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Effect of Combustion Chamber Geometry on Performance and Combustion Characteristics of Hydrogen Enriched Diesel Engine

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Abstract

The in-cylinder air motion in diesel engines generally characterised by swirl, squish and turbulence which have major impact on air-fuel mixing. To achieve controlled combustion, an attempt is made at investigating the effect of change of piston geometry on the combustion & performance characteristics of diesel engine enriched with hydrogen at optimum flow rate on four stroke single cylinder diesel engine. The combustion parameters for diesel engine enriched with hydrogen in hemispherical combustion chamber at 6lpm flow rate are increased by 7.72 %, 13.48% and 17.26% of Cylinder pressure, Heat release rate & exhaust gas temperature respectively compared to alone diesel operation at rated load and constant speed. The performance characteristics such as brake thermal efficiency is increased by 6.35% and specific fuel consumption is reduced by 33.5% as against neat diesel operation at constant flow rate of hydrogen. This is mainly due to high combustion temperatures which leads to complete burning of fuel and reduction in carbon content with addition of hydrogen. Again it is observed that with re-entrant geometry of the combustion chamber at 6lpm flow rate of hydrogen the combustion and performance parameters are further improved significantly compared to hemispherical and toroidal combustion chamber geometries

Keywords – knock, combustion chamber geometry, squish, swirl, turbulence

INTRODUCTION

Compression ignition engines play a dominant role in surface transportation, agricultural machinery and industries all over the world because of their durability and superior thermal efficiencies. In view of reduction in the availability of diesel against the increased consumption and increased stringent environmental regulations on exhaust emissions demands to develop alternative fuel [1]. Among the different alternative fuels, hydrogen is considered as suitable fuel due to its clean burning and better combustion properties. Currently, there is limitation to utilize hydrogen for combustion in case of the commercial devices due to the problems associated with its handling and limited availability of hydrogen. As per environmental legislation, which will favour clean burning technologies, the emergence of hydrogen as an energy carrier will modify current situation. Because of less pollution, non-toxic, odourless and wide range flammability, hydrogen is considered as alternative fuel for internal combustion engines by many researchers. Reduction in carbon content in dual mode operation, is the desirable feature of hydrogen fuel. Higher thermal efficiency can be achieved from combustion of hydrogen due to its higher flame speed. In transportation especially for petrol engine the usage of hydrogen was introduced since year 2000 [2]. Due to high auto ignition temperature (about 585^{0}C), to attain the same, the ignition triggering devices are used in the combustion chamber for CI engines [3]. The properties of hydrogen compared to diesel is shown in table 1.

Table 1. Properties of Hydrogen compared to Diesel

Property	***Hydrogen***	***Diesel***
Auto-ignition temperature (K)	858	543
Molecular weight (g)	2.016	170
Density of gas at NTP (g/cm^3)	0.0838	0.86
Flame velocity (cm/sec)	270	30
Specific gravity	0.091	0.83
Boiling point (K)	20.27	580-640
Heat of Combustion (kJ/kg)	120	42.4
Octane number	130	-
Cetane number	-	40-60
Stoichiometric air fuel ratio	14.92	34.3

The engines of compression ignition types perform a vital role in surface transportation, industries and agricultural machinery across the world due of their durability and higher thermal efficiencies. Compared to past, more favourable responses received from the researchers and consumers towards the gaseous fuels due to current progressive developments. Therefore, to have environmental advantage it is more economical to utilize gaseous fuel in CI engines with a concept of dual fuel mode [4]. Even though hydrogen is not a primary fuel and it must be manufactured from water with either fossil or non-fossil energy sources, it is desirable fuel due to its properties favours as a fuel for internal combustion. Hydrogen combustion entails NOx emissions apart from advantage of not emitting the particulates of HC,CO_2, CO, SO_2 etc. [5].

The burning of hydrogen with high flame speed maintain the engine to approach the thermodynamically ideal engine cycle when the stoichiometric fuel mix is used [6]. With Hydrogen as an air enrichment medium, diesel as an ignition source in a stationary compression ignition engine, the system will improve the performance characteristics and reduce emissions [7]. The abnormal combustion such as pre-ignition, knock and backfire occurs if the hydrogen fraction increases above certain extent for different engine specifications. This is due to low quenching distance and high burning velocity, the combustion chamber walls becomes hotter which causes more losses to the cooling water. Hence the amount of hydrogen being added should be optimised [8]. In case of diesel engines enriched with hydrogen, it was reported from the experimentation results that, for a higher flow rate of hydrogen admission the combustion becomes uncontrolled due to high cylinder temperatures causes knocking tendency [9]. Swirl motion of air phenomena is commonly influenced by design of intake valve, combustion bowl geometry to improve the combustion [10]. The desired swirl-squish interaction leads to a complex turbulent flow field which is possible by more intense in re-entrant combustion chamber geometries at the end of the process of compression [11]. In re-entrant chambers when compared to other type chambers, the intensification of swirl and turbulence are higher and leads to more efficient combustion [12]. From the study conducted to understand the effect of deposit in combustion chamber on engine using a two-zone model to analyse the hydrocarbon emissions contribution and concluded that the combustion chamber deposits will contribute 20-30 percent of the engine hydrocarbon emissions by the investigation [13].

Many experimental investigations have sought to understand the physical phenomena associated with evolution of the air flow inside the different combustion chamber of IC engines. Since the flow in the combustion chamber develops from interaction of the intake flow with the in-cylinder geometry, the goal of this work is to characterize the role of combustion chamber geometry on in-cylinder flow, thus the fuel-air mixing influence on combustion and performance characteristics. The combustion process depends highly on efficient fuel-air mixture. This paper aims to study the effect of piston bowl geometry on performance and combustion characteristics on hydrogen enriched diesel engine at different flow rates of hydrogen.

EXPERIMENTAL SET UP:

The experiments are conducted on single cylinder, four stroke, water cooled diesel engine. The specifications of the engine are shown in Table 2 and the schematic diagram for experimental set up is shown in Fig 1.

Table 2. Specifications of the Engine

Parameter	***Specification***
Engine Power	3.7 kW
Engine speed	1500 rpm
Cylinder bore	80mm
Stroke length	110mm
Compression ratio	16.5:1
Swept volume	550 cc
Fuel injection timing	21^0 BTDC

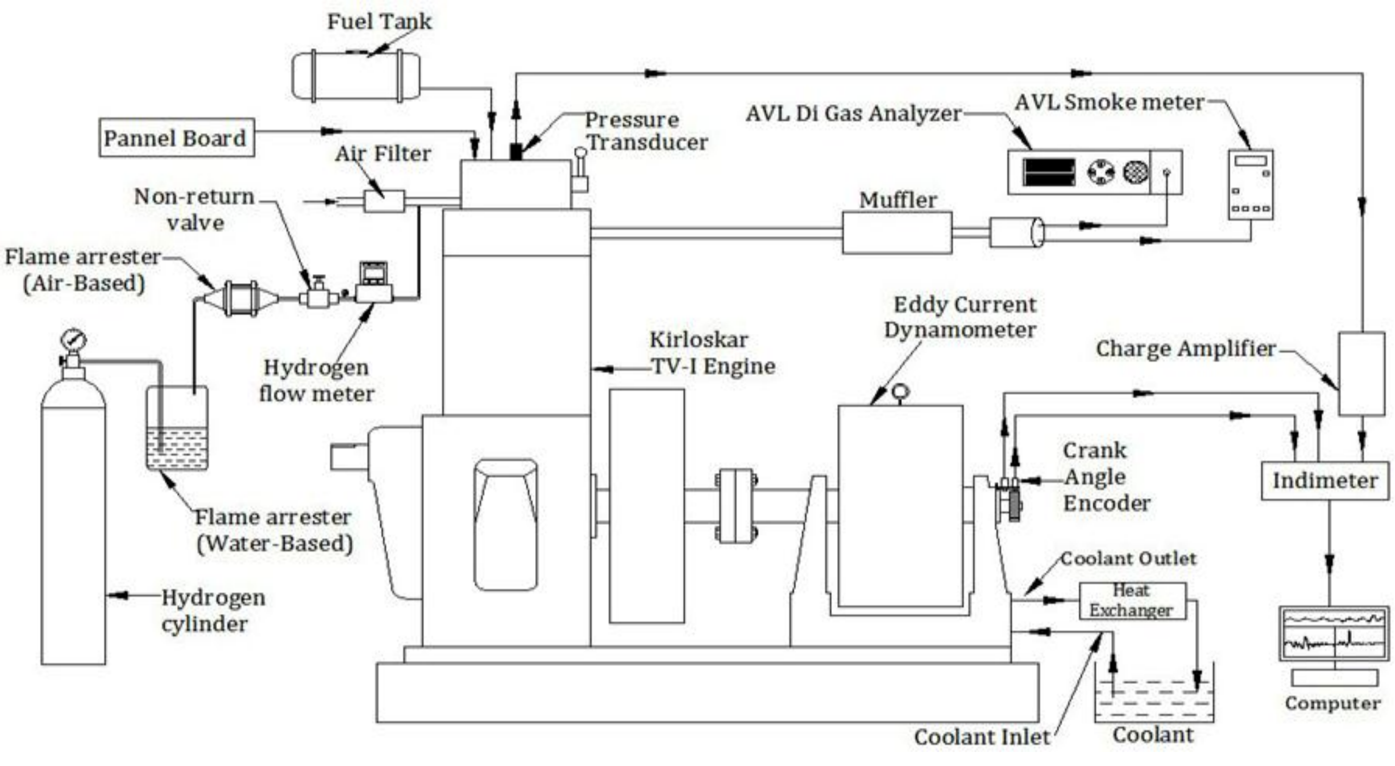

Figure 1. Schematic diagram of experimental set up

Safety measures for handling of Hydrogen

The safety measures required for hydrogen operation shown in Fig 2, as it associates with Hindenburg or Challenger disasters in its operation. To suppress the explosion inside the hydrogen cylinder a flame arrester is used. The flame arrester consists of partly filled water tank with fine mesh to prevent the flame propagation beyond the wire mesh. In case of backfire the flame gets quenched as it reaches the water surface. Also to prevent the reverse flow of hydrogen into the system a non-return valve is provided. To visualise the flow of hydrogen during the engine operations, a flow indicator is used. To measure the combustion parameters AVL combustion analyser is used.

Figure 2. Image of actual setup of inducting hydrogen indicating safety measures

Design of Combustion chamber

In the present work, the existing hemispherical (STD) piston bowl is replaced with toroidal (MP1) and re-entrant toroidal (MP2) combustion chamber piston geometries. The idea behind this attempt is to achieve a useful squish with proper air movement within the toroidal chamber. Due to this there is effective utilization of oxygen and small mask needed on inlet valve for producing powerful squish for cone angle of spray of 150^0 to 160^0. In case of re-entrant combustion chamber, the bowl lip which prevents the air squish motion pushing fuel above the piston crown, so that the majority of the fuel charge is mixed and burnt within the bowl and creates further micro turbulence within the bowl. In order to maintain same compression ratio of the engine, the combustion bowl volume is maintained same by modelling the combustion bowl geometry using CATIA V5 tool as shown in Fig 3, 4 & 5 and the fabrication of pistons were made accordingly as shown in Fig 6. The hemispherical bowl piston is replaced with toroidal and re-entrant toroidal pistons and conducted experimentation for analysis.

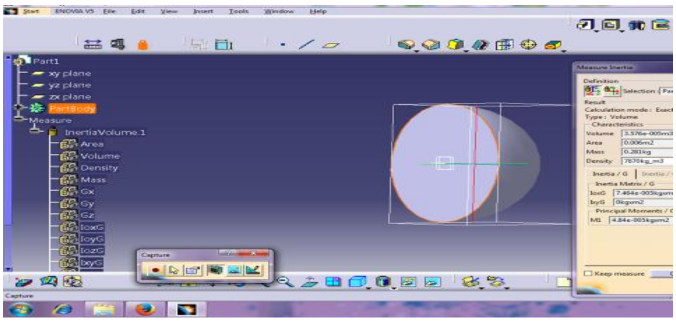

Figure 3. Image of the hemispherical piston bowl modelling

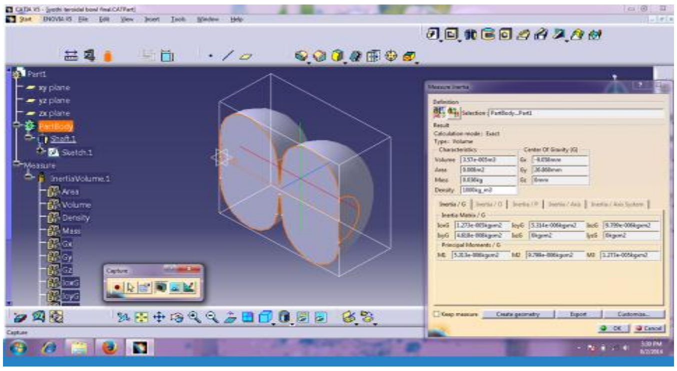

Figure 4. Image of the toroidal piston bowl modelling

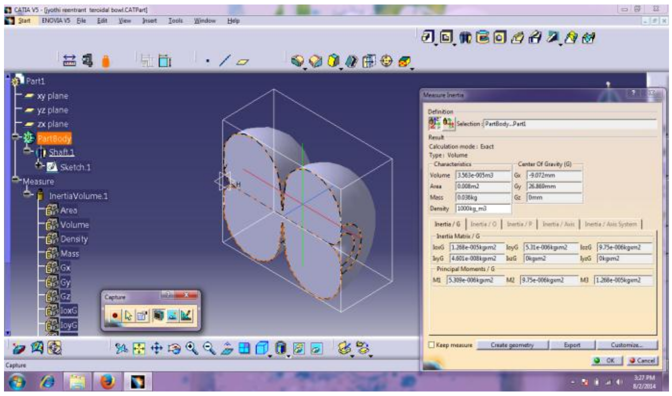

Figure 5. Image of the toroidal piston bowl modelling

Figure 6. Images of hemi-spherical, toroidal and re-entrant toroidal pistons

Experimentation Procedure

Initially the diesel engine with standard piston of hemispherical piston bowl geometry is operated at no load with rated speed for a duration to reach steady-state condition. The loads are induced to the engine running with diesel for 20%, 40%, 60%, 80% and 100% loads in steps by means of Eddy current dynamometer and performance & combustion parameters are recorded. Along with air hydrogen is inducted at constant flow rate of 6lpm (litres per minute) the experiments are conducted at different loads. For flow rates above 6lpm of hydrogen it is observed tendency of knocking due to high combustion temperatures with high flame & burning velocities. This is because more heat is being lost through cooling water from hotter walls of combustion chamber and hence results reduction in thermal efficiency. Again, the standard piston is replaced with modified piston with toroidal bowl geometry and then with re-entrant combustion chamber of same compression ratio. The performance & combustion parameters are measured by conducting experiments initially for diesel fuel alone and then by inducting hydrogen at different flow rates. Finally compared the performance & combustion parameters with all cases.

RESULTS AND DISCUSSIONS

In this investigation, the performance parameters such as brake thermal efficiency, brake specific fuel consumption and combustion parameters like cylinder pressure, heat release rate and exhaust gas temperature are determined at 6lpm flow rate of hydrogen for different piston bowl geometry with different loads.

Brake thermal efficiency (BTE):

The variations of brake thermal efficiency with brake power for different induced loads and different geometries of piston bowl is shown in Fig 7 for alone diesel operation. It is observed that, there is increase in efficiency for re-entrant toroidal piston geometry (MP2) compared toroidal (MP1) and hemi spherical geometry (STD). This is due to improved combustion with better mixture formation due to improved swirl motion of air. The BTE for engine with re-entrant toroidal geometry is increased to 26.36% as that of toroidal geometry of BTE 26.24% and normal engine of BTE 25.93% with alone diesel operation. It is observed that there is increase of 1.65 % with MP2 at rated load compared to standard piston. This is due to improved combustion with better mixture formation with improved swirl motion of air.

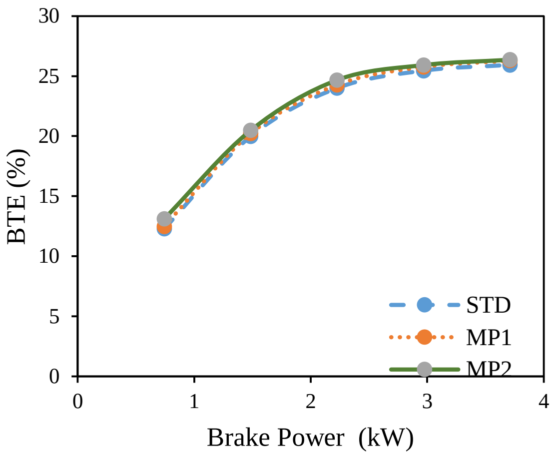

Figure 7. Variation of Brake thermal efficiency with Brake power for diesel operation

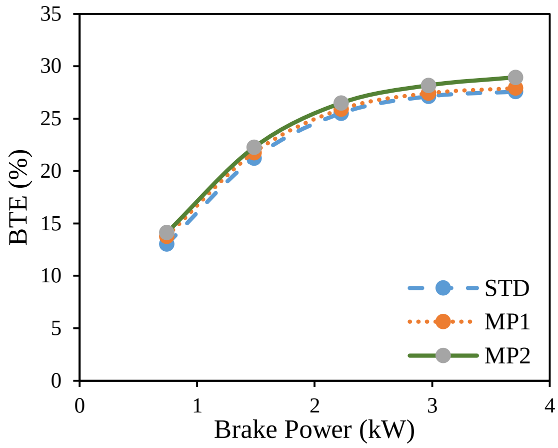

Figure 8. Variation of Brake thermal efficiency with Brake power for diesel enriched with hydrogen operation

For diesel enriched with hydrogen operation, the variation of BTE with Brake power for different combustion chamber geometries at different loads is shown in Fig 8. The hydrogen is inducted at 6lpm flow rate. It is observed that the BTE is improved to 27.57% when compared to alone diesel operation with standard piston. There is an increase of 6.35% improvement when compared to alone diesel. It is noticed BTE is improved to 27.92% with MP1 and 28.95% with MP2 with hydrogen enrichment with 6lpm flow rate. There is an increase of 5% BTE is observed with MP2 compared to standard piston with hydrogen induction at rated load. This is because of higher heating value and high flame velocity with hydrogen addition. This improves the rate formation of intermediate compounds and initiates the combustion little later than neat diesel. In turn this delay accumulates oil before combustion results better burning of fuel.

Brake specific fuel consumption (BSFC):

For alone diesel operations, the variations of BSFC with brake power for different induced loads and different geometries of piston bowl is shown in Fig 9. It is observed that, there is notable decrease in BSFC for MP2 piston when compared MP1 and standard piston. The BSFC for engine with MP2 is increased by 3.4% and 1.9% compared to MP1 and standard piston respectively at rated load with alone diesel operation. This is because the inducement of enhanced air swirl in the combustion chamber leads to the complete combustion of charge in the combustion chamber with liberation of maximum energy.

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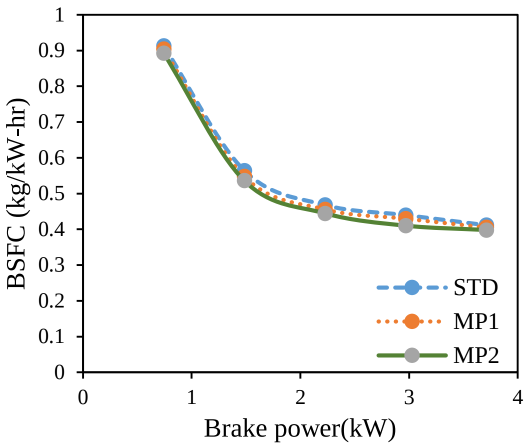

Figure 9. Variation of Brake specific fuel consumption with Brake power for diesel operation

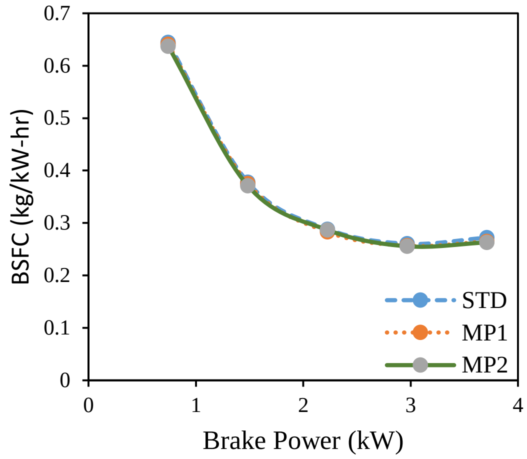

Figure 10. Variation of Brake specific fuel consumption with Brake Power for diesel enriched with hydrogen operation

The variation of BSFC with Brake power for different combustion chamber geometries at different loads with diesel enriched with hydrogen at 6lpm flow rate operation is shown in Fig 10. It is observed that there is a reduction in BSFC of 33.54% when compared to alone diesel operation with standard piston. It is noticed that there is further reduction in BSFC of 2.6% with MP1 and 2.8 % with MP2 with hydrogen enrichment with 6lpm flow rate compared to standard piston. This is because better mixing of hydrogen with air which leads to increase of temperatures in the combustion chamber resulted complete burning of fuel.

Cylinder Pressure:

The variation of cylinder pressure with crank angle is shown in Fig 11 for diesel operation for different geometries of combustion chambers. The maximum cylinder pressure is noticed for MP2 piston of 56.35 bar at 1^0 after TDC when compared to 52.09 bar and 47.2 bar at 1^0 after TDC for MP1 and Standard piston respectively. There is an increase of 19.3% cylinder pressure with MP2 compared to standard piston. This is due to better air motion and hence release of more breakdown products at rapid rate during the combustion process.

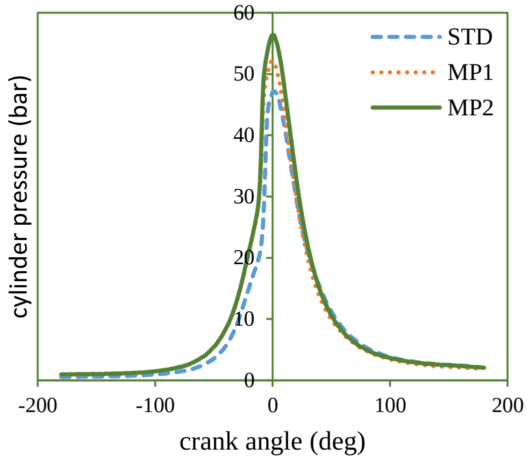

Figure 11. Variation of Cylinder pressure with crank angle for diesel operation

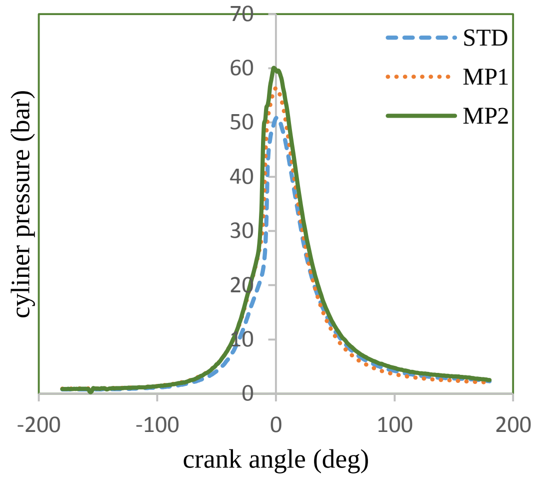

Figure 12. Variation of Cylinder pressure with crank angle for diesel enriched with hydrogen operation with Brake power for diesel operation

It is noticed that there is an increase of cylinder pressure with hydrogen enriched diesel operation for varied piston bowl geometries from Fig 12. The cylinder pressure is increased to 50.86 bar at 1^0 after TDC with hydrogen enrichment. There is an increase of 7.7% in cylinder pressure compared to alone diesel operation. This is because of highest combustion temperature of the hydrogen apart from high calorific value & high burning speed. Further there is an increase of 10.7% and 18.1% in cylinder pressure is noticed with MP1 and MP2 pistons respectively with hydrogen enrichment.

Heat release rate (HRR):

It is observed in Fig 13, the heat release rate is higher for MP2 piston compared to MP1 and standard piston for neat diesel operation. The maximum HRR of 113kJ/m^{3}deg at 11^0 before TDC is noticed with MP2 piston at rated load as against HRR of 106.6 kJ/m^{3}deg at 11^0 before TDC and 11 kJ/m^{3}deg at 11^0 before TDC for standard and MP1 piston respectively. This is because of better mixing of fuel & air mixture due high turbulence results efficient burning of the fuel.

The HRR is increased further with hydrogen enrichment for all three geometries of piston bowl compared to neat diesel operation as shown in Fig 14. There is an increase of 13.48% on HRR with standard piston. Again with hydrogen enrichment there is an increase of 6% with MP1 and 7.5% with MP2 on HRR compared to standard piston at rated load. This is due to changes in fuel combustion phenomena with sufficient ignition delay and small quenching distance with addition of hydrogen.

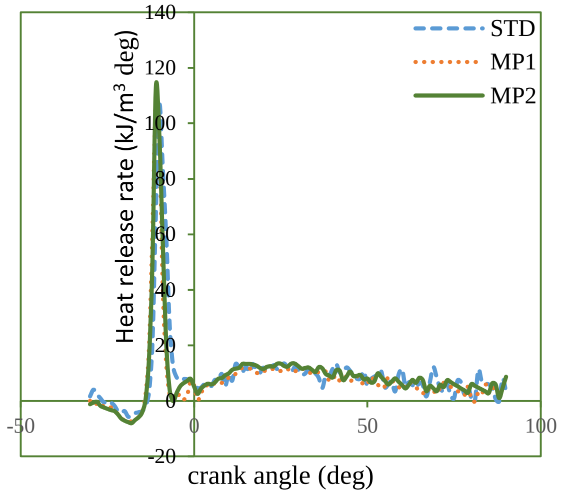

Figure 13. Variation of Heat release rate with crank angle for alone diesel operation with Brake power for diesel operation

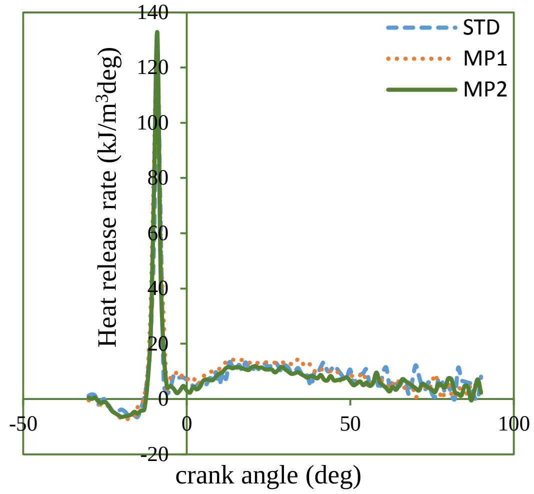

Figure 14. Variation of Heat release rate with crank angle for diesel enriched with hydrogen operation with Brake power for diesel operation

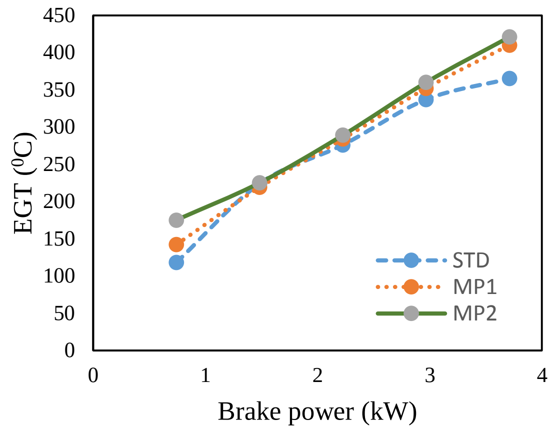

Figure 15. Variation of EGT with Brake power in alone diesel operation

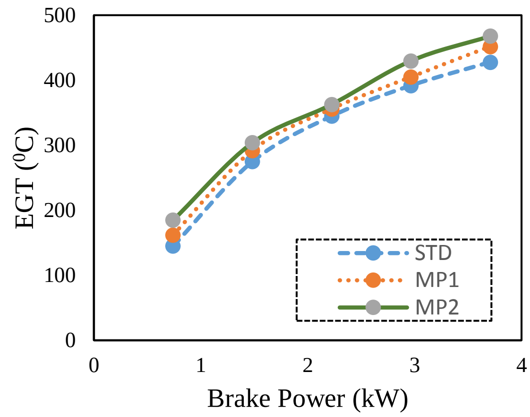

Figure 16. Variation of EGT with Brake power in diesel enriched with hydrogen operation

Exhaust gas temperature (EGT):

The variations of exhaust gas temperature with brake power for different loads in alone diesel operation are shown in Fig 15. It is observed that at rated load the EGT for standard piston is 365^0 C, for MP1 piston is 410^{0}C and for MP2 piston EGT is 421^{0}C with alone diesel. This is due to the inducement of enhanced air swirl in the combustion chamber enhances the temperature of the combustion.

With hydrogen enrichment at 6lpm flow rate, the variations of EGT with Brake power for different geometries of piston bowl are shown in Fig 16 for different loads. There is an increase of 17.2% in EGT with hydrogen enrichment for standard piston compared neat diesel operation. It is noticed there is further increase with MP1 of 5.6% and with MP2 of 9.3% when compared to standard piston with hydrogen enrichment. This is due to shorter duration in burning heavy fuel molecules than neat diesel which in turn increases the combustion temperature.

CONCLUSIONS:

This paper reports the role of hydrogen enrichment at 6lpm flow rate in existing diesel engine by changing the piston bowl geometry in analysing the performance and combustion characteristics with the objective of improving the engine performance & combustion characteristics at different loads. The main conclusions of the present study are summarized as follows:

- The Brake thermal Efficiency is increased about 6.35% for hydrogen enrichment with 6lpm compared to base diesel at rated load operation. This is due to high heating value and high flame velocity of hydrogen compared to neat diesel. Further there is an increase in BTE of 1.25% with MP1 piston and 5% with MP2 piston with hydrogen enrichment due to improved motion of air swirl.
- It is observed that the Brake specific fuel consumption is reduced with hydrogen enrichment when compared to diesel operation due to increase of temperature in combustion chamber with better mixing of hydrogen & air, which results in reduction is about 33.5% . Because of the inducement of enhanced air swirl in the combustion chamber BSFC is further reduced by 2.6% with MP1 and 2.8 % with MP2 with hydrogen enrichment with 6lpm flow rate compared to standard piston.
- The combustion parameter of cylinder pressure is increased by 7.7% at rated load and 6lpm flow rate of hydrogen due to sufficient ignition delay, high burning speed and small quenching distance compared to base line diesel operation. Due to better air motion there is further increase of 10.7% and 18.1% in cylinder pressure is noticed with MP1 and MP2 pistons respectively with hydrogen enrichment
- The heat release rate is increased by 13.5% at rated load and 6lpm flow rate of hydrogen due to sufficient ignition delay, high burning speed and small quenching distance compared to base line diesel operation. With varied piston bowl geometry HRR is further increased by 6% with MP1 and 7.5% with MP2 pistons compared to standard piston at rated load due to high turbulence results efficient burning of the fuel
- There is a growth of 17.26% in exhaust gas temperature is noticed with addition of hydrogen at 6lpm compared to alone diesel operation due to shorter duration in burning heavy fuel molecules with liberation of high energy. Further an increase of 5.6% with MP1 and 9.3% with MP2 pistons when compared to standard piston with hydrogen enrichment is due to enhanced temperature of the combustion with inducement of air swirl
- Further, at higher flow rates above 6lpm of hydrogen admission the combustion becomes uncontrolled due to high cylinder temperatures leads to tendency of knock

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