A Study on the Performance and Emission of a Compression Ignition (CI) Engine Using Syngas and Diesel Blend in Dual Fuel Mode

Njeru Gatumu, Dr. Hiram Ndiritu and Dr. Benson Gathitu

Abstract−Coal has been a major source of energy for many countries of the developed world especially where coal deposits exist. This situation is expected to persist for a long time to come; more so, in light of the rapid depletion of the world's oil reserves, underdevelopment of conversion equipment for renewable energy and the uncertainty of nuclear energy. However, most coal used today is for production of power, steel making or transportation industry and has been the major contributor to global warming and greenhouse effect. These problems associated with coal have necessitated studies to be carried out under the clean coal technology and green coal programs. Both aimed at reducing its negative environmental impact and health hazard in power and industrial plants. Also in its use as alternative fuel in ICE engines. This is partly achieved through gasification of existing coal to produce a combustible gas called synthetic gas (syngas). Syngas can meet energy needs in an economical and environmentally friendly way, especially in the areas where coal is available, the price of petroleum fuels are high, or where supplies are uncertain. Studies have shown that CI engine using syngas for fuel is very promising and even cost competitive alongside other gaseous fuels. This study is directed towards the detailed analysis of CI engine fuelled by a mixture of syngas as the primary and diesel as the pilot fuel in dual fuel mode. The paper has included information on the performance and emissions of CI engine when using syngas-diesel blend and compared it with when running on neat diesel.

Keywords−Brake thermal efficiency (BTE), emissions, Internal Combustion Engine (ICE) specific fuel consumption (BFC), Synthesis Gas (syngas).

I. INTRODUCTION

Diesel fuel has for a long time been one of the two most important fuel used in internal combustion engines (ICE). Where high torque and fuel economy are both required, diesel has pro-vide the solution. Diesel engines have considerable advantages in the aspect of engine power, durability, fuel economy and very low CO emissions. They are widely applied in vehicles [1]. However, even with these crucial advantages in the provision of energy needs, they have been found to be major contributors to environmental degradation. So much so, many countries have given a time-line of the years 2025 - 2050 by which time they will have phased out diesel vehicles from their cities and towns. However, the objection to use of diesel engines in vehicles is not based on its failure as a power plant

But only for its pollutant emissions. For that reason, there is need to develop an alternative fuel that is economical, has good performance when compared with diesel and at the same time

produces exhaust gases with limited or no acidic gases as nitrogen oxide, NO_x (NO and NO_2), SOx or other objectionable properties such as CO, CO_2, hydrocarbons (HC) and Particulate Matter (soot and smoke).

The main pollutants from the diesel engines are NO_x (NO, nitric oxide and, nitrogen dioxide), unburned hydrocarbons (HC) and smoke. To reduce these harmful pollutants alternative fuel -which does not emit other pollutants like aldehydes, ketones and SO_x. Various fuels have been considered as substitutes for the hydrocarbon-based fuel. The alternative fuels aspiring to replace the petroleum-based fuels are Alcohols, LPG, CNG,H_2, vegetable oils, biogas, syngas and LNG [2]. For all the alternatives, the one, which has proved most attractive, is the use of gaseous fuels in place of fossil diesel. This is because, combustion of gaseous fuels produces almost no oxides of Sulphur (SO_x) and relatively little oxides of nitrogen (NO_x) (the main constituents of acid rain) and substantially less carbon dioxide CO_2 (a key culprit in the greenhouse effect), than most oil products and coal [3]

Sustaining a clean environment has become an important issue in an industrialized society. The air pollution caused by IC engines is one of the most important environmental problem to be tackled [2]. The researches that have be done so far demonstrate that the syngas-diesel blend fuel can reduce the air pollution and at the same time alleviate depletion of petroleum fuels. For this to happen there is need to study the impact of these alternative fuels on the engine performance and exhaust emissions of IC engines.

The two practical sources of syngas for use in ICE are; coal gasification and biomass pyrolysis. The latter process producing what is now known as producer gas- a lower heating value type of syngas whilst and coal will mostly give rise to medium to high heating value syngas but all depending on the process used and the type of coal used [4]. An overwhelming majority of researchers concentrates their effort in the study of combustion of gaseous fuels produces almost no oxides of Sulphur (SOx) and relatively little oxides of nitrogen (NOx) (the main constituents of acid rain) and substantially less carbon dioxide CO2 (a key culprit in the greenhouse effect), than most oil products and coal [3]

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Confining the study of syngas application to ICE, Bates and Dolle¨ [6] in their review work, on use of syngas in ICE, stated that it is not possible to use syngas as a stand-alone fuel in compression Ignition (CI) engine. They explained this as being caused bythe very high auto-ignition temperature of syngas(500 C) - which way beyond the maximum temperature reached by most engine during their compression stroke.They informed that syngas should be used in dual fuel mode in which diesel fuel is used as the pilot fuel and syngas is introduced through the engine intake air after being mixed with inlet air, and therefore provides the bulk of the fuel charge. Pradhan et al. [3] who also stated that the quality of syngas gas as a fuel is significantly poorer than gasoline and natural gas explained the same observation. Therefore, the engines require certain design modifications in order to be able to run on syngas gas. Sridhar et al. [7] investigated suitability of syngas use in ICE. Their findings were that diesel fuel savings of up to 85% could be obtained in dual-fuel engines.

They also went on to show that syngas was more suited in diesel engines due to their higher efficiency. Again, this was because of the greater compression ratio, which usually varies between 12 and 24. Diesel engines also have better durability and, in some cases, require low maintenance than spark-ignition engines [8]. Further, Pradhan et al. [3] explained that because of high compression ratio and low speeds, the de-rating of diesel engines running on syngas is only between 15-30%.

As understood from a number of studies, use of syngas in ICEs lead to poorer engine performance but improved emissions. Raman and Ram [9] report that the CI engine in dual fuel mode has an energy efficiency generally of about 20% when using syngas compared with 45-54% for the conventional diesel engine running on Neat Diesel (ND). Even more disturbing is that this efficiency is only achieved when the engine is run at full power and that efficiency falls off rapidly at partial load and low engine speeds [10].

To improve the performance and exhaust emission in dual fuel engine the following factors have been cited: syngas composition, engine's compression ratio (CR), engine operating load and speed Fuel injection timing (IT).

Azimov et al. [11] investigated the effect of H_2 and CO_2 compositions in syngas on the performance and emission of a four-stroke single cylinder engine in dual fuel mode. The engine was supercharged and operated in a premixed mixture ignition in the end-gas region (PREMIER). The type of combustion was observed to increase the efficiency of the engine thus helped to enhance its performance. Moreover, they reported an increase in the mean combustion temperature, indicated mean effective pressure (IMEP), and efficiency with increase in hydrogen com-position in syngas.

Hassan et al. [12] examined the effect of fuel injection timing, engine load, speed on engine performance, and exhaust emission of a four-stroke single cylinder direct injection diesel engine operated on dual fuel mode. Their findings were: reduced consumption of diesel fuel at all engine loads, advancing of IT resulted in improvement of brake thermal efficiency for all loads, lowering of CO emission for engine speeds of 2300, NO_x increases with increase in load, lower Exhaust Gas Temperature (EGT) and reduction of CO_2 at all loads. To a large degree, Murthy et al. [1] reported the same findings in the study of the performance and emissions of dual fuel CI engine with an exhaust gas recirculation. They went further to show both hydrocarbon (HC) and Particulate Matter (PM) emissions reduced in advancement of IT. Hariram and Shangar [13] investigated the influence of CR on performance, emission and combustion characteristics of a Compression Engine (CI) in both dual fuel and diesel only modes, by

varying CR with 50% load. They obtained result showing that: EGT increased with higher compression ratios, the mechanical efficiency gradually decreased with increasing compression ratio, the brake thermal efficiency increased and brake specific fuel consumption reduced on increasing CR. The Peak cylinder pressure also increased with increase of CR. Heat release rate reduced with increase of Compression ratio. Lal and Mohapatra [14] carried out an experimental investigations on a variable CR dual fuel engine by varying compression ratios from 12-18. They observed that: as the load and compression ratio increased, NO_x emission increased, CO emission decreased as compression ratio increased, HC emission increased at higher compression ratios, and that the EGT reduce as compression ratio increased from 12 to 18. Also from their observation as the load and compression ratio increased, CO_2 emission increased in both mode of operation.
From the literature review, what is notable is the absence of investigation work for exhaust emissions and performance parameters of a diesel engine running on coal syngas-diesel fuels. Which means the effects of CR, loads and speed, IT and gas composition on the two parameters have not been clearly studied when used in a CI engine. There is need therefore, for these areas to be investigated in light of the large coal deposits in the country. For this reason, the present study was designed to experimentally investigate the performance and exhaust emissions characteristics on a single cylinder variable CR CI engine operated on dual fuel mode with syngas as primary fuel and diesel as pilot fuel.

II. EXPERIMENTAL APPARATUS AND PROCEDURE

The experiments were conducted on a 3.5 kW, 4 stroke, single cylinder, water-cooled, engine, with SI and CI mode facilities, variable compression ratio research. The geometrical and operation specifications of the engine specification are shown in Tables I and II. The engine was coupled to an eddy current-type dynamometer for loading. K-type thermocouples for tapping temperatures for incoming air and syngas, air-syngas mixture, coolant temperature, combustion chamber, inside the crankcase and exhaust gases. A bench scale fluidized,
bed-type gasifier was coupled to the engine to supply the required syngas with an air blower for supplying reactor's combustion air. Flowmeters for syngas, air and exhaust gas were also fitted. The exhaust emissions were analysed in an ecom-J2KNpro gas analyser, comprising of a control module, different gases sensors and filters, sampling probe, gas cooler, condensate evacuation unit and an integral printer.Fuel consumption was read directly from the PC monitor after obtaining the data fromflow meters and transmitters in the engine's information system.Piezo sensor and the crank angle sensor are connected respectively to engine cylinder and crankshaft and then to PC processor to provide information on in-cylinder pressure for the angle turned by the crankshaft,

TABLE I: Research Engine geometrical Specifications

Parameter	Value
Displacement(cm^3)	661
Number of Cylinders	1
Number of Strokes	4
Stroke (mm)	110
Bore (mm)	87.5
Power(kW)	3.5
Compression Ratio	VCR(18-16)
Torque(Nm)	11.5

TABLE II: Engine Operation Specifications

S/No.	Specification	Value
1	Number of cylinders	1
2	Number of strokes	4
3	Fuel	Diesel/Petrol
4	Rated power (kW)	3.5
5	Speed (rpm)	1500
6	Variable Compression ratio	12-18:1
7	Fuel consumption (g/kWh)	251
8	Injection pressure (bar)	200
9	Injection timing (btdc)	23
10	Cylinder capacity (cc)	950
11	Combustion system	Direct Injection

The connected PC and the standard accuracies of measurement as provided by the manufacturer assisted in carry out measurements. Tolerance of different equipment and the calculated uncertainties are given in Table III.

TABLE III: The accuracies of the measurements and the uncertainties in the calculated results

Parameter	Accuracy
load	+/- 0.1% Nm
Speed	+/- 0.1% rpm
flowrates	+/- 0.1% m^3=s
Time	+/- 0.25%
Temperature	+/- 0.25%
NO_x	+/- 0.1%
HCL	+/- 0.1%
CO(Nm)	+/-10 ppm
PM	+/- 1%
CO_2	+/- 0.1%
Parameters	**Uncertainty Analysis**
Speed	+/- 1.4%
BSFC	+/- 0.7%
BTE	+/- 1.28%
Load	+/- 0.9%
Combustion pressure	+/- 1.1%

The required syngas was obtained by cooling in heat exchanger and filtering the raw gas that came from the reactor. This was necessary since gas obtained from the available gasifier was produced gas at temperatures that ranged from 250 300 C. In order to use this gas for ICE applications, firstly it was cleaned of tar and dust and then cooled. A gas-to-air heat exchanger reduced the syngas temperature to almost room, which is desirable as excessively high temperature in the engine's combustion chamber facilitates formation of acidic NO_x species.

To optimise the CI engine to run at maximum efficiency, modification to its air supply system was made. A syngas-air mixture was fitted. The exhaust was fitted with a flow control to allow for extraction of a sample of exhaust gas to be extracted for testing. Fig. 1 illustrate the experimental set-up. Also adjustment were made to the engine's CR and IT to determine the best operating conditions. The research engine used in this experiment had feature for varying compression ratio of the engine.A fitted tilting-block mechanism was used to vary the compression ratio without changing the geometry of the combustion chamber. The mechanism works by varying the TDC position of the piston. To study the effect of varying CR on performance and emission characteristics of the dual fuel engine, the engine was run with CR ranges of 12-18. Performance and emissions experimental data were collected for CRs of 18, 16, 14 and 12.The standard IT of the engine is 23 CA BTDC. IT was adjusted by turning the flywheel and counting the number of teeth on the flywheel ring gear moved and noting when injector spill.

Each two teeth from a mark given by the manufacturer on the flywheel represent crank angle of 2.2^{o}. For instance, from the standard set injection timing of 23 ^{o}CA, addition of two teeth advanced injection timing to 25.175^{o} CA bTDC.

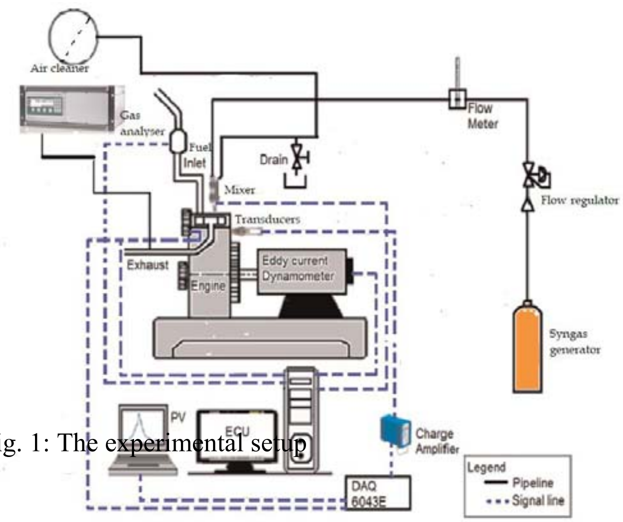

Fig. 1: The experimental setup

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To obtain figures for Brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC), are the two parameters that are the key indicators of an engine performance. The BTE is defined as the ratio of the brake power to the heat input (LHV) for the fuel or blend of fuels, whilst BSFC is the ratio of mass of fuel consumed in an engine in kg to kW of brake-power produced and is an accurate measure for efficiency of any engine.

Brake thermal efficiency (η_{Bth}) and brake specific fuel consumption (bsfc)

$$\eta_{Bth} = \frac{P_b}{m_S LHV_S + m_D LHV_D} \quad (1)$$

$$\text{bsfc} = \frac{m_s + m_D}{P_D} \quad (2)$$

where, m_S and m_D are masses of syngas and diesel consumed per second whose values are read from the flow meters. For this engine, and when running on neat diesel equations, 3 and 4 applies.

$$\eta_{Bth} = \frac{P_d\, X\, 3600\, X100}{m_D(in\ kg/(\ hr)) + LHV_D} (3)$$

$$\text{bsfc(kg/kwh)} = \frac{mD(in\ kg/hr)}{P_b} (4)$$

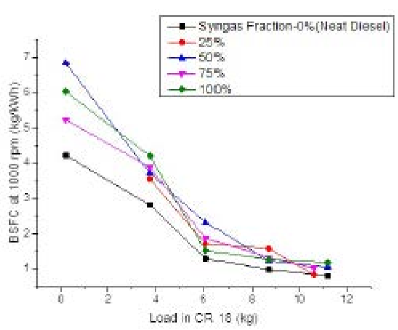
III. RESULTS AND DISCUSSION

To compare the performance and emissions of an engine when run in the two modes: dual fuel and diesel, conditions of varying load and speed, adjusted IT and CR are examined in this section. Syngas-diesel blend operated in dual fuel mode engines have been found to reduce the pollutant emissions in exhaust gases. To achieve this state some degree of modification needed to be done on the engine. Some are structural as seen in Figure 1 and others are effected by carrying out engine's adjustments. The study has shown that compression ratio and injection timing have significant effect on the engine performance and emissions of a CI engines. Therefore, in this study, the effects of compression ratio and injection timing for varying fractions of syngas substitution of diesel on the engine performance and exhaust emissions were experimentally investigated on a single cylinder CI engine with variable CR. The experimental set conditions used were three engine speeds; 1000, 1250 and 1500 rpm, four constant loads; no-load, half, three-quarter and full loads, and four injection timings 20.8, 23, 25.175 and 27.4 CA BTDC. The syngas substitution fractions, which were obtained by manual controls, were: 100%, 75%, 50%, 25% and 0 %(neat diesel). The engine's governor controlled Diesel fuel consumption. Governor regulate fuel based on the engine speed. The diesel engine governor in dual fuel mode increases or decreases the amount of diesel fuel injected as necessary to maintain engine output in the face of decreasing or increasing syngas energy content [15].

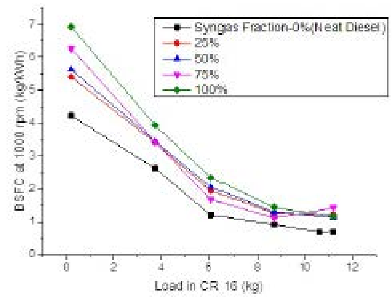
A. Brake specific fuel consumption (BSFC)

BSFC in diesel mode and dual fuel mode are calculated from calorific value and the fuel consumption of diesel and syngas gas as already explained. Fig. 2(a and b) shows the variation of BSFC at CRs of 18 and 16 respectively for neat diesel and syngas-diesel blend for syngas fractions flow rate of 100%, 75%, 50% and 25% as the load was varied at a constant speed of 1000rpm. Similar test were ran for CRs 16, 14 12 at speeds of 1250 and 1500 rpm. Results obtained showed that the lowest values of BSFC were obtained when the engine ran at full speed of 1500 rpm and CR of 18. Comparing CRs of 16 and 18, using Fig.3(a and b) the BSFC improved remarkably for no load to part load and gradually decreased on increasing loads when CR increased from 16 to 18. For CR 18 the values were: 5.5 for no-load, 0.95; quarter-load, 0.67 for half-load, 0.57kg/kWh for three-quarter load and increasing slightly to 0.82 kg/kWh at full-load. For CR of 16 and for the same condition of load results obtained were 2.86, 3.1, 1.4 0.95 and 0.9 kg/kWh respectively in diesel mode. For dual mode on 100% fractions, the results obtained showed lowest BSFC was recorded at three-quarter load for both CR 18 and 16. This could be due to the reduced heat loss at higher loads. Hariram and Shangar [13] obtained the same effects. In addition, the BSFC reduced with increase of CR. At 1500 rpm, and at full load, for neat diesel and 100% syngas flowrate, the values recorded were 0.82 and 0.6 kg/kWh for CR 18 against 0.93 and 1.7 kg/kWh for CR 16 respectively. This was better explained by the fact there was better combustion and lesser heat losses at higher CR. Murthy et al. [1] reported the same results. The reason for the BSFC increasing with increasing percentage of syngas substitution at part loads may be due to incomplete combustion of the gaseous fuel, while at higher loads BSFC improves with the increase of syngas substitution [16].

The investigation in this study has established an increase of BSFC with every increase of substitution of diesel with syngas (Figures 2). At full load, with engine speed of 1500 rpm and for CR 16, BSFC values were 0.61 kg/kWh at 0% syngas gas, 0.78kg/kW h at 25% syngas, 0.62 kg/kWh at 50% syngas, 1.72 kg/kW h at 75% and 1.7 kg/kWh at 100%. Ahmed et al. [17] obtained similar results with their study. The lowest value of 0.61 kg/kW h being obtained with neat diesel compared with 1.7 kg/kWh for 100% of flow rate. This effect is attributed to LHV per unit mass of the syngas fuel, which is considerably lower than that of the diesel fuel [18].

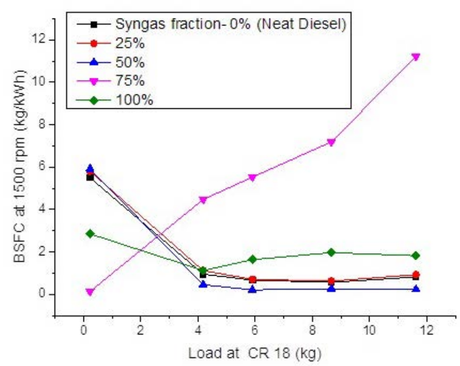


(a) BSFC for Compression Ratio (CR) 18 at 1000 rpm

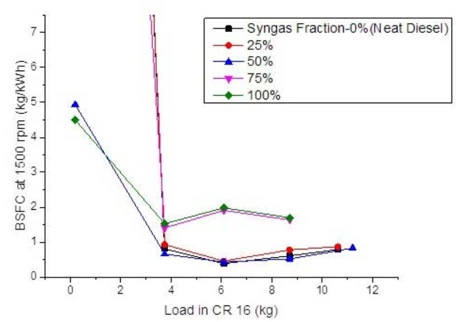


(b) BSFC for CR 16 at 1000 rpm

Fig. 2: Effect of varying engine load and compression on BSFC of CI dual fuel engine at 1000 rpm



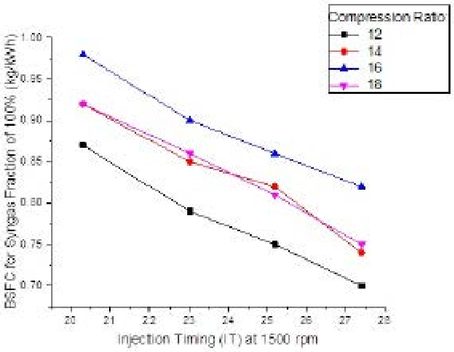
(a) BSFC at CR 18 for constant engine speed of 1500 rpm



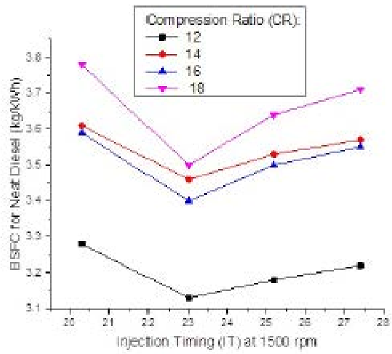
(b) BSFC for CR 16 at constant engine speed of 1500

Fig. 3: Effect of varying engine load for CRs of 18 and 16 on BSFC of CI dual fuel engine at 1500 rpm

To improve the BSFC values of the engine on dual fuel, advancing of Injection timing was done.This effect is illustrated in Fig. 4 (a and b). From standard CA 23 ^{o}to CA 25.175 ^{o} and CA 27.4 ^{o}with the latter showing better results. This was so because of the increased resident time for the fuel in the combustion chamber, thus giving it more time for combustion. Hassan et al. [12] obtained similar results with a supercharged dual fuel engine.



(a) BSFC for syngas flow fraction of 100% at 1500 rpm



(b) BSFC for Neat Diesel (ND) at 1500 rpm
Fig. 4: Effect of varying engine IT and CRs on BSFC of CI on dual fuel and diesel modes

B. Brake Thermal Efficiency (BTE)

BTE can be described as the percentage of brake power and fuel energy consumed by the engine. It demonstrates how input energy is converted efficiently into useful output energy. Fig. 5 and 6 shows that as load increases brake thermal efficiency increases this is because at low and intermediate load conditions, there is poor utilization of syngas leading to incom-plete combustion of syngas. For all loads, as the engine governor varies diesel fraction, brake thermal efficiency increases. This is because, with high quantity of pilot diesel the mixture of the syngas-air-diesel is richer resulting in complete and thus higher brake thermal efficiency in dual fuel mode. Higher brake thermal efficiency is due to better mixing of syngas with air which results in better combustion and also due to wider ignition limit and high flame propagation [16] [9]. The syngas engine delivers about two-third of the power at its maximum load as compared to the performance of engines using liquid fuel [19]. Mohammed et al. [20]reported that reduction in energy density and a reduction in compression ratio result in 20% de-rating of the engine as compared to operating it on diesel. In the present study, an efficiency of 21% was obtained with the CR 18 at a speed of 1500 rpm for syngas flow rate of 50%. This maximum power efficiency of 21% was achieved at half load (Fig. 7). It was also noted in the present study that a maximum efficiencies of 22% and 21% respectively was achieved at CR 16 at half load with neat diesel (Fig. 8) and CR 12 at three-quarter loads for 1500 rpm. This could be explained as due to factors such as; energy content offuel mixture, volumetric efficiency and balanced air-fuel mixture.Atiqh [21] reported the same results.

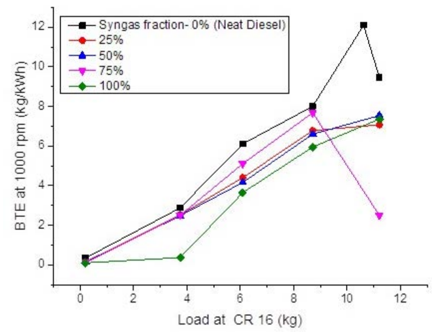


Fig. 5: Effect of engine load on BTE of Dual fuel CI engine at constant speed of 1000 rpm with a CR16

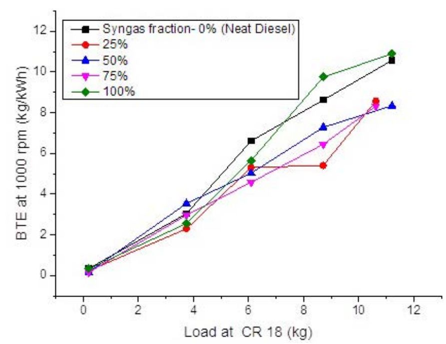


Fig. 6: Effect of varying engine on BTE of dual fuel CI engineat constant speed of 1000 rpm with a CR 18

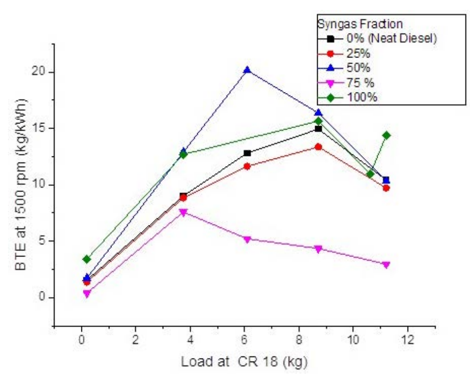


Fig. 7: BTE for varying load and syngas flow fraction for constant speed of 1500 rpm at CR 18

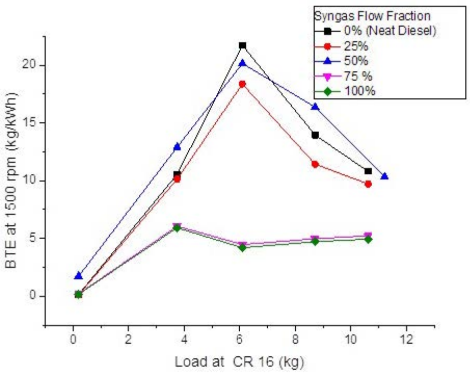
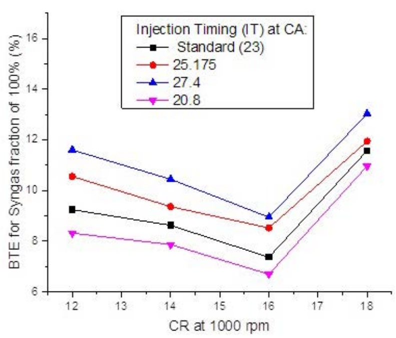
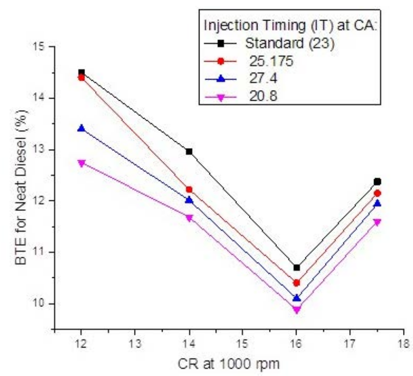


Fig. 8: BTE for varying load and syngas flow fraction for constant speed of 1500 rpm at CR 16

To improve BTE of the engine in dual fuel, studies have identified the following methods: use of syngas with high calorific value gas without adding much complications and cost or optimising adjustment to Ignition timing and compression ratio. The present study will investigate the adjustment of IT and CR on BTE. Fig.9(a and b) and Fig.10(a and b) shows that variations of the BTE with neat diesel and syngas flow fraction of 100% at different injection timings for the speeds at 1500 and 1000 rpm. The highest BTE results were obtained at standard injection timing (23^o CA) for ND and advance IT of CA 27.4 for 100% syngas flow fraction. Retarding IT to 20.8 or advanced, IT to CA 25.175 lowered the BTE values due to incomplete combustion. Optimising for CR, 18 and 16 at 1500 rpm exhibited the highest BTE, ahead of CRs of 12 and 14 (Fig. (11 and 12)). The overall BTE of 27% were obtained with Syngas Flow Fractions (SFF) of 25% and 50% at CR 18 and by advancing IT to27.4^o. The result were possible because of the longer residence time for the charge in Combustion Chamber (CC), which allowed for a more complete combustion. Sayin and Canakci [22] have reported the same observation. Hariram and Shangar [13] have observed that BTE increased with increase of compression ratio due to better combustion at higher compression ratios. Similar to what has been shown in this study, they reported that the BTE was higher by almost 13% at full load when CR was increased from 16 to 18.

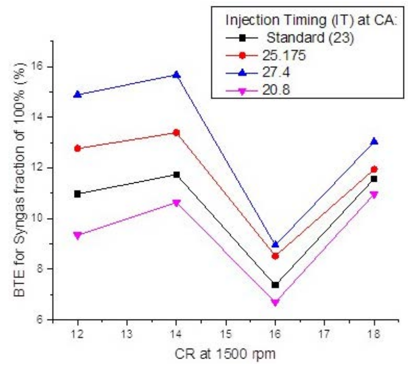


(a) BTE with Syngas flow fraction of 100%

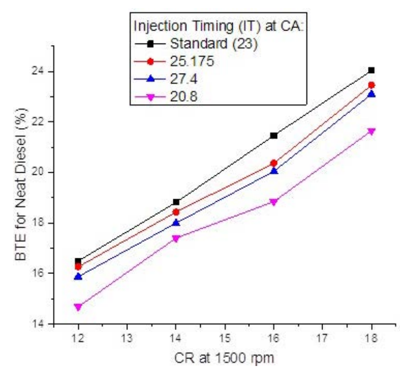


(b) BTE for Neat Diesel (ND)

Fig. 9: BTE with varying IT and CR for constant speed of 1000 rpm



(a) BTE for syngas flow fraction of 100%



(b) BTE for Neat Diesel(ND)

Fig. 10: BTE with varying IT and CR for constant speed of 1500rpm

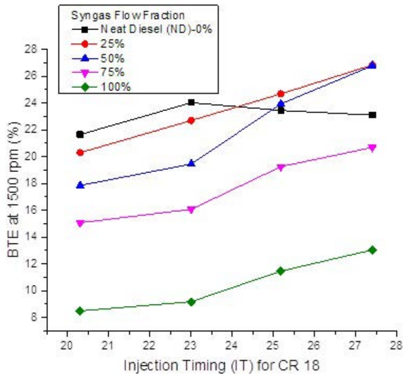


Fig. 11: BTE for varying IT at CR 18 and 1500 rpm

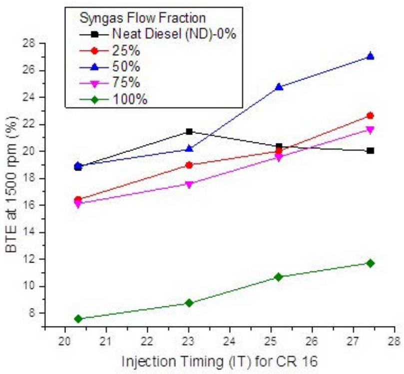


Fig. 12: BTE for varying IT at CR 16 and 1500 rpm

C. Carbon monoxide (CO)

CO is an intermediate product in the combustion of a hydro-carbon fuel, so its emission results from incomplete combustion. Emission of CO are therefore greatly dependent on the air–fuel ratio of charge. Rich mixture invariably produces CO, and emissions increase nearly linearly with the deviation from the stoichiometric [22]. Due to deficiency of oxygen, the carbon present in fuel does not contribute fully in combustion process and appear in the engine exhaust in the form of CO. The various values of CO emission with varying load for neat diesel and SFF of 50% and 100% are shown in Fig. 13,14 and 15. It was observed that with an increase in load, CO emission decreases in both mode. This may be explained as being caused by the engine governor responding to load increase by allowing for injection of more diesel fuel. The resulting richer air-fuel mixture lead to more complete combustion thus less amount of CO emission was produced.

Pradhan et al. [3] in their review work reported that CO emissions are higher for dual fuel mode at low and moderate loads but decreases with the increase of engine load. Arguing that this is most probably due to the improvement of the fuel utilization. From Fig. 16, it was seen CO emission decreases as compression ratio increases even in lower speed. Similar behaviour was observed in lower speeds. In the present test lower emission of 875, 1200 and 602 ppm for SFFs of ND (0%), 50% and 100% respectively for CR of 18 at constant speed of 1000 rpm were observed. The reasons of higher CO emission in dual fuel mode are low oxygen present in the air-syngas mixture, which caused the incomplete combustion. Shrivastava et al. [23] reported the maximum concentration of CO by 10 ppm in diesel mode and 250 ppm in dual fuel mode. Lal and Mohapatra [14] reported the maximum concentration of CO 700 ppm in diesel mode and 1300 ppm in dual fuel mode.

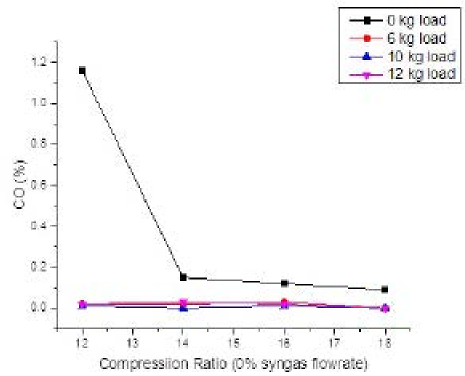


Fig. 13: CO emission dual-fuel CI engine running on ND fuel

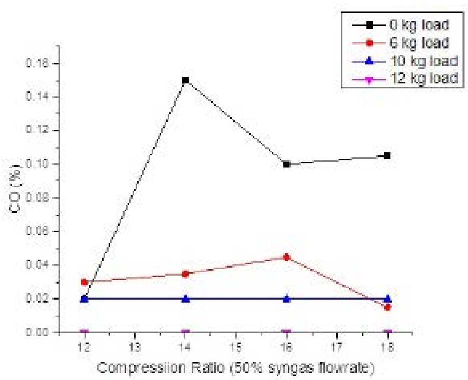


Fig. 14: CO emission dual-fuel CI engine running on 50% SFF

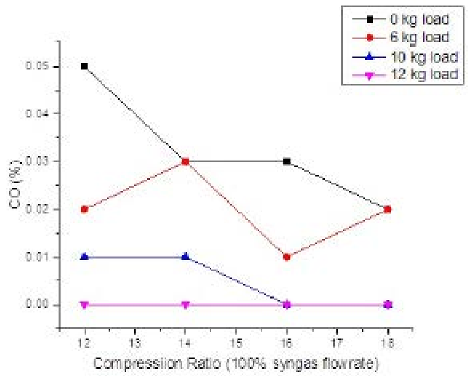


Fig. 15: CO emission dual-fuel CI engine running on 100% SFF fuel

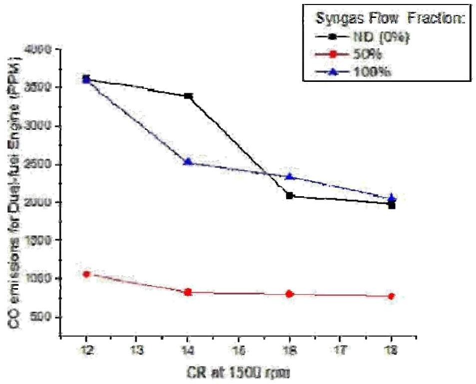


Fig. 16: CO emissions for varying CR and SFF at constant speed of 1500 rpm

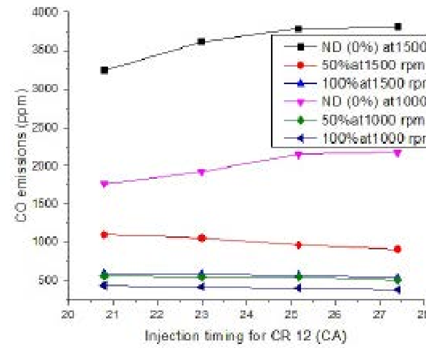


Fig. 18: CO emissions for dual-fueled CI engine for different SFF and speed at constant CR of 12

Fig. 17 and 18 shows CO emission results for different SFF fuels and injection timings at CRs of 18 and 12 and speeds of 1500 rpmand 1000 rpm. When the injection timing was advanced to CA 27.4^o,the level of CO emission decreased for SFFs of 50% and 100% but increased for ND. Even more reduction was noted when the engine was run at 1000 rpm and at CR of 12 Retardation of IT caused higher levels of CO emission in CR of 18, retarding IT from standard (CA 23^o) to CA 20.8 emissions increased by 267,15 and 85 ppm for ND, 50% and 100% SFF respectively. The probable cause of the phenomenon is the advanced IT produced higher cylinder temperature, which increased oxidation process between carbonand oxygen molecules. Nwafor, and Sayin and Canakci [2], [22] when they tested the engine with similar fuel injection timings, but different primary gaseous fuels, obtained similar results.

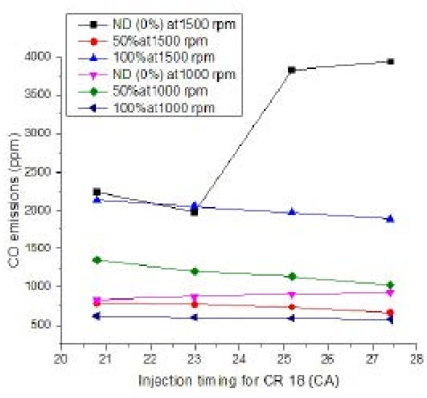
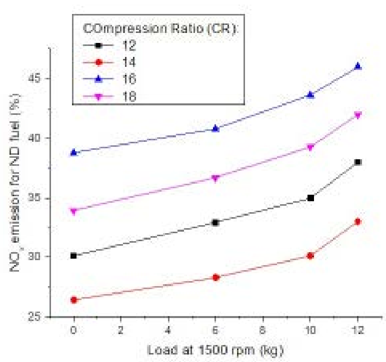


Fig. 17: CO emissions for dual-fueled CI engine for different SFF and speed at constant CR of 18

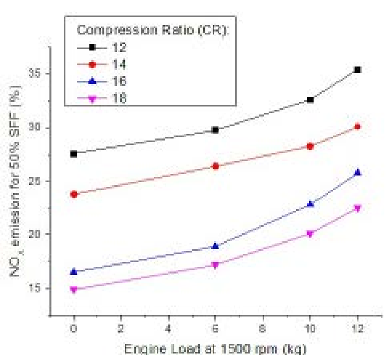
D. Nitrogen Oxide (NO_x)

It is well known that the formation conditions of $NO_x$are:high temperature, enough oxygen and sufficient reaction time [24]. The variation of NO_x emission of dual-fuelled engine ran on diesel mode and dual fuel mode with SFF of 50% and 100% for different loads are shown in Fig. 19 (a, b and c). In the current study, it was observed that as the load and CR increased, NO_x emission increased proportionately except for SFF of 50%.

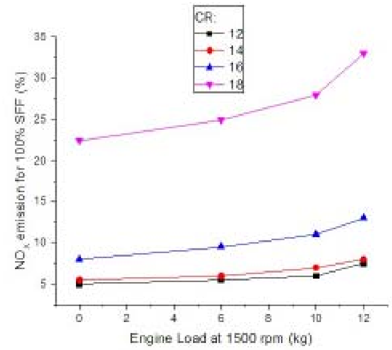
The explanation one may proffer is, more fuel was injected in higher load condition and also at higher compression ratio, temperatures and pressure in the engine cylinder were increased, the condition became conducive for NO_x species formation. As for the case of SFF of 50%, which continually exhibited better emission results, one may argue that better mixture homogeneity was readily achieved for this SFF, which lead to a gradual and well-controlledflame propagation.As such, pockets of high and low temperature in avoiding CC is avoided. Notably, the highest concentration of NO_x emission in the exhaust gas were recorded in ND mode at 46% for CR of 16 at maximum load. As expected the blend with lowest concentration were those for SFF of 100% at CR of 12 and no-load, returning 5% NO_x concentration in exhaust gases. It was observed that the NO_x emission in diesel mode was about 22% to 33% higher than dual fuel mode at ND and SFF of 100% are compared. The lower NO_x emission obtained in dual fuel mode could be explained as due to the low flame propagation in premix mixture due to the presence of a high amount of syngas gas and lower concentration of oxygen in the cylinder leading to lower temperature. For a CR of 18, other studies that were reviewed for present work have published almost similar results. Shrivastava et al. [23] reported maximum NO_x concentration of 325 ppm in diesel mode and 180 ppm in dual fuel mode at 3/4 load condition. Dhole et al. [25] reported maximum NO_x concentration of 904 ppm in dual fuel mode at 80% load condition. NO_x emission were observed to increase sharply with the advancing of fuel injection timing as shown in Figure 20(a and b). This conform with the explanation elsewhere that when the fuel injection timing is advanced, the combustion time is extended as the ignition delay period is prolonged allowing for more complete combustion and the attendant higher temperature. Similarly higher concentration of $NO_x$are experienced in ND mode, with values of 72% and 65% for advance to CA 27.4 and CA 25.175 respectively being observed. This is 51% and 41% higher than equivalent conditions in dual-fuel mode for SFF of 50%. When the IT was retarded, a decrease of NO_x emission was observed on all the three fuel mixtures studied. When the IT was retarded from standard CA (23^o) to CA 20.8^oemission decreased from 58% to 54% and 42% to 37% for CR of 18 and CR 12 respectively. Retarding the injection timing decreases the peak cylinder pressure because more of fuel burns after TDC. Lower peak cylinder pressures results in lower peak temperatures. As a consequence, the NO_x concentration starts to diminish [26] Similar results were reported from other studies: Sayin and Canakci [22] reported that when the injection timing was retarded 6 CA BTDC in comparison to original injection timing, NO_x emissions decreased by 37.3% in part-load. Hassan et al. [12] have reported similar results.



(a) NO_x emission for ND fuel

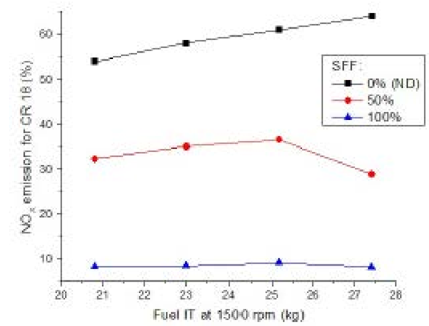


(b) NO_x emission for SFF of 50% fuel

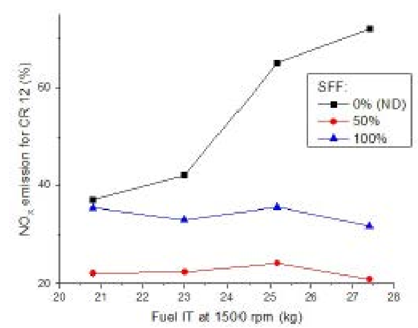


(c) NO_x emission for SFF of 100% fuel

Fig. 19: NO_x emission from dual-fuelled CI engine for varying loads at constant speed of 1500 rpm



(a) NO_x emission for CR of 18

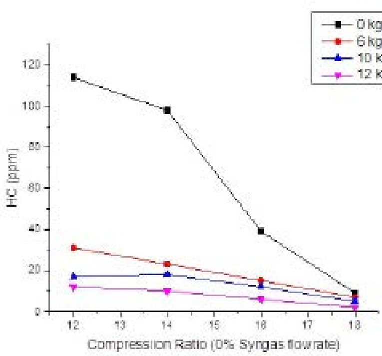


(b) NO_x emission for CR of 12

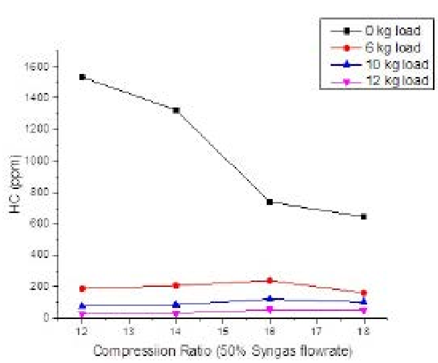
Fig. 20: NO_x emission from dual-fueled CI engine for varying Fuel IT at constant speed of 1500 rpm

E. Unburned hydrocarbon (HC) emissions

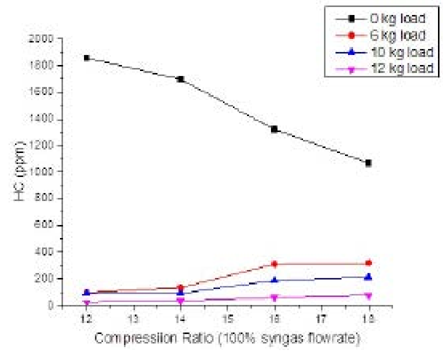
The two main causes of Hydrocarbon emission in diesel engines are: (a) during the delay period, fuel mixture is leaner than that of the lean combustion limit and (b)under-mixing of fuel. During idling and at light applied loads exhaust gases for CI engines have been found to consist of fuel that is completely unburned or partially burned. Consequently, the engine had very high concentration of HC emission [27]. It can thus be explained, HC emission arises when a part of the fuel inducted into the engine escapes combustion. At the ignition delay period, fuel-air mixture becomes too rich to ignite and burn, the unburnt mixture therefore contribute to HC emissions [15]. The comparison of HC emissions of all the pilot fuels in dual fuel mode is shown in Fig. 21(a, b and c). In dual fuel mode, the increase of engine load resulted in a drastic decrease of HC emission. Unburned HC are serious problem at light loads in CI engines. The reason given for this state is that at light loads the fuel is less likely to impinge on surfaces; but because of poor fuel distribution, large amounts of excess air and low exhaust temperature, lean fuel–air mixture regions may survive to escape into the exhaust [22]. In addition, as CR increases HC emission decreased. It is seen in Fig. 22(b and b) that advancing the injection timing reduces unburned HC emissions. Advancing the injection timing causes earlier start of combustion relative to the TDC. Because of this, time for combustion of the charge in cylinder was increased, producing relatively higher temperatures and consequent complete combustion.Hence, reducing the unburned HC emissions. On the other hand, for the same test condition, retarding IT to CA 20.8^{o} , HC emission increased. Sayin and Canakci [22] have observed that when the engine they used for the test was retarded by 6^{o} (from 27^{o} to 21^{o} CA BTDC), the unburned HC increased by 51.2%. These findings, on the effects of IT adjustment on HC emission, correlated well with the findings reported in refs [2], [22], [20].



(a) HC emission for ND fuel

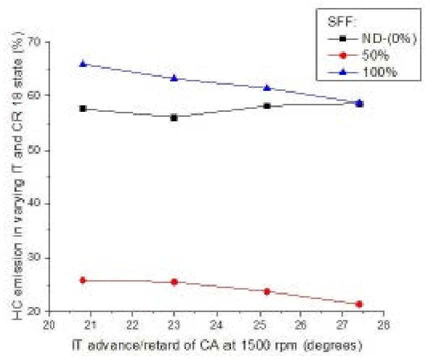


(b) HC emission for SFF of 50% fuel

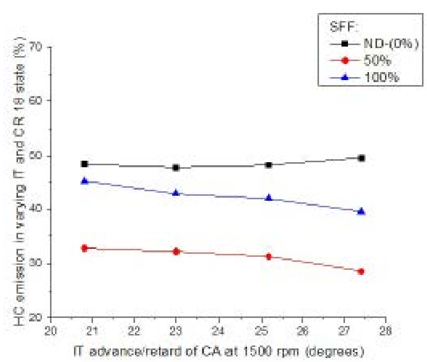


(c) HC emission for SFF of 100% fuel

Fig. 21: HC emission from dual-fuelled CI engine for varying loads at constant speed of 1500 rpm.



(a) HC emission for CR of 18



(b) HC emission for CR of 12 fuel

Fig. 22: HC emission from dual-fuelled CI engine for varying IT at constant speed of 1500 rpm

IV. CONCLUSIONS

This study investigated the effects of blending diesel fuel with syngas, a gaseous fuel, on the performance of a CI engine. Trials were done with engine fuelled by diesel blended with different fractions of syngas. This was done by operating the engine in both diesel mode and dual-fuel mode and comparing the results. Through measuring BTE and BSFC it was established, that the engine developed less power in all instances when operated in dual- fuel mode compared to when fuelled by neat diesel. Noted also was that running the engine at its optimum operating conditions of CR; 18 and speed; 1500 rpm, it de-rated by about 32% for diesel mode and dual-fuel mode of SFF 100%. The same behaviour were replicated, but in differing percentage of de-rating, for other conditions of SFF. For the above conditions, the BTE values recorded were 24.03%, 19.46% and 9.81% for Neat Diesel (ND), 50% and 100% SFF respectively. This is an indication of poor fuel-energy conversion by the engine in the dual mode. Most probably caused by poor combustion of the charge in the CC.

Studies were out carried on the effect of dual fueling on the engine emissions. The results that were obtained for similar conditions as afore-stated were: CO; 1978.00, 771.00 and 2051.00 ppm, NO_x;42%, 22.40% and 33.00% and HC; 56%, 25.5% and 63.20% for neat diesel, 50% and 100% SFF respectively. The results exhibit a remarkable improvement of engine emissions when operated in dual mode. Perhaps due to the homogeneity of the charge achieved with 50% SFF in dual mode, the performance and emission are notably better.

To alleviate the drawback of performance in the dual-fuel engine, many studies have recommended use of higher CRs, supercharging/turbocharging of the engine or/and advancing of the Fuel IT. The present study, investigated the effect of varying CR and fuel IT adjustment on a CI engine. The results obtained for BTE with a fuel IT advanced to 27.4 CA were: 23.1% 26.79% and 13.05% for ND, 50%, and 100% of SFF respectively. Similarly, the CO, NO_x and HC emissions improved to 3935, 665.00 and 1888.00 ppm, 64.00%, 29.00% and 8.20% and 49.60%, 28.60% and 39.50% respectively, for the same condition. Analysing performance and emission for CR 18 and 16 or 12 shows that, respectively the BTE of ND, 50% and 100% of SFF to be 23.1% 26.79% and 13.05% for CR 18 and 20.04%, 27.04% and 11.73% for CR 16 at 1500 rpm. No clear pattern of the advantage one has over the other, perhaps due to the closeness of their cylinder pressures and temperatures during operation. Results with CR 12 indicated lowering of BTE. NO_x increased with increase of CR; at SFF of 100% for constant speed of 1500 rpm. The emission increased from 8.2% to 31.8% for CR 12 and 18, CO from 542 ppm to 1888 ppm and HC from 58.7% to 39.5% all at IT advance of 27.4 CA. Results obtained with IT retarded to 20.8 CA, all indicate poorer performance and higher emission compared to advance IT conditions

The results reported in this studies confirms that the CI engine in dual mode is a viable alternative to traditional diesel engine albeit minor adjustment to fuel IT and CR will be required in addition to supercharging of the engine.

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