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Original Research Article

Performance, Combustion and Emission characteristics of a Manifold Injected HCNG-Biodiesel Dual Fuel Operation

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Abstract

Vehicular exhaust emission control and engine efficiency are the most important parameters in current engine design. The Euro V emission norms are directing the research towards developing new technologies for combating engine emissions. Natural gas has become a widely used alternative fuel due to its many advantages including its ready availability and its low emission levels. Addition of hydrogen to CNG improves the composition and properties of base fuel CNG. In the present work a diesel engine was converted to operate in dual fuel engine mode in which Hydrogen-compressed natural gas (HCNG) was injected into the intake manifold using a specially developed electronic control unit (ECU) while diesel/biodiesel was injected directly inside the combustion chamber during the compression stroke. Experimental investigations were then carried out on a single cylinder four stroke CI engine test rig operated in dual fuel mode using Diesel, Honge methyl ester (HOME), Jatropha methyl ester (JOME) as injected pilot fuels and HCNG as injected primary fuel. ECU was used for varying the injection timings and injection durations for HCNG injection while the optimum injection timing for diesel/biodiesel was maintained at 27°bTDC.The pilot fuel injection pressure was maintained at 230 bar. Engine was operated with constant compression ratio of 17.5 and HCNG flow rate was maintained at 0.5kg/h. The experimental results showed that an injection timing of 5°bTDC and injection duration of 40°bTDC for HCNG injection was found to be optimum based on the improved performance, combustion and emission characteristics. The brake thermal efficiency for manifold injected HCNG dual fuel operation with diesel, HOME and JOME at 80% load was found to be 28.1%, 27.2% and 26.2% respectively. The Smoke emissions, Carbon monoxide (CO), Carbon dioxide (CO_2) emissions were found to be lesser with an injection timing of 5°bTDC and injection duration of 40° bTDC in HCNG injected dual fuel operation. The use of HCNG manifold injection in the dual fuel operated CI engine improves the performance and reduces the exhaust emissions from the engine except for HC and NO_X emissions.

Keywords: Dual fuel engine; Combustion; Compression Ratio; Injection strategies; HCNG; Injector; ECU

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1. Introduction

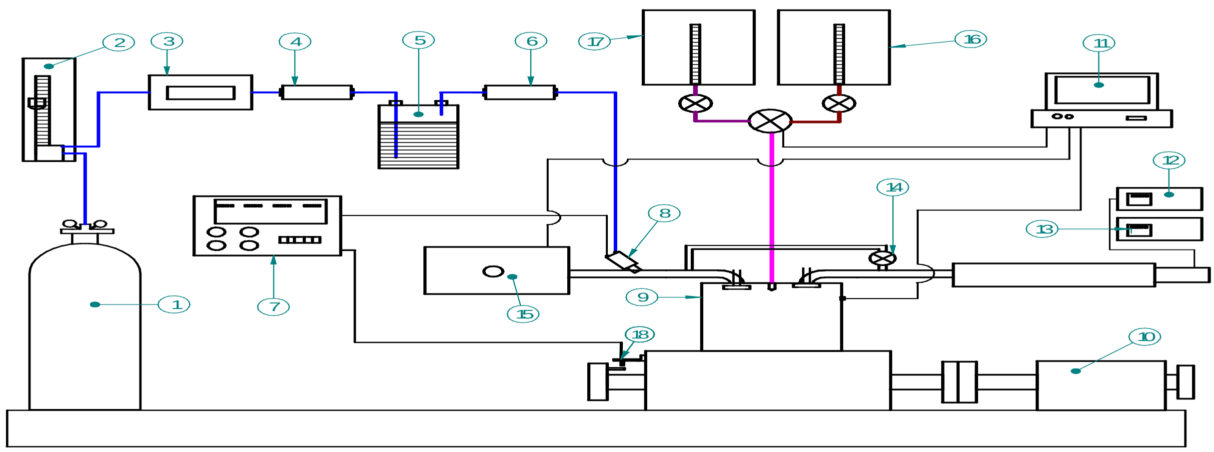
The energy crisis has necessitated the need to develop alternative sources of energy to power the automobiles. The serious problems faced by automobiles with respect to limit on reserves, recycling and polluted emissions are well known [1]. As the emission standards are becoming stringent day by day and continuous depletion of fossil fuels, make the alternative fuels (like biodiesel, CNG and Hydrogen) to gain more popularity [1-2]. Presently natural gas has become a widely used alternative fuel because of its ready availability and associated with lower emissions. Blending of hydrogen with CNG provides a blended gas termed as hydrogen-enriched natural gas. HCNG combines with the advantages of both hydrogen and methane. Natural gas and hydrogen engines have been studied with respect to different mixture percentages of natural gas and hydrogen. 5–30% (by volume) addition of hydrogen in CNG improves the composition and properties of base fuel CNG [3]. Hydrogen-compressed natural gas blends can be supplied to the engine as a mono-fuel because the phases of both hydrogen and natural gas are equal. Hydrogen has very reactive combustion characteristics, and its flammability limit is high. The production of unburned hydrocarbons can be minimized because of hydrogen’s short quenching distance, and the application of a lean-burn spark ignition engine is easy because the self-ignition temperature is high and the burning speed is fast[4,5].

High thermal efficiency of diesel engines are utilized to improve both performance and emissions [6-10]. However, diesel engines still have a major problem of higher NOx and smoke or particulate matter (PM) due to governing combustion mechanisms [3, 7, 9-11]. New methods for reducing the exhaust emissions by avoiding the combustion regions where NOx and smoke are mainly generated have been investigated [12-15]. Natural gas provides more promising alternative for conserving energy and protecting the global environment by demonstrating higher Hydrogen-to-carbon ratio than gasoline and diesel thus resulting in reduced carbon dioxide emissions [16-18]. HCNG has been mainly used in spark ignition engine applications, however, based on the properties of CNG it is best utilized in high compression ratio engine applications [18]. HCNG presents a promising prospect in dual fuel combustion (DFC) operation with pilot injection as an ignition source to increase thermal efficiency while simultaneously reducing exhaust emission in diesel engines. DFC systems are particularly promising because no much engine modifications are necessary. Dual fuel engines have an ability of reducing both PM and NOx to levels significantly lower than that of traditional diesel engines [12]. Combustion and exhaust emission characteristics of dual fuel compression ignition engines using diesel and HCNG have been reported [5, 7]. In general, an HCNG engine uses the extended flammability limit in favor of hydrogen addition so that the specific fuel consumption and the NOx emission can be improved. Further, the strategy of using an oxidation catalyst increases THC and CO emission by lean combustion [19-22]. The cylinder-to-cylinder variation affects the operation of multi-cylinder engines and can considerably influence HCNG engines because the fuel composition and excess air ratio differences can contribute to the cylinder-to-cylinder variations [22-29].Hydrogen-blended natural gas can be used for both spark ignition (SI) and CI engine applications.

Existing diesel engines may be converted readily to operate primarily on natural gas, using pilot injection of diesel/biodiesel to achieve ignition [3-6]. The injected diesel/biodiesels inside the combustion chamber will undergo combustion first which in-turn would assist to ignite the HCNG. HCNG injection assisted dual fuel engines show significant potential matching diesel engines part and full load efficiency and is equally aimed at reducing the emissions from diesel engines through dual fuel conversion by adopting different HCNG injection strategies [29].

Lower peak cylinder pressures in DFC were reported when compared to diesel single fuel combustion (SFC). Lower smoke emissions in DFC than diesel SFC were reported. Poor part load efficiency was reported for dual fuel engine operation [4-7, 25]. Flammability limit of the lean homogenous charge which leads to incomplete combustion or in the worst case absence of combustion was reported to be responsible for the observed trend [7, 25]. When the HCNG is premixed with the intake air during the induction stroke, an increase in HC emissions were also seen due to the impact of air/fuel mixture trapped in crevices in the combustion chamber during the compression stroke [7, 9, 16, 26]. Several methods of improving the poor low load characteristics, including throttling, EGR, increased inlet air temperature, or switching over to diesel mode was reported [7]. Use of micro-pilot injector to reduce exhaust emissions without unstable ignition in DFC systems has been reported [4, 11, and 27].However, most of studies on DFC systems has been conducted with diesel fuel as a pilot fuel to ignite HCNG as a main fuel. Few studies on the use of biodiesel as pilot fuel for DFC systems have been reported [7, 27-30]. Biodiesel has great potential as an alternative fuel for compression ignition engines due to its properties, which are similar to diesel. Biodiesel has a higher specific gravity and higher kinematic viscosity than diesel, as well as higher cetane number that can shorten the ignition delay. In addition, higher boiling point of biodiesel makes it easy to handle and is environmentally friendly fuel with lower smoke emissions [31-32].

Performance, emissions and combustion characteristics of single-cylinder, water-cooled, 4-stroke, direct-injection diesel engine modified to operate in dual fuel mode with HCNG manifold injection and biodiesel injection were investigated. Engine loads were controlled manually by a eddy current engine dynamometer. The dual fuel engine with suitable modifications in the injection system was undertaken to make it compatible with DFC operation. Fig. 1 shows the engine test rig equipped with a HCNG-biodiesel injection operation.



01. HCNG Cylinder
02. HCNG Rota meter
03. Gas Flow Meter
04. Dry Flame Arrester
05. Wet type flame trap
06. Dry Flame Arrester
07. Electronic Control Unit
08. HCNG Injector
09. Diesel Engine
10. Eddy Current Dynamo Meter
11. PC Interfaced to Engine
12. Exhaust Gas Analyzer
13. Smoke Meter
14. EGR Valve
15. Air Box
16. Bio Diesel Tank
17. Diesel Tank
18.Proximity sensor

Fig. 1 Schematic view of the experimental set up

The proximity sensor (TDC Encoder) is connected to cam shaft of the engine. The engine used for the experiment is 4-stroke C.I. engine and made to run at constant speed at 1500 RPM. As it is a 4-stroke engine, the number of cycles per minute is 750. The speed of the cam shaft is 750 RPM. If I do the calculations for 1o CA, the duration is equal to 0.22 milliseconds. I have considered the reference 5oBtdc as 0 milliseconds. Text and figures have been checked with this and found correct.

1.1 CNG as an alternative fuel

Major component of a natural gas is Methane. Primarily Natural Gas (NG) contains more than 90% methane (CH_4), trace amounts of carbon dioxide, propane, ethane, nitrogen, hydrogen sulphide and water vapour [30, 34-35]. In many respect natural gas is safer than gasoline and its ignition temperature is higher than gasoline and diesel [35]. Additionally, natural gas 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. Compressed natural gas is non-toxic and if spilled it will not contaminate groundwater. The main sources of natural gas are gas wells or tied in with crude oil production. Natural gas can be compressed, and hence can be stored in tanks and used as CNG. CNG requires a much larger volume to store the same mass of natural gas and requires the use of very high pressure at about 200 bar [34-38]. Advanced CNG engines promises considerable advantages over other conventional engines. CNG is a form of fossil energy and therefore non-renewable. The octane number of natural gas is about 130; hence engines could operate at compression ratio of up to 16:1 without any knock or detonation [5, 7, 34, and 39]. With encouraging CNG application policy many of the automotive manufacturers use natural gas fuelling system for transportation as consumer do not have to pay for the cost of conversion kits and required accessories. Most importantly, because of simple chemical structures (primarily methane-CH_4) which contain one carbon compared to diesel ($C_{15}H_{32}$) and gasoline (C_8H_{18}), natural gas significantly reduces CO_2 emissions by 20-25% [42-43].

1.2 Hydrogen-blended natural gas for CI Engine applications:

The non-premixed combustion of hydrogen– methane blends, as found in a diesel engine operating on late-cycle direct injected natural gas, has not been as extensively studied as the premixed case. Fundamental studies suggest that non-premixed flame stability is enhanced by the higher flame speeds and improved mixing associated with hydrogen addition [50]. In partially premixed combustion, flame thickness increases with hydrogen addition [51]. In industrial gas turbines and boilers, hydrogen addition increases NO formation (owing mainly to high H and OH radical concentrations), and flame stability is improved [52-53]. Preliminary work on a late-cycle direct injection natural gas-fuelled engine with pilot diesel ignition suggests that NOx emissions increase, while combustion stability improves and unburned fuel and PM emissions are reduced with hydrogen addition to the fuel [53].

Effects of fuelling a heavy duty diesel engine with late cycle direct injection of blended hydrogen-methane fuels and diesel pilot ignition over a range of engine operating conditions has been reported [53]. Carbon dioxide emissions were reported to be significantly reduced due to lower carbon-energy ratio of the fuel. The test results suggested that the proposed technology could significantly reduce both local and global pollutant emissions associated with heavy-duty transport applications while requiring minimal changes to the fuelling system. In this context, the experiments were conducted on a single-cylinder four stroke water-cooled direct injection (DI) CI engine operated on dual fuel mode with Honge oil methyl ester (HOME) /Jatropha oil methyl ester (JOME) and HCNG injection at optimized engine parameters and the results were compared both with diesel-HCNG operations.

2. Fuel used for the study

Diesel and biodiesel were used as pilot injection fuels and HCNG was used as the main fuel in this study. Biodiesel used in this study was produced from honge and jatropha vegetable oils. HCNG in a high pressure tank was supplied with the constant pressure of 2 bar through a double stage pressure regulator and then was injected in the inlet manifold near the intake valve with a gas injector during the intake stroke. The amount of HCNG was also controlled with a pulse generated from an Electronic control unit (ECU).Table 1 show properties of diesel, Honge (Karanja) oil, Jatropha oil and their respective esters. The important properties of natural gas used are shown in Table 2

Table 1 Properties of diesel, Honge (Karanja) oil, Jatropha oil and their respective esters and Biodiesel-Ethanol blends[13, 27, 33]

Sl No	Properties	Diesel	Honge oil	Jatropha oil	HOME	JOME	Ethanol	ASTM Standards
1	Viscosity @ 40 ^{0}C (cst)	4.59	56	50.73	5.6	4.84	1.2	ASTM D445
2	Flash point ^{0}C	56	270	240 ºC	163	192	13.5	ASTM D93
3	Calorific Value in kJ/kg	45000	35800	34000	36010	35,200	27300	ASTM D5865
4	Density kg/m^3	830	930	918	890	880	780	ASTM D4052
5	Cetane Number	45-55	40	45	40-42	40-45	----	ASTM D613
6	Type of oil	Fossil fuel	Non edible	Non edible	Non edible	Non edible	----	

Table 2 Properties of CNG and HCNG

Sl. No	Properties	CNG	HCNG
1	Density of Liquid at 15ºC, kg/ m^3	0.77	---
3	Boiling Point, K	147 K	---
4	Lower calorific value, kJ/kg	48000	47170
5	Limits of Flammability in air, vol. %	5-15	5 - 35
6	Auto Ignition Temp, K	813	825
7	Theoretical Max flame Temp, K	2148	2210
8	Flash point ^{0}C	124	---
9	Octane number	130	---
10	Burning velocity, cm/sec	45	110
11	Stoichiometric A/F, kg of air/kg of fuel	17:1	----
12	Flame temperature, ^{0}C	---	1927
13	Equivalence ratio	0.7-4 0	0.5 - 5.4

Methods of HCNG utilization in compression ignition engines

HCNG supply system plays a very vital role in its utilization especially for compression ignition engines [39]. Different techniques of HCNG supply include:

i. Carburetion
ii. Continuous/timed manifold injection
iii. Sequential/multipoint injection and
iv. High pressure direct cylinder injection

Electronically controlled HCNG port/manifold injection

The application of indirect gaseous fuel injection rather than carburetion method has provided improvements in regulating exhaust emissions and improved engine performance. Such a system introduces fuel at certain higher ambient pressure, which provides more accurate control of fuel quantity to be injected. Timed manifold injection method uses an ECU-controlled fuel injector to introduce fuel into a mixer within the intake system [35, 39].HCNG can be injected in the intake manifold by using electronically operated injector. The robust design of electronically controlled injectors has greater control over the injection timing and injection duration with quicker response to operate under high speed conditions. For both the injection systems, the ignition source could be diesel/biodiesel or ethers. The advantage of HCNG injection over carbureted system is that with proper injection timing the problems of backfire and pre-ignition can be eliminated. This leads to better control of mixture formation and quick response to changing loads can be achieved. Manifold injection system also enables fuel to be delivered precisely and provides more sophisticated technologies such as skip-firing to be used. This provides even more efficient use of the fuel at low loads, further lowering fuel consumption and unburned hydrocarbon. A high-speed gas jet is pulsed from the intake port through the open inlet valve into the combustion chamber, which causes turbulence and charge stratification, particularly at engine part load operations [34]. This system is able to diminish the cyclic variations and enhance the limit of lean operation of the engine. The flexibility of gas pulse timing provides the potential advantage of lower emissions and fuel consumption. In order to greatly reduce exhaust gas emissions, use of manifold or port injection system has been reported [35-39]. Manifold injection of HCNG produces negligible levels of CO, CO2 and NOx [34-35, 46-47]. With sequential port injection, a high-speed gas jet is pulsed from the intake port through the open intake valve into the combustion chamber, where it causes turbulence and charge stratification, particularly at engine part load operations. The flexibility of gas pulse timing offers the potential advantage of lowered emissions and lesser fuel consumption [34-39, 48].

HCNG Injector

In principle, optimal design of fuel-air mixture for producing the required power output with the lowest fuel consumption that is consistent with smooth and reliable operation is required [34-35].Sequential port injection (SPI) system has evolved into a pulse-width-modulated electronic system which uses sequentially timed individual injections at each intake port. A significant amount of work has undertaken on the fuel-air mixing process and less information has been reported on the structure of the fuel spray from SPI injectors [38].

The solenoid operated gas injector receives a constant power supply from a 12 V battery. The technical specifications of the gas injector are given in Table 3.The HCNG injector is fitted to the inlet manifold along perpendicular axis of the intake valve HCNG injector with rail and ECU showing circuit diagrams shown in Fig. 2.

Table 3 The specifications of HCNG injector

Make	BRC Italy
Supply voltage	8–16 V
Peak current	4 A
Holding current	1 A
Flow capacity	0.8 g/s at 4 bar
Working pressure	103–552 kPa
Max. Rail pressure	4bar

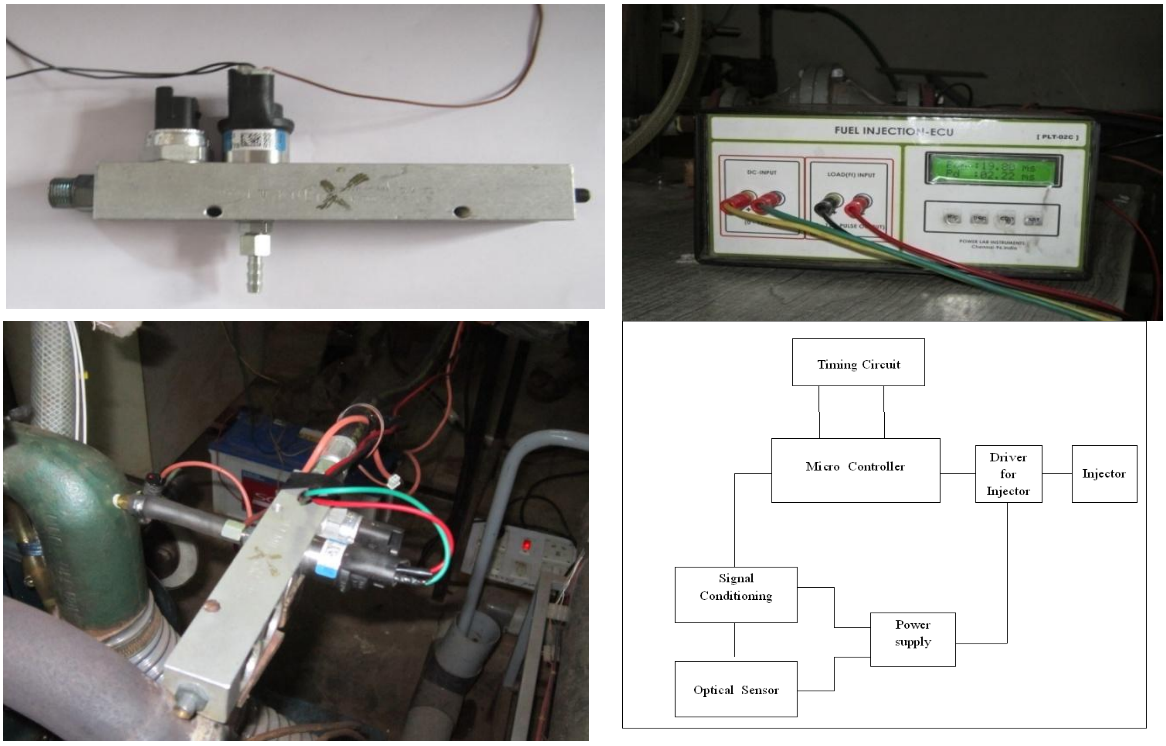


Fig. 2 HCNG injector with rail and ECU showing circuit diagram

Accuracy of measurements and uncertainty analysis

In order to ensure accuracy in measurements made, all instruments used in this study were calibrated before the experimentation. Table 4 shows the specifications of the emission measuring equipments.

Table 4 The specifications of emission measuring equipments

Sl.no	DEVICES	MAKE	ACCURACY
1	Exhaust gas analyzer	DELTA 1600S	HC = +/- 30 ppm HC CO = +/- 0.2% CO CO_2 = +/- 1% CO2 O_2 = +/- 0.2% O2 NO_X = +/- 10 ppm NO
2	Smoke meter	HARTRIDGE SMOKEMETER-4	+ / -2 % relative
3	Impeller intelligent Gas flow meter	GS-LY 25/16	+/-3%

To maintain the accuracy of the measured values, after completion of the work on a selected test condition, the gas analyzers were purged and then calibrated before the next measurement. The smoke meter was also adjusted to its zero point before each measurement. The values reported for all measured parameters are time averaged at each test condition. The consumption rate of the pilot fuel injection was averaged over 6 measurements to account for the cycle-to-cycle variations. The cylinder pressure for combustion analysis was averaged for 100 cycles at each experimental condition. To examine the accuracy of measurement and the limiting errors associated with each measured parameter, comprehensive uncertainty analysis was conducted. The uncertainty of the measured parameters could be estimated with confidence limits of $\pm 2\sigma$ (95.45% of the measured data lie within the limits of $\pm 2\sigma$ around the mean). The percentage uncertainty of the measured parameters can be estimated using the following relation:

$$\Delta xi = \frac{2\sigma x_i}{\bar{x}_i} \times 100 \text{-------}(1)$$

Where

σ = Standard deviation

Δx_i=Difference between readings

x_i=Difference between readings

$\bar{x_i}$ = *Mean difference in readings*

Engine operating method and test conditions

Engine performance and exhaust emission characteristics of dual fuel combustion (DFC) were investigated with HCNG-diesel and HCNG-Biodiesel combinations. Pilot injection pressure was maintained constant at 230 bar and the engine speed at 1500 rpm respectively. An injection timing of 27° bTDC and compression ratio of 17.5 was maintained throughout the experiment. The test engine was operated with 70 ± 2 ^{0}C cooling water temperature. The schematic view of the experimental set up is shown in Fig. 1.

The injection timings and durations for the HCNG injection were controlled through ECU. The electronic control unit acquires the signal from an infrared sensor that indicates the crankshaft position. The injection timings of the HCNG injector were varied from 10^{o}CA before top dead centre (bTDC) to 10^{o}CA after top dead centre (ATDC) in steps of 5degree CA. The trigger point was set to5^{o}CA before top dead centre. The injection duration was varied from20^{o}CA to60^{o}CA in steps of 20^{o}CA.Optimized timings for HCNG injection was then determined. The HCNG flow rate for the optimized injection timings was maintained at 12 LPM (0.5 kg/h).

HCNG was passed through a fine control valve to adjust the flow rate. The HCNG was allowed to pass through HCNG rotameter and a digital mass flow controller, which meters the flow rate of HCNG in standard liters per minute. HCNG was then passed through a dry type flame arrestor used to suppress possible fire hazards in the fuel line [7, 9, 12, 49-50]. The flame arrestor (also acts as a non-return valve) operates on the basic principle that the flame gets quenched if sufficient heat can be removed from the gas. The HCNG was then allowed to pass through the flame trap, used to suppress the flash back if any in to the intake manifold. The flame trap used for the investigation was wet-type. The HCNG from the cylinder after passing through the flame trap was injected by using a gas injector placed in the inlet port. The engine was operated with Diesel/biodiesel and HCNG fuel combinations. The pictorial view of the experimental set up is shown in Fig. 3 with gas supply arrangements used.

3. Results and discussion

The experimental investigations were carried out on a single cylinder four stroke CI engine test rig modified to operate in dual fuel mode. Earlier studies [15, 27] reported by the authors highlighted that advancing the injection timing of pilot fuel from 19^o to 27^o improved the HCNG-Biodiesel performance. In the present work, start of injection and injection duration of the HCNG were optimized based on the improved performance and emission characteristics of the dual fuel engine keeping pilot injection timing of 27^{0}bTDC, compression ratio of 17.5 and pilot injection pressure of 230bar.

3.1. Performance Characteristics

This section presents the results of investigation carried out on a single cylinder, DI engine operating on diesel, HOME, JOME along with HCNG injection in dual fuel mode of operation for different HCNG injection timings and injection durations. Engine tests have been done with the aim of obtaining comparative measures of brake thermal efficiency.

Brake thermal efficiency

Figure 4 shows the variation of Brake Thermal Efficiency (BTE) of a dual fuel engine fuelled with diesel, HOME, JOME with HCNG injection at 80% loads. HCNG being common, properties of the injected liquid blend combinations is responsible for the observed trends. HOME and JOME obviously will have reduced calorific value and higher viscosity and therefore perform poorly when compared to the diesel.

Gas supply arrangement

Gas supply to intake manifold

Fig. 3 Pictorial view of the experimental set up

Higher brake thermal efficiency of 28.1%, 27.2% and 26.2% were observed with DIESEL+HCNG (D+HCNG), HOME+HCNG (H+HCNG) and JOME+HCNG (J+HCNG) manifold injections when HCNG injection timing of 5^{o}bTDC and injection duration of 40 degree crank angle was used. The reason for this increased BTE is due to better combustion taking place inside the engine cylinder due to manifold HCNG injection timing of 5^{o}bTDC, gets adequate time to mix homogeneously with the incoming air during suction stroke. Combustion was found to be smoother with optimum injection timing in manifold injection. The general reason for the improvement in performance could be due to better mixing of HCNG with air. The higher calorific value of HCNG is also responsible for this trend. Complete burning of the injected liquid fuel as well as injected gaseous fuel leads to better combustion. Supplying gas to the intake manifold through injectors at appropriate timing provides proper air/fuel mixture available for combustion [25, 36, 51-52]. The presence of hydrogen allows the lean burn limit to be extended because of the fast burn rate of hydrogen. The expansion of the flammability limit influences the reduction in loss by high combustion temperature and heat transfer; hence, the thermal efficiency was improved. The fast burn rate of hydrogen causes the combustion duration to decrease, while the heat release rate and exhaust NOx increase with an increased percentage of hydrogen [54].

3.2 Emission characteristics

3.2.1 Smoke Opacity

Figure 5 shows the variation of smoke with injection timing and injection duration for D+HCNG, H+HCNG and J+HCNG dual fuel operation at 80% load respectively. Higher viscosity (nearly twice the diesel) of the injected HOME and JOME is responsible for the observed trends of higher smoke emissions. The heavier molecular structure of the biodiesels used results into improper fuel air mixtures compared to diesel. The smoke opacity reduces with increased HCNG injection duration. Lower smoke of 43.6, 53.3 and 56 HSU was observed with D+HCNG, H+HCNG and J+HCNG manifold injection dual fuel operation when engine was operated with HCNG injection timing of 5°bTDC and injection duration of 40°CA.The main reason for this decrease in smoke is mainly due to better combustion prevailing inside the engine cylinder [8,25,27]. 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 HCNG produces no particulates and smoke produced is mainly due to the pilot injection of diesel/biodiesels used.

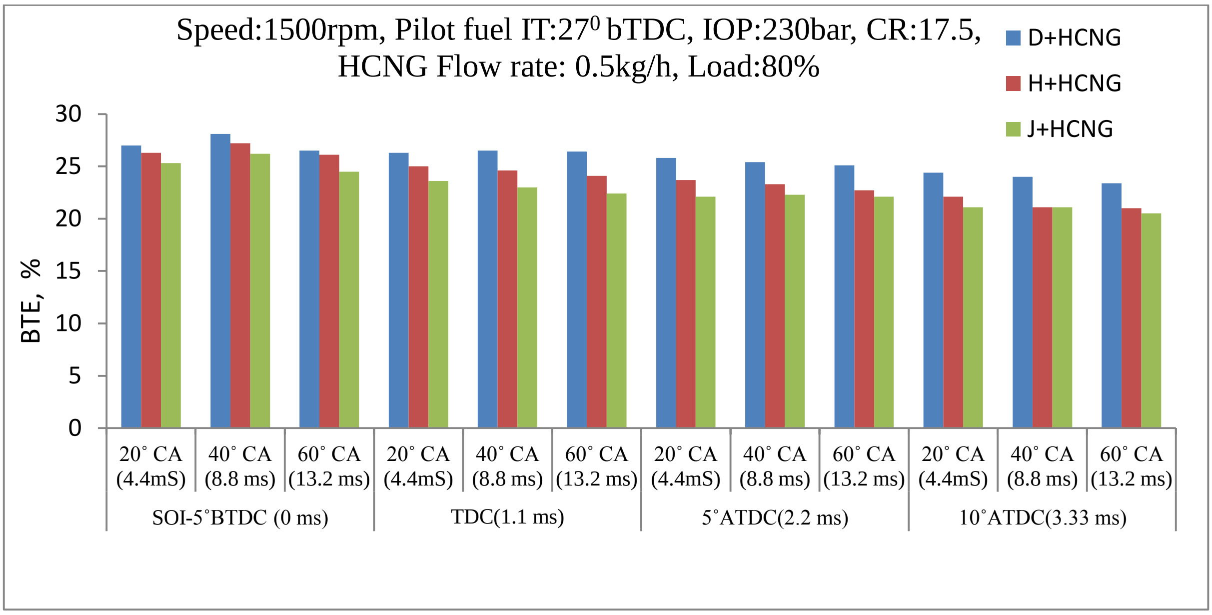


Fig. 4 Variation of BTE with HCNG injection timing and injection duration

Methane and hydrogen, the primary constituents of HCNG, has no carbon–carbon bonds with high hydrogen to carbon ratio, which leads to lower sooting tendencies [43-46].This may also be due to less carbon content and clean burning characteristics of gaseous fuels. The higher burning velocity and flame temperature of HCNG leads to more better burning compared to CNG during the dual fuel operation. Furthermore, the natural gas has enough residence time in traveling from the intake manifold to the combustion chamber to form a well-mixed mixture prior to the combustion.

3.2.2 Hydrocarbon emissions

Figure 6 shows the variation of HC with injection timing and injection duration for D+HCNG, H+HCNG and J+HCNG dual fuel operation for 80% load respectively. The overall HC emissions levels using DFC is higher due to the HCNG 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. But it is also observed that HC emissions for DFC can be improved with increased engine load. For DFC, it can be seen that there is no noticeable trend of HC in variation of pilot injection timing other than for all cases HC emissions are reduced with increased engine load [3, 12]. There are three main reasons for this phenomenon, firstly, flame quenching. When the flame approach the combustion chamber wall, the temperature of the mixture is too low to complete the combustion leaving a layer of unburned mixture. Secondly, some of the mixture is compressed into the crevices of the combustion chamber during the compression stroke, which misses the primary combustion process. Finally, the mixture is too fuel-lean for combustion to propagate throughout the charge [12, 21]. The lower BTE of the biodiesel could be responsible for the increased HC emissions compared to diesel with HCNG injection in both versions. The higher viscosity and lower volatility of the biodiesel blends could also be responsible for the increased HC emissions. HOME being common, the properties of the two gases inducted results in the behavior shown and accordingly HOME+HCNG operation results in lower HC emissions compared to HOME+CNG operation. The H_2 content gives a strong reduction of unburned hydrocarbon emission results in more complete combustion [45]. In addition HCNG engine increases the H/C ratio of the fuel, which drastically reduces the carbon based emissions. The presence of hydrogen with CNG (HCNG) has higher flame velocity and flame temperature results in better combustion compared to CNG. The HC emissions with D+HCNG, H+HCNG and J+HCNG dual fuel operation with manifold injection when engine was operated with HCNG injection timing of 5^{o}bTDC and injection duration of 40 degree crank angle were found to be 52, 55.6 and 59.7ppm respectively.

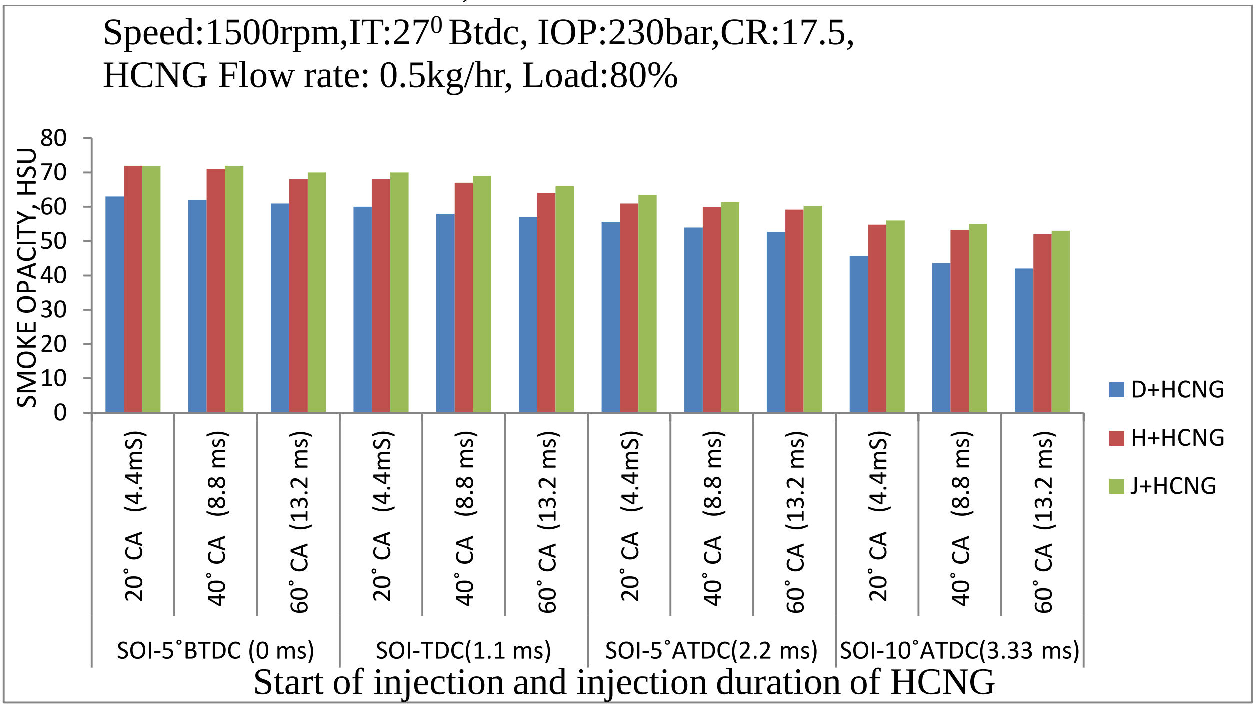


Fig 5 Variation of Smoke with HCNG injection timing and injection duration

3.2.3 CO emissions

Figure7 shows the emissions of CO for diesel / biodiesel + HCNG DFC with the variation of HCNG injection timing and duration of injection. It is observed that CO emissions were generally higher over the range of operating conditions for biodiesel blended fuels. This is mainly due to the lower combustion rates of DFC caused by increased area of quenching inside combustion chamber with BIODIESEL+HCNG dual fuels [21]. The lower BTE of the biodiesel could be responsible for the increased CO emissions compared to diesel with HCNG injection in both versions. The higher viscosity and lower volatility of the biodiesel blends could also be responsible for the increased HC emissions. The CO emissions with D+HCNG, H+HCNG and J+HCNG dual fuel operation with manifold injection when engine was operated with HCNG injection timing of 5^{o}bTDC and injection duration of 40 degree crank angle were found to be 0.062, 0.08 and 0.11% respectively. It should be noted that relatively high CO emissions were observed at low load conditions. Due to the lack of oxygen, the combustion is incomplete in the over rich region, which resulted in higher CO formation. However, in the fuel lean region, when the combustion temperature was less than 1450 K, a large number of CO was generated [21].

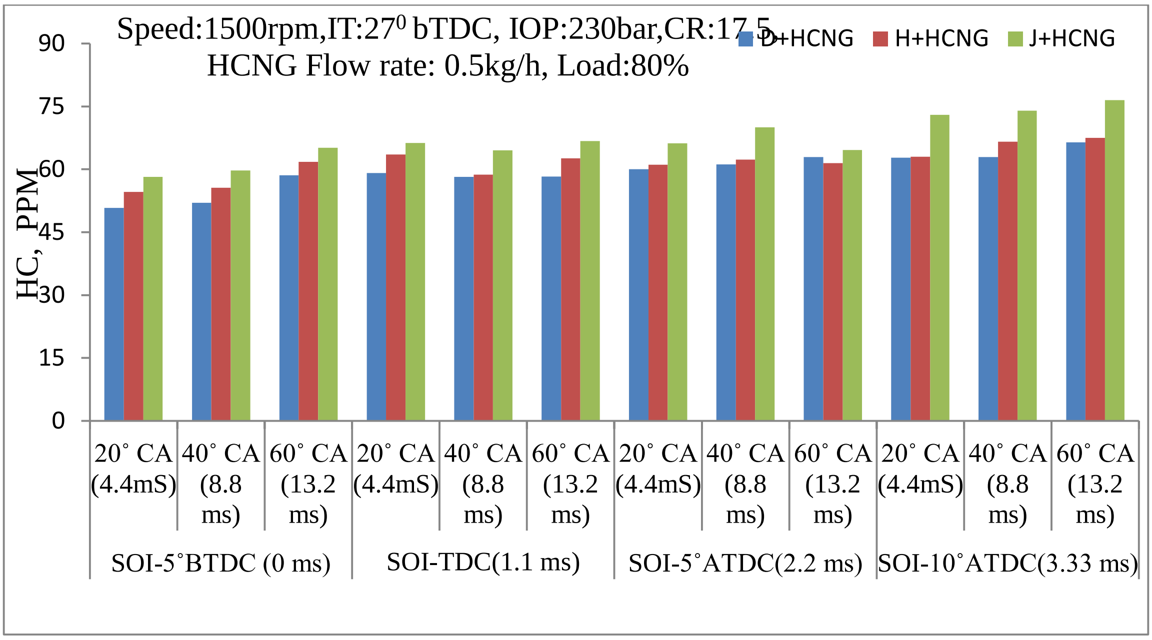


Fig.6 Variation of HC with HCNG injection time and injection duration

This temperature is also the limited temperature of the flame extinction [40]. There are three main reasons for the generation of CO, firstly the temperature of the reaction mixture being suddenly too low. Secondly the sudden lack of oxidant and finally, the suitable reaction time being too short [12].Moreover, the combustion temperatures are higher with HCNG fuel and the engine runs hotter thereby facilitating better combustion. The dual fuel operation yield higher CO emissions at low load conditions. It could be due to most of the fuel left is un-burnt and leading to poor combustion. At higher loads the CO emission are also observed to be higher compared to liquid fuel operation because of decreased combustion temperature and lower brake thermal efficiencies.

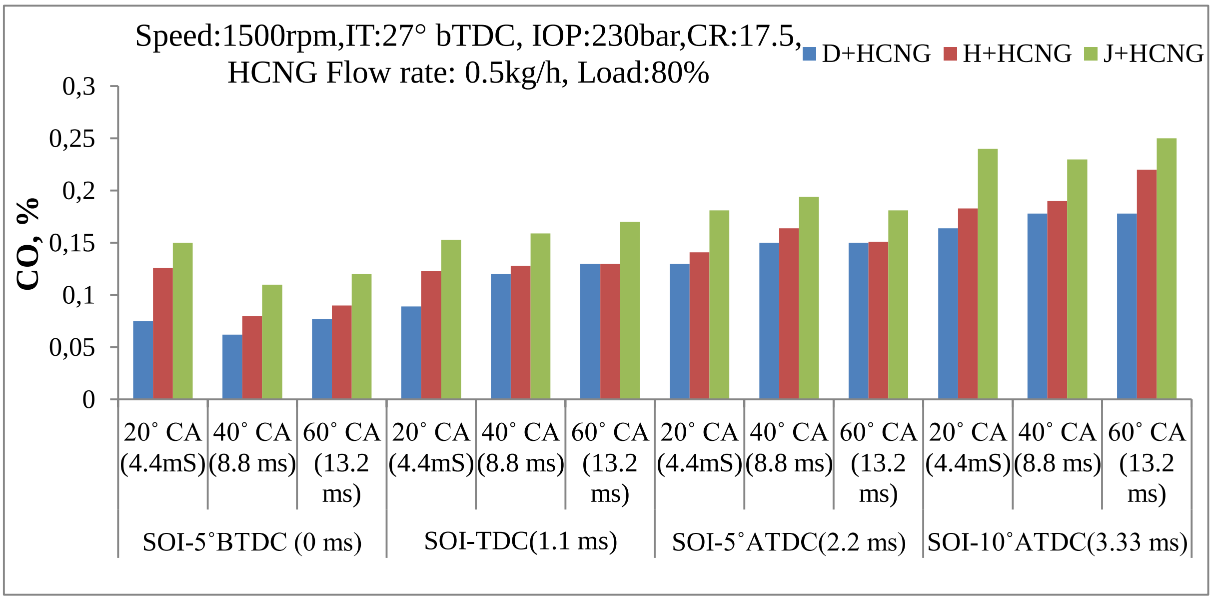


Fig 7 Variation of CO with HCNG injection timing and injection duration

3.2.4 NO_x emissions

Figure 8 shows the variation of NO_x with injection timing and injection duration for D+HCNG, H+HCNG and J+HCNG dual fuel operation for 80% load respectively. Higher NOx was observed for HCNG-

Diesel dual fuel operation than HCNG – Biodiesel dual fuel operation. This could be due to very high temperature prevailing inside the combustion chamber that leads to higher NO_x [53-54]. However this can be controlled with the introduction of suitable EGR. NOx is formed in greater quantity with high peak combustion temperatures, sufficiently high oxygen concentrations and long residence time. Significant NOx is produced when the local temperature is above 2000 K for mixtures at or below stoichiometric [12].Higher NOx formation is due to higher temperature in the flame zone under high load conditions. It has been found in other studies that the double bonds contained in biodiesel are likely to result in higher levels of certain radicals that promote the prompt NOx emissions [47]. Moreover, the induction of HCNG increases the specific heat capacity of the working fluid which thereby causes slowing of the flame propagation and lowering of the combustion temperature during the combustion process compared to the single-fuel mode [46, 53].Compared to pure HOME –CNG operation, it has been concluded that the presence of hydrogen increases the NOx emissions while reducing the HC emissions. It is observed that the NOx emissions for the HOME-HCNG operation are greater than the emissions of pure HOME-CNG operation. This is because of the elevated flame temperature due to the hydrogen. However, the NOx emissions of the HOME-HCNG are still considered relatively low compared to other diesel and HOME operation. The NOx emissions with D+HCNG, H+HCNG and J+HCNG dual fuel operation with manifold injection when engine was operated with HCNG injection timing of 5^{o}bTDC and injection duration of 40 degree crank angle were found to be 765, 785 and 850 ppm respectively.

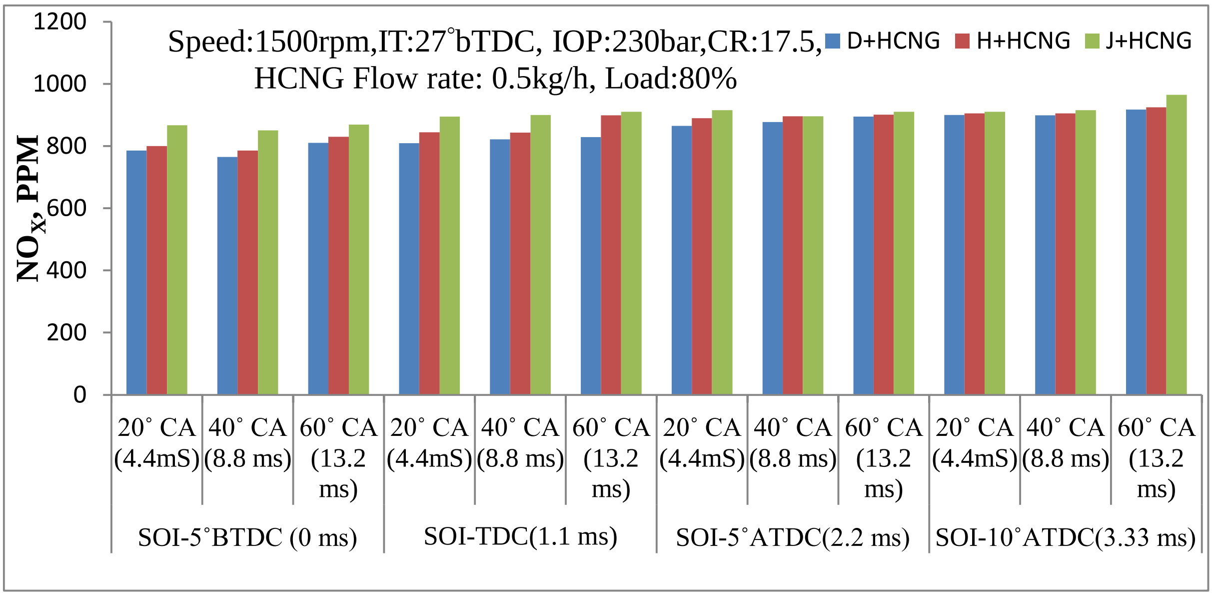


Fig.8 Variation of NO_x with HCNG injection timing and injection duration

3.3 Combustion characteristics

This section covers the various combustion aspects of the dual fuel engine operation using the selected biodiesel fuel combinations.

3.3.1 Peak pressure

Figure 9 shows variation of peak pressure with different modes of dual fuel engine operation. Peak pressure and maximum rate of pressure rise were higher with HCNG injection timing of 5^{o}bTDC and injection duration of 40 degree crank angle for D+HCNG, H+HCNG and J+HCNG dual fuel operation for 80% load respectively. Rate of pressure rise is lower in dual fuel operation as compared to single fuel operation [53]. HCNG being common, properties of the injected liquid fuels are responsible for the observed trends. Lower cetane number and lower calorific value of the biodiesels used results into lowered peak pressure when compared to diesel. The peak pressure with D+HCNG, H+HCNG and J+HCNG dual fuel operation with manifold injection when engine was operated with HCNG injection timing of 5ºbTDC and injection duration of 40 degree crank angle were found to be 76.21, 72.8 and 71bar respectively.

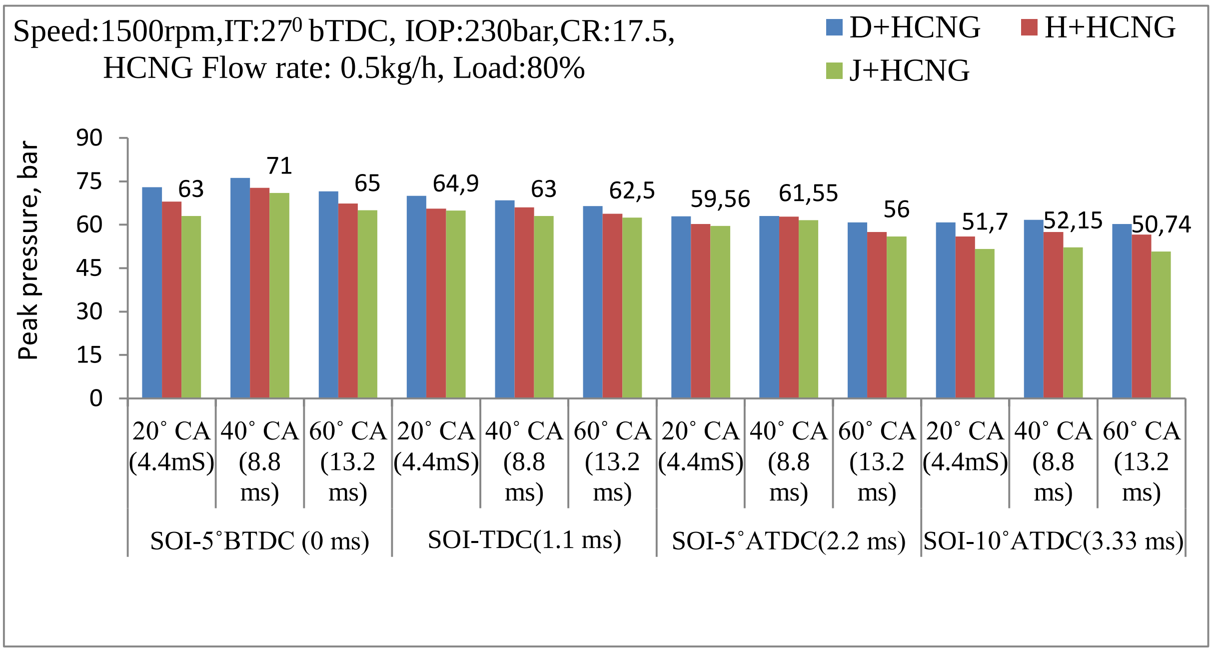


Fig.9 Variation of Peak pressure with HCNG injection timing and injection duration

3.3.2 Ignition delay

Figure 10 shows the variation of ignition delay with injection timing and injection duration for D+HCNG, H+HCNG and J+HCNG dual fuel operation for 80% load respectively. The time difference between the start of injection and ignition is termed as ignition delay in CI engines. Higher auto ignition temperature of HCNG, leads to more ignition delay. Once the HCNG ignites, further mixture ignites without any difficulty. Due to its high auto ignition temperature, initially at zero load and part load it takes more time to ignite. But at full load the ignition delay for HCNG enriched fuel is found to be lower because of the high heating value of HCNG. Higher viscosity and lower cetane number of the biodiesel shows higher ignition delay when compared to diesel. The ignition delay found to be lesser for all fuel combinations when engine was operated with HCNG injection timing of 5ºbTDC and injection duration of 40 degree crank angle. At 80% load the ignition delay was 12.3, 12.4 and 12.8°CA with D+HCNG, H+HCNG and J+HCNG.

3.3.3 Combustion duration

Figure 11 shows the variation of NO_x with injection time and injection duration for D+HCNG, H+HCNG and J+HCNG dual fuel operation for 80% load respectively. The combustion duration was calculated based on the duration between the start of combustion and 90% cumulative heat release. The combustion duration is found to be more for H+HCNG and J+HCNG compared to D+HCNG dual fuel operation. This could be due to the higher viscosity and lower calorific value of the biodiesels used. Higher combustion duration was observed with dual fuel combinations compared to single fuel operation. Improper air–fuel mixing, longer time for mixing results in incomplete combustion with increased diffusion combustion phase. H+HCNG dual fuel operation shows improvement in heat release rate compared to HOME+CNG (H+CNG) operation with increased premixed combustion phase.

Higher flame velocity, higher calorific value and fast burning rate of hydrogen in CNG causes the combustion duration to decrease while the heat release rate and exhaust NOx increase with hydrogen addition. The Combustion duration found to be short for all fuel combinations when engine was operated with HCNG injection timing of 5°bTDC and injection duration of 40 degree crank angle. At 80% load the combustion duration was 38, 40 and42°CA with D+HCNG, H+HCNG and J+HCNG.

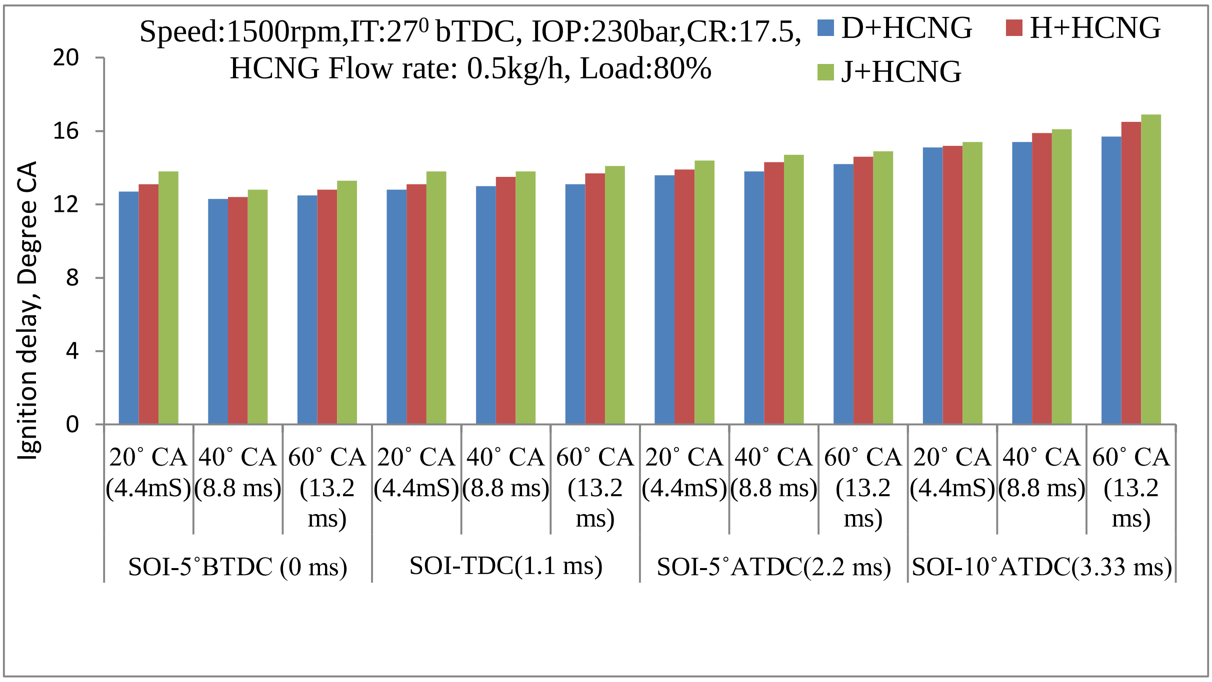


Fig.10 Variation of Ignition delay with HCNG injection timing and injection duration

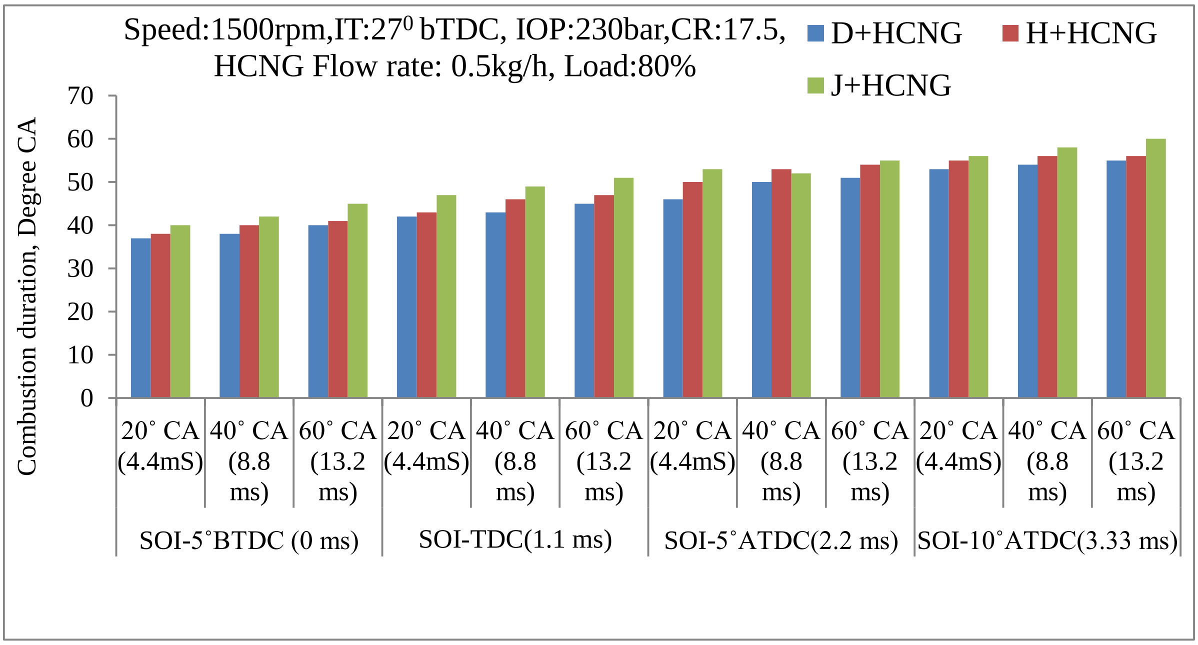


Fig.11 Variation of combustion duration with HCNG injection timing and injection duration

3.3.4 In-cylinder pressure

Figure 12 and 13shows the variation of in-cylinder pressure with crank angle for a HCNG-HOME dual fuel operation for different HCNG injection duration and HCNG injection timings. The rate of pressure rise is higher with HCNG injection time of 5°bTDC and injection duration of

40°CA. It is observed that the peak in-cylinder pressure for HOME+HCNG (H+HCNG) DFC is slightly lower than D+HCNG dual fuel combustion. The injection of HCNG during the suction process reduces the in-cylinder temperature and hence causing the primary fuel combustion to be unstable. This is primarily as a result of HOME's low lower heating value relative to that of regular diesel fuel. Similar trend was observed with J+HCNG dual fuel mode operation. The peak pressure depends on the combustion rate and that how much fuel is taking part in rapid combustion period. The uncontrolled combustion phase is governed by the ignition delay period and by the mixture preparation. Higher cylinder pressure was obtained for diesel operation due to its rapid mixing of diesel particles with air and higher heat release rate during rapid combustion phase. Combined effect of poor mixture preparation, longer ignition delay, lower calorific value and adiabatic flame temperature and slow burning nature of the HOME+CNG/HCNG resulted in lower peak pressure and maximum rate of pressure rise compared to single fuel operation. However, the higher flame velocity, calorific value and slightly increased ignition delay of HCNG during dual fuel operation leads to increased combustion during rapid combustion phase. Hence it results in to higher peak pressure and maximum rate of pressure rise. Accordingly higher peak pressure and heat release rate were observed for H+HCNG operation due to the higher flame velocity and fast burning of hydrogen content in presence of CNG.

The peak pressure developed inside the engine cylinder is found to be higher for D+HCNG, H+HCNG and J+HCNG.

3.3.5 Heat release rate

Figure 14 and 15 shows the comparison of rate of heat release rate (HRR) for H+HCNG dual fuel engine operation when operated at different HCNG injection duration and HCNG injection timings. It is observed that heat release rates for HCNG is premixed type combustion, in contrast to typical diffusion type combustion of liquid fuel operation. The reduced premixed combustion observed with biodiesel combinations could be responsible for the reduced HRR when compared to diesel operation. HCNG being common, the properties of the injected liquid fuel combinations could be responsible for the observed HRR rates. The similar trend of HRR was also observed with J+HCNG dual fuel operation. HRR for H+HCNG manifold injection dual fuel operation with HCNG injection timing of 5°bTDC and injection duration of 40°CAwas found to be 89.32J/CA respectively.

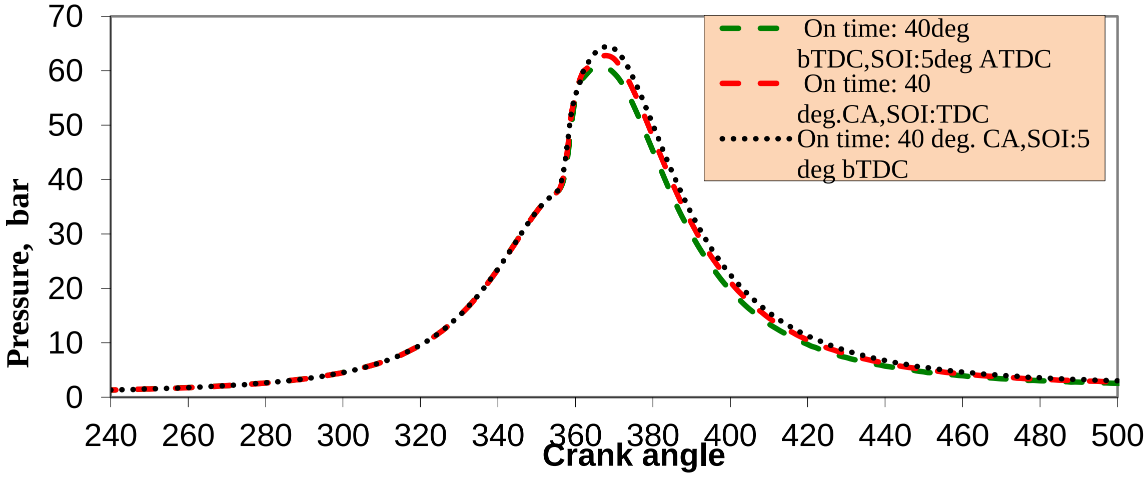


Fig. 12 Variation of In-cylinder pressure with HCNG injection timing

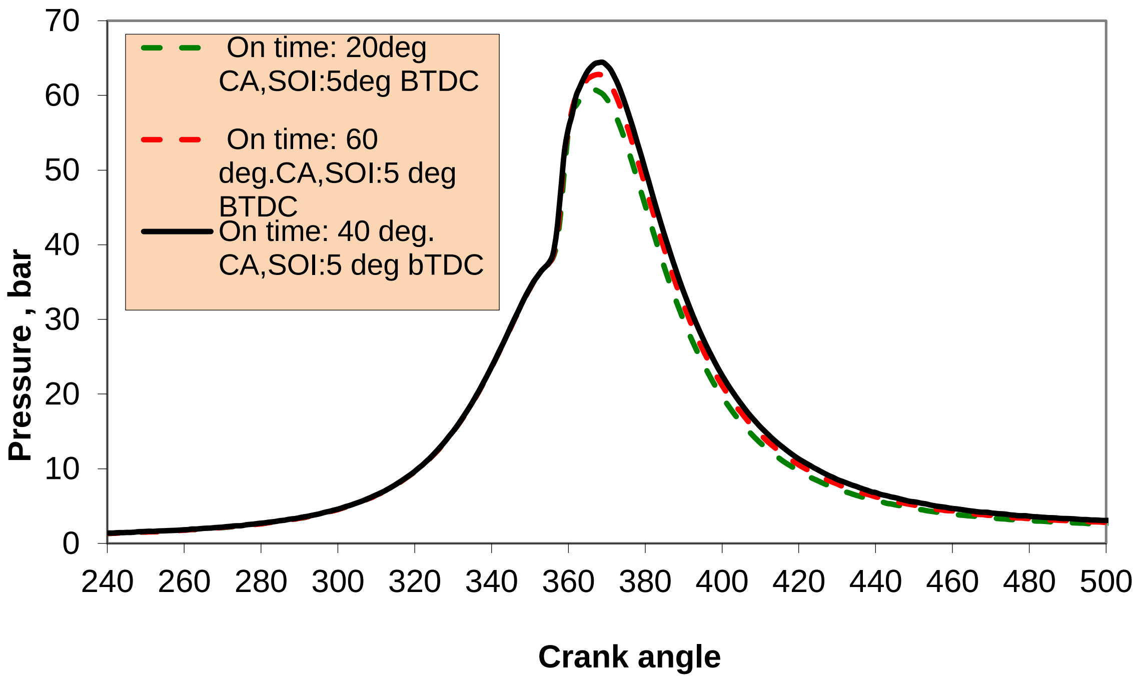


Fig. 13 Variation of In-cylinder pressure with HCNG injection duration

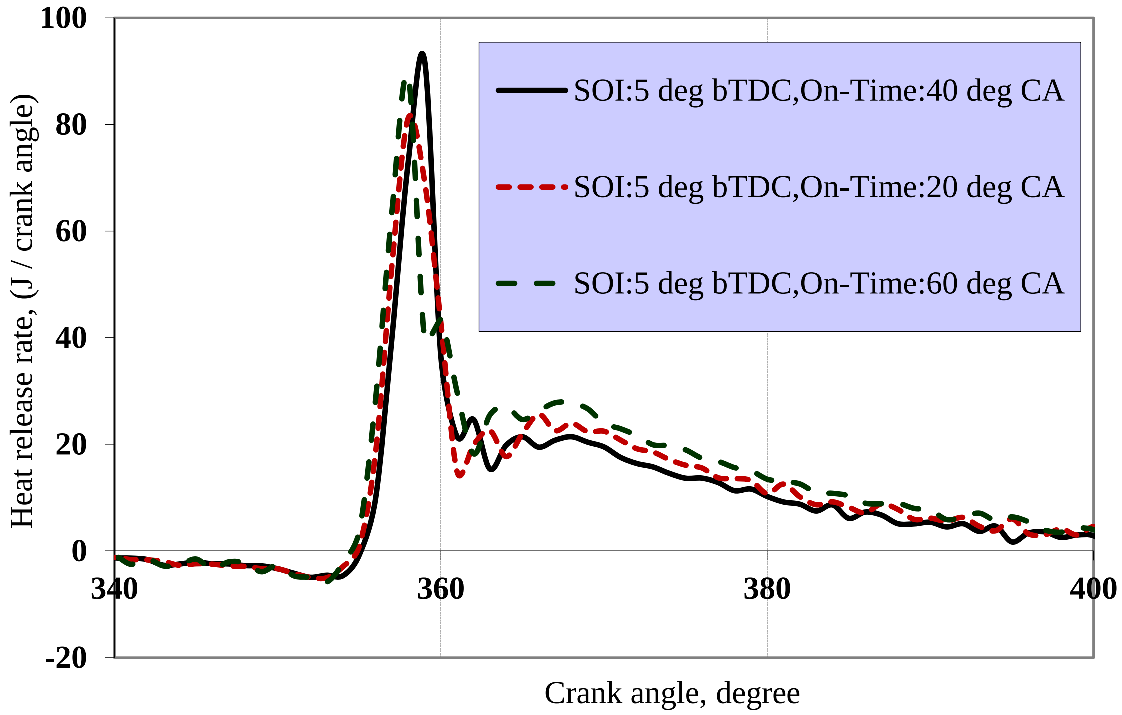


Fig. 14 Variation of HRR with HCNG injection duration

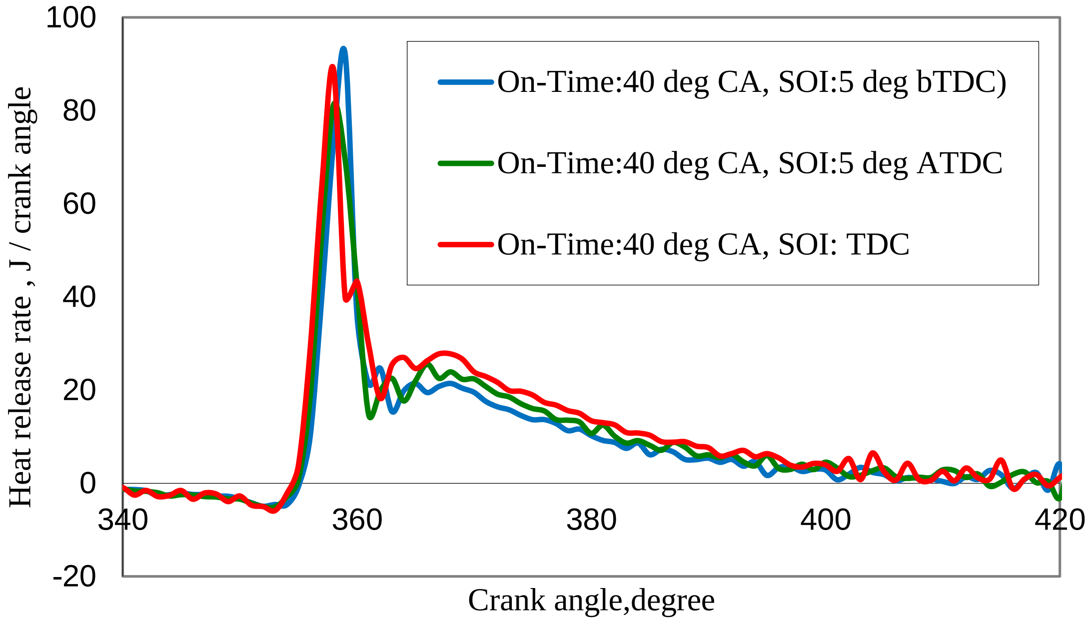


Fig. 15 Variation of HRR with HCNG injection time

4. Conclusions

Electronically controlled gaseous fuel injection system is considered to be the most viable technology to reduce emissions in HCNG dual fuel engines. This approach helps to achieve a homogeneous mixture of air and fuel before the air stream splits in the intake manifold. It is observed that HCNG manifold injection causes turbulence and charge stratification, particularly at engine part load operations and is able to diminish the cyclic variations and enhance the limit of lean operation of the engine. The gas pulse timing offers the potential advantage of lower emissions and lesser fuel consumption. The experimental results showed that an injection timing of 5^{o}bTDCwith injection duration of 40^{o}CA was found to be optimum based on the improved performance, combustion and emission characteristics. Use of renewable fuels of HOME and JOME can partially eliminates use of fossil fuels and this ensures sustained energy security for the developing countries like India. The brake thermal efficiency in manifold injected DIESEL/BIODIESEL+HCNG dual fuel operation for 80% load was 28.1%, 27.2% and 26.2% for DIESEL, HOME and JOME respectively. Reduced smoke emissions and carbon monoxide (CO) were obtained with the optimized HCNG injection timings. The use of HCNG manifold injection in the dual fuel mode in the modified CI engine improves the performance and reduces the exhaust emissions from the engine except for HC and NO_X emissions. However NOx can be effectively controlled with suitable EGR arrangement.

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