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Deployment Mechanism for CubeSat Solar Panels

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Bachelor of Mechanical Engineering
Submission date: May 2021
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Preface

This thesis is written as the conclusion of a bachelor’s degree in mechanical engineering at Norwegian university of Science and Technology, NTNU.

We would like to give our special thanks to Anna Olsen our supervisor for advice on the contents of the thesis and for providing contacts in the institute and elsewhere. We would like to thank Nathanael Ferreira Hjermann, our contact at Orbit NTNU, who suggested the thesis.

We would also like to thank family and friends who have been of support during the course of the project.

Due to covid-19 restrictions in Norway during the majority of the project, collaboration and technical assistance from the institute has been very limited. Orbit NTNU has provided us with general information and about CubeSat’s, as well as some experience from their testing, and recommended some literature. We have been to their workshop only once towards the end of the project due to pandemic restrictions.

Abstract

This thesis was written for Orbit NTNU, a student organization designing and building small satellites at the Norwegian University of Science and Technology (NTNU) in Trondheim.

The task was to explore solutions for deploying solar panels on CubeSats. The work presented in this thesis shows a presentation of possible solutions, and the development of an example solution. This development is only in theory, as workshops has not been available to us.

Orbit NTNU's motivation was to explore a system they can use in their next CubeSat missions. The solution will be combined with a yet unknown solar panel and frame. Hence, the proposed system is based on several assumptions, and is a suggested proximate solution, not a specific and final one.

The paper covers the different aspects of designing a deployment mechanism. It presents concepts and ideas to create a reliable system, that can viably be custom made for, or produced in-house by Orbit NTNU. The most promising concept has been further developed and explored. Ideas for alternative solutions to problems have been suggested for Orbit NTNU to explore and perform tests on.

Sammendrag

Denne oppgaven er skrevet for Orbit NTNU, en studentorganisasjon som designer og bygger små satellitter ved Norges teknisk-naturvitenskapelige universitet (NTNU) i Trondheim.

Oppgaven var er å utforske løsninger for utfolding av solcellepaneler på CubeSats. Arbeidet som presenteres i denne oppgaven viser en presentasjon av mulige løsninger, og utviklingen av en eksempelløsning. Denne løsningen er bare i teori, ettersom verksteder ikke har vært tilgjengelig for oss.

Orbit NTNUs motivasjon var å utforske et system de kan bruke i sine neste CubeSat oppdrag. Løsningen vil bli kombinert med et ukjent solcellepanel og ramme. Derfor er det foreslåtte systemet basert på flere antagelser, og er en omtrentlig løsning, ikke en spesifikk og endelig en.

Oppgaven dekker de forskjellige aspektene ved utforming av en utfoldemekanisme. Den presenterer konsepter og ideer for et pålitelig system som formodentlig kan lages for eller produseres internt av Orbit NTNU. Det mest lovende konseptet er videreutviklet og utforsket. Ideer for alternative løsninger på problemer er foreslått for Orbit NTNU å utforske og utføre tester på.

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Terms and Abbreviations

Abbreviation/Term	Meaning
CubeSat	A very small satellite following the CubeSat Design Specification. They consist of one or several units, one unit being 10x10cm
CSDS	CubeSat Design Specification
HRM	Hold and release mechanism. A mechanism that first holds a moving part and later releases it.
CAD	Computer Aided Design

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1 Introduction

1.1 Orbit NTNU

Orbit NTNU is a non-profit student organization whose aim is to design and build Norway's first operational student satellite. The organization consists of a big team of students of interdisciplinary backgrounds and is stationed at the Norwegian University of Science and Technology in Trondheim. Orbit NTNU is backed and sponsored by several big engineering firms and organizations.

The satellite being built today, the SelfieSat, is Orbit NTNUs first project. Their goal is to display the selfie of any person on earth on the satellite while in Orbit. The satellite consists of different subcomponents which is being designed by undergraduates from various engineering fields. The satellite is projected to launch in early 2022 on a SpaceX Falcon 9 launch vehicle.

1.2 Main objective

The thesis was proposed by Orbit NTNU. An excerpt of their proposal goes:

> *"One problem for higher power payloads is the limited surface area of CubeSats as dictated by the CubeSat standard. This creates a maximum power usage based on the solar panel efficiency. One solution to this problem is to attach deployable solar panels that fold out to expose a larger surface area in orbit.*
>
> *Orbit NTNU could need such a possibility in future mission to earth orbit. The thesis would be based around designing a system for deploying 1U solar panels once in orbit, and keeping them securely fastened to the main spacecraft during launch. Such a system has to be very thin (~7mm) and withstand large vibrations and accelerations during launch."*

We note that we later changed this to a more general solution, not specifically for a 1U CubeSat. This was done in a video meeting with our contact person at Orbit NTNU. We also established that our example satellite would be a 2U CubeSat.

Orbit NTNU are in search of a solution for higher power payloads on their satellite, by using deployable solar panels. **Our objective is to go through possible concepts and provide Orbit NTNU with a proposal for the most viable solution.**

1.3 Scope and limitations

Our scope is well defined by our main objective. We are to explore solutions that Orbit NTNU can consider when building their next CubeSat. We will provide an example solution with CAD files, as well as insights from our research in this report.

The scope for a project like this can be extremely wide, but we chose to keep it reasonably narrow. We contemplated something close to a full prototype in the beginning, but realized this was not feasible with our limited time span combined with the circumstances described in the following paragraphs.

Limitations occurred due to the pandemic restrictions of 2021. We had no in person meetings with Orbit in the project start-up phase. Only one meeting towards the end of the project. This also kept us at a distance from the resources they had offered us in the thesis proposal. (Testing equipment and facilities, space related assistance, help in discussions, community/work environment etc.)

Even though we had regular video-meetings with our supervisor from the university, the pandemic still affected our collaboration with our resources at our institute.

The reason our loads chapter elaborates on some load types we have not done further assessment on, is that this chapter was written early in the project, and we never received any technical or academic assistance from the institute on these subjects. We asked for help 42 days before the deadline. This was due to a combination of factors (including: The institute's lecturer on the subject not being available, that we did not realize soon enough that time was running out, and that collaboration with university staff was reduced to purely digital forms.) The meetings and assistance never occurred. Despite ongoing e-mailing between the parts. During the central time of the project (Late March and April.) FEA was a (read: the) major area of focus in the project. However, our system was complex and did not solve for most simulations, and when it did solve, we were unsure about the legitimacy of the results. We spent a lot of time on this, only to abandon it towards the end of the project because we were not able to acquire any assistance from the institute. This directly lessened our time available to go into technical details in some of the subsections found in this paper, and essentially pushed the mid phase of the project into the start if May. We hope the institute will take our experience into consideration for the future.

Due to no access to actual CubeSat hardware and testing equipment/facilities, this thesis is completely based on internet research and a simple 3D printed prototype. It is possible that some of the design challenges theorized, might not be of that great importance in the actual system. This is evoked by said limitations. These limitations have caused us to deal with "theorized challenges" on the design we elaborated on.

Likewise, there were challenges we could not elaborate on, due to uncertainties in what hardware to be used in combination of our system. The scope of the thesis does not include investigating or dimensioning the solar panels themselves, even if they are an essential component in the deployment mechanism.

2 About CubeSats

2.1 Origin of CubeSats

CubeSats are a subclass of nanosatellites. Their design specification is regulated by the “CubeSat Design Specification”. It will be referred to as “the CDS” from here on. The CDS was created to facilitate access to space for university students. Since its inception in 1999 it has been adopted by hundreds of organizations worldwide, including private firms and government organizations. (www.cubesat.org/about.)

2.2 CubeSats and Their Benefits

CubeSats are built up of standardized units of 10x10x10 cm, with a weight of 1kg per unit size. The small size and weight, combined with the standardization, facilitates for relatively inexpensive launch and development costs, and faster development times compared to traditional satellites. (Down to a few years.)

Traditional satellites are very large projects. They take many years to develop and manufacture. A report on US government satellite development timelines states: “The average ATP to launch duration for the programs in this study is 7½ years, ranging from a low of 3½ years to a high of 14½ years.” [1, p. 2]. Costs can range from tens to hundreds of millions of dollars. The large satellites can weigh several hundred kilograms or several tonnes.

For a lot of commercial applications, only a large satellite will do the job. For example, if you need the performance from a bigger antenna, lens, or larger solar panels etc. For applications with low enough demands in performance, of which there are many, a CubeSat will in turn provide much more attainable access to space.

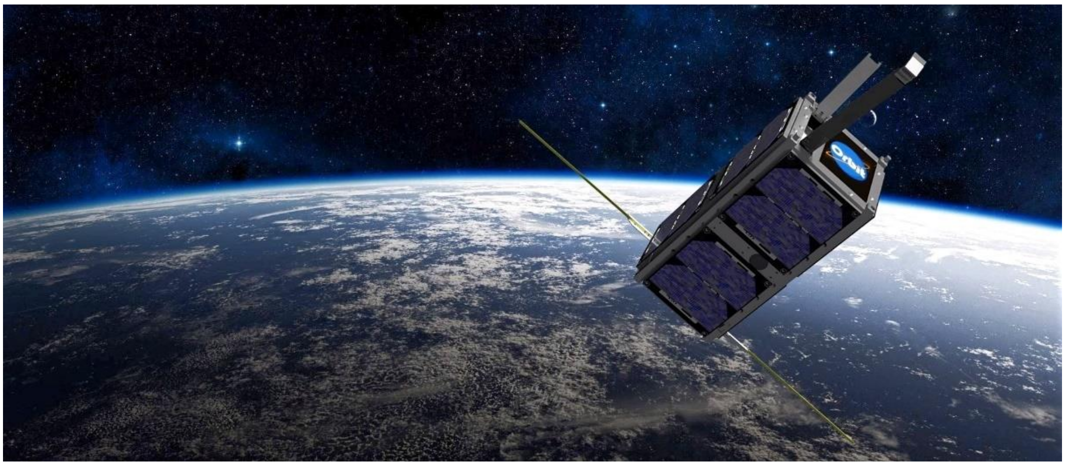


Figure 2-1: Render of a 2U CubeSat in Orbit

2.2.1 Off the shelf alternatives

There are "CubeSat kits" commercially available for purchase. These include all the basic parts you need to start a project, usually except the payloads specific to your mission. Pumpkin Inc., Interorbital Systems and EnduroSat, offer kits from USD $7,500, $11,000 and €25,000, respectively.

These are not necessarily relevant for student CubeSat projects, where the learning outcome of development is of primary importance, as well as cost savings. Academic CubeSat programmes often commit students to develop parts for the CubeSats, as is the case for our thesis.

2.2.2 Use of CubeSats today

The primary use of CubeSats today is for research missions. Usually in Low Earth Orbit (LEO). Although, in recent years, a few CubeSats have been launched into interplanetary space as well. (https://www.space.com/34324-cubesats.html.) CubeSats keeps expanding both in units launched and applications. The first CubeSat launch was in 2003, and in the last decade, CubeSats in orbit has increased dramatically. (https://www.nanosats.eu/)

2.2.3 CubeSat Launch Dispensers

CubeSats will usually be launched on excess capacity together with one or several larger "main payloads" (Figure 2-3), or "piggybacking" on the last stage of the rocket (Figure 2-4 and Figure 2-5). They can also be launched in bulk with other CubeSats (Figure 2-6).

Once the space vehicle is in orbit, the CubeSats themselves are released from dispensers, such as "the P-POD" (See Figure 2-2) and NASA's "NLAS". This is how their fitment onto a space vehicle is standardized.

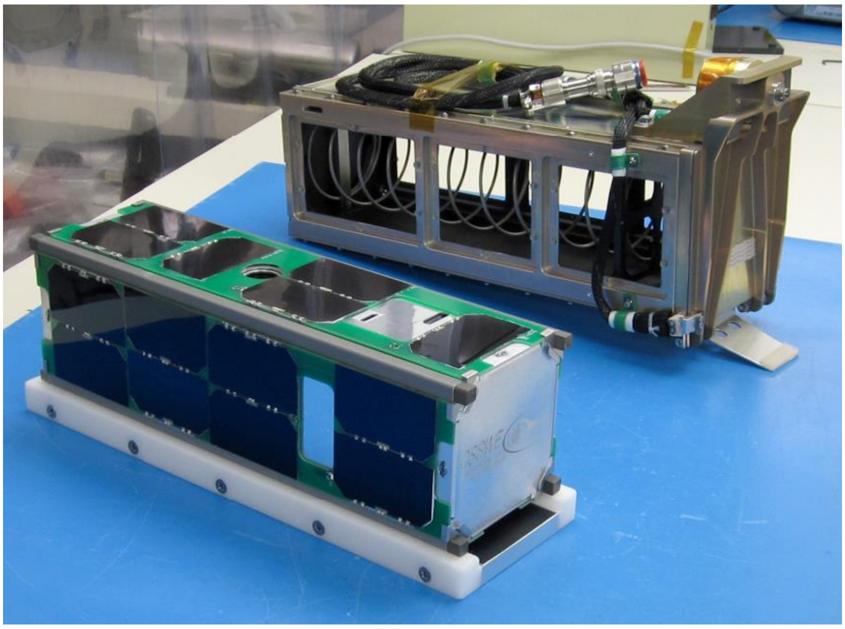


Figure 2-2: A 3U CubeSat and P-POD (background) prior to integration [2]

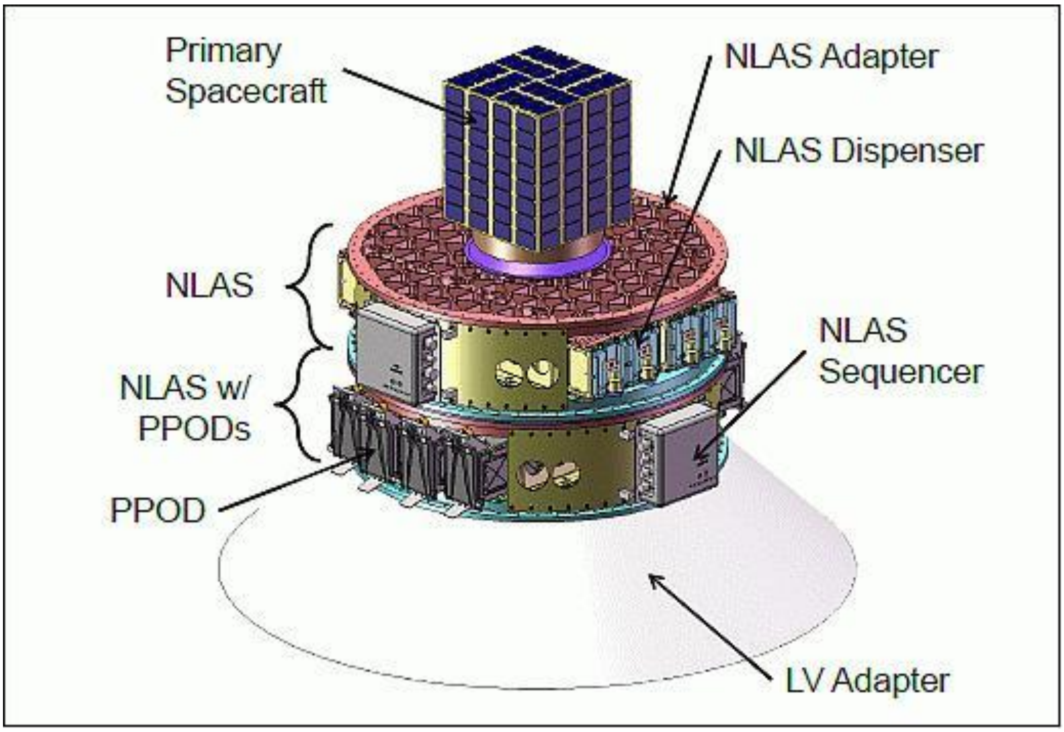


Figure 2-3: NASA's "Nanosatellite Launch Adapter System", "NLAS" [3]

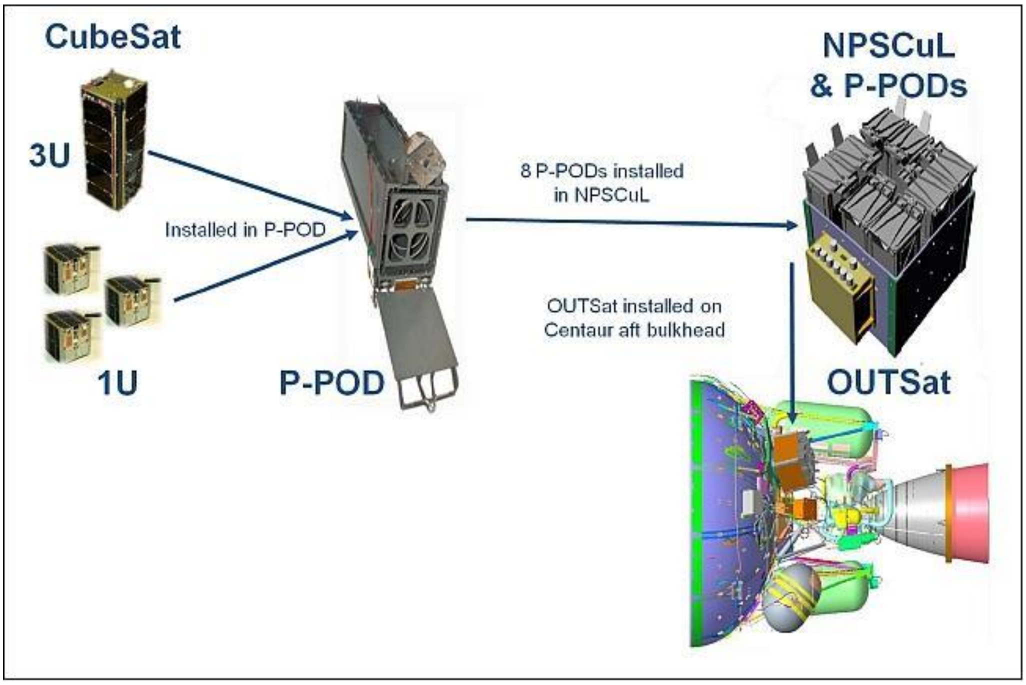


Figure 2-4: CubeSat dispensers "piggybacking" on the final stage of a rocket. [3]

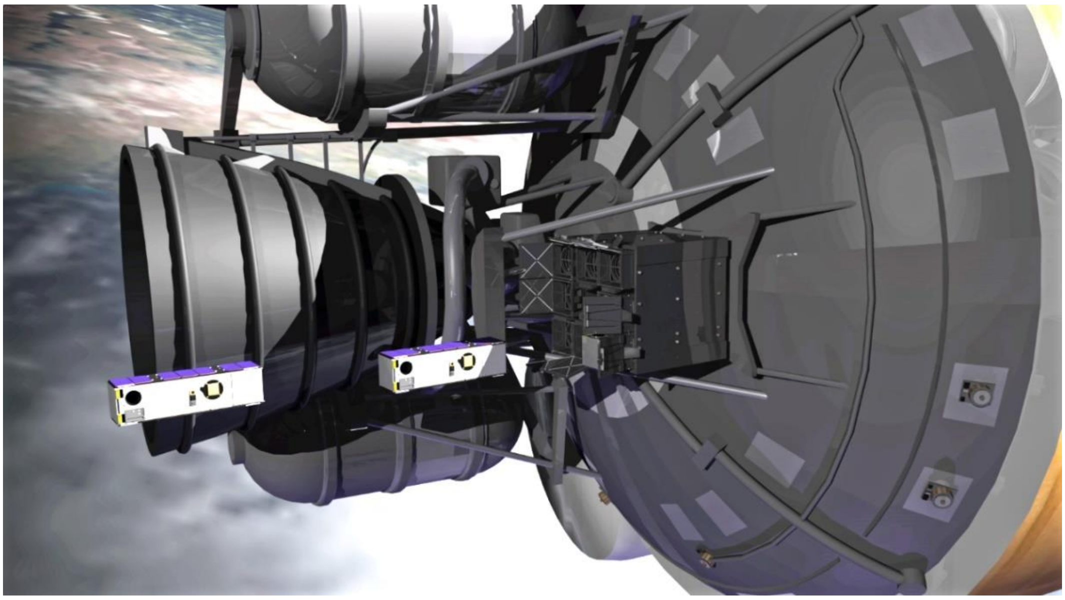


Figure 2-5: CubeSat dispensers "piggybacking" on the final stage of a rocket. [4]

Figure 2-6: SSO-A Launch depiction with SpaceX Falcon-9 stage 2. A rideshare mission with 15 microsatellite and 49 CubeSats. (image credit: Spaceflight) [4]

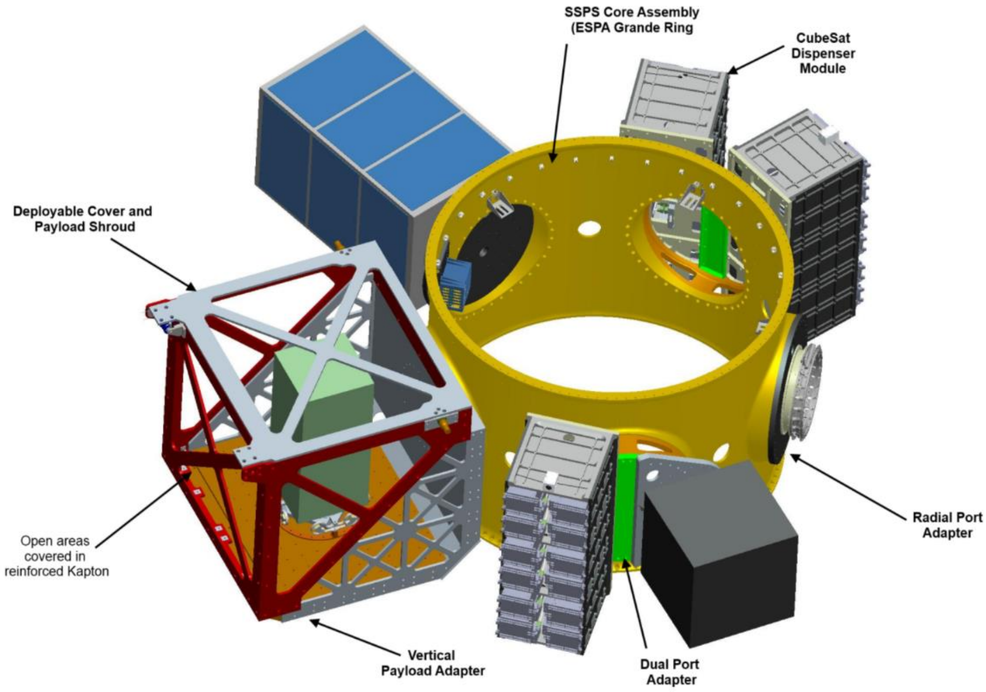


Figure 2-7: Secondary payload adapter for by "Spaceflight Inc". [5]

2.3 The coordinate system

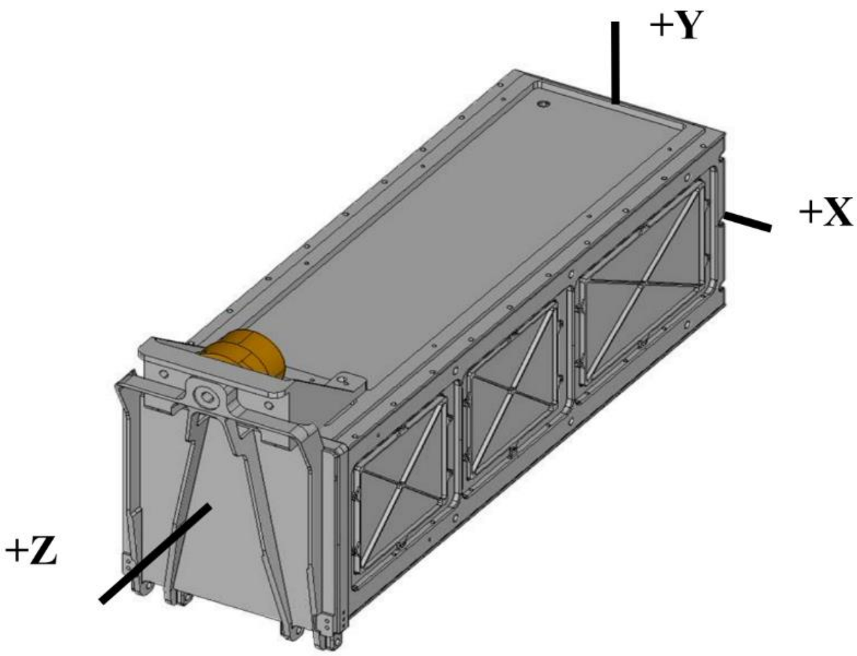


Figure 2-1: CubeSat Coordinate System

In this entire thesis, we will refer a lot to coordinates and directions. The coordinate system we will be using points to the different faces of the CubeSat. The face that goes out of the dispenser first is the Z+ face. The «horizontal» face of the P-POD is the +Y direction and the right face is the +X direction. The faces on the opposite side of the faces listed above are their minus faces, that is -X,-Y and -Z

We will treat this as Z being up, Y being in/out of the plane and X being horizontal.

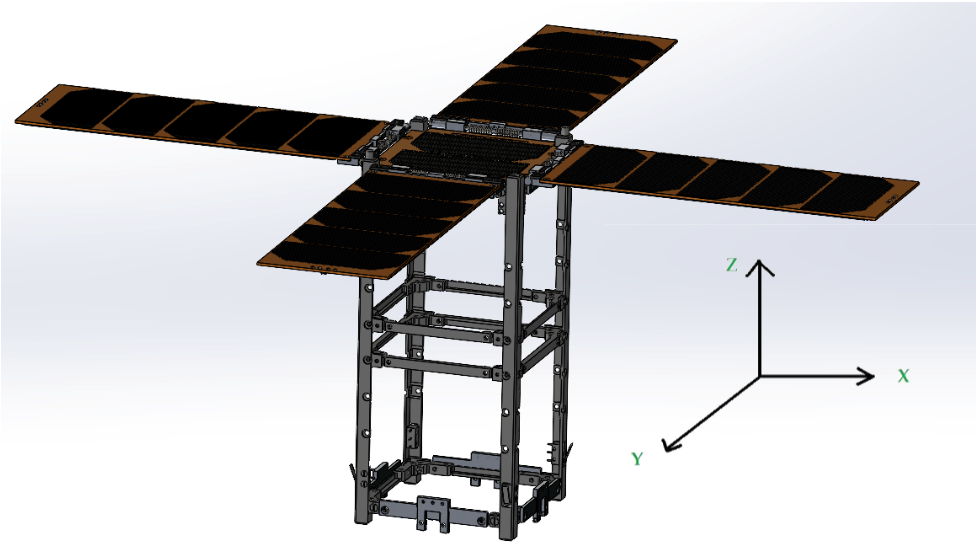


Figure 2-8: Coordinate system

3 Methods

During this project, we have exercised several engineering, product development and project management methods. This section briefly describes them.

3.1 Product development methods

3.1.1 The engineering design process

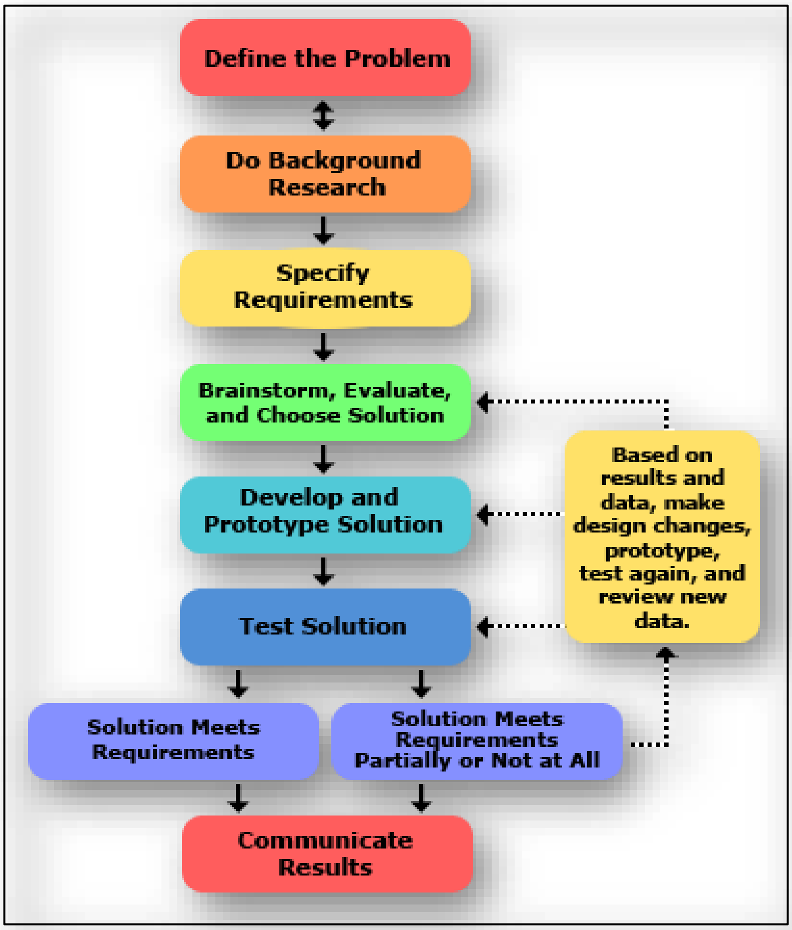


Figure 3-1 : The Engineering Design Process [6]

Figure 3-1 shows the standard engineering design process we used to solve this task. During this project, a lot of research was carried out to develop a better understanding of the given problem. Based on the specifications provided and literature study done, we tried coming up with possible solution for the system. After we had successfully come up with a promising idea, we sought out methods to further develop the idea. At last came the testing of ideas where simulations where run to determine the efficiency of the developed concept. The whole engineering process was an iterative process, which meant that all the stages were repeated continuously until we came up with a good solution. This ensured for a comprehensive paper with recurring ideas and themes throughout the project.

3.1.2 Dialogue with client

We know that a well-developed and well understood specification of the clients' requirements and wishes for the solution, are of utmost importance. Due to the ongoing pandemic and resulting restrictions, contact has been almost exclusively digital. We have had one in person meeting in Orbit NTNU during the project. First, communications were in e-mail form, and later, we arranged a group in Microsoft Teams, where we asked questions in chat form and had video meetings.

3.1.3 Product Development Journal

Internally in the group we decided to use the method called product development journal.

Product development work is documented in a book called PU journal. The journal consists of individual sheets where each sheet is identified by date, author, etc. The format is usually A3 sheets. They have one area for text and one for graphic expressions (see Figure 4.1). The text area constitutes approx. 1/3 of the sheet and the graphic 2/3. The construction drawings are often made in a larger format. Almost all communication takes place through sketches and is applied to text. The PU journal can therefore be in paper format and digital. Everything that is produced during the process is sorted in the PU journal. It will create a story line in the work, where there are opportunities to see what has been analysed, assessed, and rejected. In this way, a history is created in the development work, where the work along the way and alternative solutions are documented. [7]

Instead of A3 paper, we used A3 paper size in Word, and notes in OneNote of unlimited size with drawings made on stylus tablets.

3.1.4 Evaluation Matrix

To choose the best solution, a systematic method will be used to help make the decision. For this step, we have decided to make use of an evaluation matrix. The evaluation matrix allows to score different ideas against each other, rating them based on a set of defined criteria. The concept with the highest score will be chosen and developed.

3.2 Engineering methods

3.2.1 Software Tools

Computer Aided Design (CAD)

After we had developed an idea for what our solution would be, the tool for forming the geometry and layout of the parts was a CAD software. The software available for us through our university was SOLIDWORKS 2020.

CAD software is an aid in the creation, modification, analysis, or optimization of a design. CAD software is used to increase the productivity of the designer, improve the quality of design, improve communications through documentation, and to create a database for manufacturing.

Finite Element Analysis (FEA)

The Finite Element Method usually takes a geometric model of a part or system and splits it down into a large but finite number of elements. Once this and the boundary conditions and loads onto the part is set up in the software, it can use numerical methods to calculate approximate solutions to things like stresses and displacements between the elements in the part.

In this way we can use computational simulations to predict whether our design will perform optimally, and if it does not, we can quickly use CAD to change the design and test again.

In addition to static analysis, FEA can be used for dynamic analysis (vibration loads), buckling analysis, modal analysis (finding the natural frequencies of a model), fluid mechanics, heat transfer and electromagnetic problems.

Digital Collaboration Tools

Due to varying government and university imposed “Corona-Virus restrictions”, video-meetings and other digital communications has replaced in person meetings to a great extent during this project.

Digital tools were put into use, to make collaboration as effective as possible in periods of “restrictions” and/or campus lockdown. These include video-meetings, messaging and e-mails with our supervisor, Orbit NTNU, and between the project participants.

Among the tools used were: Zoom and MS Teams for video-meetings and messages, The Microsoft Office Suite, OneDrive, and OneNote for collaborating on documents and notes.

4 Requirements and specification

4.1 The CubeSat Design Specification

4.1.1 The Poly Picosatellite Orbital Deployer (P-POD)

Cal Poly's standardized deployment system is shown in the figures below. It can house 3 CubeSat units. Other deployers can fit up to 27-unit CubeSats. The P-POD is the standard deployer, and the one we will design for.

Once the launch vehicle is in orbit, the P-POD will open, and the CubeSat(s) pushed out along a series of rails.

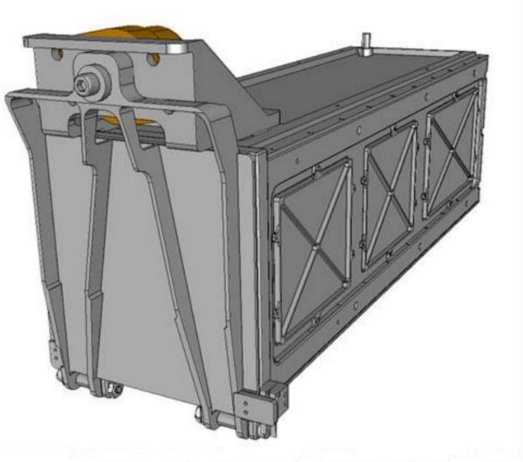


Figure 4-1: Poly Picosatellite Orbital Deployer (CubeSat Design Specification Rev. 13)

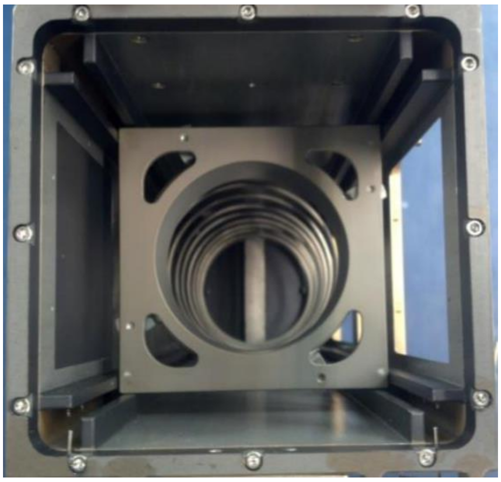


Figure 4-2: P-POD Cross Section (Kilde)

The cross section of the P-POD shows that we have available space for our solar panels, outside the 10cm by 10cm of the CubeSat itself.

4.1.2 Mechanical Requirements

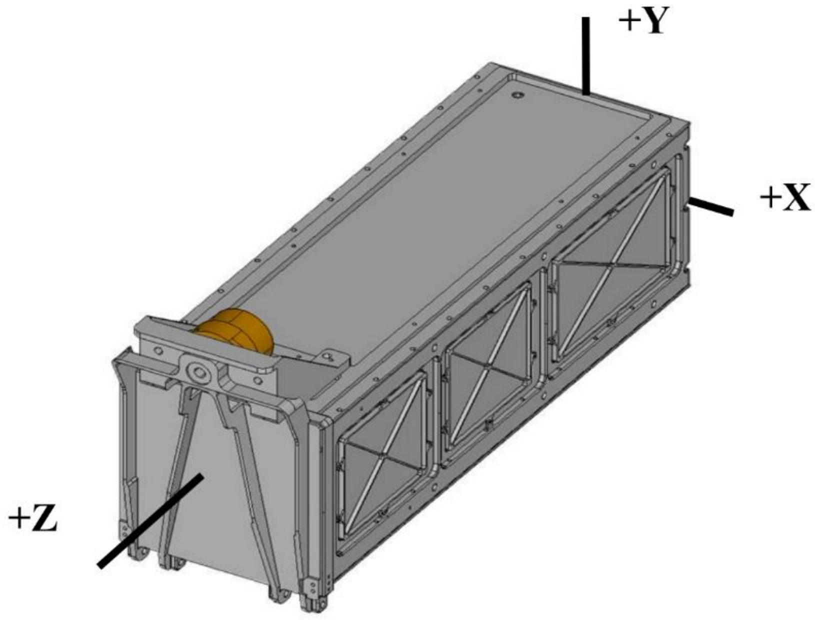


Figure 4-3: P-POD Coordinate System [8]

Figure 4-3 is referenced in the CDS and is also what we will use as the coordinate system for the CubeSat itself throughout this document.

We quote from the CDS, the mechanical requirements relevant to our project:

> *"3.2.1The CubeSat shall use the coordinate system as defined in Appendix B for the appropriate size. The CubeSat coordinate system will match the P-POD coordinate system while integrated into the P-POD. The origin of the CubeSat coordinate system is located at the geometric centre of the CubeSat."*

> *"3.2.3 No components on the green and yellow shaded sides shall exceed 6.5 mm normal to the surface."*

> *"3.2.3.1 When completing a CubeSat Acceptance Checklist (CAC), protrusions will be measured from the plane of the rails."*

> *"3.2.16 The CubeSat rails and standoff, which contact the P-POD rails and adjacent CubeSat standoffs, shall be hard anodized aluminium to prevent any cold welding within the PPOD."*

The main takeaway is the 6.5mm of space allowed on each face of the CubeSat.

4.1.3 General Requirements

We quote from the CDS, the general requirements relevant to our project:

> *"3.1.2 All parts shall remain attached to the CubeSats during launch, ejection, and operation. No additional space debris will be created."*

> *"3.1.8 CubeSat materials shall satisfy the following low out-gassing criterion to prevent contamination of other spacecraft during integration, testing, and launch. A list of NASAs approved low out-gassing materials can be found at: http://outgassing.nasa.gov."*

> *"3.1.8.1 CubeSats materials shall have a Total Mass Loss (TML) < 1.0 %"*

> *"3.1.8.2 CubeSat materials shall have a Collected Volatile Condensable Material (CVCM) < 0.1%"*

> *"3.1.9 The latest revision of the CubeSat Design Specification will be the official version which all CubeSat developers will adhere to. The latest revision is available at http://www.cubesat.org."*

We note that parts need to be safely fastened including after deployment, and that only low outgassing approved materials will be used.

4.1.4 Operational requirements

We quote from the CDS, the operational requirements relevant to our project:

> *"3.4.4 All deployables such as booms, antennas, and solar panels shall wait to deploy a minimum of 30 minutes after the CubeSat's deployment switch(es) are activated from PPOD ejection."*

From this, we know that our system must be triggered by a reliable and precise timer. In practice this will be an electric signal from the CubeSat control circuit board.

4.1.5 Testing requirements

We will not be doing testing, as that comes at a later stage in the full CubeSat development, and we are exploring solutions for the deployment of solar panels. However, we will design with regards to the loads indicated by the testing requirements.

The tests mentioned are random vibration testing, a thermal vacuum bakeout (to ensure proper outgassing of components) and shock testing.

We conclude that random vibration and shock is the most important mechanical load factors but will review other loads as well in chapter 5.

4.2 Orbit NTNU Requirements

A client's description and specification of a solution or product is of utmost importance. It needs to be well defined and understood if the solution is to achieve its purpose. In this case, the task from Orbit NTNU is quite open. We do not have a precise description of a solution, but more of a broad mission. This leaves things opens for us.

From their initial thesis proposal, and our discussion with them, we have the following points:

1. Make a system that increases the effective solar panel surface area in orbit. In other words, increase the solar panel area normal to the Sun.
2. Use minimal space inside the CubeSat frame. If possible, only use the space between the frame and the P-POD.
3. Assume a custom solar panel can be acquired. Assume a PCB panel.
4. Assume the CubeSat will orient itself towards the sun for charging. No need to take the orbit and attitude of the satellite into consideration.
5. Orbit NTNU wants a proposed mechanism including CAD files. They however do not expect a functioning prototype to be made. This is partly due to limitations in facilities and resources due to Corona-virus restrictions within the first 5 months of 2021.
6. We can use the CubeSat frame Orbit NTNU used on their last mission as a base for our solution, even though the solution is intended for use in later missions.

4.3 Adjacent Hardware's Specifications

The available off the shelf CubeSat parts, as well as the CubeSat frame, dictates the specification of our system. Here we will discuss the typical specifications of such parts, and how they will affect our design opportunities.

As mentioned earlier, the CDS gives us 6.5mm from the frame. Some frames can give us additional space. The frame we modelled the system on gave us 7.3mm, but we will use 6.5mm as our thickness nonetheless, as Orbit NTNU's next frame is unknown. A proposed in-house frame is being worked on in a similar thesis to ours.

4.3.1 Solar Panels

Our design needs to be able to accommodate different solar panels, as Orbit NTNU has not provided us with an exact panel. For the sake of this project, we were told to assume a custom panel could be ordered, and that it will be PCB (Printed Circuit Board) based. We will therefore use measurements from existing commercial off the shelf products.

The typical solar panel consists of heat sensors, sun sensors, a magnetorquer and solar cells all mounted on a PCB.

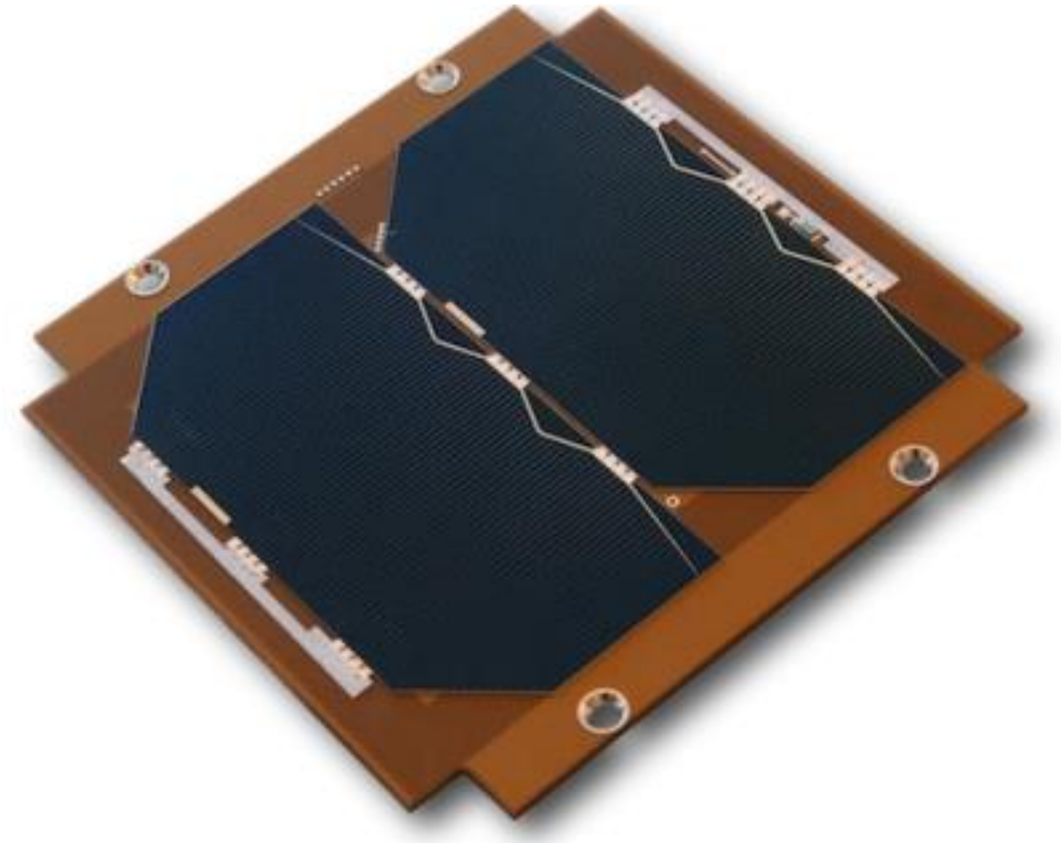


Figure 4-4: GOM space solar panel

The most common material used for the PCB is FR-4. FR-4 is a composite composed of woven fiberglass cloth with an epoxy resin binder that is flame resistant. The material has great mechanical properties with a relative high yield strength of 345 MPa and a melting point of 140 degree Celsius. [9]

Additional stiffeners can be added to the solar panels to minimize panel dynamic deflection under launch vibration loads. For instance, ISISpace has produced deployable solar panels for 6U CubeSat application, where a thin PCB made up of FR4 material of 0.18 mm thickness is stiffened by an aluminium panel. Park et al. developed an FR4 PCB-based deployable solar panel, which was stiffened by using stiffeners made up of G10 high-pressure fiberglass laminate composite material. [10]

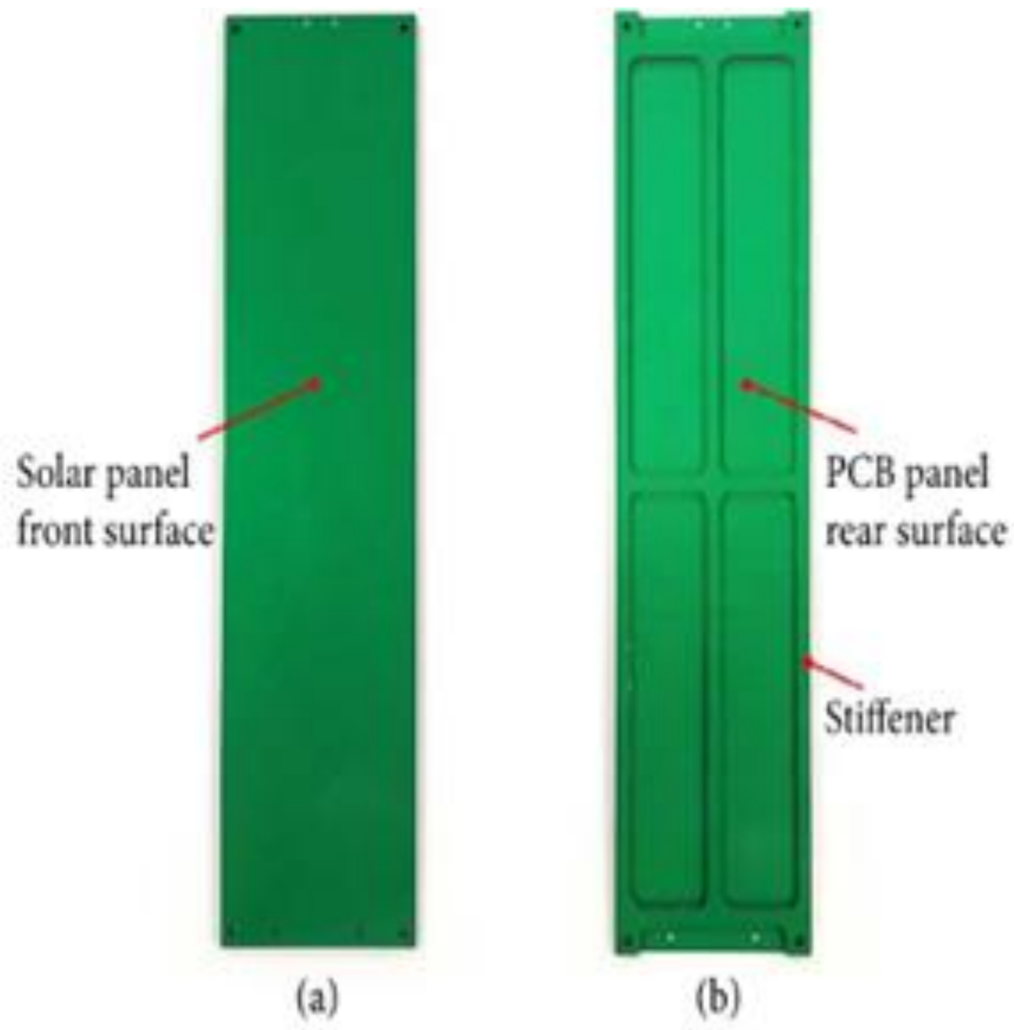


Figure 4-5- Solar panel with stiffener [24]

The solar panels found had varying masses based on what components they consisted of. For reference, a 6U solar panel developed by Park et al. with stiffeners had a mass of 560g.

PCB panels with solar cells	450g
Stiffeners x 3	110g

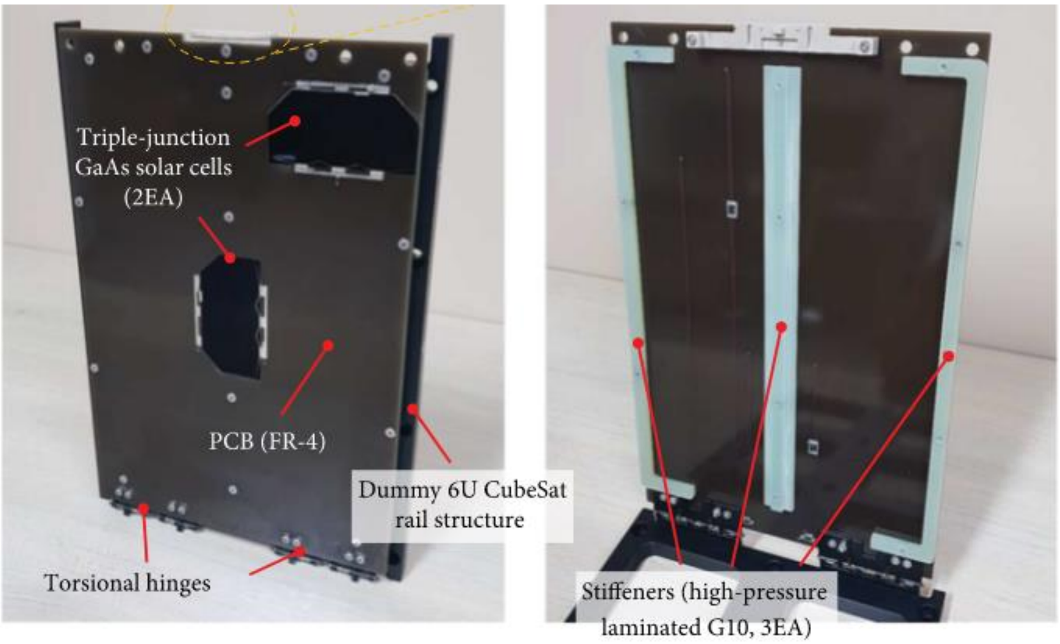


Figure 4-6 Solar panel with high pressure laminated G10 stiffeners.

ISISpace have solar panels with an aluminium substrate with a flex-PCB overlay for the solar cells. Their mass with regards to unit as shown on their shop website are as follows:

Units	Mass
1U	50g
2U	100g
3U	150g
6U	300g

Table 4-1: Panel masses

The minimum and maximum panel thickness we have seen are 1.11mm and 2.3mm.

5 Loads and environment.

In this section we discuss the loads our system shall withstand. These can be split into three main categories. Launch loads, Low Earth Orbit loads, and loads from the motion of the mechanism itself.

During launch and in low earth orbit a CubeSat will experience various static and dynamic loads. The CDS demands testing for two (mechanical) load types. These are random vibration and shock. These tests "shall be tested as defined by the launch provider" [8]. We do not know what launch providers Orbit NTNU may use in the future. Furthermore, their current launch provider has classified the specifications for their tests, hence they cannot be presented in this public thesis.

In this thesis, we will use launch loads described in "The SpaceX Falcon User's Guide". A Falcon 9 rocket is a likely launch vehicle in the coming years (from 2021), and the one Orbit NTNU's launch provider has booked for their current mission. The other most likely launch vehicle would be a rocket from the Russian "Soyuz" rocket family. There are various newcomers to the private space industry that might also become likely candidates in the near future.

Orbit NTNU has access to test facilities through some of their partners, but these tests will be performed on a fully assembled CubeSat. For our solar panel deployment mechanism, and the scope of this thesis, we will settle with considerations of these loads. (Reasons for this are explained later.)

A quick summary of the loads considered:

Launch loads:

- Acceleration
- Random Vibration
- Shock

Low Earth Orbit loads and environment:

- Thermal expansion
- Vacuum. (Danger of cold welding and low outgassing requirements.)

5.1 Loads during launch.

Launch loads depend on the launch vehicle. As declared above, we will use the Falcon rockets from SpaceX as an example in this thesis. (See Figure 5-1: SpaceX Falcon 9 and Falcon Heavy launch vehicles). The launch loads are described in the "SpaceX Falcon User's Guide" [11] (Section 4.3, page 14 to 34.) Applicable loads for our case will be mentioned in the following sub-sections.

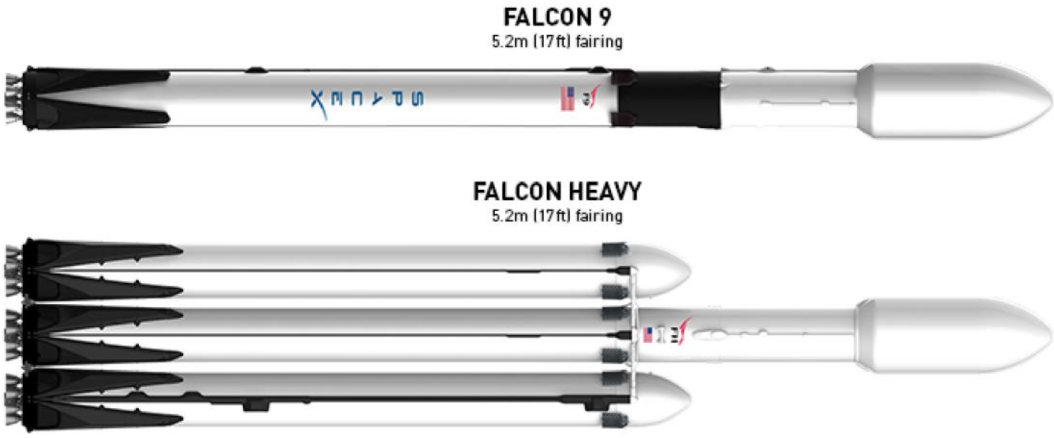


Figure 5-1: SpaceX Falcon 9 and Falcon Heavy launch vehicles

We quote the following remarks from SpaceX regarding the load factors given in the Falcon User's Guide:

> *"The load factors provided below are intended for a single payload mission; multi-payload missions should coordinate directly with SpaceX." [11, p. 15]*

> *"Actual spacecraft loads, accelerations and deflections are a function of both the launch vehicle and payload structural dynamic properties and can only be accurately determined via a coupled loads analysis." [11, p. 15]*

Yet, we assume a similar load factor to a single payload scenario for the CubeSat. This would be close to if the CubeSat is in the P-POD inside the launch vehicle fairing together with a single payload (Figure 2-3 and Figure 2-6). We have found no information on the other two scenarios where the P-PODS are on the outside of the second stage (Figure 2-4 and Figure 2-5). (The second stage separates from the first stage and main engine after the vehicle leaves Earth's atmosphere.) For this thesis, we assume the loads are similar enough.

5.1.1 Acceleration

For the quasi-static acceleration during the launch, we have data from the Falcon Users Guide, as well as from Orbit NTNU. These are quite different, but we will mention both here.

For the "Falcon User's Guide", the loads depend on the total payload mass onboard the rocket. We will assume the lowest total payload mass scenario, giving the highest acceleration. This is the load factors described by the red line in Figure 5-2.

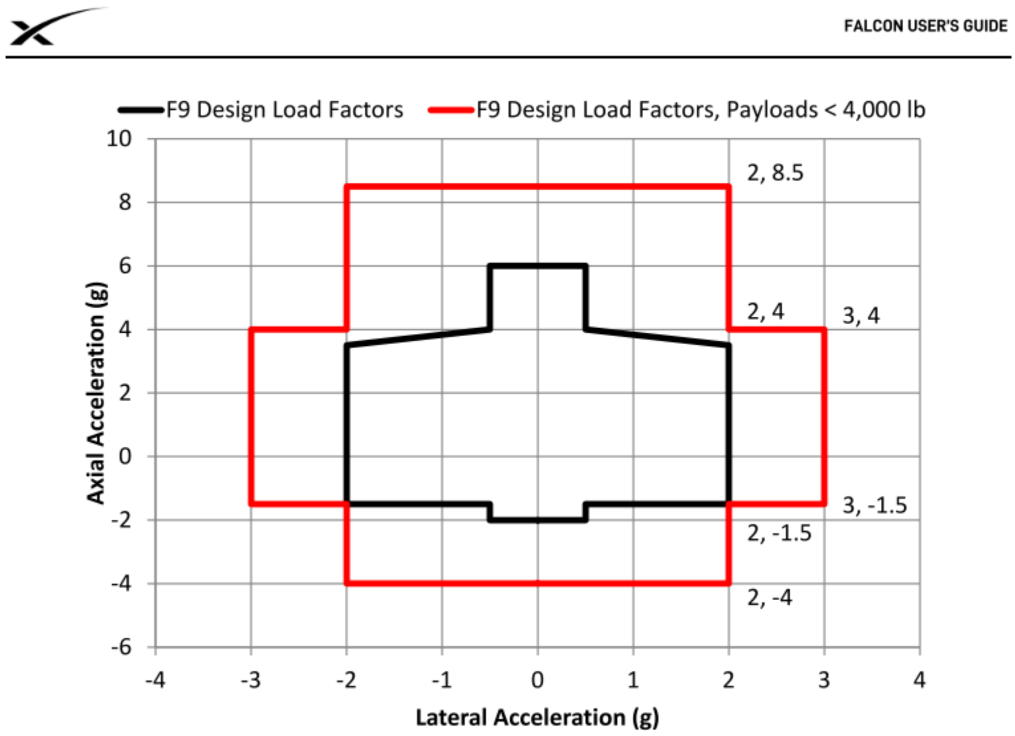


Figure 5-2: Falcon 9 payload design factors

On their current mission Orbit NTNU uses a much higher acceleration figure from their launch provider. Although Orbits NTNU's launch providers load specifications are classified, we could quote them on the quasi-static acceleration. They are using 15g's of acceleration. Onto that they choose to add a safety factor of 1.25. Thus, the dimensioning acceleration load is 18.75g. More than double that of the Falcon User's Guide.

We will dimension our system against the 18.75 g's, but the difference here shows that we will have a large safety factor if the Falcon User's Guide scenario is closer to the actual load.

5.1.2 Vibration

The rockets engine, and the aerodynamic excitation of it during flight generate vibrations. The frequency and amplitude of these vibrations is of a random nature.

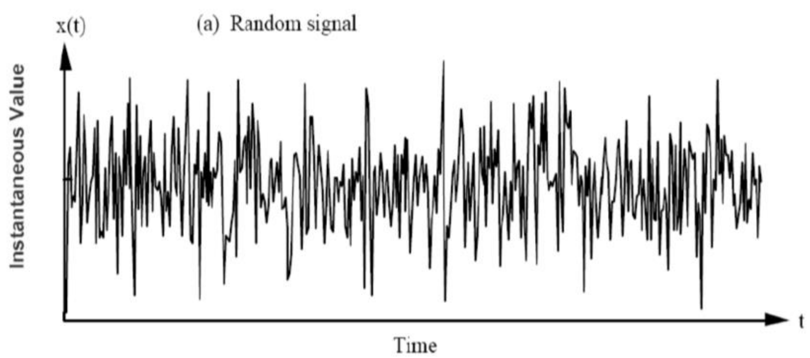


Figure 5-3: Typical Random Vibration REFREF

Figure 5-3 shows a typical random vibration signal where the magnitude of the signal as a function of the time is nondeterministic. Trying to simulate an actual random vibration curve like this for duration of the launch in an FEA simulation would be impractical. The FEA approach to random vibration is using power spectral density curves which give the relationship between the frequency and amplitude of the vibration load.

Quoting from the Falcon Users Guide:

> *"Since SpaceX accommodates a variety of payloads, results of coupled loads analysis will be used to modify these levels, if necessary, to reflect the levels at the payload interface."*

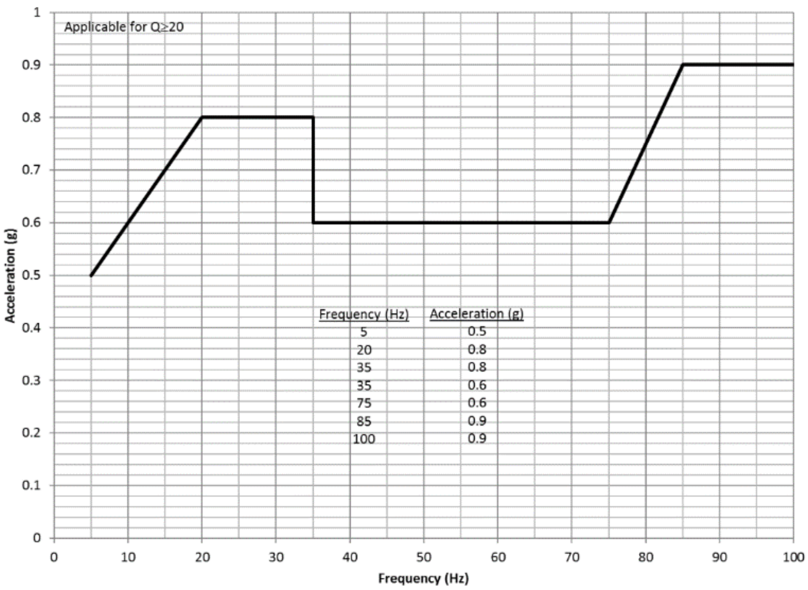


Figure 5-4: Maximum axial equivalent sine environment for Falcon 9 and Falcon Heavy

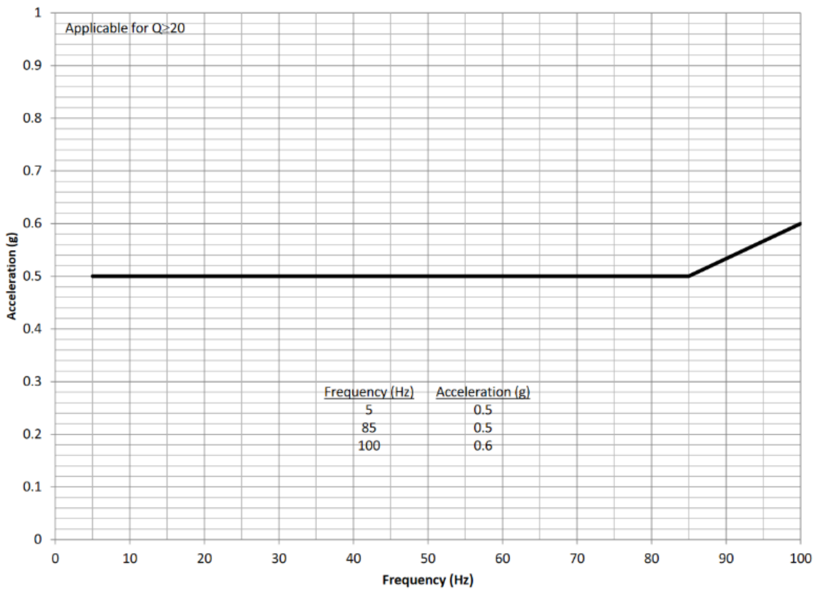


Figure 5-5: Maximum lateral equivalent sine environment for Falcon 9 and Falcon Heavy

5.1.3 Minimum resonant frequency

Quoting from the Falcon Users Guide:

> *Secondary structure designs should consider maintaining a minimum resonant frequency above 35Hz to avoid interaction with launch vehicle dynamics.*

5.1.4 Acoustic Loads

The acoustic loads are relevant only for large thin panels in the structure. Such as the solar panels. [5] The scope of this study does not include the solar panels themselves, but we will consider our solutions effect on how they cope with loads.

5.1.5 Shock environment

From the Falcon 9 payload users guide, we have:

The resulting maximum shock environment predicted at payload interface for payload fairing separation and payload separation (for a 937-mm clamp band separation system) is shown in Figure 5-3. Actual shock from the payload - specific separation system requires selection of a separation system and the associated payload mass properties.

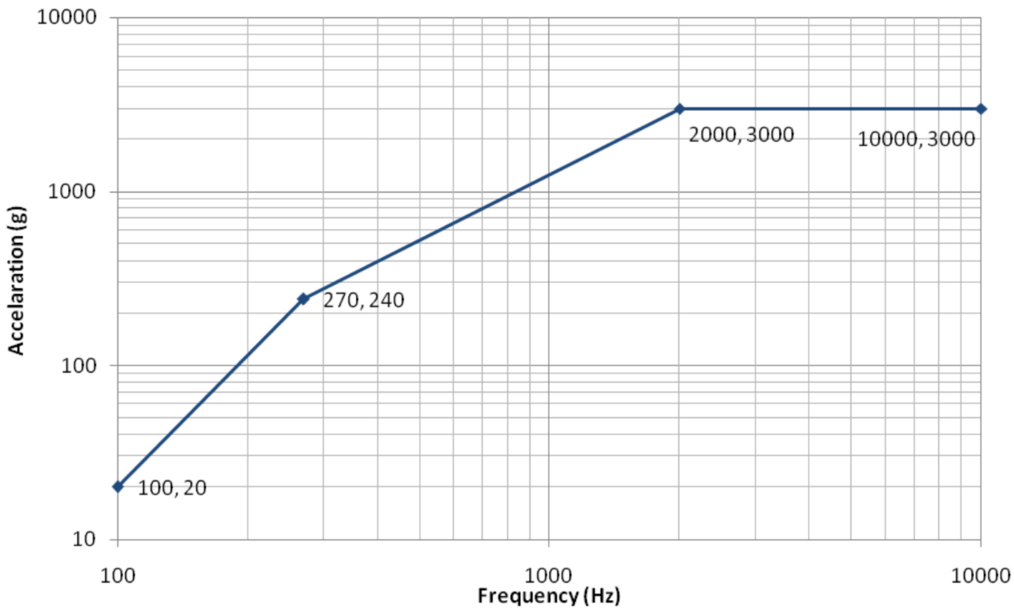


Figure 5-6: Falcon 9 Shock Response at payload interface

5.2 Loads and environment in low earth orbit.

Once placed in orbit, the CubeSat will experience no discernible external forces. Temperature variations and thermal stresses can however be very high.

5.2.1 Thermal loads/environment

A low earth orbit will typically have an orbital period of about 90 to 120 minutes [12]. Depending on the orbit, it might have up to almost half this time in "darkness", and the rest in direct sunlight, not protected by the atmosphere.

> "A metal plate in LEO will cycle from –170°C to 123°C depending on its Sun face and its time in sunlight" *[13]*

5.2.2 Cold welding

The scrubbing of metal parts against each other in a vacuum environment can cause the oxide layers and other contaminants normally found on them to wear off. If this occurs sufficiently, metal surfaces will then fuse together at touch. This is called cold welding.

In practice, this rarely happens in space constructions. The metal parts used have oxide layers on them, as well as other contaminants (dirt, grease etc.), and it would take a long time for the surfaces to wear down bare enough to cold weld.

For our mechanism, the only thinkable concern would be the last phase of launch where vibration occurs outside of the atmosphere. We do not know if this would be nearly enough to cause any cold welding, but we will consider how to avoid it anyways. Polished contacting surfaces and adding a bit of grease, as well as using materials that does not easily cold weld should be a possible solution.

6 Problem analysis

6.1 The task

Is to generate as much power as possible from solar panels, using the very limited space we have. There are solutions with several layers of panels stacked or even solutions using an origami panel, stored inside one unit (1U) of the CubeSat. However, our solution aims to use minimal volume inside the CubeSat itself. Mainly using the few millimetres between the screw holes in our CubeSat frame and the wall of the CubeSat dispenser. (Minimum 6.5mm, possibly a bit more depending on the design of the frame of the CubeSat.)

Folding out several layers/panels is feasible even with this thickness. The solution in Figure 6-2 is in 6.25mm thick when folded. This will require a more customized build and closer cooperation with solar panel manufacturers. The thickness of the usual commercially available CubeSat solar panels suggests that a single fold mechanism might be more feasible if cost-savings is a priority.

Our client told us not to plan the solar panel layout based on a specific orbit and nadir direction of the CubeSat, but to assume the satellite could point itself in the optimal angle towards the sun specifically for charging. This would also fit the scenario of a sun synchronous orbit, where the same side of the CubeSat can always point towards the sun. In essence, we want as many solar panels as possible, and we want them pointing in the same direction.

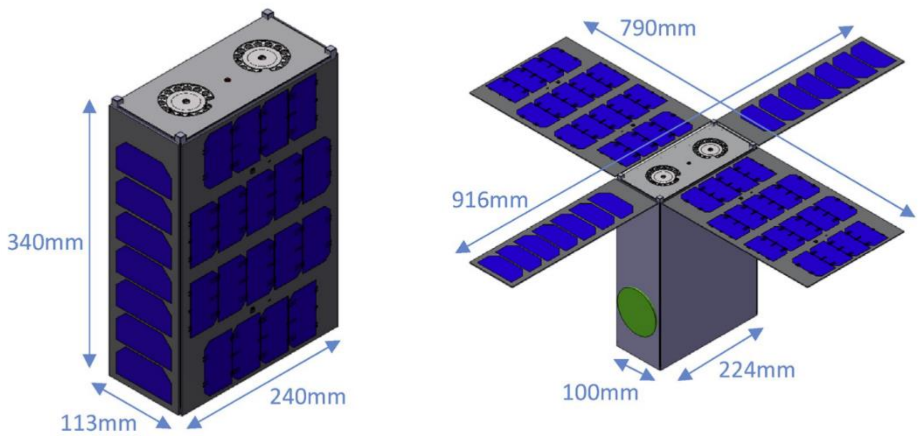


Figure 6-1: Deployable Solar Panels on a 6U CubeSat [14]

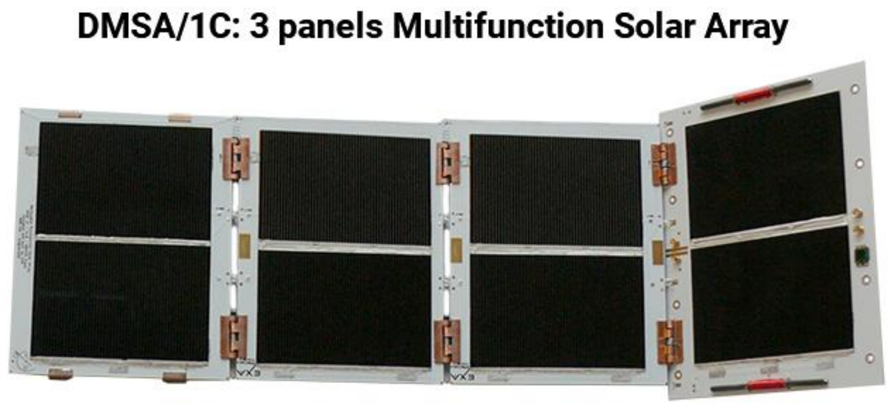


Figure 6-2: A solution by EXA for a 1U CubeSat with 3 deployable panels stacked. [15]

6.2 Our solution to the problem

Considering our client wants a cost saving over a commercially available deploying solar panel, we have chosen to first focus on a single hinge/fold deployment mechanism. We suspect that a multi panel folding mechanism would increase complexity and manufacturing costs significantly for an in-house system and choose to present our client with a single fold solution this time. Possibly, experiences from developing this design can benefit towards a possible multi-fold design being developed by Orbit NTNU later.

This means we will have four hinges and HRM's. The hinges will move from 0 to 90 degrees relative to the Z-axis (see Figure 2-8: Coordinate system). In the deployed position, all solar panels point in the same direction.

6.3 Defining sub-functions

We have split the function of the mechanism into four sub-functions.

- Actuation mechanism for deployment
- Hold and Release Mechanism
- Vibration attenuation
- Guiding/Damping mechanism for deployment

The latter two needs to be considered if found necessary.

6.3.1 The actuation mechanism

This part of the deployment system is responsible for moving the panels from its initial stowed position to its final position at 90 degrees.

6.3.2 Hold and Release Mechanism

The panels need to be held in place, and then released at the proper time. We call the responsible mechanism the HRM (Hold and Release Mechanism). It shall safely hold the panel during the loads from the rocket-launch, and then reliably release it on an electric signal.

6.3.3 Damping mechanism for deployment

If shocks from the stop of the actuation movement can damage the solar panels, a damping mechanism for the actuation will be considered.

6.3.4 Vibration attenuation

The stiffness of the panel itself will mainly handle the vibration attenuation. The panel design is not a concern of this thesis, but we will consider how our system might affect the vibration of the panel it is holding. It would be an advantage if it helps with vibration attenuation. This is a side effect of our solution, not a sub-mechanism in itself.

7 Concept Development

This section contains an overview of possible technologies for our sub-functions. At the end, we will present an evaluation matrix, and the solutions we choose to explore and develop.

7.1 Possible solutions

Actuation Mechanisms	Torsion Spring	Shape Memory Alloys	Electric Motor	Flexible struts		
Hold and Release Mechanisms	Heat/ Burn wire	Shape Memory Alloys	Electric Motor	Permanent and electro-magnets	Magnetic cone pusher	Pin re-lease w. burn wire
Damping mechanisms	Spiral spring	Rotary Damper	Electric Motor	Friction	Rubber Damper	Eccentric Bolt

Table 7-1: Sub function possible solutions

We can mention that the most widely used ones are a torsion spring for the actuation, and a nylon wire cut by a burn-resistor. We have also seen the use of shape memory alloys. However, we have not seen much use of dampening mechanisms.

All the above were considered for our solution, but not all will be discussed in detail.

Electric motors were quickly out of the picture due to their size.

7.2 The Candidates

7.2.1 Actuation mechanisms

These are our final candidates. We will discuss their properties here, and then put them through an evaluation matrix to choose one.

Flexible rubber strut

This solution involves a hinge and a rubber joint that will straighten out when free to move. (See figure). Assuming a panel thickness less than 2mm millimetres, and a 1mm thickness of the attachment, the bending radius of the struts can be no more than 3,5mm.

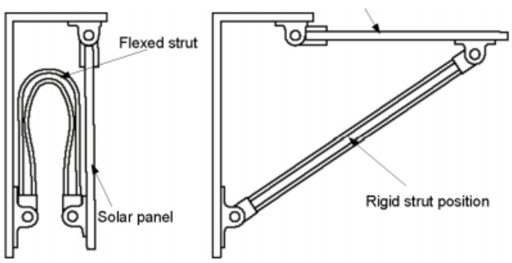


Figure 7-1: Flexible strut

Advantages

- Simple mechanism
- Locks panel in final position
- Small volume

- Inherent damping at the end of the movement

Disadvantages

- Cannot rotate to 180 degrees (if the system were to be redesigned with a mission where its orbit and nadir direction would favour this configuration.)
- Makes only a one panel solution possible, as it uses space under the panel.

Thoughts:

A simple, light, small and reliable solution. Considering the actuation dynamics, the flexible strut solution is as good as or better than torsional springs. Being made of rubber they afford an inherent dampening at the end of the motion. Where it falls short, is being open to more possibilities for different panels and deployment layouts. Say if the panel needs to be rotated 180 degrees. It is of importance that the solution is open to modifications and more “general use” by Orbit NTNU for future missions.

Torsion Spring inside hinge

A torsion spring inside the hinge will return to its natural position, when freed to move by the HRM. The hinge can stop in a metal-to-metal shock, against a dampening system, or find equilibrium between to torsional springs.

Figure 7-2: Torsion Spring

Advantages

- Reliable
- Simple mechanism
- Can be incorporated with other mechanisms.
- Can also be used for deployment damping.
- Takes up very little space on the satellite.

Disadvantages

- Undefined final position, or…
- if the final position is defined, the motion will come to a sudden stop. This might require damping.
- Small forces left at the end of deployment.
- No built-in fastening, initial release, or dampening mechanism (in the case of a locked end position).

Thoughts:

The torsion spring hinge is among the simplest, most reliable, smallest, and lightest of all possible solutions. While the flexible struts are close in these regards, it does not have the flexibility in terms of possible panel layouts. It can be adapted for more designs like 180-degree rotation while having another panel beneath, or having several deployable panels stacked.

Shape Memory Alloy (SMA)

A shape memory alloy can be "trained" to have a shape that it will return to when it is heated. Using an SMA, the actuation can be as easy as having a SMA bar in the shape of a "U" heated by leading a current through it, making it go into the shape of an "L" or an "I" depending on the desired rotation.

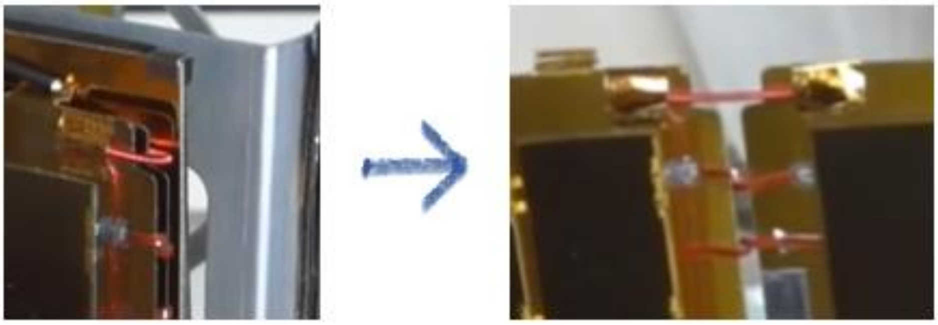


Figure 7-3: SMA Actuation [16]

Advantages

- Low volume
- Smooth actuation movement

Disadvantages

- "Memory loss" can occur if stored in "folded position" for a long time.
- Requires heat for activation. Limits panel choices.
- More complicated than others

Thoughts:

SMA actuators are best combined with an SMA HRM mechanism. The activation heat depend on which SMA material is used. They can have activation temperatures well below the solar panels melting point. However, it also needs to be well above 94 degrees C, which can be reached in the payload fairing of the rocket [5, p. 43]. We also would have to consider the 30 minutes it could be in direct sunlight in LEO before the deployment. We suspect our panel choice to be limited to metal variants, but even then, we would have to consider conduction through the metal towards heat sensitive parts. All these considerations make this solution much more advanced than the other actuation mechanisms. It also excludes low melting point materials for the solar panels. We suspect this might be the reason the SMA deployed panels from EXA (Figure 7-3) are the only ones we have seen made of titanium.

Evaluation Matrix for actuation mechanism

The group members gave individual ratings for nine criteria. The scores were then added and weighted after the importance of the following criteria.

As we are not CubeSat experts (yet), this is design based on initial research and insights into the technologies and eventual CubeSat heritage.

Reliability	How confident we are it will perform as intended after enduring the launch.
Ease of manufacture and/or procurement	Considering our client is a student organization, the tight time schedule a CubeSat project can have, etc. The convenience of making the solution a reality is important.
Development simplicity	How much time and effort we assume the engineering and testing of the system will entail.
Actuation Dynamics	How assuredly and how smoothly it moves the panel. And whether it might necessitate a damping mechanism.
Flexibility of design	Flexibility in terms of possible panel layouts and the possibility of stacking several panels.
Weight	A prime importance in any CubeSat part. Leaving more weight for the "payload itself".
Volume and interference	To what degree it interferes with other parts and systems on the CubeSat.
Cost	Our client wants a reduced cost compared to commercially available deploying panels. At the same time, the production numbers are very limited, hence this is not weighted highly.

Criteria	Weighting	Flexible Strut	Torsional Spring	SMA
Reliability	5	9	10	10
Ease of manufacture	4	7	10	5
Development simplicity	4	7	9	5
Actuation Dynamics	4	8	7	10
Flexibility of design	4	6	10	8
Weight	4	8	10	9
Volume and interference	5	7	10	9
Cost	3	10	10	6
Weighted Score:		77%	95%	79%

Table 7-2- Evaluation of Actuation Mechanism

Conclusion

We conclude that the torsional spring is the best solution for the actuation mechanism. It scores highest overall. It takes precedence over the flexible strut by being more flexible in possible panel layouts, and it is a much less complex solution than using SMA.

7.2.2 Hold and release mechanism

The panel needs to be held down safely during the launch. Both in terms of the HRM itself and the panel being safe from harm. It then needs to release the panel 30 minutes after the CubeSat leaves the P-POD. Hence an electrical signal will be used for activation.

Burn Wire

A widely used method in CubeSats due to its simplicity and reliability of activation. The panel will be secured and fastened by a wire or a thread that can be burnt off by heating a resistor. Once the wire is cut, the main actuation mechanism is free to move the panel.

The term “burn wire release mechanism” also encompasses when a nichrome wire burns itself through many lines of nylon wire. This approach might yield a higher strength HRM but would take considerably more room than the burn resistor solution.

Monofilament (nylon) or braided fishing line would be candidates for the thread. But any wire or thread with a sufficiently low melting/burn point (to not risk hurting other components) is a candidate.

Advantages

- Reliable release
- Very easy to manufacture.
- Low cost
- CubeSat heritage

Disadvantages

- The nylon wire can be a weak point when it comes to launch vibrations.

Thoughts:

This simple technology has extended CubeSat heritage, and our client has experience with a nylon wire and burn resistor mechanism from their current project. The challenge with this solution is to prove that the wire will endure the launch. The methods heritage gives us confidence in it, but the challenge will be to assert confidence in a solution including a nylon wire, without doing vibrational tests on said solution.

Shape Memory Alloy

SMA usage is widespread in the aerospace industry due to its versatility. These materials revert to their original shape when the right stimulus (such as heat or pressure) is applied. Common materials for SMAs include copper-aluminium-nickel and nickel-titanium (“Nitinol”) alloys. The transformation temperature can be controlled by the material distribution. For the initial release mechanism, these materials would be used to clamp the panel down, then move out of the way when heat is applied.

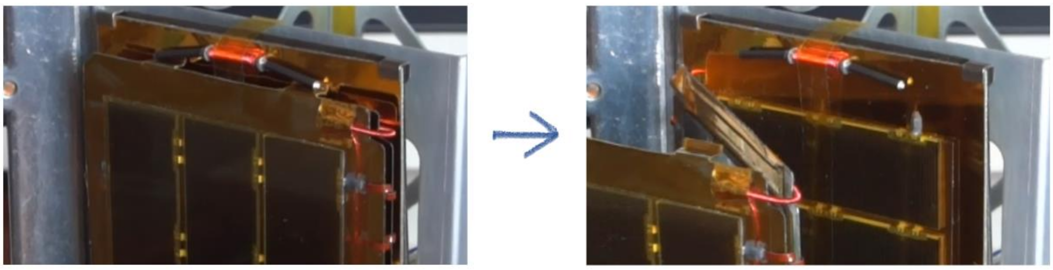


Figure 7-4: EXA deployable solar panels for CubeSats [16]

Advantages

- Reliable.
- High strength. Can clamp the panel down with high force, also providing vibration attenuation.
- Only one moving part.
- Takes little space.
- CubeSat heritage.

Disadvantages

- External energy from battery is required to heat the metal.
- Danger of forgetfulness.
- Training for retracting required.

Thoughts:

This method gives a secure/strong clamping down of the panels whilst taking little space. As mentioned in the actuation mechanism description of the SMA solution, the actuation temperature can complicate things. But maybe less so for the HRM, as the part clamped by the SMA would in our case be heat resistant (metal), even if the panel is less so.

Magnets

This would work by magnets holding the panel down, securing it in the other two dimensions using cones from the panel into holes on the frame part of the mechanism. The release would then happen by two electromagnets cancelling out the permanent ones. From a few magnets up to a line of them 80mm wide could be used on the frame-side, allowing to scalability to a very high clamping force.

Advantages

- Very high clamping forces
- Simple.

Disadvantages

- Can be heavy.
- Magnets can interfere with the electronics in the satellite.
- Cold welding possible
- Complicated design/expensive custom build.

Thoughts:

Building a custom electro-magnet for the release, thin and powerful enough would be challenging and expensive. The system would be wide and tall but can be thin with all the parts (magnets, electro-magnet, cones, and holes) placed on the side of each other. There is one red flag with this system, and that is disturbance of instruments sensitive to magnetism aboard the CubeSat. This system would be limited to CubeSat missions not sensitive to magnetic fields, which is very limiting. We are to develop a system for generic missions, so this essentially eliminates this solution for us.

7.2.2.1 Burn Wire or Magnets combined with pin release

Another idea conceived, is to have a pin or cone holding the panel, to be released by a spring released by a burn wire mechanism. (Alternatively, an electro-magnet pusher mechanism.) This would relieve the wire from holding the panel itself. The pin would secure the panel from movement in the Z-axis, the cones would secure it from moving in the X-axis and Y-axis. During acceleration and vibrations, the wire would mainly need to handle the inertia from the pin and spring, as well as the spring force.

Advantages

- Secure and steady clamping
- Relieves the wire

Disadvantages

- Adds thickness.
- More complex than simpler burn wire solution.
- Needs separate burn-system, as the burn resistor cannot be placed on the panel.

Thoughts:

By relieving the wire from the main loads, we can be more certain it will not break. The power for the resistor can be lead from the solar panel using pogo pins. (Spring loaded connectors.) It will be a challenge to execute the design elegantly.

7.2.2.2 Evaluation Matrix for Hold and Release Mechanism

The group members gave individual ratings for nine criteria. The scores were then added and weighted after the importance of the following criteria.

The criteria are the same as described for the actuation mechanism, with exception of the “actuation dynamics” being replaced by the “vibration attenuation” characteristic of the solution.

Vibration Attenuation	How tightly the mechanism holds the panel. To what degree the fastening has any vibration reducing effects.

Criteria	Weighting	Burn Wire	SMA	Pin Release
Reliability	5	8	10	10
Ease of manufacture	4	10	7	7
Development simplicity	4	8	6	8
Vibration attenuation	4	6	8	8
Flexibility of design	4	9	8	8
Weight	4	10	8	7
Volume and interference	5	10	9	7
Cost	3	10	6	7
Weighted Score:		84%	71%	81%

Table 7-3- Evaluation of Initial Release Mechanism

7.2.2.3 Conclusion

The SMA loses out to the burn wire-based solutions on simplicity. For the two burn-wire based solutions we will have to decide between a simple design with CubeSat heritage, and a more complex one.

While the pin release system is something we would like to construct, we still chose the simple burn wire system. It has extensive CubeSat heritage. While the wire might elicit some doubt when thinking about the loads during launch, we trust it to be secure due to its heritage. It is also the simplest to execute for our client.

7.2.3 Damping

If the panel stands a chance of breaking during the sudden stop of the actuation mechanism, we will need a damping mechanism. We have considered methods like hydraulic rotary dampers or elaborate friction-based latches, spiral springs, Belleville springs or stacked wave springs. But they are all disqualified due to the extremely limited space on our hinge. Only two are considered here.

7.2.3.1 Rubber Damper

A rubber damper replaces metal parts of the system that produce very high collision accelerations. While the deceleration is “short”, it is much longer than it would be with a metal-on-metal impact.

Advantages

- Simple
- Easy to fit on small surfaces. Can be glued on.
- Can be cast into shapes that maximize adhesive area or allow utilization of limited space by following the geometry of the part.
- Great shock absorber

Disadvantages

- Short deceleration
- Danger of outgassing. Material choice limited by outgassing-approved rubbers.

Thoughts:

One of two possible methods here, and the only one that has a fixed end position.

7.2.3.2 Torsions spring equilibrium (oscillating movement)

This method involves not having a sudden stop to the movement at all, but having torsion springs oscillating towards equilibrium. Safe travel of 90 degrees beyond the springs natural position in either direction is needed. The springs needs to be accurately placed so that their natural position is 90 degrees from the starting position. When the HRM releases the panel, the springs would oscillate towards equilibrium at 90 degrees from the panels initial position. This way the panel would not experience any shock. The kinetic energy of the system would dissipate into friction between parts (and some to internal friction in the spring), and it would come to a stop after a while. To try and mitigate the friction stopping the mechanism before the spring reaches its natural position, the spring should be as powerful as possible. More energy would be stored in the spring and dissipation would take longer.

Advantages

- No shock for the panel
- Simple mechanism

Disadvantages

- Undefined/Inaccurate final position.
- Low spring forces due to high safe travel. HRM separation might need to be ensured by separate mechanism.

Thoughts:

The inaccurate final position is a challenge. It must be weighed up against the challenge of making sure the panel endures the shock with the other method.

7.2.4 Choice of damper

The choice of damper comes down to whether we want a well determined fixed end position. The first two are eliminated due to space restrictions. While the oscillating solution provides superior avoidance of shock, we will propose the rubber dampers for this project as they provide the most reliable way to damp while guaranteeing the optimal end positions of the panels with the maximum area normal to the sunlight.

8 Prototyping and further development.

The development of our solution started in CAD. The hinge and torsion spring is quite basic, so we started with the HRM after designing a rudimentary hinge. Some basic design decisions based on specifications, wishes from our client, and our own observations from prototyping paved the way.

This chapter will include images of the CAD-model that was taken during the design process. As the design was iterated upon until the last few days of work, we have not updated all images in the thesis. We will present the images of the final solution at the end of this chapter.

8.1 Initial design considerations

Based on specifications, wishes from our client and our own observations and this was our initial specifications for the design.

Initial Design Consideration	Solution
The burn resistor is to be placed on the PCB panel. This was the most convenient for our client. The panel will be a custom made one.	This means the wire needs to pass through the panel twice, as the burn resistor should be on the underside of the panel to avoid additional thickness, and to make sure it is not smashed against the wall of the dispenser if the wire were to elongate from loads.
The wire shall not become space debris. It shall remain attached to the spacecraft after it is burned through.	The second point means the wire should be secured in some way, so that when the panel is cut loose, the wire cannot float into space. The CDS states no additional space debris shall be created .
The wire needs to remain pressed against the burn resistor after enduring the launch loads.	The third and fourth point, means it is an advantage having a pliable (not stiff) line. The torsional spring in the hinge, needs to pull the line out of the four holes in FIGURe81 at 0.2m.
The wire needs to easily glide out of the panel-side attachment once it is cut.	
We will use two resistors for double redundancy.	The fifth point was added late in the design process, when we heard this was common practice from Orbit NTNU.

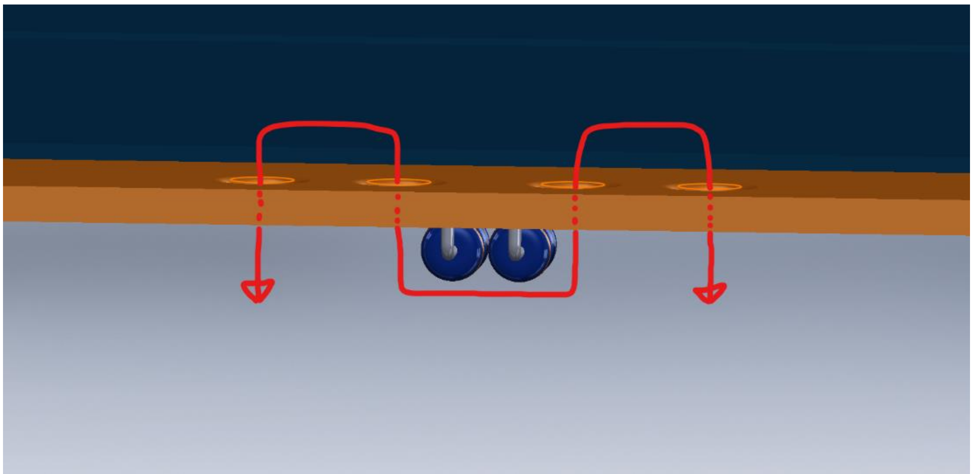


Figure 8-1: How the wire goes through the panel-side of the HRM

Figure 8-1 shows how the wire goes through the panel. On the panel side, the HRM consists only of four holes and the two resistors, powered from the PCB.

8.2 Prototyping

We prototyped using 3D-printing in plastic.

The first prototype was when we realized how small these parts really are. We also realized that a thick line can be a negative. A little force was needed to separate the panel after the wire was cut. This can be lowered with a thinner/bendier wire. We tried lines of diameter 0.4mm and 0.6mm. The force was small, but not as negligible as we had thought before prototyping.

We also found that getting the line very tightly tied around the mechanism was not trivial. We could pull the panel about 1-2mm away from the HRM due to this when using the 0.6mm line Using the 0.4mm line it was noticeably closer. It was however inconsistent. This was not good, as our cad model at the time had the panel about 1mm away from the wall of the P-Pod. We decided to aim for a larger margin between panel and P-pod, and a tighter fastening between the panel and HRM.

Figurer! (Show 3D print? Show before and after size.)

Subsequently, an additional method surfaced. We theorized an HRM, where instead of tying knots we would have a groove where the line would go through, into which some form of glue would be placed. This would also remove the weakest point in a nylon wire-based holding mechanism: The knot. On Orbit NTNU's latest vibration test a 0.2mm line was used. This was theorized to hold, but it failed at the knot.

This would also eliminate the need for an extra "round and knot" to secure the wire from becoming space debris.

8.3 Actuation Mechanism

8.3.1 Hinge

At first, we made a rather simple hinge design (Figure 8-2), but once we got our required size for our torsion springs from our calculations, things got more complicated.

Firstly, 4mm is our target height for the upper surface where the panel will sit. If the spring legs were to go into the hinge parts with a reasonable distance from the edge, only a very small spring and axle will fit.

If a design is chosen, that only requires a very small spring and axle, a solution like this may be chosen.

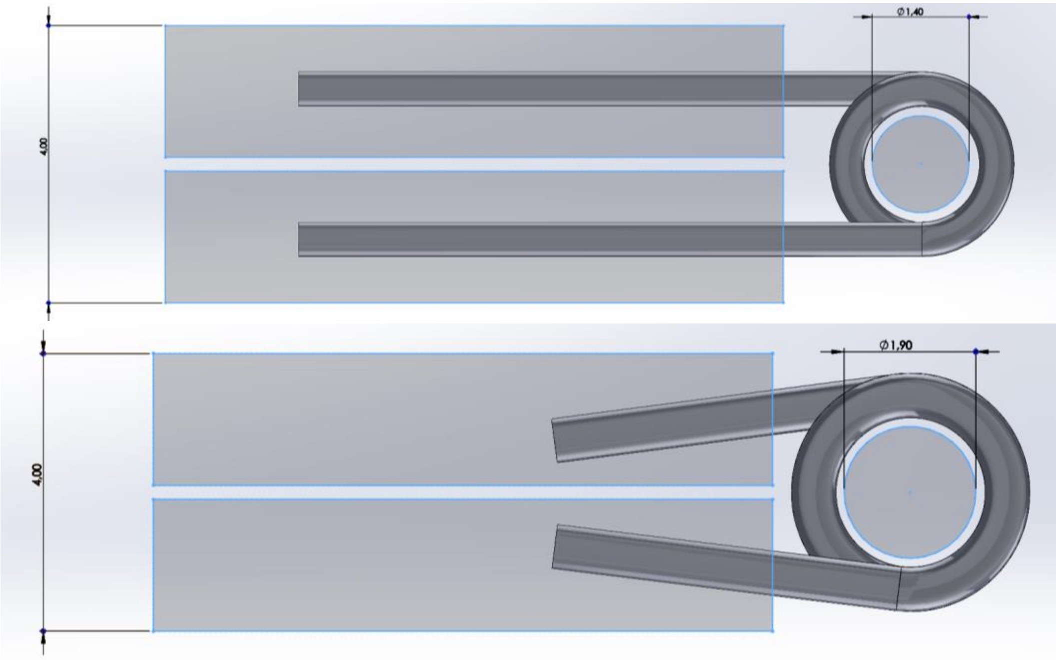


Figure 8-2: Hinge Design 1

When dimensioning our spring we decided we might use a spring that would not be afforded by this design. An attempt was made by drilling slanted holes for it like in Figure 8-2, but it would not fit. Figure 8-3 shows a section view.

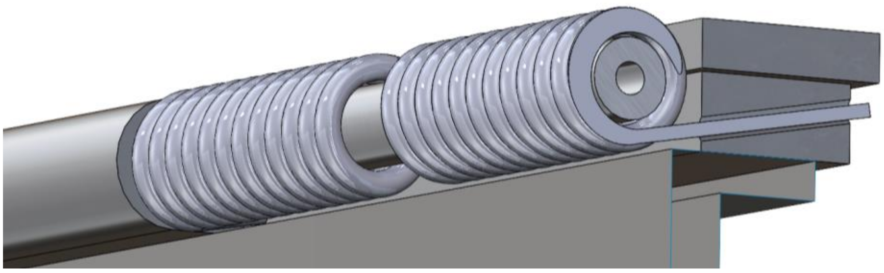


Figure 8-3: Simple Initial Design Idea

This prompted us to a full re-design where we used all available space. The CDS states that no component goes more than 6.5mm away from the yellow area in Figure 8-4

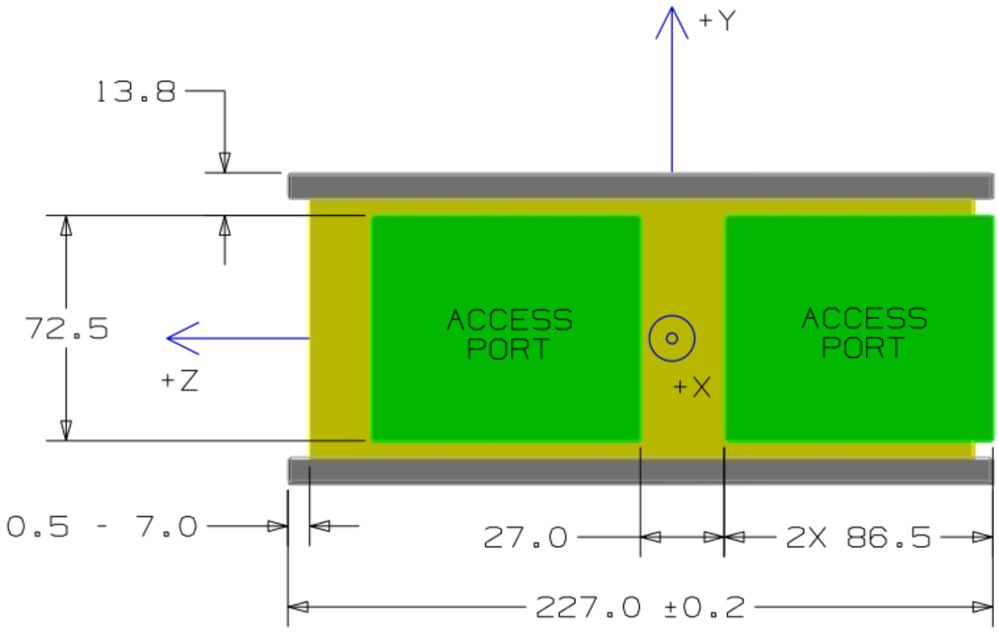


Figure 8-4: CDS Space Limitations

The following design revision led to this:

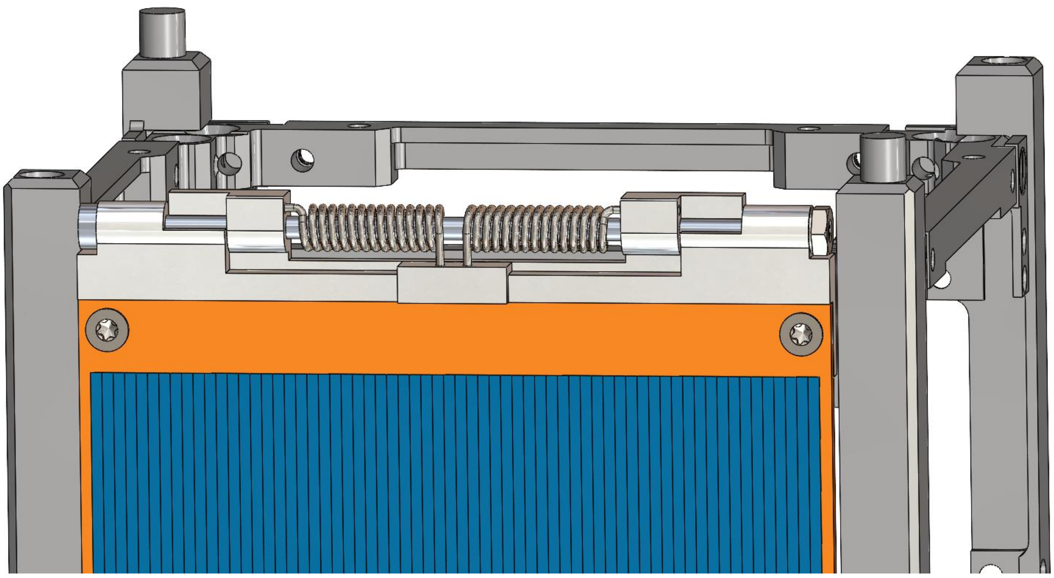


Figure 8-5: Hinge on CubeSat Frame with Mock-up Solar Panel

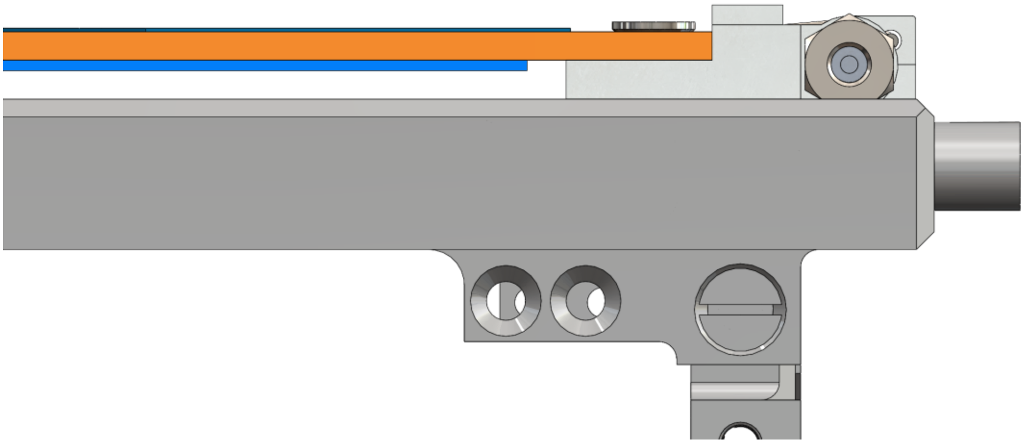


Figure 8-6: Side view of the hinge on the frame

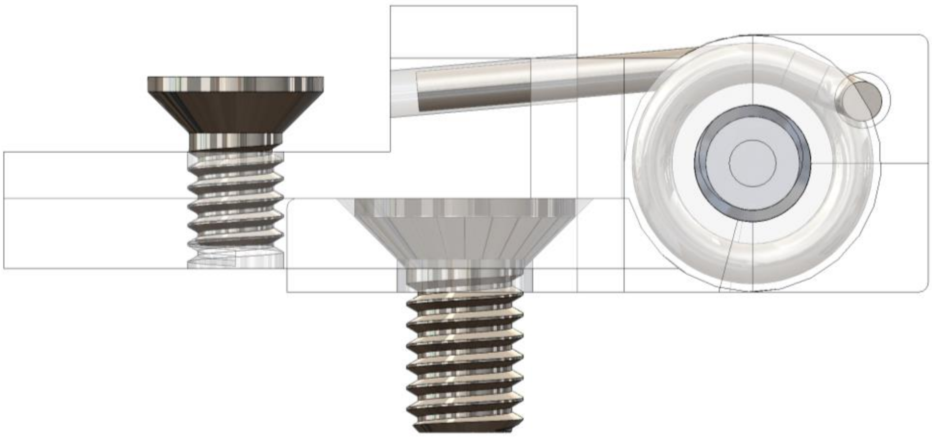


Figure 8-7: See through view of the hinge with screws, torsion spring and axle

The holes for the axial spring legs are 0.8mm from any edge.

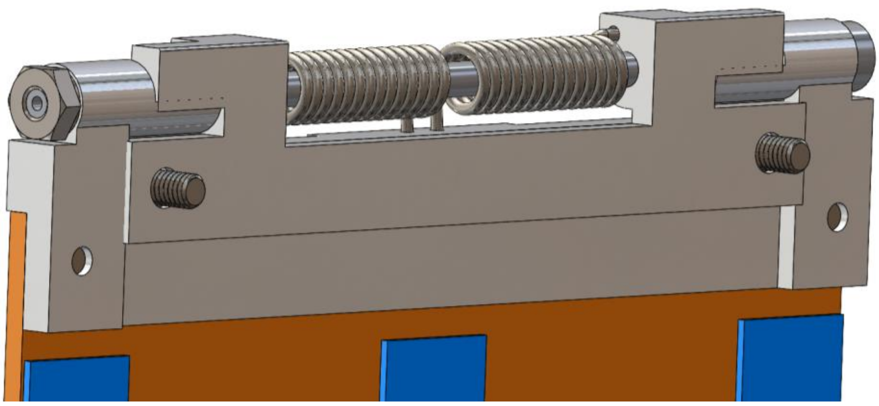


Figure 8-8: Rear view including panel and panel stiffeners

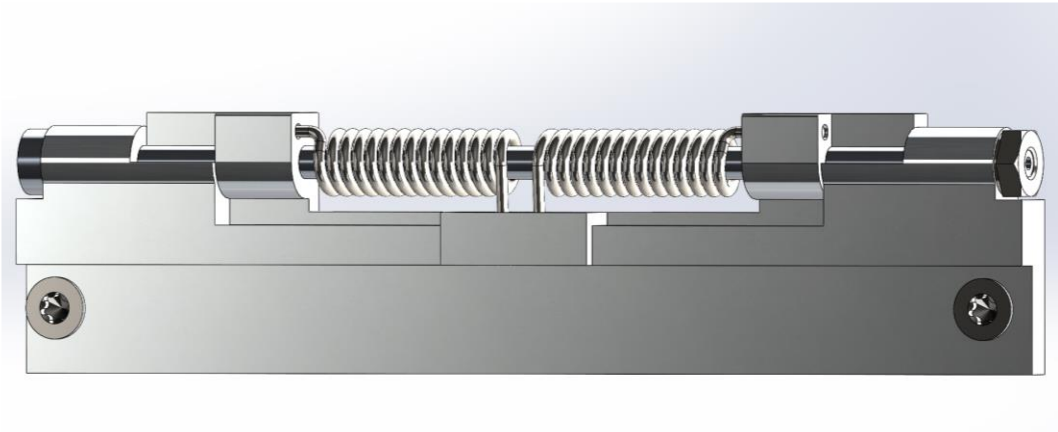


Figure 8-9: The hinge

In this design we employed all possible space to make room for an axial leg from the torsion spring. We also used all 6.5mm to fit the tangential leg. This design affords a much larger spring than the initial simpler one. Whether such a spring is needed depends on the design of the rest of the system.

We have also made a “step” down to where the panel sits. This way we can achieve our 4mm panel fitting height, while having a larger spring.

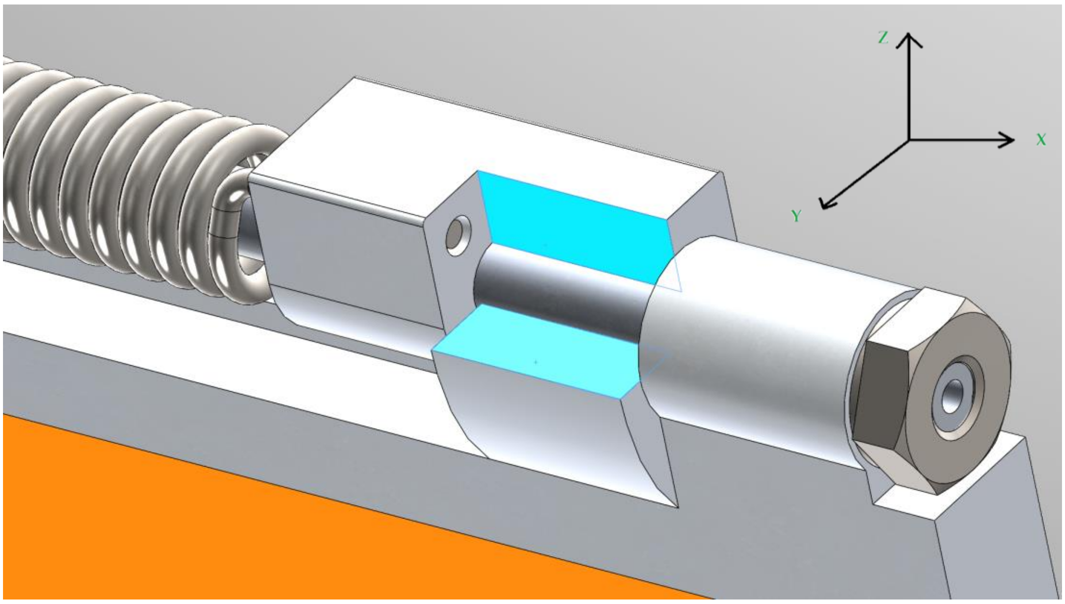


Figure 8-10: Hinge stopping mechanism without damping

The hinge stops by these two surfaces contacting. Our proposed damping solution is to place a rubber block here. This can be fastened by gluing. The appropriate surface treatment and adhesive types, depend on the hinge material and the type of rubber chosen. For our adhesive and rubber, we must look to outgassing approved materials.

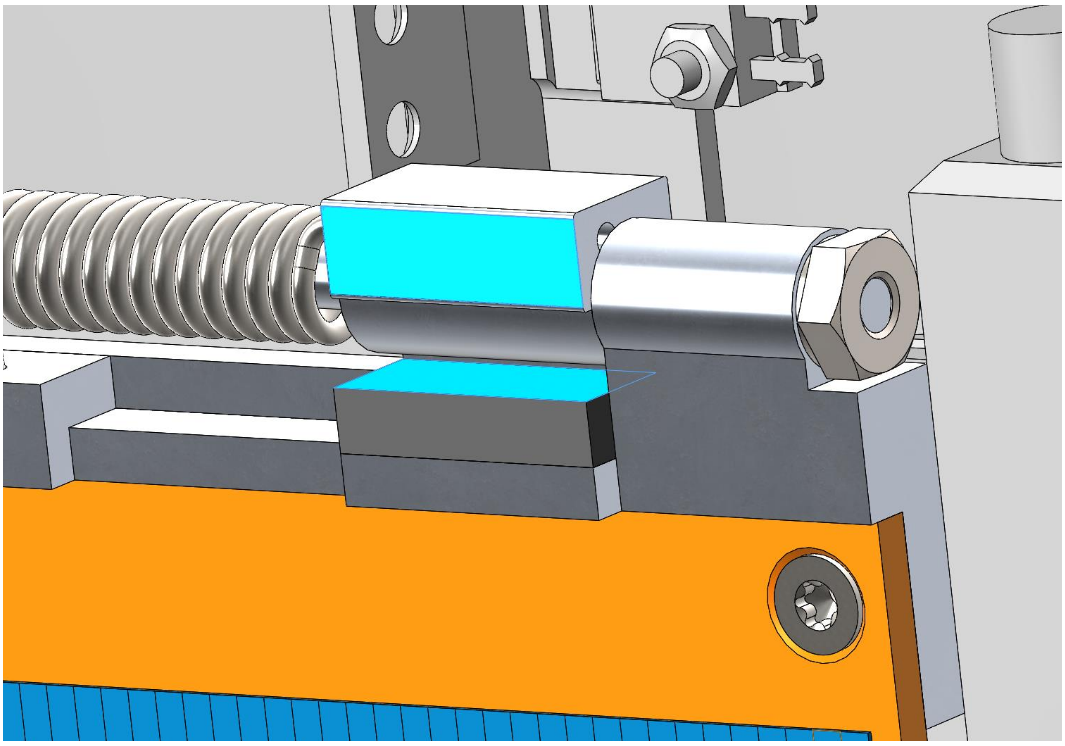


Figure 8-11: Hinge with rubber damper

This design is changed to accommodate a rubber block, which is glued onto a surface that is parallel to the impact surface. This minimizes the shear force on the adhesive bonding as the impact force is approximately normal to the adhesive surface.

If there is some doubt on the adhesive's longevity in LEO, a screw may be used to secure it from becoming space debris.

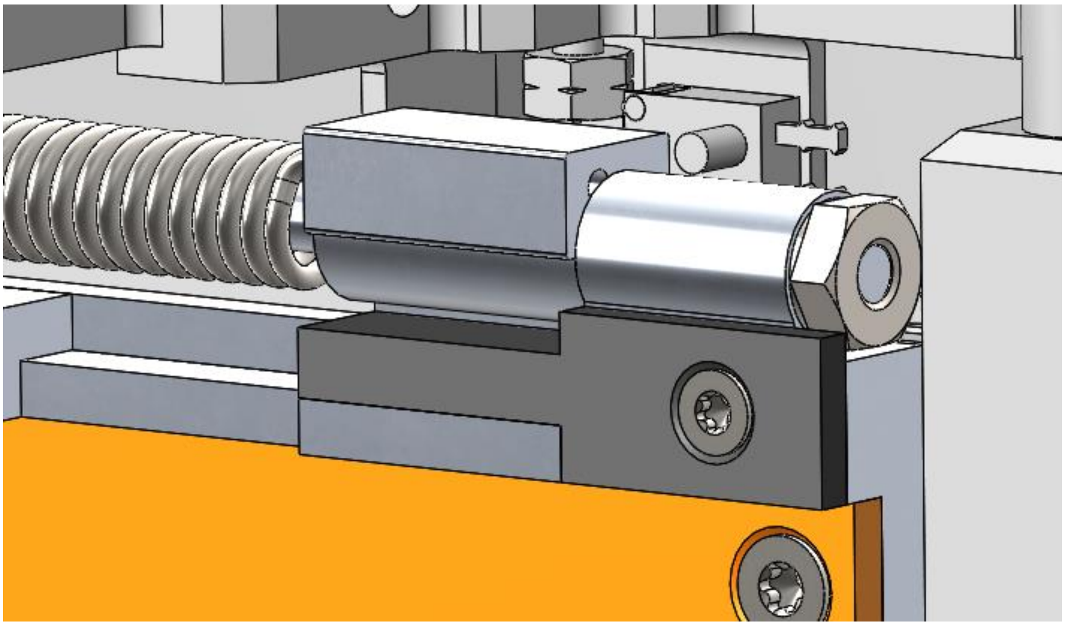


Figure 8-12: Space Debris Secured

We are not sure how the adhesives will respond to extreme repeated temperature variations in a vacuum. This will have to be further investigated, but to be on the safe side we choose the solution with screws. This also means we can prioritize the softness of the damping material instead of bonding capabilities. The adhesive is still tasked with holding the rubber in the “optimal impact position” through the launch loads.

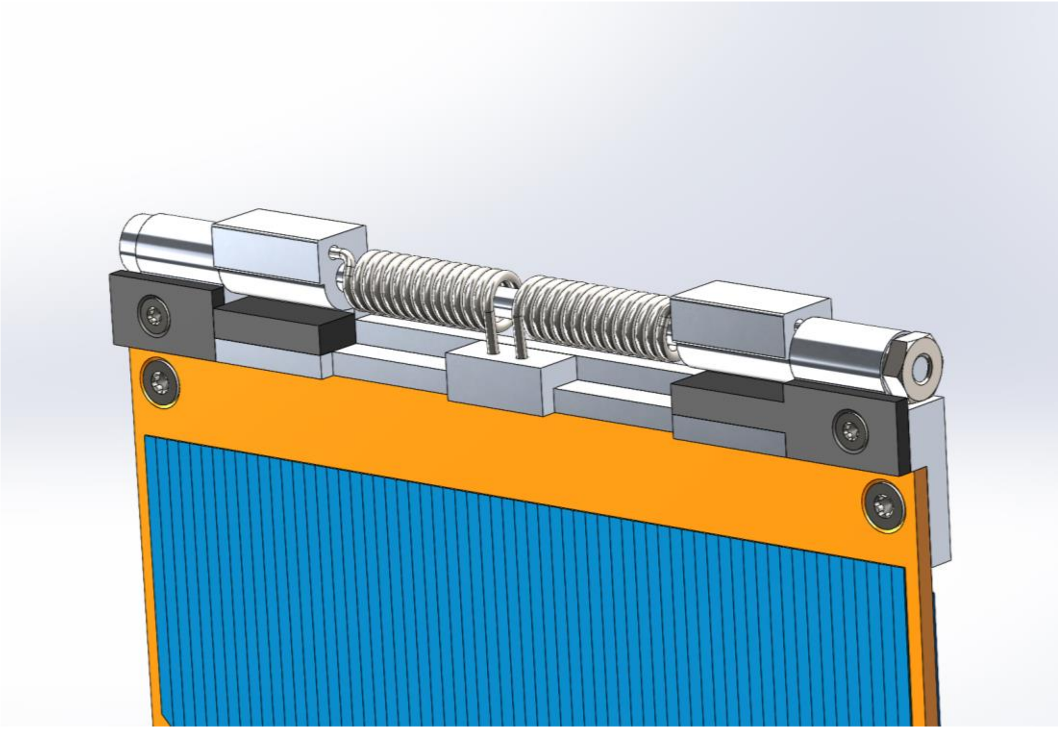


Figure 8-13: Final Hinge Design

8.3.2 Torsion Spring

The spring should accelerate the panel as slowly as possible, while at the same having the torque to pull the nylon wire out of the holes in the HRM. As previously mentioned, our first prototypes surprised us by this force not being as miniscule as we had thought. The tiny spring is to provide this at 0.2 meters distance in our case, or up to 0.3m (for potential use in 3U or 6U CubeSats.)

There are two options to ensure separation in the HRM end. Either using a torsion spring that we know provides enough force or making a “separation assuring system” at the HRM. This would be based on compression springs.

We were not able to test the separation force with a final panel, only with simple 3D printed prototypes. We did not measure the force for these prototypes, as corona virus restriction limited our access to the university workshop for parts of the semester. Therefore, we assume a force in Chapter 9. We expect this force is higher than the actual one, even with a relatively thick nylon wire.

It was an early design decision to make the torsion spring ensure separation. This was to allow for a simpler HRM part. However, as seen in section 8.3.1, fitting the larger springs meant complicating the hinge design to a large degree.

In hindsight we could have chosen to have springs in the HRM design as a “backup solution” if it is found necessary to ensure separation in actual testing. We could then have designed a simple hinge based on a small, low torque spring as first intended. (Ref. Figure 8-2: Hinge Design 1.) This way damping might not have been necessary. However, damping might still be needed, and the design affording a larger spring also has more room for damping. Hence, we will keep it for this thesis.

The hinge dictates a spring with one tangential and one axial leg. The number of coils is dictated by this equation:

$$N = n + \frac{\theta + 90 + A}{360}$$

Where N is the number of active coils. n is the number of whole coils. θ is the number of degrees between the tangents from the springs mean diameter that passes through the centre of the holes for the spring legs. (See Figure 8-14). We add 90 degrees to this angle, as this is the rotation of the hinge. Furthermore, we add A which is the additional travel we want left in the spring to hold the panel in place.

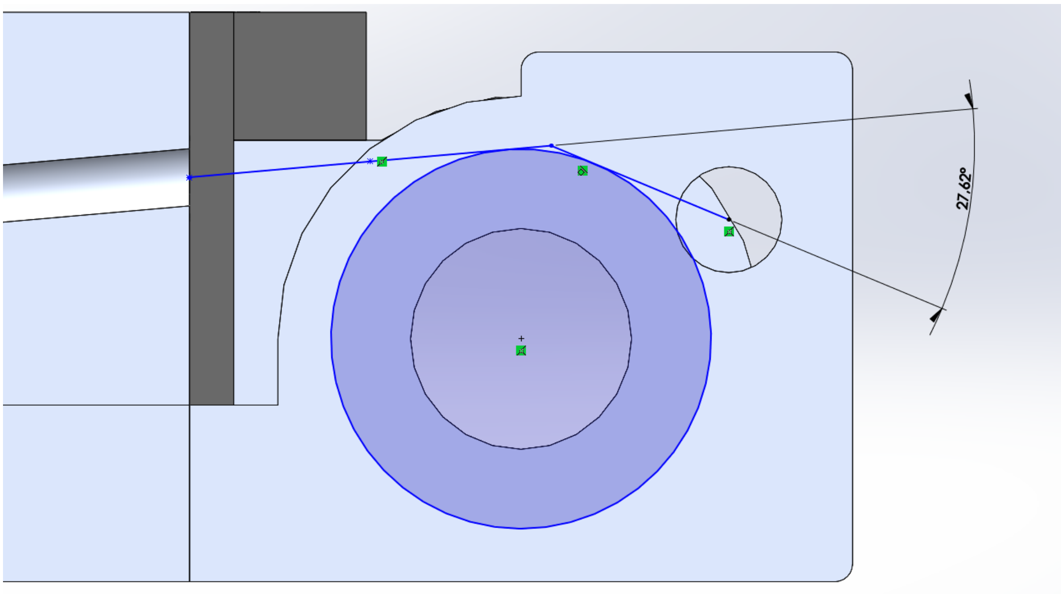


Figure 8-14

If we choose A = 10 degrees, our hinge dictates an active coil number ending in .35.

This specific coil number in addition to the axial leg, means a custom spring must be made. How to dimension a torsion spring for a specified torque and safe travel is shown in chapter 9.1.

8.4 HRM

8.4.1 Panel Side

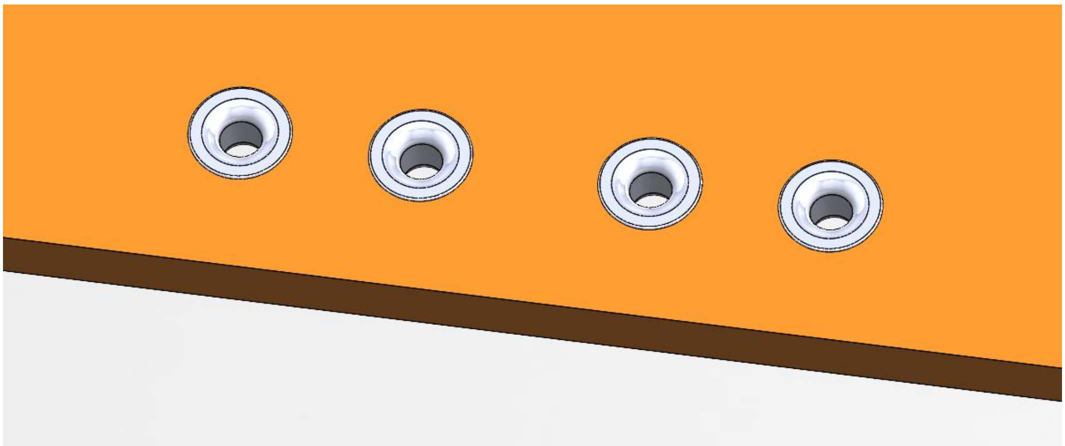


Figure 8-15

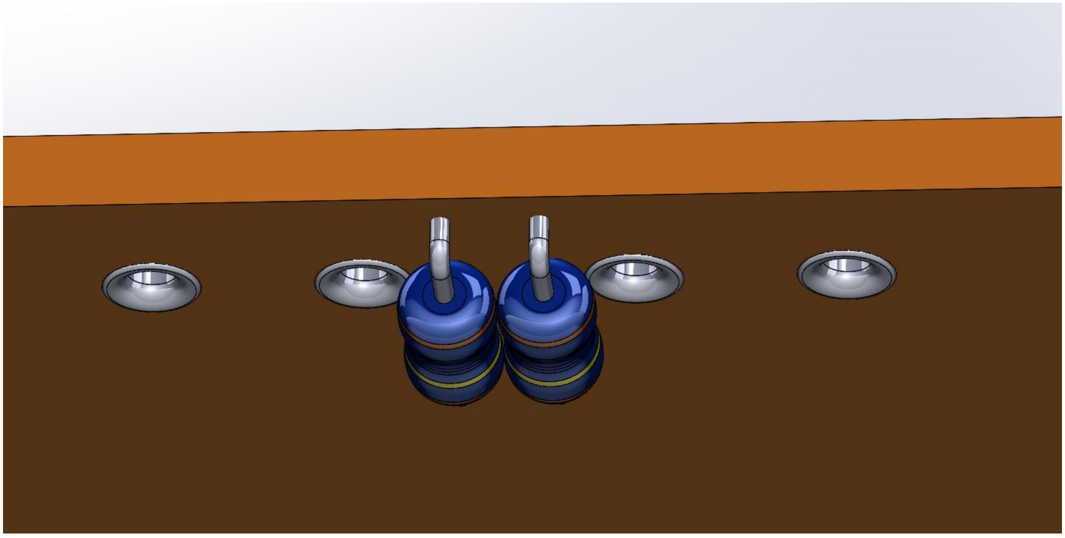


Figure 8-16: test

The holes can simply be machined out of the panel material, or we can use some sort of tubular rivet like in Figure 8-16. They need to be shaped so that the wire easily glides out at separation, and so that the wire strength is not compromised during the launch.

There are two resistors for redundancy. This normal practice in CubeSat systems. Design of the solar panel PCB/electrical systems are not in the scope of this thesis. We have therefore not detailed the fastening of the resistors in the CAD model. We leave this, as well as adapting the mechanism for the actual resistor setup, to Orbit NTNU.

We have used relatively small resistors with a diameter of 2mm and a length of 3.2mm for our CAD example. The system can be made for larger ones, or for a SMD resistor.

Figure 8-17: SMD resistor alternative

The SMD resistor is much smaller. It could be placed on top of the panel due to its thinness, allowing for just two holes in the panel. It would also need a thinner more flexible line, and/or dedicated geometry of the HRM in order to ensure the wire is pressed against it in such a way that it is burned/melted off.

We have assumed a round resistor like in Figure 8-16 will be used.

8.4.2 Frame part

8.4.2.1 Tied HRM

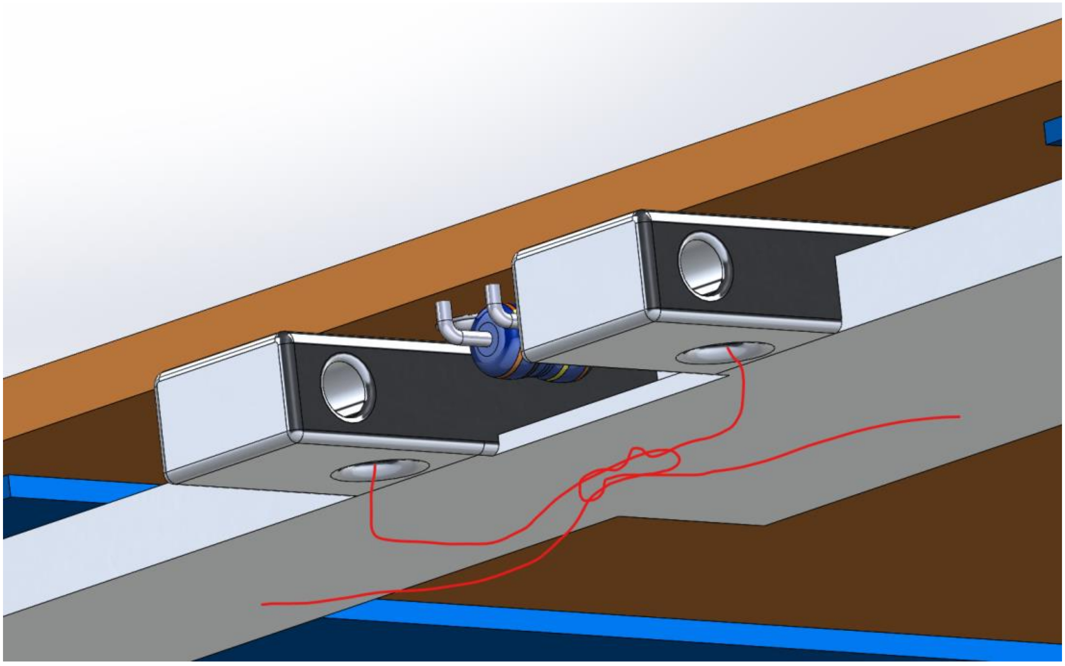


Figure 8-18: The first knot

The first surgeons knot will be tied directly under the HRM as shown in Figure 8-18. In addition, the line goes through the “horizontal” holes in the part and under it to be tied in another surgeon’s knot. The second one is to ensure it does not float away and become space debris.

8.4.2.2 Glued HRM

This method is of our own creation and we have not seen similar solutions. The red line shows the wire coming out of the hole in the bottom of the HRM. The line takes a turn under the part in order to lay centrally on the glue covered surface. The green arrows show where the line is to be covered in a deep layer of glue or resin. The geometry is constructed with the assembly of the system in mind. The “high wall” is there to ensure no glue will be spilled on moving parts during assembly of the system.

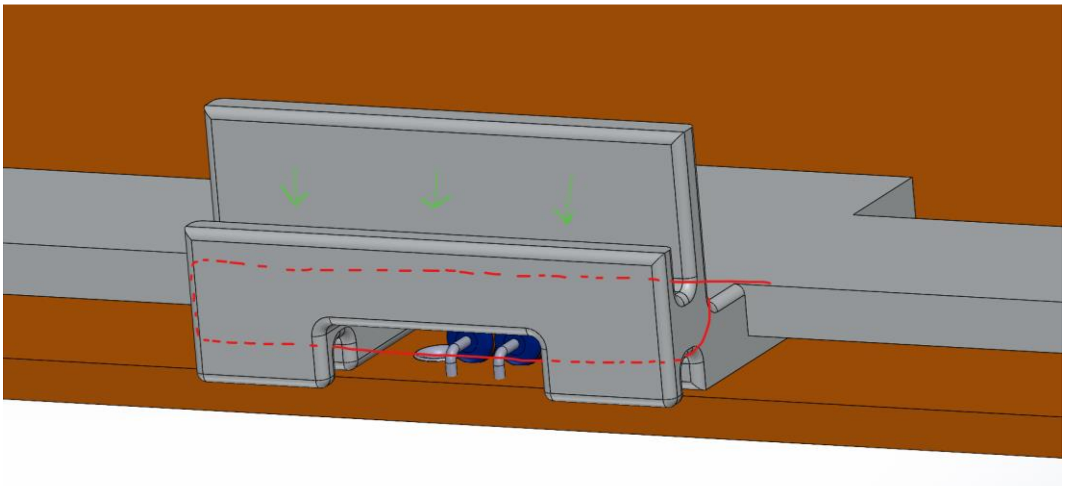


Figure 8-19: Glued HRM

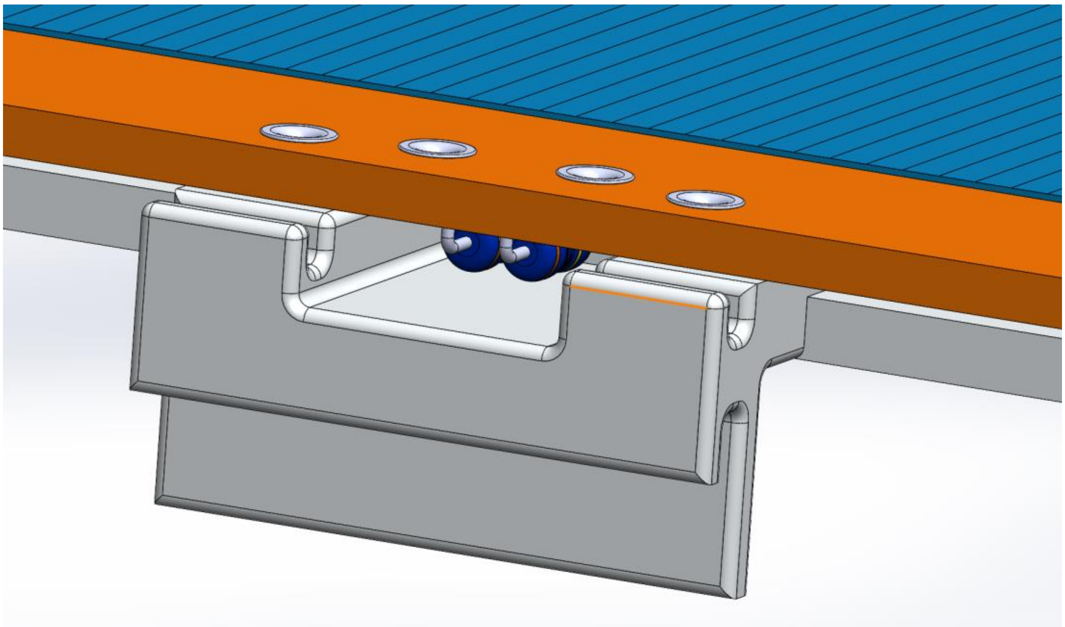


Figure 8-20: Glued HRM

Aluminium needs to be thoroughly cleaned/prepared before the application of glue. This can be difficult in a tight groove. Therefore, a different material should be used.

This material should have a yield strength well above the result of the FEA that bonds well with glues that works with the chosen wire.

Using this method removes the weak point that is the knot. Is also allows the line to be held tight while the glue cures. (For example, by two weights, or a more elaborate setup.) This way it is easier to reduce the excess wire length to a minimum. This is what we experienced with our prototyping as well.

Our prototype was 3D printed plastic, 0.6mm monofilament nylon wire and superglue. We could not tear it apart by manual force. It also had noticeably less excess wire length

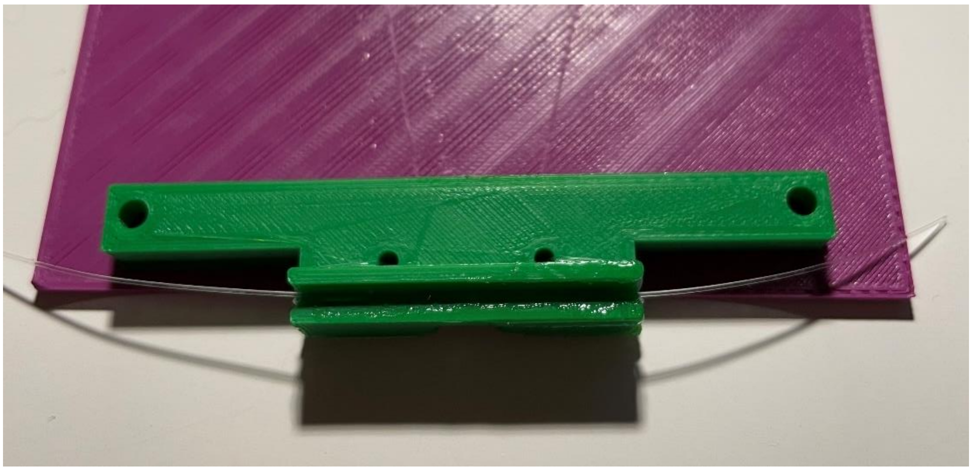


Figure 8-21: Glued HRM prototype

If this system is better at enduring the vibrational and quasi-static loads during launch, can only be determined in testing. But it removes the knot from the system.

It is however much more cumbersome to assemble. Four HRM's is to be assembled, and this needs to lay on one side and cure for each one. It also takes up space from the bottom of the CubeSat. So, it might be in the way of a payload.

We consider it a backup solution to the first one.

8.4.3 Separation

If we decide to use a thick line and a weak torsion spring, we might need a system that ensures separation of the panel once the line is cut.

We can do this with springs. A draft of such a system can look like this.

Figure 8-22: HRM with three separation springs.

If it is found that a larger force is required, another row of springs can be added. Just make the extrusion that holds the spring longer (or wider) along the CubeSat frame.

If fastening such tiny springs becomes impractical, larger diameter springs can be used. These will then have a lower force.

If we can drill a hole 1mm away from piercing our part, our example mechanism can hold a spring with a «solid height» of under 2 mm. We have seen a number of springs with this solid height.

8.4.4 Nylon wire

For the HRM mechanism, a wire able to safely secure the solar panel to the satellite and be safely burnt off when needed is required. The wire should also possess desirable mechanical properties, so as to withstand the environmental loads during launch. For this, threads used in fishing lines were considered due to its size, strength, elasticity, and shock resistant.

Fishing lines can be divided into three main types. Monofilament, fluorocarbon, and braided lines. The strength of these lines increases proportionally with the diameter size. A monofilament line is weaker than a fluorocarbon and braided line of the same diameter but has a higher dynamic strength for shock absorption. As shocks are expected during the launch phase, a monofilament line was considered to be a more viable option for our mechanism.

A monofilament line was tested for use to investigate the working principles. We looked at how the wire was fastened around the resistor and the solar panels. We tested two wires of diameter 0.4mm and 0.6mm. We encountered difficulties effectively fastening our wire around the mechanism as we went up in diameter size. This resulted in a weaker line and danger of disentanglement (loosens by itself). Going up in size for more strength, can therefore be counterproductive.

Figure 8-23: Monofilament line from Kinetic super mono

The knot was a major point of concern as it is usually the weakest point of the line. We quickly found out that the knot got less tight and weaker as we went up in size. This signifies that a minimum thread size is more desirable.

This wire should be able to be tied tightly as we have already experienced. A surgeon knot gives great knot strength. Number of turns for the knot will affect its strength. 3 to 4 turns is the most optimal for best strength [17]

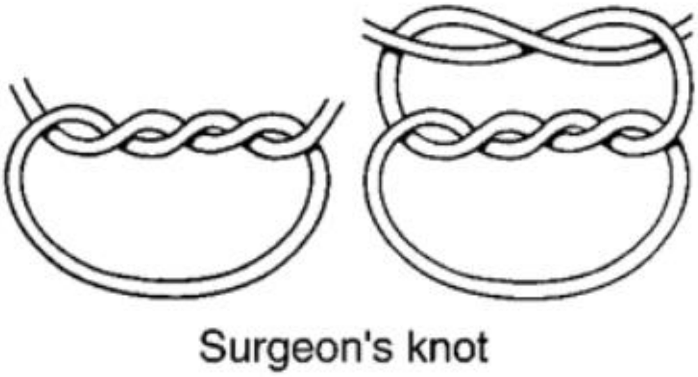


Figure 8-24: Three- turn surgeon knot.

Alternatively, braided lines can be considered. They often have 1/3 to 1/4 the diameter of mono or fluorocarbon lines at a given breaking strength test. We can have a thinner line with the same strength, or a similar line with a greater strength. They are also known to be more

Braided lines are much less elastic than monofilament lines. In general, they break at the knot before elongating. This can be a wanted characteristic of our line, as any elongation due to prolonged vibrational loads is unwanted. A mo   nofilament can elongate to a much higher degree before breaking at the knot.

8.5 The Assembly

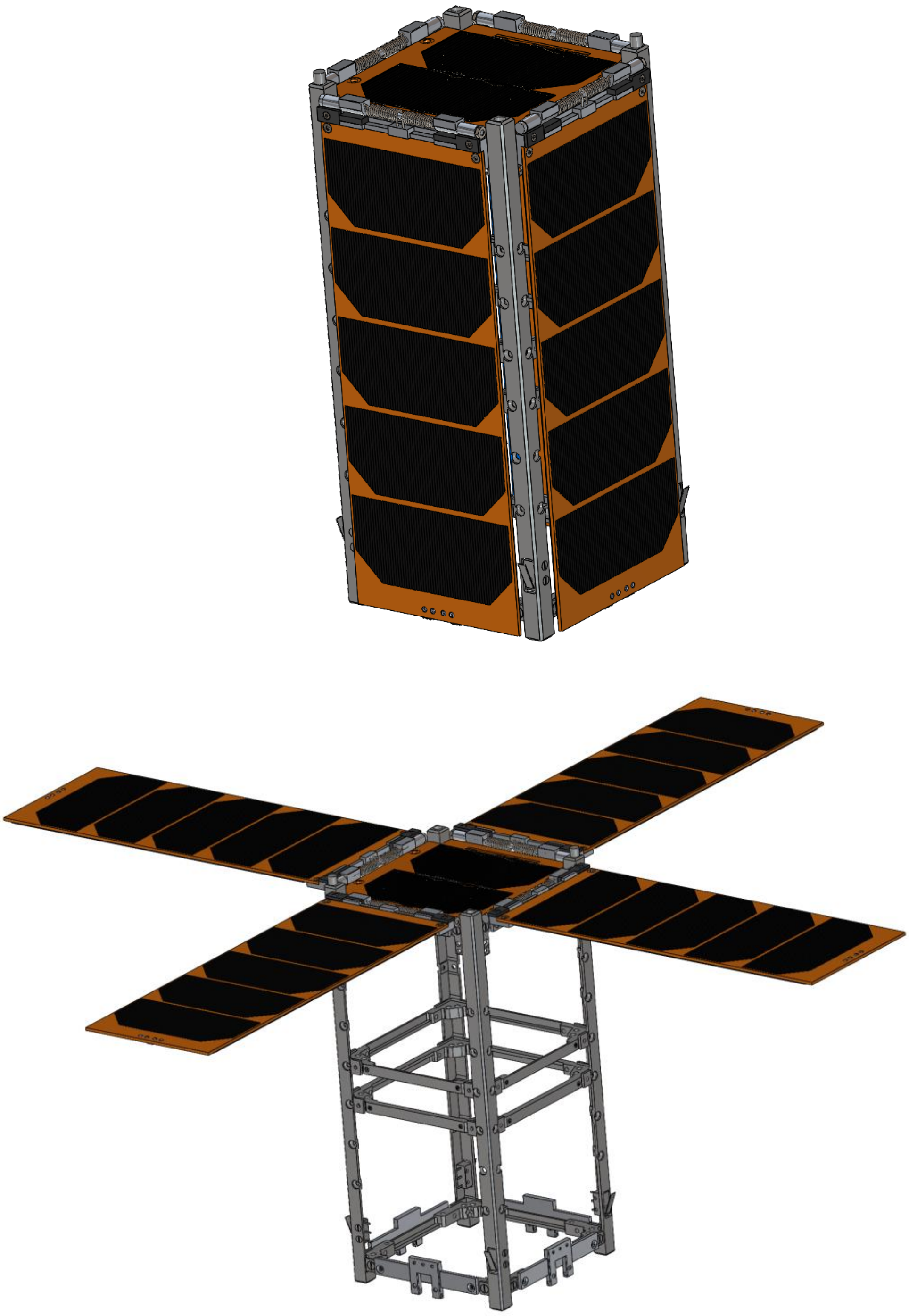


Figure 8-25: The Solar Panel Deployment System

9 Calculations and dimensioning

9.1 Torsion Spring

Our choice to derive dimensions for a custom spring is reasoned in chapter 8.1.

We do not know the force needed to separate the solar panel from the rest of the HRM. It depends on the type of wire and its diameter. This will be known after vibration testing of the system has validated a wire, which is not in the scope of this thesis. We only know from 3D printed prototypes, that while this force is small, it is not necessarily as negligible as one might think. We will assume a separation force for the sake of arriving at an example spring for our solution.

In addition to the right force at the springs starting position, we want a specified safe travel distance.

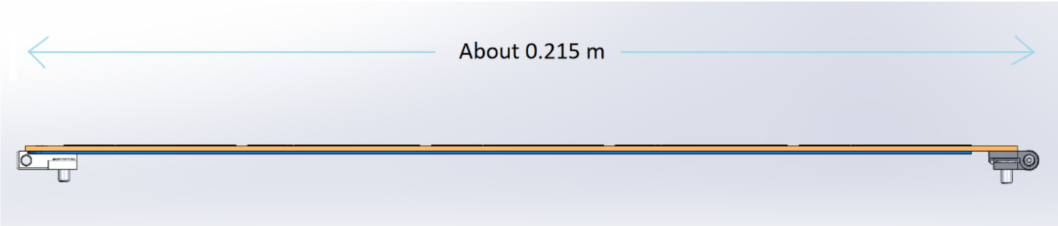
We define the following for further use:

Property	Variable	Unit	Equation/Formula
Inside Diameter	ID	mm	
Mean Diameter	D	mm	
Outside Diameter	OD	mm	
Wire diameter	d	mm	
Leg length	l	mm	
Number of active coils	N	Dimensionless	
Modulus of elasticity	E	N/mm	
		Dimensionless	
Moment/Torque	M	N*mm	
Working travel	$\Delta\alpha$	Rotations	
Spring safety factor	f_{spring}	Dimensionless	
Safe Travel	S	Degrees	$S = \Delta\alpha * f_{spring}$
Spring rate	k	N*mm/rotation	$k = \frac{M}{S} = \frac{Ed^4}{10.8DN}$
Spring index	c	Dimensionless	$\mathrm{C} = \mathrm{D/d}$
Wahl Factor	K	Dimensionless	$K = \frac{4c^2 - c - 1}{4c(c \pm 1)}$
Bending stress	σ_b	N/mm^2	$\sigma b = \left(\frac{32M}{\pi d^3}\right) * K$

Table 9-1: Spring variables and equations

Our dimensioning values

Assuming a HRM separation force of 0.5N in total, and using two springs, we get a force of 0.25N per spring. This is to occur at 0.215 meters distance from our hinge axis.



$$M = F * l = 0.25N * 215mm = 53.7 \text{ Nmm} \qquad 9.1$$

This is our moment. We then need the travel (in degrees or rotations) that our spring is supposed to work with.

$$k = \frac{M}{\Delta\alpha} = \frac{53.7 \text{ Nmm}}{100 \; degrees} \approx 0.5 \frac{Nmm}{Degrees}$$

We then need our safe travel. This has to be the working travel of 100degrees plus a safe factor. We choose the safe travel to be 180 degrees = 0.5 rotations. We assumed this to be sufficient for the spring being handled during assembly and being far off the materials yield strength during launch loads. Safety factors for the yield strength are often used, but we choose to encompass it into the safety factor here. It is normally 0.8, which means our safety factor for the safe travel is really 1.45.

$$100 \; degrees * \frac{1.45}{0.8} \approx 180 \; degrees$$

We have just assumed a safety factor for this example, and are not experts in choosing a safety factor for torsion springs. The 1.45 is intended to achieve less frailty during the assembly of the system.

Dimensioning of the spring diameters and coil number

The spring rate, or spring constant, is given by the (wire and mean) diameters, the E-modulus and the number of coils. The constant in the denominator is normally 64 for the moment per radian [18, p. 434]. This translates to 10.2 to Nm per rotation. However, testing has shown that to compensate for friction between the coils and axle, it should be increased to 10.8 [19]

$$k = \frac{Ed^4}{10.8DN} \left[\frac{Nm}{rotation}\right] \qquad 9.2$$

This formula does not encompass our safe travel, so we will derive one that does, and by settings both formulas equal, make an equation that satisfies both our wanted spring rate and safety factor. The three unknowns in this system will be the wire diameter (d), the mean diameter (D) and the number of coils (N). The rest of the variables are known from design decisions and material properties.

Using the formula for bending stress we derive the maximum moment from our materials yield stress.

$$\sigma_b = \left(\frac{32M}{\pi d^3}\right) * K \qquad 9.3$$

Rearranging and putting in our yield stress we get our max allowed moment:

$$M_{max} = \left(\frac{\sigma_y \pi d^3}{32K}\right) \qquad 9.4$$

M_{max} is to be reached at the safe travel distance of the spring, which gives:

$$k * S_a = M_{max} \qquad 9.5$$

$$k * S_a = \left(\frac{\sigma_y \pi d^3}{32K}\right) \qquad 9.6$$

$$k = \frac{\sigma_y * \pi d^3}{32 * K * S_a} \qquad 9.7$$

We substitute in the expression for k in equation 9.2 to get an equation containing all three unknowns.

$$\frac{Ed^4}{10.8DN} = \frac{\sigma_y * \pi d^3}{32 * K * S_a} \qquad 9.8$$

$$N_s = \frac{E * d * 32 * K * S_a}{10.8 * D * \sigma_y * \pi} \qquad 9.9$$

Where N_s is the coil number as determined by diameters, material properties and the springs safe travel S_a.

We now have our coil number as determined from our safe travel. Rearranging equation 9.2 we get our coil number as determined from our wanted spring rate:

$$N_k = \frac{Ed^4}{10.8 * D * k} \qquad 9.10$$

Where N_k is the coil number as determined by diameters, material properties and the spring rate.

We now have two functions N(d, D), each based on separate design factors k and S_a.

We can then plot them in the three-dimensional space of [D, d N]. To see the correlation between our remaining unknowns. Where the two expressions/planes for N intersect, is where both our spring factor k and our safe travel S_a are satisfied.

Our choice of D, d and N is further delimited by the recommended spring index "c". This is said to be between 5 and 15 [18]. The spring index is given:

$$c = \frac{D}{d} = 5 \qquad 9.11$$

Our choice is cut off by the plane where:

$$\frac{D}{d} = 5 \tag{9.12}$$

We can also delimit by the maximum diameter, which we assume to be around 6mm. (No need to calculate it exactly, but we do not want the spring to touch the P-POD 4.5mm from the hinge axis.)

At last, we need our properties to put into our equations.

Properties	Symbol	Value
Elastic modulus	E	$206\ N/mm^2$
Safe Travel	S_a	$0.5\ Rotations$
Spring Rate	k	$0.5 \frac{Nmm}{Degree} * 360 \frac{Degrees}{Rotations} = 180\ Nmm/Rotation$
Yield Stress	σ_y	$2170\ N/mm^2$

Table 9-2: Torsion Spring Properties

The most common spring material is high carbon spring steel. The one we will use as an example is "Music Wire - ASTM A 228". We have found our data on it from [20]. Its tensile yield strength is heightened because the springs are cold drawn. It changes with the diameter as such:

Music Wire - ASTM A 228

Wire diameter	Tensile yield stress
0.200 mm	2750 - 3040 MP
0.300 mm	2600 - 2880 MPa
0.400 mm	2500 - 2760 MPa
0.610 mm	2350 - 2600 MPa
0.810 mm	2250 - 2490 MPa
1.00 mm	2170 - 2410 MPa
1.30 mm	2090 - 2310 MPa
1.60 mm	2020 - 2230 MPa
2.00 mm	1940 - 2150 MPa

Table 9-3: Wire diameter and yield stress

The plane equations 9.9, 9.10 and gives the 3D plot.

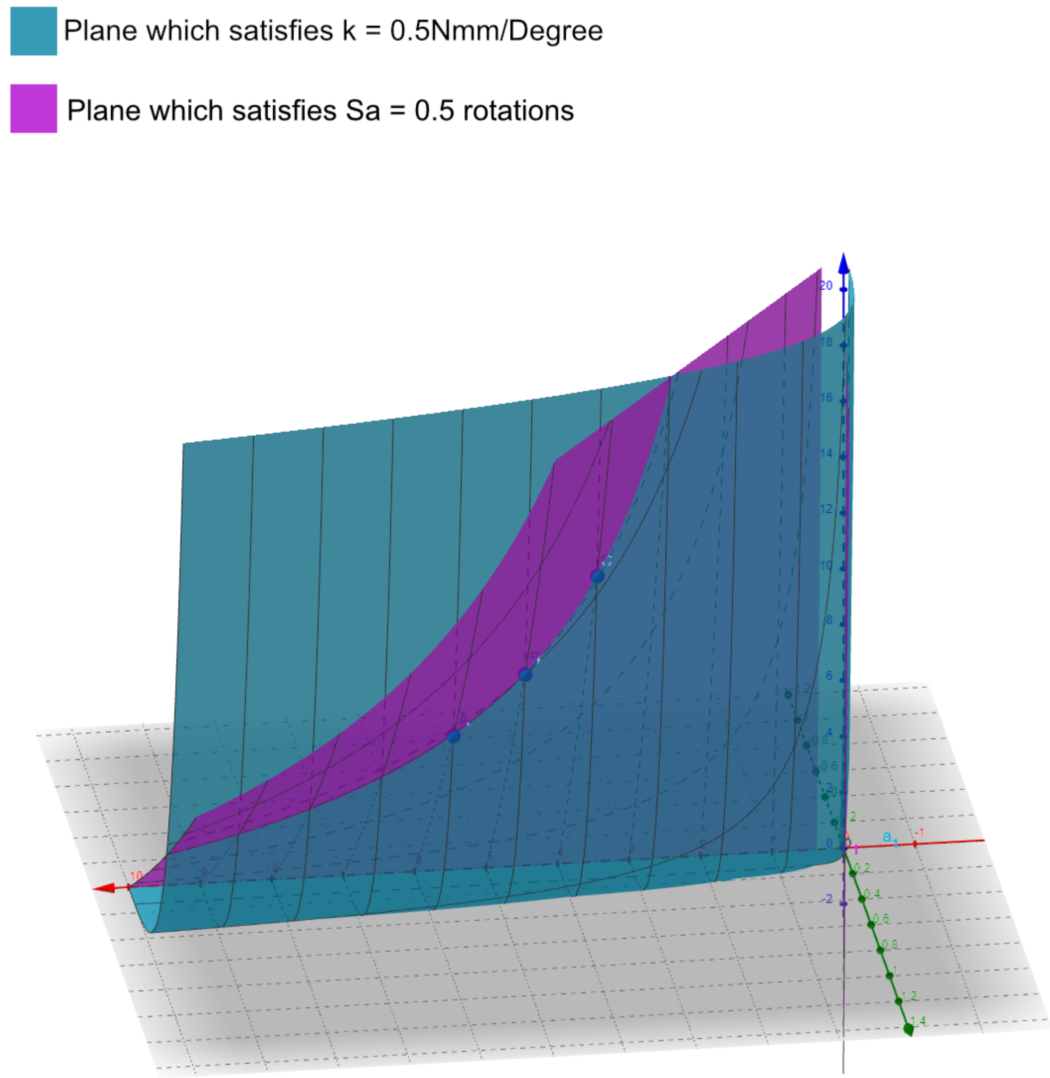


Figure 9-1: 3D plot

The green axis is the wire diameter. The red axis is the mean diameter. And the blue axis is the number of coils. The three blue dots are the intersection points at mean diameters 4, 5 and 6mm. We can see that our number of coils will be reasonable. (The top of the Z axis is 20.) We can also see that if the mean diameter were further reduced, the coil number would increase very steeply.

To choose a value for D and d we look at the plot from above, normal to the D-d plane:

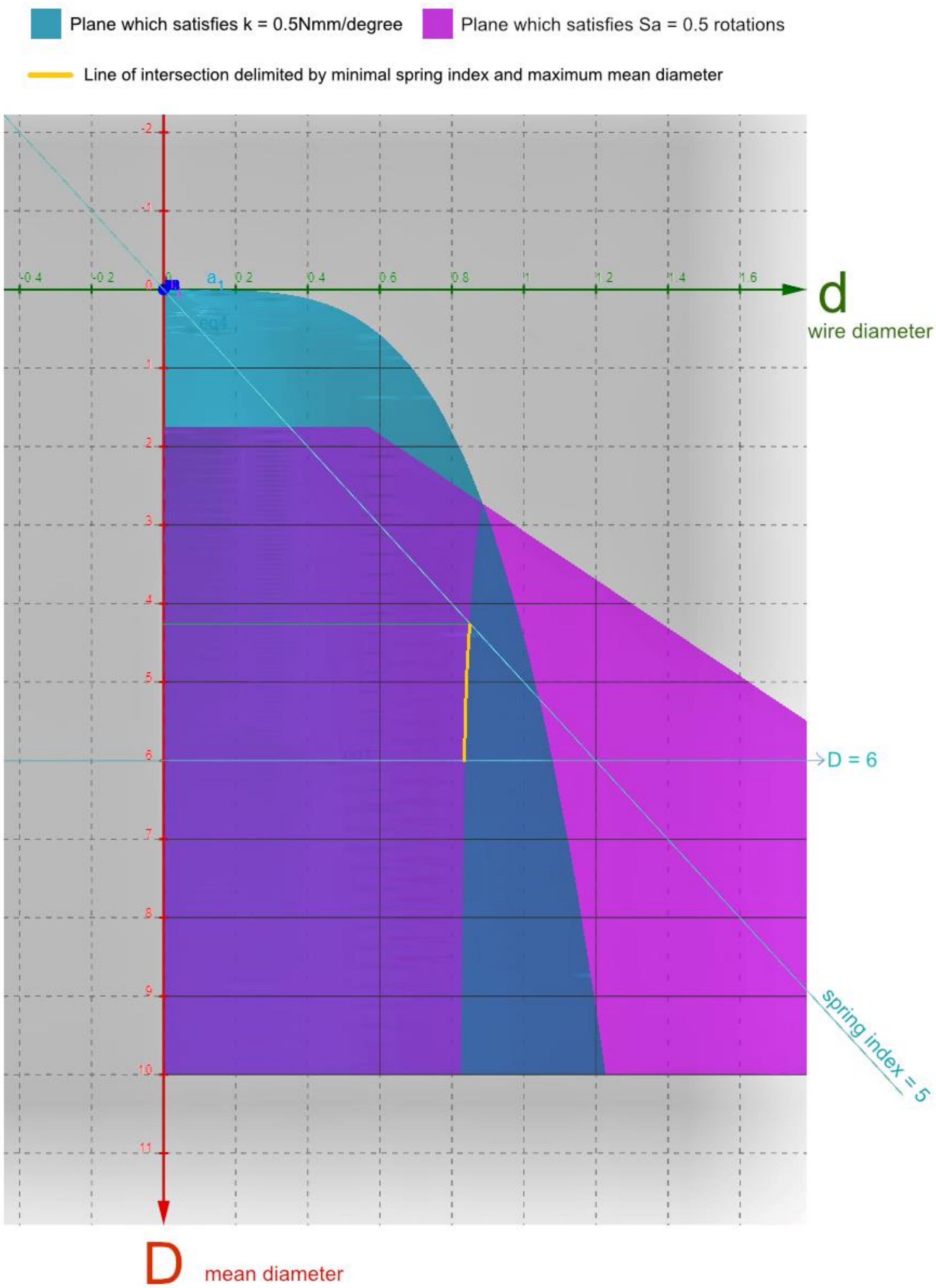


Figure 9-2: Delimited 3D plot from above

The yellow line marks where we can choose values from.

As we can see, the range of D, d and N combinations for our set spring rate and safe travel are small. The dark line from the top of the yellow line onto the D-axis marks our minimal mean diameter. We can graphically read it as 4.27mm.

Since thinness is important in our design, and we are comfortably above a reasonable axle diameter (which passes through the centre of the spring), we can choose to be at the minimal mean diameter.

We set our mean diameter as 4.3mm.

We could determine the wire diameter from the plot as well. It does not vary much with D, and it can only be machined to a realistic precision anyhow. If we are roughly on values from the yellow line, our spring will perform roughly as expected. However, we choose to show how to calculate the wire diameter exactly, in order to use it to get an exact coil number calculation afterwards.

Settings the plane equations equal, we get:

$$\frac{Ed^4}{10.8 * D * k} = \frac{E * d * 32 * K * S_a}{10.8 * D * \sigma_y * \pi}$$

$$\frac{E * d * 32 * K * S_a * 10.8 * D * k}{10.8 * D * \sigma_y * \pi * Ed^4} - 1 = 0$$

$$\frac{32 * K * S_a * k}{\sigma b * \pi * d^3} - 1 = 0$$

$$\frac{32 * K * S_a * k}{\sigma b * \pi * d^3} - 1 = 0 \qquad 9.13$$

Substituting in the Wahl factor in terms of D and d:

$$\frac{32 * \left(\frac{4\left(\frac{\mathrm{D}}{\mathrm{d}}\right)^2 - \frac{\mathrm{D}}{\mathrm{d}} - 1}{4\left(\frac{\mathrm{D}}{\mathrm{d}}\right)\left(\frac{\mathrm{D}}{\mathrm{d}} - 1\right)} \right) * S_a * k}{\sigma_y * \pi * d^3} - 1 = 0 \qquad 9.14$$

Solving for d, where D = 4.3 and using the material properties from Table 9-3, the solution for the wire diameter becomes:

$$d = 0.852488 \text{ mm}$$

For reference we read 0.852537 graphically.
Wolfram|Alpha was used to obtain the equation solution.)

We set our wire diameter to be 0.85mm.

Spring manufacturers provide diameter specifications to this precision [21]. Custom made springs can be ordered, but there is off course no need for extreme precision, as we are just trying to achieve a roughly specified force for our spring.

We proceed to find the coil number from equation 9.10.

$$N_k = \frac{Ed^4}{10.8 * D * k} = 13.9915$$

This coil number would give us our safe travel and spring rate. However, we need a coil number ending in .35 (See page 46)

Increasing the coil number will increase safe travel and lower the spring rate. The strain produced in the material is lesser when the spring is longer. We increase the coil number to 14.35. This changes the safe travel up and spring rate down about 3% (Proportional to the coil number change) which is negligible.

Our coil active coil number is to be 14.35

For further design we note the inner diameter of our spring to be 3.45mm and the outer diameter to be 5.15mm.

Having such specific requirements for the spring might not be entirely necessary, but we have showed how one can derive them if desired.

9.2 HRM Springs

If we are to define springs for the HRM instead of ensuring a powerful enough custom torsion spring, the task becomes much easier.

We have a flexible number of springs and mean diameter. We need a short solid height that can be fitted to the HRM. 1-2mm. We need a free height long enough to ensure separation. (Found by testing.)

Once these are found, head to a spring catalogue and choose the most powerful spring that fit these criteria. Use enough of them to achieve the required force.

Compression Spring Dimensions - MM	
Stock Part Number	PC178-1575-7000-MW-3302-C-N-MM
Outer Diameter (mm)	1.575
Inner Diameter (mm)	1.219
Free Length (mm)	3.302
Solid Height (mm)	1.422
Wire Diameter (mm)	0.178
Total Coils	7.000
Rate (N/mm)	0.726
Max. Load (N)	1.365

Figure 9-3: Example Stock Spring

The stock spring in Figure 9-3 would achieve twice the force that we set for the torsion spring. Assuming they sit 2mm into the HRM, meaning they are 70% compressed.

$$F_{max} * \left(\frac{H_{solid}}{H_{compressed}}\right) = 1.365N * \frac{1.422mm}{2mm} \approx 1N$$

9.3 HRM FEA

A simplified static FEA shows the occurring stresses.

First, we the loads on the part with our maximal assumed acceleration of the panel 15g and a safety factor of 1.25.

$$15g * 1.25 * 9.81\frac{N}{kg} * 0.1kg = 18.4N$$

This is the force from the weight of the panel due to acceleration. We assume it is roughly split in half between the hinge and HRM.

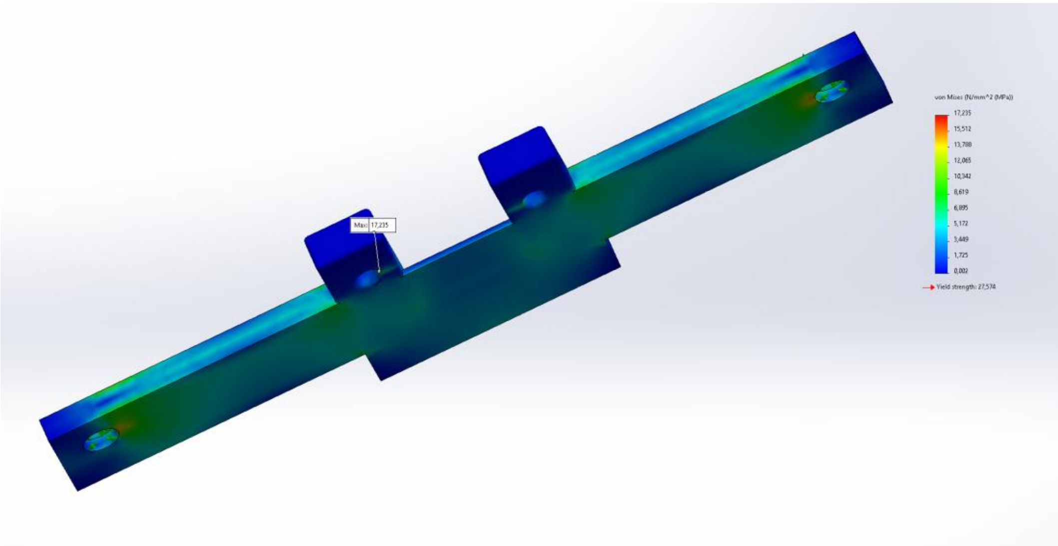


Figure 9-4: Simple FEA

The simplified case in the FEA gives us a max stress of 17MPa where the line goes our of the hole. This is well below the yield strength of aluminium and magnesium which are the lightest available metals. It is still important that the part can be produced with the exact dimensions and surface finish for the job.

For the glued version of the HRM, this indicates that some plastics may be used instead of metals, allowing for a strong and lasting bonding of the wire to the HRM.

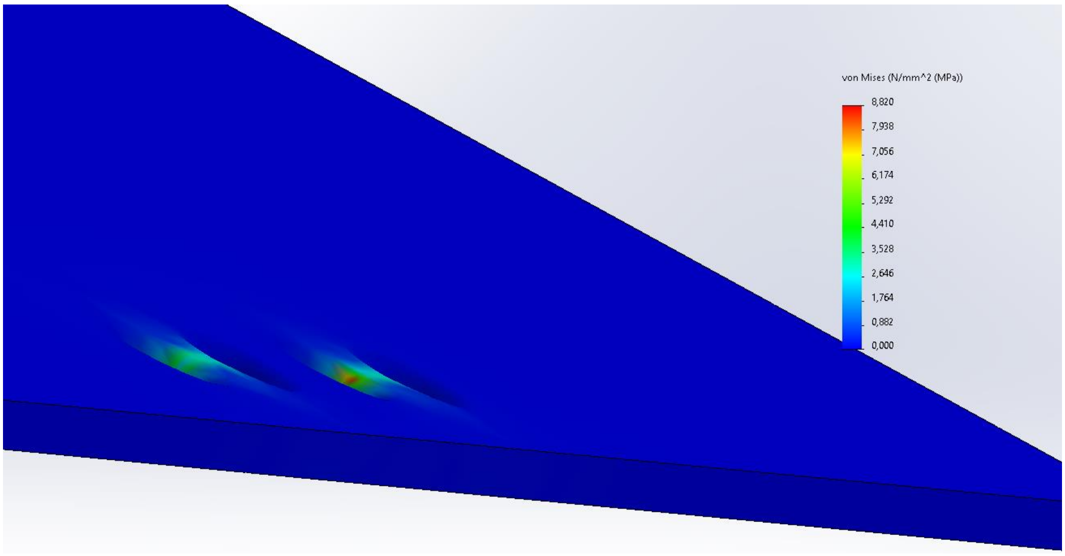


Figure 9-5: Panel holes FEA

Doing the same for the panel holes, we also find low stresses. (9MPa).

9.4 Hole and shaft (Fits)

For the hole and shaft, a free running fit can be used as accuracy is not essential and we have large temperature variations. A H9/d9 fit is recommended by the ISO 286-1 (2010) and ANSI B4.2-1978 standards.

Grade	Fits	
H9	+20	0
d9	-20	-45

Table 9-4: ISO tolerance for hole and shaft

The hole is designed to have a diameter of 2.5mm. The shaft has an upper deviation of -20 and a lower deviation of -45. This denotes a shaft size between 2.48 and 2.455. To determine which size to go for, the hole and shaft are controlled for thermal expansion.

The thermal expansion formula is as follows:

$$\Delta d = \alpha * \Delta T * d$$

$$\alpha = linear\ expansion\ coefficient, 11.5 * 10^{-6} for\ steel$$

$$\Delta T = temperature\ difference$$

The thermal expansion is controlled for the worst-case scenario; maximum shrinking (crimping) of the hole and maximum expansion of the shaft. The changes in diameter is checked for. The temperature range in a LEO orbit ranges from -150 to +120 [ref]

Change in diameter Δd after maximum crimping of the hole is $= 11{,}5 * 10^{-6} * (150 + 15) * 2{,}5 = 0.00474$

Change in diameter Δd after maximum expansion of the shaft is = $= 11{,}5 * 10^{-6} * (120 - 15) * 2{,}48 = 0.003$

Total change in diameter after the thermal expansion of the two parts is:

$0.00474 + 0.00300 = 0.00774mm \ < 0.02mm$

This means the minimum clearance between the hole and shaft at 2.48 is applicable.

In conclusion, the dimensions for the parts are:

Hole: 2.5mm

Shaft: 2.48mm

9.5 Wire.

As stated in section 8.4, monofilament lines will be used for the HRM mechanism. Monofilament wires are made from extruding melted polymers. The polymer used for manufacturing these lines is PA6(nylon 6). These lines have a high melting point of 220 degrees Celsius, optimal in low earth orbit environment.

The maximum tensile strength of the monofilament lines is based on their thread diameter size. The strength for various diameter is given under the table below.

DIAMETER	Breaking limit
0.20mm	3,0 kg
0.25mm	4,5kg
0.30mm	5,4kg
0.40mm	10,2kg
0.60mm	22,4kg
0.80mm	30,1kg

Table 9-5: Thread size chart

To determine which thread diameter should be used, the loads faced are analysed. The dimensioning load is the launch accelerations. From the CDS, the maximum load faced will be 15G*1.25 (qualification levels given by the launch provider) For a 2U panel and all its components weighing 100g, this denotes a maximum weight acting on the thread of;

$$15g * 1{,}25 * 0{,}10kg/g \ = 1{,}875 \ kg$$

As the threads are to be fastened around the mechanism using a knot, we also had to consider the loss of strength due to the knot. Typical knot strengths lie around 66 percent for a three-turn surgeon knot. [22]. To handle the loads of the panel, a minimum tensile strength of $x * (0.66) > 1{,}875, x = 2{,}84kg$ is needed.

Theoretically, the minimum thread diameter of 3kg is strong enough to hold the panels fastened. However, due to oscillating accelerations, we suspect that the wire might get fatigued, and therefore susceptible to break at a lower stress. Therefore, a fatigue behaviour analysis has been carried out on the nylon using the Palm Gren-miner method based on the axial sine frequency environment from section 5.

As long-life fatigue strength data is not generally available for polymers, a SN Curve of the same material our wire is made up from is used. We assume that the values in the graph would be similar to that of our thread and can thereby be used instead. The graph below shows a fatigue limit of around 18MPa.

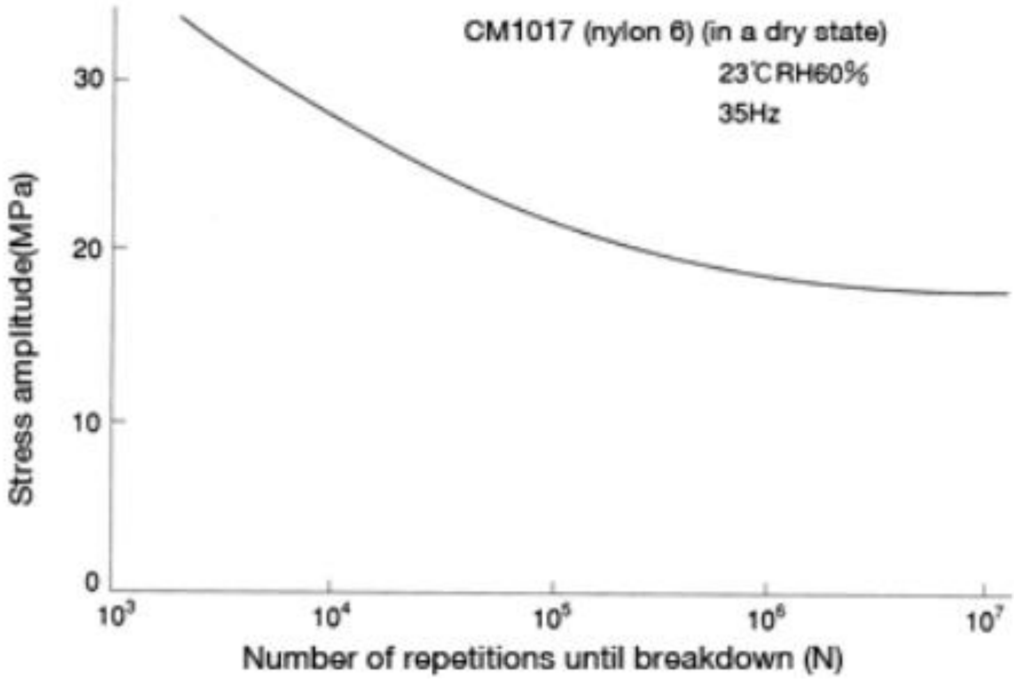


Figure 9-6: SN curve of nylon 6 [23]

Frequencies at their different accelerations are given as:

Frequency(Hz)	Acceleration(g)
5	0.5
20	0.8
35	0.8
35	0.6
75	0.6
85	0.9
100	0.9

Table 9-6: Axial sine environment data from launch provider

Number of cycles for each frequency at their respective accelerations can thereby be found as shown below. The frequency is multiplied the launch duration of 9 minutes.

Acceleration(g)	Total number of cycles
0.5	5*60*9=2700
0.6	35*60*9= 18900 75*60*9=+40500
0.8	20*60*9= 10800 35*60*9=+18900
0.9	85*60*9= 45900 100*60*9=+54000

Table 9-7: Total number of cycles

To find the maximum stress on the different thread diameters, $\frac{F}{A}$, The force at the given accelerations is found by multiplying the acceleration with the weight of the panel.

Acceleration(g)	Force(N)
0.5	0.5*9.81*0.1=0.4905
0.6	0.6*9.81*0.1=0.5886
0.8	0.8*9.81*0.1=0.7848
0.9	0.9*9.81*0.1=0.8829

Table 9-8: Axial force acting on wire.

For the thread diameter size 0.2mm of area $\pi\left(\frac{0.2mm}{2}\right)^2$, the stresses on the thread are as follow.

Acceleration	Stress
0.5	15.8
0.6	19.0
0.8	25.3
0.9	28.5

Table 9-9: Stresses for 0.2mm thread

For the thread diameter size 0.25mm of area $\pi\left(\frac{0.25mm}{2}\right)^2$, the stresses on the thread are as follow.

Acceleration	Stress
0.5	10.01
0.6	12.01
0.8	16.01
0.9	18.02

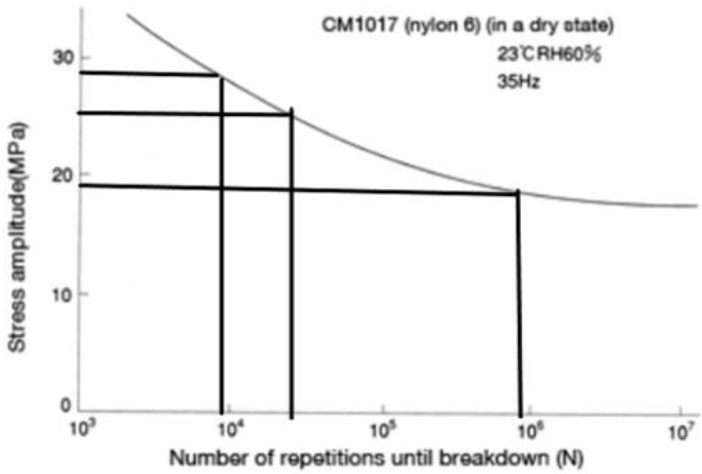
Table 9-10: Stresses for 0.25mm thread size

As the other lines of diameter above 0.25mm have stresses below the fatigue limit, they will be safe from fatigue.

To determine if it would fail from fatigue, we will use the Palmgren miner method. The Palmgren – Miner rule states that failure occurs when $\sum_{n=1}^{I} \frac{ni}{Ni} = 1$ where ni is the number of applied load cycles of type i, and Ni is the pertinent fatigue life.

To determine the if there is failure for our 0.2mm wire. The equation above is solved based on the values found in the tables over.

Acceleration	Stress	Number of cycles
0.6	19	59400
0.8	25.3	29700
0.9	28.5	99900



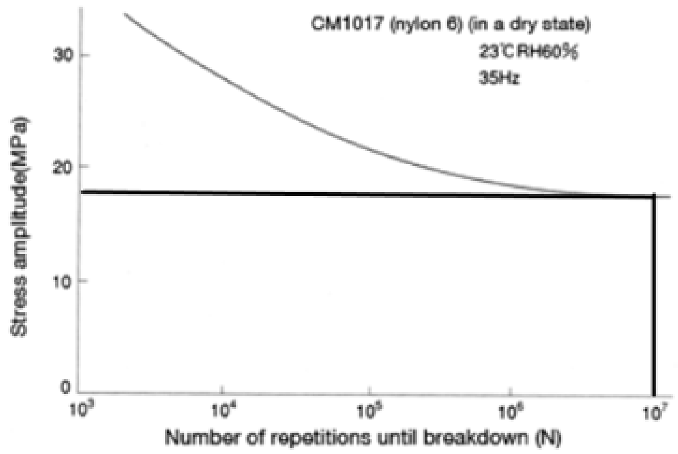
Inserted in the Palmgren-Miner equation

$$\sum_{n=1}^{I} \frac{ni}{Ni} = \frac{59400}{10^6} + \frac{29700}{4*10^4} + \frac{99900}{10^4} = 10.79 > 1$$

According to the result, we would experience fatigue failure with a wire of diameter 0.2mm

For the wire of diameter 0.25mm. We have the following values.

Acceleration	Stress	Number of cycles
0.9	18.01	99900



$$\sum_{n=1}^{I} \frac{ni}{Ni} = \frac{99900}{10^7} = 0.0099 < 1$$

As for the wire of diameter 0.25mm, fatigue will not occur as the accumulated damage is below the limit.

In conclusion, the thread size of 0.25mm will be strong enough to hold the panels, as its breaking limit of 4kg is larger than the required 2,84kg. However, a thread size of 0.30mm would be recommended for a stronger safety net, while still being able to be knotted tightly around the panel.

10 Evaluation and conclusion

We chose this project because it sounded interesting. It has been. We have experienced how the design of parts in a system is an iterative process and how our design changed drastically when we did not expect it to change more. Many ideas came towards the end of the project, and we wish we had time to execute and go deeper into technicalities instead of spending time on unfruitful undertakings in the start and middle of the work period. But this is an experience in itself. Creativity comes when one has good knowledge of possibilities, is motivated and enjoy finding and improving solutions. It takes time to build up competence in a new field, especially when there is a pandemic, and academic, organizational and community support was limited.

In hindsight we could have been more pro-active with getting requirements, wishes, specifications as well as technical support and insights from Orbit NTNU. We would have liked them to be more involved throughout. We were aware that this is a very important step at the beginning of the project. However, as both parties are students lacking in project experience, we might not have been accustomed enough in our roles as client and engineers. Mainly though, it is the Covid restrictions that caused this. We believe it would have been very different pre-Covid. Social distancing also meant academic distancing in this case, even if we had digital communications.

The project has been almost solely based on computer research and design. A little prototyping was done with 3D printing. This prototyping was very useful. Even the actual miniscule size of the parts was not apparent to us before seeing the prototype. Seeing CAD parts on a computer screen gives no perspective. Redesign ensued shortly after our first prototype.

We have learned a lot about the process of design. Our system has many interconnected design factors. One example is choosing a strong torsion spring to ensure separation of the HRM. The intent behind this was to keep the HRM simple without any springs. It however took us down the road of chapter 9.1 and prompted a full redesign of the hinge. Designing the HRM with coil springs were trivial in comparison.

Nonetheless, the torsion spring design and hinge redesign was perhaps the most rewarding part of the project. To decide the dimensions for our torsion spring, we made separate equations containing our two main requirements. We used these equations to plot the relation between our three unknown quantities as intersecting planes in three-dimensional space. We are not sure if our method can be seen as elegant, or if a simpler solution makes it unnecessarily elaborate. It was anyhow one of the more rewarding challenges of this project, and we managed to reproduce data that matched that of spring manufacturers catalogues when using this method.

We can see how it could be easier to have springs in the HRM than to make the torsion spring ensure separation. We could have avoided a convoluted hinge and torsion spring design and the acquiring of custom springs. It would make the manufacture of the HRM more complex, but we would also achieve a gentler motion of the panel. We have not investigated what kinds of impact speed the panel can withstand, but this is very dependant on what type of panel is used. We leave the decision to Orbit NTNU when the system is developed as a whole. Maybe a simple undampened hinge with a weak torsion spring, a simple spring-less HRM, and a surgeon’s knot is enough. Otherwise, we have presented possible solutions to challenges that might arise in this document.

We are happy with our design proposal. If anything, we would have liked to explore and elaborate further on the technicalities of a glue-wire based HRM.

We conclude that our main objective was fulfilled. We have presented different solutions and made a proposed example solution.

An even more exciting continuation of the design will be left over to Orbit NTNU. We leave it to them to look into which design is most feasible to produce a fully working prototype of, and to start testing.

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