Design, Build and Assess Composite Hub Bar for Gyroplane Application (Gyrate, 2008)

Darren S Lonergan

University of New South Wales at the Australian Defence Force Academy

Current rotor hub bar designs for recreational gyroplane aircraft have suffered a number of catastrophic failures. Investigations into why aluminium rotor hub bars have failed revealed that fatigue mechanisms, exacerbated by fretting and mechanical damage led to a number of premature failures and incidents of crack propagation. This project aimed to produce a replacement design utilising hand manufacture of composite materials, with properties that exceed those of current aluminium rotor hub bars. The design is intended to advance the safety of gyroplane operation. The project employed a methodical design process to produce a prototype hub bar design. Samples were manufactured and material testing carried out in order to establish the design’s potential structural properties. The experimental results show that the design requires further development and testing to fully comply with appropriate standards if it were to be a successful alternative to the current design.

Contents
Nomenclature .................................................................................................................................. 2
I. Introduction ........................................................................................................................................ 3
   A. Aim .................................................................................................................................................. 4
   B. Scope ........................................................................................................................................... 4
   C. Methodology ................................................................................................................................. 4
   D. Limitations ................................................................................................................................... 4
   E. Project Management ..................................................................................................................... 4
II. Background of Hub Bar Failure .................................................................................................... 4
III. Technical Brief ............................................................................................................................... 5
IV. Review of Previous Investigation .................................................................................................. 6
V. Gyroplane Theory and Dynamics .................................................................................................... 7
   A. Autorotation dynamics .................................................................................................................... 7
   B. Rotary Wing Complexity .............................................................................................................. 8
   C. Prerotation ................................................................................................................................... 9
VI. Design ............................................................................................................................................. 10
   A. Initial Considerations .................................................................................................................... 10
   B. Standards .................................................................................................................................... 13
   C. Design Goals ............................................................................................................................... 14
   D. Static and Dynamic load .............................................................................................................. 14
   E. Material ....................................................................................................................................... 17
   F. Form-Hub bar ............................................................................................................................... 18
   G. Form-Teeter Block ...................................................................................................................... 20

1 School of Engineering & Information Technology. Aeronautical Engineering: Project, Thesis & Practical Experience ZEIT 4500/4501.
H. Manufacturing Process ................................................................. 21
I. Hardware .......................................................................................... 23

VII. Testing ........................................................................................... 23
   A. Rationalisation .............................................................................. 23
   B. Test Equipment ............................................................................ 24
   C. Fibre Volume Determination by Immersion .................................. 25
   D. Experimental Results, Analysis and Comparison ...................... 25
      1. Three Point Bending .................................................................. 25
      2. Tension Test, Post Three Point Bending .................................... 28
      3. Torsional Stiffness determination .............................................. 29

VIII. Conclusion .................................................................................. 29
IX. Recommendations .......................................................................... 29
Acknowledgements .............................................................................. 29
References .......................................................................................... 30

Nomenclature

\[ b = \text{Width [m]} \]
\[ c = \text{Distance from the neutral axis [m]} \]
\[ C_l = \text{Elemental lift coefficient} \]
\[ C_d = \text{Elemental drag coefficient} \]
\[ D = \text{Drag [N]} \]
\[ DL = \text{Disc Loading, Gross Weight/Disc Area [kg/m}^2\text{]} \]
\[ D_R = \text{Rotor Diameter [m]} \]
\[ E = \text{Elastic (Young’s) modulus [MPa]} \]
\[ F_C = \text{Centripetal Force [N]} \]
\[ GW = \text{Gross Weight, maximum operating weight [kg]} \]
\[ h_t = \text{Teeter height adjustment [m]} \]
\[ I = \text{Second moment of inertia [m}^4\text{]} \]
\[ l = \text{Length [m]} \]
\[ L = \text{Rotor lift [N]} \]
\[ L_R = \text{Rotor Length [m]} \]
\[ m = \text{Mass [kg]} \]
\[ M = \text{Moment [Nm]} \]
\[ n = \text{Load factor} \]
\[ P = \text{Load [N]} \]
\[ PL = \text{Power Loading, GW/engine power [kg/kW]} \]
\[ r_{CG} = \text{Radius swept by rotor blade centre of gravity [m]} \]
\[ R = \text{Rotor disc radius [m]} \]
\[ S = \text{Swept area of the rotor disc [m}^2\text{]} \]
\[ t = \text{Thickness [m]} \]
\[ T = \text{Thrust [N]} \]
\[ V_c = \text{Free stream (wind) velocity [ms}^{-1}\text{]} \]
\[ V_i = \text{Induced velocity [ms}^{-1}\text{]} \]
\[ V_f = \text{Fibre volume ratio} \]
\[ V_m = \text{Matrix volume ratio} \]
\[ W = \text{Weight [kg]} \]
\[ W_e = \text{Aircraft Empty Weight [kg]} \]
\[ W_f = \text{Fuel Weight [kg]} \]
\[ x = \text{Displacement} \]
\[ \alpha = \text{Angle of Attack [degrees]} \]
\[ \beta = \text{Rotor blade angle of incidence [degrees]} \]
\[ \beta_p = \text{Pre cone angle [degrees]} \]
\[ \beta_0 = \text{Coning Angle [degrees]} \]
\[ \delta = \text{Displacement [m]} \]
\[ \theta = \text{Rotor blade pitch [degrees]} \]
\[ \rho = \text{Density [kg/m}^3\text{]} \]
\[ \sigma = \text{Stress [Pa]} \]
\[ \sigma_L = \text{Stress in longitudinal axis [Pa]} \]
\[ \sigma_T = \text{Stress in transverse axis [Pa]} \]
\[ \sigma_s = \text{Solidity of rotor disc} \]
\[ \omega = \text{Angular velocity [rad/s]} \]
\[ \text{TPP} = \text{Tip Plane Path} \]
\[ \text{AoA} = \text{Angle of Attack [degrees]} \]
\[ \text{AUW} = \text{All up weight, equivalent to Gross Weight [kg]} \]
\[ \text{FOS} = \text{Factor of Safety} \]
\[ \text{GFRP} = \text{Glass Fibre Reinforced Plastic} \]
\[ \text{GAG} = \text{Ground Air Ground} \]
\[ \text{LSA} = \text{Light Sports Aircraft} \]
\[ \text{Pax} = \text{Pilot and Passenger} \]
\[ \text{QFD} = \text{Quality Function Deployment} \]
\[ \text{VNE} = \text{Velocity Never Exceed} \]
\[ \text{FOS} = \text{Factor of Safety} \]

I. Introduction

A gyroplane is a rotary wing aircraft, also commonly referred to by the terms autogiro, autogyro and gyrocopter. The gyroplane has an unpowered rotor attached to a vertical shaft to provide lift to the aircraft. This is in contrast to the conventional helicopter which uses a powered rotor. The power is provided for forward flight by a pusher, puller, or combination of powered propellers. Lift is produced by the rotor disk being inclined at a positive angle of attack, as the aircraft is propelled forward, the resultant aerodynamic forces cause the necessary torque to spin the rotor which, in turn provides the lift required (Leishman, 2006). Figure 1 provides a basic graphic description of gyroplane fundamental principles of flight.

Most light recreational gyroplanes employ a teetering rotor head which has two blades connected via a hub bar, hinged at the rotational axis (i.e. on the rotor shaft). In contrast to the articulated rotor head employed by most helicopters, which incorporate a lead-lag hinge, flapping hinges and feathering bearing (Leishman, 2006). The focus of this project, the gyroplane rotor hub bar is part of this simple teetering rotor head assembly. The rotor hub bar is a flight critical component that provides a single load path between the rotor blades and the rotor mast spindle and will be described in greater detail in the body of this report. Figure 4 provides images of typical rotor hub bars.

The current aluminium rotor hub bar design currently in widespread use suffered a catastrophic failure in 2004 resulting in a fatality. The following investigations determined that fatigue crack development had been the cause of failure. Post this investigation a trend of apparent fatigue cracking had been established and all rotor hub bars were subsequently subject to a non destructive inspection program to allow them to remain in service if found serviceable.

An opportunity exists for the creation of a suitable replacement part, designed in a systematic way to enable safer and more reliable gyroplane operation.

Figure 1. Gyroplane Fundamentals (Leishman, Principles of Helicopter Aerodynamics 2006).
A. Aim
The aim of the project is to produce a composite gyroplane rotor hub bar design that will exceed current specifications, be more robust and reduce the possibility of further gyroplane accidents resulting from rotor hub bar failure.

B. Scope
The Scope of the project includes:
- Assessing the current state of the art, reviewing all available and relevant information pertaining to current design.
- The development of design requirements.
- Identification of relevant design standards and regulations.
- Development of a preliminary design.
- Investigation of pre-coning and optimal angles of precone integration.
- Geometric description (engineering drawing) of preliminary prototype design.
- Identifying a testing and validation plan.
- Material testing and analysis.

C. Methodology
To enable the project to meet its aim the author has employed the following simple methodology:
- Conduct a review of the issue of hub bar failures and the operation of gyroplanes generally including the principles of flight.
- Identify the static and dynamic loads the hub bar must structurally support.
- Identify the requirements that must be met to produce a successful solution, i.e. Applicable standards.
- Produce a design specification which outlines more specifically the characteristics which will govern the designs properties and form, i.e. Required strength properties, design dimensional constraints and the requirements for integration into existing aircraft systems.
- Propose a design solution by way of producing a set of design engineering drawings.
- Make improvements where necessary, as the design evolves. The design process is an iterative process involving a cycle of design, review and improvement.
- Manufacture samples to enable testing of the components, to determine relevant properties.
- Accept that the final finished product is beyond the scope of the thesis therefore identify a way forward.

D. Limitations
The project limitations include: a time frame of two semesters. The funds available to finance any required materials or external work were dictated by the UNSW. The majority of manufacturing and testing work was completed by the author and was therefore limited by the authors skills and available time, funding and facilities. The design is a case study design using a single typical gyroplane to base the design upon. This decision reflects the many various gyroplane configurations which each possess unique dimensions and geometry are not that dissimilar in their fundamental form, fit and function. The design aimed to be able to be simply adapted to extend to other gyroplane types with the majority of calculations embedded in software. During the two semester timeframe an engineering drawing will be produced, a manufacturing process developed, a sample rotor hub bar manufactured and the designs structural properties analysed.

E. Project Management
The project adopted a methodical engineering approach, integral to this was the project planning. Detailed planning assisted in meeting the desired aims within the allocated time frame. This was accomplished using management software and following a Gantt chart to review progress.

II. Background of Hub Bar Failure
The Hub bar is part of the gyroplane rotor head assembly. A catastrophic failure of a aluminium hub bar on Norley Station, near Thargomindah Queensland, in 2004 resulted in a fatality. It was found by an investigation by the Australian Transport Safety Bureau (ATSB) that the failure was due to fatigue cracking. Following this, “several subsequent gyroplane accidents and incidents that involved cracking or failure of the hub bar were reported to ATSB and/or the Australian Sports Rotorcraft Association (ASRA)” (Blyth 2006). This raised awareness of an existing issue and subsequent investigation identified a trend across the gyroplane community of apparent gyroplane hub bar cracking. Prior to the fatal catastrophic failure, the significance of hub bar cracking had not been realised by some operators and a number of incidents of cracking had gone unreported.
III. Technical Brief

In Australia the gyroplane is a popular ‘light sports aircraft’ (LSA) (Vaughan 2006, 2), part of the recreational category. The gyroplane is employed for recreation and in rural operations such as aerial observation, mustering and livestock inspection amongst other various roles. The leading Authority on the operation and use of gyroplanes in Australia is the Australian Sports Rotorcraft Association (ASRA). The mission statement claims:

“ASRA is concerned with the improvement of standards of safety, of pilot training and of aircraft. It aims to promote reasonable and responsible practices in a manner accepted as professional to other aviation bodies and the public, while retaining its own identity, reducing costs and minimising restrictions” (ASRA n.d.).

The operation of gyroplanes is subject to regulation by Civil Aviation Safety Authority (CASA).

The gyroplane aircraft generally consists of five major components, the rotor, airframe, power plant, tail surface and landing gear. A typical two seat gyroplane is shown airborne in Fig. 2. The rotor system provides lift and control for the gyroplane. Any rotor system can be employed in a gyroplane provided it is capable of auto rotation, however due to its simplicity the semi-rigid, teeter head system is the most widely used and is found in the majority of amateur-built Gyroplanes (Transportation 2000). The hub bar is one of the flight critical components that make up the rotor head assembly, two typical aluminium 6061-T6 rotor hub bars are shown in Fig 3.

The Gyroplane hub bar connects the two rotor blades to the teeter block, the hub bar is located below the teeter block in an under slung configuration and the teeter block allows the rotor assembly to pivot about the teeter bolt. The teeter bolt in turn connects the rotor head assembly to the tower plates (rotor spindle assembly) allowing rotation of the rotors about the vertical axis and transmitting the lift force to the airframe as shown in Fig. 4.

The hub bar is loaded in tension during operation due to the large centrifugal forces produced by the rotor blades which rotate at high angular velocities. The hub bar also provides control of the aircraft via the movement of the rotor plane relative to the fuselage. In flight the entire static load of the aircraft and its occupants are transmitted through the hub bar, additional dynamic loading results from flight manoeuvring, landing and taxiing. Drag and inertial loads will induce additional bending loads in the rotor plane with forward aircraft velocity (Blyth, 2006). Significantly, even for light rotor blades the centrifugal forces dominate over the aerodynamic and gravitational forces (Leishman, 2003). Figures 5 and 6 outline the primary loads and forces that require consideration.
The hub bar is subject to high cyclic stresses due to the dynamic nature of the rotating components combined with the transmission of the aerodynamic forces. This complex dynamic loading makes the hub bar susceptible to fatigue. Fatigue is the mechanism where due to repeated stress cycling it is possible for failure to occur at a stress level considerably lower than the yield strength for a static load (William D Callister, 2007).

The hub bar construction material in general use is wrought aluminium alloy 6061 T6 or 2024 T3 (Blyth 2006), with the former material emerging as the dominating material currently in use.

These materials do appear on initial consideration to have suitable fatigue properties for the hub bar application when one only considers steady state operation and assumes the aircraft limit loads are not exceeded. However a number of considerations jeopardise this initial assumption, including high energy manoeuvring, increasing maximum stress, and the incidence of fretting and mechanical damage to the aluminium hub bar. Studies of the fatigue process in metals and alloys have shown that crack initiation normally occurs at the surface. Here strain becomes localized due to the presence of pre-existing stress concentrations, such as mechanical notches or corrosion pits (Polmear 1981). This is significant as hub bars the author has so far encountered have all had some degree of mechanical damage.

IV. Review of Previous Investigation

Previous published research into the hub bar design centres largely on the investigation of why aluminium hub bars have failed. Thorough understanding of this work has provided a sound base upon which the preliminary design has been developed.

The ATSB Independent Technical Review of Gyrocopter Hub Bar Failures report, noted the implicit relationship between hub bar design (materials, physical form and construction) and its service performance, and hence, its resistance or otherwise to the development of fatigue cracking (Blyth, 2006). Furthermore it has outlined the necessity for any flight critical component to have its reliability accurately predicted and subsequently proven by testing in the interest of flight safety. These factors outline the necessity for aircraft components to be engineered with meticulous attention to detail and has been underlying in the design methodology adopted throughout this design project.

ATSB accident investigations further noted a number of cases of hub bar failure due to fatigue cracking, where “The mechanical indentations and surface fretting damage identified in the cracked or failed hub bars examined, was likely to be a significant contributing factor in the initiation of fatigue cracking” (Blyth 2006).

It is apparent that gyroplane operators conduct their own maintenance and some modifications (ASRA, 1993). There is a requirement for operators to meet the CASA regulations in this regard (CASA 2006), however the ASRA regulates the provision of training to the operators as required in conjunction with pilot training. This implies that gyroplane operators are not qualified to the same standard as professional aircraft maintainers. Anecdotal evidence of this is the widespread failure of operators to appreciate the significance of cracking and mechanical damage prior to a fatal accident, which subsequently raised the profile of the problem amongst operators, and was followed by sharp increase in the reporting of such instances (Wardill, 2009). This suggest some operators did not have an appropriate understanding of the behaviour of aluminium alloys and the relevant crack growth and fracture properties and possibly many operators still do not. These factors, and post investigation of the available information on current hub bar design and accident investigations, indicated to the author, that the hub bar design was required to be more robust and less prone to the fatigue mechanisms inherent to the use of wrought aluminium alloy. It was the intention of the designer to attempt to insulate operators from the possibility of hub bar failure due to ignorance.
The research into aluminium hub bar failures conducted by Wardill in 2009 is one of the most comprehensive documents available outside of ATSB and ASRA and therefore has been a valuable resource in establishing the design specification, in that it outlines significant problems that pertain to aircraft maintenance and operations. Particularly in the document summary the following points of consideration have influenced the new design (Wardill, 2009).

- An apparent tendency for technically inexperienced operators to carry out maintenance and modifications.
- A vast variation in aerospace product handling practices (i.e. significant potential for incidental damage to critical components).
- The inevitable likelihood of ignorance and/or arrogance amongst some operators.
- Limitations on monetary resources of operators.
- That hub bars incorporate a degree of precone.

These points underly a careful and considered approach to the new design.

V. Gyroplane Theory and Dynamics

To understand the dynamics of Gyroplane flight more fully a study of the theory of flight with respect to the gyroplane was conducted. The amount of material published on this is relatively sparse and therefore some reliance must be placed on theory relating directly to helicopter dynamics. There are many unpublished sources available; however these documents lack formal credibility, as the realm of gyroplanes largely belongs to the recreational and experimental operator. Hence, the lack of pecuniary motivation prevents copious formal publications being produced. Notwithstanding this project has made every effort not to rely on unpublished material.

A. Autorotation dynamics

A study of auto-rotational flight is critical to understanding the flight dynamics of the gyroplane aircraft. The theory of auto rotation for the conventional helicopter is best understood by considering the concept of ideal auto-rotation for vertical flight. This idea proposes that for a given thrust there is a self sustained operating state. This self sustainment is attributed to the conversion of potential energy (in the form of aircraft altitude) which, is converted to kinetic energy in the form of rotational inertia of the rotating rotor blades (rotor head Assembly). This phenomenon can be observed in nature in the flight of maple seeds, which spin as they slowly, descend, often being carried on the breeze for considerable distance. For conventional helicopters the auto-rotation is a manoeuvre that allows the aircraft to safely recover to the ground in an emergency such as loss of engine power or tail rotor failure.

For the case of the gyroplane the potential energy is replaced by kinetic energy provided by an engine driven propeller providing horizontal thrust, and the rotor disc has a positive angle of attack which tilts the lift vector aft. The relative air flow is directed up (as opposed to a conventional helicopter) through the rotor disc as the machine translates horizontally, only a low upward flow relative to the tip plane path is required for the rotor to enter into auto-rotational state. (TPP, Fig 7). The rotor experiences zero torque, about the vertical shaft. Therefore as long as the machine keeps moving through the air the rotor will continue to rotate and produce lift. If the engine power increases the aircraft will climb and if it decreases the aircraft slowly descends. Figure 8 gives insight to how the driving force to turn the rotors is derived for a blade element in auto-rotational equilibrium. Where equilibrium is given by Eq 1;

\[ Cd - \varphi Cl = 0 \]
Where; $C_d$ is the elemental drag, $Cl$ is the elemental lift, and $\phi$ is the angle between the rotor tip plane path and the relative free wind velocity. This angle is also equal to the angle that the lift force vector is tilted forward from the thrust force. When the rotor is accelerated to an angular velocity at which equilibrium is reached the rotor ceases to accelerate and maintains a constant angular velocity.

Not all of the rotor blade span can exist in this state of equilibrium at any given moment. However it is found that, at the inboard part of the blade the net AoA results in a forward inclination of the sectional lift vector, providing a propulsive component greater than the profile drag creating an accelerating torque (Driving force in Fig. 8) (Leishman, 2003).

**B. Rotary Wing Complexity**

Whenever a rotary wing aircraft operates in forward flight with the rotor passing edgewise through the air, it experiences an asymmetric velocity distribution as shown in Fig. 9. This phenomenon tends to produce a large moment due to the advancing blade having greater lift due to the higher velocity than the retreating blade which worsens with increasing velocity. The blade experiencing increased lift will tend to ‘flap up’, while the blade with decreased lift will ‘flap down’.

A further complication is the inertial and Coriolis forces due to the flapping. A bi-product of the flapping, Coriolis produces the additional lead-lag blade forces as the effective distance to the centre of gravity from the disc centre effectively decreases with the upward flapping and causes the blade to accelerate. On the downward flap the reverse is true, the lead lag adding further complication to the analysis of the rotor system. The teetering (or seesaw) rotor design uses the two blades solidly connected and has a teetering hinge on the vertical shaft axis as shown in Fig. 10.

The teetering design effectively negates the need for independent flapping hinges and attempts to balance the lift forces of each blade reducing the rolling moment about the lateral axis. As the advancing blade flaps up, its effective AoA is reduced and the moment due to its lift along the blade is transmitted about the teeter hinge to the opposing blade, this blade experiences a downward moment forcing this rotor blade down, increasing its effective angle of attack and increasing its lift. The relative bending moments are reacted as internal stresses through the hub bar.

The angle between the up-flapped span wise rotor blade axis and the rotors TPP is referred to as coning angle ($\beta_0$) and is a consequence of the upward lift force distribution. The precone angle is the angle manufactured into the rotor and is indicated in Fig. 10 by the symbol $\beta_p$. It follows that if the precone built into the hub bar equals the mean natural coning angle, the hub bar will experience a smaller maximum bending moment about the teetering axis than a hub bar without precone. Hence the maximum stress is reduced. Therefore the amount of precone manufactured into the hub bar has a direct relation with the fatigue life of the component. Operators regularly rely on trial, error and experience to set the precone for their aircraft. For the purpose of this design the author has written a Matlab code to more accurately

![Figure 8. Detail of the flow at a blade element in auto-rotational flight. (Leishman, 2006)](image)

![Figure 9. Distribution of incident velocity normal to the leading edge of rotor blade in forward flight, advance ratio = 0.3 (Leishman, 2006)](image)

![Figure 10. A teetering rotor design has two interconnected blades that behave as a single rotating system. (Leishman, 2006)](image)
determine the correct amount of required precone as a function of aircraft parameters and can be found at Appendix A.

An additional complication introduced due to natural rotor coning angle ($\beta_0$) is the rotor blades individual centres of gravity are flying above the teetering axis (teeter bolt, Fig. 4) plane, this can introduce further undesirable coupling into the rotor system through coriolis effect and results in both undesirable airframe and control-stick shake (Hollmann, 2007). Therefore the rotor is underslung below the teetering hinge to minimise the centre of gravity displacement relative to the shaft (Leishman, 2006). This is mechanically achieved by the use of the teeter block. The teeter height ($h_t$), (Fig. 11) is selected so that a line drawn from the centre of gravity (of blade in flight) of each blade would pass through the axis of the teeter bolt as shown in Fig 12. Even when using this simple analytical method for estimating proper teeter height, operators have found by trial and error that proper teeter height which produces the least undesirable vibration may be somewhat less. Hollmann, in his book on Modern Gyroplane Design, found that the optimal teeter height for his sporter gyroplane was in fact 0.85 times the analytical value (Hollmann, 2007).

Consideration of the teeter block is essential to this design, as the teeter block is the coupling between the rotor hub bar and the tower plates of the rotor spindle. The new rotor hub bar design required a new teeter block design to also be produced to this end. The redesign of the teeter block has included spacing the connecting hardware between the teeter block and hub bar further from the rotors axis of rotation to reduce the shear stress experienced at this joint when the rotor is pre-rotated.

The use of a teeter block to achieve under-sling induces a further problem known as mast bumping particularly for lightly loaded rotors (such as during a push over manoeuvre) where excessive blade flapping may cause the hub bar to contact the flap stop (shown in Fig. 11). The author has witnessed the markings left by this contact on a used hub bar in the author’s possession, and additionally on operational aircraft. The flap stop impact wear marks can be seen on the bar in Fig. 13. This will be an additional design consideration, as the contact has potential to damage the rotor hub bar or mast (Leishman, 2006).

C. Prerotation
A peculiarity associated with the gyroplane is the process of prerotation of the main rotor. This is done in order to impart the necessary initial momentum that the gyroplane rotor requires for establishing autorotation during ground operations in preparation for flight. Historically this was achieved by simply climbing the rotor mast and hand spinning the main rotor, this method proves somewhat tedious only achieving a low rotor RPM of about 20 RPM. This low RPM then requires the pilot to slowly increase his taxi speed with the rotor tilted back until sufficient RPM is achieved (Hollmann, 2007). However, as the gyroplane has evolved this process has been improved, with almost all modern gyroplane designs incorporating some mechanical
system to pre-rotate the rotor to speeds of between 90 and 100 RPM. Figure 14 shows a typical electric prerotator that uses an automotive starter motor to engage a flywheel at the base of the rotor spindle to impart the torque to the rotor. The torque and power required to accomplish this can be estimated in a relatively simple way, and for the Gyroscopic gyroplane these calculations can be found in Appendix B which show a torque of only 30Nm is required to turn the rotor. However it is not this easily determined torque that causes a problem for the rotor hub bar, but the impulse torque generated when prerotation mechanisms are initially engaged. According to research by Wardill prerotation torques can be reportedly as high as 353 Nm (260ft.lbs). This will certainly be an additional design consideration as there had been speculation as to the impact this has on aluminium hub bars and their respective service lives.

VI. Design

A. Initial Considerations

As part of the design process it was intuitive to look at the current state of the art for the rotor hub bar, due to the limited relevant published material the author endeavoured to take a survey (Appendix C) of the gyroplane community. Through the ASRA online forum the author published a survey to try and furnish as much data as possible, however participation was limited to a few vague email replies and only a single person making an attempt to furnish a survey as presented. Some of the responses were however useful, particularly information on a couple of fibreglass hub bars. Images of various hub bar designs are provided in Fig. 15.

Following this poor response the author decided that it would be more efficient to focus on a single typical gyroplane type that used a common aluminium hub bar. Additionally the author had access to a gyroplane operator in Braidwood, New South Wales who had been involved with them for many years. The operator owns a typical aircraft as well as 4 unserviceable aircraft which would provide further data on differences in configurations. For the purpose of this design the aircraft data and dimensions particular to a “Gyroscopic” gyroplane built by Ross Symes, have been used to calculate the relevant structural loads and provide the dimensional constraints, an image of the particular aircraft is provided at Fig 16.

The intention is to produce a prototype design that would be generic enough that it could easily be produced to suit the various configurations of gyroplanes that currently fly with the aluminium hub bar in question. While researching the gyroplane it quickly became apparent that there were, in general, only a small number of popular designs and configurations of hub bars in general use in Australia.
The main variables between aluminium hub bars were: the length along the long axis, the thickness of the bar, the method of introducing precone, the method of introducing rotor blade pitch, and the material used in construction. It is useful to expand on each of these characteristics and therefore have provided the following brief explanations.

Gyroplane operators select the length of their rotor hub bars in order to adjust the rotor span \( D_R \), which in turn changes the disc area \( S \) (Fig. 17.) and operating RPM. This method is used as most rotor blades are manufactured generally to one fixed length. A brief explanation of this interdependence is presented here. The rotor diameter is initially estimated using the following Eq. (2).

\[
D_R = 2 \left( \frac{GW}{DL \times \pi} \right) \text{[m]} (2)
\]

Where the gross weight \( (GW) \) is the sum of the; aircraft empty weight \( (W_e) \), pilot and passengers \( (W_{pax}) \), fuel \( (W_f) \) and oil. The disc loading \( (DL) \) is initially estimated by comparison with similar existing aircraft data, however in the case of an operational aircraft is simply found using Eq.(3).

\[
DL = \frac{GW}{S} \quad (3)
\]

The rotor diameter is proportional to the RPM using the following empirical relationship at Eq. (4)

\[
RPM = \frac{152 \times 60}{\pi \times D_R} \quad (4)
\]

Here the equation relies on the observation that most rotor tip speeds for helicopters and gyroplanes is approximately 152 m/s (Hollmann, 2007). This is due to the sharp increase in aerodynamic drag for tip velocities greater than this value. For the aircraft used in this project this method estimates a rotor speed of 340 RPM which is within 3% of the quoted cruise rotor speed observed by the operator (350 RPM), and within 5% of the mean value for the range of rotor speeds observed.

The thicknesses of aluminium hub bars, currently in use are generally restricted by commercial availability. The typical size is 2.5 in (63.5mm) wide by 1 in (25.4mm) thick. However the author has a hub bar in possession that is only 0.75 in (19mm) thick and according to the report by Wardill some hub bars are now manufactured at 1.1in (28mm) thick (Wardill, 2009). For the purpose of this design it was reasonable to consider only the 1 in (25.4mm) hub bar, as that is the dimension of the hub bar used on the case study aircraft. Furthermore the design aims to be simply extendable to other sizes and configurations. The method of introducing precone for aluminium bars is basically achieved using one of the following three methods. The first is introducing a bend to the centre of the aluminium bar so that when the optimal precone angle is incorporated the hub bar has the lowest possible bending moments acting along its length from the centre to its ends. This method is shown in Fig. 18. The centre is theoretically the best position to incorporate precone but suffers a number of practical problems in reality. The main problem with this configuration is the current use of a flat bottomed teeter blocks which do not properly mate with the hub bar when fitted and causes high localised stress concentrations where the edges of the teeter block are in contact with the hub bar (Wardill, 2009). The new design does not suffer this problem as the teeter block is matched to the hub bar, hence incorporates precone at the centre.
The second method of incorporating precone is by the use of coning blocks at the blade attachment zone of the hub bar as shown by Fig. 19. The problem with this method is that large bending moments are transmitted along most of the hub bar, peaking at the centre in the region of the teeter block. Additionally the hub bar experiences bending about the centre due these moments, and the problem of a flat bottomed teeter block on a bent hub bar as for the previous configuration still exist to some degree.

The third type uses a pitch adjustable blade cuff which has the precone machined into the blade attachment blocks as shown in Fig. 20.

The introduction of pitch (β) to the rotor system in order to provide the appropriate angle of attack (AoA) to the rotor blades is primarily incorporated by the following two methods. The first is the use of pitch blocks in a similar fashion to coning blocks, some hub bars have blocks which incorporate both precone and pitch together as shown in Fig. 21. The second way is by the ground adjustable blade cuff as shown in Fig. 20. The blade attachment blocks have elongated holes that allow rotation about a centre locating pin.

The material for aluminium hub bars is mostly wrought aluminium alloy 6061-T6, however there are still some wrought aluminium alloy 2024-T4 bars in service. Additionally during the research phase the author discovered the skywheels fibreglass hub bars, the red hub bar pictured at Fig. 5, it is a woven mat fibreglass hub bar, which showed signs of cracking about the teeter block joint in images provided with this image. In an email communication from a Mr John Evans he claimed that he and a very experienced engineer had produced two spring steel hub bars which have flown successfully and were currently working on one in 304 Stainless steel hub bar. Mr John Evans additionally expressed concern about the issues of quality control during manufacture and the early detection of failure with composite hub bars and said that on these grounds they had rejected pursuing a composite solution further. These will certainly be additional considerations in the development of a composite design, however it has been shown that unidirectional glass /epoxy beams can have many times the fatigue lives of these materials (Kaw, 1997).

The Austrian built Arrowcopter reportedly uses a composite hub bar, however the author has been unable to get specific details of this design as it is commercial proprietary information and not freely available to the public.

The current method of mating rotor blades to hub bars is through the use of steel straps of various configurations as shown in Fig. 22. A modified version of the straps will be designed to mate the designed composite hub bar with the rotor blades.

It was imperative to consider the operational environment of gyroplane operations in Australia and, it was found that they could be categorised into 3 broad groups, which were (Wardill, 2009);
1. Recreational gyroplanes; these aircraft are subject to low ground air ground (GAG) cycles, with, relatively low AUW and minimal High energy manoeuvres.

2. Trainer aircraft; these aircraft are subject to high GAG cycles, increased AUW, regular poor piloting Techniques.

3. High energy aircraft; these aircraft conduct operations that are subject to regular high energy manoeuvres such as aircraft used for mustering. These aircraft are routinely involved in high speed turns which increase g-loadings on the aircraft.

There is no information published which indicates what loadings these aircraft are experiencing and therefore the design has no reasonable motivation to design beyond the load factors provided by the relevant standards. It is of interest however that the high energy aircraft have reported a higher instance of hub bar crack detection than the other gyroplane categories and hence may warrant further investigation.

It has also been noted that it is common for Australian operators to operate from unprepared air strips, this has the potential to allow the blades to flap excessively and experience both large and negative bending moments that would not normally be experienced when the rotor is in flight and at normal operating RPM.

When taxiing at low rotor RPM, wind gusts have the potential to cause the blades to flap excessively and consequently the rotor hub may experience impact with the teeter stop, this impact must be considered in the design due to the potential for it to adversely affect the components life.

Gyroplane operators regularly dismantle their aircraft for road transport which increases the likelihood of incidental damage and wear to the gyroplanes components, including the rotor hub bar as this component is nearly always disassembled to some degree to facilitate road transport. This is due to the need to reduce the aircrafts largest dimension which is always the rotor blade span.

All of these factors mentioned, have adverse consequences for the service lives of the hub bar. Additionally reading the available reports and speaking to operators, it was apparent to the author, a qualified aircraft technician of 9 years experience, that the operator attitude to airworthiness was widely varying and in some instances concerning.

B. Standards

This design must meet all the relevant standards applicable to the manufacture of aeronautical product, modification to gyroplanes and the testing of materials intended to be used in the manufacture of aeronautical product. Below are a non exhaustive list of the relevant standards that must be applied to this design and a brief description of the relevant parts. It is important to note that standards are living documents and therefore should be independently consulted to obtain the latest amendments and no reliance should be made on specific references reproduced here or anywhere other than the proper current document.

- The operation of gyroplanes in Australia is subject to regulation by Civil Aviation Safety Authority (CASA). The applicable Civil Aviation Orders (CAO) include CAO 95.12 and CAO 95.12.1. The former of these refers to gyroplanes with an empty weight less than 250kg and the later to gyroplanes up to a maximum takeoff weight of 600kg. These orders list a number of exemptions applicable to gyroplane operation and delegates ASRA as the authority in matters relating to the operation of gyro planes in Australia (CASA, 2006).
- The regulations pertaining to the manufacture and modification of gyroplanes in Australia come under the CASA Advisory Circular AC 21-42(1). This document outlines the requirement for LSA modifications to be approved by the relevant aircraft manufacturer and further stipulates a number of specific American Society for Testing and Materials (ASTM) Standards that must be applied to Australian Gyroplanes (CASA, 2006).
- The United States Federal Aviation Administration (FAA) defines the gyroplane as follows, “Gyroplane means a rotorcraft whose rotors are not engine-driven, except for initial starting, but are made to rotate by action of the air when the rotorcraft is moving; and whose means of propulsion, consisting usually of conventional propellers, is independent of the rotor system”. Hence the gyroplane is subject to Federal Aviation Regulation (FAR) part 27, which applies to normal category rotorcraft. For Australian aircraft the ASTM standard mentioned above is more relevant due to the stipulation by the instrument of CASA AC 21-42(1) and on reading each of these
standards FAR 27 does not contradict the relevant ASTM standards albeit the ASTM standards are much more specific and directly applicable to gyroplane aircraft.

- The Applicable ASTM standards referred to by CASA AC 21-42(1) are the following; ASTM F2352-09 and F2449-09. The first of these two standards ‘ASTM F2352-09 Standard Specification for Design and Performance of light Sport Gyroplane Aircraft’ is the most instructive and restrictive of the standards applicable to this design and some relevant sections are listed below (ASTM, 2009). The second of these standards ‘ASTM F2449-09 Standard Specification for Manufacturer Quality Assurance Program for Light Sport Gyroplane Aircraft’, is relevant to the requirements which must be anticipated for component quality assurance and production acceptance if the design is to go into production and the manufacturer to act in the capacity of an aircraft component supplier. This document is less immediately relevant but awareness of its content essential to efficient development of a component intended for production in the future (ASTM, 2009).

As previously alluded to, the content of ‘ASTM F2352-09’ is the most relevant of the standards. The author has included a non-exhaustive list of the most pertinent sections and their contents in the Appendix D, an effort has been made not to cherry pick any particular standards above others but is limited only to the authors interpretation of the most relevant parts. Examination of the standards referenced in the appendix is critical to a full understanding of what is required for a design proposal to be fully compliant. The following are the absolute most significant standards of immediate concern to understanding the design evolution;

- 5. Structure
  - 5.1 General
    - 5.1.4 Design Conditions—The structural requirements of 5.1 must be met for all allowable combinations of:
      - 5.1.4.1 The maximum gross weight,
      - 5.1.4.2 Airspeeds up to VNE,
      - 5.1.4.3 The balance limitations, and
      - 5.1.4.4 The positive limit manoeuvring load factor.
  - 5.2 Flight Loads:
    - 5.2.2 Limit Manoeuvring Load Factors:
      - 5.2.2.1 The gyroplane’s rotor must be designed for positive limit manoeuvring load factor of 3.0 at all forward airspeeds from zero to the never exceed airspeed, VNE.

- 6. Design and Construction
  - 6.10 Special Factors of Safety—The factor of safety prescribed in 5.1.2 must be increased to the special factors prescribed in this paragraph.
    - 6.10.5 Rotor Components Factor:
      - 6.10.5.1 The rotor head, rotor hub bar, and blade spar structure shall have a factor of safety of two for centripetal tension loads acting alone under the critical flight loads in accordance with 5.2.2 and 5.2.3.

C. Design Goals
The goals of the project include:
1. To design a ‘composite’ replacement hub bar that will exceed the strength and reliability of current design that has a fatigue life that well exceeds the expected employment life of the part.
2. To produce a more robust, fool proof hub bar, which is less vulnerable to the same handling and operating conditions as experienced by current design.
3. The design must represent an attractive alternative to operators by being of a comparative cost to current design.
4. Ability to be incorporated as a replacement kit that is easily retrofitted, and aesthetically confidence inspiring.
5. The design must be flexible and simple enough to be extended to other configurations with relative ease and minimal tooling changes.
6. The design will conform to all relevant standards.
7. The design will satisfy spatial design constraints.
8. The design must be subject to a validation process.

D. Static and Dynamic load
The most critical consideration relevant to the design of this aircraft part is the relevant loading that is expected to be safely supported by the component. Hence the author used an analytical approach to determine the values that could reasonably be expected for the specific aircraft for which the design has been focused. To
make this design flexible the author has embedded the relevant calculations in Microsoft excel with the precone determination in Matlab code. This enables new aircraft parameters to be entered and the analysis for that new aircraft to be made with some degree of automataion.

From the study of the gyroplane theory and dynamics the highest and therefore most critical loads were found to be that of the tension-tension spanwise loading due to the centripetal force generated by the rotation of the main rotor blades about the spindle shaft axis and the bending moments about the teeter axis due to lift creation by the rotor blades. These were the first loads to be considered and were analysed as follows.

The tension force is approximately equal to the centripetal force. It was determined that the rotors could be simplified by considering their mass to act as a point mass at the rotor blades geometric centre of gravity (Fig. 23), and apply the same simplification to the rotor hub bar, then superimpose the two forces to find the total maximum tension force required to restrain the rotor blades. The equation used to find the centripetal force ($F_c$) for the rotor blade was Eq. 5.

$$F_c = m_r r_{CoG} \omega_n^2$$

(5)

Where $m_r$ is the single rotor blade mass, $r_{CoG}$ is the radius to the geometric centre of gravity of the blade and $\omega_n$ is the maximum angular velocity multiplied by a factor of 1.73 which relates to the requirement of a load factor of $n = 3$, this is explained further below. The centripetal force was found to be 157 kN due to the centripetal force of the blade only.

This calculation required determination of the accurate rotor blade mass and determination of the rotor blades centre of gravity. The author made a field trip to Braidwood, to obtain this data and other pertinent dimensional data.

To weigh the rotors the aircraft required partial disassembly so the rotors could be placed on a knife edge at one end whilst the opposite end was accurately measured using a 50kg load cell as shown in Fig. 24. The measurements were repeated at both ends in several different positions as shown in the schematic of the weighing setup at Fig. 25, and the data entered into an excel spreadsheet (provided in Appendix 45) to determine an accurate centre of gravity using Eq. 6.

$$r_{CoG} = \frac{m_{outboard}[r-(2x_{inboard})]}{(m_{outboard}+m_{inboard})} + x_{inboard}$$

(6)

Where $m_{outboard}$ and $m_{inboard}$ are the mass values at the corresponding weighing positions moving from the tips to the blade centre. $L_r$ is the total single rotor blade length and $x_{inboard}$ is the distance to the load cell sling centre from the inboard blade tip. Equation 6 was used for the three positions at each end and the average value of $r_{CoG}$ taken as the approximate true value of the rotor blade centre of gravity position and was found to be 1.719m ±1mm from the inboard edge. The rotor mass was found to be 17,740kg using this method.

The rotor was also weighed unsupported to confirm method used and was found to be 17,760kg. This was a 20 gram difference to that calculated. Some uncertainty was introduced by the inherent accuracy of the tape measure being ±1mm. The Load cell had an uncertainty of ±5 grams. Despite being inside a large shed some doors were not able to be shut and a light breeze was able to effect the environment within the shed, this has produced some
uncertainty in the data, given that the article being weighed was an airfoil. The uncertainty \(Z\) was calculated using Eq. 7.

\[ Z^2 = \left(\frac{\Delta L}{L}\right)^2 + \left(\frac{\Delta m}{m}\right)^2 \]  

(7)

Where; \(\Delta L\) is the uncertainty in length measurement and \(L\) the length measurement, \(\Delta m\) is the uncertainty in the mass measurement and \(m\) the mass value. The uncertainty found using this method was found to be ±0.5 mm and was rounded up to ±1 mm. This method failed to account for the environmental factors, hence the largest value for mass was used to remain conservative. The values used for the force calculations were \(m_r = 17.760\) kg and \(r_{CG} = 1.720\) m.

The aircraft’s aluminium rotor hub bar was found to weigh \((m_{hub\ bar})\) 3.200 kg ±0.005 kg and had a total length \((l_{hub\ bar})\) of 600 mm ±1 mm. The hub bar design intended to be a lighter hub bar therefore using this data for preliminary calculations was a conservative approach. To find the centripetal force due to the hub bars mass a similar method was used as that above and is shown here by Eq. 8.

\[ F_{c_{hub\ bar}} = \frac{m_{hub\ bar} l_{hub\ bar}}{2} \omega_n^2 \]  

(8)

Where only half the hub bar mass is considered as the spindle axis lies at the hub bar centre, and the hub bars centre of gravity, is found to be half way between the spindle axis and hub bar end. Hence the hub bars centre of gravity at \(\frac{1}{4}\)th the hub bars length from its centre. The centripetal force due to the hub bar half was calculated to be 1.05 kN.

The total value of the rotor blade and hub bar half centripetal force was superimposed together and was calculated as 158 kN. Therefore this value represented the tension force that would be considered for the design.

The other dominating forces that must be considered are the bending moments about the centre due to the lift acting on the rotor blades at some distance from the centre. In the final design it is anticipated that these moments will be minimised through the incorporation of optimal calculated precone angle as discussed under the heading rotary wing complexities, however it is essential that these moments be calculated for worst case scenario as precone is calculated for the cruise flying condition. Bending moments for this design were calculated in matlab using the matrix Eq. 9. And the following simplifications;

- That AUW is multiplied by the load factor \(n_r\).
- The Load factor is \(n_r = 3\) in accordance with ASTM F2352-09 par.5.2.2.1 (ASTM, 2009).
- The rotor hub bar has no precone.
- That the rotor is at a single azimuth position and is divided into \(n_r\) stations.

\[ \{M_{Max_{hub\ bar}}\} = [(1.0) + \omega^2[Z][ZM]]^{-1}[\{R\}L] - \omega^2[F] \{\beta_p\} \]  

(9)

Where; \(M_{Max_{hub\ bar}}\) is the column matrix containing the moment values for each station \(n_r\), \([1.0]\) is a \(n_r\) size diagonal matrix of ones, \(\omega\) has its usual meaning, \(F\) is the mass at station \(n_r\) multiplied by the radius of station \(n_r\) matrix, \(ZM\) is the station span divided by the blades elastic modulus multiplied by the second moment of inertia, \(R\) is the radius to the station \(n_r\), \(L\) is the lift at station \(n_r\) and \(\beta_p\) is the precone in the hub bar.

The output plot from Matlab is provided at Appendix E. The matlab code result was a maximum bending moment of 440Nm.

Using the thickness and width dimensions of the aluminium hub bar this maximum bending moment was used to calculate maximum stress using the flexure formula shown at Eq. 10.

\[ \sigma_{bending} = \frac{M_{Max_{hub\ bar}} c}{l_{hub\ bar}} \]  

(10)

Where; \(M_{Max_{hub\ bar}}\) is the value calculated above, \(c\) is half the hub bar thickness and \(l_{hub\ bar}\) is the second moment of inertia for the hub bar.

The maximum stress due to bending was superimposed with the maximum stress in tension due to centripetal force and this value was taken as the maximum stress that could be expected to be experienced for the worst case scenario of load combinations. This value was determined to be \(\sigma_{hub} = 184\) MPa.

Aluminium 6061-T6 has an Ultimate tensile strength of 310 MPa, and yield strength of 276 MPa (Mott, 2006). From these preliminary calculations that have accounted for the appropriate load factor as per the ASTM standard, the result suggested the aluminium 6061-T6 hub bar does meet the minimum required standard. An additional safety factor of 1.5 should be applied, for this approximation the factor of safety (FOS) is calculated.
as 1.68. The aluminium also meets the centripetal force ‘only’ requirement of ASTM 2352-09 par.6.10.5.1, which required a FOS of 2, in this case the bar had a FOS of 2.65.

From the preceding calculations the author had two significant design parameters to work with in developing the design, which were a maximum force due to centripetal load of 158 kN which required a special FOS of two taking the value to 316 kN. The second parameter was for the combined loading maximum stress which would be dependent on the designs geometry, yet enabled guidance to work towards the properties required. The calculations were laid out in excel to allow easy comparison of design geometries as each were considered and once the fibre volume of manufactured samples were known, these values were used to estimate the FOS that could be expected for glass epoxy hub bars. This is discussed further below under fibre volume determination.

Further explanation is warranted to define the factor of 1.73 that was used in equations 5 and 8 to find the \( \omega_{n-2} \). It can be shown that for a load factor of 3, that the rotor RPM would increase by a factor of 1.73 when the vertical autorotation model is assumed (Hollmann, 2007). Using that model the angular velocity \( \omega \) is found using Eq. 11.

\[
\omega = \sqrt{\frac{c_1}{n^{2/4} S^{1/2}}} 
\]

Where; \( \theta \) is the rotor blade pitch in radians, \( \lambda \) is the inflow coefficient and \( c_1 \) is a coefficient shown by Eq. 12.

\[
c_1 = \frac{2AUW_n}{0.5844R^2\sigma_uS} 
\]

Where; \( n \) is the required load factor, \( \rho \) is sea level ISA density, \( R \) is the rotor radius, \( \sigma_u \) is the rotor disk solidity and \( S \) is the rotor disk swept area. When any AUW is multiplied by 3 the \( \omega \) always increases by a factor of 1.73.

The hub bar is required to resist the torque induced forces produced by the prerotation of the rotor on start up in accordance with ASTM 2352-09 par 5.7.1.4. This is usually achieved through the use of an automotive starter motor or similar device which engages with a ring-gear attached to the bottom of the rotor spindle. The main impact of this mechanism is that on engagement an impulse force which can reportedly be as high as 350 Nm is transmitted to the rotor. The largest estimation found analytically was just under 30Nm (provided in Appendix B), while according to Wardill, he found by experiment values of 76 Nm and had information suggesting the 350Nm value. To remain conservative the 350Nm value was used as this force was of an order of magnitude larger than that found for the worst case calculation. This value was critical in the design of the teeter block and teeter block attachment due to the torque necessarily being transmitted through these components.

E. Material

The choice to design a replacement Hub bar using composite material was made to take advantage of the favourable properties that fibre reinforced products offer aerospace applications and equally as a matter of personal pursuit to investigate the potential of composite materials and gain an appreciation for the related design process.

The author chose to use unidirectional E-Glass fibre rovings in an epoxy matrix as the composite system for the initial design. Advantages of glass reinforced epoxy include:

- It has a lower specific gravity and can produce lighter components than comparable aircraft grade metals of similar dimension.
- Has up to more than twice ultimate tensile strength of Al and greater than that of titanium in the fibre direction (\( \sigma_{ut}= 2700 \text{MPa} \) (Owens Corning, 2008). Compared with Aluminium; \( \sigma_{ut}= 450 \text{MPa} \) and Ti \( \sigma_{ut}= 1100 \text{MPa} \) (Baker, et al., 2004).
- Higher impact resistance compared with other composite systems except for S-glass which is slightly higher.
- Improved damage tolerance compared with Aluminium and other metals, due to tooling marks, nicks and scratches having significant detrimental effects on failure behaviour of aerospace metals while the failure behaviour of fibre reinforced matrix composites does not suffer the same stress concentrations in conjunction with these damage modes.
- Can easily produce variety of shapes and sizes whilst remaining relatively inexpensive to manufacture, this is due to the mould construction requiring only minimal workshop machining and manufacturing.
• The low cost of glass reinforced epoxy composite system compared to other composite systems available.

The particular Rovings/fibres chosen were Owens corning SE1200 single end rovings which are specifically marketed for increased fatigue performance and corrosion resistance which allowed them to qualify for use in wind energy applications (Owens Corning, 2008). Glass fibres offer good fatigue performance but are inferior when compared with Kevlar/epoxy and Carbon/epoxy, therefore this was a significant consideration when considering glass reinforcement fibres. Owens corning trademarked Advantex glass aims to provide consistent glass fibre product properties. This is a useful product characteristic as consistency in the final product has a significant dependence on consistency of the raw materials. The glass is supplied on a centre pull 20kg doff as pictured in Fig. 26.

Epoxies generally have higher fracture toughness than other resins which can aid in improving the fatigue performance of the microstructure (Mallick, 1997). Epoxies also have relatively low shrinkage on cure and are less likely to crack during the cure. The epoxy chosen for this design is ES300 supplied by FGI industries.

The use of composite leaf springs (mono springs) for the Chevrolet Corvette provided an example of the use of glass/epoxy beam in a highly stressed high cycle application. It is claimed it provided a smoother ride and more rapid response to stresses induced by road shock, it offered a fatigue life in excess of five times that of steel in the same application, it has less chance of catastrophic failure and excellent corrosion resistance. (Kaw, 1997) The success of the spring in the Corvette has seen it become common in other makes and models.

The author recognised that other materials may possess suitable properties for the rotor hub bar application, however materials such as titanium would impose a weight penalty, which is the enemy of flight (particularly for light aircraft) and alternative composite systems can be expensive, ultimately the properties of the E-glass/epoxy system are more than adequate to provide the required strength properties required for this application and so a design compromise is made to produce an efficient safe and functional solution, the glass epoxy is proving to be superior to this end.

F. Form-Hub bar

The geometric form of the design started with sketching on paper and white board considering different potential ways to employ continuously hand laid glass rovings in a mould or cast, or over a die. The dominating structural tensile loading along the long axis of the hub bar significantly influenced the form of the design. Additionally the requirement to secure the component by way of coupling hardware to the teeter block and rotor blade cuffs or alternatively to the rotor blades directly also had strong influence. A number of additional requisites included;

1. That it must be a one for one replacement as an assembly or kit. It would need to integrate with the existing Rotor blades and rotor spindle assemblies.
2. It is dimensionally constrained by existing parts to some degree, particularly the clearance between the two spindle tower plates and the spindle bottom plate as shown by Fig. 27.
3. The choice of an anisotropic material meant that the micro-structure (i.e. The Lay-up) was a primary consideration for design, to take maximum advantage of the material properties.
4. It was important as a design philosophy to employ the KIS (Keep It Simple) principle and avoid excessive complexity, this would lead to a higher efficiency in manufacture and for the future installation to aircraft.
5. Observation of a strict weight limitation.
6. The need to meet applicable standards.

Using these considerations the author produced an initial set of design drawings in Catia drawing software, which have evolved as the project has moved forward, the design presently includes the composite teeter block which is an implied design task and 1.53 degrees of precone. An early iteration design is shown in the image at Fig. 28.

The glass fibre reinforcement orientation (microstructure) is a critical aspect in appreciating the design. The internal geometry of the hub bar is described by Fig. 29. This image shows a half sized hub bar sample manufactured to allow property testing and determine the suitability of the manufacturing process.

The red lines in the central region indicate that the fibres are first wound around the circumference of each of a set of chrome-molly sleeves which are to be embedded into the composite matrix to provide point of connection for the attachment hardware and spread the bearing load across the matrix. The continuous rovings are then wound around the next consecutive sleeve and so on until the end of the bar. This winding process is continued from end to end, until the rovings occupy the entire required thickness of the bar. The sleeves are located by bolting them to the mould in the correct location as shown in Fig. 30. The bolt holes are accurately positioned during the mould construction. The Green lines in the outer region of Fig. 29, indicate the continuous roving is subsequently wound about the outside of the entire red region until the fibres occupy the entire mould space. The mould side halves are then clamped in place to provide proper flat and smooth finish to the hub bar edge. The inter-space blue region indicates the area which is filled with chopped rovings and epoxy, this region has significant lower tensile strength properties as the fibres are not orientated in any specific direction. A manufactured sample is shown at Fig. 31 this image shows the actual fibre orientation.
The hub bar Revision 10 was designed after recent consultation with a couple of local gyroplane operators who suggested that there was room for improvement with the teeter block replacement and additionally suggested further innovation in regards to the rotor blade coupling where by the amount of lead-lag could be mechanically adjusted at this interface.

This Revision incorporates a blade attachment strap with a post welded to its edge, an adjustable link between the post and the special attachment bolt allows more accurate rigging during installation and the link means that on disassembly for transport the proper rig is easily maintained on reassembly.

The hub bar Revision 10 incorporates the required optimal precone angle calculated as 1.53 degrees. The final design will also incorporate the required blade angle of incidence manufactured into the hub bar, reducing the number of parts required. The design project has been an iterative process evolving with the designers understanding.

G. Form-Teeter Block

The teeter block has been designed to be made from hand laid woven cloth E-glass and epoxy. The weave been considered is the 7781 medium weight industrial cloth. This cloth is a high performance cloth with high strength to weight ratio. The resin system will be the same used for the hub bar itself.

The loads acting on the teeter block were calculated by considering the implications of been the single load path between the hub bar and rotor spindle tower plates. The dominating stress situations involve the teeter block being in compression due to the lift of the rotor blades supporting the weight of the aircraft, the teeter block being in tension during negative g flight manoeuvre and during ground operations and storage, and the torque due to prerotation. An additional dynamic load is due to the lateral imbalance of the rotor transmitting a high cycle oscillation (two per revolution) approximately along the long axis of the rotor and hub bar assembly. All situations must have a factor of safety of 1.5 applied. The compression case additionally requires the load factor of 3 taken into account and tension requires the negative load factor n=-0.5 taken into account.

The equations used and values found are presented in Appendix N and were calculated for the area under the highest amount of stress which is that zone about the teeter bolt where the design has the narrowest section. The conclusion was reached that for 2mm skin, with 9 laminate ply’s orientated 0,15,45,30,0,15,45,30,0 degrees. As each ply is orientated in two directions perpendicular to one another this would provide pseudo isotropic properties. Due to the complex 3 dimensional shape it was important not to reinforce in one particular axis over another as it would make the layup extremely complex operation.

An image of the latest revision for an adjustable teeter block is shown at Fig. 33. And the drawings provided at Appendix N.
H. Manufacturing Process

The manufacturing process required a significant design effort in its own right. To take the idea and then create the product in the workshop required careful consideration and creative initiative to develop a simple process on a budget of time, money and skill. The decision to use long continuous glass rovings dictated that some tooling would be required to allow continuous wetting of the glass fibre as it was hand laid into the mould. The mould would then be clamped to produce a press cure and the whole mould placed into the oven to cure the epoxy at an elevated temperature to encourage cross linking of the epoxy bonds and optimise the components strength properties. Figure 34 gives an elementary description of the process. The project supervisor Alan Fien proposed that to enable the fibres to be placed into the mould at a sufficient rate to finish the lay up within the epoxy gel time that it would be necessary to use a bundle of continuous strands. Between the literature and consultation with supervisor a simple method was devised.

The image at Fig. 34 illustrates the creel made to support up to twelve spools of glass fibre. Each spool is preloaded with fibre and weighed to ensure the amount of glass rovings are sufficient for the lay up process. The rovings are pulsed through the funnel which is periodically filled with epoxy resin and not allowed to fall below half full. The fibres then pass between the two lengths of dowel which serves to remove excess resin to keep the fibre-matrix volume ratio high, the excess is caught in suitable containers and can be reused in the funnel for wetting to reduce wastage. The area forward and below the creel is the working area where the mould is placed and the hand winding conducted.
The mould construction was achieved by forming medium density fibre board (MDF) with a router to the drawn shape required. The moulds were made in four parts. The base provided the mounting surface for the hub bar bolt sleeves and was the main part used for the winding process. Figure 36 shows a complete set of moulds made for an earlier hub bar design. The image shows the sleeves in position with large penny washers clamped above each to prevent the rovings from spilling over the edge of the sleeves as some tension is used to lay the fibres which locate the bolt sleeves. The side halves are clamped and screwed into position once sufficient fibre are wound on so that the fibres are just over the design outline drawn on the mould base. Once the sides are in place the fibre is continuously laid until the required thickness is achieved. The fibre is then cut and the voids filled before the top plate is clamped down.

Prior to going to the lab to conduct the first layup a trial process was conducted using wool wound onto the spools and glue mixed to a consistency to approximate that of the epoxy resin. The trial proved successful in that the winding process worked and the creel functioned as expected. However, when the first half hub bar sample was been wound difficulties arose due to the fibre loose ends protruding from the inside of the spool bore becoming quickly twisted around the spool shafts. Due to the enormous strength of the glass fibres one shaft actually broke away and a quick on the fly adjustment had to be made. This did not impede the outcome by any measurable significance but outlined the type of unexpected outcomes when trying a new process.

The moulds suffered a number of problems including the resin adhering to the moulds despite the moulds being coated with a fibreglass release wax. After the first sample suffered from this problem the author tried a different release agent that was sprayed on, however the resin still bonded to the MDF and the two sample moulds were destroyed in the process of freeing the samples. Figure 37 shows the first sample being removed from its mould.

The second more significant problem was the excess resin that made it into the mould for the first sample, this was easily rectified by the addition of the dowel rods to the creel shown in Fig. 35, which reduced the excess resin very effectively and can be shown by the difference in fibre volume ratios measured for each of these two samples, before and after the modification, these are shown in Appendix F.

The third problem encountered was the effective control of the sample thickness. The first sample manufactured was by chance produced with the correct thickness, the second sample however saw the top plate clamped beyond the desired depth and reduced the sample thickness by a full 1.5mm. Additionally the moulds allowed resin to escape and the filled voided regions were left with unacceptable air pockets. A possible solution to these last problems is to redesign the moulds using machined aluminium billet or similar. This process is significantly more expensive and hence was not trialled during project thus far. Finer mould dimensions, the use of release of film on the base and top plate and a more effective release agent for the mould sides should suffice for the experimental and prototyping stage. Additionally a more precise method for achieving the correct clamp up such as bolted stops should aid in
achieving the correct thickness. Once a specific design is accepted and a demand for a number to be produced exists an aluminium mould or similar, would be easily justified.

Two methods were trialled to fill the voids between the continuous rovings. The first was to press chopped rovings and epoxy into the voids and the second method used an epoxy and micro void mix. The first method tended to prevent the formation of air bubbles to the degree the micro voids allowed. However the method using chopped roving would be improved by cutting the fibres to much shorter lengths (between 3mm and 10mm) to produce a more uniform non orientated fibre zone. During the destructive testing phase the micro void filled areas appeared to have failed earlier and the damage appears more substantial than that of the chopped rovings.

I. Hardware

The hub bar design required a significant effort to design and manufacture the custom hardware required for the design. The sleeves were designed by first conducting analytical assessment of the required strength properties, this data is provided in its original excel format in Appendix G Engineering drawings were then produced in Catia and workshop resources organised. The sleeves were produced from high strength low alloy (HSLA) 4160 Chrome-moly steel as it provides the benefits of being light, hard, strong and has excellent corrosion resistance properties. The sleeves were turned on the metal working lathes within the university’s student workshop and training took about half of one day to be completed. Each sleeve manufactured took approximately 1 hour of lath time. Approximately 25 individual sleeves were produced for the sample hub bars, prototypes and teeter block replacement. A full set of sleeves for the revision 9 hub bar design is shown in Fig. 38.

VII. Testing

A. Rationalisation

The testing phase was the realisation of significant preliminary design work to prove the viability of a continuous hand wound, glass reinforced epoxy matrix replacement hub bar. The author had attempted to model the component as designed on Catia in order to analyse the part using that software’s built in analysis tools or for importing the model into Ansys. However after many hours of learning how to construct a composite model using Catia, it became apparent that the non uniform nature of the wound structure was beyond the scope of the authors skills in modelling composite structures in Catia. The elementary description of the microstructure at Fig. 29, is referred to here to communicate the complex geometry that does not easily lend itself to software modelling.

The intention from the project outset was to prove the real world viability of the replacement design and hence the author in consultation with the project supervisor was satisfied to rely on the experimental testing results to underpin the rationale behind the projects direction from that point forward.

The analytical estimations of the components tensile properties, made using the authors excel design spreadsheets, showed that the tensile properties along the principle fibre axis would well exceed that required by the dynamic loads expected for the gyroplane hub bar. Therefore with only two samples it was decided to test the stiffness properties of the component by conducting a three point bending test with the component on its narrow edge as shown by Fig. 39, and a torsional load test about the components long axis as shown below in Fig. 40.

Post carrying out the three point bending test to failure the author decided that it would be intuitive to conduct a tensile test to observe the residual tensile integrity of the design and the results are detailed below.
B. Test Equipment

To enable the conduct of testing a number of articles required either manufacture or modification. In the first case, for the three point bending test the author required four pin support blocks to be manufactured and one existing coupling modified to suit the experiment. The choice to modify an existing coupling was made in effort to conserve time and resources since the existing coupling was very close to what was required. This was a significant learning point for the engineering management part of the project, in that reliance on external resources whom have higher priorities can have significant effects on time line. The manufactured components for the three point bending test are pictured in Fig. 41.

The tensile test conducted post three point bending test required the manufacture of only four bushes to adapt existing couplings, this saved considerable time in design and manufacture which allowed the tensile tests to remain a worthwhile pursuit. The tensile test setup is shown at Fig. 49.

The torsional stiffness test required a significant design and manufacturing effort as the school did not have a testing jig capable of testing the hub bar sample. The design underwent four iterations as it went from concept conceived with the assistance of the project supervisor and lab technician to the final design which was to be built between the schools student and main workshops using available tooling and school materials as much as possible. An image of the final design is provided by Fig. 40. The expected torsional stiffness and maximum torque for the hub bar design sample were estimated in the design spreadsheets using approximations of the components structure. The Torsional stiffness jig was manufactured over a period of three weeks and cost very little to build as during the design iterations suitable materials were sourced from within the school and then the design modified to use those available materials. Much use was made of 50mm x 75mm RHS which had 5mm wall thickness. This material was used to provide all of the torsional rigidity required and was joined by mig welding the seams and ends. The sample torque coupling welded to the torque tube end was again a modified coupling which the author had previously used in the tension testing. To provide a constant lever arm length with which to twist the torque tube a quarter circle plate was welded to the torque tube and was coupled to the load cell via a length of Renold leaf chain which was sourced from a local forklift service workshop. The Torsion test setup is shown at Fig. 42. Jig drawings are provided at Appendix L.
C. Fibre Volume Determination by Immersion

To make use of the test data and assess the manufacturing process, the manufactured samples required accurate fibre / matrix volume ratio determination. To achieve this task, once the samples had been removed from their respective moulds they were taken to the universities Civil Engineering Lab. The lab is equipped with a set of scales to measure the weight of a submersed composite structures. The scale setup is shown at Fig. 43.

The process involved weighing the hub bar sample and the hub bar sleeves and then finding the submersed weight of the hub bar sample. The process then uses Eq. 13 to determine the approximate fibre / matrix volume ratio.

\[ V_f = \frac{\rho_{\text{matrix}} \cdot V_{\text{sleeve}}}{\rho_{\text{matrix}} \cdot (V_{\text{sample}} - 3V_{\text{sleeve}})} \]  

(13)

Where; \( V_f \) is the percentage of the volume occupied by glass fibre reinforcement, \( \rho_{\text{matrix}} \) is the density of the epoxy resin matrix, \( V_{\text{sample}} \) is the volume of the entire sample determined by Eq. 14 and \( V_{\text{sleeve}} \) is the volume of the sleeve determined from its geometry.

\[ V_{\text{sample}} = \frac{m_{\text{sample in air}} - m_{\text{sample in water}}}{\rho_{\text{water}}} \]  

(14)

Where \( m_{\text{sample in air}} \) is the plain weight read from the scales for the sample, \( m_{\text{sample in water}} \) is the scale reading for the sample suspended in water and \( \rho_{\text{water}} \) is the density of water. A similar approach is used to find the percentage of volume occupied by the matrix (\( V_m \)) and as a check the two values should sum to unity as in Eq. 15.

\[ V_f + V_m = 1 \]  

(15)

The calculations carried out in excel are provided at appendix F. These revealed the first sample had a \( V_f = 0.4 \) and the second had a \( V_f = 0.49 \) the difference attributed to excess resin problem mentioned under manufacturing. Knowing these values the author was able to revisit the estimations of strength made in the design spreadsheets and find the estimated maximum tensile strength for sample two was \( \sigma_1 = 1.52 \) GPa (1.7 MN tensile load for this geometry) and had a FOS of 7.4 for the combined worst case which requires FOS of 1.5 and a FOS of 11.1 for the Centripetal only case which required a FOS of 2. This represents a significant advantage over the aluminium which had FOS calculated for the same cases as 1.68 and 2.65 respectively. These calculations are presented at Appendix M.

D. Experimental Results, Analysis and Comparison

1. Three Point Bending

The first plot presented in Fig. 45, is for the final three point bending test where the testing machine was allowed to bend the hub
bar sample to failure using constant displacement. The Test set up used the 250kN maximum load, Instron 8033, Servo Hydraulic Fatigue Testing Machine. This machine was used to provide the load in all tests conducted. The displacement was measured with an EIR Laser Extensometer. The Test setup is shown at Fig. 44.

The 3 point bending test showed that the modulus of elasticity for this beam in bending to be approximately $E_B = 17.1 \text{ GPa}$. It is worthwhile to note the significance of the subscript “$b$” used to differentiate this elastic modulus from that used for homogeneous materials and those for the longitudinal ($E_L$) and transverse axis ($E_T$) of the sample bar. This is because unlike homogenous (isotropic) materials the elastic modulus in bending for composite materials is dependent on the state of stress, differences in the fibre layup through the thickness of the material and manufacturing imperfections (Peters, 1982). Additionally due to the relative short span of the sample bar (less than $l/t = 40$) the inter-laminar shear properties are likely to have a very significant effect on the failure mode. This value is higher than quoted values for transverse elastic modulus of $E_T = 12 \text{ GPa}$ and lower than quoted values for longitudinal elastic modulus of $E_L = 45 \text{ GPa}$ (Mallick, 1997). Therefore the figure is within reason as the three point bending is a combination of shear, transverse and longitudinal stress. The bending rigidity was calculated as $EI = 82 \text{ kPa.m}^4$. The elastic modulus in bending was calculated using Eq. 16, as follows (Sih, et al., 1986);

$$E_B = \frac{-P l^3}{4 u b h^3}$$ (16)

Where $P$ is the load on the centre pin, $L$ is the distance between the outside pins, $u$ is the displacement, $b$ is the thickness of the sample and $h$ is the width of the sample hub bar.

This method does not consider the non uniform nature of the bars cross section but provides a useful guide for qualitative comparison of the manufactured bar sample with that of aluminium which has a quoted elastic modulus of $E = 72 \text{ GPa}$. This indicates that the sample hub bar is over four times more elastic than aluminium and would experience smaller stress for the same deflections in flapping, leading and lagging of the rotor blade during flight. This result supports the estimated likelihood of the glass/epoxy having greater fatigue properties than that of aluminium.

Figure 45. Three Point Bending Test Data Showing Analytical Comparison to Aluminium.

Figure 46. Large Crack Post Failure in Three Point Bending.
The data for the aluminium comparison plot is purely analytical (mechanics of materials approach) and is cut off at the point of calculated ultimate load $P = 39.2$ kN, determined using $\sigma_{\text{ultimate}} = 310$ MPa for aluminium 6061-T6. If $\sigma_{\text{yield}} = 276$ MPa is used the plot would indicate the aluminium bar would start to yield at 34.9 kN.

For the composite hub bar sample the first indication of internal breakdown of the microstructure occurred in an earlier test and was indicated by the sound of fibre breakage at approximately 39.6 kN however no permanent set was observed on unloading from this first indication. The plot at Fig. 45 indicates that the bars linear elastic properties breakdown after approximately 42 kN. The full plot at Appendix H indicates that total failure occurred at a load of approximately 59.4 kN and shows the behaviour post failure which is a benefit of using the constant displacement testing, as opposed to constant load, which would reveal little about the residual properties of the component post failure. Using the maximum load of 42 kN the equivalent bending moment is equal to 9.9 kN.m. An image showing the cracking present post failure in bending is provided at Fig. 46 and demonstrates that the critical failure occurred along the inside edge of the fibre rich outer zone and the outside edge of the fibre rich inner zone. Between these two zones the fibre matrix ratio is reduced and shear stress due to bending is high in this region which is almost analogous with the web region of a steel structural beam. The close up image at Fig. 47 indicates the failure appears to have occurred due to failure in the shear planes and then failure in the transverse fibre direction. Relatively very few fibres are seen bridging this gap or splayed out from the crack edges suggesting that the failure is predominantly in the fibre volume deprived region.

Six preliminary bending tests were performed prior to the final test to failure, to observe uniformity in stiffness for bending in each direction by simply turning the sample over between tests, it was found that the
sample displayed good uniform behaviour regardless of which side of the sample was loaded resulting in a spread of 1.5% of the mean value of the elastic modulus and a range of 3% of the mean value of the elastic modulus. The first visual indication of deformation was detected post test five which had a peak load of 40.46 kN and audible indications of fracture at 39.6 kN. Figure 48 shows the visual indication seen post test five.

2. Tension Test, Post Three Point Bending.

The author saw it as intuitive to test the residual tensile properties of the first sample post testing to destruction in three point bending. The same machine was used and reconfigured for tension testing. The laser extensometer was again used to measure the displacement and this was converted to tensile strain. The Tensile test setup is shown here in Fig. 49. The hub bar sample did not exhibit total failure and maintained a linear elastic stress strain curve up to a value of 214 kN, at which the pin attaching the sample to the tension coupling failed in shear. The pin was an Unbrako socket head screw and was rated for a yield stress of in excess of 1200 MPa which would provide a maximum shear stress of approximately 600 MPa. However the bolt withstood a maximum shear stress of up to 844 MPa at which point failure in shear occurred. The test set up was limited by the bolt diameter and in hindsight should have been tightly fastened to provide a proper friction grip coupling. The data that this test provided shown in Fig 51 was beneficial allowing a pseudo longitudinal elastic modulus to be estimated and proving the ultimate structural integrity of the hub bar sample provided an ultimate tensile strength greater than 214 kN. Had the bolt not failed it was anticipated that the sample would not have failed before the 250 kN machine limit. The hub bar sample did suffer from unacceptable permanent deformation due to the sleeves crushing the epoxy matrix and bulging of the matrix about the ends of the bar sample as shown in Fig. 50. This is a significant result as it demonstrates the requirement for further testing with the sample coupled across two bolt holes as would be the case in the designed installation, and further shows that improvement is needed in structural development and design in the attachment region. One solution being considered is the use of a sleeve that has a much larger and tapered flange which would prevent the bulging of the fibre, a drawing is provided at Appendix I. The longitudinal elastic modulus was estimated to be $E_L = 30$ GPa. This is 80% of estimated values, and about 40% of the value for aluminium. Using the design spread sheets for the calculation of the maximum tensile strength of the hub bar with the as manufactured dimensions of the sample bar that was used in testing an ultimate tensile strength of 1600 kN. which would provide a FOS of more than 2.0. Figure 51 shows the Tensile Test Post Bending Test.
10. However it is now clear that the limiting factor would be the region supporting the outermost sleeves, here the compressive bearing stress of sleeve acts in a transverse orientation to the fibre direction and therefore is limited by the compressive strength of the matrix and the geometry of the joint. Empirical data must be obtained through further experiment to establish a relationship between bearing failure and this specific geometry as there no reliable analytical method to determine accurate values. It is clear for this sample that this failure occurred above 160kN and less than 214 kN.

3. **Torsional Stiffness determination.**

The second hub bar sample was subject to a torsional test to determine the torsional stiffness properties of the hub bar sample. The hub bar showed linear behaviour and no significant permanent set up to a torque of 520Nm, after this point the epoxy matrix began to breakdown particularly about the ends but interestingly the sleeves did not disbond at any stage and the integrity of this joint was maintained. The plot at Fig 52. Shows the final test to failure. To find a approximate value of the maximum shear stress in torsion an approximation was made to consider the beams cross section as an ellipse, this gave a value of $\tau_{\text{max}} = 82\text{Mpa}$, about 20% of quoted values. An aluminium 6061-T6 hub bar was setup in the torsion jig to conduct a comparison to that of the sample hub bar however the aluminium was so rigid in torsion that the spacers between the bolt hole shoulders and jig gouged into the material and therefore the data was not a true indication of the torsional stiffness. This demonstrates that the behaviour of aluminium is prone to incidental damage due to its low hardness, high stiffness and vulnerability to mechanical damage. The load rose sharply for very small angular deflection of the beam compared with that of the composite bar.

**VIII. Conclusion**

The aim of this project has been fulfilled to within the scope anticipated by the achievement of the following tasks. A review was conducted and state of the art considered, design requirements were developed and the relevant standards identified. A preliminary design was produced and developed including a comprehensive analytical estimation of the proper angle of precone. A testing regime was initiated and areas for improvement to the design identified.

The project has shown that simple hand laid glass epoxy rotor hub bars have potential strength properties in excess of aluminium and will likely have a higher resistance to the mechanisms of fatigue under the dynamic nature of the loads associated with gyroplane rotor hub bars due to their higher modulus of elasticity and superior ultimate strengths.

With further development and testing the proposed glass epoxy hub bar may provide a suitable alternative to the present design. The author has shown that minimal tooling and resources are required to facilitate production of glass epoxy hub bars. The hub bar assembly preliminary design reduces the number of parts necessary over current aluminium designs.

The analytical analysis of the dynamic loads and the estimation of composite properties indicate that the factor of safety that a glass epoxy design can potentially provide is well in excess of that of the current aluminium hub bar design.

**IX. Recommendations**

The benefits of progressing this project to a successful design has the potential to improve gyroplane safety and demonstrate that composite technology is within reach of the skill and budget of recreational aircraft operators. However the immediate recommendations for the future of this project include producing a sample incorporating modified attachment bolt sleeves. These should be tested in tension to the value of the maximum anticipated tensile load multiplied by the appropriate safety factor with the coupling bolted across two bolt holes as the design anticipates it will be installed in the aircraft. Further tensile testing should prove beyond doubt that the tensile properties exceed those of aluminium by a significant multiple.

The method of manufacture requires a high level of quality control and therefore filming, documenting and detailing the process to aid in exacting the procedure and proving consistent quality is achievable is essential to the hub bar being produced in quantities to supply end users.

Fatigue testing a full scale prototype in a machine simulating real combined loadings is essential to acceptance of the design and the author has begun to investigate the design of a simple rotating machine design to this end. Successful completion of such test would be irrefutable evidence of the designs superiority or otherwise in comparison to aluminium hub bar.

**Acknowledgements**

The author would like to acknowledge support for this project provided by Alan Fien the thesis supervisor, Mark Horan for his knowledge on gyroplanes and use of his aircraft to collect data. Steven Wardill for starting this research and inviting the author to continue the work he started. Pat Nolan for assisting with the component testing. Marcos de Almeida for assistance in the workshop. Rikard Heslehurst for instruction and assistance in
the composite lab. Specialized Forklift Services for donating the Renold leaf link chain. UNSW ADFA for workshop and financial resources. academic staff and undergraduate peers for thoughtful feedback and positive criticism. Peta Lonergan for being a supportive wife while I committed to study engineering.

References


