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Computer Simulation of the Effect of Compression Ratio on Four-Stroke Spark Ignition Engine using an Alternative Fuel

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ABSTRACT

A four stroke spark ignition engine is a mechanical engine in which its operation completes by four strokes or movements of piston in the engine cylinder. This engine can also be called gasoline or petrol engine because its basic fuel is gasoline but nowadays, alternative fuels are being considered as the engine operating fuel. The alternative can be mixture of gasoline with other hydrocarbon products like ethanol or singly be another hydrocarbon product that can offer a better performance and lesser consumption when compared with gasoline. Though, the alternative fuels offer lesser consumption but it can also be intensified by considering the engine design specification for better performance which led to the birth of this work. Compression ratio of the engine determines the engine performance, the higher the compression ratio, the greater the efficiency of a spark ignition engine. In ordinary gasoline fuel, higher compression ratio can infer another engine problem like knocking and emission of toxic combustion product but the alternative fuels have been developed to lessen all these effects. The aim of this work was to simulate the influence of compression ratio on an alternative fueled four-stroke spark ignition engine using $C^{\#}$ computer program. The simulation model was carried out under assumed varying compression ratio; 4.63, 5.10, 6.00, 6.88, 7.40, 8.28, 9.16, 9.94, 10.45, 10.98 and 11.44, from this, the volume relationship was determined. The mathematical engine simulation model was carried out using thermodynamics-based models. The result of the simulation showed that higher compression gives higher efficiency and the numerical values are in close agreement with experimental values. The percentage error is not more that 2% during combustion duration and efficiency as compared with experimental values. From the numerical results, it was observed that if the compression ratio is high, the crank angle will also be high and this will make the ignition delay period to be diminished. This also goes for the engine performance, power output or the efficiency, the higher the compression ratio the greater the efficiency. Therefore, an alternative fueled engine with compression ratio of 9.16 and above will have a greater efficiency.

Introduction

A four stroke spark ignition engine is a mechanical engine in which its operation completes by four strokes or movements of piston in the cylinder (Harish and Anthony, 2008). This engine is also called gasoline or petrol engines. Petrol engines take in a flammable mixture of air and petrol which is ignited by a timed spark when the charge is compressed. The first four stroke spark-ignition (SI) engine was built in 1876 by Nicolaus August Otto (Zografakis, 2005; Mashkour and Ahmed, 2017). It was one of Otto's associates - Gottlieb Daimler - who later developed an engine to run on petrol which was described in patent number 4315 of 1885. The shortcomings of four stroke spark ignition engine cannot be over emphasized. Since environmental impact from transport sector which mainly utilizes energy from combustion of fossil fuel awakened many people around the world, widespread global initiatives have taken place in the light of this awareness. The development of hybrid electric vehicles and solar cars is one example (Thet *et al*, 2011). The use of alternative fuels such as bio-fuel, hydrogen fuel cells, and nano-energy are among others. Because the world economy is so far dependent on oil in a way that no other energy source can claim, improving spark ignition (SI) engine performance still needs to be paid attention to (Hutton, 2004).

One of the major areas of development in the internal combustion engine is the development of computer simulations of various types of engines. According to Al-Baghdadi (2004), the economic value is in the reduction in time and costs for the development of new engines and their technical value is in the identification of areas that require specific attention as the design study evolves. Computer simulations of internal combustion engine cycles are desirable because of the aid they provide in design studies, in predicting trends, in serving as diagnostic tools, in giving more data than that that are normally obtainable from experiments, and in helping one to understand the complex processes that occur in the combustion chamber.

The compression ratio is the ratio between the cylinder volumes at the beginning and end of the compression stroke. Broadly speaking, the higher the compression ratio, the

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higher the efficiency of the engine. However, compression ratio has to be limited to avoid pre-ignition of the fuel-air mixture which would cause engine knocking and damage to the engine. Modern motor-car engines generally have compression ratios of between 9:1 and 10:1, but this can go up to 11 or 12:1 for high-performance engines that run on, say, 98 Octane petrol. In the 1950s, with low-octane fuel and less well-designed cylinder heads, compression ratios were between 6.5:1 and 7:1 (Harish and Anthony, 2009). All these effects had been limited by the use of alternative fuels in place of ordinary petrol, thus, compression can be higher in an alternative fueled engine.

Considering the fuel consumption rate of the engine, the power output compared to other stroke engine and combustion rate of the engine; it will be seen that more improvement still have to be inculcated to this engine performance ability (Thet *et al*, 2011). The aim of this work is to simulate the influence of compression ratio on an alternative fueled four-stroke spark ignition engine using $C^{\#}$ programming language. This study was limited to four stroke spark ignition engine fueled with alternative fuel (hydrogen-ethanol mixture). The fuel is considered to be 70% gasoline and 30% ethanol. The simulation model was carried out under assumed varying compression ratio: 4.63, 5.10, 6.00, 6.88, 7.40, 8.28, 9.16 as used by earlier research work and a further step of increasing it to; 9.94, 10.45, 10.98 and 11.44, from this the volume relationship was determined. The model determined the different combustion parameters and evaluated the engine efficiency under different compression ratio.

Working Principle of the Engine in Relation to Compression Ratio

The piston reciprocates in the cylinder between two fixed positions called the top dead center (TDC) and the bottom dead center (BDC). The TDC is the position of the piston when it forms the smallest volume in the cylinder and the BDC when the piston forms the largest volume in the cylinder. The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction and it is called the stroke (L) of the engine.

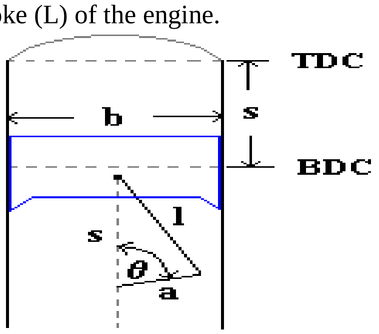

Figure 1. Piston Cylinder and Geometries.

It was assumed that a fixed mass of working fluid (fuel and air) is confined in the cylinder by a piston that moves from BDC to TDC and back. The cycle consists of four internally reversible process: An isotropic compression of an air-fuel mixture, which occur when the piston moves from BDC to TDC; a constant-volume heat addition process, where the mixture is ignited; an isentropic expansion of the combustion gases, where no heat is added and finally a constant-volume heat rejection.

The constant-volume heat rejection is a simple means of closing the cycle. It obviates the need to represent the complex expansion and outflow of combustion gases from the cylinder at the end of the cycle (Heywood, 1988).

Mathematical Analysis of the Simulation

The mathematical engine simulation model was carried out using thermodynamics-based models. Fluid-dynamics-based models are also called multidimensional models due to the fact that their formation is based on the conservation of mass, chemical species and energy at any location within the engine cylinder or manifolds at any time. Thus, the governing equations consist of ordinary differential equations (ODE) instead of partial differential equations (PDE) of multi dimensional equations. In such engine models, the combustion process is simulated in two ways for spark ignition engines. In the first approach, the rate of burning of the charge is obtained empirically by using the Wiebe function or the cosine burn rate formula. That is, combustion chamber geometry and flame geometry are not considered. Therefore, these models are called zero-dimensional models. In the second approach, the mass burning rate is determined from a mathematical model of the turbulent flame propagation. These types of thermodynamic cycle models that include detailed combustion modeling are called quasi-dimensional models.

Governing Equations

The first step in computing the simulation model was to assume varying compression ratio; 4.63, 5.10, 6.00, 6.88, 7.40, 8.28, 9.16 as used by earlier research work and a further step of increasing it to; 9.94, 10.45, 10.98 and 11.44, from this the volume relationship was determined. The minimum volume formed in the cylinder when the piston is at TDC is called the clearance volume. The volume displaced by the piston as it moves between TDC and BDC is called the displacement volume. The ratio of the maximum volume formed in the cylinder to the minimum volume is called the compression ratio ‘r’ of the engine:

$$r = \frac{\mathbf{V_{max}}}{\mathbf{V_{min}}} = \frac{\mathbf{V_{BDC}}}{\mathbf{V_{TDC}}} \qquad (1)$$

Also, from the First Law of thermodynamics (Ball *et al*., 1988), the relationship theoretically established by the equation to justify the correlation between the engine combustion and power process parameters was used. The equation is the fundamental procedure for all the process the engine underwent. The law states that the change in internal energy of a system is as result of difference the heat transferred to the system and work done by the system.

$$\Delta U = Q - W = m\frac{du}{d\theta} + \mu\frac{dm}{d\theta} = \frac{dQ}{d\theta} - p\frac{dV}{d\theta} \qquad (2)$$

Where m represents the mass of the cylinder content, μ represents the specific internal energy, ΔU represents the change in internal energy, Q represents the heat transfer, P represents pressure, V represents the volume and θ represents the crank angle.

Iteration Analysis for the Simulation of Alternative Fuels and Compression Ratio Parameters

The combustion chamber was generally divided into burned and unburned zones as earlier stated and can be separated by a flame front. The first law of thermodynamics, equation of state and conservation of mass and volume were applied to the burned and unburned zones. The pressure was assumed to be uniform throughout the cylinder charge. A system of first-order ordinary differential equations was obtained for the pressure, mass, volume, temperature of the burned and unburned zones, heat transfer from burned and unburned zone, and mass flow into and out of crevices.

The mass burning rate was modeled by the following equation according to Heywood (1989) ;

$$\frac{dM_b}{dt} = A_{ft} \cdot \rho \cdot ST \qquad (3)$$

Where M_b is the mass of burned gas (kg), A_{ft} is area of flame front (m), ρ is the density of the mixture (kg/m^3) and'ST' is the turbulent flame front speed (m/s) and it is;

$$ST = SL \cdot f \cdot \frac{(\rho_u/\rho_b)}{[(\rho_u/\rho_b)-1]Xm_b+1} \quad (4)$$

And 'SL' is laminar flame front speed (m/s), 'f' is a turbulent flame factor, defined with the following formula:

f =1 + 0.0018 × rpm (5)

The laminar flame front speed for mixtures of hydrocarbon and (or) alcohol with hydrogen, air, and residual gas can be modeled by the following equation (Bang-Quan *et al.*,2003;Blizard and Keck, 1974):

$$SL = SL_0 \cdot \left(\frac{T_u}{T_0}\right)^{\alpha} \cdot \left(\frac{P}{P_0}\right)^{\beta} \cdot \left(1 - 2.06X_r^{0.77}\right) + 0.83 \times YH_2 \quad (6)$$

Where YH_2 is an indication of the relative amount of hydrogen addition, which was defined by the following formular:

$$YH_2 = \left(\frac{[H] + \frac{[H]}{([H]/[Air])_{st}}}{[F] + ([Air] - \frac{[H]}{([H]/[Air])_{st}})}\right) \quad (7)$$

For pure hydrocarbon fuels or pure alcohol fuel, $YH_2 = 0$.

$$\boldsymbol{\alpha = 2.18 - 0.8(\varphi - 1)} \quad (8)$$

$$\boldsymbol{\beta = -0.16 + 0.22(\varphi - 1)} \quad (9)$$

Where 'φ' is the equivalence ratio and can be calculated by;

$$\varphi = \frac{(\frac{[F]}{[Air]})_{Act.}}{(\frac{[F]}{[Air]})_{st.}} \quad (10)$$

For the blending of hydrocarbon and alcohol with hydrogen fuel, the equivalence ratio can be calculated as;

$$\varphi = \frac{\left(\frac{[F]}{[Air] - \frac{[H]}{([H]/[Air])_{st}}}\right)_{Act.}}{(\frac{[F]}{[Air]})_{st.}} \quad (11)$$

Wiebe Function

A functional form often used to represent the 'mass fraction burned' against crank angle is the Wiebe Function and it can be analyzed as;

$$x_b = 1 - e^{[-\alpha(\frac{\theta-\theta_0}{\Delta\theta})^{m+1}]} \quad (12)$$

Where θ is the crank angle, θ_0 is the angle where the start of combustion occurs, $\Delta\theta$ is the total combustion duration and 'α and m' are adjustable parameters. Actual mass fraction burned curves can be fitted with α = 5 and m = 2. Obviously, the mass fraction that remains unburned is given by (1 - x_b). It is a standard functional form used for internal combustion engines which determines the overall combustion duration of the engine cylinder (Heywood *et al.*,, 1979).

Mixture Properties

In this study, during the combustion because of the existence of two distinct regions (burned and unburned) the properties of the mixture have been averaged for the purpose of simulation. Thus, the total density is :

$$\rho_{tot} = x_b\rho_b + (1 - x_b)\,\rho_u \quad (13)$$

Where $\rho_u = \frac{m_u}{V_u}$ (14)

and $\rho_b = \frac{m_b}{V_b}$ (15)

According to Dalton's Law of pressure we have :

$$P_{tot} = P_b + P_u \quad (16)$$

Total temperature can be calculated directly from the ideal gas law (Rasihhan and Wallace, 1991; Sitthichok, 2006), i.e.

$$T_{tot} = \frac{P_{tot}}{\rho_{tot} R/_{MB}} \quad (17)$$

$$\frac{dT}{dt} = \frac{1}{(m_u C_{p,u} + m_b C_{p,u})}[\dot{Q}_u + \dot{Q}_b + \dot{m}_{in}h_{in} + \dot{m}_{ex}h_{ex} - \dot{m}_u h_u - \dot{m}_b h_b + V\frac{dp}{dt}] \quad (18)$$

And

$$\frac{dp}{dt} = \left[p\left(\frac{\dot{m}_u}{\rho_u} + \frac{\dot{m}_b}{\rho_b} - \frac{dV}{dt}\right) + \left(\dot{Q}_u + \dot{Q}_b + \dot{m}_{in}h_{in} + \dot{m}_{ex}h_{ex} - \dot{m}_u h_u - \dot{m}_b h_b\right)\right]\left[V\left(1 - \frac{V}{\left(\frac{V_u C_{p,u}}{R_u} + \frac{V_b C_{p,b}}{R_b}\right)}\right)\right]^{-1} \quad (19)$$

Where C_p denotes specific heat capacity, h denotes specific enthalpy, ρ denotes gas density and V denotes cylinder volume. The terms depending on $\dot{m}_b$ and $\dot{m}_u$ are for enthalpy changes for burned and unburned zones while $\dot{m}_{in}$ and $\dot{m}_{ex}$ are for intake and exhaust processes. The $\dot{Q}_u$ and $\dot{Q}_b$ are heat losses of unburned and burned gases.

The gas pressure and temperature during compression stage are calculated using the following equations;

$$\frac{dP}{d\theta} = \frac{[-(1+\frac{R}{C_v}) \cdot P \cdot \frac{dV}{d\theta} - \frac{R}{C_v}\frac{dQ_{cr}}{d\theta} + \frac{R}{C_v}\frac{dQ_{ht}}{d\theta}]}{V} \quad (20)$$

$$\frac{dT}{d\theta} = T \cdot \left(\frac{1}{P} \cdot \frac{dP}{d\theta} + \frac{1}{V} \cdot \frac{dV}{d\theta}\right) \quad (21)$$

After spark occurrence from the whole simulation equations (carried out with Runge-Kutta method), then the delay period (DP) is calculated using this equation;

$$DP = \left[\left(\frac{6 \cdot rpm}{ST}\right) \cdot \sqrt[3]{\left(\frac{0.001V}{\pi}\right)}\right] \quad (22)$$

Performance Criteria of Internal Combustion Engine using Simulation Model

The performance criteria of the engine and their calculation by the simulation program are;

Indicated Power: It is the power actually developed in the engine cylinder or the rate at which work is done by the gas or fuel on the piston as determined by the computer program

$$\mathbf{Ip = Pi * A * L * N * Nc} \quad (23)$$

Where Ip is the indicated power at the engine cylinder, Pi is the indicated mean effective pressure, which is 120Nm, A is the area of the piston, L is the length of the stroke, N is the number of working stroke per unit time (for 4-stroke is rpm / 2) and Nc is number of cylinder

Mechanical Efficiency: It is the ratio of work-done with the energy supplied to the engine and always less than 100% because of power loss due to friction

From, $\mathbf{Pmi} = \frac{\oint \mathbf{PdV}}{\mathbf{Vd}}$, the output power can be calculated theoretically as the difference between the input power and the power loss due to friction.

Therefore,

$$\mathbf{Pme = Pmi - Pf} \quad (24)$$

$$\Rightarrow Pe = 0.5\, N * Pme * Vd \quad (25)$$

And

$$T = \frac{Pe*30}{\pi N} \quad (26)$$

Where Pmi is input power, Pme is output power, Pf is power loss due to friction, Pe is mean pressure and T is the torque transmitted (Wei-Dong *et al*., 2002).

Efficiency was calculated as;

$$\eta = \frac{b.p}{I.p} = \frac{2\pi nT}{I.p} \qquad (27)$$

where; η represents efficiency, b.p represents brake power which is $2\pi nT$ and T is the torque transmitted which is 1154Nmm or 1.154Nm by calculation.

Overall Combustion Duration of the Engine

The C# program predicted the overall combustion duration of the engine in a single cylinder as a result of crank angle between the start of combustion (SOC) and end of combustion (EOC) for varying compression ratio and the percentage error is calculated thus; The overall combustion duration was simulated by the widely used Taylor's equation for determining combustion duration in relation with compression ratio stated by Harish and Anthony, (2009) which is;

$$\Delta\theta c = 40 + 5 * \left(\left(\frac{N}{600}\right) - 1\right) + \left(166 * \left(\left(\left(\frac{12.5}{r}\right) - 1.11\right)2\right)\right)$$

and the percentage error was calculated thus:

$$\%\mathbf{error} = \frac{(\Delta\theta csim - \Delta\theta cexp)}{\Delta\theta csim}$$

Results and Discussion

The efficiency predicted by the program showed that the higher the compression ratio, the greater the efficiency.

Table 1 showed the combustion duration of the hydrogen-ethanol fuel used as an alternative fuel in place of ordinary gasoline fuel as predicted by the program which is in close agreement with the experimental result given by Harish and Anthony (2009) that was conducted on gasoline engine. The percentage error is not more than 2% which is arguably reasonable. The engine performance or efficiency was seen to be at peak (99.15%) when the compression ratio is at a greater value (11.44). The program predicted the combustion criteria which are pressure and temperature against the crank angle different that is caused by varying compression ratio. In figure 2, the average pre-ignition is observed at 10^o crank angle bottom-top dead center (BTDC) when the compression ratio is increased to 9.16. As it can be seen in the figure, the predicted pressure increases as the crank angle increases in degree which is a result of higher compression ratio.

The pressure deviates at crank angle of 200^o when the combustion process reaches its peak and this occur at the highest level of varying the compression at which a step above that the engine drops its efficiency. The pressure values predicted are in reasonable agreement with the experimental values carried out by Al-Baghdadi and Al-Janabi (1999) for an alternative fueled engine. The temperature values are also in the same trend with the pressure. Error in maximum pressures is less than 4% for both cases and the deviation in pre-ignition occurrence crank angle is less than 1^ocrank. The optimal mass fraction burned occurs at the highest compression ratio (11.44) which corresponds with the crank angle of 200^o.

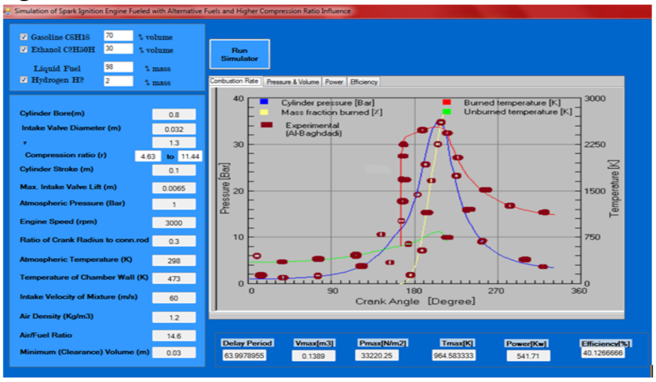

Figure 2. Visual Basic Interface of the Spark Ignition Engine Simulation Program.

This visual interface shows the relationship that exists between the combustion parameters. The hydrogen-ethanol-gasoline fueled engine operates with a stoichiometric mixture, optimum spark timing for best torque or power, 3000 rpm and a variable compression ratio. Each parameter studied was made dimensionless by relating it to its value when the engine is fueled with alternative fuel.The simulated values were compared with the experimental values carried out on

Table 1 . Simulated Values of Combustion Duration for Varying Compression Ratio.

Compression Ratio (r)	Δθc Experimental	Δθc Simulated values by C#	%Error	Efficiency (η)simulated in %
4.63	64	64	0	40.13
5.10	60	61	0.02	44.20
6.00	56	56	0	52.00
6.88	54	53	0.02	59.63
7.40	50	51	0.02	64.13
8.28	48	48	0	71.76
9.16	45	45	0	79.39
9.94	43	44	0.02	86.15
10.46	42	43	0.02	90.65
10.98	41	42	0.02	95.16
11.44	40	41	0.02	99.15

Table 2. Simulation Input Data for the C# Program.

Symbol	Value	Explanation
B	0.08	Cylinder Bore, measured in metre (m).
B_{in}	0.032	Intake Valve diameter, measured in metre (m).
γ	1.3	C_p / C_v, dimensionless.
(r)	Variable	Compression ratio, dimensionless.
L	0.1	Cylinder's stroke, measured in metre (m).
L_{in}	0.0065	Maximum intake valve lift, measured in metre(m).
P_o	1	Atmospheric pressure, measured in atmosphere (atm).
N, rpm	3000	Engine speed, measured in rounds per minute (rpm).
R_c	0.3	Ratio of crank radius to connecting rod length.
T_o	298	Atmospheric temperature, measured in Kelvin (K).
T_w	473	Temperature of chamber's wall, measured in Kelvin (K).
U_{in}	60	Intake velocity of mixture, measured in metre per second (m/s).

ordinary gasoline engine and it showed that the numerical values were valid with the experimental values.

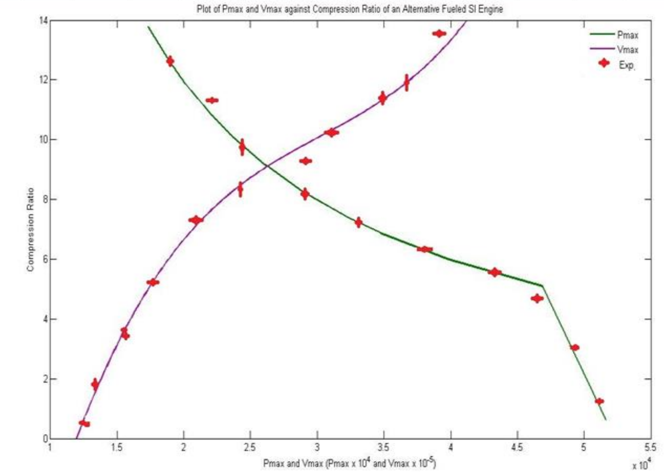

Figure 3. Predicted Pressure and Volume of the Engine at Varying Compression Ratio.

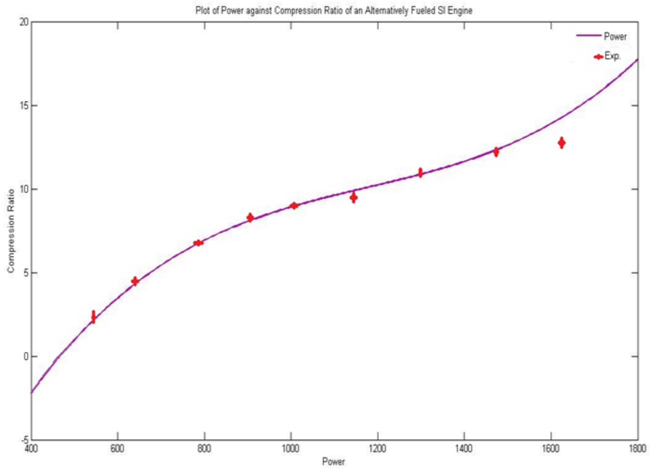

Figure 4. Predicted Output Power against the Compression Ratio.

The cylinder pressure and temperature were shown in relation with the crank angle which is determined by the compression ratio of the engine. For this type of engine simulated, the pressure firstly increased gradually as compression ratio increases but as it got optimal level of engine efficiency (i.e at compression ratio of 11.44) it diminishes. The temperature was also seen to follow the same trend with pressure but that of unburned and burned were analyzed together with declination at compression ratio of 11.44. The mass burning fraction is high at starting from compression ratio slightly above 10.

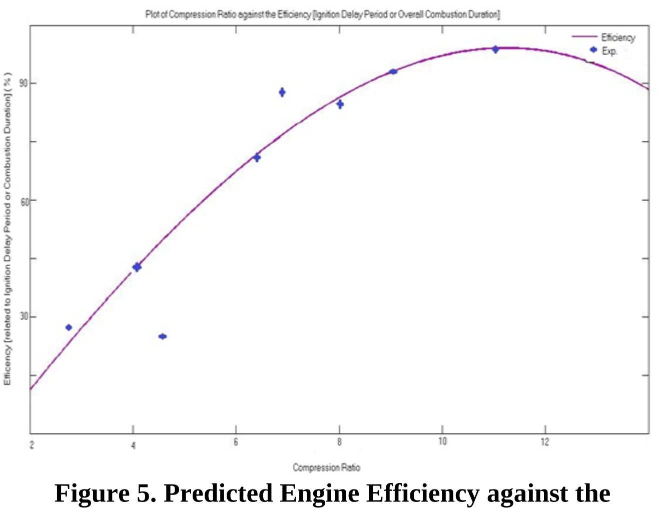

Figure 5. Predicted Engine Efficiency against the Compression Ratio.

The relationship between the compression ratio and some paramount combustion parameters like pressure, temperature, delay period, power and efficiency are illustrated in the graphs. The predicted values by the program using theoretical concept showed a close reasonable agreement with the experimental analysis of the engine performance. The simulated values were all compared with the experimental values and it was deduced that there is validation of the values with a slight error that is bearable. The predicted values were all in close agreement with the experimental values for each parameter.

The graph in figure 3 shows the effect of compression ratio on the pressure developed in the engine and the volume at the combustion chamber of the engine. When the compression ratio is high, the pressure developed will be reduced and this will in turn give high durability in an engine. An increase in compression ratio produces higher volume of the burnt fuel. The rate at which the fuel mixture undergoes combustion depends on the availability of enough combustion space in the combustion chamber.

Figure 4 shows the relationship between the compression ratio and indicated power developed during combustion. As the compression ratio increases, the indicated power increases as well. The predicted power values by the program were very close with the experimental values done ordinary gasoline fuel by Harish and Anthony (2009). Higher compression ratio gives greater engine output power even on the use of alternative fuels as it can be seen on the graph above. Engine designed with alternative fuels should also have high compression ratio more than that of contemporary gasoline ones.

Efficiency is the parameter that determines the performance of the engine as a result of the ratio of the output power of the engine to the input power. The program, using theoretical concept validates the result of experimental work when the efficiency is considered in relation to the engine ignition delay or the overall combustion duration as shown in figure 5. Both theoretical and experimental work showed that for greater efficiency, higher compression ratio is required. The simulation showed that this type of engine has highest efficiency or performance at compression ratio of 11.44, after which it diminishes. This showed that assumed working condition for this engine allows maximum compression ratio of 11.44 for higher efficiency.

Conclusion

From the numerical observation, it was deduced that compression ratio has a greater influence on the overall combustion duration of the engine with alternative fuels and

it can be designed higher than that of contemporary ordinary gasoline engine. The advantages are:

1. If the compression ratio is high, the crank angle will also be high and this will make the ignition delay period to be diminished. This also goes for the engine performance, power output or the efficiency, the higher the compression ratio the greater the efficiency.
2. When the compression ratio is 11.44, the thermal efficiency is 99.15 as predicted by the program. Therefore, engine with compression ratio above 9.16 will have a significant efficiency or performance.
3. It shows that the compression ratio determines the combustion process whereby the cylinder pressure, burned and unburned temperature were determined. The pressure gradually increases as the compression increases and later dropped back while the volume increases significantly.

4. The temperature increases as the compression ratio increase in line with the volume of the burning gas in the cylinder.

This simulation model can be used to assist in the design of a spark ignition engine for alternative fuels as well as to study many problems such as pre-ignition, fuel consumption rate and fuel mixtures. Many other parameters can be studied using this simulation model, such as the effect of combustion duration for each fuel on the performance and emission of the engine, the optimal amount of fuel supplement, and high useful compression ratio for each fuel. Also the use of hydrogen-ethanol as a supplementary fuel up to 30% of gasoline in modern spark ignition engines without major changes improves the output power and reduces the NOx emissions of a hydrogen supplemented fuel engine. The hydrogen added improves the combustion process, especially in the later combustion period, reduces the ignition delay, reduces the combustion duration and retards the spark timing. The blending of ethanol reduces the CO and NOx emissions and peak temperature. The concentration of CO is reduced, and the concentration of NOx is increased due to hydrogen blending. The engine power is increased until a hydrogen-fuel mass ratio of 2% and ethanol-fuel ratio of 30%.

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