# Single-cylinder 125 cc Stepped Piston Engine for Mobility and Portable Power Generation Applications

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**ABSTRACT**

*Two-stroke engines is far simpler than four-stroke version from its physical perspective. For a given brake output, two-stroke is lighter, easier to work on, and provide higher power-to-weight ratio than the four-stroke, making it suitable for small platform applications. However a conventional two-stroke engine has a reputation for generating smoke and unburned fuel, a situation which does not meet many emissions regulations, now enforced around the world. Thus for many decades two-stroke engines were not favored, giving ways to four-stroke engines to dominate applications, especially for mobile power-generation purposes. In the quest to improve the potential of such an engine, a group of researchers from the Automotive Development Centre (ADC), Universiti Teknologi Malaysia (UTM) has come up with a 125 cc, air-cooled stepped-piston engine to demonstrate the higher power-to-weight ratio feature, apart from overcoming emission reduction. The engine is designed to mitigate the problem of mixture short circuiting, which is the major hindrance to combustion efficiency, and for this to happen they have incorporated a three-port stratification strategy onto the engine. This paper provides the overview related to the earlier work done to infuse the necessary features and highlights some of the performance features of this unique engine design.*

**Keywords**: *Two-stroke, engine, stepped-piston, prototype, design*

## 1. INTRODUCTION

The basic two-stroke engine operates by compressing the air-fuel mixture both in the crankcase and the cylinder, and moves the mixture around by making use of the vacuum and pressure created by the piston movement. The engine performs all functions in two strokes of the piston, (down and up) and the spark plug fires each time the piston is about to approach the engine's top dead centre (TDC). Valving is provided by the motion of the piston, sliding past port openings in the cylinder wall. The above four functions must be crowded into just two piston strokes. As the piston descends on the power stroke, at roughly half-stroke it begins to uncover a large exhaust port or ports, and exhaust gas begins to leave the cylinder. Approximately a quarter-stroke later, the piston will uncover a set of fresh-charge transfer ports. Fresh air and fuel have meanwhile been drawn into the crankcase, and the descent of the piston compresses this fuel-air mixture. As the transfer ports open, this mixture begins to jet into the cylinder through them.The problem that will arise are: even with the best-possible aiming of the transfer ports (keeping the entering fresh charge away from the exhaust port, and send it on a long looping path that fills the cylinder), some fresh charge does “short-circuit”, and flow directly out the exhaust. This loss of fresh charge to the exhaust is a direct consequence of having exhaust and transfer ports open at the same time. In a four-stroke, this is prevented by the use of mechanically operated valves.How much charge is short-circuited? A good four-stroke needs about 220 grams of fuel to produce one horsepower for an hour. However a well design two-stroke (with correctly aimed transfer ports and expansion chamber exhaust) needs roughly 280 grams per horsepower per hour. The difference is about 25 percent and is due to fuel lost directly out the exhaust port [1]. This was acceptable before 1980 because fuel was cheap and the big emissions reductions were then being achieved with cars.

Two-stroke road models were not offered after 1984 [2].Two-stroke engines especially for mobility applications was not in favor for several reasons. Firstly, they produced higher levels of hydrocarbon and carbon-monoxide exhaust emissions than four-stroke engines, and reducing those emissions cannot be readily piggyback on all the work that had been expended by the automotive industry on car engines. Secondly they also produced worse fuel economy due to problem mentioned earlier. But, third, and worst of all, they were perceived as “non-green,” smoke-emitting, image disasters. A conventional car engine takes four piston movements, or strokes, to go through intake, compression, combustion, and exhaust. In a two-stroke cycle, these stages are completed with just two piston movements, delivering twice as many power strokes per revolution and requiring fewer parts. But two-stroke engines tend to spew out more unburned fuel in the exhaust, which is why the four-stroke design became more common. Once thought too polluting, the two-stroke engine makes a comeback in advance of stricter fuel efficiency standards [3].

In the last five years there were many R&D activities and investments made to revivethe use of two-stroke engines due to its power density advantage. Several innovative solutionswere introduced to address the existing short-comings [4, 5, 6]. Some have proven to be successful via the infusion of elements such as gasoline direct injection, electronic actuation of valves and new materials. With these positive developments, it is now seems possible for the small two-stroke engines to be deployedin mobility platforms (e.g. range extender for EV, UAV) and auxiliary power units.

## 2.1 EARLY STEPPED PISTON DESIGN

Bernard Hooper Engineering (BHE) was the first to develop the engine featuring stepped piston engines [7]. This engine type has been the subject of extensive testing aimed at industrial, automotive, marine and aerial applications. Engines were developed in recognitionof the benefits offered, but with the essential attribute of durability. This is afforded by the crankcase isolation provided by the stepped piston.

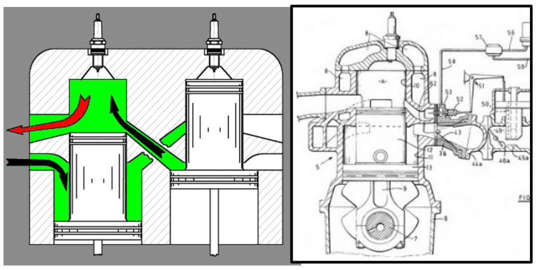


Figure.1 The early design of the stepped-piston engine [7].

Piston mass will be higher for this type of piston design. Thus its unique design means that for identical power cylinder bores the stepped piston will be 20% heavier than a conventional looped scavenged two stroke piston. This multi-section piston provides a flange stiffening effect resulting in thinner skirt sections. The mass can be reduced by composite methods. However the engine uses the two-stroke cycle and therefore with the absence of load reversals piston mass is not as critical as it is for four-stroke engines. This type of piston will provide improved load bearing and guidance resulting in low ring wear and reduced piston noise. The fresh charge and exhaust transfers are between two cylinders, as shown in Fig 1.

## 2.2 MODIFIED STEPPED PISTON ENGINE

A prototype two-stroke single-single engine with slightly different in the approach of mixture transfer design and piston dimensions was conceptualized. Using a three - port mixture transfer strategy, it was designed and developed at the Automotive Development Centre (ADC), *Universiti Teknologi Malaysia* (UTM). The primary objective of this work is to explore further research into how stepped-piston engine can mitigate current problems that restrict the role of conventional crankcase-scavenged two-stroke engines. Another aspect which was also looked into was on how to overcome the current problem of excessive use of lubricating oil (blend in air-fuel mixture) in which the subsequent effect was the excessive white smoke.

As mentioned earlier, stepped piston refers to the mating of two pistons to form an assembly. For this work the main piston has an aspect ratio (bore: height) of 1:1.5. The bigger piston coupled to the main piston, has an aspect ratio of 7.5:1. Each of this piston sections has a set of piston rings to prevent the crossing over of mixture from the upper half to the lower half of the pistons, or vice versa. The second piston has an aspect ratio of more than 2 and its only role is to inducing mixture, to draw in mixture(air-fuel-lube) when the piston undergoes expansion stroke. In other words, for one cycle, the cylinder will undergo two main processes i.e. i) combustion/exhaust and expansion and ii) compression and transfer respectively. The piston motion is similar to the conventional design butwithout the crankcase scavenging normallyassociated with typical two-stroke engines. In other words, this design shortens the travelling path of the mixture into the combustion chamber. The design is shown in Fig. 2.

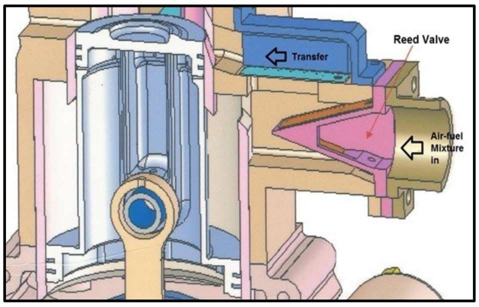



Figure 2.Engine sectional view showing intake, piston, void andtransfer port.

The induction of appropriate mixture for combustion iswhen the piston is downward motion. This is primarily due to earlier combustion that takes place in the combustion chamber resulted in gas expansion; drawing in fresh-charged mixture to fill up the void section, created between the engine wall and the smaller piston. When the pressure in the void as well as in the intake is equalized, the reed valves (shown in Fig.2) will be closed. Pressure will continue to build up as the piston assembly is moving upward with the air trapped and will enter a flap valve,(in the transfer port) and subsequently the compressed mixture will rush into the main combustion chamber.As mentioned earlier, this engine is equipped with a three-port mixture transfer to allow mixtures to thoroughly mix within the proximity of the engine's spark plug. Fig. 3 illustrates the mixing strategy employed.

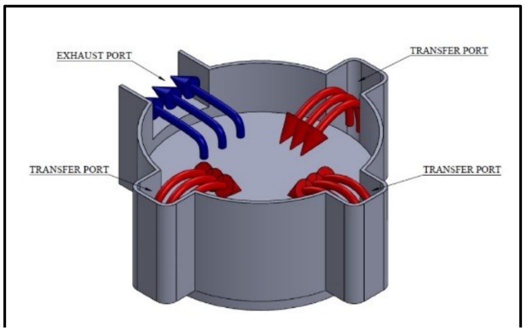



Figure 3. The cross scavenge process showing the fresh-charge (red) entering the combustion chamber, with exhaust gas product (blue) via the exhaust port.

## 3. DESIGN CONSIDERATIONS

The process of developing an engine is one of compromise and with specific attributes to achieve specific goals. There are many aspects of engineering to be considered in the design, development and testing of a prototype. Two prominent aspects were focused on i.e. i) geometrical properties and ii) thermodynamics properties [8,9]. Both of these aspects are governed by the design specifications given in Table 1.

*Table 1: Engine design specifications.*

| No. | Parameter | Features |
|---|---|---|
| 1. | Cylinder type | Single-cylinder, piston ported |
| 2. | Displacement | 125 $cm^3$ |
| 3. | Bore x stroke | 53.8 x 55 mm |
| 4. | Compression ratio | 9:1 |
| 5. | Scavenging type | Multi-port, loop scavenge |
| 6. | Exhaust port opening/closing | 93°CA ATDC/267°CA ATDC |
| 7. | Intake port opening/closing | 110°CA ATDC/250°CA ATDC |
| 8. | Design rated power | 8.5 kW@6500 rpm |
| 9. | Ignition timing | -20°CA BTDC |
| 10. | Lubrication system | Crankcase – splash type<br>Piston skirt – lube in air-fuel mixture |

### 3.1 Geometrical Properties

The following are the basic parameters used to derive for the engine reciprocating components:

i) Compression ratio, *rc*:

$$rc = \frac{max.\ cylinder\ volume}{min.\,cylinder\ volume} = \frac{Vswept + Vclearance}{Vcleance} \quad (1)$$

Where *Vswept* and *Vclearance* are the swept and clearance volume of the engine.

ii) Bore-to-stroke ratio, *Rbs*:

$$Rbs = \frac{B}{L} \quad (2)$$

Where *B* is the bore and *L* is the stroke.

iii) Connecting rod length to crank radius ratio

$$R = l/a \quad (3)$$

$L=2a$

Where *l* is the connecting rod length and *a* is the crank radius.

iv) Cylinder volume (*V*) at any crank position $\theta$

$$V=V_{clearance} + (\pi B^2/4)(l+a-s) \quad (4)$$

Where *s* is the distance between the piston pin location to the centre of the crankshaft, at engine crank angle $\theta$. The significant of calculating the instantaneous volume will assist in the computation of heat release calculation.

v) Combustion chamber surface area (*A*)

$$A = A_{ch} + A_p + \pi B(l+a-s) \quad (5)$$

Where $A_{ch}$ is the cylinder head surface area and $A_p$ is the piston surface area respectively. The significant of calculating the surface area is in the computation for heat transfer, leading towards the design of cylinder wall.

vi) Specific weight

$$Specific\ weight = \frac{Engine\ weight}{rated\ power} \quad (6)$$

The current trend is to produce high specific weight specification especially for mobile applications. For this exercise the research group targeted the specific weight not to exceed 4.0.

### 3.2 Thermodynamic Properties

The thermodynamic aspects were also given due consideration in the deriving at the geometrical properties. Among the parameters that were taken into considerations are:

i) *Delivery ratio* - The delivery ratio,η DR defines the mass of air supplied during the scavenged period as a function of a reference mass, $m_{dref}$, i.e. mass required to fill the swept volume under atmospheric conditions.

$$DR = m_{as}/m_{dref} \quad (7)$$

ii) *Scavenging ratio* - The scavenging ratio, *SR*, defines the mass of air supplied during the scavenged period as a function of a reference mass, $m_{sref}$, i.e. the mass that is able to occupy the entire cylinder volume under atmospheric conditions. It is defined as:

$$SR = m_{as}/m_{sref} \quad (8)$$

iii) *Scavenging efficiency* - Scavenging efficiency, *SE*, is the mass of delivered air that has been trapped, $m_{tas}$, in relation to the total mass of charge, $m_{tr}$, retain during exhaust port closure. The trapped charge comprises of fresh charge trapped, $m_{tas}$, exhaust gas, $m_{ex}$ and residual gas from previous cycle, $m_{ar}$.

$$SE = m_{tas}/(m_{tas}+m_{ex}+m_{ar}) \quad (9)$$

iv) Trapping efficiency - Trapping efficiency, TE, is the capture ratio of mass of delivered air that has been trapped, $m_{tas}$, to that of supplied, $m_{as}$.

$$TE = m_{tas}/m_{as} = SE/SR \quad (10)$$

v) Charging efficiency - Charging efficiency, *CE*, is the ratio of the filling of the cylinder with air, to that of filling the same cylinder perfectly with air at the onset of the compression stroke. It is written as,

$$CE = m_{tas}/m_{sref} = TE\ x\ SR \quad (11)$$

vi) Air-fuel ratio - This is an equally important parameter which is the ratio of the mass of air intake, $m_{air}$ to that of the fuel intake, $m_{fuel,}$ per unit time.

$$AFR = m_{air}/m_{fuel} \quad (12)$$

vii) Heat release during the burning process -This is a heat quantity released from the combustion of fuel induced and burnt and is designated as $Q_R$. It is calculated as:

$$QR = \eta_c\ x\ m_{tf}\ x\ LCV \quad (13)$$

Where $\eta_c$ is the combustion efficiency and *LCV* is the fuel's low calorific value.

Apart from the above parameter, heat transfer analysis is also a crucial aspect investigated in optimizting development work on the engine. Heat transfer will affect engine performance and emission. The engine specific power and efficiency will be effected by the magnitude of the engine heat transfer from the combustion chamber to reciprocating mechanism, cylinder head and wall. Under this development project heat transfer between unburned charge and the chamber wall was carefully investigated wherby the optimum designed was derived to overcome engine knock.

## 4. PERFORMANCE SIMULATION

Instead of testing every operating point on a dynamometer, a computer model was used to simulate the engine. The program used to simulate the engine design was Gamma Technology's GT-Power$^{TM}$ software suite. With GT-power, the physical dimensions of the complete powertrain system must be entered. The process of creating a GT-Power model begins with dividing the engine into its components. The major components are the carburetor, throttle, reed valve, the intake manifold, the engine, and the exhaust system. To model the intake manifold, the most important aspect is to model all of the pipe bends and flow splits. The software (version 6.0) has preset components for straight pipes, bent pipes and flow splits. Each component is defined by several parameters such as discharge coefficients, cross sectional area and lengths.

In addition to the physical properties of the engine, a combustion model must be entered. The flow dynamics created from the opening and closing of the intake and exhaust port are complex. Therefore it is difficult to replicate the intricate mixing action of fuel using a simple model. Combustion is largely dependent upon the mixing inside the cylinders and the local air/fuel ratio around the spark plug. Therefore, the in-cylinder flows are very important. Although it is possible to create a model that truly captures all of the fluid motion, the model would take a significantly longer time to converge to a solution. A long convergence time is unfavorable, so a combustion model is inputted. Using a combustion model reduces the simulation time and accounts for the in-cylinder fluid motion. The 1-D model for this engine, generalizing the properties of each of components is shown in the Fig.4 below.

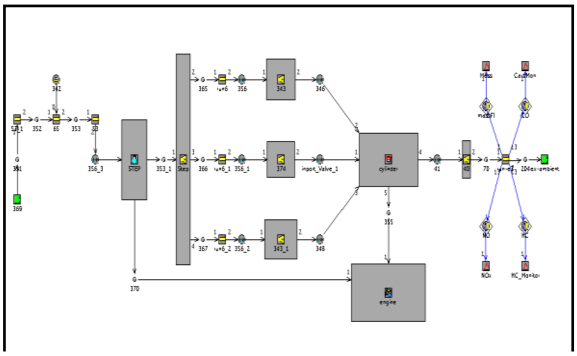


Figure 4. GT-Power 1-D model of stepped-piston engine.

The *P-θ* and *P-V* plots together with the heat release profiles were the essential tools in the optimization of the reed valve, intake passage, transfer as well as the exhaust ports. These parameters assist in the optimization of the combustion chamber configuration. Fig. 5, 6 and 7 illustrate the pressure and rate of heat release profiles generated by GT-Power based on the prescription on the properties of the engine component and geometry. These data generate peak cylinder pressure between the 1500 to the 6500 rpm setting. This speed range has been identified to generate peak torque for the designed engine. The early indication also shows that rapid rate of heat releasewill occur for the lower speed engine operation. As the speed is increased, the rate of the heat release will be lowered, as shown in Fig. 7.

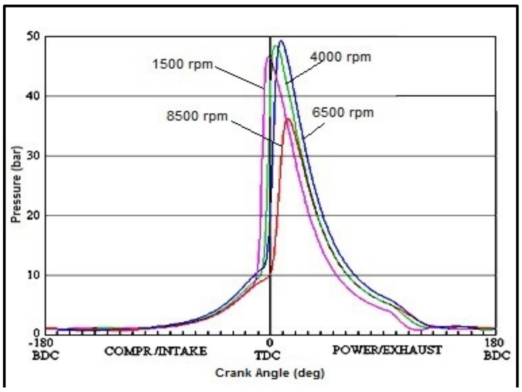



Figure 5. Pressure vs. crank angle position showing the pressure profile at various speed subjected to full load.

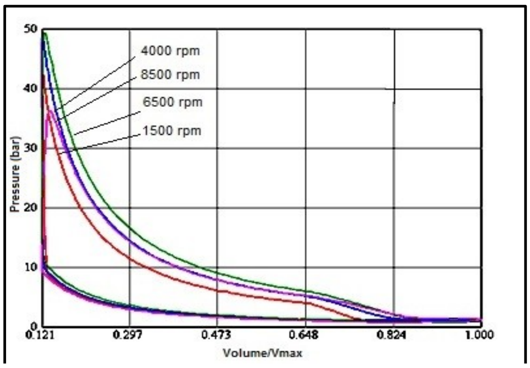



Figure 6. Pressure vs. swept volume position showing the pressure profile at various speed subjected to full load.

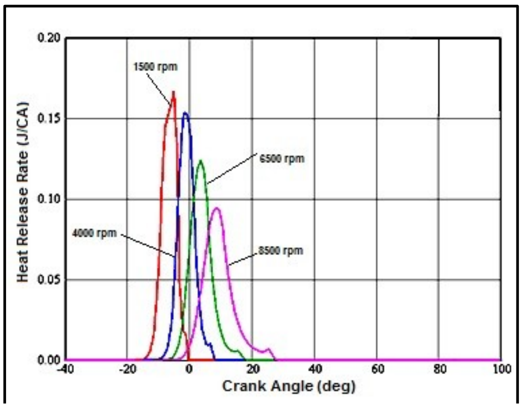



Figure 7. Heat release rate taken for the selected speeds at maximum load.

## 5. ENGINE DEVELOPMENT WORK

There are 98 components and parts that constitute the engine as a whole (refer Figure 8). They are classified as critical and non-critical, whereby some of them are standard items (of-the-shelf), which can be obtained from suppliers. The non-standard items are unique to the engine and they differentiate the engine from the ordinary crankcase scavenged type. The reciprocating parts (piston, connecting rod and crankshaft) were classified as critical. This was followed by cylinder liner, cylinder head, main body, center body (component between cylinder head and the main body) and flywheel. The critical and the less critical-type components were manufactured using local expertise and services.

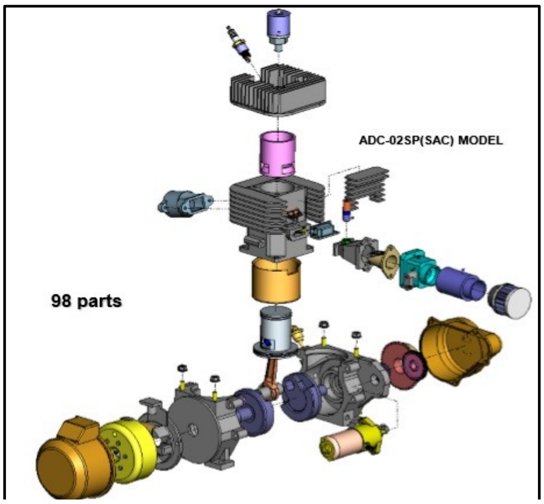



Figure 8. The exploded view showing the components of the prototype engine.

The summary on the type of materials and the process involved is given in Table 2.

*Table 2. Critical engine components.*

| Component | Material | Process |
|---|---|---|
| Piston | Aluminum alloy | Casting + dehorning + hardening + machining + grinding |
| Connecting rod | Steel alloy (T6-2024) | Drop forged + machining |
| Crankshaft Flywheel | Cast iron | Green sand mould casting |
| Cylinder head | Aluminum alloy (Al-10mg) | Green sand mould casting |
| Center body | Aluminum alloy (Al-10mg) | Green sand mould casting |
| Crankcase | Aluminum alloy (319 type) | Green sand mould casting |

These components upon completion of fabrication were then assembled. The engine was then fitted with the fuel as well as the ignition systems. The now completely assembled engine is shown in Fig. 9.

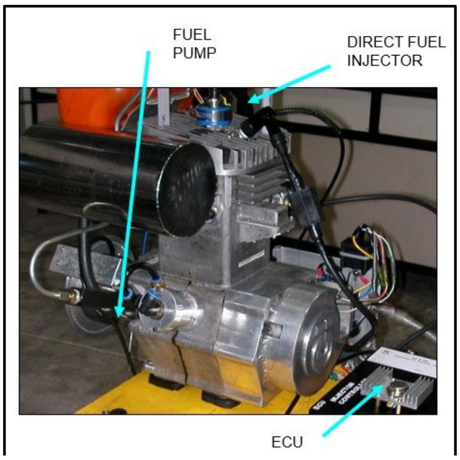



Figure 9. The 125 cc stepped-piston engine showing
the external features.

## 6. LABORATORY TRIALWORK

The laboratory investigation was done in accordance to SAE J1349 standardfor engine test. The aim was to establish the best possible performance, under typical operating conditions. A 30kW eddy-current dynamometer (*Magtrol,* WB30) was used in conjunction with a volumetric fuel flow meter (*Ono-Sokki,* FP-213s) and a portable emission analyser (*Autocheck 982*). The test was done with full open throttle, where the throttle is increased slowly until the engine produces the maximum torque. The engine speed is set to a constant value at the dynamometer controller. Dynamometer will control the engine's speed according to the value that has been set. The *M-TEST*$^{TM}$ program (control and data acquisition software) was used to record the value of speed, torque, power, and the time of the test. Average value is taken for the speed, torque and power produced by the engine at the setting speed. The setting speed value at the dynamometer and the speed produced by the engine are not the same. The values fluctuate at all the time depending on speed and ambient conditions. To overcome this irregularity, the attenuating PID (Proportional gain, integral and derivative) setting of the dynamometer controller was made for best system response.

For all the engine test the maximum speed allowed for was limited to 5500 rpm and this was solely for safety reason of operator and researchers on site. Fig. 10 is a compilation of corrected experimental results, taken for a number of cycle, showing the brake power, the brake torque and the specific fuel consumption. Fig. 10 depicts the three major constituents of the emission also at maximum load settings. The brake power registered at 5500 was 6.45 kW, maximum torque was registered at 4000 rpm and the best specific fuel consumption at 255 g/kWh at 3500 rpm respectively.

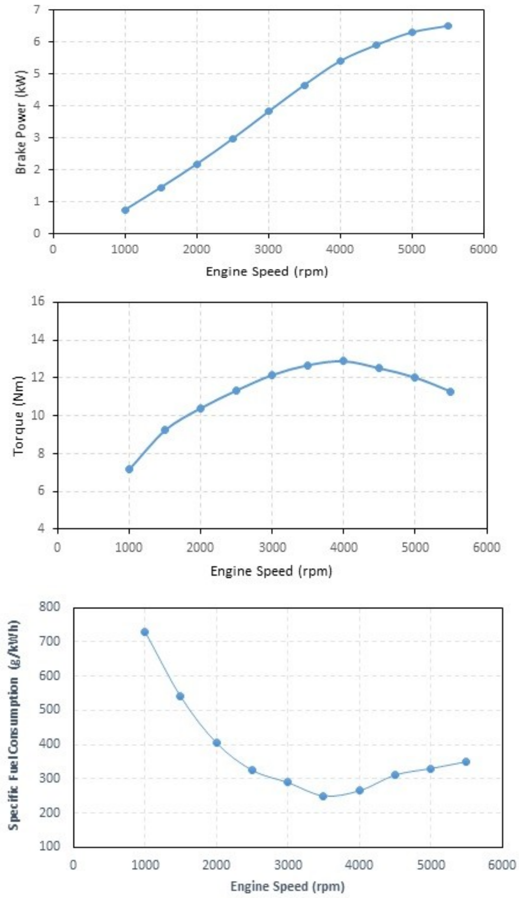
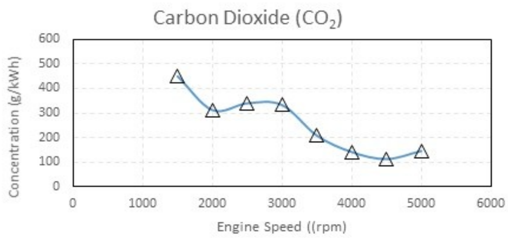



Figure 10. Engine and fuel consumption data at maximum load

Fig. 11 illustrates the results showing the tailpipe emission concentrations (g/kWh) for engine test under similar condition to those of Fig. 9. Here the trend for $CO_2$, CO and HC are in the descending order with respect to the speed increment. When the speed is increased the reduction for these three gas species is apparent. In general the concentration of $CO_2$ ranges from 100 to 500 g/kWh, CO from 100 to 500 g/kWh and HC was from 20 to 100 g/kWh respectively. These values are within the permissible level of the US Environmental Protection Agency (EPA) for off-highway two-stroke engine specifications [10].



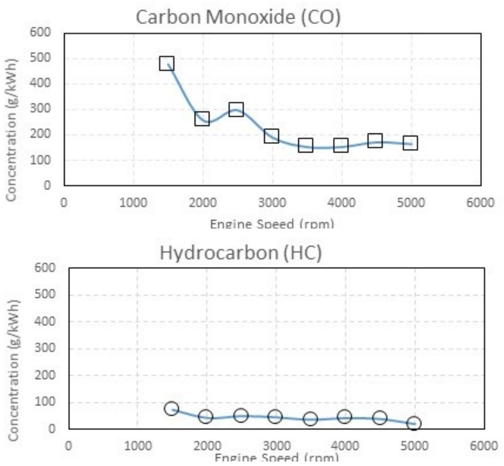



Figure 11. The tailpipe measurement for $CO_2$, CO and HC at maximum load condition.

## 7. CONCLUSION

The aspiration of producing an air-cooled, two-stroke engine of a stepped-piston type, with potential of generating 8.5 kW at 6500 rpm, has been realized - from design to prototyping. Design work starts with the desired specifications and followed by performance simulation using GT-power$^{TM}$. With the promising indication from the simulation results, this warrants the development work to commence for components, piston, cylinder head, and main body of the engine.

Once the components have been produced the fitting of the fuel and ignition systems to the prototype was made. Experimental engine trials was subsequently made to the engine using eddy-current dynamometer facility to measure its brake torque, power, fuel consumption and selected emission constituents up to 5500 rpm. The trial has produce the rate engine output at 6.5kW at this safe limit of engine trial with the best specific fuel consumption registered as 250 gram/kWh at 3500 rpm. Emission level are acceptable and within those limits specified by EPA for small engine category.

However there are still some short-circuiting problems with regard to combustion gas in which the trapping efficiency calls for further refinement. The prototype as a whole has meet up to the earlier design expectations, in terms of output at mid-speed range speed. Work are now underway to mitigate this problem by incorporating innovative rotary valve to improve the trapping mixture capability of the engine and this will lead to further combustion improvement.

## ACKNOWLEDGEMENT

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