

RAM Air Turbine – White Team

Final Report

ME 160 – Spring 2008

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Introduction

Throughout the history of aviation, safety has always been a major concern. No matter how reliable the machinery or the pilot, accidents can and do occur. These accidents are rare, but shocking when they do happen – such as the series of horrific accidents suffered by the de Havilland Comet in 1954¹, or the case of Air Transat Flight 236, which ran out of fuel over the Atlantic Ocean². In order to prevent loss of life in these situations, a number of safety devices have been built into modern commercial aircraft; one of those devices, the Ram Air Turbine (RAT), is the subject of this project.

Large commercial aircraft (like the Boeing 747 or the Airbus A380) use a hydraulic system in order to operate the flight surfaces (elevators, ailerons, rudder), which directly control the aircraft in flight. These hydraulic systems are kept pressurized by the aircraft's engines; however, if for some reason the engines cease to function, they cannot produce the pressure necessary to maintain control of the aircraft. Modern aircraft have an additional power system called the Auxiliary Power Unit (APU) which can take over responsibility for the hydraulic system in an emergency; however, the APU is itself a small engine, and needs fuel in order to operate. So, if the airplane loses its fuel, neither the engines nor the APU can keep the airplane in a controllable state.

To solve this problem, large commercial aircraft come equipped with RATs. A RAT is a small, windmill-like device that is deployed into the airstreams flowing around the aircraft. It converts a small portion of the energy in the flow into hydraulic and/or electrical power for the aircraft, allowing it to be flown even if the plane is entirely empty of fuel. This device has been responsible for saving hundreds of lives, in the case of the Air Transat flight mentioned above. Because of the ram-air turbine, the plane was able to land in the Azores despite losing all its fuel, without a single fatality².

¹ http://en.wikipedia.org/wiki/De_Havilland_Comet

² <http://aviation-safety.net/database/record.php?id=20010824-1>

Project Definition

The goal of this project is to create our own ram-air turbine design, completely from scratch. The scope of the project includes the housing, blade geometry, electrical generator and mount; due to time constraints, a deployment mechanism for the RAT was not included. More formally, our goal is to create a turbine capable of generating supplemental power for an aircraft using only the air flow around that aircraft.

Our intended market is small, low-speed aircraft, as our RAT will be undersized for large commercial aircraft applications; the wind tunnel at Duke University limits us to a diameter of 10 inches, whereas airliner RATs are on the order of two to three feet in diameter³. One purpose for our product will of course be power production for the primary electrical system in emergency situations. However, our RAT could also be used to provide a continuous supply of power in normal flight to devices that are separate from the aircraft's main electrical system; the best example of this are crop dusters, which sometimes use a ram air turbine to power their spray pumps. Applications for external pods on unmanned aerial vehicles are also possible⁴.

This project was to be completed in competition with another team (the RAT Blue Team). In accordance with this, the sponsors of the project (Dr. Robert Kielb, and Duke University as a whole) provided the teams with a set of design objectives, in order to ensure that there was a basis of comparison between the teams.

Design Objectives:

- Provide Maximum Power Output at 100ft/s airspeed.
- Max Blade Radius: 10 in
- Max Dimensions of 20 in high, 20 in wide, and 30 in long
- Should weigh less than 50 lbs.
- Must include brake so that speed would not exceed safe limits

³ http://en.wikipedia.org/wiki/Ram_air_turbine

⁴ <http://www.ghetzleraeropower.com/rat.htm>

Team Organization:

Primary responsibilities, by team member:

- Tyler – Team Leader, Blade Design, CAD, Fabrication
- Rob – Statistical Analysis, Ordering of Parts, Finances
- Stephen – Information Gathering, Fabrication, Report Compilation
- Homero – FE/CFD Analysis, CAD, Frequency Analysis

Gantt Chart:

	18-Feb	25-Feb	3-Mar	10-Mar	17-Mar	24-Mar	31-Mar	7-Apr	14-Apr	21-Apr
Finish CAD and Update Design										
Conduct FE Analysis										
Conduct Aerodynamic Analysis										
Conduct Rotordynamic Analysis										
Order Parts										
Build Prototype										
Rapid Prototype Blisk										
Design and Build Brake System										
Assemble Prototype										
Test Prototype										
Make Final Design Changes										
Prepare for DDR										

Budget:

Date	Vendor	Item	Price
2/21/2008	McMaster Carr	Coupling - 9861T8 1/2" and 5/16"	25.46
		Steel Rod for shaft 4455T111	4.18
		Aluminum tube 9056K511	42.28
		Bearing 6383K234 x4	25
		Thrust Bearing 6655K17 x4	10
		Aluminum Rod - 8974K971	74.43
		90032A035 - 5/8" - 24" threaded rod	5.84
		90566A235 - 5/8" Lock nuts - \$8.18 x1	8.18
		91078A235 - 5/8" Hex nuts	8.28
		8975K911 - 3"x12"x3/8"	11.59
	www.emotorpro.com	1/4 HP 12VDC Fasco 3" Diameter School Bus Heater Motor	52.15
3/21/2008	McMaster Carr	Steel Rod for shaft 4455T111	4.18
		Aluminum tube 9056K511	42.28
		Coupling - 9861T8 1/2" and 1/2"	25.46
		60355K15 Bearings 1/2" shaft D 1 1/8" OD, W 1/4" x4	18.88
			www.anaheimautomation.com
4/5/2008	McMaster Carr	90032A011 Zinc-Plated Steel Rod 10-32 Thread, 2'	1.41
		7633T412 Aerosol Epoxy Coating Gloss Black	5.79
		98842A035 Zinc-Plated Steel Rod 5/8"-18 Thread, 3'	9.42
		90465K671 Alloy 7075 Aluminum Rod 3/4" Diameter, 1'	9.63
		Budget:	1000
		Shipping estimate:	80
		Total:	566.34

Information Gathered

As part of the initial design phase of the project, a survey was sent to Dave Bannon, an engineer at Hamilton Sundstrand, a leading manufacturer of ram air turbines. Mr. Bannon was asked to rank a list of design features in regards to importance in a commercial RAT implementation. The full list of responses is given in Appendix C; in summary, power output, weight, and stowability were the most important features⁵. Mr. Bannon also emphasized the need for a braking system to prevent the RAT from attaining unsafe speeds; failing to regulate the turbine's speed could cause a disk burst, endangering the safety of the aircraft's passengers.

A list of design features, ranked by importance, was also obtained from the team's sponsor, Dr. Kielb. The responses obtained, in order of decreasing importance, were power, weight, stowability, durability, drag, appearance. These results matched those given by Dave Bannon, so the team moved on to the next phase – evaluation of existing designs.

First, the team researched Hamilton Sundstrand's own commercial designs. These designs are representative of the market as a whole, being featured on such planes as the Boeing 747 and the Airbus A380. These designs consisted of a large diameter (2-3 ft.), two-bladed RAT with a wide hub driving a hydraulic pump mounted behind the blades. The blades are variable pitch⁶; the pitch is controlled using a centrifugal sensor mounted in the hub, which keeps the RAT rotating at a constant speed, irrespective of the aircraft's speed. In case of failure of the centrifugal sensor system, the RAT is equipped with a backup compression brake mounted between the shaft and the hub⁷. To recover from stalls induced by momentary obstructions to the RAT's incoming airflow (such as when the landing gear deploy), the load is reduced or removed when the turbine's speed drops below a certain level⁸.

While this input was valuable, it was obvious that the team's design would have to differ significantly from Hamilton Sundstrand's to be a success, due to the constraints on disk diameter imposed by the project's organizers (a two-bladed model with a ten-inch diameter would not have extracted enough energy from the flow). Older military RATs follow this small-radius, many-blade design, but information on military ram air turbines could not be found.

⁵ Bannon, Dave. Email to author, 01/21/2008 (See Appendix C).

⁶ Patent #7306430B2, filed by Hamilton Sundstrand. <www.freepatentsonline.com>

⁷ Patent #2006/0257247, filed by Hamilton Sundstrand. <www.freepatentsonline.com>

⁸ Patent #7197870B2, filed by Hamilton Sundstrand. <www.freepatentsonline.com>

Design Concepts

Using the advice given by Dave Bannon and Dr. Kielb, and the ideas generated by existing products, the team then formulated the exact design decisions that would need to be made in the design of our prototype. The TRIZ method was first used to try and generate ideas we had not yet thought of. The team came up with three TRIZ objectives: increasing force while not diminishing reliability, increasing ease of manufacturing without diminishing strength, and increasing ease of manufacturing without diminishing power. Recommendations were then obtained by entering these objectives into a TRIZ evaluation website⁹. None of the ideas thus generated were new or useful; however, the TRIZ evaluation did serve to reinforce some of the ideas already generated by the team.

After accumulating and processing all this information, the team came up with five primary design decisions: number of blades, blade/hub material, support structure, motor size, and brake design. A QFD analysis¹⁰ was then performed, using the relative importance values obtained from Dave Bannon as weights in the matrix. The results of the QFD are given in the figure on the next page.

The five bladed rotor, while less stowable, produces significantly more power than the two-bladed rotor, which outweighs the loss of stowability. Aluminum blades, though stronger than plastic blades, are also much heavier and more expensive to machine, which offset the slight gains in safety they afford. The tripod mount, though it is heavier, has more drag, and is less stowable, is far safer than the single rod mount (due to stability issues). While a larger motor produces more power, a smaller motor has more consistency (due to lower starting torque), is much cheaper, and weighs much less. Finally, though the electro-magnetic brake is far more interesting technology-wise, the mechanical brake beats it out in nearly every category.

As a result of this QFD analysis, the team elected to use a five (later, seven) bladed rotor made of plastic rapid-prototyping material, with a small motor, mechanical brake, and tripod supports. Several of these results, such as the choice of a smaller motor and a mechanical brake, came as surprises to the team, and caused a change in our design; this showed us the value of the QFD procedure (which we also doubted initially).

⁹ <http://www.triz40.com/>

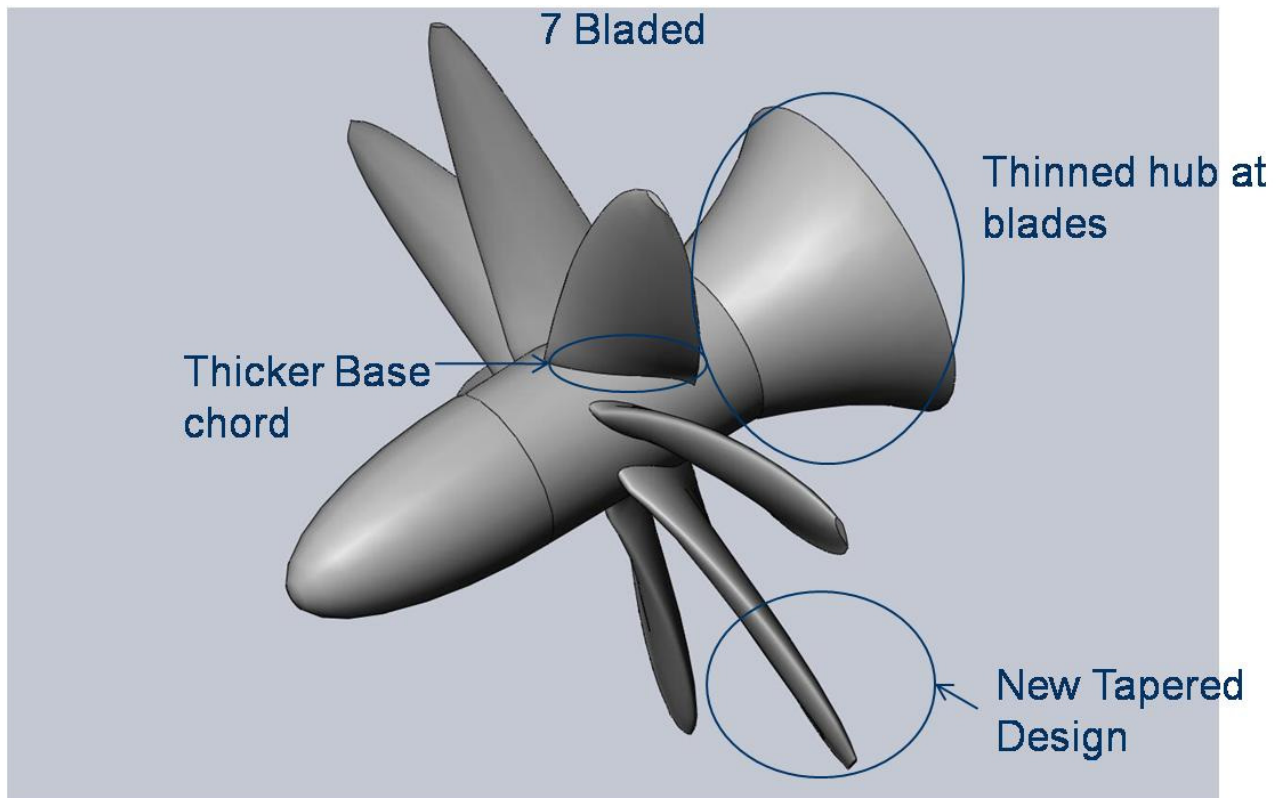
¹⁰ <http://www.gsm.mq.edu.au/cmit/qfd-hoq-tutorial.swf>

	5 Bladed	2 Bladed	Aluminum	Plastic	Tripod	Single Rod	Large Motor	Small Motor	Mech. Brake	EM Brake
Stowability x5	0	1	0	0	0	1	0	1	0	1
Power Output x5	3	0	1	0	0	0	3	0	0	0
Power Consistency x2	1	0	0	0	0	0	0	3	3	0
Drag x1	0	0	0	0	0	1	0	0	0	0
Weight x4	0	1	0	3	0	1	0	3	1	0
Deployed Size x3	0	0	0	0	0	0	0	1	0	0
Cost x3	0	3	0	9	0	1	0	3	3	0
Safety x5	1	0	3	0	9	0	0	0	3	0
Total:	22	18	20	39	45	13	15	35	34	5

Results of QFD Analysis

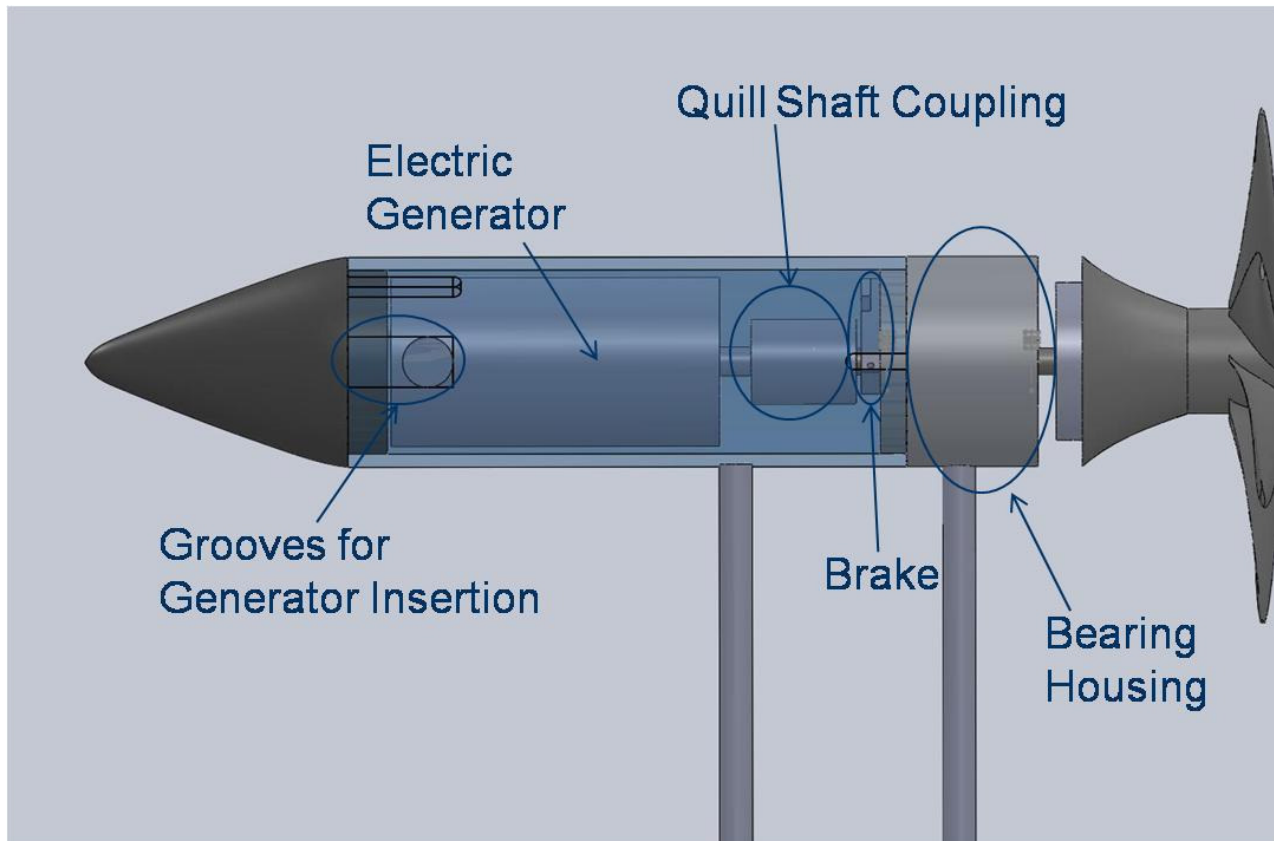
Product Architecture:

The final design arrived at for this project consists of a minimalist support structure to a multi-bladed reverse propeller type wind turbine that turns an electric generator. The blade-disk propeller design decided upon contains seven blades that are designed for near optimum gamma distribution. These blades are located on a core that has a fairly streamlined design to decrease losses induced by drastic area changes across the turbine. A rendering is included below.



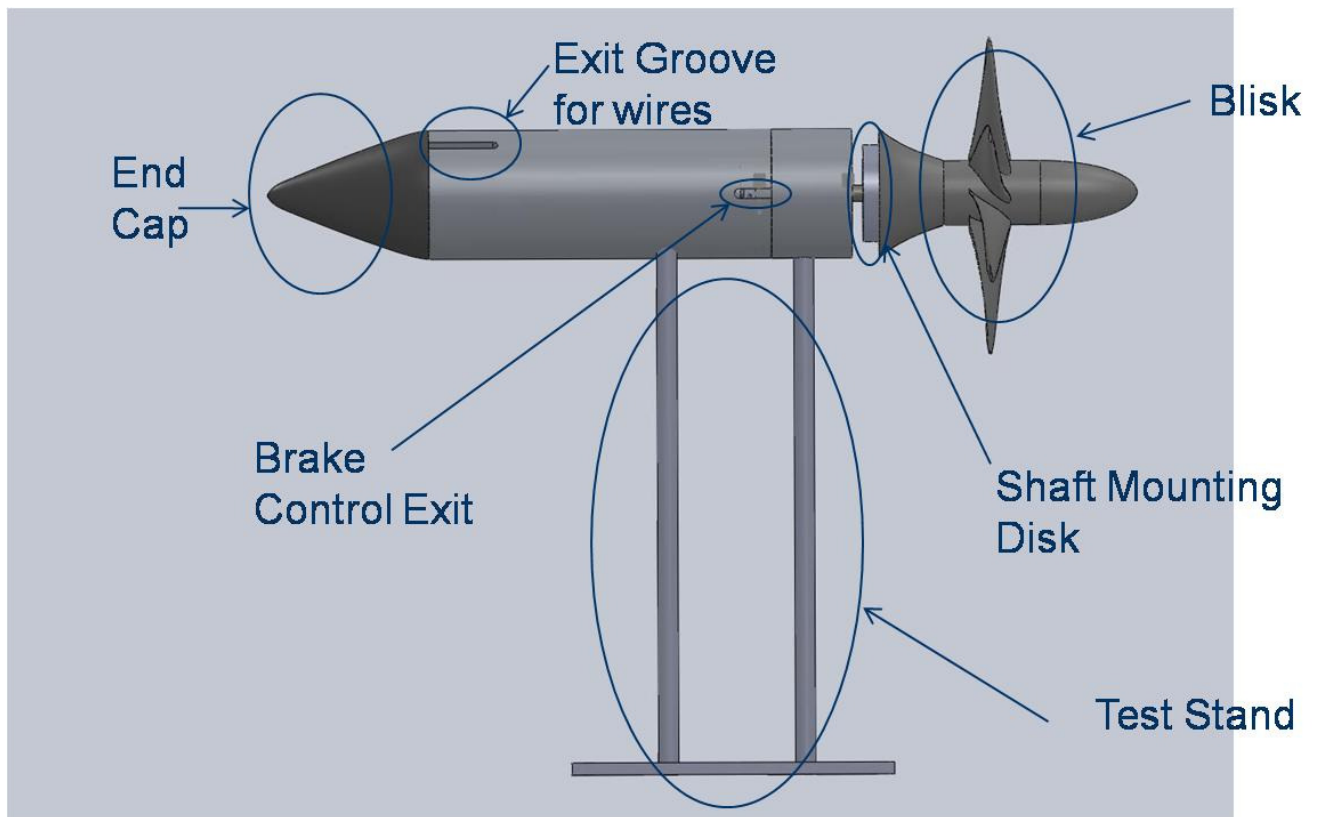
The SolidWorks design shows the layout of the prototype blisk and improvements that were made over previous initial designs. The chord distribution was generated using the FORTRAN code provided by Dr. Hall, and the rendering also shows that this ideal case was deviated from in order to increase strength at the root by maintaining the maximum thickness of the blades to the root. This will reduce some power capabilities of the design but prevent failures.

The next major aspect of the prototype that has to be explored is the setup of the internal shaft and generator support structure which includes the mechanical brake. This is shown in the following figure.



The above SolidWorks model shows the bearing housing, brake and quill shaft coupling, which are the major components that interact with the rotor shaft. The bearing housing is for all intents and purposes a solid aluminum cylinder with the center removed for the shaft to pass through and circular slots added for the bearings to be press fit into. Connected to the bearing housing by a bolt is the brake: a hinge made by two aluminum bars milled so that they are stacked flush at their connection point. The inner sides of the hinge pieces have bike brake rubber epoxied on to act as the braking material. A hole is located at the tips of each bar where the cord for a bike brake is placed through them. A spring is then placed on the cord between the two bars to keep the hinge open during normal operation. The bike cord exits the casing on one side and is terminated and kept stationary on the other side of the hinge by a nut and bolt. More of the brake is discussed in the prototype. The shaft is then coupled to the motor by a quill shaft coupling which dissipates and does not transmit vibration or unbalances while still delivering torque. The motor is the final component shown here which is fixed in position with set screws and the two grooves that hold the compartments on the motor that house the brushes.

This leads to the final part of the architecture which is the total assembly shown in the following figure.



The other major components shown here are the exit slots in the generator housing for the brake and wires for the generator. The other major components are the simplistic test stand which mainly supports the bearing housing and then balances the rest of the housing on a tripod system that then attaches to the aerodynamic force load cell. The SolidWorks drawing also shows the blisk attachment disk that attaches the blisk to the rotor shaft. The final thing shown for the product architecture is the end cap which attempts to reduce separation at the end of the Turbine.

Further improvements that would be made to the architecture once the product were to move out of the prototype phase would include a more robust design for the material of the blisk. Most likely this would involve aluminum blades made in separate sections and then placed into the hub through a locking dovetail type mechanism. Another possibility is to make the blisk out of poured steel or molded plastic, which would increase the integrity that is lost due to the layered quality of the rapid prototyping material. Another architecture improvement can take the

form of a more robust stand that can be used for deployment of the ram air turbine when it is finally implemented into its intended aircraft. Final improvements would include a more powerful and more reliable brake, and a lessening of the weight of the RAT via lighter motors and materials.

Design Analysis:

Blade / Motor Initial Design

The two main components that will define the performance of this ram air turbine are the blisk and electric motor. The optimum for a propeller type design occurs at lower advance ratios, so the faster the blade spins the better the efficiency of the design will be. For this reason, the design of the blade and motor used pushed the higher envelope of the allowed rotational speed. So, we designed and expect the RAT to perform at 4500 RPM. A motor was purchased to match this that was rated to operate at 4600 RPM.

The Betz limit was then utilized to find the total amount of energy that could be removed from the air for the designed rotor size and designed wind speed. These results produced a maximum removable amount of energy for a 10 inch diameter rotor at 100 ft/s of 519.6 W. This will be lower due to inefficiencies in our design and the motor so a good estimate we decided to go with was about 300 Watts. Our motor was rated at 324 Watts and 24 Volts. Our design using the FORTRAN code provided by Dr. Hall supplied 302 Watts of power and generated 5.36 Lbf of drag. These constitute our predictions of performance of somewhere near 300 watts of power operating at 4500 RPM at 100 ft/s wind speed.

Finite Element Analysis of Blade/Hub Assembly

To perform more complex analysis of our design, a technique called Finite Element Analysis was employed. The technique calls for a CAD model of the design broken up into thousands of smaller, interconnected pieces which make up a mesh. Finite Element Analysis then performs computations on each of those small pieces and reports back its findings.

Of particular interest in our design is the stress experienced by the blades due to rotation; we wanted to make sure that the blades would not yield or fail while it was in operation. Rotation of the blade/hub assembly is essentially an acceleration of the blades in the radial direction, which in effect produces a force on said blades. Finite Element Analysis allowed for determination of whether or not these forces on the blades were too great without the need for repeated experimentation or extra prototypes.

Also of importance were the resonant frequencies of the system. Since the system was to be driven at varying rotational speeds, the system's response could be amplified and cause undue stresses and deformations if it was driven at certain resonant frequencies; these frequencies are the rate at which the system "wants" to rotate because its resistance to rotation is as small as possible.

This resistance to motion is called the system's mechanical impedance and is composed of two components: a resistive component and a reactive component. The resistive component of the impedance represents various losses as the system translates the force applied on it to motion. The reactive component represents the system's ability to store energy; in this case the ability to store kinetic energy as potential energy. The two components work to determine what the system's response to a force will be; they determine how much of an applied force gets stored and how much of that same applied force gets dissipated as heat or some other loss. Impedance, therefore, is a frequency-dependant property of a system; that is, it is a property of the system, but it is still a function of the frequency the system is driven at. The resonant frequencies of a system are defined as those frequencies which reduce the impedance to its minimum value by reducing the resistive component's value to zero. More specifically, when a system is driven at one of its resonant frequencies, the resistive component of impedance vanishes and the system no longer dissipates any energy, and instead stores all of the vibrational

energy. Finite Element Analysis helped produce what these resonant frequencies were given the design of our blade/hub assembly and the material it was constructed out of.

Stress Analysis

Since most of the load experienced by the blades would come from the rotation of the hub, aerodynamic loads were not included in the FE Analysis. Instead, the hub was given a rotational speed and a pressure under which it would operate (14.7 psi). The restraints of the system were set so that the hub/blade assembly would have no translational motion, only rotation about its axis. The stress analysis results shown below display the blades without any rotational forces (Figure 1a) and the stress and deformations experienced by the blades as they rotate at 4600 rpm or approximately 76.7 Hz (Figure 1b).

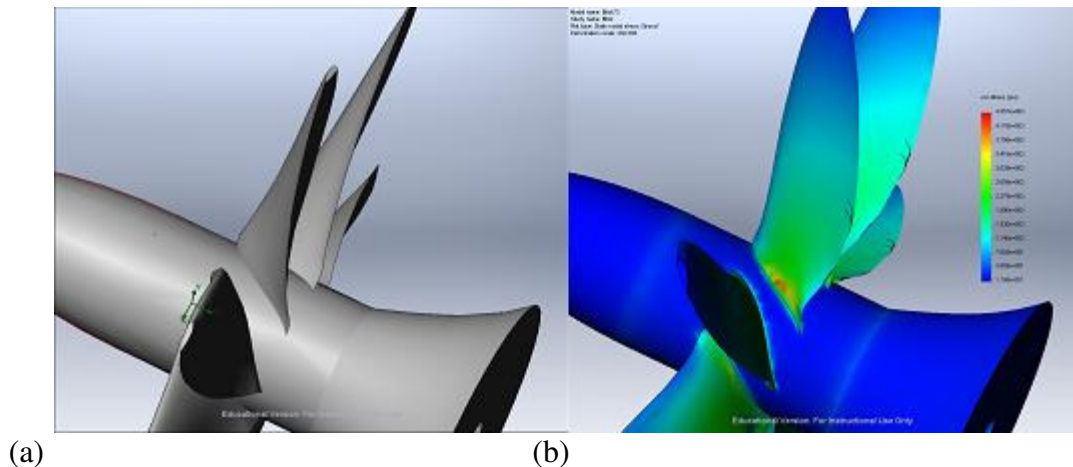


Figure 1: (a) Blades without rotation. (b) Blades rotating at 4600 RPM

The greatest stress occurred along the center of the cord of the blade, near the root where the blade was attached to the hub on the side of the blade that curves into the incoming airflow (in the image above, the blades rotate into the page at the top). The value of this stress was 424 psi, well below the ABS's yield stress of 3200 psi giving a factor of safety of 7.6. As you can see from the image, there was a stress concentration at this location; most of the other parts of the blades experience relatively mild stresses. This was probably due to the sudden cross-sectional area change that occurs here due to the differing geometries of the blades and hub. One can see the same effect (though much milder) in the portion of the hub that flares out a few

inches behind the blades. The table below gives the five highest stresses experienced by the blade/hub assembly.

Stress (psi)	X (in)	Y (in)	Z (in)	Radial Distance(in)
4.24E+02	0.87618	1.0519	-0.06479	1.37
3.93E+02	1.0143	1.0807	0.022058	1.48
3.86E+02	0.94824	1.0802	-0.02108	1.44
3.63E+02	1.0793	1.0777	0.064109	1.53
3.41E+02	0.67271	1.0271	-0.21203	1.25
3.40E+02	0.81129	1.0475	-0.10961	1.33
3.31E+02	1.1422	1.1166	0.10069	1.60

Table 1: Highest stresses experienced by the blade/hub assembly

Note how these stresses occur at around the same x-value (the x-axis is aligned with the hub's axis of rotation) and at approximately the same radial distance from the hub's axis of rotation. This further reinforces the idea of stress concentrations due to area changes introduced above. These results show that the blades were not in any danger of failing in operation.

Frequency Analysis

This portion of the FE Analysis was set up the same way as the stress analysis except that the model was tested at several different rotational speeds. These results were then plotted and a Campbell chart was constructed.

A Campbell chart is constructed out of several mode lines for a system as well as several excitation lines. Since the resonant frequencies of a rotating system are a continuous function of rotational velocity, one can derive a function that describes what the resonant frequencies will be at each rotational velocity. In general resonant frequency increases as the square of rotational velocity of the system. The excitation lines represent how many times a certain frequency needs to be excited per revolution in order to achieve resonance. Therefore, the frequencies where low excitation per revolution (i.e., 1, 2, 3, and 4 per revolution) lines intersects the mode lines are of importance on a Campbell chart; these frequencies are likely to incite a resonance in the system.

The results of the frequency analysis are displayed below. Table 2 shows the resonant frequencies of the first eight modes at ten different rotational velocities.

Rate of Revolution (RPM)										
	0	500	1000	1500	2000	2500	3000	3500	4000	4600
Mode 1	239.49	239.55	239.85	240.19	240.75	241.46	242.33	243.36	244.53	246.69
Mode 2	239.93	239.6	239.9	240.73	241.34	242.12	243.06	244.18	245.45	246.74
Mode 3	240.09	239.61	240.22	241.4	242.96	244.94	247.34	250.15	253.35	258.43
Mode 4	240.2	239.68	240.35	241.58	243.24	245.34	247.75	250.54	253.72	258.83
Mode 5	240.37	240.04	240.44	241.68	243.39	245.36	247.89	250.87	254.26	259.32
Mode 6	240.4	240.07	241.08	241.84	243.48	245.78	248.55	251.79	255.46	261.21
Mode 7	240.65	240.22	241.24	242.41	244.31	246.73	249.65	253.05	256.91	262.55
Mode 8	741.29	740.15	740.23	740.98	741.7	742.62	743.74	745.06	746.57	749.12

Table 2: Resonant Frequencies of Rotating Blade/Hub Assembly. All values are in Hertz unless otherwise specified

The first seven modes are the first bending modes of each of the seven blades in the design (Mode Shape 1.avi included with this document). The eighth mode, which is at a much higher frequency, is the first torsional mode (Mode Shape 8.avi included with this document).

Using the data in Table 2, the Campbell plot shown below was constructed.

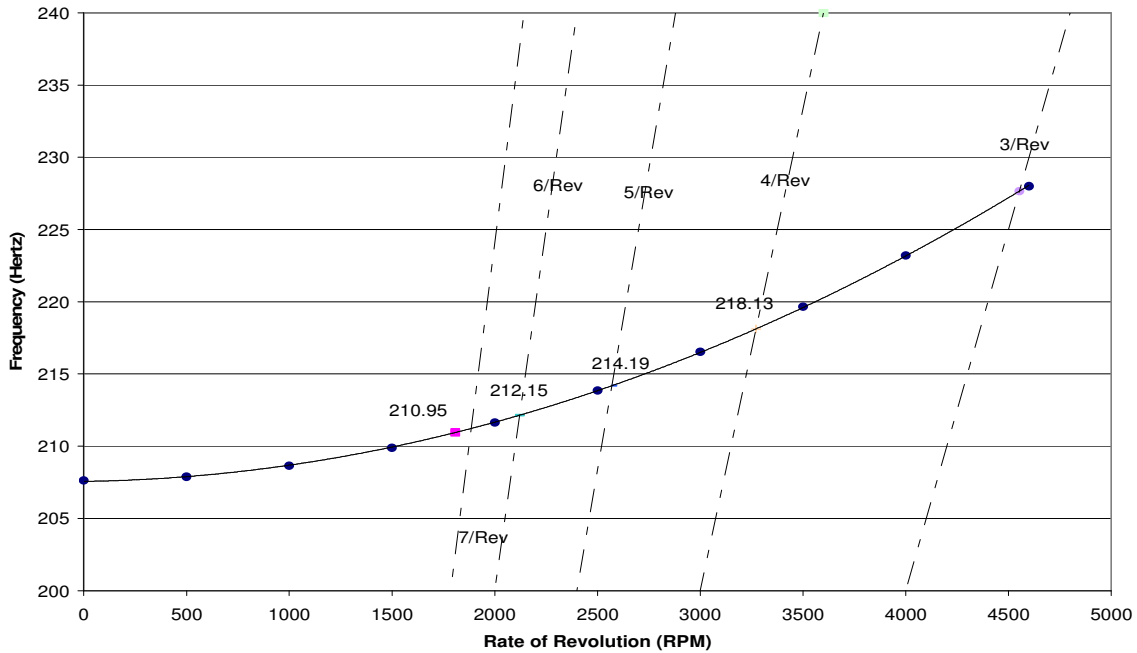


Figure 2: Campbell Chart of First Seven Modes

The graph indicates that resonance is likely to occur in the frequency range of 3657.82 RPM to 3834.4 RPM (4/Rev Line), as well as the 2906.4 RPM to 3994.4 RPM range (5/Rev Line). Therefore, testing as well as operation in these ranges should be avoided.

In addition to the frequency analysis done on the entire blade/hub assembly, a frequency analysis was performed on a single blade to obtain the decoupled resonance frequencies of the blade design. The table below shows the resonant frequencies at the same rotational velocities tested for the blade/hub assembly.

	Rate of Revolution (RPM)									
	0	500	1000	1500	2000	2500	3000	3500	4000	4600
Mode 1	207.63	207.88	208.64	209.89	211.64	213.86	216.54	219.66	223.21	228
Mode 2	657.43	657.56	657.94	658.58	659.47	660.61	662.01	663.65	665.53	668.1
Mode 3	1247.4	1247.5	1247.8	1248.3	1249.1	1250	1251.2	1252.5	1254.1	1256.1
Mode 4	1270.5	1270.6	1271	1271.6	1272.5	1273.6	1275	1276.7	1278.6	1281.3
Mode 5	2302.4	2302.6	2303.2	2304.1	2305.5	2307.3	2309.5	2312	2315	2319
Mode 6	3186.2	3186.3	3186.7	3187.2	3188	3189	3190.2	3191.7	3193.3	3195.6
Mode 7	3398.6	3398.7	3399	3399.5	3400.3	3401.3	3402.4	3403.8	3405.4	3407.6
Mode 8	3971	3971	3971	3971.1	3971.2	3971.4	3971.6	3971.8	3972.1	3972.4

Table 3: Resonant Frequencies of Rotating Blade. All values are in Hertz unless otherwise specified

The values for the resonant frequencies associated with the first mode of the blade are lower than that for the blade/hub assembly, but this difference decreases somewhat as the rate of revolution increases. The resonant frequencies for the higher modes of the blade are substantially greater than those for the blade/hub assembly as well as greater than the resonant frequencies for the first mode. Because of this, only the Campbell chart for the first mode is of any real importance; only the higher excitation lines cross the second and higher modes in the range of interest. The resonant frequencies of interest occur at rotational velocities of 2570.3 RPM (5/Rev), 3271.89 RPM (4/Rev) and 4553.3 RPM (3/Rev).

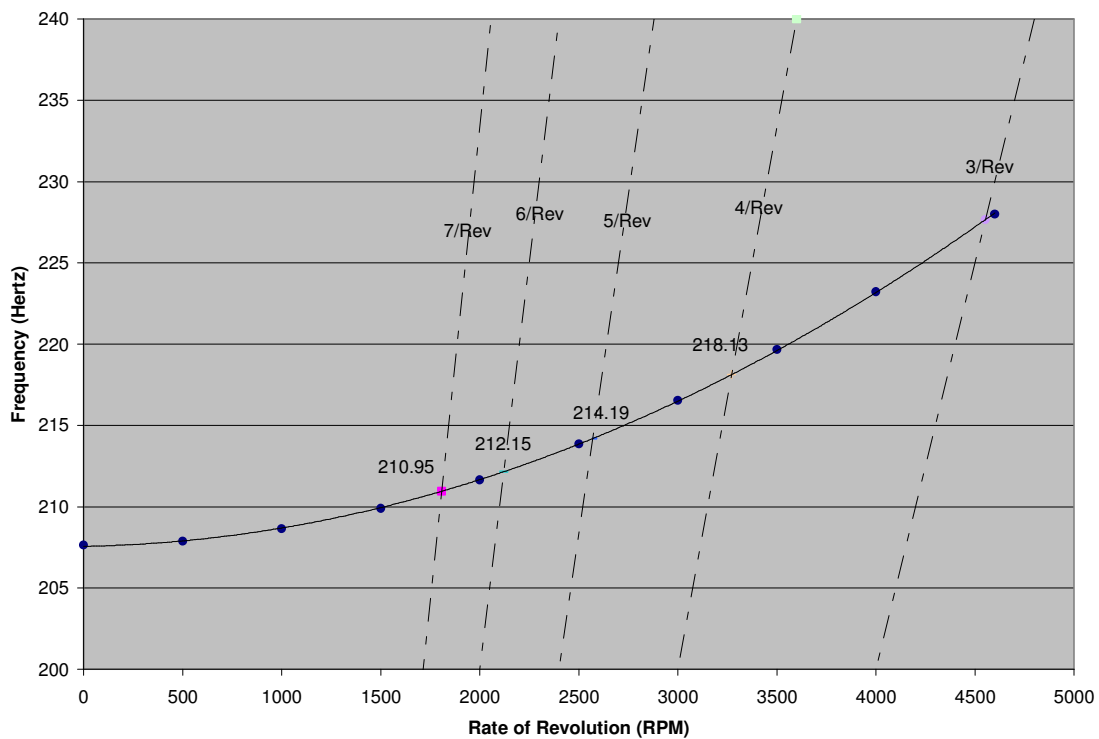


Figure 3: Campbell Chart of First Mode for the Blade design only.

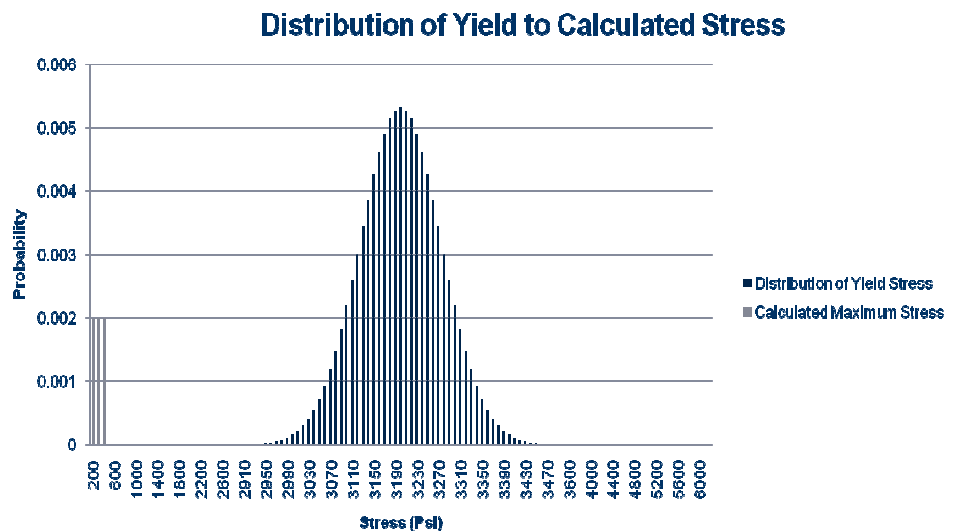
Also included with this document are the mode shapes of the first three modes (Blade Mode Shape 1.avi, Blade Mode Shape 2.avi, and Blade Mode Shape 3.avi), and an excel file containing all the data used during this frequency analysis (frequency.xls).

Statistical Analysis:

As blade or blisk failure on a RAT would mean disaster for the plane on which it was being used, we decided to statistically analyze the maximum stresses placed on the blisk and compare them to a statistical representation of the yield stress of the ABS plastic used to construct the blisk. Since the yield stress was calculated using FE analysis in SolidWorks and not through a deterministic equation, we were unable to perform a complete Monte Carlo analysis. Instead, we assigned a range of operational speeds at which the RAT could be expected to perform and calculated the maximum stress that was placed on the blisk at each speed. It was determined that the maximum stress is linearly related to the rotational speed at which it is spinning by the following equation: $Stress = .1839 * RPM - 323.74$.

We then obtained yield stress data from Stratasys, the company that makes the rapid prototyping machine used in this project. The data obtained was for a different model, but it uses the same material and layer thickness. According to this data, the yield stress of the material is 3200 psi and the standard deviation is approximately 75 psi. Using this data we constructed a normal distribution of yield stresses for the ABS material that was used. When we compared the stress data from the FE analysis to the yield stress curve, we found that the highest stress that the blisk would receive under operating conditions (at 4900 RPM) is 357.74 psi which is lower than the mean yield stress by a factor of 8.94. Extrapolating based on the previously mentioned linear relationship shows us that the blisk will fail 5% of the time at 24865 RPM, which is much higher than the rated speed of the motor. This indicates that it is incredibly unlikely that the blisk would fail before other parts of the RAT.

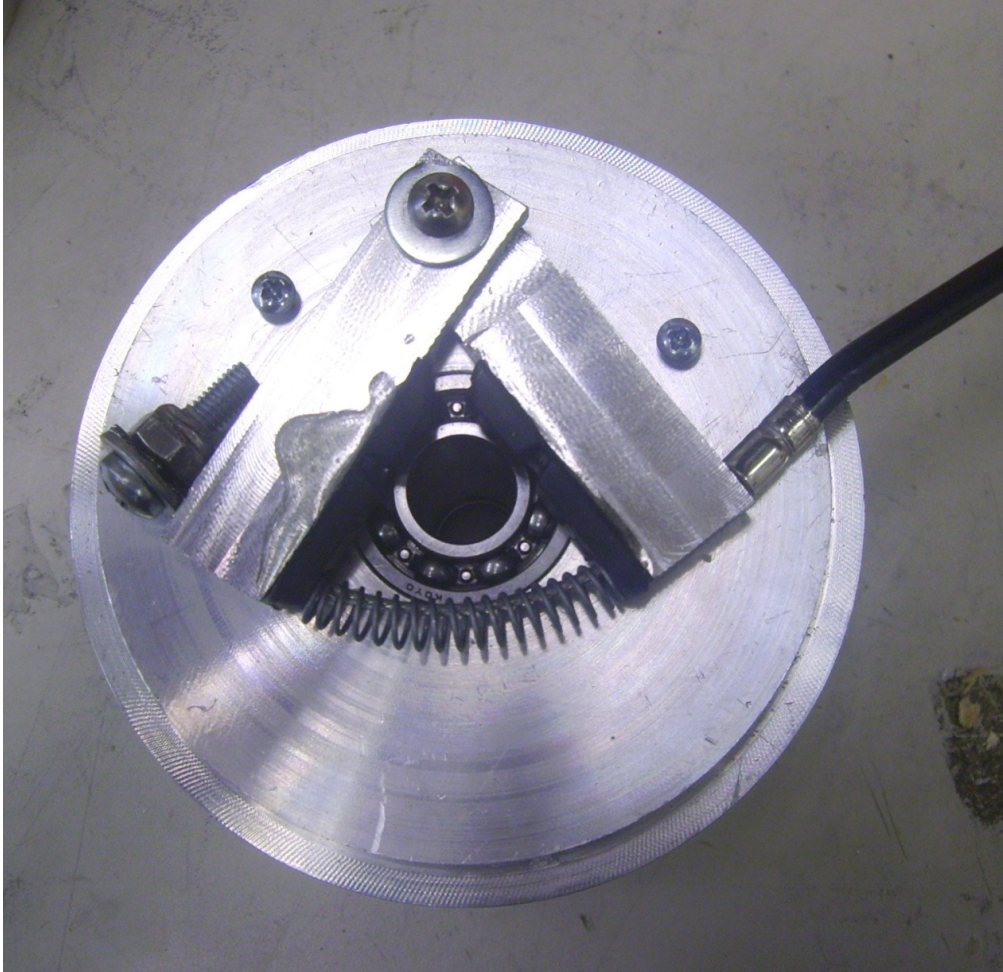
This graph illustrates that the operating stresses (represented by the bars on the left) are far outside the normal distribution of yield stresses in the middle.



Prototype:

Design

The design of the prototype followed exactly the architecture that was drawn up previously in this report. It was feasible with the time and materials provided to make and test every aspect of the project that was laid out for the product. Precautions were taken to make the prototype have all the important features discussed for this project. The prototype needed to be powerful, cheap, light, low drag, simple, dynamically stable and durable. Each important component outlined in the architecture required careful analysis and shopping to find the proper materials that would withstand the testing and perform as required. Aluminum was the most commonly used component because of its weight to strength ratio and machinability. The bearing housing was designed to hold the bearings sturdily in place and support the RAT. The shaft selection is one of the steps that shaped the majority of the rest of the design. In order to have a rotor-dynamically stable prototype it was desired to have as thick of a shaft as possible. This was obtained with the ½” diameter shaft, which determined the type of bearings to be used, and the coupling necessary. The bearing selection was also influenced heavily by the speed of operation, so a great deal overkill was put into their selection, so bearings rated for 30,000 RPM were selected. The motor size set the size necessary for the casing and the grooves that needed to be added to it, and some modifications to the coupling. The two parts that were made in the most ad-hoc manner were the casing and the brake. The brake idea developed in a fairly organic manner and was later reverse engineered. The casing was then edited and machined to make sure all the parts fit together. An up close picture of the brake assembly is shown below.



The brake was designed as a clamping hinge-like device that would provide extra damping to the spinning shaft on command. The brake was not very powerful but provided some additional stopping power.

Fabrication

The fabrication process for this prototype relied primarily on two methods of machining. The rapid prototype printer was used extensively to make two models of the blisk, one as a structural stand in, and the other the final product. The rapid prototyper was also used to make the end cap for the RAT. The end cap was made in a honeycomb manner to save on weight and printing time while the blisk was made solid to provide the necessary strength, however it was

very time intensive to make. The other main manufacturing process was through extensive utilization of the machining skills of John Goodfellow in the machine shop. The bearing housing with its press fits and large tapped hole required more machining experience and tools than we had access to. Also the casing grooves and slots for wiring were complicated to machine. The other parts were made fairly easily using basic milling and lathing, and the assembly came together well. The end of fabrication involved coating the blisk with several layers of epoxy paint to ensure that the fibrous makeup of the blisk would not unravel and to provide some extra strength. Some interesting testing for fabrication also had to be done. The disk that is used to attach the blisk to the rotor shaft was attached to the blisk via countersunk wood screws, so a test to see how the rapid prototype material holds up to connection via wood screws was examined. The success of this investigation led to a simple assembly of the blisk to the shaft. Several modifications also had to be made to ordered parts. The faceplate of the motor that arrived was originally square but this had to be cut to be circular on the band saw. Also the coupling was not quite large enough for the motor shaft, so it had to be expanded using a drill bit and the lathe.

Test Plan

The testing for our prototype had many stages. The first testing done was the ping testing which was discussed in the design analysis of the blade vibration. This testing was conducted by setting up the blisk attached to the shaft and placed in a vice to keep it fixed. It was then hit with a small hammer and an accelerometer placed on the tapped blade was then excited and provided the natural frequencies of the blisk. These were then compared to the frequency analysis from SolidWorks and used to skew the values on the Campbell chart to obtain a better approximation of vibration instabilities.

The next section of testing was done through spin testing. This involved spinning the blisk so that it experienced the same forces while removed from the wind tunnel. This was done first in the drill press with only a shorter shaft and only the blisk. This was tested up to 2900 RPM. The second stage of spin testing was done utilizing a power source and lawn mower battery to attain the rated voltage of 24 volts. This provided a spin test up to 3200 RPM which set the speed at which the wind tunnel testing would be limited to. Both tests were carried out in safe manners with metal and Plexiglas shielding, and plans for ways to turn off the tests if things go wrong.

The final stage of testing takes place as wind tunnel testing where actual wind is blown over the RAT so that it generates electricity. The setup of this portion of the experiment is described below.

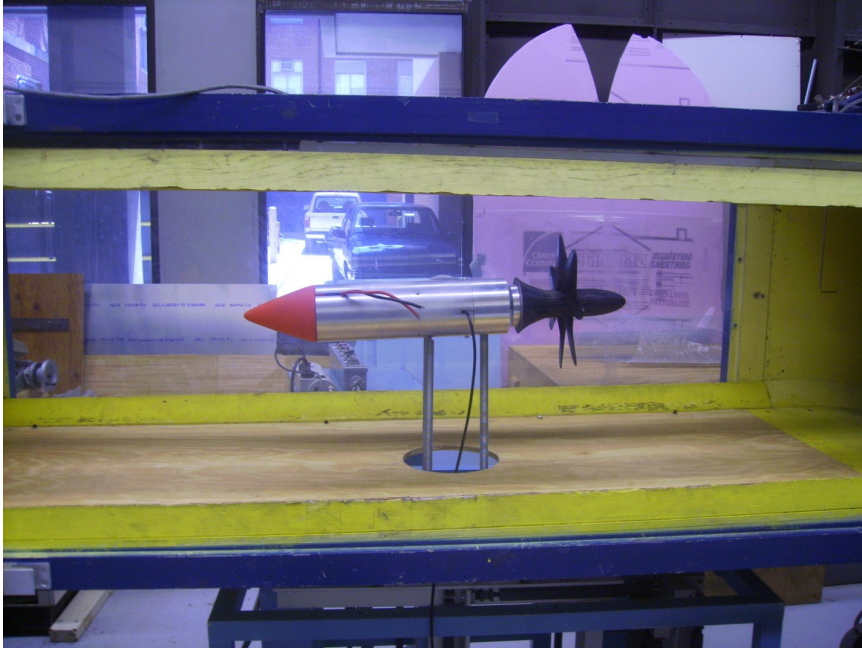
Data to record:

- Wind Velocity – Pressure difference from Pitot tube.
- Drag Force – Using load cell.
- Rotor Speed – Tachometer on Laboratory Stand.
- Power Output – Voltage and current supplied to load.

Method:

1. Assemble RAT in wind tunnel fixing the RAT to the load cell.
2. Turn on wind tunnel and slowly increase speed.
3. Take readings at regular intervals for all of the above listed measurements.
4. RAT will be tested Up to 3200 RPM, or 100ft/s wind speed, whichever occurs first.
5. If RAT still appears stable, then higher wind speeds will be explored.

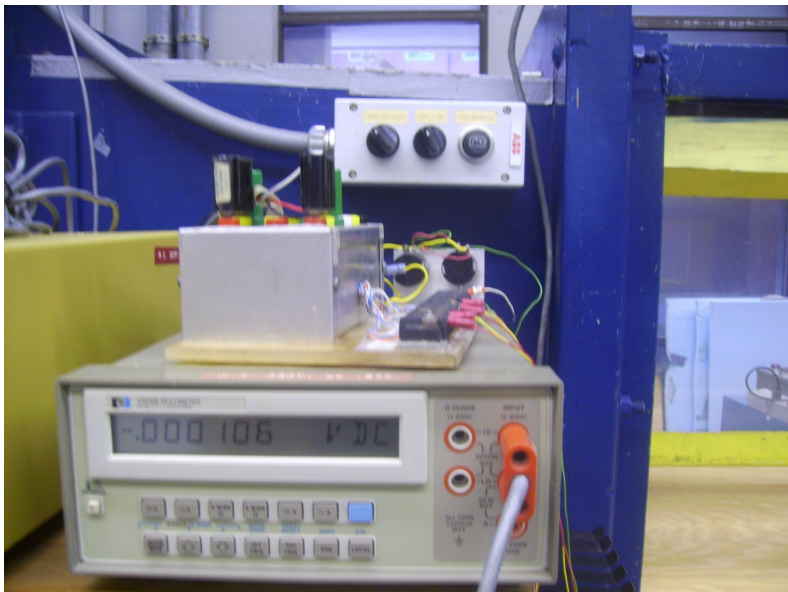
This process was the pre-test, test plan and proved fairly close to the actual testing. Documentation of the testing setup is shown below and in the instrumentation section.



RAT Placement in Wind Tunnel

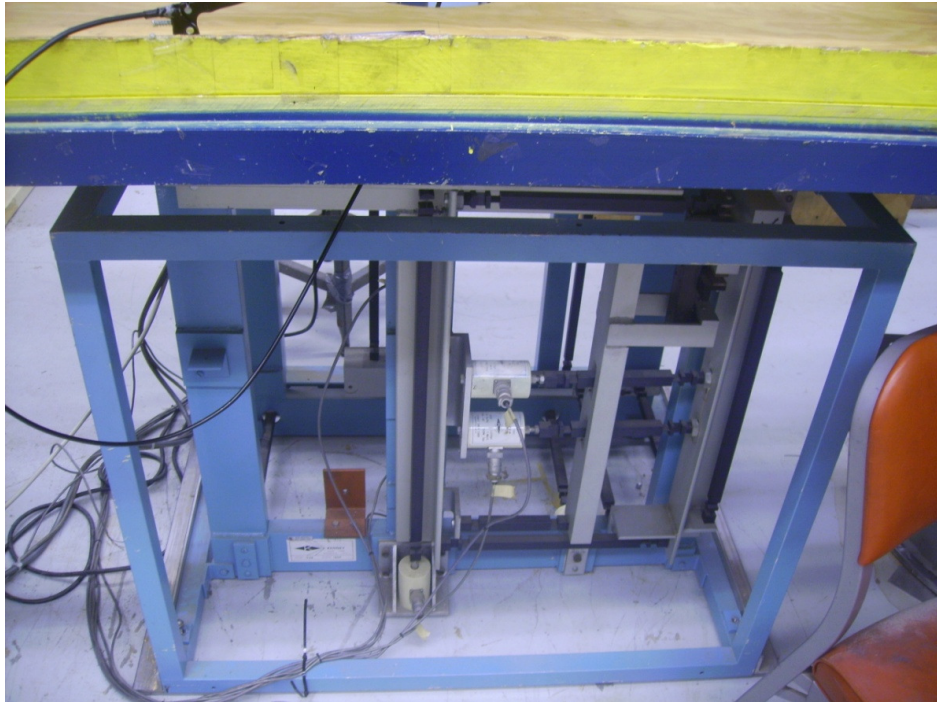
Instrumentation

The main instruments used in this testing are the Pitot Tube, the mechanical load cell, the resistor bank with attached voltmeter and ammeter, and the tachometer.

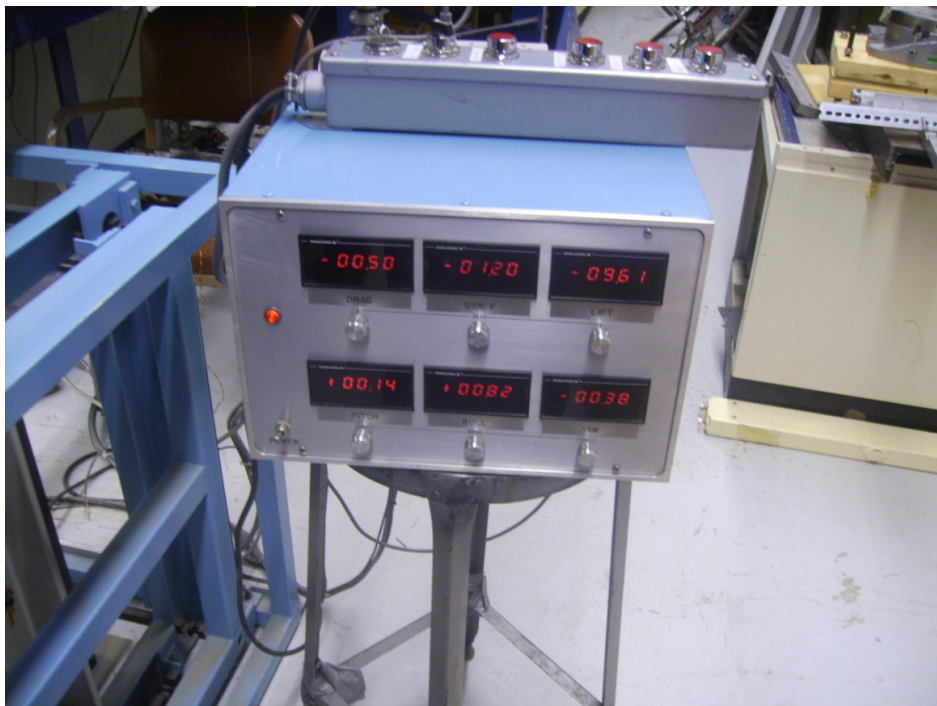


Wind tunnel Pitot tube reading

This device measured the voltage caused by the Pitot tube readings. This was then compared against a calibration chart to measure the velocities seen in the wind tunnel.

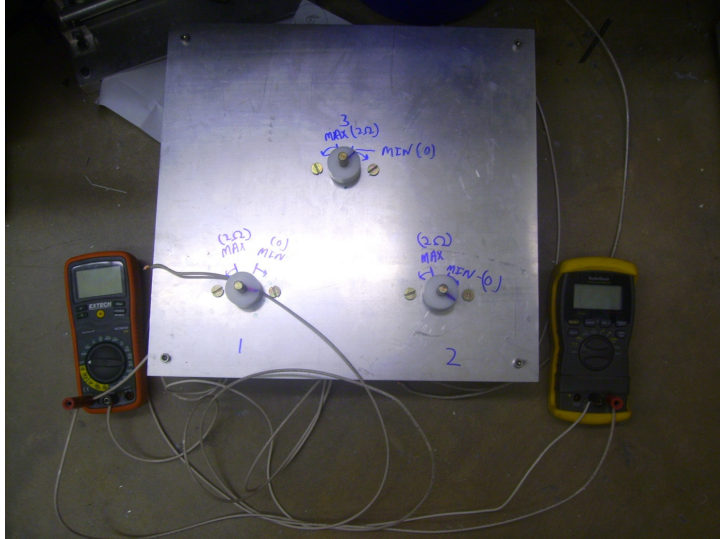


Aerodynamic forces Load Cell



Load cell display

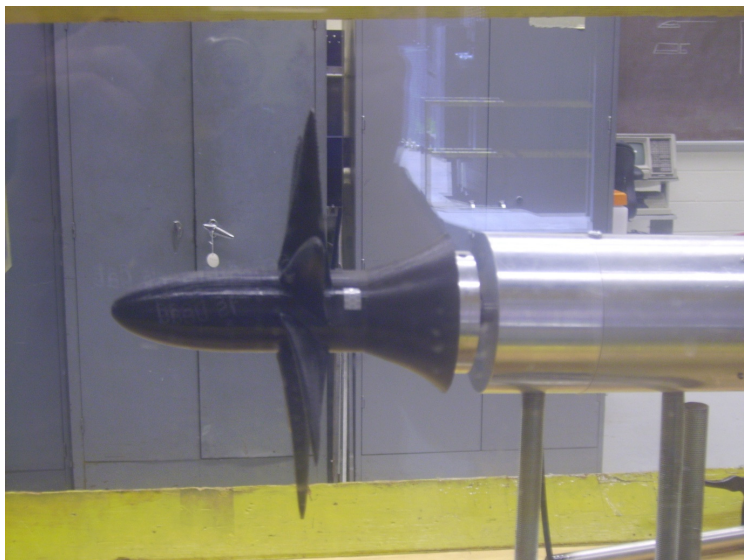
The above two pictures display the measurement system used to record the drag.



Resistor bank, Voltmeter and Ammeter

The electric load cell allowed for the variance of the load across the Turbine and provided the measurements of the voltage and current going through the resistors. These two measurements were then used to calculate the power dissipated in the load and thus the power generated by the RAT.

The tachometer was the final piece of testing equipment used which involved readings of the frequency that a shiny strip crossed the laser emitted by the tachometer. This was securely mounted on a vice and aimed at the strip placed on the blisk shown on the next page.



Measurement Site for Tachometer

Results

The results of the testing are shown below in the table. It includes all the manually collected results including the calculated velocity which was gathered from the wind tunnel calibration charts. The power generated for each test point is also calculated and shows the success of the prototype.

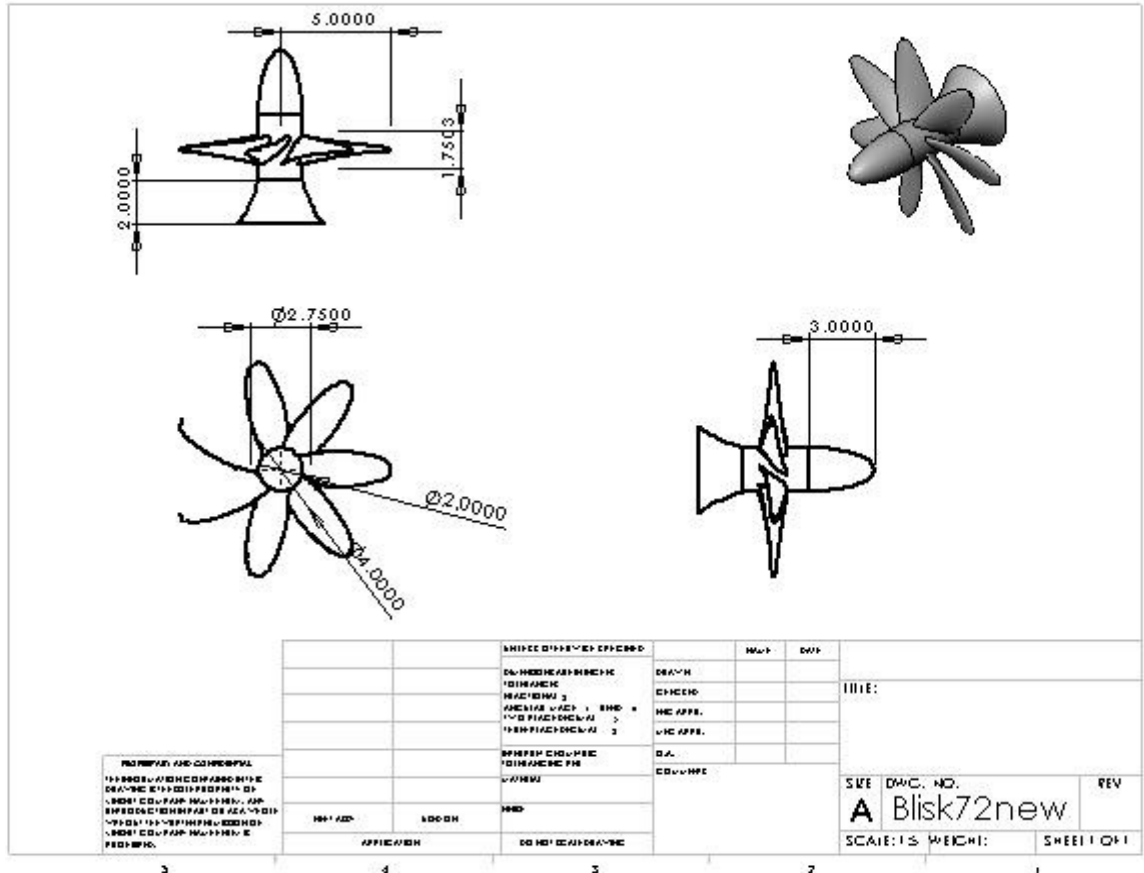
RAT White Team Prototype Test							
Pitot Voltage (V)	Velocity (ft/s)	Rotor Speed (rpm)	Drag reading	Vout (V)	Current (amps)	Resistance (ohm)	Power (Watt)
0.1175	50	2790	1.85	14.82	2.53	5.85770751	37.4946
0.1175	50	2440	1.9	12.6	3.18	3.962264151	40.068
0.1175	50	2160	1.97	10.91	3.69	2.956639566	40.2579
0.185	66	3100	3.1	15.94	5.32	2.996240602	84.8008
0.185	66	2800	3.15	14.2	5.94	2.390572391	84.348
0.185	66	2380	3.2	11.55	6.8	1.698529412	78.54
0.257	78.1	3100	4.4	15.22	9	1.691111111	136.98
0.28	81.5	3100	4.9	14.9	10.26	1.452241715	152.874
0.335	88.9	3100	5.85	14.5	12.83	1.130163679	186.035
0.538	112.533	3100	9.2	14.1	19.8	0.712121212	279.18

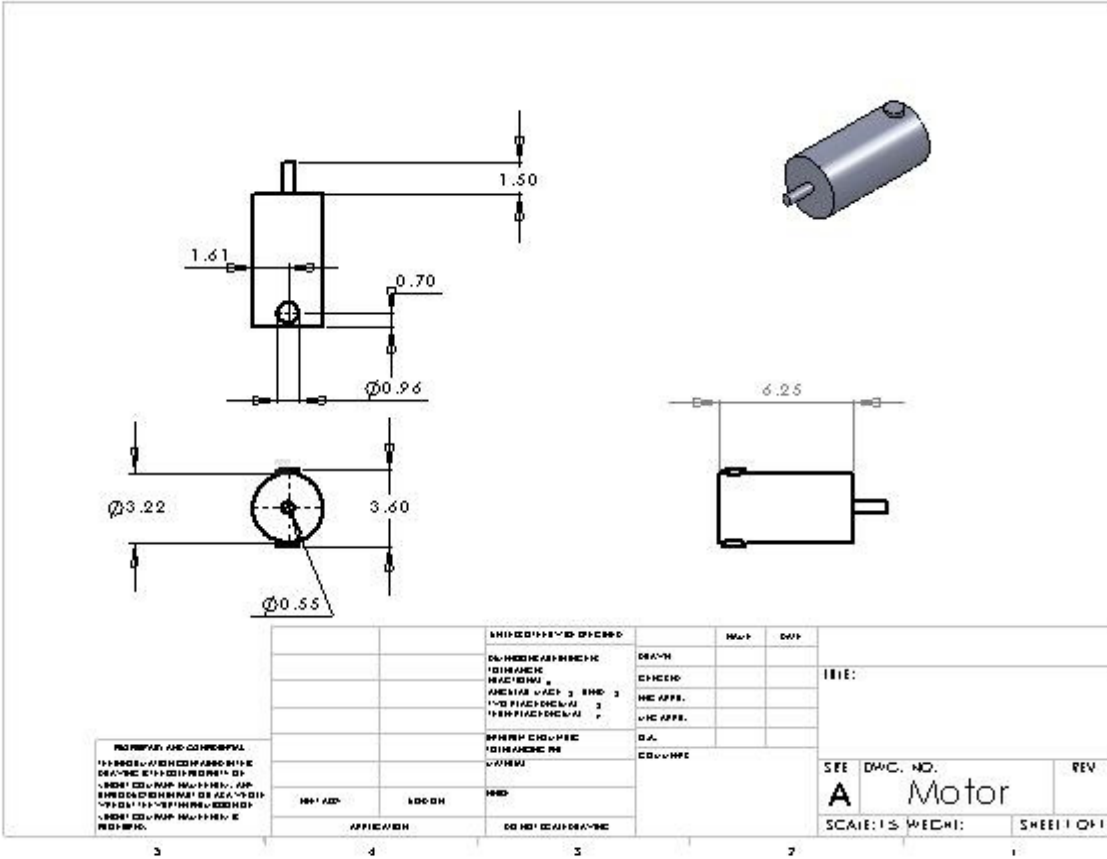
These results show the success of our design. The design power output was approached at only slightly above the designed velocity, and at drastically slower rotational speeds due to the cap imposed by the spin tests. Other observational results which are important to note are the fact that there seem to have been a couple rotordynamic problem areas at 2500 RPM and 2900 RPM. In order to avoid these issues the testing passed through these frequencies quickly. Our optimal performance resistance at 50 ft/s was recorded at approximately 3 ohms but at higher air speeds we approached our rotational speed cap. Our performance in this testing could have been improved if allowed to explore higher spin speeds, but safety limited this.

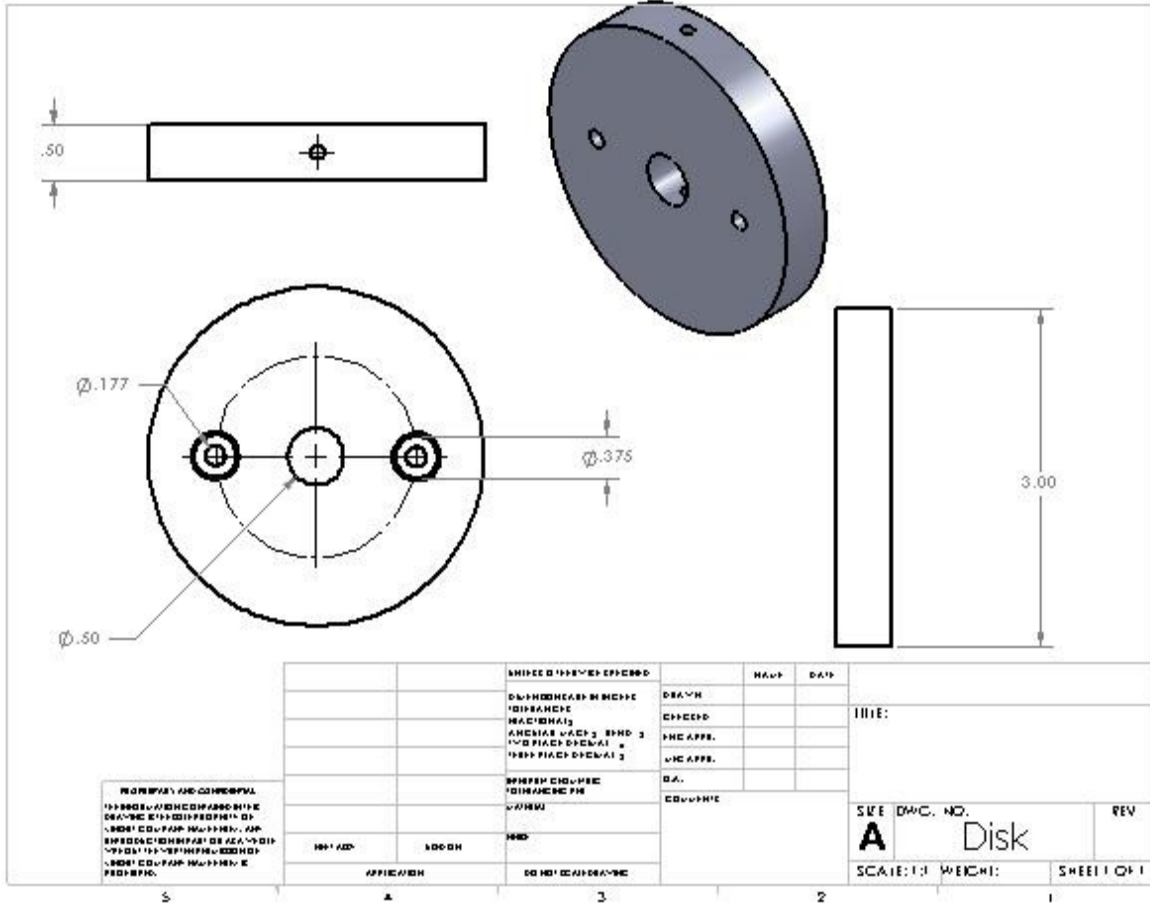
Conclusions:

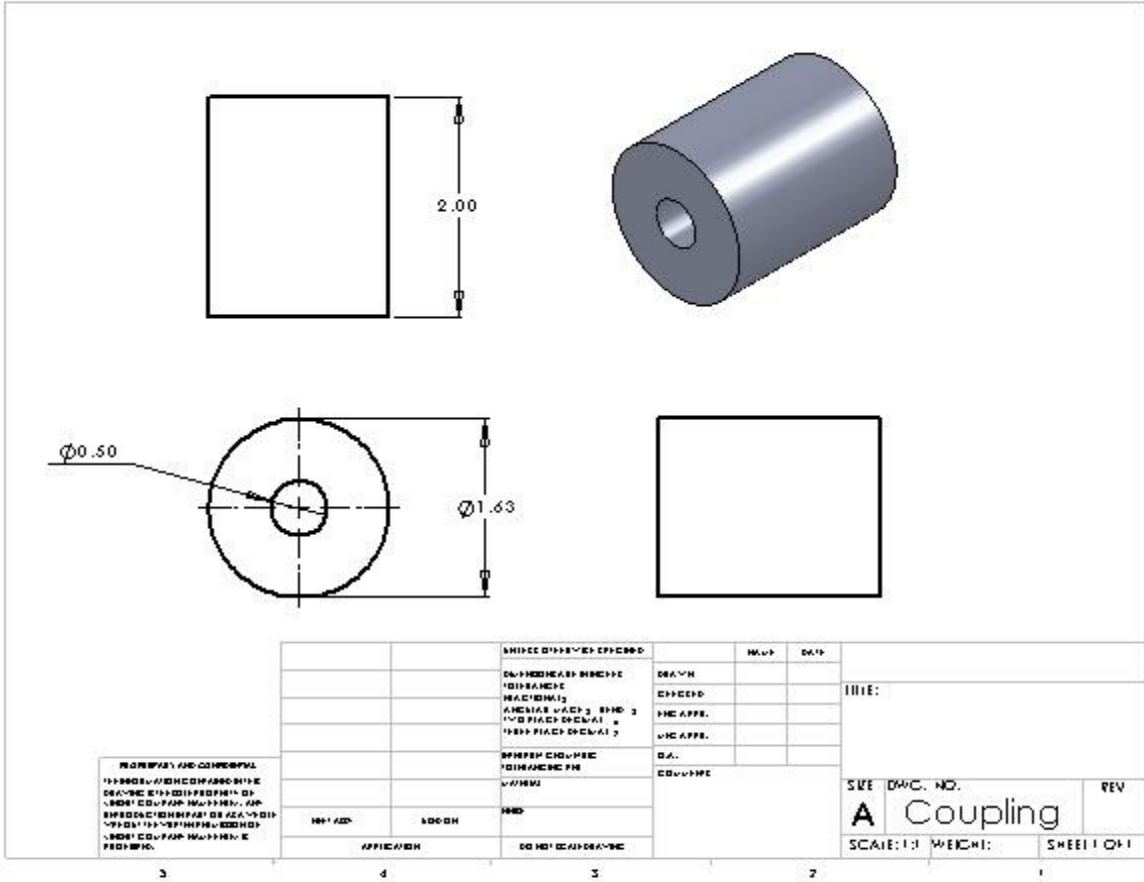
This project proved very successful at designing, analyzing, building and testing a Ram Air Turbine that could be used to power auxiliary systems or emergency systems on a small aircraft. Our data shows that a reverse propeller type of design with many blades that are designed to have an optimum gamma distribution proved very effective as a ram air turbine at low air speeds. There are many improvements already discussed including weight reductions, stand improvements and increased robustness. Our experimentation also shows that future analysis of such a project requires more analysis of rotordynamic activities. Also if this type of ram air turbine were to be commercially sold, it would require much more analysis of other components, from the bearings to the shaft. Also the construction of the support system for the motor should be more secured. For all these possible improvements, this project is still successful in that it delivered on all the design requirements set forth for the project. It has also met all the self imposed requirements listed in the product design specifications. This product was within budget, powerful, and sturdy in the testing environment, and has proven to be a very fulfilling and educational engineering design project.

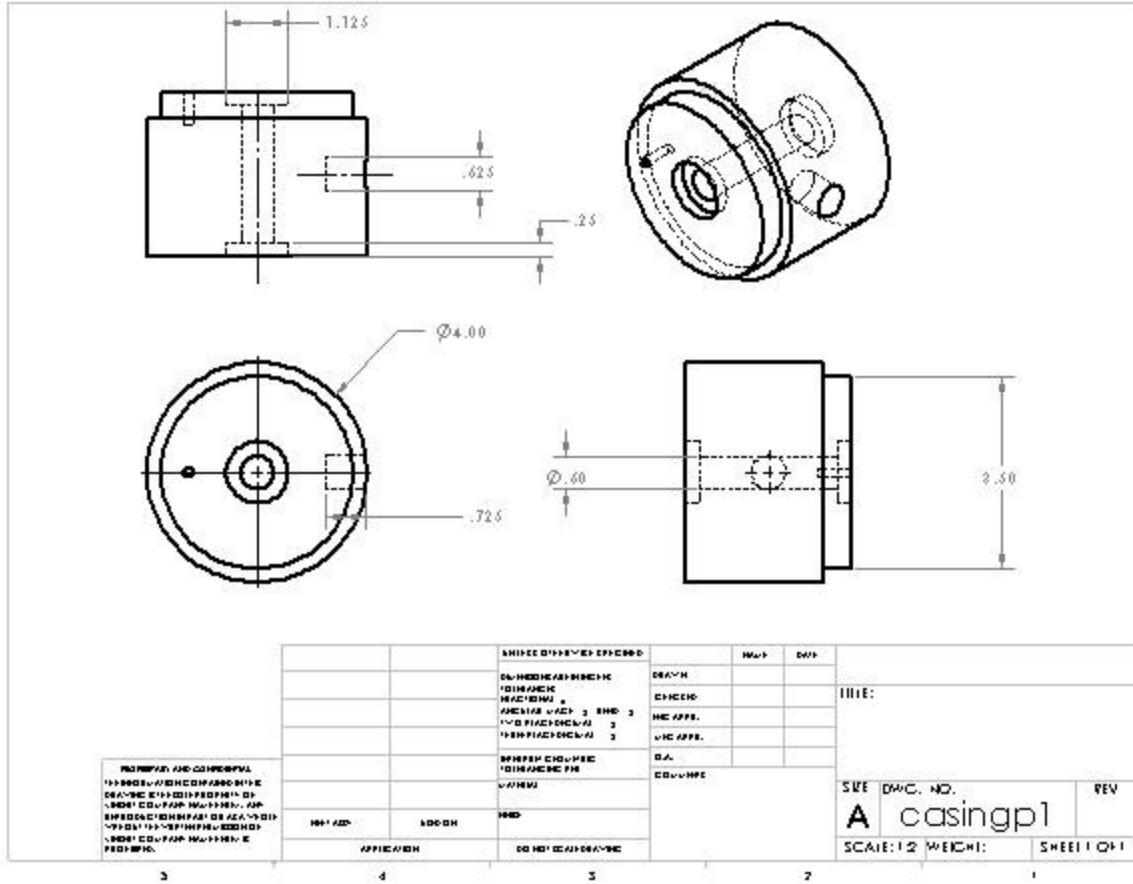
Appendix A: Solid Works Drawings

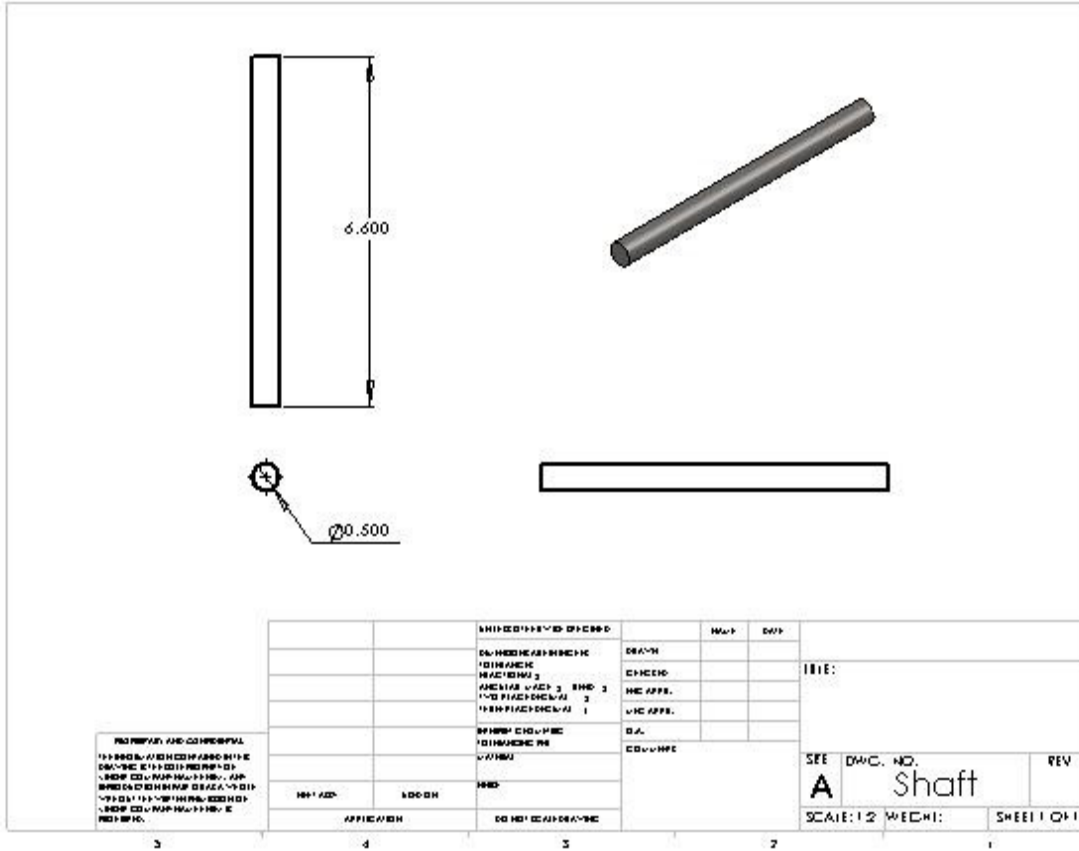


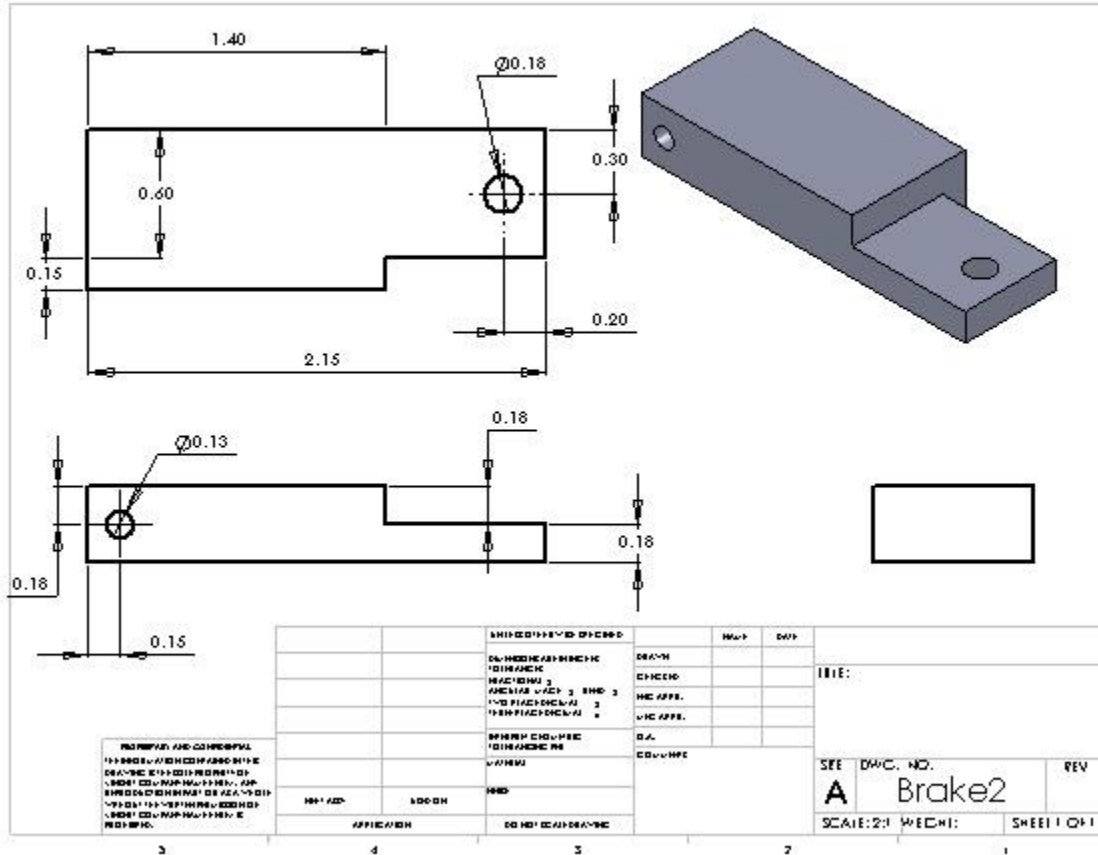


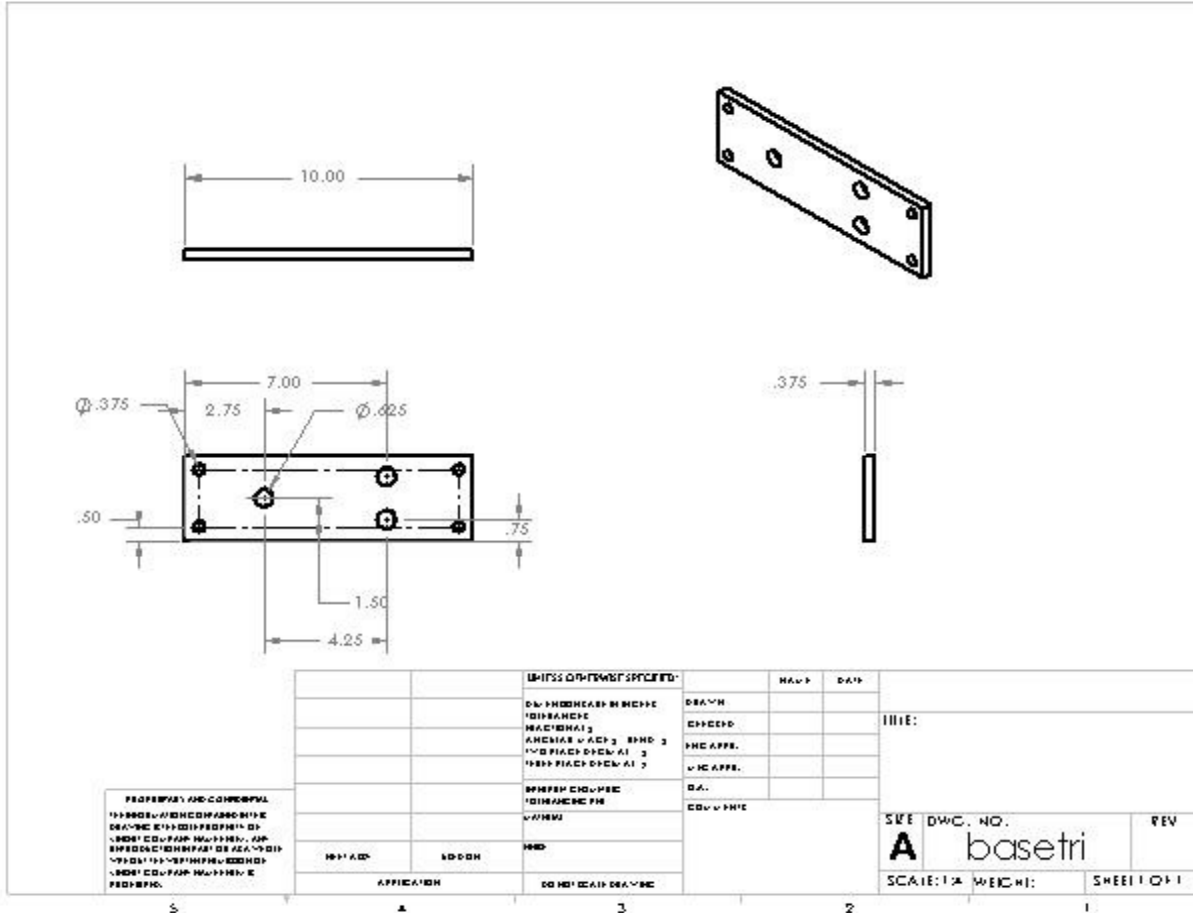












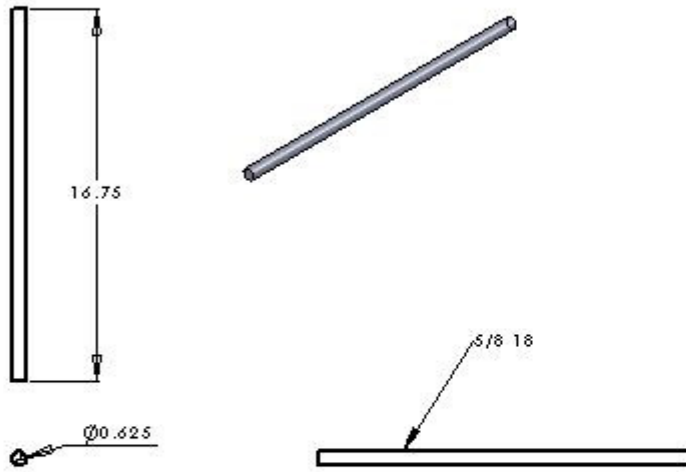
5

4

3

2

1

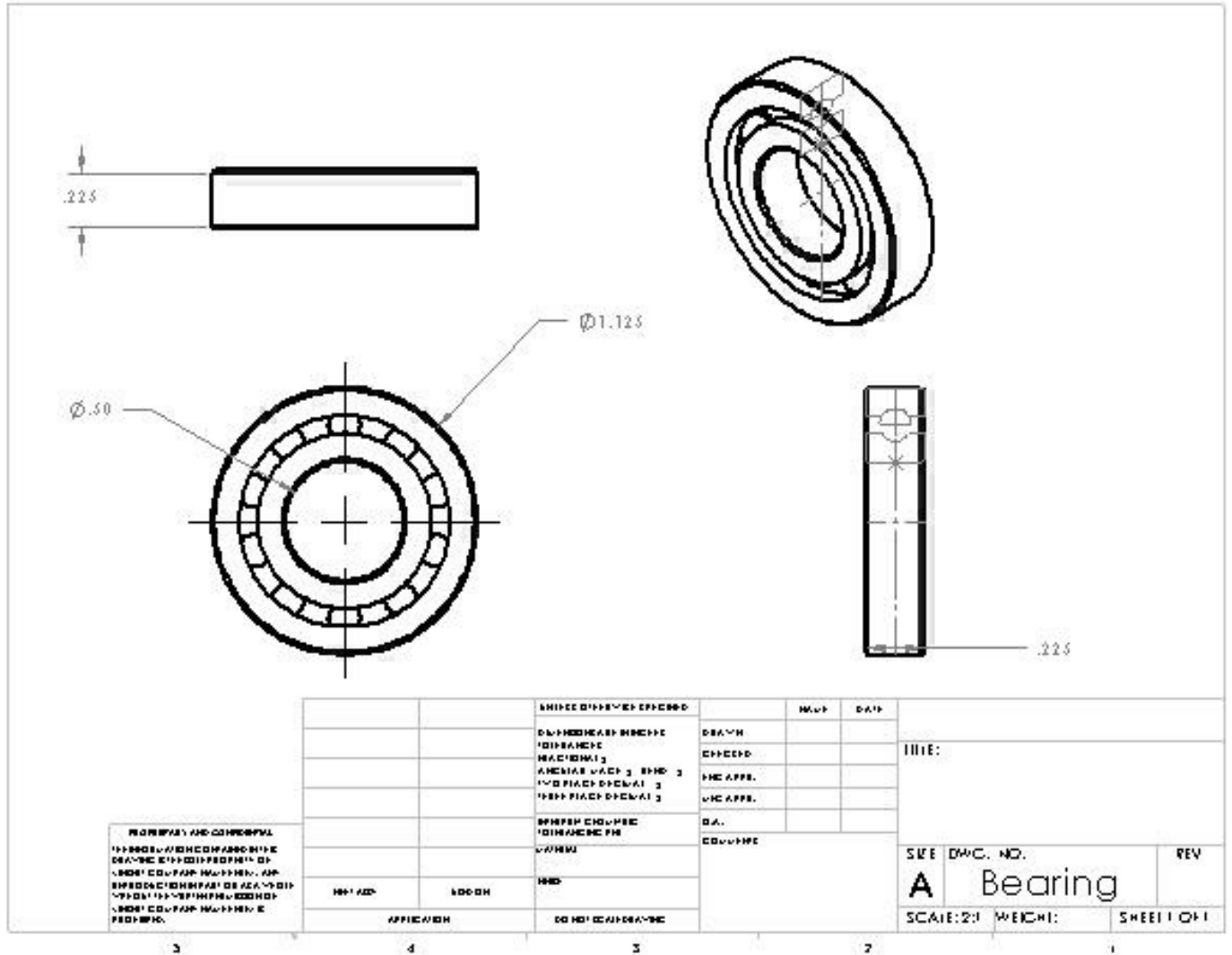


FOR REVISIONS AND COMMENTS
 TO BE MADE BY THE ENGINEER
 DRAWING ENGINEER'S SIGNATURE
 DATE
 PROJECT NO.
 SHEET NO.
 OF SHEETS

NO.	DATE	DESCRIPTION	BY	CHKD.

TITLE:		
SIZE	DWG. NO.	REV
A	CylBase	
SCALE: 1:3 (WEIGHT)		SHEET 1 OF 1

3 4 5 2 1



Appendix B: Design Specifications

Performance

- 1.1 Must be able to generate full power deliverable from generator. (Generator specs TBD)
- 1.2 Must be able to brake shaft to a stop when air is flowing at 100ft/sec.
- 1.3 Should be fairly light, less than 50 pounds.
- 1.4 Must not exceed an envelope of 20 in. high, 20 in. across and 30 in. long.
- 1.5 Must be able to be attached to Duke Wind tunnel Drag Measurement System.
- 1.6 Should have a high efficiency.
- 1.7 Must be able to produce power in a velocity range from 10ft/s to 100 ft/s.
- 1.8 Should have a profile that would be easily stowed in aircraft hull.

2.0 Environment

- 2.1 Should be useable in humid low pressure environments, such as the lower Troposphere.
- 2.2 Temperature ranges:
 - 20° F to 130°F
- 2.3 Corrosion Resistant.

3.0 Product Life Span

- 3.1 Approximately 10 years, with service for the next 5 years.

4.0 Life in Service

- 4.1 Should be able to operate for 30 minutes uninterrupted for up to five uses per year, for ten years.

5.0 Shelf Life

- 5.1 May be stored for distribution for several months.

6.0 Target Costs

- 6.1 Target manufacturing cost is 1000 dollars.

7.0 Quantity

- 7.1 One Prototype unit.

8.0 Maintenance

- 8.1 Should require minimal to no maintenance.
- 8.2 Frequent diagnostic checks should be made to certify that the product is in working order.

9.0 Marketing

9.1 None. Will be a prototype for research purposes.

10.0 Packaging

10.1 None

11.0 Size and Weight Restrictions

11.1 See Section 1.

12.0 Shipping

12.1 Product will be shipped by air.

13.0 Manufacturing Processes

13.1 Majority of custom designed parts will be machined and assembled in house.

13.2 Electric Generator, bearings, and brakes will be supplied. Vendors are TBD.

14.0 Aesthetics

14.1 Form will be guided by function.

14.2 Form will be made more attractive for grading criteria as well.

15.0 Ergonomics

15.1 Will be fairly automated.

16.0 Customer Requirements

See *Marketing*

17.0 Competition

17.1 The RAT will be competing against the Blue Team.

18.0 Quality and Reliability

18.1 Reliability is of the utmost importance due to the emergency use nature.

19.0 Standards and Specifications

19.1 Will attempt to comply with all FAA standards, especially FAA Section 121.305K.

20.0 Company Constraints

20.1 None

21.0 Processes

21.1 Will conform to standard limits and fits.

22.0 Safety

22.1 No failures should result in safety violations.

22.2 Except if the plane like crashes.

23.0 Testing

23.1 Will be thoroughly tested in Duke wind tunnel.

24.0 Legal

24.0 NMP

25.0 Installation

25.1 Must be installed to Duke wind tunnel.

25.2 Must be able to be fit into aircraft.

26.0 Documentation

26.1 Tech specification and User manual will be included

27.0 Disposal

27.1 Aircraft graveyard, except for reusable parts.

Appendix C: Hamilton Sundstrand Survey Responses

Hello RAT Team,

Good luck with you project, it should be quite interesting and challenging.

Here are my rankings (5 is most important, 1 is least important) and answers.

a) Power output: 5

When an aircraft is gliding to a landing with the RAT deployed, it must be able to execute certain maneuvers at minimum airspeed near the runway. Consequently the ability to deliver the required power is critical.

b) Power output consistency: 2

Power output consistency is not important. Being able to produce the critical amount of power at the minimum airspeed as described above is critical. While "power consistency" is not important, for a real RAT speed control is important. The rat speed has to be controlled so that at high airspeeds the RAT does not destroy itself.

c) Induced drag: 1

For a RAT deployed drag is the least important of the factors you listed. Drag is generally just a concern for the structural elements of the RAT during deployments.

d) Ability to be stowed: 5

The reason induced drag is not important is because the RAT is stowed almost all the time.

e) Weight: 4

Weight is always important with any aircraft system. In particular, aircraft manufacturers don't like to carry around a lot of weight for an emergency device.

f) Deployed Size: 3

Deployed size is somewhat important as it relates to the amount of room inside the aircraft required to stow the RAT. It is generally a challenge to find much room for a RAT in plane.

g) Lifetime: 2

Since a rat is primarily an emergency device, life is not the most important design constraint.

h) Cost: 3

Cost is important but weight and performance tend to be more important.

2) What is the preferred orientation of the RAT? (Hung underneath, deployed to the side, etc...)

RATs are generally deployed in a vertical orientation. In this way gravity can assist with the deployment.

3) What is the preferred power generation for an aircraft traveling under 100 feet per second?

I am not sure what is meant by this question but I will take a shot at it. 100 ft/sec corresponds to about 59 Knots of airspeed. This speed is below the landing airspeed of all of the airplanes where a RAT would be needed (generally jet aircraft). However, 100 ft/sec is a good choice of a senior design project because the energy in the air stream is limited and consequently it is safer to try out your designs than at higher airspeeds.

4) Any further comments or needs would be much appreciated.

A single semester project to design, build and test RAT is quite ambitious. Stay focused on your key objectives and don't get sidetracked....

Regards, Dave Bannon

RAT Engineering

Hamilton Sundstrand Electric Systems

Mechanical Technologies Engineering

815-226-7226