University of Southern Queensland

Faculty of Engineering and Surveying

Selection of an Engine and Design of the Fuelling System for a Formula SAE car

A dissertation submitted by

Travis William Mauger

In fulfilment of the requirements of

Courses ENG411 and 4112 Research Project

Towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: October, 2004

Abstract

This dissertation documents the selection of the engine for Formula SAE car. This dissertation also documents the design and testing of the intake manifold, exhaust system, carburettor and the intake restrictor for the USQ Motorsport Formula SAE engine.

Before selecting the engine thorough research into all types of engines and designs was carried out. Once the type of engine that was suitable for the Formula SAE competition was determined, all of the parameters that impacted on the selection of the engine were analysed. To accurately predict which engine was the ‘optimum engine’ a model of the Formula SAE car’s acceleration performance was created and the calculations were undertaken using Matlab.

The engine that was purchased for the Formula SAE car was sourced from a 600cc water-cooled motorcycle. Once the engine had been purchased it was possible to design the fuelling system for the Formula SAE car. In this project the fuelling system incorporated the method of aspiration, fuel mixture preparation system, the intake manifold, the intake restrictor and the exhaust system.

A feasibility study that encompassed forced air induction systems for the Formula SAE car was carried out and the utilisation of multi-point fuel injection was also examined. However due to budgetary restraints neither of these systems were feasible. Therefore it was decided that the engine would be naturally aspirated and carburetted.

The merits of fixed venturi carburettors and constant velocity carburettors were explored in order to select the most suitable type of carburettor for the Formula SAE engine.

Research was carried out in order to find the intake manifold configuration that best suited the Formula SAE car. The USQ workshop was also liased with during the initial design period in order to ensure that the machining capabilities at the USQ

could produce the required design. In order to determine if the final design of the intake manifold was feasible a prototype manifold was constructed.

In order to design the restrictor various standards that are used to design flow measurement devices were incorporated. Several prototype restrictors were constructed and tested using an airflow bench.

The design of the exhaust system was also investigated. It was found that the best solution in regards to the exhaust system was to retain the original exhaust manifold and purchase an aftermarket muffler.

As the project developed it became clear that cooling requirements of the engine were a concern. For this reason experimental procedures were devised to determine if the original motorcycle radiator would be sufficient.

Because all of the components that were designed or specified in this project are part of a system they were tested together, on the engine. Unfortunately it was not possible to obtain any meaningful test results. The reason for this was that the rest of the car was not at a stage of completion that would allow testing on dynamometer. For this reason, at the time of writing, it is uncertain whether the systems that were designed in this project matches the design criteria, which are outlined in chapter 2.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG4111 & ENG4112 *Research Project*

Limitations of Use

Certification

I certify that the ideas, designs, experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

TRAVIS WILLIAM MAUGER

STUDENT NUMBER: Q98238963

______________________________ **SIGNATURE**

______________________________ **DATE**

Acknowledgements

I would like to thank my supervisor, Selvan Pather, for all of his help throughout the course of this project.

I would like to thank Peter Penfold for his technical guidance.

I would like to thank Chris and Brian from the mechanical workshop for their construction expertise and patience.

I would like to thank Terry Seng from Paramount Performance for volunteering the use of his dynamometer and time.

I would like to thank Trevor Wolfenden from Air Direction Heads for his time and the use of his airflow bench.

I would like to thank my family and friends for their support throughout my degree and this project.

I would like to give a very special thankyou to Camellia, my fiancée, for her support and understanding throughout my project and my degree.

Table of Contents

List of Figures

List of Tables

1 Project Introduction

1.1 Introduction

This chapter will introduce the project. A brief description of the background of this project will be given and the team that is involved in the project will be introduced. The basic aims of the project will be outlined and the reasoning for the decisions to pursue these aims will also be covered. Once the basic aims have been given the specific objectives of this project will be outlined. This chapter will then conclude with an overview of this dissertation.

1.2 Background

This project is one of a group of projects that are aimed at the design and construction of a car that will compete in the Formula SAE-A competition. Formula SAE-A is an opportunity for students to apply a broad range of skills, from management to detailed design, to a 'real' engineering project. The aim of the competition is to design, construct and race an open wheeler formula style car against other universities. Figure 1.1 represents a typical Formula SAE car. In 2003 a total of twenty-four teams, including six international teams, competed in the event. In the true spirit of all engineering projects the construction of the car is restricted by budget, time and design restraints and requires team effort as well as individual endeavour.

(*Source*: University of Queensland Formula SAE, 2004)

Figure 1.1 University of Queensland's Formula SAE car.

In March 2004 a racing team was formed to enable the development of the car. The team was aptly named USQ Motorsport. The team primarily consists of fourth year mechanical engineering project students with their respective supervisors acting in an advisory capacity. Each student's project relates directly to a facet of the design and manufacture of the racing car. The topic allocations are as follows:

1. Rex Parameter – Suspension design;
2. Leslie Rayner – Steering design;
3. Jeremy Little – Power transmission;
4. John Armstrong – Project management;
5. Brad Moody – Human factors and control systems design;
6. Chris Baker – Space-frame chassis design;
7. Bruce Grassick – Monocoque chassis design;
8. Ken Nelder – Body design; and
9. Travis Mauger (the author) - Engine design.

The goal of this year's USQ Motorsport team was to design and construct a vehicle that is ready for competition by the end of August 2004. The ultimate goal of the team is to compete in the 2004 Formula SAE-A competition at Melbourne in December. However, due to financial and time constraints the former goal was not achieved. Nevertheless, it is anticipated that the car will compete in the competition in December.

1.3 Project Aim

The aim of this project is to select a suitable engine and design a fuelling system for USQ Motorsport's Formula SAE-A racing car. In this project the term 'fuelling system' will refer to the engine's induction system, extraction system and method of aspiration. An informed engine selection is obviously the most important part of racing car power development. Hence, engine selection was given first priority in this project. The remaining elements that were designed in this project represent the highest priority components in respect to developing an engine that conforms to Formula SAE competition rules.

An intake restrictor is incorporated into the Formula SAE rules. The restrictor is a *single* 20mm orifice that must be placed between the throttle and the engine. All of the air (or air and fuel) must pass through this restrictor (Formula SAE Rules, 2004). The diameter of the restrictor is approximately half the diameter of the orifice that the air (or air and fuel) usually passes through on standard engines of the same capacity as that used in the competition.

Essentially the restrictor limits the volumetric efficiency of the engine. The loss of volumetric efficiency induced by the restrictor causes the engine to loose a substantial amount of power and torque. Fortunately, the rules only state that the restrictor must be circular. For this reason the optimisation of the restrictor's shape is an important design task in Formula SAE engine development.

In 2003 the author carried out a preliminary study, which analysed the design implications that the restrictor brought about. In order for the engine to run properly, it was found that the fuel mixture and induction system would require modification or redesign. These issues were addressed in this project.

The exhaust system and method of aspiration was included in this project as the design of the exhaust system and method of aspiration impacts heavily on the performance of the engine.

1.4 Project Objectives

The objectives of this project were as follows:

1. Review engine types and designs;
2. Specify an engine for the Formula SAE car;
3. Conduct feasibility study encompassing the types of fuel delivery systems (i.e. naturally aspirated, turbocharged or supercharged);
4. Design a restrictor and manifold;
5. Design the exhaust system;
6. Select a mixture preparation system;

And if time permitted:

7. Construct an intake manifold and restrictor;
8. Test the restrictor, mixture preparation system and intake manifold;
9. Construct exhaust and test;

However as the project progressed, it was found that the cooling system is another significant aspect of designing an engine for the Formula SAE competition. In a Formula SAE car, the location of the engine's cooling system varies from the manufacturer's original design. The effect of this is that the cooling system experiences different airflow conditions and therefore the original system may not be sufficient. For this reason it was important that an analysis of the cooling system was carried out.

Therefore the following objectives were added to this project:

10. Select a location for the radiator;
11. Outline some design and testing methods in regards to the engine cooling system;

1.5 Dissertation Overview

The following is a summary of how this dissertation will be presented:

Chapter 2 will provide a background of the Formula SAE-A competition. In this chapter the design requirements of this project will also be outlined. The most significant competition rules will also be discussed.

Chapter 3 will outline the methodology that was used to undertake this project.

Chapter 4 will deal with the selection of the engine. Initially a general discussion of internal combustion engines will be presented. Comparisons will be drawn between different types of IC engines and related back to the restrictions enforced by the competition rules. This chapter will also analyse the effects of engine selection on vehicle performance and specify the optimum engine for the Formula SAE car.

Chapter 5 is devoted to the consequential effects of this project. It will include a safety analysis and a discussion of ethical and sustainability issues.

Chapter 6 will begin with a discussion of the feasibility of forced air induction for the USQ Motorsport Formula SAE car. Carburation and electronic fuel injection will also be compared and analysed and a suitable fuel mixture preparation devise will be specified. This chapter will then document the analysis of different types of intake manifolds and restrictors. The design process of the manifold and restrictor will then be covered. In addition this chapter will outline the results of the tests that were conducted in regards to the intake manifold, restrictor and fuel preparation system. The options that were available regarding the exhaust system will also be discussed in this chapter.

Chapter 7 is dedicated to the design of the cooling system for the car engine.

Chapter 8 will evaluate the designs and selections that evolved against the guidelines that were set out in chapter 2.

Chapter 9 will give a summary of the achievements of the work conducted in this project and set out the areas of this project that require further work. This section will also recommend some areas that future USQ Motorsport members may wish to pursue.

1.6 Conclusion

This chapter has introduced the project, its aims and objectives and the reasoning for pursuing the objectives that have been outlined. The following chapter will outline the specific objectives of the competition and examine the methods and designs that other teams have implemented in order to reach these objectives. The specific design objectives of this project will also be introduced.

2 Background and Design Requirements

2.1 Introduction

This chapter will outline the objectives of the Formula SAE competition. As engine systems are the focus of this project, the design requirements and objectives that must be met by the engine will be discussed. Accordingly, the most relevant rules will also be summarised. Due to the significant volume of rules, a copy of the relevant rules can be found in Appendix B. The methods and designs that other teams have used to meet the criteria set out by the competition, in regards to the engine systems, will also be discussed.

2.2 Vehicle Design Objectives

The design objectives of the Formula SAE-A competition are best stated using the guidelines set out by the Formula SAE-A rules (2004, p7), which state:

> For the purpose of this competition, the students are to assume that a manufacturing firm has engaged them to produce a prototype car for evaluation as a production item. The intended sales market is the nonprofessional weekend autocross racer. Therefore, the car must have very high performance in terms of its acceleration, braking, and handling qualities. The car must be low in cost, easy to maintain, and reliable. In addition, the car's marketability is enhanced by other factors such as aesthetics, comfort and use of common parts. The manufacturing firm is planning to produce four (4) cars per day for a limited production run and the prototype vehicle should actually cost below $25,000. The challenge to the design team is to design and fabricate a

prototype car that best meets these goals and intents. Each design will be compared and judged with other competing designs to determine the best overall car.

2.3 Judging

The competition is held on the first weekend in December every year. Before competing, the car is scrutinised by an international panel of judges. The team is not only judged on the cars performance in the racing events but also on the cost, manufacturability and design of the car. The team is also required to perform a presentation to a board. The concept of the presentation event is to evaluate the team's ability to make a presentation to the executives of a manufacturing firm. The presentation should convince the executives of the superiority of the team's design. In order for the car to compete it must satisfy stringent safety regulations and conform to the design rules. If the car satisfies the competition requirements the team then competes in a range of events including the endurance and fuel economy, autocross, acceleration and skid pad events. The allocation of points for each event is presented in table 2.1:

Static Events	
Presentation	75
Engineering Design	150
Cost Analysis	100
Dynamic Events	
Acceleration	75
Skid Pad	50
Autocross	150
Fuel Economy	50
Endurance Event	350
Total	1000

(*Source*: Formula SAE Rules, 2004, p23)

Table 2.1 Event point allocations.

2.4 Engine Design Objectives

From table 2.1, section 2.2 and section 2.3 the design criteria of the engine can be outlined. The engine design objectives are summarised in the following list:

1. **Acceleration performance** – The engine must display performance characteristics that allow the car to accelerate from a wide range of engine speeds to be competitive in the acceleration, endurance, skid pad and autocross events;
2. **Reliability** – Clearly the car must be capable of competing in all of the events without engine failure;
3. **Reproducible** – In order to be competitive in the manufacturing category the engine must be able to be reproduced easily;
4. **Fuel-efficient** - The car must maintain reasonable fuel efficiency to compete in and finish the endurance event;
5. **Cost effective** – The engine must also conform to the budget requirements of the university and the competition.

2.5 Engine Design Constraints

To obtain a better understanding of the design constraints some of the most important Formula SAE-A rules need to be discussed. The most important rules in regards to engine design include the capacity limit, operating cycle and the type of fuel that is allowed. The engine must not exceed a capacity of 610cc and must utilise a four-stroke operating cycle. The engine must also run on unleaded gasoline, which means that the engine must utilise spark ignition. The introduction of performance boosting agents into the fuel, such as Nitrous Oxide, is also prohibited as no additives are allowed in the fuel (Formula SAE Rules, 2004 pp34-35). Although the rules are strict, they still allow a great degree of freedom in terms of engine design.

2.6 The Background of Formula SAE Engine Design

2.6.1 Four Cylinder Engines in Formula SAE

In the 2003 Formula SAE-A competition the University of Melbourne and RMIT were the only teams that competed and did not utilise a four-cylinder 600cc motorcycle engine. The University of Melbourne chose to design their own engine and RMIT used a single cylinder engine in their car (Formula SAE-A, 2004). The engines that were used by all of the other teams were manufactured by Japanese motorcycle companies. These companies are considered the ‘big four’ of the motorcycle industry and include Kawasaki, Honda, Suzuki and Yamaha (Walker M. 2001, et. al.). The engines that were used in the 2003 competition were predominantly sourced from the supersport motorcycle class (Formula SAE, 2004). The supersport class consists of the Honda CBR600, Suzuki GSX-R600, Kawasaki ZX-6R and the Yamaha YZF-R6, which is depicted in figure 2.1. The motorcycles that makeup the supersport class are considered the highest performing street registered 600cc motorcycles in the world (Walker M. 2001, p35). Therefore, these engines are an obvious first choice for any Formula SAE team.

(*Source*: Yamaha Motor Company, 2003)

Figure 2.1 The Yamaha YZF-R6 and its engine.

Table 2.2 and 2.3 demonstrate that the supersport engines are all very similar in design and performance. As can be expected, the power output of these engines increase every model year, as each manufacturer tries to get the edge on the competition. Although table 2.3 displays only the maximum power and torque values it should be noted that the shape of the torque and power curves of the supersport engines are almost identical (Appendix D contains a complete compilation of all of the engines that are discussed in section 2.6.1 and section 2.6.2).

Manufacturer	Suzuki	Honda	Yamaha	Kawasaki
Model	GSXR600	CBR600RR	YZF600-R6	ZX-6-R
Operating Cycle	4 -stroke	4 -stroke	4 -stroke	4 -stroke
Capacity (cc)	599	599	600	626
Bore x Stroke (mm)	67 x 42.5	67 x 42.5	65.5x 44.5	68 x 43.8
Compression Ratio	12.2: 1	12.0: 1	12.0: 1	12.8: 1
Cooling System	Liquid	Liquid	Liquid	Liquid
No. of Cylinders	In-line 4	In-line 4	In-line 4	In-line 4
Camshafts	DOHC	DOHC	DOHC	DOHC
Number of Valves	16	16	16	16

(*Source*: BikePoint, 2003)

Table 2.2 Supersport engine specifications for the 2003 model year.

Model	Horsepower (corrrected)	Torque (ft-lb)	Top Speed (mph)	Quarter Mile	Roll-ons, 60-80mph
Yamaha R6	105.5 @ 12,750rpm	44.7 @11,750rpm	158.0	10.80sec @ 127.8mph	4.46 sec
Honda CBR600	107.2 @ 13,500rpm	45.4 @11,000rpm	162.2	10.73sec @ 129.7mph	4.23 sec
Kawasaki ZX-6R	107.5 @ 13,000rpm	46.4 @11,000rpm	158.5	10.67sec @ 131.0mph	4.34 sec
Suzuki GSXR600	103.4 @ 13,250rpm	46.5 @10,750rpm	158.8	10.87sec @ 26.0mph	5.36 sec

(*Source*: BikePoint, 2003)

Table 2.3 Supersport performance data for the 2003 model year.

A new supersport motorcycle sells for approximately $16,000 (Yamaha, 2004 et. al.). Second-hand supersport engines can be obtained from $2,500 to $5,000, which makes them well within the budget constraints of a typical Formula SAE team (Independent Wreckers, 2003, pers. comm. 12 Oct.). Therefore, the engines used by most of the teams are generally second-hand and sourced from smashed motorcycles (Formula SAE-A, 2004 et. al.). Due to the size of the air intake duct all of these engines require a restrictor. Numerous teams claim that the restricted four-cylinder engines are capable of producing around 45kW with natural aspiration (University of Queensland Formula SAE, 2004 et. al.).

2.6.2 Single Cylinder Engines in Formula SAE

In the 2003 competition only one of the teams used a single cylinder engine. However, the concept of using a single cylinder engine in a Formula SAE car was not new. Many American and European universities have implemented single cylinder engines in their designs in the past (RMIT, 2003). The engine that was used by the Royal Melbourne Institute of Technology was sourced from an enduro motorcycle, which is a class of dirt bike. The engine that the team used was sourced from a Yamaha WR450-F (RMIT, 2003), which is depicted in figure 2.2. The main competitors in the enduro class are the Honda CRF450 and the KTM 525EXC. The engines used in enduro motorcycles also display very similar performance characteristics and design (Motorcycle News, 2004).

(*Source*: Yamaha, 2003)

Figure 2.2 The Yamaha WR450F.

The main advantage that the RMIT team cited for their engine selection was a 30kg weight saving over the four-cylinder supersport engines. RMIT (2003) also explained that the small physical size of the engine would free up a large amount of room at the back of the car. RMIT (2003) also indicated that the smaller engine would allow improved weight distribution and better component packaging.

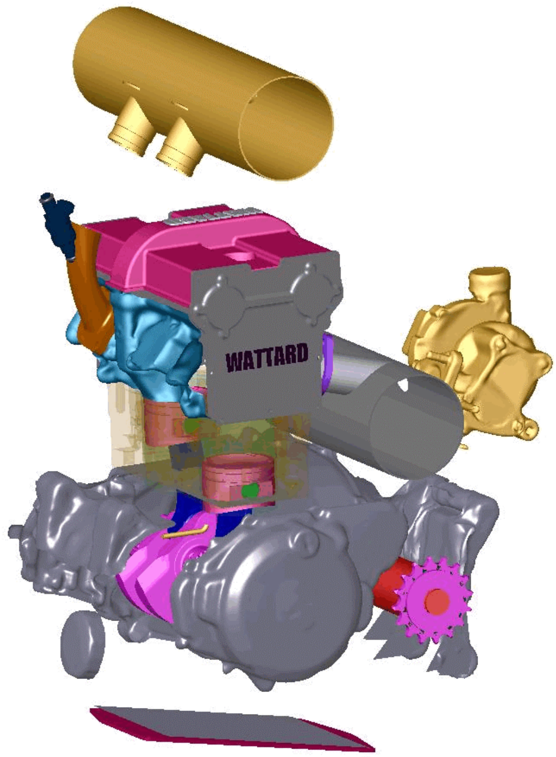
Additionally the team considered the torque characteristics of the engine more suitable for the tight tracks characterised by the Formula SAE competition (RMIT, 2003). The torque curve of the Yamaha WR450 is much 'flatter' than the supersport engines' torque curve. This 'flatness' of the torque curve corresponds to a more consistent amount of available acceleration across the engine's speed range. However the absolute value of the torque produced by the enduro engine is considerably less than the torque produced by the supersport engine. This may explain why the team placed 8th overall in the 2003 Formula SAE-A competition (Formula SAE-A, 2004). The RMIT team noted that the engine weighed 30kg less than the four cylinder engines that the other teams were using. Therefore it would be reasonable to assume that the team assumed that the lighter weight of the single cylinder would make up for its apparent lack of torque. This assumption will be analysed in depth in chapter 4.

2.6.3 'One-off' Engines in Formula SAE

With an understanding of the engine design requirements and restraints it is possible to design an engine from the 'ground up'. This is exactly what the Melbourne University Racing did for the 2003 competition. The engine is known as the WATTARD engine after its designer William Attard. The advantage of this approach is that the engine can be designed to exhibit the exact desired performance characteristics. This view is summarised by Melbourne University Racing (2003):

'It is a smaller two cylinder low friction design, optimised for the needs of a Formula SAE car rather than a motorcycle.'

Another advantage of the WATTARD engine is that the restrictor is not required as the intake manifold was designed to be the same size as the restrictor. The claimed measured output power of the twin-cylinder engine is 52kW (Melbourne University Racing, 2004). These figures rate the engine approximately 10% more powerful than a modified four-cylinder 600cc motorcycle engine. The designer also claims that the engine consumes significantly less fuel. The WATTARD engine is also claimed to be lighter than the four-cylinder engines. The engine is constructed from 7075 Aluminium and high tensile steel and is shown in figure 2.3.



(*Source*: Melbourne University Racing, 2004)

Figure 2.3 The WATTARD engine.

In the development of any 'one-off' engine or prototype there are many disadvantages. The first obvious disadvantage of this approach is that it is very expensive. Another problem that the Melbourne team has encountered is the unavailability of engine components. Because the engine is unique most of the components have to be manufactured by specialists. Even components that can be purchased 'off the shelf' have to be modified to suit the engine (MUR, 2003). This slows down the development of the engine and is very inconvenient. To undertake such a task Melbourne University Racing has received a large amount of industry support (MUR, 2003). Unfortunately, the reliability of prototype engines such as the

WATTARD engine is generally poor and would explain why MUR has consistently experienced engine failure in competition and testing (MUR Motorsport News, 2004).

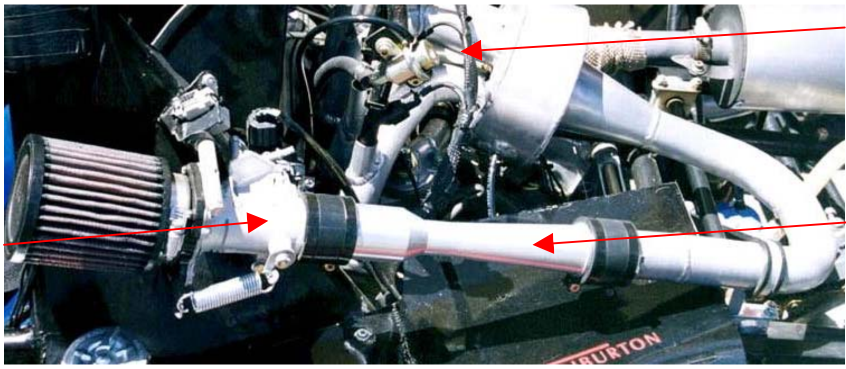
2.7 Superchargers and Turbochargers in Formula SAE

Turbochargers and superchargers are used to boost the output power of engines. Turbochargers and superchargers are allowed in the Formula SAE competition as long as the engine was not originally equipped with one. The rules indicate that the team must design the turbocharging or supercharging system. In this context, 'design' means designing the accessories and modifying the engine in order to accommodate the system. It does not mean designing the actual supercharger or turbocharger unit (Formula SAE-A Rules, 2004, pp35 –42).

In the past Monash Motorsport (2001) has fitted a turbocharger to their Formula SAE car. In the U.S., many of the entrants fit their cars with turbochargers (Monash Motorsport, 2001). U.S. teams, such as the 2001 University of Central Florida F-SAE team (2004) have also used superchargers to boost the performance of the engine and claim a peak power of 50kW from a supercharged CBR600 engine. The power increase that can be expected by using turbochargers and superchargers will be examined in detail in chapter 6.

2.8 Intake Restrictor Design Objectives & Background

The main ambition in the design of the restrictor is to maximise the volume of fluid that can pass through the restrictor. Nearly all of the Formula SAE teams use a diverging-converging nozzle shaped restrictor similar in shape to the restrictor in figure 2.4 (Curtin Motorsport, 2004, et. al.). Many other forms of motor sport, such as Formula 3, require the fitment of similar intake restrictors. Where the rules allow, most of the Formula 3 teams opt to fit a converging-diverging nozzle, with a gentle taper, to restrict the engine (Smith, H. P. 1971). The reason for this is that the diverging-converging nozzle, compared to other shapes of restrictors, allows the maximum volume of fluid to pass through it (Miller, R. 2003, et. al.). Therefore the design of a diverging-converging nozzle shaped restrictor will be pursued in this project.



(*Source*: AutoSpeed, 2004)

Figure 2.4 A typical Formula SAE restrictor.

2.9 Fuel Mixture Preparation System

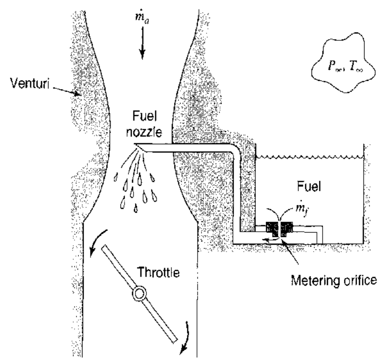
2.9.1 Design Requirements

The design requirements of the fuel mixture preparation system are as follows:

1. Cost effective;
2. Provide the correct fuel/air ratio over all operating conditions;
3. Easily tuned;
4. Easily obtainable;
5. Wide range of parts must be available.

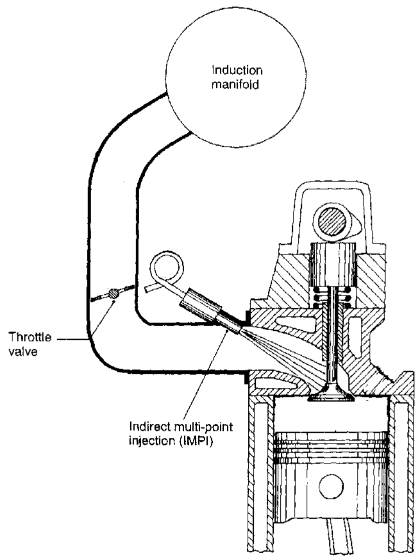
2.9.2 Background

Electronic fuel injection, as opposed to carburation is the most common fuel mixture method preparation used in the F-SAE competition (AutoSpeed, 2004). In general motorcycles have one carburettor per cylinder. As a *single* restrictor has to be placed between the throttle and the engine, the use of multiple carburettors does not facilitate an efficient or effective design. Therefore, the use of carburation on multi-cylinder engine requires discarding the standard system. Another likely reason for the use of fuel injection is the situation of the throttle. Figure 2.5 and figure 2.6 shows the relative positioning of the throttle in carburation and fuel injection systems. From figure 2.6, it can be seen that employing multi-point fuel injection means that only the air flows through the restrictor. Figure 2.4 shows the typical position of the throttle on a Formula SAE engine. Unfortunately, using a carburettor means that both the air and fuel mixture must pass through the restrictor. The advantages and disadvantages of each fuel mixture preparation will be analysed further in chapter 6.



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001, p397)

Figure 2.5 Throttle location - carburettor



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001, p395)

Figure 2.6 Throttle location - multi point fuel injection.

2.10 Intake Manifold Design Requirements and Background

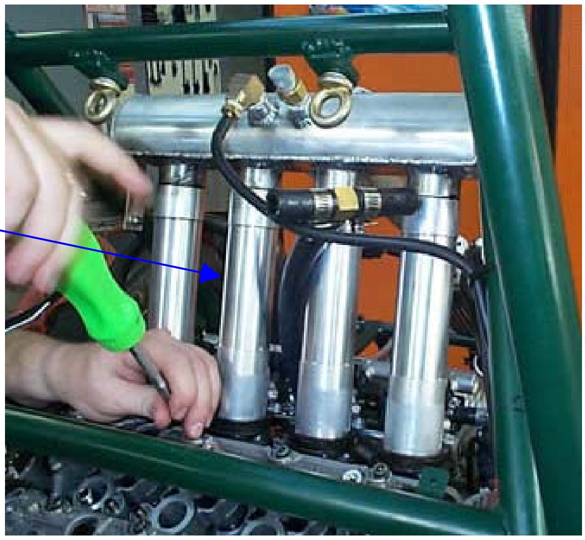
2.10.1 Design Requirements

For the Formula SAE engine, the intake manifold must fulfill the following requirements:

1. Cost effective;
2. Enhance engine performance as much as possible;

2.10.2 Background

The intake manifolds on all of the current Formula SAE-A cars are characterised by long intake runners as shown in figure 2.7 (Formula SAE-A, 2004 et. al.). This type of manifold is termed as 'tuned' and if it is well designed provides a significant increase in volumetric efficiency (Lumley, D. 1998 et. al.). Tuned manifolds have been used to enhance engine performance by Dodge as early as 1962 (Garrett T.K., Steeds W. & Newton, K. 2001 et. al.). Therefore the suitability of this type of manifold will considered when designing the intake manifold for the USQ Formula SAE car.



(*Source*: Curtin Motorsport, 2004)

Figure 2.7 Typical Formula SAE intake manifold.

2.11 Exhaust System Design Requirements and Background

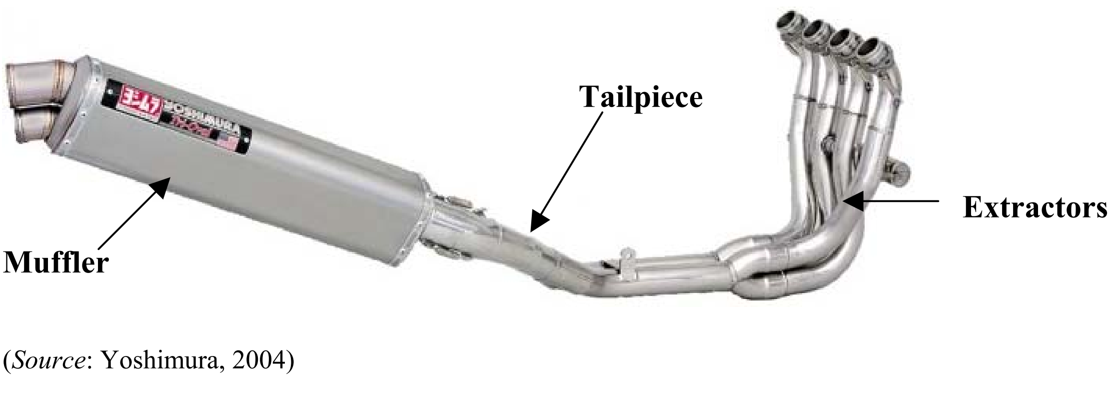
2.11.1 Design Requirements

The design requirements for the Formula SAE-A exhaust system are as follows:

1. Enhance engine performance as much as possible;
2. Meet the Formula SAE-A sound level requirement;
3. Cost effective;

2.11.2 Background

In the Formula SAE-A competition most teams employ the standard motorcycle extractors, designing only the muffler and tailpipe assemblies (Autospeed, 2004). Figure 2.8 shows the components of an exhaust system of a typical four-cylinder motorcycle engine. The fabrication of the tailpiece is required due to the difference in geometry between a motorcycle and the Formula SAE car. It can be assumed that the original extractors are retained, as there is little performance gain obtained by replacing the standard extractors (Limney, C. & J. 2002). The reason for this will be explored in greater detail in chapter 6. The design of the muffler impacts directly on the performance of an engine. Many aftermarket mufflers are available for motorcycle engines. Furthermore, the companies that produce performance mufflers allocate a significant amount of resources to the development of their products (Akrapovic, 2003). All of these manufacturers claim an increase in the output of the engine and many provide dynamometer output to validate their claims. For this reason it can only be assumed that the teams, that choice to design and construct their own mufflers, do so for financial reasons only.



(*Source*: Yoshimura, 2004)

Figure 2.8 An exhaust system for a four-cylinder motorcycle engine.

2.12 Cooling System Design Requirements and Background

2.12.1 Design Requirements

The general design requirement of the cooling system of the Formula SAE car as follows:

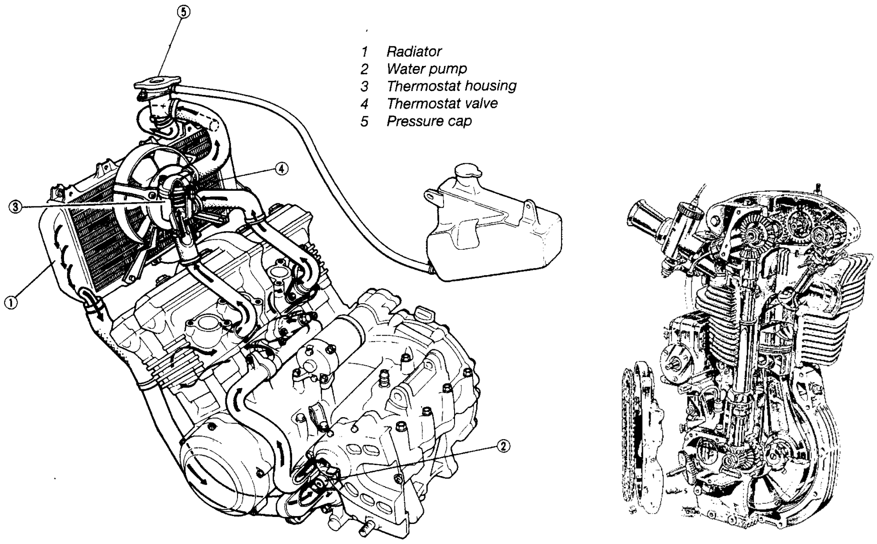
1. Maintain the engine within its operating temperatures;
2. Cost effective;

In order to fulfil these requirements some testing procedures, to test the original motorcycle radiator, will be devised.

2.12.2 Background

In chapter 1 it was stated that the cooling system was a genuine concern when transplanting a motorcycle engine into a Formula SAE car due to the characteristics of the airflow around the cooling system. Figure 2.10 displays two common types of cooling systems. The engine on the right-hand side is air-cooled, employing fins for the method of cooling. Clearly this engine requires a strong flow of air around the fins to enable effective cooling. As the engine is usually mounted behind the driver in a Formula SAE car the fins would receive minimal airflow. Consequently air-cooled engines are rarely used in the Formula SAE competition and there were no air-cooled engines in the Formula SAE-A competition last year (Formula SAE-A, 2004). For the aforementioned reasons the engine that is selected for the USQ Motorsport car will be liquid cooled.

By examining figure 2.1 and figure 1.1 it can be observed that the airflow conditions of the bike's radiator are completely different to the Formula SAE car's radiator. To this end, many of the teams have opted to upgrade the heat capacity of their radiator's or to find a mounting position that best utilises the airflow. For example, the University of Sydney team (2004) runs a naturally aspirated Honda CBR600 motor and has found that using twin radiators are required to maintain the engine at the required operating temperature. Alternatively, the University of Western Australia team (2003) uses a single radiator mounted in a side pod. The selection and position of the radiator will be examined in more detail in chapter 7.



(*Sources*: Ahlstrand, A. & Haynes, J. 1998, p6.2 & Irving, P.1971 p190)

Figure 2.9 Engine cooling systems - liquid and air.

2.13 System Design Requirements

The induction, exhaust and cooling systems are all components of the engine system and all play an important role in determining the overall performance of the engine. As well as fulfilling their individual design requirements all of the engine systems must operate in unison.

2.14 Engine Development Programs

In order to develop their engines to their maximum potential, many of the Formula SAE teams implement thorough development programs. In many cases the engine development team consists of two or three members. The programs all include the use of dynamometers and often employ computational fluid dynamics software to develop certain aspects of the engine. In the preliminary study conducted in 2003 it was discovered that the University of Queensland has an annual budget of $20,000 for the competition of which $2,000 is devoted to tuning the engine (Speering, R. 2003, pers. comm., 23 Sept.). The University of Queensland's tuning is performed by engine tuning professionals (University of Queensland, 2003). Some teams also have a large amount of industry support in terms of component supply and funding. However, this is the first year USQ Motorsport, therefore such industry support and funding has not yet been obtained. Also, given that this project encompasses many facets of engine development it is not possible to utilise the thorough analysis techniques that the more established teams use.

2.15 Conclusion

This chapter has given an overview of the general design requirements of each component that must be designed in this project. The approach that was taken by other Formula SAE teams was also discussed. The following chapter will outline the methodology that was used to try to achieve the design objectives set out in this chapter.

3 Methodology

3.1 Introduction

This chapter will cover the methodologies that were used in an effort to achieve the objectives that have been previously outlined in chapter 2.

3.2 Engine Selection Methodology

For the purpose of this project engines that are available 'off the shelf' were analysed. This approach was taken because an engine that is relatively inexpensive and reliable could be developed within the timeframe that was required. The decisions made in the Formula SAE project, like any other engineering project, are highly dependent upon the resources available. In order to complete the engine selection component of this research project it was assumed that the university had the funding to obtain the 'optimum' engine.

In order to select the engine the following approach was be taken:

1. Different types of engines were researched and classified and the type of engine that best suited the Formula SAE application was selected;

2. The parameters that affect the acceleration performance of the vehicle were researched and defined.

3. A list of all of the suitable transplant candidates was made and the relevant data was gathered. To make this task manageable only engines that were available in the 2003 model year were analysed;

4. Mathematical models of the acceleration performance and the 75 metre times of the Formula SAE car were developed;

5. Programs were written in Matlab to perform the above calculations;

6. The output of these programs was used to compare the vehicle's performance in each event with each engine and the optimum engine was specified;

7. Quotes from local motorcycle dealers and wreckers for the engine were obtained and a decision was made based on the financial constraints of the team;

3.3 Fuelling System Design Methodology

The fuelling system was designed using the following process:

1. A study was carried out to determine if it was feasible to implement a forced air induction system on the 2004 Formula SAE car;
2. A fuel mixture preparation system was specified. See section 3.5 for details;
3. A prototype manifold was designed and constructed. See section 3.6 for more details;
4. A restrictor was designed and tested. See section 3.7 for more details;
5. The exhaust system was designed and recommendations were given. See section 3.8 for details;
6. The entire system was then assembled and partially tested. See section 3.9 for details.

3.4 Fuel Mixture Preparation Specification

The methodology that was used to specify the fuel mixture preparation system was as follows:

1. Carburation and electronic fuel injection were analysed and compared;
2. Cost estimates for the utilisation of electronic fuel injection were obtained;
3. As carburation was the only financially viable alternative, research was carried out to determine the most suitable type of carburettor for the Formula SAE car;
4. Calculations were undertaken to select the correct the size carburettor;
5. Carburettor manufacturers were contacted to obtain costing and design data;
6. A selection was made based on the financial viability of each option.

3.5 Intake Manifold Design Methodology

The methodology used to design the intake manifold was as follows:

1. The USQ mechanical workshop and various workshops in Toowoomba were consulted to ascertain the fabricating methods that were available given the budget of the USQ Motorsport team;
2. Material suppliers were contacted to determine the availability of material;
3. Various configurations for the intake manifold were researched and analysed and the basic layout of the manifold was decided upon;
4. Detailed design of a prototype manifold was undertaken;
5. Sketches of the prototype were drawn and given to the USQ mechanical workshop for construction;
6. Test procedures were devised.

3.6 Restrictor Design and Testing Methodology

3.6.1 Restrictor Design Methodology

The methodology that was used to design the restrictor is as follows:

1. Research was undertaken to determine the optimum shape of the restrictor;
2. Three different shaped restrictors were designed using various standards for flow measurement devices;
3. Detailed drawings of the restrictors were drawn and taken to the USQ mechanical workshop for construction;
4. The restrictors were tested on an airflow bench;
5. The restrictors were tested on the engine;
6. The results were compared and the optimum restrictor was selected.

3.6.2 Restrictor Testing

An airflow bench is commonly used in the automotive speed shops to optimise the performance of cylinder heads. An airflow bench mimics the action of an engine. The bench draws a vacuum and measures the amount of air that passes through the orifice. An airflow bench was implemented in order to verify the optimum internal shape of the restrictor. The owner of the airflow bench conducted the testing of the restrictors. The procedure that was employed is as follows:

1. The restrictor was placed over the passage that the air is drawn through (see figure 3.2);
2. The flow range was set (see figure 3.3);
3. By adjusting the intake flow control knob the downstream pressure in the passage was regulated. The manometer was used to match the desired pressure drop with the actual pressure drop (see figure 3.3);
4. The amount of air that passed through the restrictor as percentage of the flow setting was read from the airflow meter (see figure 3.2);
5. The readings were recorded and the data was corrected using the correction factor provided by the owner;

6. Step 3, 4 and 5 were repeated;

7. Steps 2 to 6 were repeated using manometer readings between 10 inches H_2O and 30 inches H_2O at increments of 5 inches H_2O;

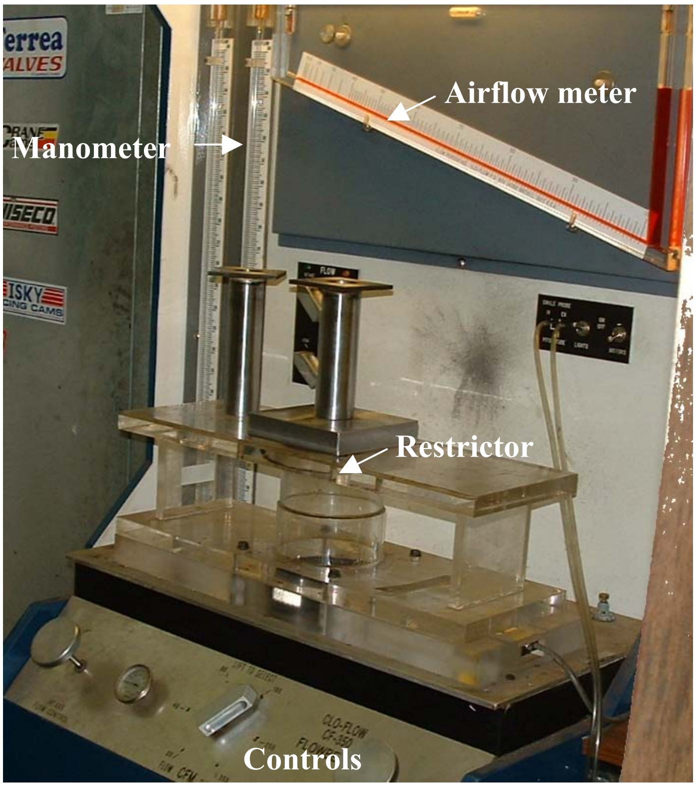


Figure 3.1 The airflow bench used to test the restrictor.

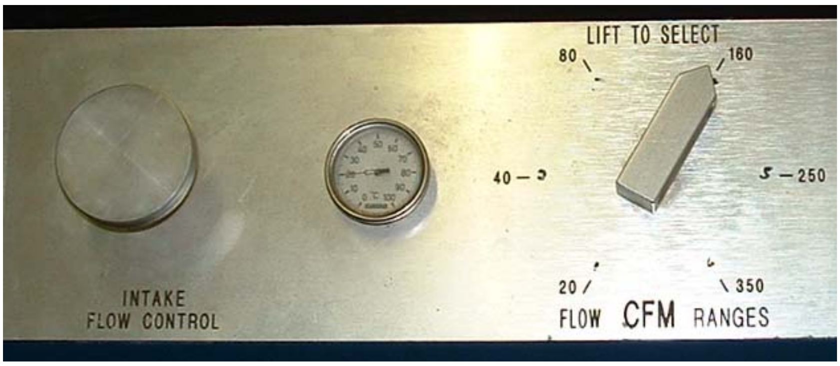


Figure 3.2 Airflow bench controls.

3.7 Exhaust System Design Methodology

The methodology that was used to design the exhaust was as follows:

1. Research was carried to determine the optimum system;
2. The exhaust system was specified;
3. Test procedures were devised.

3.8 Cooling System Methodology

It can be inferred from the discussion in the previous chapters that the actual selection or design of the radiator is very dependent on the shape of the car's bodywork. For this reason a lot of discussion with the bodywork designer was required. Once it had been agreed that the cars bodywork could be designed around the cooling system the process of designing the cooling system entailed:

1. Finding the optimum position for mounting of the cooling system;
2. Devising test procedures to ensure that the original motorcycle radiator was sufficient in the car.

3.9 System Testing

In order to test the cooling and fuelling systems, the following preliminary procedure was used:

1. The entire system was assembled without the restrictor;
2. The carburettor was tuned 'by ear';
3. The restrictor was added and adjustments were made to the carburettor as required;
4. The engine was observed for any signs of missing;
5. Further tests were devised.

3.10 Dynamometer Testing

The USQ owns two dynamometers that are capable of handling the power output of the FZR600 engine. Unfortunately, for the purposes of this project neither of these dynamometers was suitable. The dynamometer that is located under S-block was probably the most suitable for the application. Upon investigation it was found that in order to use it would require the design and construction of some electrical circuitry and also the design and construction of various mounts. Regrettably the technician that knew the most about this dynamometer suffered a serious medical incident, which prevented any further pursuit of this avenue. Another dynamometer is located at the USQ agricultural block. This dynamometer is used to measure the torque output of tractors. However the technician in charge of that particular dynamometer expressed concerns about the sensitivity of the measurements that it produces. The technician was also concerned that there was not anyone on campus that had sufficient knowledge of the equipment to use it properly. Further investigation failed to locate anyone that could use the dynamometer. For these reasons it was crucial to find sponsorship from an outside source.

In order to obtain sponsorship for testing purposes required several visits to various businesses. Initially the businesses were visited to obtain advise on various aspects of the engine. Once a 'friendly' relationship was established between the business owner and the author, sponsorship was easily obtained. Sponsorship was achieved in this way for the dynamometer testing and the airflow bench testing.

The dynamometer that the sponsor owns is classed as a chassis dynamometer. A chassis dynamometer in conjunction with a lambda meter can perform or help perform the following functions:

1. Measurement of power and torque;
2. The carburettor can be developed;
3. Fuel efficiency can be measured;
4. Air/fuel ratios of the engine at different loads can be measured;

The dynamometer would have also been useful to ascertain:

1. Relative losses or gains made by modifying the exhaust and intake systems;
2. Cooling system sufficiency;

Given the functions that a chassis dynamometer can perform, the use of this equipment is essential to conduct any meaningful analysis. In order to use a chassis dynamometer the car must be at least ‘rolling’ (i.e. the wheels must be able to be driven by the engine). For this reason the team was informed as soon as sponsorship for the dynamometer had been obtained. At the team meeting, a date for dynamometer testing was agreed upon. The engine systems were ready to be tested by the agreed date. Unfortunately, the team member in charge of the suspension suffered several months of dire health problems in the time leading up to the agreed testing date. Therefore the suspension systems were not designed or constructed and the car was not ready in time to do the dynamometer tests. At the time of writing the suspension was still under construction

It was thought that the engine could be put back in the motorcycle frame and tested on the dynamometer. But from an inspection of the frame it was found that the intake manifold would not fit in the frame. Therefore finding a sponsor to test the engine on an engine dynamometer was pursued. However, the search for a suitable engine dynamometer in Toowoomba was futile. Therefore for the purposes of this dissertation it was not possible to include the experimental data and analyses that would have resulted from testing using a dynamometer and lambda meter.

3.11 Conclusion

In this chapter the methodology was used to attempt to meet the goals of this project was outlined. The next chapter will detail the selection of the engine.

4 Engine Selection

4.1 Introduction

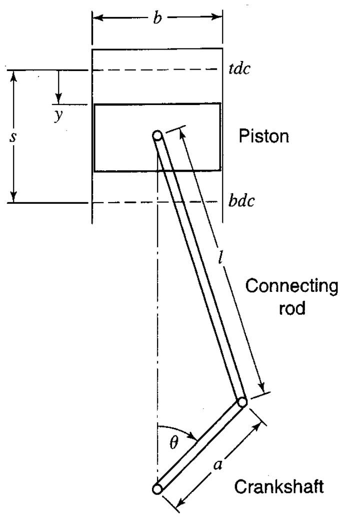
Upon reading the Formula SAE-A rules it is evident that the restrictions placed on competitors are very stringent. The first aim of this chapter is to gain an understanding of the fundamental principles of internal combustion engines and their design and then relate this understanding back to the Formula SAE-A competition rules. The short listed engines that fit the criteria outlined in chapter two will then be presented. This chapter will then turn to the discussion of the simulation method that was used to select the optimum engine for the competition and the results that the simulation produced. The other factors that affect engine selection will also be discussed.

4.2 Introduction to Engines

An internal combustion engine is defined as an engine in which the chemical energy of the fuel is released via burning or oxidization inside the engine and used directly for mechanical work. An external combustion engine utilises a separate combustion chamber to burn the fuel. An example of an external combustion engine is the steam engine. This paper will only deal with IC (internal combustion) engines.

The IC engine was unveiled to the world in 1867 by Nicolaus Otto and can be considered as one of the most significant inventions of modern society. Today IC engines are implemented in countless applications, from powering water pumps through to powering Formula 1 racing cars. Many alternatives to the IC engine have been conceived and developed since the introduction of the IC engine. However the IC engine will remain a part of our society for the foreseeable future.

In recent decades society has witnessed the advent of the rotary piston engine[1]. Nevertheless the reciprocating piston cylinder engine principle is still the most popular among the IC engine designers of today. In the reciprocating cylinder engine a piston oscillates backward and forward inside a cylinder transferring power through the connecting rod to the crankshaft thus transforming linear motion into rotary motion. Figure 4.1 depicts the reciprocating piston cylinder engine. Valves control the flow of gas in and out of the cylinder. This is the same basic principle that Otto incorporated in the very first IC engine. However over time the IC engine has evolved in terms of efficiency with some diesel engines producing thermal efficiencies in the range of 50% as opposed to the original IC engine's 10% thermal efficiency. This improvement in efficiency has come about due to advances in many fields such as fluid mechanics, heat transfer, materials and electronics.



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001, p2)

Figure 4.1 Crank slider model.

[1] See Appendix C for details of the rotary engine.

4.3 Classification of Internal Combustion Engines

Internal combustion engines can be classified in many different ways including:

1. **Application** - Automobile, truck, motorcycle, light aircraft, marine, portable power system or power generation.
2. **Basic engine design** - Reciprocating engines or Rotary engines, which can be subdivided by number of cylinders or rotors and configuration.
3. **Working cycle** - Four-stroke, two-stroke, supercharged, turbocharged or ram air inducted.
4. **Valve or port design and location** - Overhead valves, flathead, desmodromic actuation etc.
5. **Fuel** - Gasoline, diesel, natural gas, liquid petroleum gas (LPG) hydrogen, alcohol.
6. **Method of mixture preparation** - Carburettor, fuel injection which can inject fuel into the cylinder, intake port or intake manifold.
7. **Method of ignition** - Spark ignition or compression ignition.
8. **Combustion chamber design**. - Open chamber, divided chamber, hemispherical etc.
9. **Method of cooling** - Air (utilising fins) or Liquid (utilising a radiator).
10. **Method of load control -** Throttling of fuel and air together, control of fuel alone or a combination of both.

4.4 Method of Ignition

If only the method of ignition is varied, two very different engines result. These engines differ in terms of method of mixture preparation, combustion chamber design, method of load control, fuel used, details of the combustion process, engine emissions and operating characteristics (Heywood, 1988). The two methods of ignition are spark ignition (SI) and compression ignition (CI). Stratified-charge engines are hybrid engines that utilise the best characteristics of CI and SI, however this discussion will be limited to the more common CI and SI engines.

4.4.1 Compression Ignition Engines

The CI engine is commonly called a diesel engine after its inventor Dr Rudolf Diesel. The CI engine utilises a constant pressure air cycle and employs diesel as fuel. In the compression ignition system initially only air is inducted into the cylinder. Once the air has been compressed (and hence heated) in the cylinder, diesel is injected directly into the cylinder just before combustion is intended to occur. The method of load control is achieved by manipulating the flow rate of the fuel only. Compression ignition engines utilise what is termed direct injection as the method of mixture preparation.

Maximum pressures obtained in the cylinders, in modern large diesel engines approach 7MPa which is approximately three times greater than pressures reached in SI engines (Heywood, 1988). Consequently CI engines are typically more robust in design than SI engines in order to cope with the higher stress levels in the engine that result from the higher cylinder pressures. Therefore CI engines generally have a lower power to weight ratio than SI engines. Typical applications of medium and large CI engines, is in heavy industry although they have been utilised in cars by some manufacturers, such as Mercedes. As a general rule diesel engines are not suitable for racing applications.

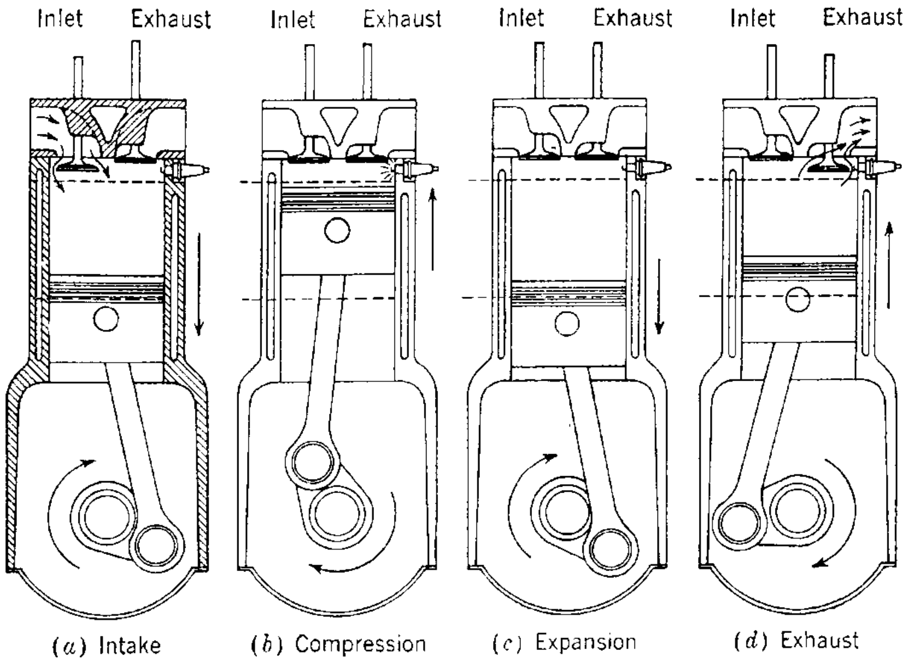
4.4.2 Spark Ignition Engines

A spark ignition engine is usually called a petrol or gasoline engine. These engines can however, run on alternative fuels. The SI engine employs a constant volume air cycle, which is often referred to as the Otto cycle. In a spark ignition engine the air and fuel is mixed in the intake system using a carburettor or fuel injection in the intake system before combustion takes place. Hence load control is achieved through throttling of the air / fuel mixture. The spark ignition engine is the most commonly used engine in the automotive industry and in particular the racing sector. Consequently the variations and the operating parameters of the spark ignition engine will be covered in more depth as this paper progresses.

4.5 Engine Operating Cycles

Internal combustion engines can be categorised in terms of their operating cycles. The two operating cycles are the four-stroke and two-stroke cycles. Both CI and SI engines can utilise either operating cycle but for the purposes of this paper all discussions will now be based around spark ignition engines.

4.5.1 Four-Stroke Engine Theory



(*Source*: Heywood, J. B. 1988, p67)

Figure 4.2 Four-stroke operating principle.

a) Intake Stroke – As the piston lowers from Top Dead Centre (TDC) the intake valve opens and the fuel / air mixture is drawn past the throttle and intake valve into the cylinder.

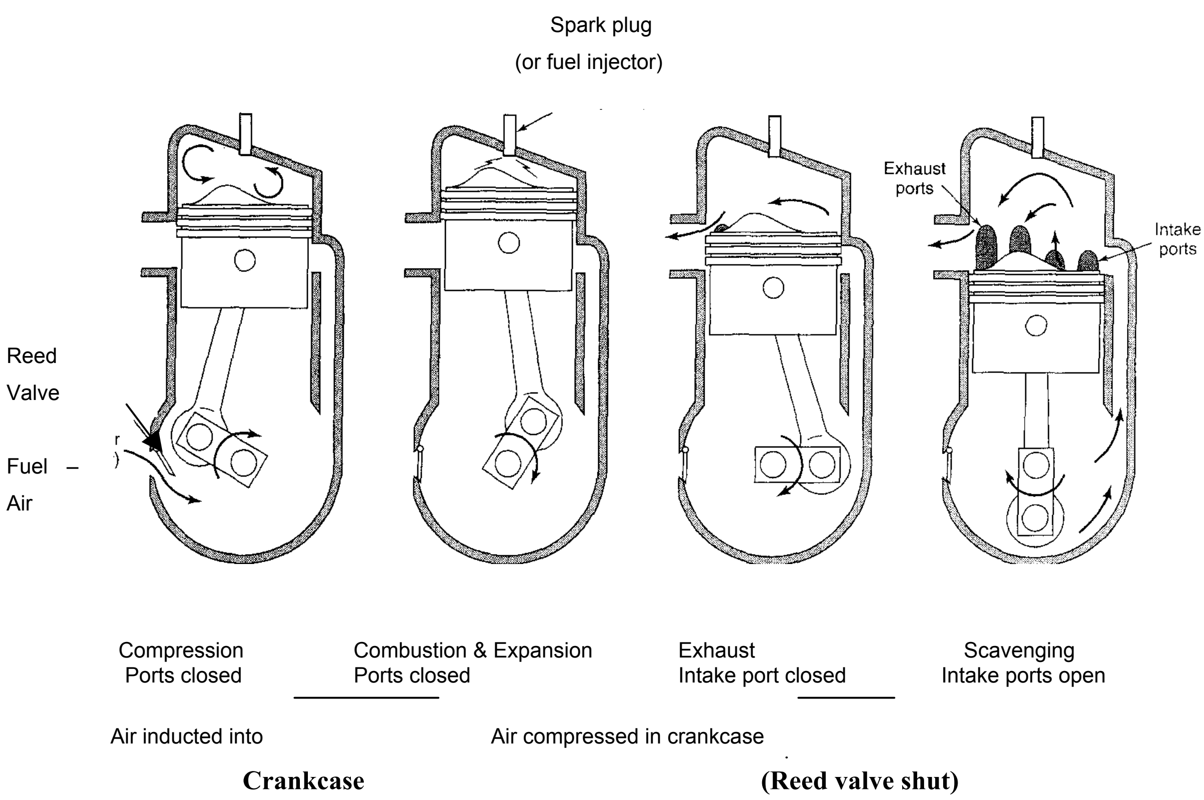
b) Compression Stroke – When the piston reaches bottom dead centre the intake valve shuts. The piston then reverses direction. As the piston raises it causes the mixture to compress and therefore rise in temperature. A spark ignites the mixture a few degrees before the piston reaches TDC.

c) Power Stroke – The piston is driven down due to the rapid gas expansion caused by the combustion of the fuel / air mixture. This causes the crankshaft to rotate. This is the only part of the cycle when useful work is done.

d) Exhaust Stroke – Once the piston reaches bottom dead centre the exhaust valve is opened and the spent gases are pushed through the valve due to the upward motion of the piston. The cycle is then ready to start again.

4.5.2 Two-Stroke Engine Theory

Observing the four-stroke engine we see that the four-stroke engine has only one power stroke for every two revolutions of the crankshaft. In the 1880’s the two-stroke was developed to obtain a power stroke for every revolution of the crankshaft. To demonstrate the two-stroke cycle a cross scavenge two-stroke engine. Will be considered.



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001, p45)

Figure 4.3 Cross scavenge two-stroke engine operating cycle.

(a) During compression of the crankcase a pressure below atmospheric pressure is created in the crankcase, which opens the reed valve. The opening of the reed valve allows air to flow into the crankcase.

(b) Once the piston reverses direction during combustion and expansion begins, the air in the crankcase closes the reed valve so that the air is compressed.

(c) As the piston travels further down the bore the exhaust ports are uncovered. This allows the used gases to start to leave. In turn the cylinder pressure drops too atmospheric pressure.

(d) The intake port is opened and the compressed air from the crankcase flow into the cylinder forcing the remaining gases to leave the cylinder. This part of the cycle is termed as scavenging.

4.6 Limitations of Two-Stroke Engines

The Formula SAE-A rules specify that the car's engine must be a four-stroke SI engine. In order to appreciate why this regulation was put in place the design objectives of the competition must be examined. According to the Formula SAE design objectives the car should be aimed at the novice weekend autocross racer. The rules also indicate that the car must have excellent performance characteristics, be inexpensive, reliable, and easily maintained whilst employing common parts (formula SAE-A Rules, 2004). With these factors in mind, two-stroke engines will be analysed.

With it high lightweight, simple design and high specific power output the two-stroke engine would seem to be the obvious choice for any high performance application. After all, the Grand Prix motorcycles adapted this type of engine up until 2002.The reason GP racers used two-stroke engine was because the two-stroke produces a lot more power than a four-stroke engine of the same cubic capacity. However the two-stroke does not produce double the power of a four-stroke as it theoretically should. Instead the two-stroke produces only about 50% more power (Eastwell, J. et. al. 2002, p5.57). Other drawbacks of two-stroke engines include high fuel consumption, unacceptable polluting emissions and a tendency to be noisy.

The relative ineffectiveness of the scavenging component of the two-stroke operating cycle is responsible for the inefficiency of the two-stroke engine. Short-circuiting, is a problem that arises and is characterized by a portion of the fresh air entering the cylinder exiting directly out the exhaust port. Some of the fresh air will also mix with exhaust gases and will also be wasted. Correct port and piston design can minimize these effects. Unfortunately designers have not yet been able to match the efficiency of the four-stroke engine. Due to the large amount of hydrocarbons produced by the two-stoke engine, currently no large two-stroke engine complies with US emission regulations (Ferguson, C.R. & Kirkpatrick A.T. 2001et. al., p45).

Considering the fuel inefficiency and loudness of the two-stroke it still appears a feasible option due to its high performance capabilities. Unfortunately students are employed to race the car and the ultimately the aim is to sell the car to the public. Why is this a problem? The reason lies in the power curve. From figure 4.4 it can be seen that the power curve of a two-stroke engine is quiet irregular. Initially the engine will feel quiet sluggish and then the 'power-band' will take effect when the throttle is rolled on. This makes the engine unpredictable and the vehicle hard for the novice to tame. To take full advantage of a two-stroke engine an expert would have to be employed to drive the car, which would defeat the purpose of this project. For this reason as well as the other characteristic problems of the two-stroke engine it can be concluded that a two-stroke engine would not satisfy the design objectives of the Formula SAE Competition.

(*Source*: Bianuci, S. 2003)

Figure 4.4 Comparison of two-stroke and four-stroke power curves.

4.7 Summary of Engine Cycles for Formula SAE

From the preceding discussions it is clear that a SI four-stroke engine would be the obvious choice for powering a Formula SAE racing car. The four-stroke engine is the most thermally efficient and 'cleanest' engine expelling the least amount of hydrocarbons. Pollution is a significant design consideration as hydrocarbons from combustion form approximately half of the hydrocarbon pollution in our air. Hydrocarbons can cause cellular mutations and contribute to ground level ozone (Ferguson & Kirkpatrick, 2001). Even though the Formula SAE regulations specify that only a four-stroke engine can be employed in the competition, from the previous analysis it can be seen that the four-stroke engine is obviously the most reasonable choice. All further discussions in this paper will now be focussed on the SI four-stroke engine.

4.8 Engine Options

In the course Systems Design the author undertook research in order to find the most suitable engines for the Formula SAE car. In this research project further research was conducted but failed to uncover any other engines that would be suitable for the car. The engines that were found to be suitable for the car fall into two main categories, single cylinder enduro motorcycle engines and four-cylinder supersport motorcycle engines. The four-cylinder engines include the:

1. Yamaha YZF – R6;
2. Honda CBR 600;
3. Suzuki GSXR 600.

The single cylinder engines include the:

4. Yamaha WR 450F;
5. Honda CRF 450F;
6. KTM EXC 525.

The specifications of these engines can be found in Appendix D.

The following engine manufacturers were considered but were found to be unsuitable:

Apart from two hundred 599cc homologation specials per year, **Kawasaki** currently only produces a 636cc supersport engine therefore does not fulfil the capacity criteria (Bikepoint, 2004).

Husqvarna and **Rotax** produce engines that fulfil all of the performance criteria. However, it was found that it is very difficult to obtain parts for these engines in a timely manner (Independent Wreckers, 2004, pers. comm. 3 Aug). For this reason it was concluded that these engines would be unsuitable for Formula SAE applications.

Harley-Davidson, BMW, Benelli, Bimota, Cagiva, Moto Guzzi, Aprilla, Ducati and **Laverda** do not build engines that conform to the displacement specifications of the Formula SAE Competition (Bikepoint, 2004).

Vespa, Triumph and **Hyosung** do not build engines that fulfil the performance requirements (Bowdler J. 2004).

4.9 Events Related to Engine Selection

In order to select the optimum engine for the Formula SAE car it is necessary to analyse the events that engine selection impact on. Table 4.1 displays the events that the selection of the engine directly relates to.

Events	**Points**	**%**
Cost Analysis	50 / 100	6.9
Acceleration	75	10.3
Skid Pad	50	6.9
Autocross	150	20.6
Fuel Economy	50	6.9
Endurance Event	350	48.4
Total	**725**	**100**

Table 4.1 Events that the engine selection impacts on.

Totalling 500 points the autocross and endurance events make up half of the entire competition points and 69% of the points that are directly related to engine selection. These two events are circuit-racing events and require the car to be able to accelerate from a wide range of velocities. Abundant acceleration is required so that the car can overtake and accelerate out of corners. The velocity range to be considered was set between zero and 120km/hr. This range was decided upon as the information provided by the official Formula SAE website (2004) indicates that the top speed obtainable on the track is approximately 120km/hr.

The other events that depend upon acceleration performance include the acceleration event and the skid pad event. The acceleration event is essentially a drag racing event held on a 75 metre long track. Obviously the engine that applies the greatest average acceleration over 75 metres would be optimum in this event. The skid pad event requires the car to perform figure eights around two witches hats, which are 18.25m apart, in the shortest possible time. In this event it was assumed that engines with the greatest acceleration in the lower velocity range are optimal.

The fuel economy event is weighted with 50 points and as its title suggests is scored on the cars fuel efficiency. From the rules it was found that 50 points of the 100 points allocated to the costing event related to engine selection. The remaining 50 points are awarded on the basis of the quality of the costing report and the manufacturing process discussion.

From table 4.1 and the previous discussions in this section it can be concluded that 86.2% of the points that are reliant on the engine selection are dependent on the acceleration performance of the car. In contrast fuel economy and costing total just 13.8% of the points dependent on engine design.

4.10 Vehicle Performance Modelling

Given that the acceleration performance of the car has the largest weighting, in terms of competition points, it is imperative that the engine that is selected has the best acceleration performance in as many events as possible. The suitable single cylinder engines, on average, weigh 30kg less than the four-cylinder engines but have less torque throughout the rev range than the four-cylinder engines. As both of these parameters directly affect the acceleration performance of the vehicle it is not possible to distinguish which engine is the most suitable by only analysing the engine performance characteristics. In order to determine the engine that would optimise the acceleration performance of the vehicle a mathematical model of the car was developed. To show how the model was created the following section will give a description of the computing equations that were used.

4.11 Development of the Acceleration Performance Model

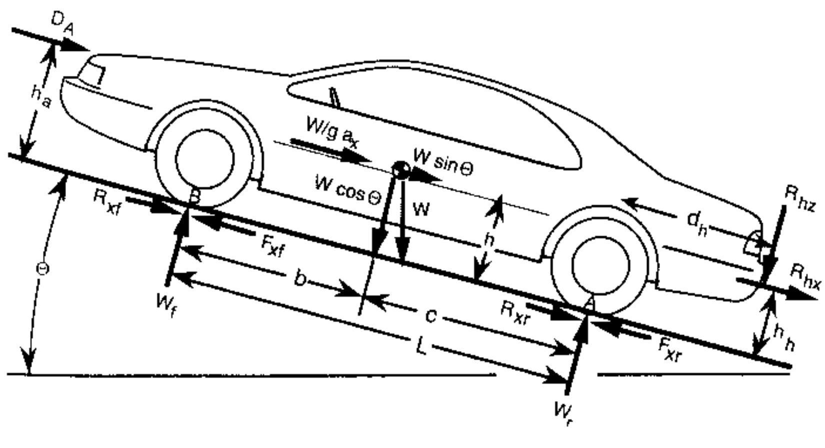
4.11.1 Assumptions

The assumptions made in the development of the acceleration model included:

1. All of the engines will loose the same percentage of torque when the restrictor is fitted and therefore it is reasonable to compare the engines in standard form.
2. No wind or side forces are present.
3. Acceleration is limited by engine performance only.

4.11.2 Equation of Motion

The arbitrary forces that are acting on a car at any time, if no side forces are present, are shown in figure 4.5.



(*Source*: Gillespie, T. D. 1992, p23)

Figure 4.5 Arbitrary forces acting on a motor vehicle.

The equation of motion states that:

$$\sum F_x = m\,a_x \qquad \textbf{(Eq. 4.1)}$$

where F_x is the force in the x direction;
m is the mass of the car; and
a_x is the acceleration of the car in the x direction.

Summing the forces in the x direction in figure 4.5 yields:

$$\left(F_{xf} + F_{xr}\right) - R_{x\,f} - R_x - D_a - W\sin(\theta) - R_{hx} = m\,a_x \qquad \textbf{(Eq. 4.2)}$$

where F_{xr} is the total tractive force applied by the engine to the rear wheels;
F_{xf} is the total tractive force applied by the engine to the front wheels;
R_{xf} is the force applied to the front tyres due to friction;
R_{xr} is the force applied to the rear tyres due to friction;
R_{xh} is the hitching forces that come about when towing a trailer etc.;
D_a is the force due to aerodynamic drag; and
$mg\sin(\theta)$ is the component of the weight acting in the x direction.

The forces induced by the car's weight in the horizontal plane were considered to be negligible as the track is relatively flat. The hitching forces were also neglected, as the car will not be used for towing. Grouping the forces applied to the front and rear wheels together and setting the hitching and weight forces to zero, results in equation 4.3:

$$F_x - R_x - D_a = m\,a_x \qquad \textbf{(Eq. 4.3)}$$

4.11.3 Aerodynamic Drag

The aerodynamic drag force acting on the car was calculated using equation 4.4.

$$D_a = \frac{1}{2}\rho v^2 C_D A \qquad \textbf{(Eq. 4.4)}$$

where A is the cross-sectional area of the car perpendicular to the air flow;
v is the velocity of the car;
ρ is the density of the surrounding air; and
C_D is the aerodynamic drag co-efficient.

The aerodynamic drag coefficient was assumed to be 0.5. This assumption was made, as no data regarding the true drag coefficient of the car was available at the time.

4.11.4 Rolling Resistance Forces

The rolling resistance forces acting on the car were calculated using equation 4.5.

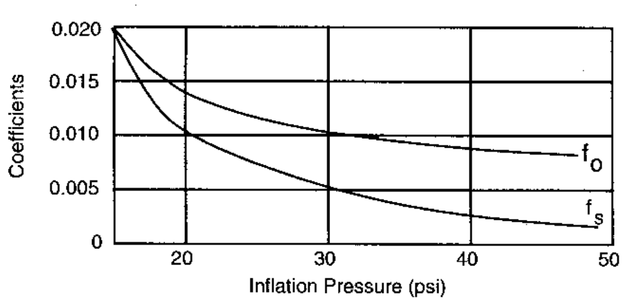
$$R_x = m\,g\,f_r \qquad \textbf{(Eq. 4.5)}$$

where f_r is the friction coefficient. The friction coefficient varies with inflation pressure and velocity (Gillespie, 1992). To calculate the friction coefficient, equation 4.6 was used:

$$f_r = f_o + 3.24 f_s \left(\frac{v}{100}\right)^{2.5} \qquad \textbf{(Eq. 4.6)}$$

where f_o is the basic coefficient;
f_s is the speed coefficient; and
v is the vehicle speed in miles per hour.

The basic and speed coefficients were determined from figure 4.6, assuming a tyre inflation pressure of 32psi.



(*Source*: Gillespie, 1992, p105)

Figure 4.6 Speed and standard coefficients versus tyre inflation pressure.

Figure 4.7 represents the magnitude of the drag force and the rolling resistance force acting on the Formula SAE car. From figure 4.7 it can be seen that as speed increases the effect of the drag forces increases drastically in comparison to the rolling resistance forces.

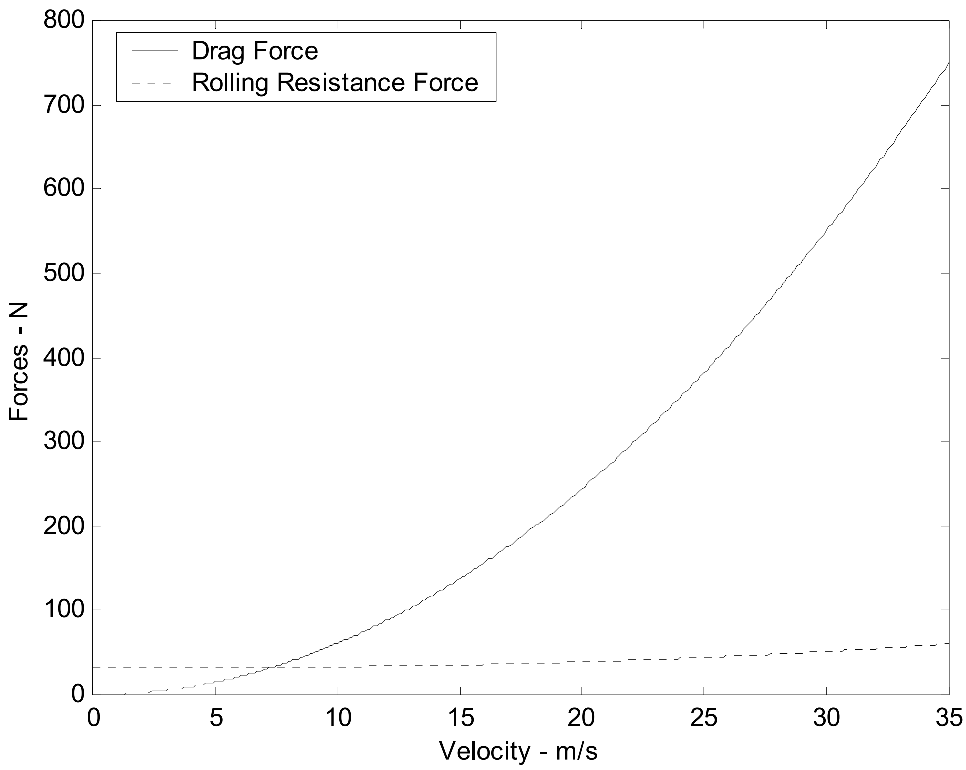


Figure 4.7 Rolling resistance and drag forces versus velocity.

4.11.5 Tractive Force

The tractive force is the force that is applied by the engine to overcome the rolling resistance and drag forces. The tractive force can be calculated using equation 4.7:

$$F_x = \frac{T_e\, n_{tf}\, \eta_{tf}}{r} - \left[(I_e + I_t) n_{tf}^{\ 2} + I_d\, n_f^{\ 2} + I_w\right] \frac{a_x}{r^2} \qquad \textbf{(Eq. 4.7)}$$

where n_{tf} is the rotational speed of the rear wheels ;

I_e is the engine rotational inertia;

I_t is the rotational inertia of the transmission;

I_d is the rotational inertia of the drive shaft;

I_w is the rotational inertia of the wheels and the axle shafts;

r is the radius of the rear wheel;

n_f is the numerical ratio of the final drive;

n_{tf} is the combined ratio of the transmission and drive;

η_{tf} is the combined efficiency of the transmission and drive; and

T_e is the torque applied by the engine.

The first term on the right hand side of equation 4.7 represents the force applied by the engine. The second term on the right hand side of equation 4.7 represents the loss of tractive force due to the inertia of the engine and drive train components. The use of equation 4.7 is the most accurate method of calculating the tractive force. In order to perform this calculation requires access to information, such as drive shaft inertia, that was not available when the model was constructed. However, equation 4.8 can be used to estimate the acceleration performance of the vehicle without knowing the value of all of the inertias.

$$(m + m_r)\, a_x = \frac{T_e\, n_{tf}\, \eta_{tf}}{r} - R_x - D_a - mg\sin(\theta) \qquad \textbf{(Eq. 4.8)}$$

where m_r is the equivalent mass of the rotating components and takes into account the loss of tractive force due to the inertia of the engine and drive train components. The combination of m and m_r is the equivalent mass and the mass factor is represented by:

$$\text{mass factor} = \frac{(m + m_r)}{m} \qquad \textbf{(Eq. 4.9)}$$

Gillespie (1989, p34) recommends the use of equation 4.10 to estimate the mass factor.

$$\text{mass factor} = 1.04 + 0.0025\, n_{tf}^{2} \qquad \textbf{(Eq. 4.10)}$$

Since the total drive train reduction ratio was known, equation 4.10 was used to calculate the mass factor. Once the mass factor was calculated, equation 4.8 was employed to calculate the acceleration performance of the Formula SAE car.

4.11.6 Time to Velocity and Distance to Velocity

In order to make a prediction of the relative performance of each engine in the acceleration event the time to velocity and the velocity to distance was calculated. The time to reach velocity was calculated using equation 4.11:

$$t_{v_1 \to v_2} = \int_{v_1}^{v_2} \frac{1}{a_x} dv \qquad \textbf{(Eq. 4.11)}$$

The distance required to reach a velocity was calculated using equation 4.12:

$$s_{v_1 \to v_2} = \int_{v_1}^{v_2} \frac{v_x}{a_x} dv \qquad \textbf{(Eq. 4.12)}$$

4.12 Program Construction

As each force and hence the acceleration of the car changes according to engine and vehicle velocity, the use of software to perform the above calculations was necessary. The torque of the engine was taken from dynamometer graphs[2]. It is generally accepted that dynamometer output varies from dynamometer to dynamometer. For this reason, where possible, the data was taken from a single source. Figure 4.9

[2] See Appendix D for torque and power charts.

displays the calculated acceleration performance of the Formula SAE car in each gear as calculated by the script *accelerations*. The cross points of the acceleration curves represent the optimum gearshift point[3].

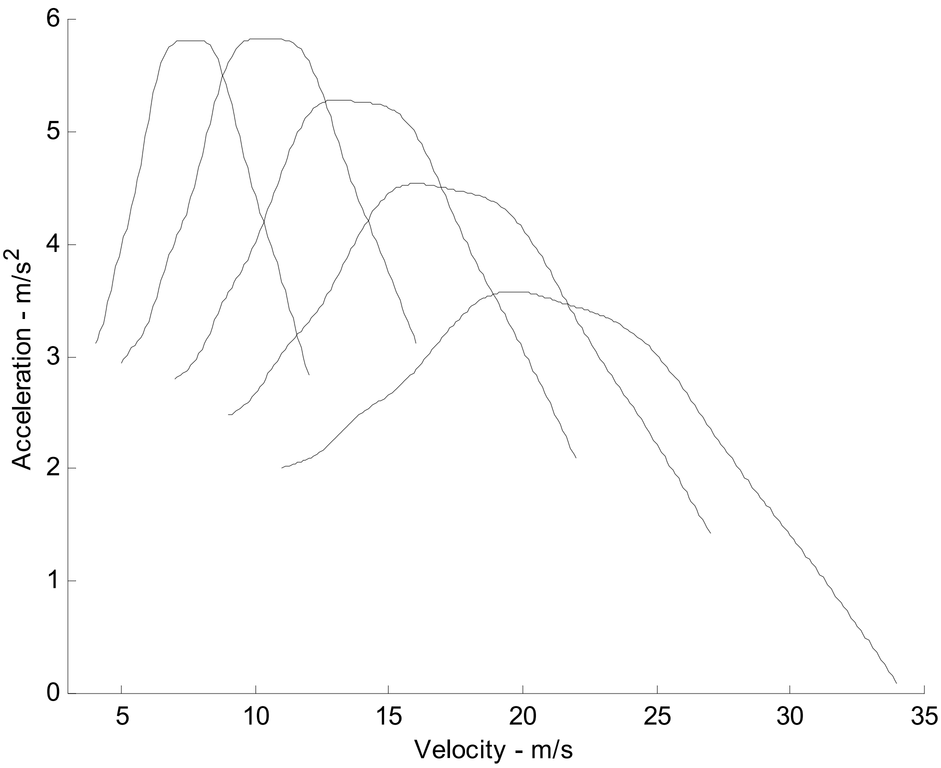


Figure 4.8 Acceleration versus car velocity for the F-SAE car fitted with a Yamaha WR450 engine.

Figure 4.9 and figure 4.10 are examples of the output of the Matlab script *GSXR75mtime*. Figure 4.9 represents the time it takes to reach any velocity between 5 and 33m/s. Figure 4.10 represents the distance required to reach any velocity between 5 and 33m/s. The starting velocity is determined from the engine speed when the clutch is released at the start line. Because the engine is not rotating at 0rpm the take-off velocity of the car is not assumed as zero.

[3] See Appendix D for a flowchart detailing the input, output and programming structure that was used to construct these programs. A Matlab file of these programs can be found on the disk containing this dissertation.

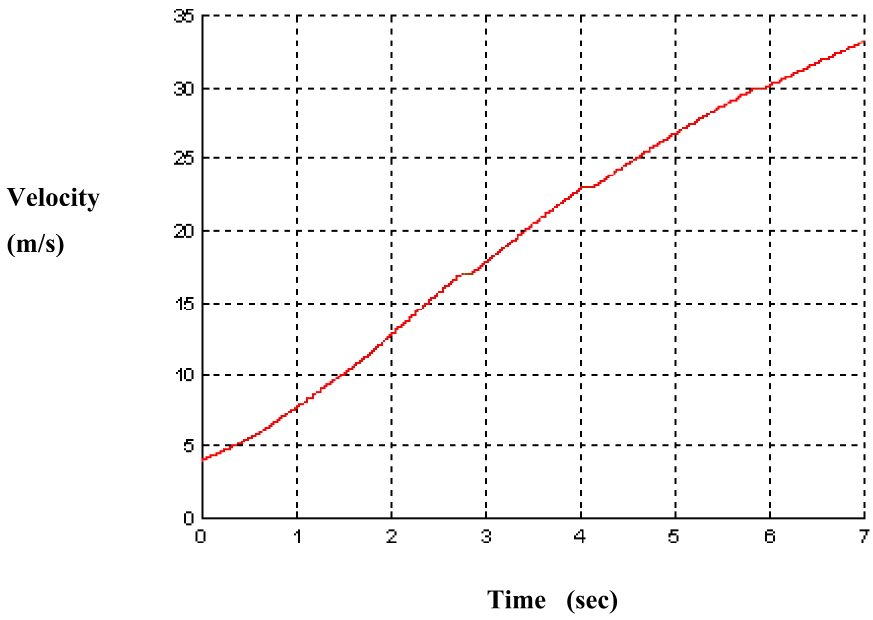


Figure 4.9 Time verus velocity for a Formula SAE with a Suzuki GSXR 600 engine.

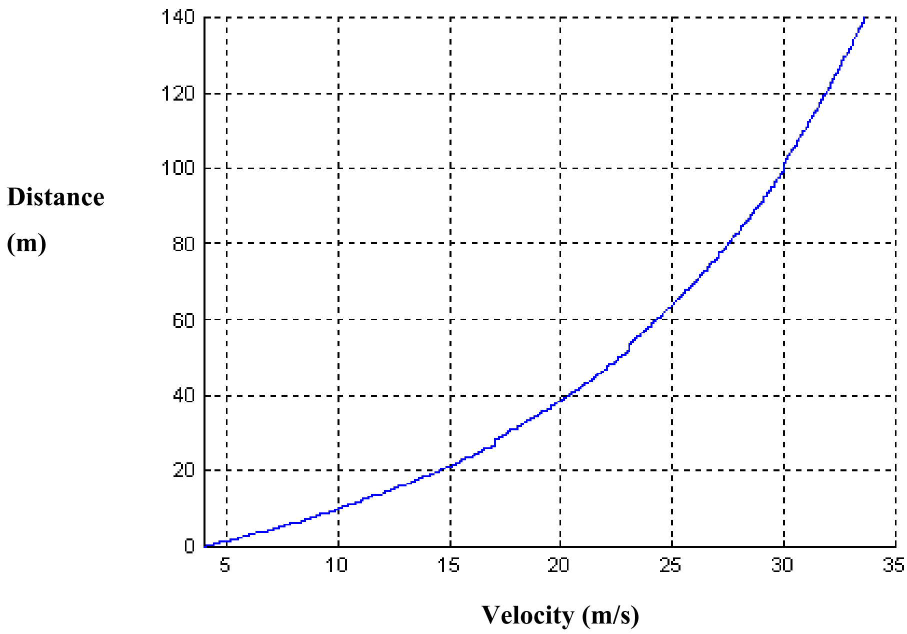


Figure 4.10 Velocity versus distance for the Formula SAE car with a Suzuki GSXR engine.

4.13 Analysis of Acceleration Performance of the F-SAE car

Initially engines of similar specification were compared to see if there were any significant differences in the acceleration performance of the car with each engine. The comparison was performed by overlaying the acceleration performance curves of the car fitted with each engine. When comparing the acceleration performance of the single cylinder KTM EXC 525, Honda CRF 450 and the Yamaha WR450 it was clear that Yamaha WR450 has superior acceleration performance across the entire velocity range. Figure 4.11 shows how striking the difference in acceleration performance was between the Yamaha WR450 and Honda CRF450.[4] From this observation it was concluded that if the car was fitted with either the Yamaha WR450, Honda CRF450 or KTM engine and was raced in the endurance, autocross, acceleration and skid pad events the Yamaha would win by a considerable amount. As these events carry 86.2% of the points that are being considered in this analysis, the KTM and Honda CRF450 were not considered any further.

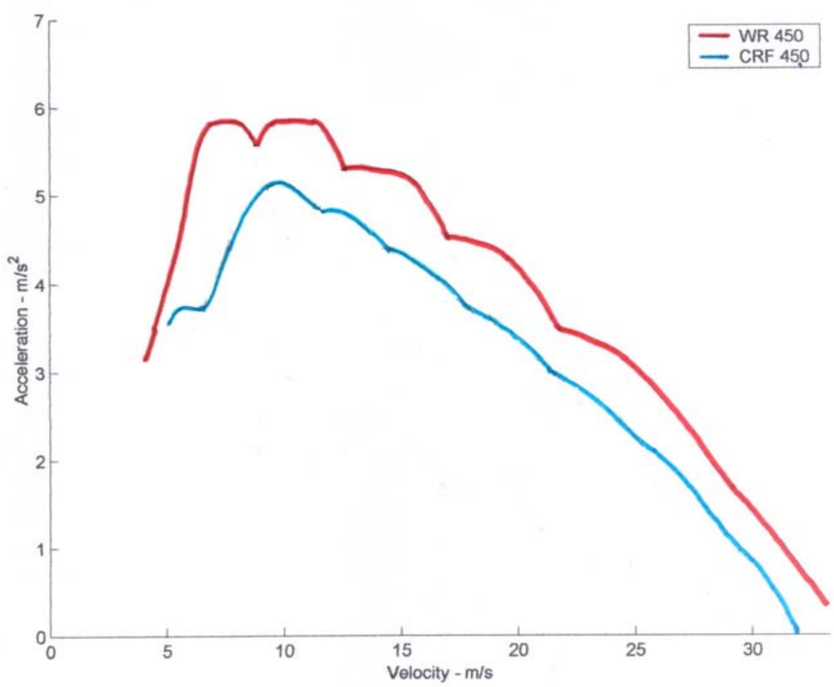


Figure 4.11 Comparison of the acceleration of the Formula SAE car with a Yamaha WR450 and Honda CRF450 engine.

[4] A complete compilation of the comparative acceleration curves can be found in Appendix D.

When comparing the four-cylinder engines, differences between the engines were not as significant, as can be observed from figure 4.12. However it was found that the Suzuki GSXR 600 did display superior performance across approximately 90% of the velocities analysed. From this it was concluded that if the car was fitted with either, a Suzuki GSXR 600, Honda CBR 600 or Yamaha YZF-R6 and raced in the endurance, autocross, acceleration and skid pad events the Suzuki would win.

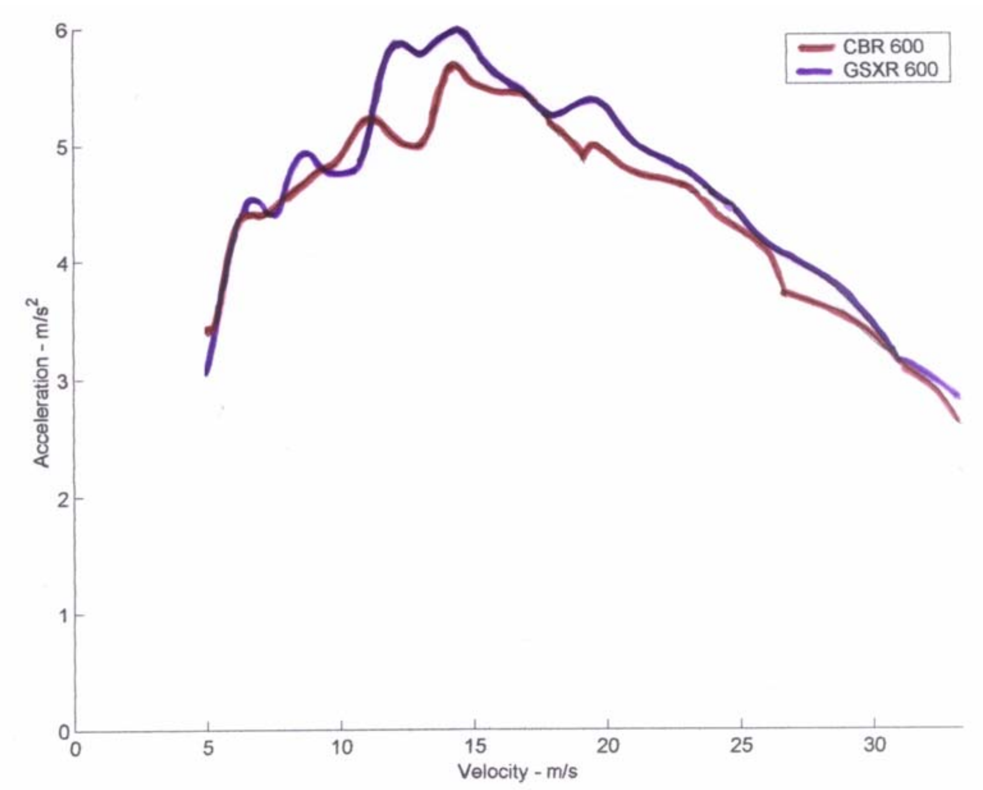


Figure 4.12 Comparison of acceleration performance of the Formula SAE car with a Suzuki GSXR 600 and a Honda CBR 600 engine.

From the two previous analyses it was found that the two highest performing engines from each category were the Yamaha WR450 and the Suzuki GSXR600. Figure 4.13 shows a comparison of the acceleration performance of the car with each of these engines fitted. From figure 4.13 it can be seen that the Yamaha WR 450 has more acceleration performance than the Suzuki GSXR600 at velocities below 44km/hr, whilst the Suzuki GSXR 600 has more acceleration performance than the WR 450 at all velocities greater than 44km/hr. Evidently this would make the WR450 ideal for the skid pad event. The ideal engine for circuit racing events is heavily dependent on the design of the track. If the car travelled at velocities below 44km/hr more than it travelled at velocities above 44km/hr the single cylinder engine would be the most suitable in these events. From the information published on various universities' Formula SAE-A websites it was found that the car did travel constantly at velocities above 44km/hr. To verify this data will require USQ Motorsport to compete at the competition. However, in this analysis it was assumed that the car will spend a lot more time above 44km/hr than below 44km/hr. Therefore, it was concluded that the GSXR600 engine would be more suitable than the Yamaha WR450 engine.

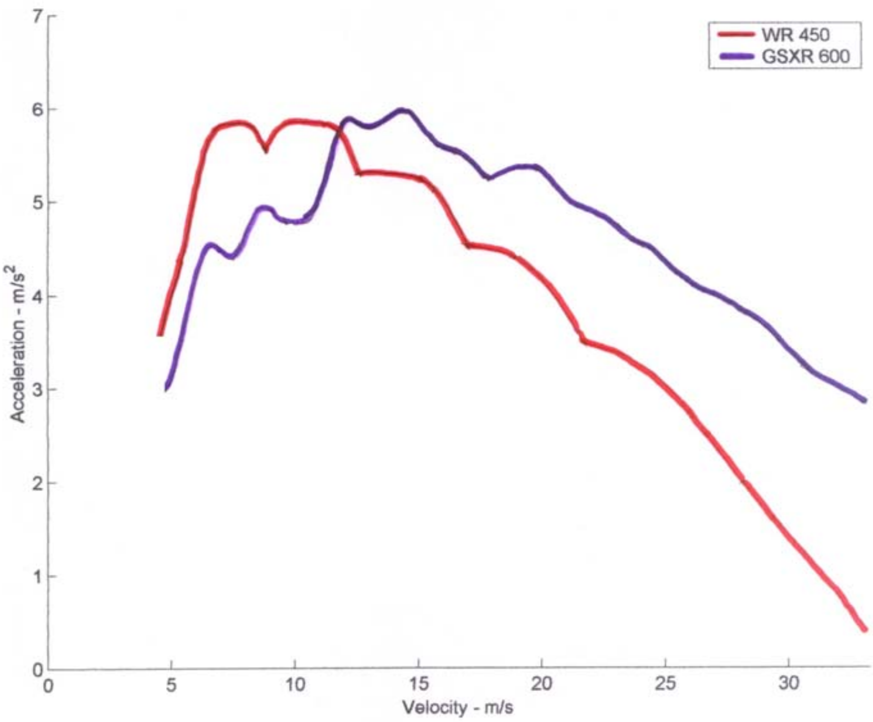


Figure 4.13 Comparison of acceleration for the Formula SAE car with a Yamaha WR450 and Suzuki GSXR600 engine.

4.14 Analysis of the Acceleration Event

By taking the time to reach velocity from the time versus velocity plot and the distance taken to reach a velocity from the distance verus velocity plots, the time taken to cover 75 metres was found. It was found that the car fitted with a Yamaha WR450 would take approximately 5.1 seconds to cover 75m whereas a Formula SAE car fitted with a Suzuki GSXR600 engine would take approximately 4 seconds. This calculation also verifies that the model is reasonably accurate as the University of Queensland's Formula SAE team quoted an approximate time of 4 seconds for the acceleration event, using a four-cylinder engine (University of Queensland, 2004).

4.15 Cost Analysis

The 50 points that are directly affected by the engine design are divided into two categories. Twenty points are allocated for the manufacturing feasibility of the car and 30 points are allocated for the lowest cost (Formula SAE-A Rules, 2004).

In terms of manufacturing feasibility it was concluded that the four-cylinder supersport engines and the single cylinder enduro engines would score similarly in this part of the event. The reason for this is that both engines are mass-produced so obviously they are both feasible to produce.

The costing of the engine and accessories consists of a separate costing for the engine, intake manifold, exhaust manifold, cooling system and mufflers. Given that both engines usually have a single muffler and water-cooling the differences in the cost of these components would be minimal. The cost of the engine is calculated by multiplying the capacity by 1.25 (Formula SAE-A Rules, 2004). Therefore the single-cylinder Yamaha would have a slight advantage in this part of the cost analysis. The cost of the exhaust manifold and intake manifold would also be less on the single cylinder engine as the complexity of these components would less on the single cylinder engine.

4.16 Fuel Economy Event

So far it has been concluded that a Suzuki GSXR600 powered Formula SAE car would be superior in the endurance, acceleration and autocross events. Therefore, the GSXR600 would score a higher proportion of 79% of the points available. It was concluded that the WR450 would probably win the skid pad event and would definitely achieve more points in the costing component of the cost analysis. This means that the WR would only score a higher proportion of 9.6% of the points available. The only event that has not been analysed is the fuel consumption event.

Comparing fuel consumption is very difficult given each of the engines were originally in two completely different styles of motorcycles. It is possible to calculate the brake specific fuel consumption of each engine. However, this does not take into account the different masses of the vehicle when the car is fitted with each engine. However the analysis of the fuel consumption was deemed unnecessary given that the Suzuki would achieve a higher proportion of 79% of the points. In order to beat the Yamaha WR450 the Suzuki would merely have to complete the race. Given that many Formula SAE cars have been fitted with a GSXR engine and have not ran out of fuel, it was concluded that this was not an issue. Therefore it was concluded that the Suzuki GSXR600 engine is the optimum engine from the 2003 model range.

4.17 Other Factors that affect Engine Selection

So far the only factors that have been included in the selection of the engine are factors that would affect the cars performance in competition. However other factors are important when selecting an engine. These factors include, reliability, availability of aftermarket parts and backup service.

4.17.1 Engine Reliability

When considering the reliability of engines for the Formula SAE car the simplest way to find out if any engine had poor reliability was to look at the history of the engine in racing conditions. Table 4.2 shows that each supersport engine under consideration has had a successful history in the World Supersport Series. It can be assumed that results of the competition demonstrate an engine's reliability under racing conditions. From the results it can be assumed that all of the supersport engines under consideration would be reliable and suitable for the Formula SAE competition. The Yamaha WR450 has only been in production for three years. In this time it has won many endurance races including the 2003 Paris-Dakar (Yamaha Design Café, 2004). Endurance events are very demanding on engines, as the motorcycles are ridden 'hard' for days at a time. Given that the Yamaha WR 450 has won such events, demonstrates that this engine is also reliable in racing conditions.

Year	Manufacturer	Model
2003	Honda	CBR 600
2002	Suzuki	GSXR 600
2001	Yamaha	YZF-R6
2000	Yamaha	YZF-R6
1999	Yamaha	YZF-R6
1998	Suzuki	GSXR 600

(*Source*: World Supersport Official Homepage, 2004)

Table 4.2 World Supersport Series winners.

4.17.2 Aftermarket Parts and Backup Service

Since the Formula SAE competition is heavily performance based, it is important to have access to performance enhancing parts. All of the engines that were considered were designed with racing in mind (Motorcycle News, 2004). For this reason a great deal of aftermarket performance enhancing parts are available. The aftermarket parts that can be purchased for each of these motorcycles include mufflers, fuel injection upgrades, camshafts, pistons etc. (Serco, 2004). Toowoomba has dealerships for each of the manufacturers considered, so back-up service is available.

4.18 University Purchase Costs

4.18.1 New Engines

From the previous analysis buying a new Suzuki GSXR600 engine was clearly the best option. Unfortunately a new engine cannot be purchased as a complete assembly. Instead the engine has to be bought in separate components. It was found that to build a complete four-cylinder Suzuki GSXR600 engine would cost in excess of $15,000 and therefore completely prohibitive. The actual cost of the Suzuki GSXR 600, new, is around $15,995 (Boyd Young Suzuki, 2003, pers. comm. 23 Oct.). Another approach would have been to buy a new bike, remove the motor and sell the rest. This would have probably resulted in paying around $7,000 for the engine and gearbox. Again, budgetary restraints prohibited this approach. Another option was to obtain sponsorship for the engine. Unfortunately attempts to gain sponsorship for the engine were fruitless, probably because USQ Motorsport is an unproven first year team.

4.18.2 Second-hand Engines

The prices for second-hand supersport engines ranged between \$2000 and \$4000, depending on vintage and condition (Independent Wreckers, 2004, pers. comm.11 Oct.). Unfortunately, the cost of purchasing a second-hand supersport engine was also beyond the budget of USQ Motorsport. The purchase cost of enduro engines is approximately the same as the cost of the supersport engines (Independent Wreckers, 2004, pers. comm. 11 Oct). Therefore, enduro engines did not pose any cost saving advantages.

Given the stringent budget constraints that were imposed on engine selection it was decided to buy the latest model four-cylinder engine that USQ Motorsport could afford. To take advantage of the extra components that a complete motorcycle offers, the team decided to purchase a crashed 1993 Yamaha FZR600 motorcycle. The wreck was purchased for \$500 from Fowles Auction Centre in Brisbane. The extra components were useful to other team members, as it was possible to adapt other parts from the wreck, such as the brakes to the car. The engine from the wreck is depicted in figure 4.14.

Figure 4.14 The Yamaha FZR 600 engine sourced from the wreck.

4.19 Conclusion

In this chapter the basic principles of engine design were discussed. The options available for the engine for the Formula SAE competition were then addressed. This chapter also detailed the model that was used to compare the performance of the car with different engines. From the results of the simulations conducted, it was concluded that the optimum engine for the car would be a new four-cylinder Suzuki GSXR 600 engine with a maximum torque output of 63Nm @10,750rpm and a maximum power output of 77kW @ 13,250rpm. However, it was found that the budget constraints of this project did not allow the purchase of such an engine. For this reason a 1993 Yamaha FZR600 engine with a maximum power output of 56kW @ 10500rpm and a maximum torque output of 58Nm @ 8500rpm was purchased. The following chapter will discuss the consequential effects of this project.

5 Consequential Effects

5.1 Introduction

Any engineering project has consequential effects. These effects include aspects of sustainability, safety and ethics. In this chapter the consequential effects of this research project will be identified and assessed.

5.2 Safety

This section will deal with the safety issues of this project. In order to accomplish this a simple safety analysis of the project components will be carried out. Safety analyses consist of hazard, risk and preventive measures. A hazard is what could cause harm. A risk is how likely the hazard will cause harm and what the outcome of any harm would be. Preventative measures minimise the hazards or the likelihood of them occurring.

5.2.1 Safety Analysis

Computers

Hazard → The radiation from the computer may cause sore eyes and ultimately a loss of vision.

Risk → Low

Preventative Measure → Frequent breaks of five minutes duration every hour should be taken.

Engines

Hazard 1 → When working on the engine knuckles and hands could be damaged if tools slip.

Risk → Medium

Preventative Measure → Tools should be used in the correct manner and due care should be taken.

Hazard 2 → When the engine has to be moved, muscle and skeletal damage could occur as engines typically weigh upwards of 50kg.

Risk → Moderate

Preventative Measure → Use a trolley when moving the engine from place to place. If the engine has to be lifted seek assistance so that the load is shared.

Hazard 3 → When engines are running they become very hot and cause serious burns if touched by the body.

Risk → Moderate

Preventative Measure → Make sure that all parts of the body are covered (i.e. wear a long sleeve shirt and trousers). Take care not to touch hot parts of the engine when it is running.

Hazard 4 → If the engine is dropped, toes and feet could be broken.

Risk → Low

Preventative Measure → Take due care and wear safety shoes when working with the engine.

Hazard 5 → Due to their nature engines have many moving parts that extremities could get caught in. Body parts can be severed or seriously damaged.

Risk → High

Preventative Measure → Keep all clothes tucked in and buttoned up. Take due care when the motor is running.

Hazard 6 → Engines exhaust harmful fumes. Excessive inhalation can cause permanent lung damage and even death.

Risk → Moderate

Preventative Measure → When working in confined spaces exhaust all fumes accordingly.

5.3 Ethics

In this project all of the work is to be conducted in accordance with the Tenets of the Engineers Australia 'Code of Ethics.' The aspect of this project that may present ethical issues is in obtaining sponsorship. It must be assured that all of the dealings with the public are carried out in an honest and ethical manner. This means not being less than truthful to potential sponsors in regards to the benefits and exposure that the company may receive.

5.4 Sustainability

The following section will examine aspects of sustainability. This assessment will be carried out in accordance with the policies set out by the Engineers Australia sustainability code, *Towards Sustainable Engineering Practice: Engineering Frameworks for Sustainability*.

The use of any device that is fired with fossil fuel raises concerns for the future development and environmental needs of future generations. There are two central concerns when using gasoline for combustion. The first issue is that gasoline is a finite resource, which raises concerns for the availability of this product for future generations. The fumes that are exhausted by gasoline-fired engines also expose current and future generations to the risk of environmental contamination.

In order to reduce emissions and fuel consumption the Society of Automotive Engineers has specified that a four-stroke conventional piston engine must be fitted if any team is to compete in the Formula SAE competition. This may be due to the fact that generally the conventional piston four-stroke is far more efficient and produces less harmful emissions than the two-stroke or rotary engines[5]. The competition also involves an event devoted entirely to reducing fuel consumption. Hence the competition itself stipulates a form of environmental protection.

In order to take a proactive approach to the harmful emissions exhausted by the engine, a catalytic converter could be implemented. At the time of writing it is the author's belief that the time stipulations of this project will not allow development in this area. Therefore, other students should consider the use of a catalytic converter or some other pollution control devices in the future.

Noise pollution is considered an issue in any environment. Engines are inherently noisy and could cause disturbances to people that are present in the surrounding environment. In this project a lot of testing will be carried out. This may be very annoying or even impact on the health of the people that work in the workshop or surrounding area. To avoid any problems, when the engine is to be tested, the people affected by the noise will be informed in advance. This will be done so that measures can be taken to work around any issues that may arise.

The guidelines set out by Engineers Australia state that the reductions in differences in living standards among women, youth and indigenous people are essential to achieve sustainability. This project does not directly impact on this issue. However it has been stated, in our meetings, that the team should strive to involve people from *all* different cultural, socio-economic or other groups. Therefore it is the team's

[5] See section 4.6 for details.

responsibility to *not* participate or support behaviour that could be perceived by anyone as discriminatory.

This project will not affect developed countries differently to undeveloped countries and does not assist any form of warfare. For this reason aspect 9 and 10 of the *Frameworks for Sustainability* are not relevant to this project work. In summary, this project will be carried out in manner that is fitting to the sustainability standards set out by Engineers Australia.

5.5 Conclusion

In this chapter the consequential effects of this project have been discussed. In the next chapter the design and specification of the fuelling system for the Formula SAE car will be detailed.

6 Fuelling System

6.1 Introduction

This chapter will document the design and testing of the fuelling system. In this project the fuelling system will imply the fuel preparation system, restrictor, induction and exhaust manifolds and muffler. A brief study encompassing the feasibility of implementing a forced air induction system on the USQ Formula SAE car will also be included in this chapter.

6.2 The Effect of Fuelling System Design on Performance

The type of aspiration, method of fuel mixture, the induction system and exhaust systems are all important factors that play a part in determining the performance of an engine. Brake power is the power measured at the flywheel by an engine dynamometer and is reflective of an engines performance. Brake power can be determined from:

$$P_b = \eta_v \, \eta_i \, \eta_m \, \eta_c \, \rho_i \, V_d \, f \, Q \, \frac{n}{X} \qquad \textbf{(Eq. 6.1)}$$

where η_v is the volumetric efficiency;

η_i is the indicated efficiency;

η_m is the mechanical efficiency;

η_c is the combustion efficiency;

ρ_i is the inlet density of the air into the intake manifold;

V_d is the engine displacement;

X is the number of power strokes per revolution of the crankshaft;

n is the engine speed;

Q is the calorific value of the fuel; and

f is the fuel:air ratio;

In this research project only the volumetric efficiency and the air inlet density will be considered.[6]

6.3 Forced Air Induction

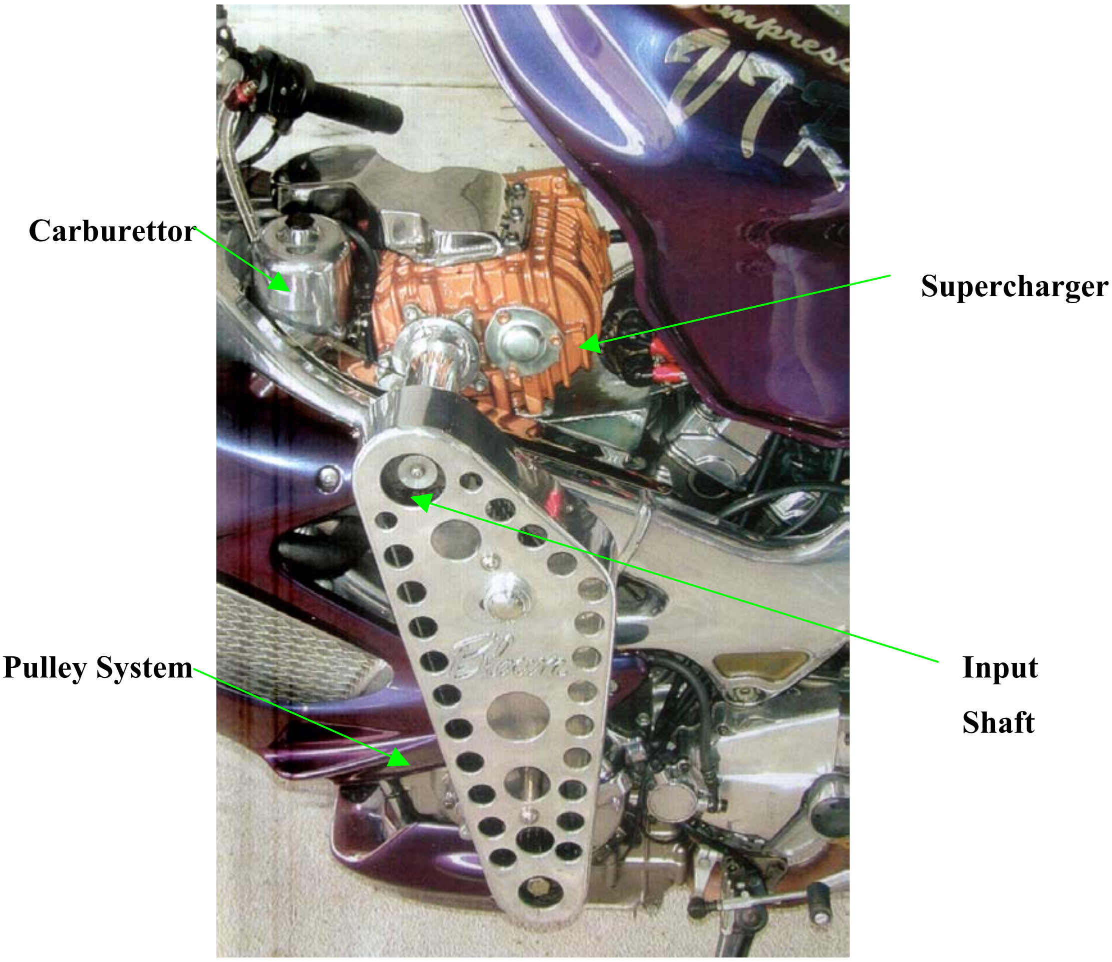
Forced air induction is a means of raising the inlet density of the air. The purpose of all forced air induction systems is to provide a pressure boost to the air inducted into the engine. The result of raising the pressure of the incoming air is a proportional increase in the density of the air (if the air temperature is kept constant). For example a naturally aspirated engine inducts air at atmospheric pressure (101.3kPa). If a forced air induction system is implemented and the air is compressed resulting in a 50% pressure increase the air would be boosted by 50.65kPa. Accordingly, the pressurised air is then also 50% denser, if the air temperature is held constant. Ignoring all the losses associated with forced air induction systems, if this air is then forced into the engine and fuel is added under stoichiometric conditions it would result in 50% increase in power output of the engine.

There are three general types of forced air induction systems including turbochargers, superchargers and ram-air systems. Ram-air systems do not increase the performance of the engine compared to superchargers and turbochargers and their effectiveness is largely dependent on the vehicle velocity (Haile, J. 2000, p15). Because the speeds that the Formula SAE car travels at are relatively low, ram-air systems will not be considered in this research project.

[6] For a detailed discussion of the other variables in equation 6.1 see appendix C.

6.3.1 Superchargers

A supercharger uses a pulley system that is attached to the engine crankshaft to drive the compressor, which pressurises the air (see figure 6.1). There are several different types of superchargers including the roots, vane, screw and G-Lader oscillating spiral displacer superchargers. Each of these systems displays different characteristics (Haile, J. 2000). However, this section will only summarise the general performance characteristics of superchargers and outline some general design considerations.



(*Source*: Limney, C. & J. 2002, p30)

Figure 6.1 A supercharged Honda VTR1000 motorcycle

The denser charge that results from the supercharger increases the brake mean effective pressure in the cylinder[7]. This is displayed in figure 6.2, where the supercharged engine has a greater vertical distance between the top and bottom

[7] For a full explanation of mean effective pressure see Appendix C.

curves. A notable feature of the supercharged engine is the relatively small increase in peak cylinder pressure, as can be seen from figure 6.2. This small pressure increase results in a relatively small increase in stress in the engine components.

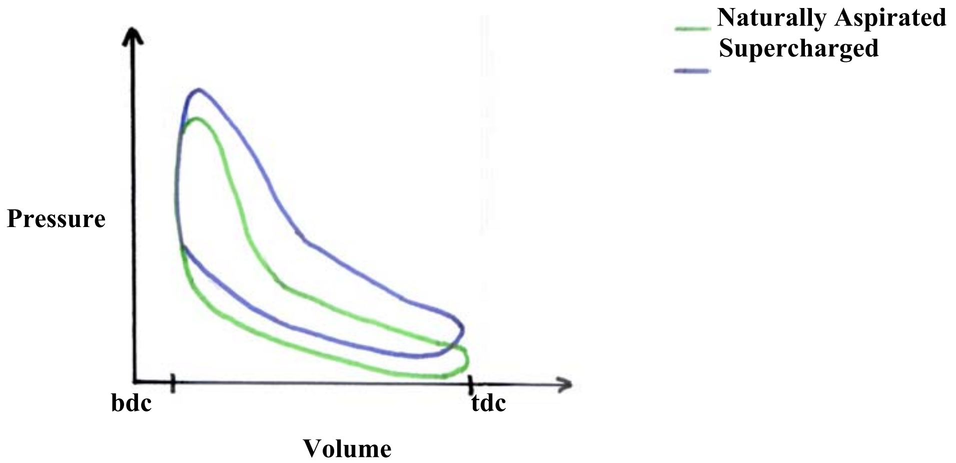
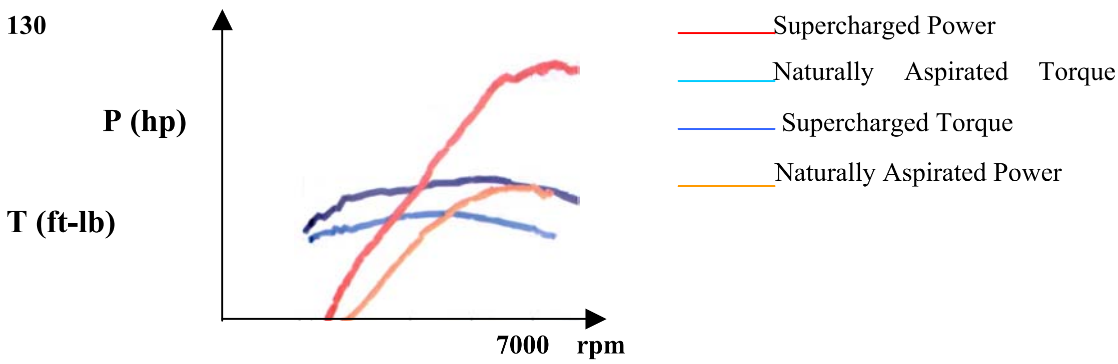


Figure 6.2 Comparison of cylinder pressure vs. volume - supercharged and naturally aspirated engines.

Another advantage of the supercharger is that it delivers boost all the way through the rev range because it is driven by the engine's crankshaft. This continuous boost through the rev range results in an increase in torque throughout the engine rev range, which is displayed in figure 6.3. Therefore a supercharged engine would deliver more acceleration throughout the rev range than a similar naturally aspirated engine.



(*Source*: Limney, C. & J. 2002, p30)

Figure 6.3 Comparative torque and power curves for a VTR1000 fitted with a supercharger.

Although the supercharger has many advantages it is also characterised by some inherent flaws. Some of the power increases that are theoretically obtainable are offset by the efficiency of the compressor and the mechanical losses that are induced by the attachment of the supercharger's pulley system to the engine's crankshaft. Also, it is generally accepted that mechanically driven superchargers are not as fuel efficient as naturally aspirated engines (Garrett, T.K., Steeds W. & Newton, K. 2001, p485).

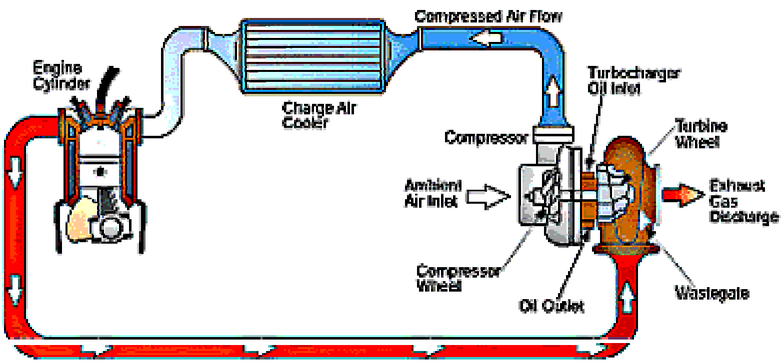
Superchargers have not been standard fitment on any mass-produced motorcycle since the 1930's (Walker, M. 2001). However, in the United States there are many companies that specialise in the production and development of superchargers for motorcycle engines. A new supercharger that would be suitable for the FZR600 engine costs $3109 (C.A.P.A Performance, 2004) and therefore beyond the means of the USQ Motorsport team.

Nevertheless it is possible to supercharge a motorcycle engine without such a large financial outlay. The supercharger that is depicted in figure 6.1 was purchased second-hand for $450. The supercharger was originally fitted to a small imported Japanese car, the Mazda MR2. The greatest cost that was involved in fitting the supercharger to this engine was the construction and purchase of the supercharger input shaft and the belt drives, which costed $700 and $300 respectively (Limney, C. & J. 2002, p30). Given the machining facilities available at the USQ, the cost to manufacture the parts required to fit a supercharger to the Formula SAE car would be minimal.

When selecting a supercharger for a motorcycle almost any small supercharger is suitable (Haile J. 2000, p33). Therefore a supercharger that was originally fitted to a small four-cylinder car would be suitable for the Formula SAE engine. Superchargers of this type are easily obtained from wreckers that specialise in imported Japanese cars.

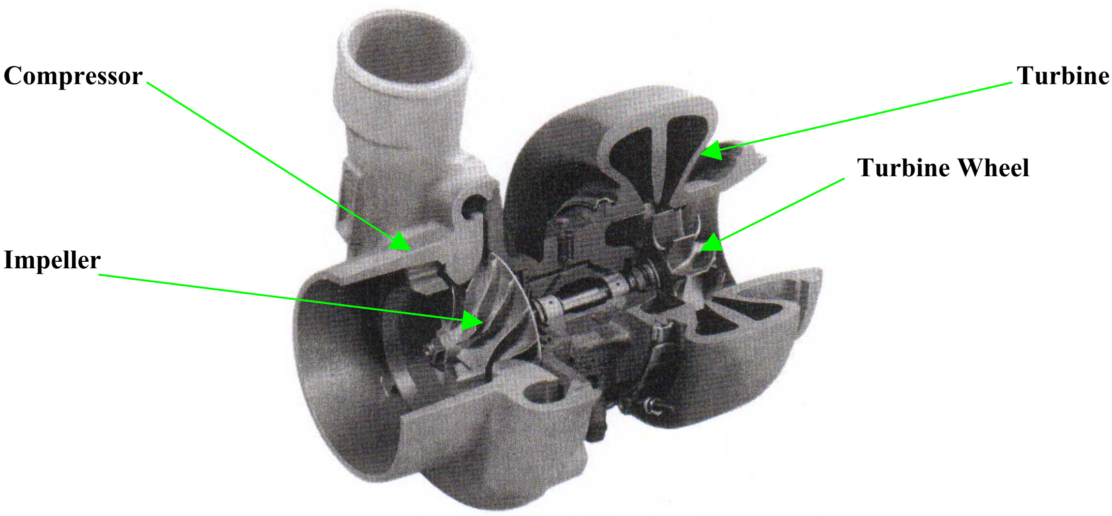
6.3.2 Turbochargers

Compared to a supercharger a turbocharger is smaller, lighter, less complicated, more durable and more efficient (Haile J. 2000, p41). The crankshaft does not drive the turbocharger; instead the turbo harnesses the energy of the outgoing exhaust gases (see figure 6.4) to drive its turbine and compressor wheel assembly (see figure 6.5). A great advantage of the turbocharger is that it actually raises the thermal efficiency of the engine. It has been reported that turbochargers can raise the thermal efficiency by up to 10%. This increase in thermal efficiency means that well designed turbocharged engines are actually more fuel-efficient than naturally aspirated engines (Garrett, T.K., Steeds W. & Newton, K. 2001, p486).



(*Source*: Turbotech, 2004).

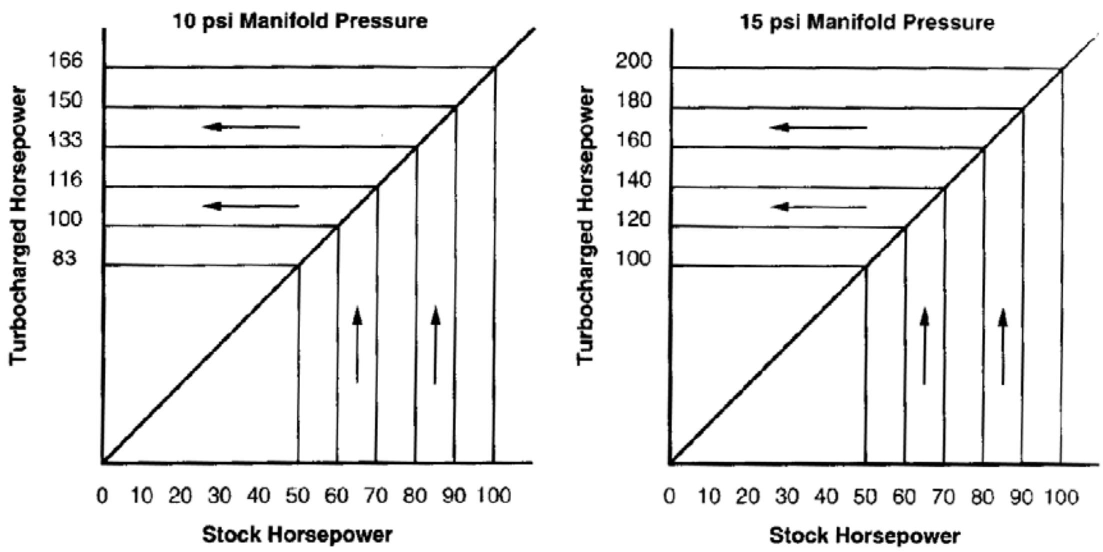
Figure 6.4 Gas flow in a turbocharged engine.



(*Source*: Japiske & Baines, 1994)

Figure 6.5 Cutaway photograph of a turbocharger.

Turbochargers, unlike superchargers, are all very similar in design. For this reason it is possible predict how much the maximum power output of the engine will increase, under a given boost pressure. Figure 6.5 shows the effect of different boost pressures on peak engine power output. So for example, if a standard Yamaha FZR600 engine with a maximum power output of 80hp were fitted with turbocharger delivering 10psi boost, the expected power output would be approximately 133hp, which is substantial.



(*Source*: Haile J. 2000, p44)

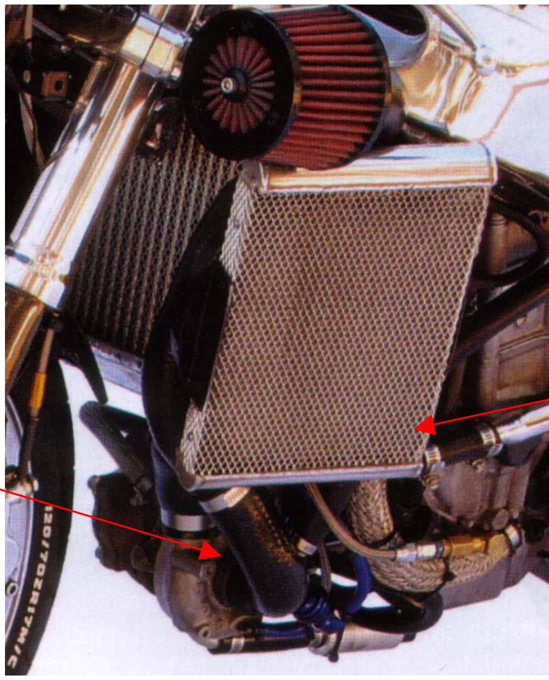
Figure 6.6 Estimated increase in maximum power output when a turbocharger is implemented.

However, the disadvantage of the turbocharger is that it causes an increase in backpressure in the exhaust manifold, which results in poorer flow in the cylinder head. Nevertheless, this loss of energy is not as a great as the energy lost due to the pulley system of a supercharger. Another disadvantage of the turbo is that it significantly increases peak cylinder pressure. For this reason, it is common practice to decrease the compression ratio of the engine. The decrease in compression ratio is proportional to the amount of boost administered to the engine. This reduction in compression ratio is disadvantageous, because at low engine speeds there is not enough energy released to run the compressor and therefore the engine will be sluggish compared to a naturally aspirated or supercharged engine. Another disadvantage is that when sudden acceleration is required, there is a small time delay commonly known as ‘lag’. This occurs because there is a time delay before the extra energy, discharged into the turbine-housing volute, can speed up the turbine wheel. In poorly designed systems lag can be as long as several seconds (Garrett, T.K., Steeds W. & Newton, K. 2001, p559). Lag could prove to be a serious disadvantage in the Formula SAE competition if a turbocharging system that was not designed correctly was utilised.

Because a turbocharger is relatively compact it is easy to fit to motorcycles, as can be seen from figure 6.7. In the 1980’s every major Japanese motorcycle manufacturer offered turbocharged engines in their model range. Some of the motorcycles that featured factory fitted turbochargers included the Kawasaki GPZ750, Honda CX650, Yamaha XJ750 and the Suzuki XN85. These motorcycles all produced over 100hp, which at the time was staggering. However, these motorcycles failed to find a market niche and the last mass-produced factory turbocharged motorcycle was produced in 1984 (Walker, M. 2001).

Intercooler

Turbocharger



(*Source*: Ware, J. 2004, p40)

Figure 6.7 Kawasaki ZXR750 fitted with aftermarket turbocharger and intercooler.

However, turbocharging remains popular in the performance industry. Today turbocharging kits can be purchased from a wide range of manufacturers. These kits provide all of the components that are required to essentially 'bolt on' a turbocharger. The total cost of the tubocharging kit that was fitted to the engine in figure 6.7 was $4000 (Ware, J. 2004, p41). Unfortunately, the rules of Formula SAE stipulate that the competitors must design the turbocharging and supercharging systems.

The selection of turbocharger is not as simple as the selection of a supercharger. The choice of turbocharger is dependent on the engine capacity, type (2-stroke or 4-stroke), maximum engine speed and maximum turbo boost (Haile, J. 200 pp42-45). The cost of a new turbocharger that is suitable for the FZR600 is $1983 (Turbotech, 2004). If a second-hand turbocharger were to be used the cost would obviously be a lot less. However, the suitability of each unit must be determined on case-by-case basis.

6.3.3 Additional Costs of Forced Air Induction

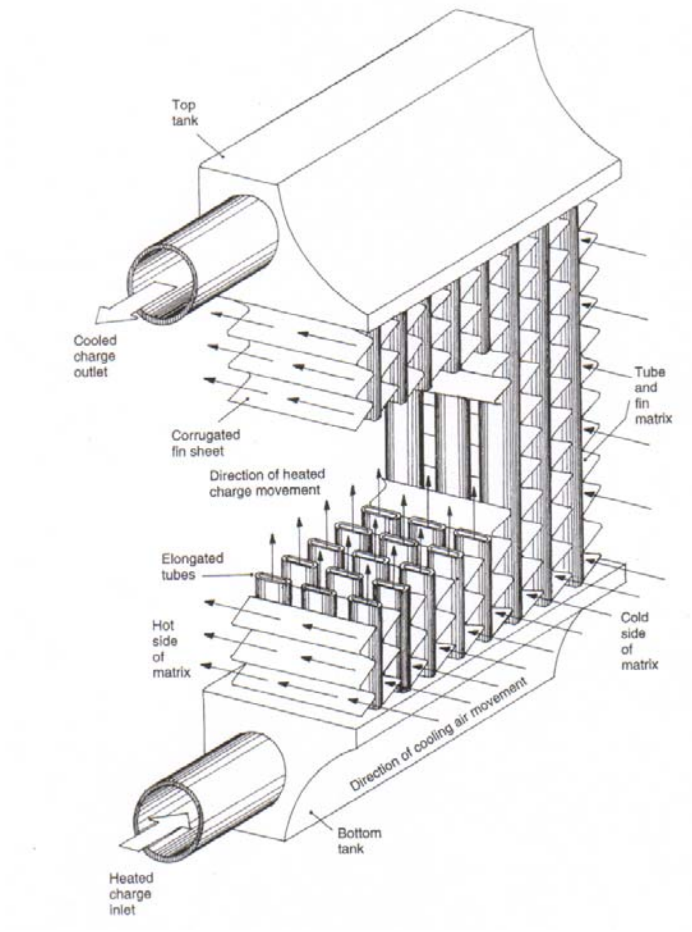
So far in this paper the only the direct costs that are associated to forced air induction systems have been discussed. In *general*, a supercharger can be fitted to a motorcycle engine without any major modifications to the engine if boost pressures are kept below 10 psi (Haile, J. 2000). If boost pressures above 10psi are used the engine will need to be strengthened, which will introduce more cost to the car.

As mentioned in section 6.3.2, in order to use a turbocharger requires the designer to lower engine's compression ratio. As with the supercharger, it can be concluded that engine strengthening would also be required if the boost exceeds a certain point. This point would largely be dependent on the compression ratio that is used.

6.3.4 Intercoolers

In the preceding sections it was assumed that the temperature of the air would remain constant as the air pressure rises. But a rise in pressure will result in a temperature increase as well as a density increase. Therefore to maximise the gains of the supercharger or turbocharger an intercooler is often incorporated to keep the temperature of the air constant. The intercooler is a heat exchanger that is fitted between the forced air induction system and the engine as depicted in figure 6.4. The

Formula SAE-A rules (2004, p48) state that only an air-to-air heat exchanger may be used. Figure 6.8 shows the construction and the airflows of a typical air-to-air intercooler. The cost of an intercooler suitable for the Formula SAE car is $795 (Turbotech, 2004).



(*Source*: Heisler, H . 1995, 351)

Figure 6.8 A typical air-to-air intercooler.

6.3.5 Forced Air Induction for the Formula SAE Engine

Given the financial constraints of this project it was concluded that forced air induction was unfeasible for this year's USQ Motorsport team. However, from a performance perspective a forced air induction system would definitely be worth pursuing. Furthermore, Haile (2000, p123) notes that the FZR600 engine is an excellent candidate for a forced air induction system.

6.4 Fuel Mixture Preparation System Design

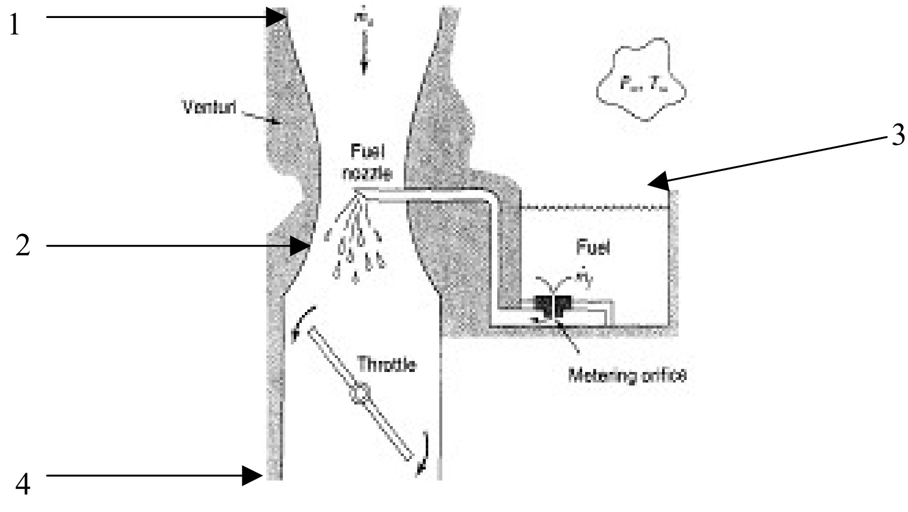
The role of the fuel mixture preparation is essentially to mix the fuel and air proportionally. Motorcycle engines generally incorporate two different systems to achieve this, carburation or multi-point fuel injection. The FZR600 engine was originally equipped with a bank of four carburettors, which are depicted in figure 6.9. Each carburettor 'feeds' one engine cylinder. Because a *single* restrictor has to be placed between the throttle and the engine, the use of multiple carburettors does not facilitate an efficient or effective design. Therefore the original system cannot be used for the Formula SAE car and the fuel mixture preparation system and induction manifold must be redesigned.

Figure 6.9 The original Yamaha FZR600 carburettor assembly.

6.4.1 Carburation

Carburation is the conventional method of fuel mixture preparation. Carburettors control the flow rate of the fuel so that it is proportional to the air flow rate. Carburettors also serve to mix the fuel into an atomised state to enable complete combustion.

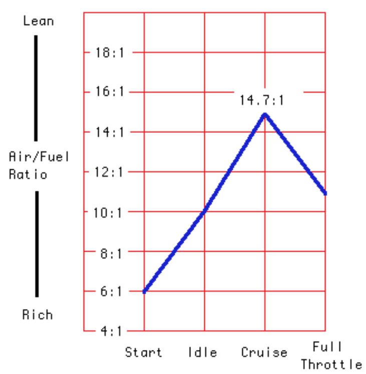
The basic principle of carburation is shown in figure 6.10. Carburettors depend on the air speed being greater than the fuel speed at the fuel nozzle to atomise the fuel. The inlet air flows through a venturi nozzle. The pressure difference between the carburettor inlet and nozzle throat is used to meter the fuel and achieve the correct air/fuel ratio. Engine speed and throttle position determine the mass flow between (1) and (4). The pressure at (2) and the air fuel ratio adjust to match the mass flow rate that the engine is demanding.



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001)

Figure 6.10 Basic principle of carburation.

However, the required fuel/air ratio varies under different operating conditions, which is shown in figure 6.11. To compensate for the different mixture requirements modern carburettors employ a circuit for each condition.



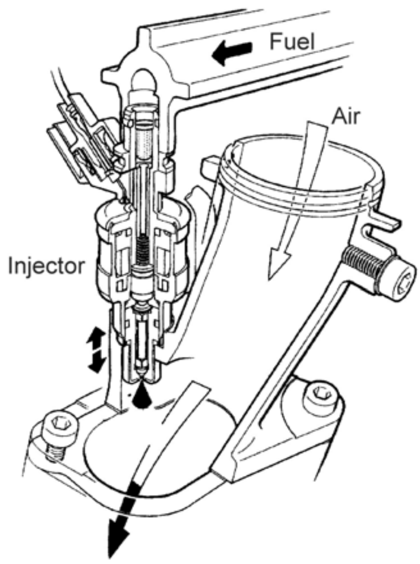
(*Source*: Howstuffworks)

Figure 6.11 Air/fuel ratio under different operating conditions.

Unfortunately, in multi-cylinder engines that are equipped with one carburettor the fuel/air ratio varies greatly between cylinders. In some engines variations of the fuel/air ratio between cylinders can be as high as 30% (Heywood, J. B. 1988). To compensate for these variations it is common practice to tune the carburettor so that the mixture in the leanest cylinder is rich enough to run reliably (Lumley, J. L. 1999). This situation is far from optimum and is detrimental to the performance of the engine. However, the use of carburation on the Formula SAE car does have some advantages including simplicity, ease of tuning and cost.

6.4.2 Electronic Fuel Injection

Motorcycles that employ fuel injection typically use a multipoint system. Figure 6.12 shows the layout of a typical multipoint fuel injector. Multi-point injection systems incorporate an injector for each cylinder. Therefore, multipoint fuel injection overcomes the mixture distribution problem that occurs when a single carburettor is incorporated on a multi-cylinder engine. It has been estimated that the output power of a multi-cylinder engine can be increased by up to 15% when fuel injection is used instead of carburation (Haile, J. 2000, p56).

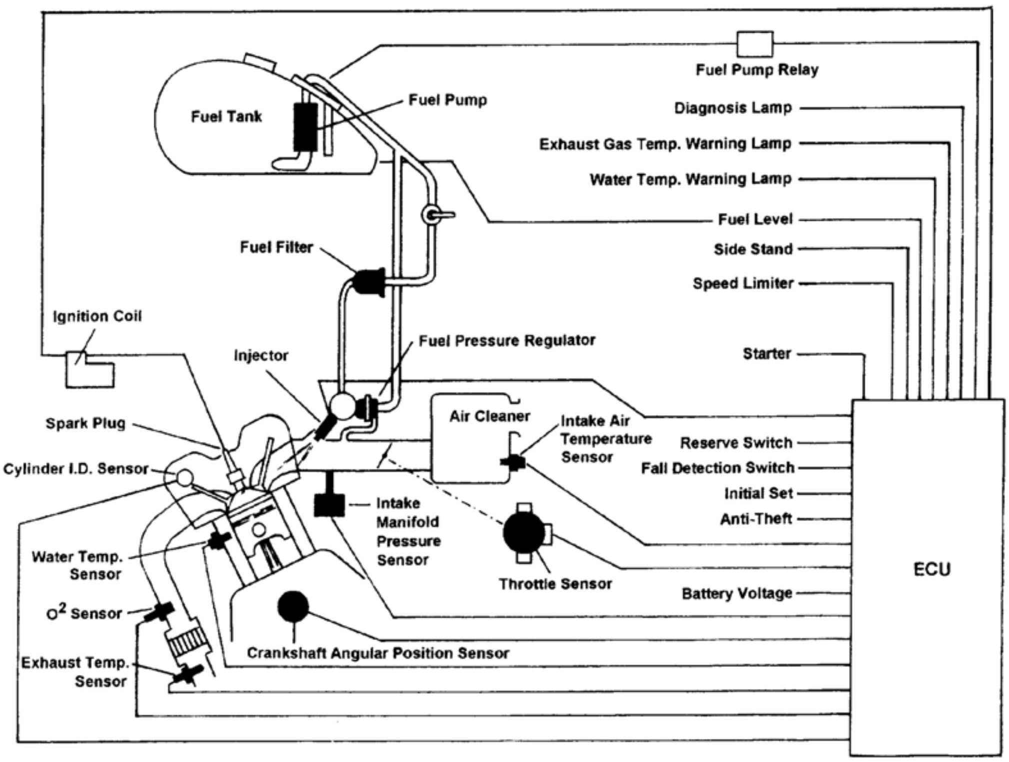


(*Source*: Haile, J. 2000)

Figure 6.12 Cutaway diagram of a typical multipoint fuel injector.

6.4.3 Electronic Fuel Injection Operating Principle

Figure 6.13 shows the layout of a typical motorcycle fuel injection system. The Engine Control Unit (ECU) processes signals from the sensors and signals the injectors how long to stay open. The duration that the injectors stay open determines the richness of the fuel/air mixture. The ECU engine 'map' is specific to a particular engine and contains the predetermined injector duration for a given input from the electronic sensors. The ECU also controls the spark ignition timing. Because the function of the ECU is to adjust for changes in atmospheric conditions it cannot sense changes within the engine's design. As the Formula SAE engine has a unique induction system a custom ECU map would have to be written. In order to write an ECU map the programmer need would to have continuous access to a dynamometer, an exhaust lambda meter and the relevant software (Squelsh, O. 2003, pers. comm. Oct. 25).



(*Source*: Haile, J. 2000, p80)

Figure 6.13 Schematic of a typical motorcycle multi-point fuel injection system.

6.4.4 Cost of Electronic Fuel Injection

The FZR600 model ran from 1987 until 1995 in Australia. The FZR600 was never equipped with electronic fuel injection at any stage in Australia or elsewhere (Ahlstrand, A. & Haynes, J. 1998). For this reason it is not possible to obtain an injection system that was specifically designed for the FZR600. The cost of a new multi-point fuel injection for the FZR600 is $3229 (Serco, 2004) and is therefore beyond the budget constraints of the USQ Motorsport team.

It *may* be possible to adapt a second-hand fuel injection system that was used on a later model four-cylinder motorcycle engine to the FZR600 engine. Some of the four-cylinder motorcycle engines that utilise multi-point electronic fuel injection include the Honda CBR600, Yamaha YZF-R6 and the Suzuki GSXR600 (Bikepoint, 2004). However, there are no modern four-cylinder motorcycle engines that were originally fitted with electronic fuel injection systems that incorporate an ECU that can be reprogrammed (Bikepoint, 2004). Therefore if a second-hand EFI system were purchased a new programmable ECU would have to be acquired. The cost of buying a new ECU that would be suitable is $995 (Serco, 2004). The cost of the ECU alone deemed the implementation of a used fuel injection system beyond the reach of the 2004 USQ Motorsport team. Hence the feasibility of fitting a used fuel injection system to the Yamaha FZR600 engine was not pursued any further.

6.4.5 Fuel Mixture Preparation System Selection

Given the performance advantages of electronic fuel injection that were outlined section 6.4.2, fuel injection is obviously the most desirable choice for the Formula SAE engine. Unfortunately, the cost of electronic fuel injection confined the fuel mixture preparation system to carburation.

6.5 Carburettor Selection

There are many different brands and models of carburettor to choose from. In order to choose a carburettor for the Formula SAE car some general selection criteria was defined. The selection criteria included capacity requirements and the carburettor manufacturer's association with the performance industry.

6.5.1 Performance Associations

When selecting a carburettor for a unique application such as the Formula SAE car it is ideal to select a carburettor brand that is associated to performance applications for various reasons. Firstly, the carburettor must be flexible in terms of tuning. In order for a carburettor to be easily tuned it must have a wide range of parts available for it. For example, there must be wide range of jets available. A wide selection of parts is generally only available from performance carburettor manufacturers. Another advantage of using a performance carburettor is that the quality of performance carburettors is generally high. In addition, if a performance carburettor is used it is easier to locate a technician with the ability to calibrate the carburettor.

6.5.2 Carburettor Airflow Capacity

The maximum airflow capacity of the carburettor must be matched to the maximum airflow capacity of the engine. In order to find the correct carburettor size some basic calculations were performed to determine the maximum air flow capacity of the engine. The airflow capacity for a given engine is calculated with equation 6.2:

$$\dot{V} = \frac{V_d \quad n \quad \eta_v}{3456} \qquad \textbf{(Eq. 6.2)}$$

where $\dot{V}$ is the volume flow rate of the engine in cfm (cubic feet per minute);
V_d is the engine displacement in cubic inches;
n is the engine speed at peak power; and
η_v is the volumetric efficiency of the engine.

The FZR 600 engine produces peak power at 10500rpm (see appendix D for the power curve) and the displacement of the engine in cubic inches is 36.57. Assuming a volumetric efficiency of 0.9 at peak engine speed and substituting values into equation 6.2 yields:

$$\dot{V}\,(FZR600) = \frac{36.57 \times 10500 \times 0.9}{3456} = 99.9\text{cfm}$$

But the airflow rating is dependent on the manifold vacuum pressure. For a race engine typical values of manifold vacuum pressure are around 0.5 inches Hg (abs) (Bell, G. 1998, p63). For single and double barrel carburettors it is standard to measure the airflow rating using a manifold vacuum pressure of 3 inches Hg (abs) (Bell, G. 1998, p63). To convert the airflow rating for the FZR600 to the standard rating, equation 6.3 is used:

$$\dot{V}_1 = \dot{V}_2 \sqrt{\frac{p_1}{p_2}} \qquad \textbf{(Eq. 6.3)}$$

where p_1 is the standard manifold pressure carburettors are rated at;
p_2 is the manifold pressure at engine peak power;
$\dot{V}_1$ is the airflow rating at p_1; and
$\dot{V}_2$ is the airflow rating at p_2.

Assuming the FZR600 has manifold pressure of 0.5 inches Hg (abs) at peak engine speed, p_2 is 0.5 inches (abs) Hg and p_1 is 3 inches Hg (abs) and substituting values into equation 6.2 yields:

$$\dot{V}(3\,inches\,Hg) = 99.9\sqrt{\frac{3}{0.5}} = 121\text{cfm}$$

Hence, the FZR600 engine requires a carburettor, which is rated at approximately 121 cfm @ 3 inches Hg (abs).

6.5.3 Carburettor Options for the Formula SAE car

The carburettor is restricted to a few manufacturers viz. Weber, SU or Mikuni carburettors. Although there are many other high quality carburettors, the aforementioned carburettors are easily obtainable in Australia (Bell, G. 1998). The Weber carburettor is manufactured in Italy and has the reputation as one of the finest carburettors available on the market. Each carburettor is quality assured and in the past were standard fitment to many Ferraris (Aird, F. & Elston, M. 1997, et. al.). The SU is a British made carburettor, which also has a strong racing heritage. In the past the SU has featured on Minis and Jaguars. Unfortunately the SU does have a reputation as being ‘finicky’ and hard to tune. Mikuni carburettors are commonplace in the motorcycle industry. Today many new motorcycles still incorporate Mikuni carburettors (Mikuni Australia, 2004). Four 32mm Mikuni carburettors were originally fitted to the FZR600 engine.

There are two general classes of carburettor. These classes include variable venturi and fixed venturi carburettors. The Weber carburettor is a fixed venturi carburettor, whereas the SU and Mikuni carburettors are of the variable venturi variety.

6.5.4 Weber Fixed Venturi Carburettors

The fixed venturi carburettor delivers stoichiometric mixture to the engine across a full range of operating conditions through the use of six systems. These systems include:

1. Float system
2. Idle and progression
3. Accelerator pump
4. Power system
5. Cold starting device
6. Main metering device

These circuits are typically made up of many components and therefore the fixed venturi carburettor is quite complex in comparison to variable choke carburettors (Heisler, H.1995). Figure 6.14 shows one of the simplest fixed venturi carburettors available. From inspection of figure 6.14 it can be seen that this carburettor is still quite complex. The advantage of Weber carburettors, as opposed to many other fixed venturi carburettors, is that all of the circuits are removable and interchangeable. A wide range of variants of each circuit is also available (Weber North America, 2004). This allows a great amount of flexibility when tuning the carburettor.

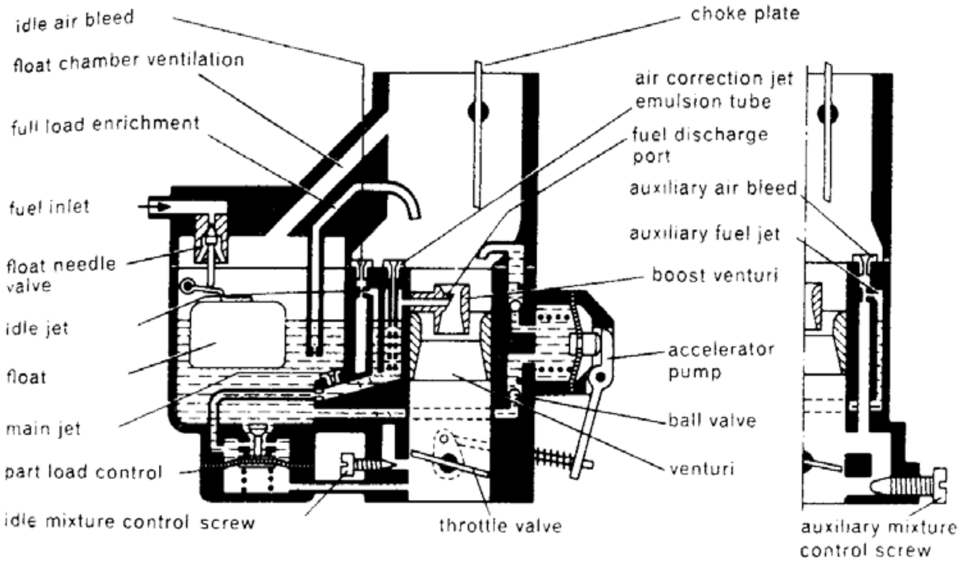


Figure 6.14 Cross-section of a single barrel fixed venturi carburettor.

The smallest carburettor that Weber currently produces is the 34 ICT, which is depicted in figure 6.15. The Weber 34 ICT carburettor has a throat size of 34 mm and a maximum airflow capacity of 126cfm @ 3 inches Hg (Weber Queensland, 2004, pers. comm. 12 Aug.). This makes the 34 ICT the ideal capacity for the Yamaha FZR600 engine. The initial cost of a new Weber 34 ICT is $355 (Weber Queensland, 2004, pers. comm. 12 Aug). However, additional costs would be encountered if the circuits that were originally fitted to the carburettor had to be replaced when the carburettor was being calibrated for the FZR600 engine.

(*Source*: Weber North America)

Figure 6.15 Weber 34 ICT single barrel carburettor.

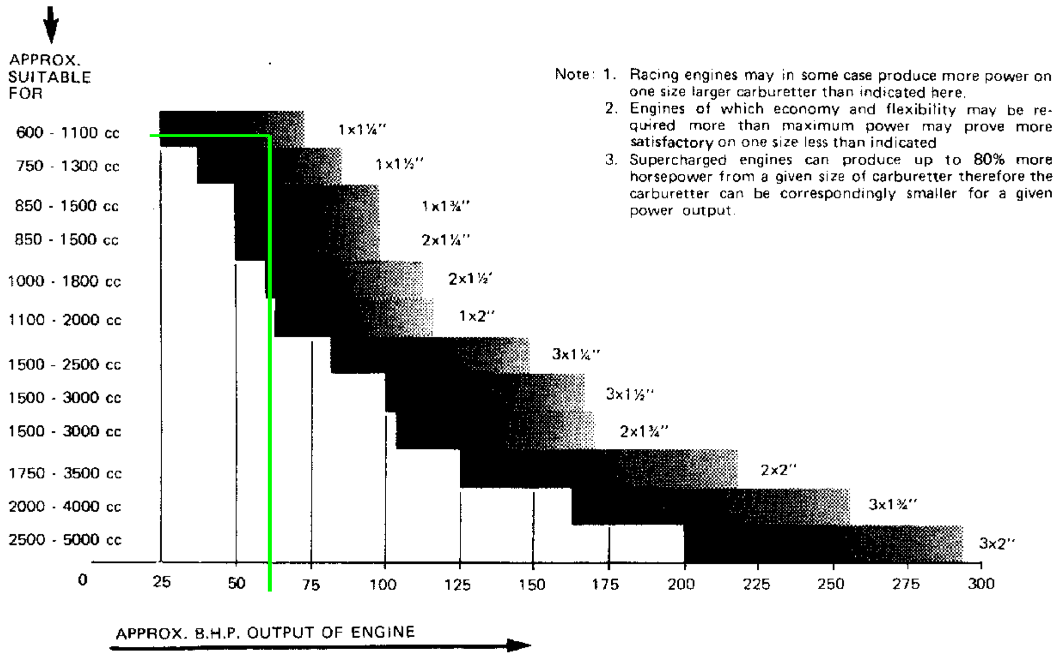
6.5.5 Variable Choke Carburettors

As stated in section 6.5.4, in comparison to the fixed venturi carburettor, variable venturi carburettors are very simple. Usually, the only components that need to be replaced for tuning purposes are the jets and needles (Factorypro, 2004). This amounts to a lot less capital expenditure, which is of high priority in this project.

Variable venturi carburettors can be categorised into two sub-groups. These groups consist of slide carburettors and constant velocity carburettors. As the Mikuni and SU carburettors are both of the constant velocity type, this paper will focus solely on constant velocity carburettors. The main difference between the SU and the Mikuni is the direction of flow through the carburettor. The SU is a side-draft carburettor and the Mikuni is a downdraft carburettor. The greatest disadvantage of using an SU carburettor for this project is that SU carburettors are no longer produced. However a large selection of needles and jets are still available (Mini Mania, 2004). It was also found that a wide variety of jets and needles for Mikuni carburettors are also available (Toowoomba Yamaha, 2004, pers. comm. 13 Aug).

To find the correct size SU carburettor the flow rating for various carburettors was found. The 1¼ inch (throat size) is the smallest carburettor made by SU and has airflow capacity of 113cfm @ 3 inches Hg (Mini Mania, 2004). SU has published graphs that help in selecting carburettors for a given application. From figure 6.16 it can be seen that the carburettor selection chart agrees with the previous calculations.

Unfortunately, data detailing the airflow capacity of Mikuni carburettors was not available from any reliable sources. The throat size of the 32mm Mikuni is almost identical to the throat size of the SU carburettor, which is 31.75mm. Because the operating principle of these two carburettors is identical it was concluded that a single 32 mm Mikuni would be sufficient. Although both the SU and the Weber carburettors may have made excellent choices it was decided that a single 32mm Mikuni would be tested initially. The reason for this was purely financial. As the bike was already equipped with four 32mm Mikuni carburettors the financial outlay to set-up the carburettor would be minimal. Figure 6.17 depicts the carburettor that was fitted to the Formula SAE engine.



Note: The estimated output power of the FZR600 engine with a restrictor is 80% of the power output of the engine without the restrictor. The maximum power output of the FZR engine is 80bhp (Larry's FZR Homepage, 2004). Therefore the maximum output power of the restricted FZR engine ≈ 80 x 0.8 = 64bhp. With this power output and a displacement of 600cc the required carburettor size is 1 ¼ inches, as shown above.

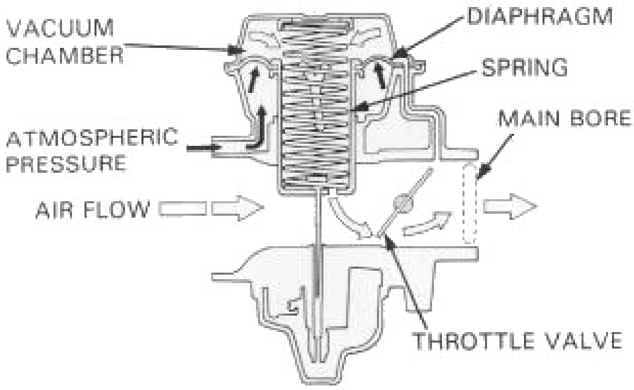
(Source: Wade, G. R. 1994, p34)

Figure 6.16 SU carburettor sizing chart.

Figure 6.17 Mikuni 32 mm carburettor

6.6 Constant Velocity Carburettor Operating Principle

A cross-sectional view of a CV carburettor is shown in figure 6.18. Although figure 6.18 displays a CV side-draft carburettor the principle of operation of the CV downdraft carburettor is the same, but the geometry of the carburettor is different. A CV carburettor is termed ‘constant velocity’ because as engine demand increases the venturi becomes larger and the velocity of the air through the venturi remains constant.

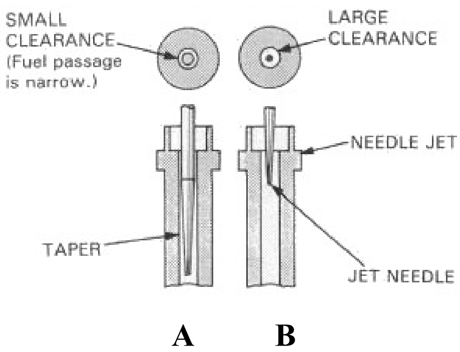


(*Source*: Bowdler, J. 2004)

Figure 6.18 Cross section of a side-draft constant velocity carburettor.

From figure 6.18 it can be seen that the vacuum piston resides in a bore that intersects the main bore of the carburettor. When the engine is idling the vacuum piston is pushed most of the way down and the spring and the main bore is almost completely obstructed. When the throttle valve is opened, the airflow in the main bore exerts a negative pressure on the lower section of the vacuum piston. At this point, air is drawn from the vacuum chamber through a hole in the bottom of the piston, overcoming the spring pressure and causing the piston to rise.

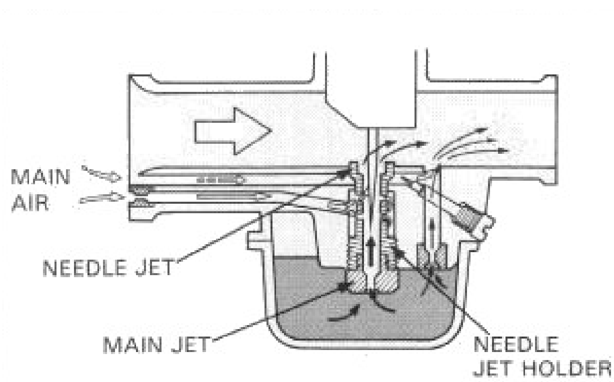
The vacuum piston carries the jet needle, which fits into the needle jet. The needle jet opens into the main bore of the carburettor and allows the fuel into the intake manifold by means of the negative pressure formed by the intake air rushing through the venturi. Jet needles are generally straight for approximately1/3 of their length and the rest is tapered. At idle and low speeds, the vacuum piston is nearly all the way down, pushing the needle into the needle jet most of the way. In this position, the straight section fills most of the space inside the needle jet tube. Thus the fuel flow is restricted to a narrow annular space around the needle (see figure 6.19A). As the piston rises with increased engine speed, the jet needle is withdrawn from the needle jet. Because the needle is tapered, the annular space through which the fuel can travel increases, allowing more fuel to match the increased airflow (see figure 6.19B).



(*Source*: Bowdler, J. 2004)

Figure 6.19 Needle jet and jet needle.

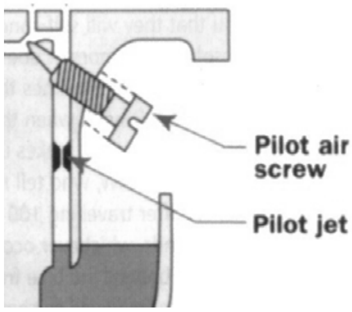
The float chamber of a constant velocity carburettor is depicted in figure 6.20. In the float chamber, the main jet controls the amount of fuel that passes through the needle jet.



(*Source*: Bowdler, J. 2004)

Figure 6.20 CV carburettor float chamber.

The pilot circuit is located on the engine side of the vacuum piston and is the main source of fuel from idle up until approximately 1/8 throttle. This pilot circuit includes the pilot jet and the pilot screw adjustment and is shown in figure 6.21.

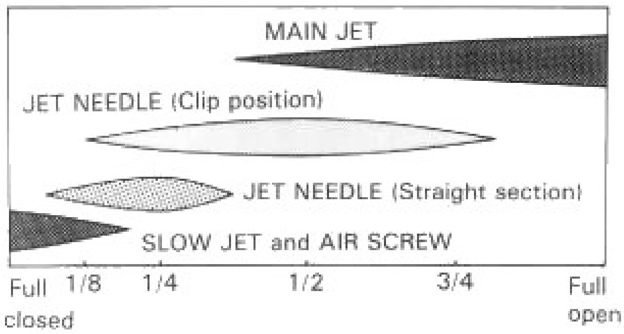


(*Source*: Bowdler, J. 2004)

Figure 6.21 CV carburettor pilot circuit.

6.7 Constant Velocity Carburettor Tuning

Figure 6.22 displays which circuit of the CV carburettor is responsible for the fuel/air ratio at a given throttle position. The height of the ‘bubbles’ in figure 6.22 is indicative of the importance of a particular circuit at given throttle position (i.e. a larger height indicates more importance). The following subsections outline what the effect of modifying each of the circuits has on the fuel/air ratio.



(*Source*: Bowdler, J. 2004)

Figure 6.22 CV carburettor circuit versus throttle location.

6.7.1 Main Jet Modification and Tuning

By enlarging the diameter of the main jet the fuel/air ratio becomes richer as more fuel flows through it. Conversely, by using a smaller main jet diameter the fuel/air ratio becomes leaner. The main jets have a standardised numbering system. For example, a jet designated 120 will have a hole diameter of 1.2mm

6.7.2 Jet Needle

Figure 6.23 shows a diagrammatic representation of the jet needle. The specification of each of the parameters of the jet needle plays an important role in determining the fuel/air ratio at different throttle positions. The effect of each parameter is as follows:

Diameter of the straight section – By using a thinner jet needle, there is a larger area between the jet needle and the needle jet. Therefore the mixture becomes richer if a thinner needle jet is used.

Length of the straight section - Determines the point that the needle taper will start relative to the clip position.

Needle Clip Position – The position of the needle clip works in conjunction with the length of the straight section. If the fuel/air mixture is too rich above 1/4 throttle, raising the needle clip will lean the mixture.

Needle Taper - A larger taper will result in a leaner mixture in the first half of the taper and a richer mixture in the last half of the needle.

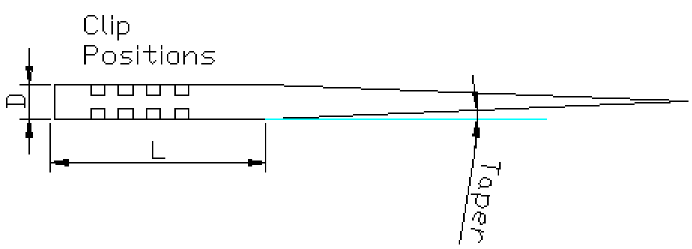


Figure 6.23 Jet needle features.

6.7.3 Idle Circuit Adjustment

If a richer air/fuel mixture is required at low engine speeds and idle, a larger pilot jet is used or the pilot airscrew is turned in. Conversely, if a leaner mixture is required a smaller pilot jet is used or the pilot airscrew is turned out.

6.8 Formula SAE Carburettor Tuning and Testing

By correctly calibrating each of the carburettor's circuits the correct fuel/air ratio can be achieved for all throttle positions. To effectively calibrate the carburettor, requires the use of a dynamometer in conjunction with a lambda meter[8]. The dynamometer is used to apply the load, which simulates driving conditions. The dynamometer is also used to measure the torque output of the engine and allows the tuner to analyse the effects of any modifications to the carburettor. The lambda meter is used to analyse the exhaust gases to determine if the correct fuel/air ratio is being obtained. It is also common practise for the final calibration of the carburettor to be performed after test-driving the vehicle (Wolferden, T. 2004, pers. comm. 12 Jun.).

Sponsorship was obtained, to allow the utilisation of dynamometer and a lambda meter for the tuning of the carburettor for the Formula SAE engine. However, due to the reasons outlined in section 3.10 it was not possible to tune the carburettor using any of the aforementioned tools.

Instead, the carburettor was mounted on the prototype intake manifold and was essentially 'tuned by ear'. John Armstrong, an experienced mechanic, carried out the tuning. The idle screw was adjusted so that the engine, 'sounded right' at idle. From listening to the exhaust note when the throttle was opened, John felt that the air/fuel mixture was lean across all throttle positions. Therefore the jet needle was adjusted to a lower position and the main jet was drilled out to 1.2mm (the original main jet hole diameter was 1.075mm). John cited that the engine *seemed* to run 'alright' but would require the use of the proper equipment to ensure that the correct fuel/air ratio was being delivered to the engine. John also placed his hand over the exhaust pipe and

[8] For a detailed description of the process that is used to tune a CV carburettor on a dynamometer see www.factorypro.com.

noted that the pulses of exhaust gases that were hitting his hand were even. This indicated that the engine was not misfiring. However very little about the carburettor or the fuel/air mixture can be concluded from these tests.

6.9 Intake Manifold Design

By the simplest definition, an intake manifold is a series of passages that connects the engine cylinders to a source of air/fuel mixture. Generally the manifold consists of a series of ducts, which extent from the engine to a common volume. These ducts are known as intake runners. The common volume that the runners are attached to is known as the plenum volume. Generally, the carburettor is placed above the plenum volume. The configuration of a typical intake manifold for a four-cylinder engine is shown in figure 6.24.

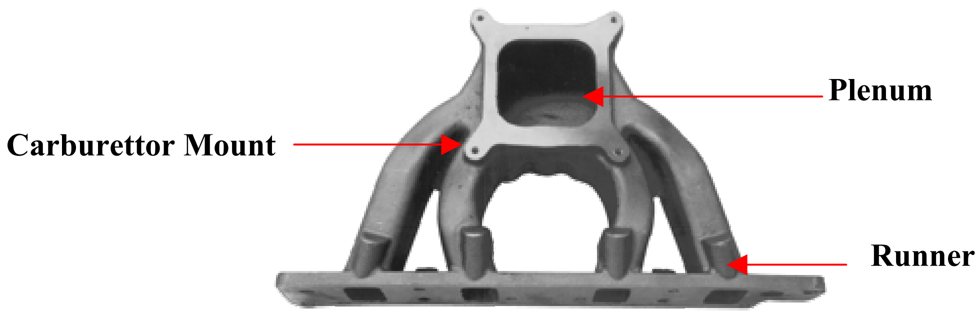


Figure 6.24 A typical four-cylinder intake manifold.

The flow regime in the intake manifold is classified as a compressible, viscid, unsteady, turbulent flow. The flow in the intake manifold also experiences some heat transfer. The equations that accurately describe the behaviour of such flow regimes are the unsteady mass, momentum, and energy conservation equations (Ferguson, C.R. & Kirkpatrick A.T. 2001, p179). Given the characteristics of the flow in an intake manifold, these equations are very difficult to solve without the aid of software. Hence, intake manifolds are generally designed with the aid of computational fluid dynamics software such as Fluent (Lumley, D. 1998, et. al.). But, due to time restraints the intake manifold was designed using empirical estimations and advice from experienced engine builders.

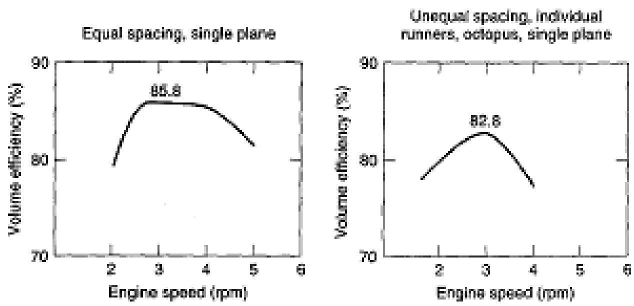
6.9.1 Volumetric Efficiency

The flow regime in the induction system significantly affects the cylinder filling capacity in two ways. Firstly, the air actually experiences a pressure drop as it travels through the inlet manifold due to viscous forces. From the ideal gas equation it can be seen that this incurs a loss in density of the charge:

$$p = \rho R T \qquad \textbf{(Eq. 6.4)}$$

where p is the pressure of the gas;
R is the gas constant;
T is the temperature of the gas; and
ρ is the is the density of the gas.

The second major affect of the flow regime is the choking that occurs in the inlet valves at high speeds due to shock waves. Choking means that the mass flow rate of the mixture cannot be increased and therefore the density of mixture in the cylinders is decreased. The volumetric efficiency takes into account these inefficiencies and is defined as the ratio of the actual mass flow rate that could be achieved with an inviscid, incompressible fluid to the flow rate that is obtained. Figure 6.25 shows, that the volumetric efficiency of the engine is largely dependent on the intake manifold design.



(*Source*: Ferguson, C.R. & Kirkpatrick A.T. 2001, p6)

Figure 6.25 The effect of intake manifold design on volumetric efficiency.

6.10 Intake Manifold Design Criteria

In order to obtain the highest possible volumetric efficiency in *each* cylinder an intake manifold should, if possible:

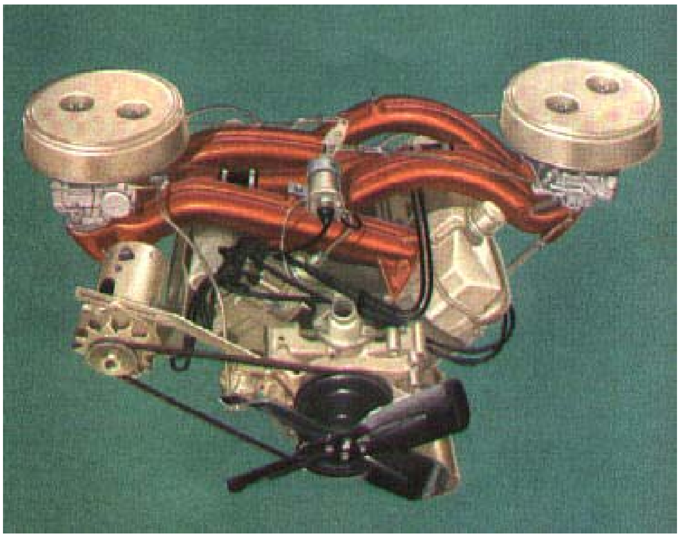
1. Provide means to prevent inter-cylinder robbery of charge;
2. Impede flow as little as possible; and
3. Be 'tuned' to increase the performance of the engine.

In addition to the previous requirements the intake manifold for the USQ Motorsport Formula SAE car, must:

1. Be cost effective;
2. Be easily attached to the existing intake manifolds;
3. Be easily manufactured, preferably utilising the facilities available at USQ; and
4. Be light in weight.

6.10.1 Intake Manifold Tuning

Tuned intake manifolds significantly increase the volumetric efficiency at high engine speeds (Ferguson, C.R. & Kirkpatrick A.T. 2001, et. al.). For this reason, tuned manifolds are commonplace on Formula SAE-A engines. In the past tuned manifolds have also been utilised by manufacturers, such as Dodge, on their performance engines. The runners that are incorporated in tuned manifolds are typically very long in comparison to the runner lengths that characterise conventional manifolds. The extreme length of tuned intake runners can be observed from figure 6.26.



(*Source*: Unknown)

Figure 6.26 1962 Dodge ram manifold.

Tuned manifolds take advantage of the compression and rarefaction pressure waves that propagate through the intake flow due to the opening and closing of the intake valves. When the inlet valve opens, the reduction in cylinder pressure produces a negative pressure wave. This pressure wave travels, at the speed of sound, through the column of gas that exists between the inlet valve and the end of the runner. When the pressure wave pulse reaches the plenum chamber, the gas at the runner entrance decreases in density and creates a pressure depression. Due to this, the surrounding gas flows rapidly in to fill the depression. This rushing effect results in a reflected positive pressure wave due to the inertia of the incoming gas. This positive pressure pulse travels back to the inlet valve port. These pressure pulses travel backwards and forwards in the intake manifold, loosing energy each time, until the intake valve closes. A tuned manifold takes advantage of this effect by incorporating runners that allow the reflected pressure waves to effectively 'ram' the charge into the cylinders in the later stages of the induction period (Heisler, H. 1998, p246).

One of the most important design parameters of a tuned manifold is the length of the runner. In order to estimate the required runner length for intake manifold tuning, it was recommended that the following equations be used (Seng T. 2004, pers. comm. 19 Jun.):

$$L_2 = \frac{132000}{n} \qquad \textbf{(Eq. 6.4a)}$$

$$L_3 = \frac{97000}{n} \qquad \textbf{(Eq. 6.4b)}$$

$$L_4 = \frac{74000}{n} \qquad \textbf{(Eq. 6.4c)}$$

where L_2, L_3, are L_4 are the pipe lengths in inches that take advantage of the ramming effect of the 2^{nd} , 3^{rd} , 4^{th} reflected pressure waves respectively. n is the engine speed. The calculated runner length is the distance between the intake valve seat and the plenum chamber. Seng (2004) cited that manifold designers generally do not attempt to utilise the first pressure wave as the intake manifold becomes too large to be effectively incorporated under the bonnet of a motor vehicle The tuned runner length is usually calculated so that the maximum ramming effect occurs at the engine speed that peak torque is developed (Seng, T. 2004) These values are used as starting point only and are only accurate within ± 50mm. The engine speed that the FZR600 produces peak torque is 8500rpm. Substituting this value into equation 6.3a yields:

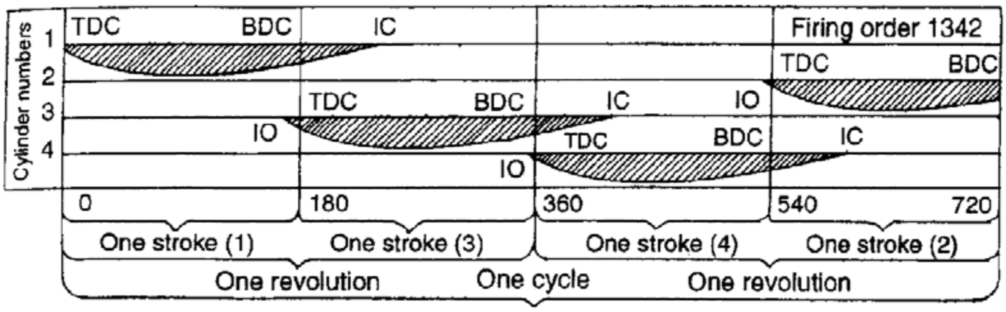
$$L_2(FZR600) = \frac{132000}{8500} = 15.5in$$

When the fuel/air mixture passes through extremely long runners it tends to slow down due to the viscous forces that oppose the flow. When the charge slows down the mixture tends to atomise, which leads to poor combustion. Therefore it is recommended that taper is incorporated into the inside of the runners to maintain the mixture in a fully atomised state. For long intake manifolds runners a 1 – 1.5° taper is recommended (Bell, G. 1998, et. al.). After further consultation with the workshop it was found that the university does not have the facilities to produce a taper in a 15.5inch long pipe. Therefore the use of long runners was deemed unfeasible. It is important to note that the other Formula SAE teams, which use tuned manifolds, have multi-point fuel injection. For this reason only air passes through the intake runners and therefore the runners in their intake systems do not require tapers.

6.10.2 Inter-cylinder Robbery of Charge

Inter-cylinder robbery occurs in multi-cylinder engines when a cylinder draws extra charge from the plenum that was intended for another cylinder. This adversely affects the cylinder filling of both of the cylinders. Inter-cylinder robbery of charge occurs when two cylinders that are adjacent to each other have overlapping induction periods (i.e. the intake valves of the two cylinders are open at the same time). The overlapping of induction periods between cylinders occurs because the intake valves open before the piston reaches top dead centre and close after the piston bottom dead centre.

The Yamaha FZR 600 engine has a firing order of 1-3-4-2. This means that when cylinder 3 is towards the end of the induction period the induction period for cylinder 4 is beginning. Because cylinder 3 and 4 are located next to each other, inter-cylinder robbery of charge occurs between these two cylinders. The same situation occurs between cylinder 2 and 1. The overlapping of induction periods of an engine with a 1-3-4-2 firing order is shown in figure is shown in figure 6.27.



(Source Heisler, H 1998, p243)

Figure 6.27 In-line 4 cylinder engine induction period timing diagram.

Because the FZR600 is a performance engine the valve opening duration is comparatively long. The inlet valve opens at 28˚ before top dead centre and 72˚ after top dead centre (Hamwood, D. 2004 pers. comm. 24 Mar.), whereas a conventional engine's inlet valve opens at 10˚ before top dead centre and closes 50˚ after top dead centre. Inter-cylinder robbery of charge diminishes at higher engine speeds because the time interval that the inlet valves stay open decreases. However, it was considered that inter-cylinder robbery of charge was an issue that required consideration in the design of the intake manifold.

6.10.3 Manufacturability

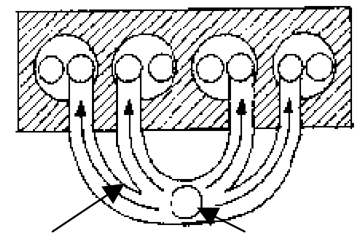
Discussions were held with Peter Penfold (Lecturer of Mechanical Engineering) in regards to the methods available at USQ for manufacturing the intake manifold. Peter cited that to cast the manifold would be a major task and would probably be beyond the capabilities of the workshop. Construction of the manifold out of fibre-composites was also considered. The use of fibre-composites was also deemed a major task that, for the relative performance gains, would require too much work and cost. For these reasons it was decided to construct the manifold out of tubing.

6.10.4 Material Selection

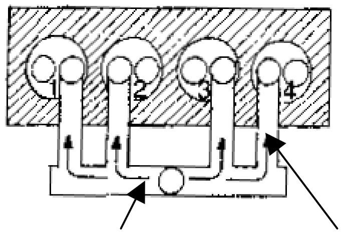
Tubing is commonly manufactured from steel or aluminium. Aluminium tubing was chosen because of its low density compared to steel. Aluminium also has high thermal conductivity. For general-purpose automotive engines intake manifolds, using a material with high thermal conductivity is advantageous as it helps provide sufficient preheating for cold starting. Of course, the disadvantage of using a material with a high thermal conductivity is that it can cause the density of the charge to drop after the engine has warmed up. Therefore manifold heating is avoided when the engine is to be used for high performance applications (McFarland, J. 2004) For this reason it was hoped that the original rubber manifolds, which would provide insulation from the engine heat, could be retained as a part of the intake manifold for the Formula SAE car.

6.11 Intake Manifold Configuration Selection

Because of the manufacturing restraints, two basic intake manifold configurations were considered. These designs included the log type manifold and the streamlined manifold. The general configuration of these two manifold types is depicted in figure 6.28.



A



B

(*Source*: Heisler, H 1998, p249)

Figure 6.28 Streamlined manifold (A) and log manifold (B).

6.11.1 Streamlined Manifolds

From figure 6.28A it can be observed that the streamlined manifold incorporates runners that gently sweep into a central plenum. It ca also be seen that and that the inner runners are shorter than the outer runners and have a tighter bend radius. The obvious advantage of using gently swept runners, as opposed to runners that change direction abruptly, is the minimisation of head loss.

When a streamlined manifold, in conjunction with a dual-barrel carburettor, is fitted to a four-cylinder engine with a firing order of 1-3-4-2, inter-cylinder robbery of charge is eliminated or vastly reduced (Heisler, H. 1998, p241). In order to eliminate charge robbery the carburettor is positioned so that the inner runners are 'fed' by one barrel and the outer runners are 'fed' by the other barrel. Furthermore, some engine modifiers introduce a divider into the plenum chamber to completely eliminate charge robbery (Smith, H. P. 1971, p67). Figure 6.29 represents the layout of a streamlined intake plenum chamber when a divider is incorporated. By fitting the divider, inter-cylinder robbery of charge is completely eliminated as the cylinders that have overlapping induction periods are 'fed' by separate ducting systems.

Unfortunately the Formula SAE rules specify that a single restrictor must be fitted between the carburettor throttle body and the engine. The restrictor would have to be placed between the plenum and the carburettor. For this reason the charge would enter the plenum through a single passage. Therefore, the implementation of a dual-barrel carburettor in conjunction with a streamlined manifold would not reduce inter-cylinder robbery of charge.

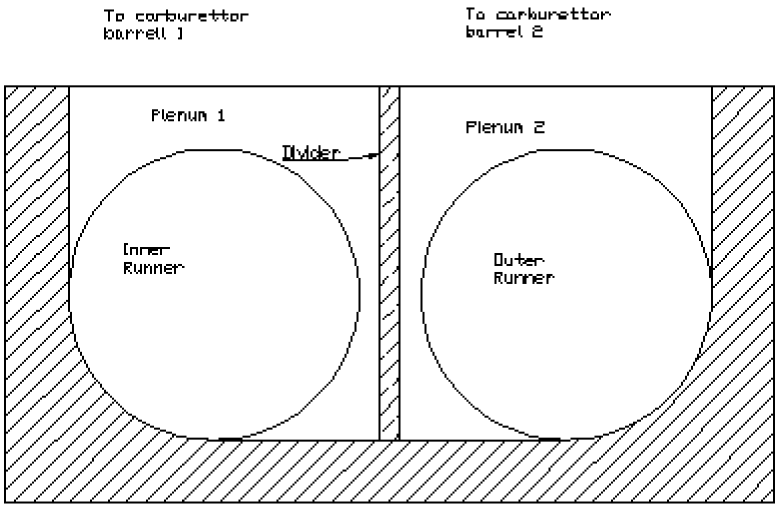


Figure 6.29 Cross-section of a streamlined manifold with a divided plenum chamber.

Due to the geometry of the streamlined type manifold it is not generally possible to have the inner and outer runners the same length. The unequal lengths and differing cause the outer cylinders to have higher volumetric efficiencies than the inner cylinders at high engine speeds. The tendency for the streamlined manifold to dramatically loose volumetric efficiency at high engine speeds can be seen in figure 6.25.

From the previous discussions it can be assumed that if a streamlined manifold were fitted to the Formula SAE engine it would display poor volumetric efficiency at lower engine speeds, due to inter-cylinder robbery of charge. Furthermore, the streamlined manifold would display relatively poor volumetric efficiency at high engine speeds due to the unequal lengths of the runners. Therefore it was concluded that the implementation of a streamlined manifold was unsuitable for the Formula SAE engine.

6.11.2 Log Manifold

From figure 6.30 it can be seen that the log intake manifold is a comparatively simple design. A log manifold consists of straight runners that intersect perpendicularly to a common plenum chamber. The plenum chamber incorporates a buffer zone at each end. Because there is a long distance between where the runners meet the plenum the tendency of inter-cylinder robbery of charge is greatly reduced (Garret, Steeds and Newton, 2001,et. al.).

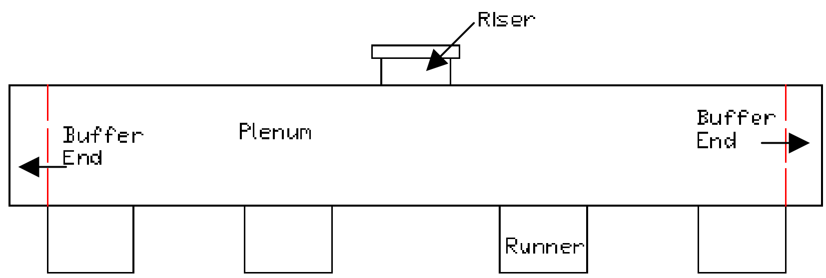


Figure 6.30 Log intake manifold layout.

The quality of mixing and the mixture strength between cylinders tends to be relatively uniform over a wide range of operating conditions when a log type manifold is utilised on an in-line four-cylinder engine (Bell, G. 1998 et. al.). The aforementioned characteristic of the log manifold is largely attributed to the formation of air pockets in the buffer ends. As the mixture leaves the plenum chamber and enters the runners the heavier fuel particles tend to overshoot the runner entrances. When the heavier fuel particles hit the air pockets in the buffer ends it is bounced back into the main air stream of the adjacent runners. Generally, the turbulent air movement created by the trapped charge in this zone improves the mixture distribution as it is projected back into the air stream.

At some engine speeds the pulsations that are caused by the opening and closing of the inlet valves can actually impede the flow of the charge into the cylinder. It is possible to tune the buffer ends to damp out these pulsations and thus improve the flow of the charge into the cylinder (Heisler, H. 1998, p238).

From first inspection, the most obvious flaw in the log manifold is that when the charge enters the runners it must make a 90˚ turn. This sharp turn obviously causes head losses and turbulence in the flow. Nevertheless, in the past it has been found that, in general, log intake manifolds have the capacity to perform better than streamlined manifolds when implemented on in-line four-cylinder engines (Bell, G. 1998 et. al.). This underpins the conclusions of section 6.10.1. Furthermore, it was found that the USQ workshop was capable of producing a log type manifold but did not have the facilities to produce the curved runners of the streamlined manifold. For these reasons it was decided to pursue the design of a log intake manifold for the Formula SAE car.

6.12 Log Manifold Design

6.12.1 Internal shape and surface finish

In order to minimise viscous forces and optimise volumetric efficiency it was decided that the internal surface finish of the intake manifold should be smooth. The only advantage that a rougher surface provides is a layer of micro-turbulence, which helps promote evaporation in cold starting situations. Since the engine is to be used for racing purposes it was decided that it would be more important to optimise the performance of the engine rather than its cold starting ability. To further minimise viscous drag it was decided to use circular tubing to construct the plenum and the runners, as a circular section provides the least surface area and hence the least viscous drag.

6.12.2 Riser Design

The diameter of the riser is the same as the throat diameter of the carburettor. When designing the entrance from the plenum to the riser it is instinctive to incorporate a rounded edge, as depicted in figure 6.31(B), to minimise head losses. However, it is common practice to retain a T-junction configuration as depicted in figure 6.31(A). It has been found that the sharp edges of the T-junction shape 'shatters' the fuel droplets and promotes evaporation. It has also been established that T-junction shape also promotes the most uniform distribution of flow (Garret, Steeds and Newton, 2001,et. al.). Therefore a riser with a T-junction configuration was used on the Formula SAE intake manifold.

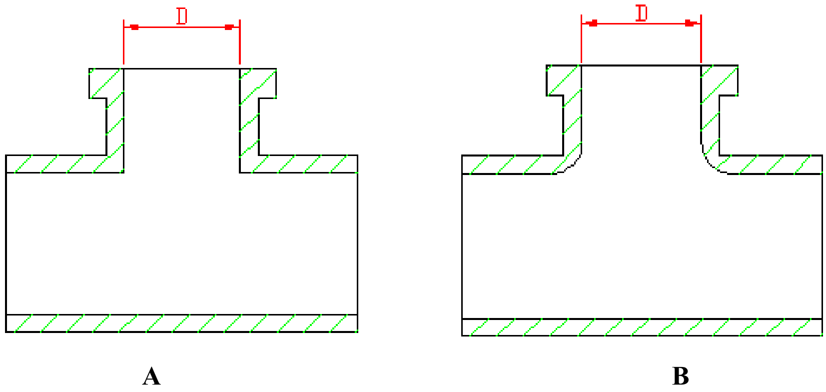


Figure 6.31 Risers designs for a log intake manifold.

6.12.3 Runner Design

In order for the buffer zones to able to function as described in section 6.10.2 it is a requirement that the buffer zones are close to the intake valves. As the existing intake manifolds are comparatively short it was possible to utilise the existing intake manifolds as the runners. Another benefit of the existing intake manifolds runners was that they already incorporate a taper. An additional advantage of using the existing runners is that they are manufactured from hardened rubber, which helps damp out the vibrations that are induced by the engine and the car.

When designing the runner entrances it was decided that a T-junction layout would be utilised. The reasoning for this layout was the same as the logic outlined in section 6.11.2.

6.12.4 Plenum Chamber Design

As a guide, for a four-cylinder engine the plenum chamber volume, for a log type manifold should be between 0.65 and 1.9 times the displacement of the engine (Garrett T.K., Steeds W. & Newton, K. 2001, p483). It was found that such a large range is recommended, as the intake manifold will function using a wide range of plenum volumes (Grape Racing, 2004 et. al.). But in order to select the most suitable plenum volume for the Formula SAE engine an experienced performance engine builder was contacted. It was recommended that in order to retain a torque across the entire rev range a smaller plenum be used. Conversely, if a large plenum is used lower end torque will suffer, but gains will be made in top end power. Given that the engine was to be used for circuit racing events it was recommended that a smaller plenum volume be implemented (Wolferden, T. 2004 pers. comm. 20 July). In order to obtain a volume of approximately 0.65 times the engine displacement it was found that a pipe with an inside diameter of 36mm was the most suitable[9]. The internal diameter of 36mm was also chosen because aluminium tubing is available in this size.

In section 6.10.2 it was stated that the buffer ends can be tuned to damp out the pulsations that are caused by the valves opening and closing. In order to effectively damp out the pulsations the volume of the buffer ends is critical (Heisler, H. 1998). Unfortunately no literature was available that allowed the calculation of the optimum volume of these buffers.

6.12.5 Prototype Intake Manifold for the Formula SAE Engine

A prototype intake manifold was designed and constructed so that experimentation could be carried out to determine the optimum buffer volume. The internal dimensions of the prototype manifold were identical to the final intake manifold, provided the prototype proved successful. The prototype intake manifold was constructed from mild steel and is depicted in figure 6.32. Although steel does not have the same thermal properties as aluminium, steel was used to minimise development costs.

Figure 6.32 Prototype log type intake manifold.

[9] The calculations that were used to arrive at this diameter can be found in appendix E.

The prototype manifold incorporated adjusters that were fitted into each end of the plenum chamber. The adjusters, depicted in figure 6.33, allowed the volume of the buffer ends to be altered. At the time the prototype manifold was designed and constructed it was envisioned that the rest of the car would be at a stage of completion that would allow dynamometer testing. Had this been the case, it would have been possible to establish what affect the buffer end volume had on the performance of the engine. It was planned that separate dynamometer runs would be performed at different buffer volume settings. It was anticipated that the trend of the torque curves, produced by the dynamometer tests, would have shown noticeably different trends. This would have allowed a selection of the buffer end length and thus the design of the final intake manifold.

Figure 6.33 Prototype intake manifold buffer end adjuster.

Because the car was not in a state that facilitated dynamometer testing, very little meaningful testing could be carried out. The carburettor was mounted on the prototype intake manifold and the engine was run, unloaded, across all throttle positions with different buffer volume settings. In these tests the restrictor was not fitted. The engine did not sound or respond any differently when the buffer volume was adjusted. Very little can be concluded from this experiment as the engine was not loaded and no measurement devices were being utilised. The most that can be concluded is that the intake manifold provided a sufficient passage to allow the air/fuel mixture to be administered to the engine.

6.13 Intake Restrictor Design

The Formula SAE rules state that a single circular restrictor with a diameter of 20mm must be placed between the throttle body and the engine (Formula SAE Rules, 2004). The diameter of the inlet tracts on the standard Yamaha FZR600 engine is 32mm. Because the USQ Motorsport engine utilises a carburettor the restrictor must be placed between the carburettor and the plenum, which is shown in figure 6.34.

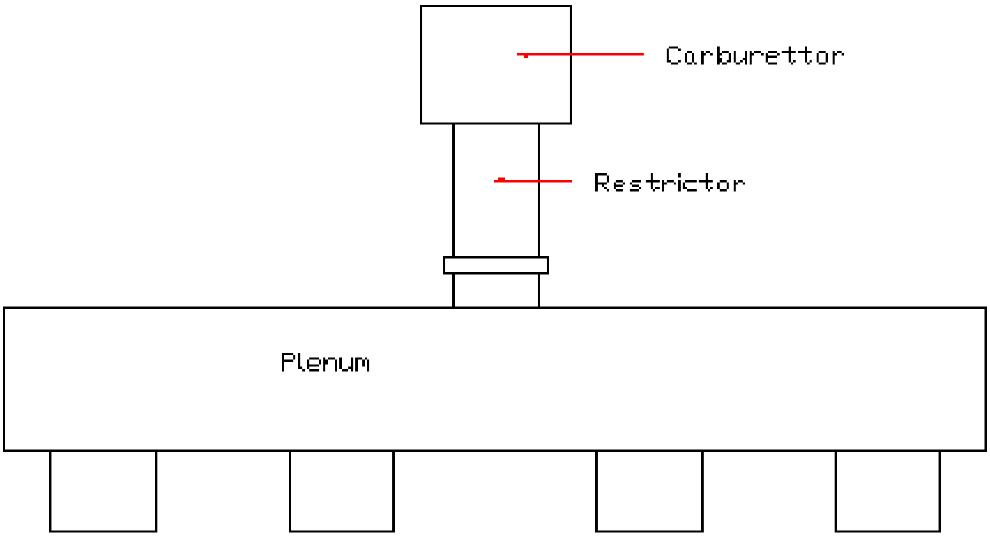


Figure 6.34 Position of the intake restrictor.

The effect of placing a restriction in the intake manifold tract is that the flow becomes choked at a lower engine speed. When the flow in the intake manifold becomes choked the mass flow rate of the mixture cannot be increased any further. Consequently the density of mixture in the cylinders is decreased. Therefore, the point at which choking occurs essentially determines the ceiling of an engines performance. By incorporating an intake restrictor into the Formula SAE rules, the organisers of the competition have essentially placed an upper limit on engine performance.

The direction of the flow and the terminology that will be used to describe each section of the restrictor is shown in figure 6.35

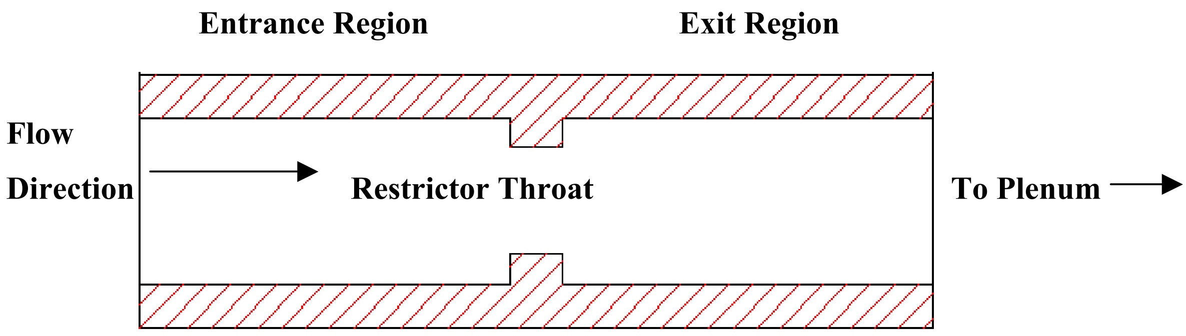


Figure 6.35 Direction of flow through the intake restrictor.

6.13.1 Intake Restrictor Design Criteria

The maximum flow rate that can be obtained through the restrictor throat is determined by equation 6.5 (Bird Precision, 2004):

$$\dot{m}_{\max} = \frac{C_d \quad A_p \quad K}{\sqrt{\frac{M\,U}{T}}} \qquad \textbf{(Eq. 6.5)}$$

where $\dot{m}_{\max}$ is the maximum flow rate that can be achieved;

Ap is the cross-sectional area of the nozzle throat;

M is the molecular weight of the gas;

U is the characteristic gas constant;

T is the upstream absolute temperature;

K is the dimensional gas constant; and

C_d is the discharge coefficient.

The discharge coefficient is defined as: $\dfrac{actual\ \ flow\ rate}{theoretical\ \ flow\,rate}$

All of the variables, apart from the discharge coefficient, in equation 6.5 are determined by the properties of the gas, atmospheric conditions or Formula SAE competition rules and cannot be easily manipulated to increase the flow rate through the restrictor. Therefore the only variable that can be optimised is the discharge coefficient.

6.13.2 Intake Restrictor Entrance Region Design

The discharge coefficient is determined by many factors. The first factor is the ratio of the nozzle throat cross sectional area and the cross sectional area before the pipe contracts. This ratio was determined by the size of the carburettor and the nozzle throat requirements. The geometry of the entrance region of the restrictor throat is also significant in the determination of the discharge coefficient. As the rules do not specify the geometry of the entrance region of the restrictor, it was possible to optimise this geometry. In order to find the geometry of the restrictor entrance region that would result in the maximum discharge coefficient, flow measurement device design standards were implemented[10]. These standards were relevant because the configuration of flow measurement devices is very similar to the restrictor. Most importantly, the maximisation of the discharge coefficient of flow measurement devices is an important design criterion. Because these standards are a conglomeration of many years of research it was considered that it would be difficult to improve on these designs.

It was found that there are three basic profiles used in the design of the entrance regions of flow measurement devices. The entrance region geometries included an elliptical, conical and straight edged shape, as depicted in figure 6.36.

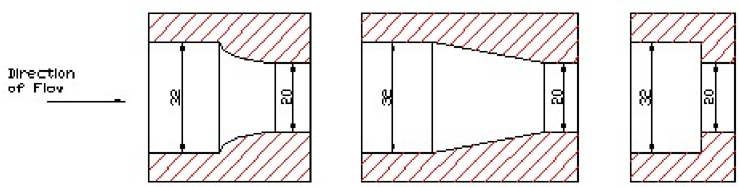


Figure 6.36 Restrictor entrance region profiles.

The discharge coefficient is also dependent on the Reynolds number of the flow as it passes through the throat. The Reynolds number is defined as:

[10] The details of the standard that was used to design each throat entrance type for the restrictors can be found appendix E, section E2.

$$\mathrm{Re}_D = \frac{\rho \quad v \quad D}{\mu} \qquad \textbf{(Eq. 6.6)}$$

where Re_D is the Reynolds number;
ρ is the density of the gas;
D is the diameter of the restrictor throat;
μ is the absolute viscosity of the gas; and
v is the velocity of the gas through the nozzle throat.

To approximate the range of Reynolds numbers that would characterise the flow through the restrictor throat, the Reynolds number was calculated at the maximum engine and minimum engine speeds. The estimated minimum and maximum discharge coefficients for each of the entrance geometries are displayed in table 6.1 (the calculations that were used to arrive these values can be found in appendix E). From table 6.1 it can be seen that the straight edge entrance displays a significantly lower discharge coefficient than the other two entrance region geometries. It can also be seen that the conical and elliptical entrances are characterised by discharge coefficients that approach unity. Additionally it can be observed that the conical and elliptical entrances have almost identical theoretical discharge coefficients. To ensure that the optimum entrance region shape was utilised in the restrictor, it was decided to construct and test three restrictors. Each restrictor incorporated a different shaped entrance (i.e square edged, conical and elliptical).

Throat Entrance Type	Minimum C_d	Maximum C_d
Straight Edge	0.62	0.63
Conical	0.96	0.99
Elliptical	0.97	0.99

Table 6-1 Approximate discharge coefficients for different entrance region types.

6.13.3 Intake Restrictor Exit Region Design

In section 6.12.2 it was shown that the entrance shape of the restrictor plays a major role in determining the maximum flow rate that can be obtained through the restrictor.

However, the shape of the exit region of the restrictor also plays an important role. If the exit region of the restrictor throat changes cross sectional area abruptly a turbulent region will form. This turbulent region downstream of the restrictor throat robs energy from the system and results in a lower discharge coefficient (Miller, R. 1996). For this reason most flow measurement devices incorporate a smooth conical exit zone.

The shape of the restrictor's exit region also determines the amount of pressure that is recovered after the fuel/air mixture after it passes through the throat. When the fuel/air mixture passes through the converging entrance section of a nozzle it is accelerated and reaches its maximum velocity at the restrictor throat. As the fuel/air mixture is travelling through the entrance section of the restrictor its pressure is decreasing. The pressure of the mixture reaches a minimum at the narrowest point in the passage, the throat. If the passage then increases in cross sectional area, the fluid decelerates and regains pressure (Fox F. W. & McDonald A. T. 1998). However, not all of the pressure is regained.

Regaining the maximum amount of pressure was an important consideration when designing the restrictor for the Formula SAE engine. The reason for this is that a loss in pressure of the charge will result in a loss in charge density. In order to regain the maximum amount of pressure it is important to minimise the pressure gradient in the exit region of the restrictor. In order to minimise this pressure gradient, it was recommended that the exit region of the restrictor should be conical in shape and be as long as possible (Mossad, R., 2004, pers. comm. 14 Aug.). It was found that the machining facilities at USQ could produce a restrictor with a conical exit region with a minimum taper angle of 4.5°, given the geometry of the restrictor. This taper angle, also fell within the *ASME* guidelines for pressure recovery in diffusers[11] (Miller, R. 1996, p10.65). The overall geometry of the restrictor is shown in figure 6.37.

[11] The conical exit region that has been described is commonly referred to as a 'diffuser.'

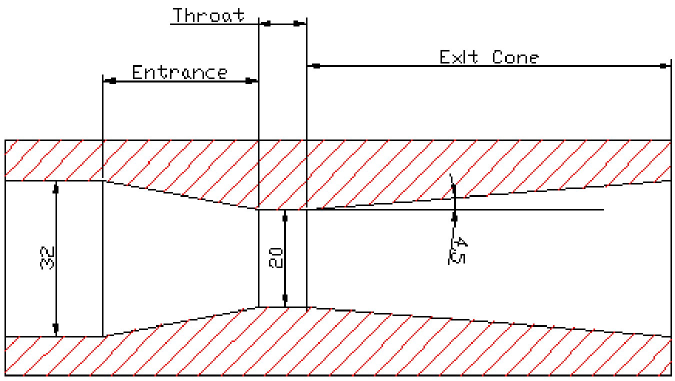


Figure 6.37 Restrictor – overall geometry.

6.13.4 Intake Restrictor Throat Design

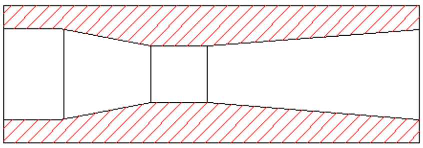
From figure 6.36 it can be seen that the throat of the restrictor is a short section of constant cross section. When this type of device is used for flow measurement, a measurement device is usually placed at this point. It was uncertain whether this section should be retained as part of the intake restrictor. However, it was stated that if the throat region was removed a turbulent zone might occur where the entrance and exit zones converge (Mossad, R. 2004, pers. comm. 14 Aug.). This may occur because a sharp edge would be formed. Furthermore, it was considered that the extra viscous drag that including this section would cause, would have an insignificant effect on the overall intake system. Therefore, it was decided to retain this section as part of the intake restrictor.

6.14 Intake Restrictor Testing

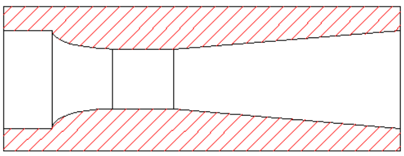
As outlined in section 6.12.2, it was decided that in order to select the optimum entrance shape for the restrictor, the restrictors were tested. Three restrictors with different entrance shapes were constructed and tested (conical, square edged and elliptical). In order to ensure that only the entrance shapes were being compared the throat and diffuser design for the restrictors with conical and elliptical entrances were kept the same. The square edged restrictor did not include a diffuser or an entrance pipe. The square edge restrictor was essentially a 20mm hole that was used as a 'benchmark'. The testing procedure that was followed is outlined in section 3.6.2. Figure 6.38 displays the flow bench that was used to test the restrictors. An airflow bench measures the volume flow rate of air that passes through the restrictor at a specified pressure drop.

Figure 6.38 Flow bench.

The profiles of the restrictors that were tested are depicted in figure 6.39:



(A) Restrictor with conical entrance and diffuser



(B) Restrictor with elliptical entrance and diffuser **(C) Straight Edge Restrictor**

Figure 6.39 Tested restrictor profiles

6.14.1 Experimental Results

Figure 6.40 displays the volume flow rates that were obtained using different pressure drops. Figure 6.40 displays the flow rates that were obtained using a conical shape entrance region and an elliptical shaped entrance region. It can be seen from figure 6.40 that the flow capacity of the restrictor with a conical entrance is slightly higher than the flow capacity of the restrictor with an elliptical entrance region.

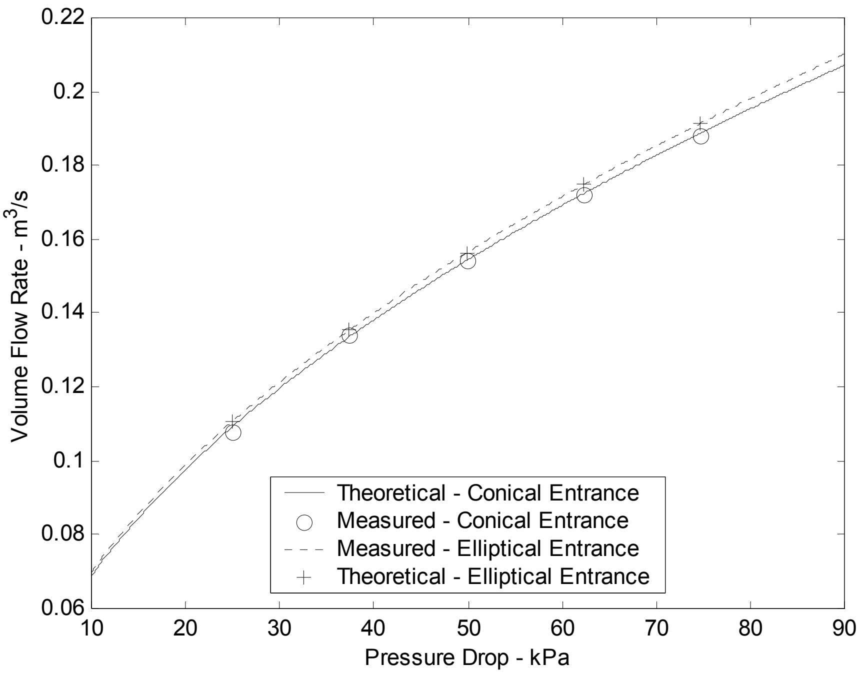


Figure 6.40 Volume flow rates for the restrictors with different entrance region geometry.

From table 6.1, theoretically the discharge coefficients of the restrictor with the conical entrance should be the same as the discharge coefficients of the restrictor with the elliptical entrance region. The values of discharge coefficients for the elliptical and conical entrance in table 6.1 do vary slightly. But it should not be possible to detect these differences using a flow bench, which is not a precision measuring device. However, because the measured values show a definite trend it can be concluded that the conical entrance does allow a higher volume flow rate than the elliptical entrance.

It was considered that the difference in flow rates came about due to the differences in entrance region surface finish. The conical entrance region was easily formed using the tooling that the USQ workshop had available. This resulted in an entrance region surface that was very smooth. To form the elliptical entrance region was far more difficult. The machinist at the workshop had to program the coordinates of each circle that formed the elliptical entrance into the CNC machine manually (Aston B. 2004, pers. comm. 14 Aug.). The coordinates that were supplied to the machinist are displayed in figure 6.41. From figure 6.41 it can be seen that the coordinates are spaced some distance from each other. It was decided to try to use the least amount of coordinates possible to cut down on workshop time. At the time it was thought that the slightly stepped finish would not significantly impact on the performance of the restrictor. However, from the flow rates that were measured it can be concluded that the surface finish of the restrictor entrance region does slightly impact on the maximum flow rate that can be achieved through the restriction.

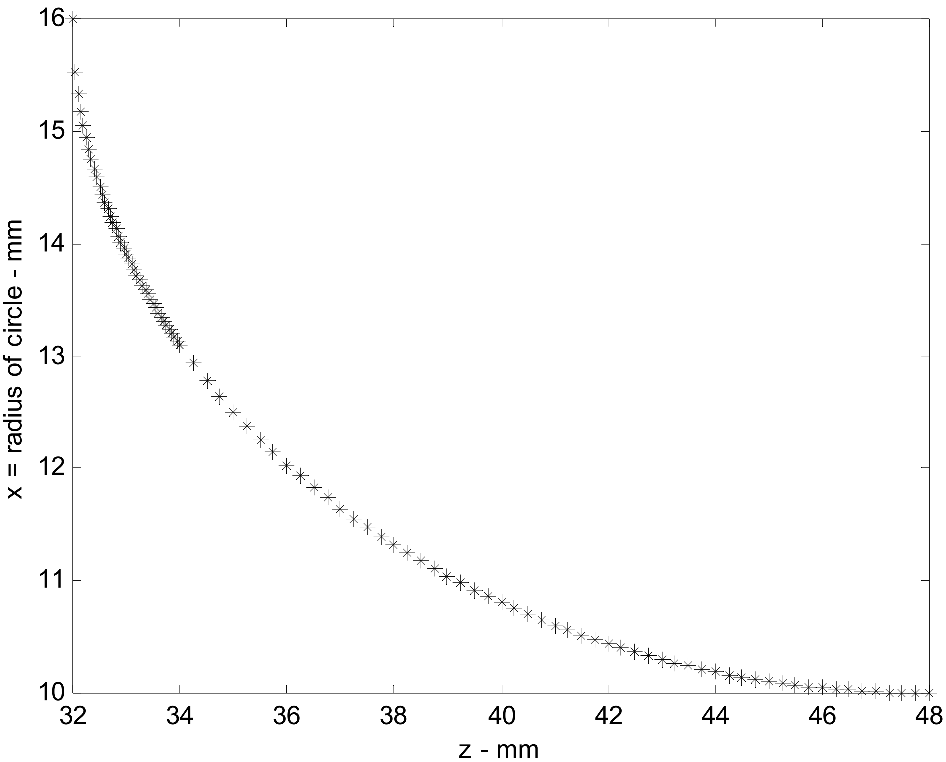


Figure 6.41 The CNC coordinates that were used to form the elliptical entrance.

In figure 6.40 a curve is fitted to the data points. This curve was obtained using equation 6.3. Equation 6.3 is repeated below for clarity:

$$\dot{V}_1 = \dot{V}_2 \sqrt{\frac{p_1}{p_2}} \qquad \textbf{(Eq. 6.3)}$$

Where $\dot{V}_1$ is the volume airflow rating at p_1;

$\dot{V}_2$ is the volume airflow rating at p_2;

p_1 is the standard pressure difference; and

p_2 is pressure difference that volume flow rate is it be calculated at.

For all of the tests that were conducted a pressure difference of 62.3kPa (manometer reading of 25 inches H_2O) was as the standard pressure difference. From figure 6.40 it can be seen that the measured values only deviate slight from the theoretical value calculated using equation 6.3. The deviation of the measured values do not display a general trend, i.e. as the pressure drop increases or decreases the measured flow rates do not move substantially further away from the calculated value in any general direction. The measured values actually fluctuate above and below the theoretical values. However, this fluctuation is only slight as can be seen from table 6.2.

	Conical Entrance		Elliptical Entrance		St. Edge Entrance	
Pressure Drop kPa	Measured Flow Rate m^3/s	Calculated Flow Rate m^3/s	Measured Flow Rate m^3/s	Calculated Flow Rate m^3/s	Measured Flow Rate m^3/s	Calculated Flow Rate m^3/s
74.7522	0.1910	0.1914	0.1880	0.1888	0.1150	0.1147
62.2935	0.1750	0.1747	0.1720	0.1724	0.1050	0.1047
49.8348	0.1560	0.1563	0.1540	0.1542	0.0940	0.0936
37.3761	0.1350	0.1354	0.1340	0.1335	0.0820	0.0811
24.9174	0.1110	0.1105	0.1080	0.1090	0.0670	0.0662

Table 6-2 Restrictors - measured and calculated flow rates.

A number of conclusions can be drawn from this observation. It can be concluded that the discharge coefficient of the restrictors is not changing significantly as the pressure drop and hence the air velocity is increasing. Therefore it can be assumed that to observe the slight changes in the discharge coefficients that occur at the airflow velocities that were used would require high precision equipment.

Given that the pressures that were used are representative of the pressures that occur in an engine's induction system it can be concluded that equation 6.3 is accurate in predicting flow rates at the flow velocities that occur in an engine. Furthermore, the owner of the flow bench stated that in his many years of experience he has always found equation 6.3 to be accurate in predicting flow rates at different pressure drops (Wolferden, T. 2003, pers. comm. 23 Sept.).

In order to prove that restrictor with conical or elliptical entrance region and a diffuser will facilitate greater flow rates, a restrictor with a straight edge entrance and no diffuser was tested to define a 'benchmark'. A comparison of the straight edge restrictor a restrictor with a conical entrance region and a diffuser is displayed in figure 6.42.

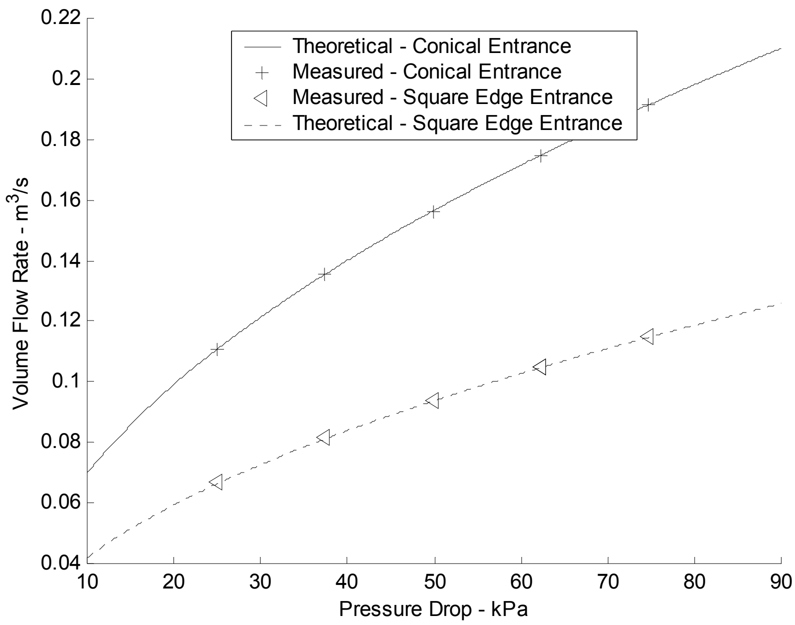


Figure 6.42 Flow rates for a straight edge and a restrictor with a diffuser and a conical entrance region.

From figure 6.42 it can be seen that the straight edged restrictor performed as predicted. The flow rates of the straight edge restrictor are approximately 60% of the flow rates that are obtained by using a restrictor with a diffuser and a conical entrance region.

In order to establish the effect that the restrictor has on the amount of air that can flow through the carburettor the carburettor was flow tested, with and without the restrictor. The results of these tests are shown in figure 6.43. From figure 6.43 it can be seen that the restrictor does significantly reduce the flow rate that can be achieved through the carburettor. By fitting the carburettor to the restrictor, the air must pass through two venturis and over the carburettor jet needle. It can be assumed that the flow becomes turbulent as it passes over the jet needle (see section 6.6 for explanation of the inner workers of the carburettor). This added turbulence in the flow would then negatively impact on the amount of air that can pass through the restrictor. This may explain why the flow capacity of the restrictor combined with the carburettor is significantly lower than either the carburettor or restrictor when tested separately. In order to alleviate this problem it may help to place the restrictor and the carburettor as far away from each other as possible. However such a solution is not feasible due to the space constraints of the Formula SAE car.

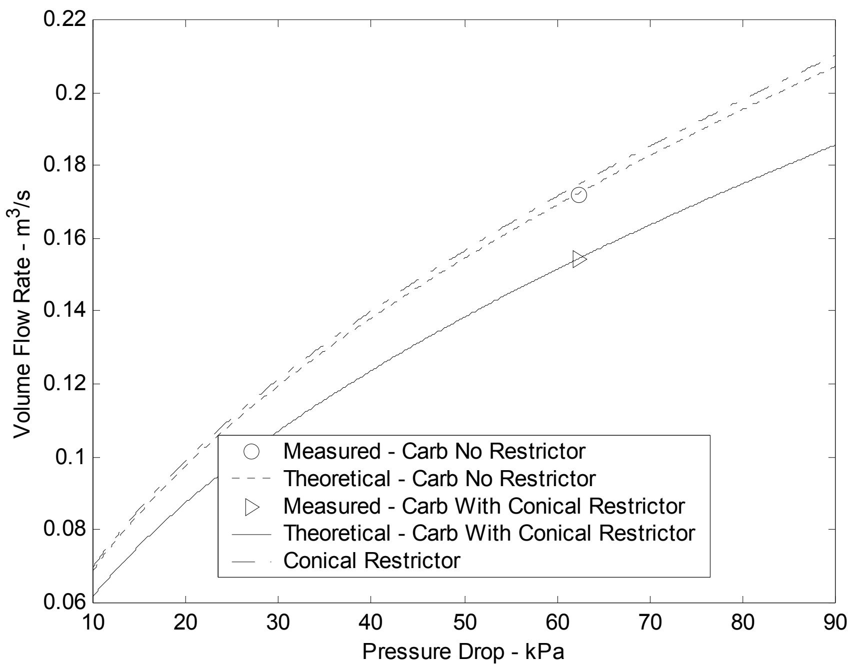


Figure 6.43 Carburettor flow rates.

6.14.2 Limitations of the Flow Test Results

One of the most limiting factors of the testing is that the flow bench does not measure the true airflow rates of the restrictor that would be obtained when the restrictor is fitted to the engine. This is due to the geometry of the testing device. Figure 6.44 shows a scaled cross sectional view of the path that the air must pass through. From figure 6.44 it can be observed that the cross sectional area of the duct changes rapidly when the air leaves the restrictor and enters the passage in the flow bench. The reason that the passage in the flow bench is so large (100mm in diameter) is that the flow bench was designed to test components that are fitted to large V8 engines that have large pistons.

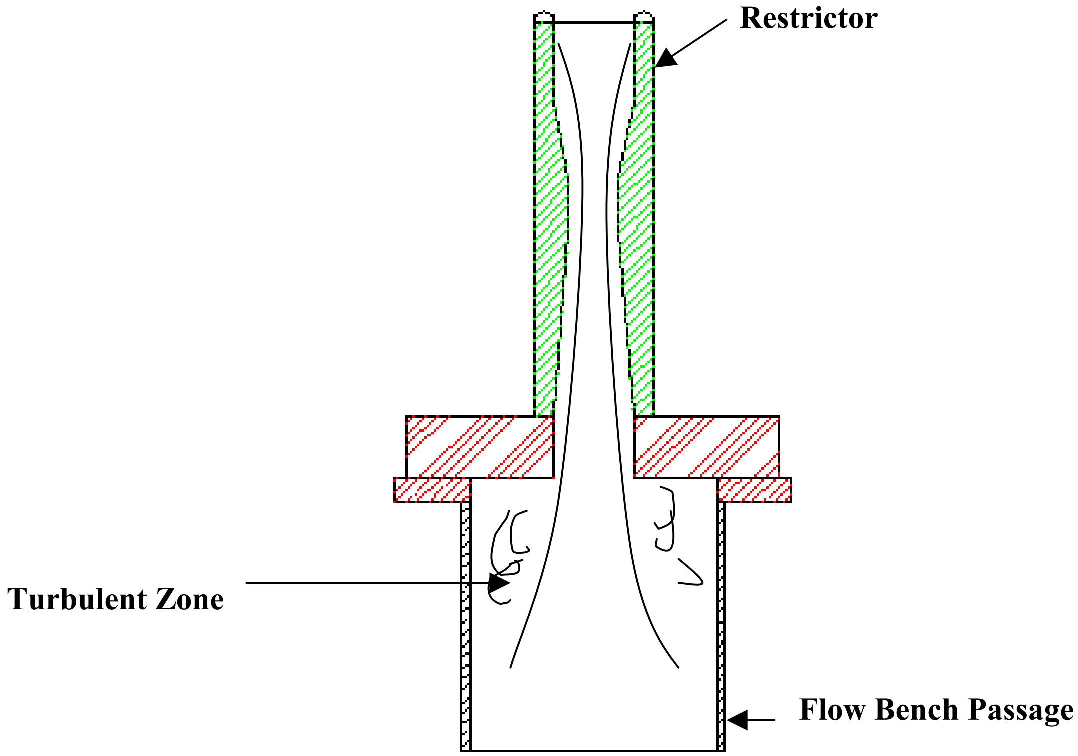


Figure 6.44 Flow bench and restrictor cross-section.

Because of the rapid change in cross section as the air passes into the flow bench passage, a turbulent zone will form down stream of the diffuser. This turbulent zone will cause the discharge coefficient of the restrictor to fall dramatically. When the air/fuel mixture passes through the restrictor and into the intake manifold plenum chamber the change in cross sectional area is not as marked, thus no comparison can be drawn between the actual engine conditions and the flow conditions. For this reason it is not possible to correlate the measured flow rates directly to the engine flow capacity. However, the previous comparative analysis is still valid.

The most obvious limitation of the testing is that the fluid that is being used for test purposes is air, not a mixture of fuel and air. In the words of the flow bench owner:

" … *Air is one thing, but the whole thing changes when you add fuel.*" (Wolferden, T. 2004, pers. comm. 12 Aug.)

In order to truly test the restrictor it must be fitted to the engine and tested on a dynamometer. Due to the reasons already stated these tests were not undertaken in this project. Nevertheless, the conical and elliptical restrictors were fitted to the engine and tested. The engine was run, with no load, across the entire range of throttle settings. No significant difference could be noticed in the way the engine sounded or ran by fitting a restrictor. There was also no observed difference in the engine by fitting either the conical restrictor or the elliptical restrictor.

By testing of the restrictor on the engine, essentially the entire induction system was being tested. A photo of the induction system is shown in figure 6.45. The fact that the engine ran, seemingly unaffected, with the restrictors fitted would seem to indicate that the design and specification of the induction system was promising.

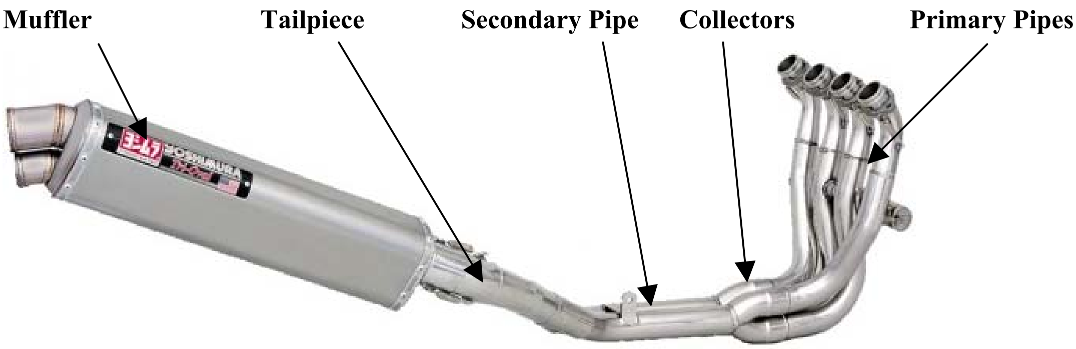
Figure 6.45 The Formula SAE induction system,

6.14.3 Intake Restrictor Specification

The restrictor with the conical shaped entrance allowed the greatest airflow rates. Although testing on a dynamometer would have confirmed or disputed the validity of the flow testing results, it was assumed that the conical restrictor would allow the engine to produce the most power. Detailed drawings of the restrictor designs and the testing apparatus can be found in Appendix H.

6.15 Exhaust System Specification for the Formula SAE Car

The role of the exhaust system, in its most basic form, is to duct the combustion products from the engine to the atmosphere. However, by correct design of the exhaust system the volumetric efficiency of the engine can be improved. Conversely, if the exhaust is poorly designed the performance of the engine can suffer. A motorcycle exhaust consists of three main components, the muffler, exhaust manifold and the tailpiece. The exhaust manifold consists of the primary pipes, secondary pipe/s and the collector/s. These components are shown in figure 6.46:



(*Source*: Yoshimura, 2004)

Figure 6.46 The components of a 4 into 1motorcycle engine exhaust system.

Because the flow regime of the gas is the compressible, viscid and turbulent the initial design of the entire exhaust system is usually carried out using CFD software (Ferguson, C.R. 1986). Once an initial design has been defined the results are verified using dynamometer and flow bench tests and adjustments are made accordingly.

6.16 Exhaust Manifold Specification

The exhaust manifold that is featured on all modern motorcycle engines is known as an extractor system. Extractors refer to exhaust manifolds that incorporate long primary pipes, generally of equal length. This paper will only deal with exhaust manifolds that incorporate extractors.

6.16.1 Extractor Exhaust System Design

Extractors utilise two important phenomena, wave scavenging and inertial scavenging, which are important to both the extraction of exhaust gases and the induction of fresh charge.

In most modern engines valve overlap is incorporated into the engine's design. Valve overlap is a period in the engine cycle when both the intake valves and exhaust valves are open at the same time. The purpose of this overlap period is to optimise the filling of the cylinder. The inertia of the burnt exhaust gases leaving the combustion chamber creates a vacuum which helps draw out the residual burnt gases whilst drawing fresh charge into the combustion chamber. This effect is known as 'inertial scavenging'. Inertial scavenging is only effective if the exhaust gases do not interfere with each other as they travel through the exhaust manifold. In order to alleviate interference between the exhaust gases of each cylinder long primary pipes are utilised (Vizard, D. 1990).

In section 6.10.1 the effects of tuning the intake manifold to take advantage of compression and rarefaction waves were discussed in detail. When designing the exhaust manifold the same theory is applied except that the aim is produce a localised region of low pressure in the exhaust manifold. The exhaust length is tuned so that at a critical engine speed, the first reflected negative pressure wave is at its minimum pressure in the period of valve overlap. This effect is referred to as 'exhaust wave scavenging'.

When designing exhaust manifolds to utilise wave scavenging the length of the extractor pipe is obviously of high importance. However there are a number of other variables that must be considered. These variables include:

1. Primary pipe diameter;
2. Collector geometry;
3. Secondary pipe diameter; and
4. Number of primary pipes per collector.

All these variables impact on how well the exhaust performs (Vizard, D. 1990). Because the relationship between the aforementioned variables is complex it will not be examined in this paper. The reason for this is that the FZR600 engine that was purchased already had an extractor exhaust manifold. The original manifold and is depicted in figure 6.47. It was considered that it would be very difficult to improve the design of this system, given the resources available to this project. It was also considered that any resources that were available to this project should be directed towards the optimisation of the induction system. Furthermore, very few Formula SAE-A teams choose to replace the standard extractors (AutoSpeed, 2004). The most likely reason for this is that there is very little performance advantage obtained by replacing the factory system (Bowdler, J. 2004). For all of these reasons it was decided that the standard extractor system should be retained.

Figure 6.47 The original FZR extractor system.

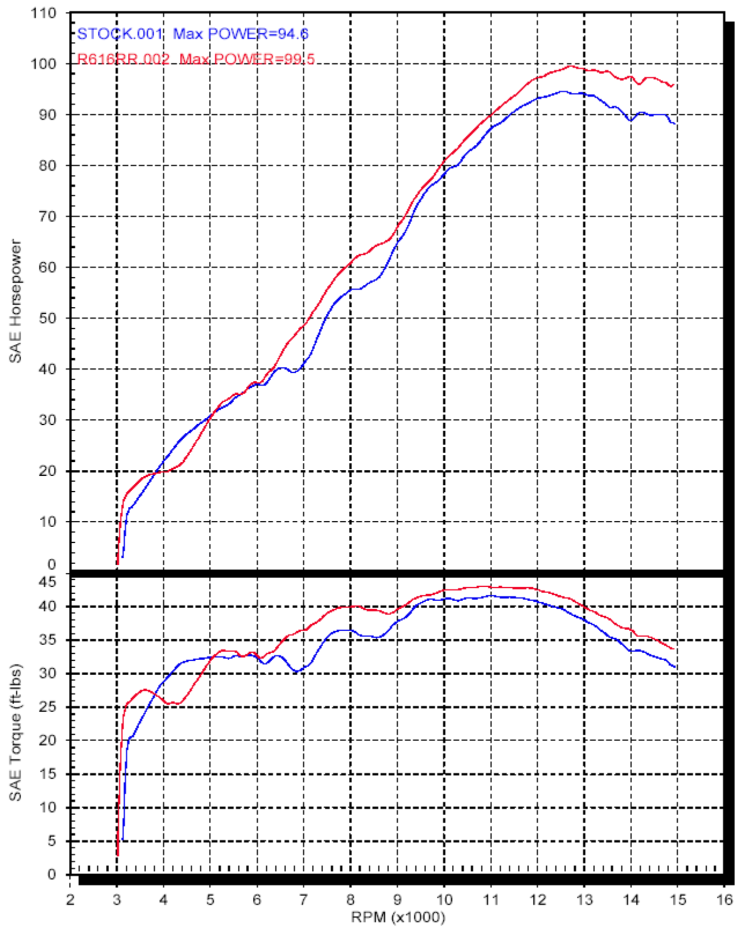
6.17 Muffler Specification

The FZR600 engine that was acquired did not have a muffler. Therefore a muffler had to be either designed or specified. The role of the muffler is simply to silence the engine. In the Formula SAE-A competition the highest sound level that is allowed is 110dB(A) (Formula SAE Rules, 2004). The main draw back of fitting a muffler is that it places a restriction in the exhaust system and impedes the flow of the exhaust gases. The muffler also increases backpressure in the exhaust system. The commonly held belief that the engine requires backpressure to run ‘properly’ is unfounded (Vizard, D. 1990 et. al). Vizard (1990) has found, through many years of dynamometer testing, that a decrease in backpressure will *always* result in an increase in engine torque output and fuel economy.

As stated previously, due to the time restraints of this project, analysis using CFD software was not possible. To use experimental methods to design and optimise the muffler would require the constant use of a flow bench and a dynamometer. Although sponsorship was obtained for both of these tools, it was thought that it would be unfair to expect the sponsors to spend many hours performing tests. Furthermore, in order to truly find an optimum design would most likely require the construction of many mufflers. It was considered that the material and labour that would be required for such an undertaking would far exceed the cost of purchasing a muffler ‘off the shelf’. The cost of a new high performance muffler to suit the FZR600 exhaust system is $395 (Akrapovic, 2004), which confirmed this assumption.

The purchase of a muffler that has been produced by specialists has many advantages, apart from cost. Muffler manufacturers typically supply a sound level rating with their products. The muffler that the quote was obtained for had a maximum sound level rating of approximately 98dB(A) (Akrapovic, 2004). This sound level is well within the requirements of the Formula SAE-A competition rules. Although it is commonly believed that a muffler with a higher sound output will increase engine performance, there is no relationship between engine performance and sound output (Vizard, D. 1990 et. al.). Therefore it can be concluded that USQ Motorsport would not be disadvantaged by using a muffler that is rated at 98dB(A) as opposed to a muffler that is rated at the competition maximum of 110dB(A).

Most importantly performance muffler manufacturers have tested their mufflers to ensure that the engine performance will increase. Most reputable muffler manufacturers provide dynamometer output charts to back-up their claims. Figure 6.48 displays a typical engine performance chart provided by an exhaust manufacturer. Figure 6.48 does not display the actual performance increase that an FZR600 engine would exhibit if a performance muffler were fitted to it. However, figure 6.48 does represent the order of magnitude of the gains that can be expected by fitting a performance muffler as opposed to the standard muffler.



(*Source:* Yoshimura, 2004)

Figure 6.48 Comparative engine performance curves for a Honda CBR600, fitted with an aftermarket muffler (red) and fitted with a standard muffler (blue).

6.18 The USQ Motorsport Exhaust System

The USQ Motorsport team decided to lower the ride hide of the car to a point where the standard extractors could not be used. This decision was made in order to lower the position of the centre of gravity of the car. An extremely low centre of gravity was pursued so that the car would pass the competition tilt test. The tilt test is performed by placing the car on a 45˚ angle (Formula SAE Rules, 2004). Other members of the team decided that the performance of the exhaust and hence the engine performance was of secondary importance compared to the position of the centre of gravity (Baker C. & Armstrong, J. 2004, pers. comm. 29 May). Nearly all of the other cars in the competition, which are all of very similar design, incorporate extractors that run underneath the car (AutoSpeed, 2004). For this reason it is uncertain why such a compromise had to be made.

Some of the other USQ Motorsport members constructed the exhaust system so that the car had an exhaust system ready for USQ open day. Hence, the 'design' and construction of the exhaust system that was built for the Formula SAE car does not form a part of this project. The exhaust system that was constructed for the car is shown in figure 6.49. The USQ Motorsport exhaust system costed $225 to construct (Armstrong, J. 2004, pers. comm. 29 July), which would have paid for a large proportion of the muffler that was specified as part of this project.

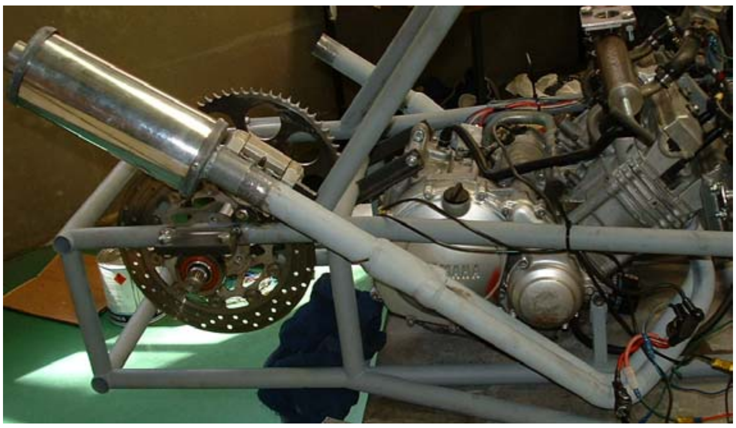


Figure 6.49 THe USQ Motorsport exhaust system.

6.18.1 Exhaust System Testing

During the testing of all of the induction system components the exhaust system that was constructed by the team was fitted to the engine. Although its affects on engine performance, adverse or otherwise, are unknown as the engine was not tested using a dynamometer. If a dynamometer had been used a comparison could have been made between the system that the team built and the system that was specified. A high performance muffler could have been easily sourced from the author's own motorcycle.

6.19 Conclusion

This chapter encompassed the design and testing of the fuelling system for the Formula SAE car. In the initial stages the feasibility of turbochargers, superchargers and multi-point electronic fuel injection was examined. It was found that all of these systems do have the capacity to significantly improve the performance of the Formula SAE engine. However, due to the financial constraints of the USQ Motorsport team it was found that new and used superchargers, turbochargers and electronic fuel injection were unobtainable.

The later parts of this chapter detailed the design and specification of the fuelling system for the 2004 Formula SAE car. It was found that a 32mm Mikuni carburettor was most suitable for the USQ Motorsport Formula SAE car. From the research that was conducted, it was concluded that most suitable intake manifold for the USQ Motorsport car was of the log variety. Through research and experiment it was found that the intake restrictor should have a conical entrance region with a 10.6˚ taper, a 20mm throat and a diffuser with a 4.5˚ taper. It was concluded that the standard exhaust manifold should be retained and an aftermarket performance muffler should be fitted. The following chapter will discuss the cooling system for the Formula SAE car.

7 Cooling System

7.1 Introduction

In this chapter the cooling system for the Formula SAE car will be analysed. The most suitable mounting position of the radiator was found first, as the positioning of the radiator largely dictates the specification or design of the radiator. This chapter will also outline the tests that were devised to determine if the motorcycle radiator is suitable for the Formula SAE car. It was hoped that the original motorcycle radiator could be retained given budget restraints of the USQ Motorsport team.

7.1.1 Radiator Positioning

The most important consideration when positioning the radiator is that the top of the radiator is elevated above the top of the engine. The top of the radiator must be higher than the engine so that any localised steam pockets that form in the engine are allowed to escape. If localised steam pockets are allowed to form in the engine it can cause catastrophic failure of engine components. To determine the optimum position of the radiator the following aspects were considered:

1. Ease of mounting at the elevation required;
2. Amount of piping required & therefore extra mass;
3. Cost;
4. Ability to upgrade the system;
5. Proximity to heat sources;

6. Airflow conditions and possibility of obstructions to the airflow;
7. Affect on aerodynamics of the vehicle;
8. Safety of the driver; and
9. Accessibility to the engine and carburettor.

Figure 7.1 shows the positions that are available to mount the radiator in the car. There are many other positions that a radiator could be placed. However all of these positions are far from optimal.

Figure 7.1 Radiator positions.

7.1.2 Position 1 - Mounted at the Front of the Car

The advantages of positioning the radiator at the front of the car include:

1. Removed from all heat sources; and
2. Easy to mount;

The disadvantages of positioning the radiator at the front of the car include:

1. Significantly longer piping required;
2. Extra weight of the water and the plumbing;
3. Higher pumping power required to circulate the water;
4. Hot tubing has to run past or under the driver, creating a safety hazard;
5. May increase the coefficient of drag of the car;
6. Extra expense incurred as the pump and the tubing would need upgrading; and
7. Extra ducting required, to remove the hot air from the radiator.

7.1.3 Position 2 – Mounted in a side pod

The advantages of mounting the radiator in a side pod are as follows:

1. The heat exchanger is in front of the engine;
2. Easy to make provisions for a second radiator without many modifications to the bodywork of the car; and
3. The ducting may facilitate a larger quantity of air the flow into the radiator.

The disadvantage of this position as opposed to placing it at the front of the car is that it is closer to the engine.

7.1.4 Summary of Radiator Positions

Table 7.1 summarises the optimum radiator positions. From table 7.1 it is clear that the optimum position for the radiator is in a side pod. This location is the most practical in terms of mounting, availability of cold air and the lack of close heat sources. It was recommended to the body designer that car incorporate two side pods with large vents to duct the air into the radiator. Although it was hoped that the car would require only the original motorcycle radiator, the second vent was recommended to vent fresh air over the engine. It was considered that should an extra radiator be required a second side pod would facilitate its placement. It was also suggested that the second pod should be designed so that the introduction of the second radiator can be mounted without significant modification to the fibreglass bodywork. It was also suggested that the pods should be designed so that the wheels and suspension components do not impede the airflow significantly.

Radiator Position	Position 1	Position 2
Cost	Satisfactory	Good
Ease of mounting	Good	Good
Airflow conditions	Good/Satisfactory	Good
Affect on aerodynamics	Satisfactory	Satisfactory
Driver safety	Poor	Satisfactory
Accessibility of drive train & engine	Good	Good
Amount of piping & overall mass	Poor	Good
Ability to upgrade	Poor	Good
Proximity to heat sources	Good	Satisfactory
Overall Ranking	Satisfactory to Good	Good

Table 7.1 Summary of radiator locations.

7.2 Heat Exchanger Design

The amount of heat energy that the radiator must absorb is largely a function of the output power of the engine. As a rule of thumb the heat energy that the radiator must absorb is equal to the mechanical power output of a gasoline engine. This is due to the characteristics of gasoline engines. The thermal efficiency of a gasoline engine is limited to approximately 28% (Garrett, Newton & Steeds, 2001, p90 et. al.). The remainder of the heat energy is lost to the exhaust gases, cooling water and in radiation. The cooling water constitutes approximately half of these losses. Therefore the power output of the engine is approximately equal to the cooling capacity required. This balance of energy is shown in figure 7.2.

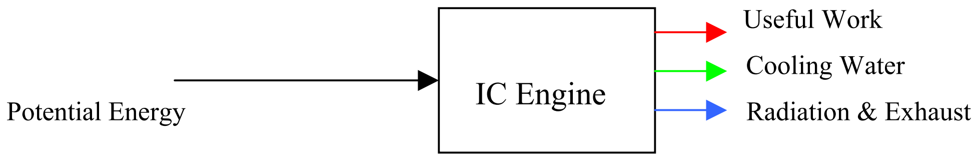


Figure 7.2 Energy balance of an internal combustion engine.

7.2.1 Heat Exchanger Specification

Heat transfer is not an exact science; it involves the use of empirical relationships to determine if a radiator will be suitable for a specific application. When specifying a radiator for an application, many factors must be considered (Kreith, F. & Bohn, S., p305). However, the factor that is of the most significance is the temperature of the water after it passes through the radiator and returns to the engine. The temperature of the water as it enters the engine is determined by the engine manufacturers guidelines and must be sustained in order to avoid overheating and premature failure of engine components. To determine the radiator water exit temperature the following factors must be known or determined:

1. The maximum heat energy that the radiator must dissipate;
2. The mass flow rate of the air into the radiator
3. The mass flow rate of the water into the radiator;
4. The effectiveness of the radiator; and
5. The inlet temperature of the air.

At first it could be thought that these values will not change between the Formula SAE car and the motorcycle that the engine was sourced from. However, the mass flow rate of the water is the only value that remains constant between the two operating conditions.

It is reasonable to assume that the radiator was designed to dissipate the maximum amount of heat that the engine can produce in the worst possible airflow situation. This situation would most likely occur when the engine is producing peak power and the vehicle is in first gear.

In the case of the Formula SAE engine the heat energy that radiator must dissipate can be *estimated* as 80% of the original maximum power output of the engine. As the power output of the original engine is 56kW the estimated power output of the Formula SAE engine is 44.8kW.

The mass flow rate of the air is calculated by finding the product of the air density, the cross-sectional area of the duct and the velocity at which the air is travelling. The maximum velocity of the motorcycle when it is in first gear and producing peak power is 58km/hr. Because the gearing of the car is different to the gearing of the motorcycle the maximum speed that is produced by the car in first gear at peak power is 49.7km/hr (velocity values were calculated using Matlab program *accelerations*).. This means that the loss of power of Formula SAE engine may be offset by the loss of air speed through the radiator.

The maximum velocities that the air will travel through the radiator at are not entirely relevant. The velocity of the air that enters the radiator is largely dependent on track conditions. If the car spends a lot of time cornering the velocity of the air entering the radiator will not be as great as it would be if the car was travelling in a straight line. The density of the air also changes due to the viscous forces that act on the air in the duct. The magnitude of the viscous forces depends on surface roughness and surface area. Therefore the magnitude of the viscous forces and hence the density of the air that enters the car's radiator will be different to the density of the air that enters the bike's radiator.

Another hurdle is determining the inlet temperature of the air. To obtain the inlet temperature of the air without experimentation, convection heat transfer methods can be used. Unfortunately heat transfer is very inexact science, which involves the use of empirical relationships to obtain temperatures. Because these relationships have been derived experimentally the answers that are obtained are only accurate within ±20%

(Kreith, F. & Bohn, S., p305 et. al.). However it can be assumed that the amount of convective heat transfer between the engine and the radiator would be less in the car than the motorcycle. The reason for this is that the radiator is situated further away from the engine in the car than it is in the motorcycle.

The effectiveness of the radiator is usually found from radiator manufacturers specifications. However heat exchanger effectiveness is a function of the airflow conditions, so varies with different operating conditions (Kreith, F. & Bohn, S., p501).

It would be possible to estimate the values of many of the aforementioned variables. However it was considered that the magnitude of the error would become too great to yield any meaningful results. For this reason it was decided that some experimental data needed to be gathered.

7.3 Heat Exchanger Testing

Throughout this dissertation it has been cited many times that budgetary and time restraints were the greatest obstacle in this project. In order to design a heat exchanger for an engine requires comprehensive testing (Ferguson, C.R. & Kirkpatrick, A.T. 2001). For this reason a 'ground-up' approach was not taken. Because of the cost involved in purchasing a replacement radiator and the fact that many of the conditions that are required to specify the radiator were unknown it was decided that the original radiator would be tested.

In order to simulate driving conditions it was hoped that a dynamometer could be used. This would allow the application of load onto the engine. The reason that this is important is because the engine produces the most power when it is loaded. It was envisaged that a large variable speed fan could be used to simulate the airflow that the car would encounter in operation. While the engine was operating under load it was planned that the temperatures would be monitored. To simulate cornering conditions, the temperatures would have also been monitored without a fan. Had the temperature outlet temperature recordings been within the limits specified by Yamaha it would have been possible to test the radiator in real driving conditions. Unfortunately due to the reasons outlined in section 3.10 these tests were not possible. Therefore it is not possible to conclude whether the standard motorcycle radiator was sufficient for the Formula SAE car.

7.4 Conclusion

In this chapter the optimum position for the radiator was found. Additionally some design considerations for the radiator have been outlined. It was found that it was not possible to accurately the predict whether the original motorcycle would be sufficient for the Formula SAE car using empirical relationships. Testing to ensure that the radiator was sufficient was planned but could not be carried out. With this in mind, it was recommended that the ducting system be designed with the potential for future upgrades.

8 Design Evaluation

8.1 Introduction

In this chapter the designs that evolved will be evaluated against the design objectives that were outlined in chapter 2.

8.2 Engine Specification Evaluation

In section 2.4 it was stated that the engine must meet a number of requirements in order for USQ Motorsport to be competitive in the Formula SAE competition. The requirements were as follows:

1. Acceleration performance;
2. Reliability;
3. Reproducible;
4. Fuel-efficient; and
5. Cost effective.

In order to specify an engine that best fulfilled these requirements, predictive modelling and an analysis of the relative importance of each of these events was carried out. Using these methods the engine that best fulfilled these requirements was specified. It is not possible to verify that the engine that was specified did in fact fulfil these requirements because the funding that was allocated to the engine did not allow the purchase of the engine that was specified. Furthermore, in order for any engine to be evaluated requires comprehensive testing on a dynamometer and on the track. At the time of writing neither of these tests were possible.

8.3 Fuel Mixture Preparation System Evaluation

The design requirements of the fuel mixture preparation system that were outlined in section 2.9 are as follows:

6. Cost effective;
7. Easily tuned;
8. Easily obtainable;
9. Wide range of parts must be available;
10. Provide the correct fuel/air ratio over all operating conditions;

A constant velocity carburettor was specified to fulfil these requirements. The carburettor was cost effective and easily obtainable as it was sourced from the engine that was purchased for the USQ Motorsport Formula SAE car. Because of the simple design of the carburettor it was found that it was easy to tune. It was also found that a wide range of parts including needles and jets were available for the carburettor. It is currently uncertain whether the carburettor is capable of providing the correct fuel/air ratio a across all operating conditions. This uncertainty has come about because it was not possible to perform dynamometer testing or an analysis of the exhaust gases. The reasons that these tests were not performed are outlined in section 3.10.

8.4 Intake Restrictor Design Evaluation

In section 2.8 it was stated that the goal when designing the restrictor is to achieve the maximum flow rate. From the flow bench tests results, it is believed that this has been achieved. However due to the lack of dynamometer testing it cannot be verified that the restrictor that showed the greatest potential on the flow bench will allow the engine to produce the most power.

8.5 Intake Manifold Design Evaluation

In section 2.10 it was stated that the intake manifold for the Formula SAE engine was required to facilitate the maximum engine performance possible and to be cost effective. The prototype manifold that was produced was definitely cost effective. The material for the plenum chamber was sourced from a gate manufacturer for $5. The material that was required to make the other features of the prototype manifold was sourced from the USQ mechanical workshop scrap material pile. Therefore the total outlay for the material was $5. Had the prototype intake manifold proved successful a final intake manifold would have been manufactured using aluminium. The cost of suitable aluminium tubing for the final manifold would have been $86 (AlQuip, 2004, pers. comm. 12 July).

The literature that was surveyed indicated that the manifold configuration that was designed would facilitate the maximum engine performance. Again, it cannot be verified, due to the lack of experimental data. Nevertheless, it can be stated that the intake manifold did perform its most basic function. The prototype intake manifold provided a path for the fuel/air mixture to travel from the carburettor to the engine cylinders.

8.6 Exhaust System Evaluation

In section 2.11 it was declared that the general design requirements of the exhaust system were as follows:

4. Enhance engine performance as much as possible;
5. Meet the Formula SAE sound level requirement of 110dB(A);
6. Resource efficient;

It was found that specifying the exhaust system rather than designing the exhaust system was the most efficient utilisation of resources. The muffler that was specified had a sound level rating of 98dB(A), deeming it within the requirements of the Formula SAE competition. The manufacturer of the muffler that was specified, had also proved that the muffler did improve the performance of the engine. The muffler that was specified was also within the budgetary constraints of the team.

The original Yamaha FZR600 exhaust manifold was specified for the Formula Sae car. In chapter 6 it was reasoned that this system was efficient and did require modification or removal. Unfortunately, it was not possible to perform dynamometer testing to verify that the exhaust system that was specified would provide superior performance.

8.7 Conclusion

In section 2.13 it was stated that all of the engine components must function as a system in order to optimise the performance of the engine. Many tests were conducted with all of the systems fitted to the engine (apart from the specified exhaust system). However it can only be concluded that all of the components that were designed and specified did function as a system. Because it was not possible to test the system on the dynamometer or in driving conditions the results of all of the tests are inconclusive.

9 Conclusion

9.1 Achievement of Objectives

The main aims of this project were to select a suitable engine and design a fuelling system for USQ Motorsport's Formula SAE-A racing car. In order to achieve these aims, a list of objectives was outlined in chapter 1. These objectives were achieved as follows:

1. A wide variety of engine types and designs were reviewed and analysed. The engine designs that were researched in this project included: combustion ignition engines, spark ignition engines, rotary engines, conventional reciprocating engines, four-stroke engines and two-stroke engines;
2. A predictive model of the acceleration performance and 75metre times of the car was constructed. Research was also undertaken to determine other factors that affect engine suitability. Using these tools and information the most suitable engine for the Formula SAE car was chosen;
3. A feasibility study was under taken to determine if forced air induction was suitable for the Formula SAE engine. The principle of operation,

cost, benefits and disadvantages of turbochargers and superchargers was covered in detail;

4. An analysis of the feasibility of implementing multi-point electronic fuel injection on the current Formula SAE engine was assessed;
5. Fixed venturi and variable venturi carburettors were investigated and a carburettor was specified for the Formula SAE engine;
6. The factors that affect the performance of the intake restrictor were researched and an intake restrictor was designed accordingly;
7. Various intake manifold configurations were examined, including log, streamlined and tuned configurations. An intake manifold was designed considering the resources that were available;
8. A review of the parameters that effect exhaust design was undertaken and an exhaust system was specified.

In addition to achieving the main objectives listed above, there were several other achievements in the course of this project. These included:

1. Sponsorship for testing facilities was obtained;
2. The optimum position for the radiator was found;
3. Tests were devised to determine the suitability of the motorcycle engine's radiator in the Formula SAE car.

In order to test the designs that evolved in this project it was planned that the designs would be constructed and tested experimentally. This was achieved with a varying degree of success, but was beyond the control of the author.

1. The intake restrictor was constructed and tested;
2. A prototype intake manifold was constructed and partially tested;
3. The mixture preparation system was obtained and partially tested;
4. The exhaust manifold was obtained and not tested;
5. The muffler that was specified was not obtained.

9.2 System Performance

It is currently unclear if the performance of the system matches the requirements that were set out in chapter 2. This is due to the lack of valid testing that has been carried out.

9.3 Further Work

It is hoped that the car will be at a stage of completion that will allow enough dynamometer testing to at least calibrate the carburettor and test the cooling system before the 2004 Formula SAE competition.

9.4 Recommendations to Future USQ Motorsport Members

If students choose to pursue the Formula SAE competition in the future, the following subsections provide some recommendations that may be useful.

9.4.1 Fuel Mixture Preparation System Recommendations

It is recommended that electronic fuel injection be implemented on the car in the future. The reasons for this are outlined in chapter 6. It may be worthwhile trying to seek sponsorship for these components. This should be easier to achieve in future years because a prototype of the car has been constructed.

9.4.2 Engine Selection Recommendations

The 1993 Yamaha engine that is currently fitted to the USQ Motorsport Formula SAE car could be considered as a ‘relic’ compared to the engines that most of the other Formula SAE teams are using. Figure 9.1 displays a prediction of the acceleration performance of the current USQ Motorsport engine compared to the performance of the ‘optimum engine’ that was selected in chapter 4. It would be possible to enhance the performance of the current engine to a high level, however it is thought that it would be more cost effective to simply obtain a later model engine.

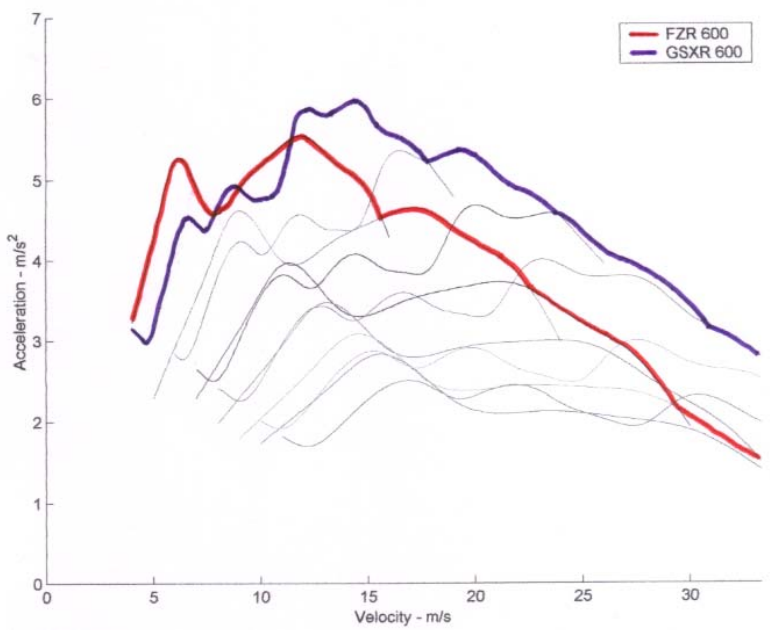


Figure 9.1 Comparative velocity versus acceleration curves.

9.4.3 Intake Manifold

Throughout this project it proved very difficult to obtain empirical relationships that accurately described the flow regime in the induction gas. It was found that CFD software is commonly used in industry to design all of the gas exchange system components. For this reason it is highly recommended that if future students wish to pursue the design of these systems that CFD software is implemented. It is also recommended that experimental testing with a dynamometer and a flow bench is incorporated in the development of the intake manifold.

9.4.4 Exhaust System Recommendations

It is suggested that future USQ engine development teams devote resources towards the induction system rather than the exhaust system. The reason for this is because more performance gains can be obtained through the optimisation of the induction system of the Formula SAE car. The reasons for this are outlined in Chapter 6.

9.4.5 Radiator Recommendations

It is recommended that a considerable amount of resources are devoted to the design and testing of the cooling system in the future if the current radiator is not sufficient.

In this project it was found that the development of a heat exchanger is a rigorous process. In order to properly design a cooling system an entire project could be devoted to it.

9.4.6 Programming Recommendations

The programs that were developed in this project were only designed to take into consideration the amount of torque that the engine can produce. However, the cars geometry and suspension set-up play an important part in determining the amount of tractive force that can be applied to the wheel without slip.

Once dynamometer testing has been performed and an accurate torque curve has been obtained it will be possible to include the aforementioned effects into the model that was produced in this project. This may help in the overall design of the vehicle. The equation that is used to take into account traction limited acceleration is provided in Appendix C.

10 References

Ahlstrand, A. & Haynes, J. 1998, Yamaha FZR600, 750 & 1000 Fours Service and Repair Manual (89-97), Haynes Publishing. USA.

Aird, F. & Elston M. 1997*, How to Tune & Modify – Carburetor Performance*, Motorbooks International, USA.

Akrapovic Exhausts [Online] Available: www.akrapovic-exhaust.com.au, [Accessed 19 August 2003]

Autospeed [Online]Available: www.autospeed.com/cms/A_0819/article.html, [Accessed 10 July 2004]

Bell, G. 1998, Four-Stroke Performance Tuning Second Edition, Haynes Publishing, UK.

Bianuci, S. *Ducati Tech Café,* [Online], Available: www.ducati.com/bikes/techcafe, [Accessed 23 September 2003]

Bird Precision [Online] Avaliable: www.birdprecision.com, [Accessed 15 March 2004]

Bikepoint [Online], Available: http://bikepoint.ninemsn.com.au/ [Accessed 23 March 2004]

Bikepoint [Online], Available: http://bikepoint.ninemsn.com.au/ [Accessed 18 September 2003]

Bowdler, J. 2004, *Performance Streetbike Number 89*, FPC Magazines, Australia.

C.A.P.A Performance [Online], Available: http://www.capa.com.au, [Accessed, 23 October 2004]

Curtin Motorsport [Online], Available: www.cage.curtin.edu.au, [Accessed 12 July 2004]

Deakin University SAE team [Online], Available: http://www.et.deakin.edu.au/SAE/ - [Accessed 21 July 2004]

Eastwell J. 2002, *Thermodynamics Study Book*, University of Southern Queensland, Australia.

Factorypro [Online] Available: www.factorypro.com/tech/, [Accessed 9 August 2004]

Faith, N. 1976, *The Wankel engine: the story of a revolutionary engine*, Butler and Tanner Ltd, Great Britain.

Ferguson, C.R. 1986, *Internal Combustion Engines*, John Whiley and Sons, USA.

Ferguson, C.R. & Kirkpatrick A.T. 2001, *Internal Combustion Engines Second Edition*, John Whiley and Sons, USA.

Garrett T.K., Steeds W. & Newton, K. 2001, *The Motor Vehicle Thirteenth Edition*, Butterworth-Heinmann, UK.

Formula SAE-A [Online] Available: http://www.sae-a.com.au/fsae/ [Accessed 18 May 2004]

Fox F. W. & McDonald A. T. 1998, *Introduction to Fluid Mechanics Fifth Edition*, John Whiley and Sons, USA.

Gillespie, T. D. 1992, *Fundamentals of Vehicle Dynamics*, Society of Automotive Engineers, USA.

Grape Racing [Online], Available: www.graperacing.com, [Accessed 9 August 2004]

Haile, J. 2000, *Motorcycle Tubocharging, Supercharging & Nitrous Oxide*, Whitehorse Press, USA

Heisler, H .1995, *Advanced Engine Technology*, Society of Automotive Engineers, UK.

Hayden, T. 2004, *Carburettor Basics* [Online] Available: www.geocites.com, [Viewed 2 September 2004]

Heywood, J. B. 1988, *Internal Combustion Engine Fundamentals*, McGraw-Hill, Inc, USA.

HowStuffWorks, *How rotary engines work* [Online], Available: http:/auto.howstuffworks.com/rotary-engine.htm/printable [Accessed 18 May 2004]

Irving, P. 1991 *Tuning for Speed 6th Edition*, Turton & Armstrong Publishers, Australia.

Japiske, D. & Baines, N.C.1994, *Introduction to Turbomachinery*, Oxford Press, UK.

Kitsios, V. 2003, *Optimisation of the gas exchange system for a formula SAE car*, University of Western Australia, Australia.

Kreith, F. & Bohn, S. 2001, *Principles of Heat Transfer Sixth Edition*, Brooks / Cole, USA.

Larry's FZR Homepage [Online] Available: (http://www.tourtimemedia.com/fzr/, [Accessed 18 August 2004]

Lewitt, E.H. 1965, *Thermodynamics Applied to Heat Engines Sixth Edition*, Sir Isaac Pitman & Sons Ltd, UK.

Limney, C. & J. 2002, *Performance Streetbike Number 71*, FPC Magazines, Australia.

Lumley, D. 1998, *Introduction to Engines*, McGraw-Hill, Inc, USA.

McFarland, J. 2004, *The Manifold Parts of the Induction System*, [Online] Available: www.N2performance.com, [Accessed 20 August 2003]

Mikuni Australia [Online], Available: www.mikunioz.com, [Accessed 13 August 2004]

Miller, R. 1996, *Flow Measurement Engineering Handbook Third Edition*, McGraw-Hill, United States.

Mini Mania [Online], Available: www.minimania.com, [Accessed July 26 2004]

Monash Motorsport [Online], Available: www.sae-a.com.au/fsae/downloads/newsletters/fsae_03_2004.pdf, [Accessed 13 May 2004]

Motorcycle News [Online] Available:
http://www.mcnews.com.au/MotorcycleRacing2004/Australian/Superbike/manufacturers.htm , [Accessed 15 September 2004]

MUR Motorsport News [Online], Available:

http://www.mame.mu.oz.au/formulasae/news/edition01.pdf, [Accessed 20 October 2004]

Royal Melbourne Institute of Technology Formula SAE 2003 [Online], Available: http://www.mm.rmit.edu.au/sae/, [Accessed 15 September 2003]

Serco [Online], Available: www.serco.com.au, [Accessed 23 September 2003]

Smith, H. P. 1971, *Scientific Design of Exhaust and Intake Systems*, Robert Bentley Automotive Publishers, USA

Turbotech [Online], Available: http://www.turbotech.com.au/, [Accessed 12 October 2004]

University of Aurburn Formula SAE [Online],Available:http://www.eng.auburn.edu, [Accessed 13 May 2004]

University of Central Florida Formula SAE [Online], Available: http://www.ucfmotorsports.cecs.ucf.edu/history.html , [Accessed 13 May 2004]

University of Queensland Formula SAE [Online], Available: http://www.uq.edu.au/fsae, [Accessed 10 August 2003]

Vizard, D. 1990, *How to Build Horsepower Volume 1*, SA Design Books, USA.

Walker, M. 2001, Performance Motorcycles Masterpieces of Engineering, Amber Books Ltd., UK

Wade, G. R. 1994, *SU Carburetters Tuning Tips & Techniques*, Brooklands Books Ltd, UK.

Ware, J. 2004, *Rapid Bikes*, Oct/Nov 2004, FPC Magazines Australia.

Weber North America [Online], Available: http://www.webernorthamerica.com/, [Accessed 12 August 2004]

World Supersport Series Official Homepage [Online], Available: http://www.superbike.it/, [Accessed 20 October 2004]

Yamaha Motor Company [Online] Available: www.yamahamotor.com.au, [Accessed 9 September 2003]

Yamaha Design Café [Online] Available: http://ymedc.introweb.nl/en/index.shtml, [Accessed 20 October 2004]

Yoshimura [Online] Available: www.yoshimura-rd.com, [Accessed 21 August 2003]