An Experimental Investigation on Performance and Emissions of a Single Cylinder D.I Diesel Engine with Manifold Hydrogen Induction

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Abstract

Hydrogen is a clean fuel for internal combustion engines as it produces only water vapor and nitrogen oxides when it burns. In this research, hydrogen is used as a blending fuel with diesel to reduce pollutants emission and to improve performance. It is inducted in the inlet manifold, (continuous manifold induction), which is of a single cylinder, four stroke, direct injection, variable compression ratio water cold diesel engine, type (Kirloskar). This technique of hydrogen blending is selected because of its simplicity and low cost. Hydrogen blending is built on the basis of energy replacement. A special electronic unit is designed and fabricated to control hydrogen blending ratio. The maximum achieved ratio is 30% of input energy and beyond that the engine operation becomes unsatisfactory. Tests are done with 17.5 compression ratio and 1500 rpm. The brake specific fuel consumption is reduced by 29% and the engine thermal efficiency increased by 16% at these operating conditions. The pollutant emissions of carbon oxides, UHC, and smoke opacity are dramatically decreased by 19.5%, 13%,and 45% respectively while NO_x emission increased by 10%.

Keywords: Hydrogen, Blended Fuel, manifold induction , Engine Performance, Engine Emissions.

الخلاصة

الهيدروجين هو وقود خالي من الملوثات عندما يستخدم في محركات الاحتراق الداخلي لأنه ينتج بخار الماء وأكاسيد النيتروجين فقط. في هذا البحث، تم استخدام الهيدروجين بعد مزجه مع وقود الديزل للحد من الملوثات وتحسين الأداء. تم أدخال الهيدروجين في مشعب دخول المحرك، ذو اسطوانة واحدة، رباعي الأشواط، حقن مباشر، تبريد مائي، نوع (كيرلوسكار). أن مزج الهيدروجين مع الديزل مبني على أساس استبدال الطاقة. تم تصميم وحدة الكترونية خاصة للسيطرة على نسبة مزج الهيدروجين وضخه أثناء شوط السحب فقط. أن أقصى نسبة مزج تم تحقيقها هي 30% على أساس الطاقة وأكثر من تلك النسبة تجعل شغل المحرك غير مرضي. أجريت التجارب بنسبة انضغاط **17.5** وبسرعة **1500** دورة في الدقيقة. تم تقليل استهلاك الوقود المكبحي بنسبة **29%**، وزيادة في الكفاءة الحرارية للمحرك بنسبة **16%**. وانخفضت انبعاثات الملوثات مثل أكاسيد الكربون، الهيدروكربون غير المحترق، والدخان بشكل كبير بنسبة **19.5%**، **13%**، **45%** على التوالي، في حين حصلت زيادة في انبعاثات أكاسيد النيتروجين بنسبة **10%**.

كلمات مفتاحية: هيدروجين، وقود مخلوط، أستحثاث المشعب، أداء المحرك، أنبعاثات المحرك.

1. Introduction

Nowadays civilization is confronted with the twin crisis: the fossil fuel depletion and environmental degradation. The increase of fossil fuel consumption has led to reduction in underground based carbon resources which trigger the search for alternative fuels, that promises a harmonious correlation with sustainable development, energy conservation, management, efficiency, and environmental preservation. For the developing countries of the world, fuels of bio-origin can provide a feasible solution to the crisis [**Vinod *et.al.*, 2013**]. Hydrogen is a promising candidate as an alternative fuel. The use of hydrogen as a fuel in internal combustion engines has been studied by a number of researchers worldwide. A review on hydrogen supplementation in compression ignition engines was conducted by

[**Haroun and Emad ,2015]**. Compared with conventional fossil hydrocarbon fuels, hydrogen offers practically an elimination of pollutants such as carbon oxides and un-burnt hydrocarbons known to pose health risks in densely populated areas. The only nontrivial pollutant from hydrogen engines is nitrogen oxides (NO_x). However the characteristics of hydrogen fuel such as a high flame speed and extensive lean-burn operation possibilities allow significant reductions in NO_x as compared to conventional fuels **[Gomes *et.al.*, 2009]**. An experimental test on the use of hydrogen as a fuel in diesel engine, by adding the hydrogen in intake port at varied load and varying percentage of conventional fuel (diesel) (10%, 30%, 50%, 70%, 80% and 90%) by volume was undertaken by [**Saravanan and Nagarajan, 2008]**. They studied the performance and emission characteristics of the hydrogen-enriched engine. They plotted the relations between the brake thermal efficiency, brake specific fuel consumption, NO_X, HC, smoke and particulate emissions with the variation of engine load. The pressure and heat released with crank angle were also plotted. The results showed an increase in brake thermal efficiency of 18.35% with 30% hydrogen blending and a reduction of 50% in the amount of particulate matter and smoke.

The optimization of the injection timing, injection duration and injection quantity of the fuel in manifold and port injected hydrogen-operated engine using diesel as ignition source for hydrogen operation were carried out by **[Saravanan *et.al.,* 2010]**. From the results it was observed that in manifold injection technique the optimized start of injection was at gas exchange top dead center (GTDC) with injection duration of 30º CA with hydrogen flow rate of 7.5 L/min. In port injection technique, the optimized start of injection was at 5º before gas exchange top dead center (5ºBGTDC) with injection duration of 30º CA with hydrogen flow rate of 7.5 L/min. It was observed that the brake thermal efficiency increases in manifold injection rather than it does in port injection. **[Eiji *et.al.,* 2001]** conducted an experimental study on a single cylinder, four-stroke diesel engine operated in dual fuel mode. Hydrogen was inducted into the intake port along with air and diesel oil was injected into the cylinder. A wide range of injection timing was studied. When the injection timing advanced, the diesel oil well mixed with hydrogen air mixture and initial combustion became mild. NOx emissions decreased because of lean premixed combustion without the region of high temperature burned gas. Emissions such as CO, HC and CO_2 decreased without emition of smoke, while brake thermal efficiency was marginally lower than that in ordinary diesel combustion.

An experimental test was conducted on a four stroke, single cylinder, water cooled, direct injection (DI), diesel engine at a speed of 1500 rpm by [**Senthil *et.al.,* 2015]**. The hydrogen was inducted in the manifold with various volume flow rates namely 4 L/min, 6 L/min and 8 L/min respectively measured by digital volume flow meter. The engine performance, emission and combustion parameters were analyzed at various flow rates of hydrogen and compared with diesel fuel operation.**[Madhujit, 2012]** studied the combustion process and engine performance parameters on single cylinder, water cooled, four stroke, direct injection diesel engine with the rated power of 5.2 kW, a compression ratio of 17.5:1 and a rated speed of 1500 rpm at different loads (2, 4, 6, 8, 10, and 12 kg). The results showed that the brake thermal efficiency increased and brake specific fuel consumption decreased for all hydrogen injection strategies as compared to base diesel at all load operations. **[Masood *et.al.,* 2007]** discussed the effect of blending hydrogen with diesel in different proportions on combustion and emissions. A comparative study was carried out to analyze the effect of direct injection of hydrogen into the combustion chamber with that of induction through the inlet manifold for dual fueling. Percentage of hydrogen substitution varied from 20% to 80%, simultaneously reducing the diesel percentages. In a study by **[Gomes *et.al.,* 2008]** two techniques of hydrogen blending with diesel were used: port and direct techniques. The heating of air to 120 ^{o}C was applied on the direct injection technique. The heating of air which was much higher than the ambient temperature gave the best specifications of engine thermal parameters such as thermal efficiency that reached to the value of 42.8% as compared to the diesel value of 27.9%. This means that the percentage of increase in thermal efficiency realized is 35%. The heating process of air to 120 degree improves the thermal efficiency of the engine, and at the same time a significant reduction in the amount of air entering the manifold occurres and give the lowest level of volumetric efficiency of the engine. This problem can be treated by charging an extra amount of air through the use of supercharge.

The hydrogen injection into the intake manifold by using an injector with electronic control unit (ECU) was experimentally studied by **[Saravanan and Nagarajan; 2009]**. The injection timing and the duration were controlled. From the results it was observed that the optimum injection timing was at gas exchange top dead center (GTDC). The efficiency improved by about 15% with an increase in NO_X emission by 3% as compared to diesel. The smoke emission decreased by almost 100% with a decrease in carbon emissions due to the use of hydrogen. By adopting manifold injection technique the hydrogen–diesel dual fuel engine operates smoothly with a significant improvement in performance and reduction in emissions. [**Saravanan *et.al.,* 2008]** investigated, the effect of hydrogen-enriched air, as intake charge in a diesel engine adopting exhaust gas recirculation (EGR) technique with hydrogen flow rate at 20 L/min on engine performance. Experiments were conducted on a single-cylinder, four-stroke, water-cooled, direct-injection diesel engine coupled to electrical generator. Performance parameters were determined and emissions with exhaust gas temperature were measured. Using of hydrogen in dual fuel mode with EGR technique resulted in a lower smoke level, and NO_x emissions.[**Biplab *et.al.,* 2012]** studied the specific composition of diesel and hydrogen for maximum brake thermal efficiency at five different loading conditions (20%, 40%, 60%, 80% and 100% of full load) individually on the basis of maximum diesel substitution rate. At the same time, the effects on brake specific fuel consumption, brake specific energy consumption, volumetric efficiency and exhaust gas temperature were also observed at various liquid and gaseous fuel compositions for all the five loadings. It was seen that a diesel engine can be run efficiently in hydrogen-diesel dual fuel mode if the diesel to hydrogen ratio is kept at 40:60.

In present research , continuous manifold hydrogen induction with air is used in a single cylinder, four stroke, direct injection (DI), variable compression ratio diesel engine to study the effect of hydrogen blending on thermal performance and pollutant emissions at various engine loads. Hydrogen-diesel blending is done on energy replacement basis.

2. Experimental set-up

A single cylinder 4-stroke water-cooled direct injection diesel engine with a displacement volume of (553 cm^3), variable compression ratio, developing rated power of 3.7 kW at 1500 rpm with an electric dynamometer is used in the present research work. The engine is fitted with conventional fuel injection system, which has a three hole nozzle of 0.2 mm separated by 120^{o}, inclined at an angle of 60^{o} to the cylinder axis. The injector opening pressure recommended by the manufacturer was 120 bar.. The engine is coupled directly to the electric dynamometer. The details of the engine specification used in the investigation are listed in Table 1.

Fig (1) displays a schematic representation of the experimental model. The blended hydrogen is supplied from a hydrogen cylinder fitted with adjusting valves to control the hydrogen supply pressure and a digital hydrogen flow meter to measure the mass or volume flow rate of hydrogen. To protect the hydrogen system from any fire or from any explosion, some safety devices are added to the experimental work such as: flame trap, flame arrestor, and non-return valve. The exhaust temperature is measured by using thermometer of type PT 100. The mass flow rate of the diesel fuel is measured by using the digital balance for each five minute. A vertical manometer is used to measure the amount of pressure difference of air which exits from a surge tank and inlet to the manifold pipe. Plate (1) shows a front view image of the experimental set up.

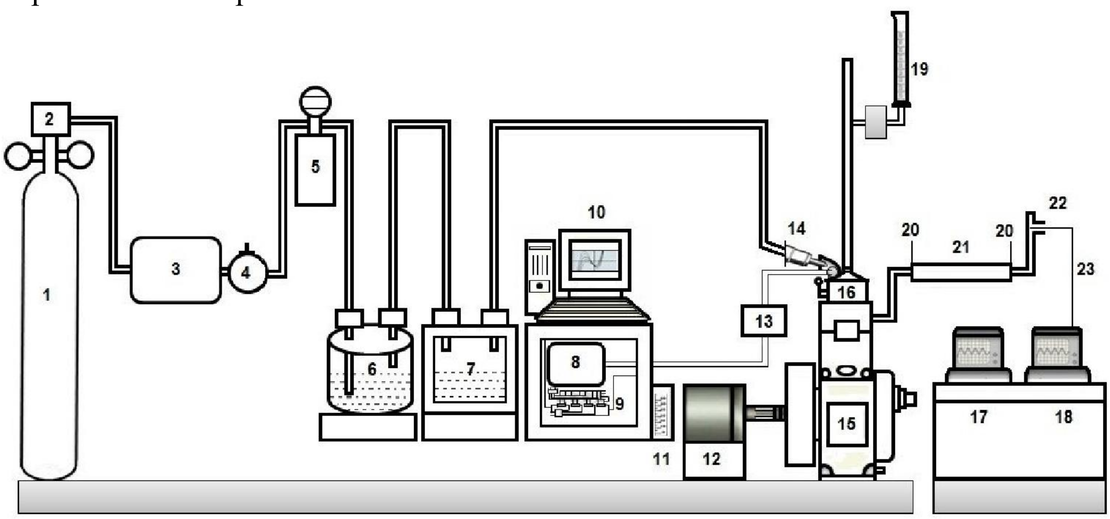

Fig 1 Schematic diagram of experimental setup.

1	Hydrogen cylinderhh	9	Data logger	17	Gas analyzer
2	Pressure regulator	10	PC to control DFC	18	Smoke meter (MED 001)
3	Hydrogen surge tank	11	Water manometer	19	Fuel tank
4	low control valve	12	Eddy current dynamometer	20	PT-100 sensor
5	Digital mass flow meter	13	Intake air	21	Calorimeter
6	Flame trap	14	Gas injector	22	Silencer tip
7	Flame arrester	15	Engine block	23	Exhaust gas probe
8	Air surge tank	16	Cylinder head		

Table 1 engine specification

Make and Model	Kirloskar AV-1
Type	Vertical, single cylinder, water cooled
Bore× Stroke	80 mm×110 mm
Swept volume	$0.553x10^{-3}$ m^3
rated power	3.7 kW at 1500 rpm
Compression ratio	12.5-17.5
Static injection timing	35 BTDC
Injection pressure	160 bar
Density of fuel	0.8275
Start of Injection	165 ^oCA
End of Injection	200 ^oCA
Injection Duration	35 ^oCA

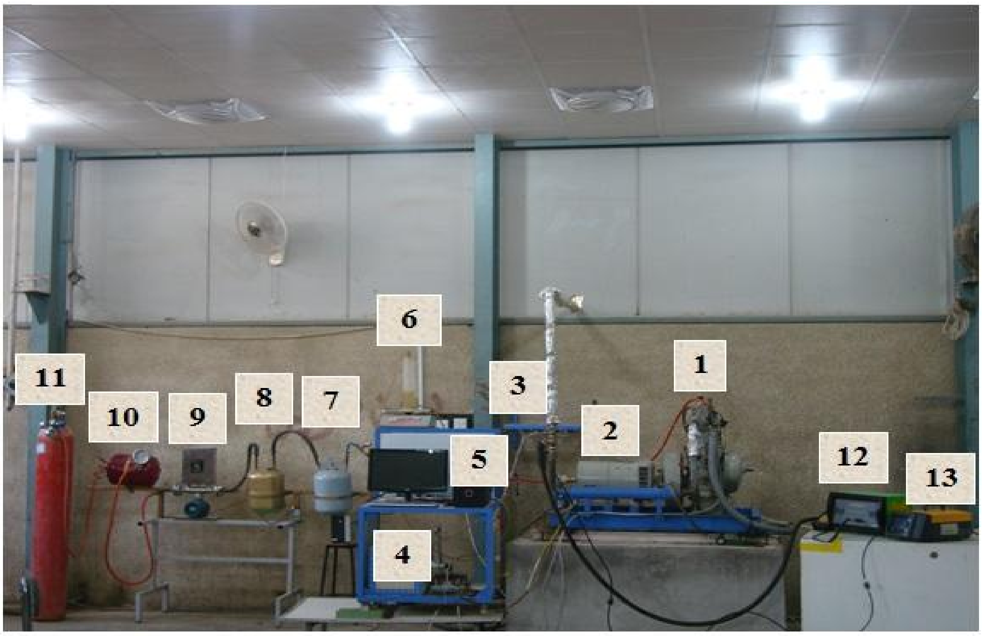

Plate 1 Front View of the Experimental Set Up

1	Single Cylinder Diesel Engine	8	Flame Trap
2	Electrical Dynamometer	9	Digital H_2 Mass Flowmeter
3	Bank of Heaters	10	Surge Tank
4	Data Logger	11	Hydrogen Cylinder
5	PC	12	Gas Analyzer
6	Fuel Tank	13	Smoke Meter
7	Flame Arrester		

3. Hydrogen induction system

Fig (2) presents a schematic representation of the hydrogen induction points through the air intake manifold of the diesel engine. Hydrogen is induced continuously into the air intake manifold from the hydrogen cylinder and is allowed to mix with the air in the manifold before the air-hydrogen mixture is admitted in to the engine cylinder. The hydrogen flow rate is regulated using a fine valve which allows the flow rate to be adjusted in steps at all speeds and loads. Two nozzles of 0.225 mm diameter with 5 holes are used for the purpose of controlling the percentage of the hydrogen induction into the intake manifold at high blending ratios. The first nozzle was used until 30% hydrogen blending ratio. Both nozzles operate when the required blending ratio exceeds 30%. They were located diagonally at 40^o, 20^o respectively. These two nozzles are directly connected to an electronic control unit (injection control system) specially designed for this purpose. This unit is used to control the induction of hydrogen into the combustion chamber during the intake stroke only. Figure (3) illustrates the block diagram of hydrogen injection control system.

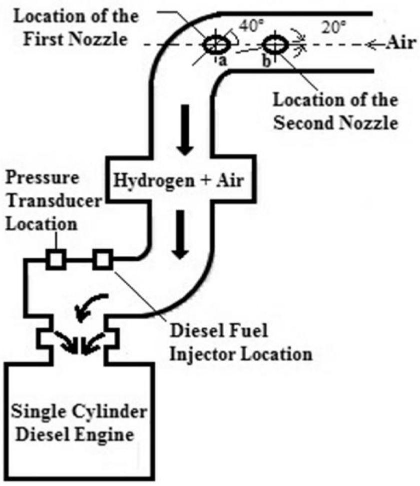

Fig 2 Locations of Continuous Hydrogen Induction Points in Intake Manifold

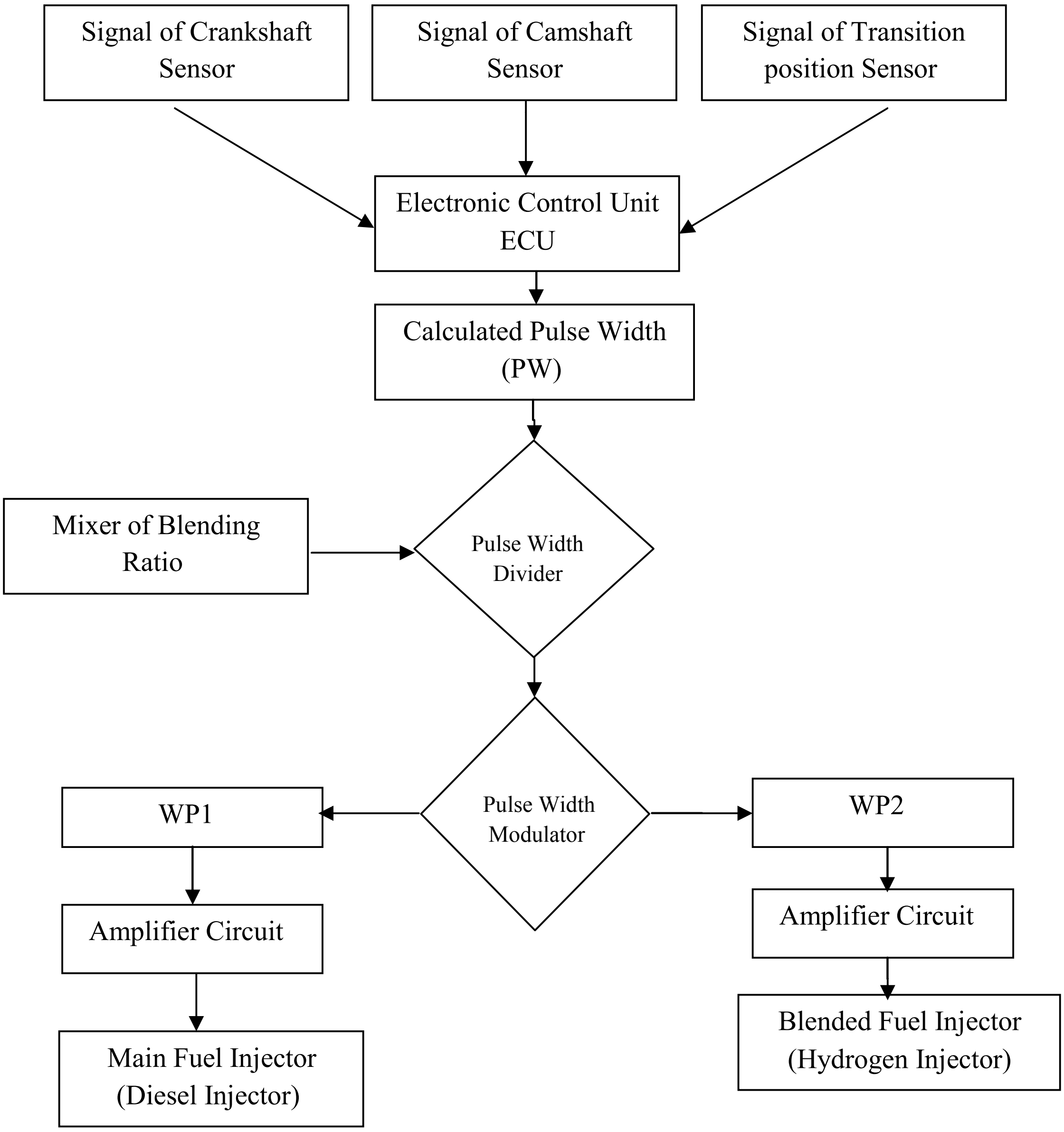

Fig. (3): Block Diagram of Hydrogen Injection Control System

4. Results and discussion

In the present investigation the effect hydrogen blending with diesel on engine performance parameters and harmful emissions is analyzed. Tests are carried out at 1500 rpm and 17.5 compression ratio at different loads with fixed diesel injection timing of -35^{o} CA. Different hydrogen blending ratios are studied namely (0%, 10%, 20%, and 30%). The study shows that with higher hydrogen blending ratio, more than 30%, engine operation becomes unstable which means engine modification is needed. The discussion of the results falls in three parts:

4.1 Performance analysis

The studied engine performance parameters are: brake thermal efficiency, brake specific fuel consumption, brake specific energy consumption, volumetric efficiency and equivalence ratio.

•Brake thermal efficiency ($\eta_{b.th}$)

The brake thermal efficiency is calculated as :

$$\eta_{b.th} = \frac{\text{Brake Power}}{(\dot{m} \times LCV)_{diesel} + (\dot{m} \times LCV)_{H_2}} \quad 1$$

The variation of brake thermal efficiency with load for different hydrogen blending ratios at a compression ratio of (17.5) is shown in Fig (4).

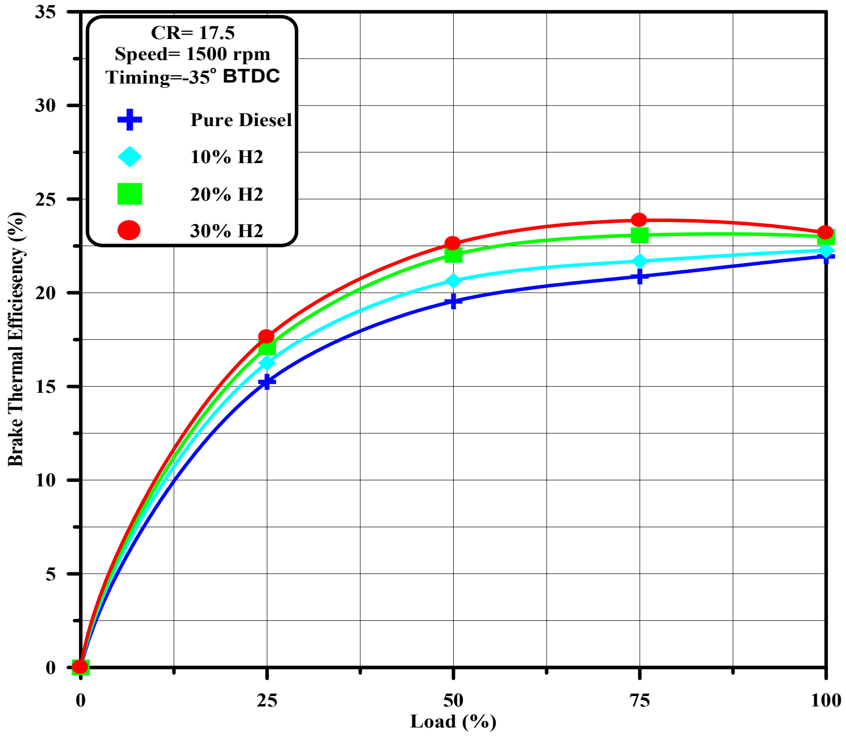

Fig 4 Variation of Brake Thermal Efficiency with Load for Different Hydrogen Blending Ratios

It is seen that, the brake thermal efficiency increases with the increase in the percentage of hydrogen addition. This is due to the improvement of combustion process caused by better mixing of hydrogen with air in addition to faster burning rate characteristics of hydrogen which improves combustion process of diesel fuel and leads to a shorter combustion duration and hence a higher cylinder temperature . At 75% load, there is 13.6% increase in brake thermal efficiency with 30% hydrogen blending ratio as compared to the pure diesel fuel mode.

• Brake Specific Fuel Consumption (BSFC)

The brake specific fuel consumption is calculated as :

$$BSFC = \left(\frac{\dot{m}_{diesel} + \dot{m}_{H_2}}{\text{Brake Power}}\right) \times 3600 \qquad \left(\frac{kg}{kW.hr}\right) \quad 2$$

Fig (5) portrays the variation of brake specific fuel consumption with load of different hydrogen blending ratios at compression ratio (17.5). The brake specific fuel consumption decreases with the increase in hydrogen blending ratio. The lowest brake specific fuel consumption of 0.29 kJ/kW-hr is obtained for 30% hydrogen enrichment at 75% load as compared to the pure diesel mode of 0.41 kJ/kW-hr when the engine is running at 1500 rpm. This is due to a better mixing of blended hydrogen with air which results in better combustion of diesel fuel and improvement of combustion process efficiency. The percentage of observed reduction of BSFC was 30% at full load at 30% hydrogen blending ratio. This is observed to be true for loadings of 25%, 50%, and 75%.

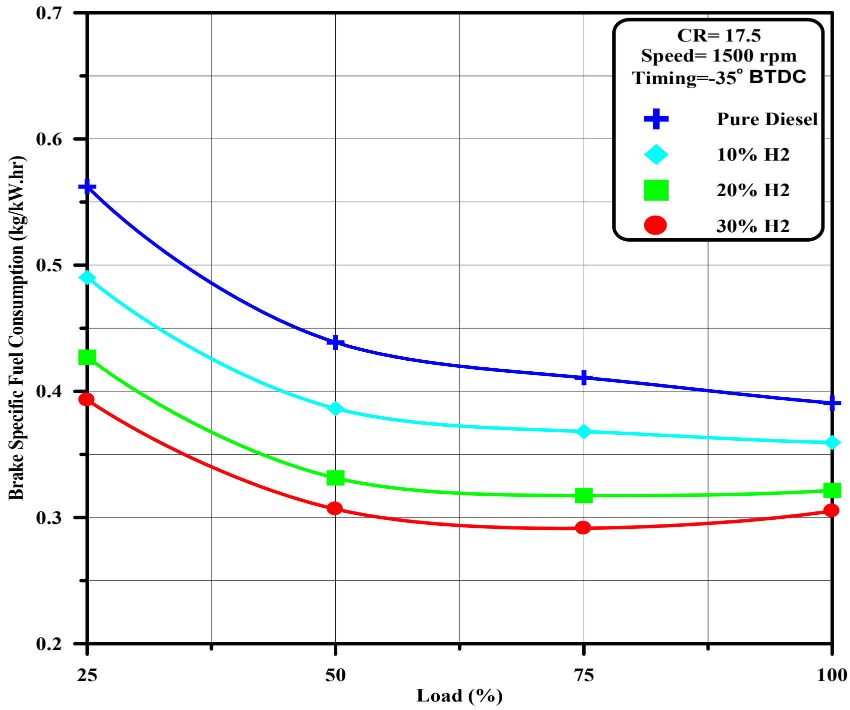

Fig 5 Variation of brake specific fuel consumption with Load for Different Hydrogen Blending Ratios

• Brake Specific Energy Consumption (BSEC)

The brake specific energy consumption is calculated as :

$$BSEC = \frac{(\dot{m} \times LCV)_{diesel} + (\dot{m} \times LCV)_{H_2}}{\text{Brake Power}} \qquad \left(\frac{kJ}{kW.sec}\right) \qquad 3$$

The variation of brake specific energy consumption with load for different hydrogen blending ratios and compression ratio (17.5) is given in Fig (6). It is observed that the inducement of hydrogen with inlet air to manifold reduces the consumption of the energy to offer a unit applicable energy. The decrease in BSEC is more important, when the hydrogen is blended with diesel, at full load and speed of 1500 rpm. This is due to more efficient combustion of diesel fuel. For a given constant speed and constant hydrogen blending ratios, (30%), the brake specific energy consumption is reduces to 2.5 times at full load.

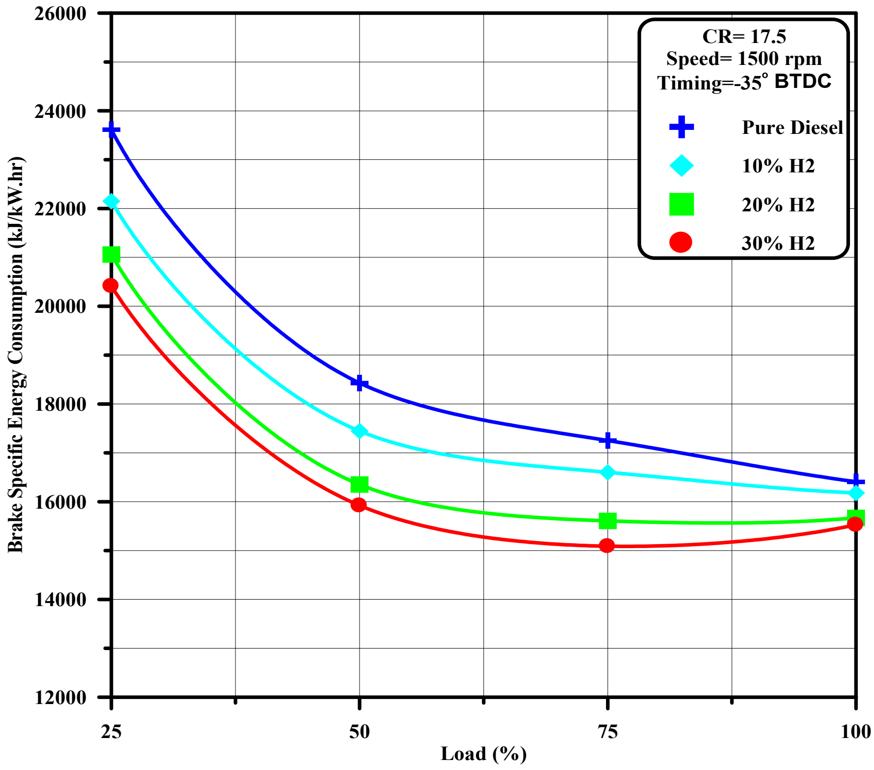

Fig 6 Variation of brake specific Energy consumption with Load for Different Hydrogen Blending Ratios

- **Volumetric Efficiency (η_{Vol})**

The volumetric efficiency is calculated as :

$$\eta_{Vol} = \frac{V_{actual}}{V_{swept}} \qquad 4$$

$$V_{actual} = C_d \times A_{oriface} \times \sqrt{2g \times h_{air}} \qquad \left(\frac{m^3}{sec}\right) \qquad 5$$

C_d = Discharge coefficient (0.65 in the present work)

$$A_{oriface} = \frac{\pi}{4}(d_{oriface})^2 \qquad 6$$

$$h_{air} = \frac{h_w \rho_w}{\rho_{air}} \qquad 7$$

$$V_{swept} = \frac{\pi}{4}\ (Bore)^2 \times Stroke \times \frac{N}{2 \times 60} \times Nomber\ of\ Cylinder \qquad \left(\frac{m^3}{sec}\right) \qquad 8$$

The influence of hydrogen blending on volumetric efficiency is shown in Fig (7). It is noted that the actual air volume sucked changes with load and with hydrogen blending ratio. The decrease of volumetric efficiency with load is due to higher cylinder walls temperature at higher loads while the decrease due to hydrogen blending is due to the displacement of part of incoming air by hydrogen gas. Thus, the higher hydrogen induction rate results in the decrease of air intake which leads to incomplete combustion of the fuel blend in combustion chamber. A reduction of 6.118% in volumetric efficiency is observed with 30% hydrogen blending under no load operation at 1500 rpm and 17.5 compression ratio. The reduction increases to about 42.6% when the load applied is 100% with engine running at the same conditions.

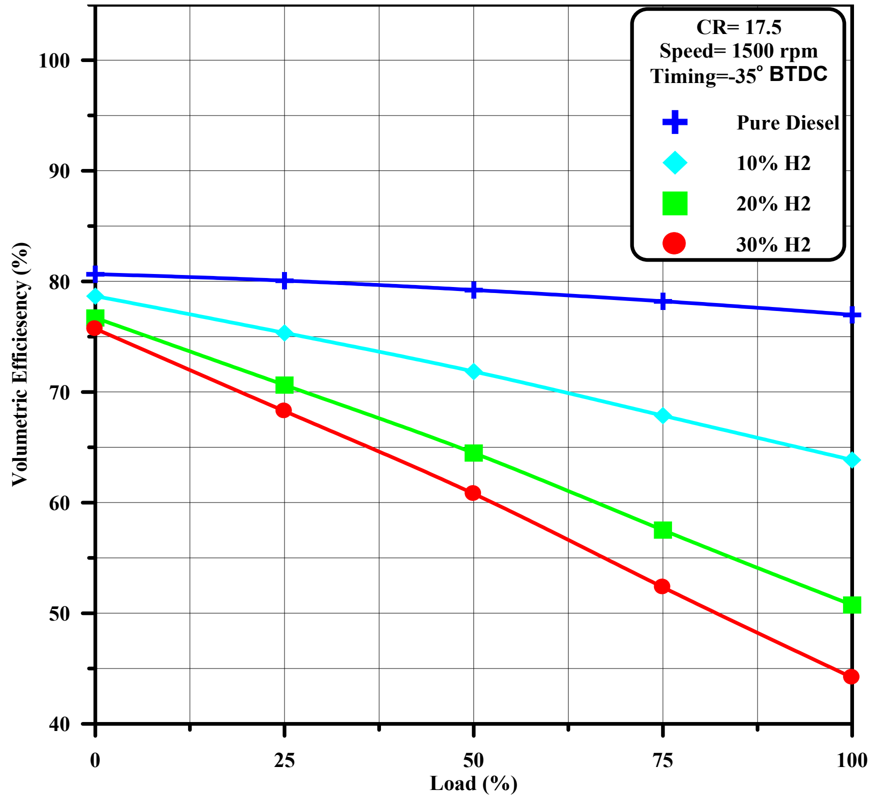

Fig 7 Variation of Volumetric Efficiency with Load for Different Hydrogen Blending Ratios

- **Equivalence Ratio** $(\emptyset)$

For the blending of hydrocarbon with hydrogen fuel, the equivalence ratio was calculated from the following formula **[3]**:

$$\emptyset = \frac{\left(\dfrac{[m_f]}{[m_a] - \dfrac{[m_{H_2}]}{\left(\dfrac{m_{H_2}}{m_a}\right)_{Sto.}}}\right)}{\left(\dfrac{[m_f]}{[m_a]}\right)_{Sto.}} \qquad 9$$

Where, $\emptyset$ is the overall equivalence ratio, $[m_f]$, $[m_a]$, and $[m_{H_2}]$ are the molar concentrations of diesel, air, and hydrogen respectively. Subscript 'st' stands for stoichiometric. Fig (8) illustrates the variation of equivalence ratio with load for different hydrogen blending ratios at a compression ratio of (17.5) and a speed of 1500 rpm. It is seen that the overall equivalence ratio decreases with hydrogen induction, especially at no load, low load and medium load conditions, and makes the charge in the cylinder leaner since the blending is done on energy replacement basis. However the overall equivalence ratio increases with the increase in hydrogen blending ratio at high and full load conditions due to the significant drop in volumetric efficiency and larger amount of injected diesel fuel. At 100% load, there is

35.18% increasing in equivalence ratio when the hydrogen blending ratio is 30% as compared to the pure diesel fuel operation.

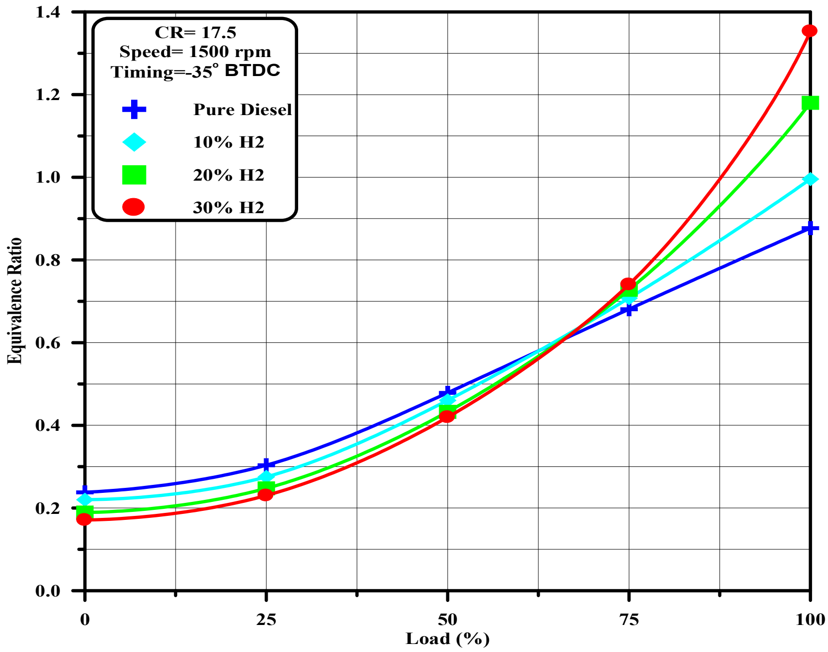

Fig. 8 Variation of Equivalence Ratio with Load for Different Hydrogen Blending Ratios

4.2 Combustion Evaluation

- **Cylinder Pressure**

The Variation of In-cylinder pressure with crank angle for different hydrogen blending ratios at1500 rpm and (17.5) compression ratio is shown in Fig (9).The start of diesel injection timing is fixed for both pure diesel and all strategy of hydrogen blending at -35^o CA. It is found that the duration of diesel injection decreases as the hydrogen blending ratio increases. This is due to the reduction in the amount of diesel injected. The end of injection for pure diesel occurs at 200^o CA, while that with hydrogen blending ratios of 10%, 20%, and 30% occurs at 195^o CA 190^o CA and 185^o CA respectively. It is also noticed that the maximum cylinder pressure increases with hydrogen blending ratio and gets closer to TDC as it is shown in Fig (9). This means that the duration of combustion gets shorter with higher blending ratios due to the better combustion process caused by the presence of hydrogen. As the hydrogen blending increases the engine cycle gets closer to Otto cycle which means higher thermal efficiency. The blending of hydrogen with 30% ratio gives the highest peak pressure of 60 bar as compared with pure diesel of 52.5 bar.

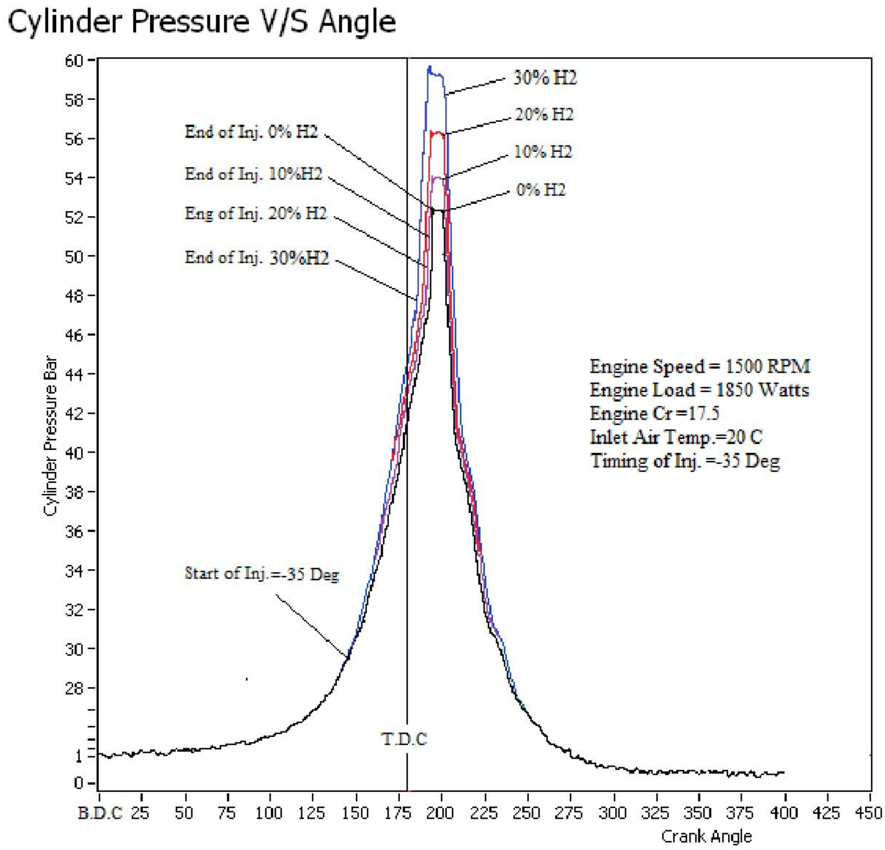

Fig **9** Variation of In-cylinder pressure with Crank angle for Different Hydrogen Blending Ratios.

- **Exhaust Temperature**

Fig (10) portrays the variation of exhaust gas temperature with load for different hydrogen blending ratios and speeds of 1500 rpm with compression ratio (17.5). The highest exhaust gas temperature (387^{o} C) is obtained at full load with 30% hydrogen blending as compared to (340^{o} C) for pure diesel. The increase of hydrogen blending results in an increase in the exhaust temperature due to the increase in peak cylinder temperature and pressure, which results from faster combustion caused by hydrogen combustion.

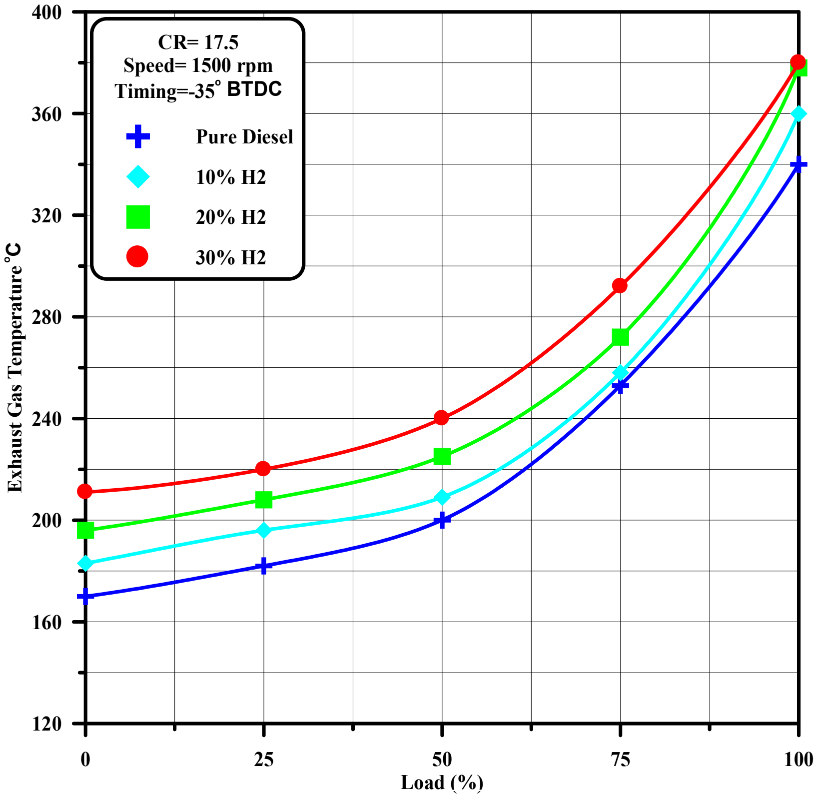

Fig 10 Variation of Exhaust Gas Temperature with Load for Different Hydrogen Blending Ratios

4.3 Emission Analysis

The effect of hydrogen blending with diesel fuel on the pollutant of the different gases of exhaust engine runs at 1500 rpm with 17.5 compression ratio at different loads are investigated in the present experimental study.Figs (11-16)depict the effect of hydrogen blending on the pollutants concentration namely carbon monoxide, carbon dioxide, unburned hydrocarbon, NO_X, and O_2. It is observed, From Fig (11),that the blending ratio of hydrogen has an important influence on carbon monoxide emission. The presence of hydrogen has two opposing effects on CO emission. It reduces the amount of carbon atoms in the cylinder which should reduce the CO concentration and reduces the amount of air which should increase the CO concentration. Fig(11)shows the net effect of both factors. However, it is noticed that at hydrogen blending ratio of 30%,the percentage of CO reduction was 18.18% at full load.Carbon monoxide concentration is also observed at other loads as shown in Fig 11.

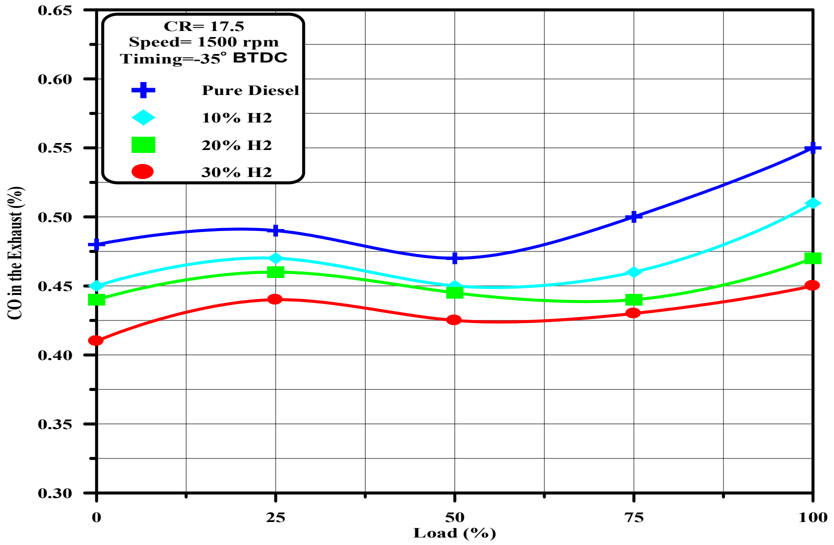

Fig 11 Variation of Carbon Monoxide with Load for Different Hydrogen Blending Ratios

The variation of CO_2 emission with load of different hydrogen blending ratios and speed of 1500 rpm with compression ratio (17.5) is given in Fig 12. The CO_2 emission in case of hydrogen enrichment is lowered because of better combustion characteristics of hydrogen fuel and the improved combustion efficiency of diesel fuel. It is also due to the absence of carbon atom in hydrogen molecule, which reduces the amount of carbon in the cylinder. A reduction of 13.25% in CO_2 concentration at full load is noticed for hydrogen operation with 30% blending ratio while at no load the reduction is about 35.48%. Similar trends were observed at other loads.

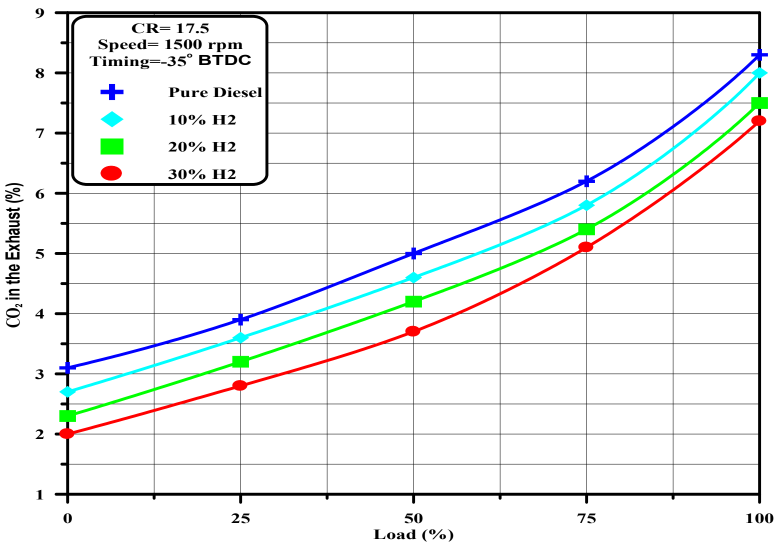

Fig 12 Variation of Carbon Dioxide with Load for Different Hydrogen Blending Ratios

Fig (13) portrays the alteration of hydrogen blending on unburned hydrocarbon UBHC emission. It is observed that the reduction in UBHC is about 18.75% at 30% hydrogen blending ratio for no load while the reduction is 14.94253% at full load.

Organic emissions (UBHC), are the result of imperfect combustion of the hydrocarbon fuel. The standard unit of UBHC in the exhaust gases is measured by parts per million (ppm) or volume percentage (as in current study). UBHC is considered as a measure of the inefficiency of the engine, but it is not considered as an important index of the polluting emissions. The reduction in UBHC emission in the case of hydrogen blending is due to the absence of carbon in hydrogen and the reduction of injected diesel fuel which means less carbon atoms in the cylinder and also better combustion . Further the burning of hydrogen enhances the temperature and leads to more complete oxidation of the hydrocarbons formed from injected fuel. The hydrogen blending ratio of 30% gives the lowest UBHC of value 74% as compared with the diesel of 83%.

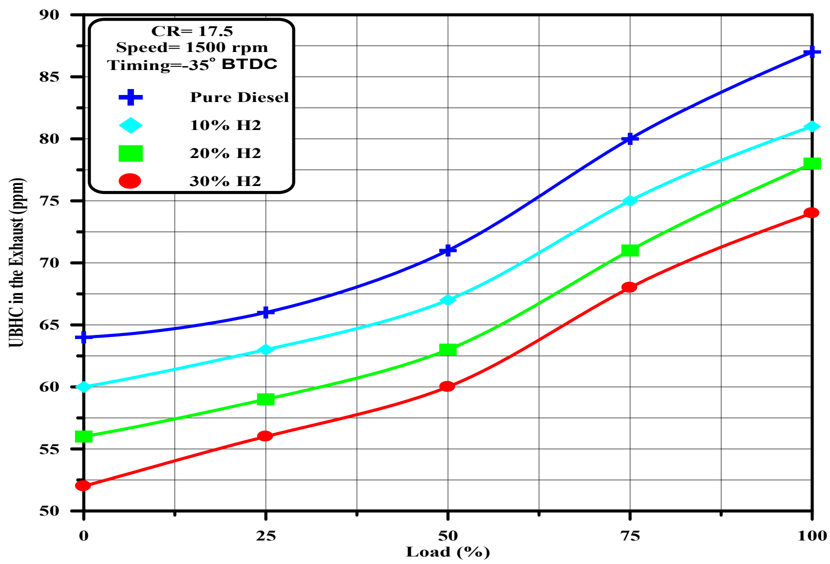

Fig 13 Variation of UBHC with Load for Different Hydrogen Blending Ratios

Fig (14) shows the effect of hydrogen blending ratio on NO_x emission. It is seen that there is about 10%-13% increase in NO_x emission at almost all loading conditions from no load to full load at 30% hydrogen blending. The reason for this higher concentration of NO_x in case of hydrogen enrichment is the increase in cylinder temperature since NO_x formation mechanism is strongly temperature dependent. This increase in cylinder temperature is due to the high heating value of hydrogen as compared to diesel fuel. The presence of hydrogen also improves the burning velocity of fuel which means shorter combustion duration and hence higher cylinder temperature.

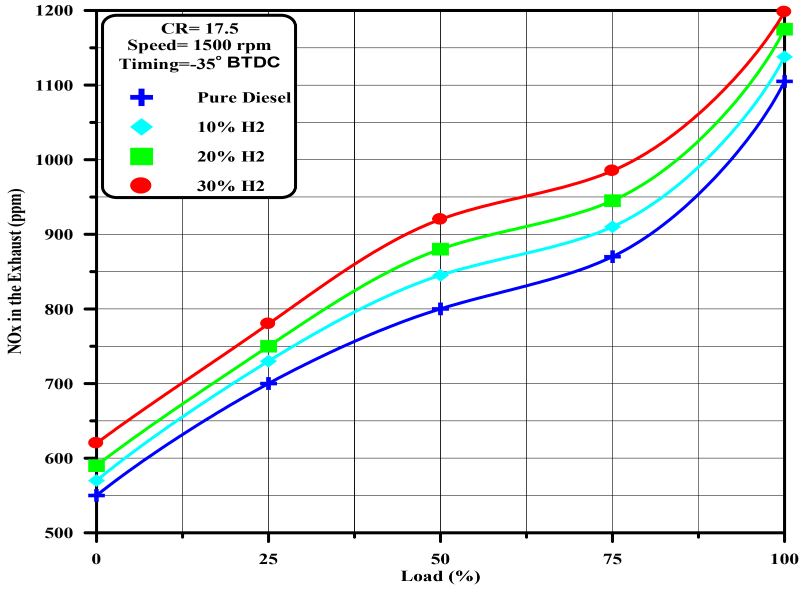

Fig 14 Variation of NO_x with Load for Different Hydrogen Blending Ratios

Diesel particulates or smoke consists principally of combustion-generated carbonaceous materials (soot) on which some organic compounds are absorbed. Most particulate material results from incomplete combustion of hydrocarbon fuel. The variation of smoke opacity with load of different hydrogen blending ratios is given in Fig 15. It can be noticed that the lowest smoke opacity of 4.5% is obtained for 30% hydrogen enrichment at no load as compared to pure diesel of 7.5%. The reduction in smoke opacity is 40%. The maximum reduction in smoke opacity observed is 65.8 at full load and 30% hydrogen enrichment. This reduction in opacity is attributed to the better mixing of fuel and air and hence combustion process caused by the presence of hydrogen. It is also caused by lower total carbon atoms present in the cylinder due to the replacement of diesel fuel by hydrogen.

Fig (16) depicts the influence of hydrogen blending on O_2 emission. It can be seen that the percentage of O_2 in the exhaust decreases with an increase in hydrogen blending ratio. A reduction of 16.67% to 21.62% was observed for different loading from the condition of 0 to 30% of hydrogen blending ratio. The increase in hydrogen induction ratio reduces the actual volume of air drawn which results in the reduction of overall O_2 content in the engine which leads to the reduction in the content of O_2 in the exhaust gases. It is also due to the fact that hydrogen consumes more oxygen during combustion than diesel does.

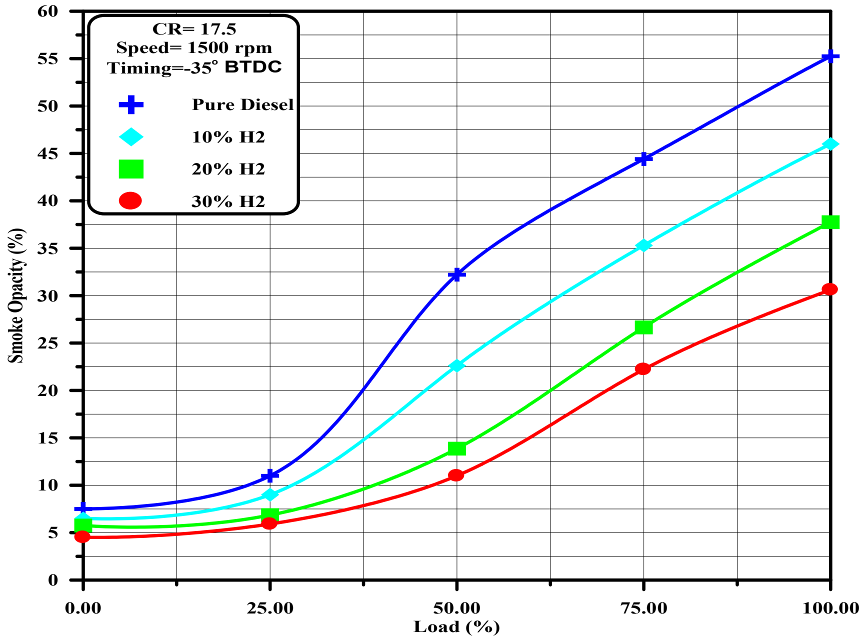

Fig 15 Variation of smoke opacity with Load for Different Hydrogen Blending Ratios

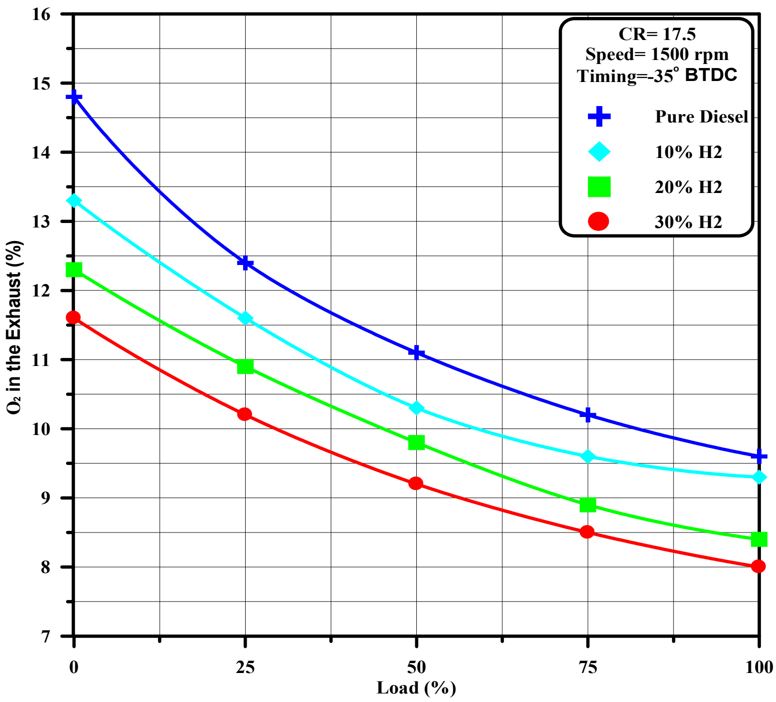

Fig 16 Variation of O_2 with Load for Different Hydrogen Blending Ratios

5. Conclusions

Depending on the results mentioned earlier, we come up with the following conclusions:

1- Hydrogen blending ratio at 30% approximately increases brake thermal efficiency by 12.5% at 75% load while the diesel fuel consumption decreases by 30% at full load.

2- At 30% hydrogen blending ratio, there is a significant reduction in volumetric efficiency of about 42.6% at full load conditions.

3- Hydrogen blending raises the maximum in cylinder pressure and exhaust gas temperature.

4- Hydrogen combustion produces high cylinder temperatures and hence high nitrogen oxides. Therefore, it is necessary to control NO_x formation by controlling the mixture concentration.

5- The maximum reduction in pollutants emissions of CO, CO_2 and UBHC, are 19.5%, 39%, and 16% respectively with hydrogen blending ratio of 30%.

6- A maximum reduction in smoke opacity is 42% with 30% hydrogen blending ratio.

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