Swirl Enhancement for Improvement of Diesel Dual Fuel Engines under Low Load Operations

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Abstract

A major challenge in low-load DDF operation was use of premixed charge with low combustion temperatures which resulted in much higher THC and CO emissions than those of conventional diesel operations. In conventional diesel engines, swirl control valve is used to promote turbulence in the in-cylinder flow field. This can promote fuel-air mixing and leads to the reduction of soot formation.

The aim of the study was to investigate the effect of use of swirl control valve on operating characteristics under low-load DDF operations. Experiments were conducted on a 4-stroke, 4-cylinder, turbocharged diesel engine with natural gas injected via a sequential port fuel injection system. The comparison was made for combustion data, engine-out emissions, and energy conversion from total fuel's energy under DDF operations at 1800 and 2500 rpm.

Results from simulation and experiment showed that use of swirl control valve could reduce THC and CO, especially at a low-speed condition. These reductions contributed to an improvement in the combustion efficiency. However, using the swirl control valve was penalized by greater pumping loss and greater heat loss. This altogether resulted in a slight reduction in the brake efficiency of the DDF engine.

Keywords: Diesel Dual Fuel, Natural Gas, Hydrocarbons, Pumping loss, Swirl Control Valve

1. Introduction

Natural gas (NG) has been recognized as one of the 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. The term "diesel dual fuel (DDF)" engine is referred to as a premixed-natural-gas diesel-ignited 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 [1–6] or stratifically by a direct injection into the combustion chamber .The pilot diesel fuel injection serving as an ignition source is usually injected during the compression stroke.

In conventional diesel engines, swirl port and swirl control valve (SCV) is used to promote turbulence in the in-cylinder flow field, thus, promoting fuel vaporizing. This can promote fuel-air mixing and leads to the reduction of soot formation. At low engine-speed operations, the SCV was activated (swirl flap closed) for promoting swirl. A thigh engine-speed operations, the SCV was deactivated (swirl flap opened) to gain more air intake into cylinder [7].

As the open angle position of the swirl flap has an impact on in-cylinder mixture formation, it affects a turbulent flow field of the charge and has a potential to control DDF combustion [8]. Srisattayakul N. [9] found that changes in the engine speed (which influenced different turbulence intensity) led to changes in the ignition dwell time and, potentially, the mixture formation in a DDF-engine cylinder. Total hydrocarbon and CO emissions from DDF engine were greater than that of conventional diesel engines [2,3,6] but NOx and soot were declined with an increase in energy replacement of diesel fuel.

The aim of the current study was to investigate the effect of use of SCV on operating characteristics under low-load operation. DDF experiments were conducted on a 4-stroke, 4-cylinder, turbocharged diesel engine. The comparison was made for combustion characteristics and engine-out emissions under different engine conditions when the SCV was activated and deactivated.

2. Experimental Setup

2.1 Test bed setup

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

Table.1 Engine specifications

Number of cylinder	4 cylinders , inline
Displaced Volume	2,494 cc
Stroke	93.8 mm
Bore	92 mm
Connecting rod	158.5 mm
Compression ratio	17.4 : 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

Fig. 1 shows a schematic of the experimental setup used in the current study. The engine was coupled to an AC engine dynamometer, AVL Schneider Electric Power Drive . The air at a room condition was drawn to the intake system. The mass of diesel was measured with AVL 733. The micro motion compressed natural gas meter, CFM010M, was used to measure the mass flow of natural gas.

Natural gas was supplied via a sequential port injection system similar to our previous work [10]. Each natural gas injector for each cylinder was connected with a rubber hose and attached to the intake runner that was connected to the swirl port as shown in Fig. 2. For each natural gas injector, the total distance from the injector tip to the inlet valve was approximately 345 mm. As suggested in the study by Jantaradach [10], EOI of natural gas was set at 390°bTDC.

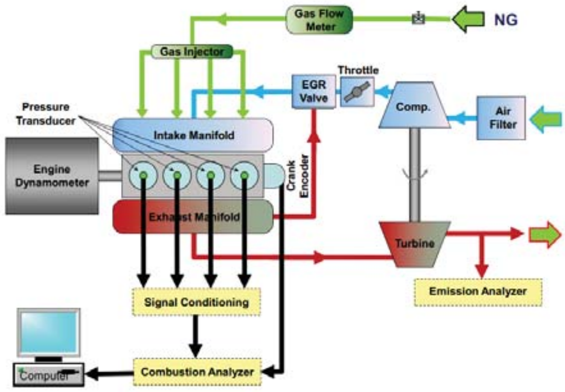

Fig.1 Schematic of experimental setup.

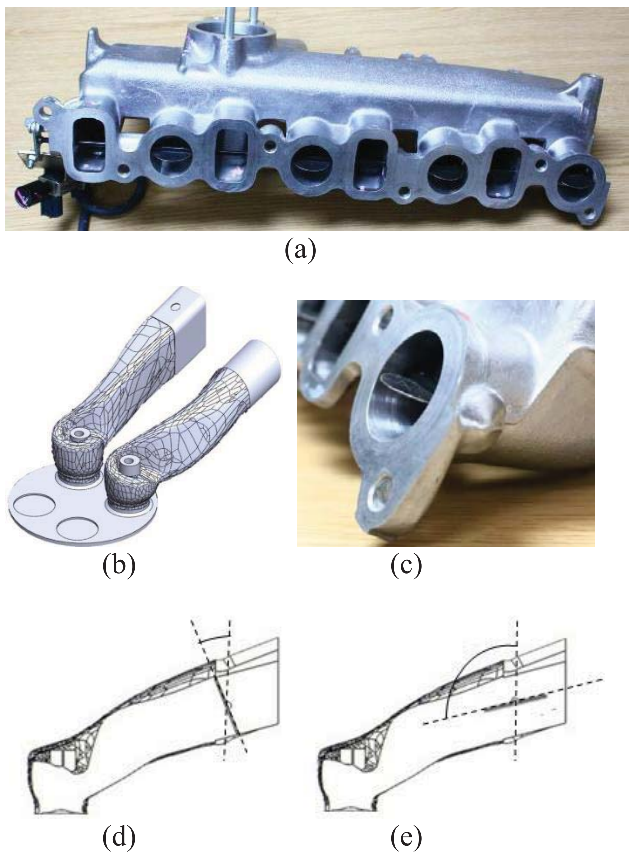

Fig.2 Location of the SCV in the intake system
(a) The intake manifold
(b) The swirl port (left) and the round port (right)
(c) Swirl flap at the deactivated position (open)
(d) SCV activated (swirl flap fully closed)
(e) SCV deactivated (swirl flap fully opened)

Each cylinder was equipped with an angle enconder, Kristler 2614A, and a quartz piezoelectric pressure transducer, Kristler 6061B. Cylinder pressure data of 150 consecutive cycles from all cylinders were recorded for combustion data analysis by DEWETRON model DEWE52 PCI-3216. Regarding our previous work [11], using cylinder pressure data more than 150 consecutive cycles did not significantly change the coefficient of variation (COV) of IMEP. Data of heat release rate (HRR) were calculated from pressure data by using DEWESoft 6.6.5.

For each cycle, the crank angle resolution of pressure data of 0.2° was recorded. Although higher resolution of crank angle would offer finer traces of pressure and HRR, we did not observe significant changes in measured pressure data. The coolant temperature was maintained at 85°C. The oil temperature was maintained at 90°C. Exhaust gas emissions were analyzed by using the Horiba MEXA 7100 DEGR gas analyzer. AVL 483 Micro Soot Sensor was used for soot measurement.

2.2 Fuels

Properties of diesel fuel and natural gas in this study are shown in Tables 2 and 3.

Table 2 Properties of natural gas used in the current study.

Methane number	83.0
Higher heating value, MJ/kg	37.71
Lower heating value, MJ/kg	34.10
Stoichiometric A/F	11.68
MW, kg/kmole	22.39
Methane, % by mole	74.2
Ethane, % by mole	5.8
Propane, % by mole	2.3
n-Butane, % by mole	0.4
i-Butane, % by mole	0.5
Larger hydrocarbons (> C_6), % by mole	0.3
CO_2, % by mole	14.5
N_2, % by mole	2.0

2.3 Test Metrics

Engine operating points were selected from driving conditions frequently used in the New European Driving Cycle (NEDC) test in a chassis dynamometer using a Toyota Hilux Vigo, a light-duty pickup truck powered by the same engine model used in this study [10].

Table.3 Properties of natural gas in Thailand (PTT sampling data in 2011)

Density, kg/m³ (calc.)	0.83
Higher heating value, MJ/kg (calc.)	45.64
Lower heating value, MJ/kg (calc.)	42.75
Stoichiometric A/F (est.)	14.41
MW, kg/kmole (est.)	170
C (est.)	12.30
H (est.)	21.92
O (est.)	0.04

Activated and deactivated SCV at engine speeds of 1800 and In this study, we explored steady-state DDF operations with 2500 rpm to observe the effects of SCV on the engine operation and emissions. Experimental conditions are provided in Table 4.

Under these engine operating conditions, the total fuel energy was kept constant and the engine torque was uncontrolled. The engine torque ranged between 36 and 42 N-m, which was considered in the low-load operating range. To discern changes in the mixture formation in the charge due to the SCV activation, EGR was not used in this study.

Table 4 Experimental matrix

No.	Engine speed	NG	Diesel	Air	SCV
	[rpm]	[mg/cycle]	[mg/cycle]	[kg/hr]	-
1	1800	9	6	117	Activated / Deactivated
2	2500	9	6	152	Activated / Deactivated

Diesel fuel was injected using a two-pulse signal: first pulse at 35°bTDC and second pulse at 10°bTDC. For investigation into the effect of SCV, mass of air intake and fuel injection were controlled by tuning the throttle valve position to maintain the same equivalence ratio when the SCV was activated and deactivated.

3. Result and Discussion

Fig. 3 shows the effect of use of SCV on the in-cylinder pressure histories and HRR. Data of ignition delay are provided in Fig. 4. At 1800 rpm, activating the SCV slightly lowered the pressure during the compression and advanced the start of combustion. These observations were less pronounced at 2500 rpm.

To gain more insight in the process prior to the combustion start, cumulative heat release data were analyzed as shown in Fig. 5 and 6. By activating the SCV, cumulative heat release was lower (more negative slope) compared to that of the SCV deactivated. Activating the SCV promoted the swirling motion of the charge, thus boosting the swirl ratio of the charge. Higher swirl ratio helps accelerate the rate of droplet evaporation, increase the turbulent mixing between fuel and air, and shorten the combustion duration (see Fig. 7) [12]. As higher swirl ratio increased rate of heat transfer [12], it was not beyond our expectation to observe a more negative slope of the cumulative heat release before the start of combustion timing. By comparison, changes in heat release were less pronounced as the engine speed was increased from 1800 to 2500 rpm.

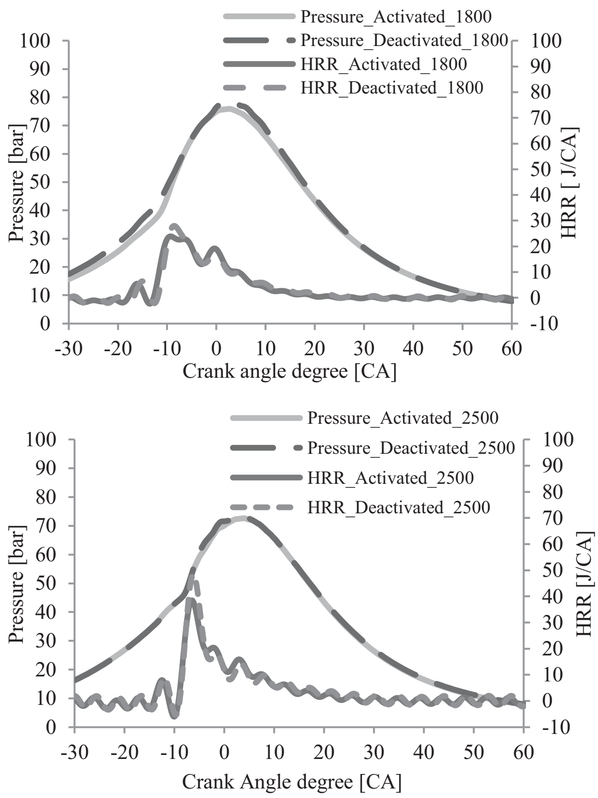

Fig.3 Heat release rate and pressure for activated and deactivated SCV at 1800 rpm (top) and 2500 rpm (bottom).

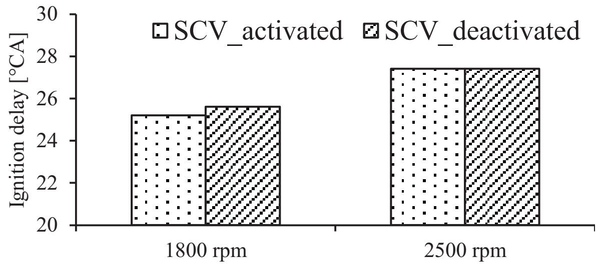

Fig.4 Ignition delay for activated and deactivated SCV at 1800 rpm and 2500 rpm

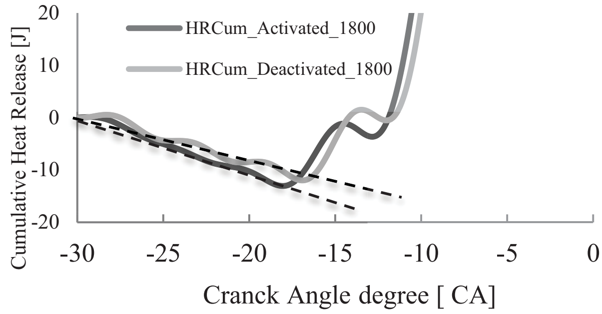

Fig. 5 Cumulative heat release before the start of combustion for activated and deactivated SCV at 1800 rpm

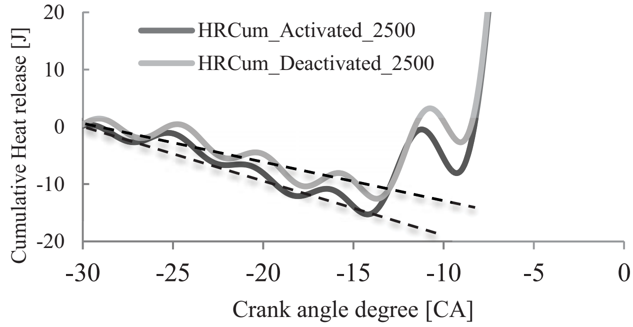

Fig.6 Cumulative heat release before the start of combustion for activated and deactivated SCV at 2500 rpm

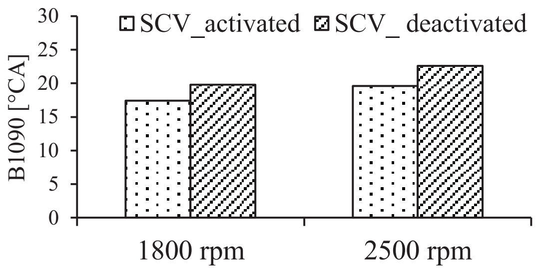

Fig.7 Combustion duration (B1090) for activated and deactivated SCV at 1800 rpm and 2500 rpm

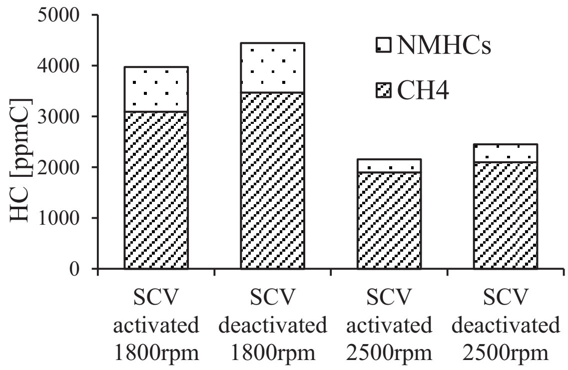

Fig.8 Hydrocarbon emissions (CH4 and Non methane hydrocarbon) for activated and deactivated SCV at 1800 rpm and 2500 rpm

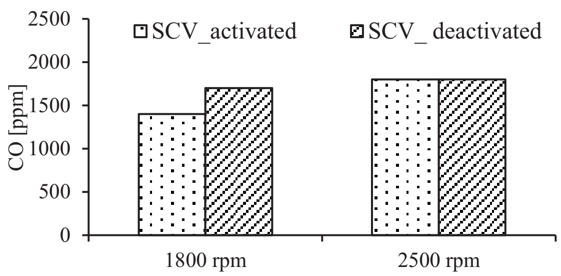

Fig.9 CO emission for activated and deactivated SCV at 1800 rpm and 2500 rpm.

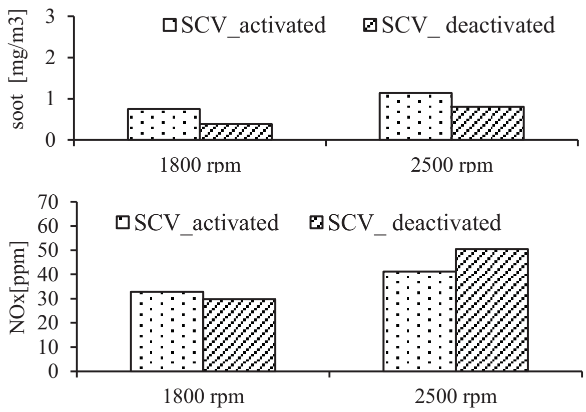

Fig.10 NO_x(top) ,and soot (bottom) emissions for activated and deactivated swirl control valve at 1800 rpm and 2500 rpm.

Data of THC and CO, NO_x, and soot engine-out emissions are shown in Fig. 8 to 10. It should be noted that CH_4, a majority in THC, was mostly derived from natural gas combustion. Activating the SCV demonstrated reductions in THC and CO, especially under the engine operation at 1800 rpm. We hypothesize that these observations are responsible by changes in the mixture formation and greater turbulence intensity, as proposed in the study of charge motion prior to the combustion by Sombut [8]. As the engine speed increased, changes in the turbulence intensity due to swirl valve activated became less pronounced as a result of a stronger turbulent flow field. THC emissions were slightly decreased with CO was approximately the same. The effect of SCV activation on NOx and soot emissions was yet not significant as the discrepancies in data were considered within the tolerance range of measurement.

Regarding the energy converted from fuel energy illustrated in Fig.11, the activation of SCV slightly deteriorated the brake efficiency. Despite an improvement in the combustion efficiency (lower exhaust chemical energy loss), the brake efficiency was penalized by greater pumping loss and heat loss. Loss due to rubbing friction was nearly unchanged as the SCV was activated (the difference less than 0.4%). In this figure, losses are a combination of heat transfer and exhaust sensible energy. Heat loss is expected to increase with greater swirl ratios, as suggested in Heywood [12].

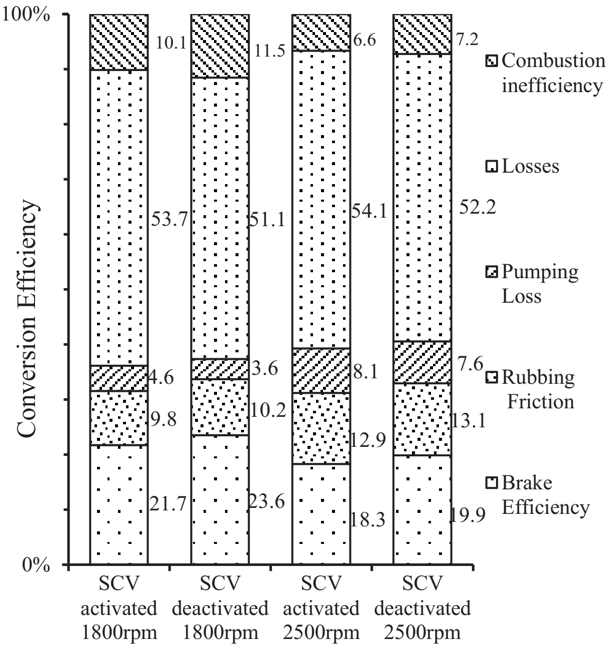

Fig.11 Energy conversion from fuels for activated and deactivated SCV at 1800 rpm and 2500 rpm.

With no data of exhaust temperature presented, activating the SCV did not significantly alter the exhaust temperature. This implies that the exhaust sensible energy would be approximately the same. Therefore, an observation of greater "losses" in Fig.11 with SCV activated would be mostly due to heat transfer, which was more pronounced at a lower speed. An increase in heat transfer was consistent with the observation found in the cumulative heat release data, presented in Fig.5 and 6.

4. Conclusion

Data have been presented for steady-state multi-cylinder DDF operations at 1800 and 2500 rpm for investigating use of the swirl control valve. The conclusion can be drawn as follows:

- At 1800 rpm, use of swirl control valve slightly advanced the start of combustion (shorten ignition delay) and slightly increased heat transfer. With this swirl enhancement, CO and THC emissions were reduced.

- At 2500 rpm, use of swirl control valve did not show significant effects on the start of combustion (same ignition delay) and CO emission. Yet, it still demonstrated slightly improvement in THC. This essentially led to a slight increase in the combustion efficiency.
- Use of swirl control valve did not offer discernable improvement in NOx and soot emissions.
- Activating the swirl control valve helped promote combustion efficiency; however, it was penalized by greater pumping loss and heat loss. This altogether resulted in a slight drop in the brake efficiency.

5.Acknowledgement

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6. References

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7. Abbreviations

A/F	Air-fuel ratio
bTDC	before firing TDC
CO	Carbon monoxide
CO_2	Carbon dioxide
COV	Coefficient of Variable
DDF	Diesel Dual Fuel
EGR	Exhaust gas recirculation
EOI	End of injection
HC	Hydrocarbons
HCCI	Homogeneous Charge CompressionIgnition
IMEP	Indicated mean effective pressure
LHV	Low heating value
NG	Natural gas
NMHCs	Non Methane Hydrocarbons
NOx	Oxides of nitrogen
SCV	Swirl Control Valve
SOC	Start of combustion
SOI	Start of injection
TDC	Top dead center
THC	Total hydrocarbons
VN	Variable nozzle (for the turbine)