PROJECT REPORT
(BMM 3912)

PREDICTION STUDIES FOR THE PERFORMANCE OF A SINGLE CYLINDER HIGH SPEED SI LINEAR ENGINE

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PREDICTION STUDIES FOR THE PERFORMANCE OF A SINGLE CYLINDER HIGH SPEED SI LINEAR ENGINE

MOHD NORDIN BIN ZAZALLI

A report submitted in partial fulfillment
of the requirements for the award
of the degree of
Bachelor Of Mechanical Engineering

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NOVEMBER 2008

SUPERVISOR'S DECLARATION

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I hereby declare that the work in this thesis is my own except for quotations and summaries which have been duly acknowledged. The thesis has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicated to my beloved

Mother,

Father,

and Friends

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Abstract

This study is prediction the performance of linear engine with spring system. To predict performance engine GT-Power software is used with small modification in friction analysis. The simulation of linear engine is done by variable speed. Performance of linear engine is determined by comparing it with conventional engine in analysis of imep, bmep, power, torque and brake specific fuel consumption. Linear engine has better performance than conventional engine.

Abstrak

Kajian ini adalah tentang prestasi enjin linear beserta sistem spring. Untuk meramal prestasi enjin tersebut, perisian GT-power digunakan dengan sedikit pengubahsuaian dalam analisis geseran. Simulasi enjin linear ini dijalankan dengan pelbagai kelajuan. Prestasi enjin linear ini ditentukan dengan membandingkan ia dengan prestasi enjin biasa dari segi analisis tentang imep, bmep, kuasa, tork dan penggunaan bahan api tentu brek.

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LIST OF SYMBOLS

Variable parameter in Friction Analysis

b = bore

D_b = bearing diameter

L_b = bearing length

N = engine speed

n_b = number of bearing

n_c = number of cylinder

P_a = atmospheric pressure

P_i = intake manifold pressure

r = compression ratio

s = stroke

U_p = piston speed

Constant in Friction Analysis

Cb = 3.03×10^{-4} kPa-min/rev-mm

Cg = 6.89

Cpr = 4.06×10^{4} kPa-mm2

Cps= 294 kPa-mm-s/m

cs = 1.22×10^{5} kPa-mm2

K = 2.38×10^{-2} s/m

K= 0.14 (spark ignition engine)

CHAPTER 1

1.1 Introduction

Most concepts of linear engines are constructed as opposed piston with complicated control devise to operate the engines. Spring has been adopted as return force of the piston movement technique. The unique of using spring as return cycle is that characteristics (stroke of the engine is not constant as conventional engine).The problem is in expansion stroke is depend on thrust force of piston. The performance of linear engine can be predicted by using GT-power software and spreadsheet. GT-power software can only simulate performance of conventional engine. However, by manipulating friction factor, the simulation can also be done for linear engine. To construct linear engine modeling, some of friction loss in conventional engine is combined into one constant fmep value and then inserted into linear engine GT-power modeling. Fmep constant value will be calculated by formulas. In order to obtain the performance of the engine, variable speed is used. Performance of linear engine that obtained from GT-power simulation is represented by graph. By compares performance of linear engine with conventional engine, the improvement of performance is known.

1.2 Problem Statement

To study and predict the performance of linear engine.

1.3 Objectives

- To study performance of linear engine by using GT-power and spreadsheet.
- To studies performance of standard conventional engine.
- To compare performance between linear and conventional engine.

CHAPTER 2

LITERATURE REVIEW

2.1 Free Piston Engine

Due to the breadth of the free piston term, many engine configurations will fall under this category. The free piston term is most commonly used to distinguish a linear engine from a rotating crankshaft engine. The piston is 'free' because its motion is not restricted by the position of a rotating crankshaft, as known from conventional engine, but only determined by the interaction between the gas and load forces acting upon it [1].

This gives the free piston engine some distinct characteristics, including (a) variable stroke and (b) the need for active control of piston motion. Other important features of the free piston engine are potential reduction in frictional losses and possibilities to optimize engine operation using the variable compression ratio [1].

2.2 Single Piston

A single piston free piston engine is shown in Fig 1. This engine is essentially consists of three parts: a combustion cylinder, a load device and a rebound device to store the energy required to compress the next cylinder charge. In the engine shown in the figure the hydraulic cylinder serves as both load and rebound device, whereas in other designs these may be two individual devices, for example an electric generator and a gas filled bounce chamber [1].

A simple design with high controllability is the main strength of the single piston design compared to the other free piston engine configurations. The rebound device may give the opportunity to accurately control the amount of energy into the compression process and thereby regulating the compression ratio and stroke length [1].

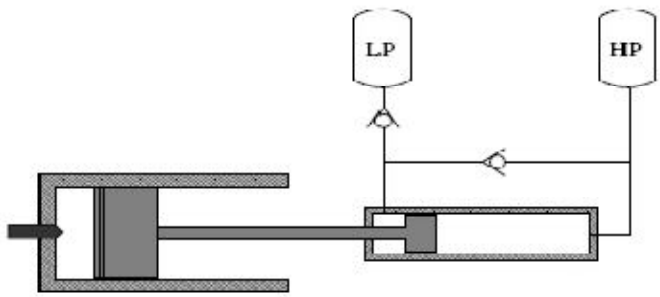


Fig. 1: Single piston hydraulic free –piston engine [1]

2.3 Free Piston Engine Features – Operating Principle.

The free piston engine is restricted to the two stroke operating principle, as a power stroke is required on every cycle. Although two stroke engines suffer from poorer performance compared four strokes, this performance gap is declining and recent years have been an increased interest in small scale two stroke engines [1].

2.4 Free Piston Engine Features – Piston Dynamics and Control

In conventional engines, the crank mechanism and flywheel serve as both piston motion control and energy storages. The piston motion control ensures sufficient compression in one end and sufficient time for scavenging in the other, while the energy storage provides energy for the compression of the next charge. In the free piston engine, the motion of the mover at any point in the cycle is determined by the sum of the forces acting upon it.

Hence, the interaction of these forces must be arranged in a way that ensures the mover motion is within acceptable limits for all types of operation if the concept is to be feasible [1].

For an engine as shown in Fig. 2a, one can derive the mover motion mathematically using a free body diagram as shown in Fig. 2b. The forces working on the mover are: combustion chamber pressure force $F_{C,}$ bounce chamber (rebound) force $F_{R,}$ load force $F_{L.}$ X denotes mover position, TDC_N and BDC_N illustrate nominal top dead centre and bottom dead centre position and ML are the mechanical limits of the motion. The mover itself will have a mass m_p [1].

Applying Newton's second law to the moving mass in Fig 2b, the piston motion can be describes with the formula below [1].

$$\sum F_i = \frac{m_p d^2 x}{dt^2} \qquad (1)$$

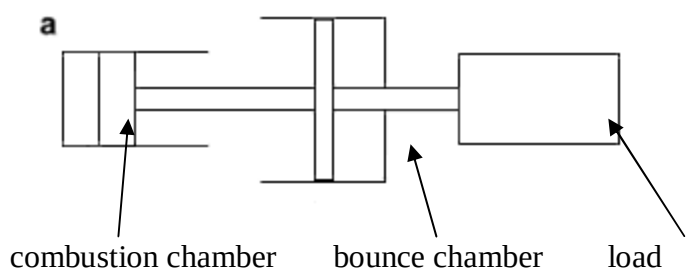


Fig. 2a: Single piston free piston engine configuration

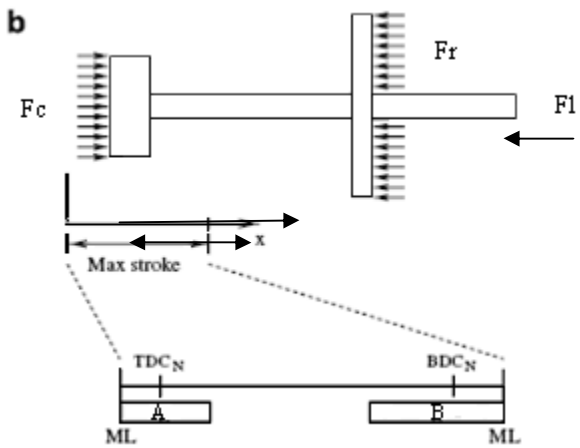


Fig. 2b: Free body diagram of the mover in a single piston free piston engine

Knowing that the combustion cylinder and the bounce chamber will have characteristics similar to those of a gas spring, it becomes clear that they will produce a bouncing type motion of the piston. Adding a load force, this must have appropriate characteristics or be subordinate the other two to ensure a reciprocating motion of the piston. If a rebound device with other force position characteristics than a bounce chamber is used, such as a hydraulic cylinder, the operational characteristics will be slightly different but the same principle will apply [1].

Fig 2b further shows the different parts of the engine stroke. Area A shows the piston position range where the compression ratio of the engine is sufficient for fuel auto ignition. For the engine to run, engine TDC must be within this area. Area B shows the piston position range where the scavenging ports are open and the burnt gases can be replaced with fresh charge. For the scavenging to be efficient, the piston needs to spend a sufficient amount of time in this area in every cycle [1].

These requirements are absolute and for the engine to be practical, an engine control system needs to be able to meet these requirements for all types of engine operation. Accurate control of piston motion currently represents one of the biggest challenges for developers of free piston engines [1].

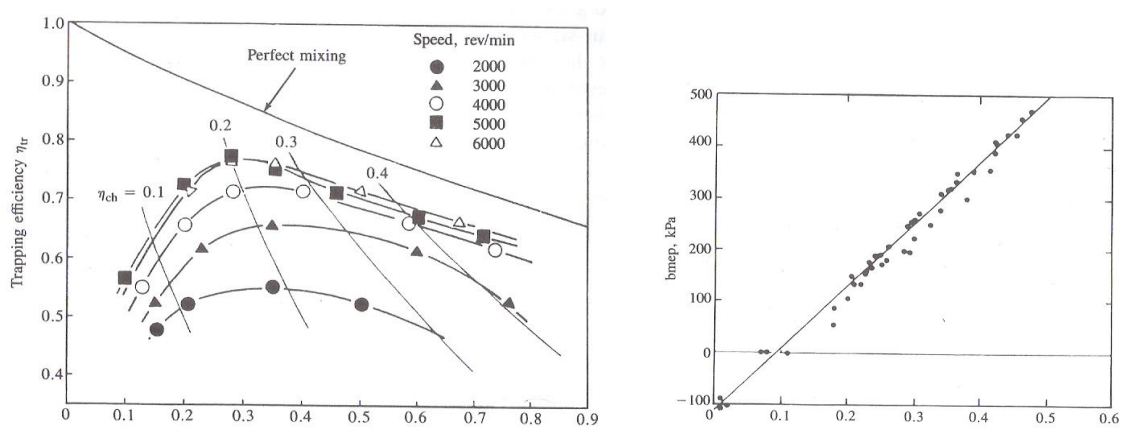
2.5 Two Stroke Cycle SI Engine.

The two stroke cycle spark ignition in its standard form employs sealed crankcase induction and compression of the fresh charge prior to charge transfer, with compression and spark ignition in the engine cylinder after charge transfer. The fresh mixture must be compressed to above exhaust system pressure, prior to entry to the cylinder, to achieve effective scavenging of the burned gases. The two stroke spark ignition engine is an especially simple and light engine concept and finds its greatest uses as a portable power source or on motorcycles where these advantages are important. Its inherent weakness is that the fresh fuel air mixture which short circuits the cylinder directly to the exhaust system during the scavenging process constitutes a significant fuel consumption penalty, and result in excessive unburned hydrocarbon emissions [2].

This section briefly discusses the performance characteristics of small crankcase compression two stroke cycle SI engines. The performance characteristics (power and torque) of these engines depend on the extent to which the displaced volume is filled with fresh mixture, i.e. the charging efficiency. The fuel consumption will depend on both the trapping efficiency. Figure 3a shows how the trapping efficiency η_{tr} varies with increasing delivery ratio Λ at several engine speeds for a two cylinder 347 cm^3 displacement motorcycle crankcase compression engine. The delivery ratio increase from about 0.1 at idle condition to 0.7 to 0.8 at the wide open throttle. Lines of constant charging efficiency η_{ch} are shown. Figure 3b shows bmep plotted against these charging efficiency values and the linear dependence on fresh charge mass retained is clear [2].

Performance curve for a three cylinder 450 cm^3 two stroke cycle minicar engine are shown in figure 4. Maximum bmep is 640 kPa at about 4000 rev/min. smaller motorcycles engine can achieve slightly higher maximum at higher speeds (7000 rev/min). Fuel consumption at the maximum bmep point is about 400 g/kW.h. Average fuel consumption is usually one-and –a-half to two times that of an equivalent four stroke cycle engine [2].

CO emissions from two stroke cycle engines vary primarily with the fuel /air equivalence ratio in a manner similar to that of four stroke cycle engines. NOx emissions are significantly lower than four stroke engines due to the high residual gas fraction resulting from the low charging efficiency. Unburned hydrocarbon emissions from carbureted two stroke engines are about five times as high as those of equivalence four stroke engines due to fresh mixture short circuiting the cylinder during scavenging. Exhaust mass hydrocarbon emissions vary approximately as $\Lambda\ (1-\eta_{tr})\ \varnothing$ is the fuel / air equivalence ratio [2].



delivery ratio Λ

(a)

charging efficiency η_{ch}

(b)

Fig. 3: a) Trapping and charging efficiencies as a function of the delivery ratio

b) Dependence of brake mean effective pressure on fresh-charge mass defined by charging efficiency

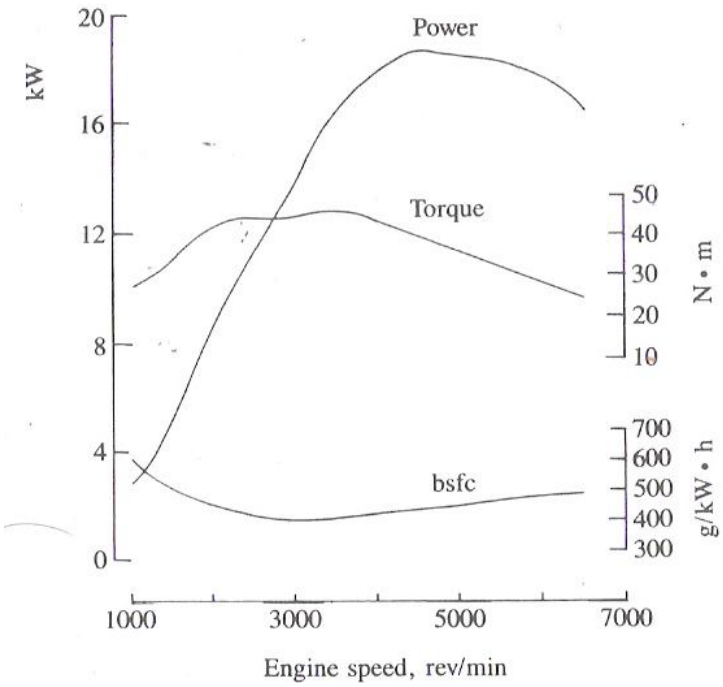


Fig. 4: Performance characteristic of a 3 cylinder 2 stroke cycle spark ignition engine

2.6 Friction

The friction forces in engine are consequence of hydrodynamic stresses in oil film and metal to metal contact. Since frictional losses are a significant fraction of the power produced in an internal combustion engine, minimization of friction has been a major consideration in engine design and operation. Engine is lubricated to reduced friction and prevents engine failure. The friction energy is eventually removed as waste heat by the engine cooling system [3].

The frictional process in an internal combustion engine can be categorized into three main components: (1) the mechanical friction (2) the pumping work (3) the accessory work. The mechanical friction includes the friction of internal moving parts such as the crankshaft, piston, rings, and valve train. The pumping work is the net work done during the intake and exhaust strokes. The accessory work is the work required for operation of accessories such as the oil pump, fuel pump, alternator and a fan [3].

We will use scaling arguments ton develop relations for the dependence of the various modes of friction work on overall engine parameters such as bore, stroke, and engine speed, then construct an overall engine friction model. The coefficients for the scaling relation are obtained from experiment data and implicitly include lubrication oil properties such as viscosity [3].

The following are the mechanical components friction for internal friction engine which are going to use in analyzing single piston free piston linear engine.

a) crankshaft-main bearings
b) crankshaft-seal
c) piston-rings
d) piston-gas pressure

a) Crankshaft-main bearings [3].

The friction mean effective pressure of a journal array with η_b bearings such as the crankshaft main bearings or the connecting rod bearings scales linearly with engine speed, assuming constant bearings clearance and oil viscosity.

$$fmep_{bearings} = \frac{c_d n_b N D_b^3 L_b}{n_c b^2 s} \qquad (2)$$

b) crankshaft-seals [3].

The crankshaft bearings seals operate in a boundary lubrication regime, since the seals directly contact the crankshaft surface. As the normal force, which the seal lip load, is constant, the friction force will be constant and the friction mean effective pressure of the crankshaft bearing seal will be independent of engine speed, and will scales as

$$fmep_{seals} = \frac{c_s D_b}{n_c b^2 s} \qquad (3)$$

Patton et al (1989) suggest a proportionality constant Cs = 1.22 x 10^5kPa –mm^2. If the bearing is not sealed, oil will leak out at the ends, so oil is pumped at relatively low pressures through internal passages to the bearings annulus.

c) piston-rings and piston-gas pressure [3].

The friction of the piston and rings results from contact between the piston skirt and the ring pack with the cylinder bore. The cylinder bore is rougher than a journal bearing bore since the cylinder bore must retain some oil during operation. The ring seals the combustion chamber, control the lubrication oil flow and transfer heat from the piston to the cylinder. In order to preserve a seal against the cylinder bore, each ring has some amount of radial tension.

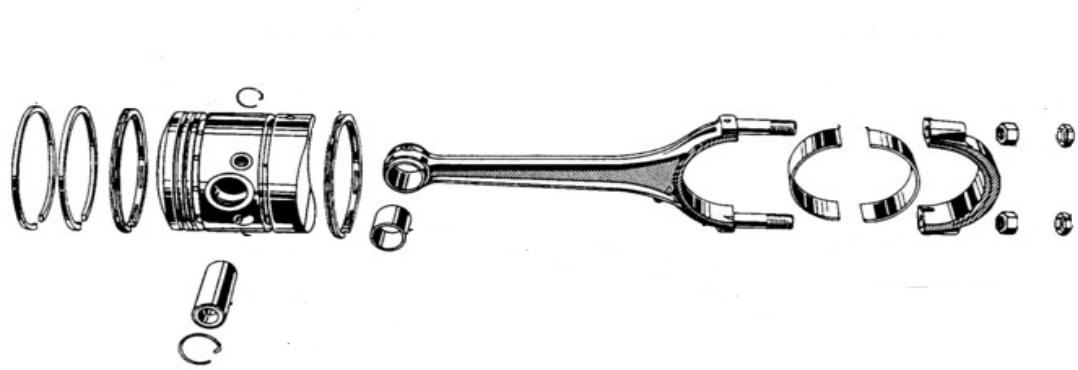


Fig. 5: Examples of piston assembly

The friction force of the piston rings has two components, one resulting from the ring tension and the other component from the gas pressure loading. The component of piston friction due to ring tension in the mixed lubrication regime will have a friction coefficient inversely proportional to the engine speed. The piston ring fmep scaling is

$$fmep_{rings} = \frac{c_{pr}}{b^2}\left(1 + \frac{1000}{N}\right) \qquad (4)$$

A correlation for the component of piston friction due to the gas pressure loading recommended by Bishop (1964) is

$$fmep_{gas\ load} = \frac{c_g P_i}{P_a}\left(0.088r + 0.182r^{(1.33 - KU_P)}\right) \qquad (5)$$

2.7 Simulation of Two Stroke Compression Ignition Hydraulic Free Piston

Engine

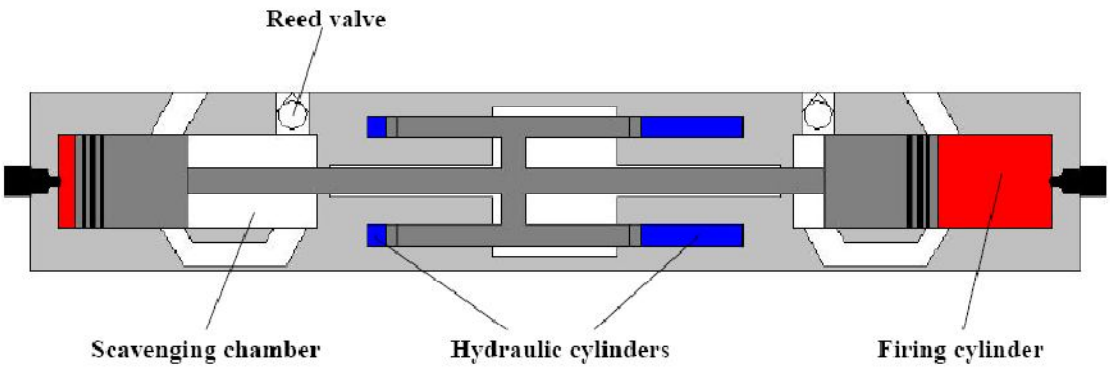


Fig. 6: The free piston engine [4]

Table 1: Free piston engine specification [4]

Bore	90 mm
Stroke	112~114 mm
Output	13-18 kW
Cycle Frequency	1700 1/min (rpm)
Outer dimension	1100 x 350 x 200 mm
Weight approximately	120 kg
Common rail fuel injection up to 1350 bar pressure	
Direct injection	

Piston Motion of the Free Piston Engine [4].

- The piston does not follow the regular crank and connecting rod motion.
- The piston motion is not symmetric around TDC/BDC.
- The piston leaves the dead center at a high speed than it approaches them.

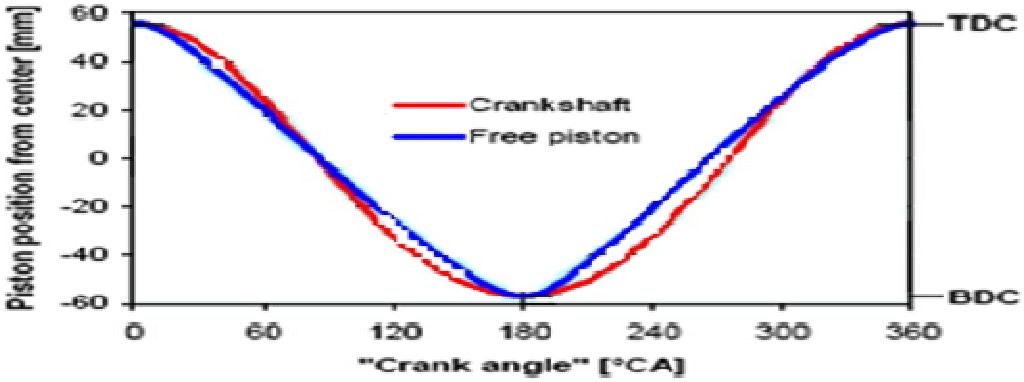


Fig. 7: The piston position against crank angle

Using Piston Motion of the FPE in GT-POWER [4].

- Piston motion object: EngCylGeomUser
- Piston motion as an XY Table.
- Piston motion either measured or simulated.
- The reference plane is at BDC.
- The XY Table starts at TDC.

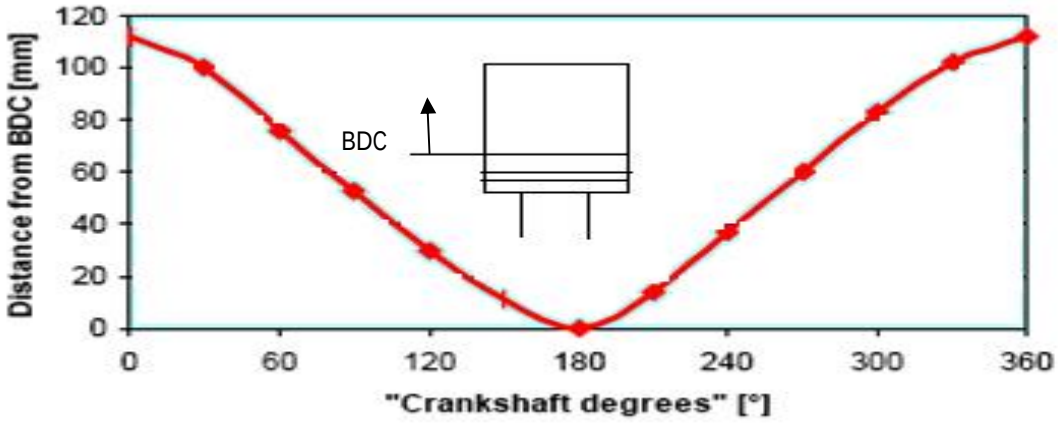


Fig. 8: Effect of distance from BDC against crankshaft degrees.

Using Piston Motion of the FPE in GT-Power [4].

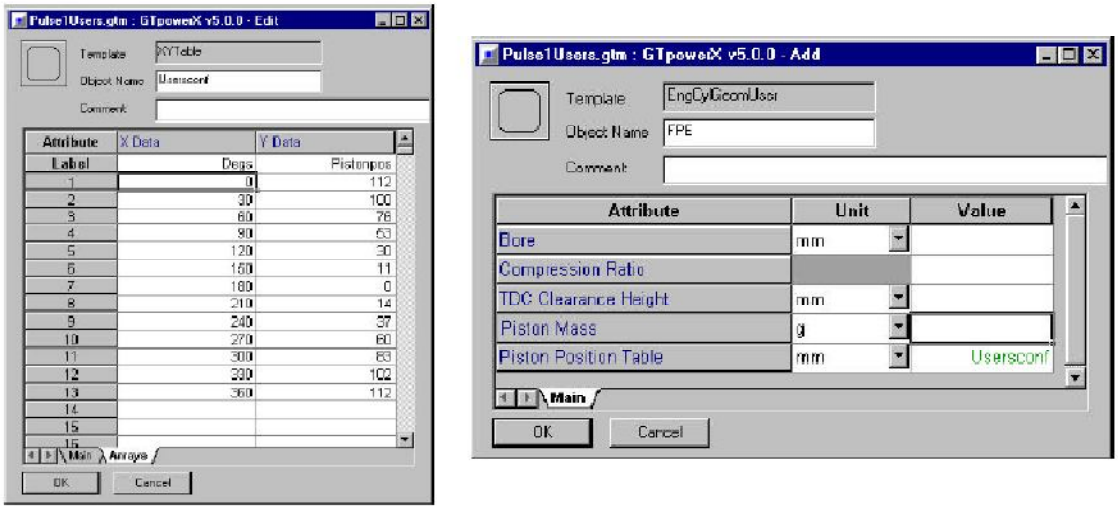


Fig. 9: Figure of piston adjustment in GT-power.

Input piston position curve: 706 points

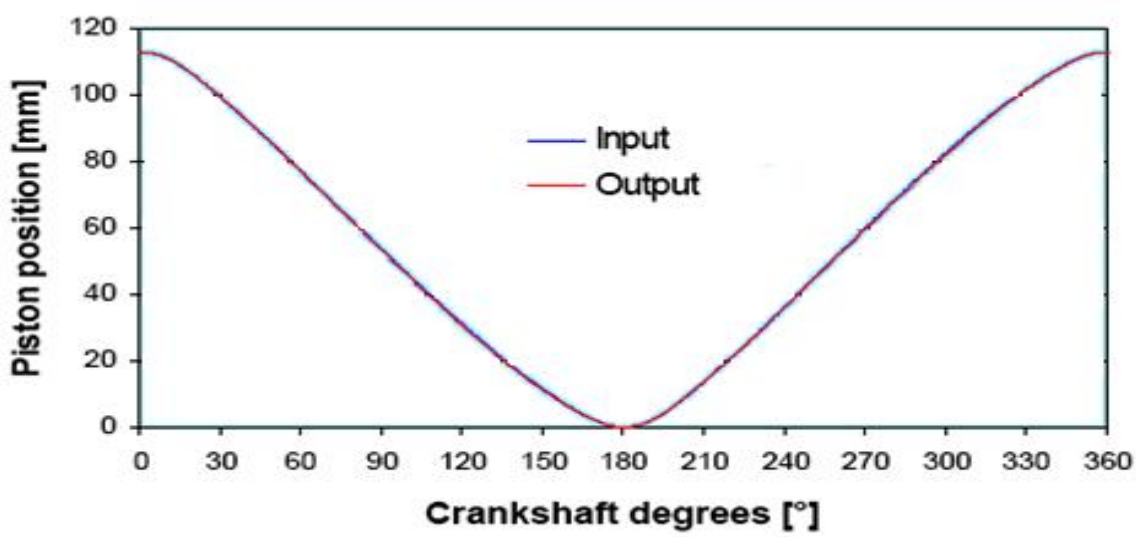


Fig. 10: Piston position with variable crankshaft degrees

Port Area [4].

- The port area in GT-Power is only defined for the down stroke as a function of crankshaft degrees.
- The port area will then be symmetric around BDC.
- A free piston engine with a non-symmetric piston motion around BDC can not be simulated accurately using a Valve Port object.

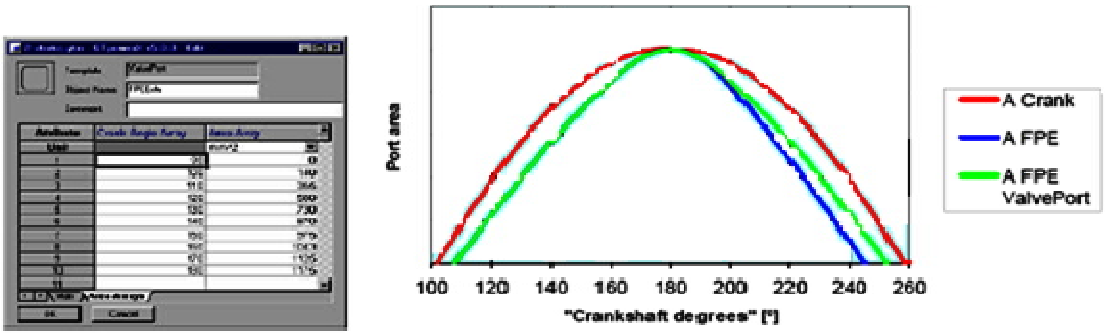


Fig. 11: Port area with variable crankshaft degrees

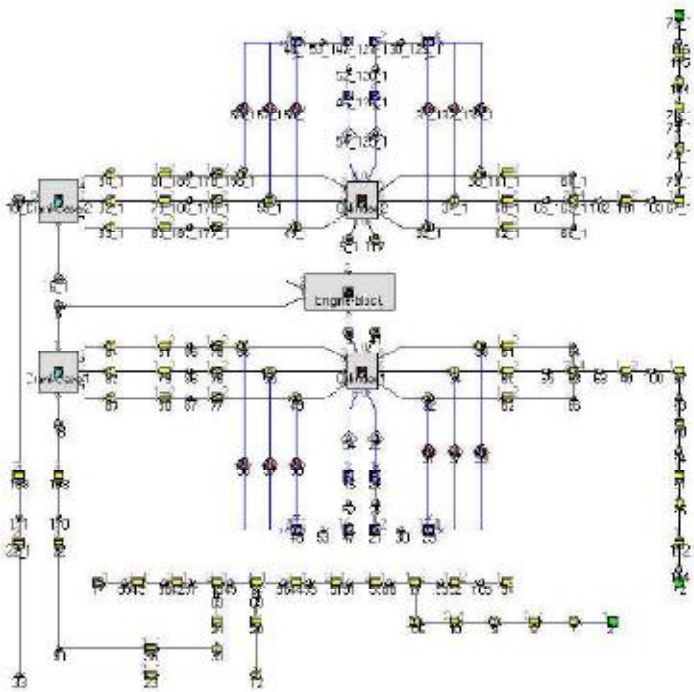


Fig. 12: The Complete Model of the Free Piston Engine

Simulation results [4].

- The results of the simulations show good correlation with measured results
- Uneven fuel injection –Uneven cylinder pressures and strokes

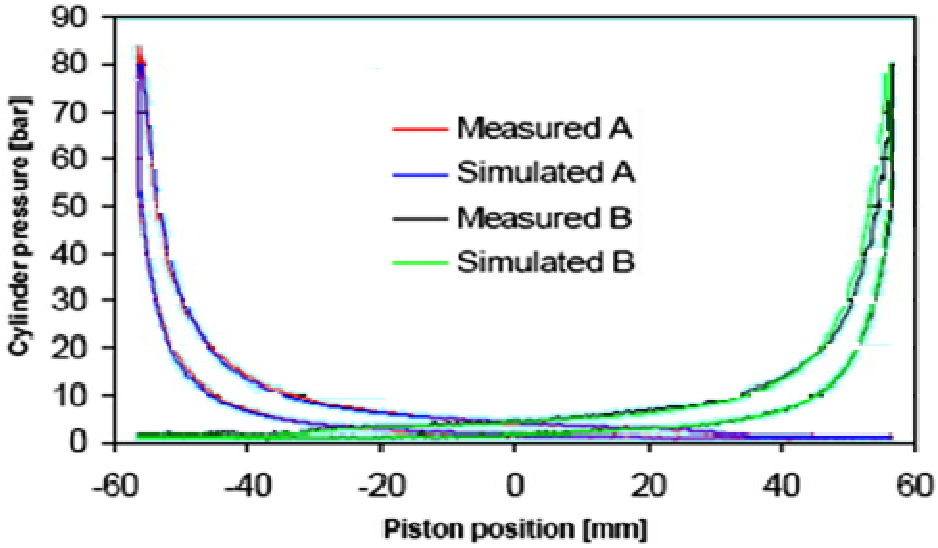


Fig. 13: Graph of cylinder pressure against piston position

- The scavenging chamber pressure show backflow in the early stages of scavenge

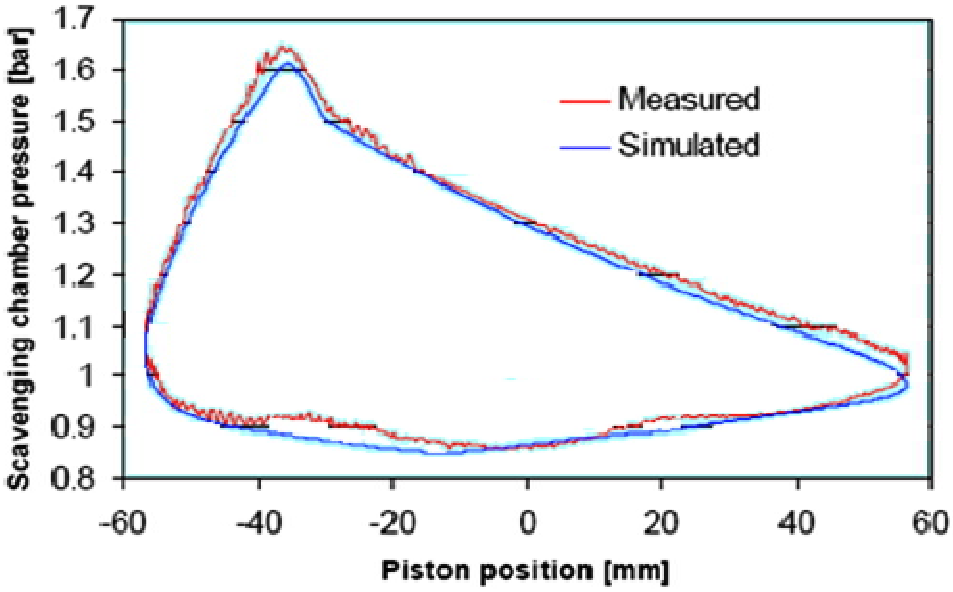


Fig. 14: Scavenging chamber pressure with variable piston position

- The exhaust gas pressure is measured in a position where the diameter of the pipe is changing Þ Hard to accurately simulate.

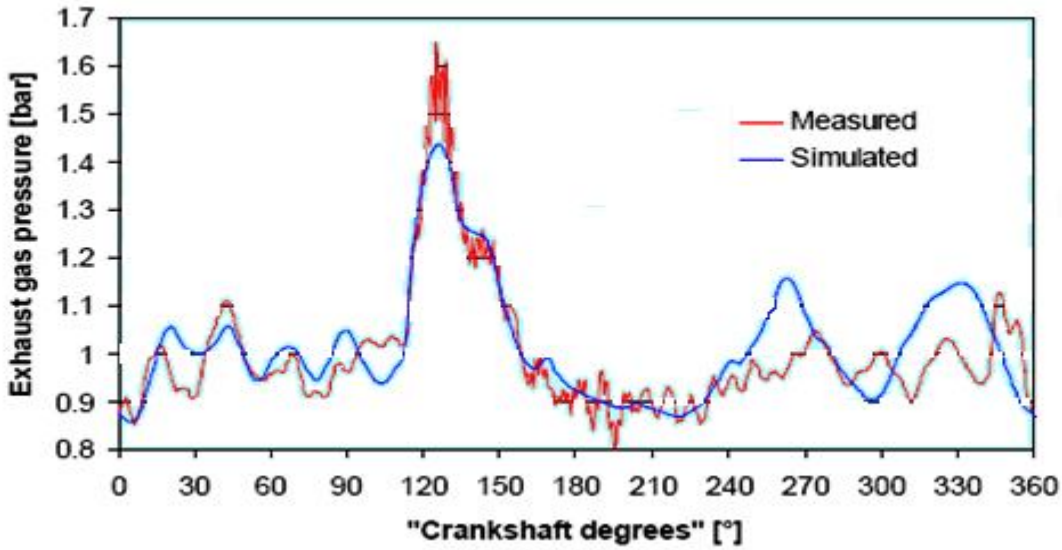


Fig. 15: Exhaust gas pressure against crankshaft degrees

2.8 GT-power Tutorial [5].

For the single cylinder SI engine model, it basically divided into three parts. Intake, exhaust and system parts. The figure below shows the details of the model.

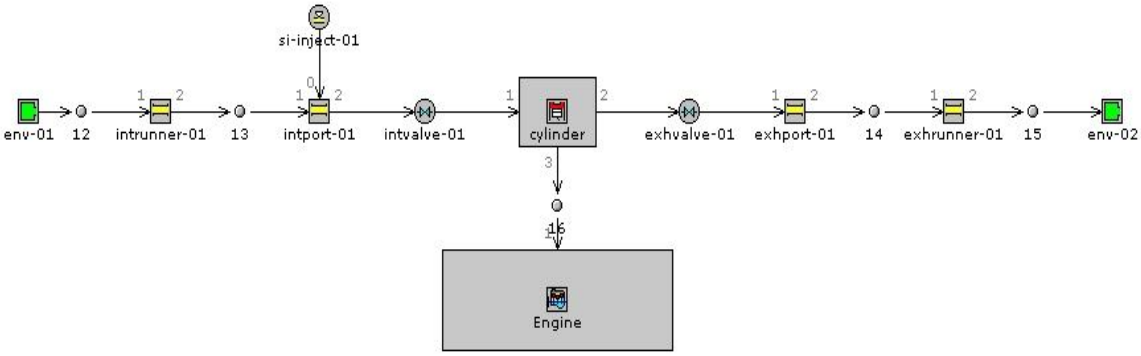


Fig. 16: Basic Single Cylinder SI Engine [5]

CHAPTER 3

METHODOLOGY

In this section, the procedures and method used in carried out this project will be explained briefly. The process are measuring the parameter of the engine, mapping engine simulation model, calculation of friction analysis, modifying friction factor, run simulation and interpreting data for conclusion. The tools and software that used in the project also be included.

3.1 Measuring the Parameter of the Engine.

There are two types of method used in measuring engine's parameter. The first method is by using conventional method which takes the reading of the parameter directly from the real engine. In order to get accurate readings, a vernier calipers is used as a tool. All the diameters in inlet and outlet have to be considered. Area also involved in measuring the parameter of the engine.

Second part is by using drawing from Solid works software as shown in Figure. 17. As we all known that there are certain parts in the engine that is hard to measure by conventional method using vernier caliper. Thus, from the drawing in Solid works, the parameter can be easily measured. The part that involved is the scavenging part.

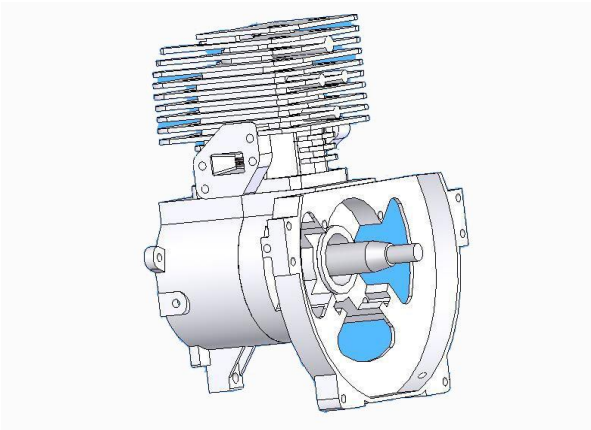


Fig. 17: Solid works drawing for two-stroke engine

3.2 Mapping Engine Simulation Model.

In Figure 18 shown that after all parameter in the engine is measured, the model for the engine is constructed in GT-POWER software. In this level, all of the elements in the simulation model are inserted with the measured value. There are also some constant in the element that has been defined.

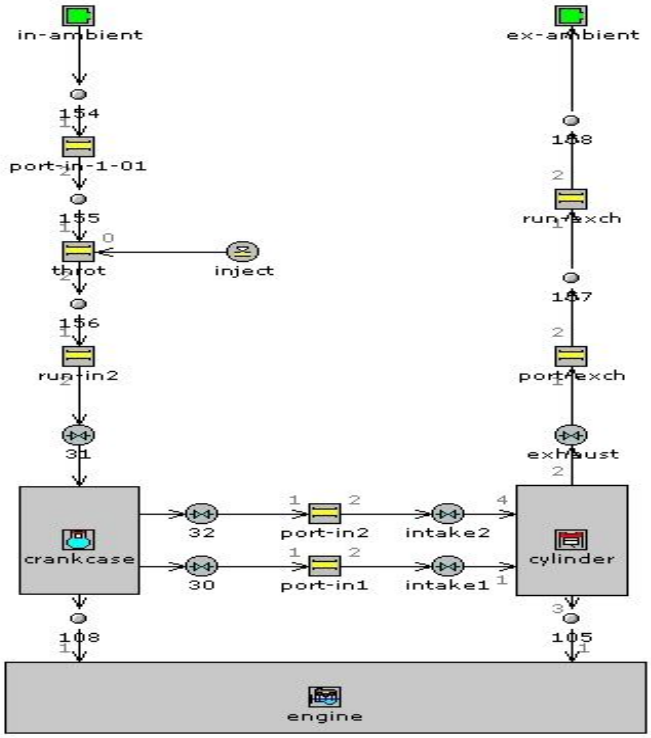


Fig. 18: 1-D mapping for 2-stroke engine

3.3 Calculation of Friction Analysis.

In order to simulate single piston free piston linear engine in GT-power software, fiction factor of the engine has been modified. In linear engine, the friction is less due less numbers of parts in the engine. Some parts in the conventional engine are replaced with the connecting shaft and spring. To calculate the friction for simulates linear engine, the friction that has been considered are at crankshaft main bearing, crankshaft-seal, piston-ring and piston-gas pressure. The formulas that used are shown in the literature review section.

3.4 Modifying friction factor.

In this stage, engine crank train part is inserted with the friction values form data base. These values are from the calculation of friction analysis that done before at Figure 19. These values are constant which represents all the friction part in the engine.

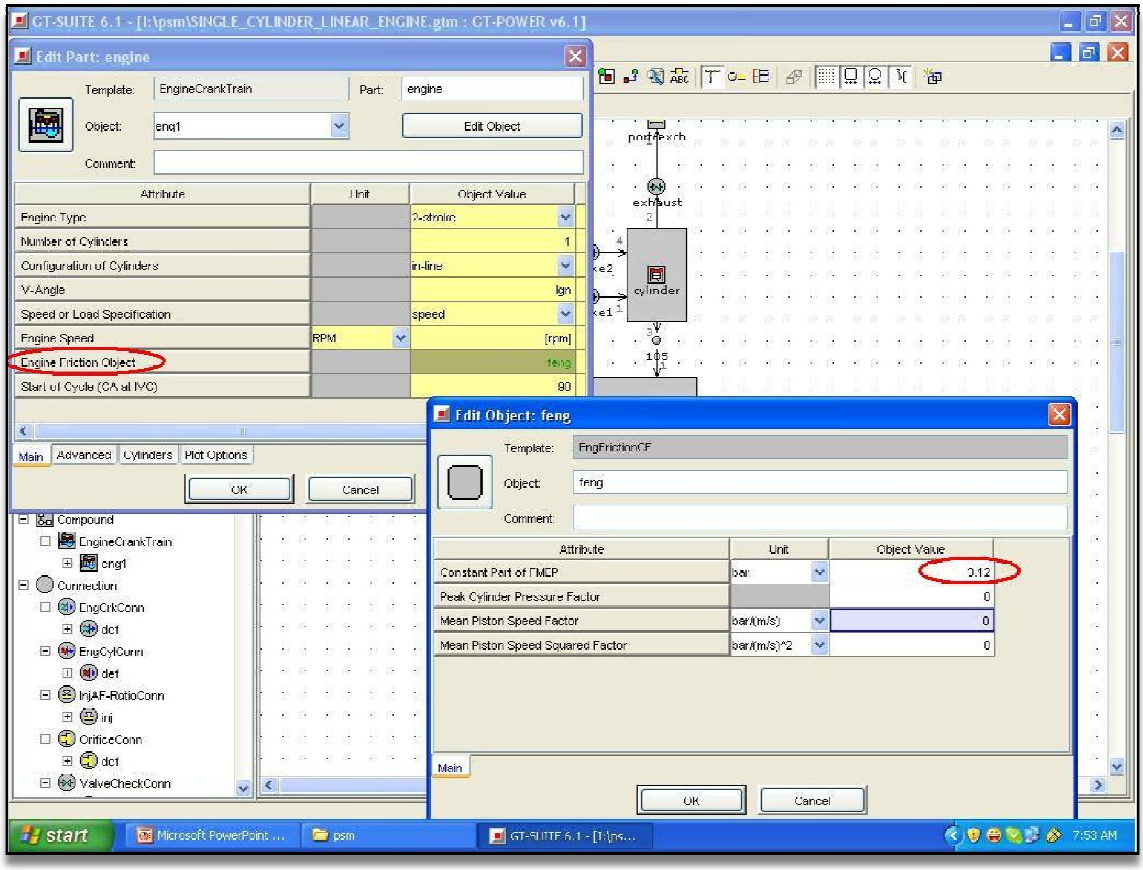


Fig. 19: Modifying friction factor in GT-power.

3.5 Run Simulation and Interpreting Data.

When the mapping model is done, the simulation has been carried out. Black box is appeared on the screen. This black box is useful to determined whether there is error or not in the simulation. After there is no error, the simulation is continued running and lastly the result will displayed. The results are collected and interpreted in the form of graph.

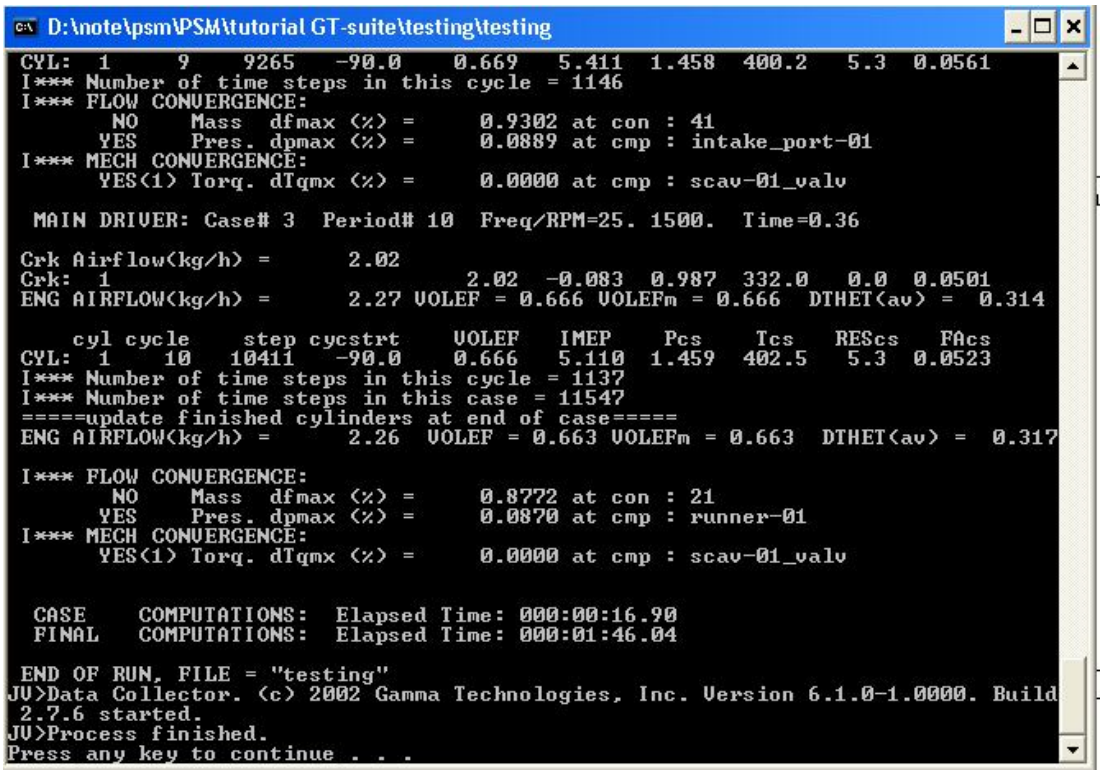


Fig. 20: Running the engine simulation .

3.6 Flowchart of The Project.

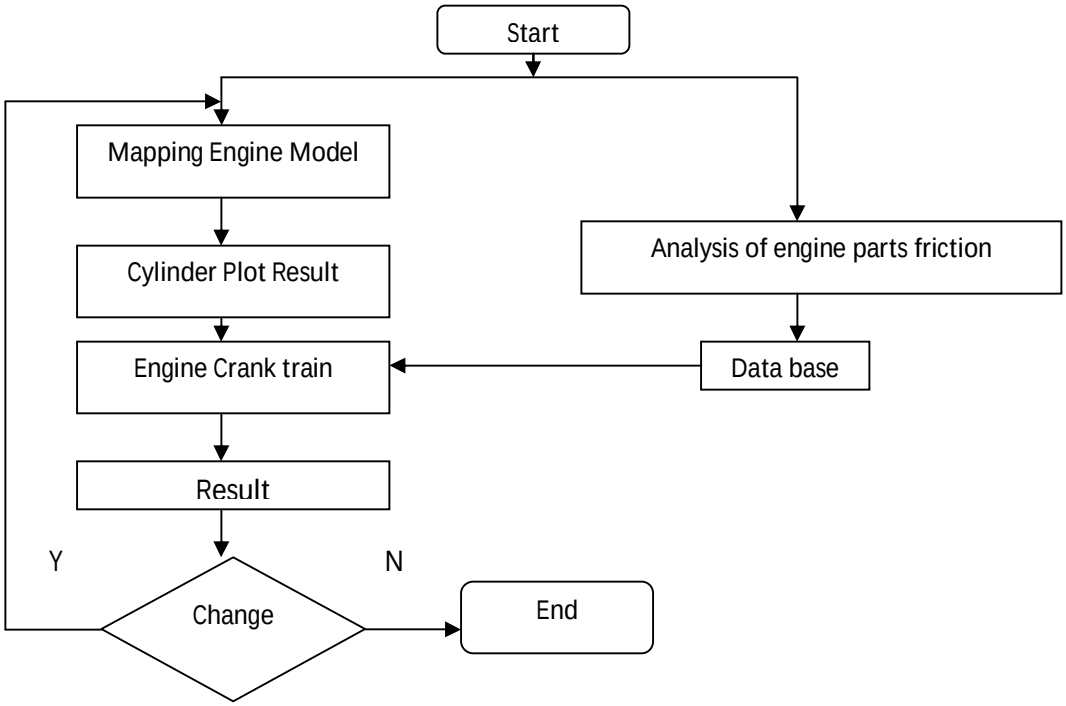


Fig. 21: Flow chart of 1-D analysis for two stroke engine

3.7 Reference Engine Specification

Model: BG-328 (Back Pack Grass Cutter)

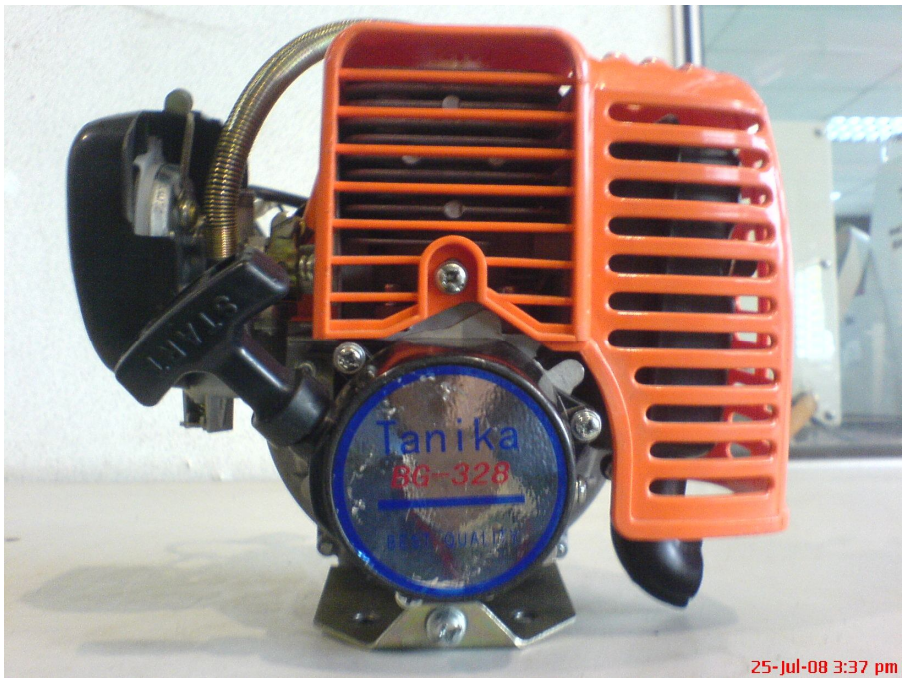


Fig. 22: Side view of BG-328 engine

Fig. 23: Front view of BG-328 engine

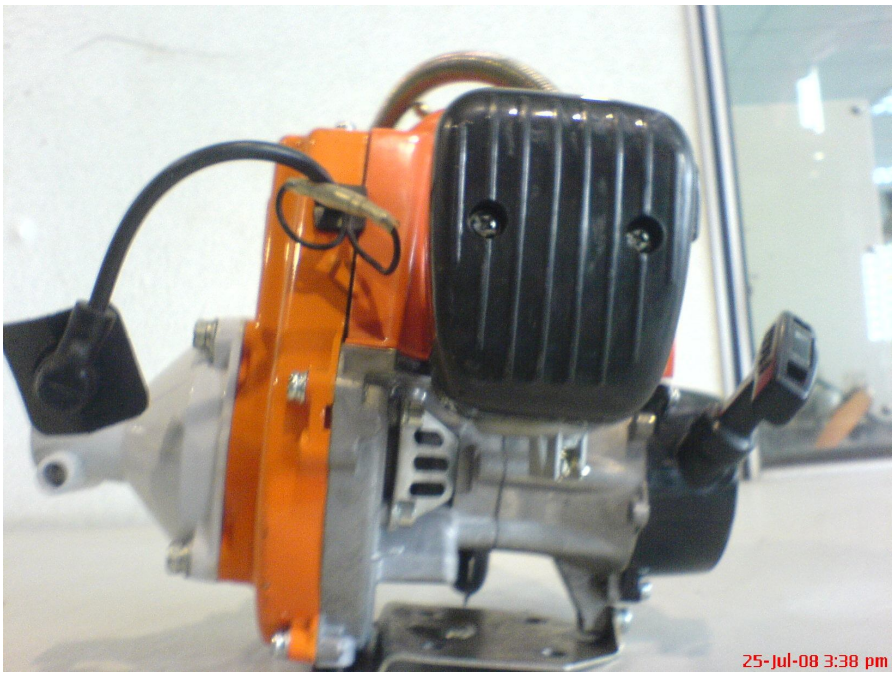


Fig. 24: Top view of BG-328 engine

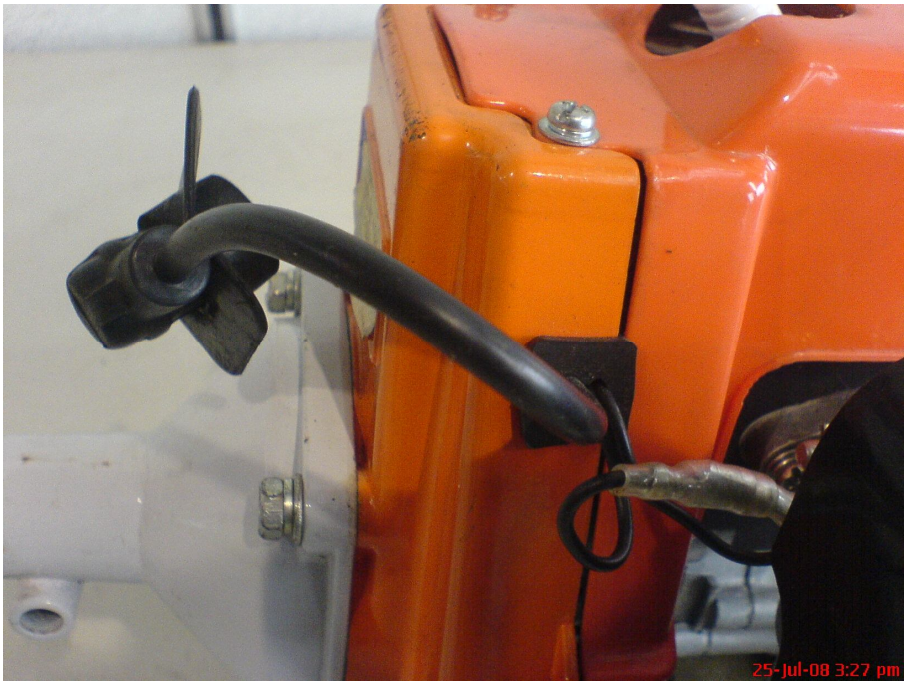
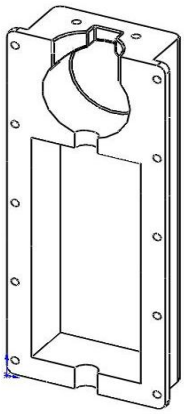


Fig. 25: Coil wire at BG-328 engine

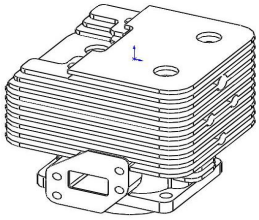
Table 2: Back pack grass cutter specification.

Model	**BG-328**
Engine	
Type	2 cycle, single cylinder, forced air cooled, gasoline engine
Displacement	30.5 cc
Max. Output	0.81 kW/ 6000 rpm
Carburetor	Float type
Ignition system	IC Ignition (Solid state)
Ignition plug	BM-7A or CHAMPION CJ6
Fuel	Mixed fuel of gasoline and 2 cycle oil at 25:1
Fuel tank capacity	1.2 liters
Body	
Drive	Flexible shaft, drive shaft, pinion and gear
Rotational direction of cutter (viewed form the top of the cutter)	Counter clockwise
Dimension (Length x Width x Height)	345 x 280 x 401 mm (Back loaded part only)
Dry weight	9.4 kg

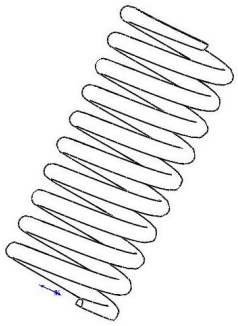
3.8 Basic component in construction of linear engine cylinder and head



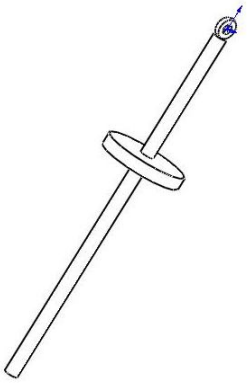
Cutting view of casing



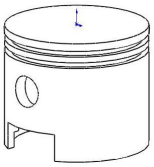
cylinder head



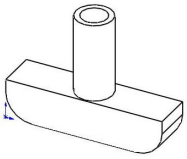
preloaded spring



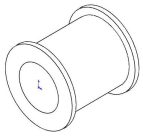
Shaft



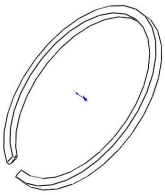
piston



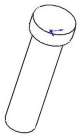
starter handle



Bearing



piston ring



screw

Fig. 26: Basic components in linear engine cylinder and head

3.9 Construction of linear engine in Solidworks

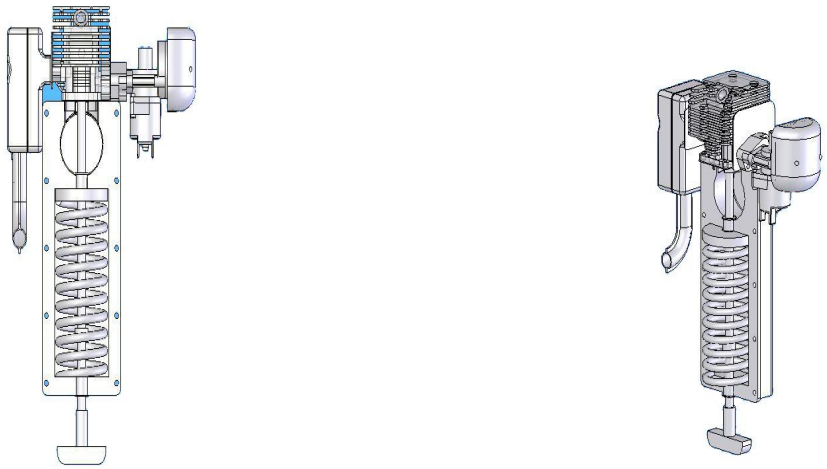


Fig. 27: Cutting view of linear engine

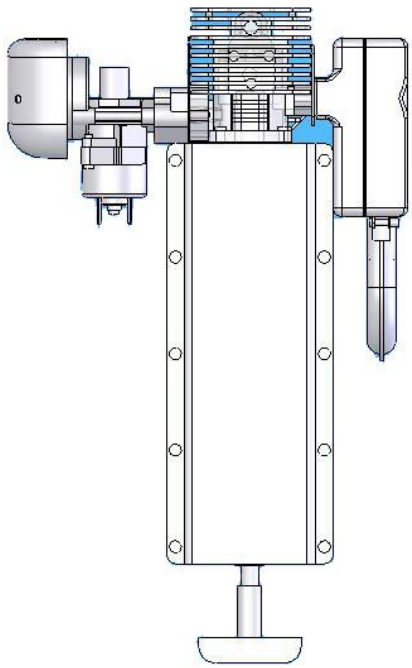


Fig. 28: Overall view of linear engine

3.10 Comparison between linear engine and conventional engine.

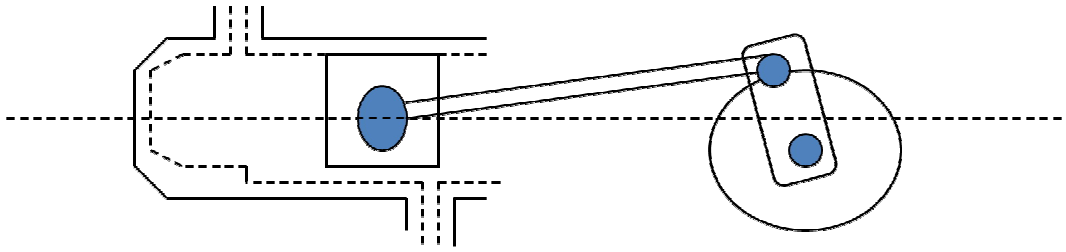


Fig. 29: Conventional engine

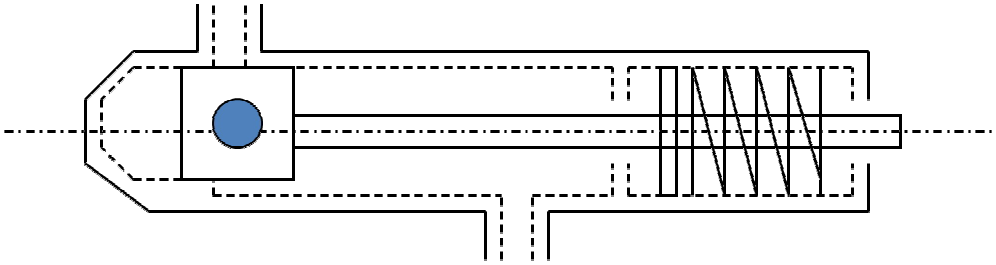


Fig. 30: Linear engine

CHAPTER 4

RESULT & DISCUSSION

In this section, the result from friction calculation and GT-power simulation is shown and described briefly. Data from calculation and simulation result is then converted into graph. All the graphs represent the performance of linear engine and conventional engine. Improvement of performance of linear engine is known by comparing it with the conventional engine performance.

4.1 Result

Table 3: Total Friction Mean Effective Pressure (Fmep) in Friction Analysis

Conventional												
RPM	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
Fmep-bar	2.4	2.2	2.2	2.3	2.4	2.4	2.5	2.6	2.7	2.8	2.9	3.0
Linear												
RPM	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
Fmep-bar	2.4	2.1	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.1	2.1	2.1

Table 3 shows the result from manual calculation for friction analysis. This calculation is done by using theoretical formulas from literature study. In order to simulate the linear engine in GT-power simulation, the friction factor has been considered.

All the conventional engine component's friction that are eliminated in linear engine is calculated and combined into one constant fmep value as shown in Table 3. This fmep constant is then inserted into GT-power modeling for linear engine simulation. To study and predict this linear engine performance, calculation of fmep is done with variable engine speed starting from 500 rpm to 6000 rpm.

Table 4: Conventional Engine Performance Simulation Data

RPM	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
Up [m/s]	0.5	1	1.5	2	2.5	3.1	3.6	4.1	4.6	5.1	5.6	6.1
Brake Power [kW]	0	0.2	0.3	0.5	0.7	0.7	0.7	0.6	0.6	0.5	0.4	0.2
Brake Torque [N-m]	0.7	1.6	2.2	2.5	2.5	2.3	1.8	1.4	1.2	0.9	0.6	0.3
IMEP [bar]	3.8	5.6	6.7	7.3	7.5	7.1	6.2	5.6	5.2	4.7	4.2	3.7
FMEP [bar]	2.4	2.2	2.2	2.3	2.4	2.5	2.5	2.6	2.7	2.8	2.9	3
BMEP [bar]	1.4	3.3	4.5	5	5.1	4.7	3.7	2.9	2.5	1.9	1.3	0.6
BSFC [g/kW-h]	2407.8	948.7	700.7	543.5	489.6	483.5	516.4	571.6	623.3	733	964.4	1691.5
Brake Efficiency [%]	3.4	8.7	11.8	15.2	16.9	17.1	16	14.4	13.2	11.3	8.6	4.9

Mechanical efficiency η_m [%]	35.9	60	66.8	68.7	68.6	65.6	59.3	52.6	47.9	40.3	30.6	17.4
Maximum pressure [bar]	36.4	41.6	44.7	45.8	45.4	42.9	38.4	35	33.3	31.3	29	26.4

Table 4 above shows the result of GT-power simulation for conventional engine. In this simulation, the friction factor is not considered. Thus fmep value for conventional engine is not used in the simulation. The modeling for this conventional engine's simulation is maintained with the standard modeling for 2 stroke engine. There is no modification done in the conventional engine modeling. The simulation is done with varies engine speed from 500 rpm to 6000 rpm to know the performance. From the simulation result, the performance parameters such as brake power, brake torque, imep, brake specific fuel consumption and others are obtained and represented in Table 4. The data from this simulation is converted into graph and explained briefly in the next discussion section.

Table 5: Linear Engine Performance Simulation Data

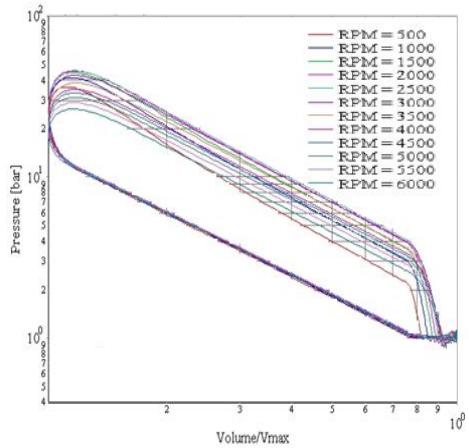
RPM	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
Up [m/s]	0.5	1	1.5	2	2.5	3.1	3.6	4.1	4.6	5.1	5.6	6.1
Brake Power [kW]	0	0.2	0.4	0.5	0.7	0.8	0.8	0.7	0.7	0.7	0.6	0.5
Brake Torque [N-m]	0.7	1.7	2.3	2.6	2.7	2.5	2.1	1.7	1.6	1.3	1	0.8
IMEP [bar]	3.8	5.6	6.7	7.3	7.5	7.1	6.2	5.6	5.2	4.7	4.2	3.7

FMEP [bar]	2.4	2.1	2	2	2	2	2	2	2	2.1	2.1	2.1
BMEP [bar]	1.4	3.5	4.7	5.3	5.5	5.1	4.2	3.5	3.2	2.7	2.1	1.6
BSFC [g/kW-h]	2286	908	667.2	512.7	456.3	440.5	451.6	473.3	490.3	524.6	585	694.9
Brake Efficiency [%]	3.6	9.1	12.4	16.1	18.1	18.7	18.3	17.4	16.8	15.7	14.1	11.9
Mechanical efficiency η_m [%]	37.7	62.7	70.2	72.8	73.5	72.1	67.9	63.6	61.1	56.3	50.5	42.2
Maximum pressure [bar]	36.4	41.6	44.7	45.8	45.4	42.9	38.4	35	33.3	31.3	29	26.4

The table 5 above is the result for linear engine simulation in GT-power. The table 5 shows linear engine performance parameters such as brake power, imep, brake efficiency and maximum pressure for the linear engine. This simulation is also run with range of engine speed from 500 rpm to 6000 rpm. The result from this linear engine simulation is obtained with some modification on conventional engine modeling. Fmep value from the previous manual friction calculation is inserted into the modeling to run simulation of linear engine. By comparing the result from table 4 and table 5, the improvement of performance for linear engine is known. The data from this simulation result is then represented by graph in the discussion section.

4.2 Discussion

In this discussion section, the graphs are built based on the simulation result on Table 4 and Table 5. The graphs represent engine performance parameters with some description on it.



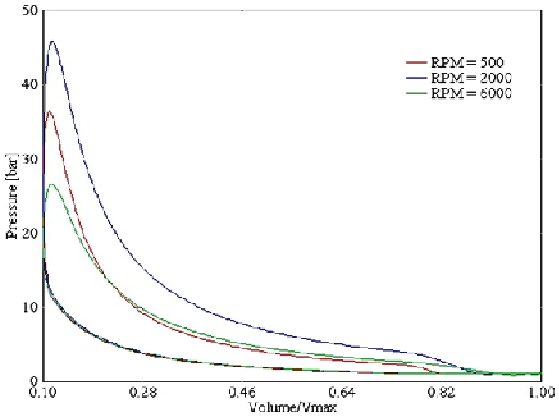


Fig. 31 (a) Log P vs log V Diagram **Fig. 31** (b) PV Diagram

Figure 31 (a) is Log P vs log V Diagram in varies engine speed and Figure 31 (b) PV Diagram for linear engine at critical condition. From the PV diagram, linear engine obtains its maximum pressure at 2000 rpm. Maximum cylinder pressure recorded is 45.75 bar. Pressure in the engine cylinder continously increases in the straight line when the engine speed increases in the beginning. However the pressure starts to decreases

with a slope after it pass its maximum value. Difference between maximum and minimum pressure recorded is 73.03 percent. Pressure and volume are in inverse relation. As one rises, the other falls. In the beginning, compression process push the piston move upwards , decreases the volume occupied and causes increases in the pressure and temperature. After reachs its maximum pressure at spark ignition phase, the pressure starts to decreases as the volume increases in the expansion process. It can be concluded that higher engine speed does not effects the increases in engine cylinder pressure.

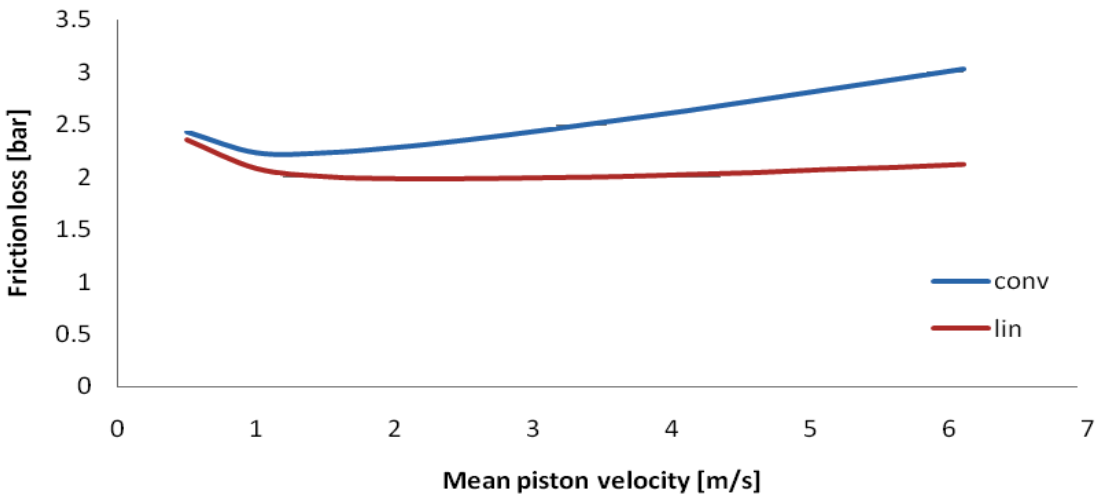


Fig. 32: Effect of friction loss at different mean piston velocity

Base on the Figure 32 shows above, linear engine records lower friction loss than conventional engine. Unlike conventional engine, linear engine friction loss start to decreases as the velocity increases and has almost constant value for higher piston velocity. The minimum value of friction loss for linear engine is 1.98 bars. Then the value obtains is around 2 bars as the speed increases. Both linear and conventional engine reaches its minimum friction loss between 1-2 m/s. Friction loss is less for linear engine because numbers in rotating parts that contribute friction is reduced.

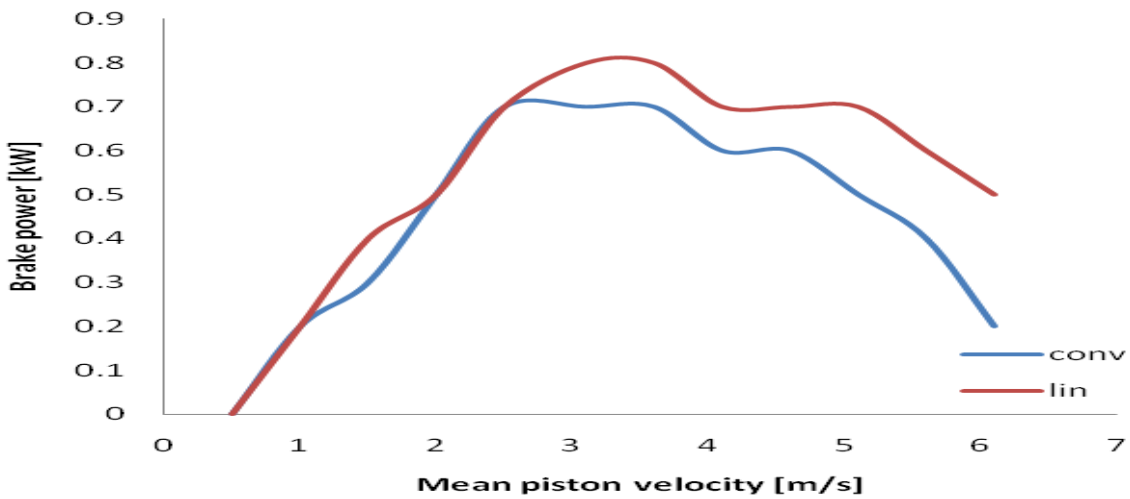


Fig. 33: Effect of brake power at different mean piston velocity

From the graph Figure 33, it can be describes that linear engine starting to gain more power at higher mean piston velocity. The linear engine power output start to increase after linear engine reaches 2.5 m/s piston velocity. The maximum power output is 0.8 kW which is 14.28 percent increases from linear engine. This maximum value occurs around 3.6 m/s. Linear engine performs better brake power due to less friction loss.

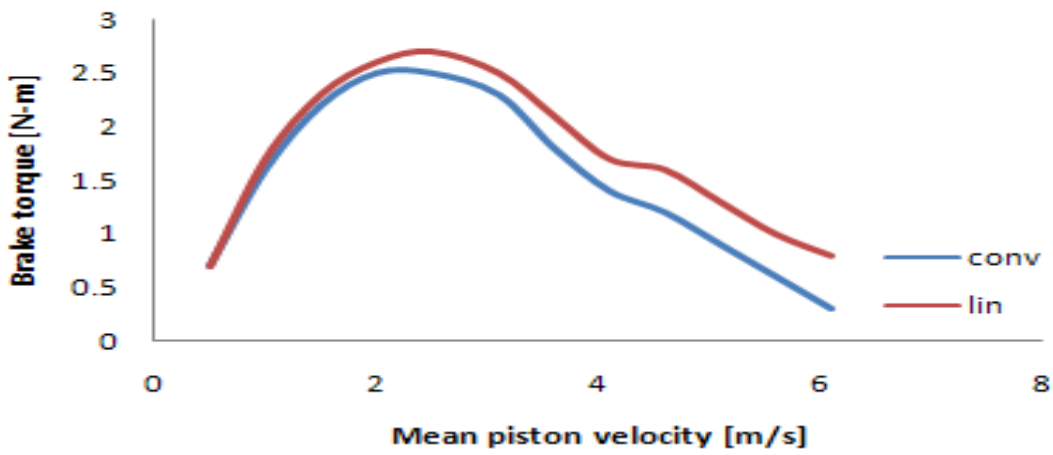


Fig. 34: Effect of brake torque at different mean piston velocity

Base on curve line performance, linear engine produces more torque compare with conventional engine starting from the beginning until the end. Maximum brake torque achieves by linear engine is 2.7 Nm which is 0.2 Nm difference from conventional engine. Maximum brake torque for conventional engine is 2.5 Nm only. The peak brake torque for both engines occurs at 2.5 m/s. From result before, it can state that brake power and brake torque occurs at different rpm. When the engine is starting, at lower rpm, it needs more torque than power to move to vehicle. After that, at higher rpm, it needs more power to operate the engine smoothly.

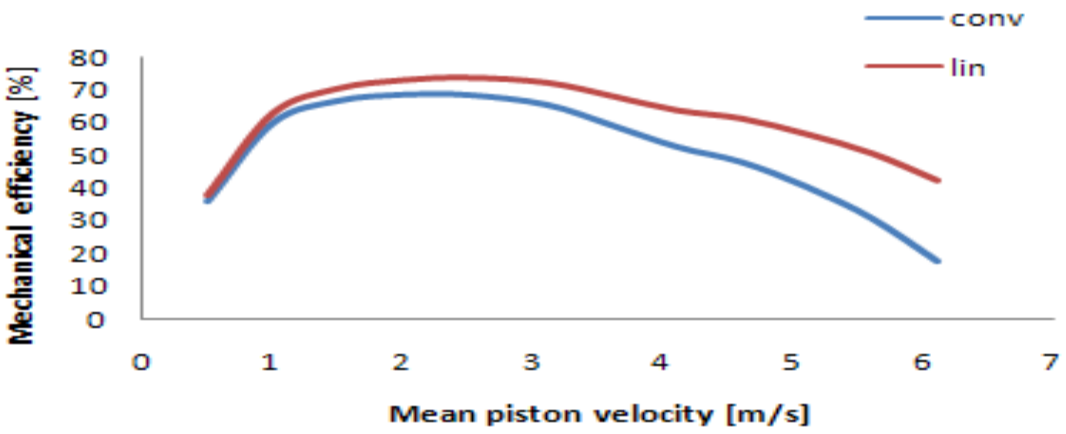


Fig. 35: Mechanical efficiency for both linear and conventional engine

The Figure 35 shows the difference in mechanical efficiency between linear engine and conventional engine at certain piston velocity. Linear engine to increases more from the beginning and its maximum mechanical efficiency occurs at 73.53 percent. Both engine starts to incline at low piston velocity and decline as the velocity increases after its peaks value. Linear engine has higher mechanical efficiency than conventional engine about 4.85 percent. Because of friction, the brake power of an engine is always less than the indicated power; hence the engine mechanical efficiency must be less than 1. Clearly, mechanical efficiencies as close to 1 as possible are desired.

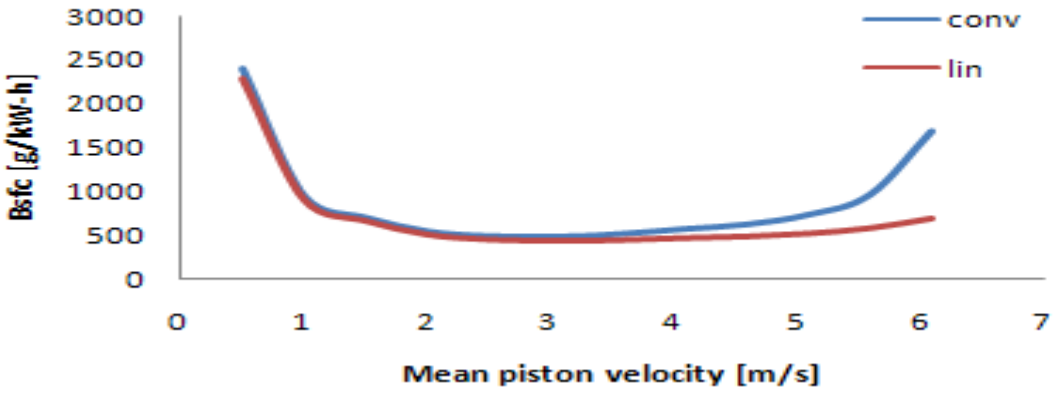


Fig. 36: Effect of brake specific fuel consumption at different mean piston velocity

The graph Figure 36 shows the brake specific fuel consumption for both engines, conventional and linear engine at elevated piston velocity. The linear engine will gives lower brake specific fuel consumption compare to conventional engine. Linear engine will start decreasing first in lower piston velocity. It has the lowest brake specific fuel consumption which the minimum value is 440.5 g/kW-h on 3.1 m/s. The difference between BSFC of linear and conventional engine is largest at 6.1 m/s which is 996.6 g/kW-h. Ergonomic area in the linear engine is between 2.5 to 4.6 m/s. It can be concluded that linear engine is more economical base on its fuel consumption.

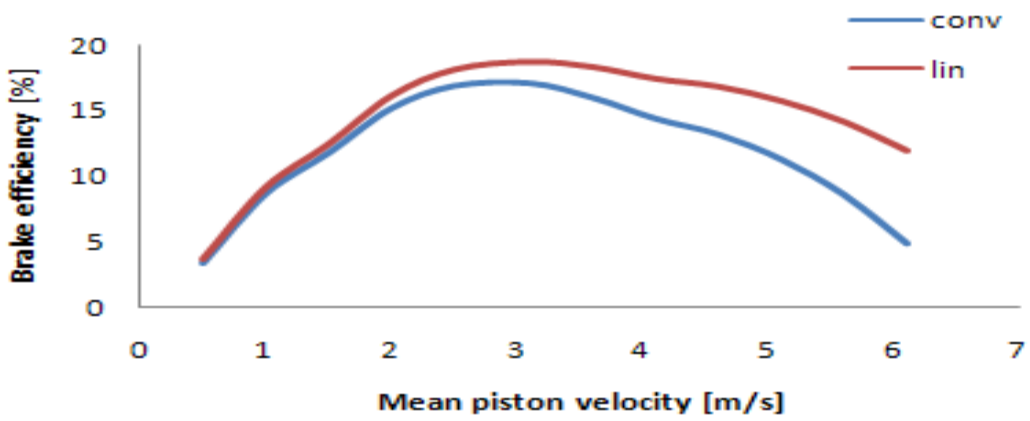


Fig. 37: Effect of brake efficiency at different mean piston velocity

The graph Figure 37 describes the performance of linear and conventional engine in terms of brake efficiency on certain mean piston velocity. Linear engine has better brake efficiency starting form the low piston velocity until high piston velocity. The highest brake efficiency produces by linear engine is 18.7 percent which is 1.6 percent higher than conventional engine. This is due to decreases numbers of rotating parts in linear that increases engine capability in braking.

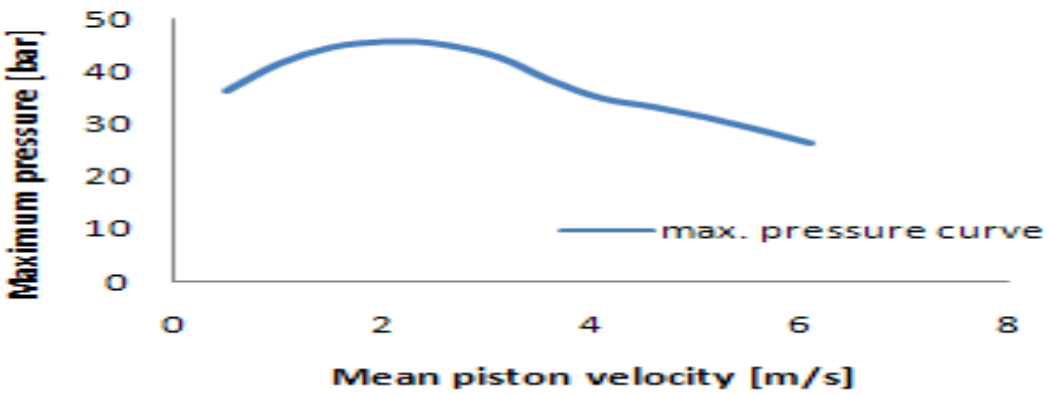


Fig. 38: Maximum pressure performance between linear and conventional engine

The graph Figure 38 above is about maximum pressure in the combustion chamber with variable engine speed. It can be seen that the pressure increases at low engine speed which is below 3 m/s. The pressure starts to incline from low piston velocity and achieves its maximum value 45.75 bars at 2 m/s. After reaches maximum value, the maximum pressure decreases as the piston speed increases. The percent of increases in maximum pressure is 25.72 percent from starting point. Maximizing cylinder pressure benefits horsepower and fuel economy. Increasing the compression ratio is one way of increasing cylinder pressure.

CHAPTER 5

CONCLUSION

In the end, performance of linear combustion engine can be predicted by modifying friction factor in GT-power simulation. Linear engine has better performance than conventional engine due to the increasing of brake power, 14 % and brake torque 8 % at their maximum point. Linear engine can be concluded more economical saving because it reduces brake specific fuel consumption (bsfc) about 9 % at the minimum point. Development of engine is more efficient by reducing number of part that contributes friction that will decreases performance of power output.

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APPENDIX

Simulation of Linear Engine in GT-power

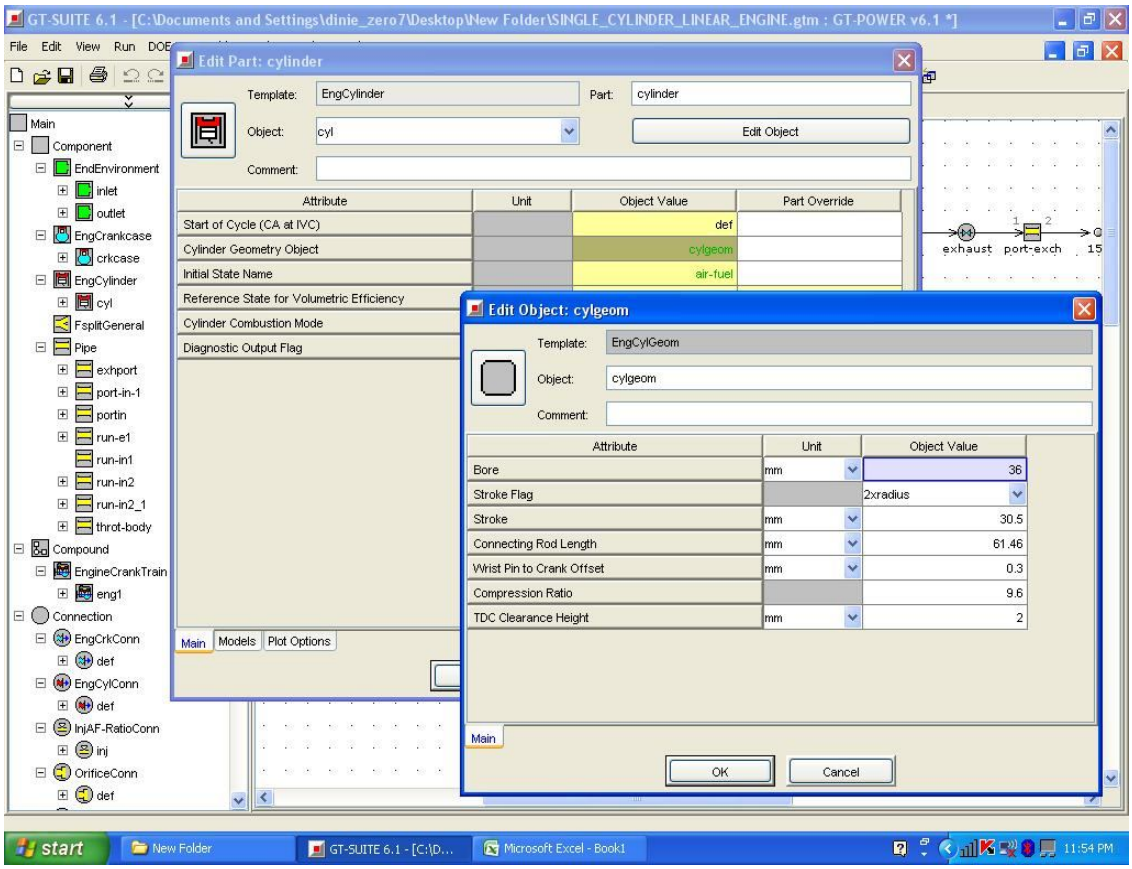

Fig. 39: Defining cylinder geometry for linear engine simulation

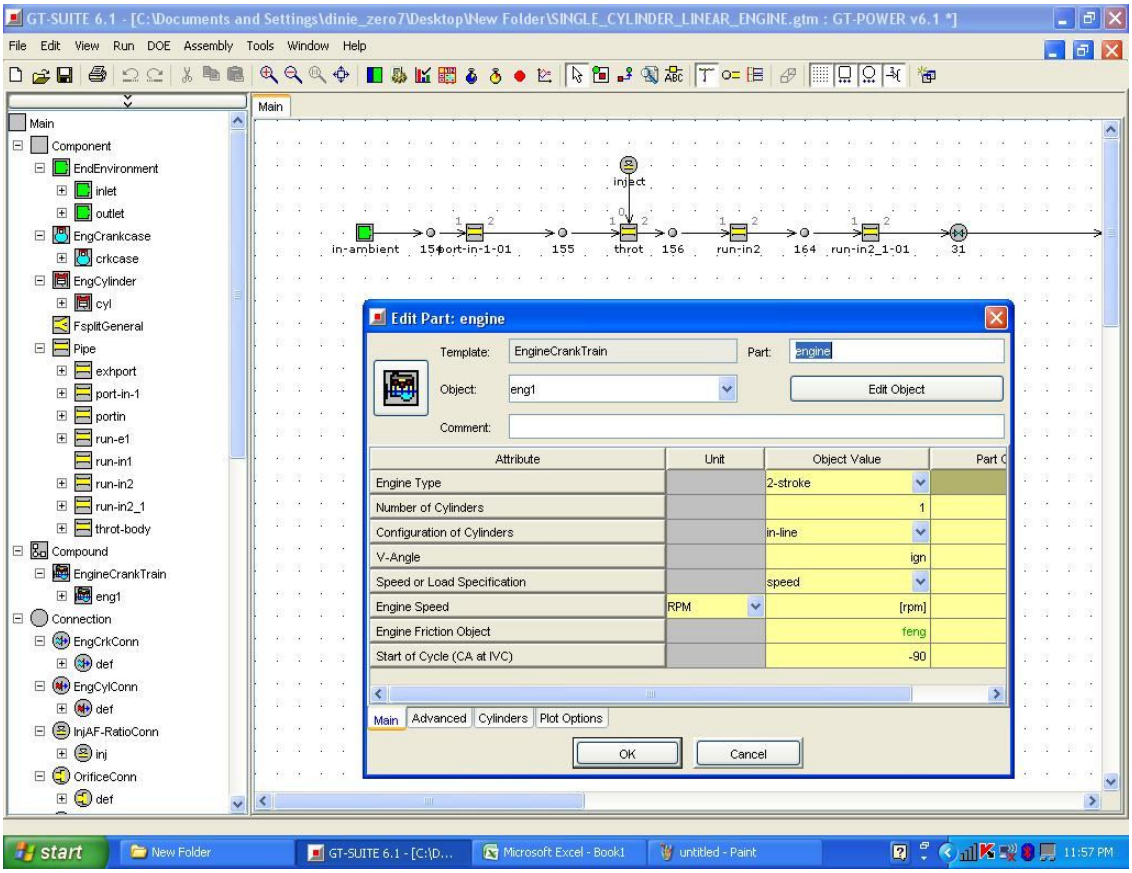

Fig. 40: Defining engine configuration

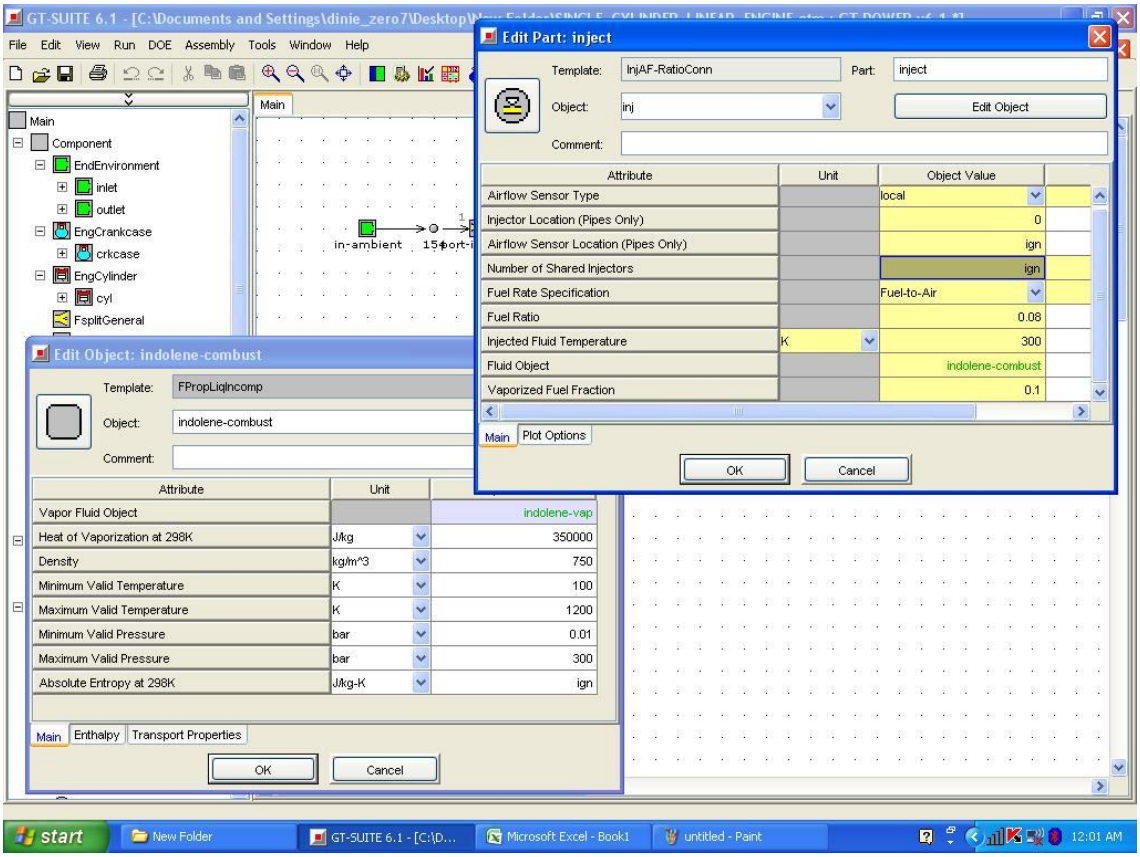

Fig. 41: Defining linear engine carburetion system

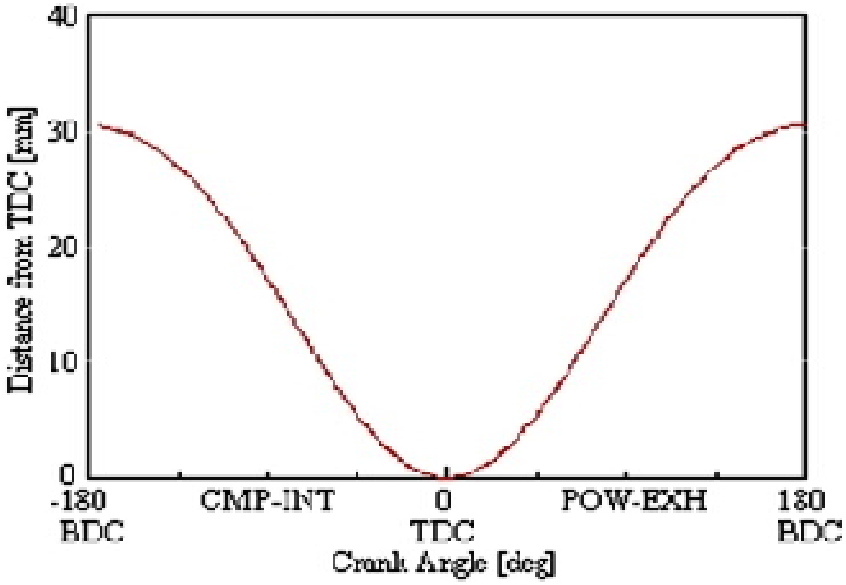

Fig. 42: Piston position at engine speed 2500 rpm on linear engine simulation

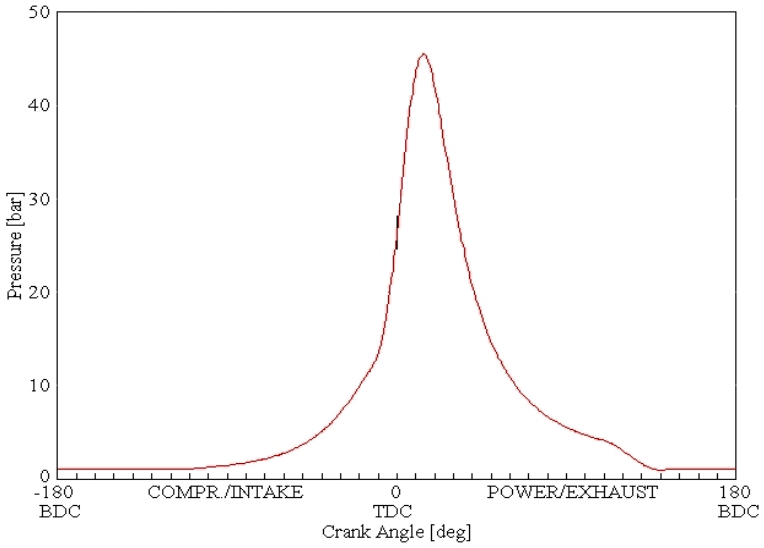

Fig. 43: Cylinder pressure on engine speed 2500 rpm

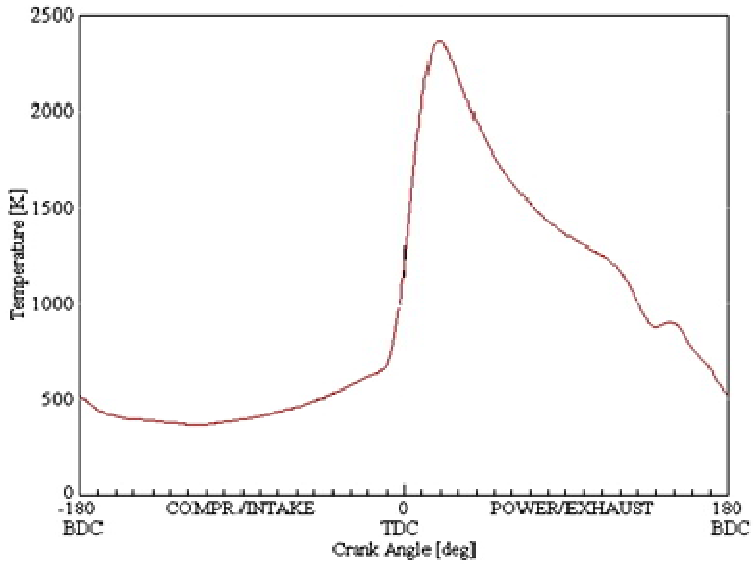

Fig. 44: Cylinder temperature on engine speed 2500 rpm

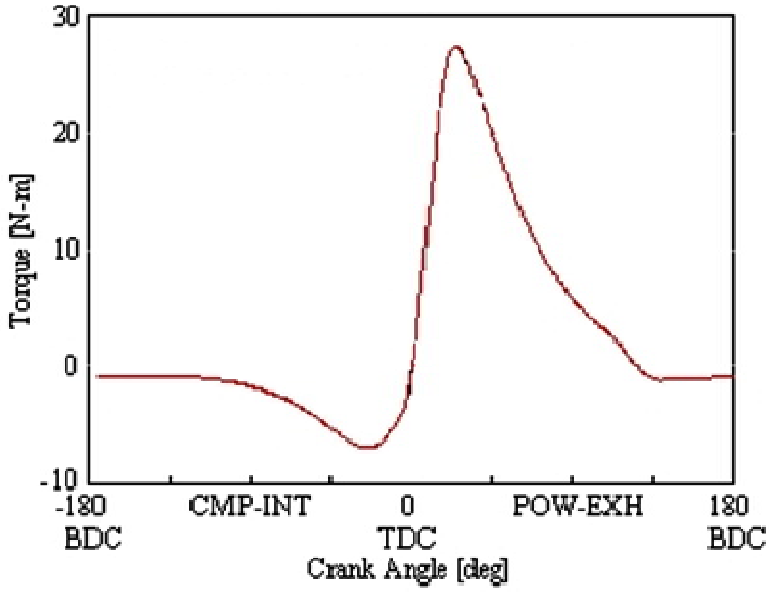

Fig. 45: Simulation of brake torque at engine speed 2500 rpm

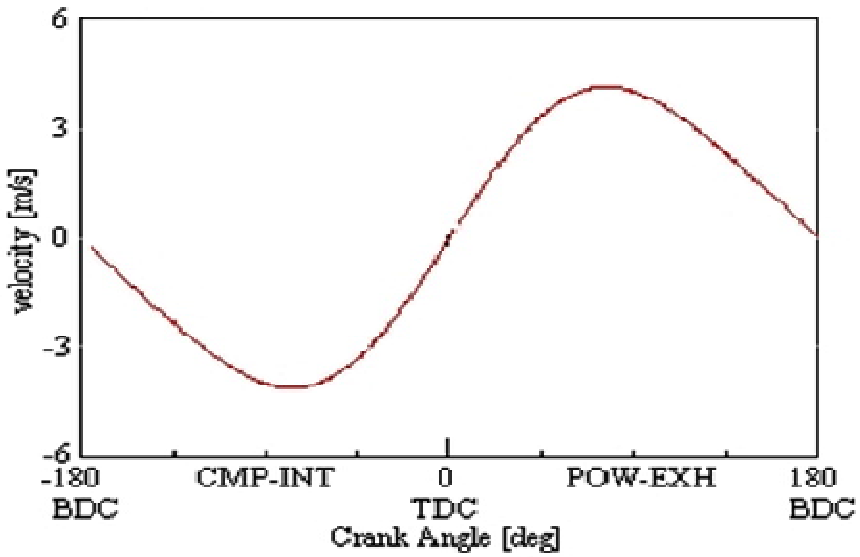

Fig. 46: Piston velocities at engine speed 2500 rpm

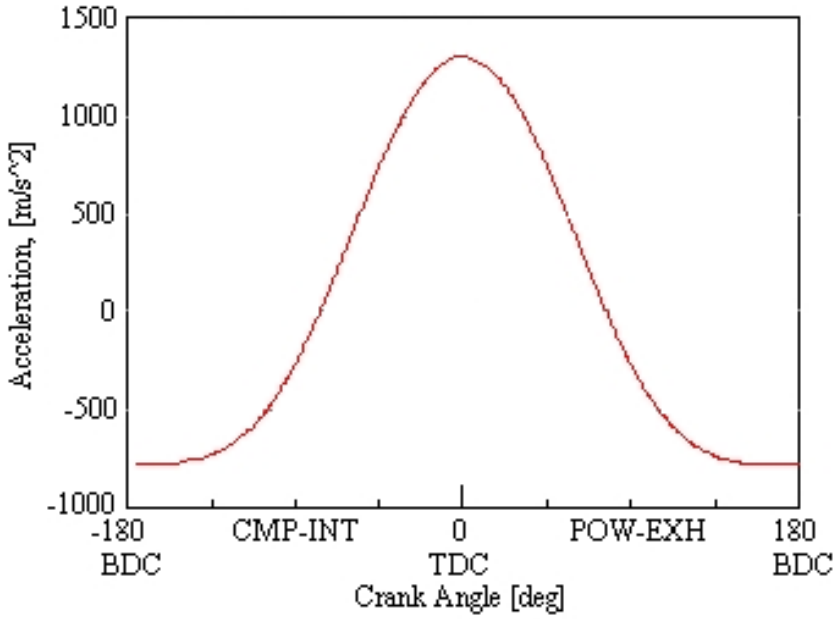

Fig. 47: Piston accelerations at engine speed 2500 rpm

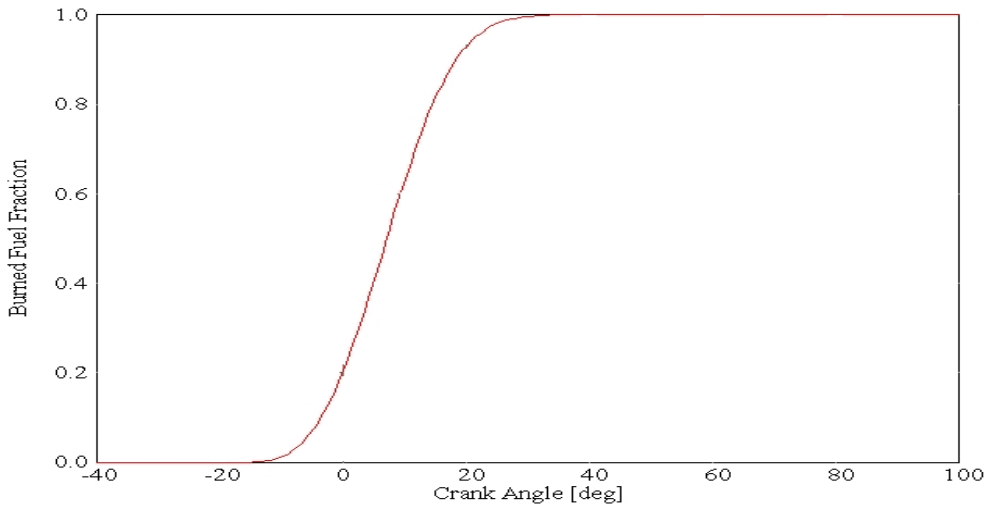

Fig. 48: Normalized cumulative burn rate at engine speed 2500 rpm

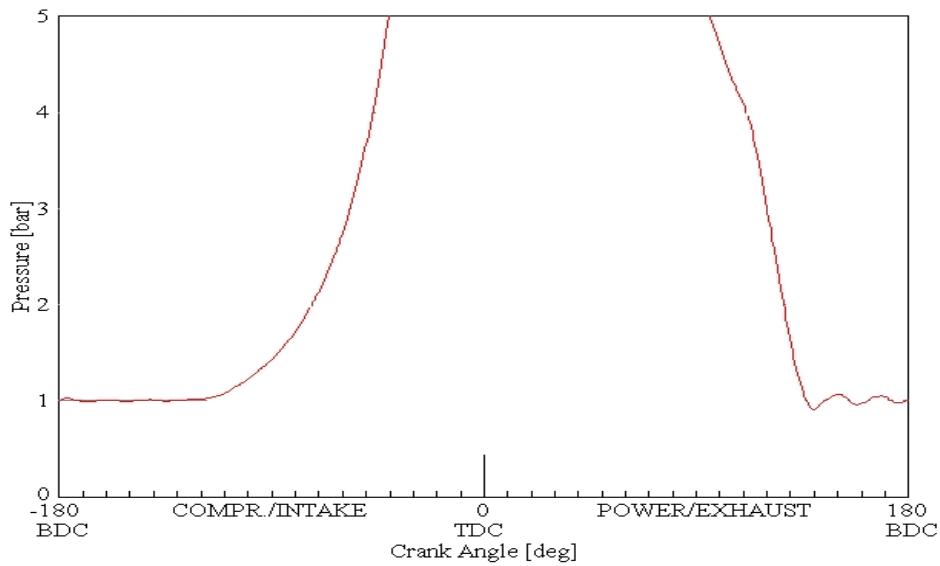

Fig. 49: Pressure pumping loop at engine speed 2500 rpm

Gantt chart Final Year Project 1 (FYP1)

Project Activities	w1	w2	w3	w4	w5	w6	w7	w8	w9	w10	w11	w12	w13	w14	w15	w16
Literature Study																
GT-power Programming																
GT-power Case Study																
Identify Problem Statement																
Define Objectives and Scope																
Detailed Methodology																
Proposal Preparation																
Proposal Submission																
FYP1 Presentation																

Fig. 50: Gantt chart Final Year Project 1 (FYP

Gantt chart Final Year Project 2 (FYP2)

Project Activities	w1	w2	w3	w4	w5	w6	w7	w8	w9	w10	w11	w12	w13	w14	w15	w16
GT-power case study (cont.)																
GT-power trial and calibration																
Data collection																
Data analysis																
Report writing																
Conclusion																
FYP2 presentation																
Report submission																

Fig. 51: Gantt chart Final Year Project 2 (FYP2)