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EFFECT OF OPERATING CONDITIONS ON PERFORMANCE AND EMISSIONS OF A DIESEL ENGINE OPERATED WITH DIESEL-HYDROGEN BLEND

Haroun A.K. Shahad[1] Emad D. Abood[2]

hakshahad@yahoo.com edashasfor@yahoo.com

[1] Department of Mechanical Engineering, College of Engineering University of Babylon

[2] Department of Automobile Engineering, College of Engineering/Al-Musaib, University of Babylon

ABSTRACT

Hydrogen is a clean fuel for internal combustion engines since 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, of a single cylinder, four stroke, direct injection, water cold diesel engine, type (Kirloskar). Hydrogen blending is done on energy replacement basis. 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 engine operation becomes unsatisfactory when the air temperature is 20 ^oC and injection timing of -35^o CA which represent the first part of this work. Inlet air heating system is built and added in the experimental work. The heating system allows to increase the air temperature up to 100 ^oC. A heating of air to 60 ^oC with injection timing of -30^o CA and 55% of hydrogen blending is executed in the second part of this study. Tests are done with 17.5 compression ratio and 1500 rpm. The brake specific fuel consumption is reduced by 29% and 46%, the engine thermal efficiency is increased with 16% and 21% for the 1st and 2nd part respectively. The pollutant emissions of carbon oxides, UHC, and smoke opacity are dramatically decreased by 19.5%, 13%, and 45% respectively for the 1st part and 41%, 38% and 65.6% for the 2nd part while NO_x emission is increased by 10% and 25% for the 1st and 2nd part respectively.

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

دراسة تأثير ظروف التشغيل على الأداء والانبعاثات من محركات الديزل التي تعمل مع مزيج الديزل والهيدروجين

الخلاصة

الهيدروجين هو وقود خالي من الملوثات عندما يســتخدم 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% عند درجة حرارة للهواء الداخل **20 ^oC** وتوقيت الحقن **-35^o CA** التي تمثل الجزء الأول من هذا العمل. تم بناء منظومة لتدفئة الهواء، وأضيفت ألى العمل التجريبي. نظام التدفئة يسمح لزيادة درجة حرارة الهواء لتصل إلى **100 ^oC**. تم تسخين الهواء إلى **60 ^oC** مع توقيت حقن **-30^o CA** و نسبة مزج للهيدروجين **55%** ,ونفذت في الجزء الثاني من هذه الدراسة. أجريت التجارب بنسبة انضغاط **17.5** وبسرعة **1500** دورة في الدقيقة. تم تقليل استهلاك الوقود المكبحي بنسبة **29%** و **46%**، وزيادة في الكفاءة الحرارية للمحرك بنسبة **16%** و **21%** للجزء الأول والثاني على التوالي.

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وانخفضت انبعاثات الملوثات مثل أكاسيد الكربون، الهيدروكربون غير المحترق، والدخان بشكل كبير بنسبة **19.5%**, **13%**, **45%** على التوالي بالنسبة للجزء الأول و **41%**, **38%**, **65.6%** على التوالي بالنسبة للجزء الثاني من العمل التجريبي في حين حصلت زيادة في انبعاثات أكاسيد النيتروجين بنسبة **10%** و **25%** للجزئيين الأول والثاني على التوالي.

INTRODUCTION

Civilization, is presently 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, which 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 Singh 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 A.K. Shahad and Emad D. Aboud (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 compared with conventional fuels, **J.M. Gomes Antunes, 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. Also, the pressure and heat released with crank angle were 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 **N.Saravanan, G.Nagarajan, (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 brake thermal efficiency increases in manifold injection rather than in port injection. **Eiji Tomita 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 was advanced, the diesel oil was 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 emitting 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, **R. Senthil Kumar 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 Deb, (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 was increased and brake specific fuel consumption was decreased for all hydrogen injection strategies 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. **J.M. Gomes Antunes et al, (2008).** Two techniques of hydrogen blending with diesel were used, in this study, which are port and direct technique. The heating of air to 120 ^{o}C was applied on the direct injection technique. The heating of air 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 diesel value of 27.9%. This means that the percentage of increase in thermal efficiency which 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 is occurred and give lowest level of volumetric efficiency of the engine. This problem can be treated by charging an extra amount of air with using 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% compared to diesel. The smoke emission decreased by almost 100% with 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 lower smoke level, particulate and NO_x emissions. **Biplab K. Debnath 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.

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 a 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 which used in the investigation are listed in Table 1. Figure (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 minutes. 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.

HEATING SYSTEM DESIGN

For improving the thermal performance along with minimum pollutants emissions, a new design system of heating inlet air to the manifold was used as shown in Plate (2) .

The purpose of heating system is to heat air at different temperature such as 60 ^oC, 80 ^oC, and 100 ^oC by using the loading system of the engine and take the advantage of four heaters that release heat to the atmospheric air. To obtain the desired air temperature, it is used both simple control system with valve regulator and by opened and closed them, the excess heat leaking out of the system.

HYDROGEN INDUCTION SYSTEM

Figure (2) gives 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) illustrate the block diagram of hydrogen injection control system.

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 for 20 ^oC of air inlet and -30^o CA for 60 ^oC of air inlet. Different hydrogen blending ratios are studied namely (0%, 10%, 20%, 30%) and (0%, 10%, 20%, 30%, 40%, 50%, 55%) for the two cases respectively. The study shows that with higher hydrogen blending ratio, more than 30% for the case one, engine operation becomes unstable which means engine modification is needed. Same trend is observed in the second stage with higher hydrogen blending ratio, more than 55%. The results and discussion is divided into three parts.

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) with and without heating is shown in Figure(4).

It is seen that, the brake thermal efficiency increases with 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 leading to shorter combustion duration and hence higher cylinder temperature. At 75% load, there is 13.6% increase in brake thermal efficiency with 30% hydrogen blending ratio compared with pure diesel fuel mode. The brake thermal efficiency increases with increase in the inlet air temperature to 60 ^{o}C. This is due to less time for the first phase of combustion for the fuel evaporate, leading to reducing the energy required for the evaporation of fuel which gives fast burning in the second phase of combustion. The brake thermal efficiency of 28% is observed with 55% hydrogen blending ratio at 50% load while the maximum of (30%) at 100% load is obtained for (30%) hydrogen enrichment compared to diesel of (28%).

- **Brake Specific Fuel Consumption (BSFC)**

The brake specific fuel consumption is calculated as :

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

Figure(5) portrays the variation of brake specific fuel consumption with load for different hydrogen blending ratios at compression ratio (17.5) for heating and without heating. 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 compared to pure diesel mode of 0.41 kJ/kW-hr. This is due to 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 with 30% hydrogen blending ratio. At 55% blending it is possible to increase the load to 50% only. Further increase in load results in deterioration of engine operation due to knock. The brake specific fuel consumption decreases with an increase in hydrogen blending ratio with air heating at 60 ^{o}C. The lowest brake specific fuel consumption of 0.196 kJ/kW-hr is obtained for 55% hydrogen blending ratio at 50% load compared to pure diesel of 0.37 kJ/kW-hr. Air heating process significantly contributed to increase the blending ratio of hydrogen to become 55%. This results to advance the injection timing to -30° BTDC and decrease the ignition delay period.

- **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) with and without heating is given in Figure(6). It is observed that the inducement of hydrogen with inlet air to manifold reduces the consuming 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 ratio (30%), the brake specific energy consumption is reduces to 2.5 times at full load._The maximum reduction of BSEC was observed to be 15.535% at 25% load between 55% hydrogen blending ratio and pure diesel when the engine running at 1500 rpm.

- **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)

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

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

$$\dot{V}_{swept} = \frac{\pi}{4}(\text{Bore})^2 \times \text{Stroke} \times \frac{N}{2 \times 60} \times \text{Nomber of Cylinder} \qquad (8)$$

The influence of hydrogen blending on volumetric efficiency with and without heating is shown in Figure(7). It is noted that the 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 resulting 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.

The increased of hydrogen blending ratio to 55% as a results of heating process of inlet air to the manifold leads to more decrease in air intake as well as the heating of inlet air leads to a decrease of the actual volume of air inside the cylinder, as the volumetric efficiency represents the ratio of the actual volume of air to swept volume so the volumetric efficiency will decrease more. The lowest volumetric efficiency of 40.139 is observed with 55% hydrogen blending ratio at 50% load.

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

For the blending of hydrocarbon with hydrogen fuel, the equivalence ratio was calculated from the following formula **[Gomes et al. 2009]**:

$$\emptyset = \frac{\left(\dfrac{[m_f]}{[m_a] - \dfrac{[m_{H_2}]}{\left(\dfrac{m_{H_2}}{m_a}\right)_{Sto.}}}\right)}{\left(\dfrac{[m_f]}{[m_a]}\right)_{Sto.}} \tag{8}$$

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. Figure(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 with and without heating. 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 liner 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 with pure diesel fuel operation.

The heating process dramatically decreases the air density inside the combustion chamber and thus led to an increase in the equivalence ratio. A maximum reduction in the equivalence ratio obtained was 57.9% at 50% load with engine running at 1700 rpm when the hydrogen blending ratio reaches 55%.

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 when the inlet air at 20 ^oC is shown in Figure(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 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 shown in Figure(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. The Variation of In-cylinder pressure with Crank Angle for different hydrogen blending ratios and speed of 1500 rpm at a compression ratio of (17.5) is shown in Figure(10). The results showed that with inlet air heated to 60 ^oC it is possible to increase the hydrogen blending ratio to 55% but the injection timing should be advanced to -30^o BTDC with satisfactory engine operation and to avoid the occurrence of a top of the curve after the ATDC. The blending of hydrogen with 55% ratio gives the highest peak pressure of 71 bar as compared with pure diesel of 57.5 bar.

Figure(11) portrays the variation of exhaust gas temperature with load for different hydrogen blending ratios and compression ratio (17.5) with and without heating. The highest exhaust gas temperature (387^o C) is obtained at full load with 30% hydrogen blending compared to (340^o C) for pure diesel. The increase of hydrogen blending gives an increase in the exhaust temperature due to the increase in peak cylinder temperature and pressure, which is due to faster combustion caused by hydrogen combustion. Exhaust gas temperature of (403 ^oC) at full load is obtained for (55%) hydrogen enrichment compared to diesel of (232 ^oC) when the engine running at 1700 rpm and the air inlet temperature is 60 ^oC.

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 and at different loads are investigated in the present experimental study. Figures (12-17) 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 Figure (12), 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. Figure(12) 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 Figure(12) with and without heating. The maximum percentage of reduction of CO is observed 82.3% at 25% load when the inlet air temperature is 60 ^oC with hydrogen blending ratio increased from 0 to 55%. The variation of CO_2 emission with load for different hydrogen blending ratios and speed of 1500 rpm with compression ratio (17.5) with and without heating is given in Figure(13). The CO_2 emission in case of hydrogen enrichment is lowered because of better combustion characteristics of hydrogen fuel and 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. A maximum reduction of 48.9% in CO_2 concentration at no load is noticed for hydrogen operation with 55% blending ratio while at full load the reduction is about 17.1% when the engine running at 1500 rpm and the inlet air temperature is 60 ^oC.

Figure(14) portrays the alteration of hydrogen blending on unburned hydrocarbon UBHC emission with and without heating. 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 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 diesel of 83%.

The heating process of air to 60 ^oC enhances combustion temperature and reduces charge dilution. These factors give better combustion and reduction in unburnt hydrocarbon emissions. The lowest UHC emission obtained was 28 ppm with 55% hydrogen blending ratio compared to 50 ppm for pure diesel at 1500 rpm and 50% load.

Figure(15) shows the effect of hydrogen blending ratio on NO_x emission with and without heating. It is seen that there is about 10%-13% increases 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 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. The highest NOx emission of 1328 ppm was obtained for 55% hydrogen blending ratio at 1500 rpm, 50% load as compared to 1001 ppm for pure diesel when the inlet air temperature is 60 ^oC.

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 for different hydrogen blending ratios with and without heating is given in Figure(16). It can be noticed that the lowest smoke opacity of 4.5% is obtained for 30% hydrogen enrichment at no load 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 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. For heating process with 60 ^oC for inlet air temperature, the maximum reduction in smoke opacity obtained was 62.6% at 50% load and 1500 rpm.

Figure(17) depicts the influence of hydrogen blending on O_2 emission with and without heating. It can be seen that the percentage of O_2 in the exhaust decreases with 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 will result in the reduction of overall O_2 content in the engine which leads to 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. The maximum reduction of oxygen content obtained was 34.5% at no load with 1500 rpm between 0 to 55% hydrogen blending ratio when the inlet air temperature is 60 ^oC.

CONCLUSIONS

Based on the above presented and discussed results the following conclusions can be drawn from this work:

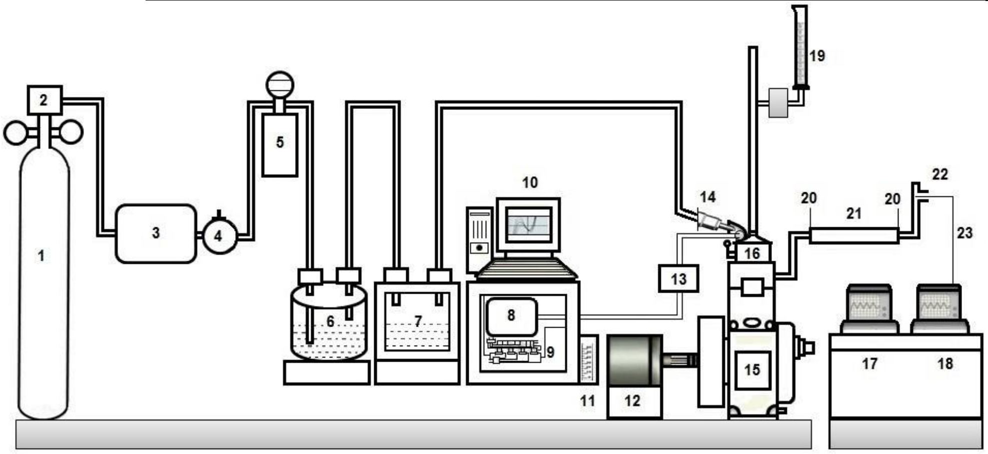
1-Hydrogen can be inducted in the air manifold of a diesel engine to improve combustion process.

2- The presence of hydrogen in dual-fuel mode of operation improves the engine thermal efficiency, and reduces engine emissions even in small induced quantities.

3- The results showed that with inlet air heated to 60 °C it is possible to increase the hydrogen blending ratio to 55% with satisfactory engine operation.
3- 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. More increased in brake thermal efficiency of 21.72% and less decreased of 46% at for diesel fuel consumption with heating process and 55% hydrogen blending ratio.
4- At 30% hydrogen blending ratio, a significant reduction in volumetric efficiency of about 18% at full load conditions while for heating process is 35.8%.
5- Hydrogen blending raises the maximum in cylinder pressure from 60 bar to 71 bar between 30% and 55% blending ratio. At the same range of hydrogen blending ratio the exhaust gas temperature increased from 387 °C to 403 °C.
6- 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.
7- In general, a reasonable reduction in pollutants emissions of CO, CO_2 and UBHC, with hydrogen blending ratio ranging from 10% to 30%. This reduction in pollutants increased with heating air to 60 °C.
8- Smoke opacity is significantly reduced in dual mode operation especially with heating process.

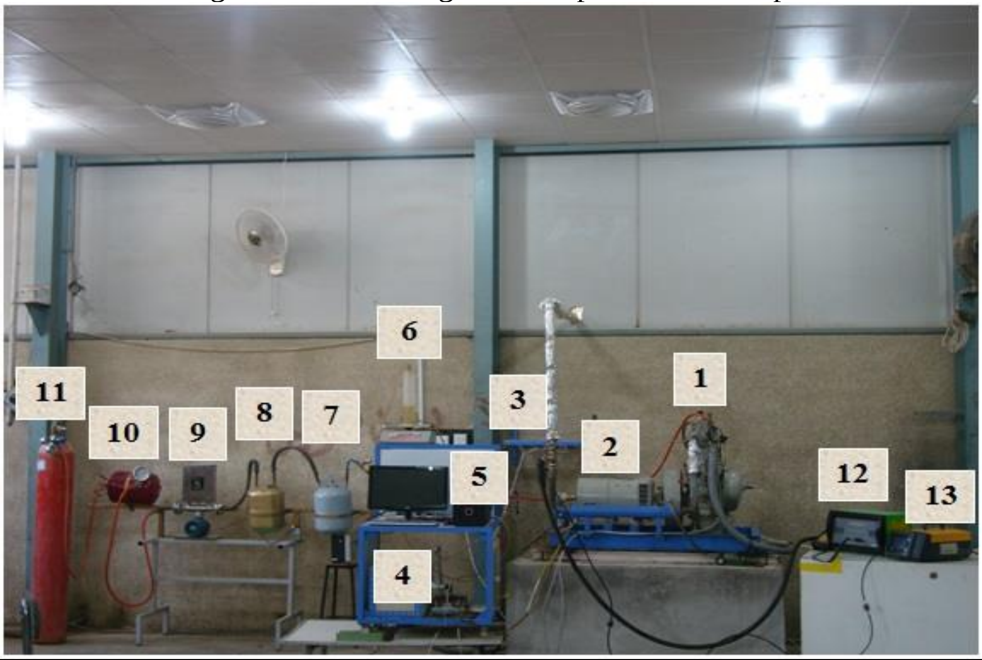
Table 1 engine specifications

Make and Model	Kirloskar AV-1
Type	Vertical, single cylinder, water cooled
Bore× Stroke	80 mm×110 mm
Swept volume	$0.553\text{x}10^{-3}$ 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 °CA
End of Injection	200 °CA
Injection Duration	35 °CA



1	Hydrogen cylinder	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		

Fig 1 Schematic diagram of experimental setup.



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		

Plate 1 The Experimental Set Up

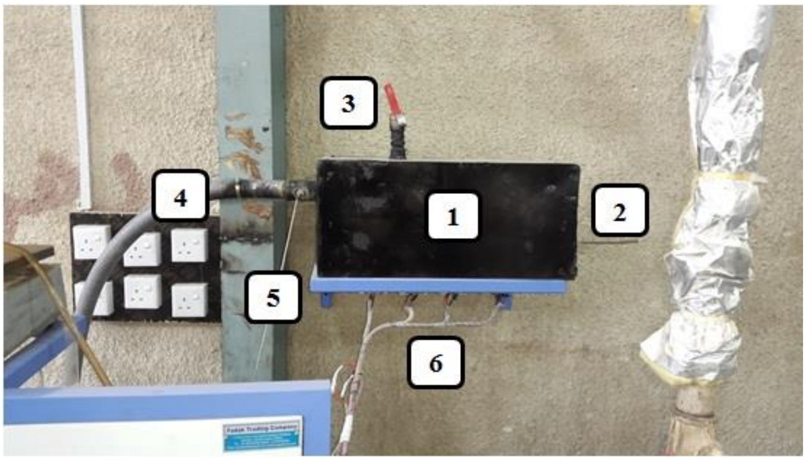


Plate **2** Air Heating System

1-Heating box, 2- Heaters control, 3-Temperature regulator, 4-Heating air pipe,
5-Temperature sensor, 6- Heaters group.

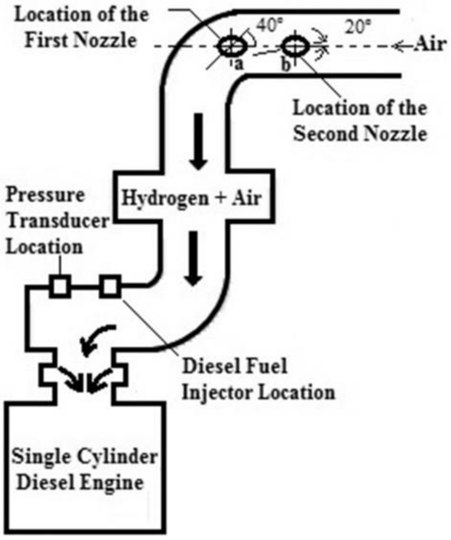


Fig 2 Locations of Continuous Hydrogen Induction Points in Intake Manifold

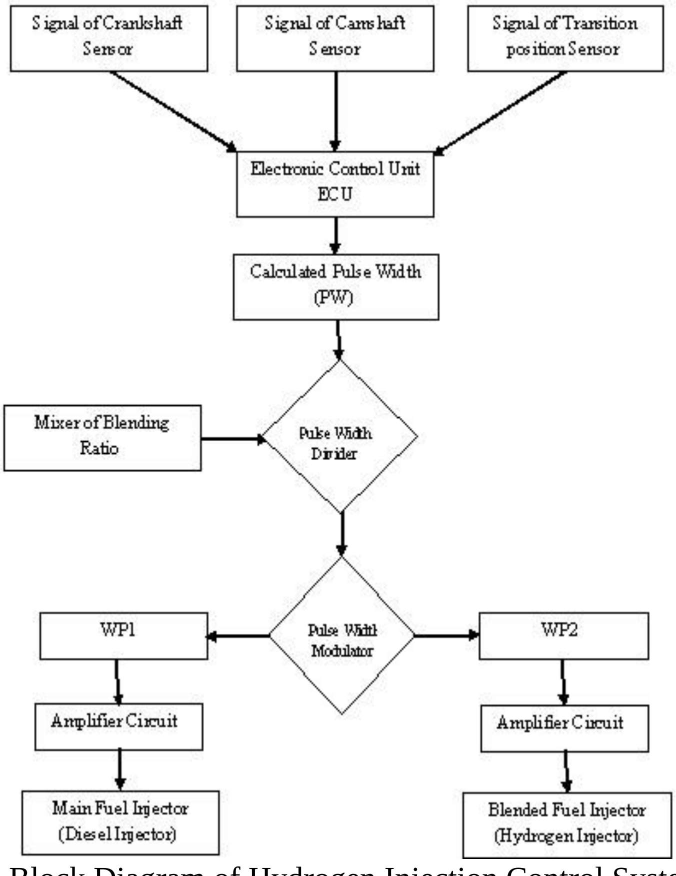


Fig. (3) Block Diagram of Hydrogen Injection Control System

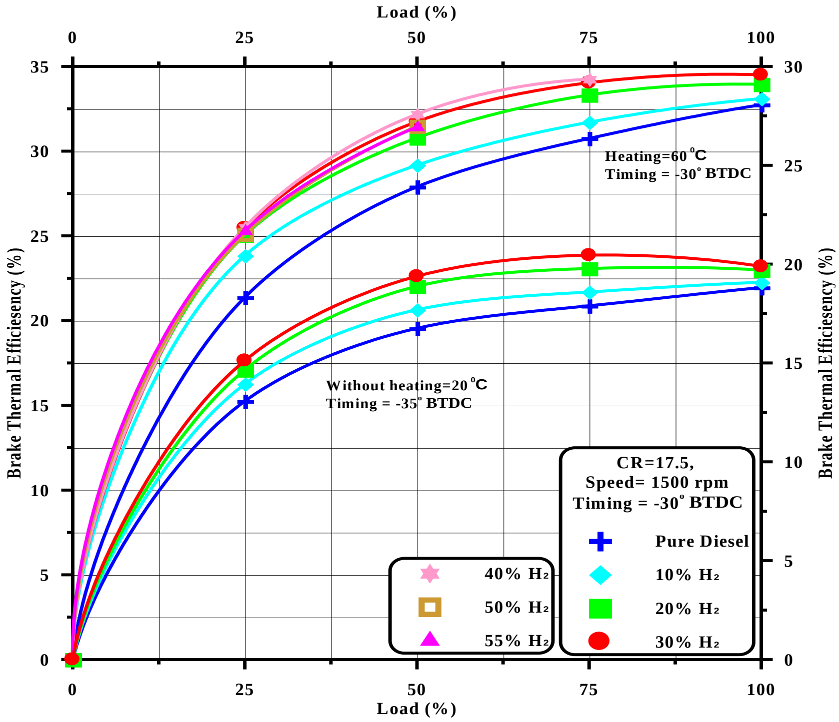


Fig 4 Brake Thermal Efficiency Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

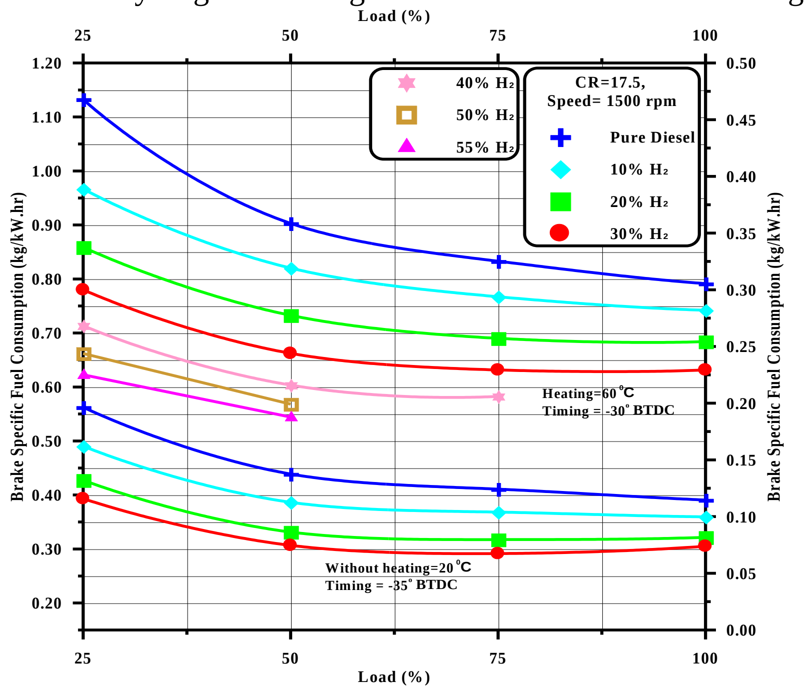


Fig 5 Brake Specific fuel Consumption Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

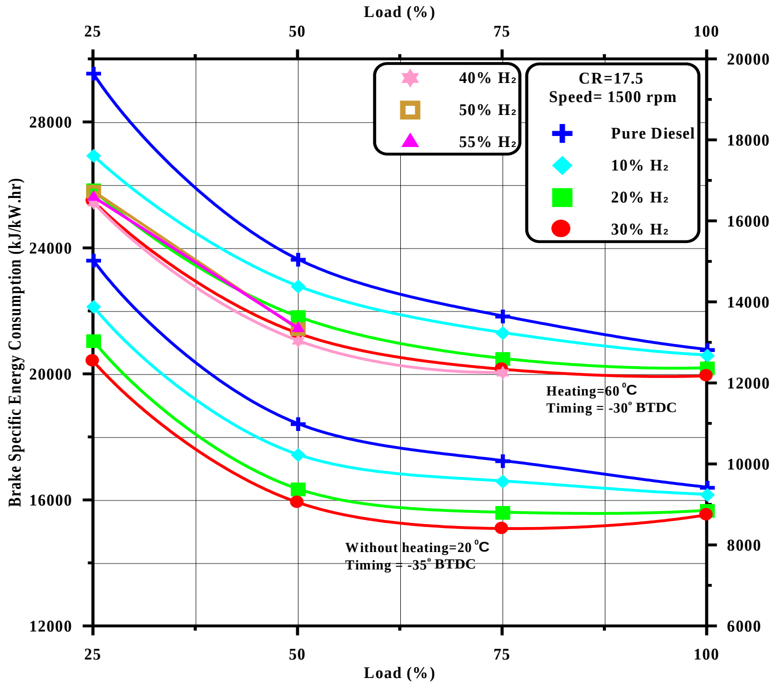


Fig 6 Brake Specific Energy Consumption Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

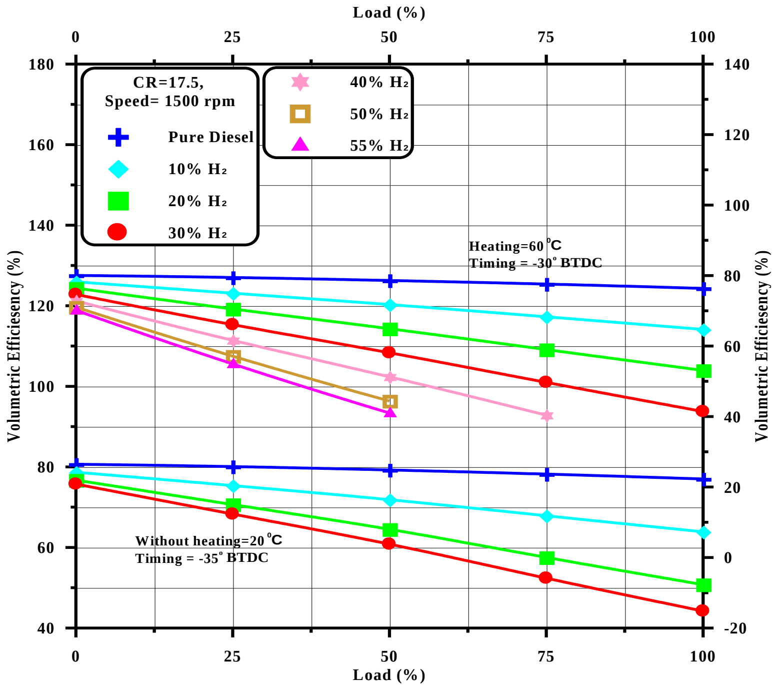


Fig 7 Volumetric Efficiency Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

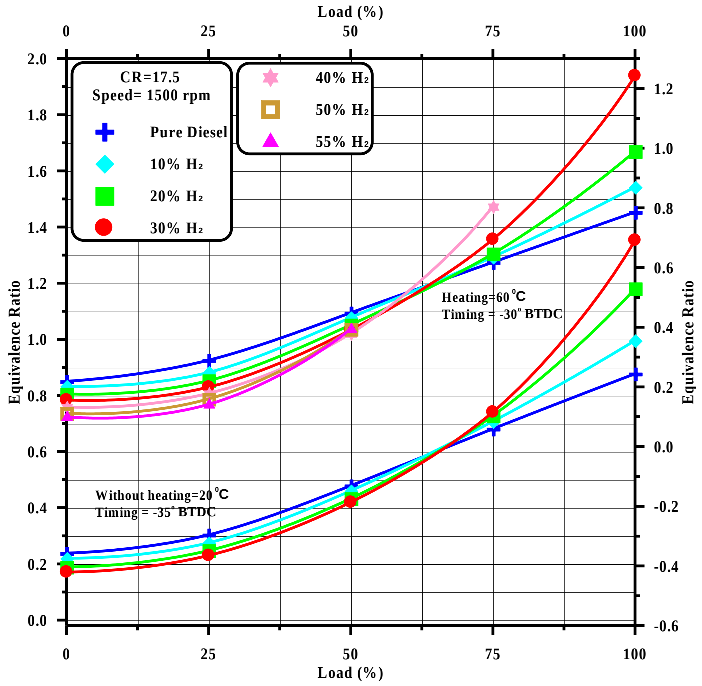


Fig. 8 Equivalence Ratio Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

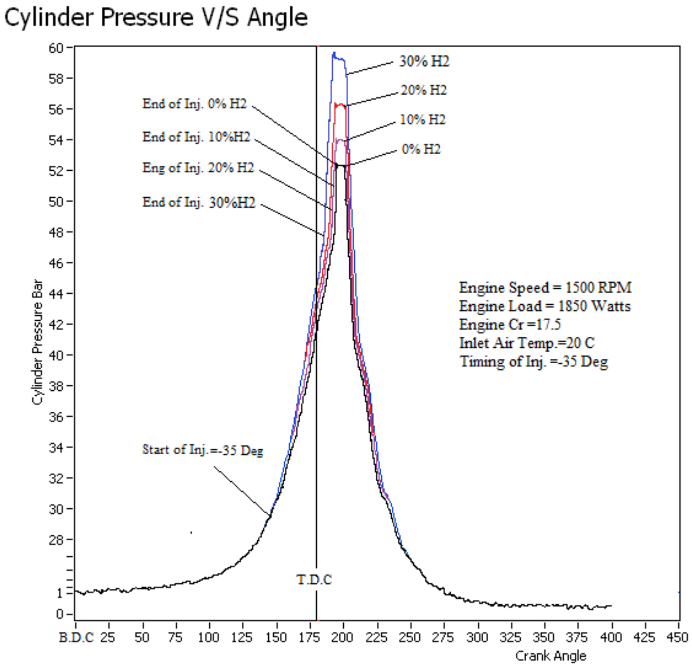


Fig **9** In-cylinder pressure History for Different Hydrogen Blending Ratios.

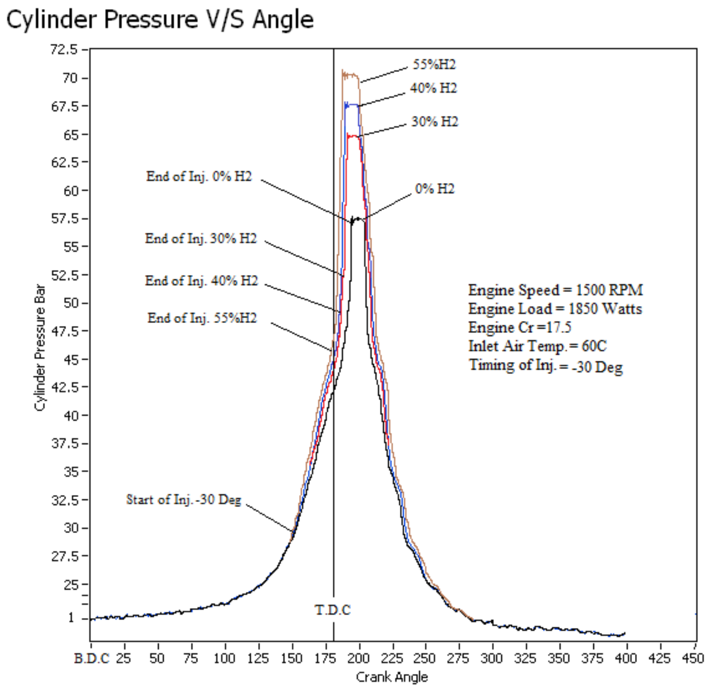


Fig **10** In-cylinder pressure History for Different Hydrogen Blending Ratios.

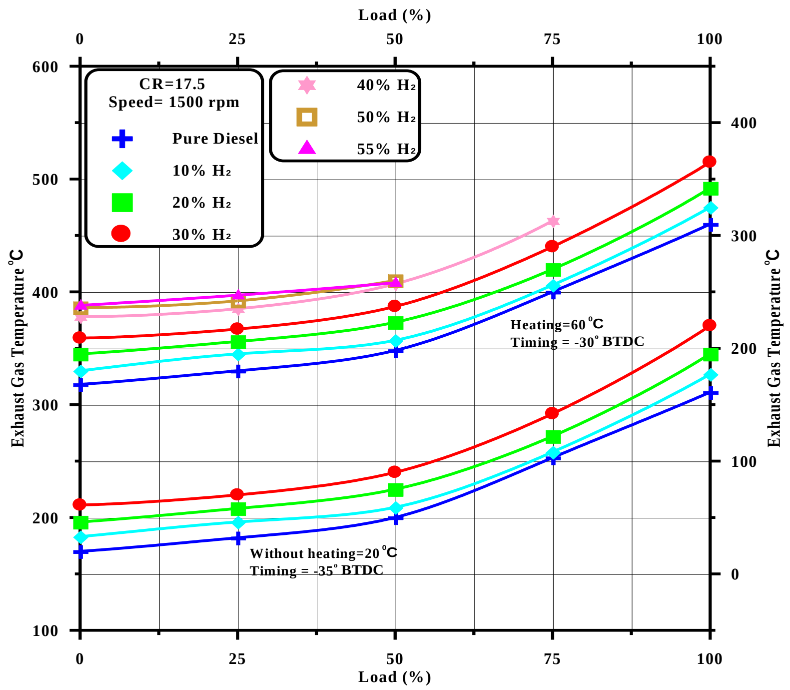


Fig 11 Exhaust Gas Temperature Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

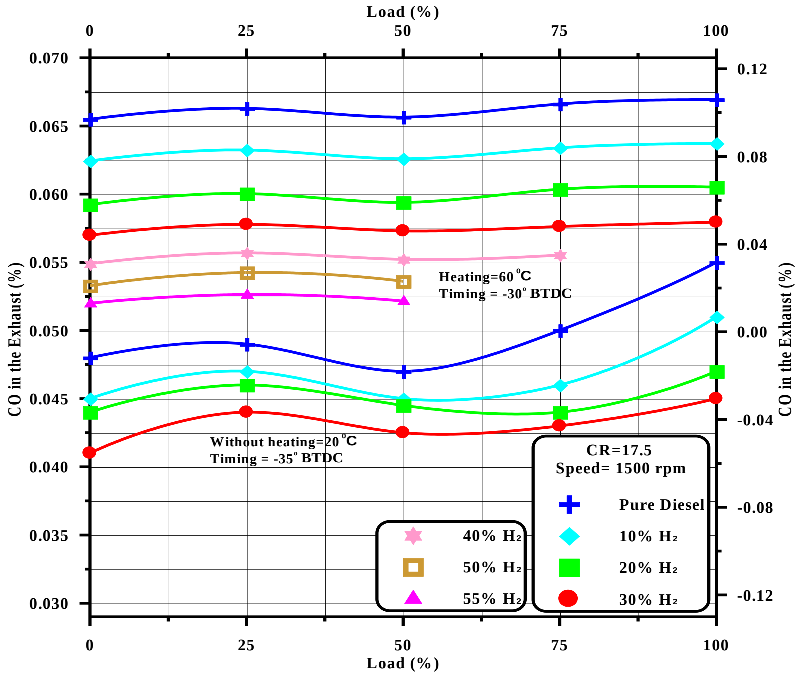


Fig 12 Carbon Monoxide Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

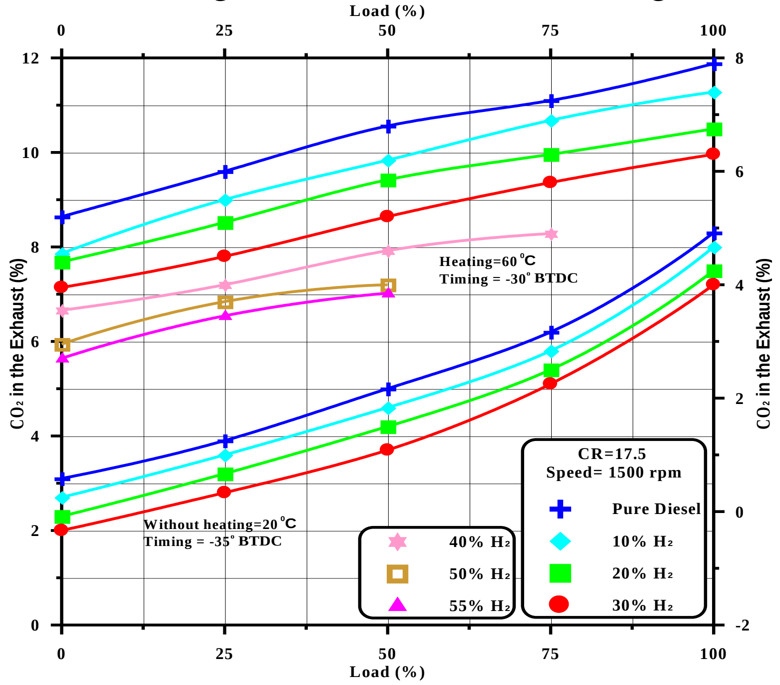


Fig 13 Carbon Dioxide Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

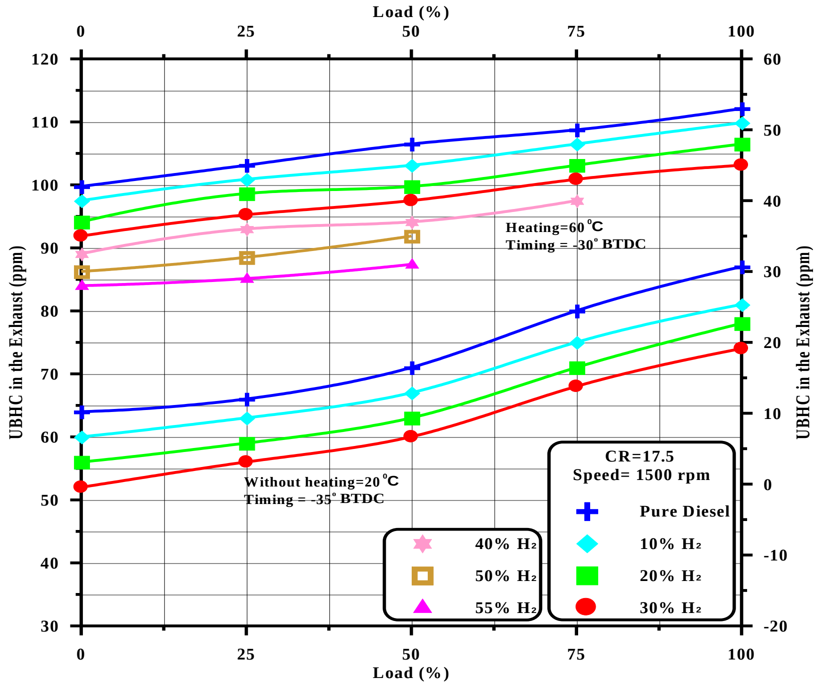


Fig 14 UBHC Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

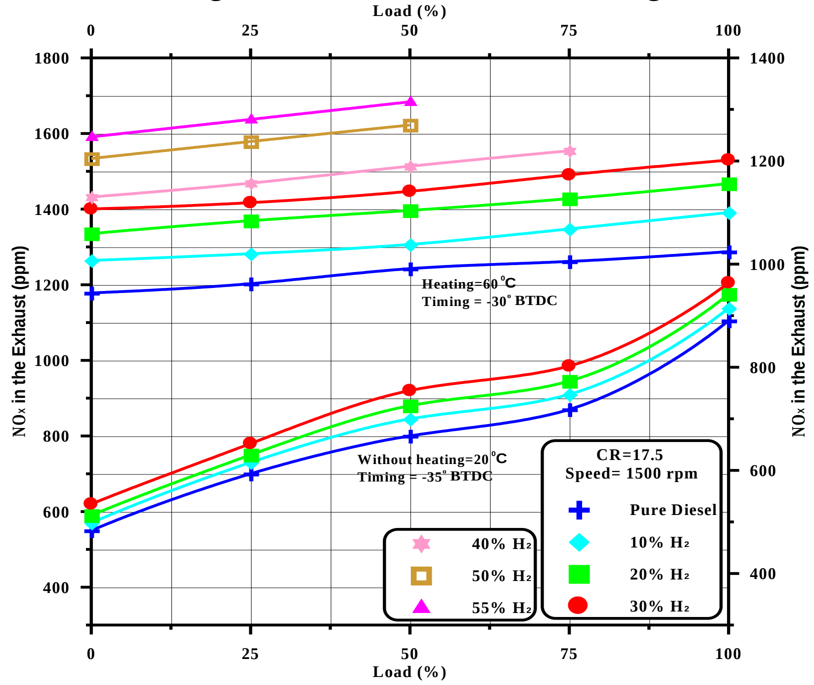


Fig 15 NO_x Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

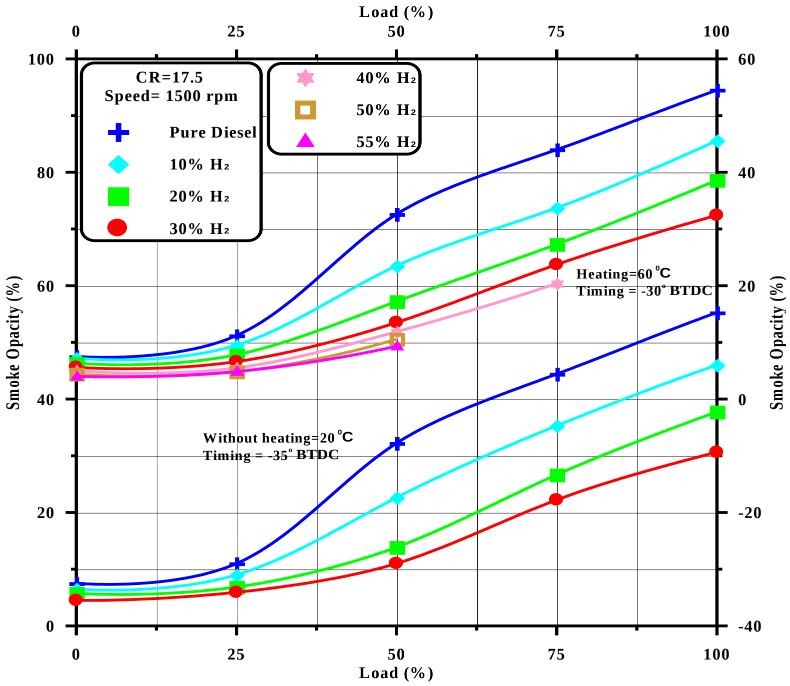


Fig 16 Smoke Opacity Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

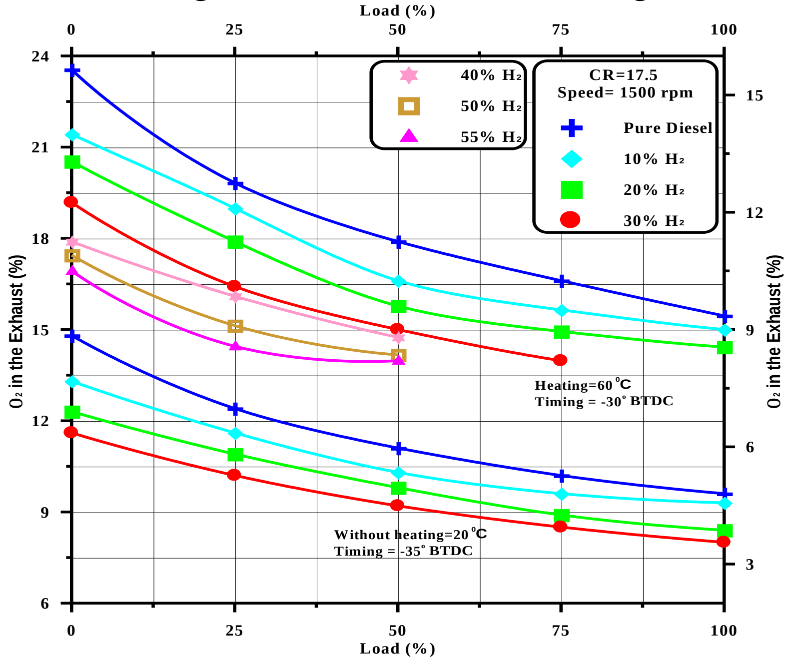


Fig 17 O_2 Variation with Load for Different Hydrogen Blending Ratios With and Without Heating

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