Universal Journal of Petroleum Sciences **2 (2014), 46-73**
www.papersciences.com

Performance, Combustion & Emission Characteristics of a Dual Fuel Engine Operated with Compressed Natural Gas (CNG) & Honge & Jatropha Biodiesels

Y.H.Basavarajappa[1], N.R.Banapurmath[2], A.J. Sangameshwar
[1]Dr.M.V Shetty Institute of Technology, Thodar, Moodbidri-574227, Karnataka, India
basavarajappa_yh@yahoo.co.in
[2] B.V.B. College of Engineering and Technology, Hubli-580031, Karnataka, India
nr_banapurmath@rediffmail.com

Abstract
Energy is an essential requirement for economic and social development of any country. Sky rocketing of petroleum fuel costs in the present day has led to growing interest in alternative fuels like vegetable oils, alcoholic and gaseous fuels. At the same time there is continuous pressure on emission control through periodically tightened regulations particularly for metropolitan cities. In this situation there is an urgent need to promote use of alternative fuels as substitutes for high speed diesel (HSD). Dual fuel mode of operation employing CNG (Compressed Natural Gas) and methyl esters of plant oils like Honge and Jatropha oils is an attractive option for countries having a large agriculture base that can be used as a feed stock to obtain newer fuels. Use of such fuels can ease the burden on the economy by curtailing the fuel imports. The current paper presents the results on the performance of a dual fuel engine fuelled with Honge and Jatropha oils methyl esters (HOME/JOME) as injected fuels along with CNG induction. Effects of Injection timing, CNG flow rate, compression ratio and Exhaust Gas Recirculation (EGR) on the performance of dual-fuel engine has been presented. From the experimental evidence it is found dual fuel engine operated with optimized parameters of injection timing of 27° BTDC, compression ratio of 17.5, 6 mm carburetor and 10% EGR resulted in overall improved performance.

Key Word: Honge Oil Methyl Ester, Jatropha Oil Methyl Ester, Compressed Natural Gas, Gas Induction, Dual Fuel Engine.

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 has 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 and provide food and energy security as well as foreign exchange savings besides addressing environmental concerns, and socio-economic issues [1]-[5]. The main drawback of CI engine is soot and nitric oxide emission levels; it can be overcome by operating the CI engine in dual fuel mode. Biodiesels derived from vegetable oils gives slightly lower performance with reduced emission levels [6][7].

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 [1][2][8]-[10]. Similarly, dual fuel engine operation with different gaseous fuels such as CNG, LPG, Hydrogen, HCNG are investigated by several researchers. 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[11]-[14]. 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.

The main advantage of CNG operated dual fuel engine include minor 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[11],[14]-[16]. Use of different biodiesels as injected fuel along with CNG induction in dual-fuel mode has been reported in the literature[13][17][21][22]. Different methods of CNG utilization in diesel engines have been reported in the literature[25]-[30].

Several engine parameters have been found to affect the performance of combustion in gas fuelled dual fuel engines[17][22][31]-[33]. These include injection timing [6][26][34], compression ratio [26][34], exhaust gas recirculation [34]-[41] and swirl augmentation techniques [33][34][42]. Better engine performance was reported for advanced injection timings. The effect of compression ratio on the performance and emission characteristics of diesel engine running on CNG in a dual-fuel mode has been reported in the literature [26]. It is reported that knock starts earlier when a high compression ratio is used, and an increased compression ratio generally increases the combustion noise. From the literature survey, it observed that use of CNG with diesel injection is more investigated and use of biodiesel-CNG combination is less reported and effect of injection timing, CR and EGR for these combinations are less investigated.

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 as injected fuels that replaces fossil diesel fuel.

2. Fuel Properties

The important properties of Honge, Jatropha oils and their esters used and CNG 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, kinematics viscosity of neat vegetable oil is several times higher than diesel oil and the transesterified oils of Honge and Jatropha show considerable reduced viscosities[43]-[46]. 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%, The fatty acids contribution, chemical formula, structure and molecular weight of the Honge and Jatropha oils used are tabulated in the Table 1. Properties of Honge and Jatropha oils and their respective biodiesels are listed in Table 2. NG 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. Table 3 shows properties of Natural Gas.

Sl No	Composition	*Fatty acid contribution		Chemical name	No. of bonds	Structure	Saturated /Unsaturated	Chemical formula	Mol. weight
		Honge	Jatropha						
1	Palmitic	10.5	1560	Hexadecanoic	----	16:0	Saturated	$C_{16}H_{32}O_2$	256.43
2	Stearic	5.56	6.42	Octadecanoic	----	18:0	Saturated	$C_{18}H_{36}O_2$	284.48
3	Oleic	49.39	32.04	Cis-9 Octadecanoic	Single	18:1	Un-saturated	$C_{18}H_{34}O_2$	282
4	Linoleic	20.37	42.47	Cis-9,cis-12 Octadecanoic	Double	18:2	Un-saturated	$C_{18}H_{32}O_2$	280
5	Linolenic	3.66	-	Cis-9,cis-12,cis-15 Octadecanoic	Triple	18:3	Un-saturated	$C_{18}H_{30}O_2$	278
6	Arachidic	1.36	0.4	Etcosanoic	-----	20:0	Saturated	$C_{20}H_{40}O_2$	312.54
7	Lignoceric	1.53	-	Tetracosanoic	-----	24:0	Un-saturated	$C_{24}H_{48}O_2$	368
8	Behenic	5.82	-	Docosanoic	-----	22:0	Un-saturated	$C_{22}H_{44}O_2$	340

Table 1 Fatty acids contribution, chemical formula, structure and molecular weight of Honge and Jatropha oils

Sl No	Properties	Diesel	Honge oil	Jatropha oil	HOME	JOME	ASTM Standards
1	Viscosity @ 40 ^{0}C (cSt)	4.59	56	50.73	5.6	5.84	***ASTM D93***
2	Flash point ^{0}C	56	270	240 ºC	163	160	***ASTM D93***
3	Calorific Value in kJ / kg	45000	35800	34000	36010	35,200	***ASTM D5865***
4	Density kg / m^3	830	930	918	890	895	***ASTM D4052***
5	Cetane Number	45-55	40	45	40-42	40-45	
6	Type of oil	Fossil fuel	Non edible	Non edible	Non edible	Non edible	

Table 2 Properties of Honge and Jatropha oils and their respective biodiesels

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 Range-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 (kg of air/kg of fuel)	17

Table 3 Properties of Natural Gas

3. Experimental Set up

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. 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 vary the compression ratio. Figure 3 shows carburetor holder with different venturi's designed for proper gas and air mixing. Exhaust gas opacity was measured using the Hartridge smoke opacity meter. The exhaust gas composition was measured using an exhaust gas analyzer (Make: MRU, Model: Delta 1600S). The measured emissions include CO2, CO, HC, NO_X, and O_2. 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 5 and 6 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.

Fig. 1 Overall view of the engine test rig test rig with dual fuel arrangement

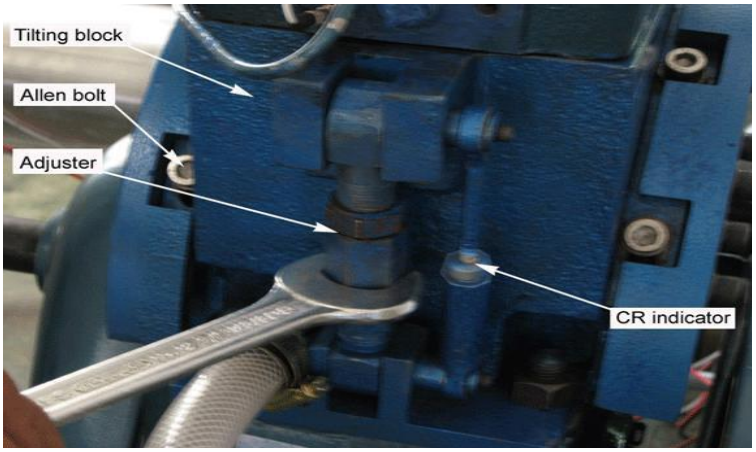

Fig. 2 Compression ratio adjustment

Fig. 3 Carburetor holder with venture

Make and Model	Kirloskar, TV1
Engine type	Single cylinder, four stroke, water cooled, direct injection, diesel engine
Orientation	Vertical
Ignition System	Compression Ignition
Bore X Stroke	87.5mm X 110mm
Displacement Volume	660 cc
Compression Ratio	17.5 : 1
Arrangement of valves	Overhead
Combustion Chamber	Open Chamber (Direct Injection)
Rated Power	5.2 kW (7 HP) @1500 rpm
Cooling Medium	Water cooled
Air measurement manometer	
Made	MX 201
Type	U-type
Range	100-0-100mm
Eddy current dynamometer	
Model	AG-10
Type	Eddy current
Maximum	7.5KW at 1500-3000rpm
Flow	Water must flow through dynamometer
Dynamometer arm length	0.180 meter
Fuel measuring unit range	0-50ml

Table 4 Specifications of the engine used

Type	**DELTA 1600S**
Object of Measurement	Carbon monoxide (CO), Carbon Dioxide (CO_2) and Hydrocarbons (HC)
Range of Measurement	HC = 0 to 20,000 ppm as C_3H_8 (Propane) CO = 0 to 10% CO_2 = 0 to 16% O_2 = 0 to 21% NO_X = 0 to 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 20^0C
Speed of Response Time	Within 15 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power Source	100 to 240 V AC / 50Hz
Weight	800 g
Size	100 mm x 210 mm x 50 mm

Table 5 Specifications of exhaust gas analyzer

Type	HARTRIDGE SMOKEMETER-4
Object of Measurement	Smoke
Measuring range opacity	0 – 100 %
Accuracy	+ / -2 % relative
Resolution	0.1 %
Smoke length	0.43 m
Ambient Temperature Range	-5^0 C to + 45^0 C
Warm up time	10 min. (self controlled) at 20^0C
Speed of Response Time	Within 15 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power Supply	100 to 240 V AC / 50HZ 10 – 16 V DC @15 amps
Size	100 mm x 210 mm x 50 mm.

Table 6 Specifications of smoke meter

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. The uncertainty of the measured parameters was estimated with confidence

limits of ±2σ (95.45% of the measured data lie within the limits of ±2σ around the mean). The percentage uncertainty of the measured parameters was estimated using the following relation:

$$\Delta xi = \frac{2\sigma_{x_i}}{\bar{x}_i} \times 100 \quad (3.1)$$

where: $\bar{x}_i$ = mean value

σ_{xi} = Standard deviation

4. 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.

Brake thermal efficiency for dual fuel engine operation operating on CNG is calculated by using the following equation:

$$\eta = B\ P/ (m_{CNG} * CV_{CNG} + m_{HOME/JOME} * CV_{HOME/JOME}) * 100 \quad (4.1)$$

where: BP= Brake power, kW

m_{CNG} =mass of CNG, kg/s

$m_{HOME/JOME}$ = mass of HOME/JOME, kg/s

$CV_{HOME/JOME}$ = Calorific value of HOME/JOME, kJ/kg

CV_{CNG} = Calorific value of CNG, kJ/kg

The heat release rate of the fuel causes a variation of gas pressure and temperature within the engine cylinder, and strongly affects the fuel economy, power output and emissions of the engine[27]. It provides a good insight into the combustion process that takes place in the engine. So, finding the optimum heat release rate is particularly important in engine research. During this work a computer program was developed to obtain the heat release rate. The heat release rate at each crank angle was calculated by using a first law analysis of the average pressure versus crank angle variation obtained from 100 cycles using the following expression given below:

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1}\left[P\left(\frac{dV}{d\theta}\right)\right] + \frac{1}{\gamma - 1}\left[V\left(\frac{dp}{d\theta}\right)\right] + \frac{dQ_{wall}}{d\theta} \quad (4.2)$$

where:

$\frac{dQn}{d\theta}$ = Net heat release rate

$\frac{dQw}{d\theta}$ = Heat transfer rate to the cylinder wall

γ = Ratio of Specific heats

4.1 Effect of CNG Flow Rate on the Performance of Diesel-CNG, HOME-CNG & JOME-CNG Dual Fuel Operated Engine

This section provides the performance of dual fuel engine operation using Diesel-CNG, HOME-CNG and JOME-CNG combination. During complete operation, injection timing, injection

pressure and compression ratio were maintained at 27^0bTDC, 230 bar and 17.5 respectively. The engine is operated with mixing chamber venturi having 6 mm hole geometry in the inlet manifold. This section mainly presents the effect of CNG flow rate on the performance of Diesel-CNG, HOME-CNG and JOME-CNG operated dual fuel engine.

4.1.1 Brake Thermal Efficiency

Figure 4 shows the variation of brake thermal efficiency with load at different CNG flow rates, when diesel, HOME and JOME was used as pilot fuel. The brake thermal efficiency is found to be higher for CNG-diesel dual fuel mode of operation compared to HOME-CNG and JOME-CNG operation at both 80% and 100% 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. As the CNG flow rate is increased from 0.25kg/hr to 1.0 kg/hr, the brake thermal efficiency [BTE] decreases for all the loads. CNG flow rate of 0.25 kg/hr results in higher brake thermal efficiency for both 80% and 100% load. As the gas flow rate is increased beyond 1 kg/h engine knock will occur resulting in poor engine performance. For the same CNG flow rate, HOME-CNG operation resulted in better performance compared to JOME-CNG operation. It could be attributed to comparatively lower viscosity and higher energy content of HOME. BTE values for HOME-CNG operation at 0.25, 0.5 and 1 kg/hr flow rate of CNG were found to be 24.6, 23.4 and 23.3% compared to 25.8% for diesel–CNG operation at 0.25 kg/hr respectively at 80% load.

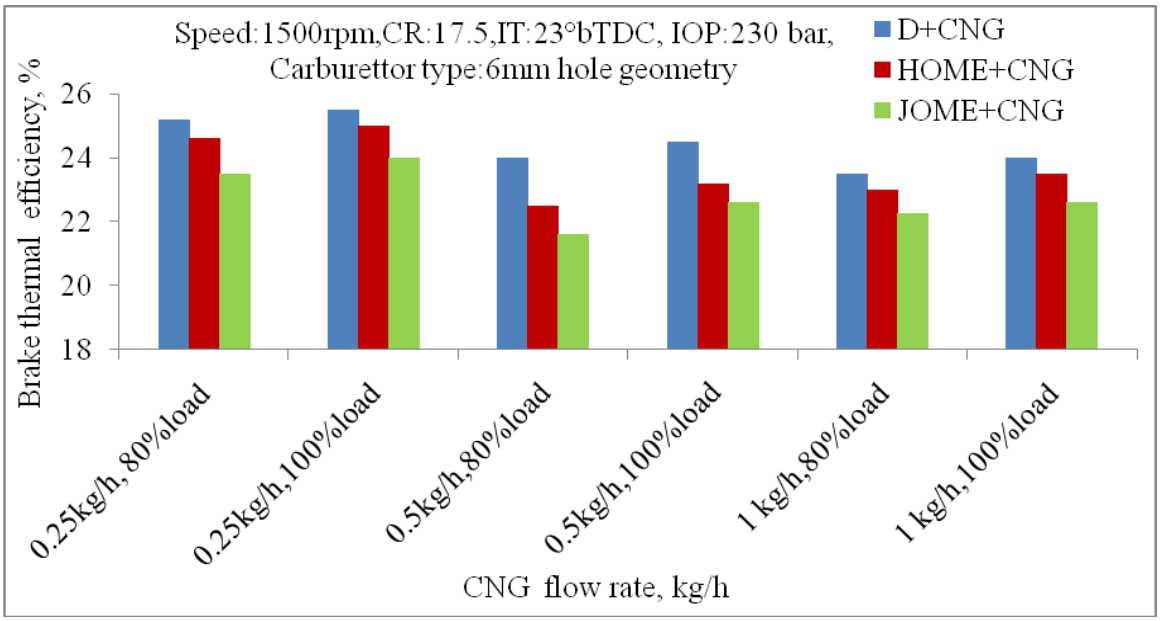

Fig.4 Variation of brake thermal efficiency with load for different CNG flow rates

4.1.2 Smoke Opacity

Figure 5 shows the variation of smoke opacity for diesel-CNG, HOME-CNG, JOME-CNG dual-fuel combinations at 80 %and 100% load. CNG-diesel dual fuel operation shows lower smoke opacity compared to HOME-CNG and JOME -CNG at 80 % and 100% load. It is considered that the biodiesel pilot fuel is mainly associated with the smoke emissions because CNG combustion consisting of mainly methane does not produce particulate emissions[4][26]. The heavier molecular structure of the injected biodiesel fuels compared to diesel results in higher smoke levels. Increasing the CNG quantity increases the quantity of heat output and the maximum temperature in the cylinder. For low gaseous flow rates the charge temperature is inferior. This results in better

combustion and hence the smoke opacity reduces with increased CNG quantity. At higher gas flow rates, the higher gas temperature promotes soot oxidation rate contributing to a further decrease of soot concentration. For low gaseous flow rates the charge temperature is inferior[14]. This results in better combustion and so the smoke opacity reduces with increased CNG quantity [14][26]. At higher gas flow rates, the higher gas temperature promotes soot oxidation rate contributing to a further decrease of soot concentration [14]. Smoke emission levels for HOME-CNG operation at 0.25, 0.5 and 1 kg/hr flow rate of CNG were found to be 71, 59 and 58 HSU compared to 62 HSU for diesel–CNG operation at 0.25 kg/hr respectively at 80% load.

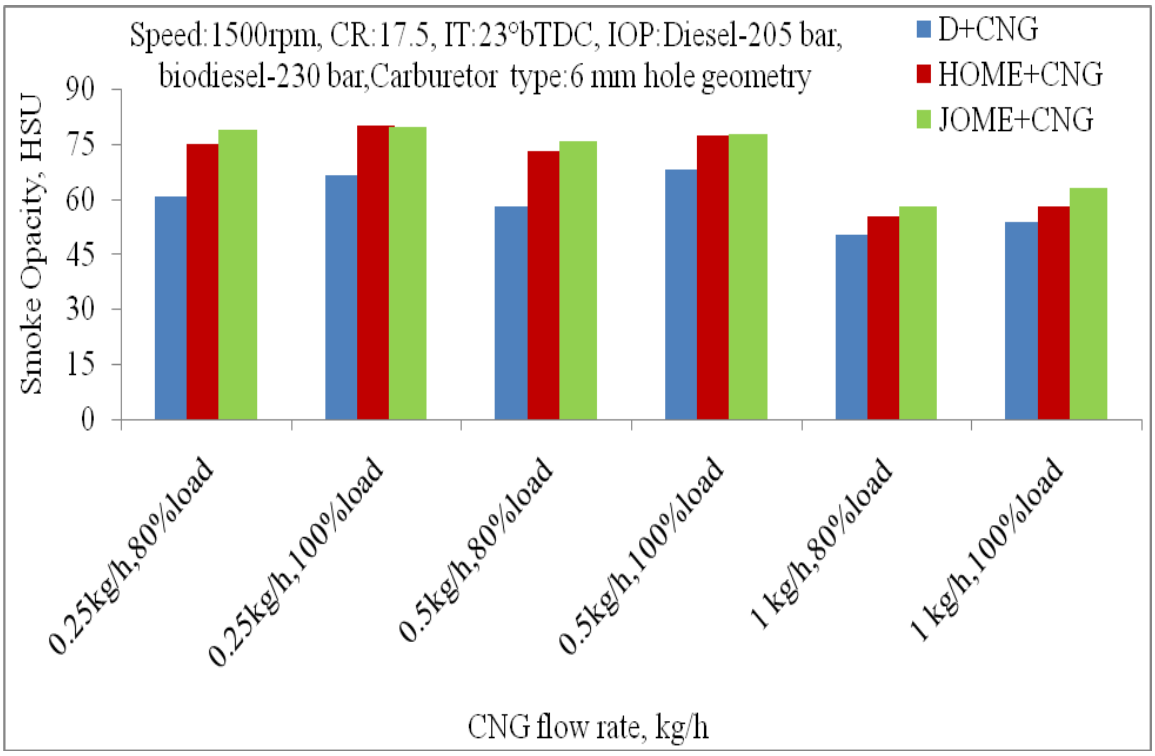

Fig. 5 Variation of smoke opacity with load at different CNG flow rates

4.1.3 HC Emissions

Hydrocarbon emissions shown in Figure 6 were higher throughout the load range for the injected biodiesel-CNG dual fuel combinations compared to CNG-diesel. The overall HC emissions levels using DFC is higher due to the CNG charge, which causes lean, homogeneous, low-temperature combustion, resulting in less complete combustion. This is because small amount of pilot fuel cannot propagate fast and far enough to ignite the whole premixed fuel mixture. At higher engine load there is an increase of HC emissions with increasing gaseous fuel rates until a certain limit where they start to decrease.

This is due to the increase of burnt gas temperature, which promotes the oxidation of unburned hydrocarbons. Under dual fuel operation, the filling of the crevice volumes with unburned mixture of air and gaseous fuel during compression and combustion while the cylinder pressure continues to rise, is an important source dominating the formation of HC emissions. HC emission levels for HOME-CNG operation at 0.25, 0.5 and 1 kg/hr flow rate of CNG were found to be 71, 74 and 81ppm compared to 80 ppm for diesel–CNG operation at 0.25 kg/hr respectively at 80% load.

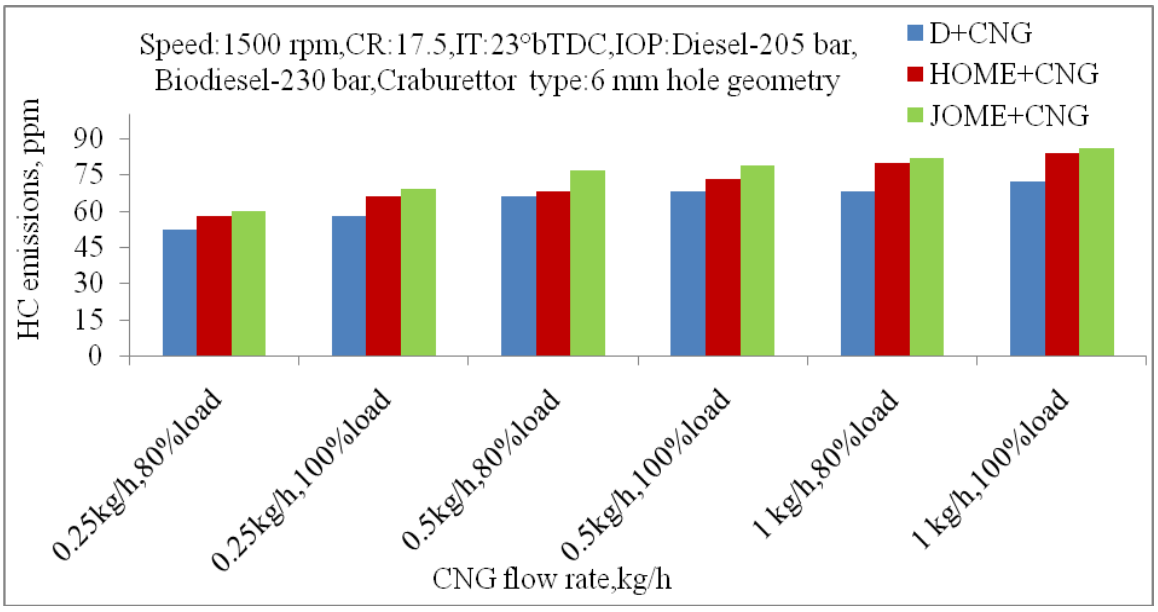

Fig. 6 Variation of HC with load at different CNG flow rates

4.1.4 CO Emissions

Figure 7 shows the variation of carbon monoxide with load at different CNG flow rates. The exhaust emissions of carbon monoxide were lower for CNG-diesel dual fuel operations compared to HOME-CNG and JOME-CNG respectively. This may be due to higher heat release during premixed burning phase rather than diffusion burning phase with CNG-diesel mixture. Emissions of CO are greatly dependant on the air-fuel ratio relative to stoichiometric proportions. Rich combustion invariably produces CO and emissions increase linearly with the deviation from the stoichiometry. At extremely low CNG fuel admission exhaust emissions of CO and fraction of gas consumed are very small. Further increased admission of CNG increases CO emission and later on these begin to decrease [4]. Increasing the CNG quantity increases the quantity of heat output and the maximum temperature in the cylinder.

Therefore, the combustion process finishes adequately and so CO emissions reduce with increased CNG quantity. Figure 7 substantiate this statement and the CO emission is observed to be lowest at 1 kg/h CNG gas flow rate. As the gas flow rate increases beyond 1 kg/h engine knock will occur resulting in poor engine performance. The onset of knock is only encountered at high compression ratios for CNG operation. CO emission levels for HOME-CNG operation at 0.25, 0.5 and 1 kg/hr flow rate of CNG were found to be 0.147, 0.157 and 0.185% compared to 0.185% for diesel–CNG operation at 0.25 kg/hr respectively at 80% load..

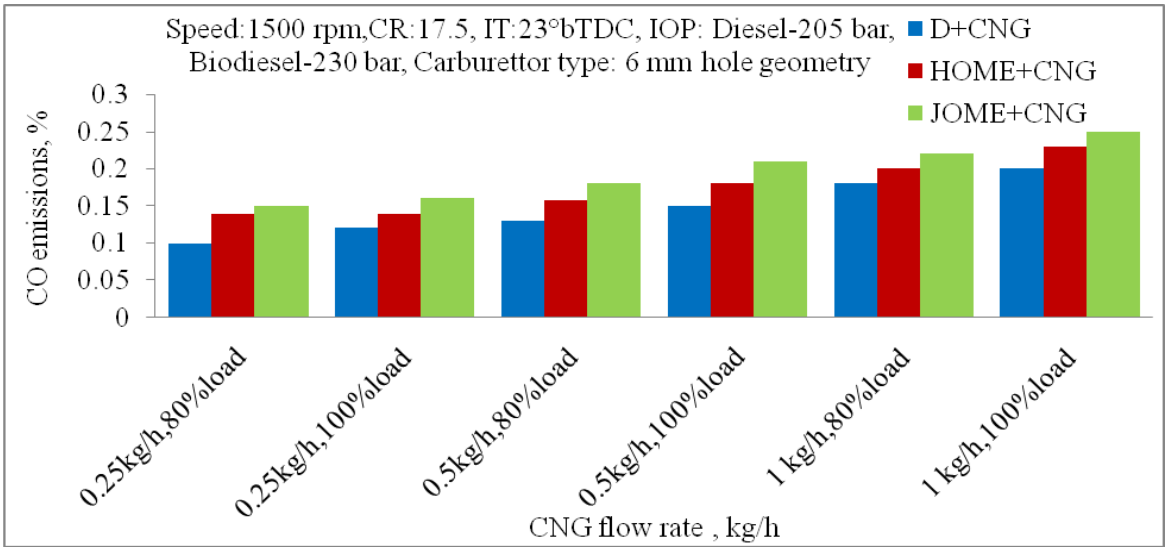

Fig. 7 Variation of CO with load at different CNG flow rates

4.1.5 NOx Emissions

Figure 8 shows the variation of NO_x with load for different CNG flow rates. As widely recognized, the formation of nitrogen oxides is favored by high oxygen concentration and higher charge temperature. NO_x emissions were higher for CNG–diesel dual fuel operation. The higher premixed combustion phase observed with diesel-CNG compared to biodiesel-CNG is responsible for the observed trends. Lower NO_x emissions for injected biodiesels of vegetable oils in dual fuel operation may be due to their lower calorific value and heavier molecular structure. Also for the same power output more biodiesel is required (due to increased specific fuel consumption) leading to delayed injection. All the dual fuel combinations show increased NO_x emissions with increase in the gas flow rate. This is mainly due to the increased quantity of heat output and the maximum temperature prevailing inside the engine cylinder with increased gas flow rate. Therefore the combustion process finished adequately results in increased NO_x emission [4][34]. NO_x emission levels for diesel-CNG, HOME-CNG and JOME-CNG dual-fuel operation at 0.5 kg/hr flow rate of CNG were found to be 910, 800 and 760ppm respectively at 80% load.

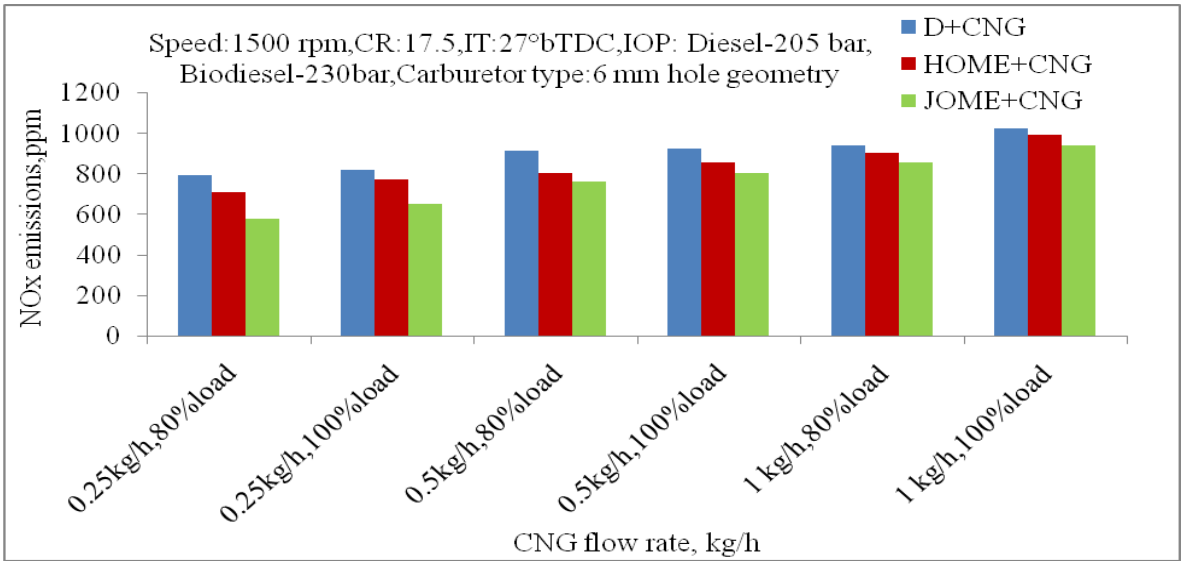

Fig. 8 Variation of NOx with load at different CNG flow rates

4.1.6 Combustion Characteristics

The pressure–crank angle diagram for the dual fuel engine operated with different CNG flow rates is shown in Figure 9. Lower cylinder pressure with increase in the mass flow rate of CNG was observed. It could be attributed to an increase in the ignition delay period of pilot fuel. Hence it is seen that ignition delay of the pilot fuel was significantly increased with an increase of natural gas flow rate. This increase in ignition delay period is mainly due to improper mixing of the fuel combination and lowering of compression temperature. However, part of air was replaced by natural gas and it reduces the oxygen concentration in the fuel mixture. Change in the specific heat of the charge mixture and its slower burning rate are also responsible for this trend[6].

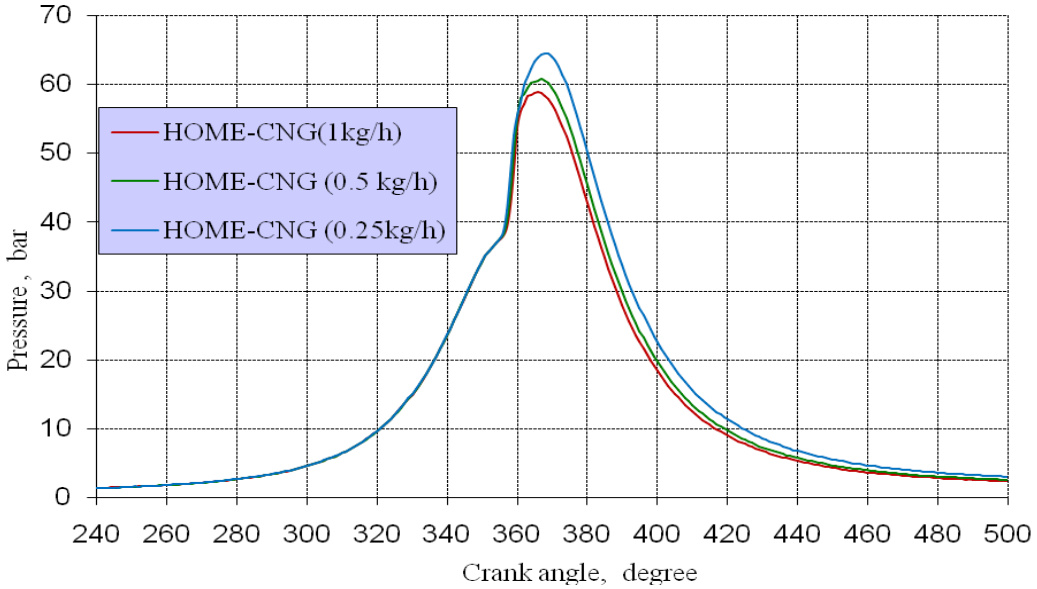

Fig. 9 Pressure verses crank angle diagram for HOME-CNG combination at 80% load

Figure 10 shows the heat release rate curves for dual fuel operation at 80% load with varying gas flow rates. Higher heat release rate was observed for CNG flow rate of 0.25 kg/hr compared to 0.5 and 1.0 kg/hr. This could be due to lower ignition delay observed with 0.25 kg/hr. Combustion of CNG deteriorates with higher CNG flow rates and lower heat release rates were obtained.

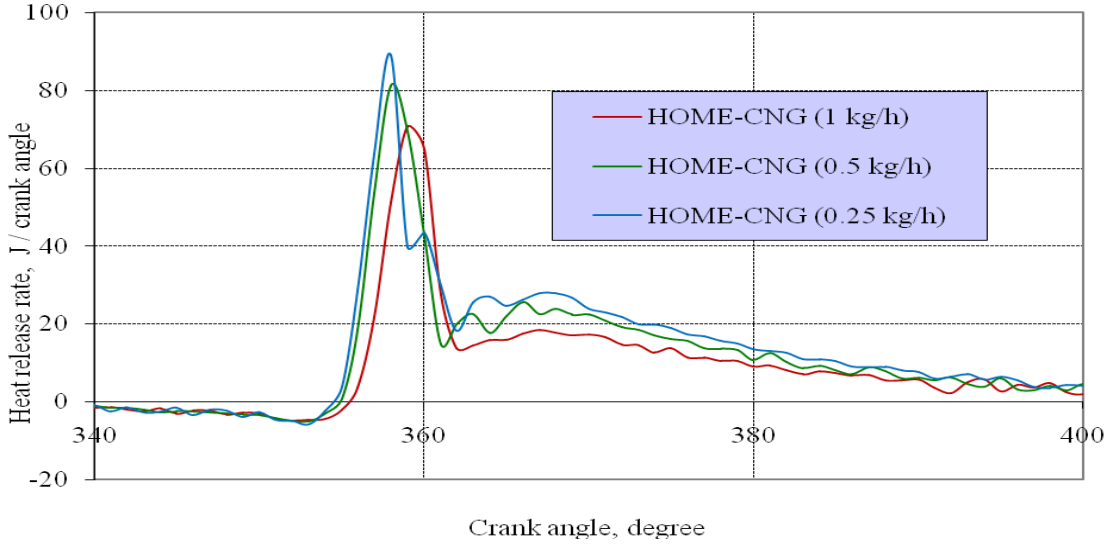

Fig. 10 Heat release rate verses crank angle diagram for HOME-CNG combination at 80% load

4.2 Dual Fuel Mode of Operation on Diesel-CNG, HOME-CNG & JOME-CNG with Variable Injection Timing

This section provides the effect of injection timing on the performance of CNG-diesel, CNG – HOME and CNG-JOME Dual Fuel operated Engine. 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 0.5 kg/hr was kept constant throughout the experiment.

4.2.1 Brake Thermal Efficiency

Figure 11 shows the variation of brake thermal efficiency with injection timing for Diesel-CNG, HOME-CNG and JOME-CNG fuel combinations operated at 60%, 80% and 100% load, as the injection timing is advanced from 19^0 BTDC to 27^0 BTDC, the brake thermal efficiency [BTE] increases for all the loads. The reason for this increase in BTE is that more time will be available for CNG fuel burning and these results in better performance and improved brake thermal efficiency. The engine performance was smooth and the ignition delay reduced through advanced injection timing but tended to incur a slight increase in fuel consumption [4]. An improvement in BTE was achieved by advancing the injection timing. Maximum pressure and pressure rise rate is higher for the advanced injection timing. BTE values for HOME-CNG operation at 19, 23 and 27 bTDC injection timing were found to be 19.3, 23.1 and 24.3% compared to 25.7% for diesel–CNG operation at 27bTDC respectively at 80% load.

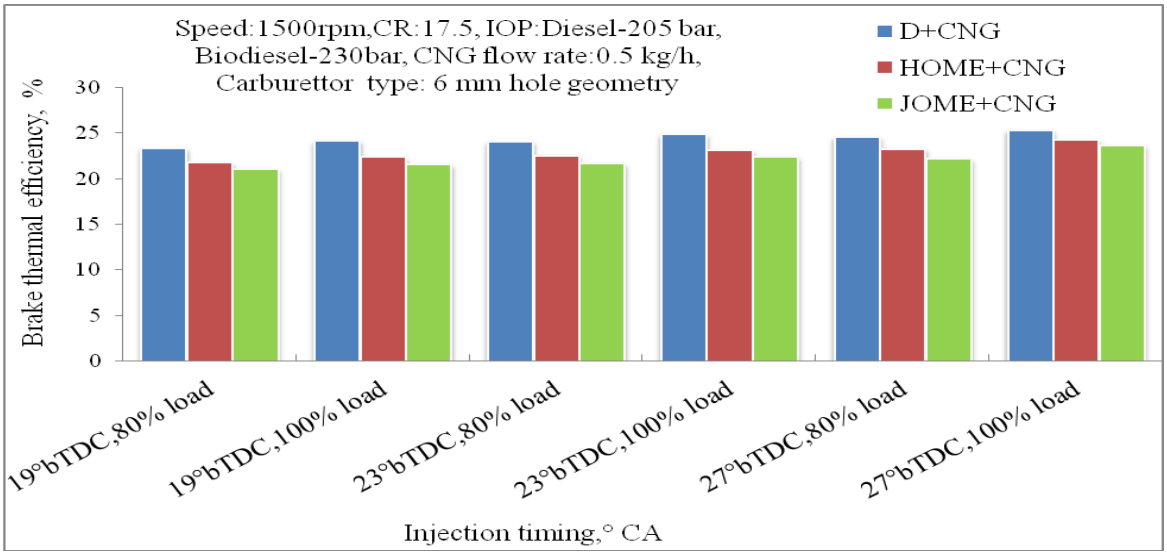

Fig. 11 Variation of brake thermal efficiency with injection timing

4.2.2 Smoke Opacity

Figure 12 shows the variation of smoke opacity with injection timing for both 80% and 100% load respectively. The smoke opacity decreases with increase in injection timing. This is because of better combustion prevailing inside the engine cylinder. It is also evident that as engine load increases, the smoke emissions increase slightly due to the decrease of air volumetric efficiency in DFC. Meanwhile, it is also observed that smoke emission of biodiesel–CNG DFC decreases with delayed pilot injection timing. Smoke levels for HOME-CNG operation at 19, 23 and 27 bTDC injection timing were found to be 70, 65 and 58 HSU compared to 51HSU for diesel–CNG operation at 27^obTDC respectively at 80% load.

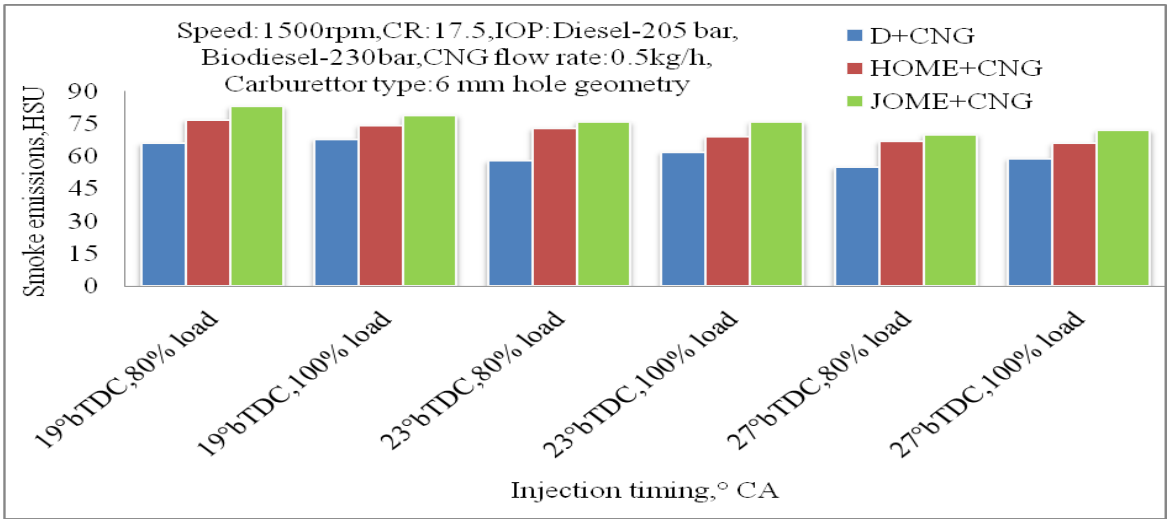

Fig. 12 Variation of smoke opacity with injection timing

4.2.3 HC Emissions

Figure 13 shows variation of HC emissions with injection timings for Diesel-CNG, HOME-CNG and JOME-CNG fuel combinations operated at 80% and 100% load dual fuel combination. As the injection timing increases the HC emission decreases considerably as seen in the figure for both 80% and 100% loads. The reason for decreased HC emissions with increased injection timing could be due to better combustion with increased BTE and more heat released during premixed combustion. However other researchers reported that advancing injection timing showed low and high HC emissions at low and high loading conditions compared to the standard injection timing operation, respectively[26]. It is also reported that advancing the pilot fuel injection timing reduces the UBHC emissions. This is due to a longer ignition delay of the mixture with the increased timing advance. The longer ignition delay allows a fuller spray penetration and development, creating a larger amount of the pilot fuel–air–gaseous fuel mixture (or flame propagation region) prior to ignition. The higher combustion rates of this larger premixed regions yields higher combustion temperatures and thus, lowers the UBHC emissions. HC emission levels for HOME-CNG operation at 19, 23 and 27 bTDC injection timing were found to be 81, 76 and 71 ppm compared to 62.1ppm for diesel–CNG operation at 27^{o}bTDC respectively at 80% load.

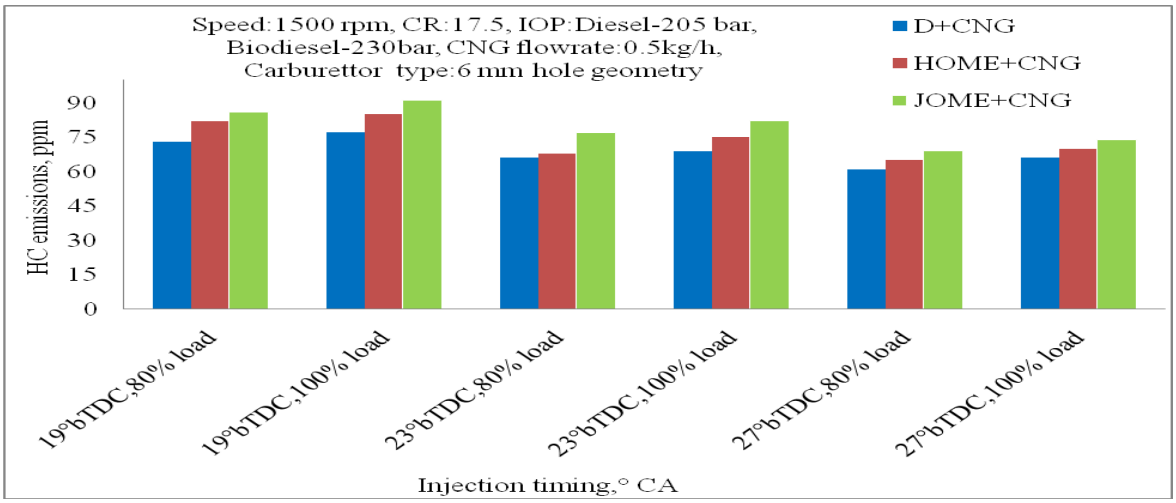

Fig. 13 Variation of HC with injection timing

4.2.4 CO Emissions

Figure 14 shows variation of CO emissions with injection timings for Diesel-CNG, HOME-CNG and JOME-CNG dual fuel combinations operated at 80% and 100% load respectively. As the injection timing is advanced from 19^0 BTDC to 27^0 BTDC the CO emission also decreased considerably as seen in the figure. With bigger injection advance, better overall combustion and the activity of the partial oxidation reactions reduce the CO emissions. The emission of CO results from incomplete combustion of HC fuel. The emission of CO greatly depends on the air-fuel ratio relative to stoichiometeric proportions. The reason for decreased CO emissions with increased injection timing could be due to better combustion with increased brake thermal efficiency. The advanced injection timing showed a significant reduction in CO emissions. compared to standard dual-fuel operation[4]. CO emission levels for HOME-CNG operation at 19, 23 and 27 bTDC injection timing were found to be 0.195, 0.162 and 0.147% compared to 0.04% for diesel–CNG operation at 27°bTDC respectively at 80% load.

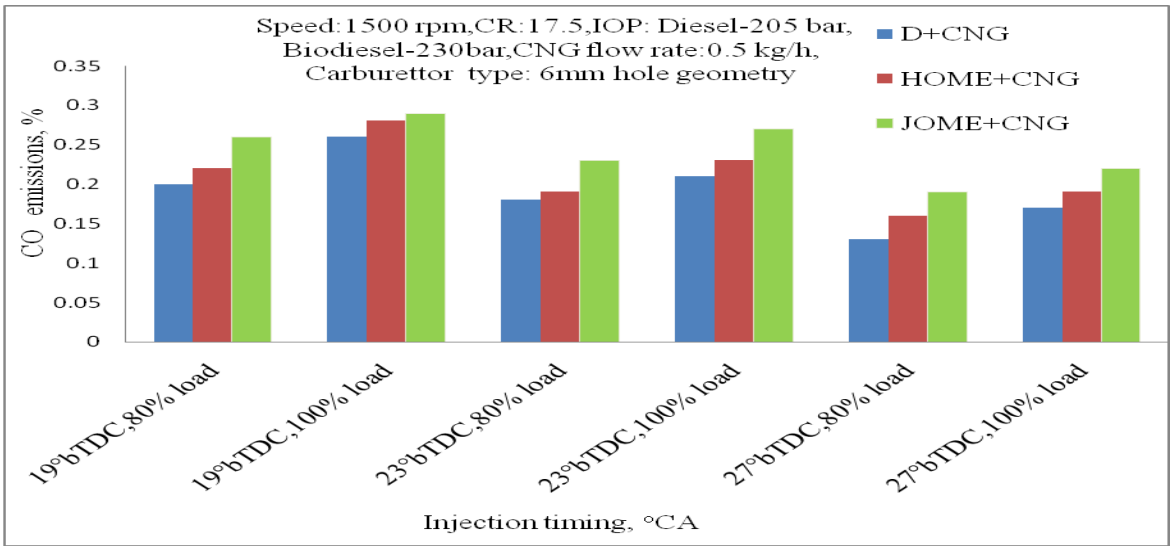

Fig. 14 Variation of CO with injection timing

4.2.5 NO_x Emissions

Figure 15 shows variation of NO_x emissions with injection timings for Diesel-CNG, HOME-CNG and JOME-CNG operated dual fuel combinations for 80% and 100% load respectively. As the injection timing increases the emission of NO_x increases considerably. The reason for increased NO_x emissions with increased injection timing could be due to better combustion prevailing inside the engine cylinder and more heat released during premixed combustion. The higher NO_x emissions for 27^0 BTDC are later controlled by appropriate use of EGR method. The variations in NO_x emissions follow changes in adiabatic flame temperature. These effects also vary with injection timing, suggesting that reaction zone stoichiometry and post combustion mixing are also influenced by fuel composition.

For biodiesel-CNG DFC it is observed that NO_x emissions are at the highest value for the pilot injection timing of 27^o BTDC and are the lowest value for the pilot injection timing of 19^o BTDC. NO_x emissions are increased with the advance of pilot injection timing due to the higher combustion temperature in flame zone caused by advanced ignition, which results in higher maximum combustion pressure. Generally NO_x emissions exhibit also a trade-off relationship with smoke emissions. NO_x emission levels for diesel-CNG, HOME-CNG and JOME-CNG dual-fuel operation at 19, 23 and 27°bTDC injection timing were found to be 910,800 and 760ppm at 80% 27°bTDC respectively at 80% load.

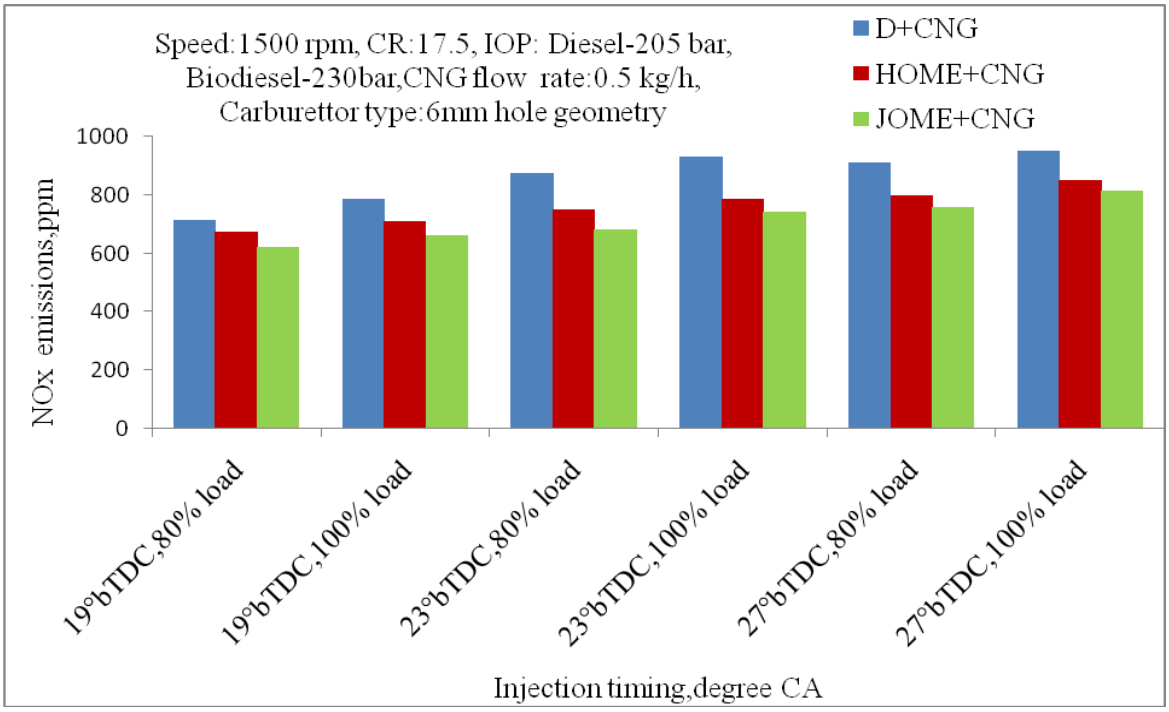

Fig. 15 Variation of NOx with injection timing

4.2.6 Combustion Characteristics

Figure 16 shows the cylinder gas pressure plotted for three injection timings of 19, 23and 27 bTDC. The cylinder pressure shows a slight decrease in peak pressure value with the retarding of fuel injection timings. Moreover, it can be seen that crank angle corresponding to the peak cylinder pressure becomes retarded at low cylinder pressure. The slow combustion rate is considered to be the main reason for low cylinder pressure [26].

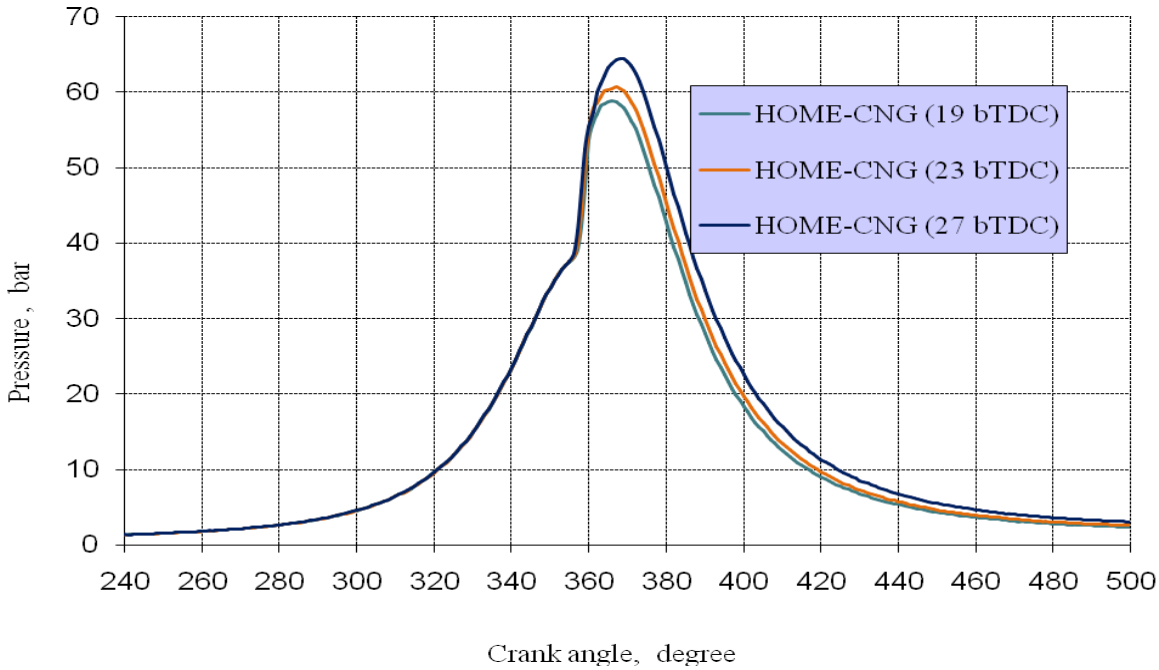

Fig. 16 Pressure- crank angle for HOME-CNG combination at 80 % load

Figure 17 shows the heat release rate with different fuel injection timings. Similar to the

behavior of cylinder pressure versus injection timings, heat release rate decreases with the retarding of injection timing. The injection timings retarded yielded lower efficiencies [26]. The most retarded injection timings resulted in combustion delayed into the expansion stroke. The phasing of the rate of heat release curves away from top dead centre combined with combustion occurring at low temperatures resulted in a loss in fuel conversion efficiency. The most advanced injection timings with their very long ignition delay period result in more fuel prepared for burning at the start of combustion compared with the retarded injection timings. Simultaneous ignition of a large proportion of the HOME fuel leads to fast combustion and a short burning time, as shown in Figure 17.

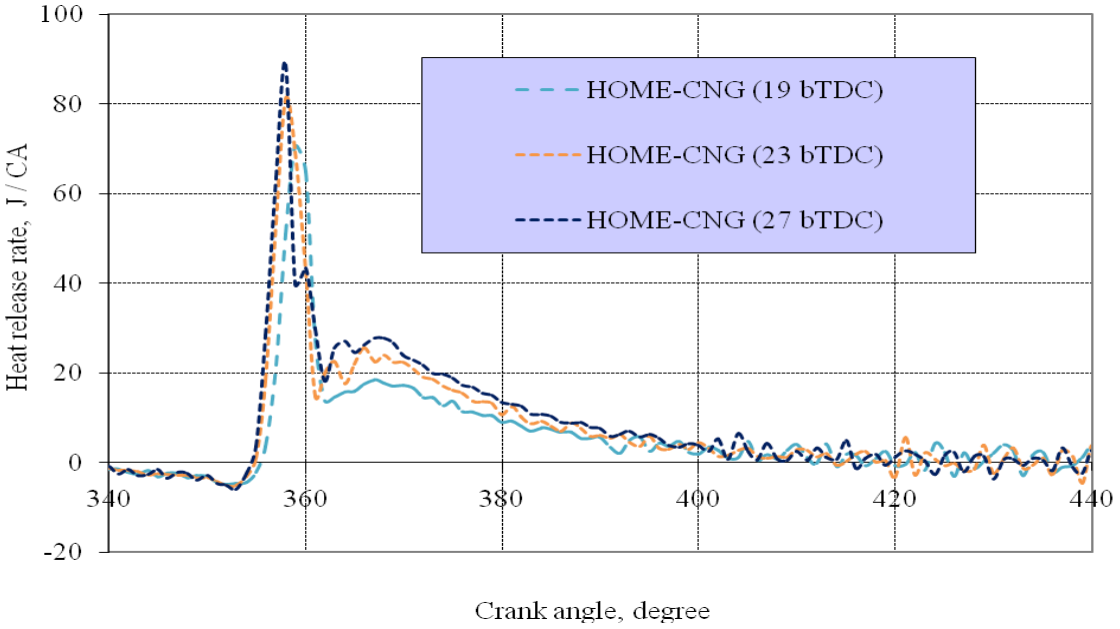

Fig. 17 Heat release rate diagram for HOME-CNG combination at 80 % load

4.3 Effect of Carburetor (Mixing Chamber Venture) on the Performance of CNG–Diesel, CNG –HOME and CNG-JOME Dual Fuel Operated Engine

This section provides the performance of dual fuel engine operation using Diesel-CNG, HOME-CNG and JOME-CNG dual fuel combinations operated at 80% and 100% load respectively During complete operation, injection timing, injection pressure and compression ratio were maintained at 270bTDC, 230 bar and 17.5 respectively. This section mainly presents the effect of mixing chamber venturi on the performance of Diesel-CNG, HOME-CNG and JOME-CNG operated dual fuel engine.

4.3.1 Brake Thermal Efficiency

Figures 18 shows the variation of brake thermal efficiency for Diesel-CNG, HOME-CNG and JOME-CNG dual fuel operation operated at 80% and 100% load respectively with two carburetors of 3mm & 6mm hole size. Brake thermal efficiency was found to be higher with 6 mm hole size carburetor compared to 3 mm hole size carburetor. This is because the 6 mm hole carburetor provides better air-CNG mixing. BTE values for HOME-CNG operation with 3 and 6mm hole geometry carburetors (CRB1 and CRB2) were found to be 22.7 and 24.3% compared to 25.7% for diesel–CNG operation with CRB2 respectively at 80% load.

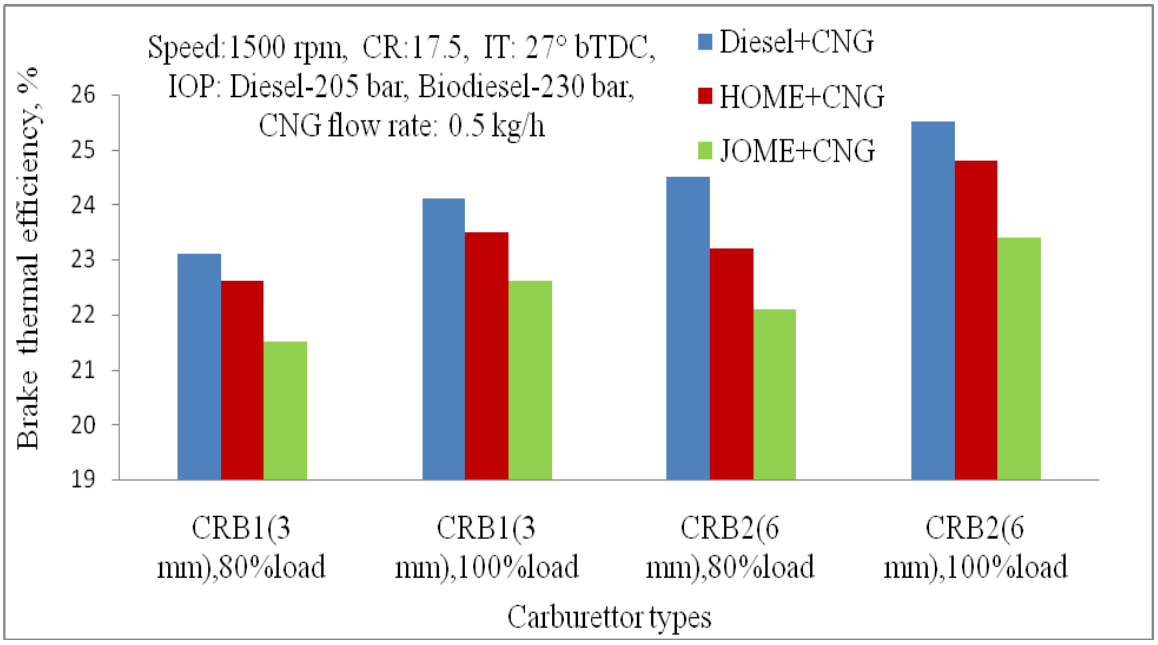

Fig. 18 Variation of brake thermal efficiency with carburetor

4.3.2 Smoke Opacity

Figure 19 show variation of smoke for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively for both 3 mm & 6 mm carburetors. The smoke opacity is lesser with mixing chamber venturi (Carb 2) having 6mm hole geometry. This is mainly due to better combustion occurring inside the engine cylinder. Smoke levels for HOME-CNG operation with 3 and 6mm hole geometry carburetors (CRB1 and CRB2) were found to be 63 and 59 HSU compared to 52 HSU for diesel–CNG operation with CRB2 respectively at 80% load.

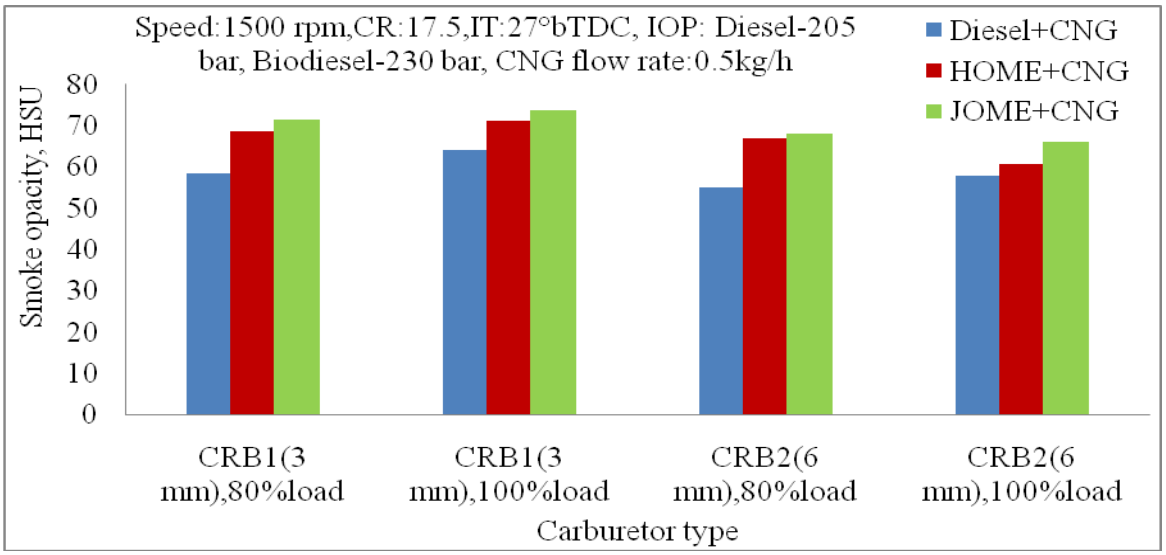

Fig. 19 Variation of Smoke opacity with carburetor

4.3.3 Hydrocarbon Emissions

Fig 20 shows variation of HC emissions with respect to 3mm & 6 mm hole carburetor for 80% & 100% load respectively. The HC emissions is least for 6mm carburetor. This is mainly because

of better combustion resulted in higher brake thermal efficiency with lower HC emissions. HC levels for HOME-CNG operation with 3 and 6mm hole geometry carburetors (CRB1 and CRB2) were found to be 71 and 72 ppm compared to 61 ppm for diesel–CNG operation with CRB2 respectively at 80% load.

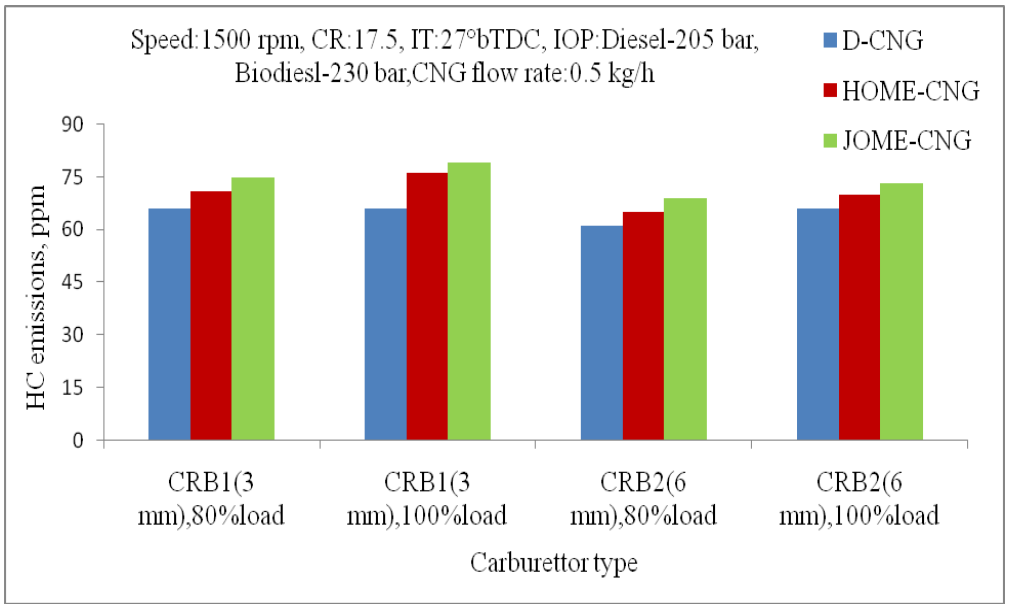

Fig. 20 Variation of brake thermal efficiency with carburetor

4.3.4 Carbon Monoxide Emissions

Fig 21 shows variation of CO emissions with respect to 3mm & 6 mm hole carburetor for 80% & 100% load respectively. The CO emissions are least for 6mm carburetor in both the cases. This is mainly because for a carburetor of 6mm hole size the exhaust gas temperature with for diesel-CNG, HOME-CNG and JOME-CNG dual fuel operation excess energy is supplied to the engine. CO levels for HOME-CNG operation with 3 and 6mm hole geometry carburetors (CRB1 and CRB2) were found to be 0.16 and 0.147 % compared to 0.04% for diesel–CNG operation with CRB2 respectively at 80% load.

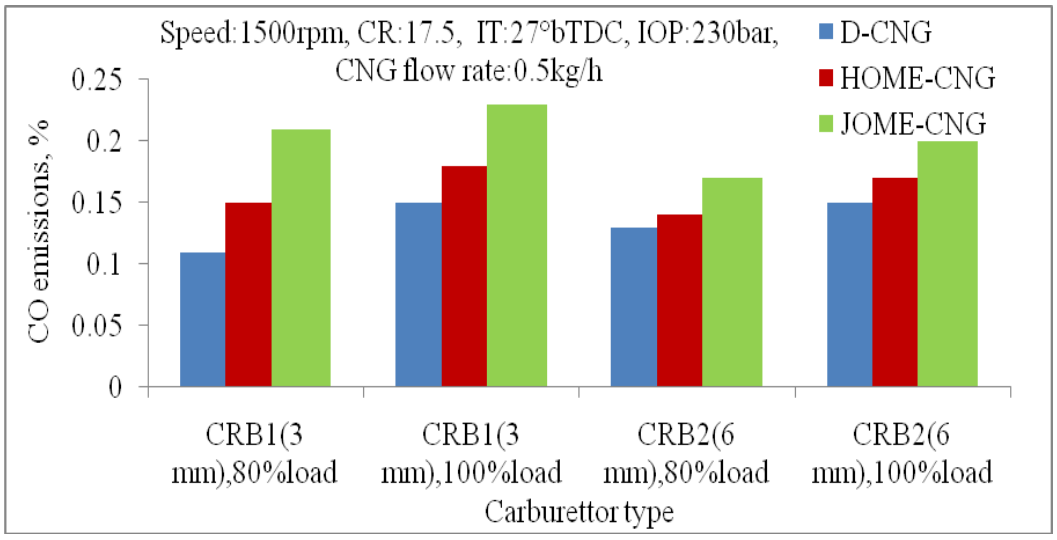

Fig. 21 Variation of CO with carburetor

4.3.5 NO_x Emissions

Fig 22 shows shows variation of NOx emissions with respect to 3mm & 6 mm hole carburetor for 80% & 100% load respectively. The NOx emissions were more for mixing chamber venturi 2. This is mainly because with this venturi increased premixed combustion phase which result in higher heat release rate. These higher NOx emissions may be controlled by suitable exhaust gas recirculation method. NOx levels for diesel-CNG, HOME-CNG and JOME-CNG operation with 6mm hole geometry carburetor (CRB2) were found to be 910, 800 and 760ppm respectively at 80% load.

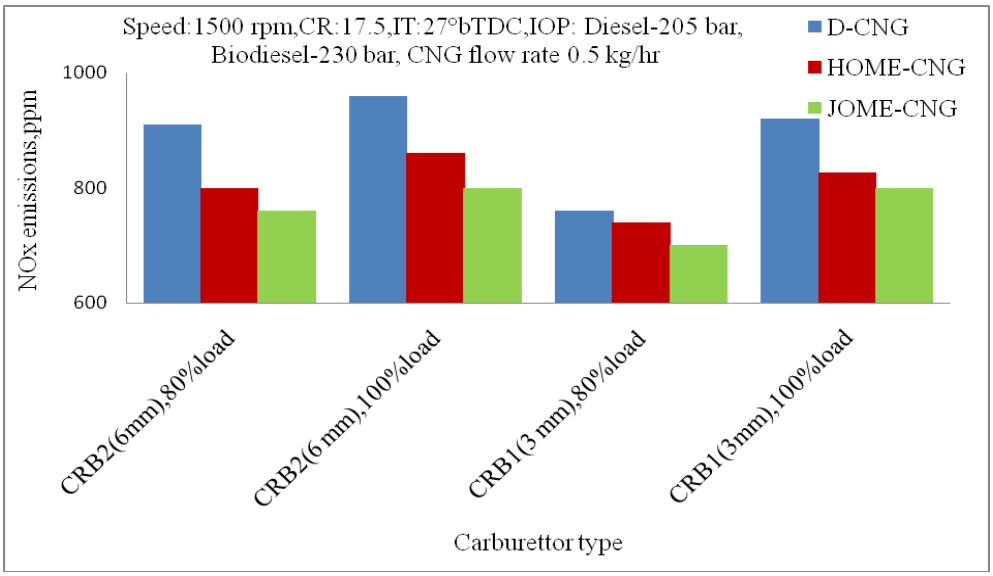

Fig. 22 Variation of NOx with carburetor

4.3.6 Combustion Analysis

Figure 23 shows the cylinder gas pressure plotted for two carburetors namely 6mm and 3 mm hole geometry. The cylinder pressure showed a slight decrease in the peak pressure value with the 3mm hole geometry carburetor. This low cylinder pressure was due to the poor mixture formation. Results revealed that effect of the carburetor type on the cylinder pressure was significantly dependent on the mixture formation tendency.

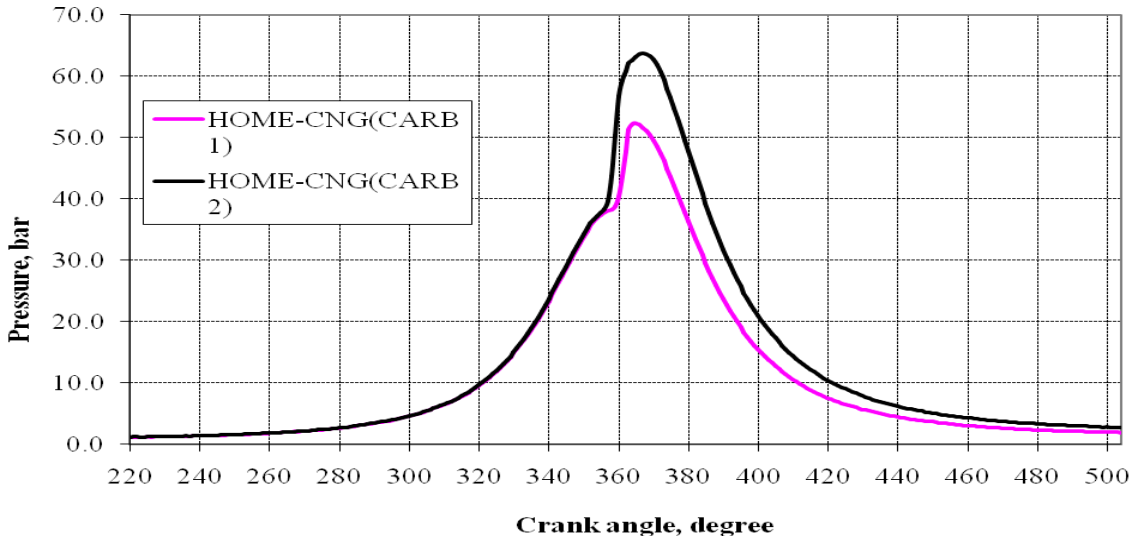

Fig. 23 Pressure-crankangle diagram for HOME-CNG combination at 80% load

Figure 24 shows the rate of heat release with different carburetor type. Similar to the behavior of cylinder pressure versus compression ratio, the rate of heat release decreased as the mixture formation tendency was poor.

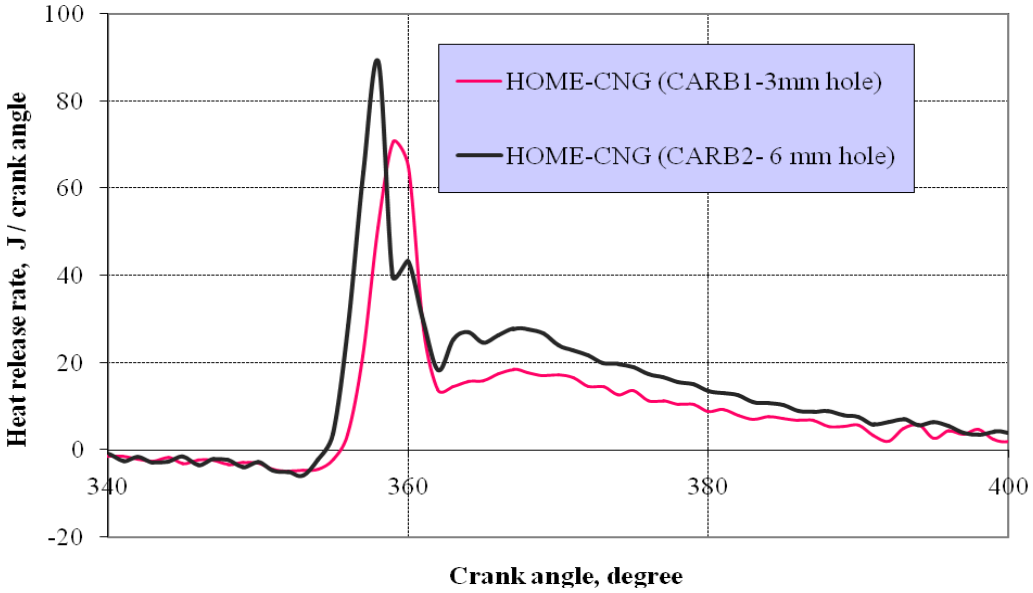

Fig. 24 Heat release rate diagram for HOME-CNG combination at 80% load

4.4 Effect of Compression Ratio on the Performance of CNG–Diesel, CNG–HOME and CNG-JOME Dual Fuel Operated Engine

This section provides the effect of compression ratio on the performance of CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively. The engine is operated at a constant gas flow rate of 0.5 kg/h, injection timing of 270 BTDC with mixing chamber venturi 2 having 6mm hole geometry in the inlet manifold. The injector nozzle opening pressure was maintained at 230 bar.

4.4.1 Brake Thermal Efficiency

Figure 25 shows the variation of BTE for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively. It is observed that Brake thermal efficiencies increases with increase in compression ratio from 15 to 17.5, the reason for this increase in brake thermal efficiency is due to better combustion taking place inside the engine cylinder. The maximum BTE was observed for a compression ratio 17.5 and its value is found to be 24.23% in case of 80% load.

The brake thermal efficiencies observed for CNG-HOME dual fuel mode of operation at 80% for 15, 16 and 17.5 compression ratios were 19.3 %, 19.7% and 24.3%, respectively. Increasing the compression ratio generally increases the combustion noise due to the higher self ignition possibility of the gaseous fuels at higher pressures and temperatures. As the compression ratio was reduced, the combustion noise also reduces with extended ignition limit. BTE values for HOME-CNG operation at 15, 16 and 17.5 CR were found to be 19.3, 19.7 and 24.3% compared to 25.7% for diesel–CNG operation at 17.5 CR respectively at 80% load.

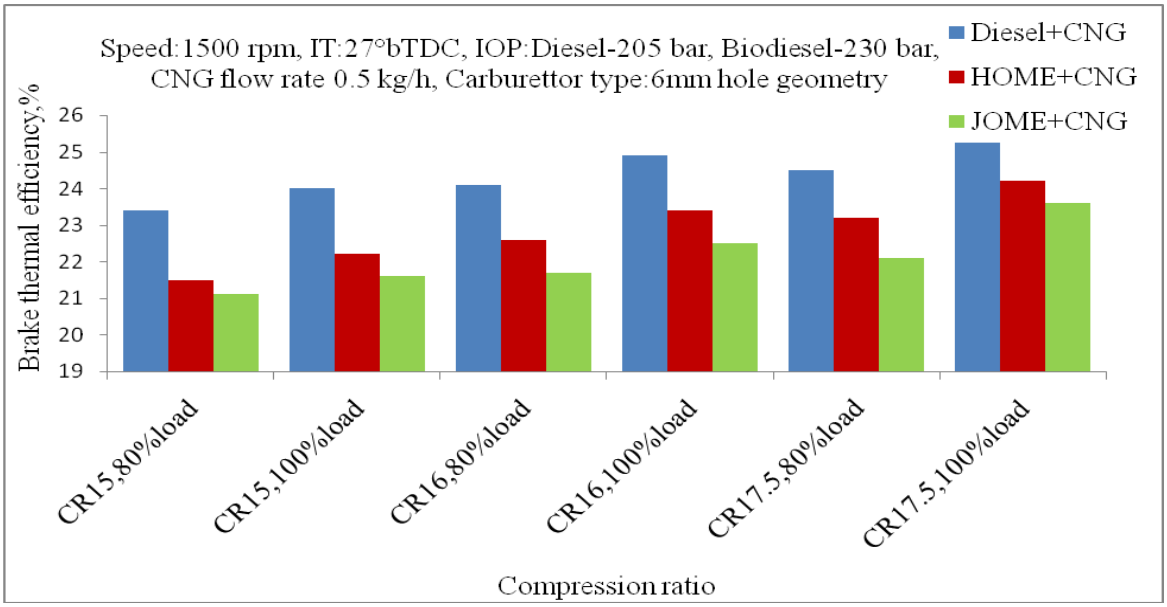

Fig. 25 Variation of brake thermal efficiency with compression ratio

4.4.2 Smoke Opacity

Figure 26 shows variation of smoke for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively for different compression ratios. The smoke opacity reduces with increase in compression ratios. The main reason for this decrease in smoke is mainly due to better combustion prevailing inside the engine cylinder. Similar observations were recorded even for full load engine operations. It is a well known fact that dual fuel operation remarkably reduces smoke emission. Moreover, combustion of CNG produces no particulates the only smoke associated are mainly due to pilot injection of diesel/biodiesel. Smoke levels for HOME-CNG operation at 15, 16 and 17.5 CR were found to be 68, 60 and 59 HSU compared to 52HSU for diesel–CNG operation at 17.5 CR respectively at 80% load.

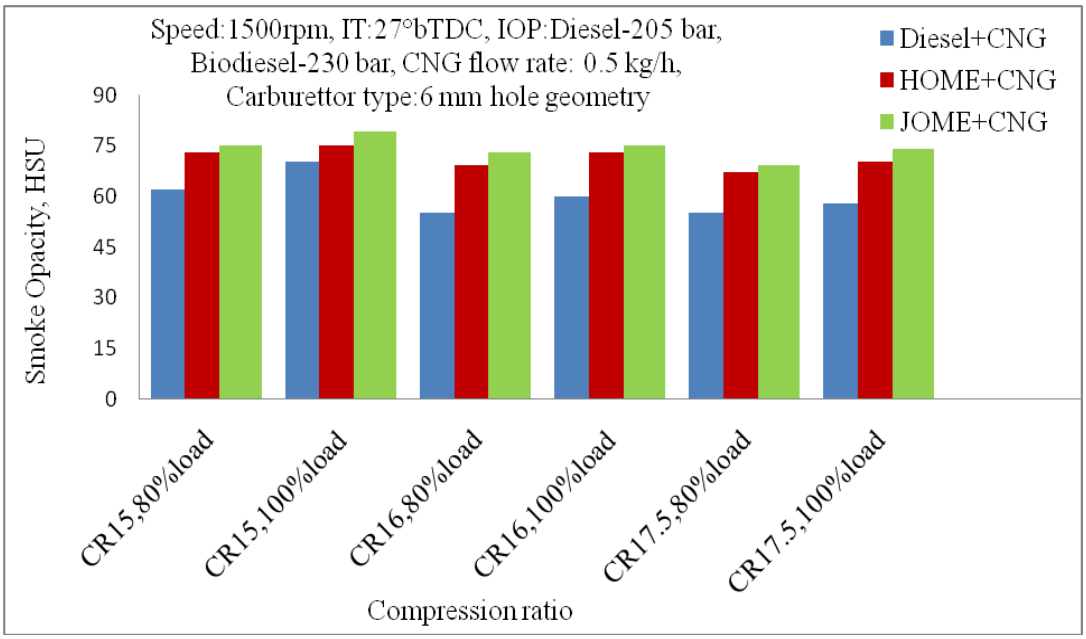

Fig. 26 Variation of smoke with compression ratio

4.4.3 HC Emissions

Figure 27 shows the variation of HC emissions for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively. It is observed that HC emissions decrease with increase in compression Ratios. The reason for decreased HC emissions with increased compression ratio is because higher brake thermal efficiency. Lower HC emissions at compression ratio of 17.5 are observed. The variation of unburned hydrocarbons in the exhaust gases is consistent with the quality of the combustion process of the engine. With the increase of compression ratio, there is a sharp decrease of HC emissions under dual fuel operation. This is the result of the increase of burned gas temperature that helps to oxidize efficiently the unburned hydrocarbons. HC levels for HOME-CNG operation at 15, 16 and 17.5 CR were found to be 81, 85 and 71 ppm compared to 61.8ppm for diesel–CNG operation at 17.5 CR respectively at 80% load.

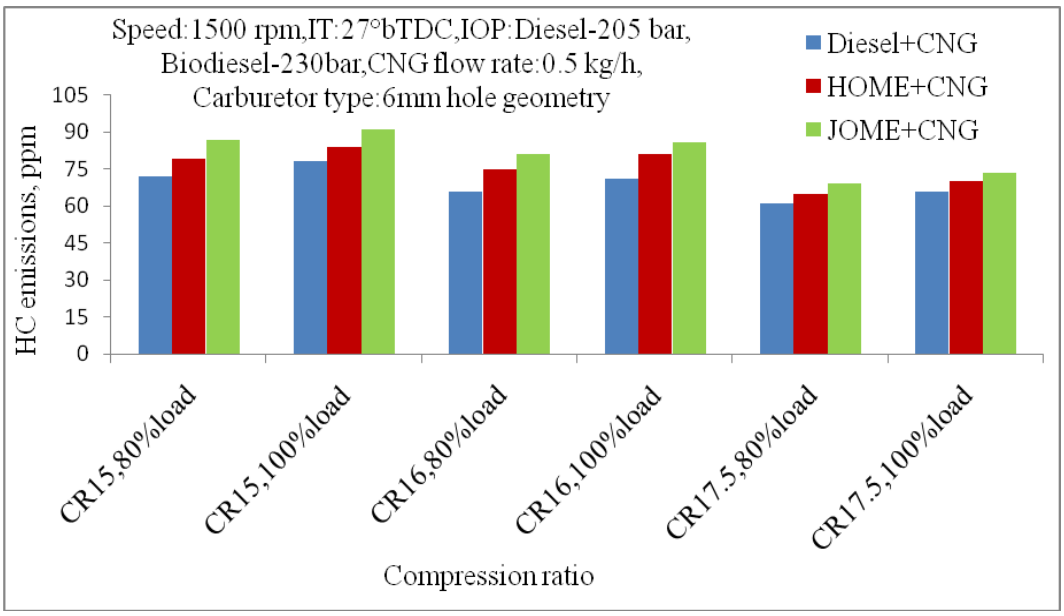

Fig. 27 Variation of HC emissions with compression ratio

4.4.4 CO Emissions

Figure 28 shows the variation of CO emissions for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation at 80% & 100% load respectively. It is observed that CO emissions decrease with increase in compression ratio. The reason for decreased CO emissions with increased compression ratio is because of higher BTE observed at these loads. The rate of CO formation is a function of the unburned gaseous fuel availability and mixture temperature, both of which control the rate of fuel decomposition and oxidation. At low engine speed, carbon monoxide concentration under dual fuel operation clearly decreases with the increase of engine load. This is the result of the improvement of gaseous fuel utilization especially during the second phase of combustion. CO levels for HOME-CNG operation at 15, 16 and 17.5 CR were found to be 0.28, 0.22 and 0.147% compared to 0.04% for diesel–CNG operation at 17.5 CR respectively at 80% load.

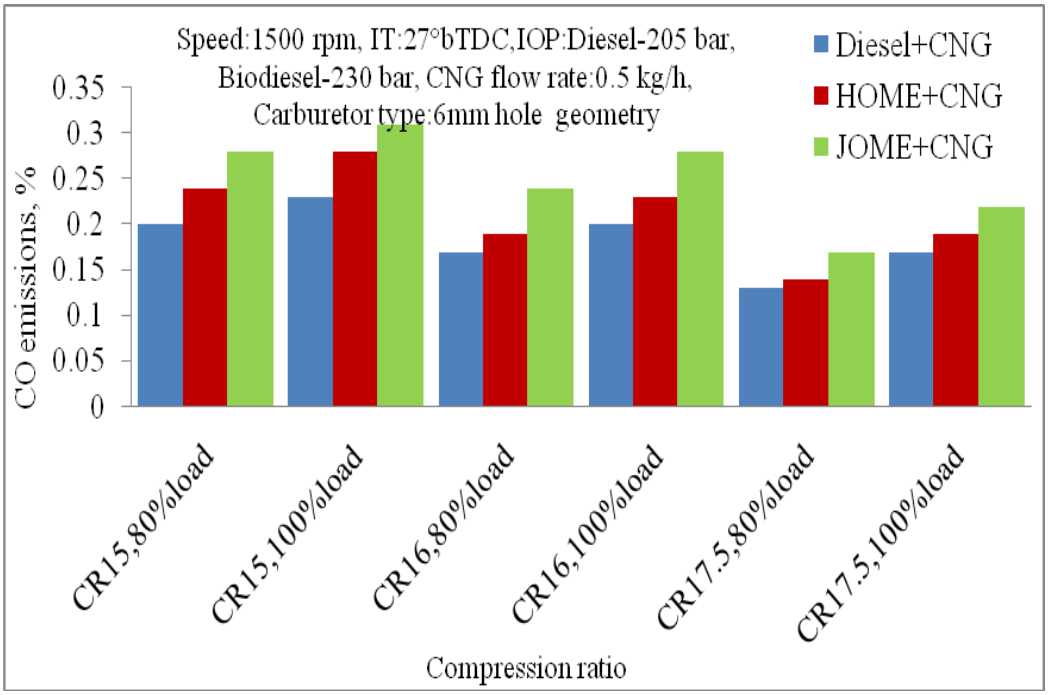

Fig. 28 Variation of CO with compression ratio

4.4.5 NO_x Emissions

Figure 29 shows the variation of NOx emissions with compression ratios for CNG-Diesel, CNG-HOME and CNG-JOME dual fuel engine operation for 80% & 100% load respectively. It is observed that NOx emissions increase with increase in compression ratios. The reason for increased NOx emissions with increased compression ratio is due to more heat released during premixed combustion phase. The formation of nitrogen oxides is favored by high oxygen concentration in the biodiesels used and high charge temperature. The lower premixed combustion observed with biodiesels compared to diesel is compensated by their higher oxygen concentration with CNG induction being common in both the cases. NOx emissions of biodiesel–CNG DFC are relatively low at the low to medium load range but are higher at higher load conditions.

This validates the idea that the simultaneous reduction of smoke and NOx can be achieved with biodiesel-CNG DFC except in the full load case. Thermal NOx reduction in DFC is a result of reduced temperature of combustion caused by the cooling effect of the CNG charge. Further exploring the causes of increased NOx formation in DFC at the full load case it is suggested that the NOx is increased greatly by increased in-cylinder temperatures. It is believed that the large increase in temperature is caused by gentle knock. At high loads, a higher engine temperature and increased turbulence inside the combustion chamber enhance the flame speed, resulting in the increase of NOx formation. NOx levels for HOME-CNG and JOME-CNG operation at 17.5 CR were found to be 800 and 910 ppm compared to 760 ppm for diesel–CNG operation at 17.5 CR respectively at 80% load.

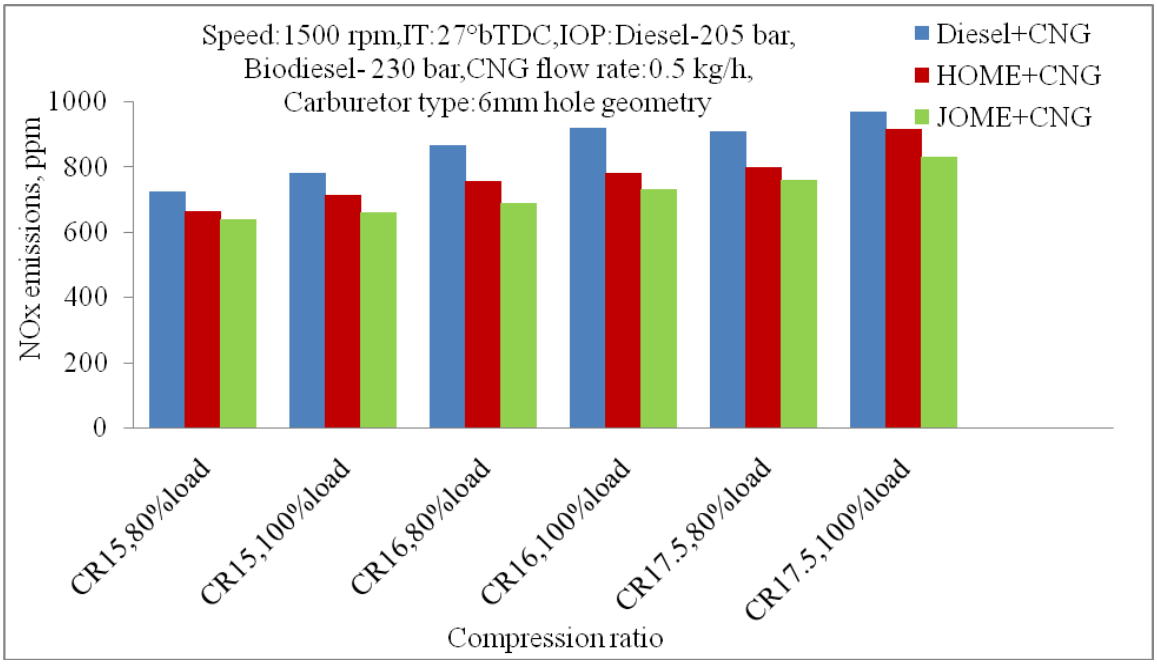

Fig. 29 Variation of NOx with compression ratio

4.4.6 Combustion Analysis

Figure 30 shows the cylinder gas pressure plotted for three compression ratios of 15, 16 and 17.5. The cylinder pressure showed a slight decrease in the peak pressure value with the reduction in compression ratio. This low cylinder pressure was due to the low compression ratio. The effect of the compression ratio on the rate of heat release was much mild. The engine operation at a higher compression ratio beyond 17.5 resulted in an unstable engine operation.

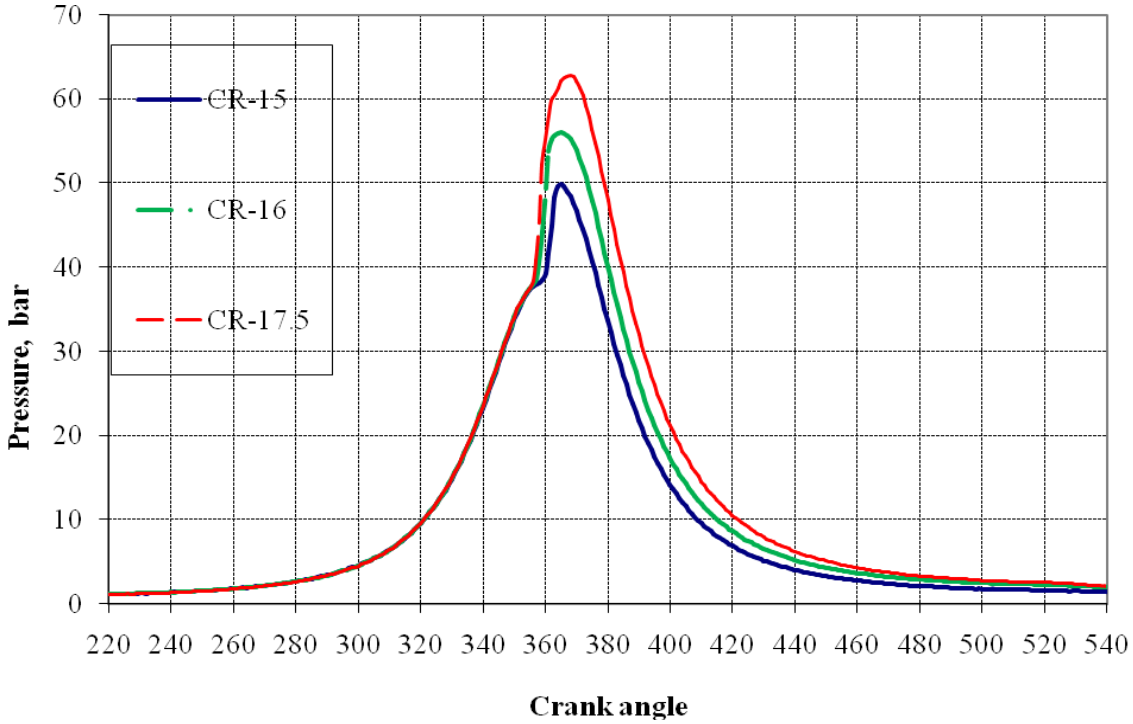

Fig. 30 Pressure- crank angle for HOME-CNG combination at 80 % load

Figure 31 shows the rate of heat release with different compression ratios. Similar to the

behavior of cylinder pressure versus compression ratio, the rate of heat release decreased as the compression ratio decreased.

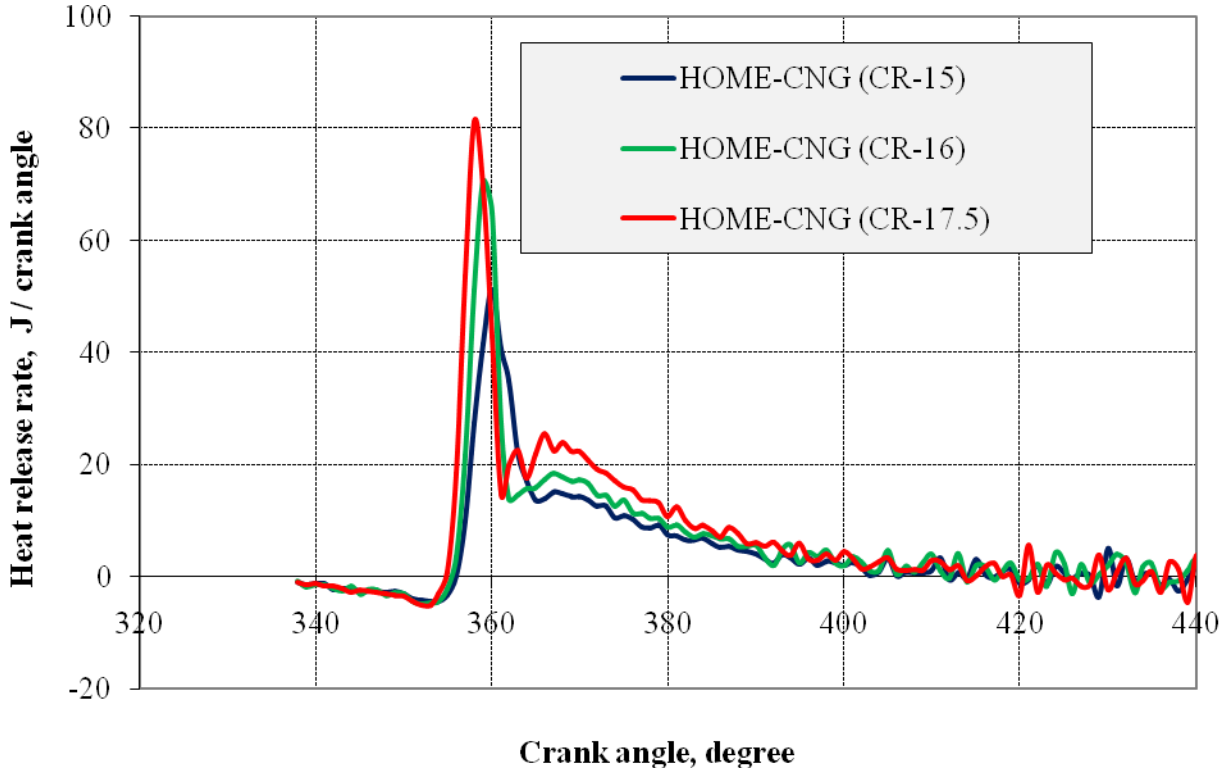

Fig. 31 Heat release rate diagram for HOME-CNG combination at 80%load

5. Conclusions

Based on the exhaustive experiments on the the dual fuel engine operation the following conclusions were drawn. The biodiesel-CNG fuel combinations resulted into lower BTE and increased emissions of smoke, HC, CO compared to diesel-CNG dual fuel operation. Increasing the CNG quantity in the dual fuel operation with biodiesels of HOME and JOME Smoke and CO emissions decreased, while the HC and NOx increased. Advanced injection timing resulted in increased BTE, NOx while smoke, HC and CO reduced. Optimizing air–CNG mixture with carburetor orifice of 6 mm improved performance and reduced emissions. Increased compression ratio increased BTE by (10%) and Nox.whereas smoke, HC and CO emissions reduced.

References

1. *Banapurmath, N.R., Tewari P.G.. Hosmath, R.S.,* 'Combustion an emission characteristics of a direct injection, compression-ignition operated on Hongeoil, HOME and blends of HOME and diesel', *International Journal of Sustainable Engineering, **1**(2) (2008), 80-93.*
2. *Banapurmath, N.R., Tewari P.G.. Hosmath, R.S.,* 'Effect of biodiesel derived from Hongeoil and its blends with diesel when directly injected at different injection pressures and injection timings in single-cylinder water-cooled compression ignition engine', *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, **223** (2009), 31-40.*
3. *Banapurmath, N.R,. Marikatti,K.M., Hunashyal,A.M., and Tewari,P.G.,* 'Combustion characteristics of a four-stroke CI engine operated on Honge and Jatropha oil methyl ester-ethanol blends when directly injected and dual fuelled with CNG induction', *International Journal of Sustainable Engineering, **4**(2) (2011), 145-152.*
4. *Lei, J., Y. Bi, and L. Shen.,* 'Performance and emission characteristics of diesel engine fueled with ethanol-diesel blends in different altitude regions', *Journal of Biomedicine and Biotechnology Article,(2011), ID417421 doi:10.1155/2011/417421.*
5. *Karim, G.A.,* 'An Examination of Some Measures for Improving the Performance of Gas Fuelled Diesel Engines at Light Load', *Society of Automotive Engineers, (1991), Paper No. 912366, USA.*

6. *Nwafor, O.M.I.,* 'Effect of advanced injection timing on emission characteristics of diesel engine running on natural gas', *Renewable Energy, **32** (2007, 2361-2368.*
7. *Banapurmath, N.R., and Tewari P.G., '*Performance combustion and emissions characteristics of a single cylinder compression ignition engine operated on ethanol-biodiesel blended fuels', *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, **224** (2010), 533-543.*
8. *Raheman, H., and A.G. Phadatare.,* 'Diesel engine emissions and performance from blends of karanja methyl ester and diese', *Biomass and Bioenergy', **27** (2004), 393-397.*
9. *Cenk Sayin, M Gumus,* 'Impact of compression ratio and injection parameters on the performance and emissions of a DI diesel engine fueled with biodiesel-blended diesel fuel', *Applied Thermal Engineering, **31** (2011), 3182-3188.*
10. *Yaliwal V.S., Daboji S. R., Banapurmath N.R., Tewari P. G.,* 'Production and Utilization of Renewable Liquid Fuel in a Single Cylinder Four Stroke Direct Injection Compression Ignition Engine, International Journal of Engineering Science and Technology', ***2(10)** (2010), 5938-5948*
11. *Srinivasan, K.K., Y. Qi, S.R. Krishnan, H. Yang, and K.C. Midkiff..* 'Effect of hot exhaust gas recirculation on the performance and emissions of an advanced injection low pilot-ignited natural gas engine'. *J. Eng. Gas Turbines Power, **128** (2006), 213-218.*
12. *Banapurmath, N.R, Budzianowski, W.M, Basavarajappa, Y.H.,. Hosmath R.S., Yaliwal V.S and P.G. Tewari.,* 'Effects of compression ratio, swirl augmentation techniques and ethanol addition on the combustion of CNG–biodiesel,in a dual-fuel engine', *International Journal of Sustainable Engineering, **7(1)** (2014), 55-70.*
13. *Carlucci, A.P., A. de Risi, D. Laforgia, F.Naccarato., '*Experimental investigation and combustion analysis of a direct injection dual-fuel diesel–natural gas engine', *Energy, **33(2)** (2008),256-263.*
14. *Papagiannakis, R.G. and D.T. Hountalas.,* 'Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel engine. *Journal of Applied Thermal Engineering', **23** (2003), 353-356.*
15. *Balasubramanian,V., Sridhara,K., Ganesan,V.,* 'Performance Evaluation of a Small Agricultural Engine Operated on Dual Fuel (Diesel + Natural Gas) System', *(1995) SAE. 951777.*
16. *Jin Kusaka, Takashi Okamoto, Yasuhiro Daisho, Ryouji Kihara, Takeshi Saito.,* 'Combustion and exhaust gas emission characteristics of a diesel engine dual- fueled with natural gas', *JSAE Review, **21** (2000), 489-496.*
17. *Selim, M.Y.E. 'Effect of engine parameters and gaseous fuel type on the cyclic variability of dual fuel engines', Fuel **84** (2005), 961-971.*
18. *Mohamed Selim, Y.E.,* 'Effect of engine parameters and gaseous fuel type on the cyclic Variability of dual fuel engines', *Fuel **84** (2005), 961-971.*
19. *Agarwal, A.K.,* 'Biofuels (alcohols and biodiesel) applications as fuels for internal combustion engines' *Progress in Energy and Combustion Science, **33** (2006), 233-271.*
20. *Ahmadi, P., I. Dincer, and M.A. Rosen., '*Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined power plants with supplementary firing*', Energy **36 (**2011).*
21. *Bahman Najafi,Vahab Pirouzpanah, Gholamhassan Najafi, Talal Yusaf, Barat Ghobadian.,* 'Experimental Investigation of Performance and Emission Parameters of a Small Diesel Engine Using CNG and Biodiesel', *SAE, (2007)Paper Number- 2007-32-0075.*
22. *Yasufumi Yoshimoto*, 'Combustion Characteristics of a Dual Fuel Diesel Engine with Natural Gas (Study with Fatty Acid Methyl Esters Used as Ignition Fuels)', *SAE(2010), SAE Paper Number: 2010-32-0050.*
23. *Karim G. A., and Burn K. S., '*The Combustion of Gaseous Fuels in a Dual Fuel Engine of the Compression Ignition Type with Particular Reference to Cold Intake Temperature Conditions', *Society of Automotive Engineers, (1980), Paper No. 800263, USA.*
24. *Karim, G. A., and Amoozegar N.,* 'Determination of the performance of a dual Fuel Diesel Engine with addition of various liquid fuels to the Intake Charge', *Society of Automotive Engineers, (1983), Paper No. 830265 USA.*
25. *Papagiannakis, R.G., C.D. Rakopoulos, D.T. Hountalas, and D.C. Rakopoulos*.,'Emission characteristics of high speed, dual fuel, compression ignition engine operating in a wide range of natural gas/diesel fuel proportions'. *Fuel, **89** (2010), 1397-1406.*
26. *Sahoo, B.B., N. Sahoo, and U.K. Saha.,* 'Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines-A critical review. *Renewable and Sustainable Energy Reviews', **13** (2009), 1151-1184.*
27. *Heywood, J.B.* '*Internal combustion engine fundamentals'*, McGraw Hill , New York, *1988.*
28. *Karim, G. A.,* 'The Dual fuel Engine', *A chapter in Automotive Engine Alternatives, Edited by R.L. Evens bleham press 1987.*

29. *Karim, G. A., Jones, W., and Raine R. R.,* 'An Examination of the Ignition Delay Period in Dual Fuel Engines'. *Society of Automotive Engineers, (1989), Paper No. 892140, USA.*
30. *McTaggart-Cowan G., W.K. Bushe, P.G. Hill, and S.R. Munshi.,* 'NO_X reduction from a heavy-duty diesel engine with direct injection of natural gas and cooled exhaust gas recirculation', *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, International Journal of Engine Research, **5**(2) (2004).*
31. *Ishida, M., T. Tagai, and H. Ueki..* 'Effect of EGR and preheating on natural gas combustion assisted with gas oil in a Diesel engine', *JSME, **46** (2003), 124–129.*
32. *Papagiannakis, R.G., D.T. Hountalas, and C.D. Rigopoulos.,* 'Theoretical study of the effects of pilot fuel quantity and its injection timing on the performance and emissions of a dual fuel diesel engine. Energy Conversion and Management', ***48** (2007) 2951-2961.*
33. *Budzianowski, W.M.,* 'Low-carbon power generation cycles: The feasibility of CO_2 capture and opportunities for integration', *Journal of Power Technologies, **91**a (1) (2011), 6-13.*
34. *N.R. Banapurmath, Basavarajappa Y.H and P.G. Tewari*, 'Effect of mixing chamber venturi, injection timing, compression ratio and EGR on the performance of dual-fuel engine operated with HOME and CNG', *International Journal of Sustainable Engineering, **5(3)** (2012), 265-279.*
35. *Pierpont, D.A, D.T. Montogomery, R.D. Reitz.,* 'Reducing particulate and NO_X using multiple injections and EGR in D.I Diesel engine', *(1995),SAE Paper No :**950217.***
36. *Budzianowski, W.M., and R. Miller.,* 'Super adiabatic lean catalytic combustion in a high-pressure reactor', *International Journal of Chemical Reactor Engineering, **7b**(2009), A20.*
37. *Budzianowski, W.M.,* 'A comparative framework for recirculating combustion of gases', *Archivum Combustionis, **30**a(1-2) (2010), 25-36.*
38. *Budzianowski, W.M.,* 'Thermal integration of combustion-based energy generators by heat recirculation', *Rynek Energii, **91b**(6) (2010), 108-115.*
39. *Prasad S.L.V., V. Pandurangadu, G. Prashanthi, V.V. Prathibha Bharathi, V.V. Naga Deepthi., '* The effect of air swirl in a diesel engine*'. African Journal of Scientific Research, **3**(1) (2011).*
40. *Love, N., Parthasarathy R.N., Gollahalli, S.R.,* 'Concentration measurements of CH and OH radicals in laminar biofuel flames. International Journal of Green Energy', ***8**(2011), 113-120.*
41. *Basavarajappa. Y.H., and. Banapurmath N.R.,* 'Effect of Exhaust Gas Recirculation on the Performance and Emissions of a Dual Fuel Engine Operated on CNG-Biodiesel-Ethanol Blends', *International Journal of Engineering Research & Technology, **2 (12)** (2013).*
42. *Benajes, J., S. Molina, J.M. Garcia, and J.M. Riesco.,* 'The effect of swirl on combustion and exhaust emissions in heavy-duty diesel engines', *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, **218(10)** (2004), 1141-1148.*
43. *Meher L.C.; Naik S.N.; Das L.M. Methanolysis of Pongmia Pinnata (Karanja) Oil for Production of Biodiesel, Journal of Scientific and Industrial Research', **63** (2004), 913-918.*
44. *Murugesan, A., C. Umarani, R. Subramanian, and N. Nedunchezhian.,* 'Bio-diesel as an alternative fuel for diesel engines - A review', *Renewable and Sustainable Energy Reviews, **13(3)** (2009),653-662.*
45. *Barnwal, B.K., and M.P. Sharma.,* 'Prospects of biodiesel production from vegetable oils in India', *Renewable and Sustainable Energy Reviews, **9** (2005), 363-378.*
46. *Mahanta P., Mishra S.C., Kushwash Y.S.,* 'An Experimental Study of Pongamia Pinnata L.Oil as a Diesel Substitute', *Proc. ImechE., Part A, Journal of Power and Energy, **220** (2006), 803-808.*