Level 4 Project

STEADY STATE RC ENGINE DYNAMOMETER

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Final Report
Date: 30/03/2007

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Executive Summary

This report documents the design and development of a dynamometer that is to be used with radio controlled nitro engines. The aim of this project was to design and manufacture a dynamometer that is capable of measuring the power and torque output of miniature nitro engines and to enable an accurate way of tuning the engines in a steady state manor.

Dynamometers have been designed in the past to be used with radio controlled engines but after investigation, it was found that the prior work does not adequately accomplish the aims of this project. Allowing the operator to tune engines to achieve maximum power and or torque in a steady state manor is a highly desirable property in dynamometers and is a feature that is not found in previous designs. The term steady state tuning indicates that an engine is allowed to run at constant operating conditions under a certain load while the engine parameters are changed to alter engine output characteristics. Previously built dynamometers used a dynamic approach to test the total output power of an engine. When using a dynamic dynamometer the power of an engine was often calculated by attaching an engine to a flywheel with known properties which is then accelerated. While dynamic dynamometers do aid in the tuning of an engine they do not give a real time outputs from the engine which is desired to enable optimised tuning.

The design of this dynamometer utilised information gained from a literature review of prior dynamometers designed for radio controlled nitro engines and also based several aspects of the design on the benchmarking of a steady state dynamometer for DC electric motors. Analysing the prior work in this field gave an insight into the requirements for a dynamometer designed for RC engines.

This project achieved in designing and building a dynamometer that is capable of tuning an engine in steady state to achieve desired output properties by utilising real time data obtained from a data acquisition system.
Disclaimer

I declare that all work contained within this document is my own, other than where it is stated to be of another author.

Heath Miller

Lewis Hewton
Acknowledgements

We would like to thank our supervisor Dr Ben Cazzolato for his support and assistance over the period of this project. We would also like to thank Bob Jarrad from Lathe-Weld Techniques and all of the workshop staff, especially George and Steve for assisting with the construction of the dynamometer.
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Glossary

cc  Cubic centimetre

DC dynamometer  *Refers to the DC dynamometer located at the School of Mechanical Engineering, University of Adelaide*

DOF  *Degree of freedom*

FEA  *Finite element analysis*

kW  *Kilowatts*

MDOF  *Multi degree of freedom*

RC  *Remote/radio controlled*

rpm  *Revolutions per minute*

rps  *Revolutions per second*

SDI  *Siskyou Design Instruments*

W  *Watts*
List of Variables

\( \delta L = \text{Change in gauge length} \)

\( \delta R = \text{Change in gauge resistance} \)

\( \varepsilon = \text{Strain} \)

\( v = \text{Poisson’s ratio} \)

\( \rho = \text{Density} \)

\( \sigma = \text{Stress} \)

\( \tau = \text{Torque} \)

\( \omega = \text{Rotational speed} \)

\( \omega_{\text{ne}} = \text{Natural frequency} \)

\( A = \text{Cross sectional area over which a force is applied} \)

\( E = \text{Young’s modulus} \)

\( F = \text{Force} \)

\( k = \text{Strain gauge factor} \)

\( r = \text{Radius} \)

\( l = \text{Length of lever arm} \)

\( L = \text{Gauge length} \)

\( \dot{Q}_{\text{total}} = \text{Heat dissipation} \)

\( R = \text{Original gauge resistance (without strain)} \)

\( R_{\text{cr}} = \text{Reynolds number} \)
1 Introduction

A dynamometer is a piece of equipment designed to measure the output power, speed and torque from an engine. Dynamometers are commonly used in the production, tuning and maintenance of automobiles and automobile engines. While full scale dynamometers are a well researched, documented and implemented area of study, there has been very little investigation into the area of miniature scale dynamometers to date. This project investigates the design and build of a dynamometer to accommodate miniature nitro powered, radio control style engines.

The purpose of this project is to develop a dynamometer that is able to be used on miniature nitro powered engines, such as those used in model cars, boats and airplanes. Generally the tuning process of these engines involves a process of trial and error which is inaccurate and time consuming. Producing a dynamometer for this application will reduce the amount of time required to tune these engines while increasing the accuracy of measurements of engine characteristics, such as power and torque output. The ability to accurately monitor the power and torque outputs of an engine during the tuning process allows maximum operating characteristics of the engine to be achieved.

Accuracy of the results gained from the use of the dynamometer manufactured in this project is crucial. The results must be reliable to ensure that the dynamometer is working correctly, and that the results are a true indication of the performance of the engine. The miniature nitro engines that are to be tested are inherently imbalanced and all attempts should be made to minimize the disturbance that the vibration creates while the engine is running. The vibration, if not appropriately dealt with will affect the quality of the results and degrade the efficiency and functionality of the dynamometer.

There are two distinct types of dynamometer, these being steady state and dynamic, however designing a steady state dynamometer is more desirable than a dynamic type. Steady state dynamometers are more desirable as they allow for engines to be tuned in real time, under constant loading conditions and also at a variety of engines speeds. Being
able to tune in real time is an advantage as the change in engine output characteristics due to a change in engine settings can be seen instantaneously allowing for a more accurate tuning process. Being able to tune at a variety of speeds is an advantage as it allows for the performance of an engine to be optimised at a range of running conditions. Dynamic dynamometers are generally operated by connecting the driveshaft of an engine to a flywheel with known properties. The output power and torque of the engine is then calculated while the engine is accelerated up to full speed. A disadvantage of this type of tuning is that engine characteristics can only be determined while the engine is accelerating. The advantages of a steady-state style dynamometer over a dynamic style dynamometer indicated that it would be valuable to create a steady state dynamometer that can determine engine characteristics at constant speed to assist in the tuning process.

Identification of the necessary sub systems to produce a RC dynamometer will be undertaken as part of the project through a literature review and benchmarking. Current dynamometer designs intended for RC engines will be analysed in the literature review. Benchmarking of a steady state dynamometer used for testing of a DC electric motor will also be used to determine the sub systems needed to produce a dynamometer capable of tuning RC engines in a steady state manor.

Because the costs involved in designing a new steady state dynamometer will be considerably large, it is desirable that the design should accommodate a wide range of engine types and sizes to ensure that it has maximum usability.

By identifying and completing the necessary design requirements, a steady state dynamometer capable of producing accurate results in real time may be produced.
2 Background

To give a better understanding of dynamometer principles and current dynamometer designs, background research into the topic was required. An investigation into the previous work on RC dynamometers was conducted through the process of a literature review and benchmarking. The principles which are used by the different types of dynamometers to determine the power and torque outputs from engines was also determined using the information from the background research.

2.1 Literature Review

RC dynamometers are a relatively undeveloped application of measuring devices. Because the application of RC dynamometers is relatively undeveloped, there is very little published literature regarding the subject and virtually no information on steady state style dynamometers is available. There is however, much more non-refereed information about inertial/dynamic style dynamometer systems available from sources such as the internet that were analysed to determine technologies and processes that are used in both research and current market place designs. The information gained from the literature review may be used to aid in the process of designing and building of the steady state dynamometer.

MWD and Associates Dynamometer

Figure 1 shows an engine mounted dynamometer designed by Brian Callahan in 1997 at the University of Michigan. The dynamometer is designed to measure the torque and power output from small, racing model marine engines. Because of the start-stop nature of remote control boat racing an inertia dynamometer was used to simulate the constantly accelerating/decelerating nature of the running conditions of such an engine. Since the intention of the dynamometer was to only measure the total power output of engines the use of an inertial style dynamometer was acceptable.
The MWD and Associates dynamometer has the facility to use an electric starter as well as a pull start which is very useful. The hemi-spherical lug, which can be seen in Figure 1 at the end of the end of the inertial disc, is used to attach a starter motor for engines that do not have a pull start. A facility for an electric starter would be desirable when designing the RC dynamometer as it would allow for compatibility with the widest possible range of engines.

The design of the MWD and Associates dynamometer does not appear to adequately allow for a variety of engine sizes due to its minimal adjustability in any direction. The lack of adjustability is primary due to the focus of the design being for RC boat engines and is not designed to be used with other engine types. The engine mounting system used does not make use of the mounting holes of the motor, but rather uses a clamp system to hold the engine supports. A simple slide arrangement is used to adjust the distance in the axial direction using bolts that can be loosened to allow movement, and then tightened to lock the assembly in place. While the RC dynamometer to be designed in this project is
to be designed for use with a wide variety of engine sizes, there is a possibility that a similar clamping style engine mounting system could be used.

Figure 1 shows that the construction of the MWD and Associates dynamometer uses components that are quite large and ‘bulky’ presumably to reduce the effect that excessive vibration has on the system. The size of the construction shows that structural rigidity should be a consideration in the design process of this project.

**Marc Decroubele Dynamometer**

This dynamometer produced Marc Decroubele was based conceptually on the design of Brian Callahan but offers several improvements including a water cooling system and data acquisition microprocessor (Figure 2).

The Marc Decroubele dynamometer allows interchangeable engine mounts to allow for 0.5 cubic centimetre (cc), 7.5cc and 15 cc motors and has different sized inertia discs to accommodate the different sized engines. The dynamometer is manufactured from aluminium and is mounted on a practical workbench. The dynamometer does not allow for a variety of engine arrangements, as there is no way to accommodate an engine whose tuned-pipe extends from the rear side of the engine (the opposite side to the driveshaft). The design utilizes a one way centrifugal clutch in the drive train to transfer power from the engine to the inertia disc. The clutch makes sure that power is transferred only from the engine to the inertia disc and not vice-versa. This ensures that when the engine reaches its maximum power and is turned off, the disc will not transfer energy back into the engine.

The dynamometer measures and displays the rotational velocity of the motor and of the flywheel, the temperature readings are taken and displayed for the motor, water-in, water-out, tuned pipe, exhaust gas and ambient temperature. Humidity and pressure of the surroundings are also measured. The measurement of ambient temperature, humidity and pressure are used to normalise the readings given by the dynamometer to constant
temperature and pressure to ensure that the dynamometer gives constant readings that are not dependent on the testing conditions.

![Image of dynamometer](image.png)

**Figure 2 - Inertia dynamometer (modelboatracing.co.uk 2005)**

It is desired to use similar electronic systems to what are used in the Marc Decroubele dynamometer for data acquisition. This is because electronic recording of data allows for electronic read-outs of torque, power and engine speed making the dynamometer more commercially viable and user friendly. The dynamometer designed by Marc Decroubele uses microprocessor and servo control on the throttle which increases the efficiency and speed of obtaining results. An arrangement capable of running engines electronically could be used on a brake dynamometer by applying similar systems to apply brake and throttle application.
DSR 810b Dynamometer

![DSR 810b Dynamometer](rcdynometersystems.com)

Figure 3 show the DSR 810b dynamometer. This is a commercially available chassis mounted inertia dynamometer that retails from US$1295 to US$1849. It features interchangeable inertia disc sizes to accommodate different engine sizes and comes complete with all software required to use the dynamometer on a home computer. The dynamometer comes in two types – roller or belt style, the belt style being the more expensive of the two options.

The DSR 810b dynamometer clearly does not accommodate for other types of RC vehicles, for example, boats and planes, hence is restricted for use with certain scale cars. The company claims that the dynamometer gives the ability to adjust the engine “on the fly” under “actual running conditions”. While this is true, the dynamometer is not able to display the changes made to the power or torque in a steady state manor as it is an inertia type dynamometer.

The principles and design features of this dynamometer are not of much use to the design of the steady state RC dynamometer. However, the DSR 810b is one of the few
commercially available dynamometers and has given an insight into the price limit that must be met to make the steady state dynamometer commercially feasible.

2.2 Benchmarking

Benchmarking was carried out on a physical dynamometer, allowing for an in-depth analysis of the sub systems required to produce a functioning, miniature scale dynamometer.

**DC Dynamometer – University of Adelaide**

The DC dynamometer was built at The University of Adelaide for use with DC electric motors (Figure 4). Several aspects of the design of the DC dynamometer can be incorporated into the design of the RC dynamometer. The DC dynamometer works on the principal of a brake dynamometer as opposed to an inertia dynamometer such as the dynamometers discussed in the literature review. An explanation of the principles used to determine output characteristics of an engine using a brake dynamometer are explained in Section 2.3.

![Figure 4 - Brake dynamometer located at School of Mechanical Engineering, University of Adelaide](image)
The physical design of the DC dynamometer uses a magnetic brake assembly, comprised of a metallic disc and rare-earth magnet taken from the hard drive of a computer, to apply a torque to the DC motor. The torque is transferred through a torque arm into a strain gauge which is used to measure the torque applied by the brake. The angular velocity is measured by an encoder which is attached to the disc assembly. The design features a self centring mounting system that centres the driveshaft of the motor with the discs using a double threaded bar. As DC electric motors are circular in shape, the mounting system uses triangular sliding clamps which run on a dual threaded bar to centre the motor as seen in Figure 5.

![Figure 5 - Self centring DC engine mounts](image)

Using a torque arm attached to a strain gauge system is a very simple idea which can be incorporated into the design of the RC dynamometer. A double threaded bar which allows for self centring through a single adjustment is very useful as well as the idea of using magnetic brakes as they provide a constant braking force and will not wear, deform or fatigue.
The dynamometer is made from machined aluminium and has been designed to be a fairly rigid structure, which helps reduce vibrational problems. Because there are greater vibrations produced by RC engines compared to DC motors due to their higher running speeds and inherent imbalance, the structural rigidity of the DC dynamometer should be kept in mind and where possible improved in the design of the RC dynamometer.

The outputs from the DC dynamometer are analysed using computer software. The raw data from the DC dynamometer which is output from the strain gauge and encoder is first amplified and then input into a computer via a dSPACE controller. A dSPACE controller is an interface between the physical system and software. The information from the controller is converted into meaningful readings of torque and speed using the Simulink toolbox of the program Matlab. The meaningful readings from Simulink are then analysed using the dSPACE Control Desk program, which provides a user friendly environment for analysis of data in real time. Because the University of Adelaide has made Matlab and dSPACE facilities available for use, using a similar data acquisition process to that which is used in the DC dynamometer is highly desirable for the RC dynamometer project as it would avoid the creation of a new software environment.

Benchmarking of the DC dynamometer has been a very useful starting point for design aspects of the RC dynamometer. The DC dynamometer has the ability to analyse data in real time and provides the ability to see results in steady state. The analysis of a steady state dynamometer, which was the only one found during the research process, has given insights into the way that torque and speed can be measured using sensors and then effectively analysed in real time.

2.3 Background Principles

There are two distinct types of dynamometer, they are inertial and brake dynamometers. While both types are able to measure/calculate power and torque, the method in which they gain the information is quite different. Each type of dynamometer can be equally as
accurate but the different types of dynamometer are more applicable for measuring different situations.

A brake dynamometer works on the concept of applying a brake to the engine, or a disc connected to the driveshaft of the engine, and measuring the torque and angular velocity that is produced when that brake is applied. The governing equation of a brake dynamometer shown in Equation 1.

\[
\text{Power} = \text{Torque} \times \text{Rotational Velocity} \tag{1}
\]

Equation 1 is used to convert a measured torque into a power which can then be plotted against the running speed of the engine to give a good representation of an engine’s performance over a range of engine speeds.

The brake dynamometer system allows the operator to apply a constant torque on the engine while it is running. The application of a constant torque allows steady state tuning of the engine because the change in power and torque resulting from changes in engine settings, such as fuel-air ratio, can be instantaneously measured. An instantaneous readout of power and or torque allows engine parameters such as air-fuel ratio as well as high and low end mixtures to be changed to maximise the performance of the engine. For this reason, a brake style dynamometer is preferable for tuning an engine at constant operating conditions.

An inertial dynamometer works by using an engine to accelerate a mass of known inertia and measuring the acceleration of the mass. The mass that is accelerated is usually a disc attached to the output shaft of an engine. The governing equation of an inertia dynamometer is shown in Equation 2 and the equation used to determine the moment of inertia about the central axis of a cylinder is shown in Equation 3.

\[
\text{Power} = \text{Moment of Inertia} \times \text{Angular Acceleration} \tag{2}
\]

\[
\text{Moment of Inertia} = \text{Mass} \times (\text{Radius})^2 \tag{3}
\]
Equation 2 is used to calculate the output power from the engine which can also be converted into torque using Equation 1. Since the measurement of torque is dependent on the angular acceleration of the mass, when the engine is at constant rpm (not accelerating) the power cannot be calculated and hence inertial dynamometers can not be used for real time tuning. Because of the inability to determine output characteristics at constant speed, the process of tuning an RC engine using an inertial dynamometer must be conducted over series of repeated runs with the changes made to engine settings being made between runs. While this style of dynamometer is less appealing for tuning, it is useful for measuring power in applications which do not generally run at steady rpm and are constantly accelerating. An advantage of the inertial method of measuring power is that the dynamometers do not require calibration as the moment of inertia of the disc can not be changed because its mass and size is always constant (dynamometer.fsnet.co.uk 2005).
3 Specifications

The specifications that the RC dynamometer must be built to were determined from the literature review and benchmarking processes. The following section details the integral components of a RC brake dynamometer and explains their importance.

**Torque Application Device (Brakes)**
A design parameter of this dynamometer is that it allows for steady state tuning of engines, requiring a brake style dynamometer. To achieve the required braking force, a braking system will be used that creates torque on the driveshaft of the engine. The reactionary torque from the engine will be measured by sensors to determine the torque and power outputs from the engine. The torque applied by the brakes must be variable to give a range of results, and the brakes must be able to apply a torque greater than the maximum torque generated by the engine.

The brakes used by the dynamometer must allow for effective heat dissipation. This is to ensure that the brakes will not overheat and warp or lose their braking capacity. Engine heat must also be accounted for, as failing to design for heat generated during operation of the engine may cause the performance of the engine to become impeded and hence give the impression that the dynamometer is giving inaccurate results.

**Engine Mounting System**
An important feature of the dynamometer design is that it must accommodate for a wide variety of engine sizes. RC engines are usually mounted with bolts through mounting plates which are located on both sides of the engine. It is desired to use these mounting plates to attach the engine to the dynamometer.

**Shaft Coupling Device**
A shaft coupling device is required that will physically couple the output shaft of an engine to the dynamometer. As the dimensions of the driveshaft for each engine will
varya, a linkage system is required that will couple varying drive shaft sizes to the brake system.

**X,Y,Z Adjustment**

Because there is a large amount of variation in engine sizes the position of an engine needs to be adjustable in all three axes. Three-dimensional adjustment allows for the driveshaft of a variety of engines to be physically coupled with the brake disc shaft on the dynamometer. X,Y,Z adjustment also allows for the driveshaft to be centred with the disc brake for effective energy transmission and to minimise vibrational effects.

Throughout this document the coordinate system shown in Figure 6 will be used to designate the X-Y-Z axes of the dynamometer. This is to clarify in which direction the displacement is occurring.

![Diagram of Dynamometer Coordinate System](image)

*Figure 6 - Dynamometer coordinate system*
Sensors
Sensors are required on the dynamometer to measure the torque and speed of the engine being tuned. As the system is assumed to be at steady state, the input to the system is equal to the output. The consequence of this assumption is that the velocity of the brake discs is equal to that of the driveshaft and the torque that is applied by the brakes is equal to that which is created by the engine. Because of this assumption, sensors will only be required for measurement of either the input or output to the system to determine the state of the system.

Vibration Isolation
The dynamometer must be able to effectively handle the vibrational strains put on it by a two-stroke nitro engine. If the dynamometer is unable to cope with the large vibration, it may lead to inaccurate results and may eventually produce fatigue induced failure of the dynamometer. The components of the dynamometer will be designed so that they are as rigid as possible and features with large resonant responses such as cantilever beams will be avoided wherever possible. Testing will be conducted to ensure that the operating speeds of the engines do not couple with resonant frequencies of the dynamometer. Filters in the analysis software can be used to reduce the effects of vibrations in the output data, while damping material can also be added if the levels of vibrations experienced during operation produce unacceptable results.
3.1 Technical Specifications

After reviewing the power and operating speed characteristics of a number of engines supplied by OS Engines (os-engines.com 2005), values were taken which were seen to represent the likely engine characteristic bounds of a radio controlled test engine. The technical design specifications of the dynamometer are listed in Table 1. It is assumed that the characteristics of virtually all RC engines will fall within these values. The accuracy of the measured torque and speed was decided based on the accuracy of the sensors to be used and their susceptibility to noise.

Table 1 - Dynamometer engine characteristics

<table>
<thead>
<tr>
<th>Dynamometer characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>2000-50000 rpm</td>
</tr>
<tr>
<td>Maximum torque</td>
<td>1 Nm</td>
</tr>
<tr>
<td>Maximum power</td>
<td>3.73 kW (5 HP)</td>
</tr>
<tr>
<td>Speed and torque accuracy</td>
<td>±5%</td>
</tr>
</tbody>
</table>
4 Design Options to specifications

After considering the design specifications, several possible concepts were devised to satisfy the design parameters. This aided in the process of designing a conceptual dynamometer.

4.1 X,Y,Z Displacement Adjustment Systems

There are several commercially available X,Y,Z adjustment systems which could be used to adjust the position of an engine shaft so that it is centrally aligned with the brake shaft of the dynamometer. Newport Corporation offers a variety of multi-degree of freedom (MDOF) stages (similar to that shown in Figure 7) which provides displacement adjustment along all three axes. An engine mounting system could be attached to one of these systems allowing for a variety of motors to be coupled to the braking system through adjustment of the positioning handles. The Newport Corporation website (2005) has a range of stages that are made for the optics industry and are extremely accurate, some with rigidity providing angular misalignment of better than 100µrad for each axis of rotation. An issue with the MDOF stages is that their range of motion is limited because they are designed to provide very fine adjustments. Because of their expensive price (approx US$1500 (newport.com 2005)) and limited adjustment, a MDOF system will not be used for the dynamometer application despite providing exceptional dimensional accuracy.

![Figure 7 - Multi-DOF adjuster (newport.com 2005)](image-url)
4.2 Individual X, Y and Z Displacement Adjustment Systems

An alternative to using a single MDOF stager to adjust the position of an engine so that it can effectively be coupled to the braking system of the dynamometer was required. It was decided to investigate the use of smaller sub systems to achieve the same amount of directional freedom as a MDOF stager. Components from the optics industry were considered as options for the X,Y and Z displacement systems because they are designed with rigidity as a high priority. These components are designed with high rigidity to minimise problems associated with vibration and so are well suited to the dynamometer application.

4.2.1 Linear Stages

A cheaper option than using a single device to adjust an engine along all three axes of displacement is to use a series of linear stages that are capable of adjusting displacement along each axis separately. Siskyou Design Instruments (SDI) supplies this type of adjuster (Figure 8). This type of adjuster would be useful in the X- and Y- directions, providing a 1” (25.4mm) range of motion. Linear stages are available from SDI for approx US$115.

![Figure 8 - Typical linear adjusters (sd-instruments.com 2005)](image-url)
4.2.2 Rails

Using a rail system is a simple idea to provide positional adjustment of an engine in the X- and Y- directions. SDI provide dovetail rails (MDR rail series) that are precision machined for good rigidity allowing for easy attachment of rail carriers (also produced by SDI) which could be used to couple engine mounts to the rails (Figure 9). The rails are available in 6” (150mm) and 12” (300mm) lengths. The length of these rails provides a wider range of adjustment in comparison to the linear and MDOF stages. The prices of the rails are listed in Table 2.

<table>
<thead>
<tr>
<th>Model</th>
<th>Description</th>
<th>Price (US$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MDR-6.0i</td>
<td>150mm rail</td>
<td>32</td>
</tr>
<tr>
<td>MDR-12.0i</td>
<td>300mm rail</td>
<td>60</td>
</tr>
</tbody>
</table>

Figure 9 - Dovetail rails (sd-instruments.com 2005)
4.2.3 Rail Carriers

Rail carriers can be used to provide a good coupling interface between the X-, Y-displacement systems and the engine mounting system. Rail carriers are used in the optics industry so that instruments can be placed onto rails. SDI provides several types of rail carriers (Figure 10) but all use a design that allows for accurate sliding along the MDR rails;

“These one-piece aluminium RC rail carriers present a low profile and allow close spacing between components. Spring loaded, hard polymer guides along the mating surface of the RC rail carriers ensure wobble-free sliding motion and dependable linear alignment along the MDR micro-dovetail rails.” (sd-instruments.com 2005)

This statement indicates that the rail carriers will provide a high level of stability for use in the RC dynamometer. The rails carriers are secured to the MDR rails using a clamp screw allowing for easy positional adjustment. Rail carriers are available from SDI for US$20.

4.2.4 Translating Rod Holder

Translating rod holders can be used to provide vertical (Z direction) alignment of the brake and engine shafts by adjusting the amount of rod that protrudes from the base of the holder (Figure 11). Spring-loaded thumbscrews are used to secure the rod in position
which, while allowing for quick adjustment of position do not have a fine adjustment option. The rod holders also use a relief cut which creates a three point loading when the thumbscrew is tightened to increase the rigidity of the connection between the rod and holder. The SDI listed price for the holders and rods are from US$15 to US$50 depending on size.

![Rod holders and rods](sd-instruments.com 2005)

**4.2.5 Adjustable Rod Holders**

Adjustable rod holders (Figure 12) provide vertical adjustment in the same way as translating rod holders but have the added advantage of 12.5mm fine vertical adjustment using an adjuster ring. The fine adjustment option promotes high accuracy when aligning sub systems, which will reduce vibrational effects by reducing the degree of misalignment between the systems. SDI produces adjustable rod holders in two sizes listed in Table 3.
Table 3 - Adjustable rod holder specifications

<table>
<thead>
<tr>
<th>Model</th>
<th>Minimum length (mm)</th>
<th>Maximum length (mm)</th>
<th>SDI listed price (US$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ARH-3.0</td>
<td>76.2</td>
<td>88.9</td>
<td>40.0</td>
</tr>
<tr>
<td>ARH-4.0</td>
<td>101.6</td>
<td>114.3</td>
<td>42.0</td>
</tr>
</tbody>
</table>

Figure 12 - Adjustable rod holders (www.sd-instruments.com 2005)

4.2.6 Four Posted Slide

As an alternative to purchasing a commercially available Z-displacement system, a four posted slide was designed. It was decided that it would be simpler to adjust the height of the braking system as opposed to the height of the engine when coupling the two systems. The assembly of the four posted slide includes a horizontal plate which the disc and brake assembly is connected to, with four slotted posts to guide the plate vertically (Figure 13). The design allows the driveshaft of the engine to be vertically aligned with the shaft of the discs to give efficient transmission of power. The slotted posts restrict the plate to only move vertically and restrict twisting in the direction of movement of the discs.
Translation in the Z-direction would be achieved by loosening the 4 locking nuts, moving the horizontal plate to the required vertical position in line with the engines driveshaft and re-tightening the locking nuts.

The assembly would be manufactured from aluminium and the degree of tolerance required would be extremely important in order to make accurate adjustments to the height of the system while maintaining structural rigidity possible.

![Figure 13 - Four post Z-direction assembly](image)

4.3 Measurement Systems

As described in the specification section (Section 3), measurement of either the input or output is required to determine the state of the dynamometer. For simplicity, it is desired to measure the output of the system (speed of brake discs and torque generated by the engine).
4.3.1 Velocity Measurement

In order to calculate the power of the engine, the speed of the driveshaft must be known and hence some form of velocity measurement is required.

**DC Tachometer**

Driveshaft speed could be measured using a DC tachometer, which generates a magnetic field using a permanent magnet while a coil is rotated between the poles. Slip rings are used to allow the signal to be recorded. An advantage of this set up is that induced voltage is proportional to velocity. The accuracy of DC tachometers can easily be affected by noise due to imperfections in electrical contact at the slip rings (Turner and Hill 1999).

**Hall Sensor**

A Hall Effect sensor uses a probe which senses the change in reluctance past a sensor (Turner and Hill 1999). The required change in reluctance can be created using a small magnet. When a change in reluctance is created, in this case due to a magnet passing the sensor, a voltage peak would be recorded. To derive the speed of the shaft, the time between each peak of the output would be measured and then converted into a rotational speed.

**Encoder**

An optical encoder can be used to measure shaft speed with exceptional accuracy. Encoders generate a specific amount of pulses per each revolution depending on their bit rate (i.e. 256, 512, 1024). If the time between each pulse is known, speed can be determined at many points per revolution. Because the speed of the engines to be used in the DC dynamometer is only required once per revolution the use of an encoder is not necessary.
4.3.2 Torque Measurement

The dynamometer must be designed so that all of the torque generated by an engine can be effectively measured, for this reason a torque measurement system was required.

Torque Arm

Benchmarking revealed that the DC dynamometer used by The University of Adelaide uses a torque arm coupled to a strain gauge system to measure the torque output of the engine (see Figure 14). This set up works well and a modified version can be used for the RC engine dynamometer. The engine (through its mounting system) is attached to a torque arm, which attempts to rotate about its swivel points (bearing supports) when a torque is applied. The torque generated is applied through a lever arm at the end of the torque arm and essentially tries to “push down” or “pull up” on the strain measuring system generating a measurable response.

4.3.3 Strain Gauges

The output torque from the engines tested can be measured using strain gauges (Figure 14). Typical strain gauges contain thin metal foil which is attached to the surface to be measured. When the surface elongates or contracts, the foil fluctuates accordingly, which changes its electrical resistance and hence, alters output voltage from the strain gauge. The change in electrical resistance is related to the strain by a gauge factor $k$ (Equation 4).
Strain gauge factor, \( k \):

\[
k = \frac{\delta R}{R} = \frac{\delta R}{\varepsilon R} = \frac{\delta R}{\varepsilon R} = \frac{\delta R}{L} = \frac{\delta L}{\varepsilon R}
\]  

(4)

where,

\( \varepsilon \) = strain

\( R \) = original gauge resistance (without strain)

\( \delta R \) = change in gauge resistance

\( L \) = gauge length

\( \delta L \) = change in gauge length.

The gauge factor is usually about 2 for foil gauges and the exact value is supplied by the manufacturer (Turner and Hill 1999).
As the amount of deformation will be very small, only elastic deformation of the strain gauges will occur, hence the stress, $\sigma$, can be calculated using Hooke’s law:

$$\sigma = \varepsilon E$$

where $E = \text{Young’s modulus (aluminium)}$.

The stress can then be converted into force $F$ using:

$$F = \sigma A$$

where $A = \text{cross sectional area over which the force is applied}$.

Finally, the torque $T$ produced by the engine can be found using:

$$T = Fl$$

where $l = \text{length of lever arm}$.

The gain factor which directly relates output voltage from the strain gauges to output torque can also be determined through calibration tests. By hanging known weights from the lever arm, the force generated in the strain gauges can be calculated using Equation 7 and the gain factor can then be determined using Equation 8. Equation 7 shows that the only variable when determining the torque is force (as the length is fixed) and the relationship is linear, hence the gain factor determined from calibration tests will be applicable for all conditions.

$$\text{Torque} = \text{Output Voltage} \times \text{Gain factor}$$
4.4 Engine Mounting System

To achieve a mounting system that is capable of adjusting to a variety of engine sizes a form of clamping or sliding adjustment is required.

4.4.1 Sliding Engine Mount

The sliding engine mount design uses two sliding halves which can be brought together until the mounting plates of the engine sit on the mounting faces of the slides (Figure 15). The mount can then be secured by screws in the base, while holes can be drilled on the mounting faces allowing for the engine to be bolted to it. The mounting faces of the unit are designed to be cheap and easy to replace when too many holes have been drilled into the mounting face preventing a new engine size from being used. A problem with these sliding engine mounts is that they are not self-centring which means a large amount of adjustment would be necessary in the Y-direction to couple the engine to the braking system. The largest downfall of this design is that effectively uses cantilevers which are very bad for vibrational behaviour as they display large displacements at their modal frequencies.

![Figure 15 - Sliding engine mount](image)
4.4.2 Replaceable Mounts

It was decided that the engine mounting system should couple into the Y-axis adjustment system to reduce the amount of adjustment which is required along this axis. As mentioned in Section 2.2, a double threaded bar is used on the University of Adelaide DC dynamometer to provide self centring which can also be used in the design of the RC engine dynamometer. The engine mounts can be designed in two halves connected by a double threaded bar (Figure 16). When the bar is turned, the two halves will move along the shaft in opposing directions, which will adjust the distance between the mounts to what is required for mounting an engine, while at the same time reducing the amount of additional Y-direction adjustment required to align the engine with the braking system.

As the dynamometer is designed to accommodate for a variety of engine sizes the mounts will either need to be replaced or altered with the introduction of each new engine. The proposed mounts contain removable inserts which an engine can be bolted on to (Figure 16). Removable inserts are preferred because the base of the mounts will be machined to enable coupling with the Y-direction displacement system, this would be more expensive and time consuming to replace than a simple insert into the top of the mounts. As the removable inserts are a simple design which will reduce costs if they need to be replaced, they are desired for use in the engine mounting system.
Engine mounts will make use of the adaptor plates on engines

Replaceable inserts

Figure 16 - Adjustable engine mount system

4.5 Brake System

Section 2.2 revealed that the brake used on the DC dynamometer was produced using a disc and rare-earth magnet obtained from a computer hard drive. It is desirable to use the same components on the RC dynamometer as magnetic brakes do not wear down and are capable of producing a constant braking force. The discs from a hard drive are very well balanced which is required to minimise vibrational effects and rare earth magnets are utilised to apply a braking force and hence torque to the system. The magnetic brakes are an advantage as they eliminate physical contact with the actual disc and will not deteriorate. The amount of brake exposed to the disc and hence amount of torque applied to the system is altered via the use of a simple worm gear system.
4.6 Drive Shaft Coupling

A coupling device is required to physically couple the output shaft of an engine to the shaft of the brake system. There are two distinct types of coupling that can be used, which are rigid and flexible. Rigid couplings have the advantage of being very cheap to manufacture, however they do not allow for any misalignment between shafts which can result in damage to the system. A flexible coupling allows for a certain amount of misalignment between shafts without causing any damage to the system and has an added advantage of reducing vibrations transferred between the shafts. For this reason a flexible coupling is more desirable than a rigid coupling.

There were several types of flexible coupling, from several manufacturers considered for use in the dynamometer. The coupling choice must be able to handle the design dynamic torque of 1Nm and operating speed of approximately 50000 rpm while isolating vibrations from the engine to the discs as much as possible. The main coupling types under consideration and their properties were:

Bellows Coupling:
- Uses a metal bellows between two hubs
- Accommodates for parallel, angular and axial misalignment
- Excellent torque transferral between shafts

Jaw Coupling:
- End hubs are separated by a jaw type insert (usually made of polyurethane)
- Provide good dampening of shock loads
- Non-metallic inserts can be used to provide electrical isolation
- Accommodates for parallel, angular and axial misalignment

Beam Coupling:
- machined from a single piece of material (usually aluminium)
- Generally have a greater torque rating than Bellows couplings

Most couplings are available with the option of two different mechanisms of tightening the coupling onto the shaft. These are set screw and clamp screw. The set screw style provides clamping by tightening a grub screw directly onto the shaft, while the clamp style tightens a screw in the coupling, reducing the bore size to provide clamping. A clamp style coupling is preferred as reducing the bore size to provide clamping ensures that the shafts of the brake system and engine will remain centred which is desired.

Possible clamp style, flexible couplings that could have been chosen for use in the dynamometer were evaluated using the part evaluation matrix shown in Table 4. To determine the most suitable coupling, several properties of the couplings were evaluated on a scale from 0-10, with 10 being most desirable. The results of the matrix show that the R+W BKL 003 coupling is the most appropriate choice of shaft coupling. The properties of this coupling are detailed in Section 5.2.

**Table 4 - Part evaluation matrix of possible flexible couplings**

<table>
<thead>
<tr>
<th>Coupling</th>
<th>Coupling Type</th>
<th>Overall Size</th>
<th>Weight</th>
<th>Bore Size</th>
<th>Cost</th>
<th>Rated Speed</th>
<th>Rated Torque</th>
<th>Availability</th>
<th>Total</th>
<th>Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ruland MFC A</td>
<td>Beam</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>5</td>
<td>8</td>
<td>5</td>
<td>46</td>
<td>4</td>
</tr>
<tr>
<td>Ruland MBC</td>
<td>Bellows</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>9</td>
<td>5</td>
<td>49</td>
<td>3</td>
</tr>
<tr>
<td>R+W EK1</td>
<td>Jaw</td>
<td>9</td>
<td>9</td>
<td>8</td>
<td>7</td>
<td>9</td>
<td>8</td>
<td>0</td>
<td>50</td>
<td>2</td>
</tr>
<tr>
<td>R+W BKL 003</td>
<td>Bellows</td>
<td>8</td>
<td>8</td>
<td>8</td>
<td>9</td>
<td>7</td>
<td>8</td>
<td>9</td>
<td>57</td>
<td>1</td>
</tr>
<tr>
<td>Lovejoy Beam</td>
<td>Beam</td>
<td>4</td>
<td>4</td>
<td>2</td>
<td>6</td>
<td>6</td>
<td>6</td>
<td>10</td>
<td>5</td>
<td>5</td>
</tr>
</tbody>
</table>
5 Final design

Following analysis of the possible design options to meet the required specifications, a concept assembly of the dynamometer was produced. After an initial concept assembly was produced, the design was evaluated to determine possible flaws and areas that could be improved, resulting in a revised design of the dynamometer. This iteration process was followed several times until the final design was produced (Figure 17). The design iterations can be seen in Appendix A.

![Figure 17 - Final dynamometer design](image)

The dynamometer has been designed to accommodate for a wide variety of engine sizes and to be capable of measuring the speed, torque and power that is created by an engine.
in real time. The dynamometer operates by applying a torque to an engine using a braking system, the reactionary torque created by the engine causes the torque transferral system to try and rotate against a strain gauge system. The information from the strain gauge system and engine speed are then analysed in real time using a data acquisition system to determine the speed, power and torque output of the engine. An outline of the individual sub systems used in the design of the dynamometer is outlined in Sections 5.1-5.5.

5.1 Brake System

A braking system was designed to create a torque on an engine, allowing for the reactionary torque generated by the engine to be determined by the sensor system. The braking system designed comprises of brake shoes and discs to create the desired torque, hubs to position the discs, a gearing system to alter the amount of brake exposed to the discs and a housing to hold the entire system in place.

5.1.1 Discs and Brakes

Discs and magnetic brakes, acquired from the hard drive of a computer are used in the design to create a torque on the driveshaft of the engine. The torque is generated by rotating the conductive metal discs in the field created by the magnets, creating opposing eddy currents (phylink.com 2005). The kinetic energy of the discs is converted into the eddy currents, which slows the discs and creates a torque. The amount of torque created is varied by adjusting the amount which the discs are exposed to the magnets. As discussed in the literature review (Section 2.1), the hard drive components are preferable as they are extremely well balanced and will not deteriorate because they operate without physical contact.

A specification of the dynamometer is that the brakes used must be capable of providing a braking force capable of stalling the engine from running condition. To achieve this force, 3 brakes and discs are to be used as shown in Figure 18. Three brakes were used to
provide for a reasonable amount of heat dissipation away from the discs when a load is applied (see section 5.1.2) while still providing for a fine resolution of the amount in which the braking load can be adjusted. Being able to stall the engine is an important property as it allows for an engines output to be determined over its entire running range.

5.1.2 Verification of Brake Choice

A heat dissipation analysis was conducted to see if the choice of three brakes was acceptable. An issue with using magnet brakes to provide a torque is that they may get hot enough to realise their Curie temperature, which for rare Earth magnets can be as low as 80°C (wikipedia.org 2005). At this temperature the magnets begin to lose their properties and will not be capable of inducing a suitable amount of torque into the system. For this reason, increasing the number of brakes used was considered. An analysis of the system was conducted to determine the radius of the discs that is needed to
effectively dissipate the heat generated when the dynamometer is operating at its specification limits. The full analysis is shown in Appendix C. The analysis showed that the design of three discs is not capable of effectively dissipating the heat generated at full power away from the magnets meaning that the brakes may reach their Currie temperature if the engine is run at full power for too long. However, as it is unlikely that that the engine will be run at full power for extended periods of time and so it is also unlikely that the Currie temperature of the magnets will be realised. Also, a large margin of safety was included in the calculations. For these reasons it was decided not to increase the amount of discs that the brakes use. However, a thermocouple could be implemented close to the discs so that the dynamometer can be shut down when the temperature of the discs becomes too large.

5.1.3 Modification of Hard Drive Housings

To enable the magnets to be effectively used as brakes, their individual housings were required to be modified. When removed from the hard drives, the housings comprised of two magnets and two plates that were separated using pins located on the plates. The gap between the magnets was approximately 3mm and the separating pins did not allow for a disc to be placed between the magnets. As magnetic field strength is proportional to the air gap between the magnets, it was desired to minimise the gap to have the largest possible braking capability. To reduce the air gap and allow for a disc to be placed between the magnets, the separating pins were removed and replaced by spacers at each end of the housings. The distance between the magnets used by the DC dynamometer was 1.7mm, while the disc used was approximately 1.3mm thick, meaning that there was an air gap of 0.2mm between magnet face to disc face. As the field strength created by the DC dynamometer was relatively strong, it was decided to use a similar air gap in the design of the RC dynamometer. The design air gap used for the RC dynamometer was 0.3mm between magnet face to disc face. As the average disc and magnet widths to be used are 1.2mm and 1.95mm respectively, a spacer width of 5.7mm was required to produce an air gap in the RC dynamometer of 0.3mm. The spacers were designed as simple rings, allowing for a shaft to be passed through them. The shafts are used to secure
the housings in place and act as an attachment to the gearing system. Modification of the housings has allowed for brakes to be produced that are capable of providing the desired amount of torque onto the engine.

5.1.4 Worm Gear Set Up

To adjust the amount of braking force applied to the system, a worm gear arrangement was used. The system uses a shaft which runs through the rear of the three individual brake housings, attached to the end of the shaft is a gear which is then attached to a worm gear. The worm gear is then turned to adjust the amount of brake that is exposed to the discs. The worm gear system and the three individual housings are held in place by posts located on the top plate of the brake housing.

5.1.5 Shaft Size

An analysis was conducted to determine the required thickness of the brake disc shaft. This was required to make sure that the stress induced on the shaft from the application of torque through the brakes does not plastically deform the shaft.

The analysis process that was used to determine required shaft size is shown in Appendix D. An assumption made for the analysis was that the input torque is 1Nm, which is equal to the specification limit that the dynamometer is designed to handle.

Possible materials for use as an axle included aluminium, steel and stainless steel. Yield stresses for these materials were taken from ANSYS and listed in Table 5. Using the material properties listed in Appendix E and solving for in the minimum required shaft radius produced the results listed in Table 5.
<table>
<thead>
<tr>
<th>Material</th>
<th>Minimum radius (mm)</th>
<th>Minimum diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>1.314</td>
<td>2.628</td>
</tr>
<tr>
<td>Steel</td>
<td>1.366</td>
<td>2.732</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>1.366</td>
<td>2.732</td>
</tr>
</tbody>
</table>

As can be seen from Table 5, there is not much variation between minimum shaft sizes for the three different materials. The results suggest that the minimum brake disc shaft size is approximately 2.7mm to resist 1N.m of torque without plastically deforming. Allowing for safety of calculations it was decided that a 5mm shaft should be used. The material chosen for the design was steel due to its high availability.

### 5.1.6 Bearing Choice

The shaft of the brake disc will be constrained using precision bearings located in the brake housing. A consequence of a 5mm diameter shaft choice is that a 5mm internal diameter bearing was required. The main constraints of bearing choice were that they must be capable of withstanding high rotational speeds and minimal size was also desired so that the bearings could easily mount into the brake housing. Many bearings are capable of fitting these requirements, however it was chosen to use the MR105A-ZZMC3 Flanged Bearings supplied by MBA bearings. These ball bearings are 5mm inner diameter with a 10mm outer diameter and are 4mm in width. The bearings are made from chromed steel and are capable of withstanding speeds of up to 61,000rpm, MBA website (2005). The price of these bearings is approx AU$4 each.

### 5.1.7 Hubs

The hub assembly is an integral part of the brake system as it sets the spacing for the brake discs as well as holding them rigid with respect to the brake axle. The hub assembly consists of a brake hub, end plate and two brake disc spacers. An exploded
view of the hub assembly is shown in Figure 19. When the parts are assembled, grub screws are used to lock the hub onto the brake axle and three bolts are used to stop the discs from rotating on the axle.

![Exploded view of hub assembly](image)

Figure 19 - Exploded view of hub assembly

Because the brake discs used are from the hard drive of a computer and are extremely well balanced, as discussed in Section 4.5, the hub assembly was designed to avoid too many machining operations to the discs as to maintain their excellent vibrational properties. The brake hub was designed to fit through the existing hole in the discs, while the discs themselves only required three small drilling operations to allow for the securing bolts. Because the discs were largely unchanged, their well balanced properties were able to be maintained.

The hub spacers were designed to a very tight tolerance as their size is crucial to the operation of the brake system and hence, the function of the entire dynamometer. To ensure that an air gap of 0.3mm between the magnet and disc faces was maintained and accounting for the size of the housing plates, a spacer size of 8.76mm was required. If the spacer size was not 8.76mm a misalignment between the brakes and discs would be generated, altering the magnetic field strength or possibly causing the discs and brakes to
rub, which would damage to the system. For these reasons, the spacers were machined to a very high tolerance to ensure the dynamometers ability to apply an appropriate braking force to the engine.

The hub, endplate and spacers are all machined from aluminium to minimise the added inertia to the system. The assembly is designed to keep the radius and mass of the components to a minimum to reduce added inertia to the system (see Equation 3). It is desired to minimise the added inertia to the rotational components to minimise the engine power required to accelerate the parts (see Equation 2). Excessive addition of mass or size will then result in inaccuracies in the engine power measured by the dynamometer due to some power being lost to accelerate the additional components. Ensuring that the hub system was designed to be as small and lightweight as possible, allowed for the added inertia to the system and hence, measurement inaccuracies of the dynamometer to be minimised.

5.1.8 Brake System Housing

The brake housing is designed to allow the discs to rotate freely, to provide mounting space and to couple the braking system to the Z-displacement system. The main body of the housing was designed using two plates to provide room for the discs. The discs and hubs are located on a shaft that runs centrally on two precision bearings through the top plate, while the bottom plate is spaced from the top plate using four PR-2.0 precision rods sourced from SDI. The rods used allow for a 10mm clearance between the tips of the discs to the bottom plate of the housing. The precision rods are used due to their high structural rigidity. A PR-2.0 precision rod is also attached centrally underneath the bottom plate. The rod is used to slide in the Z-displacement system, which adjusts the height of the braking system.
5.2 Shaft Coupling Device

A shaft coupling device was required to physically couple the output shaft of the braking system to the engine output shaft. It was decided to use a miniature flexible coupling for this requirement as they provide vibration isolation between the engine and brakes, allow for an amount of misalignment between the shafts of the two systems and they are small in size. Vibration isolation is important as RC engines are very unbalanced which could cause fatiguing in bearings of the brake housing or damage to the brakes themselves. Allowing for misalignment between shafts is a desirable feature of the flexible coupling as discussed in Section 4.6. Despite the displacement systems providing for fine adjustment of sub systems, it would be very difficult to perfectly align the brake shaft and engine output shaft meaning that a flexible coupling is required. A miniature coupling is also required to minimise the added inertia to the system reducing the accuracy lost in the measurement of power output.

![Figure 20 - A flexible coupling is used to couple the braking system to the engine](image)

The flexible coupling chosen for use in the dynamometer is the BKL 003 ECOFLEX coupling from R+W Couplings. This coupling was chosen for its low price, small size, and design features. The ECOFLEX coupling allows for up to 1mm of axial
misalignment and 2° of angular misalignment (rw-couplings.com 2005) which should be suitable for the RC dynamometer. The coupling has a mass of around 50 grams, and has an outside diameter of only 25mm, meaning that the added inertia to the system is small. The coupling is capable of handling a dynamic torque of 1.5Nm, which is 50% greater than the technical specification of the dynamometer and while the exact rated speed of the coupling is not specified it is in excess of 10,000rpm and similar couplings provided by R+W are capable of handling speeds of over 40,000rpm. The cost of the coupling was AUS48 which is much cheaper than couplings of similar performance. The features of the BKL 003 ECOFLEX coupling made it a suitable choice for use as a coupling device between the brake system and engine in the dynamometer.

![Figure 21 - BKL 003 ECOFLEX coupling](image)

The internal diameters of the ECOFLEX coupling was bored to enable it to be used with several engine output shaft sizes. One end of the coupling was bored out to 5mm, which is the size of the brake shaft, while the other end of the coupling was bored to 8mm which was assumed to the largest likely engine shaft size. If an engine shaft smaller than 8mm is to be used a sleeve split into two halves of outer diameter 8mm and inner diameter equal to the output shaft size can be placed into the coupling. The grub screws of the coupling can then be tightened to provide a clamping action on the shafts which will clamp the sleeve and shaft together.
5.3 X,Y,Z Displacement Adjustment Systems

To effectively couple the output shafts of a variety of engines to the shaft of the braking system, adjustment must be possible in all three axes allowing for the shafts of the two systems to be centrally aligned. The alignment of the two shafts is crucial to minimise the generation of undesirable movement and vibration caused by misaligned shafts.

For simplicity of design, it was decided to couple the X-, Y- displacement systems to the engine mounting system, while separately adjusting the Z- direction displacement at the braking system.

5.3.1 X-Direction Adjustment

To provide positional adjustment of the engine in the X-direction it was decided to use MDR rails (Section 4.2.2) and RC rail carrier systems (Section 4.2.3) supplied by SDI. The MDR rails are attached directly onto the torque cradle using screws. MDR-6.0i rails were chosen as they were seen to provide for an adequate level of adjustability, with the final design being able to provide a maximum of 40mm of adjustment in the X-direction. The rail carriers, which slide along the rails to provide adjustability in the X-direction, also provide mating faces to couple to the rails used in the Y- displacement system. Positional adjustment in the X- direction is made by loosening the locking screws on the rail carriers and moving the engine mounting system forwards or backwards to the desired position then retightening the locking screws to secure the system.

5.3.2 Y-Direction Adjustment

Positional adjustment of the engine along the Y-axis is achieved in a similar way to the X-directional adjustment, using rails and rail carriers. Two MDR-6.0i rails are attached to the rail cars of the X-displacement system so that the X- and Y- adjustment systems are coupled together. Rail carriers, which are coupled to engine mounting system, are then
used to slide along the rails, providing adjustability of the engine position in the Y-direction Figure 22.

![Figure 22 - X-,Y- adjustment systems utilise rails and rail carriers](image)

5.3.3 **Z-Direction Adjustment**

It was decided to use an adjustable rod holder purchased from SDI (see Section 4.2.5) to provide vertical adjustability. It was decided to couple the rod holder to the brake housing, allowing for adjustability of the height of the brake shaft, rather than the engine shaft. Adjusting the height of the brake system was done to avoid the complications required to couple the X-,Y- and Z- displacement systems together.

The rod holder used was an ARH-3.0 rod holder (Figure 23). The height of the combined brake and Z-displacement systems was desired to be minimised to increase the structural rigidity of the dynamometer. The rod holder is available in a larger size, which provides for greater adjustability, however to keep the combined brake and Z-displacement assembly as short as possible to maximise rigidity, the ARH-3.0 holder was chosen. The
optics industry that the ARH-3.0 is designed for requires a high level of rigidity to
minimise vibrational effects and a fine level of adjustment, so the rod holder is perfectly
suited to the application of the RC dynamometer.

To minimise the undesired vibrational effects of an unbalanced system, it was necessary
to place the rod holder centrally under brake housing when coupling the two systems.
This caused a design problem as the torque arm (discussed in Section 5.4.1) runs through
this location. The solution to this problem was that a bracket was designed which extends
around the torque arm (Figure 23).

The brake system is coupled to the adjustable rod holder via a precision rod that is
attached to the bottom of the brake housing as discussed in Section 5.1.8. The precision
rod is designed to slide in and out of the adjustable rod holder to provide crude height
adjustment of the brake system and the adjuster ring of the rod holder can be used to
provide fine vertical adjustment (Figure 23). A combination of the fine and coarse
adjustment allows for good vertical alignment between the shaft of the braking system to
the output shaft of the engine.

Figure 23 - Vertical adjustment system
5.4 Measurement Systems

In order to determine the power and torque output of the engine, torque and speed measurement systems were required to be implemented. A torque cradle was used to transfer the torque created by the engine into a strain gauge system, while the engine speed was measured using a hall sensor located at the brake housing.

5.4.1 Torque Arm

A torque arm has been designed to transfer the torque generated by the engine into the strain gauge system. The torque arm has been designed to couple into the X-, Y-displacement system, which effectively couples the cradle to the engine mounting system. When a torque is applied by the brakes, it is assumed that the equal and opposite torque is produced by the engine. This reactionary torque causes the torque arm to try and rotate about its bearing supports, however the arm is rigidly fixed at the strain gauge system, causing all of the torque generated to be transferred as a force through this connection. The force which is transferred through the fixed connection then creates a measurable response at the strain gauges.

The cradle of the torque arm is 300x150mm allowing for the rails of the X- displacement system to be easily attached. The lever arms of the torque arm extend 70mm each side from the centre of the torque arm. One lever arm is used to fix the torque arm at the strain gauge system while the other is used in the calibration of the system.
5.4.2 Strain Gauge System

To measure the torque output of the engine, strain gauges are used. The strain gauges used to measure the output torque are arranged in a proof load ring cell. The ring used comprises of two strain gauges set up on the inside of the ring and two on the outside. When a torque is generated by the engine, the force transmitted through the connection at the torque arm (see Section 5.4.1) causes the ring to deform, generating a measurable response in the strain gauges. When the ring deforms, the inside gauges will experience tension, while those on the outside will experience compression, generating a voltage output that can be related to torque. For an appropriate output, the strain gauges are connected to a bridge circuit giving the arrangement four times the sensitivity of a single
gauge (Turner and Hill 1999). The voltage output from the strain gauges is then amplified before being input to a computer for analysis (see Section 8).

**Strain Gauge Choice**

Choice of appropriate sensitivity of the strain gauges was important to achieve accurate results from the dynamometer. If the sensitivity of the gauges used is too low then the change in resistance registered will be small and inaccurate but if a set-up is used that has too much sensitivity then when the gauge is put under a stress it may plastically deform destroying the gauge. The wrong choice of strain gauge may also damage the amplifier chosen.

The strain gauges used must be capable of measuring forces over the entire operating range of the dynamometer. As the lever length on the torque arm is 70mm and assuming a generated torque of 1Nm, a force of 14 N will be created in the strain gauge system. For this reason, it was desired to use a strain gauge that is capable of registering, with high accuracy, forces applied to the stress ring between 0 and 14 N. An analysis was conducted in ANSYS to determine the maximum strain that the gauges will experience under a 14N force. The full analysis is shown in Appendix B. The analysis revealed that a gauge with sensitivity of approximately 180 micro strain was required. From these results an appropriate commercially available strain gauge was chosen for use in the dynamometer which was capable of registering a static torque loads of up to 2Nm.

**5.4.3 Speed Measurement System**

Measurement of engine speed, which is required to determine engine output power, is achieved using a hall sensor (see Section 4.3.1). As it is assumed that the engine is running at the same speed as the brake shaft, it was decided to measure the speed at the brake housing as there was suitable space available. The hall sensor picks up on a magnet that has been placed on the brake discs, generating a pulse once per revolution. The pulses are then amplified before being input into a computer for further analysis (see Section 8).
5.5 Engine Mounting System

An engine mounting system was required in the design to physically couple the engine to the dynamometer. It was specified that the engine mounts must accommodate for a wide variety of engine sizes and this was achieved using a replaceable mounting system that couples into the Y-displacement system as described in Section 5.3.2. Removable inserts were used, as the bottom of the mounts will require several machining operations to provide a coupling point for the Y-displacement system. The entire mounting system is displayed in Figure 25.

![Figure 25 - Engine mounting system coupled to engine and X-and Y- displacement systems](image)

To accommodate for a variety of engine sizes, the mounts are designed to move in and out symmetrically, guided by the rails of the Y-displacement system using a dual threaded adjustment bar. The double threaded bar is useful when adjusting the engine mounts to the size of the engine mounting plates. Once the mounts have been adjusted to the desired width using the bar, the engine can then be secured into the top inserts of the engine mounts.
The removable top inserts of the mounts have been designed to be easy to manufacture and replace to facilitate in the simple mounting of many different engines, which will different engine mounting plate sizes (Figure 26). All faces of the inserts have been designed flat to minimise the machining operations required in their production, which will reduce costs if they need to be reproduced to mount different engines. The inserts use a lug to locate into the bottom section of the mounts. The top inserts can then be simply screwed into holes located in the bottom section of the mounts.

The bottom sections of the mounts have been designed to securely support the removable inserts and to couple into the Y- displacement system (Figure 26). In order to couple the engine mounts to the rails of the Y-displacement system, pre-manufactured rail carriers were used (see Section 4.2.3). The rail carriers were an attractive option to couple the two systems as the mounting faces required to be machined into the bottom of the engine mounts could be achieved very easily due to the perpendicular sides of the rail carriers. The rail carriers are simply attached to the engine mounts using a screw passed through the pre-drilled bolt holes in the carrier into a tapped hole in the engine mount. The bottom of the mounts also contains threaded holes for the adjuster bar and top inserts.

The size of the mounts and inserts was designed around the size of the OS engines which are located at The University of Adelaide. These are relatively large engines used in hobby airplanes. The design allows for mounting bolts up to 55mm (not including the width of the mounting plates on the engine) and a distance between mounting holes of up to 70 mm when mounting an engine. These mounting options should allow for a very wide variety of engines to be secured to the mounts. The inserts are also designed so that the desired mounting holes can be drilled into them and then either tapped so that an engine can be screwed in or simply bolted from underneath for additional versatility.

Using the rail carriers to attach the engine mounts the Y-displacement system will provide for a very rigid connection between the two systems. The carriers are a relatively expensive solution as they cost approximately US$20 each but will remove any problems with dimensional tolerances that could arise if workshop machined parts were used. The
use of rail carriers also allows for the mounts to be easily held in place by tightening their adjuster screws.

Figure 26 - Exploded view of engine mounting system
6 Modal Analysis of Final Designs

Before the final concept designs were submitted to the workshop for manufacture, several analyses were required to minimise the risk of failure. A finite element analysis (FEA) was carried out on sections of the design to ensure that the natural frequencies of the dynamometer did not couple with the likely driving frequencies of the engines used. It is desired that the modal frequencies of the dynamometer occur outside the operating ranges of the engines used as these frequencies will lead to accelerated fatigue and early failure as well as inaccuracies in output data due to vibrations being induced in the strain gauge system.

Due to the complex nature of the dynamometer assembly, a full model analysis was not possible and so only critical sections were modelled. It was decided that the sections of the dynamometer most susceptible to vibration would be the torque arm and the combined Z-direction displacement and brake housing system, hence an FEA to determine the modal frequencies of these systems was carried out.

6.1 FEA of Z-Direction Displacement/Braking System

After visual inspection of the dynamometer design it was decided that an important area of analysis was the Z-direction adjustment/braking system. This was conducted as it was thought that the plate structures used in the housing and bracket may resonate in operating range of the engines.

Simplifications

Several simplifications had to be made to the system before it could be solved. The gears were removed as they contained a lot of fine detail which would have required a very small mesh size and would affect the accuracy of the results. Removing these parts would have little effect as they were small compact parts with little mass. The brake discs and
brakes were also removed. This was done primarily to reduce computational time. These components would have an effect on the system as they would create a reasonable mass load. If these components were in the model the modal frequencies would have probably been lowered due to the extra mass added to the system.

**Constraints**

The model of the Z-direction displacement system was fixed at its base (Figure 27). This represented the system being bolted to its base. These were the only constraints placed on the system although in reality the disc shaft would also be attached to the driveshaft. The effect of the coupling between the brake shaft and the driveshaft of the engine was ignored for this analysis.

![Figure 27 - Z-direction FEA model](image)

**Tests**

Because the Z-direction displacement system is adjustable, its modal frequencies will change every time that the length is changed. It was decided to analyse the system at its estimated shortest and highest heights to determine a range that the modal frequencies...
will occur in. It is expected that raising the height of the system will reduce the modal frequencies due to an increased cantilever effect.

**Low Model**
The first model was analysed with the adjustable rod holder set to its lowest height (76.2mm) with no additional extension rod used between the holder and brake housing (Figure 28). This places the brake shaft approximately 13mm below the top of the engine mounts which is about the limit of how low drive shafts come out of engines below their mounting plates.

![Low model](image)

**High Model**
This model was set up to simulate the Z-direction displacement system at its highest setting. The rod holder has been increased to its maximum height of 88.9mm and an extension rod has been used to increase the length of the holder a further 15mm (Figure 28). This system places the shaft of the brake discs 10mm above the top of the engine mounts.

![High model](image)
mounts. It is expected that this will be the maximum height that the adjuster will need to be increased to as most drive shafts are located equal to or below the height of their mounting plates.

A 3-D model for each case was inserted into ANSYS Workbench and solved for their modal frequencies. The results are shown in Table 6. The deflection of the first modal frequency (which was similar for both cases) is shown in Figure 29.

Table 6 - Modal frequencies of Z-displacement/brake housing system

<table>
<thead>
<tr>
<th>Modal frequency</th>
<th>Modal freq. (Hz) (low model)</th>
<th>Modal freq. (Hz) (high model)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>128.78</td>
<td>94.8</td>
</tr>
<tr>
<td>2nd</td>
<td>134.82</td>
<td>97.9</td>
</tr>
<tr>
<td>3rd</td>
<td>243.66</td>
<td>157.1</td>
</tr>
<tr>
<td>4th</td>
<td>418.19</td>
<td>360.6</td>
</tr>
</tbody>
</table>

Figure 29 - First mode shape of vertical adjuster
6.1.1 Analysis

It is expected that the first modal frequency will induce the largest response from the system so this is the frequency which should be primarily avoided. The first modal frequencies of the systems occur at approximately 95Hz and 129Hz for the high and low set-ups respectively. These frequencies correspond to approximately 5700rpm and 7740rpm. It is likely that the first modal frequency for set up heights between the upper and lower limits in this analysis will fall in this frequency range. For this reason it is likely that the accuracy of the results obtained from the dynamometer between 5700rpm and 7750rpm may be adversely affected due to vibrations. Fortunately most of the operating range for RC engines is above 9500rpm and they will quickly accelerate through this region during run up to operating speed so, the vibrational effects will be minimal. However, this range does couple with the range of some engines. For example most multi-cylinder engines such as the OS FR5-300 and fuel injected engines such as the OS 1.40 RX-FI which reach peak performance at 8,000rpm and 9,000rpm respectively listed on the OS engines website (2005) operate within the 6000-9500rpm range. Because of this, engines that are rated by the manufacturer to have peak performance which falls within the range of first-modal frequencies should be avoided for use in the dynamometer if possible.

6.1.2 Redesign and Further Analysis

To enable engines with low running speeds to be used in the dynamometer, lowering the natural frequencies of the Z-displacement/brake housing system was investigated. The natural frequency of the system can be reduced by lowering its stiffness. The easiest way to do this was to reduce the size of the bottom plate of the brake housing as this is where flexing in most of the modes originates.

As plate thickness is reduced the vibration amplitude that is induced in the plate at its first modal frequency becomes larger which will reduce the fatigue life of the system. This is due to displacement amplitude being inversely proportional to modal mass, so as the plate becomes thinner its modal mass is reduced and hence it displacement amplitude will
increase. Because of this a trade off has to be made between lowering the modal frequencies and rigidity. The original plate thickness was 5mm and the system was reanalysed with plate thicknesses of 2.5mm, 3.5mm and 4mm to determine the effects that these changes have on the modal frequencies of the adjuster system. The results are shown in Table 7.

Table 7 - Modal frequencies of system with various plate widths

<table>
<thead>
<tr>
<th>Modal frequency</th>
<th>Value (Hz) 2.5mm plate (low model)</th>
<th>Value (Hz) 3.5mm plate (low model)</th>
<th>Value (Hz) 4.0mm plate (low model)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>82.8</td>
<td>105.2</td>
<td>113.3</td>
</tr>
<tr>
<td>2nd</td>
<td>87.5</td>
<td>108.9</td>
<td>120.0</td>
</tr>
<tr>
<td>3rd</td>
<td>205.5</td>
<td>255.8</td>
<td>243.6</td>
</tr>
<tr>
<td>4th</td>
<td>248.4</td>
<td>296.6</td>
<td>334.1</td>
</tr>
</tbody>
</table>

Table 7 shows that the first modal frequencies of the plate occurs at 83 Hz (4980 cycles/min), 105Hz (approx 6300 cycles/min) and 113Hz (approx 6780 cycles/min) for the 2.5mm, 3.5mm and 4mm plates respectively. The reduction made from the use of a 2.5mm plate was very large, however it is likely that the reduction in rigidity would be too great and should not be considered as a viable option. The 3.5mm plate is probably the best option if the plate thickness was to be changed as it gave a reasonable reduction in the first modal frequency of approx 15% lower than the original system, although it resulted from a 30% reduction in thickness.

After analysing the results from the FEA it was decided not to alter the 5mm plate thickness due to uncertainty about the fatigue life of the system. The fatigue life of the system is very difficult to estimate because the input force into the system from the engine is not known as the force due to the imbalance of every engine varies. For this reason it is recommended that the plate thickness should remain at 5mm to increase fatigue life.
6.2 Torque Arm Analysis

The torque arm structure was chosen for analysis as its beam like structure is likely to cause large resonances in the dynamometer. The system analysed is shown in Figure 30.

Simplifications

There were several simplifications made in the analysis of the torque arm structure. The main simplification was the removal of the X-Y displacement systems and engine mounts which the torque arm is attached to. The model is shown in Figure 30. This simplification was made primarily to reduce computational time. The consequence of the simplifications made is that the calculated modal frequencies will be considerably higher than the modal frequencies of the physical dynamometer due to the removal of a significant amount of mass from the system.

Figure 30 - Torque arm system to be analysed
Constraints

- Bases of support posts were constrained in all DOF
- One end of the extension arm was fixed to represent a rigid connection to the strain gauge system
- Bearing restraints were created between the torque arm and support posts to represent actual bearings.

6.2.1 Analysis

The FEA results of the torque showed that it exhibits its first four modal frequencies below the operating range of most nitro engines. The modal frequencies determined are shown in Table 8. The first modal frequency of the torque arm is approximately 42Hz (2500rpm), while the fourth modal frequency occurs at approximately 160Hz (9600rpm). Due to the model simplifications made, the modal frequencies displayed by the physical dynamometer will be lower than these values. It is expected that higher modal frequencies will have little effect on the quality of the data obtained from the dynamometer due to smaller displacement amplitudes displayed at higher frequencies. It is possible however, that higher modes could induce serious fatiguing into the dynamometer, although this effect is difficult to theoretically analyse. As it is very hard to ensure that the operating range of the engines used do not couple with any modal frequencies of the dynamometer, the design of the torque arm is considered acceptable as the frequency of the most dominant modes in the dynamometer occur outside of the operating range of most engines.

<table>
<thead>
<tr>
<th>Modal frequency</th>
<th>Modal freq. (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>41.9</td>
</tr>
<tr>
<td>2nd</td>
<td>104.0</td>
</tr>
<tr>
<td>3rd</td>
<td>145.9</td>
</tr>
<tr>
<td>4th</td>
<td>159.6</td>
</tr>
</tbody>
</table>
6.3 Recommendations

It is inevitable that some modal frequencies will occur in the operating range of some engines due to the large variations in engine specifications. Because it is hard to determine what effect these modal frequencies will have on the system, an attempt can be made to minimise their effect on the dynamometer using vibration damping in the physical system and low pass filters in the software. It is also recommended that engines with operating ranges below 9,500rpm should be used with caution in the dynamometer until more is known about the possible force inputs from the motor.
7 Dynamometer Manufacture

After the final designs of the dynamometer had been created, during the manufacturing stage further consultation of certain sections were discussed with individuals with excellent practical knowledge in both designing and building of structures similar to the dynamometer. The consultation process undertaken was vital in producing a final product that worked as desired.

The dynamometer was manufactured at Lathe-Weld Technics by Mr Bob Jarrad and is shown in Figure 31 to Figure 33. During the construction period, several consultations were held between the members of the group and Mr Jarrad to clarify design concepts and make small changes to the design to make the manufacturing of the dynamometer easier. Mr Jarrad also suggested that some sections of the design should be larger, or made from thicker material to withstand forces and to minimise problems associated with vibration. The changes suggested by Mr Jarrad ensured that the dynamometer was built to be as structurally stable as possible.

The design of several sections of the dynamometer were designed and manufactured in consultation with Mr George Osborne from the engineering workshop at The University of Adelaide. The worm gear system used to apply a torque to the brakes and strain gauge systems used in the dynamometer were produced by Mr Osborne. The gear system used to apply a torque on the DC dynamometer discussed in Section 2.2 was produced by Mr Osborne and analysis revealed that the system was too loose and the structure also represented an undesirable cantilever geometry hence, was seen to be a major weakness of the DC dynamometer design. In producing a similar system for the RC dynamometer, the design was changed from a cantilever support to be supported on both ends of the worm gear. Mr Osborne also provided the manufacture of the strain gauge system including the necessary wiring required to enable it to provide output data into an amplifier before the data was analysed. The results of the work that were produced in consultation with Mr Osborne were found to operate as designed throughout testing and were determined to be successful.
The machining in the brake and disc assembly was critical to ensure the success of the dynamometer and constant consultation with workshop staff was required to ensure that the tight level of tolerance required was met. The force provided by the magnetic brake is proportional to the size of the air-gap between the brakes and the magnets in the housing and so to maximise braking capability, the design air-gap was minimised. The air-gap in the RC dynamometer was designed to be 0.3mm as a compromise between braking force and machining time to achieve a tighter tolerance. This air gap used meant that a total variance across the entire brake housing of less than 0.3mm was required to prevent the brakes from rubbing on the discs which would affect the ability of the dynamometer to apply a constant braking load and could possibly damage the dynamometer. During the manufacturing process it was found that several additional shim spacers were required to be produced to overcome the inaccuracies that occurred in machining the critical parts. However, the final braking assembly was capable of providing a significant braking load without and rubbing which was desired.

*Figure 31 - Manufactured dynamometer assembly*
Figure 32 - Manufactured XY displacement system and brake assembly

Figure 33 - Manufactured brake and gear assembly


8 Data Acquisition

In order to convert the raw data from the strain gauges and hall sensor into meaningful values of speed, torque and power in real time, a data acquisition system was required. Data analysis is conducted using a computer. In order to analyse the data on a computer, a dSPACE controller, the Simulink toolbox of the program Matlab and the dSPACE Control Desk program were required. The dSPACE controller is used as an interface between the hardware of the dynamometer (hall sensor and strain gauges) and software. A model was set up in Simulink which converted the raw data into values of speed, power and torque. The dSPACE Control Desk program was then used in conjunction with the Simulink model to analyse the data in a real time, user friendly environment.

dSPACE Controller

The dSPACE controller is used to act as an interface between the hardware of the dynamometer and the software. The controller has both encoder and analogue inputs. The voltage from the strain gauge connected through the analogue input, while the data from the hall sensor is input through the encoder input. Encoder input is used due to its high sampling rate – the analogue input is not capable of sampling at the high speeds which the engines will run at leading to incorrect speed data.

Simulink Model

A model was set up in the Simulink toolbox of Matlab which converts the data from the dSPACE controller into meaningful values of speed, torque and power. The model and process in which data is determined from the can be seen in Figure 34.
Torque Calculation

The following blocks were used to determine the torque output of the engine:

DS1140MUX_ADC – Analogue to digital converter to convert the voltage output from strain gauge into data that Simulink was capable of interpreting.

Voltage Offset – As the torque cradle which the strain gauges are attached to is not perfectly balanced, a static voltage due to this imbalance is output from the strain gauges. The voltage offset is used to remove the effect of this imbalance.

Torque Factor – This gain factor was determined from calibration testing (see Section 9.2), which converts the voltage output into torque.

Low Pass Filter – The low pass filter is used to reduce the effect that the vibration of the dynamometer have on the data, producing a smooth torque curve. The output of the low pass filter is used when analysing engine torque as it does not display the effects of high frequency vibration.
Speed Calculation

The following blocks were used to determine the engine speed of the dynamometer:

DS1104ENC_SETUP – This block is required in Simulink when using data from the encoder inputs.

DS1104ENC_POS – The data from this block represents the pulses from the hall sensor.

Pulse Divider – This gain amplifies the data from the encoder, the output from the pulse divider is then used in the Pulse Decoder to determine speed.

Pulse Decoder – This block converts the pulses into speed. The output from the pulse divider is compared against a logic function under the block, which eliminates small voltages caused from sources such as noise. Each time the logic function becomes true, a unit value is released and combined with the clock function. The frequency at which the unit values are released, equates to cycles per second determining engine speed.

Power Calculation

As power is equal to the rotational speed multiplied by torque, the output power is simply determined using a multiplying block that combines torque and power.
dSPACE Control Desk
A user interface was set up in the dSPACE Control Desk program to allow for real time analysis of torque and power outputs from the engine (Figure 35). The layout allows for real time plots of output torque and power against speed (depending on which type of plot is selected), instantaneous read-outs of torque, power and speed as well as adjustment of offset values in the Simulink model. Adjustment of offset values is required to eliminate the static imbalance of the system due to the balance of the system changing with the mounting of each new engine and the hysteresis effect from the strain gauges. The real time plots of power and torque allow for the effect of changing engine settings to be graphically seen for simple analysis during the tuning process.

Figure 35 - dSPACE Control Desk layout
9 Testing

After the dynamometer had been manufactured, several stages of testing were undertaken. Tests undertaken involved determining the natural frequency of the system, calibration, testing of a DC motor and tuning of a RC nitro engine.

9.1 Resonant Frequency Testing

Determination of the resonant frequencies of the dynamometer is required to minimise the effects caused by its resonant frequencies. As RC engines operate over a large range of frequencies, it is likely that some resonant frequencies of the dynamometer will couple with running speeds of the engines used, however it is desired that the modes with largest response will not be encountered under operating conditions. As the first resonant frequency usually generates the largest response, this is the resonant frequency that was determined in testing. If the operating speed of the engine couples with the first natural frequency of the dynamometer, there will be large levels of vibration experienced, indicating that some form of passive vibration damping will be required to prevent inaccurate results and possibly damage to the dynamometer. If the resonant frequency determined lies outside the operating range of the engines and the dynamometer does not excessively vibrate at higher frequencies, there will be no need for vibration damping to alter the vibrational response of the system.

Initial analysis of subsystems was undertaken using ANSYS during the design process in Section 6 to minimise the chance of the natural frequency of the dynamometer coupling with the running speeds of the RC nitro engines, however determination of the actual frequency was required to validate the analysis. The complex structure of the dynamometer consists of many subsystems and so a limited number of sub systems were analysed to give a rough approximation of the expected natural frequencies. For this reason it was expected that the theoretical results will be reasonably dissimilar to the theoretical analysis.
An impact test was used to determine the first natural frequency of the dynamometer. The dynamometer assembly was given a sharp hit, which theoretically excites the structure over an infinite range of frequencies, exciting the natural modes of the system (as the impact time was not infinitely small an infinite range of frequencies would not actually be experienced). The vibrations from the excited modes were recorded at the strain gauges and then analysed in dSPACE.

The first natural frequency of the dynamometer structure was determined to be at approximately 20Hz, well below the operating range of a RC nitro engine. While this frequency is out of the range of an RC nitro engine, during the commissioning stage the dynamometer was used with DC motors which are capable of running at such slow speeds and so caution was used not to operate the motors at this speed for extended periods of time.

9.2 Calibration Testing

The voltage output from the strain gauges was required to be calibrated, allowing for the data acquisition software to produce meaningful results of torque and power. The calibration tests were conducted by applying known masses onto the end of the lever arm, allowing for the force created at the strain gauges to be calculated. The test masses were hung from the lever arm using a wire to best approximate a point load on the system to provide the more accurate results. A simplification of this process is shown in Figure 36. The resulting output voltage from the strain gauges due to the loads was then seen in dSPACE Control Desk. Comparison of the output voltage and known torque created about the pivot point then allowed for the gain factor, which converts voltage to torque to be determined (see Section 4.3.3). The results of the calibration test can be seen in Table 9.
Figure 36 - Simplification of torque arm

Table 9 - Calibration test results

<table>
<thead>
<tr>
<th>Test Weight (N)</th>
<th>Lever arm length (m)</th>
<th>Theoretical torque (Nm)</th>
<th>Voltage output (mV)</th>
<th>Gain Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.00</td>
<td>0.065</td>
<td>1.04</td>
<td>0.435</td>
<td>1.195</td>
</tr>
<tr>
<td>2.84</td>
<td>0.065</td>
<td>0.37</td>
<td>0.150</td>
<td>1.23</td>
</tr>
<tr>
<td>1.96</td>
<td>0.065</td>
<td>0.25</td>
<td>0.105</td>
<td>1.21</td>
</tr>
<tr>
<td>5.82</td>
<td>0.065</td>
<td>0.76</td>
<td>0.310</td>
<td>1.21</td>
</tr>
</tbody>
</table>

Using the results from Table 9, an initial average gain factor of 1.211 was determined (this factor was changed throughout testing with each recalibration of the system).

Hysteresis effects were observed to be a problem with the torque measurement system, as prior to a test, the strain gauge would be measuring zero, and it would be found to be non-zero at the end of the test. While the values were not changing by a large amount, an offset of the torque measurement was required to be added to the software to counteract the effect. It was important to re-zero the torque measurement periodically between testing to minimise the error in the data. After the hysteresis effect was observed it was important to recalibrate the strain gauge gain factor before each use of the dynamometer as well as changing the offset value. Ensuring that the strain gauge was properly calibrated and offset ensured that the data collected from the dynamometer was correct.
9.3 DC Motor Testing

Before the dynamometer could be used with RC nitro engines, a period of testing was completed using DC motors. DC motors were used before testing with RC engines to ensure that the subsystems of the manufactured dynamometer functioned as designed and to reveal any unforeseen problems. Testing with DC motors was also used to ensure that the sensors located on the dynamometer (strain gauges and the Hall effect sensor) coupled effectively with the software and Simulink model used to analyse the output data. By ensuring that all of the sub systems of the dynamometer functioned as designed, it was also possible to meet a project goal of using the dynamometer to accurately measure torque, speed and power from the dynamometer in real time.

DC motors were ideal for these initial testing procedures because they are very well balanced and so did not create any problems from induced vibrations and their running speed can be easily controlled by varying the applied voltage using a power supply.

Mounting a DC motor on the dynamometer showed that the X-, Y- and Z-displacement systems could be used effectively to align the output shaft of the motor to the axle of the brake system. Being able to mount a DC motor was important as it showed that the dynamometer was able to mount even a very small motor as well as the larger nitro engines it was primarily designed for.

After the DC motor was effectively mounted to the dynamometer, the motor was run at various speeds to test the ability of the brakes. Starting at low voltage and hence low speed, the brakes were applied to the motor and the effect on motor speed noted. Slowly increasing the voltage, the ability of the brakes to stop the motor at higher speeds was tested, until the motor was run at maximum voltage (12V). The DC dynamometer tests proved the ability of the braking system to stall DC motors running up to 12V (giving torque outputs of around 0.1Nm) and hence, gave high expectations for the use of the dynamometer with RC nitro engines.
Testing of the DC motors gave a good insight into the performance of the dynamometer at a variety of engine speeds. The DC motors were capable of running from static conditions up to approximately 6000rpm at a peak of 12V. When the motors were running at an input voltage of approximately 3V, the speed of the motors was approximately 20Hz, coupling with the natural frequency of the dynamometer. The effect of running the dynamometer at its natural frequency was clearly apparent, causing the entire system to visibly shake. The output data at resonance was also heavily affected, not allowing for any clear analysis of torque or power to be made. If the system was left to run at its resonant frequency, severe damage to all sub systems would have been likely to occur quickly. It was found that the effects from resonance could be greatly minimised if the motors were ran quickly through the natural frequencies of the dynamometer allowing for acceptable data to be achieved.

![Figure 37 - Torque and power curves from DC motor](image)

The torque and power curves achieved from DC motor testing were in excellent correlation with the manufacturer data sheets. The DC motor used for testing is rated to a maximum torque and power of 0.1Nm and 18W respectively (penmotion.com, 2005). An example of the results achieved in testing is shown in Figure 37, in which a peak torque of approximately 0.12Nm and peak power of 21.6W were achieved. The values
determined are both 20% higher than the data from the manufacturer, while peak power was also achieved at the same speed. This indicated that the calibration factor used was incorrect and a simple recalibration of the system would have enabled for the rated torque and power to be almost identically replicated. As the rated torque and of the DC motor could be accurately determined, the initial goal to achieve measurement accuracy to within 5% could be achieved. Accuracy of speed measurement to within 5% has also been achieved, as the readings of power which were determined to be accurate are directly related to speed.

The results of testing with a DC motor have shown that the dynamometer is capable of effectively stalling motors, mounting various engine sizes and mechanically stable at speeds up to 6000rpm. The data acquisition software was shown to produce accurate results of output torque, power and speed in real time, enabling a project goal to be achieved. The results of DC motor testing were an excellent starting point to validating the major aim of the project, which is to run an RC engine on the dynamometer using real time data to determine output characteristics.

9.4  **RC Nitro Engine Tests**

The final stage of testing the dynamometer was conducted using a nitro powered RC engine. Dynamometer testing with a nitro engine allowed for tuning in real time using a steady state process by determining engine output power and torque. Analysis of the output data in real time allowed for a significant project aim of tuning in steady state to be met.

Health and safety was a large issue when testing with the nitro engines. All engine tests were conducted in an engine bay at The University of Adelaide which provided appropriate noise isolation from near by areas and air ventilation. The ventilation was provided by a centrifugal fan and flexible ducting that allowing the fumes and smoke to
be effectively removed from the area. Running the engines in a small enclosed space also created noise issues, which were rectified by the use of personal protection equipment.

**Engine Tested**

The nitro powered engine used on the dynamometer during testing was a Smartech CXP 0.15 (Figure 38). This is a motor from a miniature hobby racing car. There are three parameters that can be changed on the engine which are idle speed, air-fuel ratio and high speed mixture. Any change in these parameters will affect the output characteristics of the engine, however for simplicity only the fuel-air ratios were changed during the testing process.

![Figure 38 - Smartech CXP 0.15 engine](image)

**9.4.1 Steady State Testing Procedure**

The engine was tuned in steady state to meet a major project goal by showing that the dynamometer was capable of producing useful results in the tuning process of a nitro engine. To tune an engine in steady state the engine was run up to operating conditions with a constant throttle setting then a constant torque load was applied to the engine using the brake. Once engine speed and torque became relatively constant, as indicated by the
straight lines on the torque and power curves on the computer, an engine setting (fuel-air ratio) was changed and then the system was allowed to reach a new steady state. Analysis of the power and torque plots before and after the change in engine settings revealed the effect which the change had on output performance. The temperamental nature of the nitro engines provided some problems when tuning in steady state. Initial testing revealed that the nitro engines are very temperamental with large fluctuations in output characteristics from little or no change in engine settings. The effect of this unsteady nature was that large setting times were required after making an engine change to ensure that the change in output data was an effect of a change in engine settings rather than a fluctuation resulting from normal operating conditions. A plot of the unsteady nature of the engine is seen in Figure 39. For this test the throttle setting was held constant under no load generating an engine speed of approximately 120 rps. The changes in speed are seen to fluctuate between approximately 100 rps and 130 rps over a 50 second period. These varying output speeds are most likely to changes in the amount of fuel and air mixing that occurs between engine strokes and other factors such as the actual amount of fuel and air entering the combustion chamber varying with time. Before any test data was recorded it was ensured that the engine speed was relatively constant (engine speed also fluctuated less once the engine had warmed up).

![Figure 39 - Fluctuations in engine speed under constant throttle settings](image)

Despite constant variations in engine outputs the changes in output power and torque were able to be determined in real time in a steady state manor.
The loads applied to the engine during testing were significant enough to reduce running speed considerably from no load conditions however, a large range of engine speeds were still possible by adjusting the throttle. Under no load conditions the engine was capable of reaching speeds of up to approximately 300 rps (18,000 rpm). A peak speed reading of 340 rps (20,400 rpm) was registered during testing. The range of engine speeds which were possible after the loads had been applied in the tests was approximately 75 rps (4500 rpm) to 250 rps (12000 rpm). These load conditions were used across all steady state tests as they were assumed to be close to the loads required to overcome rolling resistance when driving the car in practice.

**9.4.2 Steady State Testing Results**

For the steady state test shown in Figure 40, the engine was run up to approximately 150 rps (approximately half throttle) and a load was applied, generating an output torque of approximately 0.14Nm, the torque applied slowed the engine speed to approximately 120 rps. Once the engine output had stabilised, the air-fuel ratio was adjusted to increase the fuel flow into the engine. The large spike seen in the plots occurs when the air-fuel ratio is changed and is primarily due to the external load applied by the operator on the engine when the air to fuel ratio was changed. After the change had been made the engine was allowed to restabilise and it can be seen that there was an increase in output torque to approximately 0.145Nm and an increase in output power of about 25W up to 100W. As both power and torque increased due to the change in engine settings, the performance of the engine at these load conditions was improved. Also, as both power and torque increased, it indicates the engine was operating below peak power as when the engine approaches peak power the output torque generally begins to decrease.
Figure 41 shows a further test that was conducted to demonstrate steady state tuning. The engine settings obtained from decreasing the air-fuel ratio (increasing fuel input) in the previous test were used while the amount of brake applied to the engine was also approximately the same. It was not possible to replicate the exact same braking input from the previous test as the braking is adjusted by hand, however possible solutions to correct this are discussed in Section 11. It can be seen that the initial output from the engine is approximately 0.15Nm and after approximately 10 seconds at steady state the air-fuel ratio was decreased and allowed to restabilise. It can be seen that this change resulted in a significant decrease in output torque and an increase in output power. The output torque decreased to approximately 0.135Nm while the power output increased from approximately 130W to 150W. The power increase was due to a large increase in engine speed with a relatively small decrease torque. A possible reason for the decrease in torque with the increase in speed is that the engine is nearing its peak power output.
9.4.3 Steady State Testing with Speed Data

To get a better indication of the effect that the change in engine speed which results from a change in engine settings, a further test was conducted that plotted torque, power and speed against time. Again the engine was set to about half throttle and a similar amount of brake was exposed to the discs as in the previous tests, however the engine settings were different as the tests were conducted on a different occasion to the previous tests.

The data for the test which records engine speed is shown in Figure 42. The initial speed, torque and power output from the engine were approximately 120rps, 0.14Nm and 100W respectively. The change in air-fuel ratio caused a noticeable increase in speed, power and torque readings. Unfortunately, the engine had not returned to steady state before the end of the test as the plots did not plateau, although the effect of the changing in engine settings can be clearly seen. The results show that the change in engine setting caused a significant increase in engine speed, while the output torque also increased (although it can be seen to reduce slightly as speed further increased due to the system not returning.
to steady state). The effect of an increasing engine speed and torque at this load also produced a considerable increase in power. The results indicate that the change made to the engine were beneficial for improving engine performance at the load conditions.

**Determining Torque and Power Curves**

After it was shown that it is possible to tune an engine in real time using a steady state method, the output characteristics of the engine over its entire throttle range was determined under a constant load.

**Testing Process**

The engine settings used in this test were different to the previous tests. A load was placed on the engine allowing it to idle at approximately 80 rps (3200 rpm) the throttle was then opened until peak engine speed was reached. The resulting torque and power curves were then plotted in the dSPACE Control Desk program.

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![Figure 42 - Steady state test with engine power, torque and speed](image-url)
9.4.4 Testing Results

The curves of engine speed versus torque and power from the test is shown in Figure 43. The results show that as engine speed increases from idle, both torque and power increase. There is a sharp increase in torque at approximately 90 rps, most likely due to the fuel mixture being too rich at idle causing the performance of the engine to be reduced. The output torque begins to plateau above 100 rps and remains relatively constant between 120 rps and 140 rps. The torque peaks at approximately 0.17Nm at about 145 rps and from this point torque continually decreases with increasing speed. The power output from the engine remains reasonably linear between idle and 170 rps. A peak power from the engine of approximately 180W is achieved at 177 rps (10,620 rpm). The plots show that as engine speed increases past 177 rps engine performance deteriorates for the particular engine setting very quickly, which is most likely to a lack of fuel at high speeds. The engine stalled at approximately 182 rps and is demonstrated by the curves of torque and power dropping off as speed is reduced after the peak speed has been reached. If the engine had not stalled, it is expected that as engine speed was reduced, the torque and power curves would follow the same shape which were produced from increasing engine speed.

Figure 43 shows that the engine is relatively well set up to run between 100 rps and 170 rps due to the relatively constant torque response in this range and linear response of output power. If higher engine speeds were desired under this load the high speed mixture ratio could be changed to allow for more fuel to enter the combustion chamber at higher speeds, while the effect of this change at the running speed of interest could be seen in real time by plotting the engine outputs against time.
9.4.5 Testing Analysis

The results of testing the dynamometer with a nitro engine have been very successful. Tuning in steady state was possible as well as determining the torque and power responses of the engine over its entire throttle range under a constant load. Minimal vibrational effects were also experienced throughout the testing process. The minimal vibrational effects are a direct result of the engine operating well above the low natural frequencies of the system. Higher natural frequencies of the system will tend to cause lower amplitudes of vibration and hence are less noticeable in the data and in vibration of the system.

Figure 43 - Torque and power curves determined from the nitro engine
Testing of the nitro engine under a constant load has shown that it is possible to tune an engine in steady state. The effect of changing an engine setting on its output characteristics was able to be determined in real time, in doing so validating a project goal. The plots determined show that a change in air-fuel ratio which results in a change in output torque as little as 0.005Nm can be determined. The output power and engine speeds were also successfully recorded during testing. Speeds of up to 340 rps (20400 rpm) were recorded while output powers of approximately 200W were recorded. It was specified that the dynamometer should be capable of determining output speed, torque and power of up to 50,000 rpm, 1Nm and 3.73 kW respectively, however testing these limits could only be possible with the use of a larger and more powerful engine.

The results from steady state testing revealed that the changes in engine output power and torque for most cases could be determined in a short period of time. The time required for the real time plots to reach steady state after a change in engine settings during most tests was between 15 seconds and 30 seconds. The time required to change engine settings was approximately 10 seconds, which could be greatly reduced, allowing the system to return to steady state much faster if servo control was implemented to alter engine parameters (discussed in Section 11). Tuning in steady state has shown that a fast tuning process is possible with the RC dynamometer.

It was noted that during testing with the nitro engines there were very little resonant effects experienced as predicted from the FEA analysis of the system, this negated the need for implementation of vibration damping. The FEA analysis (see Section 6.1) predicted that the third modal frequency of the Z-displacement system/brake housing would occur in the engine ranges tested over. The effects of this modal frequency were not very evident during testing and were reduced further with the use of a low pass filter, maintained the integrity of the results. It was initially thought that if the resonant effects experienced during testing were significant then some form of vibration damping would be required. Fortunately, the dynamometer remained relatively rigid during testing meaning that vibrational isolation was not required.
There was very little heat generation noticed in the magnetic brakes during testing. A theoretical analysis of the heat generated during braking under full load conditions indicated that the Currie temperature of magnets may have been realised (see Appendix C). If the Currie temperature of the magnets was reached they would begin to lose their properties, preventing a constant load from being applied however, testing did not reveal any significant change in brake temperatures. It is most likely that there was little change in brake temperatures during tests due to the relatively light loads applied in comparison to the design loads and the short engine run times experienced during testing. The testing indicated that the design of the braking system does effectively dissipate the heat generated over the testing period.

During testing the flexible coupling worked as intended without any damage occurring in the coupling, engine or dynamometer systems. After considerable testing time it was noted that the coupling had not deformed past its design limits and was still capable of effectively clamping onto the shafts of the brake and engine. Despite a slight misalignment between the shafts of the engine and braking system being visually determinable, the coupling was capable of withstanding the misalignment without generating any excessive forces into the dynamometer, indicating that the coupling worked as desired.
10 Conclusion

This report has documented the process of designing, building and testing of a dynamometer which can be effectively used to tune a nitro powered engine in a steady state manor. Throughout the year the majority of project aims and goals were able to be achieved and in doing so, a functioning dynamometer, designed to be stable at high frequencies, that is capable of accommodating for a variety of engine sizes and producing accurate output data in real time was produced. The previously mentioned properties of the dynamometer allowed steady state tuning of a nitro engine to be possible.

The dynamometer produced in this project operated on the principles of a brake dynamometer which makes steady state tuning possible. The benefits of producing a steady state dynamometer are that an engine can be tuned under constant operating conditions, allowing for very fine changes in engine settings to be quantified, while current dynamometer designs for RC vehicles do not possess this property. The main motivation behind this project was the fact that there are no current designs which can be used to tune an engine steady state which is more beneficial than current methods.

Ease of manufacture was a large consideration when designing the dynamometer. The parts used in the construction of the dynamometer were also designed to be as rigid as possible to minimise problems associated with vibrations. Where possible, prefabricated parts were used to minimise labour time and the parts designed were kept as simple as possible and to provide increased rigidity of the dynamometer. This design approach was very successful as very few changes were made to the dynamometer design after it had been manufactured.

A series of finite element analyses were conducted to ensure that the running speeds of the engines tested did not couple with natural frequencies of the dynamometer. A finite element analysis of the entire structure was determined unfeasible due to the complexity of the geometry but was replaced with several analyses of smaller subsystems. Resonant frequencies of key sub systems were calculated and the major modal responses were
determined to be well below the operating ranges of RC motors. The resonant frequencies experienced in testing were also in good correlation with the FEA’s, which resulted in quite low vibration effects being experienced from use with the nitro engines. The effects of vibrations were also further reduced with the implementation of a low pass filter in the Simulink model used to analyse output data. The output data determined from testing has shown that the adverse problems resulting from vibrations has been effectively accounted for in both the software used and the physical design of the dynamometer.

The dynamometer has been successfully designed to be used with a large variety of engine sizes by utilising displacement systems which allow for adjustment of the engine to be made along all three axes of displacement. The design of the displacement systems of the dynamometer allowed for the position of the engine to be adjusted to ensure that the output shaft of the engine is closely aligned to the shaft of the brake housing. A flexible coupling was then used to couple the engine to the braking system while accommodating for any slight misalignment between the shafts and at the same time reducing vibration transference between the systems, preventing any damage occurring. The versatility of the engine mounting system was shown during testing, when it was effectively used to mount several engine sizes to the dynamometer without any adverse side effects experienced. Several engine types were mounted to the dynamometer during testing including DC motors and nitro powered engines which vary considerably in size. These results demonstrated that the mounting systems allow for the dynamometer to effectively operate with various engine sizes without causing any damage to the system.

Testing of the dynamometer with both DC motors and nitro powered RC engines allowed for real time outputs of speed and torque to be produced. Tuning in steady state was achieved by making an adjustment in engine settings when the engine was running at constant load and throttle conditions. The effect of the change in engine settings on output characteristics could then be immediately seen in real time, which is the process of tuning in steady state. The data was analysed in real time using a data acquisition system that utilised speed and strain sensors, a dSPACE controller, dSPACE Control Desk program, Matlab and Simulink. A user friendly interface was set up in the dSPACE
Control Desk program which provided digital read outs of engine characteristics as well as plots against time and engine speed. The output data achieved from testing a nitro engine in this manor was capable of determining even slight changes in output power and torque, demonstrating that the dynamometer is an effective tool in fine tuning of an engine.

The torque and power curves over the entire running range of the motors and engines were able to be successfully produced by plotting these characteristics against engine speed. The power and torque curves produced aided in the tuning process of the engine as they gave the user the ability to immediately see at what engine speeds that output characteristics peaked and the ranges where outputs remained relatively constant. This data gave a better indication of what running conditions the engine could be tuned in steady state to optimise output power or torque.

The accuracy of the output data from the dynamometer was proven relatively accurate. Testing of the dynamometer with DC motors revealed that the data determined was in close correlation with manufacturer data. As there was no manufacturer data available on the nitro engine tested it was difficult to determine the exact accuracy of the results achieved during those tests however, the data determined seemed reasonable and based on the excellent results from testing with DC motors, was assumed to be accurate.

The dynamometer was not tested to its peak design limits of withstanding 1Nm of dynamic torque, operating speeds of up to 50,000 rpm and powers of up to 3.73kW (5hp) however, a large range of operating conditions were experienced. The upper limits of the torque, power and speed goals were not tested due to the lack of availability of larger engines to test. The dynamometer has been successfully tested across an engine range of 0-20000 rpm and has been able to measure torques of up to 0.17Nm and powers up to 200W. Static torques applied to the dynamometer during testing showed that the strain gauges were capable of registering torques created about the torque arm of up to approximately 2Nm, indicating that it is likely that they could accommodate for the design dynamic torque. The components used in the dynamometer have also been either
chosen or designed to accommodate for the entire ranges of power, torque and speed previously specified and it is expected that the subsystems would operate as desired. The group has met to the best of its ability the goal of having the dynamometer capable of withstanding the technical design specifications and was only limited in testing due to the lack of larger equipment.

No heat generation problems from applying a braking load using the magnetic braking system were encountered during testing. A theoretical analysis of the heat generated during braking indicated that the Currie temperature of magnets which were used as brake may have been realised if operated for extended periods under full load, however testing did not reveal any significant change in brake temperatures. It is most likely that there was little change in brake temperatures during tests due to the relatively light loads in comparison to the design loads and short run times experienced during testing. The testing process indicated that the design of the braking system can effectively dissipate the heat generated over the testing period.

The process that was followed during this project to design, build and test a dynamometer has been both challenging and rewarding. As the ability of the dynamometer to test and tune a nitro powered engine in a steady state manner has been demonstrated, it is believed that the main goal of this project has been successfully achieved.
11 Future Work

There are certain aspects of the dynamometer which, if changed, would have improved the results achieved in the testing. With some additional work, the tuning process could be even faster and more efficient.

Possible future work for the dynamometer project includes adding a closed loop control system to the assembly. Through the addition of servo motors on the torque applicator (brake) and on the throttle of the engine, the generation of power and torque curves could be automated. The advantage of a system able to automatically create power and torque curves is that for a given engine setting (air fuel ratio) the software would be able to generate the curves for pre-set throttle settings much faster which would lead to a faster, more accurate and more efficient tuning process. The engines that are to be tested are sourced from remote control models which are designed to be actuated using servo motors and so the automation of the controls would be relatively easy. The possible use of a servo motor to control the brake has been designed into the brake assembly and so the mounting of the servo motor and change to servo controlled braking should also be easy. Through the automation of the brake and throttle the dynamometer system could be completely sealed except for inlet an inlet for air to the engine and an outlet for the exhaust. This would be a much safer option as the user could be protected from the rotating components of the dynamometer. The use of servo motors to control the application of the brake will also eliminate the interference observed when the brake is applied manually. This can be seen in Section 9.4 where there are large interruptions to the data when the brake is applied or an engine setting is changed. A further advancement in the control system would be to allow servo control over the engine setting such as air-fuel ratio and high and low speed settings. Closed loop control of the dynamometer would allow a faster and more efficient tuning process.

The Simulink code could be improved to yield more accurate results. The current Simulink code does not take into account power losses from the engine due to the acceleration of the mass on the disc axle. As shown in Equation 2, power from the engine
is used to accelerate the mass on the disc axle. While this omission does affect the overall accuracy of the results obtained from the dynamometer in terms of overall power outputs it does not affect the ability of the dynamometer to tune the engine. The tuning process is achieved by comparing successive sets of data and so the error can be disregarded as a steady state error. As shown in the DC motor tests, the accuracy of the results obtained from the dynamometer is very good and this would be only a minor improvement.

The Simulink code could be improved by adding allowances for altered temperature and humidity during the testing conditions. Again this is a steady state error that will not impede the ability of the dynamometer to tune an engine in steady state, but may cause the values of power and torque to be slightly inaccurate meaning that normalisation of results is required for analysis. The addition of this allowance would require temperature and humidity sensors to be placed in the area the dynamometer is being operated in.

The Matlab, Simulink and dSPACE data acquisition system used in this project are far more complex and powerful than the project requires. Also the dynamometer is only using 2 of the many inputs available on the dSPACE controller. If the dynamometer was to be made commercially available, this data acquisition system would be unacceptable due to its prohibitive price and bulkiness. A future project for the dynamometer would involve re-designing the data acquisition system to be more suited to the dynamometer application. The system would require the two inputs from the dynamometer (torque and engine speed) to be turned into useful data and then multiplied to generate the power value. A simple user interface would be designed which allows the user to quickly calibrate the dynamometer and easily see the values and curves of engine speed, power and torque.

Some modifications could be made to the dynamometer design to increase the rigidity of the system. As was seen in Section 9 there were some issues with the vibration of the dynamometer at certain frequencies which lead to inaccuracies in the data. It was expected that there may be some problems with resonances and steps were taken to minimise the detrimental effects, but the design could be improved to further minimise
the impact on the results. The biggest problem with the rigidity of the structure is the Z displacement system. While the component is designed for the optics industry and so is made to a high tolerance and high rigidity, the level of vibration that is generated by the RC engine was probably larger than the component was designed for. The plastic bolts supplied with the adjustable rod holder were unable to clamp the rod connected to the disc and brake system and so had to be replaced with a more rugged steel bolt. The system could be made much more rigid by completely replacing the adjustable rod holder with a rigid connection. While the rigid connection will not allow the adjustability of the adjustable rod holder, it is likely that the dynamometer will only be used with a limited number of engines and so a number of rigid connections could be manufactured. The rigid connection would greatly reduce the vibration in the system and improve the accuracy of the results.

A more major redesign of the dynamometer would further minimise the vibrational effects of the RC engines. The redesign would involve mounting the brake/disc assembly on the torque cradle. Mounting the brake and disc system (with its relatively low levels of vibration) on the torque cradle would reduce the amount of vibration directly input to the strain gauge. The XYZ displacement system could also be redesigned to be completely contained on the torque cradle with the disc and brake system. Re-arranging the dynamometer in this way would also allow the engine to be more rigidly mounted to the base-plate or possibly separated from the braking system altogether, again reducing the detrimental vibrations.
12 References


Appendices

Appendix A – Design Iterations

This section details the design iterations that lead to the final dynamometer design. For each iteration, several changes were made to improve the dynamometer in some way – functionality, ease of manufacture, cost considerations and others.

Figure A. 1 - Design iteration 1
Design iteration 1 was the first complete model created of the dynamometer. It is a complete assembly utilising 12” rails in the X direction, 6” in the Y direction and a 4 post assembly to allow for Z displacement. The engine mounting assembly mounts directly onto the rails to be purchased from SDI.

Figure A. 2 - Design iteration 2

The second design iteration improves on the first iteration by decreasing the X direction displacement length as it was seen to be unnecessary and would have lead to a less rigid structure more prone to resonance and high levels of vibration. The engine mounts were changed to utilise the rail carries manufactured by SDI to mount to the rails also.
manufactured by SDI. This greatly reduced the machining time to be done by the engineering workshop as machining the slides to an acceptable tolerance to fit the rails would be extremely time consuming and difficult. It was questioned if the workshop would be able to machine the required shape with the available tools. The engine mounts will be much easier to machine to the shape of the rail carriers than it would be to the shape of the rails. For these reasons the rail carriers was seen to be a much more palatable option.
The third design iteration changed the 4 post slide design for the Z displacement system was replaced with an adjustable rod holder, again to increase rigidity of the system and to decrease the amount of high tolerance machining to be done in the workshop to achieve an acceptable result. The reliability of the 4 post slide design was questioned as the system may lead to the brake assembly not being mounted straight with respect to the base of the dynamometer and the engine/engine mounts. The use of the adjustable rod holder required the design of a mounting bracket to allow the torque arm to move freely unimpeded by the adjustable rod holder. To achieve this, a bracket was designed to fit over the torque arm that can mount the adjustable rod holder while allowing free movement of the torque arm. The use of the adjustable rod holder unfortunately raises the height of the system, however this is unavoidable and the height of the torque arm had to be raised.
Previous designs of the dynamometer did not allow for the use of an electric starter to start the RC motors to be tested. For this reason the height of the torque cradle pivoting points needed to be lowered to account for the starter. This unfortunately misaligned the axis of the pivot point and the axis of the brake creating a resistance on the pivoting action of the torque arm. This effect however is small and will not greatly impede the results gained from the dynamometer. The Z displacement system mounting was also redesigned to be smaller and more rigid to avoid resonance and vibration issues.
Appendix B – FEA of Strain Gauge System

An analysis of the aluminium load cell ring to be used in the strain gauge system was analysed in ANSYS to determine the required sensitivity of the strain gauges.

Properties of aluminium used
Young’s Modulus – 69 GPa
Poisson’s Ratio – 0.33  (roymech.co.uk, 2005)

Material was assumed to have linear, elastic and isotropic behaviour.

Loads
Max torque, \( T \), applied to ring = 1Nm = \( Fl \)

\[
F = \frac{T}{l} = \frac{1}{0.07} = 14N
\]

Ring Geometry
Outside diameter = 38mm
Inside diameter = 34mm
Width = 10mm

Element type used – Structural Mass, Solid, Triangular 6 node 2. Triangular elements were used as they are better suited to the ring shape geometry

Mesh size – Global mesh size of 4

Constraints

- Bottom node fixed in all DOF
- Unit force of -1 applied to top node
Results

To determine the maximum strain experienced by the ring, total Von Mises strain was plotted Figure B. 1. This showed that for a unit force the maximum strain experienced will be approximately 1270 micro strain. The resulting strain from a 14N force will be approximately 180 micro strain. This indicates that a strain gauge capable of measuring the widest possible range of 0-180 micro strain is desired.

Figure B. 1 - Total Von Mises strain contour plot
Appendix C- Heat Convection Analysis of Brake Discs

An analysis was conducted to determine if the choice of three brake discs was suitable to prevent the brake magnets from reaching their Currie temperature when the dynamometer is operating at its peak conditions.

Assumptions:

- A specification of the dynamometer is that it is designed to handle 5.0hp. The 91 VR-DF supplied by OS engines (www.osengines.com) is listed as producing 4.8hp at 22,000rpm.
- All power produced by the engine is dissipated as heat (for safety of calculations).
- The heat of the brake discs is equal to that of the brakes (for safety of calculations)
- Air temp = 295K.
- Surface temp of discs = 465K (just below Currie temp.).
- No additional heat transfer from environmental air flow (no wind)
- 1hp = 745.7W
- Transition from laminar to turbulent flow for a rotating disc occurs at a Reynolds number, $Re_t \approx 2.4E+05$
- Discs are zero thickness to assume no heat loss from ends of discs.

Analysis:

The following analysis is an adaptation of an analysis of rotating surfaces covered by Mills (1999). The analysis is to determine the convective heat loss from a rotating disc:

Mean surface temp \(= \frac{295 + 465}{2} = 380K\)

Properties of air at 380K (from Table A-7 Heat transfer, Mills) are:
Heat transfer coefficient, \(k = 0.0319 \text{ w/m K}\)
Kinematic viscosity, \(23.36E-06 \text{ m}^2/\text{s}\)
Prandtl number = 0.69

The disc needs to dissipate 5hp of heat $\dot{Q}_{total} = 5 \times 745.7 = 37528.5$ W

$\dot{Q}_{total} = 3.73$ kW

Angular velocity, $\omega$, at 22,000rpm = $2\pi \left( \frac{22000}{60} \right) = 2308.8$ rad s$^{-1}$

Find the radius at which laminar to turbulent transition occurs:

$$ r_{tr} = \left( \frac{(2.4 \times 10^5 \nu)}{\omega} \right)^{1/2} $$

$$ = \left( \frac{(2.4 \times 10^5)(23.36 \times 10^{-6})}{2308.8} \right)^{1/2} $$

$$ r_{tr} = 0.05m $$

Laminar heat transfer coefficient, $h_c$:

$$ h_c = \frac{(0.585)k(\omega / \nu)^{1/2}}{0.6 / Pr + 0.95 / Pr^{1/3}} $$

$$ = \frac{(0.585)(0.0319)(2308.8 / 23.36 \times 10^{-6})}{0.6 / 0.69 + 0.6 / 0.69^{1/3}} $$

$$ h_c = 95.38 \text{ W/m}^2\text{K} $$

Heat dissipated in laminar regime $\dot{Q}_{lam}$:

$$ \dot{Q}_{lam} = h_c A \Delta T $$

$$ = (95.38)(\pi)(0.05^2)(85) $$

$$ = 63.68 \text{ W} $$

Now determine the heat dissipated from the turbulent regime, $\dot{Q}_{turb}$:

$$ \dot{Q}_{turb} = \dot{Q}_{tot} - \dot{Q}_{lam} $$

$$ \dot{Q}_{turb} = 3730 - 64 $$

101
\[ \dot{Q}_{\text{turb}} = 3666 \text{ W} \]

Now, determine radius of disc in turbulent regime, \( R \):

\[ \dot{Q}_{\text{turb}} = 0.021k(\omega / v)^{0.8} \Pr^{1/3} \Delta T(2\pi / 2.6)(R^{2.6} - r_{tr}^{2.6}) \]

\[ \rightarrow 3666 = (0.021)(0.0319)(2308.8 / 23.36 \times 10^{-6})^{0.8} (0.69)^{1/3} (85)(2\pi / 2.6)(R^{2.6} - 0.05^{2.6}) \]

\[ R = 0.185 \]

Hence, total radius, \( R_{\text{tot}} \), of disc required to dissipate 5hp @ 22,000rpm and maintain a brake heat under the Currie temperature:

\[ R_{\text{tot}} = R + r_{tr} = 0.235 \text{ m} \]

The effective dissipative area (both sides) of a 0.235m disc \( A_e \):

\[ A_e = 2(\pi)(0.235^2) = 0.35 \]

The design uses three discs of 0.095m diameter, this gives an effective area, \( A_d \)

\[ A_d = (2)(3)(\pi)(0.0475^2) = 0.043\text{m} \]

As the required disc area is smaller than that used in the design, it is assumed that the brakes are not capable of effectively dissipationing the required heat. However it is unlikely that the dynamometer will be run at peak conditions and a large margin of safety has been included into the calculations. Because of this the number of brake discs used was not increased.
Appendix D – Brake Shaft Size Analysis

To determine the minimum brake shaft size required to prevent any plastic deformation occurring due to a torque of 1 Nm, the following analysis was conducted:

The stress in a rod, $\sigma$, is given by:

$$\sigma = \frac{\tau \times r}{J}$$

where,

$r$ = radial location of stress
\(\tau\) = torque induced in rod
\(J\) = polar moment inertia of rod.

The polar moment of inertia of a rod is given by

$$J = \frac{\pi \times R^4}{2}$$

where $R$ = radius of rod.

Rearranging gives:

$$\frac{\pi \times r^4}{2} = \frac{\tau \times r}{\sigma}$$

Giving:

$$\frac{\tau \times 2}{\sigma \times \pi} = r^3$$
## Appendix E - ANSYS Properties of Materials

"Aluminum Alloy" Properties

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Appendix F – Cost Analysis

Where possible, off the shelf parts were chosen to decrease the machining time required to decrease the load on the engineering workshop. This unfortunately increased the cost of the dynamometer, but the machining time required to achieve the required tolerance would have been extremely high and so expensive in terms of labour hours. The cost of components of the dynamometer is still very small in comparison to the labour costs expended. This cost analysis does not include the dSPACE controller or Simulink software which were also provided by The University of Adelaide.
## Appendix G – Timesheets

### Level 4 Project Timesheet

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**Total Costs**
- Total (Salary): $12,103
- Total (Direct): $2,793
- Total (Indirect): $4,486
- Total Cost: $19,382

**Notes:**
- Total (Salary) includes labor cost.
- Total (Direct) includes direct labor cost.
- Total (Indirect) includes indirect labor cost.
- Total Cost includes all direct and indirect labor costs.