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Effect of hydrogen-diesel combustion on the performance and combustion parameters of a dual fuelled diesel engine

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Abstract
Petroleum crude is expected to remain main source of transport fuels at least for the next 20 to 30 years. The petroleum crude reserves however, are declining and consumption of transport fuels particularly in the developing countries is increasing at high rates. Severe shortage of liquid fuels derived from petroleum may be faced in the second half of this century. In this paper, experiments are performed in a fur stroke, single cylinder, compression ignition diesel engine with dual fuel mode. Diesel and hydrogen are used as pilot liquid and primary gaseous fuel, respectively. The objective of this study is to find out the effects on combustion and performance parameters observed at diesel hydrogen fuel mixture for all the different loadings (2kg,4kg,6kg,8kg,10kg and 12kg) in the engine.

Keywords: Diesel engine; Hydrogen; Dual fuel; Efficiency.

1. Introduction

In this century, it is believed that the crude oil and petroleum products will become very scarce and costly. Day- to-day, fuel economy of engine is getting improved and will continue to improve. However, enormous increases in number of vehicles have started dictating the demand for fuel. Gasoline and diesel will become scarce and most costly in the near future. With increased use and the depletion of fossil fuels, alternative fuel technology will become more common in the coming decades. The search for an alternative fuel [1], which promises a harmonious correlation with sustainable development, energy conservation, management, efficiency, and environmental preservation, has become highly pronounced in the present context. Many researches have been directed towards the development of alternative fuels to achieve this goal. Some of the alternative fuels that have been used to replace petroleum-based fuels include vegetable oils, alcohols, liquefied petroleum gas (LPG), liquefied natural gas (LNG), compressed natural gas (CNG), bio gas, producer gas, hydrogen etc. In this context, hydrogen (H2), a non-carboniferous and non-toxic gaseous fuel, has attracted great interest and has huge potential. H2 is only one of many possible alternative fuels that can be derived from various natural resources. In a compression ignition (CI) engine, however, H2 cannot be directly used due to its higher self-ignition temperature, but it can be used in the dual fuel mode.

The concept of using hydrogen as an alternative fuel for diesel engines is recent. The self-ignition temperature of hydrogen is 858 K, so hydrogen cannot be used directly in a CI engine without a spark plug or glow plug. This makes hydrogen unsuitable as a sole fuel for diesel engines [2]. One alternative method is to use hydrogen in enrichment or induction, in which diesel is used as a pilot fuel for ignition. As hydrogen is a gas, it mixes well with air, resulting in complete combustion. Hydrogen-enriched engines produce approximately the same brake power and higher thermal efficiency than diesel engines over the entire range of operation [3, 4]. This work involves the enrichment of air with various percentages of hydrogen in a diesel engine using diesel as an ignition source. With a lesser pilot quantity of diesel, hydrogen-enriched engines give higher brake thermal efficiency with smoother combustion than a diesel engine. Increasing hydrogen beyond a certain quantity results in knocking; at the highest diesel flow rate, thermal efficiency is found to be the same as that of diesel engines. Hence, the overall behavior of the engine is similar to that of a diesel engine. Yi et al. [5] stated that thermal efficiency of intake port injection is clearly higher than in-cylinder injection at all equivalence ratios. Shudo et al. [6] stated that hydrogen combustion exhibits higher cooling loss to the combustion chamber wall than does hydrocarbon combustion because of its higher burning velocity and shorter quenching distance. Hydrogen used in the dual fuel mode with diesel by Masood et al. [7] showed the highest brake thermal efficiency of 30% at a compression ratio of 24.5. Lee et al. [8] studied the performance of dual injection hydrogen fueled engine by using solenoid in-cylinder injection and external fuel injection technique. An increase in thermal efficiency by about 22% was noted for dual injection at low loads and 5% at high loads compared to direct injection. Lee et al. [8] suggested that in dual injection, the stability and maximum power could be obtained by direct injection of hydrogen. However the maximum efficiency could be obtained by the external mixture formation in hydrogen engine. Das et al. [9] have carried out experiments on continuous carburetion, continuous manifold injection, timed manifold injection and low pressure direct cylinder injection. The maximum brake thermal efficiency of 31.32% was obtained at 2200 rpm with 13 Nm torque. Lee et al. [10] studied the characteristics of a solenoid-driven intake port injection type hydrogen injection valve. In this study, an intake port injection system was used with a solenoid valve for injection purpose to study the combustion characteristics of the fuel. It was observed that the hydrogen operated engine showed improved performance by 9% compared to normal operation. Varde and Frame [11] figured out that the brake thermal efficiency of H2diesel dual fuel mode is primarily dependent upon the amount of H2 added. The larger the amount of H2, the higher the value of brake thermal efficiency is. It has been seen in H2 diesel dual fuel mode that 90% enriched H2 gives higher efficiency than 30% at 70% load, but cannot complete the load range beyond that due to knocking problems. However, brake thermal efficiency was found to drop when the amount of H2 is less than or equal to 5%. In few analysis, an extremely lean air H2 mixture restricts the flame to propagate faster, which lowers H2 combustion efficiency. However, experimental works done later, with H2 diesel dual fuel mode, do not prove this drop in brake thermal efficiency with H2 addition as mentioned above. A study performed by Wang and Zhang [12] indicates that the introduction of hydrogen into the diesel engine causes the energy release rate to increase at the early stages of combustion, which increases the indicated efficiency. This is also the reason for the lowered exhaust temperature. According to them, for fixed H2 supply at 50%, 75% and 100% load,H2 replaces 13.4%, 10.1% and 8.4% energy respectively with high diffusive speed and high energy release rate. Guo et al. [13] studied the combustion system of a gasoline engine by using hydrogen fuel and also the effect of abnormal combustion. They found that the abnormal combustion could be controlled by in-cylinder injection instead of manifold injection. Hydrogen injector used in this system is to improve the hydrogen jet penetration and mixture formation in the combustion chamber, which in-turn reduced the rapid combustion in the combustion chamber. From their results they concluded that the shape of the combustion chamber strongly affects the performance of the engine. Furuhama [14] suggested that to prevent the pre-ignition of hydrogen in the intake manifold, the hydrogen is to be supplied into the intake system only during the suction period or the hydrogen is to be injected into the cylinder only during the intake period with a relatively low pressure which in-turn can also avoid backfire. A few researchers [15-25] have studied the variation of H2-diesel quantity for constant diesel supply at each load to improve the brake power (BP) and brake thermal efficiency. The increase in the supply of H2 in inlet manifold causes a reduction in the air flow to the engine. As a result, the volumetric efficiency and consequently the brake thermal efficiency of the engine reduces. Therefore, there is scope to study and understand engine performance by varying both H2 and diesel supply while maintaining constant BP at each load condition. In light of this fact, the objective of the present work is to study the combustion and performance parameters of the existing diesel by varying the quantity of fuel (pilot and primary) and maintaining constant speed and BP at each of the load conditions for the composition of H2-diesel at a particular injection timing of 10degree ATDC with varying injection durations. Some of the important physical and thermodynamic properties of diesel and H2 are shown in Table 1.

Table 1. Properties of hydrogen(H2) and diesel

Properties	Diesel	Hydrogen
Chemical composition	$C_{12}H_{26}$	H_2
Specific gravity	0.83	0.091
Density at 160 C and 1.01 bar (kg/m3)	833–881	0.0838
Cetane number	40–55	-
Calorific value(MJ/Kg)	42	119.81
Flammability limits (volume % in air)	0.7–5	4-75
Auto-ignition temperature (K)	553	858
Stoichiometric air fuel ratio	14.92	34.3
Energy density(MJ/Nm^3)	2.82	2.87

2. Experimental setup and procedure

The experiments are carried out in a Apex made water cooled, single cylinder, four stroke, direct-injection (DI), vertical diesel engine running at a rated power of 5.2kW with compression ratio 17.5:1 and at a rated speed of 1500 rpm installed at the Internal Combustion Engine laboratory of National Institute of Technology, Agartala, India. Figure 1 shows a schematic diagram of the engine test bed. The original engine specifications are shown in Table 2. The engine loading is performed by an eddy current type dynamometer. The liquid fuel is supplied to the engine from the fuel tank through a fuel pump and injector. The fuel injection system of the engine consists of an injection nozzle with three holes of 0.3 mm diameter with a 120^0 spray angle (Figure 2 and Figure 3). An U-tube type manometer is used to quantity the head difference of air flow to the engine, while allowing the intake air to pass through an orifice meter. The engine block and cylinder head are surrounded by a cooling jacket through which water flows to cool the engine. To measure the specific heat of exhaust gas, a calorimeter of counter flow pipe-in pipe heat exchanger is also provided. Temperature measurement is performed by K-type thermocouples, which are fitted at relevant positions. In order to convert the diesel engine test bed into dual fuel mode, some additional equipment is installed in the setup. These include: hydrogen gas cylinder with regulator, gas flow meter, flame arrester and a bubbler tank. A flow regulator in the hydrogen circuit helps to regulate the volume flow rate of hydrogen in the hydrogen circuit according to ones desire of the amount of hydrogen to be inducted per computational cycle. A flame arrestor that also acts as a non-return valve (NRV) was provided to suppress any possible fire hazard in the system. In addition a bubbler tank which acts as a flame trap is included in the circuit to dampen out any pressure fluctuations in the hydrogen supply line. The gas flow meter measures the gas flow rate of hydrogen; while the flame trap and the flame arrester are used to prevent fire hazards due to accidental engine backfire. In the dual fuel mode H2 is supplied to the engine by the induction method. Hydrogen is supplied from a high pressure cylinder (150 bars) to an outlet pressure of 1 bar using hydrogen pressure regulator. Hydrogen is then passed through a fine control valve to adjust the flow rate of hydrogen. Then it passes through the gas flow meter, which measures the flow of hydrogen gas. Then hydrogen is passed through an NRV, which prevents the reverse flow of hydrogen into the system. Such a possibility of reverse flow can occur sometimes in a hydrogen injected engine, particularly in the latter part of injection. Hydrogen is then passed through a flame arrestor, used to suppress the explosion inside the hydrogen containing system. The flame arrestors operate on the basic principle that the flame gets quenched if sufficient heat can be removed from the gas by the arrestor. It also acts as an NRV. Then hydrogen is allowed to pass through the flame trap, which is used to suppress the flashback if any into the intake manifold. The hydrogen from the flame trap is sent into the inlet manifold which is used to mix the air with hydrogen. The process of mixing air and fuel is called as enrichment. An electrical dynamometer is used to measure the brake power. Initially, the engine is allowed to run on diesel at no load condition for a few minutes to attain a steady state. The cooling water supplies for the engine and calorimeter are set to 270 and 80 liters per hour, as per the engine provider instructions. Thereafter, the load is gradually increased to 2.4 kg (20% load) and the engine is allowed to run until it reaches a steady state. Then, the inlet and outlet temperatures of engine cooling water, calorimeter cooling water and exhaust gas are measured. The diesel was supplied at a constant rate through the fuel injector. Hydrogen was then allowed to flow from the high pressure cylinder through the flame arrester to bubbler tank and from the tank to the inlet manifold through a gas injector at much reduced supplied pressure of 1bar. After the dual fuel operation started then gradually the load was varied from 2kg, 4kg, 6kg, 8kg, 10kg and 12kg and at each load condition the combustion and performance parameters were measured. Finally, the hydrogen supply is stopped completely, and the engine is allowed to run at "no load condition" prior to complete shutdown. The load is increased by the eddy current dynamometer. The experimental engine being a constant speed one, the incorporated fuel governor would adjust the fuel rack position to deliver the requisite amount of diesel fuel in the event of load change reflected by an instantaneous change in engine speed which would have been adjusted according to the motion of the fuel rack to the same value before the change of load. This would only be possible if the amount of injected diesel fuel would increase or decrease with the load, thus registering different specific fuel consumption at each setting of load.

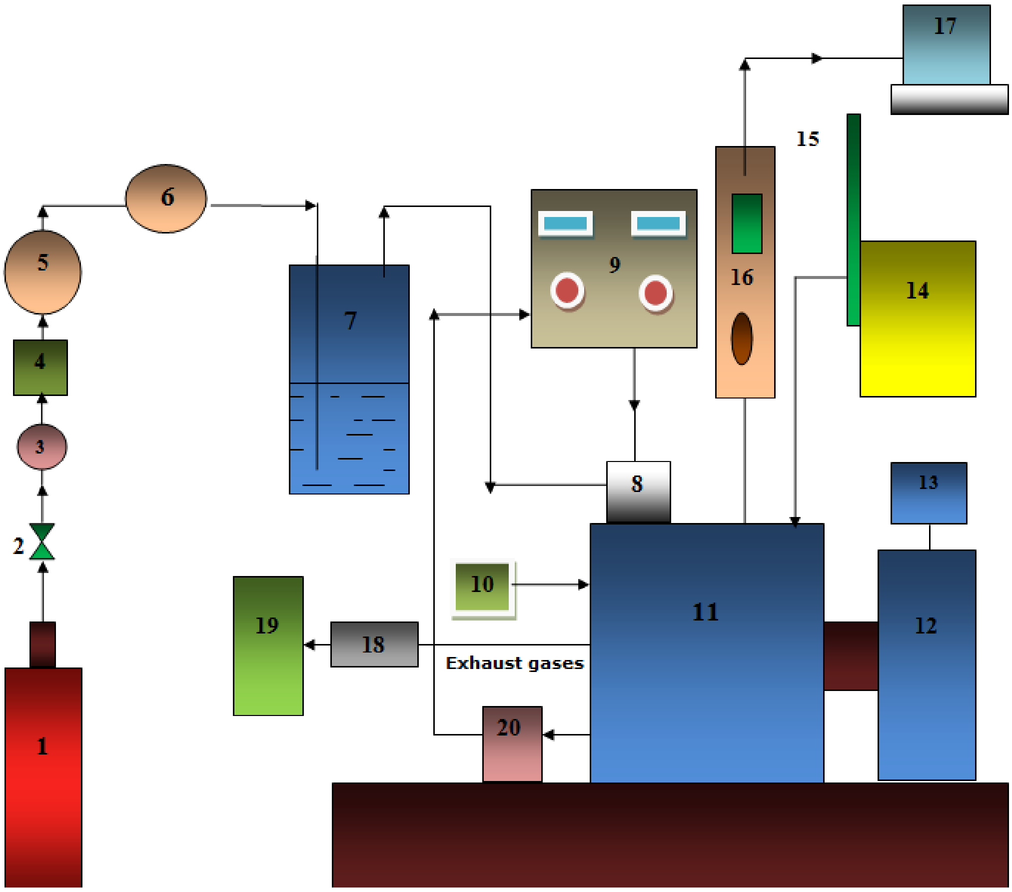

Figure 1. Schematic layout of the experimental setup
1. Hydrogen cylinder, 2. Pressure regulator, 3. Pressure gauge, 4. Flame arrester, 5. Pressure reducer, 6. Gas flow meter, 7. Water bubbling tank, 8. Gas injector, 9. DAQ station with injector driver & DAQ card and PC, 10. Air filter, 11. VCR engine, 12. Electrical dynamometer, 13. Loading device, 14. Fuel tank, 15. Burette, 16. Engine control panel with sensors, 17. Computer panel, 18. Exhaust Gas Calorimeter, 19. Gas Analyzer, 20. Crank Angle Encoder

Table 2. Diesel engine specification

Parameter	Specification
Engine type	Kirloskar TV1
General details	Single cylinder, four stroke diesel,water cooled, compression ignition
Bore and stroke	87.5X110 mm
Compression ratio	17.5:1
Rated output	5.2 kW (7 BHP) 1500 rpm
Air box	With orifice meter and manometer
Dynamometer	Eddy current loading unit, 0–12 kg
Fuel injection opening	205 bar 23^{0}BTDC static

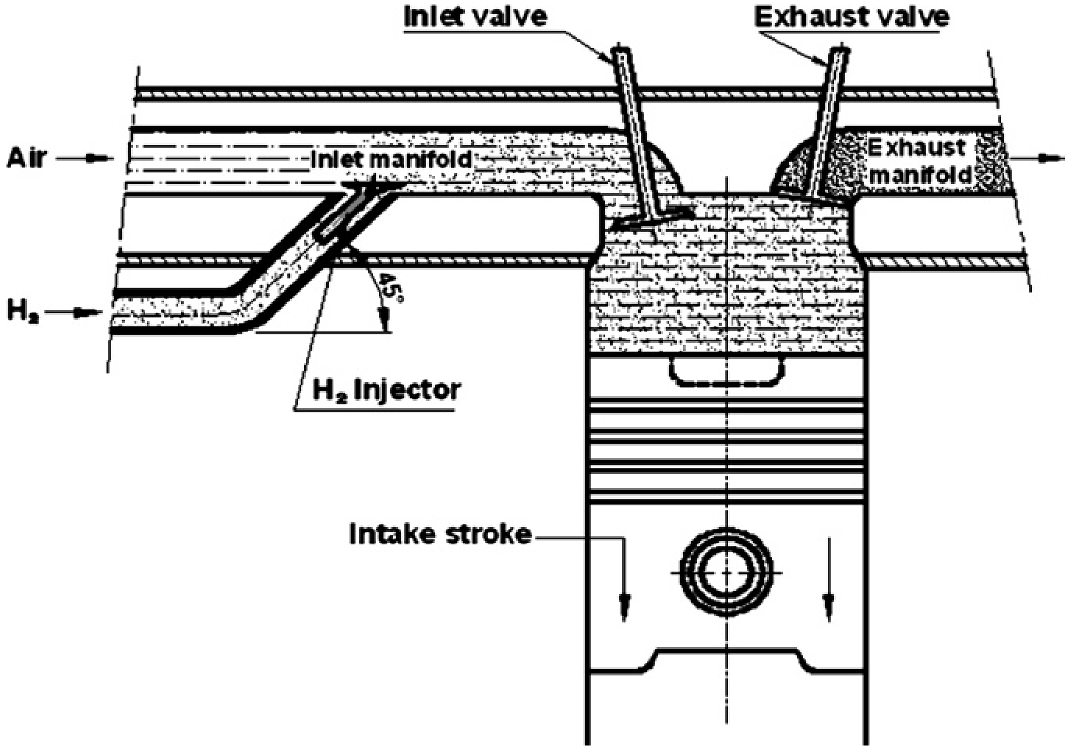

Figure 2. Schematic of the manifold layout injector

Figure 3. Photographic view of the manifold injector

3. Results and discussions
Experiments were carried out with hydrogen and diesel in dual fuel operation. We investigated the combustion and performance characteristics of a DI diesel engine enriched with hydrogen by varying injection duration of hydrogen (9exp(+3) microseconds 15exp(+3) microseconds and 21exp(+3) microseconds.) at 10^0ATDC of injection timing and compared our results with base fuel operation.

3.1 Combustion characteristics

3.1.1 Pressure Vs crank angle

From Figures 4, 5 and 6 it can be observed that the maximum value of peak pressure is observed to be 62.69 bar for 10degree ATDC at full load condition for first injection strategy of hydrogen which is very much comparable to base diesel. The peak pressure rise corresponds to the large amount of fuel burnt in premixed combustion stage and at full load condition the combustion becomes non-homogeneous as the supply of pilot fuel by the fuel injection increases with increase load on the governor, so inside the combustion chamber it becomes a rich mixture. The advance in peak pressure in hydrogen combustion may be due to the reason that hydrogen undergoes instantaneous combustion compared to diesel. As hydrogen has a higher self ignition temperature, so the mixture of hydrogen and diesel takes more time to ignite, during this period more charges get accumulated inside the combustion chamber, due to which rapid combustion takes place. At low engine load only a small amount of fuel is burned and the fuel is burned mainly in the premixed burning phase. As the engine load decreases, the residual gas temperature and wall temperature decrease which leads to lower charge temperature at injection timing, and lengthens the ignition delay.

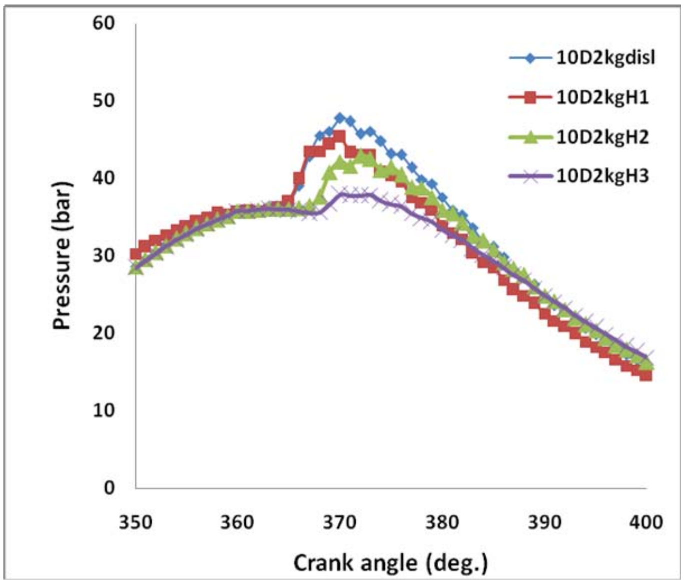

Figure 4. Pressure v/s crank angle @2kg

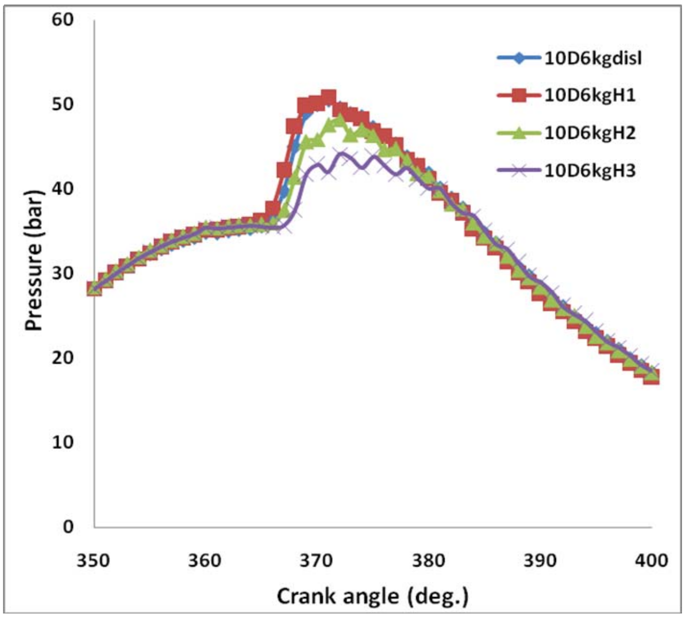

Figure 5. Pressure v/s crank angle @6kg

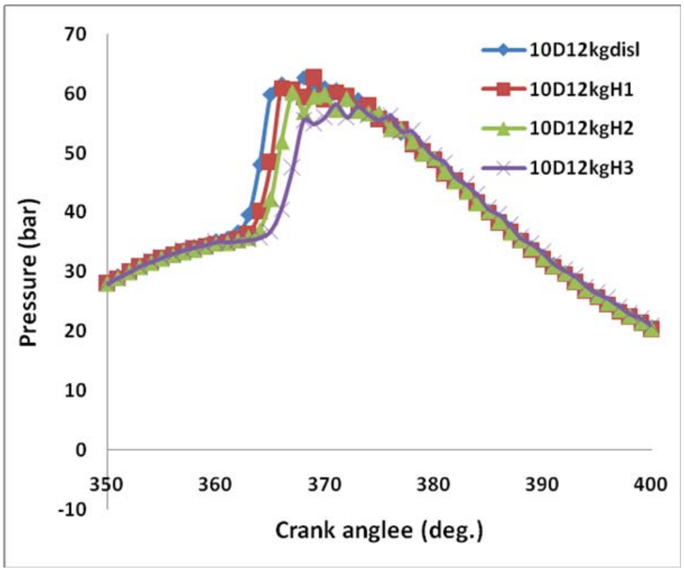

Figure 6. Pressure v/s crank angle @12kg

3.1.2 Rate of change of cylinder pressure with crank angle (dp/dθ)

From Figures 7, 8 and 9 it can be observed that the rate of pressure rise w.r.t crank angle is maximum for pure diesel only at full load condition which is of 7.553 bar per crank angle at 364 degree crank angle. In case of hydrogen injection strategies the maximum rate of pressure rise is 7.551bar at 365 degree crank angle for 10degree ATDC at first injection strategy of hydrogen i.e at 9exp(+3) microseconds which is very near to that base diesel. This increase in the pressure rise is due to high flame speed of hydrogen which promotes better combustion as compared to other injection strategies.

3.1.3 Net heat release (NHR) Vs crank angle

The variation of heat release rate at different engine load conditions is shown in Figures 10, 11 and 12. Because of the evaporation of the fuel accumulated during ignition delay period, at the beginning a negative heat release rate is observed. After combustion is initiated this has become positive. It can be seen that the maximum net heat release was observed to be 88.04KJ at first injection window of hydrogen at a crank angle of 365 degree for maximum load. After the ignition delay, premixed fuel–air mixture burns rapidly, follow by diffusion combustion, where the burn rate is controlled by fuel–air mixing velocity. The lower the ignition delay the greater the fuel accumulated in the combustion chamber at the time of premixed burning phase leading to higher net heat release rate. The higher heat release rate is also an indicative of the effective rate of completeness of combustion.

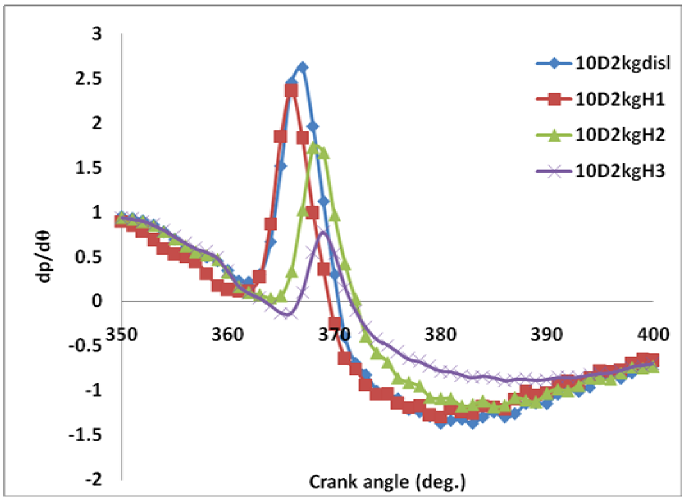

Figure 7. dp/dθ v/s crank angle @2kg

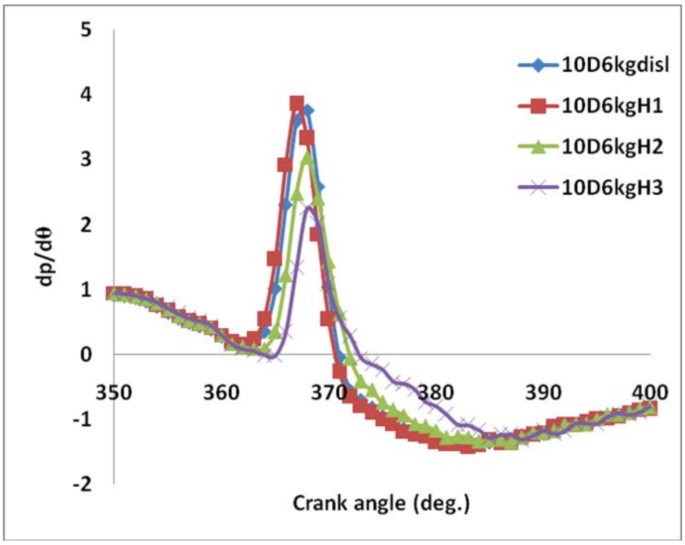

Figure 8. dp/dθ v/s crank angle @6kg

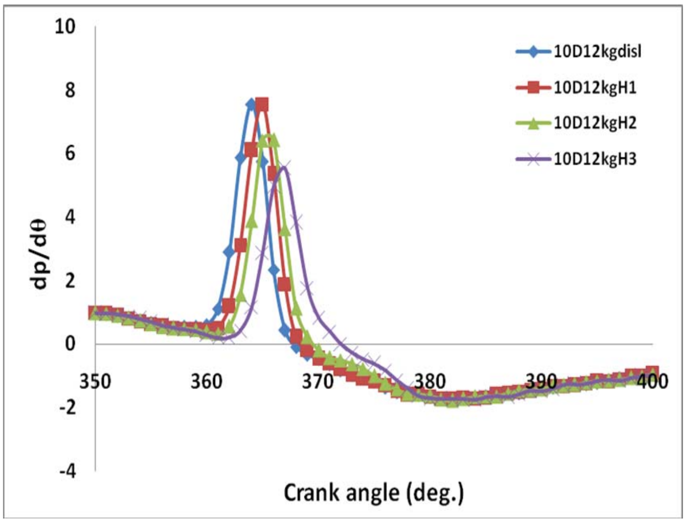

Figure 9. dp/dθ v/s crank angle @12kg

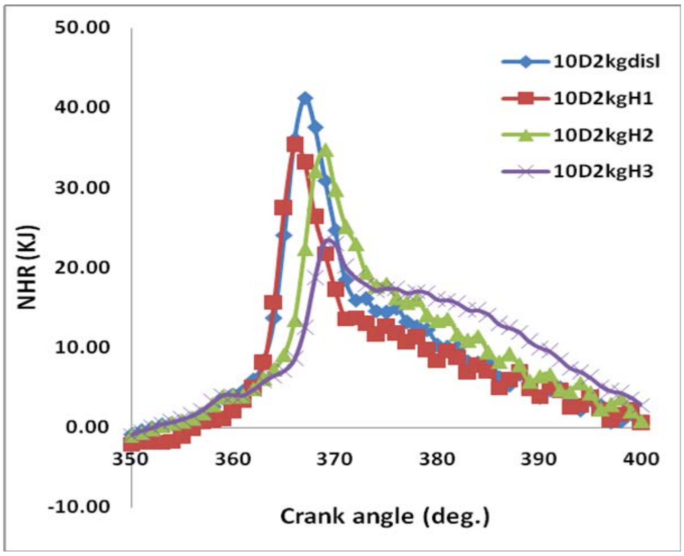

Figure 10. NHR v/s crank angle @2kg

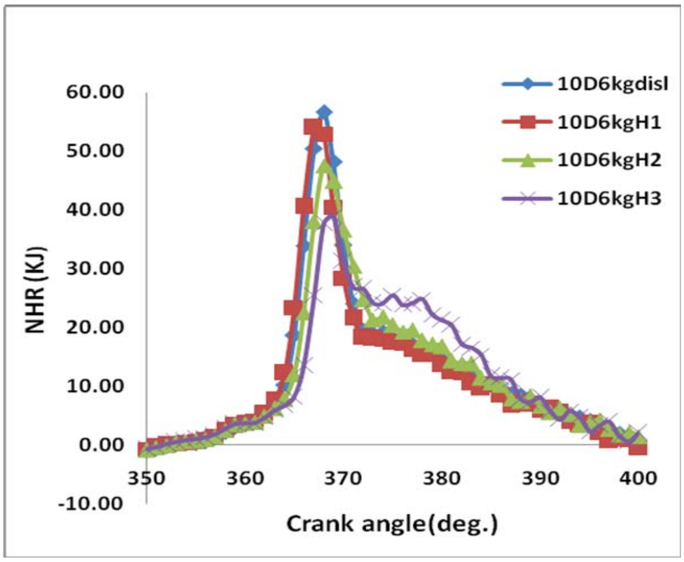

Figure 11. NHR v/s crank angle @6kg

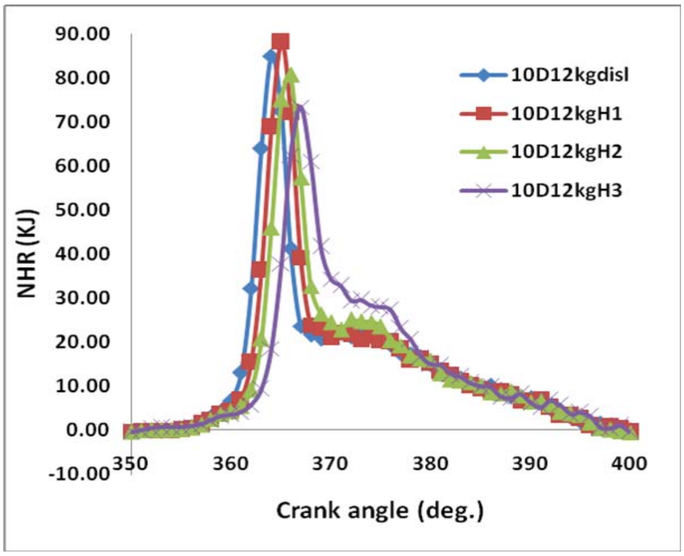

Figure 12. NHR v/s crank angle @12kg

3.1.4 Mean gas temperature (MGT)

The variation of Mean gas temperature with crank angle can be seen in Figures 13, 14 and 15 at different load conditions. it can be observed that the maximum mean gas temp. of 1924K is observed in case of 10degree ATDC with third injection strategy of hydrogen i.e at 21exp(+3) microseconds duration at full load condition with an increase of 3.2% compared to base diesel which is an indicative higher flame propagation accompanied with better combustion which may be due to better oxygenic environment. The increase in mean gas temperature is due to the increase in peak cylinder temperature and peak pressure, which is due to instantaneous combustion that takes place in hydrogen combustion.

3.2 Performance characteristics

3.2.1 Brake thermal efficiency(BTHE)

Figure 16 shows the variation of brake thermal efficiency with respect to load. It has been observed that an increase in brake thermal efficiency for all hydrogen injection strategies compared to base diesel at all load operations and is reaches to a maximum of 30% at full load conditions. The increase in brake thermal efficiency for hydrogen operation is due to hydrogen enrichment with air. Increase in thermal efficiency is attributed to improved combustion because of enhanced combustion rate due to high flame velocity of hydrogen.

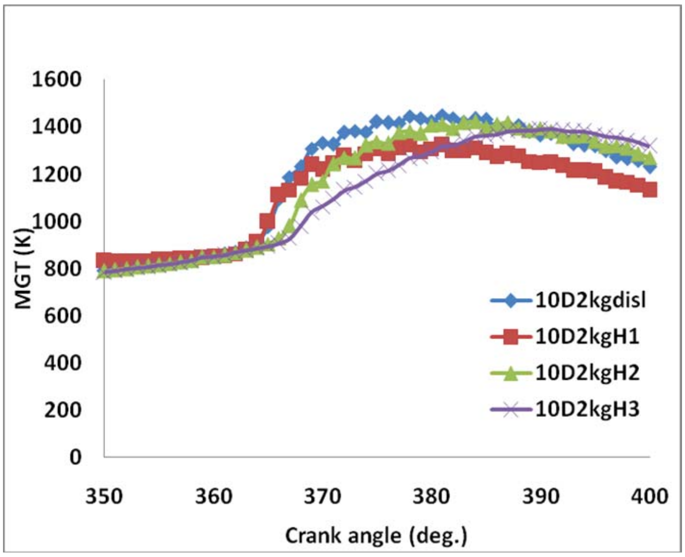

Figure 13. MGT v/s crank angle @2kg

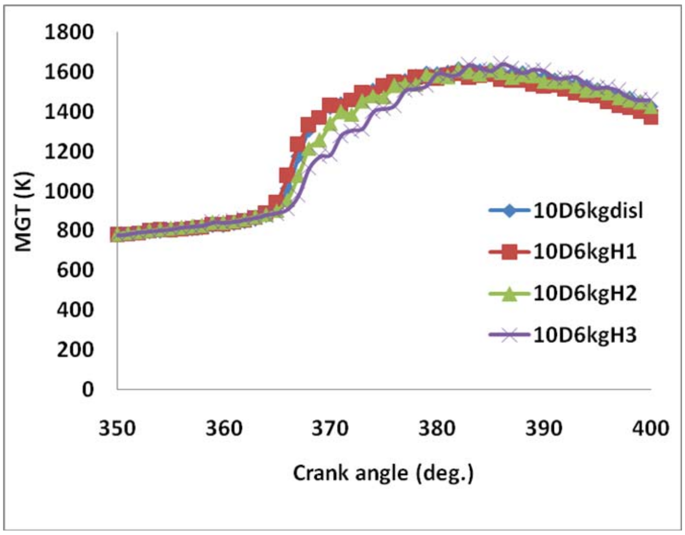

Figure 14. MGT v/s crank angle @6kg

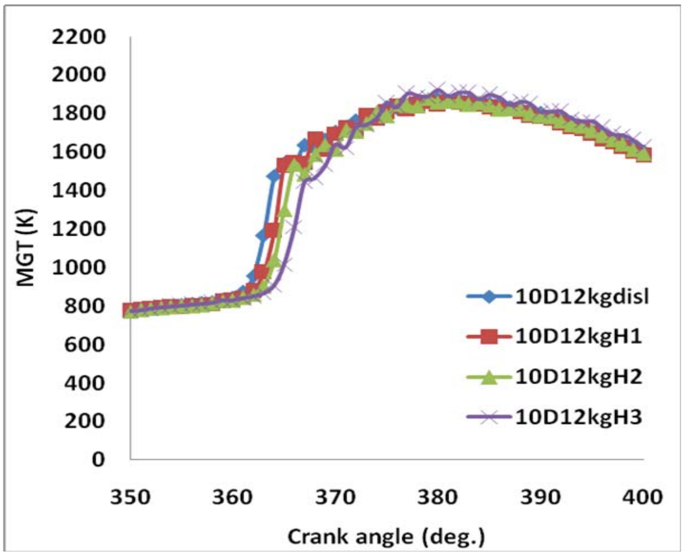

Figure 15. MGT v/s crank angle @12kg

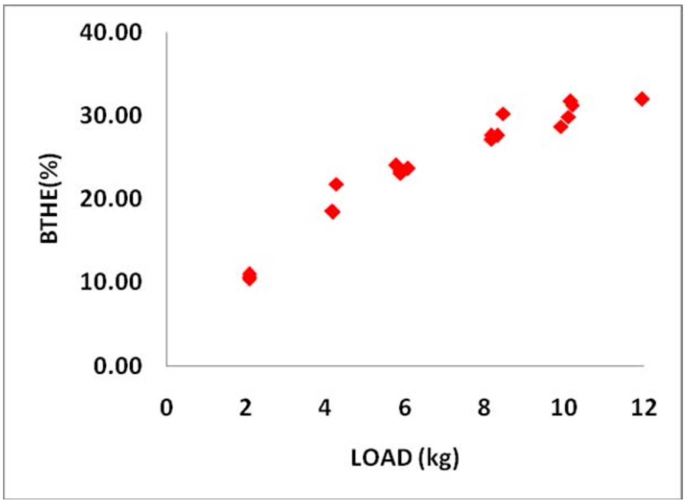

Figure 16. BTHE v/s load

3.2.2 Brake specific fuel consumption (BSFC)

The variation of brake specific fuel consumption with load for net diesel and with hydrogen enrichment is shown in Figure 17 The value of BSFC for base diesel is 0.28kg/kwh at full load conditions but with the introduction of hydrogen strategy it reduces and reaches to a minimum value of 0.18kg/kwh during third injection strategy of hydrogen i.e. at 21exp(+3) microseconds. BSFC decreases with the increase in brake power. BSFC in case of hydrogen enrichment was less compared to base diesel operation. This trend was maintained in all load conditions. BSFC reduces with respect to base diesel at all load operations which is a clear indicative of better mixing of hydrogen with air resulting in complete combustion of fuel.

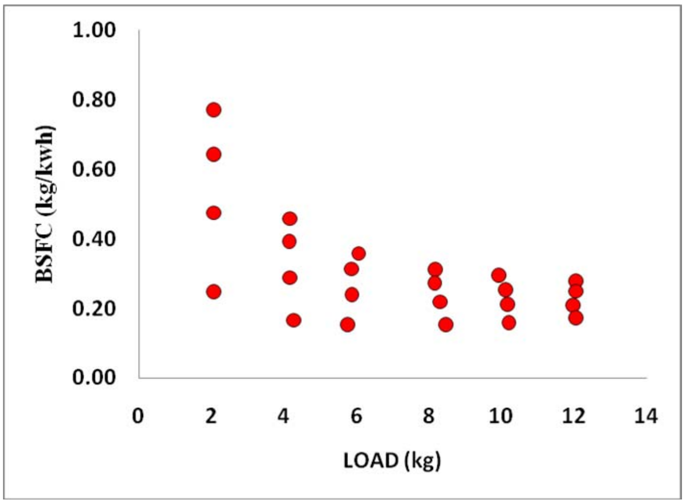

Figure 17. BSFC v/s load

3.2.3 Brake specific energy consumption (BSEC)

Figure 18 shows the variation of brake specific energy consumption with load for net diesel and with hydrogen enrichment. It was observed that BSEC with net diesel is higher at all load operating conditions compared to all hydrogen introduction strategy. The lower specific energy consumption for hydrogen-diesel dual fuel is due to better mixing of hydrogen with air resulting in complete combustion of fuel. The diesel shows the maximum energy consumption at all the loads.

3.2.4 Volumetric efficiency(Vol. Eff.)

From Figure 19 it can be observed the variation of volumetric efficiency with load conditions. The volumetric efficiency of the engine is found to be less when hydrogen is inducted with the main fuels.

The reason for low volumetric efficiency is because of the high velocity and lower density of hydrogen (0.08kg/m^3) tends to displace the air. The volumetric efficiency is calculated as the ratio of actual volume of air passed into the engine to the swept volume. The volumetric efficiency of diesel at full load condition is 81% while it reduces to 73% at third injection strategy of hydrogen since more hydrogen is taking part in the combustion which displaces more amount of air and thus reduces the breathing capacity of the engine. This trend of character can be seen in all conditions also.

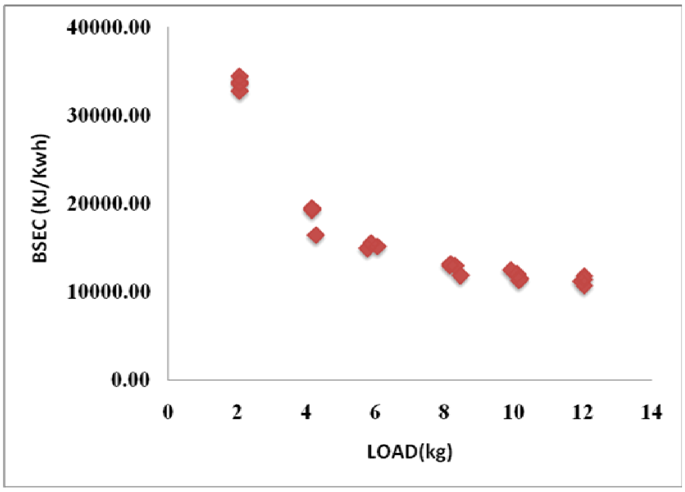

Figure 18. BSEC v/s load

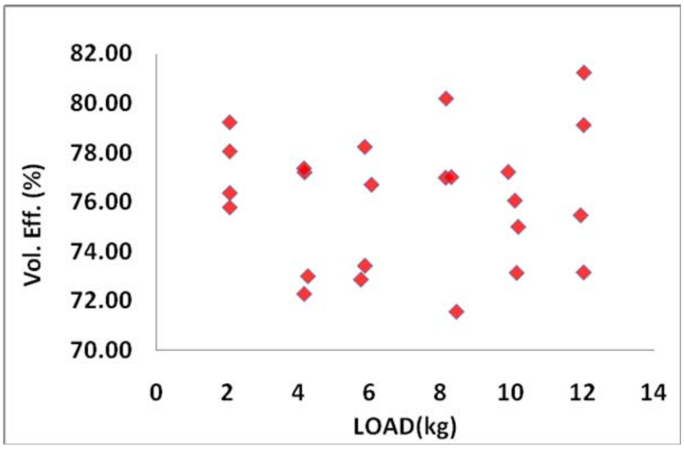

Figure 19. Vol. Eff. v/s load

4. Conclusion

Experiments were conducted on a hydrogen-enriched-air-inducted diesel engine system and based on the experiments the following conclusions can be drawn:

1. The combustion process is improved with the introduction of hydrogen in dual fuel mode.
2. The maximum pressure for the 10 degree ATDC injection was observed to be 62.62 bar at maximum load for 9 exp(+3) micro second duration of injection strategy.
3. The maximum rate of pressure rise for the 10 degree ATDC injection was observed to be 7.55 bar per crank angle at 365 crank angle for maximum load in the minimum hydrogen induction window.
4. The maximum mean gas temp for the 10 degree ATDC injection was observed to be 1924K for maximum load in the 21exp(+3)micro seconds hydrogen induction window.
5. Brake thermal efficiency increased for most of the hydrogen operation with a maximum gain of 18% at 10deg ATDC hydrogen injection for maximum injection duration.
6. Specific fuel consumption of diesel was reduced over the entire spectrum of loading with hydrogen operation where10 deg ATDC injection with maximum injection duration was able to reduce specific fuel consumption of diesel by a maximum of 87% at low loads.

7. BSEC was reduced to a maximum of 15.25% as compared to baseline diesel at 10deg ATDC injection during maximum injection proving the exceptional combustion properties of hydrogen .

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