Fuel Composition Effects in a CI Engine Converted to SI Natural Gas Operation

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Abstract

Low-carbon fuels such as natural gas (NG) have the potential to lower the demand of petroleum-based fuels, reduce engine-out emissions, and increase IC engine thermal efficiency. One of the most rapid and efficient use of NG in the transportation sector would be as a direct replacement of the diesel fuel in compression ignition (CI) engines without any major engine modifications to the combustion chamber such as new pistons and/or engine head. An issue is the large variation in NG composition with the location and age of the gas well across U.S., which would affect engine operation, as well as the technology integration with emissions after treatment systems. This study used a conventional CI engine modified for spark ignition (SI) NG operation to investigate the effects of methane and a C1–C4 alkane blend on main combustion parameters like in-cylinder pressure, apparent heat release rate, IMEP, etc. Steady-state engine experiments were conducted at several operating conditions that changed spark timing, engine speed, and equivalence ratio. The study found that NG operation increased peak pressure, IMEP, and indicated thermal efficiency compared to methane, for all the operating conditions investigated in this work. This suggests caution when translating methane-based experimental observations to real world NG operation, even for NG with mostly methane as the one used in this work. As many NG studies in the literature used methane as an NG surrogate, a better understanding of real fuel effects in diesel-like combustion environments could be important for the successful conversion of conventional diesel engines to NG operation.

Introduction

Internal combustion (IC) engine is the main power source for on- and off-road vehicles. Most of these vehicles use petroleum-based fuels therefore the recent increased interest in alternative fuels that can reduce the U.S. dependence on petroleum imports. In addition, alternative fuels could help achieve the increasingly stringent emission regulations [1, 2].

Natural gas (NG) is an attractive alternative fuel due to its high availability and low price. Compared to gasoline or diesel, NG's higher hydrogen-to-carbon (H/C) ratio can lower carbon dioxide (CO2) emissions for the same power output. NG does not need to vaporize before mixing with air, which avoids the creation of fuel-rich regions that can produce particulate matter (PM) emissions [3]. Further, the absence of aromatic compounds in NG greatly reduce the formation of soot precursors. NG's higher octane number (ON) compared to gasoline reduces the knocking potential and allows a spark-ignited (SI) engine to operate at a higher compression ratio (CR), which increases thermal efficiency [4, 5]. NG engines can operate leaner, which reduce carbon monoxide (CO) and unburned hydrocarbons (HC) emissions. More, lean operation lowers combustion temperature, which reduce nitrogen oxides (NOx) emissions. For example, Catania et al., [6] evaluated engine out emissions on a 2 L engine and found out that engine emitted NOx emissions produced by CNG are 35% lower than gasoline emissions. Mello et al. [7] found a reduction of 90% and 55% in CO and HC emissions, respectively, for light duty vehicles converted from diesel to SI NG.

Despite NG advantages over petroleum-based fuels, only a small number of NG vehicles are registered in the U.S. compared to the rest of the world. Figure 1 shows that just

![Figure 1: Natural gas vehicles distribution worldwide][1]
0.8% of total NGVs worldwide (more than 23 million vehicles) are registered in the U.S., compared to 66% in the Asian countries (about 15.7 million NGVs) [8].

While the NG consumption in the U.S. transportation sector increased by 73% over the past 10 years and it is predicted to increase by another order of magnitude by 2040 [9], there are still several issues that need addressed, such as the low number of commercial NG fueling stations [10], the large variation in the chemical composition of the NG with origin, or the limited studies in the literature of modern NG-fueled engines compared to those dedicated to petroleum-based fuels [11, 12]. This work works on addressing the varying NG composition across U.S., including the increased liquids content in shale-gas compared. For example, the main constituent of NG, methane, can vary from 75% to 96%, with balance containing heavier hydrocarbons like ethane, propane, butane, etc. along with nitrogen, carbon dioxide, and traces of sulfur compounds [13]. As a result, the chemical and thermodynamic properties of non-methane compounds in the NG can have a strong influence on the engine efficiency and emissions [14].

Several relationships are generally used to characterize the effects of the NG composition. The two most important are the Methane Number (MN) [15], which describes the gas resistance to knock, and the Wobbe Index (WI), which describes the fuel’s combustion energy output. Both MN and WI affect engine’s performance. The higher the MN, the greater its resistance to autoignition. Increasing the percentage of heavier hydrocarbons in the NG reduces MN, making the gas more susceptible to knocking [15, 16]. On the other hand, WI is an indicator of fuel interchangeability. Feist et al. tested the response of heavy-duty NG engines over a wide range of NG compositions. Their study showed that the power output of the engine increased slightly when MN decreased and WI increased [17]. A similar finding was shown by Kim et al. [18], which found that higher WI increased the cylinder pressure, which increased the power output. Lee and Kim [19] showed that vehicle’s fuel economy is proportional to the lower heating value (LHV) of the gas composition. McTaggart et al. [20] found that fuel density was higher for lower MN, which influenced the amount of fuel injected in the cylinder. As a result, a lower MN increased the peak cylinder pressure and the heat release rate of the engine. Karavalakis et al. found that a larger fraction of heavier hydrocarbons (i.e., a lower MN) increases the adiabatic flame speed, which can affect both efficiency and emissions [21]. Karim and Wierzba [22] and Amirante et al. [23] found that propane can speed up and stabilize the combustion process. El-Sherif showed that ethane produces a similar effect under lean conditions, due to formation of enhanced radicals and more hydrogen atoms [24]. Spadaccini and Colket [25] found that butane addition had a higher impact on ignition timings than ethane or propane. The reason is the effect that these larger hydrocarbons have on the flame speed. Dirrenberger et al. [26] found that laminar flame speeds of alkanes are in the order of propane > ethane > butane > methane. Whereas, Bosschaart and de Goey [27] inferred that ethane is faster burning than propane or butane, at stoichiometric conditions. Ranzi et al. [28] confirmed their findings by showing that ethane and methane have the highest and lowest flame speeds respectively.

If the effects of NG composition are well understood, the most rapid and efficient use of NG in the transportation sector would be as a direct replacement of the diesel fuel in compression ignition (CI) engines without any major engine modifications to the combustion chamber such as new pistons and/or engine head. However, it is difficult to initiate and control NG combustion process in a conventional CI engine without the use of an additional ignition source due to NG’s higher autoignition temperature compared to diesel fuel. As a result, a CI engine is usually converted to NG operation by replacing the fuel injector with a spark plug and using port-fuel injection for NG delivery. Other strategies can be used (e.g., dual fuel, partial stratification) but they generally require major engine design modifications, increase the engine-control complexity and/or the engine cost. Therefore, the main objective of this study was to investigate the effect of NG composition on the combustion characteristics of a conventional heavy-duty direct-injection CI engine converted to port-fuel injection SI operation. No changes to the combustion chamber were made and no advanced/specialized hardware was used. The fuels tested were pure methane and a C1-C4 alkane blend with a composition representative of shale gas in West Virginia.

**Experimental Setup**

The experiments were performed in a single-cylinder, four-stroke, port fuel injection, heavy-duty SI engine (Ricardo/Cussons, U.K.; Model Proteus). Figure 2 shows the details of the experimental facility. The original engine configuration is based on a supercharged, direct-injection diesel engine (Volvo, Model TD 120). The engine was modified to a SI configuration by replacing the original diesel injector with a NG spark plug (Stitt, USA, Model S-RSGN40XLBEX8.4-2). A
port fuel NG injector (Rail Spa, Italy, Model IG7 Navajo, 3 seats) positioned 55 mm upstream of the intake port delivered the fuel to the intake system. Only one of the three injector seats were used in this study, with the seat using a 3.5-mm diameter nozzle. A piezo-electric pressure transducer (Kistler, Model 6011) that was installed in the glow plug location and connected to a charge amplifier (Kistler, Model 5010) measured in-cylinder pressure. Table 1 details important engine and fuel injection specifications.

An aftermarket engine management system (Megasquirt, Model 3X) controlled fuel injection and spark timing. In-house-built data acquisition software (Scimitar) collected operating data such as engine speed, torque, air, coolant, and oil temperature, and air mass flow.

Steady-state engine experiments were conducted at several operating conditions that changed spark timing (ST) from −30 deg ATDC to −10 deg ATDC, equivalence ratio, from 0.71 to 0.80 for methane and 0.69 to 0.76 for NG (corresponding to a variation in engine controller’s fuel load from 75% to 90%), and the engine speed from 900 rpm to 1300 rpm. The engine oil and coolant temperatures were maintained at constant throughout the experiments. The plots in the results section show the average of minimum 300 cycles at each operating condition.

The fuels in this study were methane and a C1-C4 alkane blend (called NG in the narrative) with a composition representative of the shale-gas from the Appalachian region. Table 2 shows the detail chemical composition of the two fuels.

Results and Discussions

Several subsections present and discuss the experimental results. The first subsection discusses the effect of ST at constant speed and equivalence ratio. The analysis included the maximum in-cylinder pressure and its location, the start of combustion (SOC; defined as the crank angle corresponding to 10% heat release), and the combustion duration (DOC; defined as the difference between the crank angles corresponding to 90% and 10% heat release). The next subsection shows the effect of engine speed on the combustion phenomena at constant ST and equivalence ratio. Finally, a parametric study that changed the equivalence ratio, $\phi$, was performed at constant ST and engine speed.

Spark Timing Effects

ST controls in-cylinder flame development and propagation, which affects combustion efficiency and stability [29]. Advanced ST usually increases peak pressure, which can benefit engine efficiency and improve combustion stability, but it also increases in-cylinder temperature, which can result in knocking, especially at low speed and high load conditions. Retarding the spark timing can avoid knocking, but it also reduces the peak pressure while moving its location further in the expansion stroke, and increases the probability of incomplete combustion or misfiring (hence a lower thermal efficiency), particularly at high speed and low load operating conditions. In this study, the ST timing sweep from −30° CA to −10° CA ATDC at 900 rpm and a methane equivalence ratio of 0.73 captured both the effect of ST on the peak in-cylinder pressure and the maximum brake torque (MBT) timing, which are two important design parameters. Figure 3a showed that advancing ST increased peak in-cylinder pressure. More, Table 3 indicates that advancing ST first reduced the combustion duration, then increased it. It is interesting that Fig. 3b shows a double heat release peak for the most advanced ST, which is probably due to the effect that the piston bowl edges may have on redirecting the flame front. Figure 4 shows that IMEP first increased by 3% when ST advanced to −20 CAD ATDC, then rapidly decreased by 6% when ST advanced another 10 CAD. More, the same MBT timing of −20 CAD ATDC also corresponded to the maximum AHRR in Fig. 3b and to the shortest DOC in Table 3. Despite the lack of a ST sweep for the NG, it is helpful to compare NG and methane combustion processes one operating condition only (i.e., at ST = −10° ATDC, N = 900 rpm, and $\phi = 0.73$). Figure 5a shows that NG’s higher WI resulted

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1. A controller malfunction during the spark time sweep was discovered during data analysis. As a result, the spark time sweep was repeated, but only for methane (i.e., there is no spark time sweep data for NG).
in a 12% higher and slightly advanced peak pressure compared to methane, at similar ST. At the same time, Table 3 and Figs. 5b show that NG had a slightly earlier and faster combustion compared to methane. The possible reasons are explained in the next section.

**Engine Speed Effects**

The engine speed sweep was performed at constant ST and $\phi$, which means that combustion was not optimized for higher speeds. Engine speed generally affects combustion phenomena through its effect on in-cylinder gas motion, friction, the time available to complete combustion, and the heat transfer rates [29]. The heat transfer importance usually decreases at higher engine speeds. However, the speed increases the friction importance and reduces in-cylinder residence time. Figure 6a suggests that friction and reduced cycle duration affected more the combustion process than the reduced importance of the heat transfer. The peak pressure in Fig. 6a decreased with increased engine speed for both methane and NG, with the highest in-cylinder pressure observed for an engine speed of 900 rpm. At the same time, NG had in average an 8% higher peak pressure compared to methane at all investigated speeds.

**TABLE 3** Effect of spark timing and fuel on combustion parameters ($N = 900$ rpm and $\phi = 0.73$)

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Spark timing (deg ATDC)</th>
<th>P&lt;sub&gt;peak&lt;/sub&gt; (MPa)</th>
<th>$P_{\text{peak}}$ location (deg ATDC)</th>
<th>CA&lt;sub&gt;10&lt;/sub&gt; (deg ATDC)</th>
<th>CA&lt;sub&gt;10-90&lt;/sub&gt; (deg)</th>
<th>$\eta_{\text{th},i}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH&lt;sub&gt;4&lt;/sub&gt;</td>
<td>-30</td>
<td>6.45</td>
<td>5.5</td>
<td>-10.4</td>
<td>54.3</td>
<td>31.0</td>
</tr>
<tr>
<td></td>
<td>-25</td>
<td>6.07</td>
<td>6.3</td>
<td>-5.9</td>
<td>53.7</td>
<td>32.1</td>
</tr>
<tr>
<td></td>
<td>-20</td>
<td>5.68</td>
<td>9.7</td>
<td>-1.5</td>
<td>52.9</td>
<td>32.9</td>
</tr>
<tr>
<td></td>
<td>-15</td>
<td>5.04</td>
<td>13.6</td>
<td>1.0</td>
<td>54.1</td>
<td>32.5</td>
</tr>
<tr>
<td></td>
<td>-10</td>
<td>4.27</td>
<td>18.4</td>
<td>7.6</td>
<td>55.7</td>
<td>31.8</td>
</tr>
<tr>
<td>NG</td>
<td>-10</td>
<td>4.84</td>
<td>16.4</td>
<td>6.2</td>
<td>55.2</td>
<td>36.0</td>
</tr>
</tbody>
</table>

FIGURE 3 Effect of spark timing on in-cylinder pressure and AHRR (methane at 900 rpm and $\phi = 0.73$)

FIGURE 4 Effect of spark timing on IMEP (methane at 900 rpm and $\phi = 0.73$)

FIGURE 5 Effect of fuel and spark timing on (a) in-cylinder pressure, (b) AHRR (ST = -10° ATDC, N = 900 rpm, and $\phi = 0.73$).
However, the differences in peak pressure between the two fuels decreased from 11% at 900 rpm to 5.5% at 1300 rpm, which suggests smaller differences between fuels if engine speed would increase more.

In addition, Table 4 and Figs. 6b show that NG had a slightly earlier SOC compared to methane, but similar DOC except NG at 1300 rpm. This can be explained by the lower energy required to break a C-C bond compared to the energy required to break a C-H bond [30]. As ethane and propane have better ignition quality and faster laminar flame speed compared to methane, the data suggests improved early flame inception and propagation. More, the higher peak pressure indicates higher in-cylinder bulk temperature, which would affect both combustion efficiency and emissions formation and oxidation. The hypothesis of a better combustion process for NG is supported by the increased IMEP and thermal efficiency, $\eta_{th}$, for NG operation, as seen in Figure 7. However, both IMEP and $\eta_{th}$ decreased linearly with speed in the range of engine speeds investigated here, regardless of the fuel, probably due to a combination of higher engine friction and reduced in-cylinder residence time. However, the decrease was not the same for both fuels, with IMEP and $\eta_{th}$ decreasing by 8.5% and 5% for methane compared to 10% and 10% for NG, respectively. These differences are probably associated to an increase in the heat transfer to the coolant due to the higher in-cylinder bulk temperature for the NG operation.

### Engine Load Effects

This section presents the engine load effects on in-cylinder processes, at a constant spark timing of -10° ATDC and an engine speed of 900 rpm. The results are presented in terms of “fuel load”, which is the term that the engine controller used to describe an increase in the fuel injection time at constant engine speed. For example, every 5% increase in fuel load done by the engine controller corresponded to an increase in injection duration by ~2 ms, at an engine speed of 900 rpm.

### Table 4

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Speed (rpm)</th>
<th>$P_{\text{peak}}$ (MPa)</th>
<th>$P_{\text{peak}}$ location (deg ATDC)</th>
<th>CA10 (deg ATDC)</th>
<th>CA10-90 (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH$_4$</td>
<td>900</td>
<td>4.38</td>
<td>18.3</td>
<td>7.0</td>
<td>47.4</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>4.25</td>
<td>18.3</td>
<td>7.6</td>
<td>48.3</td>
</tr>
<tr>
<td></td>
<td>1100</td>
<td>4.04</td>
<td>19.0</td>
<td>8.0</td>
<td>48.9</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>3.90</td>
<td>19.5</td>
<td>8.3</td>
<td>48.2</td>
</tr>
<tr>
<td></td>
<td>1300</td>
<td>3.69</td>
<td>20.3</td>
<td>9.0</td>
<td>48.1</td>
</tr>
<tr>
<td>NG</td>
<td>900</td>
<td>4.87</td>
<td>16.8</td>
<td>6.2</td>
<td>49.1</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>4.54</td>
<td>17.9</td>
<td>7.1</td>
<td>47.5</td>
</tr>
<tr>
<td></td>
<td>1100</td>
<td>4.38</td>
<td>17.9</td>
<td>7.1</td>
<td>48.7</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>4.14</td>
<td>18.6</td>
<td>7.5</td>
<td>47.2</td>
</tr>
<tr>
<td></td>
<td>1300</td>
<td>3.89</td>
<td>20.1</td>
<td>8.6</td>
<td>43.8</td>
</tr>
</tbody>
</table>
Table 5 indicate how the fuel load increase from 75% to 90% related to the equivalence ratio of the fuel-air mixture. Hence, an increase in fuel load percentage resulted in an increase in $\phi$.

A higher engine load at constant speed usually reduce the importance of friction, but increases the heat transfer rate [29]. Figure 8 shows that the increase in fuel load raised the peak pressure for both fuels due to higher AHRR, but the increase was not similar. Methane peak pressure increase by 17% when the load increase from 75% to 90%, compared to NG who experienced a 14% increase. At the same time, the differences in peak pressure between the two fuels first increased from 6% to 10% when fuel load increased from 75% to 80%, then decreased to just 2% at a fuel load of 90%. Again, this suggests smaller differences between fuels if engine load would increase further at this speed, probably due to an increase importance of the heat transfer at this non-optimized operating condition.

Figure 9 also shows some interesting effects that fuel load had on IMEP and $\eta_{th}$. A 10% increase in fuel load corresponded to a 10% in IMEP for both fuels. However, IMEP for both fuels reached a plateau at 90% fuel load. A similar observation could be made for the thermal efficiency, which reached a maximum at 85% and 80% fuel load for methane and NG, respectively. One would expect that the increase in the fuel load would increase the turbulent flame speed, which would result in a more complete combustion process. However, both the heat transfer rates and the amount of fuel trapped in the crevices would increase at higher equivalent ratios. It is unknown how the piston shape affects near-wall combustion, hence the amount of unburned fuel or incomplete combustion. Future work will employ emissions measurements to try to answer these questions.

Engine load affected SOC and DOC. As expected, the higher fuel load improved flame development, which is shown by the faster SOC regardless of fuel. A similar Trend was observed for DOC, which reduced with fuel load.
Summary and Conclusions

An experimental investigation was conducted on a single cylinder diesel engine modified to SI operation. This was done by replacing the fuel injector with a spark plug and adding a port fuel injection to the intake manifold just before the intake valve for fuel delivery. As most NG studies used methane as an NG surrogate, this study used methane and a C1-C4 alkane blend (called NG in the text) to observe the effect of NG composition on main combustion parameters like in-cylinder pressure, apparent heat release rate, IMEP, etc. Steady-state engine experiments were conducted at several operating conditions that varied spark timing, engine speed, and equivalence ratio (noted as “fuel load” in the text to mimic the engine controller nomenclature). Following is a summary of the main findings of the study.

- The higher WI for the C1-C4 alkane blend increased in-cylinder pressure compared to methane, for all operating conditions
- While spark timing affected engine performance, there was no evidence of knocking at advanced spark timing. However, the IMEP decreased at the most advanced spark timing.
- The C1-C4 alkane blend had an earlier SOC, but similar DOC. This was probably due to the better ignition quality and higher laminar flame speed for C2-C4 alkanes compared to methane.
- IMEP and thermal efficiency both decreased linearly with increased speed, at constant spark timing and equivalence ratio, regardless of fuel. However, the fuel affected the decrease magnitude.
- Fuel load had a similar effect on IMEP and indicated thermal efficiency for both fuel. However, NG had the thermal efficiency peak at a lower fuel load, which suggests that real NG must run leaner than methane to achieve its best efficiency.

The main conclusion of the study is that the differences in combustion parameters between methane and NG suggest caution when translating methane-based experimental observations to real world NG operation, even for NG with mostly methane as the one used in this work. As many investigations in the literature used methane as NG surrogate, understanding the effect of the real fuel is important for the successful conversion of conventional diesel engines to NG operation.

Contact Information

References

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Nomenclature

DOC - Duration of Combustion
MBT - Maximum brake torque
MN - Methane number
NG - Natural gas
SOC - Start of Combustion
WI - Wobbe Index
ϕ - Equivalence ratio