Mobile Wind Turbine Blade Performance Test Rig

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As governments worldwide look to renewable energy for their electricity needs, the wind turbine has gained renewed popularity as one of several viable methods of large scale electricity production. At the point in the system where the wind is converted into useable energy is the wind turbine blade. The blade, as part of a wind turbine rotor, is a key element within the wind turbine system and as such research aimed at improving blade performance is of great importance. The main objectives of this project were to design, construct and validate the accuracy of an apparatus capable of measuring the performance parameters of wind turbine blades under realistic loading conditions. The completed test rig is now capable of field testing wind turbine rotors with up to 2.5m diameter swept areas. This will ensure that an accurate prediction of yearly power output can be ascertained in relation to each blade type and its location. At the conclusion of the project the test rig proved to be of robust design and was able to provide data on the performance of a rotor with known performance parameters, thus validating its accuracy to a certain level of error. It is hoped that the test rig will be utilised for future wind power investigation within UNSW as it has the potential to support numerous areas of related research.

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Nomenclature

\begin{itemize}
\item \( A \) = rotor swept area [m\(^2\)]
\item \( a \) = axial induction factor
\item \( C_p \) = power coefficient
\item \( C_Q \) = torque coefficient
\item \( C_T \) = thrust coefficient
\item \( E \) = modulus of elasticity [Pa]
\item \( f \) = friction coefficient
\item \( I \) = area moment of inertia [m\(^4\)]
\item \( J \) = polar moment of inertia [m\(^4\)]
\item \( L \) = length [m]
\item \( \lambda \) = tip speed ratio
\item \( \omega \) = angular velocity [rad.s\(^{-1}\)]
\item \( \dot{\omega} \) = angular acceleration [rad.s\(^{-2}\)]
\item \( P \) = power [W]
\item \( Q \) = torque [N.m]
\item \( R \) = rotor swept area radius [m]
\item \( r \) = shaft or hub/rim radius [m]
\item \( T \) = thrust force [N]
\item \( t \) = tension [N]
\item \( \theta \) = angle between each end of the De Prony brake [degrees]
\item \( v \) = upstream velocity [m.s\(^{-1}\)]
\item \( W \) = weight [N]
\end{itemize}

I. Introduction

With the ever increasing demand for alternative, renewable electricity supply, the wind turbine has gathered renewed interest. This interest has been driven by environmental debate, in particular the threat of global warming, and the recent oil supply and demand issues that evoked a greater global awareness of diminishing fossil fuel reserves. Wind derived energy has been identified by various government and commercial bodies as one, of several, viable methods used for the large scale commercial production of electricity. Wind turbine produced electricity is not the exclusive domain of the commercial sector however, with small scale turbines being routinely integrated into domestic power supplies to supplement household electricity use\(^1\). Whilst the commercial and government sectors attempt to curb fossil fuel use from the top end of the consumption spectrum there are consumers attempting to curb its use at the grass roots level. It is the grass roots level that this project’s research is aimed at supporting in an effort to eventually design and generate cheaper blades with higher performance parameters.

With a deep interest in renewable energy the author set out to research wind power and after initial discussions with his supervisor it seemed like the blades were a good place to start. The project’s conception is discussed in detail below. Within the wind turbine system it is the blades that are the initial interface with the wind’s energy making them a critical component within the system and the focus of this project. In order to test the performance of completed prototype blades a test rig needed to be designed and developed as no such device existed at UNSW@ADFA. Hopefully, now that a test rig has been constructed and tested, it will assist and encourage further experimentation and research into wind power.

Other research into blade performance has focused more on the modeling of the blades and less on the actual devices that test them. In some cases testing devices have been constructed and utilised but have not been the focus of the research therefore have been of rudimentary design with little or no documentation. There is also a trend of conducting blade performance tests in the laboratory and not in the environment where they will be used.

As the main focus of the project was the construction and operation of essentially a measuring device, a large emphasis was placed on the design, construction and testing processes. These processes, combined with a finite timeline and other competing tasks meant that careful management of the project was required to ensure its success. To achieve this, a careful balance was struck between the day to day management of the project and the other relevant processes using well established techniques and doctrine.

It should be noted that the design of the test rig was the most demanding and time consuming facet of the project. To achieve an effective and safe design every component of the rig required analysis of its structural integrity as well as its ability to perform its given task. The construction of the rig commenced once the detailed design of its base structure was complete. During construction various modifications were made and some facets completely redesigned as the practicality of certain design aspects were realised. Once the construction was
complete both field and laboratory testing and calibration were conducted resulting in some modification or redesign. The rig was then successfully operated in the field and the relevant data was collected for analysis.

The resulting data proved the rig to be capable of measuring rotor performance under real world conditions. It also showed that the expected scatter can be improved by resolving the velocity vectors into their useful components. Overall the project proved to be a useful addition to the renewable energy arsenal as well as providing the author with a wealth of knowledge in this area. It is hoped that the construction of a wind turbine test rig will encourage further research into this area within UNSW@ADFA and as such possible research topics will be discussed.

A. The Project’s Conception

In October 2008 the report’s author approached Dr Mike Harrap, senior lecturer at the Australian Defence Force Academy’s School of Engineering and Information Technology (SEIT), to discuss the feasibility of undertaking a research project relevant to wind powered renewable energy. After numerous discussions it was agreed that the design of efficient, cheap blades for smaller sized domestic turbines would be of great value to both the wind turbine industry and the author in his endeavor to gain experience in this field. The initial concept was to design blades from cheap or recycled material, in particular PVC pipe as it had shown potential to be an easy and none time consuming way to produce blades. PVC is cheap and readily available and if cut at a particular angle can be made into useful turbine blades. The major problem however, with any research into efficient blade design was the lack of suitable platform within SEIT from which to test its performance. It was at this juncture that the decision was made to design and construct a testing platform that could measure wind turbine blade performance under real world weather conditions. The test platform would provide a suitable research project as well as hopefully encourage future blade design research within SEIT.

B. The Project’s Scope

The main objectives of the project were to design, build and test a mobile apparatus that could accurately measure the performance of small wind turbine rotor prototypes. To accomplish this, the device was exposed to realistic environmental conditions such as naturally occurring wind profiles over various terrain types. In doing this the test rig’s purpose, in the context of system’s design, was to validate the accuracy of the design process prior to rotor production, after various modeling techniques had been utilised. The test rig should be able to evaluate rotor performance, when subjected to real world conditions. To determine if the test rig was producing reasonable performance data, it was fitted with a turbine rotor (the test rotor) that had known performance parameters. The test rotor’s performance data was obtained through Harrap’s research into the energy capture of an aero generator in turbulent winds. The test rotor’s performance curve, as it relates to the power it can produce, is shown at Fig. 1. The final objective of the project was to determine if the test rig was capable of outputting data that, when analysed appropriately, would produce a similar trend to that of the graph at Fig. 1. Once this was achieved the test rig would be deemed to be able to produce data for any rotor with a nominated degree of accuracy. The test rotor is based upon the Clark Y airfoil design and can be seen at Fig. 2 mounted to the test rig’s mast.

Once it was established that the test rig was capable of producing reasonable results the next step was to try and eliminate some of the expected scatter from the data resulting from...
the use of absolute velocity vectors. To do this the wind velocity vectors required reduction down to their
components normal to the rotor blades. This would eliminate the component of the velocity vector not
contributing to the power produced by the wind turbine. Therefore the use of anemometers and a wind vane to
determine the velocity magnitude and direction respectively was important for the production of reasonable
performance data.

To successfully complete the project numerous knowledge areas/disciplines were utilised, these were;
project management, risk assessment, the aerodynamics of wind turbine rotors, engineering design incorporating
CAD, structural mechanics and component design, experiment design, electronics, data acquisition and
measurement systems, wind turbine site selection, meteorology and wind profiling over various terrain.

As wind power is considered to be one of the leading forms of renewable energy an appropriate renewable
philosophy was applied to the project. This philosophy took the form of ensuring that as much as possible the
materials utilised in the construction of the test rig were either recycled from cast offs or from left over materials
from previous projects. This will be discussed further during the project’s methodology.

It should be noted that the Horizontal Axis Wind Turbine (HAWT) was the focus of this project, however,
the design process also considered the possible conversion of the test rig to test Vertical Axis Wind Turbine
(VAWT) blades in any subsequent research projects conducted by students at UNSW@ADFA.

C. The Project’s Aims

The aims of the project were based upon outcomes that were agreed upon, prior to the project’s
commencement, by the project’s manager (the author) and his supervisor as the client. These aims were then
formalized into a client brief which can be found at Appendix A. Each aim represented a milestone that when
completed lead to the eventual successful completion of the project. The aims are listed chronologically below
and were as follows:

1) Design a mobile apparatus that can accurately measure micro wind turbine blade performance.

2) Construct and test the apparatus.

3) Operate the apparatus in the field ensuring that it can produce reasonable data, resulting in the
determination of blade performance.

4) Ensure that at the completion of the project, the apparatus is in a serviceable and ready to use condition
so as to encourage future wind turbine research projects.

If the primary aims were met prior to the scheduled project completion date and it was assessed that there
was enough time remaining a secondary set of aims were to become extant. This did not occur due to time
constraints during the later half of the project. It should be noted that whilst the undertaking of the secondary
aims was not necessary for the successful completion of the project, it would have provided a solid base of
research for any future investigation into wind turbine blade design. The secondary aims will be discussed
further in the recommendations and were as follows:

1) Design a rotor from cheap, readily available material e.g. PVC pipe.

2) Construct the rotor.

3) Test the rotor’s performance using the test rig.

II. Related Research and Theory

Before the design effort could commence a theoretical baseline needed to be established. This baseline not
only established the theories and processes needed for the design process, it also set out to determine any other
related research that may have benefited or value added to the project. The purpose of this chapter is to review
related research into this and similar fields as well as detail the baseline theory that was used throughout the
project.

A. Related Research

Upon reviewing the related research it becomes apparent that there are several accepted methods for the
evaluation of blade performance. Amongst these methods, the most common appear to be; wind tunnel testing,
fixed site testing, 2D and 3D modeling using software or Computational Fluid Design packages and manual
mathematical modeling. It was also apparent that the two most common theories used are Actuator Disk Theory
(ADT) and Blade Element Theory (BET) with some newer theories beginning to emerge.
Nathan’s\textsuperscript{3} paper on simplified design methods and wind tunnel testing is a combination of manual math modeling and wind tunnel testing. Nathan uses BET to model the blades, which is beyond the scope of this project, and uses a wind tunnel to verify the modeled performance. Thrust is not considered. It is assessed that using a wind tunnel will most likely produce results for a rotor subject to near ideal conditions and limit the diameter of rotor tested. Whilst Nathan’s test rig could potentially be used in the field it would require significant redesign. Therefore the major difference between Nathan’s research and this project is that this project aims to test rotor performance under real world conditions and also includes the measurement of thrust that the rotor produces and how it affects the wind turbine’s structure. Zhiquan’s\textsuperscript{4} research follows a similar method to that of Nathan’s, however also uses optimization software for the blade design. Once again the thrust produced by the rotors is not considered, with the focus of the research appearing to be the validation of the optimization software’s accuracy using a wind tunnel. When reviewing Zhiquan’s research, it could be said that the optimization software is the test rig in some respects. However, like the wind tunnel testing, it will likely not be able to simulate real world effects in the field. The common theme of the above research is that the focus was on blade performance, as it relates to power and torque only. In each case the test rigs were of simple design intended for a narrow spectrum of indoor use. In comparison, this project’s focus is the test rig, not the blade, therefore ensuring that the test rig will be of robust design and capable of testing all manner of blades in varying, real world, wind conditions.

Neff and Meroney’s\textsuperscript{5} research into wind and turbulence characteristics due to induction effects near turbine rotors was relevant and useful to this project. Whilst their primary objective is beyond this project’s scope, there were several aspects that were informative and of some assistance in relation to the development of the test rig. The method used for the conduct of the experiment and data acquisition will be of assistance when the test rig is deployed in the field. The Prony brake design used by Neff and Meroney was simple yet effective and has contributed significantly to the design of the project’s brake. Fig. 3 is the Prony brake setup used during Neff and Meroney’s research. Note the spring, wire and ruler device used for force measurement. This was replaced by an electronic load cell attached to this project’s test rig brake assembly. Apart from this much will remain the same. As a suggestion for future projects, Neff and Meroney’s research into wind turbulence characteristics could be expanded upon using the test rig.

Hartwanger and Horvat\textsuperscript{6} discuss the use of 2D and 3D modeling in their investigation of various wind turbine model efficiencies. Whilst this paper focuses on large blades within a system of multiple wind turbines it was interesting to note their adopted process and could be of relevance if time permits the project to move into the blade design phase. Using ADT they build an ideal performance model and then compare it to the real world performance of various in service turbine models. Using the real world data they are able to modify the in service model blade designs in an effort to move towards the ideal performance parameters. This research could be useful in selecting a design method if further investigation into blade design were to occur.

Gould and Fiddes\textsuperscript{7} compare two computation methods for performance prediction. The methods discussed are the Non Linear Lifting Line (NLL) and the three dimensional method (3DM). They assess that the NLL is sufficiently accurate if reliable airfoil data is used and that the 3DM gives good power predictions but requires more work. Once again this would be useful for future blade research. Fuglsang and Madsen\textsuperscript{8} discuss an optimisation method for blade design known as inverse design methods, “where the optimum geometric shape is determined from a prescribed target distribution of some aerodynamic quantity,” and whilst it is an informative paper it is once again only relevant if time permits for blade development. In regards to the primary aim of the project the above research is beyond its scope.

Clausen and Wood\textsuperscript{9} highlighted some of the issues involved with small wind turbine development that researchers were facing at the turn of this century. From research conducted thus far it appears that these issues are still extant nine years on. Whilst most of the topics discussed in the paper were beyond the project’s scope, the issue of blade starting became relevant during the testing and calibration phase and when using the test rig in the field. The amount of torque required to overcome rotationally opposing forces, such as the friction...
associated with the shaft bearings and Prony brake, had to be accounted for with the starting torque being measured prior to deployment in the field. The other main topic of blade material and manufacturing methods will be relevant to any future projects that utilise the test rig. One other useful piece of information from Clausen and Wood’s paper is their classification of wind turbines according to blade size and energy output. This paper classifies the blades mounted on the test rig as being in the micro range (based on diameter) and provides various useful data sets for this blade group, see Table 1. This classification system will be used from this point forward when describing the test rig in the hope that a standard system of wind turbine categorization will develop.

Wright and Wood, apparently acting on one of the recommendations from Ref. 9, carried out research into the starting and low wind speed behavior of small HAWTs. Their research was a combination of field testing of a HAWT and then the comparison of the results to a “quasi-steady blade element analysis”. It was the method by which the HAWT was tested in the field that was of most interest, however the starting behavior was of great importance for both the test rig start up torque considerations as well as any future research. Wright and Wood’s test rig comprised of a three bladed rotor attached to a permanent magnet generator feeding a rectified 24V battery. The turbine was mounted at 8m above the ground with buildings below and trees of the same height in the vicinity, the result being an erratic start and stop behavior. \( \omega \), \( v \), and wind direction, turbine yaw and battery charge were all measured with the assumption that once the battery started charging that the turbine starting sequence was complete. Of interest to this project was; the locations of measuring devices in relation to the HAWT, the field test method and the behavior of a turbine that isn’t directionally fixed. All these considerations, as discussed by Wright and Wood, were taken into account during the testing and calibration and operation and data acquisition phases.

Site selection becomes an important consideration when trying to expose the test rig to wind conditions that closely match that of the actual location the rotors will be employed. McGuigan, succinctly details the considerations that should be taken into account when siting a wind turbine, the most relevant being where to site wind turbines on hills or knolls or in the vicinity of stands of trees or buildings. These considerations were used to assist in the selection of the primary and secondary test sites which will be discussed further in the chapter on testing. In relation to this a CSIRO Research Project Sheet highlights the use of hills or high ground to exploit the resulting speed up effect and flattening of the boundary layer velocity gradient through turbulence, see Fig. 4. This produces a more uniform velocity profile across the rotor’s swept area. What becomes apparent is that the top half of the rotor swept area may experience a different wind velocity than that of the bottom half, resulting in unnecessary energy losses and loading due to bending moments. According to Wizelius, this is due to the velocity gradient of the boundary layer at the earth’s surface, starting at zero and increasing with height. He also states that the rate at which velocity increases is dependant on how rough the surface is i.e. grass, trees, hills etc, see Fig. 5. The implication for the test rig was that a basic theoretical profile was developed for each test site ensuring that the velocity gradient moving through the swept area of the rotor was as even as it could be in the field. To assist with the selection of suitable test sites, a basic wind speed-up profile can be manually calculated. The ESDU mean wind speeds over hills and other topography simple model can be utilised for this purpose. Once the test site has been selected on the basis of this calculation, a more detailed model of the wind speed profile at the location can be created using the ESDU computer program for wind speed and turbulence properties. It should be noted that the software is complex and that the corresponding software guide should be utilised to assist in

### Table 1. Operating parameters for small wind turbines

<table>
<thead>
<tr>
<th>Category</th>
<th>Power (kW)</th>
<th>Radius (m)</th>
<th>Max rotor speed (rpm)</th>
<th>Typical uses</th>
<th>Generator type(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>micro</td>
<td>1</td>
<td>1.5</td>
<td>700</td>
<td>Electric fences, yachts</td>
<td>Permanent magnet (PM)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>remote houses</td>
<td>PM or induction</td>
</tr>
<tr>
<td>mid-range</td>
<td>5</td>
<td>2.5</td>
<td>400</td>
<td>remote communities</td>
<td>PM or induction</td>
</tr>
<tr>
<td>mini</td>
<td>20+</td>
<td>5</td>
<td>200</td>
<td></td>
<td>PM or induction</td>
</tr>
</tbody>
</table>

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Figure 4. Wind profile over small hills.

Figure 5. Wind Gradient over varying terrain.
the use of the program. To help validate the previous steps a tower with anemometers at heights varying from the ground to the top of the rotor’s swept area would be useful.

B. Underlying Theory

As the project was based upon wind turbine research, aerodynamic forces were used to determine the design parameters of the test rig, in particular, how these forces interacted with the structure of the rig in both direct and indirect ways. They were also used to interpret and derive the relationship between the energy in the wind and how it translated to performance parameters when the rig was operated in the field.

To determine how well various blade designs perform when compared to each other, a set of performance parameters were defined. The performance of a HAWT, as proffered by Burton, Sharpe, Jenkins and Bossanyi\textsuperscript{17}, is defined by the three parameters, $T$, $P$ and $Q$ and how they vary with differing wind velocities. The ideal values of these parameters, in particular $T$ and $Q$, were used to derive the initial design loads for the test rig for a maximum blade radius of 1.25 m from the centre of the rotor hub. The value of $P$ for a range of wind velocities, gained from the field tests, was used to determine the rig’s accuracy noting that $C_p$ is the standard used by blade manufacturers to rate their rotor’s performance.

The two most common theories, as discussed by Bianchi, De Battista and Mantz\textsuperscript{18}, used to determine the aerodynamic models for wind turbines are ADT and BET. ADT describes how energy is extracted from wind by a simplified rotor disk, whereas BET discusses how the wind exerts forces on a blade element. For its simplicity ADT was used to determine the design parameters for this project. ADT uses an ideal upper limit for $P$ extraction, the Betz Limit and when the data for the $P$ produced by the blade is gathered it can then be compared to the upper limit. ADT and the Betz Limit are discussed in detail by Wilson\textsuperscript{19} and it was the relevant areas of that article, in conjunction with Ref. 17 and Ref. 18, which provided the theoretical baseline for the aerodynamics of the project.

ADT provides an ideal model of a rotor, with infinite blades and no losses. This was used to derive the initial loading values. Using the equation

$$T = \frac{1}{2} \rho A v^2 C_T$$

where

$$C_T = 4a(1-a)$$

an ideal value of $T$, which is indicative of the worst case loading scenario, was calculated\textsuperscript{ii}. This maximum $T$ value was then applied to the test rig design to kick off the design process. $a = (v-v^*)/v$ where $v^*$ is the reduced velocity of the wind as it passes through the rotor.

To determine the maximum $Q$ for design purposes, Ref. 17 provides data that describes the typical variations for a fixed pitch wind turbine of $C_Q$ and $C_p$ with $\lambda$. From this data the maximum $C_Q$ was derived and used to design the test rig’s shaft.

During the data acquisition/test rig operation phase, $T$ measurements were provided by a load cell and $Q$ via a Prony brake/load cell combination. The $v$ was measured using a profile anemometer that was built specifically for the project. The useful component of the $v$ vector (the component of the wind’s velocity normal to the rotor’s swept area) was measured and used to solve the equations below. The velocity’s direction relative to the rotor’s axis was measured using a wind vane constructed, in part, by the author. The final parameter, $C_p$, was derived from the relationship between the equations

$$C_Q = \frac{C_p}{\lambda} = \frac{Q}{0.5 \rho v^2 AR}$$

where

$$\lambda = \frac{\omega R}{v}$$

and

$$C_p = \frac{P}{0.5 \rho v^3 A}$$

\textsuperscript{ii}The axial induction factor gives the highest value for thrust when it is equal to 0.5, resulting in $C_T$ being equal to 1.
noting that the $\omega$ was also measured by a device attached to the test rig’s shaft assembly. From this relationship Eq. (4) was substituted into Eq. (3) to obtain $C_p$ which allowed Eq. (5) to be solved to determine $P$. It was $C_p$, as it relates to $\lambda$, however, that was compared to the test rotor’s performance data, at Fig. 1, to determine that the test rig was producing reasonable results.

III. Project Methodology

The actual conduct and progress of the thesis were based upon a project management philosophy to ensure that time, finance and man hours were accounted for appropriately. The test rig and accompanying instrumentation were designed, constructed, tested and operated using appropriate systems engineering principles to ensure a successful outcome to the project. Success being measured by the accomplishment of each of the project’s primary aims. The test rig itself was designed from the ground up utilising a particular branch of wind turbine theory that was discussed in the previous chapter. This theory also formed the basis of the operation and performance data produced by the rig. This chapter aims to discuss; how the project was managed including the risk management and the conduct of the design and construction phases.

A. Management

To ensure that the project was completed on time a project management regime was utilised. The project’s management was based upon the principles detailed by Burke in his text and taught as part of the engineering program curriculum. Obviously several of the steps, procedures and disciplines were omitted as the project was undertaken by a student with no subordinates and a supervisor as the client. The student, as project manager, had access to the University’s workshops with very limited funding, workshop man hours and time. Therefore the project was managed as simply as the resources dictated resulting in the production of Gantt and Milestone charts that were used to keep the project’s progress in check. These charts are contained within Appendix A. Throughout the project’s design phase all milestones were met, it was during the construction phase that time overruns began to occur and this was magnified during the testing phase when certain aspects of the design did not perform as expected.

In regards to the engineering of the test rig, a systems engineering approach was used. This approach was based upon the principles detailed by Faulconbridge and Ryan in their text. This approach ensured that the project was conducted in a logical manner. In practice it made each transition, e.g. from requirements to design to construction etc appear to be seamless. When discussing the design process later in this section it will be broken into the various phases discussed in Ref. 21, these being conceptual, preliminary and detailed design and construction. The testing, calibration and operation of the test rig will also be discussed in detail in the subsequent chapters.

As the project involved a large device of considerable size and weight that produced relatively large kinetic forces, safety was an important consideration. Therefore a risk assessment was conducted prior to the test rig being utilised in the field. The assessment highlighted the different risks and their outcomes, if any of the risks were realised. The worst case assessment was that the risks were acceptable with periodic review. The risk assessment is contained within Appendix A and it should be noted that no incidents occurred during the project.

B. Design and Construction

Once it was established that the project would focus on the creation of a test rig the next task was to work out how and in what environment it would function. This signaled the start of the conceptual design phase. During this phase the tower concept evolved through several stages with the favored design being a result of practicality, available material (keeping in mind a renewable philosophy) and the ability of the tower design to satisfy the initial aims of the project. The original concept was to design, build and operate a rig that could be attached to a large vehicle, preferably a truck or 4x4, see Fig. 6 for the original concept sketch. The intent was to operate the rig on a runway (Jervis Bay Military Airfield) whilst attached to an Army UniMoG 4x4 utility truck. The truck would drive along the runway at various speeds with a wind vane and anemometer being used to adjust for wind speed and direction. This concept was discarded for reasons of safety, logistic complication and the blades not being exposed to realistic wind conditions.

![Figure 6. Initial test rig concept.](image_url)
The concept that followed was that of a free standing tower with a directionally pivoting rotor and Prony brake assembly, see Fig. 7. This test rig would consist of a single monopole tower held in place by guy wires, with the test apparatus mounted on top on a rotating base plate. The Prony brake and related instrumentation would all be contained within the confines of the upper base plate. This concept was rejected due to the complexity involved in mounting the instrumentation at height whilst it is rotating. Another factor that ruled this and the previous concept out was the desire to measure the thrust produced by the rotor which added another dimension of complexity to their respective designs.

The final conceptual design was that of a tower with two legs mounted to a base of some description by a pin joint. The tower would be held in place, front and back, by either light chains or wire rope. This would allow the tower to be easily raised and lowered. The blade and shaft assemblies were mounted on a fixed base plate with the Prony brake attachments being secured at ground level. See Fig. 8 for this configuration’s sketch. A major reason for this concept’s acceptance was the desire for mobility combined with the discovery of a 3m x 1.5m trailer that could adequately store, transport and act as a base for the tower.

Once this concept was accepted for further consideration and design the preliminary design phase commenced. During this phase the design underwent several evolutions prior to being ready for detailed analysis. The main factor that drove these changes was the determination of how the test rig would be configured to capture the required data. After some preliminary research into wind profiles over various terrain types it was decided that the rig would not be self aligning with the wind direction. The test rig’s instrumentation and capture software would be able to utilise wind from nearly every direction up to a limit no greater than 90° either side of the rotor’s spin axis, see Fig. 9. The limit was to be determined by the wind velocity and direction required to overcome the shaft’s starting friction and spin the rotor from stationary. It was also determined by the dynamic force (due to wind velocity and direction) required to keep the rotor spinning as it continues to oppose friction. If the wind direction were to change beyond this limit the trailer containing the test rig would be physically moved to reacquire the wind. This methodology resulted in a simplified design and eventually a rig that was easy to erect and operate in a short amount of time. The tower design that
progressed to the detailed design and analysis phase is at Fig. 10.

For the purposes of managing time constraints and the available workshop man hours the detailed design process was broken into manageable sections, these being the tower, shaft, brake assembly, cabling assembly, instrumentation and trailer configuration. The tower, as the interface between the moving parts and the trailer, was designed first. The initial assessment was that the greatest stress in the tower frame would result from the thrust produced by the turbine rotor located at the tower’s highest point. Future design changes will hopefully be more efficient than the test rotor, therefore a robust tower design using worst case loading scenarios was the underpinning boundary condition. As discussed in the basic theory of wind turbine aerodynamics, ADT produces an ideal value of $T$, which was used to estimate the approximate, worst case loadings. To do this Eq. (1) was used with $v$ of 20m/s\textsuperscript{iii}, a maximum $R$ of 1.25m and an $a$ of 0.5, the result being a $T$ of approximately 1200N. This $T$ was then applied at the top of tower which was then simplified into a beam with negligible weight, pinned at its base and attached at 2/3 of its height to a cable in tension. Fig. 11 shows this simplification, also note that the detailed analysis calculations, both manual and computer generated, and material data sheets discussed in this section are in Appendix B. Using this model, shear and moment diagrams were constructed to illustrate the points along the tower that would be subjected to the greatest stresses. As predicted the greatest bending moment occurs at the cable attachment point, then decreases to zero at top and bottom.

At this point in the analysis, a material, DuraGal 65x65mm Square Hollow Section (SHS), was selected for the fabrication of the tower’s legs and mast. It was selected for three reasons; it was in abundant supply as leftovers from other projects, satisfying the recycling commitment, it could also tolerate the stresses that it would be subjected to and it is relatively light. Deflection analysis was then conducted which showed that the tower would deflect up to 14mm rearwards of its central axis at its highest point and at least 3mm forwards at a point approximately 2600mm from its base, which was well within tolerances. Tension and compression within the SHS were calculated showing that the maximum value of each occurred in the outer front and back walls at the location where the greatest bending moments occurred. These stresses however, were well within the tolerances of the chosen SHS by a safety factor (SF) of at least 3.

The next area to consider was that of which tower component would fail first. Of the legs and mast it was assessed that the mast would be the weakest link, therefore a buckling analysis was conducted. To do this, buckling theory from Mott’s\textsuperscript{iv} text was used, in particular the alternate form of the Euler formula

$$P_{cr} = \frac{\pi^2 EI}{(kL)^2}$$

where $P_{cr}$ is the critical load at which the structure will buckle and $k$ is the end fixidity constant which is dependant on how the structure is modeled. In regards to the project’s mast it was modeled as a column fixed at one end and pinned at the other. Eq. (6) demonstrated that the combined loadings, including all associated weight, would not produce buckling with a SF of no less than 16.

The leg mounting assembly was then analysed, in particular its ability to withstand shearing forces due to the vertical loads within the tower. This assembly consisted of a leg mount, mounting pin and mounting bracket. Design books detailing all parts, assemblies and their respective dimensions are in Appendix C\textsuperscript{v}. As the leg mount and mounting bracket were fabricated from 12mm thick AS3678-300 mild steel plate it was assessed that the weakest link would be the leg mounting pin. This was due to the bracket and mount only having to withstand the compressive forces transferred between each by the pin which would be subjected to shear along its length. The pin was fabricated from 709M high tensile steel and the calculation showed that it can withstand whatever shear the tower loading will subject it to with a SF no less than 45. This SF may seem extreme however it was more beneficial to have a larger diameter as it cut down on the machining time involved.

One of the major design changes during this phase was the removal of the flange plates used to mount the mast to the legs, see Fig. 12a. When calculations were conducted to determine the stress placed on the mounting bolts, one in each corner, it was apparent that they may fail when subjected to loadings within the operational safety parameters of the test rig. This required a rethink of how the mast would be attached to the legs. The resulting design was that of a mounting spigot fabricated from 75mmx75mm SHS with metal inserts (shims) to restrict any play between the mast and the spigot, see Fig. 12b. The final step in the tower design was the

\textsuperscript{iii} For safety reasons it was decided that the rig would not be allowed to operate in winds greater than 20m/s.

\textsuperscript{iv} Note that these design books consist of parts and assemblies from the final design and may vary from those designs discussed in this section.
attachment points for the cabling assembly, one of which can be seen in Fig. 12b. This completed the tower design and from this point it was fabricated and attached to the trailer with no further changes to its configuration. The next step was the detailed design of the shaft assembly.

The shaft assembly included: the rotor mount, drive shaft, bearings and housings, shaft mounting plate and band brake drum. The rotor mount was recycled from Dr Harrap’s dissertation\(^2\) experiment and the bearings and housings from Honors Student Adam Leer’s project. This meant only the drive shaft, drum and mounting plate required fabrication. After discussions with workshop staff it was decided that instead of manufacturing a shaft and drum, a very time consuming process, a half axle with attached wheel rim from a standard motor vehicle may suffice. Prior to a shaft purchase or fabrication, analysis was required to determine the minimum shaft diameter required. Eq. (3), in combination with an ideal value of \(C_Q\) was used to calculate the worst case torque that the shaft would be subjected to. This yielded a value of approximately 120N.m that was delivered by the rotor via the shaft to the drum. Using Mott’s\(^{22}\) method of shaft size determination the eventual value of 21mm, minimum shaft diameter was calculated. The ED half axle ranged in diameter from 30mm – 35mm, therefore it was suitable for use as the drive shaft after modifications were made to accommodate the bearings and rotor mount.

As the shaft would be subject to thrust loads, SKF\(^{23}\) spherical roller bearings were selected for the design. A 30mm (YAR 206-2F) bearing was placed at the front and a 35mm (YAR 207-2F) bearing to the rear. As the rear bearing was capable of absorbing larger thrust forces a cir clip and groove were added to the design and placed so the clip was resting on the face of the rear bearing. By doing this all thrust forces are absorbed by the rear bearing and any axial forces apportioned through both as per the design calculations in Appendix B. It was interesting to note the amount of shearing force (in the form of shaft thrust) that the cir clip could absorb before failure which was well beyond anything the test rig could produce, the data sheet for the cir clip is in Appendix B. The shaft also required the addition of a keyway to accommodate the rotor mount which transferred the torque generated by the rotor via a key. As the key width was predetermined by the keyway width of the rotor mount only the key length was required. This calculation was carried out by a program written by A. Leer for his project and is also contained within Appendix B. The program was based upon theory from Mott\(^{22}\). During the construction phase the shaft assembly required the addition of a sleeve to prevent it from moving rearward, through the bearings, when thrust was applied, as the original design did not take this event into account.

At this stage of the design the Magna wheel rim was to act as the drum for the Prony brake. This meant that the brake was designed around the rim’s dimensions. The key elements in the brake’s design were the force magnitudes acting on the brake, the angle at which these forces act in relation to each other and the \(f\) of the braking material to be used. Using the relationship

\[
t = W e^\theta
\]

where \(f = 0.35\) and \(\theta = 180^\circ\), the result is that \(t = 3W\), if we then sum the torques about the shaft at the drum, see Fig. 13, the relationship becomes \(W = 60/r\). The value of \(r \approx 0.15\)m which meant that a maximum load of approximately 40kg would be needed to simulate the required load when the turbine was subjected to a maximum wind speed (20m/s). From the relationship derived via Eq. (7) this would mean that in total the brake

\(^{2}\) Value taken from http://www.engineershandbook.com/Tables/frictioncoefficients.htm and is indicative of the value between braking material and steel for dry sliding friction.

Figure 12. Mast attachment design change.

Figure 13. Sum of torques on shaft.
would have to withstand up to $4W$ or approximately 160kg. The resulting brake design would ultimately prove to be ineffective, this is discussed in detail in the chapter on testing and calibration. The final test rig design that went forward for testing and calibration is at Fig. 14.

The cabling assembly was the next area requiring design analysis and consisted of; tower cables, Prony brake cables, safety braking system, Jin pole and the winch assembly. The tower cables’ purpose is to raise and lower the tower (front cable attached to winch), hold it upright (front and rear cables) and measure the thrust produced by the rotor (front cable with attached load cell). Initially the cables were to be made from light chains however, this idea was discarded early in the design phase due to their weight and lack of availability. What was available and much lighter were the wire ropes salvaged from Dr Harrap’s experiment. These ropes were 6.4mm diameter, 6 x 19 Bright normal lay plow steel which are rated to carry loads of up to 2490kg according to Mark’s Standard Handbook for Mechanical Engineers. At this stage of the design the maximum load the tower cabling would have to support was approximately 1000kg, which was well within tolerances. Note that this load was not as a result of the tower’s weight but the angle from which the tower was to be winched upright.

The Prony brake cables were also to be fabricated from wire rope due to its abundance and fitness for purpose. On the measurement side of the brake, the cable would be fastened to the trailer tray via a load cell and mounting bracket. The weight side would be made to carry a weight rack and be easily attachable to the safety braking system. As discussed previously the maximum load these cables would have to endure was approximately 160kg which they were obviously more than capable of handling.

The purpose of the safety braking system was to slow or stop the shaft if the rotor was subjected to wind speeds above the set safety limit (20m/s). The resulting design was that of a simple hinged lever that would require minimal load on the weighted side of the brake to slow the shaft to a stop. The end of the lever would be bolted to the attachment point for the measurement side of the brake, with a second attachment point further along the lever that, in an emergency, would be attached to the base of the weight rack. If the rotor were to encounter winds at the high end of its operational spectrum then the weight rack would contain approximately 40kg. This would mean that little force would need to be applied to slow the shaft. The only difficulty with the design, which was never rectified due to time constraints, is that the safety brake could not be physically attached to the trailer during the raising and lowering of the tower as it would interfere with this process. A makeshift device was fabricated to hold the rotor still during the raising and lowering of the tower. This

Figure 14. Final test rig design.

Figure 15. The primary cabling assemblies.
situation was not ideal however it ensured that safety was not compromised.

Once the tower was almost complete it weighed in at nearly 900N which meant that if it was to be winched from a near horizontal position the winch, cabling and attachments points would be subjected to loads in the vicinity of 10kN, see Fig. 16. This situation was less than ideal as it meant that most available winches would be lifting near their vertical hoisting capacity. Dr Harrap suggested the use of a Jin pole to dramatically reduce the loads encountered during the raising and lowering of the tower. After detailed analysis was conducted it showed that theoretically the maximum loads encountered by the winching assemblies when a Jin pole was used would be reduced to no more than 2.7kN. To conduct this analysis the center of gravity of the tower assembly needed to be determined as did its actual weight with all attachments. Once these were established the forces acting along the cables and gin pole were resolved and the maximum load calculated. The next step was to select the Jin pole material and once again a length of leftover DuraGal 75mm x 25mm x 2.5mm rectangular hollow section (RHS) was recovered from a previous project. Now that the length and forces acting through the pole were known the buckling load was determined and once again a SF > 3 was achieved.

Figure 16. Drawing of the test rig and winch assembly, prior to the addition of a Jin pole, showing the orientation resulting in large loadings. See Fig. 15 for a picture of the Jin pole addition.

As previously discussed a winch was required that could safely hoist up to 2.7kN. Various winch models were researched and quotes obtained. Initially the workshops loaned the project a 12V electric winch, however it was determined that it could not safely lower the tower. The other winches researched were all manual and fit for purpose but would require some kind of mount or interface to be designed and built to accommodate them within the system. It was at this stage that a disused 10kN rated block and tackle was found in storage. A simple cable which had been previously designed and fabricated to accommodate the electric winch was slightly modified and used for the block and tackle. The block and tackle would be attached to the front of the Jin pole and its lifting chain attached to the aforementioned cable which was simply looped over the trailer’s towing mantle. Once the tower was fully erected a second cable could be attached between the cable attached to the rear of the Jin pole and the cable looped around the towing mantle, allowing the Jin pole and block and tackle to be removed. Fig. 15 is an image of the final design and set up of the winch, Jin pole and tower cable assemblies, shown next to Fig. 14 for design comparison purposes.

The final group of components to be designed or incorporated into the test rig design was the various instruments required to assist in the data acquisition process. Due to a lack of meteorological instrumentation within SEIT, electronic anemometers and a wind vane needed to be manufactured. The remaining instrumentation, load cells and an \(\text{u}^2\) measurement device, were sourced from other projects.

As discussed in the section on research into wind profiles over terrain, it was suggested that a set of anemometers at various heights up to the top of the rotor’s swept area would be beneficial for the determination of the boundary layer profile across the blade area. This would allow a more accurate calculation of the average velocity across the blades then if a block velocity profile were used. It should be noted, however that in sighting the test rig the aim was to ensure the blade swept area sees a uniform profile or as close as possible. The main qualitative specifications for the anemometer tower, henceforth to be known as a profile anemometer, were; it needed to be cheap, simple and interfaceable with Matlab’s Simulink\textsuperscript{25} software in order to record its outputted data. The resulting design, which is in Fig. 17, was constructed from a large scrap of discarded plastic tubing, six small low torque hobby motors, six 180mm hobby electric motor propellers and six 2.3mm propeller adapters. The motors had a nominal voltage of 3V at 1200rpm which was used to determine the optimal blade size for wind speeds between 0m/s – 20m/s. Once the appropriate blade was fitted the motor was able to generate voltages between 0V – 3V for wind speeds of 0m/s – 20m/s. The intention of using low torque motors was to try and reduce the wind velocity required to start spinning the anemometer blade. The profile anemometer tower was made from tent poles found in storage and stood at a height of 7.5m, with the tower being held erect by two sets of three guy wires that could be adjusted to ensure there were no bends in the tower. The anemometers were secured to the tower using hose clamps with their signals being.

![Figure 17. Anemometer design.](image-url)
fed to an analogue computer input device via recycled cabling.

The wind vane that was sourced for the project had been previously designed and constructed during a course in instrumentation by M. Singers, A. Leer and M. Hartwell. This device was calibrated during the course and interfaced to the same input device that was being used by the project for its data collection. This meant that no further work was required for the wind vane apart from it needing to be tested to ensure it still worked. The vane itself was constructed from discarded wood and metals found in the workshop bins and utilised the pulse signal from an old analogue ball mouse wheel sensor to determine the angle the device was facing. The vane is shown in Fig. 18.

Two load cells were sourced from other projects, one for use within the Prony brake assembly to help determine torque, the other for use with the tower cabling (front) to determine the thrust generated by the rotor. The torque load cell was originally rated to measure loads up to 15kN giving a fair to high amount of accuracy keeping in mind the theoretical load produced by the brake would be up to 1.2kN. The thrust load cell was rated to measure loads up to 2kN giving a much better level of accuracy than the first keeping in mind the max thrust load will be approximately 1.2kN. Each load cell was interfaced to its respective cabling via D shackles and outputted an electrical signal to the Simulink capture software. This will be discussed in greater detail in the testing and calibration chapter.

The remaining instrument was the device for measuring the $\omega$ of the rotor, see Fig. 19. This device was based around an optical sensor and was attached to the rear of the shaft assembly base plate, via an aluminum bracket. Aluminum rounds with highly reflective surfaces were glued to the reverse sides of the hub studs which acted to cut the circuit each time they passed the sensor. With 5 hub studs this meant that there would be 5 voltage pulses per revolution, this was later reduced to one to reduce the amount of data being captured. All of the instrumentation was interfaced to a laptop via a National Instruments USB 6009 digital to analogue converter. As mentioned previously Simulink was the interface software with all data being converted to arrays which could be analysed in excel.

The final design effort was that of the trailer configuration which was undertaken to ensure that the trailer would not rotate about its wheels due to the bending moment caused by the rotor’s thrust. To determine if this would happen, moments were taken about the rear of the trailer at the point where it would be supported by two caravan stands. The moment contributing to the trailer’s rotation was the thrust force generated by the rotor and transmitted via the tower’s cable assembly (front) to the trailer’s towing mantle. The moment countering this rotation was the weight distributed through the center of gravity of the test rig. Upon comparing these counter acting moments it was determined that a mass of approximately 400kg would need to act through the trailer’s center of gravity to prevent it from flipping over its end when the rotor was producing maximum thrust. It was further assessed that the combined trailer and test rig mass would be adequate to counter this, however, a chest containing all supporting equipment and measurement weights was placed at the front of the trailer to act as a second counter balance.

During the design process a search was undertaken for a suitable site to conduct the testing and data collection phases. The selection of the site was based upon the concepts discussed in the related research chapter. Initially the testing and operation were to be conducted within the confines of the Majura Defence Range Facility. Within this location was a feature that had clear approaches from most directions out to approximately 5km, little vegetation and an ideal test site located at its summit. The feature itself had a gentle sloping incline on most of the approaches which lent itself to a gradual speedup and flattening of the boundary layer that would have been ideal for running the test rig. Unfortunately, due to time overruns, the range could not be used and the sporting ovals belonging to ADFA were utilised instead. These ovals were generally flat with fairly clear approaches out to approximately 1km. This was not ideal however it was assessed that a reasonable quality of data could be gained from this location. These ovals were utilised for all testing, operation and data
collection throughout the project.

At the completion of the design process most of the aforementioned components’ fabrication was complete. Due to time constraints and other competing priorities, each component was immediately fabricated once its design was complete, with any design changes dealt with immediately during fabrication. This approach meant that the designer (the author) was able to work concurrently with the fabrication process. It was assessed had this approach not been adopted that time overruns during the construction phase may have had a larger impact on the successful completion of the project. This was not the case however and with the design phases complete the testing and calibration phase could commence.

IV. Testing and Calibration

With the design and construction phases successfully completed, the testing and calibration of various components was undertaken. Once a sub assembly was complete it would be subjected to various function tests and if required, calibration and experimentation to determine its accuracy or level of functionality. This chapter will discuss the testing and calibration that was undertaken on the rig to ensure it would be effective in the field. All calibration data and error analysis compiled during this phase is contained within Appendix D.

The first test was conducted once all the components had been fabricated and integrated into the test rig. This simple test involved erecting the rig to ensure that under its full weight the cabling would be adequate to take the load. The test also ascertained how long it would take to set the rig up and to pack it away into the trailer giving the designer a rough idea of how much operational time will be lost in setting the rig up. This was the test rig’s first shake out and there were no design changes made at that stage. The only additions were chocks for the wheels and blocks for the caravan stands for extra safety and stability.

After discussions with the client it was decided to conduct a field test to see how the test rig behaved under wind loadings, therefore no instrumentation was required at this stage. The day chosen for the test had predicted wind gusts of up to 20m/s by mid afternoon. The test commenced at 1000 hours when the rig was erected. Once it was ready for operation the safety brake was released with the first incident of note being that the blades would not spin even under gusty conditions. It was assessed that the Prony brake produced more startup torque than the wind and rotor could overcome, even with no weight load applied to it (the Prony brake). Various materials were tested as makeshift Prony brakes including the nylon rope used to secure the rig in the trailer during transportation. It was noted that the nylon rope allowed the rotor to spin even under light winds and that when a small weight load was applied it was sufficient to slow the blades, even under wind loading approaching the designed safety limits. This result meant that the Prony brake assembly would have to be re-analysed and a more functional design sought. Once the rotor was allowed to spin, under load, up to its safety limit, it was noted that the test rig’s natural frequency was reached when the rotor was exposed to mid range wind speeds. The exact value of the wind speed and shaft $\omega$ could not be determined as no instrumentation was present. When the rig reached its natural frequency the shaft and Prony brake assemblies developed a slight shuddering motion which subsided if the wind speed or $\omega$ changed slightly in either direction. By about midday it was assessed that the winds had reached the test rig’s operational safety limit and the test was abandoned. Overall the test was considered successful in that it demonstrated a shortfall in a major component’s design (the Prony brake) and a rough indication of the wind speed and $\omega$ at which the rig’s natural frequency was reached. A valuable lesson learnt from this was that when conducting field tests with a device that is driven by the wind and rotates, it is prudent that some form of wind and $\omega$ measurement is utilised to analyse the parameters present when the device’s natural frequency is reached. This will assist in any design calculations that may need to be made or adjusted to ruggedise the device against failure from resonance.

As a result of the field test the Prony brake assembly was re-analysed. It was determined that the original design loadings that determined the brake’s hub diameter and braking material were far too high. The analysis also determined that the actual or modeled performance of the blade to be tested should be used to determine the hub diameter and type of braking material used. The reason for initially using high design loads (torques) was to ensure that blades of any performance parameters with diameters up to 2.5m would not cause the test rig to fail. However it was now obvious that for each blade to be tested, a hub and...
Fig. 22. Blade balance test.  

brake material that would allow it to function under loads relevant to its specific performance parameters would be required. This resulted in the wheel rim and brake being set aside and a new pulley and brake being sourced. Using the test blade’s performance curve from Fig. 1, it was determined that under the loading it was designed to operate it would produce torques of up 12N.m, which was a magnitude less than that originally calculated (120N.m). To accommodate the reduced torque, a pulley with a smaller diameter was sourced. The SEIT Brake Dyno Lab kindly donated an ED Falcon brake drum that was suited for the task as it fit onto the shaft wheel studs and had a much smaller diameter than the original wheel rim. The new pulley is shown in Fig. 20. The second task was to find a more suitable brake material.

Testing the Prony brake in the field was very time consuming and involved logistics such as car bookings etc. It was therefore decided that a suitable location to test the brake whilst attached to the rotor shaft and tower mast would be required and as a result the Wind Turbine Laboratory (WTL) was created, see Fig. 20. The tower mast with shaft assembly was suspended in a position where the blades could rotate freely and tests could be conducted on the Prony brake without the need to leave the workshop or utilise ladders. From this point onwards all the tests were conducted in the WTL until the rig was deemed to be fully operational from a Prony brake and instrumentation perspective. One of the outcomes of the field test was that the rotors had to overcome a significant amount of startup torque to commence rotation. The resized brake drum and brake were designed to help overcome this, however it was also beneficial to investigate the test blade’s balance and the torque required to overcome the friction generated by the bearings on the shaft as this may reduce the required startup torque even further. To test the starting torque of the shaft a length of string was wrapped around it with a weight rack attached at its end, see Fig. 21 for the set up. Weight was added until the shaft began to rotate. The mass was measured at 1.2kg which meant a starting torque of 0.2N.m was required. To reduce this value, the bolts fastening the bearings to the shaft base plate were loosened and a second measurement taken. This reduced the starting torque to 0.09N.m. It was assessed that when the bearings were initially secured to the shaft mounting plate, they were not quite parallel to each other and by loosening them and spinning the blade they had self aligned. The bolts were then tightened and a final measurement taken with the final start up torque being 0.12N.m which was a significant improvement. To check if the blades were in balance a test bench was constructed from discarded material, see Fig. 22 for the set up. The test bench consisted of shock absorbing height adjustable feet so that it could be made level. A pair of metal sheets were clamped to the test bench so that their sharpened edges were facing upwards and the shaft could be placed on them and be free to roll in the direction of most weight. It was observed that the shaft did not roll in either direction and that when the blades were slightly rotated they always remained in position indicating that no balancing was required.

With the blade balanced and the shaft startup torque reduced a more suitable Prony brake material was researched. The original brake was fabricated from 1mm thick galvanized iron sheet with a braking material bonded to it, see Fig.23. This design was heavy and generated a large frictional force that the test rig could not overcome at any of the operational wind speeds. This in combination with the large diameter of the wheel rim and the wire rope used to secure it in place meant that more torque needed to be overcome than the rotor could generate. To improve on this a lighter, less frictional force generating brake, in combination with a reduced diameter pulley and the substitution of nylon rope for wire rope were required. The materials selected for testing as brake materials were; toothed gear belt, wire rope and leather. The nylon rope that
was used during the initial field test was ruled out as it melted when subjected to a short exposure to brake friction. To test each of the new brakes the WTL was utilised. A known weight was suspended from the left side of the brake with an electronic scale on the right which was fixed to the floor. The rotor would then be spun and the force generated by the brake noted. Using the rearranged form of Eq. (7)

$$f = \frac{1}{\theta} \ln \frac{t}{W}$$

an approximate value of the $f$ of the brake material was determined. The brake material with the lowest $f$ was the most preferable as it required the least startup and running torque.

The first brake to be tested was fabricated from toothed gear belt, see Fig. 24a. The gear belt brake was found to produce more frictional force than was expected and was rejected as a suitable braking material. The wire rope was then trialed, see Fig. 24b, and was found to produce much less frictional force than the gear belt brake. The final brake was made from leather, see Fig. 24c, and when tested was found to produce a frictional force in between the previous brakes. The wire and leather brakes were tested again in the field to determine which contributes better to accurate data production. Leather was eventually chosen for data collection due to its mid range $f$ and ease of manufacture.

The next tasks to be completed were; the determination of the anemometers’ startup wind velocity and their calibration. To achieve this, the high speed Student Workshops wind tunnel was utilised in conjunction with an FC0510 Micro manometer with an assessed accuracy of ± 0.04m/s. To test the anemometers in the wind tunnel a mount was fabricated with a similar shape and lateral dimensions to that of the profile anemometer tower, see Fig. 25 for the wind tunnel test setup. This was undertaken to ensure that the calibration was as accurate as possible compared to actual field use. The only outdoor field affect that was not simulated was that of a turbulent boundary layer which the profile anemometer would encounter during operational use. However it is assessed that due to the relatively small diameter of the anemometer’s blades that the change in velocity across the rotors due to turbulence would be negligible.

As discussed in the design of the profile anemometer, all efforts were made to ensure that the anemometers started spinning at as lower value of wind velocity as possible. To determine the start up velocity, the wind tunnel was set to 1m/s according to the FC0510. The wind speed was then increased in increments of 0.5m/s until the blades started to spin. Once they commenced spinning they were stopped and the wind velocity reduced to the previous, non spinning, value. The wind velocity was then increased in increments of 0.1m/s until the blades spun. The final value, which was consistent for all six anemometers, was 2.5 ± 0.04m/s.

The output of the anemometers was a voltage range of between 0V and 3V for a wind velocity between 0m/s and 20m/s as previously discussed. To determine the voltage value for a particular wind speed a calibration test was required. The anemometer being tested was interfaced with a National Instruments USB-6009 Analogue/Digital I/O device via the cable that would transmit the signal in the field. The cable was used to ensure that an accurate
resistivity was taken into account. The USB-6009 was interfaced to a laptop with Windows XP as the operating system and Matlab’s Simulink as the capture and analysis program. Initially this setup created a lot of ‘noise’ within the circuit with the voltage varying by approximately ± 50mV for a particular wind speed. It was assessed that this level of accuracy was unacceptable and a way of reducing the noise should be sought. After discussions with staff from the electronic workshops it was determined that an RC filter should improve accuracy. An RC low pass filter was built, reducing the voltage accuracy to approximately ± 5mV, which was assessed to be acceptable. The calibration commenced at each anemometer’s start up wind velocity and was initially conducted in increments of 0.5m/s. This was done until it was determined that the voltage increase relative to wind speed was fairly linear, therefore, for the remaining calibrations the increments were of 1m/s.

The voltage produced for a given wind speed was captured within the Simulink simulation environment and outputted, via Matlab, to Microsoft Excel. The data captured was a thirty second sample of 1000 data points that occurred after the anemometer had reached a relatively steady state. These data points were then averaged to determine the final voltage value for wind speeds up to 20m/s. The data had a linear trend and is represented by the graph of a typical calibration, up to 10m/s, at Fig. 26. These values were entered into a lookup table and used in Simulink when capturing the wind speed data in the field.

The optical capture device was calibrated and tested whilst attached to the tower mast in the WTL as seen in Fig. 19. The blade was spun with a known angular velocity and a reasonable result was sought within Simulink. The optical device itself did not require calibration as it only produced a pulse when cut by the reflector. The fine tuning came with ensuring that the angular velocity was multiplied with a suitable constant to ensure the final SI value was consistent with the sampling rate of Simulink, which in this case was sampling at a rate of 1000 samples/s requiring a constant of $2\pi/0.002$. In basic terms each pulse would start and stop a clock within Simulink with the time between pulses constituting one revolution. A slight lead and lag either side of the pulse resulted in some error which was calculated to be no greater than ±2% of the time value between pulses when the turbine was spinning at its maximum angular velocity ($\approx 700$ rpm).

The load cells were then calibrated to determine their output voltage for a given excitation voltage and weight attached. In the case of both the thrust and torque load cells an excitation voltage of 5V was required. Lab weights ranging from 0.1kg up to 10kg and their accompanying stand were weighed using the student workshop scales to determine their actual mass. The scales were accurate to ±0.05g meaning that the lab weight values were accurate to the same degree. These weights were then numbered and catalogued and used exclusively for the field operation of the test rig. Each load cell was set up in turn on a makeshift testing rig and their resting voltage determined. In each case, with no weight attached, they registered a zero voltage. Weights up to 30kg were applied in increments of 0.5kg up to 10kg and then increments of 1kg up to 30kg. In each case a 3 second sample of the voltage, measured at 1000 samples/s, was captured in

![Graph of Voltage Vs Wind Speed](image)

**Figure 26.** Calibration graph of anemometer 4 showing a linear trend up to 10 m/s.

![Graph of Voltage Vs Weight for the Torque Load Cell](image)

**Figure 27.** Calibration graph of the torque load cell showing a distinctly linear trend.
Simulink. The first second was discarded due to a 0.5s response time resulting from the application of a digital low pass filter to the signal in Simulink. The remaining 2s worth of samples were averaged with the result being noted against that particular weight or combination of weights. From these values a graph was produced (see Fig. 27) and like the anemometers, was linear in trend. The data from each of the load cell’s graphs was then placed in a lookup table within Simulink. Each of the load cells had an error value assigned within their data sheets, however the electrical signal produced by the setup used in the calibration had a larger error of ±0.5mV. This larger value was used for the purpose of error analysis as it was much larger than the quoted error in the data sheets. Fig. 28 shows the torque load cell and Fig. 29 the thrust load cell in their final configurations.

As discussed previously the wind vane that was used for the test rig’s operation was designed, built and calibrated two years previously. During the current testing phase it was found to be in disrepair and was quickly fixed by the electrical workshops. The vane was then tested by simply turning it about its axis to known angles from its starting point and, as was determined during its original calibration, was accurate to an error of ±2°.

With the completion of all calibration and integration of the instrumentation a detailed error analysis was conducted and is contained within Appendix D. This analysis took into account every aspect of the measurements used in calibration and value determination as well as the manufacturer’s data for the relevant devices, to produce an overall error to be assigned to the test rig for the values of $C_p$, $C_Q$ and $\lambda$ that it produces. These values are: $\Delta C_p = \pm 0.001$, $\Delta C_Q = \pm 2.25 \times 10^{-4}$ and $\Delta \lambda = \pm 0.02$. This analysis also helped to determine the appropriate significant figures for each of these values which were applied during the data collection phase.

Once the instrumentation had been calibrated and interfaced with the test rig it was decided to conduct a second field test. This test would ensure the functionality of not only the instrumentation but any modifications that had been made as a result of the first field test, such as the new braking configuration. This test included every aspect of the test rig such as the power supply, which was provided via a 12V car battery through an inverter and the use of the profile anemometer tower. The power supply was used to power the laptop and the power pack for the load cells’ double full wave rectifier’s excitation. It was originally intended that test data be captured, however due to incomplete Simulink capture code this could not be achieved. The Simulink code was later completed and tested with all instrumentation attached to ensure its functionality prior to the operation of the test rig in the field. The Simulink capture architecture can be found with the raw data at Appendix E.

The test rig was allowed to spin up with the leather brake band which provided good sensitivity for torque measurement and was durable enough to withstand continued operation with negligible deterioration. It did however produce slightly too much frictional force which impeded the blade’s ability to start up in the lower wind range. This was rectified by the construction of an angle reduction arm that reduced the angle between each end of the brake band, see Fig. 30. Previously, the angle had always remained at 180° as shown in Fig. 13. If we revisit Eq. (7) it can be seen that by reducing $\theta$, the relationship between the tension and weight side of the brake system is reduced. Once the weight is subtracted from the tension force (which is now reduced because of...
the smaller angle) and multiplied by the brake pulley’s radius it shows that a reduced torque is required to achieve the same $\omega$ of the shaft. This significantly improved the shaft’s ability to startup in the lower wind range. It should be noted that all the material used to build the arm was recycled from scrap.

It was also noted during this test that the brake hub diameter may limit the data range that can be collected for a given range of wind speed. When operating a fixed pitch rotor such as the test rotor, it is likely that to obtain data across the entire performance spectrum of the blade, gearing or varying hub sizes may be required. This allows the device to spin over a much greater range of $\omega$ for a limited range of $v$ thereby increasing the range of $\lambda$ encountered. This is discussed in detail in Ref. 16. It was anticipated however, that the single brake hub used during the data collection phase would allow enough data to be collected to determine the test rig’s accuracy in performance determination. Another observation was that there was no vibration observed throughout the test rig’s range of operation. It is assessed that the natural frequency encountered during the first test was due to the weight of the wheel rim. The rim’s replacement with the brake disk is likely to have increased the natural frequency of the test rig meaning that to achieve resonance it would have to operate at a frequency higher than that achievable with the current test rotor.

The final issue that was encountered was the production of erroneous signals from the optical sensor due to sunlight shining on the face of the brake hub. This reflected sunlight was enough to produce voltage pulses significant enough to produce larger than reasonable values of $\omega$. This was later rectified to a certain degree by the addition of sun blinds made from discarded cardboard and the painting of the brake hub face with mat black paint. When tested outside the workshops to see if these measures rectified the sunlight issue it was found that there was still significant voltage reduction because of the sun. However these measures had reduced the sun exposure enough so that some minor adjustments made in Simulink to filter the extra signal rectified the problem.

Once all the above issues had been rectified the test rig was deemed ready to operate in the field. Fig. 31 shows the ideal final configuration of the test rig. The remaining issue at that stage of the project was suitable weather conditions in which to operate the rig. As the test rig had not been ruggedised for wet weather conditions it could only operate on a fine day with winds less than the safety limit of 20m/s. This day came on the 24 Sep 09 with the test rig’s operation and results discussed in the following chapter.

V. Test Rig Operation and Data Collection

Once the test rig was deemed ready to operate in the field, the only task that remained, that would satisfy the aims of the project, was to operate it with a blade with known performance parameters and collect data that predicts these performance parameters with an acceptable level of error. The purpose of this chapter is to discuss the operation of the test rig including any issues that arose, its final configuration and the data collected.

On the 24 Sep 09 the Bureau of Meteorology predicted a fine day with prevailing winds of between 25km/h – 35km/h from the west. Prior to setup the wind direction was checked and found to be coming from West North West (WNW). The test rig was set up at 0800 hours with a WNW aspect to make full use of the prevailing winds. The rig’s set up took one person approximately an hour, this included the erecting of the anemometer tower. It should be noted that the rig can be safely and successfully operated by one person in the field. Throughout the day the configuration remained unchanged apart from the

![Figure 31. Test rig configuration during the final field test. Note the profile anemometer on the far left, the work bench to its right and the wind vane on the roof of the car.](image)
change of the test rig’s orientation as the wind moved from west to north. Fig. 32 depicts the setup on the ground from a birds eye view, this will orientate the reader to the test rig’s operational configuration which may not be apparent from imagery. Fig. 33 is an image of the operational configuration on the day. When comparing Fig. 31 to Fig. 33 it becomes apparent that the test rig was not in an ideal configuration. This was due to the operator needing to center the wind vane each time a packet was recorded. Once the rig was setup data acquisition commenced. Initially there were some instrumentation problems, however these were soon rectified through simple fault finding. It should be noted that the profile anemometer was aligned with the test rig by sight and that an error of ±10° was considered when conducting the error analysis.

The collection of data occurred in packets of up to 5 minute samples. This time limit was set because of the amount of data that accumulated in one packet. For the purposes of data collection Simulink was capturing data at a rate of 10 samples/s. This resulted in a massive amount of data per packet when taking into account the seven instruments producing data simultaneously. By limiting each packet to five minutes the data became more manageable and limited the file size for storage purposes. In regards to the method for collecting data a simple procedure was used and repeated for each packet. After a particular weight was selected and entered into Simulink, the blade would be allowed to spin up without load. Once the blade was rotating at a suitable speed the load would be placed on the brake and Simulink started. Once the blade stopped rotating or approximately five minutes was reached, Simulink would be stopped and the outputted array catalogued and stored. A new weight would then be selected and the process repeated. This was to occur throughout the day until the operator was satisfied that a broad range of tip speed ratios had been captured.

In regards to the thrust measuring load cell, the thrust produced by the wind against the tower was measured as was the resting weight of the cables on the load cell. To determine the resting weight on the load cell Simulink was run for five seconds when there was no detectable wind speed. This data was then averaged for a zeroing value that could be entered into Simulink to ensure that the thrust load cell data always started with a zero load. To determine the thrust produced by the wind against the tower the rotor was braked and thrust data collected over the range of wind velocities encountered during the test rigs operation. These values were later subtracted from the thrust data manually during the data organization and analysis phase.

The data capture process continued throughout the day with 10 packets of what was deemed to be quality data being captured before the operation was halted. Towards the end of the day it was becoming apparent that the prevailing winds were remaining within a constant velocity range, that being 8m/s – 10m/s and that this would continue for the rest of the day. This velocity range meant that only a limited load range could be placed on the test rig of up to 1kg so that the blades could continue to accelerate. Loads greater than this caused the test rig to decelerate for the duration of the packet’s capture, producing data outside the range of the performance curve i.e. negative values of torque and power. Had wind velocities up to 20m/s been experienced a greater range of loads could have been applied with a resulting extended range of $\lambda$ and $C_p$ values being captured.

Of particular note was the possible feathering of the test rotor once a critical value of $\omega$ was reached. The intended value of $\omega$ that the test rotor was designed to feather at was approximately 70rad/s. Using Simulink’s ability to display real time data as it is being collected it was determined that the test rotor was indeed decelerating once it reached a critical value. This value would fluctuate slightly between 60rad/s – 70rad/s throughout the day, however it is assessed that the position of the feather weights on the test rotor may not have been correctly positioned during the construction phase. The feathering of the test rotor will be discussed in detail in the following chapter however of relevance now is the possible limiting of the $\lambda$ range by this occurring. From a simple calculation using Eq. (4) with an $\omega$ of 65rad/s an $R$ of 1.235m and a $v$ of 10m/s it can be seen that the data range will be limited to a $\lambda$ of approximately 8 which according to the test rotor’s curve at Fig.1 is just short of its peak performance. Obviously this issue is blade specific and not a problem with the test rig, however as the blade in question was being used to validate the test rig it had the potential to hinder the validation process. Luckily this did not occur and the reasons why will be discussed in the next chapter.

Another issue that arose that had the potential to shut down the data collection prematurely was the exhaustion of the power supply. The 12V car...
battery lasted less than two hours with intermittent use which was inconvenient as the test rig needs to operate for up to several continuous hours. To complete the data collection a car had to be used to continue the power supply to the instrumentation. An appropriate resolution to this problem will be discussed further in the recommendations section.

Overall the test rig operation and data collection phase was successful. The primary aim of collecting data that would validate the rig’s accuracy was achieved and any final mechanical issues fixed. Due to time constraints, caused by the scheduled project completion date, this would be the only opportunity to collect data meaning that only a limited range of data was captured for analysis. There was however, enough data captured to paint a fairly accurate picture of the test rig’s ability to predict blade performance and this is discussed in detail in the following chapter.

VI. Data Organisation & Analysis

Once the data had been collected the final task was to analyse it and place it into a form that best represents the performance of the test rotor. To do this the data had to be organized into a useable form from which the relevant variables were used to calculate the required parameters. These parameters could then be combined and further reduced through the application of the error analysis and the averaging of related scatter. Once the data was in its final form it was graphed so that analysis could be conducted and the questions of the test rigs accuracy and the reduction of scatter could be answered.

The final output of the test rig when operated in the field is a set of arrays within Matlab which contain the raw data collected from the instrumentation including a timestamp for each sample. Using Matlab commands these arrays were converted into an excel file which could then be used to organize the data into separate packets depending on which weight they were obtained from and analysed separately. This made analysis more manageable considering the weight of data that needed to be reviewed. It should be noted that Simulink was configured to output the final performance parameters as well as the raw data from the instrumentation, however after some preliminary analysis it was determined that it was more beneficial to use the raw data from the start. Using the final parameters was difficult when trying to determine why a particular value was unreasonable whereas by using the raw data this could be established immediately. The raw data values that were measured by the test rig were: \( t \), \( T \), \( V \), \( \omega \) and the angle at which the wind approaches the rotor’s swept area.

Once the data was organized the relevant performance parameters and their supporting variables were calculated using the raw data, the first of those being \( Q \). As this value is derived only from the \( Q \) produced by the shaft it is not the total value of \( Q \) produced by the test rig. Also to be considered is the \( \dot{\omega} \) produced by the torque of the test rotor in combination with its \( J \). This blade \( Q \) and it contribution to the overall systemic \( Q \) is discussed in Ref. 2 where the value of \( J \) for the test rotor was given as 1.15m². This yields the relationship

\[
Q_{\text{total}} = Q + J \omega
\]  

(9)

which yields the total \( Q \) used to calculate \( C_P \) and \( C_T \) in combination with \( V \) and \( \omega \). To calculate \( \dot{\omega} \) finite difference is used which is basically \( \dot{\omega} = \Delta \omega / \Delta \text{time} \) which was easily applied to all the data using within excel. Applying the value of \( \dot{\omega} \) to Eq. (9) in combination with the values of \( Q \) and \( J \) the \( Q_{\text{total}} \) can then be determined. With \( Q_{\text{total}} \) known Eq. (3) can be used to calculate \( C_Q \) which is then multiplied by \( \lambda \) to yield \( C_P \). This process was repeated for all 10 packets of data.

In the case of the \( T \) produced by the test rotor it was measured directly via the \( T \) load cell. This load cell was attached to the test rig’s front cabling assembly which was aligned 60° to the horizontal (see Fig. 29) between both its attachment points. To determine the actual \( T \) produced by the test rig it was multiplied by the cosine of 60°.

The next step was to answer the question of scatter reduction through the use of the contributing component of the velocity vector. To do this the same process used to determine \( C_P \) was used, however the angle the wind approached the test rotor’s swept area was utilised to determine the actual complete velocity vector. This vector was then used instead of its useful component and his new value of \( C_P \) compared with the original.

The next step was to eliminate any data that was irrelevant or did not contribute to the test rig’s ability to predict blade performance. The first set of data to be eliminated was that which occurred in the first second of capture. As discussed previously most of the instrumentation had a 0.5s – 1s response time which rendered any data within this time frame unusable. Data that was deemed to be unreasonable was then removed, this mainly included overly fast values of \( \omega \) which were caused by additional signal pulses from direct sunlight. These were easy to detect as they would reside amongst reasonable data that helped to highlight their presence. Any data that produced negative values of \( C_Q \) and \( C_P \) was discarded as this was indicative of large values of \( -\dot{\omega} \) which was assessed to activate the feathering of the rotors at values of \( \omega \) approaching 70rad/s. Another possible cause of the \( -\dot{\omega} \) was that too much load was being applied to the test rig under the given wind conditions effectively halting the rig’s ability to accelerate or even maintain a constant velocity.
Figure 34. Scatter graphs of $C_p$ Vs $\lambda$. (a) shows the raw data without modification (b) is the data after it has been averaged and significant figure reduced.

Once the non contributing data was discarded each of the packets were combined into their relevant areas so that further graphical analysis could be conducted. These areas were; $C_p$ versus $\lambda$, $C_Q$ versus $\lambda$, $C_T$ versus $\lambda$ and $C_p$ versus $\lambda$ using the unresolved velocity vectors. The first three groups represent the performance data and the fourth the question of whether data scatter can be reduced. This data can be viewed in Appendix E. Within each of the specified areas the data was further reduced to their respective significant figures, as dictated by the error analysis contained in Appendix D and then averaged.

The main parameter used within the wind turbine industry to quantify the performance of a particular blade, rotor or turbine is that of $C_p$. With that in mind the most important relationship produced by the test rig is that

Figure 35. Graphs of $C_p$ Vs $\lambda$ comparing the test rotors actual performance curve to that predicted by the test rig. Error bars have been included to highlight the test rigs assessed accuracy.
between \( C_p \) and \( \lambda \) which is primarily used to describe the performance of the test rotor at Fig. 1. This section will examine the resulting graphs depicting this particular relationship and discuss the question of the test rig’s accuracy.

Fig. 34a is a graph of \( C_p \) versus \( \lambda \) where the raw data has been utilised to produce the graph’s scatter points. It can be seen in the graph above that even though the data is relatively raw and yet to be organized that an expected trend is readily apparent when compared to Fig. 1. The graph supports the earlier assumption that the data collected only represents \( \lambda \) from 0 to approximately 8. The graph also supports the assumption that the rotor feathering was responsible for the limited amount of data at the top end of the curve. It can be seen at Fig. 34a that there is a fairly even spread of data along the graphs gradient as it approaches the peak, however once the \( \lambda \) approaches values commiserate with \( v \approx 10\text{m/s} \) and \( \lambda \approx 8 \), which yields an \( \omega \) of between 65rad/s – 70rad/s, that the data becomes quite sparse. This will be due to feathering forcing the values of \( C_p \) to become negative through large values of \( -\omega \). However, as stated previously the data is as a result of the test rotor and not the test rig. It is assessed that there is enough data to detail a trend that is similar enough to the test rotors actual measured performance to evaluate the test rigs accuracy. Fig. 34b is a graph of the same data once it has been resolved into its correct significant figures and then values of \( C_p \) averaged along similar values of \( \lambda \). This produces a curve that is much representative of the power produced by the test rotor and through the 3rd order polynomial fit allows us to see its general trend without the influence of the rotor feathering.

Using excel the equation of the polynomial curve can be acquired, which is displayed in Fig. 34b. This equation, in conjunction with the data from the test rotor’s performance curve at Fig. 1 was then used to plot a comparison graph using Matlab, see Fig. 35. From the plots at Fig. 35 it can be seen that the test rig’s curve represents the general trend of the test rotor’s curve. Even with the error/accuracy of the test rig taken into account however, there remains a significant difference between the two curves. It is assessed that there are several reasons for this occurring, the first of which was the feathering of the rotor as previously discussed. The feathering only affected the top end of the curve and doesn’t explain the difference in the mid range data. However, had there been more data at the top end of the curve showing the peak performance and the point at which the curve flattens out and then proceeds back towards zero, a different trend may have developed. Had this occurred, it is likely that the test rig’s curve would have more closely reflected that of the test rotor. It should be noted that to obtain the test rotor performance curve a geared turbine system was used. With the use of gearing within the turbine drive assembly higher values of \( \omega \) could be obtained for lower wind speeds whereas the test rig had a fixed brake hub size for torque measurement limiting it to a certain range of operation within the wind limits encountered. With the ability to obtain higher \( \omega \) for the same or lesser wind speeds, means higher values of \( \lambda \) can be obtained resulting in the ability to explore the peak and down slope sections of the performance curve. Unfortunately this could not occur within the time budget of this project, however it would provide a solid basis for a future research which will be discussed further in the recommendations.

The final reason that may account for the difference in curves was that the test rotor’s feathering device was incorrectly positioned during the conduct of Ref. 2. This meant that not only was the test rotor able to spin past...
its designed feathering speed it may have been encouraged to increase its \( \omega \) by the orientation of the feathering weights. This would have assisted in the test rotor achieving its peak performance when the test rotor performance curve data was being obtained. For safety reasons the feathering weights were placed in their correct positions prior to the test rig’s operation. This meant that they were functioning as intended for the data collection phase. This could mean that with the feathering system functioning properly the test rotor may not be able to reach its peak performance even with gearing, unfortunately this could not be investigated further due to time constraints.

According to Ref. 16, even though the graph of \( C_Q \) versus \( \lambda \) doesn’t give the reader any more useful information than the \( C_p \) versus \( \lambda \) curve about the rotor’s performance, it does give useful information in regards to torque assessment when dealing with gearboxes and generators. Therefore this plot is also of great value, in the context of the test rig’s purpose, as it gives the blade end user an idea of the torque levels that their turbine system will encounter so that, for e.g., the drive shaft can be designed with the correct diameter to absorb the loads produced by the torque without shearing. As per the \( C_p \) curves above, Fig. 36a and Fig. 36b are respectively the raw data scatter and organized data scatter graphs for the \( C_Q \) versus \( \lambda \) curves with 3\textsuperscript{rd} order polynomial fits to display their general trends. Another purpose for these curves was to validate the test rig’s design methodology by showing that at least for this blade the design parameters and safety factors were not breeched by its maximum value of torque.

Once again the range of data available was limited by \( \lambda \)’s confinement to 0 – 7.5. The trend in Fig. 36a is not immediately apparent without the benefit of the poly fit, however with some organization it becomes more apparent in Fig. 36b. As per the data used to produce the \( C_p \) curves the peak values of this performance curve may have been affected by the blade feathering. However these curves give the reader a better idea of the approach to peak performance as the adjusted curve can be seen approaching a gradient of zero.

![Graph Comparing Test Rotor Torque Performance to Test Rig Torque Performance Prediction](image)

**Figure 37.** Graphs of \( C_Q \) Vs \( \lambda \) comparing the test rotors actual torque performance curve to that predicted by the test rig. Error bars have been included to highlight the test rigs assessed accuracy.

From Fig. 37 it can be seen that the test rig data provides a similar curve to that of the test rotors. However, once again there is a significant difference in values as the test rig’s data is less than that of the test rotor. The possible reasons for this have been explained above for the \( C_p \) versus \( \lambda \) curves and as such had a larger range of wind velocities been encountered a larger range of values may have become extant giving a better picture of the performance curve. The total torque produced by the test rig is a combination of the torque produced by the blades and that produced by the shaft due to the wind’s energy on the blades. The -\( \omega \) due to feathering or the slowing of the blades is a torque related issue which limited the power produced by the rotor. This issue could have been alleviated through the use of a smaller brake hub or some kind of gearing system, but as stated previously it is more likely to be a blade issue than a test rig issue.

The value in being able to determine the thrust produced by the rotor in a wind turbine system is in its use for a tower’s structural design. Thrust forces can significantly affect a wind turbine tower through large
moments produced at the base of a potentially long lever arm. The $T$ produced by the test rig was measured directly by a load cell. Using Eq. (1), $C_T$ was calculated and the resulting data plotted against $\lambda$. Fig. 38 is the graph of $C_T$ versus $\lambda$.

![Graph of Coefficient of Thrust Vs Tip Speed Ratio](image)

**Figure 38. Scatter graph of $C_T$ Vs $\lambda$.**

Ref. 16 states that the greater the solidity of a rotor the greater the $T$, the solidity being the number of blades. This is the reason for designing the test rig using ADT as it determines the maximum thrust assuming infinite blades giving the designer the worst case scenario. From the measured data it was ascertained that the maximum thrust produced was approximately 200N, the tower was designed to withstand 1200N with a safety factor no less than 3. The test rotor, having two blades, had a relatively low solidity, however it was designed to accept rotors of all designs including those with more than 2 blades which will result in greater levels of thrust. The other factor is that the test rig has not yet operated in winds up to its safety limit as this will also contribute greatly to the magnitude of the thrust produced.

The graph of $C_T$ versus $\lambda$ shows that the thrust fluctuates up and down with increasing values of $\lambda$, however it also shows a general trend of increasing $C_T$ with increasing $\lambda$ which was to be expected. The graph itself should only be used as a general guide as it was noted during the operation of the test rig that the steel cables were able to stretch slightly and in doing so acted as springs. This allowed the tower to rock forwards when a wind gust eased. This would have contributed to the abrupt rises and drops on the graph above. It was also noted that after a certain undetermined period of operation that the original zeroing of the load cell needed to be readjusted possibly as a result of the cabling loops settling. This may account for the rise of the graph from a $\lambda$ of 2 – 4, then its abrupt drop and then gradual rise from a lower value then where it had started. The smoother data from $\lambda \approx 4$ is probably indicative of constantly increasing wind velocity which would have resulted in a constant increase in $\omega$ and $\dot{\omega}$. These constant increasing values without the presence of gusts should result in a relatively smooth increase in thrust.

The final phenomenon that was to be investigated was to determine if there was any extra value in only using the component of the velocity vector normal to the rotor’s swept area. To achieve this a wind vane was incorporated.

![Diagram of the resolution of the useful component of velocity into the velocity vector](image)

**Figure 39. Diagram of the resolution of the useful component of velocity into the velocity vector.**
into the test rig’s configuration with the sole task of determining the approach angle of the incoming wind. As the anemometers were fixed in the same direction as the axis of the rotor they would only determine the velocity affecting the blades and not the entire magnitude of the velocity vector. With the incoming angle of the wind vector known its magnitude could be determined simply by dividing the profile anemometer’s velocity by the cosine of the angle, see Fig. 39. A standard wind turbine, through the use of a motor or guide vane changes its direction to always be facing into the incoming wind direction. When the wind’s vector is constantly changing the result is that for a significant portion of the wind turbine’s operation it is not seeing the entire vector of the approaching wind which can lead to inaccuracies in the predicted power output. This question of using only the useful component of the velocity vector aims to determine if a blade’s power output for a given time period can be more accurately determined through data scatter reduction.

The graph at Fig. 40a shows the data trend using the useful component of the velocity vector to determine $C_p$ versus $\lambda$ with the graph at Fig. 40b using the unresolved velocity vector. By comparing these graphs and disregarding the polynomial fits it can be seen that Fig. 40a depicts a tighter collection of data that gives a better representation of the graph’s trend, almost to the expected location of the performance curves peak value. Fig. 40b however, has a more dispersed scatter and only depicts the data trend to ¾ that of the other graph. This shows that there is significant benefit in only measuring the useful component of the wind’s velocity vector when trying to determine an accurate value of a turbine’s performance or power output.

In regards to the question of the test rig being able to accurately gauge a blade’s performance it is assessed that it is capable. The question of how accurate the prediction is, in regards to $C_Q$ and $C_p$, needs to be resolved however, and this may require further investigation using another test rotor that hasn’t had its configuration changed since its performance was measured. In regards to the measurement of thrust it is assessed that the system is capable of producing a rough estimate and this may be improved through the development of a more robust thrust measurement assembly. The above data has shown the benefit in using the useful component of the velocity vector to determine an accurate measurement of a blade’s ability to produce power over a given time frame. In line with this, the determination of how much performance and power output is actually attributed to a rotor as apposed to its quoted value would make an interesting topic for future research. Any suggestions made within this or any other chapter will be discussed in detail in the recommendations section of the following chapter.

**VII. Conclusions**

To conclude, this project achieved all of the goals and aims that it set out to accomplish. The main intent of the project was the design and construction a wind turbine test rig that is capable of measuring the performance of any wind turbine blade, up to 2.5m in diameter. As this kind of endeavor had not been previously documented the design process had to start from the ground up utilising a range of the basic theories used for predicting turbine performance in conjunction with some of the more fundamental component and mechanical engineering design theories. The end result was a test rig that can assist the blade designer with performance

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**Figure 40. Graphs comparing $C_p$ Vs $\lambda$ for resolved (a) and unresolved (b) velocity data.**

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determination under real world weather conditions. It should also be noted that the desire to fabricate the test rig predominately from discarded or recycled materials was also realised ensuring that the project maintained a ‘renewable’ philosophy throughout.

There is a multitude of literature available to designers to support their efforts in each of the design steps such as math modeling, computational modeling and scale modeling. There isn’t much advice on how or what to use to test a prototype blade to determine, as accurately as possible the blades actual performance prior to mass production. Hopefully the research conducted for this project will help to fill the information void in that respect. As a minimum it is hoped that this research will provide a starting point for other similar endeavors.

To support future work in this field this report documents the step by step design calculations, decisions and motivations that lead to the final physical structural design of the test rig. The final design proved itself to be robust and capable of safely operating up to its designated safety limits as was proven during the first field test. The report also details the various lessons learnt throughout the duration of the project in regards to poor procedures, equipment not appropriate for its designated task and components that need further modification to provide better results. In doing this it is hoped that future research in this area will learn from mistakes made during this project and prevent others from reinventing the wheel.

The project underwent a prolonged testing and calibration phase as various issues and faults became apparent. This phase provided an excellent chance to learn and to ensure that the test rig was going to function as intended when used in the field. Both of the field tests proved invaluable as significant design changes resulted from each, making the operation of the rig a success.

The actual operation of the test rig proved to be non problematic with only a couple of minor, easily rectified issues occurring. Within a relatively short timeframe enough data was collected to conduct an analysis on the test rotor’s performance and determine the test rig’s accuracy in this regard. An initial perusal of the data in the field showed it to be reasonable which signaled success for the operation and data collection phase of the project.

The final phase of the project was that of data analysis. For the test rig to be deemed useful it was required to produce certain performance parameters with a certain level of accuracy. Through detailed analysis of the captured data it was determined that its range was limited due to the feathering function of the test rotor and a small range of wind speeds available on the day of operation. This did not prevent enough data being captured however, to show that the test rig is capable of determining performance. To predict the performance fully though it needs to be operated over a larger range of wind conditions and unfortunately a lack of time and logistical issues prevented any further data collection. The data from the test rig was also able to show that by using only the useful component of velocity the data’s scatter can be reduced, better depicting the rotor’s performance trend as apposed to using the velocity vector.

Finally, it is assessed that the project was a success in that its stated aims were met and that the test rig is capable of determining a blade’s performance to a certain level of accuracy as determined by the error analysis. Obviously this research will benefit from further time being invested into more data collection and tweaking the system for better accuracy. Unfortunately this is outside the scope of this project now that its completion date is extant. It is hoped that research into this area and indeed the area of renewable energy will continue within UNSW@ADFA and that the test rig will provide a platform to assist in this endeavor. Suggestions for continuing research using the test rig are discussed in the recommendations section of this chapter as are any other recommendations arising from the project.

VIII. Recommendations

At the completion of the project numerous recommendations can be made in order to improve or continue the research. This section will discuss recommendations in three main areas, these being; improvement to the design, improvement to the operational methodology and suggested areas of research that could utilise the test rig.

In regards to improving the design several recommendations can be made. It was noted early on in the data collection phase that the brake hub diameter may have limited the data range for the given wind conditions on the day. Had a range of various hub sizes been available, in particular ones with smaller diameters than that used, then a larger range of data could have been obtained. Therefore it is recommended that should the test rig continue to be used then a range of brake drums of varying diameters be manufactured to accommodate differing wind and loading conditions. Another solution to this could be the design and manufacture or purchase of a gearing system. The research conducted at Ref. 2 utilised a gearing system which could be utilised for this purpose. Dr Harrap suggested the use of a tiered...
The optical device used to determine the rotor’s $\omega$ was obviously not the best device to be used in sunny conditions. As the test rig has been made to operate outdoors it is recommended that another method of $\omega$ capture should be investigated and integrated into the test rig. A possible type could be a magnetic reed switch which would be unaffected by sunlight.

The current system for measuring $T$ could be improved for a better indication of the force’s trend with increasing $\lambda$. It is recommended that a system that reduces the amount of stretch in the cabling be investigated or a system that is not so prone to the effects of the towers oscillations in gusty winds be adopted.

The method for aligning the anemometer tower with the test rig was not as accurate as it could be. It is recommended that the use of a sighting post be investigated. This method would be similar to that use by the military to ensure that indirect fire support weapons are all aligned in the same direction. This would reduce the scatter created by inaccurate wind velocity measurement.

In regards to the methodology used throughout the project, several recommendations can be made. These encompass operational techniques, data refinement and operational planning. As discussed above a single hub size has the potential to limit the range of data produced. If a range of differing sizes were produced then it is recommended that the optimal diameter of brake hub for a given wind range be researched and tabulated so that the appropriate hub may be selected for the given wind range on the day of operation. Simply put, this would be a form of manually gearing the test rig allowing for a greater range of $\omega$ for a limited range of wind speeds.

Even though enough data was collected to validate the rig’s ability to measure performance, it is recommended that another blade with known performance parameters and no feathering, be fitted and tested. This will hopefully allow the rig to operate over the entire performance spectrum giving the operator a better idea of its capabilities.

Now that the data produced by the test rig has been shown to be reasonable, it is recommended that the final parameters may be used instead of the raw data in the determination of the performance components $C_p$, $C_Q$, $T$ and $\lambda$. The raw data should still be collected so that any anomalies in the final outputs can be traced to their origins, however by using the final parameters much time will be saved during the data analysis phase. Simulink should be further expanded to eliminate the unreasonable data as discussed in the analysis chapter.

As discussed previously there are numerous areas of relevant wind energy research that can be complimented by the use of the test rig, several of which are recommended for further research below.

The most important area of continuing research would be in the area of blade design and improvement. The area of cheap blade material with high performance parameters would make for interesting and rewarding research.

In line with blade design could be the investigation of modifying the test rig to accommodate vertical axis blades for performance testing. When the rig was designed this scenario was considered and could be possible with the rig remaining on its back in the trailer. The torque measurement system would need some modification however to accommodate this configuration.

Research into the construction of an efficient generator using conventional washing machine or dryer motors could be investigated. The test rig could be modified either at its shaft assembly or within the trailers tray to accommodate a generator for this or related research.

The test rig could be modified so that a rotating assembly could be used to allow a turbine to rotate with the changing wind direction. This would allow research into how much of the winds energy is lost or not accounted for whilst the rotor is in transit between absolute wind vector directions.

The profile anemometer could be further researched into wind turbine site selection through the accurate prediction of boundary layer profiles over selected terrain. A specific topographical feature could be profiled and an adjustment factor calculated so that a single anemometer could be used to determine an accurate velocity value for an entire wind turbine rotor swept area.

These recommended areas of research are just a small indication of the potential of the test rig to support future research and it is hoped that that this type of research continues at UNSW@ADFA.

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References