Reducing greenhouse gas emissions in Sandia methane-air flame by using a biofuel

Seyed Moein Rassoulinejad-Mousavia, Yijin Mao, Yuwen Zhang

A R T I C L E   I N F O

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Sustainability
Biofuel
Reduced reaction mechanism
Sandia flame D
Greenhouse gas emissions

ABSTRACT

This article aims to study suitability and merits of employing a biofuel in methane-air jet flames for energy security and environmental sustainability purposes. A special biofuel (methyl decanoate, methyl 9-decanoate, and n-heptane) oxidation with 118 species reduced/skeletal mechanism and 837 reactions is combusted instead of methane. The biofuel was fed in the main jet inlet of the well-known Sandia flame D (SFD) while the hot pilot jet is still responsible for ignition. The open-source software OpenFOAM was used for simulating turbulent biofuel-air combustion. To check the accuracy of computational results, the system was initially validated with SFD experimental results and good agreements were found. After ignition, mean temperature distribution and species mean mass fraction at different distances in axial and radial directions were investigated. Results showed that the biofuel can be effectively used as an alternative to SFD not only for generating a reasonable temperature as methane does, but also for significantly reduction in principal greenhouse gas emissions.

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1. Introduction

Key drivers for future energy policy are climate change and energy security that sparked so much debates and concerns in many organizations including International Energy Agency (IEA) [1]. High demand and dependency of industries on fossil fuel supply, along with the fact of scarcity or disruptions of the fuel, jeopardize the energy security. Furthermore, increasing global greenhouse gases (GHGs) emissions from fossil fuels combustion and industrial processes are contributing a lot to global warming and subsequently climate change. Hence, switching from fossil fuels to biofuels in transportation by automobiles, trains, ships, and aircraft, as well as most of the industries has promising implications for energy security and mitigation of climate change. Economic developments are also expected since investment in biofuels could lead to creation of new jobs as well as new sources of income [2].

High temperature methane combustion in diffusion flames leads to the emission of major GHGs. Methane-Air Sandia flame series are turbulent piloted methane diffusion flames which are being used widely for combustion studies. Sandia Flame D (SND) experiment [3] provides a standard and high quality reference data set for validation of turbulent flame calculations. This type of turbulent piloted methane/air diffusion flame has been investigated many times since it is an important and widely used flame for various studies. For instances, recently Renzo et al. [4] carried out a comparative study among large eddy simulations performed with various grid resolutions, a Reynolds averaged Navier-Stokes simulation, and experimental data concerning the well-known Sandia D flame test case. Cai et al. [5] implemented a new non-gray radiation modeling library for combustion gases in OpenFOAM for comparisons of radiative heat transfer calculations in a jet diffusion flame using spherical harmonics and k-Distributions. Kadar [6] modeled the turbulent non-premixed combustion in industrial furnaces using an open source toolbox. Schneider et al. [7] measured the flow field of stable and locally extinguishing hydrocarbon-fueled jet flames. There are a huge number of other attractions on multiple aspect studies of the Sandia flame such as rate-controlled constrained equilibrium [8], premixed and non-premixed flamelet-generated manifolds [9], thermal radiation influence [10], Sparse-Lagrangian multiple mapping conditioning simulation [11], Large Eddy Simulation the Eulerian stochastic field method [12], multi-environment probability density function models [13], modeling of evolution equation for the sub-grid joint probability density
function [14,15], and prediction of extinction and re-ignition using a flamelet/progress variable model [16]. The popularity of these flames motivated us to consider the sustainability challenge in these combustion systems, the major GHG emissions due to use of fossil fuels, by introducing a biofuel alternative for them. Biodiesel is a promising alternative biofuel for combustion systems. It is a renewable energy fuel source that can be used in existing systems without any significant modifications. Biodiesel is primarily composed of fatty acid methyl esters and can be obtained from vegetable oils, animal fat, and waste cooking oils [17,18]. It is one of the most available alternative fuel sources for compression-ignition engines nowadays. A concise review of current status of the biofuel technologies, advantages, challenges, the key barriers to their commercial applications, and future perspectives of biofuels can be seen in the study provided by Cheng and Timilsina [19]. Many attempts have been made to decrease the emissions of hazardous gases for environment especially greenhouse gases using biofuels. As blends with diesel, Kumar et al. [20] used three advanced biofuels, iso-butanol, n-pentanol and dimethyl carbonate, for smoke/NOx reduction in a light-duty DI diesel engine. They found that NOx and smoke could be reduced at the same time but Carbon monoxide (CO) and hydrocarbons emissions might increase. Sisiiu et al. [21] recently considered emission characteristics of n-butanol and cotton seed biodiesel combustion. Their results for mechanical efficiencies and the coefficient of variation indicated stable operation with renewable fuels as well as NOx and soot reduction because of high oxygen content in the biodiesel and n-butanol. Srilhari [22] observed reduction in NOx and smoke emissions using biofuel blends. They used a pilot injector to supply a small quantity of premixed charge of fuel and air to the engine for combustion. They examined diesel and a blend of ethanol-diesel (containing 15% of ethanol by volume) and found that this mode showed that smoke emission can be reduced by up to 70%, while NOx reduction in a direct injection diesel engine [27].

In a recent investigation, ethanol was used as a main jet, the hot mixture from the pilot jet is also responsible for igniting the fuel that is injected from the main jet. The pilot, which is a lean (equivalent ratio of 0.77) mixture of C2H2, H2, air, CO2, and N2 with the same nominal enthalpy and equilibrium composition as methane/air at this equivalence ratio. The fuel nozzle has a diameter of d = 7.2 mm and is enclosed by a broad pilot nozzle with inner and outer diameters of 7.7 mm and 18.2 mm, respectively.

Pre-inlets for the fuel-jet and pilot are included in the computational domain in order to obtain fully-developed turbulent velocity profiles at the burner exit (see Fig. 1). The length of the pre-inlet fuel and pilot pipes are approximately 15D. The axial and radial dimensions of the computational domain after the inlet were set to 76.5D and 20.83D, respectively, according to recommended model by Refs. [6,35]. Reynolds-Averaged Simulation (RAS), where the turbulence was treated via a standard k-ε model [36], was applied in OpenFOAM to solve turbulent flow and heat transfer.

2. Computational details

As mentioned earlier, the SFD from the Sandia National Laboratories is a piloted methane-air diffusion flame. The central main jet consists of a methane-air mixture. The main jet is surrounded by a pilot jet and a slow coflow of air outside. Besides stabilizing the main jet, the hot mixture from the pilot jet is also responsible for igniting the fuel that is injected from the main jet. The pilot, which is a lean (equivalent ratio of 0.77) mixture of C2H2, H2, air, CO2, and N2 with the same nominal enthalpy and equilibrium composition as methane/air at this equivalence ratio. The fuel nozzle has a diameter of d = 7.2 mm and is enclosed by a broad pilot nozzle with inner and outer diameters of 7.7 mm and 18.2 mm, respectively.

The Favre-averaged continuity, momentum, and energy equations for the turbulent compressible flows are as follows:

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho \vec{u}_j)}{\partial x_j} = 0, \quad (1)
\]

\[
\frac{\partial}{\partial t} (\rho \vec{u}_j) + \frac{\partial}{\partial x_j} (\rho \vec{u}_j \vec{u}_j) = -\frac{\partial \rho}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \tau_{ij} - \rho \vec{u}_i \vec{u}_j \right], \quad (2)
\]

\[
\frac{\partial}{\partial t} (\rho h) + \frac{\partial}{\partial x_j} (\rho h \vec{u}_j) = \frac{\partial}{\partial x_j} \left[ \rho u \frac{\partial h}{\partial x_j} - \rho \vec{u}_j \vec{u}_j h \right] - \frac{\partial}{\partial x_j} (\Gamma_{ij}) + \tau_{ij} \vec{u}_j. \quad (3)
\]

The overbar denoted Reynolds averaging, while the tilde denotes Favre averaging. \( \rho \) is the density, \( p \) is the pressure, \( \vec{u} \) is the velocity, \( \tau_{ij} \) is the viscous stress tensor defined as follows.
\[ \tau_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \delta_{ij} \frac{\partial u_k}{\partial x_k} \]  

(4)

where \( \mu \) is the molecular viscosity and \( \delta_{ij} \) is the Kronecker delta.

The Reynolds stresses can be calculated using the Boussinesq assumption:

\[ \rho u_i u_j = -\mu_1 \left[ \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right] + \frac{2}{3} \rho k \delta_{ij} \]  

(5)

where \( \mu_1 \) is the turbulence viscosity.

In the mass-density-weighted energy equation (3), \( \bar{h} \) is the total enthalpy,

\[ \bar{h} = \sum_s \bar{Y}_s \bar{h}_s, \quad \bar{h} = \bar{h}^0_s + \int_{T_0}^T \frac{c_p}{T} dT \]  

(6)

where \( \bar{T} \) and \( \bar{Y}_s \) are Favre-averaged temperature and mass fraction of species \( s \). \( \bar{h}^0_s \) is the enthalpy of formation of species \( s \) at standard temperature and pressure. \( c_p \) is the heat capacity for species \( s \) in the mixture. \( \alpha \) is the thermal diffusivity, \( q_r \) indicates the radiative heat loss and \( S_{ct} \) is the source term due to combustion. The turbulence flux \( \bar{u}_i \bar{h}_j \) according to the gradient hypothesis yields:

\[ -\rho \mu_1 \frac{\partial \bar{h}}{\partial x_j} = \mu_s \frac{\partial h}{\partial x_j} \]  

(7)

where \( \mu_s \) is the turbulence Prandtl number.

Favre-averaged transport equation for the mean mass fraction of an individual species \( Y_s \) for a mixture of \( N_s \) species (where \( s = 1 \ldots N_s \)) is:

\[ \frac{\partial}{\partial t} (\rho Y_s) + \frac{\partial}{\partial x_j} (\rho Y_s \bar{u}_j) = \frac{\partial}{\partial x_j} \left( \rho D_{m,s} + \frac{\mu_1}{Sc_t} \frac{\partial Y_s}{\partial x_j} \right) + \bar{\omega}_s \]  

(8)

where \( D_{m,s} \) is the mass diffusion coefficient for species \( s \) in a mixture, and \( \bar{\omega}_s \) is the volumetric reaction rate of the species. \( Sc_t \) is the turbulence Schmidt number expressed as

\[ Sc_t = \frac{\mu_t}{Pr_D} \]  

(9)

where \( D_\alpha \) is a turbulent diffusivity.

The equations for the \( k-\varepsilon \) turbulence model are the equations for turbulent kinetic energy \( k \), dissipation of turbulent kinetic energy \( \varepsilon \), turbulent viscosity \( \mu_s \), and Reynolds’s stresses. The modeled transport equations for the standard \( k-\varepsilon \) turbulent model are

### Table 1

<table>
<thead>
<tr>
<th>Field</th>
<th>Pressure</th>
<th>Velocity</th>
<th>Temperature</th>
<th>Fuel</th>
<th>( O_2 )</th>
<th>( N_2 )</th>
<th>( H_2O )</th>
<th>( CO_2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Fuel</td>
<td>zG</td>
<td>(49.6, 0, 0)</td>
<td>294</td>
<td>0.1561</td>
<td>0.1966</td>
<td>0.6473</td>
<td>0</td>
<td>0</td>
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<tr>
<td>Inlet Pilot</td>
<td>zG</td>
<td>(11.4, 0, 0)</td>
<td>1880</td>
<td>0</td>
<td>0.054</td>
<td>0.742</td>
<td>0.0942</td>
<td>0.1098</td>
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<tr>
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<td>zG</td>
<td>(0.9, 0, 0)</td>
<td>291</td>
<td>0</td>
<td>0.23</td>
<td>0.77</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Pre-inlet Wall</td>
<td>zG</td>
<td>(0.9, 0, 0)</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
</tr>
<tr>
<td>Wall</td>
<td>zG</td>
<td>(0.9, 0, 0)</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
</tr>
<tr>
<td>Outlet</td>
<td>( 1e+5 )</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
<td>zG</td>
</tr>
<tr>
<td>Internal Volume</td>
<td>( 1e+5 )</td>
<td>0</td>
<td>291</td>
<td>0</td>
<td>0.23</td>
<td>0.77</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

zG-zeroGradient

Velocity expressed in (axial, radial, tangential) components.

Mass fractions of species used.

All quantities in SI units.

### Table 2

<table>
<thead>
<tr>
<th>Field</th>
<th>( k )</th>
<th>( \varepsilon )</th>
<th>( \alpha_t )</th>
<th>( \mu_s )</th>
</tr>
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<td>TMLDR calculated</td>
<td>zG</td>
<td>zG</td>
</tr>
<tr>
<td>Inlet Pilot</td>
<td>TIKEI</td>
<td>TMLDR calculated</td>
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<tr>
<td>Co-flow-air</td>
<td>TIKEI</td>
<td>TMLDR calculated</td>
<td>zG</td>
<td>zG</td>
</tr>
<tr>
<td>Pre-inlet Wall</td>
<td>kqRWF</td>
<td>epsilonWF</td>
<td>alphatWF</td>
<td>MutUWF</td>
</tr>
<tr>
<td>Wall</td>
<td>epsilonWF</td>
<td>alphatWF</td>
<td>MutUWF</td>
<td>MutUWF</td>
</tr>
<tr>
<td>Outlet</td>
<td>zG</td>
<td>zG</td>
<td>calculated</td>
<td>zG</td>
</tr>
</tbody>
</table>

TIKEI- Turbulent Intensity Kinetic Energy Inlet.

kqRWF- kqR Wall Function.

epsilonWF- epsilon Wall Function.

TMLDR- Turbulent Mixing Length Dissipation Rate Inlet.

MutUWF- mutU Wall Function.

alphatWF- alphat Wall Function.

Fig. 1. Computational domain and schematic of the problem.
Fig. 2. Validation of mean velocity at different locations.

Fig. 3. Validation of mean temperature distribution at different locations.
\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho k \dot{u}_j) = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_t}{\sigma_t} \right) \frac{\partial k}{\partial x_j} \right) + P_k - \rho \varepsilon
\] (10)

\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho \varepsilon \dot{u}_j) = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_t}{\sigma_t} \right) \frac{\partial \varepsilon}{\partial x_j} \right) + C_1 \frac{\varepsilon}{k} P_k - C_2 \varepsilon^2 \frac{k}{\varepsilon}
\] (11)

where \( \sigma_t, \sigma, C_1 \), and \( C_2 \) are adjustable constants. The turbulent viscosity \( \mu_t \) and the kinetic energy production \( P_k \) are

\[
\mu_t = \frac{C_\mu \rho k^2}{\varepsilon}
\] (12)

\[
P_k = -\rho \mu_t \dot{u}_j \frac{\partial \dot{u}_j}{\partial x_j}
\] (13)

The Laminar Flamelet model is applied for turbulent combustion modeling since it is found to be reasonable and accurate for Sandia Flame [35]. The initial conditions for the fuel jet (both methane and biofuel), pilot and coflow compositions are specified in terms of the species mass fractions are summarized in Table 1.

Turbulent properties applied in OpenFOAM are tabulated in Table 2. The initial turbulent kinetic energy is defined using the turbulence intensity while the initial dissipation rate is determined by the associated mixing length. Standard wall functions for turbulence kinetic energy and dissipation are used for near wall treatment.

Bidi et al. and Xu et al. [10,37] showed the significant effect of radiation in combusting flows in SFD. The well-known P1- radiation method, which is the simplest form of the more generalized P–N method (or spherical harmonics), with the Marshaks boundary condition were used for radiation modeling. In fact, it is applied for radiative heat flux calculation at the inlets and outlet and walls. The radiative heat loss can be found as

\[
-\frac{\partial}{\partial x_j} \Pi_r = \alpha_r G - 4 \varepsilon_r \sigma T^4,
\] (14)

where \( \alpha_r \) is the absorption coefficient, \( \varepsilon_r \) is the emission coefficient and \( \sigma \) is the Stefan-Boltzmann constant. The emissivity is taken to be 1 for open boundaries, including the inlet and outlet that

Fig. 4. Mean temperature contour for the applied biofuel.

Fig. 5. Mean mass fraction of some species after biofuel combustion a) GHGs b) non-GHG.
resemble black bodies. The emissivity is taken to be zero at symmetry planes and perfectly reflecting boundaries as no emissivity corresponds to zero radiative heat flux.

3.1. The incident radiation $G$ is modeled according to

$$\frac{\partial}{\partial x_j} \left( \frac{\partial}{\partial x_j} G \right) + \alpha_c G + 4e_c \sigma T^4 = 0,$$

(15)

$$\Gamma = \frac{1}{3(\alpha_c + \sigma_s)},$$

(16)

where $\sigma_s$ is the scattering coefficient.

4. Results and discussions

To validate time-averaged results from the simulations, mean velocity and temperature are compared to available data in literature. Mean axial velocities are validated against the experimental data by Schneider et al. [7], who performed flow field characterization using a two-color laser doppler velocimetry. Fig. 2 (a)-(c) show good agreements between present results with experimental ones, which confirms the robustness of present numerical simulation. In Fig. 3(a)-(c) mean temperature field at different cross-sections are compared against the experimental results of Barlow and Frank [3]. The chemistry was treated by the global single step mechanism for SFD according to [3]. As can be seen, the obtained temperature profiles match well with the experiments at different locations.

Figs. 4 and 5 demonstrate the steady state biofuel-air temperature distribution and mass fractions of combustion products in the computational domain. Fig. 4 displays the temperature distribution at steady flame state as the result of the biofuel combustion. According to the contours, it is obvious that biofuel generates almost the same temperature as methane at various locations and is also higher at some points. Observing at lower part of this figure, it is found that temperature of the upstream part of coflow is also influenced by the downstream flow. In addition, the high temperature zone gets broader due to the radiation effect. At upstream
region, the high temperature zone lies between \( r/D = 0.5 \) and \( r/D = 2 \) where combustion of main jet has not started yet and high temperature zone belongs to the pilot stream, but in downstream, maximum temperature occurs near the centerline of flame zone. Fig. 5 demonstrates some of the biofuel combustion products along the domain at different radial and axial sections for both GHGs and non-GHGs. It is evident from the contours that some of major GHGs hazardous species mass distributions have been significantly reduced by alternating biofuel in Sandia flame geometry. It is known that, biodiesel is a mixture of many oxygenated components with varying chain. Thus, the existence of chemically bound oxygen enhances the combustion rate of fuel and is expected to reduce hazardous gas emissions. In fact, oxygen content in biofuels is the biggest difference between them and petroleum feedstocks. Comparing to [3], it is evident from Fig. 5(a) that concentration and amount of \( \text{CH}_4 \), \( \text{H}_2 \text{O} \) and \( \text{CO}_2 \) at centerline and other different flame areas, dropped significantly. The maximum mass fraction of \( \text{CH}_4 \) decreased by 97% with respect to SFD. This reduction is about 23% for \( \text{H}_2 \text{O} \). Distribution and concentration of \( \text{CO}_2 \) has lowered at different parts of domain, while it has slightly increased at the hottest flame spot. At downstream, \( \text{CO}_2 \) and \( \text{H}_2 \text{O} \) mass fractions decrease smoothly from center to outward, while at downstream, its mass fraction becomes very small and gradually decreases from down to upstream. According to the contours, amount of \( \text{CO}_2 \) and \( \text{H}_2 \text{O} \) increase in the flame region and reach to a maximum at \( X/D = 20 \), where the fuel is completely consumed. Fig. 5(b) shows that mass fraction of MD is reduced axially due to its consumption by chemical reactions and reaches to zero at \( X/D = 30 \).

To compare the temperature and mass fraction of GHGs products of the biofuel and methane combustions more in details, which is the main purpose of the present work, they are compared at the centerline and different axial/radial distances in Figs. 6–9 with experimental results of methane combustion in SFD [3]. Axial variations of temperature at centerline is shown at steady state in Fig. 6 where the peak temperature for biofuel combustion is close to 2200 K.

Fig. 6(a) depicts the close temperature profile produced by the biofuel in comparison with methane. However, the biofuel is creating higher maximum temperature at centerline. According to the peaks of the two profiles, it can be concluded that full fuel
consumption occurs sooner for the biofuel at $X/D \sim 37$ while it is at $X/D \sim 45$ for methane. This confirms a faster combustion process for the biofuel than methane. However, at the range of $45 < X/D < 80$, methane combustion temperature is slightly higher than that of the biofuel. Fig. 6(b) shows that biofuel has a similar temperature profile at $X/D = 2$ at different radial distances. According to Fig. 6(c), biofuel temperature profile stands above the SFD in different radial distances at $X/D = 15$ which confirms faster ignition than methane at this region. This justification is also true for the phenomena at $X/D = 45$. However, discrepancies are less than that of $X/D = 15$ since it is farther from hottest spot where the fuel has completely burnt.

Fig. 7 (a)-(c) shows how significantly the methane, one of the principle GHGs, emission decreased at different locations with respect to SND. The comparisons between the right and left axis of the figures can confirm the merit of using the biofuel instead of methane for improvement of GHGs reduction. Water vapor, another major GHG, emission is compared at Fig. 8(a)–(d). It is clear from Fig. 8(a) that water vapor emitted by the biofuel burning is far less than that of methane combustion. This reduction in water vapor can be seen in different radial distances too. At $X/D = 2$, this reduction increases by getting away from centerline which can be seen in Fig. 8(b). According to Fig. 8(c) and (d) by closing to the hottest spot axial distance ($X/D = 37$) the water mass fraction reduces much closer to the centerline with respect to SND.

Carbon dioxide emission after biofuel combustion is compared with experimental results of methane combustion in Fig. 9 (a)-(c). Although CO2 emission decreased at some parts of centerline, it has increased from $30 < X/D < 47$ where the hottest point is located and maximum fuel burning is happening (see Fig. 9(a)). The reduction can be adjustable by manipulating the inlets mass fractions and supply of excessive air or oxygen which can be investigated in future studies. According to Fig. 9(b), CO2 emission decreased at $X/D = 2$ at different radial distances. This reduction increases by distancing from centerline. Based on Fig. 9(c) biofuel produces much less CO2 than methane as axial distances increase.

According to radial profiles of CO2 and H2O mass fractions, at downstream, their presence in the pilot stream results in a high concentration region between $r/D = 0.5$ and $r/D = 1.5$. This is
because at downstream maximum concentration occurs near the centerline because of their production in the main jet.

Fig. 10 (a)-(b) demonstrate the mean mass fraction at centerline and different axial locations of the non-GHGs resulted from the biofuel burning. The sensible amount of these products can be seen near the hottest point and distances near the maximum fuel burning. Fig. 11 shows the radial profiles of O\textsubscript{2} mass fraction at different locations of the combustion domain. At upstream, O\textsubscript{2} presents in both main jet and coflow but at downstream it consumes in the main jet. Based on Fig. 12, which displays N\textsubscript{2} concentration at different locations, the distribution of N\textsubscript{2} becomes more uniform at upstream because of coflow.

5. Conclusions

An advanced biofuel was examined in Sandia flame D instead of methane for the first time by keeping the same conditions, to consider the potential of the biofuel surrogate in methane-air jet flames for sustainability approach. Turbulent combustion using OpenFOAM was carried out by burning methane and a biofuel with a skeletal mechanism with 118 species and 837 reactions developed from a detailed LLNL mechanism. Numerical results were validated with available data in the literature and good agreements were found. It was observed that burning the biofuel can produce the equivalent temperature distribution as methane while emission of GHGs gases is decreased. Temperature and species concentrations were investigated for the biofuel at different locations of the combustion domain. Emissions of methane and water vapor as the two leading GHGs were reduced by 97% and 23%, respectively. Carbon dioxide emission, as one the most important hazardous GHGs, was significantly reduced from a huge part of domain, which represents less CO\textsubscript{2} in the environment. The oxygen content of the biofuel used (methyl decanoate, methyl 9-decanoate, and n-heptane) played a significant role in the reduction of the GHGs. Results presented in this work opens a new window for further research on biofuel combustion in methane-air jet flames which leads to mitigation of climate change and improving the energy security.
**Nomenclature**

- $c_p$: heat capacity
- $D_t$: turbulent diffusivity
- $D_m$: mass diffusion coefficient
- $e_c$: emission coefficient
- $G$: incident radiation
- $h$: total enthalpy
- $p_k$: kinetic energy production
- $p$: pressure
- $q_r$: radiative heat loss
- $S_{sc}$: source term
- $S_{St}$: Schmidt number
- $\tilde{T}$: Favre-averaged temperature
- $u$: velocity
- $\tilde{Y}_s$: Favre-averaged mass fraction

**Greek symbols**

- $\alpha$: thermal diffusivity
- $\alpha_c$: absorption coefficient
- $\delta_{ij}$: Kronecker delta
- $\epsilon$: turbulent kinetic energy
- $\mu$: molecular viscosity
- $\mu_t$: turbulent viscosity
- $\rho$: density
- $\sigma$: Stefan-Boltzmann constant
- $\sigma_s$: scattering coefficient
- $\tau_{ij}$: viscous stress tensor
- $\omega$: volumetric reaction rate

**Abbreviations**

- SFD: Sandia flame D
- GHG: Greenhouse Gas
- LLNL: Lawrence Livermore National Laboratory
- RAS: Reynolds-Averaged Simulation

**References**


