Effect of Exhaust Gas Recirculation on the Performance and Emissions of a Dual Fuel Engine Operated on CNG-Biodiesel-Ethanol Blends

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Abstract

Emissions from engine exhaust is a serious problem for environment point of view. The search for an alternative fuel, which promises a harmonious correlation with sustainable development, energy conservation, management, efficiency, and environmental preservation, has become highly pronounced in the present context. Dual fuel mode of operation employing CNG (Compressed Natural Gas) and plant oils like Honge and Jatropha oils and their esters is an attractive option as our country has a large agriculture base that can be a feed stock to this fuel technology and can ease the burden on the economy by curtailing the fuel imports. This paper presents the results of investigations carried out in studying the behaviour of Honge and Jatropha oils methyl esters [HOME/JOME] and their blends with Ethanol and subsequent testing of these oils in a four stroke, single cylinder, water cooled, direct injection CI engine in dual fuel mode with CNG induction. This paper studies the effects of ethanol addition and Exhaust Gas Recirculation(EGR) and on performance ,combustion and emissions of compressed natural gas (CNG) dual fuelled with Honge oil methyl esters (HOME) and Jathropha methyl ester (JOME) in a dual-fuel engine. From the experimental evidence it is found that combustion of HOME/JOME plus up to 15% ethanol blended with CNG in a dual fuel engine operated with optimized parameters of injection timing of 27° BTDC and compression ratio of 17.5 with 10% EGR results in acceptable combustion emissions and improved brake thermal efficiencies.

Index Terms: Honge oil methyl ester, Jatropha oil methyl ester, Ethanol blends, Compressed natural gas, Induction, dual fuel engine.

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1. INTRODUCTION

Stringent environmental policies, reduction in underground fossil fuel, escalating prices and increased demand for energy have triggered interest in more advanced and novel combustion technologies that use renewable and alternative fuels as energy sources. Compressed natural gas (CNG) in a diesel engines employing small biodiesel pilot to ignite a premixed CNG–air mixture have received considerable attention globally. Compression ignition (CI) engines have better thermal efficiency and high power output with higher soot and nitric oxide emission levels and have the ability to use high-quality renewable fuels that can be produced efficiently from biomass. Renewable and alternative fuels have numerous advantages compared to fossil fuels as they are renewable, biodegradable, provide food and energy security and foreign exchange savings besides addressing environmental concerns, and socio-economic issues as well **[1-8].** The main drawback of CI engine is soot and nitric oxide emission levels; it can be overcome by operating the CI engine on dual fuel mode. Biodiesels derived from vegetable oils gives slightly lower performance with reduced emission levels [9, 21, 22, 36-38]. Effect of different engine parameters on the performance and emissions of a single cylinder diesel engine using biodiesel and blends with diesel fuel were experimentally investigated [6, 9, 33, 42,45]. Similarly, dual fuel engine operation with compressed natural gas (CNG) is investigated by several researchers and higher thermal efficiency with reduced emission levels at higher compression ratio and advanced injection timing has been reported. CNG has better ignition qualities such as high octane number, and is therefore suitable for engines with a relatively high compression ratio [10, 17, 37-39]. The high octane number of CNG permits use of higher compression ratio without engine knocking. It can be used either as a sole fuel in spark ignition engine or can be dual fuelled with liquid fuel injection in CI engines. Due to its inability to use directly in a diesel engine, low cetane number, higher self ignition temperature, CNG cannot be operated directly in a diesel engine without injection of a small amount of diesel/biodiesel because CNG will not ignite under the prevailing conditions of temperature and pressure. However, CNG can partially substitute the diesel/biodiesel in dual-fuel engines and but have drawn considerable research attention because use of CNG can save about 70 – 90% liquid fuel and benefited in the area of pollution control [6-8,6-9]vantage of CNG operated dual fuel engine include no major modifications are required in the existing diesel engine and has a flexibility to switch back to the liquid fuel mode of operation as and when need arise. Dual-fuel mode of operation employing CNG and biodiesels of plant oils such as honge and jatropha oils is an attractive option as our country has a large agriculture base that can be a feedstock to this fuel technology and can help in improving the economy by curtailing the fossil fuel imports [7,17-23,34,38]. Use of different biodiesels as injected fuel along with CNG induction in dual-fuel mode highlights the performance of dual-fuel engine with various engine parameters that have been investigated by other researchers. Different methods of CNG utilization in diesel engines have been reported in the literature 24,25 ,26,29]. Natural gas has high octane number and therefore is suitable for engines with relatively high compression ratio [38-40]. In general, gas is used in spark ignition engines because of its relatively higher octane level. However, the compression ratio of this type of engine cannot be as high as that of diesel engine because of occurrence of knocking. To overcome this disadvantage, dual-fuel combustion system that utilizes combined diesel/biodiesel and natural gas fuel has been proposed in recent years [27,37,38,40]

In dual fuel engine, mixture of natural gas and air is induced in engine cylinder and is compressed during compression stroke. This air fuel mixture is ignited by injecting small quantity of diesel or non-edible oil, called as pilot injection in cylinder at the end of compression stroke [37,47]. This diesel or non-edible oil pilot fuel ignites due to heat of compression, just like in diesel engine. Burning of diesel or non-edible oil pilot fuel further ignites and burns compressed natural gas in the cylinder and power is produced[10-12,45] Several engine parameters have been found to affect the performance of combustion in gas engines [25,36,48]. These include injection timing [37,45]. compression ratio [12,45,48]exhaust gas recirculation [19,26,35,41,49].Various methods of employing ethanol-diesel/biodiesel dual fuel operation have been developed to enhance the engine performance with lowered emissions [10-12,32].

In this context, the experimental investigations were carried out on a single cylinder, four-stroke, water cooled, direct injection diesel engine, developing a power output of 5.2 kW at 1500 rev per minute in dual fuel mode with various fuel combinations of HOME,JOME and their blends with Ethanol as injected fuels along with CNG as the inducted fuel. The main objective of this study involves performance evaluation of single cylinder, four stroke direct injection CI engine operated on dual fuel mode using CNG with pilot injection of locally available biodiesel-ethanol blends as an injected fuel that replaces fossil diesel fuel.

2. FUEL PROPERTIES

The important properties of CNG, Honge and Jatropha oils methyl esters [HOME/JOME] and their blends with Ethanol used are found by standard methods and compared with diesel. The results show that the heating value of vegetable oil is comparable to the diesel oil, but it is slightly lower than diesel oil. However, Kinematic viscosity of neat vegetable oil is several times higher than diesel oil and the transesterified oils of Honge and Jatropha show considerable reduced viscosities[1,3,5-8].In Honge oil, oleate, Linoleate and linolenate fatty acids are said to be unsaturated and palmitic and stearate fatty acids are saturated. Saturated methyl esters possess favorable features like higher cetane number and heating value compared to their unsaturated counterparts, but it also has a higher viscosity and pour point, which is not desirable during engine operations under cold climatic conditions. The Jatropha oil mainly contains 27.1% of saturated fatty acids (SFA), 40.8% of mono-unsaturated fatty acids (MUFAs) and 32.1% of ploy–unsaturated fatty acids (PUFAs). Jatropha biodiesel is a mixture of methyl esters and mainly contains oleic (C18:1)-40.8%, linoliec (C18:2)-32.1%, palmitic (C16:0)-15.6%, and stearic (C18:0)-9.7% [Kazi (2007)].The ignition delay is affected by molecular structure of fatty acids present in the oil. Longer fatty acid chain length, higher saturation and an increase in the chain length of alcohol moiety decrease the ignition delay [3-8].

CNG is produced from gas wells or tied in with crude oil production. CNG is primarily made up of methane (CH4), but frequently contains trace amounts of ethane, propane, nitrogen, helium, carbon dioxide, hydrogen sulphide and water vapour. Methane is the principal component of CNG. Normally, more than 90% of CNG is methane. CNG can be compressed, so it can be stored and used as CNG. CNG is safer than gasoline in many respects, and ignition temperature of CNG is higher than that of gasoline and diesel fuel. In addition, CNG is lighter than air and will dissipate upward rapidly if a rupture occurs. Gasoline and diesel will pool on the ground, increasing the danger of fire. CNG is non-toxic and will not contaminate groundwater if spilled. The various properties of diesel and vegetable oils are shown in Table 1. Table 2 shows the properties of Natural Gas. \

Table- 1. Properties of diesel, Honge oil, Jatropha oil and their respective esters and Biodiesel-Ethanol blends.

Sl No	Properties	Diesel	HOME	JOME	Ethanol	HOME+ 15% Ethanol	JOME+ 15% Ethanol
1	Viscosity @ 40 °C (cst)	4.59	5.6	4.84	1.2	3.7	3.25
2	Flash point °C	56	163	192	13.5	32	33
3	Calorific Value in kJ / kg	45000	36010	35,200	27300	35550	34060
4	Density kg / m³	830	890	880	780	----	-----
5	Cetane Number	45-55	40-42	40-45	----	----	----

Table- 2 Properties of Natural Gas [12]

Properties	Natural Gas
Boiling range (K @101325Pa)	147
Density (kg/m³) at 1 atm. & 15^0 C	0.77
Flash Point (K)	124
Octane Number	130
Flammability Limits Rich Lean	 0.5873 1.9695
Flame Speed (cm/s)	33.80
Net Energy Content (MJ/kg)	49.5

Auto Ignition Temperature (K)	923 (650^{0}C)
Combustion Energy (KJ/m³)	24.6
Vaporization energy (MJ/m³)	215 – 276
Stoichiometric A/F	17:1

3.0 EXPERIMENTAL SETUP

Figure 1 shows overall view of C.I. Engine Test Rig with dual fuel arrangement. The engine tests were conducted on a four stroke single cylinder direct injection water-cooled compression ignition (CI) engine. The specifications of the engine are given in Table 4. To prepare ethanol-blended fuels, two fuels of biodiesel and ethanol were used. The biodiesels were blended with ethanol to get four different fuel blends each ranging from 0 to 15 per cent with an increment of 5 per cent. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture is homogeneous. A stirrer was mounted inside the fuel tank in order to prevent phase separation. Figure 2 shows fuel schematic layout of a C.I. Engine Test Rig with dual fuel arrangemen. Experiments were conducted on the engine with 80%and 100% loads and at a constant rated speed of 1500 rpm. There was no special arrangement to control the engine speed; the regular governor of the engine was used to control the engine speed. The engine had been provided with a hemispherical combustion chamber with overhead valves operated through push rods. Cooling of the engine was accomplished by circulating water through the jackets of the engine block and cylinder head. Experiments were conducted under a thermal steady-state condition of the engine with an inlet cooling water temperature of 80^{o}C. A piezoelectric pressure transducer was mounted flush with the cylinder head surface to measure the cylinder pressure. The cylinder pressure was measured with Piezo electric transducer fitted in the cylinder head. Figure 2 shows an arrangement to mix ethanol to biodiesel to form homogenious mixture. Figure 3 shows carburetor holder with different venturi's designed for proper gas and air mixing. Exhaust Gas Recirculation [EGR] arrangement was suitably designed and fabricated and was fitted to the engine. Figure 4 shows exhaust gas recirculation arrangement. Exhaust gas opacity was measured using the Hartridge smoke opacity meter. The exhaust gas composition was measured using an exhaust gas analyzer .
The measured emissions include CO, HC and NO_X. The basic principle for measurement of CO2, CO, HC, NO emissions is non-dispersive infrared technology and electrochemical method for NO_X and oxygen measurement. Table 4 and 5 shows the specifications of the exhaust gas analyzer and smoke meter with measurement accuracies and uncertainties. In order to reduce error in measurement of emissions five readings were recorded and averaged out readings are only presented in the graphs.

Sl. No.	Parameters	Specification
1	Type	TV, 1 Kirlosker made)
2	Governor type	Mechanical
3	Number of cylinders and number of strokes	Single cylinder, Four stroke
4	Rated power	5.2 KW (7 HP)
5	Cylinder Bore and	0.0875 m, 0.11 m
6	Compression ratio	17.5 : 1
7	Type of Dynamometer	Eddy current

Table- 3 Specifications of the Engine

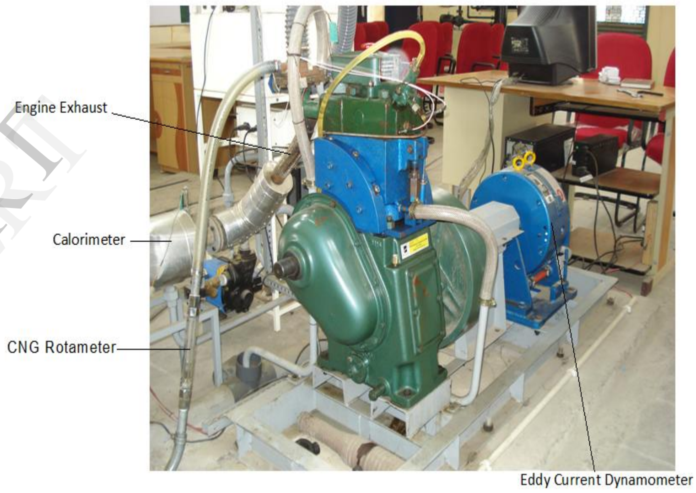

Fig- 1 Engine test rig with dual fuel arrangement.

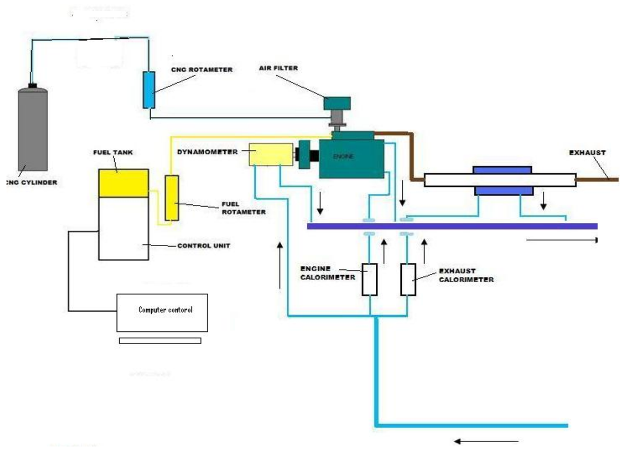

Fig- 2 Schematic **Layout of VCR Engine Test Set Up**

Fig- 3 Carburetor holder with venture

Fig-4. Exhaust gas recirculation arrangement

Table- 4 Specifications of the Exhaust gas analyser

Type	DELTA 1600S
Object of measurement	Carbon monoxide (CO), carbon dioxide (CO_2) and hydrocarbons (HC)
Range of measurement	HC = 0–20,000 ppm as C_3H_8 (propane) CO = 0–10% CO_2 = 0–16% O_2 = 0–21% NO_x = 0–5000 ppm (as nitric oxide)
Accuracy	HC = ±30 ppm HC CO = ±0.2% CO CO_2 = ±1% CO_2 O_2 = ±0.2% O_2 NO_x = ±10 ppm NO
Resolution	HC = 1 ppm CO = 0.01 vol.% CO_2 = 0.1 vol.% O_2 = 0.01 vol.% NO_x = 1 ppm
Warm-up time	10 min (self-controlled) at 20°C
Speed of the response time	Within 15 s for a 90% response
Sampling	Directly sampled from the tail pipe
Power source	100–240 V AC/50 Hz
Weight	800 g
Size	100 mm × 210 mm × 50 mm

Table- 5 Specifications of the Smoke meter

Type	Hartridge smoke meter-4
Object of measurement	Smoke
Measuring range opacity	0–100%
Accuracy	±2% relative
Resolution	0.1%
Smoke length	0.43 m
Ambient temperature range	−5°C to +45°C
Warm-up time	10 min (self-controlled) at 20°C
Speed of response time	Within 15 s for 90% response
Sampling	Directly sampled from tail pipe
Power supply	100–240 V AC/50 Hz 10–16 V DC at 15 amp
Size	100 mm × 210 mm × 50 mm

The exhaust gas analyser and smoke meter are used to measure HC, CO, NOx and smoke opacity. For air and gas mixing, suitable carburetor was used for the experimentation. All measurements were done when engine was attained steady state. For each load, five readings were generated to ensure accuracy of the data recorded and careful experimental arrangements were made to make it possible to obtain consistent and repeatable measurements.In order to reduce the error in the measurement of emissions, five readings were recorded and only their averages are presented in the graphs.

4.0 RESULTS AND DISCUSSIONS

The experimental investigations were carried out on a single cylinder four stroke CI engine test rig to operate on dual fuel mode. Engine tests were conducted on dual fuel mode using Diesel, Honge oil methyl ester [HOME] and Jatropha oil methyl ester [JOME] as injected pilot fuels and CNG as inducted fuel. Tests were conducted for 80% and 100% load conditions. In the initial stage of the work dual fuel engine operation with varying CNG flow rate, pilot fuel injection timings, carburetor venturi and compression ratios were optimised. CNG flow rate is varied from 0.25kg/hr to 1kg/hr.The injection timing is varied from 19^0 to 27^0 BTDC and the compression ratio is varied from 15 to 17.5. As the maximum compression ratio of the engine was limited to 17.5, it was not possible to study the engine performance beyond compression ratio of 17.5.

4.1 Effect of Ethanol addition on the Performance of CNG-HOME Dual Fuel Operated Engine.

This section presents the results of investigation carried out on a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. The engine is operated at a constant compression ratio of 17.5 with mixing chamber venturi having 6 mm hole geomety in the inlet manifold. The injector nozzle opening pressure was maintained at 230 bar. A CNG flow rate of 0.5kg/hr (12 lpm) was kept constant throughout the experiment.

4.1.1 Brake thermal efficiency

Figure 5 shows the variation of Brake Thermal Efficiency (BTE) of a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. at 80 % and 100% loads. The brake thermal efficiency is found to be higher for CNG-diesel dual fuel mode of operation compared to HOME-CNG, JOME-CNG operation at 80% load. CNG being common, properties of the injected fuels has a major effect on the engine performance. The injected biodiesel fuels has higher viscosity than diesel which makes atomization difficult and also has lower calorific value, which together result in lower brake thermal efficiency. The brake thermal efficiency improved with HOME-CNG -15%Ethanol and JOME-CNG-15%Ethanol at the same operating condition of injection pressure and injection timing. The reason could be that the quality of the spray obtained with blended fuels improves due to lower boiling point of ethanol than that of HOME/JOME-CNG combination. The combustion is more complete in the fuel rich zone due to the oxygen present in ethanol blended fuels. These results in enhanced combustion efficiency and decreased heat losses in the cylinder due to lower flame temperature of ethanol blended- CNG fuels than that of plain biodiesel of HOME/JOME-CNG dual fuel operation. All the dual fuel combinations show decreased BTE with increase in the gas flow rate. This is mainly due to lower pilot fuel quantity being injected with increased gas flow rate.

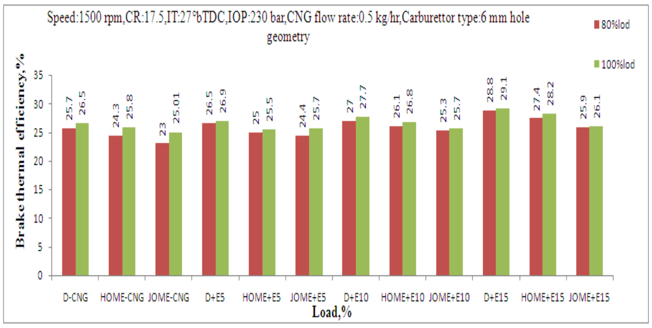

Fig-5 Variation of brake thermal efficiency with ethanol addition

4.1.2 Smoke opacity:

Figure 6 shows the variation of smoke of a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. at 80 % and 100% loads. It is revealed that dual fuel operation using natural gas is a very efficient method to reduce soot concentration at almost all conditions. The main reason is that natural gas, whose methane is the main constituent, being a lower member in the paraffin family, has very small tendency to produce soot. Thus, practically, gaseous fuel produces no soot, while it contributes to the oxidation of the soot formed from the combustion of the liquid fuel. Adding ethanol to HOME reduces smoke emissions of dual fuel engine. This is because the presence of bonded oxygen reduces the probability of soot nuclei formation in locally rich zones [2,6,10-12].The enrichment of oxygen content in the fuel due to the addition of oxygenates by both methyl ester of HOME and ethanol results in more complete combustion. The main reason for reduced smoke emission is due to improved spray characteristics of injected liquid fuels. Also CNG being common addition of ethanol to HOME lowers viscosity of the blend and increases the volatility as well.

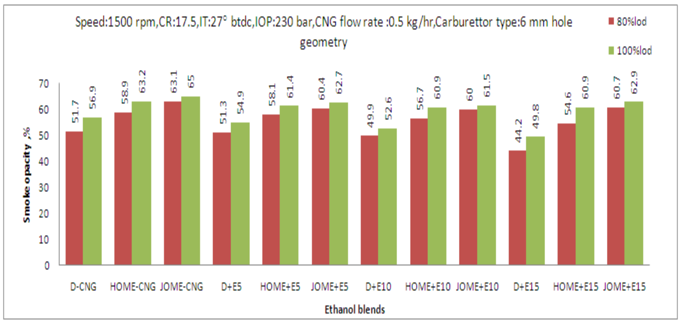

Fig-6 Variation of smoke opacity with ethanol addition

4.1.3 HC Emissions

Figure 7 shows the variation of HC emissions of a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. at 80 % and 100% loads. HC emissions in diesel engines are caused due to lean mixture during the delay period and under mixing of fuel leaving the fuel injector nozzle at lower velocity. Adding ethanol to diesel, HOME and JOME reduces HC emissions of dual fuel engine. Also ethanol addition improves overall combustion characteristics in terms of increased premixed combustion resulting in improved combustion. This is because CNG being common addition of ethanol to pilot fuel lowers viscosity of the blend and increases the volatility as well. When ethanol is added to the diesel/biodiesel fuel, it can provide more oxygen for the combustion process and leads to improved combustion. In addition, ethanol molecules are polar, which cannot be absorbed easily by the non-polar molecule lubrication oil layer and therefore ethanol can lower the possibility of production of HC emissions .HC emission is lower for ethanol 15 blend.

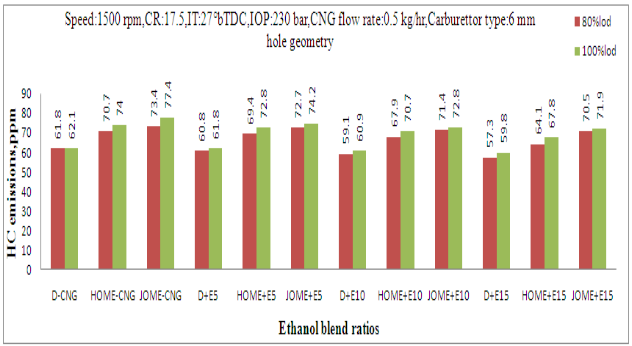

Fig- 7 Variation of HC with ethanol addition

4.1.4 CO emissions

Figure 8 shows the variation of CO emissions of a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. at 80 % and 100% loads. The dual fuel operation yield higher CO emissions at light load conditions. At low loads, most of the fuel left unburnt leads to poor combustion and ignition results in higher CO emissions. At higher loads the CO emission is lower than normal diesel operation because of better utilization. CO is a toxic by-product and is a clear indication of incomplete combustion of the premixed mixture. CO emission is mainly associated with incomplete combustion prevailing inside the combustion chamber. Adding ethanol to diesel/HOME/JOME improves combustion and hence reduces CO emissions of dual fuel engine [26-32]. This is because CNG being common addition of ethanol to HOME lowers viscosity of the blend and increases the volatility as well.

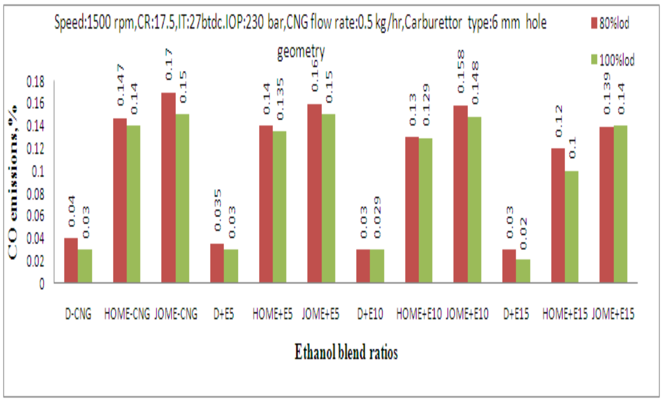

Fig.-8 Variation of CO with ethanol addition

4.1.5 NO_x emissions

Figure 9 shows the variation of NO_X emissions of a single cylinder, DI engine operating on diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation. at 80 % and 100% loads. The NO_X emission behavior for all the fuels is found to be similar at lower load with a slight increase in its magnitude at higher load. This is because at lower load, the adiabatic flame temperature of Stoichiometric air/ethanol flame is slightly lower. But the low cetane number of ethanol leads to increase in the ignition delay and the greater rates of pressure rise resulting in higher peak pressure and combustion temperatures. The exhaust gas temperature increases with increasing ethanol ratio in the fuel mixture. This high peak temperature increases NO_X emissions. Hence as ethanol concentration increases NO_X emissions also increases. Also the oxygenates of ethanol - ester fuel combinations results in higher NO_X emissions due to more complete combustion than the ester alone [6,10,12]. The exhaust gas temperature increases with increasing ethanol ratio in the fuel mixture. This high peak temperature increases NO*x* emissions. Hence as ethanol oncentration increases, NO*x* emissions also increases. Also the oxygenates of ethanol–ester fuel combinations results in higher NO*x* emissions due to more complete combustion than the ester alone.

Fig-9 Variation of NOx with ethanol addition

4.2 Effect of Exhaust Gas Recirculation on the Performance of CNG-HOME Dual Fuel Operated Engine.

This section provides the effect of exhaust gas recirculation on the performance of diesel, HOME, JOME and their blends with ethanol together with CNG in dual fuel mode of operation at 80 % and 100% loads. The engine is operated at a constant injection timing of 27^0 BTDC and constant compression ratio of 17.5 with mixing chamber venture 2 in the inlet manifold. The injector nozzle opening pressure was maintained at 230 bar. .

4.2.1 Brake thermal efficiency

From figure 10 it is observed that with EGR levels of 5% & 10% there is a small improvement in brake thermal efficiency. This could be due to improved combustion. Beyond 15% EGR level BTE reduces significantly. This could be due to predominant dilution effect of EGR leaving more exhaust gases in the combustion chamber. As the EGR amount is increased from 0 to 15 %, increase in BTE is observed at all loads (80% & full loads). There are three effects of using EGR in a diesel engine namely dilution effect, chemical effect & thermal effect [6,10-12]. This could be due to improved combustion of CNG. The inlet temperature increases when the EGR is introduced beyond 15% EGR.The chemical effect is associated with the participation of active free radicals present in exhaust gas to enhance combustion by taking part in pre-ignition reactions. However, this effect causes an increase in thermal efficiency. With more EGR substitution the thermal efficiency falls. This is due to the dilution effect of the EGR used, as it depletes the oxygen present in the combustion chamber.

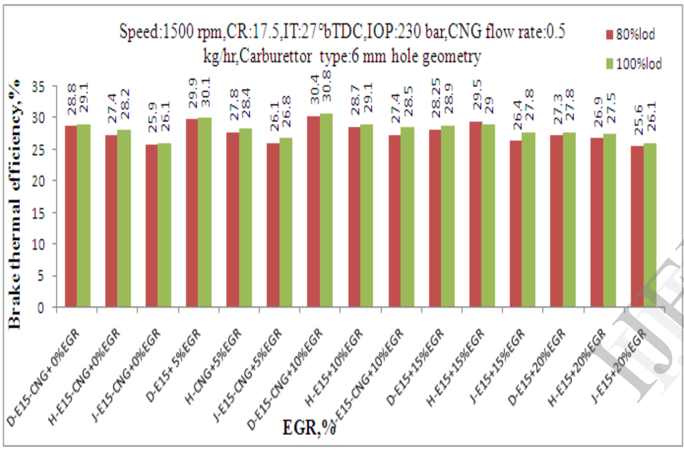

Fig-10 Variation of brake thermal efficiency with EGR

4.2.2 Smoke opacity

Figures 11 indicate that smoke opacity is higher in case of 20% EGR and is least in 0% EGR. Use of EGR has a negative effect on smoke emissions. The main reason for this is the reduction of engine air/fuel ratio supplied to the engine. It is revealed that dual fuel operation using natural gas is a very efficient method to reduce soot concentration at almost all conditions. The main reason is that natural gas, whose methane is the main constituent, being a lower member in the paraffin family, has very small tendency to produce soot. Thus, practically, gaseous fuel produces no soot, while it contributes to the oxidation of the soot formed from the combustion of the liquid fuel. Thus the use of natural gas has a positive effect of soot emission reduction.

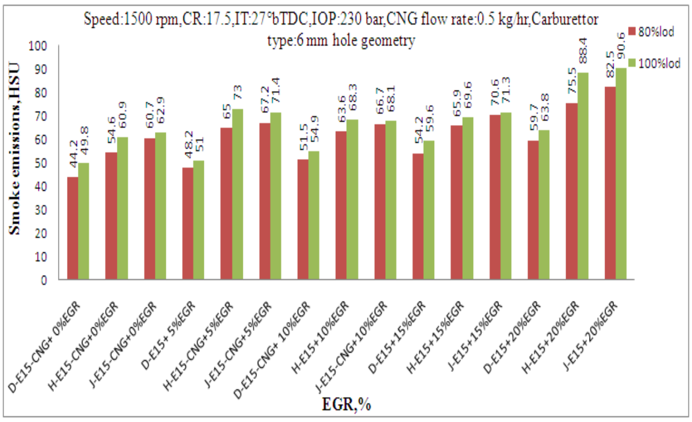

Fig-11 Variation of smoke opacity with EGR

4.2.3 HC Emissions

Figure 31 shows the variation of HC emissions with EGR. As EGR increases, the oxygen concentration in the charge & hence the temperature of combustion products both decrease. The increase in HC emission is observed with the increase in EGR percentage. This is due to poor combustion resulting inside the combustion chamber.

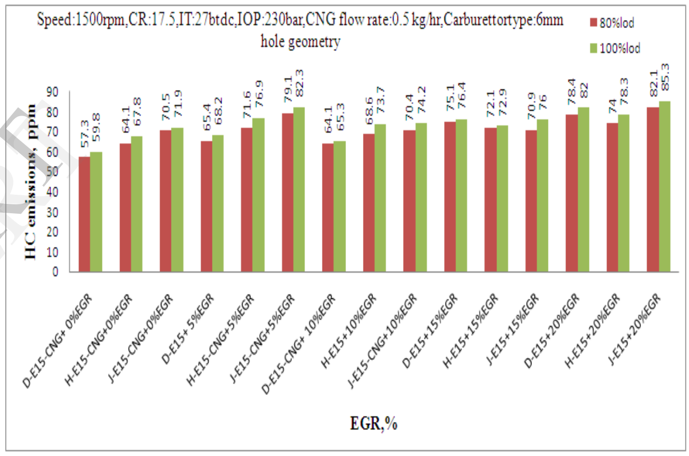

Fig-12 Variation of HC with EGR

4.2.4 CO emissions

Fig 13 indicates that Carbon monoxide emissions increase with increase in exhaust gas regulation. Higher values of CO were observed beyond 15% EGR. The reduction in the oxygen concentration is the main cause of CO and HC emissions. The dual fuel operation yield higher CO emissions at light load conditions. At low loads, most of the fuel left unburnt leads to poor combustion and ignition results in higher CO emissions. At higher loads the CO emission is lower than normal diesel operation because of better utilization of fuel. This is mainly due to high gas temperature and faster combustion rates. With EGR substitutions the CO formation is higher at full load condition than the dual fuel operation without EGR. At full load condition the availability of oxygen required for complete combustion decreases with increase in EGR ratio. Hence, the CO emission is higher with EGR addition in dual fuel operation at higher load than at the light load condition.

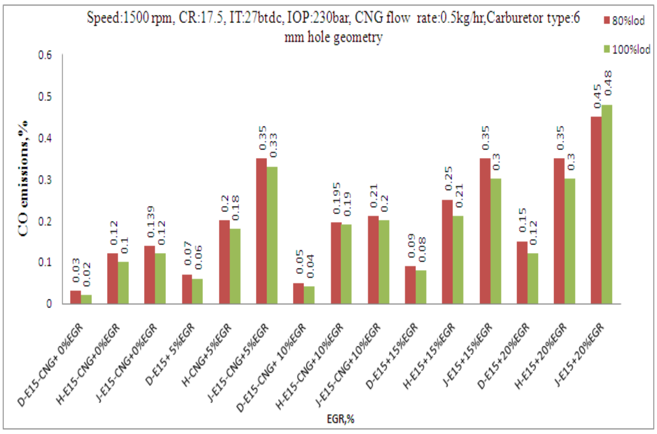

Fig.-13 Variation of CO with EGR

4.2.5 NO_x emissions

Figure 14 shows a reduction in NOx emission with increase in EGR percentage. This is because EGR reduces the oxygen concentration in the charge and reduces the temperature of combustion products due to higher specific heat capacity and hence lower NOx is observed (Banapurmath et.al 2013; Lei, Bi and Shen 2011). In dual fuel operation NOx emission is generally low but with higher substitution of CNG higher heat release rates occuring results in higher emission of NOx. Normally dual fuel operation exhibits higher emission of unburned hydrocarbon at light loads. At light loads the pilot quantity being small so flame cannot propagates fast and far enough to ignite the entire mixture. As the result it causes higher HC emissions but with increase in load the hydrocarbon emission decreases. As load progresses the pilot quantity increases and burns the surrounding fuel-air mixture sufficiently.

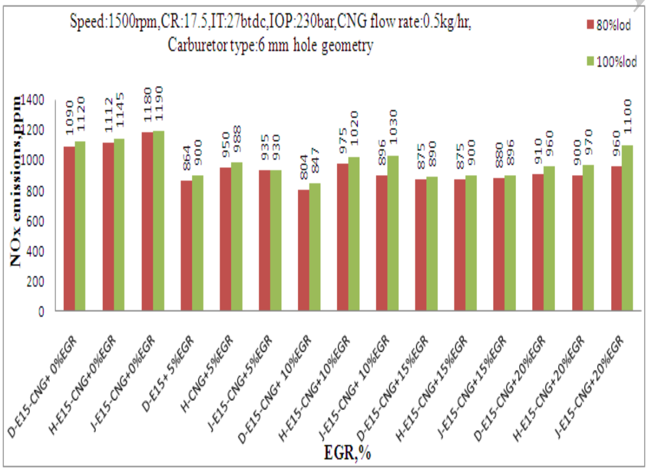

Fig-14 Variation of NOx with EGR

CONCLUSIONS

The biodiesel-CNG fuel combinations results in lower BTE and increased emissions of smoke, HC, CO compared to diesel-CNG dual fuel operation. With ethanol addition in and HOME/JOME-CNG dual fuel operation the performance improved in terms of increased brake thermal efficiency, reduced smoke, HC and CO emissions when compared to CNG and HOME/JOME dual fuel operation. However NO emission increased. Increasing the CNG quantity in the dual fuel operation with biodiesels of HOME and JOME the emissions of Smoke and CO decreases, while the HC and NOx increases. The combustion of HOME + 15% ethanol blended with CNG in the dual fuel engine operated with optimized parameters of injection timing of 27° BTDC and the compression ratio of 17.5 resulted in reduced pollutants emissions and improved brake thermal efficiency. Effect of exhaust gas recirculation shows that the brake thermal efficiency increases upto 10% EGR and beyond 15% EGR brake thermal efficiency reduces.

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