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RESEARCH ARTICLE

Comparative analysis of sequential port fuel injection and conventional manifold fuel injection using 10% HCNG blends

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Abstract

Present research examines the relative performance, combustion and emission properties of a 10% HCNG (hydrogen-enriched compressed natural gas) blend in Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) engines. The experiments were performed under the conditions of the Wide-Open Throttle (WOT) at the Maximum Brake Torque (MBT) spark timing with a range of engine speeds of 2000 to 4000 rpm. The same was done online by blending Hydrogen with CNG to provide a similar mixture preparation in both modes of injection. The findings show that SPFI has a great improvement in the efficiency of the engine and combustion performance over CMFI. A 15.87% improvement by CMFI was a 16.06% brake thermal efficiency of SPFI at 4000 rpm. There was an increase in volumetric efficiency of about 20% in SPFI mode and a decrease in brake-specific fuel consumption at all the speeds that were tested. It was found that the peak cylinder pressure (34.9 bar) of SPFI was higher than 27 bar in CMFI, which also shows enhanced homogeneity of the mixture and quicker flame spread. The accumulation of emission analysis revealed a significant decrease in CO (19.17) and HC (>50) emissions with SPFI, but the whole of NOx emission growth was attributed to the high in-cylinder temperature that comes with hydrogen enrichment. In general, SPFI exhibits high thermo-chemical efficiency and cleaner combustion properties in terms of using 10% of HCNG mixtures, which underlines its applicability in the field of sustainable use of alternative fuels.

Keywords: Hydrogen-enriched compressed natural gas (HCNG), sequential port fuel injection, combustion analysis, brake thermal efficiency, heat release characteristics, engine emission performance, alternative gaseous fuels

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1.Introduction

The increased need to have cleaner and more efficient propulsion systems has enhanced research on the advanced fuel delivery strategies and alternative gaseous fuels to Spark Ignition (SI) engines. The fuel injection system is a decisive factor that controls the efficiency of combustion, the engine performance and exhaust emissions. Two systems that have been popular in multi-cylinder SI engines include Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI), which are two different methods of preparing the air-fuel mixture. CMFI systems feed the intake manifold with fuel, which is then channeled to the cylinders via the intake runners. Despite its simplicity and low cost, CMFI is usually characterized by poor distribution of mixture, reduced volumetric efficiency in the naturally aspirated engines, increased hydrocarbon (HC) emissions and often, the tendency to backfire during the valve overlaps. These constraints have a direct impact on brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and torque properties.

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On the other hand, SPFI feeds fuel to the engine in sequence to the individual intake ports by the engine firing order. The Engine Control Unit (ECU) can use real-time sensor data, including oxygen (O_2), throttle position (TPS), manifold absolute pressure (MAP), and exhaust temperature sensors, to precisely meter out fuel to each cylinder with every engine stroke. This controlled injection has terrific effects on intake charge preparation, greater combustion stability, greater power output, and less unburned HC emissions. Nevertheless, running under some conditions, the combustion temperatures can be large enough to promote the formation of nitrogen oxides (NOx).

As shown in table 1, along with the developments with injection systems, a recent developmental fuel that has taken the limelight in terms of transitional fuel is hydrogen-enriched compressed natural gas (HCNG). Any addition of 10% of hydrogen to CNG is an efficient percentage addition to promoting combustion without significant alterations in the engine. Besides the presence of low ignition energy (0.02 mJ vs. 0.29 mJ), high flammability range (4 -75% Vol vs. 5.3 -15% Vol), and laminar flame speed (3.25 ms^{-1} vs. 0.38 ms^{-1}), hydrogen has excellent combustion characteristics with a higher laminar flame speed, a greater flammability range (4 -75% Vol vs. 5.3-15% Vol), and a greater lower heating value per unit Its high diffusion coefficient and zero carbon content are the further reasons behind the higher propagation of the flame and the negligible production of the CO_2. These features ease lean-burn operation, enhanced thermal efficiency, lower rates of carbon monoxide (CO) and HC pollution as well as cleaner exhaust profiles.

Table 1. Properties of Hydrogen Methane and Gasoline at NTP [29-32]

Property	Unit	Hydrogen (H_2)	Methane (CH_4)
Auto-ignition temperature	K	858	813
Stoichiometric air requirement (vol.)	%	29.53	9.48
Energy release per kg of stoichiometric mixture	MJ	3.37	2.56
Gas density at NTP	kg m^{-3}	0.082	0.717
Liquid density	kg m^{-3}	0.071	0.42
Diffusion coefficient in air	$cm^2\ s^{-1}$	0.61	0.189
Flammability range (equivalence ratio)	–	0.1–7.1	0.7–4.0
Energy density of stoichiometric mixture	MJ m^{-3}	3.6	3.5
Adiabatic flame temperature (λ = 1)	K	2318	2190
Higher heating value (mass basis)	MJ kg^{-1}	141.7	52.68
Higher heating value (volume basis)	MJ m^{-3}	12.1	37.71
Lower heating value (mass basis)	MJ kg^{-1}	120	46.72
Lower heating value (volume basis)	MJ m^{-3}	10.22	33.95
Lower heating value	kJ m^{-3}	10046	32573
Kinematic viscosity (NTP)	$mm^2\ s^{-1}$	110	17.2
Laminar flame speed	m s^{-1}	3.25	0.38
Smallest ignition energy	mJ	0.02	0.29
Carbon-to-hydrogen molar ratio	–	0	0.25
Normal boiling point	K	20.3	111.6
Quenching distance	mm	0.64	2.03
Research octane number	–	>130	125
Stoichiometric fuel–air ratio (mass basis)	–	0.029	0.058
Thermal conductivity	mW $m^{-1}\ K^{-1}$	182	34
Radiative heat loss from flame	%	17–25	23–33
Equivalence ratio at stoichiometric condition (φ = 1 reference)	–	0.29	0.095
Specific energy (volumetric)	kJ m^{-3}	–	3200
CO_2 emission formation	g CO_2 $kW^{-1}\ h^{-1}$	Nil	200
Flammability limits	% vol	4–75	5.3–15

1.1 Global significance

Globally, HCNG blends help in insulating energy security, environmental sustainability and economic viability. They also utilize the available CNG infrastructure and enable a progressive incorporation of hydrogen, thus aiding in meeting the strict norms of emissions and global climate agreements. Also, HCNG can be synthesized through electrolysis, which provides a route to renewable and low-carbon systems. Despite these benefits, hydrogen blending presents the following challenges as NOx inflammation in high temperatures of combustion, storage and safety, and optimization of the injection system. The HCNG combustion strongly relies on the fuel injection strategy that is applied. CMFI systems can be subjected to mixture in homogeneity and variability in the combustion process when using gaseous fuels, while SPFI can have better air-fuel mixing and combustion and cylinder to cylinder uniformity. Nevertheless, literature on the comparison of SPFI and CMFI systems with 10% HCNG mixtures in multi-cylinder SI engines is still scanty.

1.2. Aim

The overall aim of the study is to decide the potential of using 10% of hydrogen-enriched compressed natural gas (HCNG) as a cleaner and a high-energy substitute fuel in multi-cylinder Spark Ignition (SI) engines with specific attention paid to the effects of fuel injection strategy. Although HCNG blends have great potential of decreasing greenhouse gases emissions and enhancing engine performance, their practical advantages are highly dependent on how well the air-fuel mixture is prepared and distributed. One of the problems that are commonly experienced with conventional Manifold Fuel Injection (CMFI) systems on multi-cylinder SI engines is the inability to achieve uniform mixing of the fuel and air into the engine, the existence of uneven distribution of the cylinders and the excessive emission of exhausts. By comparison, Sequential Port Fuel Injection (SPFI) systems offer much better timing of injection as well as a better atomization rate, potentially resulting in higher competence of combustion and lowering of emissions. This paper will thus conduct a comparative analysis of SPFI and CMFI systems under the condition of 10 percent HCNG blends to find the most efficient injection strategy that can be used to give better performance and lower emissions.

1.3. Objectives

Thus, the current research paper is expected to perform a thorough comparative analysis of SPFI and CMFI systems on 10 % blends of HCNG in a multi-cylinder SI engine. The aims include:

1. Performance Assessment: BTE, BSFC and torque comparison during the HCNG running.
2. Combustion Analysis: Determination of in-cylinder pressure, rate of heat release, rate of increase of pressure and stability of combustion.
3. Emission Characterization: Characterization of the emissions of NOx, CO, HC and PM.
4. Air-Fuel Mixing Optimization: The study on the homogeneity of the mixture and the effects it has on the regularity of combustion.
5. Trade-off Identification: Personality of performance-emission trade-offs, especially NOx formation, with hydrogen enrichment.
6. System Optimization: Timing and parameter optimization recommendations for injection in SPFI systems and CMFI systems.
7. Sustainability View: HCNG as a green transitional fuel in automobiles.

This research develops a systematic method that allows optimization of multi-cylinder SI engines to create higher efficiency and lower emissions, and a sustainable use of energy through the integration of combustion properties of hydrogen-methane mixtures with innovative injection models.

2. Literature survey

The research on fuel injection planning and alternative gaseous engines shows that the efficiency of combustion and emission decrease in the SI engines have been improved greatly. The gasoline/isobutanol mixtures in port injection engines were examined by Irimescu [1] and showed that there was increased efficiency in the conversion of fuel, and the preparation of the mixture is of great significance in PFI systems. Khan et al. [2] thoroughly reviewed the literature on the topic of natural gas as a vehicular fuel, focusing on the beneficial effects of its use on the environment and the necessity of best injection technologies. The study of spray impingement in crossflow port fuel injection systems by Panao et al. [3] proved that the airflow dynamics is very important in deciding the homogeneity of the mixture and wetting of the walls. Comparing the results of particulate emissions of multipoint PFI and GDI engines, Lv et al. [4] concluded that the injection setting has a major influence on the number and size distribution of the particles. Zhuang and Hong [5] set up that timing in injection is critical in the reduction of knock and stability of lean burning during ethanol-gasoline engines. Yang et al. [6] investigated gas injection into natural gas ports using the diesel pilot injection and they achieved an improvement of the stability of combustion at low loads, while Huang et al. [7] numerically analyzed the dual-fuel spray processes and proved the advantages of hybrid injection configurations. Catapano et al. [8] studied the ethanol, methane, and methane-hydrogen mixtures in PFI engines, and the studies found that the enrichment of hydrogen would enhance the speed of flame and the efficiency of combustion. Geng et al. [9] conducted experimental testing of the present blends of methanol and gasoline in PFI engines and found a decrease in PM emissions when using oxygenated fuels. Qian et al. [10] suggested sequential dual-fuel combustion with port and direct injection to enhance the thermal efficiency. Huang and Hong [11] also added that heated ethanol increases the rate of vaporization and minimizes the emission. Zhu et al. [12] compared the tailpipe emissions of the GDI and PFI with various temperatures, and they showed low emissions of particulate in the PFI systems. Cheng et al. [13] paid attention to the nature of hydrogen injections in PFI engines and discussed backfire control principles that are important in the case of hydrogen-enriched fuels. He et al. [14] compared DI and MPFI technologies and showed that they differ in the number and black carbon emissions. Lee et al. [15] invented a two-port injection mechanism to have better fuel economy and to lower emissions. Jemni et al. [16] revealed that the effect of hydrogen enrichment and location of injection affects the in-cylinder flow and the formation of NOx in gaseous engines greatly. Feng et al. [17] and Lee et al. [18] examined dual-injection schemes, and by confirming the superiority of the combustion control by the ideal injection timing and spray behavior. Lee et al. [19] have compared gasoline DI and CNG PFI under high load conditions, and they found that combustion of the CNG sample with the port injection of the port was stable. Salek et al. [20] used multi-objective optimization to the hydrogen-gasoline dual-fuel engines and found out performance-NOx emissions trade-offs. The study by Kim et al. [21] set up that PFI natural gas engine performance can be improved by increasing and valve timing. Libalova et al. [22] studied the biological toxicity of the PFI engine emissions, which correlates with the composition of the fuel with the response of the cells to the particulate. Varma et al. [23] examined the roles of alcohol fuels in engines running with PFI SI and found that these engines showed better performance in terms of combustion efficiency in optimized spark timing. Yaman et al. [24] used response surface method to optimize the engine parameters during alcohol-gasoline blends and supported the idea that injection control plays a key role in providing emissions reduction. Direct and port fuel injection in hydrogen engines have been compared by Molina et al. [25] and Musy et al. [26], who conclude that the injection strategy has a great impact on backfire, efficiency, and trends of NOx. An intake manifold geometry was studied by Saaidia et al. [27] engines powered by hydrogen (H_2) and natural gas (CNG) and proved its effect on uniformity of the mixture. Koeten [28] explored the effect of hydrogen addition to the CI engines citing better combustion and higher NOx with greater hydrogen additions. Basic combustion properties of hydrogen have been well reported by Das [29], Karim [30], Lewis and von Elbe [31] and the College of Desert report [32] that report on the high speed of flame, broad flammability range, low ignition energy and large diffusivity properties of hydrogen which are considered favorable to lean burning but lead to pre-ignition and backfire effects in port injection systems. Overall, earlier investigations show that hydrogen enrichment positively affects the combustion efficiency and emission properties, while the injection strategy is the key factor in the formation of mixtures, their stability, and formation of pollutants. Nevertheless, a dedicated comparative analysis of the Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) systems using multi-cylinder SI engines with 10% HCNG mixtures has not been done with the clear aim of isolating the rationale behind the current research.

2.1. Summary of literature survey

- The literature review shows that port fuel injection (PFI), gasoline direct injection (GDI), and dual-injection methods play an important role in the combustion efficiency, fuel economy, and the emission properties of spark ignition (SI) engines [1 5,14,15]. Other works have shown that fuels naturally enriched with natural gas and hydrogen enhance the stability of combustion rates, flame rate, and reduction of carbon-related emissions because of the enhanced physicochemical characteristics of hydrogen, including high diffusivity, low activation energy, and broad flammability limits [8,13,16,2932]. The investigation of dual-fuel and hydrogen-blended systems also proves higher thermal efficiency and the possibility of lean-burn, but the problems of NO_x formation, backfire inclination, and homogeneity of the mixture also represent very important issues [19,20,25,26].
- Direct and port injection system comparisons mainly revolve around the topic of particulate emissions, spray behavior, or dual-injection geometries [4, 12, 17, 18], while studies on hydrogen or methane mixtures partially consider the system performance on single-injection schemes [8, 21, 27]. Despite the extensive research done in the addition of hydrogen in both SI and CI engines [16,28], most studies have focused either on directly injecting hydrogen engines or on general PFI systems instead of the systematic comparison of Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) under the optimized conditions of hydrogen-enriched CNG.

2.2. Research gap

Nevertheless, a comparative analysis of SPFI and CMFI systems working with a 10 % mixture of HCNG in multi-cylinder SI engines has not been carried out methodically. Minimal studies are done on the matter of cylinder-to-cylinder mixture distribution, consistency of combustion, trade-offs in emissions (especially NO_3 escalation), and measuring the performance in the same operating conditions on both injection systems. Moreover, the combustion kinetics of hydrogen-enhanced reaction and injection timing accuracy have not been fully measured.

2.3. Novelty of the present work

The originality of the current research is to conduct a more thorough experimental analysis of SPFI and CMFI systems with an optimized 10% HCNG mixture in a multi-cylinder SI engine with the same operating conditions. The paper associates the enactment (BTE, BSFC, and torque), combustion analysis (in-cylinder pressure, heat release rate, pressure rise rate) and emission summarizing (NOx, CO, HC, PM) to arrange complete interpretation of inoculation approach presentation. In addition, it highlights uniformity of air-fuel mixing and stability of combustion as some of the differences between the SPFI and CMFI systems hence generating practical implications on how to maximize the operation of hydrogen-enriched gaseous fuel in automotive SI engines.

2.4. Oxides of Nitrogen and its effects in HCNG engine

The combustion properties of hydrogen in HCNG engines are very significant in the formation of oxides of Nitrogen (NOx). Hydrogen has an increased flame temperature and burns rate than CNG, which results in elevated in-cylinder peak temperatures, which favor the formation of thermal NOx. With an increase in the hydrogen content of the mixture, particularly beyond 30%, the peak combustion temperature is likely to rise further, and this may enhance NOx emissions in stoichiometric ($\lambda = 1$) operation. Nevertheless, the broad flammability limits of hydrogen make it possible to conduct stable ultra-lean combustion, resulting in a lower combustion temperature and considerably decreasing the formation of NOx. Exhaust Gas Recirculation (EGR), best ignition timing, and sequential injection methods are some of the techniques used to control peak temperatures and reduce NOx. Also, the post-treatment methods, such as Selective Catalytic Reduction (SCR) and Lean NOx Taps (LNTs), and other modern combustion methods, such as dual-fuel injection and pre-chamber designs, further decrease the amount of NOx. Practically, having 10-20% hydrogen by volume blends is sometimes viewed as an excellent tradeoff between improved engine performance and regulated NOx emissions.

3. Experimental set up

The experimental study was conducted on a naturally aspirated 796 cc, three-cylinder spark ignition (SI) gasoline engine, which was changed to execute both the Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) operation mode in running on dual-fuel HCNG. Compression ratio kept constant at 9.2 and all the tests were done at constant speeds of 2000, 2500, 3000, 3500, and 4000 rpm under wide-open throttle (WOT) conditions and with stoichiometric equivalence ratio ($\varphi = 1.0$). The SPFI and CMFI modes were carried out as separate experimental runs and so comparative evaluation can be carried out correctly. With SPFI mode three HCNG injectors were mounted close to the inlet valves near the gasoline injectors so that they could inject sequentially, which was controlled by ECU. CMFI mode In CMFI mode, the injectors controlled by the ECU were disabled and gaseous fuel at the vaporizer was introduced with a single injector inside the intake manifold in the MAP sensor and manually flow controlled via a calibrated valve. This change has made it possible to evaluate the distribution and combustion properties of mixtures under typical manifold injections.

Figure 1. Cooled EGR system

Figure 1 (Cooled EGR System) shows the design of the installation of a cooled Exhaust Gas Recirculation (EGR) system, which is important in regulating NOx emission during combustion of HCNG. Because hydrogen enrichment raises flame temperature and peak in-cylinder temperature, cooled EGR decreases combustion temperature, which means that recirculated part of exhaust gases is cooled and exhaust gases are suppressed in thermal NOx formation at higher loads and speeds.

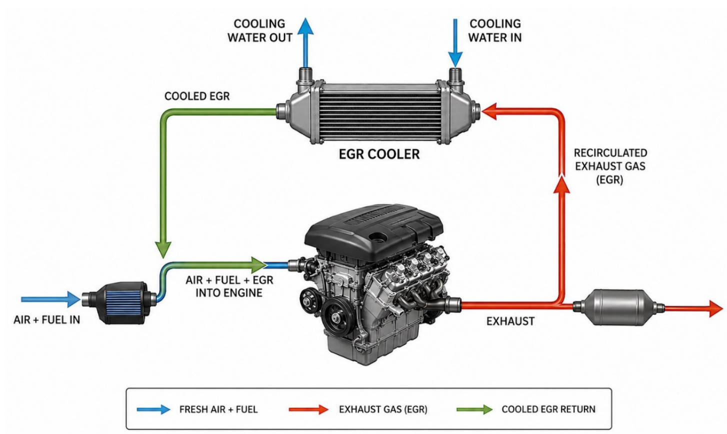

Figure 2. Flow diagram of cooled EGR system

Figure 2 (Flow Diagram of Cooled EGR System) shows the detailed direction of the flow of gases; it shows how the exhaust gases are directed through the cooler and then combines with the intake air. This arrangement guarantees that the intake charge is diluted in a controlled manner, which enhances stability in combustion, as well as upholding rates of emissions.

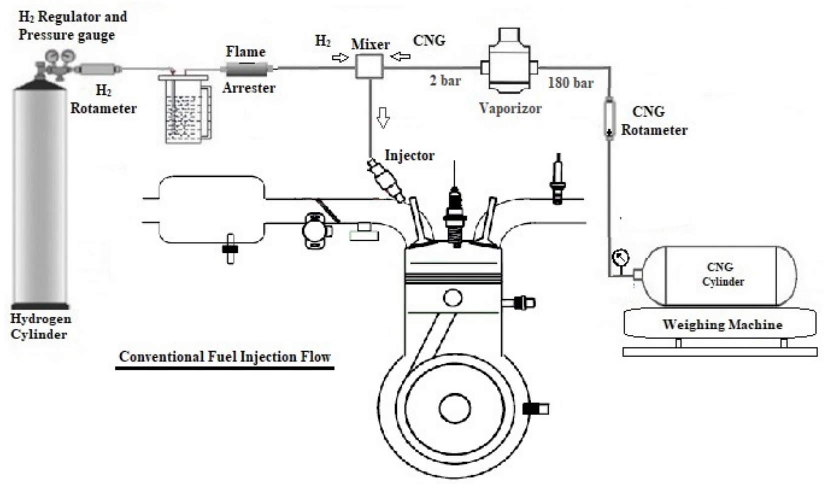

Figure 3. Conventional manifold fuel injection system

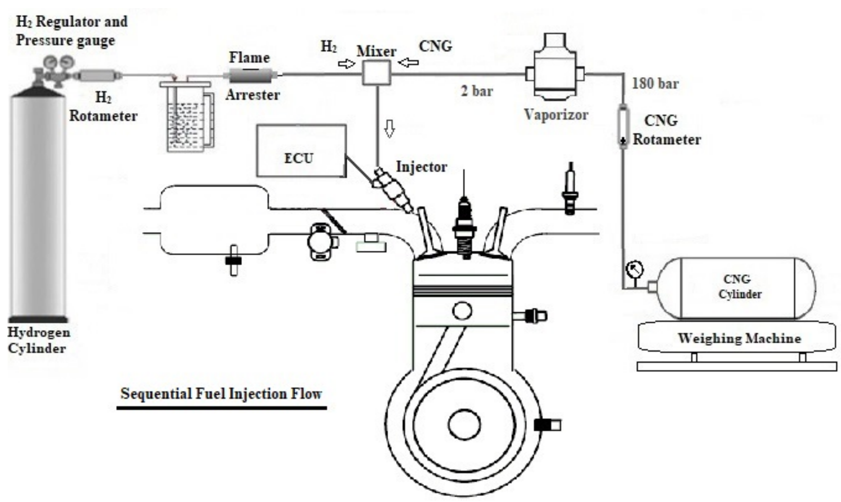

Figure 4. Sequential port fuel injection system

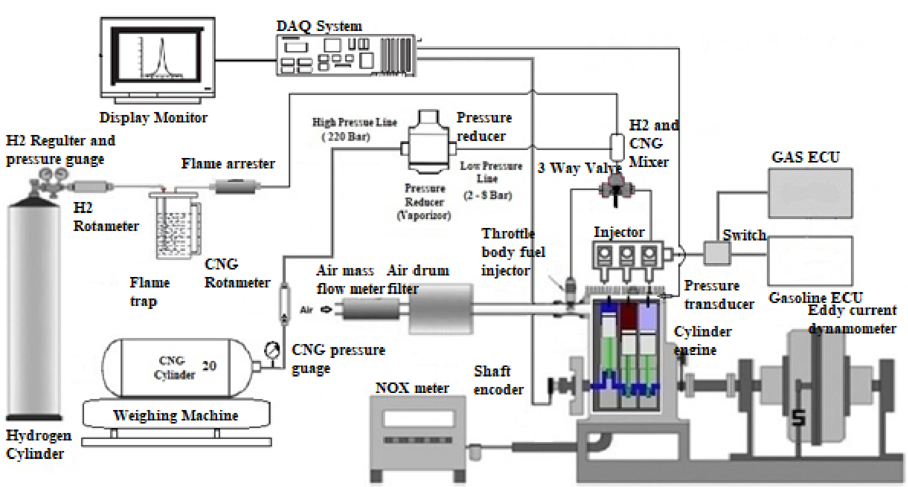

Figure 5. Experimental set up

The CMFI design is represented by Figure 3 (Conventional Manifold Fuel Injection System), in which the premixed mixture of HCNG-air is created in the intake manifold. Such a configuration is useful in testing the consequences of non-uniform mixture distribution, any backfire tendencies and variability in combustion during multi-cylinder operation.

The SPFI system is depicted in Figure 4 (Sequential Port Fuel Injection System), with injectors attached close to every intake port. This connection results in superior distribution of cylinders to other cylinders on fuel, enhanced atomization and control of timing that are fundamental to combustion of hydrogen-enriched gaseous fuels.

The entire engine-dynamometer set up is shown in Figure 5 (Experimental Setup). Brake load is measured with an eddy current dynamometer with a strain gauge; engine speed is regulated with the dynamometer load at full throttle condition. Cooling water is pumped throughout the dynamometer to cool down the heat that is produced during loading.

Figure 6. Hydrogen fuel supply setup with flame trap

Figure 6 (Hydrogen Fuel Supply Setup with Flame Taps) illustrates the hydrogen supply system, which consists of 90-liter hydrogen cylinder, 180 bar, and safety devices like flame traps and flame arrestors fitted to hydrogen and CNG lines to avoid flashback and backfire.

Figure 7. HCNG setup with engine cooling system

Figure 7 (HCNG Setup with Engine Cooling System) shows the blending system and engine cooling circuit that are linked together. A 75-liter CNG cylinder with a pressure of 200 bars and a hydrogen cylinder are adjusted using pressure reducers to 2.8 bar and mix on-line. Hot coolant water is made to circulate around the vaporizer to avoid pressure drop icing. The engine temperature is kept at 80^{0}C to have steady-state operation.

Fuel flow rates are recorded using rotameter (LPM basis) and confirmed with the help of mass measurement with a weighing machine. The air consumption is measured by air-box method when a manometer is used. Distributor is used to perfect spark timing at each speed and stroboscopic timing light is used to confirm maximum brake torque (MBT). The pressure inside the cylinders is measured with the help of a water-cooled pressure transducer in one of the cylinders and the NI-DAQ Software is used to record the Pressure -Angle of Crank (P -) diagrams, Heat Release Rate (HRR), Cumulative Heat Release Rate (CHRR), Mass Fraction Burned (MFB), Start of Ignition (SI), and Maximum Pressure (Pmax). The injection pressure and pulse width are checked by using TAMMONA software in SPFI mode.

The online measurement is taken of exhaust emissions through an AVL five-gas analyzer by measuring HC, CO, CO_2, O_2 and NOx concentrations. An RTD thermocouple is used to measure exhaust temperature, and an O_2 sensor is used to measure oxygen content. Theoretical calculations are used to calculate emission values in g/kW-hr based on SAE standards. The determination of friction power is done through the Morse test at every speed.

The engine also has the necessary sensors such as Oxygen (O_2), MAP, TPS, RPM, and Inlet Air Temperature sensors to provide the correct data. Each test lasts five minutes in steady-state conditions to get credible performance, combustion and emission data. Parameters that are measured are load, engine speed, air and fuel consumption, exhaust temperature, emission levels, and detailed combustion characteristics by use of P– 0 analysis.

Such an elaborate experimental apparatus allows a comparative study of SPFI and CMFI systems with 10 % HCNG blends to give a proper understanding of the performance, combustion phenomena, and emission nature under a controlled and standardized environment.

3.1. HCNG blending calculation by energy

The following were changed formulations that were used to calculate engine performance, fuel consumption and combustion parameters using a customized Microsoft Excel worksheet.

Brake power based on torque and speed (BP)

$$BP = \frac{2 \times \pi \times N \times T}{60 \times 1000} = \frac{2 \times 3.14 \times N \times T}{60000}$$

Where,

N = Engine speed (rpm)

T = Torque (N·m)

BP = Brake Power (kW)

Brake Power can also be expressed using Brake Mean Effective Pressure (BMEP):

$$BP = \frac{BMEP \times L \times A \times N}{60000}$$

Where:

L = Stroke length (m)

A = Piston area (m²)

Brake Mean Effective Pressure (BMEP)

$$BMEP = \frac{BP \times 60}{V_s \times N}$$

For the present engine geometry:

$$BMEP = \frac{BP \times 60 \times 10^5}{0.0725 \times \left(\frac{\pi}{4} \times 0.066^2\right) \times N}$$

Where:

V_s = Swept volume (m³)

Brake Thermal Efficiency (BTE)

$$BTE = \frac{BP \times 3600 \times 100}{m_f \times LHV_{blend}}$$

Where:

m_x = Total fuel mass flow rate (kg/hr)

$LHV_{(blend)}$ = Lower Heating Value of HCNG blend (kJ/kg)

Lower Heating Value of HCNG Blend

The calorific value of the blend is calculated using mass fraction weighting:

$$LHV_{blend} = \frac{(m_{CNG} \times LHV_{CNG}) + (m_{H_2} \times LHV_{H_2})}{m_{CNG} + m_{H_2}}$$

Where,

$$m_{total} = m_{CNG} + m_{H_2}$$

Fuel Mass Flow Rate Calculation

Mass flow rate from volumetric flow (LPM):

$$m = \frac{Q \times \rho \times 60}{1000}$$

Where:

Q = Flow rate (L/min)

ρ = Density (kg/m³)

60 = Conversion to per hour

1000 = Conversion from litre to m³

Thus,

$$m_{CNG} = \frac{Q_{CNG} \times \rho_{CNG} \times 60}{1000}$$

and

$$m_{H_2} = \frac{Q_{H_2} \times \rho_{H_2} \times 60}{1000}$$

Example Calculation

Operating Condition:

Pressure = 2.6 bar

Speed = 2500 rpm

Load = 16 kg

Hydrogen Energy Share = 5%

CNG consumption = 3.3 kg/hr

Step 1: Energy Input from CNG

Energy CNG = 3.3 × 46720 = 154.18 MJ/Hr

Step 2: Hydrogen Energy Contribution (5%)

Energy H_2 = 0.05 × 154.18 = 7.71 MJ/Hr

Step 3: Hydrogen Flow Rate Calculation

Given:

ρH_2 = 0.021 kg/m³

$LHVH_2$ = 120000 kJ/kg

Hydrogen required (mH_2) =

$$\frac{\text{Energy } H_2}{\text{LHV } H_2}$$

Corresponding volumetric flow (QH_2) =

$$\frac{m_{H_2} \times 1000}{\rho_{H_2} \times 60}$$

Therefore, Hydrogen flow rate = 5.14 LPM

Step 4: CNG Flow Rate

Corresponding CNG volumetric flow = 31.32 LPM

Thus, 5.14 LPM of hydrogen + 31.32 LPM of CNG produces 5% HCNG blend (energy basis).

Step 5: LHV of 5% HCNG Blend

Mass of CNG (mCNG) =

$$\frac{31.33 \times 1.6679 \times 60}{1000} = 3.135 \text{ kg/hr}$$

Mass of Hydrogen (mH_2) =

$$\frac{5.142 \times 0.21 \times 60}{1000} = 0.065 \text{ kg/hr}$$

LHV_5%HCNG =

$$\frac{(3.135 \times 46720) + (0.065 \times 120000)}{3.135 + 0.065} = 48.187 \text{ kJ/kg}$$

Percentage of Hydrogen by Volume = $\%H_2$ (Volume) =

$$\frac{Q_{H_2}}{Q_{H_2} + Q_{CNG}} = 14.09\backslash \text{ by volume}$$

These new expressions were entered in the Excel sheet so that the brake power, BTE, BMEP, fuel flow rates, heating values, and ratios of blends could be calculated in a systematic manner under all test conditions of the experiment.

Table 2. Computed thermo-physical and energy characteristics of HCNG blends (based on theoretical formulation)

Parameter	Pure CNG (0% H_2)	HCNG–5% (vol)	HCNG–10% (vol)	HCNG–15% (vol)
Hydrogen Content (Volume %)	0	5	10	15
Methane Content (Volume %)	100	95	90	85
Hydrogen Proportion (Mass %)	0	0.705	1.377	2.169
Methane Proportion (Mass %)	100	99.29	98.623	97.831
Hydrogen Energy Share (%)	0	1.652	3.242	5.053
Theoretical Air–Fuel Ratio (Stoichiometric)	17.19	17.23	17.26	17.284
Lower Calorific Value (MJ/kg)	50.02	50.492	50.964	51.519
Hydrogen Mass Contribution (%)	25.13	25.62	26.16	26.75
Carbon Mass Contribution (%)	74.87	74.38	73.84	73.25
Volumetric Heating Value (MJ/m³)	3.17	3.167	3.164	3.16

Table 3. Evaluated thermodynamic and composition characteristics of HCNG blends (Reference conditions: 98 kPa, 300 K – based on theoretical calculations)

Sr. No.	Parameter	Unit	CNG (0% H_2)	HCNG–10%	HCNG–20%	HCNG–30%	HCNG–40%	HCNG–50%
1	Mixture Density	kg/m³	0.5196	0.5931	0.5362	0.4793	0.4225	0.3656
2	Hydrogen Share (Mass %)	%	0	1.369	3.03	5.084	7.692	11.11
3	Methane Share (Mass %)	%	100	98.63	96.96	94.91	92.3	88.88
4	Elemental Hydrogen Content	%	25	26.03	27.27	28.81	30.77	33.33
5	Elemental Carbon Content	%	75	73.97	72.72	71.19	69.23	66.66
6	Lower Calorific Value	kJ/kg	50,000	50,954	52,084	53,556	55,380	57,772
7	Lower Calorific Value	kcal/kg	11,945	12,173	12,443	12,794	13,240	13,801
8	Volumetric Heating Value	kJ/m³	32,495	30,224	27,927	25,669	23,350	21,121
9	Carbon-to-Hydrogen (C/H) Ratio	–	0.25	0.2368	0.2222	0.2058	0.1875	0.1666
10	Hydrogen Energy Contribution	%	–	2.223	6.912	11.382	16.66	23.07
11	Theoretical Stoichiometric Air–Fuel Ratio	–	17.16	17.39	17.68	18.04	18.49	19.07

3.2. Characteristics and theoretical fuel property trends of HCNG blends

Figure 8 as shown below shows the dependence on Hydrogen Volume Fraction and Energy Contribution. The variation of the percentage of hydrogen energy replacement and the volumetric fraction of hydrogen in the HCNG blend is illustrated in figure 8.

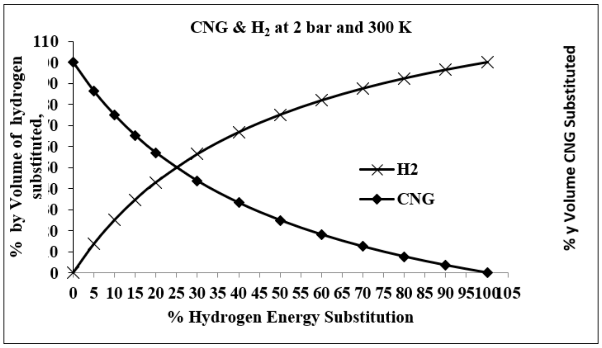

Figure 8. Percentage Hydrogen energy substitution vs. percentage volume of Hydrogen substituted

The graph shows a nonlinear rising pattern, which implies that when the hydrogen volume fraction is increased by small increments, it brings an enormous contribution to hydrogen in terms of energy since it has a high lower heating value on a mass basis. Hydrogen offers a better calorific value per unit mass than methane, but its reduced volumetric energy density means that it can be substituted by it at relatively low blending percentages. This tendency proves the significance of managing the blend composition to achieve the balance between the enhancement of performance and the emission.

Figure 9 shows the effect of the force of supply on CNG density. The density of CNG increases proportionately when there is an increase in pressure and the gas behaves ideally. Increased density of the gas under high pressure lets it pass through the mass increased by a given volumetric rate directly influencing fuel-air ratio control and engine power output. This interaction is also essential to stoichiometric operation (φ = 1) in the HCNG blending process since correct pressure control guarantees the constant metering of the fuel and stable combustion behavior.

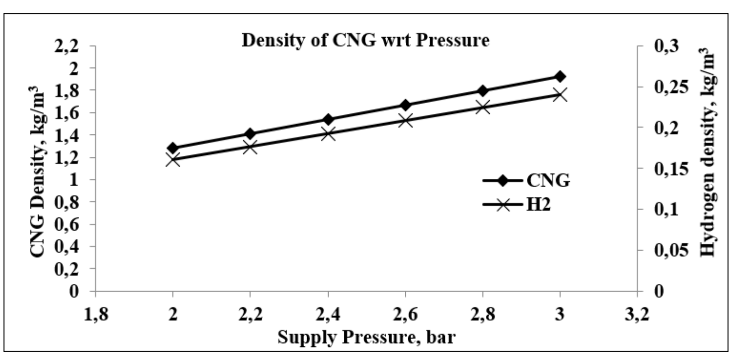

Figure 9. Supply pressure vs. CNG density, kg/m3

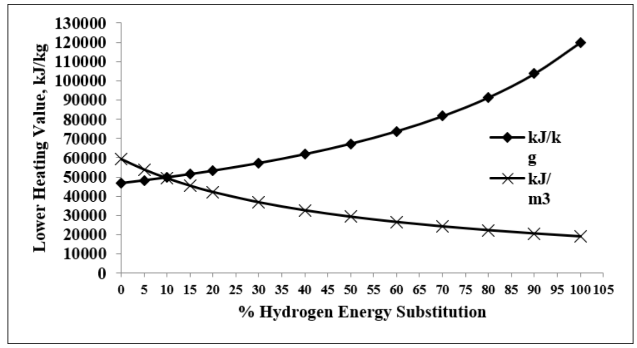

Figure 10. Percentage Hydrogen energy substitution vs. Lower heating value, kJ/kg

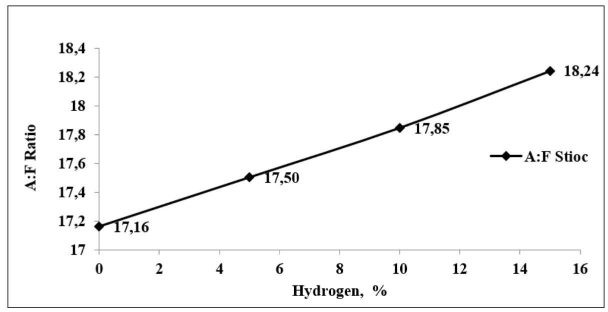

Figure 11. Hydrogen % vs A / F ratio

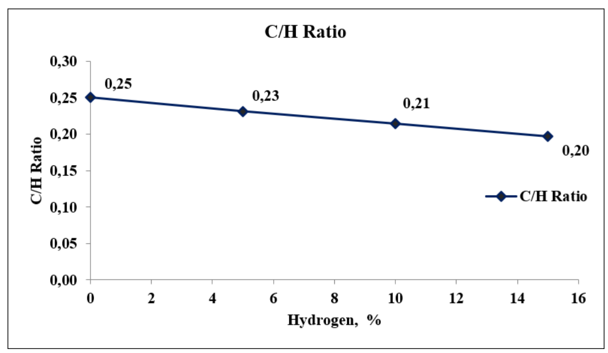

Figure 12. % Hydrogen vs. C/H Ratio

Figure 10 shows the change of the lower heating value (LHV) of the HCNG blend with substitution of the hydrogen energy. Addition of hydrogen to the LHV is gradually rising as hydrogen is much more massive-based heating value than methane. This increase in calorific value increases thermal efficiency capability and intensity of combustion. Nevertheless, the growth rate is moderate as it is caused by the reduced density of hydrogen where there is a focus on ensuring that the blend ratios are well maximized to provide maximum efficiency without reducing the volumetric content of energy.

The correlation between the percentage of hydrogen in the blend and the monitoring stoichiometric air fuel (A/F) ratio is illustrated in figure 11. Theoretical A/F ratio is slightly higher as the hydrogen concentration grows. This is since hydrogen needs another air requisite to undergo full combustion as compared to methane. This change in stoichiometric ratio influences the preparation of mixture, the calibration of the injection in the SPFI and CMFI systems and the optimization of the spark timing. The injection parameters should then be adjusted properly to ensure the best combustion and reduce the emissions.

Figure 12 shows that the ratio of carbon to hydrogen (C/H) ratio declined as the percentage of hydrogen in the blend of HCNG increased. Hydrogen has no carbon and when it is added to the mixture it decreases the total carbon level of the mixture. This decrease has a direct impact on the decrease of CO_2 formation during combustion and eases cleaner exhaust emissions. The decreasing C/H ratio confirms the environmental benefit of the HCNG blends and the possibility of these blends mitigating the carbon-based pollutants in SI engines.

4. Result and discussion

In this part, the results of the experiment with the multi-cylinder SI engine with 10% HCNG blends which was conducted in the mode of Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) at 2000 to 4000 rpm in the wide-open throttle conditions, as well as in stoichiometric conditions ($\varphi = 1$). The comparative analysis is also based on the major parameters of performance like brake power, brake thermal efficiency, brake mean effective pressure, brake specific fuel consumption, brake specific energy consumption, volumetric efficiency and exhaust gas temperature and on the properties of the emission like CO, HC and NOx. Besides that, the combustion analysis is conducted in detail using measurements of in-cylinder pressures to decide the peak cylinder pressure, the heat release rate, mass fraction burned, the length of combustion and the behavior of the spark timing and the cycle-to-cycle variation (COVIMEP). The findings identify the effect of hydrogen enrichment and injection strategy on the formation of mixtures, on stability of combustion, enhancement of efficiency and control of emissions during HCNG operating conditions.

4.1 Brake power performance

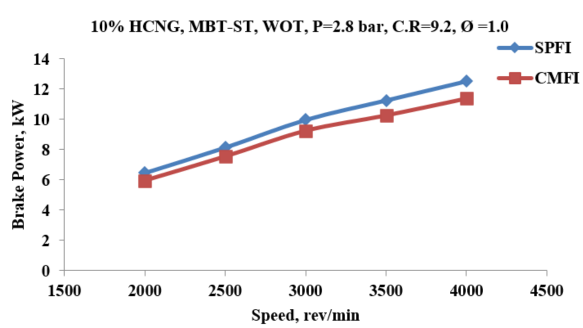

Figure 13. Brake power vs. engine speed

Figure 13 is a comparison of the variation of brake power versus the engine speed in SPFI and CMFI modes with 10% HCNG blends. Since gaseous fuels have lower density as compared to liquid fuels, charge mass induced reduces and this has a small impact on power output in naturally aspirated engines. Nevertheless, SPFI was also characterized by greater power at the brake at all the speeds, owing to the correct metering of the fuel and the enhanced distribution of the cylinder to cylinder. The highest brake power achieved when using SPFI is 12.506 kW, as compared to the 11.38 kW which CMFI gives, meaning that there is a loss of about 6 percent of power when using the manifold injection set up. The decreased production in CMFI is explained by inhomogeneous formation of mixtures and decreased volumetric efficiency.

4.2. The characteristics of brake thermal efficiency

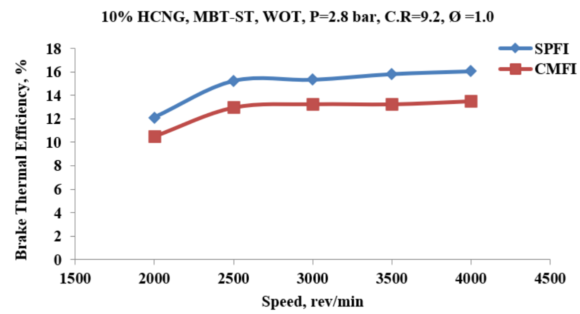

Figure 14. Thermal efficiency of the brake vs. engine speed

The graph 14 presents the dependence of the brake thermal efficiency (BTE) on the speed of the engine. BTE rises to 3500 rpm because of enhanced turbulence and combustion, and the effect decreases at high speeds since the combustion period becomes shorter. Unlike CMFI, SPF is more thermally efficient due to the correct timing of the injection and better homogeneity of the mixture. At 4000 rpm, SPFI attains 16.06 % efficiency and CMFI only reaches 13.51, almost a 16 % cut down. The enhanced atomization and the regulated injection in SPFI lead to enhanced energy conversion.

4.3. Volumetric efficiency

The volumetric efficiency varies with the diameter of the ring and is influenced by the flow rate through the pump and the gas pressure levels. Volumetric Efficiency Change The volumetric efficiency depends on the diameter of the ring, and it is decided by the rate of flow in the pump and the level of gas pressure.

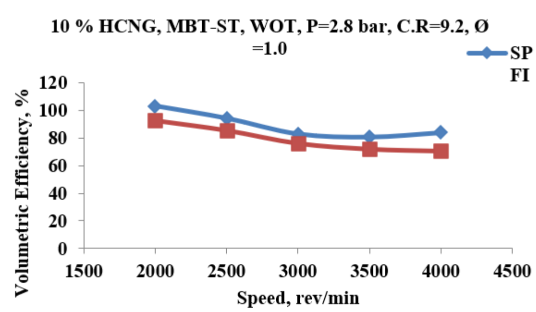

Figure 15. Engine speed vs. volumetric efficiency

Figure 15 shows a trend of volumetric efficiency versus engine speed. Rise of speed results in low volumetric efficiency because of a decrease in intake time and an increase in in-cylinder temperature. This is more in gaseous fuels due to their low density. The volumetric efficiency of SPFI is relatively higher since injection is distributed at 2.8 bar while with CMFI, there is a reduced charge filling because of single-point injection and natural aspiration. A reduction in the volumetric efficiency by about 20 percent at 4000 rpm is caused by short suction timing and the increased temperature of the cylinder.

4.4. Carbon Monoxide emissions

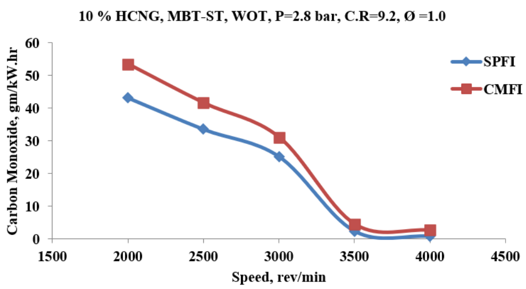

Figure 16. Engine speed and Carbon Monoxide emissions

Figure 16 presents the CO emissions at different speeds of the engine. Low and high speeds have higher CO formation because of incomplete combustion caused by poor mixing of air and fuel or lack of adequate time of combustion. Hydrogen enrichment has a lower emission of CO than pure CNG and gasoline due to enhanced flame speed and oxidation properties. The amount of gas emitted as CO is also reduced a lot with SPFI over CMFI; at 2000 rpm, CO emission is reduced to 43.17 g/kW^{-1}h (SPFI) as compared to 53.41 g/kW^{-1}h (CMFI) which is approximately 19% lower.

4.5. Hydrocarbon emissions

Figure 17 shows the trends of hydrocarbon (HC) emission. HC emissions increase at high speeds because of short combustion period and potential loss of charge during the valve overlap. Added hydrogen leads to a higher degree of combustion, and hence, less unburned hydrocarbons. The emission of HC is cut by more than 50 percent in SPFIs than in CMFI due to the even distribution of the charges and enhanced combustion stability.

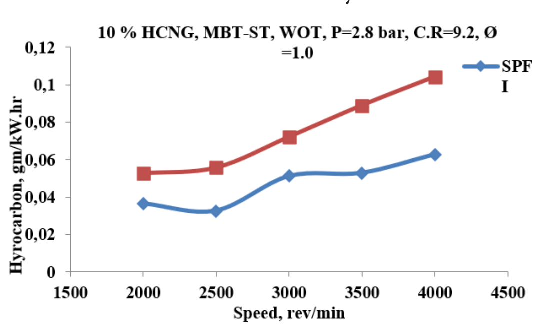

Figure 17. Engine speed versus Hydrocarbon emissions

4.6. Nitrogen Oxides emissions

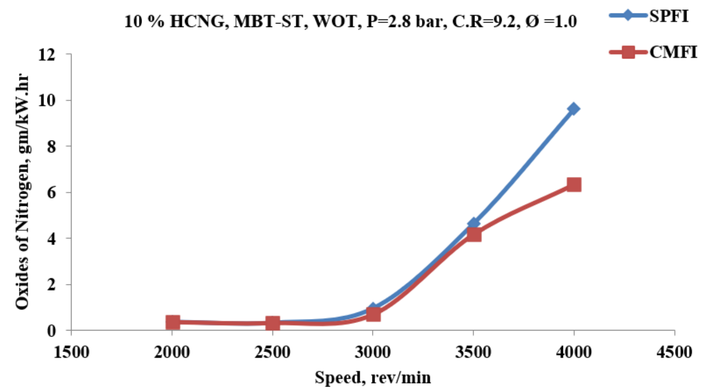

Figure 18. Engine speed and Oxides of Nitrogen emissions

Figure 18 shows the behavior of NOx at different engine rates and at different rates of EGR. Hydrogen raises the temperature of combustion and promptly releases heat, which is more likely to raise the amount of NOx. Cooled EGR is however able to reduce NOx by decreasing peak combustion temperature due to dilution of recirculated gases and increased specific heat. NOx is greatly suppressed by increasing EGR rate, at the expense of power and efficiency. Under best EGR conditions at 4000 rpm, the amount of NOx declines to 6.3 g/kW hence ranges between 9.6 g/kW and 6.3 g/kW as the EGR.

4.7. Exhaust gas temperature

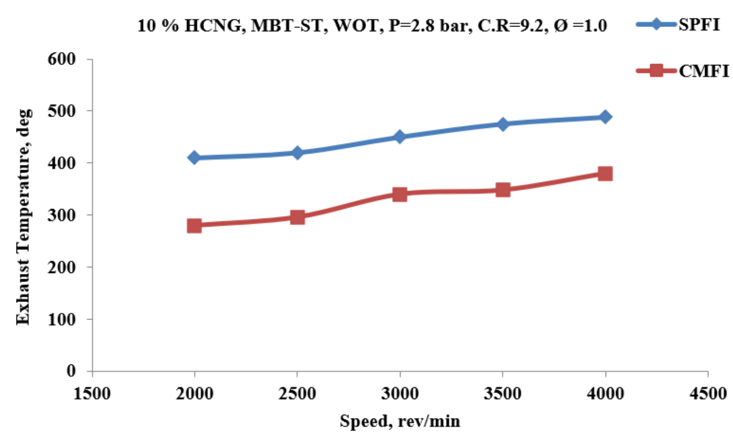

Figure 19. Engine speed versus exhaust temperature

Figure 19 illustrates the variation of temperature at exhaust with the engine speed. The speed raises the exhaust temperature in relation to more fuel burning and more combustion intensity promoted by the addition of hydrogen. The highest temperature of 488c is recorded in SPFI at 4000 rpm and CMFI at 380c being a 22 percent reduction. Reduced exhaust temperature in CMFI is associated with decreased charge intake and a relatively low intensity of combustion.

4.8. Brake mean effective pressure

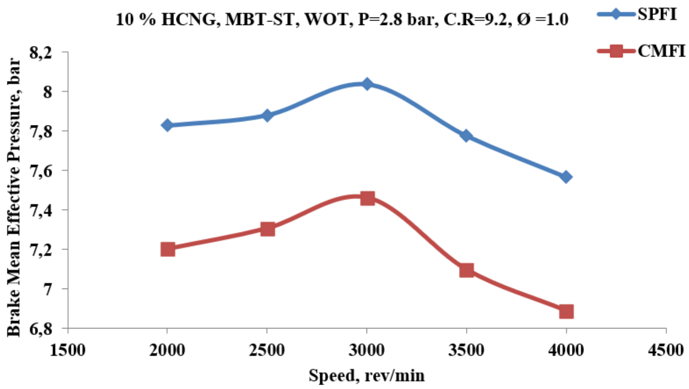

Figure 20. Speed vs brake mean effective pressure

Figure 20 uses BMEP trends of SPFI and CMFI comparatively. BMEP is at its highest point at mid-range speeds and low at extreme speeds as the combustion is not completed. During WOT at 3000 rpm, SPFI reaches 8.2 bar which is estimated at 10 percent more than CMFI. Better mixture preparation and homogeneity of combustion in SPFI leads to better mean effective pressure.

4.9. Brake specific energy consumption

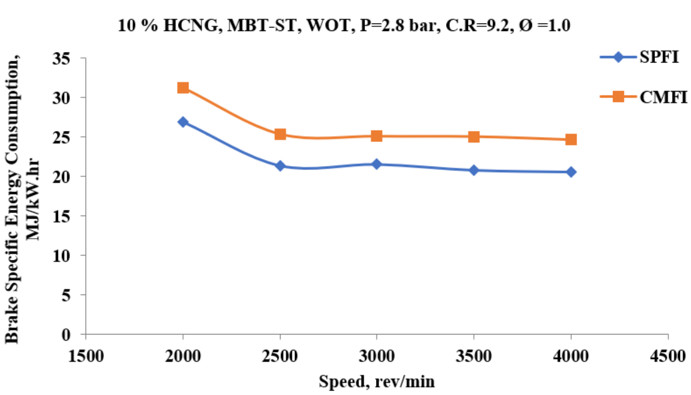

Figure 21. Speed vs brake specific energy consumption

Figure 21 shows the variation of BSEC with speed. Low speed operation results in high BSEC due to the incomplete combustion and lack of the supply of oxygen. The process of hydrogen addition causes the BSEC decrease in all the speeds through the increase of flame propagation and the decrease in the duration of combustion. At 4000 rpm, SPFI registers 20.60 MJ/kW•hr as compared to CMFI which represents about 20 percent more consumption because it has poor distribution of mixtures.

4.10. Brake specific fuel consumption

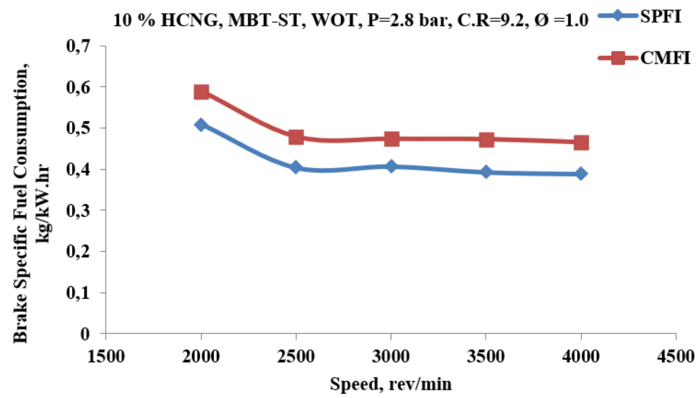

Figure 22. Speed vs. brake specific fuel consumption

Figure 22 shows that BSFC reduces with the rise in speed because of enhanced combustion and turbulence. The minimal BSFC value of 0.4 kg/kW •hr is found with SPFI at 2500 rpm. The SPFI offers more fuel economy since it has precise injection timing and enhanced mixing of fuel and air. An increase in BSFC with low speed is caused by a decrease in turbulence and complete combustion.

4.11. Cylinder Maximum Pressure

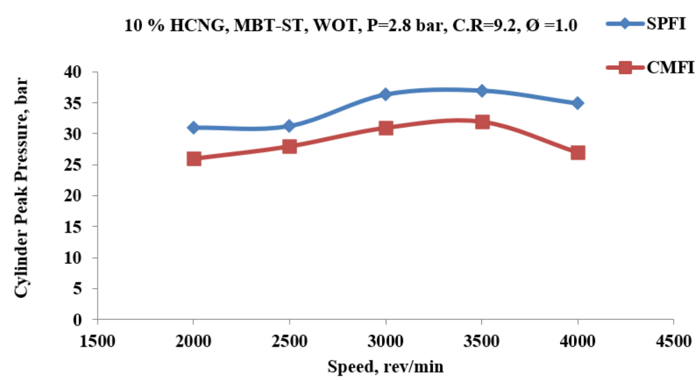

Figure 23. Speed vs. Cylinder peak pressure

The change of peak in-cylinder pressure is shown in figure 23. Addition of hydrogen raises the temperature of combustion, and raises the lean combustion ability, resulting in higher peak pressure. The highest pressure in SPF is 34.95 bar at 3200 rpm compared to CMFI which is 27 bars meaning that the ratio of uneven charge distribution and slower combustion in CMFI reduces by about 23 percent.

4.12. Combustion analysis

4.12.1. Combustion duration

Combustion analysis shows that the inclusion of hydrogen to CNG greatly increases the rate of combustion. Hydrogen enrichment also increases the spread of the flame and promotes more stable combustion especially in case the right mixture of hydrogen fraction and spark timing is sustained. The lower the concentration of the hydrogen, the more the incomplete combustion losses will decrease because of finding more time to oxidize the fuel-air mixture. At approximately 10 percent hydrogen mixture, the efficiency of combustion increases even more, leading to some partial combustion losses in the operation of pure CNG. With increased speed of combustion, real combustion losses are minimized as they are normally due to combustion taking finite time and not taking place instantly at top dead center (TDC). The rising concentration of hydrogen will also have the side effect of rising the heat transfer losses to the cylinder walls since the higher the combustion temperatures and the faster the flame moves, the nearer the flame front will be to the combustion chamber walls. Hydrogen also decreases the quenching distance of the flame that enables the flame to extend closer to the walls. Although the best design of the combustion chamber can improve thermal efficiency, it can also add heat to the walls. It is possible to counter such losses through the adjustment of engine parameters like compression ratio to ensure an efficient performance of combustion process.

4.12.2. Flame development duration

Flame development duration is the measurement of the crank angle between the spark ignition and the point at which some 10 % of the total charge in the cylinder has been used up. Hydrogen included in the HCNG mixture accelerates the first flame propagation rate because of an elevated laminar flame velocity and low ignition energy. As a result, the first phase of combustion takes a shorter period, and the time of flame development is shortened. The increased rate of starting the ignition and the first growth of the flame enhances better stability in combustion and more effective energy release in the cylinders.

4.12.3. Mass fraction burn

Mass Fraction Burn (MFB) is a term used to explain how combustion occurs and in general for example the 5 percent, 50 percent and 90 percent crank angle needed to burn the given portions of the in-cylinder charge. In the 10% HCNG mixture, the first combustion phase modelled by MFB5 is much reduced with over 30 percent reduction in duration than when pure CNG is used. Even though further addition of hydrogen concentration past this point does not necessarily decrease the MFB5 even more than that in lower blends, it improves the first combustion stage. The effect of acceleration in later stages of combustion (MFB50 and MFB90) is not so high. A difference of approximately 7 percent decrease in the duration of combustion is seen at the main combustion phase (MFB50) when combustion is conducted using a 10 percent HCNG mixture. The overall speed of combustion, measured by MFB90, shows the same trend and suggests an increase in the overall speed of combustion. The more rapid flame propagation which comes with the enrichment of hydrogen also results in the cumulative mass fraction burned increasing more rapidly and, thus, enhances better combustion efficiency and limits the chances of incomplete combustion.

4.12.4. Maximum Brake Torque Spark Timing (MBT-ST)

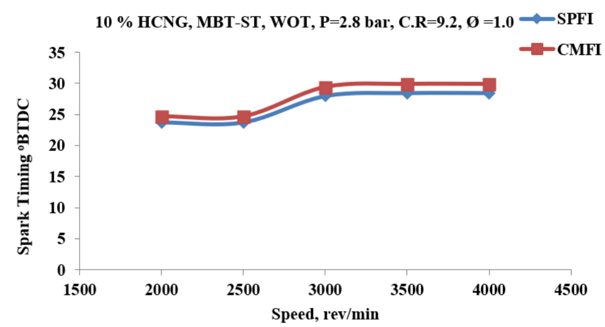

Figure 24. Speed vs. spark timing

Figure 24 shows how the spark advances to obtain Maximum Brake Torque (MBT) varies with engine speed in the case of HCNG blend. The faster the engine speed, the higher the spark timing should be to provide full combustion within the short crank angle period. Small advances in spark is normally necessary on pure CNG due to their partial slowness in flame propagation. The introduction of hydrogen, however, increases the rate of combustion and in the process, the spark advance demanded is low as compared to pure CNG operation. The increased flame rate of HCNG mixture will lead to enhanced power generation and larger thermal efficiency at the best ignition timing. In low engine speed, both SPFI and CMFI systems display a slightly retarded HCNG 10 per cent spark timing because of the faster rate at which the engine starts combustion.

4.12.5. Cylinder maximum pressure

Figure 25 shows the change of pressure within the cylinder in both SPFI and CMFI modes with the use of a 10 % HCNG mixture at 2500 rpm. The pressure trace bought by the data acquisition system shows that the pressure reaches greater peak cylinder pressure because of accelerated combustion and increased heat release attributed to enriching of hydrogen. The peak pressure of SPFI arrangement pecks at a slightly lower than the top dead center (TDC) at about 31.73 bar. Since hydrogen increases the rate at which the combustion process occurs, the peak pressure is displaced slightly away of TDC to approximately 2^0 cranks angle as shown in figure 26, Peak pressure is around 19^0 CA after TDC, which implies that combustion phasing is efficient. The pressure crank angle diagram shows also that the operation of the SPFI has been stable with no signs of knock or pre-ignition.

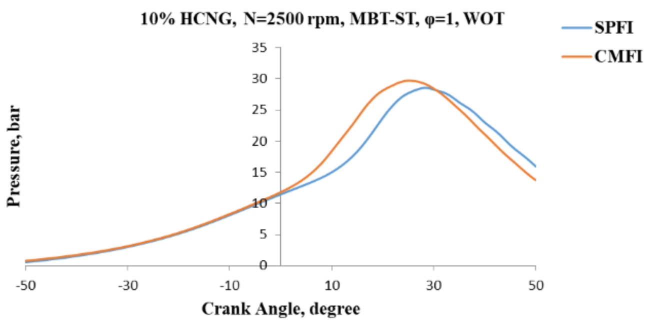

Figure 25. Crank angle vs. pressure

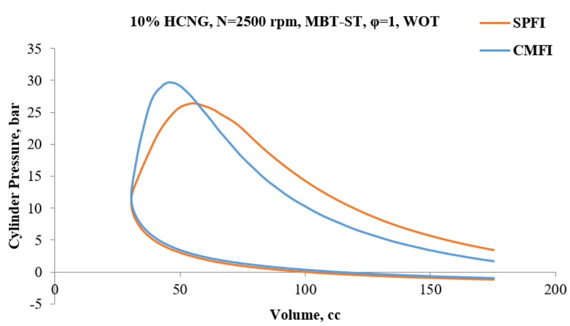

Figure 26. Crank angle vs. cylinder pressure

4.12.6. Heat release rate

Figures 27 and 28 show the change of the net heat release rate and the peak rate of the rise in pressure with crank angle in the case of HCNG blends. Heat release profile is the energy that is released in combustion in the cylinder. Fuels having lower flame propagation speed have delayed peak heat release and have low maximum heat release rates. Introduction of hydrogen enhances combustion intensity and progress in heat release because of its high laminar flame velocity and quick energy release nature. As a result, the HCNG mixture has a finer heat release profile and faster rate of pressure increase, suggesting more efficient combustion, as compared to the traditional gaseous fuel operation.

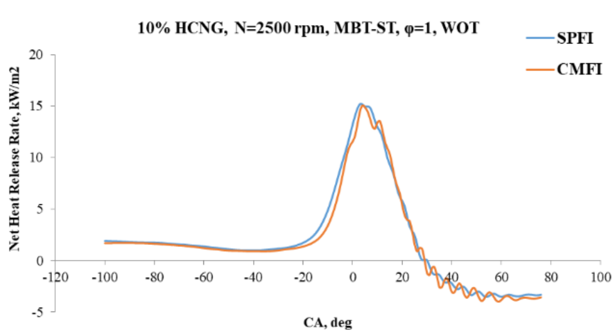

Figure 27. Crank angle vs. net heat release rate

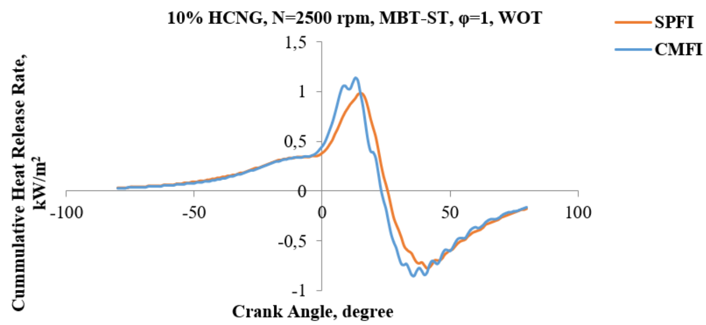

Figure 28. Crank angle vs. maximum rate of pressure rise

4.12.7. Cycle by cycle variation

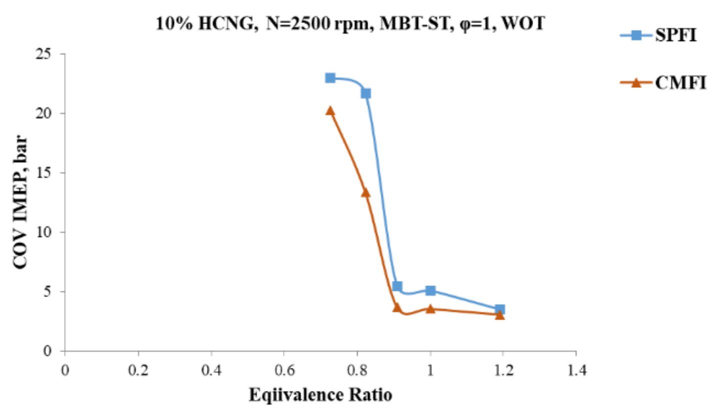

Figure 29. Speed vs. COV_{IMEP}

The coefficient of variation of the mean effective pressure reported (COVIMEP) indicating cycle-to-cycle stability in combustion is shown by Figure 29. Hydrogen addition enhances consistency of the combustion process due to faster rate of flames and better ignition properties. Engines that are powered by natural gas are often subjected to greater cyclic variability because of the slower combustion and lean burn restrictions. The hydrogen concentration within the HCNG mixture increases the rate of propagation of flames and minimizes the chances of misfire or incomplete combustion cycles. Consequently, the COVIMEP reduces as the hydrogen fraction and engine speed increase as the turbulence in the combustion chamber goes up. In the case of 10% HCNG blend that is under stoichiometric conditions, the COVIMEP stays constant at approximately 4 per cent, which means that the combustion is in a stable state.

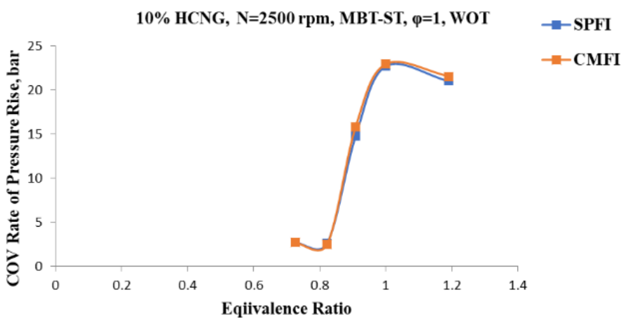

Figure 30. Speed vs. COV rate of pressure rise

Figure 30 is the plot of the coefficient of variation of the rate of pressure rise against engine speed. This parameter shows changes in intensity of combustion in one cycle to another. The findings show that enrichment of hydrogen tends to decrease cyclic anomalies by enhancing the rate of flame propagation and enhancing the success of the combustion process. But as the mixture becomes richer the rate of variable in the variability of pressure rises also increases slightly, because the rate of combustion is greater, and the rate of heat release is greater. Generally, the HCNG mix proves to be to be more stable in combustion than in normal gaseous fuel combustion.

4.13. Long term effects of HCNG blends on engine components and the fuel injection system

- The long run effects of the HCNG fuel blends on engine equipment and the injection facility are subject to the percent composition of the hydrogen in the mix, engine design and the engine working environment.
- Hydrogen presence enhances the velocity of flame propagation and the temperature of combustion which may place increased thermal loads on the parts of the engine like the pistons, cylinder walls, and valves. Unless these components are created to run in high-temperature settings, oxidation and accelerated wear of components can take place.
- At the point where there is a high ratio of hydrogen, there are possible chances that the probability of engine knocking can rise. Prolonged operation may gradually destroy pistons, piston rings, and so forth when such abnormal combustion takes place.
- HCNG mixtures with high hydrogen content, especially beyond circa 30 percent, might result in higher exhaust gas temperatures during combustion.
- High exhaust temperatures may hasten wear and tear on valve seats and exhaust valves as opposed to when running using traditional CNG. Thus, certain materials with better thermal resistance like hardened valve seats could be necessary to ensure good functioning.
- The increased hydrogen content mixtures tend to need more ignition energy to properly combust and this can increase electrode erosion in normal spark plugs. This means that spark plugs which are made of tougher elements like platinum or iridium electrodes can be desirable.
- Blends of hydrogen can also enhance the development of NOx and faster oxidation of the engine lubricating oil. Therefore, a higher frequency of the replacement of oil or lubricants with oxidation-resistant ones could be needed.
- In comparison to CNG, hydrogen does not offer any lubricating property within the fuel system that can elevate the mechanical wear in fuel injectors with time. To counter this injector elements can be coated with protective surface or materials resistant to wear.
- Conversely, HCNG combustion tends to have fewer carbon deposits making them cleaner injectors. When the fuel system is well calibrated, it will reduce the accumulation of carbon which will enhance durability of injectors.
- Hydrogen molecules are very small and therefore can permeate through some sealing materials. With the passage of time this can result in degradation of traditional rubber or polymer seals, and it will be needed to employ hydrogen-compatible materials like fluoroelastomer.
- Due to the low volumetric energy density of hydrogen in comparison to CNG, an increased volumetric rate of fuel flow could be necessary to sustain the same engine power. This can need changes in the injector design and fuel pressure controls to have the best engine performance.

4.14. Comparison of the fuel economy of CMFI and SPFI under identical conditions

Sequential Port Fuel Injection (SPFI) is much superior to engine performance as fuel is injected into one cylinder at a time when it is most suitable during the intake stroke. This is a narrow tolerance that regulates the quality of combustion and efficiency. However, when the engine speed is lower the brake specific consumption of fuel (BSFC) is expected to go up due to the less turbulence generated in the cylinder causing poor air fuel mixing hence higher HC and CO rates can be expected.

Comparing the Conventional Manifold Fuel Injection (CMFI) with the Sequential Port Fuel Injection (SPFI) in the same operating conditions, there is a clear difference in the fuel economy because of the differences in the injection timing, quality of atomization, and distribution of fuels amongst the cylinders.

Under the CMFI system, one injector placed on the intake manifold feeds the fuel to all the cylinders at once. As the fuel is supplied regardless of the personal events of intake of the cylinder, the distribution of the air-fuel mixture might not be even.

The CMFI configuration is less costly and easier to keep although in most cases it records less fuel efficiency. Since injection timing does not coincide with the intake stroke of each cylinder, not all the fuel may be involved in combustion. It may result in unequal distribution of mixtures with certain cylinders getting richer or leaner mixtures.

Besides this, fewer fuel droplets vaporize faster and therefore, the larger the droplet, the lower the combustion efficiency. As a result, CMFI engines have a fuel efficiency of about 5-15 per cent lower than sequential injections.

Conversely, SPFI systems use a system that uses specific injectors per cylinder, which allows the delivery of the fuel at the specific moment when the cylinder is under the intake process. This mode of injection helps reduce loss of fuels as well as ensuring that the mixture forms in each of the cylinders in a homogeneous way.

Very small fuel droplets generated in the SPFI systems enhance vaporization and efficiency in combustion. Additionally, correct electronic control of the injection enables improved fuel management when the engine is idle and at low-load operation resulting in low fuel consumption.

All in all, SPFI systems have better fuel economy than CMFI since they can regulate the amount of fuel provided to each cylinder based on its operating cycle. The SPFI type of engines enhances the quality of the mixture preparation and combustion efficiency, and they can offer better mileage and thus they are largely used in the current fuel-efficient cars.

4.15. Practical applications

- Hydrogen CNG blends have higher flame speed compared to traditional fuels. Sequential fuel injections allow precise air fuel mixture control, foremost to extra comprehensive combustion. Hydrogen addition enhances combustion and reduces carbon-based emission like CO_2, CO and unburned hydrocarbon. Higher calorific value of hydrogen improves thermal efficiency. Sequential injections ensure correct metering, enabling smooth operation under varying load conditions. It reduces greenhouse gas emissions due to inferior carbon content in HCNG associated with gasoline or pure CNG. HCNG can be stored and delivered using modified CNG systems. SPFI systems are compatible with standard internal combustion engines, reducing retrofitting costs. High octane rating of hydrogen gives higher compression ratio and better thermal efficiency with HCNG blends.
- Hydrogen storage needs lead pressure tanks or cryogenic arrangements, cumulative system complexity. Hydrogen's diffusivity can cause embrittlement in some materials, requiring specialized components. Precise calibration of SPFI system is critical to perfect the air-fuel mixture for HCNG blends, as hydrogen has a wider flammability range. Higher first investment in SPFI systems and hydrogen storage compared to conventional fuels. Following table 4 shows, Gaps and Attainment in present Research.

Table 4. Gaps and Attainment in present research

Sr. No.	Gap	Attainment
1	CNG fuel has slow burned velocity, and hence it is always burnt at stoichiometric conditions in IC engine. This increases NOx and reduces thermal efficiency.	In the present research CNG is blended with hydrogen to increase its flame speed. Hydrogen wider burn limit fast burn speed improves combustion steadiness and thermal efficiency.
2	Nox concentration increases with hydrogen addition if spark timing is not adjusted	• To reach this gap cooled EGR system is used at stoichiometric condition. • EGR reduces oxygen content and combustion temperature in engines which suppress Nox. MBT Spark timing advances with EGR and retard with hydrogen addition. Carbon based emission reduced with hydrogen enrichment.
3	CNG engines are more efficient at MBT spark timing.	MBT spark timing showed improvement in performance, emission and combustion characteristics for different blends of HCNG.
4	HCNG Lean air fuel ratio improves BTE and reduces Nox	Lean HCNG blends show lower Nox and increase in power compared to pure CNG. This helps to burn CNG at lean conditions with increase in brake thermal efficiency of the engine.
5	It is seen that gaseous fuel has lower volumetric efficiency which reduces power output and increases chances of backfire.	To reach this gap SPFI increased power output and reduced chances of backfire by providing precise control in injection timing leading to better performance and emission.
6	High Octane number of CNG increases thermal efficiency due to increase in compression ratio.	In the present context the trials are performed on modified gasoline engines with CR=9.2:1. Dedicated CNG engines give higher power output with increase in thermal efficiency.
7	According to a literature review it is seen that 20% hydrogen by volume (i.e. 7% by energy) with CNG is best blend for better performance and emission.	In the present research it is seen that 10 % hydrogen by volume (i.e. 3.3 % by energy) is best for improvement in performance and emission characteristics. Online fuel blending helped to take different sets of reading for varying proportion of hydrogen and CNG.

4.15. Challenges of Nox formation from HCNG Blends and its environmental impact

Hydrogen higher flame temperature and faster combustion rate compared to CNG results in elevated in-cylinder temperature, leading to increased thermal Nox formation. Increase in hydrogen percentage in blends may give rise to increase in peak combustion temperature. Lean combustion reduces Nox emission due to lower combustion temperature. Under stoichiometric and lean conditions elevated temperature of hydrogen blends promote to thermal Nox through Zeldovich mechanism. Hydrogen's wide flammability range allows for leaner air-fuel mixture operation. At lean conditions CO and HC emissions reduce, which results in higher Nox formation due to higher in-cylinder temperature. Accomplishing the perfect balance between lean operation and emissions is complicated. Too lean mixture can cause misfires, while slightly rich blends can surge Nox and HC emissions. Hydrogen's diffusivity and higher combustion temperature can cause engine material wear or hot spots. Hot spots lead to localized Nox formation, requiring advanced design materials and engine cooling systems.

Hydrogen increases the rate of chemical reactions in combustion, intensifying Nox formation. Controlling Nox becomes difficult as HCNG blends exceed 20%. Nox reacts with volatile organic compounds in presence of sunlight to form ground-level ozone, a major part of smog. It reduces air quality, affecting visibility and public health, especially in urban areas. Nox combines water vapor in the atmosphere to form nitric acid called Acid rain. It damages ecosystems, corrodes buildings, and acidifies water bodies. Nox contributes indirectly to climate change by enhancing the greenhouse effect of other gases like ozone layers. This worsens global warming and its associated environmental impacts. Nox exposure leads to health hazards respiratory issues, including asthma, bronchitis and lung cancer. It particularly affects vulnerable groups such as children, the elderly and those with pre-existing conditions. Nox deposition can lead to ecosystem imbalance by nutrient imbalance in soils and water bodies causing problems like eutrophication. Excess nutrients in aquatic system led to algal blooms, depleting oxygen and harming aquatic life. Nox contributes to ozone layer depletion. This increase in UV radiation reaching the earth leads to skin cancer and other harmful effects.

To mitigate these challenges and environmental effects, several approaches can be considered:

1. Employing technologies like Exhaust Gas Recirculation and Variable Valve Timing to control peak combustion temperatures.
2. Precise air-fuel ratio management and injection timing to reduce Nox formation.
3. Implementing SCR or LNTs to effectively reduce Nox in the exhaust stream.
4. Limiting the hydrogen fraction to a best range 10 to 15% to balance performance and emissions and combustion characteristics.
5. Enforcing policy and regulations of stringent emission norms to drive innovation in clean technologies.

Exhaust Gas Recirculation and optimized ignition timing can mitigate Nox formation. With higher hydrogen blends (>30%) and sequential Injection system helps in reducing Nox with Lean combustion. HCNG engines can be run in ultra-lean conditions due to hydrogen's wider flammability range. Optimized CNG engine for HCNG blends helps to reduce Nox. By diluting the intake air charge with inert exhaust gases (EGR) reduces combustion chamber temperature and Nox. With stoichiometric combustion (λ=1) Nox emissions are higher but can be reduced by using catalytic convertor. Advanced technologies like dual fuel injection systems and pre-chamber combustion design help to reduce Nox. 10 to 20% by volume of HCNG is often a good compromise for best power and Nox reduction.

5. Conclusions and future scope

5.1. Conclusion

- The current research compares the overall performance of both the Sequential Port Fuel Injection (SPFI) and Conventional Manifold Fuel Injection (CMFI) systems in a multi-cylinder spark ignition engine using hydrogen-incorporated compressed natural gas (10% HCNG) blends. The findings show that SPFI offers enhanced fuel delivery, depending on the engine load, and speed resulting in more correct preparation of the air-fuel mixture and higher efficiency of combustion. This enhanced combustion characteristics can be seen mostly at higher engine speeds. Consequently, SPFI produces low carbon monoxide (CO) and hydrocarbon (HC) emissions than CMFI because the injection timing is more correct and the mixture homogeneity is enhanced.
- But enhanced combustion properties in SPFI also lead to increase in combustion temperature and lead to increased NOx emissions when compared to the CMFI operation. Also, enhanced calorific content and speed of combustion of the hydrogen-enriched fuel results in high exhaust gas temperature in SPFI mode. High temperatures of exhaust can affect the long-term durability of the engine as it raises the probability of wear or damaging of the valve unless well controlled. Compared to CMFI, SPFI shows lower brake specific fuel consumption (BSFC) due to its excellent fuel atomization and better combustion process compared to CMFI that has higher fuel consumption due to incomplete combustion and uneven mixture distribution. The SPFI design also results in greater peak cylinder pressure than CMFI that is linked with greater exhaustion of combustion and release of energy in the cylinder.
- HCNG blends added added value to combustion properties, by cutting down on the time of combustion, as well as enhancing flame speed. As the hydrogen content, peak cylinder pressure, heat release rate and rate of pressure

rise, so does the required maximum brake torque (MBT) spark timing as the flames develop more rapidly. In general, the findings also suggest that the SPFI systems have better performance and emission parameters than CMFI systems because of the control of the fuel injection and mixture formation. However, the rise in NOx emissions and exhaust temperatures, which are experienced during SPFI operation, still need to be optimized further using the advanced methods of combustion control and emission cutting strategies. The combination of HCNG and SPFI systems proves that there is considerable potential to enhance engine efficiency, as well as to reduce carbon-based emissions.

5.2. Future scope

- The research on the HCNG engines can also be extended in the future through investigating more parameters and new technologies.
- The use of experimental studies with direct injection of HCNG blends can offer more information regarding the efficiency of combustion, nature of emissions and performance optimization.
- Studies of the variable compression ratio engines with HCNG blend could also demonstrate the effect of the compression ratio on combustion processes and thermal efficiency.
- Further, multi-cylinder spark ignition engine is a type of engine where numerical simulation and modeling with the usage of HCNG blends would be beneficial in predicting the combustion properties, injection-strategies, and engine design to improve performance and decreased emissions.

Conflicts of interest

The authors declare no conflict of interest.

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Figures 1, 2, 6, 7 are taken from actual experimental set up. Figures 3, 4 and 5 (Conventional Manifold Fuel Injection System, Sequential Port Fuel Injection System and Experimental Setup, respectively) are reproduced from the authors' own previously published work. These data are from the experimental work carried out by the authors and are presented here with permission. Original sources are cited and diagrams have been changed to be the experimental setups in the present study.

Author contributions statement

Vishal Meshram: Conceptualization, Methodology, Investigation, Formal Analysis, Writing – Original Draft, Visualization, Data Curation. Pravin Nitnaware: Conceptualization, Methodology, Validation, Resources, Writing – Review & Editing, Supervision, Project Administration. Ravikant Nanwatkar: Conceptualization, Methodology, Software, Formal Analysis, Investigation, Writing – Original Draft, Visualization, Data Curation. Shyamsing Thakur: Investigation, Formal Analysis, Writing – Review & Editing, Visualization, Data Curation. Sandeep Sarnobat: Validation, Resources, Writing – Review & Editing, Supervision, Project Administration. Tapobrata Dey: Conceptualization, Methodology, Software, Validation, Investigation, Writing – Review & Editing, Supervision.

Declaration of interests

The authors have no competing financial interests or personal relationships that would appear to bias the interpretation of this work reported in this manuscript. This manuscript has not received any specific financial support from any organization that might have influenced the results or interpretation presented. All authors have read and approved the final manuscript and agree to publish.

Data availability

The raw data obtained in this study is available on reasonable request from the corresponding author. Data is not available for public review or presentation for privacy/ethical reasons. The authors have ensured that all the data or analyses produced or used in this research paper are included in this published paper and in its supplementary information files.

Declaration of generative AI and AI -assisted technologies in the writing process

The authors did not use any generative AI or AI-assisted tools to aid in writing during the preparation of this work. The analysis, interpretation and drafting of all the content has been carried out by the authors only and without the support of any AI tools. The authors assert that the content of this manuscript is original, accurate and complete.

Nomenclature

θ	Crankshaft angle
φ	Equivalence ratio
m	Mass flow rate
LHV	Lower heating value
SOI	Start of injection
MBT	Maximum brake torque timing
BMEP	Brake mean effective pressure
ITE	Indicated thermal efficiency
CNG	Compressed natural gas
H_2	Hydrogen
SPFI	Sequential port fuel injection
GDI	Gasoline direct injection
CO	Carbon monoxide
THC	Total hydrocarbons
NO_x	Oxides of nitrogen

TWC	Three-way catalyst
SCR	Selective catalytic reduction
RPM	Engine speed (rev min^{-1})
λ	Excess air ratio
E	Energy
MAP	Manifold absolute pressure
WOT	Wide-open throttle
BSFC	Brake-specific fuel consumption
IMEP	Indicated mean effective pressure
COV	Coefficient of variation
HCNG	Hydrogen-enriched compressed natural gas
PFI	Port fuel injection
CMFI	Conventional manifold fuel injection
SI	Spark ignition
HC	Unburned hydrocarbons
NO	Nitric oxide
EGR	Exhaust gas recirculation
LNT	Lean NO_x trap
GHG	Greenhouse gases

References

[1] Adrian Irimescu, Fuel conversion efficiency of a port injection engine fueled with gasoline–isobutanol blends, Energy, Volume 36, Issue 5, 2011, Pages 3030-3035, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2011.02.047

[2] Muhammad Imran Khan, Tabassam Yasmeen, Muhammad Ijaz Khan, Muhammad Farooq, Muhammad Wakeel, Research progress in the development of natural gas as fuel for road vehicles: A bibliographic review (1991–2016), Renewable and Sustainable Energy Reviews, Volume 66, 2016, Pages 702-741, ISSN 1364-0321, https://doi.org/10.1016/j.rser.2016.08.041.

[3] Miguel R. Oliveira Panão, António L.N. Moreira, Diamantino F.G. Durão, Effect of a cross-flow on spray impingement with port fuel injection systems for HCCI engines, Fuel, Volume 106, 2013, Pages 249-257, ISSN 0016-2361, https://doi.org/10.1016/j.fuel.2012.11.039.

[4] Gang Lv, Chong–lin Song, Suo–zhu Pan, Jun–hua Gao, Xiao–feng Cao, Comparison of number, surface area and volume distributions of particles emitted from a multipoint port fuel injection car and a gasoline direct injection car, Atmospheric Pollution Research, Volume 5, Issue 4, 2014, Pages 753-758, ISSN 1309-1042, http://dx.doi.org/10.5094/APR.2014.084.

[5] Yuan Zhuang, Guang Hong, Effects of direct injection timing of ethanol fuel on engine knock and lean burn in a port injection gasoline engine, Fuel, Volume 135, 2014, Pages 27-37, ISSN 0016-2361, https://doi.org/10.1016/j.fuel.2014.06.028.

[6] Bo Yang, Chengxun Xi, Xing Wei, Ke Zeng, Ming-Chia Lai, Parametric investigation of natural gas port injection and diesel pilot injection on the combustion and emissions of a turbocharged common rail dual-fuel engine at low load, Applied Energy, Volume 143, 2015, Pages 130-137, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2015.01.037.

[7] Yuhan Huang, Guang Hong, Ronghua Huang, Numerical investigation to the dual-fuel spray combustion process in an ethanol direct injection plus gasoline port injection (EDI+GPI) engine, Energy Conversion and Management, Volume 92, 2015, Pages 275-286, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2014.12.064

[8] Catapano, S. Di Iorio, A. Magno, P. Sementa, B.M. Vaglieco, A comprehensive analysis of the effect of ethanol, methane and methane-hydrogen blend on the combustion process in a PFI (port fuel injection) engine, Energy, Volume 88, 2015, Pages 101-110, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2015.02.051

[9] Peng Geng, Hui Zhang, Shichun Yang, Experimental investigation on the combustion and particulate matter (PM) emissions from a port-fuel injection (PFI) gasoline engine fueled with methanol–ultralow sulfur gasoline blends, Fuel, Volume 145, 2015, Pages 221-227, ISSN 0016-2361, https://doi.org/10.1016/j.fuel.2014.12.067.

[10] Yong Qian, Qiyan Zhou, Xiaole Wang, Lifeng Zhu, Xingcai Lu, Enabling dual fuel sequential combustion using port fuel injection of high reactivity fuel combined with direct injection of low reactivity fuels, Applied Thermal Engineering, Volume 103, 2016, Pages 399-410, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2016.04.122.

[11] Yuhan Huang, Guang Hong, Investigation of the effect of heated ethanol fuel on combustion and emissions of an ethanol direct injection plus gasoline port injection (EDI+GPI) engine, Energy Conversion and Management, Volume 123, 2016, Pages 338-347, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2016.06.047.

[12] Rencheng Zhu, Jingnan Hu, Xiaofeng Bao, Liqiang He, Yitu Lai, Lei Zu, Yufei Li, Sheng Su, Tailpipe emissions from gasoline direct injection (GDI) and port fuel injection (PFI) vehicles at both low and high ambient temperatures, Environmental Pollution, Volume 216, 2016, Pages 223-234, ISSN 0269-7491, https://doi.org/10.1016/j.envpol.2016.05.066

[13] Xie Cheng, Sun Baigang, Han Zhen, Investigation on Jet Characteristics of Hydrogen Injection and Injection Strategy for Backfire Control in a Port Fuel Injection Hydrogen Engine, Energy Procedia, Volume 105, 2017, Pages 1588-1599, ISSN 1876-6102, https://doi.org/10.1016/j.egypro.2017.03.508.

[14] Liqiang He, Jingnan Hu, Shaojun Zhang, Ye Wu, Rencheng Zhu, Lei Zu, Xiaofeng Bao, Yitu Lai, Sheng Su, The impact from the direct injection and multi-port fuel injection technologies for gasoline vehicles on solid particle number and black carbon emissions, Applied Energy, Volume 226, 2018, Pages 819-826, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2018.06.050,

[15] Yonggyu Lee, Seungmook Oh, Changup Kim, Junsun Lee, Kanghun Lee, Junghwan Kim, The dual-port fuel injection system for fuel economy improvement in an automotive spark-ignition gasoline engine, Applied Thermal Engineering, Volume 138, 2018, Pages 300-306, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2018.04.027.

[16] Mohamed Ali Jemni, Sahar Hadj Kassem, Zied Driss, Mohamed Salah Abid, Effects of hydrogen enrichment and injection location on in-cylinder flow characteristics, performance and emissions of gaseous LPG engine, Energy, Volume 150, 2018, Pages 92-108, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2018.02.120

[17] Dengquan Feng, Haiqiao Wei, Mingzhang Pan, Lei Zhou, Jianxiong Hua, Combustion performance of dual-injection using n-butanol direct-injection and gasoline port fuel-injection in a SI engine, Energy, Volume 160, 2018, Pages 573-581, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2018.07.042

[18] Seung Yeob Lee, Ho Jun Lee, Yong Tae Kang, Jin Taek Chung, Effects of injection strategies on the flow and fuel behavior characteristics in port dual injection engine, Energy, Volume 165, Part A, 2018, Pages 666-676, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2018.09.026

[19] Jeongwoo Lee, Cheolwoong Park, Jongwon Bae, Yongrae Kim, Sunyoup Lee, Changgi Kim, Comparison between gasoline direct injection and compressed natural gas port fuel injection under maximum load condition, Energy, Volume 197, 2020, 117173, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2020.117173.

[20] Farhad Salek, Meisam Babaie, Seyed Vahid Hosseini, O. Anwar Bég, Multi-objective optimization of the engine performance and emissions for a hydrogen/gasoline dual-fuel engine equipped with the port water injection system, International Journal of Hydrogen Energy, Volume 46, Issue 17, 2021, Pages 10535-10547, ISSN 0360-3199, https://doi.org/10.1016/j.ijhydene.2020.12.139.

[21] Seonyeob Kim, Cheolwoong Park, Hyungjoon Jang, Changgi Kim, Yongrae Kim, Effect of boosting on a performance and emissions in a port fuel injection natural gas engine with variable intake and exhaust valve timing, Energy Reports, Volume 7, 2021, Pages 4941-4950, ISSN 2352-4847, https://doi.org/10.1016/j.egyr.2021.07.073.

[22] Helena Líbalová, Táňa Závodná, Kristýna Vrbová, Jitka Sikorová, Michal Vojtíšek-Lom, Vít Beránek, Martin Pechout, Jiří Kléma, Miroslav Ciganek, Miroslav Machala, Jiří Neča, Pavel Rössner, Jan Topinka, Transcription profiles in BEAS-2B cells exposed to organic extracts from particulate emissions produced by a port-fuel injection vehicle, fueled with conventional fossil gasoline and gasoline-ethanol blend, Mutation Research/Genetic Toxicology and Environmental Mutagenesis, Volume 872, 2021, 503414, ISSN 1383-5718, https://doi.org/10.1016/j.mrgentox.2021.503414.

[23] Jadi Raghu Varma, Sai Kiran Reddy Katepalli, Mallampati Sreeja, Balappa Hadagali, Comprehensive studies on alcohol using port fuel injection facilitated with spark plug engine, Materials Today: Proceedings, Volume 45, Part 2, 2021, Pages 3219-3225, ISSN 2214-7853, https://doi.org/10.1016/j.matpr.2020.12.379.

[24] Hayri Yaman, Murat Kadir Yesilyurt, Samet Uslu, Simultaneous optimization of multiple engine parameters of a 1-heptanol / gasoline fuel blends operated a port-fuel injection spark-ignition engine using response surface methodology approach, Energy, Volume 238, Part C, 2022, 122019, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2021.122019.

[25] Molina, R. Novella, J. Gomez-Soriano, M. Olcina-Girona, Impact of medium-pressure direct injection in a spark-ignition engine fueled by hydrogen, Fuel, Volume 360, 2024, 130618, ISSN 0016-2361, https://doi.org/10.1016/j.fuel.2023.130618.

[26] Fabián Musy, Rafael Ortiz, Inmaculada Ortiz, Alfredo Ortiz, Hydrogen-fuelled internal combustion engines: Direct Injection versus Port-Fuel Injection, International Journal of Hydrogen Energy, 2024, ISSN 0360-3199, https://doi.org/10.1016/j.ijhydene.2024.07.136

[27] Saaidia, R., Ghriss, O., Köten, H., M Alquraish, M., Bouabidi, A., & El Haj Assad, M. (2024). Effects of intake manifold geometry in H2 & CNG fueled engine combustion. Journal of Thermal Engineering, 10(1), 153–163. https://doi:10.18186/thermal.1429746

[28] Koten, H. (2019). Effect of different levels of hydrogen + LPG addition on emissions and performance of a compression ignition engine. Journal of Thermal Engineering, 5(2), 58–69. https://doi:10.18186/thermal.531704

[29] Das L. M. Hydrogen-Oxygen reaction mechanism and its implication to hydrogen engine combustion. International Journal of Hydrogen Energy 1996; 21(8): 703-715.

[30] Karim G. A, Hydrogen as a spark ignition engine fuel. International Journal of Hydrogen Energy 2003; 28(5): 569-577.

[31] Lewis B, Von Elke G. Combustion flame and explosion of gases, New York 2nd Edn. Academic Press 1961.

[32] Module 1: Hydrogen properties. A report-college of Dessert, December 2001.