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Experimental investigations of combustion, performance and emission characteristics of a hydrogen enriched natural gas fuelled prototype spark ignition engine

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HIGHLIGHTS

- Effect of hydrogen enrichment of natural gas on engine performance.
- Fuels with different H/C ratios (4, 4.22, 4.5, 4.85, 5.33 and ∞) were investigated.
- Brake thermal efficiency was superior for test fuel with H/C: 4.5.
- P_{max} increases with increasing H/C ratio of the test fuels.
- HRR was highest for hydrogen along with shortest ignition delay.

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ABSTRACT

In this study, spark ignition of hydrogen enriched natural gas (HCNG), a fast emerging alternative gaseous fuel, was experimentally investigated in a suitably modified single cylinder spark ignition (SI) engine. Port fuel injection of the HCNG engine using a high volume flow rate solenoid injector, controlled by a customized injector control unit and electronic control unit (ECU) was done and the fuel injection timings and duration were controlled for each load. Fuels with different H/C ratios in the final HCNG mixture were investigated for their engine performance, emissions and combustion characteristics. Engine investigations were carried out at constant engine speed of 1500 rpm for different H/C ratios (4, 4.22, 4.5, 4.85, 5.33 and ∞). Spark timing was kept constant (32° bTDC) for all test blends. Relative air–fuel ratio (RAFR) was kept constant for all loads during the experiments in order to avoid misfire at lower engine load. Hydrogen exhibited higher pressure peak (P_{max}) but lower maximum brake torque (MBT) compared to other test fuels due to lower knocking limit. Brake thermal efficiency (BTE) was superior for test fuel with H/C: 4.5. NOx emissions were higher for test fuel with H/C: 4.22 and relatively lower for hydrogen compared to baseline natural gas.

1. Introduction

Rapid increase in energy demand and resulting consumption of conventional fossil fuels has led to rapid depletion of underground carbon energy reserves and increasing fuel prices. This has increased dependency of all major global economies on gulf countries. In addition, it has also adversely affected air quality significantly, resulting in severe environmental degradation and climate change. Conventional fossil fuels are responsible for emission of harmful species such as unburned hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NOx), particulate matter (PM) and carbon dioxide (CO_2). These pollutants have severe health effects on human body and the global environment. Therefore commercialization of prominent low carbon or carbon free alternative fuels such as natural gas and hydrogen is necessary for the survival of humanity. These fuels have potential to reduce harmful green-house gas (GHG) emissions and could displace a portion of conventional liquid fossil fuels.

Both these fuels however offer different challenges for their utilization in internal combustion (IC) engines. For example, hydrogen requires very low ignition energy therefore utilization of hydrogen in IC engines can potentially cause pre-ignition and backfire [2]. Use of hydrogen also leads to low power output and constraints the operating load range of the engine because of very low density of hydrogen, which in-turn reduces the volumetric

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efficiency of the engine significantly. Therefore use of 100% hydrogen as a total replacement of gasoline in a spark ignition (SI) engine is rather challenging and difficult. On the other hand, natural gas is being used as an alternate fuel to conventional gasoline for last few decades. It is utilized either in the form of compressed natural gas (CNG) or liquefied natural gas (LNG). Natural gas resources have been discovered in various forms worldwide. There is tremendous interest in using natural gas on a large scale worldwide because it is relatively cleaner fuel due to its highest H/C ratio (4:1) amongst all hydrocarbon fuels. However natural gas suffers from poor lean-burn capabilities, low flame speed and poor idle stability [3], which makes CNG engine relatively low efficiency engine due to its longer combustion duration [4] and less agility. Engine's lean operation could be extended by increasing H_2 fraction in the test fuel and also by increasing intake manifold pressure [5]. In summary, both hydrogen and CNG have their own merits and disadvantages. The important properties of hydrogen and CNG are given in Table 1.

It can be seen from Table 1 that addition of hydrogen to CNG can significantly improve combustion characteristics of CNG by increasing its lean limits and flame burning velocity. Therefore in this study, performance, combustion and emission characteristics of a prototype SI engine, fuelled by various formulations of hydrogen enriched natural gas (HCNG), were experimentally investigated. This study is expected to clarify the behavior of engine fuelled with HCNG blends, in which H/C ratio was increased from 4 to ∞, or in other words, hydrogen fraction was increased from 0% (v/v) to 100 (v/v), through 10%, 20%, 30% and 40% (v/v).

Several researchers carried out investigations on hydrogen–CNG blends as IC engine fuel. Lim et al. [6] conducted experiments using 30HCNG in an engine with compression ratios of 11.5 and 10.5. They reported significant reduction in NOx emissions ($\sim$75%) and improvement in thermal efficiency ($\sim$6.5%) compared to baseline CNG. They also reported that anti-knocking tendency increased with fuel increasingly enriched in hydrogen. Significant effects of compression ratio, excess-air ratio and ignition timing were observed on NOx emissions. NOx and CO emissions increased with increased compression ratio. This increase in NOx emissions could be reduced by retarding the ignition timing or using lean combustion. Liu et al. [7] also reported similar trend and suggested that excess air ratio had a significant effect on the HC, CO, NOx, and CO_2 emissions for both, natural gas and hydrogen enriched natural gas. They reported that in lean burn operation, HC emissions decreased with increasing hydrogen fraction for a specified excess-air ratio. NOx emissions increased with increasing hydrogen fraction in the HCNG mixtures, and NOx attained its peak concentration at an excess-air ratio of 1.1. CO_2 emissions decreased with increasing hydrogen fraction in the HCNG mixture. It emerged that the addition of hydrogen in natural gas extended the lean burn limit. Thus, an engine fuelled with HCNG operating under lean mixture conditions produced fewer emissions of HC, CO, CO_2, and NOx. Ma et al. [8] carried out experiments in a turbocharged engine, introducing hydrogen fraction (0–50% v/v) in CNG at different ignition timings. They reported that increased hydrogen fraction led to decreased maximum brake torque (MBT), and increased indicated thermal efficiency (ITE). The NOx, HC and CO emissions decreased with advancing spark timing and increasing engine load. With hydrogen enrichment, NOx and CO emissions increased for same ignition timing but HC emissions decreased. If spark timing was retarded to MBT, NOx emissions exhibited no increase but thermal efficiency increased with increase in H_2 fraction in the HCNG mixture [9]. Akansu et al. [10] also studied emission characteristics of hydrogen and natural gas blends in an IC engine and reported that HC, CO and CO_2 concentrations decreased with increasing hydrogen fraction in the test fuel. However NOx emission increased as more hydrogen was added to the test fuel. Thipse et al. [11] proved the benefits of hydrogen enrichment of natural gas. Because of its excellent combustion characteristics, such as higher laminar flame speed and wider lean flammability limits, ultra-lean combustion could be achieved in HCNG. HCNG engine could comply with the NOx regulations upto EURO-6, while maintaining low levels of HC and Greenhouse gas (GHG) emissions, without use of any NOx after-treatment system. Park et al. [12,13] compared the engine experiment results of HCNG and CNG combustion. They reported that addition of 30% (v/v) hydrogen to CNG (30HCNG) was most appropriate to comply with NOx regulations of EURO-6 emission norms without affecting overall engine performance, HC and CO emissions. Moreover, the use of 30HCNG allowed sufficient range of the vehicle, compared to higher levels of hydrogen enrichment. Wang et al. [14] introduced nitrogen in the engine cylinder to reduce NOx emissions from CNG. They reported an inverse relationship between nitrogen dilution ratio and engine out emissions. Higher nitrogen dilution ratio exhibited lower NOx ($\sim$17–81%) emissions, but higher THC ($\sim$3–78%) and CO ($\sim$1–28%) emissions. Nitrogen dilution had a significant influence on combustion and exhaust emissions [8].

Tangoz et al. [15] performed experiments using 5HCNG, 10HCNG and 20HCNG at different CRs and CR = 12 was found to be optimum. The maximum in-cylinder pressure (P_{max}) increased by addition of hydrogen to CNG for all CR. As CR decreased, torque increased upon addition of hydrogen to CNG. Lee et al. [16] reported reduction (>25%) in fuel consumption rate, while using HCNG as fuel at idle, compared to CNG. THC and CO emissions decreased with HCNG at idle, because of the low carbon content and enhanced combustion characteristics of the test fuel. Lower HCNG quantity used at idling also continuously decreased NOx emissions with an increase in lambda. Huynh et al. [17] performed experiments on a modified engine by controlling valve overlap. They reported that with hydrogen, it has no significant effect on engine performance compared to fixed valve timing. However they

Table 1
Important fuel properties of hydrogen and natural gas.

Properties	H_2	CNG
Relative air–fuel ratio (Stoichiometric)	34.3	17.2
Density (kg/m^3) @ stp	0.085	0.748
Octane number	<130	120
Lower calorific value (MJ/kg)	120	50
Auto-ignition temperature (°C)	536	600 [1]
Laminar burning velocity (cm/s) @stp	265–325	37–45
Flame quenching distance (mm)	0.64	2.03
Flammability in air (%v/v)	4–75	5.3–15

Table 2
Technical specifications of the test engine.

Specifications	Before modifications	After modifications
Model/make	DM 10/Kirloskar	ERL1/IITK
Ignition type	Compression ignition	Spark ignition
Bore × stroke (mm)	102 × 116	102 × 116
Connecting rod length	232 mm	232 mm
No. of cylinders	1	1
Displacement	948 cc	948 cc
Compression ratio	17.5	11
Inlet valve opening time	4.5° bTDC	4.5° bTDC
Inlet valve closing time	35.5° aBDC	35.5° aBDC
Exhaust valve opening time	35.5° bBDC	35.5° bBDC
Exhaust valve closing time	4.5° aTDC	4.5° aTDC
Cooling system	Water cooled	Water cooled
Fuel injection type	Direct injection	Port injection
Fuel injection pressure	220 bar (in-cylinder)	3 bar (port)

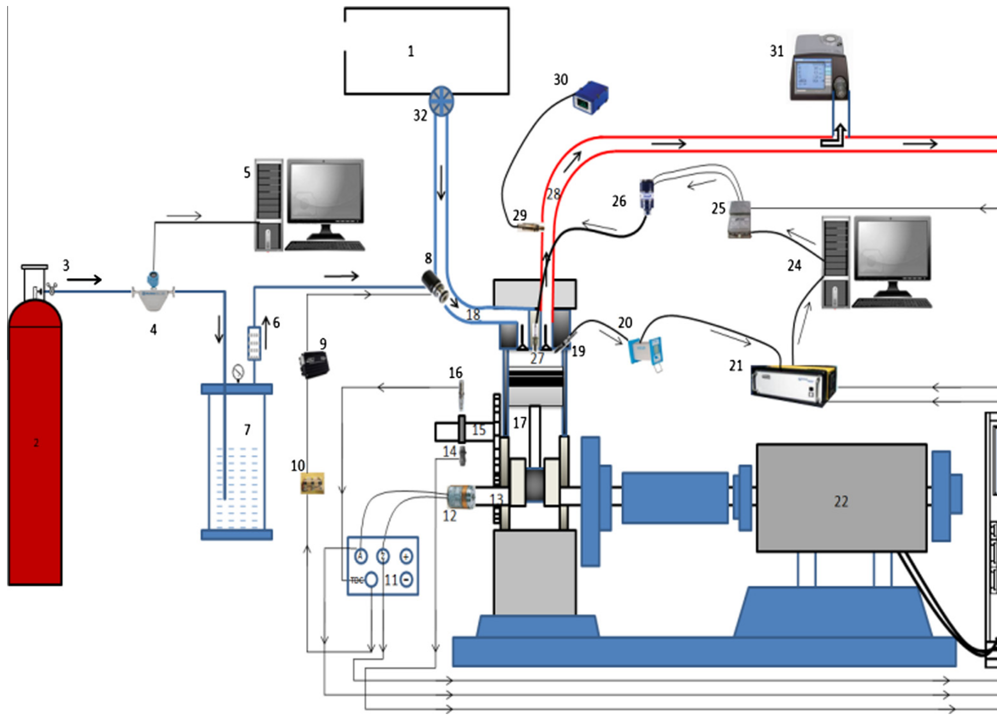

Fig. 1. Schematic of the experimental setup. 1. Surge tank, 2. gas cylinder, 3. pressure regulator, 4. Coriolis fuel mass flow meter, 5. computer for gas fuel metering, 6. flame arrestor, 7. flame trap, 8. gas injector, 9. injector driver module, 10. fuel injection control circuit, 11. BNC cable box for transmission of encoder and TDC signals, 12. rotary shaft encoder, 13. crank shaft, 14. magnetic pickup sensor, 15. Camshaft, 16. TDC sensor, 17. single cylinder engine, 18. intake manifold, 19. pressure transducer, 20. charge amplifier, 21. data acquisition system, 22. DC dynamometer, 23. dynamometer control panel, 24. computer for combustion data acquisition and spark timing control, 25. spark timing control unit, 26. ignition coil, 27. spark plug, 28. exhaust line, 29. lambda sensor, 30. lambda module, 31. exhaust gas emission analyzer and 32. air throttle.

Table 3
Experimental test matrix.

Fuel composition	CNG	10HCNG	20HCNG	30HCNG	40HCNG	H_2
H/C ratio of test fuel	4.00	4.22	4.50	4.85	5.33	∞
Engine performance tests	√	√	√	√	√	√
Combustion investigations	√	√	√	√	√	√
Emission investigations	√	√	√	√	√	√

reported that reducing valve overlap was an excellent technique to reduce engine backfire. In addition, they achieved MBT with an equivalence ratio of 1.1. Lee et al. [18] carried out experiments in a single cylinder engine using pure hydrogen at 1600 rpm and wide open throttle (WOT) conditions. Severe backfire was observed near relative air–fuel ratio (RAFR) of 1.0 and hotspots such as spark plug tip and carbon deposits in the combustion chamber were the main reasons for this backfire. Varde and Frame [19] performed experiments in a single cylinder SI engine to quantify advantages of port fuel injection of hydrogen over carbureted induction, in terms of backfire and cyclic variations. Experiments were performed at 1800 rpm and 2100 rpm. Improvements in lean-burn limits were observed for the port injection system. Thermal efficiency of port fuel injection system in lean region was also higher. Maximum NOx emissions were observed with mixtures slightly leaner than stoichiometric. Flame speed for the port fuel injection engine was higher than the carbureted engine. At 20° bTDC, backfire occurred between $\Phi = 0.9$ and 1.15 for carbureted engine, whereas for the port injected engine, backfire occurred between $\Phi = 1.1$ to nearly stoichiometric mixtures. Pal and Agarwal [20] faced backfire issues at higher engine loads. They carried out a comparative study of laser ignition (LI) and spark ignition (SI) of hydrogen and concluded that for a fixed RAFR, advancing the ignition timings led to improved combustion for both LI and SI mode. Das and Mathur [21] controlled NOx emissions from a hydrogen fuelled engine using different EGR rates. Experiments were performed in a carbureted hydrogen engine and significant reduction in NOx emission was obtained by using 15% EGR. Brake specific fuel consumption (BSFC) decreased with increasing EGR. Increasing the spark advance increased the in-cylinder temperature, which increased the NOx emission from the engine.

From this literature review, it is evident that hydrogen enrichment of natural gas is vital in improving overall performance of the gaseous fuelled engine. One can also possibly attain prominent reduction in HC and CO_2 emissions by this approach. This study is therefore aimed to explore the limits of hydrogen enrichment in the engine without substantial hardware modifications and without deteriorating overall performance.

2. Experimental setup

Engine used for this study was a modified SI engine prototype developed from a water-cooled, single cylinder, four stroke, direct injection, compression ignition diesel engine. This engine was suitably modified to operate in spark ignition mode, with the capability of using gaseous fuels in port fuel injection mode. Specifications of the test engine before and after the extensive hardware modifications are given in Table 2. Test engine was connected to a DC dynamometer. Test fuels were injected into the intake manifold using a customized, high volume flow rate solenoid fuel injector (AFS, Gs-60-05-5 H). Start of injection (SOI) timing was kept at 364.5° bTDC and injection pressure of 3 bar was maintained throughout the experiments. A custom build injector driver circuit was coupled to the injector control module (AFS) to control the peak and hold current for the solenoid injector.

Mechanically governed direct injection (DI) system was replaced with port fuel injection (PFI) system. A pressure regulator was used to maintain fuel injection pressure upto 3 bar. Fuel line was made of SS 304 and could sustain a pressure upto 250 bar. A Coriolis mass flow meter (Emerson, CMF010M) was installed to acquire real time fuel mass flow rate. Downstream of the mass flow meter, fuel line was connected to a customized flame trap, in order to avoid fire hazards. Additionally, a customized flame arrestor was also installed, which could quench the flame travelling backwards, if required. In this experimental study, in order to reduce the effect of cyclic fluctuations, combustion data of 250 consecutive engine cycles was acquired and their average data set was used for further analysis. In-cylinder pressure data was acquired using piezoelectric pressure transducer (Kistler, 6013), which was capable of measuring dynamic pressures upto 250 bar. Charge generated by the pressure transducer was converted into proportional voltage signal by using a charge amplifier (Kistler, 5015) and then this signal was acquired by the high speed combustion data acquisition system (Hi-Technique, meDAQ).

For acquiring emissions data, a raw exhaust gas emission analyzer (Horiba: MEXA 584L) was used, which measured NO, HC, CO, CO_2 and O_2 concentrations in the engine exhaust. Schematic of the experimental setup is shown in Fig. 1.

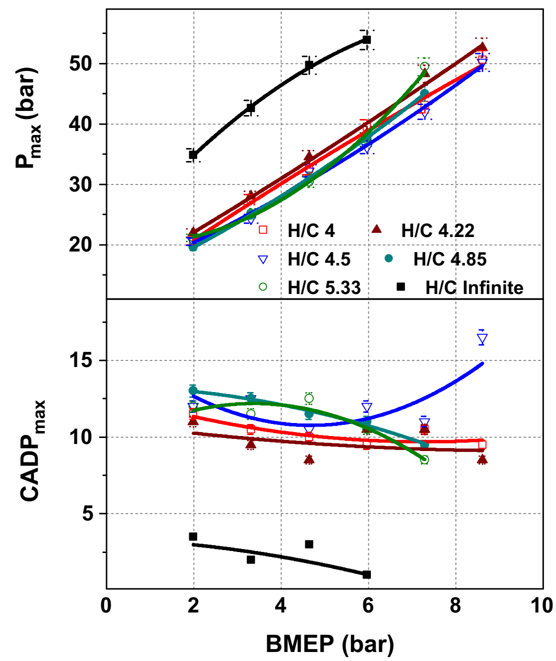

Fig. 3. Variation of peak in-cylinder pressure and its location.

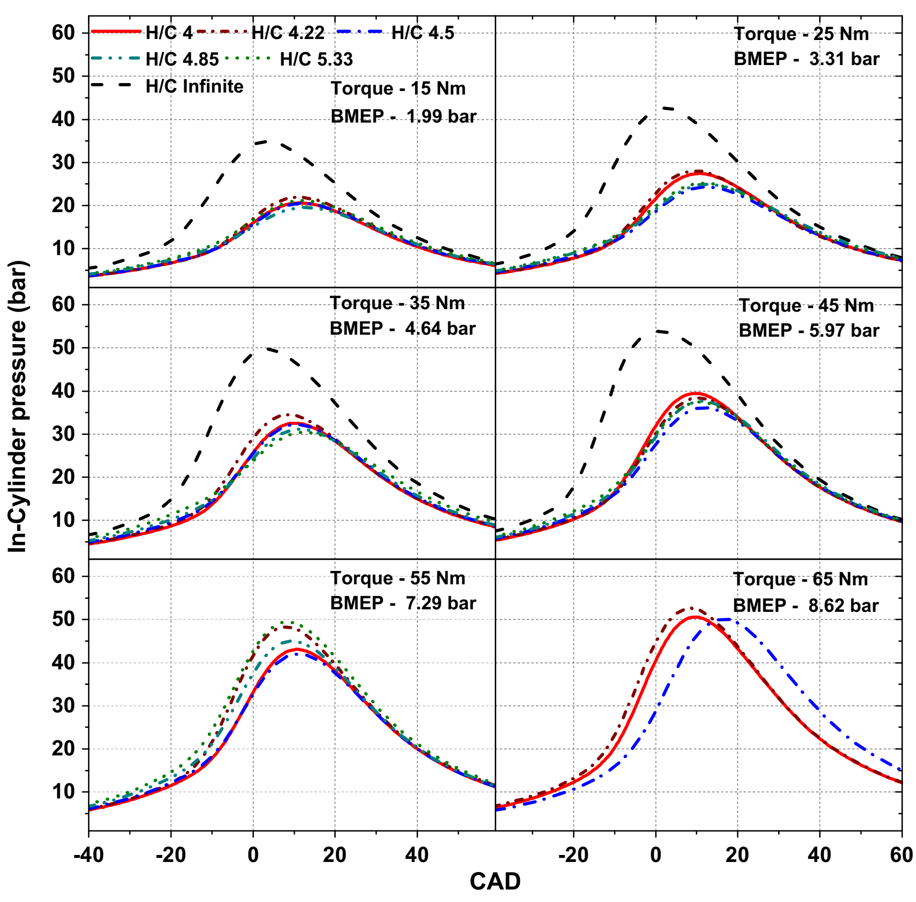

Fig. 2. Variation of in-cylinder pressure for HCNG test fuels.

3. Results and discussion

In this study, variation of H/C ratio from 4 to ∞ was investigated using hydrogen and natural gas mixtures. Operating window was decided by considering knock and misfire limits on higher and lower loads respectively. RAFR was maintained at 0.95 for all torque values by varying fuel and air flow rates. Experiments were performed at a fixed compression ratio of 11 and spark timing of 32° bTDC for all test fuels. BMEP was varied from 1.99 bar (minimum) to the MBT, which could be attained by each test fuel. For hydrogen, BMEP was in the range of 1.99–5.97 bar, due to backfire and misfire at the respective extremes. For test fuels with H/C ratios of 4, 4.22 and 4.5, BMEP range achieved was from 1.99 to 8.62 bar for each of them. 1.99 bar BMEP corresponds to 15 Nm torque at 1500 rpm and 8.62 bar BMEP corresponds to 65 Nm torque at 1500 rpm. For test fuel with H/C ratio of 4.85 and 5.33, BMEP range achieved was from 1.99 to 7.29 bar for both. In this paper, analyses and presentation of graphs is done based on H/C ratio of the hydrogen enriched compressed natural gas. Natural gas primarily consists of methane (CH_4) with traces of C2, C3 hydrocarbons. However H/C ratio was evaluated based on molecular formulae of methane in the present study. The experimental test matrix is shown in Table 3.

3.1. Combustion analysis

The engine combustion efficiency is indicator of degree of complete burning of fuel inside the engine combustion chamber [22]. Fig. 2 shows the variation of in-cylinder pressure for all test-fuels at different engine loads.

Maximum in-cylinder pressure (P_{max}) for all test fuels increased with increasing BMEP. In addition, a noticeable difference in peak pressure of hydrogen was observed compared to other test fuels. Reason for higher peak pressure for hydrogen is its relatively earlier start of combustion (SoC) compared to other fuels, due to significantly shorter ignition delay and higher flame speed (Table 1).

Crank angle position for peak pressure (CAP_{max}) was almost constant at all loads. Power output increased with increasing P_{max} at almost constant CAP_{max}. With increasing load, there was an increase in injected fuel quantity in each engine cycle, therefore P_{max} increased. However there was no advanced or retarded combustion observed because of fixed RAFR. MBT obtained for hydrogen was 45 Nm, whereas for test fuels with H/C ratio 4, 4.22 and 4.5, it was upto 65 Nm at fixed RAFR. Hydrogen enrichment of CNG was effective till H/C ratio of 4.5. For hydrogen enrichment of natural gas beyond H/C ratio of 4.5, a drop of 10 Nm in MBT was seen and the reason for this reduction was relatively lower volumetric energy density of the test fuels with H/C ratios of 4.83 and 5.33. Fig. 3 shows that hydrogen gives significantly higher peak in-cylinder pressure (P_{max}) for the same load, compared to other test fuels, although power output is similar due to early SoC in case of hydrogen. In addition, peak in-cylinder pressure increased with increasing BMEP, however its location was almost constant throughout the operating range. In other words, CAP_{max} showed no effect of increasing BMEP. Ma et al. showed that at a fixed RAFR, combustion duration shortened due to increased H_2 fraction in the HCNG mixture [9].

Fig. 4 shows the heat release rate (HRR) with respect to (w.r.t.) crank angle position for various test fuels. For hydrogen, HRR varied from 32J/CAD to 69J/CAD with increasing BMEP upto 5.97 bar, whereas for other fuels, HRR varied from 24J/CAD to 50J/CAD with

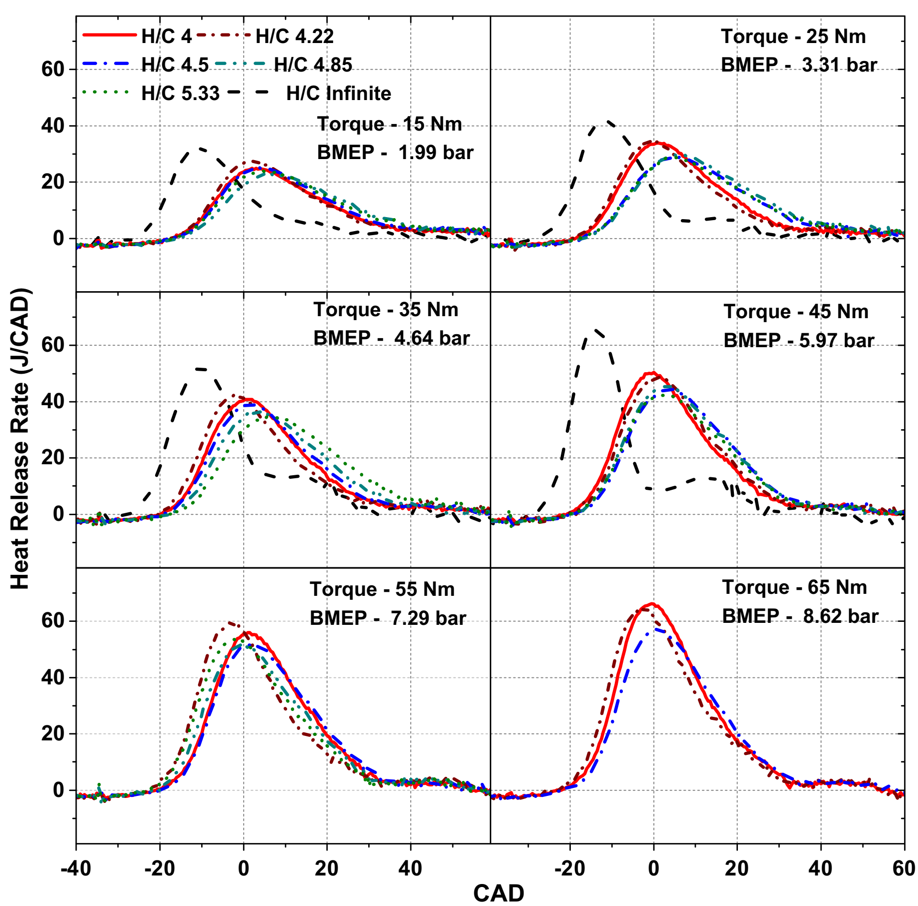

Fig. 4. Variation in heat release rate for HCNG test fuels.

increasing BMEP upto 8.62 bar. This figure shows relatively higher burn rate for hydrogen. Higher HRR of hydrogen leads to higher P_{max}, as seen in Fig. 3. At BMEP of 7.29 and 8.62 bar, HRR increased upto 68J/CAD. Crank angle position of the peak of HRR for hydrogen was at 12° bTDC for the entire load range, whereas for other fuels, it was near TDC. This also indicates higher flame speed of hydrogen. Slight hydrogen enrichment of natural gas (H/C: 4.22) helped in increasing P_{max}, especially at lower engine loads.

Fig. 5 shows the variation in cumulative heat release (CHR) w.r.t. crank angle position for different test fuels. It can be seen that maximum CHR varied from 130 to 250 J for hydrogen upto 5.97 bar BMEP at MBT. For test fuels with H/C ratio 4, 4.22 and 4.5, maximum CHR reached upto ~300 J. For test fuels with H/C ratio 4.85 and 5.33, maximum CHR reached upto ~280 J. This trend was observed because further hydrogen enrichment of natural gas beyond H/C ratio 4.5 causes reduction in volumetric energy density. In addition, relatively earlier SoC was observed in case of hydrogen because of its faster flame speeds. This increased knocking tendency at higher loads. CNG achieved highest CHR at maximum load compared to other fuels, because it had maximum volumetric energy density amongst all test fuels and the test engine was a naturally aspirated engine. Hence in terms of efficiency, H/C ratio of 4.0 was dominating.

Fig. 6 shows the rate of pressure rise (RoPR) for HCNG test fuels. Hydrogen showed highest and early rate of pressure rise due to its higher HRR and calorific value. 10% mass burn fraction (MBF) is considered as SoC and 90% MBF is considered as end of combustion (EoC). The crank angle duration between MBF_{90} and MBF_{10} is considered as 'combustion duration'. The crank angle position for 50% MBF (CA_{50}) is considered as combustion phasing. Fig. 7 shows the variations in SoC, EoC, CA_{50} and combustion duration w.r.t. BMEP. SoC for hydrogen was approximately 10° CA earlier than other test fuels. Hydrogen enrichment of natural gas led to higher ignition delay at lower engine loads for test fuels with H/C ratios 4.22, 4.5, 4.85 and 5.33. Whereas at higher engine loads, it led to lower ignition delay and relatively earlier SoC. Flame development and flame propagation periods i.e. combustion duration was relatively shorter at lower engine loads compared to other test fuels at constant spark timing of 32° bTDC. At higher engine loads, combustion duration of test fuels was almost similar. SoC for hydrogen varied from 15° to 19° bTDC for the entire load range, however for other test fuels, it varied from 1° aTDC to 9° bTDC. Combustion duration slightly decreased with increasing BMEP due to increase in the in-cylinder temperature.

Shorter flame development and flame propagation period indicates faster combustion of the air–fuel mixture as seen from combustion duration graph. Combustion phasing for hydrogen was significantly advanced compared to other test fuels and it advanced with increasing engine load for all test fuels. However at lower engine loads, combustion phasing retarded for test fuels with H/C ratio 4.5, 4.85 and 5.33, which possibly resulted in superior BTE.

3.2. Performance analysis

Fig. 8 shows the variations in brake thermal efficiency (BTE), brake specific energy consumption (BSEC) and exhaust gas temperature (EGT) w.r.t. BMEP.

BTE and BSEC were calculated from the measured parameters e.g. brake torque, fuel mass flow rate, engine speed, exhaust gas temperature, calorific values of the test fuels etc. EGT was measured in the exhaust manifold, 100 mm downstream of the exhaust

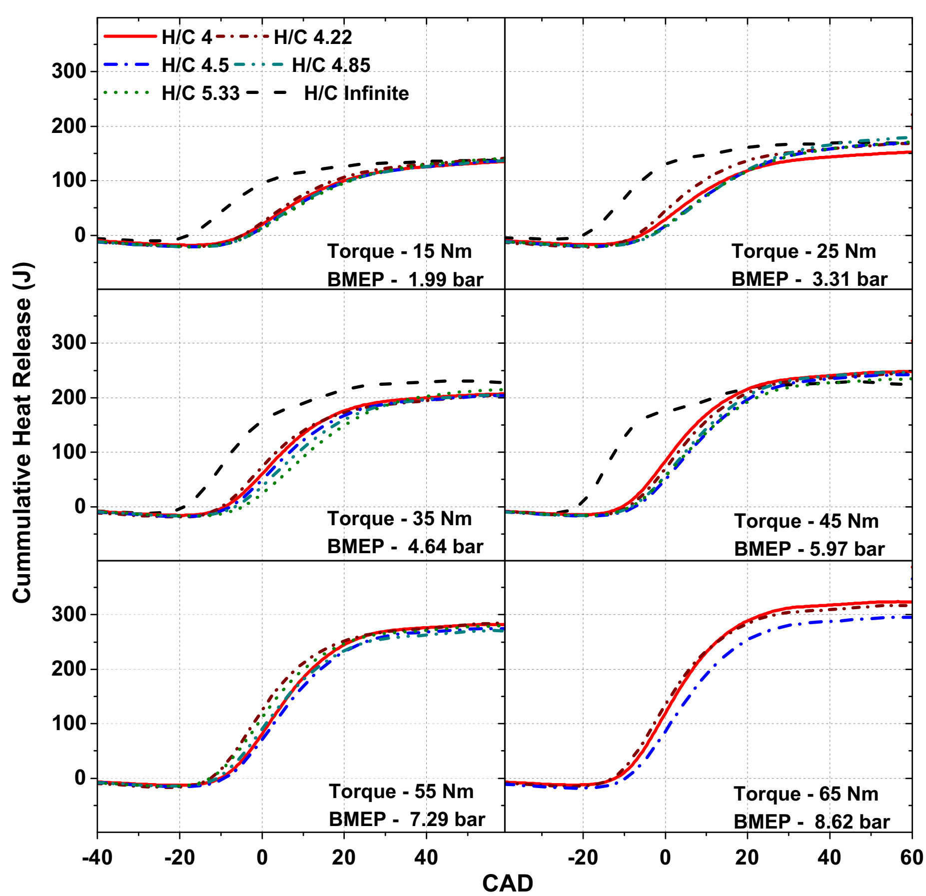

Fig. 5. Variation in cumulative heat release for HCNG test fuels.

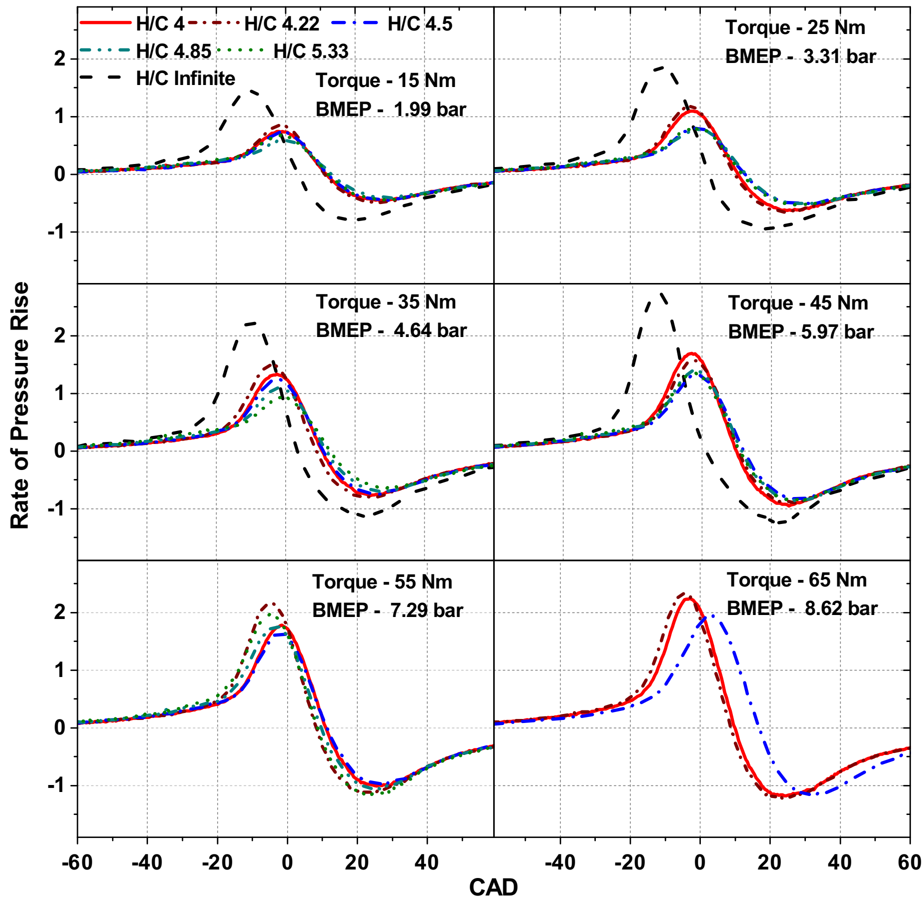

Fig. 6. Variation in rate of pressure rise for HCNG test fuels.

valve using a K-type thermocouple. The HCNG mixture fuelled engine delivered excellent BTE starting from 20% at low loads to 44% at high engine loads. This was because of the hydrogen enrichment of natural gas, which increased the flame speed and heat release closer to TDC in the expansion stroke. Test fuel with H/C ratio 4.5 was the most efficient fuel at higher loads whereas test fuel with H/C ratio 5.33 was the most efficient at lower loads. RAFR was maintained constant hence fast burning of air–fuel mixture led to superior combustion efficiency and BTE. At higher engine loads, increased H/C ratio of the test fuel upto some extent showed slightly reduced thermal efficiency because addition of hydrogen beyond a certain limit lowered the volumetric energy density of the test fuel, leading to reduction in thermal efficiency. However higher diffusivity and faster HRR of hydrogen helps natural gas in burning faster and expanding its lean limits, thus delivering superior BTE. It case of hydrogen at higher loads, BTE was relatively lower than other test fuels because of the lower volumetric energy density of the combustible mixture. However the hydrogen burns faster and this is the main reason, why higher P_{max} was observed in case of hydrogen, invariably before TDC, which results in lower BTE. Fig. 8 also reflects the trend of BSEC for all test fuels w.r.t. BMEP. With increasing BMEP, BSEC reduced i.e. the net energy consumption per unit power produced reduced. At lower engine load, BSEC is relatively higher for all test fuels, because for very small useful power produced, a large fraction of power is used in overcoming friction. The trend of BSEC is exactly opposite to that of BTE. EGT increased with increasing engine load. For hydrogen, EGT was lowest amongst all test fuels. This is because hydrogen releases its energy relatively earlier in the expansion stroke and there is enough time for hot burning gases to expand and cool down during the expansion stroke. However for H/C ratio 4 and higher, relatively higher EGT was seen throughout the operating range of the engine.

3.3. Emissions analysis

In this section, emission characteristics (mass emissions) of the hydrogen enriched natural gas fuelled, spark ignited single cylinder engine are discussed. This includes carbon monoxide (CO), unburned hydrocarbons (HC), oxides of nitrogen (NOx) and carbon-dioxide (CO_2) emissions. It can be seen that from hydrogen fuelled engine, CO, CO_2 and HC emissions were negligible. Traces of hydrocarbon emissions are seen in raw emissions of hydrogen fuelled engine, in-spite of hydrogen being a zero carbon fuel. This is possibly because of partial burning of lubricating oil film present in the engine combustion chamber, especially at the liner-piston ring interface. At high temperatures prevailing at high loads, this layer starts to partially burn and contribute to hydrocarbon emissions, however it is still be negligible from emissions point of view (see Fig. 9).

Theoretically speaking, product of hydrogen combustion should be water alone however some oxides of nitrogen are also present in its emission spectra because of localized high temperature zone present in the combustion chamber. Here the temperature reaches a point, where the atmospheric nitrogen starts reacting with oxygen to form NOx, especially at high load conditions. NOx emission for test fuels with H/C ratio of 4.0 and 4.22 were higher than other test fuels throughout the operating range. The reason for high NOx

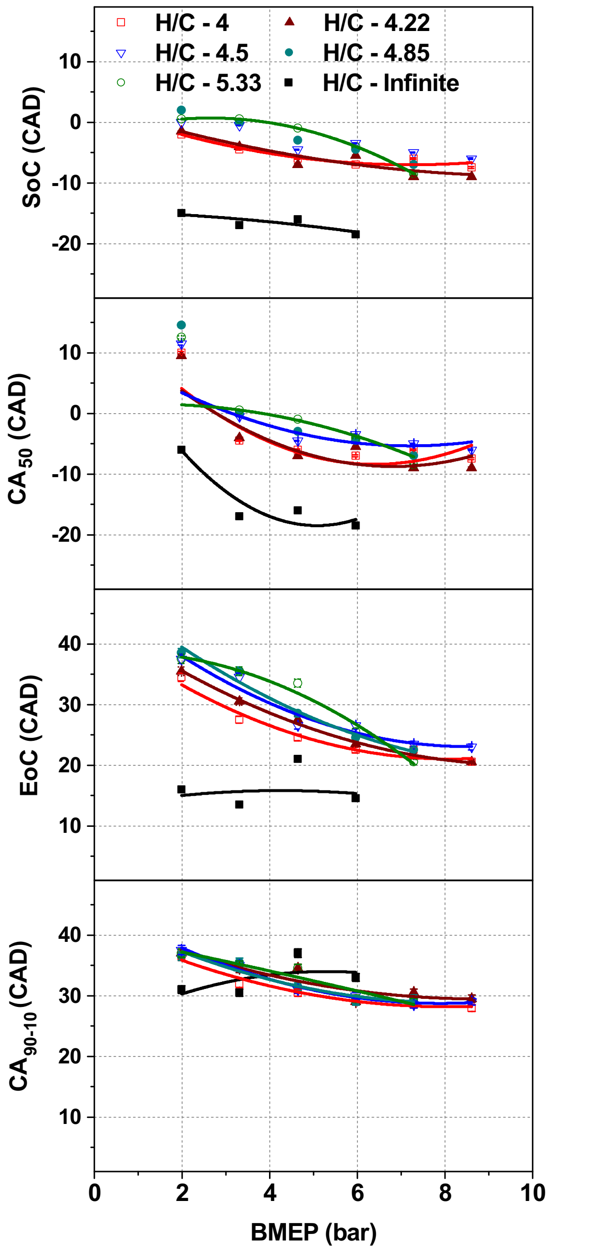

Fig. 7. Variation of start of combustion, combustion phasing, end of combustion and combustion duration for HCNG test fuels.

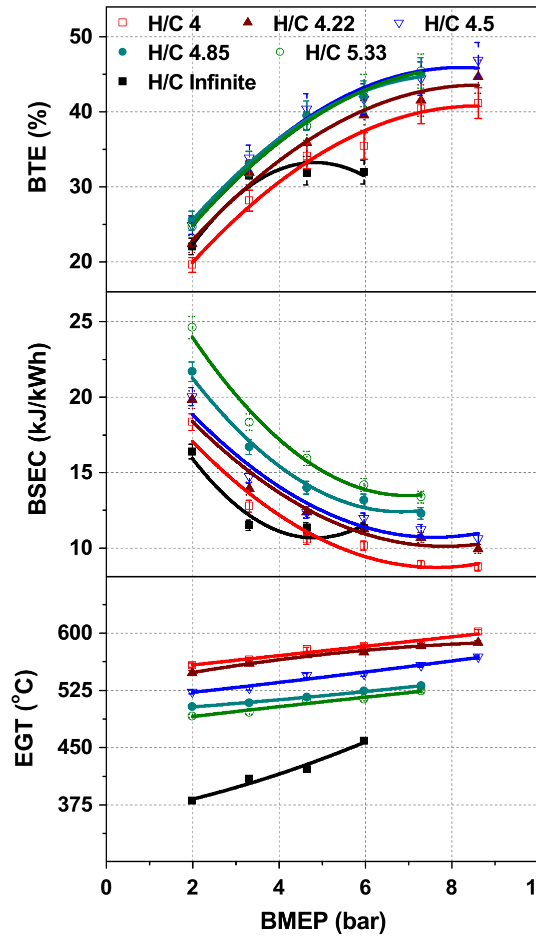

Fig. 8. Variations in brake thermal efficiency, brake specific energy consumption, and exhaust gas temperature for HCNG test fuels.

was higher EGT (Fig. 8) and higher volumetric energy density of the test fuel, delivering higher HRR and CHR (Figs. 4 and 5). It can also be seen that test fuels with H/C ratio 4.0, 4.22 and 4.5 showed lower emissions of HC and CO than test fuels with H/C ratio of 4.85 and 5.33. This is because increment in H/C ratio leads to expansion in lean limit and relatively lower quenching distance because of increased hydrogen content of the test fuel. This further helps to burning fraction of HC and CO present in the crevice volume because lubricating oil film thickness remains constant at a particular speed and increases with increasing engine speed [23]. At high loads, test fuels emit hardly any significant HC and CO emissions. Huang et al. [24] concluded that 2–4% of fuel remains unburned and its fraction is independent of injection duration. Hence CO is almost constant at full load condition and increases significantly w.r.t fuel properties. CO_2 emissions decreased with increasing BMEP. CO_2 emissions also decreased with increasing hydrogen fraction in the test fuel, with nearly zero emissions for hydrogen, similar to the results reported by Lim et al. [6] and Akansu et al. [10].

4. Conclusions

An experimental study on combustion, performance and emission characteristics of a prototype spark ignition engine was conducted with test fuels having different H/C ratios. Hydrogen was added to the natural gas and H/C ratio of the test fuels was varied from 4.0 to ∞. Engine was suitably instrumented and experiments were conducted. Lean-burn limit was directly proportional to the H/C ratio of the test fuels. It was found to be highest for hydrogen amongst all test fuels. For a fixed RAFR, thermal efficiency increased with increasing H/C ratio, except hydrogen, where it reduced because of lower volumetric energy density of the combustible charge. P_{max} increased with increasing H/C ratio of the test fuels and CAP_{max} advanced with increasing engine load. Heat release rate was highest for hydrogen and it also showed shortest ignition delay. Exhaust gas temperature increased with decreasing H/C ratio of the test fuel. NOx emissions were highest for test fuel with H/C ratio of 4.22 upto 55 Nm torque, however, NOx was highest for test fuel with H/C ratio 4.5 at full load. HC and CO emissions were negligible for the entire engine load range for all test fuels. In summary, test fuel with H/C ratio of 4.5 showed best overall performance compared to other HCNG mixtures used in this study and emerged as the best choice for hydrogen enrichment of natural gas.

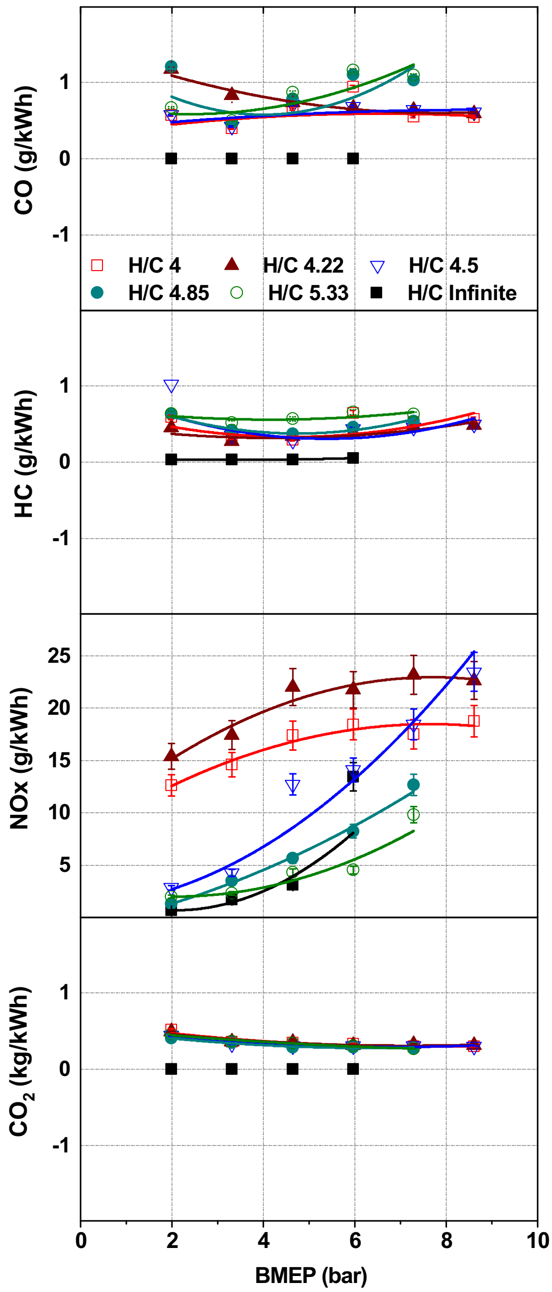

Fig. 9. Variations in CO, HC, NOx and CO_2 emissions.

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