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Effect of Hydrogen Energy Share on a Hydrogen Diesel Dual Fuel Mode using a Common Rail Direct Injection System

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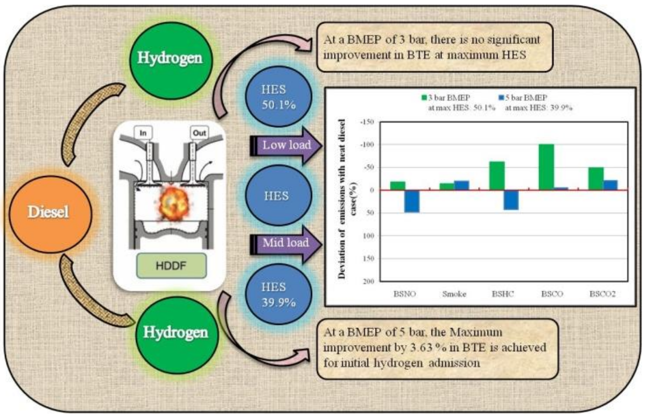
Abstract: ***This research work was aimed to investigate the maximum possible utilization of hydrogen energy in CI engine with diesel fuel replacement on energy share basis in hydrogen diesel dual fuel (HDDF) mode using common rail direct injection diesel engine. In HDDF mode, experiments were performed at two load conditions such as 3 and 5 bar of brake mean effective pressures (BMEPs). At a BMEP of 3 bar, the maximum hydrogen energy share (HES) could be reached to 50.1% without erratic engine operation, the level of NO, HC, CO and CO_2 emissions were decreased up to 18%, 62%, 56% and 45% as compared with neat diesel case. However, there was no significant change in brake thermal efficiency (BTE) and smoke at high HES. At a BMEP of 5 bar, the maximum possible hydrogen energy share was nearly 40%, beyond this energy share severe knocking was noticed. NO emission was raised by 48%, whereas smoke and CO_2 emission were decreased up to 20% and 24%. The brake specific HC emission was increased by 42% and there was no significant variation in CO emission with HES.***

Keywords* : *Hydrogen, dual fuel mode, diesel, hydrogen energy share.



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I. INTRODUCTION

Due to depleting sources of fossil fuel and stringent emission norms for the vehicular pollutions, the researches has to focus on cleaner burning, renewable and sustainable fuels. In this context, hydrogen is anticipated to be one of the promising fuel in future. Hydrogen (H2) is the superabundant chemical element, which is existing on the earth in the type of compounds. Two hydrogen atoms react with one oxygen atom resulting in greater energy liberation and generation of water; this can be done either in the type of combustion in an engine or in the type of chemical reaction in the fuel cell. In the case of combustion, hydrogen is considered as a clean and greatest alternative for fossil fuel, since it does not produce unburned hydrocarbon (HC), carbon dioxide (CO_2), oxides of sulphur (SO_x), carbon monoxide (CO) and particulate matter (PM) during combustion [1]–[3].

Hydrogen is compatible in spark ignition (SI) engine as a fuel because of its high octane rating and hydrogen utilization in SI engine may gives satisfactory performance with zero carbon emission because of high in-cylinder temperature. But, hydrogen fuelled spark ignition (H_2SI) engine operation is limited due to low thermal efficiency, low volumetric efficiency and throttling losses. In addition, SI engine is operated near stoichiometric mixture, thus it leads to high rate of combustion which produces elevated level of pressure and temperature within the cylinder. Therefore, nitric oxide (NO) emissions and knocking are also higher in the case of H2SI engine [4]–[6].

In CI engine, hydrogen cannot be utilized as a sole fuel, since the hydrogen have high auto ignition temperature and low cetane number. Therefore, CI engine need to be run in dual fuel (DF) mode if hydrogen used as a fuel. Here, less quantity of diesel is used as a combustion trigger and hydrogen is the main energy liberating fuel in the engine on the basis of hydrogen diesel dual fuel mode. This mode in CI engine leads to higher thermal efficiency due to less fuel consumption, high compression ratio (CR), no throttling losses and less green house gas emissions. In addition, CI engine does not require any major engine modification as compare with SI engine when operating with dual fuel mode. Furthermore, in CI engine, the backfire and pre-ignition are not significant and knock level is negligible compared with SI engine. This indicates CI engine can be used hydrogen safer than SI engine [7], [8]. Hydrogen has a tendency to replace the air, therefore the availability of air is reduced during diesel combustion process.

There are many techniques in hydrogen induction or injection to address this problem such as continuous manifold injection (CMI), direct hydrogen injection (DHI), carburetion and timed manifold injection (TMI). In external mixing case, the hydrogen is uniformly mixed with intake air and this method forms the mixture easily without any complexity compare with DHI, however engine power output is considerably low due to abnormal combustion such as pre-ignition, knocking and back fire at high load [8], [9].

A prior investigation by Saravanan et al. [10] on hydrogen in dual fuel operation using conventional CI engine. Their outcomes showed that the BTE was improved up to 20%, specific fuel consumption (SFC) was decreased and the level of smoke was declined remarkable level when compared with diesel base line. This was because, hydrogen uniformly mixed with incoming air, which resulted in complete combustion of entire charge. In addition, the pressure rise rate and heat release rate were elevated in dual fuel mode when match with neat diesel operation. In spite of this, the concentration of oxides of nitrogen (NO_x) emission was raised to 13 % when matching with conventional diesel combustion mode due to high combustion temperature. Hydrogen can be utilized as a supplemental fuel (small energy ratio) or high energy ratio basis in dual fuel operation to enhance the efficiency and lower the carbon related emissions, an earlier investigation by Varde and Frame [11] on the reduction possibilities of smoke by hydrogen introduction in the intake manifold of the CI engine and they reported that when hydrogen energy share at 10 to 15 %, there was a significant decrease in smoke emission (50 % at mid load and 17% at high load) but there was rise in NO_x at both mid and high load conditions. In addition to that, at very small energy shares, a adverse effect on the engine efficiency, however the thermal efficiency was improved, when increase in hydrogen energy share and this same effect was also been observed by Tsujimura and Suzuki [12].

Batmaz [13] investigated influence of hydrogen addition in single cylinder air cooled diesel engine and they could be able to reach maximum 20 % of hydrogen rate at high load condition. Authors concluded that hydrogen addition in diesel engine is not suitable because of high knocking tendency at high engine speed, elevated level of NO_x was also observed as compared with standard fuels and low volumetric efficiency. However, at medium and low speed of the engine, the SFC was improved. Lilik et al. [14] evaluated the effect of hydrogen progress in 4 cylinder, common rail, direct injection, turbocharged, CI engine on small energy share basis and they observed moderate rise in NO_x emissions, in which nitric dioxide (NO_2) being predominant than NO and a major change in NO/NO_2 ratio as increasing the hydrogen energy ratio. In addition, authors confirmed with help of numerical results that HO_2 radicals levels were increased with rise in hydrogen energy ratio, this rise in HO_2 radicals support the change of NO to NO_2. However, the level of HC, CO and soot were decreased as hydrogen energy ratio increases.

The research work has been conducted by Sharma and Dhar [15] on hydrogen supplementation in single cylinder CI engine. They noticed that NO_x emissions were decreased at 25% and 50% of load due to lowered hydrogen combustion efficiency. At this same load operations, increase in hydrogen energy ratio resulted in low NO and high NO_2. This drop in NO indicates the overall NO_x reduction at this load conditions. At high load (75%), NO_x was increased by 9% with 20% of hydrogen energy ratio and this increase was because of hydrogen effectively involved in combustion reaction caused by excessive pressure and temperature within the engine. However, when hydrogen is effectively participated in combustion, the overall emissions such as PM, CO and HC emissions were decreased.

Wu et al. [16] investigated hydrogen rich/diesel engine with change in intake temperature and EGR ratio. Here, hydrogen rich gas was supplied by methanol steam reformer into the engine, the optimal combination of parameters are indentified by using Taguchi approach for optimizing the combustion and performance characteristics with hydrogen rich gas flow rate, cooled EGR and intake air temperature. The maximum reduction rate of NO_x, smoke, HC and CO was 41.35%, 29.27%, 15% and 31.58% by using combination of optimal parameters. A study by Tomita et al. [17] on hydrogen in dual fuel mode using single cylinder, naturally aspirated, 4 stroke CI engine. Authors reported that NO_x emission was raised for retarded injection timing but as injection timing was advanced to higher level, the level of NO_x emission was lowered, since the advanced injection timing provides enough amount of available time to form lean premixed mixture, thus smooth and low combustion temperature was achieved. As hydrogen energy ratio increased, NO_x was increased, smoke was reached nearly zero for all injection timing and HC, CO and CO_2 were also lower. However, the BTE was decreased little when checking against normal diesel base line. Tsujimura and Suzuki [12] observed abnormal combustion at high engine load as hydrogen energy ratio rises above 50%. This was due to hydrogen flame propagate towards cylinder wall, which raised the cylinder head temperature to higher level and this may eventually resulted in pre-ignition.

The recent research work on hydrogen diesel dual fuel is done by Castro et al. [18] using multi-cylinder, CI engine. The highest possible hydrogen energy share for 30%, 60% and 100% engine loads were 80%, 60% and 40%. According to experimental study, highest energy shares were limited by back fire at all engine load conditions and knocking at medium and full load operation. As progress in hydrogen energy share, brake SFC was declined irrespective of the load conditions. As percentage of diesel consumption in dual fuel mode decreases, percentage of CO_2 emission was decreased. Further, NO emission was decreased with energy share at low load condition and vice versa for high load condition because of high in-cylinder temperature at elevated load condition.

The most of research work showed that significant decrease in carbon associated emissions such as HC, CO, CO_2, and PM if hydrogen energy ratio progresses. However, the level of NO_x emissions went high at medium to full load when burning the high energy content of hydrogen. Due to inconsistent and discrepancies in the literature results on HDDF operation on energy share basis, further investigation is needed for clear understanding of hydrogen energy share in HDDF mode. Therefore, it is worthwhile to study the combustion, performance, emission characteristics of HDDF mode with various hydrogen energy shares in a twin cylinder, modern common rail, direct injection, CI engine. Moreover, the HDDF mode has been studied and evaluated without any major modification in existing commercial diesel engine.

II. ENGINE INSTRUMENTATION

In this work, a twin cylinder, naturally aspirated, common rail, water cooled, direct injection CI engine was used, which was operated in HDDF mode with minor modification in the engine. Table 1 is enlisted about the engine specification. In order to calculate the engine power and also maintain the speed of the engine constant, an eddy current dynamometer was united to the engine.

The solenoid injector has 7 holes to improve the mixing phenomenon in this engine. The schematic appearance of experimental setup is depicted in Fig.1. A displacement type gas meter (Brand: FMG, Model No: FMR DN80 G100, Netherlands) was mounted near the surge tank to calculate the air flow rate of on volume basis. The hydrogen was directly inducted and flow rate was measured using thermal mass flow meter (Brand: Bronkhorst, Model No: F-113AC-M50-AGD-55-V, Netherlands) and the diesel fuel flow into the engine was measured on mass basis directly. Here, for safety purpose, the hydrogen was not directly inducted into the engine instead of this, it was sent through water trap and flame arrester to avoid the any back fire from engine to reach hydrogen cylinder. In addition, the hydrogen leak check was ensured before doing any experiment using personal hydrogen gas detector (Brand: Riken Keiki, Model No: GP-03, Japan). The exhaust gas emissions such as nitric oxide (NO), HC, CO_2 and CO from the engine are measured by using exhaust gas analyzer (Brand: AVL Di-test, Model No: 100 BL, Austria). A filter paper method of smoke measurement was done in exhaust of the engine using smoke meter (Brand: AVL, Model No: 415S, Austria).

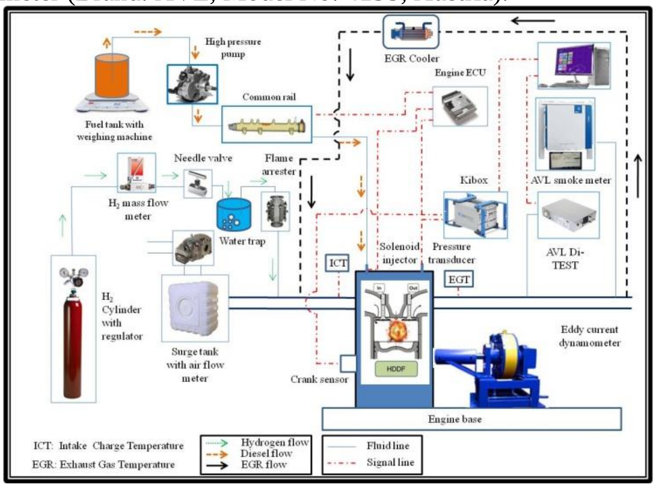

Fig. 1. Schematic appearance of the experimental setup for HDDF mode.

Table-I: Specification of the CI engine

Displacement volume	909 cc
Bore and stroke	83 and 84 mm
Compression ratio	18.5:1
No. of cylinders	2
Maximum torque	55 N-m at 1800-2200 rpm
Maximum power	25 bhp at 3600 rpm

A piezoelectric pressure transducer (Brand: Kistler, Model No: 6052C, Switzerland) was used to capture the engine in-cylinder pressure. The location of crank shaft was identified using engine crank sensor in order to evaluate the pressure data with respect to crank angle. The Kibox cockpit (Brand: Kistler, Model No: Type 2893A, Switzerland) was used to acquire the in-cylinder pressure and crank angle signals for 100 combustion cycles. The heat release rate was computed through obtaining the ensemble average for 100 successive combustion cycles. The engine coolant temperature was maintained constantly using thermostat valve and measured using resistance type thermocouple. Exhaust gas temperature measurement was done through K-type thermocouple and the back pressure valve has been developed in the exhaust manifold to control the back pressure, thereby EGR flow rate could be improved as required. The current clamp was used to check the changed injection timing (Brand: Chauvin Arnoux, Model No: E3N, France), which was mounted near the injector.

III. EXPERIMENTAL METHODOLOGY

The engine was initially operated in CI mode for any load condition. The first segment of experiments were conducted in two load conditions, one is at low load (BMEP : 3bar) and another one is at medium load (BMEP : 5bar). The engine speed of 1800 rpm and coolant temperature of 80 °C were maintained constantly all the experiments. Subsequently, the HDDF mode has been carried out by inducting the hydrogen directly through the intake of an engine. The hydrogen leak test has been performed for all experiments using personal hydrogen gas detector. The hydrogen energy share was increased by rising the hydrogen and decreasing the diesel quantity, the hydrogen was send from hydrogen gas cylinder and pressure regulator to intake manifold through water trap and flame arrester for safety purpose, here the needle valve was used in the hydrogen line to vary the hydrogen flow rate. The diesel injection timing and quantity was played by an engine ECU (electronic control unit), however quantity was adjusted according to our requirement through gas pedal. The varied injection timing by an engine ECU was noticed with help of current clamp. The hydrogen energy share is determined as follows:

$$\text{Hydrogen energy share (HES)} = \frac{EH_2}{EH_2+ED} * 100(\%)$$

Where,
EH_2 = Hydrogen energy input
ED = Diesel energy input

IV. RESULTS AND DISCUSSIONS

In this section, influence of hydrogen energy share and EGR in combustion, performance and emission characteristics of HDDF mode at two different operating BMEPs are presented and discussed in details.

A. Combustion Characteristics

Low load (BMEP: 3 bar)

Figures 2 depicts the heat release rate and injection pulse with respect to crank angle at various hydrogen energy shares in 3 bar BMEP.

At this BMEP, the engine speed and coolant temperature were kept constant, while the EGR was maintained 0 % for all the experiments, which was conducted at this BMEP. In order to improve the hydrogen energy share, the quantity of diesel need to be reduced and this was done through gas pedal. It may be noted that pre and main injection timing was varied by engine ECU as diesel quantity decreases. Here, there are two peak heat release rate for all the hydrogen energy shares which is similar to convectional diesel combustion with a dual pulse injections.

The first peak of heat release rate (pre-combustion) is indicate the combustion of pre-injection of small quantity

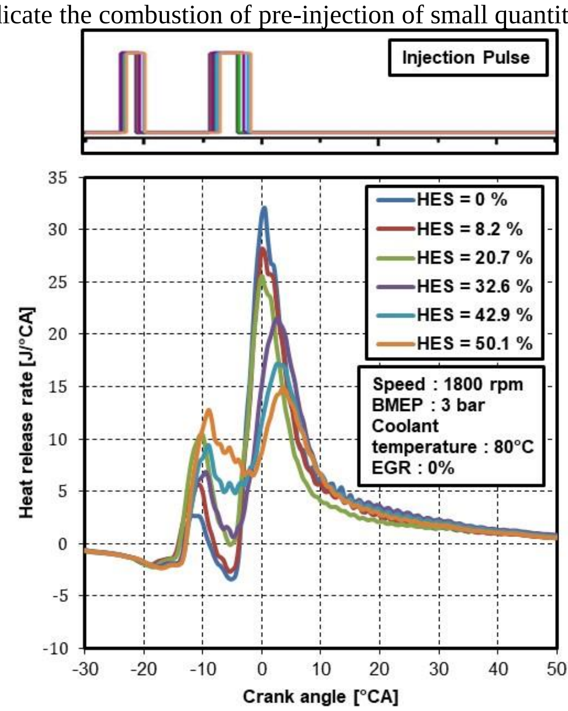

Fig. 2. Heat release rate at a BMEP of 3 bar.

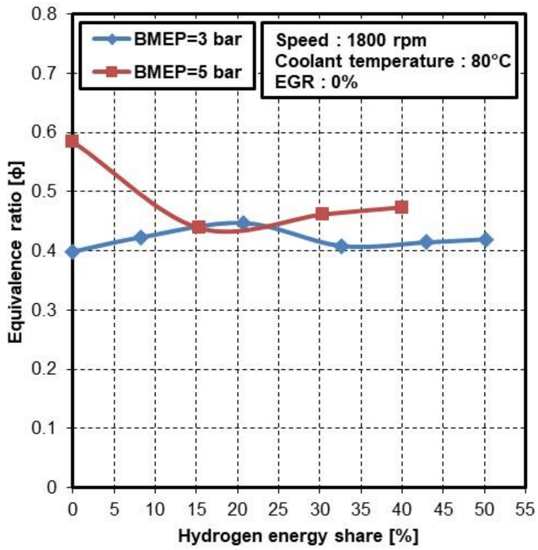

Fig. 3. Equivalence ratio at BMEPs of 3 and 5 bar.

of diesel with inducted hydrogen-air mixture. The second peak of heat release rate (main combustion) shows the combustion of main diesel injection with hydrogen-air mixture, which is prevailed in the in-cylinder. This double peak starts to combine when hydrogen energy share exceeds 42.9 %. This was because; the diesel pulse effect became lower as hydrogen energy share rises.

For pre-combustion, the rate of heat release was increasing and retarded towards TDC. However, for main combustion, combustion was retarded away from TDC and rate of heat release was also lower with hydrogen energy share. This dual combustion effects neutralize the thermal efficiency improvement and will be explained later in BTE section. It was not possible to operate the engine more than 50.1 % of hydrogen energy share. Since, above this energy share, there was erratic engine operation, the engine was not stable at this BMEP in HDDF mode. In addition, equivalence ratio moves to slight rich region as hydrogen addition rises is depicted in Fig.3. This was due to the hydrogen replaced the inducted air in the cylinder which led to the overall mixture became slightly richer.

Medium load (BMEP: 5 bar)

Figure 4 illustrates the rate of heat release for different hydrogen energy shares at a BMEP of 5 bar in HDDF mode. Here, throughout the experiment EGR was 0 % and the engine speed and coolant temperature are held constant. The two peak heat release was also observed in medium load condition for all hydrogen energy share alike in 3 bar BMEP.

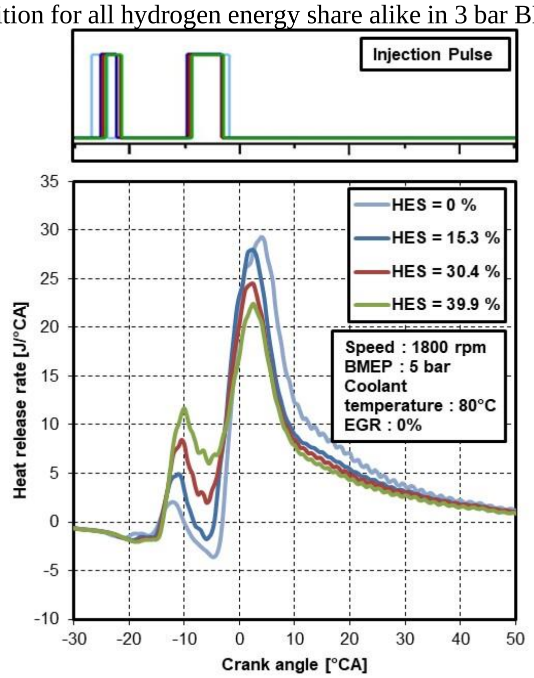

Fig. 4. Heat release rate at a BMEP of 5 bar.

From Figure 4, it is seen that the premixed combustion zone in main combustion part was not significantly decreased with hydrogen addition as compared to 3 bar BMEP, which indicates that the flame speared across the cylinder throughout the hydrogen air mixture and the hydrogen was participated effectively during combustion. Thereby, thermal efficiency was improved compare with CI mode (HES:0%), this will be discussed later in the performance characteristics section. This similar kind of effect was also observed by Tsujimura and Suzuki [12]. For hydrogen energy share of 15.3 %, the main combustion was advanced towards TDC as compared with other energy shares. This was due to the main injection timing has been slightly advanced by a engine ECU as seen in injection profile. Thus in turns improved the BTE at this energy share. The diffusion combustion zone was decreased in main combustion heat release with hydrogen addition and which was reflected positively in soot formation.

In this BMEP, the maximum possible hydrogen energy share was 39.9 %, beyond this energy share there was vigorous knocking in HDDF mode and this limits the operating range at this BMEP. The overall mixture becomes leaner when hydrogen fraction increases as showed in Fig.3. This indicates that the amount of intake air replacement by hydrogen was reduced at high engine load.

B. Performance Characteristics

Effect of hydrogen energy share in BTE at BMEP of 3 and 5 bar is shown in Fig.5.

The BTE was not significantly very with energy share at a BMEP of 3 bar. This was because, for main combustion, the rate of heat release was decreasing and combustion was retarded away from TDC with hydrogen energy share and vice versa for pre-combustion. This dual combustion effect compensates the variation of thermal efficiency. It has been reported that unburned hydrogen emission in the exhaust was linearly increase as hydrogen addition increases at low load operation and consequence of this, thermal efficiency was also deteriorated Gatts et al. [19]. In 5 bar BMEP, the BTE was improved by 3.63 % for initial HES admission as compared with CI mode. This was due to slight advanced combustion towards TDC and hydrogen was participated effectively because of higher in-cylinder temperature. As hydrogen energy progresses, the thermal efficiency was slightly decreased. This was because of retarded injection timing moves the combustion slightly away from TDC as shown in Fig.4.

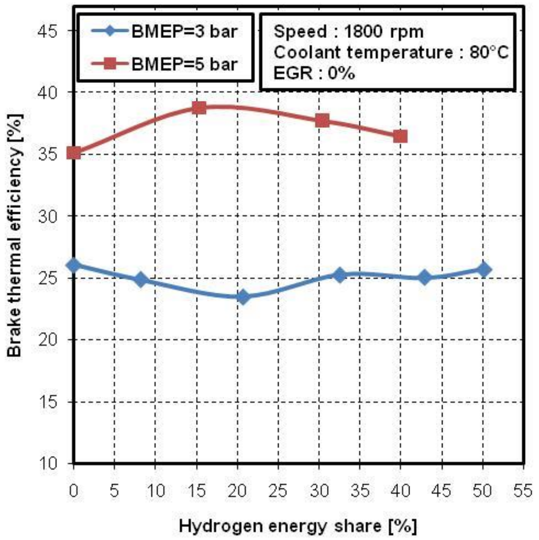

Fig. 5. Brake thermal efficiency (BTE) at BMEPs of 3 and 5 bar.

However the efficiency was not lower than CI mode. The overall utilization of hydrogen has been improved in 5 bar BMEP as compared to 3 bar and which was reflected as high thermal efficiency at medium load operation.

C. Emission Characteristics

At a BMEP of 3 bar, the brake specific NO (BSNO) emission was decreasing with energy share improves as seen from Fig.6. The BSNO emission was reduced by 18 % with highest HES when compared with CI mode. This was caused by hydrogen gas replaced the inducted air within the cylinder, thus it led to the overall mixture became slightly richer as explained earlier and the concentration of oxygen was also lower, thereby, lower in-cylinder temperature suppress the NO formation. Similar results were also noticed by Morsy et al. [20] at low load operation. It has been reported in the literature Pan et al. [21] that if unburned hydrogen present in the in-cylinder behave as a diluents which reduce NO_x formation, moreover hydrogen's specific heat capacity is slightly higher than N_2. In addition, at low load, the lower hydrogen combustion efficiency decreases NO_x emission as unburned hydrogen increases [15]. In other hand, medium load condition at 0% of EGR, NO formation was linearly increases with HES as seen from figure. This linear increase was due to lean mixture and another reason may be that the hydrogen was participated effectively during combustion because of improvement in in-cylinder temperature at medium load condition, this eventually end with high NO emission in the exhaust.

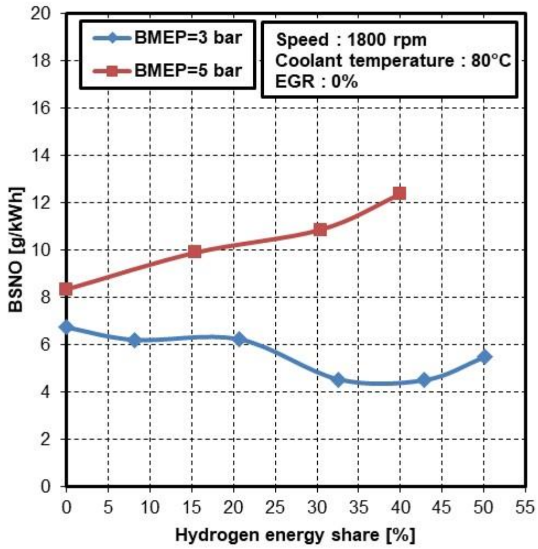

Fig. 6. NO emission at BMEPs of 3 and 5 bar for various HES

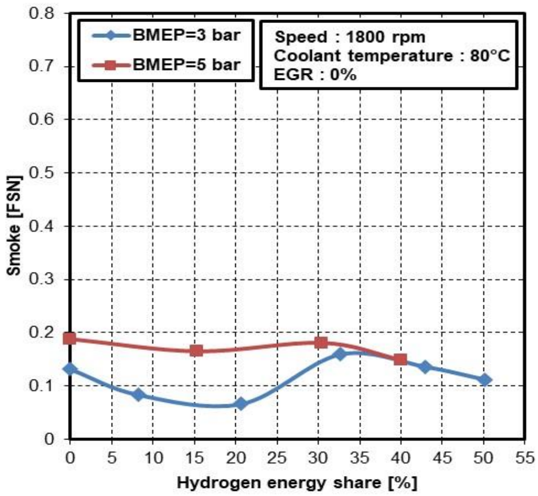

Fig. 7. Smoke emission at BMEPs of 3 and 5 bar for various HES.

In the case of smoke emission, for 3 and 5 bar, the overall smoke emission was lower as seen from Fig.7. At 5 bar BMEP, there was a decreasing trend as energy share increases, since hydrogen create the homogeneity in the mixture rather than heterogeneity and hydrogen does not contain carbon particle in it, thus the diffusion zone in the combustion was lower with HES progresses as shown earlier in combustion section.

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It may be noted that reduction in diffusion combustion zone led to lower soot formation [22]. At low load condition, for initial rise in HES, soot formation was lower as because of diesel quantity was decreased with rise in HES. However, as HES moves to 32.6 % the soot formation was increased even though drop in diesel quantity, this was because of improvement in HES reducing the oxidation rate of injected diesel fuel and decreases the heat release rate.

On the whole, there was no significant change in smoke emission as compared with CI mode in low load condition.

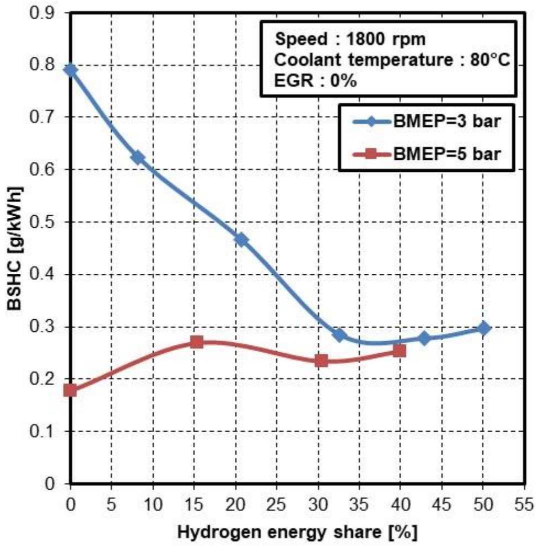

Fig. 8. BSHC emission at BMEPs of 3 and 5 bar for various HES.

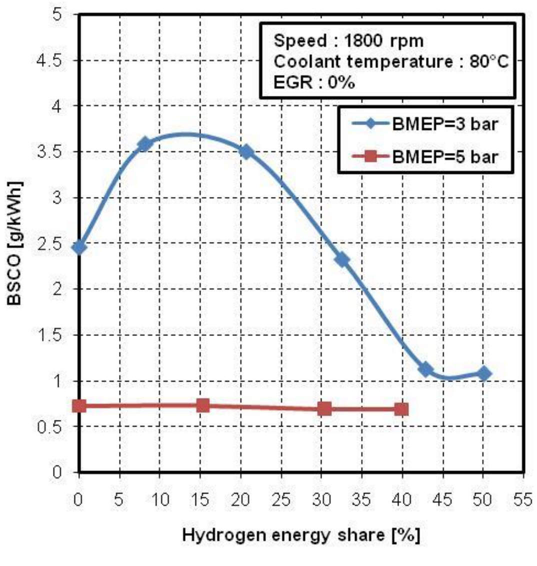

Fig. 9. BSCO emission at BMEPs of 3 and 5 bar for various HES.

The brake specific HC (BSHC) was declined up to 62% while match against with CI mode for 3 bar BMEP as depicts in Fig.8. It was seen from Fig.9 that brake specific CO (BSCO) emission was decreased by 56% as HES moves to higher energy share. This decline in carbon emissions suggest that the diesel quantity was replaced by hydrogen, the source of CO and HC emissions at low energy share was purely derived from diesel fuel. At medium load, the brake specific HC and CO emissions were lower than 3 bar BMEP because of rise in combustion temperature at elevated load condition. However, there was a increasing trend of HC with HES and there was no notable variation in CO emission with HES at 5 bar. In general, the CO_2 emission in the exhaust indicate the complete combustion of entire cylinder charge, In HDDF mode, the brake specific CO_2 ($BSCO_2$) emission was decreasing with hydrogen fraction increases as represented in Fig.10.

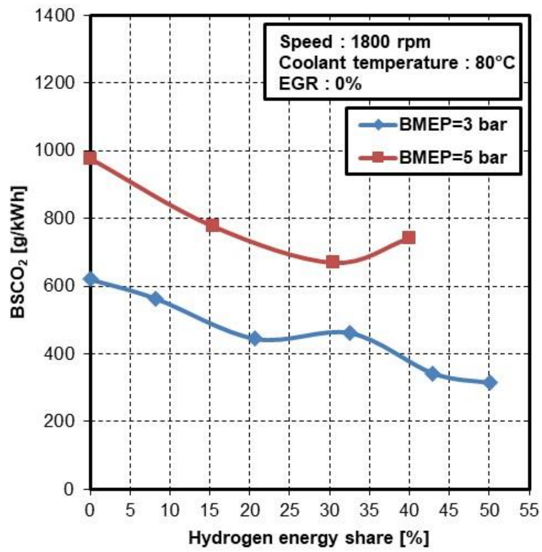

Fig. 10. $BSCO_2$ emissions at BMEPs of 3 and 5 bar for various HES.

This was caused by reducing diesel quantity with HES as well as in-cylinder charge was not completely burned. CO_2 emission was reduced up to 45% and 24% for 3 and 5 bar of BMEPs, however the concentration of CO_2 emission was higher at 5 bar BMEP as compared with 3 bar. This represents that the more complete combustion was attained at medium load as compared with low load condition because of increased in-cylinder temperature condition.

V. CONCLUSION

The following conclusions are drawn from experimental study on HDDF mode in a twin cylinder, modern automotive, common rail, direct injection CI engine.

- At a BMEP of 3 bar (low load), the maximum possible hydrogen energy share was 50.1% and above this energy share, engine was not stable and erratic engine operation was noticed. At a BMEP of 5 bar (medium load), the maximum energy share was dropped to 39.9%, since there was vigorous knocking above this energy share in HDDF mode.
- At low load condition, the BTE was not significantly change with HES, since the pre and main combustion effects were neutralize by each other and hydrogen was not effectively participated in combustion because of low in-cylinder temperature at low load operation. However, due to improvement in in-cylinder temperature at medium load condition, hydrogen may actively involved in combustion, thus the BTE was improved by 3.63% for initial HES rise as compared with neat diesel case at a BMEP of 5 bar.

- At a BMEP of 3 bar, the level of NO emission was decreased up to 18% as compared with CI mode, whereas smoke emission was lower for initial HES because of decreased diesel quantity but the combustion quality was affected as HES moves, this was because of reduction in oxidation rate of injected diesel fuel and lower heat release rate, this eventually end up with no significant reduction in smoke emission at highest HES. In spite of this, maximum reduction of HC, CO and CO_2 emissions were up to 62% 56% and 45% at maximum HES.
- At medium load condition without EGR, the brake specific NO emission was hiked up to 48% in HDDF mode when compared against CI mode. This was caused by active participation of hydrogen with improvement in in-cylinder temperature at this load condition. Smoke and CO_2 emissions were declined up to 20% and 24% while compared with CI mode. However, the brake specific HC emission was increased by 42% and there was no significant change in CO emission with HES.

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