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Hydrogen Enriched Compressed Natural Gas (HCNG) as an Alternative Fuel for Conventional IC Engines. A Review

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ABSTRACT

The continuous increasing demand of alternative energy and various stringent emission norms have led the engine researchers to develop strategic plans for the appropriate usage of available energy and reduce the carbon emission of fuels. Moving ahead with these plans, hydrogen has emerged as one of the promising energy carriers amongst the gaseous fuels but technical limitations restrict its uses. In this regard, hydrogen enriched compressed natural gas (HCNG), also called hythane, has been emerged as an attractive energy carrier for internal-combustion engines. Worldwide many countries are endeavoring hard to lower down the emission level by promoting HCNG-fueled gasoline vehicles. It has been observed that under precise circumstances, the indicated thermal efficiency of the HCNG engine is much better compared to CNG engines with identical emissions. In spite of that the H_2 enrichment to NG/CNG provides increased NO_x, it can be reduced with the application of lean burn combustion concept and also using three-way catalytic (TWC) converter. This review is an overview of HCNG engines and provides a detailed discussion on the fundamentals of characteristics of HCNG mixture, engine combustion and emissions. For covering the prospects of the HCNG engines, various approaches are discussed along with methods to increase the power output, thermal efficiency, and emissions reduction along with gaseous fuel induction strategy.

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1. Introduction

Natural gas is often thought of as the most promising alternative fuel for the transportation sector because it is the cleanest fuel; possessing high H/ C ratio, high research octane number (RON), good anti-knocking properties; is a more abundant fuel than petroleum, and produces significantly less regulated emissions {e.g., carbon monoxide (CO), carbon dioxide (CO_2), etc.} and non-methane hydrocarbon emissions investigated by Ma et al. [2010]. Amongst all other alternative fuels such as CNG, HCNG, LPG, LNG, bio-diesel, biogas, ethanol, methanol, di-methyl ether, and producer gas, the use of hydrogen as a future fuel for internal combustion (IC) engines is often thought of, but current infrastructure does not support its production, storage and dispensing. At the same time several other obstacles such as fuel induction technology and safety issues have to be overcome before commercialization of hydrogen as a fuel [Deluchi 1989]. Difficulties related with compressed natural (CNG) gas engines can be improved by adding small fractions of hydrogen and such blends of hydrogen and natural gas are termed as hythane or HCNG Patil et al. [2009], and Karim et al. [1996]. Patil et al. [2009] stated that HCNG may be considered as an automotive fuel without requiring any major modification to existing CNG engines and associated infrastructure. Table 1 illustrated by Karim et al. [2003] shows comparative important fuel properties of SI engine fuels like H_2, methane and Gasoline. CNG has many advantages compared to gasoline. It has high octane number of 130 (gasoline ~87) which results in smooth engine operation with high compression ratios without knocking. CNG operated engines emit lower emissions due to lean burning ability. Conventional engines, whether gasoline or diesel are easily convertible with minor modification to operate with CNG. Performances of such modified engines include higher thermal efficiencies and power. In addition to these advantages, there are however, disadvantages such as slow burning velocity, large cycle to cycle variation, poor lean burning ability and higher fuel consumption. Hu et al. [2009] experimented and found that to improve the lean burning capability and flame velocity, more turbulence inside the combustion chamber is required, but this measure results in higher heat losses to the cylinder walls via convection. Superior modern ignition system such as high frequency plasma, microwave and laser ignition systems may also enhance the lean burn limit but with additional cost [Idicheria, 2015]. Another method to improve this is to add hydrogen along with natural gas to achieve higher laminar burning velocity (2.9 m/s for hydrogen and 0.38 m/s for CNG). Thus, hydrogen enriched CNG can smoothen cyclic variation caused by relatively poor lean-burn capability of natural gas engine

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Table 1: Important fuel properties of SI engine fuels (Patil et al. [2009]).

Properties	Hydrogen	Methane	Gasoline
Flammability limits (v/v)	4-75	5.3-15.0	1.2-6.0
Stoichiometric composition in air (% v/v)	29.53	9.48	1.65
Stoichiometric fuel/ air ratio (w/ w)	0.029	0.058	0.0664
Minimum ignition energy (mJ)	0.02	0.28	0.25
Laminar flame speed at NTP (m/s)	1.90	0.38	0.37-0.43
Flame temperature (K)	2318	2190	2470
Auto-ignition temperature (K)	858	813	500-750
Density (g/ m^3)	82	717	5110
Heat of combustion (LHV) (MJ/kg)	119.7	46.72	44.79
Heat of combustion (HHV) (MJ/kg)	141.7	52.68	48.29
Specific heat ratio of NTP gas	1.383	1.308	1.05
Diffusion coefficient in air at NTP (cm^2/s)	0.61	0.189	0.05
Normalised flame emissivity	1.00	1.7	1.7

Table 2: Property table for different HCNG mixtures (Hythane: History: http://www.hythane.com/)

Properties	CNG	10HCNG	20HCNG	30HCNG	40HCNG
H2 [vol %]	0	10	20	30	40
H2 [mass %]	0	1.21	2.69	4.52	6.16
H2 [energy %]	0	3.09	6.68	10.94	16.04
LHV [MJkg-1]	46.28	47.17	48.26	49.61	**55.38**

according to Wong et al. [2000]. HCNG blends can reduce exhaust emissions (CO, CO_2, CH_4 and NOx), improve fuel economy and thermal efficiency. HCNG operated engines can be thought of as introduction of hydrogen into the existing CNG infrastructure and after successful implementation, HCNG engines can be directly applied to a hydrogen engine. The addition of hydrogen to natural gas also greatly reduces problems associated with evaporative emissions, and carbon monoxide and carbon dioxide emissions. Xu et al. [2010] shows the properties of different HCNG mixtures in Table 2.

2.1 Conventional HCNG engines

The H/ C ratios in HCNG blends can range from 4 to ∞ by volume. In 1989, HCI (Hydrogen Components Inc., U.S.A.) began blending various ratios of hydrogen and natural gas and tested them at Colorado State University, U.S.A. (Hythane Company, LLC, 2007). Hythane having a typical percentage of 15% blend of hydrogen in CNG by energy content was patented in 1992 by Lynch et al. [1992]. Nagalingam et al. [1983] performed experiments in an AVL engine fueled by CNG, hydrogen and HCNG different blends 20HCNG and 50HCNG. Since then, many researchers have studied the effect of hydrogen addition with CNG on engine performances and emissions. Tighter emission norms such as EURO VI can be achieved with advanced combustion techniques using lean burn combustion. In this regard, HCNG replaces CNG due to wider flammable limits as well as high flame speed of hydrogen. These two properties are the key to lean burn combustion strategies. Verhelst et al. [2013] showed the flammable limits of diesel, gasoline, methane and hydrogen in Figure 1(a).

Three main challenges for lean combustion concept are (i) higher energy is required to ignite lean combustible mixtures, (ii) once the flame is initiated, its propagation velocity is lower and (iii) it is more easily extinguished than a less lean flame as shown in Figures 1(e), as reported by Borman et al. [1998], and Warnatz et al. [2001]. After successful flame initiation, appropriate measures enhance the burning velocity so that the combustion can be completed in the shortest possible time. Keeping the combustion interval short avoids a decrease in efficiency brought on by smaller effective expansion ratio of the combustion products that occurs with longer combustion intervals.

Table 3: Typical composition of CNG supplied by Central U. P. Gas Limited; Subsidiary of Indraprastha Gas Limited (IGL) New Delhi reported by Prasad et al. [2020]

Constituents	Percentage (%)
Methane (CH_4)	85-90%
Ethane (C_2H_6)	5-7%
Propane (C_3H_8)	2%
Carbon dioxide (CO_2)	3-5%
Others	1%

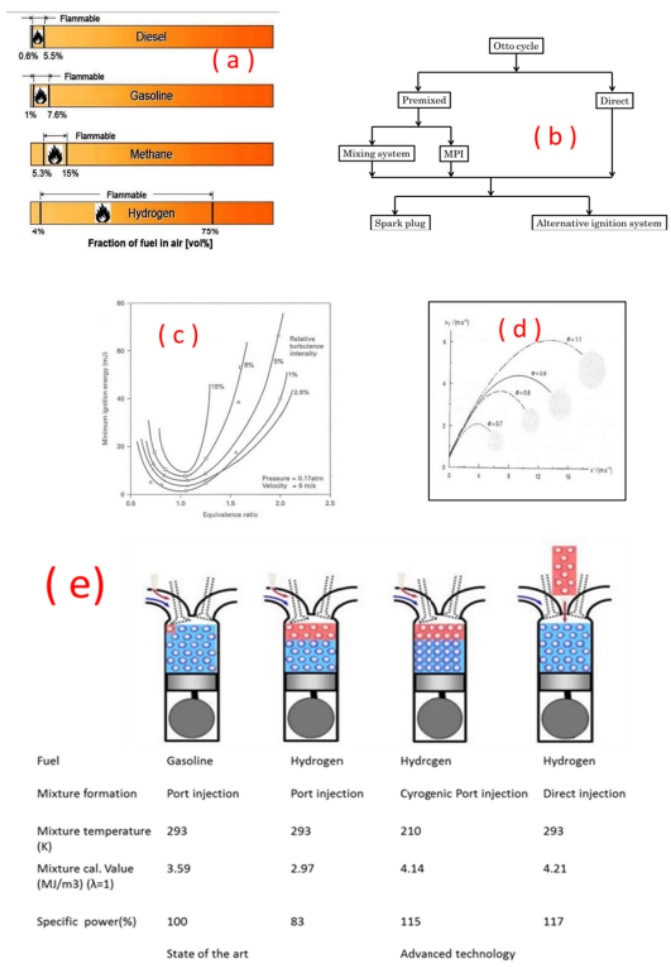


Figure1. (a) Flammability of different fuels used in engines, Borman et al. [1998]; (b) Basic structure of HCNG engine technology, Wakayama et al. [2006],; (c) Effect of equivalence ratio on MIE of propane air-mixtures, (d) Effect of equivalence ratio on turbulent flame velocity of propane air-mixtures reported by Welch et al. [2008] and Warnatz et al. [2001] ; (e) Comparison of specific power density of hydrogen mixture formation concept, Borman et al. [1998].

Verhelst et al. [2013] investigated that with multiple flames, the flame surface area is higher so the mass burning rate of air-fuel mixture is high, even though individual flames burns more slowly and each individual flame does not need to travel too far before it encounters another flame . The flame extinction due to presence of high turbulence can be avoided since one does not need to increase turbulence to the extent that is necessary with a single point flame. Warnatz et al. [2001] reported that absence of turbulence also reduces the in-cylinder heat transfer and any pumping work that would have been needed to increase the fluid motion necessary to support turbulence generation

There are several lean combustion concepts, but two among these are popular within the global research community. With spark-ignition, these are lean homogeneous and lean stratified charge combustion strategies. Nevertheless, each of these lean burn modes poses unique challenges for ignition system according to Welch et al. [2008]. The lean homogeneous strategy is applicable for port fuel injection system while lean stratified combustion is possible for direct injection system in which fuel in injected into the combustion chamber. The ignition system should enable ignition of lean homogeneous mixtures, provide adequate dilution tolerance (air or EGR), deliver adequate energy, deliver sufficiently long spark-discharge duration and withstand high in-cylinder charge motion. For second type, i.e., lean stratified, the same set of challenges are applicable as lean homogeneous, but also provide additional benefit of enhancing the reactivity and controlling the combustion phasing reported by Welch et al. [2008].

2.2 HCNG induction methodologies

Masjuki et al. [2009] experimentally found that in conventional SI engines, a homogeneous mixture of gasoline and air is inducted into the engine, compressed and ignited by a spark plug just before expansion. Figure 1(b) illustrates the basic structure of HCNG engine technology. In Otto cycle, mixture preparation of HCNG-air starts either outside the combustion chamber or HCNG is directly introduced into the combustion chamber where it mixes with air depending upon type of fuel injection strategy used for combustion. Mixture formation outside the combustion chamber can be performed using a separate mixing system or port fuel injection. In port fuel injection method, fuel is injected upstream of the intake valve with help of a solenoid gas injector where it mixes with air during suction and compression strokes.

2.2.1 Port Fuel Injection

In this method, HCNG is injected upstream of the intake valve using a solenoid gas injector where mixing occurs just before entering the engine cylinder during the suction stroke. The major drawback of this system is that fuel displaces air during the suction stroke resulting in lower volumetric and lower part load efficiencies.

2.2.2 Direct injection system

Verhelst et al. [2013] demonstrated about promising technique to increase engine performance is direct fuel injection (DI) into the combustion chamber during the compression stroke. DI engines can provide the same power output as gasoline or even higher. But, these are a few challenges to the development and use of DI hydrogen injectors:

- Reduced internal damping effect due to low density of hydrogen,
- Reduced hydrodynamic lift effect due to low viscosity of hydrogen,
- Hydrogen potentially degrades epoxy resins used to coat the piezo stack.

2.3 Mixture formation strategies

Both stringent emission norms as well as high engine efficiencies can be obtained by proper design of mixture formation of different fuel blends. Better mixture formation can be obtained in case of port fuel injection compared to direct fuel injection. Wakayama et al. [2006], and Rottengruber et al. [2004] proposed combined concepts for fuel induction technique means

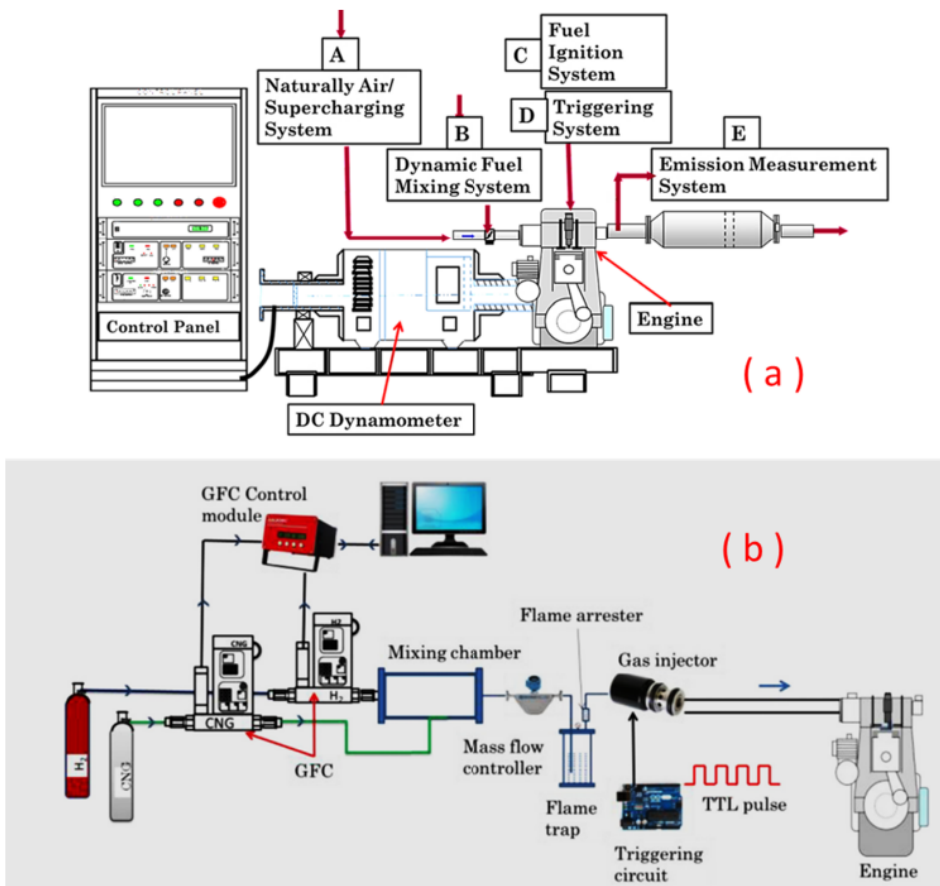


Figure 2: (a) Schematic of the engine experimental setup developed by Prasad et al. [2017, 2021a]; (b) Schematic of the dynamic fuel mixing and induction sub-system, [Srivastava, 2013]

some fuel fraction injected partly in port and remaining fraction is directly injected into the combustion chamber. Modern port fuel injectors are electronically controlled injectors and less costly compared to DI fuel injectors because DI injectors are designed to withstand more pressure and temperature according to Yi et al. [2000].

Verhelst et al. [2013] compared the theoretical maximum power output from a port fuel injection gasoline and hydrogen engine per unit engine displacement in Figure 1(e). Hydrogen in gaseous form occupies more volume compared to gasoline and results in lower volumetric efficiency. Amongst various induction techniques, maximum time available for mixing fuel –air mixtures is more in port fuel injection technique and is relatively simple. Proper mixture formation is essential for attaining fuel efficiency, reducing emissions and lubricating oil dilution.

2.4 Naturally aspirated HCNG engine

The effect of hydrogen addition to natural gas for engine application has been studied by many researchers during the last two decades' years. In this sub-section, an overview of literature related to the effect of hydrogen fraction on flame velocity, engine performance, emissions and particle size emitted from engines are provided. Some literature related to effect of compression ratio, spark timing, engine speed, and air-fuel ratio is briefly explained. Agarwal et al. [2017], and Prasad et al. [2017, 2021a] investigated the effect of hydrogen enrichment in HCNG analytically on cyclic variations in homogeneously charged spark ignition engines (refer Fig 2(a) & (b)). Prasad et al. [2021a] observed that increased hydrogen ratio was able to reduce cyclic variations while extending the operating region of the engine. Table 3 shows typical composition of CNG used in experiments

2.5 Dynamic HCNG Mixing System

HCNG is a mixture of hydrogen and natural gas on volume basis. The objective of such mixing is to overcome limitations of CNG during engine operation. A major challenge with investigating HCNG as a fuel is the variations in the fraction of hydrogen in the HCNG mixture in a controlled manner during engine operation. For this, a dedicated dynamic mixing system is needed to vary the hydrogen fraction and optimize engine behavior under desired conditions. Prasad et al. [2020] developed a dynamic mixing system as shown in Figure 2(b).

2. Ignition System

A wide range of alternate ignition technologies are available, the chief among these are high-energy inductive systems, multiple discharge (multi-strike) and dual-coil systems, high-frequency plasma, torch jet, plasma jet, turbulent jet, low-temperature plasma, microwave and laser ignition systems [Idicheria, 2015]. Most of these are yet to be implemented in high-volume engines. However, with aggressive downsizing, boosting and dilution that is necessary for near-term engines under stoichiometric operation, the conventional ignition system is being severely challenged [Srivastava, 2013]. Furthermore, to achieve lean combustion irrespective of the combustion strategy, enabling proper ignition and providing adequate controllability is of importance. Alternate ignition systems should address and meet these key challenges to be of value and relevance to the next generation of internal combustion engines. In addition to being a key enabler for lean-burn combustion, alternate ignition systems must also be cost-effective, reliable, durable and serviceable. Figure 3(a) shows the roadmap for new combustion concepts, which mainly focus on charge dilution either by EGR or by the excess air [Wenig, 2013] to improve engine efficiency. Increasing EGR creates severe conditions for both ignition systems and charge combustion, while introduction of excess air increases cycle–to-cycle variations. These approaches create disturbances in the in-cylinder flow fields, mixture homogeneity and flame kernel formation as reported by [Ball, 1999], and [Kettner, 2004], and lead to engine misfire or delayed combustion. In order to stabilize the engine operation for diluted mixtures, a wide number of new ignition concepts and new systems have been proposed over a period of time (Figure 3a).

In some of these new combustion concepts, ignition energy needed to develop flame kernel can be deposited either at a single spot or within a small volume. Types of ignition systems can be classified based on activation of flame kernel volume, as shown in Figure 3(b).

Using Single spot ignition, ignition energy can be deposited at a fixed location inside the combustion chamber, and this is not actively controlled by the ignition system. Nevertheless, a spark volume is defined by the spark channel and even enlarged by charge motion, e.g. by deflecting the ignition spark, which essentially leads to an activation of a flame volume. This is one

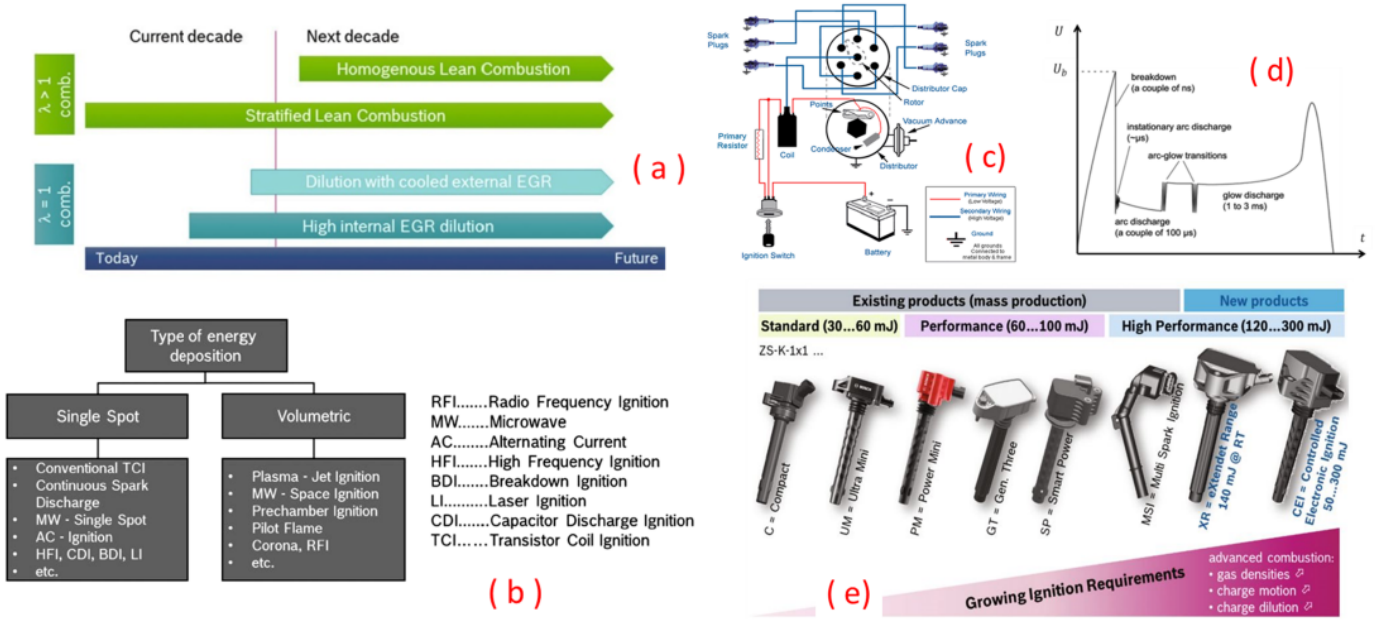


Figure 3: (a) Various new combustion concepts, Brandt et al. [2016]; (b) Classification of emerging ignition systems, Brandt et al. [2016]; (c) Typical conventional spark ignition system (http://www.carparts.com/classroom/ignition.htm); (d) Typical voltage-time characteristics of an electrical spark ignition discharge, [Bauer, 1998]; (e) Spark ignition system portfolio including latest high performance products such as MSI, XR and CEI, Brandt et al. [2016].

of the key factors, which yields the potential to ignite diluted mixtures using spark ignition systems. "Volumetric" ignition concepts, such as corona or plasma-jet ignition actively provide a flame kernel volume even in low charge motion conditions at controlled ignition timings. In IC engines, chemical energy of fuel is converted into useful mechanical work. In SI engines, combustion of fuel-air mixtures is initiated by plasma generated by a spark plug towards the end of the compression stroke (figure 3(c)). A typical conventional spark ignition system consists of a battery, switch, ignition coil, distributor, triggering device (contact breaker) and a spark plug. The ignition system generates adequately high voltage to produce a spark between the electrodes of the spark plug. For ensuring good performance and efficiency of the engine as well as lower emissions, spark is triggered at precisely the right instant in every engine cycle, towards the end of the compression stroke.

3.1 Voltage-time Characteristics of a Spark Plug

An electrical spark ignition discharge comprises of several phases namely: breakdown phase, arc discharge phase, and glow discharge phase (Figure 3(d)). During breakdown, conductive plasma is formed across the spark electrodes via electron multiplication under high voltage (several kV). After that, voltage applied between the electrodes drops from several kV to very low value in the range of ~100 V. During this period, the capacitive and inductive components of the spark plug are discharged. Breakdown is followed by in- stationary arc discharge for ~1 μs and glow discharge, which lasts for ~ few ms.

During arc discharge phase, the capacities of the high-voltage cable and coil discharge their energy and in glow discharge phase, discharge of energy stored in the ignition coil occurs. This phase always takes the form of a glow discharge. At high pressures and currents above 100 mA, however, an arc discharge can be sustained up to ~ hundreds of μs. In an intermediate current range (50-100 mA), rapid transitions occur between arc and glow regimes. Once the conditions for a self-sustained discharge are no longer fulfilled, spark breaks away and the voltage is reduced to zero [Bauer, 1998].

Despite variety of ignition systems, TCI (Transistor Coil Ignition) still remains dominating technology in the marketplace due to its simplicity, low cost and robustness. However the ignition of lean charge using TCI Systems is difficult, which puts their continued use for future lean combustion concept engines in question. To address these challenges, Bosch has proposed new high performance spark ignition systems (Figure 3(e)). The ignition energy of TCI spark ignition system can be simply increased by increasing spark current and/or by increasing the spark duration but it has adverse impact on spark plug durability due to increased wear, low ignitability in low spark current phases, limitations on size and weight of the ignition device and required electrical circuits. by Schneider et al. [2014] assessed that spark plug efficiency lies between 20–30 % under real engine conditions. To overcome these conflicts, conventional TCI system was redesigned for ~140 mJ spark energy and named as XR coil (Figure 3(e)). Conventional TCI systems have limitations in creating longer spark duration to address temporal fluctuations in the mixture formation. Therefore controlled electronic ignition (CEI) is preferred for stratified engine operations. Presence of high gas densities at full load leads to high ignition voltage demand of up to 40 kV. Such high voltage requirement decreases with reduction in electrode gap. However this results in higher quenching losses, especially with higher dilution of mixture.

3. Safety issues with HCNG

Hydrogen was considered extremely dangerous gaseous fuel after Hindenburg tragedy due to low ignition energy and wide flammability and so serious attention needed for safe operation of HCNG. Since HCNG contains some volume fraction of H_2 but it can be thought as dangerous as hydrogen. These are few measures to tackle safety issues with HCNG [Deluchi, 1989]:

- Since hydrogen is a colorless, odorless gas, it is very difficult to detect its leakage from pipe joints but CNG easily detect by its peculiar smell. In order to detect H_2 leakage, some odorant must be added in H_2 line. Also safety sensors for detecting presence of even minute concentrations of hydrogen in ambient should be used for leak detection.
- Hydrogen flames are invisible, therefore they are impossible to detect and extinguish. Flame colorant can be added to hydrogen in order to detect the flames.
- Burning velocity of H_2 is very high compared to methane. Therefore, in an enclosed space, hydrogen-air mixture will detonate more quickly compared to methane-air mixture. Therefore in order to mitigate detonation, some nontoxic chemical inhibitors can be used. However there are some intrinsic properties of hydrogen also, which make it relatively safer compared to other gaseous fuels [Deluchi, 1989].
- Natkin et al. [2002] reported that Hydrogen burns rapidly and radiates very low heat. Therefore hydrogen flames extinguish very quickly. Also, hydrogen flames do not damage the surroundings severely.
- There are several hot spot present inside the engine cylinder such as sharp corners, valve edges, spark plug electrodes; any soot particle present in crevices results backfires. Several safety devices must be installed to avoid severe accidents.
- In case of leakage in liquid hydrogen fueled cylinders, it disperses and evaporates quickly compared to gasoline, posing lower fire hazard (DOE, Energy Efficiency and Renewable Energy, Government of India).

4. Factors influencing HCNG engines Combustion, Performance and Emissions Characteristics

Since the very beginning of 20th century, the engine researchers used several approaches to improve the combustion and performance and reduce the emissions of the internal-combustion engines. This section provides the intensive discussions on the Combustion, Performance and Emissions Characteristics of HCNG engines.

4.1 Factors influencing HCNG engines Combustion Characteristics

Wang et al. [2010] investigated the effect of flame velocity experimentally and numerically by increasing the hydrogen fraction in HCNG mixtures in a constant volume combustion chamber using a conventional spark system. Images were captured for both premixed combustion and direct-injection combustion at overall equivalence ratios of 0.6 and 0.8 in CVCC by Schlieren photographic technique. They observed that flame velocity whether laminar or turbulent increased with increase in hydrogen percentage in CNG as shown in Figure 4(a, b).

Karim et al. [2003] performed a theoretical study of the effect of addition of hydrogen on methane combustion characteristics at different spark timings. They showed that ignition delay as well as combustion duration decreased with increasing hydrogen fraction in HCNG for same equivalence ratio. They also reported that addition of hydrogen could increase the flame propagation speed thus stabilizing the combustion process, especially during the lean combustion process.

Thermodynamically the net-work produced during engine combustion increases as the average value of gamma (c_p/ c_v) increases whereas gamma is a function of gas temperature and composition. Gamma decreases as the gas temperature increases, and decreases as the composition changes from air-fuel mixture to combustion products. So to maximize the net-work the in-cylinder temperatures are kept low and the product mixture dilute. The other two benefits are as follow: (a) lower in-cylinder temperatures result in lower heat transfer and (b) the higher average gamma results in lower energy being thrown away in the exhaust, because more work which is extracted during the expansion stroke results higher efficiency.

Prasad et al. [2021b] performed experiment on modified single cylinder engine having 1 litre capacity and port fuel injection. Figure 4(c, d, e) shows the variation of in-cylinder pressure, rate of pressure rise, and heat release rate for all test fuels at different engine loads. It was noticed that in-cylinder peak pressure for all fuels increasing with increase in BMEP. In addition to this, there was a noticeable increase in peak pressure of hydrogen. Reason for this is due to early start of combustion which was caused by lesser ignition delay and higher flame speed reported by Ilbas et al. [2006], and Mandilas et al. [2007]. Maximum brake torque (MBT) for hydrogen was observed to 35 N-m in optimized condition and further increasing the throttle resulted knocking. This clearly states that with retarded spark ignition timing hydrogen mixture will produce more torque under same test conditions. MBT for CNG, 10HCNG and 20HCNG was observed till 65 N-m at fixed RAFR. However, adding more fractions of hydrogen into CNG after

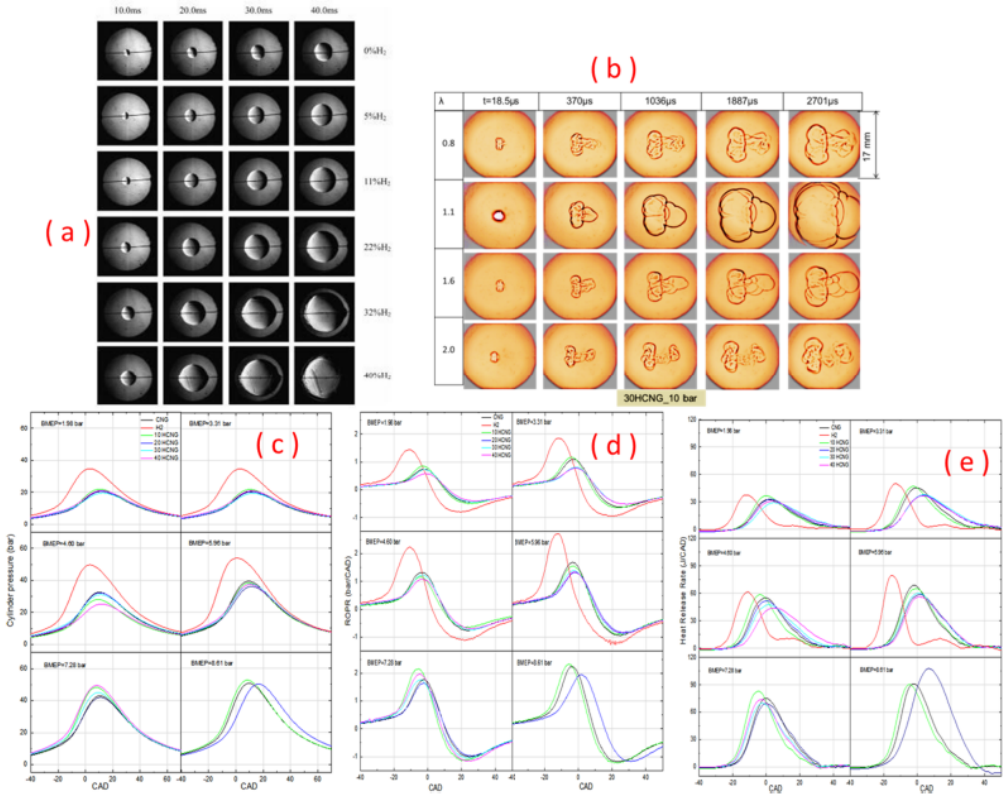


Figure 4: (a)Images of the effect of hydrogen addition to natural gas–air mixtures on flame propagating process during laminar premixed combustion (ϕ = 0.6), Wang et al. [2010] , (b) Shadowgraphs of flame kernel development for 30HCNG (0.8 < λ < 2.0), Prasad et al. [2021b]; (c) Variation of in-cylinder pressure; (d) represents the heat release rate (HRR) w.r.t CAD position; (e) Variation of HRR, Prasad et al. [2021b].

20HCNG, a drop of 10 N-m in MBT was observed. Reason being, decrement in energy density of CNG. As shown in figure 4(d), ROPR increases with increasing bmep due to higher mass of fuel and air inducted into the engine combustion chamber. Max. ROPR of 2.76 bar/CAD was observed for hydrogen at 5.96 bar bmep showing knock characteristics but HCNG blends showed no sign of Knocking because of high ocatane number. Author reported that 20 HCNG blend shows lower ROPR, better combustion characteristics in terms of knocking at constant RAFR with varying BMEP. Prasad et al. [2021b]also noticed that HRR value is varying from 37J/CAD to 79J/CAD (11-14 bTDC) in case of H2 while other blends varying from 29J/CAD to 69J/CAD with increase in BMEP value because of the higher burning rate and diffusivity of hydrogen (refer Figure 4(e)).

4.2 Factors influencing HCNG engines Performance and Emissions Characteristics

Figure 5(a) shows the variation of brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and exhaust gas temperature (EGT) w.r.t BMEP value as reported by Prasad et al. [2021b]. Author reported that 20HCNG is efficient fuel at higher engine loads whereas pure CNG was most efficient at lower engine load. This is because RAFR was maintained constant hence fast burning of air-fuel mixture led to superior combustion efficiency and BTE. However, enriching H_2 at higher engine loads up to some extent resulted superior BTE. But further enrichment led to lower BTE due to reduced energy density of fuel blend. On the other hand, high diffusivity and faster burning rate of H_2 extend the lean burning limits. Figure 5(a) also reflects the trend of BSFC for all test fuels w.r.t BMEP as reported by Prasad et al. [2021b]. They also noticed that as BMEP increase, BSFC goes superior H_2 showed superior BSFC compared to other test fuels. This is because of the higher calorific value of hydrogen than other test fuels. Also it was noticed that at MBT value for H_2 was little bit poor because of knocking tendency of fuel and early SoC. In addition EGT value was increased with increasing engine load. For H_2 EGT was f lower than other fuels.

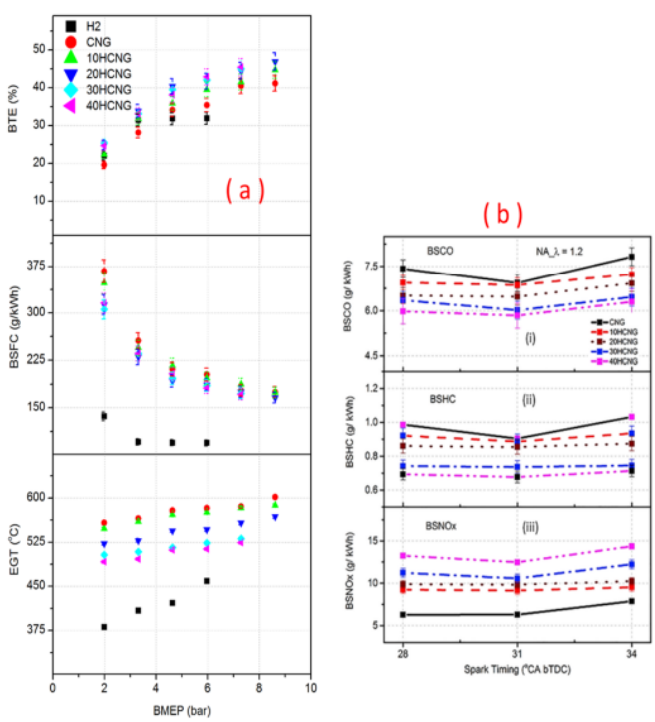


Figure5: (a) Variation of indicated thermal efficiency, indicated specific fuel consumption, indicated specific energy consumption and exhaust gas temperature, Prasad et al. [2021b]; (b) Brake specific emissions from HCNG blends in comparison to CNG, Shudo et al. [2000].

Figure 5(b) shows variations of brake specific emissions with ST. On retarding the spark timing as well as increasing the H_2 fraction in HCNG mixture, the BSCO emission, BSHC emissions, and BSNOx emissions reduced significantly. Lowest BSCO, BSHC and BSNOx were obtained at 31° CA bTDC ST for all HCNG mixtures. It was also observed that BSNOx emissions increased with increasing H_2 fraction in the HCNG mixtures for a given ST and □ . The only way to reduce NOx emissions is to operate the engine using lean mixture along with after treatment device (catalytic converter). Other researcher's investigations for suitability of HCNG for conventional engines are listed in the following paragraphs.

Shudo et al. [2000] investigated the combustion and emissions of direct injection stratified charge engine premixed HCNG-air mixture at lean condition. They showed that the engine achieved higher thermal efficiency due to higher flame propagation velocity and lower exhaust emissions. Hydrogen fraction increment resulted in an increase in combustion stability and reduction in HC and CO exhaust emissions while increased the degree of constant volume combustion and NOx exhaust emission. It was shown that the NOx emission could be maintained at a lower level with retarded ignition timing without reducing the improved thermal efficiency.

Thurnheer et al. [2009] investigated and optimized a state of the art passenger car operated using hydrogen–natural gas mixtures and high exhaust gas recirculation (EGR) rates in the major part of the engine map. It was observed that increasing the hydrogen fraction in the fuel, accelerated the combustion phenomena, which led to better efficiency.

Nagalingam et al. [1983] investigated engine parameters using HCNG and reported that power was reduced due to the lower volumetric heating value of hydrogen compared to methane. However, the higher flame speed of hydrogen (compared to methane) resulted in less spark advance to produce maximum brake torque (MBT).

Raman et al. [1994]carried out an experimental study using HCNG blends with varying H_2 from 0% to 30% in an SI V8 engine which was a 4-stroke Chevrolet Lumina, 6 cylinder, water cooled, 3.135 L with compression ratio of 8.8:1. They observed reduction in NOx and some increase in HC emission for ultra-lean combustion using 15%-20% H_2 blended CNG. The tested engine was. They found that BSFC of 15HCNG blends was less than that of natural gas. The BSFC values decreased for both natural gas and the 15HCNG mixture for advanced spark timing. The BSHC emissions of 15HCNG were lower while the BSNOx emission was higher compared to CNG.

Hoekstra et al. [1995] observed that NOx emissions decreased upon increasing the H_2 fraction upto 30% in HCNG. Beyond this H_2 fraction, no further reduction was observed. The higher flame speed and a consequent reduction of the spark advance angle to obtain the maximum brake torque, as already reported by Nagalingam et al. [1983].

Wang et al. [2010] investigated the combustion behavior of a direct injection HCNG engine operating on varying fractions of H_2. Their results showed that the brake effective thermal efficiency increased with increase of the H_2 fraction at low and medium engine loads. The combustion duration decreased while heat release rate and NOx emissions increased with increase of H_2 in the blends. They concluded that 20HCNG was the optimum H_2 volumetric fraction in natural gas (NG)-H_2 blends to achieve a compromise between both engine performance and emissions.

Kahraman et al. [2009] experimentally investigated the performance and exhaust emissions of a spark-ignition engine fueled by methane, 10HCNG, 20HCNG, and 30HCNG at different engine speeds and different □ . They concluded that CO emissions decreased with increased speed and □.. Furthermore, cylinder peak pressure decreased with increasing□ .

Ceper et al. [2009] experimentally analyzed the performance parameters and pollutant emissions of a 4-stroke SI engine operating on HCNG blends of 0%, 10%, 20% and 30% H_2 respectively at full load and 65% load for different □ . They showed that CO and CO_2 emissions decreased with increase in□ . In addition, increasing the □ led to decrease in-cylinder peak pressures. With increasing the H_2 fraction in HCNG, in-cylinder peak pressure shifted close to TDC, and the brake thermal efficiency values increased. Huang et al. [2006] conducted an experimental study for a direct-injection spark-ignition engine fueled by HCNG blends under various spark timings and lean mixture conditions. They concluded that spark timing was an important parameter for improving engine performance and combustion. Ma et al. [2008a], experimentally investigated the lean operation limit through addition of hydrogen to CNG in a 6-cylinder, throttle body injected SI engine. Four different test fuels, including CNG, 10HCNG, 30HCNG, and 40HCNG were used. They reported that the engine's lean operation limit could be extended by addition of hydrogen and increase of intake manifold pressure. The effect of engine speed on lean operation limit was smaller. At lower loads only an increase in engine speed was beneficial in extending the lean operation limit. The effects of engine speed were poor when the engine was switched on using HCNG. Spark timing optimized properly because both over-retarded and over-advanced spark timing are not advisable.

Genovese et al. [2011], performed road tests on urban transport buses to compare energy consumption and exhaust emissions for NG and HCNG blends with varying H_2 fraction (5% to 25% by volume). They found that average engine efficiency over the driving cycle increased with H_2 fraction in HCNG. The NOx emissions were found to be higher for 20HCNG and 25HCNG, despite leaner□ and delayed ignition timings. Hora et al. [2016a] investigated the effect of H_2addition to CNG and found that 30% HCNG (mixture of 30% (v/ v) H_2 and 70% (v/v) natural gas) showed most optimum performance and emissions benefits amongst the tested HCNG mixtures. They observed that H_2addition to CNG improved brake thermal efficiency

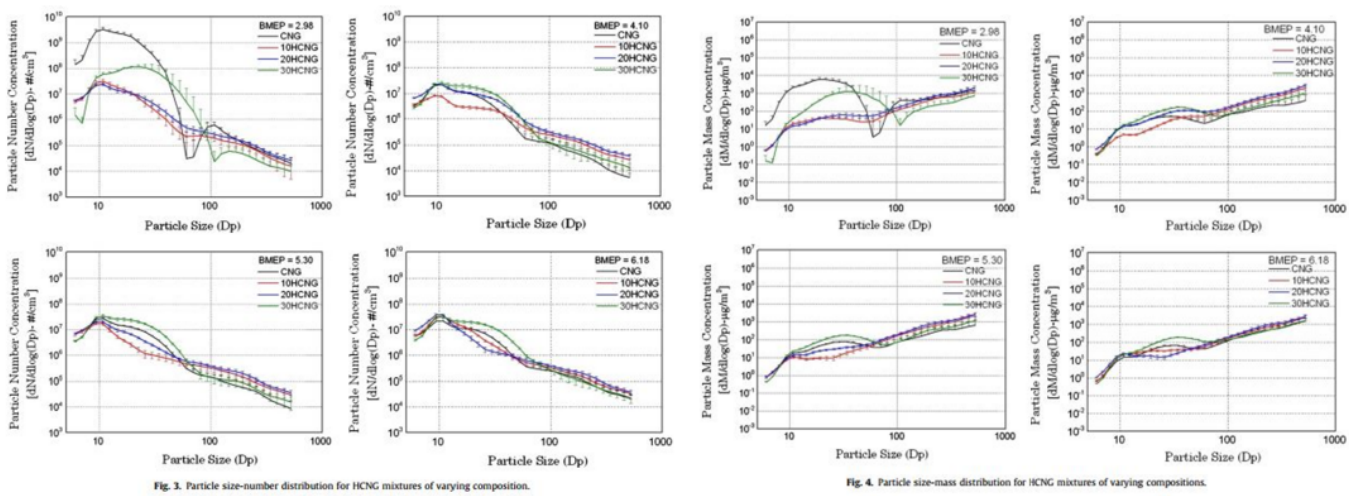


Fig. 3. Particle size-number distribution for HCNG mixtures of varying composition.

Fig. 4. Particle size-mass distribution for HCNG mixtures of varying compositions.

Figure 6: (a) Particle size number distribution for HCNG mixtures, (b) Particle size mass distribution for HCNG mixtures, Hora et al. [2016b].

(BTE), rate of pressure rise (RoPR), heat release rate (HRR) and combustion duration. It was also observed that brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC), emissions such as hydrocarbons (HC), carbon-monoxide (CO) and carbon-dioxide (CO_2) reduced while nitrogen monoxide (NO) increased. Further Combustion pressure increased with hydrogen addition to CNG.

Hora et al. [2016b] investigated the particulate emissions from a prototype single cylinder (1 litre cubic capacity) port fuel injected HCNG engine. The engine fueled by different blends of HCNG (0%, 10%, 20% and 30% v/v). They performed experiments at various operating points. At each operating point, particle size-number distribution was determined using an engine exhaust particle sizer (EEPS). They obtained higher particulate numbers in nano-particle size range compared to accumulation and coarse mode particles while their corresponding mass contribution showed an opposite trend as shown in Figure 6 (a, b). Among all tested HCNG blends, 30HCNG showed highest number of nano-particles emitted at all operating points with corresponding higher contribution to size-mass and size surface area distributions.

Park et al. [2011] performed the performance and emission characteristics of HCNG operated 11-L heavy duty engine in which lean burn combustion limit were examined. They suggested an optimization strategy for the control of excess air ratio. Spark advance timing was assessed in consideration of combustion stability. They reported that thermal efficiency increased with addition of hydrogen, allowing stable combustion under leaner operating conditions. NOx formation was closely related with excess air ratio of the mixture and spark advance. With proper optimization of these two parameters HCNG can effectively reduce NOx as much as 80%.

Mariani et al. [2013] performed experimental activity on a passenger car equipped with SI engine and fuelled with HCNG blends (15HCNG and 30HCNG). The engine was mounted on a chassis dynamometer and tested over different driving cycles. They maintained fixed ignition timing for all the tested fuels, in order to assess their interchangeability for practical purposes. They reported that CO_2 emission was reduced between 3to 6% for 15HCNG and between 13to 16% for 30HCNG compared with natural gas. Energy consumption (MJ/km) was found to be same for CNG and 15HCNG, while reductions upto 7% have been observed with 30HCNG. Heat release rate (HRR) increased with increasing H_2 fraction in the blends compared with CNG. The combustion duration (CD) has been shortened with 16% for 15HCNG and 21% for 30HCNG compared to CNG at 2.5 bar indicated mean effective pressure (imep) and 2400 rpm. As a consequence, increasing H_2 fraction resulted in a combustion phasing advance respect to CNG, reduced cycle-by-cycle variation particularly at low loads due to the positive effect of H_2 on combustion stability.

5. Turbocharged HCNG engines

Munshi et al. [2004] investigated the suitability of HCNG blends on an upgraded turbocharged lean-burn natural gas engine fuelled with blend of H_2 (fractions between 20 to 32 % V/V) and CNG. They found that 20HCNG gave desired benefits as per the engine and vehicle performance attributes. They performed steady state experiments according to AVL8 cycle after calibrating the HCNG engine at higher air–fuel ratios and retard spark timings. They observed 50% reduction in NOx and 58% reduction in NMHC emissions respectively, while the CO and CH_4 emissions were slightly reduced for same engine torque, power and fuel consumption that of natural gas fuel.

Deng et al. [2011] investigates the effect of various H_2fraction in HCNG fuels on combustion and emission characteristics of a turbocharged spark ignition natural gas engine having 6.234 L at idling conditions. They performed experiments using different H_2 fractions such as 0%, 30%, 55% and 75% by volume under various operating conditions including different excess air ratio and spark timing. They found that under various □ and spark timing, the addition of H_2 could significantly reduce methane (CH_4) emission and CO emission, although NO_x emissions increased with the H_2 addition. It was also observed that addition of H_2 can significantly reduce COV_{imep} (coefficient of variation of the indicated mean effective pressure), extend the lean burn limit, decrease the combustion duration, achieve higher thermal efficiency and reduce fuel consumption.

Ma et al. [2008b] investigated combustion and emission characteristics of a port-injection SI turbocharged engine (6.234 L) fueled with several H_2/ CNG blends (0–50 vol%) under various ignition timings. They found that with the increase of H_2fraction in HCNG, the maximum brake toque timing decreases (moves closer to TDC), combustion duration reduced and the indicated thermal efficiency increased and also increased with decreased load. The NOx, CO and HC emissions were decreased with the increase of spark advance angle, and increased with the increase of the load. Under the same spark advance, NOx and CO emissions rises while HC decreased with the increase of H_2 in blends.

Lee et al. [2014] investigated the idle performance of both CNG and 30HCNG on 11-L, 6-cylinder turbocharger engine equipped with an intercooler. They were used 30HCNG because it ensured a sufficient flame speed along with the optimal blending ratio which satisfies EURO-VI emission norms. They concluded that use of HCNG enhanced idle combustion stability and extended the lean operational limit from excess air ratio (□) 1.5 (CNG) to 1.6 and at idle condition decrease in fuel consumption (more than 25%) was observed compared to CNG. THC and CO emissions were decreased at idle with HCNG because of the low carbon content and enhanced combustion characteristics. The major pollutant in SI engine was NOx which were emitted in lower quantity when HCNG was fuelled into because of lower fuel requirement which eventually induced lower peak in-cylinder pressure for a stable idle condition compared with CNG operations.

6. Conclusions

Both Combustion chamber experiments and engine investigations conducted by various researchers the following conclusions were drawn:

1. SoC, combustion phasing, and EoC advanced with increasing BMEP in SI mode for all HCNG mixtures
2. COV_{IMEP}reduced with increasing H_2 fraction in the HCNG mixtures.
3. H_2 enrichment of CNG led to higher in-cylinder P_{max}
4. P_{max} increased with increasing BMEP and □ $_{Pmax}$ shifted towards the TDC.
5. BSHC, BSCO emissions were minimum but $BSCO_2$ emission was highest between □ = 1.2 to 1.4 for all HCNG mixtures for both LI and SI modes.
6. BSNOx emissions increased with increasing BMEP but reduced with increasing□ .
7. Existing engines with identical CRs showed improved engine performance and lean burning limit with HCNG mixtures.
8. Combustion improved (combustion pressure, RoPR, HRR and CHR) by hydrogen addition to the CNG, and due to relatively higher flame speeds achieved.
9. 10%, 50% and 90% MBF advanced with increasing H_2 enrichment. H_2 enrichment reduced exhaust emissions with identical conditions but CO, HC and NOx emissions increased with increasing CR, while CO2 emission reduced.
10. Particulate contribution of Gaseous fuel is less compared to conventional fuels

Overall, 20HCNG and 30HCNG mixtures provide superior performance and could be adopted in existing gasoline and CNG engines.

Ethics approval

Not applicable

Consent for publication

I give my consent to publish it in JEES.

Competing interests

Author declares that he does not have any personal or financial conflicts of interest.

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