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A Practical Study of Using Hydrogen in Dual – Fuel Compression Ignition Engine

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ABSTRACT

An experimental study was conducted to investigate the possibility of improving diesel engine performance by aspirating volumetric quantities of gaseous hydrogen in the intake of a diesel engine. A single cylinder Ricardo E6 with four stroke compression ignition engine and variable compression ratio, injection timing and equivalence ratios was used with hydrogen supplementation to diesel fuel in dual fuel concept. The test conditions used were: engine higher useful compression ratio and speed of 1500 rpm. The effect of a wide range of equivalence ratios and injection timing were studied. The results showed that HUCR for diesel fuel used in this study was (17.7:1), and it was increased for mixtures of two fuels. The results showed that the engine could be run with very lean equivalence ratios with supplementary hydrogen. The indicated thermal efficiency increased, and the brake specific fuel consumption reduced with increasing hydrogen volumetric ratio.

Keywords: Hydrogen volumetric fraction, brake power, brakes specific fuel consumption, thermal efficiency, injection timing, engine speed.

1.INTRODUCTION

Compression ignition engines play a dominant role in transportation and agricultural sectors all over the world, because of their superior thermal efficiency and durability characteristics. The diesel fuel availability is rapidly decreasing while the consumption of diesel oil keeps on increasing. Hence, must develop alternative fuels such as hydrogen for diesel engines [1]-[2]. Hydrogen is considered by many researchers as a fuel for internal combustion engines since the early years of the last century. But only in recent years there has been a strong, revived interest in this area [3]. The investigations of using the hydrogen as a primary or supplementary fuel for CI engines have been limited due to its high auto-ignition temperature as shown in table 1. The ignition of hydrogen-air mixture by compression alone is considered difficult unless engines with very high compression ratios are used [4]. In some studies like references [5]-[6] who have used glow plug, spark plug or pilot injection of diesel fuel to carry out a credible combustion of hydrogen in CI engines. An easy method of using hydrogen in diesel engines is the hydrogen–diesel dual–fuel mode. This method features homogenous hydrogen–air mixture compressed to below its auto-ignition conditions. Then this mixture is ignited by the injection of a pilot diesel fuel spray near the top dead center [7]-[8]. Two main problems are associated with the use of hydrogen in diesel engines in the form of dual–fuel engines using relatively small pilot fuel injection. The first is the low brake thermal efficiency of the engine when it is operated under the part load conditions as a result of poor utilization of the gaseous fuel component of the charge. The second is the loss of combustion control through knock involving spontaneous ignition of the mixture within the cylinder with higher outputs. This phenomenon consequently imposes a serious limitation on the knock limited power output [9]. Shahad concluded that very low hydrogen flow rates had an adverse effect on the engine thermal efficiency while increasing the percentage of hydrogen supplied to the engine resulted in a clear improvement in the efficiency [10]. Milen showed that their experimental results demonstrated that hydrogen addition in the intake air has an influence on improvement of engine power and energy consumption. The hydrogen addition improves the power not only quantitatively but qualitatively by the mean of combustion improvement [11]. This study aims to investigate the effect of hydrogen addition to diesel fuel as dual-fuel in compression ignition engine on engine performance, and compare the results with diesel engine operated with diesel fuel alone.

2.EXPERIMENTAL SETUP

2.1Equipment

Ricardo E6 engine with naturally aspirated, single cylinder, four strokes, variable compression ratio, injection timing and speed was used in this study. This engine is mounted on a common bed-plate with an electric dynamo-meter and equipped with controls to regulate engine speed and load. Alteration of injection timing can be made while the engine is running. Table 1 gives further details regarding this engine. Intake air flows through a 200 liter air tank connected to the induction manifold to reduce pressure pulsations. Air flow rate is measured using an orifice fitted to the air tank. The procedure used to measure diesel fuel flow rate was by recording the time required the engine to consume a 50 ml of fuel. A Piezo electric pickup sensor type (AVL 8QP 500C) was fitted into the combustion chamber. This sensor was connected to a charge amplifier, oscilloscope and Iwatsu Signal Analyzer to enable the instantaneous measurement of cylinder pressure. Fig. 1 shows schematically the layout of the test setup.

Table 1 Engine specifications

Make	Ricardo E6/US
Type	Single cylinder, vertical Water cooling, indirect injection, diesel engine
Bore (mm)	76.2
Stroke (mm)	110
Displacement volume (cm^3)	507
Compression ratios	4.5-22

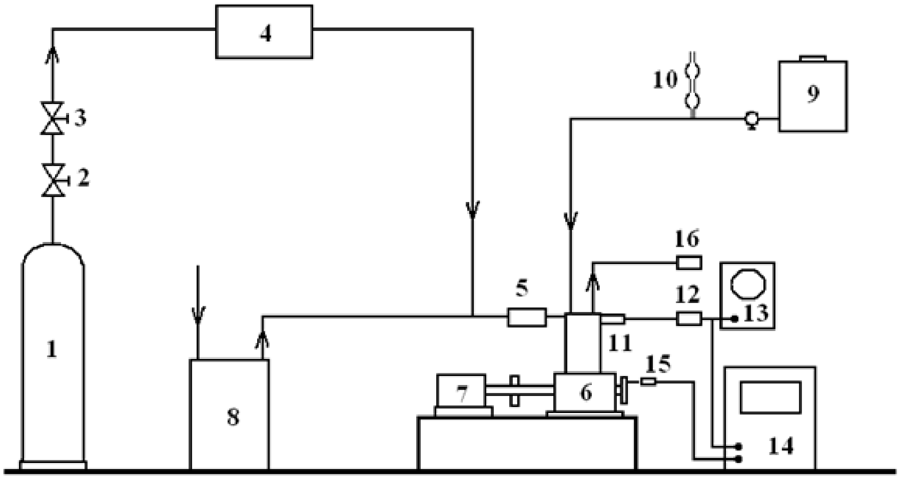

Figure 1 Schematic layout of test setup

1. Pressurized hydrogen gas cylinder, 2. Cylinder gas pressure regulator, 3. Control valve, 4. Chocked nozzle meter, 5. Carburetor, 6. Engine, 7. Electrical dynamometer, 8. Air flow meter, 9. Diesel fuel tank, 10. Diesel fuel measuring burette, 11. Pressure pickup, 12. Charge amplifier, 13. Oscilloscope, 14. Signal analyzer, 15. Crank angle induction pickup, 16. Exhaust gas meter.

2.2Types of the used fuels

Iraqi diesel fuel (cetane No. =49.5) produced by Al-Doura Refinery was supplied to the injection system. Hydrogen (99.99% pure produced by Al-Mansour Co.-Baghdad) was drawn from a high-pressure cylinder; this pressure was reduced to 1 atm by means of a pressure regulator. Hydrogen was made to pass through a control valve to regulate the gas flow. The hydrogen mass flow rate was metered using chocked nozzles meter, which was used as a flame trap also to arrest and control flash back if any. Table 2 introduces a comparison of the used fuels properties.

Table 2 Comparison of fuels properties [12]

Property	Hydrogen	Diesel
Chemical composition	H_2	$C_{12}H_{26}$
Density at ambient (kg/m^3)	0.081	824
Specific gravity	0.091	0.83
Flammability in air (% by vol.)	4-75	-
Stoichiometric A/F (mass)	34.3	14.9
Heat of combustion (lower, MJ/kg)	119.98	42.36
Auto-ignition temperature (K)	845.1	524.5
Maximum flame speed (m/s)	2.92	0.38
Cetane No.	-	40-60

2.3Data Reduction

The following equations were used in calculating engine performance parameters [13]:

1- Brake power

$$bp = \frac{2\pi * N * T}{60 * 1000} \quad kW \qquad (1)$$

2- Brake mean effective pressure

$$bmep = bp \times \frac{2 * 60}{V_{sn} * N} \quad kN/m^2 \qquad (2)$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad kg/_{sec} \qquad (3)$$

4- Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec} \qquad (4)$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60 * 2} \times \rho_{air} \quad \frac{kg}{sec} \qquad (5)$$

5- Brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr} \qquad (6)$$

6- Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW \qquad (7)$$

7- Brake thermal efficiency

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad \% \qquad (8)$$

8- Equivalence ratio [14]:

$$\emptyset = \frac{\dfrac{[D]}{[air] - \dfrac{[H]}{([H]/[air])_{st}}}}{\left(\dfrac{[D]}{[air]}\right)_{st}} \qquad (9)$$

)

2.4Tests procedure

Two sets of tests were conducted during this investigation. In the first set, engine was run at speed (1500 rpm) with neat diesel fuel. By using optimum injection timing (OIT) and full load, the higher useful compression ratio (HUCR) for this fuel was defined for each tested point. Engine performance was studied at this HUCR in detailed, to find the effect of some engine variables (load, speed, injection timing and equivalence ratio) on engine performance. The same experiments were run again in the second set of experiments, adding variable gaseous hydrogen volumetric fractions to air at inlet manifold. The same aforementioned variables affected engine performance were studied and compared with the first set of results. Knock limited power output was determined at each hydrogen volumetric fraction (HVF) to diesel fuel. Knock in hydrogen-diesel dual-fuel engine is normally associated with quite a sharp change in the operating conditions of the engine. As soon as it occurs, the noise pattern of the engine changes considerably, becoming louder and having a sharper note. Simultaneously the shape of the pressure diagram changes to display a very sharp pressure rise with much-increasing maximum pressure accompanied by oscillations on the expansion curves. The transition from normal to knocking operation is usually quite sharp and can be achieved with small changes in the hydrogen fuel concentration or increasing load. A change in the shape of the pressure diagram was used to detect the onset of knock in the present work. Fig. 2 represents a typical pressure work and diagrams just before and after the onset of knock.

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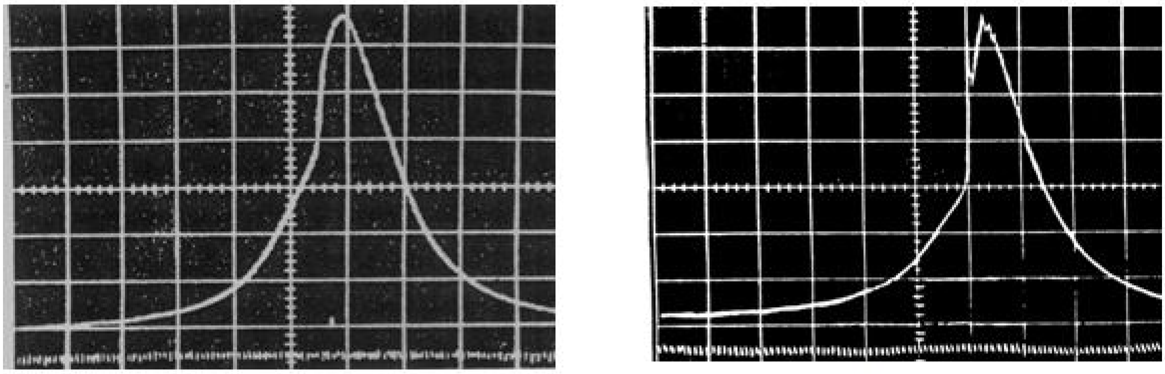

Without knocking With knocking

Figure 2 Typical pressure crank angle diagrams just before and after the onset of knock.

2.5Experimental errors and uncertainties

As long as the study is experimental and depends on several measuring devices, then it must be a degree of uncertainty. This uncertainty comes from the difference between measured and true values of the quantity. All the used measuring devices were calibrated in the laboratory, and the value of that error was assigned, **Table 3** lists the errors in the measuring devices used in the tests. The uncertainty was found less than 3% that give high accuracy for the recent results.

Table 3, Experimental Accuracy

Measurements	Accuracy in this study
Temperatures measurements	±0.045°C
Air flow measurement	± 0.07
Fuel flow measurement	± 0.95
Engine speed measurement	± 2 rpm
Engine torque measurement	± 1.24
Hydrogen flow measurement	± 0.7
Hydrogen pressure regulator measurement	± 0.022

3.RESULTS AND DISCUSSION

3.1CR effect

As has been mentioned above, the first set of experiments was conducted to explore the highest useful compression ratio for diesel fuel tested. The HUCR for diesel fuel tested was 17.7:1, as fig. 3 shows. The bp increases with CR increases from 16.5 to 17.7:1, after this at CR=18:1, bp increased in very lean mixture below Ø=0.5 and decreased in the region from Ø=0.5 to 0.71. Engine operation after Ø=0.71 was not studied because of the rough operation and the high soot exhausted from tailpipe, combined with high loss in brake power. Hydrogen addition increases CR. The HUCR for the used diesel fuel was 17.7:1. Adding hydrogen in 10% volumetric fraction, raised the HUCR for the duel fuel to18:1. This procedure continued until HVF=40%, where the HUCR for the dual fuel was 19:1, as fig. 4 shows. At HVF=50% bp increases at very lean equivalence ratios below Ø=0.4, and decreases after Ø=0.4 because of knock occurrence that influence to retard injection timing that causes a reduction in the resulted bp.

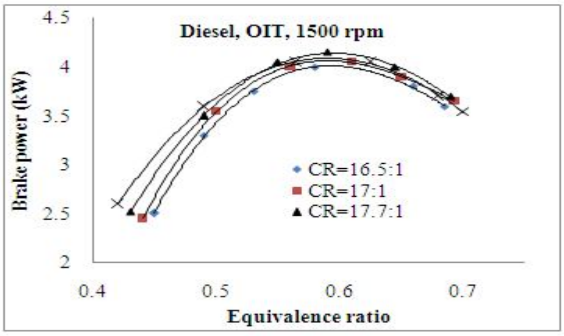

Figure 3 CR effect on bp for a wide range of equivalence ratios

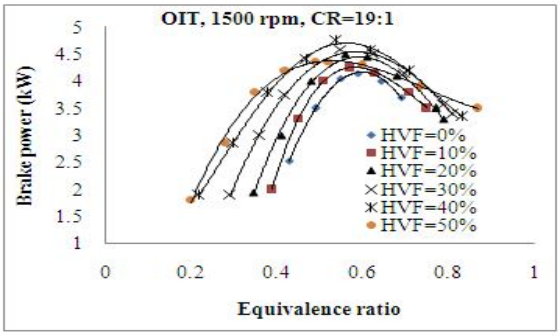

Figure 4 Hydrogen additions in volumetric fractions effect on HUCR and bp for a wide range of equivalence ratios

3.2Equivalence ratio effect

It makes an impression that the net heat release has a higher value with hydrogen addition compared to that of diesel fuel (fig. 3). The diesel fuel needs to vaporize before it ignites, and this heat of vaporization is taken from the air in the combustion chamber, and hence decreasing the net heat released. After the start of injection, hydrogen ignition takes place immediately. The released heat from hydrogen burning compensates the heat necessary for diesel vaporization. Adding hydrogen to diesel enlarged its equivalence ratios range, as Fig. 4 declares. Hydrogen possesses wide equivalence ratio range, and its effect on burning rate appears clearly. Engine operation in ultra low equivalence ratios reduces its emissions highly. Increasing equivalence ratio causes retardation for injection timing with hydrogen addition as fig. 5 shows. The reduction in injection timing is due to the high flame speed of homogeneous hydrogen-air mixture through the combustion chamber. The flame propagation rate increases due to amelioration in the propagation process between the hot air and diesel fuel vapors causing its faster completing. This diffusion process improvement causes higher heat utilization and higher net heat release respectively.

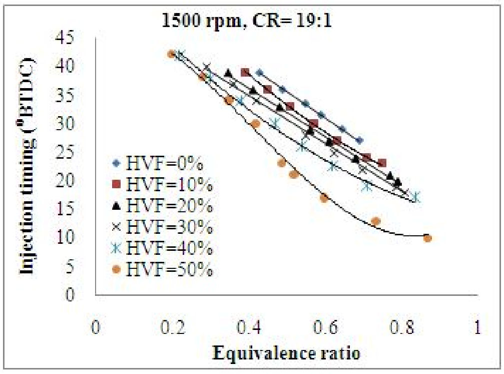

Figure 5 Hydrogen addition in volumetric fractions effect on OIT for a wide range of equivalence ratios and HUCR for each HVF

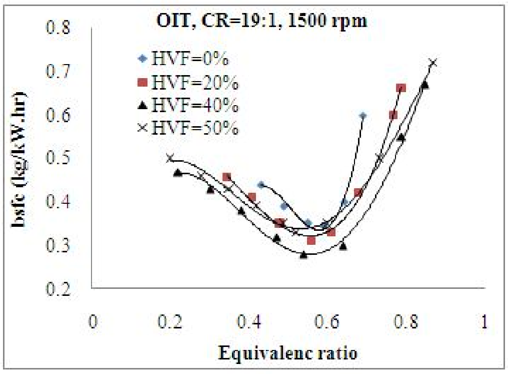

Figure 6 Hydrogen addition in volumetric fractions effect on bsfc for a wide range of equivalence ratios 1500 rpm and OIT

For HVF=50% the curve of OIT take another shape after Ø=0.4, the OIT retarded sharply with equivalence ratio increase. This retardation is due to three parameters: first high temperature inside combustion chamber, almost near self-ignition temperature for hydrogen. Secondly, the high burning velocity of hydrogen and third, the high heating value of diesel fuel. Brake specific fuel consumption (bsfc) decreases with HVF increases, as fig. 6 shows, this is due to bp increased, except for HVF=50%, where its bsfc increase due to bp decreased. Fig. 7 illustrates that there is a smooth and strong rise in the thermal efficiency with increasing HVFs. The rate of increase of thermal efficiency becomes less at higher equivalence ratios. This behavior is expected where the thermal efficiency increment at low equivalence ratios was due to combustion improvement by hydrogen addition. At higher diesel flow rates, the thermal efficiency practically remained unaltered when hydrogen was introduced. Fig. 8 demonstrates that engine volumetric efficiency reduced with hydrogen addition. Hydrogen entered the combustion chamber in a gaseous phase, so it took part from air portion and causes this reduction.

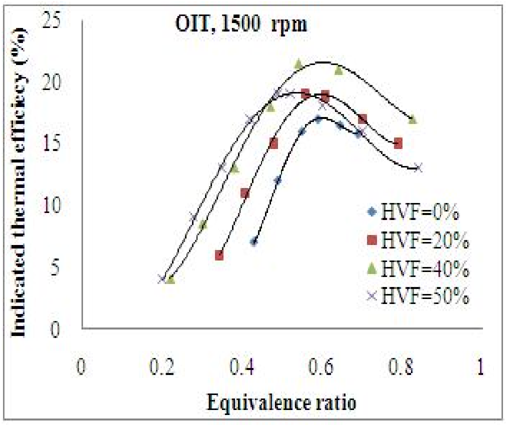

Figure 7 Hydrogen addition in volumetric fractions effects on indicated thermal efficiency, for a wide range of equivalence ratios, 1500rpm and OIT

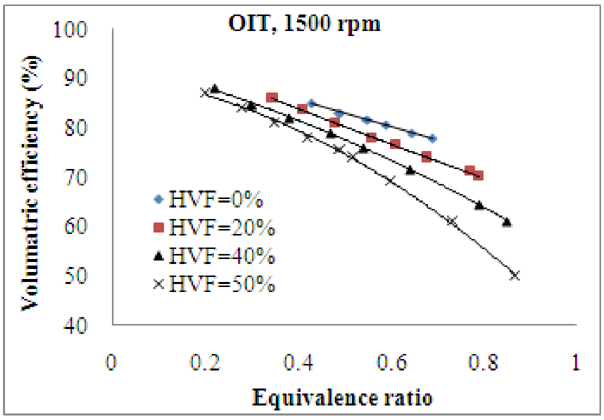

Figure 8 Hydrogen addition in volumetric fractions effect on volumetric efficiency for a wide range of equivalence ratios, 1500 rpm and OIT

Combustion improved with hydrogen addition; this improvement appears in increasing exhaust gas temperature in very lean equivalence ratio less than Ø=0.4. Above this equivalence ratio the exhaust gas temperatures decreased; this is what fig. 9 shows. Varde [15] concluded that increasing the portion of hydrogen increased the exhaust gas temperature due to hydrogen rapid flame propagation and its high temperature at a given equivalence ratio. This conclusion is true, because the researcher used constant injection timing (22° BTDC) in all their work. This timing is very retarded for many equivalence ratios as it appeared from this paper. So, the mixture combustion continued in the expansion stroke that made the exhaust gas exhausted with high temperatures.

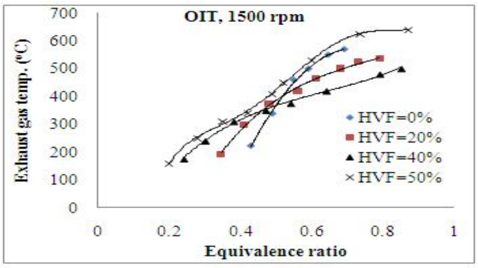

Figure 9 Equivalence ratio effect on exhaust gas temperatures for variable hydrogen volumetric fractions at 1500 rpm and OIT

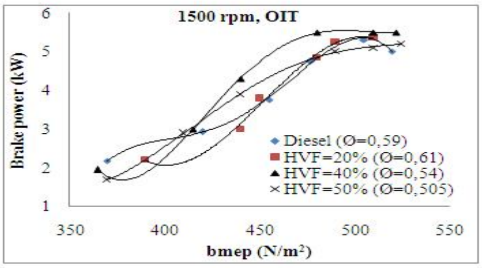

Figure 10 Load effect on brake power, for specific equivalence ratios, at 1500 rpm, OIT and HUCR

3.3Load effect

In all the previous experiments used the maximum load can be achieved without knock occurrence for each equivalence ratio. That's mean if the load was increased with a little pit knock will occur. **Figs. 10 to 13** represent the load effect on brake power; injection timing and bsfc at equivalence ratios offered the maximum brake power for the studied HVFs'. The engine was run at 1500 rpm and OIT for each point. Increasing engine load increases heat inside the combustion chamber that will reduce the ignition delay period, and the fuel-air mixture will burn faster and more efficient. As the mentioned figures demonstrate, brake power reduced with decreasing load, OIT advanced and bsfc increased. When load reduced the heat inside combustion chamber reduced so ignition delay period increases, and hence, OIT must be advanced to compensate the temperature reduction in the chamber. Engine operation with part load increased bsfc due to bp reduction. The maximum bp achieved was at OIT, as well as the minimum bsfc.

3.4 Engine speed effect

Engine speed effects were studied for specific points to clarify hydrogen addition effect. The points that offer the maximum brake power at each speed and HVF were chosen in the following figures. Fig. 13 demonstrates the relation between HVF and the maximum brake power achieved at variable engine speeds. The best operation speeds for the tested engine pertained between 1500 and 1800 rpm. Most of the energy produced from burning is utilized (increasing bp and minimizing bsfc values (Fig. 14)). But out of this range (above 1800 rpm or below 1500 rpm) there is a reduction in bp and graduate increase in bsfc in each. The cause of this reduction is the decrements in volumetric efficiency due to decreasing in air consumption rate, or reducing entering air to fuel ratio. The intake valve in this engine is designed to shut down lately after bottom dead center (38° ABDC). This procedure is suitable for high speeds, but not suitable for low speeds, because when the piston is rising the intake valve will be opened in compression stroke. So, a part of the entering charge will be retained back to the intake manifold. This charge will reduce entering air mass. The remaining hot exhaust gases in the combustion chamber will heat entering air, reducing its density. Hydrogen addition seems to have the same effect on increasing bp in low and medium speeds but in high speeds its effect is limited. The influences of injection timing and volumetric efficiency reductions caused these effects. Also, increasing engine speed increases friction power, causing engine bp reduction in high speeds

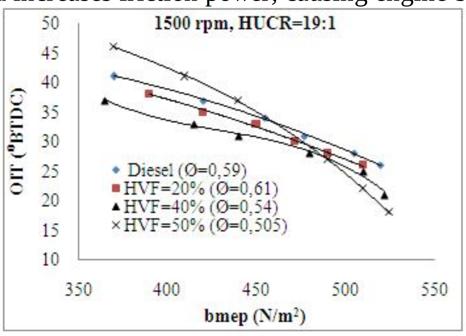

Figure 11 Load effect on injection timing, for specific equivalence ratios, at 1500 rpm and

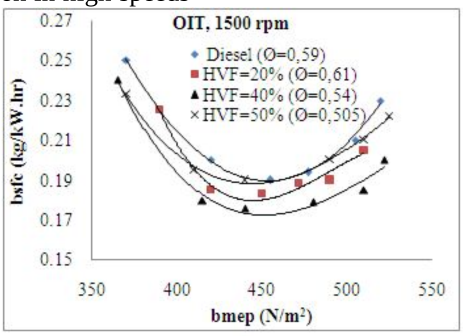

Figure 12 Load effect on bsfc, for specific equivalence ratios, at 1500 rpm, OIT and HUCR

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Fig. 15 shows the relationship between HVF and OIT for the operating points where the maximum bp achieved. Optimum Injection timing advanced with engine speed increase. Hydrogen addition causes the opposite action it forces the injection timing to be retarded. The final resultant is the OIT at the specified operating point. The results show that the retarded degrees (from HVF=0 to HVF=40%) reached about 5.5° BTDC, which means hydrogen addition effect was the dominant. Exhaust gas temperatures increase with the speed increase but hydrogen addition decrease these temperatures, the final resultant as fig. 16 shows is a reduction in these temperatures with increasing HVF. Hydrogen heating value on the volume basis is lower than that for diesel. The substitution of a part of diesel fuel by hydrogen caused this reduction in exhaust gas temperatures.

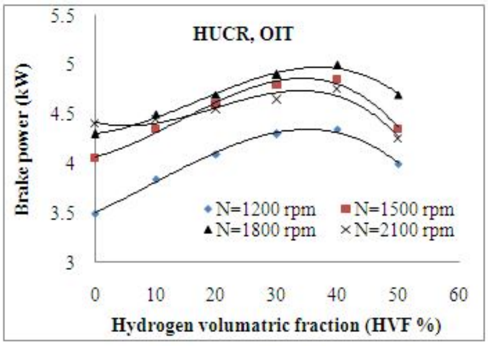

Figure 13 Hydrogen addition in volumetric fractions effects on bp, for variable engine speeds

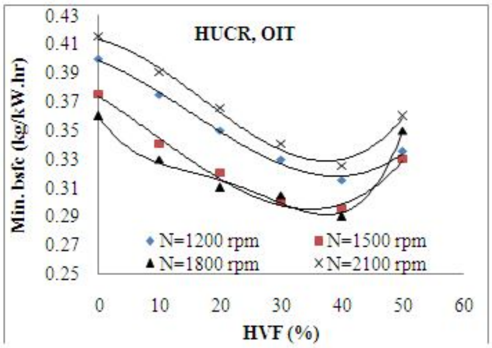

Figure 14 Hydrogen addition in volumetric fractions effect on minimum bsfc for variable engine speeds

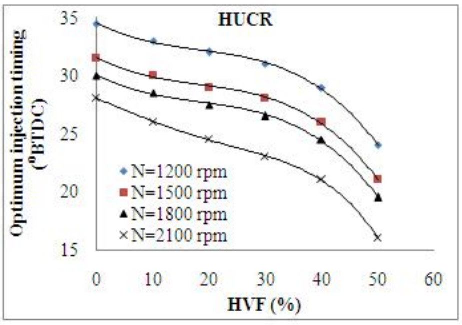

Figure 15 Hydrogen addition in volumetric fractions effect on OIT for variable engine speeds

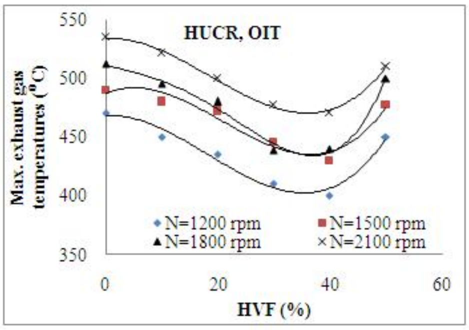

Figure 16 Hydrogen addition in volumetric fractions effect on max exhaust temperatures for variable engine speeds

3.5 Injection timing effect

Injection timing effects are represented in figs. 17 to 19, for specific equivalence ratios, took as examples for this effect. The equivalence ratio (Ø=0.25) was chosen as very lean equivalence ratio. The equivalence ratio (Ø=0.55) was chosen as the equivalence ratio which near it the maximum bp can be achieved. While the equivalence ratio (Ø=0.7) was chosen because it represents the end points equivalence ratio for diesel engine. Engine operated with hydrogen addition restricted with knock occurrences. Advancing injection timing above OIT is very limited especially in equivalence ratios above Ø=0.5, but in very lean equivalence ratio below Ø=0.3 injection timing can be advanced above OIT many degrees. But, in all conditions the bp reduced with this advance and bsfc increase also (Fig. 18). The advance of injection timing increases ignition delay period (a little pressure and temperature inside the combustion chamber at the injection moment). With hydrogen addition which have high burning velocity, when fuel starts burning it will produce high-pressure rates inside combustion chamber. This pressure will force the piston in the opposite direction for its movement (piston movement in compression stroke) producing negative work on the piston. This work will be applied against the power produced through power stroke and will minimize the resulted power. This reduction in bp returns back on bsfc, which will be increased. Also, it will reduce exhaust gas temperatures, because of increasing heat transfer rate to cooling water (Fig.19). Retarding injection timing resulted in a reduction in bp and increment in bsfc and exhaust gas temperatures. Exhaust gas temperature increments on the account of bp is due to the part of the fuel that will be burnt after the piston has reached the top dead center. So, the resulted energy from this burning will go to raise exhaust gas temperature instead of bp. Several former experimental studies were consulted to make sure of recent study results. Szwaja used a two-cylinder in-line CI F2L511 engine manufactured by Deutz, which was adapted to work as a dual-fuel engine. Diesel fuel was directly injected to the cylinder while port delivered hydrogen fueled injection (PFI) system. This system contained a hydrogen injector controlled by an electronic control unit (ECU), which adjusted injection timing with accuracy of the engine crankshaft angle of 0.5° [16]. Lilik studied a turbocharged DDC/VM engine of 2.5 L, four-cylinders, common rail and direct injection light-duty diesel engine. Hydrogen was dispersed and mixed with the boosted air using a custom built mixing manifold. Testing was conducted with the EGR valve and turbo charger enabled, to examine the effect of hydrogen on a diesel engine with modern features [17]. Debnath carried out the experiments on a Kirloskar TV1 CI diesel engine [18]. Hydrogen was supplied to the engine by means of a gas carburetor which is fixed in the intake manifold. Bose used an Apex made water cooled, single cylinder, four-stroke, direct injection, and vertical diesel engine running at compression ratio 17.5:1 and at a rated speed of 1500 rpm. Hydrogen was supplied to the engine by the induction method [12].

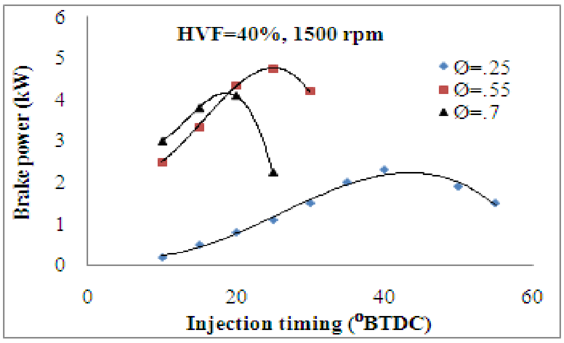

Figure 17 Injection timing effect on bp for specific equivalence ratios, HVF= 40%, HUCR=19:1 and 1500rpm

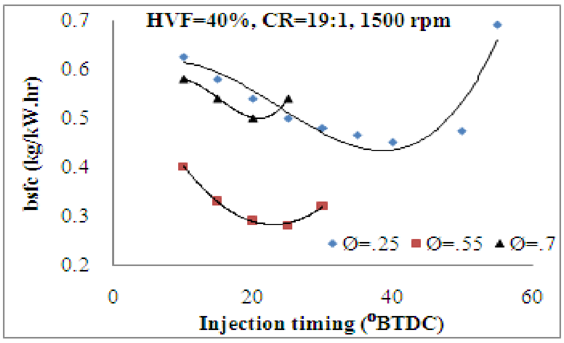

Figure 18 Injection timing effect on bsfc for specific equivalence ratios, HVF= 40%, HUCR=19:1 and 1500 rpm

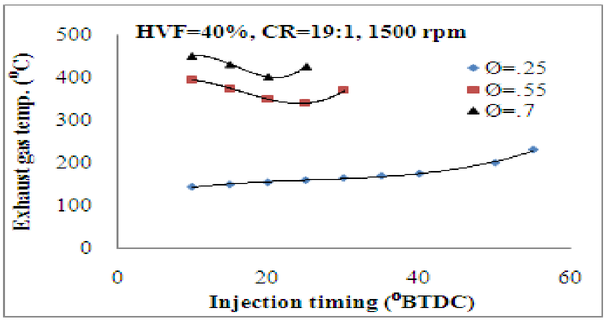

Figure 19 Injection timing effect on exhaust gas temperatures for specific equivalence ratios, HVF= 40%, HUCR=19:1 and 1500 rpm

In spite of engine size, operating compression ratio and added hydrogen fraction rate differences, all studies agreed that hydrogen addition increases brake thermal efficiency and reduces volumetric efficiency, bsfc and exhaust gas temperatures. Adding hydrogen influences the engine injection timing to be retarded.

4.CONCLUSIONS

1-This investigation establishes that hydrogen can be utilized in compression ignition engines operating them on dual-fuel principle.
2-Such operation improves thermal efficiency, makes the engine run with very lean equivalence ratios, which means very low engine emissions.
3-This operation results in lower exhaust gas temperatures leading to longer live of components such as exhaust valves.
4-Increasing hydrogen volumetric fractions accompanied with injection timing retarding, due to increasing in the flame velocity till the air is completely utilized. Further increase in HVF causes knock to appear, and OIT must be retarded sharply, which reduce the bp resulted from engine.
5-Knocking in the engine limits the proportion of energy input that can be supplied through hydrogen. The occurrence of knock in this type of hydrogen addition operation may be due to hydrogen rapid rate of pressure rise or flaming particle that is left behind due to bad engine scavenging.
6-When engine was run with hydrogen addition, bp was improved. The effect of hydrogen addition on the power enhancement was not quantitatively but qualitatively by the means of combustion improvement.

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NOMENCLATURE

BDC	Bottom dead center
Bsfc	Brake specific fuel consumption
BTE	Brake thermal efficiency
CA	Crank angle
CIE	Compression ignition engine
CR	Compression ratio
HUCR	Higher useful compression ratio
HVF	Hydrogen volume fraction
OIT	Optimum injection timing
TDC	Top dead center
η_{bth}	Brake thermal efficiency
Ø	Equivalence ratio

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