PERFORMANCE AND EXHAUST GAS EMISSIONS ANALYSIS OF DIRECT INJECTION CNG-DIESEL DUAL FUEL ENGINE

RANBIR SINGH[1*]

Research Scholar, PhD Candidate

University of Delhi, Delhi, INDIA

ranbirsharma2812@gmail.com

SAGAR MAJI[2]

Professor, Research Guide

Delhi Technological University, Bawana Road, Delhi - 110042, INDIA

smaji321@yahoo.com

[1*] Corresponding author, email: ranbirsharma2812@gmail.com, Contact No. : 09868271179

Abstract:

Existing diesel engines are under stringent emission regulation particularly of smoke and particulate matter in their exhaust. Compressed Natural Gas and Diesel dual fuel operation is regarded as one of the best ways to control emissions from diesel engines and simultaneously saving petroleum based diesel fuel. Dual fuel engine is a conventional diesel engine which burn either gaseous fuel or diesel or both at the same time. In the present paper an experimental research was carried out on a laboratory single cylinder, four-stroke variable compression ratio, direct injection diesel engine converted to CNG-Diesel dual fuel mode to analyze the performance and emission characteristics of pure diesel first and then CNG-Diesel dual fuel mode. The measurements were recorded for the compression ratio of 15 and 17.5 at CNG substitution rates of 30% and 60% and varying the load from idle to rated load of 3.5kW in steps of 1 up to 3kW and then to 3.5kW. The results reveal that brake thermal efficiency of dual fuel engine is in the range of 30%-40% at the rated load of 3.5 kW which is 11%-13% higher than pure diesel engine for 30% and 60% CNG substitution rates. This trend is observed irrespective of the compression ratio of the engine. Brake specific fuel consumption of dual fuel engine is found better than pure diesel engine at all engine loads and for both CNG substitution rates. It is found that there is drastic reduction in CO, CO_2, HC, NO_x and smoke emissions in the exhaust of dual fuel engine at all loads and for 30% and 60% CNG substitution rates by employing some optimum operating conditions set forth for experimental investigations in this study.

Key Words: Compressed Natural Gas; Dual Fuel Engine; Emission Analysis; Variable Compression Ratio Engine; Brake Thermal Efficiency; Brake Specific Fuel Consumption.

1. INTRODUCTION

Impending possible energy crisis in future, rising costs and toxic emissions associated with conventional petroleum fuels have caused researchers to search out and investigate the possibility of utilization of alternate clean and non-polluting gaseous fuels for internal combustion engines. Existing diesel engines are under stringent emission regulation particularly of smoke and NO_X in their exhaust. Much interest has centered on

Compressed Natural Gas due to its potential for low particulate and NO_X emissions. Compressed Natural Gas and diesel dual-fuel operation is regarded as one of the best ways to control emissions from diesel engines and simultaneously save petroleum based precious diesel fuel. Dual fuel engine is a conventional diesel engine which burn either gaseous fuel or diesel or both at the same time. The mode of operation is defined as straight diesel if only diesel fuel is used and dual fuel if two fuels are used at the same time. In dual fuel operation the gaseous fuel is mixed with air at lean gas-air ratios and the mixture is then compressed during the compression stroke. Near the end of compression stroke, diesel fuel is injected. After a short ignition delay the combustion of diesel occurs first, igniting the natural gas and the flame propagation begins. The introduction of CNG along with intake air changes the thermodynamic and chemical properties of the mixture in the cylinder and thus the dual fuel combustion has its own characteristics on performance and emission characteristics of a dual fuel engine. The diesel fuel which acts as a source of ignition is often referred to as pilot diesel. The quantity of pilot diesel and concentration of CNG in the intake air have important effects on the performance and emissions of a dual fuel engine.

There have been several fundamental studies on dual fuel engines. [Karim G A (1983)] reviewed the prospects, problems and solutions of the dual engine of the CI engine type. [Roydon *et al*. (1991)] studied auto-ignition of pure methane and natural gas in a simulated diesel environment using a constant volume combustion vessel for the pressure and temperature ranges of 5 to 55 atm and 600 to 1700K. [Karim G A (1991)] examined some measures for improving the performance of gas fuelled diesel dual fuel engines at light load. [G.E. Doughty *et al*. (1982)] studied natural gas fueling of a diesel engine and found that for full load operation, fuel efficiency was similar to diesel operation. [Liu Z and Karim G A (1997)] developed simulation model of combustion process in gas-fuelled diesel engines. [Guowai Li *et al*. (1991)] carried out an optimization study of pilot-ignited natural gas direct-injection in diesel engine. [Youtong Z *et al.* (2003)] formulated dual fuel engine simulation model and studied the combustion process of a diesel-natural gas dual fuel engine and good levels of agreement were obtained between measured and predicted results. [Singh S *et al* (2004)] studied the combustion and emissions of a diesel-natural gas dual fuel engine and shown that dual fuel engine combustion results in significant reduction in NO_X and smoke emissions. [Karim G A *et al.* (1980)] and [Xianhua D *et al.* (1986)] have reported that at light load, dual fuel engines usually exhibit a drop in brake thermal efficiency and power output in comparison to pure diesel operation. The emissions of unburned hydrocarbons and carbon monoxide are found higher than neat diesel operation at light loads.

The main objective of the present study is to investigate the effect of Exhaust Gas Recirculation, intake air temperature, rate of injection of pilot fuel quantity, intake air throttling and substitution of CNG at two compression ratios of 15 and 17.5, on the performance parameters and emissions of a CNG-Diesel dual fuel engine.

2. MATERIAL AND METHODS

2.1 CNG as an alternate fuel for internal combustion engines

CNG has emerged as a promising alternative fuel due to its clean burning characteristics and very low amount of exhaust emissions. In petrol engines CNG is used by installing a Bi-Fuel Conversion kit and the converted engine has the flexibility of operation either on CNG or petrol. Diesel engines can also be converted to run on CNG by installing a dual fuel conversion kit or converting the existing diesel engine into SI engine. Most existing CNG vehicles use petrol engines, modified by after-market retrofit conversions and retain bi-fuel capability. Such bi-fueled converted engines generally suffer from a power loss and can encounter drivability problems, due to the design and installation of the retrofit conversion kit. Whereas single fuel engines optimized for CNG are likely to be considerably more attractive in terms of performance and emissions. In diesel engines CNG as a fuel can be used in dual fuel mode and offers the advantage of reduced emissions of NO_X, particulate matter and CO_2 while retaining the thermal efficiency of the conventional diesel engine, [Bhandari K *et al*. (2005)].

The safety aspects of converting engines to run on CNG are of great concern to users of CNG vehicles. However, CNG has four big safety features that make it an inherently safer fuel than petrol, diesel, or LPG. Its specific gravity is 0.587 which means that it is lighter than air and even if it leaks out it just rises up and dissipates into the atmosphere. Its self ignition temperature is 540^0C compared to 227-500^0C for petrol and 257^0C for diesel fuel and higher flammability limits give the gas a high dispersal rate and make the likelihood of fire in the event of a gas leak much less than for petrol or diesel. CNG has to mix with air within small range of 4 to 14% by volume for combustion to occur which is far narrower range than for petrol or diesel fuels. CNG cylinders are designed and built with special materials to the highest safety specifications, which makes its storage far safer than petrol or diesel fuel tanks.

The life of engine increases by using CNG. Lubricating oil life is extended considerably because CNG does not contaminate and dilute the crankcase oil. A big advantage of CNG is that it is virtually pollution free. CNG has a good mixture quality with air and when correct proportions are brought together they mix thoroughly and rapidly, which improves combustion efficiency of the engine. The higher Research Octane Number (130) for CNG as compared to that of petrol (87) allows a higher compression ratio (15.6:1) and consequently more efficient fuel consumption. Due to higher compression ratio Diesel engines can also use CNG as a fuel. But it cannot replace diesel completely like petrol due to poor cetane rating of CNG, [Singh R *et al.* (2012)]. Hence CNG seems a very attractive option for its use in diesel engines. The properties of CNG as a fuel for IC Engines are given in Table 1, [Sera M.A. *et al*. (2003)].

Table1: Properties of CNG as a Fuel at 25^0c and 1 atm.

CNG properties	**Value**
Density (kg/m^3)	0.72
Flammability limits (volume% in air)	4.3-15
Flammability limits (Ø)	0.4-1.6
Auto ignition temperature in air (^{0}C)	723
Minimum ignition energy (MJ)	0.28
Flame velocity (m/sec)	0.38
Adiabatic flame temperature (K)	2214
Quenching distance (mm)	2.1
Stoichiometric fuel/air mass ratio	0.069
Stoichiometric volume fraction (%)	9.48
Lower heating value (MJ/Kg)	45.8
Heat of combustion (MJ/Kg_{air})	2.9

2.2. Development of experimental test set up

A single cylinder, 04 stroke, variable compression ratio, water cooled diesel engine installed at authors internal combustion laboratory was converted to operate on dual fuel mode by carrying out minor modifications. The CNG fuel was mixed with intake air at a point in the intake manifold just outside the cylinder. The test engine is directly coupled to an electric dynamometer, which permits the engine to operate under partial monitoring conditions representing negative brake output. For any set of operating conditions, the pilot fuel was kept constant while the amount of CNG fuel was gradually increased. The ignition delay period was established from records obtained using a water-cooled piezoelectric transducer. The injection timing was established using an electric inductance transducer. The average values obtained from several consecutive cycles were used. During the tests the injection timing was kept constant and the engine was operated at 1000 RPM, under naturally aspirated conditions. In the test set up a number of measuring and other ancillary instruments were used which included: test engine, CNG conversion kit, eddy current type dynamometer, air box with orifice meter and manometer, piezo-sensor range 5000PSI with low noise cable, crank angle sensor, data acquisition device, piezo-powering unit, digital milivoltmeter, temperature sensor, temperature transmitter, load indicator, load sensor, fuel flow transmitter, air flow transmitter, engine performance analysis software, rotameter, exhaust gas analyzer, smoke meter etc. Fig.1 shows a schematic layout of engine test set up and Fig.2 shows actual image of CNG-Diesel dual fuel engine test set up. Test engine specifications are given in Table 2.

Table 2: Specifications of test engine

Engine type	**Make Kirloskar**
Bore	87.5mm
Stroke length	110mm
No. of cylinders	01
No. of strokes	04
Type of cooling	Water cooled
Rated power	3.5 kW at 1500RPM
Engine capacity	661cc
Compression ratio	17.5
Variable CR range	12 to 18
Fuel used	Diesel, CNG-Diesel in dual fuel mode

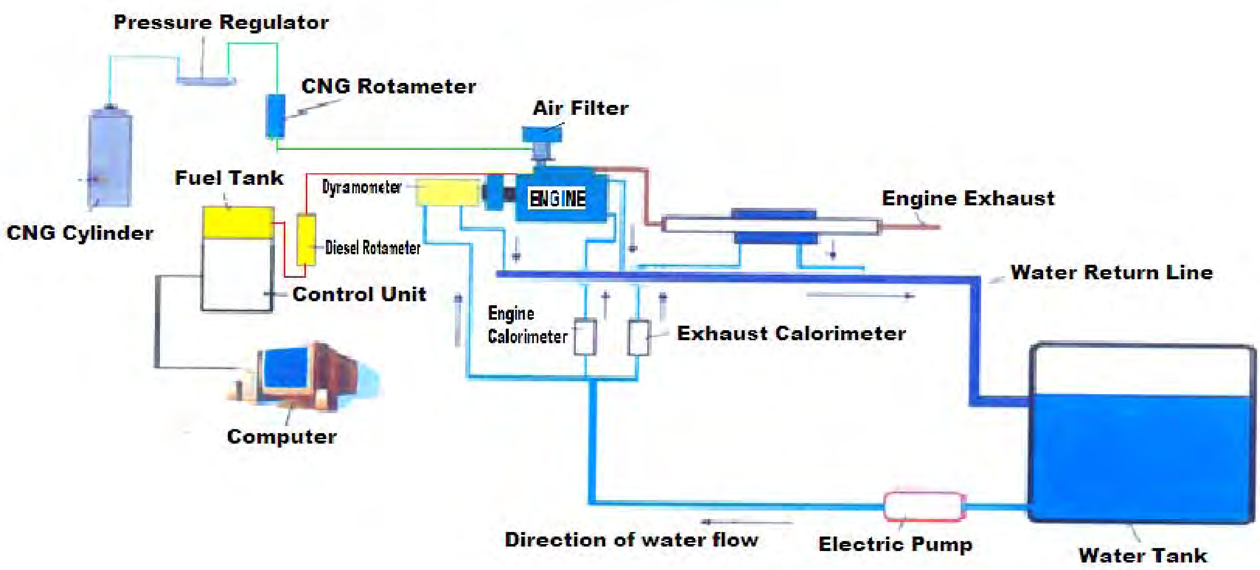

Fig.1: Schematic layout of CNG-diesel dual fuel engine test set up

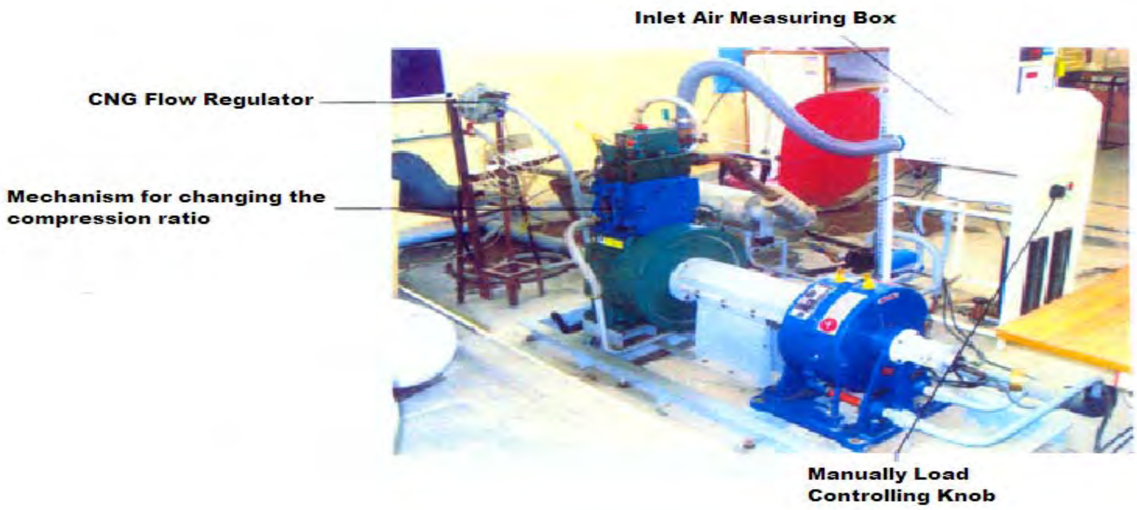

Fig.2: Actual image of CNG-Diesel dual fuel engine test set up.

2.3. Measurements and observations

A set of reading was obtained first by running the engine with diesel fuel at CR of 17.5 and varying the load from idle to rated load of 3.5kW in steps of 1 up to 3kW and then to 3.5kW. The engine performance parameters were recorded by using the Software engine Soft and other instruments.

The emissions were recorded by using Gas Analyzer (AVL Di Gas 444) and the opacity was recorded by Smoke meter (AVL437). Another set of reading was recorded for the operation of the engine in CNG-Diesel dual fuel mode. For this CNG conversion kit was switched ON and the flow of the CNG substitution rate was set at 30%. Similar set of readings were recorded for the compression ratio of 15 by changing it using the tilting head arrangement. Same set of readings were recorded for CNG substitution rate of 60%. The various performance parameters recorded were: engine load, brake thermal efficiency, brake specific fuel consumption etc. Exhaust gas emissions recorded were: CO in %, CO_2 in%, unburned hydrocarbons (UBHC) in parts per million (PPM), and oxides of nitrogen (NO_X) in PPM by using gas analyzer. Opacity of the smoke in the exhaust was measured in % by using smoke meter. As reported in the literature and during the experiments, it was observed that thermal efficiency of dual fuel engine was lower than pure diesel mode at part and low loads and emissions of CO and HC were observed higher than diesel operation. These were improved by employing larger pilot fuel quantity, using small percentage of EGR (Exhaust Gas Recirculation), increasing intake temperature and adjustment of rate of pilot fuel injection. Table 3, gives the optimum operating conditions set forth for the experimental investigations at different loads for diesel and dual fuel operation modes.

Table3: Optimum operating conditions employed for dual fuel operation.

Load, kW	**Intake Temp. K**	**Pilot fuel quantity, mg/cycle**	**Optimum EGR by volume in %**	**Throttle opening in %**
01	346	9.1	16	40
02	346	09	13	65
03	337	07	07	100
3.5	312	05	03	100

3. RESULTS AND DISCUSSION

The results obtained by performing experiments by employing optimum operating conditions mentioned in table 3, under pure diesel mode and dual fuel mode of operation are compared and analyzed by representing them graphically.

3.1 Brake thermal efficiency analysis

The brake thermal efficiency is plotted against the load applied and the curves for 30% and 60% CNG substitution rates at CR of 15 and 17.5 are plotted together both for pure diesel fuel and dual fuel modes in figures 3 and 4. It can be noticed from figure 3 that value of brake thermal efficiency of dual fuel mode with 30% CNG substitution rate, is more than diesel fuel mode by 5.11%, 5.58%, 9.77% and 10.74% at 1, 2, 3 and 3.5kW loads respectively. For 60%CNG substitution the value of brake thermal efficiency of dual fuel fuel mode is more than neat diesel mode by 9.03%,11.17%,13.43% and 12.6% at 1, 2, 3 and 3.5kW engine loads respectively. Fig.4 depicts the variation of brake thermal efficiency with variation of engine load at CR of 17.5 for the two substitution rates of 30% and 60% and at both these substitution rates, B.TH.E. of dual fuel engine is more than that of pure(0%CNG) diesel operation from no load to full load and this difference is maximum at full load and follow almost the same trend as at CR of 15 with slightly higher value at CR of 17.5.

This increase in the value of brake thermal efficiency of dual fuel engine is low at low loads but significantly high at higher engine loads because at low loads the fuel air ratio of the air-CNG mixture is very low, resulting in incomplete flame propagation and most of the fresh air-gas mixture remains unburnt. At higher engine loads, air-fuel ratio decreases, resulting in complete combustion and increase in brake thermal efficiency.

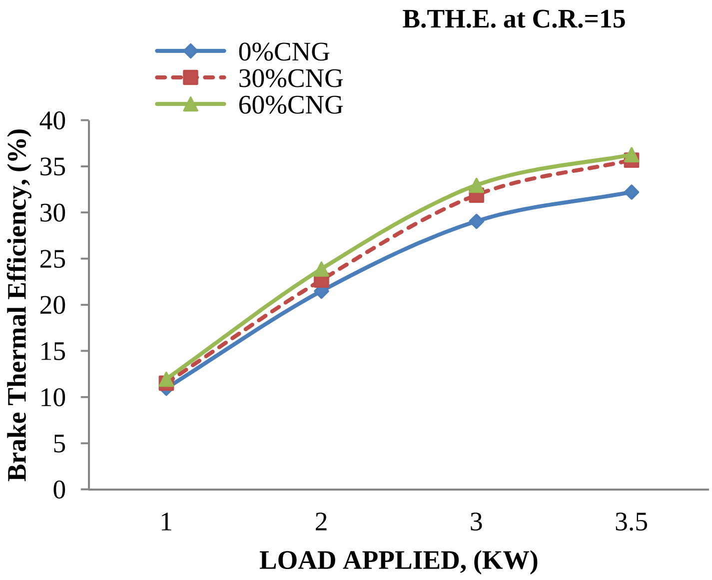

Fig.3: Brake thermal efficiency with engine load for diesel and dual fuel mode at CR=15.

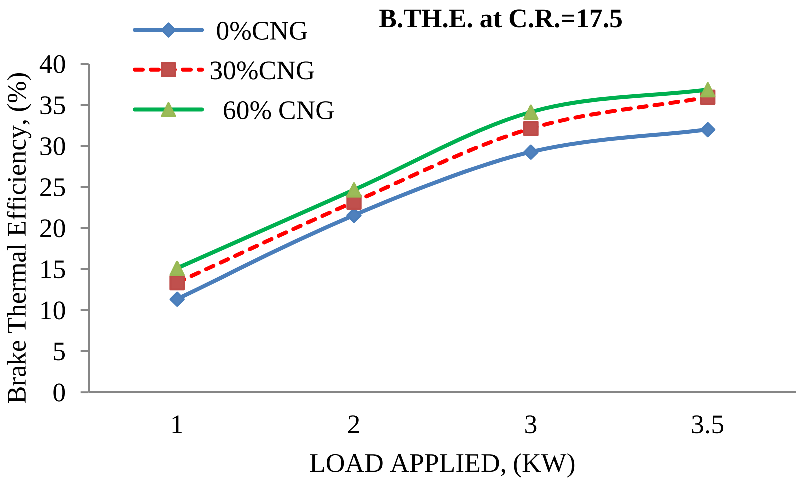

Fig.4: Brake thermal efficiency with engine load for pure diesel and dual fuel mode at C.R. =17.5.

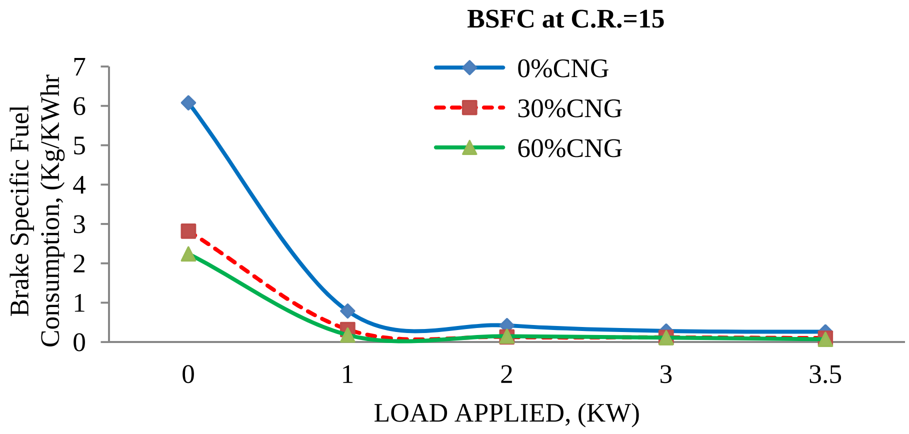

Fig. 5.:Brake specific fuel compustion with load for diesel and dual fuel mode at C.R.=15

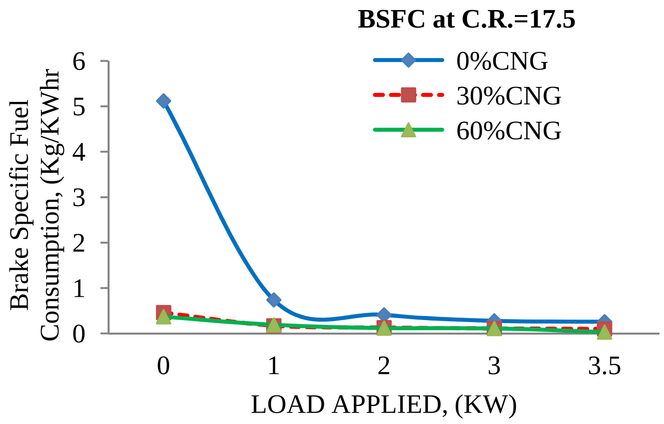

Fig.6: Brake specific fuel combustion with load for pure diesel and dual fuel modes at C.R. =17.5

3.2 Brake specific fuel consumption analysis

It is the consumption of the fuel in kg/kWhr of the brake output of the engine. The observations for BSFC for base diesel and dual fuel modes at CR of 15 and 17.5 and for 30%CNG, 60%CNG substitution rates were recorded and are represented graphically for analysis. From figure 5 it is observed that BSFC with 30%CNG dual fuel mode is less than pure diesel mode by 59.49%, 69.04%, 57.14%, 61.54% at 1, 2, 3 and 3.5kW engine loads respectively at compression ratio of 15. The same trend is observed for 60%CNG substitution rate. As the compression ratio is further increased to 17.5, BSFC value for dual fuel mode is further decreased by 77.02%, 68.29%, 60.71% and 61.54% at 1, 2, 3, 3.5kW loads respectively at 30%CNG substitution rate in comparison to pure diesel mode as depicted in figure6. It further decreases as the substitution rate of CNG is further increased to 60%. With CNG replacing diesel fuel, it contributes to extra heat energy on combustion, resulting in better BSFC than diesel mode. This could also be due to higher calorific value of CNG, better mixing of air and CNG and improved combustion efficiency.

4.EXHAUST GAS EMISSIONS ANALYSIS

Exhaust gas emissions for pure diesel and dual fuel modes were measured experimentally by exhaust gas analyser. The emissions recorded were CO, CO_2, UBHC and NO_X. The smokemeter was used to measure the opacity of the smoke.

4.1 CO Emissions analysis

Figures 7 and 8 show the effect of engine load, CNG substitution rates and changing compression ratio on CO emission concentration in diesel and dual fuel modes. It can be noticed from figure 7, that CO emissions decrease as the load on the engine is increased for diesel and dual fuel modes. CO emissions for 30% and 60% CNG substitution rates are lower than pure diesel mode in the range of 33.3%-61.9% for different engine loads at C.R.=15. There is more reduction in CO emissions at higher compression ratio of 17.5 for both CNG substitition rates in dual fuel mode as compared to pure diesel mode as shown in figure 8.This reduction in CO emissions for duel fuel operation is due to the less injected diesel fuel and its relacement with a clean burning CNG fuel.

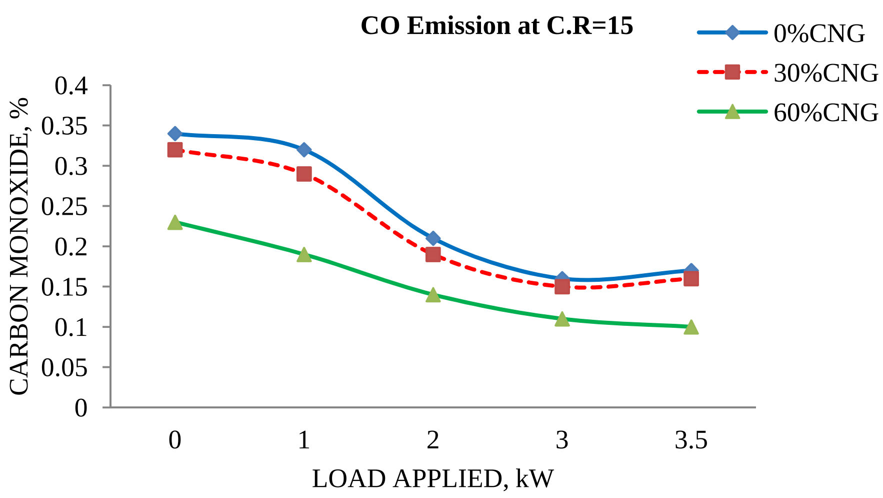

Fig.7: Carbon monoxide emission with load applied for diesel and dual fuel modes at C.R =15.

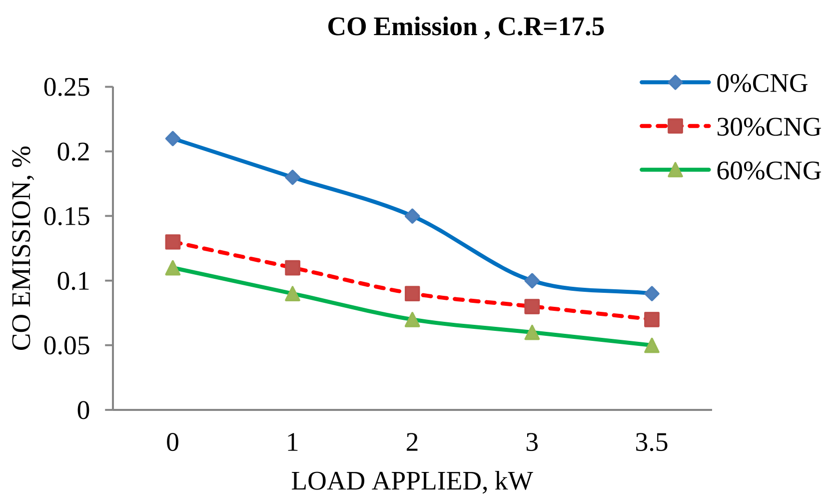

Fig.8: CO Emission with load applied for pure diesel and dual fuel modes at C.R=17.5.

4.2 HC Emissions analysis

Figures 9 and10 depict the variation of unburned HC emissions for diesel and dual fuel (with 30%CNG and 60%CNG) modes at the two compression ratios of 15 and 17.5 respectively. It is very clear from both the figures that HC concentration in the exhaust decrease with load applied for both diesel and dual fuel modes but

HC emissions are lesser for dual fuel mode than pure diesel mode by 14.55% at 1kW load and by 18.30% at 3.5kW load and by 28.16% at 1kW load and by 30.72% at 3.5kW load for 30% and 60% CNG substitution rates respectively at compression ratio of 15 as shown in figure 9. As the compression ratio is increased to 17.5, HC emissions are lower for dual fuel engine than diesel mode by 14.63% at 1kW load and 37.29% at 3.5kW load for 30% CNG substitution rate and by 17.07% at 1kW load and 44.92% at 3.5kW load for 60% CNG substitution rate as depicted in figure 10. With the use of small percentage of EGR in the engine cylinder in dual fuel mode, intake air temperature increased and as a result unburned hydrocarbon emissions in the engine exhaust decreased. At higher loads, higher compression ratio and higher CNG substitution rates, HC emissions further reduced for dual fuel operation because at these conditions the delay period decreases for pilot fuel and combustion of CNG-air mixture become fast and complete and very less amount of unburned fuel go into the engine exhaust.

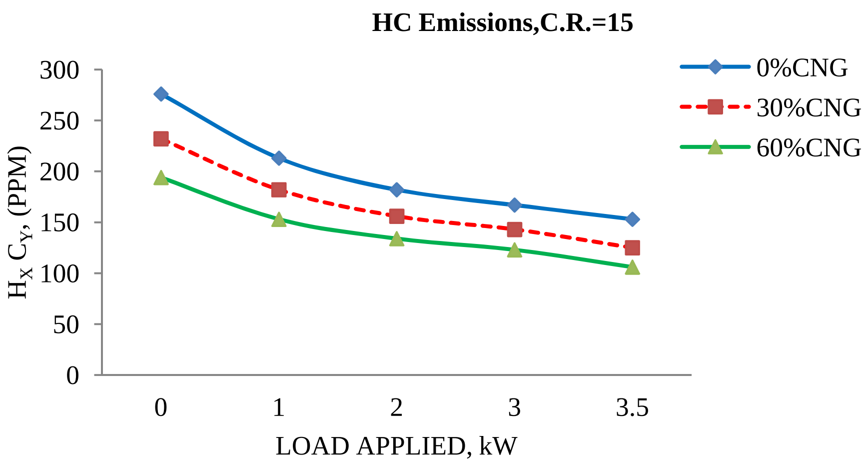

Fig.9: Hydrocarbon (ppm) emissions for diesel and dual fuel modes at C.R=15

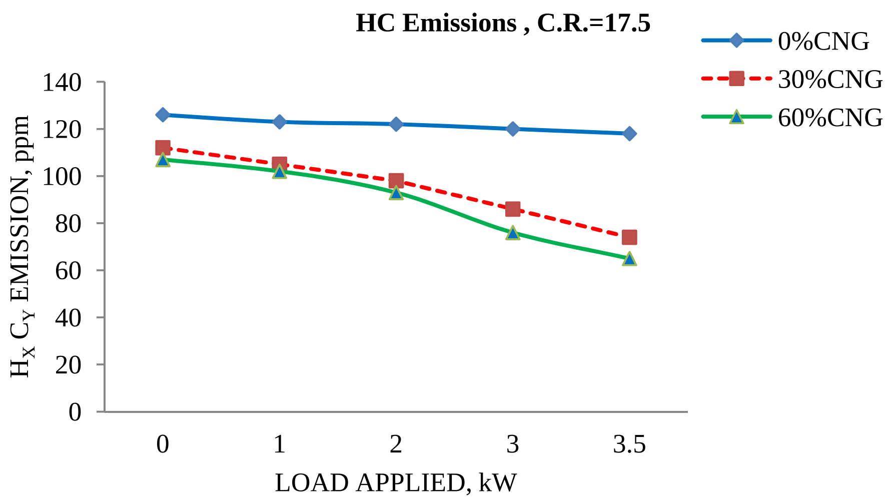

Fig.10: Hydrocarbon (ppm) emissions for diesel and dual fuel modes at C.R=17.5

4.3 CO_2 Emission analysis

Figures11 and 12 shows the effects of varying engine load on emission of carbon dioxide (CO_2). It can be observed that level of emission of CO_2 increase with increasing load both for diesel and dual fuel modes but its value for dual fuel mode is decreased by 17.24% at low load of 01kW and by 27.65% at rated load of 3.5 kW for C.R.=15 and 30% CNG substitution. At the same compression ratio, the concentration of these emissions further decrease by 44.83% for low load and by 59.57% at higher engine loads with 60% CNG substitution. CO_2 emissions further decrease with increase in compression ratio and CNG supply. This is evident from figure 12 that almost same trend is observed for 30% and 60% CNG dual fueling at higher compression ratio of 17.5. This reduction in levels of CO_2 emissions on dual fueling a converted diesel engine is beneficial in the sense that CO_2 is a greenhouse gas and its concentration in the atmosphere should be minimum. The main factors for reduction of CO_2 in exhaust of a dual fuel engine include improper conversion of CO to CO_2 due to decrease in peak temperature because of lower adiabatic flame temperature of CNG than diesel and decreased diesel fuel quantity.

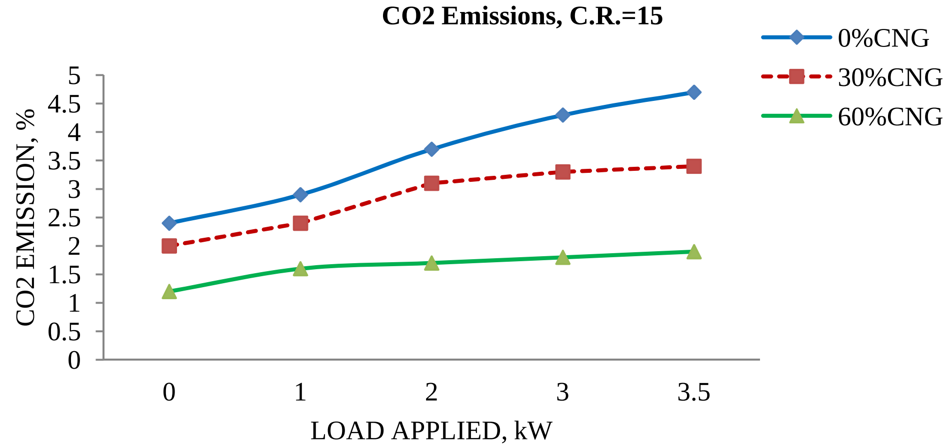

Fig. 11: CO_2 Emissions for diesel and dual fuel mode of operation at C.R. =15.

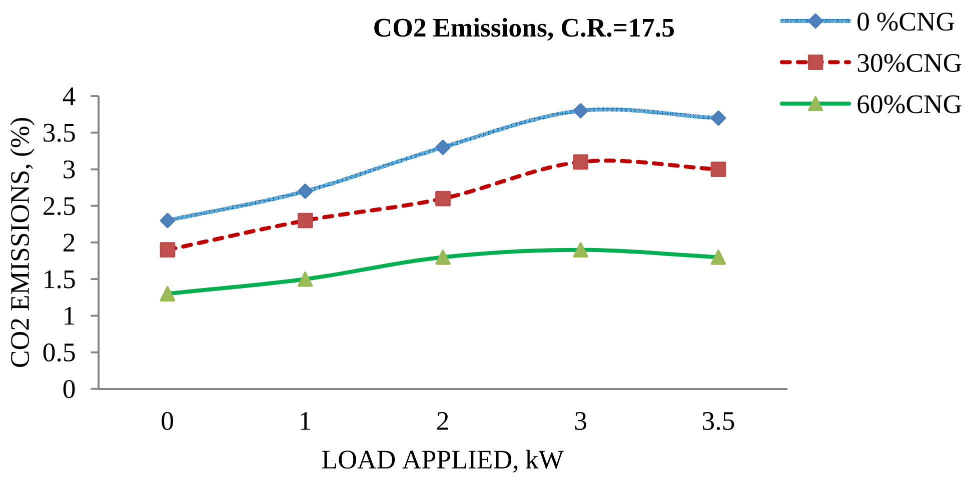

Fig.12: CO_2 Emissions for diesel and dual fuel mode of operation at C.R. =17.5.

4.4 NO_X Emissions analysis

With the introduction of gaseous fuels NO_X levels are found to be low. Figures 13 and 14 represent the effect of engine load, changing compression ratio and CNG substitution rates on NO_X emission formed inside engine cylinder for diesel and dual fuel modes. It is clear from figure 15 &16, that NO_X level increase with increase of engine load and compression ratio for both diesel and dual fuel modes but in dual fuel mode NO_X emissions are drastically reduced by 12.5% and 18.75% at low loads for 30% CNG and by 42.36% and 76.94% at high engine loads for 60%CNG at C.R. =15. NO_X concentration is further reduced by 43.78% and 77.83% at low loads for 30%CNG and 40.45% and 84.76% at rated load of 3.5kW for 60%CNG at C.R.=17.5 as depicted infigure14.

High peak temperatures and availability of oxygen are the two main factors for the formation of NO_X and it is directly related to adiabatic flame temperature. So as the CNG is introduced NO_X emissions decrease and as CNG supply is increased, NO_X further decrease. This decrease in NO_X in dual fuel engine is a positive merit in view of environmental concerns.

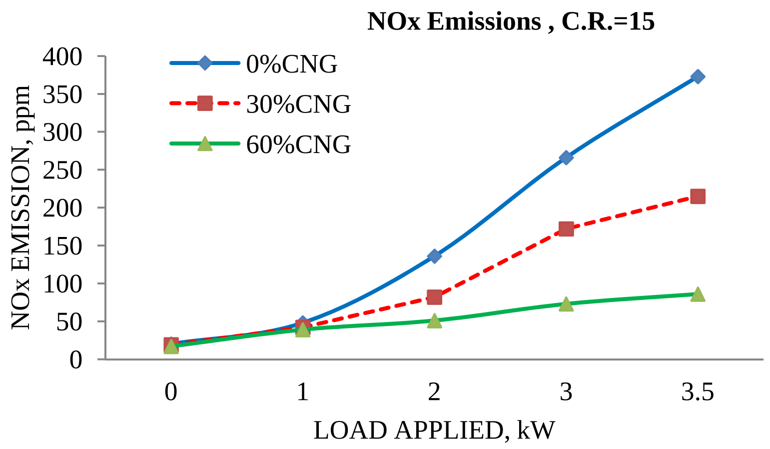

Fig.13: NO_X Emission as a function of engine load for diesel and dual fuel modes at C.R. =15.

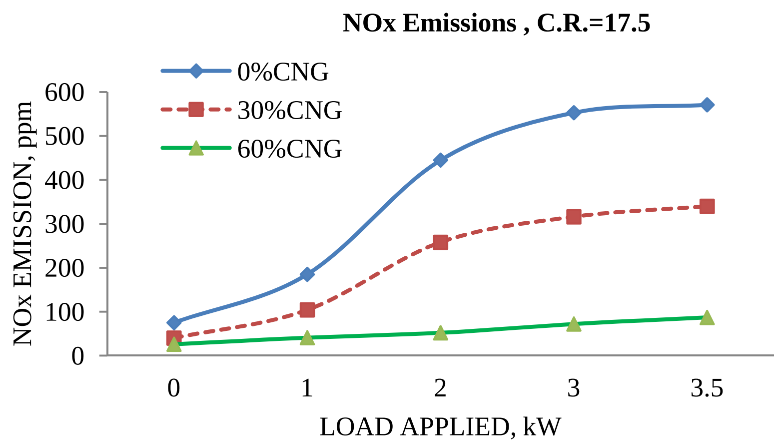

Fig.14: NO_X Emission as a function of engine load for diesel and dual fuel modes at C.R. =17.5.

4.5 Smoke opacity analysis

Smoke opacity means the degree to which the smoke reduces the passage of light. It means more smoke in the exhaust will have high smoke opacity and vice-versa in the context of diesel emissions. Figures 15 and 16, shows the effects of engine loads, variation of compression ratio and changes in CNG substitution rates on smoke opacity for diesel and dual fuel modes. Smoke opacity for pure diesel engine is inversely proportional to compression ratio i.e. if compression ratio increases smoke opacity decreases and if compression ratio decreases smoke opacity increases. Experimental results confirm this fact that its value for compression ratio of 15 is above 60% whereas for compression ratio of 17.5, its value decreases to 5.8% at the same engine load of 3.5kW. From figure 15, it can be seen that smoke opacity decreases by 25.61% at low loads and by 54.83% at rated load of 3.5kW for 30%CNG substitution and further decreases by 84.14% at low loads and 86.25% at the rated load of 3.5kW for 60% CNG substitution for compression ratio of 15. With increase in compression ratio from 15 to 17.5, results of smoke opacity are depicted in figure 16, and its value for 30%CNG reduces by 50% at low load and 84.48% at rated load of 3.5kW and it further decreases by 71.43% at low loads, 93.10% at 3.5kW load for 60%CNG supply in comparison to its values for pure diesel mode.

This decrease in smoke level with dual fueling the existing diesel engine is a positive merit in favor of dual fuel engines because diesel engines smoke reduction is the main cause of concern for researchers, manufacturers and users. The main factors of decrease in smoke emissions due to dual fueling of a diesel engine include, reduced injected diesel fuel, complete and smooth combustion of clean CNG fuel. In dual fuel engine, a flame front is formed by the ignition of small quantity of pilot fuel which sweeps the homogeneous mixture of CNG and air and exhaust contains less unburned fuel and hence less smoke. Moreover, soot particles form primarily from the carbon in the diesel fuel and in CNG, hydrogen/carbon ratio is high because of presence of smaller hydrocarbon as compared to diesel, soot formation is less and as a result Particulate Matter (PM) emission will also decrease with the use of CNG.

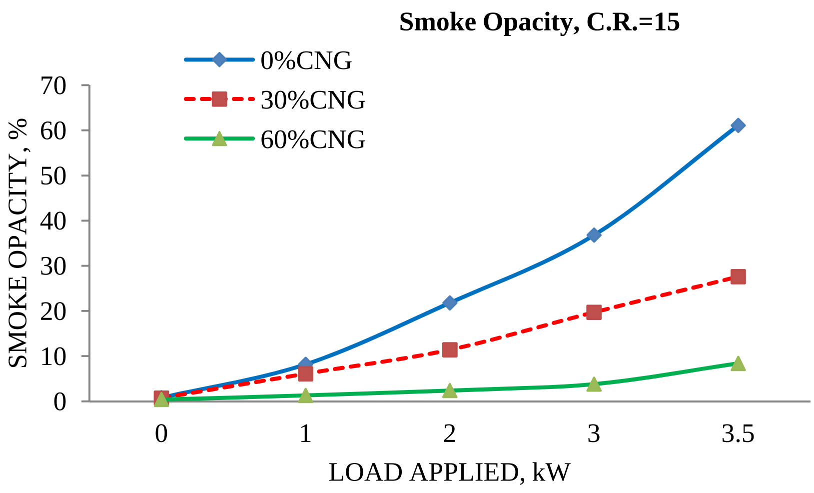

Fig. 15: Smoke opacity analysis for diesel and dual fuel modes at C.R. =15.

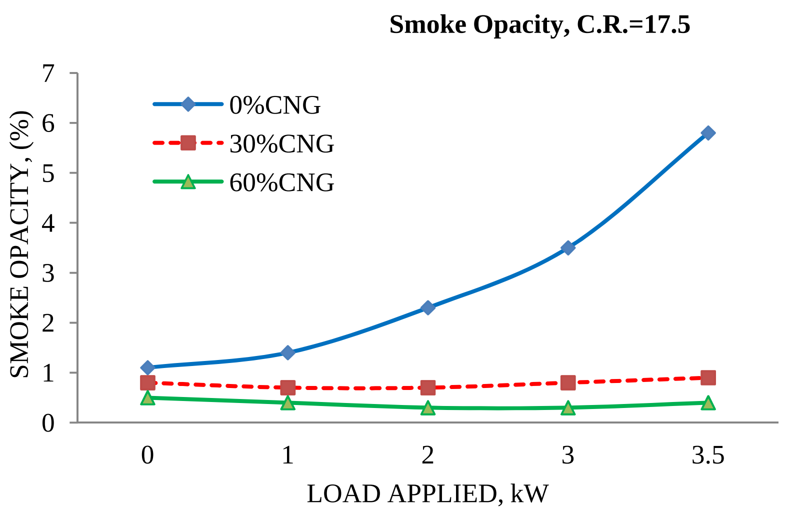

Fig. 16: Smoke opacity analysis for diesel and dual fuel modes at C.R. =17.5

5. CONCLUSION

In the present experimental research on performance and emissions on use of CNG in a converted direct injection conventional diesel engine in dual fuel mode, thorough investigations on two different compression ratios of 15 and 17.5 for 30% and 60%CNG substitution rates have been carried out at optimum operating conditions mentioned in table 3, the main conclusions are summarized below:

(1) Brake thermal efficiency of dual fuel mode with 30% CNG substitution rate, is more than diesel fuel mode by 5.11%, 5.58%, 9.77% and 10.74% at 1, 2, 3 and 3.5kW loads respectively. For 60%CNG substitution the value of brake thermal efficiency of dual fuel mode is more than neat diesel mode by 9.03%,11.17%,13.43% and 12.6% at 1, 2, 3 and 3.5kW engine loads respectively. For the two CNG substitution rates of 30% and 60%, brake thermal efficiency of dual fuel engine is more than that of pure diesel operation from no load to full load and this difference is maximum at full load and follow almost the same trend as at CR of 15 with slightly higher value at CR of 17.5.

(2) It is observed that BSFC with 30%CNG dual fuel mode is less than pure diesel mode by 59.49%, 69.04%, 57.14%, 61.54% at 1, 2, 3 and 3.5kW engine loads respectively at compression ratio of 15. The same trend is observed for 60%CNG substitution rate. As the compression ratio is further increased to 17.5, BSFC value for dual fuel mode is further decreased by 77.02%, 68.29%, 60.71% and 61.54% at 1, 2, 3, 3.5kW loads respectively at 30%CNG substitution rate in comparison to pure diesel mode. It further decreased as the substitution rate of CNG was increased to 60%.

(3) CO emissions for 30% and 60% CNG substitution rates are lower than pure diesel mode in the range of 33.3%-61.9% for different engine loads at C.R.=15. There is more reduction in CO emissions at higher compression ratio of 17.5 for both CNG substitition rates in dual fuel mode as compared to pure diesel mode.

(4) HC emissions are lesser for dual fuel mode than pure diesel mode by 14.55% at 1kW load and by 18.30% at 3.5kW load and by 28.16% at 1kW load and by 30.72% at 3.5kW load for 30% and 60% CNG substitution rates respectively at compression ratio of 15. As the compression ratio is increased to 17.5, HC emissions are lower for dual fuel engine than diesel mode by 14.63% at 1kW load and 37.29% at 3.5kW load for 30% CNG substitution rate and by 17.07% at 1kW load and 44.92% at 3.5kW load for 60% CNG substitution rate.

(5) CO_2 emissions increase with increasing load both for diesel and dual fuel modes but its value for dual fuel mode is decreased by 17.24% at low load of 01kW and by 27.65% at rated load of 3.5kW for C.R.=15 and 30% CNG substitution. At the same compression ratio, the concentration of these emissions further decrease by 44.83% for low load and by 59.57% at higher engine loads with 60% CNG substitution. CO_2 emissions further decrease with increase in compression ratio and CNG supply.

(6) In dual fuel mode NO_X emissions are drastically reduced by 12.5% and 18.75% at low loads for 30% CNG and by 42.36% and 76.94% at high engine loads for 60%CNG at C.R.=15. NO_X concentration is further reduced by 43.78% and 77.83% at low loads for 30%CNG and 40.45% and 84.76% at rated load of 3.5kW for 60%CNG at C.R.=17.5.

(7) Smoke opacity in dual fuel mode decreased by 25.61% at low loads and by 54.83% at rated load of 3.5kW for 30%CNG substitution and further decreases by 84.14% at low loads and 86.25% at the rated load of 3.5kW for 60% CNG substitution for compression ratio of 15. With increase in compression ratio from 15 to 17.5, its value for 30%CNG reduces by 50% at low load and 84.48% at rated load of 3.5kW and it further decreases by 71.43% at low loads, 93.10% at 3.5kW load for 60%CNG supply in comparison to its values for pure diesel mode.

In view of the above positive results obtained in this experimental research in favor of CNG-Diesel dual fuel engine on performance and emissions, it can be concluded that it is a promising technology for achieving better thermal efficiency and controlling both NO_X and smoke emissions in existing conventional compression ignition engines with minor engine hardware modifications, thus great saving of precious diesel fuel and saving the human and plant life from the hazardous effects of exhaust gas pollutants from the conventional diesel engines.

REFERNCES

[1] Bhandari, K et al. (2005): Performance and emissions of natural gas fueled internal combustion engine, A review, *JSIR Volume* **64** 333-338.

[2] Doughty, G E *et al*. (1992): Natural gas fuelling of a caterpillar 3406 diesel engine. *ASME* J *Engg gas turbine and power*.

[3] Guowei, L *et al*. (1999): Optimization study of pilot-ignited natural gas direct-injection in diesel engines, *SAE Paper No.* 1999-01-3556. 1739-1748
[4] Karim, G A (1983): The dual fuel engine of the compression ignition type –prospects, problems and solutions –a review. *SAE Paper No*. 831073.
[5] Karim, G A (1991): An examination of some measures for improving the performance of gas fuelled diesel engines at light load. *SAE Paper No.* 912366.
[6] Karim, G A and Burn, K S (1980): The combustion of gaseous fuels in a dual fuel engine of compression ignition type with particular reference to cold intake temperature conditions, *SAE Paper No*.800263.
[7] Liu, Z and Karim, G A (1997): Simulation of combustion process in gas-fuelled diesel engines. *Proc. Instn. Mech. Engrs. Volume,* **211** Part A, 159-169.
[8] Roydon, A. Fraser (1991): Auto ignition of methane and natural gas in a simulated diesel engine environment. *SAE Paper No*. 910227.
[9] Sera, M.A et al. (2003): CNG engine performance improvement strategy through advanced intake system, *SAE Paper No*. 2003-01-1937
[10] Singh, R and Sagar, M (2012), Dual fueling of a twin-cylinder compression ignition engine with diesel and CNG, *Journal of engineering and applied sciences*, **7,** 90-99
[11] Singh, S *et al.* (2004): Modeling and experiments of dual fuel engine combustion and emissions, *SAE Paper No*. 2004-01-0092
[12] Xianhua, D and Philip, H (1986): Emissions and fuel economy of a pre-chamber diesel engine with natural gas dual fueling, *SAE Paper No*. 860069.
[13] Youtong, Z *et al.* (2003): Modeling and simulation of a dual fuel (diesel/natural gas) engine with multidimensional CFD, *SAE Paper No*. 2003-01-0755, 336-346.