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CH4-CO2 fuel mixture combustion and emission characteristics in a combi boiler combustor

CH4-CO2 yakıt karışımının kombi yanma odasında yanma ve emisyon karakteristikleri

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CH4-CO2 Fuel Mixture Combustion and Emission Characteristics in a Combi Boiler Combustor

Highlights

- *The effect of methane ratio on combustion properties was examined experimentally and numerically.*
- *The effect of inlet temperature and O_2 concentration on combustion properties in CH4-CO2 mixtures with different ratios was examined.*

Graphical Abstract

Various CH_4 - CO_2 biogas fuel mixtures were analyzed experimentally and numerically.

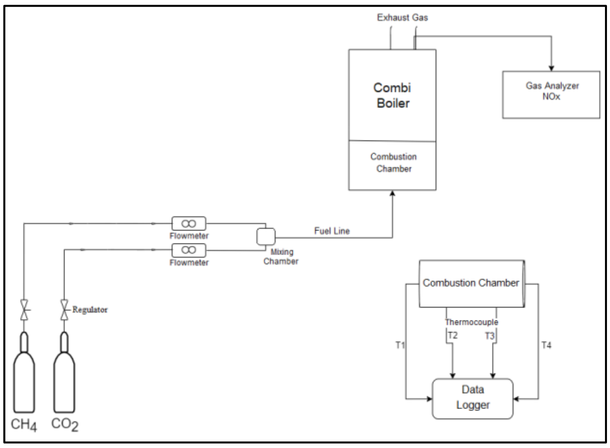

Figure. Schematic of the experimental setup

Aim

The usability of biogas in a natural gas-powered heating system was examined

Design & Methodology

In the experimental part of the study, the fuel mixtures were burned, and the temperature and the NO_X were measured. The standard k-ε turbulence model and Eddy Dissipation Concept were used in the present study's numerical part.

Originality

There are many studies regarding combi boilers and the systems using biogas in the literature, only a few studies have been conducted on the biogas usability in residential-type combi boilers. Therefore, the present research has focused on the usability of biogas in a hermetic-type gas-burning device.

Findings

CH_4 concentration in the fuel has a considerable effect on the flame temperatures. As the methane concentration in the fuel mixture decreases, the flame temperatures decrease gradually, also NOx levels decrease due to the Zeldovich Mechanism. The flame temperature levels increase with the air inlet temperature (preheating effects) for each fuel mixture. Similar conclusion occurs when the O_2 concentration in the oxizider is increased.

Conclusion

The present study experimentally and numerically presents that the CH_4 – CO_2 fuel mixtures as biogas fuel can be consumed in residential hermetic-type gas-burning devices. This can be done without any modification on the burner and the combustor

Declaration of Ethical Standards

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

CH_4-CO_2 Fuel Mixture Combustion and Emission Characteristics in a Combi Boiler Combustor

Research Article / Araştırma Makalesi

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ABSTRACT

Biogas can be obtained from agricultural, animal, and waste, replacing natural gas in heating systems. Because of that, studying its combustion characteristics will be valuable. This study investigates biogas combustion characteristics in a natural gas-powered heating system. Various CH_4-CO_2 biogas fuel mixtures were analyzed experimentally and numerically for a thermal power of 13 and 31 kW. In the experimental part of the study, the fuel mixtures were burned, and the temperature and the NOX were measured. The standard k-ε turbulence model and Eddy Dissipation Concept were used in the present study's numerical part. The model was validated with the measured data. The modeling results are in agreement with the measurements. Various fuel mixtures were then modeled to seek varying inlet temperature and O_2 concentration effects on the combustion characteristics of the biogas fuel mixtures studied. It was concluded that as the amount of CO_2 in the fuel mixture was increased the flame temperature decreased linearly from about 1900 K to 1775 K. NOx concentrations in the combustion product decreased from about 10 ppm to below 2 ppm. It was also concluded that air preheating and change in O_2 concentration in the oxidizer affected the flame temperatures and NOx considerably.

Keywords: Biogas, Combustion, Domestic Boiler, CFD, Emission

CH4-CO2 Yakıt Karışımının Kombi Yanma Odasında Yanma ve Emisyon Karakteristikleri

ÖZ

Tarımsal, hayvansal ve atıklardan elde edilebilen biyogaz, ısıtma sistemlerinde doğal gazın yerini alabilir. Bu nedenle yanma özelliklerini incelemek değerli olacaktır. Bu çalışmada doğal gazla çalışan bir ısıtma sisteminde biyogazın yanma özellikleri araştırılmaktadır. Çeşitli CH_4- CO_2 biyogaz yakıt karışımları 13 ve 31 kW ısıl güç için deneysel ve sayısal olarak analiz edilmiştir. Çalışmanın deneysel kısmında yakıt karışımları kombide yakılarak sıcaklık ve NOX değerleri ölçülmüştür. Çalışmanın sayısal kısmında standart k-ε türbülans modeli ve Eddy Dissipation Concept kullanılmıştır. Sayısal model deneysel verilerle doğrulandı. Sayısal model ile farklı yakıt karışımları için farklı giriş hava sıcaklıklarının ve O_2 konsantrasyonunun etkisi de incelenmiştir. Yakıt karışımındaki CO_2 miktarı arttıkça alev sıcaklığının doğrusal olarak yaklaşık 1900 K'den 1775 K'ye düştüğü görülmüştür. Yanma ürünündeki NOx konsantrasyonunun yaklaşık 10 ppm'den 2 ppm'nin altına düştüğü görülmüştür. Ayrıca hava ön ısıtmanın ve oksitleyicideki O_2 konsantrasyonu değişiminin alev sıcaklıklarını ve NOx değerini önemli ölçüde etkilediği sonucuna varılmıştır.

Anahtar kelimeler: Biyogaz, Yanma, Kombi, HAD, Emisyon

1. INTRODUCTION

The most critical resource in countries' economic and social development is energy. The importance of energy continues to increase due to rapid population growth, significant developments in the industry, urbanization, unplanned use of resources, and changing production and consumption methodology due to globalization. It has become one of the essential problems of today. Reasons such as the recent significant increase in the number of new products and new habits entering our lives due to technological developments, innovation, and research and development studies, as well as the inability to use existing energy resources effectively and efficiently, seriously affect excessive energy consumption. This situation increases the importance of energy for developing countries and significantly affects the strategic plans of countries, especially in economic and technological fields. Because fossil fuel reserves such as oil and natural gas are rapidly depleted, the ozone layer is thinning, and greenhouse gas emissions threaten human life. This situation has made it necessary to use existing resources efficiently, to search for new resource methods, and to turn to renewable energy sources. In a world where energy problems are increasing but

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resources are depleted, efficient energy use is gaining importance. The increasing consumption of coal, oil, and natural gas increases CO_2 emissions and causes the greenhouse effect and global warming [1]. In addition to all these, considering the problems experienced in the past regarding energy supply security, developing countries need to turn to their energy resources and ensure that these resources are renewable.

Biogas is one of the primary sources to provide energy sustainably without causing environmental pollution. Biogas energy is considered a convenient and vital source because it is inexhaustible, can be obtained everywhere, and helps socio-economic development, especially in rural areas. Specially grown plants such as corn and wheat, grasses, mosses, algae in the sea, animal feces, manure, industrial wastes, and all organic garbage (fruit and vegetable residues) thrown away from homes constitute the source of biogas. Biogas is a colorless-odorless gas mixture lighter than air and burns with a bright blue flame, resulting from the decomposition (anaerobic fermentation) of organic waste/waste in an oxygen-free environment [2]. Anaerobic processing leaves large organic materials in an oxygen-free environment and converts them into different substances through various microorganisms. These microorganisms can be expressed as hydrolyzing bacteria, bacteria with low molecules, and bacteria containing methane and acid [3]. Biogas is used in combustion systems to obtain energy with 50-70% methane in its composition, thus providing energy from waste.

Some researchers and combustion engineers have focused on biogas combustion and its effects on emissions. Leonoy and Trubaev, for instance, theoretically examined the effect of biogas composition and excess air ratio on combustion parameters such as combustion temperature and volume of combustion products [4]. It has been observed that the end-combustion temperatures decrease as the methane ratio in the mixture decreases. Nonaka and Pereira conducted numerical and experimental studies on the adiabatic laminar burning rate of the CH_4-CO_2 mixture [5]. As the CO_2 ratio in the fuel mixture increases, the burning rate decreases, and the maximum burning rate shifts from the stoichiometric point to the lean mixture side. Chan et al. investigated numerically and experimentally the effect of CO_2 (0-15%) addition on the laminar flame speed of methane-air mixture [6]. As a result of the studies, it was observed that the laminar flame speed and flame temperature of the mixture decreased with the addition of CO_2. Hinton and Stone examined the change in laminar combustion rate with the combustion of CH_4-CO_2 fuel mixture at different rates under different temperature and pressure conditions [7]. It is concluded that the burning rate decreases at high pressures and increases with increasing temperature. It has also been observed that the burning rate increases with the increase in the CO_2 ratio in the fuel mixture, and the highest burning rate point shifts towards the lean mixture region. İlbaş al. experimentally examined the combustion performances and emission properties of biogas containing different gas components in a natural gas burner with a thermal power of 10 kW and an excess air coefficient of 1.2 [8]. As a result of the experiments, it was demonstrated that the biogas' temperatures were close to the methane temperature. This result concluded that biogas could be used in natural gas-burning systems without any modification. Ma et al. numerically examined the effect of CO_2 ratio on flame temperature, flame propagation speed, and nitrogen oxide formation in biogases with low calorific value [9]. It has been observed that as the CO_2 ratio increases, the combustion flame temperature and flame propagation speed decrease. Cohé et al. conducted experimental and numerical studies on the propagation speed and flame surface density of premixed laminar and turbulent flame for a fuel-poor CH_4-CO_2-air mixture at different pressures and CO_2 ratios [10]. It has been observed that when the CO_2 ratio increases, the flame propagation speed decreases, and the combustion intensity increases with the increase in pressure and CO_2 concentration. Anggono studied the effect of CO_2 ratio and mixture initial pressure on the laminar combustion rate in spark-ignition premixed combustion of methane and carbon dioxide mixture [11]. As a result of both experimental and numerical studies, it was concluded that the laminar combustion rate decreases with the increase in CO_2 concentration and mixture initial pressure. Lafay et al. experimentally compared the stable combustion areas and flame structure for methane (100% CH_4) and biogas (61% CH_4, 34% CO_2, 5% N_2) [12]. A stable flame was obtained at high equivalence ratios for each fuel, and it was observed that the flame became unstable as the equivalence ratio decreased. It has been concluded that for the same equivalence ratio, as the CO_2 ratio in the fuel increases, the reaction/combustion intensity decreases, and the addition of CO_2 destabilizes the flame structure. Greco et al. investigated the effects of fuel composition on biogas combustion in a laminar one-dimensional premixed flame [13]. It has been observed that methane content is very effective in combustion behavior, and flame speed increases with the increase in methane content. Ghenai and Janajreh carried out numerical analysis and combustion analysis for natural gas, biogas, landfill gas, and natural gas-biogas mixture fuels in gas turbines [14]. As a result of the analysis, it was seen that as the amount of CO_2 in the fuel composition increased, the flame temperature, burning speed, and NOx amount decreased. Zuo et al. investigated the combustion characteristics with different preheating temperatures and equivalence ratios for low calorific value biogas [15]. It was observed that when the preheating temperature increased, the flame speed, flame temperature, and NO amount increased. When the equivalence ratio was 1, the highest flame temperature was obtained, and when the equivalence ratio was 1.1, the highest amount of NO was obtained at the end of combustion. Sahin experimentally studied the combustion and emission characteristics of two biogas fuels with a 55% CH_4 - 45% CO_2 ratio and a 45% CH_4

55% CO_2 ratio obtained from the biogas reactor [16]. As a result of the experiments, it was observed that the flame zone temperature levels were higher for biogas with a higher CH_4 ratio, the temperatures decreased towards the end of the combustion chamber, and there was a combustion chamber distribution closer to distributed combustion conditions. Amez et al. examined the combustion performance of the biogas-hydrogen mixture in a conventional natural gas burning device [17]. In the experiments, biogases were used in the ratios of 70:30 CH_4:CO_2 (BG70), 60:40 CH_4:CO_2 (BG60), and 50:50 CH4:CO2 (BG60), and studies were carried out with new mixtures by adding hydrogen from 5% to 25% to the biogas. As the CO_2 ratio in the mixture increases, combustion efficiency decreases, and ignition problems occur at CO_2 ratios above 40%. Although there was an increase in combustion efficiency with the addition of hydrogen, the efficiencies obtained with natural gas could not be achieved. Sahin numerically investigated the combustion characteristics of biogas with lower O_2 concentrations [18]. It has been revealed that NO_X and CO levels approach zero when the oxygen concentration in the oxidizer decreases and temperature values decrease. Additionally, when the oxygen concentration decreased, a more homogeneous thermal area was obtained in the combustion chamber. In their study, Dong et al. experimentally and numerically examined the laminar combustion properties of methane and its effects on NOx emission by adding different amounts of CO2 to methane [19]. It has been observed that, at any equivalence ratio, the laminar burning rate and flame temperature decrease as the CO2 ratio in the mixture increases. Devi et al. conducted a study on thermal and combustion efficiencies using biogas in a porous radial burner and a conventional burner [20]. While a stable flame was obtained in lean mixtures with porous radial burner, it was obtained in rich mixtures with conventional burner. Cellek et al. numerically examined the effects of pressure, excess air ratio and oxygen content on temperature and emissions in the combustion of biogases with different calorific values. [21]. The study showed that different inlet pressures, fuel calorific value and air excess ratio have a significant impact on combustion performance.

In a world where energy and emission problems are increasing and resources are being depleted, interest in renewable resources is increasing, and the usability of these resources in existing combustion systems is also important. Although there are many studies regarding combi boilers and the systems using biogas in the literature, only a few studies have been conducted on the biogas usability in residential-type combi boilers. Demonstrating the suitability of using biogas in residential combi burners will enable the use of an alternative and renewable fuel instead of natural gas. Therefore, the present research has focused on the usability of biogas in a hermetic-type gas-burning device. For this purpose, an experimental setup has been built up, and some experimental measurements were performed. Then, the modelings were carried out in the same combustion system through a commercial code. It seeks consistency between the experimental and the numerical results. After that, biogas fuel mixtures (60% CH_4 - 40% CO_2, 70% CH_4 - 30% CO_2, 80% CH_4 - 20% CO_2, and 90% CH_4 - 10% CO_2) by volume were further modeled at different inlet temperatures and O_2 concentrations to seek combustion and emission characteristics of the fuel mixtures under various combustion conditions. At the end of the modelings, temperature and NOx distributions inside the combustor were predicted and discussed along with the measurements.

2. MATERIALS AND METHOD

2.1. Numerical Modelling Details

2.1.1. Geometry and mesh structure

A typical condensation hermetic type combi boiler combustor and its 2-dimensional geometrical model, which was used to perform experimental and numerical modelings, are shown in Figure 1. The fuel injector diameter is 1 mm, along with the other dimensions shown in Figure 1. All modelings were taken place as 2-dimensional because depth is increased towards the inside of the combustor due to the symmetrical burner structure as 2-dimensional modeling reduces computational time due to the expedited computational speed. Thermal powers of 13 kW and 31 kW and an equivalence ratio of 1.0 were selected for the combi boiler combustion conditions for validation and further modelings. In addition, different inlet temperatures and oxygen concentrations were studied to examine their effects on combustion characteristics. A mesh structure was created in the meshing module of the commercial code used, and it is presented in Figure 2 (a general view of the entire domain) and Figure 3 (a detailed view closer to the burner inlets). In addition to a greater mesh structure, in numerical analysis, it is helpful to make the process independent from the number of cells in the mesh structure used to avoid problems such as insufficient computer performance and time. In this study, mesh independence modelings regarding the mesh structure were carried out, and its results are given in Figure 4.

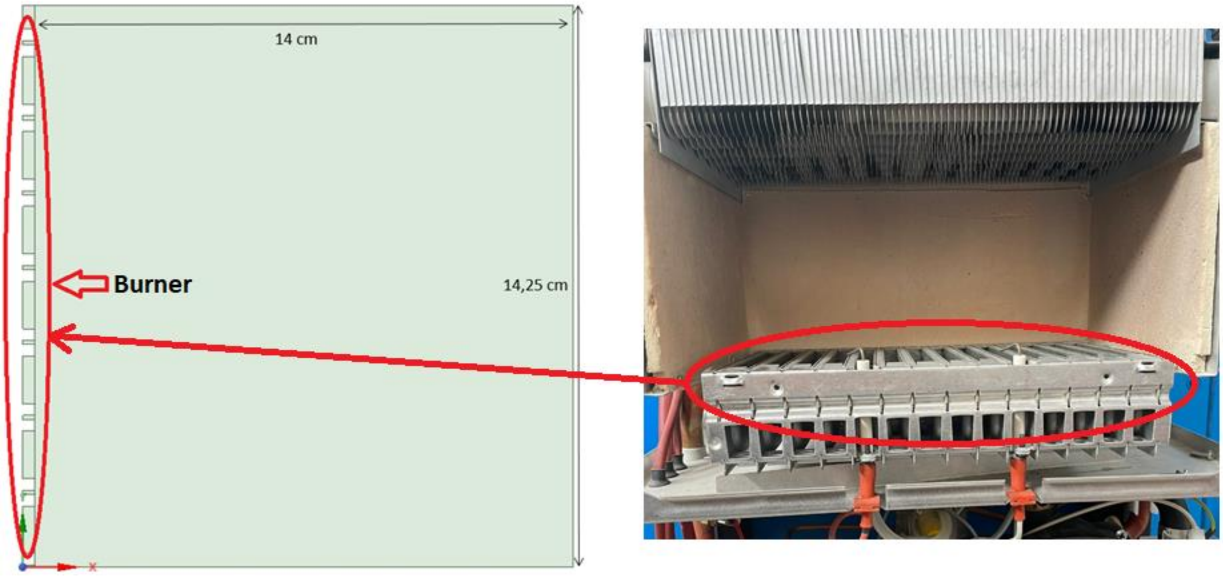

Figure 1. A hermetic-type combi boiler combustor

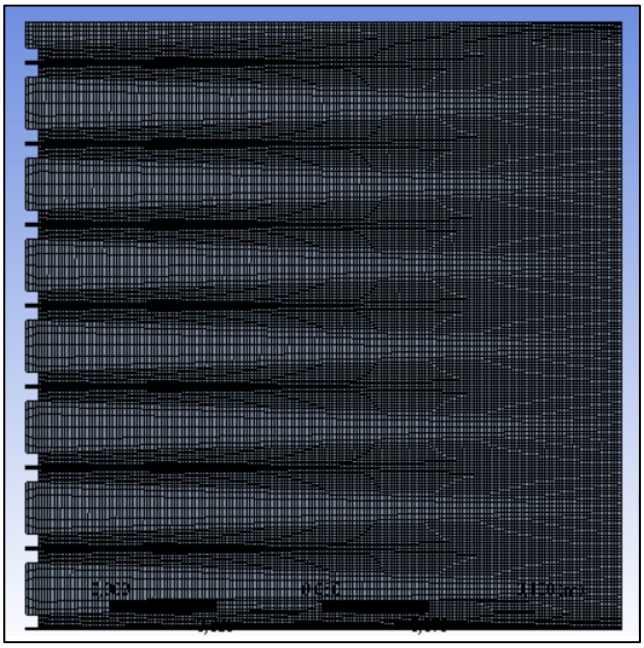

Figure 2. General view of the mesh structure

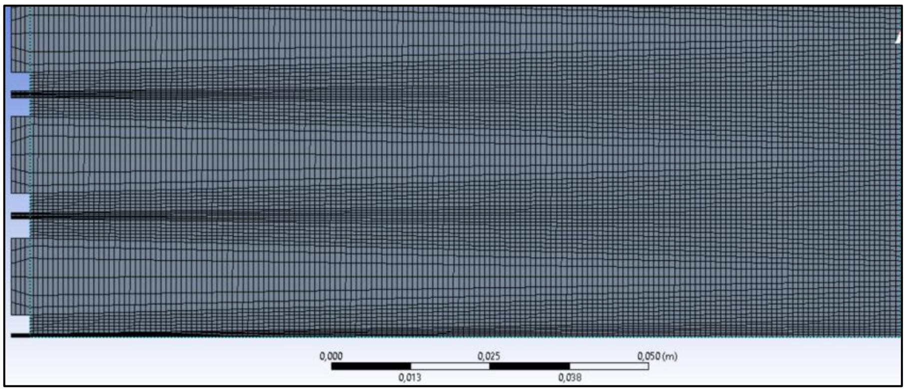

Figure 3. Detailed view of the mesh structure

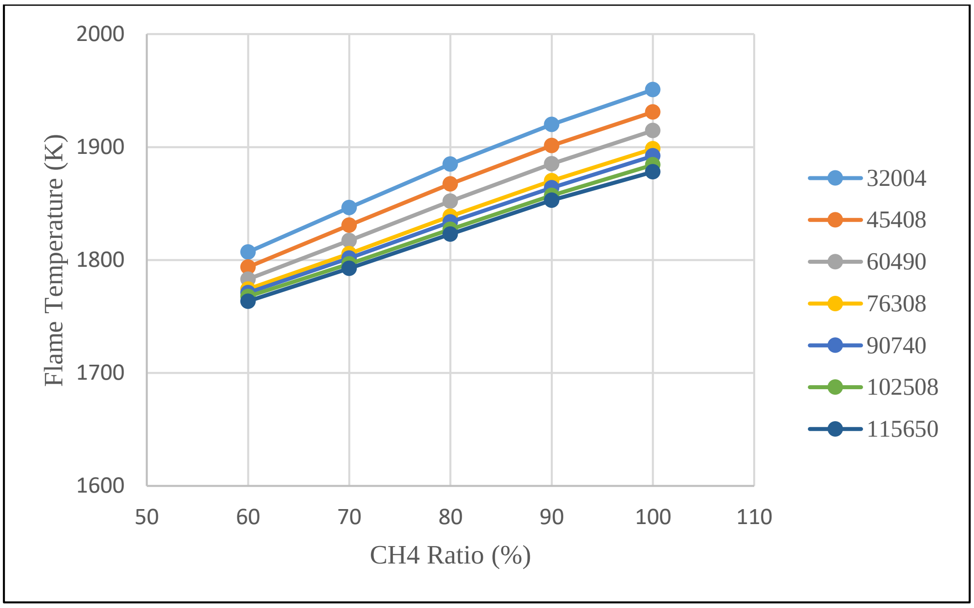

Figure 4. Temperature values according to the number of elements

For the independence study, the modelings were carried out for different mesh structures with element numbers 32004, 45408, 60490, 76308, 102508, and 115650. Figure 4 shows that the predicted flame temperature results are getting closer for all methane concentrations after the number of elements 76308 were selected for use. Since it is unnecessary to use many more elements than needed and enough due to modeling time and computer challenges, the mesh structure with 76308 elements has been chosen as the most suitable mesh structure and used for further all modelings. Data regarding mesh quality can be seen in Table 1, and it is understood that the created structure has sufficient quality for further analysis.

Table 1. The mesh structure details

	Min	Max	Average
Skewness	3.48E-04	0.2837	1.31E-02
Orthogonal Quality	0.9025	1	0.99928
Aspect Ratio	1.0002	15.238	2.4243

2.1.2. Governing equations

The mathematical model used in the simulation is based on the numerical solution of transport equations for mass, momentum, and energy [22]. Partial differential equations describing biogas combustion in the combustion chamber are presented below [23].

To solve the conservation equations for chemical species, each species' local mass fraction (Yi) is estimated by solving the convection-diffusion equation for the ith species. This conservation equation has the following general form:

$$\partial/\partial t\,(\rho Y_i) + \nabla.(\rho\vec{v}Y_i) = -\nabla.\vec{J}_i + R_i + S_i \qquad (1)$$

R_i is the net production rate of species i through a chemical reaction. S_i is the rate at which it is formed by the dispersed phase and addition from any user-defined source. This equation will be solved for N-1 species. Here, N is the total number of chemical species present in the fluid phase in the system. Since the mass fraction of the species must sum to one, the Nth mass fraction is determined as the sum of the resolved N-1 mass fractions minus one. To minimize numerical error, the Nth species should generally be chosen as the species with the most considerable mass fraction.

Mass diffusion in turbulent flows is calculated as follows:

$$\vec{J}_i = -\left(\rho D_{i,m} + \mu_t/Sc_t\right)\nabla Y_i - D_{T,i}\,\nabla \mathrm{T}/T \qquad (2)$$

Sc_t is the Schmidt number, and its default value is 0.7. μ_t is the turbulent viscosity, and D_t is the turbulent diffusivity.

For many multicomponent mixture flows, the enthalpy transport due to species diffusion is as follows:

$$\nabla.\left[\sum_{i=1}^{n} h_i\vec{J}_i\right]$$

The transport of enthalpy can significantly impact the field of enthalpy. Therefore, the analysis should address this issue when the Lewis number is far from 1 for any species.

For the pressure-based solver, species transport at the inlets consists of convection and diffusion components. The convection component is fixed by the input type mass ratio determined by the calculator. However, the diffusion component depends on the slope of the species field calculated at the input.

In Equation 1, the Eddy-dissipation model is used for the reaction rate (R_i), one of the source terms. In this model, reaction rates are assumed to be controlled by turbulence, and Arrhenius chemical kinetics calculations can be avoided.

The smaller two equations give the net production rate Ri of species i due to reaction r.

$$R_{i,r} = v'_{i,j} M_{w,i} A \rho \, \varepsilon / k_R^{min} \left(Y_R / v'_{i,r} M_{w,R} \right) \tag{3}$$

$$R_{i,r} = v'_{i,j} M_{w,i} A B \rho \, \varepsilon / k \left(\sum\nolimits_P Y_R \Big/ \sum_j^N v_{j,r}^{n} M_{w,j} \right) \tag{4}$$

Y_P refers to the mass ratio of any product species, and Y_R refers to the mass ratio of any reactant species. A and B are empirical constants equal to 4 and 0.5, respectively. ε refers to the turbulent diffusion rate, and k refers to the turbulent kinetic energy per mass.

Equations 3 and 4 govern the chemical reaction rate by the large eddy mixing time scale (k/ε). As long as turbulence exists (k/ε >0), combustion continues, and there is no need for an ignition source to start combustion.

2.1.3. Combustion equation of biogas

The stoichiometric combustion equation of biogas is as follows [10]:

$$\begin{aligned}(0.6CH_4 + 0.0025H_2S + 0.3975CO_2) + 1.204O_2 \\ + 3.76(1.204)N_2 \\ \rightarrow 0.9975CO_2 + 1.2025H_2O \\ + 0.0025SO_2 + 4.5261\end{aligned}$$

However, within the scope of the study, biogas composition is considered a mixture of carbon dioxide and methane. Accordingly, the general equation for methane-air combustion with added carbon dioxide is [10]:

$$\begin{aligned}(1-\beta)(CH_4 + (2+x)(O_2 + 3.76N_2)) + \beta CO_2 \\ \rightarrow CO_2 + 2(1-\beta)H_2O + 7{,}56(1 \\ -\beta)N_2 + x(1-\beta)(O_2 + 3.76N_2)\end{aligned}$$

Here, β refers to the mole fraction of CO_2 in the fuel, and x refers to the excess air ratio. With this equation, inlet flow rates were calculated for all mixture ratios.

2.1.4. NOx formation mechanism

Unlike many other emissions, predicting NO formation and emission from purely kinetic evaluations is nearly impossible. There are three common pathways for NO formation, and their importance depends on issues such as temperature, concentrations of free radicals in the flame and post-flame regions, fuel type, air-fuel mixture ratio, gas flow, and mixture properties [24]. NO formation occurs through different mechanisms, three important and influential in NO formation: thermal NO, fuel NO, and prompt/fast NO.

Although nitrogen and oxygen coexist in the atmosphere without reacting, they combine at high temperatures with the general reaction to form NO.

$$N_2 + O_2 \leftrightarrows 2NO$$

Molecular nitrogen is unreactive with its triple bond. Thermal nitrogen appears prominently in high-temperature combustion processes (above 1600°C). A series of high-temperature-dependent chain reactions determine the formation of thermal NO called the Zeldovich mechanism.

$$N_2 + O \leftrightarrows N + NO$$

$$N + O_2 \leftrightarrows O + NO$$

The activation energies of both reactions are very high. Therefore, formation rates are negligible at low temperatures and increase significantly at high temperatures.

Prompt NO is formed at the flame front due to the reaction of nitrogen molecules with a series of free radicals produced from the combustion of hydrocarbons. In contrast, to fuel NOx, prompt NOx formation is essential in fuel-rich regions where hydrocarbon fuel degradation is high. The amount of NO mainly formed depends on the fuel-air ratio. The highest NOx formation occurs in the fuel-rich region. The generally accepted prompt NO formation method is as follows [25]:

$$CH + N_2 \leftrightarrows HCN + N$$

$$N + O_2 \leftrightarrows NO + O$$

$$HCN + OH \leftrightarrows CN + H_2O$$

$$CN + O_2 \leftrightarrows NO + CO$$

In hydrocarbon flames, CH prompt is the main source of NOx formation.

2.1.5. Standard k-ε turbulence model

The standard k-ε turbulence model is also known as the eddy-viscosity model. It is a semiempirical model based on transport equations for turbulent kinetic energy (k) and its rate of dissipation (ε). In deriving the model, there is an assumption that the flow is entirely turbulent and the effects of molecular viscosity are negligible.

Turbulence kinetic energy, k [25];

$$\begin{aligned}\partial/\partial t\,(\rho k) + \partial/\partial x_i\,(\rho k u_i) \\ = \partial/\partial x_j \left[(\mu + \mu_t/\sigma_k)\, \partial_k/\partial x_j \right] \\ + G_k + G_b - \rho\epsilon - Y_M + S_k\end{aligned} \tag{5}$$

Rate of dissipation, ε [25];

$$\begin{aligned}\partial/\partial t\,(\rho \epsilon) + \partial/\partial x_i\,(\rho \epsilon u_i) \\ = \partial/\partial x_j \left[(\mu + \mu_t/\sigma_\epsilon)\, \partial_\epsilon/\partial x_j \right] \\ + C_{1\epsilon}\, \epsilon/k\, (G_k + C_{3\epsilon} G_b) \\ - C_{2\epsilon} \rho\, \epsilon^2/k + S_\epsilon\end{aligned} \tag{6}$$

Here, G_k indicates the turbulence kinetic energy production due to the average velocity gradient, $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are the experimental constants, and σ_k and σ_ε are the Prandtl numbers.

Turbulent viscosity μ_t is expressed as follows;

$$\mu_t = \rho C_\mu k^2/\varepsilon \tag{7}$$

Where C_μ is an empirical constant.

2.2. Experimental System

The general drawing of the experimental setup built-up is schematically shown in Figure 5. The experimental setup consisted of mass flow controllers to introduce each gas (CH_4 and CO_2) to the combustor, the thermocouples (which were positioned at an interval of 3 cm each) to measure the temperature levels on some measuring points, as can be shown in Figure 6, the flue gas analyzer to determine NO_X levels for each mixture and the combi boiler. High-pressure CH_4 and CO_2 cylinders and each regulator (to regulate and reduce the pressure to the combustion pressure being almost atmospheric) were also integrated into the system to provide each gas to the combustor. The combustion system has two separate gas lines, one of which was for CH_4, and the other one was for CO_2. Before being introduced into the combustion chamber, there was a mixing chamber to mix CH_4 and CO_2 on the fuel line.

As mentioned, each biogas mixture was obtained by mixing CH_4 and CO_2. The experiments were conducted with a CH_4 flow rate of 20 slpm (standard liter per minute), corresponding to a power of 13 kW. To keep the total power constant at 13 kW, 20 slpm CH_4 flow was introduced to the combustor by adding CO_2 through the CO_2 mass flow controller at different flow rates to achieve each biogas mixture studied (0% to 40% in an interval of 5%). The mass flow controller adjusted the flow rates, each according to the CH_4 and CO_2 concentrations in the fuel mixture, and then mixed in the mixing chamber. The combustion chamber shown in Figure 6 has a rectangular structure with dimensions of 14 cm x 28.5 cm, and Figure 6 represents a flame view (the front case of the chamber was closed during the measuring experiments).

The properties of each mass flow controller, the flue gas analyzer, each thermocouple, the data logger, and their measurement ranges and uncertainties are shown in Table 2.

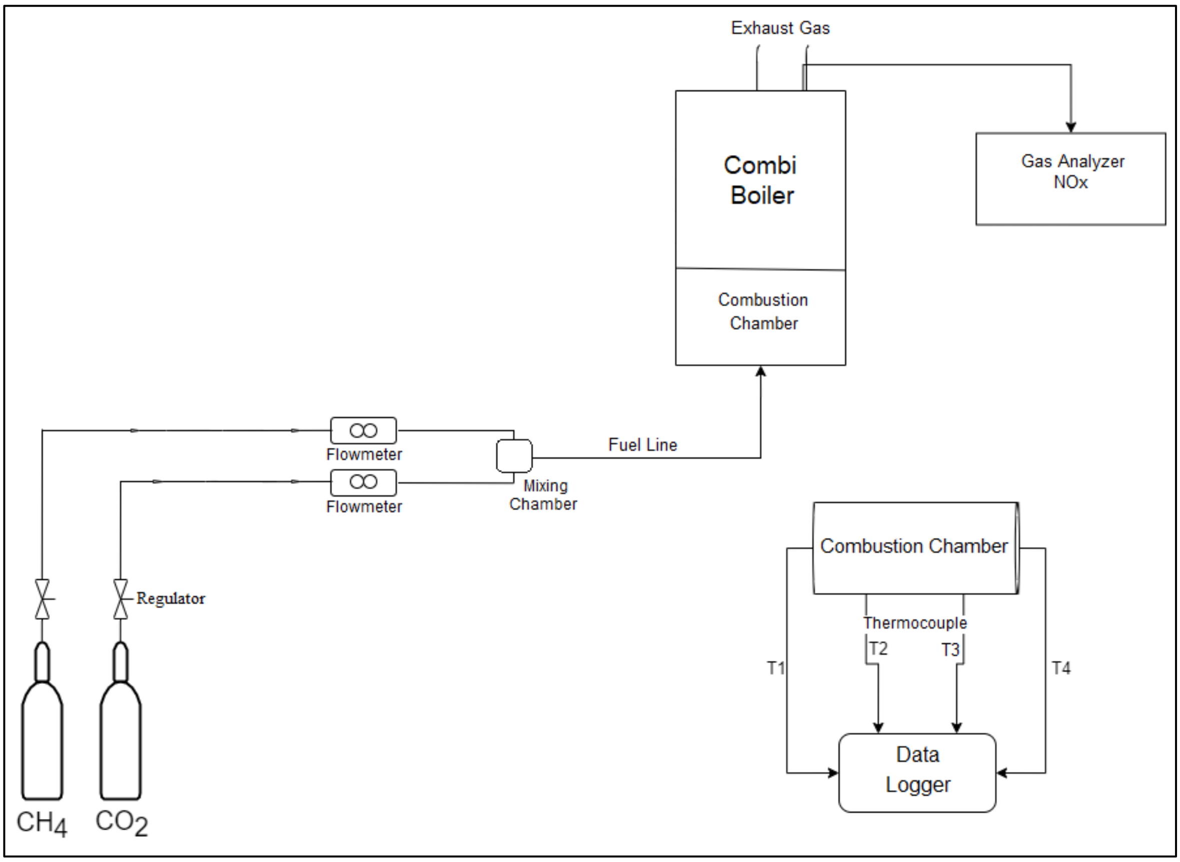

Figure 5. Schematic of the experimental setup

Figure 6. Measuring ports location

Table 2. Properties of the devices used in the experiment

Measurement	*Measuring Device*	*Measurement Range and Uncertainty*
Temperature	*K Type Thermocouple* *(Flat and mineral insulated)*	*- 200 ^{of} ~ 1300 ^{o}C* *± 4 ^{o}C*
Flow Rate	*Mass Flow Controller*	*50 L/m and 250 L/min* *± (0,8% Reading + 0,2% Full Scale)*
Data	*Datalogger E-680*	*± 0,5*
NO_X	*Gas Analyzer*	*0 – 1000 ppm for NO* *0 – 200 ppm NO_2* *Sensitivity: ±5 ppm*

3. RESULTS AND DISCUSSION

3.1. Model Validation

In this part of the present study, the experimental data measured and the predicted numerical results have been compared and discussed, as validation is crucial for the present and further modeling due to its accuracy. The combi boiler was first operated with a thermal power of 13 kW at various fuel mixtures. As stated, four thermocouples were placed in the combustor, and the validation figures are illustrated here. Figure 7, for instance, compares the experimental temperature measurements of 100% CH_4 with the numerical model results throughout the combustor. According to Figure 7, it is understood that the temperature values predicted were slightly higher compared to those of the experimental data. A similar situation is observed in Figure 8, showing the comparison between the measured flame temperatures and predicted modeling results. However, their consistency can be seen uncontrovertibly in trends. The inconsistency between the measurements and the predictions is attributed to a slight difference between the equivalence ratios studied. Because the combi boiler operates with the ambient air that is uncontrollable by the experimenter during the experiments. Nevertheless, during the modelings, the air inlet velocity was set up to correspond to an equivalence ratio of 0.75, resulting in possible differences between them. For the model validation, it is evaluated that the trend consistency emerges here. Therefore, it is concluded that the modeling results are in satisfactory agreement with the measurements within the study.

Figure 9 compares the NO_X values measured at the outlet through the flue gas analyzer and the modeling results predicted by the post-processing of the commercial code, which supports the results discussed and reported in Figure 7 and Figure 8. As seen in Figure 9, for all cases studied, there is a decrease in each NOx value with increasing CO_2 concentration in the fuel mixture due to its heating value, resulting in a reasonable consistency between the measurements and the predictions.

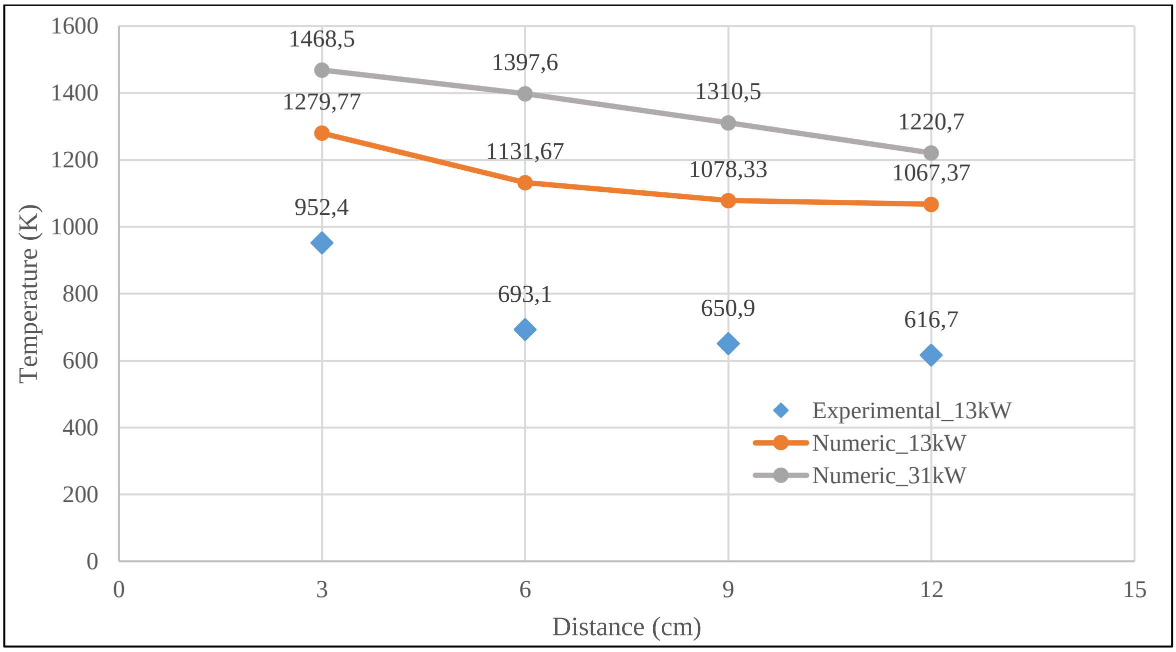

Figure 7. Model validation for temperature values at different measuring points

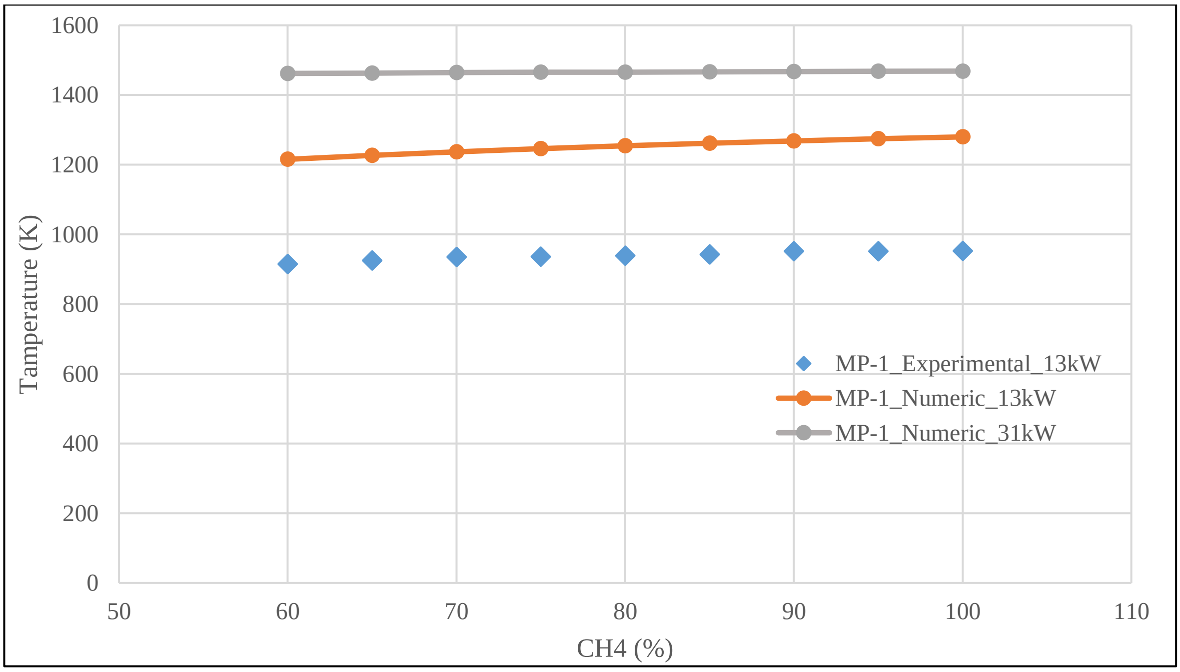

Figure 8. Model validation for different fuel mixtures' temperature values

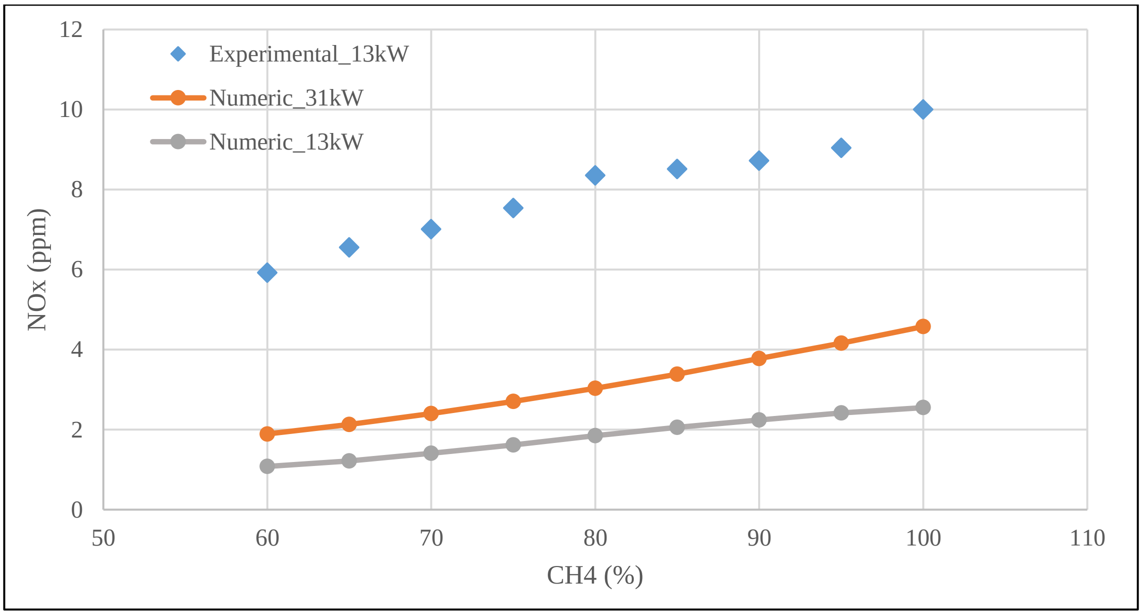

Figure 9. Model validation for different fuel mixtures NOx values

For the validation of the model created within the scope of the study, the results of different studies in the literature conducted with CH4-CO2 blended fuels were also compared.

In the study conducted by Ma et al., the effect of CO2 ratio on flame combustion temperature, flame propagation speed and nitrogen oxide formation in biogases with low calorific value was numerically examined [9]. Dong et al. investigated the combustion characteristics numerically and experimentally by adding carbon dioxide to methane [19].

In both studies, it was concluded that the flame temperature decreased as the CO2 ratio in the mixture increased as can be seen in Figure 10. In the current study, it was also concluded that the CO2 ratio reduced the flame temperature. Carbon dioxide absorbs the heat of the combustion reaction, causing the flame temperature to decrease. The results shown in Figure 10 were obtained with an equivalence ratio of 1. Differences in flame temperatures are due to flow rates. It is seen that the model used in the current study obtains results in line with the literature.

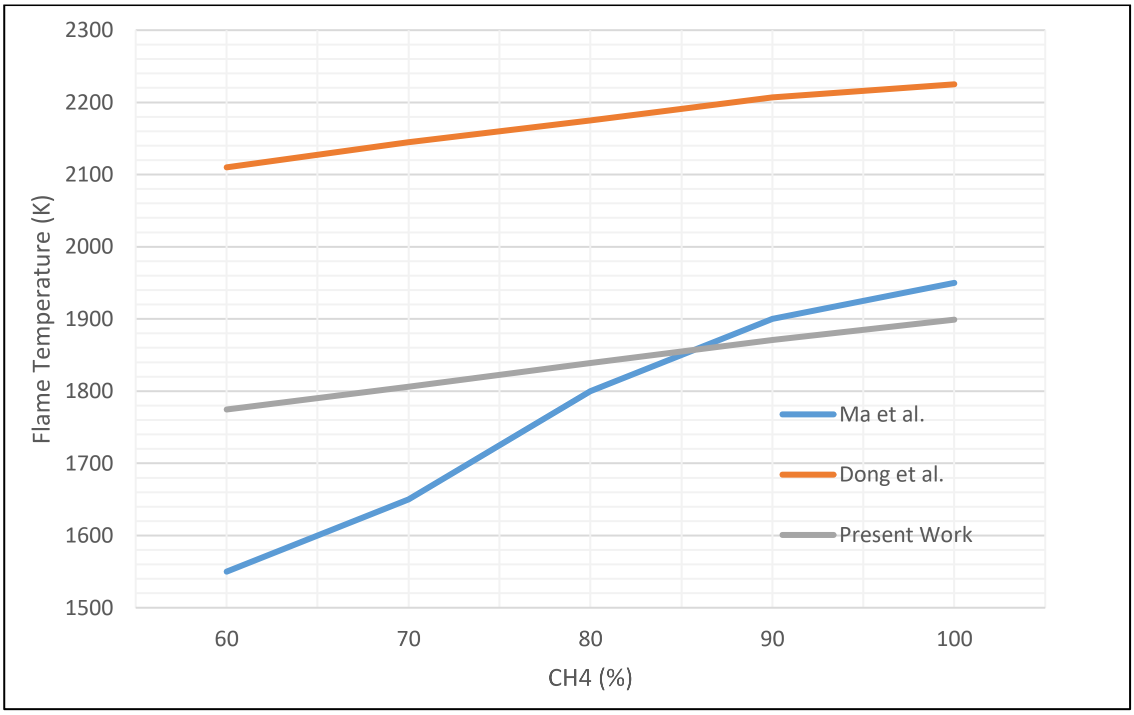

Figure 10. Comparison of different studies from the literature with the current study

3.2. Prediction Results

As the turbulence model in the numerical analyses, the standard k-Ɛ turbulence model, which is the most widely used in combustion modeling because of better compatibility with experimental data, was used [16]. The simulations used the pressure-based solver under steady-state conditions as a combi boiler operates under that. In the modeling, biogas mixtures of 60% CH_4 - 40% CO_2, 70% CH_4 - 30% CO_2, 80% CH_4 - 20% CO_2, and 90% CH_4 - 10% CO_2 by volume were consumed. Temperature and NOx distributions were obtained for different inlet air temperatures and O_2 concentrations, with an equivalence ratio of 1.0.

3.2.1. Effect of methane concentration

The effect of CO_2 addition to achieve a biogas mixture in terms of the thermodynamic properties of the mixture is mainly related to the increase of density and molar heat capacity, which tends to reduce the flame temperature [9]. CO_2 absorbs the heat released from the flame front. In addition, the concentration of small molecular intermediates such as H, O, and OH in the flame decreases as the CO_2 concentration in the fuel mixture increases. As a result, the maximum flame temperature decreases with increasing CO_2 in the fuel mixture. This dilution in the mixture reduces the concentration of reactive species in the flame front, leading to a lower rate of bimolecular reactions in the fuel oxidation reaction process. Due to the calorific value of methane, the primary component of natural gas, the maximum flame temperature during its combustion is predicted to be around 1900 K. However, as the amount of non-flammable CO_2 component in the fuel increases, the flame temperature of the consumed biogas decreases, as seen in Figure 11, consistent with the existing literature. The temperature value for a fuel mixture of 60% CH_4 - 40% CO_2 is obtained as 1774.6 K. Presence of carbon dioxide in the CH_4-CO_2 mixture reduces the rate of burning (rate of conversion of reactants to combustion products). Increasing the amount of CO_2 causes a higher energy required to decompose carbon dioxide into carbon monoxide.

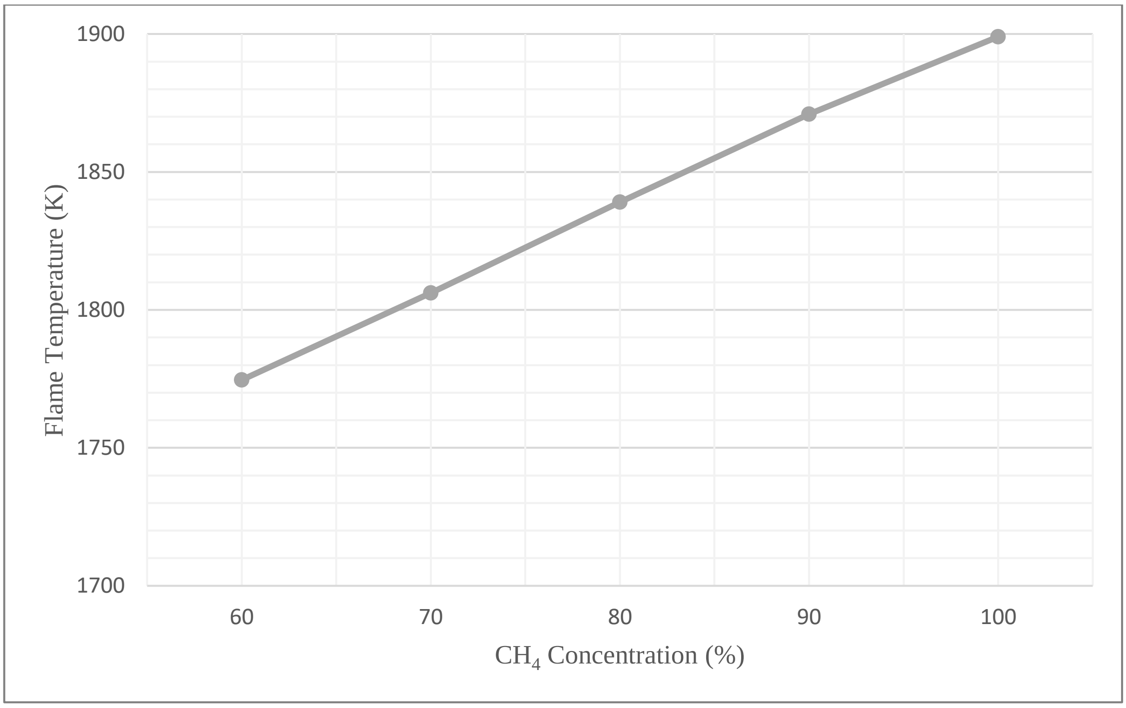

Figure 11. Methane concentration effect on flame temperature

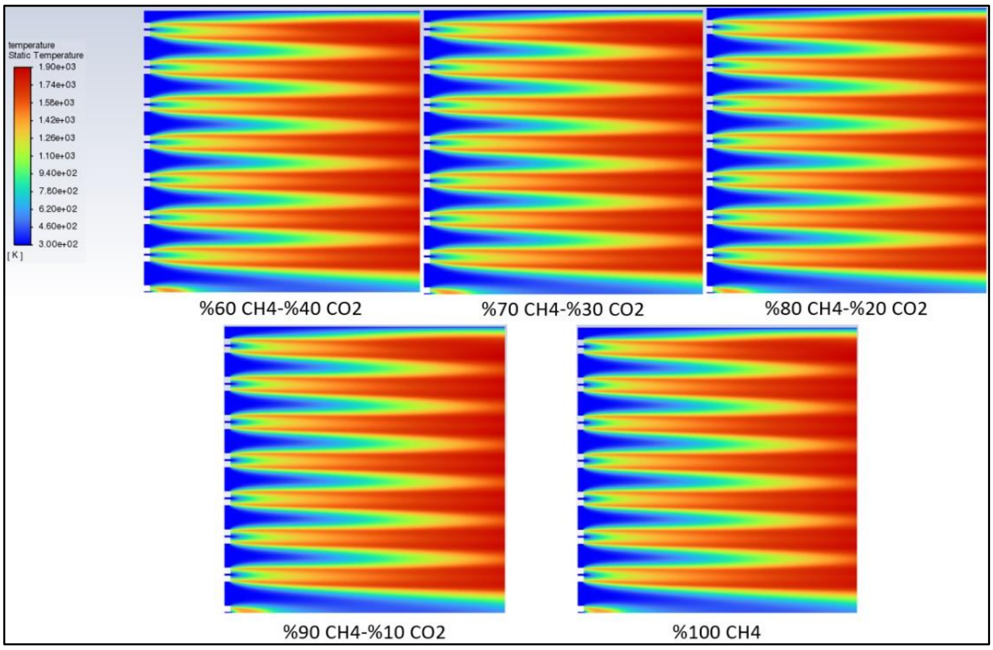

Figure 12. Temperature contours for various fuel mixtures

Temperature contours can be sought to examine flame propagation in combustion modeling studies. For this purpose, the predicted temperature contours for various blended fuels are illustrated in Figure 12. In the figure, the left side represents the inlets. In contrast, the right side represents the combustor outlet, where the hot combustion products meet a heat exchanger, and the heat release is transferred to domestic water. Although there is no considerable change in the developed flame structure, the contours give an idea about the flame structure, and it is, therefore, the flames that seem to be premixed in all cases.

Figure 13 shows the temperature change throughout the combustion chamber. When Figure 12 and Figure 13 are evaluated together, it is seen that flames started to generate near the inlet side of the combustor due to starting the reaction. Then, towards the mid-region of the combustor, the flames developed and propagated inside the combustor. The thermal effect of the flames is apparent and extends to the outlet. All flame structures inside the combustor are the same as each other because their fuel and oxidizer inlets are entirely symmetric.

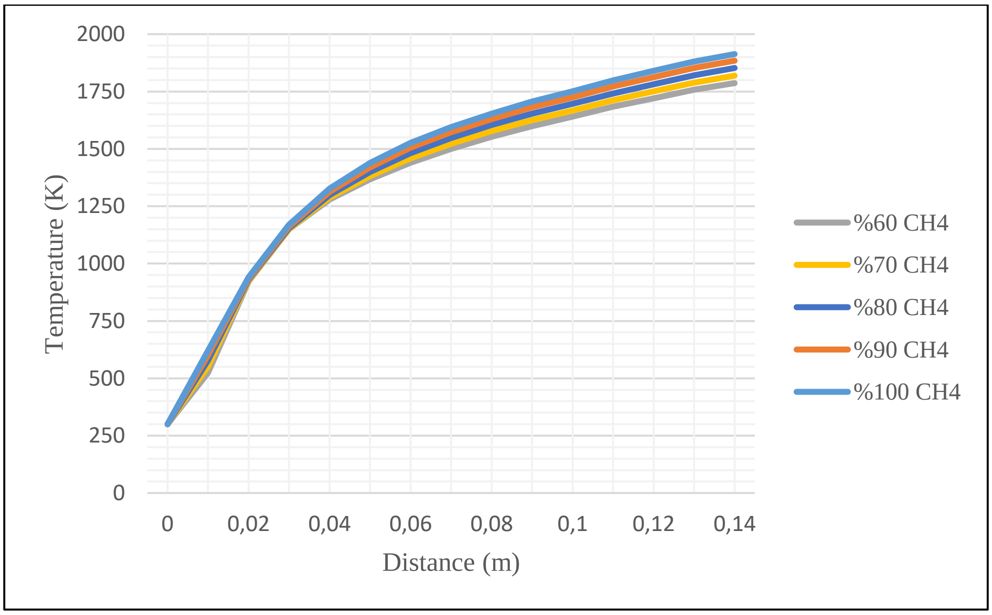

Figure 13. Temperature profile throughout the combustion chamber

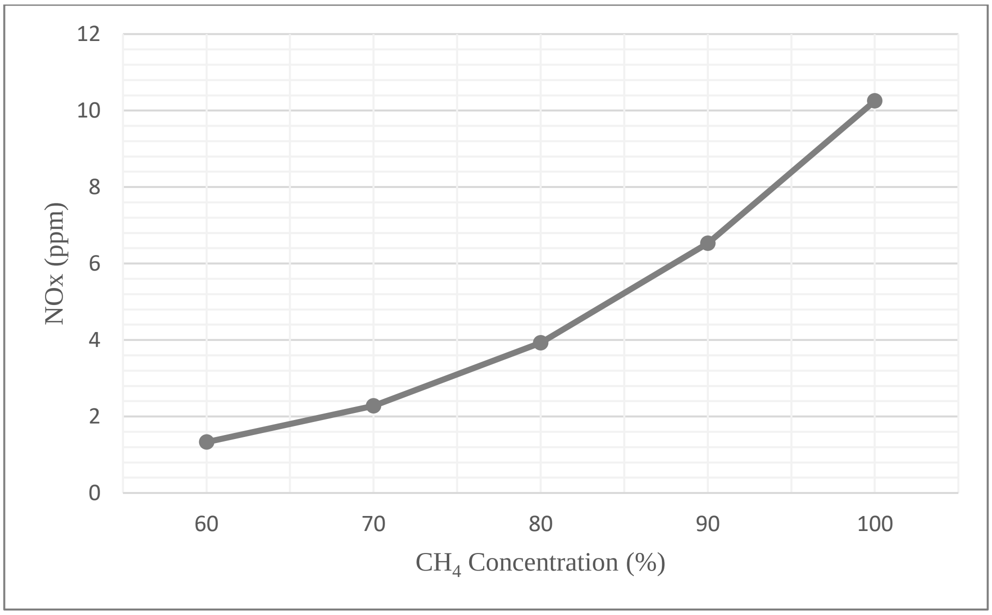

Figure 14. Methane concentration effect on NO_X emission

Nitrogen oxide emission is a problem that needs to be taken into consideration in small-capacity boilers because nitrogen oxides formed as a result of the combustion of any fossil fuel will lead to the formation of acid rain, which is an essential cause of air pollution [26]. According to the thermal mechanism, NO formation is greatly affected by temperature and oxygen concentration in the oxidizer. High-temperature zones within the flame cause nitrogen molecules in the air to separate and combine with oxygen, forming thermal NOx. NOx formation accelerates significantly above 1600°C [24]. The presence of non-flammable components such as carbon dioxide in the fuel mixture suppresses the flame temperature, and therefore, the amount of NOx can be mitigated. As seen in Figure 14, as the amount of CO_2 in the fuel mixture decreases, NOx decreases while NOx increases with increasing flame temperatures. This conclusion allows using a CH_4-CO_2 fuel mixture to mitigate it in domestic heating systems. So, the NOx value is 1.33 ppm for the 60% CH_4 - 40% CO_2 fuel mixture, which can be considered ultra-low NO_X value, while the value increases to 10.25 ppm after consuming 100% CH_4. As mentioned, thermal NOx reactions are highly temperature dependent, so the higher the flame temperature, the more NOx is formed. As the temperature value increases from the combustor's mid-region, the NOx amounts increase for each fuel mixture, as shown in Figure 15. As stated above, the expected results were obtained considering that high temperatures directly affect NOx formation.

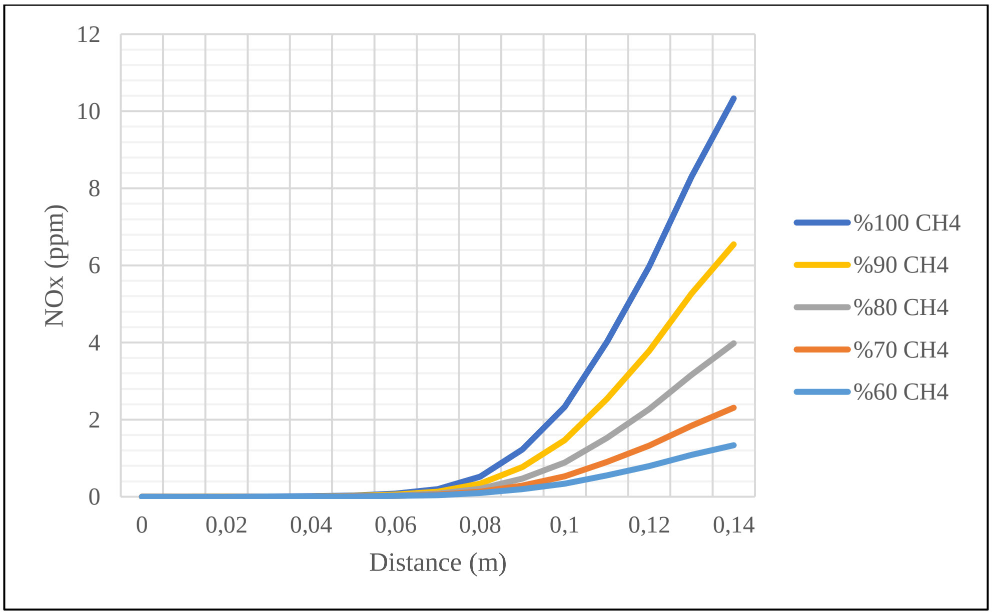

Figure 15. NOx change across the combustion chamber

3.2.2. Effect of air preheating

Within the scope of the present study, preheating effects on combustion characteristics of the CH_4-CO_2 fuel mixtures are studied. So, before the modeling was run, the preheated combustion air was set in the physics of the commercial code used, and the preheated effects on the flame temperatures were also investigated. Analyses were performed for all fuel mixtures at 300 K, 400 K, 500 K, and 600 K.

As seen in Figure 16, flame temperatures increase as the preheating value increases for all cases. This is because the increased air temperature induces the combustion reaction rate. A quadratic function relationship explaining this exists between flame propagation speed and preheating temperature [15].

$$Vf = 19.32607 - 0.12247 \times Tp + 3.2319 E - 4 \times Tp^2 \quad (8)$$

Higher flame speed causes a higher heat release rate in the combustion chamber, rapidly increasing temperatures [27]. For an inlet temperature of 600 K and 100% CH_4, the flame temperature is predicted as 1979 K, while for an inlet temperature of 300 K and 100% CH_4, the flame temperature is almost 1900 K. The main reason behind it is the expedited reaction rate of the mixture. In addition, preheating increases the enthalpy of the oxidizer and, therefore, the energy input of the flame. It also helps the fuel-air mixture to mix more homogeneously and faster. In this case, a higher flame temperature is obtained, resulting in a higher domestic water temperature in the combi boiler. The temperature contours obtained under those conditions are also presented in Figure 17.

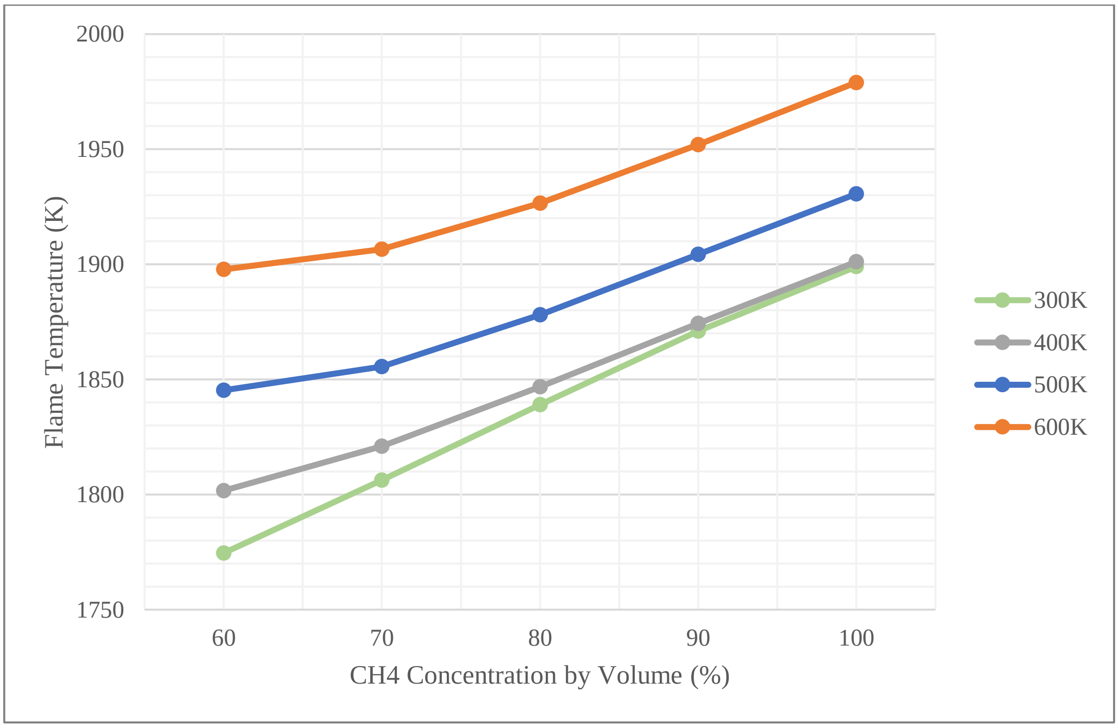

Figure 16. Preheated air effect on flame temperature

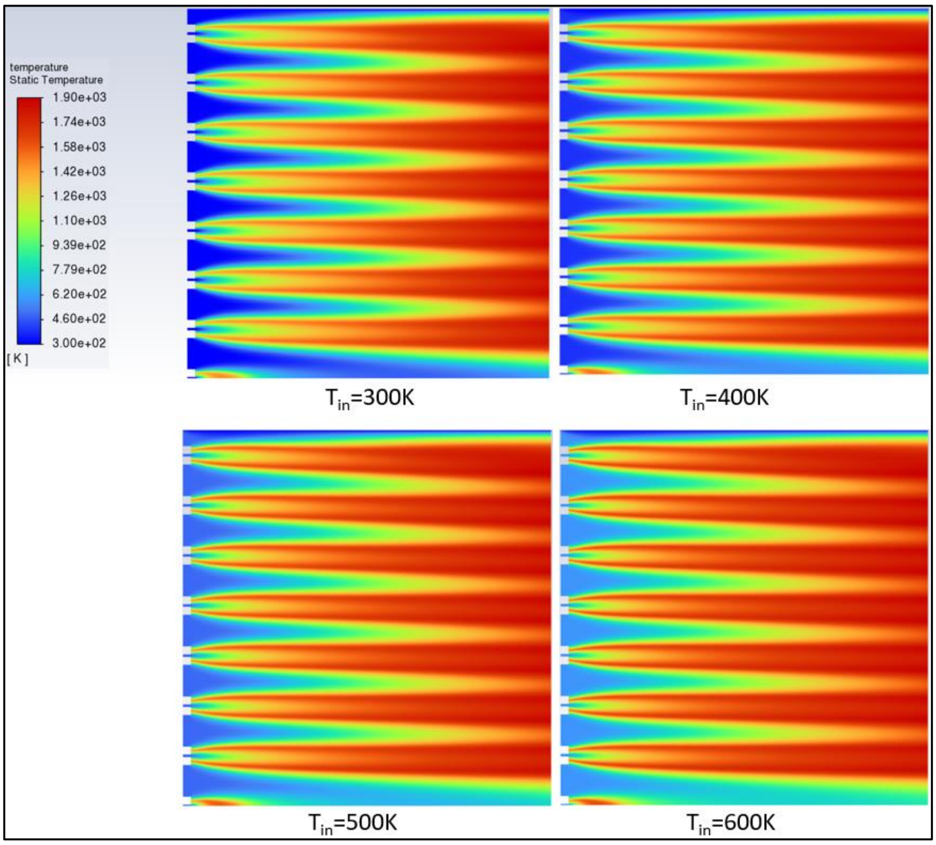

Figure 17. Preheated air effect on temperature contours for 60% CH_4 – 40% CO_2 combustion

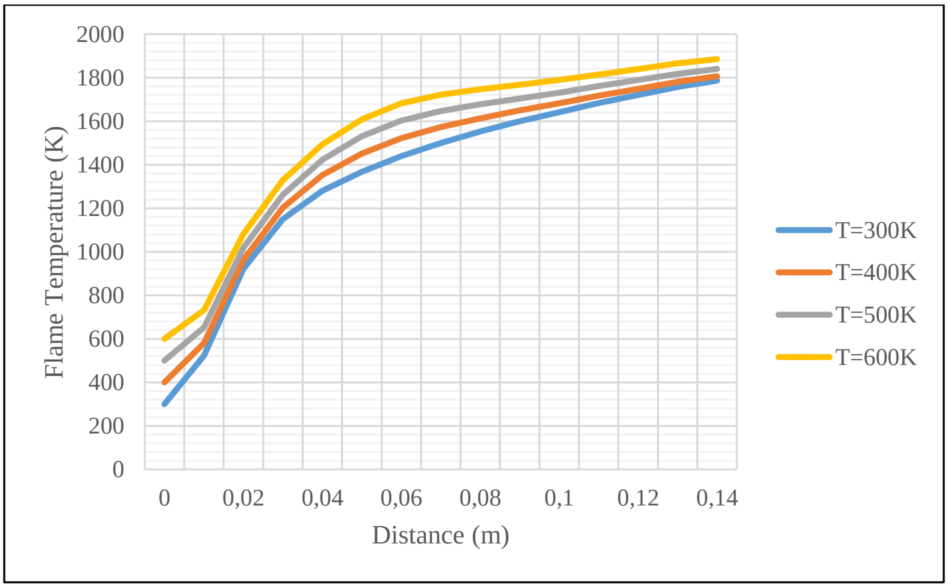

Figure 18. Preheated air effect on temperature profile for 60% CH_4 – 40% CO_2 combustion throughout the combustor

The predicted temperature profiles throughout the combustor with preheating and combustion with 60% CH_4 – 40% CO_2 mixed fuel are seen in Figure 18. Although the increase in inlet temperature increases the flame temperature, the temperature increase rate, especially in the region close to the burner, appears lower at high inlet temperatures. As can also be seen in Figure 17, the flame develops towards the mid-region of the combustor. It reaches the highest temperature value towards the end of the combustion chamber because a combi boiler combustor is designed to transfer the highest energy released to domestic water through a heat exchanger.

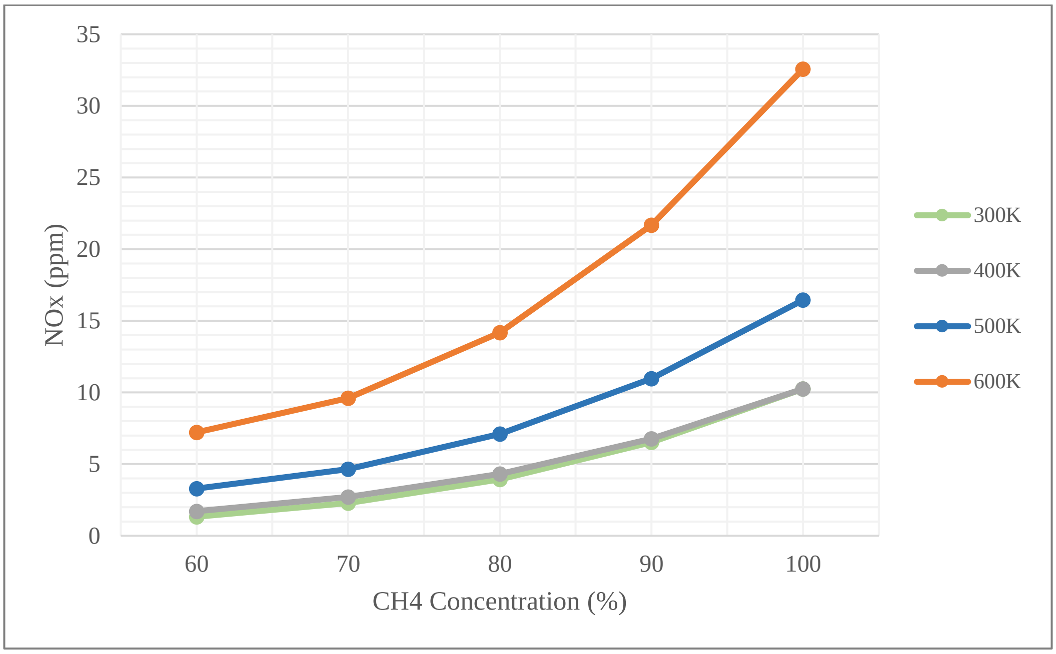

Figure 19. Preheated air and methane concentration effects on NOx emission

NOx levels in Figure 19 appear to be consistent with the predicted flame temperatures. It is worth noting that higher air inlet temperature leads to higher flame temperature, which accelerates NOx formation. In other words, as the flame temperature increases, NOx also increases. Because the thermal NOx mechanism plays a vital role in NOx formation. Besides, higher methane concentration in the fuel mixture leads to higher NOx formation in the combustion products. This is attributed to carbon dioxide in the fuel mixture absorbing higher energy released and not permitting an increase in the flame temperature. Similar results can be seen in the NOx emissions obtained throughout the combustion chamber, as shown in Figure 20.

There is also a quadratic function relationship between preheating temperature and NOx amount [15]:

$$n(NO)\% = 2.93128E - 4 + 2.89742\ E - 7 \times Tp + 2.13244\ E - 9 \times Tp^2 \quad (9)$$

Where T_p refers to the preheating temperature.

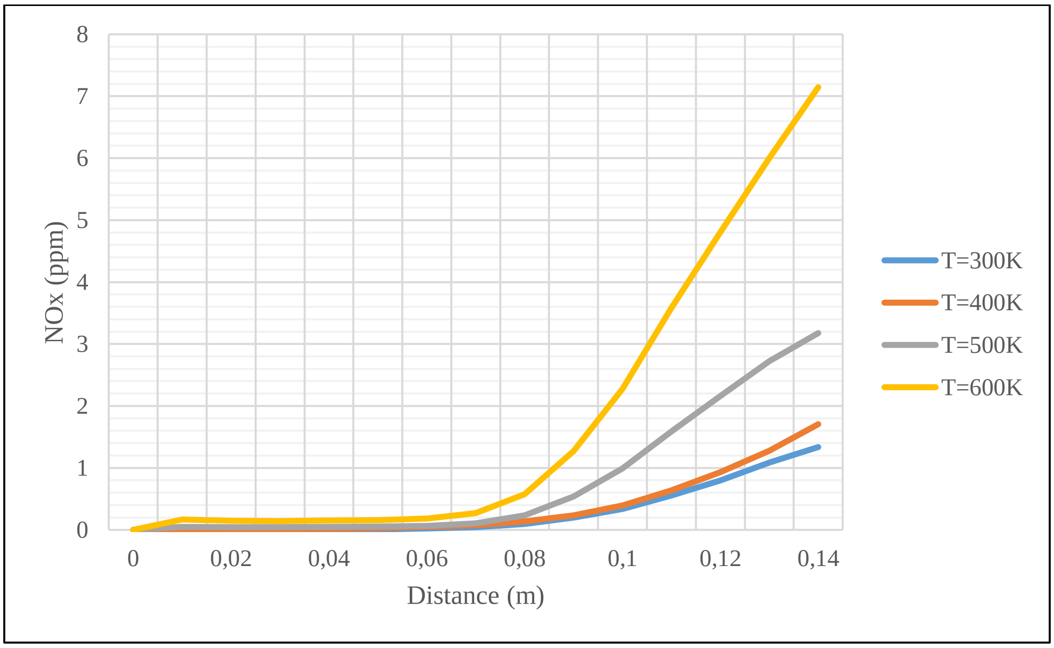

Figure 20. Preheated air effect on NOx emission for 60% CH_4 – 40% CO_2 combustion across the combustion chamber

3.2.3. Effect of oxygen concentration

Due to its depletion and environmental restrictions, many alternative gaseous fuels with lower heating values are consumed instead of natural gas. However, due to their lower heating values, the heat intensity released may need to be increased to meet the energy required for an end-user. For this reason, oxygen enrichment can be implemented. Similarly, because of environmental restrictions, the oxygen concentration in the oxidizer is reduced to mitigate pollutant emissions. The change in oxygen concentration in the oxidizer on flame temperature is examined here. The modelings were carried out at different oxygen concentrations of 15%, 18%, 21%, 24%, and 27% in the oxidizer with an inlet temperature of 300 K.

Figure 21 shows the flame temperature results obtained under various oxygen concentrations for all fuel mixtures studied. As can be seen from the figure, when oxygen concentration is reduced below 21%, resulting in a slowed reaction rate down, the flame temperatures of all biogases decrease. In comparison, these values increase once it is increased above 21%. A decrease in oxygen concentration causes a reduction in the amount of activated oxygen molecules, which inhibits the chemical reaction between oxygen and fuel. As also reported in the studies existing in the literature, it is seen that an increase in the amount of N_2, which has a dilution effect in the combustion air, reduces the flame temperature. On the contrary, when the N_2 concentration is reduced and the O_2 concentration is increased, the flame temperature increases [28]. Similar results can be evaluated based on the temperature contours predicted, which are presented in Figure 22 and Figure 23. Temperature differences increase as the flame propagates toward the middle of the combustion chamber.

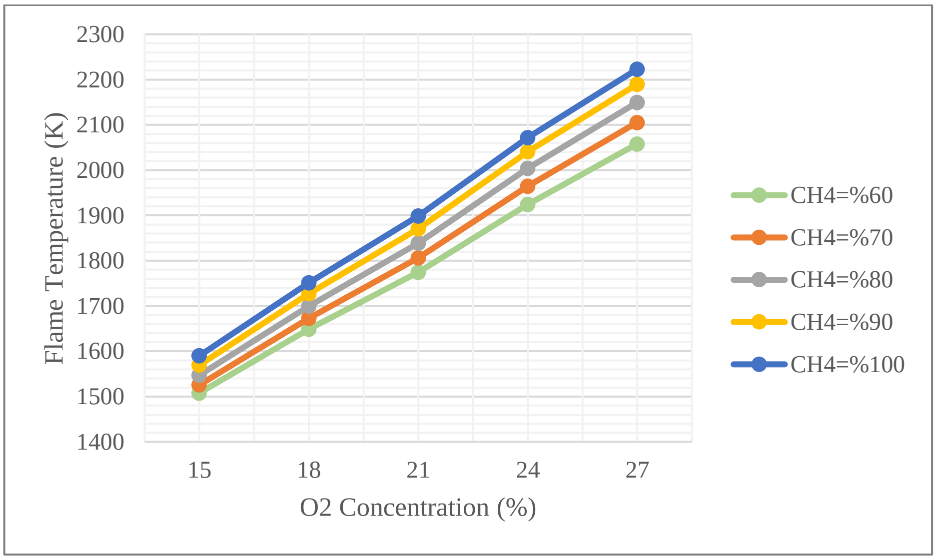

Figure 21. Oxygen concentration effect on flame temperature

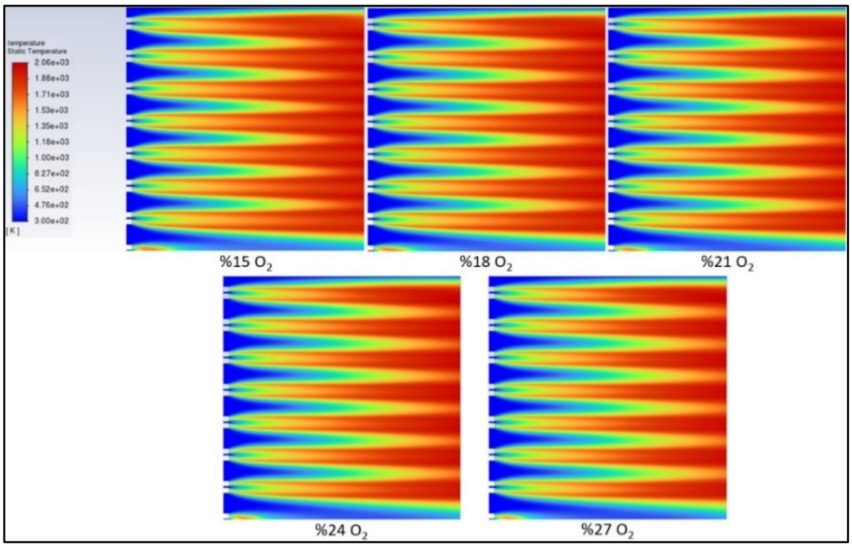

Figure 22. Oxygen concentration effects on temperature contours using a 60% CH_4 – 40% CO_2 fuel mixture

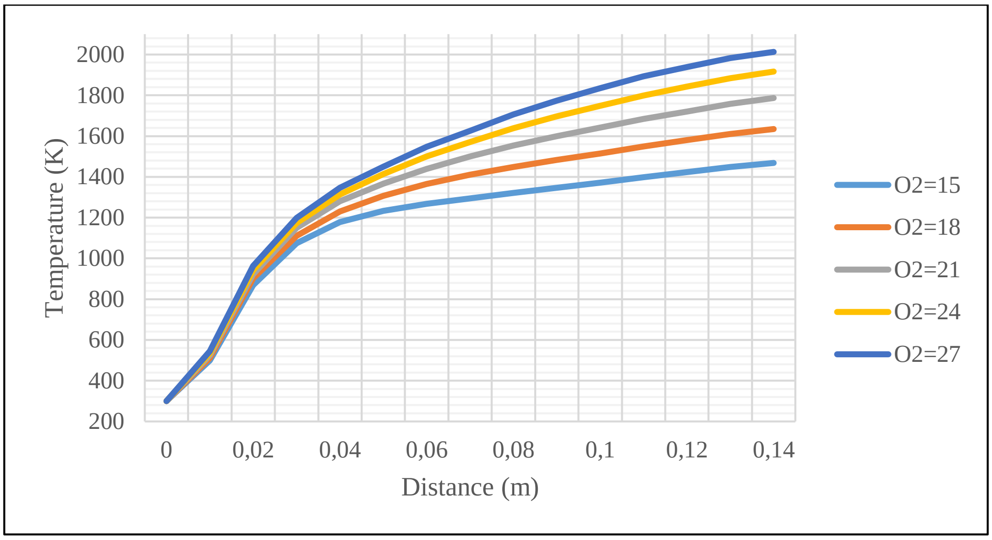

Figure 23. Oxygen concentration effect on temperature for 60% CH_4 – 40% CO_2 combustion throughout the combustor

NOx emission is affected by factors such as flame temperatures, oxygen concentration, and intermediate species [29]. An increase in flame temperatures with an increase in O_2 concentration in the oxidizer causes the amount of NOx to increase. Besides, $N_2+O=NO+N$ and $N_2O+O=2NO$ reactions are effective in NOx formation. Considering that oxygen is the reactant of the two reactions mentioned and an increase in oxygen radicals in the reaction zone, an increase in NOx is expected.

Figure 24 shows that NOx amounts increase rapidly for all fuel mixtures after 21% O_2 concentration. Since NOx formation accelerates significantly above 1600 °C and temperatures rise quickly after 21% O_2, it is considered natural for the amount of NOx to increase rapidly.

If 1600 °C is considered the approximate threshold level for the thermal NOx mechanism, a rapid increase in the rate of NOx formation is formed further. When Figure 25 is evaluated together with Figure 23 it is seen that NOx formation accelerates from the middle of the combustion chamber with the increase in temperature.

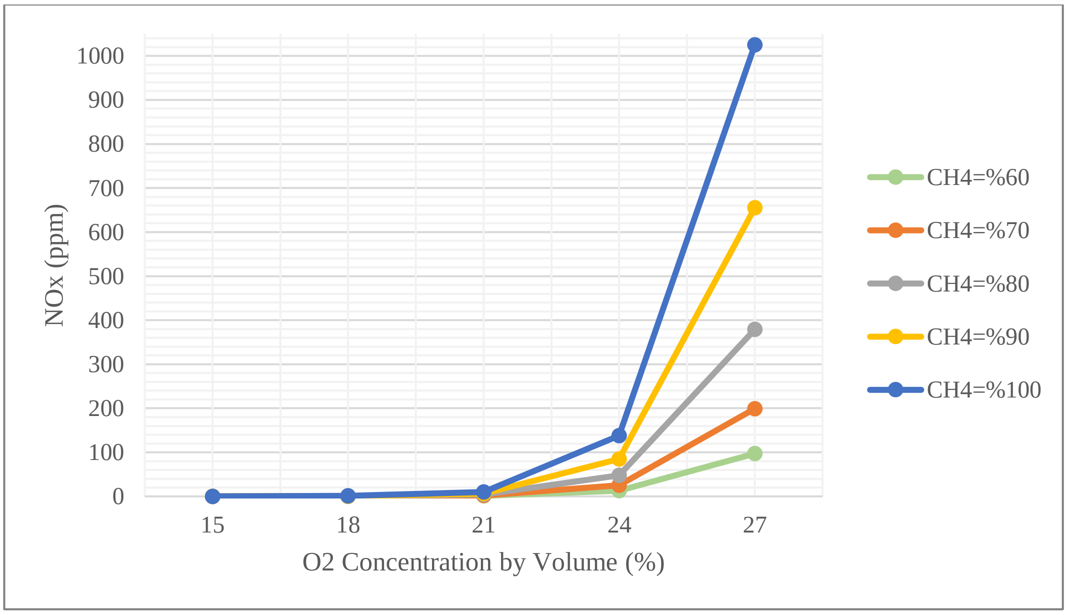

Figure 24. Oxygen concentration effects on NOx formation levels

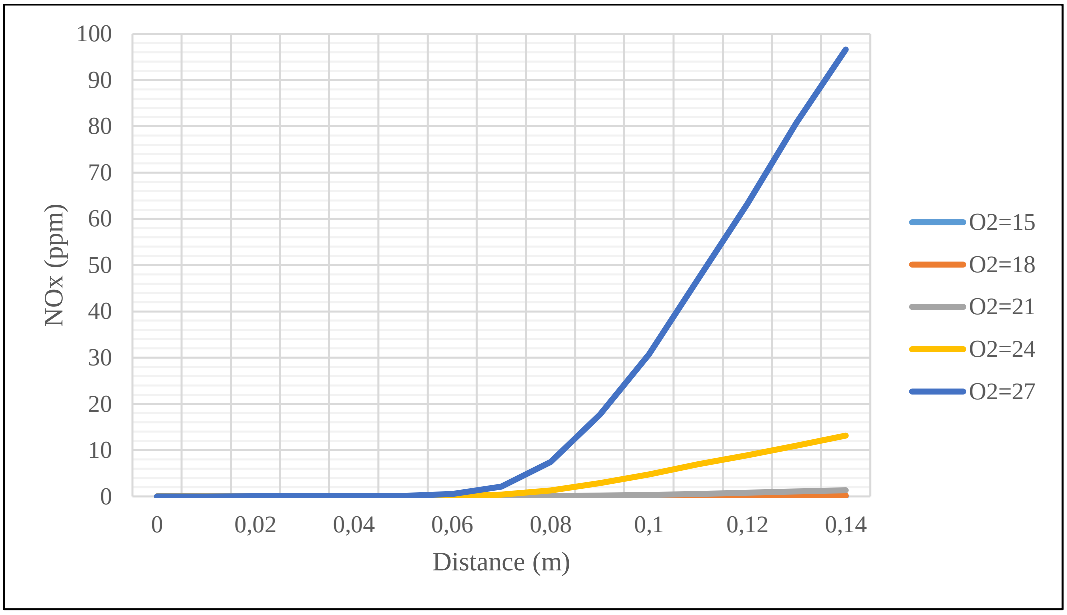

Figure 25. Oxygen concentration effect on NOx emission for 60% CH4 – 40% CO2 combustion across the combustion chamber

4. CONCLUSION

This study has focused on the usability of biogas in residential hermetic-type gas-burning devices. For this purpose, experimental and numerical studies have been performed. Biogas fuel mixtures of 60% CH_4 - 40% CO_2, 70% CH_4 - 30% CO_2, 80% CH_4 - 20% CO_2, and 90% CH_4 - 10% CO_2 by volume were used to seek their combustion characteristics in the combi boiler combustor and the outlet. Temperature and NOx profiles were first measured, and their results were compared with the predictions performed using the standard k-ε turbulence model and the Eddy Dissipation Concept. During the modelings, the air inlet temperatures and the oxygen concentration in the oxidizer were also varied to examine different combustion conditions throughout the combustor and at the outlet. The following conclusions are obtained from the results predicted:

- Although there are slight differences between the measured data and the predicted results due to possible differences in the usage of combustion air during the experiments, it is concluded that the predicted temperature and NO_X profiles are in good agreement with the measurements on the temperature and the NO_X values in terms of trends and values.
- It is revealed that CH_4 concentration in the fuel has a considerable effect on the flame temperatures. As the methane concentration in the fuel mixture decreases, the flame temperatures decrease gradually, and as a result, the predicted NOx levels decrease due to the Zeldovich Mechanism.
- It is determined that the flame temperature levels increase with the air inlet temperature (preheating effects) for each fuel mixture. This results in the flame position upstream of the burner due to the higher laminar burning velocity of the mixture. As a result, the predicted NOx levels increase because of the thermal NO_X mechanism.
- A similar conclusion occurs when the oxygen concentration in the oxizider is increased, the flame temperature levels increased. As a result of oxygen enrichment, the nitrogen in the air reacts with more oxygen to form NOx, especially around the reaction zone. The highest NOx comes out as a result of the highest oxygen enrichment. and as a result of this, the NOx levels are increased for the same reason.

In conclusion, the present study experimentally and numerically presents that the CH_4 – CO_2 fuel mixtures as biogas fuel can be consumed in residential hermetic-type gas-burning devices. This can be done without any modification on the burner and the combustor. In this way, biogas that can be quickly produced can be replaced with natural gas, which is continuously running out and not evenly distributed worldwide, in domestic devices.

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DECLARATION OF ETHICAL STANDARDS

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

AUTHORS’ CONTRIBUTIONS

Ahmet Haşim TOSLAK: Implementation of experiments, interpretation of the results and preparation of the article.

Doç. Dr. Halil İbrahim VARİYENLİ: Implementation of experiments, interpretation of the results and preparation of the article.

Doç. Dr. Serhat KARYEYEN: Implementation of experiments, interpretation of the results and preparation of the article.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

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