Engine Performance of a Gardener Compression Ignition Engine using Rapeseed Methyl Ether

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Abstract - This paper presents an experimental investigation on the influence of engine speed on the combustion characteristics of a Gardener compression ignition engine fueled with rapeseed methyl ether (RME). The engine has a maximum power of 14.4 kW and maximum speed of 1500 rpm. The experiment was carried out at speeds of 750 and 1250 rpm under loads of 4, 8, 12, 16 and 18 kg. Variations of cylinder pressure with crank angle degrees and cylinder volume have been examined. It was found that RME demonstrated short ignition delay primarily due to its high cetane number and leaner fuel properties (equivalence ratio (φ) = 0.22 at 4kg). An increase in thermal efficiency but decrease in volumetric efficiency was recorded due to increased brake loads. Variations in fuel mass flow rate, air mass flow rate, exhaust gas temperatures and equivalence ratio with respect to brake mean effective pressure at engine speeds of 750 and 1250 rpm were also demonstrated in this paper. Higher engine speed of 1250 rpm resulted in higher fuel and air mass flow rates, exhaust temperature, brake power and equivalent ratio but lower volumetric efficiency.

Keywords— combustion characteristics, engine performance, engine speed, rapeseed methyl Ether ————

1 INTRODUCTION

For several decades, fossil fuels (oil, coal and natural gas) have been serving the global energy demand due to their high energy densities. According to the analysis made by the British Petroleum (BP) in the year 2011, global oil proved reserve rose to 1652.6 billion barrels at the end of the year 2011 and was potentially adequate to meet supply in 54.2 years of the world oil production. The statistics further showed reserve to production ratio (R/P) of the world leading oil producing regions. Based on the statistics, Middle East, having 48.1% of the world proved reserve, had decreased R/P due to increased production in the region.

Even though fossil fuels are exhaustibly imminent based on the current projections on the proved/unproved reserves and contingent/prospective resources. However, rapid population, technological and economic growth of developing nations has now craved for a much higher demand of these fossil sources. However, over dependence on these conventional energy sources has now continue to pose great challenge to the environment due to emission of greenhouse gases (Carbon dioxide (CO2), methane (CH4) and nitrous oxide (NOx)). Consequently therefore, production of legislated emissions (carbon monoxide (CO), hydrocarbons (HC), NOx, particulate matter (PM)) and unlegislated CO2, water (H2O), nitrogen (N2), oxygen (O2)) emissions are inevitable (Wen, 2013).

It is therefore imperative to investigate sustainable and clean energy sources in to meet the nagging environmental concerns without sacrificing fuel economy to the minutest level. To do this, biofuels such as RME (rapeseed methyl ester) and DME (di methyl ester) have been proven to be remarkable alternatives. Since, they have demonstrated rapid reductions in some regulated emissions with increased brake mean effective pressure and decreased specific fuel consumption (Crookes, 2007). Due to higher efficiency of compression ignition (CI) over spark ignition (SI) engines because of higher compression ratio and lean combustion of fuels, RME fuels are typically used in CI than SI engines.

This paper outlines an experiment carried out on a Gardener (model 1L2) engine using an RME fuel at engine speed of 750 rpm for various loads of 4, 8, 12, 16 and 18kg. Engine performance parameters such as brake mean effective pressure (bmeanp), brake power and specific fuel consumption (sfc) will be calculated and the results will be compared with the same RME fuel run at a higher engine speed of 1250 rpm. Cylinder pressures to crank angle degrees and cylinder volume diagrams will be plotted with the target to calculating indicated mean effective pressures at various loads.

2 EXPERIMENTAL SYSTEMS AND METHODOLOGY

2.1 EXPERIMENTAL SYSTEM

The Gardener 1L2 engine is a single cylinder, four-stroke, direct injection compression ignition engine which is used to determine the effect of different fuels at different experimental conditions. It is an internal combustion engine which converts chemical to mechanical energy due reciprocating movement of piston in a cylinder. The piston is connected to a crankshaft which changes its linear vertical motion to a rotary motion. Fig.1 shows a typical Gardener compression ignition engine.

Fig.1: A diagram of a Gardener Engine Model 1L2 (Gardener Magazine, 2008)
2.2 EXPERIMENTAL PROCEDURE

The experiment was carried out using the Gardner Engine the model of which (1L2) is like the one shown in Fig.1. The engine was operated at speed of 750 and 1250 rpm fueled RME fuel and under loads 4, 8, 12, 16 and 18 kg. Table 1 shows the specifications Gardner engine assembly.

| Table 1. Gardner Engine Specifications | Specifications |
|---------------------------------------|---------------|
| Engine Model                          | 1L2           |
| Number of Cylinders                   | (σ) 1         |
| Bore (D)                              | 07.95 mm      |
| Stroke (S)                            | 152.4 mm      |
| Swept volume (capacity)(V_s)          | 1394.8 × 10^-6 m³ |
| Clearance volume (V_c)                | 115.15 × 10^-6 m³ |
| Compression ratio                     | 14:1          |
| Maximum power                         | 14.4 kW @ 1500 r/min |
| Inlet valve opening                   | 10° btcd      |
| Inlet valve closing                   | 40° abdc      |
| Exhaust valve opening                 | 50° bbdc      |
| Exhaust valve closing                 | 15° atdc      |
| Injection timing                      | 24.5° btcd    |
| Injector nozzles                      | 4             |
| Nozzle throat dia                     | 220 μm        |
| Injector opening pressure             | 16.2 MPa      |

Where: atdc = after top dead centre, btcd = before top dead centre, abdc= after bottom dead center, btdc = below bottom dead center

Atmospheric pressure was initially measured using a manometer at a room temperature and a load of 4kg was applied to the engine at a speed of 750 rpm. The engine speed was measured using analogue tachometer located on the engine. The inlet temperature was measured using a thermocouple and the time taken for the engine to consume 20 ml of RME fuel was measured with a fixed clock situated on the engine. Mercury height was noted and recorded with the aid of a manometer attached to the engine. The Gardner engine was connected to a Desktop computer where the pressure readings (fuel line pressure, cylinder pressure and TDC positions) and exhaust temperature were measured. The procedures were repeated for the remaining loads and speed of 1250rpm and readings of the engine inlet engine conditions recorded as shown in Tables 2 and 3.

2.3 PERFORMANCE PARAMETERS

The following equations in section 2.3 are for the engine performance parameters (Heywood,1998; Wen, 2013). Calculations were carried out at an engine loads of 4, 8, 12, 16 and 18kg and at a speed of both 750 and 1250 rpm.

2.3.1 Brake Power (KW)

Brake power (W_b) is given by Eq.1:

\[ W_b = \frac{L \alpha}{\mathcal{N}} \]  

(1)

Where: L = load (N), \alpha= engine speed in revolution per second (rps) and \alpha= 0.447

The load (L) is also given by Eq.(2)

\[ L = Mg \]  

(2)

Where: M= mass (kg) and g = specific gravitaional force (N/kg).

2.3.2 Brake Mean Effective Pressure (KPA)

Brake Mean effective pressure (bmep) is given by Eq.3:

\[ bmep = \frac{W_b n}{V_s N_o} \]  

(3)

where: n = number of engine revolutions per machine cycle, \V_s = swept volume \& \alpha = number of cylinders

2.3.3 Fuel Mass Fuel Rate (kg/s)

Fuel Mass fuel rate is given by Eq.4:

\[ \dot{m}_f = \frac{\varepsilon}{t} \]  

(4)

Where: \varepsilon = mass of 20 ml of fuel \& t = fuel time (s)

2.3.4 Air Mass Flow Rate (kg/s)

Air mass flow rate equation is shown in Eq. (5)

\[ \dot{m}_a = C_d A \sqrt{\frac{2 P_a \Delta P}{\rho_a}} \]  

(5)

Where: \Cd = coefficient of discharge, \A = area of the orifice, \rho_a= air density (kg/m³) \& \Delta P = pressure drop across the orifice (kPa) \& \Pa = atmospheric pressure.

2.3.5 Volumetric Efficiency

Volumetric efficiency (\eta_v) is given by Eq.6:

\[ \eta_v = \frac{m_a}{V_s P_a \rho_a} \]  

(6)

2.3.6 Mechanical Efficiency

Mechanical efficiency (\eta_mech) given by Eq.7:

\[ \eta_mech = \frac{\text{brake power}}{\text{indicated power}} = \frac{bmep}{\text{bmepp}} \]  

(7)

2.3.7 Specific Fuel Consumption (kg/s)

Specific fuel consumption (sfc) is given by Eq.8:

\[ Sfc = \frac{\dot{m}_f}{W_b} \]  

(8)

2.3.8 Thermal Efficiency

Thermal efficiency is shown in Eq. 9:

\[ \eta_{th} = \frac{W_b}{m_a \Delta H_l} \]  

(9)

Where: \Delta H_l = specific enthalpy of reaction with the product water as vapor

2.3.9 Equivalence Ratio

Equivalent ratio (\phi) is given by Eq.10:

\[ \phi = \frac{\phi_{actual}}{\phi_{stoich}} \]  

(10)

Where: \( \phi_{actual} = \text{actual fuel/air ratio} = \frac{m_f}{m_a} \) and Stoichiometric fuel air ratio \( \phi_{stoich} \)

The stoichiometric equation for the combustion of RME is given by Eq.11:

\[ \text{C}17\text{H}33\text{O}_7 + 29.5 (\text{O}_2 + 3.76\text{N}_2) \rightarrow 21 \text{CO}_2 + 19 \text{H}_2\text{O} + 110.92 \text{N}_2 \]  

(11)

2.3.10 Volume in the Cylinder

The total volume in the cylinder and compression ratio are given by Eq. 12 and Eq. 13 respectively.
\[
\frac{V}{V_c} = 1 + \frac{1}{2} \left( CR - 1 \right) \left[ \frac{1}{a^2} + 1 - \cos \theta - \left( \frac{c}{a} \right)^2 - \sin^2 \theta \right]^{1/2}
\]

(12)

Compression ratio (CR) = \( \frac{V_a + V_c}{V_c} \)

(13)

Where: \( V_a \) = displaced or swept volume, \( V_c \) = clearance volume, \( l \) = length of connecting rod, \( a \) = radius of the crank angle, \( \theta \) = crank angle

Work per cycle = \( \frac{W_i}{V_d} \)

(14)

Indicated power (\( W_i \)) = \( \frac{W_i}{n} \)

(15)

3 RESULTS AND DISCUSSION

3.1 VOLUME IN THE CYLINDER

The engine parameters recorded (inlet and exhaust temperatures, manometer height and time) from RME combustion at engine speeds of 750 and 1250 rpm under loads of 4, 8, 12, 16 and 18 kg are shown in Tables 2 and 3 respectively.

Table 2. Engine parameters for RME combustion at speed of 750 rpm

| Parameters          | Engine parameters at 750 rpm |
|---------------------|-----------------------------|
| Mass (kg)           | 4  8  12  16  18            |
| Inlet temperature (\( T_i \)) | 12  13  13  12  13 |
| Exhaust temperature (\( T_e \)) | 140.7  190.2  236  303  320 |
| Manometer height, h (mm) | 19.5  18.5  18  17.5  16.5 |
| Time, t (s)         | 122  86.67  64.9  51  45.2 |
| Load, L (mg)        | 39.2  78.48  118  157  177 |

Table 3. Engine parameters for RME combustion at speed of 1250 rpm

| Parameters          | Engine parameters at 1250 rpm |
|---------------------|-----------------------------|
| Mass (kg)           | 4  8  12  16  18            |
| Inlet temperature (\( T_i \)) | 14  14  14  14  1  |
| Exhaust temperature (\( T_e \)) | 183  233  293  371.4  383.4 |
| Manometer height, h (mm) | 46  45  43.5  41.5  37.5 |
| Fuel time, t (s)    | 67.6  47.9  37.1  28.98  26.1 |
| Load, L (mg)        | 39.2  78.5  118  156.9  176.6 |

3.2 VARIATION OF CYLINDER PRESSURE AND VOLUME WITH CRANK ANGLE DEGREE

Fig.2a shows variations of cylinder pressure with crank angle degrees and while Fig.2b shows cylinder volume all at a representative load of 4kg. The crank angle was measured on a Lab View interface using a control computer. Top dead center (TDC) position was measured alongside with 2 sets of pressures under each load conditions. These 2 pressures are cylinder or combustion pressure and fuel line pressure. Only cylinder pressure was used because it obviously indicates the pressure of RME fuel and oxygen during combustion. It is the pressure under which CO, H2O and N2 are formed as the products. Typically, at 30\(^\circ\) before TDC position, RME fuel was injected in to the cylinder of the engine at a rate of \( m_f \) while it atomizes.

The pressure with which the RME moves is called the fuel line pressure. RME vapor and air mixes while the pressure and temperature of air are above the ignition point of RME. This causes auto-ignition. That is, spontaneous ignition of the non-uniform RME/air mixture after a short delay (ignition delay) and that starts the combustion process. At this point, the cylinder pressure increases above the non-firing engine level. This behavior of fuel combustion is clearly indicated in Fig.2 as the period between the start of injection (SOI) and start of combustion (SOC) for minimum load of 4kg. In the combustion of RME with air however, the ignition delay is short due to its high cetane number of about 54.4 (Crookes, 2007). The ignition delay could also be reduced by minimizing heat transfer in the combustion chamber but increases with increase in exhaust gas recirculation (EGR). It is evident therefore as it has been suggested by Horn et al. (2006) that the main disadvantage of RME is self-ignition behavior at low loads. This occurs because of its high viscosity, surface tension and consequently, larger sauter mean diameter. Hence, the reasons for its larger fuel droplets during atomization and affinity to having high fuel rich mixture (Shoba et al., 2011).

Similarly, from Fig.2b on the cylinder pressure and cylinder volume, it is apparent that increased brake load is accomplished by increasing cylinder pressure (P), this in turn leads to decrease in total volume of the cylinder (V). For example, from Figure 2 at a minimum load of 4kg, the maximum pressure (3.681bar) at point X corresponds to a volume of 3.84 x 10\(^{-4}\) m\(^3\) while for a maximum load of 18kg, the maximum pressure of 5.317bar corresponds to a volume 1.91 x 10\(^{-4}\) m\(^3\). These results are not surprising since the compression stroke is isothermal on a T-S diagram and the cylinder pressure is
a function of crank angle degrees. Moreover, the highest pressure on the P-V diagram is associated with the smallest clearance volume ($V_c$) which further means higher compression ratio (CR). High compression ratio is a unique characteristic of most CI engines (Richard, 1985).

### 3.3 Influence of Engine Speed on Engine Performance Characteristics

![Graph](image_url)

**Fig. 3:** (a) Variation of brake power (kW) with mass flow rate (kg/s) (b) Variation of air mass flow rate (kg/s) with brake mean effective pressure (kPa)

Fig. 3a shows a graph of brake power against fuel mass flow rate. It can be deduced from Fig.3a that the higher the engine speed, the higher the fuel mass flow rate. At a speed of 1250 rpm for example, it can be observed that the mass of fuel consumed for every second is much higher than that consumed at a speed 750 rpm for the same range of loads. Increased brake load also signifies increase in brake power, which further means increasing the rate at which the fuel flows into the engine (Heywood, 1988). It can also be noticed, more distinctly, that the brake power increases much more steadily at the higher speed of 1250 rpm with a steeper slope than at lower speed of 750 rpm.

Fig. 3b shows variations of air mass flow rate with brake mean effective pressure. It can be observed from the graph that the speed of 1250 rpm has a much higher F/A and bmep than 750 rpm at the same orifice area and pressure differential.

Fig. 4a shows a graph for the variation of fuel air ratio (F/A) with bmep. It can be seen from the graph that higher F/A led to higher bmep at both speeds. However, the speed of 1250 rpm has a much higher F/A and bmep than 750 rpm at the same orifice area and pressure differential.

Fig. 4b shows the variations of volumetric efficiency with bmep. It is obvious that volumetric efficiency decreases with increase in bmep at both speeds. In addition, the speed of 750 rpm shows higher volumetric efficiency than 1250 rpm, since $\eta_v$ is in inverse proportion with speed, which might cause increase in air mass flow rate at a particular swept volume.

Fig. 5a shows the variation of Exhaust gas temperatures with bmep. At speeds of both 750 and 1250 rpm, the exhaust gas temperature was found to increase with the engine output (bmep). The relationship between the two quantities is closely linear and is almost the same with a constant difference for the two operating speeds. At a speed of 1250 rpm for example, the exhaust gas temperature is slightly higher than that at speed of 750 rpm. This result, in practical terms, means that increase in engine speed and bmep lead to corresponding increase in combustion temperature and temperature of the emission gases. Fig. 5b shows variations of equivalence ratio from RME combustion with bmep. It is obvious from Fig.5b that the higher the equivalence ratio of the fuel, the higher the output of the engine irrespective of the speed. This correlation is fairly linear with speed of 1250 rpm slightly elevated above 750 rpm at all bmeps.
Variations in Fig.5b and Fig.4a are very much similar since equivalent ration is a function of fuel-air ratio (F/A) (Heywood, 1988). Higher actual fuel-air ratio demonstrates high equivalence ratio at an increased speed and bmep. Fig.5b further delineates the lean properties of RME since all the equivalence ratios lie between the values of 0.2 to 0.8. This result corresponds to those available in most literatures (Cisek, 2010). Figs. 3 to 5 show different variations of an RME combustion to various parameters that are seminal to the engine optimal performance. A lucid comparison of these engine parameters was examined at engine speeds of 750 rpm and 1250 rpm. As it is available from most literatures an ideal combustion in a CI engine should yield high bmep, faster combustion process and higher efficiency with clean exhaust emissions and less noise (Richard 1985; Heywood, 1988). However, these requirements are entirely competing for an RME fuel. For instance, the output of the engine (bmep) was found to increase with A/F, equivalence ratio and exhaust gas temperatures with decrease in volumetric efficiency.

Although the minimum cylinder pressure was found to be 385 cm$^3$ at minimum load and is normal for a good “surface to volume ratio”, there are lots of inconsistencies with what was expected for an ideal engine. Therefore, it is quite imperative to increase the bmep of this engine by turbocharging. As with the case of high exhaust temperatures of the Gardenier compression ignition engine, this practically means much work will be extracted by the turbines at the exhaust. The engine parameters showed reasonable trend with respect to the variations in brake loads and engine speed. An increase in brake load was found to have a corresponding increase in thermal efficiency but decrease in volumetric efficiency. At the same time, increase in engine speed from 750 rpm to 1250 rpm was shown to cause decrease in volumetric efficiency. However, irregular ignition timing and faulty injector must have greatly affected the engine performance at those speeds and will likely cause the engine to underperform (detonate).

4 CONCLUSIONS
An investigation into the combustion characteristics of RME in a Gardener 1L2 compression ignition engine was presented and the effect of engine speed on the engine performance characteristics were analyzed. RME fuel has been reported to have short ignition delay due to its high cetane number and leaner fuel properties because of its equivalence ratio being less than unity. An increase in thermal efficiency but decrease in volumetric efficiency was recorded due to increase in brake loads. The effect of fuel mass flow rate, air mass flow rate, exhaust gas temperatures and equivalence ratio were well demonstrated with respect to brake mean effective pressure at 750 and 1250 rpm. Higher engine speed of 1250 rpm resulted in higher fuel and air mass flow rates, exhaust temperature, brake power and equivalent ratio but lower volumetric efficiency.

ACKNOWLEDGEMENTS
Hamisu A. Dandajeh wishes to gratefully acknowledge the Petroleum Technology Development Fund (PTDF) for sponsoring his Post Graduate Studies at the Queen Mary University of London, United Kingdom

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