Soot Emission Reduction in a Biogas-DME Hybrid Dual-Fuel Engine

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Abstract: Combustion characteristics and harmful emissions with emphasized soot emission in the new concept of a biogas-dimethyl ether (DME) hybrid dual-fuel engine were analyzed. The effects of DME content, biogas compositions and diesel injection were examined. At any biogas composition, a rise in DME content in the fuel mixture leads to an increase in indicative engine cycle work ($W_i$) and NO$_x$ but a decrease in CO and soot volume fraction ($f_v$). The effects of DME on $W_i$ and soot volume fraction are more significant for poor biogas than for rich biogas, contrary to its effect tendency on CO and NO$_x$ concentrations. With a given operating condition and DME content, the biogas compositions slightly affect the performance and emission of a biogas-DME hybrid dual-fuel engine. At a fixed global equivalence ratio, the reduction of diesel injection leads to an increase in $W_i$ and NO$_x$ concentration but a decrease in CO and soot volume fraction. The lower the diesel injection is, the more significant the effects of DME content on the combustion properties and pollutant emissions are. At a given operating condition and the same global equivalence ratio, the biogas-DME PCCI combustion mode is more advantageous than biogas-DME dual-fuel combustion mode. The substitution of diesel pilot ignition by DME pilot ignition in a biogas-DME hybrid dual engine is the optimal solution for both performance improvement and pollution emissions reduction.

Keywords: biogas; dimethyl ether; renewable energy; internal combustion engine; pollutant emissions

1. Introduction

Despite relentless attempts to introduce electricity to the transport sector, internal combustion engines (ICE) remains the main vehicle propulsion in the upcoming decades. However, they will undergo further development of technologies for controlling soot and NO$_x$ emissions [1]. Among the traditional IC engines, compression ignition (CI) engines are preferable to spark-ignition (SI) engines in view of thermal efficiency because of the high compression ratio, lean combustion, and low pumping loss. However, the drawback of CI engines is the high emissions of NO$_x$ and soot owing to the diffusion combustion characteristics [2–4]. The simultaneous reduction of NO$_x$ and soot emissions is really a challenge because these two pollutants have a trade-off relationship with each other [5]. Soot particulates are generally produced by inhomogeneous and locally fuel-rich regions of the diffusion flame, while NO$_x$ is formed in the high-temperature region around the stoichiometric air–fuel reaction regions [6]. The reduction in the diffusion combustion phase has contributed to a significant reduction in soot emissions [7]. Numerous studies report the fact that it is very hard to improve the performance and emissions of CI engines by the change of the single-fuel property [8]. It is expected that dual-fuel mode offers a better way to control the combustion than the conventional diesel CI engine. This is due to the combustion characteristics of a fuel blend–air mixture that strongly depend on the fuel vaporization, the fuel–air mixing process, and the fuel chemical reactivity of each component [9–11].
In order to reduce NO\textsubscript{x} and soot emissions simultaneously, some advanced combustion technologies have been introduced, such as homogenous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI). These technologies are intermediate between CI and SI combustion modes. Although they are attractive technologies in view of thermodynamics, it is challenging to control the ignition timing and heat-release characteristics [12]. Besides, the major drawback of the HCCI and PCCI combustion strategies is the high levels of HC and CO emissions. Subsequently, the concept of reactivity controlled compression ignition (RCCI) has been developed to overcome the problems encountered in HCCI and PCCI. In RCCI, high-octane fuel is injected into the intake manifold, while a high-cetane fuel is directly injected into the combustion chamber for ignition. The combustion properties and ignition timing of the engine can be controlled by varying injection timing and adjusting the mixing ratio of the two fuels [13].

Another attempt to control the emission of ICEs is adopting the optimal fuel blends in different operating conditions of the engine. This is the concept of the hybrid fuel engine (HFE). This technology has been proved to be a feasible approach for improving the performance of traditional mono-fuel engines [14,15]. With a large variety of properties, renewable fuels have high potential for use in the HFE. Many researchers have investigated various fuel blends in order to achieve low emissions and high thermal efficiency combustion [16–19]. Among the fuels, dimethyl ether (DME) and biogas are very attractive because of their advantageous physical chemistry properties, and being renewable fuels [20–22].

In fact, DME is considered to be a clean alternative fuel [23] and has remarkable potential for application in ICEs [24,25]. It can be produced from a variety of abundant renewable material sources or fossil fuels [26]. Likewise, it is an ideal diesel fuel substitution applied to CI engines because of its high cetane number [27]. Moreover, DME forms a liquid phase with moderate pressure (about 5 bar); therefore, it can be used in the existing infrastructures for transportation and storage of liquefied petroleum gas (LPG) [28]. DME contains about 35% oxygen. Its chemical formula only has C–H and C–O bonds, but no C–C bond, thus it produces practically no soot emission and a significant reduction of CO, HC, NO\textsubscript{x} concentrations in the exhaust gas [28]. Ji et al. [29] revealed the effects of adding DME on the gasoline engine and found that HC and NO\textsubscript{x} emissions were diminished. Similarly, Fu et al. [30] found that combustion stability was improved, and CO emission was decreased when increasing the amount of DME in the mixture with gasoline.

Furthermore, the experimental results showed that the laminar burning velocities of the traditional fuels in the air could be improved with the addition of DME [31,32]. The presence of a small amount of DME in the traditional fuel could improve the ignition quality of SI engines [33]. A study by Lee et al. [34] showed that the range of engine loads was widened with the addition of DME to LPG. They pointed out that 20% of DME-blended fuel could be used as an optimal alternative fuel for LPG engines [33].

Biogas is produced from organic wastes such as agricultural waste, cattle, and poultry manures, which are abundant in the rural areas of tropical regions. It contains mainly CH\textsubscript{4} and CO\textsubscript{2} with variable compositions depending on primary resources. Also, biogas has a high octane number and high auto-ignition temperature. Biogas is proved to be an excellent renewable fuel for stationary engines [6]. Due to the high octane number, biogas is a potential fuel for application on a high compression ratio SI engine converted from diesel engine [35,36]. NO\textsubscript{x} and smoke emissions in this engine decreased significantly with the biogas-fueled engine, whereas HC and CO emissions increased marginally [37]. Although the unique properties of biogas are beneficial for enhancing engine performance, there are still some problems that arise when the biogas engines are applied in practice. The presence of CO\textsubscript{2} in biogas improves the anti-knocking property of the fuel but reduces burning velocity, which affects the thermal efficiency and pollutant emission of the engines [38]. The application of biogas in conventional engines shows some limitations, but biogas can be used more effectively in the new combustion strategy engine, such as HCCI engines [39].
As mentioned above, to improve combustion efficiency, different fuel blends should be fueled in engines instead of a single fuel [40]. As the engine operates under low load conditions, the increase of high cetane-rating fuels in the blend with biogas is needed to improve the ignition performance. By contrast, the addition of high cetane-rating fuel should be reduced as the engine operates at a high load regime [41]. In this concept, the biogas-DME hybrid fuel engines are desirable. The properties of low-temperature oxidation and the high cetane number of DME can lead to abnormal ignition, but it can serve as an addition to improve the combustion of the biogas engine [42,43]. Due to the high cetane number of DME, its unique application in dual-fuel operation mode leads to the early start of the combustion phenomena, particularly at a relatively high load regime [27]. Wang et al. [44] studied the DME-diesel dual-fuel combustion in a naturally aspirated engine and concluded that as the DME quantity increased, the start of combustion advanced. Hence, for practical usage of DME in dual combustion mode, additional fuel for the combustion inhibitor is needed. Jang et al. [45] used LPG to the control combustion phase of the LPG-DME HCCI engine. Zhao et al. [46] used the exhaust gas recirculation (EGR) rate to control the combustion and emission characteristics of the DME-diesel dual-fuel PCCI engine.

Generally, in practical application, if DME is the main fuel, the addition of a high octane-rating fuel for a combustion inhibitor is needed. By contrast, if a high octane-rating fuel is used as the main fuel, DME can be used as a combustion enhancer. It is the case of a biogas DME hybrid dual-fuel engine. In order to optimize the operational range and improve the performance of biogas-diesel dual-fuel engines, DME is mixed with biogas due to its high cetane number and low boiling point. Based on the above properties of DME, the biogas-DME blend can be easily ignited and burnt. As biogas is a high octane number, and DME is a high cetane rating, the biogas-DME blend with the various mixing ratios has different octane ratings. Thus, under a specific operating condition, the most suitable octane-rating mixture could be obtained by adjusting the blending ratio of the biogas-DME mixture properly. This helps to achieve high efficiency and low emissions of the dual-fuel engine under different operating conditions.

Experimental results showed that in a biogas-diesel dual-fuel engine, NO\textsubscript{x} and smoke emissions decreased significantly [37]. Barik et al. [47] studied the combustion and emission characteristics of biogas-diesel dual-fuel combustion engine and found that NO\textsubscript{x} and smoke emissions were lower by about 39% and 49%, compared to that of the diesel engine. Lounici et al. [48] carried out the performance research in a compressed natural gas (CNG)-diesel dual-fuel engine and showed that a simultaneous reduction of NO\textsubscript{x} and soot could be realized. The research of Bui et al. [49,50] on dual-fuel engine fuel with hydrogen-enriched biogas showed that the soot emission could be significantly reduced with an increase in hydrogen content in the fuel mixture. Mustafi and Raine [51] indicated that soot and NO\textsubscript{x} could be reduced by biogas dual-fuel combustion by 70% and 37%, respectively, using natural gas or biogas-diesel dual-fuel engine systems. The low burning velocity of biogas causes the incomplete combustion. This in turn increases HC and CO emissions, which leads to the lack of fitness in the biogas dual-fuel engine. With the addition of DME in the biogas DME hybrid dual-fuel engine, this problem can be overcome.

In previous works, we have studied biogas dual-fuel engines [49,50,52]. We found that the dual combustion mode is an efficient solution for biogas applications on internal combustion engines. However, the variation of the fuel ratio between biogas and diesel could not improve the flame speed of biogas, which affects the engine performance and emissions in certain operating conditions. Research into technology to improve the flame speed of biogas is, thus, important for an efficient application of the fuel in the internal combustion engine. Due to the high cetane number, DME is a potential substitution of diesel fuel in the CI engine, but it is not an appropriate fuel for the SI engine. However, DME can be served as a combustion enhancer of biogas in dual fuel combustion mode because biogas has a high octane number. This is the new concept in a biogas-DME hybrid dual-fuel engine, which seems to be rarely found in the literature.
The purpose of this study is a contribution to develop the new concept of a biogas-DME hybrid dual-fuel engine in which DME is a combustion enhancer. The study will be carried out by simulation on a retrofitted mono-cylinder diesel engine. The effects of DME content, biogas compositions and diesel injection on performance and emissions, particularly soot emission, of the engine will be examined in detail.

2. Materials and Methods

2.1. Materials

The simulation of combustion and emissions characteristics of a biogas-DME hybrid dual-fuel engine was performed on a retrofit diesel engine. It was converted from a mono cylinder, 4-stroke RV165-2N Vikyno engine with 105 mm of bore, and 97 mm of stroke; the maximum output power was 16.5 HP at 2400 rpm under diesel fueling mode. The compression ratio of the retrofitted engine was reduced from 20 of the original engine to 18 to fit with the dual-fuel combustion. The gaseous fuel mixture was supplied into the intake manifold through a specific supply valve described in [53]. The specifications of the retrofitted engine are described in Table 1. Figure 1 shows the retrofitted combustion chamber and intake manifold (a) and meshing calculation spaces (b,c) of the biogas-DME hybrid dual-fuel engine.

Table 1. Specifications of the retrofit engine.

<table>
<thead>
<tr>
<th>Combustion Chamber Type</th>
<th>Spherical Shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>105</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>97</td>
</tr>
<tr>
<td>Displacement (cm³)</td>
<td>839</td>
</tr>
<tr>
<td>Connecting rod length (mm)</td>
<td>120</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18:1</td>
</tr>
<tr>
<td>Number of nozzle holes of injector</td>
<td>1</td>
</tr>
<tr>
<td>Diameter of nozzle hole (mm)</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Figure 1. The retrofitted combustion chamber and intake manifold (a) and meshing calculation spaces (b,c) of the biogas-dimethyl ether (DME) hybrid dual-fuel engine.
2.2. Methods

2.2.1. Computational Fluid Dynamics (CFD) Code

In the present paper, the commercial computational fluid dynamics (CFD) package ANSYS Fluent V18.2 was used for the simulations. The fundamental governing equations of fluid dynamics (continuity, momentum, energy, species) closed by the turbulence model for unsteady flow were solved by a segregated pressure-based solution algorithm.

2.2.2. Geometry Development and Meshing of the Computational Domain

The computational domain was modeled in the preprocessor by using workbench tools for design and meshing. The meshing of the intake manifold, the cylinder, and the combustion chamber at bottom dead center (BDC, 180 °CA (degree of crankshaft angle)) is shown in Figure 1b,c. The number of cells was 80,336 when the piston was at BDC. The dynamic mesh was applied in the cylinder and combustion chamber spaces. The admission system was deactivated at the beginning of the compression stroke to save calculation time.

2.2.3. Turbulence Model

In the present work, the well-known Re-Normalized Group (RNG) k-ε model was used for modeling turbulence. It was analogous in form to the standard k-ε model but had the advantage of including the effect of swirl, which was important for flow in the internal combustion engine. The standard k-ε model has proved to be economical, robust, and reasonably accurate, but it gives poor results for complex flows under a strong swirl.

2.2.4. Pollutants Formation Models

NOx formation was modeled by the extended Zeldovich mechanism, which is widely used in modeling pollutant emissions of the combustion process [49]. As the combustion temperature of biogas-DME is normally higher than in 1600 °K, the NOx formation rate via the thermal mechanism described by Zeldovich is dominant. Soot emission was modeled by the Magnussen mechanism in which soot formation rate is proportional to radical nuclei concentration, whereas soot combustion rate is proportional to fuel concentration, oxygen concentration, and $\epsilon/k$ [49,50]. This model has proved accurate with experimental data of the diffusion flames, combustion in the diesel engine, and combustion in an industrial furnace [49]. Soot emission is defined by soot volume fraction $f_v$. It is the volume of soot particles on the unity of combustion products volume. CO is determined by the thermodynamic equilibrium of combustion products.

2.2.5. Combustion Model

Dual-fuel combustion mode was modeled via partially premixed combustion integrated into the software. Fuels were introduced into the engine by two separate streams. The first stream was the mixture of biogas and DME composed of CH4, CH3OCH3, and CO2 species. The concentration of each species was determined by biogas compositions and DME content in the fuel mixture. The second fuel stream was C12H23 diesel. The discrete phase and diesel liquid evaporation are described via the Taylor Analogy Breakup (TAB) model [54]. The characteristics of the fuel mixture were calculated based on the thermodynamic properties of each component given in the Chemkin table.

2.2.6. Boundary Conditions

The boundary conditions include pressure and air temperature at the inlet of the intake manifold; pressure, temperature, and compositions of the fuel mixture at the inlet of the biogas-DME nozzle; and temperature, injection speed, diesel mass flow at the pilot injection. The composition of substances at the inlets is determined by mixture fraction $f$, which is between $f = 1$ (fuel mixture only) and $f = 0$.
In each calculation, the equivalence ratio is determined by the fuel and oxygen compositions introduced into the cylinder. Thus, all pressure losses on the intake manifold are considered.

2.2.7. Numerical Methodology

In the present work, the 3D pressure-based implicit unsteady solver available in ANSYS Fluent code was used to solve the basic governing equations (mass, momentum and energy). The equations were spatially discretized using the finite volume method using the standard scheme for pressure interpolation. The discretization scheme for the convective term of transport equations used the first-order upwind scheme. The pressure-velocity coupling in the discretized equations was performed using the semi-implicit method for pressure linked equations (SIMPLE) algorithm to solve the pressure field.

In the biogas-DME hybrid dual-fuel engine, the fuel compositions can be varied to fit the operating conditions. Thus, in this study, the DME content, the biogas compositions, and the diesel injection were varied. The performance and emissions of the engine were examined as DME varied in the range of 0%–40%, CH$_4$ in biogas varied in the range of 60%–80%. The energy brought into the engine by different fuel compositions was described by the equivalence ratio of the hybrid fuel mixture $\phi_{a/b}$, where $\phi_{gas} = a$ was the contribution of the biogas and DME part into the global equivalence ratio $b$. The diesel contribution in the equivalence ratio was thus, $\phi_{die} = b - a$. In this study, $\phi_{die}$ varied in the range of 0–0.2.

3. Results and Discussion

3.1. Mixture Preparation and Combustion Analysis

Figure 2a illustrates the contours of HC, oxygen concentrations, and airflow speed field in the biogas-DME hybrid dual engine at four positions of crankshaft angle 50°, 330°, 350° and 355° CA without combustion. Biogas-DME in the gaseous state after injection diffused quickly into the airflow. The fuel distribution was not homogenous in the intake manifold. The rich mixture area was on the opposite side of the biogas-DME nozzle, resulting in a slightly higher fuel concentration in the cylinder towards the intake port.

During the compression process, the fuel-air mixture distribution in the cylinder depended on the gas speed field, which was very strong at the top of the combustion chamber (Figure 2a). Therefore, at the end of the compression process, the mixture in the half-space of the combustion chamber toward the inlet port was slightly richer than the other side. After the closure of the intake valve, the amount of fuel and air introduced into the cylinder was unchanged; thus, the global equivalence ratio of the mixture remained stable until the starting of diesel injection for ignition (Figure 2b).

When diesel fuel was injected, the area with high fuel concentration was concentrated locally in the jet axis (Figure 2a). Figure 2b shows that the density of diesel droplets (DPD) appears during the diesel injection phase and disappears at the end of the injection process. This indicates that diesel droplets evaporate rapidly due to the high gas temperature at the end of the compression process. When the evaporation of liquid fuel droplets is complete, the global equivalence ratio of the mixture reaches a new stable value.

For dual-fuel mode combustion, the main combustion mixture was prepared prior to ignition; therefore, the remaining oxygen content in the combustion chamber was reduced compared to the CI diesel engine. This affected the ignition properties of the pilot injection in the dual-fuel combustion mode. Figure 2a shows that in the case of the biogas-DME hybrid dual-fuel engine, in the region of diesel jet, the oxygen concentration decreased by more than 5% compared to the remaining area and more than 15% compared to the oxygen component present in the original air. This is an essential factor that needs to be considered in the organization of the combustion process of a biogas-DME hybrid dual-fuel engine. The combustion chamber should be appropriately designed to ensure that the oxygen concentration around the diesel jet is enough for ignition.
Figure 2. Fuel-air mixing process in the biogas-DME hybrid dual-fuel engine (M7C3-30% DME, $n = 2000$ rpm, $\phi = 0.83/1.00$, without combustion). (a) Contours of HC, $O_2$ concentrations and velocity field at 50, 330, 350, and 355 °CA. (b) Variation of global equivalence ratio $\phi$ and diesel particles density (DPD) with crank angle.

Figure 3a presents the contours of temperature, fuel concentrations, soot volume fraction, CO, and NO$_x$ concentrations at top dead center (TDC) during the combustion when the engine was fueled with M7C3-20DME operating at 2000 rpm. At this moment, the diesel injection was completed, and diesel was mostly burnt. There was a small amount of unburnt diesel near the nozzle leading to a slightly rich mixture in this area (Figure 3a).
Combustion is complete. The gas temperature reaches the maximum value as the flame front propagates in the narrow kernel between the cylinder head and the piston head with a lean mixture. The gas temperature reaches about 1250 °K, as shown in Figure 3d. The formation of NOx mainly occurred after the second peak, when the piston passed TDC, and the combustion was taking place (Figure 3a). After the second peak, soot volume fraction decreased due to soot oxidation. The soot concentration in the exhaust gas was much lower than its maximum value at 370 °CA when the engine was fueled with biogas M7C3 and M7C3-40% DME operating at 2000 rpm. At this moment, the diesel injection was completed, and diesel was mostly burnt. There was a small amount of unburnt diesel near the nozzle leading to a slightly rich mixture in this area (Figure 3a).

The results of Figure 3d reveal that the CO was generated at the beginning of the combustion process. The concentration of CO increased gradually and reached its maximum value at about 380 °CA then decreased due to CO oxidation. It can be seen in Figure 3a that CO was concentrated mainly with M7C3-20DME operating at 2000 rpm. When diesel fuel was injected, the area with high fuel concentration was concentrated locally in the jet axis (Figure 2a). Figure 2b shows that the density of diesel droplets (DPD) appears during the diesel injection phase and disappears at the end of the injection process. This indicates that diesel injection phase was completed.

Figure 3b shows that the combustion process of the biogas-DME hybrid dual-fuel engine can be divided into two phases. During the first phase, the rate of fuel consumption is very high, and the diesel concentration sharply decreases, leading to a significant rise in the heat release rate curve, as shown in Figure 3c. This is due to the combustion taking place in the main space of the combustion chamber with considerable ignition energy of the pilot jet. During the second phase, the rate of fuel consumption reduces because the flame front propagates in the narrow kernel between the cylinder head and the piston head with a lean mixture. The gas temperature reaches the maximum value as the combustion is complete.

**Figure 3.** Variation of temperature and pollutant concentrations with crank angle during the combustion and expansion process (M7C3-30% DME, n = 2000 rpm, φ 0.83/1.00). (a) contours of temperature, HC, Diesel, CO, NOx concentrations and soot volume fraction at TDC; (b) variation of total fuel concentration and diesel concentration; (c) variation of in-cylinder pressure and heat release rate; (d) variation of T, NOx, CO, soot volume fraction with crank angle.
The results of Figure 3d reveals that the CO was generated at the beginning of the combustion process. The concentration of CO increased gradually and reached its maximum value at about 380 °CA then decreased due to CO oxidation. It can be seen in Figure 3a that CO was concentrated mainly in the burnt mixture, behind the flame front. CO concentration reached a stable equilibrium value, which depended on the equivalence ratio and temperature.

The formation of NOx was recorded later when the piston passed TDC, and the combustion temperature reached about 1250 °K, as shown in Figure 3d. The formation of NOx mainly occurred in the reaction region with a high temperature (Figure 3a). NOx concentration reached the maximum value at approximately the same position with the peak of temperature. It then remained almost a stable value until the end of the expansion process.

Soot concentration appeared at the beginning of diesel combustion and reached its first peak as the complete combustion of diesel was achieved, as shown in Figure 3d. The second peak occurs as the temperature peaks. These results show that soot concentration depended on diesel concentration and temperature. The highest soot concentration was in the border of the diesel jet, where diffusion combustion was taking place (Figure 3a). After the second peak, soot volume fraction decreased due to soot oxidation. The soot concentration in the exhaust gas was much lower than its maximum value (Figure 3d).

3.2. Effects of Dimethyl Ether (DME) Content

Figure 4 compares the contours of fuel concentrations, temperature, NOx, and soot volume fraction at 370 °CA when the engine was fueled with biogas M7C3 and M7C3-40% DME operating at 2400 rpm. The equivalence ratio in both cases was φ 0.77/0.97 and the same injection timing at 350 °CA. It can be observed that at the same crank angle position, the volume of burnt mixture in the case of M7C3-40% DME fueling mode was larger than that of biogas M7C3 fueling mode. This was due to the increase of flame speed with the addition of DME into biogas. The rise in fuel consumption led to a rise in temperature, and thus, an increase in NOx and soot formation at the first phase of the combustion process. The results show that the maximum temperature, NOx, and soot volume fraction in the case of biogas fueling mode were 2450 °K, 1500 ppm and 0.16 ppm as compared to 2550 °K, 3600 ppm and 0.4 ppm, respectively, in case of biogas M7C3-40% DME fueling mode.

<table>
<thead>
<tr>
<th>HC (%)</th>
<th>0 DME</th>
<th>40% DME</th>
</tr>
</thead>
<tbody>
<tr>
<td>0/7</td>
<td>0/6</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T (K)</th>
<th>900/2450</th>
<th>950/2550</th>
</tr>
</thead>
<tbody>
<tr>
<td>0/1500</td>
<td>0/3600</td>
<td></td>
</tr>
</tbody>
</table>

| NOx (ppm) | 0/0.16 | 0/0.4 |

| fv (ppm) | 0/0.4 |

**Figure 4.** Contours of fuel concentrations, temperature, NOx, and soot volume fraction at 370 °CA when the engine is fueled with biogas M7C3 and M7C3-40% DME (n = 2400 rpm, φ 0.77/0.97).
Figure 5a shows the effects of DME content in the mixture with biogas on in-cylinder pressure variation when the engine operated at 1200 rpm with the equivalence ratio of $\phi = 0.77/0.94$. It can be observed that with a given equivalence ratio, the maximum pressure increased with the DME content. This confirmed the results of Park et al. [42]. DME substitution improved the heat value of the fuel and combustion velocity leading to an increase in engine efficiency. The simulation results showed that with the same fuel supply conditions and the same equivalence ratio, the engine cycle work increased by 6% and 10%, respectively, with 20% and 40% DME addition in mixture with biogas M7C3 at an engine speed of 1200 rpm.

![Graphs showing effects of DME content on in-cylinder pressure, temperature, NOx, and CO concentration](image)

**Figure 5.** The effects of DME content in the mixture with biogas on combustion characteristics and pollutants formation (M7C3, $n = 1200$ rpm, $\phi = 0.77/0.97$). The effects of DME content on variation with crank angle of pressure (a), temperature and NOx concentration (b), diesel concentration and soot volume fraction (c), and CO concentration (d).

Under the same operating conditions, the increase of DME content led to a rise in temperature, which resulted in an increase in NOx concentration in the exhaust gas (Figure 5b). The addition of 40% DME into M7C3 biogas led to an increase of NOx concentration in the exhaust gas by 40% as compared to biogas only fueling mode. The concentration of soot depended mainly on the diffusion combustion of the diesel jet. The high temperature of the combustion products as there was increasing DME content in the mixture with biogas increased the soot production rate. However, in this case, the soot oxidation reaction rate became more intensive. Figure 5c shows that, as compared to M7C3...
biogas fueling mode, M7C3-40% DME mixture fueling mode produced almost the same soot maximum concentration but only produced a 15% value of soot concentration in the exhaust gas.

Since CO concentration was in the equilibrium state of the gas-water reaction, although the poor mixture \( \phi = 0.77/0.94 \) was used, there was an amount of CO in the exhaust gas. Figure 5d compares the variation of CO concentration with respect to the crankshaft angle when the engine was fueled with neat biogas M7C3 and with mixtures M7C3-20% DME, M7C3-40% DME. In the case of the M7C3-40% DME mixture, the CO concentration in the exhaust decreased by 30% as compared to the neat biogas case. This was due to the increase in the combustion rate with DME content in the fuel mixture, which resulted in complete combustion.

Figure 6a,b show the effects of biogas compositions on the variations of \( W_i \), CO, NO\(_x\), and soot volume fraction with DME content in the fuel mixture. At any biogas compositions, when DME content increased, \( W_i \) and NO\(_x\) increased while CO and soot volume fraction decreased. At a given DME content, the increase in \( \text{CH}_4 \) concentration in biogas led to an increase in \( W_i \) and NO\(_x\) emission. This was due to the increase in temperature resulted from the rise in the heat value of biogas. The effects of DME on \( W_i \) and soot volume fraction were more significant with poor biogas than rich biogas. As DME content rose from 0% to 40%, \( W_i \) increases by 20% and 10% with biogas M6C4 and biogas M8C2, respectively. It can be seen in Figure 6b, that as the engine was fueled with neat biogas, soot volume fraction rose by 30% as \( \text{CH}_4 \) concentration in biogas increased from 60% to 80%. When 40% DME was added to the biogas, the soot volume fraction was almost independent with biogas compositions. This was because the heat value ratio between DME and biogas was higher in the case of poor biogas than that in the case of rich biogas. By contrast, the effects of DME content on CO and NO\(_x\) concentrations were stronger with the increase of \( \text{CH}_4 \) concentration in biogas. This was due to the increase in temperature, and complete combustion resulted from the rise in laminar flame speed at high concentrations of \( \text{CH}_4 \) and DME in the fuel mixture.

![Figure 6](image_url)

**Figure 6.** Variation of \( W_i \), CO concentration (a); NO\(_x\) concentration and soot volume fraction (b) with DME content in biogas M7C3 under effects of biogas compositions \((n = 1200 \text{ rpm}, \phi = 0.77/0.97)\).  

### 3.3. Effects of Biogas Composition

With a given operating condition, equivalence ratio, and DME content, the biogas compositions only slightly affected the indicative engine cycle work and gas temperature. Simulation calculations showed that as the engine operated at a speed of 1800 rpm with biogas M6C4, M7C3, M8C2 enriched by 20% DME with \( \phi = 0.77/0.98 \), the indicative engine cycle work varied from 957 J/cyc to 985 J/cyc, i.e., it only increased by 3% when shifting from biogas M6C4 to biogas M8C2. The maximum gas temperature increased by 50 °K for the same variation of biogas compositions, as shown in Figure 7a. This can be attributed to the fact that the heat value of the fuel increased with the increase of \( \text{CH}_4 \)
content in the biogas. As shown in Figure 7a, with the same advanced ignition angle, the combustion in the case of M8C2 started earlier due to higher laminar flame speed. Because the maximum temperature occurred near TDC, the peak of pressure was higher. However, the indicative engine cycle work was slightly changed due to the increase of negative work at the end of the compression process resulting from early combustion.

Figure 7. Effects of biogas compositions on combustion characteristics and pollutants formation (20% DME, \(n = 1800\) rpm, \(\phi = 0.77/0.97\)). Variation with the crank angle of pressure (a), temperature and NO\(_x\) concentration (b), diesel concentration and soot volume fraction (c), and CO concentration (d).

Since the NO\(_x\) formation is greatly affected by the combustion temperature, the concentration of NO\(_x\) increases with the increase of the CH\(_4\) component in biogas, as shown in Figure 7b. When the CH\(_4\) content in biogas rises from 60% to 80%, the NO\(_x\) concentration increases by 50%. The increase in CH\(_4\) concentration in biogas improves the combustion rate. Thus, combustion is complete, resulting in a decrease in CO concentration in combustion products. It can be seen in Figure 7c that the concentration of CO in the exhaust gas dropped by 20% when the CH\(_4\) content in biogas increased from 60% to 80%. Soot formation depends mainly on the diffusion combustion of diesel jet, so it was slightly affected by the biogas composition (Figure 7d). The results show that the soot volume fraction was about 0.02 ppm as the engine was fueled with biogas containing 60%–80% CH\(_4\) enriched by 40% DME.
3.4. Effects of Diesel Pilot Injection

Figure 8a presents the variation of in-cylinder pressure with crank angles under the effect of diesel injection. The engine was fueled with biogas M7C3-20% DME, operating at 2400 rpm. The calculation was performed with three values of $\phi_{\text{die}}$: 0, 0.10 and 0.20 with the same global equivalence ratio $\phi = 0.97$. $\phi_{\text{die}} = 0$ corresponds to the biogas-DME PCCI engine. The maximum in-cylinder pressure increased with the decrease of $\phi_{\text{die}}$ (i.e., with the increase of $\phi_{\text{gas}}$). This was because of the rise in flame speed with gaseous fuel mixture compared to diesel fuel, which improved heat release rate, as shown in Figure 8b.

![Figure 8. Effects of diesel injection on variations of in-cylinder pressure (a) and heat release rate (b) (Biogas M7C3-20% DME, $n = 2400$ rpm).](image)

In the crucial case when the amount of diesel injection drops to 0, the biogas DME hybrid dual-fuel engine can be considered a biogas-DME PCCI engine. Figure 8a shows that as it switched from dual-fuel mode to PCCI mode with the same global equivalence ratio, the in-cylinder pressure increased sharply, and the pressure peak tended to be closer to the TDC. This led to a loss in indicative engine cycle work due to the rise in pressure at the end of the compression process.

As can be observed from Figure 9a, with a given global equivalence ratio $\phi = 0.97$, as $\phi_{\text{die}}$ decreased from 0.20 to 0.10, the gas temperature increased by 80 °K. This was due to the laminar flame speed of mixture biogas-DME being higher than that of diesel, which made the heat release rate near the TDC increase. The increase in temperature resulted in a rise of NOx, as shown in Figure 9b. NOx concentration increased by 55%, with a decrease of 10% in $\phi_{\text{die}}$. However, the growth of biogas-DME content improved the homogeneity of the mixture leading to a reduction in CO concentration in the exhaust gas. Figure 9c shows that CO concentration dropped by 30% when switching from $\phi = 0.77/0.97$ fuel mixture to $\phi = 0.87/0.97$ fuel mixture. As mentioned above, soot concentration depends mainly on diesel injection. Thus, the reduction of diesel injection resulted in a reduction in soot volume fraction. Figure 9d shows that the soot volume fraction decreased by 75% as $\phi_{\text{die}}$ dropped from 0.20 to 0.15.
Figure 9. Effects of diesel injection on variations of temperature (a), NOx concentration (b), CO concentration (c), and soot volume fraction (d) with crank angle (Biogas M7C3-20% DME, n = 2400 rpm).

As previously stated, although the amount of diesel injection for pilot ignition is small, it significantly affects the formation of pollutants due to the diffusion combustion characteristics of diesel fuel. The result in Figure 9a shows that the combustion temperature decreased by 80 °K as $\phi_{\text{die}}$ increased by 0.1, leading to a reduction in NOx concentration. Figure 9b shows that NOx concentration increased by 20% when switching from hybrid dual-fuel mode with $\phi_{0.87/0.97}$ to PCCI mode with the same global equivalence ratio $\phi_{0.97}$. Figure 9c shows that when switching from dual-fuel mode with $\phi_{0.77/0.97}$ to PCCI mode with the same global equivalence coefficient $\phi_{0.97}$, the CO concentration in the exhaust gas decreased from 1.6% down to 0.2%, meaning a decrease of 80%. This can be attributed to the fact that without diesel injection, the gaseous fuel mixture in the combustion chamber was homogeneous; thus, the combustion took place completely, resulting in a reduction of CO emission. Similarly, without a pilot jet, the diffusion combustion does not exist in the combustion chamber; therefore, the conditions of soot nuclei formation are eliminated. As a result, the soot concentration in the exhaust gas of the biogas-DME PCCI engine was almost negligible compared to soot volume fraction $f_{v} = 0.041$ ppm in the case of a biogas-DME hybrid dual-fuel engine with $\phi_{\text{die}} = 0.20$ (Figure 9d).
Figure 10a,b present the effects of DME content on the variations of $W_i$, CO, NO$_x$, and soot concentrations with $\phi_{\text{gas}}$. It can be seen from these figures, generally, at a fixed global equivalence ratio $\phi$ when $\phi_{\text{gas}}$ increased (i.e., $\phi_{\text{diesel}}$ decreases) $W_i$ and NO$_x$ concentration went up while CO concentration and soot volume fraction decreased. Otherwise, the effects of DME content on the combustion properties and pollutant emissions were more significant at high $\phi_{\text{gas}}$. This means that the addition of DME clearly improves the combustion characteristics of biogas over those of diesel because the properties of DME and diesel were not significantly different. Figure 10a shows that as the engine was fueled with biogas M7C3-40% DME, $W_i$ increased by 6% at $\phi_{\text{gas}} = 0.77$ but increased by 11% at $\phi_{\text{gas}} = 0.97$ as compared to biogas fueling mode. The reduction of CO emission was 15% and 25% at $\phi_{\text{gas}} = 0.77$ and $\phi_{\text{gas}} = 0.97$, respectively, at switching from biogas dual-fuel mode to biogas-DME hybrid dual-fuel mode with 40% DME in the mixture with biogas (Figure 10a).

![Graphs](a) and (b)

**Figure 10.** Variation of $W_i$, CO concentration (a); NO$_x$ concentration and soot volume fraction (b) with $\phi_{\text{gas}}$ under effects of DME content in biogas M7C3 ($n = 2400$ rpm).

As mentioned above, the combustion temperature was in line with DME content or $\phi_{\text{gas}}$, and it affected the NO$_x$ formation. It can be seen in Figure 10b, the NO$_x$ concentration increased by 15% at $\phi_{\text{gas}} = 0.77$ and 30% at $\phi_{\text{gas}} = 0.97$ as there was switching from biogas dual-fueling mode to biogas-40% DME hybrid dual-fueling mode. This is because DME effects on biogas combustion properties were more substantial than those on diesel fuel.

The effects of DME on the variation of soot volume fraction with $\phi_{\text{gas}}$ were significant, as shown in Figure 10b. The addition of 40% DME to biogas M7C3 led to a reduction of 60% in soot volume fraction at $\phi_{\text{gas}} = 0.77$. The soot volume fraction decreased sharply with the increase of $\phi_{\text{gas}}$. It reduced by 65% at $\phi_{\text{gas}} = 0.77$ as it shifted from biogas M7C3 fueling mode to biogas M7C3-40% DME fueling mode. The soot emission practically vanished in the exhaust gas of the biogas-DME PCCI engine. This confirms the experimental results of Park et al. [42].

The above results show that adding DME into biogas could improve the combustion characteristics of the biogas-DME hybrid dual-fuel engine. The reduction of diesel pilot injection is particularly interesting for CO and soot emissions control. Theoretically, the biogas-DME PCCI engine (without diesel pilot injection) is ideal. However, controlling ignition timing and heat release characteristics is still a challenge, as mentioned by Reitz et al. [12]. Thus, the substitution of diesel pilot ignition by DME pilot ignition in a biogas-DME hybrid dual engine is the optimal solution for both performance improvement and emissions reduction.

The present simulation research results will be beneficial for practical application of biogas and DME on internal combustion engines. They are also essential to orient the experimental research of our future works on the topic.
4. Conclusions

The following conclusions can be drawn from the above study:

- In the biogas-DME hybrid dual-fuel engine, CO was generated at the beginning of the combustion process and concentrated mainly in the burnt mixture, behind the flame front. The formation of NO\textsubscript{x} mainly occurred in the reaction region with T > 1250 °K and reached the maximum value at approximatively the same position with the peak of temperature. Soot concentration appeared at the beginning of diesel combustion. It reached the first peak at the same crank position of complete combustion of diesel and the second peak at the same position of the maximum temperature.

- At any biogas composition, when DME content increased, W\textsubscript{i} and NO\textsubscript{x} increased while CO and soot volume fraction decreased. The effects of DME on W\textsubscript{i} and soot volume fraction were more significant with poor biogas than with rich biogas, contrary to its effect tendency on CO and NO\textsubscript{x} concentrations.

- With a given operating condition, equivalence ratio, and DME content, the biogas compositions only slightly affected the performance and emission of a hybrid dual-fuel engine. With a fixed 20% DME addition, as shifting from biogas M6C4 to biogas M8C2, the indicative engine cycle work increased by 3%, the maximum gas temperatures rose by 50 °K, the concentration of CO in the exhaust gas decreased by 20%, the NO\textsubscript{x} concentration went up by 50%, while the soot volume fraction was practically unchanged.

- At a fixed global equivalence ratio \( \phi \) when \( \phi_{\text{gas}} \) increased (i.e., \( \phi_{\text{die}} \) decreased), W\textsubscript{i} and NO\textsubscript{x} concentration increased while CO concentration and soot volume fraction decreased. The effects of DME content on the combustion properties and pollutant emissions were more significant at high \( \phi_{\text{gas}} \).

- At the same operating conditions and global equivalence ratio, biogas-DME PCCI mode was more advantageous than biogas-DME hybrid dual-mode both in engine performance and pollutant emissions except for NO\textsubscript{x}. With the same global equivalence ratio \( \phi = 0.97 \), as it shifted from biogas fueling mode to biogas-40% DME fueling mode, W\textsubscript{i} increased by 6% and by 11%, CO emission was reduced by 15% and by 25% while the NO\textsubscript{x} concentration increased by 15% and by 30% for the biogas-DME hybrid dual-fuel engine and for the biogas-DME PCCI engine, respectively. The soot volume fraction decreased sharply with the increase in \( \phi_{\text{gas}} \) in the hybrid dual-fuel engine, and it practically vanished in the PCCI engine.

- Biogas-DME PCCI combustion mode without diesel pilot injection is ideal, but the control of ignition timing and heat release characteristics is a challenge. Thus, the substitution of diesel pilot ignition by DME pilot ignition in a biogas-DME hybrid dual engine is the optimal solution both for performance improvement and pollution emissions reduction.

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Nomenclature

BDC: Bottom Dead Center
°CA: Degree of crankshaft angle
CI: Compression Ignition
DME: Dimethyl Ether
Die: Diesel vapor concentration (% in volume)
f_v: Soot volume fraction (ppm)
HC: Total fuel concentration (Methane, DME, Diesel) (% in volume)
HRR: Heat Release Rate (J/°CA)
MxCy: Biogas containing 10×% CH_4 and 10y% CO_2 (in volume)
n: Engine speed (rpm)
SI: Spark Ignition
T: Mean temperature of gas mixture in the cylinder (°K)
TDC: Top Dead Center
W_i: Indicative engine cycle work (J/cyc)
φ_{a/b}: Compositions of equivalence ratio
φ_{gas}: Equivalence ratio contributed by biogas and DME, φ_{gas} = a
φ: Global equivalence ratio, φ = b
φ_{die}: Equivalence ratio contributed by diesel, φ_{die} = b – a
ϕ: Crankshaft angle (°CA)

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