Modular Composite Outrigger for Stratospheric Balloon Experiment

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Modular Composite Outrigger for Stratospheric Balloon Experiment

A senior design project submitted in partial fulfillment of the requirements for the degree of Bachelor of Science

at

HARVARD UNIVERSITY

by

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Harvard University John A. Paulson School of Engineering and Applied Sciences
Cambridge, MA
Friday, April 5, 2019
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Abstract

Mitigating climate risks posed by excess greenhouse gases (GHGs) in the atmosphere is known as solar radiation management (SRM). One such SRM method involves the addition of aerosols to the Earth’s atmosphere to potentially mitigate ozone depletion and partially offset the positive resultant radiative forcing, as explored by the Stratospheric Controlled Perturbation Experiment (SCoPEx) research team. Safely tracking the impact of aerosol loading on a large scale involves the use of weather balloon-borne sensors and relies on the design of a durable and modular landing gear system. The design in this project consists of lightweight carbon fiber tubes with high strength aluminum alloy insert materials to maximize strength as well as experiment flight duration and data collection. The fabrication of one support of the three-pronged assembly is carefully laid out, along with experimental testing to verify computational simulation results regarding the strength of the structure. Thus, a flight-ready design is developed and implemented for the purpose of creating a lightweight landing structure which can withstand several test flights, with the intention of implementing similar outrigger on propeller and gondola attachment support elements.
1. Introduction

1.1. Problem Statement

Accumulating greenhouse gases (GHGs) in our atmosphere exacerbate positive radiative forcing, altering Earth’s equilibrium for the sunlight absorbed compared to the energy radiated back into space, resulting in a net gain of energy which causes warming. This change in climate poses significant net damage costs which are expected to only increase over time. Mitigating climate risks by partially offsetting the radiative forcing from accumulating GHG emissions is known as solar radiation management (SRM), and one such management method involves the deliberate addition of aerosols to the Earth’s atmosphere to minimize ozone depletion. However, there is limited knowledge about the stratospheric processes that govern ozone depletion, and thus about both the efficacy and the risks of SRM by aerosol perturbation. To understand more about the chemistry and particle dynamics in the lower stratosphere, (where aerosol loading would take place), several laboratory investigations intended to simulate the conditions in the atmosphere have been conducted. However, laboratory systems alone cannot simultaneously meet all the conditions necessary to shed light on the uncertainties associated with the physical processes in the stratosphere. Small-scale in situ (on-site”) experiments in well-regulated circumstances can begin to remove some of the uncertainties associated with stratospheric processes. One such experiment —as put forth by the Anderson/Keith/Keutsch team in “Stratospheric controlled perturbation experiment: a small-scale experiment to improve understanding of the risks of solar geoengineering” (SCoPEx)— involves a scientific balloon suspending a propeller-driven payload which allows sensors to detect and track the impact of aerosol loading [1].
Such investigation relies on the design and fabrication of a lightweight, durable, and modular landing gear for the payload, to reduce the cost of repeated flights and enable research teams to more easily contribute to the scientific understanding of the risks of solar geoengineering.

1.2. Client Need

The Anderson/Keith/Keutsch SCoPEx project team’s payload is comprised of instrumentation for aerosol and H$_2$O generation as well as a LIDAR telescope for stratospheric chemical compound tracking. All equipment onboard, including batteries for propeller flight, is surrounded by a durable payload structure (or gondola) housing the instrumentation. Critically, the payload must be built to withstand the loads involved in flight such that (most, if not all) the structure can be used for following experiments with minimal damage to the instruments it houses. The hazards this payload will experience in flight will be explored in further detail in the following sections. In this project, the gondola design will be explored solely regarding the design of the critical element of landing gear, which shares all the requirements for modularity, durability, and component weight. As a metal payload frame can withstand the mechanical and thermal hazards experienced flight, the greatest hazards posed to the module containing all measurement tools for the experiment occur in the final stage: landing. For the estimated 600 kg total weight of the payload, the landing structure, or outrigger, should optimize both the durability and strength per unit of mass while providing maximum stability to prevent tipping over upon landing. Furthermore, given the limited government funding distributed to stratospheric research teams, the payload’s structural requirements for flight need to be considered alongside the tradeoffs between cost, mass, and modularity. Here, a lightweight, low-cost, durable, and modular payload landing system that satisfies the scope of the SCoPEx
research project is proposed. If successful in accomplishing these goals, the design of this outrigger system will be incorporated into future iterations of the payload structure, notably for the propeller supports.

Clients for such an outrigger system include those research teams, policy-makers, and independent agencies focused on experimentation which benefits from scientific balloon flight. One specific user as mentioned above is the Anderson/Keith/Keutsch SCoPEx team, part of the Keith Group at Harvard, who are working on flying a propelled scientific balloon capable of traveling a few meters per second in the lower stratosphere. As they continue to specify viable configurations, materials, and construction for the propelled payload structure their research requires, the goal of making an independent landing system that supports the onboard instrumentation is critical. The overall challenges with structural components for scientific balloons stem from the difficulty of making a lightweight structure with low-cost off-the-shelf materials with minimal customized and irreplaceable parts. Many standard payload gondolas are made with lightweight welded metals (such as aluminum) which have high durability but low modularity and need be replaced in their entirety upon failure. Furthermore, for heavier and more complex gondolas, a composite outrigger with many standard parts and a design that facilitates replacement upon failure would be an ideal solution. As the goal of this payload structure is to withstand flight in the lower stratosphere, possible additional users could include future scientific balloon research organizations with the shared goal of performing repeated in situ experiments to further improve our understanding of atmospheric chemical processes.

1.3. Methodology
The need for a landing support system that can withstand multiple test flights in order to acquire data on a larger temporal and spatial scale can be met by a variety of materials and assembly configurations. In this project, a composite outrigger system is explored, to both assess the weight-savings gained from using carbon fiber materials as well as determine the viability of extending a carbon fiber design to other gondola components. Utilizing pre-fabricated carbon fiber tubes, the goal of this project is to both design an assembly is driven by Finite Element Analysis (FEA) mechanics model simulation results as well as fabricate assembly components to for experimental testing. Due to the complex nature of composite material properties, composite-metal material interactions, epoxy bond strength, and combined bending, shear, and normal stresses that the assembly will experience, experimental testing is necessary to validate the FEA model that drives the assembly design. Such an experimentally-validated FEA model is a significant deliverable for potential users of a composite outrigger assembly, as any change in assembly geometry or outrigger design could efficiently be assessed for maximum stress under any given load.

2. Background Research

This section will first cover the Columbia Scientific Balloon Facility (CSBF) structural requirements for landing, as well as crush pad design and suggestions (section 2.1). Next, previous approaches to the fabrication of composite scientific balloon experimental payload structures will be summarized, involving: the criteria considered, critical scenarios expected, and final solution design and construction (sections 2.2 – 2.5).

2.1. Columbia Scientific Balloon Facility Structural Requirements
The CBSF is a NASA facility which provides the services of launching large, unmanned, high altitude research balloons. Upon termination of flight, scientific weather balloon payloads traditionally release a parachute to slow the descent before landing as much as possible. The release of this parachute results in vertical stress known as “chute-shock,” which is accounted for during design of the gondola by ensuring that the payload can withstand at least $10 G$ of vertical acceleration and $5 G$ of horizontal acceleration without failure [2]. If the material is defined as “brittle” (i.e. if elongation before failure is $\leq 10\%$ at $T = -60^\circ C$), these specifications must be multiplied by 1.5. In addition to these structural requirements, the factors affecting the gondola most at landing are surface winds and the terrain at the impact site. Accounting for an even terrain, (as the launch is often conducted in a dessert environment or similar low-variation settings), expected wind speeds for design verification cited by the CSBF include 15 knots vertically and up to 15-20 knots horizontally. While wind speeds can vary significantly, the upper portion of this range can be used to calculate the load experienced during flight; designing a landing system with proper support can ensure that stopping acceleration experienced does not exceed the maximum acceleration the payload is designed to withstand, i.e. chute-shock of $10 G$. The landing gear typically seen in gondola structures typically consists of solid landing legs with platforms attached to the bottom as well as some impact attenuation system. Such a system is designed to absorb gondola kinetic energy upon impact and is typically a paper crush pad or similar material that is intended to be easily fastened and replaced after each flight. The CSBF supplies researchers with corrugated cardboard material for crush pads, and the calculation for crush pad thickness is explored using the variables in Error! Reference source not found. Error! Reference source not found. A realistic calculation for the SCoPEx system will be detailed in section 3.3 of this report, using the known technical specifications for the payload.
Table 1: Crush pad design variables

<table>
<thead>
<tr>
<th>DEFINITION</th>
<th>DESCRIPTION</th>
<th>UNIT</th>
</tr>
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<tbody>
<tr>
<td>$M$</td>
<td>Mass</td>
<td>kg</td>
</tr>
<tr>
<td>$V$</td>
<td>Velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$K.E.$</td>
<td>Kinetic energy</td>
<td>J</td>
</tr>
<tr>
<td>$S$</td>
<td>Minimum allowable stopping distance</td>
<td>m</td>
</tr>
<tr>
<td>$S'$</td>
<td>Actual available stopping distance</td>
<td>m</td>
</tr>
<tr>
<td>$A$</td>
<td>Total required payload/crush pad contact area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$T_c$</td>
<td>Total required minimum thickness of crush pad</td>
<td>m</td>
</tr>
<tr>
<td>$T_i$</td>
<td>Crush pad thickness increment</td>
<td>m</td>
</tr>
<tr>
<td>$T_u$</td>
<td>Usable crush pad thickness</td>
<td>N/A (%)</td>
</tr>
<tr>
<td>$T_t$</td>
<td>Actual total thickness used</td>
<td>m</td>
</tr>
<tr>
<td>$N$</td>
<td>Integer number of crush pad thickness increments ($T_i$) to use</td>
<td>N/A (#)</td>
</tr>
<tr>
<td>$C.S.$</td>
<td>Crush strength of crush pad</td>
<td>Pa</td>
</tr>
<tr>
<td>$G$</td>
<td>Maximum allowable stopping acceleration</td>
<td>m/s$^2$</td>
</tr>
<tr>
<td>$g$</td>
<td>Acceleration due to gravity (9.81 m/s$^2$)</td>
<td>m/s$^2$</td>
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2.2. SPIDER balloon-borne telescope

The SPIDER project involved a balloon-borne telescope that was designed to measure the polarization of Cosmic Microwave background radiation [3]. Both the telescope onboard and scientific balloon maximum weights have strict requirements, which put tight constraints on the mass allowance of the outer frame gondola housing the telescope and other instrumentation. The design for this custom gondola was validated through both Finite Element Analysis (FEA) and mechanical tests, with the aim to maximize both durability and strength per unit mass of the support. The final mass of the SPIDER outer frame was 193.76 kg, accounting for only 9.4% of the total mass of the experiment. The techniques used for the SPIDER gondola are modular, and thus can extend to future scientific balloon experiments, easily adapting to geometries other than
their own. The gondola constructed consists of three main parts: the outer frame, the cylindrical cryostat, and a set of sunshields which are attached to the outer frame. The outer frame is a customized truss structure made of Carbon Fiber Reinforced Polymer (CFRP) tubes with aluminum inserts at the end, which are fastened together by multi-tube aluminum joints as seen in Figure 1 below. The goal of the triangular truss structures used in this design is to maintain axial forces and to minimize moment on truss elements. Furthermore, the customized geometry of the outer frame was defined using a series of beam mesh simulations made with the SolidWorks™ Simulation package to select the diameter and wall thickness of the tubes used. Once these were selected, the main critical scenarios considered in the design of the SPIDER gondola frame were according to the CSBF structural requirements for balloon gondolas.

![Figure 1: The SPIDER Gondola frame, simple schematic (left) and fabrication (right)](image)

The dominant scenario which translates to the largest forces on the gondola frame elements arises from the pull acceleration produced by the parachute, known as “chute-shock.” In this case, according to the CSBF, all structural components must be made to withstand a load 10 times the weight of the payload applied vertically at the suspension point. The results of simulations gave a minimum Factor of Safety (FOS) of 4.14 for gondola structural elements when comparing the
The tensile strength of the CFRP tubes (1,896 MPa) to the maximum stress experienced, given the weight of each payload component. The other critical scenario for the gondola frame with a lower FOS in the simulation was landing. To simulate the most mechanically demanding instant during landing, (first contact with the ground), the gondola was required to withstand a load equivalent to its mass at 5 G while resting on its lower plane, which resulting in a FOS of 3.77 on the truss elements. The SPIDER gondola was made without legs to land upon, due to the surface winds in Antarctica making an upright landing unlikely.

The elements of this project that most directly apply most directly to the construction of composite landing gear include frame element design and fabrication of composite joints. The CFRP tubes used were provided by CST Composites, and the specific product selected was the carbon/epoxy tubing AA70430A with a 70.4 mm inner diameter and a 3 mm wall thickness. Due to the manufacture of these composite materials, they were not weldable and could not be drilled into as drilling causes damage to the winding of the reinforced fibers. Thus, monolithic aluminum inserts (6061-T6 and 7075-T6 aluminum) were used to join the tubes, with one section designed for adhesion to the tube and the other for bolt fastening (Error! Reference source not found.). The adhesive selected for these inserts was the 3M Scotch-Weld™ Epoxy Adhesive 2216 B/A Gray, a standard for aerospace applications. Once the procedures for gluing aluminum inserts in the CFRP tubes were followed and the adhesive had cured with minimal corrosion, each assembly was pull-tested experimentally. The joints were shown to fail when the load corresponded to a minimum FOS of 1.37 and 1.43 for the 6061-T6 and 7075-T6 aluminum inserts respectively, when compared to the largest simulated axial load on the SPIDER outer frame. 6061-T6 aluminum alloy was cited as more common in aerospace applications due to its lower cost and the fact that it is
easier to machine. Finally, the geometry of the multi-tube joints in the nodes of the structure were also aluminum and determined by the orientation of the tube sets in the custom frame design.

![Figure 2: SPIDER aluminum inserts, CAD rendering (left) and joint fabrication (right)](image)

2.3. The Balloon-borne Large Aperture Submillimeter Telescope: BLAST

BLAST is a sub-orbital experiment designed to study the formation of stars in local galaxies, based on a 2-meter diameter telescope with a gondola pointing system which enables mapping positional accuracy of about 30" [4]. As in the SPIDER project, the maximum mass of the scientific payload dictated by CSBF (2268 kg) was a key constraint on the gondola system. As the BLAST 2 m primary mirror and cryostat had a total mass of 232 kg, a target weight of 2000 kg was set for the remaining payload, and the moment of inertia was estimated to be 4500 kg × m². Because of the pointing requirements for this project, the feedback rate of the control system was 10 Hz, and thus the gondola was designed with a minimum resonance frequency greater than 14.4 Hz. To accommodate this, all bearings were low friction and treated with low temperature grease, and the mechanical tolerances were set as to minimize backlash. Furthermore, all systems within the frame were designed to minimize generated torques from translations of the gondola generated by wind or from the balloon, so as not to re-orient the telescope. Finally, the CFRP tubes selected for the BLASTPol baffle were the CST composites AA20020A: a 20.0 mm inner diameter (ID) and 2.0mm wall thickness CFRP tube. The selection
of this product was made following a series of beam mesh simulations as described for the SPIDER project above, and given the similar size of the frame, the BLAST team was able to achieve a comparable weight of just under 200 kg, as seen in Table 2 below.

Table 2: Broken down total weights of the main components of the BLAST balloon-borne telescope [4]

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Frame</td>
<td>1100</td>
</tr>
<tr>
<td>Batteries</td>
<td>80</td>
</tr>
<tr>
<td>Solar Panels</td>
<td>30</td>
</tr>
<tr>
<td>Electronics</td>
<td>110</td>
</tr>
<tr>
<td>Sun Shields</td>
<td>196</td>
</tr>
<tr>
<td>Frame</td>
<td>195</td>
</tr>
<tr>
<td><strong>Total Weight at Launch</strong></td>
<td><strong>2020</strong></td>
</tr>
</tbody>
</table>

2.4. X-Calibur

X-Calibur is a balloon-borne hard X-ray scattering polarimeter which, during its 25-hour flight, took X-ray polarization to gather geometric information about compact sources such as black holes and neutron stars [5]. The carbon fiber-aluminum composite structure used for the gondola met the requirements for truss design given the relevant mechanical and thermal considerations for flight and was designed by Guarino Engineering Services. During the design phase, finite-element and thermal simulations of the truss elements were ran with cases of extreme temperature differences in an attempt to record any deflections over 3 mm. Again, because of the pointer requirements for this project, the feedback rate of the control system was 10 Hz. Thus, the truss structure was designed with the first resonant mode at a frequency greater than 10 Hz, while optimizing the weight and balance of the system. The weight is important for two reasons: first, as
the pointer system used can support telescopes weight up to 15000 lbs (or 680 kg), and second because the goal of this team was to fly X-Calibur as high as possible, which is achieved by reducing the total mass of the payload. Balance was another key concern as the pointer actuator motor system had limited torque, and thus the team had to balance X-Calibur’s center of gravity within 0.08 mm of the gimbal axis intersection point. The truss, therefore, was designed with two halves, each consisting of carbon fiber tubes and machined aluminum joints. The two halves were bolted onto a welded and machined aluminum 6061-T6 center frame, as seen in the rendering below (Figure 3). To achieve precise balance along with all three axes required for flight, about 35 lbs of brass and aluminum trim weights were mounted on the ends of the truss. Three types of carbon fiber tubes are used in different roles in the structure. The largest tubes were the four main load-carrying chords, which had an outer diameter of 2.5” and a wall thickness of 0.25”. The tubes for the side diagonal braces had an outer diameter of 1.5” and a wall thickness of 0.25” (appears blue in Figure 3), whereas the top and bottom lateral diagonals are made of 1.0” diameter tubes with a wall thickness of 0.125”, which is a standard thickness for carbon fiber tubes.

Figure 3: Rendering of complete X-Calibur truss structure [5]
Thicker tube walls increase the stiffness and allow for smaller diameter and required joint sizes, all in the effort to reduce the truss mass as well as limit the thermal expansion of the joints. The adhesive used for the aluminum joints and carbon fiber tubes was Loctite Hysol E-120HP epoxy. Furthermore, each half of the truss was a single glued part, which is stiffer than other approaches where each glued aluminum end piece is bolted to joints. However, one notable downside of this method is that the truss is far harder to repair. Three carbon fiber-to-aluminum joints were used, with the strongest type slightly differing from that of the SPIDER project, as carbon fiber tubes were bound onto the inside face of cylindrical female aluminum inserts. Because the outside surfaces of the carbon fiber tubes were ground by the manufacturer to a tolerance better than 0.002”, the inside surfaces required additional grinding before gluing. Finally, for thermal control, the trusses were painted white with appliance epoxy paint, wrapped in a fine mesh, and then in a 0.005” thick aluminized mylar. Assuming the carbon fiber properties given by the manufacturer, CST Composites, (Axial modulus of elasticity: 14 Msi, Ultimate tensile strength: 140,000 psi, Density: 0.06 lbs/in³, and Coefficient of thermal expansion: $0.18 \times 10^{-6} \frac{1}{K}$), the maximum axial tensile force in the chords was determined to be 682 lbs, and for the diagonal members 346 lbs. Using the guidelines for gondola design as provided by the CSBF, the team found that, for a 10 G vertical load, all truss members in the FEA were far below the ultimate tensile strength. For the 5 G horizontal load, the team cited maximum tube stresses of 7,700 psi. Furthermore, the area of the welded aluminum center had a yield strength of 11,000 psi, yet under the extreme cases of 10 G vertical and 5 G lateral loading, the FEA showed stresses of less than 1,000 psi for this part. Finally, the four extreme thermal cases incorporated into the FEA included the hottest and coldest cases in the two locations the flight was considered for a 24-hour period, Ft. Sumner New Mexico, and McMurdo Antarctica.
2.5. ALADIN

The space structure for the ALADIN instrument, a space-borne LIDAR telescope, had to achieve an optimal combination of mass, stiffness, and stability to meet the stringent performance requirements for this experiment [6]. The ALADIN structure consisted of a CFRP primary and secondary structure, the former supporting all large and sensitive equipment while the latter includes a large external baffle, three heat shields, and a thermal hood, all made of aluminum sheets equipped with thermal hardware for control. The overall height of the structure was 2.6 m, with baffle diameter of 1.6 m, and with an octagonal satellite interface of dimensions 1.1 m x 1.5 m.

The primary structure was determined to have to withstand very high loads due to the heavy equipment and infrastructure, summing up to 250 kg. Further, due to the dynamics of the external baffle, the primary structure needed to withstand 22.5 G in axial and 10 G in lateral directions. The main structure had a thickness of 10 cm and used 100 aluminum joints of various kinds to provide required interfaces. Finally, the 18 load transfer points identified between the main and satellite structures were realized by titanium brackets bonded into the honeycomb interface base, which has higher strength than aluminum but is typically avoided for its high cost (5-10 times greater than aluminum [2]). For the bottom CFRP tubes, a strut end design as seen in Figure 4 with tube inserts made of aluminum was incorporated such as to cope with the significant temperature variations (between −95°C and 50°C). This design is based off similar materials (CRFP and adhesives) as standard, but due to the very high loads and desired stiffness and stability, this new strut end fitting design was employed.
2.6. Key Takeaways

From the literature review, there are a number of takeaways that repeat among the variety of previous solutions for creating composite assembly components for scientific balloon experiment payloads. Generally, the examples included all mention and abide by the recommendations present in the CSBF reference material, namely that the maximum mass of the scientific payload (2268 kg) and maximum deceleration experienced by chute shock (10 G vertical and 5G horizontal) are primary considerations that cannot be compromised. Additionally, the notion that for brittle materials a minimum factor of safety of 1.5 must be considered as the height achieved by some scientific balloon applications can reach a low temperature where brittle materials are unable to deform as much. Furthermore, multiple structures were able to achieve a miraculously low weight, particularly the complex SPIDER payload which had a total mass of the outer frame that fell under 200 kg. While many applications included batteries, (which add substantial
weight and is also included in the SCoPEx experimental payload), many also included thermal and radiation shielding due to the height reached by these payloads, and such shielding will not be included in the SCoPEx payload, as the maximum height desired by the team will only be approximately 65,000 feet, where radiation is less of a concern.

Other key takeaways regarding frame attachment geometries when considering the use of carbon fiber tube supports have to do with the materials used and the attachment techniques employed. While some projects mentioned above utilized a variety of attachment methods, one persistent attachment included utilizing adhesive to bond aluminum inserts and the carbon fiber tubes. These applications all recognized the risks posed by the high electrochemical potential between aluminum and carbon fiber (which results in galvanic corrosion if the two materials come into contact), and all included procedures for minimizing such risk. While the procedures differed slightly in nature, they all included surface preparation involving abrasive techniques on the carbon fiber surface with the tighter tolerance as well as on the aluminum surface prior to bonding. Often these steps involved both sanding and degreasing of the surfaces, and one critical similarity aside from surface preparation involves the type of epoxy used. Throughout the literature review, the adhesives most commonly used were 3M Scotch-Weld™ Epoxy Adhesive 2216 B/A Gray and Loctite Hysol E-120HP. Both have similar material properties and are intended for aerospace applications, and the selection of adhesive as well as surface preparation procedure will be explored in further detail in section 6.1.1 Epoxy and CFRP surface preparation.

3. System Design Goals

3.1. Preliminary Design Criteria
In considering the system architecture upon transitioning from designing a payload flight platform to landing gear/supports, design criteria are defined as below, listed in the order of importance:

- Optimize strength
  - Withstand the loads involved upon landing and minimize any chance of tipping over or damage to the payload itself
- Modular design
  - Allows for easy assembly and disassembly, while satisfying strength requirements
  - Repeatable process for fabrication, using simple, off-the-shelf components
- Lightweight
  - Minimally impact the total mass of the payload, which is limited by Helium balloon lift and instrument weights
- Resistant atmospheric conditions
  - Thermal and radiation control methods to prevent damage to landing elements

While the technical specifications will be detailed in section 4 Design Approach, these criteria established a precedent such as to brainstorm important joint characteristics and assembly configurations.

3.2. Project Assumptions
In the pursuit of outrigger design and fabrication, several assumptions were taken into account regarding the nature of what was determined to be within the project scope. Initially, as the construction and final geometry of the payload frame had yet to be determined, the first assumptions included the notion that the payload could simplistically be represented by a $1 \text{ m}^3$ cubic aluminum frame. Additionally, the design of the payload (other than the attachment geometry of the outrigger assembly) was considered to be outside of project scope. Similarly, while calculations for the crush pad dimensions are carried out to assess the viability of achieving deceleration of $10 \text{ G}$ for the payload, (section 3.3 Crush Pad Dimensions), the construction and attachment of the crush pads also carried several assumptions.

The foremost assumption present in the crush pad calculations has to do with the limited landing situations considered for simplicity; secondary considerations regarding the attachment geometry of crush pads to outrigger bases are not explored in this project, as the crush pads are disposable and collapse completely upon every flight. While the payload experiences velocity in both vertical and horizontal components due to the high wind speeds likely present at the landscape where landing occur, the calculations for outrigger stability maintained a strictly vertical landing assumption. Furthermore, all four outrigger support legs were assumed to touch down upon landing at once, thus evenly distributing the $60 \text{ kN}$ force evenly. While the minimum Factor of Safety (FOS) considered was 1.6, the ideal specification to be achieved was to design and fabricate outrigger assembly components that could withstand over two times the maximum load expected, such that even if solely two outrigger support assemblies experienced the bulk of the landing forces, they would be capable of deforming without reaching yield. These quantitative considerations themselves stem from the assumed literature value that maximum deceleration during flight (before landing) would be that of chute shock, cited as $10 \text{ G}$ by the
However, in preliminary test flights upon parachute deceleration, the maximum chute shock forces observed corresponded to 16 \( G \) of vertical deceleration, and thus the minimum FOS of 1.6 was adopted.

With the assumptions mentioned above, the critical design considerations and calculations included within the project have to do with the outrigger geometry, degrees of freedom and attachment designs. Both the modular joints and epoxied connections had to meet the engineering threshold specified by the minimum FOS. Also, deformation of the assembly in its entirety, particularly of the composite tube supports, was assessed in regards to the assembly’s ability to withstand the maximum stress expected. The assembly degrees of freedom which affected the maximum stresses and deformations expected have to do with the spread of assembly lateral legs from one another, the angle with respect to the horizontal axis, and the length of both composite tube supports and inserts within the supports. Such criteria will be explored in depth in the following sections, upon verification that crush pads can be designed with dimensions that achieve the desired maximum 10 \( G \) of deceleration upon landing.

3.3. Crush Pad Dimensions

<table>
<thead>
<tr>
<th>Known Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( M )</td>
<td>600 kg</td>
</tr>
<tr>
<td>( V )</td>
<td>7.72 m/s</td>
</tr>
<tr>
<td>( T_i )</td>
<td>0.1016 m</td>
</tr>
<tr>
<td>( T_u )</td>
<td>70 %</td>
</tr>
<tr>
<td>C.S.</td>
<td>10 lb/in(^2) ( \Rightarrow ) 68947 Pa</td>
</tr>
<tr>
<td>( G )</td>
<td>2.5 * 9.81 m/s(^2)</td>
</tr>
</tbody>
</table>
The technical specifications in Table 3 above follow the required SCoPEx project specifications resulting from discussions with the project lead and team mechanical engineer, based on the assumption that the 600 kg overall payload weight (including the landing structures) will need to be stopped at the maximum magnitude of 10 G acceleration that the CBSF determined gondolas must be designed with in mind. Following the NASA guidelines for example crush pad design on all four legs, this results in a 2.5 G maximum allowable stopping acceleration per each support leg (assuming a worst-case two-leg landing scenario) and the minimum crush pad surface area can therefore be calculated using the data available in Table 4 to find the missing variables for Table 1.

Moreover, as shown in the sample calculation in Appendix A.2 – Four-Leg Landing Scenario for the 2.5 G stopping acceleration case, a square crush pad of 0.461 meters (or 18 inches on each square side) would account for the appropriate stopping acceleration.

Moreover, accounting for a two-leg landing scenario, rather than 2.5 G maximum allowable stopping acceleration for each leg, the 10 G maximum deceleration is distributed by 5 G on each of the two legs initially bracing for payload impact. For this scenario, a square crush pad side length 2.087 feet would suffice, (see Appendix A.3 – Two-Leg Landing Scenario for more detailed calculations) which is approximately equal to 25 inches, or 0.636 m. In the two-leg landing scenario, all parameters in the complete crush pad are listed in Table 4.

Reference source not found.: 

<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>DESCRIPTION</th>
<th>VALUE</th>
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<tr>
<td>M</td>
<td>Mass</td>
<td>600 kg</td>
</tr>
<tr>
<td>V</td>
<td>Velocity</td>
<td>7.72 m/s</td>
</tr>
<tr>
<td>K.E.</td>
<td>Kinetic energy</td>
<td>117865 J</td>
</tr>
</tbody>
</table>
### Table 1

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$S$</td>
<td>Minimum allowable stopping distance</td>
<td>1.18 m</td>
</tr>
<tr>
<td>$S'$</td>
<td>Actual available stopping distance</td>
<td>1.22 m</td>
</tr>
<tr>
<td>$A$</td>
<td>Total required payload/crush pad contact area</td>
<td>0.4048 meters²</td>
</tr>
<tr>
<td>$T_c$</td>
<td>Total required minimum thickness of crush pad</td>
<td>0.86 m</td>
</tr>
<tr>
<td>$T_i$</td>
<td>Crush pad thickness increment</td>
<td>0.1016 m</td>
</tr>
<tr>
<td>$T_u$</td>
<td>Usable crush pad thickness</td>
<td>0.7 = 70%</td>
</tr>
<tr>
<td>$T_t$</td>
<td>Actual total thickness used</td>
<td>0.91 m</td>
</tr>
<tr>
<td>$N$</td>
<td>Integer number of crush pads ($T_i$) to use</td>
<td>9</td>
</tr>
<tr>
<td>C.S.</td>
<td>Crush strength of crush pad</td>
<td>68947 Pa</td>
</tr>
<tr>
<td>$G$</td>
<td>Maximum allowable stopping acceleration</td>
<td>49 m/s²</td>
</tr>
</tbody>
</table>

3.4. Material Selection

Following calculations regarding the outrigger crush pad size, the inverted pyramid design as mentioned in the CSBF recommendations was abandoned due to added complexity with minor structural advantages [2]. Furthermore, solely rectangular prisms with square cross-sectional areas are considered, as mentioned above. For the joint elements and carbon fiber tubes themselves, several materials are considered, including: Unidirectional Carbon Fiber/Epoxy tubes (Standard, Intermediate, High, and Ultra-High Modulus), three types of aluminum, Steel 4130, and Titanium 6M-4V. Sample relevant material properties are listed below in Table 5 below, as specified by composite manufacturer Clearwater Composites [7]:
### Table 5: Material properties relevant for outrigger support tube and joint attachment components

<table>
<thead>
<tr>
<th>Material</th>
<th>Grade / Type</th>
<th>Design / Application</th>
<th>Longitudinal Tensile Strength (ksi)</th>
<th>Longitudinal Tensile Modulus (msi)</th>
<th>Shear Modulus (msi)</th>
<th>Density (g/cm³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon Fiber/Epoxy</td>
<td>Standard Modulus</td>
<td>Bending</td>
<td>300</td>
<td>15</td>
<td>0.6</td>
<td>1.55</td>
</tr>
<tr>
<td>(Unidirectional)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Carbon Fiber/Epoxy</td>
<td>Intermediate Modulus</td>
<td>Bending</td>
<td>325</td>
<td>20</td>
<td>0.6</td>
<td>1.57</td>
</tr>
<tr>
<td>(Unidirectional)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Carbon Fiber/Epoxy</td>
<td>High Modulus</td>
<td>Bending</td>
<td>250</td>
<td>30</td>
<td>0.6</td>
<td>1.59</td>
</tr>
<tr>
<td>(Unidirectional)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Carbon Fiber/Epoxy</td>
<td>Ultra-High Modulus</td>
<td>Bending</td>
<td>200</td>
<td>45</td>
<td>0.6</td>
<td>1.70</td>
</tr>
<tr>
<td>(Unidirectional)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel</td>
<td>4130</td>
<td></td>
<td>100</td>
<td>30</td>
<td>12</td>
<td>7.7</td>
</tr>
<tr>
<td>Titanium</td>
<td>6M-4V</td>
<td></td>
<td>120</td>
<td>16</td>
<td>6.2</td>
<td>4.34</td>
</tr>
<tr>
<td>Aluminum</td>
<td>6061-T6</td>
<td></td>
<td>35</td>
<td>10</td>
<td>3.8</td>
<td>2.7</td>
</tr>
<tr>
<td>Aluminum</td>
<td>2024</td>
<td></td>
<td>47</td>
<td>10.6</td>
<td>4.06</td>
<td>2.78</td>
</tr>
<tr>
<td>Aluminum</td>
<td>7075-T6</td>
<td></td>
<td>73</td>
<td>10.4</td>
<td>3.90</td>
<td>2.81</td>
</tr>
</tbody>
</table>

For carbon fiber tubes, beside the modulus of the material (i.e. stiffness or resistance to deformation), the method of fabrication also plays an important role in deciding the appropriate material for a given application. Generally, the layup of the fibers in the composite tube with respect to the radial axis have the following applications:

- \(0°\) Fibers maximize bending stiffness
• 45° Fibers maximize torsional stiffness. Typically plied together in an opposing manner, i.e. ±45

• 90° Fibers maximize crushing stiffness and generally assist in overall tube strength (i.e. radial compression)

Typically, at least 10 layers of Fiber in varying orientations are used, and as each layer is merely 0.006" in thickness, the minimum standard carbon fiber tube thickness is 0.06". Furthermore, there is a difference in fiber manufacturing techniques, which has an effect on material properties, as summarized by the radar chart in Figure 5 below (page 32).

*Figure 5: Comparison of CFRP fabrication technique and corresponding performances*
The four main manufacturing techniques considered for the selection of CFRP tubes in this particular application were as follows:

- **Roll-Wrapped**
  - Done with a carbon fiber plies pre-impregnated with epoxy resin, including at least 0°/90° layup orientations
  - Allows for maximum consistency of material properties, best in class for bending applications
- **Filament Wound**
  - Individual carbon fibers applied in a custom spiral pattern all at once by a machine over a mandrel core, which is then removed
  - Tube is then cured once the fibers have been applied, allowing for cheap CFRP which performs well in torsion
- **Pultruded**
  - All fibers applied with epoxy & spun around tube in single direction
  - Very good in tension but easily split in compression or torsion
- **Pullbraided**
  - Fibers are braided together as they are being pulled through the heated die and onto the mandrel
  - Less Expensive than roll-wrapped, similar overall performance

From this comparison, it was clear that pullbraided and roll-wrapped prefabricated tubes were ideal choices, and as a roll-wrapped tube with 10 layups of 0.006” thick CFRP plies has been narrowed down, such a unidirectional tube was chosen. The specific material properties for the tube chosen will be expressed in further detail in section 3.4 Material Selection.
3.5. Free Body Diagrams

![Free Body Diagrams](image)

The idea behind an outrigger design with a significantly wide spread of support geometry involved maintaining the structural integrity and rigidity that would be lacking if the supports were to lie perfectly perpendicular with respect to the payload edges. Each landing support is devised of three different support elements, each of which distribute the load and account for maximum rigidity of the payload. Images of a top-down view (Figure 6, above) and free body diagram (Figure 7, below) are included to demonstrate the overall design of the construction of the landing supports with respect to their orientation to the overall payload. From a top-down view four symmetrical outrigger systems at each corner of the gondola can be seen. The distance $D1$ is governed by the angular difference $\Delta \theta$ of the lateral outrigger supports from the $45^\circ$
angle that the middle support forms with each lateral edge of the payload. Furthermore, the free body diagram included below shows the lateral view of one such outrigger system, where the middle support is connected along the closest lateral edge (perpendicular to the square base at the bottom), and the two side outrigger supports are connected at a lower height to the base of the payload, and attachment of these elements will be discussed in following sections.

The view presented in Figure 7 below is helpful in understanding the added rigidity of this payload support due to using three outrigger legs as opposed to one. In the likely case of significant vertical as well as horizontal velocities occurring during landing, the side outrigger
legs at a distance $D1$ prevent the leg at $45^\circ$ from experiencing bending forces laterally. In addition to each component experiencing an axial load which is a fraction of the total $15\ kN$ vertical compressive load, there will be a significant bending moment along each outrigger support. Thus, the axial loads are as follows:

$$P = 15\ kN = \sum_{i=1}^{3} F_i = F_1 + F_2 + F_3 = F_1 + 2 \times F_2$$

$$15\ kN = F_{axial,1} \times \sin \theta_1 + 2 \times F_{axial,2} \times \sin \theta_2$$

Furthermore, with forces on the legs in bending, the three arrows in yellow at the base of the outrigger create a torque with the angles $\theta_1$ and $\theta_2$:

$$F_{bending,1} = 15\ kN \times \cos \theta_1$$

$$F_{bending,2=3} = 15\ kN \times \cos \theta_2$$

The maximum bending force that can be tolerated for the carbon fiber tubes depend on the modulus used, (standard, intermediate, high, ultra-high), as the allowable torque is dependent on the modulus of elasticity and the length of the tubes, $L$:

$$L_{2=3} = \frac{H_2}{\sin(\theta_2)}, L_1 = \frac{H_1}{\sin(\theta_1)}$$

The bending forces and minimum inner diameters needed to withstand the loads described above will be outlined in detail in the following section 3.6.2 Flexure Load.
3.6. Target Engineering Specifications

3.6.1. Compressive Load

As a first-pass at the critical loading cases, the critical compressive force is for calculated for different cross-sectional tube dimensions, involving constant length and material properties, in order to find the minimum viable tube diameter. During preliminary flight testing by the SCoPEX team with an aluminum payload frame, the high tensile forces experienced during chute-shock were slightly above the 10 $G$ vertical forces cited by the CSBF, (up to 16 $G$ in magnitude vertical accelerations were observed). Furthermore, the aluminum frame used in this test seemed well capable of withstanding these forces, and as such the minimum Factor of Safety (FOS) considered in this project will be 1.6 to compensate for the difference between the literature maximum deceleration and the observed 16 $G$ deceleration. As outrigger supports are designed with crush pads calculated to limit the assembly deceleration to the cited 10 $G$ upon landing, the supports designed in this project will be considered to experience at least a 60 $kN$ in overall landing forces, due to the predetermined 600 $kg$ payload mass. As the compressive strength of unidirectional CFRP is less than the respective tensile strength, and as the tensile and compressive strength requirements are nearly the same for most carbon fiber tube layouts, attention can be concentrated on designing the support tubes to withstand maximum axial compressive load considering strength, stiffness, and overall buckling. This first-pass consideration assures that, in the case of a non-vertical landing where all loading occurs axial to any one of the supports, the stress experienced is sufficiently below yield of the CFRP tubes. This main failure mode is known as Euler Buckling, also known as elastic instability, and occurs when a structural support is subjected to high compressive strength. The criteria that determine the critical force in buckling columns without consideration for “lateral forces” are the flexural
rigidity — the product of the moment inertia \( I = (OD^4 - ID^4) \times \frac{\pi}{64} \) and the Modulus of Elasticity \( E \) — and the effective column length, \( \kappa \times L \), where: \( OD \) is the outer diameter, \( ID \) is the inner diameter, \( \kappa \) is the column effective length factor, and \( L \) is the unsupported column weight [3]. With these criteria in mind, the equation for Euler’s critical load, \( F \), is as follows:

\[
F = \frac{\pi^2 \times (E \times I)}{(\kappa \times L)^2}
\]

Considering the worst-case scenario, which is the beam being fixed at both ends, the effective column length \( \kappa = 0.5 \), and for the necessary ballast clearance of between 2 and 3 feet the shortest leg length is limited at \( 2 \, ft = 0.61 \, m \). In terms of material properties, the composite tubing from manufacturers such as Dragon Plate™ and Rock West Composites™ have been shown to have an elastic modulus \( E = 15 \, Msi \) or \( E \approx 100 \, GPa \) (see Table 5), which is in line with the values used in projects such as the SPIDER balloon-borne telescope [3]. (For a detailed calculation with the project’s specifications, see Appendix A.1 – Calculation of Composite Elastic Modulus for SPIDER project) Furthermore, realistic inner diameters for similar loading applications application range from at least 20 \( mm \) (0.79 \( in \)) up to 50 \( mm \) (1.97 \( in \)), as seen in literature review. While the range of inner diameter values could easily exceed 50 \( mm \), as is the case in higher load-bearing CFRP assemblies, the design would become prohibitively expensive beyond such dimensions, and the purpose of this exercise is to find the minimum viable tube inner diameter for this application. Finally, a reasonable wall thickness considered for this calculation is \( t = 1.52 \, mm = 0.06 \, in \), thus \( OD = 2 \times t + ID \), establishing a relation between Euler’s critical load, \( F \), and the inner diameter, \( ID \):
\[ F = \frac{\pi^2 \times E \times (OD^4 - ID^4) \times \frac{\pi}{64}}{(0.5 \times L)^2} \]

\[ \Rightarrow F = \frac{\pi^2 \times 1 \times 10^{11} \text{ Pa} \times (((ID + 2 \times 0.00159)^4 - ID^4) \times \frac{\pi}{64}}{(0.5 \times 0.61 \text{ m})^2} \]

Simplified, this results in:

\[ F = 5.208 \times 10^{11} \frac{N}{m^4} \times ((ID + 0.00318 \text{ m})^4 - ID^4) \]

With the range of Inner Diameters considered, the resulting relationship between ID and Euler’s Critical Load, for a given length \((L = 0.61 \text{ m})\), wall thickness \((t = 1.52 \text{ mm})\), and maximum Load, \(F_{\text{max}} = 60 \text{ kN}\) are summarized in Table 6 below:

<table>
<thead>
<tr>
<th>Inner Diameter, ID (mm)</th>
<th>Moment of Inertia (mm(^4))</th>
<th>Euler’s Critical Load, F (kN)</th>
<th>FOS</th>
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</thead>
<tbody>
<tr>
<td>20.0</td>
<td>6318</td>
<td>39</td>
<td>0.41</td>
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<td>22.7</td>
<td>9017</td>
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<td>25.5</td>
<td>12394</td>
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<td>28.2</td>
<td>16524</td>
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<td>30.9</td>
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</tbody>
</table>
Along with Euler’s critical load for each of the tube cross-sectional dimensions, the corresponding Factor of Safety (FOS) is included considering a one-leg landing scenario, where a single outrigger assembly experiences the full 60 kN of force along the axial direction. While this one-leg landing scenario is outside of the scope of the remainder of the critical design scenarios, it provides an absolute worst-case calculation for these cross-sectional dimensions. Finally, increasing the thickness of the tube or the modulus of elasticity would increase the critical load, but with the calculations in this section, an inner diameter of over 30 mm is necessary to achieve the minimum FOS of 1.6.

3.6.2. Flexure Load

A more involved look at the critical design scenario involves the calculation of the maximum allowable angle for the lateral supports (which experience the most bending stress) for a given tube geometry. This involves calculation of the Moment on the tube \( M = F \times L \times \cos(\theta) \), where the \( \theta \) here is the \( \theta_2 \) in the free body diagram (section 3.5 Free Body Diagrams), and the cosine is a result of the formula for moment, which is usually \( M = P \times L \times \sin(\phi) \), where \( \phi \) is the angle between the lever arm \( L \) and the load applied \( P \), but in this case \( \theta \) is defined as the angle between the horizontal and the lever arm (length of the tube). Thus, \( \phi = 90^\circ - \theta \), and \( \cos(\theta) = \sin(\phi) \). Furthermore, for a given angle, \( \theta \), the height is given by:

\[
H = L \times \sin(\theta)
\]

Other relevant equations for the bending stresses are those for the moment of inertia for a solid round beam:
\[ I = (OD^4 - ID^4) \times \frac{\pi}{64} \]

And the distance from the beam's neutral axis to the point of interest along the height of the cross section,

\[ y = OD \times \frac{1}{2} \]

As, in this case, the neutral axis is the center of the hollow tube, and the distance to the point of interest where the bending load will be applied is simply the outer radius, or half the inner radius. From this, the bending stress can be calculated as:

\[ \sigma_b = \frac{(M \times y)}{I} = \frac{P \times L \times \cos(\theta) \times OD \times \frac{1}{2}}{(OD^4 - ID^4) \times \frac{\pi}{64}} \]

For this calculation some realistic dimensions considering the availability of composite tube layouts that meet the minimum strength desired include: the ballast clearance of between 2 and 3 feet between the bottom of the payload and the base of the outrigger support legs, and the inner diameter minimum of 30 mm. This calculation can be carried out for the known load on a single outrigger support, (considering an appropriate factor of safety), of \( F = 5 \text{ kN} \), as there are four outriggers supporting 60 kN each, leaving 15 kN for the three supports in each outrigger assembly, and 5 kN on each leg. First, in imperial measurements it is worth noting that CFRP tubes are often sold by the foot. Thus, the typical lengths for carbon fiber tubes range from 1 foot to 3, 6, 7, and 8 feet, and in order to minimize cost and maintain modularity as well as the necessary ballast clearance, a total CFRP tube length of 8 feet is chosen, cut such that the three support legs take up an equal third of the prefabricated tube (around 31.5 inches).
or 800 mm, allowing for tolerances). With a tube length of 800 mm, the minimum angle the lateral supports could have with respect to the horizontal axis is 50°, as this would account for 2 feet of ballast clearance below the payload. A 60° assembly allows for 27.3”, 65° for 28.5”, 70° for 29.6”, etc. Thus, while there is diminishing marginal gain in ballast height for increasing assembly angle, there is a significant reduction in bending stresses, and as such a range of angles will be considered. By setting equation **Error! Reference source not found.** equal to the yield strength of CFRP divided by the factor of safety, the following relation can be solved for the minimum inner diameter ID, assuming again that the minimum reasonable CFRP tube thickness used is t = 1.52 mm:

\[
\sigma_b = \frac{P \times L \times \cos(\theta) \times (ID + 2 \times 0.00152) \times \frac{1}{2}}{((ID + 2 \times 0.00152)^4 - ID^4) \times \frac{\pi}{64}} \leq \frac{\sigma_y}{FOS}
\]

Where \( \sigma_y \) is the yield strength of the CFRP tube. Here, considering a yield strength of \( \sigma_y = 2 \ GPa = 2 \times 10^9 \ Pa \), \( P = 5 \times 10^3 \ N \), \( L = 0.8 \ m \), and \( FOS = 1.6 \), the minimum required internal tube diameters and corresponding Euler critical load \( F \), for a range of angles between 50° and 80° are summarized in Table 7 below:
**Table 7: Relationship between angle of lateral legs, minimum required inner diameter, and Euler’s critical load**

<table>
<thead>
<tr>
<th>Angle, θ (degrees)</th>
<th>Inner Diameter, ID (mm)</th>
<th>Euler’s Critical Load, F (kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>39.74</td>
<td>245</td>
</tr>
<tr>
<td>55</td>
<td>37.49</td>
<td>207</td>
</tr>
<tr>
<td>60</td>
<td>34.94</td>
<td>169</td>
</tr>
<tr>
<td>65</td>
<td>32.05</td>
<td>132</td>
</tr>
<tr>
<td>70</td>
<td>28.74</td>
<td>97</td>
</tr>
<tr>
<td>75</td>
<td>24.88</td>
<td>65</td>
</tr>
<tr>
<td>80</td>
<td>20.20</td>
<td>35</td>
</tr>
</tbody>
</table>

Increasing the thickness of the tube or the modulus of elasticity would increase the critical load and decrease the minimum required inner diameter, but with the calculations in this section, an inner diameter of at least about 30 mm is necessary to achieve the minimum FOS of 1.6. For the final design, the tube selected is from Rock West Composites™, and has the following carbon fiber layup (Figure 8) and technical specifications (Table 8):

![Figure 8: Layup of carbon fiber plies for tube manufactured for bending applications](image-url)
And thus, the tube selected above is primarily intended for maximum strength for bending applications (due to the 0° carbon fiber ply layups), as well as general strength due to the included 90° fiber plies.

<table>
<thead>
<tr>
<th>Category</th>
<th>Specification</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Diameter</td>
<td>1.25</td>
<td>inch</td>
</tr>
<tr>
<td>Outer Diameter</td>
<td>1.37</td>
<td>inch</td>
</tr>
<tr>
<td>Wall Thickness</td>
<td>0.06</td>
<td>inch</td>
</tr>
<tr>
<td>Length</td>
<td>23.75</td>
<td>inch</td>
</tr>
<tr>
<td>Ultimate Tensile Strength lengthwise, (Ftu)</td>
<td>157</td>
<td>ksi</td>
</tr>
<tr>
<td>Ultimate Compression Strength lengthwise, (Fcu)</td>
<td>96</td>
<td>ksi</td>
</tr>
<tr>
<td>Flexure Strength</td>
<td>96</td>
<td>ksi</td>
</tr>
<tr>
<td>Tension Modulus</td>
<td>15.2</td>
<td>msi</td>
</tr>
<tr>
<td>Compression Modulus</td>
<td>14.9</td>
<td>msi</td>
</tr>
</tbody>
</table>

These specifications translate equivalently to a 31.75 mm inner diameter, 34.80 mm outer diameter, a 1.52 mm wall thickness and a length \(L = 603\) mm, with a \(F_{tu} = 1083\) MPa, \(F_{cu} = 662\) MPa, a flexural strength of 662 MPa as well, tension modulus \(E_t = 104.8\) GPa, and a compression modulus \(E_c = 102.7\) GPa. Will all these specifications in mind, the bending stress is of critical concern for assembly design, and the maximum feasible length while maintaining a FOS of 1.6 is calculated below, as the length of the tube used is the factor that has the most impact on the bending stress experienced, given fixed cross-sectional dimensions. Angles considered range from 50° to 87.5°, hoping to achieve a length that is similar in nature to the desired 30 inches as to achieve an appropriate ballast clearance. Additionally, it is important to note that a factor of safety of 1.6 dictates that the maximum flexural stress experienced need be \(\frac{662}{1.6} = 414\) MPa. The resulting maximum lengths (as well as clearance
height between the base of the payload and the outrigger base) are summarized in Table 9 below, assuming pure deformation for the rigid tube chosen, with lateral assembly supports ranging from 50° to 87.5° with respect to the horizontal axis:

<table>
<thead>
<tr>
<th>Angle, $\theta$ (degrees)</th>
<th>Maximum Length (inch)</th>
<th>Clearance Height (inch)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>7.3</td>
<td>5.6</td>
</tr>
<tr>
<td>55</td>
<td>8.2</td>
<td>6.1</td>
</tr>
<tr>
<td>60</td>
<td>9.4</td>
<td>6.7</td>
</tr>
<tr>
<td>65</td>
<td>11.2</td>
<td>7.4</td>
</tr>
<tr>
<td>70</td>
<td>13.8</td>
<td>8.2</td>
</tr>
<tr>
<td>72.5</td>
<td>15.7</td>
<td>15.0</td>
</tr>
<tr>
<td>75</td>
<td>18.2</td>
<td>17.6</td>
</tr>
<tr>
<td>77.5</td>
<td>21.8</td>
<td>21.3</td>
</tr>
<tr>
<td>80</td>
<td>27.1</td>
<td>26.7</td>
</tr>
<tr>
<td>82.5</td>
<td>36.1</td>
<td>35.8</td>
</tr>
<tr>
<td>85</td>
<td>54.1</td>
<td>53.9</td>
</tr>
<tr>
<td>87.5</td>
<td>108.1</td>
<td>108.0</td>
</tr>
</tbody>
</table>

It is important to note that these calculations and considerations are for the worst-case scenario, in which the aluminum attachments, inserts, and payload frame are considered rigid and non-deformable. These materials are all shown to have a lower elastic modulus than carbon fiber in any direction, and thus they will experience deformation before the composite tubes of the specified dimensions are subject to the remainder of the load alongside the aluminum components. For instance, if the aluminum compensates for so much of the load such that the tubes only experience $\frac{1}{2}$ of the 5 $kN$ in bending, the corresponding maximum length and height maintaining a FOS of 1.6 are as follows (Table 13Table 10):
### Table 10: The same calculation for maximum allowable length and height, with half the experienced load, $F = 2.5 \text{kN}$

<table>
<thead>
<tr>
<th>Angle, $\theta$ (degrees)</th>
<th>Maximum Length (inch)</th>
<th>Clearance Height (inch)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>14.7</td>
<td>11.2</td>
</tr>
<tr>
<td>55</td>
<td>16.4</td>
<td>13.5</td>
</tr>
<tr>
<td>60</td>
<td>18.9</td>
<td>16.3</td>
</tr>
<tr>
<td>65</td>
<td>22.3</td>
<td>20.2</td>
</tr>
<tr>
<td>70</td>
<td>27.6</td>
<td>25.9</td>
</tr>
<tr>
<td>75</td>
<td>36.4</td>
<td>35.2</td>
</tr>
<tr>
<td>80</td>
<td>54.3</td>
<td>53.5</td>
</tr>
</tbody>
</table>

### 4. Design Approach

In approaching the overall outrigger assembly design, several iterative design processes were devised for separate critical components. First, joint and attachment designs from a broad literature review were collected and narrowed down according to the most viable solutions for this application. Initially, the most important considerations for these designs was modularity of components and the likely stability as well as stiffness, which was later evaluated using FEA simulations. After joint designs were determined, the most important assembly degrees of freedom were addressed, including spread of lateral legs as well as the angle of the middle support leg with respect to the vertical axis. Finally, the final assembly and joint design is depicted and explored in depth in the following sections.

#### 4.1. Joint Designs

As mentioned above, the joint design was explored first, as attachment techniques posed several difficulties. Among the critical considerations, modularity and strength were of utmost importance when brainstorming joint designs. Aside from the traditional aluminum inserts which
slide into the inner diameter of CFRP tubes and are bonded with adhesive, (similar to the design of the SPIDER joints as in Figure 2 page Error! Bookmark not defined.), three alternative joint designs were considered in discussion with teaching staff and advisors. One possibility to secure aluminum shafts to carbon fiber tubes includes using high-torque hose clamps around machined oversized aluminum, with slits to allow for expansion. Additionally, a strut design similar to that employed by the ALADIN group (Figure 4) could be used in conjunction with turnbuckles to force expansion from the inside of the composite tube; alternatively, a strut that is oversized and forced into the tube as in the ALADIN project could be employed upon further research into strut design. Furthermore, to compare strut design to other methods of forced expansion, joints with tapered composite tubes and similar turnbuckle and shaft movement could be explored in future work. Due to the custom nature and reduction of modularity involved in using tapered CFRP tubes, this option was not assessed and is typically used in higher altitude applications. Finally research that has been done on different types of adhesive joints for carbon fiber tubes revealed that double lap bonded joints are the most common for tube inserts. Critical factors for adhesive joint strength include the relationship between bonding width and length, in addition to adhesive fillets which were found to reduce maximum shear stress by up to 30% [8].

Overall, for two strut end fittings are considered as general node attachment designs, seen in Error! Reference source not found. below: one ball end fitting with threaded inserts for each joint (left) and a single clevis node bolted to a double clevis for each insert (right). These designs
were taken into consideration for the node attachment of all prototypes, and most notably for the first iteration of a conceptual prototype to assist with assembly configuration.

![Image of strut end design fittings, with threaded insert (left) and double clevis inserts (right) both allowing for complex node configurations](image)

*Figure 9: Strut end design fittings, with threaded insert (left) and double clevis inserts (right) both allowing for complex node configurations [9]*

During preliminary brainstorming stages, an attachment design similar the latter double-clevis attachment technique was 3D printed for a rough prototype construction with wooden support structures, as can be seen in Figure 10 below.
The landing gear shown above was sketched during meetings with the lead mechanical engineer on the SCoPEx team, Michael Greenberg, to somewhat resemble lunar lander legs. The considerations underlying this design were based on preliminary test flights where perpendicular legs were not sturdy enough to support the payload during landing, and thus supports at an angle were investigated from the beginning of the design iterations. From these conversations, it was concluded that each corner of the rectangular payload could be supported by three composite tubes, with custom metal inserts for each point of supports [10]. The minimum clearance underneath the payload $h$ will depend on ballast design and construction, and the corresponding distance from the payload $d$ will determine the angle of the supports. Three supports will be mounted onto a metal plate, at the angles $\phi_1, \phi_2, \text{ and } \phi_3$. The crush pad of thickness $T$ will be glued to plywood which is bolted on the underside of the same plate, as per the CSBF guidelines [2]. To preserve end-to-end modularity, the joint design and lengths $\ell$ of the upper and lower
tubes will be identical in nature, such that they can be replaced with one another. However, initial conversations supposed that the middle tube will almost certainly have a different design and length, which ultimately was avoided by taking advantage of the angle of the middle support \( (\phi_2) \) as far less than that of the angles of the lateral supports with respect to the lateral supports \( (\phi_1, = \phi_3) \). Before arriving to these conclusions, the prototype in Figure 10 was created based off the model represented in Figure 11 below.

![Figure 11: Rendering for preliminary CAD model (right) with simple cubic aluminum frame, corrugated paper crush pads, and carbon fiber support elements without joints](image)

This preliminary design for the full assembly geometry was made using SOLIDWORKS™ CAD software, including a 1.5 m x 1.5 m x 1.5 m cubic aluminum frame, four support elements constructed of: two one-meter carbon fiber supports, one 0.75-meter support, four crush pads and an aluminum plate as each landing leg, or “outrigger.” The planes on the closeup to the left demonstrate the planes that divide the corrugated paper crush pads up into several sections, ideally glued together. The crush pads shown are 0.25 m x 0.25 m and are a 15° inverted pyramid design, yet as the full considerations for crush pad design discussed in section 3.3 Crush Pad Dimensions,
the inverted pyramid design was abandoned in favor of a rectangular prism, of dimensions $0.64 \, m \times 0.64 \, m \times 0.61 \, m$.

4.2. Assembly Degrees of Freedom

As mentioned previously, the degrees of freedom to be optimized for the assembly include: minimum clearance underneath the payload $h$, the corresponding lateral distance from the payload $d$, the length of the composite tubes $\ell$, and the three angles with respect to the horizontal axis of the three supports $\phi_1, \phi_2$, and $\phi_3$. Throughout the design iterations conducted, the combination of the lengths and angles with respect to the horizontal of each support had the most significant impact on the stresses experienced by the structure. For the angles as considered in section 3.6.2 Flexure Load, between 50° and 80°, given the minimum inner diameter to avoid Euler Critical loading as calculated in Table 7, the tube geometry chosen cannot be held at an angle less than 65° with respect to the horizontal. Additionally, the maximum clearance height to avoid bending at 65° is rather low, so a number of angles for $\phi_1, \phi_2$, and $\phi_3$ were explored in an attempt to achieve a clearance height $h$ of around two feet. In the schematics presented in Figure 12 below, two different configurations are shown. On the left, the outrigger assembly alone is shown with nondescript joint designs, and a conservative spread and horizontal angle. In this assembly, the lateral legs are at $\phi_1 = \phi_3 = 65°$ with respect to the horizontal axis (or 15° with respect to the vertical) and $\phi_2 = 87°$ with respect to the horizontal axis (or 3° with respect to the vertical, as marked). Additionally, the lateral legs are also at a 15° offset with respect to the middle support leg. This conservative assembly ensures a high ballast clearance and stiff structure that is unlikely to yield due to bending stresses, yet does not introduce a great deal of stability due to the narrow spread of the outrigger supports: i.e., knowing that perpendicular
supporting legs failed due to buckling in preliminary experiments, these legs would be primarily in compressive stress only when landing perfectly vertically, which is an assumption of this project and not entirely likely the case in reality. The schematics on the right-hand side of Figure 12 below depict a less conservative but more stable construction, as will likely be employed in the final flight-ready iteration of the outrigger design. In this case, the lateral legs are at $\phi_1 = \phi_3 = 45^\circ$ with respect to the horizontal axis (or $45^\circ$ with respect to the vertical) and $\phi_2 = 77^\circ$ with respect to the horizontal axis (or $13^\circ$ with respect to the vertical, as marked). Finally, the lateral legs are also at a $22^\circ$ offset with respect to the middle support leg.

![Figure 12: Schematics depicting degrees of freedom in assembly and in larger payload context](image)

4.3. Summary and Final Design

In summary of the iterative design process described in section 5, the final joint design and attachment geometry results in the renderings depicted below in Figure 13. The final design will
be machined out of 7075-T6 aluminum alloy, (due to its higher strength and machineability comparable to 6061-T6 aluminum alloy), and involves fillets for additional support and a single clevis attachment for the composite tube joint insert.

Figure 13: CAD renderings for final single clevis (right) and double clevis (left) node attachment designs

This design is a result from several prototyped iterations, considering ease of machining and overall strength, culminating in the same 3/8” hole diameter for each bolted insert between the two clevis portions, as well as a 3/8” corresponding clevis thickness on each portion. The fillet considerations will be discussed further in the technical specifications in section 5.3, but the detailed schematics for both full attachment designs can be found in appendix B.1 – Joint Attachment Design Drawings. The “node” portion of the joint, best reflected by a combination of the SPIDER joint aluminum insert geometry as well as the double clevis design represented on the right in Figure 9,(designs considered by Boeing in collaboration with NASA), yet there are some notable differences. Particularly, rather than has a double clevis design on the node and a
single clevis on the attachment component, this design reverses the two, with a single clevis node insert design.

![Figure 14: Final Tripod Outrigger Assembly Design (Left) with overall payload implementation (Right)](image)

Finally, the final outrigger assembly in its entirety is presented as rendered in Figure 14 above. Of note, the final leg lengths are 23.75 inches, as marked by dimension RD1, and have all the cross-sectional dimensions previously mentioned in Table 8: Final Tube Geometry Specifications on page 43. Furthermore, in the final designs the lateral legs are at $\phi_1 = \phi_3 = 65^\circ$ with respect to the horizontal axis (or $25^\circ$ with respect to the vertical, as marked with dimension RD3) and $\phi_2 = 75^\circ$ with respect to the horizontal axis (or $15^\circ$ with respect to the vertical, as marked with dimension RD2). Finally, the lateral legs are also at a $22^\circ$ offset with respect to the middle support leg, as marked with dimension RD4, all in the rendering to the left.
5. Design Details and Evolution

In the following sections, quantitative analyses verified by COMSOL™ FEA simulations for each stage in the iterative design evolution stages are detailed. The considerations within the scope of the project mentioned previously are assessed using FEA to determine the optimal design to satisfy the project goals in regards to: joint connection strength, attachment node design, assembly configuration stiffness, and finally overall outrigger configuration. The resulting final design is then explained in further detail, in the context of an analytical understanding of the engineering specifications and standards met for this project.

5.1. Bolted Connection

![Figure 15: Bolted carbon fiber assembly drawing, with steel cylinders acting as bolts for FEA simulation](image)

One consideration for joint design that is considered in prototype simulations during the design phase includes bolted aluminum or steel CFRP joint connectors (see Figure 15 above, and Appendix B.2 – Initial Joint Design Drawing – Bolted Insert for full schematic). Industry applications of bolted modular joint connections include aluminum screws rated up to 170,000
psi in tensile strength, as in Rock West Composites CARBONNect™ systems. As the location of drilling into carbon fiber involves high shear forces, these screws are the likely locations of structural failure in such designs, and thus computational models will determine the potential for such a joint design. The benefits of bolted connections are in the modular nature of customizable joints that can be easily assembled, yet difficulties arise if they are not withstand the loads involved in payload landing due to the nature of bolt stresses and potential carbon fiber delamination during drilling bolt holes. The stresses involved in the bolts themselves for a high load application such as in the SCoPEx project are enough to eliminate the potential for similar joint designs. Moreover, the SCoPEx team has since moved on from considering the CARBONNect™ system and delamination due to drilling into carbon fiber is outside of the scope of this project.

![Figure 16: COMSOL simulation of carbon fiber tube bolted with 4130 steel, withstanding a 5 kN axial compressive load](image)

COMSOL™ FEA simulation results regarding the stresses experienced by the bolted connection are included to address the feasibility of utilizing such a design, and different joint
attachment geometries were tested to verify the final conclusion that epoxied aluminum inserts were the most viable choice for this assembly geometry. First, bolted aluminum jointed were analyzed, as similar systems are available including Dragon Plate™ and Rock West Composites CARBONNect™ attachment systems, the latter having been explored by the SCoPEX team in initial design stages. The results of primary concern in this simulation lie in the high stresses experienced by the steel bolts in Figure 16 above. To facilitate appropriate comparison to the tube axial compression simulations, one end of the assembly (Computer Aided-Design (CAD) document imported from SolidWorks™), was kept fixed while the other side of the tube experienced a 5 kN axial compressive load. While the end-pieces on the tube were programmed with the material properties of Aluminum 7075-T6 alloy, the bolts had that of 4130 steel. The simulation was carried out considering solely linear deformations, with an “extremely fine” mesh (as defined by COMSOL™) due to the small bolt sizes relative to the remainder of the assembly.

The FEA simulation reports the “Von Mises Stress,” which is a value used to determine if a given material will yield or fracture, (typically for ductile materials, such as metals). The highest von Mises stress observed was \(1.4 \times 10^7 \text{ psi}\), or 14 msi, which is far beyond the ultimate tensile strength of steel, (approximately \(1 \times 10^5 \text{ psi}\), or 100 ksi,). While the simulation has a range up to 14 msi, the range for the colors represented in the figure are up to \(1.7 \times 10^6 \text{ psi}\), or 1.7 msi, to demonstrate that not only do the bolts deform as the stress is higher than yield, but also the composite tube would deform as the stress experienced is far higher than the maximum compressive strength of the tubes purchased, (\(9.6 \times 10^4 \text{ psi}\), or 96 ksi). In this case the full 5 kN load was experienced axially by the steel bolts, and the computational model verifies that even for the aluminum bolts rated to 170 ksi for ultimate
tensile strength as cited by Rock West Composites for their CARBONNect™ system, this joint connection method would in no way withstand loads experienced during landing.

5.2. Hose clamp Aluminum Connection

The second design assessed was inspired from performance bicycle construction with aluminum and carbon fiber components, where hose clamps and similar high-torque compression tools are used to create a gripping strength between aluminum and carbon fiber components, (see Figure 17 below for close-up view, and Appendix A.2 – Initial Joint Design Drawing – Hose Clamp for full drawing). Difficulties with this design will be discussed in the analysis section and revolve around the factors of surface area, torque, and frictional coefficient between aluminum and carbon fiber.

![Figure 17: Oversized aluminum slit and overlaid hose clamp joint design](image)

The oversized aluminum joints with slits, secured by hose tubes, were also simulated in FEA with the same boundary conditions and mesh parameters as the bolted aluminum joints.
However, in addition to the 5 kN axial compression, a radial compressive load of 4 N at each of the oversized aluminum joints, resembling the pressure that is placed upon the connections by high-torque hose clamps. The 4 N radial compressive force results from the high-torque hose clamp design achieving an allowable torque of between 8 and 28 N × m, around the 30 mm diameter tube. This resulted in a circumference of almost 100 mm, and with 8 to 28 N × m, would rest in a total force of between 1.3 and 4.4 Newtons around the circumference of the attachments when considering a factor of safety of 1.6. However, in both circumstances, the overall maximum stress experienced by the tube did not change significantly, the highest von Mises stress observed was $1.64 \times 10^6$ psi, or 1.6 msi significantly higher than the flexural strength of the composite tube (96 ksi), as seen in Figure 18 below.

![Figure 18: COMSOL™ Simulation of Hose clamp design, with 28 N-m of torque experienced at each end, as well as 5 kN axial compression](image)

5.3. Joint Design Details
When transitioning from attachment results in simulation to importing larger parts of the outrigger assembly geometry from SolidWorks™ to COMSOL Multiphysics™, one of the most difficult issues in simulation was verifying that the geometry was in fact behaving as expected with the provided finite element analysis boundary conditions. In particular, assuring the rotation of the aluminum insert with a single clevis as a separate entity from the double clevis aluminum attachment piece with respect to the pin that connected the two posed significant challenges. Understanding how COMSOL™ interpreted the imported CAD assembly lead to a more accurate and characteristic assembly, but the primary simulation results before then are depicted In Figure 19 below.

The first few attempts at simulating the insert geometry resulted in concentrated point stresses at the edges between the attachment planes and the clevis geometry, which inspired the incorporation of fillets. While this resolved some of the issues, the presence of concentrated

Figure 19: Attachment design simulation with rigid bodies, forms of contact between clevis designs unable to rotate. 3 mm fillets and 60° design (left) compared to 6 mm and 30° design (right)
regions high stresses persisted, as can be seen by the comparison between the 3 mm (left) and 6 mm (right) fillet stress reduction in Figure 19. However, the difference in maximum von Mises stress between the two resulted from a change in angle in the geometry. Not only did the base of the double-clevis aluminum attachment design deform according to the appropriate moment applied to the pin, but the surfaces of the clevis that were in contact in the assembly were treated as fixed, and not capable of sliding past one another. Thus, when the angle was decreased from 60° (left) and 30° (right), the maximum stress experienced increased, as the single clevis attachment node was in more compression under the same moment.

![Surface: von Mises stress (N/m²)](image)

*Figure 20: Rigid connected pin attachment assembly FEA simulation results*

Resolving this discrepancy involved the use of a rigid connector where the pin appeared, and applying the same moment resulted in a more characteristic rotation of the insert (Figure 20 above). However, this alone did not resolve the the maximum level of stress recorded by COMSOL™ which was $\sigma_{\text{max}} = 66.5 \, MPa$, higher than could be expected for a different joint angle configuration yet still well below the yield strength of 7075-T6 allow aluminum.
The difference between these two assembly movements will be further assessed in the response the remainder of the assembly, which has attachments most similar to these FEA simulations, as depicted in Figure 13 on page 52.

5.4. Assembly Configuration Stiffness

First attempts at entire-assembly simulations included the incorporation of imported assembly characteristics from SolidWorks™, which included the cubic payload representation, the three outrigger supports with joint insert designs, as well as the baseplate for the outrigger leg. These simulations did not utilize the “rigid connector” structural mechanics condition in COMSOL™, and thus experienced high levels of stress, due primarily to the inability of the aluminum insert designs to rotate along the axis of the pin. When only one outrigger leg was incorporated, a max surface stress of $\sigma_{max} = 3.58 \times 10^{12} Pa = 3.58 TPa$ was reported by the FEA analysis (Figure 22: Resultant concentrated stress due to forced flexure of composite tubes below). When all four outrigger supports were included, the maximum did not change
significantly, yet altering the range to display only those stresses up to $\sigma_{\text{max}} = 20.3 \text{ GPa}$ allowed for some more interesting visual results (Figure 22).

Figure 21: One Outrigger Entire Assembly Simulation

Figure 22: Resultant concentrated stress due to forced flexure of composite tubes
Upon reducing the range of stressed elements shown to a maximum nevertheless far beyond material yield, it was clear that the highest levels of stress experienced occurred at the pin designs and were due to shear upon the lateral legs, resulting from the orientation of the clevis designs at an angle. However, notable levels of stress over 2 GPa still appear in the CFRP tube itself, concentrated at the ends of the aluminum inserts within the tubes. This led to more careful analysis of the rotation behavior of the inserts, revealing that while the composite tubes were slowly bending due to their higher rigidity, high levels of stress concentrated about the rigid inserts. This was due to improper simulation parameters, not allowing for either deflection or stress to build in the inserts, as can also be seen in Figure 22 above and Figure 23 below.

Figure 23: Top-View of furthest deformed leg, demonstrating behavior of unstressed aluminum inserts, (blue) as well as pins in high shear (red)
Resolving the issue pertaining aluminum insert rotation involving the use of a rigid connector, such that the moment can be applied to one end of the outrigger assembly itself with a 15 $kN$ boundary load, while keeping the other end as a fixed constraint. The improvements in joint behavior, along with the removal of the domains representing the pins themselves, allows for a more appropriate representation of the deformation of the structure, as shown in Figure 24 below. The resulting maximum von Mises stress over the 100 step-size parametric sweep from 0 to the full 15 $kN$ boundary load was $\sigma_{\text{max}} = 273 \, MPa$, occurring in the aluminum base itself. Moreover, the modifications allowed for a great reduction in the stress experienced in the composite tubes, and still accounts for a FOS of nearly 2 when compared to the ultimate strength of aluminum $\sigma_{\text{yield, Al}} \approx 500 \, MPa$. 

Figure 24: Single outrigger assembly with baseplate, accounting for appropriate aluminum joint rotation
Furthermore, updated simulations were carried out, considering simplified joint designs without double clevis attachments on the lateral outrigger supports. What resulted were attachment inserts that were impractical to manufacture but made of solid 7075-T6 aluminum. The FEA analysis did not fully converge, (as can be seen at the top of the figure below, with the first parameter value still processing), and failed in part due in part to the rigidity of the aluminum inserts, but upon failure the maximum von Mises Stress was reported as $\sigma_{max} = 4.19 \text{ GPa}$, which is far higher than the ultimate strength of carbon fiber, as seen in Figure 25 below.

![Figure 25: Test a without double clevis design on lateral supports](image-url)
Finally updated simulations were carried out which more accurately represented the testing configuration that will be placed in the Instron. With the combination of compression and bending, the FEA simulation was first conducted with a simple 15° assembly with respect to the vertical compression capabilities of the Instron 3369 machine used. Furthermore, custom attachment pieces were developed, as detailed in section 6.2. These custom inserts were made of steel, and thus the appropriate material properties were added to the COMSOL™ file, and the simulation computed a parametric sweep for the stresses in the assembly experienced from a range of 0 to 10 kN. This range was established to both ensure that the minimum FOS was met, as well as to investigate the bending stresses encountered upon a two-leg landing scenario. As seen in Figure 26 below, the maximum von Mises Stress was reported as $\sigma_{max} = 126 \text{ MPa}$, which is far lower than the ultimate strength of carbon fiber ($\sigma_{max,cf} = 662 \text{ MPa}$) and that of steel ($\sigma_{max,steel} \approx 560 \text{ MPa}$). Importantly, the maximum stress experienced was also less than the yield strength of steel, ($\sigma_{yield,steel} \approx 460 \text{ MPa}$), thus multiple tests on the Instron machine could be conducted. On the right in Figure 26, the same simulation results are displayed with the range of stresses experienced by the components as ranging from nearly 0 Pa to merely $1 \times 10^7 \text{ Pa}$, or 10 MPa. This range facilitates a visualization of the bending stresses experienced by the composite tube, which were at most about 5 MPa.
While these stresses are far lower than the results derived in preliminary calculations for a tube alone in pure bending, (section 3.6.2 Flexure Load), the reason for this discrepancy is not intuitive and will need to be determined through verification of the COMSOL™ FEA model.

This discrepancy does not appear intuitive as steel was chosen with the consideration that using steel attachments in the Instron would force the aluminum/carbon fiber assembly to deform before the attachments. The elastic modulus (resistance to deformation) of steel $E_{steel} = 200 \text{ GPa}$ is far higher than that of the composite chosen, $E_{CF} = 15 \text{ msi} \cong 100 \text{ GPa}$ and of aluminum 7075-T6 $E_{Al} = 70 \text{ GPa}$. Such an evaluation of this discrepancy is further explored in section 6.4.
5.5. Final Outrigger Evolution

During the preliminary landing support construction, it became apparent that the precise angles and lengths of the support elements would further determine the outrigger structure and clevis components, as further explored in section 3.6. Moreover, more careful considerations of bolt and insert dimensions were required to add strength to these components, and Figure 13 demonstrates the final designs including fillets of 0.5", as well as holes placed about 2 diameter lengths away from the nearest edge of components to minimize risk of tear-out. These considerations arose during discussions with SCoPEx lead Mechanical Engineer Michael Greenberg, and design changes were made as they became apparent during prototyping and simulations. In those same conversations, the notion arose that a two-legged support structure would not provide enough lateral rigidity, and thus the final support design incorporates three different support elements, all of which have the same double and single. As discussed earlier in further detail with the free body diagrams in section 3.5, the two supports to either side of the middle support leg are symmetrical, and they will be connected to the two corresponding base edges of the payload. The middle support connects to the vertical baseload edge, (modeled as a cubic 80/20 aluminum structure), and the final lengths and angles of these supports are critical degrees of freedom in designing the assembly to meet engineering specifications.

5.5.1. Configuration Design Decisions

In attempting to mate the CAD outrigger assembly with a cubic payload design, it became apparent that while maintaining all three carbon fiber lengths the same increases modularity, the geometric considerations to allow for such a construction are non-trivial. As such, for the outrigger supports to be symmetrical in every way with respect to the central
support both a telescopic and cantilever potential design were explored, attached in various orientations to the payload itself. As seen in section 5.4 Assembly Configuration Stiffness, COMSOL™ FEA revealed that these design decisions resulted in high levels of stress along the clevis pin; this can be seen by the configuration of the clevis attachments on the lateral legs, which were no longer parallel to the vertical load, and could not rotate freely (Figure 27 below).

These initial design considerations involved the utilization of primarily 0.75 m composite tubes, purchased prefabricated and cut to that specific length, with the final dimensions and angles as specified initially in section 4.3 Summary and Final Design. In these preliminary designs, the composite tube diameter considered was 30 mm, and the central outrigger support was arranged to be at an angle of 34°, with the lateral supports in the telescopic design at a 56°. The cantilever design involved two lateral tubes of 0.375 m in length, 45° each from the central support laterally, and while they involved less composite material, the cantilever design itself poses a variety of complexities involved in the consideration of constructing such an assembly, which will be discussed in the following section.
5.5.2. Design Decisions: Final

For several reasons upon the exploration of a telescopic and cantilever design, the latter was abandoned in favor of a final telescopic design. The principle influences of this decision involved the presence of higher Von Mises Stresses ($\sigma$) in the cantilever design lateral legs, as well as the impracticality of creating composite tubes of different lengths for assembly. While these could be purchased in shorted lengths from composite tube manufacturers, shortening or bisecting the existing length tubes would risk delaminating the carbon fibers and a compromising the strength and material properties of the fibers. This additionally impacted the project specification and goal of maintaining modularity, as separate tube lengths would be needed for repairs on distinct outrigger supports (lateral or central). Furthermore, modularity and stiffness
would be impacted by the necessary design of the cantilever lateral base attachment, which would need be compatible with the double clevis attachment designs utilized throughout the assembly. Additionally, minimizing the possibility of galvanic corrosion with the added aluminum attachment resting atop the central outrigger support tube poses a series of difficulties outside of what is done in similar applications. For these reasons, in addition to the computational analysis, the final tripod outrigger assembly design was chosen (Figure 28, below).

![Figure 28: Final Tripod Outrigger Assembly Design (Right) with simplified COMSOL™ model including one corner of cubic payload (Left)](image)

6. Final Design Evaluation

6.1. Assembly Fabrication Procedure

6.1.1. Epoxy and CFRP surface preparation

6.1.1.1. Surface preparation materials
For materials in order to optimally prepare the bonding surfaces of carbon fiber and aluminum, the most commonly available aerospace grade application tools were considered. Upon evaluating the strength and sizes of commercially available epoxies, two epoxies were primarily considered for connecting composite tubes to the aluminum inserts for application involved in this design: 3M™ Scotch-Weld™ Epoxy Adhesive 2216 B/A Gray and LOCTITE® Hysol® E-120HP™ [11] [12]. Of note, the former was utilized by the SPIDER balloon-borne telescope team, and the latter by the X-Calibur telescope team. Both are flexible, two-part, room-temperature-curing epoxies with high peel and shear strengths. The aerospace-grade adhesives differ only in their working time, availability, and suppliers. While the 3M™ epoxy has a working time of 90 minutes, the LOCTITE® epoxy has a slightly higher working time of 120 minutes. Both cure in a matter of hours yet are recommended for 24-48 hour setting times and are superior in shear strength, as well as resistant to environmental effects such as vibration, temperature, and deformation, particularly in the form of contraction and expansion between plastics (such as CFRP) and metals. Due to the availability of LOCTITE® Hysol® E-120HP™ in smaller volumes (particularly in 50 mL amounts) as well as widespread applicator-gun devices for these epoxies, the LOCTITE® product was chosen and purchased for this application, (Figure 29, below).
The major concern when using an epoxied connection between CFRP and aluminum is galvanic corrosion which can occur between the two substances due to their high contact potential. Other materials, such as titanium, have corrosion-preventing features that limit this concern but many teams throughout the literature review have avoided such issues by ensuring electrical insulation between the CFRP and aluminum material surface areas. Some well-known methods for limiting the possibility of threats to joint structural integrity as well as compromised stiffness are possible using surface preparation materials and methods laid out by composite manufacturers, such as Rock West Composites. First and foremost, it is essential to abrade the bonding surfaces of both the aluminum and carbon fiber, either by hand or sand-blasted by an abrasive comparable to 100 grit sandpaper to remove the corrosive oxidizing top layer in the metal and carbon fiber to remove various impurities that may exist on the surface.
The material used to abrade the inner diameter of the composite surface is what’s known as an “Abrasive Flex Hone,” which is 120 grit sandpaper in a cylindrical series of beads, ranging from 0.876” to 2” in diameter, with the specific hone purchased for this project perfectly sized to sand the inside of a tube with inner diameter of 1.25”. The aluminum need be similarly sanded with 120 grit sandpaper, and both surfaces are then be degreased with isopropyl alcohol or acetone and cleaned thoroughly. It is most common to bond to the ID of carbon tubes rather than the OD given the tighter dimensional tolerances (+/-0.004” vs. +/-0.012”, respectively). Once the parts have been allowed to dry, glass spheres known as Bond Line Controllers (BLC) are used to minimize galvanic corrosion between two dissimilar materials (generally metals including aluminum and carbon fiber). BLC ensures the parts retain a concentric bond and are offered in various diameters that are intended to be added into the adhesive/epoxy during mixing. Typically selecting a Bond Line Controller based upon the gap between two parts specified to a machinist involves a bond line (gap) of around 0.008” to 0.013” which would use a 0.0070” and 0.0117” max diameter BLC respectively. Additionally, for every 3 ounces of adhesive, proper concentricity is ensured with approximately 2 grams of BLC, and as 50 mL of adhesive are in consideration, (i.e. 1.69 US fl oz.), 2g of BLC is more than enough [13]. Thus, for a 1.25” tube
inner diameter and a 0.008” specified bond gap, products Flex-Hone Bore 1.25” (31.7MM) - 120 GRIT, (Sku: 1106) and 0.0070” maximum diameter bond line controller (Sku: 1031-GROUP) were purchased, as shown in Figure 30 above.

With all these dimensions in mind, the final considerations for maximizing bond strength were gathered from previous research on the bond strength between carbon fiber–reinforced plastic tubes and aluminum joints. Literature reviews revealed that increasing the length of an aluminum joint insert only increases the bond strength with significant returns up to approximately 50 mm, and the bond gap increases the ultimate tensile strength of an aluminum-CFRP joint up to about 0.2 mm, or 0.008”, as shown in Error! Reference source not found. below [14].

![Graph](https://example.com/graph.png)

*Figure 31: Ultimate Tensile Force for aluminum-composite joints as a function of bond layer thickness (left) and bond layer length (right) [14]*

6.1.1.2. Surface Preparation Procedure

Finally, the process for limiting the risk of galvanic corrosion through proper surface preparation is detailed below, and is inspired by a source of similar processes, including that of the SPIDER and X-Calibur teams, as well as from recommendations by the composite supplier for this project, Rock West Composites™. The procedure must be done in a clean and well-ventilated environment, with proper protective equipment including gloves, a respirator, and safety glasses:
1. The bonding surfaces of the aluminum inserts need be hand-sanded by at least 100 grit sandpaper to remove the corrosion later in the metal and various impurities that may exist on the surface.

2. Additionally, the inside diameter of the CFRP tube can be sanded with Rock West Composites™ 120-grit abrasive flex hone, which is featured in the budget section of this project.
   a. Both surfaces are sanded to as precise a 0.008” bond gap as possible, without damaging the outer layer of the CFRP fibers in the tube.

3. Activate the two-part epoxy by combining the two elements of the formula using the epoxy applicator.

4. Add 0.007” Rock West Composites™ bond line controller spherical glass beads to limit non-concentric bond and possibility of galvanic corrosion.
   a. Utilize 2 grams of this bond line controller for every 3 oz of adhesive formula.

5. Place the activated mix in a vacuum desiccator for up to 10 minutes to remove any trapped air in the formula, when air bubbles stop rising to the surface.

6. Once the epoxy is prepared, the abrasive step is followed by wiping the surfaces down with water and then degreasing, cleaning both the aluminum and carbon fiber surfaces thoroughly with acetone, see Figure 32 below.
   a. Let the solvent evaporate and bond within 2 hours to keep the aluminum from forming an oxide layer.
7. Cover both surfaces distributing the glue in uniform thin layers.

8. Bring together both pieces, slightly rotating the insert.

9. Install the tubes in a fixed jig structure, to allow for 48-hour curing (for details on the jig utilized, see 6.1.2 below) in a fume hood or otherwise well-ventilated area,
Finally, the epoxy chosen was LOCTITE® Hysol® E-120HP™, due to availability in a variety of different sizes, as well as an application device available. The volume of epoxy needed results from the bond gap and carbon fiber tube dimensions, thus:

- For a 0.011" maximum allowable bond gap, (keeping in mind that only for 0.0035" over the maximum BLC diameter will the BLC spheres be able to allow concentricity), and using the 0.007" bond line controllers discussed above [15]
- A 1.25" outer diameter of the tube, for a 2” insert bond length results in a volume of
  \[
  V = \left(\frac{ID}{2} - \left(\frac{ID}{2} - \text{bond gap}\right)^2\right) \times \pi \times L = \left(\frac{1.25}{2}\right)^2 - \left(\frac{1.25}{2} - 0.011\right)^2) \times 
  \pi \times 2" = 0.0856 \text{ in}^3(\text{cubic inches) per insert}
  \]
  - For 8 total inserts, this translates to 0.685 in³
  - Translating to 11.23 mL, less than a quarter of the 50 mL two-part epoxy

Furthermore, summarizing the relevant project technical specifications so far:

**Table 11: Technical specifications of a single landing support leg, based on preliminary findings and results**

<table>
<thead>
<tr>
<th>PROPERTY</th>
<th>REQUIREMENT</th>
<th>REASONING</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>Payload is 600 kg overall, ≤ 10 kg for all four landing supports</td>
<td>When compared to aluminum, composites are approximately half as dense, with over three times the tensile strength, requirement based on achieved weight for SPIDER frame and proportion of carbon fiber used</td>
</tr>
<tr>
<td>Crush Pad Size</td>
<td>0.636 meters on each side, 0.4048 meters²</td>
<td>Calculations for previous table for two-leg landing scenario, resulting in a 25” square crush pad</td>
</tr>
<tr>
<td>Cross sectional area of tubes</td>
<td>0.2649 in² = 159.3 mm² &gt; 20.6 mm²</td>
<td>To withstand axial compressive load without reaching yield strength, given the selected tube dimensions, this is greater than the minimum cross-sectional area required</td>
</tr>
<tr>
<td><strong>Tube Inner Diameter</strong></td>
<td>1.25”</td>
<td><strong>Chosen CFRP tube</strong></td>
</tr>
<tr>
<td>-------------------------</td>
<td>-------</td>
<td>---------------------</td>
</tr>
<tr>
<td><strong>Tuber Wall Thickness</strong></td>
<td>0.06”</td>
<td>10 layers of unidirectional carbon fiber prefabricated layouts</td>
</tr>
<tr>
<td><strong>CFRP Tube Length</strong></td>
<td>23.75”</td>
<td>To allow for minimum ballast clearance</td>
</tr>
<tr>
<td><strong>Temperature</strong></td>
<td>$200K \leq T \leq 323K$</td>
<td>If adhesive is used, epoxy must cure at around room temperature, yet surface temperatures at launch site can reach $T = 50°C$, and atmosphere can reach $T = -70°C$</td>
</tr>
<tr>
<td><strong>Load</strong></td>
<td>15 $kN$ per outrigger</td>
<td>Four Outrigger systems, at 10 $G$ vertical loads</td>
</tr>
<tr>
<td><strong>Stiffness</strong></td>
<td>Withstand $\geq 10$ $G$ vertical load</td>
<td>Chute Shock / CSBF requirements</td>
</tr>
<tr>
<td><strong>Cost</strong></td>
<td>$360$ for Composite components</td>
<td>Based on composite material costs and dimensions, solely including composites, including costs of materials and segment cutting</td>
</tr>
<tr>
<td><strong>Altitude</strong></td>
<td>65,000 $ft$</td>
<td>When considering UV radiation, composite fibers should be resistant to physical properties in this part of the atmosphere</td>
</tr>
</tbody>
</table>

6.1.2. Jig Design and Manufacture

Below the design of an 80/20 jig design is depicted, in full construction including two CFRP tube layouts placed in the jig to allow for curing of four epoxied aluminum inserts at a time. This allows for the completion of two of the four purchased CFRP tubes at once, which accelerates the process of assembly before curing. The jig itself is comprised solely of aluminum, and includes two 24” 80/20 extrusions (1” by 1” cross-section) with two aluminum blocks precisely machined in order to allow for 3/8” steel bolts to be placed through the single clevis
inserts and into the aluminum bases such that the pieces are kept in place and the inserts are kept parallel to one another (Figure 34 below).

![Figure 34: Jig Design, capable of keeping two CFRP tubes in place while epoxied joints cure](image)

Additionally, 80/20 aluminum brackets are used to join the aluminum base pieces (5” × 2” × 1.65”) to the 80/20 extruded pieces of 24” in length. One side uses straight brackets, while the other takes advantage of 80/20 aluminum L-brackets in order to allow for dynamic movement of the 3/8” holes in the jig according to the tolerances involved with the manufacturing of both the tube and the inserts. While the tubes were specified to be cut to a 23.75” length, the tolerances dictated for roll-wrapped unidirectional tubes (such as the ones employed in this assembly) under 48” inches of length are +/− 0.06”, and thus they could vary in terms of exact distances from one clevis pin to another.

6.2. Testing Protocol
For testing of these CFRP tubes along with epoxied aluminum inserts, three main tests will take place in the Instron Model 3369 available in the Harvard SEAS department’s Active Learning Labs (ALL). The ALL Instron machine is equipped with two set of clamps, one with a maximum grip width of approximately ½", which can withstand up to 5 kN of vertical load, and the other with a ¼" grip width, which is designed to withstand the full 50 kN capabilities of the machine’s load cell (Figure 35 below).

![Figure 35: 50 kN rated Instron Clamp](image)

For the purpose of experimentally testing the procedural strength of epoxied aluminum joints, a series tensile tests were conducted, and in order to test the crushing capabilities of the assembly, a series of compression tests were conducted. In order to test the assembly to failure, a pair of the eight single clevis 7075-T6 aluminum inserts are milled to ¼” in width rather than the 3/8” widths (on the parallel faces of the single-clevis design) specified originally to the machinist. The aluminum inserts are then placed into the ¼” clamps at both ends, tested in both compression and in tension axially, up to 50 kN until failure occurs. The tests thereafter involve a more complex procedure/assembly, as is pictured in Figure 36 below.
This testing assembly involves the use of two customized steel attachment designs, which are made according to the setup of the Instron 3369 machine in the ALL. At the top of the single clevis aluminum insert, one steel double-clevis with a ¼” extrusion will be pinned to the insert and inserted into the 50 kN rated clamp. On the other end, the steel baseplate will be bolted into the Instron machine base, such as to act as a rigid body and allow for a combination of bending and compression deformation. This test will most accurately simulate the failure modes of the entire outrigger assembly, and detailed schematics of the customized steel attachments are shown in Appendix B.4 – Custom Steel Instron Attachment Component Designs. Furthermore, a strain rate protocol was adapted of 0.001/s, which was the lowest value tested in literature examining the dependence of tensile strength of carbon fiber epoxy laminate bonded with aluminum [16].
This strain rate translates to a fraction of the total assembly’s length deformation per second. With a strain rate of 1/s meaning the Instron would deform the approximately 24" or 610 mm assembly at a rate of 610 mm per second, a strain rate of 0.001/s meant a protocol of 0.61 mm/s was utilized in Instron testing. Furthermore, the dependence of material properties in tension on strain rate is depicted in Figure 37 below, as demonstrated in the paper “Effect of strain rate on tensile behavior of carbon fiber reinforced aluminum laminates,” with yield of the carbon fiber matrix occurring when the slope decreases in each case, and failure occurring shortly thereafter [16].

![Stress strain relationship, and dependence on strain rate](image)

*Figure 37: Stress strain relationship, and dependence on strain rate [16]*
6.3. Final Design Analysis

The final testing procedure detailed above was carried out with all four 23.75" tubes procured, with the same steel base plate and assembly utilized for each tensile testing and combined bending/compression test (Figure 38 above). While the original intention was to perform a tensile test axially with two of the tubes, in order to investigate the pull-out strength of the epoxied connections, the added height of the 50 kN Instron clamps meant that the tube in its full length did not fit into the height of the Instron 3369 machine. Thus, the first test was
conducted axially in tension with the top clamp rated to 50 \( kN \) and the other base clamp only rated to 5 \( kN \), due to its reduced height. While the Instron is able to report deformation (as the independent variable, set by strain protocol of 6.1 \( mm/s \)) along with the corresponding load needed to deform the structure (as the dependent variable, in units of Newtons), the axial tensile test is not included as it was only tested to 5 \( kN \). This was to reduce any risk of damage to the weaker clamp, and was conducted to validate experimentally that the assembly was capable of withstanding a \( FOS \) of at least 1. Furthermore, tests were conducted in the 15° testing assembly shown in Figure 38, such that two 50 \( kN \) clamps could be used and ideally the test could be performed to failure. The idea behind using such an assembly, other than necessity of accommodating the full height and range of loads for the composite tube, was to simulate a more realistic upward vertical load that might be experienced during chute shock. In this testing procedure, the same strain rate of 0.61\( mm/s \) was used, yet the assembly was only tested to 15 \( kN \) rather than to failure of the epoxied composite connections. The reason for stopping at such a load was due to the observed deformation of the base steel plate, which was bolted into the Intron’s base itself. Each time deformation occurred, there was an audible creaking that was due to the stripping of the threaded metric bolts used to hold the base plate down. Nevertheless, although the assembly was not tested to failure, the \( FOS \) achieved by simulating one outrigger assembly to 15 \( kN \) was 3.0, far above the desired minimum \( FOS \) of 1.6 in tension. Finally, the results of the Intron 3369 assembly testing in tension are summarized in the figures below. First, in Figure 39 below, the raw data output by the Intron is plotted, namely the displacement in the positive Y-direction (in units of \( mm \)) versus the corresponding load required to displace the assembly (in units of \( Newtons \)). The first trial of purely axial tension is compared to the average of the following four trials, one for each tube, of the tests in tension of the entire assembly.
However, in order to compare these plots to COMSOL™ simulations, as well as to the literature values included in Figure 37, the displacement in the positive Y-direction must be converted to strain, and the load must be converted to stress. In the case of the axial tension tube, strain is simply the displacement divided over the overall length, and stress is solely axial, thus it can be calculated as the pressure experienced within the tube (i.e. the force divided by the internal area of the tube). However, for the 15° assembly layout, the strain is a fraction of the axial length, and thus a factor cosine of 15° (0.966) must be included. This does not significantly impact strain, yet the addition of bending stresses in addition to pressure does complicate the calculation of von Mises Stresses. As included below in Equation 10: Calculation of von Mises Stresses, it is a weighted average of the three principle stresses experienced by a body.

Figure 39: Tension Trial setup comparisons (Load vs. Displacement)
\[ \sigma_e = \left[ \frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_2 - \sigma_1)^2}{2} \right]^{\frac{1}{2}} \]

*Equation 10: Calculation of von Mises Stresses*

In this case, a simplifying assumption that can be taken is that the primary stresses (i.e. bending and pressure) are only two, thus \( \sigma_1 = \text{bending stress} \), and \( \sigma_2 = \text{pressure} \), resulting in the following equation:

\[ \sigma_e = \left[ \frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2)^2 + (-\sigma_1)^2}{2} \right]^{\frac{1}{2}} \]

Utilizing this assumption, the following stresses are calculated (shown in Figure 40 below)

**Tubes in Tension**

![Tension Trial setup Stress-Strain comparisons](image)

*Figure 40: Tension Trial setup Stress-Strain comparisons*

Of note, the range of values for both stress and strain are quite similar to what can be expected and what has been reported from the literature, as exemplified by Figure 37. The same calculations were carried out for the compression testing, of which there were 9 trials, as can be seen in Figure 41 below.
Figure 41: Compression Testing results, including all trials and the FOS threshold

Notable differences between the compression tests and the tension tests are not only the number of individual trials conducted for each tube but also the importance of the test results to the project goals. While indeed testing for tension is important to assess the strength of the epoxied connections, it was clear that the procedure for surface preparation and adhesive used well compensated for the $FOS$ desired in chute-shock. Additionally, as chute-shock is independent of how many outrigger support legs touch the ground, it will be far better distributed
amongst the four outrigger legs, and thus a $FOS$ greater than 1.6 is not of a significant concern to validate. However, in compression there is critical concern of reaching and in fact surpassing the $FOS$ of 1.6, which can be represented by achieving at least $1.6 \times 5000N = 8\, kN$, and thus the $FOS$ threshold of $8\, kN$ is included in Figure 41 above. Furthermore, these tests were only conducted up to about $10\, kN$, again to prevent damage to the Instron machine due to the unique challenges posed by the strength of the assembly itself. Upon reaching the $FOS$ threshold, the plots are seen to increase in slope. This does not occur because yield was reached. Instead, the Instron machine itself deformed at this point, due to the assembly (while in bending), being too rigid to compress at a $15^\circ$ angle as originally and instead causing the Instron clamp to deform. The overall shape the Instron took when the testing was halted is depicted in Figure 42 below. While it can be seen that the assembly itself is clearly in some minimal range of bending, because the carbon fiber tubes are rated to the same rigidity and strength in both compression and bending, the Instron began to deform away from the neutral axis of the tubes. In fact, the clamp can be observed tending to the right, seemingly approaching an almost perpendicular moment arm, such that half-way in between axial compression and perpendicular bending, the Instron would most easily deform the structure. As both axial and bending rigidity were matched, the Instron tended to balance the two forces, as the path of least resistance, yet this caused concern of damaging the machine. It was clear from these observations that the assembly had far surpassed the desired engineering specifications, and a more detailed exploration of this notion is held in the following section, comparing the computational simulation results to the output of experimental testing.
6.4. Verification of COMSOL™ Model

In order to best assess and retroactively evaluate the COMSOL™ models that have been used in design iterations to best commit to this final design, the Instron output information first had to
be converted to both strain and von Mises stresses, as discussed above. Furthermore, the Testing assembly mentioned above had to be imported as a CAD document into the existing COMSOL™ FEA model, and then the relevant data had to be exported as a list of deformations and stresses.

The optimal method for extracting such precise data from the simulation involves taking advantage of the “derived values” function in COMSOL™’s study results sub-menu, upon the FEA study’s completion. Utilizing the nodal average, an integrated average can be taken along the entire length of the CFRP tube, to best obtain accurate data about deformation in any direction of interest, strain, and stress, as well as a myriad of physical properties not within the scope of this project. From the simulation depicted in Figure 44 below, the deformation in the negative Y-direction (in units of mm) was extracted using COMSOL™. This simulation involved all the setup procedural steps discussed in section 5.4, other than the rigid connector.

![Figure 43: Comparison of Load versus deformation of each trial and COMSOL™ FEA Simulation](image_url)
condition on the bolted connection holes. This was omitted as, when included, the steel alone
deformed and the tube did not bend properly, as discussed and demonstrated in Figure 26 on
page 67. In that figure, the simulation results depicted a visualization of the bending stresses
experienced by the composite tube, which were at most about 5 MPa. Moreover, in those series
of simulations, the composite tubes did not experience a great concentration of stress, yet the
maximum von Mises Stress occurred in the steel and was reported as \( \sigma_{max} = 1.26 \times 10^8 \text{Pa} = 126 \text{MPa} \).

\[ \text{Figure 44: COMSOL™ Simulation of a 10 kN axial compression on the assembly} \]
In this simulation, the aluminum inserts experienced the highest level of von Mises Stresses, at $\sigma_{\text{max}} = 5.09 \times 10^9 Pa = 5.09 \text{ GPa}$, which appears reasonable as the aluminum inserts have the lowest modulus of rigidity, and thus are more susceptible to deformation under the same load than steel or carbon fiber. Furthermore, this simulation results depict a visualization of the bending stresses experienced by the composite tube of about 2.5 GPa, which is above the yield of the CFRP tubes. When extracting the nodal integrated averages of this simulation data, the extension in the negative Y-direction reached approximately 10 mm rather than the 22 mm experienced in simulation. However, when the rigid connector boundary condition was included, the deformation in the Y-direction was a fraction of one millimeter, about $2.5 \times 10^{-4} \text{ mm}$, as the composite tube was not capable of bending, unlike the simulations with the aluminum joints alone as described in section 5.3 Joint Design Details.

Figure 45: Final comparison of simulations to experimental results, with Stress-Strain relationships
With the final simulation conditions established, the stress for the Instron experimental testing was calculated based on the load for each deformation as in the previous section, and this was compared with the extracted strain and von Mises Stresses from the simulation, as plotted in Figure 45 above. “Simulations 1-3” represent three different mesh sizes used for the same COMSOL™ model, with simulation 1 having the most coarse of the three meshes for the composite tube (“normal” mesh size), simulation 2 having a “finer” mesh size and simulation 3 with an “extra fine” mesh size for the CFRP tube, according to COMSOL™’s definitions of mesh sizes within the program.

The three simulations were included to demonstrate both that the integrated volumetric average is affected by the number of mesh elements and nodes in the FEA simulation, yet Figure 46 above demonstrates that as the mesh size increased, the maximum Von Mises stress converged to about $\sigma_{\text{max}} = 5.13 \times 10^9 Pa = 5.13 GPa$. Furthermore, the COMSOL™ model computational results are well validated in comparison to the Instron output assembly experimental results: while the ranges of stresses and strain differ between the two, the consistency and magnitude of the slope on the stress-strain curve is well maintained. The consistency of the slope leads to the
belief that the composite tube does not yield, as was observed in experiment. Additionally, the magnitude of the slope on this curve informs the effective resistance to deformation, what is referred to as the effective Young’s Modulus, or the effective Elastic Modulus. As mentioned previously, the elastic modulus (resistance to deformation) of steel \( E_{steel} = 200 \text{ GPa} \) is far higher than that of the composite chosen, \( E_{CF} = 15 \text{ msi} \approx 100 \text{ GPa} \) and of aluminum 7075-T6 \( E_{Al} = 70 \text{ GPa} \). Moreover, the effective elastic modulus of the entire structure can be determined from the slope of the lines in Figure 45, and is approximately \( E_{\text{effective}} = \frac{4 \times 10^8 \text{ Pa}}{0.01} = 40 \text{ GPa} \). This accounts for the ease of deformation in rotation, as well as the deformation that occurs in the aluminum and composite, which have the lower elastic moduli.

6.5. Quantitative Evaluation of Prototype

Based on the overall project technical specifications, those which could be met by this project alone are compared to the achieved values in Table 12 below.

<table>
<thead>
<tr>
<th>PROPERTY</th>
<th>REQUIREMENT</th>
<th>MET</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>&lt; 10 kg for all four outrigger landing supports</td>
<td>Each outrigger leg (with inserts) has mass of 0.41 kg, giving the total outrigger assembly a mass of 1.23 kg each, or 4.92 kg total without attachments</td>
</tr>
<tr>
<td>Cross sectional area of tubes</td>
<td>( A \geq 0.032 \text{ in}^2 )</td>
<td>Minimum Cross sectional area required to withstand compressive load axially without buckling surpassed, at ( A = 0.2649 \text{ in}^2 )</td>
</tr>
<tr>
<td>Tube Inner Diameter</td>
<td>( 1&quot; &lt; ID &lt; 3&quot; )</td>
<td>Minimum reasonable diameter to minimize assembly cost and maximize cross-sectional area ( ID = 1.25&quot; )</td>
</tr>
<tr>
<td>Tuber Wall Thickness</td>
<td>( T \geq 0.06&quot; )</td>
<td>Standard minimum thickness for high-strength and bending applications of carbon fiber reinforced</td>
</tr>
</tbody>
</table>
polymer tubes, $T = 0.06''$, containing 10 prefabricated carbon fiber layout layers

<table>
<thead>
<tr>
<th>CFRP Tube Length</th>
<th>20” $\leq L \leq$ 40”</th>
<th>$L = 23.75''$, as determined by ballast clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ballast Clearance</td>
<td>18” $\leq H \leq$ 36”</td>
<td>$H = 21.5''$, maximum ballast clearance with lengths and assembly angles discussed well within project need range</td>
</tr>
<tr>
<td>Load</td>
<td>15 kN per outrigger system</td>
<td>Single leg withstood slightly over 10 kN in compression and 15 kN in tension, rather than the 5 kN caused by chute-shock and landing</td>
</tr>
<tr>
<td>Stiffness</td>
<td>Withstand $\geq 10 G$ vertical load</td>
<td>Withstood equivalent 20 G vertical load</td>
</tr>
<tr>
<td>FOS</td>
<td>Minimum $FOS = 1.6$</td>
<td>$FOS \cong 3.0$ in tension, and $FOS \cong 2.0$ in compression</td>
</tr>
<tr>
<td>Cost</td>
<td>$$360 Material Cost</td>
<td>$$195 for Outrigger System legs without attachments or labor costs</td>
</tr>
</tbody>
</table>

For meeting these specifications, the flight-ready outrigger assembly developed and detailed in this project is incorporated into the SCOPEx team’s most recent design of the payload in its entirety, as rendered in Figure 47 below.
Figure 47: Current rendering of SCoPEx payload, with Outrigger design incorporated into base supports as well as propeller and top gondola support elements

7. Budget

7.1. Project Budget

Table 13 below represents the project budget, with materials as well as labor costs involved for project design fabrication. The material dimensions and specifications that have been finalized through research and simulations. Unknown costs for aluminum material and
development costs have been determined via further discussion with Stan, (the primary machinist for Harvard research team and graduate students), and concerns about flexural strength of composite were resolved with more quantitative calculations and qualitative discussions with Rock West Composite™ sales representative Adam Creer [15].

Table 13: Bill of Materials and overall project budget

<table>
<thead>
<tr>
<th>BOM (Bill of Materials)</th>
<th>Unit Cost $</th>
<th>Unit</th>
<th># of Units</th>
<th>Total $</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unidirectional Carbon Fiber Tubes - 1.250 X 1.370 X 96 Inch&quot; Original Length, cut into four 23.75&quot; tubes</td>
<td>$ 197.99</td>
<td>Four tubes</td>
<td>1</td>
<td>$ 211.99</td>
<td>Four Outrigger supports for experimental testing, cut from an originally 8 foot composite tube</td>
</tr>
<tr>
<td>7075 T651 Aluminum Joint Inserts (1.2395&quot; Diameter x 3.69&quot; length)</td>
<td>$ 202.67</td>
<td>One Insert</td>
<td>9</td>
<td>$ 1,824.00</td>
<td>This price reflects a total 20 hours ($1700) of machine shop labor costs, and $124 in material costs for 9 aluminum 7075 alloy inserts of a high tolerance.</td>
</tr>
<tr>
<td>Loctite Adhesive Applicator - Epoxy dispensing gun</td>
<td>$ 45.19</td>
<td>One Applicator</td>
<td>1</td>
<td>$ 45.19</td>
<td>Applicator ensuring epoxy/resin ratio is precisely 2:1</td>
</tr>
<tr>
<td>Loctite E-120HP Hysol Epoxy Bore Grit Sander (31.7 mm or 1.25&quot; in diameter) 120 grit flex-hone bore Bond line controller (0.007&quot; max diameter glass beads)</td>
<td>$ 23.21</td>
<td>50 mL of epoxy</td>
<td>1</td>
<td>$ 23.21</td>
<td>Aerospace grade epoxy used for assembly insert bonds</td>
</tr>
<tr>
<td></td>
<td>$ 26.59</td>
<td>One Sander</td>
<td>1</td>
<td>$ 26.59</td>
<td>Abrasive hone for exact inner diameter of composite tube, 120 grit</td>
</tr>
<tr>
<td>Aluminum Stock</td>
<td>2</td>
<td></td>
<td></td>
<td>$ 0</td>
<td>Two Aluminum Stock Pieces (2&quot;x2&quot;x5&quot;)</td>
</tr>
<tr>
<td>Steel Stock Plate</td>
<td>1</td>
<td></td>
<td></td>
<td>$ 0</td>
<td>Steel Stock Plate (3.5&quot;x12&quot;x0.375&quot;)</td>
</tr>
<tr>
<td>Steel Stock Rod</td>
<td>1</td>
<td></td>
<td></td>
<td>$ 0</td>
<td>Steel Stock Rod (0.5” radius x 7.5” length)</td>
</tr>
<tr>
<td>Small Steel Stock Pieces</td>
<td>4</td>
<td></td>
<td></td>
<td>$ 0</td>
<td>Four Steel Stock Pieces (0.75”x1.75”x3.5&quot;)</td>
</tr>
</tbody>
</table>
7.2. Mass Production Budget Exploration

While the material cost of the project well met the speciation for limiting the cost and maximizing the modularity of the outrigger assembly components, the material labor costs from the schematics specified by the machinist accounted for 80% of the budget overall. In the future, and in further flight-ready stages, this could be remedied by a longer time-frame and better knowledge of which dimensional tolerances are unnecessarily precise. The specification of incredibly tight tolerances on the outer diameter of the aluminum inserts was of a conservative concern to assess the strength of the composite tubes alone, with the worry that the bonded aluminum inserts would fail well before the composite tubes, but this was not the case. Thus, in future production, mass-produced aluminum inserts might be more easily machined if the schematic specifications are more carefully thought out, acknowledging that the outer diameter of the inserts can range from 0.007 to 0.011 inches can be taken into more consideration than what was hastily decided in this project due to the testing timeline and machining time.

8. Conclusions & Future Work

In conclusion, the technique of SRM involving the addition of aerosols to the Earth’s atmosphere to potentially mitigate ozone depletion and partially offset the positive resultant radiative forcing, as explored by the SCoPEx research team is facilitated by the design and fabrication of a rigid composite outrigger system. Safely tracking the impact of aerosol loading
on a large scale involves the use of weather balloon-borne sensors and relies on the design of a durable and modular landing gear system. The design and fabrication of a testing assembly involving lightweight carbon fiber tubes with high strength aluminum alloy insert materials assists in maximizing strength as well as experiment flight duration and data collection. The fabrication of one support of the three-pronged assembly is carefully laid out, along with experimental testing to verify computational simulation results regarding the strength of the structure. Thus, a flight-ready design is developed and implemented for the purpose of creating a lightweight landing structure which can withstand several test flights, with the intention of implementing similar outrigger on propeller and gondola attachment support elements.
Works Cited


[Online]. Available:


Appendix A – Further Calculations

A.1 – Calculation of Composite Elastic Modulus for SPIDER project

The SPIDER Balloon-Borne Telescope project utilized pre-fabricated CFRP tubes from CST Composites™ with a 70.4 mm inner diameter and a 3 mm wall thickness, along with a guaranteed flexural rigidity by the manufacturer of $F_m = E \times I = 43.61 \text{kN} \times \text{m}^2$ by the composite manufacturer. Thus, calculating the moment of inertia of that tube as:

$$I = (OD^4 - ID^4) \times \frac{\pi}{64} = (0.0764^4 - 0.0704^4) \times \frac{\pi}{64} = 4.67 \times 10^{-7} \text{m}^4$$

As $F_m = E \times I = 43.61 \text{kN} \times \text{m}$, calculation of the flexural modulus used is:

$$E = \frac{F_m}{I} = \frac{43.61 \times 10^6 \text{N} \times \text{m}^2}{4.67 \times 10^{-7} \text{m}^4} = 9.35 \times 10^{10} \frac{\text{N}}{\text{m}^2} = 93.5 \text{ GPa}$$

Which is a value quite comparable to the cited metrics by Rock West Composites™: a tension modulus $E_t = 81.3 \text{ GPa}$ and a compression modulus $E_c = 77.9 \text{ GPa}$

With a length of $L = 2.3 \text{ m}$ and an effective column length $\kappa = 0.5$, the Euler Critical Load for this assembly was calculated as:

$$F = \frac{\pi^2 \times (E \times I)}{(\kappa \times L)^2} = \frac{\pi^2 \times (42.31 \times 10^6 \text{N} \times \text{m}^2)}{(0.5 \times 2.3)^2} = 3.2545 \times 10^8 \text{N}$$

A.2 – Four-Leg Landing Scenario

First, as NASA performs its calculations in imperial units, convert mass ($M$), maximum acceleration ($G$), and initial speed ($V$) into imperial units:
\[ M = 600 \text{ kg} \times \frac{2.205 \text{ lb}}{1 \text{ kg}} = 1323 \text{ lb} \]

\[ V = 15 \text{ knots} \times \frac{1.688 \text{ ft/s}}{1 \text{ knot}} = 25.32 \frac{\text{feet}}{\text{second}} \]

\[ G = 2.5 \times 9.81 \frac{\text{meters}}{\text{second}^2} \times \frac{3.2808 \text{ feet}}{1 \text{ meter}} = 80.46 \frac{\text{feet}}{\text{second}^2} \]

Next, convert the crush strength of a crush pad \( \rightarrow \text{C.S.} = 10 \frac{\text{lbs}}{\text{inch}^2} \times \frac{144 \text{ inch}^2}{1 \text{ foot}^2} = 1440 \frac{\text{lbs}}{\text{foot}^2} \)

Finally, calculate the remaining variables:

\[ \text{K.E.} = \frac{1}{2} \times \frac{M}{g} \times V^2 = \frac{1}{2} \times \frac{1323 \text{ lbs}}{32.185 \frac{\text{feet}}{\text{second}^2}} \times \left( \frac{25.32 \frac{\text{feet}}{\text{second}}}{2} \right)^2 \\
= 13176.611 \text{ ft-lb} \]

\[ S = \frac{V^2}{2 \times G} = \frac{\left( \frac{25.3 \frac{\text{feet}}{\text{second}}}{2} \right)^2}{2 \times 80.5 \frac{\text{feet}}{\text{second}^2}} = 3.9 \text{ feet} \]

As \( S \) is nearly 4 feet, use this value to find the minimum thickness of the crush pad:

\[ T_c = \frac{S}{T_u} = \frac{3.9 \text{ feet}}{0.7} = 5.6 \text{ feet} = 67 \text{ inches} \]

\( T_i \rightarrow N \times T_i > T_c \rightarrow N \times 4 \text{ inches} > 67 \text{ inches} \)

\[ N = 16, T_t = 68 \text{ inches} = 5.66 \text{ feet} \]

\[ S' = T_t \times T_u = 68 \text{ inches} \times 0.7 = 48 \text{ inches} = 4 \text{ feet} \]

Finally:

\[ \text{K.E.} = A \times \text{C.S.} \times S' \]

\[ A = \frac{\text{K.E.}}{\text{C.S.} \times S} = \frac{13176.611}{1440 \times 4} = 2.2876 \text{ feet}^2 = 0.2125 \text{ meters}^2 \]
Thus, each square side length of the crush pad is 1.512 feet, which is approximately equal to 18 inches, or 0.461 m.

A.3 – Two-Leg Landing Scenario

In the two-leg lading scenario, the maximum deceleration allowable per leg is increased to 5 G, and the new crush pad dimensions can be found as follows:

\[ G = 5 \times 9.81 \frac{\text{meters}}{\text{second}^2} \times \frac{3.2808 \text{ feet}}{1 \text{ meter}} = 160.92 \frac{\text{feet}}{\text{second}^2} \]

From Appendix A.2 – Four-Leg Landing Scenario:

\[ M = 600 \text{ kg} \times \frac{2.205 \text{ lb}}{1 \text{ kg}} = 1323 \text{ lb} \]

\[ V = 15 \text{ knots} \times \frac{1.688 \text{ ft/s}}{1 \text{ knot}} = 25.32 \frac{\text{feet}}{\text{second}} \]

\[ \text{C.S.} = 10 \frac{\text{lbs}}{\text{inch}^2} \times \frac{144 \text{ inch}^2}{1 \text{ foot}^2} = 1440 \frac{\text{lbs}}{\text{foot}^2} \]

\[ \text{K.E.} = 13176.611 \text{ ft-lb} \]

But the stopping distance, S, changes due to the maximum allowable stopping acceleration

\[ S = \frac{V^2}{2 \times G} = \frac{(25.32 \frac{\text{feet}}{\text{second}})^2}{2 \times 160.92 \frac{\text{feet}}{\text{second}^2}} = 1.992 \text{ feet} \]

As S is nearly 2 feet, use this value to find the minimum thickness of the crush pad

\[ T_c = \frac{S}{T_u} = \frac{1.992 \text{ feet}}{0.7} = 2.845 \text{ feet} = 34 \text{ inches} \]

\[ T_t \rightarrow N \times T_i > T_c \rightarrow N \times 4 \text{ inches} > 34 \text{ inches} \]

\[ N = 9, T_t = 36 \text{ inches} = 3 \text{ feet} \]

\[ S' = T_t \times T_u = 36 \text{ inches} \times 0.7 = 25.2 \text{ inches} = 2.1 \text{ feet} \]
Finally:

\[ \text{K.E.} = A \times \text{C.S.} \times S' \]

\[ A = \frac{\text{K.E}}{\text{C.S.} \times S} = \frac{13176.611}{1440 \times 2.1^*} = 4.3574 \text{ feet}^2 = 0.4048 \text{ meters}^2 \]

Thus, each square side length of the crush pad is 2.087 feet which is approximately equal to 25 inches, or 0.636 m.
Appendix B – Detailed Designs

B.1 – Joint Attachment Design Drawings

Figure 48: Double clevis node attachment
Figure 49: Schematic drawings for single clevis attached to aluminum insert
B.2 – Initial Joint Design Drawing – Bolted Insert

Figure 50: Bolted carbon fiber assembly drawing, with steel cylinders acting as bolts for later simulation
B.3 – Initial Joint Design Drawing – Hose Clamp Aluminum Slit

Figure 51: Schematic for hose clamp and oversized aluminum shaft joint design
Figure 52: Design for steel Instron 3369 top attachment
Figure 53: Design for steel Instron 3369 attachment base
Figure 54: Process of machining Instron 3369 Testing base
Figure 55: Testing assembly schematic