

Article

Effects of Methane Addition on Combustion Flow Field and Combustion Characteristics of Ethanol

Hong-Tao Tang ^{1,2}, Zi-Hao Zhang ^{1,2,*}, Zhe Yang ³, Fa-Rui Zhao ^{1,2}  and Yu-Liang Liu ¹ 

¹ College of Mechanical Engineering, Tianjin University of Science & Technology, Tianjin 300222, China

² Key Laboratory of Integrated Design and Online Monitoring of Light Industry and Food Engineering Machinery and Equipment in Tianjin, Tianjin 300222, China

³ China National Accreditation Service for Conformity Assessment, Beijing 100062, China

* Correspondence: zhangzihao1@mail.tust.edu.cn

Abstract

This study employs numerical simulations to systematically investigate the combustion characteristics of methane/ethanol blended fuel–air mixtures under non-premixed turbulent conditions. The effects of the methane blending ratio on the flow-field structure, flame morphology, NO emissions, and combustion efficiency are analyzed. The results show that, with increasing methane blending ratio, the recirculation mechanism gradually shifts from near-field local entrainment to far-field transport, accompanied by a reduction in local shear intensity. Methane addition enhances flame intensity, accelerates combustion, shortens flame length, mitigates heat transfer limitations, and reduces combustion delay. At the initial 10% and the final 20% of the methane blending range, the combustion process exhibits pronounced instability. Methane addition significantly suppresses NO formation, with temperature being the dominant controlling factor, while fuel composition also plays an important role. The overall combustion efficiency is improved. However, a slight decrease is observed at low blending ratios (0–0.1), and the enhancement becomes marginal when the methane blending ratio exceeds 0.6.

Keywords: ethanol; methane blending; non-premixed combustion; flow field structure; instability characteristics

1. Introduction

In recent years, with the increasing instability of global fossil fuel supply and the growing severity of greenhouse gas emissions, the development of clean and efficient alternative fuels has become an important research direction in the field of combustion. Ethanol, as a typical biomass fuel, possesses the advantages of wide availability and strong renewability [1–8], demonstrating promising application prospects in transportation and industrial combustion systems. However, ethanol fuel still exhibits certain limitations in practical applications. For example, its flame propagation speed is relatively low, and under non-premixed combustion conditions, it is highly sensitive to flow structures and mixing states, which may result in reduced combustion efficiency and insufficient combustion stability [9–12]. To improve the combustion characteristics of ethanol, extensive studies have been conducted in recent years through approaches such as fuel blending.

Blending ethanol with highly reactive fuels, such as hydrogen, has been considered an effective approach to improving the combustion characteristics of ethanol. Zhou et al. [13] reported that the addition of hydrogen to ethanol flames can significantly enhance the laminar burning velocity. However, Li et al. [14] found that hydrogen addition to ethanol/air



Academic Editors: Cheng Shi and Zezhou Guo

Received: 12 May 2026

Revised: 11 June 2026

Accepted: 4 August 2026

Published: 6 August 2026

Copyright: © 2026 by the authors.

Licensee MDPI, Basel, Switzerland.

This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\) license](https://creativecommons.org/licenses/by/4.0/).

premixed flames markedly intensifies the intrinsic instability of the flame. With the increase in hydrogen blending ratio, flame cracks and cellular structures appear earlier, and the critical instability radius decreases significantly.

Methane, as the primary component of natural gas, possesses the characteristics of a high hydrogen-to-carbon ratio, clean combustion, and relatively low carbon dioxide emissions [15–18]. It is therefore regarded as an important transitional fuel for achieving low-carbon energy utilization, and its blended combustion with ethanol has attracted increasing attention in recent years. Blending methane with ethanol can not only alleviate the slow combustion characteristics and insufficient flame stability of ethanol, but also combine the advantages of renewable liquid fuels and clean gaseous fuels. Therefore, methane–ethanol blended fuels exhibit considerable application potential in industrial combustors, gas turbines, and dual-fuel combustion systems. Wang et al. [19] experimentally and numerically investigated ethanol/hydrogen/methane ternary blended fuels and demonstrated that fuel composition exerts a significant influence on the laminar burning velocity. An increase in ethanol content reduces the laminar burning velocity and shifts the peak value toward a higher equivalence ratio. Furthermore, Chen et al. [20] further extended this research using the spherical expanding flame method. By introducing a virtual species, FH_2 , to quantitatively separate the chemical contribution of hydrogen, they found that hydrogen mainly promotes radical generation through key elementary reactions, such as R1 ($\text{H} + \text{O}_2 = \text{O} + \text{OH}$) and R2 ($\text{O} + \text{H}_2 = \text{H} + \text{OH}$), thereby accelerating the combustion process. Their results also confirmed, from a chemical kinetics perspective, that increasing the ethanol proportion leads to a reduction in laminar burning velocity and a shift toward higher equivalence ratios. Lee et al. [21] employed the OpenFOAM platform to systematically investigate the mechanism by which ethanol blending affects soot evolution in methane laminar diffusion flames. The results showed that ethanol addition regulates the chemical formation fluxes of C_2H_2 and polycyclic aromatic hydrocarbons (PAHs) on the fuel side, thereby increasing the soot volume fraction and particle number density in diffusion flames. This trend reveals the chemical regulatory effect of ethanol on soot formation in diffusion flames. Fang et al. [22] reported that the introduction of methane into ethanol fuel exhibits a pronounced non-monotonic influence on thermoacoustic oscillation intensity. Under premixed combustion conditions, the pressure fluctuation intensity reached its maximum when the methane blending ratio increased to 30%, whereas when the methane blending ratio further increased to 50%, the blended flame exhibited relatively stable combustion behavior under all tested conditions.

Although the aforementioned studies have made important progress in the kinetic characteristics and pollutant formation mechanisms of ethanol-blended combustion, existing research has mainly focused on the combustion characteristics of premixed laminar fuel–air flames. Systematic investigations of the flow-field characteristics, flame structure, and overall combustion performance of ethanol–methane blended fuels under non-premixed turbulent combustion conditions remain relatively limited. Therefore, the present study employs numerical simulations to systematically investigate the effects of different methane blending ratios on the flow-field characteristics, flame structure, and overall combustion performance within the combustion chamber. Particular attention is paid to the coupling among fuel composition, flow-field evolution, and combustion characteristics over the entire range of methane blending ratios. Furthermore, the sensitive blending-ratio ranges associated with combustion instability are identified and analyzed. The results provide new insights into the combustion mechanisms of methane–ethanol dual-fuel systems and offer theoretical support and engineering guidance for the efficient and low-emission application of ethanol–methane blended fuels in industrial combustors.

2. Computational Domain and Computational Methods

The computational domain model consists of a fuel inlet, an air inlet, a combustion reaction zone, and an outlet. The specific structural dimensions are shown in Figure 1, and the three-dimensional model is shown in Figure 2. The velocity vector plots and temperature contours presented below are all taken from the X–O–Z cross-section of the three-dimensional model.

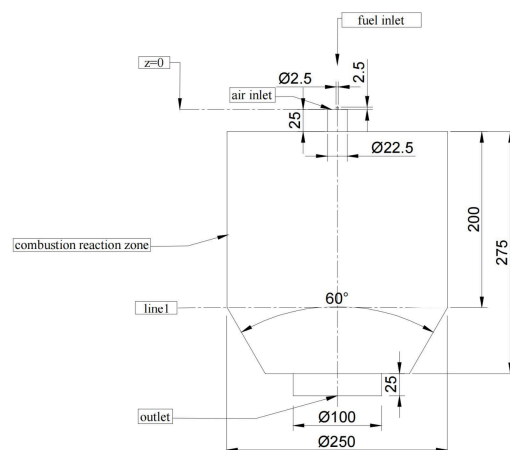


Figure 1. Computational domain.

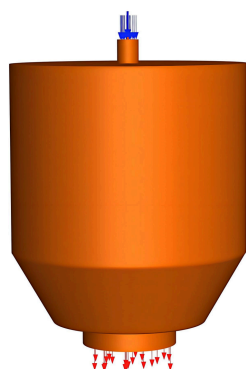


Figure 2. Three-dimensional model.

In the numerical simulation, the Realizable k – ϵ turbulence model, the Eddy Dissipation Concept (EDC) combustion model, and the CKL1.1 mechanism were employed. The CKL1.1 mechanism was constructed based on the minimized reaction network method, applies to high-temperature combustion processes of C_0 – C_4 fuels, and its satisfactory reliability has been validated against experimental data [23]. For the definition of boundary conditions in the computational domain, the fuel and air inlets were set as velocity inlets, and the combustion chamber outlet was set as a pressure outlet. The inlet velocities were determined according to the stoichiometric combustion relationship of ethanol, with approximately 10% excess air supplied to ensure complete combustion and sufficient oxygen availability. The inlet boundary conditions were kept unchanged for all operating conditions. The numerical simulation was performed with an air inlet velocity of 1 m/s and a fuel inlet velocity of 5.1 m/s. To handle the pressure–velocity coupling in solving the Navier–Stokes equations, the Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm was adopted to balance stability and memory efficiency. The spatial discretization of the convective terms in the governing equations was carried out using a second-order upwind scheme [24].

According to the law of conservation of mass, momentum, and energy, the flow process in the cylinder is described by a set of partial differential equations.

Continuous equation of component M [25]:

$$\frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m \cdot \vec{u}) = \nabla \cdot \left[\rho D \nabla \left(\frac{\rho_m}{\rho} \right) \right] + \dot{\rho}_m^c + \dot{\rho}_m^s \delta_{m1} \quad (1)$$

Momentum equation of fluid mixture [25]:

$$\frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla \cdot p - A0 \nabla (2/3 \rho k) + \nabla \cdot \sigma + \vec{F}^s + pg \quad (2)$$

Energy equation of fluid mixture [25]:

$$\frac{\partial (\rho I)}{\partial t} + \nabla \cdot (\rho \vec{u} I) = -p \nabla \cdot \vec{u} + (1 - A0) \sigma \nabla \cdot \vec{u} - \nabla \cdot \vec{J} + A0 \rho \varepsilon + \dot{Q}^c + \dot{Q}^s \quad (3)$$

Multicomponent Fickian diffusion flux equation:

$$\rho Y_i V_i = -\rho Y_i \sum_{j=1}^N D_{ij} d_j = -\rho Y_i \sum_{j=1}^{N-1} (D_{ij} - D_{iN}) d_j \quad (4)$$

where ρ is the mixture density, Y_i is the mass fraction of each component, V_i is the diffusion velocity vector, D_{ij} is the multicomponent diffusion coefficient, and d_j is the diffusion driving force vector, given by $d_j = \nabla X_j$, the gradient of the component mole fraction.

Combustion efficiency, η , is an economic indicator that characterizes the conversion of fuel chemical energy into effective thermal energy. Its core calculation model is based on the principles of carbon balance and energy conservation, and its equation is given by [26]

$$\eta = \left[1 - \frac{\sum_i (q_{m,i} \cdot \text{LHV}_i)}{q_{m,\text{fuel}} \cdot \text{LHV}_{\text{fuel}}} \right] \quad (5)$$

In the equation, the numerator corresponds to the energy loss term of the unburned components, where $q_{m,i}$ is the mass flow rate of the escaped unburned component i (CO, HCHO, CH₃CHO, CH₄, and C₂H₅OH). The lower heating value (LHV) here represents the residual chemical energy of the incomplete combustion products, with $\text{LHV}_{\text{CO}} = 10.1$ MJ/kg, $\text{LHV}_{\text{HCHO}} = 17.3$ MJ/kg, $\text{LHV}_{\text{CH}_3\text{CHO}} = 24$ MJ/kg, $\text{LHV}_{\text{CH}_4} = 50$ MJ/kg and $\text{LHV}_{\text{C}_2\text{H}_5\text{OH}} = 26.8$ MJ/kg [27]. The denominator corresponds to the theoretical energy input term of the fuel, where $q_{m,\text{fuel}}$ is the sum of the mass flow rates of the blended fuel, and its corresponding lower heating value is given by

$$\text{LHV}_{\text{fuel}} = X_{\text{CH}_4} \cdot \text{LHV}_{\text{CH}_4} + (1 - X_{\text{CH}_4}) \cdot \text{LHV}_{\text{C}_2\text{H}_5\text{OH}} \quad (6)$$

In the equation, X_{CH_4} denotes the mass fraction of CH₄.

The computational model employed a Tet/Hybrid mesh. According to the intensity of flow-field variations within the combustion chamber, a non-uniform mesh refinement strategy was implemented. Four mesh schemes with different resolutions, consisting of approximately 1.52 million, 2.04 million, 2.55 million, and 3.10 million cells, were generated for grid independence verification. Finally, considering the optimal balance among numerical accuracy, computational cost, and result reliability, the mesh with 2.55 million cells was selected for subsequent simulations.

3. Results and Discussion

3.1. Structural Characteristics of the Combustion Chamber Flow Field Under Different Methane Blending Ratios

In this study, the methane blending ratio is defined as the molar fraction of methane in the methane/ethanol blended fuel, ranging from 0 (pure ethanol) to 1 (pure methane). As shown in Figure 3, the flow field in the combustion chamber exhibits a dual-driving structure jointly dominated by a central high-velocity jet and a pair of internal recirculation vortices on both sides. A certain number of small-scale vortical structures are distributed in the corner regions near the top of the combustion chamber. As shown in Figure 3a, under pure-ethanol conditions, the flame deviates from the central axis, with a relatively low peak velocity and weak flame intensity. As shown in Figure 3f, at a methane blending ratio of 0.4, the flame gradually contracts toward the central axis, and the peak velocity increases to 6.22 m/s, indicating enhanced combustion stability. As shown in Figure 3i, when the methane blending ratio reaches 0.7, the peak velocity further increases to 6.41 m/s, the strength of the internal recirculation vortices increases, and the high-temperature reaction zone extends downstream. As shown in Figure 3l, under pure methane conditions, the flame tip is the most concentrated, the flame intensity is the strongest, and the small-scale vortices are the most fragmented. In addition, with the increase in methane blending ratio, the vortex core region as a whole moves toward the outlet and expands toward both sidewalls of the combustion chamber, and the radial distance between it and the central jet region gradually increases. This evolution indicates that the recirculation mechanism has shifted from the initial near-field local entrainment mode to a far-field transport-involved mode at a larger spatial scale. Methane blending affects the flow distribution in the combustion chamber by altering the spatial extent of the internal recirculation vortices.

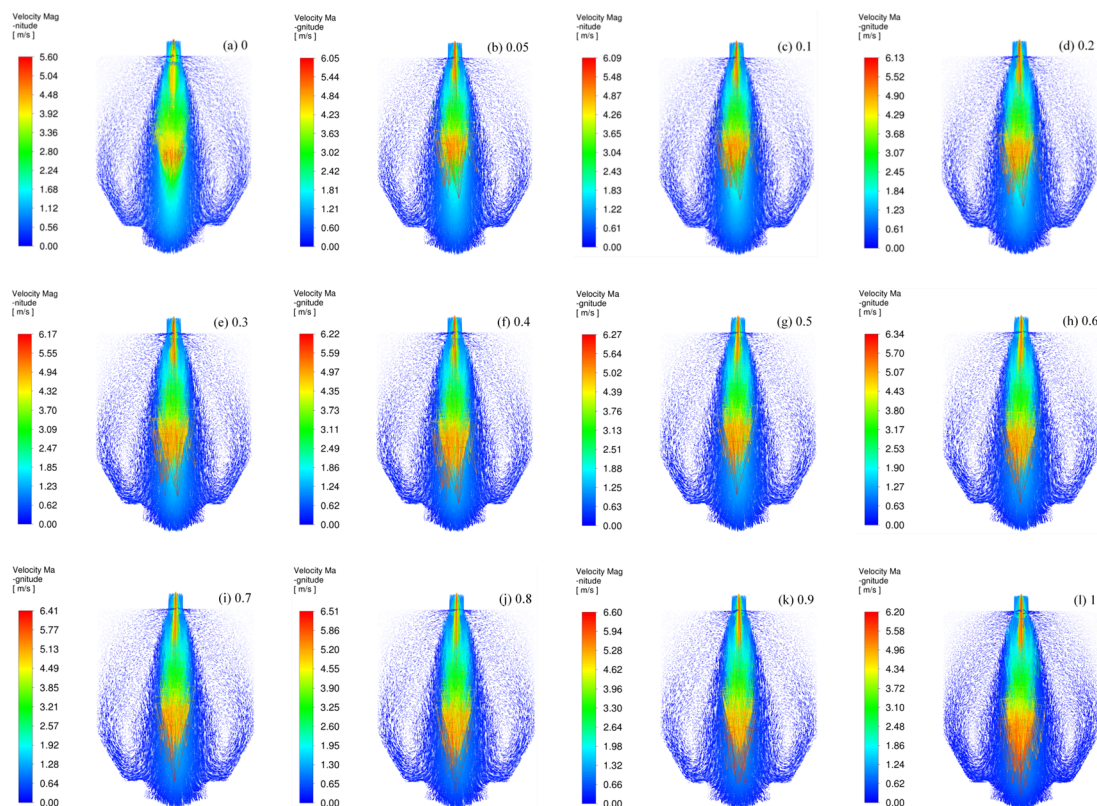


Figure 3. Velocity vector diagrams under different conditions.

Figure 4 presents the axial variation in temperature in the combustion chamber under different methane blending ratios. As shown in Figure 4a, under pure-ethanol conditions, the flame deviates from the central axis and exhibits a relatively long flame length. Heat accumulation occurs near the outlet and develops radially toward both sidewalls of the combustion chamber, forming a heat transfer stagnation region (HTSR). This region causes heat to be transported back upward to the combustion reaction zone, resulting in a relatively large heat recirculation gradient that sustains the heat required for combustion. As shown in Figure 4f, at a methane blending ratio of 0.4, the flame deviation is nearly eliminated, the flame length is shortened, and the HTSR is reduced to a certain extent. As shown in Figure 4i, at a methane blending ratio of 0.7, the flame length is further shortened, the HTSR is further reduced, and the heat recirculation gradient is significantly decreased. As shown in Figure 4l, under pure methane conditions, the flame combustion enhancement is the most concentrated, the flame length is the shortest, the HTSR nearly disappears, and the heat recirculation gradient is the smallest, allowing heat to be effectively transferred outward. With the increase in methane blending ratio, the flame intensity is improved, the HTSR is reduced, the heat recirculation gradient decreases, and the combustion chemical reactions are enhanced.

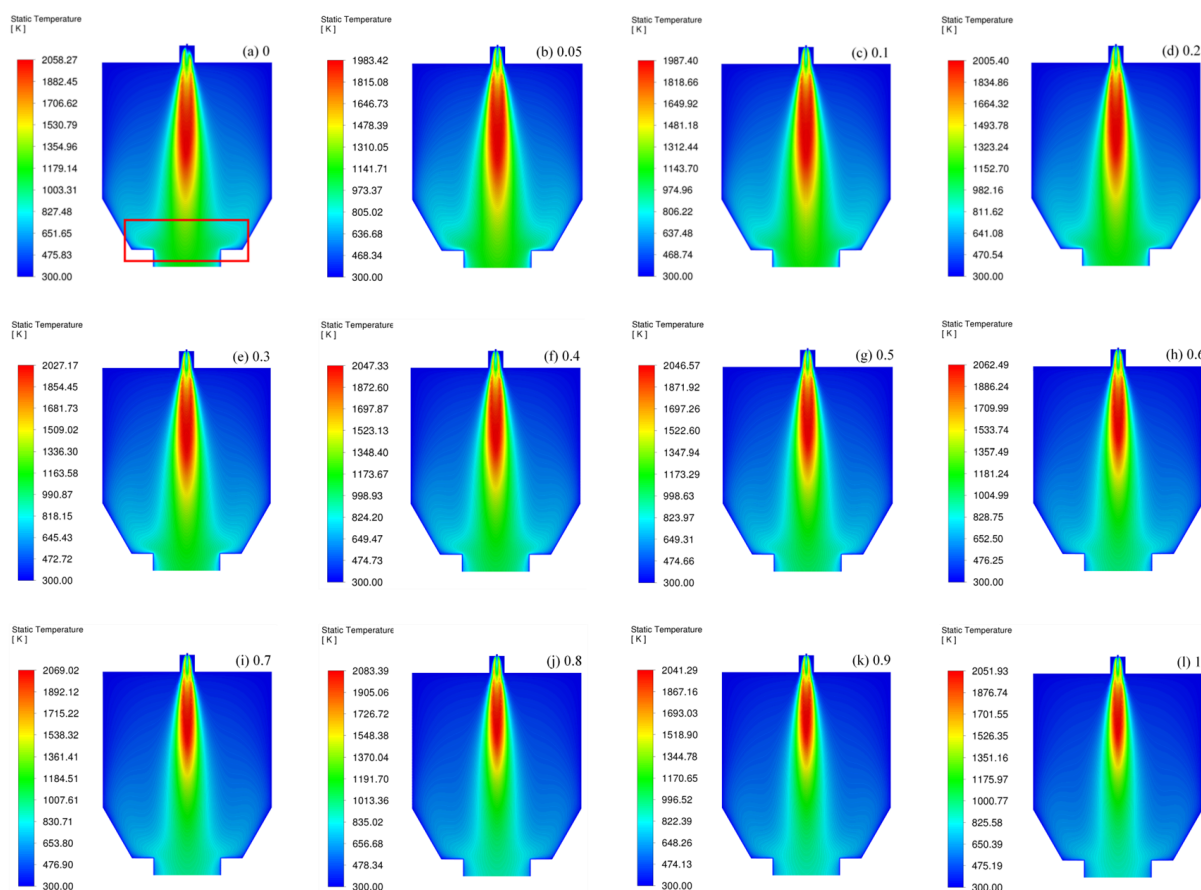


Figure 4. Temperature contours under different conditions.

As shown in Figure 5, within the range of $z = 0\text{--}0.1$ m along the central axis, the temperature curves corresponding to different methane blending ratios are arranged from top to bottom in the following order: 0.9, 0.8, 0, 0.7, 0.6, 0.5, 0.4, 0.3, 0.2, 0.1, 1, and 0.05. Within the range of $z = 0.1\text{--}0.325$ m, the vertical ordering of the temperature curves is exactly reversed, namely: 0.05, 0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 1, 0.8, and 0.9. The reversal of the temperature ranking is closely associated with the shortening of the flame length. As the methane blending ratio increases, the flame length gradually decreases, and the

high-temperature region becomes increasingly concentrated toward the combustor inlet, resulting in higher temperatures for high-methane cases in the upstream region. In contrast, in the downstream region, the temperature gradually becomes lower than that of the low-methane and pure ethanol cases because the spatial extent of the high-temperature zone is reduced. The region $z = 0\text{--}0.1$ m corresponds to the stage from initial combustion to the most intense chemical reaction. The temperature along the central axis exhibits an oscillatory and rapid increase, reflecting the extreme instability of the flame combustion chemical reaction process. This unstable oscillatory behavior promotes mass transfer and accelerates the flame combustion chemical reactions. The region $z = 0.1\text{--}0.15$ m corresponds to the intensified chemical reaction zone, where the central-axis temperature under each condition successively reaches its peak value. For $z > 0.15$ m, the flow enters the energy transfer region, where the combustion reactions are essentially completed, temperature fluctuations gradually decay, and the temperature curves decrease smoothly. However, under the conditions with methane blending ratios of 0.05 and 0.9, the central-axis temperature still exhibits a certain degree of oscillatory behavior after $z > 0.15$ m. The chemical reaction process of methane/ethanol flame combustion is an unstable oscillatory process.

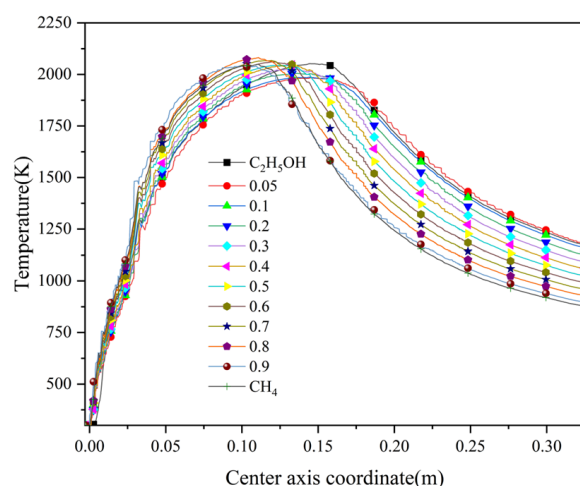


Figure 5. Temperature distribution curves of the central axis under different conditions.

The so-called comprehensive combustion characteristics refer to the combined effects of multiple factors, including the diffusion rates of reactive species, fuel heating value, combustion rate, and the combustion sequence among different components. This concept is introduced to investigate the roles of intrinsic fuel combustion characteristics under different blending ratios. As shown in Figure 6, within the initial 10% range of methane blending ratio (0–0.1), the peak temperature decreases sharply from 2052.04 K under pure ethanol conditions to 1987.03 K at a blending ratio of 0.1. Due to the small size of methane molecules, they are easy to transmit. The mole fraction gradient of methane and ethanol is large, which leads to an accelerated diffusion speed of methane molecules. This results in a non-uniform distribution of methane in the reaction zone, thereby delaying the combustion rate of ethanol. In addition, the relatively low molar heating value of methane weakens the combustion intensity of the methane–ethanol mixture, leading to a rapid decrease in the peak temperature along the central axis. The temperature decrease observed at a methane blending ratio of 0.05 precisely demonstrates the abrupt and drastic nature of this process. Within the blending-ratio ranges of 0.1 to 0.8, the peak temperature along the central axis increases gradually, influenced by the heating value per unit mass of the fuel components. At a methane blending ratio of 0.8, under the combined influence of multiple factors associated with the comprehensive combustion characteristics, the peak temperature reaches a maximum value of 2081.66 K. This indicates that an appropriate methane blending ratio

can effectively enhance the peak temperature of the methane/ethanol blended fuel system. Although the peak temperature curve exhibits certain fluctuations within the blending-ratio ranges of 0.1 to 0.8, a regression analysis was specifically conducted on the above data to enable rapid prediction of the peak temperature under different methane/ethanol blending ratios. The corrected temperature equation is expressed as Equation (7):

$$F_1(x) = -102x^2 + 220.41x + 1966.85 \quad (7)$$

where F_1 denotes the central-axis temperature, and x is the methane blending molar ratio, ranging from 0.1 to 0.8.

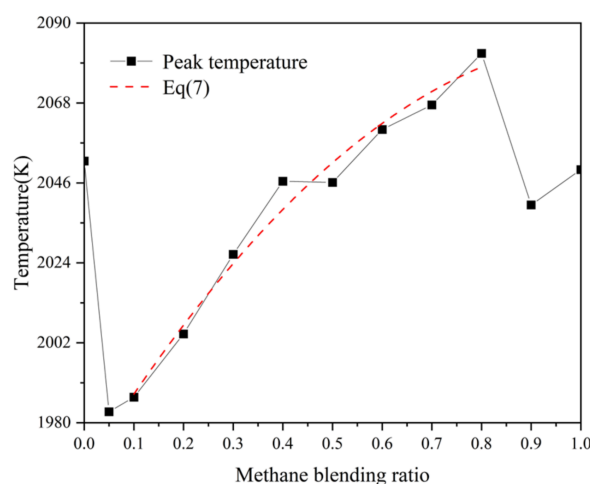


Figure 6. Peak temperatures of the central axis under different conditions.

When the methane blending ratio falls within the last 20% range (corresponding to an ethanol blending ratio of 0–0.2), the peak temperature along the central axis decreases sharply again, dropping from 2081.66 K at the 0.8 condition to 2039.94 K at the 0.9 condition. Owing to the larger molecular size of ethanol, its diffusion rate is lower than that of methane, resulting in an abrupt change in the peak central-axis temperature. Subsequently, the peak temperature increases from 2039.94 K at the 0.9 condition to 2049.66 K under pure methane conditions. Within the initial 10% and final 20% ranges of the methane blending ratio, the peak centerline temperature exhibits pronounced nonlinear variations rather than a monotonic dependence on methane composition. This behavior indicates the existence of blending-ratio-sensitive regions in the methane–ethanol fuel system, where the combustion characteristics are highly sensitive to changes in fuel composition. Similarly, Fang et al. [22] reported that variations in methane blending ratio significantly affect the combustion stability of ethanol–methane flames, with distinct combustion behaviors observed at different blending ratios. However, the locations of the sensitive blending-ratio ranges identified in the two studies are not entirely consistent. This discrepancy may be attributed to differences in combustion mode, flow-field structure, and fuel–air mixing mechanisms.

As shown in Figure 7, with the increase in methane blending ratio, the position of the peak temperature along the central axis exhibits a trend of shifting toward the inlet. However, within the initial 10% range of methane blending ratio, due to the influence of the diffusion rate of the low-fraction methane component, the peak position remains nearly unchanged at approximately $z = 0.144$ m. Under this condition, the flame length is the longest, and the flame intensity is the weakest. Within the methane blending-ratio ranges of 0.1–0.9, the peak temperature position along the central axis shifts overall toward the inlet, reaching the shortest distance of $z = 0.1$ m at the 0.9 condition. At this point,

the flame length is the shortest, and the flame intensity is the strongest. Within this range, the peak temperature position along the central axis is mainly governed by the molar heating value of the fuel components. Similarly, in order to rapidly predict the peak temperature position under different methane/ethanol blending ratios, a regression analysis is performed based on the above data, and the corresponding corrected position equation is expressed as Equation (8):

$$F_2(x) = -0.04826x + 0.1482 \quad (8)$$

where F_2 denotes the axial position (z-coordinate) of the peak temperature along the central axis, and x is the methane blending molar ratio, ranging from 0.1 to 0.9.

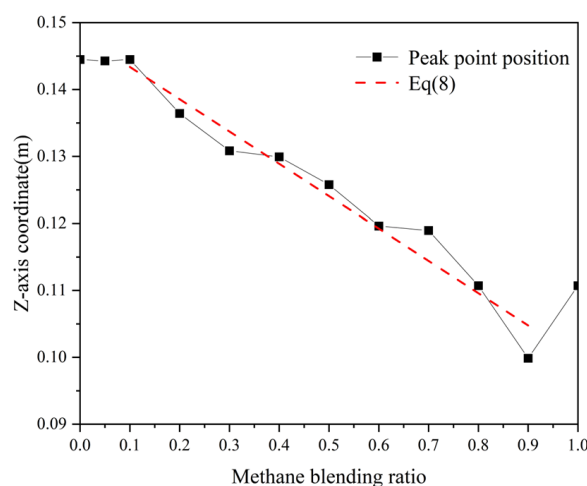


Figure 7. The peak position of the center axis temperature under different conditions.

Within the last 20% range of the methane blending ratio, the position of the peak temperature along the central axis increases from $z = 0.1$ m at the 0.9 condition to $z = 0.11$ m under pure methane conditions, due to the influence of the diffusion rate of the low-fraction ethanol component. Correspondingly, the flame length exhibits a certain degree of increase.

As shown in Figure 8, within the range of $z = 0$ – 0.0025 m, the center axis velocity of each condition rapidly increases and quickly reaches its peak. The velocity peak of the pure-ethanol condition is the lowest. Within the range of $z = 0.0025$ – 0.03 m, the central-axis velocities for different blending ratios are ranked from high to low as follows: 0.9, 0.8, 0.7, 0.6, 0.5, 0.4, 0.3, 0.2, 0.1, 1, 0.05, and 0. Under pure-ethanol conditions, the velocity remains relatively low and varies smoothly, indicating that the combustion chemical reaction in this region is relatively slow and delayed. This behavior can be attributed to the relatively complex molecular structure of ethanol, whose decomposition and oxidation processes involve more intermediate reaction steps. In addition, the diffusivity of ethanol is lower than that of methane, resulting in slower fuel–air mixing and consequently a noticeable delay in heat release. In contrast, under methane-blended conditions, this smooth transition region disappears. The addition of methane significantly enhances the combustion reaction rate of ethanol. As a small-molecule fuel, methane possesses a higher diffusion rate, enabling faster interaction with air and earlier participation in combustion reactions. Meanwhile, it promotes the formation of local high-temperature regions, thereby shortening the ignition-delay period of ethanol combustion and advancing the onset of the combustion process. The higher diffusivity of methane enhances the turbulent mixing efficiency between fuel and air, bringing the local mixing timescale closer to the chemical reaction timescale. Meanwhile, methane addition promotes the formation and transport of reactive radicals such as H, O, and OH, thereby accelerating

local oxidation reactions, which strengthens the coupling between turbulent mixing and chemical reactions. Within the range of $z = 0.03\text{--}0.05\text{ m}$, the central-axis velocity under pure-ethanol conditions increases again and becomes higher than that under methane-blended conditions. Within the range of $z = 0.05\text{--}0.325\text{ m}$, the velocity curves gradually decrease smoothly. The central-axis velocities are ranked from high to low as follows: 0, 0.05, 1, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, and 0.9. This indicates that methane addition not only increases the combustion reaction rate of ethanol but also causes the main reaction zone to shift upstream, resulting in a more concentrated distribution of the high-temperature and heat-release regions. As the combustion process is completed earlier, the velocity gradients and reaction-intensity fluctuations in the downstream region are weakened, leading to a smoother variation in the velocity curves. This indicates that methane blending improves the matching between turbulent mixing and chemical kinetics, reducing the sensitivity of the combustion process to local flow disturbances and consequently resulting in more stable combustion characteristics. Therefore, methane addition enhances the intensity of ethanol combustion chemical reactions, alleviates its inherently slow combustion behavior, improves the relative ranking of combustion velocity, and promotes a more stable behavior in the later stages of the chemical reaction process.

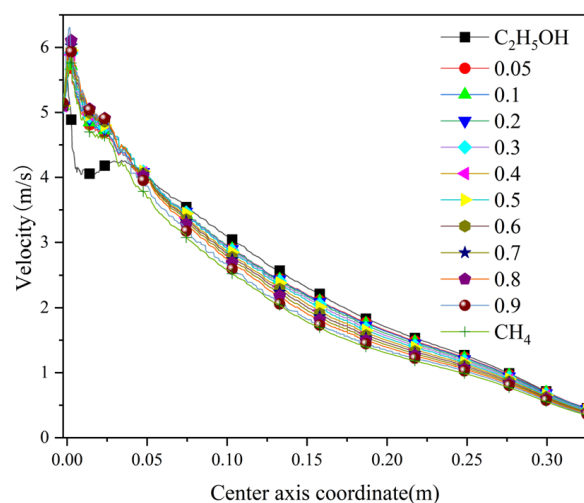


Figure 8. Velocity distribution curves of the central axis under different conditions.

Vorticity is defined as the curl of the velocity field and characterizes the local rotational intensity of fluid elements. The magnitude of vorticity is closely related to the shear intensity between the fuel jet and the surrounding fluid, as well as the mixing strength. As shown in Figure 9, along the central axis of the combustion chamber, the vorticity distribution exhibits two peak regions, one larger and one smaller. The first peak region appears near the junction of fuel and air at approximately $z = 0.0025\text{ m}$. It is mainly influenced by the density gradient and fluid expansion induced by combustion, as well as the shear effect between the high-velocity fuel jet and the low-velocity air. The second peak region appears near $z = 0.03\text{ m}$ in the upper part of the combustion chamber. This region corresponds to the interaction zone between the internal recirculation vortices and the central jet, where a velocity gradient is formed between the recirculating flow and the jet, resulting in a vorticity peak. However, under pure-ethanol conditions, the second peak region does not appear, and the intensity of the combustion chemical reaction is relatively low. Therefore, the addition of methane promotes the formation of a stronger recirculation–jet shear layer in the upper part of the combustion chamber, thereby enhancing local mixing in this region.

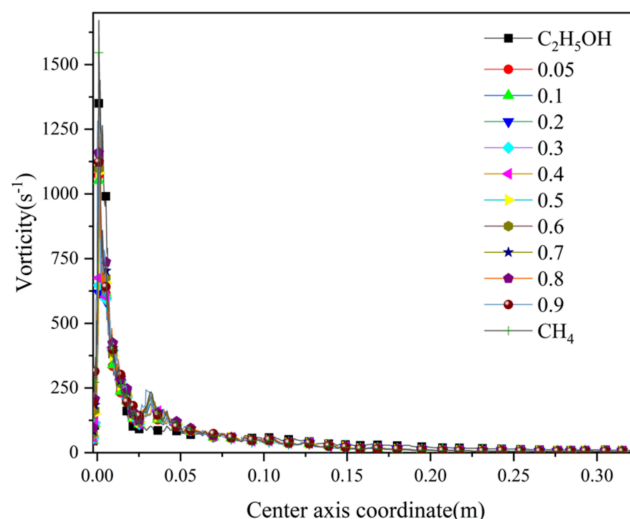


Figure 9. Vorticity distribution map of the central axis under different conditions.

As shown in Figure 10, the heat release rate along the central axis exhibits a distinct multi-peak structure, where different peaks correspond to reaction processes occurring at different spatial locations between the fuel and air. The first peak appears in the initial interaction region between fuel and air. Under methane-blended conditions, the peak position of the heat release rate is located near $z = 0.0025$ m. In contrast, under pure-ethanol conditions, the peak position is located at $z = 0.0097$ m, with a maximum heat release rate of 0.059 W. This indicates that heat release during pure ethanol combustion is relatively slow and exhibits a noticeable delay. The addition of methane promotes the generation and transport of chain-carrying radicals such as H, O, and OH, which accelerate the oxidation reactions and increase the local heat release rate. Consequently, the combustion rate of the flame is enhanced, effectively alleviating the delayed chemical reaction behavior of pure ethanol combustion. The second peak appears in the upper region of the combustion chamber ($z = 0.03$ – 0.04 m). Within the initial 10% range of methane blending ratio, the heat release rate reaches its highest value. This is attributed to the high diffusion rate of the low-fraction methane component, which enhances local mixing between fuel and air. In contrast, the peak heat release rate under pure-ethanol conditions remains at a relatively low level. The third peak is located in the middle region of the combustion chamber ($z = 0.11$ – 0.17 m). With the increase in the methane blending ratio, the peak position gradually shifts toward the inlet, consistent with the movement trend of the peak temperature along the central axis. The order of peak appearance is: 1, 0.9, 0.8, 0.7, 0.6, 0.5, 0.4, 0.3, 0.2, 0, 0.1, and 0.05. The main reaction zone is compressed upstream, and the peak position of the heat release rate becomes more concentrated toward the flame core. Overall, an appropriate addition of methane effectively enhances the combustion rate of the flame, alleviates the delayed chemical reaction behavior of pure ethanol combustion, and promotes a more concentrated and intensified flame structure.

As shown in Figure 11, the flame thickness exhibits a non-monotonic variation with increasing methane blending ratio, characterized by an initial increase followed by a subsequent decrease. Under non-premixed fuel/air combustion conditions, the flame thickness is governed by the combined effects of species diffusion and chemical reaction rates. Within the methane blending-ratio ranges of 0–0.2, the addition of a low fraction of methane enhances the diffusion capacity of the blended fuel, resulting in an increase in flame thickness. Within the range of 0.2–1, as the methane blending ratio continues to increase, the overall reactivity of the methane/ethanol mixture is enhanced, the core reaction layer becomes narrower, the reaction zone becomes more concentrated, and the

flame thickness decreases progressively. To facilitate rapid prediction of flame thickness, a regression analysis is conducted based on the above data, and the corresponding corrected flame thickness equation is given by Equation (9):

$$F_3(x) = -4.995x + 26.15 \quad (9)$$

where F_3 denotes the flame thickness, and x is the methane blending molar ratio, ranging from 0.2 to 1.

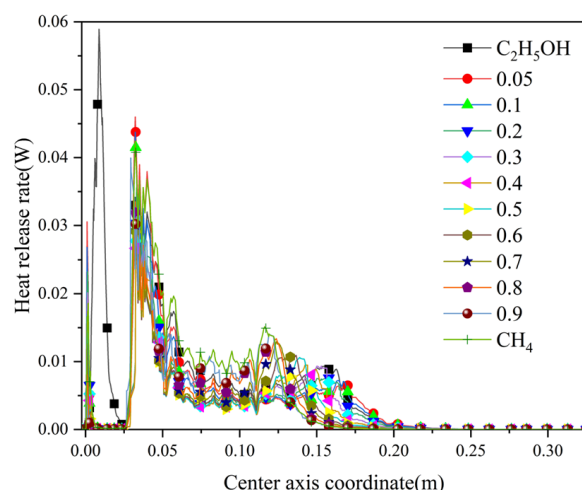


Figure 10. Distribution map of heat release rate along the central axis under different conditions.

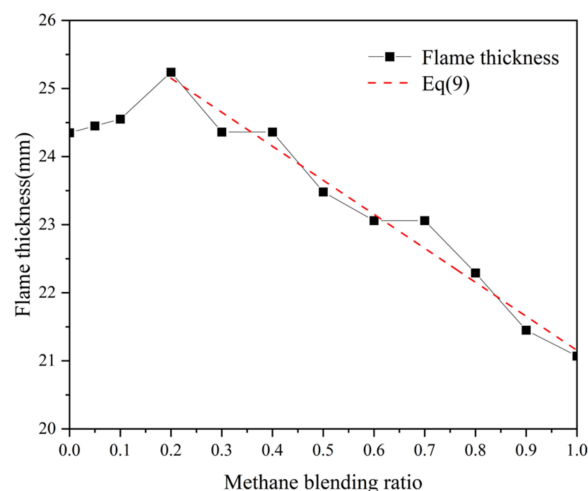


Figure 11. Flame thickness under different conditions.

As shown in Figure 12, the vorticity distribution along line 1 exhibits a distinct double-peak structure on both sides of the central axis. The left and right peak locations correspond to the shear interaction regions between the large-scale vortical structures on both sides of the central axis and the central jet. Due to the large velocity gradients in these regions, relatively high vorticity is generated. At the central axis, the interaction between counter-rotating vortices from both sides reduces the local velocity gradient, resulting in a low-vorticity region. In comparison, the vorticity peak on the left side of the central axis is higher than that on the right side, leading to a relatively higher residual vorticity near the right wall of the combustion chamber. This asymmetric distribution of vorticity peaks on both sides causes the large-scale vortices to transport recirculating reactants and products from the left side to the right side through the flame core region via internal recirculation, accompanied by mass and heat transfer. With the increase in the methane blending ratio,

the vorticity peaks exhibit an overall decreasing trend. This indicates that the local shear intensity between the central jet and the counter-rotating vortices on both sides gradually weakens with increasing methane blending ratio.

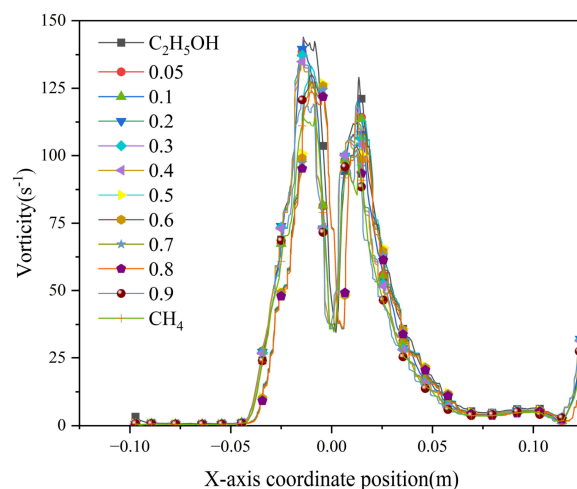


Figure 12. Vorticity distribution map along line 1 under different conditions.

As shown in Figure 13, although there are slight differences between the simulated and experimental values of the average outlet velocity, the overall trends are consistent. The average outlet velocity of the combustion chamber exhibits an overall decreasing trend as the methane blending ratio increases. Under pure-ethanol conditions, owing to the higher molar heating value of ethanol compared with methane, the overall energy input of the system is greater under the same molar flow rate, resulting in a higher average outlet velocity. Within the methane blending-ratio ranges of 0–0.1, the relatively low molar heating value and rapid diffusion of methane lead to a sharp decrease in the average outlet velocity. This indicates that the addition of a low fraction of methane has a significant influence on the average outlet velocity. Within the range of 0.1–1, the average outlet velocity decreases approximately linearly with increasing methane blending ratio. To enable rapid prediction of the average outlet velocity, a regression analysis is conducted based on the above data, and the corresponding corrected velocity equation is expressed as Equation (10):

$$F_4(x) = -0.0588x + 0.2405 \quad (10)$$

where F_4 denotes the average outlet velocity, and x is the methane blending molar ratio, ranging from 0.1 to 1.

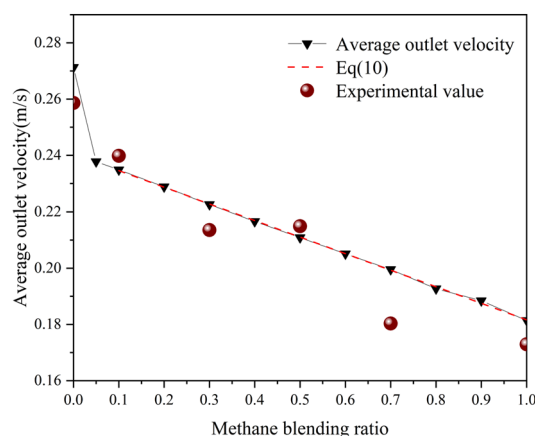


Figure 13. Average outlet velocity under different conditions.

As shown in Figure 14, the average outlet temperature agrees well with the experimental values. The average outlet temperature decreases linearly with increasing methane blending ratio. This indicates that the average outlet temperature is weakly affected by the diffusion rates of the low-fraction methane/ethanol components and is mainly governed by the molar heating value of the fuel components. To enable rapid prediction of the average outlet temperature, a regression analysis is conducted based on the above data, and the corresponding corrected temperature equation is expressed as Equation (11):

$$F_5(x) = -247.825x + 1013.169 \quad (11)$$

where F_5 denotes the average outlet temperature, and x is the methane blending molar ratio, ranging from 0 to 1.

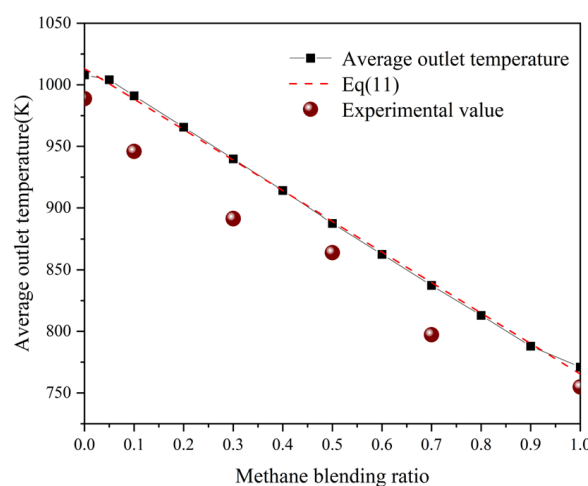


Figure 14. Average outlet temperature under different conditions.

3.2. Overall Performance Characteristics of the Combustion Chamber Under Different Methane Blending Ratios

As shown in Figure 15, along the central axis of the combustion chamber, the NO mass fraction exhibits a trend of rapid increase followed by a gradual decrease. Within the region of $z = 0-0.025$ m, the flame is still in the initial reaction stage, and the central-axis temperature is relatively low, which is insufficient for NO formation; therefore, the NO mass fraction is approximately zero. Within $z = 0.025-0.1$ m, as combustion develops, the temperature increases progressively, resulting in a rapid increase in NO formation. Within the region of $z = 0.1-0.2$ m, corresponding to the high-temperature reaction zone of the flame, the NO mass fraction under different conditions successively reaches its peak value. In conjunction with Figure 5, it can be observed that the temperature range of approximately 1579–2049 K along the central axis corresponds to the rapid NO formation region. High temperature provides the dominant driving force for NO formation. Among all conditions, the peak NO mass fraction is highest under pure-ethanol conditions, while methane addition significantly reduces the NO mass fraction. With increasing methane blending ratio, the peak position of NO gradually shifts toward the inlet. Within the region of $z = 0.2-0.325$ m, the decrease in temperature suppresses NO formation reactions, while the dilution effect induced by flow mixing is enhanced. As a result, the NO mass fraction gradually decreases and tends to stabilize near the outlet. The central-axis NO mass fractions under different conditions are ranked from high to low as follows: 0, 0.8, 0.4, 0.6, 0.7, 0.5, 0.3, 0.2, 1, 0.1, 0.05, and 0.9. Therefore, the variation in NO mass fraction along the centerline generally follows the temperature distribution, indicating that temperature remains the primary controlling factor for NO formation. However, changes

in the methane/ethanol fuel composition not only modify the temperature field but also affect the local reaction environment and the concentration of reactive species, thereby exerting an additional influence on NO formation.

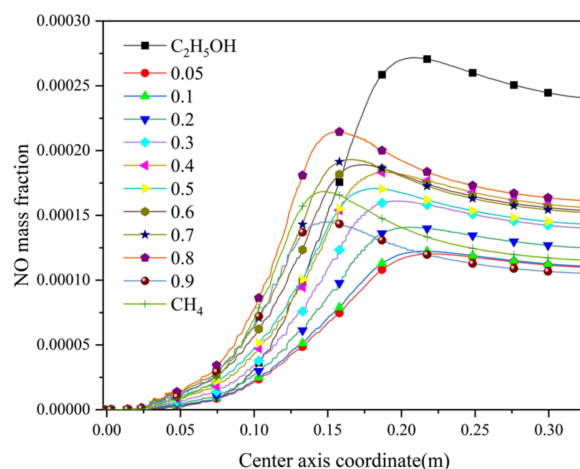


Figure 15. Distribution curves of NO mass fraction along the central axis under different conditions.

As shown in Figure 16, within the initial 10% range of methane blending ratio, the outlet NO mass fraction decreases sharply from 2.367×10^{-4} under pure-ethanol conditions to 1.1×10^{-4} at a blending ratio of 0.1. As indicated in Figure 6, the rapid diffusion of the low-fraction methane component leads to a rapid decrease in temperature, thereby significantly reducing NO formation within the combustion chamber. Within the methane blending-ratio ranges of 0.2–0.8, the peak temperature in the combustion chamber gradually increases due to the influence of the heating value per unit mass of the fuel components, resulting in an increase in NO formation and a corresponding gradual rise in the outlet NO mass fraction. To enable rapid prediction of the outlet NO mass fraction, a regression analysis is conducted based on the above data, and the corresponding corrected outlet NO mass fraction equation is expressed as Equation (12):

$$F_6(x) = 0.0004001x^3 - 0.000658x^2 + 0.000372x + 0.0000769 \quad (12)$$

where F_6 denotes the NO mass fraction at the outlet, and x is the methane blending molar ratio, ranging from 0.1 to 0.8.

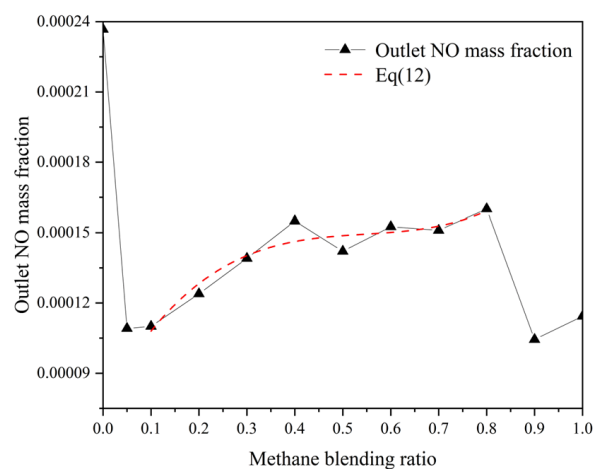


Figure 16. The mass fraction of NO at different conditions.

Within the last 20% range of the methane blending ratio, the outlet NO mass fraction decreases sharply from 1.6×10^{-4} at the 0.8 condition to 1.04×10^{-4} at the 0.9 condition, under the influence of the diffusion rate of the low-fraction ethanol macromolecules. Subsequently, it increases slightly to 1.14×10^{-4} under pure methane conditions.

As shown in Figure 17, within the methane blending-ratio ranges of 0–0.1, the combustion efficiency exhibits a slight decrease. With a further increase in the methane blending ratio, the combustion efficiency gradually increases. However, when the methane blending ratio exceeds 0.6, the increase in combustion efficiency becomes limited. To enable rapid prediction of combustion efficiency, a regression analysis is conducted based on the above data, and the corresponding corrected combustion efficiency equation is expressed as Equation (13):

$$F_7(x) = 1.7344x^3 - 4.2655x^2 + 3.5776x + 98.9151 \quad (13)$$

where F_7 denotes the combustion efficiency, and x is the methane blending molar ratio, ranging from 0.1 to 1.

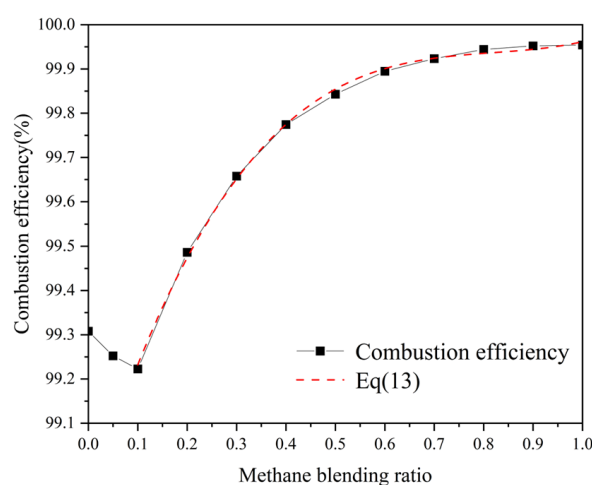


Figure 17. Combustion efficiency under different conditions.

The above results indicate that methane blending enhances the combustion efficiency of ethanol–air mixtures under non-premixed conditions. However, within the methane blending-ratio ranges of 0–0.1, the combustion efficiency is reduced to a certain extent, and beyond a blending ratio of 0.6, the improvement becomes limited.

In Table 1, R^2 represents the coefficient of determination, Adj. R^2 represents the adjusted coefficient of determination, and p -value represents the significance probability. As shown in Table 1, all regression equations exhibit high coefficients of determination ($R^2 > 0.92$) and adjusted coefficients of determination (Adj. $R^2 > 0.91$), and all pass the significance test at the $p < 0.05$ level. These results indicate that the proposed regression models possess good fitting accuracy and statistical significance and can adequately describe the quantitative relationships among the variables.

Table 1. Statistical quality parameters of the regression equations.

Eq	R^2	Adj. R^2	F-Value	p -Value
Equation (7)	0.9816	0.97425	133.39598	4.59×10^{-5}
Equation (8)	0.95603	0.94975	152.20081	5.28×10^{-6}
Equation (9)	0.97351	0.96973	257.25429	8.90×10^{-7}
Equation (10)	0.9841	0.9833	13,502.26864	3.36×10^{-14}
Equation (11)	0.9873	0.9861	7887.97788	8.01×10^{-16}
Equation (12)	0.9248	0.91839	16.39644	0.01033
Equation (13)	0.988	0.982	1664.7084	3.78×10^{-9}

4. Conclusions

(1) Methane blending affects the flow-field distribution within the combustion chamber. With increasing methane blending ratio, the internal recirculation gradually transitions from a near-field local entrainment mode to a far-field transport mode, thereby enhancing the overall regulation of the reaction zone by the recirculation flow.

(2) Methane blending alleviates the delayed chemical reaction behavior of ethanol combustion, enhances flame intensity, shortens the flame length, increases the combustion rate, and reduces the heat transfer stagnation region. The overall combustion efficiency is improved, although a decrease occurs within the low-blending-ratio ranges of 0–0.1, and further increases become limited beyond a blending ratio of 0.6.

(3) Although methane improves the combustion characteristics of ethanol, a sensitive interval of the methane blending ratio exists. Under the initial 10% and the final 20% ranges of the methane blending ratio, the combustion process exhibits pronounced instability. The comprehensive combustion characteristics of low-fraction methane or ethanol components influence the instability of the combustion process.

(4) The addition of methane significantly reduces NO formation during ethanol combustion. NO formation is affected by the comprehensive combustion characteristics of the methane/ethanol system, with temperature playing a dominant role, while fuel composition also exerts an influence.

In summary, methane blending is an effective approach for improving the combustion performance of ethanol-fueled systems. Within an appropriate blending range, methane can enhance flame stability, accelerate combustion reactions, and improve combustion efficiency. These characteristics are beneficial for industrial combustors, gas turbines, and dual-fuel burner systems utilizing ethanol or alcohol-based fuels. However, the results also indicate the existence of instability characteristics in the low-methane-fraction range (0–0.1) and high-methane-fraction range (0.8–1.0). Therefore, these blending ranges should be avoided as much as possible in practical applications. In addition, the present study was conducted using a laboratory-scale combustor under fixed operating conditions. Further investigations considering larger-scale systems, variable operating conditions, and fuel supply fluctuations are still required.

Author Contributions: Conceptualization, H.-T.T. and Z.-H.Z.; methodology, H.-T.T. and Z.-H.Z.; software, Z.-H.Z. and F.-R.Z.; validation, Z.Y. and Y.-L.L.; formal analysis, H.-T.T. and Z.-H.Z.; investigation, H.-T.T. and Z.-H.Z.; resources, H.-T.T. and Z.-H.Z.; data curation, H.-T.T. and Z.-H.Z.; writing—original draft preparation, H.-T.T. and Z.-H.Z.; writing—review and editing, H.-T.T. and Z.-H.Z.; visualization, F.-R.Z., Z.Y. and Y.-L.L.; supervision, F.-R.Z., Z.Y. and Y.-L.L.; project administration, H.-T.T.; funding acquisition, H.-T.T. and Y.-L.L. All authors have read and agreed to the published version of the manuscript.

Funding: This study was supported by the National Natural Science Foundation of China (No. 52378254, 2024).

Data Availability Statement: The data presented in this study are available on request from the corresponding author as most of the data is confidential; whether it can be disclosed will depend on the circumstances.

Acknowledgments: We are grateful to the Tianjin Key Laboratory of Integrated Design and Online Monitoring of Light Industry and Food Engineering Machinery & Equipment.

Conflicts of Interest: The authors declare no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript:

PAHs	Polycyclic Aromatic Hydrocarbons
EDC	Eddy Dissipation Concept
SIMPLE	Semi-Implicit Method for Pressure-Linked Equations
LHV	lower heating value
HTSR	heat transfer stagnation region

References

- Gupta, K.; Rehman, A.; Sarviya, R.M. Bio-fuels for the gas turbine: A review. *Renew. Sustain. Energy Rev.* **2010**, *14*, 2946–2955. [\[CrossRef\]](#)
- Mendiburu, A.Z.; Lauermann, C.H.; Hayashi, T.C.; Mariños, D.J.; da Costa, R.B.R.; Coronado, C.J.; Roberts, J.J.; de Carvalho, J.A. Ethanol as a renewable biofuel: Combustion characteristics and application in engines. *Energy* **2022**, *257*, 124688. [\[CrossRef\]](#)
- Wang, C.; Zeraati-Rezaei, S.; Xiang, L.; Xu, H. Ethanol blends in spark ignition engines: RON, octane-added value, cooling effect, compression ratio, and potential engine efficiency gain. *Appl. Energy* **2017**, *191*, 603–619. [\[CrossRef\]](#)
- Barbosa, T.P.; Eckert, J.J.; Roso, V.R.; Pujatti, F.J.P.; da Silva, L.A.R.; Gutiérrez, J.C.H. Fuel saving and lower pollutants emissions using an ethanol-fueled engine in a hydraulic hybrid passengers vehicle. *Energy* **2021**, *235*, 121361. [\[CrossRef\]](#)
- Anderson, J.E.; DiCicco, D.M.; Ginder, J.M.; Kramer, U.; Leone, T.; Raney-Pablo, H.; Wallington, T. High octane number ethanol–gasoline blends: Quantifying the potential benefits in the United States. *Fuel* **2012**, *97*, 585–594. [\[CrossRef\]](#)
- Foong, T.M.; Morganti, K.J.; Brear, M.J.; da Silva, G.; Yang, Y.; Dryer, F.L. The octane numbers of ethanol blended with gasoline and its surrogates. *Fuel* **2014**, *115*, 727–739. [\[CrossRef\]](#)
- Noh, H.K.; No, S.Y. Effect of bioethanol on combustion and emissions in advanced CI engines: HCCI, PPC and GCI mode—A review. *Appl. Energy* **2017**, *208*, 782–802. [\[CrossRef\]](#)
- Pyshyev, S.; Korchak, B.; Miroshnichenko, D.; Lebedev, V.; Yasinska, A.; Lypko, Y. Obtaining new materials from liquid pyrolysis products of used tires for waste valorization. *Sustainability* **2025**, *17*, 3919. [\[CrossRef\]](#)
- Reyes Serrano, M.; Sastre Zamora, R.; Do, H.Q.; Le Moyne, L. Impact of cellularity in the flame front on the burning velocity acceleration of ethanol/air flames in constant volume combustion bomb. *Measurement* **2026**, *259*, 119667. [\[CrossRef\]](#)
- Wang, X.; Gao, J.; Chen, Z.; Chen, H.; Zhao, Y.; Huang, Y.; Chen, Z. Evaluation of hydrous ethanol as a fuel for internal combustion engines: A review. *Renew. Energy* **2022**, *194*, 504–525. [\[CrossRef\]](#)
- Liu, F.; Liu, Z.; Sang, Z.; He, X.; Sjöberg, M.; Vuilleumier, D.; Liu, C.; Li, X. Numerical study and cellular instability analysis of E30-air mixtures at elevated temperatures and pressures. *Fuel* **2020**, *271*, 117458. [\[CrossRef\]](#)
- Zhang, N.; Zhang, F.; Zhong, S.; Peng, Z.; Yu, J.; Liu, H.; Xu, C. Numerical and theoretical investigation of ethanol/air flame instability. *Combust. Theory Model.* **2020**, *24*, 1108–1129. [\[CrossRef\]](#)
- Zhou, J.; Lu, C.; Xu, C.; Yu, Z. Experimental and numerical study on the effect of hydrogen addition on laminar burning velocity of ethanol–air mixtures. *Energies* **2022**, *15*, 3114. [\[CrossRef\]](#)
- Li, T.; Wang, X.R.; Ma, Y.; Wang, L.; Oppong, F.; Xu, C.; Jiang, G. Investigation on hydrogen/ethanol intrinsic flame instability. *Combust. Flame* **2022**, *241*, 112064. [\[CrossRef\]](#)
- Algayyim, S.J.M.; Saleh, K.; Wandel, A.P.; Fattah, I.M.R.; Yusaf, T.; Alrazen, H.A. Influence of natural gas and hydrogen properties on internal combustion engine performance, combustion, and emissions: A review. *Fuel* **2024**, *362*, 130844. [\[CrossRef\]](#)
- Tripathi, G.; Dhar, A. Performance, emissions, and combustion characteristics of methane–diesel dual-fuel engines: A review. *Front. Therm. Eng.* **2022**, *2*, 870077. [\[CrossRef\]](#)
- Sun, M.; Shao, Y.; Gong, Y.; Xu, C.; Song, T.; Lu, P.; Agarwal, R.K. Numerical investigation of flow and flame structures in an industrial swirling inverse diffusion methane/air burner. *Fire* **2024**, *7*, 237. [\[CrossRef\]](#)
- Wang, Q.; Song, Y.; Liu, K.; Li, X.; Xu, C. Laminar combustion characteristics of methane/methanol/air mixtures: Experimental and kinetic investigations. *Case Stud. Therm. Eng.* **2023**, *41*, 102593. [\[CrossRef\]](#)
- Wang, X.; Yan, C.; Zhang, Y.; Guo, H.; Xu, C.; Jiang, G. Laminar and kinetic burning characteristics of ethanol/methane/hydrogen fuel: Experimental and numerical analysis. *Renew. Energy* **2024**, *227*, 120493. [\[CrossRef\]](#)
- Chen, R.; Jiang, G.Z.; Tao, J.X. Effects of Hydrogen Blending on Laminar Combustion Characteristics of Ethanol–Methane–Air Mixtures at Atmospheric Pressure. *South. Energy Constr.* **2025**, *12*, 1–11. [\[CrossRef\]](#)
- Lee, C.C.; Tran, M.V.; Tan, B.T.; Ooi, J.B.; Chong, C.T.; Scribano, G. A numerical study on soot formation in methane–ethanol diffusion flames. *Fuel* **2022**, *328*, 125313. [\[CrossRef\]](#)
- Fang, H.; Wei, D.; Zhou, H. Experimental study on combustion and thermoacoustic instability characteristics of ethanol/methane Co-firing flames. *J. Energy Inst.* **2024**, *117*, 101768. [\[CrossRef\]](#)

23. Shentu, J.; Li, Y.; Lu, Y.; Li, J.; Mao, Y.; Li, X. Combustion Mechanism Development Based on Minimized Reaction Network Method: C₂ Fuel. *Chem. J. Chin. Univ.* **2024**, *45*, 20230386.
24. Patankar, S.V.; Spalding, D.B. A calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows. In *Numerical Prediction of Flow, Heat Transfer, Turbulence and Combustion*; Pergamon: Oxford, UK, 1983; pp. 54–73.
25. Xu, Y.; Zhang, Y.; Gong, J.; Su, S.; Wei, Z. Combustion behaviours and emission characteristics of a retrofitted NG/gasoline dual-fuel SI engine with various proportions of NG-gasoline blends. *Fuel* **2020**, *266*, 116957. [[CrossRef](#)]
26. Ding, Q.M.; Ni, J.G.; Zhang, Z.H.; Cui, Y.Y.; Guo, Y.W.; Tang, Y. Study on the effect of methane-ammonia blending ratio on low swirl flame combustion characteristics. *J. Eng. Therm. Energy Power* **2026**, *41*, 55–64.
27. Yavuz, M.; Wu, M.; Cova-Bonillo, A.; Brinklow, G.; Herreros, J.; Tsolakis, A. Study of H₂ and NH₃ Mixtures in a Gasoline-Fueled Engine. In Proceedings of the Energy & Propulsion Conference & Exhibition, Columbus, OH, USA, 12 November 2024. [[CrossRef](#)]

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.