SELECTION AND ANALYSIS OF ENGINE OF FSAE 2010

By

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Dissertation submitted in partial fulfillment of
the requirement for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

July 2009

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

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UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

December 2008

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

ROBERT HARJONI PEREIRA

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ABSTRACT

The objective of this project is to study, select and analyze the engine. This report includes introduction, literature review, methodology, result and discussions, and conclusion. The scope of study includes the usage of computer aided engineering software for the modeling and analysis purposes of the project. Theoretical calculations will also be carried out to compare the results with reality. In the literature review there will be a simple and basic explanation on engine and controllers, the specifications for Honda CBR 600 F4i, Yamaha FZ6 S2 and Aprilia SXV engines, the analysis of turbocharging small engines. The methodology section will consists of the project methodology which gives an outline of the activities taking place and also the Gantt chart which gives the milestone period for each activity. The comparison and analysis of the engines will be discussed in the result and discussion section.

CHAPTER 1
INTRODUCTION

1.1 Problem Statement

1.1.1 Problem Identification

The selection of an engine of a vehicle is an extremely important issue. The selection and analysis the engine have been made the main focus of this project. From the study of previous Formula SAE engine there are still areas where improvement can be made. Realizing this I will be undertaking this project on the selection and analysis of the engine to improve the current setup.

1.1.2 Significance of the Project

The aim of this project is to analyze and select an engine. So with this project, useful knowledge and information regarding the project will be obtained and can be used in the future. Also, this project will bring a better and upgraded design of the current engine.

1.2 Objectives

The main objectives of this project are:

- To perform selection and analysis of the proposed engine.
- To improve the current design of the engine system.

1.3 Scope of Study

The Formula SAE car will be designed in accordance with Formula SAE regulation. This project will cover the selection and analysis of the engine. For analysis purposes, computer software will be used. Theoretical calculations will also be done to ensure the reality is similar to the theoretical results.

CHAPTER 2
LITERATURE REVIEW

2.1 Engine

The engine will be selected according to the Formula SAE rules which states that the engine must be 4-stroke, displace less than 610 cc, and draw air through a circular passage of restricted diameter. Optimizing power with this restrictor makes the design of the intake system critical. Another important aspect of SAE rules is that vehicle noise levels must be less than 98 dB during any event at competition. This makes design of the exhaust system critical.

The basic principle of an engine is that the combustion of fuel occurs in a combustion chamber inside and integral to the engine. The high temperature and pressure gases that are produced by the combustion will apply force to the movable component of the engine such as the pistons.

Below is the basic process for a four stroke engine.

1. Intake
   - Combustible mixtures are emplaced in the combustion chamber.
2. Compression
   - The mixtures are placed under pressure
3. Combustion
   - The mixture is burnt. The hot mixture is expanded, pressing on moving parts of the engine and performing useful work.
4. Exhaust
   - The cooled combustion products are exhausted into the atmosphere.

All combustion engines depend on the exothermic chemical process of combustion which is the reaction of fuel, typically with oxygen from the air. The combustion process typically results in the production of a great quantity of heat, as well as the production of steam and carbon dioxide and other chemicals at very high temperature. All internal combustion engines must achieve ignition in their cylinders to create combustion. Usually engines use either a spark ignition method or a compression ignition system.

2.1.1 Honda CBR 600 F4i

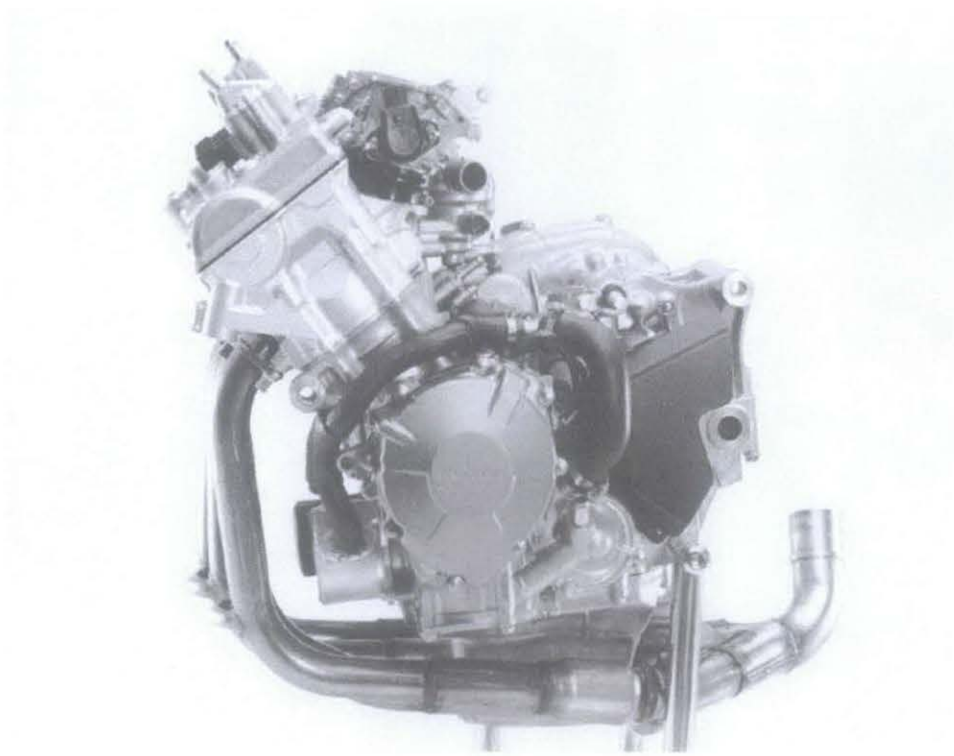


Figure 1.1: Honda CBR 600 F4i

Table 1.1: The specifications of the Honda CBR 600 F4i

Displacement	599.00 cc
Engine Type	In-line four
Stoke	4
Compression	12.0:1
Bore x Stroke	67.0 x 42.5 mm
Fuel system	Injection. Programmed Fuel Injection (PGM-FI) with automatic enricher circuit.
Valve train	DOHC
Cooling system	Liquid.
Transmission type final drive	Chain.

2.1.2 Yamaha FZ6 S2

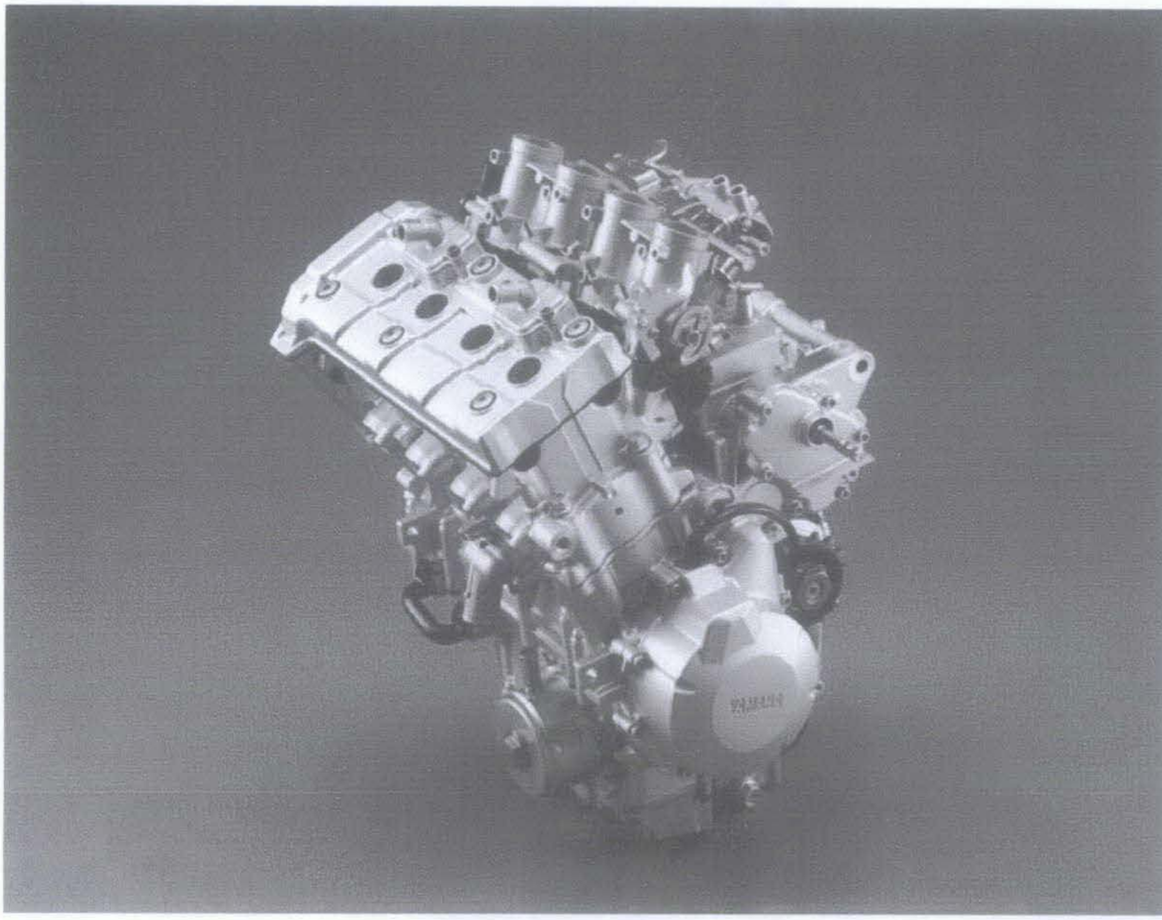


Figure 1.2: Yamaha FZ6 S2

Table 1.2: The specifications of the Yamaha FZ6 S2

Displacement	600.00 cc
Engine Type	In-line four
Stoke	4
Power	96.55 HP (70.5 KW)
Torque	63.10 Nm @ 6440 RPM
Compression	12.2:1
Bore x Stroke	65.5 x 44.5 mm
Fuel system	Injection.
Valvetrain	DOHC
Lubrication system	Wet sump.
Cooling system	Liquid.
Transmission type final drive	Chain.

2.1.3 Aprilia SXV 550

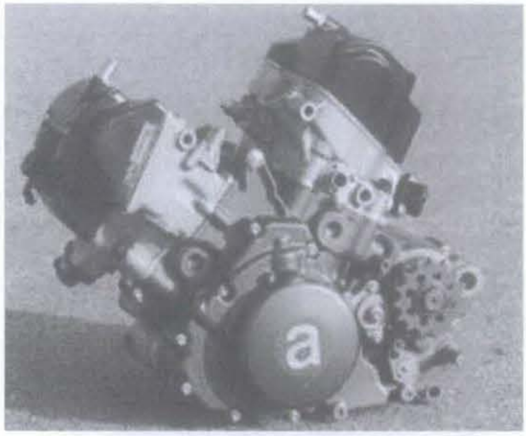


Figure 1.3: Aprilia SXV 550

Table 1.3: The specifications of the Aprilia SXV 550

Displacement	549.00 cc
Engine Type	77˚ V twin four stroke
Stoke	4
Power	66 HP
Compression	12.0:1
Bore x Stroke	80 x 55 mm
Fuel system	Injection. Integrated electronic engine management system controlling ignition and fuel injection.
Valvetrain	SOHC
Weight	33 kg.
Lubrication system	Dry sump.
Cooling system	Liquid.
Transmission type final drive	Chain.

2.2 Analysis

2.2.1 Turbocharging small engine performance limits.

Below is the finding from William Attard from the Department of Mechanical and Manufacturing Engineering, The University of Melbourne.

The aims included achieving near constant power over half the speed range together with the opportunity to increase peak power when compared to typical 0.6 L four cylinder reference engines. With appropriate design decisions, these aims could be achieved. With peak brake power limited to ~60 kW due to the limited air consumption, peak performance gains at the choked condition were only expected to be minimal and due to a combination of thermal efficiency improvements together with friction loss reductions. The strategy to reduce friction losses involved limiting the maximum engine speed to 10,000 rev/min as losses increased at the square of the speed increase. Furthermore, if the swept capacity could be reduced whilst keeping the rotating and reciprocating components as small as possible and still maintaining choked operation, delivered power would also increase due to the reduction in frictional losses associated with the smaller capacity.

The use of a suction device downstream of the restrictor allowed the maximum mass flow through the restrictor to be maintained over a wide speed range through delivering air at regulated boost, which would enable the above mentioned constant power performance aims to be achieved. Turbocharging was the preferred method of intake boosting over mechanically supercharging due to the documented ηTH improvements together with the high pressure ratios that were achievable. Furthermore, turbocharger boost limitations dictated the volume of the swept capacity while still maintaining choked flow operating conditions. Hence, the proposed engine design concept is comparable to downsized engines found in large automobile diesel and gasoline applications but on a reduced scale, with intake boosting used to compensate for the swept capacity reduction and additional mass benefits associated with the smaller engine. Whilst the requirement to achieve these results with sufficient suction on the compressor intake (needed to achieve sonic flow in the restrictor) would appear to be an excessive condition for a non-racing application, in reality it dictates better than usual turbocharger specification and performance. To determine the swept capacity, a compressible flow model was created to calculate intake air consumption based on the maximum mass flow that could be achieved in the choked restrictor condition. Hence, the engine capacity was selected with the aid of the figure below, which shows the predicted volumetric efficiency (ηVOL) needed to maintain sonic flow through the intake restriction for varying engine capacities and operating conditions.

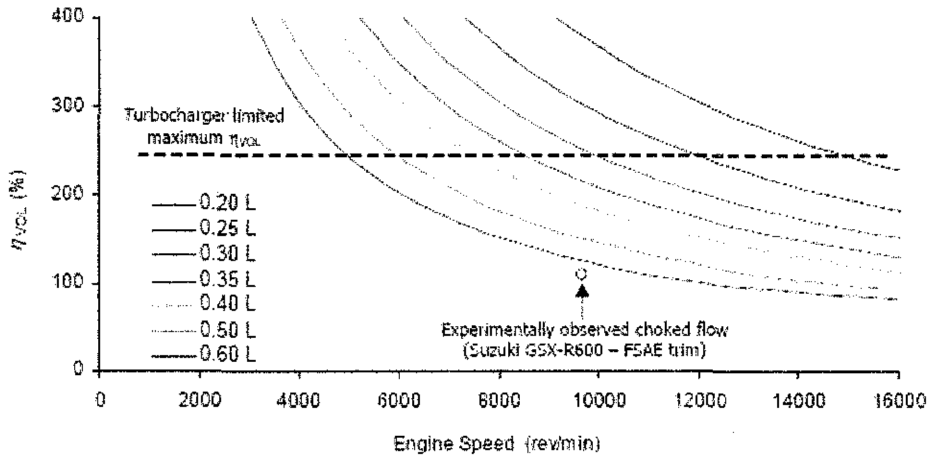


Figure 2.1: Predicted engine air consumption needed to maintain choked flow operating conditions for varying swept capacities and operating conditions with model validation from previous experimental results.

The final results

TC brake performance optimization was significantly influenced by CFD tools, with the importance of using simulation as an aid in accelerating engine development clearly demonstrated. These tools have enabled improved TC performance and understanding at reduced costs and lead times. The figure below compares the experimental and simulated TC performance data for the test engine. Experimental and simulation results show that by successfully implementing turbocharging to a flow restricted engine, near constant power can be achieved over a wide speed range. This is due to the intake restriction, which limits airflow and thus performance when operating at the choked condition. The potential also exists to further improve low speed performance (prior to the intake restriction limiting airflow) with VTG. Experimental results also show that peak power occurs at the lowest flow restricted engine speed (6,000 rev/min) due to the reduced friction losses which increases brake power.

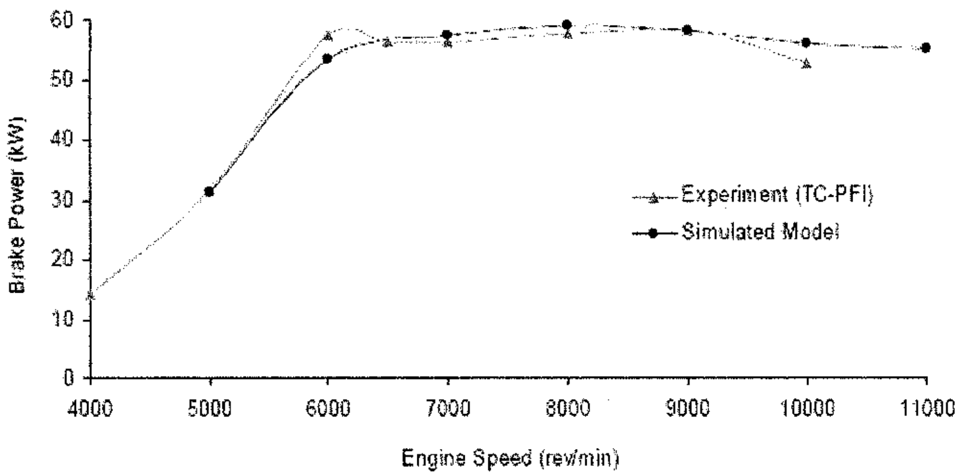


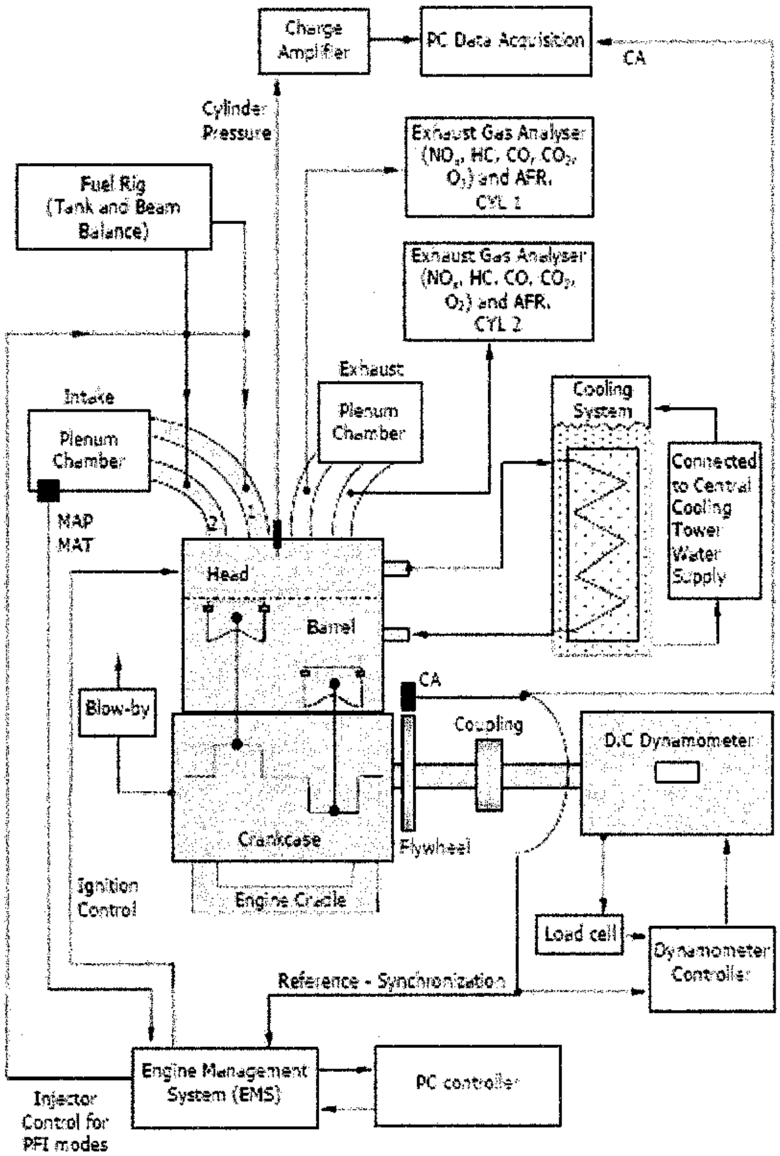
Figure 2.2: Comparison of experimental and predicted engine performance for the test engine operating in the TC mode.

The importance of using a validated simulation model in setting the engine specifications during the initial design phase is clearly demonstrated in the figure above. Excellent agreement between actual and simulated performance is evident, with minor differences at 10,000 rev/min when the actual brake power tends to fall faster than simulated. This is partly explained by higher pumping losses than simulated and increasing friction losses associated with increasing component flexure at the high engine speeds. In addition to the performance comparisons of the figure above, the development process used to improve TC performance through intake and exhaust geometries together with valve events further illustrates the agreement with simulation results.

[Reference Number 9]

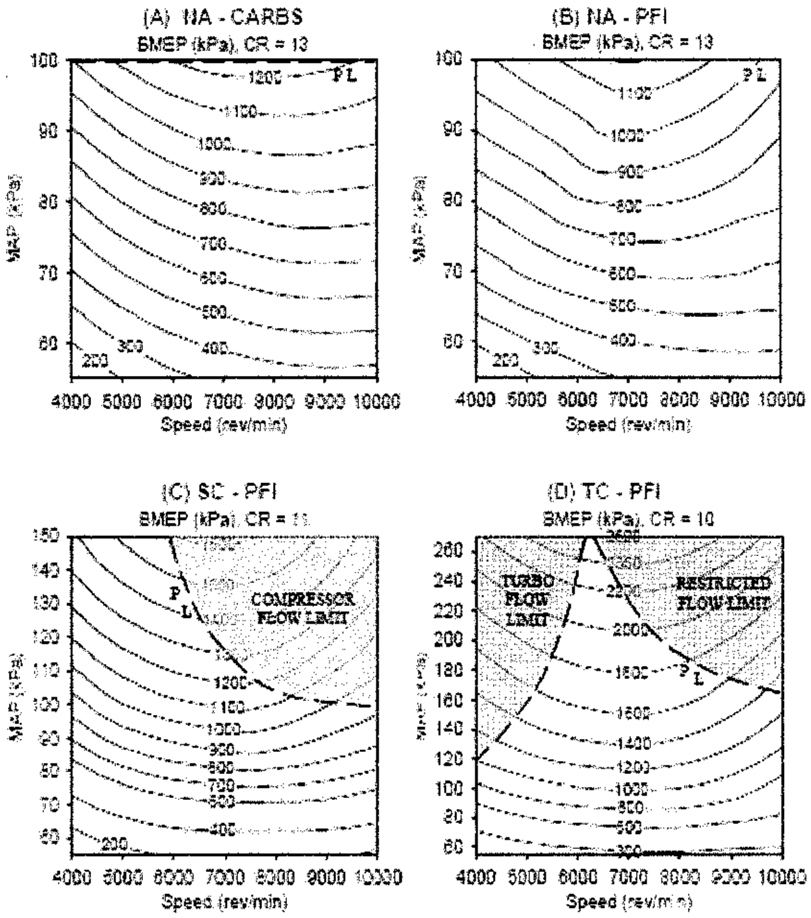
Below is the finding from William Attard, Harry C. Watson, Steven Konidaris and Mohammad Ali Khan from University of Melbourne.

The experimental setup

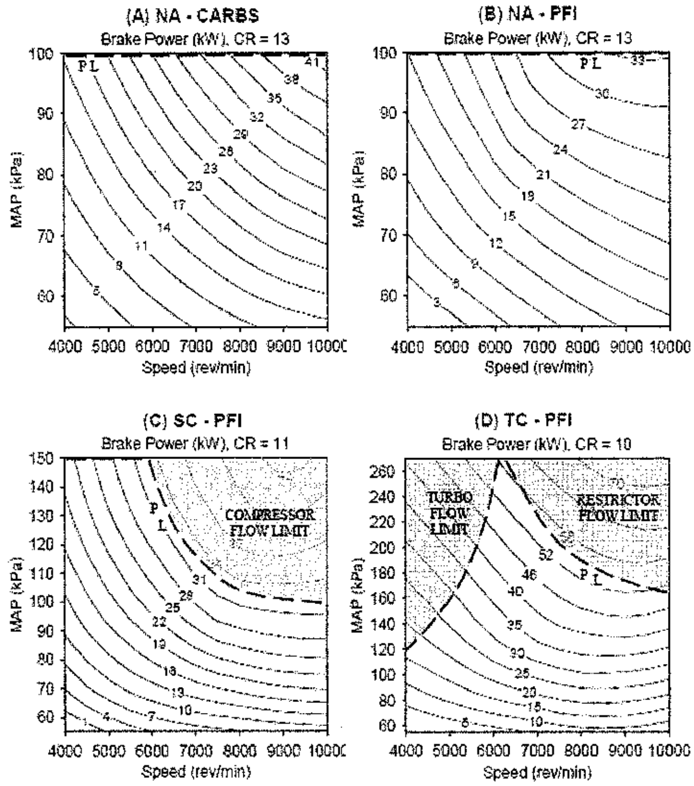


The experimental results

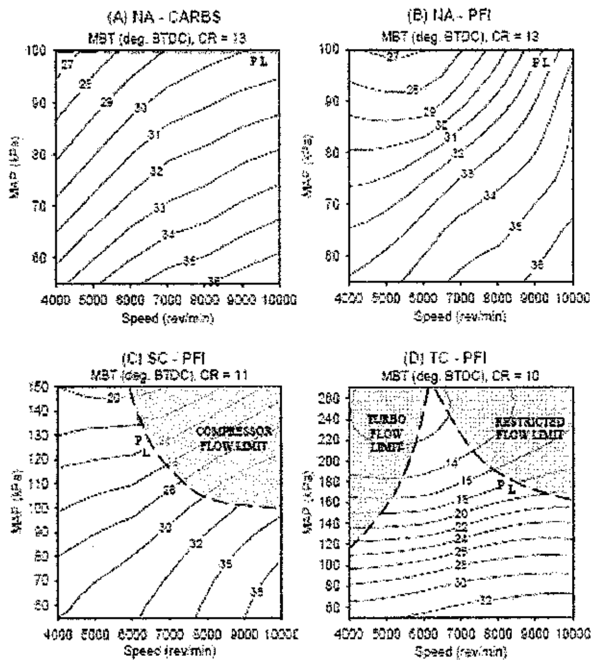
Brake Mean Effective Pressure (BMEP)



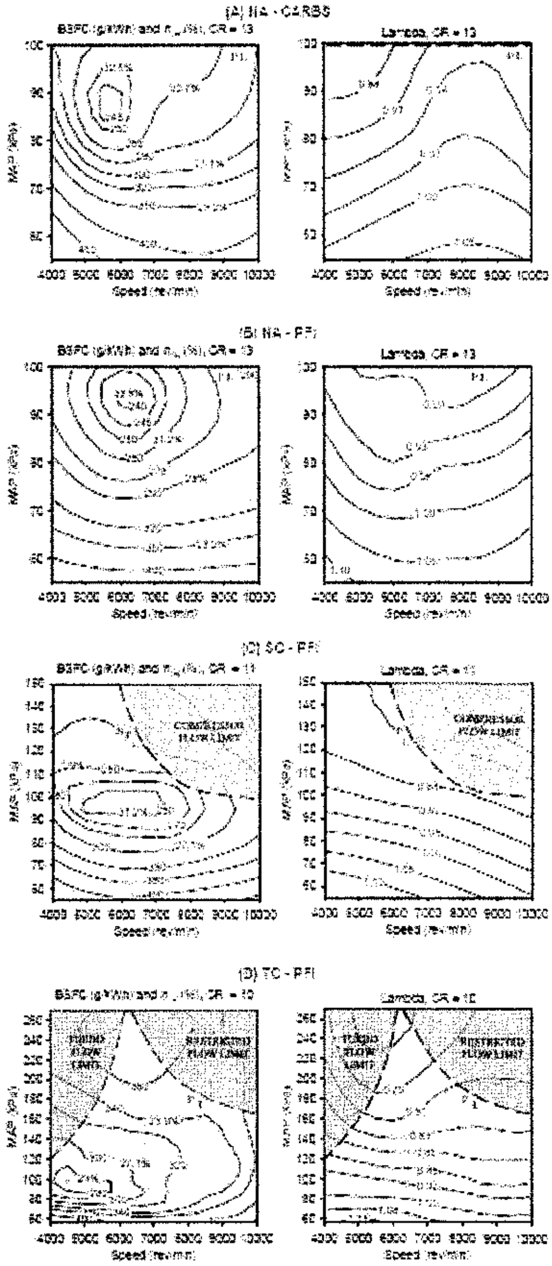
Brake Power



Spark Timing



Brake Specific Fuel Consumption (BSFC) and Thermal Efficiency (η_{TH})



[Reference Number 10]

Conclusion

The performance and efficiency of a downsized Formula SAE engine running in a variety of NA and forced induction modes on 98-RON pump gasoline has been described, with the limiting factors compared. Modes are defined by variations in the induction system and associated engine modifications, namely compression ratio optimization, which was needed to avoid uncontrolled end-gas knock and maximize η_{VOL}. These modes included:

(A) NA with carburetion

(B) NA with PFI

(C) Mildly SC with PFI

(D) Highly TC with PFI

The test engine used in experiments was specifically designed and configured for the purpose, being a 430 cm^3, twin cylinder in-line arrangement with double overhead camshafts and four valves per cylinder.

A peak value of 25 bar BMEP was achieved while running in the highly TC mode, believed to be the highest recorded value for small engines on pump gasoline. This exceeds GM's highest specific output, recently reported to be 22 bar from their 2.0-L TC Ecotec engine. The increased performance from the WATTARD engine is due to the reduced bore size, which promotes increases in engine speed and end-gas volume reductions around the periphery of the chamber. This allows CR and/or MAP values to be increased before exceeding the DL.

Knock has been highlighted as being the single most important limiting factor in defining the performance for downsized boosted engines. Testing demonstrated that spark retard and/or fuel enrichment can be used as a method of knock control for up to 1-2 CR points depending on the knock severity. Testing also showed that knock severity is highest on the intake side in a pent roof combustion chamber. This is due to the knock onset location's dependence on the flame speed, which is severely reduced as the flame travels over the cooler surfaces of the intake valves.

Experimental results showed BSFC or η_{TH} values in the order of 240 g/kWh or 34% could be achieved. Constant power could also be achieved over half the speed range in the highly TC mode due to the Formula SAE regulated intake restriction, which limited power. The engine was installed into successive MUR Motorsport vehicles in 2003 and 2004 and became the first prototype engine to successfully compete in the competition. The engine and vehicle package proved to be very competitive, finishing first in the fuel economy event at the 2004 Australasian competition.

Experimentation and competition results have proven that the performance of downsized engines can match that of their larger counterparts, with the aid of intake boosting. However, the extent to which swept volume can be reduced in any downsized application is combustion limited. If the combustion in high speed, small bore engines could be better understood or even enhanced to promote faster burning ,the severity of end-gas knock could be minimized. This would allow further increases in CR and/or MAP, resulting in increased performance and efficiency.

2.2.2 Analysis of Engine Speed Effect on Temperature and Pressure of Engine Based on Experiment and Computational Simulation.

Below are the analysis done by N.M.I.N. Ibrahim, Semin, Rosli A. Bakar, Abdul R. Ismail and Ismail Ali from University Malaysia Pahang.

The experimental results shown that the higher rpm mode produced the highest cylinder pressure compare other lower rpm as shown in Fig. 5. The higher speed may reach nearly 9.04 bar and at the speed 1100 rpm and 1400 rpm showed the value of 8.43 bar and 8.73 bar, respectively. All data is taken at motoring condition which is for flow process without firing. The pressures started increase at the -1010 BTDC for all speeds until achieve the maximum values when exhaust port started close (EPC). From this experiment also shows that when pressures reach at the 880 ATDC which is exhaust port started open (EPO), the pressures drop to negative values cause of residual form exhaust port enter to cylinder called backflow.

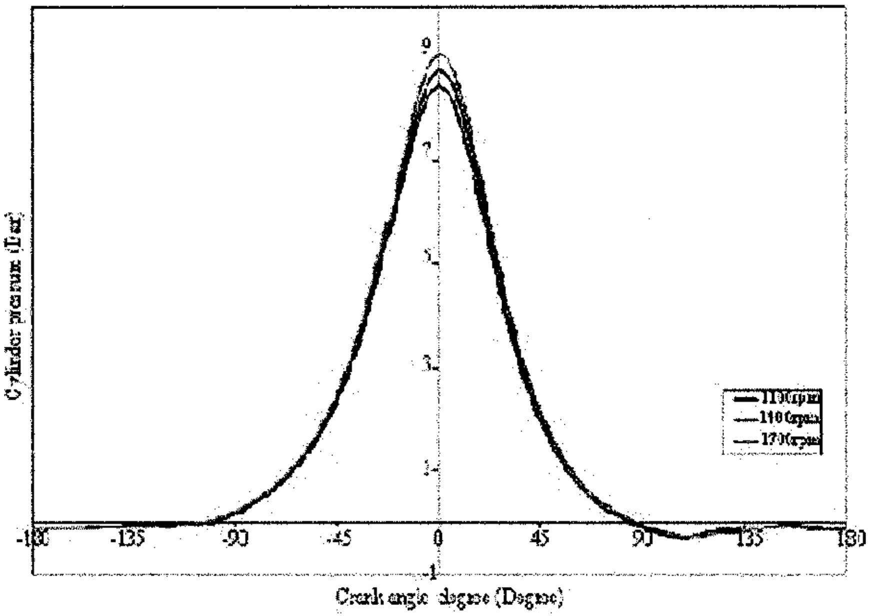


Fig. 5: The effects of cylinder pressure versus Crank angle degree

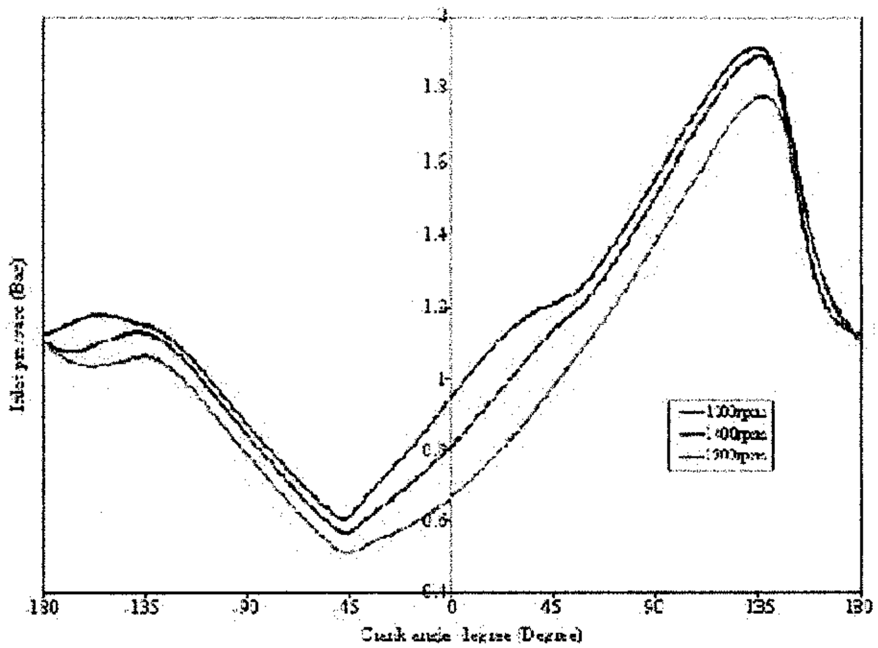


Fig. 6: The effects of intake pressure versus Crank angle degree

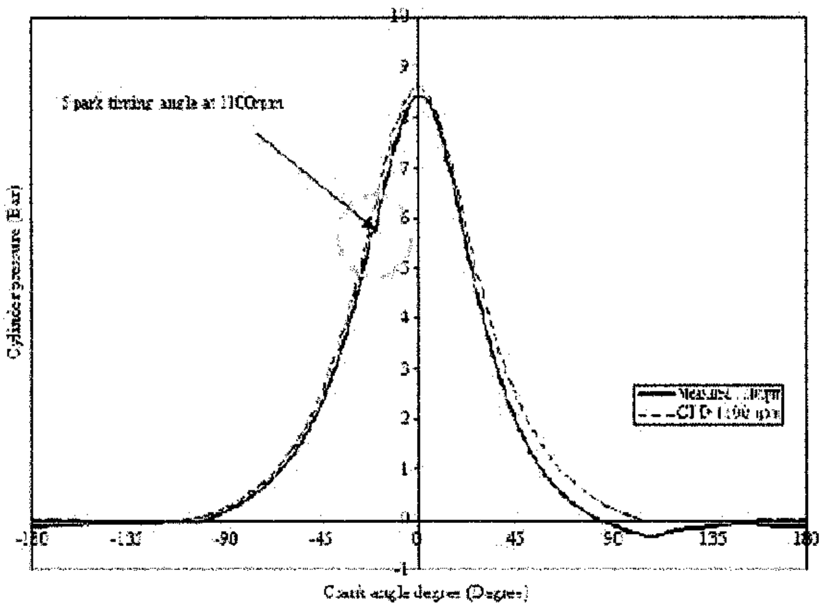


Fig. 7: The effects of cylinder pressure versus Crank angle degree at 1100 rpm

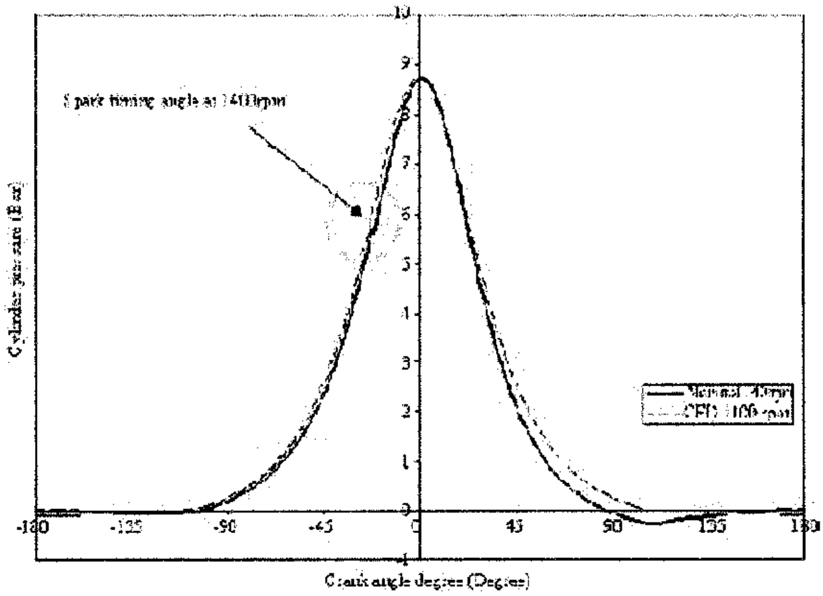


Fig. 8: The effects of cylinder pressure versus Crank angle degree at 1400 rpm

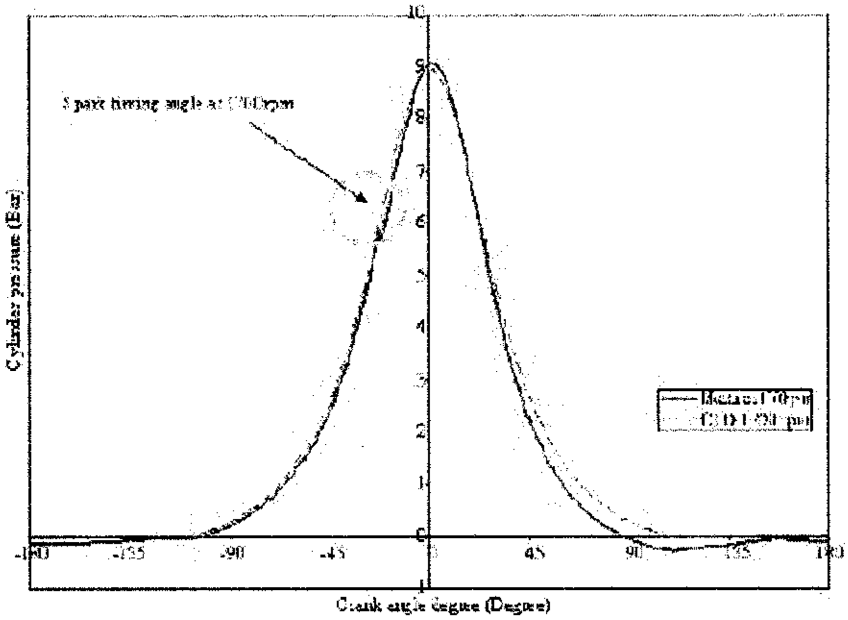


Fig. 9: The effects of cylinder pressure versus Crank angle degree at 1700 rpm

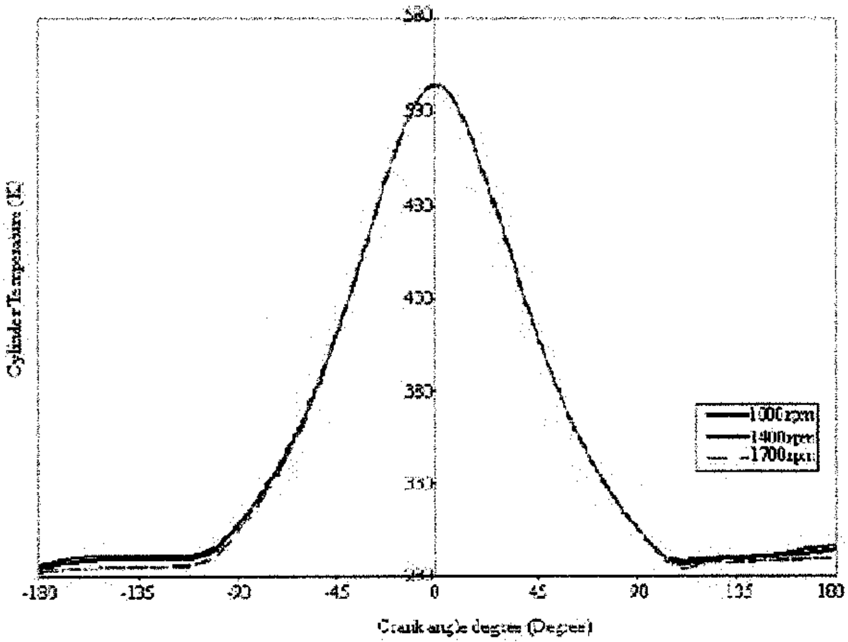


Fig. 10: Effects of cylinder temperature versus Crank angle degree

Fig. 6 shows that the intake pressures at the different of speed function of crank angle degree. The trend of profile show different compared with cylinder pressure. At the lower speed of engine can achieve the maximum pressure at the 1100 rpm and reduced for the 1400 rpm also 1700 rpm. The profile explained of process happen at the crankcase area. The crankcase pressure is similar with the intake port value of pressure. Form the overall figure, concluded that the minimum pressure occur when the fresh charge enter to crankcase through the transfer port and shown that at -450 BTDC the pressures was dropped. When the intake port started open (IPO) at 1400 ATDC, the pressure of intake port was increased and can reach maximum values. At this time the fresh charge enter to cylinder at the high velocity and reduced when piston reaches at BDC. Form this pressure profile can be concluded that the higher speed can produce minimum pressure compared the lower speed of engine. These data will used as boundary condition for simulation approach. Cylinder pressure data was measured in experimental, and introduce in simulation stages to study the characteristics of in-cylinder behavior. The cylinder pressure data provided a more than satisfactory result thus increasing the confident level to explore more details on the next stage of this study which is scavenging analysis studies. Fig. 7 and Fig. 9 were shows the cylinder pressure results as a function of crank angle degree for the different speed. Fig. 7 and Fig.8 clearly shows that the simulation higher than the experimental and opposite for Fig. 9. The experimental results show the values for 1100 rpm, 1400 rpm and 1700 rpm gave 8.43 bar, 8.73 bar and 9.04 bar, respectively. On other side, the simulation results gave 8.59 bar, 8.75 bar and 8.95 bar for the same speed as above. As theoretically explained that, the simulation results must always give higher than measured values because refer to ideal cycle and not consider factor such as heat losses. Fig. 7 given relative error is 1.86 % and followed by Fig. 8 with 0.22 %. Comparatively, results for the Fig. 9 shows that the experimental higher than simulation at 1 % of relative error compared based the experimental. The trends of Fig. 9 show that when the speed increased, the simulation results will be decrease at the small of discrepancy. When the exhaust port started open (EPO) at 880 and 1010 give at negative value for measured and simulation, respectively. The values seen that small for simulation and show clearly at high value for experimental. It explained that influenced of this modeling process. The simulation stage is complexity of geometries which is modeled based on approximation values especially at the angle port geometries. Due to the same reason, the engine port timing was also change, since two-stroke engine timing is port controlled. The approximation of geometries is measured using the Coordinate Measurement Machine (CMM) measurement and manual measurement. But CMM measurement gives an accurate measurement on plane only. The difficulty of getting accurate geometry profile occurs since the actual geometry is three dimensional. Complex geometries at the intake, exhaust ports angles, dome and volume clearance high of the engine are approximated. Manual measurement of actual geometries slightly reduced the actual compression ratio (CR) which influences the increasing of pressure in cylinder. Fig. 10 shows the cylinder temperature profiles are compared at different of speed. This figure illustrated the prediction of cylinder temperature. The profiles show similar trend with cylinder pressure figure. The higher values at the middle of process caused by the compression process started when piston at bottom dead centre (BDC). The values of cylinder temperature will decrease while piston start leaves the top dead centre (TDC). As theoretically, from the relationships between pressure P, temperature T and volume V. If the cylinder pressure increase, the temperature also increase when the volume of cylinder decrease while the piston motion move upward. At the TDC position, the maximum at 1100 rpm, 1400 rpm and 1700 rpm gave 543.9 K, 544.5 K and 544.2 K, respectively. It concluded that the speed of engine increase the temperature also increase except the last speed at 1700 rpm. In this case, the investigation of flow in cylinder not considered the combustion process and the temperature values also not show the higher compared the combustion process.

Conclusion

The experimental results showed that the higher rpm mode produced the highest cylinder pressure compare other lower rpm. The simulation result shows that the cylinder pressure results as a function of crank angle degree for the different speed. The simulation of cylinder pressure is higher than the experimental. In this temperature case, the investigation of flow in cylinder not considered the combustion process and the temperature values also not show the higher compared the combustion process.

[Reference Number 11]

2.2.3 Compression Ratio Effects on Performance, Efficiency, Emissions and Combustion in a Carbureted and PFI Small Engine

Below are the analysis done by William P. Attard, Steven Konidaris, Ferenc Hamori, Elisa Toulson and Harry C. Watson from the University of Melbourne.

The original intent of this development program was to achieve success in Formula SAE competition. However, from the research and development process, more significant findings concerning small engines have been discovered. This paper provides some insight into the CR effects, giving direction for future development of small scale engines for passenger vehicles. Specifically, the objectives are to:

- Experimentally explore the effects of CR variations on performance, efficiency and emissions for a small engine across engine speed and MAP domains.
- Compare CR effects and combustion operating limits between the small engine and larger cylinder bore engines found in passenger vehicles.
- Compare carburetion and PFI fuel delivery systems in a small engine.
- Determine combustion parameters for a small high speed engine, comparing the effects of CR and fuel delivery variations across the speed range.

It should be noted that all brake data presented in this paper corresponds to the performance at the gearbox output shaft and not at the crankshaft. This is due to the engine design featuring an integral clutch and transmission within the crankcase. Performance at the crankshaft is expected to be marginally higher, due to the reduction in parasitic losses associated with driving the transmission components. Testing for each mode commenced at the highest CR, which was limited to 13. This was the highest achievable with a flat top piston in the pent roof combustion chamber. The decision to use a flat top piston was based on manufacturability, with simple machining processes allowing reductions in CR. Hence, variations ranging from 9-13 were possible through piston crown modifications to a set of custom forged pistons. It is noted that squish areas around the periphery of the chamber were maintained to minimize the differing effects of turbulence and resulting combustion effects for varying CRs.

Results

Contour plots are used to display the large quantities of performance, efficiency and emissions experimental results. Results are presented in brake format with varying engine speed, MAP and CR parameters. The contour plot data is presented across two domains as follows:

• Engine speed versus MAP at the highest achievable CR

• CR versus engine speed at the highest achievable MAP

This allows trends to be established and visualized to quantify parameter variation effects. Moreover, plots also allow comparison to larger bore engines to quantify any performance, efficiency or emission effects attributed to the reduced cylinder capacity. These effects are summarized in Table 4, which compares the test engine's experimental results to published data for modern, larger bore engines found in passenger vehicles.

BRAKE MEAN EFFECTIVE PRESSURE (BMEP)

Figure 2.3 displays BMEP contours for both carbureted and PFI modes. When compared to larger bore engines, trends show matching BMEP effects for varying MAP and CR. Increases in BMEP are shown to be directly proportional to increases in MAP for all speeds, primarily due to increases in air consumption and ηTH. Increases in CR are shown to directly correlate to increases in BMEP for all speeds. The increases in BMEP are attributed to a combination of both increased air consumption and improved combustion. As the CR is increased, the residual gas fraction within the cylinder decreases. Reducing the hot products within the cylinder minimizes the warm up of the incoming charge during the induction process, improving charge density and air consumption. A bulk increase in charge density also enhances the combustion process with improved burning rates. A BMEP improvement in the order of 10-13% was recorded across both carburetion and PFI with a CR increase from 10 to 13. These values closely correspond to experimental results recorded in larger bore engines together with fuel-air cycle analysis. However, further performance improvements associated with CR increases are primarily limited by knock together with piston crown geometry constraints as previously outlined. At the highest test CR of 13, peak BMEP values in the region of 1200-1300 kPa were recorded at mid range speeds. These small engine values are compared to larger bore OEM engines fitted to passenger vehicles as outlined, which achieve in the order of 1000-1100 kPa BMEP. The BMEP performance discrepancies are largely associated with the CR differences associated with the differing bore sizes and engine speeds, further highlighting the performance potential for small cylinder engines.

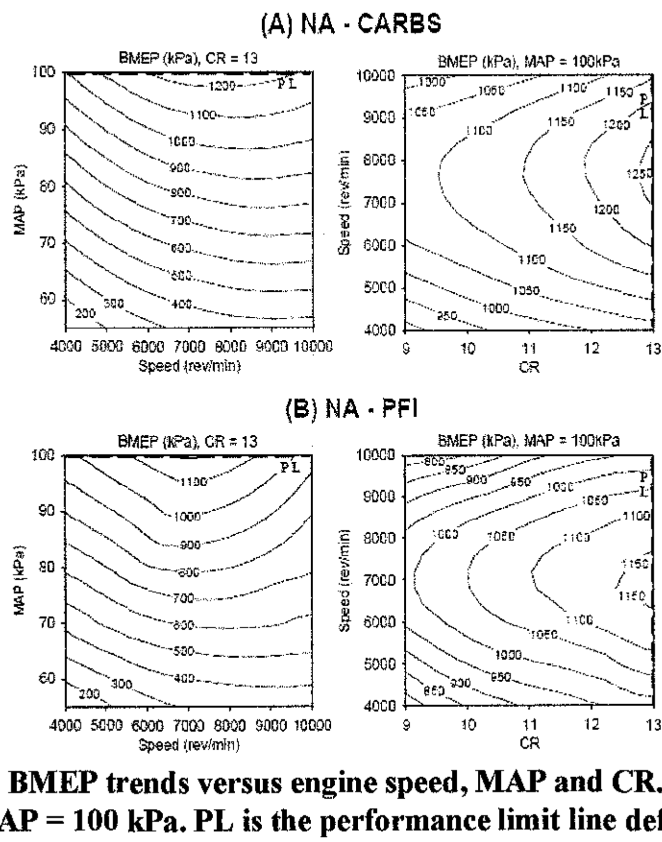


Figure 2.3: BMEP trends versus engine speed, MAP and CR. (Left): CR = 13, (Right): MAP = 100 kPa. PL is the performance limit line defined at WOT for a given CR.

PERFORMANCE, EFFICIENCY AND EMISSION COMPARISONS

Figure 12 displays CR and mode comparisons at WOT (100 kPa MAP) to evaluate the performance, efficiency and emission effects. Only three data sets are presented in graphs across the test range. However, results provide the reader with clear findings from the experiments presented in this paper. The performance comparisons (Figure 12-Upper) for equal CRs suggest similar BMEP potential between PFI and carburetion when accounting for the airflow discrepancies, which could be eliminated with intake system optimization [4,5,17]. The reduced air consumption of the PFI system is associated with fitting the mandatory Formula SAE intake restriction. Although engine air consumption rates were not high enough to cause restrictor choking, the intermittent pulsing attributed to the uneven firing order limited instantaneous peak airflow through the nozzle, resulting in reduced performance.

The fuel efficiency comparisons (Figure 12-Middle) indicate clear gains when implementing PFI over carburetion. The improved fuel break-up and atomization of the PFI system aids in vaporization, resulting in improved mixing together with minimal wall wetting and pooling. These effects improve efficiency and magnify in affect at low engine speed and load conditions. It is also reiterated that these results are attained from steady state testing, with transient testing yielding significantly different results due to fuel control and pooling issues associated with the high fuel entry position of carburetion. Hence, efficient TWC operation for emission control is difficult to achieve with this system. When comparing emissions between carburetion and PFI (Figure 12-Lower), HC emissions standout as being the major difference between the two modes of fuel delivery. Reductions in the order of 20% are shown at WOT when implementing PFI during steady state testing, with HC differences between both modes expected to increase during transient operation. The differing fuel efficiencies between carburetion and PFI are shown to have insignificant affects on combustion, consistent with the similar performance potential findings discussed. However, the poorer ηTH of the carbureted system has resulted in increased levels of engine out unburnt HC emissions. These performance, efficiency and emission results confirm previous findings from larger bore engines, with PFI generally adopted over carburetion due to improved fuel and emission control.

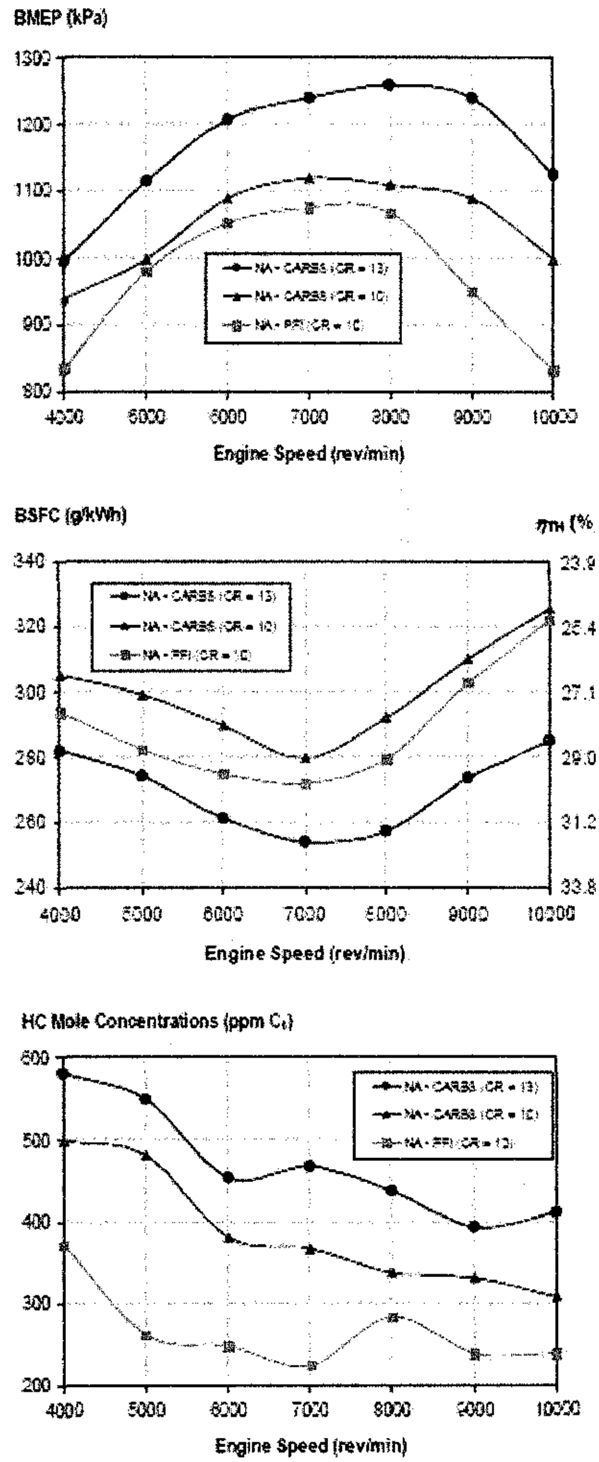


Figure 12: Performance, efficiency and emission effects for varying CRs and fuel delivery modes. WOT (100 kPa MAP). λ = 0.9 ± 0.02, MBT spark timing. (Top): BMEP. (Middle): BSFC and η_{TH}. (Lower): Engine out HC mole concentrations.

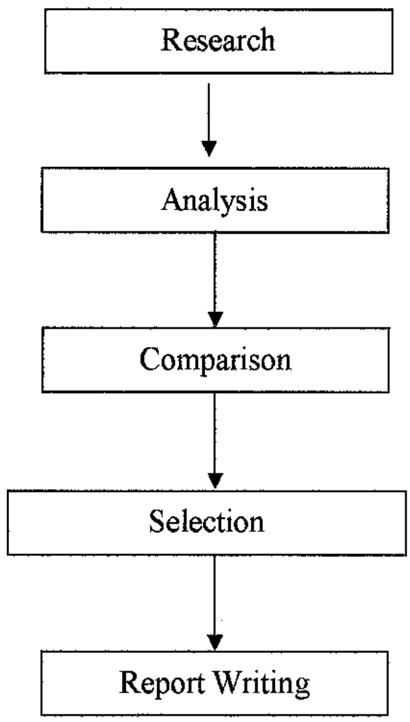
Conclusion

This paper compares the performance, efficiency, emissions and combustion parameters of a prototype two cylinder 430 cm3 SI engine. Experiments were completed over a range of CRs ranging from 9-13 for both carburetion and PFI fuel delivery systems. Results showed the potential for engine operation at a CR exceeding 13. The high CR achieved for this particular small engine is attributed to the physical size reduction, particularly the reduced bore diameter and fast burn combustion chamber. This resulted in engine speed increases together with end-gas volume reductions around the periphery of the chamber, allowing CR and/or MAP values to be increased before exceeding the DL. This enabled the engine to achieve 37% ηTH and 13 bar BMEP. When altering the CR, experimental results showed similar order effects on performance, efficiency and emissions when compared to larger bore engines. The test engine's BMEP, ηTH and O2 benefits were also found to have the potential to exceed typical larger bore engines found in passenger vehicles. However, this was only possible after CR optimization, which compensated for the higher levels of dissociation, friction and heat losses associated with the smaller cylinder capacity. When comparing carburetion to PFI, results show equal performance potential between both modes of fuel delivery. However, a 3% relative improvement in peak ηTH was observed with PFI due to the improved mixture homogeneity, as confirmed by the HC emissions. This improvement in ηTH increased with decreasing engine speed and load. In addition, reductions in PFI CO2 emissions showed similar trends to ηTH findings. However, HC emissions were shown to be the most significant difference between both modes of fuel delivery, with a reduction in the order of 20% at WOT when implementing PFI. These performance, efficiency and emission results confirm previous findings and are factors contributing to PFI's universal adoption over carburetion. Combustion results indicate that the fuel delivery system has little effect on burn rates. However, CR increases result in faster initial burn rates, which produce higher IMEP. Additionally, increases in engine speed do not linearly correlate with combustion duration, with increasing turbulence levels promoting faster flame speeds and hence faster burning rates. Peak flame speeds exceeding 40 m/s were recorded at 10500 rev/min. The future development of smaller engines for passenger vehicles is addressed in this paper, as engine downsizing grows in popularity due to rising oil prices and recent greenhouse concerns. The results presented have significant relevance to manufacturers who continue to strive for performance and efficiency benefits while meeting legislative pollutant emission standards.

[Reference Number 12]

CHAPTER 3
METHODOLOGY

3.1 Project Methodology



The first step of this project is to choose a title. The title of this project was proposed by Ir. Dr. Masri Baharom which is the selection and analysis of engine of Formula Sae. Once the title has been approved literature review is done. With the information gathered from the literature review the analysis and selection of the engine can be done. This will then lead to the report writing.

3.2 Gantt Chart

Please refer to appendix A.

CHAPTER 4
RESULT AND DISCUSSION

4.1 General comparison between the three engines.

The Honda and Yamaha engines weight about 50 kg while the Aprilia engine only weights about 33 kg. This is mainly due to the smaller displacement of the Aprilia engine. The Aprilia is supposed to have the better power to weight ratio.

The Honda and Yamaha engines uses a wet sump while the Aprilia engine uses a dry sump. A wet sump is not recommended for racing purposes. This is because when the vehicle is turning around sharp corners, this causes the oil in the pan to slosh. This will then starve the system of oil for small periods of time. This can cause damage to the engine. A dry sump has its advantages, namely increased oil capacity and a lower center of gravity for the engine. Also, dry sump designs are not susceptible to the oil starvation problems if the oil sloshes in the oil pan.

The aprilia's has a lot of issues with the starting system. In order for the italians to meet their weight goal for the motor, they used a starter off a 50cc scooter. Because that starter lack huevous, they out fitted the engine with a decompressor system that opens the exhaust valve slightly to bleed off some of the compression built up by the cylinder. Aprilia has made a lot of revisions to the starter gearing and the ecu, even putting a 6 second limit on cranking to keep the motor from burning up. The sprags are pretty beefy but get damaged when the engine backfires, which happens with a poor tune or bad starter.

4.2 FSAE noise test engine speed.

To calculate the noise test engine speed we first have to find the piston speed for each of the engine. To calculate the piston speed the following formula will be used:

$U_P = 2Ns$

Where, N = engine speed

S = stoke

For the Honda CBR F4i,

N = 12,500 rpm

s = 0.04255 m

$U_P = 2Ns$

$= 2\ (12{,}500)\ 0.04255$

$= 1{,}063.75$ m/min

For the Yamaha FZ6 S2,

N = 13,000 rpm

s = 0.0445 m

$U_P = 2Ns$

$= 2\ (13{,}000)\ 0.0445$

$= 1{,}157$ m/min

For the Aprilia SXV 550,

N = 11,000 rpm

s = 0.055 m

$U_P = 2Ns$

$= 2\ (11{,}000)\ 0.055$

$= 1{,}210$ m/min

Of the Formula SAE Rules, the noise test speed for a given engine will be the engine speed that corresponds to an average piston speed of 914.4 m/min (3,000 ft/min) for automotive or motorcycle engines, and 731.5 m/min (2,400 ft/min) for industrial engines, rounded to the nearest 500 rpm.

For automotive or motorcycle engines

The equation is: Calculated Test Speed = $\dfrac{914.4 \times 1000}{2 \times \text{Stroke (mm)}}$ rpm

Table 4.1: FSAE noise test engine speed

Model	Bore x Stroke	Displ.	Test rpm	
			Calculated	Rounded
Honda CBR F4i	67.0 x 42.5 mm	599 cc	10 758	11 000
Yamaha Z6 S2	65.5 x 44.5 mm	600 cc	10 274	10 500
Aprilia SXV 550	80 x 55 mm	549 cc	8 312	8 500

The sound of the engine is important. If it is realistic enough this sound will help you determine when it is time to change gears. Since the frequency and volume of the sound should vary continuously it is almost impossible to use recorded sound samples. The sound has to be generated online. The signal that will be sent to the sound card should be some sort of wave form. The basic frequency of this wave should be the same as the engine speed. To realize this, the engine speed is integrated and sent through a sine function. This generates a sine wave having amplitude one and the same frequency as the engine. Since engine noise consists of more than one frequency also a signal of double, half and quarter frequency is made. These signals are added up using different gains. Real engine sound is not made up of just sine functions. To make up for this, the sound is made noisier using clipping of the signal. This makes the sound more realistic. In figure 4.2 the implementation of this can be seen. The volume of the sound should depend on the position of the accelerator. This is done in a simple linear way. When the accelerator is completely floored, the sound is loudest. When the accelerator is not pushed at all, the volume of the sound is set to 30% of the maximum value.

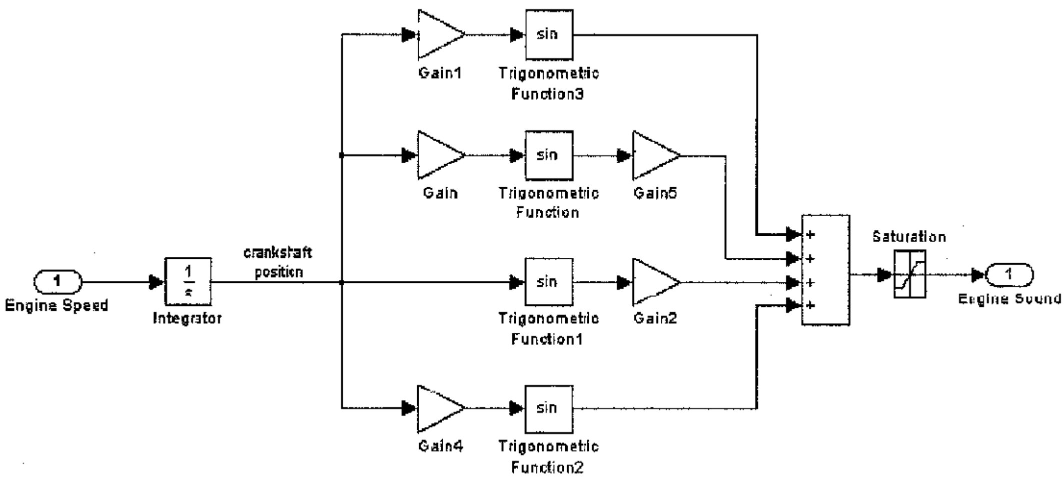


Figure 4.1: Implementation of the Sound Generator

4.3 Engine Dyno Charts

4.3.1 Honda CBR 600 F4i

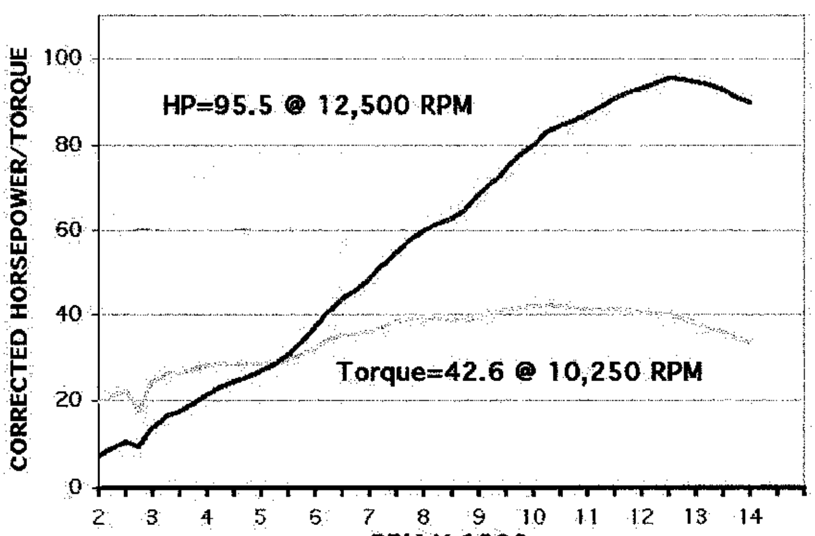


Figure 4.2: Honda CBR 600 F4 i Dyno Chart

4.3.2 Yamaha FZ6 S2

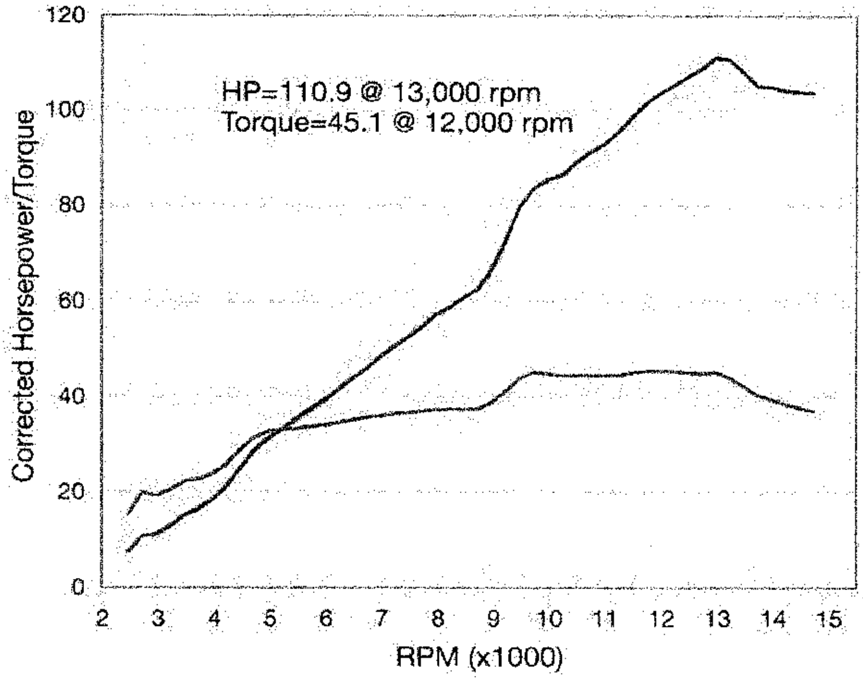


Figure 4.3: Yamaha FZ6 S2 Dyno Chart

4.3.3 Aprilia SXV 550

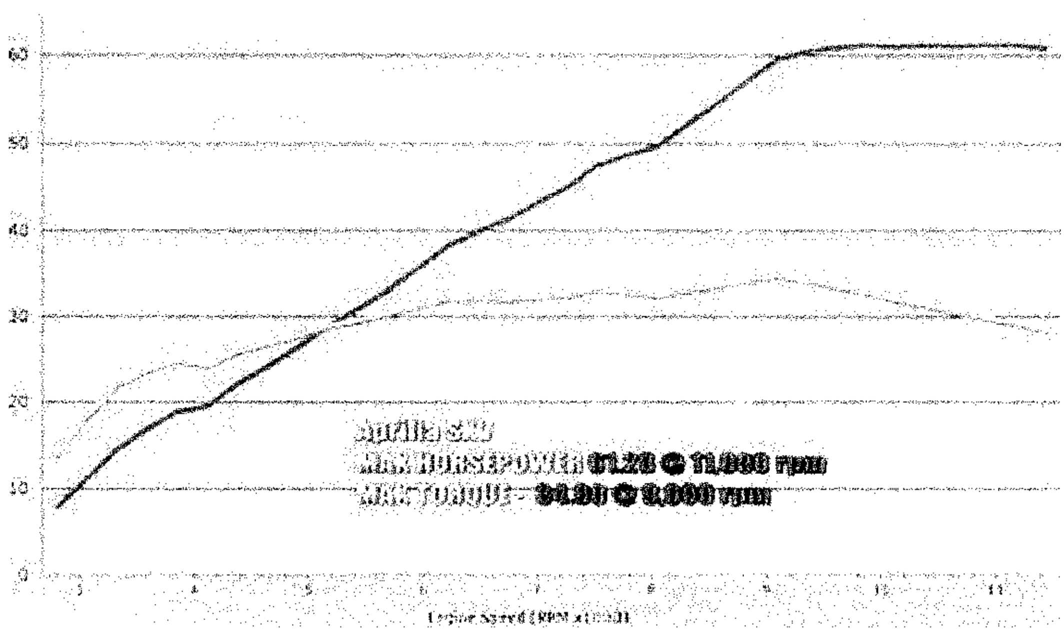


Figure 4.4: Aprilia SXV 550 Dyno Chart

4.4 Calculating Brake Mean Effective Pressure (bmep)

4.4.4 bmep for Honda CBR 600 F4i

Based on torque, the bmep is,

$$\text{bmep} = \frac{4\pi\tau}{V_d}$$

Where, torque, $\tau = 42.6$

$V_d = 599$

$$\text{Thus, bmep} = \frac{4\pi\,(42.6)}{599}$$

$$= 0.89 \text{ kPa}$$

Based on power, the bmep is,

$$\text{bmep} = \frac{2W_b}{V_d N}$$

Where, $W_b = 2\,\pi\,\tau N$

$= 2\,\pi\,(42.6)\;12{,}500$

$= 3{,}345{,}796$

$$\text{Thus, bmep} = \frac{2\,(3{,}345{,}796)}{599\,(12{,}500)}$$

$$= 0.89 \text{ kPa}$$

4.4.5 bmep for Yamaha FZ6 S2

Based on torque, the bmep is,

$$\text{bmep} = \frac{4\pi\tau}{V_d}$$

Where, torque, $\tau = 45.1$

$V_d = 600$

$$\text{Thus, bmep} = \frac{4\pi\,(45.1)}{600}$$

$$= 0.94 \text{ kPa}$$

Based on power, the bmep is,

$$\text{bmep} = \frac{2W_b}{V_d N}$$

Where, $W_b = 2\ \pi\ \tau N$

$= 2\ \pi\ (45.1)\ 13{,}000$

$= 3{,}683{,}831$

Thus, $\text{bmep} = \frac{2\ (3{,}683{,}831)}{600\ (13{,}000)}$

$= 0.94$ kPa

4.4.6 bmep for Aprilia SXV 550

Based on torque, the bmep is,

$$\text{bmep} = \frac{4\pi\tau}{V_d}$$

Where, torque, $\tau = 34.8$

$V_d = 549$

Thus, $\text{bmep} = \frac{4\pi\ (34.8)}{549}$

$= 0.79$ kPa

Based on power, the bmep is,

$$\text{bmep} = \frac{2W_b}{V_d N}$$

Where, $W_b = 2\ \pi\ \tau N$

$= 2\ \pi\ (34.8)\ 11{,}000$

$= 2{,}405{,}203$

Thus, $\text{bmep} = \frac{2\ (2{,}405{,}203)}{549\ (11{,}000)}$

$= 0.79$ kPa

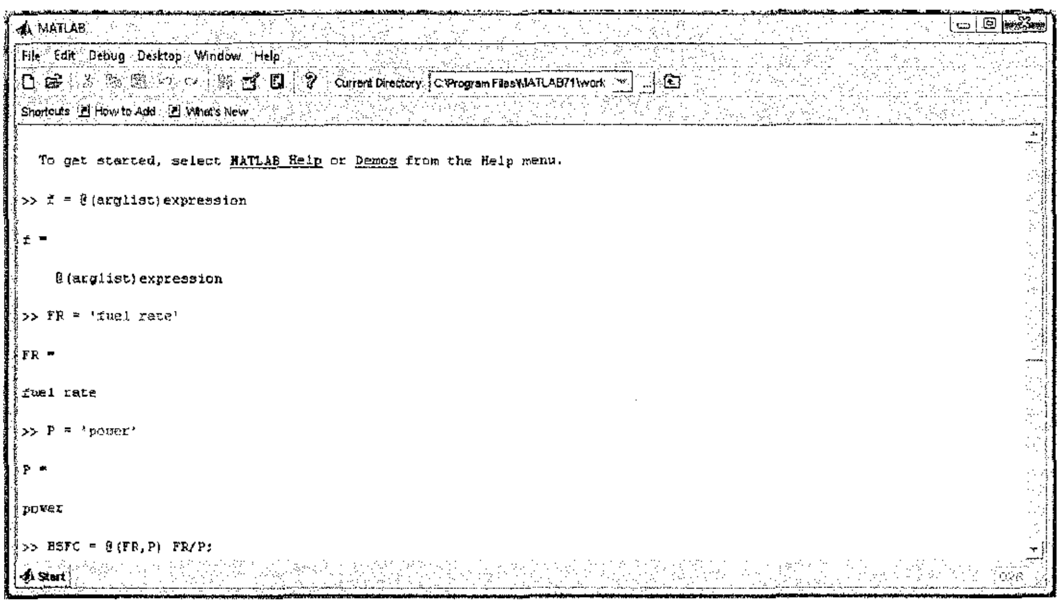
4.5 Calculating Brake Specific Fuel Consumption (BSFC)

The brake specific fuel consumption is calculated using MATLAB. The following data is needed for the calculation.

Honda
Fuel rate = 12,600 cc/hr
Power = 95.5 hp

Yamaha
Fuel rate = 13800 cc/hr
Power = 110.9 hp

Aprilia
Fuel rate = 10,800 cc/hr
Power = 61.28 hp



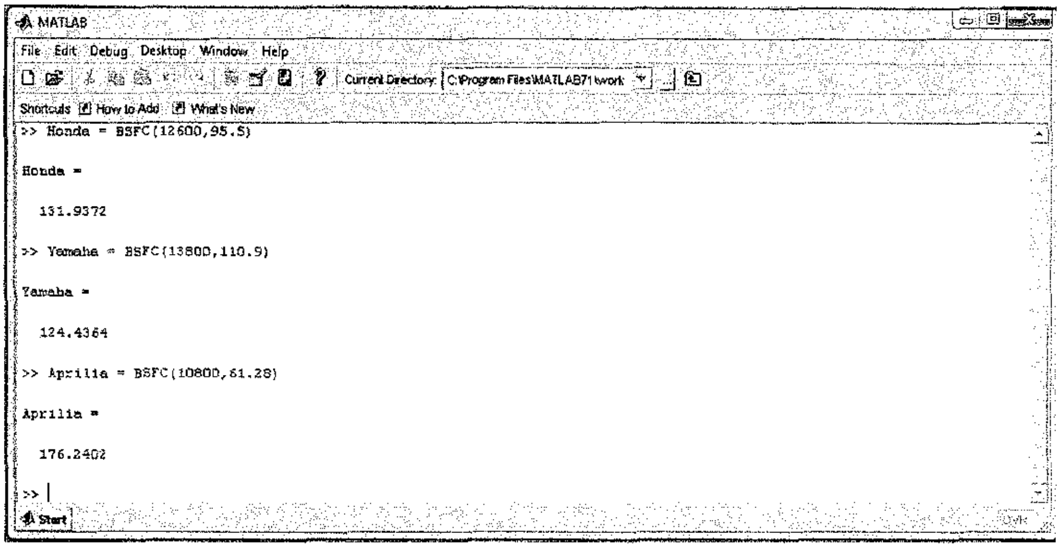


Figure 4.5: Brake Specific Fuel Consumption (BSFC)

Thus from the BSFC for the engines are:

Table 4.2: BSFC for the three engines

Engine	BSFC (cc/HP*Hr)
Honda CBR 600 F4i	131.937
Yamaha FZ6 S2	124.436
Aprilia SXV 550	176.240

CHAPTER 5
CONCLUSION

An analysis of three different engines has been done. From the dyno charts, we are able to see that the Yamaha engine tops the Aprilia and Honda engines in terms of power. While the Honda has produces more power compared to the Aprilia. But, the difference in power between the Honda and Yamaha is not too great. Also these 2 engines weight about 50 kg while the Aprilia only weights around 33 kg. Hence the Aprilia has the better power to weight ratio.

For FSAE purposes the Honda CBR 600 F4i will be the engine that is recommended. It has decent power aspects. There are also a variety of tuning kits for this engine. This is because there are a lot of these engines around thus making it quite common. Also a lot of other FSAE teams use this engine. Thus, if something were to go wrong during the competition it would not be too difficult to find spare parts. Worst case scenario, we could purchase a spare engine from another team.

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APPENDIX A

Table: FYP 1 Gantt Chart

No.	Detail / Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Selection of Project Title														
2	Research Work														
3	Project Work														
4	Submission of Preliminary Report														
5	Selection and Analysis														
6	Submission of Progress Report														
7	Seminar														
8	Submission of Interim Report														
9	Oral Presentation														

Table: FYP 2 Gantt Chart

No.	Detail / Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Project Work Continues														
2	Submission of Progress Report 1														
3	Project Work Continues														
4	Submission of Progress Report 2														
5	Seminar														
6	Project Work Continues														
7	Poster Exhibition														
8	Submission of Dissertation Final Draft														
9	Oral Presentation														
10	Submission of Dissertation (Hard Bound)														