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Studies on Port Injected Hydrogen in a Dual Fuel D.I. Diesel Engine

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Abstract – *Hydrogen is expected to be one of the most important fuels in the near future for solving not only the problem of greenhouse emissions but also carbon monoxide and unburnt hydrocarbon thereby protecting the environment in addition to, saving conventional fuels. In the present experimental work hydrogen was used in the dual fuel mode with diesel as an ignition source. Hydrogen was injected in the intake port and diesel was injected directly into the combustion chamber. The injection timing and injection duration for hydrogen injection was controlled by an Electronic Control Unit (ECU). An hydrogen injector fitted on the intake port was used to inject the hydrogen fuel during the intake stroke. For experiments a single cylinder, AV1 Kirloskar, DI Diesel engine was used. The hydrogen injection timing was fixed at suction TDC and injection duration was fixed at 30°, 60°, and 90° crank angles. The injection timing of diesel was kept constant at 23° BTDC. Results show that, the* NO_X *emission decreases from 1806 ppm for diesel to 1690 ppm at full load for 30º injection duration for hydrogen and 1606 ppm for 90º injection duration in the dual fuel mode. The smoke reduces significantly from 4.06 BSN for diesel operation at full loads compared to 2.1 BSN for hydrogen operation with 90º injection duration with a further reduction in smoke to 1.2 BSN with 30º and 60º injection duration.*

Keywords – Emission, hydrogen, injection duration, injection timing, performance.

1. INTRODUCTION

In recent days, the importance of environment and energy are emphasized and among various energy sources, the fuels for automotive use are drawing attention as they are closely related with human day-to-day life. The fossil fuels, which are widely used, have some serious problems. One of these is the limit in reserves, the second problem is they cannot be recycled and the third one is they pollute the environment [1]. Therefore, research works have been carried out on alternative fuels to find a suitable substitute for fossil fuels. Among them, hydrogen has the outstanding advantages of wide flammability range and the absence of unburned hydrocarbon, carbon monoxide and carbon dioxide in the exhaust. In order to use gaseous hydrogen as a fuel for internal combustion engine, a lot of research has been carried out on hydrogen supply system [2], [3], combustion characteristics [4], [5] and so on, and many areas of research are concerned with the adoption of with in-cylinder type injection system for high pressure hydrogen. This type of injection system can eliminate the possibility of flashback into the intake pipe and can produce more power than an intake port injection system. But this system has a very complicated structure and greater durability problem. To overcome the problems of high-pressure injection in a direct injection diesel engine a port injector system was adopted [6]. In this experiment, an intake port injection system using a solenoid was developed for hydrogen gas injection. In order to minimize the possibility of flashback, the injection duration of hydrogen was varied [7]. Hydrogen was supplied after the opening of the intake valve such that the maximum amount of air fuel mixture was inducted along the intake manifold [8], [9]. The present work involves the study of performance and emission characteristics of hydrogen injection with different injection duration in a DI dual fuel engine. Table 1 shows the fuel properties of hydrogen in comparison with diesel and gasoline.

2. EXPERIMENTAL SET-UP AND PROCEDURE

The engine used for the experimental investigation was a Kirloskar AV1, single cylinder, four stroke, water cooled, direct injection stationary diesel engine, developing a rated power of 3.7 kW at a rated speed of 1500 rpm. The specifications of the test engine are given in Table 2. The engine was coupled to an electrical dynamometer with resistance loading. An electronic control unit (ECU) controls the operation of H_2 fuel injector. One end of the positive power supply from the 12 V battery was connected to the injector; the other negative terminal of the injector to the ECU, which had the control of both the injector opening timing and duration. An infrared detector was used to give signals to the ECU for the injector opening timing. Based on the preset timing and duration the injector was opened for injection and closed after injection. The injection duration was varied within the specified range by using the knob control. The power required for opening the injector was 4A and 1A for holding the injector. Hydrogen flow was controlled by using a pressure regulator and a digital mass flow controller. As the hydrogen flow increases the governor controls the diesel flow automatically. Figure 1 shows the schematic diagram of the experimental set up and the photographic view, in Figure 2.

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Table 1. Properties of hydrogen

Sl. No.	Properties	Diesel	Unleaded Gasoline	Hydrogen
	Formula	$C_n H_{1.8n}$ $C_8 - C_{20}$	$C_n H_{1.87n}$ $C_4 - C_{12}$	H_2
1.	Auto ignition temperature (K)	530	533-733	858
2.	Minimum ignition energy (mJ)	-	0.24	0.02
3.	Flammability limits (volume % in air)	0.7-5	1.4-7.6	4-75
4.	Stoichiometric air fuel ratio (mass basis)	14.5	14.6	34.3
6.	Limits of flammability (equivalence ratio)	-	0.7-3.8	0.1-7.1
7.	Density at 16^0C and 1.01 bar (kg/m^3)	833-881	721 – 785	0.0838
8.	Net heating valve (MJ/kg)	42.5	43.9	119.93
9.	Flame velocity (cm/s)	30	37-43	265-325
10.	Quenching gap in NTP air (cm)	-	0.2	0.064
11.	Diffusivity in air (cm^2/s)	-	0.08	0.63
12.	Octane number research	30	92-98	130
	motor	-	80-90	-
13.	Cetane number	40-55	13-17	-

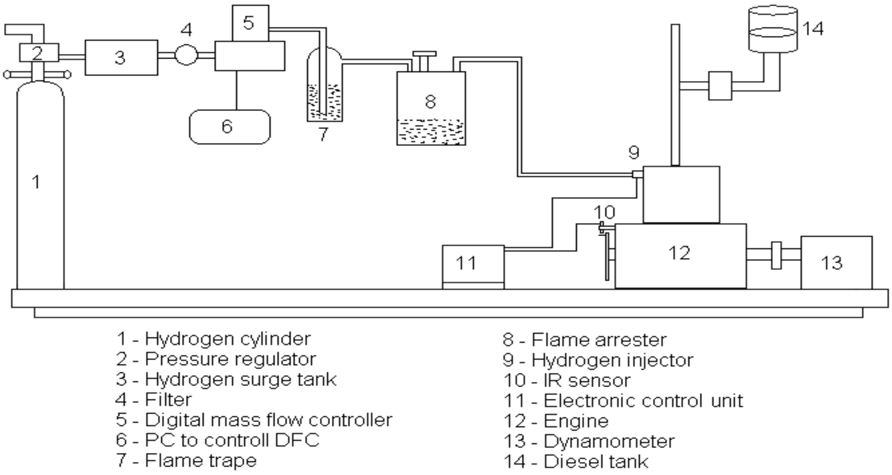


Fig. 1. Schematic diagram of the experimental set-up

Fig. 2. Photographic view of the experimental set-up

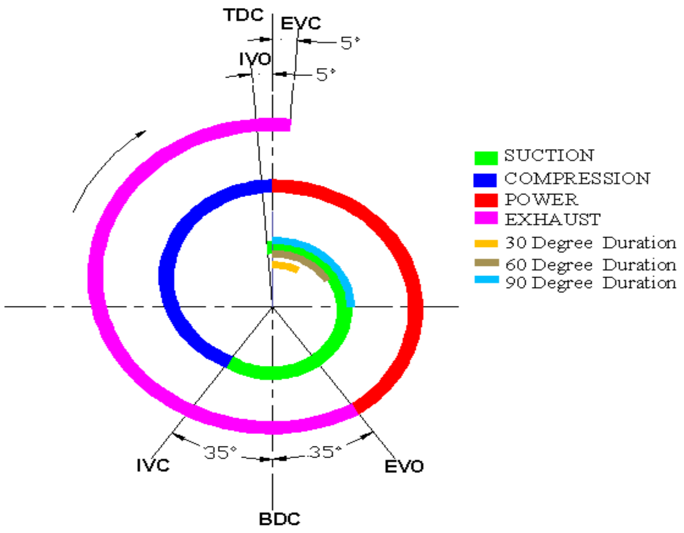
Table 2. Engine Specifications

Make and Model	Kirloskar, AV1 make
General	4-Stroke / Vertical
Type	Compression ignition
Combustion Chamber	Hemispherical open
Number of Cylinder	One
Bore	80 mm
Stroke	110 mm
Swept Volume	553 cc
Compression Ratio	16.5: 1
Rated Output	3.7 kW @ 1500 rpm
Rated Speed	1500 rpm
Type of Cooling	Water cooled

Hydrogen gas stored on a high-pressure cylinder at a pressure of 150 bar was reduced to about 3 to 4 bar by using a pressure regulator. Hydrogen was passed through a fine control valve to adjust the flow rate and then through the mass flow controller, which metered the flow of hydrogen in terms of Standard Liters per Minute (SLPM). Hydrogen was then passed through a flame arrestor, used to suppress the possible fire hazards in the system. These flame arrestors operate on the basic principle that the flame gets quenched if sufficient heat can be removed from the gas by the arrestors, which also acts as a non-return valve. Then hydrogen was allowed to pass through a wet type flame trap, which was used to suppress the flash back if any into the intake manifold. In general wet flashback arrestors work by bubbling the gas through a non-flammable and ideally non-gas-absorbing liquid, in this case the liquid used was water. Hydrogen from the cylinder after passing through the flame trap was injected through the gas injector, which was fitted in the inlet port. The engine was started with diesel and allowed to run for 10 minutes and then hydrogen was introduced in the intake port. The start of injection for hydrogen was fixed at TDC and three injection duration of 30° [3.3 ms], 60° [6.6 ms] and 90° [9.9 ms] crank angle were selected, since the fuel injector can open only for a maximum duration of 10 ms. Figure 3 shows the valve timing diagram indicating the injection timing and injection duration for hydrogen fuel injection.

3. INSTRUMENTATION

The instrumentation used for this experiment is shown in Table 3.



TDC - Top Dead Centre
BDC - Bottom Dead Centre
IVO - Inlet Valve Open
IVC - Inlet Valve Close
EVO - Exhaust Valve Open
EVC - Exhaust Valve Close

Fig. 3. Valve timing diagram indicating the injection timing and injection duration for hydrogen fuel injection

Table 3. Instruments used for experiment

Sl.No	Instrument	Purpose	Make / Model
1	Electrical dynamometer	Measurement of power output	Laurence Scott and elctromotor Ltd., Norwich and Manchester, UK, Capacity-10kW, Current Rating-43 amps
2	Exhaust gas analyser	Measurement of HC, CO, CO_2 and NO_X	QRO 401, Qrotech Corporation Limited, Korea
3	Smoke meter	Measurement of Smoke	TI diesel tune, 114 smoke density tester TI Tran service
4	Pressure transducer and charge amplifier	Measurement of Cylinder Pressure	Type 5015A, Kistler Instruments, Switzerland
5	Digital mass flow controller	Measuring the H_2 flow	DFC 46 mass flow controller AALBORG, USA
6	Hydrogen leak detector	To detect the H_2 leakage	Finch Mono II, Portable single gas monitor, INIFITRON INC, Korea

4. ESTIMATION OF UNCERTAINTY

Any experimental measurement, irrespective of the type of instrument used, possess a certain amount of uncertainty. The uncertainty in any measurement may be due to either fixed or random errors. As the fixed errors are repeatable in nature they can be easily accounted for to get the true value of measurement. However random errors have to be estimated only analytically. The details of the estimated average uncertainties of some measured and calculated parameters at some typical operating conditions are given in Table 4. It can be observed that the uncertainty ranges from 0.5 to 3.2 %.

Let R be the computed result function of the independent measured variables x_1, x_2, x_3, …..,x_n as per the relation.

$$R=f(x_1, x_2, x_3, \ldots, x_n) \quad (1)$$

and let the error limits for the measured variables or parameters be:

$$x_1 \pm \Delta n_1 , x_2 \pm \Delta n_2 ,\ldots\ldots x_n \pm \Delta n_n \quad (2)$$

and the error limits for the computed results be R± ΔR. Hence to get the realistic error limits for the computed result the principle of root mean square method to get the magnitude of error.

$$\Delta R=[(\partial R/\partial x_1\, \Delta x_1)^2 + (\partial R/\partial x_2\, \Delta x_2)^2 + \ldots + (\partial R/\partial x_n\, \Delta x_n)^2]^{0.5} \quad (3)$$

Using Equation 3, the uncertainty in the computed values such as brake power, brake thermal efficiency and fuel flow measurements were estimated. The measured values such as speed, fuel time, voltage and current were estimated from their respective uncertainties based on the Gaussian distribution. The uncertainties in the measured parameters, voltage (ΔV) and current (ΔI), estimated by the Gaussian method, are ± 0.16 A respectively. For fuel time (Δt_f) and fuel volume (Δt), the uncertainties are taken as ± 0.2 sec and ± 0.1 cc respectively. Sample calculation is given in the Appendix.

Table 4. Average uncertainties of some measured and calculated parameters

S.No	Parameters	Uncertainty
1	Speed	1.1 %
2	Temperature	0.5 %
3	Mass flow rate of air	1.3 %
4	Mass flow rate of diesel	1.9 %
5	Mass flow rate of hydrogen	1.6 %
6	Oxides of nitrogen	2.4 %
7	Hydrocarbon	2.2 %
8	Smoke	3.2 %
9	Particulate matter	3.1 %
10	Pressure	0.8 %
11	Heat Release	0.7 %

5. RESULTS AND DISCUSSION

In the present work, adopting timed port injection technique in Compression Ignition (C.I.) engine with diesel being the ignition source uses hydrogen gas-air mixture. The performance and emission characteristics were studied and compared with baseline diesel operation. In the test, the start of injection was fixed at TDC position and the hydrogen duration was fixed at 30˚, 60˚, and 90˚ CA. The hydrogen flow rate was fixed constant at 20 lpm for all the load conditions.

Brake Thermal Efficiency

The variation of brake thermal efficiency with brake power is shown in Figure 4. At 75 % load (3 kW) the highest brake thermal efficiency of 27.8 % is obtained for hydrogen duration of 90° compared to diesel of 21.8 %. At full load (3.7 kW) the brake thermal efficiency of diesel is found to be 23.4 % compared to 24.8 % for hydrogen operation at 60° injection duration. For 90° crank angle duration at full load there was an onset of knock which results in a drop in efficiency. The increase in brake thermal efficiency is attributed to better mixing of hydrogen with air, which results in better combustion and also the operation of engine at leaner equivalence ratios.

Specific Energy Consumption

The variation of specific energy consumption with brake power is shown in Figure 5. The specific energy consumption is reduced by 15 % for hydrogen with 90˚ injection duration compared to diesel at 75 % load. At full load it is observed that the specific energy consumption of hydrogen operated engine is lower than diesel for all the injection timings. The lower SEC of 14.47 MJ/kWh is observed for 60° injection duration compared to 17.1 MJ/kWh for diesel at full load. The lower specific energy consumption is due to uniform mixing of hydrogen with air resulting in better combustion than neat diesel fuel operation.

Oxides of Nitrogen

It can be observed from Figure 6, that NO_X emission in timed port injection technique is slightly higher than that of diesel. The higher concentration of NO_X may probably be due to the increase in peak combustion temperature for hydrogen . At 75 % load the NO_X emission is found to increased from 1980 ppm for diesel to 2070 ppm for hydrogen at 60° and 90° injection duration at full load. With 60° crank angle duration the NO_X emission increases by 10 % compared to diesel.

Hydrocarbons

Figure 7 depicts the variation of hydrocarbon emissions with brake power. The hydrocarbon is found to be 28 ppm for diesel at 75 % compared to 4 ppm for hydrogen operation at 90° injection duration. At full load for diesel operation the hydrocarbon is 42 ppm compared to 7 ppm for hydrogen operation at 90° injection duration. The main reason for the reduction of hydrocarbon is hydrogen is a non-hydrocarbon fuel.

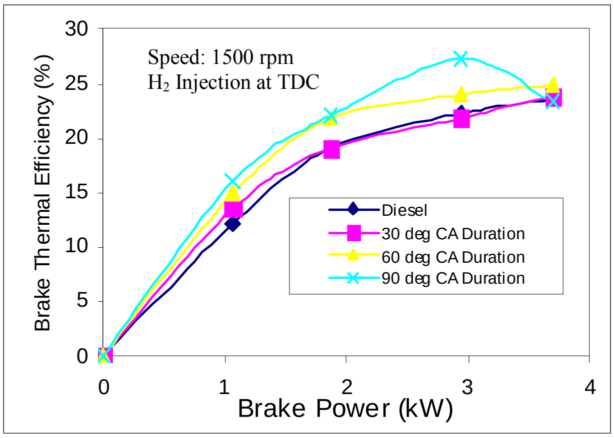


Fig. 4. Variation of Brake Thermal Efficiency with brake power

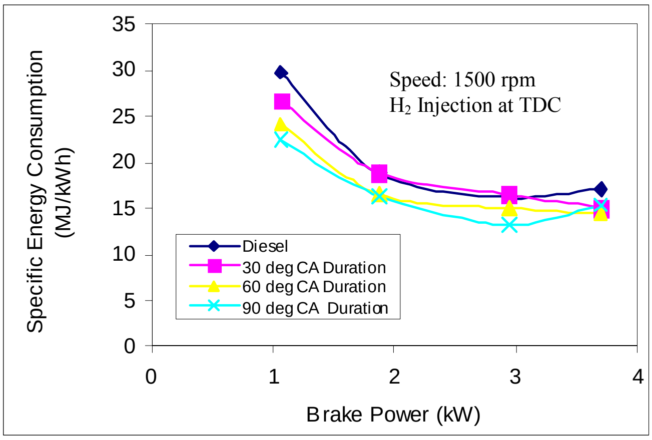


Fig. 5. Variation of Specific Energy Consumption with brake power

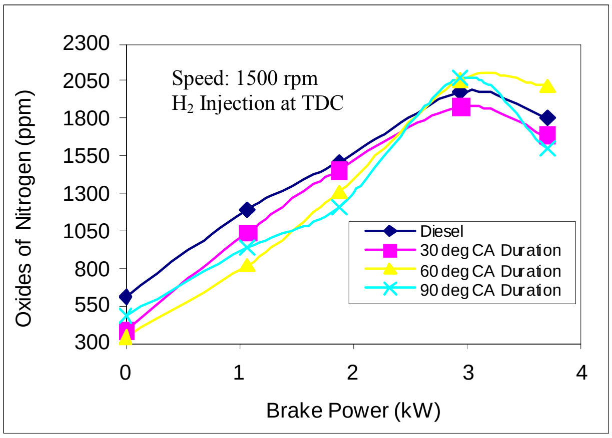


Fig. 6. Variation of oxides of Nitrogen with brake power

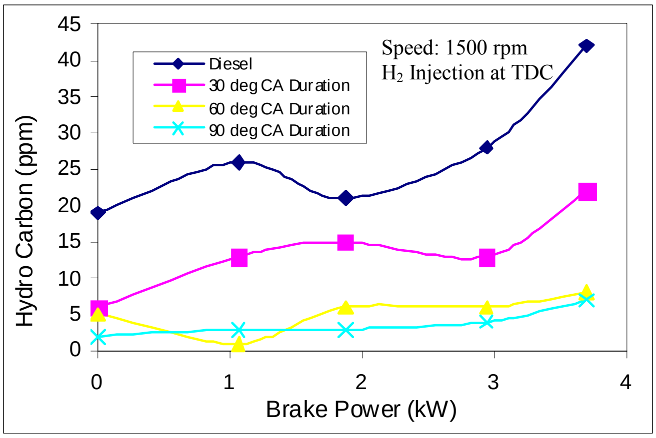


Fig. 7. Variation of hydrocarbon with brake power

Carbon Monoxide

The variation of carbon monoxide emissions with brake power is shown in Figure 8. The CO emissions are lower compared with the base line diesel, for diesel it is found to be 0.05 % by Vol. at 75 % load compared to 0.01 % by Vol. for hydrogen with 60° injection duration. At full load the CO values are far lower than 75 % load condition. The CO is found to be 0.17 % by Vol. for diesel compared to 0.05 % by Vol. for 60° injection duration for hydrogen operation. The CO emissions are lesser because of the reason that hydrogen does not contain any carbon in its structure.

Smoke

The variation of smoke level with brake power is shown in Figure 9. The smoke level is reduced at full load compared to baseline diesel. Hydrogen on combustion produces mainly water and does not form any particulate matter, hence lower smoke level. The smoke level increases with increase in diesel flow due to the formation of particulate matter by diesel fuel. In general the smoke values are decreased with hydrogen intake due to the partial replacement of diesel by hydrogen and improved combustion of diesel due to its simultaneous burning along with hydrogen. The smoke value reduces from 4.06 BSN for diesel at full load to 1.2 BSN for hydrogen injection with 60° crank angle duration.

Carbon Dioxide

Figure 10 depicts the carbon dioxide variation with brake power. The CO_2 emissions are lower compared with diesel for 60° and 90° crank angle duration. The reduction in CO_2 at 75 % load is from 9.5 % by Vol. for diesel to 2.2 % by Vol. for hydrogen operation for 60° injection duration. At full load the highest carbon dioxide is found to be 11.6 % by Vol. for diesel compared to hydrogen of 3 % by Vol. for 60° injection duration. The CO_2 emission of hydrogen is lowered because of the absence of carbon in hydrogen.

Exhaust Gas Temperature

The variation of exhaust gas temperature with brake power is shown in Figure 11. The exhaust gas temperature is higher by 40-50° C at full load for hydrogen operations compared to diesel of 247° C. At 75 % load the variations are still higher by 70-80° C for all operations of hydrogen. This may be due to the better combustion of hydrogen fuel in port injection technique.

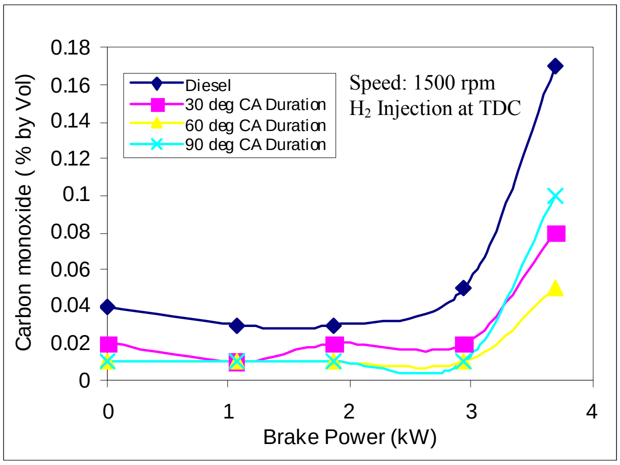


Fig. 8. Variation of carbon monoxide with brake power

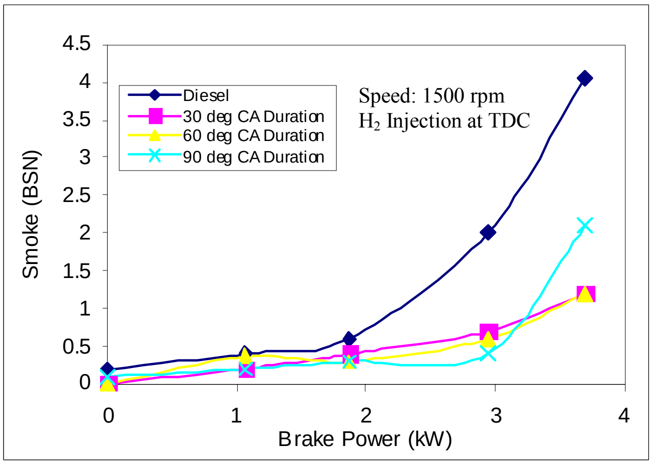


Fig. 9. Variation of smoke with brake power

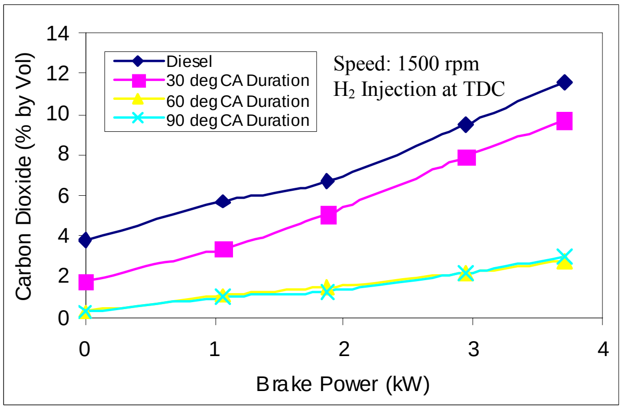


Fig. 10. Variation of carbon dioxide with brake power

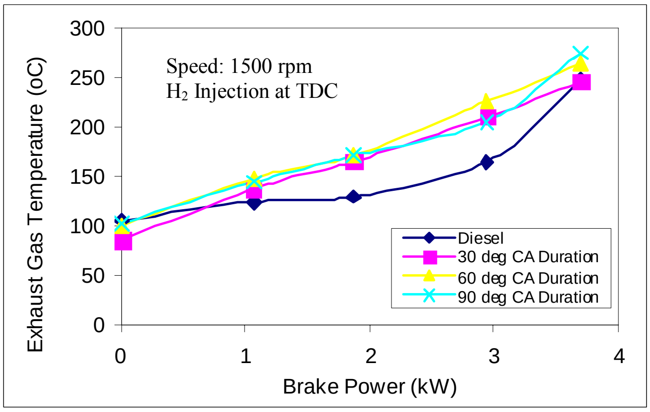


Fig. 11. Variation of exhaust gas temperature with brake power

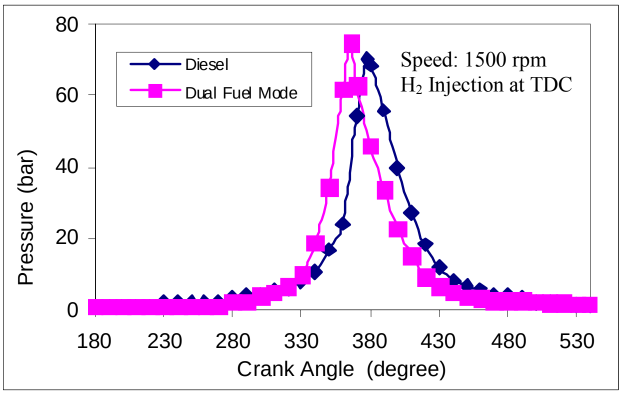


Fig. 12. Variation of cylinder pressure at full load

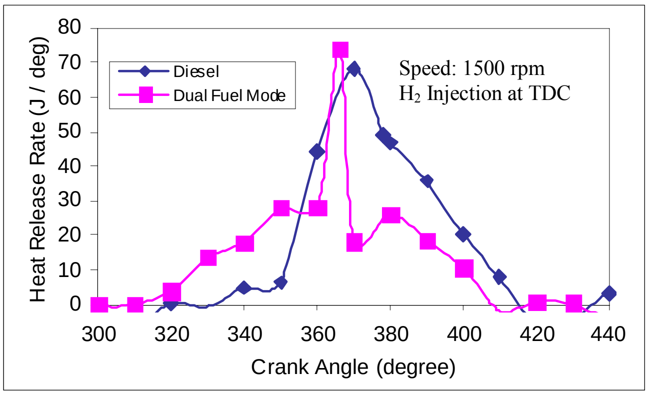


Fig. 13. Variation of heat release rate at full load

Pressure Crank Angle Diagram

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion. The pressure crank angle diagram for diesel and hydrogen with diesel dual fuel mode is shown in Figure 12. There is a delay of few crank angle degrees between the start of injection and start of combustion, as identified by the change in slope of pressure crank angle curve. It can be observed that a steep pressure rise occurs in dual fuel mode, since hydrogen burns faster than diesel fuel.

Heat Release Rate

Figure 13 shows the variation of heat release for hydrogen- diesel combustion at TDC and 90° injection duration at full load condition. It is evident that, the heat release for hydrogen is steeper than diesel. It can be observed that, hydrogen-diesel fuel mixture shosw the highest heat release rate of 75 J / degree CA compared to diesel of 68 J / degree CA. This is due to the property of quick combustion (constant volume) taking place with hydrogen fuel.

6. CONCLUSION

Experiments were conducted to study the performance and emission characteristics of a DI diesel engine using hydrogen gas by means of timed port injection technique with diesel as the mode of ignition. The emissions such as CO, CO_2, and HC are reduced drastically to negligible concentrations. There is an improvement in performance of the engine with reduction in SEC. The pressure variation shows that in hydrogen fuelled operation, the peak pressure increases rapidly. Thus the present experimental investigation on a single cylinder diesel engine indicates that by using hydrogen as a fuel adopting timed port injection technique gives better efficiency and reduced emissions compared to neat diesel fuel operation.

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APPENDIX

Sample calculation for uncertainty is given below:

Speed (N) = 1500 rpm
Voltage (V) = 230 Volts
Current (I) = 12 A
Fuel volume f_v = 10 cc
Brake power (BP) = 3.2 kW

$$BP = (VI/\eta_g \text{ x } 1000) \text{ kW}$$
$$BP = f(V,I)$$
$$\partial_{BP/}\partial I = I/(0.85\text{x}1000) = 12/(0.85\text{x}1000) = 0.014$$
$$\partial_{BP/}\partial V = V/(0.85\text{x}1000) = 230/(0.85\text{x}1000) = 0.271$$
$$\Delta_{BP} = [\surd\{(\partial_{BP/}\partial V)\text{x } \Delta V\}^2 + \{(\partial_{BP/}\partial I)\text{x } \Delta I\}^2]$$
$$\Delta_{BP} = [\surd\{(0.014X10)^2 + (0.271X0.16)^2\}]$$
$$\Delta_{BP} = 0.147 \text{ kW}$$

Therefore, the uncertainty in the brake power from equation 1 is ± 0.147 kW and the uncertainty limits in the calculation of BP are 3.2 ± 0.1474 kW.