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Research Article

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Design review of an Innovated Single Piston Diesel engine in Uganda*

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Abstract

Engineering designs are important in engine production. This research aimed at incorporating engineering principles and practices into the developed single piston diesel engine at Kevoton. Tests on the already innovated engine were carried out to evaluate the performance of the existing engines in terms of power, torque, rpm and exhaust temperature. Designs of the major components were generated together with material selection, which helped in determining whether the used material was appropriate for engine production. Results showed that the engine runs at a speed of 1800 rpm, a torque of 5 Nm, and a power of 10 HP with 2.4 liters per hour of fuel. The major components of the engine include the engine piston, the crankshaft, connecting rod, cylinder head and the camshaft. The power obtained after carrying out the new design was 13 HP, with a 5 Nm torque as well as 1 liter of fuel consumed per hour. It was realized that some components were not meeting the design specifications and the design was below the minimum required power of 13 HP. Power produced was less than 13 HP that was required to handle the purpose of engine manufacture, the engine was designed to be a prime mover to do most of the local. It was identified that the need of the engine should be considered as a primary requirement when coming up with the designs, the choice of injectors affect the amount of exhaust and engine testing requires consideration while designing an engine.

Keywords Diesel Engine, Performance, Innovation, Material Selection

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1. Introduction

An engine works on either internal or external combustion. In this study, efforts have been put on analyzing an internal combustion engine (ICE) running on diesel, locally innovated and produced in Uganda. ICEs employ compression of air-fuel mixture in a cylinder to generate the required power (Naik et al. 2019), while for the external combustion engine, a separate combustor is used to burn the fuel (Radoslav 2011). ICEs may either be diesel or gasoline fueled and are known to be reciprocating (Ken et al. 2015). They are a widely used technology for both transportation and stationary applications (Ken et al. 2015). Diesel engines are commonly used for power generation due to their higher efficiency (Breeze 2018). But these engines are accompanied with flaws like high temperature required to combust the fuel and high percentages of emissions (Shams 2019). For that matter, spark ignition engines burning natural gas have become increasingly popular for power generation in industrialized nations. Stirling engine which is also a piston engine is being developed for specialized power generation applications though it is an external combustion engine (Breeze 2018). The diesel engine was invented in 1892, by Rudolf Diesel of Munich (Michaelis 2013). Diesel engines possess a number of efficiency advantages over gasoline (petrol) engines. These arise due to the way diesel engines operate. The deceit engines do not use a throttle to control airflow into the engine or a spark plug to ignite the fuel as gasoline does. Instead the load is controlled by the amount of fuel injected. The timing of fuel injection controls combustion timing as the fuel ignites almost immediately after being injected into the hot compressed gas within the cylinder. As a result, diesels eliminate the significant pumping losses that come from forcing air past the throttle in gasoline engines (Aaron Isenstadt 2017). Uganda is among the countries that have taken up the diesel technology for their applications. The country is gifted with natural resources and a salubrious climate with low industrialization and value addition. The country faces the challenge of poverty alleviation, a high human population growth rate low technology, science, engineering and innovation levels (Mutambi 2013). Uganda has invested in a number of sectors that use electricity and engines for running their activities. However, all these engines are imported to the country due to the little potential to produce them within the country. There is no import restriction on cheaper low quality engines for example the average age of light duty diesel passenger cars was 16.4 years by 2014 (UN Environment 2018). In February 2019, the Ministry of Finance, Planning and Economic Development compiled a report that shows Uganda's expenditure on imports to be US$ 507.14 million and incomes from exports to be US$ 299.57 million (MoFPED Report 2019). Engines are very useful in industries for running heavy machines, in agriculture, transportation and others. These sectors greatly contribute to the country's GDP with good margins. For example, the industrial sector's contribution to GDP is 19.8%, the services sector which consists communication, transport, education, health as major players makes the biggest contribution to GDP, standing at 47.8%, with agriculture taking the second place at 24% (Muwanguzi et al. 2018). KMEL a Ugandan local company developed a diesel engine running on a four stroke single piston. The challenge with the engine produced at KMEL, is that it lacked the preliminary engineering designs required for benchmarking in future manufacturing at the company and possibly acting as a stand for correcting the flaws in the engine. Some of the flaws included; high vibrations, emissions, low power and low torque.

2. Ease of Use

The Design review of a locally innovated diesel engine in Uganda involved studying the developed engine and then generating better designs to help new researchers in constructing engine parts.

2.1. Abbreviations and Acronyms

CI- Compression Ignition, GDP- Gross Domestic product, ICE- Internal Combustion Engine, ISO- International Standards Organization, KMEL – Kevoton Motions Engineering Company Limited, MAPRONANO - Africa Center of excellence in Materials Product Development and Nanotechnology, MoFPED – Ministry of Finance, Planning and Economic Development, MTIC - Ministry of Trade, Industry and Co-operatives, NEMA – National Environmental Management Authority, UNBS - Uganda National Bureau of standards.

3. Materials and Methods

2.1 Materials

The materials used in this research were limited to locally casted pieces of engine components of Aluminum 6061, for example the piston, cylinder head, engine block and purchased fogged steel components for example, the connecting rod, crankshaft and the camshaft. Cast iron was also considered for the case of bearings and the flywheel

2.2 Sample Preparation

The locally innovated engine was as well tested for brake power, torque, rpm, fuel consumption and exhaust temperatures at the Materials Laboratory, Makerere University. The engine was suspended to a height of 75 cm to achieve the level of the dynamometer shaft for alignment with the engine shaft. A TD200 dynamometer of maximum power absorption 10 HP at 7000 rpm, and maximum torque 15Nm was used in carrying out this test as shown in figures 4.1 and 4.2. Before the testing, the engine was started and allowed to get warmed up for about 5 minutes. The engine was placed to a test bench where its output shaft was coupled to the rotor of the dynamometer. Mechanical friction helped in braking the rotor. In determining the major components of the developed single piston diesel engine, emphasis was put on component dimensions and measurements. The main components identified according to literature included; the engine piston, piston rings, connecting rod, crankshaft, camshaft, flywheel, cooling system, cylinder head and others. Focus is drawn to the physical, mechanical and chemical properties of the material of the engine. The samples were prepared in three ways to do the tensile stress, compression strength at Central Materials Laboratory, Ministry of Works and Transport in Kireka, Kampala and the chemical properties of the material used in building the components of the engine were done at Steel and Tubes Industries Ltd Laboratory, Kampala. For tensile test, the sample was sized as; Length = 50 mm, Width = 50 mm and Height = 400 mm. The test sample was prepared into dimensions of 50 mm x 50 mm for compression tests. Chemical tests were done by dimensions of material of 50 mm x 50 mm.

2.3 Engine Specifications

The specifications of the locally innovated engine are as highlighted in Table 1;

Parameter	Specification
Swept volume, cc	32.52
Stroke, mm	106
Cylinders	1
Bore, mm	96
Strokes	2
Compression ratio	22:1
Fuel	Diesel
Emission standards Euro	4

Table 1: ***Specifications of the locally innovated engine***

2.3 Major Engine Components

According to the National Council of Education, Research and Training (NCERT), NCERT, (2019)an IC engine is composed of the following major components; Engine Block, Cylinder, Cylinder Head, Piston, Connecting Rod, Camshaft and the Crankshaft.

2.4 Design of the Engine

Engine components identified in section (2.3) were designed as per the following formulae;

2.4.1 The Crankshaft

The crankshaft was designed according to the following procedure applied by (Reddy 2019).

The combustion pressure = $(\frac{\pi}{4})\, \mathbf{D^2 \text{ x } P_{max}}$ [1]

Angle of inclination of the connecting rod with the line of stroke**α**, $\mathbf{\sin\alpha} = \frac{\sin\theta}{L/R}$ [2]

Where, L is the Stroke, R is the Crankshaft radius, θ is the Maximum crank angle

The Tangential force on the crankshaft, $\mathbf{F_T = F_Q \ \sin(\theta + \alpha)}$ [3]

Radial force on the crankshaft $\mathbf{F_R = F_Q}\ \cos(\theta + \alpha)$ [4]

2.4.2 The connecting Rod

The Rod of the Engine was designed according to (Parkash, Gupta, and Mittal 2013).

Normal pressure (P) = $\mathbf{P_o \cos\theta}$ [5]

Where P_o is the Normal pressure constant

Mass of the Rod M = ρAL [6]

Fatigue analysis $\mathbf{F/A} \leq \partial_e$ [7]

Area of a connecting rod $A = 11t^2$ [8]

The least radius of gyration $\mathbf{K_{xx}}^2 = 3.18t^2$ [9]

$\mathbf{K_{yy}}^2 = 0.995t^2$ [10]

Height of the section = 5t [11]

Height of the big end = 1.25t [12]

Width of the section = 4t [13]

Maximum tensile force $\mathbf{F_{recip,total} = (M_{piston} + M_{conrod,recip})\ r.\omega^2.(\cos\beta + \lambda\cos 2\beta)}$ [15]

Inertial bending forces on the beam of the Rod $\mathbf{F_{beam} = 2/3\ M_{rodr}.\omega^2}$ [16]

Where $\mathbf{P_o}$ is the Normal pressure constant, A is the area of the rod, ρ the density of material for the rod and L is the length of the rod, ∂_e the endurance stress of steel, *F/A* fatigue stress on the rod, t the thickness of the rod, $\lambda = r/l$

2.4.3 The Piston

The engine piston was designed as follows, according to Rastogi, (2017)

The piston head (Grashoff's formula) $t_h = \sqrt{(3P.D^2)/(16\partial_t)}$ [17]

P – Maximum gas pressure, ∂_t – Tensile strength of the material of piston, D – Diameter of the piston

Thickness for optimal heat dissipation,$\mathbf{t_H} = \frac{H}{12.56K(T_c - T_E)}$ [18]

The heat flow through the piston head, H = C x HCV x M x BP [19]

H- Heat flow through the piston head, K is the thermal conductivity, T_c is the temperature at center, T_E is the temperature at the edge

Radial Thickness of Ring (t_1): $\mathbf{t_1} = D\sqrt{\frac{3P_w}{\partial_t}}$

P_w = Pressure of gas on the cylinder wall, D is the cylinder bore

The thickness of the rings $\mathbf{t_2} = 0.7\mathbf{t_1} \leq \mathbf{t_2} \leq \mathbf{t_1}$ [21]

Width of the Top Land, $\mathbf{b_1 = t_H}$ *to* $\mathbf{1.2t_H}$ [22]

Width of other Lands (b2), $\mathbf{b_2} = 0.75\ t_2$ to 3 mm [23]

Maximum Thickness of Barrel (t_3), $\mathbf{t_3} = 0.03B + b + 4.5$ [24]

b = radial depth of the piston ring groove $b = t_1 + 0.4$ [25]

Piston wall thickness towards the open end $t_4 = 0.35t3$ [26]

Piston pin diameter, $\mathbf{d_o} = 0.03B$ [27]

Inside diameter of piston pin, $d_1 = 0.6d0$

The stress acting in MPa on piston crown, $\partial_b = M_b/W_b = P_{zmax}(r_i/\partial)^2$ [28]

Where, $M_b = (1/3)\,P_{zmax}\,r_i^3$ is the bending moment, $W_b = (1/3)\,r_i\,\partial^2$ is the moment of resistance to bending of a flat crown, $P_{zmax} = P_z$ is the maximum combustion pressure, $r_i = [D/2-(s + t_1 + d_t)]$ is the crown inner radius,

Mean piston speed $S_p = 2LN$ [29]

2.4.4 The combustion chamber design

The combustion chamber of the engine was designed following Heywood,. (2018)

Compression ratio $r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c}$ [30]

V_d Is the displaced or swept volume and V_c is the clearance volume

Ratio of cylinder bore to piston stroke; $\mathbf{R_{bs} = B/L}$ [31]

Ratio of connecting rod length to crank radius R = l/a [32]

The cylinder volume V at any crank position θ is $V = \mathbf{V_c} + \frac{\pi B^2}{4}(l + a - s)$ [33]

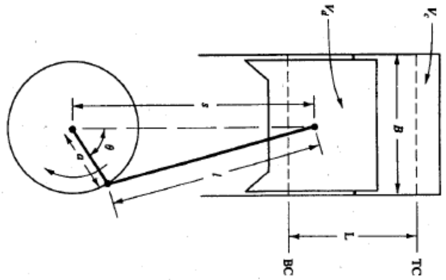

Figure 1: *Geometry of cylinder, piston, connecting rod and the crankshaft*

B – Bore, L- stroke, l- connecting rod length, a – crank radius θ – crank angle, s is the distance between the crank axis and the piston pin axis and is given by

$\mathbf{S = a\cos\theta + (l^2 - a^2\ \sin\sin^2\theta)^{1/2}}$ [34]

The above equation becomes $\mathbf{V/V_c = 1 + 1/2\,(r_c - 1)\,[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}}$ [35]

Combustion chamber surface area A at any crank position**θ**, $A = \mathbf{A_{ch} + A_p + \pi B(l + a - s)}$ [36]

2.5. The camshaft

The empirical relation according to (Ahamed, Shilpa, and Roshan 2019),

Diameter of camshaft = (0.16 x B) + 12.7 x 10^{-3} [37]

Width of cam (W_c) = (0.09 x B) + 6 x 10^{-3} [38]
F= Force of follower + force on the rocker arm [39]
Forces on inlet cam,
Force on rocker arm = Fs + Fa + $\mathbf{F_f}$ [40]

$$Where\ Fs = \frac{\pi}{4div^2\ x\ ps}$$

3. Results and Discussions

3.1 Tests on the locally innovated engine

Variable	Value (Idle state)	Value (On-load)
Speed, rev min^{-1}	1800	1000
Torque, Nm	3	5
Fuel consumption Liters per hour	1.58	2.4
Power HP	5	10
Emissions, °C	169	200

Table 2: Summary of the test results of the locally innovated single piston diesel engine

The locally innovated engine was designed to operate at an rpm of 1800 rpm to 2000 rpm producing a brake power no less than 13 HP, Torque of not less than 5 Nm and a fuel consumption of not more than 1 L every hour yet the actual output was less than the design expectations as shown in Table 2.

3.2 Major Engine Components and Their Dimensions

3.2.1 The piston

The piston of the locally innovated single-piston diesel engine was constructed using locally available Aluminium (scrap) on the market in Uganda. The piston of the single piston diesel engine had specifications and measurements as per Table 3

Variable	Dimension
Thickness of the piston head (H), mm	10
Heat flow H, KJ/s	5.13
Thickness of the piston t_h, mm	26.6
Radial thickness of the ring t_1, mm	2.9
Maximum thickness of barrel t_3, mm	10.83
Piston wall thickness towards the open end	3.79
Piston pin diameter, mm	27.3

Table 3: Specifications for the Piston of the Locally Innovated Engine

3.2.2 The Crankshaft

The crankshaft of this engine was purchased from the market after assessing minor considerations and relationships with the constructed piston and the connecting rod that was being used. This was done using a trial and error method which left many shafts destroyed by failure. This lacked any form of preliminary calculation to aid in knowing the requirements of the crankshaft to be purchased. However, there is no proof that the crankshaft currently in the engine can withstand the pressure for the entire life of the engine. The biggest issue with the shaft was with balancing the crank web with the rest of the reciprocating weight of the engine. In the locally constructed engine, balancing was done by a try-and—and-error method through trying to weld onto the web different weights and then running the engine severally. This was done until a uniform and balanced rotation was achieved.

Specification	Dimension
Crank pin radius, mm	29
Thickness of the crank web, mm	22.8
Length of the crank pin, mm	26.6
Crankshaft diameter, mm	33
Crankpin journal width, mm	30

Table 4: Dimensions for the used crankshaft in the locally innovated single piston diesel engine

3.2.3 Connecting rod

The rod was purchased from the market after several attempts to come up with an aluminum cast rod built at Kevoton workshop but yielded little result. Several failures occurred due to fatigue and buckling which weakened the aluminium rod in a few operational hours. This could have been due to limited studies carried out on failures that could happen and taking little precautions on the reciprocating mass of the engine. The used rod therefore is made of fogged steel. Its dimensions are as per Table 5.

Specification	Dimension
Thickness of the rod, mm	6.2
Height of the section, mm	26.5
Width of the section, mm	21.3
Height of the big end, mm	6.8
Height of the small end, mm	4.7
Mass, kg	0.6
Bigger radius mm	29
Small radius mm	27.3

Table 5: Dimensions of the Connecting Rod in the Locally Innovated Engine

3.2.4 Camshaft

The camshaft was obtained from the market with specifications meant to operate in line with those of the other parts of the engine. Due to the limited engineering knowledge in making the selection, it could not be concluded that the camshaft chosen for the engine was that meant to run a 13hp engine, although later proven to be a 10hp engine due to the errors in the construction. Dimensions of the camshaft of the engine locally developed at Kevoton are in Table 6

Specification	Dimension
Camshaft width, mm	15
Camshaft diameter, mm	18
Cam height, mm	41.3
Base circle diameter, mm	40
Total lift of cam, mm	7.65
Journal diameter, mm	70

Table 6: Camshaft dimensions of the locally innovated engine

3.2.5 Cylinder Head

The cylinder head of the engine was locally developed at Kevoton workshop through casting of aluminium alloy (6061). On this head, the injector inlet and exhaust valves were put. According to this head, the injector manifold is suitable for a 20hp engine. This created a lot of unburnt fuels being exhausted due to the excess fuel that was being injected into the combustion chamber. The angle of fuel injection was not also well calculated since it is about 80□which is quite high for fuel injection. The exhaust manifold is not smooth enough to clear the exhaust gases of the engine. This causes a lot of soot to remain in the manifold, which could even cause blockage within the manifold hence interrupting with combustion of the fuel. The cylinder head was made leaving a larger clearance of about 8mm after fitting in the gasket, which reduced the compression ratio, hence delaying ignition.

3.2.6 Flywheel

The flywheel of the single piston diesel engine was purchased from the available cast iron flywheels on the market. This flywheel underwent some machining to be fit for the recommended work of balancing the weights of the reciprocating mass and restoring the kinetic energy of the crankshaft. This flywheel is big and may not be applicable for automotive equipment since it requires a lot of space to be able to operate.

3.3 Development of Design of the Major Engine Components

After the preliminary studies carried out on the locally innovated engine at Kevoton, various engine components were designed according to various standards to serve its purpose. A number of parameters were put in consideration together with the engineering product development procedures. This followed performing the preliminary designs, Conceptual Design, geometrical design of the engine, and material selection. According to Duffy., (1912), the design process may undergo the following stages of production; Concept Development, Scheme Design, Detail Design, Manufacture, and finally Post-production. Now this is not different from the procedure being used in this research, for example, during the scheme design, activities like drawing are generated in the interest of the client which is also used as in the study during the preliminary design. However, according to Wiley (2005), the design process follows the following processes; understanding the need, analysis of the problem, statement of the problem, conceptual design, selected schemes, embodiment of schemes, detailing and working drawings. According to Eder (2010), the design process follows stages like; planning the product, design specifications and planning preliminary layout, release for detailing, manufacture, test, and develop prototype. All these lead to a successful product.

3.4 Geometrical Design of the Engine

Major engine components were designed geometrically following the formulae seen in section (2.4)

3.4.1 The Engine Block and Its Components

The required engine was supposed to have a brake power of 13 HP and a revolution of approximately 2000rpm, the length of the connecting rod was 160 mm. Therefore, calculation of the combustion chamber and other components of the engine;

The brake power (P_{brake}) therefore in becomes 13 HP

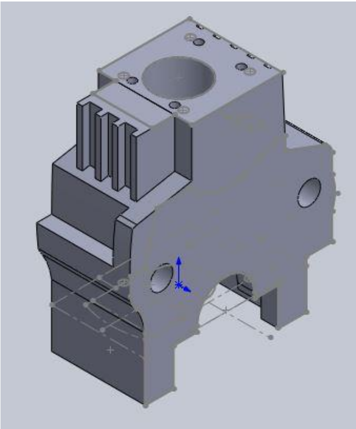

Figure 2: 3d of the Engine Block of the Engine

The brake mean effective pressure (BMEP)

According to Moreira (2015), the BMEP of an ICE can be calculated through considering the following formula;

$$(\mathbf{P_{brake}}) = \frac{\mathbf{BMEP\ V_d\ N}}{\mathbf{n_c}} \quad [41]$$

Where, V_d is the displaced volume, N is the rpm, and n_c is the number of revolutions per cycle, since rpm is 2000 and the engine is four-stroke, then n_c=1000

$V_d = \frac{\pi B^2}{4}$ L, but from the engine specifications, bore (B) =91 mm, stroke (L) = 106 mm and the N = 2000

So $V_d = \frac{\pi\, x 9.1^2}{4}$ x10.6 = 689.4 cc

Substitute V_d into equation $(\mathbf{P_{brake}}) = \frac{\mathbf{BMEP\ V_d\ N}}{\mathbf{n_c}}$ [41]

$9.7 = \frac{\text{BMEP X } 0.6894 \text{ X } \frac{2000}{60}}{1000}$

Therefore, BMEP of the engine = 422.1 kPa

3.4.2 Cylinder Design

Consider (Fig 1), the calculations for the cylinder geometry is as below;

Compression ratio

Consider equation [30], where swept volume, $V_c = \frac{\pi \text{ x } 9.1^2 \text{x } 0.5}{4}$ = 32.52 cc

$$r_c = \frac{689.4+32.52}{32.52} = 22$$

So the engine compression ratio is 22:1, which is in the range of diesel engines (14 - 24) to produce an ignition.

Crank radius was calculated as,

$$a = \mathbf{stroke(L)}/\mathbf{2} \quad [42]$$

$$= 10.6/2 = 5.3 \text{ cm}$$

Ratio of cylinder bore to piston stroke

$$\mathbf{R_{bs}} = \mathbf{B}/\mathbf{L} \quad [43]$$

$$= 91/106 = 0.86$$

Ratio of connecting rod length to the crank radius according to Ratio of connecting rod length to crank radius R = l/a [32]

16⁄5.3 = 3.02

3.4.3 Piston Design

The piston was designed after analysis from a number of authors like (Koszałka 2016) and (Sonar and Chattopadhyay 2015). Details of the design are as follows;

Thickness of Piston Head (t_H)

The piston thickness of the piston head is calculated using the following Grashoff's formula as detailed in equation 17

P – Maximum pressure in N/mm² that aluminium can withstand is 8 N/mm²,

∂_t – Allowable tensile stress for the material of the piston (Al Annealed 6061 with zero temper, ∂_t = 124Pa

$$t_h = 91\sqrt{(3 \text{ x } 8)/(16 \text{ x } 124.4)} \text{ so } t_h = 10 \text{ mm}$$

Heat flow through the Piston Head (H):

The heat flow through the piston head was calculated using the formula in equation

The heat flow through the piston head, H = C x HCV x M x BP [19]

C- Constant heat supplied to engine (ratio of the heat absorbed by the piston to the total heat developed in the cylinder about 5% or 0.05),

HCV= 44800, K= 205 W/mK for Aluminium,

Te = Temperature at edges of piston head in °C factor of safety of 5

The required engine consumes 1L of fuel every 1hour, this is equivalent to 2.777778x10^{-7} m^3/s. this is the volumetric flow rate of the fuel. Now the mass flow rate is (density of diesel x the volumetric flow rate of fuel) = 849 x 2. 777778x10^{-7} = 2.36x10^{-4} kg/s

H = 0.05 x 44,800 x 2.36x10^{-4} x 9.7 = 5.13 kJ/s

But basing on heat dissipation, the thickness of the piston is given by;

$$\mathbf{t_h} = \mathrm{H}/12.56 \text{ x K x } (T_c - T_e)$$

But $(T_c - T_e)$ = 75°C for Aluminium alloy

$= 5.13 \text{ x } 1000/12.56 \text{ x } 205 \text{ x } 75$ =0.0266 m equivalent to 26.6 mm

Radial Thickness of Ring (t_1):

Using Radial Thickness of Ring (t_1): $t_1 = D\sqrt{3P_w/\partial_t}$, Where, P_w = (nearly taken as 0.025 MPa to 0.042 MPa).

$$\text{So } t_1 = 91\sqrt{\frac{3 \text{ x } 0.042}{124.4}} = 2.9 \text{ mm}$$

Axial Thickness of Ring

$\mathbf{t_2} = 0.7\ \mathbf{t_1}$

[45] = 0.7 x 2.9 = 2.03 mm

Width of the Top Land b_1

The width of the top land varied from

$\mathbf{b_1} = \mathbf{t_H}$ to 1.2 $\mathbf{t_H}$

[46]

So b_l becomes = 10 mm

Width of other Lands (b_2)
$\mathbf{b_2}$ = 0.75 t_2 to 3 mm, so $\mathbf{b_2}$ = 3 mm [47]
Maximum Thickness of Barrel (t_3)
$\mathbf{t_3}$ = 0.03B +b + 4.5 [48]
b = radial depth of the piston ring groove, b = t_1 + 0.4 = 2.9 + 0.4 = 3.6 mm
So t_3 = 0.03 x 91 +3.6 +4.5, t_3 = 10.83 mm
Piston wall thickness towards the open end
t4 = 0.35 $\mathbf{t_3}$ [49]
= 0.35 x 10.83, t_4 = 3.79 mm
Piston Pin Diameter
d_o = 0.03B = 0.03 x91 = 27.3 mm
Inside diameter of piston pin
$\mathbf{d_1}$ = 0.6 d0 [50]
$\mathbf{d_1}$ = 1.638 mm

Theoretical Stress Calculations
Considering equation The stress acting in MPa on piston crown, $\partial_b = M_b / W_b = P_{zmax}(r_i/\partial)^2$, P_{zmax} = is the maximum combustion pressure = 11.96 MPa
r^i= [D/2-(s + t_1 +d_t)] is the crown inner radius, m;
However, the thickness of the sealing part s = 0.05B = 0.05 x 91 = 4.55 mm
Radial clearance between the piston ring and channel: d_t =0.0091 m
Therefore, r_i= [0.091/2-(0.00455 +0.0029 + 0.0091)] = 0.029 m
The thickness of the piston crown ∂ = (0.08 to 0.1) x B = 0.085 x 91 = 7.7 mm
So substituting in the equation (28) above, $\partial_b = 11.96\,[(\frac{0.029}{0.0077})^2)]$ = 70.9MPa.
so the required theoretical stress obtained from the calculation is 70.9 MPa.
However, for the design to be failsafe, the value of the theoretical stress must be less than the allowable stress. Since the previously obtained allowable stress was 124.4MPa, which is greater than the obtained stress 70.9Mpa. This means that the design is safe, that is, obtained stress < allowable stress
Mean piston speed
Consider the equation Mean piston speed Sp = 2LN [29]
= 2 x 0.106 x 2000/60 = 7.07 m/s

Inertia Force of the Piston
The inertia force on the piston unit was calculated according to Zheng et al., (2013);
$\mathbf{F_1 = m_p\, R\omega^2}$ $(1+\lambda)$ [51]

Where, m_p is the mass of the piston, piston pin and the oil ring, R is the crank radius, ω is the angular velocity, λ is the crank link ratio, $\lambda = l/R$ = 160/53 = 3.02, so
F_1 = 1.055 x 53 x 209.44^2 (1+3.02) = 9859.93 N

Pressure on the Piston
Pressure due to the combusted gas acting on the top of the piston which enables it to move downwards. There is also a pressure that acts on the piston at the bottom hole of the piston pin due to the force acting on the piston pin.

Figure 3: CAD drawing of the piston

Computed values for the piston		Results from literature with Author names		
Specification	**Dimension**	(Gopal et al. 2016)	(Sonar and Chattopadhyay 2015)	(Prasad, Achari, and Goud 2016)
Heat flow through H, kJ/s	5.13	2.58	0.6527	4.22
Thickness of the piston t_h, mm	26.6	15.57	24	9.58
Radial thickness of the ring t_1, mm	2.9	2.69	3.0	1.63
Maximum thickness of barrel t_3, mm	10.83	14.35	10.7	8.03
Piston wall thickness towards the open endt_4, mm	3.79	5.023	2.675	2.81
Piston pin diameter d_0, mm	27.3	17.04	24	13
Theoretical stress∂_b, MPa	70.9	139.02	82.9	92.31

Table 7: Piston Specifications

3.4.4 The crankshaft
The forces acting on the crankshaft of the engine include;

The forces acting on the crankshaft of the engine include;
Thrust Force in the Connecting Rod

FQ = Fp⁄cos$_{\beta}$ [52]

Where $_{\beta}$ is the angle of inclination of the connecting rod on the line of stroke, F_p is the force acting on the piston

Force acting on the piston

F_p = Area of bore x Max combustion pressure

The density of diesel at say 15.556°C is 849kg/m^3, Molecular weight of diesel $C_{12.3}H_{22.2}$ = 170g/mole

Mass of fuel in the combustion chamber = density of diesel x volume of cylinder

$V = \frac{\pi \times 0.091^2 \times 0.106}{4} = 6.89 \times 10^{-4}$, Mass = 849 x 6.89 x 10^{-4} therefore, M = 0.585 kg

From gas laws,

PV = mRT [53]

$P = {[0.585 \times \frac{8.3143}{0.17} \times 288]}/{6.89 \times 10^{-4}}$ = 11.96 MPa, So Fp = $[\frac{\pi 0.091^2}{4}] \times (11.96 \times 10^6)$= 77786.4 N

But according to Fig 1,

$\cos\theta = \frac{53}{160}$, so θ = 71.7 °, and $_{\beta}$= 18.3 °

The thrust force $F_Q = {77786.4}/{\cos 18.3}$ = 81930 N

The tangential force on the crankshaft

Applying equation The Tangential force on the crankshaft, $\mathbf{F_T} = \mathbf{F_Q}\,\mathbf{sin(\theta + \alpha)}$ [3]

, $\mathbf{F_T}$ = 81930 x $\mathbf{sin(71.7 + 18.3)}$ = 81930N

The radial force acting on the crankshaft,

According to equation Radial force on the crankshaft $\mathbf{F_R} = \mathbf{F_Q}\,\mathbf{cos(\theta + \alpha)}$ [4]

, the radial force F_R becomes = 81930 x$\mathbf{sin(71.7 + 18.3)}$ = 81930 N

Reactions at the bearing (1&2) due to tangential force is given by

${F_T}/{2} = {81930}/{2}$ = 40965 N

Reactions at the bearings (1& 2) due to radial force

${F_R}/{2} = {81930}/{2}$ = 40965 N

Selected cast steel for the crankshaft and crankpin has allowable bending stress of 75N/mm^2 and an allowable sheer stress of τ = 35N/mm^2

The total resultant load on the crankshaft was calculated according to Fatemi (2014) as;

$F = \int_{\frac{-\pi}{3}}^{\frac{\pi}{3}} \mathbf{P_o}\ \mathbf{\cos\emptyset\ rtd\emptyset} = \mathbf{P_o r_t \sqrt{3}}$ [53]

However, force F = 81930N, r is the radius of the crankpin, t is the length of the crankpin

So $P_o = {F}/{r_t\sqrt{3}} = {81930}/{[53 \times 26.6 \times \sqrt{3}]}$ = 33.6 MPa

Crankpin journal width = 0.35 x 91 = 31.9 mm, Web thickness = 0.25 x 91 = 22.8 mm

Analysis of the Stress History at Different Locations at the Engine Speed of 2000 Rpm

Forces applied to the crankshaft cause bending and torsion. Fig 1 demonstrates the positive directions and local axis on the contact surface with the connecting rod. Fig 4 shows the variations of bending and torsion loads plus the magnitude of the total force applied to the crankshaft as a function of crankshaft angle for the engine speed of 1800 rpm. The maximum load happens at 355 °C where combustion also takes place. The force acting on the crankshaft at this moment is bending force as the direction of the force is exactly towards the center of the crank radius (Fy = 0 in Fig 4)

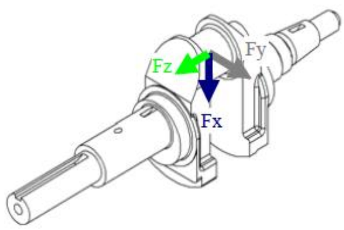

Figure 4: Crankshaft Stress Analysis

F_x		F_y		F_z	
Stress, MPa	Crankshaft angle (°)	Stress, MPa	Crankshaft angle (°)	Stress, MPa	Crankshaft angle (°)
-30	0	0	0	40	0
-10	70	10	30	10	45
-20	90	0	45	45	90
-50	180	-25	90	45	135
0	290	0	180	50	180
175	360	50	270	51	270
0	405	75	315	175	360
-40	495	0	360	10	405
-45	540	-40	360	45	495
-5	630	-25	405	35	550

Table 8: Variations of stress with the crankshaft angle of rotation

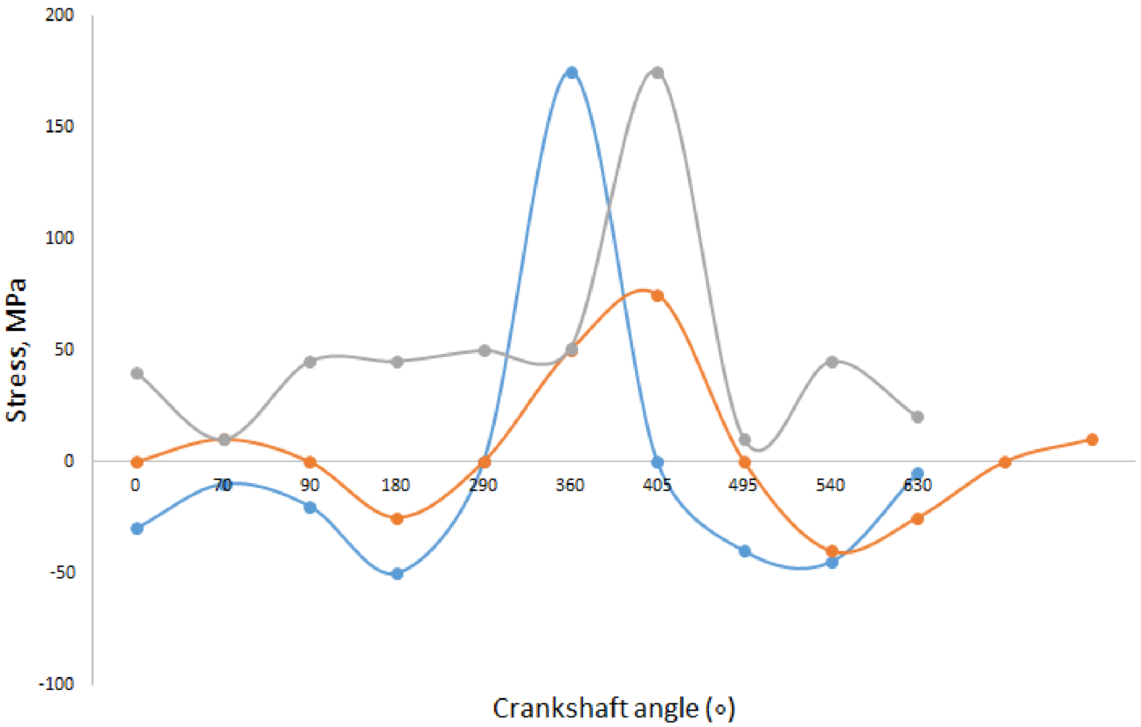

Figure 5: Variations of Stress with the Crankshaft Angle of Rotation

Computed values for the crankshaft		Results from literature with Author name
Specification	**Dimension**	(Jayachandraiah 2013)
Crank pin radius, mm	18.2	22.6
Thickness of the crank web, mm	21.3	21.336
Bore diameter, mm	91	53.73
Length of the crank pin, mm	34.4	43.73
Maximum pressure, MPa	11.96	3.5
Crankshaft diameter, mm	34.9	37.925
Crankpin journal width, mm	31.9	

Table 9: Dimensions of the crankshaft

3.4.5 The Camshaft

The empirical relation according equations

Diameter of camshaft = (0.16 x B) + 12.7 x 10^{-3}

Width of cam (W_c) = (0.09 x B) + 6 x 10^{-3}

[38], F= Force of follower + force on the rocker arm

[39] and Force on rocker arm = Fs + Fa + F_f

[40], the;

Diameter of camshaft = (0.16 x B) + 12.7 x 10^{-3} = (0.16 x 91) + 12.7 x 10-3 = 14.57 mm

Base circle = diameter of camshaft + 3 (mm) = 14.57 + 3 = 17.57 mm

Width of cam (W_c) = (0.09 x B) + 6 x 10-3 = (0.09 x 91) + 6 x 10-3 = 8.196 mm

Forces, F= Force of follower + force on the rocker arm

Forces on inlet cam, Force on rocker arm = $F_s + F_a + F_f$, Where Fs = π/(4div² x ps), Fs is the spring force, Ps is the maximum pressure < atmospheric pressure = 0.01N/mm2
So div = dip + 2(0.05dip + 0.07dip), But dip = 0.0197 + 2(0.05 x 0.0197) = 0.022m = 22 mm
So div = 0.022 + 2(0.05 x 0.022 + 0.07 x 0.022) = 0.0273m = 27.28 mm

$$\text{Therefore, Fs} = \frac{\pi}{4 \times 27.3^2 \times 0.01} = 1.05 \times 10^{-1}\ \text{N}$$

3.4.6 The Connecting Rod

The connecting rod was designed considering forces like, Force on the piston due to gas pressure, force due to inertia of the connecting rod and reciprocating mass, force due to friction of the piston rings and the piston as analyzed below;

Force due to gas pressure,
The maximum force due to gas pressure;
F= 81930 N
While designing the connecting rod, it was assumed that the rod does not fail under either fatigue or buckling due to force F on the piston. Mass of the rod is given by equation Mass of the Rod M=ρAL [6
(ρ_{steel} = 7850 kg/m³, L = 160 mm
Fatigue Analysis
Fatigue was analyzed according to equation **Fatigue** analysis $F/A \leq \partial_e$ [7
Where, ∂_e = 290 MPa is the endurance stress of steel, F/A the fatigue stress on the rod but F/A = ∂_e (maximum stress on the rod)
F/A = ∂_e (for maximum stress), Therefore substituting A = F/(∂_e) into
A = 81930/290 = 282.5 mm²

$$M = \rho \, {}^{F}\!/_{\partial_e} L$$

$$\text{Then } M = 7850 \times \frac{81930}{290 \times 10^6} \times 0.16 = 0.35\text{kg}$$

Area of a connecting rod may also be
A = 11t²
282.5 = 11t², so t = 5.06 mm
Buckling Analysis
Buckling load F_r = maximum explosion x factor of safety
[(81930)/290] x 5 = 1412.5 N
The least radius of gyration, mm
$\mathbf{K}_{xx}{}^2$= 3.18t2, $\mathbf{K}_{xx}$= 81.4 mm
$K_{yy}{}^2$ = 0.995t², K_{yy} = 25.5 mm
Height of the section = 5t = 5 x 5.06 = 25.3 mm
Height of the big end = 1.25t = 6.3 mm
Width of the section = 4t = 20.24 mm
Height of the small end = 0.9t = 4.55 mm
The Maximum tensile forces on the connecting rod is given as;
As per equation Maximum tensile force $F_{recip,total} = (M_{piston} + M_{conrod,recip})\, r.\omega^2.(\cos\beta + \lambda \cos 2\beta)$ [15
$F_{recip,total}$ = (0.585 + 0.35)5.3x10⁻³ x 209.442 (cos18.3 +5.3/160 cos2x18.3) = 212.2N
Inertial bending forces on the beam of the connecting rod
This arises due to triangular distribution and is calculated as shown in equation Inertial bending forces on the beam of the Rod $F_{beam} = 2/3\, M_{rod}\, r.\omega^2$ [16]
F_{beam}= 2/3 x 0.35 x 0.0053 x 209.44² = 54.25 N

Specification	Dimension
Thickness of the rod, mm	5.06
Height of the section, mm	25.3
Width of the section, mm	20.24
Height of the big end, mm	6.3
Height of the small end, mm	4.55
Mass, kg	0.35
Bigger radius, mm	18
Small radius, mm	13.5
Bending force, N	54.25
Maximum tensile forces, N	212.2

Radiator

Table 10: Specifications of the Connecting Rod

The exit temperature of hot fluid from the radiator (T_2) and the exit temperature of cold fluid from the radiator T_3, was determined according to Subbiah & Harish (2016);
Specific heat of cold fluid C_{pc} = 1.005 KJ/kgK, mass flow rate of cold fluid m_c = 4.63 kg/s, Specific heat of hot fluid C_{ph} = 4.718 KJ/kgK, mass flow rate of hot fluid m_h = 1.09 kg/s
Specific heat ratio
Specific heat capacity C_{cold} = mc x C_{pc} = 4.63 x 1.005 x 10³ = 4656 W/K
Specific heat capacity C_{hot} = m_h x C_{ph} = 1.09 x 4.718 x 10³ = 5173 W/K

$$C_{min} = 4656\ \text{W/K},\ C_{max} = 5173\ \text{W/K},\ C = {}^{C_{min}}\!/_{C_{max}} = 4656/5173 = 0.9$$

Assume the effectiveness of the cross flow heat exchanger ε = 0.128

$$0.128 = {}^{(T_1 - T_2)}\!\big/_{(T_1 - T_3)},\ 0.128 = {}^{(90 - T_2)}\!\big/_{(90 - 25)},\ 8.32 = 90 - T_2$$

T_2 = 81.68 °C, the exit temperature of hot fluid from the radiator
Heat lost by the hot body = Heat gained by cold body
$m_h \times C_{ph} \times (T_1 - T_2) = m_c \times C_{pc} \times (T_3 - t_1)$
$1.09 \times 4.187 \times (90 - 81.68) = 4.63 \times 1.005 \times (t_2 - 25)$
T_3= 33.16 °C, exit temperature of cold fluid from the radiator

Description	Radiator design
Air inlet temp, T_2	25 °C
Inlet temp, T_1	90 °C
Air outlet temp, T_3	33.16 °C
Effectiveness, ε	0.128

Table 11: Radiator specifications

3.4.7 Cylinder Head

The cylinder head of the single-piston diesel engine was designed to handle fuel injection, exhaust emissions, plus allowing cooling water to pass through it. This houses the injector and exhaust valves that work at a rate that can handle a 13hp engine without releasing excess fuel into the combustion chamber as well as releasing the burnt fuel-air mixture without affecting the entire engine block.

Specification	Cylinder head design
Cylinder head height (mm)	60
Intake valve angle (°)	71.5
Exhaust valve angle (°)	70
Wall thickness area water jacket (mm)	5

Table 12: Specification for the Cylinder Head

3.4.8 Flywheel

The flywheel of the single piston diesel engine was designed with consideration of the weight that it has to balance while the engine is operating. Kinetic energy stored in the flywheel according to Hoag & Dondlinger (2016) is obtained as;

$$E = \frac{\mathbf{I\omega^2}}{\mathbf{2}} \quad [55]$$

Where I is the polar moment of inertia, ω is the rotational velocity

$$\text{But } I = \frac{\mathbf{1}}{\mathbf{2}}\mathbf{mr^2} \quad [56]$$

m = mass of the reciprocating parts = (mass of piston = 0.585 kg and its components + mass of the connecting rod = 0.35 kg + mass of crankshaft)
But mass of each oil ring = 15 g x 3 =45 g, mass of piston pin = 75 g
M = 0.585 + 0.12 + 0.35 + 11.5 = 12.56 kg
Radius of the flywheel r = 0.1 m
I = 0.5 x 12.56 x (0.1) ^2
$I = 6.3 \times 10^{-2}$ kgm^2
Since the maximum rpm = 2000, then $\omega = 2\pi \times 2000/60 = 209.44$ rad/sec
So the kinetic energy stored in the flywheel, $E = [\frac{6.3 \times 10^{-2} \times 209.44^2}{2}] = 1376.8J$
According to Shigley (1996) a certain amount of fluctuation in shaft speed will not cause harmful torques or reduce the usefulness of a machine. On designing the flywheel of the single piston diesel engine, such considerations were looked at. So the coefficient of speed fluctuation $\mathbf{C_s}$ is defined as

$$\mathbf{C_s} = \frac{\boldsymbol{\omega_{max} - \omega_{min}}}{\boldsymbol{\omega_{Avg}}} \quad [57]$$

Where $\boldsymbol{\omega_{max}}$ = 2000 rpm, $\boldsymbol{\omega_{min}}$ = 1800 rpm
$C_s = \frac{2000-1800}{1900}$ = 0.11(moderate variation)
The mass moment of inertia J of the flywheel was obtained from;

$$J = \mathbf{E}/[\boldsymbol{\omega_{avg}^2}]\mathbf{C_s}$$

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$= 1376.8/[198.9675^2 \times 0.11]$, The mass moment of inertia for the flywheel J = 0.32 Ns^{2}m

Section	Sizing, mm
Containment material	Grey cast iron
Flywheel rim outer diameter	425
Flywheel inner Diameter	330
Containment inner radius	285
Containment thickness	60

Table 13: Subscale Flywheel parameters

4. Material Selection

Material for building an engine requires selection according to the ISO standards and guidelines for internal combustion engines plus other material selection standards from literature. Major components of the engine that were constructed at Kevoton workshop were put under various material analyses. This went under various material tests of compression, tensile strength, and shear stress and then obtained value compared to that recommended by ISO. Factors that guided the material selection process included; minimizing (cost, weight, volume, maintenance and repair, environmental impact) and maximizing (structural efficiency, fatigue life, damage tolerance, impact damage resistance, durability and operating life),

4.1. Engine Block

The engine block was constructed at Kevoton through sand casting and machining was also done by the same team to obtain a compatible block for housing the reciprocating parts of the engine. Aluminum alloy (Al 6061) was used in construction of the engine block since it is available, low weight and easily releases heat. Materials of aluminum obtained from the Ugandan market were tested to see whether they conform to the standard material specifications as recommended by ISO. A sample was cut from the block measuring (mm) 6.9 x 26.7 with an area of 184.23mm^2 and taken for a tensile test and the results are summarized in Table 7. This was done to relate it with the recommended figures obtained by other authors for engine block design.

Because of the shape of the stress-strain curve, some tensile properties in principle were determined with a higher degree of precision than others, for example, the upper yield strength R_{eH} was only dependent on the tolerances for measurement of forces and cross sectional area while the proof strength R_p was dependent on the force, strain (displacement), gauge length and the cross-sectional area.

4.2 Experimental Set Up For Tensile Strength Tests

The set up plus sample preparations have been explained before in section (2.2). These gave results of tensile strength, compression, and chemical composition of the material used in engine construction

Results by from literature review		Results from tests performed on the various components casted at Kevoton		
Composition	Tensile Strength [MPa] Literature	Component tested	Obtained result UTS [MPa] Obtained	Variation from other Authors
		Engine Block		
Al 6061	128	1	110	14%
Al 6061 +6%SiC	150	2	154	2.6%
Al 6061 + 6%SiC + 3%Others	141	3	132	6.4%
		Chain cover		
Al 6061 + 6%SiC+6% Others	148	1	124	16.2%
Al 6061 +	156	2	280	79.5% better
6%SiC+9% Others	206	3	178	14%
		Crankshaft		
		1	460	
		2	530	
		3	432	

Table 14: Tensile Tests for the Engine Block, Crankshaft and the Chain Cover

Load 1 kN	Displacement 1 mm	Load 2 kN	Displacement 2 mm	Load 3 kN	Displacement 3 mm
10.2	0.275	0	0	10	0.25
89.3	1.225	115.9	1.275	105	1
80	1.25	102	1.3	90	1.3
95	2.775	120	2.3	102	2.225
110	3.75	155	3.5	132	3

Table 15: Tensile strength of the engine block

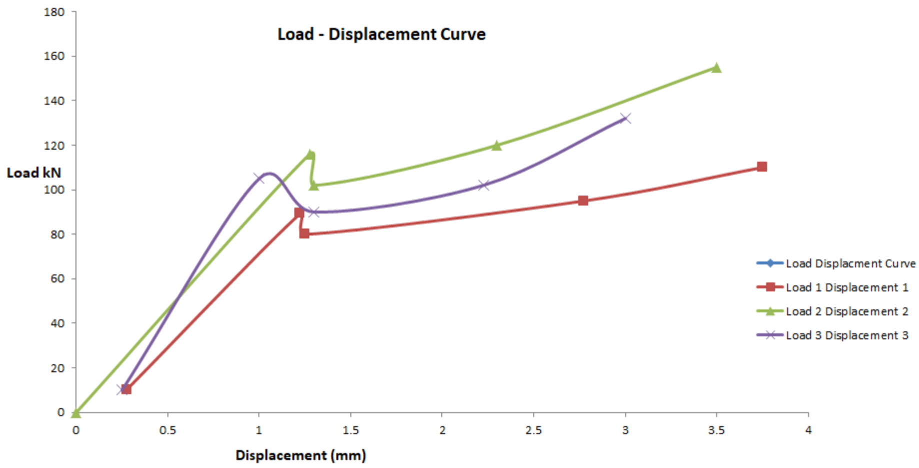

Figure 6: Load – Displacement curve for tensile strength of the engine block

Load 1 kN	Displacement 1 mm	Load 2 kN	Displacement 2 mm	Load 3 kN	Displacement 3 mm
0	1	0	0	0	0
85	8.9	150	5.1	105	6
75	10	125	6	90	6.5
105	13.75	235	9.1	175	7
		275	14		

Table 16: Tensile Strength of the Material of the Crankshaft

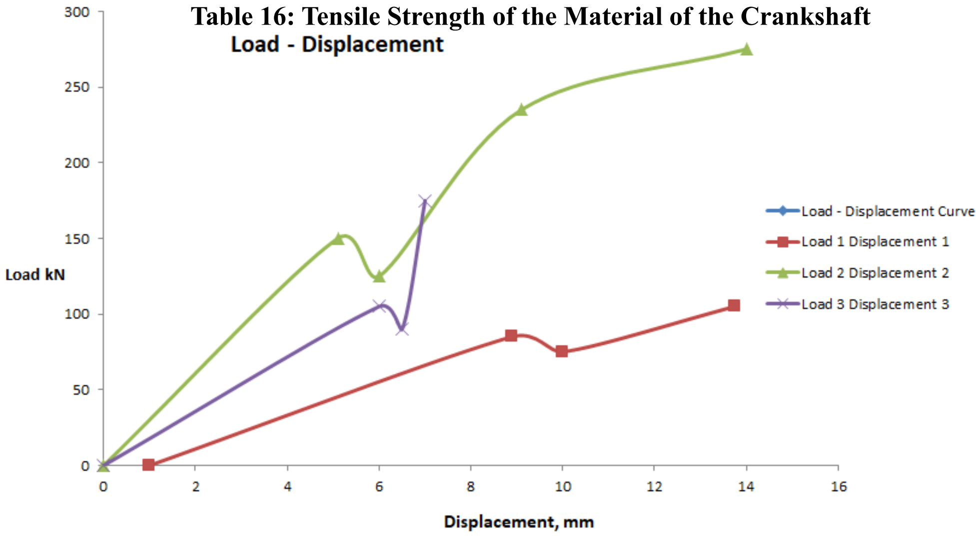

Figure 7: Load – Displacement Curve For the Tensile Strength of the Material Of the Crankshaft

Load 1 kN	Displacement 1 mm	Load 2 kN	Displacement 2 mm	Load 3 kN	Displacement 3 mm
20	0	40	0.3	0	0
420	1.1	460	1.4	390	2.9
400	2.4	500	2.5	400	3.6
430	4.8	520	3.7	430	4.8
460	7.6	-	-		

Table 17: Tensile Strength of the Material of the Chain Cover

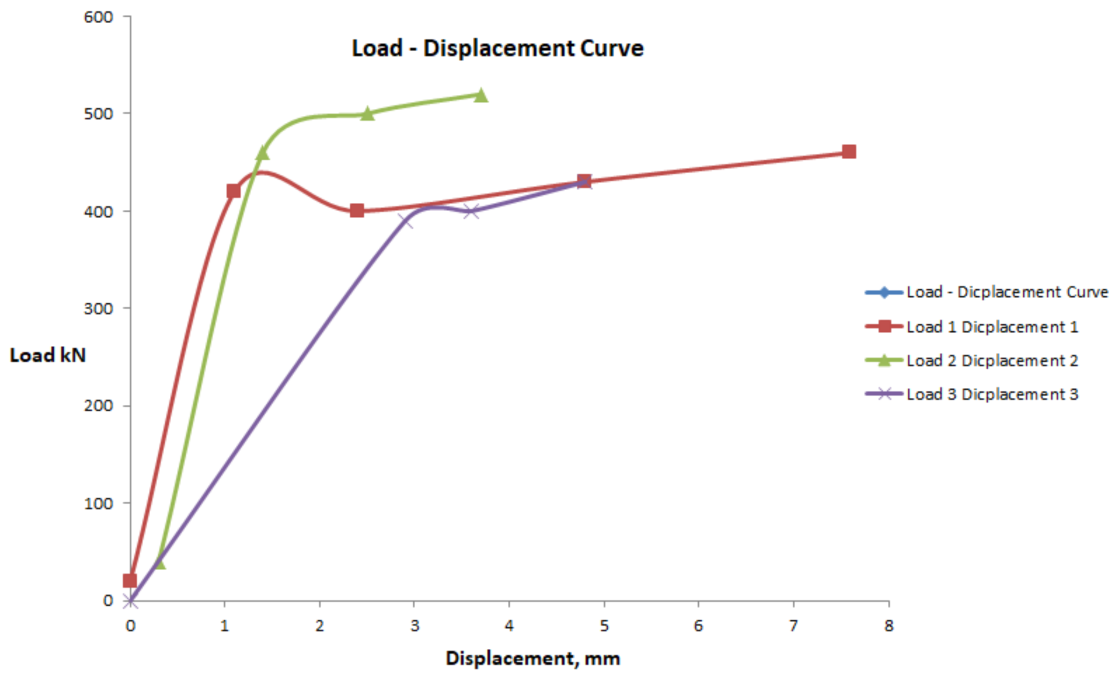

Figure 8: *Load – Displacement curve for the tensile strength of the material of the chain cover*

However, according to the results obtained from the tests performed on Aluminium and its alloys by Davis, (2001a), tensile strength for Al 6061 varies in ranges as per Table 18

Samples	Ultimate Tensile Strength Range [MPa] Literature	Obtained result UTS [MPa] Obtained value	Variation from other Authors
Al 6061	70 – 175	110	26%
Al – Cu – Mg – Si (3 – 6% Cu)	380 – 520	280	5.7%
Al – Mg (1 – 2.5% Mg)	140 – 280	132	17%
Al – Mg – Si, Heat treated	150 – 380	124	4%
Al – Cu – Ni – Mg	186 – 221	178	3%
Al – Si – Cu – Mg	159 – 269	154	

Table 18: Tensile Properties of Cast Aluminum 6061

While Guzmán et al., (2019) was investigating the mechanical behavior of 6061 Aluminium alloy welded by pulsed Gas metal arc welding (GMAW) with filler metals and heat treatment, the following results in Table 19 were obtained that actually as per the obtained results for the single piston diesel engine are related According to the International Journal of Pure and Applied Mathematics by Ajith et al.,(2018a) while enhancing the mechanical properties of aluminium 6061 metal matrix composite, the following results in Table 20 were obtained for the tensile tests of the various samples that are related to the obtained figures after testing the single piston diesel engine.

Samples	UTS (MPa) Literature	Obtained result from the tests UTS [MPa]	Variation from other Authors
1	68.164	-	-
2	70	-	-
3	110.151	132	19.9%
4	113.9	124	8.9%
5	64.938	-	
6	78.548	-	
7	101.922	110	8.9%
8	119.914	178	48.5%
9	153.932	-	-

Table 19: Mechanical Behavior of 6061 Aluminum Alloy Welded By Pulsed Gmaw

Samples	Yield strength [MPa]	Ultimate tensile strength [MPa]	Obtained UTS [MPa]	Variation from other Authors
Al 6061	48	93	110	15%
Al 6061 + 2% Graphite + 3% TiO_2	80	115	132	14%
Al 6061 + 2% Graphite + 13% TiO_2	83	119	124	4%
Al 6061 + 2% Graphite + 8% TiO_2	79	115	154	33.9%

Table 20: Mechanical Properties of Aluminum 6061 Metal Matrix Composite

4.3 Compression Tests on the Material of the Engine

Obtained results from the tests on the single piston diesel engine				Results from l
Specimen	Force (kN)	Extension (mm)	Compressive Strength Fc (MPa)	Compress Strength Fc (
Engine Block	444.25	8.04	320.8	329.2
	520.3	9.20	372.6	360
	582.3	10.8	412.7	429.7
Piston	193.60	2.00	304.6	298.5
	217.40	3.5	443.3	408
	249.10	4.4	286	254.2

Table 21: Compression Test Results for the Material of the Engine

In context to the compression results obtained in the performed tests on given components of the single piston diesel engine locally made in Uganda which were carried out under normal conditions as per Table 21. However other results obtained by given authors on aluminium 6061, there is a close relationship as per Table 22.

According to (Raj 2013), the compression test results obtained from different samples of aluminium alloy were;

Sample	Compression strength, MPa	Extension, mm
1	250	0.2
2	180	0.2
3	195	0.2
4	200	0.2
5	250	0.5

Table 22: Compression Test Results Obtained At the National Institute of Technology

According to the Journal of structure on the stress- strain compression of aluminum alloy at room temperature by Scari et al., (2014), the results obtained are as follows;

Samples	Compression strength, MPa	Extension, mm	F [N]
1	227.7	0.078	44525
2	254.2	0.027	49714
3	289.1	0.187	56542
4	294.4	0.198	57570
5	329.2	0.899	64384
6	360.0	0.921	70400
7	429.7	3.012	84035

Table 23: Compression Results of Aluminum

Load, kN	Extension, mm
0	0
195	2
250	3.5
245	4.4

Table 24: Compression Test for the Piston

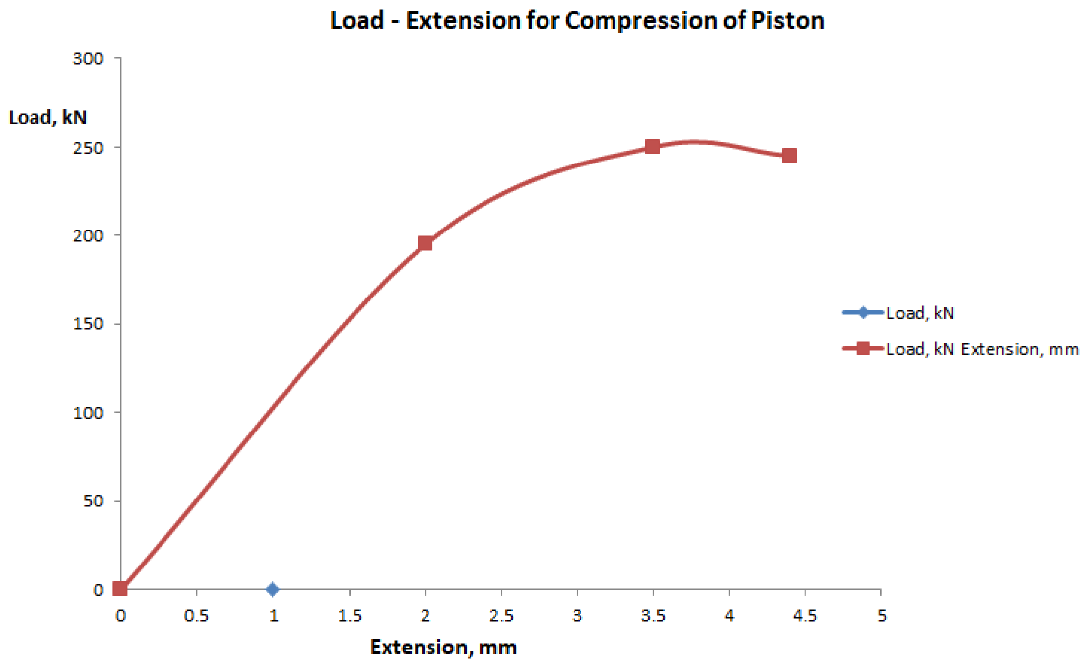

Figure 9: *Load – Extension Curve for a piston*

Load, kN	Extension, mm
0	0
440	8
520	9.1
590	11
580	13

Table 25: Compression of the Engine Block

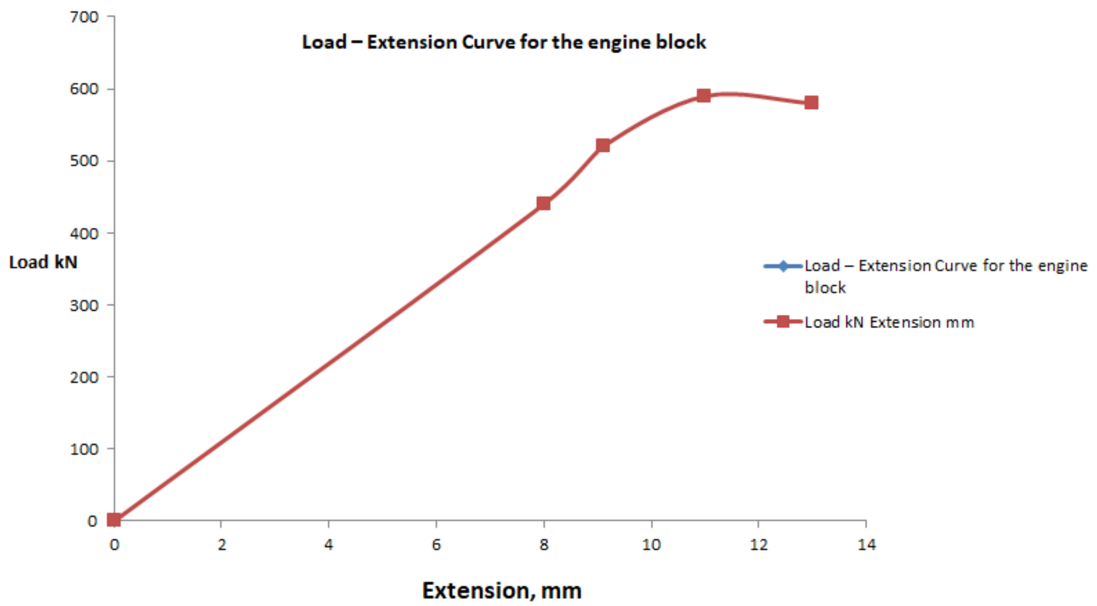

Figure 10: Load – Extension Curve for the engine block

4.4 Composition of the materials for the engine block, piston, cylinder head and the crankshaft

The compositions for the materials of the engine parts were determined as highlighted in section (2.2).

Specimen	Si	Mn	Zn	Pb	Cr	Mg	Ni	Fe	Cu	others	Al
Engine block	8.63	0.252	0.5	0.02	0.030	0.950	0.201	0.53	0.265	0.119	Bal.
Engine piston	7.34	0.55	0.07	0.52	0.005	0.980	0.415	0.15	0.112	0.211	Bal.
Cylinder head	6.93	0.0348	0.25	0.325	0.025	0.890	0.310	0.06	0.073	0.320	Bal.
Crankshaft	9.23	1.70	-	-	-	-	-	Bal.	-	1.82	-

Table 26: Chemical Composition of the Engine Material

Comparisons are made to the various results obtained by different authors; a similarity in the results that were obtained on the tests carried out on the materials of the single piston diesel engine to those obtained by various authors is witnessed. Yadav et al., (2017) carried out tests on Al 6061/ 15wt. % MoS2, Scari et al., (2014) conducted tests on AA6082-T6 Aluminum at room temperature, Heysler et al., (2001) conducted tests on AlSiMgCuNiFe, Ahamed et al.,(2019) performed tests on Al-6061 metal matrix composite, Anilchandra et al., (2017) tested the composition of AlSi9Cu3(Fe) and all these authors made deductions as per the results in Table 27.

Specimen	Si	Mn	Zn	Pb	Cr	Mg	Ni	Fe	Cu	others	Al
Al 6061/ 15wt. % MoS_2	3.47	0.325	0.445	0.096	-	0.023	0.082	0.85	0.62	0.119	Bal.
AA6082-T6	7.34	0.46	0.002	0.001	0.0005	0.24	-	0.21	0.017	0.023	Bal.
AlSiMgCuNiFe	7.21	0.50	0.139	-	-	0.41	0.51	0.43	0.37	0.056	Bal.
Al-6061 metal matrix	0.65	0.03	0.08	-	0.14	0.88	-	0.24	0.23	0.1	Bal.
$AlSi_9Cu_3$(Fe)	8.23	0.261	0.895	0.083	0.083	0.252	0.081	0.799	2.825	0.426	Bal.
$ASi_7Mg_{0.3}$	6.5	0.007	0.006		-	0.3	0.003	0.1	0.002	0.1	Bal.

Table 27: Chemical Tests Carried Out On the Various Aluminum Alloys For Engine Construction

5. Conclusion and Recommendations

This study focussed at reviewing the designs of the engine prroduuced at KML so as to come up with a reproducable design for future developments. Based on the results obtained from the tests carried out on this engine, it was realized that the choice of the injectors matters in the exhaust fumes produced by any engine as well as the combustion pressure, the purpose of the engine should be considered as well since it helps customizing the design to avoid it requiring modifications to have it perform the purpose, the engine should be built with the ability to be coupled for testing.

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