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Thermodynamic Performance of an Engine by Modifying Piston Bowl Geometries Fuelled by SME-100, LA-100, KB-100 Biodiesel Blends, and Diesel

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ABSTRACT

Simulation of Direct injection (DI) compressed ignition (CI) engine working on diesel thermodynamic cycle on diesel-Rk software carried out for evaluating the effect on thermal performance. The analysis of an engine was worked out by applying different bowl geometrical shapes and by testing with different fuels. Further same compositions were tested on an experimental test rig having a set-up of (compressed ignition, single cylinder, four strokes, air-cooled, direct injection) diesel engine at constant crank speed. Hemispherical (HCC), Shallow depth (SCC), Re-entrant (RCC), Double wedge shallow, and Toroidal (TCC) piston geometries were created in solid-work and analyzed with B-100 blend of SME (Soybean Methyl Ester), KB (Karanja Biodiesel), LA (Roselle Biodiesel) and diesel further they were analyzed and their effects have investigated experimentally and numerically at fully loaded condition, with a constant crank speed of 1500 rpm and by setting constant compression ratio at 17.5. BSFCs were higher by 21.03%, 12.97%, and 12.96% for SME100, KB100, and LA100 with hemispherical, toroidal, and re-entrant type combustion chambers compared with the diesel fuel. Indicated thermal efficiencies and ignition delay periods were reported slightly lower for different blends and pure biodiesel than diesel at specific load conditions whereas combustion durations were reported higher compared to diesel.

1. Introduction

The demand for energy and fuel due to rapid growth in the human population worldwide leads to a hike in the prices of petroleum [1]. This increase in need also leads to a reduction in the stock of fossil fuels as reserves of fossil fuel are limited and maybe it lasts for a few decades. In the majority of the commercial sector transportation is a key factor. The transportation sector consumes the second largest amount of energy which is considered a key factor for projections of energy demand [2,3]. Aiming for improvement in fuel consumption and to achieve emission standards there have been some techniques applied like improvisation in fuel quality by little

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modifications in properties, a small alteration in engine design, and treatments on the exhaust. The objective of modifying or redesigning work on the engine is meant to enhance turbulence kinetic energy and which in turn leads to the engine performance along with matching emission standards. Modification in the engine includes various parameters like bowl shape, injection pressure, injection timing, and ignition delay. There has been a gigantic amount of research in the area of elective fuels and different piston geometries.

Assessment of thermodynamic performance, which includes study and comparison based on thermal parameters like combustion duration, ignition delay, and specific fuel consumption, is a widely used methodology in IC engine research area. Many researchers have attempted the same study with lower blends of biodiesel (B20, B30) but for higher blends (above B70), and by modifying piston bowl shape, rarely anyone has attempted. Here an attempt has been made to fulfill the said research gap. Thermodynamic performance assessment of different B100 fuels and diesel carried out numerically and compared by using Diesel-RK software.

1.1 Literature Survey

Yang *et al.,* [4] modified shapes of the cavity on the piston head and carried out an analysis of the performance of combustion and other important factors of an engine. Three sets of bowl geometries including HCC (hemispherical combustion chamber, hemispherical cavity), SCC (shallow combustion chamber, shallow hemispherical cavity), and OCC (omega combustion chamber, omega-shaped cavity) were installed at different engine speeds (1200, 2400, and 3600 rpm). An improved pattern of squish upgrades the equivalence ratio at higher motor speeds. They derived the following results at low crank speed, shallow geometry gives better combustion, performance and fewer values of emission parameters, while at moderate speed, omega shape CC results in a good value of heat release rate, less CO emissions value, and high peak pressure measured than HCC and SCC. However, NOx emissions were found to be in high amounts. Jaichandar *et al.,* [5] reviewed an empirical study on the impact of a re-entrant type cavity on a DI-CI diesel engine powered by Pongamia oil-based biofuel also known as POME. They regulated two sets of different bowl geometries one was Toroidal Re-entrant Combustor (TRCC) and another was Shallow Re-entrant Combustor (SCC) fueled with B20 and petroleum-based diesel fills (PBDF). They obtained higher thermodynamic performance values along with a sharp diminishment in the value of UBHC, PM, and CO while utilizing a 20% blend in TRCC. The only negative aspect of their study was rising in NOx value. Ravikrishna [6] experimented and led to the emission reduction that occurred by introducing a high vortex re-entrant cup geometry with a bladeless injector. In addition, they applied a simulation-based study with the help of the CFD tool for finding the combustion and emission parameters. The modification resulted in proper combustion with lower emissions readings. Injection timing, 6° CA bTDC was found optimal which resulted in a 26-27% reduced NOx emissions and 84-85% reduced soot levels compared to the base configuration. Jafarmadar [7] demonstrated in his experiment that the combustion process gets highly influenced by the equivalence ratio followed by the homogeneity factor during the charge mixing process. Also, combustion initiation is influenced due to the heat release rate quantity and the pressure curves peaks which are other two important factors, which play an important character during the combustion phenomenon. Miniature cup size creates balanced squish and vortex formation besides that it creates less spray penetration and a shorter ignition delay period. The results show that the toroidal chamber with B 20 blend showed a significant rise in the value of thermal efficiency and a fall in the value of SFC compared to SCC & HCC. Mamilla [8] illustrated in his experimental work that the optimized B20 blend proved better in performance and executed the lowest pollutant readings like HC, CO, and smoke with the usage of toroidal geometrical shapes. However, NOx resulted significantly low in other modified shapes compared to toroidal shapes and that found higher compared to the conventional hemispherical combustion chamber with diesel. Venkanna [9] performed a test on a CI engine using B100 of Calophyllum oil methyl esters to evaluate performance, emission, and combustion characteristics. High injectors (DI) entering pressure exhibited improvement in brake thermal efficiency and reduction in UBHC, HC, CO, and smoke emissions values. The promising result was received by them while using various bowl cavity shapes along with B20 blend fuel. The further combined effect of changing compression ratio and different bowl cavity shapes was studied and compared based on various combustion and performance parameters. Benajes *et al.,* [10] carried out experimentation on DI-RCCI diesel-fueled engines to investigate the consequences of piston bowl cavity shapes on RCCI functioning and pollutant results at low to high variations in engine cranking loads. They used the three-piston geometry called kick, step, and tub they measured a slight increase in CO and HC while performing experiments at low load. While working with medium load, the fuel consumption and exhaust gases were minimal for the bathtub shape cavity at high load, the stepped piston showed the better result as lower BSFC and lower exhaust gases. Ganji *et al.,* [11] presented the effect of various cavity shapes on piston heads named conventional HCC (hemispherical combustion chamber), SCC (Shallow depth), and TCC (Toroidal) on the performance and emissions parameters of DI CI diesel engine with a constant value of compression ratio kept around 17.5. The Converge CFD tool coupled with the SAGE combustion model for numerical analysis has been applied and they derived that the Toroidal piston cavity provides better mixing of charge inside the cylinder, which leads to smooth charge distribution. Further trials (iterative study) on software were conducted to analyze the most preferable bowl cavity shape by modifying the depth of Toroidal geometry. The result showed that with 1.26mm decrease in the depth of the bowl from the baseline (TCC) design resulted in better emissions reading and acceptable performance output. Bapu *et al.,* [8] conducted an experimental study on an adjustable compression ratio DI CI single-cylinder engine run on blends of Calophyllum Inophyllum Methyl Ester (CIME). Reduced emissions and improved process outcomes were achieved by improvising the fuel preparation, charge mixing process, and turbulence by modifying the geometrical specifications of the piston bowl. The formation of the A/F mixture inside the chamber space was simulated at different piston positions (like TDC, BDC, and Midway) using Ansys Fluent software recognized that MHCC (modified hemispherical combustion chamber geometry) is an ideal design of the piston bowl shape for the B20 blend compared to that of conventional HCC design.

2. Methods

2.1 Numerical Simulation

Diesel-RK software tool used here for simulation and the thermal analysis of a DICI diesel cycle. Full-cycle engine simulation can be done professionally with precision by the Diesel-RK software. It gives freedom to cover a variety of practice tasks, starting from the design of engine systems and up to the complex multi-cylinder engine thermal evaluation [12-14]. Diesel-RK simulates and solves turbocharging, gas-exchange model, EGR, heat transfer in engine elements, friction, and water condensation, which means almost all aspects of full cycle simulation. Initially, different bowl geometries were created in SolidWorks CAD software and the same had been imported to the Diesel-RK environment for further analysis. DIESEL-RK software as a tool provides a virtual environment for conducting numerous trials and combinations of effective parameters and different fuels [14]. High-speed data processing has allowed DIESEL-RK to take optimization tasks as well, including optimizing the crown piston shape and fuel injection system to achieve better combustion, performance, and low NOx, and smoke emissions [15].

The fuels used for the said investigation were prepared as per ASTM [16] and also properties were tested for further comparison to one another and the same data were taken as input for numerical simulation by Diesel-RK software [18,19]. Table 1 shows the model and mechanism used for the simulation of a diesel engine.

Table 1
Model used in a simulation

The model used in the Simulation	
Dimension/ Geometry	Model
Ignition Delay Period	Tolstov's Mechanism
Combustion	Multi-Zone Model

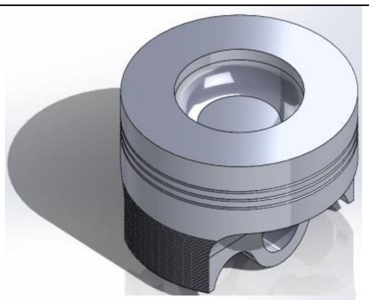
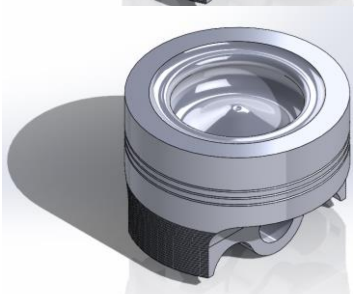
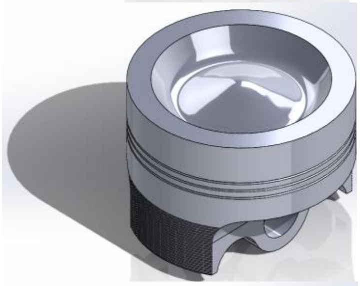
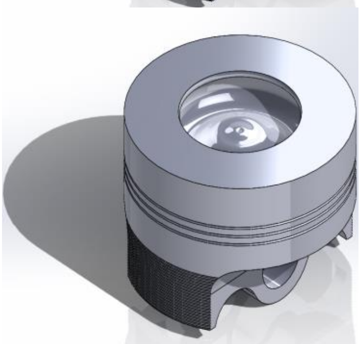
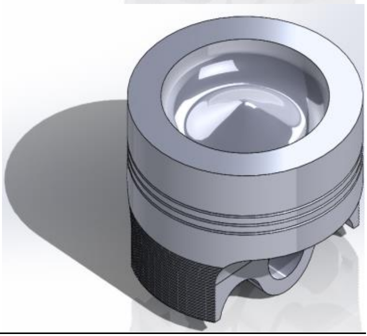
Table 2 gives an idea about fundamental equations which as a software program runs and computes the required desired output.

Table 2
Governing Equation used by Diesel-RK software

Eq. No	Name of equation	Governing equation for Diesel-RK model
i	Mass conservation	$\frac{dm}{dt} = \Sigma_j \dot{m}_j$
ii	Conservation Equation for Species	$Y_i = \sum_j \left(\frac{\dot{m}_j}{m}\right)(Y_i^j - Y_i^{cyl}) + \frac{\Omega_i w_{mw}}{\rho}$
iii	Energy Conservation	$\frac{d(mu)}{dt} = -p\frac{dv}{dt} + \frac{dQ_{ht}}{dt} + \sum_j \dot{m}_j h_j$
iv	Equivalence Ratio	$\alpha_1 = \frac{\left(\frac{A}{F}\right)}{\left(\frac{A}{F}\right)_s} = \frac{(\dot{m}_a/\dot{m}_f)}{(\dot{m}_a/\dot{m}_f)_s}$
v	Mean Effective Pressure (Frictional)	FMEP=$\alpha + \beta P_{max} + \gamma V_p$
vi	Specific Fuel Consumption	SFC=$\frac{\dot{m}_f}{P_b}$
vii	Heat Release Rate (ignition delay)	$\tau = 3.8 * 10^{-6}(1 - 1.6 * 10^{-4}.n)\sqrt{\frac{T}{P}} exp\left(\frac{E_a}{8.312T} - \frac{70}{CN + 25}\right)$
viii	Heat Release Rate (premixed combustion)	$\frac{dx}{d\tau} = \Phi_0 * \left(A_0\left(\frac{m_f}{v_i}\right) * (\sigma_{ud} - x_0) * (0.1 * \sigma_{ud} + x_0)\right) + \Phi_1 * \left(\frac{d\sigma_{ud}}{d\tau}\right)$
ix	Heat Release Rate (controlled combustion)	$\frac{dx}{d\tau} = \Phi_1 * \left(\frac{d\sigma_u}{d\tau}\right) + \Phi_2\left(A_2\left(\frac{m_f}{v_c}\right) * (\sigma_u - x) * (\alpha - x)\right)$
x	Heat Release Rate (late combustion)	$\frac{dx}{d\tau} = \Phi_3 A_3 K_T(1 - x)(\xi_b \alpha - x)$

Table 3 shows images of different piston bowl geometries which were drafted by using Solidworks software. This geometry is fed into Diesel-RK software for finding the thermodynamic performance of an engine.

Table 3
Geometries used for investigation

Model Geometry Name	3D Geometry
Hemispherical	
Toroidal	
Shallow Depth	
Re-entrant	
Double Wedge Shallow	

2.2 Biodiesel Preparation and Experimental Setup

In this study total of four different fuels were used for evaluation. Engine performance was tested on three different biodiesel B100 blends and diesel. Out of three biodiesel, one blend was derived from edible feedstock whereas the other two biodiesel blends were derived from the non-edible feedstock. Soybean has been used as one of these biodiesels in the engine, it is a very versatile grain and is used for the extraction of oil which is usually consumed by the food industry and agrochemical industry [17,20,21]. Other biodiesel, which is used, is Karanja and Roselle biodiesel as these are two non-edible oil sources. Karanja oil is normally extracted from seeds of the Millettia pinnata tree belonging to Australian and Indian region [21]. Roselle (Hibiscus sabdariffa) is botanical plant species of the Malvaceae family [22]. Physical and chemical properties of Karanja, Roselle, and Soybean oil along with its blends were tested and recorded [24-27]. Further, these properties are used as input for software studies.

Table 4
Fuel properties recorded by the test

Properties	Roselle biodiesel (LA)	Karanja Biodiesel (KB)	Soybean biodiesel (SME)	Diesel	Test method (ASTM)
Relative density	0.878	0.88	0.876	0.830	D 287
Viscosity (Kinematic) (mm^2/s)	4.6	4.86	4.62	3.0	D 445
Heating Value (MJ/kg)	38.78	38.91	36.22	42.5	D 4809
Cetane Number	52.2	52.1	51.3	48	D 613
C%	78.71	77.96	77.31	87	D 6866
H%	12.12	13.05	11.88	12.6	D 613
O%	9.23	8.97	10.81	0.4	D 4629

The trial runs were conducted on a diesel engine which is shown in Figure 1 (four-stroke, single-cylinder, direct injection) by using Pure Roselle, Karanja and Soybean based biodiesel (B100) at fixed injection timing (23.5° bTDC) and compression ratio (17.5:1). Table 4 gives important fuel properties. Fuels were tested for tabulated properties in a standard laboratory as per ASTM standards.

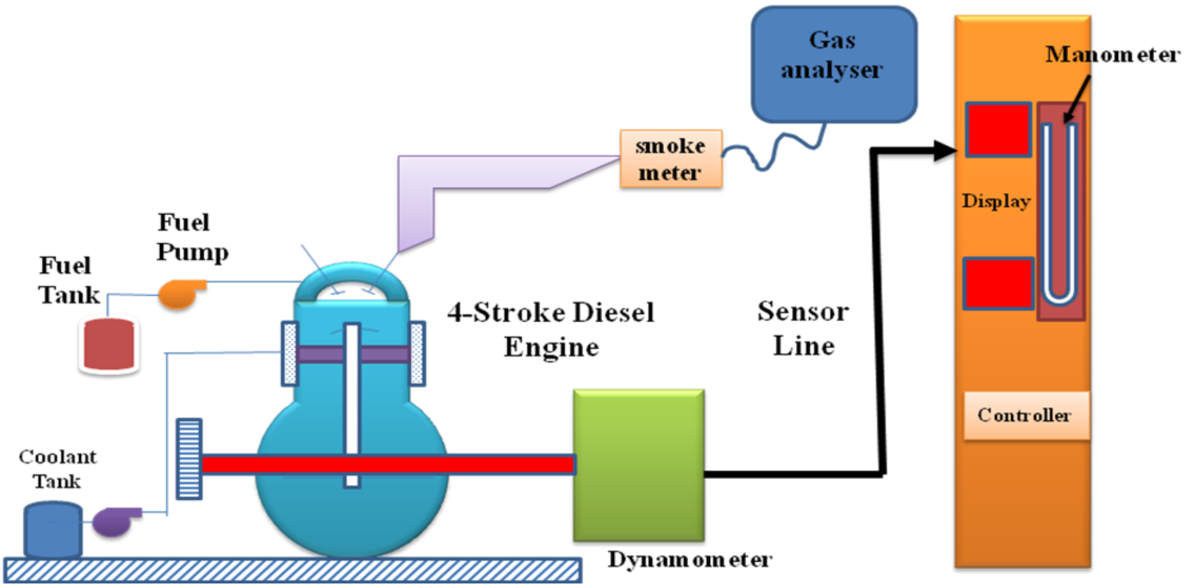


Fig. 1. Experimental setup schematic diagram

Table 5 shows the detailed specification of an engine that was run with specified varieties of fuel and tested to get output in the form of thermodynamic performance. Trials were conducted on diesel engines by changing fuels (LA B-100, KB B-100, SME B-100, and Diesel) and by varying different piston bowl geometries. The setup consists of a single-cylinder, four strokes, DI VCR engine coupled with an eddy current-type dynamometer for a varying load. Air temp, coolant temp, mass air pressure, and trigger sensor are connected to Open ECU which controls fuel flow, fuel injector, and fuel pump. Set up is provided with necessary instruments for combustion pressure and crank-angle measurements. These signals are interfaced with a computer for pressure crank-angle diagrams. Instruments are provided to interface airflow, fuel flow, temperatures, and load measurements. Rota meters are provided for engine cooling water and calorimeter water flow measurement. The setup enables the study of VCR engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, specific fuel consumption, A/F ratio, heat balance, and combustion analysis. Lab-view-based Engine Performance Analysis software package "Enginesoft" is provided for online engine performance evaluation. Observation and result tables were obtained from software output files and the same results were compared with the results from numerical (Diesel-RK software) analysis.

Table 5
Engine Details used for Experimentation and Numerical simulation

Engine Specifications	Single Cylinder, 4-stroke Diesel, water-cooled, Power (P) 3500 W at Speed (N) 25 rps, stroke (L) 11 cm, bore (D) 8 cm, CR 17.5
Connecting rod length	23.5 cm
Injection timing of fuel	23.5° before TDC
Swirl Ratio	1.5
Opening of the Intake valve (IVO)	16° before TDC
Closing of the Intake valve (IVC)	42° after TDC
Opening of Exhaust valve (EVO)	42° before TDC
Closing of Exhaust valve (EVC)	16° after TDC

4. Result Analysis
4.1 Brake-Specific Fuel Consumption

It is termed as a ratio of a fuel supplied in terms of mass flow rate to the shaft power achieved at a crankshaft which signifies how effectively the fuel is getting burned to obtain desired brake power [28,29]. BSFC quantifies the efficiency of the load in the engine and is therefore a crucial term for analysis. Furthermore, BSFC shows how accurately the engine's fuel is converted into shaft work [27]. Studies show that raising the mixing ratio of the biodiesel oil in the mixture raises, the Latent Heat Value and BSFC also decreases. This is why the BSFC increases when biodiesel oil is mixed with pure diesel fuel. BSFC of Diesel, SME, KB, and LA are as follows with different piston geometry with full load conditions at 1500 rpm [24].

The bar graph for diesel and various biodiesel has shown below in Figure 2 with modified piston geometries for bsfc. Table 6 gives bsfc (kg/kWh) values for modified pistons and for different fuels that are observed and calculated by diesel-RK software and compared with experimental output. The BSFC first increases under low motor load conditions and then gradually decreases with the increasing motor load. Maximum BSFC for SME100, KB100, and LA100 is 0.2955 (HCC), 0.27583 (TCC), and 0.2758 (re-entrant) kg/kWh higher respectively than diesel with their piston geometry. The thermal values of biodiesel are smaller than diesel.

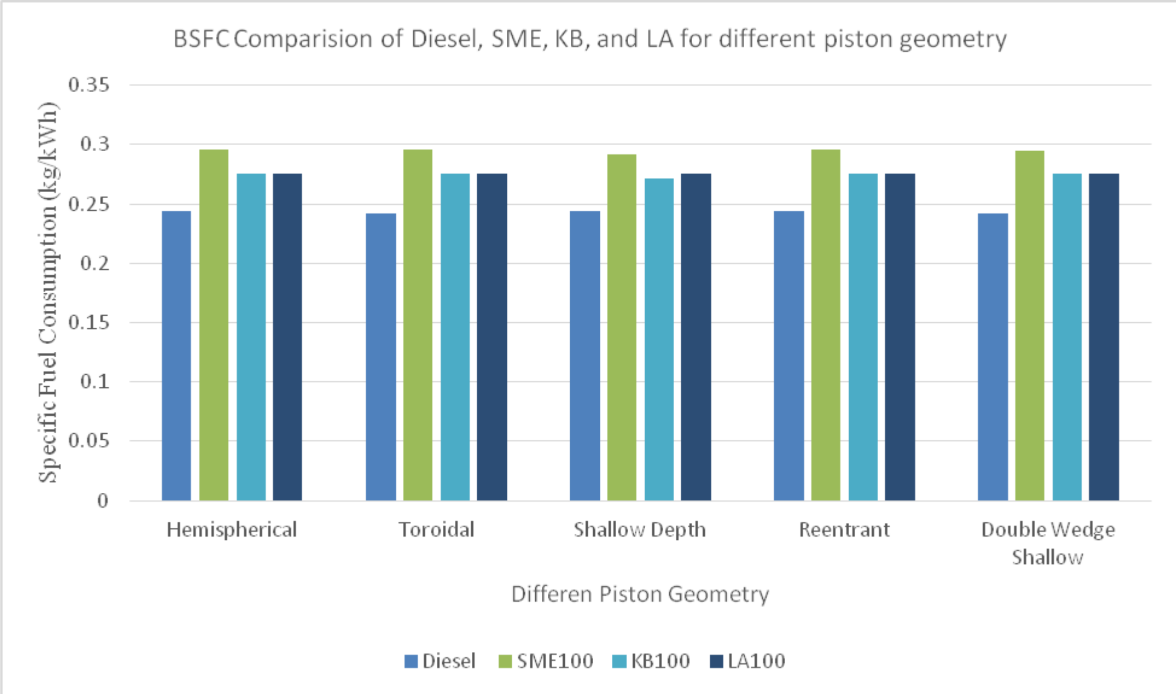


Fig. 2. BSFC Comparison of Diesel, SME, KB, and LA for different piston geometry

Table 6
Specific Fuel Consumption (kg/kWh)

	Diesel		SME100		KB100		LA100	
	Num.	Exp.	Num.	Exp.	Num.	Exp.	Num.	Exp.
Hemispherical	0.24365	0.247	0.2955	0.3000	0.27549	0.279	0.27561	0.280
Toroidal	0.24213	0.246	0.29529	0.3000	0.27583	0.277	0.27552	0.280
Shallow Depth	0.24415	0.245	0.29214	0.2999	0.27149	0.276	0.27552	0.279
Reentrant	0.24386	0.244	0.29544	0.2940	0.2756	0.278	0.27580	0.276
Double Wedge Shallow	0.24173	0.240	0.29513	0.3003	0.27546	0.277	0.27531	0.277

4.2 Indicated Thermal Efficiency

The ratio of indicated power generated through the engine to the power developed by the burning of the fuel is termed an ITE. It quantifies the accounted power used by the piston to that of available total fuel power. It has also determined by the ratio of the power produced by fuel combustion and the power produced by the engine. Changes in specific thermal characteristics for different fuel combinations with different piston geometries were recorded in this research work. With different piston geometries, it has been noticed that there is a small difference between the efficiency shown for biofuel compared to conventional petroleum diesel oil [28]. The results show that the maximum specific indicated thermal efficiency of SME100, KB100 and LA100 is about 41.19% (shallow depth), 41.114% (shallow depth), 41.88%, and 40.53% (double wedge shallow) respectively. The regulatory efficiency of pure diesel oil is increased due to the HCV of the fuel compared to biodiesel. The bar graph for Diesel and various biodiesel is shown below [24]. The bar graph for diesel and various biodiesel has shown below in Figure 3 for different piston geometries versus ITE at 1500 rpm. Table 7 gives ITE (%) values for modified pistons and for different fuels that are observed and calculated by diesel-RK software.

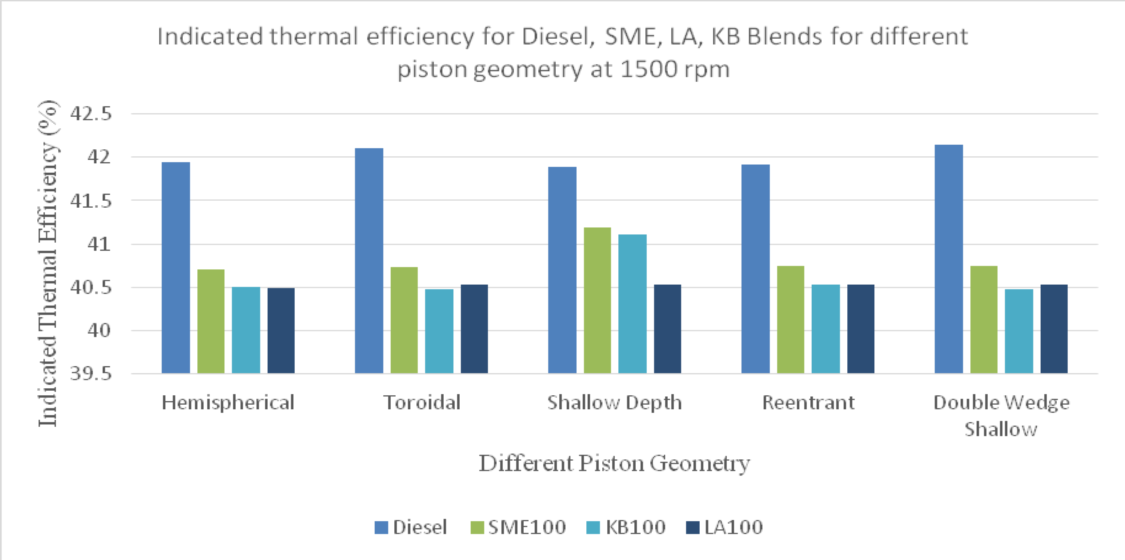


Fig. 3. Indicated thermal efficiency for Diesel, SME, LA, and KB Blends for different piston geometry at 1500 rpm

Table 7
Indicated Thermal Efficiency (%)

	Diesel		SME100		KB100		LA100	
	Num.	Exp.	Num.	Exp.	Num.	Exp.	Num.	Exp.
Hemispherical	41.95	43.44	40.713	41	40.506	41	40.495	41.1
Toroidal	42.1	43	40.739	41.4	40.481	40.9	40.527	41.125
Shallow Depth	41.89	42.5	41.19	41.5	41.114	41.2	40.527	41.5
Reentrant	41.92	41	40.741	40.9	40.533	40.65	40.526	41.5
Double Wedge Shallow	42.149	41.5	40.744	40.8	40.473	40.85	40.53	41

4.3 Ignition delay

It shows the reading of the time interval between the start of fuel injection and the start of fuel combustion. This is a crucial terminology used to quantify the quality of combustion (cetane number) which directly affects the start-up and combustion flow. The shorter delay interval affects the combustion performance of an engine [28,32]. The ignition delay of the CI engine fuel behaves inversely proportional to its cetane number [30]. Hence, optimization of the ignition delay parameter improves the better performance of the engine. Changes in the mechanical delay load change for different combinations with different combustion chambers were recorded. Fuel with a higher cetane content has shorter combustion delays and better fuel diffusion compared to the combustion phase of the mixture [31]. Biodiesel and its compounds show higher cetane values compared to that diesel. Observation shows that there was a decrease in combustion delay for all tested fuels. In table 8 at full load, the combustion delay periods for diesel, SME100, KB100, and LA100 has given for different combustion geometry [24]. The bar graph for diesel and various biodiesel has shown below in figure 4 for different piston geometries versus ignition delay period (deg) at 1500 rpm.

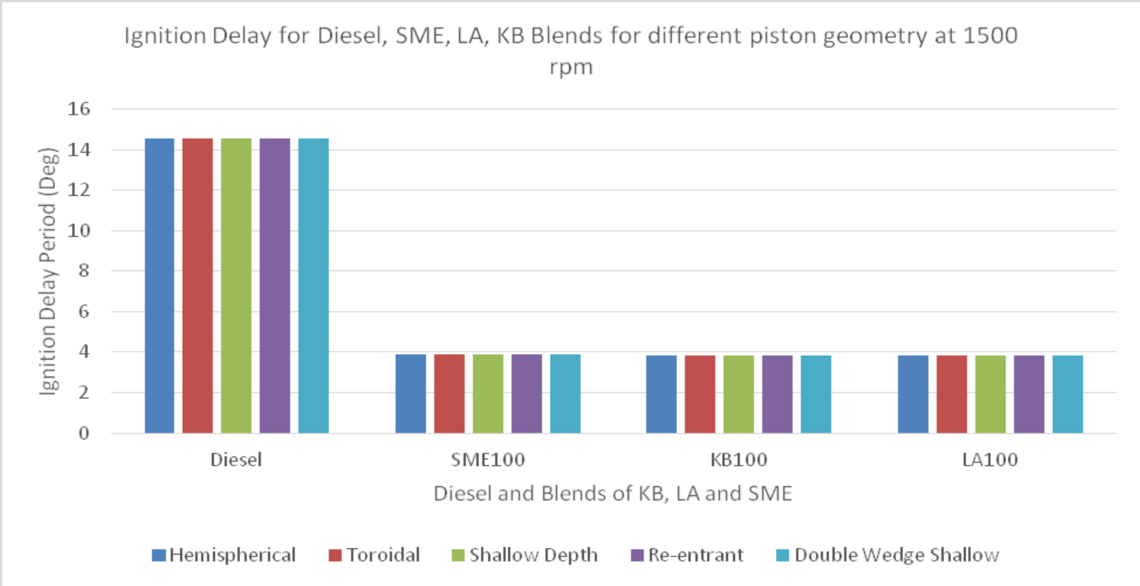


Fig. 4. Ignition Delay for Diesel, SME, LA, KB Blends for different piston geometry at 1500 rpm

Table 8
Ignition Delay Period (Deg)

	Diesel		SME100		KB100		LA100	
	Num.	Exp.	Num.	Exp.	Num.	Exp.	Num.	Exp.
Hemispherical	14.554	14	3.8919	3.9	3.8525	3.91	3.8459	3.79
Toroidal	14.561	14.3	3.8934	3.9	3.8495	3.91	3.8409	3.81
Shallow Depth	14.551	14.25	3.867	3.9	3.8198	3.85	3.8409	3.81
Re-entrant	14.569	15	3.8864	3.95	3.8405	3.85	3.8338	3.82
Double Wedge Shallow	14.567	15	3.8967	3.85	3.8592	3.85	3.844	3.82

4.4 Combustion Duration

It is the duration in which about 90 % of the mixture gets mixed and burns thereafter. It is basically due to turbulent kinetic energy transferred by the piston to combustion space [25]. This parameter helps the designer set the equivalence ratio. The observation through a set of experiments shows that the combustion time of Soybean, Roselle, and Karanja biodiesel is measurably longer compared to diesel oil at full load [24]. The distributed volume of the fuel injection droplets raises due to the low instability and high viscosity of the other biofuels. Low-volatile fuel (biodiesel) with large-size droplets requires more duration to mix with atoms and air (oxygen) and subsequent combustion, which indicates a significant rate of energy in the late phase of combustion [32]. Therefore, by adding a small amount of biodiesel to diesel oil, the combustion time increases. Test at full load conditions for observing combustion duration (CA) for the various fuels has given in Table 9. The bar graph for diesel and various biodiesel has shown below in figure 5 for different piston geometries versus combustion duration (deg) at 1500 rpm that is observed and calculated by diesel-RK software.

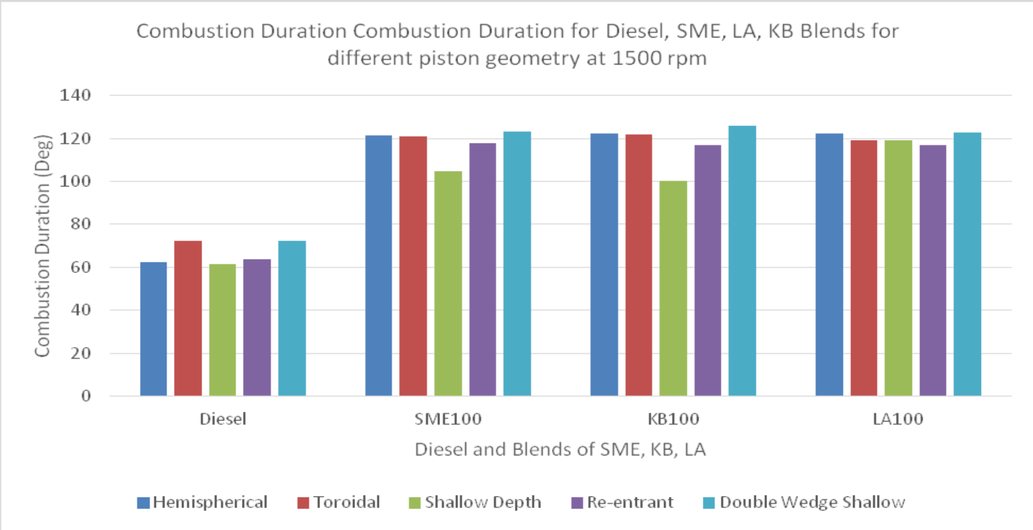


Fig. 5. Combustion Duration for Diesel, SME, LA, KB Blends for different piston geometry at 1500 rpm

Table 9
Combustion Duration (Deg)

	Diesel		SME100		KB100		LA100	
	Num.	Exp.	Num.	Exp.	Num.	Exp.	Num.	Exp.
Hemispherical	62.21	60	121.52	120	122.39	125	122.24	124
Toroidal	72.412	70	120.72	119	121.79	124	119.04	121
Shallow Depth	61.61	60	104.92	105	100.38	103	119.04	121
Re-entrant	63.611	65	117.72	120	116.98	115	117.04	120
Double Wedge Shallow	72.412	70	123.32	120.5	125.79	122	122.84	124

5. Conclusion

Assessment of the technical feasibility of utilizing Soybean, Roselle, and Karanja as optional fuel has been explored by making a thermodynamic analysis based on an experimental and numerical study in this research work. The feasibility of the compressed ignition, the direct injection diesel engine was evaluated at a fixed CR of 17.5 and injection timing of 23.5 ° before TDC, and a constant engine speed of 1500 rpm.

Increased BSFC with an increase in the percentage of biodiesel blends observed during the study. BSFCs found increased by 21.03%, 12.97%, and 12.96% for SME100, KB100, and LA100 with hemispherical, toroidal, and re-entrant combustion geometry while compared with the diesel fuel with the hemispherical combustion chamber. The shallow depth and Double-wedge Shallow depth show a measurable reduction in BSFC value compared to other piston bowl geometries.

Indicated thermal efficiencies were slightly lower for biodiesel and their blends when compared with diesel at full load conditions. Shallow depth geometry recorded a bit high indicated thermal efficiency compared with other bowl geometries in biodiesel run whereas toroidal and double-wedge shallow depth showed marginally improved results for diesel.

The ignition delay period was found to be lower at 100 percent load for biofuels and their different blends whereas their combustion durations phase was reported to be longer compared to diesel fuel. Shallow depth geometry measured shorter ignition delay compared to other geometries with all the varieties of fuel used. Longer combustion duration was measured for double wedge shallow depth geometry whereas shorter duration was recorded for shallow depth geometry.

Also, numerical studies showing nearby results while compared to experimental outcomes show around 95% accurate results, thus diesel-RK gives nearby results and is considered an important tool for saving time, cost, and material.

Through detailed experimentation and its outcomes are considered for comparison and validation purposes still more thorough experimental work is suggested for more focused outcomes.

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