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Effect of injection timing on modified direct injection diesel engine performance operated with dairy scum biodiesel and Bio-CNG

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ABSTRACT

The optimization of engine parameters, best nozzle hole and piston bowl geometry is highly marked for bio-fuel operation as they have slightly higher viscosity and lesser heating value than the petroleum diesel. In the first phase of work, at single fuel operation, study optimized the best fuel blend as B20 (among B10, B20, B30 and B100), injector opening pressure as 230 bar (among 210, 220, 230 and 240 bar), injection timing as 26. deg.bTDC (among 20, 23, 26 and 29. deg.bTDC), nozzle as 5 holes (among 3, 4 and 5 holes) and piston bowl geometry as re-entrant toroidal piston bowl geometry (among Hemispherical piston bowl geometry (HPBG), Straight sided piston bowl geometry (SSPBG), Toroidal piston bowl geometry (TPBG) and Re-entrant toroidal piston bowl geometry (RTPBG)). Hence, baseline engine is modified with all these optimized parameters and then modified engine is carried further for dual fuel experiments. In second phase of the work, study unfolds the effect of injection timing (IT) on dual fuelled (B20+Bio-CNG (enriched methane)) modified engine. From the dual fuelled engine study, it is revealed that 29. deg.bTDC IT has shown the improved performance, combustion and emission characteristics when compared to 20, 23, 26 and 32. deg.bTDC injection timings.

1. Introduction

1.1. Single fuel operation in direct injection diesel engine

As transportation and industrial sectors are growing day by day, looking into alternate fuels for petroleum products which are locally available and emits lesser emissions than diesel are highly pronounced to curb the foreign exchange [1]. In this vision, the transesterified biodiesels from non edible oils would substitutes the conventional diesel fuel with generating minimum exhaust emissions [2]. The biodiesels are sustainable, renewable and reveals comparable properties as of petro-diesel [3]. In other hand, biodiesels exhibits higher NOx emissions at higher biodiesel share as they are rich in oxygen content in comparison with petroleum diesel [4]. The employment of biodiesels may not affect the weight loss and surface of the fuel injector equipment (FIE), hence cause lesser wear when compared with mineral diesel [5].

The minor modifications in diesel engine parameters such as injector opening pressure (IOP), injection timing (IT), compression ratio (CR) and nozzle geometry would give the improved performance for the diesel engine especially operated with biodiesels. The higher viscous and denser biodiesels requires higher IOP to ensure the better fuel atomization followed by improved fuel-air mixing with smaller fuel droplets to achieve the improved combustion and emission behaviors of biodiesel run diesel engine [6–10]. In addition, for biodiesel operation, advanced IT may also enhance the performance of the engine and lowers the HC, CO emissions with compromised NOx emission in comparison with petroleum diesel [11–13]. The diesel engine operated with biodiesels at higher compression ratio improves the performance, emission and combustion behaviors when compared to the lower compression ratio (CR). It is because of higher CR contributes the more cylinder temperature and pressure which leads to complete combustion [14–17]. From the literature, it is noticed that the design of nozzles hole is really a critical parameter to get improvement in spray characteristics, succeeding emissions and performance in combustion chamber when both experiment and simulation results are analyzed [18]. The 20% biodiesel with conventional diesel would decreases the HC, CO emissions except NOx

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emission even it is a detrimental gas. Modifications in nozzle geometry (modifying injector nozzle hole from 5 (base) to 6 holes) would give a feasible solution to reduce NOx emission of biodiesel operated diesel engines [19]. Cavitation and turbulence presence in nozzle has a noteworthy influence on the succeeding spray behaviors [20]. At 250 bar injection pressure, brake thermal efficiency, brake specific fuel consumption are improved and emissions are decreased with modified nozzle (5 holes) when compared to baseline nozzle (3 holes) of the standard engine [21]. In 8 holes nozzle, output power, brake specific fuel consumption and engine torque are increased as a result of higher diesel injection velocity and improved fuel-air mixing with reduced NOx and CO emissions when compared to 6, 7 and 9 holes [22]. The injector nozzle with smaller orifice diameter would cause the considerable engine performance, combustion and emission behaviors than the larger orifice diameter [23].

Modification in the piston bowl geometry along with modifications in injection strategies is sturdily influences the performance and emission characteristics of the diesel engine [24]. From the numerical (CFD) simulations, it is observed that the symmetric toroidal vortices of re-entrant piston bowl geometry enhance the squish-swirl and turbulence kinetic energy (TKE) during compression stroke when compared to open-piston bowl shape which have bigger bowl diameter. Therefore lesser emissions can be achieved with toroidal piston. Hence swirl, squish, turbulence are muscularly influenced by the piston bowl geometry in diesel engines [25–27]. The combustion chamber with bottom corner will results in improved thermal efficiency and reduced NOx, CO and soot emissions when compared to lip shape and wall distanced combustion chambers [28]. The diesel engine operation with 20% jatropha methyl ester has shown the enhanced performance and reduced HC, CO and smoke with re-entrant toroidal combustion chamber in comparison with hemispherical and toroidal combustion chambers [29]. The toroidal re-entrant combustion chamber (TRCC) geometry has shown the improved brake thermal efficiency, peak pressure and heat release rate and reduced brake fuel consumption, ignition delay period, CO, HC and smoke as it contributes more air motion, squish and fuel-air mixing rate when compared to hemispherical combustion chamber [30].

1.2. Dual fuel operation in direct injection diesel engine

Currently in India, the complete substitution of conventional diesel for transport and agriculture sector is the major task to reduce both cost and emissions of the petroleum diesel. In this regard, biomass derived alternative fuels would give the feasible solutions for these challenges. However, dual fueled diesel engine could reduce the nitrogen oxide and smoke when compared to the conventional diesel mode. Whereas, dual fuel operation in diesel engine exhibits the inferior brake thermal efficiency and superior brake-specific energy consumption as well as exhaust gas temperature when compared to sole diesel fuel operation [31,32]. The biogas flow rates affect the brake thermal efficiency of the engine when it is operated with dual fuel. The brake thermal efficiency of the diesel engine would increase and specific energy consumption may decreases with increase in the methane concentration in the biogas. However, specific fuel consumption of the CI engine is higher for dual operation when compared to sole petro-diesel. The HC and CO emissions of the engine are higher with increased biogas flow rate [33]. The engine would give 50% lower thermal efficiency for dual fuel (Biogas and Diesel) operation when compared to single fuel (diesel) operation. It could be attributed to higher CO_2 concentration in the biogas cause in incomplete combustion. However, biogas is a promising substitute fuel and it can be produced easily at any place [34].

The application of dual fuel (biodiesel and biogas) in diesel engine would exhibits lower NOx emissions and particulate matter with significantly reduced thermal efficiency when compared to diesel fuel operation. The optimum injection timing and optimum compression ratio would play a prominent role on diesel engine performance operated with biogas-diesel dual fuel [35,36]. The improved performance, combustion and emission characteristics were observed with the diesel engine operated with rice bran oil biodiesel and biogas at higher compression ratio [37]. The application of biogas in diesel engine is technically viable for power generation at rural areas [38]. Higher CO_2 in biogas could reduce the brake thermal efficiency, NOx emissions, heat release rate and cumulative heat release, whereas, HC and CO emissions are increased. However, these HC and CO emissions can be controlled using different suitable methods. Hence, raw biogas can be easily used in the diesel engine at dual fuel operation [39]. The engine performance can also be increased with optimized injection parameters at different CNG flow rates [40]. The biogas production, up gradation and utilization of compressed biogas (CBG) or Bio-CNG would reduce the green house gases and global warming than the fossil fuels [41]. The biogas has lower calorific value than CNG, however; NOx emission is lesser for biogas operation than the CNG. Hence, enriched biogas is an eco-friendly and renewable fuel that can play a promising alternative fuel for the vehicles in coming days [42]. The liquefied biomethane (LBM) or liquefied biogas (LBG) which is obtained from upgradation and liquefaction of biogas would be the promising substitute fuel for the transportation. LBM has three times more energy dense than the compressed biomethane (CBM) [43]. The engine exhaust emissions namely HC, CO and NOx are slightly higher with enriched biogas (Bio-CNG) than base compressed natural gas (CNG) with no significant change in fuel consumption for both enriched biogas and CNG. However, the emission of enriched biogas in engines meets the BS IV emission norms. The properties of renewable enriched biogas are similar to fossil CNG hence it exhibits the better performance as of CNG. Therefore, enriched biogas could be used as an auto fuel for spark ignition vehicles [44,45].

The exhaustive literature directed that, many researchers used the simulation and numerical techniques to reveal the effect of combustion chamber geometry to assess the diesel engine performance. However, there are limited works explored the influence of the piston bowl geometry along with optimized engine parameters and nozzle hole geometry of the diesel engine when it is operated with dairy scum biodiesel and Bio-CNG (enriched methane). In this view, present study formed the constructive hopes to realize the full potential usage of dairy scum biodiesel and Bio-CNG in the conventional diesel engine (Kirlokar 3.5 kW, direct injection (DI), water cooled) by modifying the engine parameters, nozzle hole geometry and piston bowl geometry. In addition to this, substandard dual fueled engine performance could be improved by providing powerful squish, improved turbulence, rapid and proper fuel-air-gas mixing in the combustion cavity with robust design in piston bowl geometry, modifications in nozzle hole geometry and engine parameters (IOP, IT & CR). In this vision, present investigation is carried to comprehend the effect of injection timing on the modified dual fuel (B20+Bio-CNG) engine with optimized engine parameters (injector opening pressure (IOP), injection timing (IT), compression ratio (CR), nozzle hole (NH) and piston bowl geometry (PBG)).

2. Materials and methods

2.1. Biodiesel preparation and its properties

The novel approach of developing and exploitation of dairy scum oil methyl ester (DSOME) would play a crucial role in shrinking of conservative energy consumption and their environmental contamination. Mainly, treating of dairy waste scum into fuel would be the value added benefit for the milk dairies in terms of economics, solution to disposal problem, alleviation of global warming and independency on fossil fuels. Hence, production of biodiesel from waste scum would give the feasible solutions for the above challenges. The white color and semisolid dairy scum (Fig. 1) is heated up to 50–60 °C to convert it into liquid and then liquid oil is passed to the transesterfication reactor followed by filtration. In the reactor, the scum oil is treated with KOH and methanol to get glycerin and crude biodiesel. Later, crude biodiesel is washed with water three times to remove the acid, soaps and residual catalyst followed by preheating at 110 °C to remove moisture from the produced biodiesel. The procedure is continued until getting pure oil methyl ester. The biodiesel production process is demonstrated in Fig. 2.

The Right fuel with right proportion and slight engine modifications may helps in getting better usage of fuel in the engine. In this regard, transesterified dairy scum biodiesel is considered for experimental investigations. The properties of DSOME and its blends such as viscosity, calorific value, density, flash point and fire point are determined as per the ASTM- 6751 standards [48–52]. Table 1 shows properties of the used diesel and biodiesel blends. The density of biodiesel blends are measured using a hydrometer at a temperature of 30 °C. The Flash points of the fuels are computed by using Pensky-Martens apparatus in 40–250 °C temperature range. The Bomb calorimeter is used to calculate the calorific value of the various fuel blends used in the study. The Redwood viscometer is used to calculate the kinematic viscosity of the biodiesel blends at a 40 °C temperature. Table 2 shows the properties of the Bio-CNG.

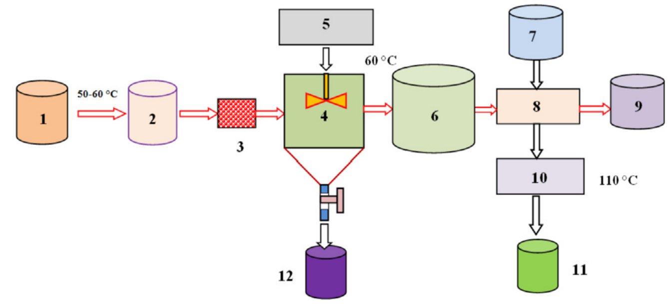


Fig. 2. Biodiesel preparation.
1: Dairy waste scum, 2: Scum oil, 3: Filter, 4: Transesterification reactor, 5: KOH + Methanol, 6: Crude biodiesel, 7: Water storage, 8: Washing, 9: Acids, soap and residual catalyst, 10: Drying, 11: Dairy scum biodiesel collector, 12: Glycerin collector.

Table 1
Properties of fuel blends.

Properties	**Methods IS 1448**	Diesel	B20	B100
Density (kg/m^3)	P:16	830	840	870
Viscosity at 40 °C (cSt)	P:25	2.9	2.98	4.36
Calorific value (kJ/kg)	P:6	43000	40890	38012
Flash Point (°C)	P:69	50	58	130
Fire point (°C)	P:69	60	68	142

Table 2
Properties of Bio-CNG.

Properties	
Methane (CH_4)	95–95.5%
Carbon dioxide (CO_2)	3–4.5%
Hydrogen sulphide (H_2S)	Less than 5 ppm
Water vapour (H_2O)	Nil
Oxygen (O_2)	Nil
Heating value (HV)	40 MJ/kg
Density (kg/m^3)	0.75

2.2. Modifications in direct injection diesel engine

The parameters of the standard diesel engines are suitable for diesel fuel operation. However, scrupulous research is required to optimize the engine parameters for biodiesel operation as they have different origins and properties. The tests are conducted on a kirloskar standard engine (SE) (IOP: 210 bar, IT: 23. deg.bTDC, CR: 17.5 and NH: 3) with using dairy scum biodiesel blends with diesel at various blend ratios (10, 20, 30 and 100%) and they are named as B10, B20, B30 and B100 respectively. Thereafter, the optimized fuel blend is carried for whole experimental study. Later, the standard engine IOP is varied from 210 bar to 240 bar in step of 10 bar (210, 220, 230 and 240 bar) and fuel injection timing is changed in step of 3° such as 20, 23, 26 and 29. deg.bTDC. Similarly compression ratio (CR) is also modified without altering the combustion chamber geometry to optimize the best compression among 16, 17 and 18. Thereafter, on the basis of the brake thermal efficiency of all the above stated parameters the best engine parameters are optimized as IOP: 230 bar, IT: 26. deg.bTDC, CR: 18. Then, these optimized parameters are carried further to examine the influence of the nozzle holes. The specifications and photographic views of the nozzles are given in Table .3 and Fig. 3 & Fig. 4. From the effect of nozzle study, it is observed that the nozzle with 5 holes has exhibited the better BTE when compared to 3 and 4 hole nozzles, hence 5 holes nozzle is optimized. The engine with optimized parameters (IOP: 230 bar, IT: 26. deg.bTDC, CR: 18 and NH: 5 holes) is carried further to investigate the effect of piston bowl geometry on

Fig. 1. Waste scum.

Table 3
Specifications of employed piston and fuel injectors.

Specifications	NH3	NH4	NH5
Number of hole	3	4	5
Part name	DLL110S639	DLLA150S1211	DLLA142S1033
Hole diameter(mm)	0.280	0.210	0.240
Area (mm^2)	0.0616	0.0346	0.0452
Spray angle (θ °)	110	150	142

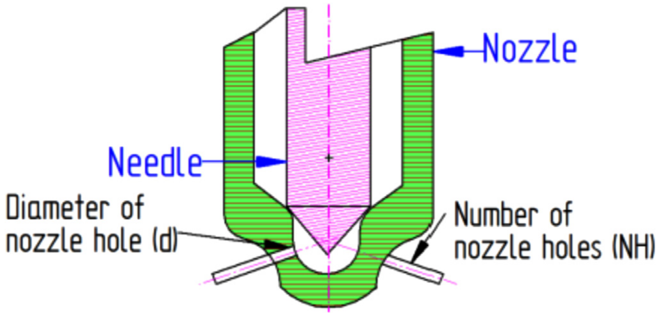


Fig. 3. Schematic view of the fuel injector nozzle.

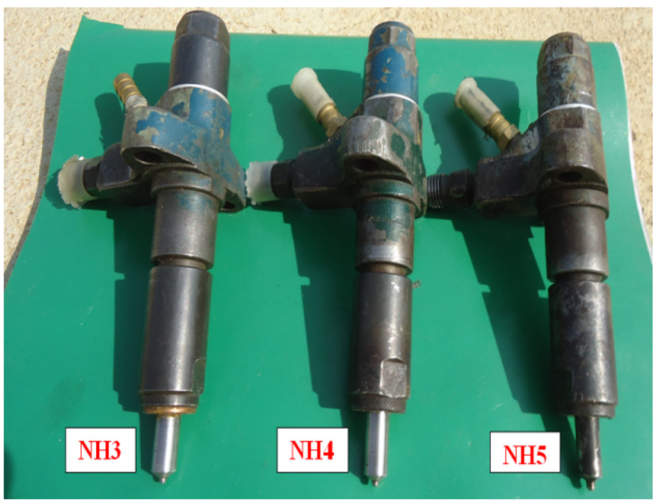


Fig. 4. Photographic view of 3, 4 and 5 holes nozzle fuel injectors.

diesel engine performance operated with B20 fuel blend. Finally, based on experimental results the baseline engine is modified with optimized engine parameters.

To guarantee the improved air-fuel mixing (swirl and squish) inside the cylinder is mainly depends on the piston bowl geometry in the direct injection diesel engine. In this study, the standard engine Hemispherical piston bowl geometry (HPBG) which is centrally positioned is modified into Straight sided piston bowl geometry (SSPBG), Toroidal piston bowl geometry (TPBG) and Re-entrant toroidal piston bowl geometry (RTPBG) without altering the bowl volume. The perception behind straight sided, toroidal and re-entrant toriodal piston bowl geometry is to exploit the entire oxygen with better mixing of fuel and air by providing the powerful squish and swirl inside the combustion chamber. Cross-sectional view, schematic diagram with dimensions and photographic view of the different piston bowl shapes are represented in Fig. 5, Fig. 6 and Fig. 7 [50]. The specifications of various piston bowls are given in Table .4. The tests are conducted on diesel engine at various loads (20%, 40%, 60%, 80% and 100%) to appraise the performance, combustion and emission behaviors of the diesel engine operated with 20% dairy scum biodiesel. Then after experiments are carried to unfold the effect of different engine parameters, nozzle hole geometry and piston bowl geometry (PBG) on diesel engine performance. For assessment and comparison, average of three readings is considered in the current study. The photographic views of piston disassembling and pistons after experimentation are shown in Fig. 8.

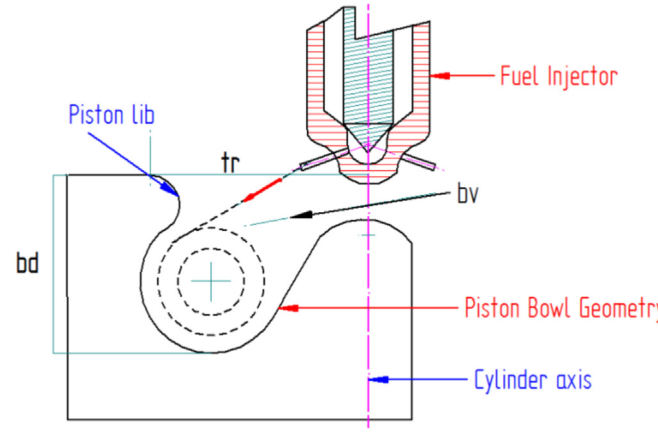


Fig. 5. Schematic view of re-entrant toroidal piston bowl and fuel injector.

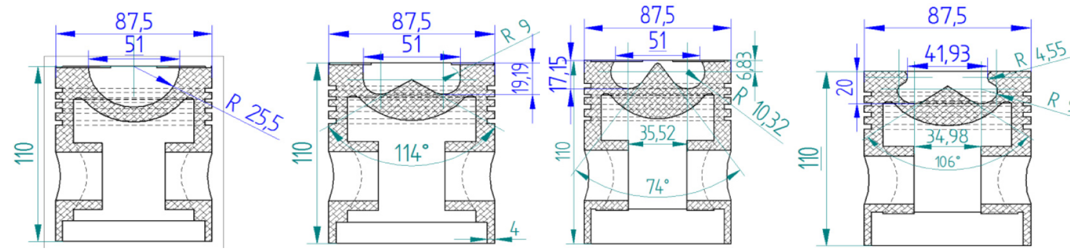
2.3. Experimental test setup

Kirloskar, 3.5 kW (TV1) diesel engine (Fig. 9) is used to conduct the experiments. The standard engine specifications are given in Table 5. Instruments used for the defined work are diesel engine-test rig, ECU, exhaust gas analyzer, burette and stopwatch, digital manometer, Chromel Alumel (K-Type) thermocouples. “Engine soft LV” software is employed for online combustion analysis. The combustion pressure at various crank angles is determined using Piezosensor and crank angle sensors. The test engine is operated at a constant speed of 1500 rpm for throughout experimentation. The engine is assembled with a dynamometer for loading the arrangement. The water circulation is provided to avoid the over-heating of the engine. Various temperature sensors are used to measure the water jacket temperature and calorimeter water temperatures. The fuel and air flow rates are determined using flow sensors. Data acquisition system is employed to determine the combustion characteristics of the diesel engine which is operated with dual fuel. Airrex Automotive Emission Analyzer HG-540 is used to measure the exhaust emissions such as Hydrocarbons, Carbon Monoxide and Oxides of Nitrogen. At very bnning of the experimentation, baseline engine readings are drawn for comparison when the engine is fueled with pure diesel. At the end, average of three readings is considered for the examination and comparison.

In the dual fuel operation, B20 fuel blend is used as liquid fuel (pilot fuel) and Bio-CNG is used as a gaseous fuel (primary fuel) which is filled in a cylinder at a pressure of 160 bar. This high pressure Bio-CNG is converted to 2 bar using a two stage pressure regulator. Rotameter is employed to measure the flow rate of the Bio-CNG. The T-type gas-air mixing chamber is used to mix the air and Bio-CNG properly before it entering into the engine cylinder. The flash back flame arrestor is used in gas flow line to avoid the flash back and fire. At the beginning of dual fuel experimentation, the effect of effect of Bio-CNG flow rates (in step of 0.12 kg/hr such as 0.12, 0.24, 0.36, 0.48, 0.60 and 0.72 kg/hr) on diesel performance is also studied and optimized the best Bio-CNG flow rate. At the end, optimized Bio-CNG flow rate (0.48 kg/hr is kept as constant) is further carried to explore the effect of injection timing on the performance of the dual fueled engine.

3. Results and discussion

The basic performance (BTE) of the diesel engine run with different DSOME fuel blends (namely B10, B20, B30 and B100) is studied and optimized the best fuel blend among all the fuel blends.



Note: all dimensions are in mm

Fig. 6. Schematic view of different piston bowl geometries.

(a) HPBG **(b) SSPBG** **(c) TPBG** **(d) TRPBG**

Fig. 7. Photographic view of different piston bowl geometries.

Table 4
Specifications of employed piston.

Specifications	HPBG	SSPBG	TPBG	RTPBG
Bowl volume (bv) (mm^3)	34727.9	34727.0	34727.4	34724.5
Throat diameter (tr) (mm)	25.5	25.5	25.5	20.965
Bowl depth (bd) (mm)	25.5	19.19	17.15	20.0
Piston diameter (mm)	87.5	87.5	87.5	87.5

Fig. 8. Photographic view of piston dissassembling and pistons after experimentation.

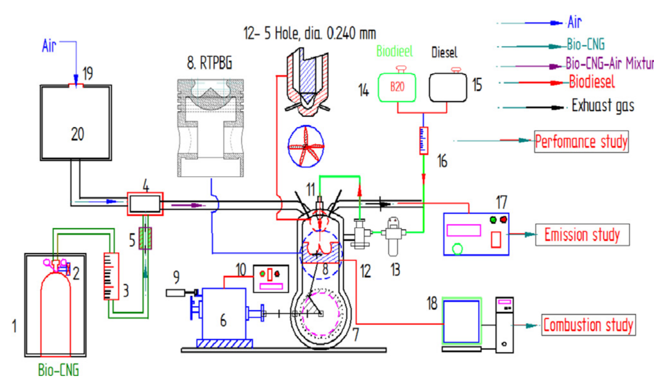


Fig. 9. Schematic diagram of experimental set up.
1. Bio-CNG cylinder, 2. Gas pressure regulator, 3. Rotameter, 4. Bio-CNG and air mixture, 5. Flame arrestor, 6. Dynamometer, 7. Engine, 8. Piston, 9. Crank encoder, 10. Control panel, 11. Fuel injector, 12. Fuel pump, 13. Filter, 14. Biodiesel tank, 15. Diesel tank, 16. Fuel burette, 17. Exhaust gas analyzer, 18. Data acquisition system, 19. Orifice meter, 20. Air box.

Further, the optimized fuel blend is carried to optimize the various engine parameters namely IOP, IT, CR, nozzle holes and piston bowl geometry at different loads. Later, the existing baseline diesel engine is modified with all above optimized engine parameters. At the end, the experiments are conducted on the modified engine to unfold the effect of injection timing on the performance, combustion and emission characteristics of the modified engine run with dual fuel (at constant Bio-CNG flow rate of 0.48 kg/hr).

3.1. Brake thermal efficiency of different fuel blends and engine parameters

Variation of BTE with brake power for Diesel and different DSOME fuel blends is depicted in Fig. 10. From the test results, it is

Table 5
TV1engine specifications.

Parameters	Specifications
Engine suppliers	Apex Innovations Pvt. Ltd
Type	TV1 (Kirloskar)
Cubic capacity	661 cc
Bore and stroke length	87.5 mm × 110 mm
Injector opening pressure	210 bar
No. of Nozzle holes	3 holes of 0.280 mm diameter
Piston bowl geometry	Hemispherical
Rated power	3.5 kW
Injection timing	23° bTDC (diesel)
No. of cylinder/stroke	One/Four
Compression ratio	17.5
Dynamometer	Eddy current
Software used	Engine soft

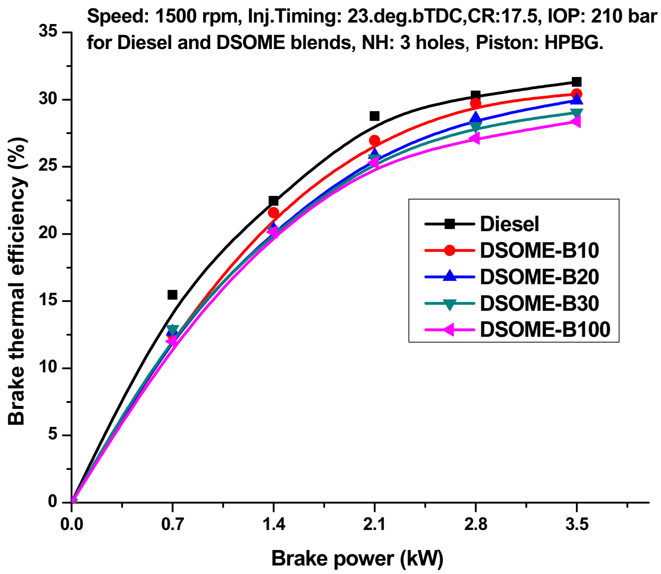


Fig. 10. Brake thermal efficiency versus brake power for different fuel blends.

watched that BTE for biodiesel is lesser than petroleum diesel operation for all loads. This could be accredited to lower calorific value, higher specific gravity and greater viscosity. The experimental result shows that the B20 operation results in better performance when compared to B30 and B100 fuel blends. This could be attributed to increased calorific value, lower viscosity and lower density of the B20 fuel blend, hence causing in improved performance. From results, it is observed that diesel has highest BTE of 31.32% followed by B10 of 30.42% BTE and it is slightly more than B20 BTE of 29.93%. BTE of B20 is near to B10 when compared to B30 (29.04%) and B100 (28.37%) fuel blends. Hence, B20 fuel blend is optimized as best fuel blend among others.

Variation of BTE versus BP for different IOPs is presented in Fig. 11. From graph, it is viewed that, as IOP is increased BTE is also increased. It could be attributed to improved atomization, vaporization; better air-fuel mixing leads to better combustion. At 230 bar IOP, enhanced BTE is noticed, it might be due to better air-fuel mixing during higher loads at compression process leads to maximum BTE. The higher BTE of B20 (30.55%) at 230 bar pressure is near to the petro-diesel (31.32%) at full load condition. However, B20 operation with 240 bar IOP, the BTE is decreased (27.97%) when compared to 230 bar and 220 bar IOP (29.94%) at all loads. The reason might be due to, too increased IOP would reduce the fuel droplet size, hence too finer (very small size) fuel droplets would cause lower momentum, delayed ignition and lowered relative velocity. Introduction of more fuel at greater loads may results improper combustion by its own combustion products [9]. The 210 bar IOP has shown the lower BTE of 29.63% than 230 bar at full load. Among all IOPs, 230 bar IOP has shown the improved BTE, hence it is optimized and carried for further investigations.

Fig. 12 depicts the variation of BTE against brake power for different ITs. The 23. deg.bTDC (engine manufacturer) IT is better for petro-diesel operation. BTE is reduced for B20 operation when compared to the petroleum diesel at 23. deg.bTDC IT. The lower calorific value and higher density of the B20 fuel blend may require more fuel to produce the same output power as of pure diesel. The improved BTE is noticed with advanced IT of 26. deg.bTDC when compared to baseline IT of 23. deg.bTDC. This could be ascribed to more time availability for air-fuel mixing in the combustion chamber results in improved combustion process and releases the more heat during combustion. Whereas, by retarding the IT from 23. deg.bTDC to 20. deg.bTDC, the BTE is decreased, this might be due to reduced interaction time of air and fuel hence causing in sluggish combustion with lower heat release. From the experimental study, it is revealed that BTE values for Diesel-23. deg.bTDC, 20. deg.bTDC, 23. deg.bTDC, 26. deg.bTDC and 29. deg.bTDC are found to be 31.32, 28.21, 30.55 31.03 and 29.15% respectively for B20 operation at full load. For B20 operation, 26. deg.bTDC IT has resulted higher BTE of 31.03% which is near to the Diesel BTE of 31.32%. Hence, 26. deg.bTDC IT is optimized as best IT for B20 operation.

The variation of BTE versus brake power for various compression ratios is presented in Fig. 13. The BTE for diesel is highest when compared to B20 with different compression ratios. It is because of higher calorific value and lower viscosity of the diesel fuel. The increased BTE is noticed with the increased compression ratio for

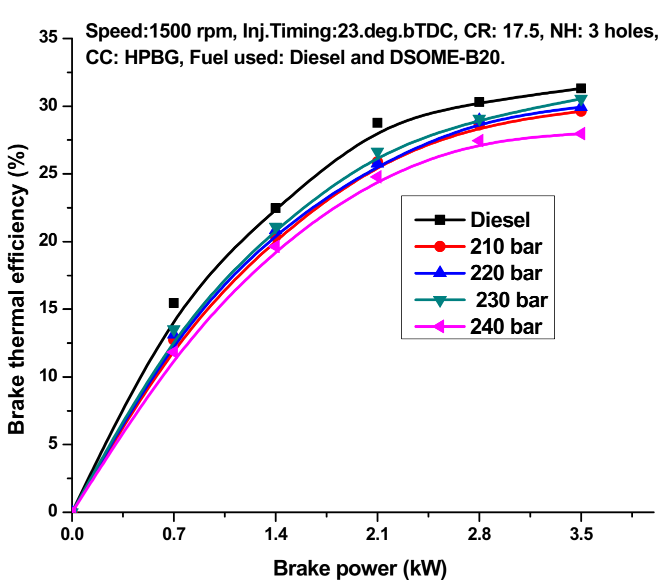


Fig. 11. Brake thermal efficiency versus brake power for various injector opening pressures.

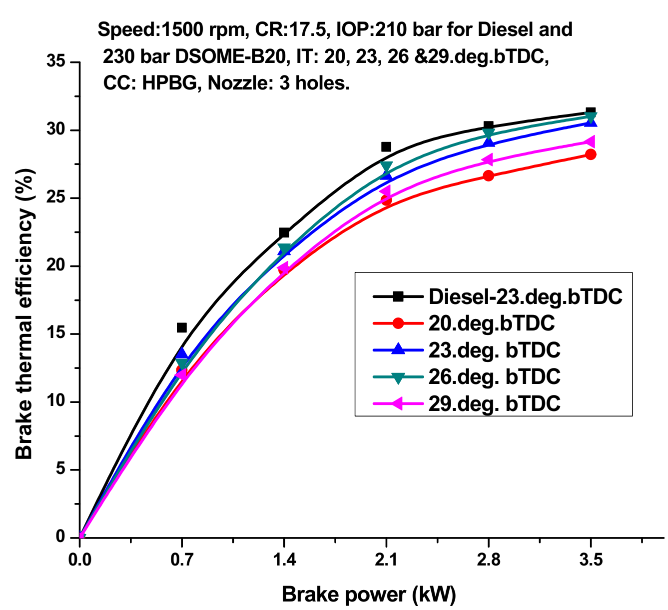


Fig. 12. Brake thermal efficiency versus brake power for different injection timings.

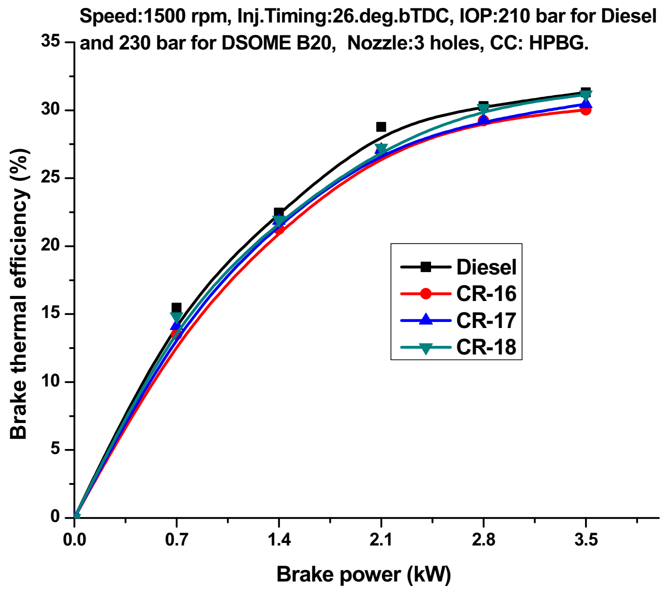


Fig. 13. Brake thermal efficiency against brake power for different compression ratios.

all loads. This might be due to higher air temperature and better mixing of fuel and air inside the cylinder results in faster evaporation and complete combustion. At 100% load, when the compression ratio is increased from CR 16 to CR 18, the BTE is also increased from 30.03% to 31.15% respectively. CR 18 has revealed the higher BTE which is closer to the BTE of diesel (31.32%). The least BTE is noticed with lower CR. The reason for this might be dilution of residual gas which causing in sluggish burning of the products during combustion. From the test results, it is cleared that CR 18 is the best compression ratio for B20 operation among other compression ratios.

Fig. 14 depicts the variation of BTE against brake power for different nozzles. The baseline engine run with diesel has confirmed the highest BTE of 31.32% when compared to other 3 and 4 holes nozzles operated with B20 fuel blend at 100% load. For optimized parameter engine (IOP: 230 bar, IT: 26. deg.bTDC, CR: 18), 5 holes nozzle has resulted the higher BTE of 31.5% than 4 holes of 29.30% and 3 holes of 31.15%. This could be attributed to enhanced atomization and fuel-air mixing inside the cylinder leading to rapid evaporation and burning. The too smaller diameter fuel droplet would have the lesser relative velocity and momentum hence causes partial suffocation with its own combustion products [13,15,16]. The larger diameter (higher denser) fuel droplets have higher penetration and lesser velocity hence poor fuel-air mixing rate causes the incomplete combustion. About, 4.98% of BTE is increased with 5 holes nozzle with optimized engine parameters (IOP, IT and CR) when compared to baseline engine. Therefore, 5 holes nozzle with IOP: 230, IT: 26. deg.bTDC and CR: 18 are further carried to examine the effect of combustion chamber geometry on the performance of diesel engine run with 20% biodiesel fuel blend.

Fig. 15 shows the distinction of BTE with brake power for standard, modified engine with different piston bowl geometries fueled with B20 fuel. The lowest BTE (29.93%) is noticed with the B20 fuel blend in baseline (standard) engine operation. The reason for this might be lower calorific value, higher specific gravity and greater viscosity of B20 fuel blend. Whereas, modified engine (IOP:230 bar, IT:26. deg.bTDC, CR:18, NH:5 holes, CC:HPBG) has shown the improved BTE (31.50%) than standard engine (IOP:210 bar, IT:23. deg.bTDC, CR:17.5, NH:3 holes, CC:HPBG) operated with B20 [50]. This could be due to improved fuel atomization, increased cylinder temperature and more time availability for air-fuel mixing rate leads to faster oxidation and evaporation process, hence causing in better combustion. Whereas, in case of modified engine with TPBG has revealed the grater BTE of 31.69% than HPBG (30.50%) and SSPBG (30.29%) for maximum load range. This could be attributed to improved air motion in the TPBG leads to better air-fuel mixture formation and evaporation causing in complete combustion. The RTPBG has shown highest BTE (32.28%) than the all various piston bowls. This could be attributed to better swirl, as entering of swirl air which spreads downwards and outward into the undercut region and then divides into streaming up the bowl sides and stream flowing along the bowl base. And also re-entrant cavity with round lip generates larger spray volumes and spray spreading. The fuel hits just on the lip corner produces the maximum spreading area and also corner radius helps to disperse

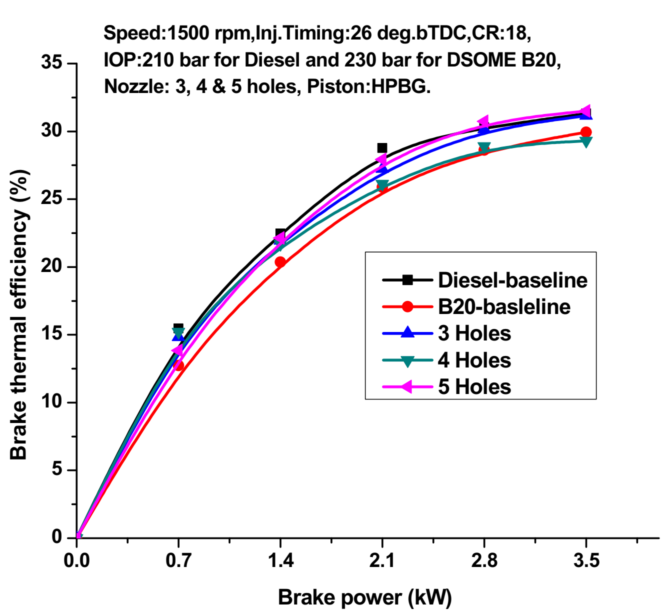


Fig. 14. Brake thermal efficiency versus brake power for different nozzle holes.

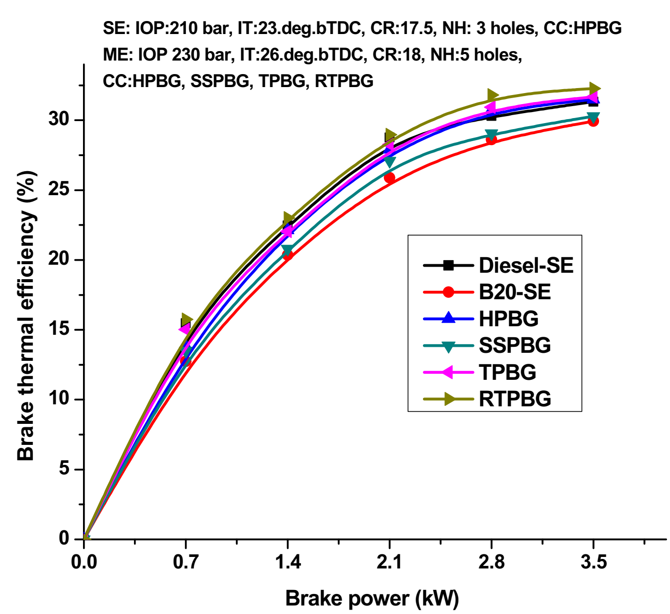


Fig. 15. Brake thermal efficiency versus brake power for various combustion chamber geometries.

the fuel accumulated at the bottom corner. Hence, RTPBG is optimized as best piston bowl geometry as it causing better air-fuel mixing and combustion.

Based on the experimental test results, the baseline engine (IOP: 210 bar, IT: 23. deg.bTDC, CR: 17.5, NH: 3 holes and CC: HPBG) is modified with the optimized engine parameters (IOP: 230 bar, IT: 26. deg.bTDC, CR: 18, NH: 5 holes and CC: RTPBG) [50]. This modified is engine is further carried to unfold the effect of fuel injection timing on the performance of diesel engine operated with dual fuel (B20+Bio-CNG).

Fig. 16 shows the distinction of BTE with brake power for HPBG standard and modified engine fueled with B20 fuel blend. In standard engine (SE) operation, it is observed that BTE for B20 is lower than petroleum diesel at all loads. The modified engine (ME) has shown the improved BTE than the SE engine operated with B20 fuel blend. This could be ascribed to improved fuel atomization, increased cylinder temperature and more time availability for fuel-air mixing leads to faster oxidation and evaporation process hence cause better combustion. The BTE results for Diesel-SE, B20-SE and B20-ME are found to be 31.32, 29.93 and 32.28% respectively at maximum load. Whereas, in case of modified engine operated with B20 (B20-ME) has revealed the highest BTE than baseline engine operated with B20 fuel blend (B20-SE) for entire load range. This could be attributed to improved air motion and heat transfer in the RTPBG leads to proper mixing of burned and unburned fluid particles along with better air-fuel mixture formation and evaporation causing in complete combustion. Whereas, BTE is lower in HPBG-SE (B20-SE) than RTPBG-ME as it has insufficient turbulence (air swirl) inside combustion cavity results in poor burning efficiency. The dual fuel (B20+Bio-CNG) operation (DF) has resulted in lower brake thermal efficiency when compared to single fuel (B20) [31,32,34–37]. This could be attributed lower pilot fuel injection and reduced oxygen content for the combustion when Bio-CNG is admitted into the cylinder. It also observed from graph that, increase in Bio-CNG flow rate increases the ignition delay hence, results the slower flame propagation in gas and air mixture when the Bio-CNG flow rate is greater than pilot fuel (B20). However, with higher pilot introduction into the combustion chamber would helps to burn Bio-CNG properly and completely. The lower Bio-CNG flow rate has shown the improved brake thermal efficiency as it exhibits better combustion. Whereas, in case of higher Bio-CNG flow rate the thermal efficiency of the engine is lower. This could be attributed to higher gas flow rate would reduce the oxygen supply during the combustion. In dual operation, the BTE is increased to 3.84% when Bio-CNG flow rate is reduced to 0.12 kg/hr from 0.72 kg/hr. The highest BTE is obtained in dual fuel operation with 0.12 kg/hr flow rate of 23.9% among 0.24 kg/hr (23.5%), 0.36 kg/hr (22.75%), 0.48 kg/hr (22.155%), and 0.60 kg/hr (21.475) bio-CNG flow rates. From the experimental results, it came to know that BTE of 0.12 kg/hr Bio-CNG operation is decreased to 23.69% lesser than the diesel-SE operation, 20.14% lesser than B20-SE operation and 25.96% lesser than B20-ME operation. However, increase in Bio-CNG flow rate more than 0.48 kg/hr, engine started to knock and observed the engine vibration at higher loads. Hence, 0.48 kg/hr Bio-CNG flow rate is optimized and carried further to study the effect of IT on dual fuel engine. Introduction of gaseous fuels results in very fast reaction rates hence, causing very high pressure rise rate and uncontrolled combustion leads to knocking. A small amount of increase in the gaseous fuel beyond a limit could result in very severe knocking [46,47].

3.2. *Effect of injection timing on diesel engine performance, combustion and emission characteristics operated with B20+Bio-CNG*

3.2.1. *Brake thermal efficiency*

Fig. 17 shows the distinction of BTE with brake power standard and modified engine fueled with B20 fuel. In standard engine (SE)

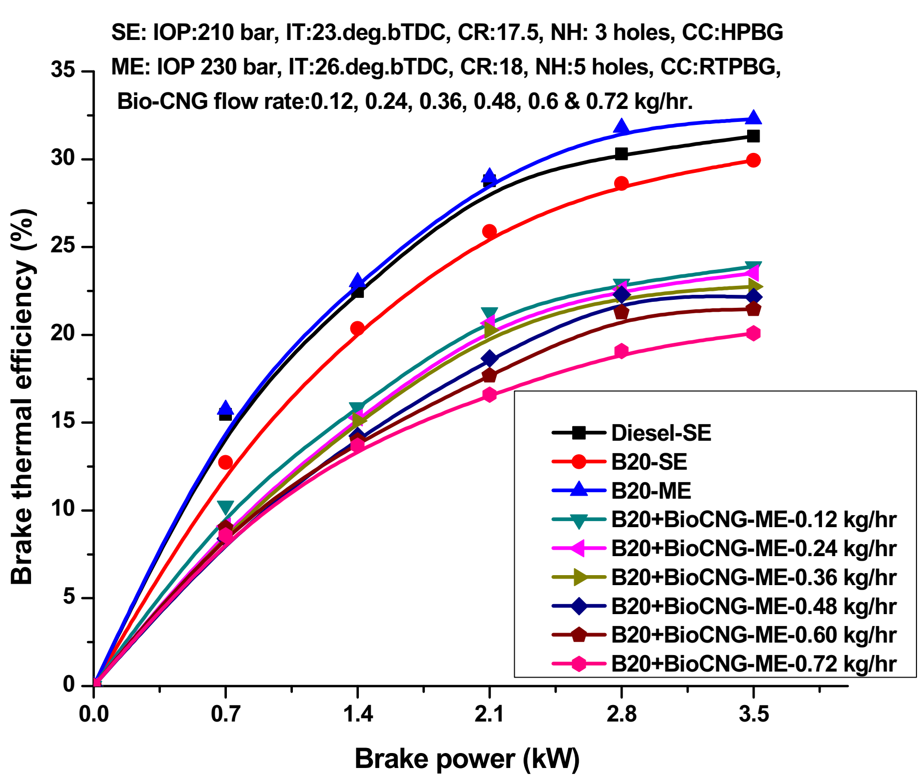


Fig. 16. Brake thermal efficiency versus brake power for various Bio-CNG flow rates.

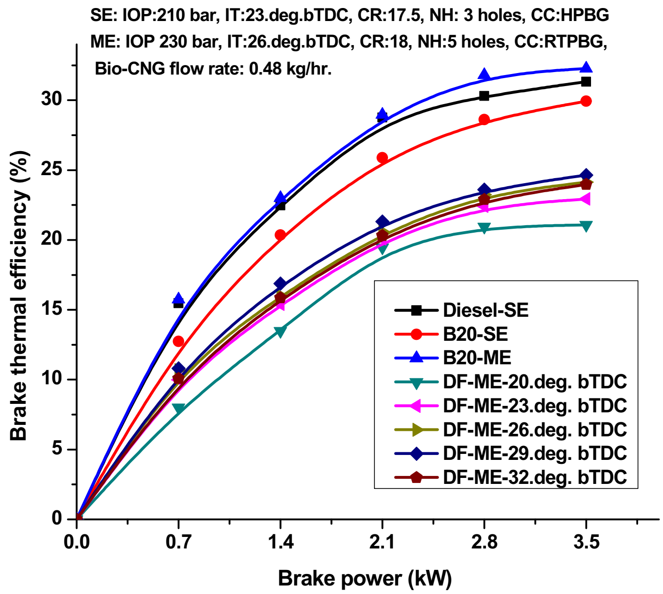


Fig. 17. Brake thermal efficiency versus brake power for various injection timings.

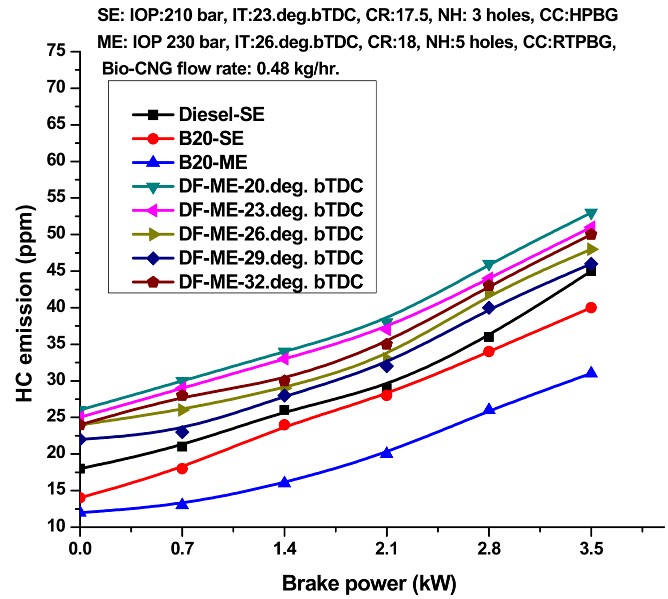


Fig. 18. HC emission versus brake power for various ITs.

operation, it is observed that BTE for B20 is lower than petroleum diesel for all loads. This could be due to lower calorific value, higher specific gravity and greater viscosity of B20 fuel blend. Whereas, modified engine (ME) has shown the improved BTE than SE engine operated with B20 fuel blend. This could be due to improved fuel atomization; increased cylinder temperature and more time availability for fuel mixing rate leads to faster oxidation and evaporation process hence, cause better combustion. The BTE results for Diesel-SE, B20-SE and B20-ME are found to be 31.32, 29.93 and 32.28% respectively at maximum load. Whereas, in case of modified engine with B20-ME has revealed the greater BTE than B20-SE for entire load range. This could be attributed to enhanced air motion and heat transfer in the RTPBG leads to proper mixing of burned and unburned fluid particles along with better air-fuel mixture formation and evaporation causing the complete combustion. From the experimental results it is observed that, BTE is increased with advancing the injection timing. This could be attributed to rapid combustion process at premixed combustion phase leads to improved BTE. However, as the IT is retarded BTE is decreased, because retarded IT gives the less time for air-fuel mixing leads to improper air-fuel mixing results in slow burning of charge which is introduced in the combustion chamber. The DF-ME-29. deg.bTDC has shown the greater BTE of 24.64% among other ITs at constant Bio-CNG flow rate of 0.48 kg/kW.hr. Further, advancing of IT, BTE is reduced and experienced the knocking at medium and higher loads. The experimental results for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 21.07, 22.93, 24.14, 24.64 and 23.96% respectively at full load condition.

3.2.2. Hydrocarbon emission

Fig. 18 depicts the distinction of hydrocarbon (HC) emission levels with brake power for standard and modified engine. The HC emissions are lower for DSOME operation when compared to HS diesel for standard engine operation. This might be due to increased gas temperature in the cylinder and more oxygen presence in the B20 in comparison with conventional diesel. The modified engine with B20-ME has resulted in lower HC emission than standard engine B20-SE. This could attributed to optimum engine parameter's effect in turn better fuel atomization, faster evaporation rate, and increased cylinder temperature. And also generation of more turbulent kinetic energy inside the re-entrant toroidal combustion chamber with improved air-fuel mixing also leads to better oxidation and combustion process by burning the complete fuel which is admitted in the combustion chamber. This may also responsible for reduced HC emission in B20-ME when compared to B20-SE operation. The HC emission values for Diesel-SE, B20-SE and B20-ME are found to be 45, 40, and 31 ppm respectively at full load. The HC emission for dual fuel operation is more than the single fuel operation. This could be attributed to insufficient oxygen availability in combustion process leads to decreased volumetric efficiency and incomplete combustion with introduction of Bio-CNG into the cylinder when compared to single fuel operation. However, the complete combustion can be achieved with supplying more amounts of air and injecting more amount of pilot fuel into the cylinder. In dual fuel operation, for all loads, the HC emissions for all Bio-CNG flow rates are higher when compared to neat conventional diesel. This could be due to lower air-Bio-CNG mixture temperature leads to slower combustion [40]. The HC emissions are decreased, as the IT is advanced from 23. deg.bTDC to 29. deg.bTDC. This could be attributed to longer ignition delay leads to improved pilot fuel spray atomization, improved intensity of turbulence and increased heat transfer to the unburned charge. This increased ignition delay would cause the higher spray penetration and improved fuel-air-gas mixture before ignition, hence causing in improved combustion of compressed air-fuel mixture during rapid combustion phase. Therefore, a larger premixed region yields the higher combustion rate with increased combustion temperature helps in reducing the HC emissions. The experimental results of HC for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 53, 51, 48, 46, 50 ppm respectively at maximum load condition.

3.2.3. Carbon monoxide emission

The comparison of carbon monoxide (CO) emission with brake power for standard and modified engine is represented in Fig. 19. CO emission of biodiesel and their respective blends are lower than the conventional diesel. This might be due to presence of more oxygen content in the biodiesel leads to complete oxidation and

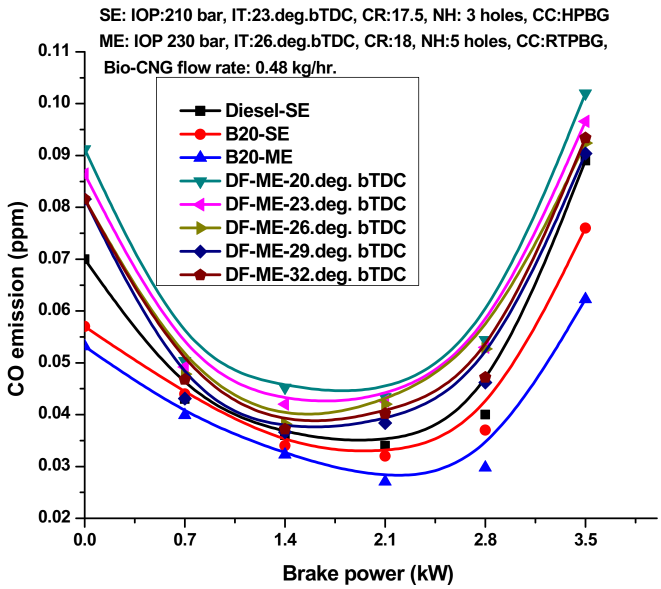


Fig. 19. CO emission versus brake power for various ITs.

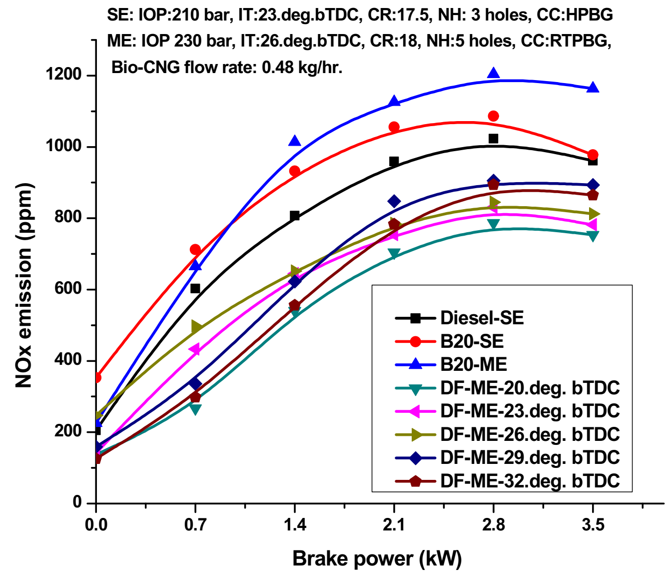


Fig. 20. NOx emission versus brake power for different ITs.

combustion. The modified engine with B20 operation results in lower CO emission than standard engine as the modified engine provides greater cylinder temperature to accelerate the evaporation of the charge, thereby its result in faster oxidation to CO into CO_2 hence reduces the CO emission. The standard engine operation with pure diesel (Diesel-SE) has resulted in highest CO emission of 0.089% when compared to biodiesel operation (Both B20-SE and B20-ME operations). The modified engine (B20-ME) has shown the lower CO emission of 0.062% when compared to standard engine (B20-SE) of 0.076%. This could be attributed to improved air motion and squish inside the cylinder releases the more heat with improved oxidation process thereby improves the combustion process. The CO emission levels for dual fuel operation at all loads is higher than single fuel operation with modified and without modified engine [40]. Due to early injection of pilot fuel into the combustion chamber results in lower CO emissions. As early injection of pilot fuel provides better premixing of air and fuels before the top dead centre (TDC) providing sufficient time for oxidation process with higher cylinder temperature results in complete combustion and lower CO emissions. The experimental results of CO emission for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, and DF-ME-32. deg.bTDC are found to be 0.102, 0.0966, 0.0924, 0.0904, and 0.0934%.

3.2.4. Oxides of nitrogen emission

Fig. 20 depicts the variations of NOx emissions for standard and modified engine at different loads operated with B20 fuel blend. Higher cylinder temperature and oxygen presence in the fuel would results the NOx formation in diesel engine. NOx emission is found to be greater for B20 fuel when compared to petro-diesel for all loads. Higher HRR during premixed burning phase observed with biodiesel fuels leads to greater cylinder temperature and improved combustion. This higher HRR and temperature might be responsible for higher NOx formation. The NOx emission results for Diesel-SE, B20-SE and B20-ME are found to be 961, 978 and 1164 ppm respectively at maximum load. The modified engine (B20-ME) has exhibited the highest NOx emission in comparison with standard engine (B20-SE) when both engines are operated with B20 fuel blend. This could be attributed to improved turbulent motion of air with higher oxygenated B20 fuel intensifies the more heat release and heat transfer to the burned and unburned parts leads to increased combustion temperature. Therefore, it causes greater NOx formation in the B20-ME when compared to B20-ME. At higher loads, the NOx emission is lower for dual fuel operation when compared to single fuel operation. It is evident from the graph that, NOx emission decreases with increased Bio-CNG flow rates at increased loads for dual fuel operation. This might be attributed to lower gas-air mixture temperature and slower burning speed during the combustion as increased amount of Bio-CNG flow rates reduces the oxygen concentration in the charge. Lowest NOx emission is noticed with retarded DF-ME-20. deg.bTDC of 753 ppm. However, highest NOx is observed with advanced DF-ME-29. deg.bTDC IT of 893 ppm. This could be due to lower combustion temperature of charge in the cylinder leads to slower and improper combustion [46]. The NOx emission is increased with advanced IT. This might be attributed to increased cylinder temperature and premixed charge (proper mixed pilot fuel-air-Bio-CNG) temperature during uncontrolled combustion zone leads to higher NOx emission. Further, increase in IT beyond 29. deg.bTDC, the NOx emission is reduced while engine knocking rate is increased. At full load, the experimental results of NOx emissions for DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-32. deg.bTDC are found to be 782, 812 and 865 ppm respectively.

3.2.5. Cylinder pressure

The deviations of cylinder pressure versus crank angle for standard engine and modified engine at full load are presented in Fig. 21. Graph showed that, peak cylinder pressure is increased as the load is increased. From the experimental results, it is observed that B20-ME has resulted in highest cylinder pressure when compared to B20-SE. The higher cylinder pressure with B20-ME might be attributed to more turbulence intended higher flame speed would helps to cause rapid combustion of fuel droplets which are entered into the combustion cavity. Hence, B20-ME causing in rapid combustion with increased pressure waves when the piston is at top dead centre (TDC) results in highest cylinder pressure when compared to non-turbulent B20-SE. The cylinder pressure values for Diesel-SE, B20-SE and B20-ME are found to be 53.63, 52.29 and 55.42 bar respectively at higher load. The highest cylinder pressure with B20-ME is due to higher turbulence, increased chemical reactions, and increased flame front velocity

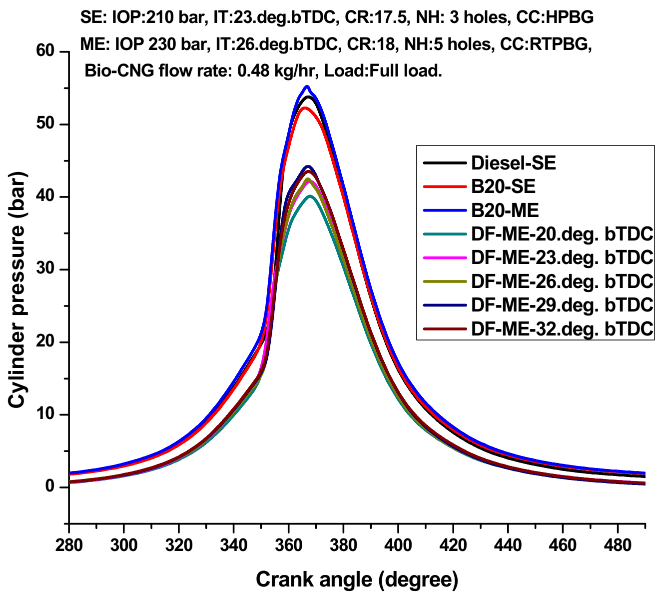


Fig. 21. Cylinder pressure versus crank angle for various ITs at full load.

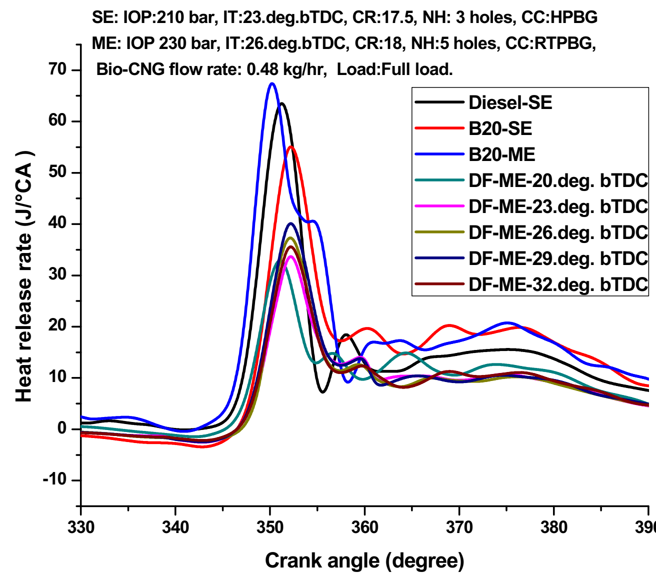


Fig. 22. Heat release rate versus crank angle for different ITs at full load.

results in rapid rise of pressure in the cylinder leads to increased cylinder pressure. The maximum cylinder pressure and cylinder temperatures are observed for advanced IT. This exhibits the improved flame front and combustion of air-gas mixture which results in higher cylinder temperature and pressure. However, lower cylinder pressure and temperature is observed with retarded IT. This could be ascribed to slower burning of induced gaseous fuel in the combustion chamber during rapid combustion phase. The retarded injection timing cause the lower charge and cylinder temperature therefore, this temperature is not enough for the flame propagation in the complete gaseous fuel-air mixture, hence leads to incomplete combustion [46]. At maximum load, the experimental cylinder pressure results for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 40.14, 42.20, 42.63, 44.27 and 43.56 bar respectively.

3.2.6. Heat release rate

Fig. 22 depicts the rate of heat release profile versus crank angle for standard and modified engine at full load. Higher heating value and lower viscosity of the diesel results in higher heat release rate (HRR) than the B20-SE when engine operated at standard engine parameters. There is maximum HRR is observed with B20-ME (modified engine) operation when compared to B20-SE (standard engine) operation. This might be attributed to the improved air-fuel mixing rate, faster evaporation and combustion with optimized parameters of the engine fueled with B20. The HRR results for Diesel-SE, B20-SE and B20-ME are found to be 65.43, 57.14, and 70.18 J/degree crank angle respectively at maximum load. Engine operation with B20-ME has exposed the highest HRR in comparison with B20-SE. The reason for this might be enhanced chemical reaction with intimate mixing of fuel and air during the compression process causing in higher turbulence which results the complete burning of weak charge hence increases the heat releases and heat transfer to the cylinder wall. Whereas, in case of open type combustion chamber of B20-SE (HPBG), HRR is least amount as they have more surface-volume ratio which causes in lower cylinder pressures and HRRs at different locations of the piston cavity hence, cause lesser flame speed. Increase in the liquid fuel would increases the heat release rate and lowers the ignition delay. The reason might be increased liquid fuel would utilize the complete oxygen which is introduced in the cylinder during combustion leads to improved combustion which results in increased HRR, reduced combustion duration and rapid heat transfer to the cylinder wall. Whereas, in case of advanced IT, ignition delay increases and HRR decreases which might be due to higher self ignition temperature of Bio-CNG and more heat loss during pre combustion with increased ignition delay. The HRR values for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 30.49, 35.36, 38.64, 41.78 and 37.07 J/degree crank angle respectively at full load condition.

3.2.7. Ignition delay period

Fig. 23 depicts the variation of ignition delay with brake power for standard engine and modified engine at maximum load condition. From the test results, it is observed that ignition delay for B20 is greater than the petroleum diesel. This could be attributed to

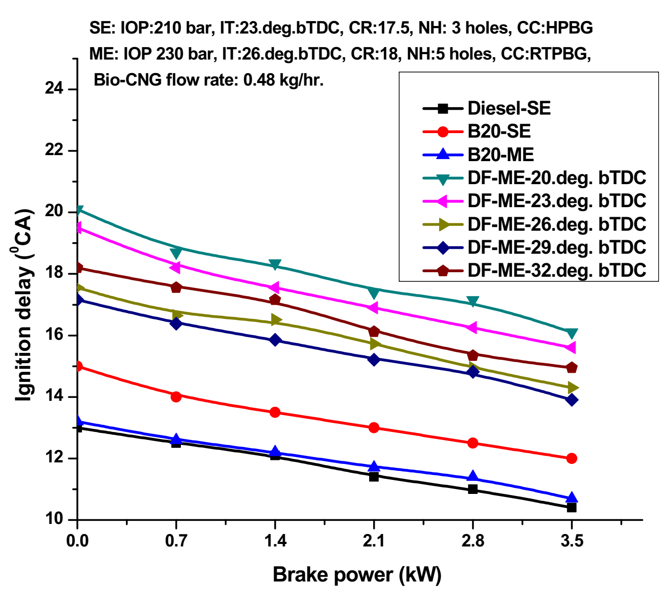


Fig. 23. Ignition delay period versus brake power for various ITs.

higher viscosity of B20 cause poor atomization and air-fuel mixing during premixing phase. Standard engine with pure diesel (Diesel-SE) has resulted the lowest ignition delay of 10.40 deg. crank angle when compared to B20-SE of 12 deg. crank angles at maximum load condition. The lowest ignition delay of 10.7 deg. crank angle is observed with B20-ME (RTPBG) when compared to B20-SE of 12 deg. crank angle. The lower ignition delay with B20-ME might be attributed to higher turbulence of the B20-ME during compression process. This higher turbulence would accelerate the chemical reactions with appropriate mixing of oxygen and fuel in combustion cavity. Therefore, higher turbulence and higher cylinder wall temperature helps to ignite the fuel in advance hence, weak mixture in the combustion chamber would burn completely in a shorter time. For entire load range, the ignition delay for retarded IT is more when compared to adanvced IT. This could be attributed to higher self ignition temperature and lower cetane number of Bio-CNG leads to increased ignition delay. Whereas, in advanced IT the ignition delay is reduced. The reason for this might be advanced IT would help to mix the Bio-CNG with air easily and forming the homogeneous mixture results in reduced physical and chemical delay. The experimental results of cylinder pressure for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 16.1, 15.6, 14.3, 13.91 and 14.95 deg. crank angle respectively at full load.

3.2.8. Combustion duration

Fig. 24 represents the variation of combustion duration with brake power. For diesel, as it has lower viscosity leads to faster atomization and air-fuel mixing results in rapid combustion since combustion duration is lower when compared to B20. The experimental results of combustion duration for Diesel-SE, B20-SE, and B20-ME are found to be 38, 44 and 38 deg. crank angle respectively. The lowest combustion duration is observed with B20-ME, which might be attributed to enhanced flame front velocity due to higher air swirl (turbulence) in the re-entrant toroidal combustion chamber hence decreases the combustion time. From the graph it is noticed that, combustion duration is reduced with advanced IT. This could be due to proper mixing of Bio-CNG with air during premixed combustion phase with accelerated flame propagation speed leads to improved flame development and combustion.

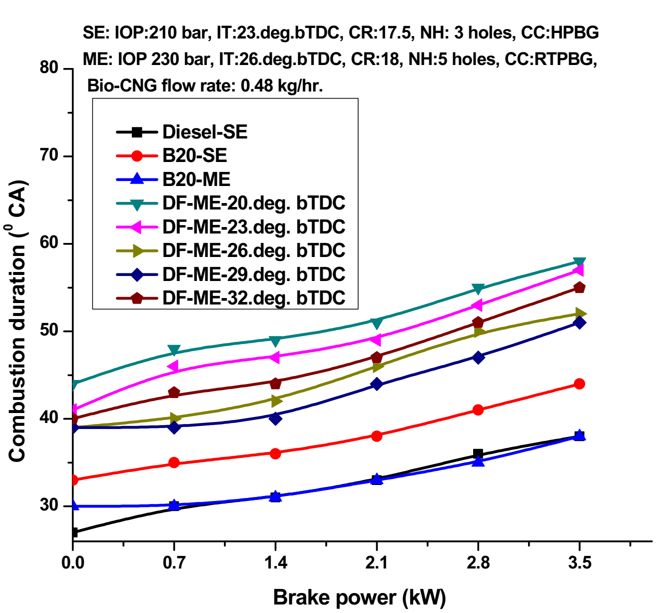


Fig. 24. Combustion duration versus brake power for various ITs.

Whereas, in case of retarded IT, due to admission of more amount of Bio-CNG in the premixed combustion process results in increased ignition delay hence requires more time (as Bio-CNG has higher self ignition temperature and lower cetane number) for later phase of combustion. The combustion duration results for DF-ME-20. deg.bTDC, DF-ME-23. deg.bTDC, DF-ME-26. deg.bTDC, DF-ME-29. deg.bTDC, DF-ME-32. deg.bTDC are found to be 58, 57, 52, 51 and 55 degree crank angle respectively at full load.

4. Conclusion

In the current study, experiments are conducted on the four stroke direct injection diesel engine. The following conclusions are drawn based on the experimental results for both single and dual fuel operations.

4.1. Single fuel operation

1. From the experimental study, it is observed that DSOME-B20 resulted in greater BTE than the DSOME-100, DSOME-B30 and lower BTE than DSOME-B10. However BTE of DSOME-B20 is closer to DSOME-B10, hence DSOME-B20 is optimized.
2. From the optimization study, it is revealed that enhanced BTE is observed with IOP of 230 bar (among 210, 220, 230 and 240 bar), IT of 26. deg.bTDC (among 20. deg.bTDC, 23. deg.bTDC, 26.deg.bTDC and 29. deg.bTDC) and CR of 18 (among CR 16, CR 17 and CR 18), Nozzle of 5 holes (among 3, 4 and 5 holes) and piston of RTPBG (among HPBG, SSPB, TPBG and RTPBG).

Finally, the modified engine (IOP: 230 bar, IT: 26. deg.bTDC, CR: 18, NH: 5 holes and CC: RTPBG) has exhibited the improved performance when compared to the standard engine (IOP: 210 bar. IT: 26. deg.bTDC, CR: 18, NH: 3 holes and CC: HPBG). Therefore, dual fuel experiments are carried on the modified engine to unfold the effect of injection timing on the dual fueled engine (B20+Bio-CNG).

4.2. Dual fuel operation

1. The BTE is increased from 22.93% to 24.96% when IT is advanced from 23. deg.bTDC to 29. deg.bTDC at full load. However, reduced BTE (23.96%) is observed with IT of 32. deg.bTDC. Whereas, IT of 20. deg.bTDC has resulted in least BTE of 21.07% when compared to all other ITs.
2. For 29. deg.bTDC IT reduced HC of 46 ppm and CO of 0.0904% is observed when compared to HC emission of 20. deg.bTDC (53 ppm), 23. deg.bTDC (51 ppm), 26. deg.bTDC (48 ppm) and 32. deg.bTDC (50 ppm) and CO emission of 20. deg.bTDC (0.102%), 23. deg.bTDC (0.0966%), 26. deg.bTDC (0.0924%) and 32. deg.bTDC (0.0934%) ITs at full load operation.
3. However, increased NOx emission of 893 ppm is observed with 29. deg.bTDC IT when compared to 20. deg.bTDC (753 ppm), 23. deg.bTDC (782 ppm), 26. deg.bTDC (812 ppm) and 32. deg.bTDC (865 ppm) at full load operation
4. Highest cylinder pressure is observed with 29. deg.bTDC IT (44.27 bar) when compared to 20. deg.bTDC (40.41), 23. deg.bTDC (42.20 bar), 26. deg.bTDC (42.63) and 32. deg.bTDC (43.56 bar) ITs at full load operation. At full load, highest HRR is noticed with 29. deg.bTDC (41.78 J/deg. crank angle) when compared to 20. deg.bTDC IT (30.49 J/deg. crank angle), 23. deg.bTDC (35.36 J/deg. crank angle), 26. deg.bTDC (38.64 J/deg. crank angle) and 32. deg.bTDC (37.07 J/deg. crank angle) ITs.
5. The combustion duration for 29. deg.bTDC IT is lower (51 deg. crank angle) when compared to 20. deg.bTDC (58 deg. crank angle), 23. deg.bTDC (57 deg. crank angle), 26. deg.bTDC (52 deg. crank angle) and 32. deg.bTDC (55 deg. crank angle) ITs at full

load operation. The ignition delay for 29. deg.bTDC IT is lower (13.9 deg. crank angle) when compared to 20. deg.bTDC (16.1deg. crank angle), 23. deg.bTDC (15.6 deg. crank angle), 26. deg.bTDC (14.3 deg. crank angle) and 32. deg.bTDC (14.95 deg. crank angle) ITs at full load operation.

On the whole, in single fuel operation modified engine has exhibited the better performance (BTE) when compared to the standard engine. From the dual fuel (B20+Bio-CNG) experimental study, it is concluded that modified engine with advanced injection timing of 29. deg.bTDC has shown the improved performance, combustion and emission characteristics when compared to the other injection timings.

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Abbreviations

ASTM	American society for testing and materials
BMEP	Brake mean effective pressure (bar)
BSFC	Brake specific fuel consumption (kg/kW.hr)
bTDC	before Top dead centre
BTE	Brake thermal efficiency (%)
Bio-CNG	Enriched methane/Compressed biogas
B10	10% biodiesel+90% diesel
B20	20% biodiesel+80% diesel
B30	30% biodiesel+70% diesel
B100	100% Biodiesel
B20-SE	20% Biodiesel operated standard engine
CBM	Compressed bio-methane
CC	Combustion chamber
CHRR	Cumulative heat release rate (KJ)
CI	Compression ignition
CNG	Compressed natural gas
CO	Carbon monoxide (%)
CO_2	Carbon dioxide (%)
CR	Compression ratio
deg.	degree
DI	Direct injection
Diesel-SE	Standard engine operated with diesel
DSOME	Dairy scum oil methyl ester (biodiesel)
ECU	Electronic control unit
EGR	Exhaust gas recirculation
HC	Hydrocarbons (ppm)
HCC	Hemispherical combustion chamber
HPBG	Hemispherical piston bowl geometry
HS	High speed
HRR	Heat release rate
IDP	Ignition delay period
IOP	Injector opening pressure
IT	Injection timing
KOH	Potassium hydroxide
kW	killo Watt
LBG	Liquefied biogas
LBM	Liquefied bio-methane
ME	Modified engine
NH	Nozzle hole
NOx	Nitrogen oxide (ppm)
PBG	Piston bowl geometry
RTPBG	Re-entrant toroidal piston bowl geometry
SE	Standard engine
SSPBG	Straight sided piston bowl geometry
TPBG	Toroidal piston bowl geometry
TRCC	Toroidal re-entrant combustion chamber

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