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Combustion and emission characteristics of a natural gas-fueled diesel engine with EGR

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ABSTRACT

The use of natural gas as a partial supplement for liquid diesel fuel is a very promising solution for reducing pollutant emissions, particularly nitrogen oxides (NOx) and particulate matters (PM), from conventional diesel engines. In most applications of this technique, natural gas is inducted or injected in the intake manifold to mix uniformly with air, and the homogenous natural gas–air mixture is then introduced to the cylinder as a result of the engine suction.

This type of engines, referred to as dual-fuel engines, suffers from lower thermal efficiency and higher carbon monoxide (CO) and unburned hydrocarbon (HC) emissions; particularly at part load. The use of exhaust gas recirculation (EGR) is expected to partially resolve these problems and to provide further reduction in NOx emission as well.

In the present experimental study, a single-cylinder direct injection (DI) diesel engine has been properly modified to run on dual-fuel mode with natural gas as a main fuel and diesel fuel as a pilot, with the ability to employ variable amounts of EGR. Comparative results are given for various operating modes; conventional diesel mode, dual-fuel mode without EGR, and dual-fuel mode with variable amounts of EGR, at different operating conditions; revealing the effect of utilization of EGR on combustion process and exhaust emission characteristics of a pilot ignited natural gas diesel engine.

1. Introduction

With the increasing concern regarding diesel engines emissions, including NOx, smoke, and PM, and the rising cost of the liquid diesel fuel as well, the utilization of alternative fuels in diesel engines seems to present attractive solution for both environmental and economical problems.

Among the alternative fuels, natural gas is very promising and highly attractive. Beside its availability in several areas worldwide at encouraging prices, natural gas is eco-friendly fuel that has clean nature of combustion. It can substantially reduce the NOx emissions by approximately 50–80% while produces almost zero smoke and PM; which is extremely difficult to achieve in DI diesel engines. It can also contribute to the reduction of carbon dioxide (CO_2) emissions, due to the low carbon-to-hydrogen ratio. In addition, natural gas has a high octane number, and hence high autoignition temperature. Therefore, it is suitable for engines with relatively high compression ratio without experiencing the knock phenomenon. Moreover, it mixes uniformly with air, resulting in efficient combustion to such an extent that it can yield a high thermal efficiency comparable to the diesel version at higher loads [1–3].

The most common natural gas–diesel operating mode is referred to as the pilot ignited natural gas diesel engine; where most of the engine power output is provided by the gaseous fuel, while a pilot amount of the liquid diesel fuel, represents around 20% of the total fuel supplied to the engine at full load operation (energy basis), is injected near the end of the compression stroke to act as an ignition source of the gaseous fuel–air mixture. The injected spray ignites several points in the gaseous fuel–air mixture, forming multi flame-fronts that travel throughout the entire mixture. The engine power output is controlled by changing the amount of the primary gaseous fuel, while the pilot fuel quantity is kept constant [4–6].

In some applications, natural gas is directly injected into the cylinder shortly before the end of the compression stroke. This technique provides better fuel economy and more efficient combustion, and maintains the power output and the thermal efficiency of an equivalently-sized conventional diesel engine [7,8]. However, direct injection of natural gas requires the development of special high-pressure gaseous injectors. Therefore, in most applications to date, natural gas is inducted or injected in the intake manifold to mix uniformly with air, and the homogenous natural gas–air mixture is then introduced to the cylinder as a result of

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Nomenclature

Latin

C_p	specific heat at constant pressure (J/kg k)
C_v	specific heat at constant volume (J/kg k)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/h)
p	in-cylinder pressure (N/m^2)
Q	heat (J)
V	cylinder volume (m^3)

Greek

γ	specific heat ratio (–)
θ	crank angle (°)
ϕ	equivalence ratio (–)

Superscripts

stoic	stoichiometric

Subscripts

D	diesel
i	intake
NG	natural gas
tot	total

Abbreviations

ABDC	after bottom dead center
A/D	analog-to-digital
AFR	air to fuel ratio (kg air/kg fuel)
ATDC	after top dead center
BBDC	before bottom dead center
BTDC	before top dead center
CA	crank angle
CAD	crank angle degree
CI	compression ignition
C/H	carbon to hydrogen ratio
CNG	compressed natural gas
CO	carbon monoxide
CO_2	carbon dioxide
COV	coefficient of variance
DI	direct injection
EGR	exhaust gas recirculation
EI	emission index
HC	unburned hydrocarbon
HHR	heat release rate (J/CAD)
NDIR	non-dispersive infrared
NO	nitric oxide
NO_2	nitrogen dioxide
NOx	nitrogen oxides
PC	personal computer
PM	particulate matters
ROPR	rate of pressure rise (bar/CAD)
TDC	top dead center

the engine suction. A typical four-stroke engine has one suction stroke per cycle while there is no suction in the other three stokes. For that reason, the measurement of the gaseous fuel flowrate becomes a point of doubt and should be emphasized and carefully treated, in order to avoid the use of inappropriate measurement technique that does not take into account that the actual gaseous fuel consumption takes place in only one stroke per cycle; i.e. the suction stroke. As the gaseous fuel should be inducted into the cylinder as a result of the engine suction only, its pressure should be kept as low as possible to prevent the flow while there is no suction. Some flowrate measuring instruments, such as rotary flowmeters and variable area flowmeters, involve a considerable pressure drop, and therefore they require the increase of gas pressure in order to overcome this pressure drop. The increase of gas pressure may lead to continuous gas supply during the four strokes while the actual consumption takes place in only one stroke. In such a case, the measured value would not represent the actual consumption. Hence, these instruments cannot be used in measuring the gaseous fuel flowrate in reciprocating internal combustion engines.

During the last years, the implementation of pilot ignited natural gas diesel engines has been investigated, experimentally and theoretically, by numerous researchers. Combustion and exhaust emission characteristics of this type of engines have been examined in various studies [9–13]. Several predictive models have also been developed in order to provide better understanding of the combustion process in gas–diesel engines and some of their performance features and emission characteristics [14–16]. Moreover, the effects of some important parameters, such as pilot diesel fuel quantity, pilot injection timing, natural gas percentage, natural gas composition, and intake air temperature have also been studied [17–21].

It has been reported that the main drawback of this operating mode, in contrast with conventional diesel mode, is the negative effect on engine efficiency, CO and HC emissions, particularly at low and intermediate loads. At high load, the improvement in gaseous fuel utilization leads to corresponding improvement in both engine performance and CO emissions, and the thermal efficiency becomes comparable to that observed under conventional diesel operation. Alternating some engine parameters, such as the increase of pilot fuel quantity and the advance of injection timing, has positive effect on engine performance, CO and HC emissions, but it adversely affects NOx emission.

In order to overcome these drawbacks while provide further reduction in NOx emission at the same time, EGR may be used. By employing EGR, portion of the unburned gas in the exhaust from the previous cycle is recirculated, and expected to possibly reburn in the succeeding cycle; resulting in a reduction in the unburned fuel with simultaneous improvement in thermal efficiency and reduction in CO. Furthermore, the application of EGR involves replacement of some of the inlet air with EGR. The consequences of this replacement include a dilution of the inlet charge and an increase in its heat capacity. These two effects lower the combustion temperature. The simultaneous reductions of oxygen concentration, combustion temperature, and flame propagation speed reduce NOx substantially. However, as NOx is reduced, PM is increased; due to the lowered oxygen concentration. When EGR further increases, the engine operation reaches zones with higher instabilities, increased carbonaceous emissions, and even power losses. [22–25].

The aim of the present work is to investigate, experimentally, the potentials of the use of EGR in pilot ignited natural gas diesel engines. A complete set of measurements is conducted for various engine operating mode; diesel, plain dual-fuel (without EGR), and dual-fuel with EGR, at different operating conditions. Detailed results are given for combustion characteristics, engine performance, and exhaust emission analysis.

2. Experimental apparatus and conditions

2.1. Experimental apparatus

The present study has been conducted on a Petter PH1W single cylinder, naturally aspirated, four-stroke, water cooled, high speed,

Table 1
Engine specifications.

Model	Petter PH1W diesel engine	
Engine configuration	Single cylinder, four-stroke, naturally aspirated, water cooled	
Bore	87.3 mm	
Stroke	110 mm	
Compression ratio	16.5:1	
Rated power and speed (B.S. continuous rating)	8.2 bhp @ 2000 rpm	
Fuel injection system	Direct injection (DI)	
Injection pressure	200 bar	
Number of nozzle holes	3	
Nozzle hole diameter	0.25 mm	
Spray angle	120°	
Valve timing	Opening	Closing
Intake	4.5° BTDC	35.5° ABDC
Exhaust	5.5° BBDC	4.5° ATDC

Table 2
Properties of diesel fuel and natural gas.

Fuel	Diesel	Natural gas
Chemical formula	$C_{10.8}H_{18.7}$	–[a]
Density (kg/m^3)	830	0.695[b]
Lowe heating value (MJ/kg)	43	49
Flammability limits (% vol.)	0.6–5.5	5–15
Laminar flame speed (cm/s)	5	34
Octane number	N/A	120
Cetane number	52	N/A
Autoignition temperature (°C)	220	580
Stoichiometric air–fuel ratio (AFR^{stoic}, kg air/kg fuel)	14.3	16.82

[a] Natural gas consists of various gas species; from which methane (CH_4) is the main constituent (methane represents about 91% (v/v) of the natural gas used in the present work). The equivalent chemical composition of natural gas may be expressed as $C_{1.16}H_{4.32}$ [26].
[b] At normal temperature and pressure.

DI diesel engine with a bowl-in-piston combustion chamber. The engine specifications are given in Table 1. Schematic diagram of the test bed is shown in Fig. 1. The engine is properly modified to suit dual-fuel operation; with natural gas as a main fuel and diesel as a pilot. The properties of both fuels are given in Table 2. The engine intake system is modified via the installation of a specially designed venturi-type gas mixer that allows the introduction of natural gas, and EGR when being employed, and mix them with the fresh air. The mixture is then induced to the cylinder as a result of engine suction. A damping reservoir and orifice system is used to measure the mass flow rate of the inlet air supplied to the engine; eliminating the pulsation effect of the engine suction. The natural gas is supplied through high-pressure (200 bar) commercial CNG bottles; typical to those used in vehicular applications. A three-stage pressure regulator is used to reduce the CNG pressure to sub-atmospheric level suitable for the engine suction. The gaseous fuel flow rate is measured by a specially-designed Pitot-tube connected to an Omega low pressure transducer, model PX277, having a maximum range of one inch of water. The pressure transducer converts the measured pressure to an analogue electrical signal, which is further manipulated via a TTi; model 1906, digital multimeter with computational functions, to be presented in the units of mass flow rate. The gaseous fuel, before entering the engine cylinder, passes through a small tank to damp the pressure fluctuation resulting from the engine suction. The pilot diesel fuel is supplied to the cylinder through the conventional diesel fuel system. A Cole–Parmer variable-area rotameter is used to measure the diesel fuel flow rate. A three-hole injector nozzle, each hole has a diameter of 0.25 mm, is used to inject the pilot diesel under a pressure of 200 bar.

The EGR system consists of piping arrangement taken from the engine exhaust pipe, EGR cooler with independent cooling circuit, moisture trap and condensate drain valve, cartridge-type soot precipitator, and control valve; to change the amount of EGR introduced to the cylinder. Schematic diagram of EGR system is shown in Fig. 2.

The temperatures of exhaust gas, cooled EGR, inlet air, and engine cooling water are measured using type-K thermocouples.

A PCB Piezotronics, model 112B10, combustion pressure sensor is used to measure the pressure inside the engine cylinder. A PCB Piezotronics, model 443A01, dual mode charge amplifier is used to condition and amplify the signal from the engine combustion sensor.

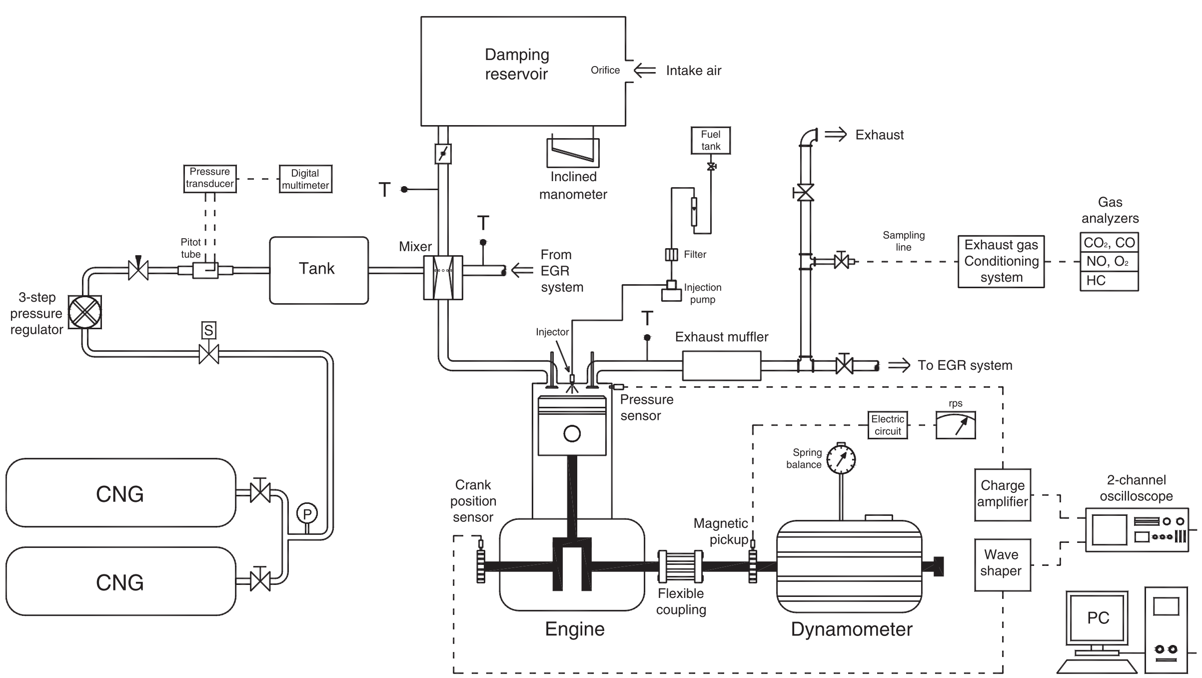

Fig. 1. Schematic diagram of the test bed.

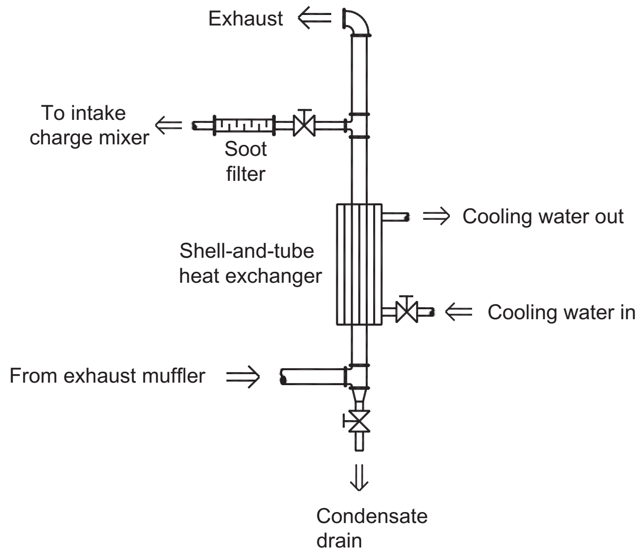

Fig. 2. Schematic diagram of EGR system.

An inductive magnetic pickup sensor having a one degree resolution is used to indicate top dead center (TDC) position and regular intervals of crank angular position as well. A wave shaper is used to manipulate the sinusoidal wave, produced by the sensor, to display the crank shaft angular location.

A Tektronix, model TDS 430A, two-channel, high-speed (400 MHz), digitizing, real-time oscilloscope, is used to present, analyze, and record the output signals from the amplifier and the shaper. A pressure/crank angle diagram was continuously displayed on the screen of the oscilloscope while the engine is running, thus enabling the effect of a change in conditions to be observed immediately. The oscilloscope is outfitted with an eight-bit analog-to-digital (A/D) converter for each channel, to allow presenting, analyzing, and recording of high-speed phenomena. The stored data is then retrieved and transferred to a PC for further computation.

An ADC multi-gas analyzer, model MGA3000, is used for measuring exhaust gas concentrations from the engine during operating conditions. Typically, NO, CO and CO_2 emissions are measured using single-beam non-dispersive infrared (NDIR) technology, while O_2 concentration is measured using paramagnetic cell technology. A CAI flame ionization detector, 600 series, is used to measure the HC emissions.

2.2. Test conditions

The experimental tests have been conducted at constant engine speed of 1600 rpm for a wide range of engine load; ranging from 43% up to 95% of the engine full load at this speed. At each load point, three operating modes have been studied: conventional diesel, plain dual-fuel (without EGR), and dual-fuel with variable amounts of EGR.

For both plain dual-fuel operation and dual-fuel with EGR, the pilot amount of the liquid diesel fuel is kept constant at 20% of the rated value under conventional diesel operating mode, while the power output of the engine is adjusted through controlling the amount of the gaseous fuel. The total equivalence ratio (i.e. that takes into account both fuels) is calculated as:

$$\phi_{\text{tot}} = \frac{\text{AFR}_{NG}^{\text{stoic}} \cdot \dot{m}_{NG} + \text{AFR}_{D}^{\text{stoic}} \cdot \dot{m}_{D}}{\dot{m}_{\text{air}}} \tag{1}$$

where $(\text{AFR}_{NG}^{\text{stoic}})$ and $(\text{AFR}_{D}^{\text{stoic}})$ are the stoichiometric air–fuel ratios (mass basis) for natural gas and diesel fuel; respectively, and $(\dot{m}_{NG})$, $(\dot{m}_{D})$, and $(\dot{m}_{\text{air}})$ are the mass flow rates of natural gas, diesel fuel, and air; respectively.

For the dual-fuel with EGR operating mode, three ratios of EGR have been examined: 5%, 10% and 20%. The percentage of exhaust gas recirculation employed (%EGR) is defined on mass basis as the percent of the total intake mixture that is recycled exhaust [27]:

$$\%\text{EGR} = \left(\frac{m_{\text{EGR}}}{m_i}\right) \times 100 \tag{2}$$

where (m_{EGR}) is the mass of the exhaust gas recycled, and (m_i) is the mass of the total intake: $(m_i = m_{air} + m_{fuel} + m_{\text{EGR}})$.

The net heat release rate (HRR) can be calculated by the traditional first law equation [27]:

$$\frac{dQ_{\text{net}}}{d\theta} = \frac{\gamma}{\gamma - 1} \cdot p \cdot \frac{dV}{d\theta} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta} \tag{3}$$

where (θ) is the crank angle (CA), (p) is the in-cylinder pressure at a given crank angle, (V) is the cylinder volume at that point, and (γ) is the specific heat ratio (C_p/C_v). The value of (γ) varies with the variation of the gas temperature inside the cylinder, and therefore it is calculated from a polynomial function of bulk gas temperature; see Appendix A (and, for more details, Ref. [28]). The net HRR represents the rate of energy release from the combustion processes less wall heat transfer and crevice flow losses. If the crevice flow losses are disregarded, the net HRR represents the combustion energy release less the heat loss to the cylinder walls; as represented by Eq. (3) [27–29]. This type of heat release model is referred to in the literature as zero-dimensional model.

For each operating point examined, five consecutive pressure–CAD diagrams have been recorded. The arithmetic average of these five curves has been taken to represent the final pressure–CAD diagram; which is used to calculate the net HRR. The net HRR curve is smoothed by arithmetic averaging of groups of every five consecutive points on the curve. More details about the methodology of determining the experimental HRR can be found in another work of the author [30].

For all experiments, the inlet air temperature is 25 °C, the engine cooling temperature is kept at 70 °C ± 3 °C, and the EGR temperature, when being employed, is kept at 35 °C.

The tests have been conducted in accordance with ISO standards. NO emission concentration is corrected for ambient humidity and temperature according to calculations presented in ISO 8178-1 Section 13.

3. Accuracy of measurements and uncertainty analysis

To ascertain the accuracy of measurements, all the instruments used are tested and calibrated, under the same operating conditions of the actual tests, before conducting the experiments. Special emphasis is given to the exhaust gas emissions measurements. All gas analyzers are purged after each measurement, and then calibrated before the next measurement using reference gases from a certified source.

To examine the repeatability of measured values, the experiments have been conducted such that five measurements of each parameter have been recorded; for each operating point. The values reported for all measured parameters, which are then used for further computations, are the arithmetic mean ones of the five measurements. The coefficient of variance (COV) for each measured value is computed, to estimate the repeatability of measurement and the accuracy of procedure. It has been found that the value of COV of each main measured parameter is less than 0.5%. Accordingly, the measurements precision is quite high.

To estimate the limiting error associated with each measured parameter, comprehensive uncertainty analysis is conducted;

Table 3
Absolute error and uncertainty of measured parameters.

Measured parameter	Absolute error	Uncertainty (%)
Inlet air flow rate	0.357 m^3/h	2.05
Diesel fuel flow rate	8.27×10^{-3} kg/h	2.7
Natural gas flow rate	1.284×10^{-2} m^3/h	2.06
Engine speed	0.25 rev/s	1
Engine torque	0.6 N m	2
EGR temperature	0.55 °C	1.57
NO emission	2 ppm	2.35
CO emission	0.002%	2.5
CO_2 emission	0.15%	3.57
O_2 emission	0.025%	0.69
HC emission	3 ppm	3.06

based on the accuracy of the instrument used and the measured value [31]. Table 3 summarizes the uncertainty analysis of the measured parameters in the present study.

4. Results and discussion

To visualize the various effects of the utilization of EGR in pilot ignited natural gas diesel engines, comparative results are given in the following subsections for different operating modes: diesel, plain dual-fuel (without EGR), and dual-fuel with variable amounts of EGR. With regard to the in-cylinder pressure and heat release rate, the experiments have been conducted for only two loads, equivalent to 52% and 87% of the engine full load at the operating speed, and comparative results are given for different operating modes. With regard to engine performance and emissions, the experiments have been conducted for all cases examined as mentioned in section 2.2, and the results for different operating modes are analyzed and presented graphically for brake thermal efficiency, total equivalence ratio, NO, HC, CO, and CO_2 emissions, and O_2 concentration.

4.1. Cylinder pressure and heat release rate

4.1.1. In-cylinder pressure and ignition delay

Figs. 3 and 4 show the pressure–crank angle degree (CAD) diagram for both conventional diesel and plain dual-fuel modes at 52% and 87% of the engine rated load at the operating speed; respectively, and the motoring pressure as well. It can be seen that at all loads, conventional diesel mode exhibits higher in-cylinder pressure and earlier start of combustion than dual-fuel mode. This is attributed to the nature of the combustion process in each mode.

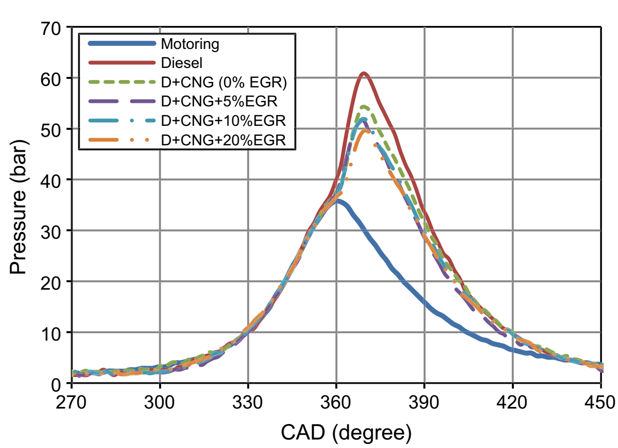

Fig. 3. Pressure–CAD diagram for different operating modes at 52% of the engine rated load.

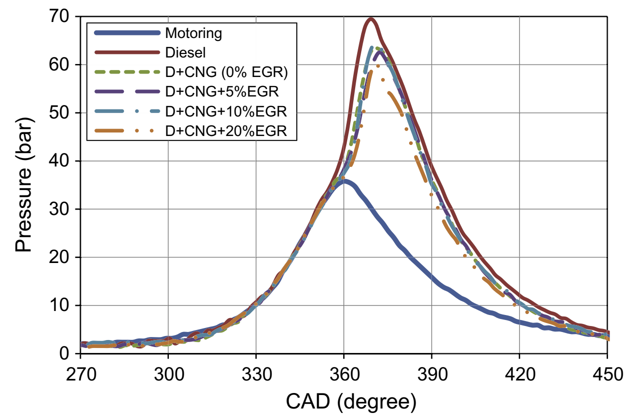

Fig. 4. Pressure–crank CAD diagram for different operating modes at 87% of the engine rated load.

Conventional diesel mode is characterized by a heterogeneous mixture, where the engine charge is only air while the diesel fuel is directly injected into the cylinder near the end of the compression stroke. Broadly, the heterogeneous mixture undergoes a non-premixed combustion process; except for the initial stage where a rapid combustion of some fuel that has mixed with air within the flammability limits during the delay period takes place rapidly in a few crank angle degrees [27]. It is well known that the non-premixed flames are not sensitive to the air–fuel ratio (AFR) value; as the combustion domain contains a variety of air–fuel ratios. Therefore, the flame is well-anchored depending on the value of the local AFR; irrespective of the value of the overall AFR, which can reach a value of 100 kg air/kg fuel. The anchored flame and the associated high combustion efficiency result in a high peak of in-cylinder pressure [27].

On the other hand, dual-fuel mode is characterized by non-premixed combustion of pilot diesel fuel, followed by premixed combustion of the main gaseous fuel; as the charge is a homogenous mixture of natural gas and air that is ignited by the injection of the pilot diesel near the end of the compression stroke. The in-cylinder conditions at that moment causes the pilot diesel fuel that has high cetane number to spontaneously ignite, providing an ignition source for the subsequent flame propagation within the surrounding gaseous fuel–air mixture. That is, there are two distinct flames, resulted from the combustion of two different fuels; each has its own properties [5,15]. Unlike non-premixed flame, premixed flame is very sensitive to AFR. In other words, the combustion efficiency for premixed flames is the best when the AFR is around the stoichiometric condition, and deteriorates as AFR moves away from that condition. Observing the values of AFR in Table 4, which are calculated from the flow rates of intake air, diesel fuel, and natural gas at the specified conditions, it is clear that the premixed combustion in dual-fuel mode suffers from very lean mixture, which is reflected negatively on combustion efficiency, and, consequently, results in lower peak value of the in-cylinder pressure.

Table 4
Air–fuel ratio (AFR) values at different operating conditions.

Engine load (% rated load)	Operating mode	AFR_{Diesel} (kg air/kg diesel)	$AFR_{Natural\ gas}$ (kg air/kg natural gas)	Peak value of in-cylinder pressure (bar)
52	Diesel	31	–	60.8
	Dual-fuel	87.5	48	54.1
87	Diesel	22.5	–	69.5
	Dual-fuel	85.5	36	63.3

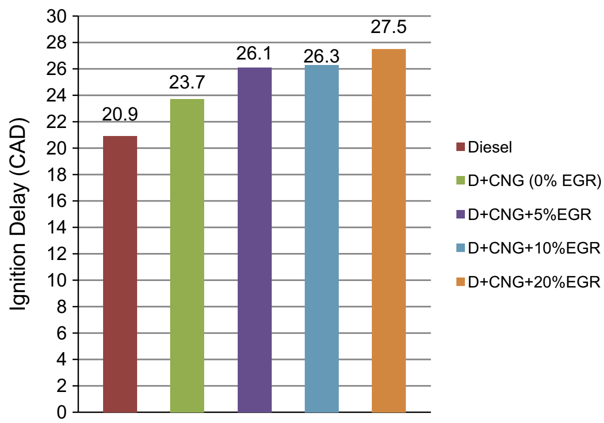

Fig. 5. Duration of ignition delay (CAD) for different operating modes at 52% of the engine rated load.

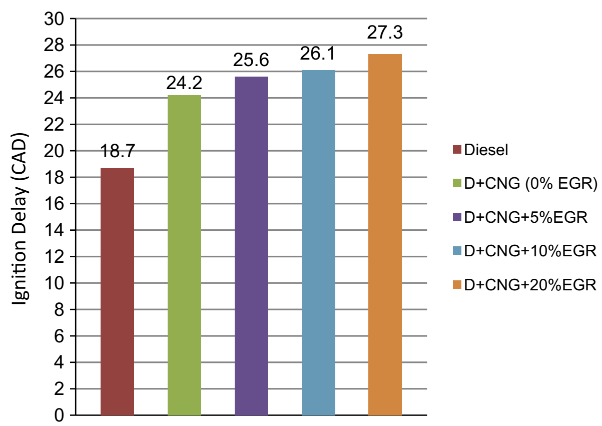

Fig. 6. Duration of ignition delay (CAD) for different operating modes at 87% of the engine rated load.

In addition, the late start of combustion; i.e. the longer delay period, of dual-fuel mode causes the whole combustion process to be shifted further into the expansion stroke. Accordingly, the pressure rise is moderated as the piston moves in the expansion stroke down away from the top dead center (TDC); increasing the volume and reducing the peak pressure.

The application of EGR to dual-fuel mode decreases the cylinder pressure. The effect is more obvious with high EGR percentages of 20%, where larger amount of O_2 is replaced by CO_2 and H_2O. This suppresses the combustion process and damps the pressure rise. Consequently, the peak pressure becomes lower.

The ignition delay period is defined as the period between the start of fuel injection into the combustion chamber and the start of combustion, identified by the change in the slope of the p–θ diagram [27]. In the present work, the start of combustion is identified as the point at which the firing diagram separates from the motoring diagram. Figs. 5 and 6 show the duration of ignition delay, expressed in degrees, for different operating modes at 52% and 87% of the engine rated load at the operating speed; respectively. Dual-fuel mode demonstrates longer delay period than conventional diesel mode. This is because the introduction of gaseous fuels to the intake air of a diesel engine increases both the physical and chemical processes of the ignition delay period. The extension of the physical process of the delay period results from the decrease in the charge temperature, the decrease in the partial pressure of oxygen, and from the absorption of some of the pre-ignition energy release; as the gaseous fuel–air mixture has a higher specific heat capacity than the pure air. The extension of the chemical process of the ignition delay results from the chemical interactions between the diesel vapor and the gaseous fuel. This chemical effect has been examined [32] on the basis of adiabatic reaction conditions at mean temperature and pressure values similar to those during the delay period in diesel engines, while employing detailed reaction kinetics for the oxidation of dual-fuel air mixture. It has been shown that the type of gaseous fuel and its concentration in the cylinder charge considerably affect the ignition delay period, while the physical properties of the mixture is maintained. In fact, the changes in the ignition delay period of dual-fuel engine show that the extension of the chemical process of the ignition delay with the admission of the gaseous fuel is the main rate-controlling process during the delay period of dual-fuel engine [32].

The application of EGR to dual-fuel mode additionally increases the ignition delay. The effect is enlarged as EGR percentage is increased. This is because the application of EGR involves the replacement of some of the intake air with combustion products. That is, the mixture is diluted and its heat capacity is increased, as a part of the oxygen available for combustion is replaced by carbon dioxide and water vapor. This will partially obstruct the combustion initiation and will absorb some of the heat relapsed by the combustion of the pilot fuel as well. As a result, ignition delay is increased.

4.1.2. Heat release rate (HRR)

Figs. 7 and 8 show the net HRR (Joule/CAD) for both conventional diesel and plain dual-fuel modes at 52% and 87% of the engine rated load at the operating speed; respectively. For both cases, it can be seen that the start of combustion in conventional diesel mode takes place earlier than that of dual-fuel mode; as revealed by the sudden rise in HRR at an earlier position. This is because conventional diesel mode exhibits a shorter ignition delay.

At engine part load; i.e. 52% of the engine rated load, Fig. 7 shows that the trend of HRR curves of both plain dual-fuel (no EGR) and dual-fuel with EGR is analogous to that of conventional diesel. That is, the HRR curve is characterized by the presence of two peaks; the first peak is for the premixed rapid combustion phase and the other one is for the mixing-controlled combustion phase. This is because at low loads, a small amount of natural gas is being utilized, and as the diesel pilot amount is quantified as a percentage of its rated value at conventional diesel operation, the pilot diesel fuel quantity in this case represents a considerable portion of the total fuel mass introduced to the cylinder. Therefore, the traditional HRR curve pattern of conventional diesel dominates.

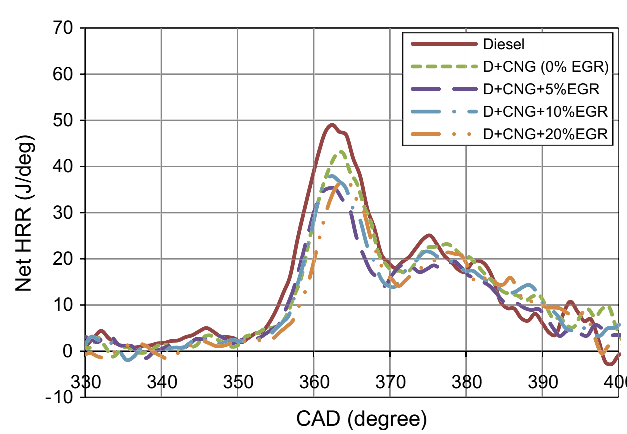

Fig. 7. Net HRR (Joule/CAD) for different operating modes at 52% of the engine rated load.

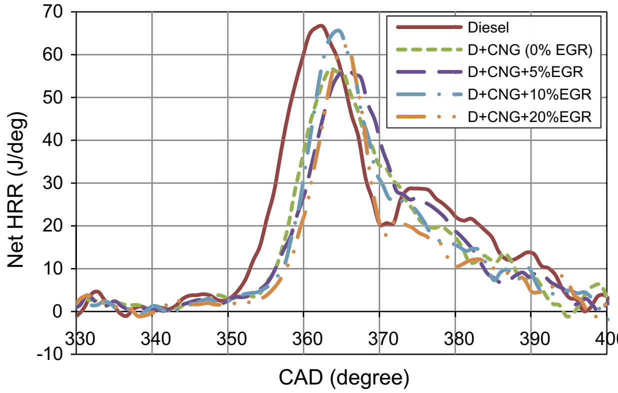

Fig. 8. Net HRR (Joule/CAD) for different operating modes at 87% of the engine rated load.

It can also be noted form Fig. 7 that dual-fuel mode exhibits lower values of HRR, compared with conventional diesel mode. This is because of the very lean mixture of gaseous-fuel and air at part load, and the associated poor fuel utilization efficiency. In such a case, large portion of natural gas escapes from the combustion process, as revealed by the low HRR. A slow burning rate of natural gas is also observed. The presence of EGR absorbs a considerable amount of the heat release, as a result of substituting some of O_2 by CO_2 and part of H_2O in the exhaust gas. This increases the charge heat capacity and dilutes the mixture [33,34]. It also suppresses the burning rate of natural gas [35]. As a result, a further reduction in heat release is observed when EGR is employed; compared with plain dual-fuel with no EGR.

In contrast, at engine high load; i.e. 87% of the engine rated load, a large amount of natural gas is being used in dual-fuel mode, while the pilot amount is kept constant. Increased mixture strength leads to an improvement in fuel utilization efficiency; as the premixed natural gas–air mixture becomes closer to the correct mixture conditions, and the burning rate of the natural gas becomes very fast and the combustion duration becomes shorter [35]. Consequently, the major portion of the combustion process is in the rapid burning phase; as illustrated by Fig. 8, where the HRR curve of dual-fuel mode is characterized by the presence of only one peak for the rapid combustion phase. However, after reaching the peak value of HRR, the curve falls down with a slower rate than that of its rise; because of the mixing-controlled combustion of the pilot diesel residue that releases some heat and hence causes the HRR curve to diverse somehow and not to fall sharply. Nevertheless, the peak of HRR in conventional diesel mode remains higher than that of dual-fuel mode. This is because conventional diesel mode utilizes a large amount of fuel at high loads; which results in a higher in-cylinder temperature, and hence, higher rates of fuel evaporation and mixing. Consequently, a larger portion of the premixed mixture in the preparation zone is formed. The premixed rapid combustion of this large amount of diesel fuel releases large amount of energy, and hence results in a higher peak value of HRR.

At engine high load, in addition, the EGR contains a sufficient amount of active radicals and unburned fuel molecules. The active radicals are expected to improve the combustion process [36,37] while the unburned fuel molecules are expected to reburn in the mixture [38,39]. The combination of these two effects causes the HRR to increase, compared with the plain dual-fuel mode, particularly with high EGR percentages of 10% and 20%. At a low EGR percentage of 5%, however, the effect of the presence of active radicals and unburned hydrocarbon is moderated by the dilution effect of EGR and by the increase in the mixture heat capacity. HRR therefore remains almost unchanged with low EGR ratios.

4.1.3. Rate of pressure rise (ROPR)

The pressure–time data is used to calculate the rate of pressure rise, or the slope of the pressure–CAD curve, at each data point. To obtain ROPR curve, the actual pressure–CAD curve is divided into several segments, and the equation of each segment is obtained as $p = f(\theta)$. The differentiation of each equation with respect to the independent variable (θ) gives the rate of pressure rise ($dp/d\theta$) at the certain segment. Complete ROPR curve is then constructed. A typical ROPR curve against CAD for a given pressure–CAD curve is shown in Fig. 9. It can be seen that the ROPR increases during compression and early stages of combustion until it reaches its highest value at a certain CAD then starts to decrease, while the pressure is still increasing till the peak pressure point. The maximum value of the ROPR data is then taken and recorded, in units of bar/CAD, to represent the combustion noise at the corresponding conditions.

Figs. 10 and 11 show the maximum ROPR that may represent combustion noise, for different operating modes, at 52% and 87% of the engine rated load at the operating speed; respectively. It can be seen that there is evident coincidence between the combustion noise and the maximum HRR. Conventional diesel mode involves an intense combustion of large amount of diesel fuel that releases a large amount of heat, as demonstrated by high peak of HRR, associated with rapid pressure rise rates. Consequently, the combustion noise of conventional diesel mode is always higher than that of dual-fuel mode.

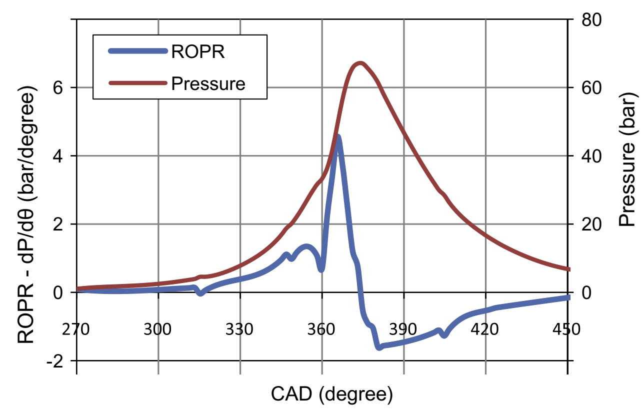

Fig. 9. Typical ROPR curve against CAD for a given pressure data.

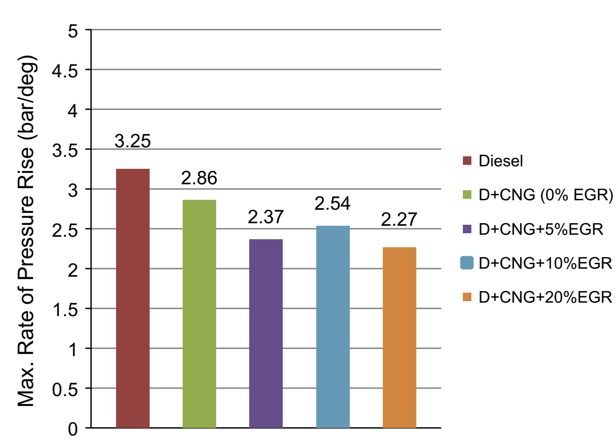

Fig. 10. Maximum ROPR (representing combustion noise) for different operating modes at 52% of the engine rated load.

Concerning the application of EGR to dual-fuel mode, it has been found that a low EGR percentage of 5% causes a reduction in the combustion noise; at all engine loads. This is attributed to the reduced oxygen concentration of the mixture (dilution effect of EGR), and to the increase in the mixture heat capacity (thermal effect of EGR). With higher EGR percentages of 10% and 20%, however, the effect of the presence of active radicals and unburned hydrocarbons (radical effect of EGR) improves the combustion conditions, as revealed by a relatively higher peak of HRR, and consequently, it generates a higher combustion noise, but its value continues to be inferior than that of the plain dual-fuel mode at part loads; as shown in Fig. 10. At high loads, however, the radical effect of EGR becomes considerable, as EGR in such conditions contains sufficient amount of active radicals and unburned fuel molecules. This is demonstrated by considerably higher peak of HRR for the EGR ratios of 10% and 20%, and the combustion noise in such cases exceeds that of plain dual-fuel; as shown in Fig. 11. However, it can be seen that the maximum ROPR (representing combustion noise) with 20% EGR is lower than that with 10% EGR, although the former contains lager amount of active radicals and unburned hydrocarbons. This is attributed to the substantial reduction in oxygen concentration in the cylinder with high EGR ratios at high loads, as the reduced oxygen concentration adversely affect the combustion process and the possibility of reburning the unburned hydrocarbon. As a result, 20% EGR exhibits lower peak of HRR and, consequently, lower combustion noise than 10% EGR.

4.2. Engine performance analysis

4.2.1. Brake thermal efficiency

Fig. 12 shows the brake thermal efficiency trends for different operating modes, throughout a wide range of engine loads; from 43% up to 95% of the engine full load at the operating speed. It can be seen that the dual-fuel engine suffers from lower brake thermal efficiency at part loads, in comparison with conventional diesel mode. This is because of the very lean mixture of gaseous fuel and air at part load and the associated poor fuel utilization efficiency. That is, in this case, only the part of gaseous fuel–air mixture that is very close to the diesel preparation zone is entrained in such a zone that subsequently burns; while the rest of the gaseous fuel escapes from the combustion process, since it forms a very lean mixture with air that cannot be burned, and goes with the exhaust. At high loads, on the contrary, a larger amount of gaseous fuel is being introduced to the cylinder, while the pilot diesel quantity is kept constant. Consequently, the mixture strength is increased; leading to an improvement in fuel utilization, as the gaseous fuel–air mixture becomes able to form a sustainable flame, and hence, a larger amount of the gaseous fuel is involved in the combustion process. In addition, the very fast burning rate of the natural gas causes a larger portion of the combustion process to take place closer to the TDC; i.e. at the beginning of the power stroke. This results in producing more power from the dual-fuel combustion at high load conditions, compared with diesel combustion at the same conditions, as the latter is characterized by a longer combustion duration where a considerable portion of the combustion takes place in late stages of the power stroke; reducing the useful power obtained. Moreover, the high peak of HRR associated with the diesel combustion increases the radiative heat loss to the cylinder walls and to the formed exhaust, due to the high emissivity rate, and this also adversely affects the brake thermal efficiency.

Concerning the application of EGR to dual-fuel mode, Fig. 12 shows that the utilization of a low percentage of EGR of 5% causes almost no change in the brake thermal efficiency at low loads. At medium loads, a slight improvement in the thermal efficiency is obtained with 5% EGR. This may be attributed to the reburning of some of the hydrocarbons that is contained in the EGR. At high loads, however, slight decrease of the brake thermal efficiency is observed, due to the reduced oxygen concentration that adversely affect the combustion process. With high percentages of EGR of 10% and 20%, a considerable improvement in the brake thermal efficiency is observed; at part and medium loads. This is because a larger amount of active radicals and unburned hydrocarbons is admitted into the cylinder when high percentages of EGR are used. Also, the partly-cooled EGR acts like a pre-heater for the intake charge; improving the combustion conditions. These effects are more prudent at high percentages of EGR. Therefore, 20% EGR exhibits a higher brake thermal efficiency at these operating conditions. At high loads, however, the amount of fuel supplied to the cylinder is increased at a higher rate, and the oxygen available for combustion gets reduced substantially. The presence of EGR further aggravates the problem and the combustion process is deteriorated. The brake thermal efficiency is therefore reduced. Moreover, the increased CO_2 concentration with high the EGR percentages increases the heat capacity of the mixture and absorbs more heat. Again, the effects become more voluminous as EGR percentage is increased, and therefore 20% EGR exhibits a lower efficiency at high loads.

4.2.2. Equivalence ratio (ϕ)

The preference of using the equivalence ratio (ϕ) to present the results of the current study rather than the use of air to fuel ratio (AFR) is due to the fact that every particular fuel has its distinctive

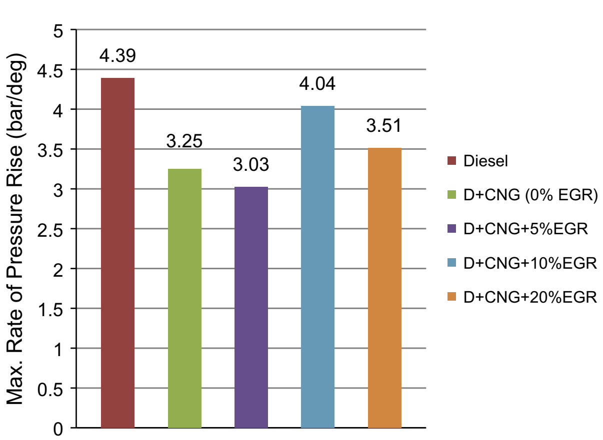

Fig. 11. Maximum ROPR (representing combustion noise) for different operating modes at 87% of the engine rated load.

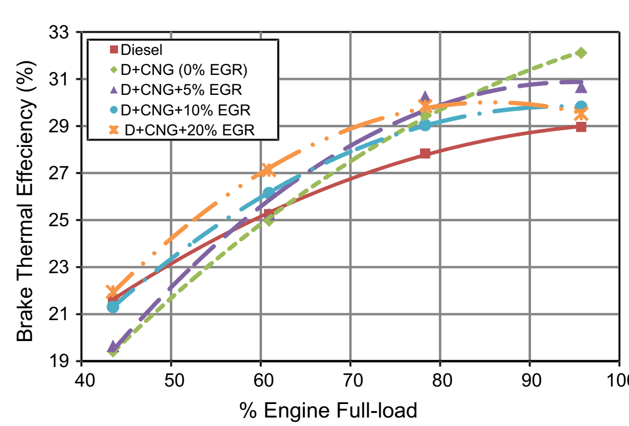

Fig. 12. Brake thermal efficiency trends for different operating modes.

stoichiometric air to fuel ratio (AFR^{stoic}); that differs from those of other fuels. The present work involves dual-fuel experiments, and therefore the use of the equivalence ratio to represent the results has the merit of taking into account the variation of AFR^{stoic} form diesel fuel to natural gas; while the use of AFR does not; as it only considers the masses of air and fuels.

Fig. 13 shows the equivalence ratio (ϕ) for different operating modes; estimated from stoichiometric combustion equations and from actual flow rates of air and fuels. It can be noted that the conventional diesel operation exhibits lower equivalence ratio than that of the plain dual-fuel operation; at low and medium load conditions. This is because the diesel combustion process involves the utilization of a large amount of excess air, due to the heterogeneous mixture. That is; a leaner mixture is required. At high load conditions, on the other hand, the plain dual-fuel mode demonstrates a lower equivalence ratio than conventional diesel mode. This is because dual-fuel mode at such conditions has lower specific fuel consumption, as revealed by the higher thermal efficiency. Although the dual-fuel operation involves reduction in the amount of air introduced to the cylinder as a consequence of the utilization of the gaseous fuel, the reduction in the total amount of fuel used is larger. As a result, the equivalence ratio becomes inferior.

The application of EGR to dual-fuel mode affects the equivalence ratio in two different manners; depending on the load conditions. At low and intermediate loads, the effect of the reduction in fuel consumption as a result of the presence of active radicals and reburning of unburned hydrocarbons predominates. Therefore, the equivalence ratio is lower than that associated with the plain dual-fuel. The effect becomes more visible as the EGR percentage is increased. At high loads, in contrast, a large amount of gaseous fuel is used, and the combustion air is reduced. The application of EGR further worsens the situation; as it replaces a considerable amount of the air available for combustion. In addition, the combustion process deteriorates, and the specific fuel consumption increases. As a result, the equivalence ratio becomes higher than that associated with the plain dual-fuel. Again; the effects are more apparent with high EGR percentages.

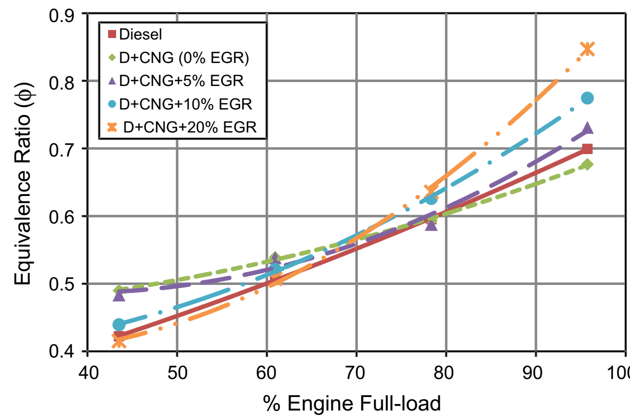

Fig. 13. Equivalence ratio (ϕ) for different operating modes.

4.3. Exhaust emission analysis

4.3.1. Nitric oxide (NO)

The formation of the nitric oxide in the combustion zone is due to two mechanisms; typically, the thermal mechanism (Zeldovich mechanism) and the prompt mechanism (Fenimore mechanism). The thermal NO formation is established by high-temperature combustion; i.e. when the combustion temperature goes higher than 1400 K. In this mechanism, the formation rate of NO increases exponentially with the increase in the combustion temperature, and vice versa. On the other hand, the prompt NO formation is established within the rich, low-temperature combustion zones; where a reasonable amount of active radicals is available.

Fig. 14 shows the nitric oxide (NO) emission for different operating modes; expressed as emission index (EI NO). It can be clearly noted that conventional diesel mode emits the largest amount of nitric oxide. This is because the conventional diesel combustion is characterized by the formation of a preparation zone, which is a thin layer of nearly stoichiometric air–fuel mixture, surrounded by a rich mixture zone. As combustion starts, the flame is anchored throughout the preparation zone at nearly stoichiometric conditions; resulting in high-temperature combustion. This is responsible for the formation of thermal NO. In turn, the combustion of the surrounding rich mixture zone is responsible for prompt NO formation. To sum up, due to the nature of diesel combustion, a large amount of NO is formed according to two distinct formation mechanisms, although prompt NO for diesel engine is very small; compared with thermal NO.

On the other side, dual-fuel combustion is characterized by the presence of two combustion configurations; typically, non-premixed combustion of the pilot diesel, and premixed combustion of the gaseous fuel. The non-premixed combustion of the pilot diesel spray is responsible for the formation of some thermal and prompt NO, but in a much smaller quantities than those associated with the conventional diesel combustion, as the pilot diesel quantity is considerably small. Besides, the premixed combustion of the gaseous fuel produces only tiny amount of NO, because of the very lean mixture that results in low-temperature combustion, while there are not any rich zones; particularly at part load. At high load, however, NO is noticeably increased in dual-fuel mode, as a result of the increased mixture strength that results in enhanced combustion at high temperature, but its value remains inferior to that of diesel combustion.

The most appraised effect of the application of EGR to dual-fuel engines is its significant contribution to the decrease of nitric oxide emissions. As widely recognized, the formation of nitrogen oxides is favored by high oxygen concentration and high charge temperature [26,27]. In dual-fuel engines, the application of EGR highly dilutes the mixture and increases its heat capacity; as a part of O_2 is replaced by CO_2 and some H_2O. Consequently, the oxygen concentration is reduced and the combustion temperature is lowered. This combined effect therefore suppresses NO formation. The higher the percentage of EGR is employed, the larger the reduction of NO is achieved.

4.3.2. Unburned hydrocarbon (HC)

Fig. 15 shows the unburned hydrocarbon (HC) emission for different operating modes; expressed as emission index (EI HC). As conceded, the variation of the unburned hydrocarbon emission in exhaust gas is consistent with the quality of the combustion process [26,27]. Observing Fig. 15, it is obvious that dual-fuel mode suffers from significantly a higher HC emission, compared with conventional diesel mode, particularly at part load conditions. This is because of the very lean mixture of gaseous-fuel and air at such conditions and the associated poor fuel utilization efficiency, since a large portion of the natural gas escapes from the combustion process; increasing the HC emissions. At high loads, the increase in the mixture strength and the improvement in the fuel utilization cause a dramatic reduction in HC emission, but its value continue to be higher than that of conventional diesel mode.

The application of EGR to dual-fuel mode reduces the HC emission levels, particularly at part load. This is because with EGR, a portion of the unburned hydrocarbon is recirculated and reburned

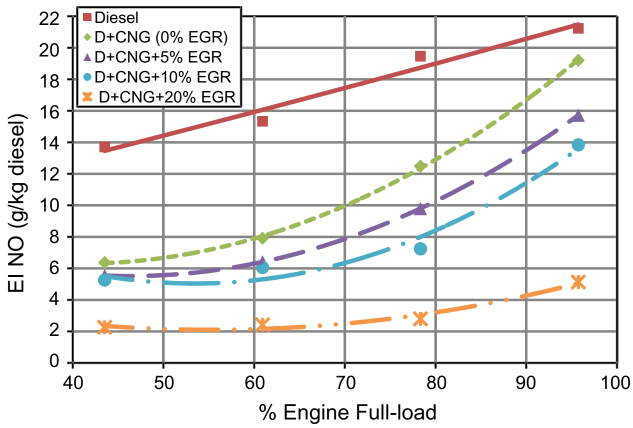

Fig. 14. Nitric oxide emission index (EI NO) for different operating modes.

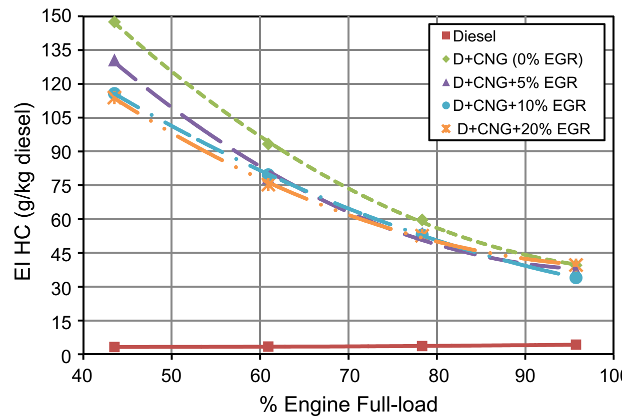

Fig. 15. Unburned hydrocarbon emission index (EI HC) for different operating modes.

in the mixture; due to the presence of a sufficient amount of oxygen in the combustion chamber at part loads. This effect is more evident with high percentages of EGR, as they contain a larger amount of unburned hydrocarbon. At high loads, however, the reduced oxygen concentration adversely affects the possibility of reburning the unburned hydrocarbon; especially with high percentages of EGR. That is, the capability of EGR to reduce the HC emissions via the reburn of some of the unburned hydrocarbon is contingent upon the excess oxygen availability in the combustion chamber. As a consequence, the effect of EGR in reducing HC emission from the dual-fuel engines at high loads is negligible.

4.3.3. Carbon monoxide (CO)

Fig. 16 shows the carbon monoxide (CO) emissions for different operating modes; expressed as emission index (EI CO). As known, the rate of CO formation is a function of the unburned gaseous fuel availability and mixture temperature, both of which control the rate of fuel decomposition and oxidation [26,27]. Observing Fig. 16, it can be clearly noticed that CO emission with dual-fuel mode is always higher than its counterpart with conventional diesel mode. This is because dual-fuel mode suffers from a poor fuel utilization that leads to incomplete combustion and high HC emission. That is, the process of fuel decomposition and oxidation is not optimized, and consequently, CO emission is increased. As the load is increased, the improvement in the combustion process reduces CO emission; as more fuel experiences a complete combustion.

The utilization of EGR in dual-fuel engines contributes to a further reduction in CO emissions, as it provides the opportunity to reburn a part of the unburned hydrocarbon, increasing the possibility of complete combustion. Also, the active radicals present in EGR improve the combustion conditions. Further, as the temperature of partly-cooled EGR is slightly higher than the atmospheric temperature, the application of EGR involves an increase in the intake charge temperature, contributing to a lower CO emission. To sum up, the addition of EGR involves the reburn of some of the unburned hydrocarbon and slightly increases the charge temperature. This combined effect causes a reduction in CO emission. The trend is almost the same for both EGR percentages of 5% and 10%; and the effect is more apparent with the latter. Although higher EGR percentage of 20% exhibits the same trend at part loads, it demonstrates a very high CO emission at high loads, as a result of a massive reduction in oxygen concentration of the charge and the associated low AFR, since a large amount of EGR is introduced in place of the intake air.

4.3.4. Carbon dioxide (CO_2)

Fig. 17 shows the carbon dioxide (CO_2) emission for different operating modes; expressed as emission index (EI CO_2). It can be

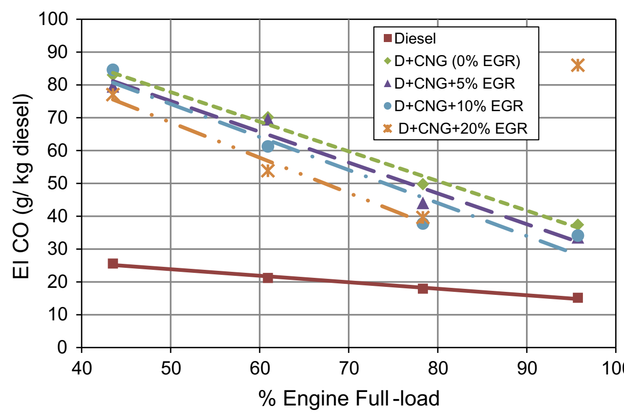

Fig. 16. Carbon monoxide emission index (EI CO) for different operating modes.

seen that dual-fuel mode emits considerably lower CO_2 emission, compared with conventional diesel mode. This is because of the clean nature of combustion of the natural gas, due to the lower carbon-to-hydrogen ratio (C/H); for one reason. The other reason may be the high HC emission of dual-fuel mode and the incomplete combustion, as revealed by the high CO emission; particularly at part loads. At high loads, however, the improvement in the combustion process causes CO_2 emission to increase, but its value remains inferior to that of diesel combustion.

The application of EGR to dual-fuel mode increases CO_2 emission at part loads. This is attributed to the increased CO_2 concentration in the intake charge as a result of the application of EGR. In addition, the improvement in the combustion process due to the presence of active radicals and reburning of some unburned hydrocarbon causes CO_2 emission to increase; and the effect becomes stronger as the EGR percentage is increased. At high loads, however, the reduced oxygen concentration as the EGR is employed adversely affects the combustion process, and therefore CO_2 emission is reduced. The reduction in CO_2 at high loads is noticeable with high EGR percentage of 20%, as the combustion deteriorates at such conditions because of the considerable lack of oxygen.

4.3.5. Oxygen (O_2)

When a sufficient amount of air is used to burn a certain amount of fuel in a CI engine, some of the oxygen content of the air is used to oxidize the fuel, while the excess oxygen goes with the exhaust as it is. In addition, the part of air away from the com-

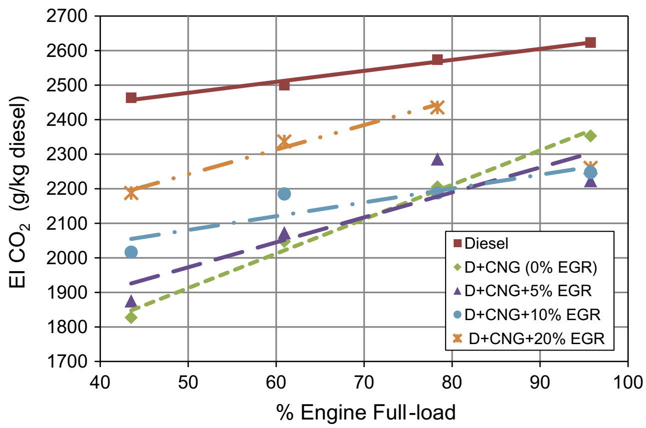

Fig. 17. Carbon dioxide emission index (EI CO_2) for different operating modes.

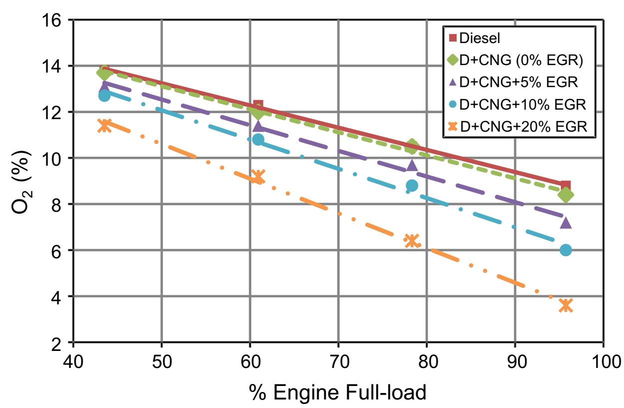

Fig. 18. Oxygen concentration (% O_2) for different operating modes.

bustion zone does not experience the combustion process at all, but goes with the exhaust as it is. The oxygen found in the exhaust gas comes from these two sources.

Fig. 18 shows the oxygen (O_2) concentration in the exhaust gas for different operating modes. For all modes, the oxygen concentration in the exhaust gas notably reduces with the increase in the engine load, as a consequence of the utilization of a larger amount of fuel that consumes a more amount of the oxygen present in the cylinder. The oxygen concentration in the exhaust gas is a direct reflection of cylinder charge composition. That is, conventional diesel mode exhibits the highest oxygen concentration in the exhaust gas, since the intake charge is only air. Part of the intake air oxygen is used to burn the diesel fuel, while the excess oxygen exits with the exhaust as it is. On the other side, dual-fuel mode involves the replacement of some of intake air by the gaseous fuel. As a consequence, the oxygen content of the charge is reduced, and therefore, the oxygen concentration in the exhaust gas will be reduced. However, the amount of the gaseous fuel that replaces the oxygen is much smaller, compared to the cylinder charge, and therefore the oxygen concentration in dual-fuel mode is comparable to that in conventional diesel mode.

The utilization of EGR in dual-fuel engines involves the replacement of an additional amount of the intake air with combustion products. That is, the oxygen available in the cylinder is considerably reduced; especially with high EGR ratios at high loads. Therefore, the oxygen concentration in the exhaust gas is substantially reduced, and it principally comes from the air that escaped from the combustion process.

5. Conclusions

The present work aims at investigating the effect of utilization of partly-cooled EGR on the combustion process and exhaust emission characteristics of a pilot ignited natural gas diesel engine. A comparative study between various engine operating modes; conventional diesel mode, plain dual-fuel mode (without EGR), and dual-fuel with variable amounts of EGR; typically 5%, 10%, and 20%, has been conducted at different running conditions. The principal findings from this study are:

(1) The cylinder peak-pressure of a diesel engine can be reduced by applying the dual-fuel strategy. The utilization of EGR further reduces the peak pressure and hence extends the engine life. The effect increases with the increase of the EGR percentage.
(2) Dual-fuel mode exhibits longer ignition delay than that of conventional diesel mode. The use of EGR further extends the delay period. The duration of the ignition delay period can be arranged in an ascending order for different operating modes as: diesel, plain dual-fuel, dual-fuel with 5% EGR, dual-fuel with 10% EGR and dual-fuel with 20% EGR.
(3) The value of the maximum rate of pressure rise with dual-fuel mode is lower than that with conventional diesel mode. The application of EGR to the dual-fuel mode causes a further reduction in that value.
(4) Dual-fuel mode suffers from a lower thermal efficiency than conventional diesel mode at part loads. However, the case is reversed at high loads; where dual-fuel mode demonstrates a higher efficiency. The use of EGR alters the thermal efficiency by increasing or decreasing; depending on the load conditions. Dual-fuel mode with EGR, in general, exhibits thermal efficiency comparable to conventional diesel mode.
(5) Dual-fuel strategy is a very promising solution to reduce NOx emissions from diesel engines. Moreover, the application of EGR to dual-fuel mode causes an additional reduction of NOx emissions significantly. The higher the percentage of EGR is employed, the larger the reduction of NOx is achieved.
(6) HC and CO emissions of conventional diesel mode are lower than those of dual-fuel mode; especially at engine part loads. Nevertheless, HC and CO emissions of dual-fuel mode reduce with the increase of the engine load. The application of EGR to dual-fuel mode slightly reduces HC and CO emissions, but their values are still considerably higher than that of conventional diesel mode.
(7) CO_2 emission of dual-fuel mode is noticeably lower than that of conventional diesel mode; at all loads. The application of EGR to dual-fuel mode increases CO_2 emission, but its value remains inferior to that of conventional diesel mode.

Appendix A

The value of the specific heat ratio (γ) used to calculate the net heat release rate (HRR) varies with the variation of the gas temperature inside the cylinder, and it can be calculated from the following equation [28]:

$$\gamma = \left(1 - \frac{R}{C_p}\right)^{-1} \tag{4}$$

where (R) is the gas constant; (J/kg k), and (C_p) is the specific heat of the gas at constant pressure; (J/kg k). The value of $\left(\frac{R}{C_p}\right)$ can be calculated from:

$$\frac{R}{C_p} = (A_0 + A_1 \cdot T + A_2 \cdot T^2 + A_3 \cdot T^3 + A_4 \cdot T^4)^{-1} \tag{5}$$

where (A_0), (A_1), (A_2), (A_3), and (A_4) are constants, and their values are [28]:

$$A_0 = 3.04473$$
$$A_1 = 1.33805 \times 10^{-3}$$
$$A_2 = -4.88256 \times 10^{-7}$$
$$A_3 = 8.55475 \times 10^{-11}$$
$$A_4 = -5.70132 \times 10^{-15}$$

and (T) is the bulk gas temperature; (K), at a given crank angle.

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