Effects of Exhaust Valve Timing on Diesel Dual Fuel Engine Operations under Part Load Conditions

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Abstract

A natural gas engine with direct diesel injection has shown potential as an alternative combustion mode. Many research studies have reported poor diesel dual fuel (DDF) operation characteristics under part load conditions. Under these conditions, the engine can suffer from high HC emissions, mostly methane.

In the current study, a four-cylinder turbocharged diesel engine has been converted into a dual-fuel engine operating under premixed natural gas and common-rail direct diesel injections. Experiments were performed to investigate use of different exhaust valve timings for improvement of low-load DDF operation. For all engine conditions, the flow rate of natural gas was maintained, on average, approximately 70% by energy of the total fuel supplied.

Results showed that different exhaust valve timings changed the fractions of EGR and the charge temperature. Advancing the exhaust valve timing was most beneficial for low-load DDF operations. Under these operations, HC and CO were significantly reduced. As the engine load increased, the exhaust timing advance might lead to excessive combustion rates and high NOx emissions. Under these operations, the suitable exhaust valve timing should be shifted to towards the original setting for conventional diesel operations. The findings from the current work offered a possibility for expanding the DDF operating range.

***Keywords**:* Diesel Dual Fuel, Premixed charge compression ignition, Natural gas, Exhaust valve timing.

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 compression, a compression-ignition engine can produce a higher efficiency than a spark-ignition engine. These reasons have drawn interest in investigating a diesel engine using high NG replacement ratios. In such a dual fuel engine, NG is served as a main fuel and diesel fuel is served as a pilot fuel. Accordingly, the term "diesel dual fuel (DDF)" engine is referred to as a dual fuel natural gas-diesel engine in the current study.

In dual fuel engines, the main fuel is introduced either homogeneously by a mixer or port injectors in the intake system [1–7] or stratifically by a direct injection into the combustion chamber [9]. The pilot diesel fuel injection serving as an ignition source is usually injected during the compression stroke [1–9]. When converting a diesel engine to a dual fuel engine, the combustion chamber is normally remained unchanged and the compression ratio is therefore maintained on relatively high levels. A high auto-ignition resistance of the main fuel is desirable to avoid problems with knock.

DDF engines are capable of reducing both particulate matters and NO_x to levels considerably lower than those of conventional diesel engines. This is combined with a comparable engine efficiency and, at certain operating points, even a possible higher efficiency than that of diesel engines [1,2,6]. However, DDF engines suffer from poor low load characteristics. This is mainly because of the lean flammability limit of the homogeneous charge which leads to incomplete combustion or, in the worst case, misfire.

Similar to stratified charge HCCI, but probably less sensitive, the onset of DDF combustion is influenced by thermodynamic state histories of the charge mixture. Several research studies demonstrated effective techniques to control HCCI-like combustion is by means of complex variable valve control systems [10–13]. Variable valve actuation systems can be used for regulating the amount of residuals trapped in the cylinder (i.e. internal EGR), which changes the volumetric efficiency and affects the charge temperature. Adjusting the amount of exhaust residuals will heat up the fresh charge and help promote the auto-ignition process and improve the engine performance, operating stability and emissions.

Our previous investigations on DDF engines have demonstrated DDF operation in a range of engine speeds, loads, and diesel injection techniques [6,7]. In this current work, the objectives were to investigate changes in the exhaust valve timings on combustion characteristics and engine-out emissions.

2. Methods

All experiments were performed at PTT Research and Technology Institute. A turbocharged Toyota 2KD-FTV diesel engine was used in the current research. The engine specifications are shown in Table 1.

Table. 1 Engine specifications (standard valve timings)

# Cylinder	4 cylinders, inline
Displaced volume	2,494 cc
Stroke	93.8 mm
Bore	92 mm
Connecting rod	158.5 mm
Compression ratio	18.5:1
Number of valves	16 valves (DOHC)
Exhaust valve open	30° BBDC
Exhaust valve close	0° BTDC
Inlet valve open	2° BTDC
Inlet valve close	31° ABDC

Figure 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. In a natural gas supply unit was similar to that used in our previous work [6]. In the current work, a natural gas injector for each cylinder was connected with a rubber hose and attached to the intake runner, which have the total length of approximately 120 cm away from injector tip to the inlet valves. Cylinder pressure data of 150 consecutive cycles from cylinder 1 was recorded for combustion data analysis. For each cycle, the crank angle resolution of pressure data of 0.2° was recorded. The coolant temperature was maintained at 85°C. The oil temperature was maintained at 90°C. Gaseous emissions were analyzed by using the Horiba MEXA 7100 DEGR gas analyzer. The AVL 439 opacimeter was used to measure particulate matters.

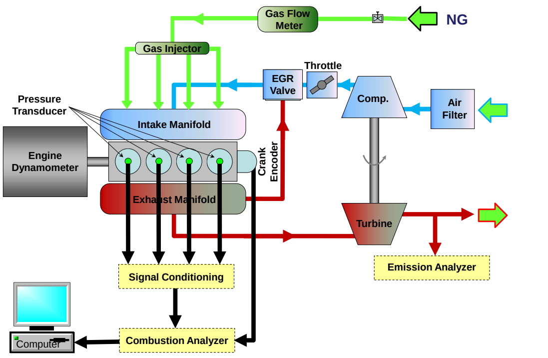

Fig. 1. Schematic of the experimental setup

Properties of natural gas 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. In the present work, %NG was set approximately at 70% for all engine conditions.

$$\%NG = \frac{\dot{m}_{NG} \cdot LHV_{NG}}{\dot{m}_{D} \cdot LHV_{D} + \dot{m}_{NG} \cdot LHV_{NG}} \times 100 \qquad (1)$$

For each engine operating point, the amount of EGR was calculated from CO_2 concentration data measured in the intake manifold and in the exhaust system. The volumetric definition of %EGR was defined by (adapted from [14]):

$$\%EGR = \frac{[CO_2]_{int} - [CO_2]_{ref}}{[CO_2]_{exh} - [CO_2]_{ref}} \times 100 \qquad (2)$$

where

$[CO_2]_{int}$ = measured concentration of CO_2 in the intake manifold.

$[CO_2]_{exh}$ = measured concentration of engine-out CO_2 at the outlet of the exhaust manifold.

$[CO_2]_{ref}$ = measured concentration of CO_2 in the fresh air-NG mixture.

The internal EGR is the residual mass from a previous cycle trapped in the cylinder at IVC. The burned gas fraction is defined by the fraction of total combustion products in the in-cylinder mixture at IVC. By this definition, the summation between external EGR and internal EGR is thus the burned gas fraction.

3. Engine Cycle Simulation

An engine cycle simulation AVL-BOOST model was built and tuned with experimental data to provide additional information for thermodynamic state histories of the mixture in the engine system. Figure 2 shows the layout of our engine model.

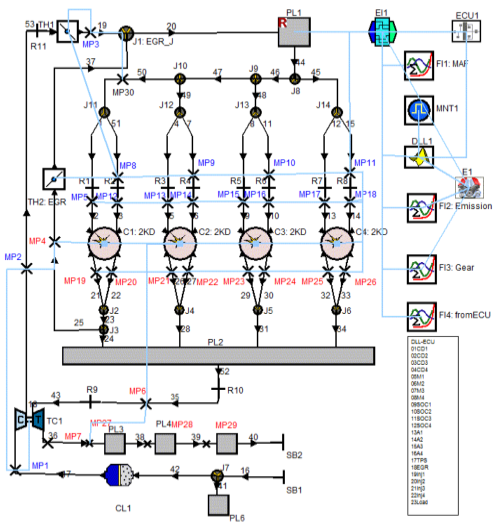

Fig. 2 Layout of the AVL-BOOST engine model

In order to estimate the mass-averaged temperature at IVC, we tuned the AVL-BOOST simulation model by using mass flow data and cylinder pressure traces from experiments. The mass flow rates of air and fuels, intake temperatures and pressures, and exhaust temperatures and pressures were calibrated with measured data.

4. Engine Test Conditions

Table 2 lists five steady-state engine operating conditions in the current study selected from data of load versus speed on NEDC as shown in Figure 3 [6]. These operating points were selected from our NEDC data [8]. Since the chemical kinetics of the air-fuel mixture played an important role in the DDF combustion, we maintained the overall inlet equivalence ratio (diesel + NG) being constant at 0.6. This equivalence ratio of the mixture allowed us to investigate a boarder DDF operating window in this engine and compare results under different loads. To achieve this equivalence ratio, we controlled the air flow rate, and the EGR by means of the throttle valve and the EGR valve. This DDF technique was already demonstrated in our previous work [6].

It should be noted that the net IMEP calculated over a complete engine cycle was uncontrolled. The engine load changed with the combustion characteristics.

Preliminary results from engine cycle simulation suggested that changes in the exhaust valve timing altered the IVC temperature. For the same cam profile, the current cylinder head configuration settings allowed us to adjust the cam shaft in a step of which corresponded to 19° of crank angle. From this, three different exhaust cam settings were selected including (i) the standard OEM valve timing, (ii) the 19° timing advance, and (iii) the 38° timing advance. Advancing the exhaust valve opening from the 38° timing advance resulted in overly preheated intake mixture and decreased expansion work. Figure 4 shows all valve lift profiles used in the current work. For each natural gas injector, the end of injection timing was kept at 270° before firing TDC. A single-pulse diesel injection was employed for the current study. The SOI was varied during late compression as suggested in our previous study [7].

Table. 2 Engine test conditions

Test Set	Speed [rpm]	EV advance [°CA]	Air [kg/h]	NG [kg/h]	Diesel [kg/h]	Diesel SOI [°CA bTDC]	pressure [MPa]	Net IMEP [bar]	Overall ϕ
1	1900	0°, 19°, 38°	44.30	1.58	0.54	50° to 20°	39.6	2.0 – 2.5	0.6
2	1900	0°, 19°, 38°	52.60	1.91	0.63	50° to 30°	42.7	2.5 – 3.0	0.6
3	1900	0°, 19°, 38°	63.25	2.30	0.77	50° to 30°	48.0	3.5 – 4.0	0.6
4	2400	0°, 19°, 38°	66.43	2.40	0.80	50° to 30°	49.8	2.5 – 3.0	0.6
5	2400	0°, 19°, 38°	80.23	2.90	1.01	50° to 30°	55.8	3.5 – 4.0	0.6

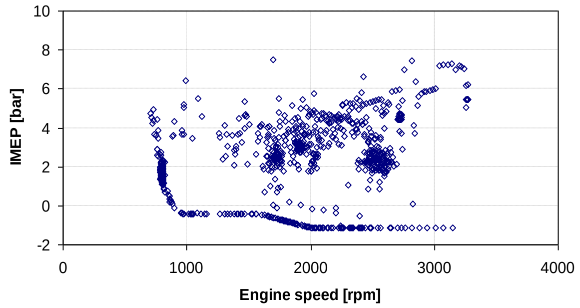

Fig. 3 IMEP vs. engine speed on the NEDC [6]

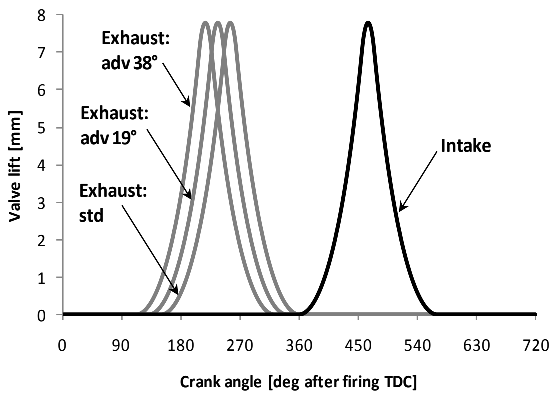

Fig. 4 Intake and exhaust valve lift profiles

5. Results and Discussion

5.1 EGR and Temperature at IVC

Figure 5 shows the amounts of external EGR at IVC calculated by Equation (2) and the mass-averaged temperature at IVC calculated by the AVL-BOOST model. In the model, we used data of mass flow rates, temperatures and pressures in the intake system, and mass burning rates as inputs for the simulation. The model validation was performed under motored and fired conditions to ensure the predictions of thermodynamic state histories of the air-fuel mixture.

In general, the external EGR (defined by %EGR) was slightly decreased as the exhaust valve timing advanced from 0° to 19°. Changes in the amount of external EGR were discernable in only Test Set 4. For all test conditions, the decrease in the external EGR was more pronounced as the exhaust valve timing advanced further to 38°. Based on discussion from our previous work [8] under these engine conditions, as the exhaust valve timing was advanced from 19° to 38°, the residual mass fraction was significantly increased and the total mass fraction burned (i.e. residual mass + external EGR) was nearly unchanged. As the residual mass was hotter, the greater residual fraction resulted in a hotter charge temperature at IVC. By comparing data in Figure 5, as the load increased, the external EGR was reduced.

It should be note that the mass flow are of the air was constant for the same test set. As a result, the total residual fraction would be roughly constant for different exhaust valve settings with all other engine parameters being the same. From this assumption, the residual trapped at IVC (a so-called "internal EGR") should be increased with advanced exhaust valve timings.

From the temperature results, advancing the exhaust valve from 0° to 19° did not have a significant change in the IVC temperature. The change in IVC temperature became more obvious as the exhaust valve timing advanced from 19° to 38°. Depending on engine load and speed, the 38° exhaust valve timing advance could raise the IVC temperature approximately 10 to 20 K compared to the other valve timings.

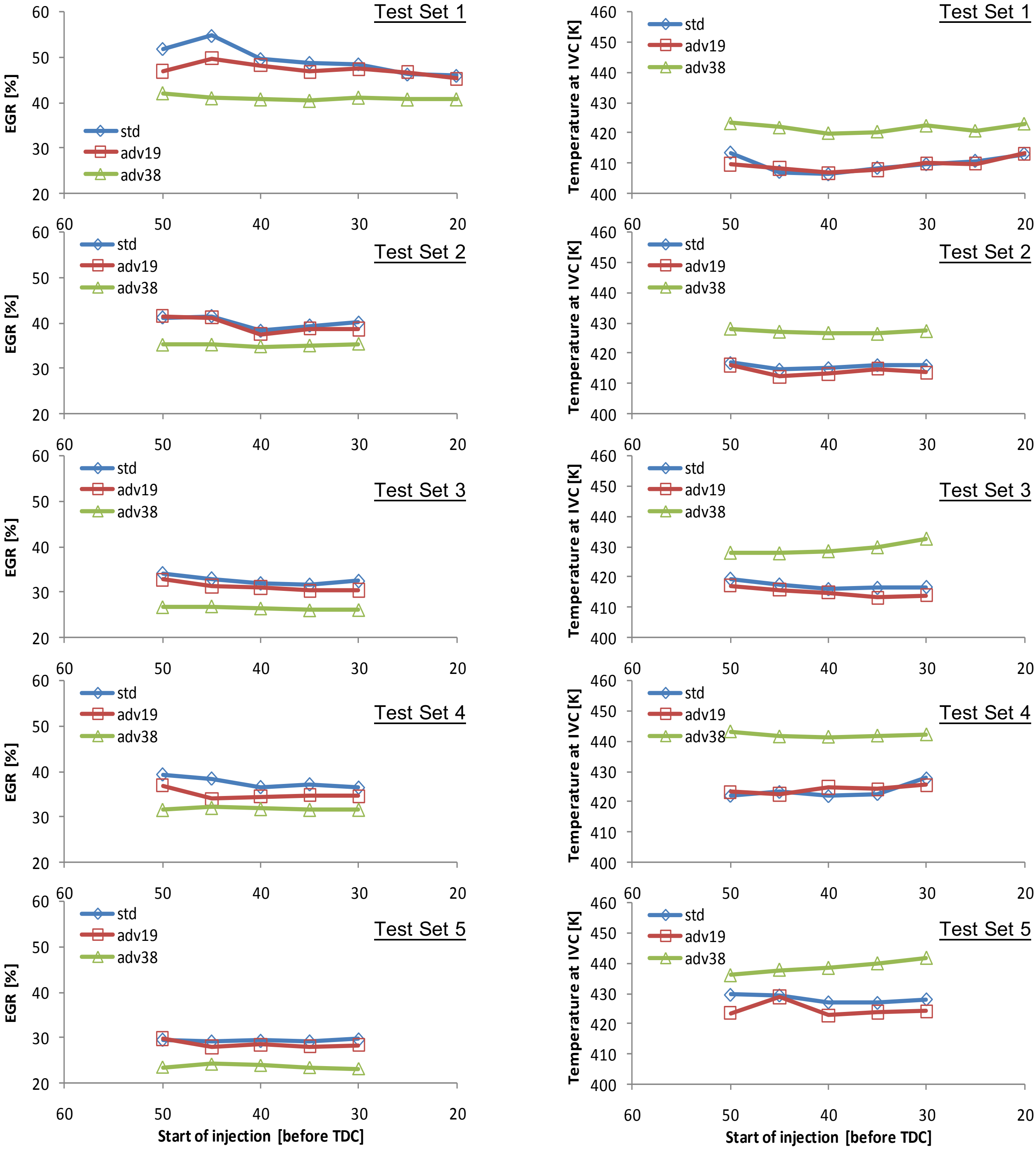

Fig. 5 External EGR and calculated temperature at IVC for different start of injection timings

5.2 Combustion Characteristics

Figure 6 presents histories of pressure, heat release rates, and mass-averaged temperature of cylinder 1 at different diesel injection timings for Test Set 2 with standard exhaust valve timing. As the SOI was retarded from 50° to 30°

before TDC, the heat release rates became more rapid. Retarding the SOI to 30° caused the mass-averaged temperature during the combustion to increase.At SOI = 50° before TDC, the combustion stability was very poor (COV of IMEP = 40%), where there were quite a few misfire cycles.

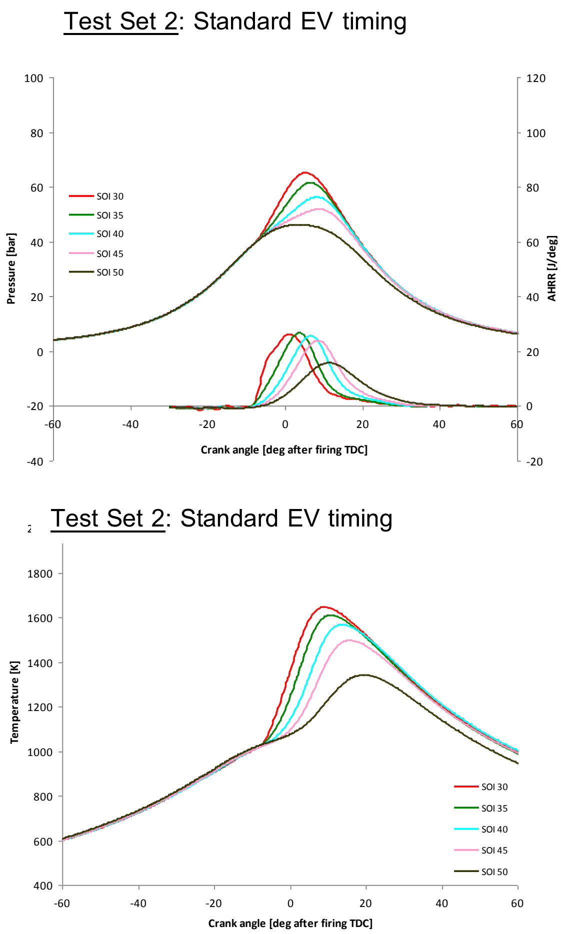

Fig. 6 Pressure, apparent heat release rates, and mass-averaged temperature histories for Test Set 2 (1900 rpm) using the standard exhaust valve timing

In contrast, Figure 7 shows results from Test Set 5 with 38° exhaust valve timing advance. Under this engine condition, both engine speed and load were greater than those shown in Figure 6. In Figure 7, the combustion stability was very stable (COV of IMEP > 2.5%) for all data points. Changes in the combustion timing were less sensitive to changes in the SOI. Compared to other SOI timings, the earliest SOI of 50° before TDC showed the greatest rates of heat release where the maximum rate of pressure rise was about 9 bar/°CA.

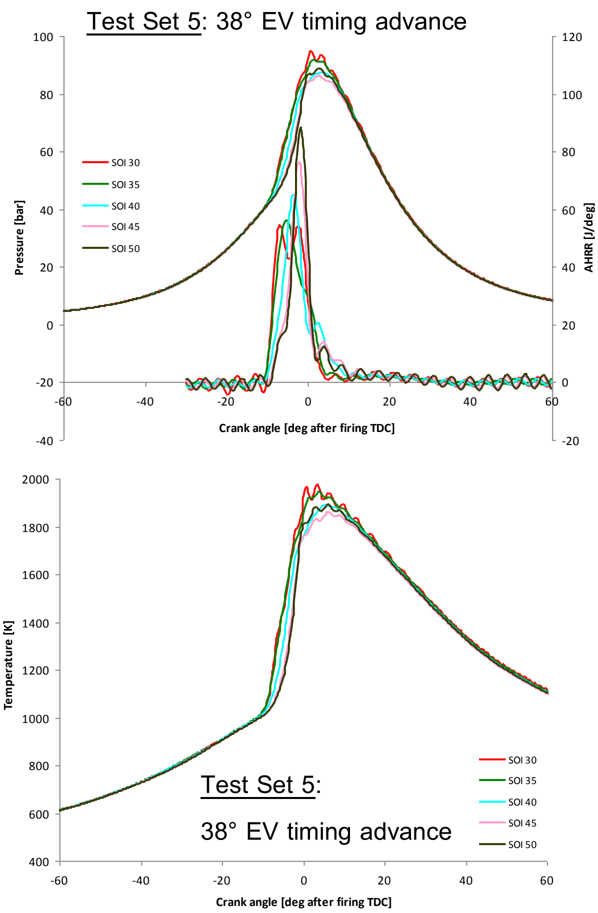

Fig. 7 Pressure, apparent heat release rates, and mass-averaged temperature histories for Test Set 5 (2400 rpm) using the 38° exhaust valve timing advance

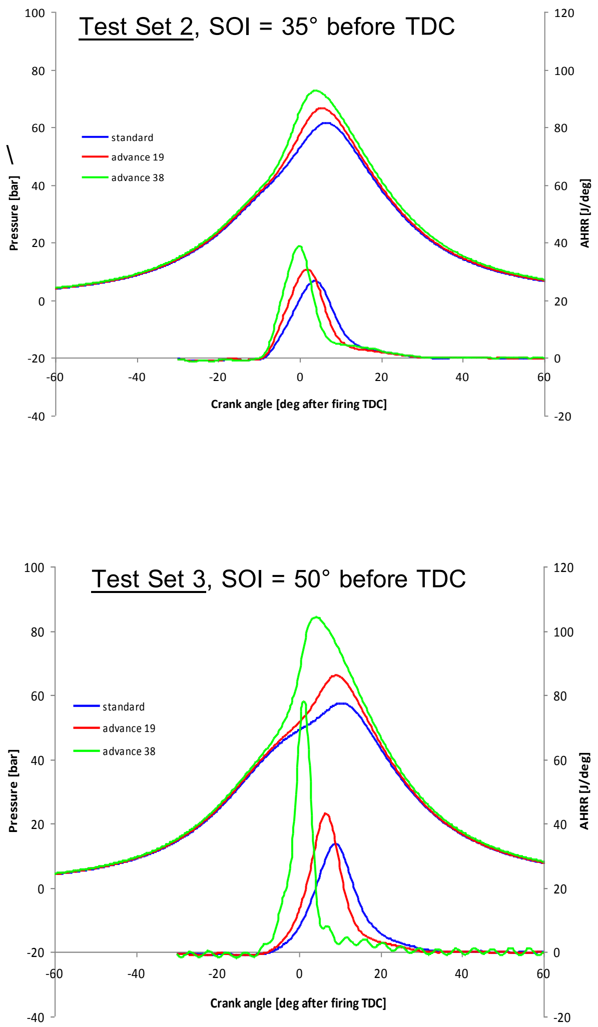

Fig. 8 Changes of pressure and apparent heat release rate histories with exhaust valve timing for Test Set 2 with SOI = 35° before TDC and Test Set 3 with SOI = 50° before TDC

Figure 8 compares traces of cylinder pressure and apparent heat release rates for different exhaust valve timings under two selected engine conditions. Advancing the exhaust valve timing resulted in more rapid heat release rates. We hypothesize that the more rapid combustion rate may involve two main reasons. One is the charge being hotter which makes the mixture more prone to react. The onset of auto-ignition is thus easier to reach from multiple locations in the cylinder. The other main reason involves residual gas effects. A greater amount of residual gas heats up the entire mixture. The low temperature oxidation process of charge mixture can be affected by greater concentrations of intermediate species [15,16]. Furthermore, it also promotes stratification effects in the mixture as a result of residual gas mixing which influences the combustion phasing [17]. Regarding the effect of a residual gas fraction distribution on the combustion, the computational HCCI study by Babajimopoulos et al. [12] showed that the effect of composition stratification on the onset of ignition was noticeable only at high levels of residual gas fraction.

5.3 Emission Characteristics

Since data in each test set have a variation in loads due to combustion phasing, we present gaseous emissions in terms of indicated specific emissions. Indicated specific amounts of CO, HC, and NO_x (ISCO, ISHC, and $ISNO_x$), combustion efficiency, and filter smoke number (FSN) from Test Set 1 (1900 rpm, net IMEP = 2 – 2.5 bar) and Test Set 3 (1900 rpm, net IMEP = 3.5 – 4 bar) are presented in Figure 9.

As the injection timing was retarded, the combustion efficiency was increased. This was mainly a result of reduction in HC and CO emissions. Comparing different exhaust valves, the 38° timing advance can significantly improve the combustion efficiency (i.e. less HC and CO). However, at higher load, the 38° timing advance was penalized by greater NOx (> 100 ppm with the SOI of 40° before TDC) and slightly greater filter smoke number (~ 0.2 FSN with the SOI of 40° before TDC). Although the charge was hotter with the most advanced exhaust valve timing, the greater exhaust residual trapped in the cylinder (presented in our previous work [8]) would enrich the mixture air-fuel ratio in the cylinder. This could be the reason for a slightly greater smoke compared to the other two valve timings in Test Set 3. It should be also noted that with an early diesel injection technique, as the charge became hotter, HC and CO emissions were reduced. The reduction in HC and CO was more pronounced than the slight increase in FSN. As a result, the combustion efficiency was greater than other valve timings.

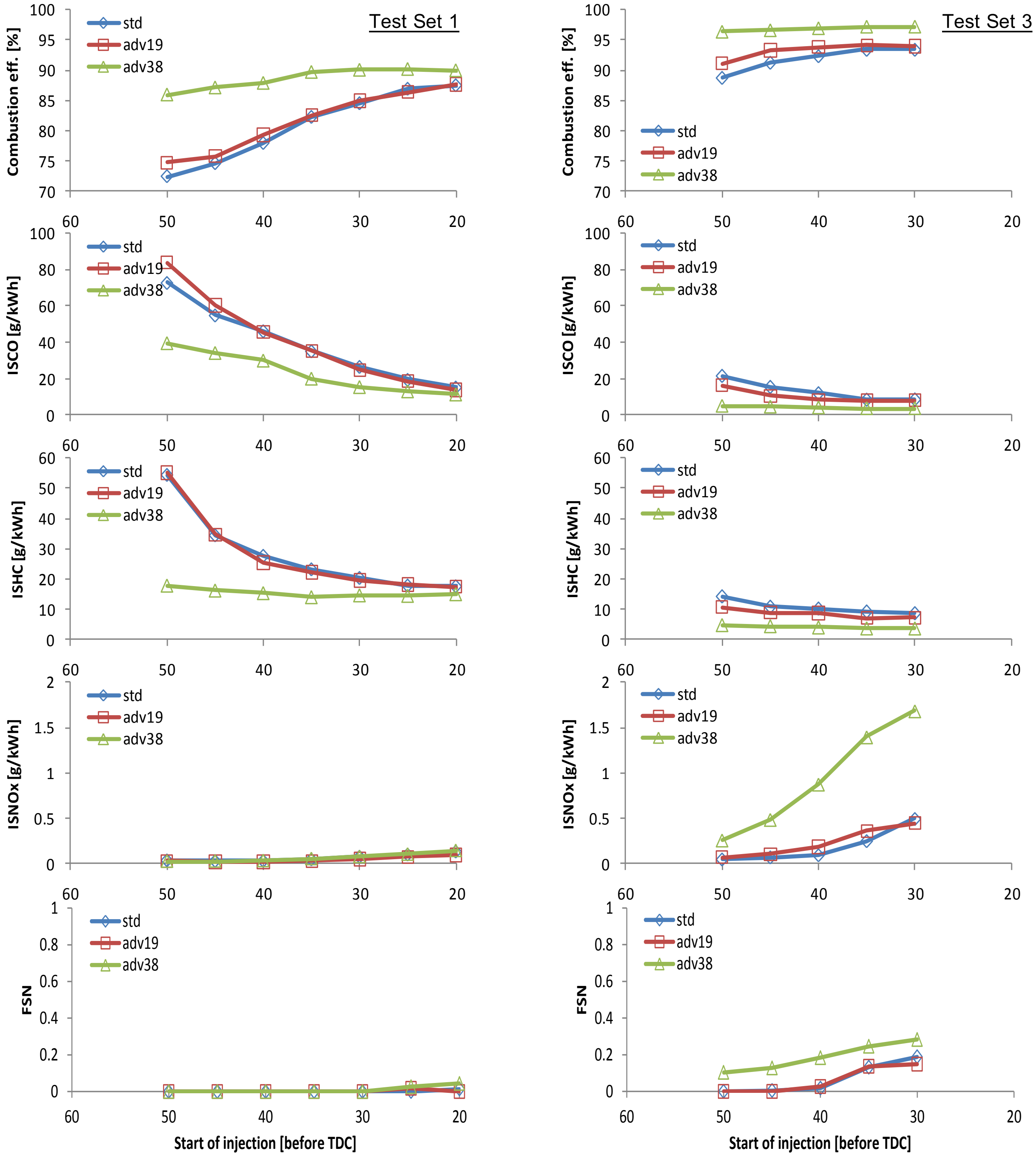

Fig. 9 External EGR and calculated temperature at IVC

5.4 Discussion

It appeared that use of early exhaust valve timing was most beneficial for low load DDF operations. With early exhaust valve timing, a greater amount of exhaust residual (and probably enhanced residual gas mixing and charge inhomogeneities) helped promote the combustion stability and the combustion efficiency. An early exhaust valve open also gives higher exhaust gas temperatures (shown in Figure A1 in the Appendix). Moreover, a hotter exhaust gas temperature provide is favorable for an oxidation catalyst operation.

It should be noted that although the 38° exhaust valve timing advance was used, the exhaust gas temperature was still ~200°C. This temperature level was too low for methane conversion in an oxidation catalyst [18,19]. Further investigations into hydrocarbon (mainly methane) emission control should be performed which may involve both engine parameters and additional exhaust aftertreatment devices.

As the engine load increased, early exhaust valve timing was imposed by too rapid energy release rates during combustion and greater amounts of NO_x and smoke emissions. This was mainly a result of excessively heated charge mixture. To avoid this problem, the optimum valve timing in this study should be the original OEM timing.

6. Conclusion

Data have been presented for DDF operation in a four-cylinder turbocharged, direct injection engine. Experiments were conducted under engine speeds of 1900 rpm and 2400 rpm using 3 different exhaust valve timings, including (i) the standard timing, (ii) 19° timing advance and (iii) 38° timing advance. Flow rates of the air, natural gas, and diesel were kept constant for the same test set conditions.

Results showed that exhaust valve timing advance changed the amounts of both internal EGR and external EGR, and altered the mixture temperature at IVC. Under low-load operations (IMEP = 2 to 3 bar), data suggested that:

- With an optimum SOI timing, the 38° timing advance improved the combustion stability and provided lowest HC and CO emissions with relatively low NO_x emissions (< 50 ppm) and smoke emission (< 0.1 FSN).
- Advancing the exhaust valve timing raised the exhaust gas temperature which was favorable for the oxidation catalysts.

With higher load (IMEP = 3 to 4 bar), advanced exhaust valve was penalized by overly heated charge mixture which results in too rapid energy release rates. This also resulted in high NOx emissions (> 100 ppm) and greater amounts of smoke emission. The findings from the current study provide a guideline of exhaust valve timings for improving DDF engine operations.

7. Acknowledgement

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10. Abbreviations

AHRR	Apparent heat release rate
COV	Coefficient of variation
DDF	Diesel dual fuel
EGR	Exhaust gas recirculation
FSN	Filter smoke number
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated mean effective pressure
ISCO	Indicated specific carbon monoxide
ISHC	Indicated specific hydrocarbon
ISNO	Indicated specific nitrogen oxides
IVC	Intake valve closure
LHV	Lower heating value
NEDC	New European Driving Cycle
SOI	Start of injection

11. Appendix

Table. A1 Properties of natural gas used in the current work

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 HC (> 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 (est.)	12.30
H (est.)	22.13

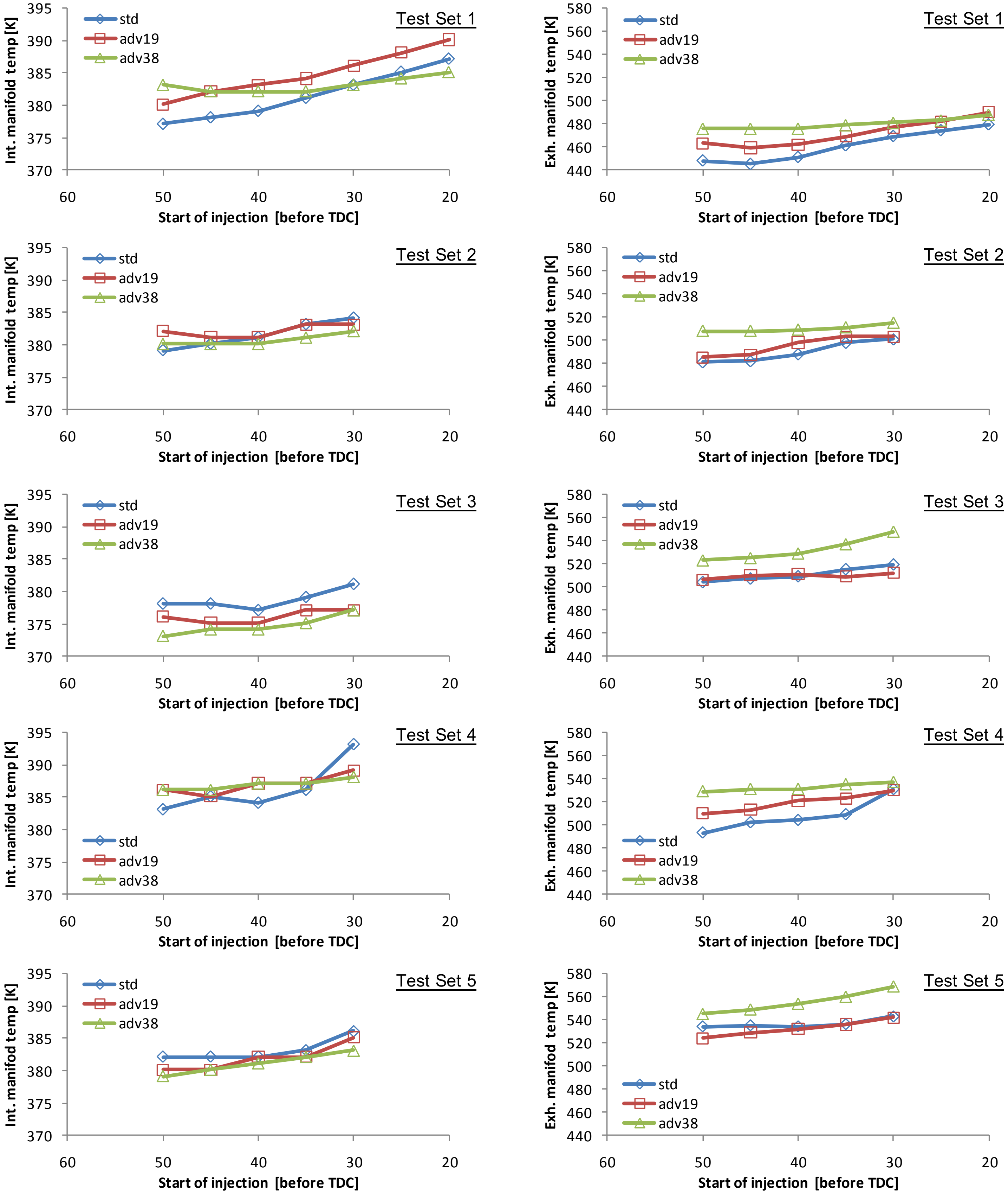

Fig. A1 Measured intake and exhaust manifold temperatures