Mechanical Design and Development of a Modular Drag Sail for the CanX-7 Nanosatellite Mission

by

Jesse Hiemstra

A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
Graduate Department of Aerospace Science and Engineering
University of Toronto

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2014

Abstract

The drag sail developed for the CanX-7 nanosatellite mission is an aerodynamic deorbiting device suited to small spacecraft. The drag sail enables spacecraft within a range of bus and orbit properties to comply with international regulations that require newly-launched satellites to be removed from low Earth orbit after their end of mission, with the intent of reducing the rate of orbital debris formation. The overall design uses a modular approach that enables integration with two existing spacecraft busses, and also allows for distributed mounting on notional busses yet to be designed. Additive fabrication allows for complex part geometry that enables the drag sail components to be compactly stowed within a small volume. A model of the deployment mechanism dynamics in terms of basic design parameters reproduces a simplified picture of the operation of the real system at room temperature.
Acknowledgments

The development of the CanX-7 mission is a collaborative effort in which I have had the honour and privilege of being granted a creative role. I thank UTIAS-SFL founder and administrator Dr. Robert E. Zee for taking a chance by admitting a candidate to an institution at which I am outclassed by any objective figure of merit. I also thank project manager Grant Bonin for his incredible patience and understanding over the difficult course of the mission thus far, and which is only outshone by the inspiration he instills in his capacity as a friend and mentor.

Others at SFL played an essential role in the CanX-7 experience: I thank fellow students Barbara Shmuel, Vince Tarantini, Thomas Sears, Brad Cotten, and John Chung; all of whom were integral participants in the drag sail design and development process. I hope to someday have the opportunity to help each of you as much as you have helped me. SFL facilities manager Chris Coggon provided extensive support during practical day-to-day lab activities, and is unquestionably the person at the lab with the most interesting and unexpected perspectives on a huge variety of topics.

Free agents on the periphery of the SFL ecosystem provided assistance far out of proportion to the scope of their formal involvement. Doug Sinclair was never without an idea (or twelve) when I needed one. Kieran Carroll provided essential early support by providing background on deployable sails.

Most importantly I thank my parents, family, loved ones, teachers, and all others who have invested in me, believing (rightly or wrongly) that I might accomplish something of value, and thereby propagate something of themselves forward in time.

Finally I acknowledge that the opportunities I experience come at the cost of the systematic deprivation of uncountable numbers of other human and non-human entities as well as living and non-living systems by the dominant cultures and forces of history; in the name of competition, natural selection, manifest destiny, and all of the other processes and ideologies used to justify the exploitation and extermination of that which is not itself. While it is not popular to do so, I feel that it is important to emphasize the ugliness and despair and inequity and ignorance that we strive to transform but so infrequently discuss in our affluent public lives. I submit that our collective priorities are misplaced, and that if we were not so comfortable, we would better understand the gap between what we claim we are trying to accomplish as sentient beings, and the reality of what we inflict upon the universe.
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<td>$a$</td>
<td>Acceleration</td>
</tr>
<tr>
<td>$a_{1\ldots 5}$</td>
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<td>$A$</td>
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<td>$d_r$</td>
<td>Bore, roller assembly bearing</td>
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<td>Bending stiffness</td>
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<td>$F_{BS}$</td>
<td>Force due to boom and sail</td>
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<tr>
<td>$m$</td>
<td>Mass, general or of spacecraft</td>
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<td>$m_{booms}$</td>
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<td>$M^*$</td>
<td>Steady end moment</td>
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<td>Opposite-sense steady end moment</td>
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<td>$n_b$</td>
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<td>$n_l$</td>
<td>Number of layers of paired tapes</td>
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<tr>
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<td>Thickness of an individual tape spring</td>
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<td>Outer radius of coil</td>
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<tr>
<td>$r_{med}$</td>
<td>Median coil layer radius</td>
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<td>$r_r$</td>
<td>Radius of roller</td>
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<tr>
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<tr>
<td>$t$</td>
<td>Time</td>
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<td>Temperature</td>
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<td>$v$</td>
<td>Speed</td>
</tr>
<tr>
<td>$x$</td>
<td>Deployed length</td>
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<td>$\alpha$</td>
<td>Angular acceleration</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Angular position</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>Boom mass per length</td>
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<tr>
<td>$\mu_a$</td>
<td>Coefficient of friction, axle bearing</td>
</tr>
<tr>
<td>$\mu_r$</td>
<td>Coefficient of friction, roller assembly bearing</td>
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<td>Description</td>
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<td>-------------</td>
</tr>
<tr>
<td>2U</td>
<td>2-Unit cubesat</td>
</tr>
<tr>
<td>3U</td>
<td>3-Unit cubesat</td>
</tr>
<tr>
<td>ACA</td>
<td>Active Collision Avoidance</td>
</tr>
<tr>
<td>ADR</td>
<td>Active Debris Removal</td>
</tr>
<tr>
<td>ADS-B</td>
<td>Automatic Dependant Surveillance-Broadcast</td>
</tr>
<tr>
<td>AO</td>
<td>Atomic Oxygen</td>
</tr>
<tr>
<td>COTS</td>
<td>Commercial Off-The-Shelf</td>
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<tr>
<td>CanX</td>
<td>Canadian Advanced Nanospace eXperiment</td>
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<tr>
<td>CuBe</td>
<td>Copper-Beryllium</td>
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<tr>
<td>DLR</td>
<td>German Aerospace Center – Braunschweig, Germany</td>
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<tr>
<td>EM</td>
<td>Engineering Model</td>
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<tr>
<td>GNB</td>
<td>Generic Nanosatellite Bus</td>
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<td>GSE</td>
<td>Ground Support Equipment</td>
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<td>IADC</td>
<td>Inter-Agency Space Debris Coordination Committee</td>
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<tr>
<td>ICD</td>
<td>Interface Control Document</td>
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<tr>
<td>LEO</td>
<td>Low Earth Orbit</td>
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<tr>
<td>MS</td>
<td>Margin of Safety</td>
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<td>Mk</td>
<td>Mark</td>
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<tr>
<td>NEMO</td>
<td>Nanosatellite for Earth Monitoring and Observation</td>
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<tr>
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<td>Nanosatellite Launch Service</td>
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<td>Post-Mission Disposal</td>
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<td>Qualification Model</td>
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<td>SLS</td>
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<td>Triangular Rollable And Collapsible</td>
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<td>University of Toronto Institute for Aerospace Studies</td>
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<td>UV</td>
<td>Ultraviolet</td>
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Chapter 1

Introduction, Scope, Background, and Overview of the Work

The term *orbital debris* refers to all human-made objects in Earth orbit that are not functioning spacecraft [1]. Among other sources, orbital debris arises increasingly from random collisions between existing debris, existing spacecraft, or both [2]. Theoretical predictions [3] and real-world evidence [2] show that beyond a critical density, debris from random collisions comes to dominate the population in a slow but accelerating cascade, and that regions of low-Earth orbit have already passed that threshold. If nothing is done to stabilize or reduce the debris population, the situation in low Earth orbit may, at some difficult-to-predict time in the future, deteriorate well past the fuzzy boundary beyond which remediation to a usable state becomes impractical [4] [5].

The orbital debris problem is complex, and becomes increasingly difficult to solve the longer it goes unaddressed [6]. At the very least it seems necessary to halt the behaviour that led to the problem in the first place, which is the practice of abandoning defunct spacecraft in orbit after their end of mission. Instead, new spacecraft should be removed from orbit by an appropriate means and within an appropriate timeframe, where the definition of “appropriate” is subject to debate.

In the meantime, the Inter-Agency Space Debris Coordination Committee recommends that spacecraft be removed from low-Earth orbit within 25 years of their end of mission [7]. Increasing enforcement of these guidelines by government agencies necessitates a means of compliance for actors in low Earth orbit, including the Space Flight Laboratory (SFL) at the
University of Toronto Institute for Aerospace Studies (UTIAS). Therefore, the mission of the seventh Canadian Advanced Nanospace eXperiment (CanX-7) is to develop a deorbiting technology suitable for spacecraft designed and built at UTIAS-SFL that would not otherwise deorbit within this timeframe.

Using a nanosatellite platform, CanX-7 will demonstrate the practicality of a modular aerodynamic drag sail as a deorbiting solution for a range of operational small satellite missions. The design takes a multi-module approach that allows for flexible bus integration, occupies a minority of the spacecraft bus volume when stowed, and employs stored strain energy for deployment. The resulting sail has an area of 4.2 m², and is designed to be entirely passive in operation. This thesis presents the mission requirements, design evolution, and qualification-model payload that arose over the course of the spacecraft development process.

1.1 Scope and purpose of the work

Generally, this thesis discusses the deorbiting payload onboard the CanX-7 nanosatellite. Specifically, it presents the author’s contribution to the design and development of the payload, covering a span of time beginning from the final stages of conceptual design up until a point just before the commencement of qualification testing. This thesis first describes the requirements and most aspects of the preliminary design, and then focuses in particular on the detailed design, development, and analysis of the deployment mechanism and sail module structure.

The aim of this work is to create a record that presents the evolution of several topics that encompass a large span of the spacecraft design and development process, and to show why this process has resulted in a payload that is ready for qualification testing. Also, this thesis serves to document the initial goals and ensuing trades that lead to the final design, for the benefit of readers seeking to learn about the evolving field of nanosatellite development. Consequently, this thesis intentionally devotes space to “dead-end” design options, with the goal of illustrating the extent of the problem space that was explored before striking out in the chosen direction.
Other publicly-available works by the CanX-7 team members collectively describe the entire mission, including aspects of the drag sail module that are not covered here. This includes topics such as:

- the deorbiting and attitude analyses that determined the size of sail [8] [9],
- the selection of the material for the drag sail itself, as well as the final boom material [10],
- the detailed design for the drag sail electronics [11], and
- other spacecraft systems and ground support equipment (GSE) [10] [12] [13] [14].

(Publication details for the works cited here may be found in the list of references.)

1.2 Background on the orbital debris problem

This section discusses the orbital debris problem, and the role of the CanX-7 mission among the solutions that have been proposed.

1.2.1 What is orbital debris?

Orbital debris consists of human-made objects in orbit that are not functional spacecraft [1, p. 20]. The conventional classification by origin divides debris into the categories of derelict spacecraft, rocket bodies, mission-related debris, and fragmentation debris. The conventional scheme also divides debris into small, medium, and large categories based on their size and mass:

- Objects large enough to be catalogued and tracked from the ground are typically greater than 10 cm in size, 1 kg in mass, and number in the tens of thousands.
- Largely undetectable but inferred medium-sized objects down to 1 mm in size and 1 mg in mass number in the millions.
- Smaller objects detected by in-situ measurements number in the trillions [1, p. 63].

Human use of space actively contributes to the debris population when spacecraft or launch vehicles shed parts by design or by accident during operation. In Earth orbit, these mission-related debris account for approximately one eighth of the total number of objects large enough
to be catalogued.¹ Spent rocket bodies (such as the upper stages used to loft satellites in the final leg of their journey to orbit) account for another eighth of the total. To date, most spacecraft are not removed from orbit at their end of mission, and non-functional spacecraft account for approximately a quarter of the total objects; outnumbering functional spacecraft by approximately four to one. The remainder of catalogued objects consist of fragmentation debris.

This existing debris population just described continuously grows in number without human intervention as operational spacecraft become disabled, deteriorate due to environmental wear, undergo fragmentation due to explosion (in the case of spent rocket bodies), or undergo fragmentation due to collision. The majority of the catalogued debris population in low Earth orbit (LEO) now consists of fragmentation debris, from which the orbital debris problem arises.

1.2.2 What is the orbital debris problem?

The problem is twofold:

- First, orbital debris has the potential to disable active spacecraft, presenting a hazard to valuable space assets.
- Second, orbital debris grows in number through a cascading process of collision and fragmentation.

Each process feeds the other. Even a minor collision with a tiny fragment of debris can disable an active spacecraft. Uncontrollable disabled spacecraft are more likely to be involved in a collision, and also tend to be replaced with new spacecraft, which then face a greater risk of collision. While the cost to replace them can be significant, disabled spacecraft are harmful for reasons beyond immediate financial loss: They also contribute to the ongoing collisional cascade.

Debris in low Earth orbit are continuously undergoing a process of collision and fragmentation that is fundamentally similar to the one currently underway in the asteroid belt, and mostly completed in the rings of the gas giant planets [2]. Evidence of this process can be seen in the

¹ Using 1994 data [1].
steadily rising number of fragmentation debris catalogued since the beginning of the space age, which has grown at a sixty-year-average rate of approximately 300 objects per year.

Even with no new launches, the size of the present population of debris will only trend upward. In LEO, atmospheric drag removes the smallest fragments generated by collisions. However, the present population of large fragments alone is sufficient to create smaller fragments faster than they are removed by drag, resulting in a climbing population of small fragments that will persist for quite some time until the large fragments are depleted. This is the “Kessler syndrome,” the result of a cascading process in which debris produced by random collisions has come to dominate the number of catalogued objects in orbit [15].

The debris cascade is ongoing, and early predictions of its outcome remarkably match the current reality. In 1978, Kessler and Cour-Palais predicted that collisions between catalogued objects would become the dominant source of fragmentation debris around the year 2000 [3], a prognostication startlingly confirmed by the catastrophic Iridium-Cosmos collision in 2009 [16]. However, the cumulative number of lesser-known observed collisions has been following this trend for some time now, and closely matches what Kessler et al originally predicted would result from the eventual real-world debris growth-rate [2].

1.2.3 Why is there an orbital debris problem?

The debris problem exists predominantly as a consequence of objects not being removed from orbit after the end of their operational life. Roughly the first five decades of spaceflight operated under the “big sky theory,” in which collisions between the initially small number of objects in orbit were considered to be vanishingly improbable [17] [18]. Without an enforced impetus to remove them, abandoned satellites have accumulated in orbit ever since, and now outnumber active ones by an increasing fraction. Consequently, observable random collisions are no longer without precedent.

At a higher level of abstraction, the behaviour and consequences that lead to and result from the abandonment of spacecraft in orbit is an example of the tragedy of the unregulated commons.

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2 Note that under “business as usual” conditions, the process is exacerbated when fragmented satellites are replaced with new ones.
This phrase describes phenomena in which rational self-interested humans act contrary to the best interests of the whole population, and in so doing, deny themselves the use of a shared resource in the future by their actions in the present. Depending on the nature of the system in question, the mechanism for future denial may be simple depletion, (e.g. in the case of a non-renewable resource,) or arise from the effects of forcing a high-latency feedback loop (e.g. in the case of a system otherwise in equilibrium over certain time scales, perhaps longer than the typical career length of an engineer or bureaucrat). It should be unsurprising that a tragedy of the commons is underway in low Earth orbit; historically such phenomena often manifest on the frontiers of human expansion [19] [20] [21].

1.2.4 What can be done about orbital debris?

Proposed solutions to the orbital debris problem fall into three categories:

- Active Collision Avoidance (ACA)
- Post-Mission Disposal (PMD)
- Active Debris Removal (ADR)

Spacecraft operators today practice active collision avoidance (ACA) to some extent by performing conjunction analyses using software tools such as SOCRATES [22], and performing debris-avoidance maneuvers when practical. Uncertainty in orbit tracking and propagation limits the present utility of ACA, and it cannot prevent collision and fragmentation between existing debris.

If it were exercised from the beginning of the space age, post-mission disposal (PMD) would have been sufficient to prevent the formation of all debris except that generated by accidents and explosions. Given the density of the debris population today, however, researchers have shown that PMD alone would still result in a steady increase in the number of catalogued objects [2, p. 13] [23].

Active debris removal (ADR) is an intuitive solution to the orbital debris problem. Assuming that the enormous total number of objects precludes the removal of each and every one, a more practical approach is to remove the much smaller number of objects that present the largest potential to generate debris. Liou et al predict that the removal of only the five most-threatening
objects per year, combined with aggressive post-mission disposal, would be sufficient to cap the debris population at its present size [5].

Once begun, if the expensive practice of ADR is to eventually become unnecessary, then the population of extant spacecraft must be limited as well. Kessler et al have shown that for certain regions of LEO there exists a critical maximum number of spacecraft that may be sustained indefinitely, beyond which the fragmentation process tends to produce an infinite number of objects [2]. Below this number, the debris population will either diminish, or reach equilibrium. This critical density limit has already been exceeded in large swaths of LEO; therefore, limits on the number of spacecraft in those regions are necessary to cap the eventual debris population and prevent them from becoming eventually unusable [24].

1.2.5 What is currently being done about orbital debris?

At the moment, worldwide orbital debris mitigation efforts are incomplete:

- Actors in LEO are only incompletely practicing ACA and PMD.
- No ADR technologies are operational.
- Limits on the number of extant spacecraft have not been set or enforced.

The intuitive solution to the orbital debris problem is to remove orbital debris. An ideal solution would be to:

- dispose of new spacecraft after their end of mission,
- impose limits on the number of functioning spacecraft in critical regions,
- remove the most dangerous objects in the orbital debris population, and
- remEDIATE the low Earth orbit environment to something closer to its pre-spaceflight state, or at least below the critical cascade threshold.

In the meantime, the governments of space-faring nations appear willing to enforce the guidelines published by the Inter Agency Space Debris Coordination Committee (IADC) that limit the generation of mission-related debris, and require post-mission disposal of new spacecraft. To that end, the drag sail developed over the course of the CanX-7 mission will enable the UTIAS Space Flight Laboratory (SFL) to meet this requirement for the lab’s own missions as well as those of partners, collaborators, and clients.
1.3 Overview of the CanX-7 mission

CanX-7 is an experimental mission to demonstrate a practical deorbiting solution for small satellites. The spacecraft, depicted in Figure 1, carries three payloads:

- a deployable aerodynamic drag sail,
- a sail inspection camera, and
- an experimental radio receiver to collect signals from aircraft.

Engineers at SFL will operate these payloads over the course of the mission, during which telemetry and tracking data will verify the effectiveness of the drag sail. The following two subsections describe the payloads and their operation sequence.

![Figure 1: Illustration of the CanX-7 nanosatellite.](image)

1.3.1 Payloads

The deployable drag sail is the primary payload onboard CanX-7. It consists of four identical sail modules, each of which deploy an independent sail segment with an area of 1.05 m². In each module, two tape springs (similar in shape to everyday tape-measures or carpenter tapes) deploy the sail segment stowed within using stored strain energy, and form cantilevered booms that support it. The remaining chapters in this thesis discuss the primary payload extensively.

The secondary payload is a radio receiver that collects traffic-control signals broadcast by certain aircraft. This experimental payload will be one of the first to demonstrate the feasibility of
receiving these Automatic Dependant Surveillance-Broadcast (ADS-B) signals from low-earth orbit, which are normally intended for reception by airborne and terrestrial receivers. The transmissions encode the position, heading, and velocity of the broadcasting aircraft, among other information. Orbital ADS-B data collection has applications in improved traffic management and surveillance in presently unmonitored airspace above the oceans and high arctic. The ADS-B receiver secondary payload is designed and built at the Royal Military College of Canada [25].

The tertiary payload is an inspection camera that will image the deployed drag sail. It is a compact boom-mounted device incorporating three miniscule commercial off-the-shelf (COTS) chip-scale-packaged wafer-level-construction image sensors operating in the visible band. The intent of the inspection camera is to supplement telemetry from the drag sail modules by providing conclusive photographic evidence of successful deployment. Work by Sears [10] describes the inspection camera in detail, which is designed and built at SFL.

1.3.2 Mission operations sequence

The operational phase of the CanX-7 mission will span at least a year, with the spacecraft itself remaining in orbit for much longer.

First, a launch vehicle will place the CanX-7 spacecraft into a polar orbit as a secondary payload brokered through SFL’s Nanosatellite Launch Service (NLS). The spacecraft is designed to operate in a range of orbits, and NLS maintains partnerships with several launch service providers; therefore, the specific orbit and launch vehicle operator will become known closer to the completion of mission development. A flight-proven XPOD\(^3\) separation system designed and built at SFL will eject CanX-7 from the launch vehicle, whereupon it will automatically power-up and await contact from the ground.

After separation, spacecraft operators will establish contact with CanX-7 from the SFL ground station in Toronto. The operators will conduct checkout and commissioning activities over a period of days to weeks to establish reliable and consistent operation of the onboard subsystems.

\(^3\) eXoadaptable PyrOless Deployer
After commissioning, the spacecraft will host operations of the secondary payload for a period of six months. During this phase, the active attitude control system will favourably orient CanX-7 to collect data from aircraft beneath its ground track using the ADS-B receiver payload.

After the conclusion of secondary payload operations, engineers will command the drag sails to deploy one-by-one. The sail modules will capture telemetry during deployment, and the sail inspection camera will capture images of each deployed segment in its field of view. The onboard computer will collect and store this telemetry to be later relayed to the ground by the communication subsystem.

A measurable change in the spacecraft orbit should become clearly detectable within one month after deployment. Publicly-available tracking data from ground-based radars will provide up-to-date orbital elements in order to verify that the sail is causing the spacecraft to de-orbit at the expected rate. Depending on solar activity and its initial orbit (which will only be known precisely once manifested on a launch) CanX-7 will likely re-enter the atmosphere within five to ten years [9].

1.4 Overview of the CanX-7 spacecraft

CanX-7 is a small spacecraft compared to most examples from the six decades of spaceflight that precede it. The spacecraft bus has the shape of a rectangular prism measuring 34 cm long with a square cross-section measuring 10 cm on a side. The spacecraft subsystems are all designed and built in-house at SFL:

- The modular power subsystem incorporates photovoltaic cells for power generation in a parallel-regulated direct energy transfer topology.
- An onboard computer performs command and data handling tasks supporting all spacecraft subsystems.
- The communication subsystem employs an always-on UHF band\(^4\) command uplink receiver and an S band\(^5\) downlink transmitter.

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\(^4\) Ultra High Frequency, 300 MHz to 3 GHz

\(^5\) 2 GHz to 4 GHz
• Four spring-loaded antennas, as well as an instrument boom containing the sail inspection camera and three-axis magnetometer, deploy upon ejection from the launch vehicle separation system.
• In combination, the magnetometer and solar panels (which allow for coarse sun-sensing) provide three-axis attitude knowledge.
• Three vacuum-core magnetorquers provide two-axis attitude control.
• Passive thermal control surfaces maintain all components of the spacecraft within their survival temperature range in all attitudes.

The majority of these subsystems use flight-proven unit designs developed incrementally over the course of several missions and nearly a dozen operational spacecraft to date. Work by Singrayar [13], Tarantini [9], Cotten [12], Chung [14], and Sears [10] discuss their implementation on CanX-7.

1.5 Comparable missions

CanX-7 is just one mission among several to develop deployable sails for small spacecraft. Aerodynamic drag sails are a straightforward method for deorbiting spacecraft in low Earth orbit, and experimental development is currently popular in the small satellite community. There are now several more examples of deployable sails at the nanosatellite scale than when the mission was first proposed in 2009. Similar to CanX-7, most of the following examples presented in this section employ a bus measuring 10 cm x 10 cm x 30 cm, referred to as a 3U bus, according to published standards in which a single unit (U) measures 10 cm x 10 cm x 10 cm.

Broadly speaking, deployable sails can be classified according to the method by which they support the sail membrane. Examples from two classes are discussed here: Tensegrity sails derive stiffness from a balance of tension and compression forces in separate members, achieved by forcibly extending stiff booms against taught sails. In contrast, cantilevered sails derive stiffness from the shape and material properties of their booms, which are loaded in bending; the sails, for the most part, are slack. Most sails with an aim toward eventual functionality as solar sails employ the tensegrity approach due to the need to tightly stretch a reflective membrane. Those that function only as drag sails (also called aerobrakes in some literature) often employ the simpler cantilevered approach.
Nanosail D-2 deployed the first sail from a cubesat platform on 20 January 2011 [26]. The sail, intended to demonstrate technologies developed for solar sailing, occupied the majority of a 3U bus. The four segments of its square tensegrity sail were deployed from a centralized mechanism by means of rolled metallic booms with a Y-shaped cross-section.

RAIKO incorporates the first purpose-specific drag sail to be placed on orbit, and was released from the International Space Station on 4 October 2012 [27]. Among other payloads, the 2U cubesat incorporates a sail module that deploys a single-segment square cantilevered sail measuring 0.5 m on a side [28]. The sail module employs a centralized circumferential tape spring deployment mechanism similar to the one considered during the conceptual design phase of CanX-7 (See Chapter 3, section 3.1.3.1).

SPROUT incorporates a single-segment triangular cantilevered sail deployed by means of inflatable polymer tubes [29]. The sail appears to employ a variant of the Miura fold, which has long been proposed for stowing thin membranes on spacecraft [30]. The spacecraft was launched in May 2014 [31].

Lightsail-1 is the latest spacecraft in the Planetary Society’s solar sailing program [32]. The sail, booms, and deployment mechanism of the 3U cubesat are similar to those of Nanosail-D2. The spacecraft has yet to be launched at the time of writing.

CubeSail will deploy a four-segment square tensegrity sail measuring 5.0 m on a side and occupying the majority of a 3U bus. It will use a novel sail actuation platform to demonstrate solar sailing [33]. The centralized deployment concept is similar to that of Nanosail-D2 and Lightsail-1. The metallic lenticular cross-section booms used on this spacecraft inspired those in the centralized deployment mechanism considered during the conceptual design phase for CanX-7. Later, the same booms provided the inspiration for the custom-built thermo-formed copper-beryllium alloy tape springs eventually employed in the final design on CanX-7, after moving away from commercial off-the-shelf steel tape springs. The spacecraft has yet to be launched at the time of writing.

Deorbitsail has a similar overall design to CubeSail, but will deploy its sail using booms formed from carbon-fibre reinforced polymer [34]. The spacecraft has yet to be launched at the time of writing.
AEOLDOS is a modular drag sail product commercially marketed by the Clyde Space company, associated with the University of Glasgow. Like that on CanX-7, the modular drag sail employs metallic tape springs, sail cartridges, and occupies a minor fraction of a 3U cubesat bus. The centralized deployment mechanism deploys four sail segments to form a square tensegrity sail measuring 1.7 m on a side [35]. The device has yet to be placed on orbit at the time of writing.

Aerodynamic drag sails are not the only deorbiting technology suited to small spacecraft, and discussion of the mission requirements in Chapter 2 compares several alternatives. While it has been proposed that drag-based deorbiting devices may not reduce the overall probability of collision for the spacecraft to which they are attached [1, p. 147] [36] [37], if nothing else they represent a tangible example of the collective progression toward a more effective practice of environmental stewardship in Earth orbit.

1.6 Organization of the thesis

The remainder of this thesis is organized as follows:

- Chapter 2 describes the development of the requirements that pertain to the payload.
- Chapter 3 presents the prototyping efforts and evolution of the payload over the course of the conceptual, preliminary, and detailed design phases.
- Chapter 4 presents the final drag sail module design.
- Chapter 5 develops a mathematical model for the deployment mechanism, and compares its predictions to experimental results.
- Chapter 6 revisits the requirements, and presents themes, lessons learned, recommendations, suggestions for follow-on work, and conclusions.


Chapter 2

Design, Functionality, and Performance Requirements

This chapter explores the requirements that have been defined for the CanX-7 mission and their role in the design and development of the drag sail. In microspace design at UTIAS-SFL\(^6\), *requirements* define the goals that a mission must reach in order to achieve success. Careful construction enables requirements to specifically define what all of the mission elements must accomplish, while at the same time not constraining the engineering team unnecessarily. A familiarity with the topic of requirements in general is essential to understanding their role in the design and development process, and how they influence the decisions made over the course of the mission.

Broadly speaking, the functionality and performance requirements for CanX-7 stipulate:

- that the mission shall demonstrate a drag-based deorbiting device,
- that the device shall enable compliance with IADC guidelines, and
- that it shall be suited to a range of spacecraft over a range of orbits.

The requirements themselves refine these statements in detail. In terms of design, the requirements state that the device itself shall be modular, reusable, testable, and activated only by ground command.

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The following sections in this chapter deal with:

- the topic of requirements in general,
- their role in the canonical spacecraft design and development process, and
- the specific requirements pertinent to the CanX-7 drag sail.

### 2.1 Requirements, in general

This section presents a brief overview of requirements in general. Two major topics are presented here: First, well-written requirements have specific qualities that make them capable of objectively defining exactly what a system must achieve. Second, requirements can be organized or categorized in ways that specifically emphasize certain qualities. The following subsections describe these qualities and categorization schemes.

#### 2.1.1 Qualities of good requirements

Good requirements are commonly described as “necessary, sufficient, traceable, and verifiable.” Briefly, this implies the following:

- Each requirement describes a quantity, behaviour, or set of conditions that is strictly necessary to solve the design problem or meet the mission objective. To allow for flexibility, good requirements express *what* is to be done, but not *how* to do it.
- The set of requirements together must be sufficient to define the necessary performance; there can be no missing information.
- Each requirement must be traceable to a higher-level requirement; the highest of which arise from the basic mission goals (typically contained in the mission proposal) or accepted conventions.
- Requirements must be expressed in a way that can be objectively verified by inspection, analysis, test, or demonstration. This often implies that they are expressed quantitatively and with explicit tolerances.
2.1.2 Content of requirements

Categorization by content communicates whether individual requirements affect the general design, functionality, or performance of a system:

- Functional requirements specify what the system must do.
- Performance requirements specify (usually quantitatively) how well the system must accomplish those tasks. The “extent of deployment” requirement for the CanX-7 drag sail is an example discussed later in this chapter.
- Design requirements specify methods, preferred practices, and standards that the team follows during the design and development process. Paraphrased, a requirement for “no expendable parts” is one example of a design requirement for the CanX-7 drag sail.

Categorization by content reveals whether or not a set of requirements together are necessary and sufficient. For example, an overabundance of design requirements compared to functional and performance requirements may imply that the solution space is more constrained than strictly necessary, compared to the space that would exist if all possible design practices were allowed.

2.1.3 Granularity of requirements

Categorization by granularity or scope emphasizes the level of abstraction at which requirements apply to a system. An early step in the design and development of a space mission like CanX-7 is to define requirements at the mission level, system level, and subsystem level:

- Mission level requirements define what the mission as a whole must accomplish.
- System level requirements define what the various components of the system must accomplish in concert.
- Subsystem level requirements define what each subsystem must accomplish in isolation.

Categorization by granularity or scope also reveals the traceability between requirements, the extent to which subsystems interact, and the relative complexity of the system at each level of abstraction.

2.1.4 Strength of requirements

Categorization by strength communicates design priorities and allows for flexibility in the design and development process. It is useful to distinguish between areas where performance is
mandatory, and other areas where performance goals exist, but are recognized to be mutable. Three strengths of requirements are “shall,” “should,” and “will.”

“Shall” implies a strict requirement. These describe non-negotiable characteristics that define the system in question. Though their presence or absence is not negotiable, it can occur that the final system is non-compliant with a “shall” requirement, which might be deemed acceptable under some circumstances if it can be shown that the general intent of the requirement is met. Such instances of non-compliance necessitate careful examination to ensure that the functionality of the system is not compromised.

“Should” implies a desire to achieve a given performance or functionality goal. These stretch goals often serve to capture an envisioned design feature that would be useful or convenient, but as often as not may become difficult or impossible to achieve within the constraints of a mission. Judiciously abandoning these requirements (sometimes playfully called “desirements”) may become appropriate as the design progresses.

The “will” requirements are weaker still, and might be described as implying intent or providing clarification.

Categorization by strength emphasizes which requirements are critical, and which may be sacrificed for reasons of practicality, cost, or schedule.

2.2 The role of requirements in the design and development process

Having discussed the topic of requirements in general in the previous section, this section deals with their role throughout the design and development process. Requirements play changing roles over the course of a mission, starting from the early definition phase, moving to the design phase, and later during integration and testing. The following three subsections discuss these roles in more detail.

2.2.1 Defining requirements

Before the design of a mission begins in full, the engineering team defines the requirements in consultation with the mission stakeholders. The requirements capture the goals that the mission is intended to achieve.
At this stage, the process of “trading on requirements” (a colloquialism peculiar to the field of spacecraft engineering) balances performance against practicality. In one direction, increasing the performance of a system to meet a requirement may result in diminishing returns and increased cost. In the other direction, accepting decreased performance may meet the same goals and be less taxing to affected subsystems. On the other hand however, the resulting system may be less robust and operate with less available margin in the face of uncertain operating conditions.

The specificity and precision that are essential to writing useful requirements can be difficult to achieve without first knowing what information is important, which quantities are appropriate to specify, and what values are realistic. This presents a problem for unfamiliar systems at a low level of technology readiness. If prior experience is lacking, then creating flexible requirements, or refining them iteratively, can allow for adaptation to problems that may result from imperfect definition. For example, an early CanX-7 system-level requirement specified a drag sail area; later it was found that a requirement for ballistic coefficient better captured the design goals. Later sections in this chapter on the specific CanX-7 payload requirements discuss this topic in more detail.

Once the requirements are acceptably mature, a formal review can serve as a useful milestone before moving on to refine the design in greater detail.

### 2.2.2 Designing to requirements

In the majority of the design phase, the engineering team designs a system to meet the relevant set of requirements. The function of requirements in the design process depends on the maturity or technology readiness level of the system or class of system in question – simply put, how well it is understood. At one extreme, the requirements may serve as inputs to flight-validated design calculations. At the other, they may serve as pass/fail criteria for the output of analyses and tests yet to be defined. When the engineering predictions converge to values that meet the requirements, the design is finished and the next phase begins.

### 2.2.3 Verifying compliance with requirements

As the components of the system are assembled, integrated, and tested, the engineering team compares the real-world design, functionality, and performance of the system against the values
set out in the requirements. When the behaviour of the final system matches that specified in the requirements, it is said that the system has been verified\footnote{The different but related process of system validation occurs during the operations phase, and determines whether the system and the requirements it meets are, in practice, appropriate for the end-use.} to meet requirements.

In some cases, the system may not meet all of the relevant requirements. This might occur after emergent behaviour indicates that meeting a requirement is not possible, given eventual real-world constraints. In these cases of late-stage noncompliance, the performance of the system is evaluated against the intent of the higher-level requirements, in some cases all the way up to the mission objectives. For a well-engineered system, it may be possible that the high-level mission objectives can still be met while not meeting specific system- or subsystem-level requirements.

\section*{2.3 Requirements for the CanX-7 drag sail}

Having discussed the role of requirements in the previous section, this section discusses the specific mission-, system-, and subsystem-level requirements that pertain to the drag sail.

- The mission-level requirements deal mostly with the method of deorbiting and the notional future spacecraft for which the sail is designed.
- The system-level requirements for the payload flow from the mission-level requirements. They describe the functionality and performance of the drag sail in the context of the whole spacecraft.
- The subsystem-level requirements flow from both the mission- and system-level requirements. They describe the characteristics of the drag sail system necessary to meet the system requirements, as well as characteristics that serve to enhance the usability of the subsystem in the context of other missions.

The following subsections discuss these requirements in more detail. For the most part, the specific wording and quantitative values set out in the requirements are paraphrased and described here qualitatively; numeric values are given only where they are particularly relevant and do not distract from the discussion at hand.
2.3.1 Requirements at the mission level

For the most part, the mission requirements flow from primarily from two documents. First, the IADC guidelines [7] specify how quickly the deorbiting system must operate. Second, the mission proposal [38] identifies the development of a drag sail as the appropriate first step on the path toward useful deorbiting technology for small spacecraft.

2.3.1.1 Method of deorbiting

The first mission-level requirement encapsulates the basic mission definition and addresses the choice of deorbiting method. The requirement flows from the CanX-7 mission proposal, which approaches the problem of deorbiting small spacecraft as a choice between active and passive methods:

- **Active methods** imply the presence of a feedback system to execute a controlled shift into an orbit with a shorter lifetime. They may involve the application of an internally-powered force to accomplish this goal in a short time.
- **Passive methods** accomplish the same goal without feedback or the continual use of power, and rely on an external force typically acting over a longer period of time.

On one hand, active methods include devices such as reaction engines (e.g. rockets or thrusters), steered solar sails, or powered electrodynamic tethers. Active methods generally require some high level of spacecraft functionality to operate the deorbit mechanism:

- Rockets and solar sails require some ability to properly orient their thrust vector, which may rely on sensors or actuators that are at greater risk of failure toward the end of mission operations.
- Powered electrodynamic tethers or other forms of electric propulsion require power at a time in the spacecraft’s life when its aging batteries and solar arrays will perform their worst. Such engineering challenges can be overcome, but at a cost.

On the other hand, passive methods include devices such as unpowered electrodynamic tethers, and aerodynamic drag devices such as strips, sails, and balloons. These passive methods rely on external forces to continuously remove energy from the spacecraft’s orbit.

The CanX-7 proposal justifies the choice of passive methods as appropriate for nanosatellites by citing the reduced number and functionality of spacecraft subsystems needed to support a passive
deorbiting device, compared to an active one. In principle, the simplest imaginable passive
device would need only the ability to activate once, and a functioning command uplink or
watchdog device to trigger it.

The proposal justifies the choice of a sail by eliminating other drag-based alternatives:

- Tethers have a large characteristic dimension and success has been limited at the cubesat
  scale [39].
- Ribbons involve complex flexible body attitude dynamics in addition to the same
  materials and mechanical challenges that exist for sails [40].
- Balloons are conceptually vulnerable to puncture and deflation in the absence of
  sophisticated materials. Furthermore, the use of pressurized (or otherwise energetic)
  substances for inflation impose non-trivial complications and/or risk to the spacecraft and
  launch vehicle.

For these reasons, the first mission requirement states that CanX-7 “shall demonstrate a passive
drag-sail based deorbiting technology for nanosatellites.” A further requirement goes on to state
that “the spacecraft’s primary payload shall consist of a deployable passive drag sail device.”

2.3.1.2 Unpowered operation

A later mission requirement states that the drag sail shall be capable of meeting the deorbiting
requirements in an unpowered state. This requirement defines the meaning of the “passive”
operation just mentioned.

2.3.1.3 Reference spacecraft

A further set of mission requirements define the characteristics of a “reference spacecraft” for
which the sail is designed to be effective. CanX-7 is much smaller than this notional spacecraft,
and serves as a platform to demonstrate a sail with capabilities suited to spacecraft that are much
larger than itself.
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The reference spacecraft is representative of the NEMO-class\textsuperscript{8} spacecraft designed and built at SFL, and has the following characteristics:

- Mass of 15 kg.
- Starting orbit altitude anywhere in the range of 600 km to 800 km.
- Orbit inclination anywhere in the range of 80° to 100° with an unconstrained right ascension of the ascending node (encompassing a range of sun-synchronous polar orbits).

2.3.2 Requirements at the system level

Several of the CanX-7 system-level requirements involve the drag sail.

2.3.2.1 Spacecraft bus compatibility

An important system-level requirement driving the payload design as well as the bus design specifies that the sail payload shall be compatible with the CanX-2 and GNB-class\textsuperscript{9} busses designed and built at SFL, and shown in the following figure. This requirement arose early in the mission, when the appropriate choice of bus was still unclear. The ability to attach the drag sail to various busses implies modularity of the payload; Chapter 4 deals with this topic in detail.

Figure 2: Left: CanX-2 (foreground), and a GNB-class satellite (background). Right: GNB-class satellite (foreground), and CanX-2 (background).

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\textsuperscript{8} Nanosatellite for Earth Monitoring and Observation

\textsuperscript{9} Generic Nanosatellite Bus
2.3.2.2 Commandability

Another system-level requirement stipulates that triggering deployment of the sail shall only be possible via ground command. The specific concern behind this requirement is that a premature autonomously-commanded deployment has the potential to jeopardize the primary mission of the notional parent spacecraft. The inclusion of this requirement at the system-level implies that other spacecraft systems may be involved in triggering deployment of the sail, namely the communications and power subsystems. This is in contrast to other deorbiting systems that are triggered by a watchdog timer and internally powered.

2.3.2.3 Telemetry

A pair of requirements deal with gathering telemetry at the system level:

- “The system shall gather engineering telemetry that can be used to infer sail deployment.”
- “The system should be capable of determining the coarse percentage of sail deployment, e.g. 0%, 50%, 100%.”

The inclusion of these requirements at the system level allows spacecraft systems other than the payload to contribute to gathering this telemetry, specifically the boom-mounted inspection camera.

2.3.2.4 Mass

A system-level requirement specifies that the total mass of the drag sail payload shall be no greater than 0.5 kg. Including this requirement at the system level allows the mass at the subsystem level to be specified with more granularity.

2.3.2.5 Testing

Two system-level requirements pertain to testing. One requirement specifies that the payload shall be tested during thermal-vacuum loading using a particular standard temperature profile used at SFL, which implies a specific testing pattern and methodology for selecting the temperature extremes. The other requirement specifies that the payload shall be tested at the unit level after vibration loading using a specific vibration spectrum derived from that used to test the
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XPOD\textsuperscript{10} separation system, which envelopes the load levels of a number of potential launch vehicles.

2.3.3 Requirements at the subsystem level

2.3.3.1 Deorbiting performance

The first requirement concerning the payload at the subsystem level addresses its deorbiting performance, and flows from the higher mission-level requirement specifying a maximum allowable deorbit time of 25 years. The requirement states a maximum allowable value for the mean ballistic coefficient that the drag sail shall cause the reference spacecraft to achieve after deployment.

The \textit{ballistic coefficient} is a quantity that describes the tendency of a moving object to overcome air resistance; or as the name suggests, to move like a bullet: On Earth, air resistance causes bullets fired level to slow down while they fall due to gravity. A bullet with a low ballistic coefficient will slow down and impact the ground sooner than a bullet with a high one, all else being equal. On orbit, aerodynamic drag continuously acts in the direction opposite to a spacecraft’s velocity, which gradually decreases its orbital energy and thus altitude. The ballistic coefficient $C_B$ depends on the spacecraft mass $m$, its drag coefficient $C_D$, and its cross-sectional area $A$, and may be expressed by the following formula:

$$C_B = \frac{m}{C_D A} \quad (2-1)$$

Objects with a high ballistic coefficient, such as a bullet or compact spacecraft, tend to have a greater mass, have a lower drag coefficient, (i.e. are more “streamlined,”) and have a smaller cross-sectional area than objects with a low ballistic coefficient, such as a feather or spacecraft equipped with a drag sail.

For an object in a moving fluid, the \textit{drag coefficient} is a dimensionless quantity specific to a particular shape that relates the drag force on an object of that shape to the fluid’s density, relative speed, and the area of the object itself. In everyday experience, the drag coefficient

\textsuperscript{10} eXoadaptable PyrOless Deployer
describes the extent to which an object is blunt; although for spacecraft, which operate in the free molecular flow regime, the relationship between drag coefficient and shape is less intuitive.

With the mass of the reference spacecraft fixed, specifying the ballistic coefficient allows deployment of the sail to increase both its drag coefficient and cross sectional area. Analysis performed by Shmuel [8] determined a required value for the mean ballistic coefficient that will cause the reference spacecraft starting from the highest required altitude (and all lighter spacecraft in lower orbits) to deorbit in 25 years or less. Further analysis performed by Tarantini [9] showed that a tumbling spacecraft incorporating a sail with an area of 4.0 m$^2$ can achieve that required value in an acceptable majority of starting orbits under worst-case conditions. For conservatism, both analyses neglect the contribution of the spacecraft bus to the cross sectional area, and also neglect the likely effect of an increased drag coefficient due to the sail$^{11}$.

A complication arises in the distinction between the planar sail area, and the mean cross sectional area over long periods when tumbling. Analyses and design changes addressed this distinction when it was examined during the preliminary design process. The outcome of this process influenced the design in a critical way discussed in Chapter 3.

2.3.3.2 Stowage life

Flowing from the typical required length of SFL missions, a subsystem-level requirement specifies the required stowage life of the drag sail module. The requirement specifies that the drag sail payload shall be designed to survive and successfully deploy following a three year stowage period at any temperature within the survival range of its constituent parts.

2.3.3.3 Bus interface

Flowing from the system-level bus compatibility requirement, a subsystem-level requirement specifies conformity to an interface control document (ICD) that describes the required mechanical and electrical interfaces. The ICD also specifies the required operating temperature range of $-40 \, ^\circ C$ to $80 \, ^\circ C$, which becomes relevant in Chapter 5.

$^{11}$ The analyses use the typically-accepted value of $C_D = 2.2$. On-orbit evidence suggests the actual average value for spacecraft with a high aspect ratio to be at least 10 percent greater [62].
2.3.3.4 Commandability

Flowing from the system-level requirement that specifies exclusive ground-commandability, a subsystem requirement states that “each drag sail module shall require separate arm and fire commands sent via command uplink to deploy. Deployment shall only be triggered if the commands are sent in the correct sequence.” Chapter 3 discusses the effect of this requirement on the preliminary design of the sail module electronics.

2.3.3.5 Telemetry

Two subsystem-level requirements flow from the higher system-level requirement to infer successful sail deployment, but also have roots in the later stand-alone utility of the drag sail module. The two requirements specify that the payload shall telemeter the motion of its booms, and also whether its door is open or closed. (Obviously, both requirements were written after the conceptual design included both booms and a door.) Together, these two telemetry points alone provide evidence independent of the inspection camera that the sails have deployed successfully. This is useful in the context of the notional reference spacecraft that the sail is eventually intended to serve.

2.3.3.6 Testing

A single payload subsystem requirement pertains to testing, and requires that the sail shall be fully testable under Earth gravity. For the drag sail, this ability provides compelling evidence that the deployment mechanism and structure are mechanically robust, and therefore more likely to function in the lightly-loaded space environment. It also reduces the cost of testing, as microgravity flights are not needed to demonstrate functionality.
2.3.3.7 Reusability

Reusability is an important goal for the drag sail subsystem. Reusability reduces the number of sail modules needed throughout the mission to perform all of the necessary qualification testing, and in principle reduces the mission development cost (provided that the additional effort of designing a reusable module is not onerous). One payload subsystem requirement stipulates that “it shall be possible to stow, arm and deploy each drag sail payload a minimum of 10 times without disassembly or refurbishment.” Three other requirements then flow from this reusability requirement and address details of the re-stowage process:

- A weak “will” requirement sets the goal of a one-hour re-stowage time in a cleanroom environment.
- A stronger requirement states that the drag sail should not use expendable parts. Such parts add recurring cost to each deployment.
- Finally, a “shall” requirement mandates that re-stowage shall not require any disassembly of the spacecraft.

This last requirement precluding disassembly flows from the spacecraft acceptance testing methodology, and is an important design driver. As part of the acceptance testing campaign, the fully-integrated spacecraft undergoes functional tests to verify that it still operates after being loaded in a manner representative of launch. Within this campaign, the vibration and thermal-vacuum tests serve, in part, as workmanship tests intended to verify that the spacecraft is assembled correctly. Disassembling the spacecraft after these tests invalidates the workmanship verification that they achieve. Therefore, this subsystem requirement precludes disassembly as part of the rearming process, which presents a challenge for the mechanical design.

2.4 Chapter summary

This chapter presented the requirements pertinent to the CanX-7 payload at the mission, system, and subsystem levels. The following chapters deal with the ensuing design and development of a system to meet these requirements, and the trade-off processes in which expectations of system performance were adjusted to match what was realistically achievable. Later, Chapter 6 revisits these requirements and summarizes the current state of compliance.
Chapter 3

Design Evolution and Prototyping

This chapter presents the evolution of the CanX-7 drag sail from concept to qualification model. Due to the relatively low level of technology readiness for a deployable drag sail at this scale, the design and development process emphasized rapid prototyping to verify the functionality of major design revisions. This chapter presents the results of this process over the course of three phases of design:

- During the conceptual design phase, the engineering team selected a multi-module concept incorporating sails supported and deployed by tape springs. Prototypes produced during this phase had the intent of determining which of the concepts under consideration could be implemented simply and reliably.

- During the preliminary design phase, the design adopted a compact configuration incorporating a removable cartridge, and a deployment mechanism employing a rotating reel and rollers. A full-scale prototype demonstrated the design’s basic functionality.

- The detailed design phase involved refinement of the deployment mechanism, development of a release mechanism, and the incorporation of ground-support equipment to rewind the booms for re-stowage. Prototypes constructed during this phase contributed to an iterative testing process leading up to the fabrication of several units for a planned qualification test campaign.

The following sections discuss these phases in greater detail.

3.1 Conceptual design: A modular drag sail using tape springs

The CanX-7 drag sail is a deployable space structure. In contrast with structures that arrive on orbit in their single fully-functional configuration, (such as some satellite busses,) deployable
structures undergo an automated transition from a stowed configuration to a deployed configuration, without physical human intervention. The challenge in the design of a deployable structure lies in showing that it will operate reliably in the space environment, which cannot be closely replicated on the ground where gravitational loads are difficult to eliminate.

In general, the challenge of conceptual design lies in identifying which design features are essential to meeting the driving requirements in various contexts, and then proposing methods to implement them. Starting with the goal of developing a useful aerodynamic drag sail compatible with two spacecraft busses, the conceptual design phase gave rise to several candidate concepts. Each concept aims to answer two major questions:

- What is the structure and deployment mechanism for the sail?
- How can the design concept be integrated within the constraints of each spacecraft bus, as well as notional busses yet to be defined?

The initial concept for a modular drag sail arose during early development of the CanX-7 mission. Beginning with this concept, the engineering team constructed a series of prototypes based on a segmented sail by supported by perimeter tension. A second generation of prototypes explored different deployment mechanisms and structural concepts, including one based on a circumferential deployment mechanism, as well as one using a reel-based deployment mechanism. The team selected a final concept based on both the reel-based deployment mechanism and the original the multi-module segmented form-factor. The following subsections discuss this process in more detail.

### 3.1.1 The initial concept

The CanX-7 mission proposal authored by Zee [38] and early technical memos authored by Pranajaya [41] describe an initial concept characterized by four small circular sails deployed from identical modules stacked at one end of a CanX-2-class bus. In this concept, the sail segments incorporate a perimeter member formed from a stiff elastic material formed into a loop. Stored strain energy in the perimeter member powers deployment and provides perimeter tension that holds the circular sail material in a flat shape. The original requirement for the total sail area, based on early deorbiting analyses, was 2.0 m², implying an area for each segment of 0.5 m².
Shmuel constructed several prototypes based on this initial concept [8], patterned after collapsible flying-disc toys formed from stiff wire and flexible fabric. Later prototypes replace the stiff wire with flexible tape springs in various candidate stowage configurations. While conceptually simple, the full scale prototypes from this stage are insufficiently stiff to convincingly demonstrate reliable deployment.

### 3.1.2 Second-generation concepts and prototypes

Abandoning the deployment mechanism concept employed by the first-generation prototypes, Shmuel later developed a concept for a mechanism incorporating commercial off-the-shelf (COTS) tape springs wound into a coil around a rotating reel [8], shown in the following figures. The early prototypes based on this concept demonstrate deployment using only strain energy, and suggest that the concept holds promise. While exceedingly compact, the narrow 6.3 mm (0.25 inch) tape springs provide only a limited amount of torque to rotate the reel, and the sail stowage volume within them is limited. Wider and stiffer tape springs in subsequent prototypes allow for more reliable deployment.

![Figure 3: Deployment of an early second-generation prototype.](image)

Among the first of many prototypes constructed by the author is the one shown in the following figure, incorporating COTS tape springs with a width of 12.7 mm (0.5 inch), following curved channels inside an enlarged module. The same Teflon sliding surfaces that allow preceding prototypes to function decreases the friction in this example slightly, but not enough to achieve deployment driven only by strain energy. This prototype suggests that complex tape paths with more than a single bend would benefit from rolling anti-friction devices.
3.1.3 Alternate design concepts

Following the early prototypes, a dedicated concept exploration effort spawned a handful of alternate design possibilities. Some of these reached the prototype stage, but ultimately the choice was made to develop Shmuel’s baseline concept to maturity. The following two subsections describe some notable alternate concepts.

3.1.3.1 A centralized prototype

Tarantini [42] proposed a design concept characterized by a centralized circumferential deployment mechanism, shown in the following figure. Compared to distributed ones, centralized concepts offer the possibility of increased space for sail stowage within a given volume, by eliminating the repeated structure and mechanisms used to support all four sail segments. Other researchers have independently proposed similar circumferential deployment mechanisms [28].

Early prototypes based on this concept used a single tape spring for each boom, which are susceptible to buckling under terrestrial loads in certain directions. Such buckling introduces complications during testing. By augmenting each tape spring with a second, in which the convex sides face together, the booms can be made much stiffer. This is the approach taken by Adeli and Lappas on the CubeSail spacecraft [43] [33], brought to the attention of the CanX-7 team at the suggestion of Carroll [44]. However, practically implementing this configuration becomes complicated, because rolling the resulting booms in a coil causes a steadily increasing shear between the two tapes in each pair along their longitudinal axis. Two potential solutions arise: One approach involves stiffly bonding the two tapes in each boom together, such that the
inner and outer tapes strain by the same amount. This is the approach taken by the TRAC\textsuperscript{12} booms [45] used on Nanosail-D2 and other spacecraft, and the rollable booms developed by DLR\textsuperscript{13} [46] and used on Deorbitsail [47]. On the other hand, Lappas et al solve the shear problem by using a custom polyimide sleeve in which the tapes are free to slip [48]. In the early SFL prototype, bands of polyimide film fulfill this purpose. The resulting stiffness is lower, but it avoids the challenge of creating a strong but flexible connection between the two tapes in each pair.

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**Figure 5:** Deployment of a centralized prototype.

### 3.1.3.2 Some canted design concepts

At high altitudes, disturbance torques due to solar radiation pressure, gravity gradients, and the geomagnetic field may cause a spacecraft equipped with a drag sail to tumble, reducing the effective cross-sectional area and thus the deorbiting rate. A handful of design concepts proposed

\textsuperscript{12} Triangular Rollable And Collapsible

\textsuperscript{13} German Aerospace Center – Braunschweig, Germany
during this stage offer the potential to achieve better aerostability by canting the sail segments at an angle relative to the spacecraft’s longitudinal axis. However, later post-deployment attitude analyses would show that simply enlarging the sail is sufficient to increase the effective area. Such an approach also has the benefit of being far simpler to implement. Other researchers have proposed canted sail concepts, including one innovative example that uses a deployment mechanism possibly inspired by early published descriptions of the one on CanX-7 [49].

3.1.4 Choosing a concept

Choosing between the baseline concept and the centralized concept was not an easy matter. On one hand, rapid prototypes of the centralized concept are simple to implement and operate reliably. On the other hand, selecting the centralized concept would mean losing some flexibility in terms of bus mounting, and could necessitate the development of separate versions for both the 3U and GNB busses. The desire to design, build, and test only a single type of unit for both busses tipped the balance in favour of the baseline concept. (Chapter 4 describes how the selected concept eventually integrates with both busses.)

3.2 Preliminary design: A compact module with a removable cartridge and reel-based deployment mechanism

In general, the challenge of preliminary design lies in creating communicable descriptions of all the features that are implicit in the conceptual design. For example, the final conceptual design does not include descriptions of the electronics, or methods for attaching, folding, and stowing the sail; whereas the final preliminary design adds these features. Pre-existing features that would be refined during this phase include the specific components of the deployment mechanism, and the overall form-factor.

3.2.1 Increase in sail size

An early event in the preliminary design phase was an increase in the size of the sail, brought about by refinements to the attitude analysis which reveal that both CanX-7 and the reference spacecraft could tumble for most of their time on orbit. These analyses indicate that a sail larger than the originally-proposed 2.0 m² is needed to achieve a greater overall effective drag area and meet the required the ballistic coefficient. Thus, the total sail area in the preliminary design is double that of the conceptual design, at 4.0 m², based on analyses performed by Tarantini [9].
Accordingly, the size of the module is larger as well, with the bus design at this time adapting to accommodate the larger sail modules.

### 3.2.2 Shift to triangular footprint

Tarantini suggested [42] a compact triangular shape for the preliminary design of the sail module that allows for a straight path from the reel to the exterior of the module. With only a single bend where the tape springs leave the reel, straight tape paths substantially reduce deployment resistance compared to curved paths. The disadvantage of straight paths in a square module is that much of the module volume goes unused, and that the volume for sail stowage between the booms is low. Accompanied by an increase in height such that the module volume remains unchanged, the revised triangular footprint creates a much larger sail stowage volume suitable to the enlarged sail. The increase in height also allows for the use of wider tape springs, with an accompanying increase in boom stiffness and deployment torque.

### 3.2.3 Addition of sail cartridge

The existence of a triangular volume between the booms leads naturally to the inclusion of a triangular sail cartridge. In the preliminary design, the cartridge contains the tightly-packed sails, and allows for re-arming without the need to disassemble the spacecraft. An integrated door contains the sail within the cartridge. The inclusion of the door as part of the cartridge, rather than the rest of the module, later affects the design of the release mechanism.

To stow the sail, the preliminary design identifies a simple pattern involving two sets of perpendicular accordion folds as being appropriate to the triangular cartridge shape. For attachment, the design also proposes flexible lanyards joining the sail to the booms and cartridge. Aluminized Mylar and Kapton polymer films were leading candidates for the sail material at the time, with Mylar being used for the prototypes up to that point.

### 3.2.4 Refinement of the deployment mechanism

In the preliminary design, a set of shaft-mounted rollers constrain the coil of tape springs and reduce the resistance to deployment by replacing sliding friction between the coil and the enclosing structure with rolling friction. Inside these rollers however, sliding friction still exists where the stationary shafts and structure constrain each bushing-mounted monolithic roller in
place. This source of sliding friction would later give rise to stalling and jamming failure modes in prototypes based on this design.

To release the stored energy of the tape springs and allow the deployment mechanism to function, the preliminary design sets aside volume for a notional release mechanism and accompanying actuator. However, the design does make a selection from any of several proposed candidates. The release mechanism would later become completely defined during the detailed design phase.

3.2.5 Preliminary electronics

The preliminary design of the sail module electronics addresses all of the command and telemetry requirements described in Chapter 2.

To meet the requirement for ground commandability, the modules receive commands over the spacecraft uplink data bus originating in the communications subsystem. The independently commandable modules respond to unique arm and fire commands sent in the proper sequence. The commands are defined in the form of long binary sequences called firecodes, interpreted by low-level microcontrollers residing in various spacecraft subsystems. The firecode system allows basic commands to be addressed to specific spacecraft subsystems in the event of an onboard computer failure, which is exactly the kind of situation in which the drag sails are likely to be needed. At the level of the release mechanism, separate arm and fire lines driving the actuator reduce the susceptibility to a radiation-induced single-event upset that might inadvertently cause deployment when powered. (The modules themselves are typically un-powered, and time-out if both commands are not received in quick succession.)

Physically, the modules are multi-dropped to shared power, command, and telemetry busses (rather than connected in a daisy-chain) to prevent a failure in one module from cutting off access to others downstream. (A daisy-chain harness is otherwise tempting because it can occupy less volume compared to one with branched or star topology.) A telemetry bus connected to the onboard computer (separate from the command bus connected to the receiver) accepts commands to output stored telemetry for later relay to the ground. Command functionality over the bi-directional telemetry bus is not included by design, to prevent a spurious command issued by a malfunctioning onboard computer from inadvertently triggering deployment.
To meet the requirements for onboard telemetry collection, the preliminary electronics design includes two sensors. First, a notional door sensor provides coarse telemetry that implies the release mechanism and deployment mechanism have functioned by pushing the door open. Second, a notional boom sensor provides fine telemetry that indicates the movement of the deployment mechanism over time. While not strictly mandated by the requirements, this fine telemetry allows notional future spacecraft to infer the quality of deployment in the absence of an inspection camera. Again, the preliminary design does not specify the specific sensors from the set of those proposed; they would be selected later during the detailed design phase.

Conceptually, the electronics design congealed rapidly thanks to guidance from the large body of electronics experience that exists at SFL. The preliminary characteristics described here persist in the detailed design and implementation by Bonin and Cotten [12].

3.2.6 A full-scale functional prototype

The preliminary design process culminated in a series of full-scale prototypes that demonstrate sail stowage and deployment. One of these prototypes, shown in the following figures, incorporates commercial off-the-shelf (COTS) carbon steel tape springs with a width of 19 mm (0.75 inch) forming the booms. The deployment mechanism employs shaft-mounted monolithic polymer rollers to constrain the coiled tape springs. The sail material is Mylar film with a thickness of 5 microns.

![Figure 6: Solid model of the preliminary design (left), and a partially assembled prototype (right).](image-url)
This prototype functions repeatably, but can be induced into a jamming mode by manually over-rotating the reel while restraining the boom tips. The over-rotation causes the coil to press outward on the rollers, increasing the load upon them and therefore the sliding friction within. Additionally, the prototype exhibits a stalling mode in which deployment does not resume if artificially interrupted. Both the susceptibility to jamming and the marginal performance at room temperature indicate that a flight-like deployment mechanism with an identical design would be unlikely to function across the required operating temperature range, or after exposure to vibration loading.

With these results, the prototypes based on the preliminary design fulfill their purpose in demonstrating the basic functionality of the deployment mechanism. The first step in the later detailed design of the deployment mechanism would be to eliminate the causes of the observed failure modes.

### 3.3 Detailed design: A low-friction, damped “jack-in-the-box” deployment mechanism

In general, the challenge of detailed design lies in refining the features defined by the preliminary design such that they can be implemented using real-world hardware. In large part this consists of examining the preliminary design, which necessarily involves some speculation, and devising and implementing solutions to the puzzles posed therein; thus proving by demonstration that the implicit goals set during preliminary design are in fact realistic. Ideally this process occurs only once, and results in perfect compliance with the design requirements. In reality it may be advantageous or necessary (in terms of performance, reliability, schedule, or cost,) to iterate several times, or accept noncompliance with certain requirements.
Refining the preliminary design to the point of fabricating units based on a detailed design suitable for qualification testing took three cycles of iteration:

- The first iteration resulted in an Engineering Model (EM) of the drag sail module. Risk-reduction environmental testing of this EM revealed an unexpected jamming mode.
- Results from debugging the EM informed the second design iteration that resulting in the fabrication of a single Qualification Model (QM). Functionality testing of the first QM revealed insufficient cycle life due to boom failure.
- Results from debugging the mark-1 QM resulted in changes to the design of a second iteration. Six mark-2 QM units have been produced and await qualification testing.

The following sections in this chapter describe the results of this process, and how it gave rise to a system expected to achieve qualification for space flight.

### 3.3.1 Results from the deployment mechanism testbed

At the beginning of the detailed design phase, investigations to address the stalling and jamming failure modes identified in the preceding prototypes (based on the preliminary design) began with the construction of a deployment mechanism testbed. As shown in the following figure, the testbed accommodates a variety of roller and boom positions:

![Deployment mechanism testbed](figure8.jpg)

**Figure 8: Deployment mechanism testbed.**

When configured in the same way, the testbed is capable of reproducing the two jams exhibited by the prototypes described earlier. Experimentation with various roller configurations indicates that the position of the final rollers constraining the tape spring coil can be responsible for much of the friction in the mechanism, and the stalling failure mode it causes. In the preliminary design, the two final rollers are located at the tangent points where the inner and outer layers of
tape spring depart from the coil and exit the module as the two booms. The testbed shows that constraining the transition regions of tape that exist at these points leads to substantial outward reaction force, which leads to friction in the roller. Moving the final roller for each boom away from the transition region eliminates the excessive reaction force.

![Diagram of final roller location](image)

**Figure 9: Example of location for final roller. For clarity, this illustration depicts only a single tape spring.**

With the transition region unconstrained, the resisting torque in the testbed decreases to the point where the stalling mode vanishes. If interrupted, deployment always resumes using the available unwinding torque. However, this configuration is still susceptible to the jamming mode induced by over-rotation of the reel.

The jamming mode can be attributed to the remaining friction torque in the rollers. The friction in each roller is roughly proportional to the radial load, applied in this case by the coil when over-rotated. The asymmetric bulging coil shape that exists at the unconstrained tangent locations causes the loads to be unevenly shared by the rollers, meaning that the maximum allowable friction torque in any single roller is small. Increasing the outer- to inner-diameter ratio of the rollers within the volume and strength constraints of the sail module reduces the friction torque somewhat, but the coefficients of friction for sliding contact between metals and polymers have a practical minimum achievable limit, even when lubricated. Rolling-element bearings, on the other hand, have coefficients of friction that are orders of magnitude lower.
Therefore, replacing the bushing-mounted rollers with assemblies that incorporate rolling-element bearings reduces their friction torque substantially. By mounting the inner ring of each bearing on a stationary shaft, the outer ring serves as a rolling surface that contacts the coil of tape, and the balls inside completely replace sliding friction with rolling friction. On the testbed, this reduces the total resisting torque to the point where the jamming mode due to over-rotation disappears. The selected bearings are rated for static and dynamic loads that are upwards of ten times greater than those expected during launch, and are therefore not expected to be damaged by vibration.

The testbed also answers the question of the class of release mechanism best suited to the drag sail module. Recall that the preliminary design only defines the existence of a release mechanism, but not its nature or implementation. The successful elimination of the rotation-induced jamming mode in the testbed indicates that a “jack-in-the-box” scheme is feasible, in which the motion of the deployment mechanism is restrained or released by applying or removing force at the boom tips. (Recall that applying a force at this location, accompanied by the kind of reel rotation that results from environmental loading, can otherwise result in a jam in certain configurations.) Such an approach has the appeal of conceptual simplicity, in that it can release both the door and the deployment mechanism using a single actuated linkage. Earlier proposed concepts used two linkages; one to release the door, and one to release the reel.

With these results, the testbed fulfills its purpose of informing the arrangement and type of deployment mechanism components to include in the subsequent detailed design and prototypes, as well as the class of release mechanism to be implemented.

3.3.2 The Engineering Model

The Engineering Model (EM) is the first iteration of the sail module constructed during the detailed design phase. Shown in the following figures, it incorporates the first implementation of flight-like structural components, flight-like deployment and release mechanisms, and flight-like sail features.
The EM represents a substantial mechanical design effort. A detailed parametric solid model (in which each critical dimension is defined only once) captures the structural design of the EM and all of the iterations that follow it. (Chapter 4 provides more details on the role of the parametric solid model in the design process.)
The EM deployment mechanism incorporates a reel enclosure that contains roller bearing assemblies to constrain the coil, similar to those devised for the testbed. Inside the enclosure, the reel incorporates flanges to prevent the upper and lower surfaces of the moving coil from contacting the stationary housing, which can otherwise result in jams when rewinding the booms. The toothed flanges also enable the eventual use of ground support equipment (GSE) to rewind the reel when the module is integrated with the spacecraft. (This GSE is a feature of the later Qualification Model (QM) units; the EM itself is rewound manually.) The teeth also form part of an incremental rotary encoder for speed measurement.

The EM incorporates the first iteration of the release mechanism selected for flight. The purpose of the release mechanism (which is absent in the preceding prototypes,) is to release the sail and allow the deployment mechanism to function. The selected design takes its inspiration from the XPOD separation system. It uses a woven polymer cord loaded in tension to restrain the door, and a resistance wire heater that cuts to cut the cord to initiate deployment.

### 3.3.3 Results from the Engineering Model

For the most part, the Engineering Model and the features it incorporates perform adequately. However, it initially exhibited two jamming modes during early tests after environmental loading, which have since been corrected.

The first jamming mode observed in the EM (as originally constructed) is due to an interference issue not present in the testbed. In this mode, it is possible for vibration loading and thermal loading at high temperature (applied individually) to cause settling and expansion of the coil. In turn, the bulging coil comes into contact with the surrounding stationary module structure at a low-clearance location, resulting in sufficient static friction to match the deployment torque. The jam can be repeatably induced and observed during tests configured to do so. A design change to remove the interfering structure (involving a slight decrease to the stowage volume in the sail cartridge) eliminates this jam in subsequent revisions of the sail module design.

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14 eXoadaptable PyrOless Deployer
The second jamming mode observed in the EM (as originally constructed) occurs during deployment tests at the cold extreme of its operating range. At this temperature, friction torque in the roller assembly bearings due to the lubricant increases to the point where, in combination with the force due to friction between the sail and cartridge, the deployment mechanism is unable to overcome the total static resisting torque. Since the bearing lubricant is specifically selected for its wide temperature range and vacuum compatibility, reducing the bearing friction is not practical. Instead, a redesigned door hinge removes obstructions from the interior of the sail cartridge, at the cost of some stowage volume, and reduces the sail friction sufficiently to eliminate this jam in subsequent revisions of the sail module design.

With risk reduction tests eventually demonstrating reliable operation, the Engineering Model fulfills its purpose in verifying that the features incorporated in the detailed design function as expected, setting the stage for the subsequent design and fabrication of qualification model units.

3.3.4 The Mark-1 Qualification Model

With the jamming modes in the Engineering Model (EM) eliminated, the next steps in the detailed design and development process resulted in the fabrication of a single qualification model (QM) unit. Because the planned qualification tests involve multiple units, fabricating an initial pathfinder unit offers the possibility to avoid the cost of reworking multiple units should the need arise to correct design flaws revealed during initial functionality testing.

The design of the QM is the first to accommodate functional ground support equipment (GSE) to rewind the deployed booms back into the module while integrated with a spacecraft. As shown in the following figure, the rewind GSE functions by rotating the reel backwards using gear teeth located around the circumference of the reel flanges. In order allow the teeth to mesh, tight-tolerance rolling-element bearings constrain the position of the reel within the module, and replace the looser integral bushings that serve the same purpose in the EM.

Furthermore, the QM design incorporates flight-like electronics designed and implemented by Bonin and Cotten [11]. It also incorporates a refined release mechanism; this time designed to work with the onboard electronics, instead of specialized testing GSE.
Figure 12: The suite of Qualification Model hardware.

Ground support equipment designed and built by Sears [10] allows for faster cutting, folding, and stowing of flight-like sails compared to the earlier manual methods. The sails are a high performance polyimide film with a thickness of 12 microns. (Chapter 4 presents more detail on the sail material.)

The QM accommodates booms made from both the original COTS\textsuperscript{15} tape spring material used in previous models, as well as purpose-built tape springs formed from copper-beryllium (CuBe) alloy in a process implemented by Sears [10]. These CuBe booms are selected to achieve the required magnetic cleanliness of individual units, which was determined to be impractical using the original COTS tape springs. Mechanically, they perform comparably to those formed from carbon steel. (Chapter 4 provides more details of these features.)

\textsuperscript{15} Commercial Off-The-Shelf
3.3.5 Results from the Mark-1 Qualification Model

In most respects, the mark-1 qualification model (QM) unit functions adequately. Initially however, the booms in early tests failed to survive the required ten deployment cycles, and would instead fracture at the roots, where the tape springs attach to the reel. Examination of the failed booms reveals characteristics of overload and low-cycle fatigue. Similar failures would occur in both the earlier carbon steel booms as well as the new copper-beryllium (CuBe) booms selected for flight.

In the QM as originally constructed, as well as the prototypes that precede it, rapid deceleration at the end of deployment results in high loads. As the deployment mechanism reaches the end of its range of motion, the energy in the moving booms and reel is dissipated rapidly into the structure. In the case of the mark-1 QM, the resulting loads were high enough to cause the booms, and occasionally other parts of the structure, to fracture. There are three contributing factors to this failure:

- Over-rotation of the reel due to its angular momentum causes the outer boom root to bend beyond its intended stopping position.
- Tension in both boom roots due to the linear momentum in the booms causes the two tape springs to be pulled apart at their attachment point.
- The shape of the cartridge and reel enclosure, as well as the boom-root mounting method, exacerbates the bending loads on the tape springs and the loads on the surrounding components. Two highly-loaded regions of the structure were observed to fail.

Independent work later published by Harkness et al [35] reports the same type of behaviour in a similar system. Chapter 4 goes into more detail about how the basic shape of the deployment mechanism influences the space of possible solutions to this problem.

Earlier prototypes, going back to the deployment mechanism testbed, reveal bending in the boom roots at this location, but not failure. Due to reduced friction, the mark-1 QM as originally constructed achieved higher final deployment speeds, which results in deceleration loads sufficient to cause fracture. The decrease in friction is attributable to the reel bearings that enable the functionality of the rewind GSE.
Proposed methods for increasing the cycle life of the QM design fall into a few basic categories:

- Strengthening the structure and mechanism to better withstand deceleration loads.
- Dissipating kinetic energy during a final fraction of the range of motion.
- Reducing overall deployment speed.

The main constraint on all these methods is to have a negligible or tolerable impact on the initial net deployment torque, so as not to impact the ability of the deployment mechanism to function. Chapter 4 describes some of the candidate solutions, a few of which reached the prototype stage.

The selected approach for increasing the cycle life in the QM design incorporates boom roots redesigned for increased flexibility, and a commercial off-the-shelf (COTS) viscous rotary damper. Testing of this damper, conducted and described by Sears [10], builds confidence in its ability to operate in the required environment. Testing to date consists of functional tests under atmospheric pressure and over temperature, functional tests under vacuum at room temperature, and outgassing tests of the constituent materials. Functional testing of the Mk-1 QM retrofitted with the redesigned boom roots and rotary damper indicates a cycle life well in excess of the requirement, and the fracture issue no longer occurs. Therefore, the second iteration of the qualification model design incorporates these design changes.

### 3.3.6 The Mark-2 Qualification Model

Following the elimination of the boom root fracture issue, six qualification model units have been fabricated for use during the planned qualification testing campaign, which includes tests after vibration loading as well as during thermal-vacuum loading. The subsequent chapter presents the design of the Mk-2 QM in detail.

### 3.4 Chapter summary

This chapter presented the broad evolution of the drag sail from the initial concept to the final qualification model units. The basic concept of a multi-module drag sail deployed using strain energy persisted throughout the design process, which resulted in a series of prototypes incorporating a removable sail cartridge and a rotating deployment mechanism driven by tape springs. The next chapter presents the qualification model design in more detail, and discusses some design considerations pertaining to the sail structure and deployment mechanism.
Chapter 4

Drag Sail Module Mechanical Design

The drag sail developed for the CanX-7 mission is a deployable structure made up of four independent sail segments, each one consisting of a flexible membrane supported by a pair of stiff booms deployed from a compact module. The combined sail is roughly square in shape, and measures 2.0 m on a side, with a total area of 4.2 m². The sail modules are designed to be stowed during launch and initial operations, and later deployed to deorbit the spacecraft. The following illustration depicts the spacecraft with all four sails deployed.

Figure 13: Illustration of the CanX-7 spacecraft with all four segments of the drag sail completely deployed.
4.1 Basic characteristics of the drag sail module

The basic unit of the drag sail is an individual drag sail module. CanX-7 incorporates four such modules to form a roughly planar sail. A single drag sail module is a triangular prismatic object with a side-length of 9.0 cm and height of 4.0 cm. The module deploys two 1.5 m long booms that support a trapezoidal sail with a base length of 2.0 m, a width of 1.0 m, and an area of 1.05 m². Each module has a mass of approximately 200 g. Figure 14 shows a drag sail module in the stowed state, and Figure 15 shows the deployed state.

Figure 14: Drag sail module, stowed.

Figure 15: Drag sail module, deployed.
4.2 Modularity and integration

The drag sail employs two levels of modularity. First, the structure of the sail payload is separate from the spacecraft bus structure. Second, the payload is composed of multiple homogeneous units, which are distributed at several locations across the spacecraft bus. These two different levels of modularity have distinct impacts on the design process.

The first level of modularity is fairly common for spacecraft payloads in general, and serves to partially decouple the payload design from the bus design. For missions developing a new payload like CanX-7, and/or where the bus design is also somewhat uncertain, developing the two separately can provide a schedule advantage. For missions using a mature bus design like the Generic Nanosatellite Bus\(^{16}\) (GNB), modularity reduces the impact of a changing payload design on the development and analysis of the bus, so long as the interfaces between them remain constant.

For the drag sail in particular, the second level of modularity enables a greater number of solutions for integration. The parts of a distributed sail payload can be mounted on the periphery of a large spacecraft, whereas a centralized sail will have some of its area covered by the bus. Furthermore, a centralized sail mounted on a flat panel must traverse the panel and its protrusions during deployment, which might present a risk of snagging or tearing; a distributed sail faces no such risk. The large sail on TechDemoSat-1, for example, is stowed in a channel that wraps around the perimeter of its bus [50]. However, the second level of modularity means that repeated mechanisms and components are present that could, in principle, serve all four sail segments. For example, the CanX-7 sail has four release mechanisms, four deployment mechanisms, and four microcontrollers; whereas a similar centralized design might only require one of each. Each of these repeated devices occupies volume, adds to the payload mass, and must be assembled, tested, and refurbished.

\(^{16}\) The GNB is a bus commonly used for missions designed and built at UTIAS-SFL. See Chapter 2, section 2.3.2.1.
As intended, the drag sail modules integrate naturally with the two required busses. On CanX-7, four modules are stacked beneath a structural panel that forms one of the end faces of the spacecraft bus. Pass-through channels allow wiring access to the exterior panel, as shown in the expanded views in Figure 16.

Figure 16: Mechanical integration (left) and electrical integration (right) with CanX-7.

On a GNB-class spacecraft, the sail module is designed to be mounted onto one of the faces that remain exposed when loaded into its XPOD\textsuperscript{17} separation system. Figure 17 shows the baseline conventional configuration in which the modules are attached to one of the main structural trays. Furthermore, unconventional spacecraft orientations within the XPOD have been proposed for other GNB-class missions, and the sail module is compatible with these, too.

\textsuperscript{17} eXoadaptable Pyroless Deployer
On larger spacecraft, the modules might be distributed toward the extremities of the bus, as in the NEMO-class\textsuperscript{18} example shown in Figure 18.

Significantly, compatible spacecraft in general need only incorporate the number of modules appropriate to their mass and altitude, freeing up precious space within the volume-limited cubesat form factor. Nor is the mounting orientation restricted to symmetric or even planar configurations: Since geomagnetic, gravity gradient, and solar radiation pressure torques can overwhelm aerodynamic torques at high altitudes [9], a tumbling spacecraft with a large time-averaged projected area in all directions might deorbit as effectively as one with limited aerostability obtained from a symmetric sail configuration.

\textsuperscript{18} Nanosatellite for Earth Monitoring and Observation, built by UTIAS-SFL.
4.3 Major parts and mechanisms

The drag sail module contains a folded sail stowed in a removable cartridge, two booms coiled around a reel and mounted within an enclosure and cap, as well as support electronics; all contained within a single housing. Mechanical ground support equipment (GSE) facilitates rewinding the booms when the modules are integrated with a spacecraft. The following figure shows an expanded view of the drag sail module depicting these major parts.

![Figure 19: Major parts of the drag sail module.](image)

The drag sail module accommodates three mechanisms that provide its essential functionality, supported by a collection of structural parts:

- The deployment mechanism, depicted in Figure 20, extends the stowed booms to deploy the sail.
- The release mechanism, depicted in Figure 21, releases the restraining loads on the sail and deployment mechanism.
- The rewind mechanism, depicted in Figure 22, enables the booms to be retracted while on the ground, thus allowing the sail module to be repacked and rearmed after testing.
Figure 20: Components of the deployment mechanism.

The various parts of the deployment mechanism work in concert to deploy the booms and sail:

- The reel supports the booms in both their stowed and deployed states.
- When coiled around the reel, the tape spring booms form a rotary spring. The strain energy residing in the stowed booms powers the deployment mechanism.
- Roller assemblies surrounding the reel allow for smooth motion of the attached coil.
- When moving, the rotary damper provides a resisting torque roughly proportional to the angular speed, which slows down deployment without greatly affecting the net torque when stationary.
- The reel enclosure and the sail cartridge provide mounting points for the roller assemblies.
- The housing and cap provide mounting points for the bearings that support the reel axle.
- The housing and cartridge also form paths that channel the booms into the V shape that forms the sail structure.
Figure 21: Components of the release mechanism. (Wiring harness not shown.)

The various parts of the release mechanism work in concert to release the restraints on the sail and deployment mechanism.

- The hinged door applies a reaction load to the booms and sail. A wear plate on the door bears the load due to the unwinding torque applied by the inner boom.
- A load-bearing expendable polymer cord holds the door in place.
- A resistance wire heater severs the cord when powered.
- Spring pins form a switch that telemeters whether the door is completely closed.
- The door protects the cord, heater, and switch from direct exposure to ultraviolet (UV) radiation and atomic oxygen (AO).
- The sail module electronics drive the heater.

Figure 22: Components of the rewind mechanism.
The rewind mechanism consists of both ground support equipment (GSE) as well as interfaces on the module itself that support attachment and mechanical power transmission. The various parts of the rewind mechanism work in concert to drive rotation of the deployment mechanism in reverse:

- The rewind rotor meshes with gear teeth on the flanges of the deployment mechanism reel.
- A ratchet prevents the rotor and reel from moving in the deployment direction when engaged.
- When the booms are completely rewound, the cartridge is removed and replaced with a dummy cartridge, such that the rewind GSE may be safely detached and used elsewhere while the cartridge is re-packed.

### 4.4 Solid model

While only consisting of a few major parts, the drag sail module contains over one hundred individual components. Managing these components requires an appropriate method for ensuring that the equally large number of mechanical interfaces remain consistent throughout the design evolution. Therefore, a detailed parametric solid model captures the mechanical design of the CanX-7 drag sail module.

In general, parametric solid models define the driving relationships between basic geometric entities and design features, and accommodate variation in the dimensions and other parameters to achieve the size and positioning needed for a specific application. The approach aims to reduce the re-work time that necessarily results from changing dimensions that affect multiple parts. Accordingly, well-built parametric solid models capture not just the design of a single system, but also the potential design of all systems with the same geometric structure. Associativity between separate parts and assemblies in the model maintains consistency at critical interfaces. Compared to an assembly like a spacecraft bus, which achieves high packing density by finding a compact configuration of pre-existing fixed shapes, an assembly like the sail module can achieve the same thing through the flexibility of its constituent parts.

The solid model for the CanX-7 drag sail module uses a hierarchical structure of parts and assemblies. It defines critical interfaces using a top-down system of layout sketches and inter-
part links, as well as horizontal links for a minority of relationships where the top-down approach becomes counterproductive. Geometric relations and simple expressions capture design intent, and each driving dimension is defined only once. The solid model enables the process of experimentation that is necessary to achieve high packing density within the given volume constraints.

4.5 Fabrication

Most of the purpose-built mechanical components of the drag sail module are made using an additive fabrication process, with a minority made using subtractive processes. Additive fabrication allows for parts with complex 3D geometry to be made in small batches at a lower cost compared to many subtractive processes, even those that are partially or completely automated.

The additive fabrication process selected for the engineering, qualification, and flight models of the CanX-7 drag sail module uses selective laser sintering (SLS) of carbon-fibre filled polyamide. The CRP USA company performs this process using the proprietary Windform XT 2.0 material, selected at the suggestion of Sinclair [51]. Current and upcoming space missions are establishing flight heritage for this material [52] [53], and it passes standard outgassing tests. Resilience against degradation due to ultraviolet (UV) radiation and atomic oxygen (AO) on exposed surfaces is as-yet undetermined. On orbit however, AO- and UV-resistant thermal control tapes will cover all exposed surfaces of the drag sail modules.

Application-specific post-processing prepares the Windform parts for spaceflight. The service provider performs some of these steps, mostly consisting of upsizing printed holes to standard diameters, and other operations are performed at SFL to adjust critical dimensions on a module-by-module basis. Helical and press-fit threaded inserts installed in printed holes allow for the installation of threaded fasteners. Controlled-volatility epoxy adhesive increases the strength of the resulting hole. Careful cleaning to remove un-sintered support material results in parts acceptable for cleanroom assembly.

4.6 Sail structure and components

The deployed structure of each sail segment consists of the booms, sail membrane, and interconnecting components. Work by Sears [10] presents details of cutting, folding and stowing
the sail membranes into the removable cartridges, as well as the process and apparatus for fabricating the booms. This section discusses the structure of the sail in general, and the methods for attachment to the booms and cartridge.

4.6.1 Structure of the sail

According to the classification developed in Chapter 1, the CanX-7 sail is a cantilevered structure in which the bending stiffness of the booms holds each separate sail segment in place against the external loads that act upon it. On Earth, the sail structure has the ability to support its own weight when oriented horizontally in free space. While not a requirement, the ability to support its own weight establishes confidence that the sail can easily withstand the eventual on-orbit loads. The tip forces due to the weight of the sail membranes and the distributed forces due to their own weight load the booms in a combination of bending and torsion. On orbit, the much lower aerodynamic drag loads at the boom tips are expected to result in a similar combination of bending and torsion.

Deformation of the booms and sail membrane results in a slight loss of drag area. Under load, the membrane assumes a shape similar to a catenary and the projected area decreases compared to the surface area of the material itself. On earth, the projected area loss under gravity when oriented horizontally is due to the weight of the booms (approximately 0.30 N each) and sail (approximately 0.20 N), and amounts to approximately 10 percent of the sail material area. On orbit, the maximum drag loads occur at the lowest operational altitude required, and are predicted to be less than 1.0 percent of the sail membrane weight alone on Earth. Therefore, the area loss due to drag loads on orbit will be well below that due to terrestrial gravity.

In the design presented here, the area of one sail segment is 1.05 m², and is the largest that can be practically accommodated within the limits of folding and stowage inside the sail cartridge. The cartridge itself is as large as can be practically accommodated within the CanX-7 bus. The total area of the sail material is 5 percent greater than that considered in the deorbiting and attitude analyses performed by Shmuel [8] and Tarantini [9]. The conservatism in these two analyses, combined with the additional sail area, allows for a margin of error in the eventual mean ballistic coefficient on orbit that accounts for the miniscule projected area reduction expected under drag load.
During tests on the ground, an additional area reduction effect exists that is expected to diminish on orbit. Upon deployment, residual stress in the folds of the sail material causes tension in the partially-folded membrane that pulls the booms together. Thermal tests show that the folds tend to relax when heated, resulting in a smoother surface. Thermal modelling shows that the sail will routinely reach temperatures upwards of 100 °C [12], which is sufficient to result in fold relaxation during sub-scale ground tests. Therefore, it is anticipated that the planar sail area will come to approach the flat material area when deployed on orbit.

4.6.2 Sail Material

The sail material is a metallized polyimide film called Upilex-S. It is comparable to DuPont Kapton but has a wider allowable operating temperature range, in addition to other high-performance properties. The material has a thickness of 12.7 microns. An aluminum coating on both sides has the purpose of protecting the substrate from erosion by atomic oxygen. The trapezoidal sail shape is a consequence of both the width of the available rolls of material, as well as the need to reduce the worst-case temperature of the sail’s hottest location due to radiative interaction with the bus (by offsetting the root while maintaining the sail area).

During conceptual and preliminary design, aluminized Mylar was the leading candidate for the sail material. However, early thermal analyses indicated that the predicted worst-case hot temperature for thin aluminized membranes would result in melting of this substrate. Later, more refined thermal finite element analyses performed by Cotten [12] indicated that even Kapton, the replacement choice, could degrade over time, leading to the selection of Upilex. Works published by Sears [10] and Shmuel [8] describe the analyses and trade studies that led to the final selection of the sail material.

4.6.3 Attachment of the sail

Flexible lanyards made from metal wire attach the sail to the booms and cartridge. At the outboard corners of the trapezoidal sail, two lanyards at each corner attach the sail to the squared-off ends of the nearest boom tip. At the root of the sail, two lanyards attach the inboard corners to a pin inside the cartridge.

Crimp tubes secure the lanyards in place. On the sail side, the lanyards loop through metal grommets installed at the corners the sail, and attach to themselves using a crimp tube. On the
cartridge side, they loop around the internal pin. On the boom side, the lanyards pass through holes drilled at the corners of the boom tips, and are terminated with a crimp tube. This results in a degree of freedom for the lanyards to slide through the tip holes, which simplifies boom retraction and stowage.

The use of two lanyards to attach each outboard sail corner (suggested by Tarantini [42]) enables the booms to support loads in both directions relative to the sail plane normal. Under gravitational loads during ground testing, one lanyard in each pair transmits most of the load on the sail to the boom tips. The other lanyard in each pair remains mostly slack. The loading of a single lanyard at one corner of each boom tip results in torsion that favourably twists the booms such that they undergo opposite-sense bending, as shown in the following illustration, in which the sense of the bending moment is opposite to that of their transverse curvature. When the direction of the load on the sail is reversed, the opposite lanyard becomes tense. Therefore, the booms are expected to withstand loads applied on orbit from both forward and aft directions relative to the sail plane normal.

![Figure 23: Illustrations showing sail deformation under load (left), and loading at boom tips due to lanyards (right). The dominant loading of a single lanyard depending on the direction of the drag load results in a favourable combination of bending and torsion in the booms.](image)

### 4.6.4 Material and fabrication of the booms

The booms that support the sail and power the deployment mechanism are purpose-built tape springs formed from copper-beryllium (CuBe) alloy. The choice of copper-beryllium arises from magnetic cleanliness issues, and in-house fabrication is necessary due to the expense of procuring ready-made tape springs in this material.
Up until the qualification model (QM) design, the booms were made from commercial off-the-shelf (COTS) tape springs (and later, raw tape springs provided by a major manufacturer,) made from carbon steel; a ferromagnetic material. However, attitude analyses performed by Tarantini [9] and Cotten [11] place an upper limit on the allowable spacecraft dipole, with testing showing that the steel booms alone would be well in excess of this limit. While it is possible to degauss carbon steel, the process would require subsequent magnetic cleanliness practices up to the time of launch, imposing logistical complications. Furthermore, such booms might be susceptible to magnetization during launch or over their on-orbit life. A seemingly simple solution is to use a non-ferromagnetic boom material, hence the adoption of copper-beryllium; a material widely used for deployable booms in space missions and inspired here by its use in a similar application on the CubeSail mission [33]. Mechanically, the performance of the copper beryllium tape springs is comparable to or better than those made from carbon steel. Furthermore, the thermo-forming process implemented by Sears [10] (based on the CubeSail example [48]) allows for variable geometry at the roots, which simplifies their attachment to the reel.

The booms are coated with reflective metal plating in order to achieve an expected worst-case-hot temperature below that at which the CuBe material may anneal and weaken, and also to prevent excessive heat transfer to the sail module structure. Up until the QM design, both the COTS tape springs (and later the bulk material obtained from a manufacturer) simply retained their stock coating of baked-on yellow paint, as has been the case for many cubesat antennas. Being that drag sail booms are not antennas, however, and are load bearing structures that contact moving parts, the potential issues of outgassing and adhesion exist if the stock paint is retained. Furthermore, the thermo-optical properties of the paint would result in high temperatures and heat transfer to the sail module and bus. Therefore, a coating with engineered properties is indicated even for steel booms. The selected coating for the flight booms is high-phosphorus electroless nickel (EN), which has an amorphous microstructure that results in both non-magnetic and anti-galling properties [54] [55, p. 124].
4.7 Deployment mechanism design considerations

4.7.1 Basic concept of operation

The deployment and release mechanisms together operate similarly to a “jack-in-the-box” toy, in which constant deployment forces act at one end of a moveable body, and reaction forces at the opposite end are removed to initiate deployment. When the mechanism reaches the limit of its range of motion, the structure reacts against the deployment forces and the motion stops. Somewhat more precisely, the deployment mechanism can be described as being formed from a flexible body, with:

- a driving torque at one end,
- a restraining force at the other, and
- constraining and resisting forces and torques that act in between.

The driving torque arises from the coiled tape spring booms, which exert an outward force against the rollers inside the reel enclosure. Reaction forces from the rollers incompletely constrain the coil, which results in a net torque in the deployment direction. The surrounding structure applies constraining forces that transform the unwinding torque into linear force on the booms and door. The restraining force applied by the cord via the door prevents the booms from moving, and balances the force that results from the net torque due to the coil. Removing this restraining force causes the system to become unbalanced and accelerate. Furthermore, friction acts opposite to the direction of the net forces and torques, and torque due to the damper reduces the mechanical power once the system starts to move.

4.7.2 Broad classes of deployment mechanisms

The deployment mechanism uses an internal source of stored strain energy to power deployment. This mechanism might be described as internally-driven, in contrast to notional externally-driven mechanisms that use an electric actuator to provide deployment force. Although the initial concept for the drag sail module incorporated an internally-driven deployment mechanism from the start, externally-driven mechanisms were considered during the concept exploration stage. In the absence of an unambiguous reason to use one or the other, the choice between the two classes might be seen as something of a philosophical one.
On one hand, an internally-driven mechanism requires only a small input of energy to trigger deployment, perhaps provided by a simple uni-directional actuator, or the degradation of an expendable link. Therefore, it has the appeal of conceptual simplicity, which might be assumed to correspond with reliability and ease of design. The internally-driven approach deals with uncertainty by designing to conservative performance requirements that are verified by testing.

On the other hand, an externally-driven mechanism offers a form of control, and more deployment force might be available than provided by the internal source alone; allowing for deployment to proceed in the face of unexpected friction or other resisting forces. In addition to serving as the release mechanism, if reversible the deployment actuator might also act as the rewind mechanism. Also, it offers the possibility of overcoming unexpected contingency modes (namely jams) that fall outside of the expected operating envelope. Therefore, it has the appeal of adaptability in the face of uncertainty.

This distinction would not be a concern were it not for the fact that space mechanisms sometimes fail despite testing, and failures involving one paradigm might seem like a strong reason to favour the other. Notably, spring-loaded mechanisms can jam, for reasons that may be difficult to diagnose on orbit. The separation system that ejected the Nanosail-D2 spacecraft, for example, jammed upon activation; the satellite was only recovered after thermal strain cycles (presumably) worked it loose. Of course, externally-driven mechanisms can fail too, even during the kind of contingency situations in which they offer adaptability: The Galileo high gain antenna deployment anomaly, for example, might have ended differently if the nominally bi-directional Dual Drive Assembly was capable in practice of driving the unexpectedly jammed mechanism to which it was attached in both directions, instead of only the single direction for which it was configured [56]. The perceived ability to “get out and push” if needed has a certain allure, but it may come at the cost of complexity that can itself become a cause of failure. On-orbit results continue to provoke surprise. As it would turn out in the case of the drag sail module, volume constraints made it clear that the internally-driven approach was practical. Perhaps for the same reasons, at least one other cubesat-scale drag sail has adopted a similar approach [35].

### 4.7.3 Deployment arresting solutions

The boom-root fracture issue discussed in Chapter 3 necessitates a means of either increasing the strength of the failed components, or arresting deployment so as to reduce the loads that cause
the components to fail. The basic shape of the deployment mechanism is a contributing factor to the boom root fracture issue, and influences the selection of a solution. Within the constraints of the sail module form factor, the asymmetric shape is designed to optimize the starting configuration to achieve low friction. It is not designed to optimize the stopping configuration for low force. The rotationally-asymmetric orientation of the booms with respect the reel means that the momentum of the booms tends to forcibly continue rotation of the reel, instead of working against each other to stop it, as they do in the symmetric mechanism developed by Harkness et al (incorporating a redesign after encountering the same issue [35]). The proposed solutions fall into three categories described below, the most promising of which reached the prototype stage.

Prototypes implementing the first and seemingly simplest category of solutions, which involve strengthening the boom roots and structure, all failed. Practical implementations within the volume and flexibility constraints of the deployment mechanism were insufficient to increase the fatigue life of the booms. Notably, one proposed solution that was perhaps mistakenly discounted without being prototyped involves pin-mounted boom roots with a rotational degree of freedom, similar to the solution later described by Harkness et al [35].

In the second category, the approach of dissipating energy at the end of deployment involves a notional device that operates only near the maximum extent of the mechanism’s range of motion. One proposed solution incorporates a viscoelastic member integrated into the last portion of the tape springs. Analysis suggests this would be able to reduce the final deceleration to manageable levels; however, prototypes indicate that fabricating such a brake, and attaching it to the flexible tape spring material, would be highly challenging. Two separate proposed solutions involve contact between parts to create a friction brake. One of these uses the expanding inner diameter of the tape spring coil to trigger engagement. Another involves a flexible wedge. The difficulty in confidently predicting friction properties in the operational environment complicates this type of approach.

In the third category, solutions that continuously dissipate energy are conceptually simpler, but any static contribution to the deployment torque balance must be low. A rotary spring is one simple example; however, consultations with manufacturers indicate that constructing a spring to achieve the required maximum torque and range of motion within the volume available inside
the sail module could be difficult. Among many other concepts, Sinclair [51] suggested the use of an escapement, as used on the Viking Mars Lander antenna deployment mechanism [57]. Miniature viscous dampers have the appeal of a COTS\textsuperscript{19} solution. Small linear dampers exist, but might necessitate an additional cam-type linkage in order to act on a rotating system. Miniature rotary dampers seem fit for purpose, but require environmental testing, which is described in work by Sears [10].

The final design incorporates a rotary damper into the bottom of the sail module housing, shown in the following figure. Chapter 5 discusses the effect of the damper on the operation of the deployment mechanism.

![Cutaway view showing the rotary damper (blue).](image)

### Figure 24: Cutaway view showing the rotary damper (blue).

#### 4.8 Chapter summary

This chapter presented the design of the final iteration of the qualification model drag sail module. The module deploys a cantilevered sail structure using a mechanism conceptually similar to a jack-in-the-box toy. The mechanism operates under the influence of driving, constraining, and retaining forces and torques that arise from its components and structure. The next chapter develops a mathematical model of the deployment mechanism, and compares the results to experimental data in order to determine the extent to which the explanation given in this chapter is correct.

\textsuperscript{19} Commercial Off-The-Shelf
Chapter 5

Analysis of the Deployment Mechanism

The purpose of the analysis presented in this chapter is to develop a model for the deployment mechanism that describes the motion of its major parts and their dominant behaviour. The extent to which the model is valid will be determined by comparing its predictions to test data.

5.1 Modelling the system

The analysis in the following subsections models the deployment mechanism as a system composed of two rigid bodies. One body undergoes one-dimensional rotation and represents both the reel and the coil of tape springs wound around it. The other body undergoes one-dimensional translation and represents the two booms, which are formed by the portions of tape spring that have unwound from the coil. The two bodies move in synchrony, with the rotation of the reel mapping directly to the position of the booms, very much like a rack and pinion – except that the two bodies exchange mass as they move.

5.1.1 Basic equations of motion

Two equations of motion that describe rigid bodies are simply Euler's second law in the case of one-dimensional rotation and Newton's second law in the case of one-dimensional translation. For a rotating body, the equation relating the net external torque $\Sigma \tau$, moment of inertia $I$, and angular acceleration $\alpha$ is:

$$\Sigma \tau = I \alpha$$  \hspace{1cm} (5-1)
For a translating body, the equation relating the net external force $\Sigma F$, mass $m$, and acceleration $a$ is:

$$\Sigma F = m \cdot a$$  \hfill (5-2)

The moment of inertia of the rotating body corresponds to that of the reel and coil, and the mass of the translating body corresponds to the mass of both booms together. These are represented by:

$$I = I_{reel} + I_{coil}, \quad m = m_{booms}$$  \hfill (5-3)

(The properties of both bodies are shown here as constant, which is sufficient for the basic definition in this section. However, the mass and moment of inertia change when the tape springs wind on or off coil as they move; this is addressed in a subsequent section.)

Two space variables describe the system: The angular coordinate $\theta(t)$ represents the rotation of the reel and coil, and the linear coordinate $x(t)$ represents the deployed length of either boom at any instant. The deployment direction is positive for both coordinates. The angular and linear acceleration can be represented in terms of the space coordinates by:

$$a = \frac{d^2}{dt^2} \theta(t), \quad a = \frac{d^2}{dt^2} x(t)$$  \hfill (5-4)

It is useful to consider the forces and torques at the interface between the two bodies separately from the other external forces and torques that act on them. (These quantities at the interface become relevant when the system rapidly decelerates at the end of deployment, which can cause the boom roots to fail under some conditions.) At the interface, an external torque due to the booms $\tau_B(t)$ acts on the reel and coil, and an external force due to the reel and coil $F_C(t)$ acts on the booms. Elsewhere, other external forces and torques act on the two bodies: External unwinding and resisting torques $\tau_E$ act on the reel and coil; other external forces $F_E$ act on the booms.

Using the preceding definitions for the external torques, moment of inertia, and angular acceleration, the equation of motion for the reel and coil becomes:

$$\tau_E + \tau_B(t) = (I_{reel} + I_{coil}) \frac{d^2}{dt^2} \theta(t)$$  \hfill (5-5)
Likewise, using the preceding definitions for the external forces, mass, and linear acceleration, the equation of motion for the booms becomes:

\[ F_E + F_C(t) = m_{\text{booms}} \frac{d^2 x(t)}{dt^2} \]  

(5-6)

Two more equations complete the system. At the interface where the booms join the coil at its constant outer radius \( r_c \) (determined by the fixed size of the reel enclosure), the interface torque on the coil and the interface force on the booms act in opposite directions.\(^\text{20}\) The unknown functions representing the torque on the coil due to the booms \( \tau_B(t) \) and the force on the booms due to the coil \( F_C(t) \) are related by:

\[ \tau_B(t) = -r_c F_C(t) \]  

(5-7)

Also, because the two bodies are directly joined, the coordinates describing the angular position \( \theta(t) \) and the linear position \( x(t) \) are simply related by:

\[ x(t) = r_c \theta(t) \]  

(5-8)

The following diagram illustrates the coordinate system implied by the preceding equations:

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\(^\text{20}\) The torque \( \tau_B \) and force \( F_C \) are reactions to each other in different coordinate frames. The inclusion of the minus sign in the coordinate relation (as opposed to either of the equations of motion) allows the sign convention to remain consistent when the system is driven in both directions: When the motion of the coil is forced by an applied torque \( \tau_E \), the interface torque \( \tau_B \) acts as a reaction torque opposite to the applied torque. Meanwhile, from the perspective of the booms, the interface force \( F_C \) drives the motion of the booms in the direction opposite to the reaction torque on the coil \( \tau_B \). On the other hand, when the booms are forced by an external force \( F_E \), the interface force \( F_C \) is a reaction force due to the coil, and the resulting interface torque on the coil \( \tau_B \) drives the coil in the opposite direction as the reaction force. The sign relation therefore is analogous to that between the forces linking two collinear bodies \( A \) and \( B \), which is \( F_{AB} = -F_{BA} \).
Note that while all other quantities in the preceding figure are depicted as acting in the positive direction, \( \tau_B \) and \( F_C \) must act in opposite directions. The directions shown here imply that the external torque \( \tau_E \) drives the boom interface force \( F_C \), which corresponds to deployment (although other configurations are valid).

The model up to this point consists of the four preceding equations, and four unknown functions of the independent variable \( t \): These are the two space coordinates \( \theta(t) \) and \( x(t) \), as well as the interface torque \( \tau_B(t) \) and force \( F_C(t) \). This system alone would permit a solution if the external forces, torques, masses and moments of inertia were constant. These quantities are not constant, however, due to a dependence on one or more of the space variables and their derivatives.

The next several sections expand the system of differential equations just presented to accommodate the remaining variables that are indirectly functions of time, introducing more equations and unknowns as necessary. Later, a static simplification of the resulting system compares the unwinding torque against the various resisting torques and develops a margin of safety. Finally, a numerical solution presents the evolution of the system over time.

### 5.1.2 External forces and torques

The system presented in the preceding section includes several variables that require definition. These variables are the external torque applied to the reel and coil \( \tau_E \), its moment of inertia \( I(\theta) \), the external force applied to the booms \( F_E \), and their mass \( m(t) \).

---

21 The two bodies are connected by a single kinematic joint with only one degree of freedom. Therefore, it is possible to describe the system using only a single equation of motion and a single space variable. However, using a greater number of dimensions allows the forces, torques, and positions to be conveniently expressed in terms of the native space coordinate for each body, rather than converting back and forth between whichever coordinate happens to be favoured for a given task.
The external torques applied to the reel and coil $\tau_E$ consist of the unwinding torque $\tau_{UW}$ from the tape springs, and the resisting torque $\tau_R$ from various sources. The unwinding torque serves to rotate the reel and coil, and acts only in the positive direction. The resisting torque opposes any motion of the reel and coil, and in this case acts in the negative direction:\footnote{22 The constant negative direction used here is sufficient for this analysis because the directions of velocity and applied torque do not reverse. However, in situations where the torque or motion do reverse, such as earlier versions of the deployment mechanism where a viscoelastic brake allowed oscillation at the end of deployment, the model must capture these changes of direction. Generally, the damper torque acts opposite to the velocity and has a sign given by $-\text{signum}(\omega)$ in the reel’s coordinate frame. The various notional friction torques $\tau_f$ act against the direction of applied torque and have signs given by $-\text{signum}(\Sigma \tau - \tau_f)$ in the reel’s coordinate frame.}

$$\tau_E = \tau_{UW} - \tau_R \quad (5-9)$$

The resisting torque $\tau_R$ arises from several sources:

- The reel enclosure exerts a friction torque $\tau_e$ that depends on the load on the rollers.
- The damper exerts a torque $\tau_d$ that depends on the angular velocity.
- The axle bearings exert a friction torque $\tau_a$ that depends on the load on the axle.

Thus, the resisting torque may be expressed as:

$$\tau_R = \tau_e + \tau_d + \tau_a \quad (5-10)$$

The external forces acting on the booms $F_E$ consist only of a resisting force $F_R$ that acts in the negative direction, against deployment:\footnote{23 As in the preceding footnote, more generally the notional friction forces $F_f$ act against the direction of applied force and have signs given by $-\text{signum}(\Sigma F - F_f)$ in the booms’ coordinate frame.}

$$F_E = -F_R \quad (5-11)$$

This resisting force is due to the force of friction between the booms and structure $F_{BS}$, and the force of friction between the sail and cartridge $F_S$:

$$F_R = F_{BS} + F_S \quad (5-12)$$

The following sections go on to derive equations for the several new forces and torques just introduced.
5.1.3 Unwinding torque and the steady end moment

First among the new torques to be defined is the unwinding torque $\tau_{uw}$. The unwinding torque arises from the steady end moments of the two tape springs that make up the coil. When a tape spring is bent past the point of snapping from its transversely-curved unstressed state to a longitudinally-curved stressed state, a moment $M^*$ applied at both ends will hold the fold in place. This end moment is described as “steady” because it does not vary with the angle $\theta$ through which the ends of the tape are rotated. Seffen and Pellegrino describe the bending and folding of tape springs in a detailed 1999 monograph [58], and Soykasap gives a concise overview in a 2007 paper [59].

There are two different values for the steady end moment depending on the direction in which a tape spring is folded, as depicted in the illustration that follows (Figure 26). On one hand, same-sense folds are weaker than their opposite-sense counterparts, and occur when the ends of a tape spring are bent such that the resulting curvature of the longitudinal axis has the same sense as the transverse curvature of the unbent portion. Put another way, the resulting outward normal vectors on a single side of the tape (including the bent and unbent portions) either all converge or all diverge (with the normal vectors on the reverse side doing the opposite). Applying a same-sense steady end moment $M^*$ to both ends of the folded tape (in opposite directions on either end to prevent it from rotating) will maintain the fold in place. On the other hand, opposite-sense folds are stiffer, and on either side of the tape the longitudinal curvature of the fold has the opposite sense as the original transverse curvature: The outward normal vectors of the bent portion on one side of the tape converge or diverge opposite to those of the unbent portions on the same side. Applying an opposite-sense steady end moment $M^*$ to both ends of the tape (again, in opposite directions) will maintain the fold in place.
Figure 26: Sign convention for end moments $M^+$ applied to tape springs to maintain same-sense and opposite-sense folds.

The steady end moments depend on the bending stiffness (also called flexural rigidity) $D$ of the flat cross section of the tape spring, the included angle $\phi$ of its transverse cross section, and the Poisson ratio $\nu$ of the material. For an opposite-sense fold, the steady end moment is given by:

$$M^+ = (1 + \nu) D \phi \quad (5-13)$$

For a same-sense fold, the steady end moment is given by:

$$M^- = -(1 - \nu) D \phi \quad (5-14)$$

(Note that the sign of this moment is negative, by definition.)

From Soykasap's paper [59], the bending stiffness (flexural rigidity) $D$ for a plate of thickness $p$, elastic modulus $E$, and Poisson's ratio $\nu$ is given by:

$$D = \frac{E p^3}{12 (1 - \nu^2)} \quad (5-15)$$

Each layer of the coil is made up of two stowed tape spring booms, with one boom forming an inner sub-layer and the other boom forming the outer sub-layer overtop it. The inner tape spring
undergoes same-sense folding, and the outer tape spring undergoes opposite-sense folding. Together, the reactions to the moments exerted by these tapes produce the unwinding torque $\tau_{uw}$ that acts on the reel and coil, as depicted in the following illustration:

![Figure 27: Simplified diagram of the forces and torques acting on the stowed tape spring coil.](image)

When stowed, the forces and torques on the coil are balanced, and it does not move. The perpendicular forces applied by the structure at the boom tips are reactions to the end moment exerted by of the free ends of the tape. These reactions give rise to sliding friction, discussed in a later section. The radially-oriented forces on the coil in the figure above represent the distributed constraining forces from the reel enclosure. They are reactions to the outward force exerted by the coiled tape, and also serve to balance the off-centre forces from other sources. These reactions give rise to rolling friction, discussed in the next section. The longitudinal forces at the tips represent constraining forces from the door, and are reactions to the unwinding torque $\tau_{uw}$ from the coil, discussed in this section. They exert torque on the coil that acts against deployment. When these forces are removed by the release mechanism, the net torque becomes unbalanced and the coil rotates.

The unwinding torque $\tau_{uw}$ arises from the same-sense steady end moment $M^+$ and opposite-sense steady end moment $M^-$ of the tape springs within the coil. For the purposes of this analysis, it is assumed that the relationship between the unwinding torque and the end moments
is independent of the number of layers wound around the coil: It is well established that the end
moment exerted by a tape spring folded through a small angle is the same as one folded through
a large angle. Therefore, an empty reel that is partially rotated and consequently wound with an
incomplete layer of tape exerts the same unwinding torque as one that has made a full rotation,
and presumably a coil that is wound to include many layers does the same. Simple measurements
support the approximation\(^{24}\) that the unwinding torque \(\tau_{uw}\) is constant throughout deployment,
and equal to the sum of the absolute values of the steady end moments of both tape springs:
\[
\tau_{uw} = M^* = |M^-| + |M^+|
\]
(5-16)
The absolute value operators serve to overcome the sign convention of the published formulae,
in which the same-sense moment is negative. To accommodate this sign convention, the equation
becomes:
\[
\tau_{uw} = -M^- + M^+
\]
(5-17)
Using the equations for the end moments given earlier, the equation for the unwinding torque of
the coil simplifies to:
\[
\tau_{uw} = 2D \phi
\]
(5-18)
The preceding equation depends only on the basic fixed parameters of the tape springs; therefore
the unwinding torque in this model has a constant value throughout deployment.

5.1.4 Reel enclosure torque, part 1

Continuing to define the torques that act on the reel and coil, the reel enclosure contains several
roller assemblies that are collectively a source of friction torque. The torque due to the enclosure
\(\tau_e\) arises from the \(n_r\) individual torque contributions \(\tau_i\) from each roller:
\[
\tau_e = n_r \tau_i
\]
(5-19)

---

\(^{24}\) Although early measurements made using a coil formed from tape springs with uniform properties support this
assumption, later experiences with springs having non-uniform properties show that the underlying layers of tape do
affect the unwinding torque, and that assumption of \(\tau_{uw} = M^*\) is simplistic. The actual torque seems likely to be
greater than that predicted here due to the action of the underlying layers of tape spring. It may be the case that
sliding friction between the inner layers results in them having a diminished effect compared to the outer layer,
perhaps similar to the capstan effect.
At the outer radius of the coil $r_c$, each roller exerts a tangential interface force $F_t$ related to the individual roller torques $\tau_i$ about the reel and coil:

$$\tau_i = r_c F_t$$  \hspace{1cm} (5-20)

Assuming no slip, each interface force $F_t$ in turn arises from a resisting torque about the roller axle $\tau_r$ that is inversely proportional to the radius of the roller $r_r$:

$$F_t = \frac{\tau_r}{r_r}$$  \hspace{1cm} (5-21)

Each resisting torque about the roller axle $\tau_r$ arises from the $n_b$ bearings in each assembly, which individually exert a friction torque $M_b$. (The chosen symbol $M$ in this context denotes the torque in a bearing, in deference to the convention of prominent publications on the subject [60], [61].) The torque about each roller axle can be expressed as:

$$\tau_r = n_b M_b$$  \hspace{1cm} (5-22)

Combining the preceding four equations, the resisting torque on the coil due to the reel enclosure $\tau_e$ comes to depend on the bearing friction torque $M_b$:

$$\tau_e = n_r \frac{r_c}{r_r} n_b M_b$$  \hspace{1cm} (5-23)

At this point, it can be seen that the ratio between the radii of the coil and rollers results in leverage of the friction torque $M_b$ from each bearing in the roller assemblies. The next section presents a derivation for this torque.

### 5.1.5 Reel enclosure torque, part 2: Bearing friction

In the absence of test data, this analysis uses a parametric model that incorporates basic characteristics of the roller assembly and axle bearings to provide an approximation of their friction torque. A common model for the friction torque $M$ in a bearing incorporates the radial load $F$, the bearing bore $d$ (also called inner diameter), and a friction coefficient $\mu$:

$$M = \mu F \frac{d}{2}$$  \hspace{1cm} (5-24)

The friction coefficient typically used in this model is a general-purpose value applicable to lightly-oiled bearings at room temperature, and depends on the basic type of bearing as well as its operating mode (i.e. starting or running). The ball bearings in the sail module, on the other
hand, include a thick vacuum-compatible grease and operate across a range of temperatures. The viscosity of the lubricant changes substantially with temperature, resulting in a different friction torque for the same load. This analysis captures the variation in friction torque as a change in the friction coefficient $\mu$.

In the case of the roller-assembly bearings in the reel enclosure, the friction torque model uses the load on each bearing $F_b$, their bore $d_r$, and the friction coefficient $\mu_r$ associated with the lubricant in question:

$$ M_b = \mu_r \frac{F_b d_r}{2} \quad (5-25) $$

(An equation with the same form applies to the bearings on the reel axle.)

The model for the bearing friction coefficient $\mu_r$ depends on the temperature $T$ and uses an exponential function with empirical coefficients $g_{1,2}$ obtained from test data for the selected lubricant, discussed in appendix A:

$$ \mu_r = g_1 e^{g_2 T} \quad (5-26) $$

In reality, the running friction of the bearing is also weakly a function of rotational speed, but temperature effects are assumed to dominate at the low speed resulting from the damper.

The load on each bearing $F_b$ arises from the load on each roller assembly $F_r$, and is assumed to be shared equally between the number of bearings per assembly $n_b$:

$$ F_b = \frac{F_r}{n_b} \quad (5-27) $$

Thus the equation for the roller assembly bearing friction torque comes to depend on the roller load $F_r$:

$$ M_b = \mu_r \frac{F_r}{n_b} \frac{d_r}{2} \quad (5-28) $$

Like the unwinding torque, the load on each roller $F_r$ arises from the steady end moment, and is derived in the following section.
5.1.6 Reel enclosure torque, part 3: Roller assembly load

As just stated, the load on each roller assembly $F_r$ arises mostly\textsuperscript{25} from the outward force exerted by the tape spring coil, which arises from the steady end moment of the tape springs. An equation approximating the load on each roller can be obtained from a thought experiment in which a tape spring coil is held in place by two initial pinching forces in opposition, which are then replaced by several smaller circumferentially-distributed forces that represent the rollers.

The first step in the thought experiment results in an equation for the pinching force. Measurements confirm that if a moment is applied to a folded tape spring in order to maintain the fold as shown in the following illustration, then the resulting moment is very close to the value predicted by the steady end moment formulas.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure28.png}
\caption{Relationship between end moment and force.}
\end{figure}

Therefore, the relation between the end moment and the force $F$ exerted on a tape spring at a particular position $l$ along the tape's length is simply:

$$l F = M^*$$  \hspace{1cm} (5-29)

It can also be confirmed by measurement that the same holds approximately\textsuperscript{26} true when $l$ is reduced to a smaller value, say, closer to the radius of the tape spring coil $r_c$ (which is approximately equal to the natural fold radius of the tape springs, by design). Defining a

\textsuperscript{25} For example, this analysis ignores the centrifugal loads from the flexible coil.
pinching force $F_p$ as one that is applied at the natural fold radius, the magnitude of the pinching force can then be expressed using the previous relation as:

$$F_p = \frac{M^*}{r_c}$$  \hspace{1cm} (5-30)

The following illustration shows such an arrangement, in which end moments are applied to both ends of a coil of tape by means of two pinching forces that are analogous to two rollers:

![Diagram showing pinching forces](image)

**Figure 29:** With two rollers, the reaction load on each roller $F_r$ is equal to the pinching force $F_p$.

The total inward force exerted by the rollers is $2F_p$. If the number of rollers were increased, as shown in the following illustration, the total inward force would be divided between the number of rollers $n_r$, resulting in a reaction force on each expressed by:

$$F_r = \frac{2F_p}{n_r}$$  \hspace{1cm} (5-31)

\[26\] In reality, as the perpendicular force is applied closer and closer to the fold, the actual radius of the fold decreases beyond the natural radius, and the force necessary to maintain the tape in position increases beyond that predicted by the formula. However, the coil within the enclosure undergoes no such reduction in radius, because the size the enclosure that applies the pinching is fixed. Therefore, the discrepancy in the case of the coil and enclosure is assumed to be small.
Figure 30: If the two pinching forces were replaced by a greater number of forces to achieve the same effect, the overall inward force would be distributed amongst them, and remain unchanged.

Using the definition of the pinching force (5-30), the preceding equation for the reaction force on each roller becomes:

$$F_r = \frac{2M^*}{r_c n_r} \quad \text{(5-32)}$$

The end moment $M^*$ represents the steady end moment of the multi-layer coil consisting of two tapes back-to-back, which was approximated earlier as being equal to the unwinding torque $\tau_{uw}$. Therefore the equation for the reaction force on each roller becomes:

$$F_r = \frac{2\tau_{uw}}{r_c n_r} \quad \text{(5-33)}$$
5.1.7 Reel-enclosure torque, part 4

Recall that the results of the preceding three sections were:

\[ \tau_e = n_r \frac{r_c}{r_r} n_b M_b, \quad M_b = \mu_r \frac{F_r d_r}{n_b} \frac{d_r}{2}, \quad F_r = \frac{2\tau_{uw}}{r_c n_r} \]  

(5-34)

Combining these, the equation for the resisting torque due to the reel enclosure \( \tau_e \) becomes:

\[ \tau_e = \frac{\mu_r \tau_{uw} d_r}{r_r} \]  

(5-35)

Note that:

- Both the number of rollers and the number of bearings per roller cancel out, and have no effect on the total reel enclosure torque in this model. The total torque from the bearings is proportional to their number, but their individual torques are proportional to the applied load which is shared amongst them. This could be true only if the friction torque in each bearing were entirely due to rolling friction. In reality, the friction torque includes a contribution from the lubricant that is independent of the applied load. Therefore the model does not completely capture the added contribution from the lubricant.

- Bearings with a wider bore \( d_r \) result in greater leverage of their friction torque. Therefore, small-bore bearings are preferable.

- Space permitting, rollers with a larger radius \( r_r \) lead to decreased resisting torque \( \tau_e \). Likewise, a coil with a larger radius \( r_c \) would lead to increased resisting torque, were it not for the inverse proportionality to the same parameter in the roller load \( F_r \) that cancels out the effect.

5.1.8 Axle torque

Continuing to define the torques that act on the reel and coil, the axle torque \( \tau_a \) arises from friction torque \( M_a \) in each of the \( n_a \) bearings upon which the axle is mounted:

\[ \tau_a = n_a M_a \]  

(5-36)

Using the same model as for the roller assembly bearings, the friction torque depends on the bearing bore \( d_a \), load \( F_a \), and coefficient of friction \( \mu_r \):

\[ M_a = \mu_r \frac{F_a}{2} \frac{d_a}{2} \]  

(5-37)
The axle bearings use the same lubricant as the roller assembly bearings, and this analysis uses the same model for the friction coefficient to capture the variation in friction torque over temperature. As just discussed in Section 5.1.7, this model only incompletely captures the contribution due to lubricant viscosity. However, the impact of the discrepancy is less severe for the axle bearings, since a smaller number of them are present.

Assuming that speeds are low enough such that dynamic forces are small, the load on each axle bearing \( F_a \) arises only from the tape springs that connect the coil to the reel. This load arises from the end moments \( M^* \) of the inner ends of both tape springs together, which act at approximately the inner radius of the coil \( r_i \), and are assumed to be divided evenly between the number of axle bearings \( n_a \):

\[
F_a = \frac{-M^+ + M^-}{n_a r_i}
\]  

Later, Section 5.1.11 derives an equation for the changing inner radius of the coil \( r_i \) in the course of modelling its moment of inertia.

### 5.1.9 Damper torque

Continuing to define the torques that act on the reel and coil, an empirical formula models the damper torque \( \tau_d \) and depends on the speed \( \omega \), temperature \( T \), and several coefficients \( a_{1...5} \). The model has the form of a bivariate polynomial defining a surface with parabolic curvature in both orthogonal directions:

\[
\tau_d(T, \omega) = a_1 + a_2 \omega + a_3 T + a_4 \omega^2 + a_5 T^2
\]  

The model is extrapolated from available performance data for the damper, and discussed in more detail in Appendix A.

### 5.1.10 Boom forces

Moving on to the forces that act on the booms, recall that the resisting forces applied to the boom \( F_R \) are due to friction between the booms and structure \( F_{BS} \), and the friction between the sail and cartridge \( F_S \):

\[
F_R = F_{BS} + F_S
\]
The force due to friction between the sail and cartridge is difficult to model with much accuracy due to the complex behaviour of the unfolding sail. Crude measurements show that it decreases from an initial value $F_{S,i}$ to zero as the cartridge empties. It can be approximated by a simple piecewise function incorporating a first-degree polynomial:

$$
F_S = \begin{cases} 
F_{S,i} - \frac{F_{S,i}}{l_{\text{boom}}} x, & x \leq l_{\text{boom}} \\
0, & x > l_{\text{boom}}
\end{cases} \quad (5-41)
$$

The force due to friction between the booms and structure arises from both the inner and outer booms, where the end-moment of each one presses the sliding tape against the unmoving structure:

$$
F_{BS} = F_{\text{inner}} + F_{\text{outer}} \quad (5-42)
$$

The friction forces are modelled as Coulomb dry friction, arising from a normal force $F_\perp$ and a friction coefficient $\mu$. The maximum value of such a friction force $F_\parallel$ is given by:

$$
F_\parallel = \mu F_\perp \quad (5-43)
$$

In the case of either boom, the normal force $F_\perp$ arises from the end moment of the tape spring $M^*$ acting at the distance from the coil $l$ where the tape spring contacts the structure, with a friction coefficient $\mu_{ts}$ between the two materials. For either boom, the friction force may be expressed as:

$$
F_\parallel = \frac{\mu_{ts} M^*}{l} \quad (5-44)
$$

Thus the friction force from both booms together may be expressed as:

$$
F_{BS} = -\frac{\mu_{ts} M^*}{l_{\text{inner}}} + \frac{\mu_{ts} M^*}{l_{\text{outer}}} \quad (5-45)
$$

(Taking into account the sign convention that defines $M^*$ as negative.)

At this point, all of the forces and torques that act on the bodies have been defined. The next section completes the definition of the remaining variables in the system.

### 5.1.11 Masses and moments of inertia

As the reel and coil rotate, they exchange mass with the body representing the booms. Furthermore, the moment of inertia of the body representing the reel and coil changes as the tape
on the coil redistributes itself such that its outer radius remains in contact with the rollers of the reel enclosure.

The moment of inertia of the reel and coil arises from a constant contribution from the reel and a varying contribution from the coil that varies with the deployed length. The contribution from the reel $I_{reel}$ may be obtained from the solid model. The contribution from the coil can be approximated using the formula for the moment of inertia of a tube about its axis, which depends on its mass $m$, inner radius $r_i$, and outer radius $r_o$; and is given by:

$$I_{tube} = \frac{1}{2} m (r_i^2 + r_o^2)$$  \hspace{1cm} (5-46)

The outer radius of the coil always contacts the inner radius of the reel enclosure and is constant. The inner radius expands as the booms deploy, and depends on the length of tape present on the coil $l_{coilt}$ and the thickness of each layer $s$.

The coiled length $l_{coilt}$ may be expressed as the product of the number of layers $n_l$ and their median circumference $C_{med}$:

$$l_{coilt} = n_l C_{med}$$  \hspace{1cm} (5-47)

The number of layers can be expressed as:

$$n_l = \frac{r_o - r_i}{s}$$  \hspace{1cm} (5-48)

The median circumference is proportional to the median radius $r_{med}$, and can be expressed as:

$$C_{med} = 2\pi r_{med}$$  \hspace{1cm} (5-49)

In turn, the median radius can be expressed as:

$$r_{med} = \frac{r_o + r_i}{2}$$  \hspace{1cm} (5-50)

Using the preceding four equations, the coiled length can be expressed as:

$$l_{coilt} = \frac{\pi (r_o^2 - r_i^2)}{s}$$  \hspace{1cm} (5-51)
Solving the preceding equation for \( r_i \) results in:

\[
 r_i = \sqrt{\frac{r_o^2 - \frac{l_{coil}}{\pi} s}{r_c^2 - \frac{(l_{boom} - x) s}{\pi}}} \quad (5-52)
\]

Apart from the definition stated earlier, the coiled length \( l_{coil} \) can be also expressed using its relationship to the deployed length \( x \):

\[
l_{coil} = l_{boom} - x \quad (5-53)
\]

Using the preceding result, and replacing the symbol for the outer radius \( r_o \) with the symbol for the coil radius \( r_c \), the final version of the equation for the inner radius of the coil \( r_i \) is:

\[
r_i = \sqrt{\frac{r_c^2 - \frac{(l_{boom} - x) s}{\pi}}{r_c^2 - \frac{l_{coil}}{\pi} s}} \quad (5-54)
\]

The remaining quantity to be defined in the general formula for the moment of inertia of a tube (5-46) is the mass of the coil \( m_{coil} \), which can be expressed as the product of the number of booms \( n_B \), the mass per unit length \( \kappa \) of a single boom, and the coiled length:

\[
m_{coil} = n_B \kappa l_{coil} \quad (5-55)
\]

Thus, using general formula given earlier, the moment of inertia of the coil can be expressed as:

\[
I_{coil} = \frac{1}{2} m_{coil} (r_i^2 + r_c^2) \quad (5-56)
\]

Lastly, the mass of the booms at any instant arises from their mass per unit length \( \kappa \) and their total length, which includes the deployed length \( x \) of the \( n_B \) booms, as well as the short straight portions at their tips that are not wound on the coil when completely stowed (with lengths \( l_{inner} \) and \( l_{outer} \), respectively):

\[
m_{booms} = (n_B x + l_{inner} + l_{outer}) \kappa \quad (5-57)
\]

At this point, definitions for all the variables included in the system initially constructed in Section 5.1.1 have been presented. The next section deals with a simplification that applies to the static case.
5.1.12 Static figure of merit: Torque margin

A static evaluation of the system reveals whether sufficient torque is present to initiate deployment at the instant when the release mechanism operates. In order for deployment to begin, the maximum possible static resisting torque \( \tau_{R,S} \) (including torques not just acting directly on the coil, but also torques due to forces on the booms) must be less than the unwinding torque \( \tau_{UW} \):

\[
\tau_{R,S} < \tau_{UW}
\]  
(5-58)

A margin of safety is a useful figure of merit to compare the unwinding torque to the resisting torque. The margin of safety is equivalent to the signed net torque on the reel and coil expressed as a fraction of the static resisting torque. At the instant when the release mechanism removes the restraining forces on the boom tips, a positive nonzero torque margin indicates that the system will begin to deploy. Such a torque margin \( MS \) may be defined as:

\[
MS = \frac{\tau_{UW} - \tau_{R,S}}{\tau_{R,S}} = \frac{\tau_{UW}}{\tau_{R,S}} - 1
\]  
(5-59)

The static resisting torque \( \tau_{R,S} \) arises from the resisting torques \( \tau_R \) applied directly to the coil, and the torque on the coil due to the resisting forces \( F_R \) on the booms:

\[
\tau_{R,S} = \tau_R + \tau_c F_R
\]  
(5-60)

Recall that the resisting torque \( \tau_R \) was previously stated to be due to the enclosure, damper, and axle:

\[
\tau_R = \tau_e + \tau_d + \tau_a
\]  
(5-61)

Also recall that the resisting force \( F_R \) was stated to be due to the sail, as well as friction between the booms and structure:

\[
F_R = F_S + F_{BS}
\]  
(5-62)

For the purpose of comparing them to the other torques acting on the reel (in the next section), it is useful to express these two individual forces as the equivalent torques they exert on the coil:

\[
\tau_S = \tau_c F_S, \quad \tau_{BS} = \tau_c F_{BS}
\]  
(5-63)

Values for the constituent forces and torques in this section can be obtained from the equations previously derived, using static coefficients of friction where appropriate.
This concludes the development of a model describing the deployment mechanism. The next two sections present the results obtained from the model using parameter values that correspond to the real system, and a later section compares those results to bench-top test data.

### 5.2 Static results

The torque margin of safety $MS$ is calculated using static friction coefficients for the bearing lubricant as well as for the interface between the tape springs and structure. (Appendix A lists the values used for all the parameters in the model.) The following plot shows the resulting predicted torque margin across the operating temperature range:

![Static Torque Margin](image)

**Figure 31: Predicted static torque margin over operating temperature range.**

The deployment model predicts that the torque margin is positive over the operating temperature range of $-40$ °C to $+80$ °C. The torque margin becomes negative around approximately $-75$ °C. (The beginning of a downward trend past $+90$ °C is an artifact from the damper torque model, and probably not representative of the real trend.) These results agree with tests performed at atmospheric pressure. Tests under thermal-vacuum conditions will be performed as part of the qualification test campaign.
Rising resisting torques from several sources cause the decrease in torque margin seen toward the cold end of the operating temperature range. The following plots compare the unwinding torque $\tau_{UW}$ against the total static resisting torque $\Sigma \tau_{RS}$ and its constituent terms:

![Static Torques](image)

**Figure 32:** Comparison of unwinding and resisting torques.

![Static Torques](image)

**Figure 33:** The same data as the previous plot, but with a logarithmic scale to better show the smaller torques.
The resisting torque due to static friction between the booms and structure \( \tau_{BS} \) dominates the total resisting torque \( \Sigma \tau_{R,S} \) throughout the operating temperature range. The static damper torque \( \tau_d \) and sail friction torque \( \tau_S \) are the next largest. The torque from the axle bearings \( \tau_a \) and reel enclosure bearings \( \tau_e \) (as modelled) are the least significant.

## 5.3 Dynamic results

The model developed in the previous sections forms a system of differential equations that may be solved numerically. The solution presented here predicts performance that qualitatively matches the dominant behaviours of the real system. The following two plots show the predicted speed and acceleration over the time it takes for the booms to deploy, using the best available values given in Appendix A for model parameters that correspond to room temperature.

![Deployment Speed, \( T = 20 \, ^\circ\text{C} \)](image)

**Figure 34: The predicted speed achieves a nearly-constant rate of increase.**

As the system begins to move it is severely unbalanced and the velocity rises quickly. As the damper torque increases with speed it tends to pull the system toward an equilibrium velocity; to the naked eye observing the deployment of a real sail module, the velocity appears to reach a constant value. The system as modeled never reaches equilibrium, however, and the velocity continues to increase slowly due to the non-zero acceleration shown in the next plot.
CHAPTER 5. ANALYSIS OF THE DEPLOYMENT MECHANISM

Figure 35: The predicted acceleration rapidly decreases to a small nonzero value. After the acceleration falls rapidly from its initial value, it reaches a non-zero minimum value and begins to slowly climb at a nearly-constant rate until the end of deployment. The presence of the damper in the system is the main reason for the initial decreasing trend, which in isolation would cause the acceleration to proceed asymptotically towards zero. The rising trend that continues to the end of deployment is primarily a result of the system failing to reach equilibrium, as shown in the following plot which compares the unwinding torque $\tau_{UW}$ to the total resisting torque $\Sigma \tau_R$:

![Graph showing acceleration over time](image1.png)

Figure 36: The unwinding and resisting torques remain unbalanced throughout deployment.

---

27 The model as presented here is not formulated to capture the sudden deceleration at the end of deployment. However, previous versions of the model employing other arresting mechanisms needed to deal with this scenario.
The torque imbalance that causes the system to persistently accelerate after nearing equilibrium is very small and decreases over time. The sail friction torque \( \tau_S \) for example, seen in the next plot, constitutes a ramp input that continuously unbalances the system. The redistribution of mass within the coil contributes to a higher-order effect as its inner radius increases.

The following two plots compare the unwinding torque \( \tau_{UW} \) against the total resisting torque \( \Sigma \tau_R \) and its constituent terms at a larger scale throughout the course of deployment:

![Torque From Various Sources, T = 20 °C](image)

**Figure 37: Plot of torque from various sources throughout deployment.**

As intended, the damper serves to substantially decrease the deployment power without greatly affecting the static torque balance. Once the system begins to move, the damper torque \( \tau_d \) quickly becomes the dominant contribution to the resisting torque \( \Sigma \tau_R \) during the majority of deployment. The torque due to friction between the booms and structure (\( \tau_{BS} \), which was the dominant source of friction in the static case,) is far below that of the damper. The other torques are small in comparison.
Despite their low magnitude, the effect of the other resisting torques is not invisible: The torque due to friction between the sail and cartridge $\tau_S$ varies with the deployed length and is the main cause for the persistent disequilibrium shown previously in Figure 36. The axle torque $\tau_a$ also varies with the deployment distance due to its dependence on the inner radius of the coil, and contributes to the same effect.
Like the real system, the model predicts performance that changes noticeably with temperature. The model exhibits the expected trend in which the overall speed increases with temperature, as seen in the predicted angular speed histories shown in the following plot:

![Deployment Speed Over Operating Temperature Range](image)

- **Figure 39**: The overall predicted deployment speed increases with temperature.

### 5.4 Comparison with test data

Most of the deployment telemetry captured to date has been for the purpose of damper characterization, a task made difficult by the inclusion of a sail. Therefore the experimental data referenced in this section do not include a sail, and values for the model parameters are adjusted to suit. Under these circumstances, the model predicts performance at room temperature that resembles the behaviour of the real system, shown in the following plot by speed data calculated from position telemetry collected during one of several deployment tests.
An obvious difference between the model and the test data is the hint of a periodic component to the deployment speed. The deployment telemetry consistently reveals a variation in angular speed that occurs with a spatial period equal to one revolution, and therefore seems likely to be associated with the attachment point of the tape springs to the reel. The reel enclosure does not completely constrain the flexible coil; instead it bulges slightly at the two gaps where the inner and outer booms wind off. Inside the coil, where the roots of the tape springs attach to the reel, the smaller fold radius causes the coil to protrude even further through the gaps as the attachment point passes beneath. This change in the bulge may cause a variation in the unwinding torque, resisting torque, or both; giving rise to the periodic variation in speed. However, since flexure of the elastic coil might be non-dissipative for the most part, it could be the case that the net effect on the deployment speed over time is small.

Other un-modelled losses and inaccurate parameter values may account for the remaining error. For example, the model does not include terrestrial friction due to gravity as the booms (and sail, when present,) slide across the surface of deployment GSE\(^{28}\). The agreement at room
temperature in the absence of a sail suggests that the magnitude of this difficult-to-model force is relatively small. Furthermore, a study of the sensitivity to changes in uncertain parameters would reveal the extent to which the agreement with experiment is simply coincidental.

The agreement of the speed history with test data is better at room temperature than at the extremes of the operating range. The model over-predicts the plateau speed at the cold extreme of temperature, and under-predicts the speed at the hot extreme. Accordingly, the predicted deployment time follows the opposite pattern. The following table compares the difference between the model results and test data calculated from position telemetry:

<table>
<thead>
<tr>
<th>Plateau Speed [rad/s]</th>
<th>Temperature [°C]</th>
<th>-40</th>
<th>20</th>
<th>80</th>
</tr>
</thead>
<tbody>
<tr>
<td>Observed</td>
<td></td>
<td>16 to 25</td>
<td>38 to 50</td>
<td>81 to 94</td>
</tr>
<tr>
<td>Predicted</td>
<td></td>
<td>36</td>
<td>48</td>
<td>52</td>
</tr>
<tr>
<td>Model Error</td>
<td></td>
<td>125 % to 44 %</td>
<td>26 % to -4 %</td>
<td>-36 % to -45 %</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Deployment Time [s]</th>
<th>Temperature [°C]</th>
<th>-40</th>
<th>20</th>
<th>80</th>
</tr>
</thead>
<tbody>
<tr>
<td>Observed</td>
<td></td>
<td>3.8</td>
<td>1.6</td>
<td>0.8</td>
</tr>
<tr>
<td>Predicted</td>
<td></td>
<td>2.0</td>
<td>1.5</td>
<td>1.4</td>
</tr>
<tr>
<td>Model Error</td>
<td></td>
<td>-47 %</td>
<td>-6 %</td>
<td>75 %</td>
</tr>
</tbody>
</table>

Since the plateau speed is mainly attributable to the damper, the errors suggest that the variation in predicted damper torque at the speeds encountered is too insensitive to cold temperatures, and too sensitive to hot temperatures. The deficiency in the model arises from limited performance data, which do not span the range of speeds over which the system operates. In service, the damper operates well above its rated speed range, where trends must be extrapolated from available data. The extrapolation appears valid at room temperature, but becomes increasingly inaccurate at colder and hotter temperatures. Although its performance is difficult to predict beyond the range over which data is available, testing to date indicates that the damper operates reliably over the required temperature and speed range at atmospheric pressure, and also while in vacuum at room temperature. Tests over the entire required range under thermal-vacuum conditions will be performed as part of the payload qualification test campaign.

The model for the bearing friction torque due to lubricant viscosity is another source of error that grows toward the cold end of the operating temperature range. As previously mentioned, the model treats the changing lubricant friction as an increase in rolling friction, which does not scale with the number of bearings. Thus the bearing friction model may be adequate for a system
with a single bearing, but not more. A better model for the friction torque \( M_b \) would include separate terms for the rolling friction \( M_r \) dependent on load \( F \), and the lubricant friction \( M_v \) dependent on speed \( \omega \):

\[
M_b = M_r(F) + M_v(\omega)
\]  

(5-64)

Better agreement with real-world results might be achieved by refining the analytical models for the resisting forces and torques, by determining more accurate values for uncertain parameters through measurement, and/or by performing regression analyses to determine better values from available telemetry.

### 5.5 Chapter summary and conclusion

The analysis of the deployment mechanism presented in this chapter models the system as two linked bodies with a single degree of freedom that move and exchange mass under the influence of changing torques and forces.

A simplification for the static case developed a figure of merit that compares the forces and torques at the instant when the deployment mechanism is released. The results predict that the deployment mechanism will operate at the extents of its required temperature range, which matches the outcome of real-world testing.

The form of the dynamic results (without the sail) matches a simplified picture of the behaviour of the real system. This agreement with experiment implies that the model captures the most important performance drivers under these conditions. The model shows better agreement with real-world performance at room temperature than toward the extremes of the operating temperature range. The model predicts trends in performance over temperature that match the direction of those observed in the real system, but undershoots their magnitude.
Chapter 6

Requirements Compliance, Themes, Lessons Learned, Recommendations, and Conclusions

This chapter concludes the thesis by revisiting the requirements outlined in Chapter 1, presenting some common themes developed over the course of the design and development process, and offering general recommendations and suggestions for follow-on work.

6.1 Compliance with requirements

For the most part, The CanX-7 drag sail fulfills the intent of the critical requirement calling for a cubesat-scale drag sail suited to larger spacecraft. The design meets most of the applicable requirements, with a few exceptions.

The final payload mass is approximately 0.9 kg, greater than the maximum 0.5 kg set out in the requirements. Since the mass of a single sail module is dominated by the booms, there is little opportunity to adjust the design to meet the requirement. Instead, the spacecraft mass budget has been revised to accommodate the larger payload mass. The total mass of the spacecraft, with contingency, is within the limit to which the launch vehicle separation system is qualified.

The drag sail module does not completely meet the requirements pertaining to reusability. Compliance with the weakest “will” requirement for a one-hour re-arming time has yet to be achieved for flight units. Also, the system is noncompliant with the “should” requirement to not use expendable parts. The drag sail module uses two expendables parts; the release mechanism
cord and the crimp tubes on the sail lanyards. Fortunately the sail itself turns out not to be expendable, as threatened to be the case before the folding process was refined. Finally, the system is technically noncompliant with the “shall” requirement precluding disassembly of the spacecraft when re-arming. The system meets the intent of the requirement by using a removable sail cartridge, which does not invalidate the workmanship verification achieved by system-level acceptance testing. Compliance with this requirement ends up being a trade-off involving risk to the spacecraft. While it is possible to imagine a sail module that can be rearmed without involving a removable component, doing so might place the spacecraft at greater risk than it already is, by increasing the intensity of sail re-stowage work performed in close proximity to its delicate exterior surfaces.

6.2 Themes and lessons learned

There were several emergent themes in the design and development process. First, the process was somewhat exploratory. The level of technology readiness for the system in question means that there are not yet any established design rules. Despite the best of intentions, exploratory design can end up consisting of more trial and error than originally planned, which can be time consuming and expensive.

Another theme is one of continuously striving for simplicity. There exists a general conservatism with regard to perceived complexity that favours the simplest approach from the outset, where “simplest” usually equates with the lowest part count. This is easier said than done. If the simplest solution among a set of candidates turns out to be nonfunctional, then it might become necessary to begin a time-consuming cycle of iteration and complication. Obviously, wisdom, judgement, intuition and experience all play a role in distinguishing between options that are simple as in “elegant,” and simple as in “simplistic.” Starting with this goal in mind, the process of simplification yielded beneficial results. Some examples include:

- the lanyard-sail attachment scheme, which originally involved delicate reusable split rings instead of expendable crimp tubes,
- the sail folding scheme, which originally used the Miura fold and later moved to the perpendicular accordion folds, and
- the release mechanism, original aspirations for which involved a complicated latching behaviour.
There are several lessons learned that are worth mentioning. The most important has to do with the risks associated with changing a tested system during a staged design process.

In large part, the features of the drag sail module were developed in a staged fashion, in which the most fundamental features were refined and prototyped first, (e.g. the form factor and deployment mechanism,) followed by support features (e.g. the rewind mechanism and release mechanism). In this type of process, if features developed during the later stages of design affect features tested during the earlier stages, then the confidence in the design established by earlier testing is invalidated. This increases the risk associated with the subsequent design change.

In the case of the drag sail module, it so happened that a later feature (meshing gears in the rewind mechanism) necessitated a change (reducing play in the reel axle by adding bearings) to an earlier tested feature (the deployment mechanism). The risk associated with the change was accepted by reasoning that the expected side-effect would not adversely impact the existing system, (based on the understanding of the system at the time,) as it was consistent with the direction of the development effort up to that point. Specifically, the expected side-effect of adding the needed bearings was to reduce friction in the deployment mechanism, which was the goal of earlier deployment mechanism features.

There are two points where this kind of projection can diverge from reality: Either the side-effect(s) of the design change can be different from expected, (e.g. no change in friction, but some other emergent behaviour,) or the impact of the side-effect can be different from expected (e.g. decreased reliability, not increased). As it happened, the impact of further decreasing the friction was to cause to parts to fail that earlier testing had implicitly verified as sufficiently strong. To their credit, this possibility was raised by some observers of the process, but not acknowledged as a likely outcome. Again; wisdom, judgement, intuition and experience play a role in this type of assessment.

It can be the case that a design change to a tested system is inescapable. This type of risk can be better managed when the design is flexible, and is therefore more tolerable during earlier stages of design and development than later ones. An effect of the conservatism toward change in the staged design process is that later features become increasingly constrained as the design progresses, which results in more time needed to find solutions. Therefore, the approach to reduce risk under this paradigm is to judiciously incorporate elements of all the planned features
early on, particularly focusing on those areas where features overlap, with the intent of eliminating the need to change earlier features in order to implement later ones.

A counterpart to the staged design process (in which the design begins to cool and freeze from the inside out – or along some other vector field) is a concurrent one (in which the design cools uniformly and gels). It would seem that a concurrent process is perhaps better suited to systems for which the operation is well understood (whole spacecraft), and less well suited to systems being developed for the first time (new payloads). To propose a change to the one based on an experience with the other would be reactionary.

Much design and development of terrestrial self-contained mechatronic systems (mobile robots, aircraft, etc.) benefits from the ability to get a “second chance” to correct a design flaw discovered in service. It is usually possible to conduct field testing, revise production designs, and perform rework. Even systems that operate in extreme environments are typically retrieved if possible. In contrast, there are no second chances on orbit. All the second chances that might ever exist only exist on the ground. It has been said that many spacecraft failures arise not because of “random” component failures, but because the spacecraft was designed to fail. Therefore, representative ground testing, combined with other verification measures to go beyond the limits of that testing, is the primary means of establishing confidence in a design. It is this verification process that justifies the conservatism toward change in the staged design process, which applies from the component level up to the system as a whole.

On a different topic, the size of the system in question also results in some more lessons:

- Compact assemblies can become complicated in order to achieve the required strength. Simple part geometry is often strong, but takes up volume. If space is at a premium, then the geometry needed to achieve adequate strength while accommodating all the needed parts can become complex. This forms a compelling incentive to aim for a low part count to begin with.

- Small assemblies have a limited suite of COTS\(^\text{29}\) hardware to draw from. For example, many types of fasteners are not available in the smallest of standard sizes.

\(^{29}\) Commercial Off-The-Shelf
Finally, the design considerations for multiple units differ from those for a single unit:

- Building multiple units is yet another reason to keep the part count low. The multiplication of parts mentioned in Chapter 4 is a downside of the multi-module approach in this regard.
- A single successful batch or unit does not validate the process used to create it. Batch-to-batch and unit-to-unit variation can only be assessed after building up a sufficient sample population.
- Assembling a single workable unit is different from assembling multiple units. A manual processes that is tolerable for a single unit can become unsuitable when repeated.

### 6.3 Recommendations and suggestions for follow-on work

A few general recommendations arise from the work:

- Use additive fabrication and an appropriate material for complex, lightly-loaded parts that can tolerate the limitations associated with the selected material. The parts can be less expensive and faster to procure than equivalent ones machined and assembled from polymer or metal.
- Do not use 0-80 size screws unless absolutely necessary. The paucity of threaded inserts in this size constitutes an unwelcome design constraint.

Some suggestions for worthwhile work in the future pertaining to the analysis of the deployment mechanism are as follows:

- Improve the bearing friction model to better capture the variation in viscous friction over temperature, instead of modelling it as a change in the rolling friction coefficient.
- Compare the results of the dynamic model to telemetry from deployments incorporating the sail.
- Given the abundance of telemetry from each deployment, use regression techniques to estimate better values for those model parameters that are more uncertain than others, such as the friction and damping coefficients.
There are two overarching conclusions to this work:

- The aspects of the CanX-7 drag sail described in this thesis are designed to meet all of the critical requirements applicable to the system. The payload is ready for qualification testing and subsequent steps toward flight.
- The model of the deployment mechanism presented in this thesis reproduces a simplified picture of the behaviour of the real system. Suggestions for improvement have been proposed.
List of References


Appendix A: Parameter Values for Deployment Mechanism Model

List of Parameters

The following are the parameters values used to obtain the results presented in the analysis of the deployment mechanism in Chapter 5.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom length, nominal</td>
<td>$l_{boom}$</td>
<td>1.5</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Boom length, tests without sail</td>
<td>$l_{boom}$</td>
<td>1.24</td>
<td>m</td>
<td>For comparison against test data</td>
</tr>
<tr>
<td>Elastic modulus</td>
<td>$E_b$</td>
<td>131</td>
<td>GPa</td>
<td>[62]</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>$\nu$</td>
<td>0.3</td>
<td></td>
<td>[62]</td>
</tr>
<tr>
<td>CuBe density</td>
<td>$\rho$</td>
<td>8.36</td>
<td>$\frac{g}{cm^3}$</td>
<td>[62]</td>
</tr>
<tr>
<td>Transverse radius</td>
<td>$r_t$</td>
<td>16.75</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Boom thickness</td>
<td>$t_b$</td>
<td>0.127 [0.005]</td>
<td>mm [inch]</td>
<td>Tape sold in imperial sizes.</td>
</tr>
<tr>
<td>Boom width</td>
<td>$w$</td>
<td>19.05 [0.75]</td>
<td>mm [inch]</td>
<td>Tape sold in imperial sizes.</td>
</tr>
<tr>
<td>Transverse angle</td>
<td>$\theta_t$</td>
<td>1.137</td>
<td>rad</td>
<td></td>
</tr>
<tr>
<td>Boom mass per length</td>
<td>$\kappa$</td>
<td>0.0183</td>
<td>$\frac{kg}{m}$</td>
<td></td>
</tr>
<tr>
<td>Layer thickness of both booms</td>
<td>$s$</td>
<td>0.24</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Outer boom static length</td>
<td>$l_{outer}$</td>
<td>70.42</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Inner boom static length</td>
<td>$l_{inner}$</td>
<td>40.75</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Moment of inertia of reel</td>
<td>$I_{reel}$</td>
<td>3.36627205</td>
<td>kg mm$^2$</td>
<td>From solid model</td>
</tr>
<tr>
<td>Quantity</td>
<td>Symbol</td>
<td>Value</td>
<td>Units</td>
<td>Notes</td>
</tr>
<tr>
<td>----------------------------------</td>
<td>--------</td>
<td>---------------</td>
<td>---------</td>
<td>-----------------------------------------------------------------------</td>
</tr>
<tr>
<td>Radius of coil</td>
<td>( r_{coil} )</td>
<td>17.29</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Number of bearing per roller</td>
<td>( n_b )</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of booms</td>
<td>( n_B )</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of rollers</td>
<td>( n_r )</td>
<td>11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Radius of roller</td>
<td>( r_r )</td>
<td>1.98 [0.0781]</td>
<td>mm [inch]</td>
<td>Imperial bearings used.</td>
</tr>
<tr>
<td>Roller assembly bearing bore</td>
<td>( d_r )</td>
<td>1.19 [3/64]</td>
<td>mm [inch]</td>
<td>Imperial bearings used.</td>
</tr>
<tr>
<td>Axle bearing bore</td>
<td>( d_a )</td>
<td>3.57 [9/64]</td>
<td>mm [inch]</td>
<td>Average of both axle bearings. Imperial bearings used.</td>
</tr>
<tr>
<td>Sliding coefficient of friction between tape and structure</td>
<td>( \mu_{ts} )</td>
<td>0.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Static coefficient of friction between tape and structure</td>
<td>( \mu_{ts} )</td>
<td>0.8</td>
<td></td>
<td>Static coefficient assumed to be double the sliding coefficient.</td>
</tr>
<tr>
<td>Initial sail resisting force</td>
<td>( F_{S,i} )</td>
<td>0.2</td>
<td>N</td>
<td>From coarse measurements.</td>
</tr>
<tr>
<td>Damper coefficient</td>
<td>( a_1 )</td>
<td>0.0068904</td>
<td>N m</td>
<td>See below.</td>
</tr>
<tr>
<td>Damper coefficient</td>
<td>( a_2 )</td>
<td>( 0.93264 \times 10^{-3} )</td>
<td>( \frac{\text{kg m}^2}{\text{s}^2} )</td>
<td>See below.</td>
</tr>
<tr>
<td>Damper coefficient</td>
<td>( a_3 )</td>
<td>( -0.11931 \times 10^{-3} )</td>
<td>( \frac{\text{kg m}^2}{\text{s}^2 , ^\circ\text{C}} )</td>
<td>See below.</td>
</tr>
<tr>
<td>Quantity</td>
<td>Symbol</td>
<td>Value</td>
<td>Units</td>
<td>Notes</td>
</tr>
<tr>
<td>--------------------------------------</td>
<td>--------</td>
<td>---------------</td>
<td>------------------</td>
<td>-------------</td>
</tr>
<tr>
<td>Damper coefficient</td>
<td>$a_4$</td>
<td>$-4.7225 \times 10^{-7}$</td>
<td>$\text{kg m}^2\text{s}^{-1}$</td>
<td>See below.</td>
</tr>
<tr>
<td>Damper coefficient</td>
<td>$a_5$</td>
<td>$6.6290 \times 10^{-7}$</td>
<td>$\text{kg m}^2\text{s}^{-2}\text{C}^{-2}$</td>
<td>See below.</td>
</tr>
<tr>
<td>Bearing friction model coefficient, starting case</td>
<td>$g_1$</td>
<td>0.010864</td>
<td></td>
<td>See below.</td>
</tr>
<tr>
<td>Bearing friction model coefficient, starting case</td>
<td>$g_2$</td>
<td>$-0.051003$</td>
<td>$\frac{1}{\text{C}}$</td>
<td>See below.</td>
</tr>
<tr>
<td>Bearing friction model coefficient, running case</td>
<td>$g_1$</td>
<td>0.0043596</td>
<td></td>
<td>See below.</td>
</tr>
<tr>
<td>Bearing friction model coefficient, running case</td>
<td>$g_2$</td>
<td>$-0.048478$</td>
<td>$\frac{1}{\text{C}}$</td>
<td>See below.</td>
</tr>
</tbody>
</table>
**Friction Coefficient Model**

A common model for the friction torque $M$ in a bearing incorporates the load $F$, the bearing bore $d$ (also called inner diameter), and a friction coefficient $\mu$:

$$M = \mu F \frac{d}{2}$$

The model used here for the bearing friction coefficient $\mu$ depends on the temperature $T$, and uses an exponential function with empirical coefficients $g_1$ and $g_2$. It has the form:

$$\mu_r = g_1 e^{g_2 T}$$

To determine values for the coefficients $g_{1,2}$, the model is fit to friction coefficient data over the temperature range of interest. The data are obtained from standard tests of the lubricant as well as typically-accepted values based on the type of bearing.

At cold temperatures, the manufacturer’s datasheet gives ASTM D1478 test values for the friction torque $M$ in a test bearing packed with the lubricant in question. It is possible to calculate values for the coefficient of friction $\mu$ in each case by using the friction torque model just stated, and a description of the test setup to obtain values for the bore $d$ and load $F$. Table A2 lists the test data for the lubricant in question. Figure A1 plots the resulting coefficients of friction.

At room temperature, Table A3 lists typical values for the coefficient of friction $\mu$ for the type of bearing in question.

Using these data for cold and room temperatures, the exponential friction coefficient model is fit to the data to obtain values for the empirical coefficients $g_{1,2}$. The data are weighted to give a better fit over the temperature range of interest, $-40^\circ C$ to $80^\circ C$. Table A1 (at the beginning of this appendix) lists the resulting values for the coefficients $g_{1,2}$. Figure A1 plots the model and data together.
Table A2: ASTM D1478 test data for roller assembly bearings.

<table>
<thead>
<tr>
<th>Temperature [°C]</th>
<th>Friction Torque [gram-force cm]</th>
<th>Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>−100</td>
<td>1430</td>
<td>Starting</td>
</tr>
<tr>
<td>−62</td>
<td>585</td>
<td></td>
</tr>
<tr>
<td>−100</td>
<td>637</td>
<td>Running</td>
</tr>
<tr>
<td>−62</td>
<td>228</td>
<td></td>
</tr>
</tbody>
</table>

Test load $F = 20.1$ N

Test bore $d = 20$ mm

Table A3: Standard friction coefficient values for radial ball bearings.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room temperature bearing friction coefficient, starting case</td>
<td>$\mu$</td>
<td>0.003</td>
<td></td>
<td>[66]</td>
</tr>
<tr>
<td>Room temperature bearing friction coefficient, running case</td>
<td>$\mu$</td>
<td>0.0015</td>
<td></td>
<td>Starting coefficients are approximately twice as high as running coefficients [67].</td>
</tr>
</tbody>
</table>

Figure A1: Friction coefficient data and model.
**Damper Model**

The damper torque varies with speed and temperature. To predict the damper torque over the operating range of interest, a bivariate formula is fit to performance data from the damper manufacturer. The formula depends on the speed $\omega$, temperature $T$, and several empirical coefficients $a_{1...5}$. It has the form:

$$\tau_d(T, \omega) = a_1 + a_2 \omega + a_3 T + a_4 \omega^2 + a_5 T^2$$

Table A1 lists values for the coefficients obtained from the manufacturer’s data. The plots in Figure A2 show the manufacturer’s data superimposed over the model fit to the data.
Three views of the damper model and data: Over the operating speed range of the drag sail module (above), and over the extent of the data (below).