Effect of Diesel Injection Parameters on Diesel Dual Fuel Engine Operations with Charge Preheating under Part Load Conditions

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Abstract

Diesel Dual Fuel (DDF) is an alternative operating mode for conventional diesel engines. DDF engines provide a potential alternative solution for PM and NO_X emissions reduction from typical diesel engine operations. However, DDF engine operations suffer from high HC emissions and poor operation characteristics under part load conditions.

The current study investigated the effects of diesel injection parameters and charge temperature on DDF combustion and emission characteristics in a common-rail direct injection, single-cylinder research engine. The natural gas was supplied at 9.5 mg/cycle. The diesel fuel injection timings were varied under injection pressures of 200 and 500 bar. The diesel injection duration was tuned to obtain the injected mass of 3.25 mg/cycle. This corresponded to 70% energy ratio of natural gas to the total fuel energy supplied to the engine for all engine conditions. The intake port temperature was controlled at 45ºC, 60ºC, and 80ºC. All experiments were tested under steady state engine operations at 1600 rpm.

Our data indicated that using high intake charge temperature has potential to reduce HC emissions. However, the higher charge temperature resulted in increase of NO_x emissions and the rate of pressure rise. To achieve good DDF operation, which produce lowest HC emissions possible; the injection parameters should be tuned. Increasing the injection pressure promoted the mixing shifted the optimum injection timing later towards TDC. Use of the injection timing at later crank angles would help lessen the amount of fuel impinged on to the cylinder liner. The findings from this research provide a guideline for optimizations of injection parameters and intake charge temperature for improvement of DDF engine operations.

***Keywords**:* DDF, PCCI, Diesel Dual Fuel, Injection Pressure, Injection Parameters, Charge Temperature

1. Introduction

In Thailand, natural gas has played an important role in the substitution of oil-based fuels in the transportation section due to its lower price and its availability in natural resources. An application for natural gas in spark ignition engines has been successfully utilized due to its similar fuel properties to those of gasoline fuel. However, for the compression ignition engines, natural gas cannot use as a single fuel because of fuel's ignitability. So, the diesel-ignited natural gas concept has been brought into our attention instead.

In diesel-ignited natural gas engines or diesel dual fuel (DDF) engines, the mixture of gaseous fuel and the intake air are compressed at a diesel-like compression ratio, and then a small amount of diesel fuel is injected to ignite the mixture. The diesel fuel ignites in the same way as in compression ignition engines, and the gaseous fuel is consumed by flame propagation in a similar manner to spark ignition engines. This ignition system has an advantage of large energy source and stable ignition, thus lower cycle-to-cycle variation, compared to the spark ignition system [1].

The DDF engines provide a potential alternative solution for PM and NO_x emissions reduction from conventional diesel engines operations. However, DDF engines operations suffer from high HC (total hydrocarbons) emissions and poor engine performance at part load conditions [2, 3].

The previous investigation on our single-cylinder research engine [4] has demonstrated DDF operation in a range of injection timings and injection pressures by using a single-pulse injection signal at earlier injection timings compared to conventional diesel engines. HC emissions were extremely high in all the DDF experiments. As mentioned before, this poor exhaust emission characteristic can be improved by increasing the charge (air + natural gas) temperature. The higher charge temperatures will widen mixtures flammability limits and sustain flame propagation within relatively leaner mixtures. Thus, the flame can propagate easier through the gaseous fuel air mixture. This can possibly reduce HC emissions in the exhaust [5].

Use of the early injection technique has potential to substantially reduce NO_x emissions and the in-cylinder pressure rise rate. However, since the injection occurs under a lower ambient density and lower temperature condition, the increase of HC emissions and the wall wetting could be a severe problem. One solution for these issues is to increase the injection pressure. Research studies on premixed charge diesel engines [6, 7] suggested that with higher injection pressure the diesel fuel could be injected at later timings than the lower injection pressure operations for the same degree of homogeneity and leanness. So, the increase of injection pressure can reduce the mixing time required for mixture formation due to the turbulent mixing rate enhancement.

The focus of this study was to investigate the combustion and exhaust emissions characteristics of DDF engine operations. The investigation was done by varying injection parameters coupled with charge preheating. This research work has been conducted using a single cylinder research engine. Results of this work will offer more insights into combustion phenomena and facilitate engine calibration work on other commercial engines.

2. Experimental setup

All experiments were performed at PTT Research and Technology Institute. A naturally aspirated, single cylinder, four-stroke, common rail direct-injection diesel engine was used in the current research. The compression ratio was reduced from our previous study [5] as we replaced a different cylinder-head gasket. The engine specifications are shown in Table 1.

Table 1. Engine specifications [4]

Engine Type	2-Valve DI Diesel Ricardo Hydra
Fuel system	Common-rail
Aspiration	Naturally aspirated
Compression ratio	17.91:1
Displacement	449.77 cc
Bore	80.26 mm
Stroke	88.90 mm
Connecting rod	158.0 mm
Valve timings: Intake valve open Intake valve close Exhaust valve open Exhaust valve close	 8° BTDC 42° ABDC 60° BBDC 12° ATDC

Fig. 1 shows a schematic of the experimental setup. The engine was coupled to a DC dynamometer. Natural gas was introduced into the intake port through a gas injector. The original diesel injection system was replaced by a common rail injection system. Cylinder pressure data of 0.2° crank angles (CA) were recorded for 100 consecutive engine cycles. Coolant and oil temperatures were set at 85°C. Gaseous emissions were analyzed by using the AVL AMA i60 emission analyzer. Although data of particulate emissions were not available in this work, previous studies showed very low PM emitted from DDF engines [1, 4].

Properties of natural gas and diesel fuel used in the current study are shown in the appendix. It should be noted that a variation of natural gas composition may be largely different from place to place. The ratio of natural gas replacement was therefore defined as an energy flow of natural gas to the total fuel energy flow (diesel + NG):

$$\%NG = \frac{\dot{m}_{NG} \cdot LHV_{NG}}{\dot{m}_D \cdot LHV_D + \dot{m}_{NG} \cdot LHV_{NG}} \times 100 \quad (1)$$

To investigate the effects of diesel injection parameters with charge preheating on DDF operation characteristics, DDF experiments have been done and compared at the same amount of total fuel energy. The injection of diesel fuel was controlled by a common rail system. The pressure in the common rail was set at 200 and 500 bar. The intake port temperature was varied from 45°C to 80°C. The injection timing of the pilot diesel fuel was varied from the latest timing at which the pressure rise rate was not exceed 8 bar/°CA, to the timing where misfire operation occurred. A single-pulse injection was employed throughout the experiments and the injection duration was tuned to obtain constant total fuel energy at the 70% energy ratio (diesel 3.25 mg/cycle, NG

0.456 mg/cycle). This corresponded to the total fuel energy supply of 463.75 J/cycle.

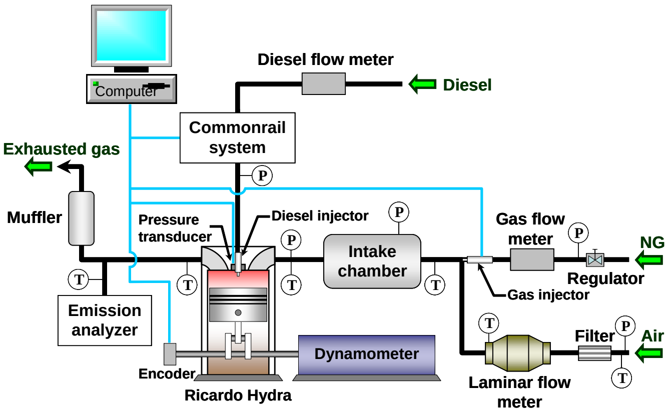

Fig. 2 Schematic of the experimental setup.

3. Results and Discussion

3.1 Effect of intake charge temperature

Fig. 2 illustrates the in-cylinder pressure and rate of heat release (ROHR) data for the engine operations at -46 °ATDC injection timing with the injection pressure of 200 bar and charge temperatures of 45, 60 and 80°C.

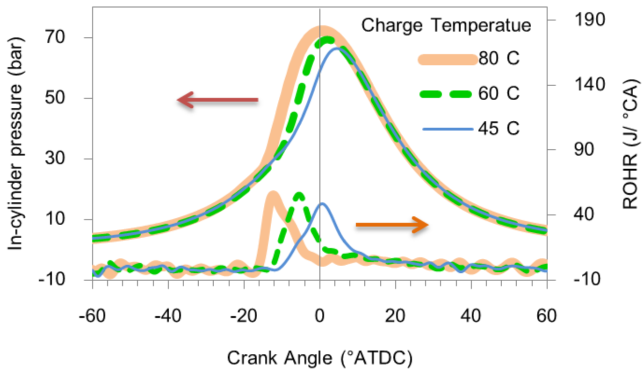

Fig. 2 In-cylinder pressure and rate of heat release data for different charge temperature at -46 °ATDC injection timing with 200 bar injection pressure.

The DDF combustion significantly changed with charge temperature. By observation of ROHR histories, the peak of heat release was higher as charge temperature was increased. This indicated that higher charge temperature accelerated the chemical reactivity in the air-fuel mixture. Fig. 3 shows combustion timings (in terms of CA50), IMEP, COV of IMEP, and ROHR for difference charge temperatures at the 200-bar injection pressure. For the same injection timing, changes in charge temperature significantly changed the combustion timing. Increasing the charge temperature from 45 to 80°C caused the combustion timing to advance. This was a result of higher inlet temperature accelerated the overall kinetics [8]. In addition, there was an increase in the ROHR due to the increased reaction rate. However, raising the intake temperature caused the pressure rise rate to increase. A greater pressure rise rate could lead to a combustion noise concern.

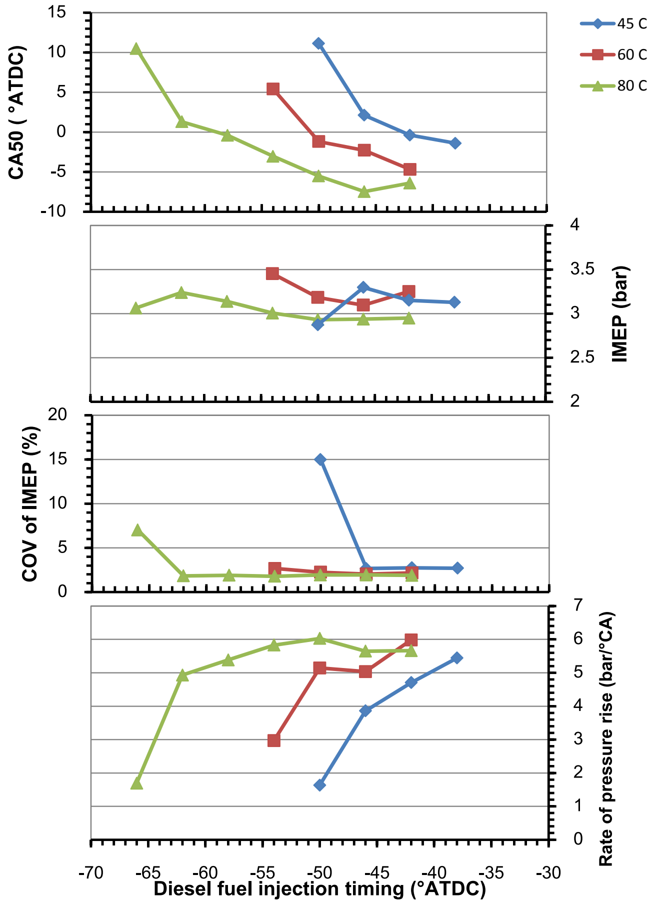

Fig. 3 Combustion timing (CA50), IMEP, COV of IMEP, and rate of pressure rise (ROHR) for

different charge temperature at -46 °ATDC injection timing with 200 bar injection pressure.

In Fig. 3, it was found that IMEP was related with the combustion timing. From our data, IMEP was increased when the combustion timing became retarded. However, for the injection timing at -50°ATDC with intake port temperature of 45°C and the injection timing of -66°ATDC with the intake port temperature of 80°C, IMEP was decreased due to misfired operation in some engine cycles.

Fig 4 shows the comparison of exhaust emissions characteristics for different charge temperature at the 200-bar injection pressure. HC emissions were reduced significantly as the charge temperature was increased from 45 to 80°C. Thus, the hotter charge temperature was favorable for the flame propagation through the premixed gaseous fuel/air mixture. Therefore, raising the charge temperature could improve poor emissions characteristics (mainly HC). Data also demonstrated HC emissions being slight greater when the injection timings were retarded for all charge temperature conditions. With late injection timings, diesel fuel did not have enough time to mix with the charge mixture. As the diesel spray and the un-burned charge mixture did not well mix enough, there would be portions with poor mixing where the incomplete combustion occurred [4, 9]. It should be noted that for the injection timing at -50°ATDC with intake port temperature of 45°C and the injection timing of -66°ATDC with the intake port temperature of 80°C, HC emissions were slightly increased as a result of misfired operation in some engine cycles as previously mentioned.

Data in Fig 4 showed that changes in the inlet charge temperature did not significantly change the amount of CO emission. However, CO emission at the earliest injection timings of all charge temperature conditions was slightly increased significantly. With early injection timings, diesel fuel had more time to mix with charge mixture. This resulted in an increase of excessively lean fractions of the mixture. The excessively lean fractions of the mixture increased the possibility of incomplete combustion [10].

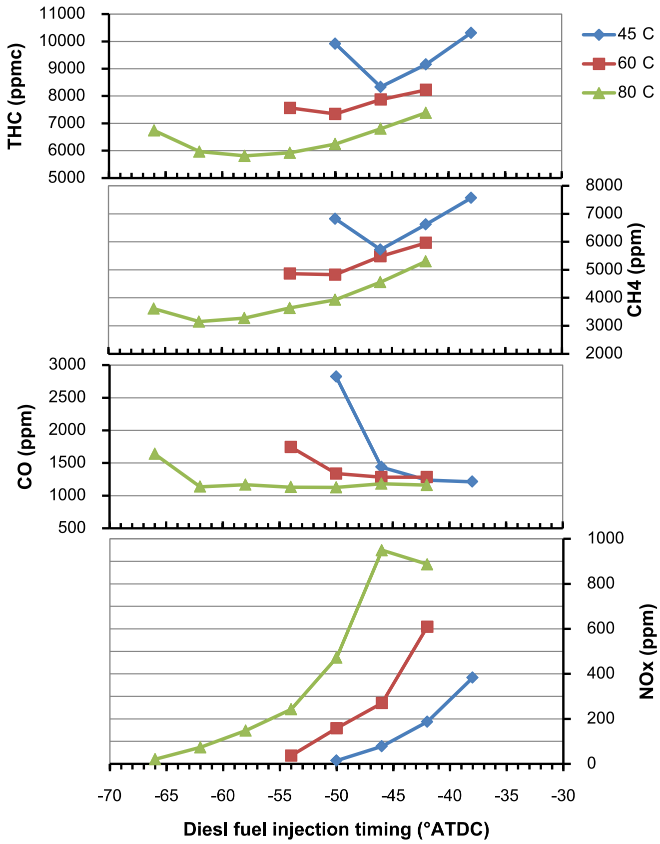

Fig 4 THC, methane, CO, and NO_x emissions for different charge temperatures with 200 injection pressure.

As shown in Fig. 4, NO_x emissions were greatly influenced by charge temperature. By increasing the intake temperature from 45 to 80°C, NO_x emissions were rapidly increased.

The penalty of improving the combustion process by increasing the charge temperature was the increase of NO_x production. With higher charge temperature, we expect to have the increased compression temperature and, hence, higher flame temperature. Therefore, an increase of the maximum cycle temperature would promote NO_x formation [5, 11].

To reduce HC emissions by increasing the charge temperature, the DDF operations would suffer from noisier operation and produce more NO_x emissions.

3.2 Effect of injection parameters

Fig. 5 illustrates the in-cylinder pressure and rate of heat release (ROHR) data for the engine operations at -46 °ATDC injection timing with the injection pressures of 200 and 500 bar. The intake charge temperature was kept constant at 60 °C.

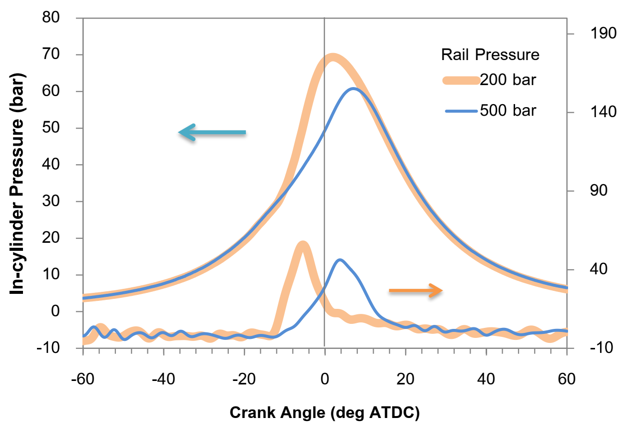

Fig. 5 In-cylinder pressure and rate of heat release data for different injection pressures, the injection timing of -46 °ATDC and the charge temperature of 60 °C.

By comparison of ROHR histories, the peak of heat release was low and the combustion duration became longer as the injection pressure was increased. We hypothesized that this in turn could indicate less portion of the mixture around stoichiometry, which has high flame speed [7]. Increasing injection pressure would increase the turbulent mixing rate and, hence, increasing in degree of leanness could be expected.

Data of the combustion timings are shown in terms of CA50 in Fig. 6. At the same injection timings, use of the 500-bar injection pressure caused the combustion timing to retard, compared to those of the 200-bar injection pressure. This implied that the 500-bar injection pressure condition could produce higher degrees of leanness mixture, which had a lower flame speed.

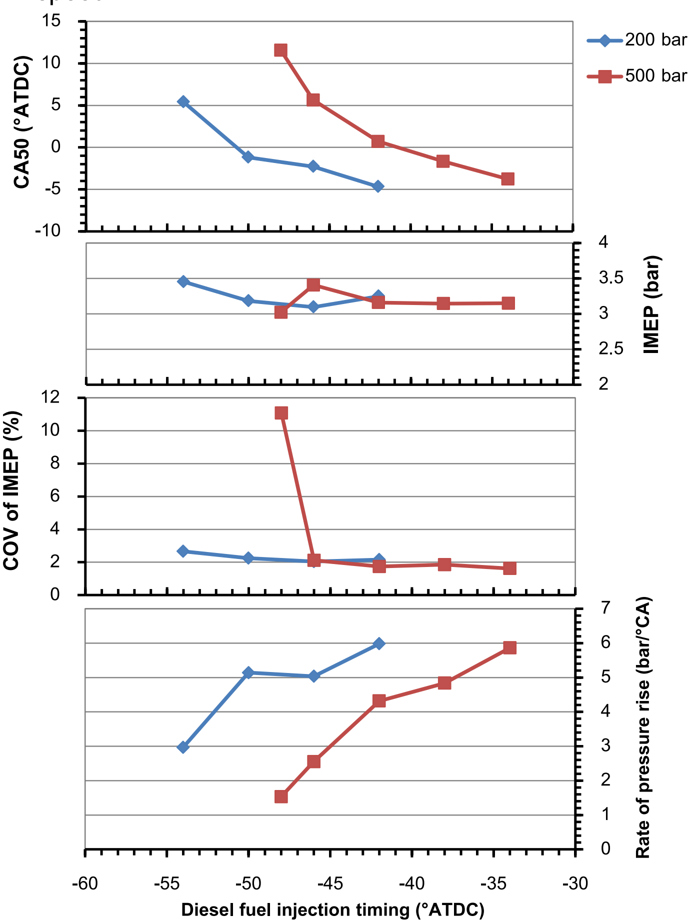

Fig. 6 Combustion timing (CA50), IMEP, COV of IMEP, and rate of pressure rise (ROHR) for different injection pressure with 60 °C charge temperature.

Increasing the injection pressure had no significant change in IMEP, as shown in Fig. 6. However, IMEP was slight greater as the combustion timing was retarded after TDC. It should be noted that for the injection timing at -50°ATDC with intake port temperature of 45°C and the injection timing of -66°ATDC with the intake port temperature of 80°C, the decreased IMEP was due to the misfire operation and, hence, the COV of IMEP of this operating point was about 11%.

Fig. 6 shows, for the same injection timing, the pressure rise rates with the 500-bar injection pressure being lower compared to those of the 200-bar injection pressure conditions. By observation of the COV of IMEP and the pressure rise rate diagram, the injection timing window of the injection pressure of 200 bar was shifted to the earlier injection timings. This observation was similar to our previous experiments [4].

Fig. 7 shows the comparison of exhaust emissions characteristics for different injection pressures. Increasing the injection pressure from 200 to 500 bar, the total hydrocarbon (THC, in ppmC) and CH_4 emissions at -46 and -42 °ATDC injection timings were reduced by about 2000 ppm. Integrating this observation to those found by Minato et al. [7], the potential to reduce HC emissions by increasing the injection pressure can be described as follows. The increase of injection pressure increases the turbulent mixing rate and reduces the injection duration. These can produce the mixture which has more uniform distribution in the equivalence ratio and shorter mixing time. A more uniform mixture has fewer portions of very lean regions where the temperatures are too low for complete oxidations. Thus, HC emissions can be reduced. With an overall-lean and more uniform mixture, we would also expect very low NO_x formation.

For CO emission, increasing the injection pressure had no significant change in CO emission. However, the CO emission was increased with earlier injection timings as mentioned previously.

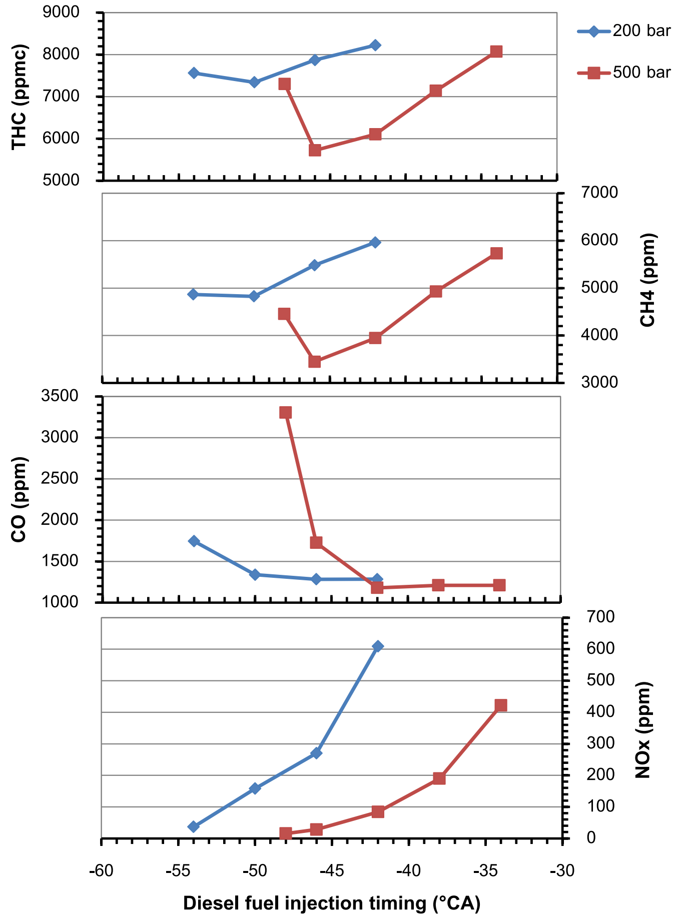

Fig. 7 THC, methane, CO, and NO_x emissions for different charge temperatures for different injection pressure with 60 °C charge temperature.

It was found that NO_x emissions were greatly influenced by injection parameters as shown in Fig. 7. By increasing rail pressure from 200 to 500 bar, NO_x emission were rapidly decreased. As one can see from Fig. 7 at -42 °ATDC injection timing the NO_x emissions were reduced from 609 ppm to 84 ppm when the injection pressure was increased to 500 bar. The decreased NO_x emission in high injection pressure conditions implied better mixing quality in this condition.

From the above discussion, the use of high charge temperature has the potential to reduce HC emissions. However, it led to greater NO_x emissions and more rapid rate of pressure rise. To reduce HC emissions, the injection pressure should be tuned. By increasing the injection pressure from 200 to 500 bar, the start of injection could set at later injection timings. With later injection timings, the injected fuel jet can penetrate through higher ambient density and higher temperature environment in the cylinder condition. Therefore, rapid vaporization and momentum exchange between the injected fuel and the premixed (natural gas + air) charge result in rapid fuel mixing and wall wetting reduction. These produced more uniform distribution of the equivalence ratio and, potentially, lower HC and NO_x emissions.

It should be noted that, the geometry of the combustion chamber was asymmetric; thus the diesel fuel always hits the cylinder head and the liner. In order to have assessment to the diesel spray distribution, we performed a multi-dimensional CFD simulation by using AVL-FIRE. The combustion chamber was asymmetric and the injector position was inclined from the vertical axis. Fig. 8 shows CFD results under the DDF engine condition at 1600 rpm, diesel = 3.25 mg/cycle with the start of injection at 40°BTDC and the injection pressure of 200 bar. Although there was no detailed information of the spray parameters (only injection timing, duration, and injected mass were available), we could roughly see that the diesel jet always hit the cylinder head and the liner.

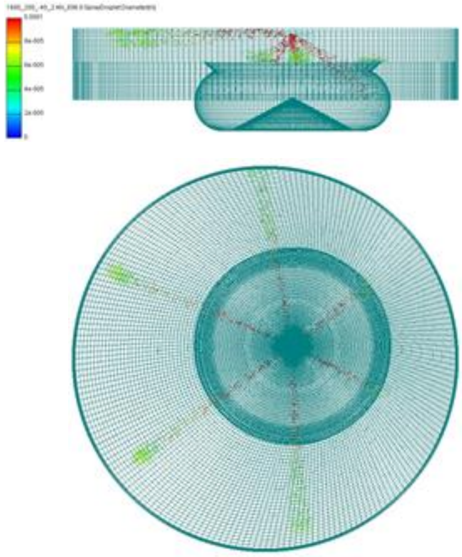

Fig. 8 Combustion chamber geometry and the injected spray direction of the tested engine. Engine condition: 1600 rpm, intake temperature = 60°C, SOI = 40°BTDC, diesel injection pressure = 200 bar, diesel = 3.25 mg/cycle, NG = 9.5 mg/cycle, overall equivalence ratio = 0.4.

4. Conclusion

Experiments have been conducted with the objective of improving the DDF operation under part load conditions by varying injection parameters coupled with charge preheating. Natural gas, as a primary fuel, was supplied to the engine at 70% energy ratio. Experiments were run by controlling the constant mass flow rate of each of natural gas and diesel fuel. The conclusions are summarized as follows:

- Increasing the intake charge temperature caused the combustion timing to occur earlier and increased the maximum rate of pressure rise.

- The use of high inlet charge temperature has potential to reduce HC emissions but increase in NO_x emissions.
- Increasing the injection pressure has potential to reduce the rate of pressure rise and NO_x emissions.
- The use of injection pressure could produce more uniform distribution of the equivalence ratio in the mixture, which has potential to reduce HC emissions.
- To reduce HC emissions by using high intake charge temperature, the injection parameters should be tuned to reducing NO_x emissions and achieving acceptable combustion noise.

6. Acknowledgement

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8.Appendix

Table A1. Properties of natural gas in Thailand (PTT sampling data in 2010)

Lower heating value, MJ/kg	34.14
Stoichiometric A/F	11.71
Specific gravity	0.77
MW, kg/kmole	22.20
Methane, % by mole	74.89
Ethane, % by mole	5.57
Propane, % by mole	2.10
n-Butane, % by mole	0.39
i-Butane, % by mole	0.48
n-Pentane, % by mole	0.06
i-Pentane, % by mole	0.12
Larger hydrocarbons (> C6), % by mole	0.12
CO_2, % by mole	14.30
N_2, % by mole	1.97

Table A2. Properties of diesel fuel (B2) used in the current work

Lower heating value, MJ/kg	42.8
Stoichiometric A/F	14.5
Specific gravity	0.83
MW, kg/kmole	170
C (calculated)	12.30
H (calculated)	22.13