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EFFECT OF EGR ON COMBUSTION AND EMISSION CHARACTERISTICS IN DIESEL BIOETHANOL DUAL-FUEL ENGINE

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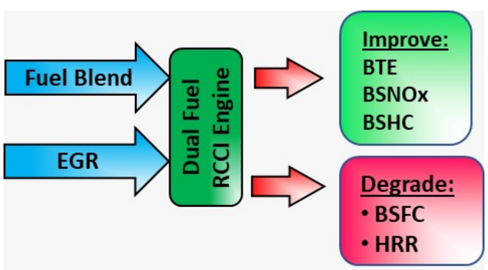
Graphical abstract



Abstract

The diminution of global fossil fuel reserves is a main concern and alternative fuel sources are needed to tackle the issue of global warming caused by greenhouse gases. Biofuels have been considered as an alternative to the depleting oil resources. This study investigates the effect of exhaust gas recirculation (EGR) on combustion, performance and emission characteristics in a single-cylinder reactivity-controlled compression ignition engine using diesel-bioethanol as fuel. A dual-fueling strategy is achieved such that bioethanol is introduced into the intake manifold using a port-fuel injector (PFI) while diesel is directly injected into the combustion chamber. Various percentage of EGR rate were considered and compared. The important parameters that were taken into consideration are: engine efficiency, combustion characteristics, and exhaust gas emissions. Experimental results revealed that the EGR variation had a noticeable effect on the engine performance, emissions, and combustion characteristics with diesel-bioethanol dual fuel operation. Introduction of EGR has effectively reduced the combustion pressure rise rate along with keeping the NOx emission within the legislation norm. Besides, the results also revealed that the diesel-bioethanol dual fuel combustion has low smoke emission across all EGR ratios. Lastly, the results showed that the engine operating under diesel-bioethanol fueling could achieve high efficiency with near zero nitrogen oxides (NOx) and smoke emissions.

Keywords: Bioethanol, Dual-fuel, Diesel, EGR, NOx .

1.0 INTRODUCTION

Exhaust gas recirculation (EGR) is now the most widely used method for NOx control strategy. The use of fossil fuels as energy source has great impact on the ecology of living beings. Most importantly, emission is one of the serious aspects to be considered as research in emission reduction has become a hot topic for past several years. But alternative fuels are now poised to make an entry into the mainstay of automobile fuels which predominantly occupied by fossil fuels. As graphically explained in Figure 1, a recent publication by a top company from energy sector has projected an increase in primary energy demand across different end-use sectors and regions. This motivates us to search for better alternatives for consumption especially in the transportation sector for a greener environmental footprint.

Bioethanol can be prepared from numerous biomass sources such as dates, grapes wastes, grape pomace, sweet potato, sweet peet pomace, rice straw, wheat straw, oil pam fronds, sugarcane bagasse, apple pomace and reed [1, 2]. Researchers have suggested using low reactivity fuels like ethanol instead of gasoline for reactivity controlled compression ignition (RCCI) which requires the use of high reactive fuel like diesel or biodiesel for extended load range operation [3]. Several studies had been conducted the influence of EGR for combustion of various fuels. The test results observed with EGR equipped engine running on diesel resulted in reduction of nitrogen oxides (NOx) lower by 25% whereas hydrocarbon (HC) and particulate matter (PM) raised by 25% [4]. When using alternative fuels test results were observed to be favorable with NOx reduction by 20% for jatropha oil methyl ester [5], 57% for pentanol fuel [6] and 50-55% for jojoba methyl ester fuel [7]. Based on the literature review conducted, this study is set to focus on the effect of varying EGR ratio on combustion, performance and emission characteristics of in a RCCI engine fueled by a combination of diesel and bioethanol.

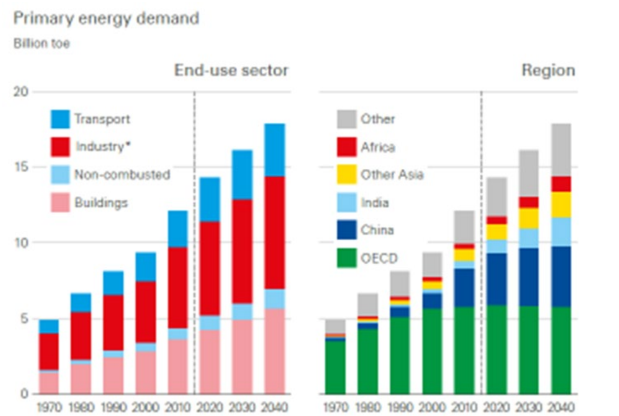


Figure 1: Primary energy demand by different end-use sectors and regions [8].

2.0 EXPERIMENTAL APPARATUS AND PROCEDURE

2.1 Test Fuels

In this study, the neat diesel fuel is procured from local fuel supplier in Malaysia. Bioethanol fuel produced by the fermentation of sugarcane with more than 99.8% purity was acquired from Chemical Industries (Malaya) Sdn. Bhd., Malaysia. The fuel properties are listed in Table 1.

Table 1 The Fuel Properties of Petroleum Diesel and Bioethanol

Properties	Unit	Diesel	Bioethanol
Kinematic viscosity @ 40ºC	mm^2s^{-1}	3.34	1.08
Density @ 15ºC	$kg\ m^{-3}$	851.9	794.0
Calorific value	$MJ\ kg^{-1}$	45.31	29.0
Flash point	ºC	71.5	14
Octane number	-	-	>120
Cetane number	-	52	5 - 8

2.2 Engine and Operating Conditions

To ensure the success of this study, the experiments were conducted on a retrofitted single-cylinder CI engine, which equipped with port fuel injector (PFI) and direct injector (DI) dual injection system. Details on the modification of the engine's fuel delivery system have been reported in previous work [9]. The engine specifications are shown in Table 2 and the schematic diagram is shown in Figure 2. During the intake stroke bioethanol is injected through PFI and premixed with air in the combustion chamber. Diesel is injected through the DI into the cylinder during the compression stroke. The PFI fuel line pressure was kept at 400 kPa using a pressure regulator. All the fuel injection parameters and engine control were managed using programmable microcontroller and graphical user interface (GUI) using LabView software.

Table 2 Characteristics of the single-cylinder engine test setup

Parameter	Specification
Engine displacement	638 cm^3
Bore x Stroke	92 x 96 mm
Ratio of compression	17.7 to 1
Peak power @ speed	7.8 kW @ 2400 rpm
Fuel injection type	Common-rail fuel injection system

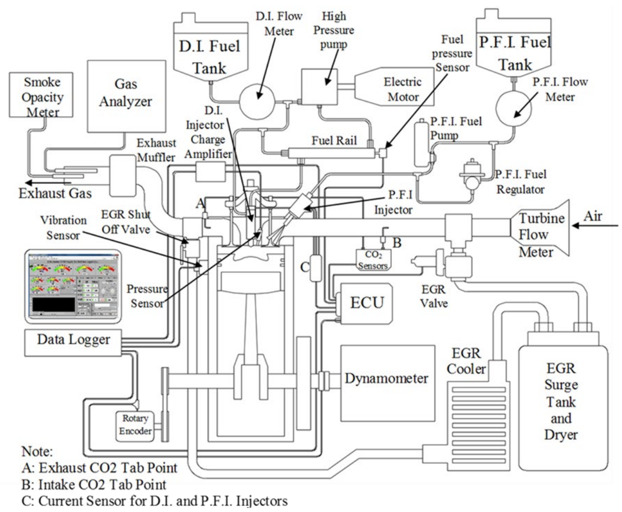


Figure 2: Schematic diagram of the experimental setup [10]

All tests were performed under a fixed speed of 1500 rpm and with 600 bar of DI fuel pressure. The engine load was maintained at around moderate load of 20 Nm for all EGR rates. This load level was chosen because under a normal operating condition, the engines used in light-duty application are typically operated under partial load condition. The performance, emissions and combustion characteristics are compared. In this study, the port injection of bioethanol fuel onto the opened intake valve timing at -360º ATDC. All experiments were conducted at a constant start of injection (SOI) timing of -11º ATDC.

The injection fuel quantity was set to 7 and 15.9 mg/stroke for diesel fuel and bioethanol, respectively across all EGR rates. Besides, due to the introduction of dual fuels, a parameter, Rp, represents the ratio of energy of the premixed fuel Qp to the total energy Qt, which can be obtained from the following equation:

$$R_p = \frac{Q_p}{Q_t} = \frac{m_p h_{up}}{m_p h_{up} + m_d h_{ud}} \quad (1)$$

where m_p is the mass of the premixed bioethanol fuel, m_d is the mass of the directly injected fuel, h_u is the calorific value and subscripts p and d denote premixed and directly injected fuel, respectively.

In this study, the bioethanol ratio was maintained at 0.6 for obtaining of total supplied fuel energy of approximately 760 J/cycle. Besides, the exhaust gas recirculation (EGR) rate was altered from 0 to 50%. The EGR calculation method was used from a study on advanced diesel combustion cycles using EGR [11]. The operating condition for this injection strategy is shown in Table 3. As can be seen, the dual-fuel engine tested without EGR was used as the baseline for comparison. The tests were conducted under steady-state conditions at room temperature with satisfactorily warmed exhaust gas and water coolant temperature.

Table 3 Experimental Conditions

Parameter	Setting
Engine speed (rpm)	1500
DI rail pressure (bar)	600
DI timing (º CA ATDC)	-11
EGR rate (%)	0 - 50
DI fuel type	Diesel
DI fuel quantity (mg per stroke)	7.0
PFI fuel type	Bioethanol
PFI fuel quantity (mg per stroke)	15.9
Premixed ratio	0.6

2.3 Instrumentation

A synchronous dynamometer of 7.5 kW capacity was fitted to the engine to provide load and maintain desired engine speed. A K type thermocouple was fixed in the exhaust downstream to acquire the exhaust gas temperature data. To measure the in-cylinder gas pressure, Kistler 6125B type pressure sensor was installed. For the purpose of acquiring the engine crank angle timing, an incremental encoder with 0.125º crank angle resolution was employed. A computer system installed with a high-speed data sampling card was used to record the cylinder pressure data and encoder signals. The data acquisition card used has a 14-bit resolution, 2MS/s, and with 4 channels of analogue input. The recorded signals were then computed and analyzed using MATLAB software. The average of collected 100 consecutive combustion cycles of pressure data was calculated. An AVL DICOM 4000 gas analyzer was employed to check various composition of exhaust emissions.

3.0 RESULTS AND DISCUSSION

3.1 Combustion Analysis

It is understood that from Figure 3, that maximum pressure rise rate (MPRR) decrease with increasing EGR rate. Researchers explain that is due to higher heat capacity of recirculated exhaust gas compared to fresh atmospheric air, flame front propagation is slower thus reducing in-cylinder temperature and subsequent pressure [12-14].

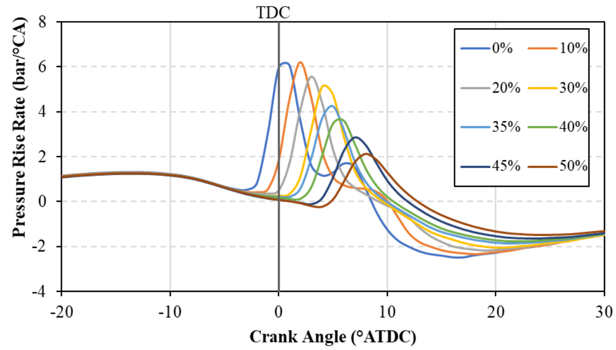


Figure 3: Influence of EGR ratio on max. pressure rise rate w.r.t crank angle.

As seen from Figure 4, there is an increasing trend on mass fraction burned (MFB) with rising EGR rate. A study reports that this is due to slower combustion process due to lower speed of flame front propagation in the combustion chamber which is enriched by recirculated exhaust gas [13].

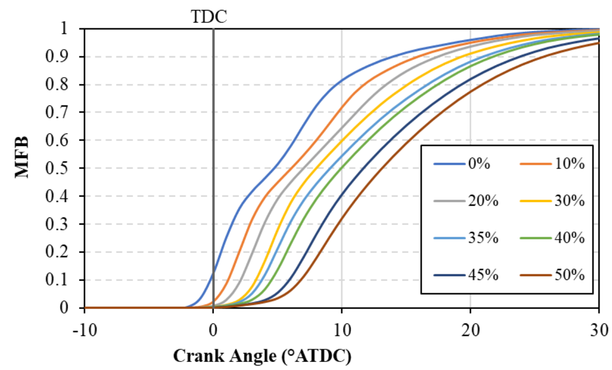


Figure 4: Influence of EGR ratio on mass burn fraction w.r.t crank angle

It is understood that from Figures 5 and 6, that in-cylinder pressure and heat release rate decreases with increasing EGR rate. This is understood that increased amount of recirculated exhaust into the combustion chamber lowers combustion quality.

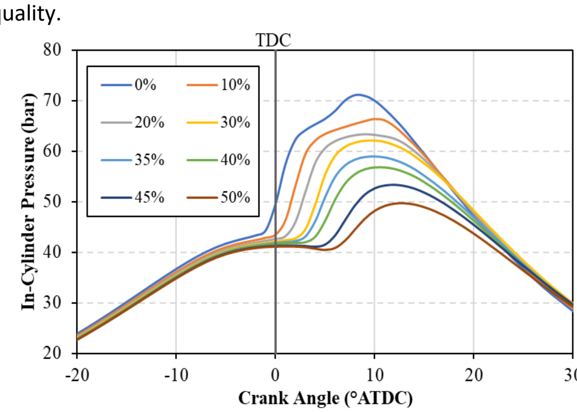


Figure 5: Influence of EGR ratio on in-cylinder pressure w.r.t crank angle.

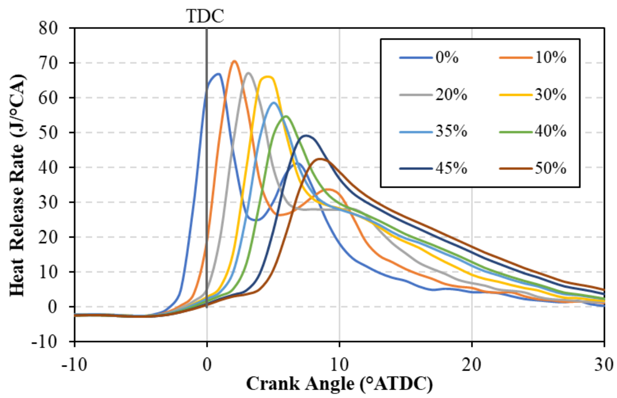


Figure 6: Influence of EGR ratio heat release rate on w.r.t crank angle

3.2 Performance Analysis

Brake specific fuel consumption (BSFC) is observed to decrease as seen in Figure 7 with higher EGR rate and then moving on to a higher trend line. The behavior of fuel consumption can be correlated with brake thermal efficiency (BTE), where at 30% EGR rate highest BTE and lowest BSFC is observed to coincide. This shows that the engine running at optimum thermal efficiency gives lower fuel consumption. Fuel consumption rises to sustain engine power delivery for the load demand and this can be due to poor combustion quality brought by increased combustion duration and lower MPRR at high EGR rate.

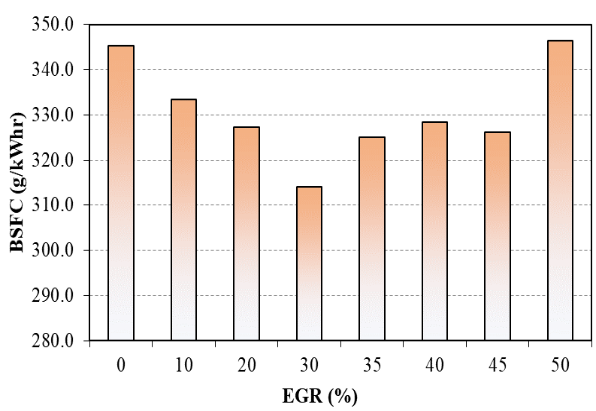


Figure 7: Influence of EGR ratio on brake specific fuel consumption

As seen from Figure 8, increasing of EGR rate improves BTE until 30% EGR rate. Further increase in EGR rate has resulted in the decrement in BTE and it can be attributed by higher EGR rate causing lower in oxygen content and thus reducing the combustion burning rate.

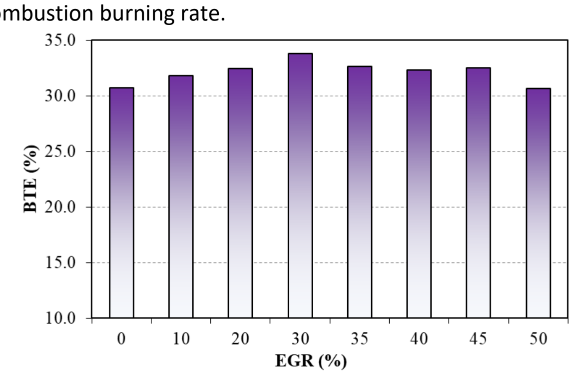


Figure 8: Influence of EGR ratio on brake thermal efficiency

3.3 Emissions Analysis

Brake specific nitrogen oxides (BSNOx) formation is noted to decrease in Figure 9 with lower in-cylinder temperature at higher EGR rate. Notably, at higher EGR rate of beyond 30%, the NOx emission was lower than the permitted EURO VI emission of 0.4 g/kWhr. This trend agrees with other study conducted by other researcher [15].

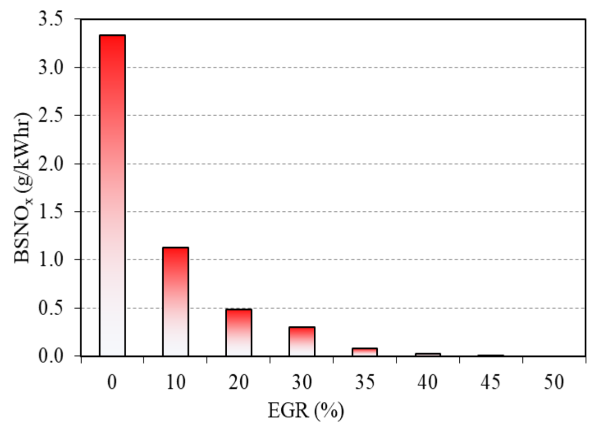


Figure 9: Influence of EGR ratio on brake specific nitrogen oxides

From Figure 10, it is noted that initially increasing of EGR rate seems not to cause much effect, at 30 – 35% EGR rate brake specific carbon monoxide (BSCO) is significantly reduced due to better combustion which is indicated by higher BTE, after which it raises again where lower presence of oxygen makes combustion unstable which forms CO and lacks excess O_2 to form CO_2. This statement is in agreement with other study with EGR [16].

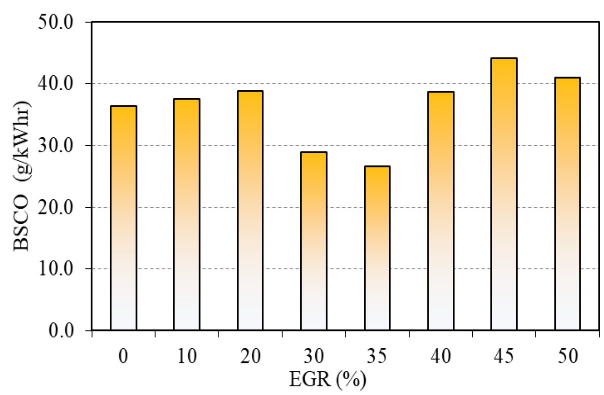


Figure 10: Influence of EGR ratio on brake specific carbon monoxide

From Figure 11, it is seen that using EGR reduces brake specific hydrocarbon (BSHC) emission which is evident at 30% EGR rate due to combustion stability leading to high BTE. Lower oxygen content at higher EGR rate increases the unburned hydrocarbon emission. Also other researcher reports that lower cetane number present in bioethanol increases auto ignition delay duration of fuel which induces flame quenching at leaner combustion zone leading formation of higher unburned hydrocarbon emission [17]. As seen from Figure 12, it is known that EGR reduces smoke at optimum BTE at 30% EGR rate. A study reports that raise in smoke opacity can be attributed to as lower oxygen concentration and higher local equivalence ratio gives improper combustion leading to soot formation [16].

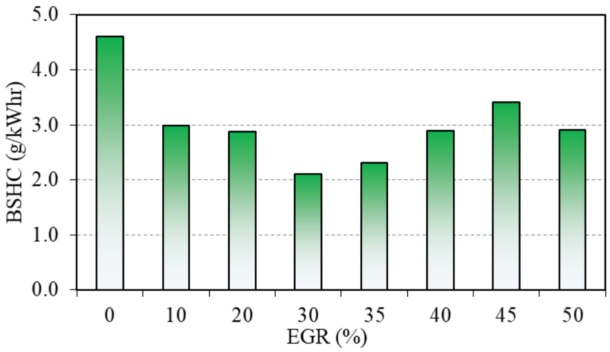


Figure 11: Influence of EGR ratio on brake specific hydrocarbon

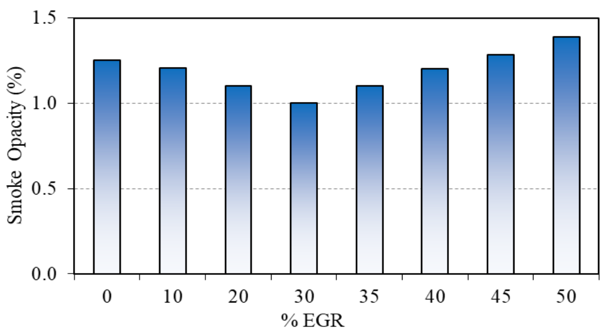


Figure 12: Influence of EGR ratio on smoke opacity

4.0 CONCLUSION

The effect of EGR was studied on a single cylinder engine running on bioethanol. The study concludes that 30% EGR rate is the identified optimum EGR rate because of high BTE, low BSFC performance wise along with lower overall emission levels. The use of bioethanol and increased EGR rate has a high tendency of triggering low temperature combustion (LTC) mode. Smoke and HC achieved lowest levels while CO is reportedly slightly higher at 30% than at 35% EGR rate. But NOx is drastically reduced well below EURO VI emission standards.

Acknowledgement

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