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Experimental investigation of hydrogen port fuel injection in DI diesel engine

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Abstract

During the last decade the use of alternative fuels for diesel engine has received renewed attention. The interdependence and uncertainty of petroleum based fuel availability and environmental issues, most notably air pollution, are among the principal forces behind the movement towards alternative source of energy. Hydrogen is expected to be one of the most important fuels in the near future for solving greenhouse problem for protecting environment and saving conventional fuels. In this experimental investigation, a diesel engine using hydrogen as fuel was investigated with diesel as an ignition source for hydrogen. Hydrogen was injected into the intake port, while diesel was injected directly inside the cylinder. The parameters such as injection timing and injection duration of hydrogen were varied for a wider range at a constant injection timing of 23° before injection top dead center (BITDC) for diesel. The hydrogen flow rate was kept constant at 10 lpm for varied load conditions. The maximum brake thermal efficiency of 29.4% was obtained at full load for the optimized injection timing at top dead center (TDC) with injection duration of 90° crank angle (CA). The oxides of nitrogen (NO_X) emission tends to reduce to a lower value of 705 ppm at full load condition for the optimized injection duration at TDC and with an injection timing of 60° CA compared to neat diesel fuel operation. The smoke emission reduces by three fold for the hydrogen operated engine at optimized conditions. Using port-injected hydrogen there is an increase in brake thermal efficiency of the engine with a greater reduction in emissions.

Keywords: Hydrogen; Injection duration; Injection timing; Performance; Emission; Combustion

1. Introduction

The main pollutants from the conventional hydrocarbon fuels are unburned/partially burned hydrocarbon (UBHC), carbon monoxide (CO), oxides of nitrogen (NO_X), Smoke and particulate matter. Regulations on exhaust emissions such as NO_X, hydrocarbon and carbon monoxide from engines are being tightened. It is very important to reduce exhaust emissions and to improve thermal efficiency. The higher thermal efficiency of diesel engines certainly has advantages for conserving energy and also solving the greenhouse problem. However, in a diesel engine there must be a trade-off between smoke and NO_X, since it is difficult to reduce both simultaneously. Government regulations have also stimulated research and development programs for alternative fuels and alternative fuel vehicle systems. Promising alternative sources of energy are alcohols, vegetable oil, compressed natural gas (CNG), liquefied petroleum gas (LPG), liquefied natural gas (LNG), producer gas, biogas and hydrogen. Among these fuels, Hydrogen is a long term renewable, recyclable and non-polluting fuel. Hydrogen has some peculiar features compared to hydrocarbon fuels, the most significant being the absence of carbon [1]. The burning velocity is so high that very rapid combustion can be achieved. The limits of flammability of hydrogen varies from an equivalence ratio (Ø) of 0.1 to 7.1, hence the engine can be operated with a wide range of air/fuel ratio. The properties of hydrogen are given in Table 1.

Some researchers tested diesel engines, using hydrogen as a sole fuel however; it was very difficult to operate a diesel engine with hydrogen just by increasing the compression ratio, due to its high self-ignition temperature. Therefore, glow plug or spark plug is often used [2]. Ikegami et al. [3] investigated the hydrogen combustion with a special injector that was equipped with a glow plug and they obtained moderate engine performance.

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Table 1
Properties of hydrogen

Sl. no.	Properties	Diesel	Unleaded gasoline	Hydrogen
	Formula	$C_nH_{1.8n}$ C_8–C_{20}	$C_nH_{1.87n}$ C_4–C_{12}	H_2
1	Auto-ignition Temperature (K)	530	533–733	858
2	Minimum ignition energy (mJ)	—	0.24	0.02
3	Flammability limits (volume % in air)	0.7–5	1.4–7.6	4–75
4	Stoichiometric air fuel ratio on mass basis	14.5	14.6	34.3
6	Limits of flammability (equivalence ratio)	—	0.7–3.8	0.1–7.1
7	Density at 16 °C and 1.01 bar (kg/m^3)	833–881	721–785	0.0838
8	Net heating valve (MJ/kg)	42.5	43.9	119.93
9	Flame velocity (cm/s)	30	37–43	265–325
10	Quenching gap in NTP air (cm)	—	0.2	0.064
11	Diffusivity in air (cm^2/s)	—	0.08	0.63
	Octane number			
12	Research	30	92–98	130
	motor	—	80–90	—
13	Cetane number	40–55	13–17	—

2. Hydrogen in internal combustion engines

Hydrogen is used as a sole fuel in spark ignition engines. Hydrogen powered S.I. engine has a comparable power output with gasoline, with higher efficiency [4]. The problems that are to be overcome in hydrogen operated S.I. engines are backfire, pre-ignition and NO_X emissions [5]. By proper control of hydrogen using electronic fuel injection or induction system these problems can be reduced.

The concept of using hydrogen as an alternative to diesel fuel in C.I. engines is a recent one. As the self-ignition temperature of hydrogen (858 K) is higher than diesel (453 K), hydrogen cannot be ignited by compression. Hence it requires the use of external ignition source like spark plug or a glow plug. One of the alternative methods is to use diesel as a pilot fuel or by using ignition improvers like DEE for ignition purpose [6]. The methods of using hydrogen in C.I. engines are hydrogen enrichment in air, hydrogen injection in the intake system and in-cylinder injection [7].

Numerous research works were carried out on dual fuel engines to run with alternative sources of energy with diesel as a pilot source of ignition. In the present research work hydrogen was used as a source of energy and diesel as a source of ignition [8]. The hydrogen flow rate was maintained constant at 10 lpm (liters per minute) to obtain the optimized injection timing and injection duration [9].

3. Test engine

The engine used for the experimental work was a single cylinder, water-cooled, four stroke, vertical, naturally aspirated, stationary, D.I. diesel engine developing a rated power of 3.76 kW at 1500 rpm having a compression ratio of 16.5:1. The combustion chamber employed was a hemispherical open bowl combustion chamber with a swirl ratio of 1.2. The nozzle is a four-hole nozzle having an injection pressure of 240 bar with 0.01 mm nozzle hole diameter. The diesel is injected at 23° before injection top dead center (ITDC). The engine was modified to operate with hydrogen by positioning the injector on the cylinder head. The specifications of the test engine are shown in Table 2.

4. Modifications for port injection

The head of the engine was fitted with the solenoid operated hydrogen gas injector. Hydrogen was injected into the intake port during the suction stroke of the engine. The cross section of the hydrogen injector is shown in Fig. 1. The injector was placed just above the intake valve at a distance of 13 mm from

Table 2
Engine specifications

S. no.	Parameters	Specifications
1	General details	Single cylinder, four stroke, compression ignition, constant speed, vertical, water cooled, direct injection
2	Bore	80 mm
3	Stroke	110 mm
4	Swept volume	553 cc
5	Clearance volume	36.87 cc
6	Compression ratio	16.5:1
7	Rated output	3.7 kW at 1500 rpm
8	Rated speed	1500 rpm
9	Injection pressure	240 bar

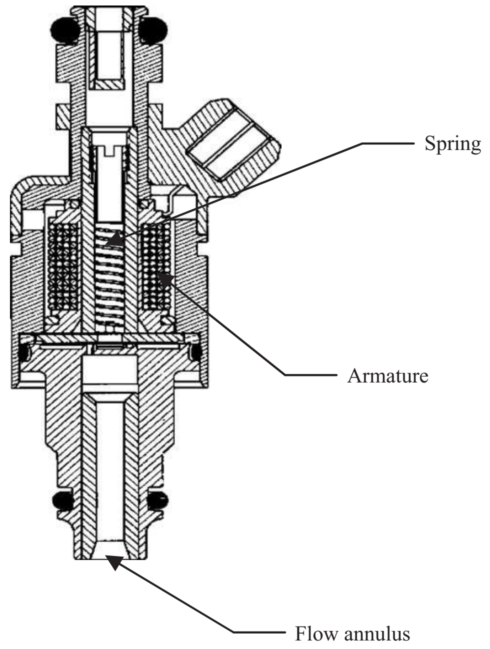

Fig. 1. Hydrogen gas injector.

the intake valve seating position. The inlet valve opening timing is 5° BITDC, which is taken as the start of injection timing for hydrogen. The electronic control unit (ECU) controls the start of injection from 5° BGTDC to 15° AGTDC and injection duration from 30° CA to 90° CA [10]. The specifications of hydrogen injector are given in Table 3. An infrared (IR) sensor is used to sense the position of the crankshaft. The start of the injection and the duration of the injector opening are controlled by using the ECU [11]. The schematic layout of the experimental set-up is shown in Fig. 2.

5. Experimental procedure

Hydrogen was stored in a high-pressure storage tank at a pressure of 130–150 bar reduced to 1–4 bar by using a double stage pressure regulator. The hydrogen from the pressure regulator was passed through a shutoff valve, which can be controlled and closed if any backfire results in the fuel pipeline. The hydrogen after passing through the shutoff valve was allowed to pass through the digital mass flow controller (DFC). The DFC precisely measures the flow rate of hydrogen in standard liters per minute (SLPM). Since the hydrogen supplied to the injector should be free from any impurities, a filtering device was provided in the flow line. The filtered hydrogen was allowed to pass through a flame trap. The flame trap acts as a non-return valve and also as a visible indicator for hydrogen flow. The hydrogen from the flame trap was then passed to the flame arrestor, which consists of wire mesh. The wire mesh will shut off the flame during the backfire condition. The flame arrestor also consists of bursting diaphragm, which punctures when the pressure inside the system exceeds 10–13 bar during backfire conditions and it also acts as a non-return valve (NRV) to prevent the reverse flow of hydrogen. The hydrogen from the flame trap was passed to the 2-way valve. One end of the 2-way valve was connected to the fuel injector and the other end was let into the atmosphere to remove the excess hydrogen present in the fuel line during engine shutoff time.

Table 3
Hydrogen fuel injector specifications

Make	Quantum technologies
Supply voltage	8–16 V
Peak current	4 A
Holding current	1 A
Flow capacity	0.8 g/s at 483–552 kPa
Working pressure	103–552 kPa

The ECU gets an input signal from the IR sensor. Based on the signal from IR sensor the ECU will process the signal depending on the requirement, which will have a control over the start of injection as well as the injection duration. An optical pickup sensor was used to give a voltage pulse exactly when the pointer interrupts the signal. This sensor consists of infrared diode and phototransistor [12]. A pointer was fixed on the disc, which was aligned on the crankshaft. When the engine runs the pointer interrupts the infrared signal delivered by phototransistor. The IR light reflected due to interference was observed by the infrared diode. The infrared diode gives a voltage signal to the ECU, which is an indication of the position

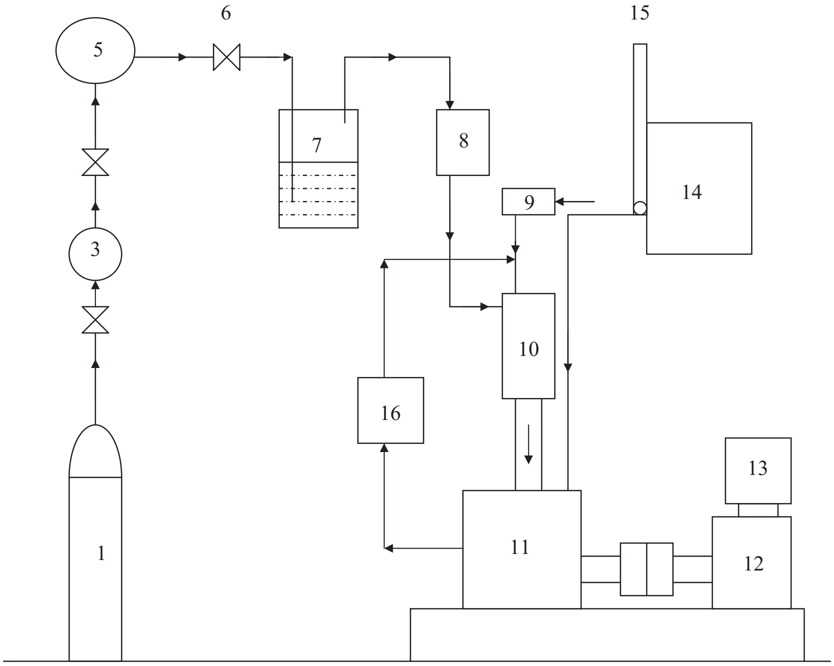

Fig. 2. Schematic diagram of experimental setup. 1 — Hydrogen cylinder; 2 — Pressure regulator; 3 — Hydrogen surge tank; 4 — Filter; 5 — Digital mass flow controller; 6 — PC to control DFC; 7 — Flame trap; 8 — Flame arrestor; 9 — Hydrogen injector; 10 — IR sensor; 11 — Electronic control unit; 12 — Engine; 13 — Dynamometer; 14 — Diesel tank.

of crankshaft. The pointer was fixed on the disc at a point of 10° BGTDC. The start of injection was varied in steps of 5° from 5° BGTDC. The IR detector was used to give the signal to the ECU for the injector opening. Based on the preset timing and duration the injector will be opened for injection and closed after injection. The injection timing and injection duration can be varied within the specified range with the help of ECU. Based on the presetting value the hydrogen flow will be taking place and the flow controlled by using the pressure regulator and also by using the digital mass flow controller.

For determining the best optimum timing and duration, the duration and start of injection were varied. Three injection durations (30° CA [3.3 ms], 60° CA [6.6 ms], 90° CA [9.9 ms]) were selected since the fuel injector can open for a maximum duration of 10 ms. The injector opening timing used were (−5° BTDC, TDC, 5° ATDC, 10° ATDC, 15° ATDC) [13]. Since the valve opening was 4.5 BTDC it was taken as the initial reading; the injection duration was varied by 5° CA till the optimum condition was attained. Due to the difficulties in representing the graph in three dimensions, as the number of legends is more and to have clear picture on readings, the graphs are drawn only for three injection durations of −5° BTDC, TDC and 5° ATDC. The hydrogen flow rate was set at 10-l/min for the entire range of operation and optimized timing and duration were obtained based on the performance and emissions [14]. From the results it was observed that the optimized timing was 5° ATDC with an injection duration of 90° crank angle [15].

6. Instrumentation

The power output of the test engine was measured by an electrical dynamometer. The power of the dynamometer was 10 kW with a current rating of 43.5 A. Carbon monoxide, carbon dioxide and unburned hydrocarbon emissions were measured by using a nondispersive infrared (NDIR) gas analyzer. NO_X emissions were measured by using an analyzer that works on the principle of electrochemical cell method.

Smoke emissions were measured using a Bosch type smoke meter. Gas samples were trapped on the filter paper and the filtered smoke was evaluated by using a photocell reflector, which gives a smoke value from 0 to 10. The engine cylinder pressure was measured with Kistler piezoelectric pressure sensor. The amplified signals were correlated with the signal from crank angle encoder and the data were stored on a personal computer for analysis. Hydrogen flow was measured by using a digital mass flow controller, which has the flow range of 0–100 lpm.

A hydrogen sensor was used to monitor the hydrogen gas in the environment and also used to sense any leak of hydrogen in the pipelines. The hydrogen sensor has a beeping circuit which comes into action and gives a single beep/double beep sound continuously. If the hydrogen in the atmosphere exceeds 100 ppm a single beep sound is produced, and double beep sound is produced continuously if it exceeds 1000 ppm. Any hydrogen leak from 1 to 2400 ppm can be sensed easily by using the hydrogen leak detector.

7. Estimation of uncertainity

Any experimental measurements, irrespective of the type of instrument used, possess a certain amount of uncertainty. The uncertainty in any measurement may be due to either fixed or random errors. As the fixed errors are repeatable in nature they can be easily accounted for to get the true value of measurement. However, random errors have to be estimated only analytically. The details of the estimated average uncertainty of some measured and calculated parameters at some typical operating conditions are given in Table 4. It can be observed that the uncertainty ranges from 0.6% to 3.1%.

8. Results and discussion

The performance and emission characteristics were studied with different injection timing of hydrogen and injection duration of hydrogen. The various timings and duration adopted are listed in Table 5. Based on the different injection timings and durations 10 parameters have been identified and measurements were taken at all the load conditions. For the purpose of comparison, tests were also conducted using diesel fuel No. 2. Table 6 shows the energy share ratio for hydrogen and diesel at different load conditions. The schematic view of the experimental view is shown in Fig. 2. The photographic view of the experimental setup and the hydrogen flow arrangements are shown in Figs. 3 and 4, respectively.

Table 4
Average uncertainties of some measured and calculated parameters

S. no.	Parameters	Uncertainity (%)
1	Speed	1.2
2	Temperature	0.6
3	Mass flow rate of air	1.3
4	Mass flow rate of diesel	2.1
5	Mass flow rate of hydrogen	1.2
6	Oxides of nitrogen	1.7
7	Hydrocarbon	2.2
8	Smoke	2.1
9	Particulate matter	3.1

Table 5
Start of injection and duration

Number	Start of injection	Injection duration
I (diesel)	23° BITDC	23°
II	5° BTDC	30°
III	TDC	30°
IV	5° ATDC	30°
V	5° BTDC	60°
VI	TDC	60°
VII	5° ATDC	90°
VIII	5° BTDC	90°
IX	TDC	90°
X	5° ATDC	90°

Table 6
Energy contributions by hydrogen and diesel under optimized conditions

Power output (kW)	Hydrogen energy (%)	Diesel energy (%)
0	38.6	61.4
1.06	24.2	75.8
1.87	19.3	80.7
2.97	14.4	85.6
3.74	12.9	87.1

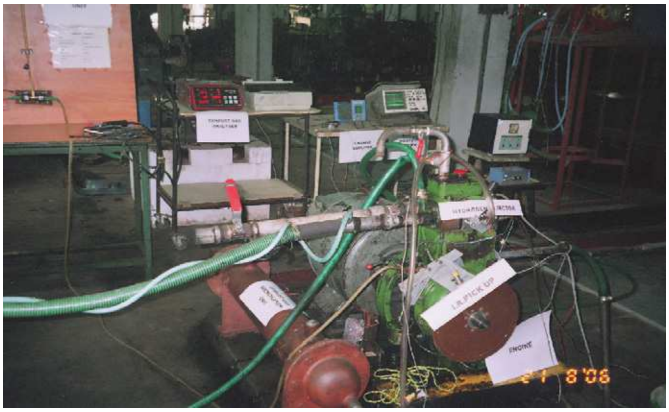

Fig. 3. Photographic view of the experimental setup.

Fig. 4. Photographic view of the hydrogen flow arrangement.

9. Specific energy consumption

The variation of specific energy consumption with load for various start of injection and injection duration is depicted in Fig. 5 . The observation of the figure reveals that the start of injection at 5° ATDC with injection duration of 90° CA is the most efficient one. For the injection timing of 15° ATDC with an injection duration of 60° CA the SEC is lower at full load compared to other operating conditions, but in this condition there is a high onset of knocking and smoke emissions are also on the higher side. Compared to neat diesel fuel operation the SEC decreases from 4.7 to 3.4 at full load for the injection timing of 5° ATDC with an injection duration of 30° CA. The reduction in SEC is due to the uniformity in hydrogen mixture formation with air resulting in better combustion and also hydrogen assists diesel during the combustion process.

10. Brake thermal efficiency

The variation of brake thermal efficiency with load for different operating conditions is presented in Fig. 6. The brake thermal efficiency increases from 23.6% to 29.4% for the injection timing of 5° ATDC with an injection duration of 90° CA compared to diesel. But the maximum efficiency obtained is 31.67% at 15° ATDC with 60° crank angle duration but knocking was observed at this condition. The increase in brake thermal efficiency is due to better mixing of hydrogen with air that results in enhanced combustion. Diesel combustion during the rapid burning stage initiates the hydrogen combustion and it was instantaneous making the delay period to reduce, which in turn increases the overall thermal efficiency of the engine.

11. NO_X emission

Fig. 7 depicts the variation of NO_X emissions at different loads, start of injection and injection duration. The NO_X emission is found to be minimum of 783 ppm for the injection timing of 5° ATDC with an injection duration of 60° CA compared to diesel of 1981 ppm at 75% load. The NO_X value for the neat diesel fuel operation at full load is 1806 ppm, whereas it reduces to 888 ppm in diesel fuel mode with an injection duration of 90° CA with 5° ATDC timing. The lowest NO_X of 705 ppm is obtained at full load with 60° CA while the injection commenced at TDC. The reduction is due to the operation of hydrogen engine at leaner equivalence ratios. At full load due to unstable combustion the NO_X emission is reduced.

12. Hydrocarbon

Fig. 8 depicts the variation of hydrocarbon emissions with load for different injection timings and injection durations. It can be observed that at no load the HC levels of hydrogen operated engine are lesser than diesel. At no load the HC is 3 ppm for the start of injection at 5° AGTDC with 90° injection duration for hydrogen operation compared to 19 ppm for diesel. At full load the hydrogen operated dual fuel engine results in an increase in HC compared to diesel. For hydrogen operation it is 7 ppm at an injection timing of 5° AGTDC with 90° injection duration compared to diesel which is 42 ppm. The reduction in HC is due to the higher burning velocity of hydrogen, which enhances the diesel burning at no load. The absence of carbon in hydrogen fuel also reduces the HC emissions to a greater extent.

13. Smoke

Observation of Fig. 9 that shows the variation of smoke emissions indicates that the smoke emission of hydrogen operated dual fuel engine is reduced significantly compared to diesel-operated engine. At 75% load condition the hydrogen-operated engine at an injection timing of 5° ATDC and with 90° injection duration the smoke value is 0.4 BSN whereas for diesel

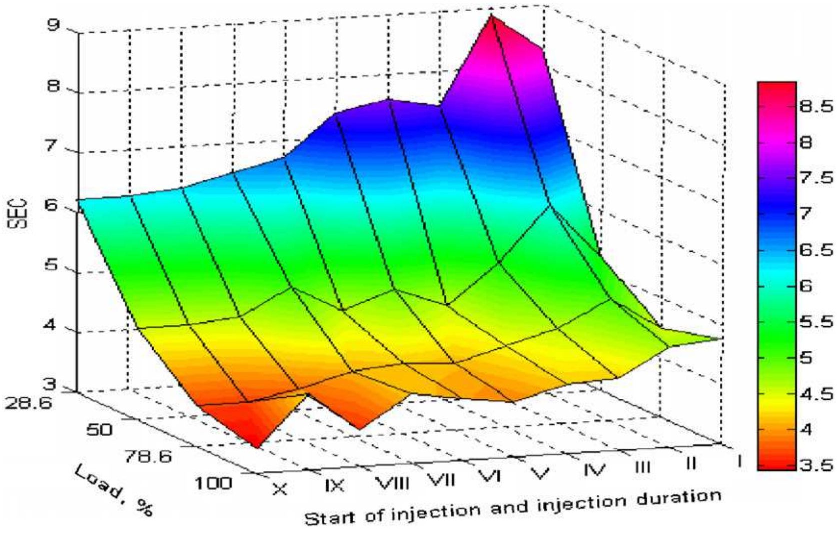

Fig. 5. Variation of specific energy consumption with load.

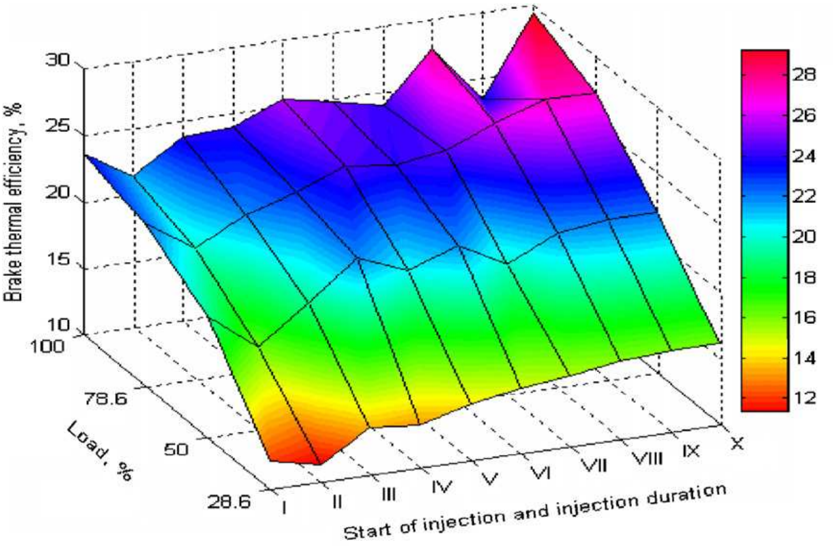

Fig. 6. Variation of brake thermal efficiency with load.

it is 2.0 BSN; the lowest smoke value of 0.3 BSN is observed at an injection timing of TDC with 60° injection duration. At full load for the start of injection at 5° ATDC with 90° injection duration smoke value of 0.3 BSN is noticed compared to diesel of 3.8 BSN at full load. The reduction in smoke is due to the absence of carbon in hydrogen and also hydrogen forms a homogeneous mixture during combustion rather than a heterogeneous mixture like diesel. Hydrogen assists diesel during the combustion process to have a greater reduction in the smoke value.

14. Carbon monoxide

Fig. 10 portrays the variation of carbon monoxide emissions with load for various start of injection and injection duration. It can be observed that at no load the CO levels of hydrogen-operated engine at all operating conditions is lesser than diesel. For the start of injection 5° ATDC with 90° injection duration for hydrogen operation there is no CO compared to 0.04 vol% for diesel at no load. At full load the hydrogen operated dual fuel engine results in an increase in CO compared to part load operations, the value of CO being 0.01 vol% for hydrogen operation at injection timing of 5° ATDC and 90° injection duration compared to that of diesel of 0.17 vol%. The reduction in CO in hydrogen-operated dual fuel engine is due to the absence of carbon in hydrogen fuel. At no load since the engine is operated at lean equivalence ratio, a reduction in CO is observed for hydrogen operation at no load. But at full load the oxygen concentration reduces significantly resulting in an increase in CO formation rate, which makes the overall CO concentration to increase compared to diesel.

15. Carbon dioxide

The variation of carbon dioxide with load is depicted in Fig. 11. The CO_2 values are lesser for hydrogen operated dual

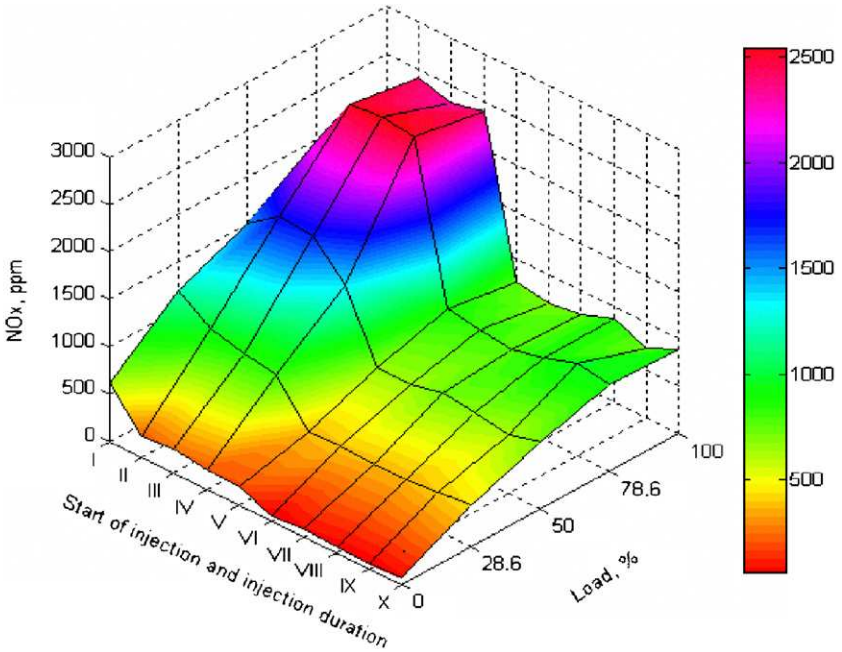

Fig. 7. Variation of NO_X with load.

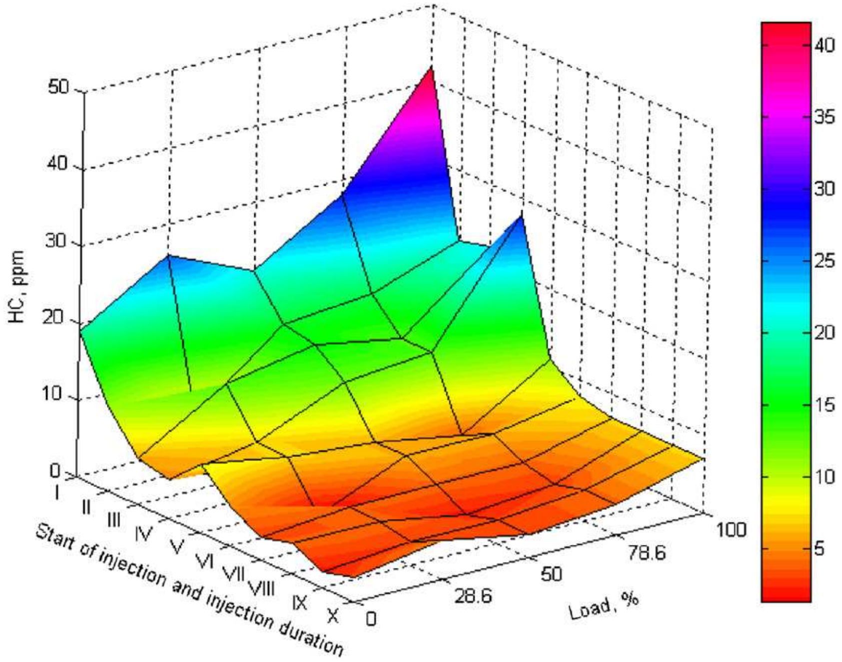

Fig. 8. Variation of hydrocarbon with load.

fuel engine compared to diesel-operated engine. The CO_2 for hydrogen with start of injection at 5° ATDC with 90° injection duration for hydrogen operation is 2.1 vol% compared to 9.5 vol% for diesel at 75% load. At full load for hydrogen with start of injection at 5° ATDC with 90° injection duration the CO_2 concentration is 2.4% compared to 11.6% for diesel. The reduction in CO_2 concentration is due to the absence of carbon in hydrogen fuel.

16. Peak pressure

The variation of peak pressure with load is shown in Fig. 12. At 75% load the peak pressure is 64.3 bar for the start of injection at 5° ATDC with 90° injection duration for hydrogen compared to diesel operation of 68.9 bar. At full load the hydrogen operated dual fuel engine results in an increase in peak pressure compared to diesel. The peak pressure for hydrogen

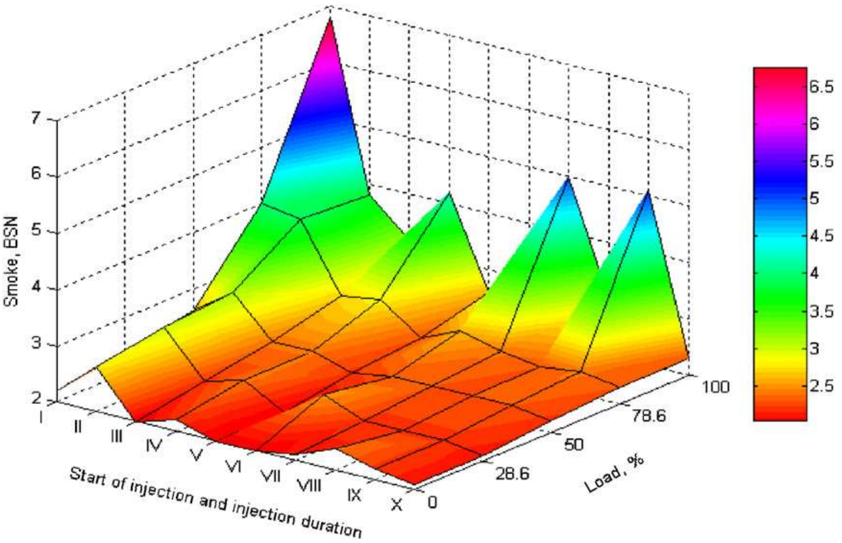

Fig. 9. Variation of smoke with load.

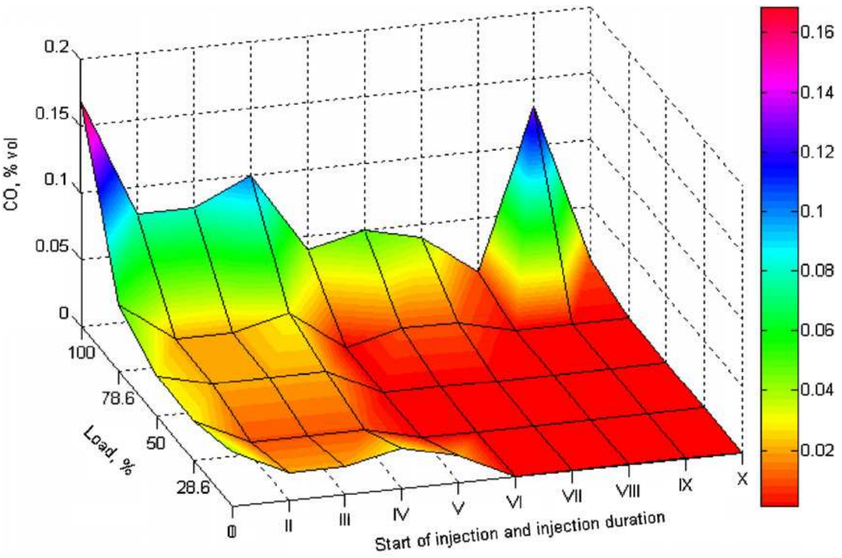

Fig. 10. Variation of CO with load.

operation is 71.7 bar at an injection timing of 5° ATDC and 90° injection duration compared to diesel of 73.7 bar. The increase in peak pressure is due to higher burning velocity of hydrogen, which makes combustion to be almost instantaneous, resulting in increased pressure.

17. Pressure crank angle diagram

Fig. 13 portrays the measured pressure data for hydrogen operated dual fuel engine at 5° ATDC, 90° CA and diesel operation at full load. The peak pressure obtained for hydrogen operation at full load is 82.7 bar whereas in the case of diesel it is 82.2 bar. The peak pressure for hydrogen-operated dual fuel engine is advanced by 5° compared to the peak pressure of diesel at full load. The advance in peak pressure for hydrogen combustion may be due to instantaneous combustion of hydrogen compared to diesel. The rate of pressure rise is also high for hydrogen operated dual fuel engine compared to dieseloperated engine due to the instantaneous combustion of hydrogen fuel.

18. Heat release rate

Fig. 14 shows the rate of heat release for hydrogen operated dual fuel engine at 5° ATDC, 90° CA and diesel engine at full load. The combustion starts at the same crank angle of about 10° BITDC for both the dual fuel and diesel operation. It can be observed from the figure that in the case of diesel operation there are three phases of combustion, namely premixed combustion phase, mixing controlled combustion phase and late

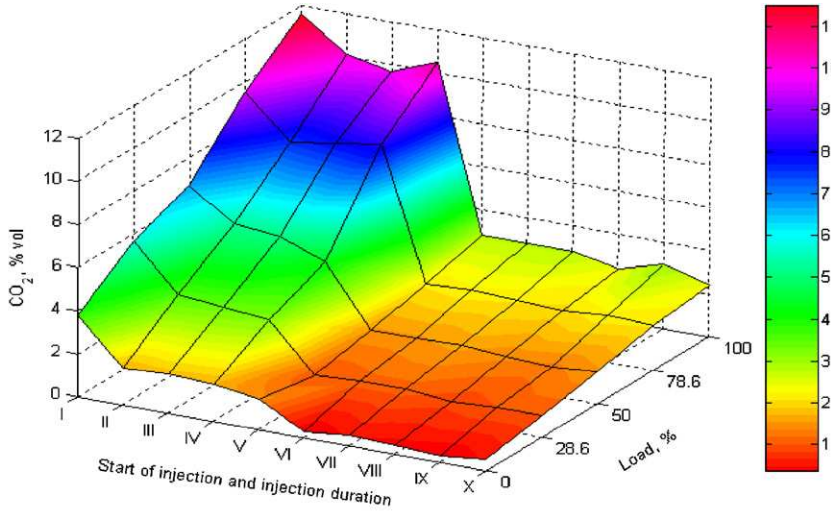

Fig. 11. Variation of CO_2 with load.

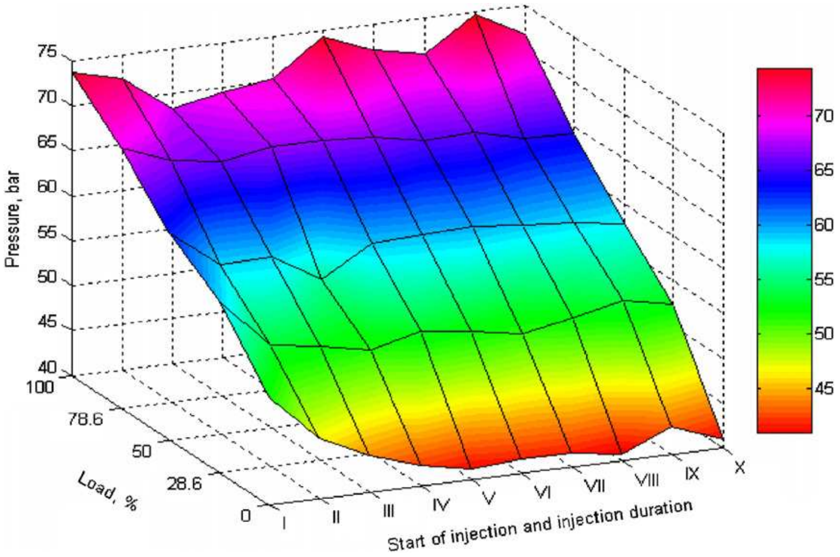

Fig. 12. Variation of peak pressure with load.

combustion phase. The heat release for hydrogen shows distinct characteristics of explosive, premixed type combustion, in contrast to typical diffusion type combustion of diesel fuel. The peak heat release rate of hydrogen-operated engine is 87.6 J/° CA compared to diesel of 81.5 J/° CA. The maximum heat addition occurs nearer to ITDC for hydrogen operation, which also makes the cycle efficiency to increase. Premixed combustion of hydrogen also has the problem of detonation but the operation of engine at leaner equivalence ratio keeps, knocking within the limit.

19. Conclusion

From the experiments conducted it is concluded that an injection duration of 90° crank angle and the start of injection at 5° ATDC give the best results both performance and emission wise.

The smoke emission reduces from 6.8 BSN to 2.3 BSN and NO_X emission decreases from 1806 to 888 ppm at full load. Hence simultaneous reduction of NO_X and smoke is possible using hydrogen in the dual fuel mode. Efficiency increases from 23.59% to 29% at optimized conditions. The emissions such as CO, CO_2 and HC reduce significantly with negligible concentrations.

The pressure variation shows that due to hydrogen fuelled operation the peak pressure increases but with proper injection timing and injection duration the peak pressure can be reduced and maintained within the limit to achieve knock free combustion.

In general, in the operation of hydrogen in the dual fuel mode the performance can be improved with significant reduction

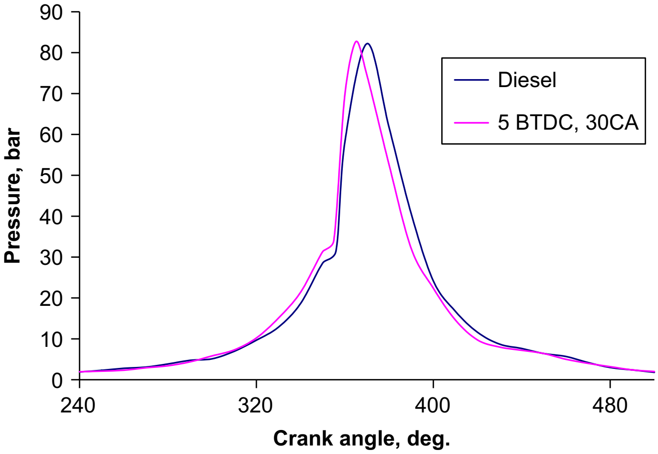

Fig. 13. Variation of pressure with crank angle at full load in optimized condition.

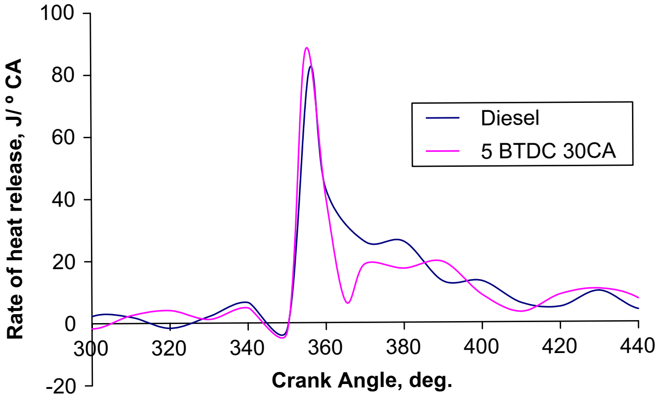

Fig. 14. Variation of heat release rate at full load in optimized condition.

in emissions. Hence by optimizing the injection timing and duration the hydrogen fuelled engine can be operated smoothly in the dual fuel mode.

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