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Original Research Article

An experimental investigation on performance and emissions of a single cylinder dual fuel Diesel-CNG engine combined with EGR

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Abstract

Natural gas plays an important role as an alternative fuel for gasoline and diesel engines. It has a promising future especially with the world crisis in fuel and the lower prices of natural gas compared to the prices of gasoline and diesel fuels. It can be used as a sole fuel in spark ignition engines because it has nearly similar properties as gasoline, but in diesel engines it is used as a main fuel while the diesel fuel is injected to the combustion chamber as a pilot fuel to be an ignition source.

This paper investigates the performance of a small dual fuel engine where a single cylinder, four stroke water cooled diesel engine is converted to run on a dual fuel of diesel and compressed natural gas (CNG). An exhaust gas recycling system is used to introduce two ratios of EGR (Exhaust gas recirculation) to the engine. Effects of percentage of natural gas content and EGR ratio on engine performance and emissions are investigated for different engine loads.

Based on the obtained experimental results, it can be concluded that the total reduction in engine thermal efficiency increases as the ratio of the used natural gas increases. A maximum reduction of about 5 % in engine efficiency is recorded when 80% of diesel fuel is replaced by natural gas to deliver the same engine power. Also, NO_x emissions are decreased while CO is increased with utilization of EGR.

Key Words: Diesel, Dual fuel engine, EGR, Performance, emissions

Nomenclature

Symbol	***Definition***
T	Temperature
T	time
I	Current
V	Voltage
$\cos\emptyset$	Phase Factor
RPM	Revolution per minute
BP	Brake Power
Bmep	Brake mean effective pressure
BSFC	Brake specific fuel consumption
η_{th}	Thermal efficiency
DDF	Diesel dual fuel
A/F_a	Actual air to fuel ratio
A/F_{th}	Thermal air to fuel ratio

L.H.V	Lower heating value
H.H.V	Higher heating value
Λ	Excess air Factor
ρ	Density
$\dot{m}_f$	Mass flow rate
N.G	Natural Gas
Cp	Specific heat at constant pressure
G_i	Mass fraction
EGR	Exhaust gas recirculation
CO	Carbon monoxide
CO_2	Carbon Dioxide
O_2	Oxygen
NO_x	Nitrogen oxides

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1. Introduction

Nowadays there are tremendous numbers of internal combustion engines that are converted from using commercial liquid fuels only to use alternatives gaseous fuels. Natural Gas (NG) is used regularly as a supplement due to its availability and its inherent clean nature of combustion. Natural gas has become the second only to crude oil as a supply of energy. In general the use of alternative gaseous fuels in internal combustion engines has many advantages including their abundance, cheaper prices, adaptability as engine fuels, higher energy content, and less exhaust emissions when compared to conventional liquid fuels [1]. It has a high octane number, and therefore, it is an excellent candidate for engines with relatively high compression ratio. Moreover it mixes uniformly with air, resulting in efficient combustion and a substantial reduction of emissions in the exhaust gas [2-4]. Utilization of natural gas in petrol engines has been optimized and implemented widely particularly with taxi cars and it is quite seen in Egypt. Use of gaseous fuels in diesel engines, on the other hand, has not been optimized yet and waiting for this specific development to be utilized in many applications such as trucks, buses, and stationary power generation units or locomotives. In a natural gas-fuelled diesel engine (Dual Fuel Engine), the gaseous fuel is mixed homogeneously with the air within the intake manifold by using a modified intake mixer [5-7] and is ignited by injecting a small amount of diesel fuel (the pilot) sprayed in the usual way. The pilot fuel rapidly auto ignites and provides multiple ignition sources for flame propagation within the surrounding gaseous fuel-air mixtures. Other studies proposed that injecting the natural gas using controlled injectors can achieve better fuel consumption and higher efficiency [8, 9].

The performance of this type of converted engines which covers many aspects such as efficiency, knock, injection timing and ignition delay has attracted many researchers.

Carlucci et al. [10] reported that the operating parameters that mainly influence dual fuel combustion are those related to the diesel fuel injection and the in-cylinder bulk flow which is associated with the swirl port that leads to a more rapid development of the dual fuel combustion and to an earlier extinction of the oxidation process. Beside the previous factors it was found that injecting the gaseous fuel into the inlet manifold, varying its pressure, position and forming a homogeneous mixture has an effect on the combustion process [11].

In dual fuel engines the power output is reduced due to several factors. The most important factor is the reduction of the main fuel used in the CI engine. In most dual fuel operating systems the engine power output is controlled by changing only the amount of the primary gaseous fuel added to the air during the induction stroke. Abdullah et al. [12] found that operating the dual-fuel engine at low loads has lower thermal efficiency and higher unburned hydrocarbons emissions while the efficiency can be improved by increasing the percentage of pilot fuel, but that will lead to early knocking. Ehsan et al. [5] found that the flame propagation speed of natural gas mixture caused ignition delay and increased exhaust gas temperature, exhaust heat loss and produced less complete combustion of fuel. Tomita et al. [13] reported that at constant injection pressures, there is an optimum injection quantity for the maximum power and indicated thermal efficiency. It was also found that advancing the injection timing by 11 or 13 degrees BTDC at a specific pressure value for each case can obtain higher values of power and efficiency but it has its drawbacks on the emissions especially NO_x. In addition, increasing the advance in injection timing more than 13 degrees results in engine knocking [14]. It was reported by Wannatong et al. [15] that the use of double-pulse injection has potential to reduce the rate of pressure rise, the maximum in-cylinder pressure where the first-pulse injection should be used with a large portion (more than 70% by total diesel mass injected) and the smaller amount of the second-pulse injection which was injected closer to TDC could help promote the stable combustion and higher efficiency. On the other hand, some experimental studies [16, 17] conclude that thermal efficiencies in dual fuel operation are similar to those with normal CI engine operation (at max BMEP levels).

Simple conversion of diesel engines to dual fuel mode without controlling the injection timing will result in a longer ignition delay and reduction in peak temperature of combustion that will reduce the NO_x emissions. On the other hand, the use of controlled injection timing will reduce ignition delay, increase in peak temperature, improve efficiency and increase NO_x emissions and this can be solved by using EGR in the controlled mode.

Longer ignition delay and lower concentration of O_2 in simple dual fuel modes are the main reasons for the formation of higher CO and HC emissions. The longer ignition delay makes the combustion occurs late in the expansion stroke where there is no sufficient temperature and pressure for achieving complete combustion; so the increase in ignition delay increases the emission of CO and HC [14, 18].

Good emission control on dual fuel engines can be achieved by installing a throttle valve and using EGR to get close to stoichiometric operating conditions [14, 19].

The objective of the present work is to investigate the performance of a single cylinder diesel engine which is converted to run on dual mode of natural gas and diesel fuel by using a simple conversion mechanism without controlling the injection timing or using any sophisticatedcontrolling mechanisms. This objective has attracted some researches in the country such as Abdelaal and Hegab [20] especially in the period when the owners of small engine generators faced a hard time to save the required needs of diesel fuel to run their engines.

2. Experimental setup

The experiments were conducted on a naturally aspirated DI-single cylinder diesel engine (model SJ-65 Peter type) with a bowl-in-piston combustion chamber. Table 1 summarizes the engine specifications.

The engine is coupled to a DC generator type STC-6 with a maximum power of 6 KW to load the engine. Figure 1 shows a schematic diagram of the experimental setup.

To allow the engine to operate in dual fuel mode, a conversion kit is used where the natural gas is manifold inducted using a gas mixer (appendix A) to form a homogeneous mixture with air. The mixer was designed based on the regulations published by Sangsereki, et al [21].

A commercially available conversion kit was used to supply the natural gas at 1.2 bars in the intake manifold of the engine. The two fuels flow rates were controlled to achieve the desired ratio of diesel and natural gas. Natural gas was controlled using a regulator valve on the feeding line before the mixer. The flow rate in m^3/h was measured by a CNG glass type flow meter (model is LZJ-15F). The Diesel fuel flow rate was estimated by calculating the time required to consume a certain amount of diesel fuel.

The engine's output signals for RPM, temperatures, and cylinder pressure were coupled to a suitable DAQ system. The engine emissions of CO and NO_x were measured using a gas analyzer type MR 2800 P Series.

Properties of fuels used in the current study are shown in appendix B.

Table 1. Engine specifications

Model	***Specifications***
Configuration	Single cylinder, vertical, compression ignition, and four stroke engine.
Bore	85mm
Stroke	110mm
Compression ratio	17:1
Cooling system	Water cooled
Rated output	3.8 KW at 1500 rpm
Number of valves	2
injector	Single injector with three holes who delivers a pressure of 200 bar
Injection pressure	200 bar

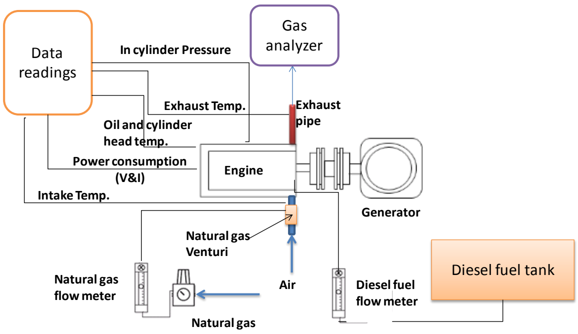


Fig. 1. Schematic diagram of the test rig

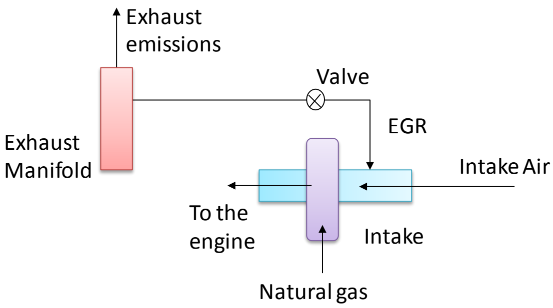


Fig. 2. EGR circuit in the engine

2.1. Mass flow rate of EGR based on the heat balance

The EGR flow rate was measured by applying a heat balance methodology where a heat balance equation between air, EGR, natural gas and the total mass of the mixture as shown in Figure 3 was developed for this purpose as follows:

$$\dot{m}\, h_1 + \dot{m}_{EGR} h_2 + \dot{m}_{N.G} h_3 = \dot{m}_{total} h_4$$

Where h is the enthalpy, m is the mass flow rate.

$$\dot{m}_{total} = (\dot{m}_{air} + \dot{m}_{EGR} + \dot{m}_{N.G})$$

By assuming the same reference temperature for all gases:

$$h = Cp.T$$

$$m_{air} Cp_{air}.T_1 + \dot{m}_{EGR} Cp_{EGR}.T_2 + \dot{m}_{N.G}\, Cp_{N.G}.T_3 = \dot{m}_{total} Cp_{av}.T_4$$

As all temperatures are known then the$\dot{m}_{EGR}$ can be obtained.

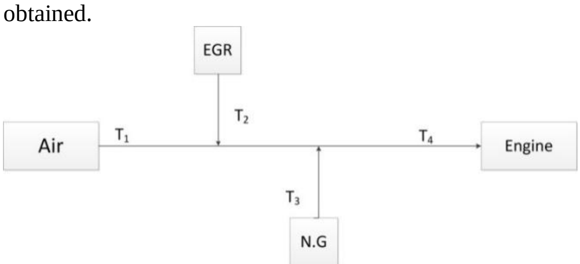


Fig. 3.Flow chart for the gases that feed the engine

2.2. Errors and uncertainty analysis

Based on the individual errors for basic measurements, Table 3 shows the maximum errors for different parameters by utilizing the propagating error analysis [22].

Table 3 Maximum error of measured and calculated parameters

Parameters	Error (%)
Diesel fuel flow rate (kg/h)	1
Natural gas flow rate (kg/h)	3
Brake power (BP)	5
Brake thermal efficiency (BTE)	1.6
BSFC	1.8
NO emission	5
CO emission	5
O_2emission	1

3. Results and discussion

Two different methods have been considered to estimate the total fuel consumption in the present study; based on mass fraction and based on thermal equivalent diesel fuel consumption. In the first method the ratio of diesel fuel to natural gas was estimated based on the absolute mass fraction of both fuels while in the second method the heating values of both fuels are considered to obtain the diesel equivalent consumption of natural gas based on energy content. In this work, the second method was adopted. The difference in heat capacities of gas mixtures is also considered.

The engine was tested and the performance was analyzed through two stages; firstly as a dual fuel engine without EGR and secondly as a dual fuel engine with different EGR ratios.

All experiments were conducted at constant speed of 1350 rpm.

3.1. Performance of the engine using dual fuel without EGR

In this stage of study the engine was tested as a dual fuel engine at different Diesel/NG ratios.

Figure 4shows the brake specific fuel consumption (BSFC) based on diesel thermal equivalent as mentioned earlier. The figure shows clearly an increase in the BSFCas the natural gas fraction increased. This can be explained by the higher ignition delay of natural gas compared to diesel fuel [5, 23] which results in a late peak combustion pressure obtained in the expansion stroke. In addition to that it is well known that instead of having a spontaneous combustion of diesel fuel, a flame propagation speed has to be considered in case of NG. Accordingly, for mixtures with higher ratios of natural gas it is expected to have slower speeds. This will lead to delay the energy released from the combustion of natural gas towards the end of expansion stroke which results in lower pressures and less outputs. This is supported by figure 5 where the peak cylinder pressure is reduced as natural gas is used. When using the same amount of fuel consumption (0.76 kg/h) with different percentages of (diesel/natural gas), different values of BMEP were generated (3.6 bar at 100% diesel, 3.2 bar at 60% diesel and 2.8 bar at 30% diesel). The fuel consumption was measured in diesel thermal equivalent values. Accordingly, the fuel consumption has to increase to maintain the same output power.

Figure 6 shows a maximum reduction of 5-8 % in the absolute engine efficiency over the entire window of applied loads and NG/diesel ratios.

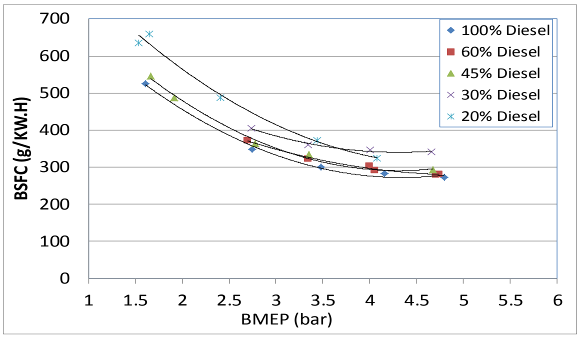


Fig. 4 Specific fuel consumption at different loads and different NG/diesel ratios

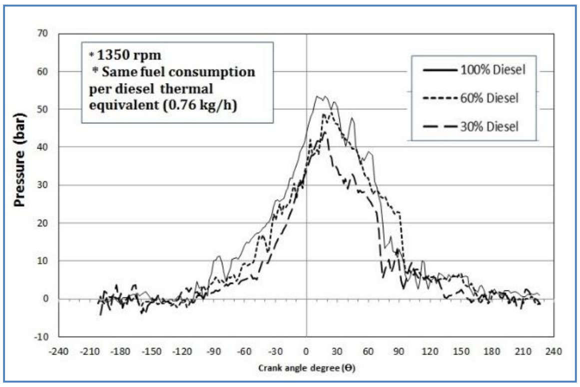


Fig.5 In-cylinder pressure variation with crank angle degree at two different NG/Diesel contents compared to 100% diesel fuel

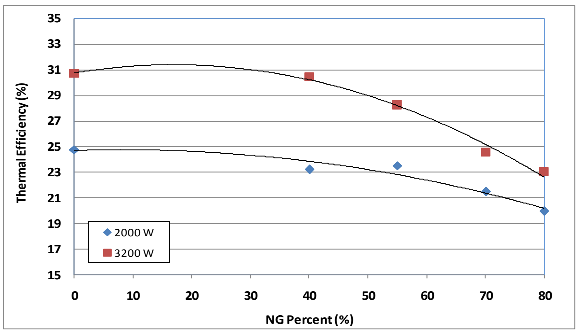


Fig. 6 Thermal efficiency at different NG percentages and selected two engine loads of 2 kW and 3.2 kW

Figure 7 presents the variation in lambda values at different diesel/NG percentages. As shown in the figure, the values of lambda decrease as the NG contents increase due to replacing part of the intake air by NG which reduces the volumetric efficiency.

As the only way to control the output power is the amount of fuel, so the fuel consumption is expected to increase as the engine is loaded. On the otherhand, the mass flow rate of air is constant as a result of running the engine at the same speed of 1350 rpm. Accordingly, the actual engine air-fuel ratio $(A/F)_a$ decreases and the engine tends to run richer which reduces the gap between running on 100% diesel and/or running on 20% diesel as shown in figure 7.

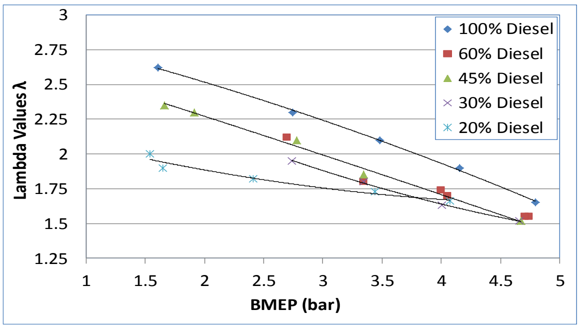


Fig. 7 Lambda values at different loads and different NG/diesel ratios

3.2. Performance of the engine using dual fuel with EGR

The following five major factors that impact the combustion reaction have to be considered when EGR is introduced:

- Higher intake temperature which enhances the reaction.
- Small percent of radicals which are important for starting the reaction.

- An increase in the specific heat of the mixture when EGR and NG contents increase which reduces the combustion temperature and consequently impact the reaction negatively.
- Non reactive gases (CO_2, H_2O, N_2) which damp the reaction
- A reduction in O_2 contents due to the reduction in the volume availability.

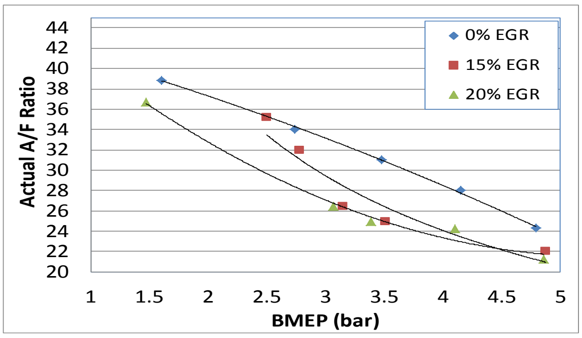
In this stage of study the engine was running on dual fuel mode using three different NG/Diesel ratios (0/100, 30/70 and 40/60) along with two different percentages of exhaust gas recirculation (15, 20 % EGR).

The addition of EGR in the intake manifold reduces the amount of air as indicated by the amount of air-fuel ratio in figure (8a) when the engine was tested using diesel fuel only. The same results were obtained in case of duel fuel mode with higher reduction in the air availability when NG is used as shown in figure (8b)

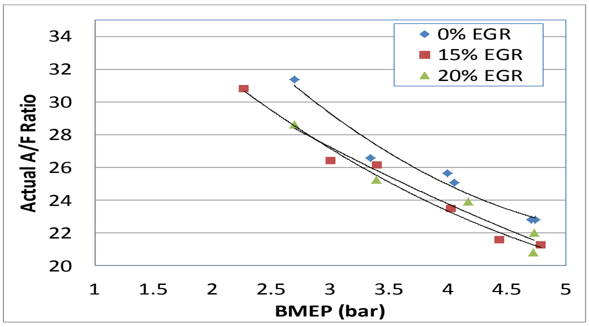
There is also a decrease in the fuel consumption with the EGR operation. This can be explained by the higher intake temperatures and by the availability of small portion of radicals when EGR is used as indicated earlier. Figure 9 shows a maximum increase of about 26 and 40 ºC in the intake temperature when 15 and 20 % EGR were used respectively at high load.

Another reason can be considered is the availability of unburned fuel in the EGR which produces some heat and saves some of the fuel as indicated in references [24, 25].

Figure 10a for a medium load shows that there is a positive effect of increasing the EGR percent for different NG percentages. For high load on the other hand, figure 10b shows some increase in fuel consumption (from 0% to 15% EGR) followed by a decrease (15% to 20%). This can be explained by the combination of the five parameters mentioned earlier with opposing effects.



(a) 100% Diesel



(b) 60 % Diesel

Fig. (8) Effect of engine loads on actual A/F ratio at different EGR ratios for (a) 100% diesel fuel, and (b) 60 % diesel fuel

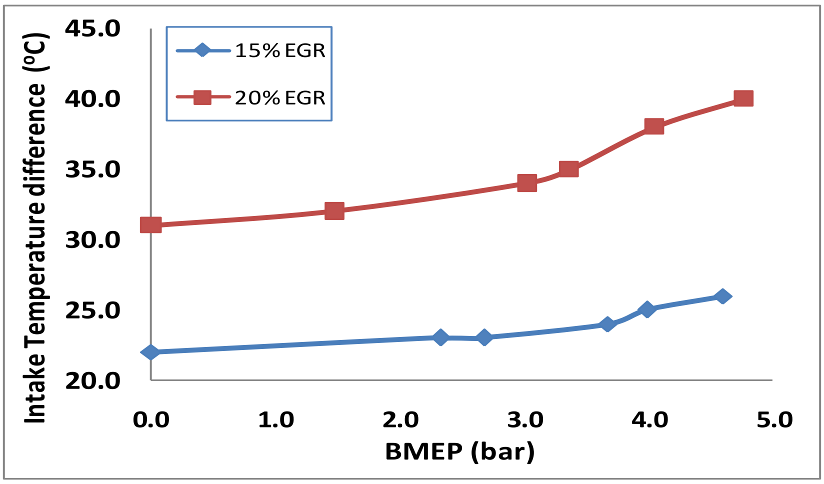
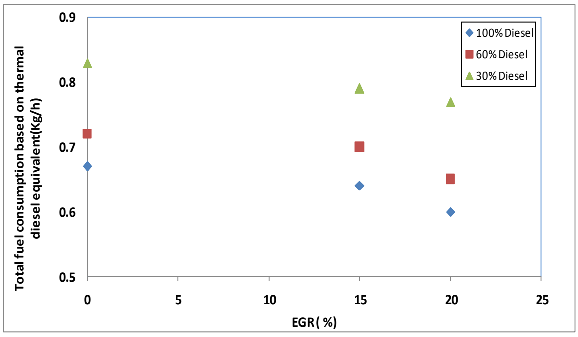
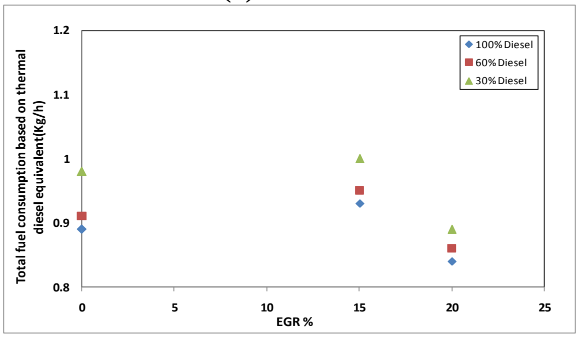


Fig. 9 Average increase in intake temperature at 15 and 20 % EGR relative to 0 % EGR when diesel was used by 100 %.



(a) 2000W



(b) 3200W

Fig. 10 Total fuel consumption based on thermal diesel equivalent vs. EGR percentages for different diesel contents at (a) 2000W, and (a) 3200W engine loads

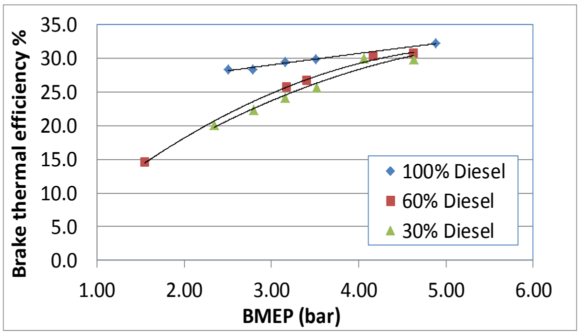
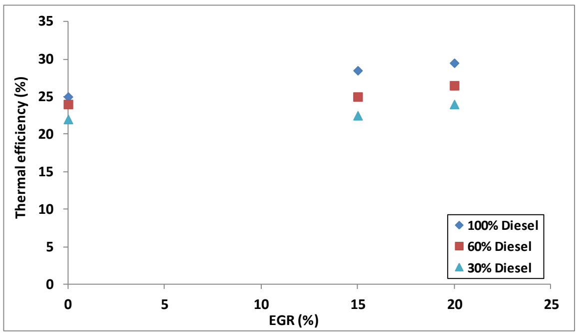
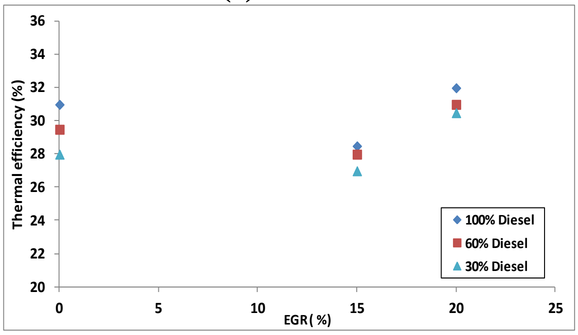


Fig. 11 A comparison of engine thermal efficiency for two NG contents to original Diesel operation at 15 percent injected EGR.

The engine thermal efficiencies for two different NG contents were compared to original diesel operation at 15 percent EGR. Figure 11. More than 28 % reduction in the thermal efficiency when NG was used at relatively low loads which reduced to the half (about 14 %) at high loads. As the natural gas content increases the efficiency decreases while EGR on the other hand has an effect on the efficiency as a reflection of its impact on fuel consumption discussed earlier. Figure 12a illustrates this point where the efficiency slightly increased with EGR for an intermediate load. Figure 12b for a high load on the other hand, showed some decrease followed by an increase in efficiency.



(a) 2000W



(b) 3200W

Fig. 12 Thermal efficiency vs. EGR percentages for different diesel contents at (a) 2000W, and (b) 3200W engine loads

3.3. Engine exhaust emissions for dual fuel operation with and without EGR

Dual fuel engines have a role in solving emissions problem in diesel engines while retaining the performance of these engines almost the same. Previous researches indicated that dual-fuel engines produce lower particulate matter and lower nitrogen oxides than diesel engines [2, 5].On the other hand simple conversion to dual fuel mode without controlling the injection timing increases the CO and HC emissions [19, 26]. The objective of the present work is to check such results and try to explain the reasons and expand that to cover EGR role. Thus, a recommendation of a set of engine operating conditions may be suggested to get optimum performance.

3.3.1. CO Emissions

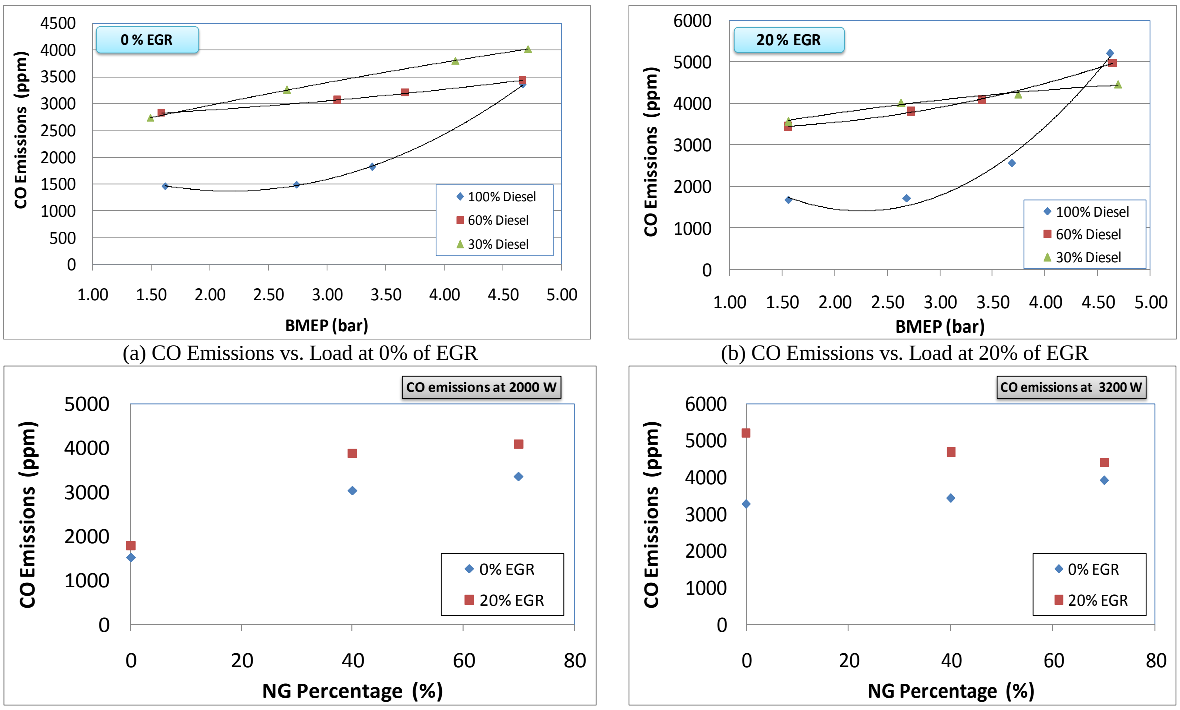
Figure 13 shows the CO emissions from the tested engine at different NG/Diesel contents and with/without EGR. As discussed earlier the increases in NG fraction and EGR generally reduce the rate of reaction and thus increase CO and HC emissions. The longer ignition delay makes the combustion occurs late in the expansion stroke; a point at which the temperature is not high enough for complete combustion. Similarly the reduction in O_2 concentration in the intake manifold reduces the rate of reaction and thus produces higher CO and HC emissions.

Unlike figure 13c at 2 kW load, figure 13d at high load of 3200 W, CO emissions tend to decrease with adding EGR. This may be attributed to the previously mentioned positive effects of the EGR being dominant than the negative ones.

By analyzing the premixed lambda values, it was revealed that there is excess air beyond the flammability limit in this case as shown in figure 14.This might be taken as another reason for the high CO. Obviously by increasing the NG portion, the lambda values decrease as shown in the figure.

This can be solved with installing a catalyst in the exhaust system so it can convert most of CO and HC emissions to CO_2 and H_2O [14, 18, 19]. As shown from figure 15, the oxygen percent in the exhaust can be used with the catalyst for oxidizing the CO and HC.

The oxygen content tends to decrease at higher loads because of more fuel is injected. With higher ratios of NG the oxygen availability tends to be lower because of the volume occupied by NG as shown in figure 15. The case will be worse in case of EGR due to the more volume occupation by EGR which decreases the total amount of oxygen available for combustion.



(a) CO Emissions vs. Load at 0% of EGR

(b) CO Emissions vs. Load at 20% of EGR

(c) CO Emissions vs. NG percent (with/without) EGR at 2kW

(d) CO Emissions vs. NG percent (with/without) EGR at 3.2kW

Fig. 13 CO emissions at different NG/Diesel contents and with/without EGR.

3.3.2. NO_X Emissions

Figure 16a shows that as the load increases NO_x increases because of the higher consumption of fuel which increases the peak temperature. It is clear that the increase of diesel replacement by NG decreases the NO_x. The reduction of NO_x formation is a result of the reduction of peak temperature due to ignition delay.

figure 16b shows the effect of using 20 % EGR on NO_x emission at different NG/Diesel ratios. The NO_x emission decreases especially at low loads. The effect of EGR on NO_x reduction is almost due to large ignition delay, increased heat capacity and dilution of the intake charge with inert gases. The ignition delay hypothesis asserts that because EGR and dual fuel mode cause an increase in ignition delay and thus decrease the combustion temperature.

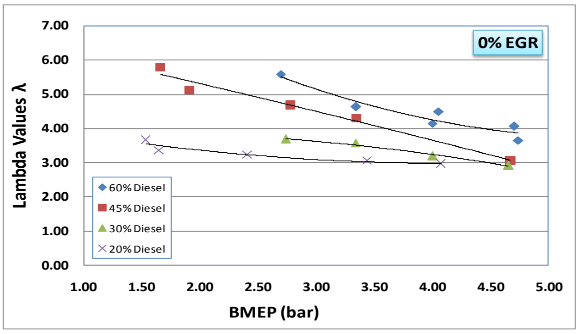


Fig. 14 Premixed lambda values for different Loadsand NG/Diesel ratios at 0 % EGR

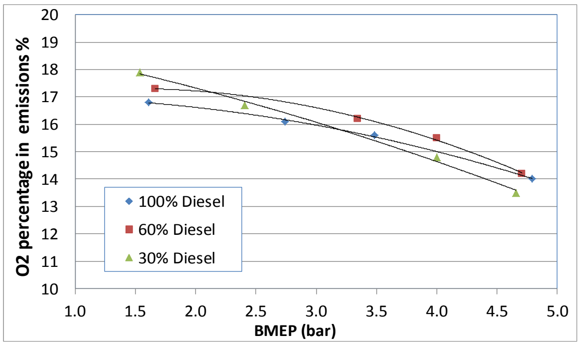
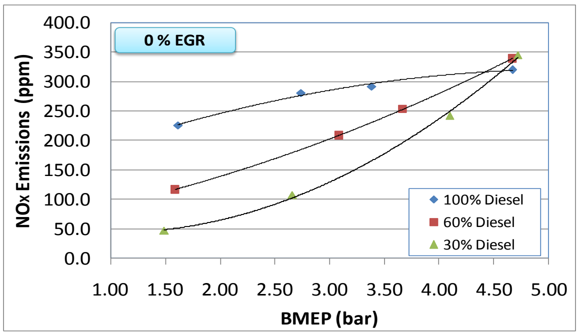
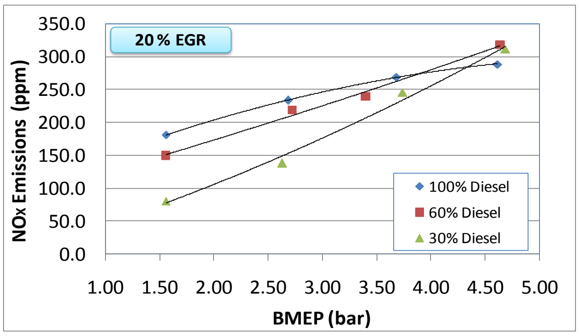


Fig. 15 O_2 Percentage vs. Load at 0% EGR

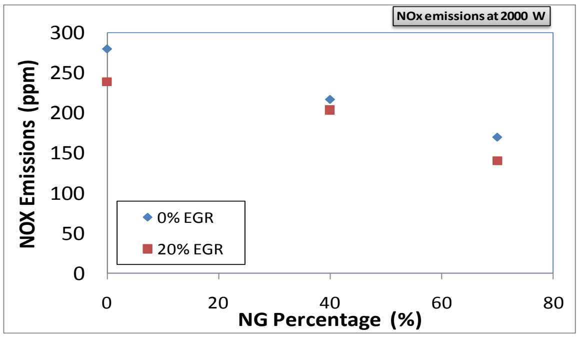
The addition of the inert exhaust gas into the intake increases the heat capacity (specific heat) of the non-reacting matter present during the combustion and thus leads to lower peak combustion temperature. According to the dilution theory, the effect of EGR on NO_x is caused by increasing amounts of inert gases in the mixture, which reduces the adiabatic flame temperature [27]. Installing a throttle with EGR to get close to stoichiometric operating conditions and then a catalyst will make a good emission control on dual fuel engines. Excess air has to be avoided, otherwise NOx formation quickly increases. On the other hand, rich mixtures have to be avoided; otherwise PM emissions (besides HC and CO) may increase [19]. Figure 16c shows that NO_x decreases as NG contents increase at 2KW, medium load. Also, NO_x decreases with EGR due to the previously mentioned reasons. While at high loads, 3.2KW as shown in figure 16d, NO_x emission tends to increase with increasing NG contents due to higher combustion temperature that leads to NO_x formation. The effect of EGR is the same on NOx in this case.



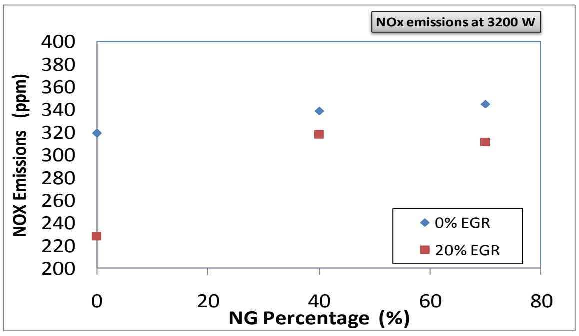
(a) NO_x Emissions vs. Load at 0% of EGR



(b) NO_x Emissions vs. Load at 20% of EGR



(c) NO_x Emissions vs. NG percent (with/without) EGR at 2000W



(d) NO_x Emissions vs. NG percent (with/without) EGR at 3200W

Fig. 16 NO_x emissions at different NG/Diesel contents and with/without EGR.

4. Conclusion

This paper investigates the performance of a small dual fuel engine where a single cylinder diesel engine is converted to run on a dual fuel of diesel and compressed natural gas (CNG). Exhaust gas recirculation (EGR) was utilized in this study. Based on the obtained experimental results, it can be concluded that:

- Although NG is an excellent candidate to overcome the shortage in diesel fuel especially for small power engines, it was noticed that the total reduction in engine thermal efficiency increases as the ratio of the used natural gas increases. A maximum reduction of 5-8 % in the absolute engine efficiency was recorded over the entire window of applied loads and NG/diesel ratios.
- When 15 % EGR was considered, more than 28 % reduction in the thermal efficiency was recorded when NG was used at relatively low loads. At high loads however, the adverse effects of NG addition is less (about 14 % reduction in thermal efficiency).
- The increases in NG fraction and EGR generally reduce the rate of reaction and thus increase CO emission except at high loads, where CO emissions tend to decrease with adding EGR.

NO_x decreases as NG contents increase at medium loads while at high loads NO_x emission tends to increase with increasing NG contents. Also, NO_x decreases with EGR for the entire range of loads however, at high loads the increase in NG content becomes dominant and takes the rule of increasing the final NO_x emission.

5. References

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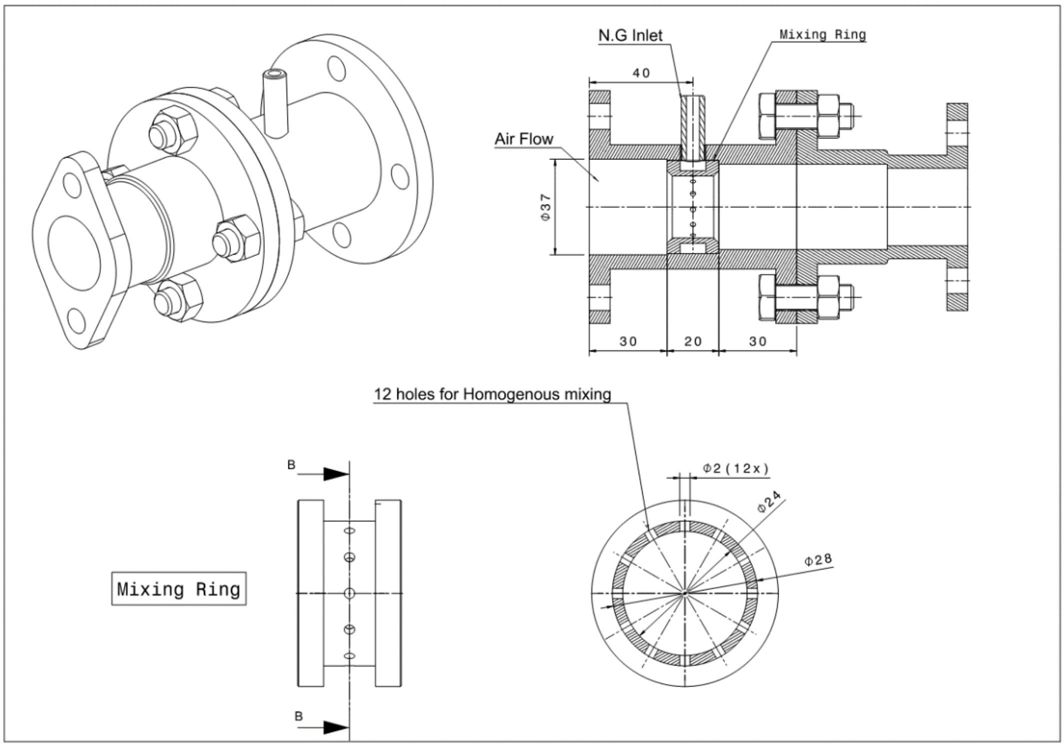
Appendix (A) Gas Mixer features



Appendix (B) Fuel properties [20]

Approximate Properties	Natural gas	Diesel
Chemical Formula	CH_4	C_8 to C_{25} ($C_nH_{1.8n}$)
Molecular Weight	16.04	200
Composition, weight %	75% Carbon 25% Hydrogen	87% Carbon 13% Hydrogen
Specific Gravity @15 °C	0.424	0.85
Boiling temperature °C	-128	180-340
Higher Heating Value (HHV) Kj/Kg	55,530	45,500
lower Heating Value (LHV) Kj/Kg	50,050	42,800
Freezing Point °C	-146.6	-4.44
Flash point °C	-148.8	60-80
Auto-ignition Temp °C	482-632	315
Specific Heat Kj/Kg.K @20 °C	2.2	1.9
Stoichiometric air/fuel, weight	17.2	14.7