IMPACT OF CONICAL PISTON CROWN EQUIPPED COMPRESSION IGNITION ENGINE ON PERFORMANCE

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ABSTRACT

The significant roles being played by compression ignition engines in the energy and transportation industries is been taken aback due to low thermal efficiency and high emission which remains to be fully addressed despite the continuous redesigning of its combustion chamber. Literature is sparse on the impact of the use of non-cylindrical piston crown in addressing these setbacks. The performance of a conical piston crown equipped compression ignition is investigated numerically in this study. Numerical model was developed from mass balance, momentum, energy and k-ε turbulent equations using finite element technique with an equivalence ratio of 0.5, initial pressure of 10000 N/m^2, and initial temperature of 313 K. Performance parameters; engine power, thermal efficiency, specific fuel consumption and carbon-monoxide emission fraction were estimated for cone and inverted cone-shaped piston crowns with cone base-angle of 25°, 30°, 35°, 40° and 45° respectively. The 40° cone-shaped piston crown equipped engine showed the best improvements in performance. Numerical estimates of the performance parameters for the standard engine were 481.64 J, 30.90 %, 0.1938 kg/kWh and 1.0 respectively, and for 40° cone-shaped piston crown equipped engine was 486.41J, 30.91 %, 0.1918 kg/kWh and 0.9996 respectively. Conical piston crowns equipped compression ignition engine resulted in improved performance.

Keywords: Compression-Ignition Engine, Numerical Model, Piston Crown, Reduced n-Heptane.

INTRODUCTION

Internal combustion engines research emphases are mainly on higher efficiency and lower emission parameters. It is believed that by optimizing the shape the combustion chamber of internal combustion engines, a lot of advantages in terms of performance can be achieved.
Despite some nation’s government and different groups clamour for the adoption of electric powered vehicles, the attendant cost of outright purchase and its maintenance is in the order of ten to 15 times the cost of internal combustion engine powered vehicles. The debate is also still ongoing as to how catastrophic its crash would be because of the high weight of the batteries. Due to the aforementioned, and because of the relatively low cost of the internal combustion engine, its relevance is still been expected to continue for some tens of years to come.

One of the several methods of improving the performance and emission characteristics of the compression ignition engine is by the redesigning of the combustion chamber. The combustion chamber design has effects on the air velocity, thereby dictating the in-cylinder motion.

The pistons of internal combustion engines have been modified in many ways by researchers and its impact on performance and emission characteristics studied. These have been achieved by cutting of grooves on the piston head to get different geometries like the toroidal, spherical, hemispherical, re-entrant, trapezoidal e.t.c. All this piston crown modification still makes use of a cylindrical piston. Literature is sparse on the use of a non-cylindrical piston.
This research is thus focussed on the numerical studies of the performance characteristics of a compression ignition engine fitted with a conical piston crown.

LITERATURE REVIEW

The utilisation of dual fuels in compression ignition engines has been found to be beneficial especially in the reduction of toxic emission and in reducing the dependency on the dwindling reserves of crude oil.

Carbon-monoxide (CO), unburnt hydrocarbon (HC), and NOx emission decrease with the use of dual fuel in homogeneous charge compression ignition engine [1],[2],[3]. Increased brake thermal efficiency, reduced smoke, and brake specific fuel consumption were further reported by[1] and[3].

A large number of hydrocarbons, organic matter, aromatics and other compounds form the practical fuels such as diesel and gasoline. The combustion characteristics of automotive fuels are often represented using blends of hydrocarbons known as primary reference fuels (PRF) because the inclusion of all the components of the practical fuels in the modelling of the combustion process is not practicable. The composition of the automotive fuels depends on the fuel’s source and production history. Basic hydrocarbon fuels such as methane, ethane, propane, n-heptane, iso-octane etc are known as primary reference fuel. [4]

The development of chemical kinetic models for n-alkanes up to sixteen (16) carbon atoms has been carried out. Experimental and modelling work on lower molecular weight surrogate components such as n-decane and n-dodecane which are most relevant to jet fuel surrogates, but are also relevant to diesel surrogates where simulation of the full boiling point range is desired have been undertaken. [5]

Detailed chemistry represents a fundamental pre-requisite for a realistic simulation of a diesel engine combustion process. The use of any reduced mechanism must hence be a true representation of it. A validation of a reduced mechanism of n-dodecane which accurately described its high and low-temperature reactivity for a wide range of operations has been made.[6]

Diesel fuel can be represented with n-Heptane since its properties are close to that of typical diesel fuels, asides being its simplest representative, well-validated mechanisms exist for it; its cetane number (CN~56) is close to that of typical diesel fuels (CN~50) giving an indication of similar ignition characteristics (ignition delay). This eliminates the complexity of modelling several hundreds of components. [7],[8]

The composition of a typical commercial diesel fuel is too complex to model exactly, surrogate fuels which are simpler mixtures that capture the essential performance characteristics of commercial diesel fuel to a sufficient accuracy are therefore usually created; however for the sake of computational tractability; it contains only approximately ten or fewer pure “palette” compounds. [9]

The combustion chamber geometry of internal combustion engines have a major influence on the in-cylinder fluid motion and thus affect the combustion process which ultimately determines its performance and emission characteristics.
Using Jatropha oil-diesel blend as fuel in the study of the effect of combustion chamber design on the performance and emission characteristics of a single cylinder constant speed air-cooled 4-stroke direct injection compression ignition engine, the toroidal combustion chamber led to an improvement of the overall engine performance. [10]

Simulation studies on different combustion chamber geometry of a single cylinder compression ignition engine showed that the use of toroidal and re-entrant geometries showed improved performance in terms of power and thermal efficiency. An improvement in overall efficiency was also observed. [11], [12]

Simulations using KIVA-3V and experiments were conducted at engine speeds of between 1500 rev/min and 4000 rev/min using different modified bowl geometries and re-entrants and it was discovered that bowl geometries have more pronouncing effects on emissions at lower engine regimes, while highly re-entrant bowl showed less sensitivity to engine speed and affects engine emission characteristics. [13]

The performance and emission characteristics (HC, smoke, and CO) improved with the use of toroidal shaped combustion chamber operated on biodiesel fuels and least for trapezoidal shaped chambers however in comparison to using diesel in a traditional compression ignition engine it still fell below. [14], [15]

Grooved pistons operated on the blended fuel showed better performance in terms of brake thermal efficiency, specific fuel consumption., CO, HC and smoke emissions in comparison with the diesel operated engine without tangential groves [16], and improved performance occurs when the comparison was between a diesel fuel standard engine and a diesel operated grooved piston. [17]

METHODOLOGY

The modelled engine is tailored towards the Kirloskar TV1 engine, fitted with cone-shaped and inverted cone-shaped piston crowns having inclination angles of 25°, 30°, 35°, 40° and 45° while maintaining the engine's standard compression ratio. COMSOL Multiphysics 5.0 was used to model the engine in 2-D axisymmetric coordinates.

The modelled engine parameters are as depicted in Table 1 below;

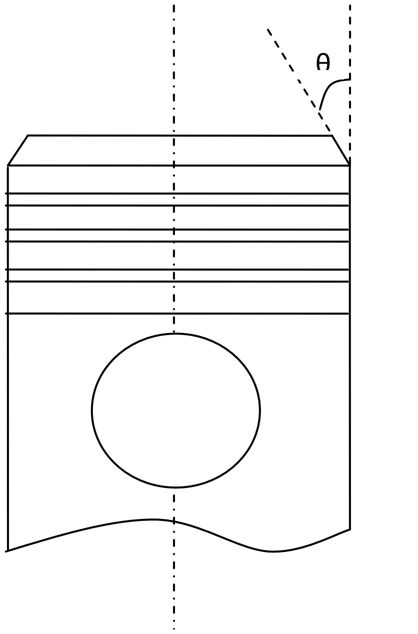
Table 1: Modelled engine parameters

Parameters	Variable	Value
Initial Temperature (K)	T_o	313
Initial Pressure (N/m^2)	P_o	1e5
Bore Diameter (m)	D	0.0875
Stroke (m)	S	0.11
Connecting rod length (m)	L	0.238
Crank arm length (m)	La	0.055
Engine speed (RPM)	N	1500
Compression ratio	CR	17.5:1
Equivalence ratio	ER	0.5

ISSN 2056-5860

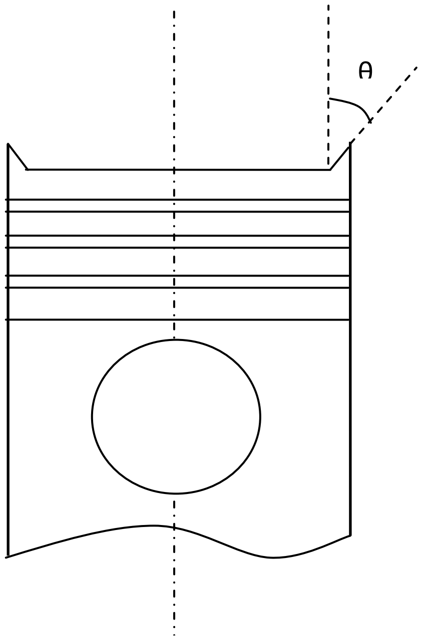
The initial mole fractions of Oxygen was 0.20801, Nitrogen 0.78253 and n-Heptane 0.0094589 while the initial concentration is 7.9931 mol/m^3 for Oxygen, 30.069 mol/m^3 for Nitrogen and 0.36347 mol/m^3 for n-Heptane.

The cone-shaped and inverted cone-shaped piston crown geometry is as depicted in Figures 1 and 2 below;



Θ is the inclination angle

Figure 1. Cone-shaped piston crown



Θ is the inclination angle

Figure 2. Inverted cone-shaped piston crown

The combustion of fuel in a compression ignition engine is governed by the Navier-Stokes equation and the heat equation to model the thermo-fluid activity undergone in the chamber;

$$\rho \frac{\partial u}{\partial t} + \rho\, u.\nabla\, u = -\nabla.\rho + \nabla.\left[\mu\left(\nabla u + (\nabla u)^T\right) - \frac{2}{3}\mu\nabla(\nabla.u)\right] + E \quad (1)$$

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p\, u.\nabla T = \nabla.(k\nabla T) + q \quad (2)$$

Where ρ is the density
E is the source term
μ is the kinematic viscosity
c_p is the specific heat capacity at constant pressure
q is the heat energy generated per unit volume
To ensure its convergence, the governing equation was simplified by averaging it using the Reynolds Averaged Navier-Stokes (RANS) employing the k-ε turbulent model to close it.

$$\rho \frac{\partial u}{\partial t} + \rho\, u.\nabla\, u = \nabla.\left[-\rho 2I + (\mu + \mu_T)\left(\nabla u + (\nabla u)^T\right) - \frac{2}{3}(\mu + \mu_T)\nabla(\nabla.u)I - \frac{2}{3}\rho k I\right] + E \quad (3)$$

The 'k' equation is;

$$\rho \frac{\partial k}{\partial t} + \rho(u.\nabla)k = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\nabla k\right] + p_k - p_\varepsilon \quad (4)$$

And the 'ε' equation is;

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho(u.\nabla)\varepsilon = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\nabla \varepsilon\right] + C_{\varepsilon 1}\frac{\varepsilon}{k}P_k - C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k} \quad (5)$$

Where

$$\mu_T = \rho C_\mu \frac{k^2}{\varepsilon} \quad (6)$$

$$P_k = \mu_T\left[\nabla u : \left(\nabla u + (\nabla u)^T\right) - \frac{2}{3}(\nabla.u)^2\right] - \frac{2}{3}\rho k \nabla.u \quad (7)$$

The turbulent model parameters are spelled out as depicted in Table 2 below.

Table 2: Turbulent parameters

Parameters	Value
$C_{\varepsilon 1}$	1.44
$C_{\varepsilon 2}$	1.92
C_μ	0.09
σ_k	1
σ_ε	1.3
k_v	0.41
B	5.2

The removed volume of the piston crown material for the geometry modification which was added back to the piston crown top to ensure constant compression ratio for some of the simulated cases was calculated using the relation;

$$V_r = \pi R^2 h_o - \left(\frac{\pi R^2 h}{3} - \frac{\pi}{3}(R - Ro)^2(h - ho)\right. \quad (8)$$

Where V_{rem} is the volume of material removed for modification
R is the radius of the piston
Ro is the radius of the tapered section
ho is the height of the tapered section and
h is the height of the cone formed by tapering the piston by an angle.

$$h = \frac{R}{2tan\theta} \quad (9)$$

Where θ is the angle of taper

The increase in piston crown height was calculated using the formula:

$$V_{cylinder} = \pi R^2 \quad (10)$$

$$Height\ increase = \frac{V_r}{\pi R^2} \quad (11)$$

The performance indicators of concern to this study; power, thermal efficiency, and specific fuel consumption respectively were calculated using the following relations;

$$P = \frac{P_m LAN}{n_c} \quad (12)$$

$$\eta_{Th} = \frac{W_{net}}{Q_{in}} \quad (13)$$

$$SFC = \frac{\rho_{atms}}{(A/F)P_m} \quad (14)$$

W_{net} is the network output from the engine
Q_{in} is the heat energy generated by the combustion of the fuel-air mixture
V_d is the swept volume
A is the area of the cylinder
N is the number of revolutions per minute
n_c is the number of cycles required to make a complete revolution. (For a four-stroke engine, n_c = 2)
ρ_{atms} is the atmospheric air density
(A/F) is the air-fuel ratio.
ρ_{atms} used for the computation was 1.1886 kg/m^3, while (A/F) based on an equivalence ratio of 0.5 was 30.33.

RESULTS

Using reduced n-Heptane as the diesel fuel representative, the performance of a compression ignition engine tailored towards the Kirloskar TV1 engine was modelled and referred to in this study as 0° inclination angle. The model was adopted for use with cone-shaped and inverted cone-shaped piston crowns having the stated inclination angles.
The derived net work and heat energy obtained from the fuel combustion in the simulations for the cone-shaped and inverted cone-shaped piston crowns are depicted in Tables 3 and 4 respectively.

Table 3: Cone-shaped piston crown output

Inclination Angle	Net Work (J)	Heat Energy (J)
0°	481.64	1558.62
25°	483.25	1564.90
30°	480.93	1560.00
35°	483.97	1564.74
40°	486.41	1573.88
45°	483.05	1564.94

Table 4: Inverted cone-shaped piston crown output

Inclination Angle	Net Work (J)	Heat Energy (J)
0°	481.64	1558.62
25°	480.64	1557.74
30°	480.86	1558.52
35°	480.06	1556.82
40°	483.95	1564.74
45°	483.81	1564.68

The power generated by the engine "P" and the mean effective pressure "Pm" values were determined using Eq. (12). The derived mean effective and the corresponding plots for the cone and inverted cone-shaped piston crowns are depicted in Table 5 and Fig. 3 respectively.

Table 5: Mean effective pressure of simulated engine

Inclination Angle	CSPC ($*10^5 N/m^2$)	ICSPC ($*10^5 N/m^2$)
0°	7.282	7.282
25°	7.306	7.266
30°	7.271	7.270
35°	7.317	7.258
40°	7.354	7.317
45°	7.303	7.314

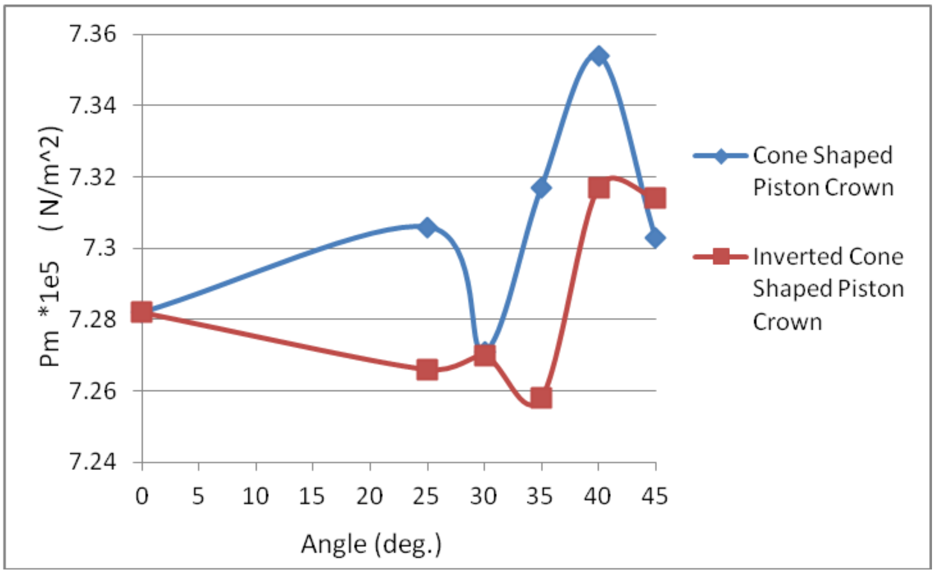


Figure 3. Mean effective pressure plots of simulated engine

The generated power estimates and the corresponding plots are as depicted below in Table 6 and Figure 4 respectively.

Table 6: Generated power estimates of the simulated engine

Inclination Angle	**CSPC (KW)**	**ICSPC (KW)**
0°	6.021	6.021
25°	6.041	6.008
30°	6.013	6.011
35°	6.050	6.001
40°	6.080	6.050
45°	6.038	6.047

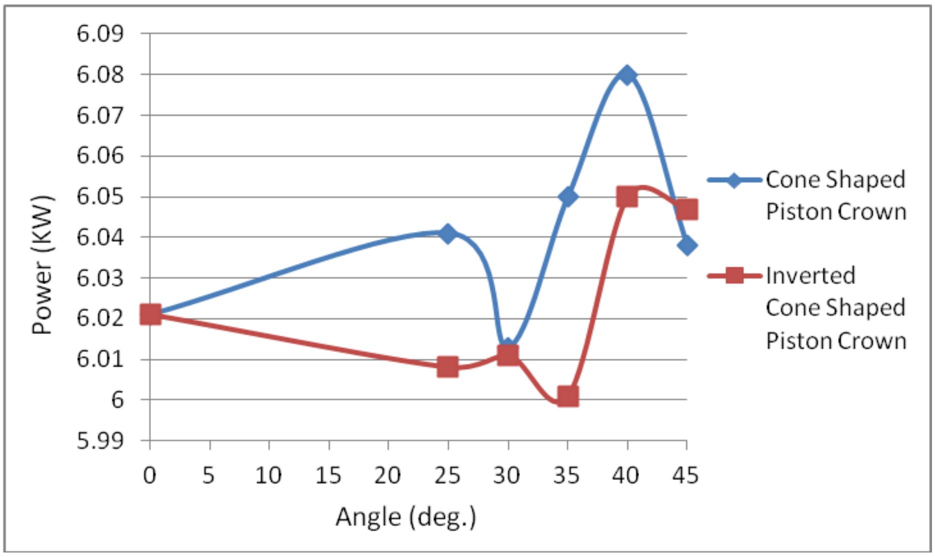


Figure 4. Power plots of simulated engine

The thermal efficiencies of the modelled engine using the modified piston crowns were calculated using Eq. (13), and the obtained values are as shown in Figure 5.

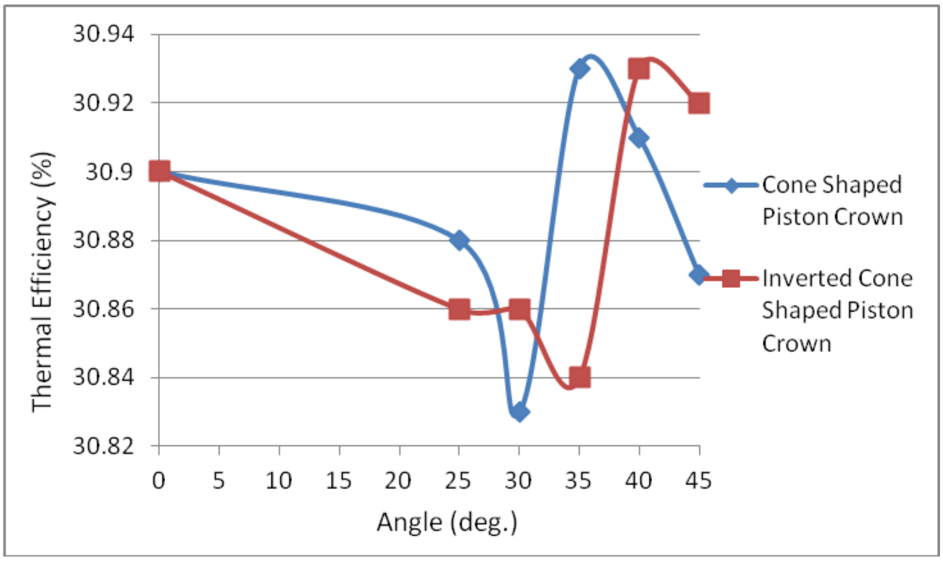


Figure 5. Thermal efficiency plots of simulated engine

The specific fuel consumption of the modelled engine fitted with the modified piston crown was derived using Eq. (14). The plot below in Figure 6 represents the specific fuel consumption of the modelled engine using the cone and inverted cone-shaped piston crowns of the stated inclination angles respectively.

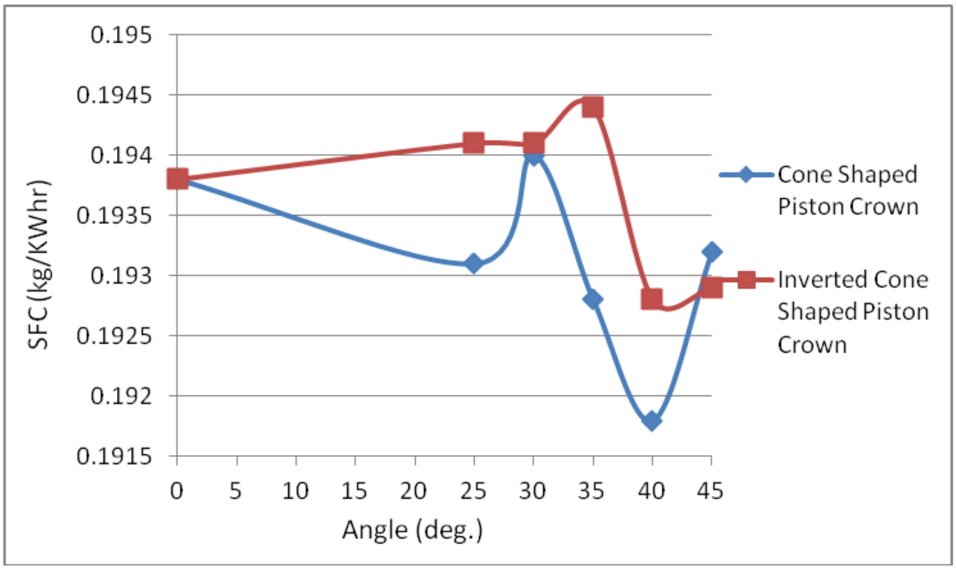


Figure 6. Specific fuel consumption plots of simulated engine

The CO emission values relative to that gotten for the unmodified piston crown fitted engine is plotted as shown in Figure 7.

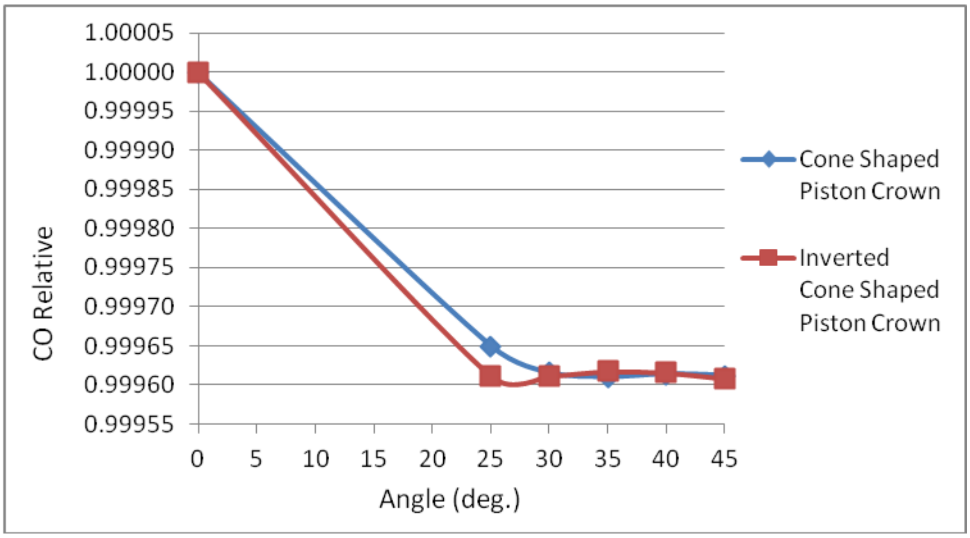


Figure 7. Carbon-monoxide relative value plots of simulated engine

DISCUSSION

The cone-shaped piston crown fitted engine showed improved performance over the unmodified piston crown fitted engine in mean effective pressure, power and specific fuel consumption for all inclination angles except for 30°. It also fared better in thermal efficiency at an inclination angle of 35° and 40°, and for all inclination angles, it had lower CO emission values.

The inverted cone-shaped piston crown fitted engine showed improved performance over the unmodified piston crown fitted engine in mean effective pressure, power, thermal efficiencies and specific fuel consumption for inclination angles of 40° and 45°. It also fared better in terms of CO emissions, even as it was found to have recorded the least CO emission value of all the simulated cases at an inclination angle of 25°.

The improvement in performance of the modified piston crown equipped engine can be attributed to the improved fuel combustion nearing complete combustion in the combustion chamber resulting from better in-cylinder fluid motion which might have been caused by increased turbulence as a result of the combustion chamber redesign due to the piston modification.

Results Validation

The weighted values for the engine simulation using the cone and inverted-cone shaped piston crowns with inclination angles of 25°, 30°, 35°, 40° and 45° is shown in Table 7 below.

Table 7: Weighted Values

Performance Parameters	**CSPC and ICSPC**
Power (KW)	6.032±0.023
Thermal Efficiency (%)	30.89±0.03
Specific Fuel Consumption (kg/KWh)	0.1934±0.0008
Mean Effective Pressure (10^5 N/m^2)	7.2950±0.0282
Torque (N/m)	38.40±1.5
Carbon-monoxide ratio	0.999681±1.56E-4

The numerical estimates derived from the simulation compared favourably with experimental results obtained from some literature. The estimated and literature value for the engine power have a percentage difference of 13.64% [18], [19], [20] [21]. The estimated thermal efficiency and values for literature have values of 30.75±0.21 and a percentage difference of 0.97% [20] a percentage difference of 1.29% and 2.83% respectively [18,] [21] The estimated specific fuel consumption and the values from literature at a brake power of 5.2KW are 0.22±0.03 with a percentage difference of 19.25%. [21]. This is depicted in Table 8 below.

Table 4.8: Numerical model estimates validation against literature

Performance Parameter	**Sundarraj**	**Sreedhar and Prassad**	**Sundarraj and Saravannan**	**Velappan and Sivaprakasam**
Power (KW)	Mean values: 5.61±0.58 % Difference: 13.64	Mean values: 5.61±0.58 % Difference: 13.64	Mean values: 5.61±0.58 % Difference: 13.64	Mean values: 5.61±0.58 % Difference: 13.64
Thermal Efficiency (%)	Mean values: 31.35±0.64 % Difference: 1.29	- -	Mean values: 30.75±0.21 % Difference: 0.97	- -
Brake Specific Fuel Consumption (kg/kWh)	- -	- -	- -	Mean values: 0.22±0.03 % Difference: 19.25

CONCLUSIONS

- The study showed that the result of the numerical simulation using n-heptane as fuel was a true representation of the real compression ignition engine process fuel with AGO.

ISSN 2056-5860

- The performance of a compression ignition engine can be enhanced with the use of conical piston crown with the appropriate angle of inclination.
- The cone-shaped piston crown with an inclination angle of 40° showed the best improvement in performance over the other selected inclination angles and the cylindrical piston type; the power improved by 1%, specific fuel consumption witnessed an improvement of 1.04%, thermal efficiency increased by 0.01% and a 0.04% reduction in CO emission.

ACKNOWLEDGEMENTS

This is to acknowledge COMSOL Multiphysics for making available some of its tutorials online; it really helped in understanding how to use the software and basic equations.

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