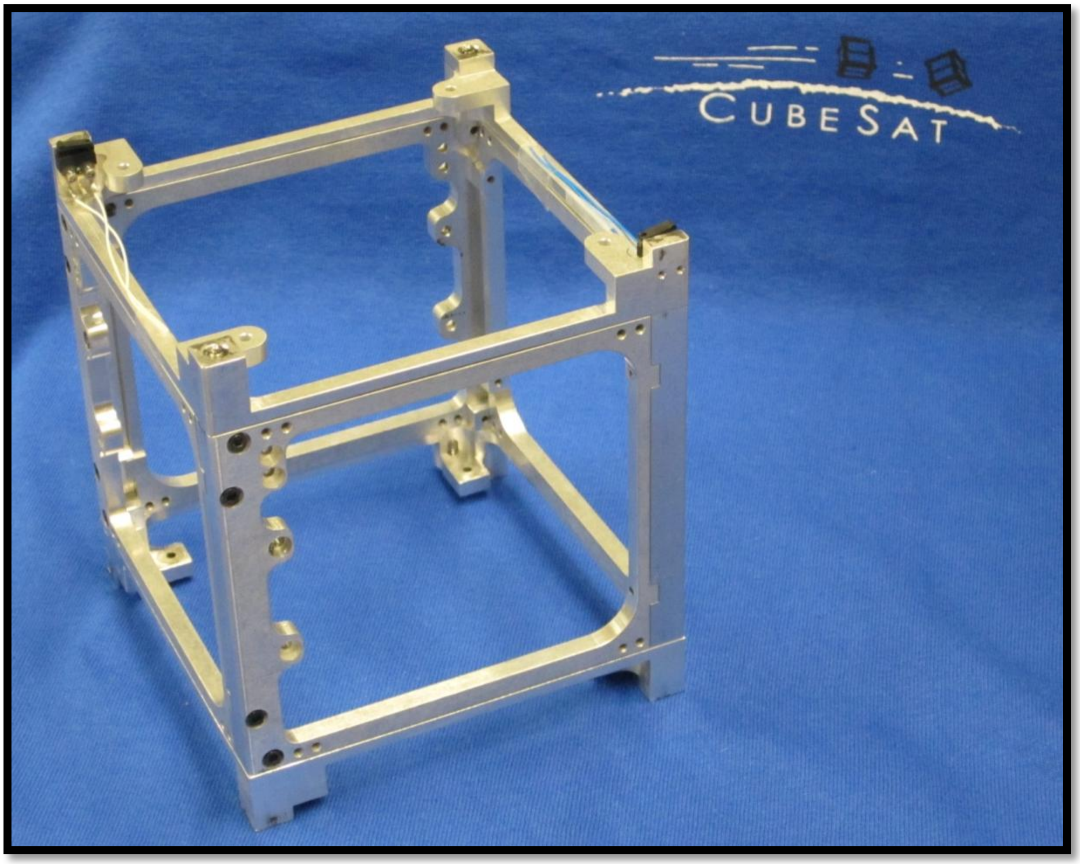
THE NEXT GENERATION CUBESAT

A MODULAR AND ADAPTABLE CUBESAT FRAME DESIGN



SPONSORS:

Dr. Jordi Puig-Suari

Aerospace Department Faculty Member

Austin Williams

Electrical Engineering Graduate Student

TEAM MEMBERS:

James Dolengewicz

Mechanical Engineering
Undergraduate Student

jdolenge@calpoly.edu

Lucas Whipple

Mechanical Engineering
Undergraduate Student

lwhipple@calpoly.edu

Stephanie Wong

Mechanical Engineering
Undergraduate Student

sdwong@calpoly.edu

STATEMENT OF DISCLAIMER

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CHAPTER 1: INTRODUCTION

EXECUTIVE SUMMARY

The goal of this project is to develop an improved next-generation CubeSat structure for Cal Poly's PolySat program. Notable achievements include significantly increased ease of access, design to optimize payload space, improved machinability, increased modularity and a platform which allows for easy integration of future payloads.

SPONSOR BACKGROUND AND NEEDS

The goal of the HyperCube project is to design an updated frame structure for the PolySat program that is built around the next-generation communications electronics module being developed by Austin Williams, an electrical engineering graduate student. The Cal Poly PolySat Project was founded in 1999 and serves to introduce students to a real-world aerospace environment. PolySat's primary task is to design and build very small satellites (picosatellites) to "[…] perform a variety of scientific research and explore new technologies in space" (http://polysat.calpoly.edu/). Since the next-generation electronics module is being developed to be as small as possible, a new structure is desired to maximize this efficiency and provide a larger payload volume.

FORMAL PROBLEM DEFINITION

The objective is to design, build and test a new structure for PolySat's new and improved frame. The frame will be designed around newly developed communications and power boards that have been made in order to maximize available space for science payloads. For easy integration, the design will have good access to mount these improved circuit boards as well as any payloads. Maximizing the utilization of the allowed 6.5 mm protrusion space on each face of the cube is also a goal of this project. Furthermore, the design will be extended to preliminary models in a 2U and 3U size configuration. This frame should maximize the available payload volume by efficiently mounting the communications board and other needed boards (such as solar panels and batteries) while providing expandability up to the 3U size.

Another important goal for this project is improving ease of assembly. Assembly testing and prototype development information is described in detail in the sections below.

OBJECTIVE/SPECIFICATION DEVELOPMENT

There are many considerations to take into account when designing a satellite's structure. Synthesizing these considerations into a design specification allowed us to ensure that our final design fulfilled all design objectives.

The total mass of the CubeSat cannot exceed 1.33 kg, so making the frame as lightweight as possible is a concern. Based on previous frames and the recent increase in allowable mass, we plan to shoot for 200 grams maximum.

Since the satellites change from mission to mission, mounting the new payloads and other mission specific boards is also a concern. We will aim to provide adjustable mounting points for mission specific payloads and electronics. Previous frames such as those for CP5 and CP6 provide only a few fixed points to mount payloads within the cube.

Ease of access into the internal volume of the cube with minimal disruption to other components is another extremely important goal which must be considered in our overall design. Previous designs have been notoriously difficult to access for repairs or rework, and our design should minimize this difficulty as much as possible without compromising other design specifications.

This satellite must undergo a series of tests before it can be launched aboard the P-POD – an assembly test, an integration test, and a NASA GEVS vibration test. These tests and their results are described in detail in chapter 3.

A majority of our specifications come from the CDS, or Cubesat Design Specification. The CubeSat Design Specification was created by Cal Poly CubeSat program members and is used by groups around the world as a basis for the creation of standard picosatellites. We developed requirements for mass, mounting points, and ease of integration using information from past CubeSat designs, as well as a house of quality, presented in appendix A. These requirements are detailed in Table 1.

TABLE 1: TECHNICAL SPECIFICATIONS OF CUBESAT STRUCTURE

Spec #	Parameter	Requirement	Tolerance	Risk	Compliance
1	Cube Size (see CDS)	100x100x113 mm	± 0.1 mm	L	AI
2	Protrusion	6.5 mm	Max	L	AI
3	Modularity	1,2,3U	N/A	M	AIS
4	Mass	200 gm	Max	M	AI
5	Mounting Points	6	Min	M	A
6	Vibration Test	NASA GEVS	Survive	M	AT
7	Ease of Access (integration)	Easy	N/A	H	AI

By relating the customer requirements to engineering requirements we were able to determine the engineering requirements that we need to focus our attention on. Also by benchmarking the customer requirements against previous frames will let us easily determine the parts of each frame system we should investigate further. This allows us to create targets for our quantifiable engineering requirements.

CHAPTER 2: FINAL DESIGN

CURRENT DESIGN ITERATION

Our ultimate design consists of six modular panels that fit together to form the faces of a cube. The top panel, which has been designed specifically to hold the newly redesigned communications and power electronics boards developed by PolySat, four identical side panels and a bottom panel that can be easily customized to the unique requirements of each payload. Through this modular approach, this structure is flexible enough to handle a wide variety of missions and payloads.

Internal volume has been further optimized by designing the faces of the cube to be open, which allows for electronics or payload items to be "sandwiched" between inner and outer cube face boards.

The structure has been designed so that portions of it can be easily modified to fit specific missions. The side rail mounting tabs can be easily moved if need be for a specific payload without affecting the structural properties or assembleability of the frame. As well, the bottom panel can be customized to fit specific payloads if the need arises.

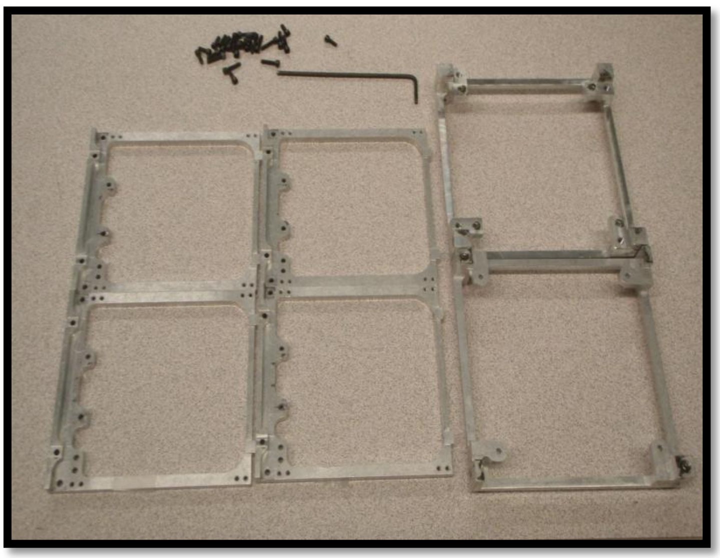


FIGURE 2: DISSASEMBLED CUBE

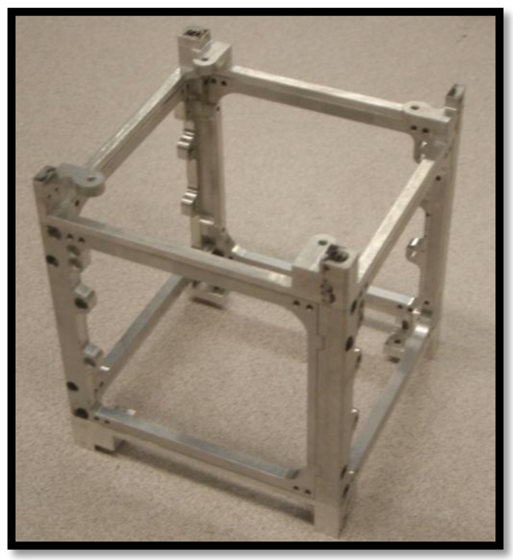


FIGURE 1: ASSEMBLED CUBE

DESIGN DETAILS

Our structure has several additional advantages over previous frame designs. The modularity of the structure allows the panels to be customized if the mission dictates it. Most notably, the shoe will likely be customized for each mission to properly hold the payload. This will allow the removal of a payload with minimal hassle and disruption of other components. As well, because the electronics and structure have been designed concurrently, the structure has an unprecedented available payload volume as is shown in Table 2.

FULFILLMENT OF DESIGN OBJECTIVES

This ultimate design fulfills our design objectives. The table below shows how our design compares to previous iterations and the specification verification checklist shows that we conform to the design specification we developed at the beginning of the project.

TABLE 2: IMPROVEMENTS MATRIX

Category	Previous Design	Hypercube
Ease of Accessibility to interior components	Structure did not come apart, exterior components must be removed to access interior components	Structure disassembles easily allowing for quick access to internal volume from fully assembled state
Machining Difficulty	Machined from two large pieces of aluminum, design did not optimize use of base materiel volume	Majority of machining done on one face, holes must be drilled on all faces. Holes are in repeating pattern for easy fixturing.
Cost to Machine	2500 (to tolerance)	1900 (to tolerance)
Modularity	None	Six parts which can be individually swapped
Accommodates Multiple Payloads with Minimal Changes:	Entire structure must be redesigned for new payloads	Only shoe must be customized for each payload, rest of structure remains the same.
Accommodates New Communications Electronics:	No	Yes
Usable Internal Volume, Before Comm Boards	838000 mm^3	950000 mm^3
Comm Board Stack Depth	30 mm	15mm

SPECIFICATION VERIFICATION CHECKLIST

TABLE 3: SPECIFICATIONS VIBRATION CHECKLIST

Spec #	Parameter	Requirement	Tolerance	Risk	Compliance	CHECK:
1	Cube Size (see CDS)	100x100x113 mm	± 0.1 mm	L	AI	✕
2	Protrusion	6.5 mm	Max	L	AI	✕
3	Modularity	1,2,3U	N/A	M	AIS	✕
4	Mass	200 gm	Max	M	AI	✕
5	Mounting Points	6	Min	M	A	✕
6	Vibration Test	NASA GEVS	Survive	M	AT	✕
7	Ease of Access	Easier than Previous	N/A	H	AI	✕
8	Ease of Integration	Easier than Previous	N/A	H	AI	✕
2.2.4	Maximum X and Y dimensions (rails)	100 mm	± 0.1 mm	M	N/A	✕
2.2.5	Maximum Z dimension	113.5mm	± 0.1 mm	M	N/A	✕
2.2.6	Maximum protrusion	6.5 mm	Max	H	N/A	✕
2.2.9	Minimum rail dimension	8.5 mm	Min	L	N/A	✕
2.2.11	Rail edge radius	1 mm	Min	L	N/A	✕
2.2.12	Rail end area	6.5 x 6.5 mm	Min	L	N/A	✕

DYNAMIC ANALYSIS

Our final dynamic verification consists of a 3-axis NASA GEVS vibrational test with sine sweeps before and after each axis has been tested to detect any anomalies. Our test unit was outfitted with a 1.3 kg test mass model which was rigidly secured to our structure at eight locations (bottom panel and mid side panel) and the structure was loaded into a Test POD (Test Picosatellite Orbiter Deployer) which represents the conditions during deployment. Our final prototype survived this test easily and conformed to all CDS requirements. The results of this test can be viewed in appendix E.

The results of our FEA (finite element analysis) suggest that our design will be strong enough to survive a Minotaur launch with a worst-case factor of safety of 84. Torque specifications were also developed for all the fasteners. Analysis indicated that the screws will be safely tightened at 4 lb-inch for all frame screws. Details of these analyses can be viewed in appendix E.

COST ANALYSIS

The only significant cost of our project is the cost to perform the final production machining. With this as our only cost reduction mechanism, we took great care from the outset of our project to optimize our designs for easy machining and repeatable fixturing. Our final design uses six parts. The side panels are all identical to reduce fixturing and programming costs, and our top and bottom panels have minor geometric changes that allow fixtures and tooling to be used for both parts. These and other design features helped to reduce the overall cost to machine our final prototype.

One potential area for cost reduction in the future is in our tolerances. Our relatively tight tolerances (+/-.01 mm in some locations) came at a cost premium. In future designs, it is likely that some tolerances can be decreased. However, the CDS requires an overall tolerance of +/- .1mm, so reductions in tolerance will be somewhat difficult to justify.

TABLE 4: COST ANALYSIS

Date	Item	Source	Cost	Approved By
3/5/2010	Rapid Prototype 1	Hypercube Project Team	No Charge	Professor P, Professor Meagher
4/10/2010	Rapid Prototype 2 (Objet prototype)	Hypercube Project Team	No Charge	Austin Williams, Professor P
5/6/2010	Rapid Prototype 3 (Objet Prototype)	Hypercube Project Team	No Charge	Austin Williams, Professor P
9/4/2010	Rapid Prototype 4 (Flight Spacing Prototype)	Hypercube Project Team	No Charge	Austin Williams, Professor P
11/3/2010	Machined Prototype (6 Hats, 8 Side Panels)	Sponsor	4800 (1800/cube)	Austin Williams, Professor P

MATERIAL SELECTION

The material of the frame is limited by the CDS to aluminum alloys 6061 or 7075. Aluminum zinc alloy 7075 has a higher hardness rating and yield strength than 6061 making it our choice for the frame material. However, the price of 7075 can run more than twice the price of 6061 and for this reason we have elected to machine all parts out of 6061 Aluminum that is hard anodized post-machining.

SPECIAL FABRICATION AND ASSEMBLY INSTRUCTIONS

All fabrication instructions are contained in our mechanical drawings. With a machined aluminum prototype complete, no fabrication issues have been reported by the machinist nor were any detected during our internal prototyping. An assembly manual is included in appendix G, but the current iteration is such that it cannot be assembled incorrectly.

SAFETY CONSIDERATIONS

There are very few additional safety considerations that must be made for this project. Care must be taken not to pinch skin in between the parts during assembly. Screws must be tightened to specified torques.

MAINTENANCE CONSIDERATIONS

There is very little maintenance required on the CubeSat. Bolts must be tightened and all CDS specifications must be met before flight. Once in space the satellite is difficult to repair and must burn up after a specified period of time, effectively negating maintenance considerations

CHAPTER 3: DESIGN VERIFICATION PLAN

To verify our design, we plan to run a series of tests to evaluate the fitness of our concept as well as to ensure that it meets the project's functional criterion. These tests include an assembly test, an integration test, and a vibrations test on the final aluminum prototype.

TEST DESCRIPTIONS

ASSEMBLY TEST

An assembly test must be completed. An assembly test consists of a timed test of an untrained user assembling our design. Subjective ease of assembly is compared to previous designs. Quality of assembly is also evaluated (perpendicularity, parallel members, overall size tolerances, etc).

INTEGRATION TEST

An integration test must be performed. An integration test consists of the integration team testing subjective ease of integration relative to previous designs. Opinions will be gathered from several team members using a scoring metric and those values will be compared to previous designs.

VIBRATIONS TEST

A vibrations test profile will be to random qualification as indicated in NASA GEVS (General Environmental Vibration Specification). The test will occur at Cal Poly facilities in building 41 per approval of the faculty and student in charge of the aerospace department's vibration table and equipment. An accelerometer will be obtained from Dr. James Meagher. The CubeSat program will procure the documents to run the shaker table and the mounting plates. As far as testing in a one unit Test POD, this will depend on the availability. Thus, advance notice be will needed to the CubeSat program for a check on the unit. Please take note that the integration may take two to three days.

TEST RESULTS

ASSEMBLY TEST

An assembly test was performed by asking members of the PolySat team to assemble the cube with only the assembly manual as guidance. From four independent tests, the overall opinion of our test assemblers was that our cube was easy to assemble and fit together intuitively.

INTEGRATION TEST

Integration was done with a representative mass model and laser-cut electronic board mockups. Integration was performed by members of the PolySat team and their reports on the ease of integration were positive, indicating an improvement over previous designs.

VIBRATIONS TEST

Our final vibrations test was a 3-axis NASA GEVS vibration test with sine sweeps before and after each random vibration has been tested to detect any anomalies. Our test unit was outfitted with a 1.3 kg test mass model which was rigidly secured to our structure at eight locations (bottom panel and mid side panel) and the structure was loaded into a Test POD which represents the conditions during deployment. Our final prototype survived this test easily and conformed to all CDS requirements. The detailed results of this test can be viewed in appendix E.

CHAPTER 4: PROJECT MANAGEMENT PLAN

MANAGEMENT PLAN

At the outset of our project, our management plan exploited the strengths of our group members. Our initial management plan, which is shown below, has largely held true. All members of the team contributed to prototype machining. Stephanie Wong was the liaison between our sponsor and our group, as well as the leader on FEA modeling and vibration testing. Lucas and James did a majority of the solid modeling and prototyping. Our management plan allowed our project to go very smoothly and facilitated communication between our team and our sponsor.

INITIAL PROJECT MANAGEMENT PLAN

Because of the unique skill sets of individual members of our group, it will be useful to split up certain tasks among group members. Certain tasks, such as design conceptualization and research phases are most efficiently handled as a group, but other responsibilities, such as leading the weekly status meeting with our advisor will be rotated weekly basis. Stephanie is currently taking a FEA class, so the majority of the responsibility for generating the FEA model will fall upon her. James and Lucas have prototyping and machining experience, and the bulk of the modeling and manufacturing will be tackled by them. When the prototype is completed, the whole team will join together to test it on Cal Poly's vibes table. If our initial design shows problems in the vibes table, the team will redesign the frame using the knowledge gained.

CHAPTER 5: BACKGROUND

EXISTING PRODUCTS

There have been many attempts to make the optimal CubeSat structure by different organizations such as universities, governments and military personal. Large corporations, such as Boeing, have even released open-source designs. There are also kits available for groups without the resources to design and build their own structures.

SPECIFIC TECHNICAL DATA

Due to the standardization process of launching pico-satellites, there are a lot of constraints that CubeSats must adhere to so they will both fit and launch properly from the P-POD. These specifications are provided in the CubeSat Design Specifications (CDS). An excerpt drawing is shown in Figure 1 from the CDS labeling the sides of the standardized CubeSat. The specific design requirements presented in the CDS include the specific size of the overall cube as well as the minimum dimensions of the rails. In addition, the CDS mentions the maximum protrusion from the cube dimensions. These dimensions are summarized in Table 1 below. The CDS also contains maximum mass requirements (1.33 kg), material requirements (aluminum 6061 or 7075), and other requirements such as the location of the separation springs and deployment switches.

TABLE 5: CDS RELEVANT DIMENSIONS FOR ALL CUBESAT STRUCTURES

Dimension location	Dimension	Tolerance	CDS reference #
Maximum X and Y dimensions (rails)	100 mm	± 0.1 mm	2.2.4
Maximum Z dimension	113.5mm	± 0.1 mm	2.2.5
Maximum protrusion	6.5 mm	Maximum	2.2.6
Minimum rail dimension	8.5 mm	Minimum	2.2.9
Rail edge radius	1 mm	Minimum	2.2.11
Rail end area	6.5 x 6.5 mm	Minimum	2.2.12
Rail surface contact	75% (85.1 mm)	Minimum	2.2.13 ,2.2.13.1

CHAPTER 6: DESIGN DEVELOPMENT

CONCEPT - OPTION A

The modular rail design consists of four rails mounted to a cross braced ring or cross bracing struts. The advantages of this design are easy disassembly, fairly simple manufacturing, vertical modularity, and ease of access. This designs lack rigidity due to the multiple pieces associated and does not offer many mount points. Some initial designs keep part counts down, but these designs have the potential to have very high part counts, especially in some more flexible bracketed iteration.

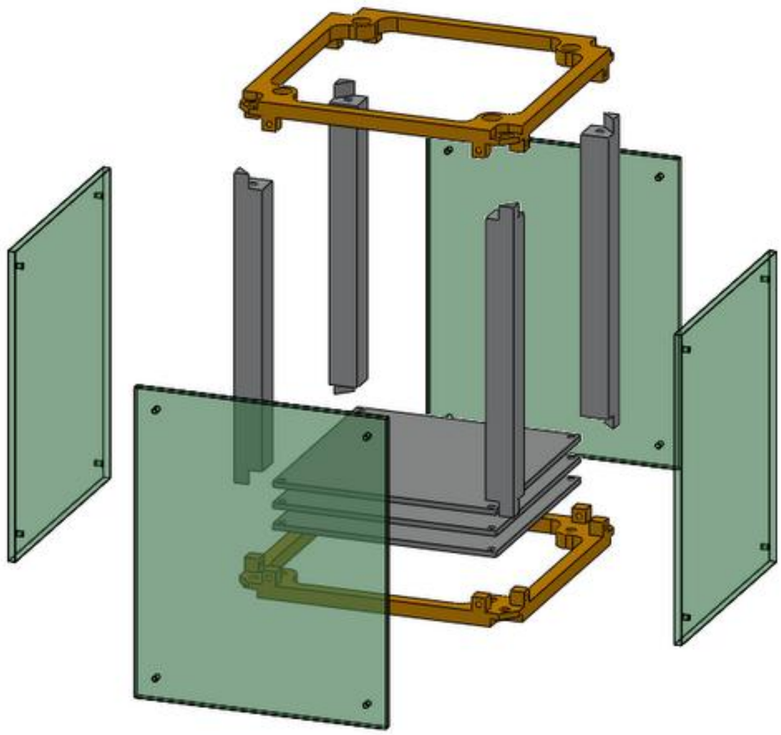


FIGURE 3: RING AND RAIL ITERATION

One example of this type of design can be seen in Figure 3. The rings which mount to top and bottom are attached to the rails with countersunk socket head cap screws that mount through the top of the rings and into the body of the rail. The part count is low, with four identical rails and two rings. The rings can be used as a carriage for electronics, allowing easy removal of the entire stack with after removing only four bolts. As well, the exterior machined surfaces of the rings constrain the movement of the rails regardless of initial misalignment, forcing the structure to be perfectly square at assembly and preventing assembly errors. Disadvantages of this design include possible machining difficulties for situations where multiple rings are needed, lack of mounting options, high total part count at a minimum of six machined parts with eight fasteners, and poor shear strength.

CONCEPT - OPTION B

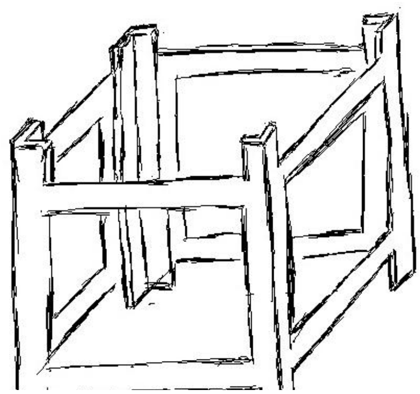


FIGURE 5: OPTION B, CONCEPT DESIGN, 'L' SHAPED RAILS WITH SUPPORT BARS ON TOP AND BOTTOM

The frame is simple and consists of four ‘L’ rails and eight structural posts. Holding the cube shape, the structure can withstand loading in different axes.

The advantages of this concept are that it is lightweight, modular, and has flexible mounting points. It is lightweight due to the inside material taken away from the rail portion of the CubeSat which is advantageous for the limit of 1.33kg in the CubeSat Design Specification. The concept is modular due to its symmetry and can be modeled into a 1U, 2U, 3U and 1.5U lengthwise. The major plus on this structure is that the frame has the ability to have flexible mounting points along the rails. This is done by attaching small brackets on the inside of the rails. The brackets provide easily adjustable mounting points for new board sizes, new clearance in between boards, and new payloads.

The disadvantages of the unit are that the frame has multiple pieces and is tougher to integrate electric boards. Although the frame can be lightweight, if the pieces are all separate, the screws to hold the pieces together may weight a substantial amount. Also the more pieces in the structure may lose a lot of rigidity in the structure and cause unnecessary stress and strain on the electrical boards. In addition, finding the placements of the screws to hold the frame together may be an issue as well as bring the cost of manufacturing up for the different pieces and jigs to manufacture the frame. Besides the issue of having multiple pieces as the base of the frame, the assemble frame may be difficult to integrate the electrical boards and payloads if the only way to tighten a mounting screw is from the top or bottom of the CubeSat.

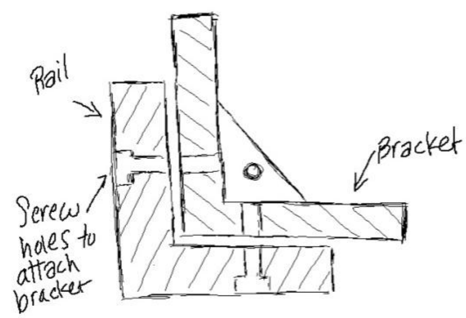


FIGURE 6: TOP VIEW, CROSS SECTION, OF ONE OF THE RAILS WITH A BRACKET TO SHOW POSSIBLE MOUNTING POINTS.

Overall this structure, though a seemingly simple concept, may have a lot of issues in manufacturing and integrating. Seeing as how integrating electrical boards is a top test topic, this frame may not be the best way to go for this new priority level.

CONCEPT - OPTION C

The panel design consists of four side panels that connect together at the corner rails to form the main body of the CubeSat with a "hat" piece that connects to the top and bottom and provides mounting for the core electronics stack and payload. This idea is similar to the CP-X structure but we have done away with the diagonal supports and minimized beam cross sections. We replaced them with sturdier cross supports to increase usable space within the cube. The advantages to this design are a low part count for both assembly and manufacturability, as well as the ability to construct the solar board sandwiches with ample space for batteries or other thick components. The four panels are the same piece and the "hat" fits on top and bottom of the structure. This makes for a total part count of six for assembly and only two distinct parts for machining.

Different designs with this overall scheme are possible. One example, in Figure 5, again borrows from the design of CP-X. The "hat" piece on this is designed similar to a combination of the stand-offs and top support piece of CP-X. The tops and bottoms of the rails have notches to accept matching tabs that protrude from the bottom of the "hat", which attach together with recessed screws. This structure provides a positive method of aligning the hat and box structures together but restricts the deployment switch possibilities, leaving only 11mm for them. Another benefit of this design is the method of assembly. Each side panel would have the solar board sandwiches assembled to them separately. The core electronics would be assembled with one "hat" and the payload on the other. The panels could then be assembled into a box that the "hats" would slide into and be attached or each panel could be assembled onto the two "hats". Any single panel would be removable with six screws after the whole cubesat is assembled.

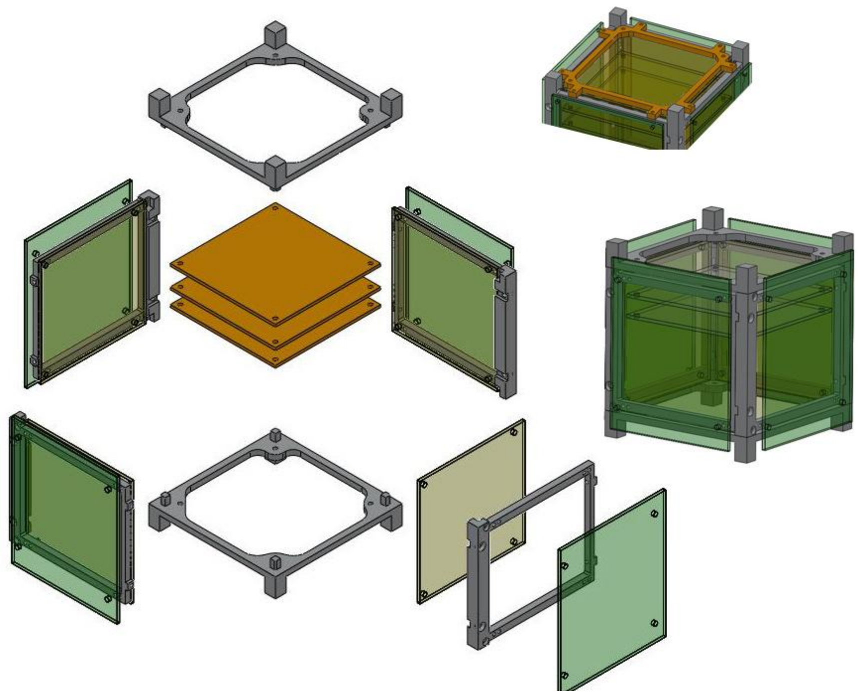


FIGURE 7: OPTION C, MOST IDEAL CONCEPT DESIGN, CONTAINS DIFFERENT VIEWS WITH BOARDS INTEGRATED TO SHOW HOW THE 'HAT' DESIGN AND SOLAR PANELS WOULD MOUNT ON THE STRUCTURE

A second "hat" design is also shown (top right) in which the rails extend the entire height of the cube allowing for other deployment switch designs. This hat simply attaches to the top of the cross beams and provides the same mounting holes as the previous "hat".

Another advantage would be the expandability of this design. To achieve 2U or 3U sizes the panels would only need to be extended. Additional cross bracing and mounting holes would also be required but those could be added to the design very easily. This would also provide for 1.5U size cubes to be developed. The "hat" pieces would be able to remain unchanged for any size.

Regardless of "hat" design, these parts would all be made of aluminum 7075 or 6061 as per the CDS. Each piece would be machined out of a plate of aluminum; various machines could be used based on the capabilities of the machine shop.

CONCEPT SELECTION

Option C is our top design choice and was the concept our final design was based upon. It was selected with a Pugh diagram that compared various overall cube structures (available in Appendix B). We compared our brainstorming ideas to the CP-6 frame in many categories including, manufacturing cost/time, number of parts, and ease of assembly among others. The rail-panels, Hat, and inverted Hat designs were the top three choices and all were built off a panel design as presented here with different options for attaching the core electronics and payload.

Significant proof of concept work has been done to validate our design. A FEA model was developed, and the results of that analysis can be seen in appendix E. Five rapid prototypes were developed, assembly and vibrational testing was performed and our design was reviewed by the sponsor several times throughout the project.

CHAPTER 7: CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

After testing the final design, the HyperCube team feels confident in its mission readiness. Missions are already using the frame structure and several more missions have been proposed as of the time of this writing. Minor modifications may be required for final flight units depending on the mission, but the final design has a great deal of flexibility with regard to payload mounting. If a mission requires a specific mounting scheme, the entire bottom panel can be can be restructured for payloads and the rail mounting points can be moved up or down as needed.

RECOMMENDATIONS:

The hypercube ream recommends that this frame design be adopted for as many future missions as possible. As well, the team strongly recommends that our project be put into space.

REFERENCES

Acronyms, Definitions, and Abbreviations

1U, 2U, 3U, 1.5U: indicates the size of the structure, current P-POD configuration goes up to 3 units (in a row), U=unit

A: Analysis Compliance

ASD: Acceleration Spectral Density in G^2/Hz

CAC: CubeSat Acceptance Checklist

CDS: CubeSat Design Specification, created by the CubeSat Program at Cal Poly

CP: California Polytechnic State University, San Luis Obispo

FEA: Finite Element Analysis

Grms: Root-Mean-Square Acceleration

H: High risk

I: Inspection Compliance

L: Low risk

M: Medium risk

N/A: Not applicable

NASA: *National Aeronautics and Space Administration*

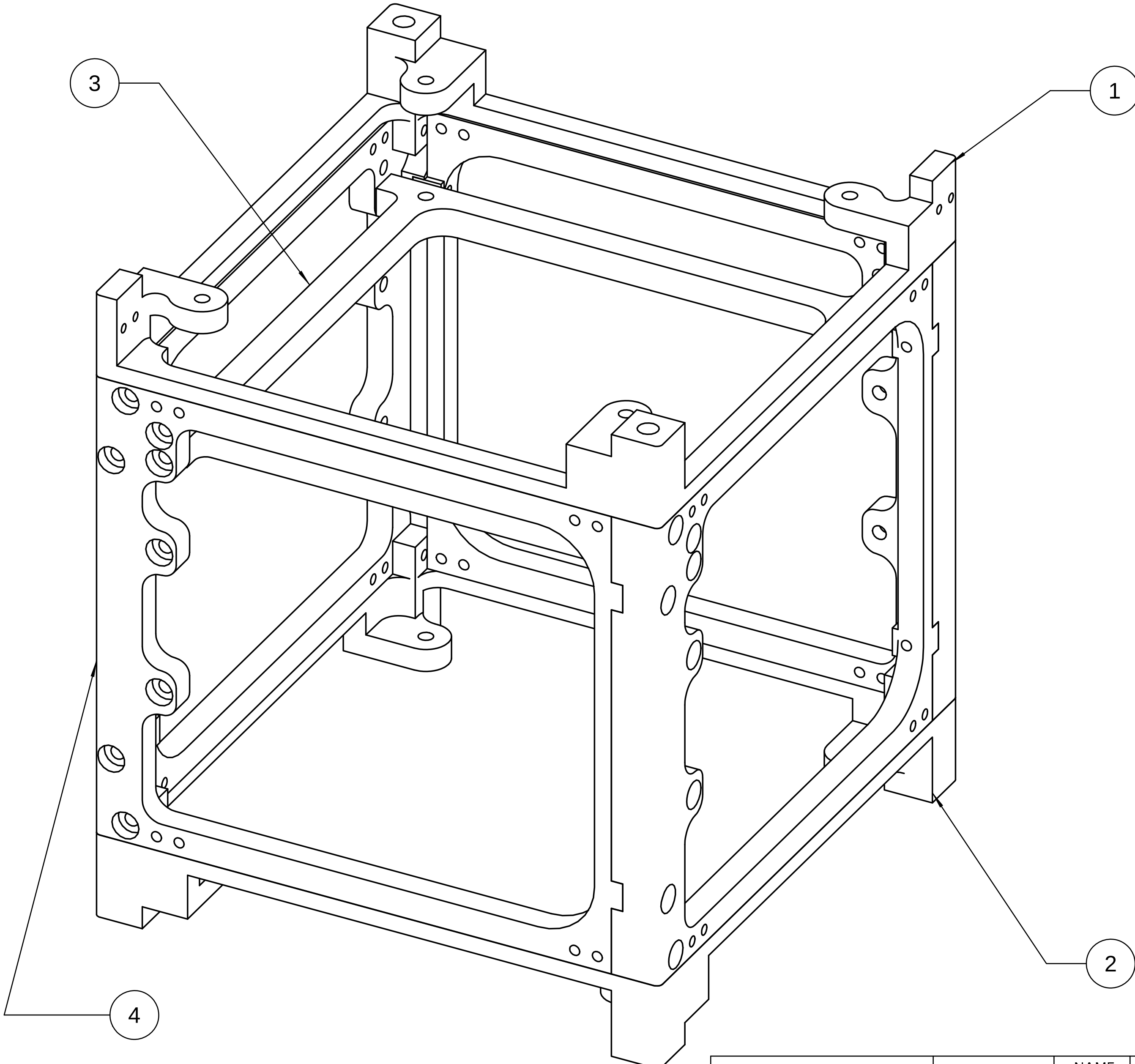
NASA GEVS: General Environmental Verifcation Standard, upheld by NASA

P-POD: Poly-Picosatellite Orbital Deployer, developed by the CubeSat Program to deployer picosatellites, interfaces with the launch vehicle, consists of a box and spring to eject the picosatellites once in orbit

S:Similarities Compliance

T: Test Compliance

LIST OF APPENDICES



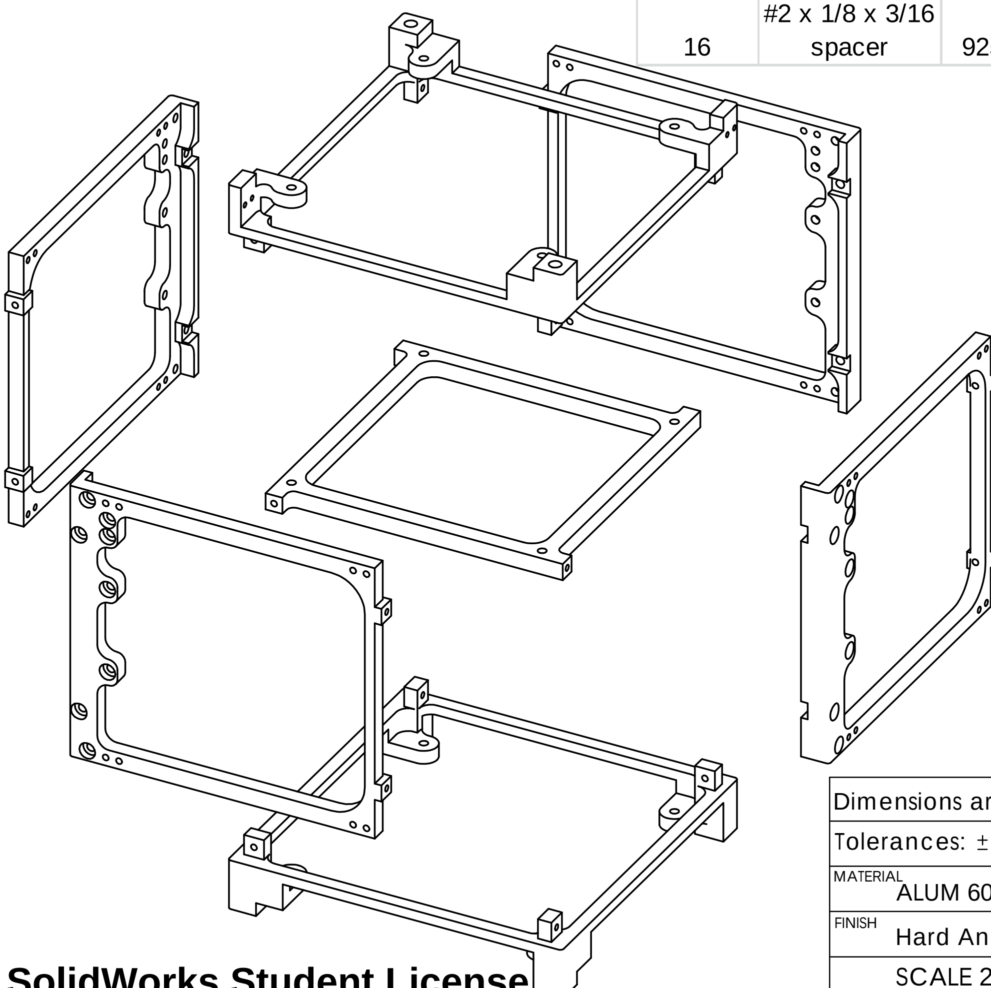
ITEM NO.	PART NUMBER	QTY.
1	HC1101	1
2	HC1102	1
3	HC1104	1
4	HC1103	4

Dimensions are in mm		NAME	DATE	TITLE:
Tolerances: ± .1 mm	DRAWN	JD	9/28/10	1U Cube
MATERIAL ALUM 6061-T6	COMMENTS:			
FINISH Hard Anodized				SIZE A · DWG. NO. 2101
SCALE: 1:1				REV 2 · SHEET 1 OF 2

4 3 2 1

Full Bill of Materiels

Quantity	Part	Part Number	Supplier	Cost	Note	Spec
1	Hat	HC1101	HyperCube			
1	Shoe	HC1102	HyperCube			
4	Side Panel	HC1103	HyperCube			
?	Cross Support	HC1104	HyperCube		Optional payload mounting bracket	
4	0-80 x 3/16	92200A054	McMasterCarr	$6.70 per 10	Deployment switch screw	MIL 16996-2
8	2-56 x 1/4	92200A077	McMasterCarr	$3.31 per 10	Hat to Panel screws	MIL 16995-2
8	2-56 x 3/8	92200A079	McMasterCarr	$3.43 per 10	Panel to Panel screws	MIL 16995-3
4	2-56 x 1/2	91255A081	McMasterCarr	$4.18 per 25	Core mounting screws	MIL 16995-4
16	2-56 x 3/16	91255A076	McMasterCarr	$3.54 per 25	Sandwich panel screws	MIL 51959-2
16	2-56 x 5/16	91255A078	McMasterCarr	$3.30 per 25	Solar panel screws	MIL 51959-4
16	#2 x 1/8 x 3/16 spacer	92510A687	McMasterCarr	$0.20 ea = $3.20	solar panel spacers	



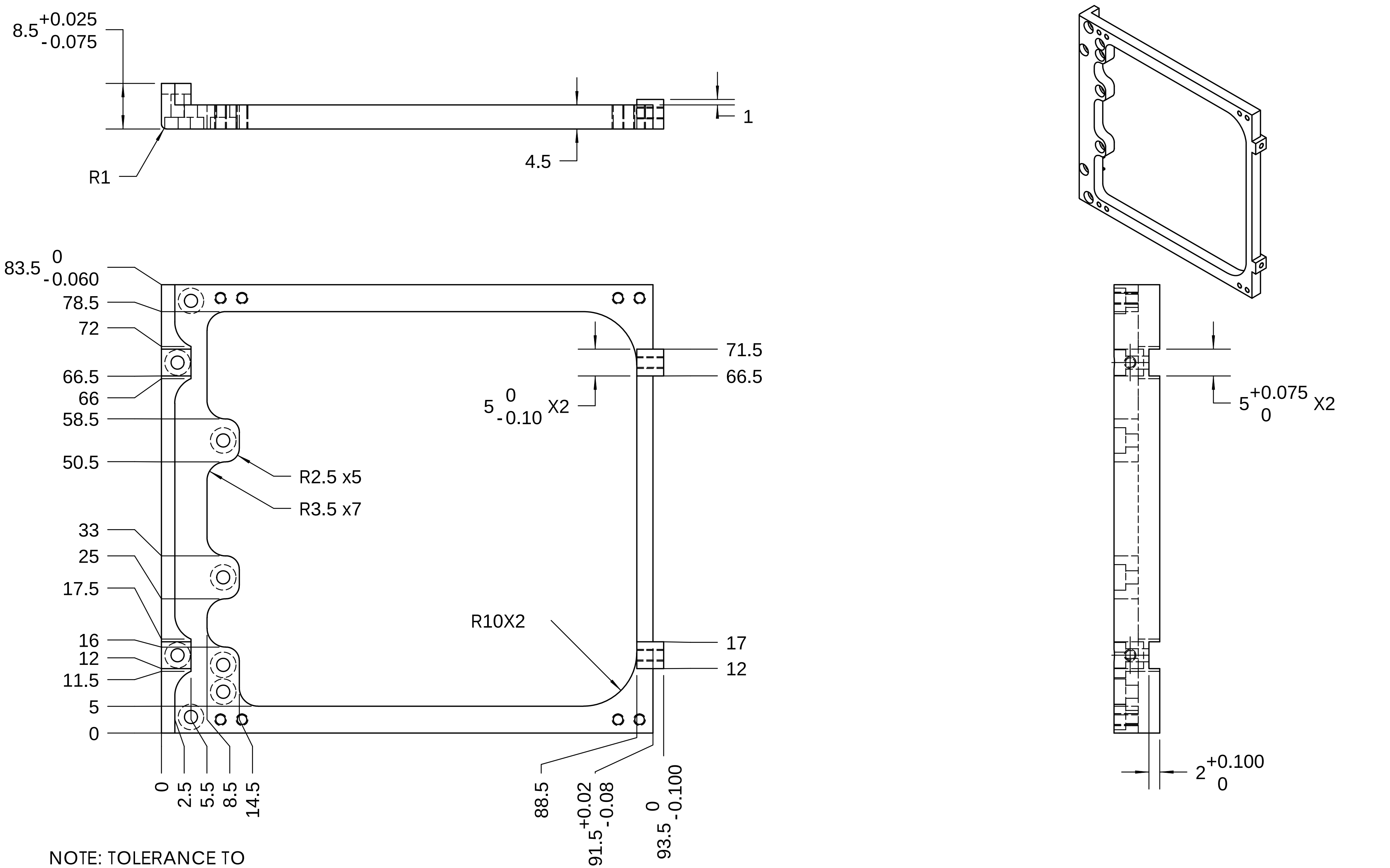
Dimensions are in mm		NAME	DATE
Tolerances: ± .1 mm	DRAWN	JD	9/28/10
MATERIAL ALUM 6061-T6	COMMENTS:		
FINISH Hard Anodized			
SCALE 2:3			

TITLE: 1U Cube

SIZE A | DWG. NO. 2101

REV 2 | SHEET 2 OF 2

4 3 2 1



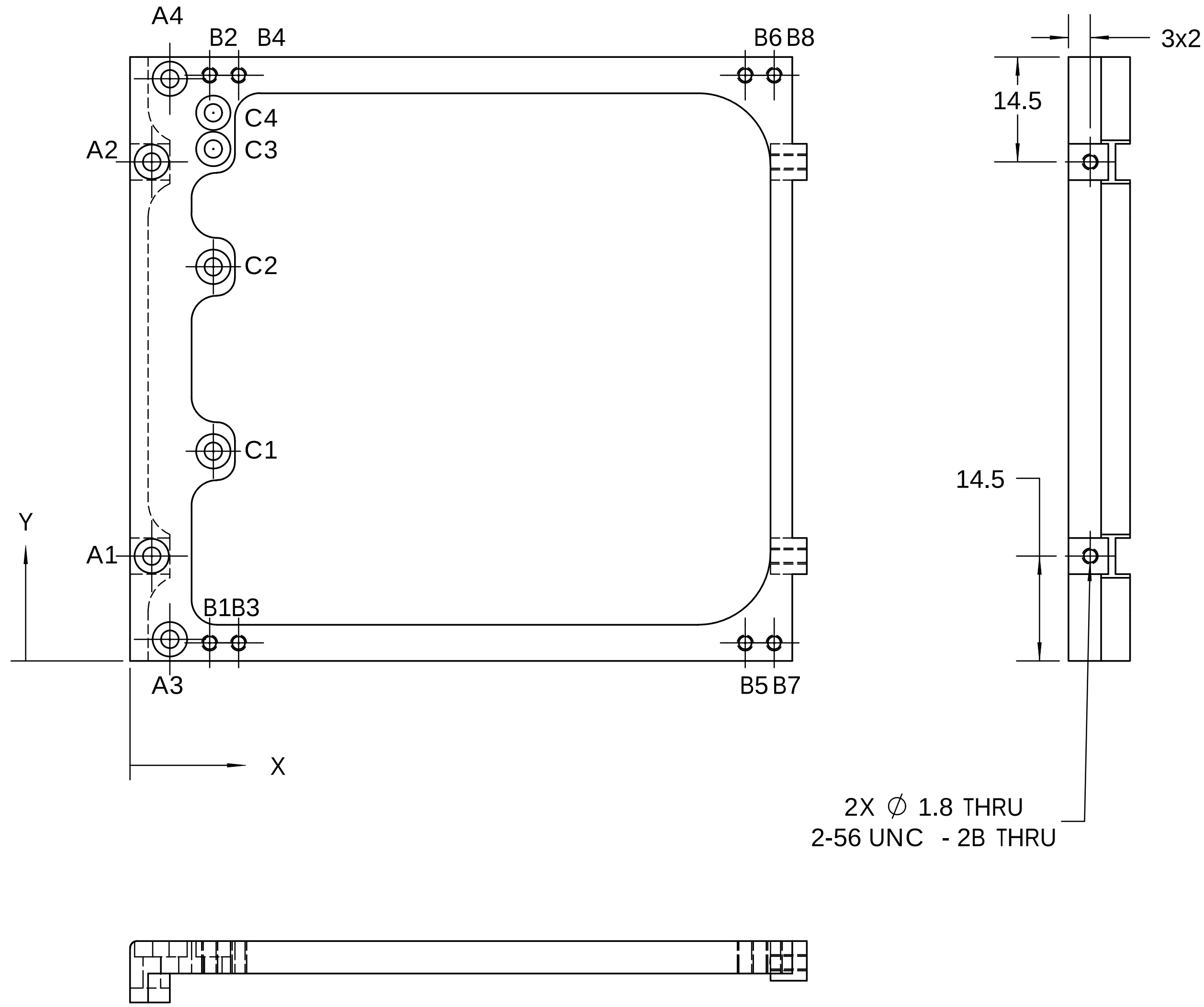
NOTE: TOLERANCE TO COMPENSATE FOR THICKNESS GAINED IN ANODIZING (ASSUME ANODIZING THICKNESS OF .025 MM, 0.5 THOU)

HYPERCUBE

Dimensions are in mm		NAME	DATE	TITLE:	
Tolerances: ± .1 mm	DRAWN	LW	4/20/10	Side Panel	
MATERIAL ALUM 6061-T6	COMMENTS:				
FINISH Hard Anodized				SIZE A	DWG. NO. 1103
SCALE: 1:1				REV 7	SHEET 1 OF 3

4 3 2 1

TAG	X LOC	Y LOC	SIZE
A1	3	14.50	⌀ 2.4 THRU ⌴ ⌀ 4.8 ↧ 2.2
A2	3	69	⌀ 2.4 THRU ⌴ ⌀ 4.8 ↧ 2.2
A3	5.50	3	⌀ 2.4 THRU ⌴ ⌀ 4.8 ↧ 2.2
A4	5.50	80.50	⌀ 2.4 THRU ⌴ ⌀ 4.8 ↧ 2.2
B1	11	2.50	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B2	11	81	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B3	15	2.50	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B4	15	81	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B5	85	2.50	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B6	85	81	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B7	89	2.50	⌀ 1.8 THRU 2-56 UNC - 2B THRU
B8	89	81	⌀ 1.8 THRU 2-56 UNC - 2B THRU
C1	11.50	29	⌀ 2.4 THRU ALL ⌴ ⌀ 4.8 ↧ 2.2
C2	11.50	54.50	⌀ 2.4 THRU ALL ⌴ ⌀ 4.8 ↧ 2.2
C3	11.50	70.80	⌀ 2.4 THRU ALL ⌴ ⌀ 4.8 ↧ 2.2
C4	11.50	75.80	⌀ 2.4 THRU ALL ⌴ ⌀ 4.8 ↧ 2.2



NOTE: DIMENSIONS IN TABLE ABOVE IN MILIMETERS EXCEPT FOR THREAD CALLOUTS, WHICH ARE STANDARD THROUGH HOLES A1-B4 ARE ALL STANDARD #2 THROUGH HOLES

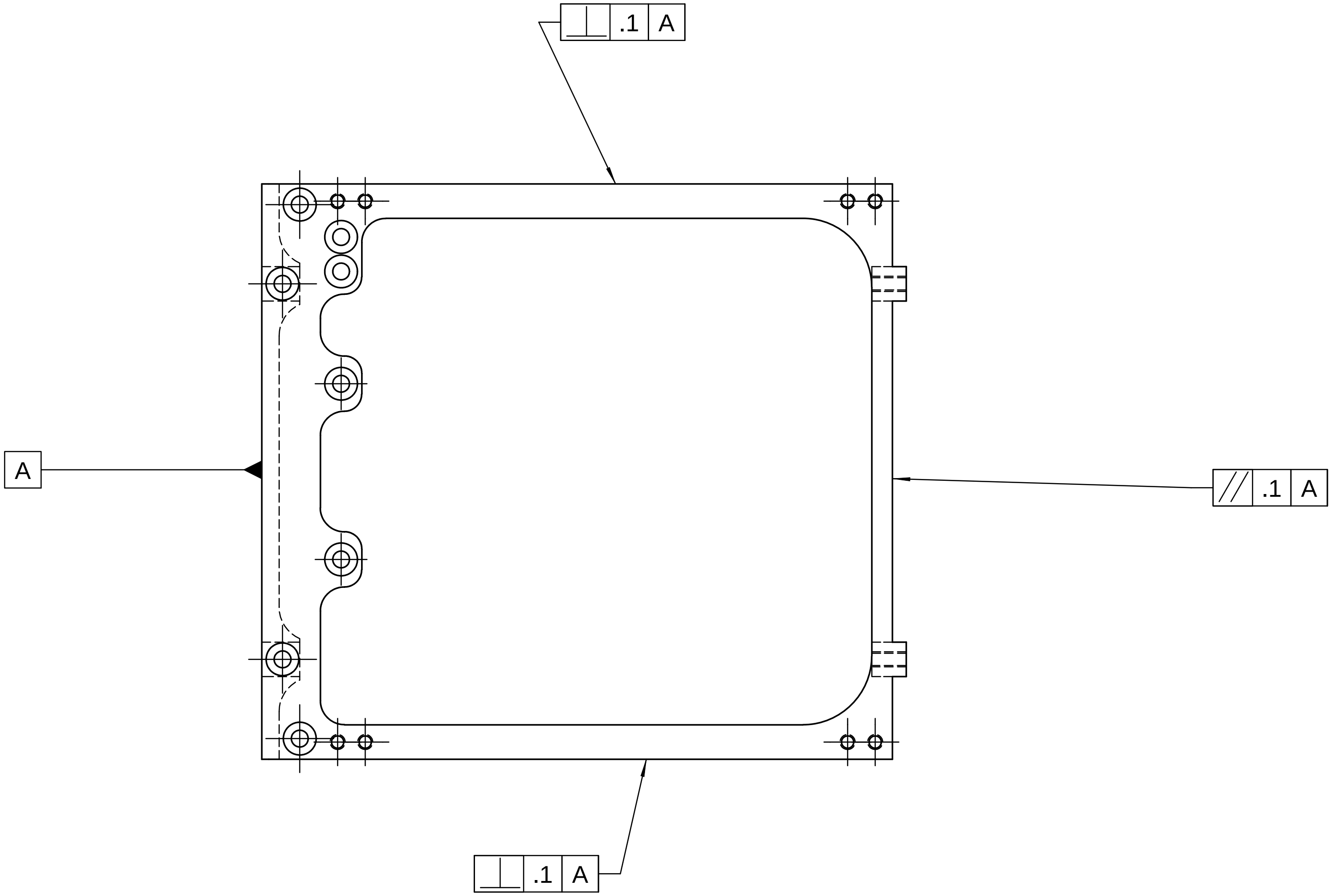
HYPERCUBE

Dimensions are in mm		NAME	DATE
Tolerances: ± .1 mm	DRAWN	LW	4/20/10
MATERIAL ALUM 6061-T6	COMMENTS:		
FINISH Hard Anodized			
SCALE: 1:1			

TITLE: **Side Panel**

SIZE	DWG. NO.	1103
A	REV 7	SHEET 2 OF 3

4 3 2 1

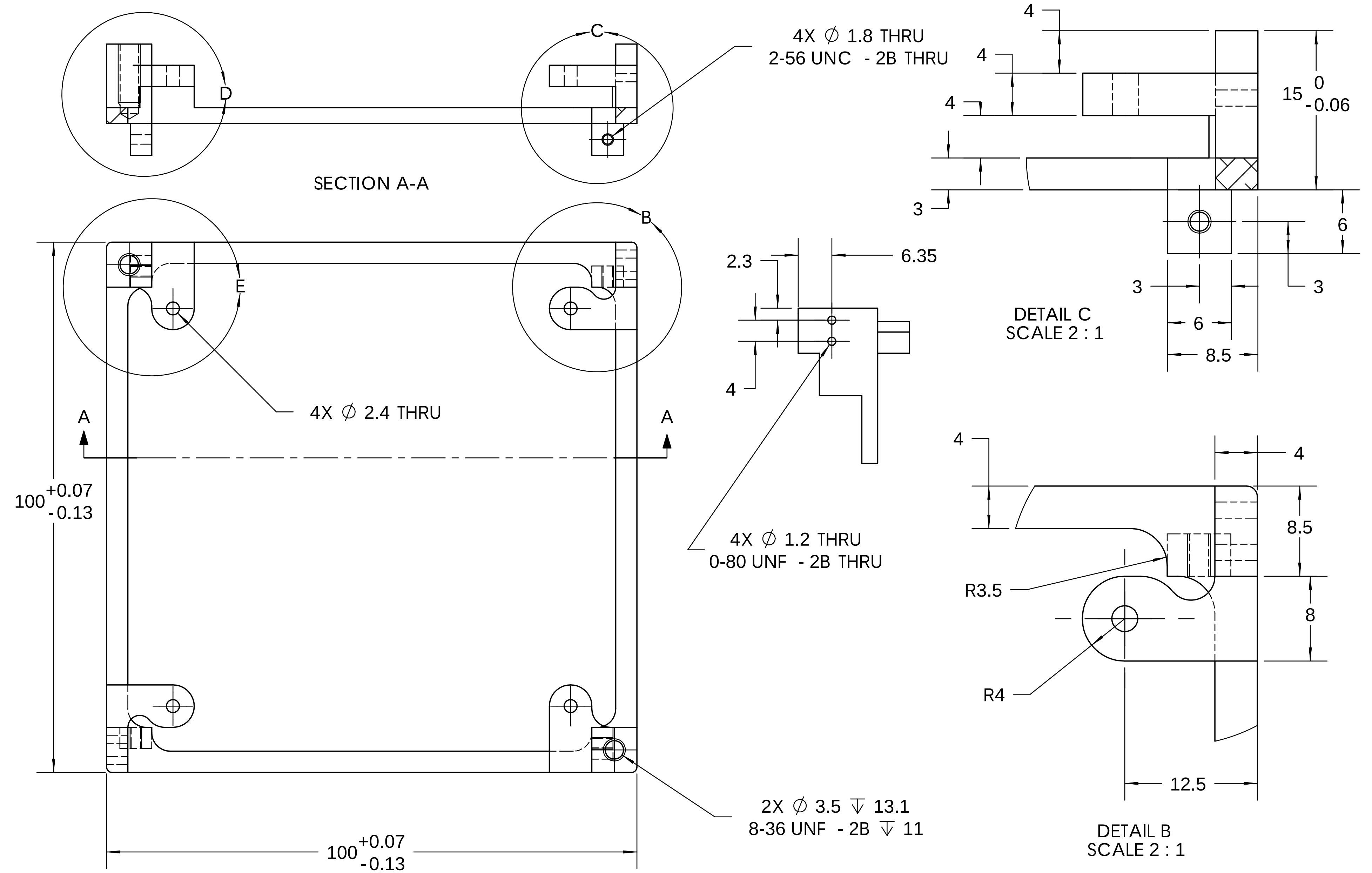


Note: Areas between 2 mounting tabs is okay to be out of tollerance.

Dimensions are in mm		NAME	DATE	TITLE:
Tolerances: ± .1 mm	DRAWN	LW	4/20/10	Side Panel
MATERIAL ALUM 6061-T6	COMMENTS:			
FINISH Hard Anodized				SIZE A / DWG. NO. 1103
SCALE: 1:1				REV 7 / SHEET 3 OF 3

HYPERCUBE

4 3 2 1



Note: Hole callouts are english sizes with dimensions in mm

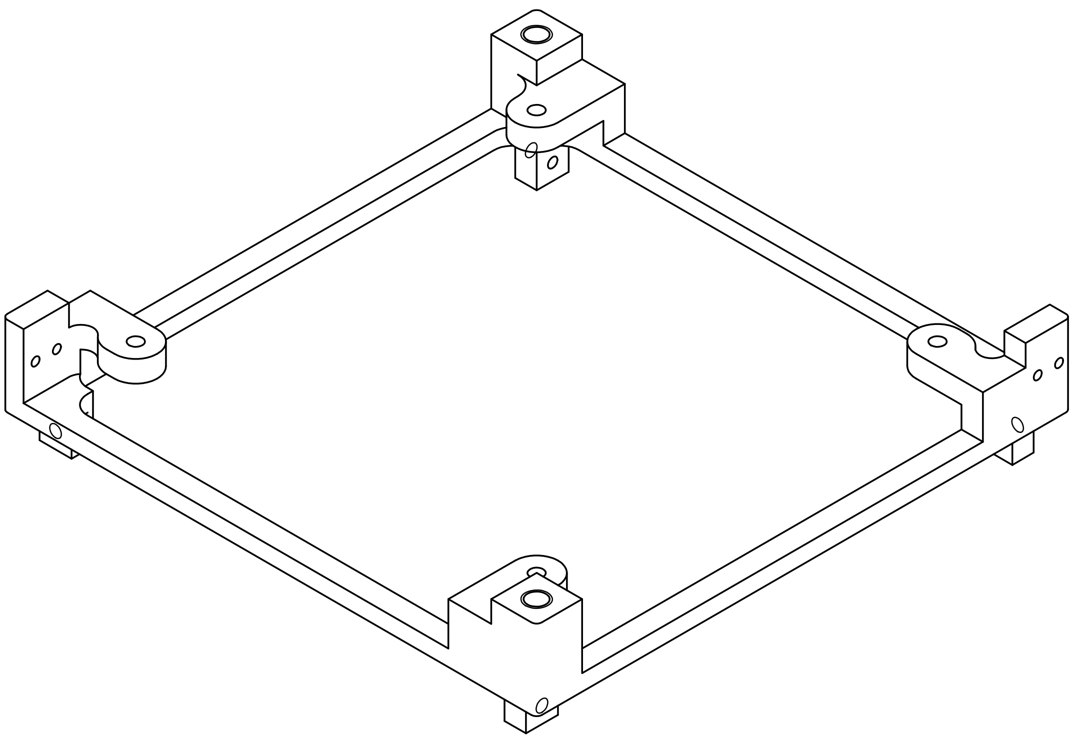
Dimensions are in mm		NAME	DATE
Tolerances: ± .1 mm	DRAWN	JD	9/28/10
MATERIAL ALUM 6061-T6	COMMENTS:		
FINISH Hard Anodized			
SCALE: 1:1			

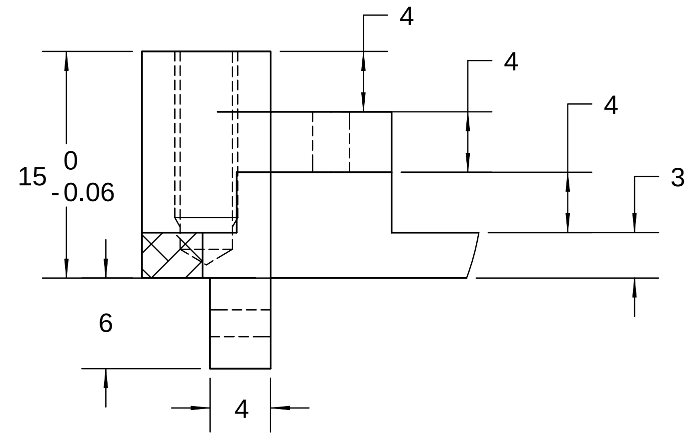
TITLE: Hat

SIZE A | DWG. NO. 1101

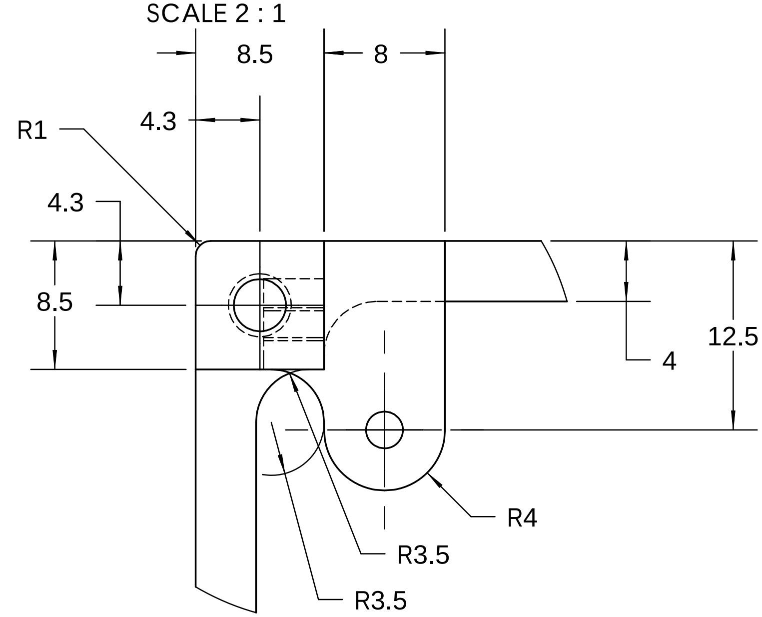
REV 1 | SHEET 1 OF 2

5 4 3 2 1





DETAIL D
SCALE 2 : 1

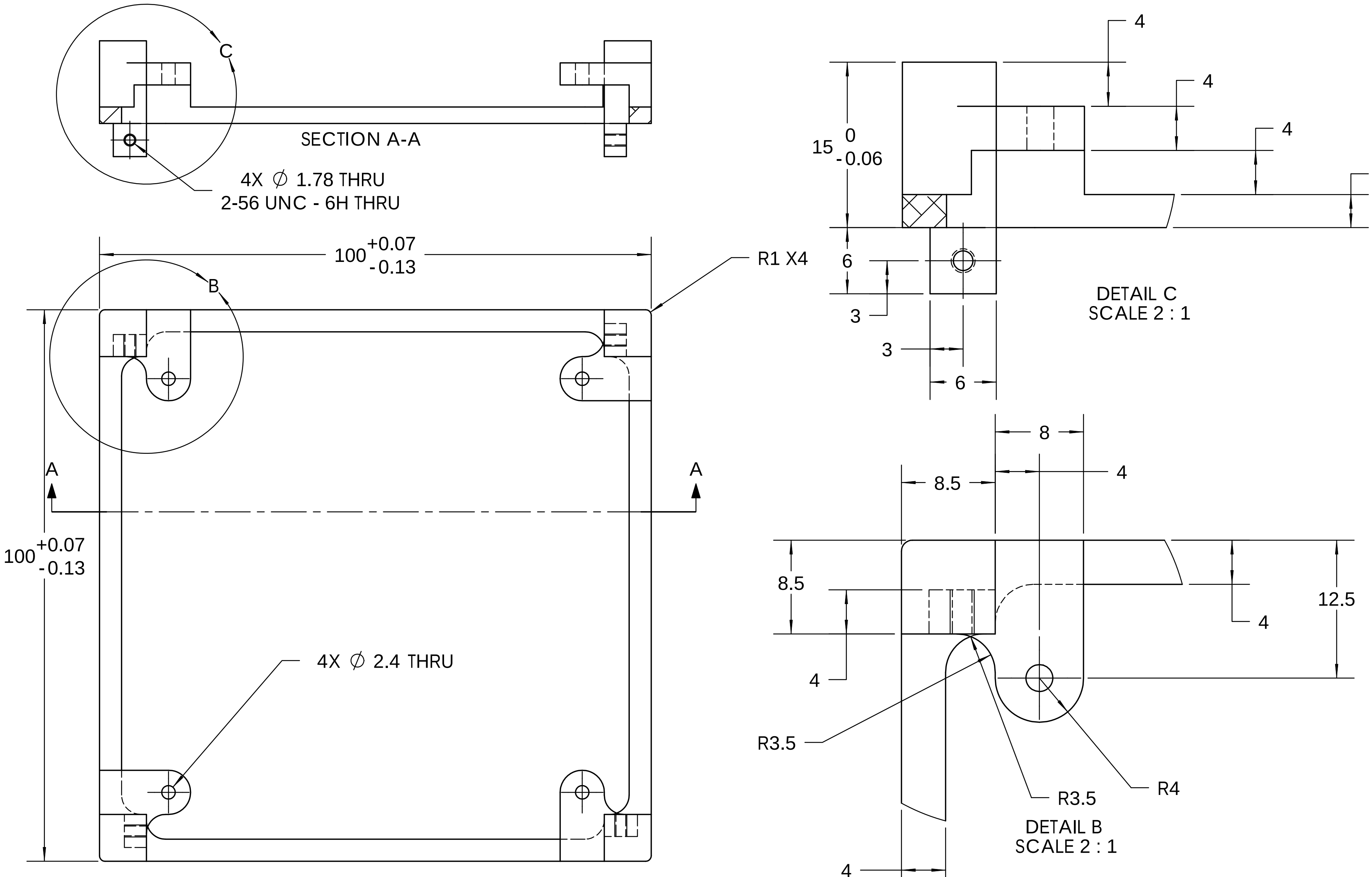


DETAIL E
SCALE 2 : 1

HYPERCUBE

Dimensions are in mm		NAME	DATE	TITLE:
Tolerances: ± .1 mm	DRAWN			Hat
MATERIAL ALUM 6061-T6	COMMENTS:			SIZE A · DWG. NO. 1101
FINISH Hard Anodized				REV · SHEET 2 OF 2
SCALE: 1:1				

5 4 3 2 1

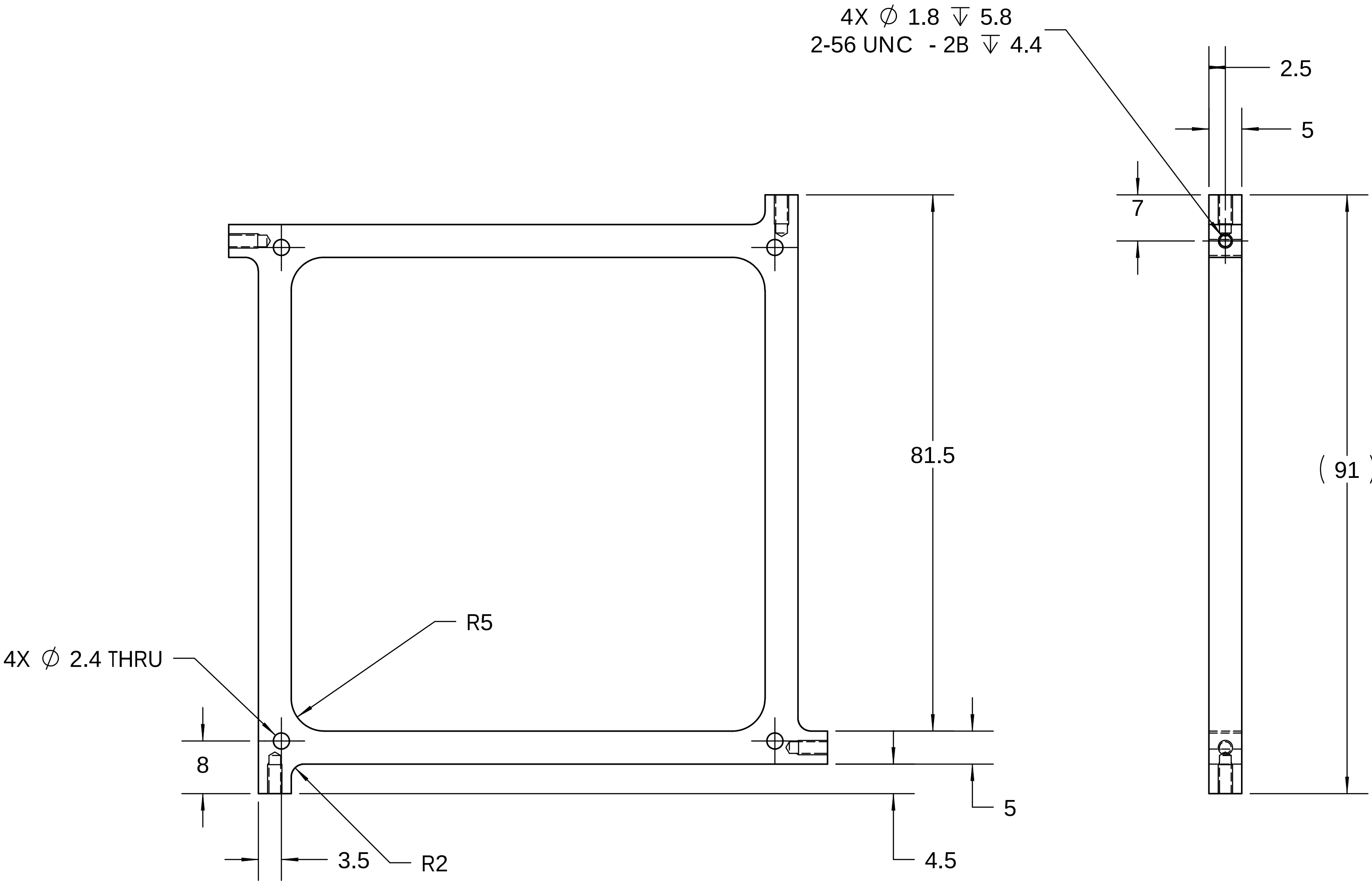


Note: Each corner is same just rotated

Dimensions are in mm		NAME	DATE	TITLE:
Tolerances: ± .1 mm	DRAWN			Shoe
MATERIAL ALUM 6061-T6	COMMENTS:			SIZE A
FINISH Hard Anodized				DWG. NO. 1102
SCALE: 1:1				REV
				SHEET 1 OF 1

HYPERCUBE

5 4 3 2 1



Dimensions are in mm		NAME	DATE	TITLE: Cross Support
Tolerances: ± .1 mm	DRAWN	JD	9/28/10	
MATERIAL ALUM 6061-T6	COMMENTS:			SIZE A · DWG. NO. 1104
FINISH Hard Anodized				REV 1 · SHEET 1 OF 1
SCALE: 1:1				

HYPERCUBE

5 4 3 2 1

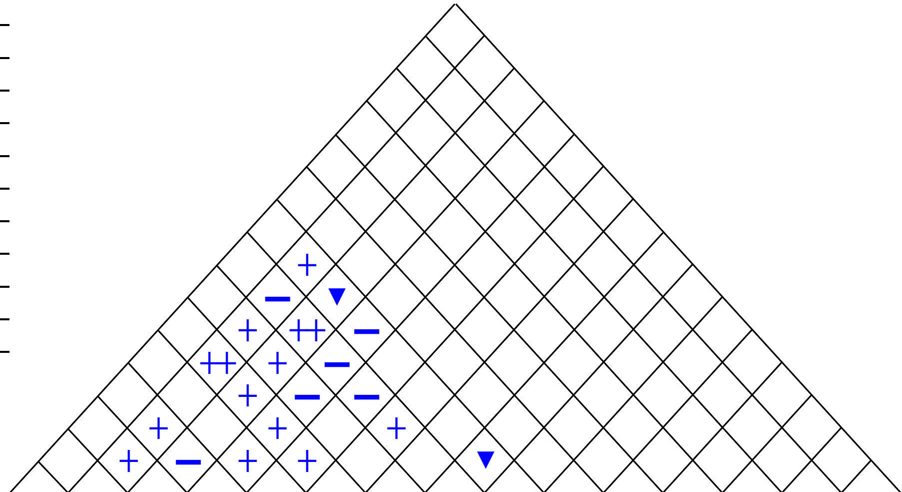
Title: HyperCube

Author: James Dolengewicz, Lucas Whipple, Stephanie Wong

Date: 1/27/2010

Notes:

Legend		
Θ	Strong Relationship	9
O	Moderate Relationship	3
▲	Weak Relationship	1
++	Strong Positive Correlation	
+	Positive Correlation	
—	Negative Correlation	
▼	Strong Negative Correlation	
▼	Objective Is To Minimize	
▲	Objective Is To Maximize	
x	Objective Is To Hit Target	



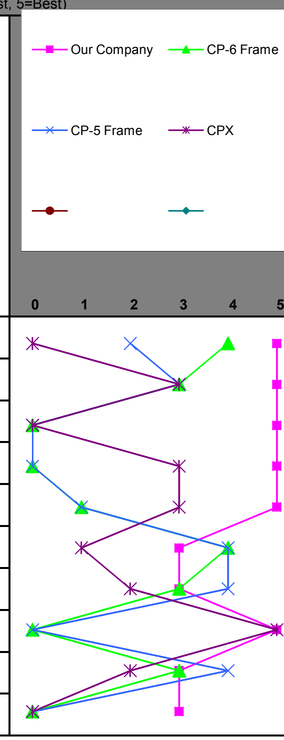
Direction of Improvement: Minimize (▼), Maximize (▲), or Target (x)

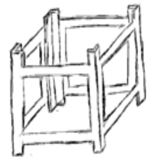
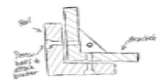
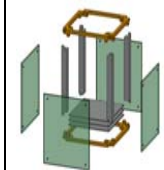
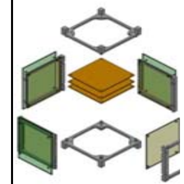
Quality Characteristics (a.k.a. "Functional Requirements" or "Hows")

Demanded Quality (a.k.a. "Customer Requirements" or "Whats")

Competitive Analysis (0=Worst, 5=Best)

Row #	Max Relationship Value in Row	Relative Weight	Weight / Importance	Demanded Quality	1	2 Modular Rails	3 Board-integral structure	4 Adjustable Mount Points	5 2-D Machined Parts	6 Mission-specific cross members	7 Design around new electronics board	8 Minimal stucture to support loading	9 Easy disassembly (for access)	10	11	12	13	14	15	Our Company	CP-6 Frame	CP-5 Frame	CPX
1	9	15.0	15.0	Ease of Access		O				O			Θ							5	4	2	0
2	9	15.0	15.0	Volume Maximization			Θ		▲	Θ	Θ	O								5	3	3	3
3	9	15.0	15.0	Exploitataion of new electrical hardware			O				Θ									5	0	0	0
4	9	15.0	15.0	Flexable mount points		Θ		Θ		O										5	0	0	3
5	9	10.0	10.0	Outward expansion into PPOD			▲	Θ												5	1	1	3
6	9	8.0	8.0	Ease of production		Θ			Θ											3	4	4	1
7	9	8.0	8.0	Low cost		Θ			Θ											3	3	4	2
8	9	8.0	8.0	1-2-3 u modularity		Θ				Θ										5	0	0	5
9	9	4.0	4.0	low mass			Θ					Θ								3	3	4	2
10	9	2.0	2.0	1/2-1-1/2 u modularity		Θ				Θ										3	0	0	0
				Target or Limit Value		1,2,3U	N/A	6+ Points	N/A	N/A	N/A	200 grams	4 Sections										
				Difficulty (0=Easy to Accomplish, 10=Extremely Difficult)		3	7	5	5	7	2	4	6										
				Max Relationship Value in Column		9	9	9	9	9	9	9	9										
				Weight / Importance		414.0	226.0	225.0	159.0	315.0	270.0	81.0	135.0										



	1	2	3	4	5	6	7	8	9	10
	1 piece	2 halfs	clamshell	tabbed rail	L rail	profiled	notched Rail	Rail-panels	Hat	inverted Hat
										
1	-	s	D	-	s	-	-	+	+	+
2	+	s	A	-	-	-	-	s	-	-
3	-	-	T	s	s	-	+	-	+	+
4	s	s	U	-	s	-	+	+	-	+
5	+	s	M	s	+	+	+	+	+	+
6	-	-		-	+	+	s	s	-	+
7	+	s		s	-	s	s	+	s	s
8	-	s		s	+	+	s	+	+	+
9	+	s	D	-	-	-	-	+	+	+
10	s	s	A	+	+	+	+	+	+	+
Σ+	4	8	T	5	4	4	4	7	6	8
Σ-	4	2	U	1	3	5	3	1	2	1
Σs	2	0	M	4	3	1	3	2	1	1

1	Manufacturing cost/time
2	Number of parts
3	Ease of assembly
4	Ease of attaching solar panel stacks
5	Large payload space
6	Ease of using a large electronics stack
7	Lightweight
8	Able to be modular
9	Stiffness/strength
10	Makes use of avalible stickout space

Finite Element Analysis on the Next Generation PolySat Structure

Stephanie Wong
Member of the CubeSat Program and HyperCube Senior Project Team
Analysis done for the HyperCube Senior Project Team
Aided by the Applied Finite Element Analysis Class and Dr. James Meagher
Winter 2010 at Cal Poly

ABSTRACT

The concept of a small satellite that can fit in ones hand has brought about a tremendous amount of interest and support due to the increase in accessibility of secondary payloads to hitch a ride to space. The HyperCube senior project team has been presented with constructing Cal Poly's next generation satellite. Currently there are three Cal Poly, student built, satellites orbiting the Earth.

The purpose of this finite element analysis is to verify that the current HyperCube design can withstand launch loads, particularly the vibration. Analyzing the buckling and deflection values of the structure, the results of the critical buckling load is at a factor of safety of 44, not including the factor of safety of 2.5 applied to the calculated load. The deflection is also well within range of less than 0.1 millimeter. With a structure this rigid, how can anything go wrong?

INTRODUCTION

Picosatellites have become a racing phenomenon majorly due to the art of standardization. Contained in a ten centimeter cube is all the components necessary for a functional satellite. These components include the structure, communications boards, power boards and solar panels. Weighing roughing one kilogram, one may wonder what on Earth can anyone do in that small of a space. With the advancements in technology in miniaturizing components for computers and cell phones, these tiny satellites have become a power platform for scientifically rich experiments. From biology experiments to revolutionary propulsion systems that require no fuel, these small structures have been harboring great opportunities which are a cheaper alternative to multimillion dollar satellites. The idea started in 1999 with Dr. Jordi Puig-Suari (Cal Poly Aerospace Department professor) and Dr. Bob Twiggs (former Stanford University professor) who thought of the idea to give students an opportunity to develop and launch satellites rapidly at a low cost. This resulted in a new class of picosatellites – called CubeSats – with the main propose of having students heavily involved in the complete life cycle of a space mission.

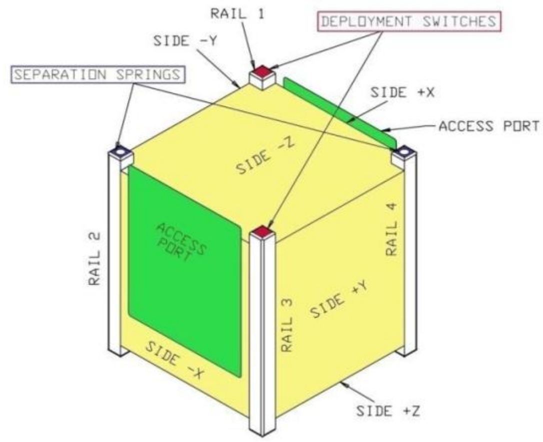


Figure 1: CubeSat Design Specification drawing that shows one example of the standardization process (Reference 2)

Presented with the HyperCube's Senior Project of analyzing Cal Poly's next generation picosatellite structure, the goal is to ensure that the structure will not fail during worst case launch environments, in this case the random vibration requirements. Below is the random vibration profile according to NASA GEVS (Reference 3) for random qualification.

Table 1: Random Qualification Vibration profile

Frequency [Hz]	ASD [G^2/Hz]
20	0.026
80	0.160
500	0.160
2000	0.026

Random vibrations simulate launch environments and depending on the launch vehicle and placement of the satellites, the ride can either be smooth or simply unbearable. Figure 2 shows the typical vibration profile for random qualification that the structure needs to survive.

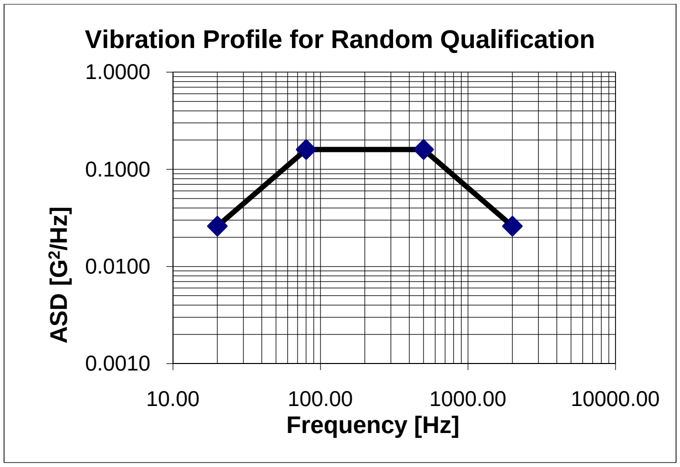


Figure 2: Random Qualification vibration profile with respect to acceleration spectrum density and frequency

Although most primary satellites on a launch vehicle is mounted directly onto the rocket, the picosatellite requires a deployment system, developed by Cal Poly students, called the Poly-Picosatellite Orbital Deployer (P-POD). The P-POD is a hollow box, consisting of a spring and door similar to that of a "Jack-in-a-Box" to deploy the picosatellite. Once the launch vehicle sends an electrical pulse to the door actuation device, the door will open and the spring will deploy the picosatellites out of the P-POD. With the ability to hold three picosatellites, the worst case scenario for the loading onto the satellite structure would be if the P-POD is upright during launch and the Cal Poly structure is located at the bottom of the other two picosatellites.

In order to characterize the structure, the dynamic environmental loads were converted into a representative static load in order to do a static load analysis in the finite element analysis program, W. Lan's thesis will be used to calculate the maximum acceleration (G's) from the launch vehicle environment (Appendix A and Reference 1). The structure's integrity will go through buckling analysis and the worst case applied loads analysis to find critical loads and deflection.

SIMPLIFIED STRUCTURE

Due to the complexities of the picosatellite structure, various steps are taken to develop a simplified model. First the actual structure will be analyzed as an assembly and then the loads will be analyzed based on simple geometric shapes that are derived from the model. The goal is to simulate the actual structure in a simplified matter using basic shapes. That way the simplified model will be able to mesh quickly and run the analysis faster, rather than having the software deal with the complex geometries and large amounts of elements due to multiple uneven surfaces and holes.

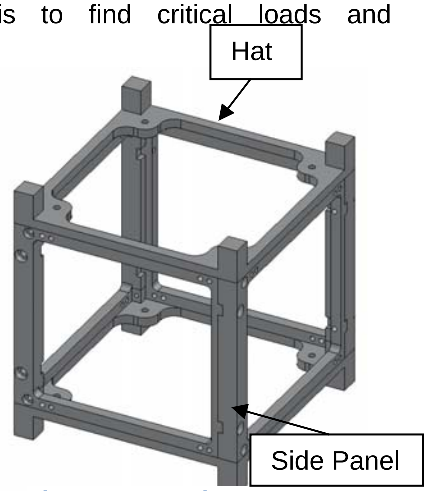


Figure 3: Actual structure assembly

Figure 4: Side Panel CAD model

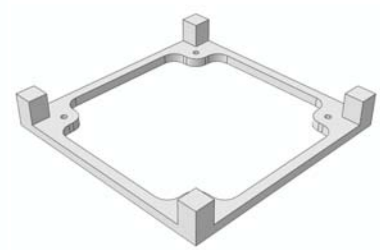


Figure 5: Hat Part CAD model

MODEL DEVELOPMENT – Made up of two main components, the actual structure contains four Side Panels and two Hat parts. The Side Panels create the frame work of the "walls" of the satellite. Solar panels and electronic boards will mount onto this part which is why the structure needs to be rigid enough to ensure that the boards do not break. In addition, there are two Hat parts that sandwich the Side Panel assembly together. One of the Hat parts will also hold a stack of electronic boards that will make up the brains of this satellite. The stack will also be wired to the solar panels and boards that are connected to the Side Panels. For this phase of the project, the electronic boards and solar panels will be neglected for two main reasons: the boards is not a structural component of the system and should not be considered as such, and the exact placement of the boards is still yet to be determined.

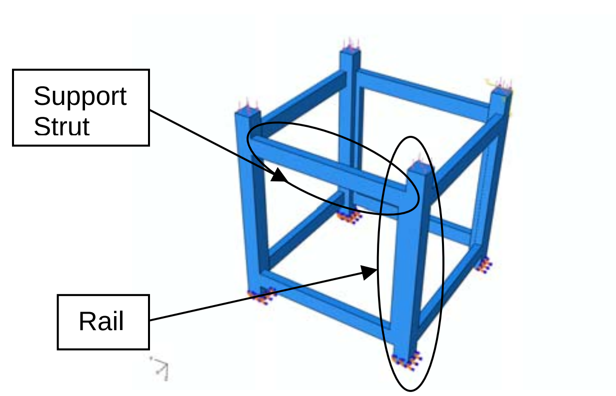


Figure 6: Simplified structure indicating main geometries

Since the assembly mated and preloaded together in designated slots and surfaces, the assembly will be considered as one part for simplicity's sake. Once put together, the structure begins to look similar to that of four rails and eight support struts. The four square columns will be called the rails since they are the points

that the satellite will interface with the P-POD. Thus the support struts will just hold the rails parallel to each other. In addition, to make the model FEA friendly, the rails and supports will morph into one rigid structure to avoid the complications of mating and part interaction.

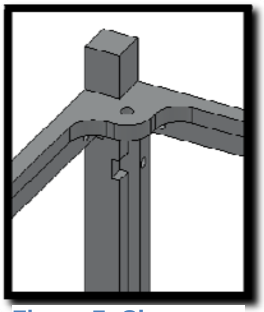


Figure 7: Close-up of an attachment point for an electrical board

Besides uniting the parts as one, the electronic board attachment points (Figure 7) will be cut extruded around the rail since they are not load supporting features. The holes in the model will also be filled for smooth surfaces.

MESH DEVELOPMENT – The mesh element sizes for this structure is tetrahedral since the assembly is not a simple geometric part. However, the tetrahedral elements are small enough to get into the tight spaces between the non-uniform features. For a more precise mesh, hexagonal elements could have been made on each individual surface on the structure. That is if the surfaces are partitioned around the joining of the rails and support members. In addition to the tetrahedral sizes, the meshes are linear and will evaluate 3D stress without the help of reduced integration, incompatibility mode and any other options the 'Tet' menu offers. The total count is 15 thousand elements at 3.5 millimeters which has about 90 thousand degrees of freedom. Below is a table showing the different mesh verifications ABAQUS offers.

Table 2: Mesh Quality Criteria

Element Failure Criteria	Values
Worse Shape Factor	0.198193
Worse Min Angle	19.58
Worse Max Angle	118.49
Worse Aspect Ratio	3.09
Worse Geometric Deviation Factor	2.21E-14
Shortest Edge	1.5
Longest Edge	6.19
Smallest Time Increment	56.5

ANALYSIS – Since the picosatellites are compressed in the P-POD, the forces are acting mainly on the rail portions of the structure. A design factor of safety of 2.5 will be used on the applied forces which will give the maximum force on the structure to be about 3180 N, see Appendix A for calculations. This load comes from the worse case random vibration loads seen by the Minotaur I launch vehicle. This force will be distributed between the four top surfaces and placed as a pressure amounting to 11 MPa on each top square surface. The boundary condition on the structure will be fixed at the four bottom surfaces which will simulate rigid compression in the P-POD.

The structure will be analyzed for buckling conditions, displacement, stress, and strain during worse case applied loads.

Buckling - The critical forces are found using the buckling step in the linear perturbation option in ABAQUS. To find the critical forces on the different models, the boundary conditions remain fixed at the bottom and a one unit total load is applied to the top. Since the cross-sectional areas of the model have square pressure points, the one unit total load is placed on one of the corner nodes because there is not a center node. Even though there may be discrepancies on the placement of the force load, this corner load presented a worse case load placement rather than placed in the center of the square cross-sectional area.

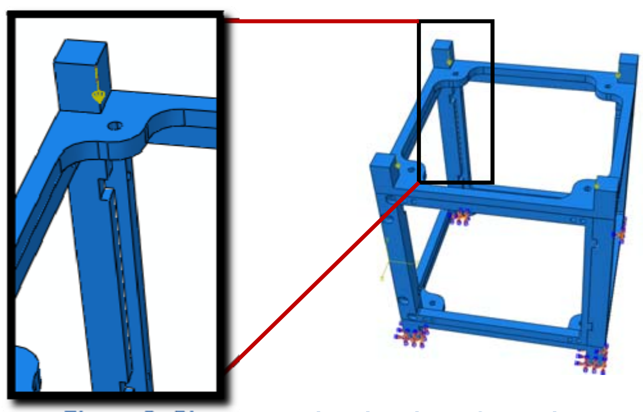


Figure 8: Close-up and entire view of actual structure subjected to Buckling loads

Besides the force placement, there is one error that the program sees is when the seeding gets smaller and smaller. When the seeds get too small, ABAQUS decides to aborts the analysis job due to too many iterations to get the Eigen value. When this occurs, the seeding in mesh elements are reduced. However this error is not consistent with the original prediction of small seeding because the large seeding meshes also have the same problem. To fix this problem, the seeding is changed to the normal sized seeds experienced with this size of an object.

In any case, the buckling value found will help find the critical forces the structure will be able to see without buckling. Thus, the converged value is 140 thousand Newtons. This value will help find the factor of safety between the critical forces and the actual forces so we can be safe to say that the structure will not break.

Applied Loads – In addition to the critical loads, the predicted characteristics of the structure is useful to know in case the structure is compromised in other places than the rails. ABAQUS's static/general command is used to find displacement, stress and strain when the model is under an applied load. A pressure of 11 MPa is placed on each of the four top surfaces and the bottom four surfaces are fixed. Below is the Table of the desired values with the applied force are at a factor of safety of 2.5.

Table 3: Summary of the displacement, stress and strain undergoing applied loads that have a factor of safety of 2.5

Desired Variables	FEA Values
Displacement (Magnitude)	1.76E-02 mm
Stress (Maximum Principal)	6.19 MPa
Strain (Maximum Principal)	81.6 µ

MESH CONVERGENCE – In order to ensure that the model is converged, the element sizes are changed until the values remain constant. ABAQUS can evaluate multiple elements but the less it has to evaluate, the faster the program can run. Thus, it is helpful to proceed with a mesh convergence study to see the number of elements that is necessary to get an acceptable reading.

To find the buckling value, the graph below indicates that the mesh converged at around 10 thousand elements. The model is meshed various times by changing the seed sizes. After numerous iterations, the outcome revealed a consistent number.

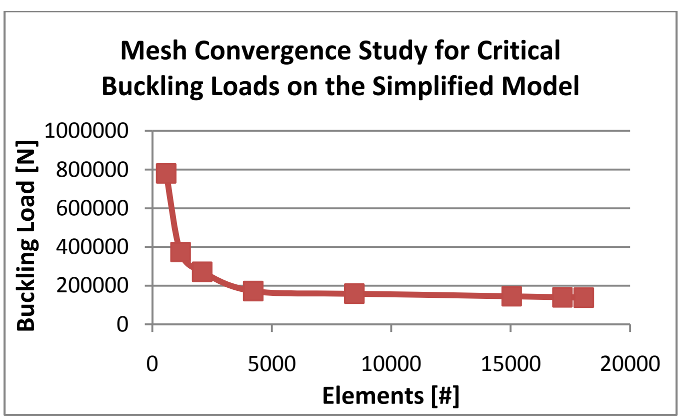


Figure 9: Buckling load at different element sizes which converges over larger elements

As for the deflection, the magnitude converges to 1.76E-2 millimeters at around 15 thousand elements. More testing has gone into trying additional element but to no avail. The small seeding is too much for the software to iterate. However, the graphs both show a steep slope and then a cutoff element where the elements start to steady out.

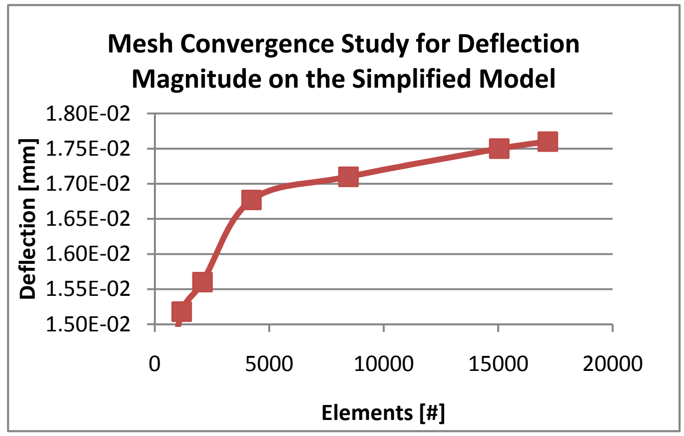


Figure 10: Converged value of deflection over several element size changes

This study goes to show that different variables may not have the same converging point. Thus, multiple convergence studies need to be made so that the largest element number may be chosen as the converged model for all variables. For these reasons, the stress and strain readings will not be considered since they do not converge.

The limitations of the seeding, for both the buckling and deflection values, are the amount of iterations the software has to process. The model seeding at 30 and 3 millimeters is unable to process due to the Eigen value iterations, as told by the command window in ABAQUS.

Overall the model underwent analysis in about a minute at 3.5 millimeter seeds. Compared to the actual structure's FEA, the simplified model had a much shorter run time.

RESULTS

Although the results in this section may seem all too easy to believe, it must be remembered that the finite element analysis just scratches the surface of the layers of analyzing imperfect and complex structures.

BUCKLING – Starting with the rail component of the structure, one can easily analyze the critical force in a square column (Appendix A). Multiplying the rail force by four, since there are four rails on a structure, the critical force comes out to be 24 thousand Newtons. Comparing that to the buckling finite element results, the buckling in the simplified structure will occur at 140 thousand Newtons. These two and very different calculations illustrate the importance of the supporting struts in the assembly, and that they assist the rails in buckling quit effectively.

According to the design load which is 3179 Newtons, and already at a factor of safety of 2.5, the addition

factor of safety is a huge 44 compared to the simplified structure! This value is well above the safety factors needed to ensure that the structure will not buckle under the pressure. The simplified model is also compared to a finite element model of the actual structure that was done as a case study to find the differences between the two (Appendix B). In the plot below, both structures look like they convergence at the same value due to the scaling. The simplified structure buckles at 140 whereas the actual buckles at 107 thousand Newtons. Even though the simplified model should be enveloping the actual model for worst case situations, the addition factor of safety to the 2.5 already in place, is 34. So the buckling issue is really not an issue at all.

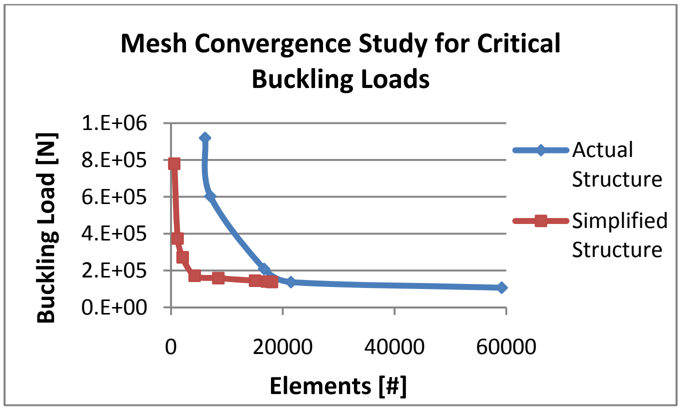


Figure 11: Mesh convergence comparison of the actual structure and simplified structure analyzed

Looking over Figure 11 again, the element cutoff for both structures is very different. This fact just showcases that a simplified structure uses less elements to converge to a steady value.

APPLIED FORCES – Applying the projected load onto the simplified and actual structure, the displacement values are the only values that are of interest, Table 4. The displacement will indicate if the structure will deflect during harsh loads.

Table 4: Displacement values from the finite element analysis of both the actual and simplified structure

Desired Variables	FEA Values of the Actual Structure	FEA Values of the Simplified Structure	Percent Error
Displacement	2.11E-2 mm	1.76E-2 mm	16.6%

The displacement of the actual and simplified structure differs at about 16.6%, assuming that the actual structure is the true value. Considering that the deflection is much less than one millimeter, the percent difference will not be an issue, though it is understood that the simplified structure should be more conservative than the actual one. These values justify the life of the electronic boards during their journey into space though the true test will be during a vibration test.

The stress and strain were going to also be evaluated except the mesh convergence values were oscillating. Meshing the parts with different seeding to its full range, the simplified model seems to go on a roller coaster ride whereas the actual structure seems to decide which way to go about mid way.

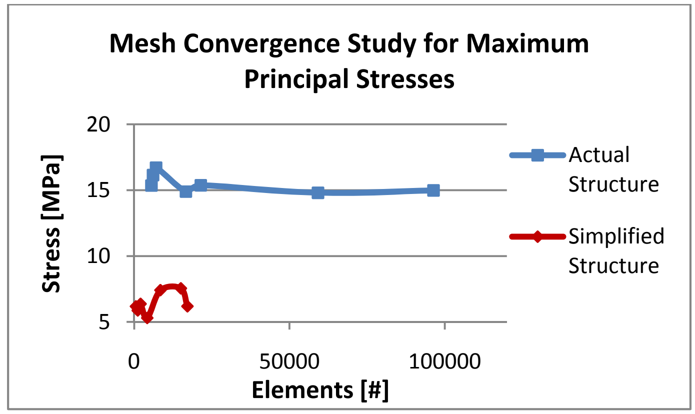


Figure 13: Maximum principal stresses in both actual and simplified structures

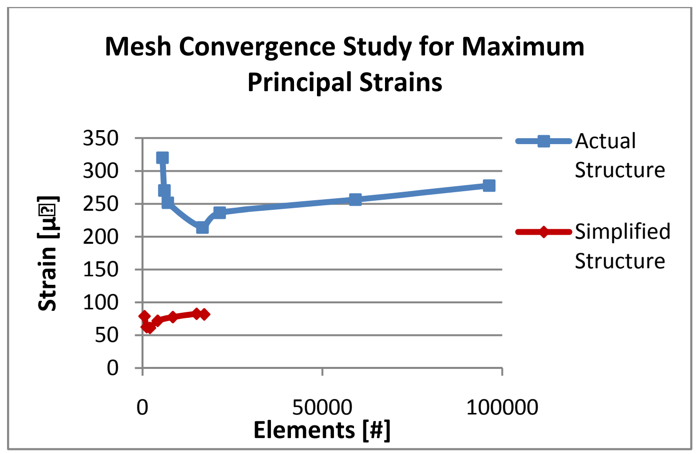


Figure 14: Mesh convergence for the maximum principal strains in the actual and simplified structures

Overall the stress and strain values are not critical to document expect if there is a lower factor of safety in buckling. The maximum stress and strain are also good indicators visually to input more rigidity or to alleviate stress concentrations on the structure like where the rail and support struts meet.

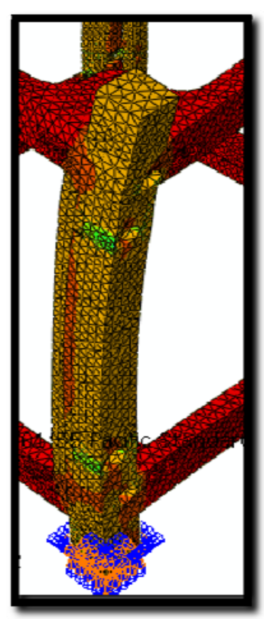


Figure 12: Close-up of the rail in stress on the actual structure

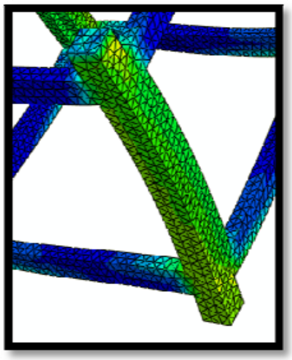


Figure 15: Close-up of the strain analysis on the rail of the simplified structure

DISCUSSION

As far as discrepancies go on the test data, the only ones that are concerning is the stress and strain values since the actual and simplified model values did not match. Given that the non-converged data was thrown out in the results section, this discrepancy did too. Other than that, the results matched the requirements of random qualification vibration.

Looking over the buckling results, it is pleasantly surprising how large the factor of safety is on the simplified structure. It has a factor of safety of 44 with an applied load to have a 2.5 factor of safety built in. Although the simplified model did not envelop the worst case scenarios, the factor of safety is so high that the applied loading does not matter unless it is near critical loading. This is assuming that the finite element model on the actual structure is correct and converged properly. In any case, it seems crazy that this little frame can handle so much during harsh vibrations. However, it will have to not only hold itself together, but hold the other subsystems attached to its frame.

The issues with the structure is going to be in the fastener placements and material around the fasteners seeing as the structure will be made of Aluminum 7075 and the screws will be stainless steel. The simplified structure eliminated the screws and mating pieces, but these calculations can easily be done using simple bolt and screw calculations done in Intermediate design classes at Cal Poly.

A recommendation is that the mass could be cut down since it is known that the weight of the picosatellite is limited to 1 or 1.33 kilograms (depending on the mission) according to the CubeSat Design Specfication. Also to avoid strain and stress between the rail and support struts, fillets will be good to use to avoid possible nicks.

CONCLUSION

Overall, the goal of the project is to analyze a simplified model of the actual structure to ensure it will not buckle, deform or break. Given a factor of safety of 44 on top of a factor of safety of 2.5, the structure will be able to survive fairly well on its own. Buckling is not an issue and once it is, other components are sure to break before that time. Deflection is minimal and thus deformation is too since the yield strength is high for Alumimum 7075. These are the two main issues that are caused by compression and because the picosatellite goes through compress only, the likely hood of the structure breaking is very unlikely.

Besides verifying the structure's integrity, the finite element model will be useful engineering tools since the PolySat team plans to use this structure for muilple future missions. Thus, the simplified structure will provide for easy manipulation if mission loads are changed. Given that the load has a factor of safety of 2.5, the structure will be very rigid.

ACKNOWLEDGMENTS

Special acknowledgments to HyperCube team who worked on the design of the structure. Thanks to Dr. Meagher for helping the confused (aka the writer) and listening patiently to CubeSat and PolySat presentations over and over. Another thanks to the CubeSat and PolySat teams for being open to ideas and once in a while letting me think outside the box. Also thank you Riki for teaching me a bit of NX Nastran and the confusion you face with that program daily.

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1. Lan, Wenschel. *Redesign and Modal Analysis of the Poly Picosatellite Orbital Deployer*. Thesis. San Luis Obispo, Calif. : California Polytechnic State University, 2008. Print.
2. CubeSat Acceptance Checklist*
3. Simmons, Ryan. "FEMCI Book - Calculating Grms." *FEMCI - Finite Element Modeling Continuous Improvement at NASA Goddard Space Flight Center*. Web. 01 Mar. 2010. <http://femci.gsfc.nasa.gov/random/randomgrms.html>.

*Documentation only available through the CubeSat

program at Cal Poly San Luis Obispo

CONTACT

Stephanie Wong is an undergraduate mechanical engineer at Cal Poly and is part of the CubeSat program where she spends most of her time working on missions regarding the P-POD.

sdwong@calpoly.edu

DEFINITIONS, ACRONYMS, ABBREVIATIONS

Here are the definitions and explanations of terms regarded in the report. If more information is desired, you can contact the author via email. Goggle is pretty useful too.

ASD: Acceleration Spectral Density in G^2/Hz

Cal Poly: California Polytechnic State University, San Luis Obispo

CDS: CubeSat Design Specification created by the CubeSat Program at Cal Poly

CubeSat: Cube Satellites made from the CubeSat Design Specifcation

FEA: Finite Element Analysis, computer aided numerical analyzer that takes a part(s) alone or in a system and analyzes the structure characteristics using numeric methods

Grms: Root-Mean-Square Acceleration, Reference 3, average of the square acceleration over time at a certain frequency, creates a magnitude from the range of acceleration I order to characterize the overall vibration profile

NASA: National Aeronautics and Space Administration

NASA GEVS: General Environmental Verification Standard, upheld by NASA

P-POD: Poly-Picosatellite Orbital Deployer, developed by the CubeSat Program to deploy picosatellites, interfaces with the launch vehicle, consists of a box and spring to eject the picosatellites once in orbit

APPENDIX A- HAND CALCULATIONS

Design Load Factor Calculation (W. Lan, Reference 1)

From NASA GEVS

$N_i = S_i \pm \sqrt{L_i^{\,2} + R_i^{\,2}}$;

N_i = combined load factor

S_i = steady-state load factor

L_i = low frequency dynamic load factor

R_i = high frequency random vibration factor

Worst-case (so far) is Minotaur I (numbers from LV Users guide):

$S_{longitudinal}$ = 7- 13g

$S_{latitudinal} = \pm 3.3g$

L_i = 5g (ICD)

R_i = 14.1g (from NASA GEVS)

$$N_x = N_y = \pm 3.3g \pm \sqrt{5.0g^2 + 14.1g^2} = \pm 18.26g$$

$$N_z = 13g \pm \sqrt{5.0g^2 + 14.1g^2} = 1.96g, 27.96g$$

$$N = \sqrt{N_x^{\,2} + N_y^{\,2} + N_z^{\,2}} = \sqrt{2(18.26g)^2 + 27.96g^2} = 38.06g$$

Design Force Calculation (on the picosatellite in the worst case position where the P-POD is vertical and the picosatellite in question is at the bottom) - Stephanie calculations

$F_{applied} = F_{satellites} + F_{springs}$;

where $F_{springs}$ includes the force of the main spring and spring plungers from the P-POD

$F_{satellites} = 9.8(mass)(\#\_of\_Satellites)(N) = 9.81(1.33kg)(2)(38.06g) = 995N$

$F_{main\ spring} = 44.5N$ (for a nominal CubeSat exit velocity of 1.8 m/s)

$F_{spring\ plunger} = 57.8N$ (max force from supplier specification)

$F_{applied} = 995N + 44.5N + 4(57.8N) = 1271.4N$

F.S. = 2.5

$F_{design} = 2.5(1271.4N) = 3178.6N$ (max design load on the entire structure including the factor of safety of 2.5)

$F_{design\_load\_on\_rail} = \dfrac{3178.6N}{4} = 794.6N$ (load on each of the four rails)

$F_{design\_pressure\_on\_rail} = \dfrac{794.6N}{(8.5E-3m)^2} \approx 11MPa$ (pressure on the rails, taking into account the cross-sectional area of the rails)

Critical Buckling Load

$$P_{cr} = \frac{\pi^2 \mathrm{EI}}{\mathrm{L}_{\mathrm{eff}}^2}$$

DEFINE TERMS
P_{cr}= Critical buckling force

E= Modulus of Elasticity for Al 7075

I= Moment of Inertia, simple square column

$\mathrm{L}_{\mathrm{eff}}$=Effective length for a fixed and free situation

DEFINE TERMS
E= 71.7 GPa (Note: This value is the average of tension and compression, compression modulus is 2% greater than tension modulus) Thus, E=72.417 GPa

$$I = \frac{bh^3}{12} = \frac{(0.0085m)(0.0085m)^3}{12} = 4.35E - 10\ m^4$$

$$\mathrm{L}_{\mathrm{eff}} = 2 * \mathrm{Length} = 2 * .1135\mathrm{m} = 0.227\mathrm{m}$$

$P_{cr} = \frac{\pi^2(72.417\mathrm{E}9)(4.35\mathrm{E}-10)}{(0.227)^2} = 6033.7\mathrm{N}$ (Critical force for one rail to buckle)

APPENDIX B- ACTUAL STRUCTURE ANALYSIS

Before getting the actual structure into ABAQUS, the parts need to be imported from Solidworks. Solidworks is a modeling program that can create 3D structures which is the program the HyperCube team uses for their picosatellite model. The assembly of parts is saved as a STEP file and imported into ABAQUS. The parts are then constrained to each other using the Tie command.

Mesh Development - The actual structure contains tetrahedral elements due to the complexity of the surfaces. There are more than 96 thousand elements, at seed size of 2 millimeters, due to the multiple surfaces of the Side Panels and Hat parts. Since all the elements can move in any direction except the limitation due to the boundary conditions and loads there are more than 576 thousand degrees of freedom. The mesh needs to have small seeding due to the complex geometry of the part, see Figures 16 and 17.

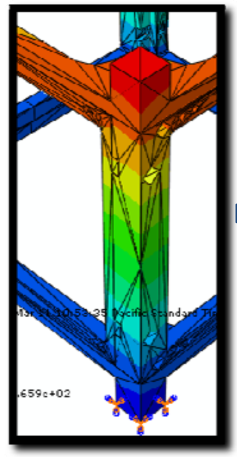

Figure 17: Tetrahedral elements with 5 millimeter mesh

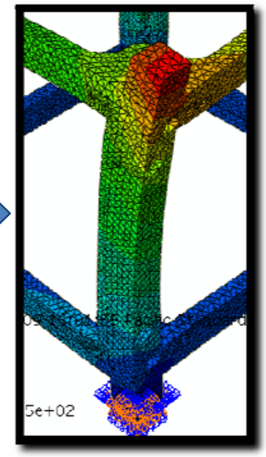

Figure 16: Tetrahedral elements with 3.15 millimeter seeds

Displacement

Table 5: Variables of the actual structure at different seed sizes

Elements [#]	Seeds [mm]	Deflection Magnitude [mm]	Stress Max Principal [MPa]	Strain Max Principal [μ]
5552	50	1.49E-02	15.4	320
6083	30	1.55E-02	16.2	270
7041	20	1.60E-02	16.7	252
16659	10	1.78E-02	14.9	214
21455	5	1.92E-02	15.4	236
59180	2.5	2.07E-02	14.8	256
96367	2	2.11E-02	15.0	278

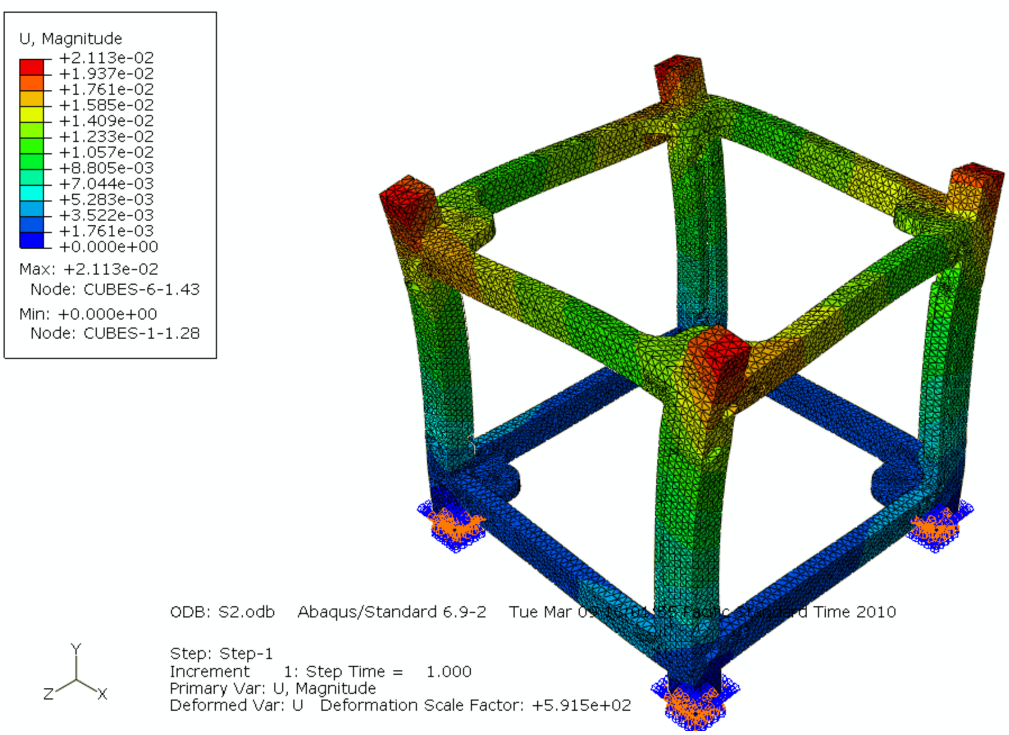

Figure 18: Displacement Magnitude of the actual structure with a 3.15 millimeter seed mesh

Buckling

Table 6: Elements, seed size and Eigen values show the different iterations in convergence

Elements [#]	Seed [mm]	Eigen value [N]
6083	30	9.20E+05
7041	20	6.03E+05
16659	10	2.09E+05
21455	5	1.37E+05
59180	2.5	1.07E+05

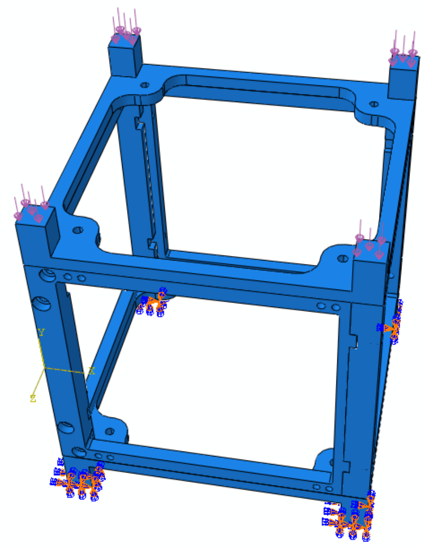

Figure 19: Pressure loads on the actual structure with fixed boundary condition at the bottom

Frequency

Table 7: Various mode frequencies using the converging mesh method, the larger elements have the converged values

Elements	Seeds	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
580	20	1766	1802	2506	2877	5881
1179	10	1278	1309	1623	1952	3965
2083	7	1067	1074	1399	1623	3100
4224	5	862	864	1122	1372	2749
8457	4	827	829	1060	1238	2296
15043	3.5	791	793	1012	1172	2229
17166	3.15	777	779	987	1067	1919

Cal Poly Structures & Composites Lab
Swept Sine Test Report

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RUN NAME: Hypercube Post X
USER/PROJECT FOLDER: Class Tests
SAVE NUMBER: 1

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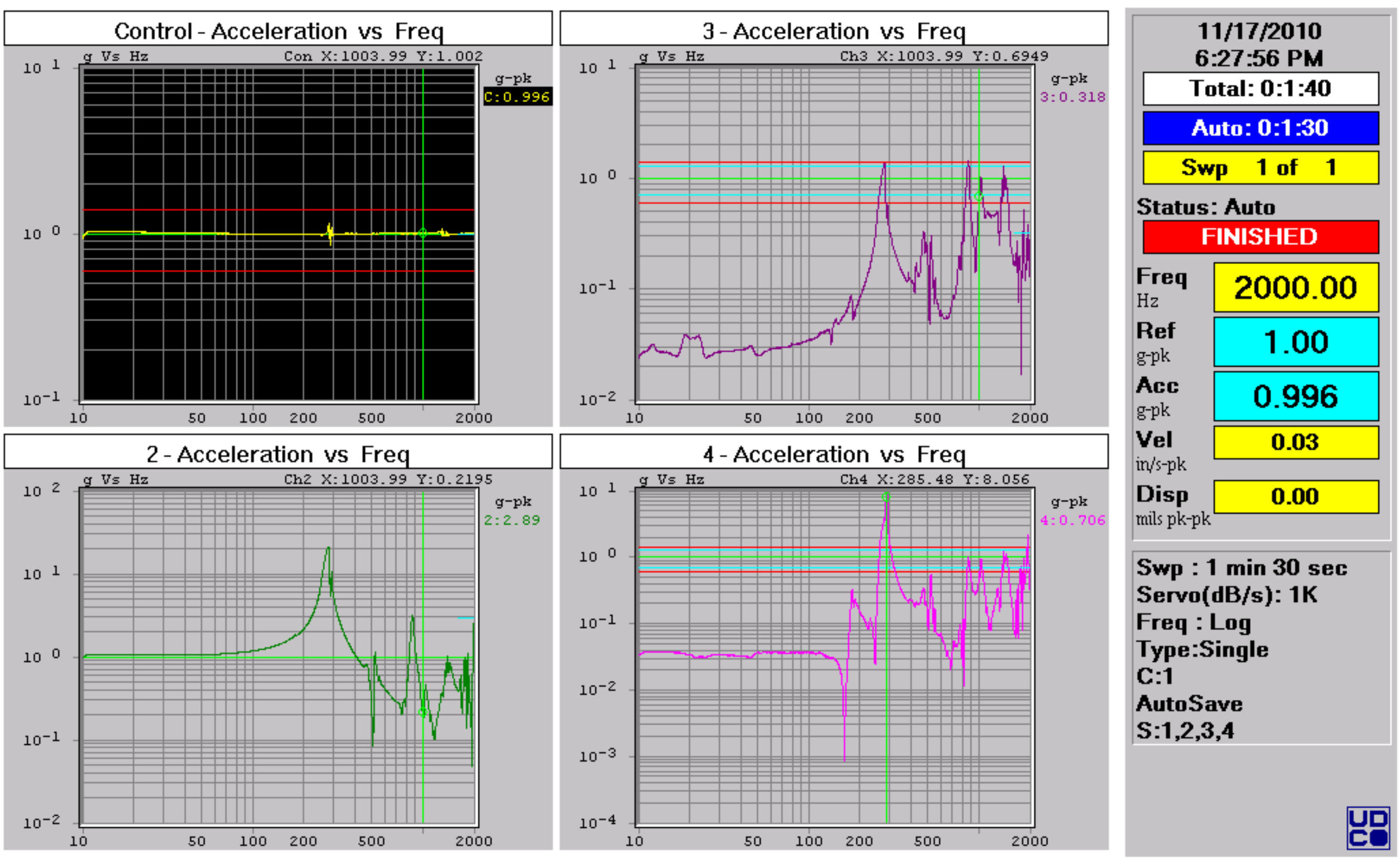
CONTROL VELOCITY: 0.03 in/s pk
CONTROL DISPLACEMENT: 0.00 mil pp



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LOWER FREQUENCY: 10.00 Hz
UPPER FREQUENCY: 2000.00 Hz
SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	**Sensitivity (mV/g)**	**Coupling (AC/DC)**	**Max.Range (g pk)**
1	9.74	AC	100.00
2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

Chan #	**Description**
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2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Random Test Report

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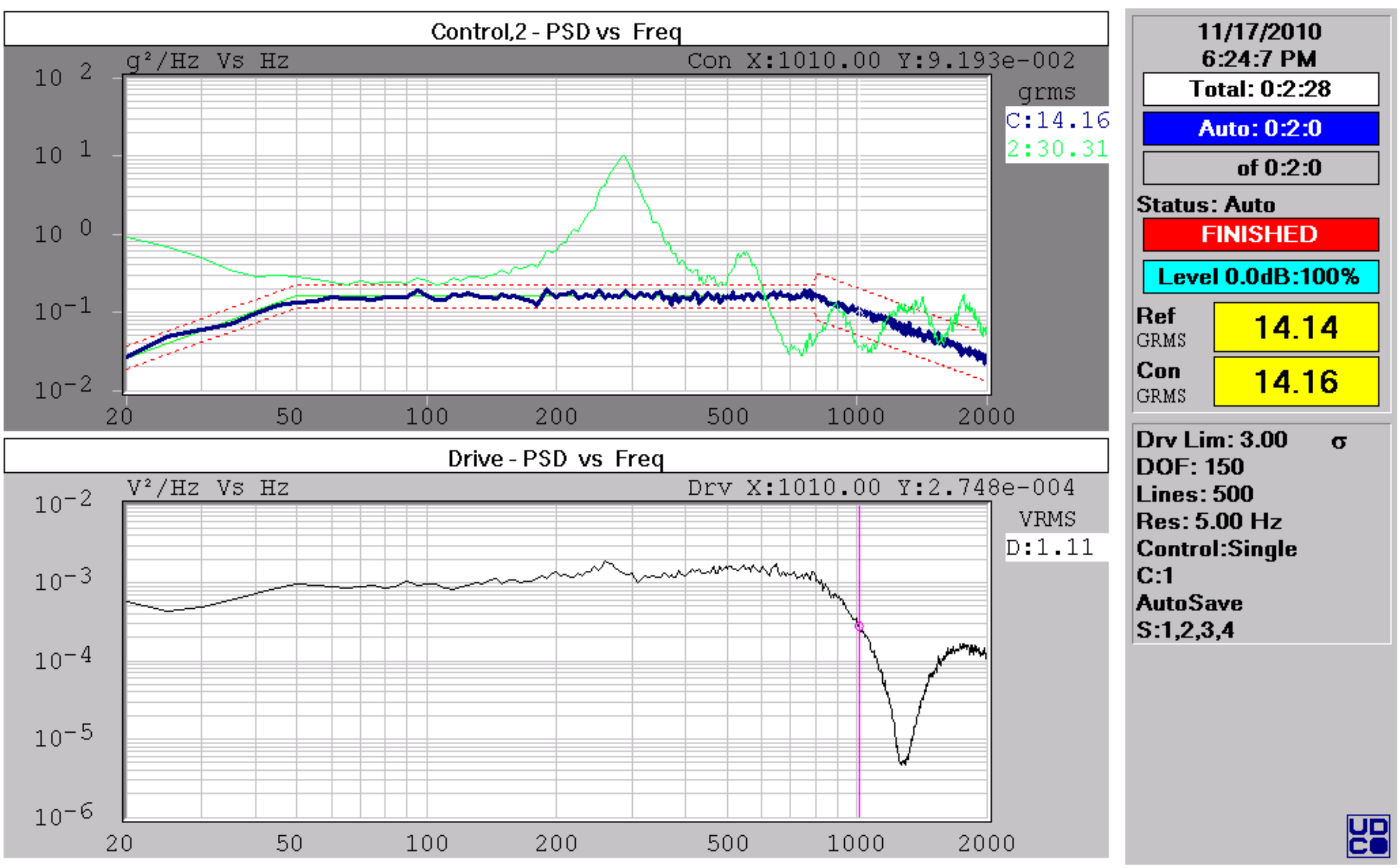

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2	103.70	AC	19.00
3	105.20	AC	19.00
4	102.90	AC	19.00

INPUT CHANNEL DESCRIPTION

Chan #	**Description**
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3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Swept Sine Test Report

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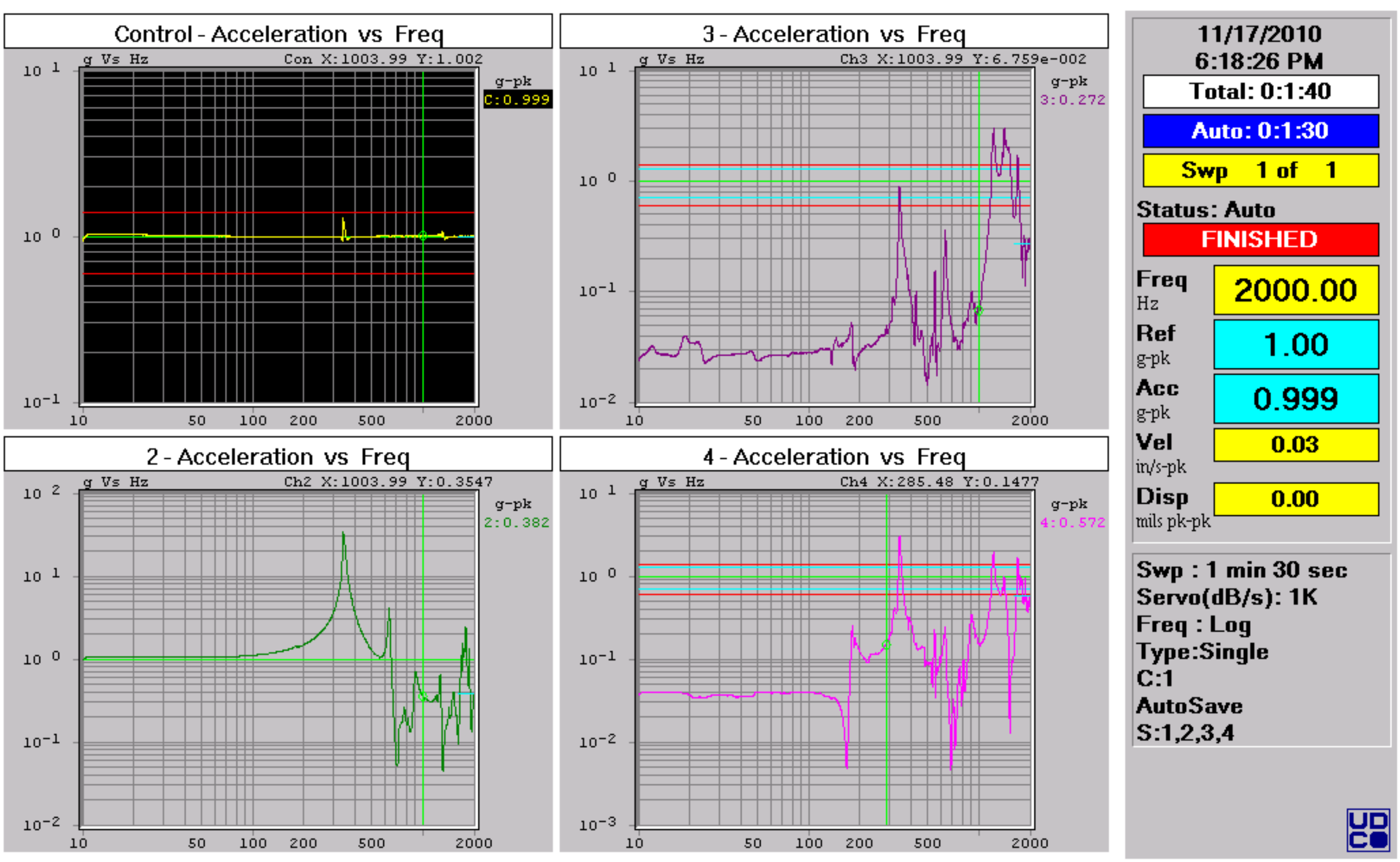

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SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	Sensitivity (mV/g)	Coupling (AC/DC)	Max.Range (g pk)
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2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

Chan #	Description
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Swept Sine Test Report

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RUN NAME: Hypercube Pre Y
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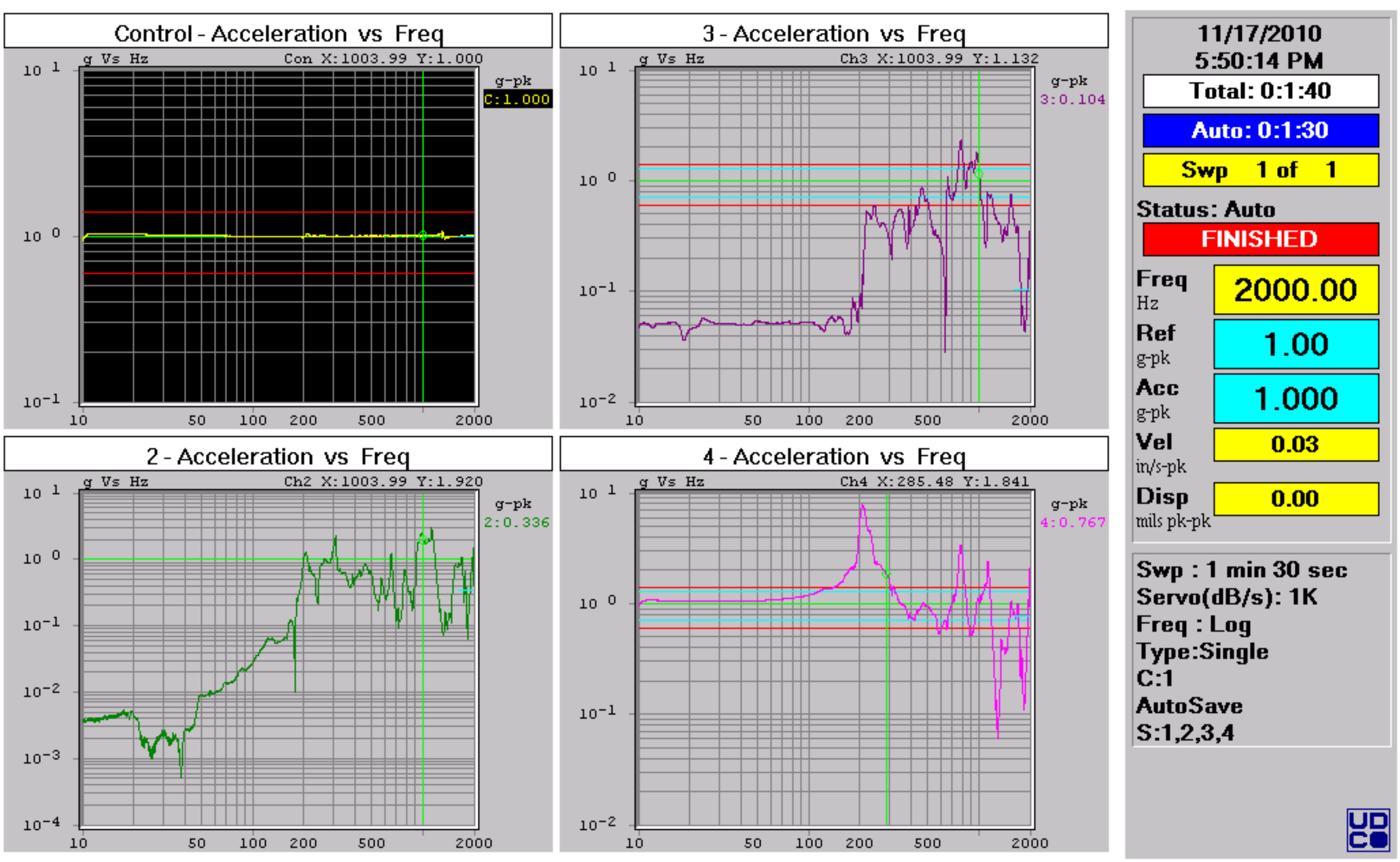
CONTROL DISPLACEMENT: 0.00 mil pp



CONTROL PARAMETERS

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LOWER FREQUENCY: 10.00 Hz
UPPER FREQUENCY: 2000.00 Hz
SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	**Sensitivity (mV/g)**	**Coupling (AC/DC)**	**Max.Range (g pk)**
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2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

Chan #	**Description**
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2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Random Test Report

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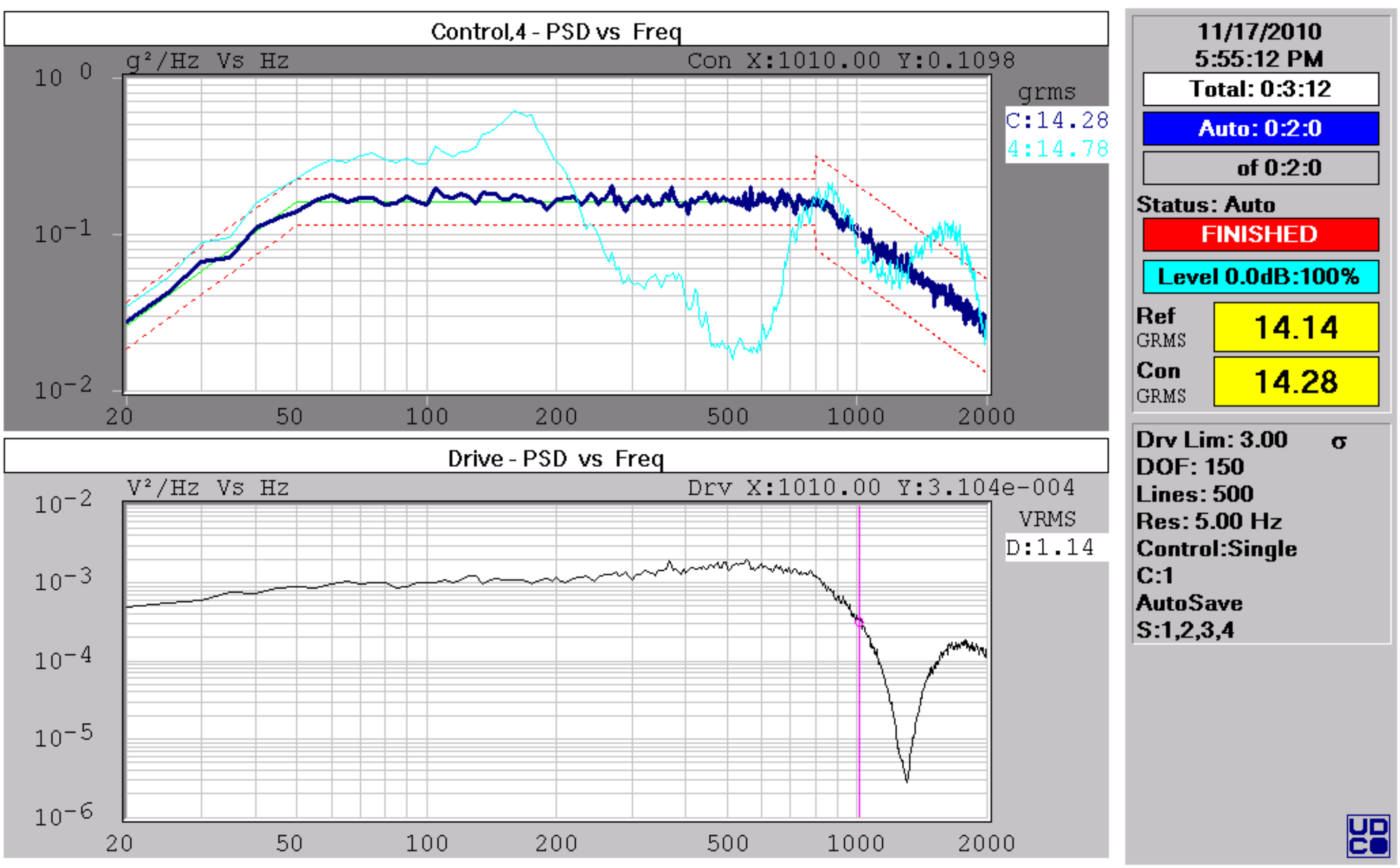

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FREQUENCY RESOLUTION: 5.000 Hz
DOF: 150
SIGMA DRIVE LIMITING: 3.00

INPUT CHANNEL PARAMETERS

Chan (#)	Sensitivity (mV/g)	Coupling (AC/DC)	Max.Range (g rms)
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2	103.70	AC	19.00
3	105.20	AC	19.00
4	102.90	AC	19.00

INPUT CHANNEL DESCRIPTION

Chan #	Description
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Swept Sine Test Report

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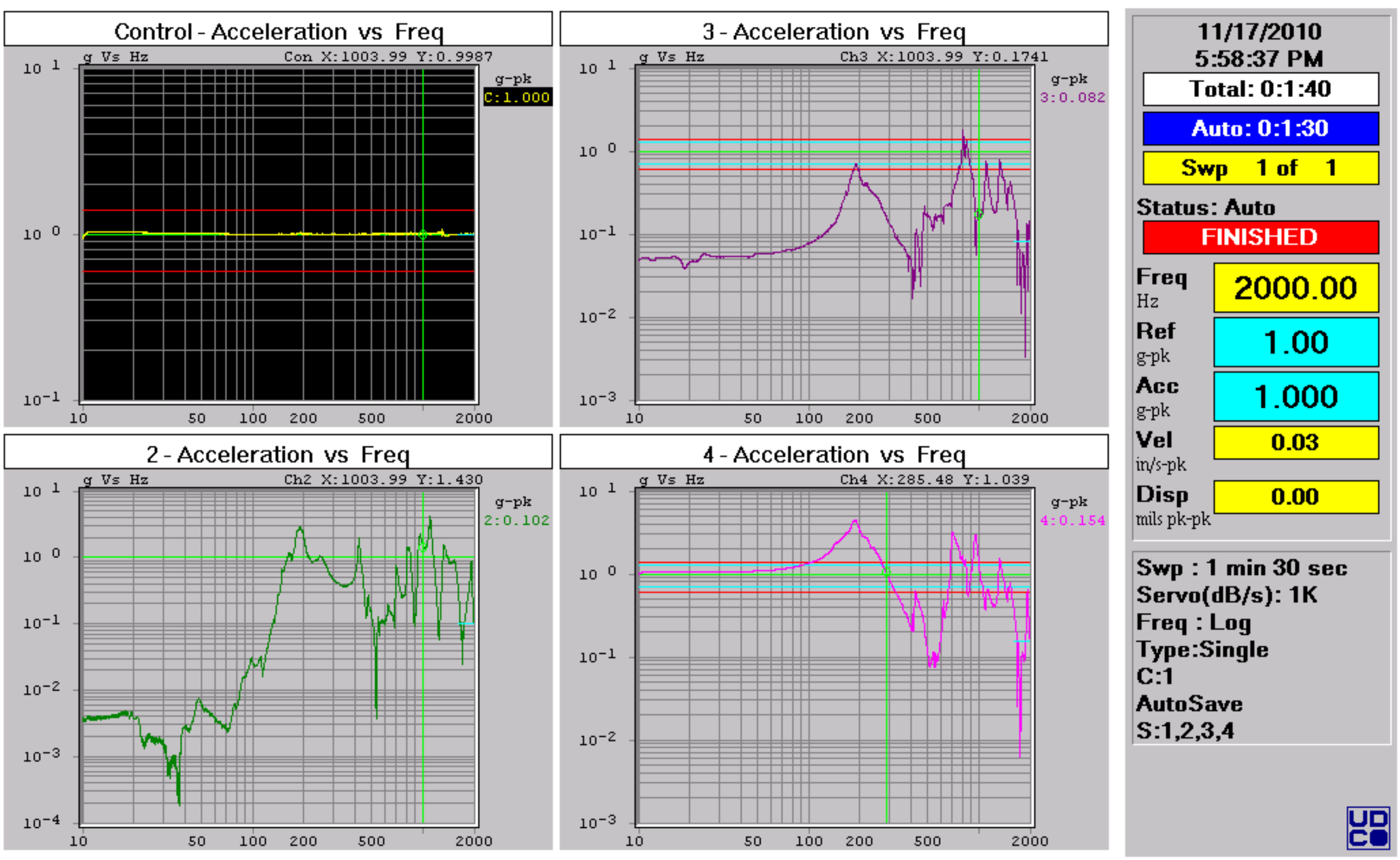

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LOWER FREQUENCY: 10.00 Hz
UPPER FREQUENCY: 2000.00 Hz
SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	**Sensitivity (mV/g)**	**Coupling (AC/DC)**	**Max.Range (g pk)**
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2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

Chan #	**Description**
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Swept Sine Test Report

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RUN NAME: Hypercube Pre Z
USER/PROJECT FOLDER: Class Tests
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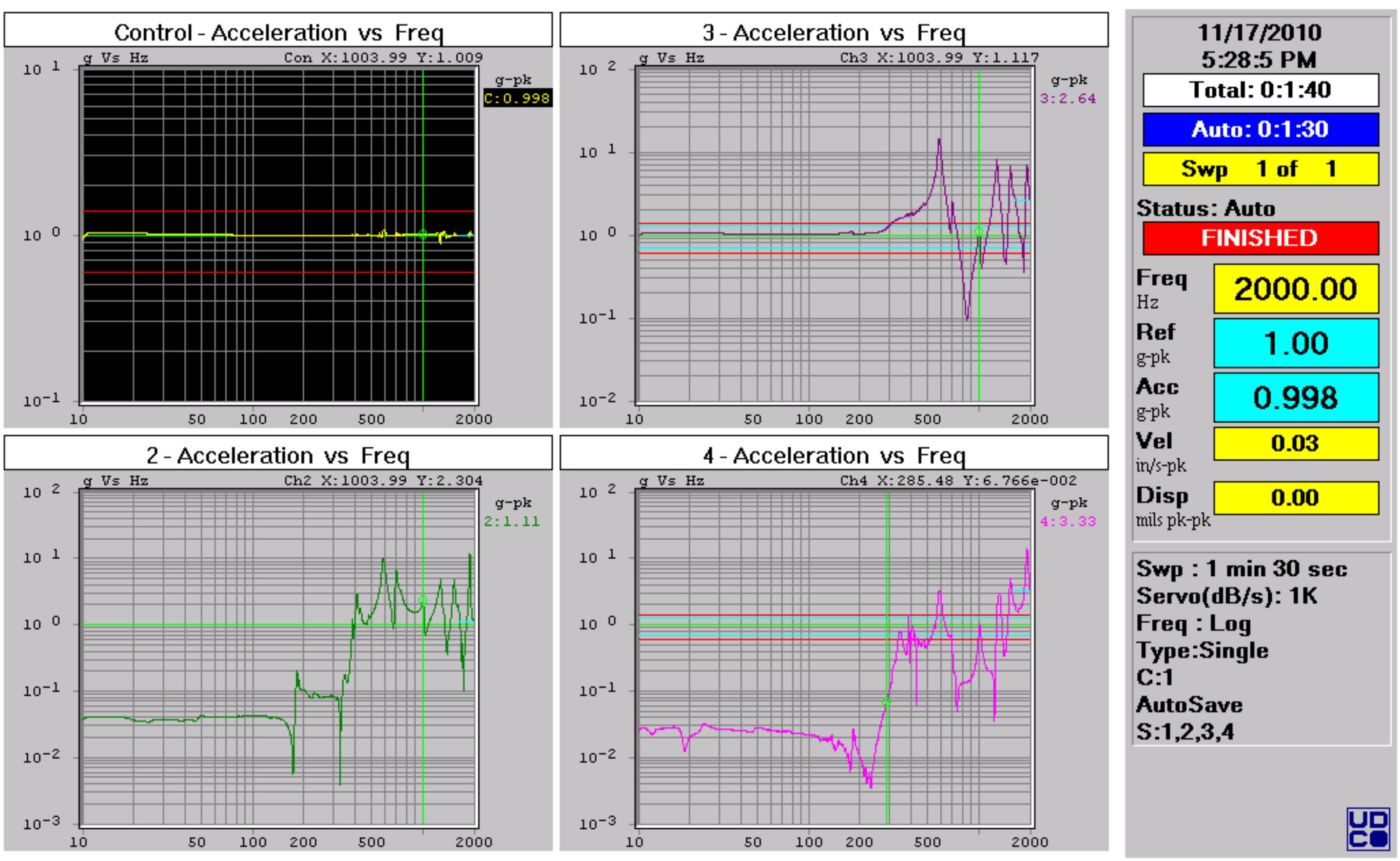
STATUS INFORMATION

TEST EVENT TIME: Wednesday, November 17, 2010 at 05:28:05 PM
TEST STATUS: FINISHED
TEST MODE: AUTO
TOTAL TIME ELAPSED (HH:MM:SS): 0:1:40
AUTO TIME ELAPSED (HH:MM:SS): 0:1:30
SWEEP #: 1
FREQUENCY: 2000.00 Hz
REFERENCE: 1.00 g pk
CONTROL ACCELERATION: 1.00 g pk
CONTROL VELOCITY: 0.03 in/s pk
CONTROL DISPLACEMENT: 0.00 mil pp



CONTROL PARAMETERS

CONTROL CHANNEL(S): 1
CONTROL TYPE: SINGLE
SWEEP TIME: 1 min, 30 sec
SWEEP TYPE: LOG
STARTING SWEEP DIRECTION: UP
STARTING FREQUENCY: 10.00 Hz
LOWER FREQUENCY: 10.00 Hz
UPPER FREQUENCY: 2000.00 Hz
SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	**Sensitivity (mV/g)**	**Coupling (AC/DC)**	**Max.Range (g pk)**
1	9.74	AC	100.00
2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

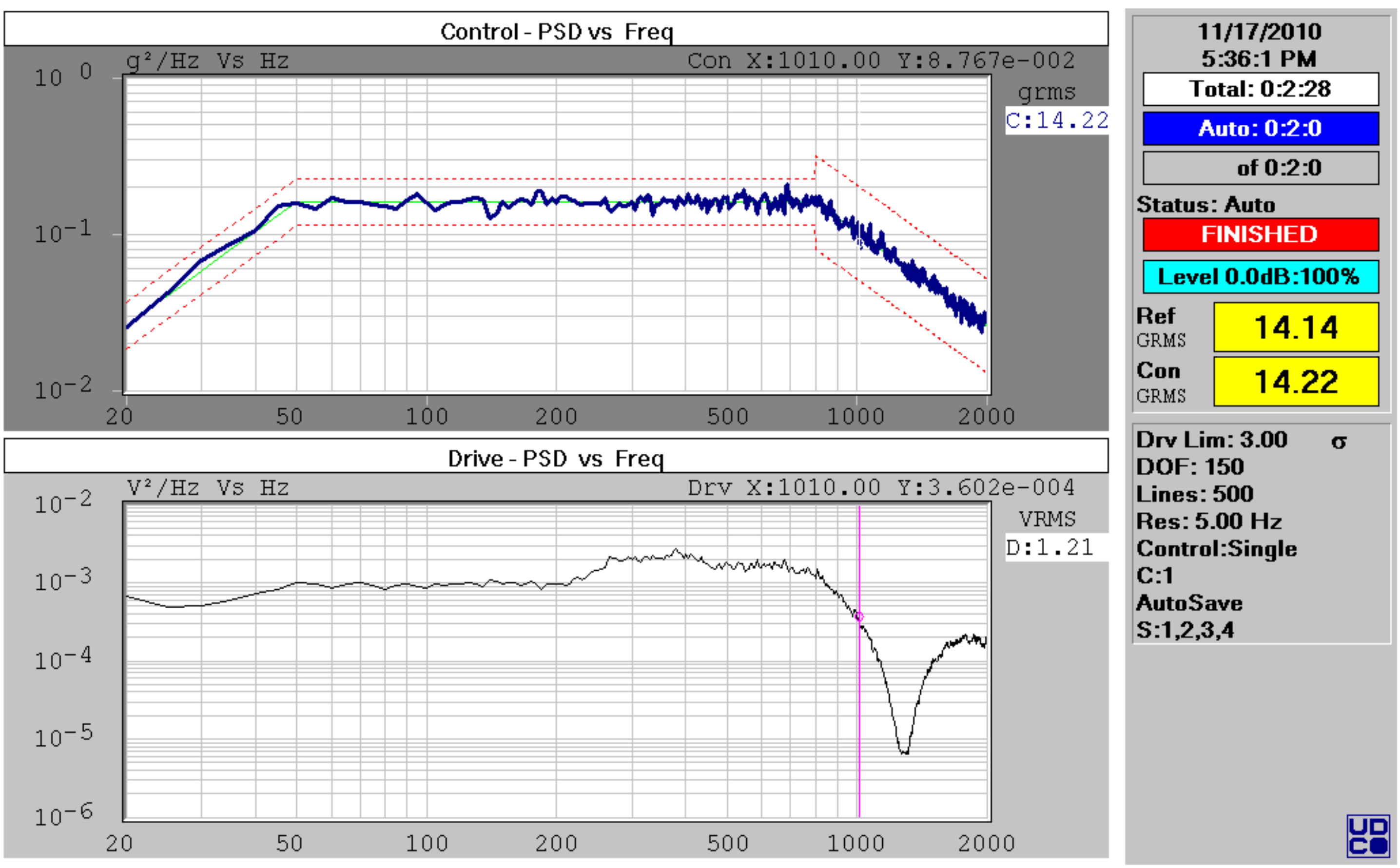
Chan #	**Description**
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Random Test Report

SETUP NAME: NASA GEVS wtih Triaxial
RUN NAME: Hypercube Random Z
USER/PROJECT FOLDER: Class Tests
SAVE NUMBER: 1

STATUS INFORMATION

TEST EVENT TIME: Wednesday, November 17, 2010 at 05:36:01 PM
TEST STATUS: FINISHED
TEST MODE: AUTO
TOTAL TIME ELAPSED (HH:MM:SS): 0:2:28
AUTO TIME ELAPSED (HH:MM:SS): 0:2:0
TEST LEVEL: 0.0 dB
REFERENCE: 14.14 g rms
CONTROL: 14.22 g rms



CONTROL PARAMETERS

CONTROL CHANNEL(S): 1
CONTROL TYPE: SINGLE

FREQUENCY RANGE: 2500 Hz
NUMBER OF PSD LINES: 500
FREQUENCY RESOLUTION: 5.000 Hz
DOF: 150
SIGMA DRIVE LIMITING: 3.00

INPUT CHANNEL PARAMETERS

Chan (#)	Sensitivity (mV/g)	Coupling (AC/DC)	Max.Range (g rms)
1	9.74	AC	50.00
2	103.70	AC	19.00
3	105.20	AC	19.00
4	102.90	AC	19.00

INPUT CHANNEL DESCRIPTION

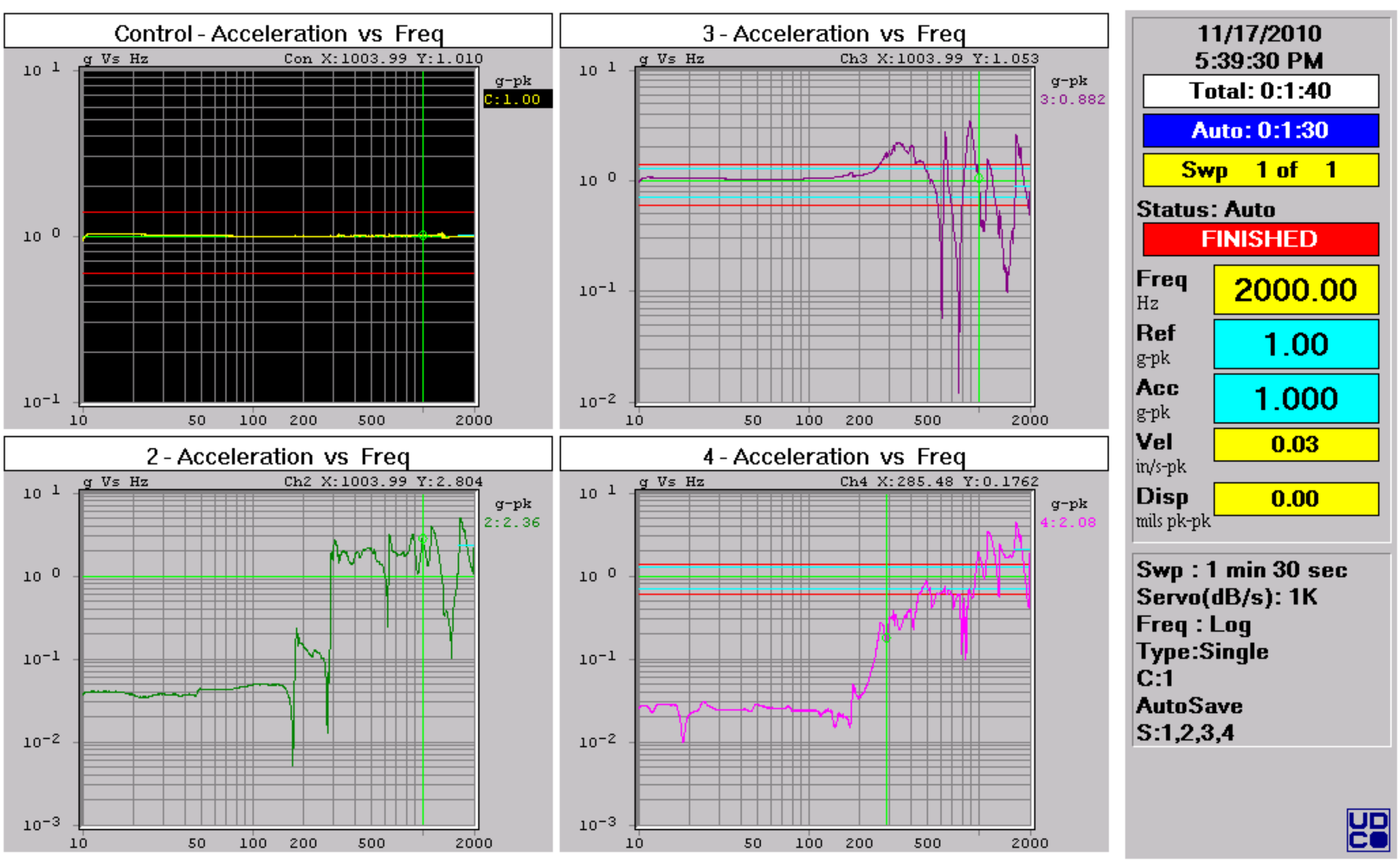
Chan #	Description
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Cal Poly Structures & Composites Lab
Swept Sine Test Report

SETUP NAME: Sine Sweep with Triaxial
RUN NAME: Hypercube Post Z
USER/PROJECT FOLDER: Class Tests
SAVE NUMBER: 1

STATUS INFORMATION

TEST EVENT TIME: Wednesday, November 17, 2010 at 05:39:30 PM
TEST STATUS: FINISHED
TEST MODE: AUTO
TOTAL TIME ELAPSED (HH:MM:SS): 0:1:40
AUTO TIME ELAPSED (HH:MM:SS): 0:1:30
SWEEP #: 1
FREQUENCY: 2000.00 Hz
REFERENCE: 1.00 g pk
CONTROL ACCELERATION: 1.00 g pk
CONTROL VELOCITY: 0.03 in/s pk
CONTROL DISPLACEMENT: 0.00 mil pp



CONTROL PARAMETERS

CONTROL CHANNEL(S): 1
CONTROL TYPE: SINGLE
SWEEP TIME: 1 min, 30 sec
SWEEP TYPE: LOG
STARTING SWEEP DIRECTION: UP
STARTING FREQUENCY: 10.00 Hz
LOWER FREQUENCY: 10.00 Hz
UPPER FREQUENCY: 2000.00 Hz
SERVO SPEED: 1K dB/s

INPUT CHANNEL PARAMETERS

Chan (#)	**Sensitivity (mV/g)**	**Coupling (AC/DC)**	**Max.Range (g pk)**
1	9.74	AC	100.00
2	103.70	AC	80.00
3	105.20	AC	80.00
4	102.90	AC	80.00

INPUT CHANNEL DESCRIPTION

Chan #	**Description**
1	Control
2	Triaxial X
3	Triaxial Y
4	Triaxial Z

Hypercube Assembly Manual

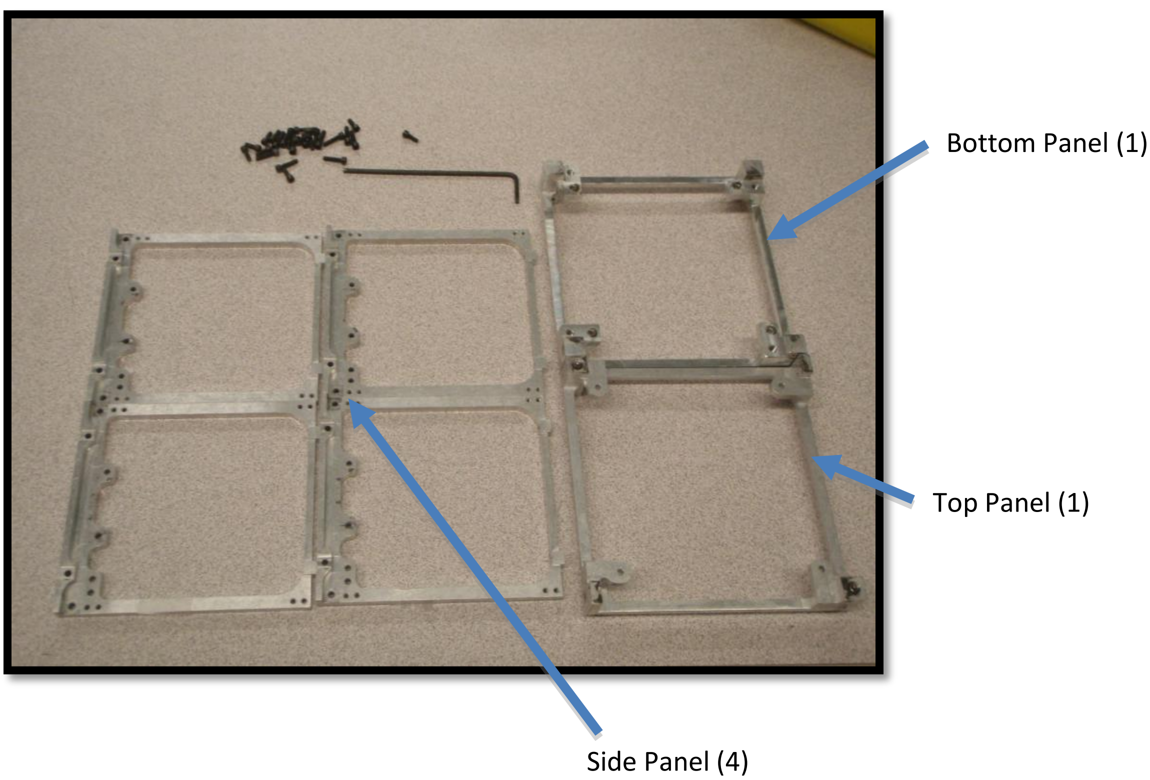
Purpose and Scope of document

This document is intended to aid in the assembly of the Hypercube frame structure. This guide does not consider the integration of solar panels, communications boards, or other payloads. Refer to the documentation provided for the specific payload or module for integration instructions.

Step 1: Verification of Parts

Verify that all parts listed in the parts list below are present.

Quantity	Part Description	Part Number	Supplier	Location Used	Spec
1	Top Panel	HC1101	HyperCube		
1	Bottom Panel	HC1102	HyperCube		
4	Side Panel	HC1103	HyperCube		
8	2-56 x 1/4	92200A077	McMasterCarr	Hat to Panel screws	MIL 16995-2
8	2-56 x 3/8	92200A079	McMasterCarr	Panel to Panel screws	MIL 16995-3

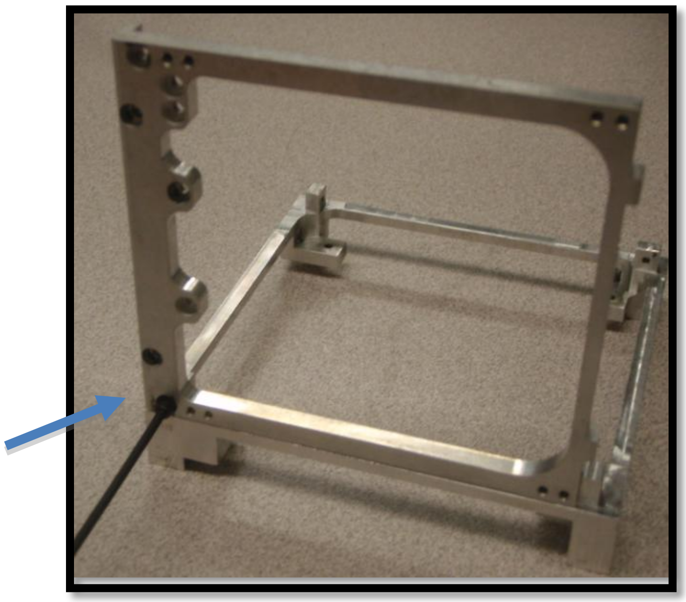


Step 2: Prepare the Bottom Panel for assembly

Place the bottom panel on a clean work surface. Verify that the bottom panel is free of damage and that all threaded holes are unobstructed.

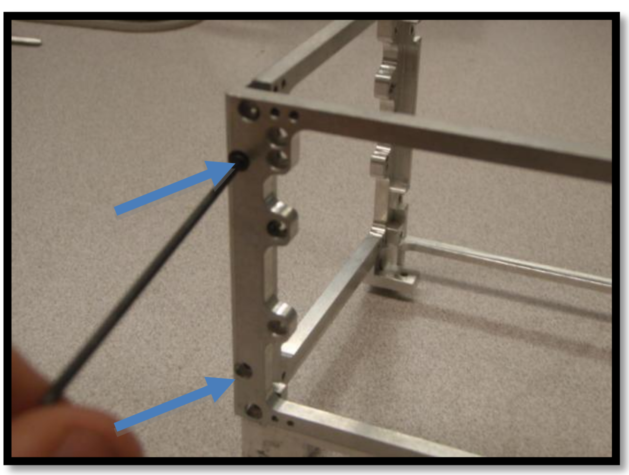
Step 3: Begin Assembly

Begin by placing one side panel on the bottom shoe as shown in the picture below. Prepare one 2-56 x ¼" screw by applying potting compound to its tip. Loosely install one 2-56 x ¼ screw in the location shown below



Step 4: Continue Assembly

Install a second side panel in the manner shown below, repeating the procedure in step 3. Prepare two 2-56 x 3/8" screws for assembly by applying potting compound to their tips. Loosely install these screws in the locations shown below.

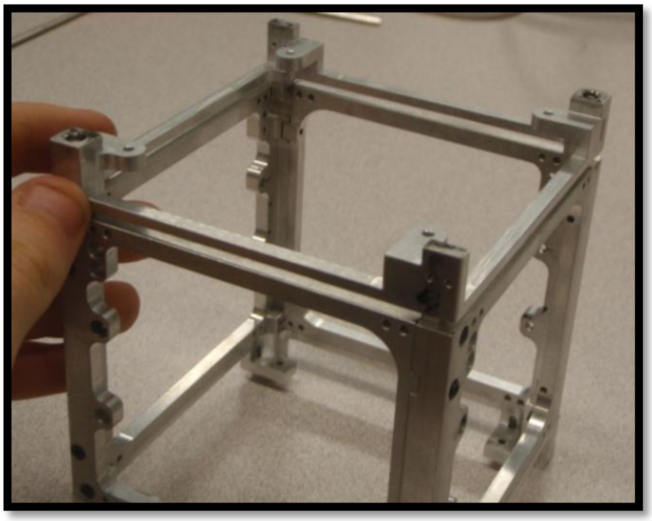


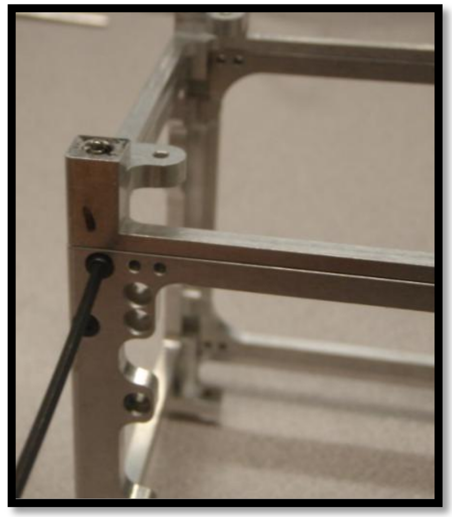
Step 5: Finish Side Panel Assembly

Repeat steps 4 and 5 for the remaining two side panels.

Step 7: Top Panel Installation

Insert the hat into the structure in the manner shown below. Prepare four 2-56 x ¼" screw by applying potting compound to their tips. Install these screws into the locations shown below.





Step 8: Torque Fasteners

Torque all fasteners to 4 inch-lbs.