University of Southern Queensland
Faculty of Engineering and Surveying

Engine Optimisation and Performance Characteristics for a Formula SAE Race Car

A dissertation submitted by

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towards the degree of

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Abstract

This dissertation documents an investigation into determining methods of achieving optimised engine performance. Designs for improved intake and exhaust systems are also presented, and have formed an important focus of the project overall. Methods for modifying a dynamometer and results of engine testing using this equipment comprise another important aspect of this dissertation.

Optimisation of engine performance in this case was required for a specific application. Therefore, preliminary research involved identifying specific engine performance requirements in the application of Formula SAE, together with determining full specification of engine characteristics for a 1994 Yamaha YZF600 engine.

The most crucial aspects of engine operation requiring improved performance were identified, to determine directions for pursuing modifications to existing components. As a result, custom intake and exhaust systems have been designed for use on the YZF600, according to the relevant fundamental engineering principles and empirical correlations.

A mechanism for determining engine operating characteristics and quantifying and comparing engine performance also formed an integral component of this project. Research into potential dynamometer options was carried out to determine the most suitable option to be modified for use in Formula SAE engine testing.

Modification of this dynamometer enabled determination of benchmark performance characteristics; however, attempts to test the 2004 inlet manifold indicated that the dynamometer itself was not suitable for determining the performance of an engine in this configuration. Further research is required to determine conclusively the suitability of this dynamometer for use in this particular application.

The contribution to performance of the intake and exhaust systems designed in this project has yet to be determined. Testing of these components will proceed in the form of 'in-vehicle' testing, once the vehicle is complete and running. Therefore, determination and quantification of these systems' overall contribution to improved engine performance will be enabled.

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Limitations of Use

Certification

I certify that the ideas, designs and 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.

Melinda Rachel Plank

Q1122062X

Signature

Date

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Chapter1
Introduction

1.1 Introduction

This chapter provides an overview of the intentions and objectives of this research project. An introduction to the Formula SAE event and some associated regulations is also presented. Some of the achievements relating to the 2004 USQ Motorsport Team forms an important focus, and provides the basis for an understanding and appreciation of the need for this research project.

1.2 Formula SAE - The event and its Regulations

Formula SAE is a worldwide event aimed at the professional development of university students from a variety of disciplines. The competition encourages participation from engineering, business and marketing disciplines, and provides the opportunity for students to apply their skills and knowledge to a 'real-world' project.

The design brief offered by the Society of Automotive Engineers (SAE) commissions the conception, design and construction of a small, high-performance, low-cost, open-wheeled, open-cockpit, formula-style racing car, with marketing and business appeal. As such, this competition necessitates innovative and practical engineering design, supported by an intelligent business plan, and combined with a clever marketing strategy. Accordingly, the event's judging criteria rewards entries with the most intelligent combination of these attributes.

The entries must conform to the event's general regulations and safety requirements, as dictated by the *Formula SAE Rules* for the current year. Importantly, event rules state that the car must be conceived, designed and constructed entirely by students, in order that the members are presented with the best opportunity to acquire knowledge in all facets of the projectobjectives. As well as general regulations, there are also limitations placed on specific design areas. However, the event very much encourages innovation in design, and the regulations are strongly in support of this.

1.3 Project Background

1.3.1 USQ Motorsport 2004

USQ competed in Formula SAE-Australasia for the first time in 2004. USQ Motorsport was founded that same year as a facility for ensuring the continual development and growth of the project. This organisation also provides a means of recruiting students from various levels and disciplines to work together towards creating the optimal Formula SAE package for USQ.

In 2004, USQ Motorsport consisted primarily of final year Mechanical Engineering students, each completing research projects involving various aspects of the car's design. USQ's inaugural Formula SAE entry was designed entirely by these nine students, and constructed with great assistance from the USQ Mechanical Engineering Workshop. The team's design philosophy focussed strongly on maintaining simplicity in design, therefore ignoring any undue and unnecessary complexity. This design strategy intended to use commonly available (and reproduceable) components, ensuring ease of maintenance, and aimed to achieve affordability, aptly naming their creation, *Jettison 1* (Figure 1.1).

Figure 1.1: University of Southern Queensland's inaugural (2004) Formula SAE race car

The team successfully achieved their intended objectives by designing and constructing an integrated and reliable package, with their efforts rewarded with a commendable performance at the Formula SAE Australasian Championships in late 2004, in Werribee, Victoria.

1.3.2 Project Origin

In 2005, USQ Motorsport is focussed on designing and constructing a more competitive entry, through an evolutionary process of refinement and optimisation. Much useful information was gained from USQ's first experience participating in the event, and the team's aim this year is to focus on improving and refining certain elements of the foundational design in order to optimise the overall package. Consequently, this research project aims to achieve optimised engine performance for USQ’s 2005 race car.

In 2004, a final year research project was undertaken to determine a suitable engine for use in a Formula SAE race car. Having determined the characteristics required of an engine for use in Formula SAE, the next step is to determine methods of extracting maximum performance from this engine, in relation to the Formula SAE event. The process of achieving optimised engine performance will necessarily involve thorough exploration of the various engine modifications available for performance improvement, and identification of the most crucial aspects of engine performance requiring optimisation. Consequently, this research project aims to determine benchmark performance characteristics and explore various methods of modification to optimise engine performance.

1.4 Aim

All classes of motorsport demand certain performance requirements of the competing cars (Figure 1.2). These requirements vary according to the category involved, and require that all components on a particular race car be purpose-bred for optimum performance in the given conditions. Formula SAE, however, is a low-budget, student-based event, and as such, limited research and development resources are available for optimising various facets of car design.

Unlike most forms of motorsport, there are no purpose-built engines for use in Formula SAE. Short of designing and developing a custom SAE engine, the most ideal solution is to choose an engine most suited to this application, and then to perform particular modifications and design enhancements in order to optimise the performance of the chosen engine.

Figure 1.2: USQ's 2004 entry negotiating one of the many turns characteristic of the Formula SAE track, at the 2004 Australasian event

Engines used in Formula SAE competition are limited in capacity to no greater than 610 cc (*Formula SAE Rules* 2005); motorbike engines are therefore most suited to this application, as they are readily available and offer prices within team budgets. The restrictions placed on engines (refer to section 2.2.1) are intended to limit engine output in order to maximise safety. Therefore, to obtain an engine whose operation and performance is most suited to the application of Formula SAE, particular modifications are required. Identification of the demands of an engine in this application is integral in determining the particular modifications suitable, and these performance requirements are presented in section 2.5.

The aim of this project is to determine benchmark performance characteristics for the 2005 USQ Formula SAE engine, identify the requirements of an engine in the application of Formula SAE, and hence determine the most suitable modifications for achieving optimised engine performance. Achievement of these objectives will result in an engine with characteristics specifically suited to the demands of Formula SAE.

1.5 Objectives

A number of objectives have been determined to facilitate the progressive completion of this project. These objectives comprise:

1. Conduct background research into Formula SAE Rules and 2005 addendum to these rules, as well as FSAE limitations and guidelines pertaining to engine performance.

2. Determine mechanism for measuring engine performance, and assist with establishing related equipment.

3. Establish full specification and performance characteristics of 2005 engine.

4. Investigate possible methods of improving engine performance.

5. Design systems to implement improvements.

6. Conduct appropriate testing to quantify performance improvements (using equipment established in Objective 2).

7. Hence determine overall performance improvement gained.

As time and resources permit:

8. Implement further performance improvement methods.

9. Conduct appropriate testing to quantify further performance gains.

1.6 Dissertation Overview

This dissertation provides a record of the methodologies implemented to achieve the specified objectives, as well as discussion of the results obtained through the course of this project, and recommendations for further work. An outline of this dissertation follows:

Chapter 2 provides the basis from which engine performance optimisation will be developed, presenting background information regarding engine limitations as regulated by Formula SAE, in addition to the performance requirements of engines in this competition.

Chapter 3 presents the methodology to be employed in the pursuit of achieving the specified project objectives.

Chapter 4 presents a discussion on a range of engine modifications available for increasing engine efficiency, power and overall performance.

Chapter 5 presents fundamental design theory relating to intake systems to form the basis from which design of the 2005 intake system will proceed.

Chapter 6 focuses on the 2005 intake design. The design procedure for each of the components forming the overall intake system is detailed, with a number of preliminary designs presented. This chapter shows the evolution of these preliminary designs into the final intake design for the 2005 race car.

Chapter 7 explains the fundamental operating principles of the two available methods of fuel delivery: carburetion and electronic fuel injection. Each system is analysed in terms of relative advantages and disadvantages, to enable determination of the most suitable system for delivering fuel to the car's engine.

Chapter 8 details the principles governing operation and design of exhaust systems. Design theories, including wave tuning, flow restriction minimisation and reduction in charge contamination, are explained to lay the platform for performance exhaust system design.

Chapter 9 extends the fundamental theory presented in Chapter 8 to form the design of the 2005 exhaust system. Empirical data and fundamental engineering principles are combined to achieve a design that provides benefits for increased charge potential, fuel efficiency and power production.

Chapter 10 discusses the need for a reliable and suitable engine testing facility, to enable continued engine development and improvement. The process involved in modifying an engine dynamometer for use in developing Formula SAE engine performance is detailed.

Chapter 11 presents the results of engine testing performed as part of this project. These results are discussed in terms of their significance to engine performance, in addition to the suitability of the dynamometer used.

Chapter 12 provides a conclusion to the methods carried out in this project and recommendations for further work. This chapter concludes with discussion of the achievement of project objectives, which indicates the success of this project in achieving optimised engine performance.

1.7 Conclusion

This chapter has provided an introduction to the Formula SAE Competition, some general rules pertaining to overall car design and the main intentions desired to be achieved through this project. Background information, forming the basis for development of engine performance, follows in Chapter 2.

Chapter 2
Background

2.1 Introduction

This chapter presents the boundaries within which engine modifications can be performed for the Formula SAE competition. The particular regulations pertaining to engine design are discussed, and the performance requirements of an FSAE engine are identified. Possible methods of performance improvement are also discussed. Overall, this chapter provides a basis for further research and development for determining design of improved systems for the 2005 race car.

2.2 2005 Formula SAE Competition Rules

The 2005 rules provide many general, and some specific, regulations pertaining to the various areas of design for cars participating in the competition. These rules are intended to encourage innovation in design and ensure safety for all competitors.

2.2.1 Engine Regulations

Rules pertaining specifically to engine design are minimal, providing significant opportunity for achieving improvements in performance. These regulations govern maximum engine capacity, engine type, air delivery method and fuel type, and are presented in the *2005 Formula SAE Rules* as:

1. Engine Type

- *engine(s) must be piston engine(s), petrol-fuelled and operate on a four-stroke cycle*

2. Engine Capacity

- *maximum engine displacement must not exceed 610 cc*

3. Method of Air Delivery

- *all airflow to the engine must pass through a single circular restrictor, 20 mm in diameter, located between the throttle and engine*

4. Fuel Type

- *only the fuel supplied by event organisers may be used by vehicles in competition (no fuel additives allowed)*

5. Forced Air Induction

- *forced air induction (turbocharging and supercharging) is allowed, but competitors must adhere to regulations regarding placement of the restrictor in relation to the throttle and compressor, and the system must be student-designed*

These regulations outlined will largely dictate the methods pursued for improving engine performance.

2.3 Engines Used in Formula SAE

Acknowledging the above restrictions, the most suitable engines for Formula SAE are motorbike engines. Generally, teams source these engines from motorbikes in the Supersport category, produced by the 'big four' Japanese bike manufacturers: Yamaha, Honda, Suzuki and Kawasaki.

Most commonly, 600 cc, four-cylinder engines are employed, although occasionally single-cylinder (endurance bike) engines have been used (RMIT 2004). The advantages of using a single-cylinder engine are reduction in engine weight and good torque characteristics. However, the requirement that the engine must breathe through a single restrictor at intake may have a greater detrimental effect on air flow ability for a single-cylinder compared to a multi-cylinder engine. In particular, a four-cylinder engine's flow may be much less impeded by the inclusion of this restrictor, than that of a single-cylinder. This is perhaps the reason for the majority of FSAE teams running four-cylinder engines.

2.4 2005 USQ Engine

To promote the continual development of knowledge and encourage familiarity with a particular engine, a decision was made early in this project to attempt to acquire an engine similar to that used in 2004 (1993 Yamaha FZR600). An extensive search indicated that acquiring such an engine at a reasonable price would be very difficult. However, in late May, the team acquired a 1994 Yamaha YZF600 motorbike (Figure 2.1), from which the engine (Figure 2.2) and a number of other components, including the electrical system, brake rotors and brake callipers, were sourced.

Figure 2.1 Donor bike - 1994 Yamaha YZF600

Torque and power charts available for this engine show good high-end performance characteristics (refer to Appendix C). Modifications will be necessary to improve low- to mid-range performance, as required by the nature of Formula SAE.

Through observation and research, it has been determined that this engine is very similar to the FZR engine used by the team in 2004, in terms of performance characteristics, physical size and weight, and internal components. There are a number of advantages in obtaining this similar engine, viz:

1. Familiarity

2. Engine performance able to be anticipated (from 2004 engine performance)

3. Compatibility with 2005 engine (enables interchangeable components)

4. Technical relationship formed with manufacturer

Figure 2.2: 1994 Yamaha YZF600 engine for use in 2005 USQ race car

2.5 Engine Performance Requirements

Certain performance characteristics are demanded of an engine in the Formula SAE event. Identification of these requirements is critical in optimising engine performance.

An excellent indication of the performance requirements of an FSAE engine was gained through USQ's participation in the 2004 event. The SAE track generally incorporates a number of tight turns, narrow width and a relatively short straight. The performance characteristics demanded of an engine for use in the FSAE competition are identified as:

1. Good low-end torque and power

2. Excellent acceleration ability

3. Fuel efficiency

4. Reliability

Further explanation of these requirements follows.

2.5.1. Good low- to mid-range torque and power

Good torque and power throughout the rev range is most desirable; however, due to the nature of the SAE track, low-end torque is of critical importance. The track layout is such that engine speeds will generally be somewhere in the mid- rev range (4 000 – 8 000 rpm); typically, motorbike engines produce peak torque and power at about 10 000 – 11 000 rpm. Therefore, it is required that suitable modifications be performed such that low- to mid-range power and torque are increased.

2.5.2. Excellent acceleration ability

The tight turns characteristic of the FSAE track require excellent acceleration capabilities from the vehicles. Some corners require the cars to turn almost 90º, meaning that the cars slow considerably through these corners. Therefore, good acceleration ability is required to regain speed out of corners.

Acceleration performance is also required to enable effective overtaking. The endurance event is the only FSAE event that facilitates overtaking.; therefore, to ensure the USQ entry has adequate ability to overtake rivals, it must possess excellent acceleration characteristics.

2.5.3. Fuel efficiency

Economy of fuel is an important characteristic of a performance engine. The endurance event offers points rewards for the most fuel efficient entry, and as such, fuel efficiency is of high priority in engine performance design.

2.5.4 Reliability

Reliability is one of the most important aspects of engine performance. An engine's ability to produce very high amounts of torque and power requires that its components be sufficiently strong to handle these excessive forces and stresses.

2.6 Intentions for Optimising Engine Performance

There are numerous modification options available for improving engine performance. A general list of modifications, which will be considered further in Chapter 4, includes:

- intake system improvement (flow optimisation)
- exhaust system improvement (improve scavenging ability)
- implementation of electronic fuel injection
- porting
- varying camshaft (valve) timing
- use of forced air induction
- use of high-performance internal components
- varying compression ratio
- varying firing order

2.7 Conclusion

The Formula SAE Rules regarding engine design have been introduced, and the performance requirements of an engine in the application of FSAE identified. The types of engines generally used in Formula SAE cars were discussed, and the engine acquired for use in 2005 was introduced. Chapter 3 presents the methodology intended to facilitate achievement of performance optimisation.

Chapter 3
Methodology

3.1 Introduction

An appropriate and logical methodology is integral to the successful completion of any research project. Following a clear and concise method of work will enable the progressive attainment of project objectives. Here, the methodology proposed for this project is presented, and its role in achieving the project objectives is discussed.

3.2 Mechanisms of Methodology

Attainment of the project objectives is very much reliant on the successful implementation of the proposed methodology. This methodology will largely rely on the following mechanisms:

1. Review and use of previous work

2. Theoretical Analysis

3. Empirical Analysis

The main mechanism of measuring performance improvement in this project will be via empirical methods. Theoretical analysis will be employed, where applicable, to substantiate experimental results, determine the viability of proposed methods of performance improvement, and quantifiably estimate anticipated gains in situations where repeated empirical analysis may not be viable. Previous work in this field will provide a basis and direction for continued work. Review and critical evaluation of such previous work is essential to enable informed decisions, and to avoid 'reinventing the wheel'.

3.2.1 Review and use of previous work

Previously established ideas and methods relevant to this project will contribute to providing a basis and direction for future work. This existing information will be used where possible to enhance the research and development process. Some useful data on engine performance exists for the engine under scrutiny. However, as Formula SAE is a somewhat specialised application, much of the fundamental research and data acquisition relating to engine performance in this application will necessarily be performed through the course of this project.

3.2.2 Theoretical Analysis

Theoretical methods of analysis will be used in determining the anticipated effect of various engine modifications on overall engine performance. Fundamental engineering theory will be applied to develop directions for designing improved engine-related systems. The main advantage of employing theoretical analysis is that, for a given situation, all parameters affecting engine performance can be varied successively and repeatedly, and their affect theoretically determined, for relatively low cost and little time, compared with experimental analysis of the same nature. However, theoretical results must be treated with some caution, as in practice, a combination of varied parameters may occasionally produce unexpected results.

3.2.3 Empirical Analysis

As mentioned, the main mode of determining relative engine performance gains will be via empirical methods. Dynamometer testing will be performed initially to determine benchmark performance characteristics, and will be the foremost mode of determining relative performance gains. Once particular methods of modification have been chosen and applied to the engine, their effect on performance will be measured through further dynamometer testing.

3.3 Methodology for Determining Modifications for Implementation

Recognition of need is the first critical step in all engineering design processes (Ertas & Jones 1996). Thus, in order to determine methods of achieving optimal engine output, the requirements of an engine in the application of the Formula SAE competition must first be identified. Once known, ways of achieving this output can then be researched and compared, to determine the most effective methods of optimising engine performance.

The general process for determining the modifications to be implemented is:

1. Determine benchmark performance characteristics of engine
2. Identify requirements of an engine in the application of Formula SAE
3. Therefore, determine particular areas of performance that require improvement
4. Conduct research into methods of achieving this performance
5. Determine modifications to be implemented according to performance requirements

Optimisation of engine performance will be pursued through:

1. Intake system design

2. Exhaust system design

The standard motorbike intake system comprises separate carburettors feeding each cylinder. Therefore, a custom intake system will be designed, with the intention of optimising flow of intake charge to the cylinders (Chapter 6).

A custom extractor system will be designed for the exhaust system, with lengths tuned for low speeds, to achieve improved low-end torque and power (Chapter 9).

3.4 Intake and Exhaust Design Optimisation

Due to the restrictions placed on the amount of airflow available to the engine by inclusion of a restrictor, it is critical that the available air is used to its full potential. Integral to this is optimising the intake restrictor shape and flow through the intake manifold.

3.4.1 Intake Restrictor Design

In 2004, the research project undertaken by Travis Mauger, *Selection of an Engine and Design of the Fuelling System for a Formula SAE Car*, incorporated flow measurement testing of various restrictor designs. From those tested, the restrictor shape producing the highest flow was determined to be the conical converging-diverging nozzle. Restrictor design is an area of critical importance in ensuring maximum flow to the engine, and as such, this project will devote appropriate research and development time to this particular aspect of design. Results of flow testing various restrictor designs in 2004 will be used as a significant reference point in further optimising restrictor design in 2005.

3.4.2 Intake Manifold Design

Inspection of the 2004 car's intake manifold indicates some areas requiring further development and improvement. The intake (and exhaust) manifolds are critical factors in improving overall engine performance. Therefore, this project will focus on optimised design of these systems, by conducting research into various manifolds. The methodology for this section will include:

1. Research various intake manifold designs to determine suitable options

2. Utilise fluid mechanics theory to determine flow characteristics

3. Design various intake configurations

4. Determine final intake design

5. Construct intake system

6. Perform dynamometer testing to quantify performance of intake design

3.4.3 Exhaust Manifold Design

Similar to the intake manifold, the exhaust system will be further developed and redesigned in 2005. Improving exhaust system operation can provide significant gains in engine power. Key to exhaust system design will be improving the engine's exhaust scavenging abilities and analysing wave harmonics.

The methodology for optimising the exhaust system is similar to that for intake design:

1. Research various exhaust manifold designs to determine suitable options

2. Design various exhaust configurations

3. Determine final exhaust system design

4. Construct exhaust system

5. Perform dynamometer testing to quantify performance of exhaust system

3.5 Methodology for Measuring Performance Improvement

The fundamental mode of determining relative performance gains will be through conduction of experimental methods. The most crucial aspect of these experiments will be to determine *relative* gains in performance. In other words, the specific amount of, say, additional power produced by a particular modification may not be as crucial as the relative output produced by that modification in comparison with another modification.

The basic methodology for determining the relative effect of varying certain parameters and implementing various modifications on engine performance can be generalised. Obviously, individual modifications may require slightly varied testing methodologies, but, for the most part, these methodologies can be generically specified:

1. Determine benchmark performance characteristics

2. Implement improved system/modification

3. Conduct appropriate testing to measure new output/performance

4. Determine relative gain or loss

5. Make appropriate changes to system with aim of achieving desired result (optimised performance)

6. Re-test system until desired result is achieved

3.6 Conclusion

The various methodologies to be employed in this project have been detailed and explained. Appropriate implementation of these methodologies should enable achievement of optimised performance.

Chapter 4
General Engine Performance Optimisation

4.1 Introduction

In the pursuit of improved engine performance, there are a variety of modifications available that can contribute to increased power, economy and overall performance. Choice of performance upgrades must be in accordance with the performance benefits desired. In this chapter, a range of engine modifications are presented, with some explanation regarding their contribution to performance enhancement.

4.2 Determining Appropriate Modifications

The range of modifications available vary in their abilities to improve performance, cost and complexity to implement. Most fundamentally, the requirements of engine performance must be clearly defined, in order that the possible modifications be accurately analysed to determine the most appropriate upgrade(s) to implement.

Individual modifications take significant time to implement. Some modification methods can generically be applied to a range of engines; however, more often, design for a specific engine is required, in order to achieve application-specific improvement.

Technical expertise is also quite often necessary, as well as additional testing and tuning after initial implementation.

In many cases, it is often more beneficial to develop fully a single modification, than to generically apply a range of upgrades. That is, by identifying the most significant aspect of engine performance requiring improvement, and developing a modification accordingly, more significant benefits are generally seen than in implementing upgrades which have not been fully developed or tested.

The process of identification, design, implementation and testing can be lengthy. Initially, this project was determined to explore and implement a number of engine modifications. However, it soon became apparent that the allocated time would not justify full development of several upgrade options. Identification of the anticipated most beneficial design modifications, in terms of achieving improved performance in the SAE competition, was then required to determine the direction for progress.

Requirement of an intake restrictor dictated that achieving maximum intake flow to the engine would be crucial; also, it was deemed that design of a custom exhaust system would further improve engine efficiency, power and overall operation. Therefore, it was determined to focus on designing custom intake and exhaust systems, and to develop a means of testing these systems, in order to achieve significantly improved engine performance, and overall vehicle performance, for USQ's 2005 race car.

4.3 General Engine Modifications

In addition to redesigning intake and exhaust systems, general engine modifications that may be pursued, given more time, include:

- porting
- varying camshaft (valve) timing
- use of high performance internal components
- varying intake manifold length (use of spacers)
- varying method of fuel delivery
- use of forced induction systems

4.3.1 Porting

The term 'porting' refers to redefining the shape of inlet ports. Inlet ports provide the final passage for charge entering the combustion chamber. The shape of these ports is therefore crucial in assisting the maximum amount of charge to enter the cylinders. Inlet ports must enable sufficient intake velocity, while providing facility for fuel to remain in an atomised state.

A more steeply angled intake port relative to the valve, such as in the Lotus cylinder head shown in Figure 4.1, compared with the Fiat cylinder head illustrated in Figure 4.2, provides a much more direct flow path.

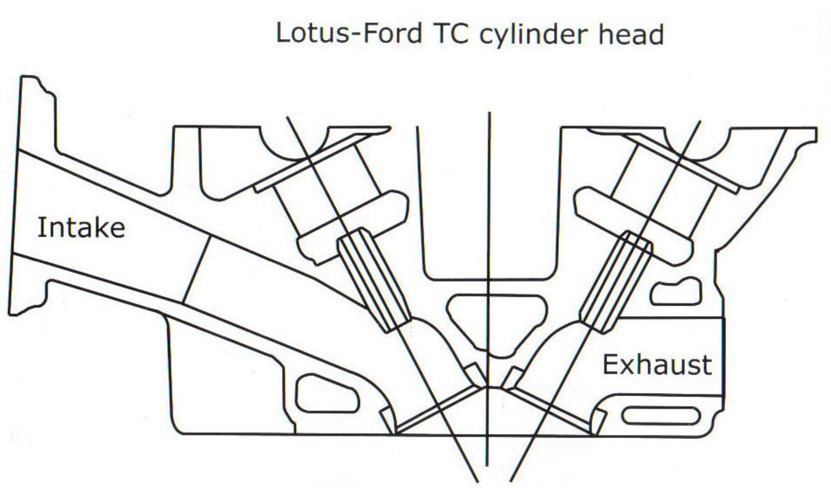


Figure 4.1: 1960's Lotus-Ford high performance inlet port

(Source: *Race Performance Magazine* Issue 1 2005)

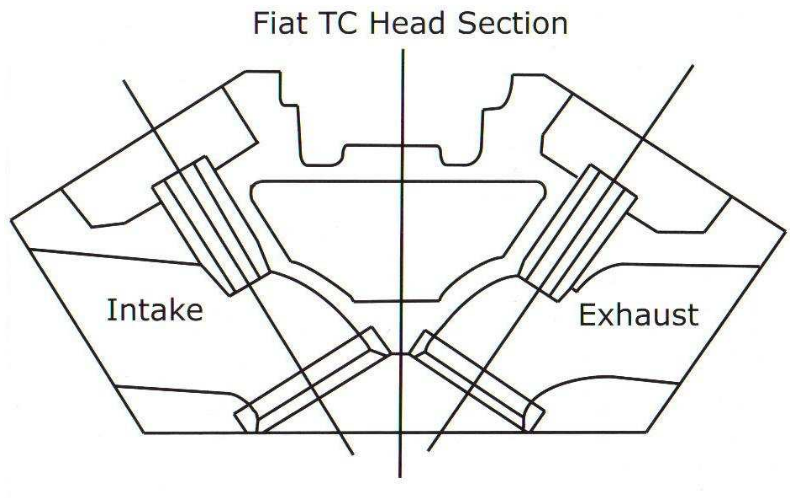


Figure 4.2: 1960's Fiat inlet port – flat, inefficient intake path

(Source: *Race Performance Magazine* Issue 1 2005)

Race Performance Magazine (Issue 1 2005) states the advantages of this steeper intake shape as:

- straighter path into combustion chamber may help to maintain intake velocity

- effective valve area increased

- may improve fuel distribution due to larger effective valve area, leading to better combustion

Engine porting therefore involves modifying the shape of inlet tracts to create more efficient flow. Until some decades ago, inlet ports on standard road cars had shapes similar to that of the 1960's Fiat shown in Figure 4.2 (*Race Performance Magazine*, Issue 1 2005), characterised by flat flow paths, followed by a sharp turn into the combustion chamber. Such port shapes provide decreased charge momentum and increased losses due to high bend radii (Fox & McDonald 2003).

More efficient intake ports can be achieved through a combination of machining and using 'port putty'. Port putty is an adhesive substance that can be moulded into virtually any shape, and dries hard, becoming suitable for sanding or grinding. Port putty therefore provides an ideal method of increasing bend radii, thus reducing flow losses (Figure 4.3). Redirection of the flow path is achieved by machining out the cylinder head, to create a new, more direct inlet tract (Figure 4.4).

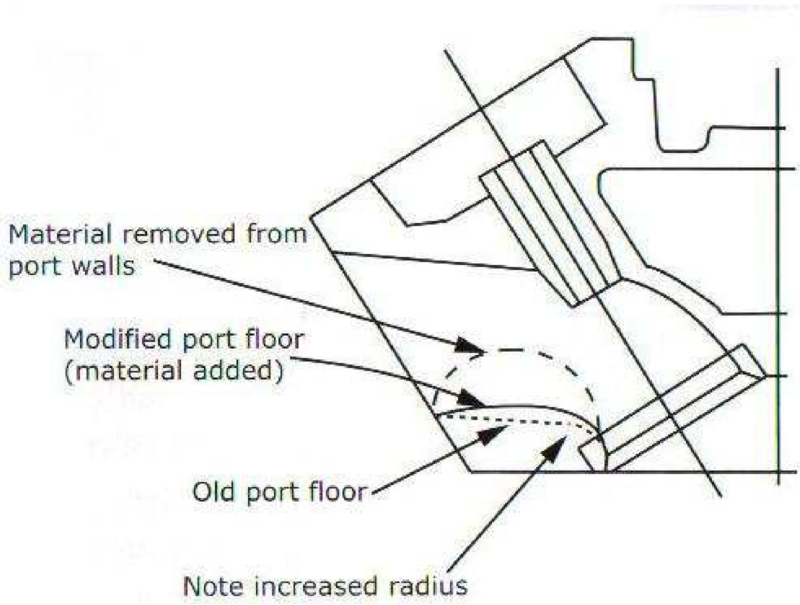


Figure 4.3: Use of port putty to increase bend radii

(Source: *Race Performance Magazine* Issue 1 2005)

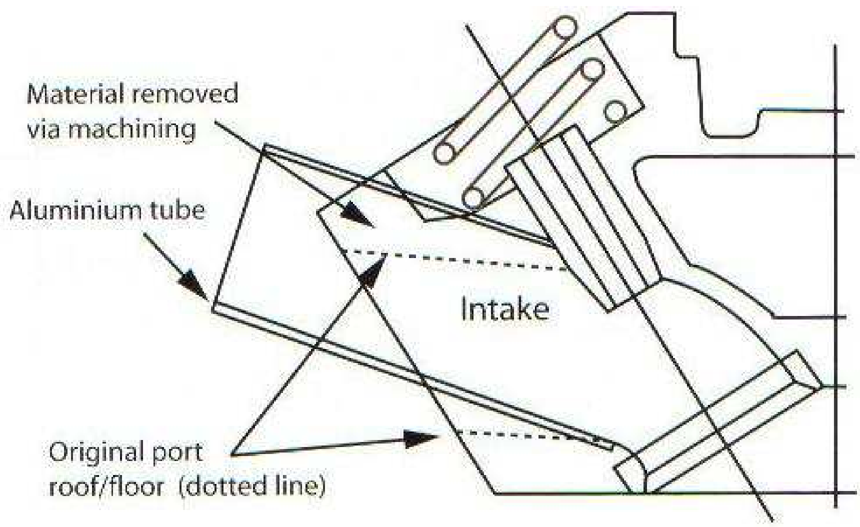


Figure 4.4: Machining a new flow path to increase port efficiency

(Source: *Race Performance Magazine* Issue 1 2005)

Use of port putty and re-machining ports thus enables creation of much more efficient inlet ports, and can markedly improve fuel economy and power.

Inlet ports in modern engines (c. 1990 onwards) are generally very efficient, providing direct, loss minimising paths for intake charge (Toowoomba Yamaha Service Manager 2005, pers. comm., 7 July 2005). This is evidenced in Figure 4.5, showing a photo of the very direct YZF600 inlet tract. Porting of an engine such as a 1994 Yamaha YZF600 would therefore contribute very minimally to performance improvement, if at all. However, some advantage may be gained by using port putty to reduce the diameter of inlet tracts, thus increasing intake velocity and improving combustion efficiency.

Figure 4.5: Steeply angled, direct inlet port of the 1994 YZF600

4.3.2 Camshaft (Valve) Timing

Camshafts control valve timing, and therefore dictate when inlet and exhaust valves open, as well as the duration of these openings. Valve opening durations affect the amount of intake charge able to enter the cylinders, as well as the ability of combustion gases to be removed.

Lobes (or ‘eccentrics’) on the camshaft (Figure 4.6) convert rotation of the cam into up and down movement of valves (Burgess & Gollan 2003).

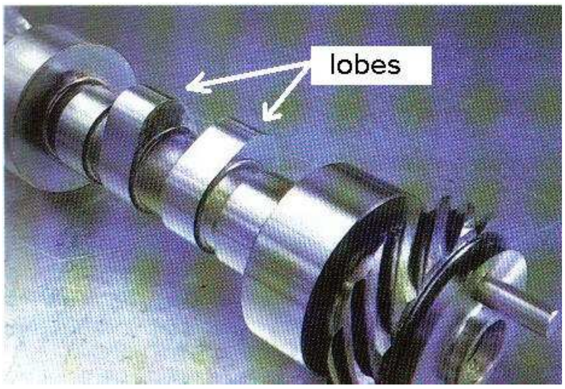


Figure 4.6: A typical camshaft

(Source: Edgar 2001)

Figure 4.7 provides some important terminology associated with camshaft lobes.

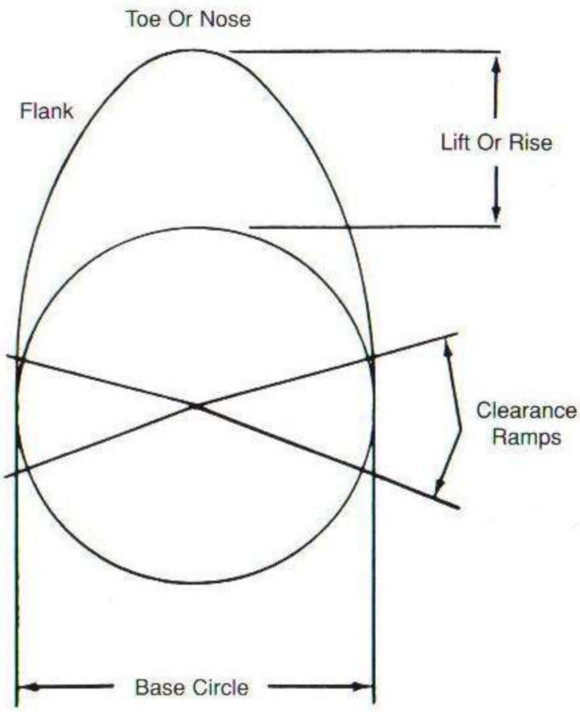


Figure 4.7: Lobe terminology

(Source: Edgar 2001)

The overall profile of a lobe is defined by the attributes shown in Figure 4.7, with this profile determining valve opening and closing characteristics. Therefore, in order to change timing, modifications can be made to lobe profiles. Some of these attributes and their critical functions are explained below:

- **Toe or Nose**

  Nose determines valve opening durations; a flatter nose increases the time for which valves remain open.

- **Lift or Rise**

  Lift or rise determines the vertical distance valves move from their seats (called 'valve lift'). A greater rise results in higher valve lift, which increases the physical opening for charge to flow into the combustion chamber.

- **Flank**

  The flank is the part of the lobe that enables opening and closing of the valves. The shape of the flank heavily influences the nature of valve movement. A steeper flank causes rapid opening or closing, while a more gently curved flank results in more gradual valve movement. Lobes can be ground with different flanks on the opening and closing sides, called 'asymmetric lobes', which may be used to induce rapid opening of the inlet or exhaust valve (steeper flank), while the closing movement occurs more slowly (Burgess & Gollan 2003).

Depending on the requirements of the engine, modifications to camshafts can be used to advance or delay valve opening, increase or decrease opening duration and increase the period of valve overlap (refer to section 8.4.4.1 *Valve Timing*). Ignition timing must also be considered in relation to valve timing, as changing opening times may require earlier or later ignition accordingly.

Appropriate modifications to valve and ignition timing can generally result in increased engine efficiency and overall power.

4.3.3 High Performance Internal Components

Standard internal engine components, such as pistons, connecting rods, valves and rings, can be replaced by high performance equivalents (Figure 4.8). These high performance parts usually offer reduced weight and higher strength.

Figure 4.8: High performance internal engine components, including valves, pistons and conrods, are used for reduced weight and additional strength
(Source: *www.manleyperformance.com* 2005)

Overall vehicle performance is improved with a better power-to-weight ratio. Reducing weight in all available areas of the vehicle therefore provides benefits to performance. A percentage of the power produced by an engine is lost through movement of the various internal components. The magnitude of this loss is determined largely by the mass of the components. Therefore, use of lightweight internals reduces the energy lost through this motion, increasing the power transmitted to the wheels.

As most modifications are intended to increase engine power, an important consideration is the strength of the engine's components themselves, along with other related vehicle components. In some applications, engine modification can result in substantial power increases. The engine's internals are therefore subjected to greater forces and higher temperatures, and must be rated accordingly.

Replacement kits, containing parts such as pistons, conrods, rings and valves made from high-strength alloys or titanium, are commonly available, and can withstand harsher operating conditions. Some modifications also necessitate replacement of the crankshaft, to a higher-strength version.

Compared to standard internals, these parts can be relatively expensive, but are necessary in some situations. Furthermore, upgrading to high-performance internals in accordance with a specific engine modification should prevent the possibility of severe damage being caused by inadequate-strength component failure.

4.3.4 Varying Intake Manifold Length

The effective length of inlet runners can be increased or decreased, according to tuning requirements. Rather than redesigning and re-manufacturing the manifold to produce the desired runner lengths, spacers can facilitate this function.

Bell (1997) presents empirical results to show that longer lengths generally improve low- to mid-range power. Spacers can be added at the entrance to inlet runners to increase the overall effective length of the intake (Figure 4.9).

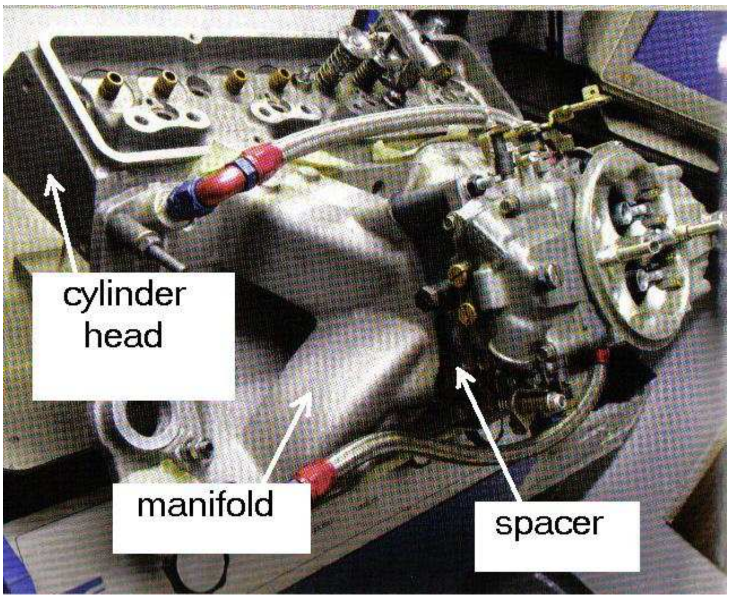


Figure 4.9: Positioning of manifold spacer at entrance to intake runners

(Source: *Performance Buildups Magazine* Vol. 15 No. 3)

A variety of spacers are available, varying in height, to add less or more length overall (Figure 4.10). The principle of these spacers is to change the wave tuning characteristics of the manifold; manifold length affects the timing of pressure waves (refer to section 8.4.5), and so the length is varied to take advantage of these waves at higher or lower speeds, as appropriate, resulting in a broader power band.

Figure 4.10: Some of the variety of manifold spacers available, each offering different wave tuning characteristics

(Source: *Performance Buildups Magazine* Vol. 15 No. 3)

Performance Buildups Magazine (Vol. 15 No. 3) lists some of the advantages of using manifold spacers as increased fuel delivery, better (throttle) response and improved track times. Testing, such as on a flow bench, is required, however, to determine the appropriate spacer(s) for a particular engine and application.

4.3.5 Varying Method of Fuel Delivery

Until electronic fuel injection (EFI) was developed, carburettors performed the function of fuel metering and delivery. In carburettors, air/fuel ratio is largely governed by the amount of air flowing through the carburettor, with an 'appropriate' amount of fuel being drawn from the fuel float, according to this rate of airflow.

EFI systems allow a variety of air/fuel ratios to be programmed into the computer that controls the engine (Engine Control Unit, ECU), according to the operating conditions and requirements, in a process called 'fuel mapping'. Injectors (Figure 4.11) spray fuel into the inlet ports, with the duration of this spray dictated by the pre-set fuel map.

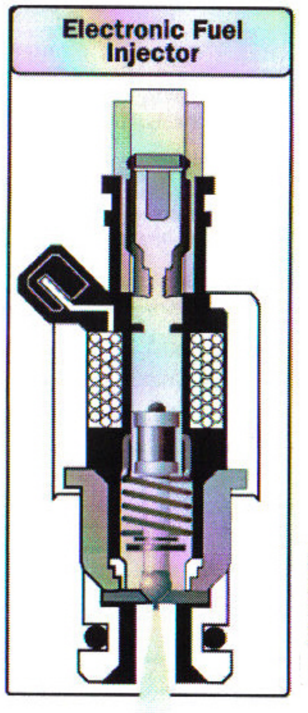


Figure 4.11: Typical electronic fuel injector

(Source: www.howstuffworks.com 2005)

The major benefit of EFI is that engine operation and ambient conditions are monitored constantly by the ECU, and variations to ratios can therefore be performed automatically, as necessary.

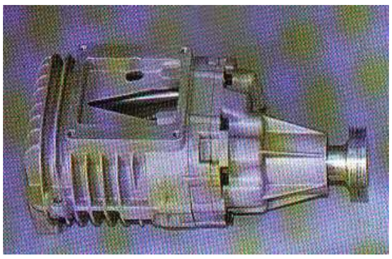
Fuel injection systems generally provide more accurate fuel metering, improved efficiency and better response than carburettors. A more detailed explanation of carburettor and EFI operation and performance is presented in Chapter 7.

4.3.6 Forced Induction Systems

Superchargers and turbochargers are referred to as *forced induction* systems, as opposed to naturally aspirated engines. These systems dramatically increase the volume of charge entering the cylinders, thereby producing much improved power.

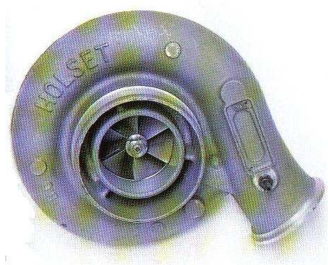
Both systems ‘force’ air into the engine, creating a higher pressure in the intake manifold relative to atmospheric air (Edgar 2001). Superchargers and turbochargers are essentially forms of air pumps, with each operating in a different manner.

Turbochargers redirect exhaust gases to spin a turbine which is mounted in-line with a compressor, forcing air into the engine (Edgar 2001). Therefore, the more exhaust gases an engine produces, the greater the benefits gained from use of a turbocharger. Conversely, superchargers are driven directly from the crankshaft, via a belt-drive (Edgar 2001). Superchargers can be classified into two broad categories, centrifugal and positive displacement, with two types, the Roots design and screw-type, forming the two main designs in the latter category (Figure 4.12).



(a)

(b)



(c)

Figure 4.12: Supercharger types: (a) and (b) show positive displacement Roots and screw-types, while (c) shows a centrifugal-type

(Source: Edgar 2001)

Installing a turbocharger requires significant modification to the existing exhaust system. The turbine and compressor must also be compatible with the engine to ensure optimum operation. Turbochargers are, however, highly efficient and produce better fuel consumption than naturally aspirated engines (Edgar 2001). Their effect, though, is more directed at increasing higher-end torque and power.

Turbochargers also suffer from 'lag' due to their reliance on being driven by exhaust gases. In order to produce more power, a greater amount of exhaust gas is required; however, additional exhaust gas only becomes available once the throttle has been opened further. Therefore, response is not instant, and there is a short time delay before the turbo's increased effect is felt, resulting in a period of lag.

Superchargers are much simpler to install than turbochargers, requiring connection of the drive system (involving the crankshaft) as well as a modified inlet manifold (Edgar 2001). The components needed for this set-up can usually be purchased in kit form, providing all the essential components. Superchargers also offer high efficiency, are easily matched to various engine types and offer ease of tuning, in addition to increased fuel economy over a naturally aspirated engine. Depending on the type of supercharger chosen, improved performance at higher or lower speeds can also be achieved.

4.4 Conclusion

The modifications presented in this chapter have been described in brief detail, along with some of the major performance benefits that can be expected from their implementation. This list of modifications is by no means exhaustive, as a great variety of modifications exist. However, this chapter is intended to indicate some of the more common performance upgrades available and their potential benefits, and to provide some direction as to future research options that may provide improved performance to USQ's Formula SAE race car.

Chapter 5
Intake System Fundamentals

5.1 Introduction

Having identified the requirements for engine performance in the application of Formula SAE (section 2.5), it follows on to design the various components required to achieve these objectives. The intake system performs the crucial role of transporting charge to the engine, sufficient to create the necessary power. All the components in this system must be designed to work in unison to provide the maximum potential for power generation. This chapter provides the design theory behind the creation of the intake system.

5.2 Function of an Intake System

The primary function of an intake system is to deliver a charge of air/fuel to the cylinders in the correct proportions. The various system components work to meter the air and fuel, facilitate charge mixing to achieve optimum fuel atomisation, and deliver the charge to each cylinder in equal amounts, while achieving optimum flow, with minimal losses. As shown in Figure 5.1, intake systems typically comprise:

- air filter (not shown)
- carburettor or throttle body
- plenum
- intake runners (manifold)

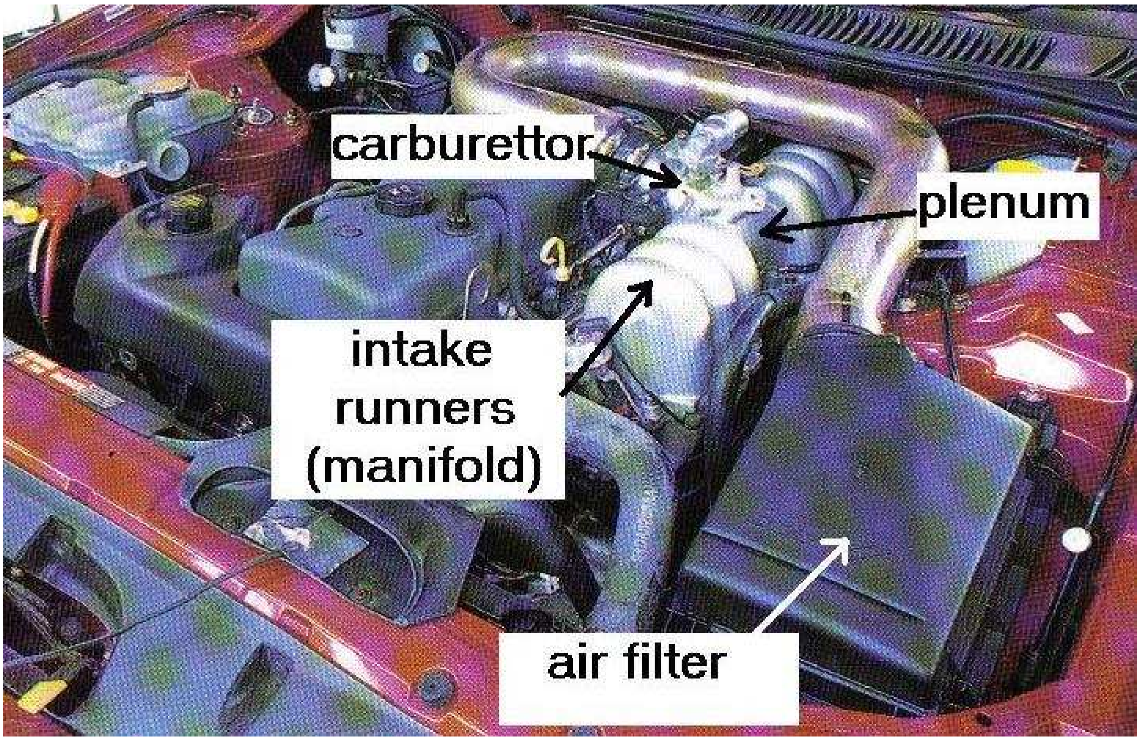


Figure 5.1: Components of a typical intake system

(Source: Edgar 2001)

5.3 Intake System Components – Purposes and Performance Benefits

The introduction to this chapter mentioned that all intake system components must work *together* to provide maximum benefit to power. That is, the overall design does not necessarily aim to produce a number of individually 'perfect' components; but rather, aims to ensure that each component, in performing its own function, also caters to the needs of all related components.

For example, large-throated carburettors are obviously capable of producing greater flow rates than their small-throated counterparts (for equal velocities), which may lead to the conclusion that using the biggest throat size available will maximise flow to the engine. However, the flow rate through a carburettor must match the amount of flow the remaining engine components can accommodate. That is, attempting to supply more air/fuel to the engine than it can handle results in no beneficial effects, and may actually cause detriment to the ability of the engine to accept the required charge flow. Ultimately, every component within the intake system must achieve compatibility with its related components. Overall, this design philosophy should produce a system that operates as an integrated unit, rather than simply forming a connection between individual parts.

Each component within the intake system performs a vital function in itself, while also accommodating the requirements of related parts. The function of these components can contribute significantly to overall engine performance. The functions and contributions of these components are presented below.

5.3.1 Air Filter

The air filter is the first component in any intake system, and performs the function of providing clean air to the engine. Most vitally, the air filter prevents foreign objects from entering the inlet tract, protecting valves, pistons and cylinders from potentially serious damage.

Open-wheeled race cars use airboxes, situated relatively high, above the driver's head (Figure 5.2). However, air intakes on standard road cars are situated under the bonnet, generally in close proximity to the engine, and therefore breathe relatively hot air (Figure 5.3). Hotter air is, of course, less dense than cool air, reducing the quantity of intake charge per volume, and resulting in decreased power.

Figure 5.2: Airbox used in open-wheel racing

(Source: *F1 Racing Magazine* October 2004)

Figure 5.3: Air intake in standard road car, situated close to engine, increases intake air temperature

(Source: Edgar 2001)

To achieve maximum potential for producing improved performance, air filters should therefore be positioned in an unobstructed location, providing the greatest potential to accept sufficient airflow; and should be situated so as to accept cool intake air.

5.3.2 Carburettor/Throttle Body

Carburettors and throttle bodies perform the function of metering air and fuel (or air) through an intake system. Carburettors have long formed the standard vehicle fuel metering devices, while throttle bodies are used on vehicles which employ electronic fuel injection (EFI).

Carburettor housings contain a movable throttle, which creates a greater opening at higher air velocities, and draws in fuel from a float bowl by means of a pressure differential, created by a venturi throat (Figure 5.4). Similarly, a throttle body contains a throttle plate to meter airflow.

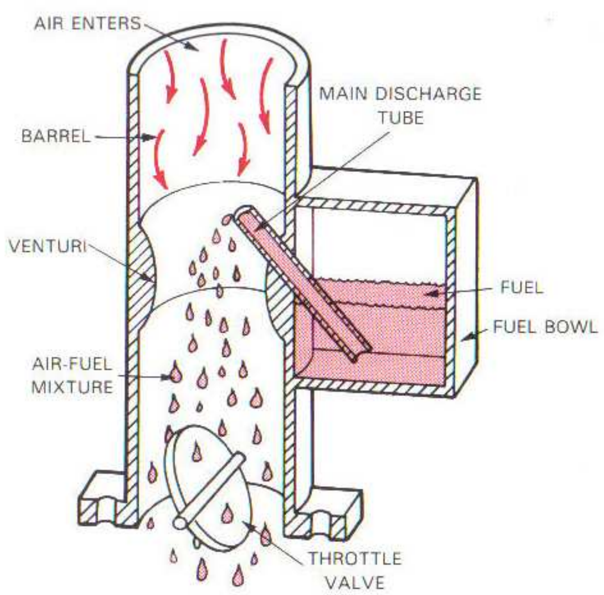
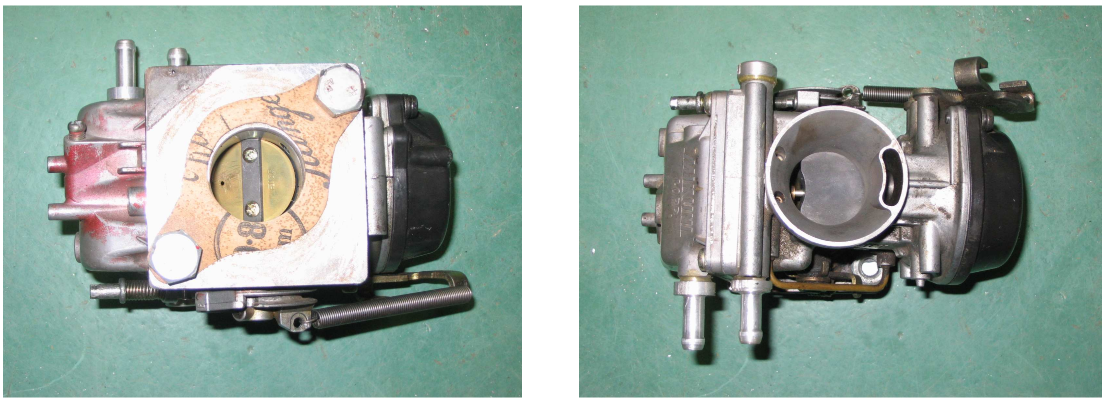


Figure 5.4: Typical components of a carburettor
(Source: Stockel, Stockel & Duffy 2001)

Carburettors and throttle bodies must meter air/fuel or air in the correct proportions to enable peak performance at various speeds. If the mixture supplied is too lean (air content too high) or too rich (fuel content too high), the cylinders may ‘run hot’, causing engine overheating, spark plug damage and, more seriously, detonation, or may cause engine misfiring. Also, if a carburettor/throttle body with too large a throat is employed on a particular engine, excess amounts of intake charge will be supplied, which may lead to charge reversion. This involves the charge travelling in the opposite direction, back along the intake tract, leading to mixture supply problems.

Carburettors and throttle bodies contribute to optimum engine performance by supplying air/fuel ratios appropriate to engine operating conditions. EFI systems use fuel mapping to determine the required air-fuel ratios at various speeds, and hence are generally able to produce superior performance to carburetted systems (Bell 1997). Fuel metering devices must also be able to deliver sufficient, equal flow to all cylinders.

The 1993 Yamaha FZR600 engine used in USQ’s 2004 race car breathed through four 32 mm Mikuni carburettors in standard motorbike trim, and one of these is shown in Figure 5.5.



(a) **(b)**

Figure 5.5: Mikuni carburettor from 1993 Yamaha FZR600 motorbike, showing (a) 32 mm throat and (b) entrance to venturi

Carburetion and throttle body operation and performance are discussed in further detail in Chapter 7.

5.3.3 Plenum

The plenum is a common volume from which intake runners extend (Figure 5.6). Intake charge flows from the carburettor (or throttle body) and accumulates in the plenum, creating a 'reserve' from which individual intake runners draw charge.

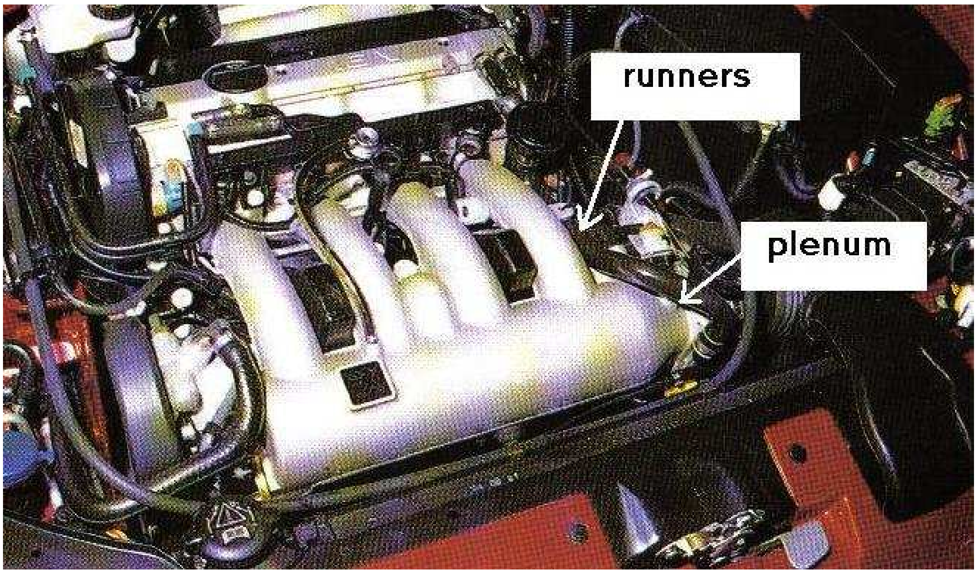


Figure 5.6: Intake runners extending from a common volume, or plenum
(Source: Edgar 2001)

Creating a chamber in which inlet charge is able to accumulate provides a constant supply of charge in close proximity to the inlet runners. This reserve of charge provides additional momentum to the air/fuel flowing into cylinders, creating a more efficient intake process, increasing inlet charge and hence providing greater power production.

Plenum volumes are specific to each engine, with larger-capacity engines requiring larger plenum volumes. Empirically, plenum volume and carburettor (or throttle body) throat size have been found to be inter-dependent, with smaller carburettors generally producing better performance when combined with larger plenums, and vice versa (Bell 1997).

5.3.4 Intake Runners (Manifold)

Intake runners form the final passage in the intake system, delivering intake charge to the engine's cylinders. Most importantly, intake runner design should incorporate:

- excellent flow characteristics
- promotion of fuel atomisation
- use of tuned lengths to increase intake potential

Flow through an intake system is of critical importance. Any losses caused by an intake system reduce the amount of charge flowing to the engine, resulting in power losses. Intake runners should therefore contain no dramatic changes in direction or variations in section, and should feature smooth inside surfaces to minimise flow losses. Runner diameters should enable sufficient flow velocity to ensure fuel maintains its vaporised state, increasing volumetric efficiency. Equal-length runners should also be achieved where possible, to ensure equal volumes of charge are delivered to each cylinder. Tuned-lengths provide assistance in admitting additional inlet charge into each cylinder, by utilising wave phenomena, thereby increasing the amount of power produced.

Overall, flow from the air filter to the inlet ports should follow as direct and smooth a path as possible, minimising flow losses to maximise intake potential.

5.4 Types of Intake Manifold

There are broadly two types of intake manifold: log-type and streamlined-type. These systems each offer different characteristics, as discussed in the following two sections.

5.4.1 Log-Type

Log-type manifolds are very common due to their compact configuration. This type of manifold basically comprises a plenum, on which the carburettor or throttle body is mounted, with intake runners extending from this common volume (Figure 5.7). The charge therefore flows from the air/fuel metering device into the plenum, where it accumulates, ready for induction into each cylinder.

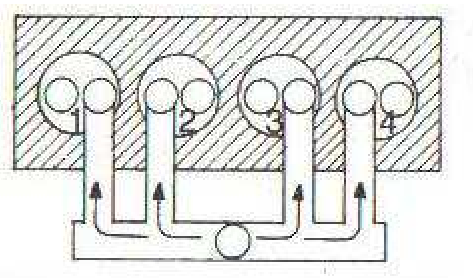


Figure 5.7: Typical log-type manifold for inline four-cylinder engine
(Source: Heisler 1995)

The generally short intake runners, which are most often a product of space restrictions, result in log-type manifolds commonly experiencing inter-cylinder charge robbery problems (Bell 1997) (refer to section 6.6.2).

5.4.2 Streamlined-Type

Streamlined manifolds are characterised by long runners, and are usually designed with tuned lengths (Figure 5.8). Streamlined-types provide a number of benefits over log-type manifolds, viz:

- increased flow
- reduction in charge robbery
- improved fuel atomisation
- use of tuned lengths

Long, separate runners to each inlet port increase flow to individual cylinders, as there is less competition between cylinders for inlet charge. Individual runners also promote separation of flow, decreasing interference of charge between cylinders and thereby reducing charge robbery (Bell 1997).

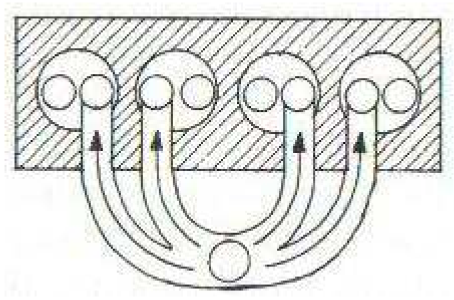


Figure 5.8: Typical streamlined-type manifold for inline four-cylinder engine
(Source: Heisler 1995)

A difficulty arising due to the design of streamlined manifolds is that equal-length runners are difficult to achieve. As shown in Figure 5.8, with all four runners extending from a central plenum, outer runners must travel a longer distance to reach their respective inlet ports than inner runners. This will generally lead to non-equal amounts of charge reaching the inner and outer cylinders, and produce uneven firing. Log-type runners can easily achieve equal length, as each runner feeds from a plenum which extends along the length of all the runners (Figure 5.7).

Equal-length, streamlined manifold runners can be achieved, and will generally necessitate introduction of a bend into each of the inner runners. These bends will induce some flow losses; however, these losses should be offset by the benefits achieved through equal-lengths, namely producing equal flow and smooth power production.

Long length, often high velocity, runners also promote fuel atomisation by increasing relative movement between particles. This benefits fuel economy, by flowing more combustible mixture into the cylinders.

The individual runners employed in streamlined systems allow for lengths to be designed for tuning at certain speeds. This makes such manifolds able to be designed for more specific applications than log-types, as tuning can be accomplished at higher or lower speeds, depending on the operating requirements of the vehicle.

Log manifolds often dictate flow to turn 90° from the plenum chamber into the runners, causing losses and flow restriction (Fox & McDonald 2003) (Figure 5.9). Streamlined manifolds feature very directional runners such that intake charge follows a direct path from plenum to port.

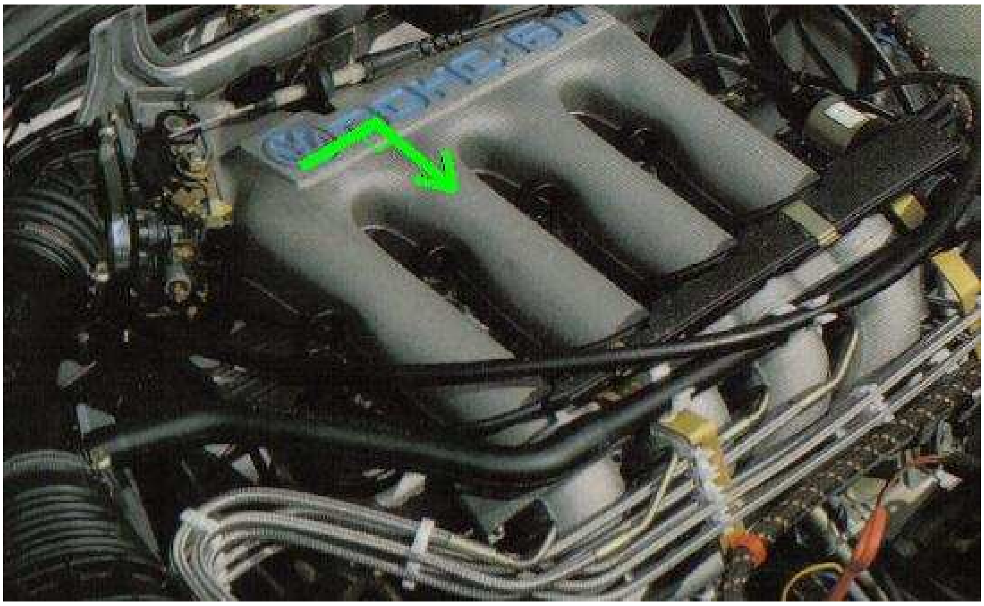


Figure 5.9: Charge must turn 90° from air intake to enter inlet runners

(Source: Bell 1997)

Overall, streamlined-type manifolds provide more efficient flow and better capabilities for 'tuneability'. These manifolds therefore provide benefits in performance applications, where specific requirements can be catered for in design of the intake manifold.

5.5 Nature of Flow Through an Intake System

The flow regime through an intake system is classified as internal, compressible, viscous flow. This flow is subject to pressure losses, causing a reduction in charge density and a corresponding decrease in output power. Therefore, to design an intake system that focuses on increased and more efficient production of power, flow characteristics must form the basis of design.

Intake charge is exposed to pressure differentials due to variations in flow area and the effect of viscous (frictional) drag (Fox & McDonald 2003). Losses in pressure result in decreased charge density, meaning that a reduced amount of air/fuel mix is available to all cylinders, decreasing the potential for power production.

Flow losses occur through constant-area sections of pipe and also through areas of changing section (Fox & McDonald 2003). There are many features within an intake system that have the potential to induce these types of flow losses. The simple act of charge flowing through intake pipes induces losses due to friction. Any bends and curves in these runners increase these flow losses. Changes in section also contribute to overall losses in flow; sectional changes through an intake system are seen where intake charge flows from a carburettor throat into a larger diameter plenum, and from this plenum into smaller diameter runners. These losses combine to form the system's total head loss, and, as mentioned, are attributed to two frictional effects, viz:

- frictional flow through constant-area sections (Major Losses), and
- frictional flow through non-constant areas and fittings (Minor Losses)

Flow through constant-area intake runners accounts for pressure decreases due to major losses, while, as will be explained in section 6.7.3 *Intake Restrictor Design*, flow through entrances and exits and contractions and expansions accounts for minor losses, and can be attributed to the inlet restrictor (Fox & McDonald 2003).

5.6 General Design Principles – Intake System

Most crucially, an intake system must comprise excellent flow characteristics to facilitate efficient delivery of inlet charge. That is, the charge must flow at high velocity with minimal losses and reach the cylinders in equal amounts to provide the greatest potential for maximum power production. Key criteria in achieving these objectives comprise:

- minimum restriction to flow
- facility for maintaining fuel atomisation
- reduction in inter-cylinder charge robbery
- use of tuned pressure waves for increased charge intake

The methods implemented to facilitate these aims include:

- using gently swept, smooth bends
- using appropriately-dimensioned runners (diameter, length)
- companion cylinder grouping
- achieving tuned, equal-lengths

These methods form the basic principles governing intake system design, and are discussed in detail in the following sections.

5.6.1 Flow Restriction, Fuel Atomisation Tuned Lengths

Any losses induced through an intake system result in corresponding losses in power, due to a reduced amount of charge being available to the cylinders. Flow losses must therefore be minimised. A number of factors influence the amount of restriction through an intake system, including runner shape, diameter, length and direction, and these are discussed below.

5.6.1.1 Runner Direction

Ensuring that the path taken by intake charge is as direct as possible, from air intake to port, is important in optimising efficiency of flow (Figure 5.10). The more unnecessary twists and turns the charge is forced to negotiate in making its way to the port, the more energy it expends in doing so. This energy is wasted, where it could have been used to increase the volume of charge reaching the cylinders. Straight, direct runners are therefore desired to create a minimal-loss journey of charge from intake to cylinders.

Figure 5.10: Direct path followed by intake charge from plenum to cylinders
(Source: Edgar 2001)

5.6.1.2 Runner Dimensions

The dimensions of intake runners, namely length and diameter, have critical effects on the behaviour of flow through the system.

● Diameter

Runner diameter influences the velocity of flow, and also contributes to the amount of restriction. Flow rate, Q, is determined by flow velocity and flow area, v and A, respectively, as given in Equation 5.1 (Fox & McDonald 2003):

$$Q = vA \tag{5.1}$$

Larger diameter pipes provide increased flow area, therefore reducing restriction to flow. However, as Equation 5.1 dictates, an increase in area must correspond to a reduction in flow velocity. Velocity of flow is important in two respects: maintaining fuel atomisation and increasing charge momentum.

▪ Maintaining Fuel Atomisation

Fuel droplets suspended in air in intake charge must remain in a vaporised state to allow combustion to occur. Low velocities cause these droplets to fall out of suspension, reforming a liquid state, causing incombustible, or partially incombustible, mixture. Higher velocities also induce more relative movement between particles as the charge flows through the induction system, which is conducive to maintaining vaporisation.

▪ Increasing Charge Momentum

Momentum is another key factor contributing to optimising flow of intake charge. While charge is induced into an engine's cylinders by way of a pressure differential (refer to section 10.4.1 *Air Pressure*), additional force 'pushing' more charge into the cylinders increases the amount of mixture available for combustion.

The momentum, G, of a volume of intake charge depends on its mass, m, and speed of travel, v, as related in Equation 5.2 (Meriam JL & Kraige LG 1998).

$$G = mv \tag{5.2}$$

Assuming constant mass, momentum can then be increased by higher velocities. Smaller diameter intake runners result in higher flow velocities, according to

Equation 5.1, and shown in Figure 5.11 below. Therefore, reducing pipe diameters increases velocity, causing a corresponding rise in momentum, which can help force additional intake charge into engine cylinders.

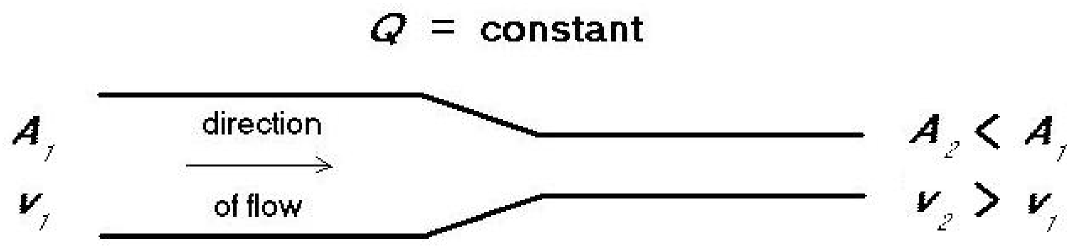


Figure 5.11: Effect of section change on flow velocity, with constant flow rate

Minimising restriction and increasing velocity can both positively influence induction system behaviour. However, the mechanism for each dictates conflicting design criteria. Therefore, a compromise should be reached, achieving sufficient velocity of flow, while maintaining minimal restriction. Empirical formulae are often used to determine inlet runner diameters, and consider such criteria in their determinations (Edgar 2001).

● **Length**

In non-performance applications, intake pipe lengths are determined by the space available. However, runner lengths can be used to markedly increase intake charge potential, and the desired lengths should be accommodated where possible.

▪ **Tuned Lengths**

'Tuned length' refers to intake runners whose lengths have been designed to take advantage of particular wave tuning effects. Depending on a number of variables individual to the engine, induction charge can make use of pressure waves arising in the inlet passage to increase the volume of charge entering the inlet ports. High-performance applications generally employ intake systems tuned for a particular speed, which results in runner lengths corresponding to this particular speed.

A phenomenon known as 'induction ramming' is the desired effect of inlet manifold tuning (refer to section 8.4.5.2 *Case 2*). Induction ramming uses appropriately timed pressure waves to provide additional force to intake charge entering the combustion chamber. This ramming effect also helps to expel remaining exhaust gases during the period of valve overlap (refer to section 8.4.4.1 *Valve Timing*), while increasing charge intake volume.

Achieving not only tuned lengths, but equal-length runners provides further benefits to the induction process, by providing equal volumes of charge to each cylinder. This equality enables the engine to run more smoothly, as each cylinder is theoretically producing equal amounts of power. Ultimately, the result of wave tuning an induction system is an increase in power production.

Longer pipe lengths induce greater losses compared with shorter lengths, and the often necessary bends added to achieve the desired lengths also contribute to flow losses, as explained in *Runner Shape*, below. However, the benefits offered by achieving well-formed, tuned lengths outweigh these additional losses, and should be utilised wherever possible (Edgar 2001).

5.6.1.3 Runner Shape

Inlet manifolds generally comprise runners of circular cross-section. On road cars, the length of these runners is constrained by the space available, and so the runners are typically very short and direct. However, in applications where longer lengths are desired, the runners must often contain bends to achieve these lengths within the available space. Also, the position of the plenum can necessitate that runners feeding the outer cylinders be curved to provide the required path from plenum to inlet port.

Bends in pipes induce greater head losses than do straight sections of pipe (Fox & McDonald 2003). Therefore, some charge will be lost simply by flow through a bend in an intake runner. These losses can, however, be minimised by using large bend radii (r/D, see Figure 5.12) and methods of bending that retain integrity of pipe shape.

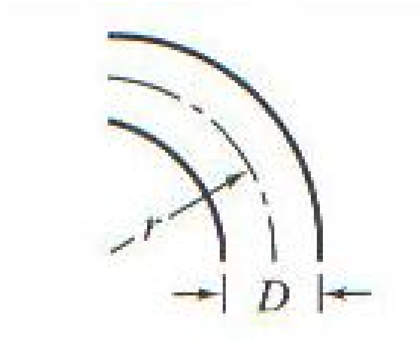


Figure 5.12: Definition of bend radius, *r*/*D*
(Source: Fox & McDonald 2003)

Less severe bends (greater *r*/*D*) cause less dramatic directional changes in flow, thereby reducing losses (Fox & McDonald2003). Therefore, for constant runner diameter, any necessary bends in an intake system should use as high a radius of bend as is practical, to minimise losses due to flow through bends.

Bend method also influences the severity of loss caused by a bend (refer to section 8.4.3.3). For applications requiring tighter bends, mandrel bending produces flow characteristics comparable to a straight length of pipe (section 8.4.3.3, Figure 8.6). Mandrel bends involve forcing a die internally through a length of pipe, similar in diameter to the pipe being bent. This method produces bends of excellent quality, in regard to maintaining sectional integrity and minimising flow loss (Edgar 2001).

Using bending methods that produce good flow characteristics has important effects for tuned-length applications. Producing bends of high quality enables the associated flow losses to be minimised, which further increases the beneficial impact of tuned lengths on induction potential.

5.6.2 Inter-Cylinder Charge Robbery

Inter-cylinder charge robbery occurs when one cylinder breathes in intake charge intended for another cylinder. This results in uneven charge delivery, and therefore produces uneven delivery of power, with cylinders running hot or rich, accordingly.

Charge robbery is a concern most often associated with mass-produced cast inlet manifolds. These manifolds typically comprise very short intake runners, which offer minimal provision for directing and maintaining separation of flow (Figure 5.13). Long, streamlined intake runners offer much advantage for reducing charge robbery (Smith PH & Morrison JC 1971). However, this problem can be further minimised by implementing companion cylinder grouping.

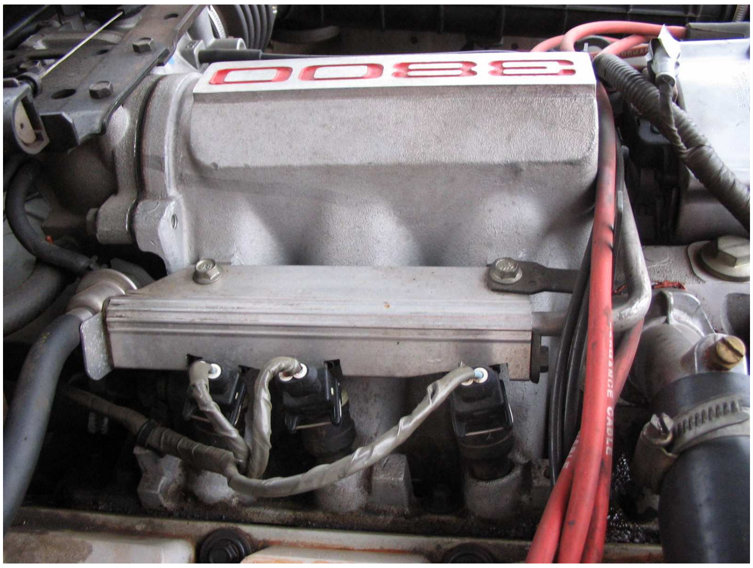


Figure 5.13: Typical cast inlet manifold, promoting charge robbery

● Companion Cylinder Grouping

Grouping of companion cylinders works on the basis of an engine's firing order. In a four-cylinder engine, there will always be two pairs of cylinders operating on the same crankshaft revolution. Each pair is known as a companion cylinder pairing.

As each pair operates on the same revolution of the crankshaft, their pistons each travel up and down in synchronisation. Therefore, the pair must use camshaft timing 180° out of phase with each other to allow successive firing of all four cylinders.

This means that as one of the cylinders is on its induction stroke, its pair will be on a power stroke, 180° ahead in terms of cam timing. Simultaneously, one of the second pair will be performing a compression stroke, while its respective pair is on an exhaust stroke.

By this arrangement then, companion cylinders do not (can not) fire successively. This is an important fact in reducing charge contamination, as explained below.

The order in which cylinders perform their power strokes is expressed by an engine's firing order. This order therefore shows the succession of firing, or which cylinder follows another's power stroke. Cylinders which fire in succession also perform their induction strokes in succession. Therefore, these non-companion cylinders will attempt to draw in induction charge within very short periods of each other. If these cylinders happen to be physically situated in close proximity, i.e. next to each other, such as cylinders 1 and 2, or 2 and 3, or 3 and 4 in a straight-four engine configuration, their respective inlet ports will each be 'breathing' from the same region in the induction passage, or plenum. The problem of inter-cylinder charge robbery then becomes apparent, as there is accordingly less charge volume available to the second cylinder following this first cylinder's induction stroke.

The idea of companion cylinder grouping then is to physically connect pairs of cylinders that do not fire in succession, i.e. companion cylinders. This connection, at the inlet to the respective intake runners, allows more time for charge to accumulate after the induction stroke of the first in the pair, making a greater volume of charge available to the second cylinder in the pairing for its induction stroke. A greater volume of charge provides greater potential for increased power, and so companion cylinder grouping can contribute significantly to overall performance improvement.

Companion cylinder grouping can also be used in exhaust systems to increase flow by reducing restriction, hence drawing out additional combustion gases and, at the same time, increasing induction potential. Companion grouping of exhaust pipes for USQ's 2005 race car is covered in section 9.2.2.

Companion cylinder grouping will form part of the design of the 2005 intake system for USQ's Formula SAE car, with details regarding this design presented in section 6.6.

5.7 Conclusion

Fundamental design theory for intake systems has been presented here, including engineering flow principles and empirical correlations. These theories will now be used to design the 2005 intake system (Chapter 6), with the aim of achieving optimised engine performance.

Chapter 6
2005 INTAKE SYSTEM DESIGN

6.1 Introduction

Having established the fundamental criteria for intake system design (Chapter 5), this theory is now applied to determine a suitable design for the 2005 intake system. Design of the various components comprising the intake system will result from these design criteria, to create a system which should achieve the specified objectives, as outlined in section 5.6.

6.2 Design Objectives

To reiterate, design of the intake system must achieve the following objectives:

- minimum restriction to flow
- facility for maintaining fuel atomisation
- reduction in inter-cylinder charge robbery
- use of tuned pressure waves for increased charge intake

Achievement of these criteria should produce an intake system that offers performance benefits to USQ's 2005 race car, in terms of increased power, improved fuel economy and suitability for the intended application.

6.3 Intake System Design

Design of the intake system must be inclusive of all system components. Therefore, the design focus incorporates:

- fuel delivery device
- restrictor
- plenum
- intake runners

In addition to each of these components satisfying engine operating requirements, they must also fulfil another important design criterion: system packaging. That is, all components must fit into the physical space available. Compromises will necessarily have to be made in accordance with the overall space available for fitment.

The overall design of the intake system features a vertical configuration; that is, the intake system will essentially be built 'directly upwards' from the engine. This means that the runners will extend vertically upwards from the inlet ports, with the plenum, restrictor, carburettor (or throttle body) and air filter being positioned one on top of the other. Measurements taken with the engine in the chassis show the available height to be approximately 700 mm. Initial calculations regarding optimum runner length, plenum volume and restrictor length have been used to reach compromises in regard to the space available, resulting in approximate allowable heights for each component, as represented in Figure 6.1.

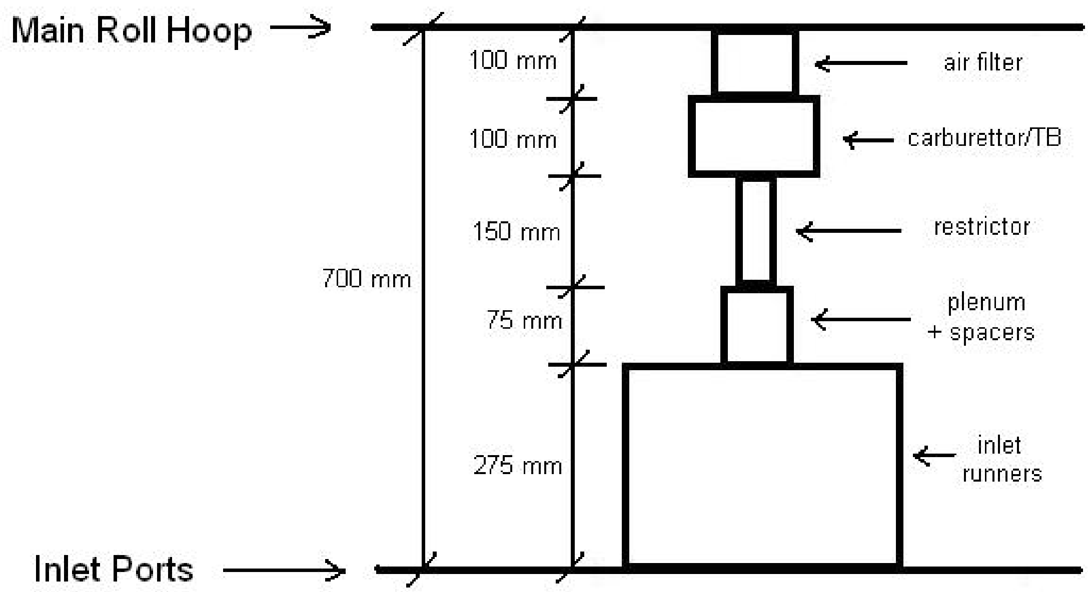


Figure 6.1: System Packaging for Intake Components

Each of these components now forms the focus of detailed design, as outlined in the following sections.

6.4 Fuel Delivery Method

The choice of fuel delivery method is vitally important, as substantial performance benefits can be achieved according to the method chosen, including reduced fuel consumption and increased power across a specific rpm band. The choice is often necessarily dictated by budget, technical knowledge and suitability of the system to be integrated with existing equipment. The two choices, carburetion and electronic fuel injection (EFI), each have individual advantages and disadvantages which must be considered in the decision-making process.

Carburetion is a well-known and well-established air-fuel metering method, and offers adjustability and tuning potential for experienced persons.

Electronic fuel injection is a relatively recent invention (Edgar 2001), and relies on computer controlled signals to provide the engine with the correctly proportioned mixture. EFI systems require fairly high levels of technical competence, and are also considerably more expensive than carburettors, but can offer superior performance when tuned correctly.

The method of fuel delivery forms an important component of the overall intake system, and is afforded a broader treatment in Chapter 7. Operating principles and potential performance benefits are discussed, providing a basis for deciding which method to employ.

6.5 Intake Runners

Intake runners perform a vital function by delivering intake charge to the cylinders. The design of these runners must therefore ensure that the mixture arrives in a highly combustible state, while inducing as few losses as possible throughout the mixture's journey from carburettor to intake port.

A number of factors must be considered to determine the overall runner design, including manifold type, runner dimensions and configuration.

- **Manifold Type**

  The intake manifold will take the form of a streamlined-type design (Figure 6.2), according to the benefits presented in section 5.4.2. Use of long, streamlined runners will create a direct flow path to the inlet ports, increased momentum, ability to tune for lower speeds and provision for companion cylinder grouping.

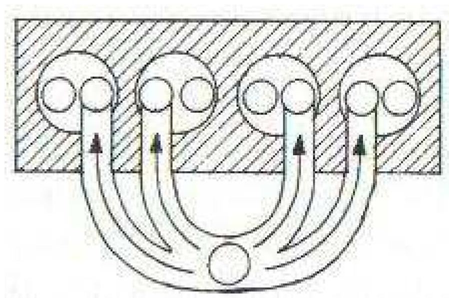


Figure 6.2: Typical streamlined-type manifold forming the basis of the 2005 intake system

(Source: *Advanced Engine Technology*, 1995)

● Runner Dimensions

The inlet manifold will incorporate tuned, equal-length runners. Achieving equal-lengths in a streamlined design will necessitate the inclusion of additional bends in the inner runners; however, as explained in section 5.4.2, achieving equal-lengths should provide improved engine performance, with high-precision bending methods employed to minimise losses.

Empirical formulae will dictate runner lengths for a desired tuned speed. However, the actual runner lengths will necessarily be a compromise between the desired tuning range and the space available (refer to Figure 6.1).

Bell (1997) recommends long-length pipes to tune for maximum torque at low- to mid-range speeds. The recommendation is to use as long a length as possible within the allocated space. Smaller diameters have also been empirically found to increase torque and power in the low- to mid-ranges (Bell, 1997). The configuration of the Formula SAE competition track enables speeds no greater than 100 km/h, with average speeds of around 50 - 60 km/h (*Formula SAE-A Event Program* 2004). Therefore, engines used in this application require good low- to mid-range torque. Long pipes and small diameters will therefore characterise intake manifold design, with specific dimensions to be determined by empirical formulae, in section 6.7.2.

● Configuration

Configuration of the intake manifold refers to the connection and orientation of the individual intake runners. Companion cylinder grouping will form a feature of this intake design, and is facilitated by the choice of a streamlined-type manifold. The connection of the runners at the end opposite the inlet ports is therefore highly important.

The YZF600 has a firing order of 1-2-4-3 (*YZF600 Service and Repair Manual*). Companion cylinder grouping dictates connection of cylinders in non-successive firing order. Therefore, companion cylinders in this case are:

- cylinders 1 and 4

- cylinders 2 and 3

Connection of inlet runners to the plenum will therefore preclude non-companion cylinders from breathing charge from similar regions. This requirement will be facilitated by either physically joining companion pairs at the non-port ends, forming dual 2-into-1 branches, or by making provisions to separate flow within the plenum into two regions, by addition of a dividing plate, to feed each companion pair.

6.6 Preliminary Manifold Designs

Some preliminary manifold designs were determined prior to defining the design for the 2005 intake system. These design variations mainly differed in the arrangement of the runners, runner-to-plenum connection and plenum shape. These manifold designs are now presented along with their basic design intentions.

6.6.1 Preliminary Design 1

Initial manifold design focussed on achieving companion cylinder grouping. This first design stemmed from the theory of 4-2-1 extractor design (refer to section 8.5.2), and featured a 'reverse 4-2-1' configuration (Figure 6.3). Extending downwards from the restrictor to the inlet ports, this manifold was designed with a single (short) runner extending from the restrictor, branching into two separate pipes; these two branches then separate into a further two pipes each, forming the four individual inlet runners feeding each inlet port.

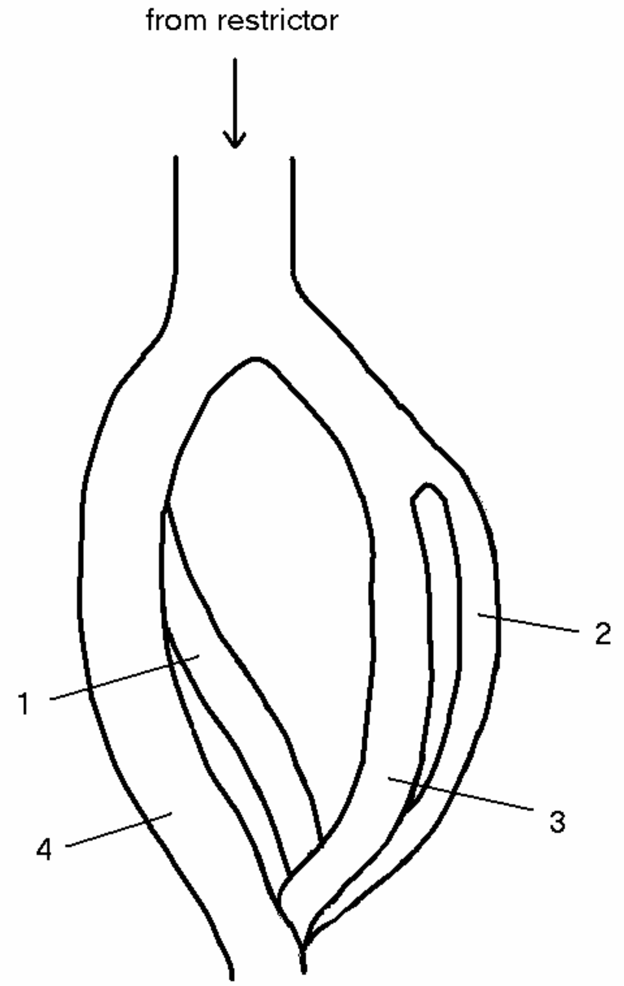


Figure 6.3: 'Reverse 4-2-1' preliminary inlet manifold design

Runners to cylinders 1 and 4 were designed to extend directly from their common branch to the outer ports; runners 2 and 3, however, were designed to curve outwards for some distance from their shared branch, to achieve the equal-lengths desired for all four pipes.

This design involved no defined plenum, with the two initial single branches breathing charge directly from the restrictor. The effect of the exclusion of a plenum cannot be defined without actually testing the system, and comparing it with a similar system that makes use of a plenum.

This design should virtually eliminate inter-cylinder charge robbery, as non-companion pairs each breathe from separate intake branches on their respective intake strokes.

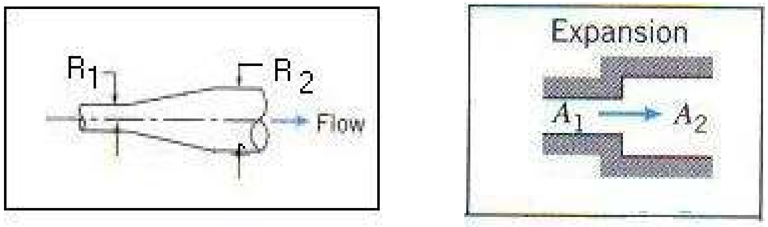
Theoretically then, this manifold design should achieve many of the prescribed objectives. However, it was indicated that the required connection of pipes, along with the bending needed to achieve the desired equal lengths would be relatively difficult, and also quite expensive, requiring mandrel bends for facilitating the 2-into-1 connections (O'Neill AM 2005, pers. comm., 8 August). Therefore, an alternative manifold design was pursued, with greater focus on manufacturing ability.

6.6.2 Preliminary Design 2

Achieving companion cylinder grouping and tuned, equal-lengths was identified as being able to be facilitated through various other designs. Therefore, it was decided to determine an alternative design that offered improved manufacturing ability.

The connection of companion cylinders as 2-into-1 branches has been discounted, due to manufacturing difficulty. Therefore, the revised method of connection will involve each of the four inlet runners connecting independently to a common volume (plenum), with companion cylinder grouping facilitated by inclusion of a dividing plate within the plenum.

Firstly, the plenum design was considered. In addition to accommodating all four inlet runners and providing facility for fuel atomisation, an important consideration is design of the connection from the restrictor to the plenum. The restrictor offers a 37 mm diameter exit (refer to section 6.7.3 *Restrictor Exit Region*); the overall size of the plenum is required to be considerably larger, to accommodate connection of all four inlet pipes. Therefore, inlet charge will be required to flow through a change in section as it exits the restrictor and enters the plenum. Fox and McDonald (2003) show that use of a diffuser shape to accommodate the variation from a smaller section to a larger section reduces flow losses, compared with forcing fluid directly from a small to a large area, through a sudden expansion (Figure 6.4).



(a) (b)

Figure 6.4: Losses can be minimised by flow through a diffuser, (a), as opposed to a sudden expansion, (b)

A conical plenum shape was therefore chosen, extending from the restrictor. The four inlet runners were equally spaced around the base of the plenum, with runners from cylinders 1 and 4 connected to one side of the plenum, and runners from cylinders 2 and 3 to the other (Figure 6.5). A divider plate within the plenum would then direct flow into two separate streams, one to each set of companion cylinder pairs (Figure 6.6).

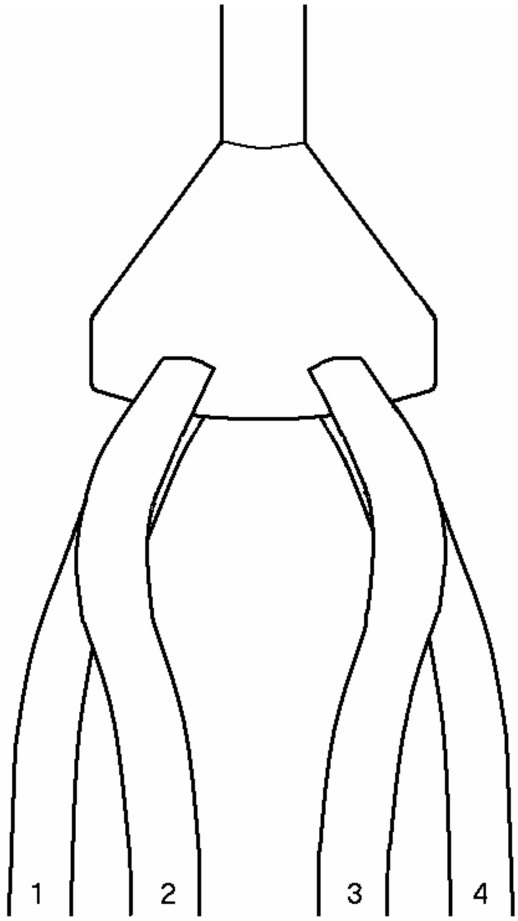


Figure 6.5: Preliminary manifold design 2

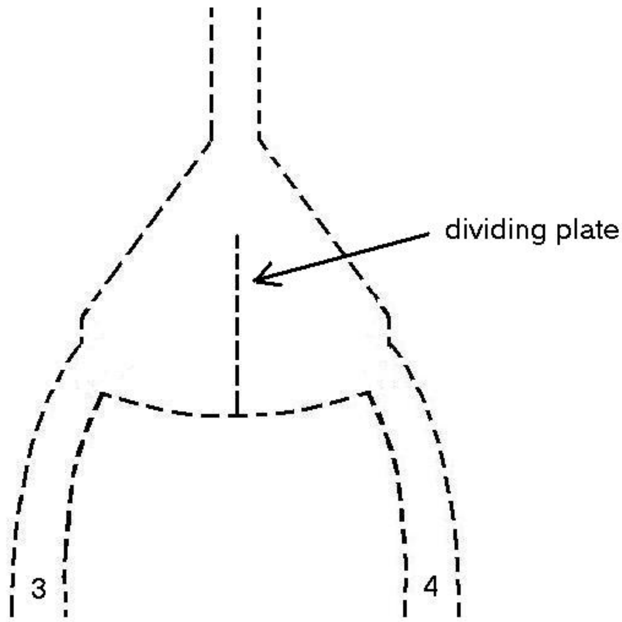


Figure 6.6: Plenum divider to reduce inter-cylinder charge robbery

The base of the plenum was designed as a relatively flat plate. This aspect caused some concern regarding inlet charge losing velocity and hence fuel droplets falling out of suspension when coming into contact with the flat base. As will be seen in section 6.6.3 *Preliminary Design 3*, this design aspect was reconsidered.

While this revised plenum design should provide advantages through use of a gradual section change, it was anticipated that some inefficiencies in combustion of charge may result from the flat-bottomed plenum. Another plenum design was therefore pursued, and is presented in the following section.

6.6.3 Preliminary Design 3

This design focussed on creating a plenum that was more likely to provide benefits to inlet charge velocity and fuel atomisation. The diffuser shape at the entry to the plenum as in Preliminary Design 2 was retained, with a nozzle shape replacing the flat base plate. The overall design then was essentially a gently tapered diverging-converging arrangement (Figure 6.7).

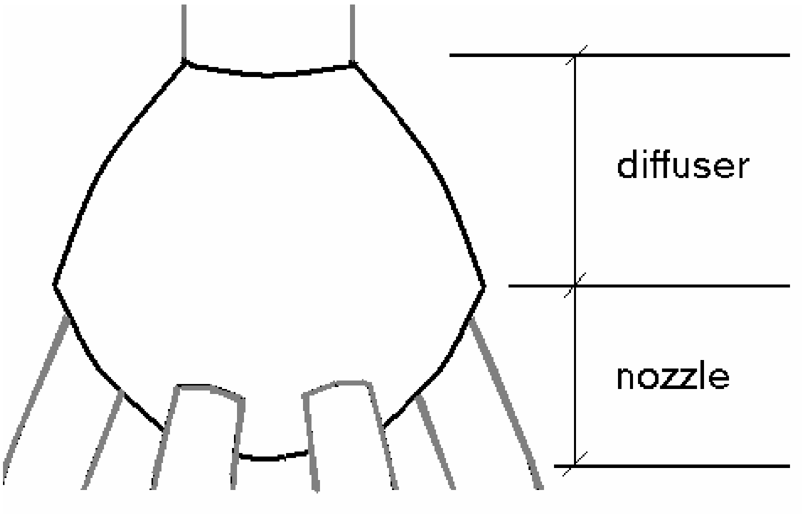


Figure 6.7: Diverging-converging plenum design

Providing a gently curved path into the intake runners will theoretically enable intake charge to maintain velocity and hence remain atomised. This plenum design should therefore provide improved combustion, resulting in creation of more power and better fuel economy.

An alternative arrangement for the intake runners also evolved, to provide another option for achieving equal lengths and grouping companion cylinders. The runners from cylinders 2 and 3 featured a 'cross-over' arrangement, extending the length of their individual paths from plenum to port, to enable equal-length runners (Figure 6.8).

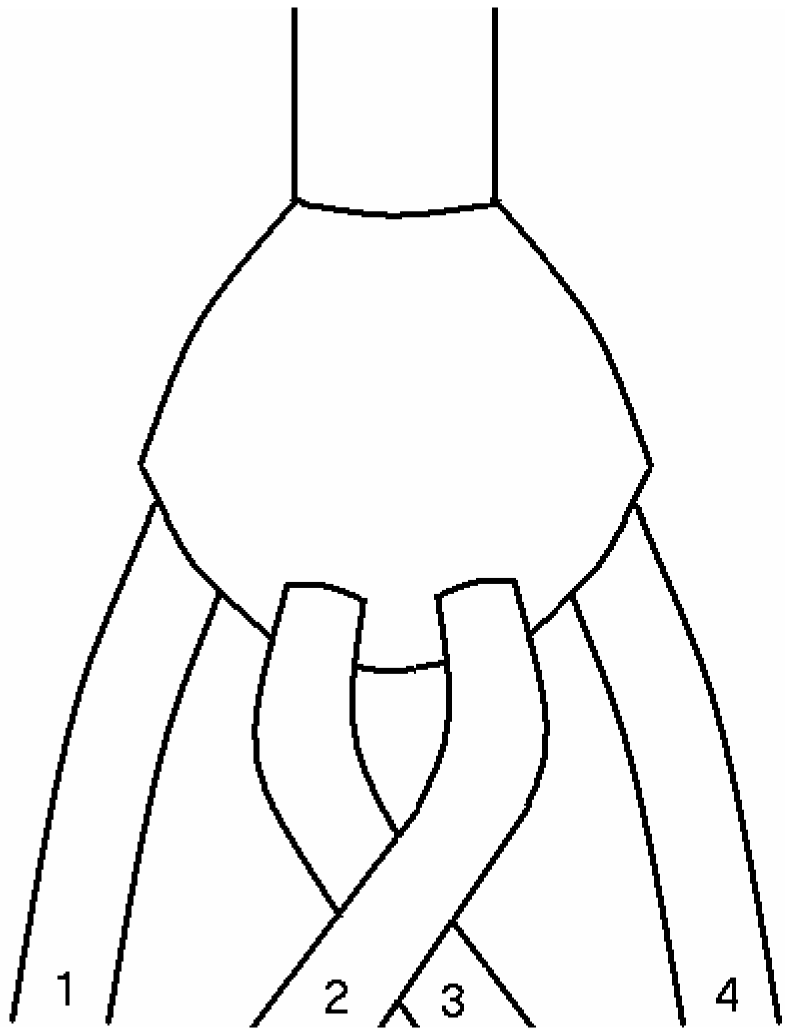


Figure 6.8: Diverging-converging plenum and cross-over intake runner design

Inlet runners are again equally-spaced around the perimeter of the plenum, however no divider plate is used. Instead, it is anticipated that the diverging-converging conical plenum shape will cause a downward spiralling effect on the flow, assisting increased delivery of charge to each inlet runner in turn. The runners are also positioned around the plenum in an order to best accommodate the engine's firing order and make use of companion cylinder grouping benefits.

Manufacture of this plenum was determined to best be achieved by creating two separate halves, making the required cone shapes by rolling rectangular pieces of sheet metal. Joining of the two halves would be facilitated be welding; however, a sharp edge and sudden direction change would be formed at the join, which would probably detrimentally affect charge flow, by creating a turbulent region around the join (Fox & McDonald 2003), as shown in Figure 6.9.

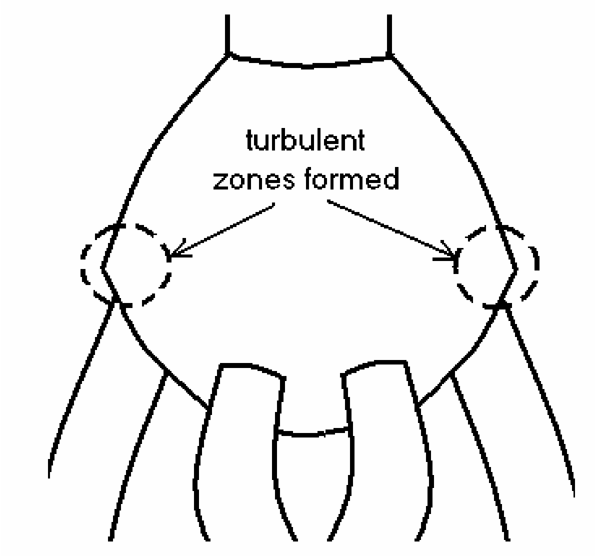


Figure 6.9: Turbulent regions caused by sharp edges and sudden direction change

A constant diameter 'ring' could be introduced between the two halves, providing a more gradual directional change (Figure 6.10), however this flat-walled section may reduce intake velocity.

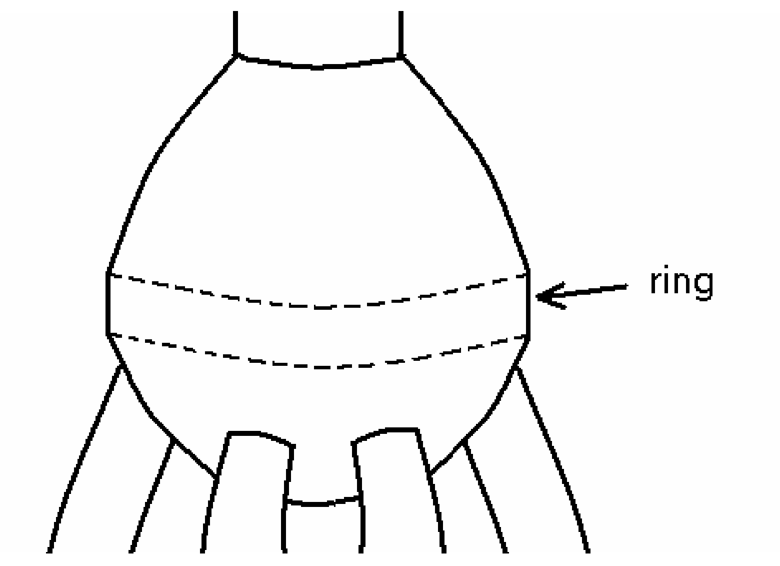


Figure 6.10: Additional 'ring' to reduce turbulent zone

Consultation with the workshop indicated that the plenum shape would be challenging to manufacture (Aston B. 2005, pers. comm., 8 August).

A variable plenum volume was also an attractive design objective, as conflicting recommendations for plenum volume were encountered. The shape of this plenum would cause some difficulty in achieving variable volumes, compared to a rectangular- or square-shaped arrangement. Considering this difficulty together with that of manufacturing the plenum overall, a decision was made to again pursue another plenum design, still focussing on intake velocity and efficiency, while achieving an easily-manufactured component.

This final design option ultimately formed the intake system implemented on the 2005 car, and design of its various components is presented in the following sections.

6.7 Final Intake System Design

Design of the intake system for the 2005 car involves design of the intake restrictor, plenum and inlet manifold, each of which now follows.

6.7.1 Plenum

● Shape

To promote ease of manufacture, enable provision for varying plenum volume and increase fuel vaporisation potential, the plenum was determined to offer a V-shaped design (Figure 6.11). Intake runners should offer the most direct path possible for flow of charge; therefore, the inclined plates forming the V-shape offer a straighter connection of runners to the plenum, than would be achieved by requiring the runners to curve around and connect at the side of the plenum.

Figure 6.11: V-shaped plenum base

Also, the flat base of the 2004 plenum caused fuel to re-liquefy, resulting in puddles of unburnt fuel accumulating in the plenum chamber. This causes reduction in fuel efficiency, decreased power and poses a possible fire hazard in the event of the engine backfiring. Therefore, a plenum base that maintains charge velocity and hence promotes improved fuel atomisation, while also providing a very direct path into manifold runners is required. The V-shaped design of this plenum should provide such benefits.

● **Dimensions**

Edgar (2001) recommends a plenum volume 0.8 times the size of the engine, which, in this case, results in a plenum volume of 480 cc. However, other sources recommend volumes should be 2.5 times greater than engine capacity. Discussion with an experienced engine builder indicated that for a 600 cc engine with a restricted inlet, a relatively small plenum would be sufficient (pers. comm., O'Neill, T., 8 August 2005). It was therefore decided to create the plenum base just large enough to accommodate connection of all four inlet runners, with 'spacers' used to vary plenum volume to determine the most suitable plenum size.

Section 6.7.2 shows that 28.8 mm OD pipes were determined to provide optimum flow characteristics for the intended application; therefore, two rectangles made from 2 mm steel plate were manufactured, each measuring 40 x 80 mm, with dual 29 mm diameter holes drilled through each (Figure 6.12).

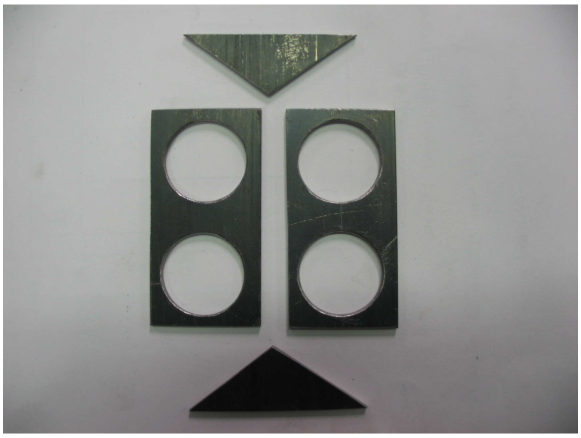


Figure 6.12: Plenum base and side plates

The angle of these plates with respect to the horizontal was determined as a compromise between providing a relatively large internal angle to avoid as much flattening of the plenum base as possible, and reducing the plates' angle with respect to the horizontal to provide a straight, direct connection of inlet runners to the plenum. An anticipated good compromise was chosen as a 35° angle to the horizontal (Figure 6.13).

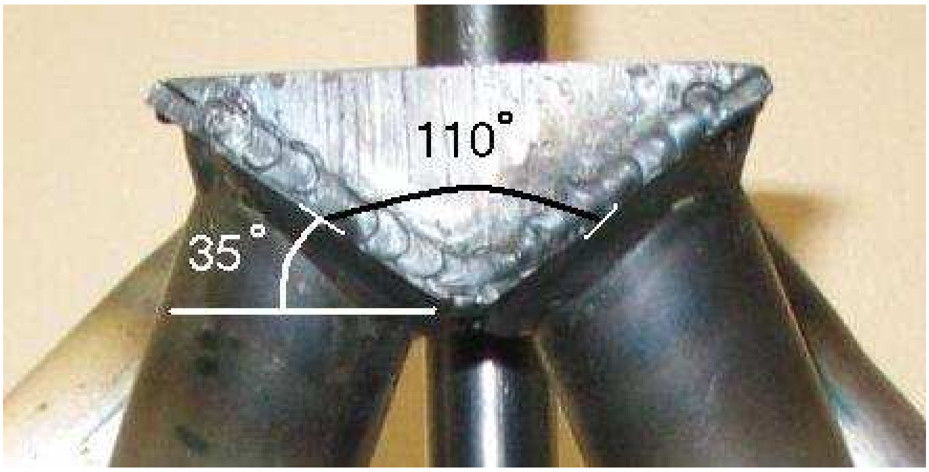


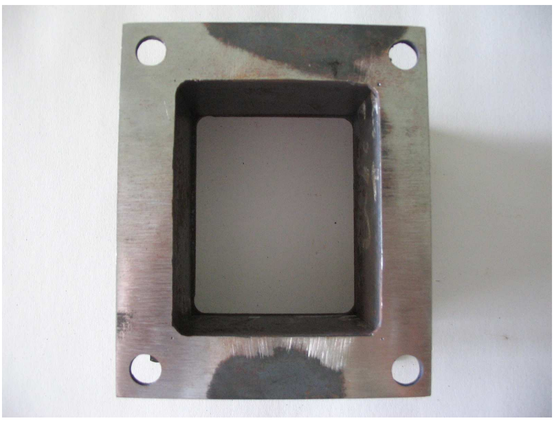
Figure 6.13: Orientation of plenum base plates

● **Spacers**

Research into plenum volumes did not provide a conclusive indication of appropriate plenum size. Therefore, it was decided to develop a variable volume plenum, by using various bolt-on spacers. The intention was then to test these various volumes, either using a dynamometer or on-track testing, to determine the most suitable volume. Rectangular spacers, 25 mm and 50 mm high, were manufactured to provide this function. Each was made from 2 mm steel plate, formed into a rectangle by welding together four separate sides (Figure 6.14). The height of these spacers, which naturally influences the volume of each, was recommended given the size of the engine (O'Neill AM 2005, pers. comm., 8 August).

(a)

(b)



(c)

Figure 6.14: (a) 25 mm and (b) 50 mm spacers, and (c) spacer connecting plate

Each of these spacers is intended to be a simple bolt-on addition, positioned between the restrictor and plenum base, with connecting plates attached to each spacer and correspondingly to the restrictor exit and plenum base. A photo showing the assembly of the manifold, both spacers and restrictor is presented in Figure 6.15.

Figure 6.15: Restrictor, plenum spacers and manifold assembly

A number of options therefore exist with regard to plenum arrangement. The intake manifold has the option of comprising:

1. V-shaped plenum only
2. plenum plus 25 mm spacer
3. plenum plus 50 mm spacer
4. plenum plus both 25 mm and 50 mm spacer

This plenum design therefore offers variable volume, in the range of 60 – 450 cc.

● **Companion Cylinder Grouping**

For ease of manufacture, runners 1 and 2 were each connected to one side of the plenum, with runners 3 and 4 connecting to the other side. As these pairings are not in accordance with companion cylinder grouping, use of a divider plate within the plenum is necessitated. The orientation of this plate is shown in Figure 6.16.

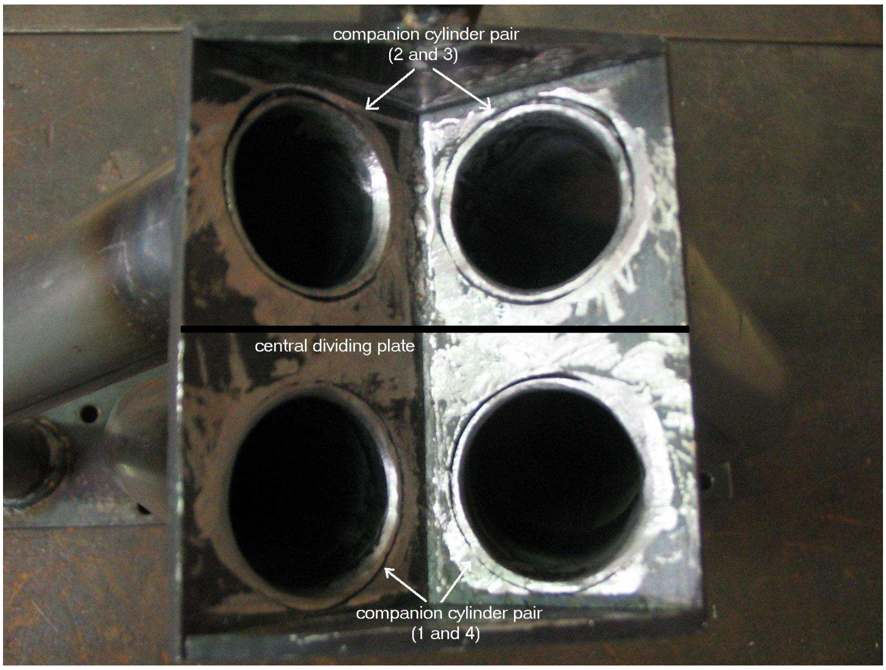


Figure 6.16: Plenum divider plate used to reduce charge robbery

6.7.2 Intake Runners

● **Shape**

In order to provide as direct a path as possible, the orientation of the manifold runners is essentially dictated by the positioning of the central plenum and the four inlet ports. Each runner will necessarily contain bends to achieve these paths, and so using high radius bends and maintaining integrity of pipe shape is integral to producing efficient flow.

● **Length**

With the general orientation of the runners decided, determining the appropriate length for tuning at the desired speed now becomes the crucial aspect of the manifold's design.

Intake ramming (refer to section 8.5.4 *Case 2*) is the desired affect of tuning intake runners. The pressure waves travelling through the intake system actually generate a number of reflected waves. That is, when the first reflected wave travels from the runner entrance and contacts the inlet valve, it generates another reflected wave, which is sent back toward the runner entrance. When this wave reaches the entrance, another reflected wave is again generated. This succession of reflected waves, comprising first, second, third, fourth and so on pulses, therefore provides a variety of tuning options. Each successive reflected wave takes more time than the previous to arise and travel through the system; therefore, depending on the desired tuned speed, and taking the engine's valve timing into consideration, tuning for a certain pulse can be achieved.

Atherton (1996) and Edgar (2001) provide similar empirical equations for determining tuned intake lengths, with recommendations to tune for the third reflected pulse. It is indicated that this pulse should provide appreciable benefit in forcing out the last of the exhaust gases late in the period of exhaust valve opening, while also significantly boosting intake charge volume.

Equation 6.1 (Edgar 2001) is used to determine the tuned inlet length (in inches) for the third reflected pulse.

$$L = \frac{97000}{n} \tag{6.1}$$

where L is the tune length (in inches)

n is the desired tuned speed (rpm)

Tuning for between 4 000 and 8 000 rpm is desired for USQ's race car in Formula SAE-A competition (refer to section 8.6.1.2). Therefore, lower and upper length limits will be calculated, according to these two speeds.

● **Lower Limit: Tuned speed = 8 000 rpm**

$$L = \frac{97000}{8000}$$
$$= 12.1 \text{ inches} = 308 \text{ mm}$$

● **Upper Limit: Tuned speed = 4 000 rpm**

$$L = \frac{97000}{4000}$$
$$= 24.3 \text{ inches} = 616 \text{ mm}$$

Taking a compromise between these two lengths, which is effectively tuning for an average speed of 6 000 rpm, produces a desired tuned length of:

● **Tuned Inlet Length: Tuned Speed = 6 000 rpm**

$$L = \frac{97000}{6000}$$
$$= 16.2 \text{ inches} = 411 \text{ mm}$$

Hence, a tuned length of approximately 400 mm should provide increased charge volume and improved expulsion of exhaust gases at 6 000 rpm.

However, another compromise must be made, regarding the amount of space available for the intake manifold. As indicated in section 6.3, there is a maximum vertical space of 275 mm available to accommodate the intake runners. To achieve the desired tuned length, the runners could curve out some distance from the plenum before flowing into the intake ports; this arrangement is highly undesirable however, as loss of charge and a reduction in inlet velocity would be incurred due to the numerous bends required for this arrangement. Therefore, the length of the inlet runners will be decreased to 275 mm each, according to the space available.

The effect of this reduction in length will be to tune the manifold for a higher speed, which can be determined by rearranging Equation 6.1, as follows:

$$
\begin{aligned}
L &= \frac{97000}{n} \\
\therefore n &= \frac{97000}{L} \\
&= \frac{97000}{(275/25.4)} \\
&= 8959 \text{ rpm}
\end{aligned}
$$

Therefore, the manifold runners will be tuned for approximately 9 000 rpm, which is acceptable, given that the upper limit of the tuned speed range is 8 000 rpm. Although tuning for the desired speed could not be achieved due to space restrictions for the inlet manifold, tuning at a speed of 9 000 rpm should still provide appreciable benefit in terms of charge ramming and additional expulsion of combustion gases.

The intake manifold to be used on the 2005 race car is shown in Figure 6.17.

Figure 6.17: 2005 USQ Formula SAE intake manifold

6.7.3 Intake Restrictor

Formula SAE Rules mandate the inclusion of a single, circular restrictor in the inlet system, no more than 20 mm in diameter (2005 Formula SAE Rules). This restrictor is required to be placed between the throttle and the engine, with all airflow passing through this contraction, with the purpose of limiting engine power. Restrictor design therefore forms an integral component of the overall manifold design, to attempt to achieve optimum airflow through this section and hence increase the potential for improved power and performance.

• Intake Restrictor Design

Fluid dynamics largely governs design of the intake restrictor (see Section 6.4 above). A decreased flow area results in a reduced amount of charge flowing to the engine. Therefore, to extract maximum power from the engine, the restrictor must be designed to produce maximum flow through the 20 mm contraction. Restrictor design is governed by:

- specified dimensions (20 mm diameter)
- flow characteristics
- available space in intake system

Flow rate through the restrictor depends on:

- entrance shape
- exit shape
- throat shape
- length
- surface finish
- charge density (affected by pressure, temperature)

USQ's entry uses a naturally aspirated engine, meaning that the engine breathes ambient air. Therefore, the density of the intake charge will depend upon ambient air conditions, which are of course influenced by the weather on any given day. If a supercharger or turbocharger system were incorporated, the intake charge density could be increased (refer 4.3.6), resulting in more power. However, in this case, flow rate through the restrictor will be determined by those factors listed above that can be controlled, which excludes charge density.

The inlet restrictor can be viewed as comprising three separate sections, with the design of each aiming to meet specific objectives, as shown in Table 8.1:

COMPONENT	OBJECTIVE
Entrance Region	Accommodate 32 mm to 20 mm section change with minimal frictional losses
Throat	Provide mandatory 20 mm restriction while maintaining optimum flow
Exit Region	Accommodate sectional change from throat diameter to plenum area with minimal pressure gradient

Table 6.1: Break-down of requirements for intake restrictor

The methods for achieving these objectives are detailed in the proceeding sections.

• Entrance, Exit and Throat Shape

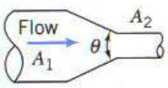
Losses in pressure occur when flow travels through a sudden expansion or contraction in a pipe. However, losses due to area change can be somewhat reduced by installing a nozzle or diffuser between the sections of interest (Fox & McDonald 2003).

The restrictor must be placed between the throttle and engine, and have a maximum diameter of 20 mm. In this case, the carburettor (or throttle body for injected systems) will lead into the restrictor, which will then follow on to the plenum. Therefore, a reduction in section from the carburettor to the restrictor is required, and similarly, an increase in section from the specified 20 mm diameter to the diameter of the plenum opening is also required.

Consequently, the most suitable shape for the restrictor becomes a conical converging-diverging (nozzle-diffuser) arrangement. The flow characteristics of these shapes are shown in Figures 6.18 and 6.19.

▪ **Entrance Region Design**

Figure 6.18 shows that a gently tapered cone will produce less severe losses than an acutely angled conical nozzle.



	Included Angle, θ, Degrees						
A_2/A_1	10	15–40	50–60	90	120	150	180
0.50	0.05	0.05	0.06	0.12	0.18	0.24	0.26
0.25	0.05	0.04	0.07	0.17	0.27	0.35	0.41
0.10	0.05	0.05	0.08	0.19	0.29	0.37	0.43

Figure 6.18: Loss Coefficients for Gradual Contractions: Round and Rectangular Ducts

(Source: Fox & McDonald 2003)

In cases of constant area ratios (AR = A_2/A_1), the nozzle length (the length between the two straight sections of pipe) then becomes critical:

- for constant AR, a shorter nozzle length requires a greater included angle θ, resulting in an increased loss coefficient K

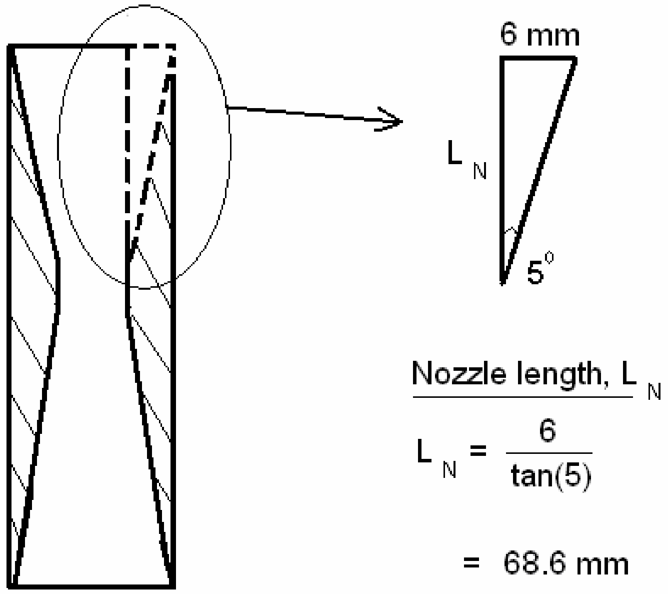
It is evident then, that to minimise losses through the restrictor entrance, a compromise must be taken between the length of the section and the taper angle.

The restrictor entrance region must provide for a reduction in section from the carburettor throat (32 mm) to the restrictor throat (20 mm). The area ratio is thus:

$$AR = A_2/A_1 = 20/32 = 0.625$$

As this AR is not given specifically in the table, the loss coefficients corresponding to AR = 0.5 will be used.

Using the minimum included angle provided (θ = 10°; taper angle = θ /2 = 5°), the length of the nozzle must be as shown in the following diagram:



This is a preliminary calculation, to be revised after investigating the exit diffuser length, and taking into consideration the overall length of the restrictor. It can, however, be seen from Figure 6.18 that the loss coefficient does not change appreciably for included angles between 10 and 40°; consequently, it is conceivable that the taper angle could be increased from anywhere between 5° and 20° without resulting in any further loss in pressure. This issue will be revisited at the end of Section 6.8.2.2.

▪ **Exit Region Design**

Forcing the intake charge to flow through sections of pipe of varying areas results in losses in pressure, as previously mentioned. After flowing through the mandatory 20 mm restricted zone, the flow will then be forced out through a conical diffuser to minimise the pressure drop. Regaining the maximum amount of pressure possible after this restriction is critical to providing a charge with the most potential for power production. In other words, achieving a minimal pressure differential between the restrictor inlet and outlet will produce a denser charge, with greater power potential.

Figure 6.19 gives pressure recovery coefficients, C_p, for diffusers. This figure shows that, for constant dimensionless length ratios, N/R_1, increasing taper angles produce higher coefficients of pressure recovery (for constant AR also). Higher C_p values indicate greater drops in pressure, resulting from higher pressure differences between diffuser inlet and outlet.

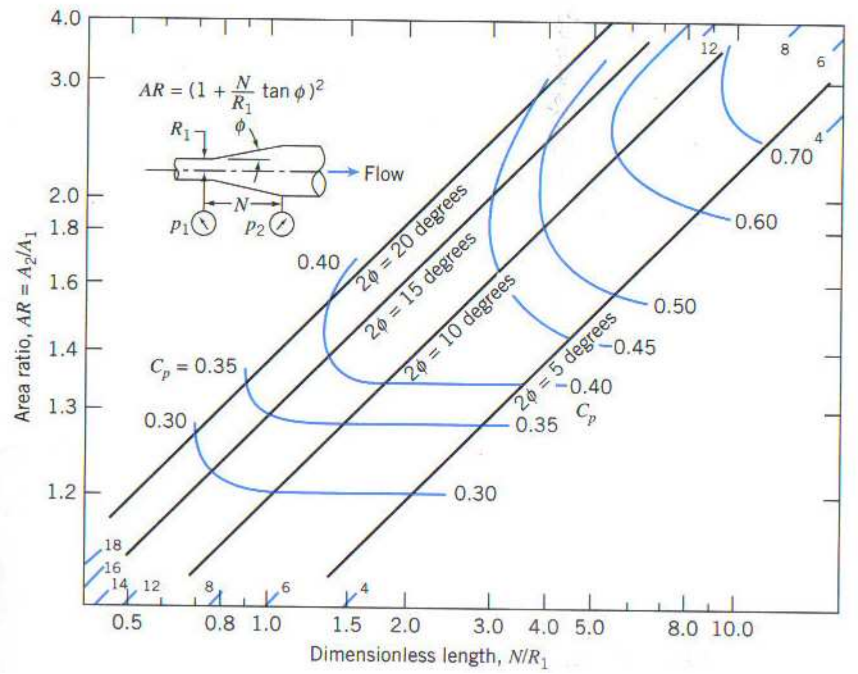


Figure 6.19: Pressure Recovery for Conical Diffusers

(Source: Fox & McDonald 2003)

Therefore, it is necessary to minimise taper angle or increase diffuser length to achieve minimal pressure loss

The overall length of the restrictor is a critical factor. A compromise must be made, however, to ensure that the restrictor entrance and exit regions are long enough to minimise the pressure gradient, while the overall restrictor must be kept to a suitable length to ensure all the intake system components fit within the allocated space.

The orientation of the engine within the car makes available approximately 700 mm of vertical length in which to locate all the intake system components (extending from the inlet ports to no higher than the main roll hoop). These components must include: air filter, carburettor/throttle body, restrictor, plenum (plus spacers) and intake runners.

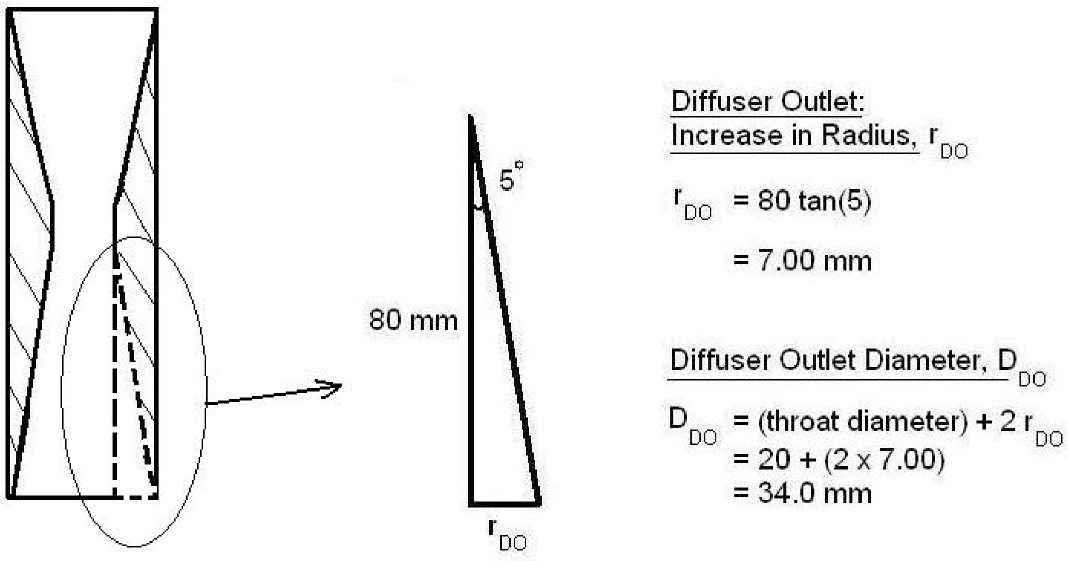
The length of the runners is desired to be kept to a maximum to take advantage of pressure waves which increase low- and mid-range torque (refer to Section 8.4.5).

With known heights for the filter, carburettor and plenum (plus attachments), some judgement is then used to allocate (maximum) space to the remaining components, as shown in Figure 8.1 (note that additional space required for joining plates, tubes, etc has been taken into account in this allocation).

The intake restrictor has been allocated a total length of 150 mm, aiding with packaging requirements and ensuring maximum length for the runners. This total length must then be proportioned into separate lengths for the three restrictor components. Aiming to maintain maximum length for the exit region, lengths were allocated in the following proportions:

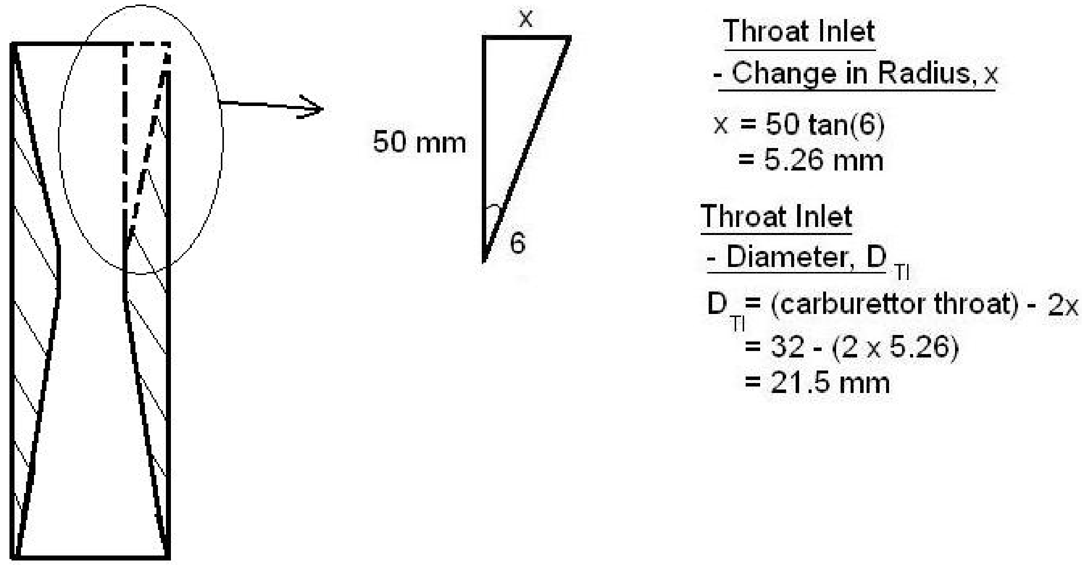
1. Entrance: 50 mm
2. Throat: 20 mm
3. Exit: 80 mm

Continuing with the exit region design, the maximum length available for the diffuser is now known, and progressing from the determination above that taper angles should be minimised, the diameter at the diffuser outlet can now be determined, as follows:



So, the restrictor exit region tapers from 20 m to 34 mm, over an 80 mm length.

The length of the entrance nozzle has now been specified as 50 mm, rather than the calculated 68.6 mm in *Entrance Region Design*. The taper angle will be increased to 6°, and the diameter of the entrance to the throat region is now:



The region where the diffuser outlet joins the plenum inlet is another critical point in the inlet flow path. Any sudden changes in area must be avoided to limit frictional losses; therefore, the restrictor outlet must taper gently into the plenum volume. Design considerations for this region are discussed in detail in *Exit Region Design*.

▪ Throat Design

Design parameters for the throat region dictate:

- maximum diameter of 20 mm
- optimisation of flow rate

Essentially, the throat must provide least restriction to flow by minimising frictional losses.

The inlet charge can be considered as flowing through an entrance region when travelling from the nozzle outlet into the throat. Therefore, entrance region design principles will govern the design of the throat. Sudden changes in area or directional flow changes are advised to be avoided, while rounded entrances are claimed to provide significantly reduced loss coefficients (Fox & McDonald 2003).

Following this advice, and considering the nozzle-diffuser arrangement, if the nozzle were to taper in to 20 mm and simply be joined to the diffuser at a point, a turbulent region would more than likely form at this point due to the sudden change in direction of flow (Figure 6.20).

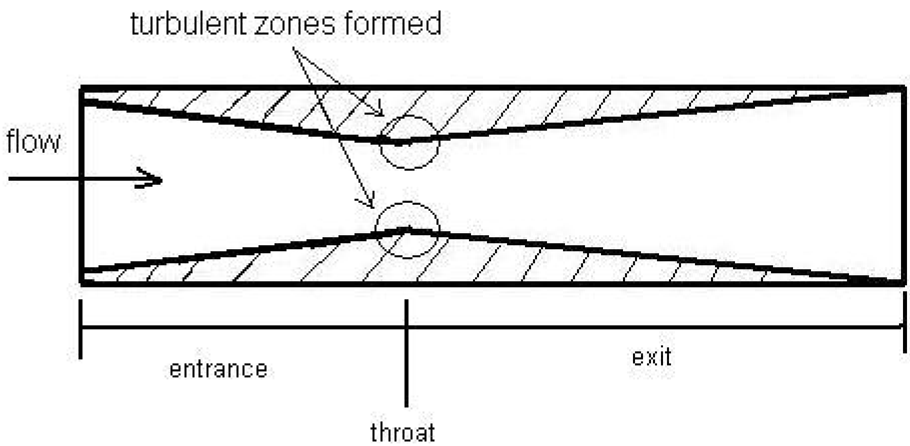
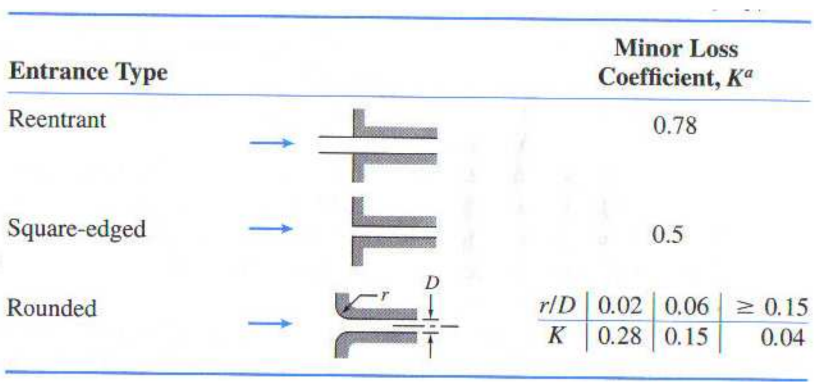


Figure 6.20: Turbulent regions formed around areas of sudden direction change

Therefore, the throat region, which provides the required 20 mm restriction, must be designed to minimise flow losses. Design of the throat region thus will incorporate a large bend radius, rounding into the minimum throat size of 20 mm, and leading back into the diffuser. Figure 6.21 shows almost negligible loss coefficients for r/D >= 0.15.



Entrance Type	Minor Loss Coefficient, K^a
Reentrant	0.78
Square-edged	0.5
Rounded	r/D: 0.02, 0.06, $\geq$ 0.15; K: 0.28, 0.15, 0.04

Figure 6.21: Loss Coefficients for Entrance Regions

(Source: Fox & McDonald 2003)

Given that the geometries of the restrictor entrance and exit have been determined, and knowing that the minimum throat area will be 20 mm, a bend radius of 70 mm is determined to provide a suitable throat shape. For this configuration, r/D is:

$$r/D = \frac{70}{20} = 3.5$$

This ratio is much greater than 0.15, as given in the table above. Therefore, it can be said with confidence that any losses resulting from this change in section will be negligible.

The design of the restrictor has now been determined and is represented diagrammatically in Figure 6.22 below:

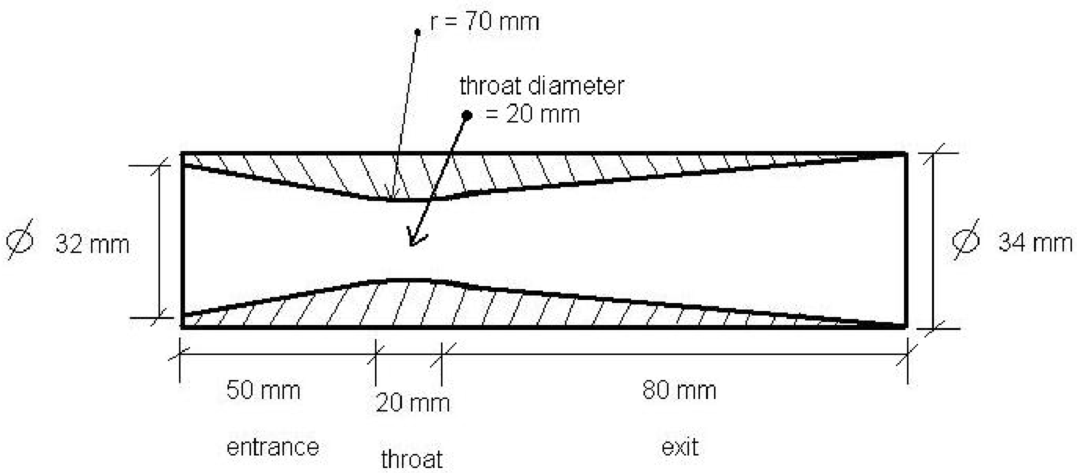


Figure 6.22: Overall design of intake restrictor

This restrictor was manufactured by the Mechanical Engineering workshop at USQ, by machining a single block of aluminium. Material choice in this case was important for minimising weight and achieving favourable heat flow characteristics. A further improvement of this component, to reduce weight, would be to machine the outside of the tube, forming a taper that follows the internal shape of the restrictor, leaving a wall thickness of around 5mm. The proposed outside shape of the restrictor is shown in Figure 6.23:

Figure 6.23: Aluminium restrictor, with proposed outside taper to reduce weight

6.8 Conclusion

The overall manifold design has evolved through a series of preliminary designs, and should provide efficient flow and combustion, with improved fuel efficiency and increased power output. While compromises have been necessary in some aspects of this design, the overall design philosophy has been maintained, with the intake system designed in accordance with engineering principles and empirical data.

The dynamometer modified as part of this research project was intended to facilitate testing of the intake manifold. However, as will be discussed in Chapter 10, this dynamometer proved ultimately to be unsuitable for testing the YZF600 engine with the Formula SAE intake configuration. Thus, performance gained by this intake design was unable to be quantified, but the design methodology implemented should ensure benefits to engine performance.

Chapter 7
Fuel Delivery Methods

7.1 Introduction

Method of fuel delivery is an important consideration in the overall design of an intake system. Carburetion and electronic fuel injection (EFI) are the two methods available, and the choice of method for a particular application is dependent on engine operating requirements, cost and engine compatibility.

Development of an electronic fuel injection system is not an objective of this project; however, some fundamental theory regarding EFI operation and performance is presented in this chapter, to provide an overview of the basic operation of injection systems.

7.2 Requirements of a Fuel Delivery System

A fuel delivery system's fundamental purpose is to provide fuel to an engine in the correct proportions. This system must provide the following facilities:

- accurate metering
- adjustability
- fuel efficiency

Fuel delivery systems should achieve a compromise between delivering charge for maximum power and maintaining fuel economy. The following sections explore the abilities of carburetion and EFI systems to achieve these objectives.

7.3 Types of Fuel Delivery Systems

7.3.1 Carburetion

Carburettors have long been the primary means of metering fuel. Their design and operation has been refined over a substantial period of time to become one of the most highly developed aspects of the internal combustion engine (Shoemark, 1981). The chief components contained within a carburettor housing include a throttle plate, venturi, fuel bowl and jet (Figure 7.1).

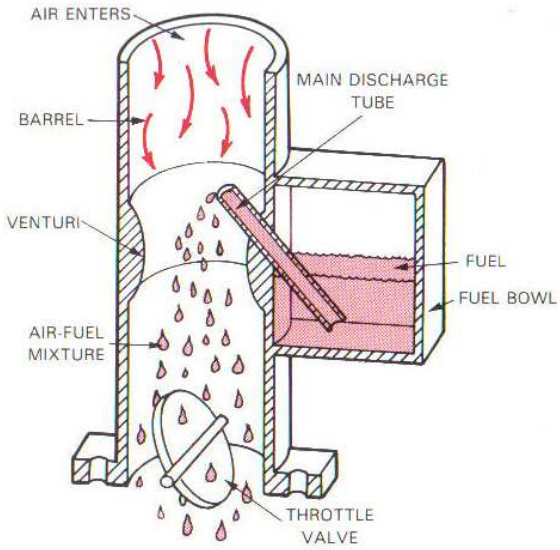


Figure 7.1: Basic components of a carburettor

(Source: *Stockel, Stockel & Duffy* 2001)

7.3.1.1 Basic Operation

Fuel is introduced into the airflow within the carburettor housing, proportionate to the volume of air flowing through the throat. The reduction in section of the venturi throat causes an area of low pressure; a passage connects the fuel reservoir, maintained at a constant height by a float valve, to the throat at the point of maximum constriction (Figure 7.2); this pressure differential caused by the venturi draws fuel from the higher pressure region of the fuel bowl into the lower pressure airstream, by way of the system attempting to maintain pressure equilibrium. The main jet is used to control the amount of fuel entering the airstream by extending or retracting further to create a smaller or larger passage for fuel to travel through.

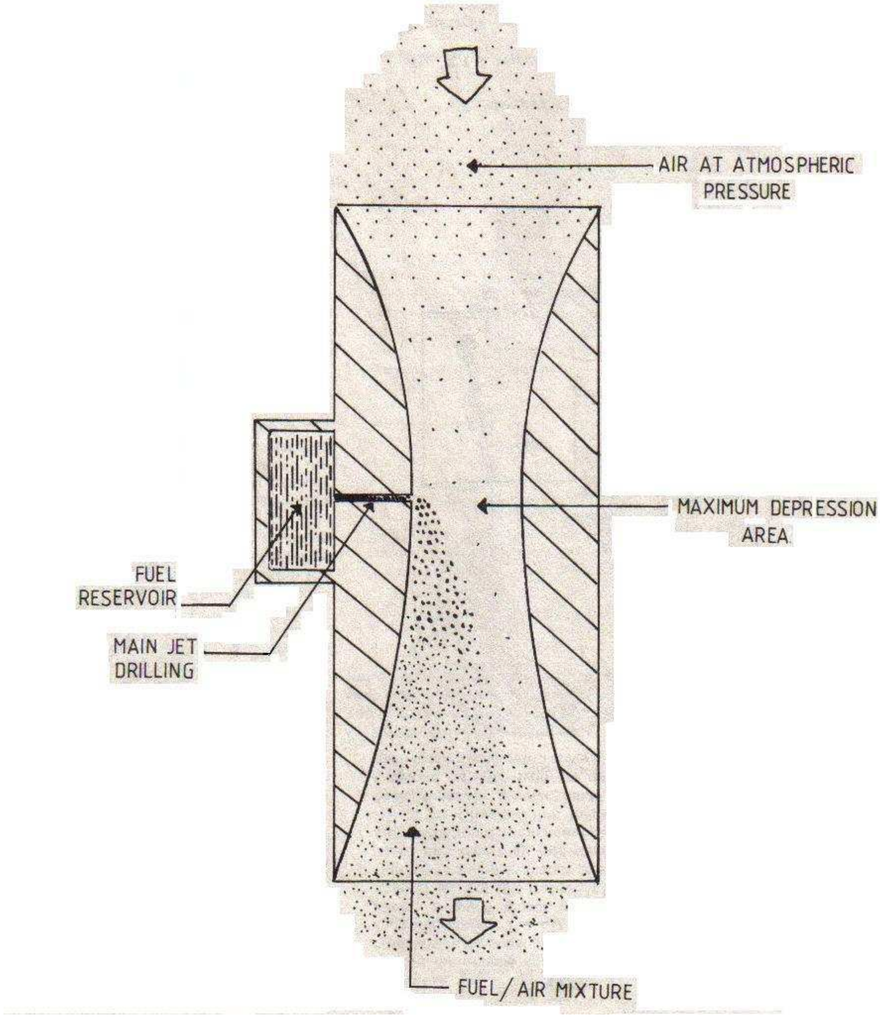


Figure 7.2: Venturi throat induces low pressure region, drawing fuel into airstream

(Source: Shoemark 1981)

The velocity at which the fuel is drawn into the airstream enables mixing of the two fluids, and promotes fuel atomisation. This air/fuel mix is metered through the throttle plate, and then enters the inlet tract, flowing to each cylinder.

7.3.1.2 Carburettor Types

A number of carburettor types exist, with the appropriate type for a particular application being determined largely by the engine's operating requirements.

Commonly available carburettors include downdraught, Weber, SU and CV (constant vacuum).

7.3.1.3 Carburettor Throat Size

Choosing the most appropriate throat size for the intended application is crucial to achieving optimum fuel delivery. Carburettor throat (throttle) diameter (Figure 7.3) must be compatible with the intake requirements of the engine, providing sufficient flow through all ranges of operating requirements.

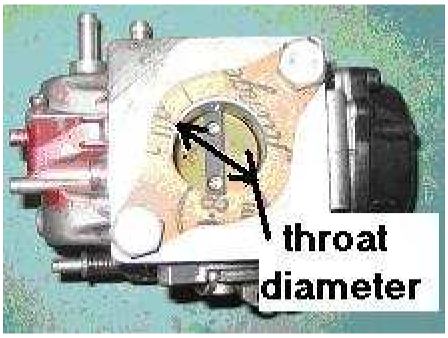


Figure 7.3: Choosing a carburettor throat size compatible with the engine is vital to overall engine performance

For low-power, restricted airflow applications, Bell (1997) recommends small throat sizes, in the range of 28 - 34 mm. These small carburettors are commonly available on small road cars, and can be purchased from car wreckers for around \$40 - \$50 each (Malpress R 2005, pers. comm., 4 October).

7.3.1.4 Tuning

The ability to properly tune a carburettor is essential to improving overall engine and vehicle performance. Carburettor tuning must provide sufficient flow to produce peak power at certain speeds, while also conserving fuel at lower speeds.

Modification to the main jet can be used to achieve the desired fuel flow rate, and hence mixture characteristics, with the general method outlined below.

- **Modifying the Main Jet**

  The main jet within a carburettor extends or retracts due to pressure variations, to control the amount of fuel entering the airstream. Movement of this jet creates a physically smaller or larger passageway through which fuel can travel. Main jets remain straight along most of their length, but are tapered towards the end. It is this tapering that provides the mechanism of controlling the amount of fuel flow, as the taper angle and position of the jet creates a smaller or larger clearance for the fuel to travel through (Figure 7.4).

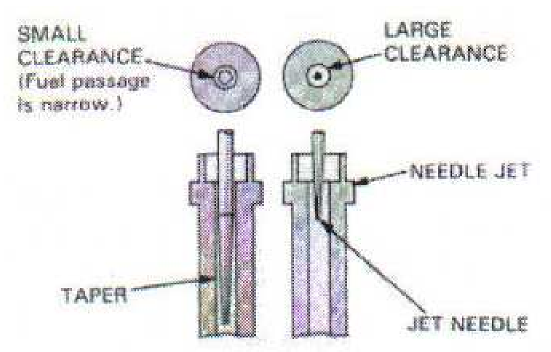


Figure 7.4: Jet taper, enables variation in size of fuel flow passage
(Source: *Performance Streetbike Magazine* No. 89)

The taper angle of a jet can therefore be modified to produce different mixture properties. For example, increasing the taper angle provides a richer fuel ratio, which can be used to achieve increased power.

7.3.2 Electronic Fuel Injection (EFI)

Fuel injection systems are computer-controlled and disperse fuel via injectors. A throttle body is used to meter air, similar in principle to a carburettor, with a throttle plate controlling air flow.

In addition to the injectors and throttle body, EFI systems require a number of components for correct operation, including fuel rail, pressure regulator, Engine Control Unit and fuel pump. A variety of sensors are also required, enabling operating conditions to be monitored, and hence fuel requirements to be determined. Variables commonly measured by these sensors include oxygen percentage, manifold pressure, airflow, throttle position and air temperature.

7.3.2.1 Basic Operation

Fuel injection systems offer the ability to pre-program into the ECU a set of air/fuel ratios for various operating conditions. An engine dynamometer can be used to generate a fuel map, by determining the ratios required at numerous operating conditions, for economy, full power, or a comprise between the two. During operation, sensors in the engine and exhaust system deliver information to the ECU, which is compared with the pre-set data, enabling the computer to communicate the requirements for air/fuel delivery to the related components.

Fuel injectors are fed pressurised fuel by a fuel hose at one end, and contain a metering nozzle at the other (Figure 7.5). Fuel is injected via an electromagnetic valve-type mechanism. The body of the injector is surrounded by a coil, which, when current passes through it, generates a magnetic field. A valve connected to a spring in the injector nozzle is pulled into the injector body by the magnetic field, creating an orifice for pressurised fuel to travel through. When the current supply is stopped, the spring causes the valve to return to its initial position, preventing fuel flow.

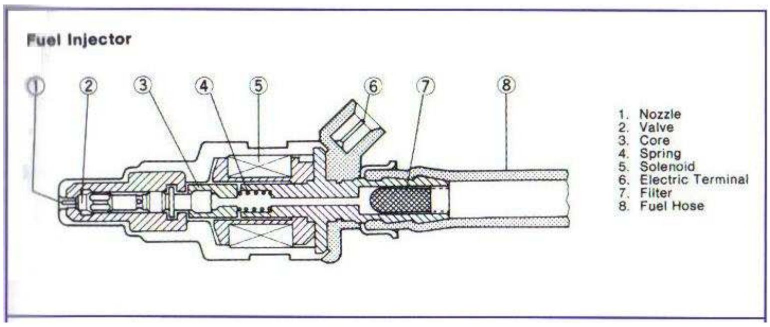


Figure 7.5: Components of a fuel injector

(Source: Wade 2004)

7.3.2.2 Pulse Width

The period of time for which fuel is able to flow through the injector nozzle is called the injector *pulse width*. It is these pulse widths that are determined during the fuel mapping process and programmed into the ECU, thus delivering the required amount of fuel for particular operating conditions.

Fuel injection systems also monitor operating conditions constantly, and can therefore adjust pulse widths accordingly. This adjustment is performed on the basis of the information provided to the computer by various sensors.

7.3.2.3 Role of Sensors

Sensors play a key role in EFI systems by constantly monitoring engine operating conditions to provide the ECU with instantaneous data relating to fuel supply requirements. A number of sensors perform these vital roles and are discussed briefly below.

- **Throttle Position Sensor**
  The throttle position sensor determines how far the throttle is open, and therefore the amount of air entering the engine.

- **Oxygen Sensor**
  An oxygen sensor (Figure 7.6) located in the exhaust system detects the percentage of oxygen in the gases of combustion, and therefore determines whether the mixture is rich or lean.

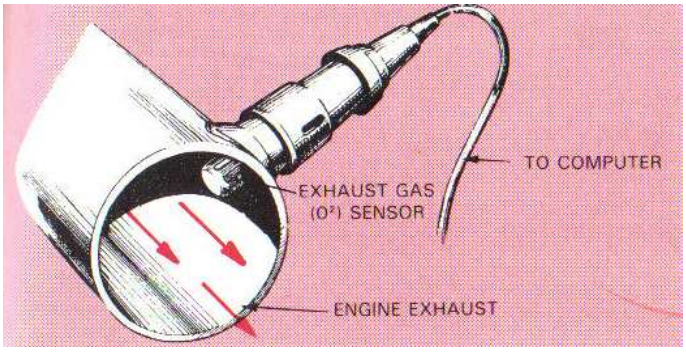


Figure 7.6: Positioning of an oxygen sensor within an exhaust manifold
(Source: Stockel, Stockel & Duffy 2001)

● **Mass Airflow Sensor**
The mass airflow sensor is positioned at the end of the air intake tract, just prior to the throttle body, and detects the amount of air entering the system.

● **Manifold Absolute Pressure (MAP) Sensor**
The MAP sensor monitors pressure in the inlet manifold, and from this is able to indicate the power being produced by the engine. The more air drawn into the engine reduces the pressure in the manifold; a greater volume of air entering the engine also enables greater power production. Therefore, by monitoring the manifold pressure, the MAP sensor is able to estimate the power produced by the engine.

● **Temperature Sensor**
A temperature sensor monitors the temperature of inlet air, and from this determines air density. Air density affects the mass of air in a given volume, and this information is used, together with that provided by the mass airflow sensor, to determine the volume of air entering the engine.

● **Engine Speed Sensor**
The engine speed sensor monitors the speed of the engine. This information contributes to determination of injector pulse widths.

7.3.2.4 Types of Injection

A variety of injection methods exist, including multi-point injection and throttle body injection. Each of these methods offers advantages and disadvantages.

- **Throttle Body Injection (TBI)**

Throttle body injection systems, also known as single-point or central injection, require a single injector, mounted in the throttle body housing. TBI was the first method of fuel injection developed (*www.howstuffworks* 2005) and effectively provides a 'bolt-on' replacement option to carburetion systems.

Similar to carburettors, fuel is introduced into the airstream within the throttle body housing. However, the injector significantly improves fuel atomisation, providing a very combustible mixture to the cylinders. TBI systems therefore provide much improved fuel efficiency and power over traditional carburetion systems.

As will be discussed in the next section, multi-point fuel injection offers further benefits over single-point systems; however, single-point injection requires fewer components than multi-point, and is therefore a cheaper alternative.

- **Multi-Point Injection**

Multi-point, multi-port or direct injection employs one injector in each cylinder. Injectors are generally located close to inlet valves, and therefore offer better efficiency as atomised fuel is injected directly in the vicinity of the combustion chamber, with losses due to flow through the manifold reduced. Atomisation is also improved as fuel is sprayed at the inlet valve, causing dispersion.

Multi-point injection can be further classified as *sequential* or *bank firing*. Sequential injectors spray fuel only when their respective cylinder is about to begin its intake stroke; that is, injectors follow the engine's firing order, and each sprays only once per full set of cylinder firing.

Bank firing causes all injectors to spray at a single time. Therefore, in a four-cylinder, four-stroke engine using bank firing, injectors will each spray four times for two crankshaft revolutions, whereas in a sequential system, injectors spray only once each for two revolutions of the crankshaft.

Sequential injection offers more efficient fuel consumption and can better meet the fuel requirements of each cylinder. However, these systems are somewhat more complicated to implement than bank firing systems, requiring additional programming of the ECU.

7.3.2.5 Injector Design Criteria – Positioning and Size

Determining optimum positioning of injectors within the intake system is an important design criterion. Bell (1997) recommends (for multi-point systems) that injectors be placed as close to intake valves as possible. The reasons behind this positioning are that increased atomisation will be achieved through aiming injectors directly at the valves, while maintaining atomisation will also be more likely as fuel has to travel only a short distance to enter the combustion chamber. Fuel efficiency is also increased due to minimised flow losses and improved vaporisation.

Choosing an appropriate injector size is also crucial. The size of an injector relates to the amount of flow it can provide, commonly rated in cc/min. The injector(s) must be able to provide an adequate volume of fuel to satisfy peak operating requirements, while also being capable of supplying a reduced volume at idle, so that the engine does not run rich.

7.3.2.6 EFI Performance

Electronic Fuel Injection provides improved performance over carburetion. EFI generally provides better combustion, improved fuel efficiency and increased power, by using a more direct, efficient method of fuel delivery, and offering the capability to adapt to changing conditions automatically.

7.4 Recommendations

Electronic Fuel Injection is expensive to implement, requires a variety of additional components, takes time to properly tune and usually requires expert technical knowledge to implement and tune appropriately. However, the advantages offered by this method of fuel injection over carburetion, in terms of increased fuel efficiency, power and tuning ability, determine fuel injection to be the much preferred method of fuel delivery for the Formula SAE car.

Yamaha developed its first model with fuel injection as standard in 2000 (Toowoomba Yamaha Service Manager 2005, pers. comm., 7 July). The 1994 Yamaha YZF600 engine to be used in this year's car is therefore not fuel injected, and comes standard with quad, 32 mm Mikuni carburettors; conversion to injection would thus be required. Toowoomba Yamaha does not generally perform this type of conversion (Toowoomba Yamaha Service Manager 2005, pers. comm., 6 July); however, components from Yamaha models offering EFI as standard can be purchased and used to attempt a conversion from carburetion to injection. Schematic diagrams showing injection components for a 2004 model YZF-R1 (1000 cc) are provided in Appendix B.

Some brief recommendations regarding injection method and component choice are now presented, to provide a direction for future development work.

- **Injection Method**

  Sequential, multi-point fuel injection is recommended, as research has indicated this system type to provide the most efficient, effective form of fuel delivery. Tuning for individual cylinders is facilitated, enabling more economic use of fuel and more balanced creation of power.

- **Injector Positioning and Size**

  Recommendations for injector positioning are provided by Bell (1997). From these recommendations, positioning of injectors within the inlet ports, spraying directly at intake valves, is the most desirable option, to increase fuel atomisation as well as the volume of charge entering the cylinders. Injectors can be mounted via bosses on each of the intake runners to accommodate this desired positioning, being fed by a common fuel rail at the alternate end (Figure 7.7).

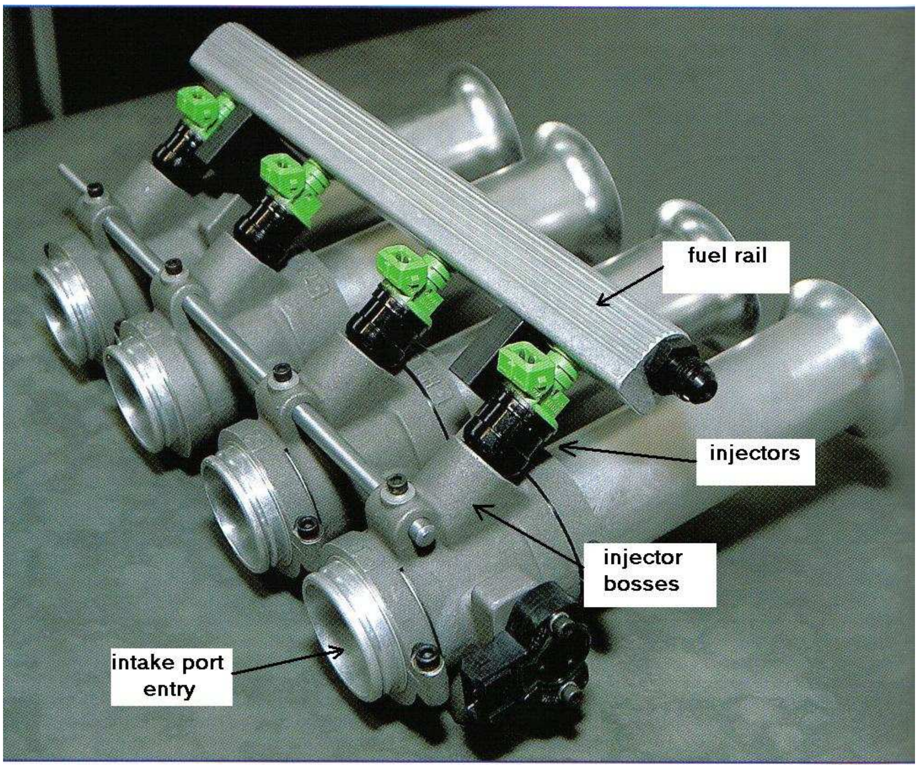


Figure 7.7: Connection of fuel rail and injectors to inlet manifold (multi-point injection system)

(Source: Wade 2004)

In the application of Formula SAE, considering engine capacity (< 610 cc) and air intake ability (limited by the restrictor), very small injectors are required. It has been recommended that injectors used on a 1.2 L Hyundai Excel are capable of providing good idling characteristics, as well as sufficient flow for peak power requirements (Molloy R 2005, pers. comm., 10 August).

● **Throttle Body Size**

Throttle body sizes are rated according to throat (throttle plate) diameter. Bell (1997) states that empirical results have shown that increasing throttle bore size correspondingly increases power at high rpm. Conversely, small throat sizes have been found to produce more finely atomised fuel droplets and allow more uniform burning, to produce greater power in the low- to mid-ranges. Therefore, a throttle body with a small throat diameter, such as 32 mm, is recommended.

7.5 Conclusion

Implementation of a fuel injection system is strongly recommended for the SAE car, as EFI can provide substantial benefits over carburetion, as presented in section 7.3.2.6. Full development, implementation and testing of a fuel injection system can follow on from the design recommendations outlined in section 7.4, for use in future years' race cars at USQ.

Chapter 8
Exhaust System Fundamentals

8.1 Introduction

The car's exhaust system can have a significant impact on its engine performance. Just as the intake system must deliver air and fuel to all cylinders as quickly and efficiently as possible, the gases produced by combustion must also be extracted expediently to optimise the cylinders' potential for reception of new intake charge. Providing an unrestricted path for expulsion of gases and achieving appropriately tuned lengths to optimise wave scavenging potential can markedly improve overall engine power and throttle response.

8.2 Exhaust System Fundamentals

The primary function of an exhaust system is to provide a means of removing gases of combustion from the combustion chambers. The purpose is to create an empty volume above the piston head which can then be filled with a fresh intake charge, ready for firing.

As illustrated in Figure 8.1, exhaust systems typically comprise:

- exhaust manifold
- exhaust pipes
- tailpiece
- muffler

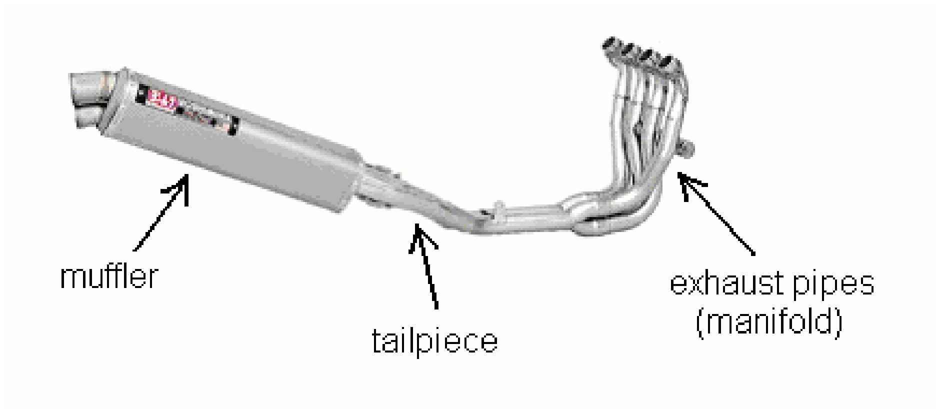


Figure 8.1: Components of a typical exhaust system

(Source: www.yoshimura-rd.com)

It is desirable for any exhaust system to perform this function in the most efficient and effective manner possible, providing least restriction to flow, and hence producing benefits for improved induction potential and creation of power. These objectives are achieved through the following functions:

- minimising restriction to flow (section 8.4.3)
- minimising back pressure (section 8.4.3.1)
- reducing charge contamination (8.4.4.1)
- use of wave dynamics through tuned lengths (section 8.4.5)

Specific applications require very specific performance goals from an exhaust system, and accordingly there are many different types of exhaust manifolds to suit various purposes. Most often, performance applications employ custom designed and made exhausts, called headers, to extract maximum power from an engine; on the other hand, mass-produced road cars prioritise ease of manufacture and fit, low-cost and minimal emissions, making such exhaust systems very limited in their ability to contribute to improved performance. Various manifold types and their applications are explored in section 8.3.

8.3 Types of Exhaust System

The appropriate exhaust system depends very much on the intended application. In the broadest sense, these systems fall into two general categories: exhaust (cast manifold) systems for mass-produced vehicles, and extractor systems for performance applications. Within these categories there are many variations. The following two sections present discussions manifold types, with the focus on extractor systems, as such a design will be pursued for the 2005 USQ Formula SAE race car.

8.3.1 Exhaust (Cast Manifold) Systems

In general terms, an exhaust system comprises an exhaust manifold, exhaust pipe(s), catalytic converter, muffler and tailpipe. Such systems, as represented in Figure 8.2, are commonly found on standard road vehicles.

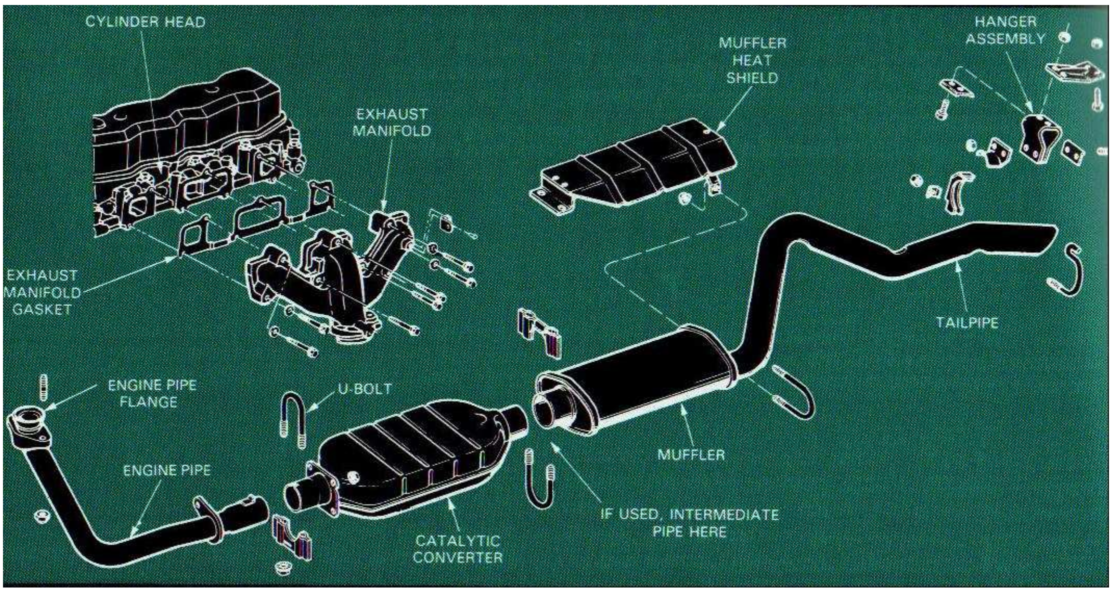


Figure 8.2: Typical exhaust System Components

(Source: Stockel, Stockel & Duffy)

In this type of standard exhaust system, the exhaust manifold is a metal casting that bolts over the exhaust ports on the cylinder head (Stockel, Stockel & Duffy 2001). Exhaust gases from each port are typically combined at a common collector, and fed out through a single exhaust pipe. The resulting volume then flows through a catalytic converter and into a muffler, and, eventually, out into the atmosphere.

Standard exhaust systems are mass-produced, often with one design being fitted to a number of vehicles, and prioritise ease of manufacture and fit, low emissions and cost effectiveness. In other words, their contribution to engine performance is minimal, and, in fact, often detrimental.

Restriction to flow is a common symptom of these systems, due to the usually single exhaust pipe, which is often small in diameter and contains bends and other impedances. Contamination of charge is another typical complaint, resulting from the usually short manifold runners, which encourage gas mixing and backflow. These issues both contribute to reduced engine performance, and are discussed in detail in upcoming sections.

Some modifications can be made to standard manifolds in the pursuit of performance, such as the addition of an internal dividing plate to separate flow and increase effective pipe length (refer to section 8.4.4.1). However, racing and performance applications most often use a set of custom headers to improve performance.

8.3.2 Extractor Systems

Exhaust systems for race applications differ from road usage in that they must contribute to the vehicle's overall pursuit for optimum performance. High performance exhaust systems therefore make use of free-flowing pipes for unrestricted gas flow, and tuned lengths to maximise wave scavenging abilities.

Extractors (commonly called headers) utilise separate pipes for each exhaust port, and are typically characterised by long-length pipes. Depending on the configuration, a set of extractors may consist of primary and secondary pipes, or primary pipes alone (Figure 8.3).

Primary pipes extend directly from the exhaust ports, and may be joined at a single, common collector which then flows into a single pipe prior to the muffler; or, the primaries may flow into multiple collectors which are followed by secondary pipes, before combining into a single pipe. A common four-cylinder extractor configuration is shown in Figure 8.3. This 4-into-2-into-1 arrangement connects two pairs of primary pipes into dual secondaries, before combining all gas flow into a single exhaust pipe, leading into a muffler.

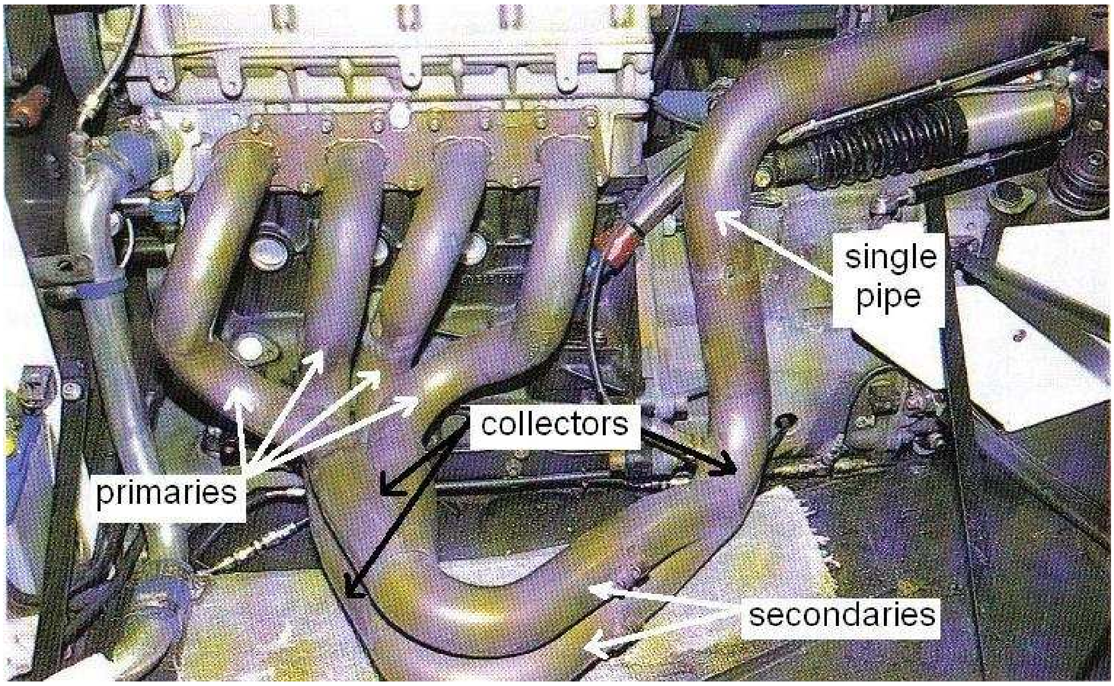


Figure 8.3: 4-2-1 race extractors

(Source: Edgar 2001)

Extractors primarily improve exhaust flow characteristics by reducing restriction. They also utilise tuned lengths which take advantage of reflected pressure waves (section 8.4.5), assisting in the extraction of additional exhaust gases from the combustion chambers, after the initial exhaust stroke. The benefit is seen in making available a greater volume to be filled by intake charge, while the lower pressure area induced also has the effect of drawing in additional charge. Ultimately, the result is increased power through more efficient extraction and induction.

The principles governing extractor performance improvement involve a variety of factors, and are discussed in detail in the proceeding sections.

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8.4 Extractor Design

8.4.1 Objectives

An exhaust system's main objective is to remove gases from the immediate vicinity of the exhaust ports and to pipe these gases out of the system in as little time as possible. Extractors aim to perform this function more effectively than standard manifolds, with efficient removal of combustion gases also assisting in inducing intake charge. Paramount to this objective is improving flow through the exhaust pipes, which, together with use of tuned pressure waves, provides extractors with their basis for improved engine performance.

The main objectives of an extractor system are then to:

- minimise flow restriction
- reduce charge contamination
- induce induction ramming effects
- employ exhaust scavenging abilities

The methods employed to achieve these objectives are now discussed.

8.4.2 Operating Principles

The main operating principles of an extractor system are to attempt to maintain separation of flow between individual exhaust ports for the maximum length possible;

use large, well-directed, free flowing exhaust pipes; and use tuned waves to increase induction potential. Ensuring that exhaust gases follow a very direct and unimpeded path from port to collector, and minimising interference of flow between cylinders is crucial to producing improved power characteristics.

Numerous factors affect the ability of these principles to work optimally, and interact with each other to contribute to overall system performance.

8.4.3 Minimising Restriction to Flow

The removal of combustion gases from the cylinders must occur as quickly and efficiently as possible, as induction potential is decreased with any remaining amount of spent gases. Therefore, the flow path from exhaust port to pipe end must be as unrestrictive as possible.

Minimising restrictions in the exhaust system is perhaps the most critical function performed by an extractor system. Restrictions to flow cause pressure to build up within the system, reducing expulsion ability, encouraging flow mixing and leading to charge contamination. Overall, pressure build-ups cause losses in power (Edgar 2001).

A number of factors contribute to exhaust system flow characteristics, and must be considered in design for performance applications.

8.4.3.1 Back Pressure

The term ‘back pressure’ is often used in relation to exhaust systems and has an effect on engine power. Pressure is built up within the system due to the motion of combustion gases regularly pulsating from exhaust ports. This back pressure restricts the motion of gas flow, and is an undesirable quantity when chasing increased engine performance. Back pressure typically reduces engine power (Stockel, Stockel & Duffy 2001; Edgar 2001) and should be minimised at all times. Enlarging exhaust pipes and using free-flowing mufflers reduces back pressure and helps to increase power (Stockel, Stockel & Duffy 2001).

8.4.3.2 Flow Area

The first point of concern for maximising flow is to enlarge the exhaust pipes. Larger exhaust pipes provide a greater flow area for transportation of combustion gases, decreasing restriction to flow. Fox and McDonald (2003) show that for a constant rate of flow, Q, an increase in flow area, A (cross-sectional area of exhaust pipe), will result in a proportionate decrease in flow velocity, v, according to Equation 8.1:

$$Q = v\mathrm{A} \tag{8.1}$$

Therefore, a compromise must be reached to ensure that exiting gases have adequate flow area available to travel effectively unrestricted through the exhaust pipes, while also travelling at sufficient speed to ensure efficient expulsion from the system.

The speed of events occurring within a typical engine should generally provide more than sufficient velocity to the intake charge. However, an excessively large diameter pipe extending from the exhaust system may show some detrimental effect on the velocity of the exiting gases, and would be otherwise impractical.

For all intents and purposes, it will be assumed that the speed of the gas at expulsion from the combustion chambers will be sufficient to ensure adequate flow, and so the main concern regarding flow restriction will be in determining appropriate pipe diameters and lengths. Detailed determination of these parameters is covered in section 9.6.1.2.

8.4.3.3 Pipe Shape

The second factor relating to minimal restriction concerns pipe shape, and includes maintaining integrity of shape. Shaping of exhaust pipes is generally dictated by the space available for fitment in the vehicle. Therefore, exhaust systems invariably contain numerous bends and curves to ensure proper fitment and adequate clearance. Fox and McDonald (2003) state that the head loss through a bend is larger than for fully developed flow through a straight section. It is virtually unavoidable then that some losses will be incurred due to the requirement that the pipes be bent to fit around various components within the vehicle. Minimising these losses is then achieved through using larger bend radii and high quality methods of bending.

The method used to achieve the desired bends can play a critical role in minimising flow losses. Equipment used in the bending of straight pipe often results in mis-shapen pipes, either through flattening, causing a reduction in section, or other issues such as crimping (Figure 8.4).

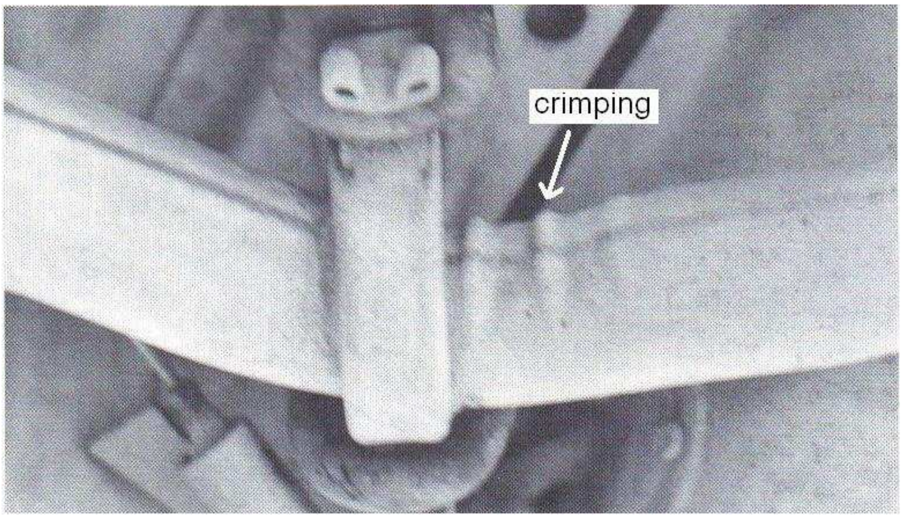


Figure 8.4: Crimping, resulting from poorly formed bends, reduces power
(Source: Bell 1997)

Bell (1997) states that poorly formed bends, resulting in crimping or creasing, cost horsepower right through the power range, with the effect more pronounced the closer the bend is to the engine.

Any imperfection in pipe shape created by bending and fitting results in an increase in pressure of the fluid travelling through the system, leading to increased restriction to flow and a reduction in gas speed. Maintaining integrity of shape is therefore critical in limiting flow restriction.

Two methods of bending exhaust pipes are generally available: pipe bending and mandrel bending. These methods are discussed below.

- **Pipe (Press) Bending**

  Pipe benders perform their bending action by feeding a length of pipe between two rigid edges, forcing the pipe to follow the prescribed shape (Figure 8.5). This 'external' method usually results in flattening of the pipe on one or both sides contacted by the rigid guides. Hence, the resulting bends do not retain their full internal cross-section (Edgar 2001), causing restriction to flow.

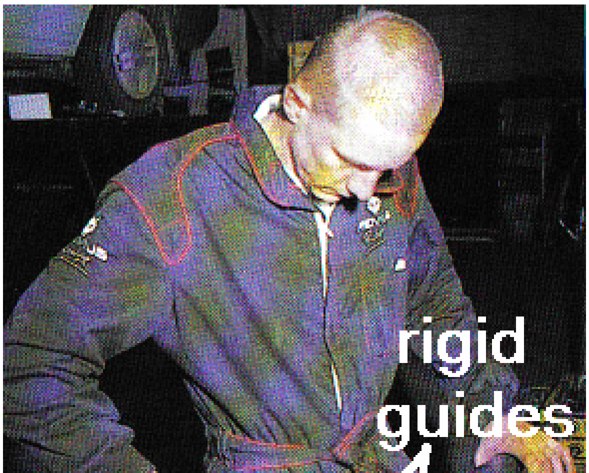


Figure 8.5: Typical press bending process

(Source: Edgar 2001)

- **Mandrel Bending**

  Mandrel bending uses a 'die' which is forced internally through the section of pipe. This spherical die maintains roundness of the section all times, resulting in a bend that maintains integrity of shape.

● **Comparison of Bending Methods**

Pipe benders can offer quite acceptable results, sometimes with little effect on flow performance. However, mandrel bends are generally regarded as being much superior to press bends (Edgar 2001), particularly for tighter bend radii. Mandrel bends are, however, significantly more expensive than pipe bends, as they require special equipment and more labour time. In applications where maintaining integrity of section is crucial though, mandrel bends are usually required.

Flow testing can be performed to determine the effect of bending method on pipe flow capabilities. Edgar (2001) has carried out such testing and presents results for a range of mandrel and press bends, in 64 mm (OD) pipe, ranging from 45° to 180°. In each case, an equivalent length of straight pipe is used as a control to determine the relative reduction in flow resulting from the method of bend (see Figure 8.6).

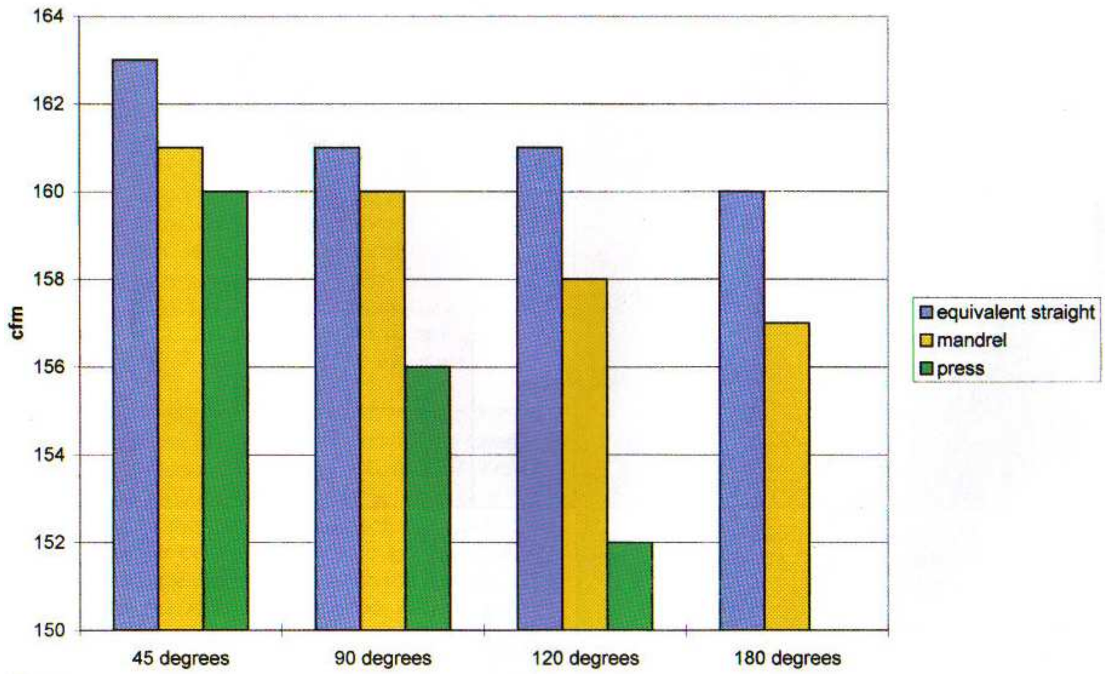


Figure 8.6: Comparison of flow rate between mandrel and press bends, and equivalent lengths of straight pipe (note that flow rate data is presented in cubic feet per minute, cfm, in this instance)

(Source: Edgar 2001)

This chart shows that both types of bend produce very good flow characteristics for smaller bend angles, with very little reduction in flow capability compared with that of an equivalent length of straight pipe.

The difference in flow rates becomes more pronounced with increasing angles of bend, with mandrel bending clearly showing its superiority over press bends.

A very significant point to note is the very minor reduction in flow capacity through a mandrel bend compared to that of an equivalent straight pipe. This testing shows that the effect on flow of introducing a bend into an exhaust pipe can be significantly minimised by using good quality bends, and further emphasises the importance of maintaining integrity of shape in such systems.

A common method of constructing custom extractor systems is to use a number of mandrel bends, cut and welded to achieve the necessary curves, to form a single pipe of the desired length. While it is well-accepted that mandrel bends produce superior flow characteristics to press bends, and the flow testing results above agree, the quality of inside surface finish is also a significant factor. Numerous welded joints will often result in protrusions, or welding ‘dags’, on the inside of the pipe, as well as ‘steps’ resulting from the joining of numerous lengths of pipe. These imperfections will adversely affect flow through the pipe, and should be avoided as much as possible to produce a smooth inside surface finish. In some cases, such imperfections can cause mandrel-joined exhaust pipes to flow more poorly than press bent pipes (Edgar 2001).

8.4.4 Separation of Flow

8.4.4.1 Charge Contamination - Causes and Effects

Maintaining separate flow paths for combustion gases from individual cylinders reduces charge contamination and ultimately leads to improved power production.

Charge contamination occurs when new intake charge mixes with exhaust gases, and results in a general drop in power. Smith & Morrison (1971) state that this problem can occur to the extent that the mixture becomes ‘unfireable’, causing the engine to “miss a beat or two”.

● **Valve Timing**

Charge contamination is a result of valve timing and exhaust system configuration. It is common practice for many engine manufacturers to use timing where the exhaust and inlet valves are open simultaneously, at least for some amount of time. That is, as the piston reaches Top Dead Centre (TDC) on its exhaust stroke (with the exhaust valve obviously open), the inlet valve begins to open also. As the piston then begins to descend on its intake stroke, and starts to draw in fresh intake charge, the exhaust valve remains open for a short period of time. For this time then, both the inlet and exhaust valves are open simultaneously, in what is called ‘valve overlap’. During this time, any (negative) pressure forcing exhaust gases to flow back towards the exhaust port will result in charge contamination, by virtue of the open exhaust valve. Changes to valve timing can therefore reduce the extent of contamination; however, exhaust system configuration can also contribute to a reduction in contamination and is often a simpler method of reducing flow mixing and producing improved power.

Mixing of flow results in two detrimental effects: reduction in intake potential and prevention of expulsion of exhaust gases. The basic idea is that if flow from one exhaust port makes its way into the path of another, it causes pressurising of the affected cylinder. Therefore, there is a reduction in potential for this cylinder to accept intake charge, and power is reduced.

The misdirected flow also has the affect of preventing exhaust gases from leaving the combustion chamber completely, and forces these gases back into the chamber. This results in charge contamination, and generally reduced power production. In some cases, the charge can become completely unsuitable for firing, causing the engine to misfire.

● **Exhaust System Configuration**

Separating exhaust gas flow reduces the tendency for pressure build-ups and encourages better intake potential. Many cast exhaust manifolds contribute to contamination of charge and mixing of flow through short, ill-directed runners. In these systems, combustion gases often collect at a common point only a short distance from where the gases initially leave the individual ports. Common configurations also often encourage combustion gases from one port to rush back into another cylinder's port, rather than following a direct path to exit the system.

Maintaining separate flow for a prescribed distance is also crucial to taking advantage of reflected pressure waves, for improved combustion gas removal and assisted intake potential (see section 8.4.5).

The following illustrations indicate methods of maintaining separation and increasing runner length.

Figure 8.7 shows a typical connection between cylinders 1 and 4 on a cast exhaust manifold. It is clear from this illustration that exhaust gas rushing out of cylinder 1 will have a great tendency to flow into the more 'straight-on' connection to cylinder 4, rather than directly exiting the system, as required.

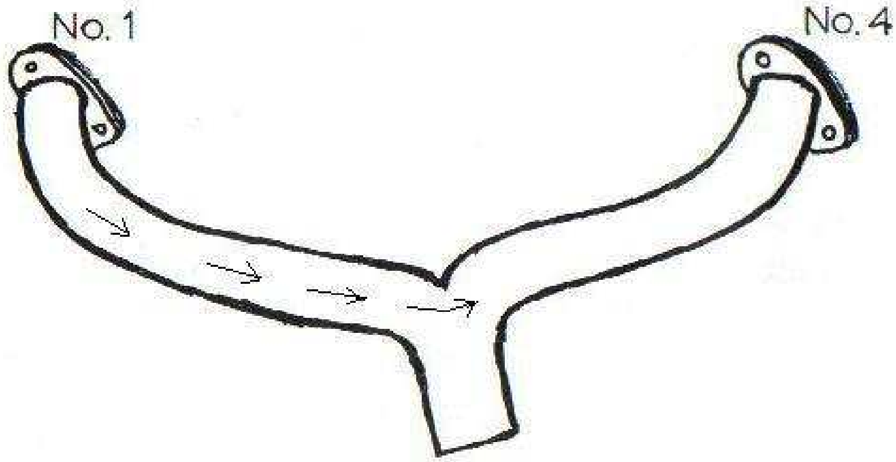


Figure 8.7: Path likely to be taken by exhaust gas exiting cylinder 1, causing back-flow into cylinder 4 and resulting in charge contamination

(Source: Bell 1997)

Figure 8.8 shows a simple modification, which redirects flow and greatly reduces backflow. The addition of a plate, welded between the pipes extending from cylinders 1 and 4, dramatically reduces backflow between these cylinders and therefore helps to prevent charge contamination.

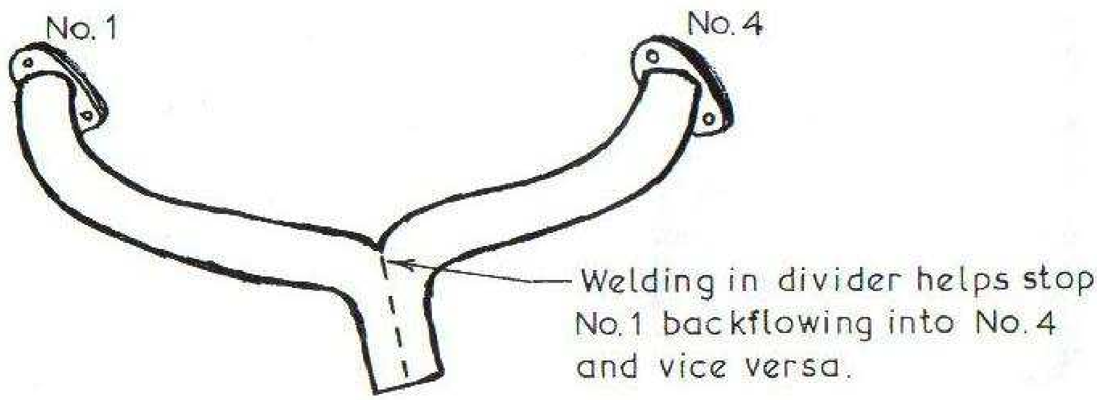


Figure 8.8: Simple modification to greatly reduce backflow
(Source: Bell 1997)

Inclusion of a dividing plate in a standard cast manifold in some way approaches header design, by encouraging separation of flow. Headers are designed with individual pipes extending from each exhaust port and generally extend quite a way before meeting at a common collector. In this manner, header design aims to significantly reduce exhaust gas mixing and therefore alleviates many problems associated with charge contamination.

8.4.5 Wave Tuning

Fluids travelling through induction and exhaust systems are in an unsteady state. These fluids are constantly subjected to regions of changing pressure, temperature and velocity. Lateral movement of pistons in cylinders throughout the four-stroke cycle causes continual pressure, temperature and velocity variations, and so affects the behaviour of the induction charge and exhaust gases flowing through the system's passages. In addition, the movement of these fluids themselves contribute to the formation of pressure waves which flow through the intake runners and exhaust pipes. The ability to take advantage of such pressure waves by correctly tuning intake and exhaust systems to maximise induction ramming and exhaust scavenging effects constitutes a vital aspect of extractor design.

Pressure wave behaviour is a complex phenomenon. There are myriad of factors contributing to wave creation and behaviour, many of which cannot easily be controlled and/or monitored. Accurate tuning of an induction and/or exhaust system therefore relies heavily on extensive and reliable testing. Atherton (1996) suggests there are only two methods able to determine effective tuned pipe lengths and diameters: dynamometer testing and pressure wave simulation, using an appropriate computer simulation package.

It is very difficult then, without reliable analysis equipment and extensive testing, to accurately determine appropriate lengths and diameters for inlet and exhaust pipes. There are, however, some widely-accepted empirical equations that can be used to determine initial estimates for pipe lengths and diameters. Design of the inlet and exhaust systems for the 2005 USQ Formula SAE race car therefore relies on this theoretical analysis to produce 'best estimates' for pipe diameters and lengths, which are theoretically tuned for maximum effect. These equations will be presented section 9.2.2.2, where they will be used to calculate appropriate pipe lengths and diameters for tuned intake and exhaust runners. Unfortunately, testing to determine the accuracy or effectiveness of these initial dimensions can only take place once the systems are mounted on the engine in-vehicle, where on-track testing may provide some indicators of performance enhancement.

Wave tuning of an internal combustion engine is considerably complex, with an assortment of factors affecting wave behaviour and hence engine performance. Atherton (1996) suggests a preliminary list of these factors may include such things as: valve timing, cam profiles, piston speed, pipe lengths, valve discharge coefficients and cylinder pressures, with a full analysis obviously incurring a much more extensive list. The complex nature of this tuning, compared with the analysis and testing resources available, necessitates a more basic analysis of wave tuning phenomena. In this section, an introduction to wave tuning terminology and phenomena is presented, to provide an understanding of the fundamental aspects of wave tuning techniques.

8.4.5.1 Types of Waves Generated in Internal Combustion Engines

Pressure waves are formed in IC engine systems due to the unsteady nature of flow through the induction and exhaust systems. It has been determined, through laboratory analysis, that these waves are finite-amplitude in nature, and are energy-charged with extremely high pressure ratios. These very high intensity finite-amplitude waves are markedly different from acoustic waves, which result in the general sounds we hear day to day.

In internal combustion applications, finite-amplitude waves take two forms: compression waves and expansion waves (Figure 8.9).

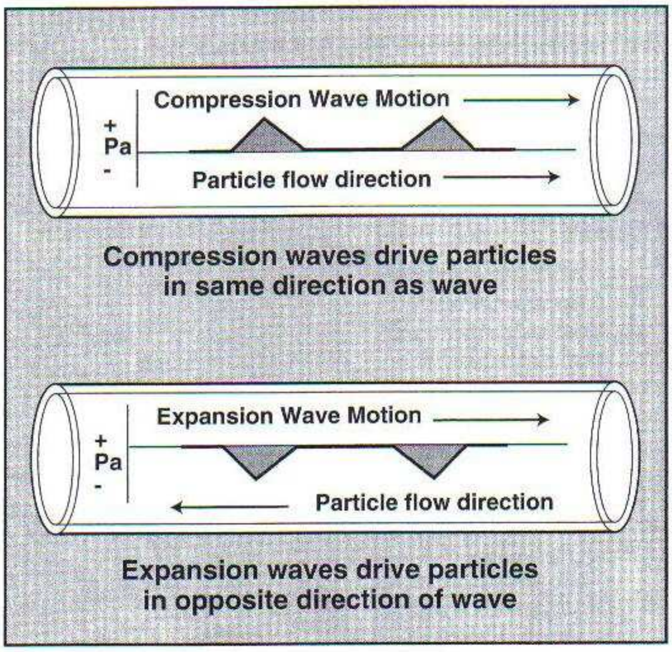


Figure 8.9: The nature of compression and expansion waves in internal combustion engines

(Source: Atherton 1996)

Compression waves create a positive pressure displacement, and cause motion of particles in the same direction as the moving wave.

Expansion waves constitute a marked drop in ambient pressure, with the resulting motion of particles occurring in the opposite direction to that of the wave motion.

Section 8.4.5.2 shows how the effects of both compression and expansion waves can be used to significant advantage in engine performance applications.

8.4.5.2 Effects of Open- and Closed-Ended Pipes

Movement of these pressure waves results in the formation of reflected waves, which also flow through the system. The nature of these reflected waves is dependent on the conditions within the system, and most notably, whether the initial wave encounters an open or closed end within the surrounding pipe.

Various regions within induction and exhaust systems can be modelled as lengths of pipe with either open or closed ends. For example, charge flowing through an intake runner toward the inlet port will encounter a closed end when the inlet valve is closed. Similarly, on the exhaust side, gases exiting the combustion chamber and travelling along a primary length to a common collector can be modelled as flow through a pipe approaching an open end.

Some interesting phenomena arise when flow approaches open and closed ends within pipes. These phenomena are discussed in the following two cases.
(Reference to Figures 8.10 and 8.11 is recommended to complement the ideas presented in these cases).

- **Case 1**

  When a compression wave encounters a closure at the end of a pipe, a reflected wave is sent back, in the form of another compression wave, which travels back in the opposite direction to the initial wave. As explained previously, the nature of these compression waves is such that their effect on particle motion is to move particles in the same direction as that of the pressure waves.

This reflected compression wave then travels toward an opening at the opposite end of the pipe. When this positive compression wave contacts the sudden opening, the result is another reflected wave, sent back toward the closed end. However, this time the reflected wave is an expansion wave. Although this expansion wave is travelling toward the closed end, the resulting particle motion is in the direction of the open end, due to the nature of expansion waves.

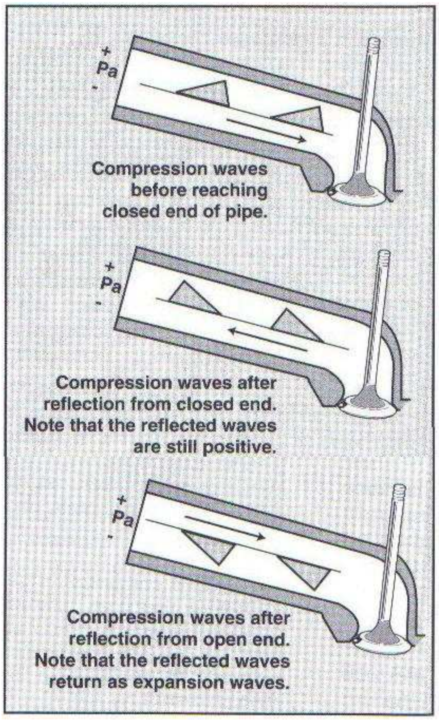


Figure 8.10: Effect of closed and open pipe ends on behaviour of compression waves

(Source: Atherton 1996)

The net result for particle movement is an initial move toward the closed pipe end, followed by a dual movement toward the open end. This result has significant implications for exhaust system wave tuning, and enables the phenomenon known as 'exhaust wave scavenging'. Compression waves travelling through primary exhaust headers towards open collectors will generate a secondary force that effects the further removal of exhaust gases, secondary to the initial expulsion occurring during exhaust valve opening.

This additional assistance in removal of combustion gases, or wave scavenging, aims to remove as much of the remaining gases as possible. If timed correctly, this action occurs as the piston is beginning its intake stroke, and thus has two desirable benefits: it provides increased volume to be filled by fresh intake charge, and the 'sucking out' of additional exhaust gas also draws in a greater amount of intake charge. This technique of wave scavenging is a very desirable action, and the exhaust system design will incorporate tuning of exhaust pipe lengths in accordance with maximising wave scavenging potential.

● **Case 2**

This case looks again at a wave travelling first towards a closed pipe end, followed by travel in the direction of an open ended pipe, however this time the initial wave is a negative expansion wave.

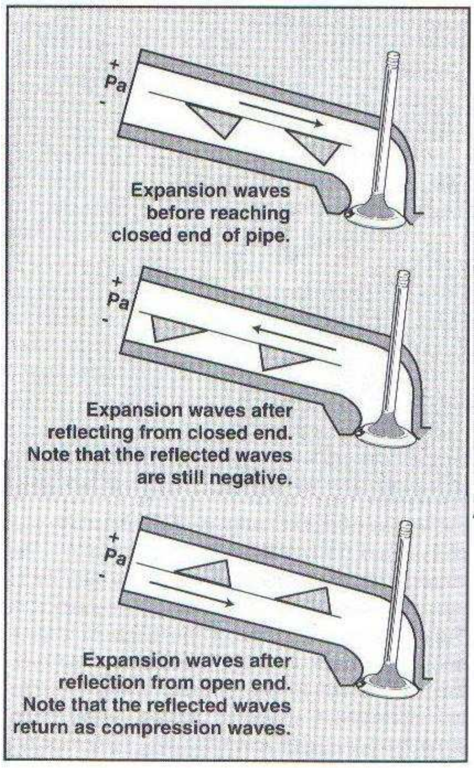


Figure 8.11: Behaviour of expansion wave travelling toward closed and open pipe ends

(Source: Atherton 1996)

As this expansion wave contacts the closed pipe end, a reflected wave is sent back through the passage, as a similar expansion wave. Contact with the open end then transforms this negative wave into a positive compression wave, which travels back toward the closed pipe end (see Figure 8.11).

This phenomenon has the ability to significantly improve induction potential. The resulting particle motion is an initial movement toward the open end of the pipe, followed by a double movement toward the closed end, similar in principle to Case 1. Imagining the open end of the pipe to be the point where intake runners extend from the plenum, and the closed end as being similar to a closed intake valve at the opposite end of the runner, expansion waves travelling toward the intake valve will produce a 'charge ramming' effect. Tuning to take advantage of the appropriate wave pulse then has great potential to significantly induce a greater volume of charge into the combustion chamber, with a corresponding increase in power production.

8.4.5.3 Timing Considerations

The crucial factor underlying wave tuning techniques is timing. In order to take full advantage of wave ramming and scavenging effects, waves must be timed to arrive at the appropriate point in the system at the time most likely for these waves to impart maximum effect.

Waves travelling through the system take time to move from one point to another. The speed at which these waves travel is variable, and depends on engine speed. Therefore, the time taken for a wave to travel between imaginary points A and B in the system is reduced or increased as engine speed climbs higher or lower. Also, for constant engine speeds, pipe length affects wave travel time, with shorter pipes incurring less time for travel than longer pipes.

Engine behaviour is affected by many factors, and so every engine requires individual wave tuning treatment. Engines are generally tuned for a specific speed, or a specific speed range. With a known tuning speed, the velocity with which the gases flow through the system can be approximated, and hence the time taken for waves to travel finite distances is also known. Thus, determining appropriate pipe lengths to take maximum advantage of these operating conditions is possible.

As previously mentioned, many empirical formulae exist for predicting tuned lengths, based on desired engine tuning speeds. These formulae will be presented in the forthcoming sections detailing primary and secondary header design.

8.5 Extractor Configurations

Some common and well-developed extractor configurations exist for four-cylinder engines. These two common arrangements are:

- 4-into-1
- 4-into-2-into-1

8.5.1 4-into-1 Configuration

Four-into-one extractor systems consist of four primary pipes collecting combustion gases from each of the four exhaust ports, connected at a common collector, as illustrated in Figure 8.12. Following this collector is a single pipe transporting the flow through the muffler and expelling the gases to the atmosphere.

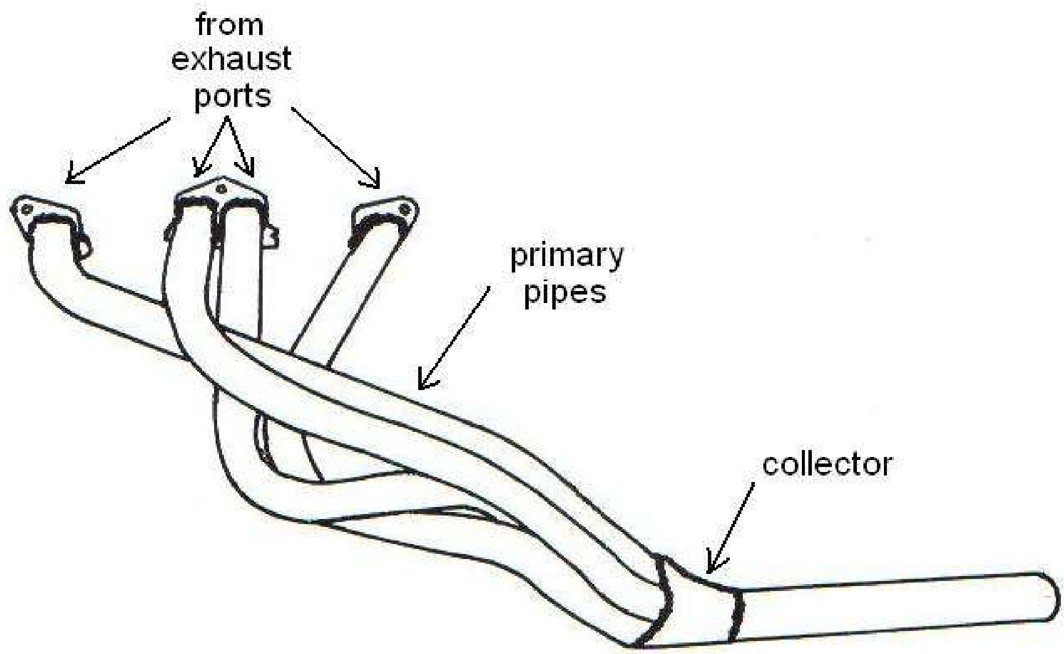


Figure 8.12: 4-into-1 extractor configuration

(Source: Bell 1997)

4-1 arrangements have been commonly used in performance motorbike applications for some time. These systems utilise a single collector, compared to 4-2-1 configurations which make use of three collectors.

8.5.2 4-into-2-into-1 Configuration

Four-cylinder engines using primary and secondary exhaust pipes form a 4-2-1 arrangement. Four primary pipes are again implemented, forming two pairs, combined at two collectors (Figure 8.13). These two secondary pipes then flow into a single exhaust pipe, again through a common collector.

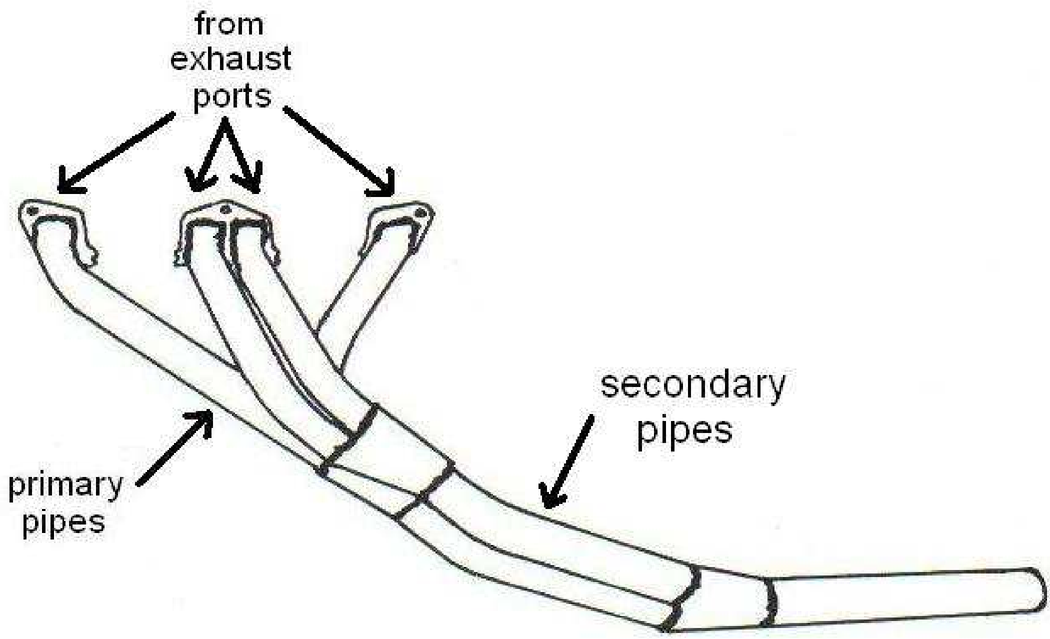


Figure 8.13: 4-into-2-into-1 extractor configuration

(Source: Bell 1997)

8.5.3 Comparison of Configurations

Determining which of these two configurations provides better performance involves a number of factors. Each of the two arrangements has its own advantages and disadvantages, namely: 4-1 induces less losses due to pipe entrance loss coefficients by virtue of single collector versus three collectors; 4-2-1 provides less restriction to flow through additional flow area (*two* secondary pipes).

8.5.3.1 Losses Due to Collectors

Firstly, the collectors required to group multiple pipes into single pipes induce losses due to changes in flow area, variations in entrance regions and joining methods (Fox & McDonald 2003). Poor collector-to-pipe joining methods result in quite significant reductions in flow (Bell 1997). Therefore, 4-1 systems are subject to only a single loss through this connection. Conversely, 4-2-1 systems can suffer multiple losses through their three collectors. Until recently, collector joining methods produced relatively poor results, meaning that using fewer such joiners in an exhaust system culminated in less flow losses overall (Bell 1997). For this reason, 4-1 configurations have been the preferred choice for performance motorbike manufacturers until fairly recent times (O'Neill AM 2005, pers. comm., 18 April).

However, techniques to achieve excellent quality joins are now available, and result in effectively zero flow losses. One common method to achieve free-flow joiners is to cut and weld 180° mandrel bends. It was shown in section 8.4.3.3 *Pipe (Press) Bending* that mandrel bends offer excellent flow characteristics. Therefore, using a mandrel bend and producing a neat, smooth weld will produce a joiner with excellent flow characteristics (Figure 8.14).

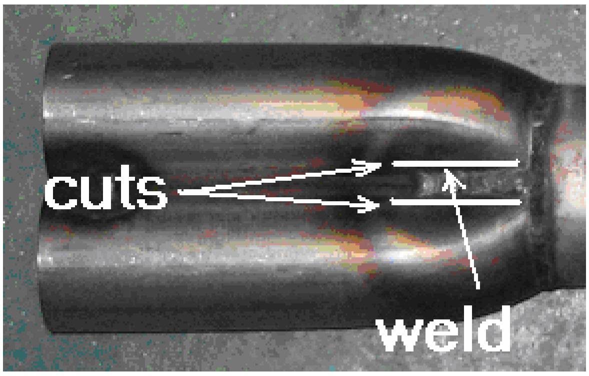


Figure 8.14: Free-flow joiner created by cutting and welding 180° mandrel bend

8.5.3.2 Flow Area

Minimising restriction to flow has already been stated as a critical objective in extractor design, with enlargement of exhaust pipes (increase in flow area) a sound method of reducing restriction. In a 4-2-1 system, the four primary pipes feed as pairs into two secondary pipes; hence, there are just two primary pipes flowing into a single secondary pipe, compared with four pipes flowing into one in a 4-1 arrangement. This additional flow area reduces restriction and therefore provides greater potential for increased power.

8.5.3.3 Wave Tuning Characteristics

Wave tuning characteristics for each of these configurations are also quite different (refer to section 8.4.5 for wave tuning criteria). Wave pulses travel through the primary pipes in each configuration. As they meet the collector(s), reflected waves are sent back through the pipes. Obviously, in a 4-2-1 configuration, there are two collectors at the end of the primaries, compared with one in the 4-1 system. Hence, quite different wave characteristics are generated in each system.

In a 4-1 system, reflected waves are generated at a single point. Motion of these waves back through the system will accordingly affect each of the four primary pipes simultaneously. In a 4-2-1 configuration, reflected waves generated at the first pair of collectors will travel back through two pipes at a time only. Following this, flow through the secondaries will then encounter another collector, again producing reflected pressure waves.

8.5.3.4 Companion Cylinder Grouping

Companion cylinder grouping was explained in section 5.6.2. Here, it was shown how grouping of companion cylinders can provide significant benefits in the pursuit of performance. Another advantage of the 4-2-1 configuration is that companion cylinder grouping is easily facilitated. By virtue of the connection between the primaries and secondaries as pairs, it is easy to group each of the two appropriate pairs, according to firing order, aiding in reduction of charge contamination and assisting exhaust gas expulsion.

Generation of reflected waves at both the end of the primary and secondary pipes also lends itself to tuning for different rpm ranges. The first set of reflected waves can be tuned for a different speed to that of the second wave set, resulting in an exhaust system that has a broader tuned range. This concept is explored further in the next section, covering design of the 2005 exhaust manifold.

8.6 Conclusion

The fundamental theory governing the operation of exhaust systems has been presented and explained in this chapter. The methods that can be utilised to take advantage of certain exhaust system behaviour have formed a major focus, as these methods can be used to greatly improve engine performance. Chapter 9 uses the theory presented here to develop the design for the 2005 exhaust system.

Chapter 9

2005 Exhaust System Design

9.1 Introduction

The main principles governing custom header design have been discussed, with the aim to provide a basic understanding of the major influences affecting exhaust system performance. Another important intention has been to provide an indication of the fundamental principles governing the development of the exhaust system design for the 2005 USQ Formula SAE race car, and to provide justification for decisions regarding this design. Development of the design now follows.

9.2 Exhaust System Design

9.2.1 System Type

Extractors have been shown to provide substantial performance benefits over standard exhaust systems. A set of extractors can be purchased 'off-the-shelf' from custom exhaust shops, with a range of pipe diameters, lengths and configurations available to suit a variety of engines. However, the advantage of designing and manufacturing a set of custom, tuned-length headers, is that it allows a system to be developed that specifically suits the particular engine, its operating conditions and application.

The Yamaha YZF600 used in USQ's 2005 car requires power and torque through the mid-range, rather than at the high rpm for which the engine is designed in standard bike trim. Therefore, a set of custom headers will be designed to increase low- to mid-range power and torque, by tuning for speeds in these ranges.

The basic design intentions comprise:

- 4-2-1 configuration
- tuned length primaries
- tuned length secondaries
- free-flow joiners

Development of this design now follows, in accordance with the areas outlined above.

9.2.2 Configuration

As per the YZF600's firing order of 1-2-4-3, companion cylinders are:

- 1 and 4, and
- 2 and 3

The 4-2-1 configuration will accommodate these pairings, with primaries 1 and 4 grouped into a single secondary pipe, and primaries 2 and 3 joined at a collector to form another secondary pipe.

This grouping allows the exhaust gas expulsion sequence to switch between primary pairs; that is, cylinder 1 discharges to the first primary pair, cylinder 2 then expels to the second pair, cylinder 4 discharges again to the first branch, finishing with 3 discharging again to the second pair (Heisler 1995). This alternating expulsion sequence minimises interference, improves exhaust flow and, in combination with tuned, equal-length pipes, promotes equal wave scavenging effects.

9.2.2.1 Extractor Pipes

It is estimated that in competition, USQ's Formula SAE car will run most frequently in the range of 4 000 - 8 000 rpm, based on the final sprocket and drive shaft design (O'Neill AM & Harber M 2005, pers. comm., 12 July). Therefore, excellent power and torque characteristics are needed in this range to provide the vehicle with optimum speed and acceleration ability.

Tuned lengths for both the primary and secondary pipes are utilised, taking advantage of wave phenomena. A feature of the exhaust system will be to broaden the engine's power band by tuning the primary and secondary pipes to take effect of wave characteristics at different speeds. This 'dual tuning' should theoretically provide benefits in terms of torque and power abilities at both higher and lower engine speeds.

● Overall Header Length

Empirical formulae exist for determining the overall length of extractor pipes. For a 4-2-1 configuration, this length is defined as the distance from the exhaust ports to the final collector, just prior to the tailpiece (see Figure 9.1). In this case, therefore, overall header length includes both the primary and secondary pipes.

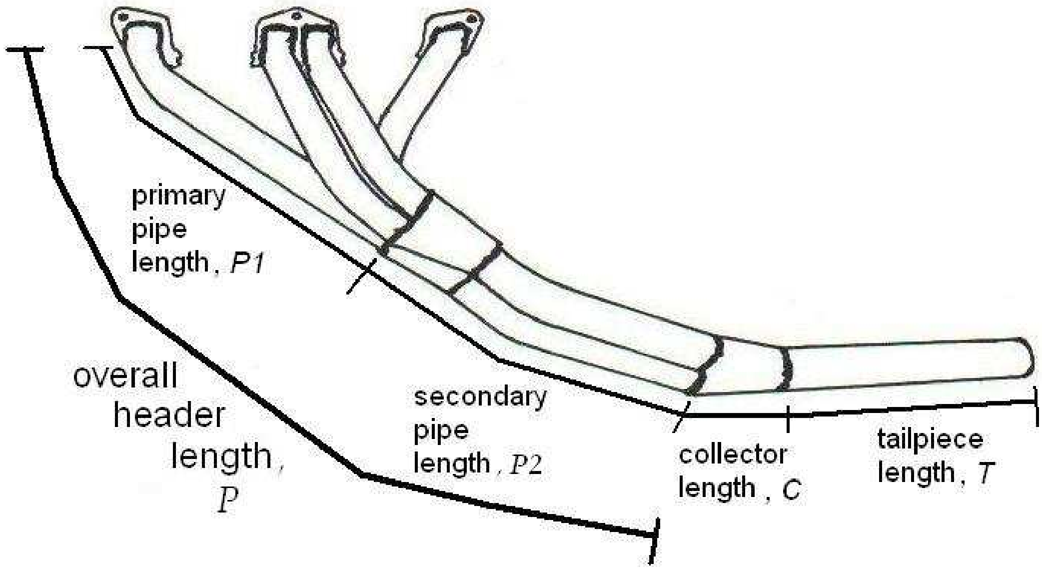


Figure 9.1: Terminology for pipe lengths

(Source: Bell 1997)

In general terms, shorter pipes have empirically been found to produce better high range torque, with longer pipes generally resulting in improved mid-range power and torque. Similarly, smaller pipe diameters consistently produce better low-to-mid-range characteristics in testing than do larger diameters (Bell 1997). A governing factor concerning pipe length is the overall space available to accommodate the headers. It is very common therefore to see, particularly on race cars, extractor pipes that contain numerous curves, twists and bends, in order to achieve extended pipe lengths in the space available. Of course, these bends must be of excellent quality, with very tight bend radii avoided, in order to minimise head loss (refer to section 8.4.3.3); however, even with the inclusion of these numerous bends, achieving the long lengths desired is usually of much greater benefit than compromising with shorter pipe lengths.

Overall header length is determined by consideration of a number of variables, and depending on the source of the empirical equation, includes:

- desired tuned speed
- number of degrees exhaust valve opens before BDC
- speed of combustion gases exiting exhaust port

▪ Calculation of Overall Header Length, *P*

Edgar (2001) specifies an empirical formula for determining overall header pipe length, *P*, in inches, as given in Equation 9.1:

$$P = \frac{850 \times ED}{rpm} - 3 \tag{9.1}$$

where P is the total header length (in inches), as defined in Figure 9.1

ED is 180° plus the number of degrees the exhaust valve opens before BDC

rpm is the desired tuned speed of the engine

▪ Parameters

Data from Yamaha indicates that for a 1994 YZF600, the exhaust valve opens around 70° before Bottom Dead Centre (BDC). Also, the specified tuning range has been determined as 4 000 – 8 000 rpm. Calculation of the overall length can thus be performed; in this case, a minimum and maximum length will result, using the upper and lower tuned rpm limits, respectively.

● Lower Limit: Tuned speed = 8 000 rpm

$$P_{\min} = \frac{850 \times ED}{rpm} - 3$$

$$= \frac{850 \times (180 + 70)}{8000} - 3$$

$$= 23.5 \text{ inches} = 598 \text{ mm}$$

● **Upper Limit: Tuned speed = 4 000 rpm**

$$P_{\max} = \frac{850 \times ED}{rpm} - 3$$

$$= \frac{850 \times (180 + 70)}{4000} - 3$$

$$= 50.0 \text{ inches} = 1273 \text{ mm}$$

A range for total header length has now been determined. An educated estimate must now be made to determine a suitable overall length for the exhaust system.

Considering that it is low-to-mid range torque and power that is desired to be improved, and that motorbikes typically generate peak power and torque at high revs, tuning for improved performance at lower speeds is required. Also, empirically it has been shown that long pipes produce better torque figures in the lower ranges. Therefore, the decision is made to use the longest pipe length calculated, and the overall header pipe length is thus:

$$\boldsymbol{P} = \mathbf{1\ 275\ mm}$$

This overall length must now be distributed appropriately between the primary and secondary pipes.

● **Primary Pipes**

Referring again to the 'dual tuned' exhaust design, the desired tuned speeds must be specified. It is sensible to tune one set of pipes for an expected average speed, while tuning the second set for revs in the low range. Considering the vehicle will run at between 4 000 and 8 000 rpm, an average tuned speed will be around 6 000 rpm, while a low tuned speed shall be 4 000 rpm.

As the YZF600 produces peak torque and power in the high rev ranges (at 9 500 rpm and 10 500 rpm, respectively), superior power and torque are generated at these higher speeds. Therefore, to take further advantage of these characteristics, choice of an 'average' tuned speed will be increased to around 6 500 rpm. It is feasible to tune the primaries for this slightly higher 'average' speed, as this value still remains in the expected operating range of 4 000 – 8 000 rpm.

Reflected waves will first be generated at the end of the primary pipes, before the flow reaches the secondary collectors, generating the second set of reflected pressure waves. Therefore, the primary pipes will be tuned for the average expected rpm, as this tuning will provide benefit through a broader range of speeds. Thus, the secondary pipes will be tuned to optimise low rpm reflected pressure waves, at a speed of 4 000 rpm.

Primary pipe length is then determined by proportions, according to Equation 9.2:

- **Primary Pipe Length, *P1***

$$P1 = \frac{rpm_{P1} - rpm_{P2}}{rpm_{P2}} \times P \tag{9.2}$$

where $P1$ is the primary pipe length (in inches)

rpm_{P1} is the primary pipe tuned rpm

rpm_{P2} is the secondary pipe tuned rpm

P is the total header length (in inches

The length of the primary pipes is then:

$$P1 = \frac{(6500 - 4000)}{4000} \times 50$$

$$= 31.3 \text{ inches} = 795 \text{ mm}$$

These primary pipes are designed to utilise maximum wave tuning effects at a speed of 6500 rpm.

● **Primary Pipe Diameter, *D1***

Bell (1997) also offers empirically derived equations for header pipe diameters. The primary pipe internal diameter, *D1*, in inches, is expressed by Equation 9.3:

$$D1 = \sqrt{\frac{cc}{(P+3)\times 25}} \times 2.1 \qquad (9.3)$$

where *D1* is the primary pipe diameter (in inches)

cc is the swept volume per cylinder (in cc)
i.e. (engine capacity) / (no. of cylinders)

P is the total header length (in inches)

The primary pipe diameter is then:

$$D1 = \sqrt{\frac{150}{(50+3)\times 25}} \times 2.1$$

$$= 0.707 \text{ inches} = 17.9 \text{ mm}$$

The primary pipes for the 2005 exhaust manifold are shown below:

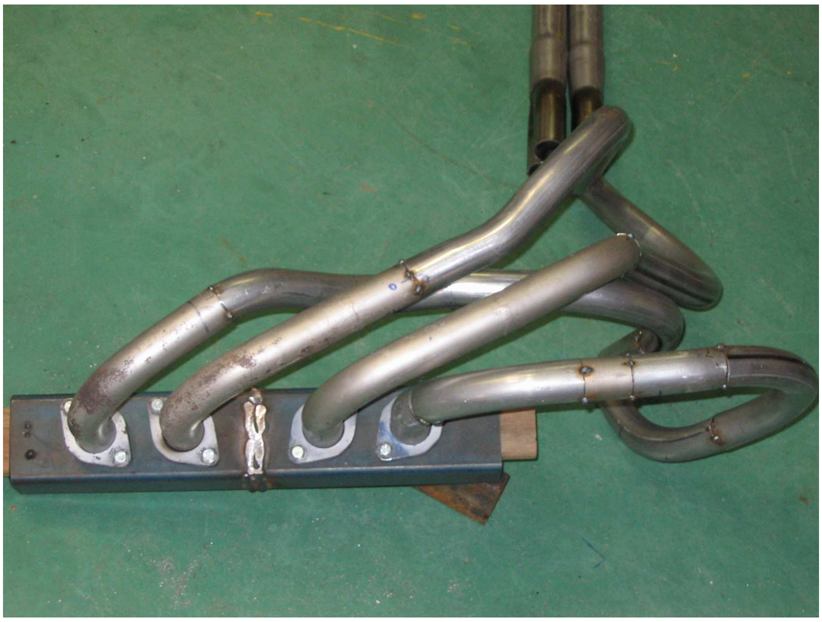


Figure 9.2: Primary pipe configuration for 2005 exhaust system

- **Secondary Pipes**

The secondary pipes are designed to be tuned at 4 000 rpm. Determination of the secondary pipe length is similar to that for the primaries.

- **Secondary Pipe Length, *P2***

$$P2 = \frac{rpm_{P1} - rpm_{P2}}{rpm_{P1}} \times P$$

where $P2$ is the secondary pipe length (in inches)

rpm_{P1} is the primary pipe tuned rpm

rpm_{P2} is the secondary pipe tuned rpm

P is the total header length (in inches)

The length of the secondary pipes is then:

$$P2 = \frac{(6500 - 4000)}{6500} \times 50$$

$$= 19.2 \text{ inches} = 488 \text{ mm}$$

● **Secondary Pipe Diameter, *D2***

Bell (1997) provides a formula for calculating the secondary pipe diameter, *D2*, in inches, as given by Equation 9.4:

$$D2 = \sqrt{\left(D1^2 \times 2\right)} \times 0.93 \tag{9.4}$$

where *D2* is the secondary pipe diameter (in inches)

D1 is the primary pipe diameter (in inches)

The secondary pipe diameter is then:

$$D2 = \sqrt{\left(0.707^2 \times 2\right)} \times 0.93$$

$$= 0.929 \text{ inches} = 23.6 \text{ mm}$$

The following figure shows the secondary pipes, together with the first set of collectors, which will facilitate joining of the primaries to the secondaries.

Figure 9.3: Secondary pipes for 2005 exhaust system

The primary and secondary pipe lengths and diameters have now been determined. For ease of manufacture, these dimensions are rounded to the nearest whole inch or nearest 10 inches in the case of pipe lengths, and then to equivalent whole metric measurements. The dimensions of the primary and secondary pipes are then:

- **Primary Pipe Length, *P1*:** 760 mm (30 inches)
- **Primary Pipe Diameter, *D1*:** 18 mm (ID)
- **Secondary Pipe Length, *P2*:** 510 mm (20 inches)
- **Secondary Pipe Diameter, *D2*:** 24 mm (ID)
- **Total Header Length, *P*:** 1 270 mm

- **Tailpiece**

As defined in Figure 9.1, the tailpiece is a section of single exhaust pipe, fed by the two secondary pipes, and flowing into the muffler. Figure 9.4 shows the connection of the two primary pipes into the single tailpiece. Similar to the four primaries and two secondary pipes, the tailpiece may be referred to as the single, tertiary exhaust pipe. The main objective of the tailpiece is to facilitate merging of all exhaust flows into a single volume, while providing minimal restriction to flow. The tailpiece should be long enough to provide adequate momentum to the combined gases to flow through into the muffler, and large enough to minimise restriction.

Figure 9.4: Secondary pipes feeding into 'tertiary' tailpiece

Bell (1997) recommends tailpiece lengths in the order of 12 – 13 inches (305 – 330 mm), as determined by exhaust flow testing. It is also recommended to choose tailpiece diameters larger than secondary pipe diameters, to aid in maximising flow. Tailpiece diameters corresponding to various engine capacities are provided by Bell (1997), and for a 600 cc engine, a tertiary pipe diameter of 1 ¼ inches is recommended.

The above recommendations are the result of extensive testing, and dimensions in this range have proved, in practical applications, to produce optimum flow conditions (Bell 1997). Therefore, these recommendations will be adhered to for design of the tailpiece, and the chosen dimensions are therefore:

- **Tailpiece Length, *T*:** 330 mm

- **Tailpiece Diameter, D_T:** 30 mm (ID)

● **Collectors**

As discussed in section 8.5.3.1, collectors induce losses in exhaust systems due to changes in section, welding imperfections and misaligned joins. These losses, however, can be significantly minimised depending on the joining method employed.

Mandrel bends have been shown to offer excellent flow characteristics (section 8.4.3.3). An excellent method for producing free-flowing exhaust pipe collectors is to cut and weld 180° mandrel bends, as shown in Figure 9.5. Taking the desired diameter mandrel bend and cutting parallel to the straight sections of pipe on each side, produces two halves which are then welded flush along the cuts. When quality welds are produced, this method results in joins with excellent flow characteristics, no leakage and virtually zero losses through the collectors. Therefore, configurations that use multiple collectors, such as a 4-2-1, benefit from the improved wave flow characteristics associated with this arrangement, with effectively no decrease in flow through pipe joins.

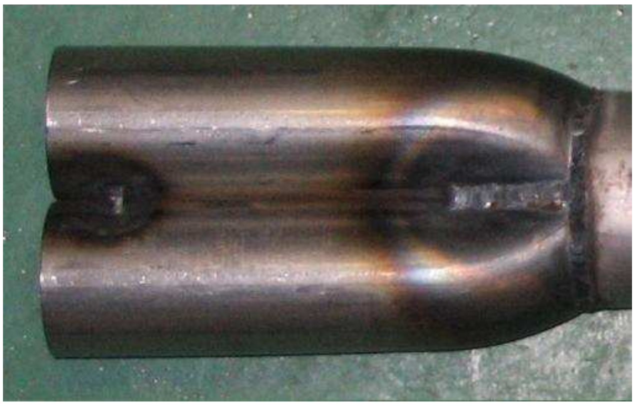


Figure 9.5: Free-flow collectors, created from 180° mandrel bends

9.2.2.2 Muffler Design Criteria

A muffler, or silencer, is a device designed to suppress to an acceptable level the noise created by the exhaust gases expelled from an engine’s cylinders (Heisler 1995). Silencer design influences not only the audibility of pressure waves pulsing from an exhaust system, but can also markedly affect power production, according to the amount of restriction offered by the flow path for combustion gases.

The noise generated within an exhaust system is the result of vibrating pressure waves, caused by the pulsations of gas flow from exhaust ports. These vibrating waves produce excessive noise, and thus require muffling, or silencing, through an appropriate device.

There are two types of noise generated within an exhaust system: noise arising from the periodic expulsion of combustion gases from the cylinders, called *fundamental noise*, and *secondary noise*, created by the actual movement of gases through the pipes (Heisler 1995). Fundamental noise is directly affected by the speed at which the engine is running, while secondary noise often results in rattling and vibrating of various end-plates and components throughout the exhaust system. Silencers therefore must effectively dissipate both types of noise.

More than just silencing devices, however, mufflers affect the amount of power an engine is able to produce (Edgar 2001). The most fundamental reason is that mufflers often constitute yet another restriction in the flow path of exhaust gases. Silencers, in general, also increase undesired back pressure (Smith PH & Morrison JC 1971). Therefore, in performance and racing applications, muffler design is of critical importance.

Muffler design for a Formula SAE race car must achieve:

- least restriction to flow
- adequate silencing, to comply with FSAE regulations
- maximum benefit to power

Generally, mufflers consist of a perforated tube, enclosed by a muffler body. This body contains packing, or absorption material, usually in the form of stainless steel wool and/or fibreglass (Edgar 2001). The basic operating principle is that exhaust gases flow into the tube and are then forced out through the many perforations, where these gases expand. Here, the gases are then absorbed by the packing, resulting in dramatic reductions in noise.

Muffler design has evolved over a relatively long period of time, resulting in some well-developed and well-tested designs being available today (Smith PH & Morrison JC 1971). There are broadly three muffler types available, and these are now discussed.

- **Baffle Type**

  Baffle type mufflers contain a series of obstructions, or baffles, within the muffler housing. These baffles are often in the form of a number of perforated tubes, one inside the other. Blank walls inside the tubes cause impedance to flow and force gases out through the perforations. Expansion occurs as the gases leave one tube, before flowing into another, resulting in noise reduction.

  Baffle type mufflers, in general, produce reasonable noise suppression, but are very restrictive to flow, by virtue of multiple impedances.

- **Reverse Flow Type**

  Reverse flow mufflers appeared in the 1960's (Edgar 2001) and are commonly referred to as 'turbo mufflers'. These are characterised by long, complicated flow paths through the muffler, forcing the flow to turn twice through 90° before exiting, as shown in Figure 9.6. Flow restriction is reduced, compared to baffle types, as baffles are not used. Instead, the longer flow path taken by combustion gases allows greater dissipation of pressure waves to the surrounding packing. However, the severe bends cause losses in flow.

Figure 9.6: Cut-away view showing complicated path of exhaust gases through reverse-flow muffler

(Source: Edgar 2001)

● **Straight-Through Type**

A straight-through, absorption type muffler is very common among performance applications as it offers minimal restriction to gas flow. The straight-through muffler transports combustion gases 'straight through' a single tube inside the muffler body, from entry point to exit. This single tube is often orientated diagonally within the muffler housing, as shown in Figure 9.7, creating a longer flow path, and hence improving noise suppression. Perforations around the tube allow dissipation of gases into the surrounding packing material. This single, direct path from muffler entry to exit creates very little restriction to flow, compared to baffle and reverse-flow type mufflers.

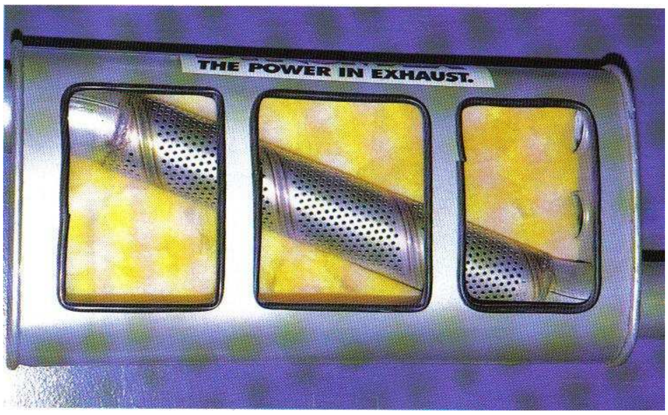


Figure 9.7: The very direct flow path taken by gases in a straight-through muffler design

(Source: Edgar 2001)

Minimal flow restriction is a very important muffler design criteria, with the presiding factors being the flow path and muffler pipe diameter. Flow bench testing provides an ideal method of comparison between muffler types, showing any severe restrictions to flow. Edgar (2001) presents flow bench results for the three basic muffler types.

In each case, flow through the various types is compared with flow through an equivalent length of straight pipe. Table 9.1 presents the results from this testing:

MUFFLER TYPE	% FLOW COMPARED WITH EQUIVALENT STRAIGHT PIPE
Baffle	38
Reverse-flow (turbo)	59
Straight-through	92

Table 9.1: Comparison of flow for various muffler types

This testing indicates significantly better flow through a straight-through type muffler compared with the other two designs. This result should be expected, due to the unimpeded flow path offered by this muffler type, minimising restriction. Not only is the straight-through type shown to offer far superior flow to both baffle and reverse types, but it also achieves an incredible 92% of the flow through an equivalent length of straight pipe.

● **Noise Considerations**

Mufflers must reduce engine noise to an acceptable level. An 'acceptable level' in Formula SAE terms means that the noise produced must not exceed the limit set by FSAE regulations. A sound level test will be performed at the event, with the regulations stating that the maximum permitted sound level is 110 dBA (*Formula SAE Rules* 2005). Muffler choice must therefore satisfy this noise level criterion.

● **Muffler Choice**

In the pursuit of optimised power, all induction and exhaust components must work to produce the best potential for increased power. Therefore, a straight-through muffler, offering minimal flow restriction, is chosen for its anticipated ability to offer superior performance compared with other tested designs.

Such a muffler may be manufactured 'in-house'; however, a very cost-effective alternative, both in terms of finances and time, is to purchase an 'off-the-shelf' version. Such mufflers can be purchased from motorbike shops for around $50 (Toowoomba Yamaha 2005, pers. comm., 9 June), and provide very acceptable performance and silencing characteristics. This silencer must be rated to not exceed 110 dBA, to comply with FSAE regulations.

9.3 Overall Exhaust System Design

The overall design for the 2005 USQ Formula SAE exhaust system has now been specified, including configuration, pipe lengths and diameters and muffler type.

Formula SAE regulations (2005) also stipulate conditions concerning positioning of the exhaust outlet, including:

- *exhaust must be routed so that the driver is not subjected to fumes at any speed considering the draft of the car*

- *exhaust outlet(s) must not extend more than 60 cm behind the centreline of the rear axle*

- *exhaust shall be no more than 60 cm above the ground*

In the 2005 USQ car, the exhaust system sweeps rearward of the cockpit, tracing the right-hand side of the car, with the exhaust outlet positioned at the rear. The outlet is mounted in compliance with height and length regulations.

A pictorial representation of this system is shown in Figure 9.8, illustrating critical dimensions:

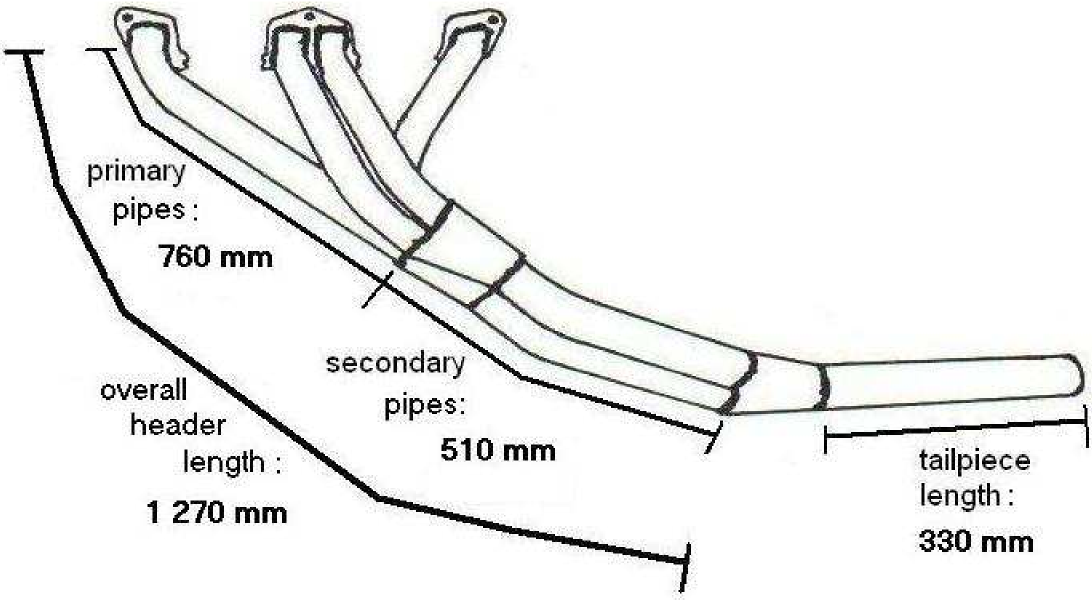


Figure 9.8: Overall exhaust system design, 2005 USQ Formula SAE race car
(Source: Bell 1997)

Figure 9.9 shows the actual exhaust manifold mid-construction. Note the orientation of the pipes, to achieve companion cylinder grouping and equal-lengths, as well as the gently swept bends, minimising flow restriction.

Figure 9.9: The 2005 exhaust system mid-construction, made from mild steel

The orientation of the exhaust system within the vehicle is illustrated by Figure 9.10, showing the position of the primary headers as they flow from the exhaust ports.

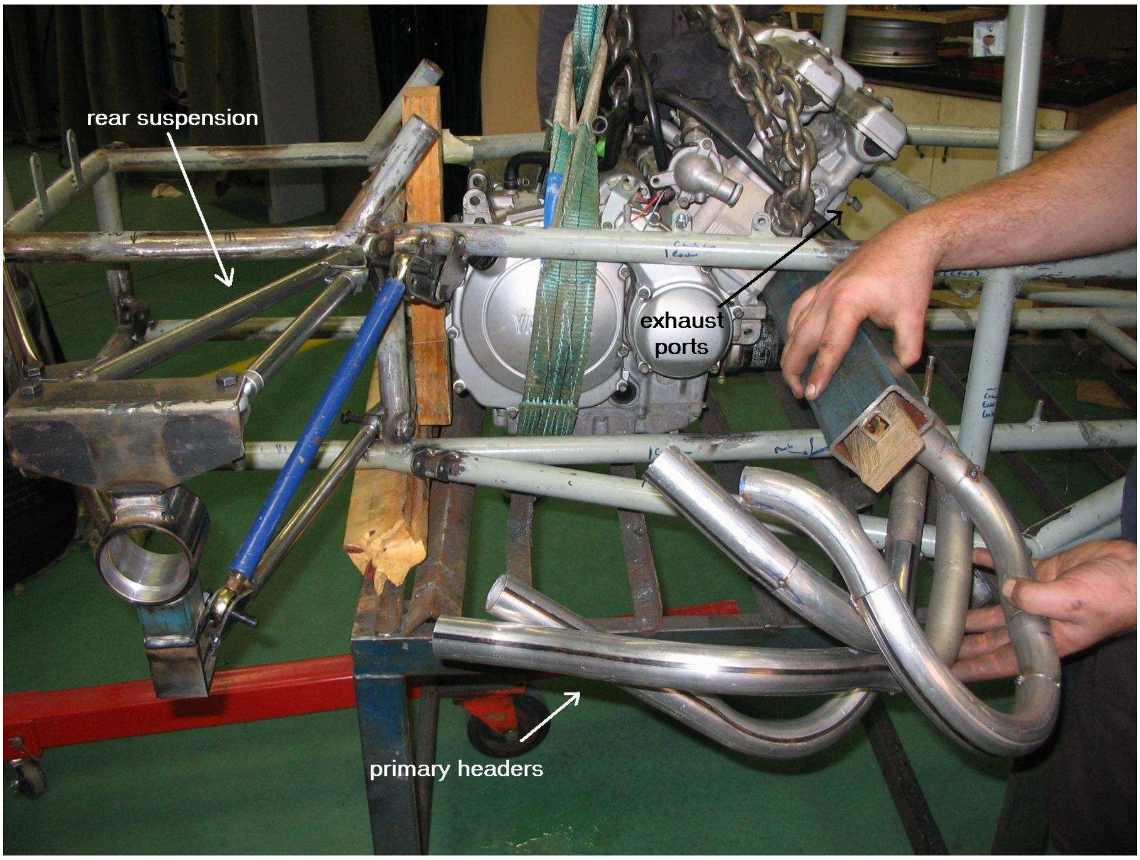


Figure 9.10: Orientation of exhaust manifold within vehicle

9.4 Conclusion

The exhaust system has been designed by combining fundamental engineering flow theory with empirical data. Empirical correlations have been heavily relied on in determining pipe lengths and diameters. It seems reasonable to take this design approach, as data obtained through implementation and testing of various systems shows general trends to indicate appropriate critical extractor dimensions.

The main point of concern is in ensuring the correctness of empirical data. Much data is obviously available through numerous sources, including books, websites and personal communications. Therefore, determining the value and credibility of such data in relation to its source must be seriously considered. In this case, multiple sources were consulted, with independent testing in each case producing similar empirical results. Therefore, such empirical data can be utilised as a design tool with relative confidence.

Exhaust system design involves many factors, as discussed in Chapter 8. Designing an optimum exhaust system for a particular application must therefore involve continued developmental work, including testing, modification and refinement. Flow bench testing can produce very useful results; however, the best indication of the contribution of exhaust system design to overall engine performance is generally determined through dynamometer testing. This allows a range of variables to be altered sequentially to determine optimum combinations of lengths, diameters and other parameters, and also takes into consideration other influences, such as engine speed, inlet configuration and general engine operating conditions.

Unfortunately, time constraints have meant that dynamometer testing of this exhaust system was not feasible; nor was it guaranteed that the dynamometer developed as part of this project would have been suitable for producing useful results (refer to Chapter 11).

However, the fundamental engineering principles applied in the design process, together with the use of suitable empirical data, should produce an exhaust system that performs basic expulsion duties both effectively and efficiently, and provides a significant contribution to increasing the power produced by the YZF600.

Chapter 10

Dynamometers

10.1 Introduction

A dynamometer represents a vital testing and analytical tool in the engine development process. Dynamometers play a key role in indicating directions for improvement and achieving optimised performance. This chapter presents various dynamometer types and their basic operations, and also looks in detail at the process involved in modifying an existing dynamometer for use in Formula SAE engine development at USQ.

10.2 What is a Dynamometer?

The term *dynamometer* covers a variety of devices which are used to determine the power produced by an engine. ‘Dynamometer’ literally means ‘to measure a moving force’. Dynos, as they are often called, measure the torque being produced by an engine at various speeds. Throttle control is performed remotely, usually in a ‘viewing’ room outside the dyno room, which also houses the necessary instrumentation. Through application of the appropriate formula, the power corresponding to this torque and speed can then be calculated. Thus, dynamometers do not measure power directly, but rather measure the force applied by the engine over a distance, or torque.

There are several different types of dynamometers, differing in their modes of operation. However, all measure speed and the torque produced by an engine in order to determine power. The next section looks in more detail at the various types available and the particular operating methods of some of the most common types.

10.3 Dynamometer Types

Dynamometers can be classified into two broad categories: engine-type and chassis-type. Within these two categories exists a number of further variations.

10.3.1 Engine Dynamometers

Engine dynamometers (Figure 10.1) record output via a direct connection between the engine and dyno. This connection, fitted directly to the flywheel or crankshaft pulley, therefore records data at the actual flywheel.

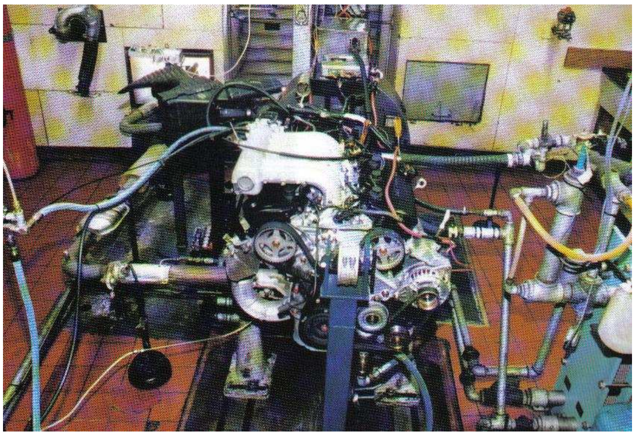


Figure 10.1: Typical engine dyno
(Source: Edgar 2001)

Engine dynamometers are most commonly used for major engine development and tuning (Bell 1997). This is especially the case for large automotive companies developing more powerful and efficient motors, as well as in racing applications for developmental work, as fine-tuning the engine for optimum power is facilitated.

10.3.2 Chassis Dynamometers

Chassis-type, or rolling road, dynamometers require the engine to remain in the vehicle being tested. This type offers convenience, as the vehicle is simply driven onto rollers, where the power produced at the wheels is measured (Figure 10.2).

Figure 10.2: Rolling road (chassis) dyno

(Source: Edgar 2001)

Rolling road dynos are very common in general performance and tuning shops (Bell 1997) as they offer the customer greater convenience, and initial outlay is also low. However, only minor tuning can be performed on these types, as they offer far less sensitivity than engine dynos.

10.3.3 Comparison of Engine- and Chassis-Type Dynamometers

The major point of difference between these two types is that engine dynos measure torque at the flywheel, whereas chassis dynos measure driven wheel torque. This is important, as depending on where in the system torque is measured, higher or lower figures will result. This is because the transmission of power from crankshaft to driven wheel inherently induces losses, through mechanical connections such as gears and shafts, etc. Therefore, an engine tested in-vehicle on a chassis dyno will produce lower torque and power figures than for the identical engine tested on an engine dyno. This is not to say that the engine loses power in moving from one dyno to another, but that the rolling road dyno accounts for losses through mechanical power transfer, by taking measurements at the wheel.

Chassis dynos therefore offer results that more realistically represent the power actually available at the wheel, in a road or track situation. Engine dynos, however, are used to optimise engine power output.

10.3.4 Brake Dynamometers

Brake dynamometers, also known as steady-state dynamometers, are so called because they operate by attempting to 'hold' the engine at a constant speed (rpm). A braking device, commonly driven by water, applies a force to the engine, attempting to slow it down, and the engine reacts by applying a force to continue the brake spinning at a constant speed. The torque being applied by the engine to maintain this speed is measured via a lever connected to the brake. The engine can maintain this speed up to a point, applying the necessary torque to achieve constant rotational speed. The torque produced by the engine at this point is recorded, along with its speed, and from this, a figure for power is determined.

10.3.5 Inertial Dynamometers

Inertial dynamometers involve an engine spinning a flywheel, with the angular speed and acceleration of the flywheel being monitored. The engine must apply more power to increase the flywheel's rotational speed and, together with the wheel's known moment of inertia, the amount of power being applied is able to be determined.

10.4 Validity of Testing Results

The results obtained from any dynamometer test must be treated with some caution, as there are many variables which may affect the readings. This section explores some of the factors commonly affecting dynamometer output readings.

10.4.1 Influence of Atmospheric Conditions

Atmospheric conditions can significantly affect engine performance. Therefore, results produced from the same dynamometer on different days, or even at different times on the same day, can vary markedly. Commonly, various correction factors are applied to the calculated power figures to account for variations in conditions at the time of testing. Variations in atmospheric conditions most significantly affecting dynamometer output are now discussed.

- **Air Pressure**

  The mechanism of drawing air (and fuel) into an engine's cylinders is by way of a pressure differential. For a naturally aspirated (NA) engine, the inlet charge is at atmospheric pressure (101.1 kPa). The downward motion of pistons on their intake strokes causes a vacuum to be created within the cylinders (any pressure lower than atmospheric is considered a vacuum) (Stockel, Stockel & Duffy 2001). Basic physics dictates that air at higher pressure will rush to regions of lower pressure to maintain equilibrium. Therefore, the motion of intake charge being drawn from the inlet manifold into the cylinders is courtesy of this pressure differential.

  The greater this differential, that is, the lower the pressure in the cylinders relative to atmospheric pressure, the more air must fill the vacuum to achieve equilibrium. Therefore, increasing inlet charge pressure creates a 'higher' vacuum, forcing more air into the cylinders. This is also the basic principle of operation governing supercharging and turbocharging systems (refer to section 4.3.6).

- **Air Temperature**

  Air temperature affects the density of inlet charge, and therefore influences the amount of power produced. Temperature and pressure share an inversely proportional relationship; therefore, an increase in temperature causes air particles in the inlet charge to expand, resulting in a decrease in pressure. This, of course, means that colder air contains more particles per volume than does hotter air. So, for air entering the inlet tract, a greater mass will be admitted at lower temperatures than at higher temperatures. Therefore, a greater volume of charge entering the cylinders results in more power being produced. Conversely, increasing the temperature of inlet charge produces a decrease in power.

Monitoring these two variables is therefore important, as differences in dynamometer outputs can sometimes be accounted for according to inlet air pressure and temperature variations.

Therefore, when conducting dynamometer testing for this project, equipment such as a barometer and thermometer shall be used to measure and record these variables, in order to monitor possible effects of atmospheric conditions on testing results.

10.4.2 Comparison of Dynamometer Output

It is not advisable to attempt a direct comparison of power readings between different dynos (Bell 1997) due to the intrinsic differences between each, and the variations in testing conditions. Applying appropriate correction factors to the results is intended to standardise calculated power figures to some degree; however, direct comparisons can still be problematic.

As such, one of the most critical factors in performing dyno testing is achieving consistent, repeatable results. If successive testing can produce consistent readings, then achieving a 'repeatable' dynamometer is very likely. Such a dyno will then allow for comparative testing, which can show the relative performance gains or losses from one engine configuration to another.

The methodology implemented to determine the repeatability of the dyno used for testing the YZF600 engine at USQ will therefore involve successive testing of the engine in ‘standard’ trim. This testing will be used to generate control-type figures, and to attempt to achieve consistent, repeatable results. This ‘standard’ trim involves the engine and related components being transferred essentially directly from the motorbike to the dyno. That is, the engine, exhaust and intake systems, complete with quad-carburettors feeding each cylinder, will be removed from the bike and connected to the dyno, and torque figures measured. This procedure should contribute to achieving genuine confidence and validity in results.

Once repeatability has been achieved, confidence can then be had in the dynamometer offering reliable results. Therefore, testing can commence to compare various set-ups, such as different intake manifold configurations, including various plenum volumes, carburettors and restrictor shapes, as discussed in Chapter 11.

10.5 Role of Dynamometer Testing in Engine Development

In the pursuit of optimised engine performance, there are many possible modifications that can be determined theoretically. Implementing these ‘appropriate’ modifications should then provide benefits in terms of increased power, torque and fuel economy. There are a range of well-developed and fairly reliable generalised theories on performance improvement methods (Bell 1997); however, engines do not operate independently of the many other components within a vehicle. That is, they form an integral part of an overall package, interacting specifically with the drive train, and their performance therefore is influenced by the behaviour of these other various components, in addition to their own intricacies in design and operating conditions.

Engine related components, namely the inlet and exhaust manifolds, contribute directly to engine performance, and with intelligent design can dramatically improve power, torque and efficiency. These components must work in unison to provide the best combination for overall performance improvement.

On-track testing obviously provides excellent indicators for how each component contributes to overall vehicle performance, and allows testing of various set-ups to determine optimum combinations. However, track testing is inherently expensive, often prohibitively so, in terms of finances, time and resources. An ideal alternative, then, is to employ dynamometer testing.

Dynamometers allow abundant amounts of testing to be performed at a much reduced cost than track testing. Testing apparatus can be conveniently contained entirely within a single room, and the cost of transporting equipment and personnel, as required for track testing, is avoided. The cost of consumables, such as fuel and tyres, is also significantly less compared with on-track activities.

Dynamometer testing allows various ancillary systems to be tested, enabling determination of the effect of each system individually; this method also facilitates successive testing, by interchanging various components to determine the performances of a range of configurations. In this way, dyno testing provides indications of areas for improvement, and allows continued development to eventually reach a point of overall optimisation.

10.6 Importance of Dynamometer Testing to the USQ Formula SAE Project

Formula SAE at USQ is a long-term project, with the intention of continuous development and improvement. Analysis tools which can be used to quantify various aspects of the car's performance are therefore likely to provide significant benefits in terms of determining causes and effects of performance gains and indicating directions for continued development.

As previously mentioned, the real test comes on the track, where the engine is able to interact with the car's characteristics overall, and respond to the dynamic nature of the interaction between the car and track itself. However, dynamometer testing provides a more efficient and cost-effective alternative to this testing procedure.

While a dynamometer is able to determine a relative figure for the power produced by an engine, the dyno's real benefits are in providing comparative measurements and indications of improvement. Testing an engine in its 'standard' form to acquire benchmark power figures provides a control against which any modifications can then be compared, hence showing clearly any gains made from modification to the engine and related systems.

Overall, dynamometers facilitate huge amounts of testing within relatively short periods of time, compared with on-track testing, and can therefore produce substantial results quite quickly. Dynamometers therefore form invaluable testing, diagnosis and analysis tools for engine performance optimisation, and are absolutely vital components in any engine development program.

Considering the long-term nature of the Formula SAE project at USQ, availability of a suitable dynamometer for engine development is vital if significant performance gains are to be achieved.

10.7 USQ Dynamometer Options

The USQ campus houses a number of existing dynamometer options. In determining which of these is the best candidate for modification into a suitable testing facility, there are some key criteria that must be fulfilled. Presented here is an explanation of the most vital objectives to be met by a dynamometer in this particular application.

10.7.1 Criteria

For the long-term development of the project, and for convenience, it was decided that any dynamometer available to the team be located at the USQ campus. The benefits of this decision are identified as:

- location/convenience
- availability/accessibility
- suitability for application
- longevity and permanency
- facilitation for comparisons between configurations
- opportunity for use as educational tool
- assistance in establishment of general engine development program at USQ

● **Suitability**

Suitability is the most important criteria for assessing available options. 'Suitability' refers to the testing equipment's capability of achieving the desired intentions and objectives. There is no value offered by a dyno that produces repeatable, reliable results, yet is catering for a different application to that intended. For example, a dyno may run unsteadily at low engine speeds, in an application where tuning for low revs is the main objective. Therefore, in all scenarios involving dynamometer choice, suitability for the intended application must constitute the most important criteria.

● **Location**

Having a dynamometer located on-campus means that testing facilities are available to the team generally at their convenience. Also, transportation of equipment, personnel and the engine or vehicle to be tested is kept to a minimum. In 2004, there was an option available to make use of a rolling-road chassis dyno from an automotive performance shop in Toowoomba. However, for the reasons expressed, the importance of having a dyno available for the team's use at their convenience is of critical importance.

Locating a dynamometer on-campus also has benefits for the longevity of the project. Providing the team with access to a dynamometer each year presents a major advantage in terms of improving engine performance, through facilitation of extensive testing.

● **Repeatability**

Another key factor relates to achieving repeatability, which encourages consistent results. Therefore, use of the same dynamometer for long-term testing is preferable, if not a necessity, and should enable comparisons to be drawn between testing of various configurations.

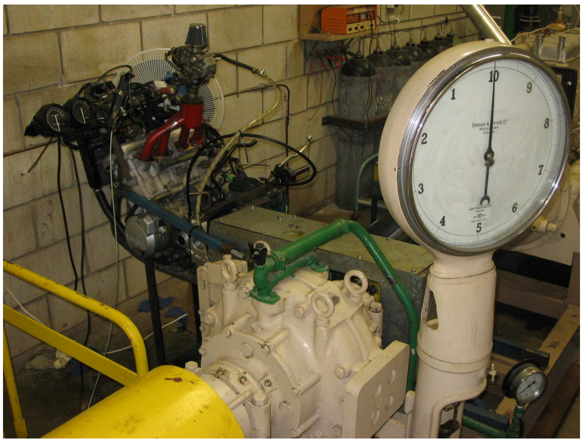
● **Educational Benefits**

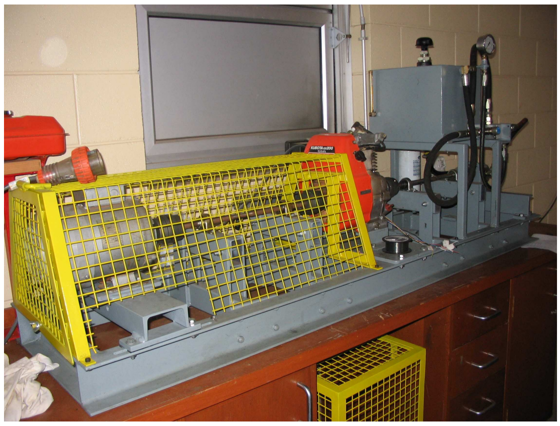
Achieving set-up of an engine testing facility that produces reliable and useful results not only benefits the Formula SAE project, but provides further educational opportunities at USQ. Engine testing could conceivably form part of the standard curriculum, with many general engineering principles able to be demonstrated through testing and operation of an engine. Establishment of an engine development program would also be feasible, perhaps aimed at post-graduate research. Areas such as thermodynamics, fluid mechanics, wave theory and many others would benefit from such a program.

10.7.2 Process for Determining a Suitable Dynamometer

Research was conducted to determine all viable options regarding dynamometer use. During this time, it quickly became evident that the process would not involve simply choosing a dyno, connecting the engine and producing the desired results; rather, the process would be somewhat more involved, and entail choosing the *most suitable* dynamometer and consequently carrying out all necessary modifications to produce a piece of equipment suitable for engine testing in this application. From this, a number of likely candidates arose:

- Heenan-Froude Engine Dynamometer (S Block) (Figure 10.3 (a))
- Tractor Engine Dynamometer (Agricultural Engineering Lab)
- David Buttsworth's Pump Engine Dynamometer (Hydraulics Lab) (Figure 10.3 (b))
- 'Design and Build Own' Engine Dynamometer





(a) **(b)**

Figure 10.3: Two of the available options, (a) Heenan-Froude Engine Dynamometer and (b) Pump Engine Dynamometer

Each option carried with it advantages and disadvantages which were taken into consideration in determining the most appropriate option to be pursued. The main issues concerned suitability, reliability, cost and available time. The particular advantages and disadvantages of each option are discussed in Table 10.1 (note the rating system employed, which is explained following this table).

Note: the option to simply purchase a dynamometer fulfilling the requirements as specified in section 4.6.1 was deemed financially unviable, at least within the time constraints of this project, and was not considered any further.

	DYNO OPTIONS			
CRITERIA	**Heenan-Froude Dyno**	**Tractor Dyno**	**Pump Dyno**	**'Design and Build' Dyno**
(Current) Availability	1	2	1	5
Location	1	3	1	1
Suitability	2	4	4	1
Modifications needed	2	2	4	1
Cost	2	2	5	4
(Anticipated) Reliability	3	3	2	2
Existing knowledge/technical skills	3	3	2	1
OVERALL RANKING	14	19	19	15

Table 10.1: Determining Suitability of Dynamometer Candidates for Modification

● **Rating System**

The rating system used employs a scale from 1- 5, as follows:

1 = Good 3 = Average 5 = Unsuitable

This means that, for example, if the dynamometer was housed in a convenient location, it would score a 1 for the ‘Location’ criteria. Whereas, if the cost to modify a particular dyno to render it suitable for the intended application was very high, it would rank a 5 for ‘Cost’. Likewise, the actual amount of modification and related effort required to bring the dyno to a suitable, useable state would rank 1 for minimal effort, and 5 if excessive modifications were required.

This means then, that the dynamometer with the lowest (numerical) overall ranking theoretically represents the most suitable option to be pursued.

10.7.3 Dynamometer Choice

This rating system indicates that the most likely option to produce satisfactory results is the Heenan-Froude engine dynamometer, currently existing in S Block.

Pursuing the ‘Design and Build’ option would allow very precise specification of objectives, and should accordingly result in a piece of testing equipment that produces the desired results, and offers a large initial technical and knowledge base, with advantages for ongoing maintenance. However, the cost associated with constructing such a device inhibits pursuing this option in this case. Another major disadvantage relates to the immediacy of the required equipment. A significant amount of time would likely need to be devoted to design and manufacture of an ‘own’ dyno; for the purposes of this project, a dynamometer is needed in the very short-term.

The rating system employed in section 10.7.2 considered the most crucial aspects of dynamometer choice, and ranked each objectively. This system was therefore a very useful tool in deciding to pursue modification of the Heenan-Froude dynamometer. In addition, further objective scrutiny of this existing dyno provided endorsement that there was good potential for achieving the desired testing objectives.

To reiterate, choice of the Heenan-Froude engine dynamometer (Figure 10.4) as the preferred option was based on:

- relative least cost in terms of modifications required
- ability to handle power produced by motorbike engine
- suitability to application (as designed for automobile engines)
- necessary instrumentation already in existence
- offer of technical support from Engineering technical officers
- location (on-campus)
- availability (not currently in use)

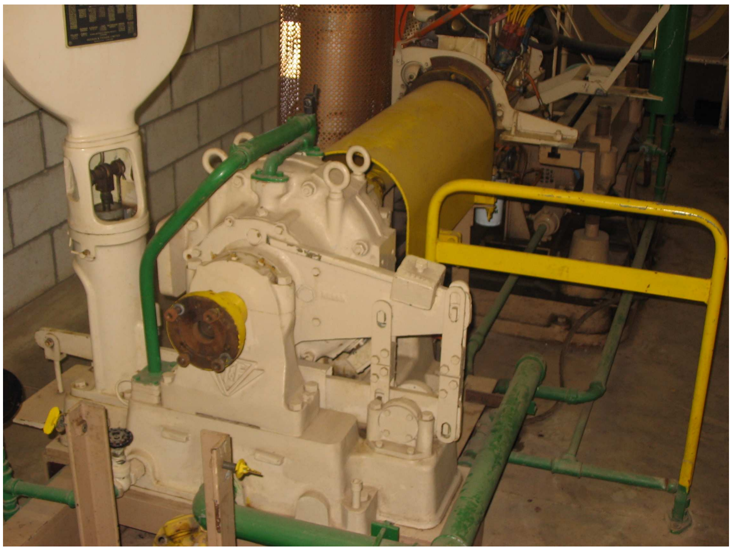


Figure 10.4: The Heenan-Froude Engine Dynamometer chosen for modification

10.8 Modification of an Existing Dynamometer for use in Formula SAE Engine Development

Determination of the most appropriate dynamometer option has been achieved. The modifications needed to accommodate the requirements specified for dynamometer use in this project must now be determined. First, specifications regarding the operation and intended applications of the Heenan-Froude dyno will be presented, and from this, conclusions will be drawn regarding the modifications necessary.

10.8.1 Heenan-Froude Engine Dynamometer Specifications

The dynamometer located in S Block (USQ) was manufactured by Heenan-Froude Ltd in 1971. It is an engine dynamometer with a water-applied brake. Intended for development of small to medium motor vehicle engines, it measures the torque applied by the engine via a lever, and registers a corresponding reading on a needle dial (Figure 10.5).

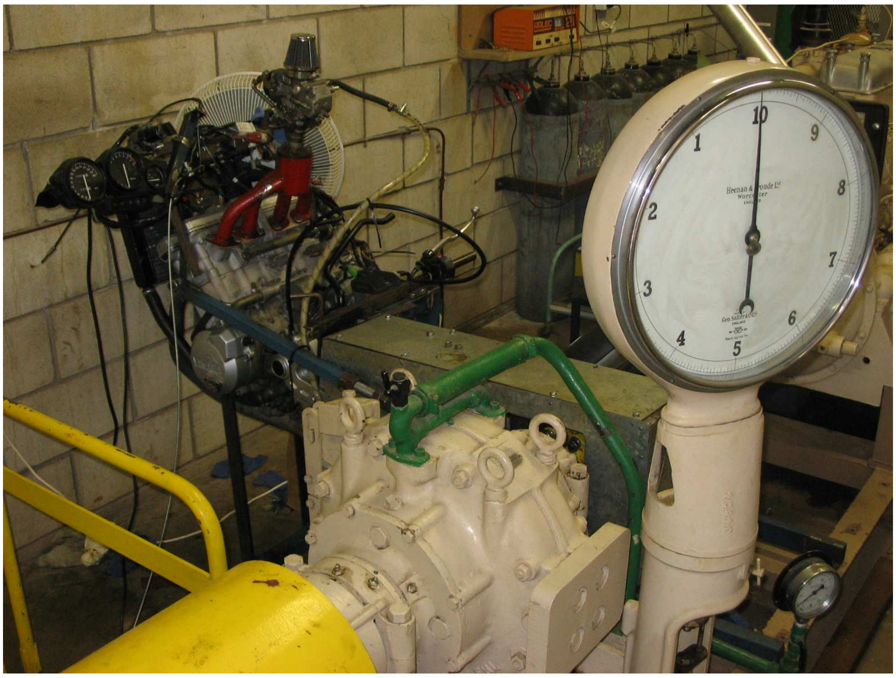


Figure 10.5: Dial gauge for registering torque application

The dyno was procured by USQ in 1971 as an educational tool, and formed part of a practice course for some time (Blokland A 2005, pers. comm., 21 June). During its use as part of this course, it ran a Holden petrol engine and a Leyland diesel engine.

The existing instrumentation is, of course, all analogue, consisting of a needle gauge for torque measurement, as mentioned, and a control panel, located outside the testing room, facilitating load control and speed monitoring.

The fundamental operation of the dynamometer involves running the engine up to a pre-determined, constant speed, and then applying load through the Load Control knob.

Load is increased until the engine can no longer maintain speed. The reading on the torque gauge at this point is recorded, along with the speed of the dyno, as recorded on the tachometer located on the control panel. From these recordings, the corresponding power produced by the engine under these conditions can be determined (refer to section 11.3.

An operator's manual for the Heenan-Froude water-brake dynamometer is located in Appendix E.

10.8.2 Modification Process

Some brief indications of the modifications necessary for testing the YZF600 engine on the Heenan-Froude were presented above. Collectively, the modification process included:

- modification of existing exhaust system
- design and manufacture of frame to support engine
- design and manufacture of chain guard-cover
- design and manufacture of sprocket for chain drive
- replacement of existing fuel system (for different fuel type)
- re-routing of fuel from existing engine to bike engine
- implementation of remote throttle control

Each of these objectives were required to be completed before the motorbike engine could be tested on the dynamometer. A more detailed look into the modification process for dyno set-up now follows.

10.8.2.1 Exhaust System

The existing exhaust system comprised single exhaust pipes feeding from each of the two diesel engines. Each of these pipes was connected to one of two wall-mounted outlets, piping combustion gases outside (Figure 10.6). The exhaust manifold and muffler from the motorbike were to be used during testing; thus a means of connecting the muffler end to one of the wall outlets was required.

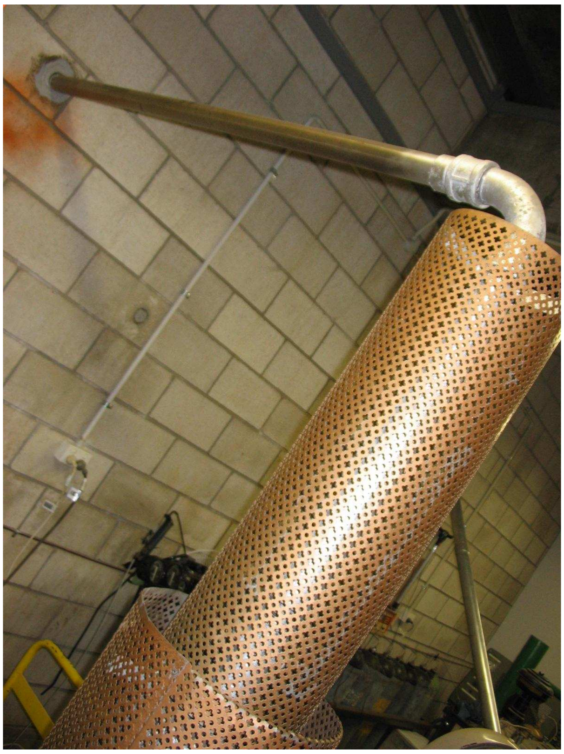


Figure 10.6: Original dynamometer exhaust system

Implementing this new connection necessitated removal of one of the existing exhaust pipes. Due to the era in which the pipes were installed, concerns were raised regarding the high probability that the pipe contained asbestos (Scott J 2005, pers. comm., 12 July). Therefore, no work could proceed until removal and testing of the pipe material, by professionals, was completed. Once the pipe was removed, however, the process of modification began. Ultimately, results from testing proved negative to asbestos content (Scott J 2005, pers. comm., 27 September).

A 2-metre length of exhaust pipe was purchased from *A1 Exhausts*, Toowoomba, costing $70. This pipe was formed in two halves, each with one swaged end: one half contained a 90° bend at the opposite end, while the other similarly featured a 45° bend. These angles allowed for correct positioning, given the orientation of the muffler and exhaust outlet at the wall. Forming the pipe in two halves with swaged ends allowed for relative rotation between the two, facilitating fitment and correct orientation.

A valid concern was the possibility of inducing severe restriction to flow through the addition of this pipe. Obviously, on the motorbike and in the FSAE vehicle, exhaust gases exit the muffler directly into atmospheric pressure. Forcing gases of combustion from a muffler into the confines of another pipe would conceivably cause. However, under the supervision of safety officers, the exhaust pipe beyond the muffler was removed and a dyno run performed. This run produced no variation in results to those achieved when the exhaust pipe was connected (refer to section 11.4.3), and it was hence concluded that the pipe offered no detectable restriction to flow.

10.8.2.2 Engine Support Frame

Connection of the engine to the dynamometer required manufacture of an appropriate support frame. It was decided that the most convenient orientation for the engine was to position it beside the dyno, rather than in-line with the dyno as per the existing orientation. This meant that the engine-to-dyno connection was not able to be via direct drive; the drive transfer method became chain drive, similar to that on the bike.

The engine support frame was required to meet the following objectives:

- rigidity
- correct height (for required fitment of chain to sprocket)
- moveability
- fitment in dyno room

An important design criteria for the frame was that it provide sufficient rigidity to prevent any bending or movement during engine operation. It was also very important that the engine, when located in the frame, sat at the correct height to ensure correct placement of the chain on the sprocket. In addition to this, it was crucial that the position of the frame be fairly easily varied, so that very precise alignment of the chain and sprocket be facilitated. Another requirement, concerning practicality, was that the frame be capable of being easily fitted in the dyno room, preferably with engine attached.

The frame was constructed from 25 mm square hollow section (SHS), with the engine connected via front and rear engine mounts. The support frame is shown in Figure 10.7.

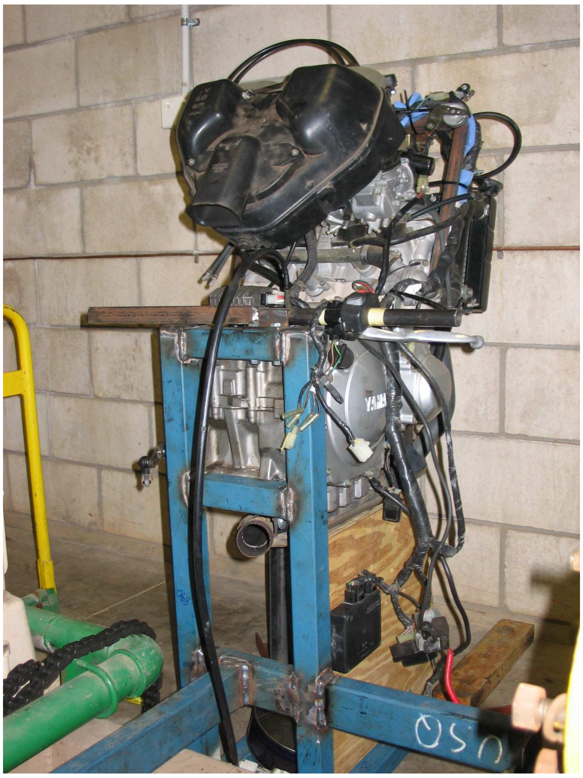


Figure 10.7: Engine support frame

10.8.2.3 Chain Guard Cover

Motorbike chains, in operation, move with excessive speed and therefore pose a risk to safety. A moving chain that dislodges while spinning can cause great damage to surrounding equipment, as well as inflicting severe injuries on any persons in the immediate vicinity of the incident. Therefore, as on a motorbike, the chain drive linking the engine to the dyno was required to be sufficiently covered to minimise damage in the event of the chain coming loose. The basic design criteria involved:

- ease of manufacture
- adequate material strength
- provision for performing maintenance on chain
- provision for connection to engine frame and dyno frame

The design of the chain guard was desired to be as simple as possible in order that manufacturing time be minimised. Therefore, a simple box-type shape was chosen, constructed from 1 mm steel plate (Figure 10.8). Provision for connection to the engine support frame at one end was accommodated by the addition of a length of angle; at the opposite end another length of angle formed a support leg, extending from one corner of the chain cover to the existing frame housing the dyno. A diagonal brace, between the engine frame and chain cover support leg, added additional torsional strength. A bolt-on, removable cover was incorporated to facilitate maintenance of the chain if required, and was made from 2 mm steel plate for additional strength and protection (Figure 10.9).

Figure 10.8: Chain guard cover

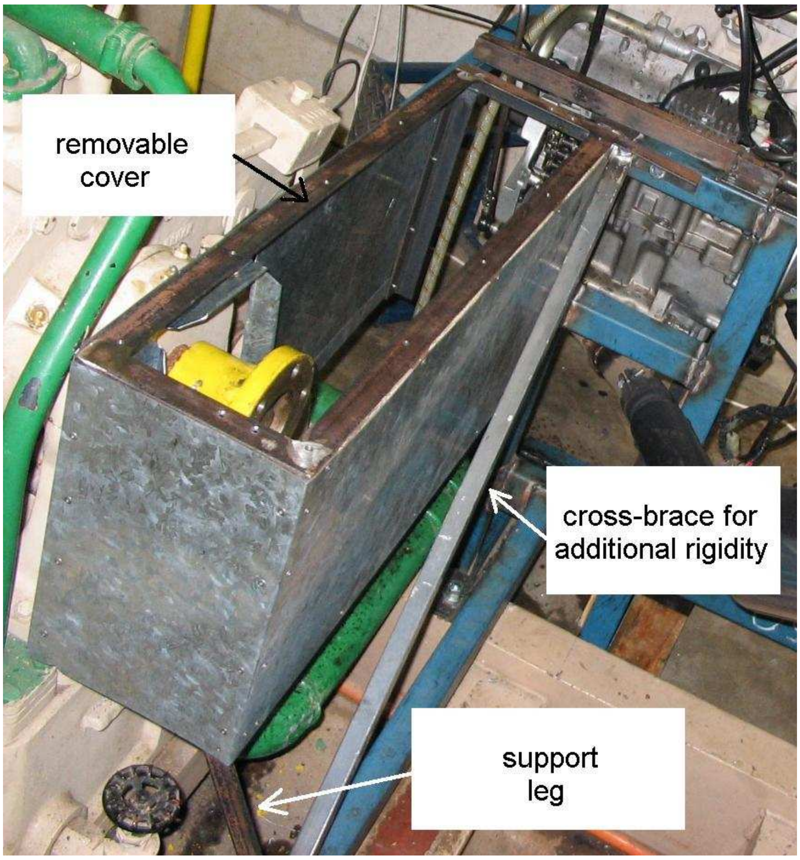


Figure 10.9: Some design features of the chain guard cover

10.8.2.4 Sprocket

The chain drive mechanism required that a suitable sprocket be manufactured to connect the engine to the dyno (Figure 10.10). The main design criteria for this sprocket was to produce a gear ratio that resulted in appropriate dynamometer speeds. A 46-tooth sprocket was designed, producing a gear ratio of 3.1:1 (46/15). In other words, the sprocket was designed to gear down the dyno to run at speeds of about one third actual engine speed.

After some initial testing, the dyno appeared to show some unsteady behaviour (as evident on the torque gauge) at lower (engine) speeds. It was therefore decided that running the dyno at higher speeds may produce some more consistent results. Another sprocket was manufactured, providing a gear ratio of approximately 1:1; however, it soon became clear that this gearing did not improve the stability of the torque readings, and that there were other causes for these varying readings (refer to section 11.4.4).

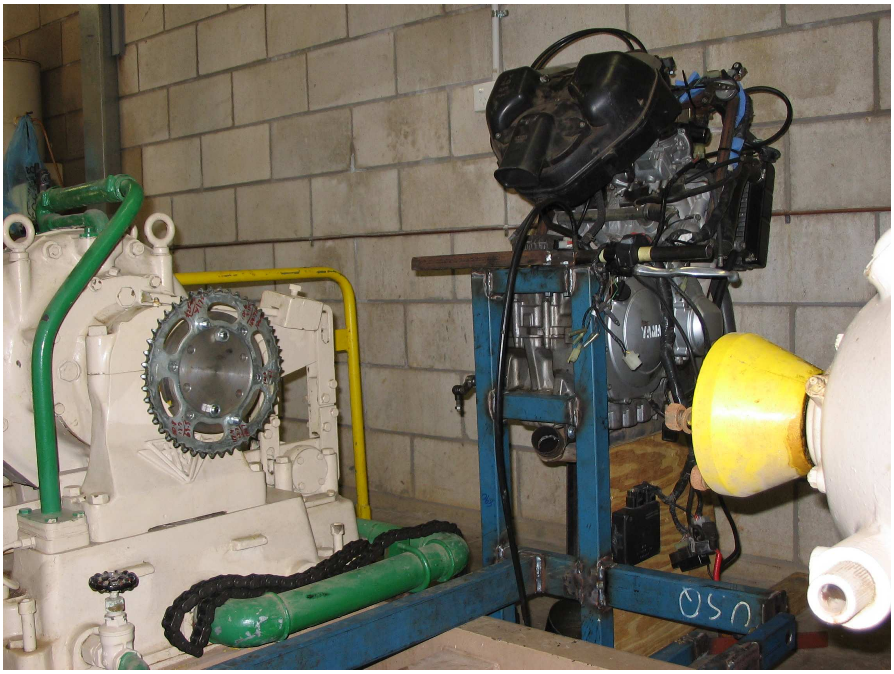


Figure 10.10: Dyno set-up, showing engine in frame and engine-to-dyno sprocket before chain connection

10.8.2.5 Fuel system

The existing fuel system comprised a fuel tank outside the dyno room, with a fuel pump and hoses delivering fuel to each of the Holden and Leyland engines. This system carried leaded fuel and hence modification was required to feed the YZF600 with the required unleaded fuel. An additional fuel tank was purchased, supplying 98 octane unleaded petrol to the YZF. Re-routing of the fuel beyond the fuel pump was then required, and was achieved using a length of fuel hose, feeding the bike's fuel supply system.

10.8.2.6 Remote Throttle Control

To optimise safety, it was critical that provision be made to enable control of engine speed outside the dyno room. Therefore, a system for remote throttle control had to be implemented. This involved extending a length of throttle cable from the engine throttle to outside the dyno room, a distance of around 6 m. Push-pull type cable was used, and connected at the non-engine end to a lever arrangement to facilitate speed control (Figure 10.11). A 'killswitch' was also accommodated on this lever to enable virtually instant shut-down of the engine in the event of a dangerous incident.

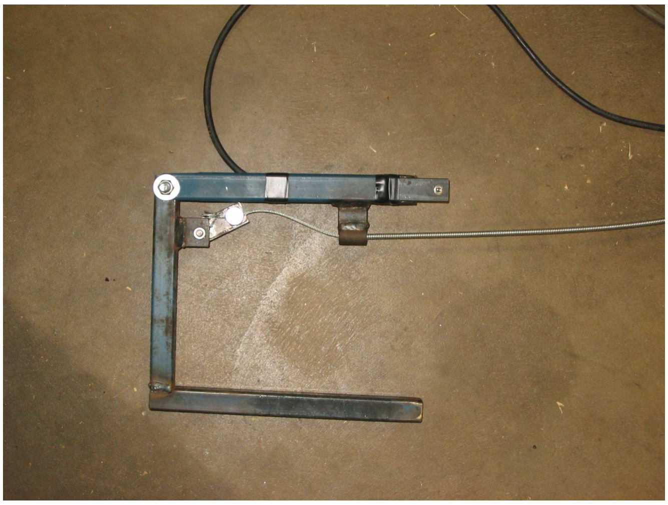


Figure 10.11: Remote throttle control lever

It was found that after around three months of weekly testing, the throttle cable became quite 'sticky', making smooth speed control very difficult. An alternative means of remote throttling is therefore recommended, using a higher gauge cable, or an outer sleeve over the cable.

10.9 Conclusion

As expected, establishment of a dynamometer for use in this application proved to be a relatively lengthy process. Design and implementation of the necessary modifications was not without challenges; however, eventually, a state was achieved that allowed connection of the YZF600 engine, ready for testing.

Chapter 11 presents the testing method employed, along with the results and discussion of testing the YZF600 on the Heenan-Froude dynamometer.

Chapter 11
Engine Testing and Analysis of Results

11.1 Introduction

Engine testing forms an integral part of any engine development program. Dynamometer testing can be used in determining performance characteristics, indicating directions for development and achieving comparative performance measurements. This chapter presents the methodology implemented in testing the YZF600 engine, along with the results obtained and the conclusions drawn from these results.

11.2 Intentions

Chapter 10 presented the process involved in selecting and modifying a dynamometer appropriate for use in development of USQ's Formula SAE engines. Having performed the necessary modifications, the dynamometer was then ready for the testing process to begin. Testing of the 1994 YZF600 engine was intended to achieve some important objectives, namely:

1. **Development of a reliable, consistent, repeatable dynamometer**
   Consistency and repeatability are the most important attributes of a dynamometer. The device must be capable of producing consistent results repeatedly, in order that modifications made to the engine or related components are able to be successfully tested, and the various arrangements compared.

2. **Establishment of benchmark performance characteristics**
   Establishing benchmark, or fundamental, performance characteristics for any engine is important for a number of reasons. Benchmark characteristics may indicate areas of performance that require improvement, thereby indicating directions for development; also, benchmark figures provide vital data which can be used as control-type measurements, against which testing of alternative engine set-ups can be compared.

3. **Determination of relative performance gains (or losses)**
   By its very nature, engine *development* relies on determining methods for improving performance. Therefore, it is absolutely necessary that results from testing enable determination of the factors contributing to performance enhancement or impedance.

The parameters intended to be measured included:

- power
- fuel consumption
- combustion efficiency

The configurations to be tested included:

- standard bike intake system
- 2004 USQ FSAE-A intake system
- 2005 USQ FSAE-A intake system

Further information regarding the aims of these tests is offered in the sections presenting the results of the respective tests.

11.3 Testing Methodology

Methodology is vitally important in ensuring the best potential for consistent results. Following a standard methodology for all tests performed enables variations in the testing procedure to be eliminated as a factor in inconsistent results.

The method developed for testing the YZF600 is as follows:

1. Run engine in top (6^{th}) gear
2. Open throttle to achieve desired (pre-determined) engine speed, read from the dynamometer tachometer
3. Maintain speed
4. Load engine gradually (via dyno load control switch), while monitoring dyno tacho, dyno torque gauge, bike tacho and engine temperature gauge
5. Continue loading engine until it can no longer maintain speed
6. At this point, record dyno speed (from dyno tacho) and torque gauge reading, with verification from at least a second observer
7. Repeat steps 1-6 for all desired speed intervals
8. Calculate power produced by engine (in kilowatts), according to equation 11.1, as supplied with the dynamometer operating manual:

$$P = W \times N / 536 \tag{11.1}$$

where P is the calculated power (in kW)
W is the torque gauge reading
N is the speed (rpm) from the dyno tacho

Manufacturers generally quote engine power in terms of horsepower (hp). To convert to horsepower, kilowatts is by multiplied by (4/3).

Dyno testing is generally performed by taking readings across an engine's power band, in increments of 250 or 500 rpm. Therefore, it was decided to take readings across the engine's power band, from 6 000 to 10 000 rpm, in steps of 500 rpm, for testing in each case.

11.4 Testing Results

11.4.1 Establishing the Dynamometer's Consistency and Repeatability

Initial, repeated testing of the engine in its configuration as on the motorbike was determined to be the most suitable method for establishing whether the dynamometer was capable of producing consistent results. Consistent results would indicate the repeatability potential of the dyno (refer to section 10.4.2); that is, its potential to be used as a comparative measuring device, with the ability to draw comparisons between various engine arrangements.

The engine, configured with motorbike air intake, 32 mm Mikuni carburettors feeding each cylinder and 98 octane fuel supply, will be referred to as being in 'standard bike trim' (Figure 11.1). This configuration was tested several times, across a number of days, and indicated very consistent torque and power readings in each session. These results are presented in the next section.

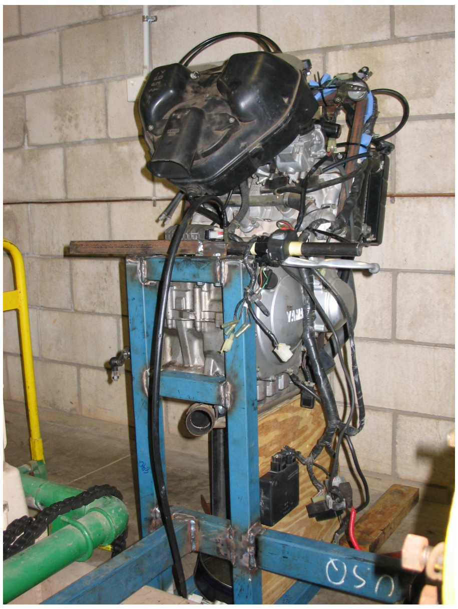


Figure 11.1: YZF600 in 'standard bike trim'

11.4.2 Determining Benchmark Performance Characteristics

Torque and power figures for various Yamaha motorbikes are available from a variety of sources, including the official Yamaha Motors website (www.yamaha-motors.jp), as well as numerous other private websites, from consumers who have had their machines dyno tested, or have dyno tested their own, and published the results.

Testing of the YZF600 on the Heenan-Froude engine dyno at USQ can provide an interesting comparison with these published figures, and may indicate some shortfalls in the testing procedure used.

Published torque and power figures, or torque and power curves, were not able to be found for a model directly matching the 1994 Yamaha YZF600 used by USQ in 2005. The closest match was found in the form of a 1997 Yamaha YZF600R (www.geocites.com). Research regarding performance figures for various other models indicated that an appreciable difference in performance between these two bikes would be unlikely (www.yamaha-motors.jp). Therefore, the figures for this later model bike will be used for comparison purposes.

- **1997 Yamaha YZF600R**

Power and torque curves for a 1997 YZF600R are presented in Appendix C.

- **1994 Yamaha YZF600**

Testing of the 1994 YZF600, in standard bike trim, on the Heenan-Froude dyno, yielded power and torque figures (refer to Appendix D for testing results). These results are presented graphically in Figures 11.2 and 11.3, showing comparisons with those prescribed for the 1997 model YZF.

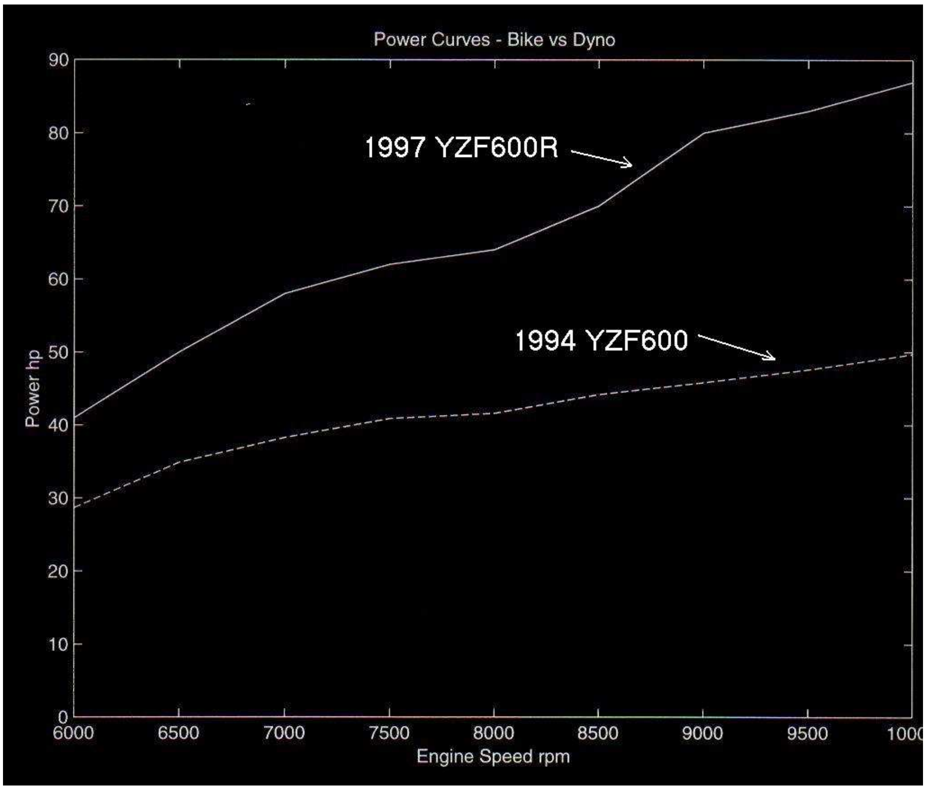


Figure 11.2: Comparison of power curves for 1994 and 1997 model YZF600 motorbikes, between 6 000 – 10 000 rpm

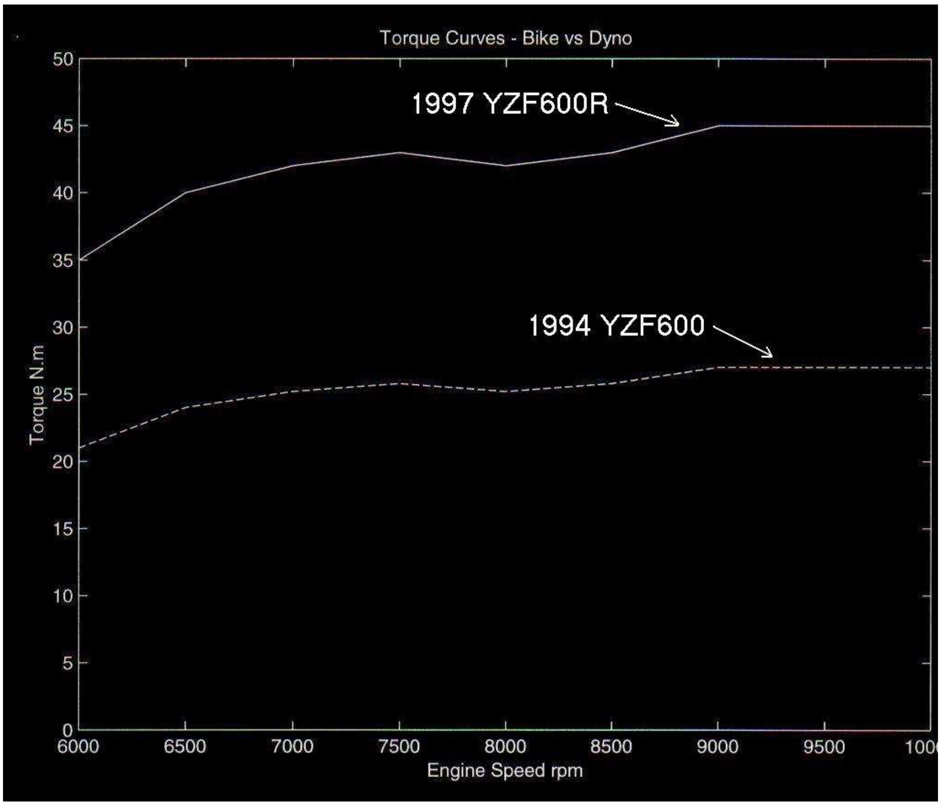


Figure 11.3: Comparison of torque curves for 1994 and 1997 model YZF600 motorbikes, between 6 000 – 10 000 rpm

This engine testing in standard bike trim indicated a maximum power of 48 hp, produced at 9 500 rpm, with peak torque delivered at 7 000 rpm. While this initial testing produced very consistent results, it also indicated an apparent drop in torque and power of approximately 40 %, compared with the figures given for the similar 1997 model YZF.

11.4.3 Exploring Causes of Apparent Power Loss

Power and torque data for high performance 600 cc motorbike engines generally indicate figures more in the range of those for the 1997 YZF. Therefore, it was concluded that the Heenan-Froude dyno was more than likely producing low readings, due to some combination of factors relating to the configuration of the engine and dyno. The most likely contributing factors were identified as:

- possible flow restriction caused by additional exhaust pipe after muffler (refer to section 10.7.2.1)
- possible reduction in intake air density, as air is breathed from within dyno room
- losses through chain drive mechanism

In combination, these factors have the potential to generate significantly reduced power readings, possibly as high as 20 % (Malpress R 2005, pers. comm., 9 August).

● **Restriction to Flow**

A severe restriction beyond the muffler will result in a high pressure build-up, causing restriction to combustion gases attempting to exit the exhaust system. This will result in decreased capacity for charge to enter the cylinders, due to exhaust backflow, resulting in reduced power.

● **Higher Intake Air Temperature**

During operation, the engine produces a significant amount of heat, which increases the air temperature within the dyno room. The engine breathes this air; therefore, with an increased inlet air temperature, a less dense charge will be delivered to the cylinders. This will result in decreased power production.

● **Mechanical Losses**

Some mechanical loss thorough the drive mechanism is unavoidable;
however, if there are substantial losses through this drive, there will be a corresponding substantial decrease in power.

The drive mechanism between the engine and dyno is similar to the arrangement on a motorbike. Therefore, significant losses of this form are not likely to arise.

Testing was carried out to determine if the reduced power readings were due, partially or wholly, to the effects listed above.

Under the supervision of USQ Occupational Health and Safety Officers, the engine was run on the dyno with the additional exhaust pipe removed. Results indicated no change in power to that produced with the exhaust pipe connected. It was concluded therefore, that this additional pipe caused no appreciable restriction.

Air from outside the dyno room (at a temperature of 19°) was then made available to the engine's intake, by means of a generic 'leaf blower' pumping air through a hose to the air intake. Again, dyno results were unchanged, and it was concluded that the rise in temperature of the intake air was not contributing to power losses.

Air temperature and pressure were monitored during dyno runs, as these variables can each cause losses in power (refer to section 10.4.1). Air pressure readings showed little variation, while air temperature increased a maximum of 5°, from 19° to 24°, over a period of about half an hour, in which the dyno was run several times. This relatively minor temperature increase would probably have little influence on power readings, and this assertion was supported by the invariance in results when (cooler) outside air was delivered to the engine.

The dyno's torque measuring device was also investigated as a possible cause for the apparently low readings; however, the device appeared to be correctly calibrated.

These tests indicated that the most likely factors thought to be contributing to reduced power readings were, in fact, ineffectual in this case. The cause of these apparent power losses therefore remains undetermined.

Being unable to determine unequivocally whether the engine was actually producing the power that it was claimed to be capable of was somewhat disconcerting; however, the engine appeared to be in good running order, and was guaranteed to be as such at the time of sale (Toowoomba Yamaha Sales Manager 2005, pers. comm., 21 May). That is, there were no obvious reasons for the engine to actually be producing reduced power, and it was concluded that other factors were the cause for these unexpected readings, yet were unable to be defined.

This inability was essentially ineffectual, as a set of baseline readings was able to be produced consistently by the dyno, and would therefore stand as a control against which power readings for alternative engine set-ups could be measured.

11.4.4 Testing of the 2004 USQ FSAE-A Intake System

Once it was established that the dynamometer offered consistent readings, testing of the 2004 intake system began. The aim of this testing was to quantify the anticipated change in engine performance characteristics due to this manifold. These readings would indicate the loss in power due to this arrangement compared with the standard bike intake system, and would also be used to compare the 2004 design with the 2005 intake system design, to determine and quantify the anticipated improvements. The 2004 intake manifold mounted on the engine in preparation for dyno testing is shown in Figure 11.4.

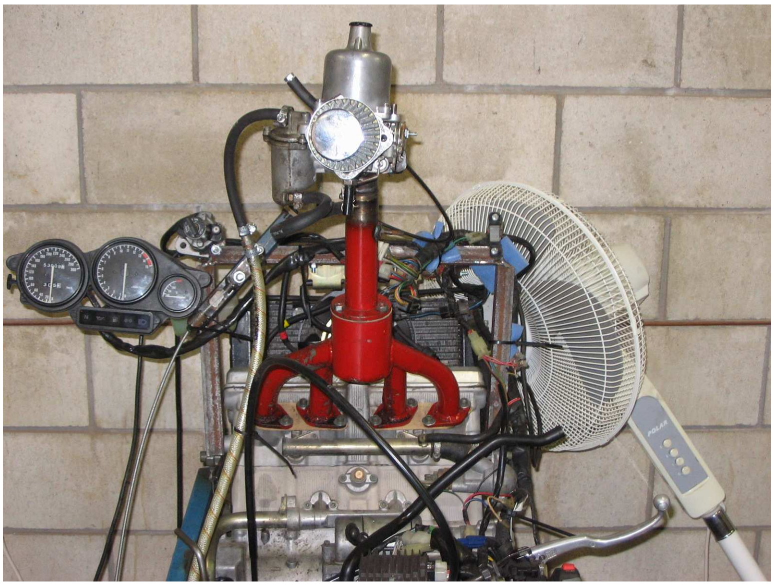


Figure 11.4: Dyno testing set-up of 2004 intake manifold

The methodology outlined in section 11.3 was again followed for this testing, running the engine at speeds between 6 000 and 10 000 rpm, as specified in section 11.3.

The first run, at a speed of 6 000 rpm, produced very unsteady readings on the torque gauge once the load was applied. The engine was then run at the next incremental speed (6 500 rpm), however the same instability resulted. The engine was then run up to a significantly higher speed (9 500 rpm), to determine whether this greater speed would overcome the unstable torque readings; no change was evident.

When the engine was run in standard bike trim, the needle on the torque dial moved constantly higher before peaking at a definite value, with minimum and maximum readings determined as 4.7 and 5.4 across the range of speeds tested (6 000 – 9 500 rpm). However, when the 2004 inlet system replaced the motorbike intake, the needle moved rapidly between very low and relatively high readings constantly. The range covered by the needle was approximately 2.4 up to 6.2, and the torque needle did not settle on a specific value at any point during the loading.

It was unclear as to which factor was specifically affecting the wildly varying torque readings, as in switching from the motorbike intake to the 2004 FSAE-A intake, multiple variables were changed simultaneously. Therefore, investigation into the possible causes was required.

The most obvious causes to pursue focussed on the changes introduced by the 2004 inlet manifold, namely:

- 20 mm intake restrictor
- different carburettor type
- different manifold type

It was determined to attempt to eliminate each of these factors individually, as the cause of the torque dial variance. Changes were made to the manifold configuration accordingly, with further dyno testing performed across the engine's power band after each modification, and these methods are outlined below.

● **First Trial**

The first trial involved removing the 20 mm restrictor, and replacing it with an equivalent length of straight-through pipe. It was thought that this restrictor may be severely limiting airflow to the engine, perhaps contributing to the inability to produce a definite torque reading.

The carburettor feeding the 2004 manifold when initially connected to the dyno was a Weber-type carburettor. This type of carburettor is required to be mounted vertically, thus requiring an elbow-type joint for connection to the intake system (Figure 11.5 a). It was unsure as to how much restriction this 90° bend may be causing; however, the first step was to determine whether it was the restrictor providing the greatest obstruction to flow, and so the shape of the carburettor-to plenum connecting pipe was retained (Figure 11.5 b), complete with elbow and SU carburettor.

(a) **(b)**

Figure 11.5: (a) Existing SU carburettor connection with 20 mm restrictor tube; (b) replacement connection, in 36 mm ID straight-through pipe

This replacement connection was made from 36 mm ID pipe, in accordance with the carburettor throat diameter.

● **Testing Results**

Replacement of the restrictor with a straight pipe, while retaining all other intake features, provided no change in the instability of the torque readings.

● **Second Trial**

The second trial focussed on carburettor type, replacing the current SU carburettor with a downdraught type, while retaining the restrictor.

It was unknown how much restriction the elbow joint was causing in the manifold; it was also preferable to run the same carburettor type on both manifolds, even though the bike manifold uses four separate carburettors in comparison with the FSAE-A manifold's single carburettor. The elbow was removed from the top of the restrictor, and an adaptor plate added to incorporate a 32 mm downdraught (Mikuni brand) carburettor, as used on the 1993 FZR600 from last year's car.

- **Testing Results**

  Testing across the engine's power band produced no contribution to achieving a stable torque reading.

- **Third Trial**

Thirdly, the restrictor was replaced with a straight-through pipe of equal length, and the downdraught carburettor was maintained (Figure 11.6). The restriction caused by the single 20 mm hole for airflow was possibly causing great enough restriction to flow to severely limit the power being produced, which was thought to possibly have some effect on the attempts by the dyno to produce a stable torque reading.

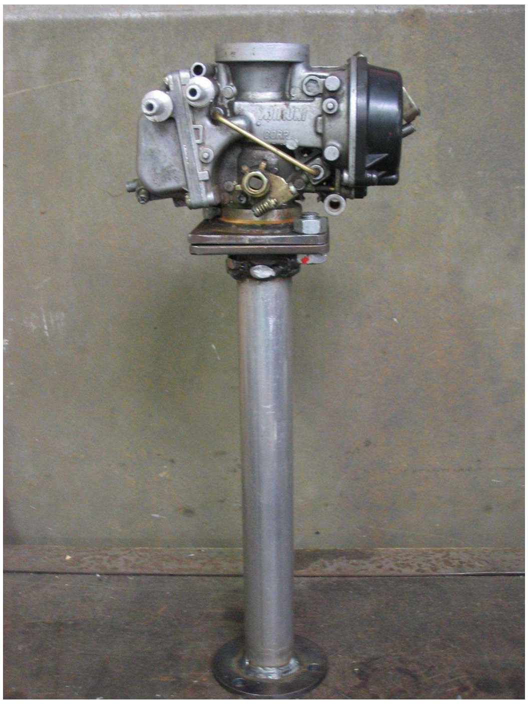


Figure 11.6: Elbow replaced by straight pipe, with carburettor type retained

A 32 mm ID pipe, matching the carburettor throat size, equivalent in length to the restrictor, was connected between the plenum and carburettor.

- **Testing Results**

  Testing again showed no difference in the unstable results produced on the torque gauge.

- **Fourth Trial**

The fourth trial involved removal of the pipe between the carburettor and plenum altogether, as the pipe itself may have been causing restriction to flow. The 32 mm Mikuni carburettor was retained, and mounted to feed the plenum directly.

- **Testing Results**

  Further testing followed this third modification; however, the torque readings still varied dramatically, with no stable reading resulting.

Implementation and testing of these modifications offered no apparent benefits in improving the stability of the readings on the torque dial. It was then decided to re-test the engine with the standard motorbike manifold attached, to ensure that the same readings produced in the first set of testing were still achievable. This would eliminate the possibility that the unstable readings were the result of an equipment malfunction, relating to the dyno, rather than being caused in some way by the configuration of the 2004 intake manifold.

The bike intake system was repositioned on the engine, and subsequent testing produced very similar results to the initial phase of testing. This indicated that some aspect of the 2004 manifold configuration was likely to be the cause of the inability to produce stable results.

11.4.5 Pursuing Alternative Testing Mechanisms

As this dynamometer testing was intended to provide the primary means of quantifying engine performance characteristics and providing a method of comparison between intake system designs, an alternative testing method was required. There was no short-term solution to enable useable torque and power data to be produced by the engine for the FSAE-A type intake manifold. However, other parameters able to be measured during engine operation could be used to compare operating characteristics produced by various manifold types.

Analysing exhaust gases can provide substantial information about the effects of a particular manifold on flow, delivery of charge and production of power. Exhaust gas analysers typically measure the percentages of oxygen (O_2) and carbon monoxide (CO) present in the products of combustion. From this, it can be determined whether an engine is running lean or rich, with high percentages of oxygen indicating inadequate supply of fuel.

Combustion efficiency can also be determined, with high amounts of unburnt fuel producing higher emissions. This information therefore indicates the ability of the intake manifold to maintain intake charge velocity and hence promote atomisation of fuel, with lower emissions indicating a more efficient intake system, which should then offer improved fuel efficiency and greater power.

Therefore, it was intended to run each of the 2004 and 2005 intake systems on the dyno, and use exhaust gas analysis to determine the superiority of one design over the other.

A gas analyser was procured for this purpose. However, lack of time prevented proper calibration and useful operation of the analyser.

At this point, it was decided to end pursuits to use the Heenan-Froude dynamometer as a device for determining engine performance, at least in regard to this particular project, largely due to lack of time.

11.5 Conclusion

Numerous avenues were pursued in attempting to produce useful results from the Heenan-Froude dynamometer over a three-month period. Ultimately, it was concluded that considerable more time would need to be devoted to this system to determine definitely its suitability for testing engines and related systems used in the application of Formula SAE.

The dynamometer proved capable of producing consistent results for the standard bike intake system configuration, and also proved reliable through a relatively long period of testing.

However, the age of the dyno means that all its instrumentation is analogue, relying on observers to correctly record testing results, which is likely to increase errors attributed to incorrect observations. Modern equipment offers more advanced digital instrumentation, and although requiring to be correctly and accurately calibrated, provides facility for recording engine and dyno output, hence producing records of engine operation and performance.

The Heenan-Froude dynamometer is also limited in its ability to indicate fractional changes in power and torque, as its sensitivity is low. This system can offer approximate, broad indications of power output, but for applications seeking optimum performance through fractional gains, a much more sensitive dynamometer is required.

There are many opportunities for performance gain related to the various aspects of engine operation. Engine performance forms an important aspect of overall vehicle performance in all racing applications, including Formula SAE. A quality engine development program is crucial in pursuing and achieving optimised performance. To facilitate future research projects in this area, a suitable dynamometer is absolutely essential, as extensive testing of various configurations is the most efficient and effective method of achieving optimum engine performance.

Chapter 12
Conclusion

12.1 Introduction

The execution of this project aimed to achieve a number of important objectives. Gaining familiarity with the engine used in USQ's Formula SAE project was paramount to determining and implementing suitable methods to enhance engine performance. Developing a suitable facility to enable ongoing testing, diagnosis and analysis of engine performance also formed an integral component of this research.

The methods employed in conducting research enabled achievement of the objectives specified in Chapter 1, and also produced some interesting and practical information to indicate directions for further work. Discussion of achievement of these objectives now follows.

12.2 Achievement of Objectives

1. The environment and boundaries within which engine modification is permitted in terms of the Formula SAE-Australasia Competition was determined by conducting research into the rules and regulations associated with the event. The requirements of engine performance specifically relating

to this application were also identified, through review of the performance of USQ's 2004 FSAE-A entry, in addition to exploring the demands of each of the competition's events. A basis for development of improved engine performance was therefore established.

2. A variety of methods suitable for increasing performance and achieving the specified objectives were researched. From this, a conclusion was drawn recommending that considering the time allocated to this project, it was more beneficial to concentrate on a single area for improvement, rather than attempting to apply a range of modifications. This method would enable adequate design, implementation, testing and development time, with the area for focus to be determined by the aspect of current engine configuration identified as most in need of modification, with regard to the competition's performance requirements.

3. The requirement of a single restrictor at inlet mandates that the intake system for the engine in competition be significantly modified from that of the configuration on the motorbike. The changes necessitated by this regulation were seen as potentially causing a significant reduction in performance, due to a substantially reduced potential for intake charge to enter the engine. Design of an intake system specifically aimed at maximising intake potential, fuel efficiency and power, was identified as a crucial component in achieving the performance characteristics required by the nature of the competition. Design of a custom exhaust system to complement the intake configuration was also deemed as integral to improving overall performance characteristics.

4. Access to a reliable and suitable engine testing facility was identified as integral to the progress of an engine development program. Testing of various engine configurations and establishing benchmark performance characteristics is most suitably facilitated by an engine dynamometer.

Objective analysis of all available options, according to the criteria outlined in Chapter 10, identified the dynamometer most likely to facilitate the intended engine testing.

This dynamometer proved capable of producing consistent results, and a set of performance data was obtained for the engine in standard bike trim. However, despite considerable time and effort devoted to modification of the dyno and pursuing various testing configurations, the end result indicates that this dynamometer is not suitable for testing of engines in Formula SAE trim; that is, with inlet restrictor and custom intake manifold fitted.

An engine dynamometer is seen as being absolutely integral to continued development of engine performance; thus it is strongly recommended that a suitable dynamometer be purchased, not only for use in the Formula SAE project, but to facilitate establishment of an engine development and research program within the Faculty of Engineering and Surveying.

5. Many alternatives were pursued to achieve a dynamometer that could be used to indicate performance differences between various intake and exhaust configurations. However, extensive testing indicated that the equipment itself was not suitable for the intended application. Therefore, quantification of engine performance related to redesign of the intake and exhaust systems was not able to be performed. This also precluded the ability to establish conclusively, with testing verification, that the 2005 intake and exhaust systems offered improved performance characteristics over their 2004 counterparts.

However, the redesigned intake and exhaust components for this year's car were designed in accordance with fundamental engineering principles, and with consideration of certain aspects of the 2004 designs identified as requiring improvement. Therefore, a conclusion can be drawn that, in accordance with the design approach implemented, the 2005 intake and exhaust systems should provide improved performance characteristics over last year's designs, in terms of better fuel efficiency, increased power and improved reliability.

6. While a facility for quantifying overall engine performance improvement as a result of the designs implemented was unable to be achieved as desired, thorough research enabled identification of performance requirements, methods for attaining this performance and design methods for application-specific inlet and exhaust configurations. Therefore, this approach should enable improved engine performance, achieving the objectives of the project.

12.3 Further Work and Recommendations

Some aspects of design relating to various engine ancillaries are yet to be fully determined. Definitive positioning of the radiator must be considered further, with the most likely options being to mount the radiator either towards the rear of the vehicle, near the rear axle, or on the right-hand side of the vehicle, towards the front of the cockpit, at driver shoulder height.

Analysis of the thermal effects of each of these positions would be required, by implementing heat transfer methods, to determine whether each position provided adequate cooling. Positioning of the radiator at the side of the cockpit would enable greater cooling effects, due to increased exposure to airflow; however, this greater exposure would also make the radiator more susceptible to damage from low-flying objects, such as loose rocks on the track, and the radiator would therefore require shielding. Additional pipes, clamps and other ancillaries would also be required to facilitate pumping of radiator fluids over a greater distance in this configuration.

Positioning the radiator at the rear of the vehicle would minimise the lengths of pipes needed for transfer of fluids. However, this positioning would also provide less airflow to the radiator, which could significantly affect its cooling ability. The magnitude of this effect would need to be determined, to ensure that this positioning would offer adequate cooling, to prevent engine overheating. These factors must all be considered to determine the most viable option for radiator positioning.

Carburettor choice is another aspect requiring further development time. This situation has arisen due to the inability to test various carburettors with the given engine configuration. Therefore, testing of different carburettor types will take place once the vehicle is completed, with on-track performance dictating carburettor choice and tuning potential.

Flow bench testing is a mechanism that may be able to provide an indication of the improvement in design of the 2005 systems compared with last year's. Testing of the restrictors and inlet manifolds from both years will be pursued, to determine the flow abilities and characteristics of each. It is anticipated that such testing will provide conclusive data describing fuel atomisation and combustion characteristics, in addition to power production potential, of the 2005 intake design.

Execution of this project has also identified a number of areas that should be afforded further research and development resources, as their potential for significantly improving overall engine performance is regarded as very high. These recommendations are outlined below.

- **Intake and Exhaust System Testing and Optimisation**

  Intake and exhaust system design are two relatively complex areas, with many variables influencing each system's ability to contribute to optimum performance overall. In particular, the nature of flow through intake and exhaust systems is complicated, with the ability to produce some interesting behaviour. If the flow through these systems can be directed to take advantage of fuel atomising, velocity maintaining and wave tuning characteristics, very significant performance improvement can be achieved.

  Therefore, analysis of various intake and exhaust configurations can provide indications of methods for improving performance by varying parameters such as manifold length, diameter and shape. It is recommended that a computational fluid dynamics program be used in intake and exhaust system design in order to extensively analyse various configurations, and determine the best combination of parameters for each design, to achieve the intended performance characteristics. A program such as *Fluent*, which is available at USQ, would be suitable for this development.

Further improvement of these systems would be achieved through dynamometer testing, particularly in relation to wave tuning characteristics. Various lengths of intake runners and exhaust pipes could be tested, facilitated by use of spacers, rather than manufacturing a number of complete manifolds, to tune the systems for the desired speeds.

● **Fuel Delivery Method**

As discussed in Chapter 7, fuel delivery method is an important aspect of overall intake design. Electronic fuel injection has been shown to offer superior performance over carburetted systems, in terms of improved fuel economy and power. In 2005, an ECU was donated to USQ, and effectively all other components necessary for implementation of a fuel injection system have now been acquired. Therefore, it is strongly recommended that an EFI system be developed, tested and tuned for implementation on next year's car.

● **Testing Facility**

The need for a suitable, reliable, repeatable means of engine testing and comparison has been discussed in Chapters 10 and 11, and previously in this concluding chapter. It is recommended that further research be conducted to determine a suitable engine dynamometer for use in Formula SAE engine testing, as well as in general engine development programs. It is also recommended that this equipment be purchased as soon as possible, in order that further developmental work relating to engine performance be facilitated in the short-term.

- **Other Engine Modifications**

  Research into methods to achieve improved engine performance was presented briefly in Chapter 4. There are many options available in pursuit of this goal, and it is recommended that further research be conducted into each of these, to determine additional methods of increasing engine performance. Particularly, changing camshaft and ignition timing to suit the engine's operating conditions in Formula SAE, and analysing the benefits of implementing a forced induction system are likely to offer significant benefits to performance. Thorough research into these methods in relation to the performance objectives identified is required to maximise the benefit of implementing further modifications.

12.4 Personal Appraisal

While the objectives of this project were devised to enhance the overall performance of USQ's entry in Formula SAE-A 2005, this project has presented me with an invaluable opportunity to increase my knowledge and explore my passion for engine performance optimisation.

Through the various research, design, manufacturing and testing aspects comprising this project, I have expanded my knowledge considerably throughout the course of the year. My understanding of the operating principles of the internal combustion engine has been significantly enhanced, and exposure to its various intricacies has afforded me a better appreciation of the numerous variables contributing to overall engine performance.

The opportunity to perform extensive dynamometer testing has been invaluable, with exposure to this aspect of engine development forming a basis for my planned future work in this field. Evolving into a substantially large 'debugging' exercise, modification of the dynamometer also provided an opportunity to further develop my problem solving skills.

Working within a team and reaching compromises to design an overall optimised racing package has been very challenging, but extremely rewarding. The opportunity to design, build and test various engineering creations has facilitated a steep learning curve, but also provided invaluable experience and immense satisfaction in creating systems that provide functionality, while enhancing performance.

12.5 Conclusion

Execution of this project has been intended to determine methods for improving engine performance for a Formula SAE race car, and furthermore, to implement these methods to achieve enhanced performance. The objectives specified to facilitate the overall project aim have been achieved, with some valuable conclusions able to be drawn from these results.

Familiarity with the operating characteristics of the engine used in USQ's 2005 car has been gained, and the requirements for performance in Formula SAE identified, enabling determination of the methods for achieving the required performance.

The mechanisms by which intake and exhaust systems affect overall engine performance have been thoroughly investigated, with design of these systems following accordingly, which should achieve significantly improved engine performance.

Methods for facilitating quantification of engine performance and comparison between various engine configurations have also been explored, and have culminated in the modification of a dynamometer for use in engine development. However, the suitability of this facility for the purpose of Formula SAE engine testing has yet to be conclusively determined.

Overall, implementation of methods to achieve the specified objectives has enabled successful attainment of the project aim. This achievement should result in significantly improved engine performance, in addition to improved overall vehicle performance of USQ's 2005 race car.

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Appendix A: Project Specification

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR: **Melinda Rachel PLANK**

TOPIC: Engine Performance Characteristics for a Formula SAE Race Car

SUPERVISORS: Dr Selvan Pather
Mr Derek Mulder

SPONSORSHIP: USQ Faculty of Engineering and Surveying

PROJECT AIM: To determine the performance characteristics of the current USQ Formula SAE race car engine, and to explore methods of Performance improvement to maximise engine output.

PROGRAMME: Issue B, 29 May 2005

1. Conduct background research into the Formula SAE Rules, changes to the rules in 2005, and FSAE limitations and guidelines pertaining to engine performance.

2. Determine mechanism of measuring engine performance, and assist with establishing related equipment.

3. Establish full specification and performance characteristics of 2005 engine.

4. Investigate possible methods of improving engine performance.

5. Design systems to implement improvements.

6. Conduct appropriate testing to quantify performance improvements.

7. Hence determine overall performance improvement gained.

As time and resources permit:

8. Implement further performance improvement methods.

9. Conduct appropriate testing to quantify further performance gains.

AGREED: ________________________ (student) ______________________ (supervisor)

(dated) ___/___/___ ___/___/___

Appendix B: Yamaha Electronic Fuel Injection System Components

B

This appendix contains schematic diagrams showing the components of an Electronic Fuel Injection System for a 2004 Yamaha YZF-R1 (1000 cc) motorbike.

Printed 21/06/2005

Ref #	Part No	Description	Qty	Remarks
1	90201-12790	WASHER, PLATE	1	
2	8CC-85790-01	THERMOSENSOR ASSY	1	
3	5VY-81673-00	ROTOR	1	
4	90282-03001	KEY, STRAIGHT	1	
5	90201-103U4	WASHER, PLATE	1	
6	90105-10290	BOLT, FLANGE	1	
7	5VY-85820-00	SERVO MOTOR ASSY	1	
8	5VY-85829-00	COVER, SERVO MOTOR	1	
9	97707-50514	SCREW, TAPPING	2	
10	5VX-82380-00	SENSOR, PRESSURE	1	
11	97707-50020	SCREW, TAPPING	2	
12	5VY-82310-00	IGNITION COIL ASSY	4	
13	5VY-82309-00	WIRE, SUB LEAD	1	
14	3GB-82540-01	NEUTRAL SWITCH ASSY	1	
15	90430-10148	GASKET	1	
16	5VY-85720-00	OIL LEVEL GAUGE ASSY	1	
17	95027-06016	BOLT, FLANGE	2	
18	90465-08013	CLAMP	1	
19	4XV-83755-01	SENSOR, SPEED	1	
20	90464-51001	CLAMP	1	
21	90110-06041	BOLT, HEXAGON SOCKET HEAD	1	
22	90465-08028	CLAMP	1	
23	5VY-81670-00	PICK-UP ASSY	1	
24	5VY-15589-00	PLATE, CLUTCH COVER	1	
25	90110-06041	BOLT, HEXAGON SOCKET HEAD	1	
26	90465-08191	CLAMP	1	
27	5VY-85896-00	SENSOR, CAM POSITION	1	
28	90110-06117	BOLT, HEXAGON SOCKET HEAD	1	

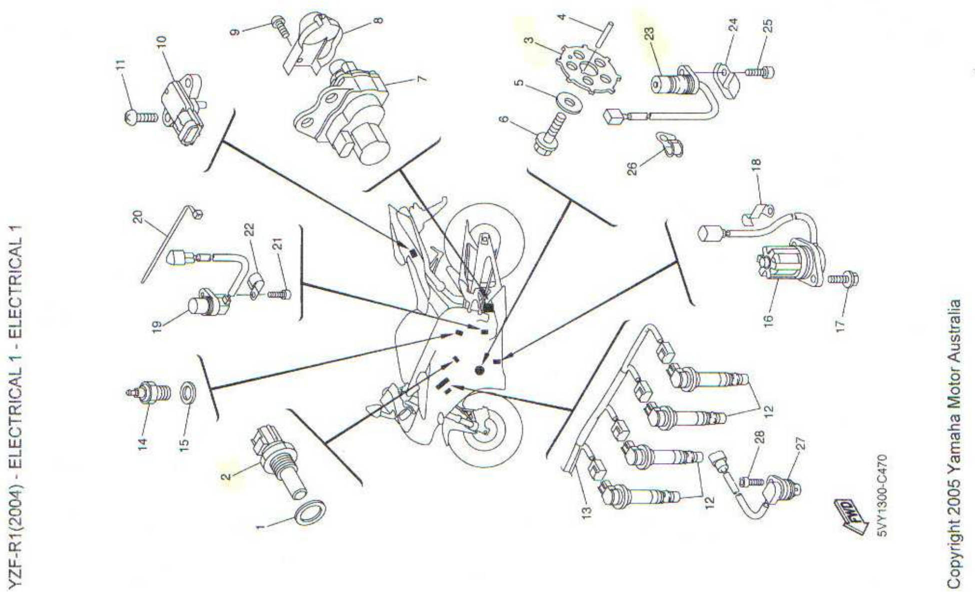

Figure B.2: Electrical system components required for YZF-R1 EFI system

(Source: Toowoomba Yamaha 2005)

Printed 21/06/2005

Ref #	Part No	Description	Qty	Remarks
1	5VY-13595-00	JOINT	1	
2	90450-62001	HOSE CLAMP ASSY	2	
3	5VY-13596-00	JOINT, CARBURETOR 2	1	
4	90450-62001	HOSE CLAMP ASSY	2	
5	5VY-13597-00	JOINT, CARBURETOR 3	1	
6	90450-62001	HOSE CLAMP ASSY	2	
7	5VY-13598-00	JOINT, CARBURETOR 4	1	
8	90450-62001	HOSE CLAMP ASSY	2	
9	5VY-14411-00	CASE, AIR CLEANER 1	1	
10	5VY-14421-00	CASE, AIR CLEANER 2	1	
11	5VY-14451-00	ELEMENT, AIR CLEANER	1	
12	5VY-14419-00	PIPE	1	
13	31A-14435-00	GROMMET	1	
14	5CA-85886-00	SENSOR, AIR TEMPRATURE	1	
15	5VY-14435-00	GROMMET	1	
16	5VY-14452-00	SEAL	1	
17	5VY-14462-00	SEAL	1	
18	5VY-14457-00	SEAL	2	
19	5VY-15393-00	PIPE, BREATHER 2	2	
20	90468-08104	CLIP	1	
21	90480-08264	GROMMET	2	
22	5VY-14498-00	STAY 1	1	
23	5VY-14467-00	SEAL	1	
24	5VY-1440B-00	JOINT ASSY	2	
25	90109-05005	BOLT	3	
26	90154-05004	SCREW, BINDING	10	
27	5VY-14862-00	PLUG	1	
28	5VY-14425-00	DAMPER 1	1	
29	90467-14015	CLIP	1	
30	5VY-12682-00	PROTECTOR, HEAT	1	

YZF-R1(2004) - INTAKE - INTAKE

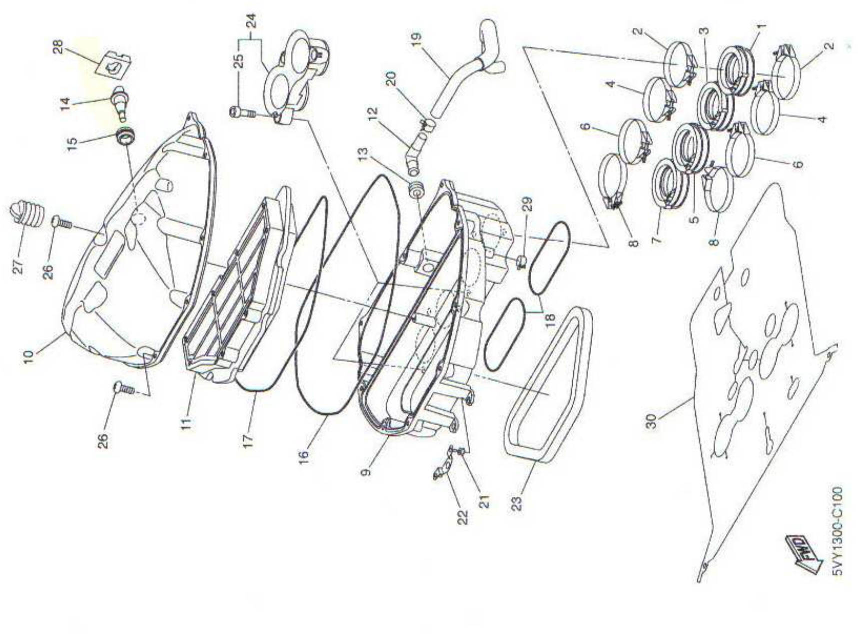

Figure B.3: Intake system components required for YZF-R1 EFI system

(Source: Toowoomba Yamaha 2005)

YZF-R1(2004) - INTAKE 2 - INTAKE 2

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Ref #	Part No	Description	Qty	Remarks
1	5VY-13750-00	THROTTLE BODY ASSY	1	
2	5FL-85885-01	THROTTLE SENSOR ASSY	2	
3	5VY-85850-00	GEARED MOTOR COMP	1	
4	22U-14952-00	WASHER	2	
5	91492-12010	PIN, COTTER	2	
6	98507-06012	SCREW, PAN HEAD	2	
7	98507-05008	SCREW, PAN HEAD	2	
8	5VY-13906-00	REGULATOR, PRESSURE	1	
9	4KM-14137-00	CLIP	5	
10	5VY-82386-00	EXTENSION, WIRE HARNESS	1	
11	68V-24376-00	PIPE, JOINT 1	2	
12	5JW-14396-00	PIPE	1	
13	5PW-1410F-00	CONNECTOR	1	
14	98507-05014	SCREW, PAN HEAD	4	
15	5VY-14171-00	PLUNGER, STARTER	2	
16	5VY-14204-00	CASE, STARTER	1	
17	5VX-82380-00	SENSOR, PRESSURE	1	
18	98507-05016	SCREW, PAN HEAD	2	
19	92907-05100	WASHER, SPRING	2	
20	92907-05600	WASHER, PLATE	2	
21	5FL-14349-10	PIPE	2	
22	98507-03016	SCREW, PAN HEAD	4	
23	92990-03100	WASHER, SPRING	4	
24	5PW-14349-00	PIPE	2	
25	5PS-14987-00	HOSE	3	
26	5PW-14251-00	NIPPLE	2	
27	5DM-14936-00	SEAT, SPRING	1	
28	5JW-14589-00	SPRING	1	
29	5SL-14349-00	PIPE	1	
30	5VY-14348-00	PIPE 1	1	
31	8CC-14216-10	SCREW	4	
32	5PW-14103-00	THROTTLE SCREW SET	1	
33	5VY-13761-00	INJECTOR ASSY	4	
34	5PS-14147-00	O-RING	1	
35	90430-09002	GASKET	4	
36	5JW-14104-00	AIR SCREW SET	2	
37	5VY-14905-00	LEVER, STARTER	1	
38	98507-06020	SCREW, PAN HEAD	2	
39	5VY-13972-00	PIPE, FUEL 2	1	
40	90467-120A1	CLIP	1	
41	5VY-13543-00	HOSE, AIR	2	
42	90464-30040	CLAMP	1	
43	5VY-13971-00	PIPE, FUEL 1	1	
44	90464-16005	CLAMP	1	
45	90467-12003	CLIP	1	
46	5VY-14413-00	STAY	1	

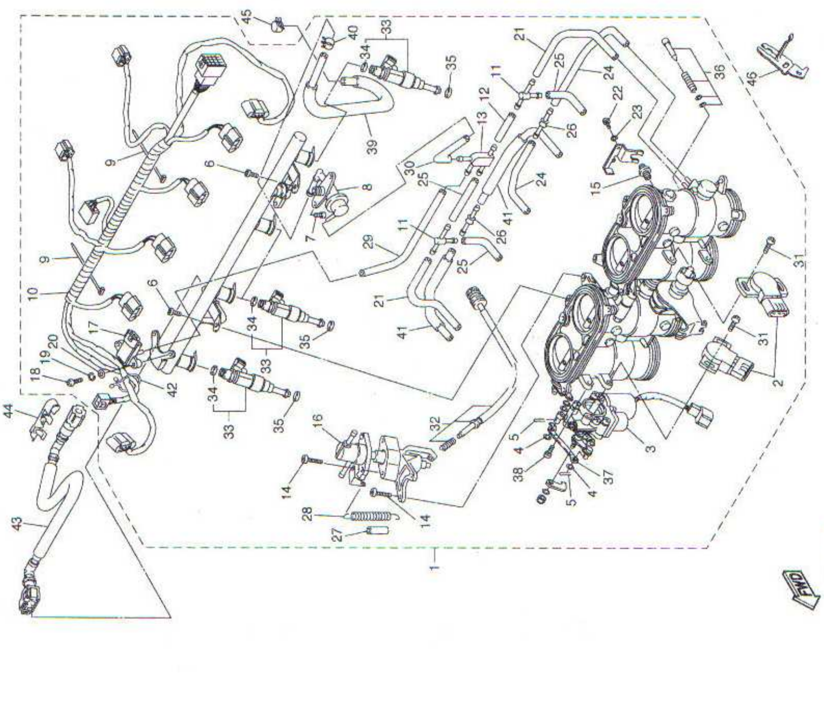

Figure B.4: Further intake system components required for YZF-R1 EFI system

(Source: Toowoomba Yamaha 2005)

Appendix C: 1997 Yamaha YZF600R Performance Data

C

The following two graphs show power and torque curves for a 1997 Yamaha YZF600R. This data was used as a comparison for the benchmark performance characteristics achieved for the 1994 Yamaha YZF600 through dynamometer testing in this project.

● **Power Curve**

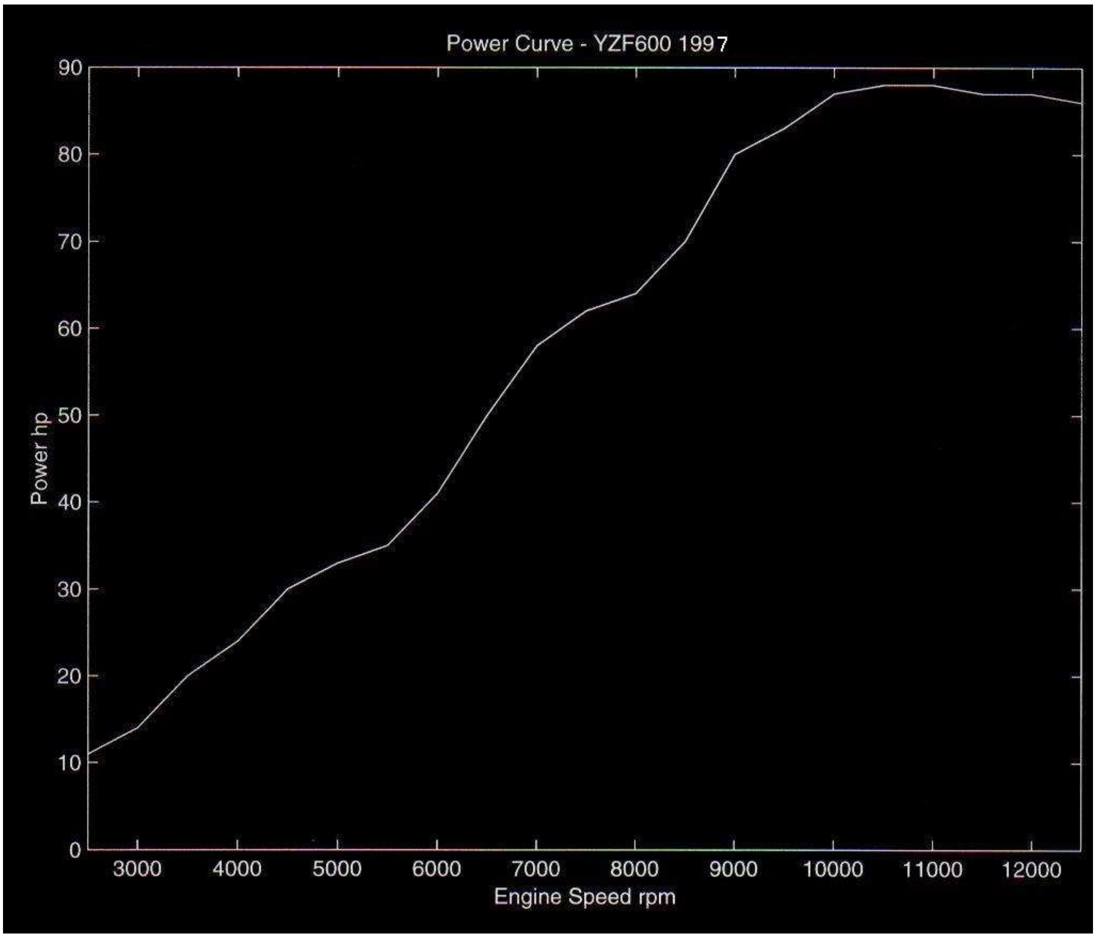

Figure C.1: Power curve for a 1997 Yamaha YZF600R
(Source: www.geocities.com 2005)

● **Torque Curve**

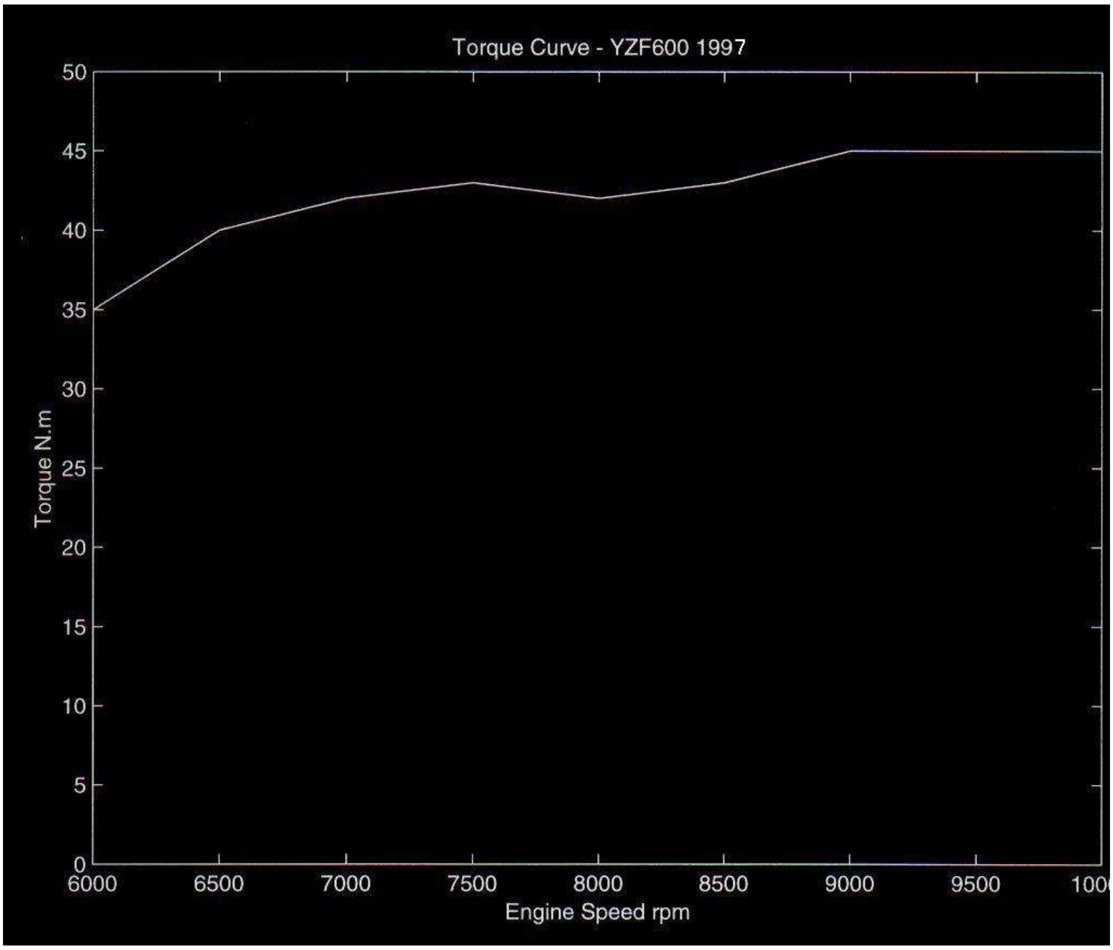

Figure C.2: Torque curve for a 1997 Yamaha YZF600R
(Source: www.geocities.com 2005)

Appendix D: Engine Testing Results

D

The following table presents data obtained from dynamometer testing to determine benchmark performance characteristics.

BIKE REVS (rpm)	ENGINE SPEED (rpm)	BRAKE SPEED (rpm)	TORQUE READING	POWER (kW)	POWER (hp)
6000	2439	2400	4.8	21.5	28.7
6500	2642	2700	5.2	26.2	34.9
7000	2846	2850	5.4	28.7	38.3
7500	3049	3100	5.3	30.7	40.9
8000	3252	3280	5.1	31.2	41.6
8500	3455	3550	5.0	33.1	44.2
9000	3659				
9500	3862	3750	5.1	35.7	47.6
10000	4065				

(Note: Blank fields indicate no testing was performed at these speeds).

Table D.1: Dynamometer testing data for 1994 YZF600 in standard bike trim (benchmark performance characteristics)