Human Powered Helicopter

Cal Poly Aircraft Construction Club

Brenton Haven - bhaven@calpoly.edu
Daniel Hudson - dhudson@calpoly.edu
Eli Knight - eknight@calpoly.edu

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# Table of Contents

List of Figures ................................................................................................................................. 4  
List of Tables ..................................................................................................................................... 7  
Executive Summary ............................................................................................................................ 8  
Introduction ....................................................................................................................................... 9  
Objectives ......................................................................................................................................... 10  
Background ....................................................................................................................................... 12  
The DaVinci III Design ...................................................................................................................... 12  
Evolution of the DaVinci Series Helicopters ..................................................................................... 13  
DaVinci I .......................................................................................................................................... 13  
DaVinci II ......................................................................................................................................... 13  
DaVinci III ....................................................................................................................................... 13  
New Problems ................................................................................................................................. 14  
The DaVinci IV ................................................................................................................................ 14  
Human Powered Flight ..................................................................................................................... 15  
Conventional Helicopter Comparison ............................................................................................ 15  
Design Development ....................................................................................................................... 19  
System .............................................................................................................................................. 19  
Fuselage .......................................................................................................................................... 22  
Introduction ..................................................................................................................................... 22  
Concepts ......................................................................................................................................... 23  
Analysis .......................................................................................................................................... 24  
Conclusion ....................................................................................................................................... 33  
Drivetrain ......................................................................................................................................... 33  
Drivetrain Requirements .................................................................................................................. 33  
Chain Drives ................................................................................................................................... 36  
Belt Drives ....................................................................................................................................... 37  
Linear Oscillating Transmission ........................................................................................................ 38  
Electrical .......................................................................................................................................... 39  
Research Conclusions ..................................................................................................................... 40  
Concept Analysis .............................................................................................................................. 40  
Efficiency ......................................................................................................................................... 40
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frictional losses due to a bushing</td>
<td>44</td>
</tr>
<tr>
<td>Ball Bearings Losses</td>
<td>45</td>
</tr>
<tr>
<td>Interfacing the bearing design</td>
<td>47</td>
</tr>
<tr>
<td>Gearing</td>
<td>47</td>
</tr>
<tr>
<td>Drive Shaft sizing</td>
<td>49</td>
</tr>
<tr>
<td>Spool</td>
<td>51</td>
</tr>
<tr>
<td>Conclusion</td>
<td>51</td>
</tr>
<tr>
<td>Rotor Hub</td>
<td>52</td>
</tr>
<tr>
<td>Introduction</td>
<td>52</td>
</tr>
<tr>
<td>Loads</td>
<td>54</td>
</tr>
<tr>
<td>Hub Concepts</td>
<td>55</td>
</tr>
<tr>
<td>Hub Analysis</td>
<td>57</td>
</tr>
<tr>
<td>Bearing Analysis</td>
<td>59</td>
</tr>
<tr>
<td>Rotor Shaft Analysis</td>
<td>64</td>
</tr>
<tr>
<td>Rotor Connection Analysis</td>
<td>65</td>
</tr>
<tr>
<td>Manufacturing</td>
<td>67</td>
</tr>
<tr>
<td>Fuselage</td>
<td>67</td>
</tr>
<tr>
<td>Introduction</td>
<td>67</td>
</tr>
<tr>
<td>Wax Investment Mold</td>
<td>67</td>
</tr>
<tr>
<td>Composite Layup</td>
<td>70</td>
</tr>
<tr>
<td>Rotor Hub</td>
<td>72</td>
</tr>
<tr>
<td>Short Fiber Composites</td>
<td>72</td>
</tr>
<tr>
<td>Molds, Tooling, and Post Machining</td>
<td>76</td>
</tr>
<tr>
<td>Test Verification Plan</td>
<td>84</td>
</tr>
<tr>
<td>Results</td>
<td>86</td>
</tr>
<tr>
<td>Conclusion</td>
<td>90</td>
</tr>
<tr>
<td>Management Plan</td>
<td>93</td>
</tr>
<tr>
<td>Conclusion and Recommendations</td>
<td>94</td>
</tr>
<tr>
<td>Works Cited</td>
<td>96</td>
</tr>
</tbody>
</table>
List of Figures

Figure 1: Drive train system from DaVinci III. ................................................................. 12
Figure 2: DaVinci III during record setting flight. ............................................................. 14
Figure 3: Coning angle which is the defined as the angle of each blade measured from a line perpendicular to the axis of rotation in hover. Image courtesy of http://www.cavalrypilot.com/fm1-514/Ch2.htm .......................................................................................................................... 16
Figure 4: Cal Poly DBF Club model airplane wing that inspired the current rotor design for DaVinci IV. .. 17
Figure 5: Scale model rotor for aerodynamic testing building in progress .................................. 17
Figure 6: End view of scale model rotor wing showing the profile as well as the carbon fiber spars and Aramid fiber bracing ............................................................. 17
Figure 7: Figure from page 73 from [Prouty] showing how the rotor flapping works to stabilize the helicopter ................................................................. 18
Figure 8: Gyroscopic stability control. The center figure shows the conceptual configuration sketch of how a gyroscopic control system might look. The figure at left from page 100 of [Prouty] shows the gyroscopic stability control designed by Lockheed-Martin in the 1950's. The figure at right shows the effects of the same gyroscopic stability control on the helicopter from page 547 of [Prouty]. ........................................... 19
Figure 9: Isometric view system overview ............................................................................. 20
Figure 10: Front view system overview ................................................................................. 21
Figure 11: Side view system overview .................................................................................... 21
Figure 12: Body configuration angles (Reiser, Peterson, & Broker, 2001) ........................... 23
Figure 13: Conceptual drawings Teardrop (top left), Pole (tor right), Upright (bottom left), Hammock (bottom right) ......................................................................................... 24
Figure 14: Second iteration concepts evaluated for weight .................................................... 25
Figure 15: Final fuselage geometry ........................................................................................ 26
Figure 16: Hinge joint ............................................................................................................. 27
Figure 17: Final fuselage iteration .......................................................................................... 27
Figure 18: Pole design static analysis .................................................................................... 28
Figure 19: Bottom bracket section analysis .......................................................................... 28
Figure 20: Seat joint section analysis .................................................................................... 29
Figure 21: Pole section analysis ............................................................................................ 29
Figure 22. Material Orientation of the fuselage Frame ......................................................... 30
Figure 23. Mesh convergence on the deflection of node 59 on the seat of the frame ............. 31
Figure 24. Mesh Convergence of node 131 near the bottom fillet of the frame .................. 31
Figure 25. Stress distribution on the final layup .................................................................... 32
Figure 26. Deflection of the frame under worst case loading ............................................... 33
Figure 27. Spectra Fiber wound around a propeller shaft .................................................... 35
Figure 28. Diagram spectra fiber feed into the rotor shaft .................................................... 35
Figure 29. Spectra fiber running to a spool on a belt driven driveshaft ............................... 36
Figure 30: Gates Carbon belt drive technology applied to bicycle technology .................... 38
Figure 32. Conceptual drawing of the electrical drive system ............................................... 39
Figure 31: Linear Oscillating drive system design ................................................................. 39
Figure 33: Diagram of the frictional forces in the spectra thread .......................................................... 40
Figure 34. Free body diagram of one of the eyelets ................................................................................. 41
Figure 35. Loading case for a wing with half the lift force at the centroid triangular lift distribution ....... 42
Figure 36. Free body diagram approximating the friction is a slipping belt (left) Free body diagram of the belt length ds (right) ...................................................................................................................... 44
Figure 37. The figure on the left show the forces and moments required to maintain constant rotation. The figure on the right shows the frictional circle. (Hibbeler, 2007) ................................................................. 44
Figure 38. Simply supported beam modeling the shaft for the ball bearings to rotate around ............... 47
Figure 39. FBD in the xy plane .................................................................................................................. 49
Figure 40. FBD in the xz plane .................................................................................................................. 49
Figure 41. Bending moments on the shaft based on location on the shaft .................................................. 50
Figure 42: The final design of the rotor hub systems for the DaVinci IV. The hub is comprised of Graphlite pultruded carbon fiber tubes configured into a truss to carry loads out to the rotor. The eight rings at the corners of the truss are a short fiber carbon composite as are the central rings that join the truss. The rotor shaft is a carbon fiber tub reinforced where the spectra thread enters. The rotor bearing assembly is bonded into the bottom of the shaft housing two SKF51106 thrust ball bearings........................................ 52
Figure 43: DaVinci III Rotor Hub. This figure shows that the fuselage is tied to the rotor which allows the fuselage to swing beneath the rotor. For reference some fishing swivels have been included in the bottom right hand corner of the figure. ........................................................................................................................................ 53
Figure 44: Rigid rotor hub from a Eurocopter BO-105. This illustrates a rotor hub bearing attachment where the fuselage can exert a force on the rotor disk and change the attitude of the rotating disk. The rigid rotor system of this particular helicopter makes it suitable for aerobatic maneuvers. Photo courtesy of the Wikimedia commons......................................................................................................................... 53
Figure 45: Free body diagram and mass acceleration diagram of the fuselage and rotor at maximum instability.................................................................................................................................................. 54
Figure 46: Free body diagram and mass acceleration diagram of the fuselage using the model of a pendulum and applied forces that are sufficient to move the fuselage ................................................................. 55
Figure 47: This figure shows the three main rotor hub concepts that led to the final version. Each of these concepts was eliminated due to excessive weight. ........................................................................ 56
Figure 48: The final DaVinci IV rotor hub system showing the connection between the rotor spars and the rotor hub system. .................................................................................................................. 56
Figure 49: The rotor hub truss system without the rotor shaft................................................................. 57
Figure 50: This figure illustrates the two part decomposition of the rotor hub to simplify the statically indeterminate structure for analysis. ..................................................................................................... 58
Figure 51: Free body diagram used to analyze the loads in the rotor hub.................................................. 58
Figure 52: Free body diagram and mass acceleration diagram of the fuselage and rotor during stable hover.................................................................................................................................................. 60
Figure 53: Excavator turntable bearing used to illustrate how a heading hold orientation system might be adapted to future DaVinci iterations. http://img.diytrade.com/cdimg/726150/5858397/0/1236611648/Turntable_bearing.jpg ............. 61
Figure 54: Section view of the DaVinci IV rotor hub bearing showing the four point contact bearing and the assembly to capture that bearing between the two rotor shafts .................................................. 62
Figure 55: Exploded view of the final selection rotor bearing using a Kaydon KAA17XLO. This design provided sufficient stiffness with the least weight.................................................................62
Figure 56: SKF 51106 Hub Bearing Assembly. This assembly shows assumes a rigid attachment which can handle a very severe moment load on the shaft however this design was not selected due to manufacturing complexity and extra weight.................................................................63
Figure 57: Two different rotor bearing concepts. The KB020XP0 assembly is a 4-point contact bearing that was deemed too heavy for this application. The SKF GZE008ES assembly assumes a spherical bearing which would make the rotor able to flap which is undesirable in this aircraft.................................................................63
Figure 58: Free body diagram of the rotor hub modeled as a cantilever beam.................................65
Figure 59: Exploded view of the rotor hub connections which permit removal of the rotors from the aircraft...........................................................................................................66
Figure 60: Machining initial wax test pieces to derive feeds and speeds for production......................68
Figure 61: Hand shaping wax cylinders (this is the bent tube in the middle of the frame)......................68
Figure 62: Final wax mold right before the layup........................................................................69
Figure 63: Fuselage mold vacuum bagged after a layup................................................................69
Figure 64: Final part hung in the over for wax melt-out.................................................................70
Figure 65: Wetting out carbon fiber strips for a layup....................................................................71
Figure 66: Complete fuselage ......................................................................................................72
Figure 67: Solidworks model showing exploded spar ring mold next to a spar ring.......................73
Figure 68: Spar ring mold disassembled and covered in Frekote....................................................74
Figure 69: Spar ring mold assembled and open, ready for composite. Note that the top half of the mold will reduce the volume of the female mold cavity when the two halves come together..........................74
Figure 70: Spar ring mold filled with composite matrix. Note the coarse appearance of the fiber........75
Figure 71: Spar ring mold in hydraulic press. Note the excess resin oozing from the sides of the mold...76
Figure 72: Photo of spar ring mold showing the channel cut to allow excess resin to escape the mold...77
Figure 73: Polishing spider ring mold with Tripoli compound and a felt bob in a die grinder. Note the finish is lustrous but not mirrored..........................................................78
Figure 74: Completed spider ring mold after 8 parts. .................................................................78
Figure 75: Rotor hub truss in fixture while pultruded rods are being fit............................................79
Figure 76: Test fixture supporting completed DaVinci IV fuselage and drivetrain...........................80
Figure 77: Completed rotor hub with strain gauges still bonded....................................................81
Figure 78: Drill fixture for spar rings. Note that the parts rest against 1/4" SHCS that orient them with respect to the axis of the rotor spars. There are screws on both sides of the part which allow both right hand and left hand parts to be drilled.................................................................82
Figure 79: Drilling spar rings on fixture in Haas mill........................................................................83
Figure 80: Strain gauges locations to determine rider input loads..................................................84
Figure 81: Team member Eli Knight safely sits in the DaVinci IV...................................................87
Figure 82: Strain gage testing setup with team member Eli Knight pedaling 45 lb weights..............87
Figure 83: Strain gage testing setup for dynamic load data............................................................88
Figure 84: Dynamic strain gage data from the first test run...........................................................89
Figure 85: Microstrain oscillation at rider cadence 60 rpm and 80 rpm.......................................90
Figure 1: Gossamer Albatross propeller mechanics........................................................................100
List of Tables

Table 1: Pole and Teardrop concept surface area's and weights .......................................................... 26
Table 2: Fuselage parts and weight ........................................................................................................... 29
Table 3. Drivetrain Design Requirements for the DaVinci IV ................................................................... 34
Table 4: Table of efficiency and weight values for the two best competing continuous drive components .......................................................... 38
Table 5. Table of the curvature and forces at each of the wing eyelet locations ....................................... 43
Table 6. Efficiency results from the 90 degree turn analysis .................................................................... 45
Table 7. Overall system efficiencies and weight for possible drive systems ............................................. 46
Table 8. Gates belt sprocket sizes ............................................................................................................ 48
Table 9. Possible Gearing Options with Gates Carbon Drive Belt systems .............................................. 49
Table 10. Bending Moments on the drive shaft linking the belt driven system with the spectra system ........ 50
Table 11. Comparison between bushings and needle bearings at the drive shaft ........................................ 51
Table 12: Rotor bearing weight comparison ............................................................................................. 64
Table 13: Carbon fiber specifications ....................................................................................................... 70
Table 14: Layup scheme ............................................................................................................................ 70
Table 15: Assembly weights ..................................................................................................................... 86
Table 16: Part weights ............................................................................................................................... 86
Table 17. Static experimental and theoretical results .................................................................................. 88
Table 18. Frequency data for dynamic strain run 1-3 ................................................................................ 89
Table 19. Test Plan .................................................................................................................................... 92
Executive Summary

Fall of 2009, The Cal Poly Aircraft Construction club restarted Cal Poly's quest for the Sikorsky prize. The Sikorsky prize rewards the first a human powered helicopter to sustain controlled hover for one minute without stored energy. Throughout the 1980’s, Cal Poly made three attempts; the most successful being the DaVinci III. Also, the DaVinci III was the first ever publically recognized successful human powered helicopter to leave the ground. This scope of this project is to improve the DaVinci III fuselage and drivetrain for the DaVinci IV.

The DaVinci IV adopts the DaVinci III system layout and improves both weight and efficiency. The helicopter will be a single tip driven rotor with a single pilot. The tip propellers will be turned by unspooling thread at the propellers with a winch spool that the rider will be supplying power to. Specifically this document contains the fuselage, rotor hub, and drive train for the DaVinci IV.

The fuselage functions as the support structure for the drivetrain and rider. The fuselage will maintain an efficient riding position while providing a rigid structure for the drivetrain to transmit power from the rider to the propellers. The rotor hub will attach the fuselage to the rotor while the rotor rotates above the pilot. The drive train will be a winch driven by the rider’s pedal strokes. The winch will spool the thread from the propellers, thrusting the propellers forward. The DaVinci IV will be 30% lighter and better drivetrain efficiency.

This project will provide the framework for Cal Poly's next attempt at the Sikorsky prize. This project includes valuable research from the previous DaVinci helicopters, uses engineering techniques to understand unknown flight characteristics, and provides a recommendation to the future DaVinci series helicopters. Upon completion, this project will test fuselage, drivetrain, and rotor hub with data to help future designers model the system.
Introduction

This project is a continuation of the DaVinci III human powered helicopter that first flew in December 1989 and set the record as the world’s first human-powered helicopter to leave the ground. Conceptually, the DaVinci series of human powered helicopters are a large (approximately 100ft.) rotating airplane wing driven by two wingtip mounted propellers; one on each side. Instead of a traditional torque transmission device as in an engine driven helicopter, the design uses a high tensile thread wrapped around the propeller shafts and unwound on a pedal-driven take-up-reel mounted on the fuselage to minimize weight and the need for anti-torque measures. The fuselage is effectively a recumbent bike hanging from the wing center and does not provide any stability for the craft. On DaVinci III, the rotor hub was essentially a string that tied the fuselage to the wing in order to use momentum for fuselage anti-torque. See Figures 3, A.6, and A.7 the appendix for photos of previous DaVinci aircraft.

This project will begin the next attempt from Cal Poly at winning the Sikorsky Prize. The previous DaVinci proved too unstable for sustained flight, and hence, there is very little conclusive data on system performance and in particular stability. There are three major components and a total of two critical systems. Our responsibility is the fuselage component which contains the rotor hub and drive train systems. We are not responsible for the main rotor or tip propeller components. The overall club goal is to win the Sikorsky prize the first step is to create a working models and collecting data. The wing group will begin by building 1/10th scale wings and testing for stiffness and lift characteristics. Our immediate goal is to build a full size and working fuselage, drive train, and rotor hub. These components and systems will be working and flyable as soon as full scale wings are built. Our complete project includes building the full scale parts as well as performance testing them.

Our airframe and power system will benefit the Cal Poly Aircraft Construction club by providing a useable platform for testing and data collection so that the next group can investigate stabilizing this aircraft concept and test those ideas on a working structure.
Objectives

The goal of this project is to build a full sized and functional chassis and power transmission system for the Da Vinci IV human powered helicopter. The wing and propeller groups are working on testing scale models which are relatively easy to scale by the properties of fluid mechanics. The fuselage on the other hand would be difficult to test as a scale model since it is built for a human, and also certain components such as bicycle chain do not easily scale. Due to this scaling issue this project will be a full scale airworthy fuselage and power train. The attached house of quality shows us that the main focus for this project is weight. This is directly related to the amount of lift required for flight and the human power requirement to generate the lift. In the specification table below weight is listed as a high risk objective because this is the most critical part for our future vehicles flight. Previous DaVinci teams focused on analysis and construction and did not collect data to use in future DaVinci designs. In addition to the requirements listed in the specification table, this project requires data collection for the power train and power distribution systems. The table below outlines the requirements for the entire DaVinci IV fuselage and power train while more specific requirements for each subsection derived from the table are listed in Table 1.

Table 1: Design specifications - Risk levels: high (H), medium (M), low (L). compliance analysis method: analysis (A), inspection (I), testing (T).

<table>
<thead>
<tr>
<th>Specification</th>
<th>Minimum</th>
<th>Target</th>
<th>Maximum</th>
<th>Risk</th>
<th>Compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Target Weight [lbs.]</td>
<td>N/A</td>
<td>16</td>
<td>20</td>
<td>H</td>
<td>T</td>
</tr>
<tr>
<td>Average Power To Wingtip Propellers [Watts]</td>
<td>350</td>
<td>400</td>
<td>750</td>
<td>M</td>
<td>A,T</td>
</tr>
<tr>
<td>Power Transmission Efficiency [%]</td>
<td>80</td>
<td>94</td>
<td>N/A</td>
<td>L</td>
<td>A,T</td>
</tr>
<tr>
<td>Coning Angle From Rotor Hub [degrees]</td>
<td>N/A</td>
<td>0.286</td>
<td>0.55</td>
<td>M</td>
<td>A,T,I</td>
</tr>
<tr>
<td>Height Of Rotor Center [ft.]</td>
<td>4.5</td>
<td>5</td>
<td>5.5</td>
<td>L</td>
<td>I</td>
</tr>
<tr>
<td>Rotor Hub Power Loss [Watts]</td>
<td>N/A</td>
<td>10</td>
<td>25</td>
<td>M</td>
<td>A,T</td>
</tr>
<tr>
<td>Tip Mounted Propeller Speed [Rev/Min]</td>
<td>60</td>
<td>180</td>
<td>500</td>
<td>L</td>
<td>I</td>
</tr>
<tr>
<td>Rotor Tip Speed [ft./sec]</td>
<td>48</td>
<td>50</td>
<td>52</td>
<td>L</td>
<td>A</td>
</tr>
<tr>
<td>Pedal Cadence [Rev/Min]</td>
<td>80</td>
<td>100</td>
<td>120</td>
<td>L</td>
<td>A,T</td>
</tr>
<tr>
<td>Compatibility 1</td>
<td>Must accept carbon wing spar pattern</td>
<td>M</td>
<td>I</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compatibility 2</td>
<td>Must meet Sikorsky prize rules</td>
<td>L</td>
<td>I,A</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compatibility 3</td>
<td>Rotor removable for transport</td>
<td>L</td>
<td>I,T</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

1. Develop a lightweight structure to support the pilot and transmit human pedaling power to the drive train. The fuselage will be subject to the following design criteria:
   - Total weight goal 6.4 lbs. (Not including drive-train or rotor hub)
   - Maximum weight goal 8.0 lbs.
   - Stiff enough for 98% power transmission efficiency
   - Positions CG directly below rotor hub
   - Comfortably position the rider for optimum power output
   - Center of airfoil no more than 5.5ft. from the ground
2. Develop a rotor hub that minimizes reaction torque the fuselage and can efficiently distribute the bending loads imposed by the rotor. The rotor hub will be subject to the following design criteria:
   - Total weight 3.2 lbs.
   - Maximum weight goal 4.0 lbs.
   - Maximum coning angle due to rotor hub is $\theta=0.286$ degrees with bending loads of 4000ft-lb.
   - Must minimize rotor reaction transmitted to the fuselage
   - Must counteract any residual rotor reaction torque so that pilot does not rotate
     - Power consumption not to exceed 25 Watts
   - Accepts four carbon spars from each wing. Spar sizes to be determined
   - Allows wing removal for transportation

3. Develop a power transmission system that can transmit pedal power to wing tip mounted propellers on the rotor disk. The power system will be subject to the following design criteria:
   - Total weight goal 6.4 lbs.
     - Maximum weight goal 8.0 lbs.
   1. Must transmit an average of 400 Watts to the rotor tip mounted propellers
   2. 90% efficiency
   3. Rotates tip propellers at 120 RPM
   4. Rotates rotor tips at 50 ft/sec.
   5. Smooth out sinusoidal pedaling motion so that force applied to the propellers varies by no more than 5%
   6. No energy storage per Sikorsky Prize rules

4. Collect data on the power transmission system to assess parasitic losses and numerical data for future calculations. The project hopes to benefit from the following data:
   7. Total power output
      - Pilot power output
      - Drive-train efficiency
      - Propeller output
   8. Fuselage rigidity
      - Deflection due to rider input
      - Fuselage vibration frequencies and oscillations
   9. Rotor hub torque reaction
      - Power is required to counteract the rider precession
      - Rotor hub power loss
Background

The DaVinci III was the first human powered helicopter to officially leave the ground and since then only the Yuri I from Nihon University in Japan has taken flight. Neither of these two entrants even came close to satisfying the Sikorsky prize rules which are posted on the American Helicopter Societies website [http://www.vtol.org/awards/hphregs.html](http://www.vtol.org/awards/hphregs.html). There are three main parts to the Sikorsky prize rules. First, the flight last for at least one minute. Second, the flight must be sustained totally with human power, nothing lighter than air and no stored energy. Third, the lowest part of the aircraft must instantaneously reach an altitude of three meters from the ground.

The DaVinci III Design

The DaVinci III consisted of a two bladed single rotor design with tip propellers (See Figure 1: Drive train system from DaVinci III.). The thrust of the propellers spin the rotor blades providing a torque less reaction on the fuselage. The propellers are wound with spectra thread. This Sprectra® thread in an open loop configuration runs down the wing to a wench spool located within the fuselage. The winch spins on shaft connected to a sprocket attached to a chain drive receiving the rider input (See Figure 2: DaVinci III during record setting flight.). A flywheel was added for momentum in the system on the same shaft as the winch and rear sprocket.

The rider support structure hangs from the rotor hub. The wings are supported by an aluminum truss system with casters on the bottom allowing the wings to spin around the rider with little added friction. The frame structure for the rider is constructed from two by fours wrapped in carbon as can be seen in the figure showing the drive train system.

Figure 1: Drive train system from DaVinci III.


**Evolution of the DaVinci Series Helicopters**

The DaVinci helicopter series began in 1981 under the leadership of Dr. Patterson at California Polytechnic University in San Luis Obispo. Three possible systems were investigated, a single tip driven rotor system, a dual rotor coaxial system or tandem drive. Due to the slow rotation speed of the rotor blades the DaVinci rotor design seems more similar to windmill design rather than a helicopter. The efficiency of windmills increases with the fewer number of blades due to flow interference. As a result some windmills have been designed with a single blade. Based on efficiency this favors the single tip driven rotor design. However, driving the propellers a large distance away from the power source will accrue significant losses. A coaxial system will be much more efficient and compact, but would require a minimum of four blades. A tandem system would also require four blades and a more complex support structure. Thus, the group decided to attempt a single tip driven rotor system due to fewer blades required and simpler support structure.

**DaVinci I**

The original DaVinci design positioned the rider above the wings to take advantage of ground effects. Upon test the wing structure flexed significantly demanding a use of a guy wire system. However this raised the wings three and a half feet which eventually led to the decision to move the rider below the aircraft.

**DaVinci II**

The next iteration of DaVinci design kept the same tip driven propeller system with addition of a guy wire system. During testing a Vandenberg AFB the helicopter failed to fly and suffered failure in one of the main spars. The spar was wrapped with unidirectional tape on the top and bottom which eventually buckled in the guy wire system.

**DaVinci III**

The DaVinci III design supported the wing structure with aluminum tubing on casters, leaving the rider and drive train suspended from the rotor hub. A flywheel was added because riders complained about trying to pedal smoothly. In response they added a fly wheel for additional momentum in the drive train system. With these improvements the DaVinci III became the first human powered helicopter to leave the ground seen below in Figure 3: Coning angle which is the defined as the angle of each blade measured from a line perpendicular to the axis of rotation in hover. Image courtesy of http://www.cavalrypilot.com/fm1-514/Ch2.htm.
Figure 2: DaVinci III during record setting flight.

New Problems

Achieving flight opened the door to new challenges to the aircraft including stability and control. There was also an additional problem; even with the advanced guy wire system, the wing structure experienced a significant amount of flexing and inconsistent stiffness. The next iteration DaVinci series aircrafts will be a lighter and stiffer version of the DaVinci III design. The intent behind this design is to eliminate some stability issues caused by inconsistencies and allow for accurate analysis of the helicopter.

The DaVinci IV

Currently, the Aircraft Construction Club is in the process of redesigned the wing structuring for testing. The new wing uses four small carbon spars, replacing the large carbon single spar. Tensioned Kevlar string will provide a shear web between the ribs and carbon rods. This structure places the carbon rods in tension and resists torsion.

The main limitation with human powered flight is the power output. As a result the traditional main rotor and tail rotor arrangement is not feasible and the typical design approach for a regular helicopter does not directly apply to this situation. In particular, the large main rotor is designed to spin very slowly to minimize drag (due to only small pressure differences) and reduce the amount of power required to spin the rotor. Due to the slow speeds a large plan-form area is required in order to generate the required lift to get the helicopter airborne.
Human Powered Flight

The two human powered helicopters that have succeeded in lifting off have been limited in flight duration by instability. Based on videos of the flight the DaVinci III with its huge rotating wing had sufficient power to lift the craft but sustained flight for only 8 seconds before becoming unstable. This is a potential problem with this design since the craft is essentially a rotating wing that is tethered in the middle and so in the previous designs there are no forces to counteract variations in lift. The Yuri design from Nihon hovered for 19.46 seconds but suffered from a different type of instability. The Yuri design by Professor Niato of Tokyo created four separate rotor systems each suspended from trusses because he claimed that his design will have the best thrust to power ratio. Yet he will admit the aerodynamics of the flight are still to be examined and much can be learned from others attempts [IHPVA vol.11 no. 4, pg19]. Additionally this design proved stable from a tipping because the lift is being applied from four rotors that act as legs. However the Yuri had no control surfaces to change location with respect to the ground and drifted away from its original location and was forced to stop.

Several other attempts have been made to build a human powered helicopter but the common causes of failure are excessive weight, mechanical complexities, and insufficient disk area. The most notable so far is the University of British Columbia which has two co-axial counter rotating rotors with two blades each that spin at 6 RPM. They have not been able to fly due to the limitations listed above. Many other configurations have been tried ranging from garage builds to school projects but no other projects have come close.

Other types of human powered aviation have been more successful. Paul MacCready designed the Gossamer Condor which won the first Kremer prize. A later iteration of the design the Gossamer Albatross, flew across the English Channel to win the second Kremer prize. The MIT's Daedalus pedaled 74 miles across the Aegean Sea [http://www.ihpva.org/air.htm]. Each of these successes was designed with a similar formula: large wing plan-form, low velocity, minimal weight, and simplistic drive-train. The Gossamer Albatross has been highly documented and we have looked at many schematics for inspiration for this next craft in the DaVinci series (Figures A.2-A.4 located in the appendix).

Conventional Helicopter Comparison

While this project is called the human powered helicopter it is probably more appropriately called a human powered rotor-craft since the only trait that this vehicle shares with a conventional helicopter is that the lifting surface rotates about a stationary fuselage. Instead this machine behaves more like a very slow airplane similar to the Gossamer Albatross human powered airplane which flew across the English Channel. The reason for this similarity is the power output limitation imposed by using a human as a power-source. A traditional helicopter has relatively small rotor blades that spin very fast where the DaVinci has very large rotor blades that spin at very slow speeds. Drag on the rotor blade increases approximately proportional to the square of the rotor speed and accordingly the power required to turn a traditional rotor through the air is significant but can be compensated for by increasing the engine power output. Since the power output for a human powered craft is essentially fixed, the DaVinci series of aircraft is designed around a very large rotor that spins at a very low angular velocity. For example where a Bell UH-1 rotates at approximately 500RPM with tip speeds reaching the speed of sound, the DaVinci IV is designed for a 10RPM operating point with tip speeds of 50 ft/s.
Because the rotor is spinning very slowly the rotor blade operates with very low lifting pressure of approximately 0.5lbf/ft\(^2\), compared to lifting pressures on a conventional helicopter as high as 11lbf/in\(^2\) [Prouty].

The high blade loading on a conventional helicopter is possible because of the centrifugal force generated by the high angular velocity of a conventional rotor system which works to minimize deflection of the cantilever beam shaped rotor called coning angle outlined in Figure 3. The DaVinci series helicopters on the other hand do not benefit from centrifugal force and so the rotors must be designed as a rigid cantilever beam system that can withstand 5000ft-lbf bending moments. This requirement is one of the central design themes for the DaVinci IV and is addressed with the wing concept introduced in Figure 4-Figure 6. While this concept is the responsibility of a different group, it must be thoroughly understood since it is the driving design criteria for this project. The wing is designed as a beam with four carbon fiber spars that are tensioned into a truss with an Aramid fiber thread. From the large separation distance between the spars, in this case 14”, the bending moment is effectively turned into tension and compression loads down the axis of the carbon fibers which exploits the benefits of carbon fiber with its exceptional tensile strength. Torsion and shear are controlled by lacing the spars and rotor ribs together with the high tensile thread. This concept puts the carbon fiber spars in tension/compression of approximately 2.5ksi compared to the tensile of 105ksi. This calculation is detailed in appendix E.

Figure 3: Coning angle which is the defined as the angle of each blade measured from a line perpendicular to the axis of rotation in hover. Image courtesy of http://www.cavalrypilot.com/fm1-514/Ch2.htm
Figure 4: Cal Poly DBF Club model airplane wing that inspired the current rotor design for DaVinci IV.

Figure 5: Scale model rotor for aerodynamic testing building in progress.

Figure 6: End view of scale model rotor wing showing the profile as well as the carbon fiber spars and Aramid fiber bracing.
The slow rotation speed which requires the substantial cantilever wing section also presents some substantial control issues that are not present on a traditional helicopter. The high rotation speed of a conventional helicopter and the mass of the rotor blade provides for gyroscopic forces which act to keep the rotor disk stable due to the effects of gyroscopic precession. This restoring force is decidedly absent in the DaVinci IV rotor system due to the low rotation speeds so the stability inherent to conventional rotor disk is left for other systems to deal with. This requirement presents itself again in selecting the rotor-hub bearing. The other consideration area where the slow rotation of the rotor changes the behavior compared to a conventional helicopter is in rotor blade flapping. As illustrated in Figure 8, the rotor disk tilts with respect to the helicopter fuselage in order to equalize the angle of attack for each rotor so that the list is equalized between the two rotor blades. The problem is that the traditional helicopter model assumes a rotating disk where the rotor for DaVinci is more like a slow rotating wing. The distinction between a wing and a disk is significant. The disk model behaves like a gyroscope with the associated gyroscopic stability, while the wing acts like a slowly rotating bar that has a low moment of inertia about the rotor axis and is therefore subject to pitch instability. Additionally for a traditional helicopter the pitch of the blades changes with respect to the helicopter whereas for DaVinci there is no cyclic and consequently no pitch change with respect to the fuselage. While a control system is explicitly not part of this project, these physical limitations must be recognized in designing each system.

Figure 7: Figure from page 73 from [Prouty] showing how the rotor flapping works to stabilize the helicopter.
The DaVinci III used a flywheel to smooth out the pedaling input from the rider. We analyzed a potential concept which located the flywheel above the rotor disk and constructed it with a large moment of inertia to act as a stabilization device and a possible re-circulating drive system. This concept is described pictorially in Figure 8 and compared to a viable concept presented by Lockheed-Martin in the early 1960’s. Some simple calculations included in the appendix show that for the flywheel to have a significant control input, the size would exceed both weight and practicality requirements. Additionally this concept would require a shaft to protrude through the rotor hub which would apply an additional torque to the fuselage and would require too much power to counteract.

![Figure 8: Gyroscopic stability control. The center figure shows the conceptual configuration sketch of how a gyroscopic control system might look. The figure at left from page 100 of [Prouty] shows the gyroscopic stability control designed by Lockheed-Martin in the 1950’s. The figure at right shows the effects of the same gyroscopic stability control on the helicopter from page 547 of [Prouty].](image)

**Design Development**

**System**

In order to successfully fulfill the requirements of this project, we must have a design that holds a rider, captures energy from the rider, transmits the energy to the propellers, and lets the rotor spin freely above the rider. We have broken this project into one major system (fuselage), and two minor sub systems (drivetrain, rotor hub).
The function of the helicopter fuselage is to provide power to the propellers located at the tips of the wing. This is accomplished by a winch system driven by the riders pedaling which winds up spectra thread spooled at the propellers. The fuselage's role is to carry the rider and hold him in an efficient riding position. The fuselage is connected to the rotors through the rotor hub which is used to keep the rider facing forward while the rotors rotate over the riders head. All three of these parts make up the major mechanical system of the DaVinci IV.

Figure 9: Isometric view system overview
Figure 10: Front view system overview

Figure 11: Side view system overview
Fuselage

Introduction

The fuselage has some very strict requirements to fulfill in order to be an improvement over the DaVinci III.

1. Total weight 6.4lbs, max 8lbs
2. Position the center of gravity directly below the rotor hub
3. Position rider for maximum power output
4. Stiff enough to provide 98% power transmission efficiency

The main restriction is weight. This craft must weigh less than 8lbs and our goal is to weigh less than 6.4 lbs. We estimate this as a 30% decrease in weight over the DaVinci III which is a substantial improvement. The more weight on the craft results in a higher power requirement in order to generate flight. The rider's center of gravity must be directly under the rotor in order to not cause any induced pitch due to the rider's weight changing the orientation of the rotors. The next major requirement is that the rider must be positioned in his most efficient riding position. This ensures that the rider is comfortable and has no resistance to power output due to the design of the structure. Another goal is to provide a rigid fuselage in order to not dissipate rider output. We feel that these requirements are very strict and provide an excellent framework to design a successful fuselage around.

In order to stay with the concept of DaVinci III we have decided to maintain some major concepts from the DaVinci III. The rider will still provide all of the power to the craft with his legs similar to a bicycle. Our fuselage only needs to support the weight of the rider and loads from the drive train. The weight of the wings will be supported by a simple truss structure that will be attached to the bottom of the wings. This reduces the weight of the rotor hub and fuselage and also provides a relatively easy method to add stiffness to the wings. The choices help keep the scope manageable as well as give us a good shot at significantly improving over the DaVinci III.

Rider requirements helped drive the design initially because we could easily integrate rider geometry and performance at the time we started to design our fuselage. We have a Kinesiology student working with us to find an optimum rider and rider position and provide those rider's power output figures. From the preliminary research our rider will be a 5'9" male weighing 150lbs or lighter. The riders Body Configuration Angle (BCA, Figure 12: Body configuration angles) needs to be between 130-140 degrees. This has been proven to allow for most riders to output at their maximum power output. Rider efficiency is critical in any human powered vehicle and this is a major design constraint. Allowing the rider to produce his maximum amount of power removes unnecessary design constraints on other subsystems.
Concepts

Concepts for DaVinci IV were derived from previous DaVinci series designs, bicycles, and other human powered vehicles. The main considerations were placing the rider in an efficient riding position, allowing room for a drivetrain, and providing a means to connect the fuselage to the rotor hub. These constraints allowed for a variety of concepts but only a few designs proved to be feasible.

The first concept (Figure 13: Conceptual drawings Teardrop (top left), Pole (tor right), Upright (bottom left), Hammock (bottom right)) was essentially the same as the DaVinci III with a more efficient structure; DaVinci III had carbon fiber sandwich panels glued together whereas the new concept is a more efficient structure in a similar shape. This design can be good for integrating the seat and drive train structures into one large structure. There can be some potential weight savings and strength gains through a more organic hoop shape, improved manufacturing, and better materials.

The second concept was the pole design (Figure 13: Conceptual drawings Teardrop (top left), Pole (tor right), Upright (bottom left), Hammock (bottom right)). This design concept is essentially a pole hanging down from the rotor hub. The drivetrain will be held out in place by a structure that attaches to the bottom of the pole along with the seat. Additional support can be added by creating a truss structure connecting to the pole in a different location to the bottom bracket. This design has potential for being the easiest to connect to the rotor hub as it would essentially be a shaft attachment to the bottom of the rotor hub.

Our third concept was an upright design (Figure 13: Conceptual drawings Teardrop (top left), Pole (tor right), Upright (bottom left), Hammock (bottom right)). This design puts most riders in a familiar position, and initially we were not sure about efficiency differences between the recumbent and upright seating positions. We have now learned that at high rpm (80-120rpm) where the rider is most efficient there is negligible difference between the two different seating positions. In order to keep the
center of gravity lower and make the helicopter more stable we have selected the recumbent design. Another concern about the upright seating position is that pedaling up and down might induce vibrations into the long lengths of our wings.

The fourth concept was a hammock design. This design concept has a structure directly attaching the drive train components to the rotor shaft above the rider. The rider will be suspended below the structure in a hammock-like cloth which is attached to the structure. Some potential strengths of this design is the weight savings of a cloth seat and potential adjustability. This design was ruled out because the hammock is not fixed and pedaling could create a rocking motion in the hammock. We want the rider to be able to sit statically and not induce any extra vibrations into the system.

Analysis
From the initial concept phase we eliminated some designs for not being practical and decided to focus on two designs. The upright design was ruled out in order to eliminate the vibrations from the pedaling causing the helicopter to jump or bounce with the rider’s pedaling. The hammock design was eliminated because the soft back of the design could allow the rider to sway as he pedals. The tear drop
and pole designs were selected because they caused the fewest potential issues and had the most potential to be a successful iteration of the fuselage.

Analysis for the fuselage was done on first iterations after the preliminary concepts. After some basic analysis some major flaws were identified in each design and addressed to allow the two structures to be analyzed and compared closer to their final iterations. The two designs were compared based on weight and build complexity. Both second concept iterations were designed with ergonomics in mind in order to ensure that both concepts would put the rider in his most efficient position and most comfortable pedaling position. Early in our concepts and analysis it was decided that in order to save weight we would create a composite part to allow us to optimized our fuselage as much as we could. A composite structure allows us to adjust the strength of each section based on the loads that each specific section sees. We used static analysis to identify our critical locations of the frame. At these critical sections we used composite beam theory to identify a general layup scheme followed by optimization in MATLAB and further optimization using FEA. (Figure 14: Second iteration concepts evaluated for weight and Table 1: Pole and Teardrop concept surface area’s and weights).

The pole design was selected (Figure 15: Final fuselage geometry) for our final design due to being lighter than the teardrop design as seen in Table 1: Pole and Teardrop concept surface area's and weights. After close analysis of both structures it was apparent why the pole design was a lighter design than the tear drop design. The tear drop had two sections of carbon wrapped around the rider rather than one supporting in the middle. Because of carbon fiber’s strength and the relatively low loads that this craft sees it would be much harder to optimize the structure in the teardrop because of the significantly larger surface area than the pole design. During the initial analysis the load paths for the drivetrain in the teardrop design were significantly better. Because the weight of the rider was located directly under the rotor hub and in the center of the teardrop, most moments within the structure were eliminated. This was a major advantage for that design until the development of the concept of a hinged joint structure that integrated adjustability, winch shaft, and rotor hub all within the fuselage joint. Now both structures have their main components in the directions of the loads which create
efficient load paths. The pole structure was easier to integrate the adjustability into the design with the development of the hinge joint design. The hinge joint design places the winch spool on the end of the fuselage thus moving the belt forces in line with structural members of the pole design. With this change it allowed the pole design to shed some weight and weigh 0.5 pounds less than the comparable tear drop design as seen in Table 1: Pole and Teardrop concept surface area’s and weights.

The addition of the hinge in the fuselage design allowed us to integrate three features into one component. The hinge is the joint between the fuselage and the rotor hub. This part adds adjustability as the hub can be unlocked allowing the rider to come to equilibrium with his center of gravity directly below the rotors. Inserting bolts through the adjustment holes locks the rider into this position. Bushings and bushing nuts provide the adjustment point for the fuselage. These allow the spool shaft and bearings to be located as well as the hinge to be able to pivot. Putting the spool in the center of the rotor hub eliminates a bend in the spectra fiber and removes another component that could introduce losses to the drive train. This location of the drivetrain components also eliminates some bending forces seen in some of the fuselage structure. These three functions integrated together saves weight, reduces part count, and further optimizes the fuselage.

<table>
<thead>
<tr>
<th>Concept</th>
<th>Surface Area (in$^2$)</th>
<th>Weight (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pole</td>
<td>715</td>
<td>1.8</td>
</tr>
<tr>
<td>Teardrop</td>
<td>1150</td>
<td>2.3</td>
</tr>
</tbody>
</table>

Table 1: Pole and Teardrop concept surface area’s and weights

Figure 15: Final fuselage geometry
Studying the pole design showed us that there are three locations that experience the greatest load. These were identified through static analysis of the structure seen in Figure 18: Pole design static analysis. We analyzed those locations in detail and designed our layup scheme around those locations. These locations are shown in Figure 19: Bottom bracket section analysis, Figure 20: Seat joint section analysis, and Figure 21: Pole section analysis. After the detailed analysis the critical location is at the location where the pole connects with the truss structure. This location has the greatest moment due to the distance away that the rider weight is applied. This location requires 5 plies of plain weave carbon fiber to handle the loads in a layup pattern of [0, 45, 0, 0].
Figure 18: Pole design static analysis

Figure 19: Bottom bracket section analysis
Figure 20: Seat joint section analysis

Figure 21: Pole section analysis

Table 2: Fuselage parts and weight

<table>
<thead>
<tr>
<th>Part</th>
<th>Weight (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuselage</td>
<td>1.8</td>
</tr>
<tr>
<td>Seat</td>
<td>1.5</td>
</tr>
<tr>
<td>Hinge</td>
<td>1.4</td>
</tr>
<tr>
<td>Total</td>
<td>4.7</td>
</tr>
</tbody>
</table>
This final pole design matches up well with our specifications that have been driving our design. We can see from the total weight of the fuselage (Table 2: Fuselage parts and weight) that we hit our weight goal with a ~30% cushion for weight drift due to glue and parts coming out heavier than anticipated. The hinge joint allows us to easily adjust where the rider center of gravity is without any guessing. This is critical in order to not induce any effects from the rider into the rotor. Our final rider BCA for our ideal male rider is 135 degrees which is within our target BCA to allow the rider to be most efficient. Hitting all of these goals ensure that this DaVinci iteration will make forward progress toward winning the Sikorsky prize.

**FEA optimization of the a composite frame**

The model was developed as a solid and then converted to a shell. The initial layup was determined using a CST code applying classic laminate theory. The material orientation was determined based on the manufacturing scheme layup scheme. The figure shows the 1-direction of the fibers. As shown, the fiber 1-directions will cross at the joints. Many partitions were created to separate the various regions and accommodate the appropriate fiber direction. The step was a static load case based on the worst possible load case when the rider is mashing hard on the pedal with 200 lb of force. Many of the loading cases on the frame would result in torque on a shell edge loads and therefore the bottom bracket and the frame are two separate parts and then tied.

![Figure 22. Material Orientation of the fuselage Frame](image)
The frame was meshed with quadrilateral shell elements. The criteria for quality mesh element was inside angle greater than 35 degrees and no angle larger than 135 degrees. The aspect ratio for the elements should be less than five. The quantity of element with poor mesh characteristic was less than a tenth of a percent. The graphs on the next page show the mesh convergence based on deflection at two different points based on degrees of freedom. As seen by the graph below, the model converges around 200000 degrees of freedom. This standard will be used for the remainder of the analysis.

Figure 23. Mesh convergence on the deflection of node 59 on the seat of the frame

Figure 24. Mesh Convergence of node 131 near the bottom fillet of the frame
After successfully determining the appropriate mesh, analysis was done to optimize the composite structure. The initial layup was 5 layers of cloth covering the frame which resulted in 1.6 inches of deflection by the seat. Thus more layers of cloth were added to the top by the hinge and down by the seat. In the end the optimum scheme has 5 layers of cloth on the bottom bar, 7 on the curve, 11 on the angle and 13 down by the bottom fillet, and 9 on the seat. On the worst case loading, the deflection of the seat is 0.7 inches.

The reason for the discontinuous stress concentration is due to the varying number of cloth layers. The highest stress concentrations occur because of bending on the seat. The overall weight with of the cloth layup was 2.4 lbs for the fuselage. Using a Uni layup, the design will save about a half on a pound over the cloth.

Figure 25. Stress distribution on the final layup
Conclusion

We feel this design is a large step in the right direction. The weight reductions and part integrations make this a very integrated fuselage with no extra parts. Having an optimized fuselage to fit the needs of the drivetrain and rider is critical to the success of the DaVinci IV and we feel that this fuselage fulfills this criteria.

Drivetrain

The main goal of the drivetrain is to transmit power from the rider to the propeller. Designing the next DaVinci helicopter, the Aircraft Construction Club decided to pursue iterating the existing tip-propeller design, rather than developing a new configuration. As a result the rotor blades will each be 50 ft long with propellers mounted on the tips. Since the design is torque-less, the drive system should not place a torque on the rotor shaft which is not counteracted.

Drivetrain Requirements

The specifications presented in Table 3 list the club’s design requirements for the power transmission. The result should propel the team forward towards developing feasible concepts.
Table 3. Drivetrain Design Requirements for the DaVinci IV

<table>
<thead>
<tr>
<th>Drivetrain Design Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Power Output</strong></td>
</tr>
<tr>
<td>400 Watts to the rotor tip-propellers</td>
</tr>
<tr>
<td>Support 1000 Watts power input</td>
</tr>
<tr>
<td>90% efficiency</td>
</tr>
<tr>
<td>Rotates tip propellers at 120 RPM</td>
</tr>
<tr>
<td>Smooth out pedaling motion</td>
</tr>
<tr>
<td><strong>Gearing</strong></td>
</tr>
<tr>
<td>Rider input between 100 and 120rpm</td>
</tr>
<tr>
<td>Allow for interchangeability of gearing</td>
</tr>
<tr>
<td><strong>Weight</strong></td>
</tr>
<tr>
<td>Maximum weight goal 8.0 lbs</td>
</tr>
<tr>
<td><strong>Other</strong></td>
</tr>
<tr>
<td>No energy storage per Sikorsky Prize rules</td>
</tr>
</tbody>
</table>

The previous designs accomplished this feat by winding spectra around the propeller shaft and then feeding the thread below the rotor blade to a drive shaft. This design is incredible lightweight. The length of spectra needed to power the helicopter for 2 minutes is approximately 180 ft. The weight of 180 ft of spectra is merely 10 ounces. Thus, continuous drive systems were discussed but not included in the report due to simplicity and minimal weight of the spectra system. The follow figures diagram the path on the spectra. First the Spectra is wound around the propeller shaft (Figure 27. Spectra Fiber wound around a propeller shaft). The spectra then runs beneath the wing into the rotor shaft (Figure 28. Diagram spectra fiber feed into the rotor shaft). In Figure 29, the fiber runs down the propeller shaft to the spool and main driveshaft.
Figure 27. Spectra Fiber wound around a propeller shaft

Figure 28. Diagram spectra fiber feed into the rotor shaft
The DaVinci II wound the spectra from the propeller straight to the bottom bracket, eliminating the need for a drive shaft. However, the designers of the DaVinci II encountered two problems:

1. The rider complained about lack of inertia in the system
2. The spectra continued to crush the spool

Therefore, in request of the club, this new design includes a non-continuous spectra system (propellers to the drive shaft) and a continuous drive system (Cranks to the drive shaft). Due the limited power output of a human, the DaVinci series aircrafts will trade weight and power.

After the findings from the DaVinci II, we concluded the new design should include a continuous drive system and thus we investigated the following four transmissions: Chains Drives, Belt Drives, Linear Oscillating and Electric Generator.

**Chain Drives**

Chain drives operate using a steel roller chain together with a chain wheel and a rear wheel cog. Even though chain drive efficiency is critical to cycling, little work has been published. Through testing, chain transmissions are known to yield efficiencies of 98 percent and higher. James Spicer reported in “On the Efficiency of Bicycle Chain Drives” four significant findings (Wilson, Bicycle Science, 2004):
1. Transmission efficiency decreases as the size of the rear sprocket is reduced. Specifically the efficiency will be reduced on cogs with less than 21 teeth.
2. Efficiency diminishes as the amount of torque transferred (or chain tension) is decreased.
3. The maximum efficiency attained is at relatively high power and low pedal rpm and in the lowest gear at just over 98%.
4. The additional losses due to chain offset are negligible.
5. The type of lubrication has little effect on efficiency.

Spicer tested the efficiency as it relates to chain tension and found the maximum efficiency occurred at 305N (98.6%) and the lowest at 76.2 N(80.9%). The losses in chain drives presumably occur through friction of the bearings, and also through slip loss. The chains themselves see losses primarily through inner link bushing and chain pin contact and between sprocket tooth and link roller. The major modifications to the DaVinci III would be lighter components due to improvements to modern cycling components.

**Belt Drives**

Roller chains while very efficient and adjustable, weigh a substantial amount in order to retain the strength and stiffness. Synchronous belts otherwise known as toothed belts were designed in the 1940’s to address many of the issues in chain drives such as noise, elongation and lubrication. However, the early technology was unsuitable for low speed high torque applications. HTD (High Torque Drives) belts brought the next generation of synchronous belt drives but however became impractical due to the large width of the belts. In the last two decades Gates designed the Gates Poly Chain GT belt which combines a polyurethane tooth structure and a carbon fiber tensile member. This carbon cord has higher fatigue resistance than steel, glass or aramid fibers and substantially better strength to weight ratios than roller chains. Reported weight values for the belt drives are 80 grams compared to the latest dura ace chain weigh of 252 grams. Gates reports efficiency values of 98% on toothed drives which are equivalent or slightly greater than reported efficiency values of roller chain drives. Gates Poly Chain GT belt have already been designed compatible to bicycle drive train systems. Various sizes of sizebelts and other carbon bicycle drive information can be found at [www.carbondrivesystems.com](http://www.carbondrivesystems.com).
Integrating the Gates Carbon drive into the new DaVinci design would only require swapping parts. Gates has parts compatible with bike parts on their website [www.carbondrivesystems.com](http://www.carbondrivesystems.com). Carbon belts are 98% efficient while comparable chain drives are around 94% efficiency if they are well maintained. Carbon belts also require no lubrication and are about a quarter of the weight of chain drives (see Table 4). Also from the Gates website a variety of belt sizes are offered which would be compatible with our system. Due to the obvious advantages of belt drive systems against chain drives, belts seem like an obvious upgrade to the current DaVinci system.

**Table 4: Table of efficiency and weight values for the two best competing continuous drive components**

<table>
<thead>
<tr>
<th>Components</th>
<th>Efficiency</th>
<th>Weight (grams)</th>
<th>Weight with Components (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bicycle Chain</td>
<td>96%</td>
<td>366</td>
<td>2.4</td>
</tr>
<tr>
<td>Gates Carbon Drive Belt</td>
<td>98%</td>
<td>116</td>
<td>1.7</td>
</tr>
</tbody>
</table>

**Linear Oscillating Transmission**

Another form of transmission is a linear oscillating transmission. This transmission allows the rider to pedal in a natural up and down motion. To transmit the torque smoothly the design requires a one-way clutch without backlash. The design is effective at only slow speeds because energy is lost decelerating the legs finishing a pedal stroke. Incorporating a fuse used in spring wound clocks has been suggested. A diagram of the design is shown in Figure 31: Linear Oscillating drive system design.
Linear Oscillating drive is much like the chain drive except it has the ability to transfer axis of rotation much easier than a chain drive. The linear oscillating drive converts up and down motion of the rider rather than rotational. However this design has many disadvantages. First the design has not been fully tested and efficiency values are unknown. Belt drives already a very efficient alternative at 98%. Secondly, the linear pedal strokes are unfamiliar to professional riders and therefore the same wattage values will be incapable of achieving. Thirdly, Gearing the drive system is less compatible then belt or chain drives and more difficult to adjust.

Electrical

This design appeared very desirable because using circuitry we could spin the propeller at a constant rpm and would open up the door to solving stability and control issues. Also the electrical system would eliminate the need for spectra thread driving the propellers. This design the rider spins a generator which then converts the energy to electricity. Implementing a circuit board between the between the generator and the motors at the tips will allow a constant rotational torque to the propellers. Thus, this system already has no need for a flywheel. A slip ring will allow us to transfer the rotational energy across the rotational shaft.

![Figure 31: Linear Oscillating drive system design](image1)

![Figure 32: Conceptual drawing of the electrical drive system](image2)
When investigating this potential design we learned technology has not fully caught up with this design. As technology develops this design could become the optimum design for this application. Current lightweight generator technology is about a maximum of 70% efficient according to local experts. Electric motors have become increasingly efficient over the decade and are about 95% efficient. Other losses will also accrue throughout the circuitry. An estimated efficiency for the system is around 65%. This efficiency fails to satisfy the current design specifications.

**Research Conclusions**

After the initial concept development belt drives, chain drives and electrical will be considered for final design but with further analysis. The electrical design would be a great design for future control systems of the helicopter however the efficiency values of the electrical are low. Efficiency values of the mechanical systems will indicate whether or not the electrical design is worth pursuing at the current time. Both designs have relatively similar efficiency characteristics but belts are over half the weight of chains. The Linear- Oscillating design will not be investigated further because there is no proven efficiency values, poor gear interchangeability, and more difficult manufacturability.

**Concept Analysis**

The reason behind the analysis is to determine the best design for this project. Effectively the best project is the one with the greatest efficiency for the least amount of weight. Initially, the results will analyze each of the possible systems excluding the electrical system. Given the low efficiency values of the electrical system, if the comparable mechanical systems efficiencies are greater than 70%, the electrical system need not be explored further. After calculating the efficiency of the drive components, we will discuss the gearing and implementation of system, specifically shafts and bearing.

**Efficiency**

The system loses efficiency due to friction (Figure 33: Diagram of the frictional forces in the spectra thread). The losses occur in two different locations frictional losses on the eyelets and the 90 degree turn where the spectra line twist and then are spooled up.

![Figure 33: Diagram of the frictional forces in the spectra thread](image)
The frictional losses due to the eyelet occur to the deflection of the wing structures. Shown below is a free body diagram of the eyelet. After finding the normal force on the eyelet, we can then find the frictional force. Now we will select an appropriate beam theory model to find the slope of the wing at locations of the eyelets.

Summing the forces in the x and the y directions assuming the part is in equilibrium.

\[
\sum F_x = 0 = -T_1 \cos \theta + F_f \cos \theta - N \sin \theta + T_n
\]  
(1)

\[
\sum F_y = 0 = T_1 \sin \theta - F_f \sin \theta - N \cos \theta
\]  
(2)

Substituting \(F_f\) for \(\mu N\) then \(N\) is found

\[
N = \frac{T_1 \sin \theta}{\cos \theta + \mu \sin \theta}
\]  
(3)

Then the frictional force is

\[
F_f = \mu N
\]  
(4)

Using beam theory, treating the wings as cantilevers with have the half lift force at the centroid of the lift distribution. According to our rotor blade designer Greg Gradwell the lift distribution can be well approximated with a triangular lift distribution. Assuming the beam sees 5 feet of deflection.
\[ \delta_{\text{max}} = \frac{Fa}{6EI} (a - 3l) \]  

(5)

Where \( a \) is the distance from the root of the blade to the centroid, \( l \) is the length of the rotor blade, \( F \) is half the lift, \( E \) is the elastic modulus, and \( I \) is the moment of inertia. Solving for the effective EI

\[ EI = \frac{Fa}{6\delta_{\text{max}}} (a - 3l) \]  

(6)

Thus the effective EI is 3.66 Mpsi

The equations for the deflection of the beams are

\[ \delta = \frac{Fx^2}{6EI} (x - 3a) \quad \text{for } 0 < x < a \]  

(7a)

\[ \delta = \frac{Fa^2}{6EI} (a - 3x) \quad \text{for } a < x < l \]  

(7b)

Differentiating the deflections with respect to \( x \), we found (Wilson, Bicycle Science, 2004) the slope of the wing at any given point \( x \).

\[ \frac{d\delta}{dx} = \frac{Fx}{3EI} (x - 3a) \quad \text{for } 0 < x < a \]  

(8a)

\[ \frac{d\delta}{dx} = -\frac{Fa^2}{2EI} \quad \text{for } a < x < l \]  

(8b)

Now using the tension force \( T_1 \) coming into the eyelet the normal force can be found using statics on Figure 34.
Table 5. Table of the curvature and forces at each of the wing eyelet locations

<table>
<thead>
<tr>
<th>Rotor Location (ft)</th>
<th>Slope</th>
<th>Change in Slope</th>
<th>Normal Force (lbs)</th>
<th>Frictional Force (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>-0.05111</td>
<td>2.928224</td>
<td>9.618465</td>
<td>0.577108</td>
</tr>
<tr>
<td>20</td>
<td>-0.09932</td>
<td>2.762475</td>
<td>9.048607</td>
<td>0.542916</td>
</tr>
<tr>
<td>30</td>
<td>-0.12439</td>
<td>1.436487</td>
<td>4.702079</td>
<td>0.282125</td>
</tr>
<tr>
<td>40</td>
<td>-0.12857</td>
<td>0.239415</td>
<td>0.784306</td>
<td>0.047058</td>
</tr>
<tr>
<td>50</td>
<td>-0.12857</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Sum</td>
<td></td>
<td></td>
<td></td>
<td>1.449207</td>
</tr>
</tbody>
</table>

As can be seen in the table the frictional losses due to the eyelet are relatively small. The major of the losses are caused by the 90 degree turn. We designed the losses to enter the fuselage through slots in the rotor hub shaft and turning downward toward the take-up spool. We investigated the losses for three different designs discussed below:

1. Teflon curved surface
2. Bushing
3. Ball Bearing

**Teflon Curved Surfaces**

Frictional losses due to the Teflon surface are similar to a belt slipping on. Drawing a free body diagram of the belt we see that the belt friction opposes the direction of motion and thus T2 will be greater than T1. (The SKF Group, 2010)
Figure 36. Free body diagram approximating the friction is a slipping belt (left) Free body diagram of the belt length \(ds\) (right)

Solving for \(T_2\) knowing the initial belt tension \(T_1\) and the coefficient of friction \(\mu\) and the \(\beta\) the angle of belt to surface contact, measured in radians. (Hibbeler, 2007)

\[
T_2 = T_1 e^{\mu\beta}
\]

In our case the belt angle of contact is 90 degrees and Teflon coefficient of friction is 0.06 and initial tension of 190 lb results in a final tension value of 208 lbs. Thus the efficiency of a curved Teflon surface even with minimal friction results in an efficiency of 90%. This design would benefit due to the limited weight required to needed to build the system but the efficiency is rather low.

**Frictional losses due to a bushing**

Figure 37. The figure on the left show the forces and moments required to maintain constant rotation. The figure on the right shows the frictional circle. (Hibbeler, 2007)

A bushing experiences frictional losses at a point of contact where the shaft slips against location of the reaction force. By summing the moments around the axis of rotation we find the equation shown below where \(\varphi_k\) is the angle between the normal force and the reaction force.
Thus simplifying the equation

\[ M - (R \sin \theta_k) = 0 \]  \hspace{1cm} (10)

If the bushing is partially lubricated then the angle is considered a small angle and the moment can be simplified to

\[ M = R \sin \theta_k \]  \hspace{1cm} (11)

\[ M \approx Rr\mu_k \]  \hspace{1cm} (12)

For our design the roller is limited in size to 1.5 inches. Also the roller shaft has to be relatively large to support the large loads in the spectra. Sizing an appropriate roller shaft is discussed in the later section. The results are a 12mm pin giving efficiency values of 81%.

Ball Bearings Losses

SKF website states frictional moment \( M \) is found as on half the coefficient of friction times the load and \( d \) the bearing bore. The coefficient of friction for the ball bearings is 0.0015 and needle bearings is 0.0045.

\[ M = 0.5\mu P d \]  \hspace{1cm} (13)

The conditions on the site include the bearing load should be approximately 0.1 the load rating of the bearings (The SKF Group, 2010). (Nisbett, 2008)The design condition of 400 Watts out to the propellers yields a bearing load at about 142 lbs. If the desired condition for the formula is \( P \) approximately equals 0.1 times the dynamic bearing load rating then the \( C \) should be about 1420 lbs. Knowing the remaining information about the bearing, the frictional moment is a little under 0.05 lb-inches. Subtracting the value from the spooling torques, the efficiency of the bearing is 99%.

Table 6. Efficiency results from the 90 degree turn analysis

<table>
<thead>
<tr>
<th></th>
<th>Theoretical Efficiency</th>
<th>Input Required (Watts)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Teflon Surface</td>
<td>89%</td>
<td>446</td>
</tr>
<tr>
<td>Bushing</td>
<td>90%</td>
<td>445</td>
</tr>
<tr>
<td>all Bearings</td>
<td>98%</td>
<td>406</td>
</tr>
</tbody>
</table>
Table 6. Efficiency results from the 90 degree turn analysis gives the efficiency values of spectra loop. In actuality, the appropriate design will have the best power to weight ratio. Table 7 includes the system efficiencies and weight comparisons. Using cursory thrust calculations using the rotor disc area the power required is shown below:

\[ P_R = \frac{3W^{3/2}}{R\sqrt{32\pi \rho}} \]  

Now rewriting the equations to solve for a change in power based on a change in weight

\[ \Delta P_R = \frac{3}{R\sqrt{32\pi \rho}} \left( (W_0 + \Delta W)^{3/2} - (W_0)^{3/2} \right) \]  

\( W_0 = \) original weight  
\( \Delta W = \) change in weight  
\( \Delta P_R = \) change in power required  
\( \rho = \) density of air (0.002377 slugs/ft\(^3\) at Sea Level)  
\( R = \) rotor radius, ft

The estimated weight of the craft is around 200lbs. Adding 1 pound will require the rider to output an additional 2.61 Watts. So even though the bushing design has the least amount of weigh, the system losses 41 Watts compared to the belt. The design requires a rider to output 400 Watts out to the propellers. The results of this project will determine the whether the club needs an Olympic athlete or a local rider.

Table 7. Overall system efficiencies and weight for possible drive systems

<table>
<thead>
<tr>
<th>Drive System</th>
<th>Theoretical Efficiency</th>
<th>Input Required (Watts)</th>
<th>System Weight (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Belt Drive</td>
<td>95.6%</td>
<td>414</td>
<td>5.3</td>
</tr>
<tr>
<td>Chain Drive</td>
<td>93.6%</td>
<td>423</td>
<td>5.7</td>
</tr>
<tr>
<td>Belt Drive with Bushing</td>
<td>86%</td>
<td>465</td>
<td>5.0</td>
</tr>
</tbody>
</table>
Interfacing the bearing design

Several designs were considered when implementing the ball bearings into the system. After the spectra is feed through the rotor shaft, the bearing roll the spectra roll towards the rotor shaft. The most efficient way for packing the two bearing is a common shaft (refer to Figure 28). Since the bearings rotate in opposite directions, the shaft will remain stationary and the spectra slides on over the bearings. The pin was sized to minimize deflection to less than 0.0025 inches. Using a simply supported beam, as shown in the figure below, the maximum deflection is in the center of the beam. The formula for the maximum deflection is

![Figure 38. Simply supported beam modeling the shaft for the ball bearings to rotate around](image)

\[ \delta_{\text{max}} = \frac{Pl^3}{48EI} \] (16)

Given a solid shaft the and a length \( l \) of 2.12 inches, the elastic modulus \( E \) of 30 Mpsi the diameter of the shaft is 11mm. adding a small factor of safety we selected a 12 mm shaft and selected a pair a bearing which satisfy the basic load rating requirements.

Gearing

The requirements for the drivetrain gearing include gearing a rider from spinning at 100 rpm to a final propeller speed of 120 rpm. This gearing could theoretically occur in only the belt system, or the spectra system, or potentially both systems. Thus, we investigated the limiting factors in the gearing. The main two factors were the:

1. Strength of the spectra
2. Maximum diameter of the spool

The available space in the frame structure limited the driveshaft spool diameter to 1.5 inches. The spectra fiber is limit by the breaking strength. Spectra thread is used in various applications including fishing and kites. Propower fishing line offers a line with a breaking strength of 250 lbs and Honeywell offers a fiber with a breaking strength of 270 lbs.

With a breaking strength of 250 lb, a limit load on the spectra is a power output of 1000 W out to the propellers. the next step is to find the minimum diameter of the propeller spool.

\[
\begin{align*}
P & \text{ Max Power 500 W} \\
\omega & \text{ Prop Speed is 120 rpm} \\
F_{\text{max}} & \text{ limit Load 250 lb}
\end{align*}
\]
T- Torque on the propeller shaft

We have the equations:

\[ P = T \cdot \omega \]  \quad (17)

\[ F = T \cdot \frac{D_{\text{min}}}{2} \]  \quad (18)

Combining the two equations and solving for the minimum diameter of the spool

\[ D_{\text{min}} = \frac{P}{\omega \cdot F_{\text{max}}} \]  \quad (19)

We found the minimum diameter to be 2.81 in.

After design iterations to the frame structure and to the gearing system we found the take-up spool needed maximum diameter spool we allowed was 1.5 inches in diameter due to fuselage size requirements. Applying the same equations and finding the rotational speed of the shaft.

\[ \omega = \frac{P}{(D_{\text{max}} \cdot F_{\text{max}})} \]  \quad (20)

The rotational speed is 225 rpm and the minimum gearing 2.25

There are many sprocket sizes to accomplish this result.

<table>
<thead>
<tr>
<th>Table 8. Gates belt sprocket sizes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gates Front Sprocket</td>
</tr>
<tr>
<td>----------------------</td>
</tr>
<tr>
<td>39</td>
</tr>
<tr>
<td>46</td>
</tr>
<tr>
<td>50</td>
</tr>
<tr>
<td>50</td>
</tr>
<tr>
<td>55</td>
</tr>
<tr>
<td>60</td>
</tr>
</tbody>
</table>

One of the design requirements is an input of 100 rpm and an output of 120 rpm. Given a gearing of 2.5 in the belt drive then the spectra gearing system needs to have 0.48 gearing between diameters of the take-up spool and propeller spool. Since the take-up spool is 1.5 in diameter, the
propeller spool needs to be 3.125 in diameter. Below in Table 9 are various alternatives for gearing using the Gates sprocket sizes.

<table>
<thead>
<tr>
<th>Front Sprocket</th>
<th>Rear Sprocket</th>
<th>Belt Gearing</th>
<th>Spectra Gearing</th>
<th>Propeller Spool Diameter (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>46</td>
<td>22</td>
<td>2.09</td>
<td>0.57</td>
<td>2.61</td>
</tr>
<tr>
<td>50</td>
<td>22</td>
<td>2.27</td>
<td>0.53</td>
<td>2.84</td>
</tr>
<tr>
<td>55</td>
<td>22</td>
<td>2.5</td>
<td>0.48</td>
<td>3.13</td>
</tr>
<tr>
<td>60</td>
<td>22</td>
<td>2.72</td>
<td>0.44</td>
<td>3.41</td>
</tr>
</tbody>
</table>

Looking at the table, the smallest propeller diameter option while still satisfying the design requirements is a 50 tooth front sprocket and rear sprocket 22 tooth.

**Drive Shaft sizing**

Now the two systems connect through a common drive shaft. As the rider pedals and powers the rear sprocket, the driveshaft rotates spooling the spectra. Below are free body diagrams of the loading of the shafts. Table 10 records the values of the bending moments based on the location x on the shaft.
Figure 41. Bending moments on the shaft based on location on the shaft

Table 10. Bending Moments on the drive shaft linking the belt driven system with the spectra system

<table>
<thead>
<tr>
<th>Shaft Location (in)</th>
<th>xy plane (lb-in)</th>
<th>zx plane (lb-in)</th>
<th>Mtotal (lb-in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>1.74</td>
<td>-276.66</td>
<td>479.1892</td>
<td>553.32</td>
</tr>
<tr>
<td>2.74</td>
<td>-405.9873837</td>
<td>252.6786</td>
<td>478.1968</td>
</tr>
<tr>
<td>3.74</td>
<td>-36</td>
<td>0</td>
<td>36</td>
</tr>
<tr>
<td>15.74</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Selecting Chromoly Steel 4130 for the material of the drive shaft, the yield stress is 75 ksi. Knowing the maximum bending moment of shaft of 553 lb-inches we found the normal stress. The minimum wall thickness of the tube is 0.028 in found on McMaster Carr’s website. After calculating the moment of inertia the, the normal stress is 27.3 ksi. Applying MSS failure criteria the safety factor is 2.7. This safety factor is more than sufficient however we will need to shave some metal off to allow room for bearings and attaching the sprocket. Thus we selected a 0.049 wall thickness.
Needle bearings are necessary for the shaft. Applying the equations from the 90 degree turn analysis for both bushings and needle bearing the power lost for bushing is significantly greater than with needle bearings. Even with properly lubricated bushing the loads are too severe for the bushings to be effective. Shown below is a table of the frictional moments. From the data we can conclude the need for needle bearings.

<table>
<thead>
<tr>
<th></th>
<th>Frictional Moment (lb-in)</th>
<th>System efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bushings</td>
<td>25.1</td>
<td>85.1%</td>
</tr>
<tr>
<td>Needle Bearings</td>
<td>0.64</td>
<td>97.5%</td>
</tr>
</tbody>
</table>

### Spool
Comparing the tension forces to an external pressure on the spool we found the stress values for a single wrap of spectra. Given the spool is 1.75 in long and the spectra is 0.035 in diameter the spectra will wrap over itself multiple times. For a flight time of two minutes the spectra will wrap around the spool about 10 times. Assuming the pressure force scales with the number of wraps the maximum resulting pressure is 8750 psi. Using thick walled cylinder equations the hoop stress is 16.7 ksi. The longitudinal stresses found are 12.3 ksi. Using Von Mises failure criteria the safety factor of the part is 4 for Aluminum with yield strength of 18 ksi.

### Conclusion
After analysis of the conceptual designs on the drivetrain the appropriate design is a continuous drive Gates carbon belt drive with and a non-continuous belt spectra system. Below are the list of components needed for the drivetrain.

- Carbon Crankset
- Bottom Bracket
- Gates Carbon Drive belt
- Front 50 tooth sprocket
- Rear 22 tooth sprocket
- 1 in diameter Chromoly 4130 Steel shaft
- 1.5 diameter PEEK shaft
- 150 Yards of ProPower Spectra
- 2 SKF K 24x28x17 Needle Bearings (0.9844)
- 2 SKF 6201 deep groove ball bearings (0.4744 in)
- 10 eyelets
**Rotor Hub**

**Introduction**

The rotor hub is the junction between the rotor and the fuselage. This structure connects the lifting surfaces to the rest of the vehicle, allows the rotor to rotate independently of the fuselage and provides the conduit for power transmission to the tip-mounted propellers. Figure 42 shows the rotor system design for the DaVinci IV helicopter with a rigid rotor shaft which would allow the fuselage to exert a restoring moment on the rotor and thus positively contribute to craft stability. Figure 43 shows the rotor hub from the DaVinci III which was designed as a “teetering” rotor system but actually behaved like a rigid rotor because the bottom of the fuselage was tethered to the rotor in four places. We chose the rigid rotor system configuration because this provides the only control attachment to the rotor system unlike a conventional helicopter which controls the rotor pitch with cyclic control inputs. Figure 44 shows a “rigid” rotor head from a MBB BO-105 helicopter where you can see the pitch links connected to each rotor blade for comparison. As mentioned previously we specifically defined this project to exclude a control system.

![Diagram of rotor hub system](image)

*Figure 42: The final design of the rotor hub systems for the DaVinci IV. The hub is comprised of Graphlite pultruded carbon fiber tubes configured into a truss to carry loads out to the rotor. The eight rings at the corners of the truss are a short fiber carbon composite as are the central rings that join the truss. The rotor shaft is a carbon fiber tub reinforced where the spectra thread enters. The rotor bearing assembly is bonded into the bottom of the shaft housing two SKF51106 thrust ball bearings.*
Figure 43: DaVinci III Rotor Hub. This figure shows that the fuselage is tied to the rotor which allows the fuselage to swing beneath the rotor. For reference some fishing swivels have been included in the bottom right hand corner of the figure.

Figure 44: Rigid rotor hub from a Eurocopter BO-105. This illustrates a rotor hub bearing attachment where the fuselage can exert a force on the rotor disk and change the attitude of the rotating disk. The rigid rotor system of this particular helicopter makes it suitable for aerobatic maneuvers. Photo courtesy of the Wikimedia commons.
**Loads**

The rotor hub and bearing analysis proceeded by treating the rotor as a black box that provides sufficient power to lift the fuselage and ignoring the specific aerodynamic methods the rotor uses to generate those forces. Figure 45 shows a free body diagram that considers the lift forces and translational acceleration provided by the rotor. Since we are interested only in the forces that act on the rotor hub, we replaced the free body diagram of Figure 45 with the free body diagram in Figure 46 and assumed that the fuselage is a simple pendulum that is acted on by a force-moment couple. Using this model we looked at the video of the DaVinci III flight and measured the angular displacement at maximum instability as well as the time required to reach that instability. Assuming a constant acceleration and using eqn(20) we calculated $\alpha=0.061 \text{ rad/s}^2$.

\[
\theta = \theta_0 + \omega_0 t + \frac{1}{2} \alpha t^2
\]  
(20)

This last piece of information allowed us to calculate the forces and moment at point O in the figure where $R_{on}= 158.8\text{lbf}$, $R_{ot}= 20.4\text{lbf}$, and $M_{wing}=61.4\text{ft-lbf}$ to be used in sizing rotor hub members.

![Figure 45: Free body diagram and mass acceleration diagram of the fuselage and rotor at maximum instability.](image-url)
Figure 46: Free body diagram and mass acceleration diagram of the fuselage using the model of a pendulum and applied forces that are sufficient to move the fuselage.

Hub Concepts

Figure 47 shows the initial three rotor hub concept iterations with three different rotor bearing configurations. The first two concepts on the left of the figure assume a core made from Rohacel 31G foam and wrapped in carbon fiber. These two designs however were removed from consideration primarily based on weight. A similar composite only shell was compared to the truss section that we selected and the shell alone weighed almost 3lbf, not including any shear webbing, pockets for the wing spars, or the rotor shaft while the entire truss-hub we selected is calculated to weigh only 2.8lb including bearings. Additional problems arose from load path design. The 14” square box section described by the rotor spars covers a large area and the rotor loads must be transmitted between these points through plane panels in the form of a shear loading which is not the optimum way to use a carbon fiber composite. The third concept from Figure 47 led to the final truss-style rotor hub design in the process of figuring out the most efficient way to transfer loads to the individual composite panels. Recalling the truss based analysis from statics we determined that the most efficient method to transfer the load would be through a straight line from point to point which defines a truss structure. Figure 48 shows the resulting rotor design complete with rotor spars and spar connections which is the synthesis of this design process.
Figure 47: This figure shows the three main rotor hub concepts that led to the final version. Each of these concepts was eliminated due to excessive weight.

Figure 48: The final DaVinci IV rotor hub system showing the connection between the rotor spars and the rotor hub system.
Hub Analysis

Figure 49 shows the truss structure without the spar-tubes or rotor-shaft. This structure is comprised of a series of 0.25” diameter Graphlite brand pultruded carbon fiber rods which have a tensile strength of $S_{tu} = 320$ksi, the material specification sheet appears appendix D. These extremely stiff rods are also light due to the low volume they occupy compared to a box shaped structure. Also these tubes are readily purchased for easy fabrication. The rods attach to the rotor spars by way of eight short fiber composite rings that are molded to accept the rods. Because the truss must be capable of withstanding loads from several directions, the truss is statically indeterminate and so we made several assumptions for a simplified analysis. First we looked diagonally at the truss system and assumed it was a plane truss. Next we decomposed that truss into two smaller symmetrical trusses as in Figure 50. We assumed this simply supported and simply loaded truss accounts for $\frac{1}{4}$ of the total truss structure and therefore allocated $\frac{1}{4}$ of the total hub loading to this truss and assigned those loads to the free body diagram in Figure 51. Using a simple plane truss analysis of the truss as presented in Figure 51 gives the most highly loaded member as $F_{BC} = 39$lb which translates into $\sigma_{ave} = 1.4$ksi assuming 55% fiber volume in the Graphlite rods. Since the compressive force is much less than the tensile strength of the rod, the probable failure mode is column buckling. We will address this by wrapping the Graphlite rods where they cross with fiber to reduce the effective column length. In any case this analysis makes so many assumptions that the best we can say is that the design seems reasonable pending testing.

Figure 49: The rotor hub truss system without the rotor shaft.
Figure 50: This figure illustrates the two part decomposition of the rotor hub to simplify the statically indeterminate structure for analysis.

Figure 51: Free body diagram used to analyze the loads in the rotor hub.
Bearing Analysis

Figure 52 shows the free body diagram of the interaction between the rotor and the fuselage during a stable hover where the only interaction between the rotor and the fuselage is the frictional moment from the rotor hub bearing. This is a critical component of the design because like DaVinci III we plan to rely on the inertia of the pilot and fuselage as well as the precessional moment of the flywheel to prevent the pilot from rotating with the rotor. From Newton’s second law which appears in angular form as eqn(21) we know that the pilot must accelerate since there is in fact an applied moment.

\[ M = I \alpha \] (21)

However given the very short anticipated flight duration from this aircraft we expect that the rotation should be minimal. This conclusion was validated with DaVinci III. Theory from SKF for minimally loaded bearings gives the following equation for bearing frictional moment where \( M \) is the frictional moment, \( \mu \) is the constant co-efficient of friction, \( p \) is the dynamic bearing load, and \( d \) is the bearing race diameter:

\[ M = 0.5\mu pd \] (22)

This equation shows that minimizing the lever arm decreases the frictional moment on the fuselage. Summing moments in Figure 52 and equating the terms from the free body diagram and the mass acceleration diagram gives the following relationship:

\[ \alpha = \frac{0.5\mu pd}{I} \] (23)

As an example consider the swivel type of rotor hub attachment using Eqn. 23 with \( d=0.689 \text{in} \) diameter thrust ball bearing, \( \mu=0.0013 \), and \( p=180\text{lbf} \), where the frictional moment \( M \) is predicted as 0.081in-lbf which shows that using momentum alone to counteract rotation is not an unreasonable strategy although the pilot must accelerate if only at a slow rate. A rigid rotor attachment would require a larger diameter bearing to handle the resulting torque load as illustrated in Figure 46 which would produce a larger frictional moment than the swivel type bearing. As an example consider again Eqn. 23 where \( d=1.22 \text{in} \), \( \mu=0.0013 \) and \( p=491\text{lbf} \) the rigid rotor would impart a theoretical maximum moment of 0.532lbf-in to the fuselage. While the magnitude of the forces required to counteract these moments are small, this vehicle is power limited and so parasitic losses also had to be addressed. Consider the following equation where \( P \) is the power dissipated, \( M \) is the magnitude of the frictional moment, and \( \omega \) is the angular velocity:

\[ P = M \omega \] (24)

Using the results obtained in the previous two examples yields a power loss of 0.01 Watt and 0.063 Watts respectively. Based on these calculations we confirm that even with increased frictional
moment the resulting from the rigid rotor attachment is small enough to yield a minimal rotational acceleration.

We performed some initial calculations about the power required to mount a small electric motor on the fuselage that acted on a ring gear attached to the rotor similar to the excavator turntable bearing in Figure 53. If this motor were geared correctly it could be controlled to speed up and slow down such that it would rotate the fuselage in the opposite direction as the rotor thereby maintaining the orientation of the pilot. An appropriate controller would be a heading-hold gyro from a radio-controlled helicopter. We calculated that such a system would only require 1 Watt to orient the fuselage especially compared to the substantial power loss involved in operating a tail rotor as on a conventional helicopter. We decided that this concept could be added and investigated for future senior projects because of the significant amount of analysis required to design a control system and a sufficient power source for such a system.

Figure 52: Free body diagram and mass acceleration diagram of the fuselage and rotor during stable hover.
Figure 53: Excavator turntable bearing used to illustrate how a heading hold orientation system might be adapted to future DaVinci iterations. http://img.diytrade.com/cdimg/726150/5858397/0/1236611648/Turntable-bearing.jpg

Figure 54 shows a sectional view of the rotor bearing where the fuselage is supported by a four point contact ball bearing from Kaydon Bearing. Figure 56 shows an exploded assembly view of the same bearing system. The shaft is thin carbon fiber and the coupling is machined from anodized 2024 aluminum. This will provide sufficient stiffness for the application with a minimum weight. The design originally called for a magnesium coupling but magnesium and carbon fiber have a strong galvanic reaction which can prevent epoxy from curing or contribute to future degradation in such a joint. Figure 55 shows the final rotor bearing design using a lightweight KAA17XL0 four point contact bearing. This design saves almost a third of a pound from the other designs as is Table 12: Rotor bearing weight comparison and is easier to build. We initially thought that the design required two thrust bearings as detailed in Figure 56 because there is sufficient tension in the drive system during flight that the rotor shaft will be in compression. We also assumed two thrust bearings were required to handle the moment induced on the shaft at maximum aircraft instability as in Figure 45. Initial calculations showed that these forces exceed the load capacity and invalidate the use of eqn(22) for smaller non-thrust bearings. Similarly we investigated the bearings in Figure 57 and concluded that for a four-point contact bearing the radius required to handle the moment loading would cause too much frictional moment and at the same time weigh too much. However the initial calculations assumed that the center of mass of the fuselage would be three feet below the rotor bearing where as the final design locates the center of mass only 18in below the bearing which decreases the moment on the bearing by a factor of two. Also from looking at the footage from DaVinci III, once the pilot reaches instability the run would be over so the bearings do not need to operate optimally at this condition. The spherical bearing on the right hand side of Figure 57 was eliminated because we deemed a rigid rotor shaft was more important.
Figure 54: Section view of the DaVinci IV rotor hub bearing showing the four point contact bearing and the assembly to capture that bearing between the two rotor shafts.

Figure 55: Exploded view of the final selection rotor bearing using a Kaydon KAA17XLO. This design provided sufficient stiffness with the least weight.
Figure 56: SKF 51106 Hub Bearing Assembly. This assembly shows assumes a rigid attachment which can handle a very severe moment load on the shaft however this design was not selected due to manufacturing complexity and extra weight.

Figure 57: Two different rotor bearing concepts. The KB020XP0 assembly is a 4-point contact bearing that was deemed too heavy for this application. The SKF GZE008ES assembly assumes a spherical bearing which would make the rotor able to flap which is undesirable in this aircraft.
Table 12: Rotor bearing weight comparison.

<table>
<thead>
<tr>
<th></th>
<th>Truss Weight [lb]</th>
<th>Main Shaft Weight [lb]</th>
<th>Rotor Connection Weight [lb]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coupling Style</td>
<td>KAA17XL0</td>
<td>KB020XP0</td>
<td>SKF51106</td>
</tr>
<tr>
<td>Coupling Weight [lb]</td>
<td>0.15</td>
<td>0.34</td>
<td>0.18</td>
</tr>
<tr>
<td>Bearing Weight [lb]</td>
<td>0.05</td>
<td>0.19</td>
<td>0.30</td>
</tr>
<tr>
<td>Total System Weight [lb]</td>
<td>2.32</td>
<td>2.65</td>
<td>2.60</td>
</tr>
</tbody>
</table>

Rotor Shaft Analysis

In order to size the shaft we assumed that the shaft was a cantilever beam as in Figure 58 and used standard beam theory to size an aluminum shaft in order to find the shaft diameter with the best ratio of the moment of inertia versus weight. We made an Excel spreadsheet to iterate the different diameters and wall thicknesses and settled on 2.5” diameter. Next we used classical laminate theory (CLT) to predict the effective stiffness term using the Matlab code included in the appendix. The input for the CLT code is $N_x$ from eqn (25) which is the maximum load per unit thickness of the laminate. The code will output the $A*$ matrix term which is the compliance matrix, also known as the inverse of the stiffness matrix for the laminate. The result from the compliance matrix is used in eqn (26) to determine the effective beam stiffness and by extension the deflection of the cantilever beam.

$$N_x = \frac{M_y}{R^3} = \frac{M}{R^2}$$  \hspace{1cm} (25)

$$EI = \pi R^3 \frac{1}{A_{11}}$$ \hspace{1cm} (26)

We ran the CLT code several times with different layup schemes and wrote an Excel spreadsheet to compare deflections and selected the following layup [0,90,0] using ACG LTM45-1 epoxy pre-preg.
Figure 58: Free body diagram of the rotor hub modeled as a cantilever beam.

Rotor Connection Analysis

Figure 59 shows an exploded view of how we will attach the rotor hub to the rotor as well as the method of attaching the two rotor sections together. Per the figure, threaded sleeves will be bonded to the rotor spar ends. This will permit the use of a coupling nut to connect the two rotor halves and will operate in a manner identical to a turnbuckle. In a similar manner the spar rings of the rotor hub will slide over a tube bonded to each rotor spar. The spar ring will be un-threaded but will have a thin jam nut on either side to permit the hub to be locked in place relative to the rotor spars. While this design seems complex it meets two objectives for this project. First the rotor blades must be removable and thus the shaft and rotor blades must be separate pieces. Second this design prevents excessive concentrations of stress on the thin and lightweight parts. This distribution is accomplished by allowing the rotor hub to move far away from the rotor shaft centerline thereby reducing actual loading, the threaded tube adds additional material to the spar to increase the wall thickness and reduce localized fiber buckling and finally the jam nuts may be spaced to allow the rotor to stretch without unduly stressing the smaller rotor hub.
Next we picked the thread size required for the bonded tubes to handle the appropriate tension force in the rotor spars but still have a minimal cross-section for weight savings. We approached this task by comparing the tensile stress area required by three different materials: aluminum, magnesium and titanium. The required tensile stress area is given by eqn(27). We adapted from Shigley’s, the equation for tensile stress area of a hollow bolt listed in eqn(28) where \(d_p\)=pitch diameter, \(d_m\)=minor diameter, \(d_s\)=spar diameter. We then wrote an Excel spreadsheet and input different fine pitch standard thread forms and iterated to find the tensile stress area which had the best ratio of strength to weight ultimately settling on 1 1/8-28UN threads using magnesium.

\[
A_t = \frac{P}{\sigma} 
\]

(27)

\[
A_t = \frac{\pi}{4} \left[ \left( \frac{d_p^2 + d_m^2}{2} \right) - d_s^2 \right] 
\]

(28)

Finally we analyzed the glue joints used to bond the threaded barrels to the rotor spars as well as to join the Graphlite tubes to the spar rings. We checked the glue joints using eqn(9) which showed that assuming a bonded shear strength of 2.5ksi and a safety factor of 3.0 the coupling was required to be 0.82” long, where \(\sigma_{ave}\) is the average shear strength of the glue and \(P\) is the load applied to the joint.

\[
l = \frac{P}{\pi d_s \sigma_{ave}} 
\]

(8)

A similar analysis showed that the glue joints for the rotor hub Graphlite rods required only 0.060” of penetration.
Manufacturing

Fuselage

Introduction
The fuselage will be made from carbon fiber so that the ply count can be optimized to the desired structure. The fuselage will be made using a lost-wax composite molding technique. This will allow for the fuselage to be a hollow, one piece construction. There are significant weight savings due to eliminating the need for core material as well as a weight savings in adhesive required to bond the two halves together of a two part fuselage. A positive plug of the fuselage will be made from wax and the carbon fiber will be laid up directly on the wax. We had initially selected and ACG low temperature prepreg, however after reaching our go/no-go date and had not heard about getting the prepreg donated we changed our plans. We moved on to our second option which was a wet layup. This involved hand wetting out the fiber with resin and then laying the wet fabric strips over the mold.

Wax Investment Mold
Our lost wax molding technique was carried out by constructing a set of cylindrical molds to pour a series of wax cylinder blanks. These wax cylinder blanks are then machined to the correct diameter and the ends of the cylinders are prepared with a male-female coupling arrangement connection similar to Legos as shown in Figure 60. The wax cylinders have a press fit at the joint which can be joined by hand and the joint can be additionally strengthened by melting some of the wax together. This modular design allows us to save a significant amount of time in tooling. No tooling molds are required to make our final mold.

In order to get curved members within our frame we initially thought we could heat the wax to near melting point and hand bend the curves into the modular structure. After two attempts at performing this operation it was found to be too difficult. While bending the wax cylinders many of the cylinders cracked and after we got enough non-cracked cylinders we realized after assembling them together that they formed a three dimensional curve. This complex curve was not acceptable for our final part. To combat this issue we miter cut five sections and assembled them seen in Figure 60. Although it is not a perfect curve it is a two dimensional curve which was critical to our mold.

To combine the modular sections of straight wax the ends needed to be hand shaped in order to connect each piece. This proves to be extremely time consuming but worked as planned. We also hand sculpted and molded fillets into the mold. The fillets were constructed from a triangular blank mold that was larger than the size of the desired fillet. After the blanks were made they were shaped and a significant portion of the wax was melted away with a hot knife tool. After the fillet had been shaped it was melted onto the larger wax mold. Remaining gaps were filled with hot wax dripped into the voids. After all the pieces were assembled and all of the major gaps were filled some mesh sand paper was used to smooth the surface. Sanding the wax with mesh sand paper seemed to work relatively well when compared to all of the different techniques that we tried to get a better surface finish. When the mold was finished and sanded it yielded a fairly smooth surface finish for us to layup on which can be see in Figure 62.
The whole structure was assembled on top of a full scale plot to ensure correct dimensions. After the part had been cured on the mold, it was taken to the composites lab where the part was hung in the oven and the wax was melted out from the inside (seen in Figure 64). The wax cylinders fell out as soon as there was enough of a film of melted wax on the inside of the part. The wax was collected by large pots placed on the floor beneath the openings in the part.

Figure 60 Machining initial wax test pieces to derive feeds and speeds for production.

Figure 61: Hand shaping wax cylinders (this is the bent tube in the middle of the frame)
Figure 62: Final wax mold right before the layup

Figure 63: Fuselage mold vacuum bagged after a layup
Composite Layup

From our initial part design and layup specifications we selected a plain weave fabric and unidirectional fabric for reinforcements (see Table 13). West Resin system was selected based on prior experience with the system. FEA and MATLAB to optimize our layup scheme (see Table 14). From this we finalized our material quantities and placed our order with 150% of the materials we needed to build it. This extra 50% is to allow for scrap fabric and broken parts.

Table 13: Carbon fiber specifications

<table>
<thead>
<tr>
<th>Fabric</th>
<th>Thickness (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain Weave</td>
<td>0.012</td>
</tr>
<tr>
<td>Unidirectional</td>
<td>0.007</td>
</tr>
</tbody>
</table>

Table 14: Layup scheme

<table>
<thead>
<tr>
<th>Section</th>
<th>Area (in²)</th>
<th>Plain Weave Ply Count</th>
<th>Uni Ply Count</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom</td>
<td>165.38</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td>Curve</td>
<td>93.17</td>
<td>4</td>
<td>2</td>
</tr>
<tr>
<td>Angle</td>
<td>153.77</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Front Fillet</td>
<td>17.85</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td>Top Fillet</td>
<td>21.64</td>
<td>5</td>
<td>0</td>
</tr>
<tr>
<td>Bottom fillet</td>
<td>15</td>
<td>5</td>
<td>0</td>
</tr>
<tr>
<td>Seat Back</td>
<td>90.9</td>
<td>5</td>
<td>6</td>
</tr>
</tbody>
</table>
To practice our layup procedures and get comfortable with the materials we were using, we chose to start by building the rotor mast tube which is a long straight tube. Initially we attempted to roll the carbon fiber over the wax cylinder. However during the curing process many wrinkles were introduced into the part (likely due to the vacuum bagging). In order to combat this problem we decided to do half of the tube at a time which would require two separate layups and cures in order to make the complete rotor mast. This worked significantly better than rolling carbon fiber over the tube and there was only a few minor wrinkles in the final product. There was still one problem with this part, we had used our minimum layup scheme which yielded a part that was too fragile to use. To solve this problem an additional ply of fabric was added to the final layup scheme. Our third attempt at this part came out exactly as anticipated with only minor wrinkles.

The same technique of half of the structure at a time was also used for the layup on the fuselage. The main structure was done in five layups. The first two was the minimum amount of ply’s for the entire fuselage and the following two were to add additional layers to the critical parts of the fuselage. One unanticipated layup occurred due to some thin areas caused by not overlapping the ply’s enough. This issue was caused by not being able to tell how far each overlap went onto the other side of the fuselage. The final layup was to attach the seat to the fuselage. This seat was used following the Human Powered Vehicles specifications and using their molds. We then cut one tube in half for the seat to mount onto. The seat was first epoxied onto the fuselage and then a layup over the entire half tube. Overall the fuselage came out well for a first part while developing a investment wax molding system.

Figure 65: Wetting out carbon fiber strips for a layup
Attaching the fuselage to the rotor hub was accomplished through the hinge joint. This joint was attached to each piece by bonding an aluminum piece to the inside of the tube. The composite parts had a hand sanded surface preparation while the aluminum parts were particle blasted. Before bonding the parts in each surface was thoroughly cleaned with acetone. We used cartridges of 3M DP460 as our adhesive. Due to not having an adhesive gun for the cartridges, the backs of each cartridge were pulled off and the entire contents dumped out and mixed to ensure that we had the proper ratios of adhesive and hardener. Each of the composite parts was sanded on a lapping table in order to ensure that we had a flat reference surface. These flat reference surfaces are what was used to locate each of the parts that are bonded in. After the parts were bonded in they were left for 48 hours to ensure a full cure.

Figure 66: Complete fuselage

**Rotor Hub**

**Short Fiber Composites**

For complex parts in the rotor hub we selected short fiber composites. This provides strong lightweight parts which are ideal for the rotor hub. This technique is frequently done in industry using complex machines and injection molding. Our team decided we could accomplish the same quality of parts using a less complex method.

Due to our method these parts did take a significant amount of time to produce. As it stands each short fiber part costs about 2 hours of labor not including mold and fixture making. Considering there are 10 short fiber parts on the assembly plus scrap parts there was a significant amount of time
invested into these parts. Production time for these parts could have been reduced by machining them from metal or cast plastic. However the parts that we produced were light weight and function well under our initial test conditions. We selected short fiber composites to ensure a sound bonded joint with no risk of corrosion or inconsistent bonding. We initially chose machined magnesium for these parts but abandoned that choice due to the potential of galvanic corrosion. Plastics that met our strength requirements such as PEEK do not bond well so they could not be used for our parts. While the method we used to make the short fiber composite parts was simple, the parts came out with reasonable consistency. Figure 67 shows the mold next to one of the short fiber parts.

Figure 67: Solidworks model showing exploded spar ring mold next to a spar ring.

To create our parts the process began by applying Frekote 700NC mold release agent to the entire surface of each mold including the outsides. Frekote 700NC forms a chemical boundary that prevents the resin from bonding as well as providing chemical cleaning to the mold surface. Since Frekote 700NC is a chemical boundary it does not appreciably affect the tolerance of the parts. Frekote was very effective as the parts were fairly easy to remove from the mold. Figure 68 shows a disassembled mold covered in Frekote.
The short fiber composites were made using the same resin system and cut up fabric as the rest of the fuselage. The fabric was cut into .5-1" pieces so that it was easy to pack into the molds. We learned after from a structural engineer from Boeing (David Espasta) that there have been studies published about the optimum fiber length for these types of parts. He suggested that 1" might be too long especially given the small scale of the mold cavities. He noted that the longer fibers might be more inclined to lay smoothly in the mold parallel to one another instead of getting tangled together inside the part. We did not pursue this issue any further but it should be an area of further research.

The next step involved mixing the resin system with the graphite. Our goal was to have a 60% fiber by mass but we are not entirely sure of the properties due to several key deficiencies. First the scale we used did not consistently maintain its reading and would slowly decrease well into negative weights. Second is that the compression molding process causes resin to squeeze out of the molds altering whatever resin content we would have started with. There are some more advanced ways to
measure the resin content but they were not available to us at Cal Poly. In the end we hand-mixed the resin using the scale and then added the carbon and chose a consistency where the carbon was completely coated but there was not significant excess resin. We then hand packed the carbon mix into the mold using popsicle sticks, aluminum welding wire and some aluminum sheet metal. The molds were designed to reduce the volume of the cavity by about 16% from the filled to level condition as in Figure 69. For this reason when filling the molds we tried to cram in enough fiber that this small reduction in mold volume would be sufficient to compress the part as detailed in Figure 70. A typical injection molding process uses a pressurized plunger to force the fiber matrix into the mold cavity whereas our process fills the mold under ambient conditions. As a result our parts included some visible porosity which was due to the resin hand mixing. This could have been mitigated by pulling a vacuum on our parts before they were pressed to allow for the bubbles to flow out of the mold. Given the small size of our molds we could have put the entire mold into a Bell jar and pulled a vacuum to allow gases trapped through mixing as well as some of the volatiles to be removed. However the porosity is very minor and we do not expect it to hinder the material properties of our parts.

Figure 70: Spar ring mold filled with composite matrix. Note the coarse appearance of the fiber.

The final step included placing the mold into a hydraulic press and applying 4000 pounds of force to the mold platens. The resulting pressure squeezed any extra resin through small channels cut into the mold. Figure 71 shows excess resin being pressed out of a mold. This was an essential indicator which showed that enough of the composite matrix was added to the mold. The molds were allowed to cure at room temperature over night. As with the graphite, David Espasta also suggested that there might be more effective types of resins for this application. He mentioned also that some of the resins that must be stored at freezing temperatures can be higher quality as well as have mechanical
properties more suited to a short fiber composite. For this project we used the same resin that we had for the fuselage.

![Figure 71: Spar ring mold in hydraulic press. Note the excess resin oozing from the sides of the mold.](image)

After curing, the parts were removed from the mold with the arbor press and the mold flash was removed using a carbide burr knife and a diamond encrusted file. The key to machining composites is to turn it into powder using fast surface speeds, diamond tooling and slow feed rates. Due to budget constraints we drilled the parts with a high speed steel drill and ran the spindle as fast as it would go (4000 RPM) but there was still some tearing on the exit sides of the hole. For the scope of this senior project it was not an issue, but it is not the best solution.

**Molds, Tooling, and Post Machining**

The short fiber composite molds were made in three distinct parts including the part profile, and the top and bottom platens which contain end features and apply pressure to the parts. This proved to be an good design for our method of creating parts because the two platens were easily removed using jack screws leaving the cured part in the profile plate which was removed with an arbor press. As mentioned above this is not an efficient method to make these parts but for the small quantity of parts it was suitable. The following paragraphs outline the important lessons learned from mold making.

One of the most important things we did to improve the processing was to grease the 3/8" dowel pins before assembling the mold. The Frekote minimized friction with the completed part but the resin penetrated between the dowel pins and the molds and set which eliminated the clearance designed for dowel pin removal. In the platen for each style of part we machined a groove that was approximately .050" deep by .125" wide to allow extra resin to escape as the mold was compressed.
This channel was cut this small so that fibers could not flow out. Not only did resin relieve through the channels, resin also traveled through any available clearance, opening or other relief. The resin is under high pressure so at least a thin film travels through most of the mold.

![Figure 72: Photo of spar ring mold showing the channel cut to allow excess resin to escape the mold.](image)

To achieve an appropriate surface finish as well as to facilitate part removal the molds were polished. There are many different types of polishing compounds and abrasive materials available including diamond lapping compounds which are generally recommended. The diamond compounds are optimum but also expensive and out of our budget so we selected less expensive products. Our polishing process consisted of first using a 320 grit wet/dry sandpaper to remove tool marks and then incrementally progressing to 1500 grit wet/dry sandpaper using denatured alcohol as the solvent to remove the cuttings. The final step was to use felt bobs in a attached to a Dremel with polishing compound. We purchased some 3000 grit silicon carbide lapping compound and a “cut and color” Tripoli compound polishing bar. Our aluminum molds responded best to the Tripoli bar with a more lustrous finish and less mess. I expect that molds made from a harder material would respond better to the silicon carbide lapping paste. The finish had a high luster but was not mirrored.
Figure 73: Polishing spider ring mold with Tripoli compound and a felt bob in a die grinder. Note the finish is lustrous but not mirrored.

Figure 74: Completed spider ring mold after 8 parts.
We spent a good deal of time making fixtures to assure part fitment. In order to fit up the truss that comprises the rotor hub we built the fixture in Figure 75 to hold each of the 36 parts together while the glue dried. The aluminum piece at the bottom of the fixture locates the rotor mast and centers the central rings. The large vertical plates are the correct width of the truss and the 1” chromoly tubes emulate the spar box for the rotors. The turnbuckles welded to the sides of the fixture are essentially adjustable gussets so that any misalignment after welding can be corrected. Figure 76 shows how the rotor hub fixture was incorporated into the testing fixture.

Figure 75: Rotor hub truss in fixture while pultruded rods are being fit.
Figure 76: Test fixture supporting completed DaVinci IV fuselage and drivetrain.
The spar rings were difficult to post machine because they include holes tilted at three different angles as well as rotated about the axis of the part. To drill these parts we made a fixture that held three spar rings, one in each position, and wrote a CNC code to drill one hole in each part every cycle. The fixture is shown in clamped to the mill table in Figure 78. There are two spar ring part numbers that include a left handed part and a right handed part. These parts are designed such that there is an imaginary point at the intersection between the outward plane of the spar ring and the centerline of the
rotor spar that all holes go through. The coordinates for G55, G56, G57 align the z-axis of the milling machine to that point for each angled hole. This way the spar rings are simply rotated to the correct orientation for either right handed or left handed parts and the same short code can be utilized for both parts. The G-code for the spar rings is included in the appendix and the zeros are scribed onto the fixture itself.

Figure 78: Drill fixture for spar rings. Note that the parts rest against 1/4” SHCS that orient them with respect to the axis of the rotor spars. There are screws on both sides of the part which allow both right hand and left hand parts to be drilled.
Figure 79: Drilling spar rings on fixture in Haas mill.
**Test Verification Plan**

The original intent of building an updated version of the DaVinci III was to use the final project as a test vehicle for further research. The main areas of interest will eventually be control and stability but in the present time functionality, system weight, and drivetrain efficiency will all be more closely examined.

Initial testing will be to weigh the rotor hub concept and the frame supporting structure before assembling the full model. This process will allow us to verify we hit our target weight requirements for each of the components and for the whole system. After final assembly, weigh the assembled structure before performance testing.

The next test will be visual inspection of the drive components to insure there is no unnecessary noise or large amounts of frame flexing. Other testing includes building a supporting structure and suspending the assembly to test rider loading on the structure. The results of this test will help the final design determine how to control the craft. Strain gauges will be added to all major components of the assembly to observe loading conditions as well as to find rider input frequencies. Strain gauges on the frame will provide insight into frame stiffness and damping. Strain gauges on the rotor shaft will provide the rider input loads and frequencies for future control system modeling.

![Figure 80 Strain gauges locations to determine rider input loads.](image-url)
The main testing will be the efficiency of the drivetrain. A power meter at the crank will provide the rider input power in watts. An additional power meter at the end of the drivetrain will measure the output after all losses. Comparing these values will provide the overall drivetrain loss. Removing and adding different components will allow us to pinpoint the exact loss values for every component. We will test both a belt drive as well as a chain drive for accurate comparison to ensure that we are using the most efficient system.

Additional testing will include bearing testing to find the frictional moment in the rotor hub bearing. This will tell us how much friction needs to be counteracted in order to allow the rider to remain in the forward position. By inspection we will check rider comfort, center of gravity position and rotor removal.

Testing Summary

1. Weight individual systems
2. Weight final assembly
3. Suspended fuselage test
4. Drivetrain efficiency testing
5. Rotor hub friction
Results

The initial test was to weigh each of the components and compare to our target weight values. The overall craft weight was 12.84 lbs compared to the target weight of 16 lbs. The table below show the weight values per component and compares the target weight.

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
<th>Target Weight (lbs)</th>
<th>% Weight Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Assembly</td>
<td>12.84</td>
<td>16</td>
<td>19.8</td>
</tr>
<tr>
<td>Frame</td>
<td>4.06</td>
<td>6.4</td>
<td>36.6</td>
</tr>
<tr>
<td>Rotor Hub</td>
<td>1.84</td>
<td>3.2</td>
<td>42.5</td>
</tr>
<tr>
<td>Drivetrain Components</td>
<td>5.24</td>
<td>6.4</td>
<td>18.1</td>
</tr>
</tbody>
</table>

As seen by the table the drivetrain components contribute the most weight to the system. A more detail weight breakdown of the drivetrain components are shown in the table below.

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crankset</td>
<td>1.24</td>
</tr>
<tr>
<td>Bottom Bracket Assembly</td>
<td>0.62</td>
</tr>
<tr>
<td>Flywheel/ Shaft</td>
<td>2.02</td>
</tr>
<tr>
<td>Chain</td>
<td>0.82</td>
</tr>
<tr>
<td>Hinge</td>
<td>0.36</td>
</tr>
<tr>
<td>Rear Sprocket</td>
<td>0.18</td>
</tr>
</tbody>
</table>

The crankset chosen for the helicopter was for a fixed gear bicycle. These cranksets tend not to be as stiff because they will not receive the same wear as a road cycle and therefore less material tends to be used on the crank arms. While the cranksets are not quite as stiff as other cranksets, they tend to weigh less than some carbon cranksets. The typical carbon crankset runs around 700-800 grams. The fixed gear crankset is around 640 grams. Areas for weight reduction could include the bottom bracket shell and assembly or in the Flywheel and shaft. The current flywheel was constructed from a 700c wheel. Most of the spokes have been removed but because the wheel does not receive the same type of loading, less mass further out would be equally acceptable. The bottom bracket was purchased as a cartridge and weighed 0.38 lbs alone. Creating a bottom bracket compatible with cups or external bearing races would be lighter. Not to be forgotten the belt was replaced for a chain drive to speed the manufacturing and testing process.

The next phase of the testing was static load testing of the structure. Our team member Eli was selected to test the fixture because he was the closest to the appropriate build at 145 lbs. Structure sufficiently held the rider as seen in the pictures (Figure 81. Team member Eli Knight safely sits in the DaVinci IV). A raised platform was quickly assembled as a safety procedure for the rider. Once we were sure the rider held we removed the safety platform and allowed the rider to pedal. Visible input into the rotor hub was observed and thus strain gages were applied to the rotor shaft to understand the inputs from the rider.

The testing setup for static and dynamic loads is shown in Figure 82 and Figure 83. Spectra fiber is feed through a system of pulleys and then tied to 45 lbs weights. The thread is wrapped three times before tied to the 45 lb weight in order to achieve longer time duration from the test.
Figure 81. Team member Eli Knight safely sits in the DaVinci IV

Figure 82. Strain gage testing setup with team member Eli Knight pedaling 45 lb weights
Before the dynamic testing we attempted a series of static load testing. The corrected average strain value 52 microstrain. The theoretical value of strain in the rotor mast is 50 microstrain.

**Table 17. Static experimental and theoretical results**

<table>
<thead>
<tr>
<th></th>
<th>Theoretical</th>
<th>Experimental</th>
<th>% Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strain (µ)</td>
<td>50</td>
<td>52</td>
<td>4.0</td>
</tr>
<tr>
<td>Load (lbs)</td>
<td>158</td>
<td>163</td>
<td>3.2</td>
</tr>
<tr>
<td>Stress (psi)</td>
<td>500</td>
<td>520</td>
<td>4.0</td>
</tr>
</tbody>
</table>

The average testing time was about 10 second before the weights reached the top of the fixture. The result of the first dynamic testing run is shown in Figure 84.
Gage 4 monitors the motion of the rotor mast front to back while Gage 3 monitors the motion from side to side. In this initial run the average cadence of the crankset was 58.8 RPM. The period of oscillation for Gage 4 is 0.5 second and the period for Gage 3 is 1.05 seconds. The values converted to RPM values are shown in Table 18. The frequency value seen in the strain gages correspond to frequency of a rider’s pedal stroke. Gage 4 oscillates twice for every pedal repetition because it experiences both pedal strokes for every repetition. Gage 3 experiences the right and the left pedal strokes differently. As seen in the other data runs the differentiation subsides with a faster pedal speed as shown in Figure 85. A rider able to apply smooth pedal stroke will create a sinusoidal function which oscillates at every pedal stroke.

**Table 18. Frequency data for dynamic strain run 1-3**

<table>
<thead>
<tr>
<th></th>
<th>Gage 3 Frequency (RPM)</th>
<th>Gage 4 Frequency (RPM)</th>
<th>Average Frequency (RPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Run 1</td>
<td>60</td>
<td>30.7</td>
<td>58.8</td>
</tr>
<tr>
<td>Run 2</td>
<td>79</td>
<td>41.4</td>
<td>83.2</td>
</tr>
<tr>
<td>Run 3</td>
<td>*</td>
<td>43.6</td>
<td>83.2</td>
</tr>
</tbody>
</table>

*Strain Gage failure*
Another key observation of the dynamic data is the microstrain which the period oscillates around. In Figure 84, Gage 4 oscillates about 50 microstrain; however Gage 3 the mean strain is 21 microstrain. The reason for this is the offset of the rider weight positioned more toward the chain. The frame tubes are 2.5 inch in diameter and needs to position the rider on the outside of the chain also. As a result, there is a tendency to apply more weight over the drive side and place the strain gage in compression rather than tension. Reducing the data using Classical Laminate Theory the moment side to side is 471 lb-in and the moment front to back is 79 lb-in.

![Dynamic Data](Dynamic Data.png)

*Figure 85. Microstrain oscillation at rider cadence 60 rpm and 80 rpm*

Other testing include the efficiency of the bearings. The efficiency of bearing is measured by the tension in divided by the tension out. A weight of known quantity 2.2 lbs tensions the spectra. The spectra is wound around a bearing and pulled out at 90 degrees to simulate the actual setup. A fish scale is attached to the free and pulled. The reading on the scales was 2.24 lbs. As a result the efficiency of the bearings were 98% which is what was expected.

**Conclusion**
As a result of the testing data we determined:

- The rider weights shifting their weight side to side is significant and should be addressed when considering control.
- The rotor mast experiences inputs front to back also
- The oscillations of the strain reflects the cadence of the rider
• The reason the side to side strain oscillates at a mean strain different from front to back is due to weight distribution of the rider
• CG correctly adjusted front to back
• Weight savings could be considered before flight in the areas of the driveshaft, flywheel and chain
<table>
<thead>
<tr>
<th>Item No</th>
<th>Specification or Clause Reference</th>
<th>Test Description</th>
<th>Acceptance Criteria</th>
<th>Test Responsibility</th>
<th>Test Stage</th>
<th>Pass/Fail</th>
<th>TIMING</th>
<th>Start date</th>
<th>Finish date</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fuselage Structure Weight</td>
<td>Weigh fuselage, not including drivetrain or rotor hub</td>
<td>8.0lb Maximum, 6.4lb Target</td>
<td>Brenton DV</td>
<td>Pass</td>
<td>4.06 lb</td>
<td>6/1/10</td>
<td>6/4/10</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>CG Position</td>
<td>Test that rider CG is directly below rotor shaft</td>
<td>5% Error</td>
<td>Brenton DV</td>
<td>Pass/ Fail</td>
<td>FB SS</td>
<td>6/1/10</td>
<td>6/4/10</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Comfort</td>
<td>Test rider pedaling comfort</td>
<td>Yes/No</td>
<td>Brenton CV</td>
<td>Pass</td>
<td></td>
<td>6/1/10</td>
<td>6/4/10</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Drivetrain Weight</td>
<td>Weigh drive components</td>
<td>8.0lb Maximum, 6.4lb Target</td>
<td>Daniel DV</td>
<td>Pass</td>
<td>5.24 lb</td>
<td>6/1/10</td>
<td>6/4/10</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Rotor Hub Weight</td>
<td>Weigh rotor hub</td>
<td>4.0lb Maximum, 3.2lb Target</td>
<td>Eli DV</td>
<td>Pass</td>
<td>1.84 lb</td>
<td>6/1/10</td>
<td>6/4/10</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Rotor Bearing Power</td>
<td>Test frictional moment</td>
<td></td>
<td>Eli CV</td>
<td>Pass</td>
<td>0.7 lb-n</td>
<td>6/1/10</td>
<td>6/2/10</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Hub Structure</td>
<td>Make sure hub carries design loads</td>
<td></td>
<td>Eli DV</td>
<td>Pass</td>
<td></td>
<td>6/1/10</td>
<td>6/2/10</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Rotor Removal</td>
<td>Designed to allow rotor removal?</td>
<td>Yes</td>
<td>Eli CV</td>
<td>Pass</td>
<td></td>
<td>6/1/10</td>
<td>6/2/10</td>
<td></td>
</tr>
</tbody>
</table>
Management Plan

Our management plan divides the project up into three major parts: the drive train, rotor hub, and fuselage. We organized this way to have one specific team member focusing on every decision made on that component. While each team member will contribute to all components they do have one piece of the project to focus on which allows for sufficient collaboration and still allows for one person to make a final decision if need be. To aid in collaboration with different group members we are using an online storage server called DigitalDropbox. To maximize the potential of this system each group member downloaded an application to sync the Dropbox files to our individual computers and automatically update every time a file was updated. This system has the advantage that each group member is assured that the files are the most up to date and that the files are all consistently organized which is a requirement for the parametric file system used by SolidWorks. From a collaborative standpoint our group has experience in manufacturing tooling and composites as well as knowledge in finite element analysis. In this way our group functions as a consultancy where we have experts in each area to probe for help and new ideas.

Subsystems

Fuselage - Brenton Haven

Drive Train - Daniel Hudson

Rotor Hub - Eli Knight

While our senior project group was divided into respective subsystems our whole group is working closely with the rest of the Aircraft Construction Club to develop our concepts and to guide the direction of our design. We have weekly meetings with our advisors, the rotor group and the kinesiology group where we can get feedback and validation on our decisions. In addition to these weekly meetings we have had “field trips” to meet with industry experts in the areas of rotor design, helicopter theory and composite manufacturing.
Conclusion and Recommendations

As specified by our sponsor DaVinci IV will be an updated version of the DaVinci III to match the change in technology over the last 20 years. The whole craft will be designed around a new wing concept that consists of a large box-truss spar structure which will improve rotor stiffness and therefore vehicle control. The fuselage for DaVinci IV is designed to maximize the power output of the pilot, and deliver the maximum amount of the input power to the propellers while minimizing weight. The drivetrain was optimized to minimize part count and power consuming frictional interfaces. The rotor hub is designed to distribute the fuselage loads to the rotor, minimize localized stress loading in the thin composite member, and minimize friction at the rotor hub bearing to allow the rotor to rotate independently of the rotor.

A large challenge for this project has been determining an appropriate scope. Since no one has ever met the requirements for the Sikorsky prize, the project could easily extend to problems we are not equipped to solve. For example, the biggest issue with a hovering rotorcraft is stability which is a huge dynamics and controls problem exceeding the capability of our current level of education. Instead we focused our effort on some of the more immediate problems facing this current attempt at the Sikorsky prize. As mentioned before there is no appreciable quantity of data from the DaVinci III from which to base analysis or to create an analytical model for control purposes. For this reason our group designed focused on building a functional fuselage and drive system so data can be collected for a future analytical model. Additionally a project focused on machine design is a better fit for the format of the Mechanical Engineering senior project. However, in the course of our design work we have identified several other areas that would make appropriate senior projects: an electrical drive system, a heading hold control system, and an aerodynamic stability control system.

An electrical drive system would replace the current spectra thread based system with an electrical generator which would convert pedal power to electricity and transmit that power to electric motors at the wing tips. The advantage of such a system is that electrical controls could be easily integrated. The drawback is that the efficiency of such a system seems to be lower than a direct drive system. From the standpoint of this project we decided not to pursue an electrical system because this would be a substantial project on its own. In a similar light the heading hold gyro exceeded the scope of this project and would be more appropriately integrated into a craft that has an electrical drive system. Finally in conjunction with the Aircraft Construction Club we decided not to address the stability issue for the craft at this time so that we could focus on producing a physical, testable configuration to obtain more information for future design.

Recommended Adjustments:

- Grooves on the hub roller need to be much deeper. The thread keeps popping off if the it is not pulled exactly parallel to the grooves.
• New flywheel. The flywheel currently on the machine is a spare bicycle wheel. DaVinci III used a large hoop laced with Kevlar thread and covered with monocote. This design provides a larger moment of inertia while actually saving weight over our current part.
• The winch spool needs to be machined smaller because to lighten the part and allow for more thread to spool on it. This would change the gear ratio and to compensate for this a sprocket with one less tooth could be used for the “rear” sprocket. Another modification that would help is to have larger ends on the spool.
• The holes in the hinge which are to set the hinge angle do not have enough clearance to tighten a nut. We suggest that when the appropriate pilot is found and balanced that and two new holes get drilled in a more accessible location. This is a necessary element as the craft has a small tendency to rock when pedaling. However the hinge does a good job of balancing the craft so the screws have very low loading.
• Several drivetrain components could be updated before flight. The current bottom bracket is a cartridge. Lighter options are available including external race cups. Also, due to our sponsor's request we selected a chain drive. Carbon 8mm pitch belt drives are available in appropriate sizes.

In conclusion we would like to emphasize that the purpose of this project was to redesign DaVinci IV for 2010 and to understand and document the decisions made for this aircraft. Thus this project was meant to lay the ground work for future attempts at the Sikorsky prize instead of trying to meet all of the prize objectives with this aircraft iteration. In this way our design satisfies the need of our client, the Cal Poly Aircraft Construction Club, by providing a testable aircraft fuselage with sufficient documentation that future groups can progress the design instead of retracing the original design.
Works Cited


Appendix

Figure A.1: Task Management Chart
Figure A.2: Gossamer Albatross Frame Set
Figure A.3: Gossamer Albatross drive train
Figure A.4: Gossamer Albatross Propeller Drive
Figure A.5: Da Vinci III Schematic

Figure A.6: Da Vinci III Flight
Da Vinci III
Human Powered Helicopter Project
Cal Poly San Luis Obispo

DRIVE TRAIN

System: Non-continuous pedal driven.
Speed: 85 - 100 operational r.p.m at crank.
Sprocket: Overdriven chain derailleur.
Flywheel: Momentum, for smooth power input.
Power: Allied Spectra thread.

MAIN ROTOR

Span: 50 ft. radius in 3 sections, 2 ft. radius hub.
Span: 0 and 45 degree wrap carbon graphite.
Airefoil: FX63-137.
Chord: 5.10 ft. constant to 16 ft. then taper to 2.50 ft. at tip.
Twist: Linear -8 degree.
Ribs: Divinycell every 2 ft.
Skin: Heat-shrink Melinex.
Speed: 8.5 - 9.0 operational r.p.m.

PERFORMANCE

Weight: Helicopter: 97 lbs.
        Pilot: 130 lbs.
Power: Available: 0.8 h.p. for 1 minute
       Required: 0.3 - 0.75 h.p.

PROPELLER

Prop: 2 ft. radius, two bladed.
Chord: 3.5 in. maximum.
Twist: Non-linear 60 degree.
Speed: 550 operational r.p.m.
       44.5 ft/sec. flight speed.
Efficiency: 84 percent.
Figure A.8: Yuri I, only other successful flying human powered helicopter.
Figure A.10: First page of sample calculations showing why a large flywheel above the rotor is not practical.
From torsion of axis theorem:

\[ I = I_{dc} + m\theta^2 \]

where \( \theta \) = distance from axis of mass

Assume masses on the ends of light beams:

\[ I = m\theta^2 \]

If we limit the weight to 2 lbs,

\[ \frac{2 \text{ lb}}{2 \text{ ft}^2} = 0.0621 \text{ slugs} \]

\[ \text{1 slug} = 16 \text{ lb}, \quad \text{1 ft} = 0.33 \text{ ft} \]

\[ \theta = 1 \text{ slug} = 16 \text{ lbs} \cdot \frac{0.33 \text{ ft}}{1 \text{ slug}} \]

\[ d = \sqrt{\frac{I}{m}} \]

\[ = \sqrt{\frac{2.7 \text{ lbs ft}^2}{0.0621 \text{ slugs}}} \]

\[ = 6.6 \text{ ft} \rightarrow \text{oops, too big.} \]

\[ \therefore \text{flywheel on top of the wing is probably not a feasible option because we need a lightweight structure and we are limited by how fast a human can spin the flywheel.} \]
% O36901
(SPAR RING HOLES)
(TOOL#6)
(#3(.250IN) CENTER DRILL)
(SHOP DRILL HOLES IN ALL 3 PARTS)
(G54 IS AT THE TOP RIGHT CORNER OF THE FIXTURE)
(G54 Z0 IS AT THE TOP OF THE FIXTURE)
(ALL OTHER COORDINATE SYSTEM MEASUREMENTS)
(WRT G54 IS INCLUDED IN THE CODE)
(STARTS AT 45DEG. HOLE AT G55)
(X-0.619,Y-1.625,Z.444 )
T6M6
G0G80G40G90G55X0Y0
S4000M3
/M8
G43H6Z.1
G81Z-.2R.1F8.
G80
G0Z2.
M1
(DRILLS VERTICAL HOLE AT G56)
(X-1.25, Y-2.769, Z-.374)
S4000M3
/M8
G0G80G40G90G56X0Y-.3
Z.1
G81Z-.2R.1F8.
G80
G0Z3.
M1
(DRILLS 67 DEG HOLE AT G57)
(X-2.316,Y-1.625,Z.144)
G0G80G40G90G57X0Y0
S4000M3
/M8
Z.1
G81Z-.2R.1F8.
G80
G0Z2.
M9
G0G80G91G28Z0.
G90
M5
(DRILL THROUGH HOLES)
(TOOL #7)
(LETTER K (.281 DRILL))
(DRILLS HOLES IN ALL THREE PARTS)
T7M6
G0G80G40G90G55X0Y0
S4000M3
/M8
G43H7Z.1
G81Z-.42R.1F8.
G80
G0Z2.
M1
(DRILLS VERTICAL HOLE AT G56)
S4000M3
/M8
G0G80G40G90G56X0Y-.3
Z.1
G81Z-.42R.1F8.
G80
G0Z3.
M1
(DRILLS 67 DEG HOLE AT G57)
G0G80G40G90G57X0Y0
S4000M3
/M8
Z.1
G81Z-.42R.1F8.
G80
G0Z2.
M9
G0G80G91G28Z0.
G90
M5
M30
%