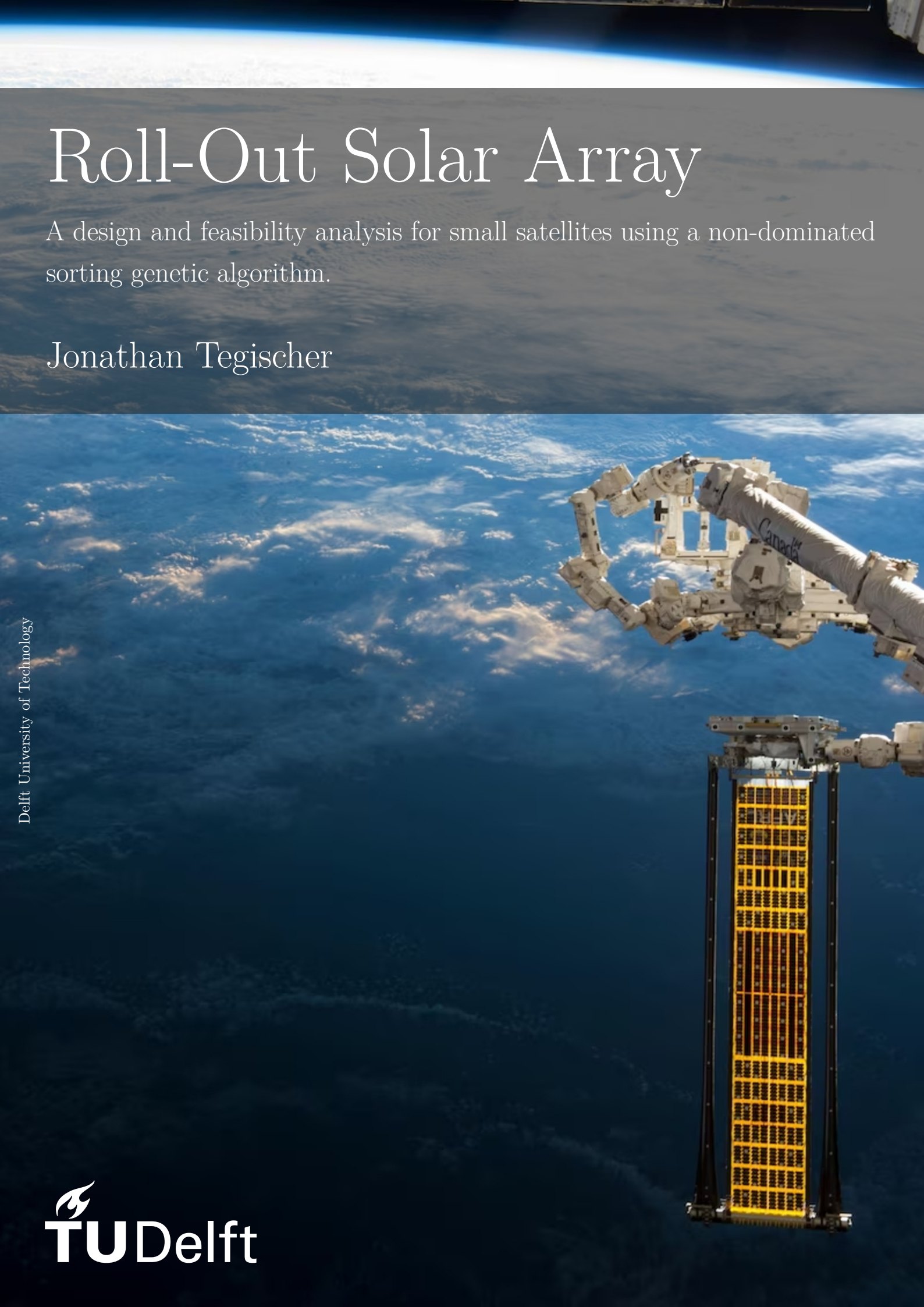


Roll-Out Solar Array

A design and feasibility analysis for small satellites using a non-dominated sorting genetic algorithm.

Jonathan Tegischer



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by

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Cover photo: Roll-out solar array experiment by Redwire and NASA.

Preface

The following thesis details the work carried out as part of a collaboration between Delft University of Technology and Revolv Space. The thesis was a challenging process, but rewarding in the end. All of this would not be possible if it were not for several people. Firstly, I would like to thank Jasper Bouwmeester for guiding me along this journey. Michał, thank you for dedicating your time and effort to mentoring me the whole way with your invaluable advice and insights. I'd also like to extend my gratitude to the whole team at Revolv Space for their encouragement and assistance over the last months. A special thank you to Lara for supporting me from the beginning and for being there during the long, studious days. Last but not least, I'd like to thank my family, Mamma, Papa, Raphy, for their everlasting support and unconditional love. I could not be more grateful.

Jonathan Tegischer
Delft, September 2024

Summary

The rapidly increasing utilization of small satellites has led to a growing demand for improved solar array technologies. Traditional solar arrays are limited by their mass and volume constraints, and thus are struggling to meet the increased demands. This feasibility study explores the potential of roll-out solar arrays as a solution to enhance the power generation capabilities of small satellites. These innovative systems are characterised by their unfurling method of deployment, and have demonstrated improved performance compared to traditional solar arrays. By analysing the feasibility of designing a roll-out solar array for small satellites, it is hoped that the operational capabilities of small satellites can be enhanced.

A comprehensive literature study revealed a significant performance gap in high power-to-mass ratios for satellites with a power range of $300 - 1000\text{ W}$, leading to the development of a roll-out solar array design specifically for this power range. The study combined an analytical model with the non-dominated sorting genetic algorithm (NSGA-II), an efficient multi-objective optimisation algorithm, to generate a set of optimal designs by maximising the power-to-mass ratio and stowed power density, with the latter parameter being another key performance indicator for roll-out solar arrays. The analytical model incorporated the core design elements of a roll-out solar array like deployment mode, deployment mechanism, solar cell type, and stiffening method. A multitude of constraints ensured the feasibility of the designs, while several verification and validation tests were conducted to confirm the accuracy of the analytical model.

The optimisation results revealed that a variety of design combinations exist that could outperform the state of the art solar array capabilities. Optimal designs demonstrated power-to-mass ratios exceeding 145 W/kg and stowed power densities over 55 kW/m^3 . These designs would deliver over double the power-to-mass ratio of traditional solar arrays in the $300 - 1000\text{ W}$ range, while remaining comparable to the $120 - 140\text{ W/kg}$ state of the art performance of arrays outside of this range. Furthermore, optimal designs offer up to 40% increases in stowed power density compared to the 40 kW/m^3 state of the art benchmark. Despite the promising gains, the study reveals some key limitations to roll-out solar arrays. These are the increased development costs, the higher risks, and their bulky geometry being potentially inefficient for smallsats. Ultimately, the study provides for a strong foundation for the development of a roll-out solar array. The results confirm the potential of such a system, and future research is recommended to solidify the feasibility of this innovative technology.

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Nomenclature

Abbreviations

Abbreviation	Definition
BOL	Beginning Of Life
CAoS	Chinese Academy of Science
CFRP	Carbon Fiber Reinforced Polymer
CIGS	Copper Indium Gallium Selenide
COBRA	Composite Beam Roll-out Array
COTS	Commercial Off The Shelf
CTM	Collapsible Tube Mast
DART	Double Asteroid Redirection Test
ECSS	European Cooperation for Space Standardization
EOL	End Of Life
ESA	European Space Agency
ESPA	EELV Secondary Payload Adapter
FAA	Federal Aviation Administration
FEM	Finite Element Method
FOSA	Fold Out Solar Array
FPC	Flexible Printed Circuit
FRUSA	Flexible Rolled Up Solar Array
GaAs	Gallium Arsenide
GEO	Geostationary Orbit
HTJ	Heterojunction
ISS	International Space Station
LEO	Low Earth Orbit
NASA	National Aeronautics and Space Administration
NSGA	Non-dominated Sorting Genetic Algorithm
PCB	Printed Circuit Board
(i)ROSA	(ISS) Roll Out Solar Array
SADA	Solar Array Drive Assembly

Abbreviation	Definition
SAW	Solar Array Wing
SMPC	Shape Memory Polymer Composite
(BI-)STEM	(Bistable-) Storable Tubular Extendible Member
TASSA	Tape Spring Solar Array
TJ	Triple Junction
TRAC	Triangular Rollable and Collapsible

Symbols

Symbol	Definition	Unit
$a_{y/z}$	Acceleration along the x or y axis	$[m/s^2]$
A	Area	$[m^2]$
A_l	Longeron cross sectional area	$[m^2]$
b	Tape width	$[m^2]$
C_1	Correction coefficient	$[-]$
C_f	Coefficient of friction	$[-]$
c	Furthest distance from neutral axis	$[kg]$
D	Flexural rigidity	$[Nm^2]$
$D_{x/y}$	Flexural rigidity about the x or y axis	$[Nm^2]$
d	Diameter	$[m]$
E	Youngs modulus	$[Pa]$
$E_{x/y}$	Youngs modulus along the x or y axis	$[Pa]$
F	Force	$[N]$
f	Natural frequency	$[Hz]$
G	Shear modulus	$[Pa]$
H	Stowed height	$[-]$
I	Second moment of area	$[m^4]$
$I_{x/y/z}$	Mass moment of inertia about the x , y , or z axis	$[kg\ m^2]$
$I_{xx/yy/zz}$	Second moment of area about x , y , or z axis	$[m^4]$
J	Mass moment of inertia	$[kg\ m^2]$
J_s	Distributed mass moment of inertia	$[kg\ m^2]$
K	Torsional constant	$[m^4]$
k	Gear ratio	$[m^4]$
k_f	Factor of safety	$[-]$
L	Length	$[m]$

Symbol	Definition	Unit
M	Moment	$[Nm]$
m	Mass	$[kg]$
N_{cr}	Number of compression rollers	$[-]$
n	Number of	$[-]$
L	Length	$[m]$
l	Length	$[m]$
P	Power	$[W]$
P	Force	$[N]$
R	Radius	$[m]$
r	Radius	$[m]$
$r_{x/y}$	Radius about x or y axis	$[m]$
S	Solar flux	$[W/m^2]$
T	Torque	$[Nm]$
t	Thickness	$[m]$
U	Perimeter	$[m]$
U	Strain energy	$[J]$
V	Volume	$[m^3]$
v	Poisson ratio	$[-]$
w	Width	$[m]$
w	Distributed load	$[N/m]$
α	Dynamic overshoot factor	$[-]$
δ	Deflection	$[m]$
ϵ	Strain	$[-]$
η	Efficiency	$[-]$
ζ	Volume scaling factor	$[-]$
θ	Angle	$[rad]$
$\ddot{\theta}_{x/y/z}$	Angular acceleration about the x , y , or z axis	$[rad/s^2]$
$\kappa_{x,y}$	Curvature about x or y axis	$[-]$
ρ	Density	$[kg/m^3]$
σ	Stress	$[Pa]$
τ	Shear stress	$[Pa]$
ξ	Power-to-mass ratio	$[W/kg]$
ψ	Stowed power density	$[kW/m^3]$

1

Introduction

In recent decades, the field of satellite technology has witnessed a significant increase in the utilization of small satellites, also known as smallsats [1]. Characterised by their reduced mass and size, they have opened up possibilities for new research and expanded the accessibility to space to many entities beyond the governmental sector. Smallsats, defined as any satellite below 600 *kg*, have become so popular that they comprised 96% of all satellites launched in 2022, albeit largely due to many launched satellites pertaining to constellations such as Starlink and OneWeb [1]. With this rise in their usage, their operational requirements have increased as well, and as a result, the demand for improved power generation systems has grown [2]. Current solar array technologies for small satellites are limited by their mass and volume, constraining their power generation capabilities and limiting mission operations, highlighting the scarcity of efficient power generation solutions tailored to the unique requirements of smallsats. One promising solution that tackles these increasing power demands are roll-out solar arrays. These innovative solar arrays, characterised by their unfurling deployment method, were first flown in 1971 but have since been incorporated into only a few missions [3]. Despite their limited flight heritage, roll-out solar arrays have demonstrated significant performance improvements due to their efficient storage and use of lightweight materials. However, existing roll-out solar arrays have predominantly been designed for larger satellites, unveiling a gap in the market for such systems tailored to smallsats.

Addressing this gap, the aim of the following study is to analyse the feasibility of designing a roll-out solar array for small satellites. Developments of lightweight composite materials, flexible solar cells, and deployment mechanisms suggest that a roll-out solar array could be designed for smallsats. By integrating these novel technologies into a deployable structure, a roll-out solar array has the potential to offer higher power-to-mass ratios and stowed power densities, thereby increasing operational capabilities of small satellites and enabling more power demanding missions. Despite their promising performance increases, roll-out solar arrays have complex design architectures, and concerns over their added value requires a study to determine their viability.

Through a comprehensive feasibility analysis that includes a conceptual design evaluation, this study seeks to propose a solution that could significantly increase the power generation capabilities for small satellites. With this in mind, the main research question is defined:

How can a roll-out solar array be designed for small satellites, such that it outperforms current solar array technologies?

A set of subquestions are derived that allow for a clearer exploration and analysis. These are presented as follows:

1. *What are the requirements for solar arrays in low earth orbit?*
2. *What solar cell technology would make a roll-out solar array for small satellites feasible?*
3. *How can the deployment of the roll-out solar array be achieved, while ensuring structural stiffness post deployment?*
4. *How does the power-to-mass ratio and stowed power density of a roll-out solar array design compare with that of current solar array technologies for small satellites?*
5. *What are the limitations associated with a roll-out solar array, and in what ways could these limitations impact the evaluation of its feasibility?*

By answering these questions, it is expected that a set of roll-out solar array designs will be proposed which outperform the currently available solar arrays. These will be composed of certain core design elements which ensure that the deployment sequence is possible, and that operational requirements thereafter are met. Nevertheless, it is possible that due to the complex architecture, the limitations associated with a roll-out solar array may outweigh the performance gains.

The report commences by describing some background information on the key topics of this thesis in chapter 2, followed by a detailed definition of the problem in chapter 3 which also presents the requirements. Following this, chapter 4 focuses on the concept generation of the core design elements of a roll-out solar array. Then, chapter 5 defines the entire analytical model and how it is integrated with a multi-objective algorithm to form an optimisation model. Next, chapter 6 presents all the tests and analyses made to verify and validate the analytical model, followed by chapter 7 which presents all the results of the optimisation model and makes a comprehensive comparison with traditional solar arrays. Finally, the conclusion and recommendations are presented in chapter 8, wherein the research question is answered and suggestions for future work are proposed.

2

Background

The following chapter explores the theoretical background essential for understanding the context and framework of this study. It begins by providing an overview of the current ecosystem of small satellites in section 2.1, followed by an exploration of the state of the art rigid-, foldable-, and roll-out solar arrays in section 2.2.

2.1. Small satellite ecosystem

Ever since the first human made object was put into space with the historic launch of Sputnik 1 in 1957, satellite development has showcased a significant variety in terms of size and mass [4]. However, recent trends show that small satellites, also known as smallsats, are the preferred satellite solution. Developments in miniaturization technologies, reduced launch costs, and increased demand in a multitude of industries has caused a surge in the deployment of smallsats. To put into perspective how fast the smallsat market has grown in the past decade, 2013 saw 130 smallsats put into space, whereas 2022 saw 2,402 smallsats launched, demonstrating over a 1,700% increase. Likewise, the share of the smallsat market has also shown significant increases. While 2013 showed a smallsat share of 61% of all satellites, 2022 had the satellite launch landscape dominated by smallsats, which accounted for 96% of the total 2,510 satellites launched that year, highlighting the increasing preference for smaller, more cost effective satellites for today's space missions. Even if constellation satellites of Starlink and OneWeb are disregarded, the share of smallsats in 2022 still amounts to 84%. Forecasts also predict that by 2032, the market size of smallsats would increase five fold from 2022, indicating the continued growth of this industry [5]. [1]

A notable trend that is present for small satellites is the drastic increase in power demands over the past years [2]. This is largely driven by more capable payloads and the increasing popularity of using propulsion systems on smallsats. Nevertheless, characteristics of smallsats differ quite significantly from larger satellites, and in general have higher volume constraints due to their small size, but in turn

also have significantly lower development times and development costs due to readily available COTS technologies [6]. Furthermore, smallsats tend to have fixed solar arrays as opposed to sun tracking ones, and additionally are commonly body mounted. Due to this, the power generation of smallsats is much more limited with respect to larger satellites. All of these limitations of smallsats have created increased demands for innovative solutions in order to increase their power generation capabilities. To meet the growing demands, innovative technologies are being explored by satellite manufacturers.

2.2. State of the art solar arrays

In the rapidly evolving world of solar technology, state of the art solar arrays are redefining the benchmarks for satellite payload capabilities. By leveraging new materials and innovative designs, solar arrays are setting new standards in power-to-mass ratios and stowed power densities, significantly improving satellite functionality and efficiency. The following section defines the state of the art solar arrays that are available on the market, giving a focus on rigid, foldable, and roll-out solar arrays in subsection 2.2.1, subsection 2.2.2, and subsection 2.2.3, respectively. Except for the roll-out solar arrays, the following section does not provide an exhaustive list of all solar arrays, and instead is intended to give an overview of what the currently available state of the art technologies are. A comparison of these solar array technologies is given in subsection 2.2.4.

2.2.1. Rigid solar arrays

The most common types of solar arrays that are used on satellites are rigidly deployed or body mounted ones. For these types of solar arrays, the solar cells are usually placed on panels made of printed circuit boards (PCB), carbon fiber reinforced polymers (CFRP), or aluminium [7]. Power generation systems are also typically designed to target high power-to-mass ratios in order to minimize mass, and as the size of a satellite decreases, volume considerations become more relevant. Table 2.1 presents a few of the state of the art solar arrays currently available on the market, as offered by various leading space companies. Since smallsats include power generation systems with a large range of power outputs, the table includes a few state of the art solar arrays for various power output ranges. Results show that the average power-to-mass ratio of these rigid solar arrays is 65 W/kg . The only outliers are the solar arrays from MMA Design, which have exceptionally high power-to-mass ratios, with the highest being 120 W/kg .

Table 2.1: State of the art rigid solar arrays, arranged in ascending order of power.

Company	Name	Type	Power-to-mass ratio (W/kg)	Power output (W)	Ref.
ISISPACE	6U	Body mounted	57	17	[8]
DHV	6U	Body mounted	64	19.3	[9]
Endurosat	8U	Body mounted	51	24	[10]
MMA Design	zHaWK	Rigid deployed	102	36	[11]
EXA	DMSA 6U	Rigid deployed	94	60	[12]
MMA Design	eHaWK	Rigid deployed	120	72	[13]
MMA Design	ESPA	Rigid deployed	62	100	[14]
Pumpkin Inc.	DCSA	Rigid deployed	44	375	[15]
DHV	MDSAW	Rigid deployed	57	697	[15]
Redwire	HiPASS	Rigid deployed	59	1000	[16]
Sparkwing	/	Rigid deployed	52-73	100-1000	[17]
Lockheed Martin	SSA	Rigid deployed	55	175-1080	[18]

2.2.2. Foldable solar arrays

A foldable solar array comprises of a thin, flexible solar blanket that is deployed from its stowed configuration in a folding manner. While the solar cells are not put under bending loads, the blanket that they are placed on is fully flexible, and therefore allows for unique folding techniques. A notable example of a foldable array is the Solar Array Wing (SAW) of the International Space Station (ISS), with each saw generating over 31 kW of power at a power-to-mass ratio of 28 W/kg [19]. This solar array uses an articulated mast to deploy and then stiffen the solar blanket, depicted in Figure 2.1. A newer generation of foldable solar arrays is the Lockheed Martin Multi-mission Modular Solar Array (MM Solar Array). The cells in this design are again placed on a thin flexible blanket which unfolds in a similar manner as depicted in Figure 2.1. However, instead of an articulated mast being used as the deployment and stiffening structure, a boom is used instead. Furthermore, the MM Solar Array is designed with modularity in mind, meaning they can be easily adapted to fit spacecrafts with power requirements anywhere between 6 – 12.5 kW [20]. It was launched on three heavy ($>5000\text{ kg}$) GEO satellites [21]. A smaller foldable solar array for cubesats has also been developed by Exoterra, which uses a similar deploying boom method as the MM Solar Array to deploy and stiffen the blanket [22]. Their scalable Fold-Out Solar Array (FOSA) is able to generate 75 – 300 W with a power-to-mass ratio of 140 W/kg [22].

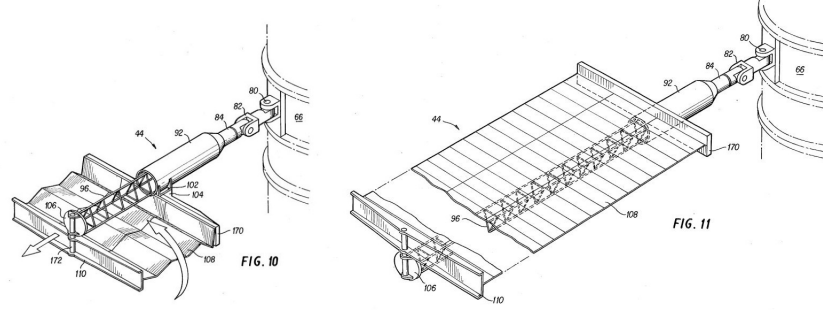


Figure 2.1: Deployment method of a foldable solar array. In this design, a truss is extended in the middle, which deploys and stiffens the blanket. [23].

Another type of folding technique follows the Chinese fan. Northrup Grumman's UltraFlex, used in various missions such as the Cygnus Spacecraft (1750 W) and Mars Phoenix lander, employs a considerably space efficient design [24]. As seen in Figure 2.2, it deploys from a thin structure to a fully circular solar array. A bigger version of this, called the MegaFlex, is predicted to be able to reach power-to-mass ratios of up to 200 W/kg for specific missions, although no such solar array has yet been used in space [25]. While the power-to-mass ratio is very promising, an issue with this technology is the deployment reliability, as in two separate missions one of the arrays failed to deploy [26]. This highlights the increased risk that comes with more complex deployable solar arrays.



Figure 2.2: Deployment of the UltraFlex foldable solar array [24].

The final type of folding technology which has had more development over the recent years is the origami-style folding technique. This technique involves a complex sequence of folding that minimises the packing volume, with the Miura fold being a common example illustrated in Figure 2.3. Due to the complex nature of origami, there can be a multitude of folding methods, but two companies have come up with designs incorporating this technique. DCUBED has developed a 100 W origami solar array which has a power-to-mass ratio of 83 W/kg [27], while the SolarCube of Astradyne offers a 70 W solar array with a 88 W/kg power-to-mass ratio [28]. These designs, while having significantly high power generation capabilities, do not show competitive power-to-mass ratios, likely due to their complexity requiring additional mass to allow for the folding technique to work. A design for high power ($>150\text{kW}$) called the HanaFlex has been proposed, allowing for a power-to-mass ratio of 120 W/kg and stowed power density of $25 - 40 \text{ kW/m}^3$, but no such system has yet been put into practice [29].

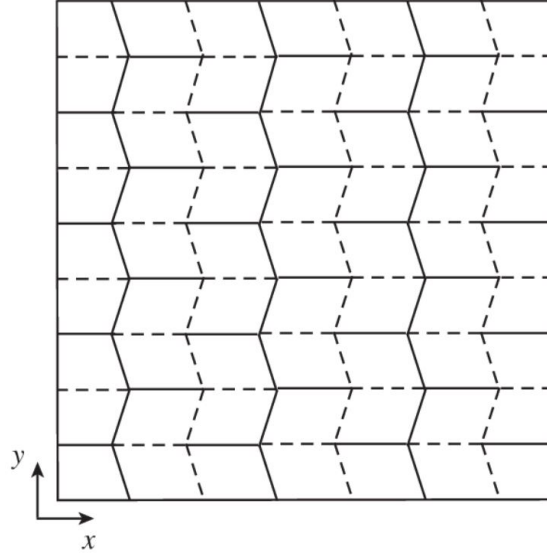


Figure 2.3: Miura fold origami technique [30].

2.2.3. Roll out solar arrays

A roll-out solar array is a type of solar array technology which employs a flexible solar blanket that unfurls from a cylindrical mandrel. Similar to the foldable arrays, roll-out solar arrays employ flexible solar cells, but instead of being folded, they are rolled around a cylindrical mandrel for stowage. The first publication of a roll-out type solar array was made in 1969, in which it was theorised that this new technology could have mass and volume savings over traditional rigid solar arrays [31]. Shortly after this, the first roll-out solar array was designed by the Hughes Aircraft Company and put into orbit in 1971 with the launch of the Space Test Program 71-2 mission [3]. This roll-out solar array, called the Flexible Rolled Up Solar Array (FRUSA), proved the feasibility of this new technology, and implemented the design as seen in Figure 2.4. The design consisted of two solar cell blankets which unfolded from a single mandrel in the center. Extendable BI-STEM booms (explained in subsection 4.5.1) were used to deploy and stiffen the solar array. This design had a power-to-mass ratio of 40 W/kg [3], and a BOL power generation of 1500 W [32]. Considering that the efficiency of flexible solar cells at the time of launch was 10%, a FRUSA with today's technology would reach a power-to-mass ratio of $>80 \text{ W/kg}$ [33]. ESA and General Electric also made proposals for more roll-out solar arrays during this time [3] [34].

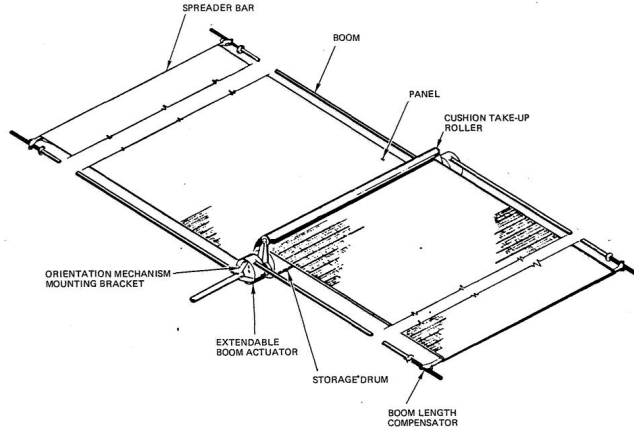


Figure 2.4: The Hughes' FRUSA, the first roll-out solar array deployed in space [3].

The first roll-out solar array that was used for a significant mission was for the Hubble space telescope. The array used a double solar blanket design similar to the FRUSA, giving it a power-to-mass ratio of 73 W/kg , and providing more than 4000 W to the telescope [35]. Using a roll-out design was more of a requirement as the tight mass and volume constraints did not allow for a rigid solar panel to be used [36]. Unfortunately, the flexible technology was sensitive to temperature changes caused by the space environment, and as a result produced vibrations on the telescope which affected the required stability for observations [37]. In 1993, a stiffer roll-out solar array which was also less prone to thermal cycling effects was put in place of the old one. As this solar array degraded and the power requirements increased with new instruments, the roll-out solar arrays were replaced yet again by a set of rigid solar arrays in 2002 [37].

Since then, a handful of new roll-out solar arrays have been developed. The most notable one of them is the Roll-Out Solar Array (ROSA) by Redwire. This roll-out solar array uses stored strain energy to deploy by itself, meaning that no additional mechanisms or actuators are required [38]. This significantly improves the reliability of the system and decreases design complexity. Deployable booms are used on the sides of the array as its primary stiffening structure and method for tensioning the solar blanket. The power-to-mass ratio of the ROSA is between $100 - 120 \text{ W/kg}$, and the stowed power density is 40 kW/m^3 [39]. Its notable feature comes from its considerable flight heritage. Since its demonstration mission in 2017, 6 of these ROSAs have been deployed on the ISS to increase its power generation by $20 - 30\%$ [40], and a pair of ROSAs were also used as the solar arrays for NASA's Double Asteroid Redirection Test (DART) spacecraft, see Figure 2.5. Its modularity means that a single ROSA can be configured to provide anywhere from $1 - 30 \text{ kW}$, allowing it to be also used for smallsats [39]. The flight heritage is set to continue as more ROSAs are currently under development for the power and propulsion element of the Lunar Gateway and Blue Origins Blue Ring [39].

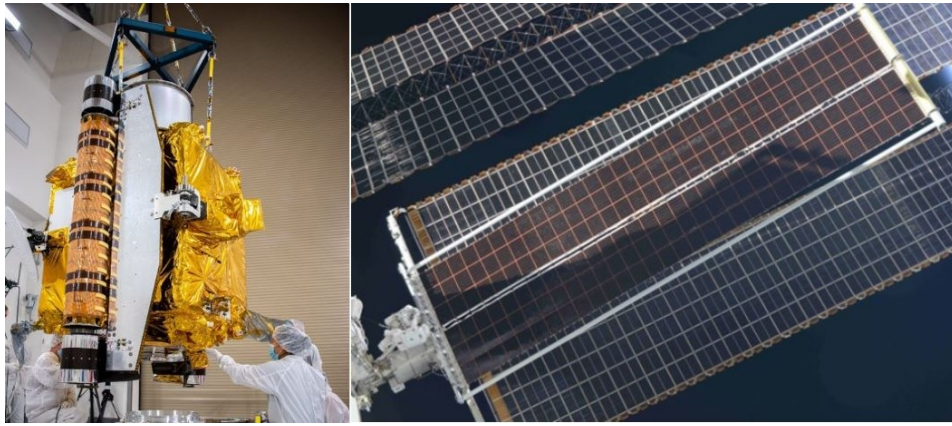


Figure 2.5: Stowed ROSA for the DART mission (left), and fully deployed ROSA used on the ISS (right). [41]

The latest roll-out solar array that completed its demonstration mission is Momentus' TApe Spring Solar Array (TASSA) [42]. This is one of the smallest roll-out solar array that has been designed to date, with the company claiming that the solar blanket is wrapped around a 4 inch mandrel, as opposed to the 12 inch ones used by other roll-out solar arrays [43]. Momentus stated that in the future, their TASSA could be used as the solar arrays for its Vigoride satellite, a transporter satellite with a wet mass without payload of 250 *kg*, making it a viable roll-out solar array for smallsats [44]. Their patented tape spring design allows for the removal of stiffening booms, frames, and tensioning devices as the design ensures both lateral and longitudinal stability for itself [45]. Without the need of these additional structures and mechanisms, the TASSA becomes inherently more reliable as there are less parts that could fail, and with less parts the whole system becomes more lightweight. While the stored potential energy of the tape spring is said to be enough to deploy the panel unassisted and keep it stiff without additional booms or structures, a motor was used in the demonstration mission such that it could be rolled back into its stowed configuration and have it tested for multiple iterations [42]. The most apparent limitation of this design is the volume it takes in its stowed configuration. As seen in Figure 2.6, the TASSA is not fully rolled around its mandrel, and instead has a bit of its array already deployed with a supporting structure. This causes the stowed power density to be significantly lower than other roll-out solar array designs, and could be problematic when requirements on the volume are constrained. Additionally, the curvature of the solar blanket causes the array to be less efficient than a flat one of the same size due to the incident light not being completely perpendicular to the panel. In subsection 6.5.1, this reduction in power is calculated to be approximately 9%, which is not insignificant. Figure 2.6 also shows two deployed TASSAs that were used in the demonstration mission, displaying clearly the concave structure of the solar blanket.

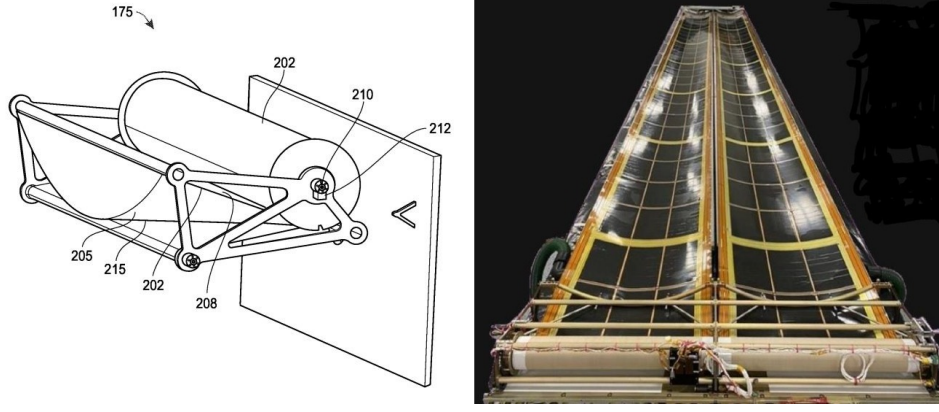


Figure 2.6: Patented design of the TASSA (left) [45], and deployment test of the TASSAs used in the Vigoride 6 demo mission (right) [46].

A design similar to the TASSA was also designed by Alliance Spacesystems, and then by SolAero. The patented design, called the COmposite Beam Roll-out Array COBRA, also features the tape spring concept as the stiffening method for the solar blanket, as seen in Figure 2.7. A design of 1 kW is predicted to have a stowed power density of $20.2 \text{ kW}/\text{m}^3$ and a power-to-mass ratio of $174 \text{ W}/\text{kg}$, displaying much higher performance characteristics than other roll-out solar arrays. Nevertheless, since 2019, there have been no updates on the development of COBRA, and no flight heritage has been achieved. [47]

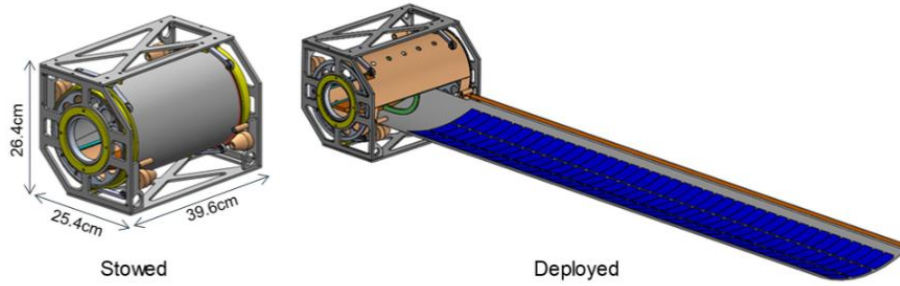


Figure 2.7: Design of SolAero's 300 W COBRA concept [47].

Finally, Thales' SolarFlex, currently under development, is a new roll-out solar array that is set to launch on the ASTRA-1Q GEO satellite in 2024 [48]. This solar array employs a new design which incorporates a "double facing tape spring" deployment mechanism, in which the tape spring is bent into two [49]. This concept has been patented by Thales, and works in the way depicted in Figure 2.8, wherein the tape spring is bent in two, fixed at one end of the tape spring, and uncoils from the other end. As the tape spring lengthens, the solar blanket attached to the rotating tape spring unfurls from its cylindrical mandrel. The use of this bent double tape spring concept has a significantly higher stiffness of the whole array when compared to the stiffness of a roll out solar array which uses a single boom on each side of the array [49]. The fully deployed SolarFlex can be seen in Figure 2.8. An issue

that occurred during the design of this solar array was that thermoelastic tests caused considerable deformations and twist in the carbon fibre tape spring [49]. This was due to the tape spring being made of a very thin layer of carbon fibre which was susceptible to thermal deformations. This issue was eventually overcome by designing a new lay-up method for the material. A version of the SolarFlex that was used during testing had a mass of 103 *kg* and a power generation of 11.8 *kW*, giving it a power-to-mass ratio of 115 *W/kg*, considerably higher than the power-to-mass ratio of rigid deployable solar arrays [49]. This value should only be taken as a rough indication of its performance, as 7 years of development have since passed, and the mass considered only the solar blanket and tape spring, meaning the power-to-mass ratio is expected to be different. Currently, the SolarFlex is planned to only be used on GEO satellites, meaning that the design has been optimised for heavy satellites with high power requirements.

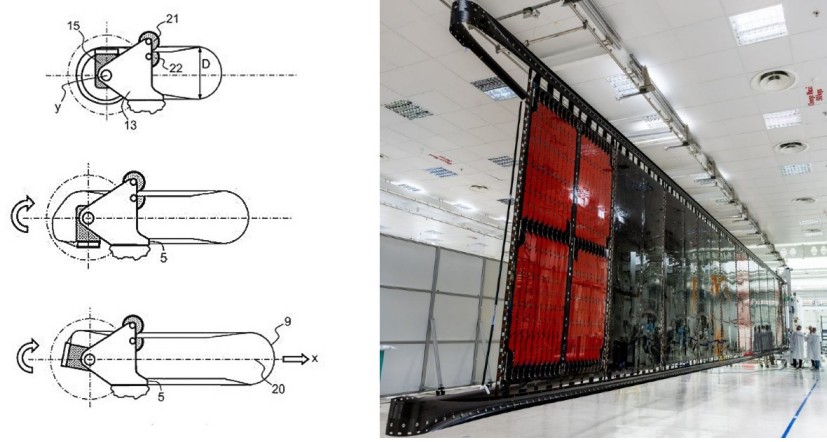


Figure 2.8: Deployment mechanism of the SolarFlex (left) [50], and fully deployed SolarFlex (right) [51].

2.2.4. Comparison and literature gap

Having collected all the data for the state of the art solar arrays, a comparative graph can be made to visually contrast the various types of arrays, highlighting their performance differences and revealing gaps in the market. When analysing solar array technologies for smallsats, a key parameter to use for comparisons is the power-to-mass ratio. This directly incorporates various performance characteristics and shows how efficiently a solar array can generate power relative to its mass, which is critical for space missions where both power supply and mass are limiting factors. Figure 2.9 tabulates the solar arrays by their power and power-to-mass ratio. Note: some solar arrays such as the TASSA and MM Solar Array are not included due to a lack of available data. Also, conceptual solar arrays which have only been proposed in literature are distinguished with a square marker instead of a circular one.

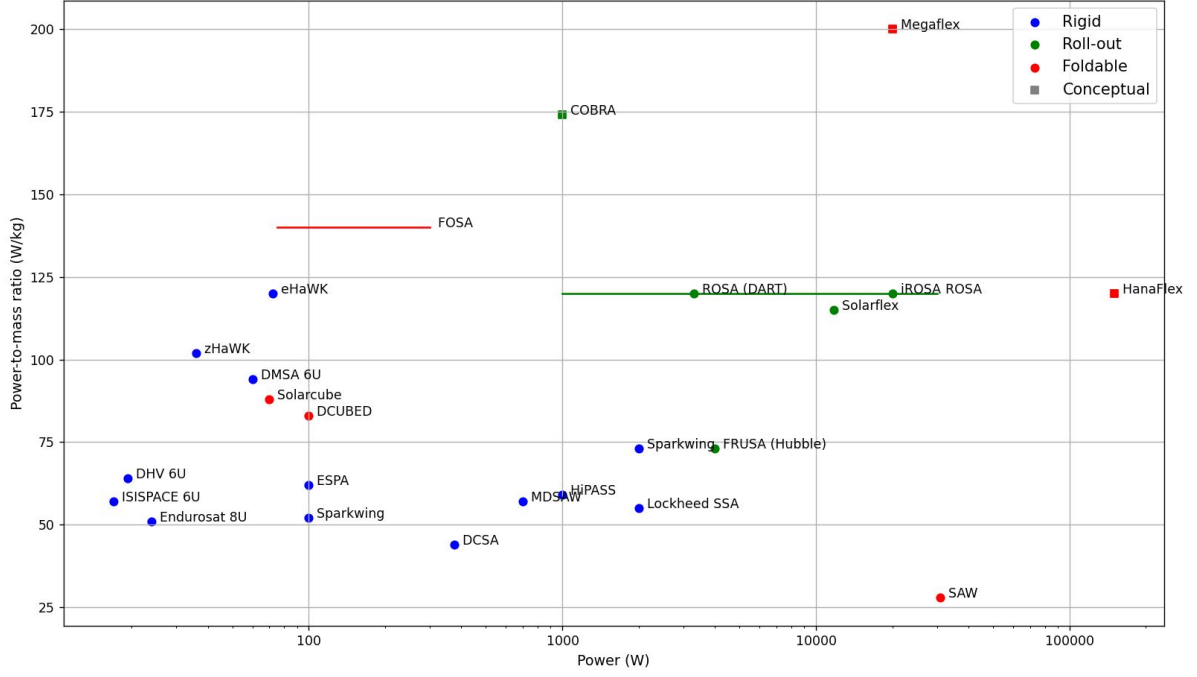


Figure 2.9: State of the art solar array comparison.

From the graph, some insightful observations can be made. Firstly, it is evident that rigid solar arrays in general have lower power-to-mass ratios compared to other types. Other than the eHawk, zHawk, and DMSA 6U designs, rigid solar arrays have a power-to-mass ratio less than 75 W/kg . On the other hand, foldable arrays have a wide range of power-to-mass ratios, although the SAW is old and outdated. Excluding the outdated FRUSA, roll-out solar arrays show promising performance as they cluster around the 120 W/kg mark. There are two designs which offer very high power-to-mass ratios, namely the COBRA and MegaFlex. These however are both just proposed in literature, and thus are not applicable for the comparison with existing solar arrays. From the designs which have undergone the complete design process and are qualified for space, Exoterra's FOSA appears to be the design with the highest power-to-mass ratio (140 W/kg). This design however only accommodates satellites with a solar array power requirement from $75 - 300 \text{ W}$. On the other hand, Redwire's ROSA offers a slightly lower power-to-mass ratio of 120 W/kg , but is scalable from $1 - 30 \text{ kW}$ size, meaning that designing a roll-out solar array for power application above 1 kW is impractical. This leaves a gap in the market for a $300 - 1000 \text{ W}$ solar array with high power-to-mass ratios, highlighting the need for a novel, high performing solar array for smallsats.

A limiting factor of this comparison is that it only depicts the BOL performance. Disregarding the EOL power hides degradation factors due to environmental effects, and may lead to a wrong assessment of the long-term performance. It may happen that at EOL, a certain solar array may have more efficient solar cells which degrade less and therefore have an overall better performance than other solar arrays. Another limiting factor is the negligence of the stowed power density. This again is due to the given data

being limited, particularly because the stowed power density is a critical metric predominantly used to characterise foldable and roll-out solar arrays. Although some benchmarks can be set with the known data of other roll-out solar arrays, the stowed power density cannot be compared with traditional solar arrays.

3

Problem definition and requirements

The following chapter outlines the focus of the thesis in more detail. Firstly, the customer needs are identified in section 3.1. With these, the research scope and methodology of the study is defined more thoroughly in section 3.2, followed by a complete list of requirements for the roll-out solar array in section 3.3.

3.1. Customer needs

For the context of this thesis, which is being conducted in partnership with Revolv Space, the first step in deriving the requirements for the roll-out solar array involves understanding the needs of the company. Table 3.1 lists the needs that are given by the customer, Revolv Space.

Table 3.1: Customer needs

ID	Need
1	The system shall be able to generate power for the satellite
2	The system shall be able to integrate with small satellites larger than Cubesat size
3	The system shall be able to deploy in an unfurling manner
4	The system shall have a lifetime of at least 5 years in LEO
5	The power-to-mass ratio of the system shall be equal to or greater than the state of the art
6	The stowed power density shall be equal to or greater than the state of the art
7	The system shall handle loads from small satellite operations

3.2. Research scope and methodology

Having investigated the literature for roll-out solar arrays and knowing the needs from the customer, a research scope for the thesis can be defined. From the state of the art solar array comparison given in section 2.2, a gap was identified for a high power-to-mass ratio solar array for a power range of $300 - 1000\text{ W}$. Taking this into account, the research objective is to enhance operational capabilities of small satellites by designing a roll-out solar array in the $300 - 1000\text{ W}$ range that surpasses the power-to-mass ratio and stowed power density of currently available solar arrays. The stowed power density is added as it is another key performance indicator for roll-out solar arrays, and assesses how efficiently the solar array's packaged volume is utilised relative to its power generation. This parameter is especially critical for smallsats which are known to have stringent volume requirements.

While the broader state of the art foldable and roll-out solar arrays reach a power-to-mass ratio of 140 W/kg , solar arrays like the MDSAW and HiPASS, which currently lead the $300 - 1000\text{ W}$ range market, reach at most 73 W/kg . By targeting a power-to-mass ratio of 140 W/kg , the solar array aims to surpass the capabilities of existing solutions, while simultaneously offering a performance which is at the level of the state of the art. For the stowed power density, a target of 40 kW/m^3 is chosen, as it aligns with the state of the art performance of similar arrays like the ROSA. Since this performance parameter is not used for traditional solar arrays, it cannot be compared to the solar arrays in the $300 - 1000\text{ W}$ range.

Considering the objectives, the study will aim to find a set of optimal roll-out solar array designs and analyse their feasibility. To accomplish this, a few key steps will be followed to ensure a systematic and comprehensive engineering design process. Figure 3.1 shows a summary of the steps that will be taken during the research phase of the thesis. After a requirement definition and finding possible concepts for the design, an analytical model will be set up that designs a roll-out solar array based on certain inputs. After verifying and validating the analytical model, it is integrating into an optimisation model that follows the Pareto optimisation theory and optimises the power-to-mass ratio and stowed power density of the solar array. From this, a handful of optimal designs are expected to generated or produced which abide to all the constraints. These steps are all planned in mind such that the research questions will be answerable at the end.

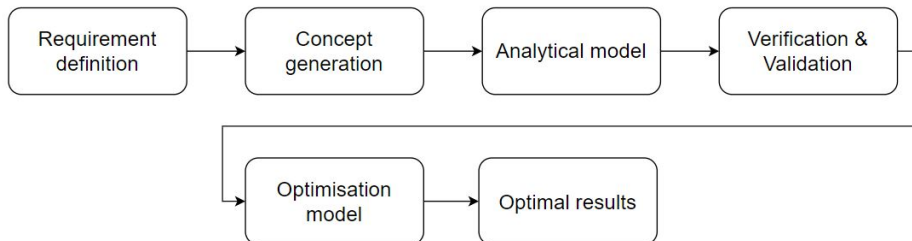


Figure 3.1: Summary of the methodology of the thesis

3.3. Requirement definition

Table 3.2 gives the top level system requirements for the roll-out solar array. These were formulated by taking into account both the customer needs and the general requirements for a solar array.

Table 3.2: System top-level requirements

ID	Requirement	Rationale
1	The system shall have a stowed configuration and deployed configuration	
2	The system shall unfurl from a rolled up state during deployment	Definition of a roll out solar array
3	The system shall be able to transfer power from the array to the satellite	
4	The deployment mechanism shall be fault tolerant	
5	The system shall be designed to prevent unintentional deployment during launch or prior to intended deployment.	
6	The system shall be designed using solar cells qualified for space	
7	The power-to-mass ratio of the system shall be at least 140 W/kg	State of the art
8	The stowed power density of the system shall be at least 40 kW/m^3	State of the art
9	The system shall have a fixed power output capacity within the range of $300 - 1000\text{ W}$	FOSA is up to 300 W and ROSA is from 1000 W
10	The system shall have a maximum width of 96cm	Integratable with ESPA class satellites [52]
11	The system shall withstand the equivalent of 5 years of radiation in LEO	
12	The system shall be able to operate nominally from -60 to $+90\text{ deg C}$	Max testing temperatures for the ROSA [38]
13	The system shall be able to withstand atomic oxygen	
14	The impact of a micrometeorite on the array shall not result in system failure	

Table 3.2 continued from previous page

ID	Requirement	Rationale
15	The deployed system shall withstand 28 N of thrust in the lateral directions	Hydrazine thruster [7]
16	The deployed system shall have a first natural frequency above 0.6 Hz	Sparkwing fundamental frequency for 1000 W configuration [17]
17	The deployed system shall withstand angular accelerations of 2.5 deg/s^2 in all directions.	Maximum for agile satellites [53]
18	The deployed system shall not twist by more than 5 deg	Set by Revolv Space, explanation provided below
19	The system shall automatically lock in its deployed configuration	Prevent unintended retraction
20	The system shall use materials which show a relative mass loss of <1% and collected volatile condensable material of <0.1% as outgassing properties	ECSS-Q-ST-70-02C. Prevent contamination on cells and satellite

The twist angle requirement is one which is not common for traditional solar arrays but imperative for roll-out solar arrays. This requirement is responsible to ensure that the array is stiff enough to not deform significantly during satellite manoeuvres. There might be a flexible design wherein the angular acceleration does not cause high stresses in the structure, but causes the array to deform and twist considerably. This is undesirable, and thus a maximum twist angle must be set. Due to being irrelevant for rigid solar arrays, a standard maximum twist angle does not exist in literature or in ECSS, and thus is set by Revolv Space at 5 deg. This ensures a low enough value which does not affect the geometry of the array under dynamic satellite manoeuvres.

Another important parameter for solar arrays is the voltage, but this is not explicitly set as a requirement as this can change depending on the mission requirements. Furthermore, a couple different voltages are usually proposed to customers like the Sparkwing solar arrays [17]. The possible voltages are thus omitted from the requirements and left to be determined for the later stages of the design.

Concept generation

The following chapter analyses the various options that the roll-out solar array could have for its key design elements. The purpose of this step is to filter out some of the unfeasible options, such that the optimisation can be simplified by only having relevant design options to analyse. The chapter starts with an overview of the design elements of the roll-out solar array in section 4.1. Then, the solar cell options are analysed in section 4.2, followed by a explanation of the deployment mode options in section 4.3. The deployment mechanisms are analysed in section 4.4, and finally an overview of the possible stiffening structure options is presented in section 4.5.

4.1. Roll-out solar array core design elements

For the roll-out solar array, four main elements are identified from existing roll-out solar arrays which determine the overall design. These are the solar cells, the deployment mode, the deployment mechanism, and stiffening structure, as seen in Figure 4.1. These elements make up a roll-out solar array, such that a conceptual design can be derived. For each design element, the various options will be presented and discussed in the following sections. Unfeasible options will also be eliminated to allow for a simplified simulation process.

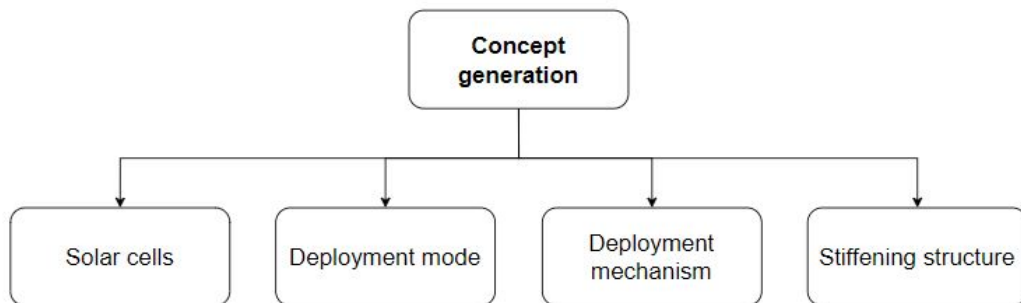


Figure 4.1: Core design elements of a roll-out solar array.

4.2. Solar cells

Solar cells have long been the primary method for generating power in satellites, and as of 2021, over 90% of all small satellites utilized solar cells [15]. This section gives an introduction to and explains solar cell technology in subsection 4.2.1, followed by an overview of the most commonly used solar cells in the space sector in subsection 4.2.2. Lastly, the solar cells which will be used in the optimisation model are identified in subsection 4.2.3.

4.2.1. Intro to solar cells

Photovoltaic cells, referred to in this study as solar cells, are semiconductors which convert sunlight into electrical energy. Based on the photovoltaic principle, they function by having a photon strike the cells surface to excite electrons in the valence band with energy. If the added energy is more than the energy required to cross the band gap, the electrons move into the conduction band of the solar cells, in which the electrons are free to move. This movement of electrons generates an electrical current when the cell is connected to a circuit. [54]

A solar array is a collection of solar cells which are able to convert sunlight into electricity on a larger scale. While there are many types of structures of solar arrays, a typical one involves a couple of components as seen in Figure 4.2 (left), which shows a cross section of a rigid solar array. The array begins with a substrate (shown as the honeycomb structure), which acts as the base layer to provide structural support and durability. For flexible solar arrays, the substrate is usually much thinner than that of rigid arrays, and is commonly made of a polyimide film such as Kapton, a flexible insulating film used for various space applications [3]. Solar cells are then added and bonded to the substrate with an adhesive. These adhesives are used to ensure mechanical stability and must be highly durable to environmental stresses for long periods. Finally, a layer of transparent cover glass is placed on top of the solar cells to protect them from environmental factors like atomic oxygen and radiation. This transparent layer is commonly made of borosilicate as it has a low thermal expansion coefficient, and does not have discolorations when exposed to radiation [55].

One of the most important performance characteristic of a solar cell is its efficiency, or the percentage of incident sunlight which is converted to electrical energy. While it is a key attribute, it is also one of the most challenging to increase. This is because the efficiency is depended on the semiconductor's band gap, which determines which photons the cell is able to convert into electrical energy. If the photon does not have enough energy to excite the electron past the band gap, energy is wasted as a transmission loss. Likewise, if the energy of the photon is much higher than that of the band gap, the additional energy is wasted as heat, also known as a thermalization loss. The band gap should ideally match the energy of the photon to maximise the efficiency of the cell. However, since the solar spectrum includes a wide range of energies, it is impossible to harness it all since the band gaps are limited by the material they are made out of. Figure 4.2 (right) shows how much energy of the solar spectrum a typical silicon solar

cell at sea level harnesses and how the rest is lost. Some solar cells, specified in the upcoming section, use methods to harness a wider range of the solar spectrum to improve their efficiency. [56]

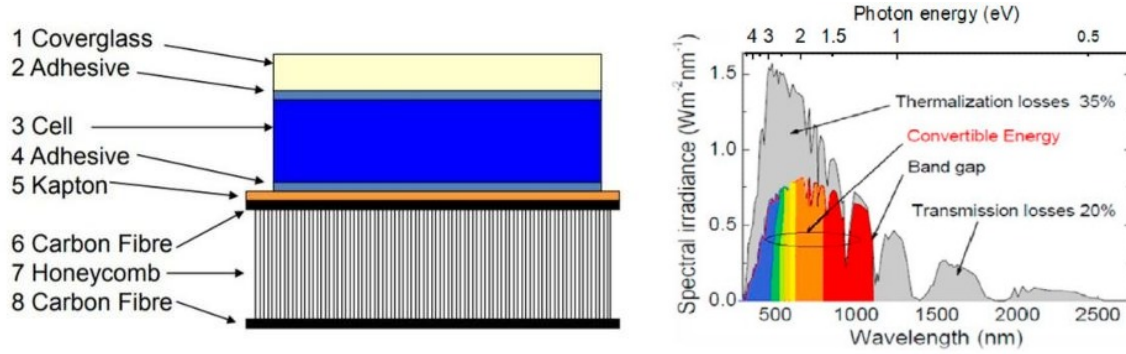


Figure 4.2: Structure of a solar array (left) [57], and the energy conversion of a silicon solar cell at sea level with losses (right) [58].

A large ongoing challenge in solar cell research is the ability to upscale laboratory cells for them to be used for commercial purposes. While commercial solar cells are optimised for mass production, longevity, and cost effectiveness, their efficiency is much lower than cells made in labs, which focus on demonstrating the technological potential rather than commercial viability. Nevertheless, laboratory cells are often very small, incur high costs, and don't offer long term longevity, making them unsuitable for real life applications. As a result, the efficiencies of laboratory cells are frequently taken as a metric for what future commercial cells might have, while commercial cells represent current state of the art. [54]

4.2.2. Solar cell types

The first solar cell produced was made out of silicon, a common semiconductor which currently also contributes to over 85% of the solar cell market [59]. Silicon solar cells can come in two types: monocrystalline or polycrystalline. The former one is a lab grown single crystal of silicon, which makes it efficient but also considerably expensive. Polycrystalline cells are made by melting together multiple silicon fragments, making them cheaper to produce but also less efficient. While this type of semiconductor dominates the solar cell market, their limited band gap makes their efficiency have a theoretical limit of 33%, but practical efficiencies are still about 20% [15].

More efficient types of solar cells are the so called III-V compound solar cells. These semiconductors, named after the elements from the 13th and 15th group of the periodic table, such as gallium arsenide (GaAs) or gallium indium phosphide, offer superior efficiencies over traditional silicon-crystalline solar cells. This is because their material properties allow them to have a more efficient band gap, converting a greater portion of the solar spectrum into electricity, resulting in higher efficiencies. These types of solar cells also offer high radiation and temperature resistance, and as a result have been commonly used in space applications. [54]

Known as the second generation of solar cell technology, thin films have also been developed to be a lighter and cheaper alternative to crystalline silicon solar cells. As the name suggests, they are made with materials that allow the cell to be significantly thinner and more lightweight than crystalline silicon solar cells, offering also flexible properties. The most used materials for thin films are amorphous silicon, copper indium gallium selenide (CIGS), and cadmium telluride. While the lighter mass, flexible characteristics, and decreased cost are favorable, the market is still highly dominated by silicon cells as thin films have lower efficiencies (6-18%), making them overall less effective. [59]

Multi junction cells are made of two or more junctions, allowing each layer to absorb different portions of the solar spectrum and thus have higher efficiency limits. Among these multi junction cells, the triple junction configuration is most commonly used in space applications, effectively harnessing a broader range of the solar spectrum by stacking three semiconductor materials. Since they are commonly made with III-V semiconductors like GaAs, they also exhibit very good resilience to radiation. To date, state of the art commercial triple junction solar cells have efficiencies over 30%, and four junctions cells in labs have managed to reach efficiencies of over 45% [33]. While a cell can have unlimited amount of junctions, the more junctions a cell has, the more complex and expensive it becomes, and so triple junction solar cells are the most commonly used commercial cells nowadays, balancing efficiency and cost to be the optimal solar cell candidate for space applications. [59]

The last types of solar cells belong to the emerging technologies group. These are the types which are relatively new but have significant potential in the future once research progresses. In particular, the most promising technologies include perovskites and those made of organic materials. Despite having low efficiencies, they offer advantages in the form of being lightweight, flexible, and low cost, making them ideal candidates for roll-out solar arrays. While there are companies investing in these new technologies, commercial panels made of these technologies do not exist yet, and research regarding their radiation hardness has yet to be done. Dual junction, also known as tandem variations, also exist which aim to increase the efficiencies of these cells. The most researched type is the silicon perovskite tandem solar cell, which has a record efficiency of 33.9% in a lab environment [33]. Furthermore, the material composition also allows the tunability of its band gap, meaning perovskite cells can be optimised to match the solar spectrum. Other emerging solar cell technologies include dye sensitized and quantum dot solar cells. While these have already been researched for several decades, they still don't offer high efficiencies, and will require further research and testing before they become viable alternatives. [59]

4.2.3. Solar cells for roll-out solar arrays

Most solar cells can be made into a flexible cell by incorporating thin film technologies and using flexible substrates, but there are only a few which could be viable for a roll-out solar array. Considering that there are two primary constraints for cells of a roll-out solar array, a lot of cells' characteristics become impractical for the given application. The first constraint relates to the environmental resistance required in space, effectively making some cells not viable for this use case. The second constraint relates

to the cell being commercially available. Many solar cell technologies are still under development, particularly those designed for flexibility, meaning that not all types of solar cells can be used for a roll-out solar array. Taking these constraints into account, the following solar cells are ones which exhibit high levels of flexibility, are viable for space applications, and are commercially available. The complete list of the viable solar cells with their respective performance characteristics can be found in Table 4.1.

The most popular solar cell type which meets the aforementioned criteria are triple junction solar cells. One state of the art flexible triple junction solar cell from CESI has an efficiency of 29.5%, a minimum bending radius of 3 *cm*, and a cell mass of 84 *mg/cm*² [60] [61]. Redwires ROSAs use Spectrolabs' XTJ Prime cells, which feature an efficiency of 30.7% [62]. While the bending radius is not specified, their usage in a roll-out solar array validate their flexibility. The last triple junction solar cell is Rocket lab's ZTJ cell. This cell is 29.5% efficient and has a comparable areal mass as CESI's cell [63].

Silicon cells based on a heterojunction technology are also promising candidates for a roll out solar array. Solestial offers this type of solar cell technology for a range of applications [64]. Their cells can be variable in size, and offer an efficiency of 20%, with a thickness of 20 μm . Furthermore, the Chinese Academy of Science (CAoS) created a solar cell module based on crystalline silicon with an efficiency of 24.5% [65] [66]. This single junction solar cell module has a minimum bending radius of 0.4 *cm*, and a cell thickness of 60 μm [65]. These silicon cells which are based on a heterojunction structure are also made to be radiation resistant enough for space applications [67].

The last two solar cells are based on the CIGS and GaAs technologies. The former is made by Ascent Solar Technologies, and features a flexible cell for space with an efficiency of 17.6% [68] [69]. This solar cell, which has been used on Momentus' TASSA, has a maximum bending radius of 5 *cm*, and an areal mass of 0.108 *kg/m*². The GaAs solar cell candidate is made by Ubiquity Solar and offers a 23% solar cell with low areal weight of 0.114 *kg/m*² [70].

Having identified the state of the art flexible solar cells, their performance characteristics can be compared. While multi junction solar cells have the highest efficiencies, it is important to note that for a roll-out solar array, the power-to-mass ratio and power density are more relevant than the efficiency, provided that the power requirements of the satellite are still met. Thus, cells with lower efficiencies than the triple junction cells could still perform better, given that their mass can be sufficiently reduced, or their bending radius is low enough for the solar array to be more compact.

Table 4.1: State of the art flexible solar cells

Company	Name	Cell type	Efficiency (%)	Thickness (μm)	Bending radius (cm)	Cell mass (kg/m^2)	Area ($cm \cdot cm$)	Reference
CESI	CTJ30-80	TJ	29.0	80	3	0.5	$7.0 \cdot 4.0$	[60] [61]
Spectrolab	XTJ Prime small	TJ	30.7	80	3^1	0.84	$6.9 \cdot 4.0$	[62]
Spectrolab	XTJ Prime large	TJ	30.7	80	3^1	0.84	$13.9 \cdot 7.0$	[62]
Rocket Lab	ZTJ	TJ	29.5	80^1	3^1	0.84	$10.2 \cdot 5.1$	[63]
Solestial	-	Silicon HTJ	20.0	185^2	5	0.42^2	$5.5 \cdot 5.5$	[64]
CAoS	-	Silicon HTJ	24.5	60	0.4	0.5^1	$15.6 \cdot 15.6$	[65] [66]
Ascent Solar	Titan	CIGS	17.6	52	5	0.108	$42.4 \cdot 21.0$	[68] [69]
Ubiquity Solar	-	GaAs	23.0	80^1	5	0.114	$5.0 \cdot 2.0$	[70]

¹ Value estimated due to lack of data² Value includes coverglass

4.3. Deployment mode

The following section describes possible modes in which the roll-out solar array could deploy. Furthermore, the electrical connections for each modes are considered and evaluated to find the optimal solution. For the first two modes, rollers would be necessary near the casing exit to ensure the solar array deploys correctly in the right direction.

4.3.1. Normal deployment

In the normal deployment mode, the casing of the solar array, along with the mandrel which coils the solar blanket, is fixed to the satellite. Once deployment is initiated, the solar array starts unfurling and leaves the casing as seen in Figure 4.3.

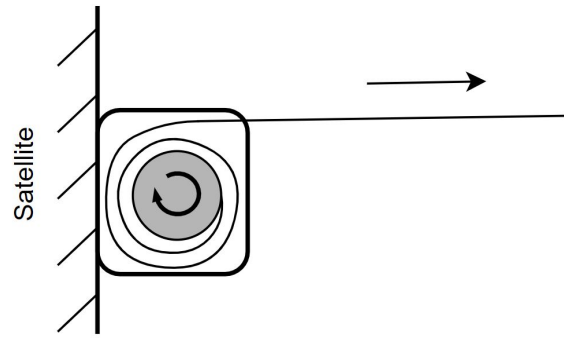


Figure 4.3: Roll out method for the normal deployment mode.

One drawback of this mode is that the electrical connection to the satellite is not so straightforward. This is because the electrical connection is made with the innermost ring of the coiled solar blanket, which gets the most rotations as the solar array deploys. One solution is to use a slip ring, which directly allows the transfer of power from the rotating system to the satellite. Another method is to use a flexible printed circuit (FPC). This method, depicted in Figure 4.4, functions similar to a spiral torsion spring wherein the spiral shaped coil allows rotational movement. As the solar array unfurls from its central mandrel, the FPC uncoils, increasing in size and becoming less tight.

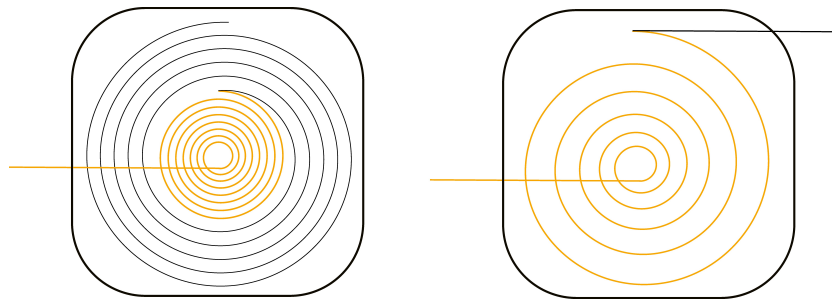


Figure 4.4: Electrical connection with an FPC before deployment (left), and at the end of deployment (right). The orange spiral represents the FPC, and the black spiral the solar array.

While a slip ring offers a straightforward solution to the electrical connection problem, it comes with a few undesirable limitations. Firstly, slip rings that are qualified for space are very expensive, even more so when they have to be custom made for a particular system. The addition of a slip ring would also incur extra weight and volume to the system. On the other hand, an FPC, usually made of Kapton as the insulating substrate and copper for the conductive material, are not only extremely lightweight, but also highly cost effective to manufacture. Furthermore, designing an FPC does not require a significant amount of time or complex process. Due to these reasons, a complex trade off or analysis of both solutions is not required, and the FPC solution will be used for the electrical connection of this deployment mode.

4.3.2. Inverted deployment

In this mode, the tip of the solar array is fixed to the satellite, while the rest of the roll out solar array system moves outwards. Figure 4.5 depicts how the casing and mandrel move away from the satellite, causing the solar array to unfurl. For this mode, the electrical connection would be direct, meaning that no extra volume is required to accommodate for an external connector. On the other hand, this mode will have additional mass at the end of the solar array, effectively making the system more susceptible to external loads. Having additional mass at the end of the solar array would mean that there would be greater deflections and lower natural frequency of the system, both of which are disadvantageous for the structure. One method to mitigate this would be to jettison the mandrel and its attachments from the solar array. While this would make it less susceptible to accelerations from the satellite manoeuvres, it adds to the rapidly increasing issue of space debris, and thus is disregarded as a possible option. Therefore, the effect of the additional tip mass will be incorporated and analysed in the analytical model for this mode.

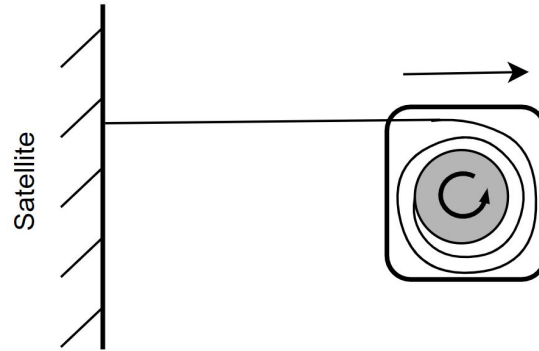


Figure 4.5: Roll out method for the inverted deployment mode.

4.3.3. Free deployment

The last deployment mode follows the process visualised in Figure 4.6. When stowed, the solar array is rolled up and enclosed in a casing. As the deployment is initiated, the casing opens and the rolled up array is released. Due to the stored energy in the structure, the array would automatically deploy to its

full length. This method has the benefits of the two previous modes since it would have a direct electrical connection, and the lack of a mandrel would cause the array to be less prone to dynamic manoeuvres. Despite its advantages, this type of deployment has been shown to be highly unpredictable and follow a chaotic dynamic behaviour, causing lots of disturbances and stress concentrations on the stiffeners [71]. Since deployment has to be controlled to minimise damage, this deployment mode is discarded as a design option for the roll-out solar array.

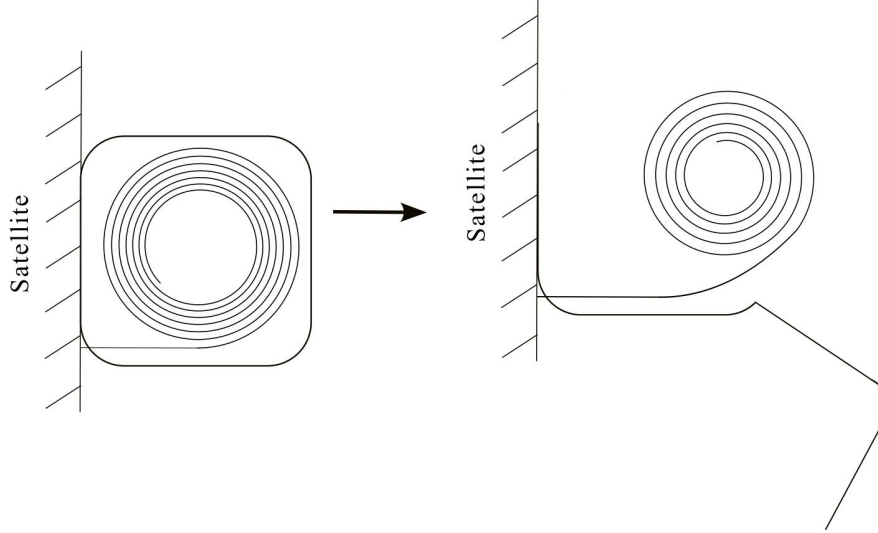


Figure 4.6: Free deployment mode for a roll-out solar array.

4.4. Deployment mechanism

Deployment mechanisms can be broadly categorised into two forms: constraint-driven and energy-driven [72]. These two concepts have their own advantages and mechanical characteristics, but differ primarily in how the deployment action is initiated and controlled. The following sections explore the various mechanisms that the roll-out solar array could employ for deployment.

4.4.1. Constraint-driven

Constraint-driven mechanisms are characterised by their reliance on a motor or actuator for deployment. These mechanisms offer high precision and control due to their mechanical constraint offering adjustable speeds and positions, enabling fine tuning of the deployment sequence and reversible actuation. Satellites and spacecraft frequently employ constraint-driven mechanisms, like Thales' Solarflex, described in section 2.2, which deploys with the assistance of a motor, enabling a controlled and reversible deployment sequence. The reliability and precision of this type of mechanism makes it very favourable as a deployment method, especially for such systems that need to be retractable. Nevertheless, constraint-driven mechanisms can become complex given the need for additional mechanical components. The use of a motor for deployment of the roll-out solar array would be possible, and thus is a viable design

option that will be considered in the simulation. A drawback of this method is that the volume and weight of the motor would have to be considered into the design process, and therefore could decrease the overall performance of the solar array.

4.4.2. Energy-driven

Energy-driven mechanisms follow the principle of utilizing the total potential energy stored within a system to achieve deployment. The stored strain energy is used to facilitate expansion, extension, or unfolding of the structure. While this non-reversible actuation mechanism is less complex than a constraint-driven mechanism, the deployment can be very rapid and follow unpredictable behaviour, thus decreasing the overall reliability of this mechanism. A practical example of an energy-driven deployment mechanism is Redwire's ROSA, described in section 2.2. The entire system deploys automatically and relies on stored strain energy, meaning no actuators are used for the deployment sequence [38]. The following subsection explores the various energy-driven deployment mechanisms that can be implemented in the analytical model as design options.

The first method relies on the strain energy stored in the system to achieve self deployment. This method, which is used by Redwire's ROSA, functions similar to a tape spring, enabling it to store elastic potential energy when deformed or coiled. If it begins to uncoil, the structure will continue until it is fully deployed. A limitation of this type of mechanisms is that once deployment is initiated, the uncoiling sequence is fast and uncontrollable. Designs exhibiting neutral stability, wherein the structure maintains equilibrium in intermediate positions, have also been proven, but this requires an external actuating mechanism to function, which would consequently make it a constraint-driven mechanism [73]. This mechanism is a viable option for the roll-out solar array as it is reliable, and wouldn't require power for deployment. Nevertheless, to deal with the uncontrolled deployment of this type of mechanism, a mechanism that slows down the deployment sequence would be required such that the solar array deploys in a safe manner, thereby minimising risks of the system. Some various options are proposed to tackle this problem.

The initial idea to slow down the self deployment sequence was a spring loaded system, and a few different solutions were found. The first method is called a ratchet spring braking system, and works like a ratchet wheel and pawl system as seen in Figure 4.7 (left). When the solar array deploys, the teeth of the wheel cause the pawl to displace, which in turn slow down the angular velocity of the mandrel. Once the design would be finalised, it would be very cheap and reliable part of the system. This type of braking system can also be highly customised since the springs, ring teeth, and ratchet can all be easily adapted in the design phase. Another method of braking the deployment of the roll-out solar array be facilitated by the energy stored in a constant torque spring or clock spring. The drawback about conventional torsion springs is that the output torque of the spring is not linear, meaning that the rate of deployment would be very low at the beginning and very high at the end. This is not ideal as the whole duration of deployment should be slow and controlled. Constant torque springs and clock

springs, shown in Figure 4.7 (middle, right) output a near constant torque throughout their rotation, and thus can be designed such that they counteract part of the deployment torque caused by the stored energy. These spring based solutions are very cheap, lightweight, and would be very reliable systems.

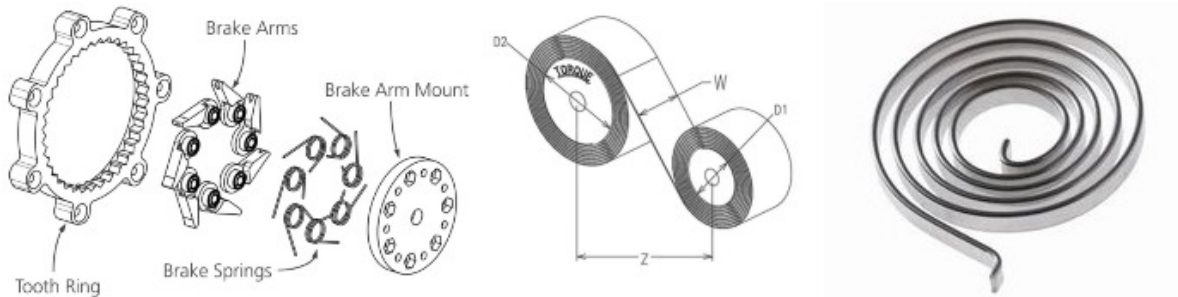


Figure 4.7: Spring based braking mechanisms: Ratchet spring brake (left) [74], constant torque spring (middle) [75], clock spring (right) [76].

Although the aforementioned spring based braking systems are straightforward and simple to integrate, they present a key limitation. Since springs have a constant predetermined torque at the beginning and end of deployment, they would have to be sized such that the end deployment rate is within the allowable range. That is, if it sized such that the final deployment rate is a set value, there would be no room for uncertainty. For example, if the deployment torque would be less than predicted and the resistive torque of the spring would be higher, there is a chance that the deployment would never initiate. Similarly, if the deployment torque in reality is larger in practise, and the resistive torque is lower than predicted, the solar array would deploy quicker than intended due to the spring not being able to adjust its resistive torque. This presents inherent risks in the system, and as a result, the spring based braking methods become inviable. Instead, a mechanism which acts as a damper is necessary. This ensures that the resistive torque is not predetermined, but instead varies with the angular velocity of the deploying solar array. A mechanism based on the principle of escapement was found to be the ideal solution to this problem.

An escapement is a type of mechanism which is essential to the functioning of clocks and watches. The primary purpose of the escapement is to regulate the angular velocity for a given purpose, which for watches is ensuring accurate time measurement. The mechanism consists of two main parts, the escapement wheel and the detent, labelled 3 and 2 in Figure 4.8, respectively. As the escapement wheel is driven by a torque, the detent alternately locks and unlocks the motion of the wheel, and as a result acts as a brake to slow down the system. When the resistive torque is increased, the mechanism responds similar to a rotary damper [77]. This type of damping mechanism would be ideal as it provides a cheap and reliable method to control the deployment of the roll-out solar array. It has also been tested and proven in an aerospace boom deployment application [78]. This damper, seen in Figure 4.8, used the escapement mechanism to dampen the deployment of its micro boom antenna for a NASA mission. Because of its effectiveness and proven use, this method is a viable design option that will be implemented in the analytical model.

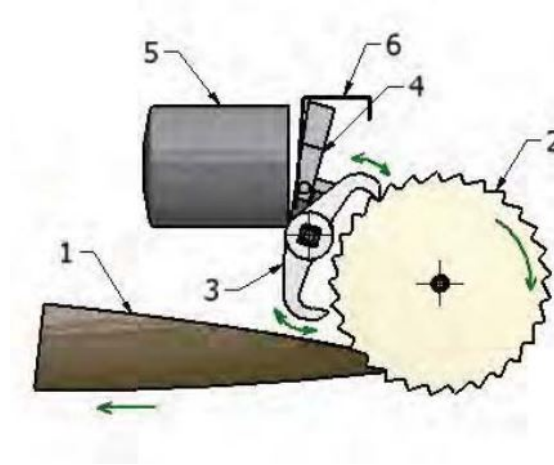


Figure 4.8: Damping mechanism used for a micro boom antenna deployment [78].

Another solution to slow down the self deployment sequence would be to utilize commercial off the shelf rotary dampers, which are specifically design for this type of use. Rotary damper function similar to conventional linear dampers but are instead specifically engineered to limit and constrain movement in a rotational direction, meaning one could be used to control the speed of deployment of the roll-out solar array. Eddy current rotary dampers from CDA Intercorp and Honeybee are already space qualified, meaning they could just be used as a plug and play solution [79][80]. However, in comparison the mechanical braking system, this rotary damper has significantly more limited customizability since commercial off the shelf items have standardized specifications and adaptability. Furthermore, a space qualified COTS rotary damper would be significantly more expensive than an escapement damper – potentially by orders of magnitude. This cost would increase even more if the COTS damper would have to be customised specifically to meet certain requirements. Due to these limitations, the COTS rotary damper becomes an impractical solution in comparison, and thus is omitted as a method to control the autonomous deployment sequence. As an additional advantage, the escapement mechanism ensures that the solar array does not retract, directly satisfying requirement 19 in Table 3.2.

The last type of energy-driven deployment mechanism method relevant for a roll-out solar array relies on the shape memory phenomena. Shape memory polymer composites (SMPC) have a unique material property that allows them to revert back to a predefined shape when heated. They are ideal for use as deployment and actuation mechanisms since they can also function in the presence of high loads [81]. In space applications, SMPCs have been used as a deployment mechanisms for an on orbit demonstration of a roll-out type solar array [82]. The experimental mechanism was attached to China’s SJ20 geostationary satellite, and successfully demonstrated the viability of SMPCs to be used for such a use case. The most apparent limitation of this mechanism is the need for high power requirements needed for the resistive heating needed for deployment. The demonstration on the SJ20 satellite used 40 W despite having a small deployed envelope of $34 \times 42 \text{ cm}$ [82].

While the SMPC technology can be used as a deployment mechanism, it introduces a few drawbacks compared to other types of deployment mechanisms. Firstly, SMPCs present a much more complicated system compared to self - or motor deployment, since the material behaviour requires precise control and monitoring to ensure proper function, especially in the extreme environmental conditions of space. The use of this mechanism would also require significantly more parts and systems to ensure correct and stable power distribution to the SMPC, all of which would significantly increase the cost of development. Additionally, the limited flight heritage of SMPCs used in roll-out solar array applications make this an unfavourable solution compared to other deployment mechanisms with extensive flight heritage. This is because the deployment mechanism must be extremely reliable, as it is part of a critical satellite system. Finally, the long-term reliability of SMPCs in extended missions also remains uncertain, further decreasing their suitability as a roll-out solar array deployment mechanism. Given these considerations, it is concluded that the use of SMPCs is not recommended, and further testing is necessary before their potential application in a roll-out solar array can be considered.

4.5. Stiffening structure

This section analyses the possible structures which would stiffen the roll-out solar array after being deployed. After an intro on deployable structures in subsection 4.5.1, two types of stiffening methods for the roll-out solar array will be analysed: external stiffeners (subsection 4.5.2), and inherent stiffeners (subsection 4.5.3). Both types of structures would be able to be used with any deployment mechanism, meaning a multitude of design combinations are possible for the roll-out solar array. The possible materials used for the stiffeners are then identified in subsection 4.5.4.

4.5.1. Deployable structures

Deployable structures serve many applications, employing unique types to facilitate deployments of space systems. The following will explore two predominant types of deployable structures: deployable masts and deployable booms. While not exhaustive, this exploration aims to highlight the most common deployable masts and deployable booms. Also, other deployable space structures like tensegrity structures exist, but would be unfeasible to use for a roll-out solar array structure and thus are not analysed [83]. Unless stated otherwise, the following information is derived from Pellegrino [72].

The deployable masts can be categorised into two types: coilable and articulated. In general, masts are high load-bearing frames interconnected by a combination of flexible membranes and cables. They are commonly used to elevate sensors away from a spacecraft to minimize interference, but they have also been used as solar array deployers. Coilable masts are made up of three primary longerons connected by supporting buckled membranes called battens, creating an almost triangular cross section. Interconnecting the repeating frames are pre-stressed cables which further add support and stiffness to the whole structure. Coilable masts can be deployed in two ways, as visualised in Figure 4.9, by either

deploying a lanyard centrally which drives the top away from the bottom and lengthens the membranes, or by rotating an inner canister which automatically deploys the stiffening membranes. These types of masts offer compact storage capabilities due to their slender canister, making them particularly suitable for volume critical applications.

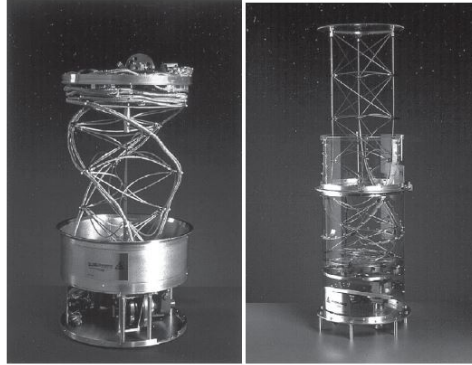


Figure 4.9: Coilaible mast deployed by releasing a central lanyard (left), and by rotating the inner structure (right). [72]

Articulated masts on the other hand employ a square configuration as opposed to the triangular one of coilaible masts, as shown in Figure 4.10. They are based on a repeating cubic frame that is folded to a compact storage volume, and consists of a set of vertical and horizontal membranes with support from interconnected cables. After deployment, a tensioning system is used to stiffen the structure. A notable use of these masts are in the old solar arrays of the ISS.

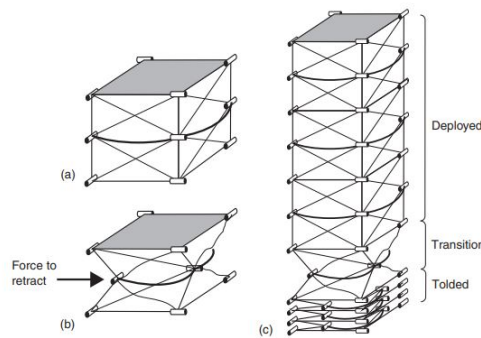


Figure 4.10: Working principle of an articulated mast. [72]

Deployable booms, in contrast, serve similar functions as masts but are distinguished by their tubular shape. The first type is the coilaible boom, which follows the concept of a tape spring but has a much larger curved cross-section. The Storable Tubular Extendible Member (STEM), invented in 1965, is a lightweight and compact coilaible boom that functions similarly to a tape spring but differs in having overlapping curved edges. A closed form structure of the STEM can be achieved by having an interlocking feature at the edges such that it forms a circular tube-like cross section. A coilaible boom consisting of two STEMs, known as the BI-STEM, offers a more symmetric structural response to loads,

and allows the rolled-up width and transition zone length to be shortened. A further improvement of the BI-STEM is the interlocking BI-STEM which secures the two individual STEMs together. The third type of coilable boom is the lenticular mast, which features a closed cross section. This type has much higher torsional stiffness compared to the previous two open designs, but is significantly more complex to manufacture as the sides have to be bonded together from two separate parts. The three types of coilable booms are visualised in Figure 4.11. Coilable booms are a popular deployable structure as seen from their flight heritage. Hubbles' FRUSA first used BI-STEM booms to deploy its solar arrays. Currently, Redwire's ROSA and Thales' SolarFlex both use forms of coilable booms as deploying and stiffening structures. This shows the effectiveness of coilable booms and their viability to be used as deployable structures for roll-out solar arrays.

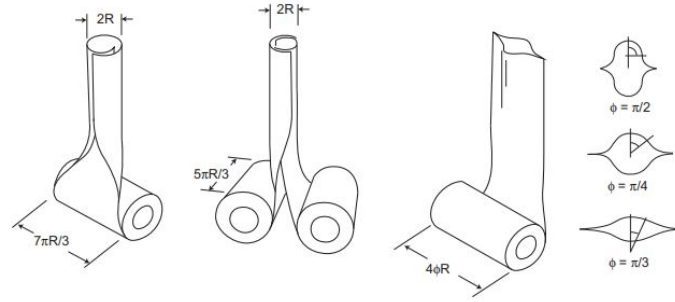


Figure 4.11: Types of coilable booms: STEM (left), BI-STEM (middle), and lenticular (right). [72]

Another type of deployable boom is the inflatable boom, which follows the concept of pressurising a hollow tubular structure to deploy it and maintain rigidity. Moreover, various concepts of deploying or method of inflating have been studied for this type of boom. One concept is Z-folding, wherein a depressurised boom is folded onto itself for compact stowage. Once the booms starts to get pressurised, it unfolds automatically until it is fully inflated. Another method of storing a depressurised boom is by coiling the thin material into a roll, wherein the deployment sequence once pressurisation starts, follows that of a party horn. The last concept is similar to a sleeve being pulled up the arm. This method allows for controlled axial deployment as the outermost fold gets deployed first due to the higher pressure acting on largest diameter. Figure 4.12 displays the three inflatable boom concepts.

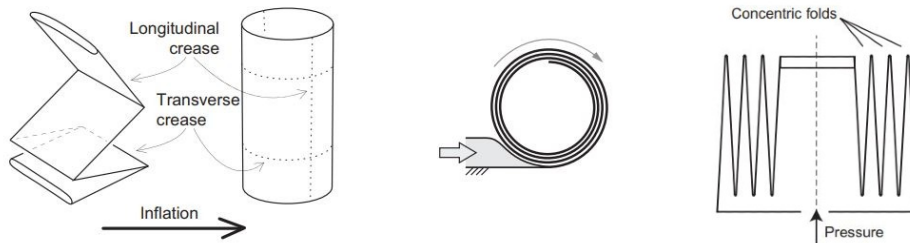


Figure 4.12: Z folded (left), coilable (middle), and pressure sequenced (right) inflatable boom. [72]

The last type of deployable boom is the telescopic boom, which is characterised by its segmented, extendable design. Unlike coilable and inflatable booms, telescopic booms extend linearly through a sequential deployment of rigid sections that are store within another. These sections then deploy similar to a telescope by sliding out from one another and lock into place to form a complete load bearing structure, as seen in Figure 4.13. The deployment process is usually facilitated in two ways. The first method employs a motor that pulls a cable, and with a series of pulleys integrated at the ends of each segment, the boom is able to deploy. The second method is through a coilable boom in the middle which extends the segments, similar to the lanyard in the coilable mast. This type of boom has been used in various applications involving deployable antennas or extending sensitive payloads away from the satellite.

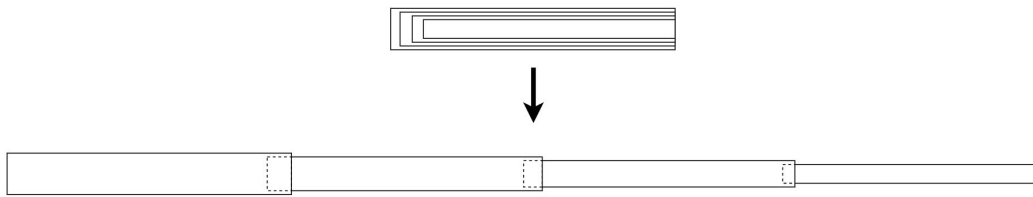


Figure 4.13: Deployment of a telescopic boom.

4.5.2. External stiffening structure

This type of configuration relies on external deployable structures to stiffen the deployed solar array. In other words, the solar blanket would unfurl from its mandrel, along with one or more stiffeners which would act as the load bearing structure post deployment. The following subsection will explore the feasibility of using the aforementioned deployable structures as external stiffeners. Since STEM, lenticular, interlocking and TRAC booms have flight heritage and are commonly used for this application, they will be automatically considered as possible design options.

Masts are another type of stiffening structure which could be used for the roll-out solar array, but before implementing them in the optimisation model, a simplified deployable mast will be designed and compared with a STEM boom in order to check whether they are a suitable competing candidate. For the purpose of this preliminary design investigation, a mast with four longerons is assumed, and its architecture is given by Figure 4.14.

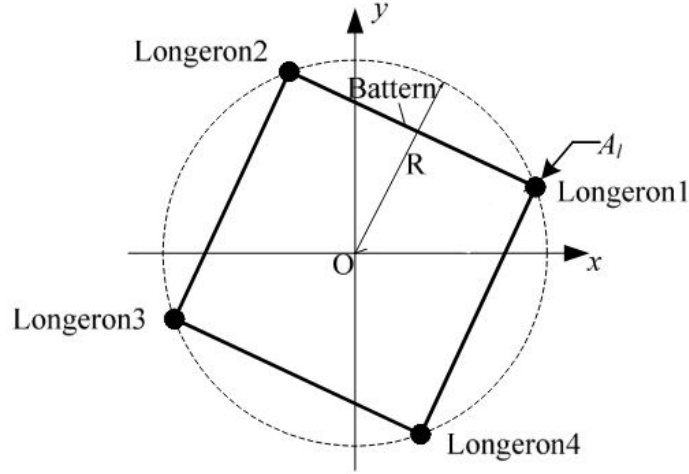


Figure 4.14: Cross section of a mast with four longerons. [84]

For the following comparison, the solar array is assumed to be $L = 5 \text{ m}$. For masts of this size, the diameter of the longeron becomes a critical limit due to the constraints in manufacturability. For CFRP, the minimum diameter of a COTS rod is 0.5 mm [85]. To ensure that the longerons coil elastically when packaged, the radius of the mast shall follow the constraint given by Equation 4.1 [86].

$$R > \frac{d}{2\epsilon} \quad (4.1)$$

With the strain calculated with $\epsilon = \frac{\sigma}{E}$, the minimum radius of the mast can be obtained. The final missing component that defines the architecture of a mast is the length of the longerons, which is demonstrated to be optimal when $l = R$ [87]. With this, the stowed height of the mast can be calculated with Equation 4.2.

$$H = \left(\frac{L}{l} + 1\right)d \quad (4.2)$$

While this assumes that the longerons coil perfectly without any packaging losses, it excludes the height of the driving mechanism, so this is still taken as a lower bound. Similarly, the mass of the structure is calculated as follows:

$$m = 4C_1\rho A_l L \quad (4.3)$$

With A_l representing the area of a longeron, and C_1 representing a correction coefficient which considers the mass of battens, cables, latches, and other components, assumed to equal 4 [84]. Lastly, the bending stiffness is calculated using Equation 4.4, with $E \approx 130 \text{ GPa}$ [84].

$$EI = 2EA_l R^2 \quad (4.4)$$

To make a comparison with a deployable boom structure, a design is made such that the bending stiffness is equal to that of the mast. For this comparison, a STEM boom with a circular cross section is assumed, such that it resembles a thin walled tube. Furthermore, the material properties are kept equal. With the bending stiffness known, the radius of the boom can be calculated using Equation 4.5.

$$EI = E\pi R^3 t \quad (4.5)$$

Since CFRP sheets can be made much thinner than rods due to different manufacturing processes, the analysis will include thicknesses of 0.1, 0.2, and 0.3 *mm* such that a larger design space is analysed. To compare the two structures, their masses and stowed volumes are compared. The mass, assuming that $t \ll R$, is calculated as follows:

$$m = A \rho L = 2\pi R t \rho L \quad (4.6)$$

With $L = 5 \text{ m}$ as for the mast. For the stowed volume, it is assumed that the boom is coiled around a spool with a radius r equal to the minimum allowable bending radius. This is calculated similar to Equation 4.1, with the diameter replaced with the sheet thickness. The volume is then calculated by taking the product of the spool area with the flattened width of the boom. The extra radius from the coils of the boom is assumed to be negligible.

$$V_{stowed} = \pi r^2 2\pi R \quad (4.7)$$

With the important performance parameters calculated, Figure 4.15 compares the volume of masts and booms of varying thickness as a function of the bending stiffness. The increase in the mast's bending stiffness is due to the enlargement of the longeron diameter from the previously identified minimum value.

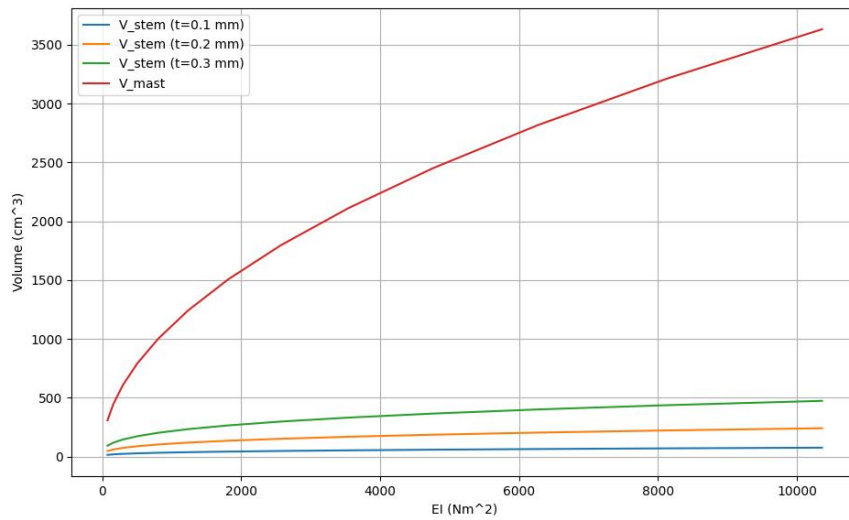


Figure 4.15: Comparison of the stowed volume of masts and booms as a function of their bending stiffness, assuming a constant deployed length of 5 *m*.

From Figure 4.15, it is evident that masts are significantly less volume efficient than booms. This is because the sheets used for the booms can coil around a significantly smaller mandrel since they can be manufactured thinner than the longerons. When comparing the masses as in Figure 4.16, the mast becomes the more favourable option. However, since both mass and volume are important for a roll-out solar array, taking both into consideration is vital. Since the difference of masts and booms is much larger in volume than it is in mass, booms would in any case be the favourable option.

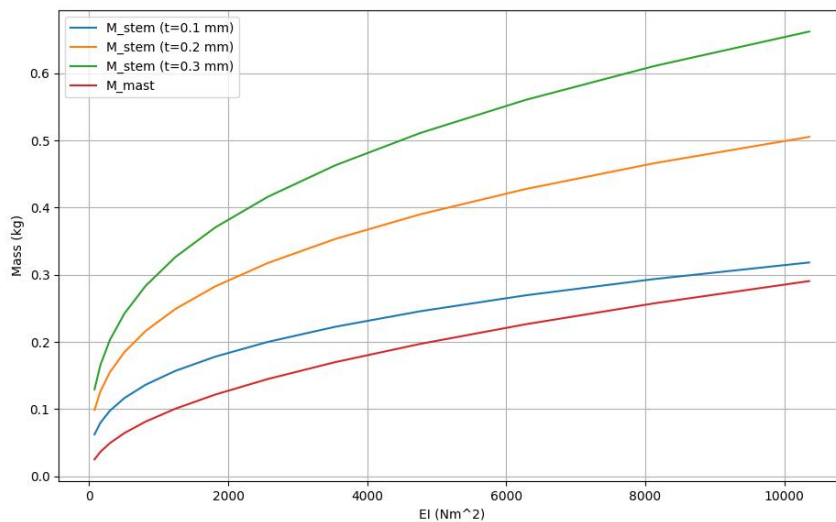


Figure 4.16: Comparison of the mass of masts and booms as a function of their bending stiffness, assuming a constant deployed length of 5 *m*.

Furthermore, when analysing the radius of the base, the masts show an even greater limitation in comparison with the booms. Figure 4.17 shows how the radius at the base varies with bending stiffness. With a bending stiffness of 3400 Nm^2 , the mast already has a radius of 10 cm , greatly exceeding the dimensions expected for the final design. Using a larger mast for an ESPA class smallsat would consume a significant portion of the available space on the smallsat, rendering this option infeasible. Furthermore, this assessment does not include the additional volume required for the housing of the mast, and thus is estimating a best case scenario.

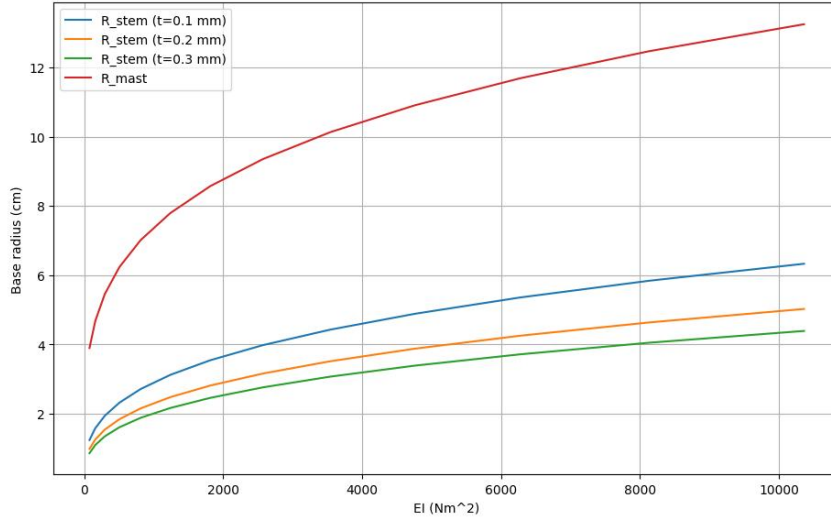


Figure 4.17: Comparison of the radius of the base of masts and booms as a function of their bending stiffness, assuming a constant deployed length of 5 m .

Overall, as the structural performance of the masts can be achieved with various types of STEM booms, the significantly higher stowed volume makes them unfavourable candidates. While masts offer better mass characteristics, the way the masts are stowed with their large base radius makes them infeasible candidates to be used for the stiffening structure of a smallsat roll-out solar array. Due to these reasons, masts shall not be considered as design options for the following study. Furthermore, Puig et al. suggest that masts are generally not made for lengths shorter than 10 m [88]. This relationship is also evident in real missions involving deployable structures: ISS's legacy solar arrays use masts due to their large size, while applications of smaller size such as solar sails and antenna deployers favour booms due to their compact packaging. Since the solar array is expected to be in the range of $5 - 10 \text{ m}$, this further justifies the omission of masts as a stiffening structure option.

Telescopic booms are another method that could be used to deploy the solar array. While their structural properties are comparable to coilable booms, their packaging efficiency is significantly lower. The stowed length of telescopic booms can be anywhere from 10 to 50 percent of the deployed volume, meaning that a 5 m boom would be as a best case scenario 50 cm long in its stowed configuration [3] [89]. Combined

with the fact that the method of packing the telescopic boom would have to be perpendicular to the stowed solar array, the packaging efficiency of this structure is greatly reduced. Since having it stowed outside the satellite is clearly not feasible, a solution could be to have the telescopic boom packaged into the satellite, as seen in Figure 4.18. However, not only would this take a lot of internal satellite volume away, but it would also greatly increase the complexity in the case that the solar array would be integrated with a SADA, since the root of the telescopic boom would be fixed deep in the satellite structure.

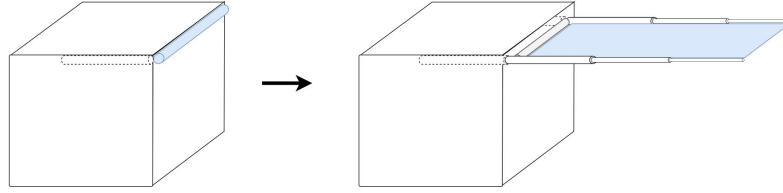


Figure 4.18: Stowed (left), and deployed (right) state of a roll-out solar array with a telescopic boom.

One possibility to resolve this packaging drawback would be to have the telescopic boom side by side with the mandrel such that the stowed state would be more volume efficient. This method would look similar to Figure 4.19, wherein the deployment would commence with the deployment of the telescopic boom by a hinge or motor, followed by the elongation of the boom. A string or attachment to the tip of the solar array could then be retracted to unfurl the solar array from its rolled configuration. While more volume efficient, this solution presents some other limitations. One drawback of this design is that the connection from the solar array to the boom would only be through a cable, which might not constrain the solar array enough. Furthermore, given that this solution involves numerous motors, hinges, and other moving components, it becomes unfavourable for mission critical applications that demand high reliability such as a solar array. Thus, the use of telescopic stiffeners are not considered as a viable design option for the roll-out solar array.

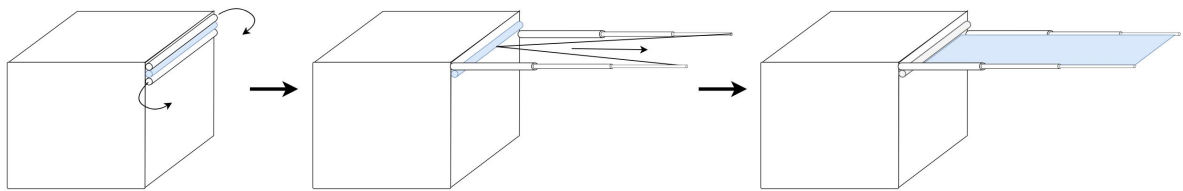
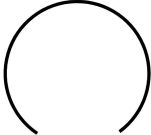
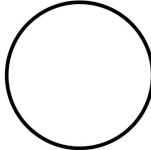
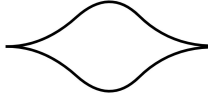
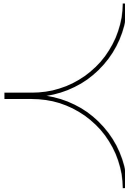


Figure 4.19: Proposed solution to improve the stowed geometry of the telescopic stiffeners.

Finally, inflatable booms are also deemed unfeasible for this application for a number of reasons. Firstly, inflatable structures, especially of the z folded type, have very low deployment accuracy compared to other deployable structures [72]. Additionally, since they have not been used for solar array deployment

applications, their technology readiness level is significantly lower compared to booms or masts, both of which have been employed for solar array deployment applications. Consequently, inflatable booms will not be investigated, and the final list of stiffeners that could be used for the roll out solar arrays are tabulated in Table 4.2.

Table 4.2: Possible stiffener structures viable for the roll-out solar array

Name	Cross section
STEM	
Interlocking	
Lenticular	
TRAC	

While the type of boom is a critical design choice for the external stiffening method, the consideration of the amount of booms is also crucial. Figure 4.20 shows the configurations that the roll-out solar array could employ. Note that although booms are shown in the figure, they could be replaced with any stiffener identified previously. Similarly, a design employing three stiffeners does not represent the maximum allowable design space, as the solar array may end up with more if the simulation calculates it to be the most optimal design. A design with one boom may seem to be more lightweight and space efficient, but it may not be able to be as long as a design employing two or three stiffeners due to having lower structural stiffness. Therefore, the amount of stiffeners will also be a discrete design option that will drive the design of the roll-out solar array.

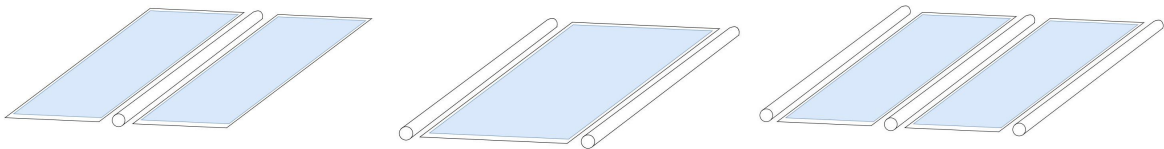
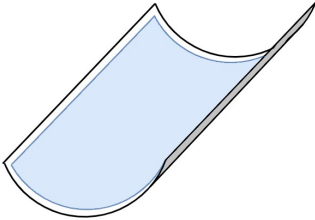
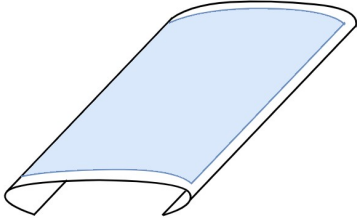
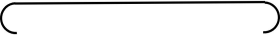
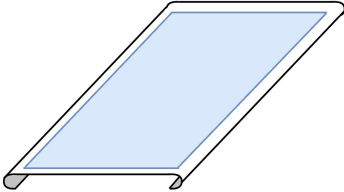
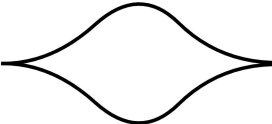
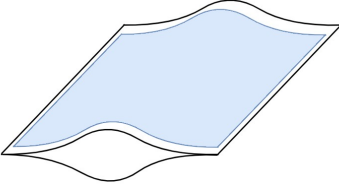


Figure 4.20: Roll-out solar array with single (left), double (middle), and triple (right) stiffener configurations. The blue area represents the solar blanket.

4.5.3. Inherent stiffening structure

The following designs are additional options that will be analysed during the simulation. Since the structure itself is stiffening, they don't require external stiffeners, meaning that there are only a limited amount of designs applicable. Without having the option of choosing the type and amount of stiffeners, only a few options remain as design contenders. One consideration for these designs which might be limiting is the method of bonding the solar blanket to the structure.

Table 4.3: Designs with inherent stiffening structures. The blue area represents the solar blanket.

Name	Cross section	3D representation
Tape spring		
Convex spring		
Flat spring		
Lenticular		

The convex spring stiffener will not be considered due to its similarity to the tape spring and flat spring. Including this would likely not contribute to additional insight or value, while coming up with geometrical properties of this shape would be complex and time consuming. Therefore, omitting this stiffener avoids redundancy and simplifies the optimisation model without compromising the accuracy of the results.

4.5.4. Material selection

The choice of material for the stiffening structure is critical as it will have an effect on the mass, volume, and overall design of the solar array. While carbon fiber reinforced polymers (CFRP) have the most flight heritage for extendable booms and deployable space structures, the specific composite can vary significantly, and the use of other types of materials remains a possibility. Flexibility characteristics, density, and other material properties will all have an effect on the design, and thus a range of materials shall be considered given the absence of a clear material choice.

While researching for CFRP materials and their mechanical properties, significant challenges were encountered in finding multi-ply laminates that had comprehensive data suitable for the optimisation model. Books and research papers lacked extensive mechanical properties of laminates, while databases and manufacturer websites only had information on single-ply composite. Furthermore, the use of classical laminate theory was excluded as this was out of the scope of this study. Due to this limited availability of data on multi-ply CFRP laminates, only single-ply composites are used for the optimisation model. While information on a lot of different composites has been collected, it lacks depth on potential advantages that multi-ply laminates have over single-ply composite. It is therefore recommended that further evaluation is done with the use of multi-ply laminates, such that a broader design space is investigated. Nevertheless, the single-ply composite that were found still had thicknesses in the range that deployable stiffeners would have, and therefore are still valid design options.

Apart from CFRP composites, a space qualified kevlar composite was also found in a materials database and included as a design option as it has favourable density characteristics compared to CFRP. Furthermore, glass fiber reinforced composite and stainless steel are also used due to their suitability for this application. Aluminium was omitted since CFRP outperforms it in all mechanical property characteristics. While aluminium is much cheaper than CFRP, this criteria is not critical for this feasibility study, and thus is excluded from the design space. Another material which is omitted due to limited flight heritage and to descope the study is hybrid carbon/glass composite. While this type of material would be suitable for this application, tests would have to be done to understand the mechanical properties of this material. Table 4.4 lists all the materials that will be used as design options in the optimisation model, along with their respective mechanical properties.

Table 4.4: Mechanical properties of various materials.

Material	Young's Modulus		Shear Modulus	Ultimate Strength		Shear Strength	Density	Thickness	Reference
	(GPa)		(GPa)	(MPa)		(MPa)	(kg/m ³)	(mm)	
	E_1	E_2	G_{12}	σ_{u_1}	σ_{u_2}	$\tau_{u_{12}}$	ρ	t	
T300/RS-3	66.9	66.9	4.8	834	834	139	1600	0.13	[90]
T300/913	159.9	11.7	3.9	2005	68	150	1600	0.2	[91]
HT/Epoxy	140	12	5	2100	50	90	1600	0.13	[92]
CUD/125NS	127.6	7.58	4.05	2650	65.5	86.2	1500	0.153	[93]
CPW/3327	48.3	48.3	3.81	548.9	548.9	86.2	1500	0.25	[93]
CPW/E773FR	138	7	4.02	2085	65.5	79	1600	0.142	[94]
T40/5320-1	156	9.7	5.2	2700	83.4	110	1510	0.142 ¹	[94]
M55J/2040	297	5.62	4.6	1620	35.2	56.5	1400	0.28 ¹	[94]
T700S/2511	59.2	56.7	4.01	1000	993	86.2	1510	0.218	[94]
M46JUD/2511	225	9	3.79	1682	51	68.9	1520	0.213	[94]
M46JPW/2511	95.1	95.1	3.93	572	572	64.1	1520	0.239 ¹	[94]
AS4C-3K/M92	50.2	50.2	2.5	687	687	94	1430	0.271	[94]
HSPW/950-1	68	61	4.5	724	710	103	1463	0.326 ¹	[94]
M55J/950-1	307	5.9	3.1	1770	24.8	84.1	1440	0.272 ¹	[94]
Kevlar/950-1	32	28.2	1.93	535	434	86.9	1350	0.405	[94]
E-Glass/Epoxy	39	8.6	3.8	1080	39	89	2100	>0.1	[95]
Stainless Steel ²	200	200	80	350	350	125	7850	>0.05	[94]

¹ Value calculated using Toray's ply thickness calculator. [96]² Value/s estimated based on typical mechanical properties.

5

Methodology

This chapter explains the complete details of the analytical model and how it is implemented in the optimisation model. At first, the parametrization of the design variables and coordinate system is given in section 5.1, followed by an explanation of all the constraints passed to the analytical model in section 5.2. Then, the effects of the deployment mechanism and deployment mode are given in section 5.3 and section 5.4, respectively. The calculation of how the mass of the solar array is explained section 5.5, followed by the volume calculation in section 5.6, and the power calculation in section 5.7, all of which are used to calculate the design objectives in section 5.8. Finally, the functionality of the optimisation model is explained in section 5.9.

5.1. Parametrization

The following section details the parametrization of the analytical model. This introduces the necessary information that is needed to understand the workings of the analytical model.

5.1.1. Design variables

A critical aspect of the analytical model are the design variables. These define all the different possible parameters that make up a roll-out solar array. These will be passed as inputs in the analytical model, such that a design is made possible from the variables. As the designs differ significantly between external and inherent stiffener designs, two separate analytical models are made. The variables for the external stiffener optimisation model and inherent stiffener optimisation model are given in Table 5.1 and Table 5.2, respectively.

Table 5.1: Input variables for the external stiffener analytical model

Variable	Name	Type	Description
x_1	Deployment mode	Binary	Choice between normal and inverted deployment
x_2	Deployment mechanism	Binary	Choice between motor and escapement damper
x_3	Width	Integer	Number of solar cells widthwise
x_4	Length	Integer	Number of solar cells lengthwise
x_5	Material	Discrete	Choice between the materials given in Table 4.4
x_6	Solar cell	Discrete	Choice between the cells given in Table 4.1
x_7	Mandrel radius	Real	Geometric variable used to determine mandrel geometry
x_8	Stiffener type	Discrete	Choice between STEM, Interlocking, Lenticular, TRAC
x_9	Number of stiffeners	Integer	
x_{10}	Stiffener radius	Real	Geometric variable used to determine stiffener geometry

Table 5.2: Variables for the inherent stiffener analytical model

Variable	Name	Type	Description
x_1	Deployment mode	Binary	Choice between normal and inverted deployment
x_2	Deployment mechanism	Binary	Choice between motor and escapement damper
x_3	Width	Integer	Number of solar cells widthwise
x_4	Length	Integer	Number of solar cells lengthwise
x_5	Material	Discrete	Choice between the materials given in Table 4.4
x_6	Solar cell	Discrete	Choice between the cells given in Table 4.1
x_7	Mandrel radius	Real	Geometric variable used to determine mandrel geometry
x_8	Stiffener type	Discrete	Choice between Tape Spring, Lenticular, Flat Spring
x_9	Angle	Real	Geometric variable used to determine stiffener geometry

Note that for some designs, a certain variable may not be applicable. Nevertheless, the design variables are given by:

$$X = [x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8, x_9, (x_{10})] \quad (5.1)$$

5.1.2. Coordinate system definition

The coordinate system that is used for the analytical model is given in Figure 5.1. The x axis runs longitudinally through the length of solar array, while the y axis is defined to run parallel to the width of the solar array. Finally, the z axis is perpendicular to the previous two axes.

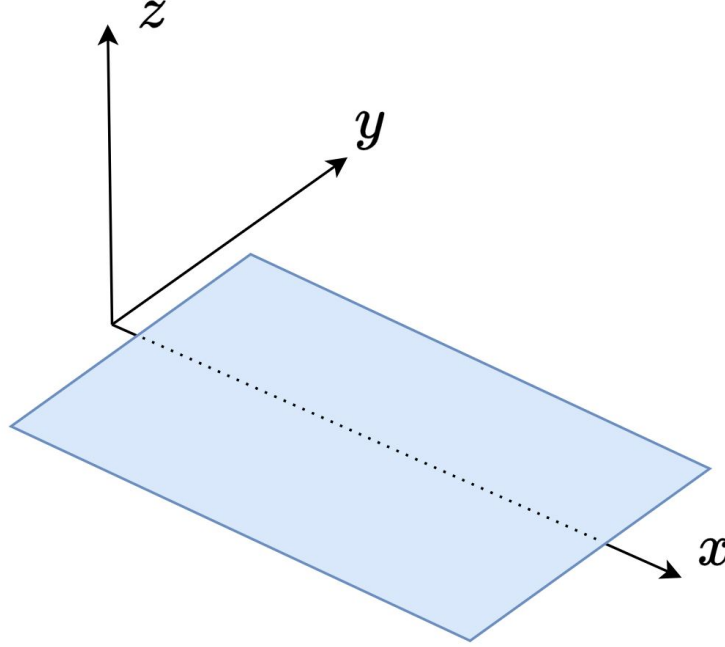


Figure 5.1: Coordinate system used in this study for the solar array.

5.2. Constraints

The following section explains all the constraints imposed on the analytical model. In optimisation problems, there can be soft constraints and hard constraints. The former is a type of constraints which is preferred but not absolutely required, while the latter type imposes a strict requirement that must be met. For the following optimisation problem, all constraints will be considered to be hard constraints. Furthermore, the standard convention of passing constraints to the algorithm is by having them be less than or equal to 0. This way of standardising requirements and constraints makes it simple to handle different types of constraints without needing to alter the handling mechanics. This also allows the penalties imposed by violating constraints to be activated simply when any constraint function returns less than 0. Therefore, constraint functions $g_i(x)$ are all passed to the algorithm in the following way:

$$g_i(x) \leq 0 \quad \text{for } i = 1, 2, 3, \dots, n \quad (5.2)$$

5.2.1. Minimum and maximum power

The first two constraints are set by requirement 9 in Table 3.2. This requirement states that the power of the solar array shall be within 300 and 1000 W . This allows for a clear definition for the first two constraints of the model.

$$g_1 = 300 - P_{tot} \leq 0 \quad (5.3)$$

$$g_2 = P_{tot} - 1000 \leq 0 \quad (5.4)$$

with P_{tot} representing the total power output of the solar array.

5.2.2. Maximum width

Based on requirement 10 in Table 3.2, the system shall have a maximum width of 96 cm such that it will be able to be used for ESPA class satellites. This means that the total width w_{tot} of the solar blanket, along with stiffeners or any additional components or hardware attachments, must not exceed this specified limit. A reduction of 10% is applied to account for uncertainties, resulting in a design target width of 86.4 cm to allow for potential variations and unforeseen additions in the future. Therefore, the constraint is as follows:

$$g_3 = w_{tot} - 0.864 \leq 0 \quad (5.5)$$

5.2.3. Minimum fundamental frequency

The fundamental frequency of the system when it is deployed is constrained by requirement 16 in Table 3.2, which states that the first natural frequency in the deployed configuration shall be above 0.6. The natural frequency is calculated by assuming that the solar array acts as a cantilever beam, as depicted in Figure 5.2, with an array mass m_{array} and tip mass m_{tip} . The array mass includes the solar blanket mass and stiffener mass. When more than one stiffener is used, the second moment of area scales with n_{stiff} .

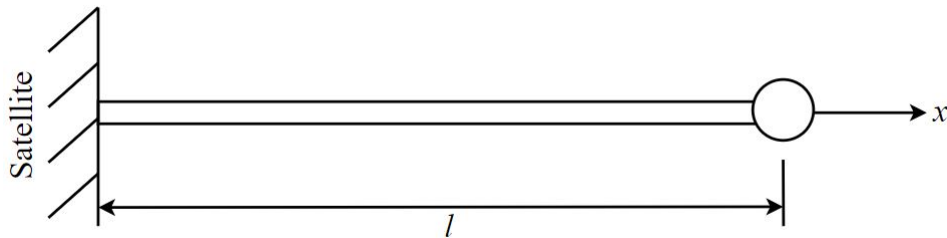


Figure 5.2: Simplification of the solar array problem as a cantilever beam. The circle at the end of the beam represents the tip mass.

The formula for calculating the fundamental frequency f_z of this simplified scenario, where one end is fixed and the other is free, is taken from Roark's Formulas for Stress and Strain [97].

$$f_z = \frac{1}{2\pi} \sqrt{\frac{3E_x I_{yy} n_{stiff}}{(m_{tip} + 0.236m_{array})l^3}} \quad (5.6)$$

With E_x representing the longitudinal modulus of elasticity of the stiffening material, I_{yy} the second moment of area about the y axis, and l the length of the solar array. The geometric properties of the stiffeners can be found in Appendix A. Similarly, the formula for the natural frequency in the y direction is calculated with:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{3E_x I_{zz} n_{stiff}}{(m_{tip} + 0.236m_{array})l^3}} \quad (5.7)$$

Finally, the natural frequency in the x direction, also known as the torsional frequency, is computed with Equation 5.8 [97].

$$f_x = \frac{1}{2\pi} \sqrt{\frac{GK}{(J + J_s/3)l}} \quad (5.8)$$

With K representing the torsional constant of the stiffener, G the shear modulus, J the mass moment of inertia of the end mass, and J_s the distributed mass moment of inertia of the beam. With this, the constraint is set as follows:

$$g_4 = 0.6 - \min(f_z, f_y, f_x) \leq 0 \quad (5.9)$$

5.2.4. Maximum stress due to boom flattening

When stiffeners transition from their deployed configuration to a stowed one, large deformations take place which load the material in various ways. The first type of deformation is the flattening of the booms as they transition from a deployed cross section to a flat one. When the stiffeners deform in this way, the stresses are influenced by the transverse modulus of elasticity, thickness of the stiffener, and radius of the boom, as depicted in Equation 5.10.

$$\sigma_y = \frac{E_y t}{2r} \quad (5.10)$$

With r representing the radius of the stiffener. If the stress exceeds the material's transverse strength, it can lead to failure, making it crucial to include it as a constraint. This constraint has direct effect on the boom thickness and radius of the stiffener, and thus must be satisfied as follows:

$$g_5 = \sigma_y - \sigma_{y_{max}} \leq 0 \quad (5.11)$$

5.2.5. Maximum stress due to boom coiling

After the stiffeners have flattened, they must coil around the mandrel to complete deployment. While this also incurs stresses in the material, it only stretches material in the longitudinal direction, and thus the respective modulus of elasticity has to be used. For isotropic materials this wouldn't change, but for composites that are anisotropic, the modulus of elasticity applicable to this is different than for the flattening constraint.

$$\sigma_x = \frac{E_x t}{2R} \quad (5.12)$$

With R representing the radius of the mandrel. This will have a direct effect on the thickness of the boom and radius of the mandrel. Similar to the flattening stress constraint, the following relation will have to be satisfied:

$$g_6 = \sigma_x - \sigma_{x_{max}} \leq 0 \quad (5.13)$$

5.2.6. Maximum shear stress due to flattening and coiling

The maximum shear stress in the boom is calculated using Mohr's circle by assuming that the two previously calculated stresses are the principal stresses. The maximum shear stress is consequently calculated with Equation 5.14.

$$\tau = \frac{1}{2} \left(\frac{E_x t}{2r} - \frac{E_y t}{2R} \right) \quad (5.14)$$

Thus, the constraint shall satisfy:

$$g_7 = \tau - \tau_{max} \leq 0 \quad (5.15)$$

5.2.7. Minimum mandrel radius

This constraint arises due to the bending limits of the solar blanket when it is coiled around the stowage mandrel. In particular, this constraint ensures that the radius of the mandrel is larger enough so that the solar blanket does not fail due to bending. The radius that a given composite material which consists of many components can bend around without failing is given by the following expression:

$$R_{\text{composite}} > \frac{E_{\text{composite}} t_{\text{composite}}}{2\sigma_{y,\text{composite}}} \quad (5.16)$$

When two different materials are bonded together like in a solar blanket, their combined bending behavior involves the interaction of their Young's modulus and ultimate strengths, which in turn can lead to a composite structure that behaves differently from either material alone. Since these values of the composite structure can only be discovered through experimentation, the largest bending radius of the individual subcomponents of the solar blanket is taken as the constraining mandrel radius. To account for this uncertainty, a factor of safety of two is taken.

The two constraining components of the solar blanket are the solar cells and coverglass. For the solar cells, the minimum radius is given by the solar cell manufacturer, and thus no calculation is needed. For the coverglass, the minimum bend radius is calculated with Equation 5.17. Since the properties of the coverglass are fixed, this will directly constrain the mandrel radius such that the coverglass does not fail when coiling.

$$R \geq \frac{Et}{2\sigma_y} \quad (5.17)$$

For the coverglass, it is assumed that the material is tempered glass. The elastic modulus of tempered glass is 70 *GPa*, and the yield strength is 120 *MPa* [98][99]. The other components of the solar blanket are assumed to not be limiting as they are either much thinner than the coverglass and solar cell (interconnect, kapton), or have a more flexible nature (adhesive). Consequently, the next constraint given to the optimisation model is as follows:

$$g_5 = \max(r_{cg}, r_{sc}) - r \leq 0 \quad (5.18)$$

5.2.8. Maximum root bending stress due to linear acceleration

According to requirement 15 in Table 3.2, the solar array shall withstand a root bending moment imposed by a 28N thruster acting on the satellite. This thruster creates a linear acceleration which causes the solar array to bend similar to the way depicted in Figure 5.3 and Figure 5.3, depending on the direction of thrust.

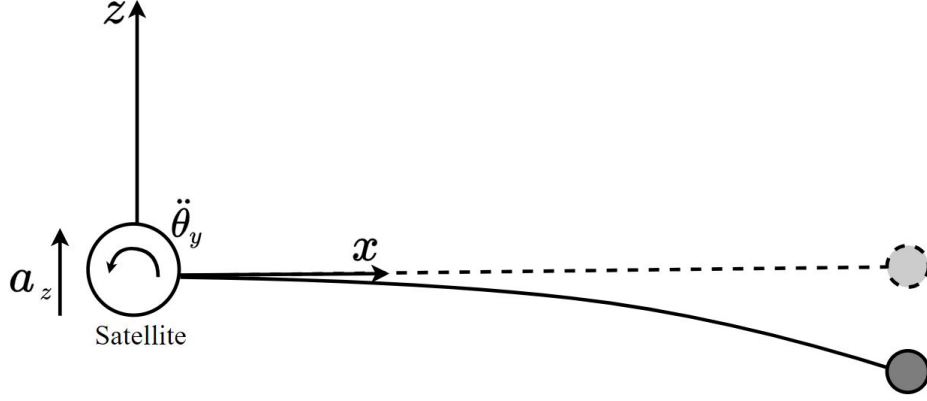


Figure 5.3: Solar array deflecting due to linear acceleration a_z or angular acceleration $\ddot{\theta}_y$.

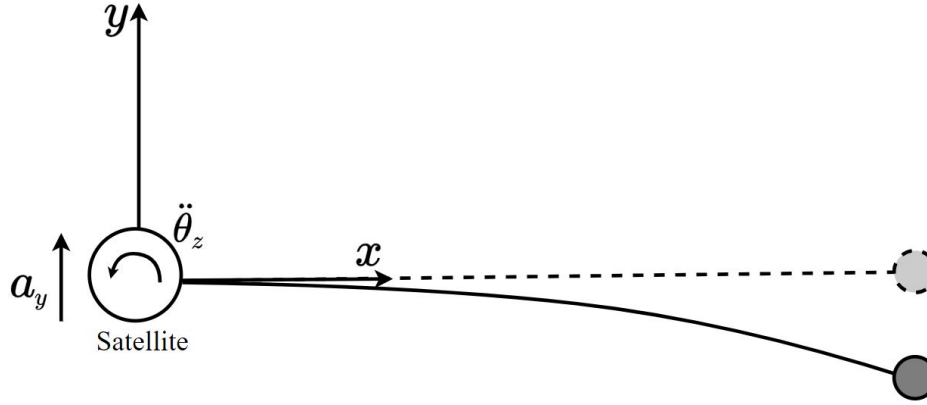


Figure 5.4: Solar array deflecting due to linear acceleration a_y or angular acceleration $\ddot{\theta}_z$.

To calculate the linear acceleration,

$$a = \frac{F}{m_{sat}} \quad (5.19)$$

Where F represents the thrust force, and m_{sat} the mass of the satellite. Assuming the worst case scenario, the smallest mass of the satellite shall be chosen such that the acceleration is highest. A mass of 50 kg is chosen. Since a_y and a_z are equal, the root bending moment with a linear acceleration is calculated as follows:

$$M_{bend} = a (m_{tip} l + 0.5(m_{sb} + m_{stiff} n_{stiff}) \frac{l}{2}) \alpha \quad (5.20)$$

With α being the dynamic overshoot factor and assumed to be equal to 2 [86]. If more than one stiffener is used, the moment is divided by n_{stiff} . The stresses can now be calculated as follows:

$$\sigma_z = \frac{M_{bend} c}{I_{yy} n_{stiff}} \quad (5.21)$$

and

$$\sigma_y = \frac{M_{bend} c}{I_{zz} n_{stiff}} \quad (5.22)$$

Finally, the constraint is calculated using the maximum of the two stresses, as follows:

$$g_6 = \max(\sigma_z, \sigma_y) - \sigma_{max} \leq 0 \quad (5.23)$$

5.2.9. Maximum root bending stress due to angular acceleration

The maximum root bending due to angular acceleration is defined by a similar physical problem, but with the only variation being the type of acceleration. This constraint is driven by requirement 17 in Table 3.2, which states that the solar array shall withstand angular accelerations of $\ddot{\theta} = 2.5 \text{ deg/s}^2$ in any direction. The root bending moment due to the angular acceleration $\ddot{\theta}$ including dynamic overshoot factor is then calculated as follows:

$$M_{bend} = \ddot{\theta} (m_{tip} l^2 + \frac{1}{3} (m_{sb} + m_{stiff} n_{stiff}) l^2) \alpha \quad (5.24)$$

With the bending moment known, the maximum stresses in both directions are computed. When more than one stiffener is used, the moment is distributed accordingly.

$$\sigma_z = \frac{M_{bend} c}{I_{yy} n_{stiff}} \quad (5.25)$$

and

$$\sigma_y = \frac{M_{bend} c}{I_{zz} n_{stiff}} \quad (5.26)$$

The constraint is then driven by the maximum of the two stresses and given as follows:

$$g_7 = \max(\sigma_z, \sigma_y) - \sigma_{max} \leq 0 \quad (5.27)$$

5.2.10. Maximum torsional stress due to angular acceleration

The final stress constraint is related to the torsional stress produced by the angular acceleration in the x direction, as seen in Figure 5.5.

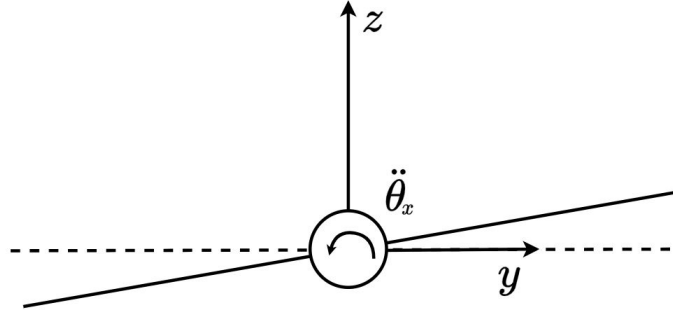


Figure 5.5: Solar array deflecting due to angular acceleration $\ddot{\theta}_x$. The solid line represents the position of the solar array at the root, while the dashed line represents the position at the tip.

The torque transferred to the solar array due to the angular acceleration is as follows:

$$T = I_x \ddot{\theta}_x \quad (5.28)$$

With I_x representing the mass moment of inertia of the array which is dependent on its configuration and includes a few simplifications. For example, the solar blanket is assumed to be a uniform mass rotating around the x axis, while external stiffeners are assumed to be a point mass. With this, the shear stress can be computed using simplified approximations for thin walled structures from Roark's Formulas for Stress and Strain [97]. For open structures with uniform thickness, the maximum stress is given by Equation 5.29.

$$\tau = \frac{T(3U + 1.8t)}{U^2 t^2} \quad (5.29)$$

With U representing the perimeter of the structure. For closed structures with uniform thickness, the stress, which is uniform if t is small enough, is given by:

$$\tau = \frac{T}{2tA} \quad (5.30)$$

With A representing the area enclosed by the structure. Having computed the stress due to torsion, the constraint can be formulated such that the shear stress does not exceed the maximum shear strength (τ_{max}) of the material.

$$g_8 = \tau - \tau_{max} \leq 0 \quad (5.31)$$

For design configurations where there are two or more external stiffeners, the torsional stress due to angular acceleration cannot be calculated with Equation 5.29 or Equation 5.30 anymore as the problem

changes significantly. Nevertheless, a simplification can be made such that the problem can be computed without the need of advanced models. A design which has two stiffeners can be simplified as depicted in Figure 5.6. In this scenario, it assumes the twist due to the angular acceleration is small enough such that the displacement of the two beams is vertical, causing them both to bend.

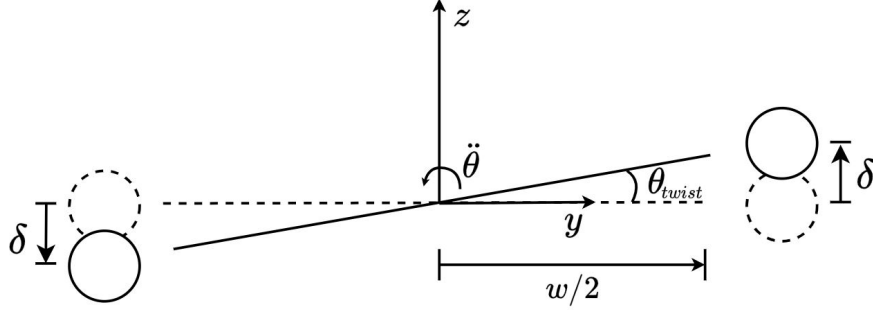


Figure 5.6: Simplification of the twist when two external stiffeners are used. The dashed lines represents the initial position, and the solid line represents the final position after twisting.

The root bending moment can then be calculated with Equation 5.32.

$$M_{bend} = P L \quad (5.32)$$

Where L is the length of the stiffener, and $P = \frac{T}{w/2}$, the torsional stress due to the angular acceleration for designs with more than one stiffener can be calculated using Equation 5.21, while keeping in mind the appropriate number of stiffeners to use depending on the design. The constraint can then be formulated as follows:

$$g_8 = \sigma_z - \sigma_{max} \leq 0 \quad (5.33)$$

5.2.11. Maximum twist angle

For the torsional moment, the shear stress is not the only component which shall be analysed, and a maximum twist is set by requirement 18 in Table 3.2. The twist due to a torsional moment caused by an angular acceleration in the x direction is calculated with Equation 5.34.

$$\theta_{twist} = \frac{T l}{K G} \quad (5.34)$$

With T representing the torque, K the torsion constant, and G the shear modulus of the stiffeners' material. The torsion constant is geometry dependent and for each stiffener shape can be found in Appendix A. When more than one stiffener is used for a design, the problem is changed again to the simplification visualised in Figure 5.6. The maximum deflection at the tip is calculated as follows:

$$\delta = \frac{Pl^3}{3E_x I_{yy}} \quad (5.35)$$

With $P = \frac{T}{w/2}$. Since there are two beams in bending, the deflection is halved. By using a relation between the deflection and twist, the twist can be calculated for configurations which use two stiffeners.

$$\theta_{twist} = \frac{\delta}{w} \quad (5.36)$$

The twist is set to be a maximum of 5 deg as required by Revolv. Also, any angle larger than this would also not classify it as small enough for the simplification in Figure 5.6 to be valid anymore. The constraint is thus defined as follows:

$$g_9 = \theta_{twist} - 5 \leq 0 \quad (5.37)$$

5.3. Deployment mechanism

The deployment mechanism is one of the four major design elements of the roll-out solar array, and will be discussed in detail in the following section. During deployment of the roll-out solar array, there are a few factors which affect the mechanism which initiates and controls the deployment sequence. First, a mechanism which is necessary to prevent a type of deployment failure mode called blossoming is detailed in subsection 5.3.1. Then, two inherent torques identified to be present in the deployment of the stiffener are the self deployment torque and frictional torque, each of which are described in subsection 5.3.2 and subsection 5.3.3, respectively. Lastly, the two possible deployment mechanisms which are implemented as design variables in the optimisation model are discussed in subsection 5.3.4 and subsection 5.3.5.

5.3.1. Anti-blossoming mechanism

A prevalent issue with deployable booms is blossoming. This type of event is critical to investigate, as overlooking it can result in the failure of the roll-out solar array. When a stiffener is deployed by rotating a center mandrel, there is a possible scenario where the mandrel turns but the stiffener does not deploy. Instead, the coils will increase in radius, eventually leading the innermost layer to fail or be damaged irreversibly. This is called blossoming, and a visualisation of this type of deployment failure is seen in the left image of Figure 5.7. Nevertheless, blossoming can be prevented by ensuring that the coils have enough inward pressure to not blossom outwards when the mandrel turns. A simplification of this solution is seen in the right image of Figure 5.7, wherein a set of compression springs create enough inward force such that blossoming does not occur.

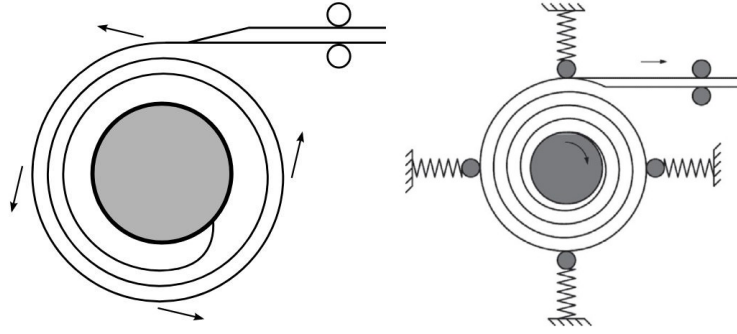


Figure 5.7: Blossoming of a deployable stiffener (left) [100], normal deployment with compression rollers (right) [101].

To achieve the effect of springs acting on the coiled stiffener, designs have opted to use a flexure spring design. While the use of normal springs, as shown in the right image of Figure 5.7, is possible, it is outshined by the benefits of a flexure spring design. This method uses the principle of bending a metal sheet to create an equivalent spring stiffness, meaning that this design offers a much more compact solution since the metal sheets can be designed to curve and fit the design of the stowed roll-out solar array. Thus, it is concluded that a design with flexure springs is the most optimal and assumed to be used as the anti-blossoming device.

One spring assembly to mitigate the blossoming effect of deployable booms has been designed and validated [102]. This design, seen in Figure 5.8, features a circular frame with rollers attached to thin strips which act a spring that apply a force inwards towards the center of the mandrel, effectively preventing the boom to blossom outwards. The design is made for a 2 m lenticular boom, restricted around a 5 cm radius mandrel, weighs 60 g, and each roller has a spring stiffness of 5 N/mm. During the development of the optimisation model, results showed that the boom would not be smaller than 5 cm, and the boom would be coiled around a 6 cm mandrel, meaning that the anti blossoming assembly is expected to be larger than the one presented in the paper. In order to size this to an arbitrary design, a scaling function is necessary.



Figure 5.8: Flexure spring mechanism to prevent blossoming of a deployable stiffener. [102]

To size the anti-blossoming design, scaling functions are developed. For the volume, the rollers are estimated to have a radius of 1 *cm* and are used as the limiting part since the frame is much thinner than the rollers. Therefore, an additional 2 *cm* is added to the final radius of the stowed array, since the diameter of the roller is added to the final radius. For the mass, a scaling function based on the circumference is developed. It is assumed that the device is a tube with equal thickness. First, the mass per unit circumference is calculated based on the known mass and size of the device. With a mass of 60 *g* and an outer radius of 5.35 *cm* (including the added radius of the coiled radius), the mass per unit circumference is 1.78 *g/cm*, or 0.178 *kg/m*. With this, the mass of an anti-blossoming device with different final radius can be estimated with the scaling function given in Equation 5.38.

$$m_{bloss.} = 0.178 (2\pi R_{final}) \quad (5.38)$$

In reality, the rollers and attachment gap would introduce variations in thickness, leading to localized changes in the mass distribution. Nevertheless, the estimation will suffice for a design at a conceptual level. Furthermore, since the rollers in reality would not scale with the radius, the estimation would be an overestimation, which is favourable at this stage of the design.

5.3.2. Self deployment torque

The self deployment torque is inherent to the system and arises due to the strain energy stored in the stiffener when it is flattened and coiled around the mandrel. The stored strain energy in a tape spring made of an isotropic material is given by Equation 5.39 [103].

$$U = \frac{1}{2}DA [(\Delta\kappa_x)^2 + (\Delta\kappa_y)^2 + 2\nu(\Delta\kappa_x\Delta\kappa_y)] \quad (5.39)$$

With D representing the flexural rigidity of the material, and given in Equation 5.40, A the contact area, and $\kappa_y = \frac{1}{r_y}$ and $\kappa_x = \frac{1}{r_x}$ representing the curvatures of the tape spring about their respective axes, as seen in Figure 5.9.

$$D = \frac{Et^3}{12(1-\nu^2)} \quad (5.40)$$

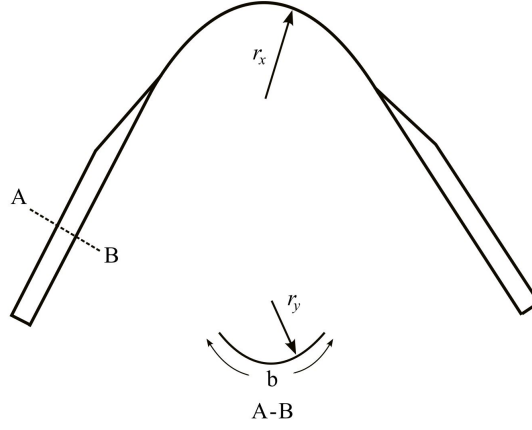


Figure 5.9: Geometry and curvatures of a tape spring in bending. This drawing is inspired by figure 5 in [103].

Tape springs can be bent in two ways as seen in Figure 5.10, and are called equal sense and opposite sense bending depending on the direction of bending. The way in which they are bent has an effect on the stored energy, with the stored energy in opposite sense bending given in Equation 5.41.

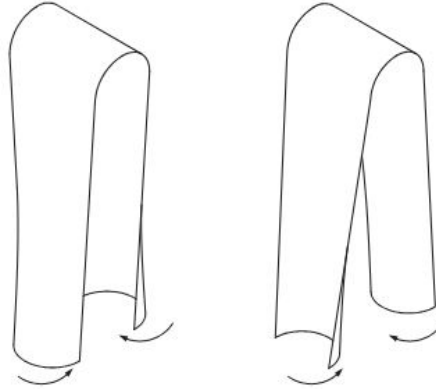


Figure 5.10: Equal sense bending (left), vs opposite sense bending (right). [101]

$$U_{op} = \frac{Db\theta}{2} \left[\frac{1}{r_x} + \frac{r_x}{r_y^2} + \frac{2\nu}{r_y} \right] \quad (5.41)$$

With equal sense coiling, the κ_x is reversed and taken as $-\kappa_x$, and thus the strain energy is given by Equation 5.42.

$$U_{eq} = \frac{Db\theta}{2} \left[\frac{1}{r_x} + \frac{r_x}{r_y^2} - \frac{2\nu}{r_y} \right] \quad (5.42)$$

To get the torque, the change in energy per change in angle is calculated as given in Equation 5.43.

$$T = \frac{dU}{d\theta} \quad (5.43)$$

This results in the self deployment torque for isotropic materials to be following:

$$T_{sd_{iso}} = \frac{Db}{2} \left[\frac{1}{r_x} + \frac{r_x}{r_y^2} \pm \frac{2\nu}{r_y} \right] \quad (5.44)$$

For anisotropic materials like carbon fiber composites, Equation 5.39 changes slightly to accommodate for the different mechanical properties that the material has in different directions [104].

$$U = \frac{b\theta}{2} \left(\frac{D_x}{r_x} + \frac{D_y r_x}{r_y^2} \pm 2D_1 \frac{1}{r_y} \right) \quad (5.45)$$

$$T_{sd_{aniso}} = \frac{b}{2} \left(\frac{D_x}{r_x} + \frac{D_y r_x}{r_y^2} \pm 2D_1 \frac{1}{r_y} \right) \quad (5.46)$$

With the flexural rigidity's defined as follows:

$$D_x = \frac{E_x t^3}{12(1 - v_{xy}v_{yx})} \quad (5.47)$$

$$D_y = \frac{E_y t^3}{12(1 - v_{xy}v_{yx})} \quad (5.48)$$

$$D_1 = v_{xy}D_y \quad (5.49)$$

With these sets of equations, the self deployment torque of any stiffener can be computed by breaking down the stiffeners to a combination of tape springs and sheets and summing their individual deployment torque. For example, the torque due to the strain energy in a TRAC stiffener with anisotropic material properties can be calculated in this way. Firstly, the structure is split into its three main components: the upper flange which experiences opposite sense bending, the lower flange which experiences equal sense bending, and the bond region, for which $\Delta\kappa_y = 0$. Consequently, the strain energy is given as follows:

$$U_{TRAC} = \frac{b\theta}{2} \left(\frac{D_x}{r_x} + \frac{D_y r_x}{r_y^2} + 2\frac{D_1}{r_y} \right) + \frac{b\theta}{2} \left(\frac{D_x}{r_x} + \frac{D_y r_x}{r_y^2} - 2\frac{D_1}{r_y} \right) + \frac{w\theta}{2} \frac{D_x}{r_x} \quad (5.50)$$

Simplifying the equation and taking the derivative with respect to θ , the self deployment torque for the TRAC stiffener is given by Equation 5.51. This final formulation is consistent with a derivation from another paper, validating the method of calculating the self deployment torque [105].

$$T_{TRAC} = b \left(\frac{D_x}{r_x} + \frac{D_y r_x}{r_y^2} \right) + \frac{w}{2} \frac{D_x}{r_x} \quad (5.51)$$

5.3.3. Frictional torque

The frictional torque arises due to the contact region between the outermost layer of the coiled stiffener and the compression rollers. While rollers generally have low frictional coefficients, they are not assumed to produce negligible friction in this study as they are the only resistive contribution to the deployment process, meaning that even a small amount is important to include in order to understand the complete system behaviour. The frictional force from each compression roller is calculated using the normal force and the frictional coefficient of the rollers. The normal force, or compression force, is calculated by taking the product of the spring stiffness and the displacement of the flexure spring, which is determined by the number of wraps around the mandrel. To find the resulting torque due to the friction, the frictional force from all the compression rollers is added and multiplied by the radius at which they act on.

$$T_f = k_f N_{cr} F_{cr} C_f r_m \quad (5.52)$$

With N_{cr} representing the number of compression rollers, k_f the factor of safety, F_{cr} the compression roller force, C_f the coefficient of friction of the stiffener material, and r_m the mandrel radius. For the coefficient of rolling friction, a steel roller on steel plate has high estimate of 0.005, and thus is assumed [106]. From ECSS-E-ST-33-01C, the safety of factor for frictional forces or torques is taken as 3 [107]. An important note is that as the compression rollers act as a spring loaded system, the normal force on the outermost layer decreases during deployment. This is because the compression rollers loses compression, therefore reduce the overall friction torque as the solar array deploys.

This relation, however, has the limitation of neglecting the additional frictional forces in the case that the system would start blossoming. This could be a critical risk that would have to be tested further to ensure that it does not occur at any stage during deployment. Since blossoming is assumed to not initiate during deployment, the frictional torque calculated shall be accepted as a reference for the torque resisting deployment. Nevertheless, experiments or detailed FEM analyses are recommended to capture the complete and real interactions of this mechanism.

5.3.4. Motor sizing

With the self deployment torque and frictional torque computed, the residual torque in the system can be calculated as in Equation 5.53 to find the torque required to deploy the solar array.

$$T_r = T_{sd} - T_f \quad (5.53)$$

One method to achieve this remaining torque to drive deployment is to use a gearbox and motor system. Motors will operate at high speeds but in turn output low torque, and are often not powerful enough to deploy systems by themselves. A gearbox addresses this issue by having a high torque while decreasing the speed. To size the gearbox and motor, empirical data will be used to facilitate the design process. While this is not a completely accurate process, it gives a suitable method to size these system at a conceptual level. Hillebrandt amassed catalogue data from numerous companies in order to come up with empirical approximations for the mass of gearboxes and electric motors against their torque [74]. All of the data collected were from commercially off the shelf products qualified for space use, and the empirical approximation for both the gearbox and motor mass are plotted in Figure 5.11.

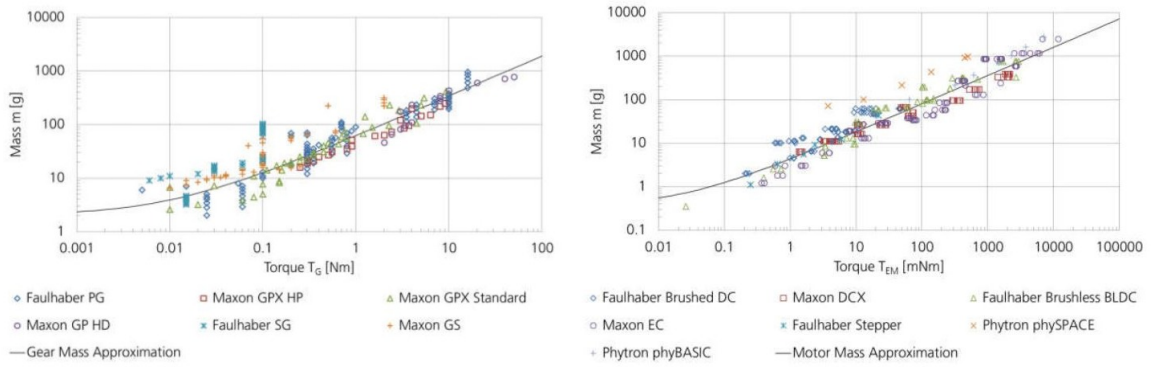


Figure 5.11: Gearbox mass vs torque approximation (left), and electric motor mass vs torque approximation (right). [74]

Before calculating the respective masses, the torque of the electric motor is still required. This can be obtained with Equation 5.54, where the gearbox efficiency η is assumed to be 0.55, while the gear ratio k is assumed to be 1:300 as is estimated in the paper.

$$T_{em} = \frac{T_r k}{\eta} \quad (5.54)$$

With the gearbox torque equal to T_r , and with the torque of the electric motor known, the masses of the two mechanisms can be computed with the empirical relations given below, which are derived from the trend line approximations in Figure 5.11.

$$m_{gb} = 60 T_r^{0.75} + 2 \quad (5.55)$$

$$m_{em} = 4 T_{em}^{0.65} + 0.35 \quad (5.56)$$

The motors and gearbox are assumed to be small enough to be integrated inside the mandrel, and thus the volume calculations are not necessary since they would not affect the total volume of the roll-out solar array. Similar systems which deploy a stiffener or similar deployable structure have achieved this, hence validating this assumption [47] [102].

5.3.5. Mechanical damper sizing

The sizing of the mechanism is largely based on the system shown in Figure 4.8, but will be based on a scaling function such that it is sized according to an input variable. The width w of the escapement wheel is assumed to be 1 cm, and the material is also assumed to be made of an aluminium alloy with a density of 2700 kg/m^3 . Furthermore, the detent protrudes about 40% from the outermost point of the escapement wheel in Figure 4.8, but is assumed that this additional area will already be accounted for in the additional volume considerations of the deployment mode. The functionality of the mechanism with these assumptions would have to be verified with experimentation to make sure that the impact of the detent on the teeth of the escapement wheels can be withstood. The mass of the mechanical damper can then be calculated by estimating the volume of the mechanism and taking the product of it with the density, mathematically layed out in the scaling function of Equation 5.57. An additional mass of 15% is added as part of the detent, which is calculated by analyzing the area of the detent as a percentage of the escapement wheel.

$$m_{md} = (w(\pi R^2)\rho)1.15 \quad (5.57)$$

This method of sizing the mechanical damper is limited in the sense that it is based on one existing mechanism. While it is for the same application of deployable booms, the sizing of it in the detailed design stage may cause it to have very different characteristics. Nevertheless, it does provide a simple method for giving a preliminary estimate of the mass that it would require. One immediate improvement which would increase the performance of this deployment mechanism would be to include lightening holes. These could be included such that the mass of the mechanism reduces without hindering the structural performance of the mechanism. Again, it is recommended that specific FEM analyses or experimental tests are conducted to verify the functionality of the final mechanism.

5.4. Deployment mode

This section explores the effects that the two deployment modes have on the design of the roll-out solar array. Depending on the mode, a certain additional volume will be required, and the method for determining this is given in subsection 5.4.1. Furthermore, when an external stiffener design is employed with the normal deployment mode, a spreader bar is required, and its design is explained in subsection 5.4.2.

5.4.1. Additional volume considerations

For deployable structures and stiffeners, an important characteristic which impacts the stowed volume is their ploy region. This is the region when the stiffener transitions from a coiled state to its final state. Usually, designs employing deployable structures have opted to include the ploy region in their stowed design such that a fully constrained root can be achieved [45] [108]. This however takes a lot of extra space due to the significant length of the ploy region, and the most common alternative is to have the stiffener not fully transitioned when it is stowed. Once the deployment initiates, the stiffener continues to transition until the ploy length has been achieved and it has transitioned to its final shape. This solution comes with the limitation that the stiffener is not fully constrained, since it still has to undergo a transition period. This means that the fully fixed root constraint assumption is not entirely correct. For the conceptual design phase, this is an acceptable difference, and would have to be analysed more extensively in the following stages of design.

Due to the stiffener protruding out from the circular envelope of the stowed roll-out solar array, it has to be accounted for in the stowed volume consideration. The additional volume that is required comes from the extra space outside of the stowed solar array, and can be visualised in Figure 5.12. The extra space is sized with the factor ζ , and scales with the final radius of the roll-out solar array, which includes the layers of the wound up solar blanket and stiffeners.

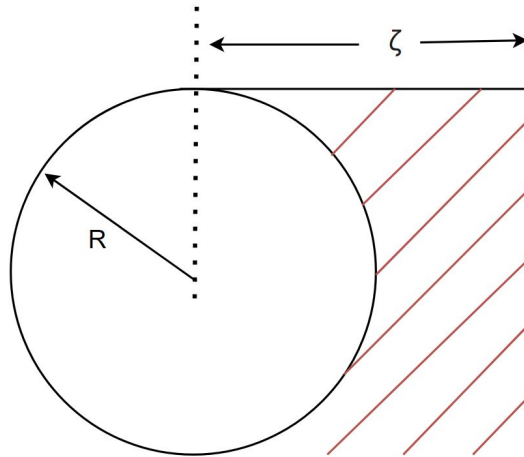


Figure 5.12: Extra volume required for roll-out solar arrays, represented by the red area.

From existing roll-out solar arrays, the ζ can be estimated. For the normal deployment, two designs which employ this type of deployment were analysed, and averaged to 2.33 [109][110]. For the inverted deployment, a factor of 1.75 is used and based on the iROSA design. While a single data point to source this factor is not robust, no other roll-out solar arrays which employ the inverted deployment exist. Table 5.3 summarises the extra space needed by the two deployment modes. These are used in the optimisation to estimate the total volume that will be needed by the roll-out solar array when it is stowed.

Table 5.3: Extra volume factor for the two deployment modes

	Normal deployment	Inverted deployment
ζ	2.33	1.75

5.4.2. Spreader bar

For designs with external stiffeners and normal deployment mode, a spreader bar is required to be added at the beginning of the array. This component ensures structural stability, and acts as an attachment point for the solar blanket. One method to size this is to base it on the tensioning load that would be required to keep the solar blanket taught during and post deployment. As seen in Figure 5.13, which shows the tensioning load for a single stiffener design, the problem acts similar to a beam that is under a uniform distributed load. The tensioning is caused by constant force springs at the root of the solar array, which then goes through the blanket and creates a distributed load on the spreader bar.

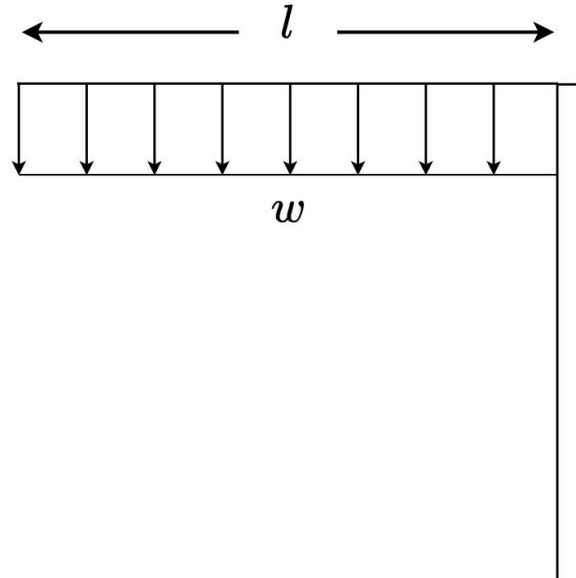


Figure 5.13: Tensioning load on the spreader bar.

A tensioning load w of 140 N/m is used and based on the tensioning load that is used for the legacy fold out solar arrays [111]. Since a roll-out solar array is a similar structure with a flexible solar blanket and supporting beams, the same tensioning is assumed. With this, some other assumptions are made in order to be able to size the spreader bar. Firstly, it is assumed that the bar is a hollow tube made out of aluminium. Then it is assumed that the bar would have a radius of 1.5 cm , ensuring it is compact enough to integrate with the system. With these assumptions, the thickness can be computed such that the load does not exceed the yielding properties of the bar. For the problem shown in Figure 5.13, the maximum moment M at the root is given by Equation 5.58.

$$M = \frac{w l^2}{2} \quad (5.58)$$

With w representing the uniformly distributed load, and l being half the length of the spreader bar. A factor of safety of 2 is added to the distributed load due to uncertainties in this early stage of the design. Furthermore, the maximum stress is computed with:

$$\sigma = \frac{M c}{I} \quad (5.59)$$

With I representing the second moment of area, given by $\frac{\pi(R^4 - (R-t)^4)}{4}$, and c representing the furthest point of the beam from the neutral axis, which in the case of the bar would be 1.5 cm . Expanding the second moment of area, and rearranging Equation 5.59, the thickness t can be computed with Equation 5.60.

$$t = R - \sqrt[4]{R^4 - \frac{4 M c}{\pi \sigma}} \quad (5.60)$$

For a solar array with a spreader bar length of 1 m , the thickness is required to be at least 1.4 mm to be able to withstand the load and not yield. This would also cause the tip to deflect a maximum of 2.4 mm , which is an acceptable deformation. It is expected that actual values for optimised arrays would be smaller than this considering the length of the bar would be shorter. The mass is then computed with the density of aluminium and cross sectional area of the bar. In the optimisation model, this is included in the total mass of the solar array, and is assumed to be the tip mass for designs using external stiffeners and normal deployment mode. For configurations with two stiffeners, the problem is slightly altered such that the bar acts as a beam with two fixed ends. Lastly, a spreader bar is assumed to not be required for inherent stiffener configurations. This is because the structure is self supporting, and the blanket does not need to be tensioned.

5.5. Mass calculation

The following section describes in detail how the mass of the solar array is divided amongst its components and calculated. Firstly, a breakdown of all the components of the solar blanket is given in subsection 5.5.1, along with the function to calculate the blanket mass. Then, the stiffener mass formulation is given in subsection 5.5.2, followed by a method to compute other unaccounted for masses in subsection 5.5.3. Finally, the derivation to compute the total mass is given in subsection 5.5.4.

5.5.1. Solar blanket mass

The mass of the solar blanket is calculated by considering each layer in it. Figure 5.14 shows each individual layer of the solar blanket. While this might be adapted in real life, it gives an indication of a common breakdown of a flexible solar blanket [64].

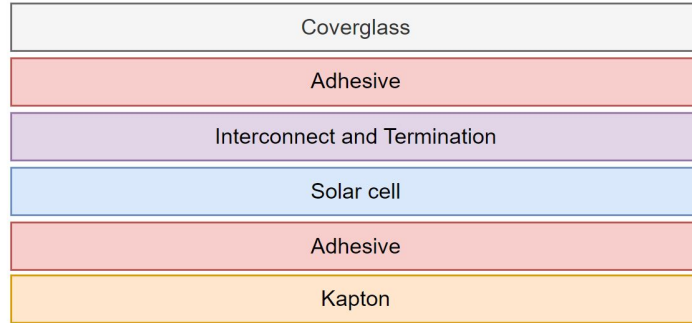


Figure 5.14: Components of the flexible solar blanket. Relative sizes are not to scale.

Additionally, the material properties of the components of the solar blanket can be found in Table 5.4. The mass and thickness of the solar cells can be found in Table 4.1.

Table 5.4: Properties of the solar blanket components.

Component	Mass (kg/m ²)	Thickness (mm)	Ref.
Coverglass	0.238	0.1	[55]
Kapton	0.038	0.05	[64]
Adhesive	0.107	0.05	[64]
Interconnect	0.0097	0.025	[64]
Termination	0.0257	0.050	[64]

Since each mass is given per unit area, the mass of the solar blanket is calculated as follows:

$$m_{blanket} = A_{sb}(m_{cg} + 2m_{adhesive} + m_{interconnect} + m_{termination} + m_{sc} + m_{kapton}) \quad (5.61)$$

5.5.2. Stiffener mass

For the stiffening structures, their mass is calculated as follows:

$$m_{stiff} = (\rho w A) n_{stiff} \quad (5.62)$$

with ρ representing the material density, w the width of the array, and A the cross sectional area of the stiffener. If more than one stiffener is used, the mass of a single stiffener is multiplied by the amount of stiffeners n_{stiff} present. Since the optimisation model does not analyse combinations of different stiffeners, this method is valid.

5.5.3. Other masses

Designing a roll-out solar array involves many different components and requires careful consideration of all structural constituents. While the major structural elements like the stiffeners are estimated, there are additional parts that may be overlooked and not accounted for. These include harnessing and cabling, attachments, release mechanisms, and tensioning springs. Furthermore, while they might not have a large individual mass, their summed mass is not insignificant, and therefore should be accounted for. Although these components are difficult to size, especially in the early stages of design, they still require mass allocation, and thus a scaling function is required for this additional unaccounted for mass.

One method to estimate the additional structural mass is to use a mass fraction such that a scaling function can be formulated. The structural mass fraction of solar arrays relates the structural mass to the solar blanket mass. Since all structural components except the additional mass is known, a scaling function can be found through this. According to NASA, a structural mass fraction of 0.2 is deemed as a feasible target for efficient solar arrays such as roll-outs and fold-outs [112]. While the ISS solar arrays and Ultraflex have a structural mass fraction of around 0.4, these are old systems, and therefore it can be assumed that an innovative system today would have a lower structural mass fraction. Thus, a relation between the structural mass of the roll-out solar array and the mass of the solar blanket is given by Equation 5.63.

$$\frac{m_{structure}}{m_{blanket}} = 0.2 \quad (5.63)$$

Expanding the structural mass into its separate constituents and rearranging Equation 5.63, the mass for additional structure can be computed.

$$m_{other} = 0.2 m_{blanket} - (m_{dep.mech} + m_{stiff} + m_{drum} + m_{spreader} + m_{bloss.}) \quad (5.64)$$

5.5.4. Total mass

Finally, all the previously calculated masses can be summed to find the total mass of the solar array. Like for any space mission, margins are added to account for unforeseen events that may require additional mass to the system. This also accounts for the uncertainties in the design calculations which may differ as the design is evaluated in more depth. The amount of extra margins may depend on the design, type of spacecraft system, and level of risk that is associated with the system, but in general a margin of 20% is commonly used [113] [114]. Consequently, an additional margin of 20% is applied to the total mass. The total mass of the roll-out solar array is then calculated with Equation 5.65.

$$m_{tot} = 1.2 (m_{blanket} + m_{stiff} + m_{dep.mech.} + m_{drum} + m_{spreader} + m_{bloss.} + m_{other}) \quad (5.65)$$

5.6. Volume calculation

To find the volume of the solar array, two main components are needed: the width of the solar array and the final radius. The width is calculated by combining all the different components such as the solar blanket width, flattened stiffener width, FPC width, etc. These are ultimately constrained such that requirement 10 in Table 3.2 is complied with. The contributions from the deployment mechanism are described in section 5.3. The total width is calculated as follows:

$$w_{tot} = w_{sb} + w_{flex} + w_{stiff} n_{stiff} + w_{dep.mech} \quad (5.66)$$

For certain design configurations, certain widths may not be applicable. Then, to calculate the radius, a method to compute the amount of wraps around the mandrel is first made, which is dependent on the length of the deployed solar array. Furthermore, for inherent stiffeners, the solar blanket thickness and stiffener thickness will be combined, while for the external stiffeners the larger thickness is used. Note that the lenticular and TRAC stiffeners will have a thickness that is doubled due to their flat configuration. A packing efficiency of 80% is also applied, which assumes that the coils have some free space between them. Thus, the final radius is computed as follows:

$$r_f = r_{mandrel} + t_{eff} n_{wraps} \quad (5.67)$$

With t_{eff} representing the effective thickness including the packing efficiency. With the final width and final radius known, the total volume including the deployment mode scaling factor can be computed. Similar to the mass margin, a 20% margin on the total volume is applied to account for uncertainties. The final volume is then calculated with

$$V_{tot} = 1.2(w_{tot} r_f^2 (\frac{\pi}{2} + 2\zeta)) \quad (5.68)$$

5.7. Power calculation

The following section describes how the power of the solar array is calculated. For the calculations, it assumes a solar flux of 1361 W/m^2 , based on the value for the mean Earth-Sun distance. Furthermore, for a simplified analysis, it assumes that the solar array is perpendicular to the sunlight, and that shadowing from the satellite is not present. The method for calculating the total power is given in subsection 5.7.1, and the method for determining the power reduction due to a curved surface is given in subsection 5.7.2.

5.7.1. Solar array power

The power of a single cell is calculated with the following equation.

$$P_{cell} = S A_{cell} \eta \quad (5.69)$$

With S representing the solar flux, A_{cell} the area of the solar cell, and η the efficiency of the cell. The total power of the solar array is calculated by multiplying the number of cells in the lengthwise (n_l) and widthwise n_w direction by the power of a single cell, as given in Equation 5.70. A packing density factor of 90% is also added for both the lengthwise and widthwise direction. This is the ratio of cell area to total area to allow for gaps between cells [3]. Lastly, solar arrays in the 1 kW range usually have a 1% wiring and interconnect power loss, and thus is also added as reduction factor [3].

$$P_{tot} = 0.99 n_w n_l P_{cell} \kappa \quad (5.70)$$

With κ the percentage reduction of power due to a curved surface. This is only applicable to some inherent stiffener structures, and explained in more detail in subsection 5.7.2.

5.7.2. Power reduction due to curved surfaces

For some of the inherent stiffener structures, the solar cells will be placed on a curved surfaces, meaning that the incident sunlight will not be perpendicular to the solar cells anymore. To calculate the reduction in power generated due to curvature, the power of a small length on a curvature is calculated and then integrated over the whole structure. This model is visualised Figure 5.15.

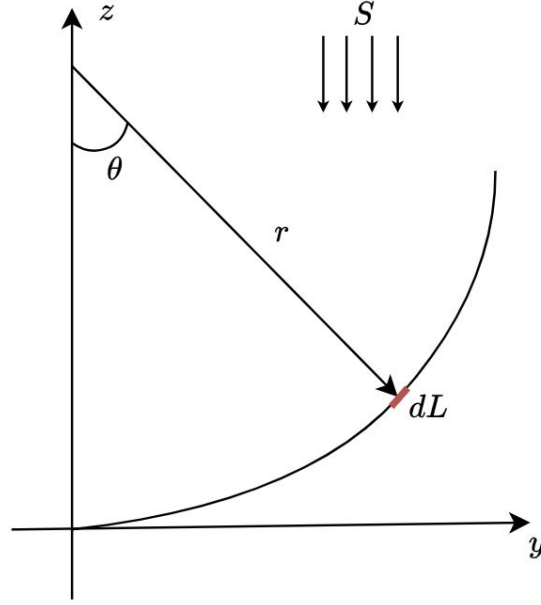


Figure 5.15: Small length (red) of a curved surface with radius r subject to incident sunlight S .

The power generated by the small length dL is the following:

$$P = S \cos(\theta) dL \quad (5.71)$$

This can be further changed with $dL = r d\theta$:

$$P = S \cos(\theta) r d\theta \quad (5.72)$$

To calculate the power over the whole surface, the integration over the required angles can be made.

$$P = Sr \int_{\theta_i}^{\theta_f} \cos(\theta) d\theta \quad (5.73)$$

With θ_i and θ_f being the initial and final angle, respectively. For example, finding the power generated by the tape spring inherent concept, assuming that it is a full half circle, results in the following power generated.

$$P = Sr \int_{-\pi/2}^{\pi/2} \cos(\theta) d\theta = 2Sr \quad (5.74)$$

Comparing that to the power generated of a flat surface of the same length, the power reduction due to the curvature is:

$$\kappa = \frac{P_{curved}}{P_{flat}} = \frac{2Sr}{\pi Sr} = 0.64 \quad (5.75)$$

Meaning that a half circle tape spring would only generate 64% of the power a flat surface of the same length would. Since this is quite a significant reduction in power, this method will be used for the inherent stiffeners which have solar cells on a curved surface. Additionally, Equation 5.73 can be used in a similar way for the lenticular stiffener as well.

5.8. Design objectives

The outputs of the analytical model will be the power-to-mass ratio and the stowed power density. The objectives are defined with the following symbols: ξ for the power-to-mass ratio and ψ for the stowed power density, and are calculated as follows:

$$\xi = \frac{P_{tot}}{m_{tot}} \quad (5.76)$$

$$\psi = \frac{P_{tot}}{V_{tot}} \quad (5.77)$$

5.9. Optimisation model

The most important challenge of this problem is the fact that it is a multi objective optimisation problem. While single-objective optimisation requires only the minimum or maximum solution to be found based on a single criteria, multi-objective optimisation involves finding a solution which satisfies multiple objectives at once. It is common that the objectives have competing goals, meaning that increasing the performance on one objective will negatively affect the performance on the other objectives. This optimisation problem is further complicated by the usage of different types of variables, also known as the mixed variable problem. For example, the type of boom for the external stiffeners is a choice with discrete options, while the dimensions such as width and length are determined by the integer-valued counts of solar cells. On the other hand, variables like the subtended angle of the STEM or TRAC booms can be any real value within a range. This diversity presents a more complex optimisation problem in comparison to classic mathematical optimisation problems, which typically involve only real-numbered variables. The following section presents the algorithm which is used to solve this problem, along with an explanation of how it is integrated with the analytical model.

5.9.1. NSGA-II algorithm

This multi-objective, mixed variable problem can be tackled using the popular Non-dominated Sorting Genetic Algorithm II (NSGA-II) [115]. NSGA-II uses a modified evolutionary algorithm which sorts and ranks solutions based on Pareto optimality principles, allowing the algorithm to converge to a set of

optimal results, also known as the Pareto front. This front contains what are known as Pareto optimal solutions, which are defined to be non-dominated, meaning that an improvement in one objective cannot be made without degrading another objective. Figure 5.16 shows an example of a Pareto front of an arbitrary double-objective optimisation problem, wherein the red dots represent the non-dominated solutions. This front presents a trade-off which decision-makers use to evaluate the final solution based on weighted scoring or using a different set of criteria.

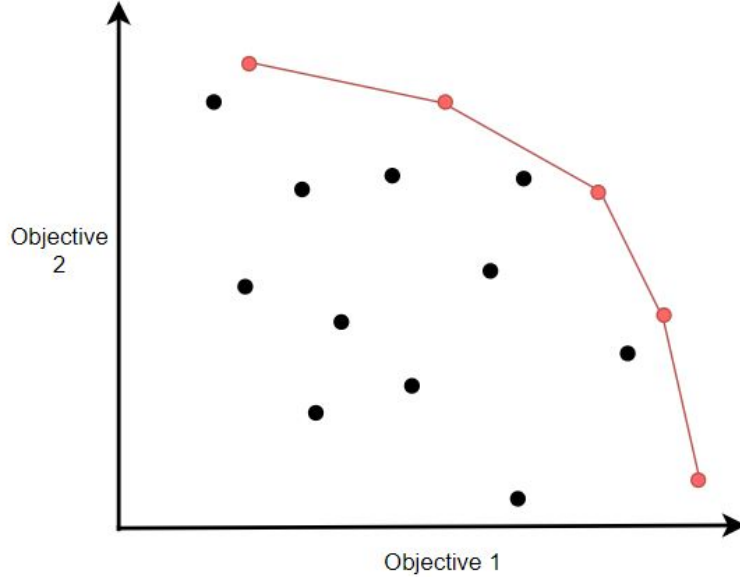


Figure 5.16: Example of a Pareto front made up of optimal solutions (red dots). Each solution cannot be improved in one objective without degrading the other objective.

Before converging to a final optimal solution set, the model uses a complex selection process for advancing generations which is based on a ranking and crowding distance method. This tries to optimise which solutions are the best to be used as parents of the next generation based on the solutions' level of dominance over others. For each generation, the algorithm will group the solutions into varying non dominated fronts, each one less dominating than the previous. Then, the algorithm chooses which solutions will be used for the offspring of the next generation by balancing diversity (crowding) in the solution set and promoting solutions in high fronts (ranking). This means that even if a solution is not in the highest front, it may still get promoted to keep diversity and allow for a wider design space search.

5.9.2. Implementation

The NSGA-II algorithm is implemented using the pymoo library [116]. Designed for the Python programming language, this library also supports mixed variable problems, making it a perfect candidate to be used as the optimisation solver. The code follows the pymoo documentation, and in simple terms works as depicted in Figure 5.17. The variables and constraints are fed into the analytical model, giving a solution with a certain power-to-mass ratio and stowed power density. Once the population size,

which refers to the number of solutions in a population, is reached, the optimisation algorithm sorts and ranks the solutions. If the termination constraint is reached, the model returns the optimal solutions. Otherwise, the algorithm continues to look for better solutions by altering a few crucial components of the evolutionary algorithm.

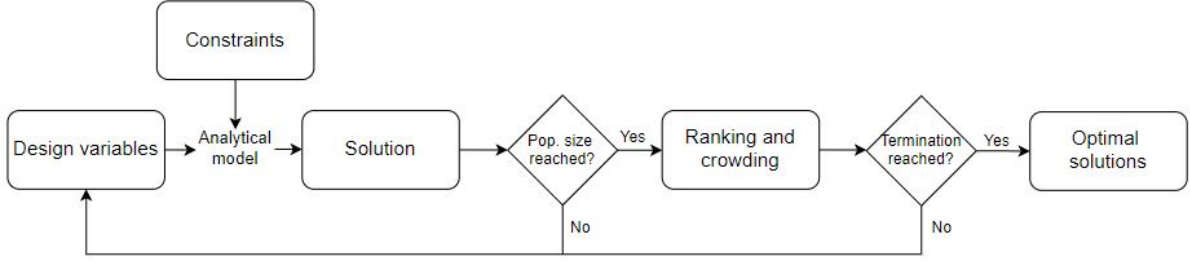


Figure 5.17: Overview of the how the optimisation algorithm works in conjunction with the analytical model.

There are a few main components of the algorithm which can be tuned to drive the search process to the set of absolute optimal solutions, with the first one being the population size. A higher population size is beneficial as it impacts the genetic diversity in the solutions, as the more solutions there are, the more variability in the solution space. On the other hand, increasing the population size directly impacts the computational cost of the algorithm. A right balance is beneficial to limit the computation time, while ensuring enough genetic diversity in each population such that the optimal solutions can be found. Another tunable component in the algorithm is the mutation rate, which determines the chance that a solution will undergo mutation, or a change in one of its variables. The mutation rate is important as it ensures variability in the population and allows for higher genetic diversity. Again, a balance is important, as setting a very high mutation introduces too much randomness in the solutions, while setting a low mutation rate could lead to a premature set of optimal solutions due to some areas of the solution space being unexplored. A final component of the algorithm which can be changed is the crossover, which determines what traits from parent solutions get passed to a new solution of the next generation, known as the offspring. This is vital in allowing the offspring to get good characteristics of the parents such that a potentially even better solution is found. For the optimisation model in this study, the population size and number of generations will be tweaked, while the crossover and mutation rate are kept at their default values.

Another related component of the algorithm is the aforementioned termination constraint, which determines when the algorithm stops running. There are a few ways in which this is done. The first and simplest method is done by defining a maximum number of generations. Another is by selecting a convergence criteria, which means that the algorithm will keep running until further generations do not produce better solutions. Lastly, a maximum run time may be set to limit the computational resources of the algorithm. For the optimisation, a termination constraint will be used, and a convergence analysis will be made to see at what point the absolute optimal designs are achieved.

6

Verification & Validation

This chapter focuses on verifying the methods in the analytical model that are shown to be the most constraining. This ensures that the constraints driving the designs are correct, which ultimately enhances the model's credibility. The frequency constraint is verified in section 6.1, followed by the verification of the torsion in section 6.2. As open stiffeners were not able to be verified, a deeper analysis is made in section 6.3. Then, the coiling and flattening stress constraints are verified in section 6.4. Finally, the analytical model is validated in section 6.5 against the ROSA and TASSA roll-out solar arrays.

6.1. Frequency analysis verification

The following section evaluates the verification of the frequency analysis. During development of the analytical model, it became apparent that the natural frequency was one of the limiting constraints that was driving the designs. To ensure that this limiting constraint is correct, the calculations and formulas used in the analytical model shall be verified using the Ansys FEM software. As the formulas used in the analytical model are simplifications, it is assumed that the results from Ansys are the correct ones, meaning that a disagreement between the analytical model and Ansys would assume the former to be incorrect. For each case, the problem was modelled in the modal environment of the program. The section starts by describing the verification of closed stiffeners in subsection 6.1.1, followed by the verification of open stiffeners in subsection 6.1.2. Then, the two external and two inherent stiffener design verification's are also described in subsection 6.1.3 and subsection 6.1.4, respectively.

6.1.1. Closed stiffeners

The verification of the closed stiffeners consisted of the interlocking and lenticular cross sections. The analyses for the frequency verification were set up such that they resembled the problem in the analytical model. One end would have a fixed constraint, while the other end would be free. A few cases were made for each type of stiffener such that no conclusions would be made on single data points. While

some design variables such as the material and length were kept constant for all analyses, other variables like the radius, thickness, and tip mass were different for each test case. A detailed description of each test case with their respective design variables and results can be found in Table 6.1.

Table 6.1: Frequency analysis of closed external stiffeners

Stiffener	Radius (mm)	Thickness (mm)	Tip mass (kg)	Frequency (Hz)		Difference (%)
				Analytical	FEM	
Interlocking	30	0.5	0	2.46	2.38	-3.3
	30	0.3	0.5	1.75	1.72	-1.7
	60	0.3	0.5	4.01	3.96	-1.6
Lenticular	30	0.5	0	1.73	2.07	19.7
	30	0.3	0.5	1.42	1.59	12.0
	60	0.3	0.5	3.11	3.59	15.4

From the results, it is apparent that the natural frequency of the interlocking structure is being predicted very well by the analytical model, with the largest difference being -3.25% . The lenticular stiffener, while still in the acceptable range, shows higher differences, with a maximum difference of 19.65% . Despite the larger difference, the analytical model consistently under-predicts the frequency, which is better than over-predicting the frequency, and therefore the analytical model effectively is oversizing the stiffener. Also, the lenticular structure is much more complex than the round interlocking structure, which may explain the larger discrepancies. Additionally, the mode shapes predicted by the analytical model matched those of Ansys. An example of the first mode of the lenticular cross section can be seen in Figure 6.1, which also conveys how the model was set up. It is therefore assumed that the closed stiffeners are verified and can be used in the optimisation model.

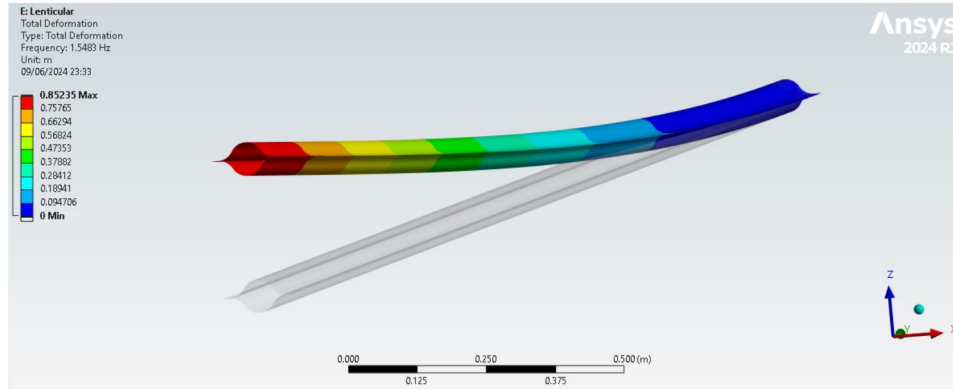


Figure 6.1: First mode of the lenticular cross section.

6.1.2. Open stiffeners

The open stiffeners consisted of the TRAC and STEM type structures. These were modelled in the same way as the closed stiffeners: a cantilever type setup with one end fixed and the other end free. Due to time limitations, each test case was made with the same material, and the opening angle for

the STEM and TRAC were kept at a constant 150° and 90° , respectively. Other variables like length, radius, thickness, and tip mass were used as design variables. Table 6.2 shows the results of the open stiffener verification.

Table 6.2: Frequency analysis of open structures

Stiffener	Length (mm)	Radius (mm)	Thickness (mm)	Tip mass (kg)	Frequency (Hz)		Difference (%)
					Analytical	FEM	
STEM	2500	30	1	0	7.94	5.38	-32.2
	2500	30	1	0.5	6.33	4.30	-32.1
	2500	30	0.3	0	2.57	4.24	65.0
	2500	30	0.3	0.5	2.57	2.55	-0.8
	5000	30	0.3	0	1.29	1.16	-10.1
	5000	30	0.3	0.5	1.29	0.84	-34.9
	2500	100	1	0	2.50	13.4	436.0
	5000	100	1	0	1.29	3.44	166.7
	5000	30	0.3	0	0.82	1.09	32.9
	5000	30	0.3	0.5	0.68	0.75	10.3
TRAC	2500	30	0.3	0	1.65	4.15	151.5
	2500	30	0.3	0.5	1.65	1.27	-23.0
	5000	30	1	0	0.86	0.92	7.0
	5000	30	1	0.5	0.81	0.74	-8.6
	2500	30	1	0	3.45	3.68	6.7
	2500	30	1	0.5	3.04	2.57	-15.5

Immediately, it is evident that a lot more test cases were performed for open stiffeners than for closed stiffeners. This is because preliminary results from Ansys of open stiffeners were not matching the analytical calculations, and more data was necessary to confirm if the discrepancies were consistent or not. In the end, it is apparent that the calculations by the analytical model were not able to be verified by Ansys. While some test cases gave promising results with differences under 10%, other test cases showed significantly higher discrepancies, with over three test cases having differences exceeding 100%, the largest of which was a 436% difference.

From Figure 6.2 and Figure 6.3, it is evident that the eigenmodes for these types of structures involve a combination of bending and torsion. On the other hand, the analytical model is limited in the sense that it only calculates the bending frequencies and torsional frequency through simplified formulas. This coupling of mode shapes makes it impossible for the analytical model to correctly predict the natural frequency of these shapes, and explains the high volatility in the difference between the frequency predicted by the analytical model and Ansys.

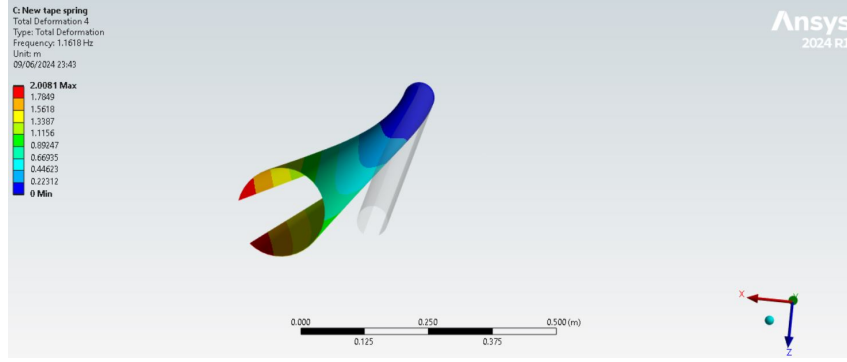


Figure 6.2: Eigenmode of the STEM cross section.

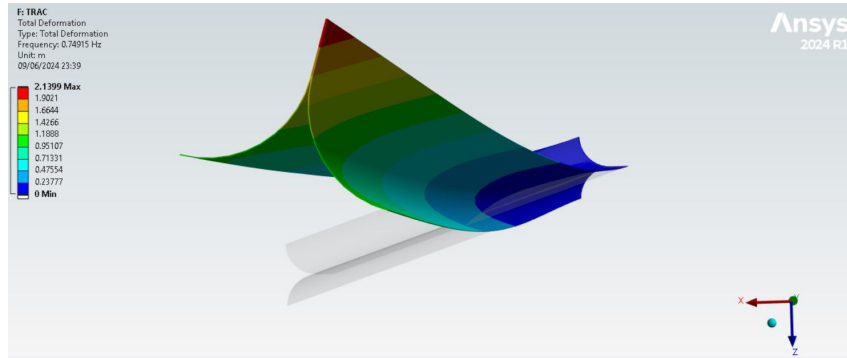


Figure 6.3: Eigenmode of the TRAC cross section.

The results also indicate that changes in the design variables, such as length, thickness, radius, and the presence of tip mass, significantly affect the accuracy of the analytical model. This high volatility in results is unacceptable, and thus, the analytical model cannot be used to predict the natural frequency of open structures. In order to calculate the frequencies more accurately and capture the true mode shapes, a significantly more complex method would be required, which for this study is considered to be out of the scope. Nevertheless, a solution to this unresolved problem is explained in detail in section 6.3.

6.1.3. Double external stiffener

This verification analysed the modal response in the case when two stiffeners are used in the structure. The analysis was modelled by having two stiffeners side by side, and joined at the tip with a massless beam with infinite stiffness. All other boundary conditions and analysis setup parameters remained the same. For this analysis, a model with two interlocking stiffeners and two STEM stiffeners are used, such that both closed and open stiffeners would be verified, as seen in Figure 6.4. Furthermore, multiple analyses were made for the same stiffener type to verify the effect of various design characteristics, such as the length, radius, and tip mass. All stiffeners had a thickness of 0.5 mm and were modelled using an isotropic material, and the STEM stiffeners had a subtended angle of 120° . Other types of stiffeners were not analysed due to time limitations of this study. The results of the analyses are tabulated in Table 6.3.

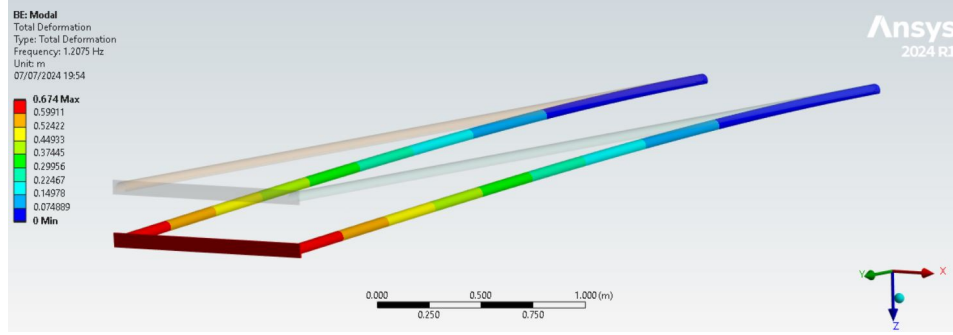


Figure 6.4: Eigenmode of the double external STEM stiffeners.

Table 6.3: Frequency analysis of double external stiffener configuration

Stiffener	Length (mm)	Radius (mm)	Tip mass (kg)	Frequency (Hz)		Difference (%)
				Analytical	FEM	
Interlocking	1000	10	0	21.0	20.5	-2.4
Interlocking	1000	10	1	6.8	6.7	-1.5
STEM	1000	30	0	33.7	40.5	20.2
STEM	1000	30	1	17.2	17.7	2.9
STEM	5000	30	0	1.35	1.62	20.0
STEM	5000	30	1	1.08	1.2	11.1

Similar to the single stiffener analyses, the interlocking stiffener, or closed structure, shows results of the FEM which match very closely that of the analytical model. Even with the addition of tip mass, the analytical model has a maximum error of -2.4% . The open structure, or STEM stiffener, has results which have a much greater discrepancy with respect to the closed structure, with a maximum recorded error of 20.2% . Nevertheless, these results are significantly better than the single stiffeners, and given that the FEM estimates natural frequencies higher than the analytical model consistently, it is safe to assume that a design with two stiffeners which lies on the boundary of the frequency constraint will be structurally stiff enough.

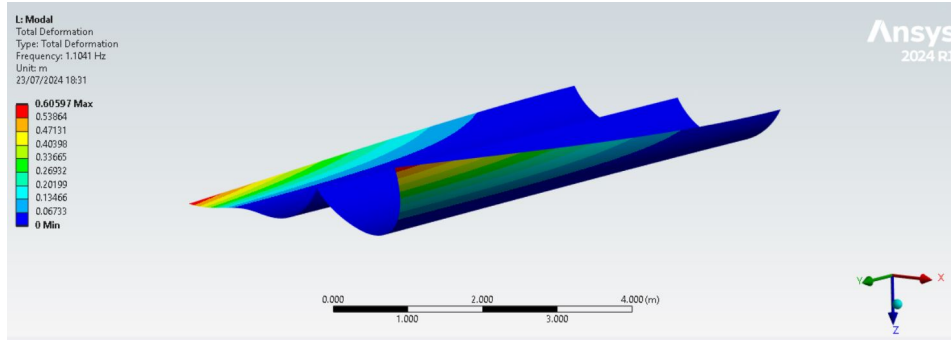
6.1.4. Double inherent stiffener

This type of design follows a similar one to the TASSA, which in a simplified model is two large tape springs connected to each other. The constraints are the same as in previous analyses, with the only additional constraint being that the two tape springs are connected longitudinally. For this analyses, the design variables that are changed include the length, the opening angle, radius, and tip mass. The results are tabulated Table 6.4.

Table 6.4: Frequency analysis of double inherent stiffener configuration

Length (mm)	Angle (deg)	Thickness (mm)	Tip mass (kg)	Frequency (Hz)		Difference (%)
				Analytical	FEM	
5000	30	0.5	0	1.88	1.69	-10.1
5000	30	0.3	0.5	1.85	1.31	-29.2
8000	60	0.5	0	3.91	1.11	-71.6
8000	60	0.3	0.5	3.87	0.85	-78.0

The results show that this configuration cannot be verified due to the significantly high differences between the analytical model and Ansys. While one design achieves an accuracy of -10% , the other three designs show that the analytical model is not able to capture other more complex phenomena, with the largest difference being over -78% . Since Ansys also shows that the real eigenfrequency is always lower than what the analytical model is predicting, this makes the method even less robust since it underestimating the true value. The differences can be attributed to the fact that the eigenmode of the stiffener is not characterised by a bending or torsion, and is rather a different complex mode that is specific to that shape. This can be seen clearly in Figure 6.5, where the eigenmode is specific to that shape.

**Figure 6.5:** Eigenmode of a double inherent stiffener design.

6.2. Torsion analysis verification

The following section evaluates the torsional analysis verification, another constraint which was found to be limiting the designs. Similar to the frequency verification, Ansys is used to verify the analytical model, and is similarly used as the true basis. Furthermore, all of the analyses are made in the static structural environment of the program. The section begins by describing the verification of closed stiffeners in subsection 6.2.1, followed by the verification of open stiffeners in subsection 6.2.2. The double external and inherent configurations are evaluated in subsection 6.2.3 and subsection 6.2.4, respectively.

6.2.1. Closed stiffeners

The torsional analyses were configured to closely resemble the conditions in the analytical model, and included the interlocking and lenticular cross sections. As in the previous section, one end of the stiffener was fixed, while the other end was left free. Multiple test cases were conducted for each type of stiffener, with certain design variables such as material and length kept constant, while others like radius, thickness, and moment varied. A comprehensive description of each test case, along with their respective results, can be found in Table 6.5.

Table 6.5: Torsional analysis of closed external stiffeners

Stiffener	Radius (mm)	Thickness (mm)	Moment (Nm)	Stress (MPa)		Difference (%)
				Analytical	FEM	
Interlocking	30	0.5	10	3.54	3.76	6.2
	30	0.3	10	5.89	6.31	7.1
	60	0.3	20	2.95	3.07	4.1
Lenticular	30	0.5	10	3.88	3.33	-14.2
	30	0.3	20	6.46	5.48	-15.2
	60	0.3	20	3.23	2.82	-12.7

From the results, it is evident that the interlocking stiffener exhibits a good correlation between the analytical calculations and Ansys, with the largest difference being 7.1%. This indicates that the analytical model is accurate and reliable in predicting the stress for interlocking stiffeners. Similarly, while the lenticular stiffener shows a larger discrepancy, with the maximum difference being -15.2% , this is still within an acceptable range. The consistent underestimation by the analytical model for the lenticular stiffener also suggests a conservative approach, which can be beneficial for ensuring structural safety. Therefore, the calculations of the analytical model when run with the interlocking and lenticular stiffeners can be considered verified and suitable for use in the optimization model.

6.2.2. Open stiffeners

The torsional analysis of open structures tested the STEM and TRAC stiffeners. For simplifying the design space, some characteristics like the opening angle, material, and length were kept constant. The radius, thickness, and moment were varied across the analyses. The setup also was the same as for the closed structures, and modelled again in Ansys. Table 6.6 gives the results of the tests.

Table 6.6: Torsion analysis of open structures

Stiffener	Radius (mm)	Thickness (mm)	Moment (Nm)	Stress (MPa)		Difference (%)
				Analytical	FEM	
STEM	30	0.3	0.01	2.10	0.31	-85.2
	30	1	0.1	1.90	1.94	2.10
	50	1	0.1	1.15	0.66	-43.0
	50	2	0.5	1.43	1.83	28.0
	50	0.3	0.1	12.70	0.94	-92.6
	50	0.3	0.5	63.70	4.70	-92.6
TRAC	30	1	0.1	2.70	3.27	21.1
	30	0.3	0.01	3.00	2.80	-6.70
	30	0.3	0.02	6.00	5.62	-6.30
	50	1	0.5	8.10	9.73	20.1
	50	0.3	0.01	1.80	2.87	59.4

Similar to the frequency analysis verification, the open structures had a very high variability between the analytical calculations and results from Ansys. With the highest variability being -92.6% and 59.4% , it shows that the analytical model fails to capture the true behaviour of open structures under torsional loads. In some cases, the structures warped, causing out of plane stress concentrations that were not being captured by the analytical model. A solution would be to include warping considerations in the analytical model. However, this would incur a lot of time and effort, which for such a small part of the study would not be beneficial. Additionally, it is probable that the simplified formula is not robust enough to capture the full effects on these structures, meaning that open structures cannot be verified for the torsional tests.

6.2.3. Double external stiffeners

The following analysis verified the stresses due to torsion for the double external stiffener case. The analyses were setup similar to the one described in subsection 6.1.3, except that the connecting bar is not added such that the stiffeners can stretch as they are being deformed. Consequently, one side of the stiffeners is fixed, while at the other end a torsional moment is applied such that it acts on the longitudinal axis that runs along the middle of the two stiffeners. The resulting deformation and stress contours due to an acting torque can be seen in Figure 6.6. For a counterclockwise acting moment, the right stiffener displaces upwards, causing compression at the top and tension at the bottom of the stiffener. The opposite effect happens on the left stiffener, which displaces downwards, causing tension at the top and compression on the bottom. Analyses were conducted for the interlocking stiffener which represented closed structures, and STEM stiffener as the open structure representative stiffener. To simplify the study, only a few parameters were tweaked, and other characteristics such as the material and opening angle of the STEM were kept constant throughout the analyses. Table 6.7 tabulates the results from the analyses conducted in Ansys.

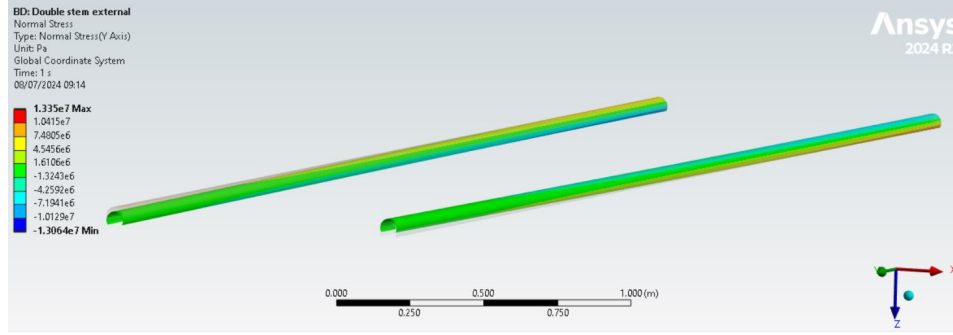


Figure 6.6: Stress contours of the double external stiffener configuration in response to torsional moment.

Table 6.7: Torsion analysis of double external stiffener configuration

Stiffener	Moment (Nm)	Length (mm)	Radius (mm)	Stress (MPa)		Difference (%)
				Analytical	FEM	
Interlock	1	1000	10	31.0	39.9	28.7
Interlock	1	5000	30	3.5	4.5	29.6
Interlock	10	5000	30	34.5	44.7	29.5
STEM	1	5000	30	23.2	26.5	14.2
STEM	1	5000	30	39.0	44.4	13.8
STEM	1	5000	60	5.7	7.2	25.7
STEM	0.5	5000	60	4.9	5.9	20.2
STEM	5	3000	60	29.5	38.1	29.2

Overall, the analyses reveal that the stresses obtained from Ansys are consistently higher than those predicted by the analytical model, with the maximum discrepancy being 29.5%. While this difference is not ideal, it is acceptable at the conceptual stage that the design is in. Nevertheless, the analytical model can be improved with this data to enable a more accurate optimization. Consequently, a factor of safety of 30% is added to the analytical model to compensate for this difference and ensure that the final stresses are not underestimated.

6.2.4. Double inherent stiffeners

The double inherent stiffener configuration is modelled in the same way as in subsection 6.1.4. The two tape springs are constrained longitudinally on the sides, and the rest acts as a cantilever model. The design variables that vary are the length, angle, thickness, and torsional moment. The results are tabulated in Table 6.8.

Table 6.8: Torsional analysis of double inherent stiffener configuration

Length (mm)	Angle (deg)	Thickness (mm)	Moment (Nm)	Stress (MPa)		Difference (%)
				Analytical	FEM	
5000	30	0.5	1	11.5	0.58	-95.0
5000	30	0.3	2	63.7	2.57	-96.0
8000	60	0.5	1	11.4	0.15	-98.7
8000	60	0.3	2	63.7	0.67	-99.0

From the results, it is apparent that the analytical model significantly overestimates the stress values for double inherent stiffener designs compared to Ansys, with differences ranging from -95.0% to -99.0% . This high variability indicates that the analytical model fails to accurately capture the behavior of double inherent stiffeners under torsional loads, and the invariability between designs suggest that the simplified analytical approach is not robust enough to predict the stress in these configurations. This could be due to complexities in the geometry and loading conditions that are not accounted for in the analytical model. Therefore, the analytical model cannot be considered verified for this type of configuration, and a different method shall be considered to allow this configuration to be included.

6.3. Review of open stiffeners

In section 6.1 and section 6.2, it was shown that the open stiffener calculations of the analytical model were not able to be verified as the results from Ansys differed significantly with the calculations from the analytical model. With these structures not verified, they cannot be used in the optimisation model, and a solution is needed. The first proposed solution which would allow the open structures to still be part of the optimisation model was to calculate the complete frequency analysis for the open structures instead of using the simplifications. While this method would ensure accurate and reliable results, it would be a very time consuming approach. As the direct solutions to the analytical model posed too complex methods, it was decided that the open structures could not be used in the optimisation model, and another method would be required.

Another method to still be able to incorporate them as a design option would be to do a type of reduced manual optimisation. This would involve doing a set of iterations in Ansys to find an optimal result by testing out designs with varying variables. This method however would also be very time consuming, and not that accurate, considering that there are a significant amount of design variables that can be optimised, and thus only a limited design space would be tested. As a results, it was decided to omit this method as well.

In the end, a significantly less time consuming and complex method was thought of and applied. This method involves comparing the structural performances of the open and closed stiffeners in the case that they would be used in the same roll-out solar array. More specifically, the two stiffeners are compared,

assuming that the solar array has the same power-to-mass ratio, effectively ensuring a fair evaluation. This is achieved by having some design variables constant for all stiffeners, and then computing the rest of the design variables such that the equal comparison is possible. The detailed explanations and results, along with the discussions, are explained in the following sections.

6.3.1. External stiffeners

Firstly, the external stiffeners are investigated and compared. To ensure that power-to-mass ratio would remain equal between designs, some parameters would have to be predetermined. These variables included the length, flat width, and mass of the stiffener. This ensured that the stowed area of the stiffener would be the same in each case, allowing equal power generation for the solar array. The opening angle of the stiffeners was chosen based on commonly used angles. Then, the thickness and radius were calculated based on the previously chosen design parameters. The thickness was determined by the mass of the stiffener, and the radius was computed through the flat width and opening angle. With all the physical parameters known, the stiffeners could be modelled and analysed in Ansys. The mass of the stiffeners were also verified to be correct through Ansys, ensuring that the calculated parameters were indeed correct. Results of the analyses are tabulated in Table 6.9.

Table 6.9: Comparison of open and closed external stiffeners of same solar array power-to-mass ratio.

Parameter	STEM	TRAC	Interlocking	Type
Length (m)	5	5	5	Chosen
Flat width (mm)	200	200	200	Chosen
Mass (kg)	1	1	1	Chosen
Angle (deg)	120	90	180	Chosen
Thickness (mm)	0.99	0.50	0.98	Calculated
Radius (mm)	47.7	93.0	32.0	Calculated
Frequency (Hz)	5.2	3.87	7.27	Ansys
Stress at 1N (MPa)	32	17.5	0.16	Ansys

The results show that for the same type of solar array, a closed stiffener will have better structural characteristics compared to a design employing an open stiffener. While the natural frequency does not differ much between stiffeners, the stresses from the open stiffeners due to a torsional moment are two orders of magnitude higher than the closed stiffener. These results are not surprising considering that thin walled open stiffeners lack significant torsional rigidity compared to closed stiffeners. Their much lower polar moment of inertia, along with unequal distribution shear stress distribution causes them to have much higher stress concentrations and make them much more prone to warping.

6.3.2. Inherent stiffeners

The analysis of the inherent stiffeners is made in a similar way as the external stiffener in the previous subsection. First, the length, flat width, and mass are chosen to be constant for each stiffener. Then, instead of choosing an angle at random, this would have to be calculated. This is because the addition of the curvature of certain structures made it so that the angle would have to be determined in order for all the solar arrays to have the same power output. Once determined, the thickness and radius could be computed using the previous design variables. The analyses were made in Ansys, and results are shown in Table 6.10.

Table 6.10: Comparison of open and closed inherent stiffeners of same solar array power-to-mass ratio.

Parameter	Tape spring	Double Tape spring	Flat spring	Lenticular	Type
Length (m)	5	5	5	5	Chosen
Flat width (mm)	960	960	960	960	Chosen
Mass (kg)	2	2	2	2	Chosen
Angle (deg)	25	25	180	30	Calculated
Thickness (mm)	0.42	0.42	0.40	0.21	Calculated
Radius (mm)	1100	550	9.6	328	Calculated
Frequency (Hz)	2.90	4.83	0.83	8.35	Ansys
Stress at 1N (MPa)	3.38	1.79	18.37	0.04	Ansys

Similar to the external stiffener investigation, inherent open stiffeners also exhibit much lower structural characteristics compared to a closed stiffener. Furthermore, the flat spring performs the worst among the three open stiffeners, with both its natural frequency and stress levels being an order of magnitude worse than those of the tape spring. While two tape springs beside each other have increased natural frequency compared to the single tape spring, they still are less stiff than the lenticular structure. When the stress is taken into account, it is evident that the lenticular stiffener is the better performing structure, exhibiting stresses of two order of magnitude lower than the tape spring designs.

6.3.3. Discussion and review of the analytical model

From the investigation in this section, it is apparent that closed stiffeners are the optimal structures for the roll-out solar array application. This conclusion was reached by comparing the structural performances of open and closed stiffeners on a solar array, ensuring that the power-to-mass ratio of the solar array was constant regardless of the stiffener used. Not only do closed stiffeners exhibit higher natural frequency properties, the torsional rigidity of these structures are significantly higher than open structures. While there might be a combination of physical design variables which cause the open stiffeners to have a better performance, this method follows an unbiased example, having chosen initial parameters randomly. The following study therefore proceeds with the assumption that for any open

stiffener, a better performing closed stiffener design exists. As a result, the final optimisation model does not include open stiffeners as design options for single external stiffening cases and all external stiffening cases. Nevertheless, open stiffeners will still be analysed in the case that two or more external stiffeners are used, since this design variation has been verified.

The inability to use open stiffeners does pose the limitation of restricting the design space. This means that the optimisation on the full spectrum is not truly possible, and therefore the final optimal design cannot be guaranteed to be true. Furthermore, the TASSA, a flight proven roll-out solar array, employs the double inherent open stiffener design, which questions the decision to remove open structures as design options. On the other hand, their decision for that particular design might have involved many other trade off variables besides the power-to-mass ratio. For example, although a lenticular closed structure may offer superior structural properties for the same power-to-mass ratio, the added difficulty of producing a lenticular stiffener due to bonding two CFRP parts might be significant enough to rule it out as an option. Additionally, their supporting structure and method of deployment might have also driven the decision to use open structures. It is important to realise that this complex system may be influenced by more than just individual performance parameters, and in practical terms, some suboptimal designs may be more appropriate.

6.4. Coiling & flattening stress verification

To further verify the calculations in the analytical model, an analysis is made on the coiling and flattening stress of the stiffeners. The stresses that arise from this loading were identified to be a critical constraint, and therefore verifying their accuracy is crucial. Both tests were carried out with Ansys and modelled in the static structural environment of the program.

The first test case pertains to the coiling stress of the stiffener that occurs when it is wrapped around the mandrel for stowage. The analysis is made by having a thin plate that would resemble a flattened stiffener be placed between a circular body and a fixed body with a circular cavity, as seen in the top image of Figure 6.7. The model only coils a small part of the plate to save computation time, and since the coiling stress remains equal along the whole radius of the coiling mandrel, analysing one part ensures that the stress along the entire plate is captured and modelled. In the analysis, the top body displaces downwards such that the plate takes the shape of the circular object as it is constrained between the two rigid bodies. The final state of the analysis can be seen in the middle image of Figure 6.7, wherein the plate is assumed to be coiled around the circular object. Two models of different thicknesses (0.3, 0.5 mm) were analysed for redundancy. Table 6.11 compares the results from the analytical model with those of the FEM.

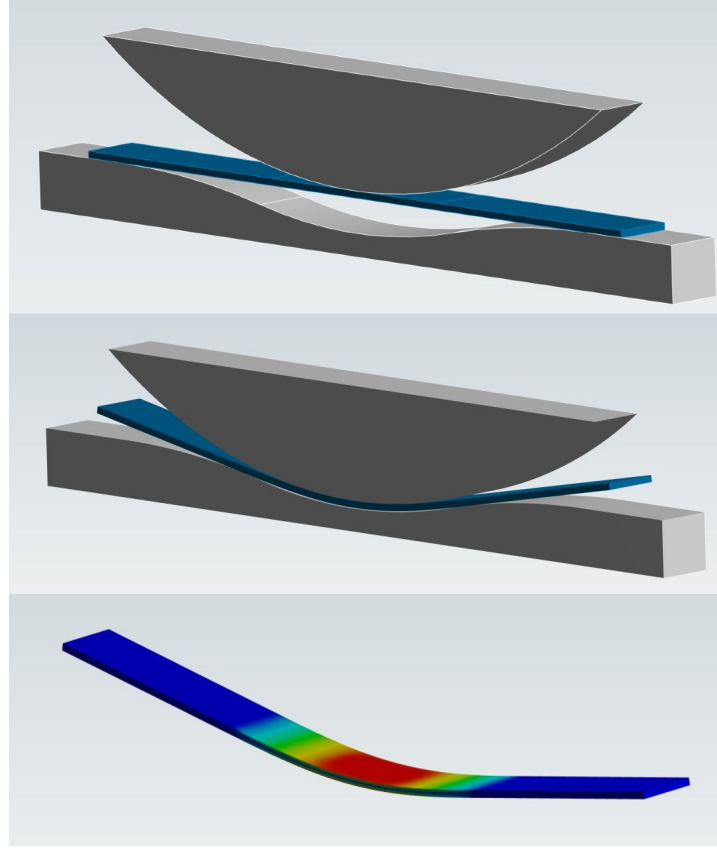


Figure 6.7: Undeformed model of the coiling verification analysis (top), coiled model (middle), and its stress contours (bottom).

Table 6.11: Analytical and FEM results for the coiling stress analyses.

Thickness (mm)	Stress (MPa)		Difference (%)
	Analytical	FEM	
0.5	10.00	10.66	6.6
0.3	6.00	6.32	5.3

To analyse the flattening stress, a model is set up to simulate a lenticular stiffener cross section transitioning from a deployed state to a flattened state. Similar to the coiling stress analysis, the problem is reduced to save computational time. This is done by reducing the length of the stiffener and analysing only one half of the lenticular cross section. Nevertheless, the following analysis is still valid since the cross sectional shape is independent of the length, and therefore captures the stresses that would be present along the entire length of the stiffener. Furthermore, the symmetry of the stiffener ensures that analysing only one half of the cross section is sufficient to capture the whole system. The simulation is set up as seen in Figure 6.8, wherein the top rectangular block displaces downwards to flatten the stiffener. The model is constrained such that it cannot move out of plane, and the flanges at the end of the boom are constrained such that they do not separate with the bottom body during the flattening process. Figure 6.8 also shows the model after it has transitioned to a flattened state due to the dis-

placement of the top rectangular body. Again, two models of different thicknesses were analysed, and Table 6.12 shows the comparison between the results of the analytical model and those of the FEM.

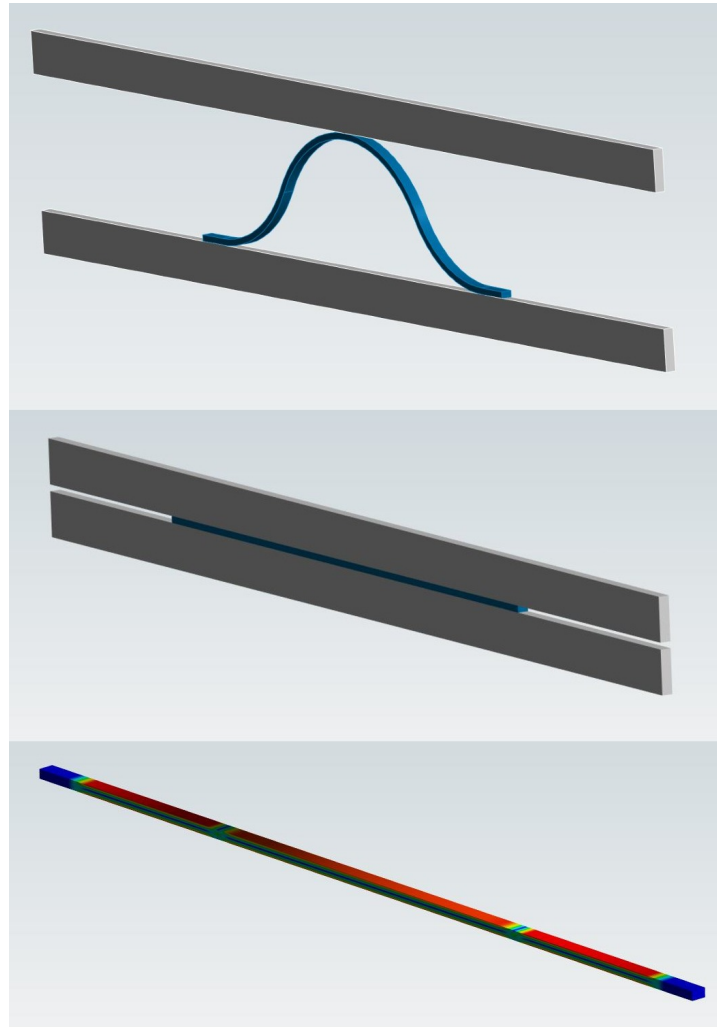


Figure 6.8: Undeformed model of the flattening verification analysis (top), flattened model (middle), and its stress contours (bottom).

Table 6.12: Analytical and FEM results for the flattening stress analyses.

Thickness (mm)	Stress (MPa)		Difference (%)
	Analytical	FEM	
0.5	45.48	50.11	10.2
0.3	26.32	26.25	-0.3

Overall, for both the coiling and flattening processes, the analytical model matches the results of the FEM, with a maximum difference of 6.6% for the coiling stress, and a maximum difference of 10.2% for the flattening stress analysis. Therefore, the method used in the analytical model is verified and confirmed to be accurate.

6.5. Analytical model validation

Finally, a holistic validation of the analytical model is made by comparing the results of the model against existing roll-out solar arrays. This is made by changing the inputs of the analytical model such that they represent as closely as possible the real roll-out solar array. The roll-out solar arrays that will be used for validation are the TASSA, and iROSA, each described in detail in section 2.2. As they both are different to each other in terms of design and size, verifying the model against these two roll-out solar arrays will give a good understanding of the level of accuracy of the model. Note that the COBRA and SolarFlex are not used to validate the model as there is too little information available to the public on these designs.

6.5.1. TASSA

The first roll-out solar array which will be used to verify the model is Momentus' TASSA. Before starting the validation, some assumptions and estimations are required in order to have all the inputs necessary to run the analytical model. Firstly, the TASSA can visually analysed to get some required inputs. From Figure 2.6, it is evident that the solar array is an inherent stiffening, normal deployment roll-out solar array. It is also evident that it uses a double tape spring design, with an assumed curvature of 30 deg. As the TASSA is retractable, a motor is chosen as the deployment mechanism. It is also known that the TASSA uses Ascent Solar Technologies' Titan module, a solar cell based off of the CIGS thin film photovoltaic technology [117]. Their solar module is one square foot in size, weighs 10 *g*, and comes with an efficiency of 17.55%. Assuming that the module is twice as long as it is wide, as seen in Figure 2.6, the size of the module is estimated to be 0.424 *m* by 0.212 *m*. It is also said to produce 17 *W* per module [117]. From the picture of TASSA, it can be seen that 6 modules span widthwise, while 27 modules span along the length of the array, meaning that a total of 162 modules would be used, totaling a power output of 2754 *W*. Due to an estimated subtended angle of 20 *deg*, approximately 9% of the power is lost, resulting in a final power output of 2507 *W*. Momentus' Chief Executive Officer has stated that the TASSA is "intended to allow Smallsats to generate kilowatts of power", reaffirming that the calculated power of 2.5 *kW* is a realistic value [118]. The final assumption relates to the mandrel radius, which is sized to the minimum bending radius of the solar cell, resulting in a mandrel radius of 5 *cm*.

The only unknown is the type of material which is used in the stiffening structure, and so the analytical model will be run with all the materials available in order to have a general idea of what the range of the performance parameters would be. In terms of comparing to the real values, the power-to-mass ratio of the TASSA is unknown and cannot be computed since the real mass is not publicly known. On the other hand, the stowed power ratio can be estimated. From the patented design paper, the size of each side of the structure can be estimated if one side is known by taking the relative sizes of each side to the known side. This assumes that the encapsulating volume of the array is a cuboid, which may not

be completely accurate. Nevertheless, the volume of the TASSA using this estimation method turns out to be 0.0911 m^3 , resulting in a stowed power density of 27.5 kW/m^3 . This is a low stowed power density for roll-out solar arrays, but is justified by the low efficiency solar arrays and the additional volume that is taken by the structure surrounding the mandrel as seen in Figure 2.6. Running the analytical model with the aforementioned inputs and all the available materials, a dataset of possible TASSA designs is obtained and graphed in Figure 6.9. Materials which don't abide to the flattening and coiling constraints are omitted. It is also important to note that none of the designs adhered to the frequency and torsional constraints set by the analytical model. Since the method for calculating these constraints was not able to be verified in section 6.1 and section 6.2, they are omitted as part of the constraint checking process.

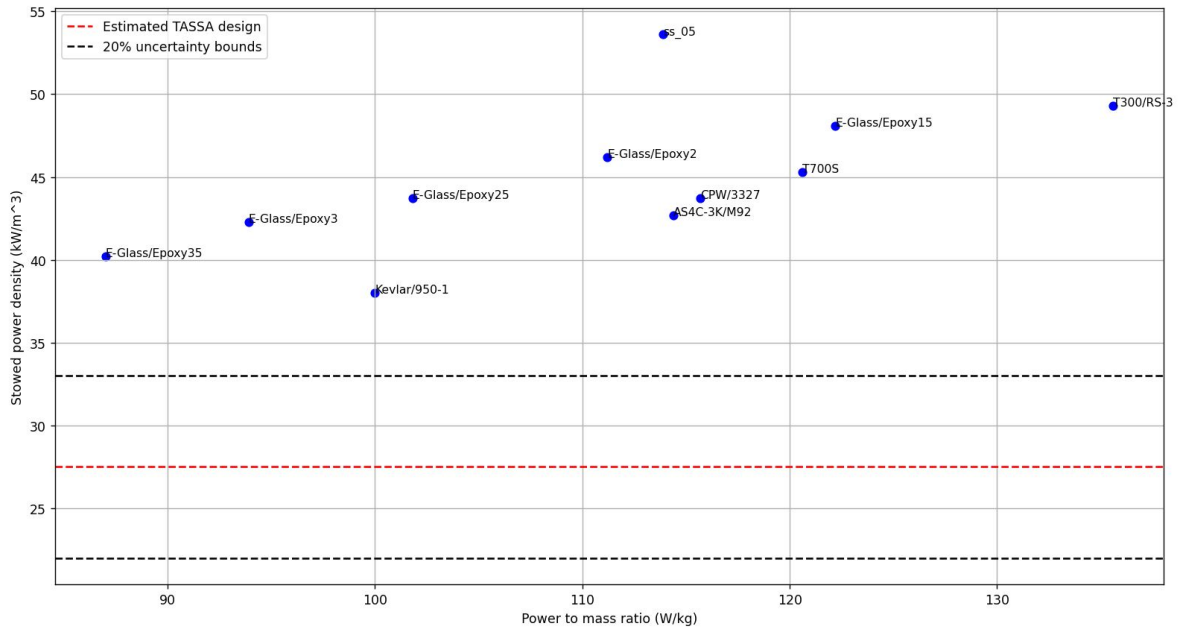


Figure 6.9: Performance of different TASSA designs using different stiffener materials. The red horizontal line indicates the estimated performance range of the real TASSA design, and black lines indicate the uncertainty bounds.

Overall, the designs of different materials have noticeable variety on the power-to-mass ratio and stowed power density, proving that the material has a substantial effect on the performance parameters. The estimated power-to-mass ratio ranges from 85 W/kg to 137 W/kg , aligning well with what is expected for a roll-out solar array utilising a low efficiency cell technology. While the range is still higher than that of traditional solar arrays, it is expected that it would not exceed the 140 W/kg mark due to the cell efficiency limitation. On the other hand, the stowed power density is evidently overestimated significantly, with all designs being outside of the uncertainty bounds. The reason for this is due to the significant additional volume that the design of the TASSA takes compared to other roll-out solar arrays. The model bases the additional volume on a scaling factor, explained in detail in subsection 5.4.1. For a roll-out solar array with a normal deployment mode like the TASSA, the factor ζ is 2.33, when in

reality it is roughly 4. This is visualised in Figure 6.10, and shows how much volume is not being considered by the analytical model. This is a significant difference, and hence the initial calculations underestimate the additional volume required for the TASSA. Another set of designs of the TASSA are made with the ζ equal to 4, and the results are plotted in Figure 6.11.

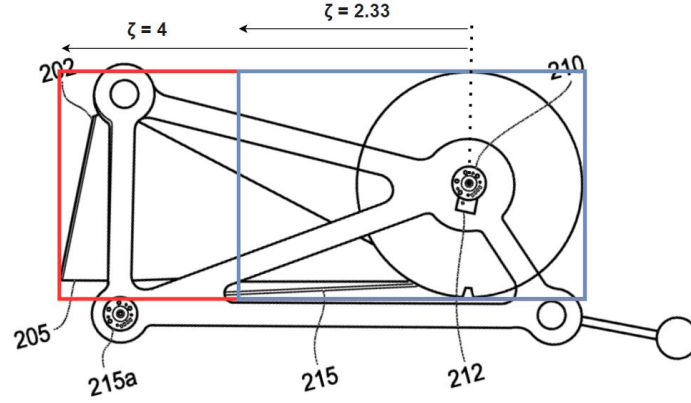


Figure 6.10: Additional volume scaling factor ζ for the analytical model vs the actual ζ required. Figure modified from [45].

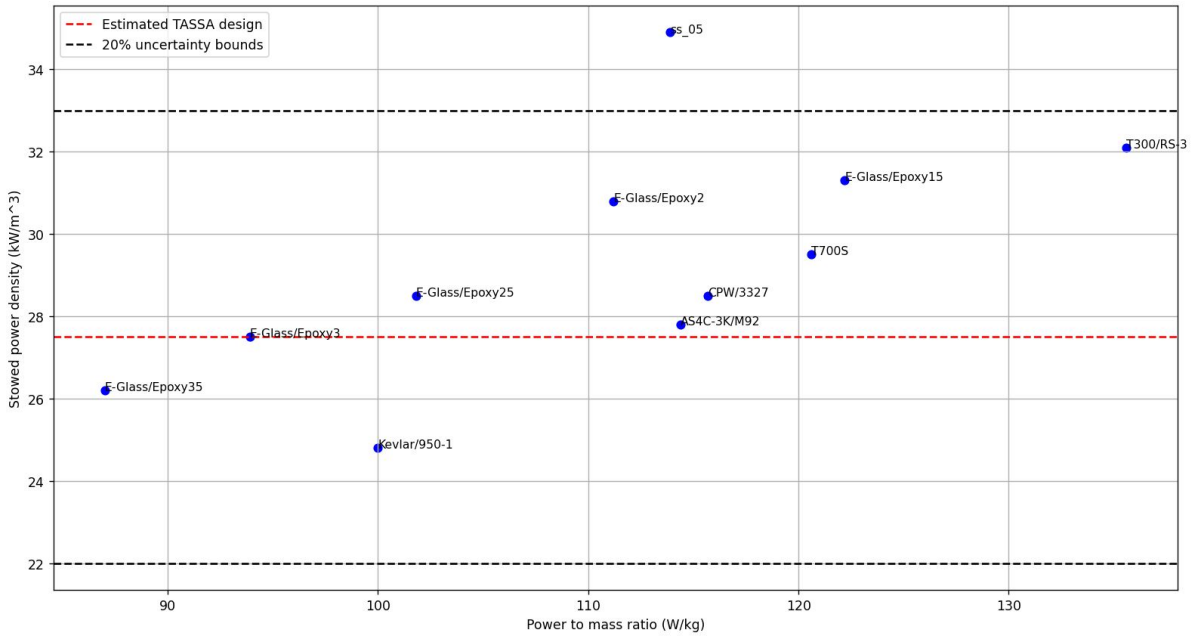


Figure 6.11: Performance of different TASSA designs using different stiffener materials, and with an increased ζ . The red horizontal line indicates the estimated performance range of the real TASSA design, and black lines indicate the uncertainty bounds.

With the modification, the designs show a much better match with the estimated real design of the TASSA, with all except one design made of stainless steel being within the uncertainty bounds. As roll-out solar arrays are generally made using carbon fiber composites, the stowed power density does not differ by more than 20% for the designs with CFRP materials. This difference could still be explained

by the additional volume that is not taken into account under the blue and red boxes in Figure 6.10, since the volume is largely based on the final radius of the stowed solar array. Nevertheless, this shows that the model is able estimate the performance parameters close to the real TASSA performance parameters, which in turn helps validate the model. While the ζ value had to be adjusted, this is an acceptable change considering that this roll-out solar array has a much greater enclosed volume than other roll-out solar arrays which the model is based upon.

6.5.2. iROSA

The iROSA's are the roll-out solar arrays used on the ISS, and are the largest roll-out solar arrays that have been flown to date. By inspecting pictures of an iROSA, it is evident that it employs the inverted deployment method, and uses two STEM type stiffeners. Even though they are STEM stiffeners, magnets are placed such that the ends of the cross section lock together, causing the structure to lock and act as a closed tubular stiffener [119]. Additionally, the stiffeners are known to have a radius of 8.25 cm [120]. Based on various pictures of the iROSA, the exact number of cells that are used can be calculated: 36 cells in width and 256 cells in length. Furthermore, it is known that the iROSA's use Spectrolabs' XTJ Prime triple junction solar cells, which offer an efficiency of 30.7%, and have a thickness of 140 μm [120]. When calculating the total power from these cells, a total output of 28.1 kW is estimated. NASA states that the beginning of life power of a ROSA is approximately 28 kW, which validates the amount of cells that are estimated, along with the type of cell [121].

The three remaining inputs are the deployment mechanism, the stiffener material, and the radius of the mandrel. Firstly, an assumption has to be made for the deployment mechanism, which in the case of the real iROSA's, is facilitated through self deployment, and damped with an eddy current damper [38]. As this type of mechanism is not included in the analytical model, a motor is assumed to be used instead. Secondly, since the mandrel radius is unknown, a test can be made which simulates the effect of a range of mandrel radii. Furthermore, the solar blanket is thicker than any stiffener materials, meaning that the stowed power density becomes independent of this variable, and can thus be analysed separately. Figure 6.12 shows the stowed power density of the iROSA as a function of the mandrel radius.

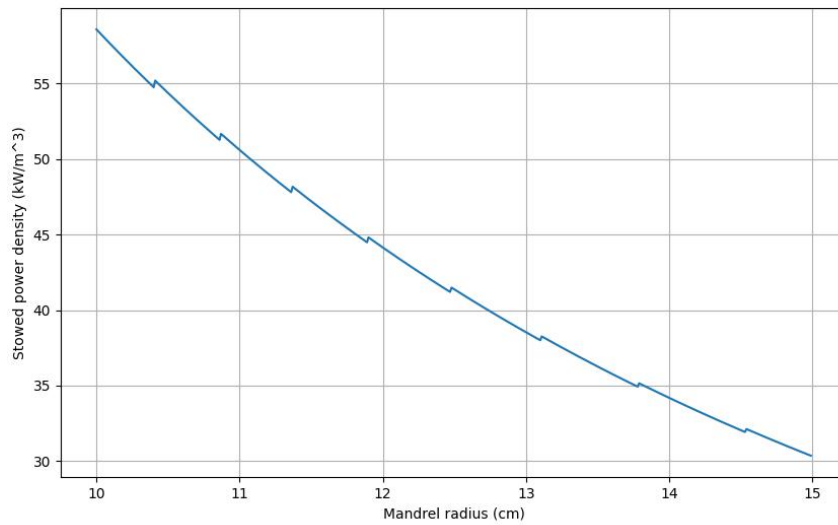


Figure 6.12: Effect of the mandrel radius on the stowed power density of the iROSA.

Generally, as the mandrel radius increases, the stowed power density decreases as expected. While the relationship is negative, it does not do so linearly or smoothly, and unusual dents or abrupt rises in the stowed power density can be observed. This phenomenon occurs due to how the final volume is calculated based on the number of coils that are wrapped around the mandrel. The dents in the graph happen when a slight increase in the mandrel radius causes one less coil to be wrapped around the mandrel. This reduction of coils decreases the volume more than the increase due to the mandrel radius increase, and thus the sudden increase in stowed power density is observed. Furthermore, the solar array can be sized to match the real stowed power density of 40 kW/m^3 if the mandrel radius is set to 12.7 cm . While this is likely not the real mandrel radius, it shows that there exists a mandrel radius which is realistic, and would achieve the same stowed power density as the real iROSA. This radius would be much larger than the minimum bend radius of the cells or coverglass, but other factors could have determined the mandrel radius for the iROSA. For example, the mandrel could have been constrained by the size of the eddy current dampers, or due to the flattening and coiling constraints of the composite material. Attempts were made to find the radius of the mandrel utilising pictures for verification purposes, but no proper estimation could be obtained.

Assuming a mandrel radius of 12.7 cm , the power-to-mass ratio can now be analysed by comparing the performance of the solar array with different stiffener materials. As the stiffeners are known to be made of CFRP composites, the other materials such as stainless steel and E-Glass are omitted for this analysis. A few materials did not meet the coiling and flattening stress constraints, and thus Table 6.13 shows the results of the materials that meet all the constraints. The power-to-mass ratio, as well as the natural frequency, are tabulated to give an indication of the effect of the materials on the performance characteristics.

Table 6.13: Power-to-mass ratio and natural frequency of the iROSA based on different stiffener materials.

Material	Power-to-mass ratio (W/kg)	Natural frequency (Hz)
T700S	140.4	0.102
T300/RS-3	142.3	0.083
T300/913	140.5	0.160
HT	142.3	0.121
CUD/125NS	142.0	0.125
CPW/3327	139.7	0.098
CPW/E773FR	142.0	0.126
T40/5320-1	142.2	0.134
AS4C-3K/M92	139.5	0.104

Among these materials, the T300/913 composite exhibits the highest natural frequency of 0.160 Hz with a power-to-mass ratio of 140.5 W/kg , while the T300/RS-3 composite has the lowest natural frequency of 0.083 Hz but the highest power-to-mass ratio of 142.3 W/kg , shared also by the HT composite. The natural frequency for most materials is significantly higher than the requirement of 0.06 Hz , validating the suitability of these materials and design method [122]. Overall, the power-to-mass ratios are relatively close, with the highest and lowest values differing by less than 3 W/kg . This shows that the power-to-mass ratio is not sensitive to the stiffener material since it has a low contribution to the overall mass of the solar array. Nevertheless, this value is significantly larger than the actual power-to-mass ratio of the iROSA, which is estimated to be between 100 – 120 W/kg [39]. Assuming a power-to-mass ratio of 110 W/kg , the analytical model would be overestimating the performance of the roll-out solar array by 27.3%. This additional mass may be due to a number of reasons. Firstly, the model may be underestimating the mass of some parts, or assuming wrong contingencies and margins. Furthermore, the attachments to the ISS, which are brought up with the iROSA's, are also unaccounted for in the model, since these are specific for the iROSA's. Finally, the eddy current dampers which are used on the solar arrays may also be much heavier than the motor mass estimated by the model. Overall, the results of the analytical model match well with the real values of iROSA, and aids in the validation of the model. While the power-to-mass ratio is overestimated by 27.3%, this may be due to additional unaccounted for masses that are specific to the iROSA. Nevertheless, for a preliminary estimation model, this discrepancy is acceptable.

6.5.3. Summary

From the previous validations, the analytical model is able to estimate the real performance of the roll-out solar arrays, with some acceptable discrepancies. With an adjustment to the volume scaling factor, the TASSA design was estimated with a 20% or lower difference for the stowed power density, while the power-to-mass ratio was estimated at a realistic range. The iROSA validation had consistent

results for the stowed power density, but showed an overestimation of the power-to-mass ratio of 30%. Despite there being differences with the real designs, it is important to note that roll-out solar arrays can take very different forms. As they are an emerging technology, there are many different designs which may have different requirements and employ different design decisions. Capturing accurately all these different designs is very complex, and would be difficult to simulate in a single model. It was especially evident for the TASSA, where the stowed power density predicted by the analytical model was significantly overestimated.

To conclude, for a model which analyses the conceptual design of the roll-out solar arrays, the discrepancies shown by the validation are acceptable. Given the early stage of design, it is expected that the performance and characteristics of a design would have significant changes as it proceeds to more detailed design phases. Since the analytical model achieves an accuracy that doesn't differ significantly from reality, it gives the model substantial credibility such that it can be used to perform an optimisation. With this, the results from the model are expected to be realistic and close to how the designs would perform in reality. It is nonetheless recommended that the model shall be validated further in the case that more information on other roll-out solar arrays is made public, as it would ensure more data points to compare the accuracy of the analytical model.

With the analytical model verified and validated, the optimisation model can be run to find out what the optimal designs are. The chapter commences by doing a critical analysis on the preliminary results in section 7.1. This is followed by section 7.2 wherein the set of optimal designs are obtained and analysed in more detail. Finally, a comparative analysis with traditional solar arrays is given in section 7.3.

7.1. Preliminary results

The following section presents the initial results of the optimisation model, along with analyses and evaluations to identify any anomalies and assess their validity. Initial results of the optimisation model are presented and analysed in subsection 7.1.1, followed by a convergence analysis in subsection 7.1.2 which evaluates at what population size and generation number the results have converged to the absolute optimal ones. Finally, some analyses are made on the complete design space in subsection 7.1.3 to better understand the optimisation model and verify the general results of the model.

7.1.1. Initial optimisation results

As explained in chapter 5, the optimisation model was split into two separate models: one for inherent stiffeners and one for external stiffeners. This was necessary as there were a lot of variables independent to each other, and the setup of the NSGA-II algorithm made this infeasible. Consequently, the results of each individual model would have to be combined in post processing, and a new Pareto front would have to be identified. This means that the Pareto front of each model are combined, such that a new final Pareto front can be identified using the optimal results of each optimisation model. The initial results after the model was completed and validated are presented in Figure 7.1.

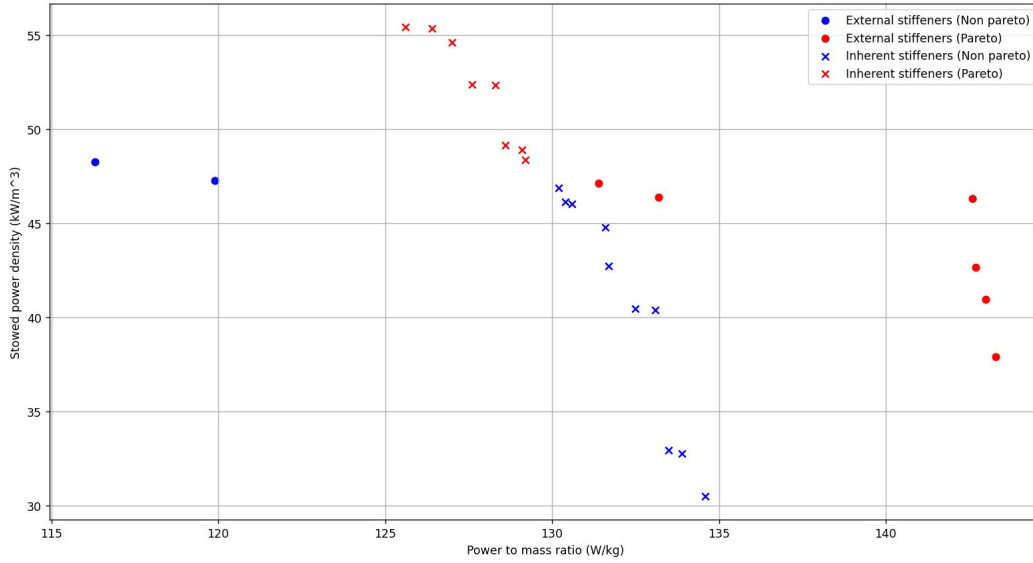


Figure 7.1: First results of the optimisation model. Number of generations = 200, population size = 200.

The graph combines the optimal results from the inherent and external stiffener models, and some conclusions can be drawn. If the optimal results are analysed separately, it is evident that the external stiffener Pareto front is very sharp, whereas the inherent stiffener Pareto front shows a more curved shape. The sharp right angle is not ideal, as it means that an increase in one objective results in a significant cost to the other objective. With a shape like this, the design in the corner of the right angle becomes the obvious best design. The optimal results of the inherent stiffeners show a much more ideal Pareto front since the curved pattern presents a more gradual trade off analysis between the objectives which allows for a more critical trade off. Nevertheless, when the two optimal sets are combined, the new Pareto front, represented by the red points in Figure 7.1, provides an interesting new set that combines the strong aspects of the individual design sets. Overall, the Pareto optimal results which have higher power-to-mass ratio are dominated by external stiffener designs, whereas the external stiffeners perform better for the stowed power density. Specifically, from the 48 kW/m^3 mark, the inherent stiffeners dominate the rest of the external stiffener designs. This presents a clear trade off which has a large impact on the design of the roll-out solar array.

While a detailed analysis of the design variables and elements will be analysed in more depth during the final results section, some relevant patterns can be identified. For the external stiffener designs, they all use the inverted deployment mode, and the majority of designs use the T300/RS-3 material and Cesi solar cell. Similar to the inherent stiffener designs, the external stiffener optimal designs are dominated by the T300/RS-3 material and Cesi solar cell, but unlike the other set, some designs employ the root drum deployment mode. Finally, for both sets the deployment mechanism varies. The reasons and implications of these patterns will be evaluated in more detail in the later sections.

7.1.2. Convergence analysis

Two important factors which have a large influence on the optimal designs are the population size and number of generations that are being run by the algorithm of the model, both of which are explained in detail in subsection 5.9.1. As both of these parameters are increased, the design space increases, and thus the probability of finding the absolute optimal results increases. However, the trade off for increasing these parameters is computational resources. This presents the opportunity to study the convergence of increasing the two parameters: at what point does increasing the population size and number of generations stop benefiting the optimal results found? Thus, the effect of increasing these parameters shall be analysed for convergence to ensure confidence in the optimality of the final results. To test the convergence, the number of generations and population size will be increased incrementally and the optimal results from the optimisation model will be analysed. To simplify the convergence analysis, the population size and number of generations will be kept equal to each other. While there might be certain combinations of varying population size and generation number, optimising this would not be beneficial and would take significant time to achieve. The results of the analysis are graphed in Figure 7.2.

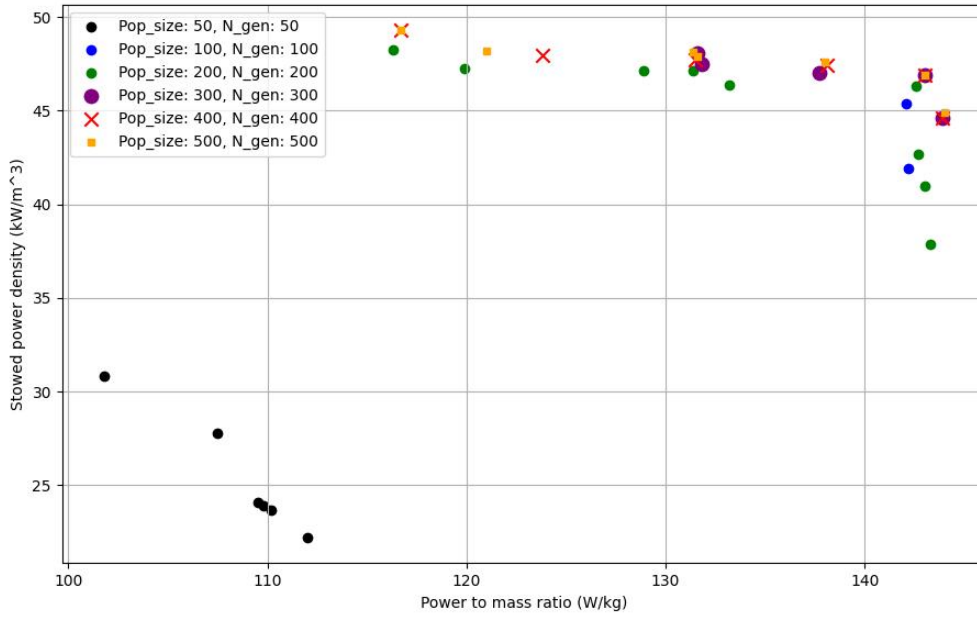


Figure 7.2: Convergence study of the optimisation model for external stiffeners. Pareto optimal results for each analysis are plotted.

From the convergence study, some notable conclusions can be drawn. Firstly, as expected, the performance of the designs has an improving relationship as the population size and number of generations increases. It is evident that the first study (50 generations and populations size) results in very sub optimal results compared to the other studies. As the population size increases beyond the 100 popula-

tion size and generation number mark, convergence is already noticeable, and the performance increase begins to decrease significantly. Nevertheless, improvements are still apparent until the study with a population size and generation number of 400. The results from the 500 study do not show improvements compared to the previous analysis, and thus the results can be considered as converged. Therefore, running the optimisation model with 400 generations and a population size of 400 is guaranteed to yield results which are very close to the absolute optimal while limiting the computational effort. This effectively requires the model to analyse 160,000 designs, each of which are run through the analytical model.

7.1.3. Complete population analysis

The following subsection will examine the design space of all of the populations which are produced by the model. Doing so will help better understand how the optimisation works, and if there are any anomalies which need further investigation. Firstly, the optimal results will be plotted alongside the complete design space, meaning that all individual designs of all populations will be plotted too. For resource and time optimisation, the study will only be made on the external stiffener optimisation model. Nevertheless, the results from the inherent stiffener would generate similar graphs as the analytical model is based on the same algorithm and other than the design variables, is very similar. The graph will all of the populations of the optimisation model is plotted in Figure 7.3.

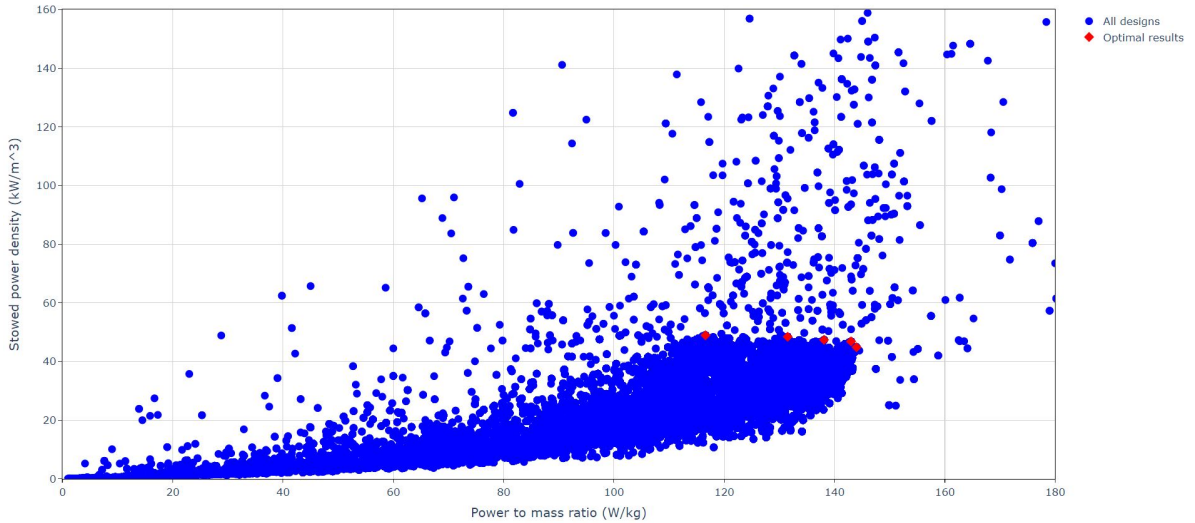


Figure 7.3: Complete population design space for external stiffener optimisation. Number of generations = 400, population size = 400.

Immediately, some abnormal results and patterns can be identified. The first unusual result from the graph is that there are many designs which perform better than the optimal results highlighted in red. An hypothesis which would explain why there are designs which perform better than the optimal results but are not optimal is because they don't abide to all of the given constraints. This means that although they don't abide to all constraints, they are still graphed as the optimisation model simulated

this design through the analytical model. A way to verify this statement is to highlight all of the designs which reject one or more constraints. If it doesn't highlight a design which performs better than the optimal designs, this would show a fault in the model. The adapted graph with the infeasible designs is visualised in Figure 7.4.

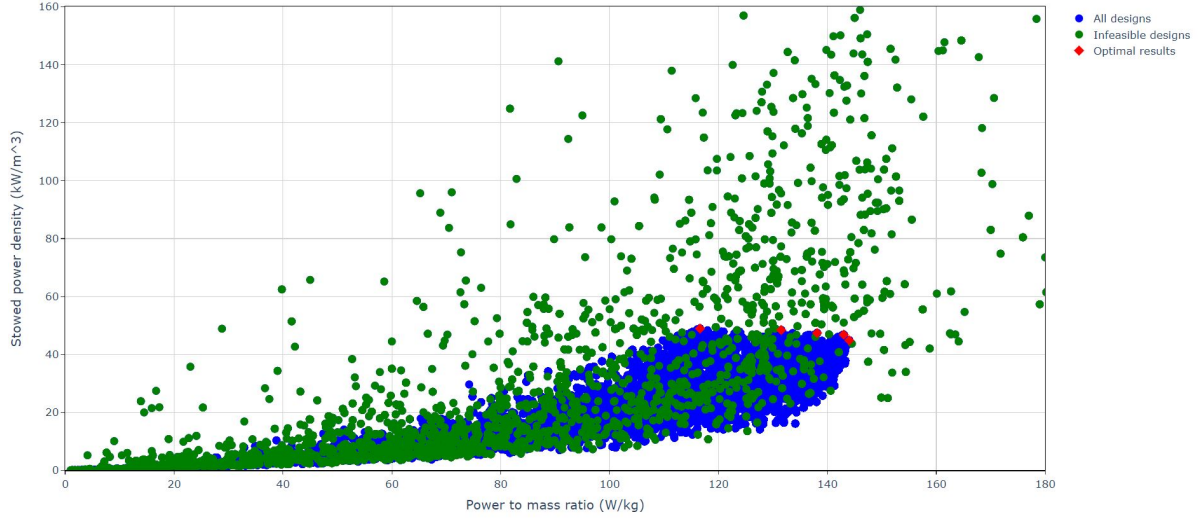


Figure 7.4: Complete population design space for external stiffener optimisation with infeasible designs which do not abide to one or more constraints. Number of generations = 400, population size = 400.

From the graph, it is evident that the initial hypothesis predicting the designs which perform better than the optimal results are infeasible is correct. In other words, there are no designs which perform better than the optimal results which don't abide to one or more of the constraints. This shows that the optimisation model works as intended and chooses the optimal results correctly. This also closely relates to how the optimisation algorithm functions. When the algorithm is started, it chooses a random value for each design variable within the specified range, meaning that first population would be scattered, and more importantly, have many designs which would not abide to the constraints. To verify this statement, the first population is highlighted and plotted in Figure 7.5. The last population is also highlighted for a better understanding of the optimisation model evolution of generations.

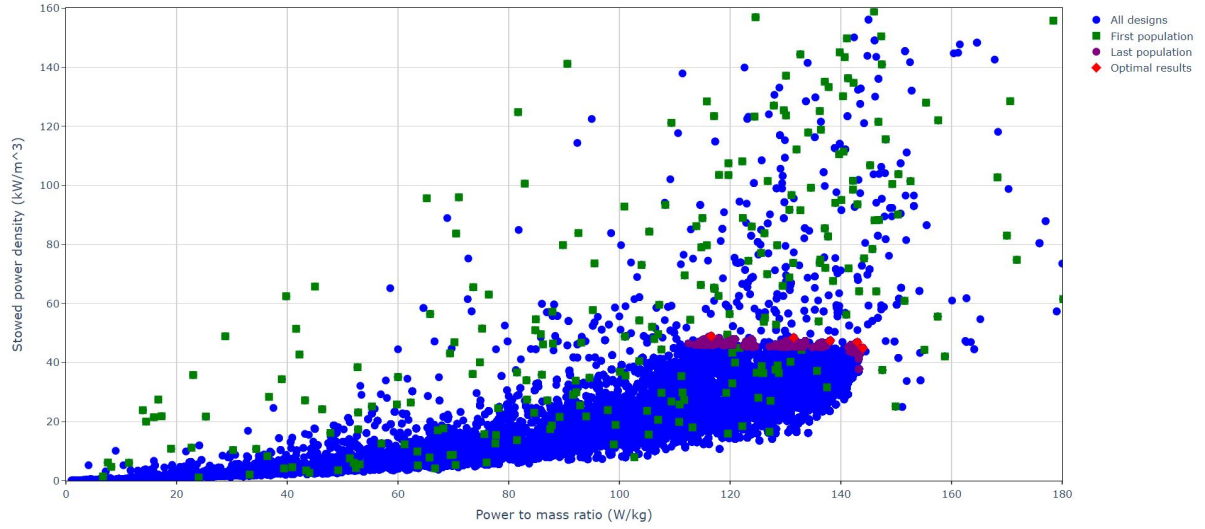


Figure 7.5: Complete population design space for external stiffener optimisation with highlighted first and last population. Number of generations = 400, population size = 400.

As seen from the graph, the first population, shown by the green points, are very scattered and random. Moreover, this gives a good representation of the evolution of populations and shows how the algorithm functions: it first calculates designs based on random initial variables, and over the generations refines the results based on the constraints. This is further justified when examining the final population, shown in purple in the graph. It is evident that the final population has designs which are all very close to the optimal designs, showing how the algorithm finds the final optimal results.

Another anomaly that can be seen in the graph is that designs are not able to have a low stowed power density, and a curved line made from the points suggests there's a minimum stowed power density for each power-to-mass ratio. This should not be the case as the stowed power density in theory could be made as low as possible if the mandrel radius were to keep increasing. It is predicted that this phenomenon is happening as the range of the mandrel radius is constrained such that it can't increase beyond a certain point, effectively acting as an artificial constraint as designs with a lower stowed power density would require a mandrel radius larger than the given range. During the development of the optimisation model, it was clear that there weren't any designs which had a mandrel radius over 8 cm, and thus the range for this variable was constrained with a maximum radius of 10 cm. To verify that this is the correct reason, the range of the mandrel radius is increased from 10 cm to 50 cm, and the new results are plotted in Figure 7.6.

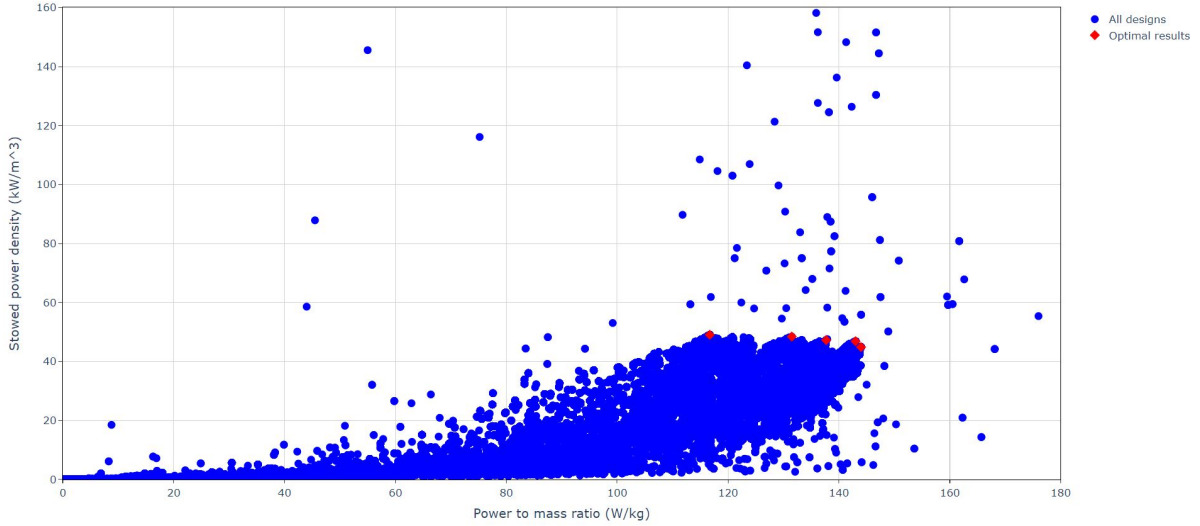


Figure 7.6: Complete population design space for external stiffener optimisation with wider mandrel radius range. Number of generations = 400, population size = 400.

From the graph, it is evident that increasing the allowable range does indeed allow designs under the curve to be created, and a much more evenly distributed design space is seen. This shows that the limited range was creating an artificial constraint as predicted. Nevertheless, the optimal results do not significantly vary since a larger radius is generally not beneficial to the performance of the solar array. As the results are not sensitive to this change, the previous method of constraining the range within a smaller range is preferred, since there is a higher likelihood to find optimal results faster.

7.2. Final results

Having verified that the optimisation model produces reliable and correct results, and knowing the correct number of generations and population size to run the model with, a final analysis can be conducted. This is done in the same way as in subsection 7.1.1, wherein the Pareto front of each optimisation model is combined and a new final Pareto front is found with a mix of external and inherent stiffener designs. The new results are plotted in Figure 7.7.

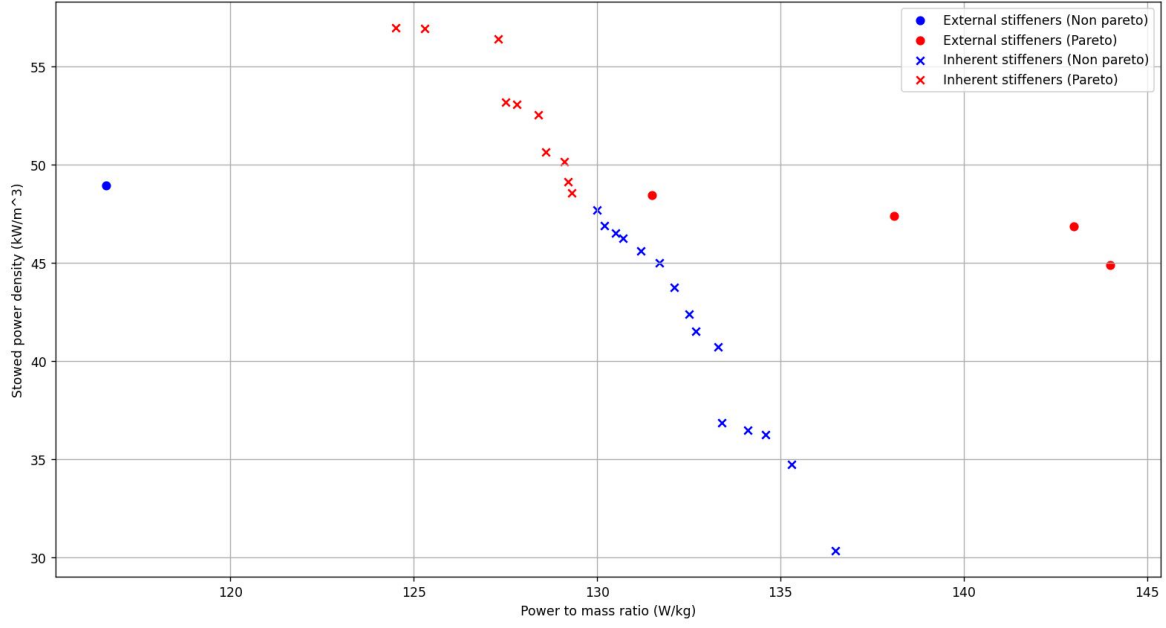


Figure 7.7: Final results of the optimisation model. Number of generations = 400, population size = 400.

The results from the graph are very similar to the preliminary results in Figure 7.1, with a few differences. This is to be expected, as the convergence analysis showed that running the model with 200 generations and a population size of 200 already resulted in nearly converged absolute optimal results. The main difference is that there are more optimal designs for the inherent stiffeners, while the external stiffener design space remains less populated. Nevertheless, the general conclusion from the initial results holds true: inherent stiffener designs have higher stowed power densities, while the external stiffener designs have the advantage for the power-to-mass ratio.

A limitation of this analysis is that many of the external stiffener designs are exactly 1000 W, suggesting that they are being constrained by the power requirement from Table 3.2. This means that even if there are higher performing results, they are not shown in the outcome as they would be conflicting with a constraint. An insightful analysis would be to remove this constraint, and examine how the results change. This is simulated and the new results are plotted in Figure 7.8. Again, all the optimal designs from each optimisation model are combined such that a final Pareto front can be identified. For comparison, the optimal designs from Figure 7.7 are plotted as well.

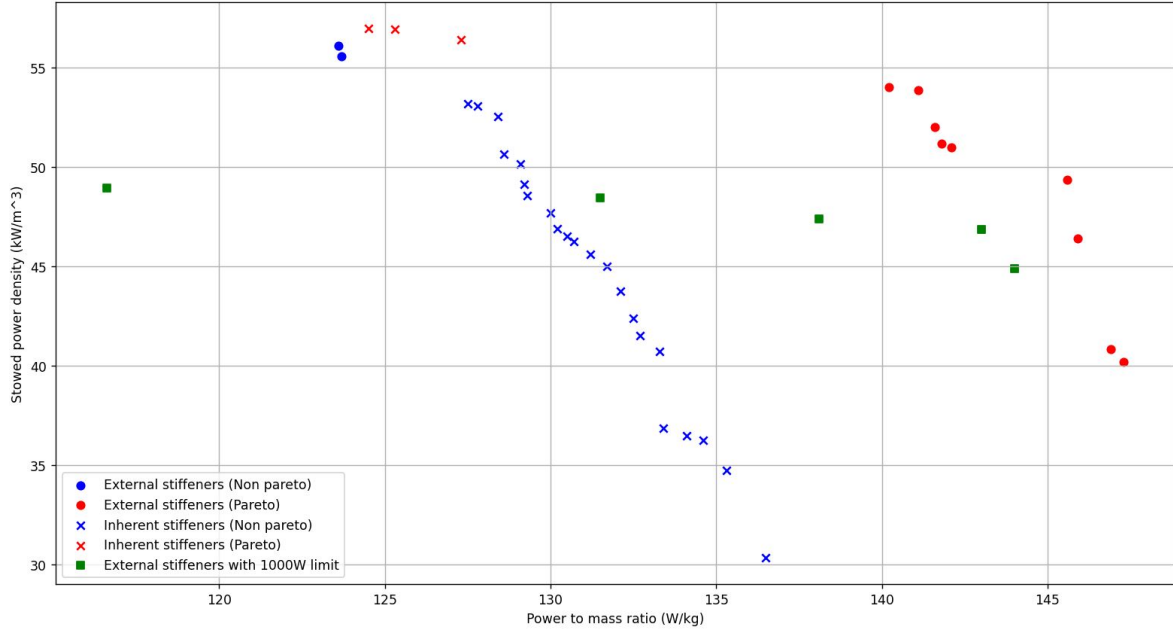


Figure 7.8: Optimal results of the optimisation model without the 1000 W power constraint. Number of generations = 400, population size = 400.

From the updated graph, it is evident that the new unconstrained external stiffener designs have improved performance over the old designs. Notably, there exist designs which exceed 145 W/kg and 55 kW/m^3 . On the other hand, the inherent stiffener designs did not change at all since they were not reaching the 1000 W limit. An increase in the performance of external stiffeners, combined with the unchanged performance of external stiffeners means that the external stiffeners dominate the design space significantly, with only three external stiffener designs left as part of the final Pareto front. These designs become advantageous as the stowed power density becomes the more critical or important objective. This means that if volume becomes a critical factor for a certain mission, and the stowed power density is deemed significantly more important than the power-to-mass ratio, external stiffeners may be the more suitable choice. Taking this approach however does not abide to the minimum power-to-mass ratio requirement, so a critical trade off would have to be made to justify this choice.

Although the external stiffeners have increased performance, the maximum power-to-mass ratio shows a percentage increase of less than 3%, while for the stowed power density the increase is roughly 10%. Despite the improvement, it shows that the removal of the requirement does not have such a significant effect on the overall performance. Furthermore, the power-to-mass ratio of external stiffeners is higher only by about 15% compared to inherent stiffeners. Considering that the designs are made at a conceptual level, these differences lie within the uncertainty bounds of the calculations, meaning that the performance of inherent and external stiffener designs may be much more comparable than the optimisation model suggests. Overall, while it can be concluded that designs employing external stiffeners have an advantage for most of the design space, inherent stiffener designs cannot be disregarded.

as possible options. It is recommended that for the following design stages the non optimal results highlighted in blue in Figure 7.8 are analysed too.

A more detailed examination of the designs can be made by investigating the parameters which make up the designs. This is useful to see if there are any elements or design variables which dominate others, or if there is an even distribution of them amongst the optimal designs. This is made by analysing the constituent parameters of the optimal designs highlighted in red in Figure 7.8. For a broader and comprehensive analysis, if any parameters are included in the non-optimal results (highlighted in blue in Figure 7.8), these will be annotated around brackets. The reason for this is to show that while they might not be the optimal results from the optimisation model, they have potential, and thus may be useful to keep into consideration when continuing to later design stages. The summary of the parameters which make up the optimal designs is found in Table 7.1.

Table 7.1: Summary of the parameters from the optimal designs.

Parameter	External stiffeners	Inherent stiffeners
Power	1157 - 1307	469 - 642
Material	T300/RS-3, T40/530-1	T300/RS-3
Solar cell	CTJ30-80	CTJ30-80
Deployment mode	Inverted	Inverted
Deployment mechanism	Motor, Escapement	Motor, Escapement
Mandrel radius	5.9 - 7.0	6.1 - 6.2
Stiffener type	Lenticular, Interlocking	-
Stiffener radius	5.2 - 6.7	-
Stiffener subtended angle	-	36 - 55

From the breakdown summary, some interesting results can be observed. Firstly, it is evident that the inherent stiffener designs offer overall lower power designs, having a highest power of 642 *W* compared to 1307 *W* from the external stiffener designs. CESI's CTJ30-80 cell dominates all other cells for all optimal designs, which is to be expected from a triple junction cell with very high efficiency. Nevertheless, some non Pareto designs (blue marks in Figure 7.8) do include Spectrolabs' XTJ Prime cells, so some variability may be possible. Despite there being other cell technologies which offer thinner and lighter solutions, their lower efficiency make them overall less optimal compared to the triple junction technology. The T300/RS-3 CFRP composite is also the most optimal in the majority of cases due to its low thickness, which allows the mass and volume to both be minimised. The T40/5320-1 material is used in the high power-to-mass ratio external stiffener designs due to its low thickness and low density, making it ideal for minimising mass. For the deployment mode, it is clear that the inverted mode is preferred, largely due to the lower additional volume scaling factor and direct electrical connection. Nevertheless, some non-optimal inherent stiffener designs also use the normal deployment mode, mean-

ing that it has potential to be optimal. The deployment mechanism is not dominated by one type, and in both inherent and external stiffener designs the mechanism is evenly populated. This indicates that the deployment mechanism have a similar effect on the performance parameters of the solar array. Lastly, an even distribution of lenticular and interlocking stiffener types are also visible for the external stiffeners, indicating that there is no superior type. The breakdown shows that there exists considerable parameter variation between the designs, and overall gives a good indication and range of what the best designs are to investigate further.

7.2.1. Mass distribution analysis

To get a better understanding of how the system is broken down into its components, a mass distribution can be made. This breakdown also serves as a verification check to see if the percentages are in the realistic ranges. In other words, it is expected that the solar blanket would have the largest percentage of the total mass, whereas the deployment mechanism and anti blossoming device are expected to have relatively small mass dividends. Therefore, an optimal external stiffener design as well as an optimal inherent stiffener design are evaluated to see how their complete mass is distributed amongst their components. The designs are chosen at random from the optimal designs in Figure 7.8. Note that the external stiffener optimal design employs an inverted deployment mode, meaning that the spreader bar is not required. Similarly, inherent stiffeners do not need a spreader bar, meaning that this component is removed from the mass distribution of this design. The complete breakdown of the mass for an optimal external and inherent stiffener design is given in Table 7.2 and Table 7.3, respectively.

Table 7.2: Mass distribution of an optimal external stiffener design.

Array element	Mass (g)	Percentage of total mass (%)
Stiffener	685	9.5
Solar blanket	4905	68.3
Mandrel	852	11.9
Deployment mechanism	85	1.2
Spreader bar	0	0
Anti blossoming device	78	1.1
Additional mass	577	8.0
Total	7182	100
Margin	1436	20
Total with margin	8619	120

Table 7.3: Mass distribution of an optimal inherent stiffener

Array element	Mass (g)	Percentage of total mass (%)
Stiffener	864	28.2
Solar blanket	1558	50.8
Mandrel	216	7.0
Deployment mechanism	94	3.1
Anti blossoming device	89	2.9
Additional mass	247	8.0
Total	3068	100
Margin	613	20
Total with margin	3680	120

Analysing the mass breakdown of the external stiffener design, some observations can be made. In general, the percentage distribution of all the components is what is to be expected, with the solar blanket assuming the majority of the system's mass. Additionally, the mandrel and stiffener mass have significant mass contributions of 11.9% and 9.5%, respectively, which again is to be expected since they also make up a large part of the system, but have a small cross sectional area. The deployment mechanism and anti blossoming device each make up a bit over 1% of the solar array mass. This may seem like a small contribution, but considering their small size or use of lightweight materials is justifiable. Lastly, the remaining subcomponents which are part of the additional mass are calculated to have an allocation of 8%.

The mass distribution for the inherent stiffener design shows similar results as for the external stiffener design mass breakdown, with some differences. The most apparent change is that the percentage of the total mass of the solar blanket is almost 18% lower, while the stiffener mass contribution is almost three times larger. Despite this large difference, this is to be expected since the inherent stiffener design stiffener is much larger in comparison to the external stiffener design stiffener. This design however weighs half as much as the external stiffener design as it is less wide and long, so even if has an optimal power-to-mass ratio + stowed power density combination, its smaller area means it produces significantly less power. The identical additional mass of 8% is unusual, but further inspection reveals it to be purely coincidental, as other optimal inherent stiffener designs show different values.

In conclusion, the mass breakdowns of both types of roll-out solar arrays are in the expected ranges. For both cases, the solar blanket is revealed to contribute to the largest percentage of the total mass, with the external stiffener design having a higher solar blanket mass percentage. Other than the stiffener mass for the inherent stiffener design having also a high percentage of the total mass due to its large size, the rest of the components for both cases have significantly low contributions. The additional mass allocation is unusually consistent for both designs, though this appears to be coincidental rather than a fixed design outcome.

7.2.2. Performance evaluation

Having analysed the final results and verified their mass distribution, their overall power-to-mass ratio and stowed power density performance can be analysed and evaluated. For the power-to-mass ratio, their placement in the state of the art solar array comparison graph of subsection 2.2.4 can be made. This will visually show where the final design is expected to place amongst the rest of the solar arrays. The placement is made with an area rather than a point to indicate the potential range where the final design could be. The expected placement of the roll-out solar array can be seen in Figure 7.9.

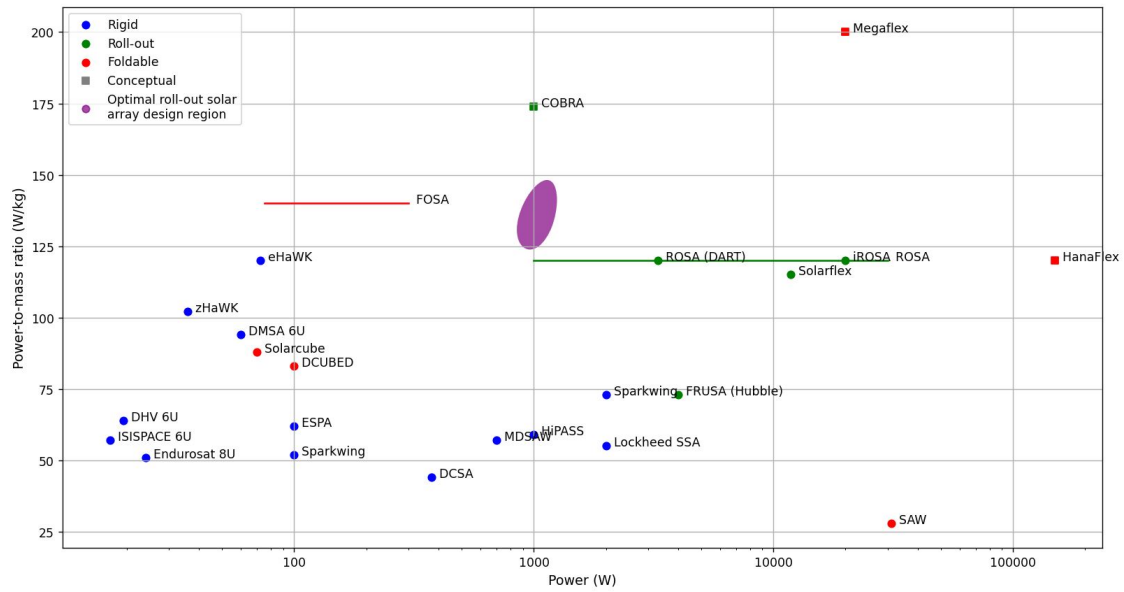


Figure 7.9: Expected placement of the roll-out solar array design in comparison to other solar arrays on the market.

Without taking into account the conceptual designs, it is evident that the potential design is expected to be a high performing solar array in the overall solar array market. More specifically, the proposed design would perform better than both existing rigid and roll-out solar arrays, and most foldable solar arrays. Of all the possible designs, Exoterra’s FOSA is the only one which has a comparable power-to-mass ratio. Nevertheless, even if the FOSA had a higher power-to-mass ratio than the final design, its lower power output would not make it a direct competitor. Compared to the solar arrays in the same power output range, the roll-out solar array would be the highest performing solar array. Compared to the traditional solar arrays, it would offer over double the power-to-mass ratio, effectively offering half the weight of the system for the same power output. The closest competitor would be Redwires ROSA, which thus far has only claimed to be able to make a 1 kW array, and thus may be considered a conceptual design. Similarly, while the COBRA is stated to have an even higher power-to-mass ratio of 175 W/kg, it is only a conceptual design which hasn’t flown yet, and thus is not considered as a design for comparison.

In terms of the stowed power density, the 40 kW/m³ benchmark set at the beginning of the study has been shown to be achievable and surpassable. Unfortunately, as this performance indicator is not used for traditional solar arrays, it cannot be compared to the state of the art solar arrays in the 300–1000 W range which are populated by traditional solar arrays. Nevertheless, with a maximum stowed power density of 57 kW/m³, optimal designs indicate that a 40% increase over the state of the art benchmark can be achieved with a roll-out solar array for small satellites. Overall, the development of the system has a lot of potential and could offer significant improvements to the performance of the solar array market.

7.2.3. Requirement compliance

The following subsection investigates how well the requirements formulated in Table 3.2 have been complied with. This is important to understand to what degree the goals and requirements have been met in the current design stage by the optimal designs. Compliant requirements which need no further action are scored with a ✓, while non-compliant requirements are scored with a ✗, indicating a need for re-evaluation. As the thesis focuses on the feasibility and conceptual design, some requirements may not be able to be assessed against their compliance. These will be scored with a - and will need to be re-assessed for compliance in later stages of design. The compliance assessment of each requirement is given in Table 7.4.

Table 7.4: Compliance of the roll-out solar array requirements established in Table 3.2.

ID	Requirement	Compliance	Comments
1	The system shall have a stowed configuration and de- ployed configuration	✓	
2	The system shall unfurl from a rolled up state during deployment	✓	
3	The system shall be able to transfer power from the array to the satellite	✓	Flex PCB ensures power transfer
4	The deployment mechanism shall be fault tolerant	✓	
5	The system shall be designed to prevent unintentional de- ployment during launch or prior to intended deployment.	-	Testing required
6	The system shall be designed using solar cells qualified for space	✓	
7	The power-to-mass ratio of the system shall be at least 140 W/kg	✓*	*Some optimal designs reach below 140 W/kg
8	The stowed power density of the system shall be at least 40 kW/m^3	✓*	*Some optimal designs reach below 40 kW/m^3
9	The system shall have a fixed power output capacity within the range of $300 - 1000 \text{ W}$	✓*	*Some optimal designs reach above 1000 W
10	The system shall have a maximum width of 96cm	✓	Ensured with optimisa- tion model
11	The system shall withstand the equivalent of 5 years of radiation in LEO	-	Analysis or testing re- quired

Table 7.4 continued from previous page

ID	Requirement	Compliance	Comments
12	The system shall be able to operate nominally from -60 to +90 deg C	-	Analysis or testing required
13	The system shall be able to withstand atomic oxygen	✓	Coverglass ensures solar cell resistance
14	The impact of a micrometeorite on the array shall not result in system failure	-	Analysis required
15	The deployed system shall withstand 28 <i>N</i> of thrust in the lateral directions	✓	Ensured with optimisation model
16	The deployed system shall have a first natural frequency above 0.6 Hz	✓	Ensured with optimisation model
17	The deployed system shall withstand angular accelerations of 2.5 deg/ <i>s</i> ² in all directions.	✓	Ensured with optimisation model
18	The deployed system shall not twist by more than 5 deg	✓	Ensured with optimisation model
19	The system shall automatically lock in its deployed configuration	✓	Fully deployed stiffeners are stable
20	The system shall use materials which show a relative mass loss of <1% and collected volatile condensable material of <0.1% as outgassing properties	-	

From Table 7.4, it is immediately evident that most of the requirements have either been met or have pending assessment as they cannot be evaluated yet. The quantifiable requirements such as 15-17 are compliant since they directly became constraints in the optimisation model, meaning that any optimal design must abide to them. On the other hand, assessing and verifying the radiation or thermal requirements would require simulations or tests to be conducted. While the individual materials and components chosen for the roll-out solar array were ensured to be space qualified, it does not mean the whole array system behaves the same way, especially for the thermal environment requirement. Likewise, the outgassing requirement for the materials would only be able to verified once the final materials are chosen.

Requirements seven through nine relate to the performance parameters, namely the power-to-mass ratio, stowed power density, and power output. These are marked as complied with an asterisk, as it depends on which design is chosen. This is because in section 7.2 it was found that some optimal designs had

performance parameters outside the range given in the requirements. This is an acceptable deficiency as designs in this early conceptual stage are still worthwhile to investigate. Furthermore, designs above the 1000 W requirement yielded higher performance parameters, introducing a trade off between sticking to the requirements, or improving the performance of the solar array.

Overall, there are still a few requirements which cannot be fully assessed, which is not ideal. Nevertheless, all other requirements have met the goals, and given that this is still in the early stage of the design, it indicates that a roll-out solar array can be achieved such that it abides to the initially set requirements. Moreover, it aids as a foundation for further development since the unassessed requirements can be a starting point for the more detailed design stages.

7.3. Comparison with a traditional solar array

In order to fully answer the initial research question, a critical comparison between the optimal roll-out solar array designs and a traditional solar array is required. For the purpose of a fair comparison, a hinge deployable array is assumed to be the traditional solar array. This is done as it would be the closest competitor in the satellite solar array market for the 300 – 1000 W range.

When examining what roll-out solar arrays have to offer, some immediate advantages can be identified. Firstly, the investigation in subsection 7.2.2 showed that the expected power-to-mass ratio of the roll-out solar array would be double that of rigidly deployed arrays. This is a significant increase that could provide satellites with increased power generation systems, ultimately enabling more power demanding payloads. In terms of the stowed power density, it is difficult to compare the calculated value with that of traditional solar arrays. This is because for hinge deployed solar arrays, this metric is not used as a performance parameter. Nevertheless, the compact nature of roll-out solar arrays ensures that they are space efficient. A last advantage of roll-out solar arrays is that they are also inherently more modular and can be scaled in size to fit the requirements of missions. Roll-out solar arrays are made such that they can be made shorter or longer by only altering stiffener and solar blanket length, given that the constraints are still met. This also enables a system which could rapidly be sized for specific missions without the need for large redesigns.

Despite the clear advantages in terms of the performance, developing a roll-out solar arrays also has its limitations. Firstly, the initial costs of developing a roll-out solar array would be much higher as it is a novel and complex design. Compared to traditional solar arrays which have already been optimised with little room for innovation, developing a roll-out solar array from scratch would incur a lot of research and development costs. These would include all the necessary engineering and testing costs to create the new system. In general, a critical trade off and cost analysis would have to be conducted to find out if the development of a roll-out solar array and its potential advantages would outweigh the additional costs. The second limitation of a roll-out solar array compared to a traditional array stems from their unusual geometry. While traditional solar arrays are flat and commonly share the areal size of the side of a satellite, roll-out solar arrays are bulky cylindrical structures. While their stowed power density

can be very high, the geometry may still make it less viable as an option due to integration challenges. This disadvantage is necessary to take into account, because even a high performing roll-out solar array may not be feasible if it does not integrate seamlessly with a satellite.

Another effective yet different way to compare the two types of solar arrays is through a risk analysis. This comparative risk analysis aims to evaluate potential vulnerabilities and limitations associated with each solar array, which serves to better understand their performance from a more qualitative and reliability driven point of view. By examining the potential paths to failure, the analysis ultimately helps provide a more comprehensive and broader comparison between the two types of solar arrays.

The risk analysis is conducted by first identifying the general risks for satellite solar arrays, followed by mapping them appropriately on a risk matrix for both types of solar arrays. Seven are identified, and listed in Table 7.5. These range from mechanical risks such as deployment failure, to environmental risks due to thermal variation and radiation damage, to operational risks such as the assembly and testing failure risk. Each of these risks are then ranked by estimating the likelihood of the risk happening, and the severity it would have if it would occur. Once all the risks have their likelihood and severity estimated for both types of solar arrays, they can be mapped out on a risk matrix. The risk matrices of the traditional deployable solar array and roll-out solar array are visualised in Table 7.7 and Table 7.8, respectively.

Table 7.5: Risks for a deployable satellite solar array.

Risk	Description
A	Deployment failure
B	Space debris collision
C	Thermal cycling failure
D	Failure due to radiation
E	Manufacturing defects
F	Failure against competition / market longevity
G	Assembly and testing failure

Depending on the severity and likelihood of the risk, its placement on the risk matrix will determine the necessary response strategy. For example, risks with high likelihood and high severity would require the solar array to undergo an immediate review, while risks with low severity and low likelihood would not require any action. This also helps identify where engineering resources are required the most by prioritising the improvements needed to lower the intolerable risks. A legend of the response strategy for each type of risk is outlined in Table 7.6.

Table 7.6: Legend for the risk matrix colours.

Colour	Response strategy
	Acceptable. No action required
	Tolerable. May need to be reviewed for improvements.
	Not Acceptable. Immediate action required.

Table 7.7: Risk matrix for a traditional deployable solar array.

5-Almost certain					
4-Likely					
3-Moderate			D		
2-Unlikely		C, G	E	A, B, F	
1-Rare					
Likelihood/ Severity	1-Insignificant	2-Minor	3-Moderate	4-Significant	5-Mission failure

Table 7.8: Risk matrix for a roll-out solar array.

5-Almost certain					
4-Likely					
3-Moderate		G	E	D	A
2-Unlikely				B, C	
1-Rare				F	
Likelihood/ Severity	1-Insignificant	2-Minor	3-Moderate	4-Significant	5-Mission failure

It is evident from the matrices that a traditional deployable solar array is in general a system with lower risk compared to a roll-out solar array. For both solar arrays, the majority of risks are in the tolerable area, which is not ideal but understandable given that they are a critical system of a satellite. However,

other than the failure against competition risk, the roll-out solar array risks rank either the same or higher in terms of likelihood or severity. For the thermal cycling and radiation risks, their severity is higher due to the natural higher risk of material damage that CFRP's have compared to metals [123] [124]. Furthermore, the use of CFRP's also increases the likelihood of manufacturing defects since the production of CFRP's is more complex than metals.

A standout from the analysis is that the deployment failure risk for roll-out solar array ranks too high in its likelihood and severity, placing it as an intolerable risk which requires immediate action. This risk includes both the risk of the hold down and release mechanism failing, as well as the deployment mechanism failing. The reason it ranks a level higher in both likelihood and severity compared to a traditional deployable solar array is for two main reasons. The likelihood scores higher due to it being a more complex system with higher amounts of moving parts and deployment processes. Furthermore, if a traditional solar array would fail to deploy, the outer solar array could still be used like a body mounted solar array, and thus the mission may still continue to a certain degree. If a roll-out solar array would fail to deploy, there would be no active cells to generate power, meaning that mission failure would be certain. The intolerable risk would require an immediate response to reduce either the likelihood or severity. Some mitigation strategies are possible to implement. The first idea would be reduce the likelihood by adding redundancy. Although this may decrease the likelihood, it adds complexity to the system, and thus the reliability may decrease. A careful analysis would have to be made to check whether adding redundancy actually decreases the likelihood considering the increased complexity of the system. Another method would be to ensure high reliability in each component of the system and system as a whole through testing. In any case, implementing a mitigation strategy would be necessary such that the likelihood would change from moderate to unlikely or rare, and ultimately change the risk to a tolerable one.

Overall, the two risk analyses provide insightful results which ultimately highlight the fact that roll-out solar arrays generally have higher risks involved with their development and use on satellites compared to traditional solar arrays in many aspects. The risk analysis particularly highlights that roll-out solar arrays are more prone to risks associated with deployment failure, environmental risks, and manufacturing defects amongst others, largely due to their higher complexity and use of CFRP's. Despite their higher risks, mitigation strategies can be employed to make all the risks tolerable.

In conclusion, if the additional costs are deemed to be acceptable, the integration is proven to be feasible, and the risks can be reduced to an acceptable level, a roll-out solar array would be worthwhile to investigate further. With the ever increasing capabilities and power demands from smallsat payloads, innovative solutions such as roll-out solar arrays become necessary. An optimal roll-out solar array design is expected to achieve double the power-to-mass ratio of a traditional rigid solar array, meaning that a customer could either have the same power for half the mass, or double the power for the same mass. Furthermore, the modularity and variation in designs also shows that a roll-out solar array can be tailored to meet the specific requirements of different missions.

Conclusion and future work

The following chapter discusses the conclusions drawn from the feasibility study by analysing how the results have helped answer the research question. The recommendations are also proposed for future work, including an outlook on the most critical aspects of the design that would need revision and deeper analysis.

8.1. Conclusion

This feasibility study aimed to answer the question: How can a roll-out solar array be designed for small satellites, such that it outperforms current solar array technologies? The research focused on finding a set of roll-out solar designs which achieve a higher power-to-mass ratio and stowed power density compared to existing solar arrays in the $300 - 1000\text{ W}$ range. The results indicate that roll-out solar arrays for small satellites have the potential to outperform the current solar array technologies, and to answer the research question, a roll-out solar array has many possible design combinations which would result in a higher performing power generation system. Optimal designs utilising lightweight materials such as CFRP and efficient triple junction solar cells consistently outperformed other design combinations, and the inverted deployment mode proved to be the optimal design option. Furthermore, designs employing external stiffeners would never require more than one stiffener, and overall were more favourable for higher power-to-mass ratios. On the other hand, inherent stiffener designs offered higher stowed power density at the expense of lower power-to-mass ratios. The type of stiffener and deployment mechanism of optimal designs had varied outcomes, indicating that neither design option had a clear advantages over the other.

In terms of their performance, optimal designs demonstrated power-to-mass ratios exceeding 145 W/kg and stowed power densities capable of surpassing 55 kW/m^3 . For the $300 - 1000\text{ W}$ power range, the state of the art power-to-mass ratio of traditional solar arrays is $\sim 60\text{ W/kg}$, indicating a more than double potential increase in power-to-mass ratio performance from a roll-out solar array. Compared to the general state of the art solar arrays which reach $120 - 140\text{ W/kg}$, the optimal designs power-

to-mass ratio would be comparable. For the stowed power density, optimal designs could surpass the general state of the art solar array performance by over 40%. While this indicates the potential for a compact and space efficient design, a comparison to traditional solar arrays in the 300 – 1000 W cannot be made due to a lack of data. Despite the promising performance advantages, roll-out solar arrays are inherently more complex systems, which drive up the research and development costs, as well as the risks. Notably, the risk of deployment failure presented a significant intolerable risk that would require a mitigation strategy. Furthermore, the bulky cylindrical geometry of a roll-out solar array poses additional integration challenges and could still be less space efficient despite their high stowed power density.

Verification analyses showed that open stiffener structures were not able to be verified due to their limited modelling technique. Nevertheless, a comparative analysis showed that closed stiffeners, which were verified, were in any case more favourable than open stiffeners for a roll-out solar array application. Thus, the open stiffeners were ultimately discarded as design options. Validation against existing roll-out solar arrays demonstrated the ability of the analytical model to estimate actual performance with acceptable discrepancies of $< 30\%$. Even so, the validation against the TASSA highlighted the difficulty in modelling unique design features of roll-out solar arrays, while the validation against the iROSA indicated an overestimation in the power-to-mass ratio from the analytical model.

Overall, the study successfully demonstrated the potential that roll-out solar arrays hold for the small-sat market. Compared to traditional solar arrays, they offer significantly increased performance, and compared to the broader state of the art solar arrays on the market, it is expected that an optimal design could offer comparable or higher performance. Despite some limitations of roll-out solar arrays, this highlights the potential to drastically improve the operational capabilities of smallsats by offering efficient and compact solar arrays. Other than ensuring the feasibility of such a system, the study also provides a strong foundation for future development should Revolv Space decide to continue researching into this promising innovative technology.

8.2. Recommendations

Based on the findings and methods of this study, several recommendations are proposed for future research. Firstly, as the study focuses on the conceptual design phase, future research should involve developing detailed designs, including considerations for manufacturing, assembly, and testing. To address the requirements which were not able to be assessed for compliance, prototype development and relevant environmental testing would be essential. This would also serve as additional validation steps for the analytical model.

Secondly, while closed stiffener structures were determined to be more structurally efficient, an open structure tape spring design has been used in the TASSA, suggesting that it should not be completely disregarded as a design option. Further research into optimising the design of open stiffeners could

potentially unveil certain configurations which have improved performance characteristics.

A key limitation of the optimisation model is the restricted range of material options. Although there were more than 20 design options to choose from, it still presents a limited range compared to the hundreds of possible composites available. Furthermore, the stiffeners will likely be made up of multi-ply laminates like the ones used for the iROSA stiffeners. Expanding the model to incorporate multi-ply laminate would however require classical lamination theory to be incorporated, and some testing would also be necessary to find out certain material characteristics. Nevertheless, expanding the model to incorporate multi-ply laminates would allow for a more comprehensive and realistic analysis which could yield higher performing designs, particularly increasing stiffness characteristics.

The stowed power density presents another limitation to the comparison of roll-out solar arrays. The problem with this metric is that it solely focuses on the power-to-volume ratio of the stowed array, without considering the overall geometry of the system. While a high stowed power density indicates a volume efficient design, the inherent cylindrical bulky shape of the solar array can lead to integration challenges with certain satellites. For example, a traditional solar array may integrate much easier with a satellite as they usually have a similar geometry as the face of a satellite. On the other hand, a cylindrical roll-out solar array with higher stowed power density may be harder to integrate into a cuboidal satellite structure. Addressing and analysing the feasibility of this geometric limitation is important to understand if smallsats are suited to accommodate such systems.

Lastly, the feasibility of roll-out solar arrays is highly dependent on their cost-effectiveness compared to traditional solar arrays. Incorporating cost into the optimisation model as a third objective to optimise would offer valuable insight into how the design would change with this additional factor. Estimating the cost of materials and manufacturing costs is very time consuming, and would in turn would require careful analysis and market research to be able to fairly incorporate it into the model. Nevertheless, this change would especially benefit the solar cells which have lower efficiency but have significantly low costs, or the materials which cost less but have worse structural characteristics. An optimisation model with cost as the third objective could reveal that there are some design combinations which would make more sense to implement such that a high performing yet cost effective solution is achieved.

The aforementioned recommendations offer substantial ways in which the design of the roll-out solar array could be investigated further. By addressing these recommendations, a roll-out solar array can be developed with high performance, allowing it to penetrate the smallsat market and offer enhanced operations for the next generation of satellites.

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A

Geometric properties of stiffeners

This chapter outlines the geometric properties of the stiffeners that are used in the analytical model. For each stiffener, the moments of inertia and torsional constants are calculated.

STEM, Tape spring, Interlocking

The geometric properties of the STEM, tape spring, and interlocking structures all share similar formulas as they can be all set as a function of the subtended angle θ , as seen in Figure A.1. The interlocking structure is achieved by setting $\theta = \pi$. Similarly, $\theta > \pi$ can be used to achieve STEM structures

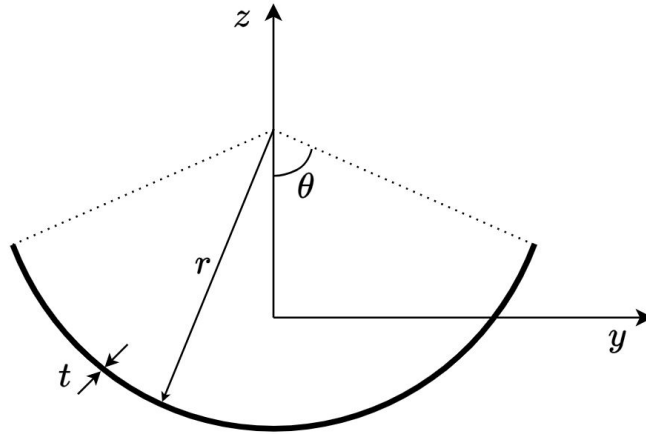


Figure A.1: Cross section of a tape spring.

The second moments of areas about the y and z axes are given by Equation A.1 and Equation A.2, respectively [74] [125].

$$I_{yy} = r^3 t \left(\theta + \frac{\sin(2\theta)}{2} - \frac{2\sin^2(\theta)}{\theta} \right) \quad (\text{A.1})$$

$$I_{zz} = r^3 t (\theta - \sin(\theta)\cos(\theta)) \quad (\text{A.2})$$

The torsional constant is calculated using the simplified formulation for a given open structure as in Equation A.3 [97].

$$K_{open} = \frac{1}{3} U t^3 \quad (\text{A.3})$$

With U representing the length of the boundary. The torsional constant for the STEM and tape spring are then calculated as follows:

$$K_{STEM, tape} = \frac{1}{3} (2 \theta r) t^3 \quad (\text{A.4})$$

For the interlocking structure, the torsional constant changes as it is assumed to be a closed structure. The formula for the torsional constant of closed structures is given by Equation A.5 [97].

$$K_{closed} = \frac{4 A^2 t}{U} \quad (\text{A.5})$$

With A representing the enclosed area of the structure. Substituting the appropriate values for an interlocking stiffener, the following torsional constant is achieved:

$$K_{inter} = \frac{4(\pi r^2)^2 t}{2\pi r} \quad (\text{A.6})$$

TRAC

The geometric parameters which make up the TRAC cross section are visualised in Figure A.2. For the following calculations, the curved sections are labelled as sections '1', while the flat part is referred to as section '2'.

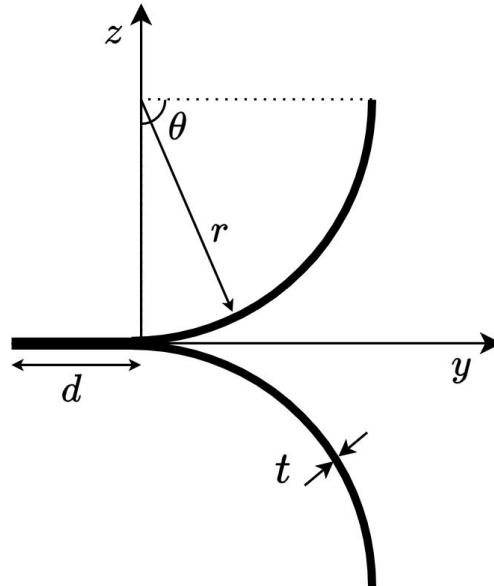


Figure A.2: TRAC stiffener cross section.

The second moment of area of the TRAC about the y axis can be decomposed by the sections of the structure:

$$I_{yy} = 2I_{yy,1} + I_{yy,2} \quad (\text{A.7})$$

The individual $I_{yy,1}$ and $I_{yy,2}$ are given by Equation A.8 and Equation A.9, respectively [126].

$$I_{yy,1} = \frac{t}{48} \begin{bmatrix} -32(t+r)(t^2+3tr+3r^2)\sin\theta \\ +6(t+2r)\{(5t^2+10tr+6r^2)\theta \\ +t^2\cos\theta\sin\theta + r(t+r)\sin 2\theta\} \end{bmatrix} \quad (\text{A.8})$$

$$I_{yy,2} = \frac{2t^3 d}{3} \quad (\text{A.9})$$

The second moment of area of the TRAC about the z axis can be found with Equation A.10. As the structure is not symmetric about the axis, the parallel axis theorem has to be used to find the second moment of area about the centroid of the whole stiffener.

$$I_{zz} = 2(I_{zz,1} + A_1 y_1^2 + I_{zz,2} + A_2 y_2^2) \quad (\text{A.10})$$

With $y_{1,2}$ representing the distance from the centroids of the sections to the global centroid. The global centroid of composite areas $\bar{y}$ can be found using the weighted average of each area as in Equation A.11.

$$\bar{y} = \frac{\sum_{i=1}^n (A_i \cdot y_i)}{\sum_{i=1}^n A_i} \quad (\text{A.11})$$

Then, the second moment of areas of the two sections about the z axis are as follows:

$$I_{zz,1} = \frac{(R+h)^2 h (2R+h) \theta}{2} - 2(R+h)(1-\cos\theta) \frac{(R+h)^3 - R^3}{3} + \left(\frac{\theta}{2} - \frac{\sin(2\theta)}{4} \right) \frac{(R+h)^4 - R^4}{4} \quad (\text{A.12})$$

$$I_{zz,2} = \frac{d^3 t}{12} \quad (\text{A.13})$$

The torsional constant is calculated using Equation A.3 as it is an open structure and results in Equation A.14.

$$K_{TRAC} = \frac{1}{3}(d + 2\theta r) t^3 \quad (\text{A.14})$$

Lenticular

The geometry of the lenticular structure with the defining parameters can be seen in Figure A.3. For the external stiffener design, the angle θ is set to $\frac{\pi}{3}$ in order to simplify the geometry and reduce the amount of inputs in the analytical model. Furthermore, this angle was found to be optimal in a paper analysing the geometry of this stiffener [108].

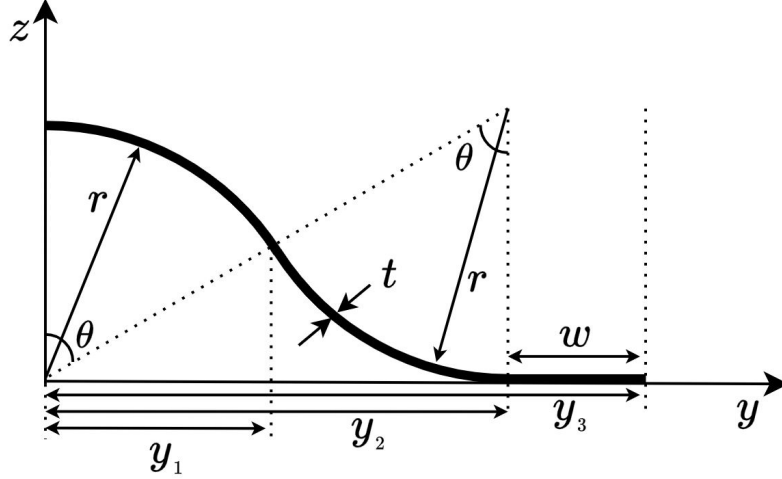


Figure A.3: Quarter cross section of the lenticular stiffener. Inspired by figure 3 in [108].

The second moment of areas about the two axes are then given by Equation A.15 and Equation A.16 [127].

$$\begin{aligned}
 I_{yy} = & 2tr \left[y_1 \sqrt{r^2 - y_1^2} + r^2 \sin^{-1} \frac{y_1}{r} \right] \\
 & + 2tr (y_2 - y_1) \left[\sqrt{r^2 - (y_2 - y_1)^2} - 4r \right] \\
 & + 6tr^3 \sin^{-1} \frac{y_2 - y_1}{r}
 \end{aligned} \tag{A.15}$$

$$\begin{aligned}
 I_{zz} = & 2tr \left(r^2 \sin^{-1} \frac{y_1}{r} - y_1 \sqrt{r^2 - y_1^2} \right) + \frac{4t}{3} (y_3^3 - y_2^3) \\
 & + 2tr \left[(3y_2 + y_1) \sqrt{r^2 - (y_2 - y_1)^2} - 4y_2 r \right] \\
 & + 2tr (2y_2^2 + r^2) \sin^{-1} \frac{y_2 - y_1}{r}
 \end{aligned} \tag{A.16}$$

As the lenticular stiffener is a closed structure, Equation A.5 can be used to find the torsional constant.

$$K_{lent} = \frac{4 \left(\frac{4ry_2}{2} \right)^2 t}{8\theta r} \tag{A.17}$$

Flat spring

The cross section of the flat spring is visualised in Figure A.4. Due to its irregular shape, the second moments of area about both axes will have to be calculated through the global centroid of the structure. The global centroid is calculated with Equation A.11. The curved sections are referred to as section '1', and the straight part is section '2'.

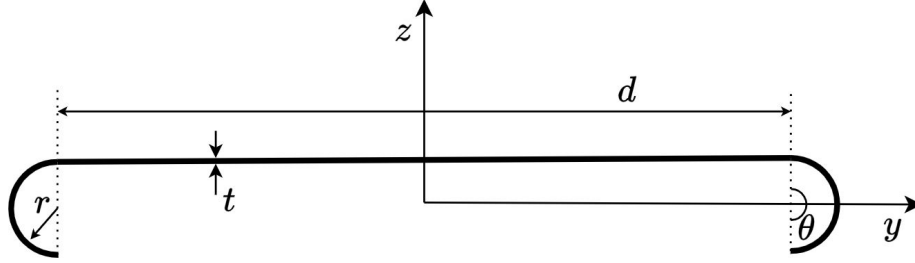


Figure A.4: Cross section of the flat spring.

The second moment of area of the stiffener about the y axis can be computed as follows:

$$I_{yy} = 2(I_{yy,1} + A_1 z_1^2) + I_{yy,2} + A_2 z_2^2 \quad (\text{A.18})$$

with

$$I_{yy,1} = \frac{t}{48} \left[-32(t+r)(t^2 + 3tr + 3r^2) \sin(\theta) + 6(t+2r) \right. \\ \left. ((5t^2 + 10tr + 6r^2)\theta + t^2 \cos(\theta) \sin(\theta) + r(t+r) \sin(2\theta)) \right] \quad (\text{A.19})$$

$$I_{yy,2} = \frac{dt^3}{12} \quad (\text{A.20})$$

Then, the second moment of area about the z axis can be computed using Equation A.21.

$$I_{zz} = 2(I_{zz,1} + A_1 y_1^2) + I_{zz,2} \quad (\text{A.21})$$

with

$$I_{yy,1} = \frac{(r+t)^2 t (2r+t) \theta}{2} \\ - \frac{2(r+t)(1 - \cos(\theta)) ((r+t)^3 - r^3)}{3} \\ + \frac{\left(\frac{\theta}{2} - \frac{\sin(2\theta)}{4} \right) ((r+t)^4 - r^4)}{4} \quad (\text{A.22})$$

$$I_{zz,2} = \frac{td^3}{12} \quad (\text{A.23})$$

The torsional constant can be calculated using the formula for open structures in Equation A.3. Substituting the parameters, the torsional constant is as follows:

$$K_{flat} = \frac{1}{3}(2\theta r + d)t^3 \quad (\text{A.24})$$