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Effect of Hydrogen-Diesel Dual Fuel Combustion on the Performance and Emission Characteristics of a Four Stroke-Single Cylinder Diesel Engine

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***Abstract*—**The present work attempts to investigate the combustion, performance and emission characteristics of an existing single-cylinder four-stroke compression-ignition engine operated in dual-fuel mode with hydrogen as an alternative fuel. Environmental concerns and limited amount of petroleum fuels have caused interests in the development of alternative fuels like hydrogen for internal combustion (IC) engines. In this experimental investigation, a diesel engine is made to run using hydrogen in dual fuel mode with diesel, where hydrogen is introduced into the intake manifold using an LPG-CNG injector and pilot diesel is injected using diesel injectors. A Timed Manifold Injection (TMI) system has been developed to vary the injection strategies. The optimized timing for the injection of hydrogen was 10^0 CA after top dead center (ATDC). From the study it was observed that with increasing hydrogen rate, enhancement in brake thermal efficiency (BTHE) of the engine has been observed with reduction in brake specific energy consumption (BSEC). Furthermore, Soot contents decrease with an increase in indicated specific NOx emissions with the enhancement of hydrogen flow rate.

***Keywords*—**Diesel engine, Hydrogen, BTHE, BSEC, Soot, NOx.

I. Introduction

THE employment of hydrogen (H_2) as an alternative fuel in internal combustion engines has been studied by a number of research groups globally in retort to increase in governmental stipulation and consumer requirement for more environmentally responsive fuel handcuffs. In contrast with conventional, fossil hydrocarbon fuels, hydrogen tenders almost an exclusion of pollutants for instance carbon monoxide and smoke particulates which are acknowledged to pretense health risks in thickly populated regions. The only nontrivial contaminant from hydrogen engines is nitrogen oxides (NOx); nevertheless the characteristics of hydrogen fuel, such as a high flame speed and prevalent lean-burn operation potential, tolerate significant reductions in NOx in contrast to conventional fuels [1]. The objective of the present work is to use hydrogen (by injection in the intake port) in dual fuel mode with diesel and to study the performance, combustion and emission characteristics in comparison with baseline diesel.

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II. Hydrogen: Its Combustion Characteristics

The unique combustion properties of hydrogen (Table I) make it an ideal choice for its use in compression ignition engine. Hydrogen (H_2) is a colorless gas that has no direct environmental or health impact but can play a role in a number of reactions in emission control catalysts. Hydrogen has significantly different combustion characteristics than other hydrocarbon fuels. The burning velocity of hydrogen-air mixture ranges from 153 to 232 cm/s for its stoichiometric mixture [2], [3]. This results in a more isochoric, thus thermodynamically more favorable combustion than conventional diesel engines which experience a pressure rise spread over several degrees of crank angle. Hydrogen has wider flammability limits of 4-75% by volume in air compared to diesel of 0.7-5% by volume. One of the significant advantages of hydrogen is that the engine can be theoretically operated up to an equivalence ratio of 0.1 [4]. The minimum energy required for ignition of hydrogen-air mixture is 0.02 mJ only. This enables hydrogen engine to run well on lean mixtures and ensures prompt ignition. The density of hydrogen is 0.0837 kg/m^3, which is lighter than air that it can disperse into the atmosphere easily. Hydrogen has the highest energy to weight ratio of all fuels. The flame speed of hydrogen is 270 cm/s that may cause a very high rate of cylinder pressure rise. The diffusivity of hydrogen is 0.63cm^2/s. As the hydrogen self ignition temperature is 858 K, compared to diesel of 453 K. Moreover, the high auto-ignition temperature of hydrogen encourages the use of higher compression ratios as prevalent in diesel engines. The hydrogen-diesel dual fuel concept method combines the advantages of the high part load efficiency, lower specific fuel consumption of a diesel engine and the clean combustion characteristics of hydrogen. However, hydrogen with its cetane-number being very low, are not directly suited to compression ignition engines. Some source of ignition has to be created in the combustion chamber to ensure ignition [4]-[8]. To this end, a small amount of pilot diesel fuel is injected by means of the existing fuel injection equipment near the end of the compression stroke to ignite the gaseous mixture. The advantage of this type of engine is derived by using the difference of flammability of two fuels at different stages of the combustion process.

TABLE I
PROPERTIES OF HYDROGEN IN COMPARISON WITH DIESEL

Sl. No.	Properties	Diesel	Hydrogen
1.	Formula	$C_nH_{1.8}nC_8–C_{20}$	H_2
2.	Auto ignition temperature (K)	530	858
3.	Minimum ignition energy (MJ)	-	0.02
4.	Flammability limits (volume % in air)	0.7-5	4-75
5.	Stoichiometric air fuel ratio on mass basis	14.5	34.3
6.	Molecular weight (g/ mol)	170	2.016
7.	Limits of flammability (equivalence ratio)	-	0.1-7.1
8.	Density at 16°C and 1.01 bar (kg/m^3)	833–881	0.0838
9.	Net heating value(lower) MJ/kg	42.5	119.93
10.	Flame velocity (cm/s)	30	265–325
11.	Quenching gap in NTP air (cm)	-	0.064
12.	Diffusivity in air (cm^2/s)	-	0.63
13.	Octane number	30	130
14.	Cetane number	40-55	-
15.	Boiling point (K)	436–672	20-27
16.	Viscosity at 15.5 °C, (centipoise)	2.6–4.1	-
17.	Specific gravity	0.83	0.091

III. EXPERIMENTATION

The experimentation was conducted on an existing single cylinder four stroke CI engine. The detailed specifications of the test engine are shown in Table II in compliance to the Indian Standards IS: 11170-1985. The engine was attached to an eddy current dynamometer of Saj Test Plant® make and model AG10 for load measurement and integrated with speed sensing unit designed to run at a constant speed of 1500±10 rpm. The instrumentation was interfaced to a computer through a NI Labview® based centralized DAQ platform synchronized with a crank angle encoder onto a GUI based Engine Soft post processing software. The DAQ was programmed to acquire in-cylinder and fuel injection pressure data at 1 deg crank intervals and present the data smoothened over 100 consecutive cycles to compensate for cyclic variations at a particular case of engine operation. The exhaust gas temperature, cooling water inlet and outlet temperatures were also reported as an average value over the period of data acquisition while common performance parameters were reported as calculated values for the same period of sampling. The speed scanning interval of the dynamometer was set to 2000 ms to smoothen undesirable signal fluctuations. The specific fuel consumption of diesel fuel was carried out in fuel burette of all tests were fuel consumption for a time interval of 60 s was recorded in a fuel burette of 12.4 mm diameter. Mass flow rate of inducted air, was recorded as a calculated value (kg/h) from the sensed manometric depression in the integrated air box. Extra care was taken in keeping the speed constant (±10 rpm) during data acquisition at each case of engine load stepping by adjusting the screw given with the fuel pump rack. All procedure pertaining to experimental testing preparation, performance test and recording of test results conformed to the Indian Standards specifications *(IS 10000(Part 5):1980, IS 10000(Part 8):1980, IS 10000(Part 6):1980)* [9]-[11]. The average ambient temperature, cooling water temperature and relative humidity during experimentation was recorded at 25^{0}C, 18^{0}C, and 55%, respectively. In this study, the test engine was operated in diesel mode, and the compression ratio was set to 17.5:1 A schematic view of the test bench is shown in Fig. 1.

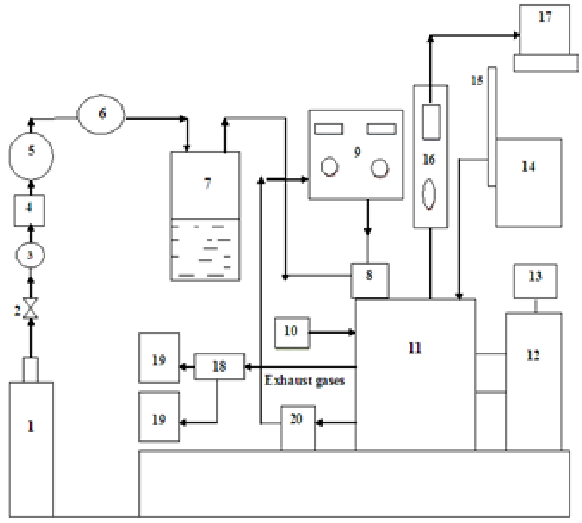

Fig. 1 Schematic diagram of complete experimental circuit
1. Hydrogen cylinder **2.** Pressure regulator **3.** Pressure gauge **4.** Flame arrester **5.** Pressure reducer **6.** Gas flow meter **7.** Water bubbling tank **8.** Gas injector **9.** DAQ station with injector driver & DAQ card and PC **10.** Air filter **11.** VCR engine **12.** Electrical dynamometer **13.** Loading device **14.** Fuel tank **15.** Burette **16.** Engine control panel with sensors **17.** Computer panel **18.** Exhaust Gas Calorimeter **19.** Gas Analyzer **20.** Crank Angle Encoder **21.** Opacimeter

IV. HYDROGEN INDUCTION SYSTEM

If an existing engine were to be fuelled by hydrogen, problems, such as backfire, pre-ignition, high rate of pressure rise and even knock can occur [12]. So, considering the temperamental nature of hydrogen fuel, sufficient care was taken to incorporate redundant safety measures in the hydrogen induction circuit. To this effect, a special in-house developed intake manifold extension was designed with inbuilt flame arrestor consisting of circular plates with perforations of ≤0.064 cm diameter corresponding to the approximate minimum quenching diameter of hydrogen. A remotely operated emergency shut off v/v with temperature relay from the inlet manifold, a flame trap, surge tank and interconnecting flame arrestors as depicted in the schematic Fig. 2 constituted the safety measures in the circuit. The hydrogen (BOC India Ltd: UN 1049 as per CAS: 1333-74-0; Purity ≥99.5% v/v) compressed at around 205 bar, was routed to the solenoid injector via a two stage pressure reduction system comprising of the high pressure reducing stage provided by the pressure regulator/reducer of CONCOA make, capable of 280/14 bar reduction and a low pressure reducing stage located after the surge tank to reduce the pressure manually to the required working pressure of 1.2 bar at the solenoid injector of DYMCO make**.** The injector was mounted onto the inlet manifold at a distance of ≤15 Di from the inlet port [13]. The distance was chosen on the basis of a minimum window allowance to the inducted hydrogen to form a homogenous mixture with the incoming charge and also to ensure at the same time that undesirable pre-ignition induced flashback was dissuaded considering the easily ignitable tendencies of a hydrogen-air mixture. The injector was controlled by means of an electronic control unit (ECU). An Infrared sensor was used to sense the crank angle position. The start of the injection and the duration of the injector opening were controlled by using the ECU. The TMI system was programmed to ensure that hydrogen was inducted only after the scavenge period (IVO-EVC) of 10 deg as depicted in to minimize any loss of the inducted fresh hydrogen charge to the exhaust which further minimized any unwanted pre-ignition of the inducted hydrogen on contact with the hot exhaust gas Moreover, the designed time ensured that hydrogen was injected only after a sufficient flow of cooler ambient air was inducted in the cylinder during the opening of the inlet valve, this being critical in quenching any residual hot spots remaining in the cylinder thereby removing any possibility of flash back [10].

TABLE II
SPECIFICATIONS OF THE ENGINE

Make and Type	KirloskarTV1-Single cylinder, Four stroke, Water cooled.
Bore and stroke	87.5 mm × 110 mm
Type of ignition	Compression Ignition
Compression Ratio	17.5:1
Peak pressure	76 bar
Cubic Capacity	661cc
Maximum power	5.2 kW (@1500rpm)
Fuel Injection pressure	205 bar
Dynamometer	Eddy current type (Make: SAJ test plant pvt. Ltd)
Crank angle sensor	Model 8.3700.1321.0360 (Make KUBLER)
Pressure transducer	Piezoelectric type Model-6056A31U20 (Make: KISTLER),

V. EMISSION ANALYSIS INSTRUMENTS

An AVL Digas 444 5-gas analyzer was used to analyse the gaseous emissions of CO, THC, CO_2 and NOx and an An AVL 415S Smoke Meter was used to measure the smoke emissions.AVL 437 opacimeter are used to analyze the exhaust of the engine. The 5-gas analyzer is used to measure the emissions of CO, CO_2, and O_2 in terms of volume percentage. Although the AVL Digas 444 5-gas emission analyser supplies data as volume % for the CO and CO_2 values and as ppm for the THC and NOx values and the AVL 415S supplies data as FSN or mg/m^3 for the smoke emissions, all of the emission values are converted to g/kW-hr.

VI. EXPERIMENTAL PROCEDURE

As cited earlier, all of the experiments were performed in a single-cylinder, 4-stroke, diesel engine which was first tested with pure Diesel, which provides a baseline data set for comparing different Diesel-Hydrogen combinations in dual fuel mode. Diesel fuel was pulverized in the cylinder at 23^{0}BTDC constant diesel injection advance. H_2 was injected into the intake port using a DYMCO make solenoid injector. In the present experimental investigation hydrogen was injected into the intake manifold via a TMI approach wherein baseline diesel acted as the pilot fuel for the subsequent dual fuel combustion. The TMI approach possesses the capability to commence fuel delivery at a timing position sometimes after the beginning of intake stroke to ensure a pre-cooling effect and thus make the pre-ignition sources ineffective. Furthermore, it helps to quench and dilute any residual combustion products that could be present in the compression space close to TDC [10]. The 10 deg crank angle window being provided to allow hydrogen induction to commence after the scavenge period of the experimental engine together with the reduction of the possibility of a backfire by allowing hotspots contents to cool within a crank window of 4.5 degree ATDC beyond the exhaust v/v closing timing. This creates a $(4.5^0+4.5^0) = 9^0$ of valve overlap. So the induction period remains $(180^0+ 35.5^0+ 4.5^0) - 9^0=221^0$. In order inject the H_2 right at the start of combustion, it was injected 10 degree after TDC, which reduces the effective induction period to $(221^0-10^0) =211^0$. Four different hydrogen injection strategies were studied with 6500µs, 7500µs, 8500µs and 9500µs. The arithmetic mean was calculated based on at least 100 experiments. Before starting this experiment, the engine was heated to a constant engine regime temperature. The cooling water intake and output temperatures were measured using K-type thermocouples. The test was first implemented with no hydrogen content, that is, with only diesel fuel, at 1500 rpm, which was the full load. After achieving the stable operation of the engine, hydrogen was sent into the engine at (1500±10) rpm, which was the full load. First, this procedure was applied to the case of 6500µs injection duration strategy 11and then this procedure was then applied for the other H_2 strategies. In all of the experiments, the engine was at full load condition. Backfire, engine knock, or pre-ignition were not encountered in the experiments.

VII. RESULTS AND DISCUSSIONS

Experiments were carried out with hydrogen and diesel in dual fuel operation and the engine was operated with pure diesel at a constant speed of (1500±10) rpm at 20% full load, 60% full load and 100% full load conditions. Then the engine is switched to dual-fuel mode of operation with hydrogen as the main fuel and diesel as the pilot fuel under the same operating conditions. At each load condition on various injection strategies combustion, performance and emission analysis have been studied during experimentation.

A. Performance Analysis

1. Brake Thermal Efficiency (BTHE)

The variation of the brake thermal efficiency with several hydrogen strategies is shown in Fig. 2. The brake thermal efficiency of the engine with hydrogen enrichment reaches to a maximum of 32.24% at full load condition with hydrogen injection strategy-3 (DH3), whereas with diesel alone was 26.79%. In the experiment it was observed that as flow rate of hydrogen started increasing there was decrease in flow rate of diesel. It indicates that hydrogen is taking part in the combustion. The increase in brake thermal efficiency for hydrogen operation is due to enhancement of hydrogen in air. Increase in thermal efficiency is credited to improved combustion because of superior combustion rate due to high flame velocity of hydrogen.

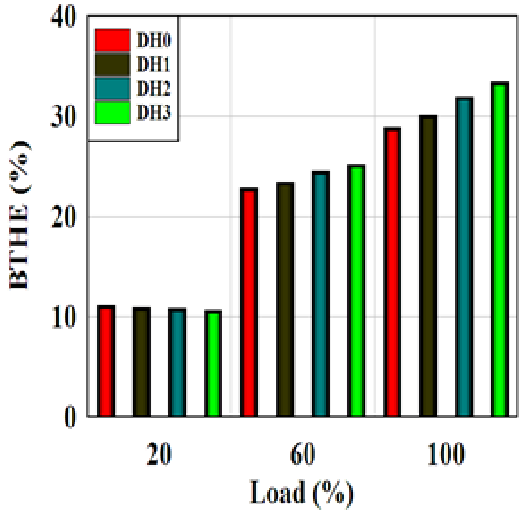

Fig. 2 Variation of BTHE (%) with load (%)

2. Brake Specific Energy Consumption (BSEC)

Fig. 3 shows the variation of BSEC (brake specific fuel consumption) with hydrogen injection strategies. In the present study, it has been observed that there was a persistent trend of reduction of BSEC with increase in load for a given hydrogen enrichment strategy. This is an indicative of the enhanced combustion of conventional diesel combustion on account of high flame velocities and high calorific content of the participating hydrogen with air resulting in complete combustion of fuel and a little more engine power due to the increase in the amount of hydrogen fuel. This trend was maintained in all cases of observations which are a clear indicative of better mixing of hydrogen with air resulting in more efficient burning process of fuel mixture. This is an indicative of the enhanced combustion of conventional diesel combustion on account of high flame velocities and high calorific content of the participating hydrogen with air resulting in complete combustion of fuel and a little more engine power due to the increase in the amount of hydrogen fuel.

B. Emission Analysis

1. Soot

Fig. 4 shows the variation of soot formation with an increasing H2 energy content. The soot formation reduces surprisingly with hydrogen addition. It has been observed that in this TMI technique the maximum soot reduces by 38.62% when compared to diesel with the augmentation of hydrogen share with the main fuels at higher load. Hydrogen enrichment in diesel engine acts as a soot inhibitor in many ways. Primarily, it enhances the overall H/C ratio during the dehydrogenation of the hydrocarbon fuel, thus inhibiting soot nucleation, and the elevated combustion temperatures together with the profusion of OH radicals made available through the hydrogen–oxygen reaction mechanism under such hydrogen enrichment operation serve as potent soot oxidizers.

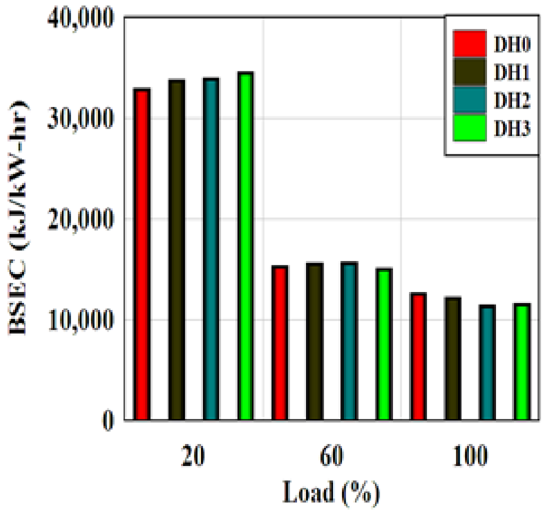

Fig. 3 Variation of BSEC (kJ/kW-hr) with load (%)

2. Oxides of Nitrogen

The variation of indicated specific NOx emissions with the increasing H_2 content is shown in Fig. 5. The development of NOx is highly dependent on in-cylinder temperature, oxygen concentration and dwelling time for the reaction to take place. The NOx emission is found to be max of 10.35kg/kW-hr at full load condition for hydrogen-operated engine at maximum injection durations of hydrogen (DH3) compared to diesel of 3.15kg/kW-hr. The reason for this higher concentration of NOx in case of hydrogen enrichment due to peak combustion temperature and high residence time of the high temperature gases in the cylinder. Formation of NOx is associated with the in-cylindrical temperature rise, oxygen concentration and reaction duration. Hydrogen has higher flame speed propagation than that of diesel which will enhance the probability of complete combustion in dual fuel mode. Complete combustion leads to higher in-cylinder peak pressure and which in turn increases the in-cylinder temperature. Hydrogen has lower heating value (LHV) compared to diesel fuel which is the further reason for high NOx emission, which results in an enhancement in the peak in-cylinder temperature.

C. Combustion Analysis

1. Variation of In-Cylinder Pressure with Crank Angle

Figs. 6-8 represent the calculated pressure data for several H_2 injection strategies. The peak cylindrical pressure value was observed to be 58.92 bar for the third injection strategy of hydrogen (DH3) at full load condition. The rise in peak cylindrical pressure the rise in peak cylindrical pressure directly with the added hydrogen quantity as hydrogen increases the in-cylinder pressure after the combustion is initiated. Owing to elevated laminar flame speed of hydrogen, the peak pressure and the rate of pressure rise reaches a higher value with instantaneous combustion of hydrogen.

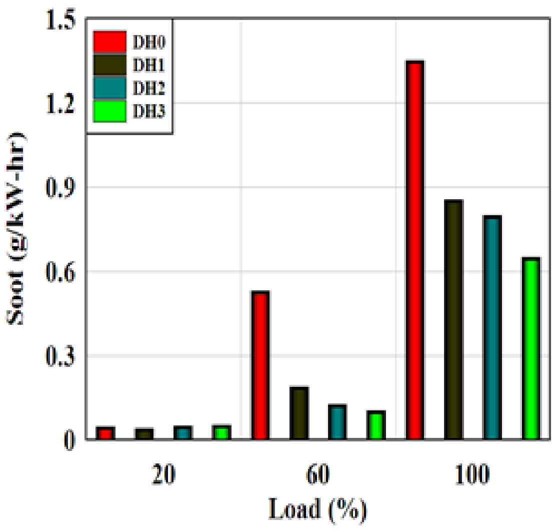

Fig. 4 Variation of Soot (g/kW-hr) with load (%)

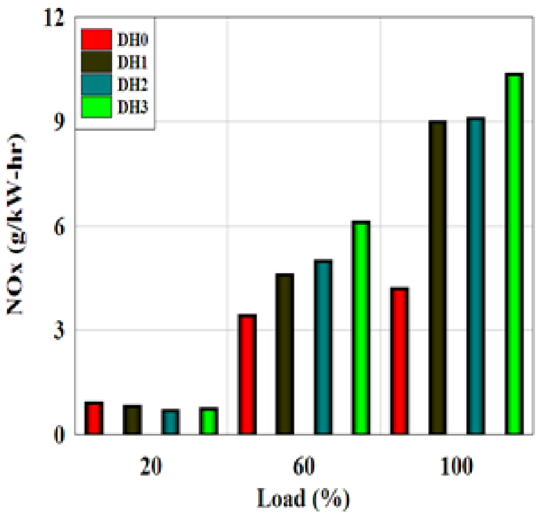

Fig. 5 Variation of NOx (g/kW-hr) with load (%)

2. Variation of Rate of Heat Release with Crank Angle

The rate of heat release for pure diesel and various H_2 strategies are shown in Figs. 6-8. In case of pure diesel (DH0), three phased combustion can be observed as the premixed combustion phase, the mixing controlled combustion phase and late combustion phase. If the quantity of hydrogen increases due to the high flame speed of hydrogen, then in contrary of the mixing-controlled combustion phase, i.e., a premixed combustion phase in explosive type is observed. The premixed combustion of hydrogen has the difficulty of causing detonation; however, in this experimental study, three hydrogen injection strategies have been incorporated i.e. DH1, DH2 and DH3. The peak heat release rate is 57.08 J/^{0}CA for the third injection strategy of hydrogen (DH3) at full load condition. This rise in peak heat release rate was observed to be amplified with increase in H_2 content compared pure diesel, and it decreases in all other H_2 strategies due to inability to burn the hydrogen with the stipulated period of time. The addition of hydrogen was shown to speed up the initiation of the premixed combustion. Compared to diffusion combustion, hydrogen was found to have a more momentous effect on the premixed combustion. With the hydrogen accumulation in fuel enhances the heat release rate.

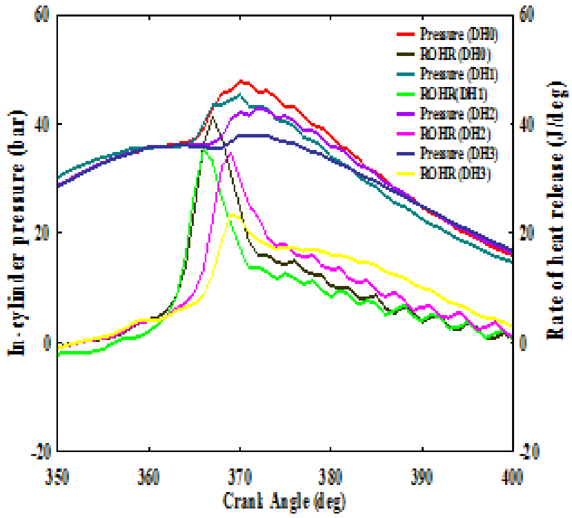

Fig. 6 Variation of In-cylinder pressure and Rate of heat release versus crank angle at 20% full load condition

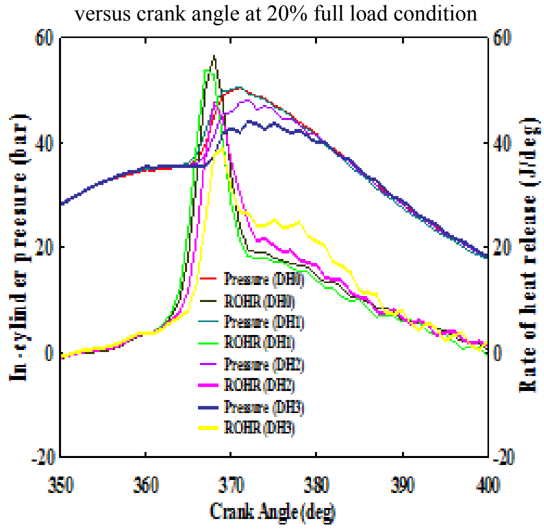

Fig. 7 Variation of In-cylinder pressure and Rate of heat release versus crank angle at 60% full load condition

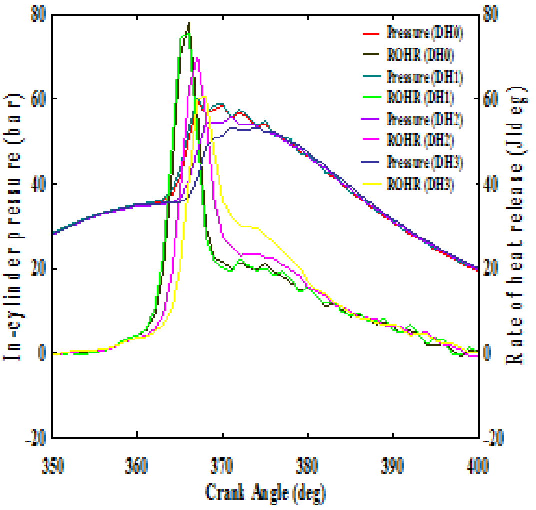

Fig. 8 Variation of In-cylinder pressure and Rate of heat release versus crank angle at full load condition

VIII. Conclusions

In this experimental study, the performance characteristics of a single-cylinder diesel engine are observed with its combustion characteristics at a (1500±10) rpm engine speed, 5.2kW constant indicated power and different hydrogen injection strategies. The test outcome is acknowledged below:

- The brake thermal efficiency of the engine with hydrogen enrichment reaches to a maximum of 32.24% at full load condition with hydrogen injection strategy-3 **(DH3),** whereas with diesel alone was 26.79%. Increase in thermal efficiency is credited to improved combustion because of superior combustion rate due to high flame velocity of hydrogen.
- As hydrogen is enriched in the cylinder, there was a persistent trend of reduction of BSEC with increase in load for given hydrogen injection strategies. This is an indicative of the enhanced combustion of conventional diesel combustion on account of high flame velocities and high calorific content of the participating hydrogen with air resulting in complete combustion of fuel and a little more engine power due to the increase in the amount of hydrogen fuel.
- The soot formation reduces surprisingly with hydrogen addition. It has been observed that in this TMI technique the maximum soot reduces by 38.62% when compared to diesel with the augmentation of hydrogen share with the main fuels at higher load. The increase in hydrogen induction enhances the overall H/C ratio during the dehydrogenation of the hydrocarbon fuel and thus reduces soot.
- With the addition of hydrogen, an increase in NOx emission was observed and which reaches to a maximum of 10.35kg/kW-hr at full load condition for hydrogen-operated engine at maximum injection durations of hydrogen (DH3) compared to diesel of 3.15kg/kW-hr. The reason for this higher concentration of NOx in case of hydrogen enrichment due to peak combustion temperature and high residence time of the high temperature gases in the cylinder.
- The peak heat release rate is 57.08 J/^{0}CA for the third injection strategy of hydrogen (DH3) at full load condition. This rise in peak heat release rate was observed to be amplified with increase in H_2 content compared pure diesel, and it decreases in all other H_2 strategies due to inability to burn the hydrogen with the stipulated period of time.
- The peak cylindrical pressure value was observed to be 58.92 bar for the third injection strategy of hydrogen (DH3) at full load condition. The rise in peak cylindrical pressure. Due to high flame speed of hydrogen which produces an instantaneous combustion.

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