EFFECTS OF ETHANOL PORT INJECTION TIMING AND DELIVERY RATE ON COMBUSTION CHARACTERISTIC OF A HEAVY-DUTY V-12 DIESEL ENGINE

by

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Using biobased fuels in Diesel engines is an effective solution reduce the generation of toxic components in the exhaust gas. One of them, alcohol, is the potential fuel to reduce emissions and dependence on fossil fuels. So, this study is aimed to investigate the effects of ethanol port injection timing and delivery rate on the combustion characteristic of a heavy-duty V-12 Diesel engine when ethanol substitution percentage is 30% to reach the original diesel quantity at full load. The combustion characteristic analysis indicates that the variation in cylinder gas pressure and temperature decreases when retarded ethanol injection timing and decreased ethanol delivery rate, the engine works more smoothly due to the maximum rate of pressure rise decreases. However, the changes are greater when changing the ethanol injection timing as compared to ethanol delivery rate case.

Key words: *biobased fuels, ethanol port injection timing, ethanol delivery rate, V-12 Diesel engine, combustion characteristic, heavy-duty*

Introduction

Vehicle pollutants are emitted from the engine by three main sources [1]: the firstly, the crankcase where piston blow-by fumes and oil mist are vented to the atmosphere, the secondly, the fuel system where evaporative emissions from the carburettor or petrol injection air intake and the fuel tank are vented to the atmosphere, and finally, the exhaust system where the products of incomplete combustion are expelled from the tail pipe into the atmosphere. Spark-ignition (SI) and Diesel engines are major sources of air pollutants. The SI engine exhaust gases contain NO_x (nitric oxide – NO, and small amounts of nitrogen dioxide – NO_2, collectively known as NO_x), CO, and organic compounds, which are unreacted or partially reacted fuel HC. In Diesel engine exhaust, concentrations of NO_x are comparable to those from SI engines [2]. Improvement to Diesel engine combustion process and after-treatment technologies is expected for reducing the emissions of new vehicles. For in service vehicles, the use of alternative fuels has been widely investigated [3].

Alcohols, mainly methanol and ethanol, in combination with diesel fuel, have been widely investigated for reducing the NO_x and the particulate matter (PM) emissions [4]. The alcohols and the diesel fuel are mostly applied together either in the blended mode or in the

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fumigation mode, while other approaches such as dual fuel injection are less commonly investigated. In comparison with the blended mode, the fumigation method seems to be more flexible despite extra fuel injection system is required. Thus, the fumigation approach is used in this investigation. This paper aims to survey effects of ethanol port injection timing and delivery rate on combustion characteristic of a heavy-duty V-12 Diesel engine. In order to realize dual-fuel engine, the ethanol port injection system was added for ethanol injection into the intake manifold, while retaining the original diesel direct injection. The combustion parameters such as the in-cylinder pressure and temperature, heat release rate, 50% burned crank angle (CA), burn duration 0-50%, and burn duration 10-90% of the ethanol/diesel dual-fuel engine were studied with various ethanol injection timing (EIT) and ethanol delivery rate (EDR) values. The results of this study are expected to provide a basic understanding of effects of EIT and EDR on combustion of a heavy-duty V-12 Diesel engine in dual fuel mode.

Ethanol can be used in Diesel engines in the fumigation mode with diesel fuel injected directly into the engine cylinder and with ethanol injected into the air intake. Fumigation mode has been widely investigated and reported in [5-9]. It has been found that the application of ethanol fumigation technique leads to a significant reduction in the more environment concerning emissions of CO_2, NO_x, and PM. However, increase in CO and HC emission have been found after use of alcohol fumigation. Alcohol fumigation also increase the BSFC due to having higher heat of vaporization. Brake thermal efficiency decreases at low engine load an increases at higher engine load.

However, there are some investigations aimed at comparing the effects of EIT and EDR on combustion, engine performance and exhaust emissions in Diesel engine. Such us, Guedes *et al*. [9] investigated the effects of different injection timings for diesel-biodiesel-ethanol blends on engine's performance parameters and combustion characteristic. Their results showed the specific fuel consumption could be reduced 4%, while ethanol specific energetic conversion was up to 30% improved at optimized injection timings. Ning, *et al*. [10] investigated the effects of methanol injection timing (MIT) and methanol substitution percentage on the combustion and emissions of methanol/diesel dual-fuel engine. Their results indicated that methanol addition (direct injection) and retarded MIT allowed the diesel injection timing to be properly advanced. So, the study represents the attempt to survey the effects of ethanol port injection timing and EDR on combustion characteristic of a heavy-duty V-12 Diesel engine.

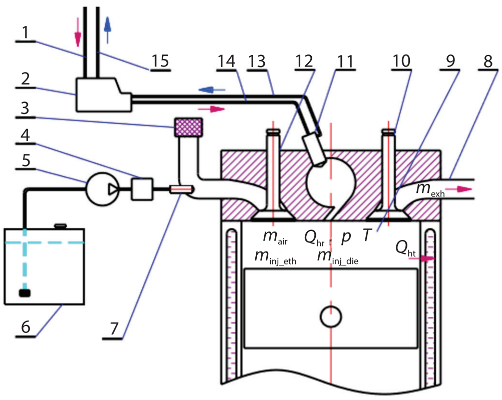

Figure 1. Schematic diagram of ethanol fumigation system, [11]:
1 – diesel fuel line to high pressure pump, 2 – high pressure pump, 3 – air filter, 4 – ethanol filter, 5 – ethanol pump, 6 – ethanol tank, 7 – ethanol injector, 8 – exhaust manifold, 9 – combustion chamber, 10 – exhaust valve, 11 – diesel injector, 12 – intake valve, 13 – fuel return line, 14 – high pressure fuel line, 15 – fuel return line back to diesel tank; m_{inj_eth} – the amount of ethanol fuel, m_{air} – the amount of air, m_{inj_die} – the amount of diesel fuel, m_{exh} – exhaust gas, Q_{hr} – the radiant heat, Q_{ht} - the heat transferred to the combustion chamber walls, p – in-cylinder pressure, and T – in-cylinder temperature

Modelling and simulation of dual-fuel engine operation with ethanol fumigation

The diesel-ethanol dual fuel delivery system consists of two independent working systems: the diesel fuel delivery system and the ethanol port injection system as detailed in fig. 1 [11].

This study is aimed to investigate the effects of EIT and EDR on the combustion characteristic of a heavy-duty V-12 Diesel engine at the maximum power speed of 2000 rpm and 100% engine load. The properties of the fuels are presented in tab. 1.

Table 1. Properties of the tested fuels, [6, 12]

Parameters	Diesel	Ethanol
Molecular formula	$C_{14}H_{30}$	C_2H_5OH
Molecular weight, [gmol^{-1}]	198.4	46.07
Density at 20 °C, [gcm^{-3}]	0.856	0.785
Lower heating value, [MJkg^{-1}]	41.66	26.87
Cetane number	51	≈ 8
Carbon content [%.mass]	87	52.2
Hydrogen content [%.mass]	13	13
Oxygen content [%.mass]	0	34.8

The engine of this study was a V-12 engine that consists of 12 cylinders where two banks of six cylinders are arranged in a V configuration around a common crankshaft. The V-12 engine were equipped on the tanks of Russia and Vietnam. The main specifications of the V-12 diesel engine are summarized in tab. 2.

Table 2. Specifications of V-12 diesel engine under study [13]

Parameters	Symbol	Value	Unit
Number of cylinders	i	12	–
Engine type	V-12	Diesel, the V arrangement, the 12 cylinders are arranged in two banks of six, with a 60° angle between their axis	
The working order of the cylinders	1^L-6^R-5^L-2^R-3^L-4^R-6^L-1^R-2^L-5^R-4^L-3^R		
Compression ratio	ε	15±0.5	
Maximum power	–	387.4/2000	kW per rpm
Maximum torque	–	2256.3/1200	Nm per rpm
Intake valve open	IVO	340	°CA aTDC
Intake valve close	IVC	-132	°CA bTDC
Exhaust valve open	EVO	132	°CA aTDC
Exhaust valve close	EVC	380	°CA aTDC
Specific fuel consumption	$G_{e.min}$	265±5	[gkW^{-1}h^{-1}]

Simulation model of the V-12 engine was made by GT-POWER software as shown in fig. 2.

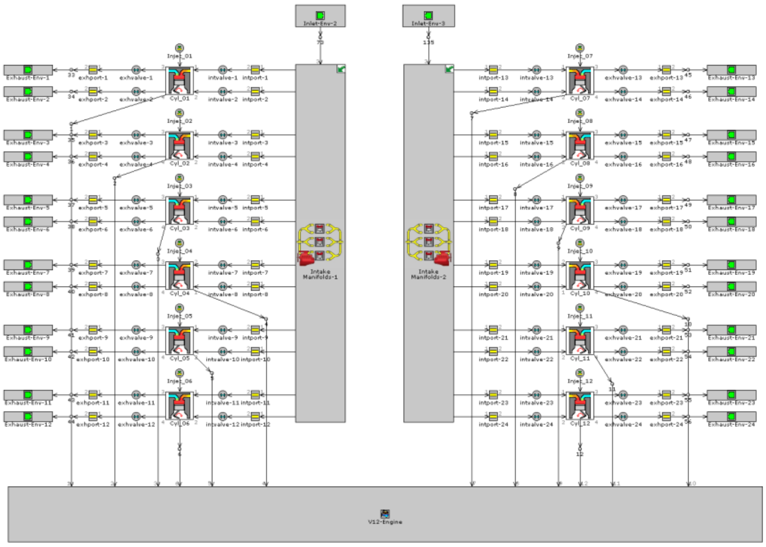

Figure 2. The V-12 engine simulation model

To validate the simulation model, the comparison results between simulation and experiment are shown in figs. 3 and 4.

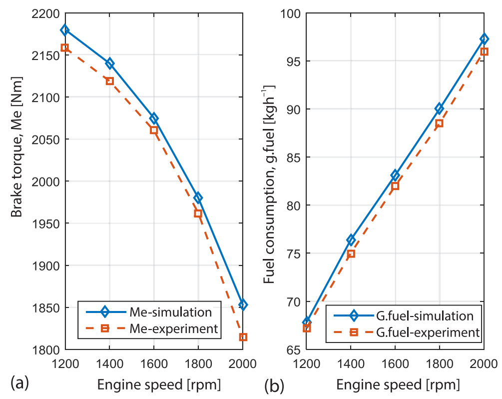

Figure 3. Brake torque and fuel consumption in an hour *vs.* simulation and experiment at 100% load and engine speeds from 1200-2000 rpm; (a) brake torque and (b) fuel consumption

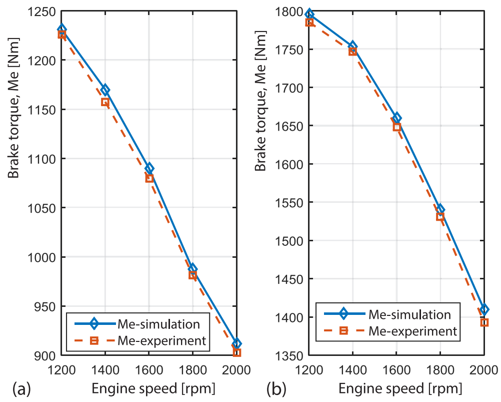

Figure 4. Brake torque *vs.* simulation and experiment at 50%, 75% load and engine speeds from 1200-2000 rpm; (a) brake torque: 50% load and (b) fuel consumption: 75% load

With the simulation and experimental results of V-12 engine on the figures from figs. 3 and 4, it can be seen that: the simulation results of the V-12 engine are quite consistent with experimental results. So, the V-12 engine simulation model made by GT-Power software has high accuracy and reliability. Therefore, this model can be used to simulate a thermodynamic cycle of the engine as well as to set up dual-fuel engine operation with ethanol fumigation. In order to establish this model, it is necessary to build an ethanol injector element into the intake manifold and set the properties of the ethanol fuel to be injected. Ethanol injector element, the model of intake manifold after the additional set-up of ethanol injectors are shown in figs. 5 and 6, respectively.

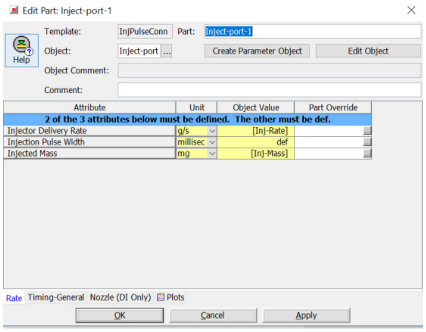

Figure 5. Input interface window of ethanol injector element

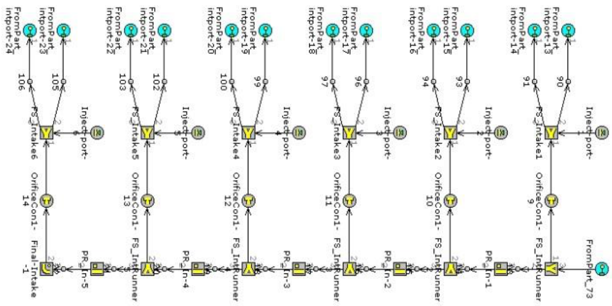

Figure 6. The intake manifold of the dual-fuel engine operation with ethanol fumigation

Results and discussion

In this study, the simulation was performed at the maximum brake power speed of 2000 rpm and at 100% engine load, the ethanol substitution percentage was defined as the percentage of the diesel replaced by ethanol fuel to reach the original diesel quantity. With the ethanol substitution percentage is 30%, firstly the author is going to survey at the different six EIT: –360, –340, –320, –300, –270, and –240 CA° before top dead centre (bTDC) as shown in fig. 7. Then the author is going to survey at the different four EDR into intake manifold as shown in fig. 8. The operating condition sets are shown in tab. 3. Simulation results of the effects EIT and EDR on engine combustion characteristic are analyzed clearly as following.

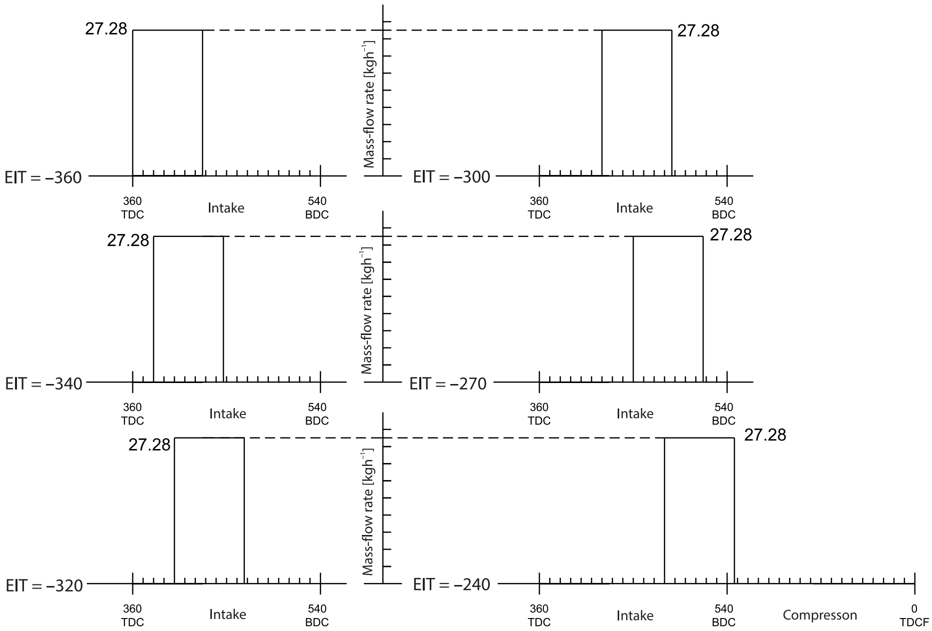

Figure 7. The different EIT

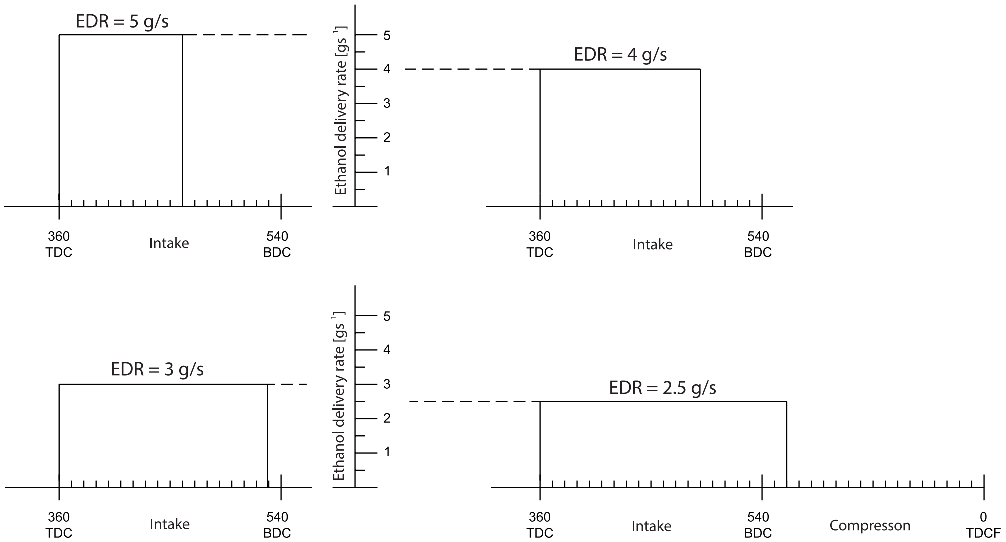

Figure 8. The different EDR

Table 3. Operating condition sets

Parameters	Value
Indicated mean effective pressure (IMEP)	8.59 bar
Engine dpeed	2000 rpm
Engine load	100%
Ethanol substitution percentage (ESP)	30%
Diesel injection timing	30 °CA bTDC
Injected ethanol fuel mass	41.67 mg
Injected diesel fuel mass	97.23 mg
Injected ethanol temperature	300 K
Injected diesel temperature	300 K

In-cylinder gas temperature

The effects of the ethanol port injection timings into intake manifold on the in-cylinder gas temperature and the combustion parameters are shown in fig. 9 and in tab. 4, respectively. Through the results shown on it, it is noticed that the temperature tendency of dual-fuel engine operation with ethanol fumigation decreased compared to when using pure diesel and more clear when the retarded ethanol port injection timing. Besides, CA at maximum temperature does have a small change at the various EIT. When the same amount of ethanol is shared with mineral diesel, if the start of ethanol port injection timing delayed, the maximum temperature declines slightly, while the combustion parameters (50% burned CA, burn duration 0-50% and burn duration 10-90%) of dual-fuel engine have a climb as compared to Diesel engine.

Table 4. The combustion parameters of the ethanol/diesel dual-fuel engine according to EIT

	EIT						
	–360	–340	–320	–300	–270	–240	Diesel engine
Maximum temperature [K]	1978	1971	1962	1949	1920	1868	2098
The CA at maximum temperature [°]	14.30	15.10	14.30	15.10	14.32	13.54	15.08
50% burned CA [°]	3.60	3.70	3.80	3.90	3.95	4.15	–1.59
Burn duration 0-50% [°]	30.6	30.6	30.6	30.6	30.6	30.6	26.4
Burn duration 10-90% [°]	36.5	36.5	36.5	36.5	36.5	36.5	31.8

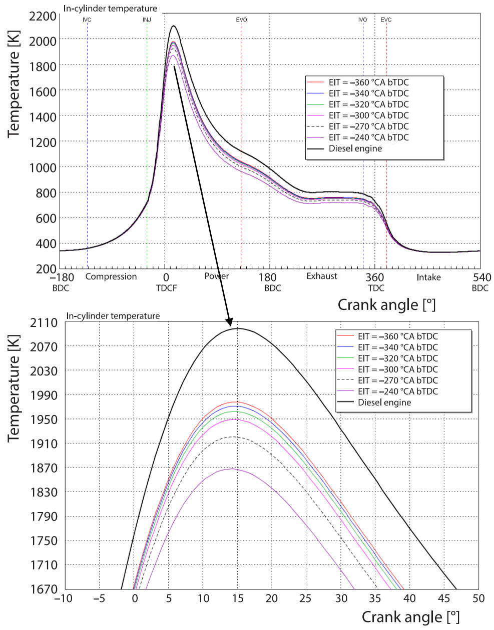

Figure 9. In-cylinder gas temperature of dual-fuel engine operation with ethanol fumigation according to the different EIT

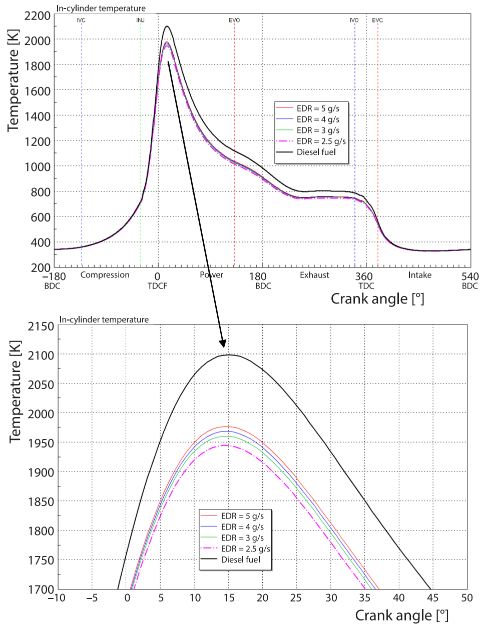

Figure 10. The in-cylinder gas temperature of dual-fuel engine operation with ethanol fumigation at the different injection rates

The effects of different EDR on the in-cylinder temperature and the combustion parameters are shown in fig. 10 and tab. 5, respectively. Through these results, it was found that com-

Table 5. The combustion parameters of the ethanol/diesel dual-fuel engine according to EDR

	EDR				
	5 [gs^{-1}]	4 [gs^{-1}]	3 [gs^{-1}]	2.5 [gs^{-1}]	Diesel
Maximum temperature [K]	1977	1968	1960	1944	2098
The CA at maximun temperature [°]	15.09	15.09	14.29	14.30	15.08
50% burned CA [°]	3.90	3.96	4.02	4.05	-1.59
Burn duration 0-50% [°]	30.6	30.6	30.6	30.6	26.4
Burn duration 10-90% [°]	36.5	36.5	36.5	36.5	31.8

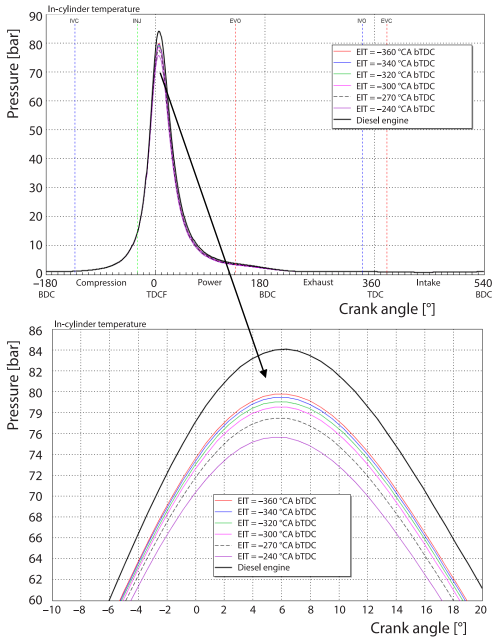

Figure 11. Variation in cylinder pressure with CA at full load according to EIT

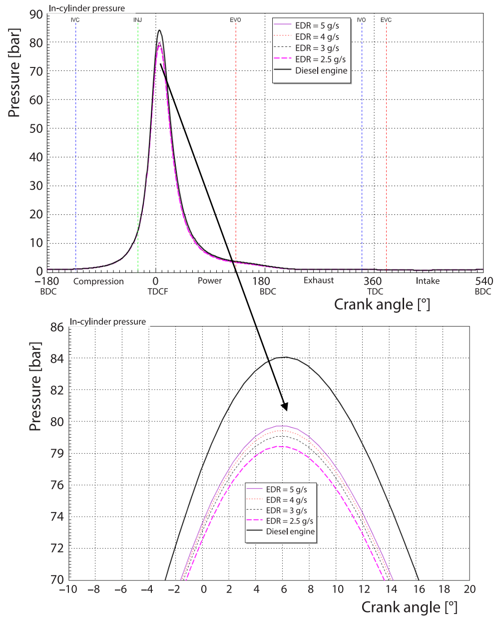

Figure 12. Cylinder pressure of dual-fuel engine operation with ethanol fumigation *vs.* ethanol injection rates under simulation condition

pared to pure diesel fuel, the in-cylinder gas temperature tendency decreases slightly. However, the lower the ethanol port delivery rate is, the more clearly the temperature changes. This can be explained by the fact that at the lower delivery rates, the EIT finished at the end of intake process even at the beginning time of the compression stroke, so the quality of evaporation, mixing with air is not good compared to the higher delivery rates.

In-cylinder gas pressure

When changing the EIT into the engine's intake manifold, it is found that they all have the same effect on the in-cylinder gas pressure as shown in fig. 11. When the EIT is later, the variation in cylinder pressure has a gradual reduction as compared to the mineral Diesel engine, the high values occur around the peak pressure, at other stages of the working cycle this decrease is relatively small. Moreover, as shown in tab. 6, the maximum pressure, CA at maximum pressure and maximum rate of pressure rise have a slight fluctuation when the EIT is retarded.

Table 6. The maximum pressure, CA at maximum pressure and maximum rate of pressure rise of the ethanol/diesel dual-fuel engine according to EIT

	EIT						
	–360	–340	–320	–300	–270	–240	Diesel engine
Maximum pressure [°]	79.78	79.47	79.06	78.54	77.49	75.65	84.07
The CA at maximum pressure [°]	5.9	5.9	5.9	5.9	5.8	5.8	6.1
Maximum rate of pressure rise [bar per °]	2.990	2.975	2.955	2.929	2.875	2.780	3.193

As shown in fig. 12, it is found that the variation in cylinder pressure of the dual-fuel engine is lower as compared to Diesel engine and concentrates most around the peak pressure point. When reducing the EDR into the intake manifold, the ethanol injection quality and the fuel-evaporation rate would be decreased compared with higher injection rates this should reduce the peak pressure. In addition, the maximum pressure, CA at maximum. Pressure and maximum rate of pressure rise have a gradual volatility when the EDR goes down, as shown in tab. 7.

Table 7. The maximum pressure, CA at maximum pressure and maximum rate of pressure riseof the ethanol/diesel dual-fuel engine according to EDR

	EDR				
	5 [gs^{-1}]	4 [gs^{-1}]	3 [gs^{-1}]	2.5 [gs^{-1}]	Diesel engine
Maximum pressure [bar]	79.74	79.43	79.07	78.44	84.07
The CA at maximum pressure [°]	5.9	5.9	5.9	5.9	6.1
Maximum rate of pressure rise [bar per °]	2.988	2.973	2.955	2.923	3.193

Heat release rate

As shown in fig. 13, the heat release rate of dual-fuel engine is lower than that of Diesel engine, especially with retarded EIT. This is because ethanol fuel has a much lower calorific value than mineral diesel fuel. In addition, with retarded EIT from –360 °CA bTDC to –300 °CA bTDC, the heat release rate lines did not have a significant difference, but after that time, the heat release rate decreased noticeably.

The effects of four different delivery rates on heat release rate under simulation condition were presented in fig. 14. In the simulation case, there are double peaks in the HRR diagrams, these maximum HRR of four EDR is lower than that of pure Diesel engine and in this case, there is a slight reduction as compared to the EIT case.

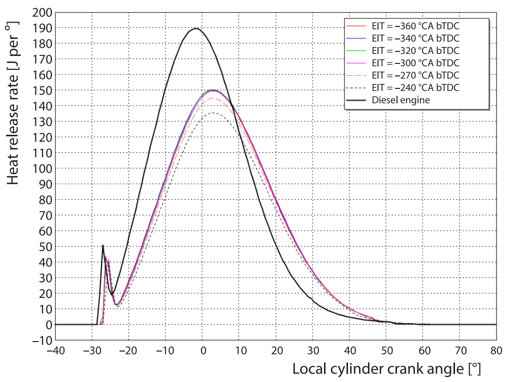

Figure 13. Variation in heat release rate of dual-fuel engine operation with CA at full load according to the different EIT

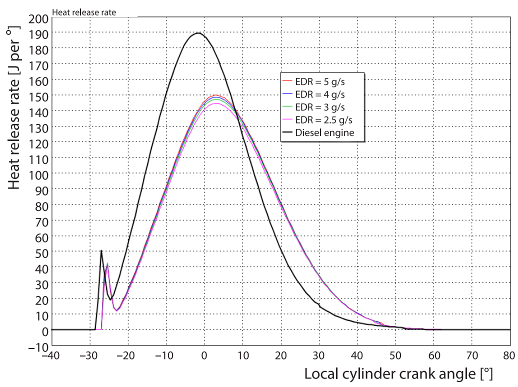

Figure 14. Variation in heat release rate of dual-fuel engine operation with CA at full load according to the different EDR

Conclusions

Based on the aforementioned simulation results on the effect of the EIT and EDR into the intake manifold on the combustion characteristic in a heavy-duty Diesel engine in dual fuel mode, the main conclusions of the investigation are summarized as follows.

- The in-cylinder temperature and pressure curves have the same tendency. Variation in cylinder pressure and temperature is a slight decline as compared to Diesel engine and rapidly drop when the retarded EIT and decreased ethanol EDR. The maximum rate of pressure rise decreases slightly. However, these changes are less than when compared to the EIT case.
- The CA at maximum temperature and pressure is almost unchanged between ethanol/diesel dual-fuel engine and Diesel engine. This can be explained by ethanol burn rate is faster than diesel fuel although the ignition delay of ethanol/diesel dual-fuel engine increases.
- The CA of 50% mass fraction burned (CA50) and the combustion duration which is the time interval elapsed or CA traversed between 10% mass burned and 90% mass burned increase slightly along with the retarded EIT and the reduced EDR.

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