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ORIGINAL RESEARCH ARTICLE

MODELLING THE PERFORMANCE CHARACTERISTICS OF FOUR STROKE INTERNAL COMBUSTION RENAULT ENGINE CYCLE USING MATLAB SIMULATION TOOL

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ABSTRACT

The trends in Internal Combustion Engine (ICE) cycle is gradually changing due to the quest for optimum performance, efficiency and zero emission level which are often achieved through a series of experimental procedures. To reduce the huge experimental cost, time and resources, MATLAB 2018b was employed in the modelling and simulation process of a four stroke internal combustion Renault engine (Mercedes-Benz 250SE) W108 model. The displaced volume was $4.65x10^{-4}$ mm^3 ($4.65x10^{-7}$ cm^3) while the minimum volume occupied by the charge was $0.5x10^{-4}$ mm^3 ($5x10^{-8}$). Moreover, the maximum velocity occurred at a crank angle of 72°, having a value of 19.65 m/s while the minimum velocity occurred at a crank angle of 288° with a value of -19.65 m/s. However, maximum cylinder pressure of 28 bar was observed at crank angle of 20°, followed by gradual decline up to 0.2 bar at subsequent crank angles of 100, 200, 300°. The results showed that maximum peak pressure between simulated data and experimental data were 5347 and 5320 KPa, while maximum spark pressure in cylinder before combustion between simulated data and experimental data were 1849 and 1730 KPa. In addition, highest crank angle at maximum pressure for simulated and experimental data were 30 and 22°. It has been established that developing mathematical models to simulate IC engine operation cycles can help offset the cost, time and resources required for experimental set up, testing and data acquisition from the engine.

1.0 Introduction

Internal combustion (IC) engines consist of various types which may include; two stroke, four stroke and six stroke engine. Four stroke petrol engine finds its application in mechanical components such as cars, airplanes, motorcycles etc. (Akpobi and Oboh, 2007). In four stroke IC engines, reciprocating motion of the piston is converted into rotary motion of the crankshaft via a connecting rod (Georgiev, 2011). The combustion of air-fuel mixture in internal combustion engine occurs in the combustion chamber of the engine where mechanical work is achieved (Ikpe et al., 2016a; Ikpe et al., 2016b). The heat generated in the process reacts with the mixture inside the cylinder to form a hot in-cylinder pressurize gases that cause reciprocating movement of the piston to be translated into rotating motion of the crankshaft (Soda, 2013; Ikpe and Owunna, 2020). The working principles of four stroke IC engine mostly occur in four stages namely: 1st stroke (suction stroke), 2nd stroke (compression stroke), 3rd stroke (power stroke) and 4th stroke (exhaust stroke).

During the first stroke, the inlet valve opens due to cranking and pure air is drawn into the cylinder through the inlet valve and piston moves from top dead center (TDC) to bottom dead center (BDC) while the exhaust valve is closed. For the second stroke, the piston moves from bottom dead center (BDC) to top dead center (TDC). At this stage, both the inlet and exhaust valves are closed and the air drawn into the cylinder during suction stroke is compressed as the piston ascends. The compression ratio which varies from 14:1-22:1 occurs under high temperature and pressure ranging from 30-45kg/cm^2 and 650-800°C. As the ascending piston approaches the end of the TDC, liquid hydrocarbon fuel is sprayed into the combustion chamber which ignites and combusts with the compressed air. Both valves remain closed and the piston moves from TDC to BDC in the 3rd stroke, as the hot product of combustion consisting mainly of carbon II oxide and nitrogen in the compressed air expands and forces the piston downward (Palanivendhan et al., 2016). This causes the cylinder pressure to fall from its maximum (47-55 kg/cm^2) to minimum (about 3.5-5 kg/cm^2) near towards the end of the stroke. During this stroke, power is gained by the engine for mechanical work. In the 4th stroke, the exhaust valve opens while the piston travels upwards and the exhaust gases are swept out via the open valve by the piston traveling upward the cylinder. Several studies have been carried out by different researchers in recent times to investigate the performance of IC engines in different areas of application.

Partel and Rathod (2013) studied the performance of high-speed supercharger in four stroke internal combustion engine and found that, 30-35% increase in Brake Mean Effective Pressure (BMEP) is achievable with supercharged engines compared to naturally aspirated engines. It was also found that the engine power increases with increase in supercharge pressure.

Davis et al. (2015) experimentally investigated the performance parameters (Break Specific Fuel Consumption (BSFC), Total Fuel Consumption (TFC), break thermal efficiency and mechanical efficiency of a single cylinder four stroke spark ignition engine by preheating the intake air using a thermocouple module. While the mechanical efficiency was observed to increase from 17.79-20.12% at part load condition, the BSFC and TFC were found to decrease with preheated air from 0.981-0.877 Kg/kWhr at full load and from 0.430-0.380 Kg/hr at full load condition, indicating a reduction in fuel consumption.

Karabulut and Ersoy (2012) investigated the vibration aspect of a two-cylinder four-stroke internal combustion engine and found the flywheel moment of inertia as the major component to be minimized in order to reduce the angular speed fluctuation of the crankshaft. The in-cylinder gas pressure and the piston mass were also found to be the major factors responsible for the rotational and translational vibration of the engine. The findings also revealed that the vertical vibrations of the engine could be minimized by means of subjecting the crankshaft to counter weight.

Islam et al. (2016) employed k-e turbulent model in ANSYS FLUENT software in modelling and simulating spark ignition premixed combustion via variations in pressure, temperature, velocity and swirl ratio in four stroke single cylinder. At 1500 rpm crank angle rotational speed using gasoline as fuel, the maximum pressure after compression of air-fuel mixture was 1.3 MPa. The maximum temperature as a result of expansion was 710 K while approximately 490 W brake power was achieved.

Iliev and Hadjiev (2012) investigated the effects of engine speed on the performance of four-stroke direct injection gasoline engine using Computational Fluid Dynamics (CFDs). The results showed highest cylinder pressure of 57.76 bars at engine speed of 3000 rpm and lowest cylinder pressure of 45.1 bars at engine speed of 1000 rpm. In addition, highest pumping Mean Effective Pressure (MEP) of 0.0357 bar at 1000 rpm engine speed and minimum of 1.7497 bar at 6000 rpm where obtained. However, highest brake power of 63.76 Kw at 4500 rpm and indicator power of 77.09 Kw at 4500 rpm speed were obtained for the four-stroke port injection gasoline engine modelling.

The working principles of Internal Combustion Engine over the years have improved significantly due to several reforms and developments incorporated by experts in automotive industries. However, automobile industries are continuously driven by the cravings for effective performance, better fuel efficiency and less emissions (Ikpe et al., 2021), which are mostly determined by experimental methods. Although numerical methods are also employed to determine the performance of an IC engine as stated in the aforementioned literature, it is possible that validating the numerical with experimental results could yield more accurate and useful data for further studies in this area.

Hence, MATLAB (Matrix Laboratory) software which is a numerical modelling tool, was employed in modelling the performance characteristics of four stroke internal combustion Renault Engine cycle. It is a programming tool that is designed for easy, simple as well as complex calculation and analysis, and its versatility in solving numerically related problems cuts across several field of studies.

2. Materials and Methods

The experimental data was collected from a Renault production engine (Mercedes-Benz 250SE) W108 model at different engine speeds and inlet pressures. The data includes P_{spark}, P_{peak} and $\theta_{peak\ pressure}$. The geometric parameters of the engine were entered into MATLAB 2018b software to predict the pressure and other performance characteristics of the engine. The geometric data of the engine according to (Ayub et al., 2015) are presented in Table 1. The angle of spark firing and burning duration angle were not explicitly known but their approximate values were obtained from the investigation by Kuo (1996) who also used this experimental data for comparison.

Table 1: Geometric parameters of Mercedes-Benz 250SE Renault engine

Number of cylinders	6
Number of valves per cylinder	2
Displacement (cc)	2500
Bore x Stroke (mm)	82 x 78.8
Compression ratio	9.8:1
Ratio of connecting rod length to crank radius	3.2

Intake valve open before TDC (degree)	11°
Intake valve close after BDC (degree)	53°
Maximum valve lift (mm)	8
Inlet valve diameter (mm)	41.2

MATLAB software was used for modeling and simulation of the four (4) stroke IC engine model. MATLAB codes were written to provide access to matrix and data structures provided by the LINPACK (Linear system package) and EISPACK (Eigen system package) projects. MATLAB is a high-performance language which is capable of performing technical computing. It combines visualization, computation and programming environment. The MATLAB software provides a modern programming language environment as well as complex data structures containing in-built editing and debugging tools. MATLAB software also support OOP (Object Oriented Programming). The simulation of the IC engine model was done with MATLAB 2018b software. The model is a four stroke internal combustion engine with an in-line configuration. The engine is naturally aspirated with 6 cylinders, and uses petrol as fuel. The engine features a single overhead camshaft, 2 valves per cylinder. Data employed for the simulation is presented in Table 2, and was obtained from a technical report on Renault production engine (Mercedes-Benz 250SE) W108 model (Ayub et al., 2015).

Table 2: Data employed for the simulation

Parameters	**Value**
Intake Manifold Pressure (KPa)	101325
Specific heat ratio	1.4
Cylinder temperature during Intake(K)	300
Engine speed(rpm)	3000
Range of θ_s	-30° to +30° with respect to TDC
Range of θ_b	30° to 120°
Heating value of Gasoline	44000KJ/Kg
Air to fuel ratio	14.6/1

2.1. MATLAB Codes for the IC Engine Modelling

The codes used in the experiment were generated by considering the processes that makes up the full engine. For a four stroke engine, four processes were involved; namely: intake stroke, compression stroke, power stroke and exhaust stroke. Developing mathematical models to simulate the IC engine helped in testing certain parameters of the engine rather than performing an experimental set up to see the effects of such parameters on the engine. The mathematical model does not give accurate or perfect results but is helpful in speeding up the engine development process before producing a real prototype. The IC Engine modelling and simulation in this study was divided into two categories namely: The first was fluid-dynamic based model and the second was a thermodynamic-based model.

Fluid-dynamic based model which is also called the multidimensional model because its formation is primarily due to the conservation of mass and the state variables of the engine at certain locations in the engine. That is, these variables are independent of time (time independent variables) and are tied to space variables, of which the governing equations are more or less ordinary differential equations or partial differential equations.

The thermodynamic-based models are function of time or crank angle and this model is based on the laws of thermodynamics. In these models, the first law of thermodynamics was applied to the cylinder and the charge in it. The system is treated as an open system during the intake and the exhaust stroke while the compression and power stroke was treated as a close system.

The thermodynamic model used in this case was zero-dimensional because it only depends on time or crank angle. The resulting equations from this model are ordinary differential equations instead of partial differential equations. In the engine models developed in this study, the combustion model was simulated according to Weibe's function. However, the geometry and flame propagation were not considered.

2.2 Modelling of the IC Engine Strokes

The intake stroke of the four stroke engine is modelled as a one dimensional isentropic flow through a restriction, the restriction being the poppet valve of the engine. The equation governing this process is given by Equation 1.

$$\frac{dm}{d\theta} = \frac{C_D A_R P_O}{6N(RT_O)^{0.5}} \left(\frac{P_T}{P_O}\right)^{\frac{1}{k}} \left\{\frac{2k}{k-1}\left(1-\left(\frac{P_T}{P_O}\right)^{\frac{k}{k-1}}\right)\right\}^{0.5} \quad (1)$$

Where, A_R is the reference area given by Equation 2, P_T is the cylinder pressure, L_v Is the valve lift, C_D is the Discharge coefficient, P_O and T_O are the intake manifold pressure and temperature respectively.

$$A_R = \pi D_v L_v \quad (2)$$

The above differential equation was then solved using the inbuilt MATLAB differential equation solver, which is the code45 that solves differential equation based of the Runge Kutta fourth order differential equation. After solving the equation, the pressure for the intake model was computed using the Equation 3.

$$P_{intake} = \frac{mRT}{v} + (pressure\ due\ to\ change\ in\ cylinder\ volume) \quad (3)$$

The mass, m is gotten by solving the differential equation above while T is the cylinder temperature and V being the volume at various crank angle. The flow chart for the intake model is presented in Figure 1.

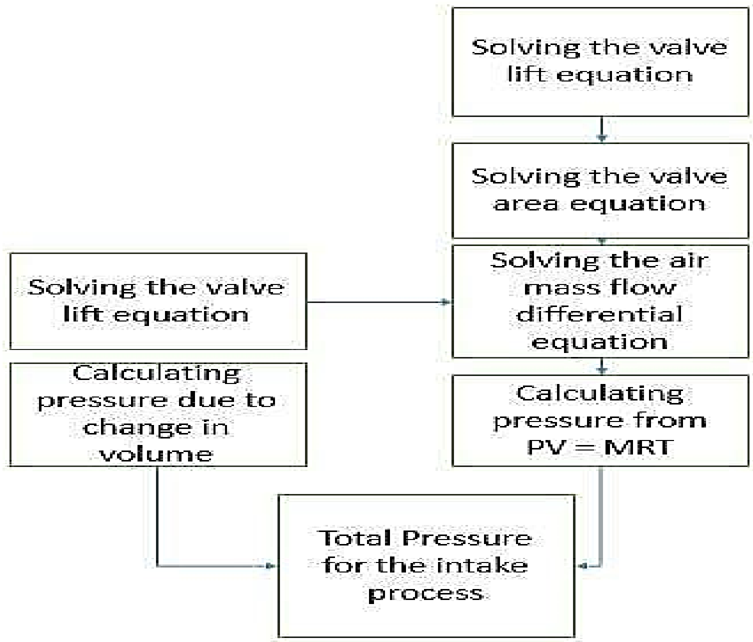


Figure 1: Flow chart for the intake process

The drag coefficient is given by the correlation in Equation 4, the valve lift is given by equation 5 while the crank angle is given by Equation 6.

$$C_D = 107.78\left(\frac{L_v}{D_v}\right)^4 - 77.204\left(\frac{L_v}{D_v}\right)^3 + 14.1\left(\frac{L_v}{D_v}\right)^2 - 1.01\left(\frac{L_v}{D_v}\right)^1 + 0.6687 \quad (4)$$

$$L_v = \frac{L_{vmax}\left(1+cos(\emptyset)\right)}{2} \quad (5)$$

$$\emptyset = \frac{\pi(IVO-IVC+2\theta+540)}{IVO-IVC+180} \quad (6)$$

where: L_V is the valve lift function, L_{vmax} is the maximum valve lift, θ is the crank angle, IVO is the inlet valve open angle before Top Dead Center, IVC is the inlet valve close angle after Bottom Dead Center. For our work, it was assumed that the intake starts at 0° and end at 180°.

2.3. Modelling the compression and the expansion process

The compression and expansion process was modelled according to Equation 7.

$$\frac{dp}{d\theta} = -k\frac{p}{v}\frac{dv}{d\theta} + (k-1)\frac{Q_T}{v}\frac{dx}{d\theta} - k\frac{c}{w} \quad (7)$$

This differential equation is solved to yield the pressure as a function of crank angle, where Q_T is the heat supplied from the combusted fuel, c is the blow-by constant, *w* is angular velocity. The temperature for this process can also be calculated using Equation 8.

$$\frac{dT}{dQ} = T(k-1)\left[\left(\frac{1}{pv}\right)\left(\frac{dQ}{d\theta}\right) - \left(\frac{1}{v}\right)\left(\frac{dv}{d\theta}\right)\right] \quad (8)$$

This differential equation is solved to yield the temperature as a function of crank angle. The exhaust process was simply modelled as a constant pressure process.

3. Results and Discussion

Experimental and simulated results obtained from the four stroke IC engine are presented in this section. Table 3 shows the experimental results obtained from the four stroke Renault engine. Similarly, the results obtained from running the code in MATLAB environment for the four stroke engine are presented in Figures 2-10.

Table 3: Experimental result obtained from four strokes Renault engine

Cases	N(rpm)	P_{inlet}	Spark Advance θs	Burn Duration θb	P_{peak}	$\theta_{peakpressure}$	P_{spark}
1.	2000	63.7	-18	71	3050	16 ATDC	890
2.	2000	102.2	-10	65	3780	22 ATDC	1730
3.	4000	45	-27	75	2720	11 ATDC	546
4.	4000	100.2	-23	78	5320	20 ATDC	1420

Figure 2 shows the plot of volume occupied by the charge in the cylinder vs the crank angle. The results clearly show that the plot follows a sinusoidal path, having a maximum volume at 180° or (180 + 360n) where n can be 0, 1, 2... and minimum value at 0° or (360n) where n can be 0, 1, 2, 3… The maximum volume occupied by the charge is also known as the displacement volume. Interestingly, the displaced volume is 4.65×10^{-4} mm^3 (4.65×10^{-7} cm^3) while the minimum volume occupied by the charge is 0.5×10^{-4} mm^3 (5×10^{-8}). This volume is also known as the clearance volume which indicates the volume remaining above the piston of an IC engine when it reaches Top Dead Center (TDC). The variation in volume of the charge also allows for variation in the density of the charge. With the minimum density occurring at the BDC and the maximum density occurring at the TDC during the compression stroke. The change in the density of the charge is part of what causes the pressure to rise during the compression stroke. As the piston moves upward to compress the charge, it squeezes the charge to occupy a small space which invariably increases the density which in turn increases pressure and temperature (Owunna and Ikpe, 2021).

The density of the charge should be carefully monitored so that auto ignition temperature will not be reached during compression of the charge to clearance volume. The auto ignition temperature of the charge (fuel-air mixture) varies from 520K to 553K.

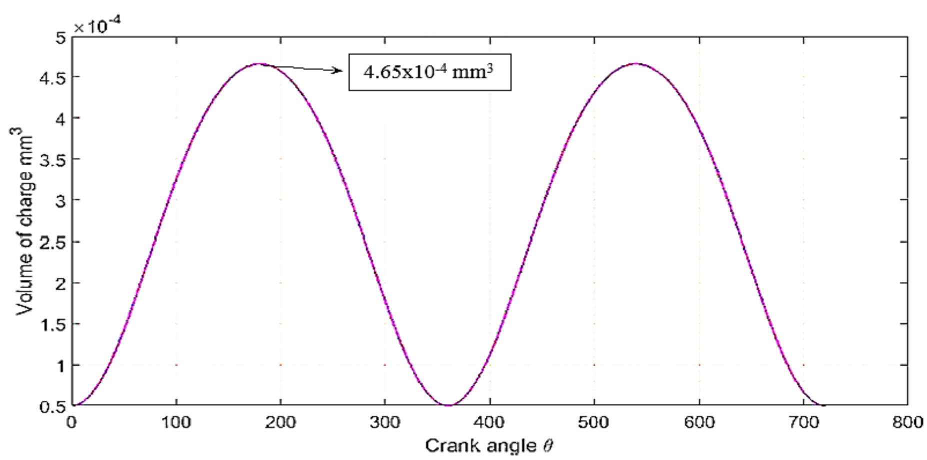


Figure 2: Plot of volume occupied by charge in cylinder against crank angle

Figure 3 shows the piston velocity vs crank angle for a rotating speed of 4000 rpm of the crank shaft. It can be observed that the maximum positive velocity occurs at a crank angle of 72° or corresponding crank angle (72 + 360n). Moreover, the maximum negative velocity occurs at a position of 288° or corresponding crank angle (288 + 360n).

The position of zero velocity occurs at turning points of crank angle 0° or multiples of 180°. Careful observation shows that from BDC, the piston accelerates from the bottom of the cylinder upwards till it reaches a maximum velocity of 19.65 m/s at a position of 72°. Still moving upwards, the piston now decelerates from 19.65 m/s to zero velocity at 180°, which is the BDC of the engine. The same process is repeated as the piston travels upward the cylinder (from the BDC to the TDC).

Air-fuel charge is introduced into the cylinder as the piston travels from TDC to BDC. As a result of the piston movement from TDC to BDC, the in-cylinder pressure is reduced to a value in the range of atmospheric pressure (760 mm Hg) (Eastop and McConkey, 1993). As the piston approaches the bottom of the cylinder, air is channeled through the carburettor, of which a metered amount of fuel is added to the air. The inlet valve closes at this point, but this only occur when the piston has traveled midway along the return stroke. The higher the piston velocity, the higher the output power delivered by the piston. Furthermore, the higher the stroke ratio, the lower the piston velocity and the lower the stroke ratio, the higher the piston velocity at the same crankshaft speed

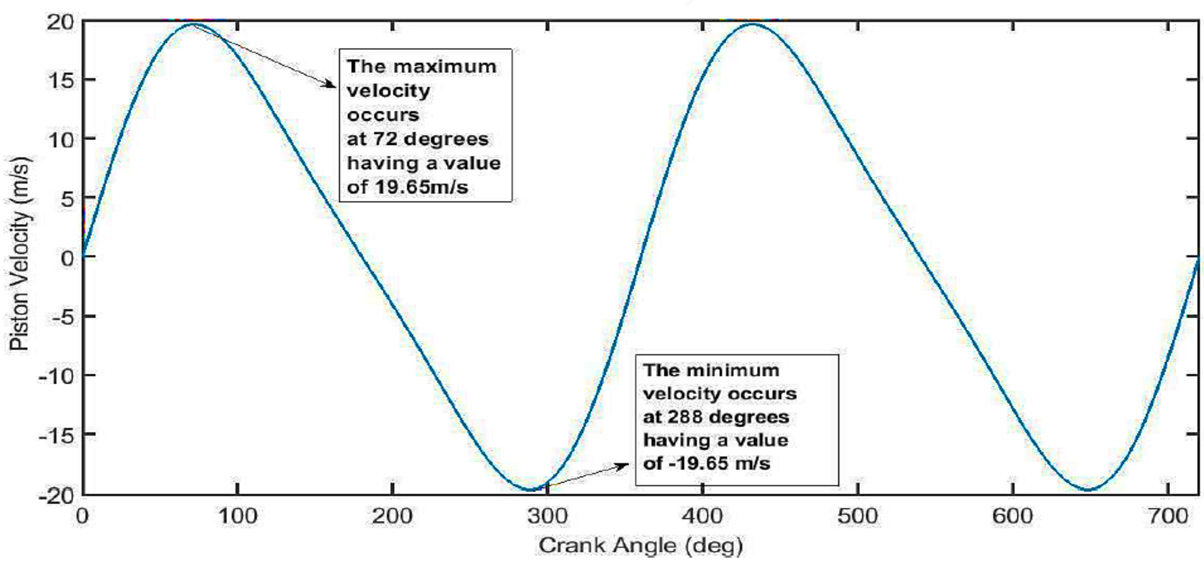


Figure 3: Plots of piston velocity against crank angle

Figure 4 shows the plot of pressure of the charge against crank angle. This plot vital in IC engine application as it helps auto mechanics to know the pressure distribution in the cylinder, position of maximum pressure. Considering different piston crank angles, Mahmound and Ahmed (2017) in similar study plotted an IC engine cylinder pressure against crank angle, and the plot maintained almost the same trend as that obtained in this study. This implies that at crank angle of 0 degree, the piston traveled from BDC to TDC where it attended a peak pressure of 29 bar. In this case, the air-fuel mixture available when the piston is at the bottom of the cylinder is compressed as the piston travels upward towards TDC. In addition, Figure 4 indicates that engine pressure reduces as the crank angle increases. This implies that at increasing crank angles, available space in the cylinder decreases, and the amount of air-fuel mixture compressed by the engine is proportional to the available space.

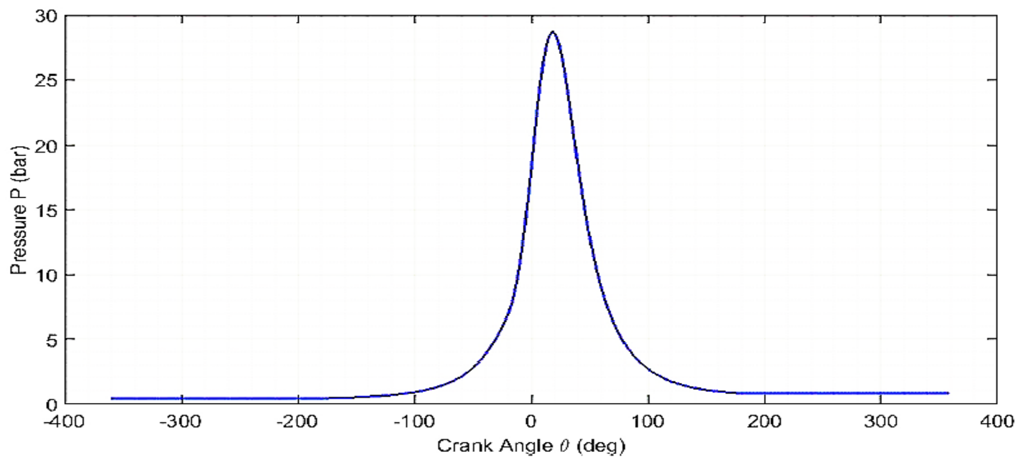


Figure 4: Plot of cylinder pressure against crank angle

Figure 5 represent the plot of pressure against volume occupied by the charge. The pressure volume diagram shows each of the process, here, the straight line at the base represent the exhaust process. This is because during the modelling, assumption was made that the exhaust phase is a constant pressure process. The area enclosed by the pressure volume diagram represent the work output delivered by the engine during one complete thermodynamic cycle.

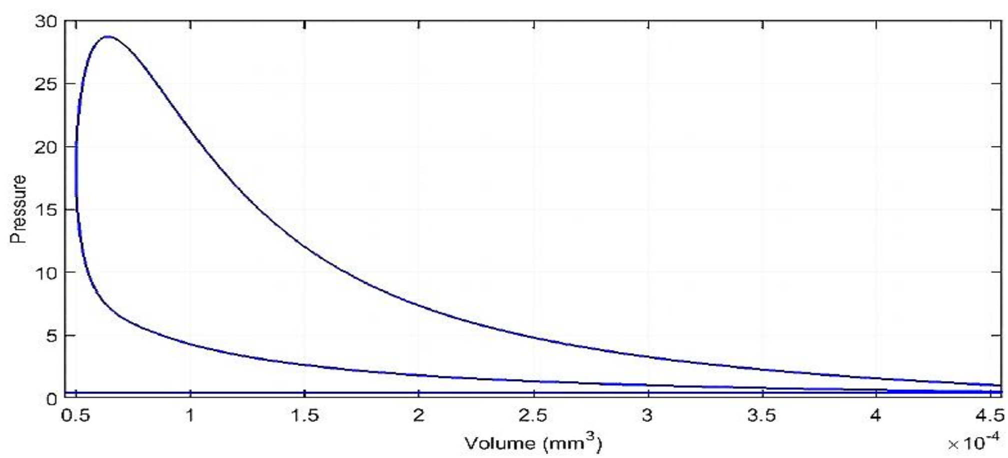


Figure 5: Plot of cylinder pressure against pressure volume

Figure 6 shows the cumulative work against the crank angle. This is crucial as it helps determine how much work is spent on compressing the charge, how much work is produced by the expansion process and the useful work made available to the crankshaft. With this, careful optimization can be done in order to optimize the useful power and reduce the power consumed in compressing the charge. The broken line plot represents the amount of heat loss to the environment. It can be seen that significant heat loss commenced during temperature rise and a jump in the heat loss occurred immediately the combustion process started occurring. The heat loss to the environment is one of the major inefficiencies in internal combustion engines.

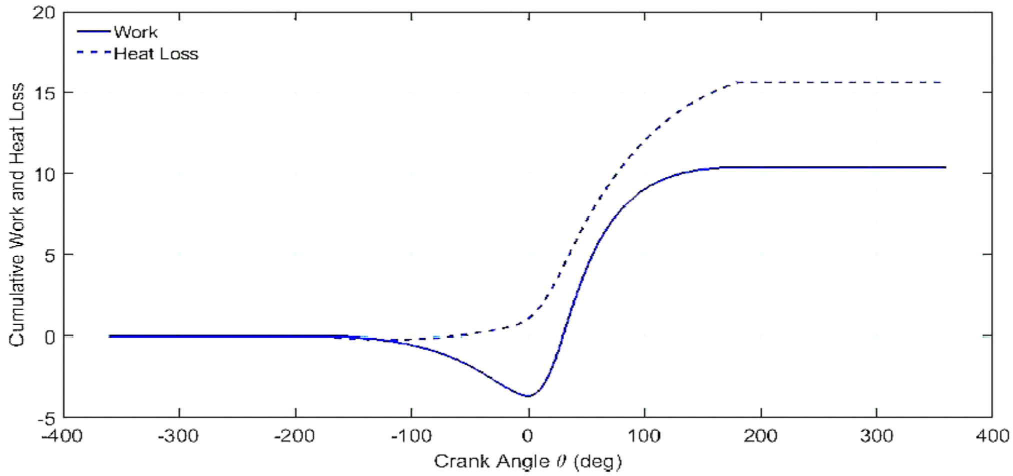


Figure 6: Plot of cumulative work and heat loss against crank angle

Figure 7 shows the experimental and simulated cylinder gas peak pressure. It was observed that the cylinder pressure increased as the combustion of in-cylinder air-fuel mixture occurred. This correlate with the findings of Challen and Baranescu (2003). On the other hand, in-cylinder pressure changes with crank angle due to cylinder volume change, heat transfer to chamber walls, flow in and out of crevice regions and leakages (Semin et al., 2010). The cylinder gas peak pressure occurred inside the cylinder and were computed during combustion in four crank angles cases of 10°, 20°, 30°, and 40°. The results indicates that maximum peak cylinder gas pressured occurred at the lowest crank angle degree and vice versa. Therefore, maximum experimental peak cylinder gas pressured of 5320 kpa and maximum simulated peak cylinder gas pressured of 5347 kpa occurred at the lowest crank angle of 10°. This agrees with the findings of Kuo (1996) who studied cylinder pressure in a spark-ignition engine.

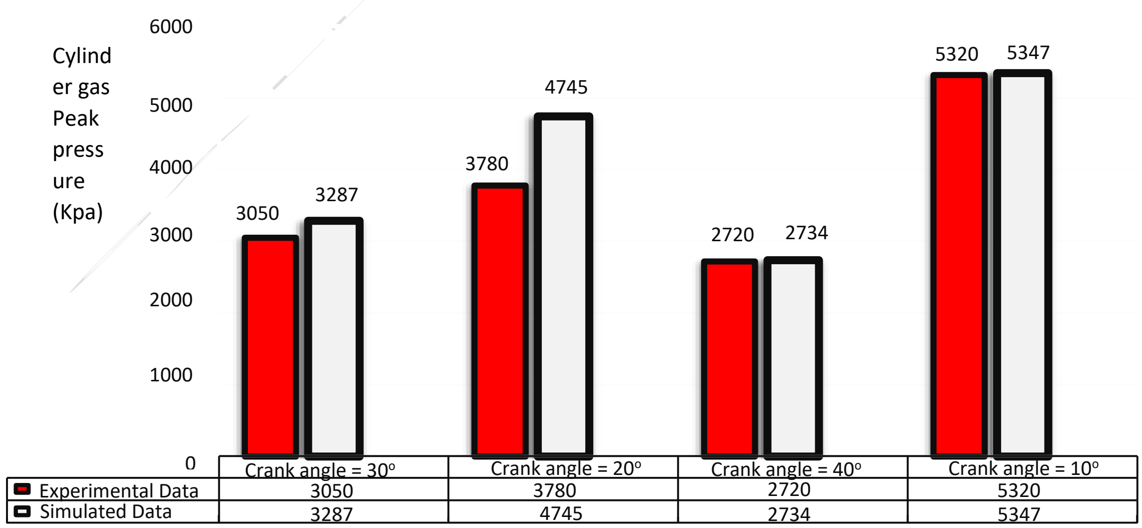
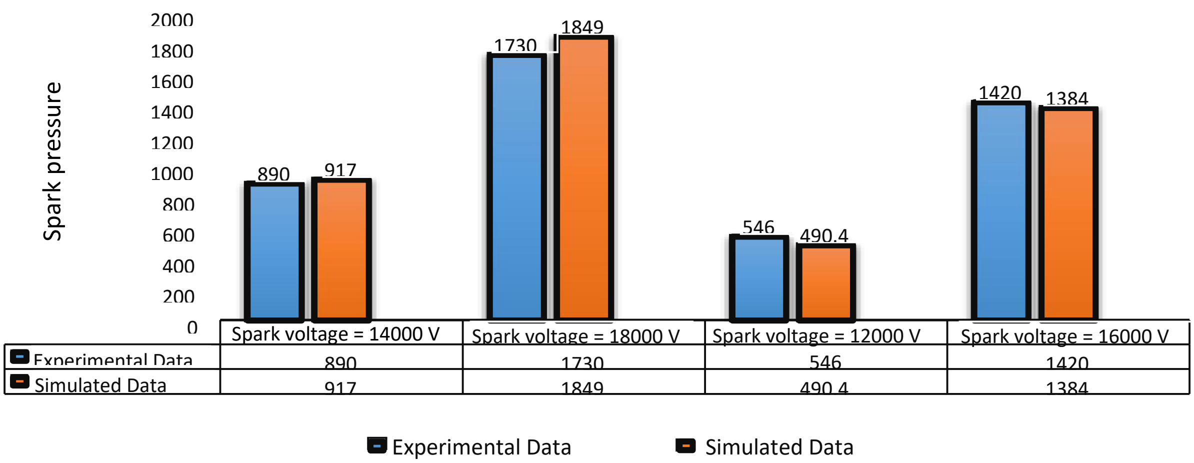


Figure 7: Peak pressure between simulated data and experimental data

Figure 8 presents the experimental and simulated cylinder spark pressure results. Spark plug is connected to high voltage generated by coil or magneto. As current flows from the coil, voltage is developed between the central and outside electrodes. Initially, current cannot flow because the air-fuel mix gases in the plug gab serves as an insulator. As soon as the voltage exceeds the dielectric strength of the gases, the gases becomes ionized. The ionize gas becomes a conductor and allows current to flow across the gab. The resultant heat and pressure force the gases to react and produce fire ball in the spark gab as the gases burn on their own. Therefore, maximum experimental cylinder spark pressure of 1730 kpa and maximum simulated cylinder spark pressure of 1849 kpa occurring at maximum spark voltage of 18000 V is required for proper engine firing. This indicated that spark pressure increased as the spark voltage also increased. Hence, the supply of higher results in a hotter and longer spark duration.



	Spark voltage = 14000 V	Spark voltage = 18000 V	Spark voltage = 12000 V	Spark voltage = 16000 V
Experimental Data	890	1730	546	1420
Simulated Data	917	1849	490.4	1384

Figure 8: Spark pressure in cylinder before combustion for simulated and experimental data

Figure 9 shows a comparison of experimental and simulated results for heat flow rate to cylinder liner at different crank angles. Crank angle is the measure of the position of an engine's crankshaft in relation to the piston as it travels along the cylinder walls. It is observed that heat flow rate to cylinder walls is maximum in the combustion stroke at crank angle between 0-180° with a maximum value of 250 W for the simulated result and 160 W for the experimental result. This is more likely the stroke and crank angle where the cylinder accommodates the highest heat rate as a result of air-fuel mixture combustion. The combustion stroke (also known as power or expansion stroke) is the third phase in IC engine cycle where the ignited air-fuel mixture expands and pushes the piston downwards. The force created by this expansion is responsible for the engine's power (Ikpe and Owunna, 2021). After combustion stroke, the heat flow rate to cylinder walls reduced from a maximum value of 250-160 W for simulated and experimental result. Also, the heat flow rate to cylinder walls reduced from a maximum value of 160-155 W for simulated and experimental result at crank angle between 180-360° in the exhaust stroke. The result showed that heat flow rate from the combustion chamber is higher in the combustion stroke than the exhaust stroke. This is because the hot gasses exiting the combustion chamber is extremely hot (600-1000°C), but gradually cools as it passes through the exhaust stroke or when exposed to atmospheric air.

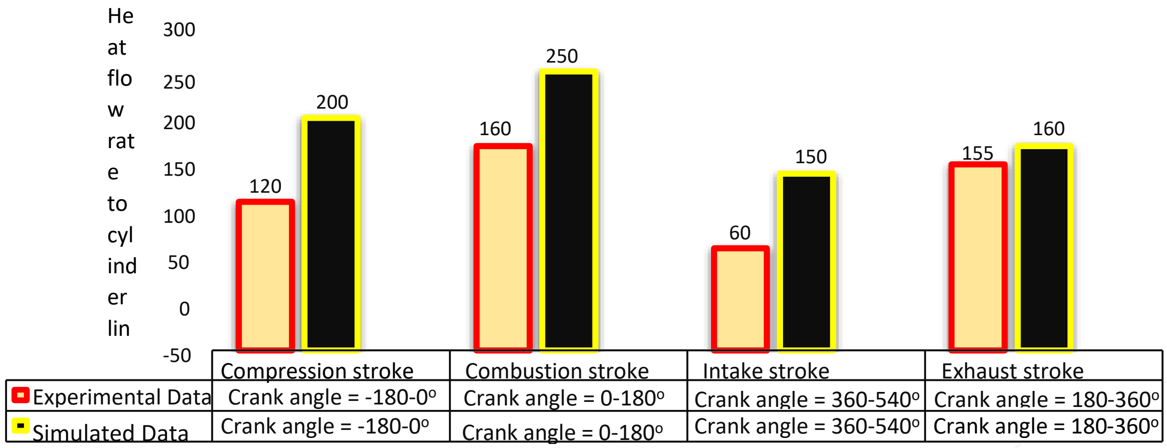


Figure 9: Heat flow rate to cylinder liner at different crank angles

Figure 10 shows the average percentage error between the simulated and experimental data. Accuracy of the MATLAB model is optimum, as the average percentage error for peak pressure and spark pressure is well below 10%. The angle of crank where peak pressure occurs showed a significant difference with percentage error of about 34.3%. Discrepancy between results from the MATLAB model and the experimental finding is as a result of the assumptions that were made. In this case, the effect of valve lift, ram effect, and exhaust valve timing were neglected in the model. Heat transfer from cylinder walls and the chemical kinetics of combustion reactions were not computed in the model.

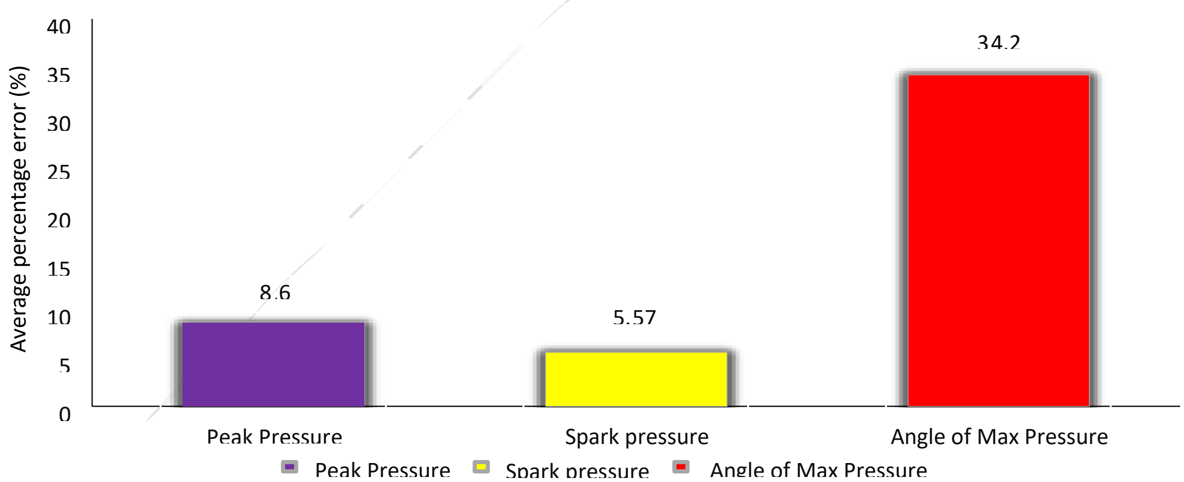


Figure 10: Average percentage error between the simulated and experimental data

4. Conclusion

In this study, the performance characteristics of a four stroke internal combustion Renault engine cycle was successfully modeled using 2018b MATLAB simulation tool. The codes were written where simulations were run for one cycle of the engine using MATLAB. The model can be seen as accurate because, the average percentage error for the peak pressure (8.6%) and the spark pressure (5.57%) were below 10%. The angle of crank where peak pressure occurred indicates considerable difference from the result, with percentage error of about 34.2%. Discrepancy between the simulated and experimental result may be due to cylinder pressure measurement.

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