Cylinder-to-Cylinder Variations in Diesel Dual Fuel Combustion under Low-load Conditions

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Abstract

Diesel dual-fuel (DDF) operation is a promising alternative engine operating mode. DDF combustion can achieve lower soot and NOx emissions compared to conventional diesel engine operations. However, DDF engine operations suffer from high HC (mainly CH₄) emissions and poor engine operating stability, especially under low load conditions. The current study investigated cylinder-to-cylinder variations in the DDF combustion in a four-cylinder turbocharged direct-injection diesel engine. All experiments were performed under steady-state engine conditions at 1800 rpm for a range of diesel injection timings and EGR rates.

Data showed that the diesel injection timing and the EGR played an important role in controlling both cyclic variation and cylinder variation in the cylinder outputs. High %EGR was needed to achieve stable DDF combustion in all cylinders. Several factors could render cylinder-to-cylinder variation in the combustion and the combustion stability in each cylinder including the mixture lambda, the EGR distribution, the thermal stratification in the intake, and the charge inhomogeneities. As the cylinder-to-cylinder variation in the combustion became small, the engine efficiency together with HC and CO emissions can be reduced.

Keywords: Diesel dual fuel, Natural gas, Direct Injection, EGR, Cylinder-to-cylinder variations

1. Introduction

Natural gas (NG) has been recognized as one of the promising alternative fuels for internal combustion engines. With the ability to achieve a high efficiency, a compression-ignition engine produces a higher efficiency than a spark-ignition engine. These reasons have drawn interest in investigating a premixed NG charge diesel-ignited engine. In such a dual fuel engine, NG can be supplied into the intake system or injected into the cylinder. A small amount of diesel fuel is directly injected into the cylinder to promote the autoignition. When converting a diesel engine to a dual fuel engine, the combustion chamber is normally remained unchanged and so is the compression ratio. In the current study, the term “diesel dual fuel (DDF)” engine is referred to as a premixed-natural-gas diesel-ignited engine [1].

DDF engines are capable of reducing both soot and NOx to levels considerably lower than those of conventional diesel engines. However, at light load, DDF engines are fed with a very lean air-fuel mixture which is hard to auto-ignite and slow to burn, leading to significant amounts of THC and CO emissions [2, 3]. As a premixed charge autoignition mechanism similar to the HCCI combustion, the DDF combustion is governed by both chemical kinetics and the mixing between air and fuels. The chemically kinetic controlled combustion implies that the combustion is affected by thermodynamic state histories that the air-fuel mixture traverses in the cycle. Also, the composition of the charge mixture is one of the main factors influencing the combustion process.

In HCCI engines, the combustion initiation is dependent of several parameters that are not easily controlled like the temperature and pressure histories in the cylinder. So, in a multi-cylinder engine, achieving the same ignition condition in all the cylinders is a challenge. Factors such as the gas exchange, the compression ratio, the cylinder cooling, the fuel supply, and the inlet air temperature can differ from cylinder to cylinder. These differences cause both combustion phasing and load variations between the cylinders, which in the end affect the engine performance and engine-out emissions [4-9].

2. Experimental Setup

2.1 Test Bed Setup

All experiments were performed at PTT Research and Technology Institute. The latest version of a four-cylinder, VN turbocharged Toyota 2KD-FTV common-rail diesel engine was used in the current research. The engine specifications are shown in Table.1.

Table. 1 Engine specifications

<table>
<thead>
<tr>
<th>Engine type</th>
<th>2KD-FTV</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>4 cylinders, inline arrangement</td>
</tr>
<tr>
<td>Number of valves</td>
<td>16 valves (DOHC)</td>
</tr>
<tr>
<td>Manifold</td>
<td>Cross-flow with an intercooler and a VN turbocharger</td>
</tr>
<tr>
<td>Fuel system</td>
<td>Common-rail, Direct Injection</td>
</tr>
<tr>
<td>Displacement volume</td>
<td>2494 cc</td>
</tr>
<tr>
<td>Bore × Stroke</td>
<td>92 mm × 93.8 mm</td>
</tr>
<tr>
<td>Connecting rod</td>
<td>158.5 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.4:1</td>
</tr>
<tr>
<td>Exhaust valve open</td>
<td>30° BBDC (+150° after firing TDC)</td>
</tr>
<tr>
<td>Exhaust valve closed</td>
<td>0° BTDC (+360° after firing TDC)</td>
</tr>
<tr>
<td>Inlet valve open</td>
<td>2° BTDC (+358° after firing TDC)</td>
</tr>
<tr>
<td>Inlet valve closed</td>
<td>31° ABDC (~149° after firing TDC)</td>
</tr>
<tr>
<td>Firing order</td>
<td>1-3-4-2</td>
</tr>
</tbody>
</table>

As an attempt to improve low-load engine operations, the current study investigated the cylinder-to-cylinder variation in the DDF combustion in a four-
cylinder turbocharged direct-injection diesel engine. The focus of this work was to investigate cylinder-to-cylinder variations in the cylinder outputs and address some important factors for a range of diesel injection timings and EGR rates under steady-state engine conditions.

Fig. 1 shows a schematic of the experimental setup. The engine was coupled to a DC dynamometer. The air at a room condition was drawn to the intake system. A natural gas injector for each cylinder was connected with a rubber hose and attached to the intake runner of the swirl port at which the total length from the NG injector tip to the inlet valves was approximately 345 mm. Diesel fuel was supplied through a common-rail direct injection system. Cylinder pressure data of 150 consecutive cycles from all cylinders were recorded for combustion data analysis. For each cycle, in-cylinder pressure data were recorded with the crank angle resolution of 0.2°. The coolant temperature was maintained at 85°C. The oil temperature was maintained at 90°C. Four wide-band lambda probes were installed at the spacer attached between the exhaust ports and the exhaust manifold to measure exhaust A/F of each cylinder. Exhaust gas emissions were analyzed by using the Horiba MEXA 7100 DEGR gas analyzer. Data of smoke emissions were not available to present in this paper. Although smoke is one of the major common emissions from conventional diesel engines, it was found in our previous study [1] that acceptable DDF combustion produced much less smoke than conventional diesel combustion.

Fig. 1  Schematic of the experiment setup.

Table. 2  Properties of natural gas in Thailand (PTT sampling data in 2011)

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane number</td>
<td>83.0</td>
</tr>
<tr>
<td>Higher heating value, MJ/kg</td>
<td>37.71</td>
</tr>
<tr>
<td>Lower heating value, MJ/kg</td>
<td>34.10</td>
</tr>
<tr>
<td>Stoichiometric A/F</td>
<td>11.68</td>
</tr>
<tr>
<td>MW, kg/kmole</td>
<td>22.39</td>
</tr>
<tr>
<td>Methane, % by mole</td>
<td>74.2</td>
</tr>
<tr>
<td>Ethane, % by mole</td>
<td>5.8</td>
</tr>
<tr>
<td>Propane, % by mole</td>
<td>2.3</td>
</tr>
<tr>
<td>n-Butane, % by mole</td>
<td>0.4</td>
</tr>
<tr>
<td>i-Butane, % by mole</td>
<td>0.5</td>
</tr>
<tr>
<td>Larger hydrocarbons (&gt; C6), % by mole</td>
<td>0.3</td>
</tr>
<tr>
<td>CO₂, % by mole</td>
<td>14.5</td>
</tr>
<tr>
<td>N₂, % by mole</td>
<td>2.0</td>
</tr>
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</table>

Table. 3  Properties of diesel fuel (B3)

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, kg/m³ (calc.)</td>
<td>0.83</td>
</tr>
<tr>
<td>Higher heating value, MJ/kg (calc.)</td>
<td>45.64</td>
</tr>
<tr>
<td>Lower heating value, MJ/kg</td>
<td>42.75</td>
</tr>
<tr>
<td>Stoichiometric A/F (est.)</td>
<td>14.41</td>
</tr>
<tr>
<td>MW, kg/kmole (est.)</td>
<td>170</td>
</tr>
<tr>
<td>C (est.)</td>
<td>12.30</td>
</tr>
<tr>
<td>H (est.)</td>
<td>21.92</td>
</tr>
<tr>
<td>O (est.)</td>
<td>0.04</td>
</tr>
</tbody>
</table>

2.3 EGR Calculations

The EGR line was connected from the exhaust runner of the first cylinder (upstream from the exhaust manifold) to the intake line (right after the throttle valve). The EGR flow rate was controlled by adjusting the opening position (EGR set point) of the solenoid valve. The EGR set point could be varied from 0% (fully closed) to 100% (fully opened). One popular technique to estimate the EGR amount is using CO₂ concentrations in the intake manifold and in the exhaust manifold by using Eq. (1) [10]:

\[
\%\text{EGR} = \frac{[\text{CO}_2]_{\text{int}} - [\text{CO}_2]_{\text{exh}}}{[\text{CO}_2]_{\text{exh}}} \times 100
\]

Notice that as the actual EGR flow rate approaches zero, %EGR calculated by Eq. (1) does not go to zero. To adjust this offset for low EGR cases (say less than 20%), we use the following equation [11]:

\[
\%\text{EGR} = \frac{[\text{CO}_2]_{\text{int}} - [\text{CO}_2]_{\text{ref}}}{[\text{CO}_2]_{\text{exh}}} \times 100
\]

where \([\text{CO}_2]_{\text{ref}}\) is the measured CO₂ concentration at the reference point, which is taken as the ambient CO₂ concentration in the laboratory room.

Note that Eq. (1) and Eq. (2) are approximations for the EGR molar ratio and should be used with caution. Theoretically, the EGR rate is defined by the ratio between the mass of the EGR and the total mass of the charge mixture entering the cylinder [12]. However, the mass flow rate of EGR was not measured. Using Eq. (1) and Eq. (2) offered a simplified approach to estimate our EGR amount. To allow a better understanding of the characteristics of our engine system, Fig. 2 presents a relation between the EGR set point and the calculated EGR rate (%EGR) at low-load diesel and DDF operations at 1800 rpm. The difference found between diesel and DDF operations was due to the open position of the throttle valve.

Fig. 2  EGR rates at different EGR set points at low-load diesel and DDF operations at 1800 rpm.
3. Test Matrices

The experiments were divided into 2 matrices with both having the engine speed of 1800 rpm. The first matrix was to observe the DDF combustion characteristics and cylinder output variations as the start of diesel injection (SOI) was varied. In this matrix, the amounts of diesel and natural gas supplied per cycle were kept constant as shown in Table 4. In the second matrix, we varied the EGR set point to observe combustion characteristics under DDF operations as shown in Table 5. In this matrix, the throttle was also adjusted to achieve the same air mass flow rate for each EGR set point. Note that for with zero \%EGR, we could not achieve stable combustion, thus no data presented.

Table 4 Engine parameters for the first matrix

<table>
<thead>
<tr>
<th></th>
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</tr>
</thead>
<tbody>
<tr>
<td>DDF</td>
<td>1800</td>
<td>57.1</td>
<td>0.63</td>
<td>2.36</td>
<td>45°, 40°, 35°, and 30°</td>
</tr>
</tbody>
</table>

Table 5 Engine parameters for the second matrix

<table>
<thead>
<tr>
<th>Mode</th>
<th>Speed [rpm]</th>
<th>Air [kg/h]</th>
<th>Diesel [kg/h]</th>
<th>NG [kg/h]</th>
<th>EGR set point [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>DDF</td>
<td>1800</td>
<td>57.1</td>
<td>0.63</td>
<td>2.36</td>
<td>20%, 40%, 60%, 80%, and 100%</td>
</tr>
</tbody>
</table>

For each data point, the injection signal for each individual fuel injector (for both NG and diesel) was controlled independently to minimize the deviation of injected mass between injectors. The injection timing (in terms of the end of injection) for NG was fixed at 270° bTDC for all DDF runs. For the second matrix, we used a single-pulse diesel signal with the SOI of 38° bTDC.

4. Results and Discussion

This section presents results and discussion of engine performance parameters, pressure data analysis, and engine-out emissions.

4.1 Effects of the Start of Diesel Injection

The DDF operation was investigated using a single-pulse injection signal for all of the test conditions in this work. As the start of injection (SOI) timing was varied from 45° to 30° before TDC (bTDC), the brake torque was increased. Fig. 3 shows torque data and the corresponding engine efficiency for different injection timings.

As the SOI was advanced, the engine torque (also, the efficiency) was reduced. This was due to changes in the combustion in each cylinder. Data of the net IMEP for each cylinder and its cylinder-to-cylinder variation (termed by “IMEPn variation”) are shown in Fig. 4. The combustion stability is presented in terms of the cycle-to-cycle variation of the net IMEP recorded over 150 consecutive engine cycles as shown in Fig. 5.

Results showed that the combustion stability in cylinder 1 was least sensitive to the injection timing change compared to other cylinders. This implied that the charge in cylinder 1 could be ignited easier. On the other hand, the combustion stability in cylinder 4 was most sensitive to changes in the injection timing. This also revealed that the charge in cylinder 4 was most difficult to reach autoignition.

Data of the combustion start were recorded and referred to as CA10 as shown in Fig. 6. As the SOI became advanced from 30° to 45° bTDC, the combustion timing was retarded. Regarding data of CA10 (Fig. 6) and IMEPn (Fig. 4) together with its COV (Fig. 5), it appeared that as the onset of combustion shifted later, the combustion stability got worse resulting in a lower cylinder output. At the latest SOI timing, the combustion timings in cylinders 2, 3 and 4 shifted even later from TDC. This corresponded to a significant increase in the COV of IMEPn.
Cylinder-to-Cylinder Variations in Diesel Dual Fuel Combustion under Low-load Conditions

Fig. 6 Start of combustion (CA10) for different SOI timings.

Fig. 7 and 8 compare histories of the cylinder pressure and heat release rates for each cylinder for the injection timings at 45° and 30° bTDC. Notice that the deviations in cylinder pressure and heat release rates between cylinders were more discernible at the SOI of 45° bTDC. For this SOI condition, one could easily observe that cylinder 4 demonstrated the weakest combustion energy release, with cylinder 1 showing the strongest among the four cylinders.

Fig. 7 Pressure and apparent heat release rates with SOI = 45° bTDC.

Fig. 8 Pressure and apparent heat release rates with SOI = 30° bTDC.

Based on the discussion in our previous study [13], we found that the cylinder-to-cylinder variation in the combustion was most likely due to the mixture lambda between cylinders. However, we found in the current study that the mixture lambda between cylinders was nearly the same with cylinder 4 having a slightly leaner mixture as shown in Fig. 9. So, in the current study, the mixture lambda would be less pronounced. The dominant factors responsible for the cylinder variations can be the thermodynamic state histories of the local mixture that traverses in the cycle and charge inhomogeneities in each engine cylinder. Note also that with the mixture lambda between cylinders being nearly the same, the charge inhomogeneities, the local charge composition, and its thermodynamic state histories in each cylinder do not need to be identical.

Fig. 9 Measured exhaust lambda (clustered bars) and its variation between cylinders (dotted line) for different injection timings.

Fig. 10 Intake temperature at the circular port of each cylinder for different injection timings.

Fig. 11 Intake temperature at the swirl port of each cylinder for different injection timings.
As one possible approach to assess the initial thermodynamic states of the mixture, one can observe data of intake temperatures at the circular port and the swirl port of each cylinder as shown in Fig. 10 and 11. Results showed that the temperature of each cylinder was different. The deviation of port temperatures might be due to the distribution of EGR flow at each intake port being different. As the temperature difference between the two ports became greater, it would promote more thermal stratification in the cylinder charge. The HCCI study by Iverson et al. [14] found that thermal stratification in the intake had a significant impact on the onset of combustion. As such, we proposed that changes in the temperature stratification in the intake mixture can lead to the combustion variations between cylinders in a DDF engine.

Fig. 12 shows data of engine-out emissions. Regarding the emission standpoint and the brake engine efficiency (Fig. 3), there appeared an operational window for achieving a low-emission DDF operating condition. We selected the timing of 35º bTDC for the best operation for this test condition. This injection timing operation resulted in only 18 ppm of NOx, total hydrocarbons (HC, in ppmC) of 7000 ppm, and CO of 0.17%. We prioritized hydrocarbons (especially CH4) over CO emissions as it would be easier to treat by using an oxidation catalyst compared to CH4 reduction.

4.2 Effects of EGR

For this data set, we started our preliminary investigation in conventional diesel operation without EGR to reduce the complicity of the operating parameters. Fig. 13 shows diesel mass flow and exhaust lambda of each cylinder with the EGR valve being fully closed. One can see that the diesel supplied to each cylinder was approximately the same, but the measured lambda was varied from cylinder to cylinder. These data suggested that our engine had some variation in the air trapped in each cylinder after IVC.

From the observation above, we now turn our focus to DDF operations. Fig. 14 shows data of brake torque and intake manifold pressure for different %EGR set points. As the EGR rate was reduce, the intake manifold pressure dropped. This caused the pumping work to reduce which then penalized the engine torque. Data of the corresponding efficiency is shown in Fig. 15.

Data of the net IMEP and its cylinder-to-cylinder variation are shown in Fig. 16. It appeared a similar trend as found in the previous test matrix that cylinder 1 produced greatest IMEP and cylinder 4 produced the smallest. Furthermore, cylinder 4 had the poorest combustion stability (represented by the cyclic variation of IMEP in Fig. 17). Note that with the 20% EGR set point, we could achieve DDF combustion in only some engine cycles. This resulted in very high COV (also, a drop in the net IMEP) in all cylinders.

Fig. 12 Hydrocarbons, CO and NOx engine-out emissions at different SOI timings.

Fig. 13 Lambda and diesel mass flow of each cylinder when the EGR valve is fully closed (zero %EGR) under diesel operation.

Fig. 14 Brake torque and intake pressure for different EGR set points.

Fig. 15 Brake efficiency for different EGR set points.

Fig. 16 Net IMEP for each cylinder (clustered bars) and its variation between cylinders (dotted line) for different %EGR set points.
Fig. 17  Coefficient of variation (COV) in net IMEP for different %EGR set points.

Fig. 18  Start of combustion (CA10) for different %EGR set points.

Fig. 18 shows the start of combustion (in terms of CA10) for different %EGR set points. As the EGR was reduced, the combustion timing was retarded. Fig. 19 and 20 emphasize these changes found with the cylinder pressure and heat release rates for the two extreme cases: 20% and 100% EGR set points. By comparing with data in Fig. 2, the 20% EGR set point corresponded to a nearly zero %EGR rate and the 100% EGR set point (i.e. fully open) had approximately 50% EGR rate.

Fig. 19  Pressure and apparent heat release rates for the EGR set point = 20%.

Fig. 20  Pressure and apparent heat release rates for the EGR set point = 100%.

Data of the measured exhaust lambda at each cylinder exit are shown with its cylinder variation in Fig. 21. It was found that the overall lambda was slightly reduced with greater %EGR. Cylinder 4 appeared to have leaner mixture compared to other cylinders for all EGR conditions, especially at lower %EGR. The leaner mixture is more difficult to reach autoignition and produces weaker energy release during the combustion. This is in agreement with the smallest IMEP and the most retarded combustion timing found in cylinder 4.

Fig. 21  Measured exhaust lambda (clustered bars) and its variation between cylinders (dotted line) for different %EGR set points.

Data of intake temperatures at the circular port and the swirl port of each cylinder are shown in Fig. 22 and 23. As one would expect, the intake temperature was increased with %EGR. With 20% EGR set point, the deviation of these port temperatures was minimal. As the EGR rate was increased, the discrepancies in port temperatures became larger. This suggested that, not only the variation in the air trapped in each cylinder, but there would be also some variation in the EGR flow into each port and each cylinder. At 100% EGR set point, cylinder 4 had the greatest temperature difference between the two ports which was approximately 16ºC. Therefore, the thermal stratification in the intake would be most pronounced in this cylinder.
It should be noted that the cylinder-to-cylinder variation in the IMEP of diesel dual fuel operations can involve several factors such as the air trapped, the variation of EGR ratio in each cylinder, the thermal stratification in the intake, the deviation in the diesel mass between injectors, the variation in NG trapped in each cylinder, the variation in the residual gas fraction, the turbulent mixing effect on the mixture inhomogeneities, and so on. Further investigation will be required to quantify each of these factors.

Fig. 24 shows engine-out emissions for DDF operations. It was found that hydrocarbon and CO emissions were reduced with greater %EGR. NOx were very low and slightly increased with %EGR as each cylinder had higher combustion temperatures. One can see that as the cylinder variation and the cyclic variation in the combustion was reduced, reductions in hydrocarbon and CO emissions could be achieved. As such, regulating high EGR was required to obtain good DDF combustion characteristics.

5. Conclusions
Data have been presented for DDF operations in a four-cylinder turbocharged, direct-injection engine. Experiments were conducted under low-load, steady-state operations at 1800 rpm for a range of diesel injection timings and EGR rates. The important findings are summarized as follows:

- The diesel injection timing and the EGR played an important role in controlling both cyclic variation and cylinder variation in the cylinder outputs.
- There could be several factors rendering cylinder-to-cylinder variation in the combustion and the combustion stability in each cylinder including the mixture lambda, the EGR distribution, the thermal stratification in the intake, and the charge inhomogeneities.
- EGR was the important factor responsible for the thermal stratification in the intake.
- High %EGR was required to achieve stable DDF combustion in all cylinders.
- If the cylinder-to-cylinder variation in the combustion is kept minimal, the engine efficiency together with HC and CO emissions can be reduced.

6. Acknowledgement
The authors are sincerely thankful to PTT Public Company Limited for their financial supports throughout the course of this work.

7. References

8. Definitions, Acronyms, and Abbreviations
A/F  Air-fuel ratio
bTDC  before firing TDC
CO  Carbon monoxide
CO2  Carbon dioxide
DDF  Diesel Dual Fuel
EGR  Exhaust gas recirculation
EOI  End of injection
HC  Hydrocarbons
HCCI  Homogeneous Charge Compression Ignition
IMEP  Indicated mean effective pressure
LHV  Low heating value
NG  Natural gas
NOx  Oxides of nitrogen
SOC  Start of combustion
SOI  Start of injection
TDC  Top dead center
THC  Total hydrocarbons
VN  Variable nozzle (for the turbine)