

Design and Dynamic Validation of a Membrane-Frame Solar Array Module

A collaboration between
GTM-AS & TU Delft

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by

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Cover: Artistic visualization of a deployable semi-flexible solar array in a space environment, juxtaposed with an illustration of the physical prototype tested during the experimental campaign.

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Preface

This thesis marks the final step of my Master’s degree in Aerospace Engineering at TU Delft, with a specialization in Aerospace Structures & Materials. The research presented in this thesis was conducted during the latter part of my internship at GTM Advanced Structures (GTM-AS), motivated by a desire to bridge academic theory and industrial practice. During my internship, I became fascinated by the challenges associated with large-scale flexible solar arrays. At the same time, I wanted my graduation project to have a strong practical component. In particular, I was interested in developing a physical prototype and correlating its behavior with a finite element model to gain a deeper understanding of both the possibilities and limitations of the technology. This project allowed me to pursue all of these objectives, and I am grateful for the opportunity to have worked on it.

First and foremost, I would like to sincerely thank my university supervisor, Prof. Prem Sundaramoorthy. Whenever I found myself struggling with the complexities of the project, his advice to systematically simplify the problem helped me regain focus and move forward. I would also like to thank Kees Honseelaar, CTO of GTM-AS, for making this thesis possible and for sharing his valuable industry insights throughout the project. In addition, I would like to thank my third supervisor, Henk Cruijssen, for his technical guidance, enthusiasm for flexible solar arrays, and assistance in transporting equipment between TU Delft and GTM-AS.

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I would like to express my deepest gratitude to my family, my girlfriend, my roommate, and my friends. Thank you for your unwavering support, encouragement, and patience throughout this journey, as well as for listening to countless discussions about my project.

Aleksei Esser

The Hague, August 26, 2026

Author’s Note on AI Usage: During the preparation of this thesis, the author utilized Gemini strictly for language editing, stylistic refinement, and LaTeX formatting assistance. The AI was not used to generate data, draw structural conclusions, or formulate the core theoretical concepts. The author has thoroughly reviewed all AI-assisted edits and takes full personal responsibility for the final content and integrity of this document

Summary

Driven by the increasing demand for lightweight and cost-effective spacecraft, semi-flexible solar arrays offer an attractive alternative to conventional rigid architectures. Their large photovoltaic surface area is enabled through lightweight, pre-tensioned thin-film membranes supported by a rigid structural frame. However, the low bending stiffness of these membranes makes the arrays particularly susceptible to dynamic out-of-plane deformation during launch, potentially resulting in excessive oscillations and damage to solar cells or structural interfaces. Accurate prediction of their dynamic behavior is therefore essential for structural design and qualification.

This research investigates to what extent the dynamic characteristics of a semi-flexible membrane-frame solar array module can be predicted using an as-built finite element (FE) model. To support this objective, a representative small-scale membrane-frame prototype (presented in Figure 1) was designed, manufactured, assembled and experimentally characterized. Experimental Modal Analysis (EMA) was performed using an LDS V550 electrodynamic shaker and a Polytec PSV-500 Scanning Laser Doppler Vibrometer (SLDV) (seen in Figure 2) to determine the natural frequencies and mode shapes of both an unpopulated and populated prototype configuration. The corresponding numerical model was developed in Abaqus using S4 shell elements, while element birth-and-death techniques were employed to replicate the physical manufacturing and tensioning sequence.

For the unpopulated structure (which did not carry any solar cells), the first three membrane-dominated modes were predicted within 9% absolute frequency error, while frame-dominated modes were predicted within 14%. Furthermore, a Modal Assurance Criterion (MAC) of 0.78 was achieved for the fundamental membrane mode, demonstrating good spatial agreement between the numerical and experimental mode shapes. For the populated prototype, a simplified distributed-mass approach was adopted to represent the photovoltaic layer without explicitly modeling the local stiffness contributions of the solar-cell interfaces. Using an effective mass corresponding to approximately 40% of the physical photovoltaic mass, the FE model predicted the first three membrane modes within 5% of the experimentally measured frequencies, while frame-dominated modes remained within 13%. The photovoltaic layer reduced the fundamental membrane frequencies by approximately 54–62% and altered the observed modal deformation patterns through localized stiffening effects.

The results demonstrate close agreement between numerical predictions and experimental measurements for the fundamental dynamic behavior of both configurations. However, the spatial correlation obtained for the populated prototype remained limited, indicating that the simplified distributed-mass approach captures the global frequency response more effectively than the local stiffness-driven mode-shape characteristics. Discrepancies in the experimental correlation were primarily driven by manufacturing-induced imperfections, asymmetric pretensioning, and the finite compliance of the physical boundary conditions. In particular, the finite stiffness of the supporting test rig and its overhanging mounting configuration introduced additional compliance and dynamic amplification effects that were not explicitly represented in the numerical model. These observations highlight the sensitivity of membrane-frame structures to as-built conditions and boundary-condition representation. Despite the simplifications required to model the photovoltaic layer, the developed FE framework successfully captures the global dynamic characteristics of the membrane-frame architecture. The findings demonstrate that an as-built finite element approach can provide a reliable framework for the dynamic assessment, validation, and early-stage design of future lightweight membrane-frame solar-array systems.

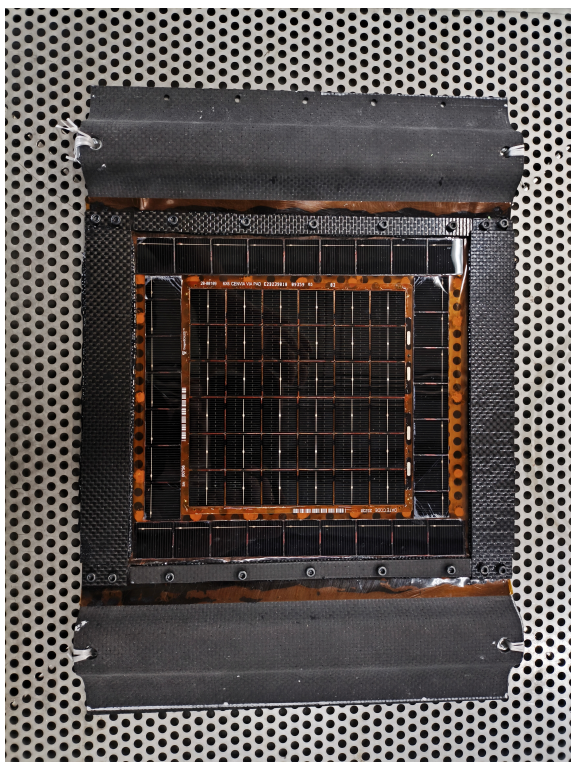


Figure 1: Assembled membrane-frame prototype.

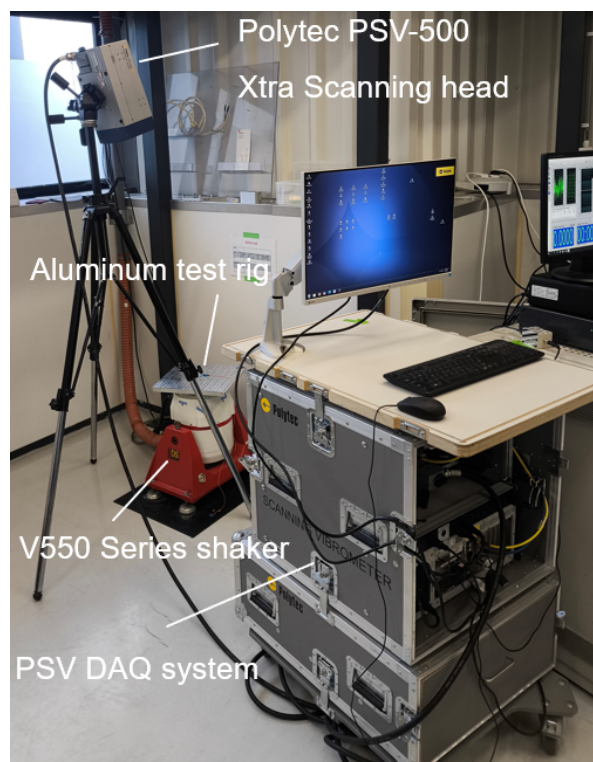


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Nomenclature

Abbreviations

Abbreviation	Definition
BOL	Beginning of Life
CAD	Computer-aided design
CFRP	Carbon Fibre Reinforced Polymer
CTE	Coefficient of Thermal Expansion
ELSA	Extensible Low-Profile Solar Array
EMA	Experimental Modal Analysis
ESPA	Evolved Expendable Launch Vehicle Secondary Payload Adapter
FE	Finite Element
FEM	Finite Element Model
FRF	Frequency Response Function
GaAs	Gallium Arsenide
GTM-AS	GTM Advanced Structures B.V.
LEO	Low Earth Orbit
LDV	Laser Doppler Vibrometry
MAC	Modal Assurance Criterion
MQ	Main Research Question
PSV	Polytec Scanning Vibrometer
PV	Photovoltaic
PVDF	Polyvinylidene Fluoride
RMS	Root Mean Square
ROSA	Roll-Out Solar Array
SLDV	Scanning Laser Doppler Vibrometer
SQ	Sub-Research Question
USD	United States Dollar

Symbols

Symbol	Definition	Unit
A	Area	[m ²]
$[\mathbf{C}]$	Global damping matrix	[Ns/m]
E	Young's modulus	[Pa]
F	Force	[N]
f_n	Natural frequency	[Hz]
G	Shear modulus	[Pa]
g	Gravitational acceleration	[m/s ²]
$H(\omega)$	Frequency response function	[(m/s ²)/N]
$[\mathbf{K}]$	Global stiffness matrix	[N/m]
k	Stiffness	[N/m]
L	Length	[m]
$[\mathbf{M}]$	Global mass matrix	[kg]
m	Mass	[kg]
N_x, N_y, N_{xy}	In-plane membrane forces per unit length	[N/m]

Symbol	Definition	Unit
P	Pretension load	[N]
t	Thickness	[m]
T	Membrane tension	[N/m]
u, v, w	Translational displacements	[m]
V	Velocity	[m/s]
w	Out-of-plane deflection	[m]
x, y, z	Cartesian coordinates	[m]
δ	Deflection	[m]
ϵ	Strain	[-]
ζ	Modal damping ratio	[-]
η	Loss factor	[-]
$\theta_x, \theta_y, \theta_z$	Rotational displacements	[rad]
λ	Eigenvalue	[-]
μ	Mass per unit area	[kg/m ²]
ν	Poisson's ratio	[-]
ρ	Density	[kg/m ³]
σ	Normal stress	[Pa]
τ	Shear stress	[Pa]
$\{\phi\}_r$	Mode shape vector for mode r	[-]
ω	Angular natural frequency	[rad/s]

Introduction

1.1. Research Background

The space industry is undergoing a paradigm shift driven by the development of reusable launch vehicles, moving from a strictly mass-dominated to a more cost-dominated market. Between 1970 and 2010, launch costs to Low Earth Orbit (LEO) were typically between 10,000 and 20,000 USD/kg (with respect to 2018 USD) [1]. The Falcon Heavy demonstrated a reduction to approximately 1,400 USD/kg [1], with further reductions anticipated as fully reusable launch systems mature [2].

Furthermore, the current state-of-the-art solar cells used in many space solar arrays are Gallium Arsenide (GaAs) due to their high efficiency, with Beginning-of-Life (BOL) efficiencies of approximately 29% [3]. Silicon solar cells offer lower efficiency, with BOL efficiencies of approximately 18% [4], but benefit from greater material availability and lower cost. As launch costs decrease and the demand for large-area space power systems increases, the use of lower-cost Silicon technology becomes increasingly attractive. However, the lower efficiency of Silicon requires a larger surface area to generate an equivalent amount of power, with approximately 1.5–2 times the area required compared with GaAs technology.

Scaling state-of-the-art rigid composite honeycomb architectures to accommodate this increased area introduces several engineering challenges. Specifically, the use of Silicon on rigid panels can result in lower specific power, lower deployed first-mode natural frequencies, and increased stowed volume [5]. Increased stowed volume can become particularly problematic for rideshare missions, where the available payload envelope is constrained. For commercial missions utilizing platforms such as Falcon 9 Rideshare, increased stack thickness can require a transition from a standard 15-inch ESPA-class slot to a larger ESPA Grande configuration, potentially increasing mission cost [6]. Larger rigid-array architectures may also require larger hold-down and release mechanisms, introducing additional parasitic mass.

Therefore, while Silicon provides an attractive path toward lower-cost space solar power generation, realizing its cost advantage at larger required surface areas requires a highly stowable structural architecture that does not introduce excessive structural or deployment overhead.

A common alternative to rigid composite honeycomb structures is the flexible solar blanket. A flexible blanket is a pliable subassembly consisting of interconnected photovoltaic cells mounted on a flexible polymer or mesh substrate [7]. It can be stowed compactly through rolling or folding and relies on a separate deployment structure to provide the required tension and deployed stiffness. A prominent example is the Roll-Out Solar Array (ROSA), a dual-sided, rectangular, dimensionally stable blanket deployed by elastic composite booms or motors [8]. While ROSA has been shown to reduce stowed volume by 70% compared with traditional rigid arrays, it provides only a 20% overall mass reduction due in part to the parasitic mass associated with its deployment and tensioning mechanisms [9]. Furthermore, these mass savings are most significant for large, high-power array configurations and may degrade for power outputs around 1 kW or lower due to non-scalable structural overhead [10].

This trade-off highlights a technological gap between conventional rigid panels and mechanism-intensive flexible blankets. A semi-rigid membrane-frame architecture provides a potential intermediate solution, combining a lightweight flexible photovoltaic substrate with a structural frame capable of maintaining

the deployed geometry without requiring a large dedicated deployment structure.

1.2. Research Motivation

A semi-flexible membrane-frame solar array uses a flexible substrate to support photovoltaic modules within a lightweight structural boundary frame. This architecture retains the compact stowage potential of flexible structures while providing a more rigid deployed configuration than a conventional blanket. A related approach is gaining industry traction through concepts such as Redwire's Extensible Low-Profile Solar Array (ELSA). ELSA utilizes a patent-pending nested frame architecture intended to reduce stowed volume by 50% compared with equivalent rigid panels [11]. Furthermore, decoupling the structural frame from the photovoltaic blanket can allow aspects of the manufacturing workflow to be performed in parallel, potentially reducing production time [11].

While this architecture offers an attractive solution to volume-constrained missions, its low-profile configuration introduces an important structural-dynamics challenge. In particular, the membrane may experience out-of-plane deformation and oscillation during launch, potentially increasing the risk of cell-to-cell contact or interference between closely stacked panels. This risk becomes increasingly relevant as the inter-panel stack height is minimized to maximize stowage efficiency.

Controlled structural pretension provides one potential method for increasing membrane stiffness and limiting out-of-plane deformation [12]. However, pretension introduces a coupled structural trade-off. While increasing the membrane pretension improves dynamic stability, it simultaneously increases the loads transmitted to the boundary frame. Consequently, this may require larger or heavier frame elements to prevent excessive deformation or buckling.

Consequently, the dynamic behavior of a membrane-frame solar array cannot be considered independently of its pretension, structural geometry, manufacturing state, boundary conditions, and attached photovoltaic hardware. These coupled effects make the accurate prediction of the dynamic characteristics of an as-built membrane-frame structure a non-trivial modeling and experimental problem.

The central objective of this research is therefore to develop and experimentally characterize a representative membrane-frame solar-array prototype and to determine to what extent its dynamic behavior can be predicted using an as-built Finite Element Model (FEM). The physical prototype provides the experimental reference required for this validation. Concurrently, the FEM provides the numerical framework needed to investigate the effects of structural design, pretension, manufacturing, and photovoltaic integration.

Experimental Modal Analysis (EMA) is performed using a PSV-500 scanning laser Doppler vibrometer in conjunction with V550 Series shakers. The non-contact measurement approach is particularly suited to the lightweight membrane structure because it minimizes the influence of measurement hardware on the dynamic response. Both the bare prototype and a fully populated configuration are investigated to characterize the changes in dynamic behavior associated with the integration of mPower Silicon Dragonscales.

The resulting research framework therefore consists of three interconnected stages:

1. **Structural design and physical realization:** development, sizing, manufacturing, and tensioning of a representative membrane-frame prototype.
2. **Experimental and numerical characterization:** development of a non-contact EMA methodology and an as-built FEM for the physical prototype.
3. **Dynamic validation and application:** correlation of the experimental and numerical modal characteristics, investigation of the sources of discrepancies, and evaluation of the effect of photovoltaic integration.

This structure ensures that the design, manufacturing, pretensioning, and experimental investigations directly support the central objective of validating the dynamic behavior of the membrane-frame architecture.

Research Contributions

This research provides the following contributions:

- **Physical membrane-frame prototype:** Design and realization of a representative membrane-frame solar-array prototype, including a repeatable manufacturing and tensioning methodology for lightweight, nested structures.
- **Non-contact experimental characterization:** Development and validation of a high-resolution Experimental Modal Analysis methodology using scanning laser Doppler vibrometry to characterize the dynamic behavior of a lightweight, tensioned membrane-frame structure under base excitation.
- **As-built finite element modeling:** Development of an Abaqus Finite Element Model representative of the manufactured and tensioned prototype, including the structural characteristics required to reproduce its fundamental dynamic behavior.
- **Experimental-numerical correlation:** Quantitative comparison of experimentally measured and numerically predicted natural frequencies and mode shapes, including identification of the principal limitations affecting the correlation between the as-built structure and the FEM.
- **Photovoltaic integration:** Experimental and numerical characterization of the change in dynamic response resulting from the integration of mPower Silicon Dragonscales onto the pre-tensioned membrane.

1.3. Research Questions

The primary objective of this research is to validate the structural dynamics of a scaled, nested, semi-rigid solar-array panel by comparing the experimentally measured behavior of a physical prototype with an as-built Finite Element Model. The design, pretensioning, manufacturing, and experimental investigations described throughout this thesis support this central validation objective.

Consequently, the Main Research Question (MQ) guiding this thesis is defined as:

MQ: *To what extent can the dynamic characteristics, specifically natural frequencies and mode shapes, of an as-built Finite Element Model be predicted in agreement with Experimental Modal Analysis of a physical membrane-frame prototype?*

To answer the main research question, the following sub-questions are divided into three interconnected areas: structural design, experimental and numerical methodology, and dynamic validation.

Structural design

SQ-1: *What is the primary mechanical risk governing the conceptualization of a robust modular membrane-frame solar-array structure?*

SQ-2: *What is the relationship between membrane pretension, maximum load factor, and out-of-plane deflection for a membrane-frame structure?*

Experimental and numerical methodology

SQ-3: *Which finite element modeling techniques provide the most representative dynamic response for the single-panel configuration without solar cells attached?*

SQ-4: *What experimental setup and measurement methodology enable Experimental Modal Analysis of a membrane-frame structure?*

SQ-5: *What experimental limitations and boundary-condition inaccuracies are encountered when performing vibrational shaker testing on a small-scale membrane-frame configuration?*

Dynamic validation and photovoltaic integration

SQ-6: *How does the dynamic response change when solar cells are attached to the single-panel structure?*

Together, these sub-questions establish a progression from structural design and pretensioning, through experimental and numerical methodology, to the final experimental-numerical correlation and evaluation of photovoltaic integration.

1.4. Scope and Limitations

The scope of this research is strictly bounded to the fundamental structural dynamics of a semi-rigid membrane-frame architecture. The physical prototype is not intended to represent a flight-ready solar-array system. Instead, it serves as a controlled experimental article designed to isolate the coupled frame-membrane dynamics and enable quantitative correlation between experimental measurements and numerical predictions.

To maintain a controlled experimental environment, the following boundaries and structural simplifications are established:

- **Environmental Loading:** This study evaluates the structural response exclusively under dynamic vibrational loads. Other mission-critical environments, including thermal-vacuum cycling, acoustic loading, radiation degradation, and zero-g deployment kinematics, are identified as relevant to a complete flight design but are excluded from the present analysis.
- **Hardware Configuration:** The physical test article is restricted to a single-panel prototype. Multi-panel stack interactions, inter-panel hinges, and associated deployment mechanisms introduce additional multi-body and potentially nonlinear dynamics that fall outside the scope of this fundamental modal correlation.
- **Geometric Scaling:** A scaled prototype is utilized to accommodate the physical mounting dimensions and mass capacity of the shaker armature. This scaling enables the use of controlled and repeatable boundary conditions during base excitation while avoiding additional fixture dynamics associated with a larger test article.
- **Frequency Bandwidth:** The nominal experimental and numerical analysis bandwidth is 0–500 Hz. However, reliable experimental correlation is restricted to the approximately 20–460 Hz range because of low-frequency coherence limitations and test-rig resonances that begin to influence the measured structural response at higher frequencies.
- **Numerical Discretization and Homogenization:** The finite element model is constrained to a 1000-node computational threshold. This limitation requires simplified representations of selected structural components and motivates the use of homogenized material representations where appropriate.

These simplifications introduce limitations on the direct applicability of the results to a full-scale flight system. Nevertheless, they are necessary to isolate the coupled membrane-frame dynamics and enable controlled Experimental Modal Analysis of the physical prototype. The resulting model and experimental methodology are therefore intended to provide insight into the dynamic behavior and modeling requirements of the architecture rather than constitute a complete flight qualification analysis.

1.5. Report Outline

This thesis is structured into three primary phases, as presented in Figure 1.1, progressing from theoretical and structural conceptualization to experimental hardware realization and finally to dynamic validation.

Chapter 2, Chapter 3 & Chapter 4 establish the theoretical and engineering framework of the research. These chapters review the relevant state-of-the-art solar-array architectures, establish the conceptual trade-off between rigid, flexible, and membrane-frame solutions, and perform the preliminary structural sizing required to define the prototype.

Chapter 5 & Chapter 6 cover the physical realization of the research article. These chapters document the detailed design and manufacturing of the membrane-frame prototype, together with the development of the specialized assembly and tensioning rig required to establish the desired pretension state.

Chapter 7 constitutes the experimental and numerical validation phase. Experimental Modal Analysis is performed on both the bare and populated structures using scanning laser Doppler vibrometry. The measured dynamic characteristics are subsequently correlated with the as-built Finite Element Model, allowing the predictive capability and limitations of the modeling approach to be evaluated.

Finally, **Chapter 8** synthesizes the findings, answers the research questions, identifies the principal

limitations of the study, and provides recommendations for future development of the membrane-frame solar-array architecture.

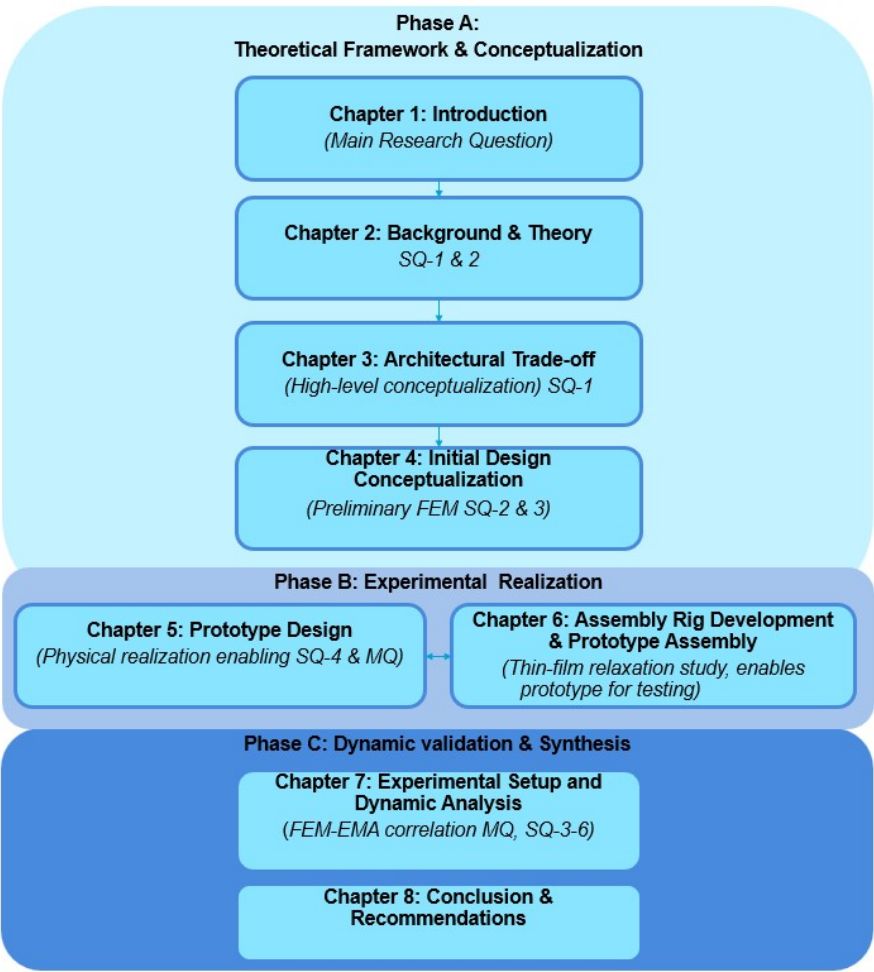


Figure 1.1: Structural outline of the thesis. Chapters 5 and 6 are depicted in parallel to reflect the iterative development of the numerical model and physical prototype.

Background and Theory

2.1. State-of-the-art Solar Array Architectures

State-of-the-art flexible solar arrays can be classified into three primary folding configurations: Z-folded, fan-shaped, and roll-out solar arrays [13]. The Z-folded flexible solar array typically consists of numerous flexible panels, which are unfolded using a large supporting truss structure and a driving mechanism for deployment [14]. Because the tension is typically introduced through an active driving mechanism, Z-folded solar arrays are able to impart high geometrical stiffness to the flexible substrate. However, due to the requirement of a large truss for achieving stiffness, this configuration is only used for large-scale spacecraft (to ensure the specific power (W/kg) remains competitive with other flexible and rigid configurations) [9], [14].

An example of a Z-folded configuration is the Advanced Photovoltaic Solar Array (APSA) (depicted in Figure 2.1), developed from 1985 to 1989 at the NASA Jet Propulsion Laboratory [15]. The flexible blanket consists of a 50 μm thick Kapton polyimide film, which is carbon-loaded to prevent electrostatic charge buildup [15], [16]. During stowage, the accordion-folded blanket is contained in a housing structure built from graphite/epoxy-faced aluminum honeycomb sandwich panels [15]. To mitigate dynamic launch loads, the inside of the lid is encased in a thick flexible polyimide foam layer (wrapped in a thin polyvinylfluoride sheet) to prevent localized damage to the solar cells [15]. This top lid is clamped to the pallet structure with braided steel cables at four different locations [15]. Structurally, the array utilizes a continuous deployable lattice mast constructed from fiberglass, which is reinforced diagonally with stainless steel braided cables [15]. The flexible substrate panel is populated with silicon solar cells [15].

To validate the survivability of this stowed configuration against the dynamic launcher environment, component-level vibration testing was conducted on an 8-panel section of the prototype blanket [15]. While load-relaxation tests confirmed that the polyimide foam was acceptable for cushioning the stowed array, the vibration tests, which simulated shuttle vibroacoustic launch environments between 10 and 30 g's (rms), revealed significant structural vulnerabilities within the flexible substrate [15]. Even prior to vibration testing, visual and infrared inspections revealed that approximately 5% of the ultra-thin cells and 10% of the coverglasses were already cracked [15]. Following a series of fifteen one-minute dynamic load cycles, an additional 3% of the solar cells and 14% of the coverglasses suffered fracture damage [15]. This extensive physical degradation empirically quantifies the severe mechanical risks associated with flexible array dynamics, highlighting the critical necessity of limiting out-of-plane deflections and localized bending stresses during launch. Furthermore, beyond the immediate launch environment, it has been established that bare carbon-loaded Kapton structurally degrades and is susceptible to tearing over extended mission lifetimes [7]. This material degradation, coupled with a relatively high coefficient of thermal expansion that induces severe on-orbit dimensional instability, renders the unreinforced film sub-optimal as a primary structural load path [7].

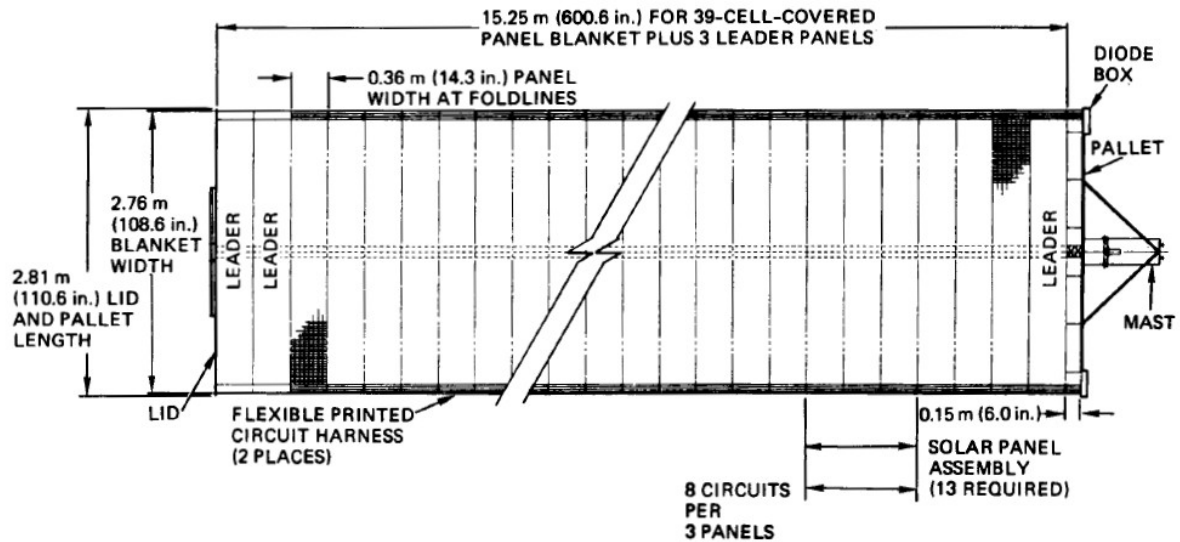
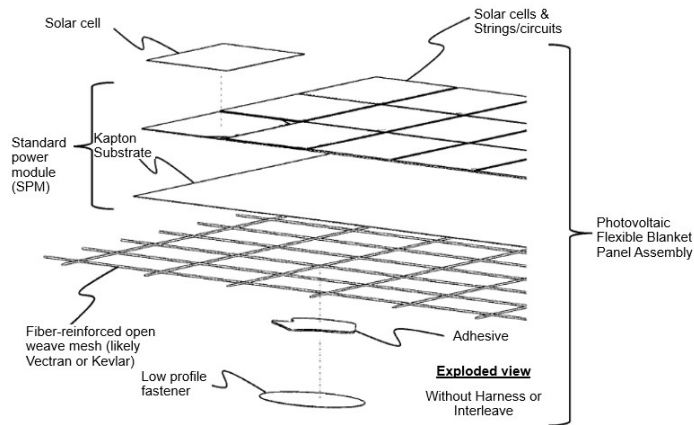


Figure 2.1: 5.3 kW (BOL) GEO Deployed Wing Configuration of the Advanced Photovoltaic Solar Array (APSA) [15].

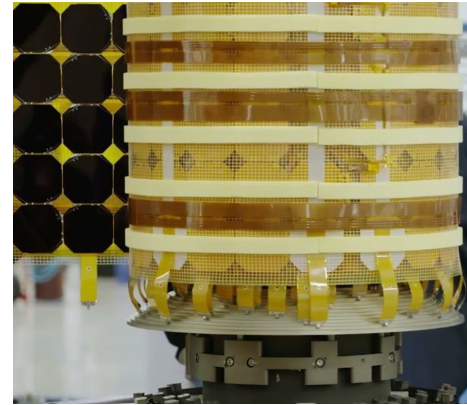
The Roll-Out Solar Array (ROSA), developed by Redwire (formerly Deployable Space Systems (DSS)), is a state-of-the-art flexible architecture that successfully utilizes a strain-energy roll-up mechanism in orbit [17], [18]. One of the earliest precursors to this concept was the Flexible Rolled-Up Solar Array (FRUSA) from Hughes Aircraft Company in 1971, which later inspired the SA2 arrays (visualized in Figure 2.3a) utilized on the Hubble Space Telescope (HST) in 1990 (a joint NASA and ESA project) [19]. The dual-wing SA2 arrays also supported supported silicon solar cells, generating approximately 5.1 kW with a specific power of 18.75 W/kg per wing [19].

While the HST utilized a tensioned, glass-reinforced Kapton blanket, its deployment relied on motor-driven metallic bi-stem booms (as depicted in Figure 2.2b). Consequently, the flexible arrays faced severe structural challenges due to thermal-elastic flutter and transient dynamic disturbances, a phenomenon commonly referred to as "thermal snap", occurring during rapid temperature fluctuations at orbital eclipse transitions [7], [20]. To overcome the dynamic instabilities and high mass penalties inherent to mechanically driven masts, the ROSA architecture was developed to deploy passively [19]. Instead of a central mast, ROSA utilizes two composite slit-tube edge booms that unroll via stored strain energy, providing perimeter tension to the flexible blanket and achieving high geometric stiffness, which led to its highly successful implementation on the International Space Station (iROSA) [21].

The ROSA architecture utilizes a modular, lightweight photovoltaic flexible blanket assembly; an exploded view of this assembly is presented in Figure 2.2a, and a high-resolution screenshot of the rolled array is shown in Figure 2.2b. The flexible blanket assembly consists of a fiber-reinforced open weave mesh (likely glass-fiber, Vectran or Kevlar) that acts as a dimensionally stable and stiff planar tensile surface. Discrete, low-profile fasteners are distributed across this mesh and bonded with an adhesive to a dielectric polyimide Kapton substrate layer. The photovoltaic cell strings are subsequently mounted to this Kapton layer [7].



(a) Exploded view of the modular, lightweight photovoltaic blanket assembly utilized on the ROSA [7]



(b) High-resolution screenshot detailing the rolled packaging of the 63 kW ROSA by Redwire [22]

Figure 2.2: Roll-Out Solar Array (ROSA) architecture showing (a) the exploded modular blanket assembly and (b) the stowed flight configuration on the deployment mandrel.



(a) The ESA/B Ae flexible solar array utilized on the Hubble Space Telescope, relying on central bi-stem masts for tensioning [23]



(b) The modern Roll-Out Solar Array (ROSA), utilizing perimeter tensioning via composite slit-tube edge booms [17]

Figure 2.3: Evolution of flexible roll-out solar array architectures, contrasting historical central-mast deployment (a) with state-of-the-art perimeter tensioning (b).

2.2. Launch Environment and Dynamic Loading

The dynamic launch environment is imposed on the payload by the launch vehicle's propulsion and aerodynamic forces, which propagate through the primary rocket structure [24]. Each launch vehicle introduces a unique dynamic environment. Consequently, the rocket structure, adapter, and payload must be designed to ensure that detrimental resonance and critical frequency coupling with the launch vehicle are avoided, thereby mitigating the risk of high dynamic amplification or structural failure [25]. The physical parameters governing this dynamic response can be fundamentally understood by approximating the array as a single-degree-of-freedom (SDOF) spring-mass-damper system, governed

by the equation of motion:

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (2.1)$$

where m is the modal mass of the object [kg], c is the viscous damping coefficient [N·s/m], k is the structural stiffness [N/m], and $F(t)$ is the external time-dependent applied force [N].

By assuming an undamped ($c = 0$), free vibration ($F(t) = 0$) condition where the displacement response takes a standard harmonic form of $x(t) = A \sin(\omega_n t)$, the equation simplifies to yield the circular natural frequency $\omega_n = \sqrt{k/m}$. Converting this to the ordinary natural frequency (f_n) gives $f_n = \frac{1}{2\pi} \sqrt{k/m}$. This mathematical derivation establishes the fundamental proportional relationship governing structural dynamics:

$$f_n \propto \sqrt{\frac{k}{m}} \quad (2.2)$$

This demonstrates that to actively tune the fundamental frequency of the solar array and avoid primary launch resonances, the ratio between the structural stiffness and the mass must be carefully controlled during the design phase [25].

While the SDOF model provides the fundamental mathematical basis for frequency tuning, the actual external forcing function, $F(t)$, experienced during ascent is rarely a simple, single-frequency harmonic. A launch vehicle subjects the payload to a highly complex, multi-modal dynamic environment. Events such as engine ignition, maximum aerodynamic pressure, transonic buffeting, and pyrotechnic stage separations each introduce distinct mechanical loads across a wide spectrum of frequencies.

To ensure that payloads can survive these extreme conditions without structural failure, launch providers dictate strict environmental bounding limits. For modern commercial rideshare missions, the *SpaceX Falcon 9 Payload User's Guide* explicitly defines the maximum expected mechanical environments that the payload adapter and spacecraft must withstand. Because spacecraft are continuous structures, different components will resonate and fail under different types of excitation.

Consequently, the structural sizing (the required mass and stiffness) of a spacecraft is driven by categorizing these dynamic loads based on their frequency domains and physical characteristics. As outlined in Table 2.1, massive primary bus structures are typically sized by low-frequency Quasi-Static Loads (QSL) and sine vibrations. Conversely, components with large, lightweight surface areas, such as stowed flexible solar arrays, are exceptionally susceptible to mid-to-high frequency random vibrations and vibro-acoustic noise, often requiring specialized damping and tensioning strategies to survive (such as polyimide foam layers utilized in ROSA [7]). However, Table 2.1 serves as a general guideline; secondary loads must still be evaluated, as the governing worst-case load condition is ultimately determined through comprehensive structural analysis [25].

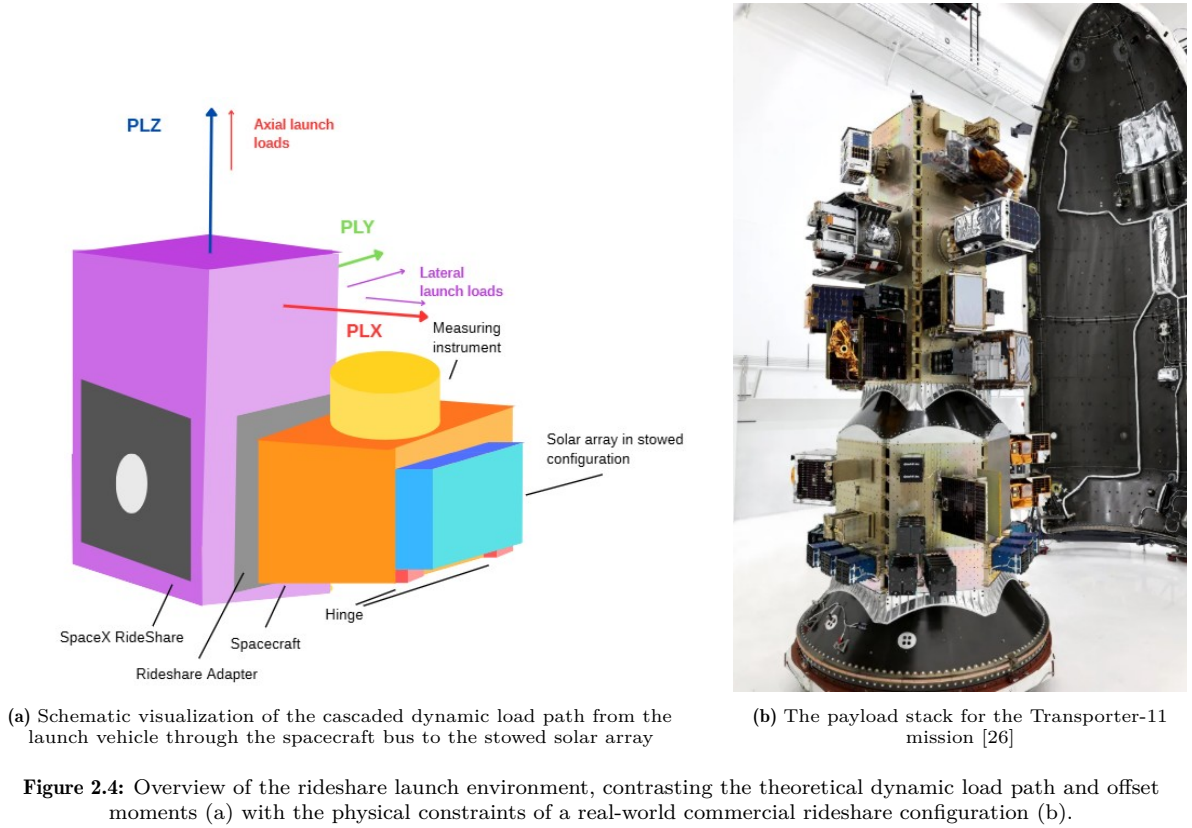
Table 2.1: Launch load types and their typical sizing domains [25].

Load type	Typically sizing for...
QSL	Primary structures (of medium/large Spacecraft).
Sine (up to 100Hz)	Primary structures, secondary structures, big payloads.
Vibro-acoustic (up to 2000-3000Hz)	Elements with large surfaces or comparatively thin structures such as solar arrays, antenna reflectors, etc.
Random (up to 2000Hz)	Small satellites, payloads, instruments, units.
Shock (up to 10000Hz)	Mechanisms, electronics, optics (high susceptibility, though rarely primary sizing).

Furthermore, it is critical to recognize that satellite boundary conditions are defined strictly at the primary launch vehicle interface. For example, small satellites and CubeSats often utilize a rideshare

adapter (as seen in Figure 2.4b) to allow dozens of payloads to be launched simultaneously [26]. On SpaceX rideshare missions, two standard payload attach fittings (15-inch and 24-inch diameter rings) are typically provided [6], through which dynamic launch loads are introduced into the spacecraft.

Because these satellites are mounted outboard on the adapter, a lateral offset moment (lever-arm) often arises between the primary load path and the satellite, dynamically amplifying the accelerations experienced at the interface. Similarly, because solar arrays are structural appendages mounted to the exterior of the spacecraft bus, they experience a secondary cascading amplification of these loads. The magnitude of this final amplification is heavily dependent on the orientation of the array and the specific structural dimensions of the host satellite (as depicted in Figure 2.4a). Due to this geometric dependency, baseline vibration profiles provided in launch vehicle user's guides generally serve as preliminary design envelopes rather than final qualification limits [6]. Consequently, a spacecraft-specific coupled loads analysis and a tailored component-level or full-system vibration test campaign are typically required to fully verify the structural integrity of the stowed solar array [25].



2.3. Structural Mechanics of Pre-tensioned Membranes

The paper by E.V. Morozov & A.V. Lopatin discusses the non-linear analytical formulation for the out-of-plane deformation of a membrane stretched across a spacecraft solar array frame [12]. The authors derive an analytical equation for the maximum central out-of-plane displacement for both isotropic and orthotropic membranes by applying the Galerkin method to a generalization of Föppl's equations [12]. The full computational implementation is provided in Appendix B in the form of a Python script.

Assume a single membrane panel has dimensions $(a \times b)$, where the aspect ratio is restricted to $c = \frac{a}{b} < 2$ to limit shape mismatch errors. Treating the material as a continuous, unreinforced isotropic film, the governing equations rely on the Young's modulus (E), Poisson's ratio (ν), and thickness (h). The load experienced by the blanket is expressed as a transverse pressure $p = B_\rho n_z g$, where B_ρ is the mass per unit area of the membrane material (including the solar cells) $[\text{kg}/\text{m}^2]$, n_z is the load factor, and $g = 9.81 \text{ m/s}^2$ is the gravitational acceleration [12]. The resulting deformation can be mitigated through the application of in-plane pre-tension forces T_x and T_y $[\text{N}/\text{m}]$.

Assuming the boundaries of the membrane are fixed (i.e., $u = v = w = 0$), the Galerkin method is applied. The approximating function for the out-of-plane deformation of the membrane is given by:

$$w(x, y) = W \sin\left(\frac{\pi x}{a}\right) \sin\left(\frac{\pi y}{b}\right) \quad (2.3)$$

where W is an unknown coefficient representing the central deflection. By resolving this, the non-linear deformation of the isotropic membrane under equal pretension ($T_x = T_y = T$) is governed by a cubic algebraic equation that balances material stiffness, in-plane tension, and transverse pressure [12]:

$$\frac{\pi^4}{a^4} E h r W^3 + 16 \frac{\pi^2}{a^2} T W - p \frac{128}{\pi^2} = 0 \quad (2.4)$$

where W is the maximum out-of-plane deflection at the center of the membrane [m], a is the side length of the square membrane cell [m], E is the Young's modulus of the film [Pa], h is the membrane thickness [m], T is the uniform in-plane stretching force [N/m], and p is the transverse applied pressure [Pa] [12]. The dimensionless material constant r is derived purely from the Poisson's ratio (ν) of the isotropic material, defined as [12]:

$$r = \frac{1}{1 - \nu^2} \left[1 + 2\nu + \frac{64(25 - 9\nu^2)}{64(1 + \nu) + 9\pi^2(9 - \nu)} \right] \quad (2.5)$$

To isolate the central deflection W , Equation 2.4 is mathematically reduced to the standard cubic form [12]:

$$W^3 + S W - R = 0 \quad (2.6)$$

where the geometric and loading coefficients S and R are substituted as:

$$S = 16 \frac{a^2}{\pi^2} \frac{T}{E h r}, \quad R = \frac{128 p a^4}{\pi^6 E h r} \quad (2.7)$$

Because the discriminant of this system, defined as $Q = (S/3)^3 + (R/2)^2$, is strictly greater than zero ($Q > 0$), the polynomial yields one real root and two complex conjugate roots. Utilizing Cardano's formula, the explicit analytical solution for the real central deflection W is obtained [12]:

$$W = \sqrt[3]{\frac{R}{2} + \sqrt{Q}} - \sqrt[3]{\sqrt{Q} - \frac{R}{2}} \quad (2.8)$$

Using Equation 2.8, the structural compliance of the membrane can be directly predicted and iteratively tuned prior to finite element discretization.

The relationship between equal pretension ($T_x = T_y = T$) and the resulting out-of-plane deflection is presented in Figure 2.5. Notably, higher load factors (n_z) produce significantly larger out-of-plane deflections. Furthermore, the deflection response to increasing tension varies depending on the load factor; specifically, at lower values of n_z , the initial reduction in maximum deflection occurs much more rapidly than at higher load factors.

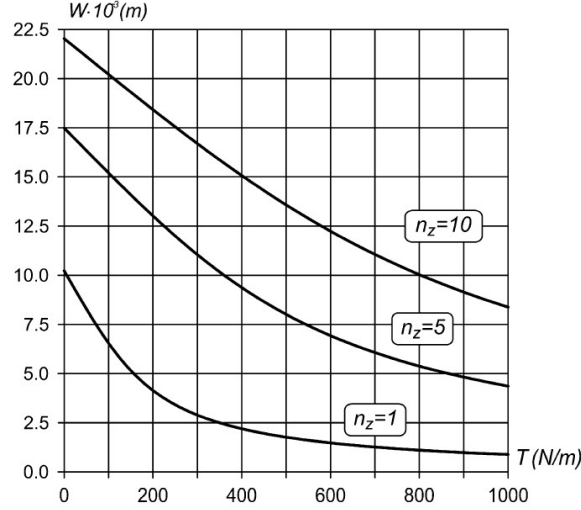


Figure 2.5: Function $W(T)$ for an isotropic membrane [12].

2.4. Non-contact Experimental Modal Analysis (EMA) for Thin-film Structures

2.4.1. Considerations for Thin-film Modal Analysis

In traditional experimental modal analysis (EMA) of rigid structures, piezoelectric accelerometers are typically attached to the test object to measure its dynamic response [27]. However, when evaluating fragile or ultra-lightweight structures, any physically attached sensor introduces localized mass and stiffness. This phenomenon, known as the mass-loading effect, fundamentally alters the dynamic properties of the structure, artificially shifting its natural frequencies and distorting its true mode shapes [28], [29]. To prevent this structural interference, the use of non-contact measurement techniques, such as Scanning Laser Doppler Vibrometry (SLDV) or Digital Image Correlation (DIC), is highly incentivized [27], [29].

A Laser Doppler Vibrometer (LDV) is a non-contact optical instrument that measures surface vibration by directing a focused laser beam onto a target and extracting frequency and amplitude data from the Doppler shift of the reflected light. In the case of transparent or highly absorptive surfaces, white or retro-reflective dots are added to ensure a sufficient signal return [29]. By scanning a predefined grid of points, the full spatial structural response can be captured. LDV can be performed in 1D, utilizing a single scanning head to measure the out-of-plane response, or in 3D using multiple scanning heads to capture the complete spatial displacement vector [30].

Digital Image Correlation (DIC) is an alternative non-contact EMA technique that utilizes image processing algorithms to track surface displacements from high-speed camera footage, which can then be differentiated to determine accelerations [31]. A significant advantage of DIC is its ability to measure the full 2D surface field simultaneously. However, DIC requires the application of a stochastic speckle pattern across the entire measurement area. This adds measurable mass and localized stiffness to the film, which is a critical drawback when compared to the minimal mass of a few retro-reflective dots used in LDV [32].

2.4.2. Excitation Methods and Signals

To perform experimental modal analysis, a controlled excitation source is required to drive the structure into resonance. For thin-film membranes, this excitation is typically introduced using either a modal impact hammer, an electromagnetic shaker, or an acoustic speaker [29], [33].

The modal hammer is a portable and relatively inexpensive tool used to approximate a perfect Dirac impulse ($t \approx 0$). Theoretically, an ideal impulse excites all structural modes with equal energy by inducing a constant amplitude across the frequency domain. The hammer is equipped with a force transducer (load cell) at its tip to measure the exact input force introduced into the structure. However,

manually exciting a highly flexible membrane with an impact hammer poses significant challenges; human error often leads to inconsistent input forces, off-axis strikes, or inadvertent "double-hits." To mitigate this input variability, an automated, mechanically triggered modal hammer can be utilized to ensure highly repeatable strikes [29], [34].

An electromagnetic shaker utilizes the same principle as a standard audio speaker, namely the Lorentz force. There are both small, portable modal shakers, which typically introduce a low force input, and larger industrial shakers capable of much higher force inputs, designed to simulate severe dynamic environments. An example of the latter is the LDS V550 series vibrator, which features a magnetic structure that houses and supports the armature and field coils [35]. To prevent overheating during continuous testing, these coils are air-cooled using an externally located fan [35]. While shakers require significantly more setup time and capital investment than impact hammers [36], they provide highly controlled continuous excitation. Furthermore, they allow for the precise determination of the load direction while delivering a substantially higher level of input energy [36].

In cases where the excited structure possesses low or zero bending stiffness, such as thin-film membranes, vibro-acoustic excitation through a speaker is also a highly viable alternative [33]. By connecting a speaker to an amplifier and a signal generator, the structure can be driven into resonance via acoustic pressure waves in the surrounding air. Unlike a shaker or an impact hammer, the acoustic force is not localized to a single discrete point; rather, it acts as a distributed pressure field over the entire surface of the test subject. However, because a 0.05 mm thick membrane is ultra-lightweight, the surrounding air effectively acts as an added virtual mass coupled to the structure [29]. In turn, this fluid-structure interaction introduces aerodynamic damping that can artificially lower the measured natural frequencies compared to the membrane's true dynamic behavior in a vacuum environment.

2.4.3. Signal Processing and Data Acquisition

When utilizing a Scanning Laser Doppler Vibrometer (SLDV), it is critical to account for the aliasing effect [30]. Aliasing is a distortion or error that occurs when a continuous signal is sampled at an insufficient rate. If the sampling frequency is too low, high-frequency signal components cannot be correctly translated and instead appear as false, lower-frequency components in the spectrum [30]. To prevent this distortion, it is essential to satisfy the Nyquist-Shannon sampling theorem:

$$f_{sample} \geq 2f_{signal} \quad (2.9)$$

Once the data is sampled correctly, the dynamic characteristics of the structure can be evaluated using the Frequency Response Function (FRF). The FRF is a frequency-domain function that defines the mathematical relationship between the input excitation force and the resulting structural output for a linear time-invariant system [37]. Across a range of frequencies, resonance peaks manifest in the FRF magnitude, directly indicating the natural frequencies of the structure [37]. Because real-world measurements inevitably contain noise, mathematical estimators are employed to calculate the FRF. The H_1 estimator is typically preferred when noise is predominantly present on the output measurement (such as LDV velocity noise), whereas the H_2 estimator assumes noise is concentrated on the input force.

An essential metric for assessing the quality and reliability of these measurements is the coherence function. Coherence yields a real value bounded between 0 and 1 across the frequency spectrum. A value approaching unity indicates a clean, highly linear relationship between the input and output with minimal noise contamination. Conversely, a drop in coherence signifies measurement errors, which can stem from background noise, spectral leakage, or unmodeled non-linearities. For thin-film membranes, broadband drops in coherence frequently occur near low-stiffness boundaries or drive points due to local non-linear responses [29], making rigorous coherence monitoring essential for reliable modal parameter extraction.

Architectural Trade-off

In Chapter 3 the trade-off of the straw-man membrane-frame solar array architecture is presented. The straw-man architecture is defined as the initial design philosophy which is deemed most prominent. First, in Section 3.1 the membrane-frame solar array architecture is dissected, for which a dissection of four existing architectures is elaborately discussed in Appendix E. Next, in Section 3.2 the architectural trade-off is performed on the different straw-man architectures, derived from existing architecture.

3.1. Component Definition

The semi-flexible (or membrane-frame) solar array architecture is a hybrid concept that combines the reliable deployment kinematics of traditional rigid arrays with ultra-lightweight flexible thin-film substrates. Based on the literature search detailed in Appendix A, only four existing concepts are found to fit the definition of a membrane-frame solar array: Redwire’s ELSA, MBB’s historical ULP, Airbus’s Eurostar Neo semi-rigid array, and JAXA’s ultra-lightweight prototype. A comprehensive technical expansion and historical dissection of these specific architectures is provided in Appendix E. To systematically evaluate these designs, a hierarchical component-level dissection was established, as detailed in Table 3.1 and visualized in Figure 3.1. This systematic breakdown provides the baseline framework for the architectural trade-off.

The functional role of the components within each tier is discussed below, alongside their most critical design drivers and associated structural risks. To provide engineering context, these critical drivers have been translated into a preliminary mission requirements document, provided in Appendix D. It should be noted that this list serves as a representative baseline to identify the primary physical boundaries governing membrane-frame solar arrays, keeping the focus strictly on structural dynamics rather than full mission validation.

The primary structure consists of all load-bearing elements essential for maintaining global and local structural stability in both the stowed and deployed configurations. In the stowed configuration, a minimum fundamental frequency is strictly defined to prevent dynamic coupling with the primary launch vehicle structure (as detailed in Appendix D, PER-REQ-003) [11]. Furthermore, to prevent dynamic coupling with the spacecraft’s Attitude and Orbit Control System (AOCS) during orbital maneuvers, a minimum deployed first eigenfrequency is also required (PER-REQ-004) [11]. Expanding upon the fundamental dynamic proportionality established in Equation 2.2 ($f_n \propto \sqrt{k/m}$), this deployed eigenfrequency can be approximated using a cantilever beam model. Assuming the hinges and longerons form the structural span, the frequency is governed by the array length ($f_1 \propto L^{-2}$), the material’s specific stiffness ($f_1 \propto \sqrt{E/\rho}$), and the second moment of area ($f_1 \propto \sqrt{I}$) of the longerons:

$$f_1 = \frac{1}{2\pi} \cdot \frac{3.5156}{L^2} \sqrt{\frac{EI}{\rho}} \quad (3.1)$$

Furthermore, the primary structure must be designed to withstand continuous compressive loads introduced by the membrane’s pretension. Finally, the material selection for the frame elements must

account for the Low Earth Orbit (LEO) thermal cycling environment to minimize Coefficient of Thermal Expansion (CTE) mismatches, which could otherwise induce dimensional instability and cause fluctuations in the baseline pretension (as defined by ENV-REQ-001 and ENV-REQ-002).

The external interfaces define the mechanical boundary conditions of the solar array system (captured by INT-REQ-001 through 006 in Appendix D). Because this research focuses primarily on the structural and dynamic response of a single panel, the deployment hinges and bus attachments are treated as functional black boxes with predefined stiffness characteristics. Conversely, the Hold-Down and Release Mechanism (HDRM) requires strict definition, as it provides the primary structural load path between the stowed panel stack and the spacecraft bus during launch. By establishing precise preload constraints, the HDRM mitigates relative dynamic displacements and inter-panel gapping during ascent.

The membrane interface consists of the structural thin-film substrate and the discrete mechanical attachments (such as constant-force springs or hooks) that secure it to the frame. Because a highly flexible membrane possesses negligible bending stiffness under compression, a strict pretension must be applied to establish geometric stiffness [12]. This pretension is governed by a tightly bounded design envelope: if the tension is too low, the membrane will suffer from excessive out-of-plane dynamic deflections during launch, potentially damaging the payload. Conversely, if the pretension is too high, the substrate becomes highly susceptible to severe random or vibro-acoustic coupling (resonance), potentially inducing excessive stress on the mechanical attachments (as defined in MECH-REQ-003 and MECH-REQ-004 in Appendix D) [6]. This also places unnecessary compressive strain on the primary structure, which could induce frame bending and paradoxically relieve the membrane pretension. The attachments themselves serve to maintain this delicate tension balance while structurally decoupling the membrane from the thermal-elastic deformations of the rigid frame.

The power system consists of the Solar Power Modules (SPMs), which comprise the photovoltaic cells, cover glasses, and electrical interconnects. For this research, mPower DragonScales modules are utilized as the baseline payload. Because brittle components (such as cell cover glasses, silicon wafers, and electrical solder joints) are highly sensitive to localized bending stresses and thermal-elastic strains, they must not act as a primary load path. Therefore, it is critical that the power system remains structurally decoupled from the global frame dynamics, riding passively on the tensioned mechanical membrane [7].

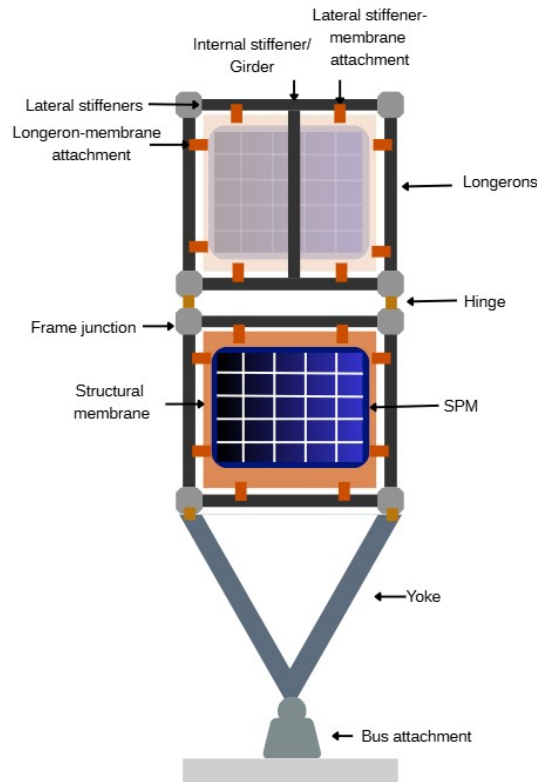


Figure 3.1: Membrane-frame solar array component nomenclature.

Table 3.1: Systematic Dissection: Membrane-Frame Structural Components.

Tier	Component	Definition
1. Primary Structure	Frame Elements	Rigid slender members; longitudinal "longerons" and lateral "cross-members" providing global stiffness.
	Frame Junctions	Corner nodes coupling frame elements; serves as the primary load transfer point between frames.
	Yoke	Transition structure connecting the inboard panel to the spacecraft bus interface.
2. External Interfaces	HDRM (Hold-Downs)	Mechanisms providing the primary load path during launch; constrains the stowed stack against the bus.
	Deployment Interface	The "black box" kinematic link (hinges/actuators) that defines inter-panel connectivity and stowed/deployed constraints.
	Bus Attachment	The mechanical interface to the satellite body; serves as the fixed boundary condition for the deployed system.
3. Membrane Interface	Membrane Attachments	Clips, springs, or tensioning elements that decouple the membrane from frame thermal/elastic deformations.
	Structural Membrane	The thin-film substrate (e.g., Kapton/GF) or net mesh providing the tensioned surface for solar cell integration.
4. Power System	SPM (Solar Power Module)	The active layer consisting of the solar cells, cover glass, and electrical interconnects.

3.2. Straw-man Trade-off

In Section 3.2, the qualitative trade-off is presented between the different straw-man concepts. The aim of this trade-off is to obtain an initial design that best adheres to the specification sheet of the to-be-designed membrane-frame solar array, presented in Appendix D. To achieve an objective comparison, the design features of ELSA, ULP, Airbus Neo, and JAXA's ultra-lightweight solar panel are transferred to the same template, which is described in Section 3.2.1. This is followed by Section 3.2.2, where the straw-man concepts are presented. Next, Section 3.2.3 presents the qualitative criteria and their corresponding sub-criteria. Finally, in Section 3.2.4, the final trade-off result is presented.

3.2.1. Straw-man Template

Each existing membrane-frame solar array described in Appendix E has different features that originate from their own unique set of requirements. Similarly, the to-be-designed membrane-frame solar array has its own required specifications as presented in Appendix D. To ensure that the concepts are comparable, existing concepts are transformed to fit the same-size template, presented in Section 3.2.1.

The most dominant feature of the template is ensuring each straw-man concept utilizes the same SPMs, resulting in identical performance. For the template, silicon mPower DragonScales modules are utilized, targeting an end-of-life power of $P_{EOL} = 1000$ W across a 3-panel array. A conservative assumption can be made that for a silicon solar array in LEO, the power production decreases by 3.75% per year [38]. In turn, for the specified 5-year mission in the requirements, this gives a lifetime degradation of:

$$L_d = (1 - \text{degradation/year})^{\text{duration}} = (1 - 0.0375)^5 = 0.82 \quad (3.2)$$

This means that the EOL efficiency of the mPower silicon cells will drop from 18.8% to 15.42% [4]. Assuming a packing factor $PF = 0.85$, the total required area is:

$$A_{array} = \frac{P_{EOL}}{\sigma \cdot PF \cdot \eta_{EOL}} = \frac{1000}{1361 \cdot 0.85 \cdot 0.1542} = 5.606 \text{ m}^2 \quad (3.3)$$

Distributed across three panels, the required area per panel is 1.87 m^2 . Constrained by a maximum allowable panel width of $W_{\text{panel}} = 1.5 \text{ m}$, the required structural length becomes $L_{\text{panel}} = 1.25 \text{ m}$. This dimensional envelope encompasses the entire structural frame, not just the active cell area. To isolate the panel architecture during this trade-off, the following boundary assumptions are applied:

- The yoke is treated as a black box and is therefore excluded from the trade-off.
- The hinges are treated as black boxes and are therefore excluded from the trade-off.
- Deployment mechanisms are not considered in this trade-off template and are therefore excluded.
- All frame elements are manufactured from the same CFRP laminate, making the trade-off independent of the frame material properties.

3.2.2. Straw-man Candidates

In Section 3.2.2, the straw-man candidates are presented, obtained by adjusting the existing membrane-frame solar array concepts into the template described in Section 3.2.1. The straw-man concepts are presented in Figure 3.2 where ELSA, ULP, Airbus Neo, and JAXA's ultra-lightweight panel are presented (from left to right). Additionally, the cross-sectional stowed configuration is presented below each illustration. The most notable features of each design are presented in Table 3.2.

As seen in Figure 3.2 and Table 3.2, an important distinction can be made that a minimum frame design is used by ELSA and JAXA's ultra-lightweight panel, while the ULP and Airbus Neo also have additional internal stiffeners. The minimum frame design is advantageous because of its lower mass and a low manufacturing complexity, which will lead to faster lead times and much lower costs. On the other hand, the big disadvantage is that in case the structure is in resonance with the launcher, the maximum out-of-plane deflection will be significantly larger if there are no internal stiffeners constraining the blanket's displacement. To limit the displacement, the pretension needs to be significantly increased, which will lead to a larger compressive load onto the frame. This compressive load acting on the

Table 3.2: Straw-man Candidate Structural Characteristics.

Characteristic	ELSA	ULP	Airbus Neo	JAXA
Stiffeners per panel	4	7	6	4
Profile type	Longerons: Hat Lateral: Rectangular	Channel stiffeners	Hat stiffeners	Square stiffeners
Membrane type	Net mesh	Sheet	Sheet	Sheet
Frame-membrane attachment	Hook/tension springs	Hook/tension springs, locally supported by internal stiffeners	Frame bonded directly to membrane	Hook/tension springs
Stowage type	Nested	Stacked	Stacked	Stacked
Original Scaling Factors	Length: 1.25 Width: 1.0	Length: 1.09 Width: 0.47	Length: 0.55 Width: 0.46	Length: 2.27 Width: 2.14

frame will result in the bending of the beams, which could result in a non-uniform tensile force acting on the blanket, potentially leading to a higher maximum displacement and wrinkling of the blanket. Additionally, the maximum allowed pretension is also determined by the buckling limit of the whole frame of the solar array [12]. To still ensure structural stability, ELSA utilizes both a wide and thick longeron with a high specific stiffness (EI) together with the utilization of a low CTE mesh net membrane. Because the net is most likely made of unidirectional fibers, the stiffness of the net is also high, allowing for structural stability.

JAXA's ultra-lightweight panel is considered an even more modular and minimal frame design than ELSA, due to the repetition of the same profiles in both longitudinal and lateral directions. However, as presented in Table 3.2, it is the only panel that required its dimensions to be increased by more than twice in its dimensions. The relation between the eigenfrequency of a 1D rope is presented in Equation 3.4 and can be used to interpret the relation between the length and pretension. Namely, if the length increases by a factor of 2, the pretension needs to increase by a factor of 4 to maintain the same first eigenfrequency ($f_1 \propto \frac{\sqrt{T}}{L}$). This could cause problems for the frame because it would induce a too high compressive force onto a frame designed for a significantly lower pretension.

$$f_1 = \frac{1}{2L} \cdot \sqrt{\frac{T}{\rho}} \quad (3.4)$$

Scaling down a concept to fit within the template does not result in the same issues as scaling up, assuming the initial design met its stiffness requirements. The ULP and Airbus Neo concepts require scaling down and would rather be affected by being over-designed in terms of the stiffeners count required for an efficient solution. For example, the ULP needs to be reduced (approximately) by a factor of 2 only in width. Originally, the ULP incorporated seven internal stiffeners. When transferred directly to the current template, this results in an over-designed configuration, with stiffener spacing reduced to only 16.7 cm (compared to the original 36 cm). Therefore, four internal stiffeners are removed, allowing for a spacing of 37.5 cm (which is much closer to the original 36 cm spacing). The Airbus Neo does not require similar alterations to fit the straw-man template because both its length and width scale down by approximately the same amount.

Another distinction can be made between the stowage type utilized by ELSA compared to the other concepts. ELSA has a nested frame design, which means that the longerons are stowed such that they can be stacked compactly onto each other. The other frames are stacked without nesting, which results in a significantly lower stowage efficiency. Compact stowage is beneficial because a smaller offset bending moment will be experienced on the spacecraft during launch. The down-side of a nested frame design is that there is a risk that if the preload of the hold-down and release mechanisms is too large, the frames become stuck. Also, the only logical location for the hold-down points has to be at or very close to the hinge locations. The freedom in determining the hold-down locations increases significantly

in the case of the Airbus Neo, which can be altered such that a hold-down point could even be added in the middle of the panel.

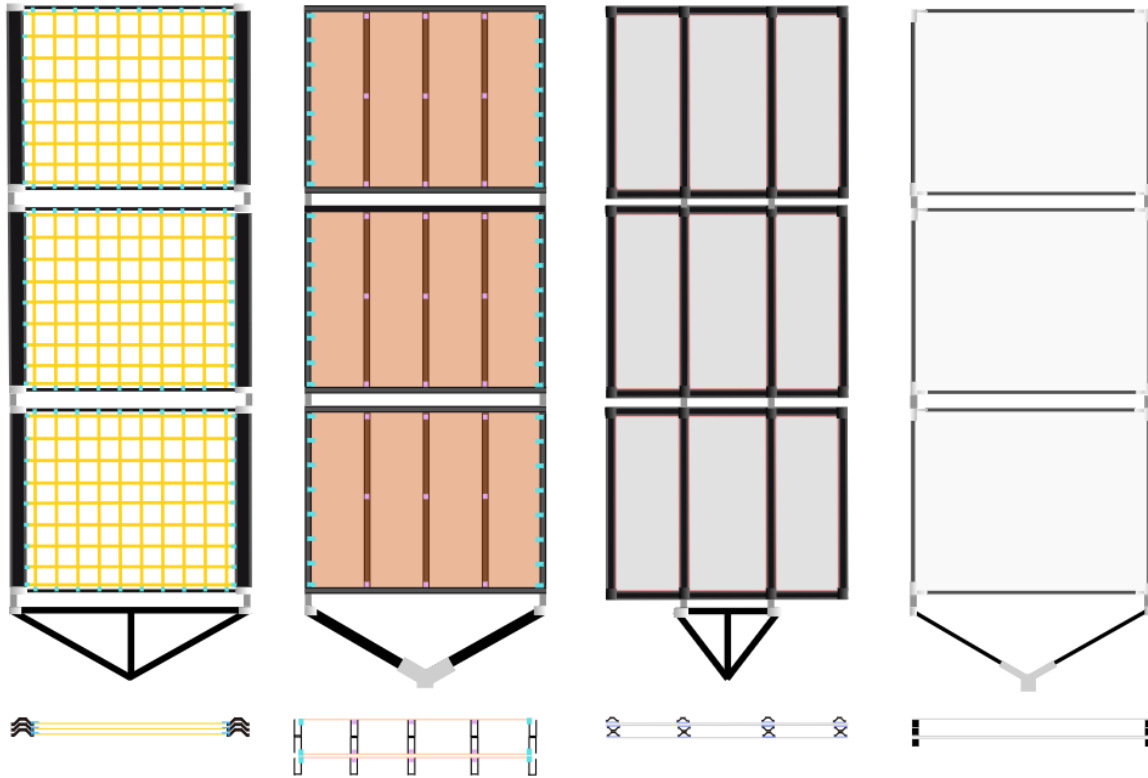


Figure 3.2: Compared straw-man concepts derived from ELSA, ULP, Airbus Neo and JAXA's Ultra-lightweight panel (left to right).

3.2.3. Trade-off Criteria

In Section 3.2.3, the chosen trade-off criteria are defined and presented. The trade-off criteria chosen are top-level qualitative criteria because each straw-man concept is identical in dimensions and thus also in performance. The criteria are assessed by means of sub-criteria, each with their own weight, which together make up the score of the top criteria. This method creates a comprehensible structure and rating of the known features of the straw-man candidates. The three main criteria of this trade-off are: the design flexibility/adaptability, the stowage efficiency, and the manufacturing/modularity.

The first criterion is the design flexibility/applicability (weight = 0.4). This criterion assesses how well the concept aligns with the specification sheet created for the membrane-frame solar array system. Ideally, the to-be-designed structure is versatile such that increasing its surface area (by adding panels and/or increasing its panel dimensions) is viable. Also, it would be preferred that the starting point of the risk-based iteration meets or approaches the requirements. In turn, the sub-requirements comprising the score of design flexibility/applicability are: Mission profile compatibility, accommodation penalty, and scalability. The following list presents the definitions, sub-weights, and the leading parameters for these sub-criteria:

- Mission compatibility (sub-weight = 0.3) is judged by its orbit type; if designed for LEO/GEO, it adheres to the environmental conditions and also evaluates whether the panel has any flight heritage (and thus a higher TRL).
- Accommodation penalty (sub-weight = 0.4) relates to how many (negatively impacting) alterations need to be performed to fit the array into the specified template. This includes effects like upscaling, creating design penalties that flow from 'the snowball effect', and downscaling, which is less severe but could result in an over-designed starting point.
- Scalability (sub-weight = 0.3) is defined by how easily the design can be altered in total surface

area by adding panels without inducing large redesign penalties. Adding panels is more effective in increasing the surface area but could increase the stowage height and, in turn, an increased offset bending moment. Increasing dimensions (in length and width) is also an effective measure but to a lesser degree than adding panels.

Initially, the solar cell compatibility was also assessed assuming there will be a significant difference. However, after performing measurements on the Silicon mPower modules and areal density calculations between GaAs strings, the difference is deemed negligible (see Section F.1 for more detail).

The second criterion is the stowage efficiency (weight = 0.3), which assesses how compactly an array can be stowed during launch. Ultimately, it is beneficial to be able to stow as much effective surface area while minimizing the stowage volume to minimize the offset bending moment during launch. Preferably, it should also be able to have some flexibility in applying the hold-down points, which provides design freedom. Lastly, at the locations where the preload of the hold-down is placed, it is favorable to have somewhat of a stiff frame which is able to resist deformations during the dynamic loading environment introduced by the launcher. The following list presents the definitions, sub-weights, and the leading parameters for these sub-criteria:

- The panel stowage height (sub-weight = 0.3) is defined as how effectively the longitudinal stiffening profile is stowed. What is most notable in this criterion is whether nesting is implemented, which significantly improves the stowage efficiency.
- The active area packing density (sub-weight = 0.2) is defined as how much of the designated $1.25\text{ m} \times 1.5\text{ m}$ area is usable by the SPMs. Factors that limit this area are longerons with a large width or obstructions within the plane due to internal stiffeners.
- The hold-down compatibility (sub-weight = 0.3) is defined as how much freedom there is for positioning the hold-down points. This is rated by how many feasible hold-down locations can be implemented.
- Frame stiffness (sub-weight = 0.2) refers to how well the panel resists deformations under various loading conditions. This frame should be able to resist HDRM preload, the compressive force introduced by the tensioned membrane, and dynamic loads under which the structure wants to deform. The ranking of this sub-criterion is governed by the number of stiffening elements used per panel and the profile type used, while assuming all concepts are made from the same material.

The third criterion is manufacturability/modularity (weight = 0.3), which is defined as the expected effort required to produce and assemble the design. This criterion implicitly accounts for the manufacturing and integration cost by assessing to what extent modularity is incorporated into the concept. For example, if a design consists of a few parts which are duplicated and uses bolts at the interfaces, it indicates less time is required for the assembly and it can be disassembled too if needed. Because the exact materials are not selected yet and flow from later analyses, the straw-man concepts are not judged on the availability of the materials in this stage. The sub-criteria that make up this criterion are:

- Directional dependency (sub-weight = 0.3) is defined as how dependent the design is on a specific (un)folding direction and bus attachment. When a panel has high directional dependency, the panel design is unable to change its orientation because there is inherent design bias to a specific attachment onto the bus.
- Assembly/disassembly ease (sub-weight = 0.7) is defined as the amount of effort that is expected in integrating the whole array structure. The drivers for the score of this criterion are how many parts the array consists of (and in turn the number of interfaces) and the type of interfaces.

3.2.4. Trade-off Results

In Section 3.2.4, the results of the performed trade-off are presented. A relative qualitative approach has been applied for which a score is chosen: orange = 2 (lowest), yellow = 3, light-green = 4 and dark-green = 5 (highest). The score of each criterion is determined by the weighted sum of its sub-criteria. Similarly, the final score of each concept is determined by the weighted sum of its criteria.

The final scores generated per concept are presented in Table 3.3. The best straw-man candidate that acts as a starting point for the risk-based iteration is the ELSA, followed by Airbus Neo, ULP,

and lastly JAXA’s ultra-lightweight panel. ELSA’s main strengths are that due to its nested minimal frame design, it offers a good balance between high stowed efficiency and assembly/disassembly ease. The disadvantage is that inherent bias is introduced in terms of hold-down locations, and significant pretension would be required to ensure dimensional stability of the SPMs.

The ULP and Airbus Neo, which utilize internal stiffening elements, follow closely after. While these elements do limit the maximum out-of-plane deflection of the membrane, they affect the assembly speed significantly. Also, because nesting is not utilized, adding extra panels will have a bigger impact on the total stack height compared to ELSA. If it is too difficult to mitigate the maximum allowed deflection of the membranes in the minimum frame design, the internal stiffeners could be added.

Lastly, JAXA’s ultra-lightweight panel follows closely after the ULP. While this design also utilizes the minimum frame design, it does not act as the best starting point for the risk-based iteration. The reason for this is that its design features suffer from the accommodation penalty because it had to scale up by a factor of approximately 2. In turn, the design suffers from lower panel stiffness and having limited hold-down locations, even though the design is highly modular and easily manufacturable. For a more elaborate reasoning behind this trade-off, see Section F.2.

Table 3.3: Straw-man trade-off score, where orange = 2 (lowest), yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion / Sub-criterion	ELSA	ULP	Airbus Neo	JAXA
<i>Design flexibility / adaptability (weight = 0.4)</i>				
Mission profile compatibility (0.3)	5	5	5	3
Accommodation penalty (0.4)	5	4	5	3
Scalability (0.3)	5	3	3	2
Subtotal	5.00	3.80	4.20	2.60
<i>Stowage efficiency (weight = 0.3)</i>				
Panel stowage height (0.3)	5	3	3	4
Active area packing density (0.2)	3	4	3	5
Hold-down compatibility (0.3)	2	4	5	2
Panel stiffness (0.2)	4	5	5	3
Subtotal	3.50	3.90	4.00	3.40
<i>Manufacturability / Modularity (weight = 0.3)</i>				
Directional dependency (0.3)	2	4	4	5
Assembly / disassembly ease (0.7)	5	3	3	5
Subtotal	4.10	3.30	3.30	5.00
Final score (weighted total)	4.28	3.68	3.87	3.56

Initial Design Conceptualization

Following the architectural trade-off in Chapter 3, the ELSA minimum-frame concept was selected as the baseline straw-man architecture. As identified in the trade-off, the primary structural vulnerability of this minimum-frame approach is its heavy reliance on high membrane pretension to mitigate out-of-plane dynamic deflections during launch. Before initiating a highly detailed mechanical design of the frame and hinges, it is critical to establish a preliminary structural baseline to understand the complex, non-linear interaction between frame compliance, membrane pretension, and transverse deflection.

Therefore, this chapter presents the initial design conceptualization and preliminary structural analysis of the baseline panel. First, Section 4.1 the required inputs and settings are presented. Finally, to validate the structural behavior and assess the impact of frame compliance, the numerical results are compared against theoretical rigid-boundary analytical models in Section 4.5.

4.1. Initial Finite Element Model Setup

This section defines the structural inputs, material properties, and computational settings required to establish the baseline numerical model of the membrane-frame architecture.

Material and Geometry Properties

The macroscopic dimensions of the baseline panel are constrained to a width of $W_{\text{panel}} = 1.5$ m and a length of $L_{\text{panel}} = 1.25$ m, directly inherited from the standardized straw-man sizing calculations established in Section 3.2.1. This geometric envelope accommodates the three primary structural components of the array:

- **The membrane:** A thin-film substrate providing the tensioned surface for the solar cell modules.
- **The longerons:** The primary frame elements positioned parallel to the deployment length of the solar array.
- **The lateral cross-members (stiffeners):** The transverse frame elements positioned perpendicular to the longerons.

For these initial finite element calculations, Kapton HN has been selected as the membrane material. The relevant mechanical properties are presented in Table 4.1. While Kapton HN (a polyimide film) exhibits slightly orthotropic behavior in reality, it is assumed to behave isotropically for these early design calculations [39]. Kapton HN was selected because its elastic modulus and yield strength are among the highest of available space-grade polymer films, alongside excellent radiation and atomic oxygen resistance. Furthermore, its Coefficient of Thermal Expansion (CTE) is relatively low compared to other fluoropolymers.

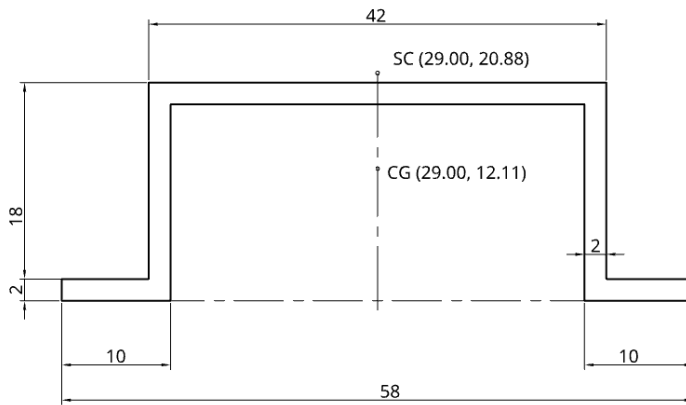
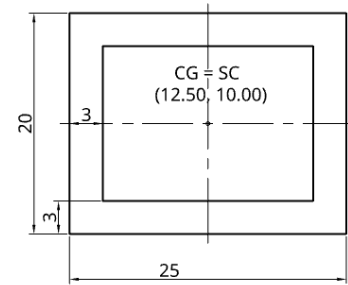
Because the ELSA minimum-frame concept serves as the baseline for this design iteration, the following geometric modeling assumptions and simplifications have been adopted for the preliminary frame profiles:

- The cross-section of the longerons utilizes a hat profile (Figure 4.1), while the lateral cross-members utilize a hollow rectangular profile (Figure 4.2).

Table 4.1: Consolidated material properties for the preliminary finite element analysis [40].

Symbol	Property	Kapton HN	CFRP (Idealized)	Unit
E	Young's modulus	2.5×10^9	200×10^9	Pa
ρ	Density	1420	1550	kg/m ³
ν	Poisson's ratio	0.34	0.15	–
σ_y	Yield stress	6.6×10^7	–	Pa
CTE	Coefficient of thermal expansion	2.0×10^{-5}	–	K ⁻¹
h	Nominal thickness	1.27×10^{-4}	–	m
–	Material model	Isotropic	Isotropic	–

- In the actual ELSA design, the webs of the longeron hat profiles are slightly slanted to facilitate nesting of the folded frame. For this initial analysis, the geometric and structural properties of the slanted profiles have been mathematically approximated as equivalent to their non-slanted counterparts.
- Manufacturing constraints requiring a minimum corner radius for CFRP composite profiles have not been taken into account during the calculation of the idealized section properties.
- In the finite element models, the edge of the membrane is assumed to coincide directly with the center of gravity of each frame profile. This simplification allows the membrane edges to be represented physically as stringers attached perfectly at the neutral axis.
- As a direct result of the above assumption, secondary effects—such as torsional loading of the profiles due to slightly off-center membrane attachments and the minor reduction in effective membrane area—are neglected in this current phase of analysis.

**Figure 4.1:** Hat profile dimensions used for longerons (in mm).**Figure 4.2:** Hollow rectangular profile used for lateral members (in mm).

Preliminary Finite Element Settings

To simulate the structural response, specific element formulations and boundary conditions were defined, as summarized in Table 4.3. The Kapton membrane is discretized using M3D4R elements—4-node quadrilateral membrane elements utilizing reduced integration and hourglass control. Because these elements lack bending stiffness and only transmit in-plane forces, they accurately represent the highly flexible behavior of a thin film. The supporting frame is discretized using B21 elements, which are 2-node linear beams assigned to the edges of the membrane.

The pretension on the membrane is artificially applied by predefining an initial temperature field at $T = 0$ K and calculating the required thermal contraction to achieve the specified uniform stress. For example, to achieve an in-plane pretension of $T_x = T_y = 1000$ N/m, the required stress across a fully

Table 4.2: Comparison of Section Properties.

Property	Hollow Rectangle	Hat Profile
Area	234 mm ²	188 mm ²
Centroid x_c	12.5 mm	29.0 mm
Centroid y_c	10.0 mm	12.106 mm
I_{xx}	12322 mm ⁴	10616.54 mm ⁴
I_{yy}	18039.5 mm ⁴	61342.67 mm ⁴
I_{xy}	—	2476 mm ⁴
J (Torsion Constant)	—	250.67 mm ⁴
Shear Center x_{sc}	12.5 mm	29.0 mm
Shear Center y_{sc}	10.0 mm	20.878 mm
e_y (from centroid)	0.0 mm	1.878 mm

fixed membrane is:

$$\sigma = \frac{T_F}{t} = \frac{1000}{1.27 \times 10^{-4}} = 7.874 \text{ MPa} \quad (4.1)$$

Generating this stress via thermal contraction for Kapton HN (which possesses a CTE of $\alpha = 2.0 \times 10^{-5} \text{ K}^{-1}$) requires a prescribed temperature change (ΔT) defined by:

$$\Delta T = \frac{\sigma(1-\nu)}{E \cdot \alpha} = \frac{7.874 \times 10^6 \cdot (1-0.34)}{2.5 \times 10^9 \cdot 2.0 \times 10^{-5}} = 103.94 \text{ K} \quad (4.2)$$

Table 4.3: FEM Model Setup and Boundary Conditions.

Component	Element Type & Details
Sheet / Membrane	M3D4R — 4-node quadrilateral membrane element, reduced integration, hourglass control. Transmits in-plane forces only (no bending stiffness).
Edge / Frame	B21 — 2-node linear beam element (used on boundaries).
Pretension	Isotropic pretension = 1000 N/m $T_x = T_y = T$, equivalent $\Delta T = 103.94 \text{ K}$
Loading	Gravity loading Component 1 = 0, Component 2 = 0, Component 3 = $-9810 \times n_z \text{ (mm/s}^2\text{)}$
Analysis Settings	NLGEOM = On (Large displacement formulation) Time period = 1 Maximum number of increments = 10,000 Initial increment size = 1×10^{-5} Minimum increment size = 1×10^{-5} Maximum increment size = 1
Boundary Conditions	
Fixed Edge Model	All edges fully fixed: $u_1 = u_2 = u_3 = 0$
Fixed Point Model	Only corner points pinned (translations restrained)

The preliminary results derived from this setup are divided into three distinct analyses: an analytical model of the fixed-edge case (Section 4.2), a numerical model of the fixed-edge case (Section 4.3), and a numerical model of the fixed-node case (Section 4.4). The primary objective of this tri-fold analysis is to evaluate how different boundary condition assumptions affect global deformation, and to correlate the analytical approximations with high-fidelity numerical responses. Finally, an overarching comparison is synthesized in Section 4.5.

4.2. Analytical Results

The analytical solution utilized proposes an algebraic cubic relationship with respect to the out-of-plane deflection at the center of a pretensioned membrane [12]. This solution is briefly discussed in Section 2.3 with the full code provided in Appendix B. This approach explicitly assumes that the out-of-plane displacements along the rigid boundaries are exactly zero. By modeling these analytical relationships, the structural response between applied pretensions, stress resultants, and applied load factors can be rapidly derived.

Using the material and geometric inputs established in Section 4.1, the variations in T_x , T_y , and maximum central deflection are plotted in Figure 4.3 for a load factor of $n_z = 1$, and in Figure 4.4 for $n_z = 3$. The red transparent plane represents the maximum allowable deflection limit, with the red intersection line indicating the exact (T_x, T_y) pretension combinations that satisfy this boundary. As expected, higher combinations of T_x and T_y yield lower maximum deflections. However, the reduction in deflection experiences diminishing returns as pretension increases. Furthermore, as the load factor increases to $n_z = 3$ (Figure 4.4), the applied surface pressure rises, drastically increasing the out-of-plane deflection such that the allowable limit is no longer satisfied at lower tensions.

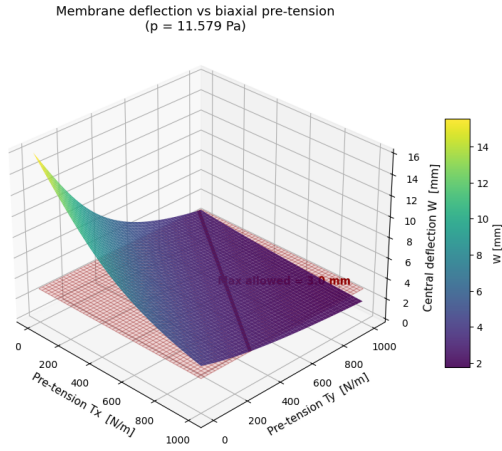


Figure 4.3: Out-of-plane deflection u_3 for a variation of T_x and T_y at $n_z = 1$. The red surface indicates the maximum allowed deflection.

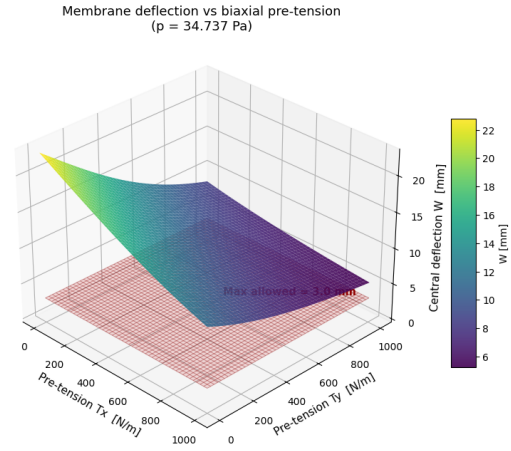


Figure 4.4: Out-of-plane deflection u_3 for a variation of T_x and T_y at $n_z = 3$. The red surface indicates the maximum allowed deflection.

Figure 4.5 isolates the variation in maximum out-of-plane deflection against an increasing load factor n_z , assuming a uniform in-plane pretension ($T_x = T_y = T$). Across all cases, the deflection increases with n_z , reflecting the classic nonlinear geometric response of a membrane under transverse loading. Notably, at $T = 1000$ N/m, the structural response is significantly stiffer than at lower tensions, demonstrating a gradual decrease in slope as n_z rises. This confirms the nonlinear stiffening behavior inherent to highly tensioned thin films.

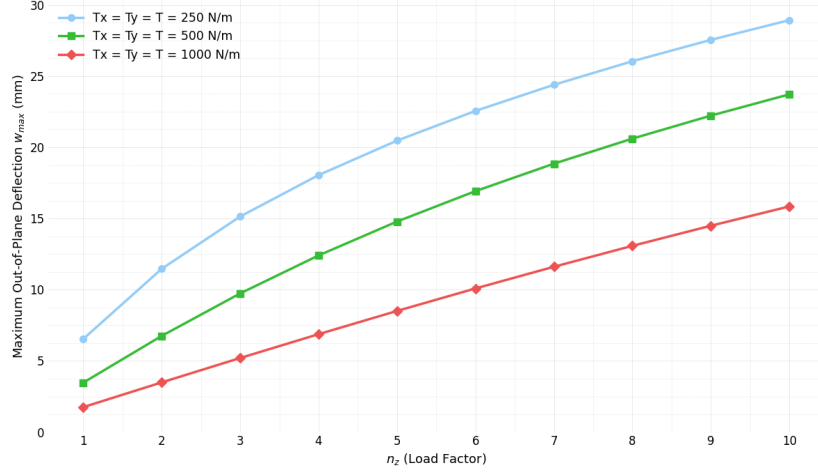


Figure 4.5: Analytical results of maximum out-of-plane deflection vs. load factor for $T_x = T_y = 250, 500$, and 1000 N/m.

4.3. Numerical Results: Fixed-Edge Boundary Condition

Section 4.3 presents the finite element results for the fixed-edge model. In this configuration, the boundaries are fully constrained in translation but left free in rotation (i.e., $u_1 = u_2 = u_3 = 0$). The resulting out-of-plane displacement u_3 for $n_z = 1$ and $T_x = T_y = 1000$ N/m is illustrated in Figure 4.6 from front, side, and isometric perspectives. To aid visualization, the deformed shape has been scaled by a factor of 100.

As anticipated by the pinned boundary conditions, the edges experience zero deformation. Consequently, the geometric center of the membrane experiences the absolute maximum out-of-plane deflection, measuring -1.576 mm under the predefined loading.

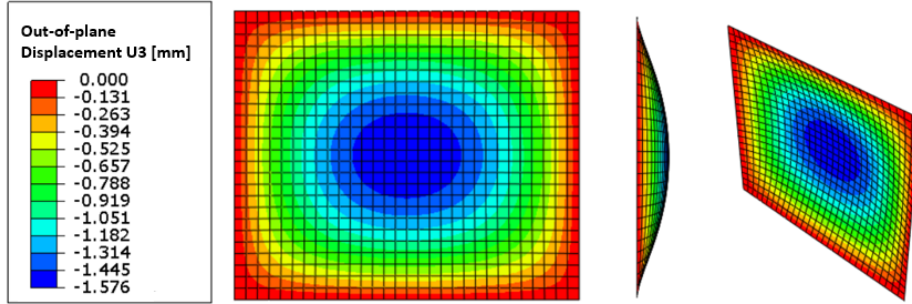


Figure 4.6: Out-of-plane displacement u_3 for the fixed-edge model at $n_z = 1$ and $T_x = T_y = 1000$ N/m. The deformed shape is scaled by $100\times$.

Furthermore, Figure 4.7 and Figure 4.8 map the corresponding normal stress resultants. It is clearly evident from these contours that the variation in normal stress is exceptionally low (ranging strictly between 7.87 and 7.89 MPa). This homogeneity highlights a physical inaccuracy introduced by the idealized boundary condition: as described in Section 4.1, by applying a pretension of 7.874 MPa and artificially fixing the displacement of the membrane edges, the initial state is forced to remain perfectly uniform. As a result, the FEM artificially generates a stiffer structural response, keeping the additional out-of-plane deflection induced by the $1g$ gravity load at a lower order of magnitude.

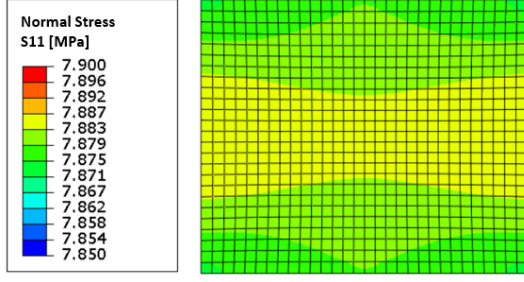


Figure 4.7: Normal Stress S11 [MPa] of the fixed-edge model for $n_z = 1$ and $T_x = T_y = 1000$ N/m.

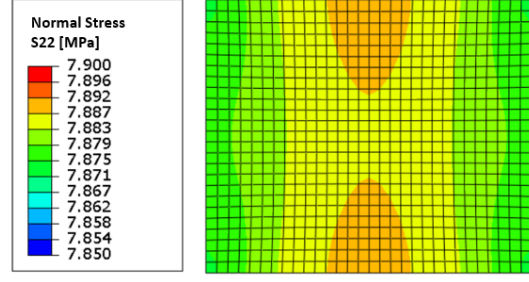


Figure 4.8: Normal Stress S22 [MPa] of the fixed-edge model for $n_z = 1$ and $T_x = T_y = 1000$ N/m.

4.4. Numerical Results: Fixed-Corner Boundary Conditions

Section 4.4 transitions to the numerical results for the fixed corner node model. In this configuration, a pinned boundary condition ($u_1 = u_2 = u_3 = 0$) is imposed strictly at the four corner nodes, simulating discrete HDRM hold-down points during the stowed configuration.

The out-of-plane deflection for the $n_z = 1$ and $T_x = T_y = 1000$ N/m load case is presented in Figure 4.9. Unbound by continuous rigid edges, the frame elements exhibit clear in-plane bending. Because the boundaries are compliant, the maximum out-of-plane deflection experienced at the center of the membrane increases to -3.348 mm—more than twice as large as the deflection predicted by both the fixed-edge FEM and analytical models. This massive divergence proves that the assumption of a purely rigid frame drastically underestimates global deflection for the selected structural profiles.

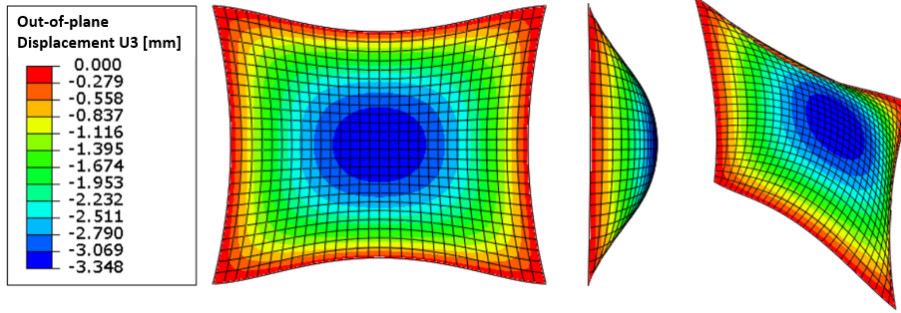


Figure 4.9: Out-of-plane displacement u_3 for the fixed-corner model at $n_z = 1$ and $T_x = T_y = 1000$ N/m. The deformed shape is scaled by $100\times$.

Figure 4.10 and Figure 4.11 detail the normal stress distributions in the axial and lateral directions, respectively. Unlike the near-perfect uniformity observed in the fixed-edge models (Figure 4.7 and Figure 4.8), there is a severe delta between the minimum and maximum stress values here. This non-uniform stress field is a direct consequence of the frame elements bowing inward under the pretension load. As the frame deflects, the initial uniform pretension relaxes, creating localized zones of structural slack.

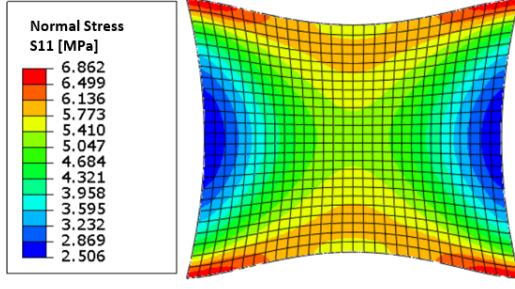


Figure 4.10: Normal Stress S11 [MPa] of the fixed-corner model for $n_z = 1$ and $T_x = T_y = 1000$ N/m.

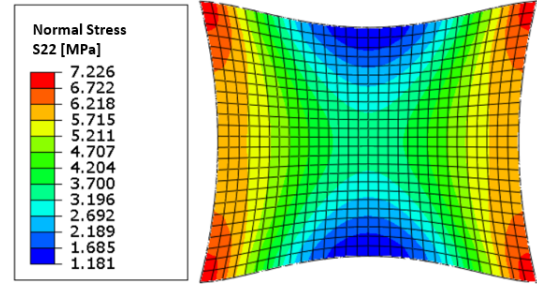


Figure 4.11: Normal Stress S22 [MPa] of the fixed-corner model for $n_z = 1$ and $T_x = T_y = 1000$ N/m.

To quantify this relaxation, Figure 4.12 and Figure 4.13 plot the drop in in-plane pretension along the x - and y -axes before the gravitational load is even applied. It is evident that the pretension loss is most severe at the mid-span of the frame elements. Because maximum inward frame deflection occurs at the center of the beam span, the attached membrane elements experience the largest relaxation. This drop is noticeably larger along the lateral cross-members than the longerons due to their slightly longer span and lower area moment of inertia (I_{yy}). This mechanical relaxation fully explains why u_3 is exacerbated when edge compliance is accounted for.

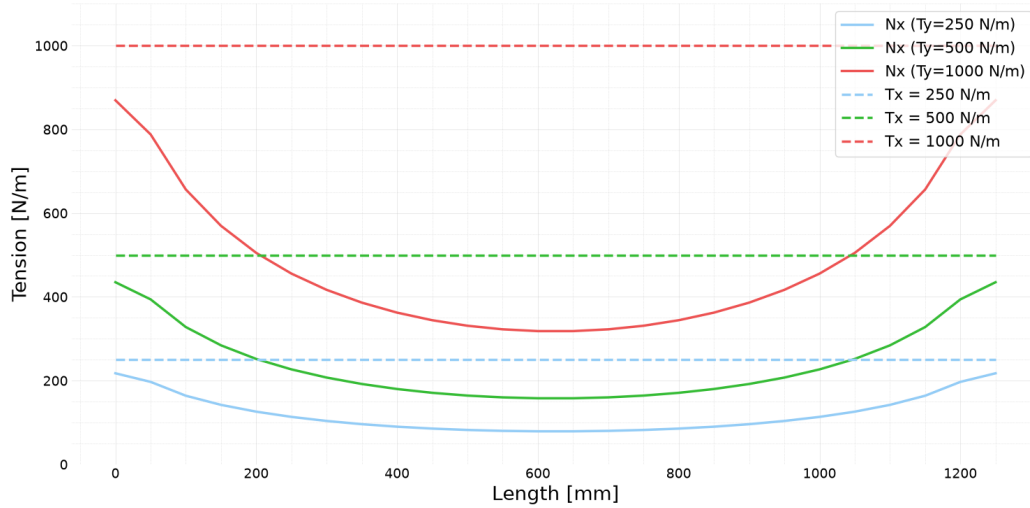


Figure 4.12: Applied vs. experienced pretension for $T_x = T_y = 250, 500, 1000$ N/m along the longeron length.

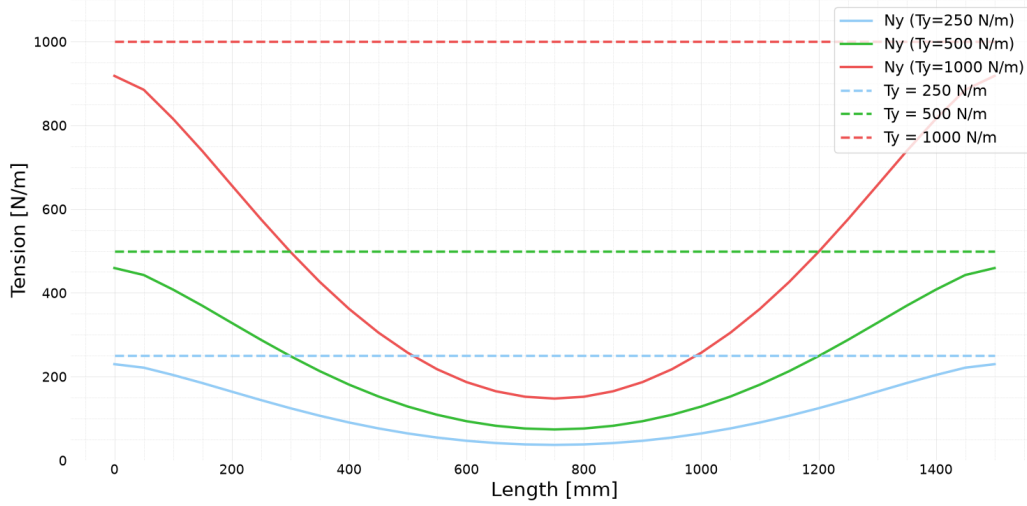


Figure 4.13: Applied vs. experienced pretension for $T_x = T_y = 250, 500, 1000$ N/m along the lateral element length.

4.5. Comparison of Boundary-Condition Configurations

In Section 4.5, an overarching comparison between the three distinct modeling approaches is presented. Figure 4.14 tracks the increase in maximum out-of-plane deflection (u_3) across ascending load factors (n_z). The blue line represents the numerical fixed corner node model, the green line represents the numerical fixed-edge model, and the red line represents the analytical fixed-edge formulation. The following critical observations are made:

- The numerical fixed corner node model deflects 78% to 112% more than the numerical fixed-edge model (with the percentage delta decreasing at higher n_z loads).
- The numerical fixed corner node model deflects 59% to 91% more than the analytical fixed-edge approximation.
- The numerical fixed-edge model actually deflects 11% to 11.6% *less* than the analytical rigid-edge model.
- A near-perfect second-order polynomial fit can be applied to characterize the nonlinear response for all three data sets.

The mechanical reason the numerical fixed-edge model is less compliant than the analytical fixed-edge model lies in the nature of the assumed deflection shape. The analytical solution assumes the out-of-plane deflection follows a smooth, continuous sine wave: $w(x, y) = W \sin\left(\frac{\pi x}{a}\right) \sin\left(\frac{\pi y}{b}\right)$. At the midplane ($x = a/2$), this simplifies to $w(y) = W \sin\left(\frac{\pi y}{b}\right)$.

However, the true out-of-plane deflection shapes plotted in Figure 4.15 tell a different story. The numerical solution exhibits a much steeper deformation gradient near the clamped boundaries, paired with a flatter, plateaued central region. This geometry is characteristic of membrane "pillowing" under transverse loading [41]. Because the M3D4R membrane elements have zero bending stiffness, they must rotate so the vertical vector of their tension physically pushes back against the transverse pressure. At the constrained edges, the adjacent elements must angle upward much more sharply than elements in the center. Because these boundary elements take the brunt of the deformation gradient, the internal elements remain comparatively flat. The idealized sinusoidal formula used in the analytical approximation cannot internalize this localized pillowing effect, resulting in a slight overestimation of global displacement. This discrepancy perfectly mirrors previous FEM comparisons by Morozov and Lopatin, confirming that this numerical deviation is an expected limitation of the "quick" analytical solution [12].

Finally, the massive divergence of the fixed-node model compared to both fixed-edge cases is driven entirely by the in-plane compliance of the CFRP frame profiles, which severely relaxes the stabilizing mid-tension. This drastic increase in transverse displacement under high initial pretension highlights

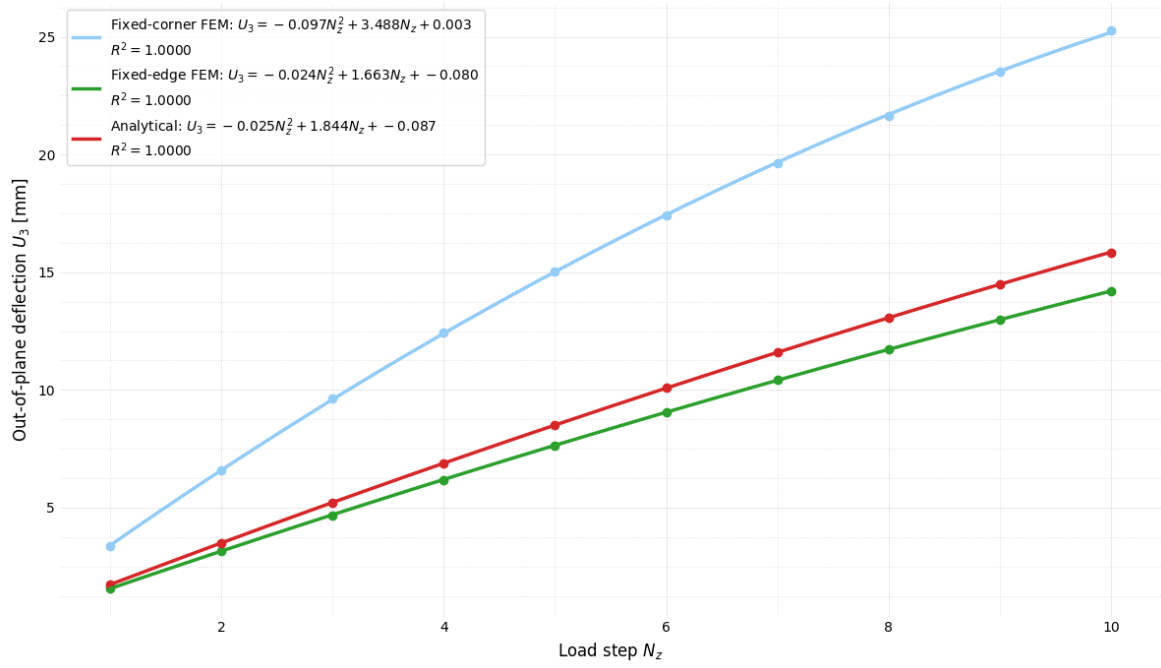


Figure 4.14: Out-of-plane deflection u_3 across an increasing load factor n_z for the fixed corner FEM (blue), fixed-edge FEM (green), and analytical model (red).

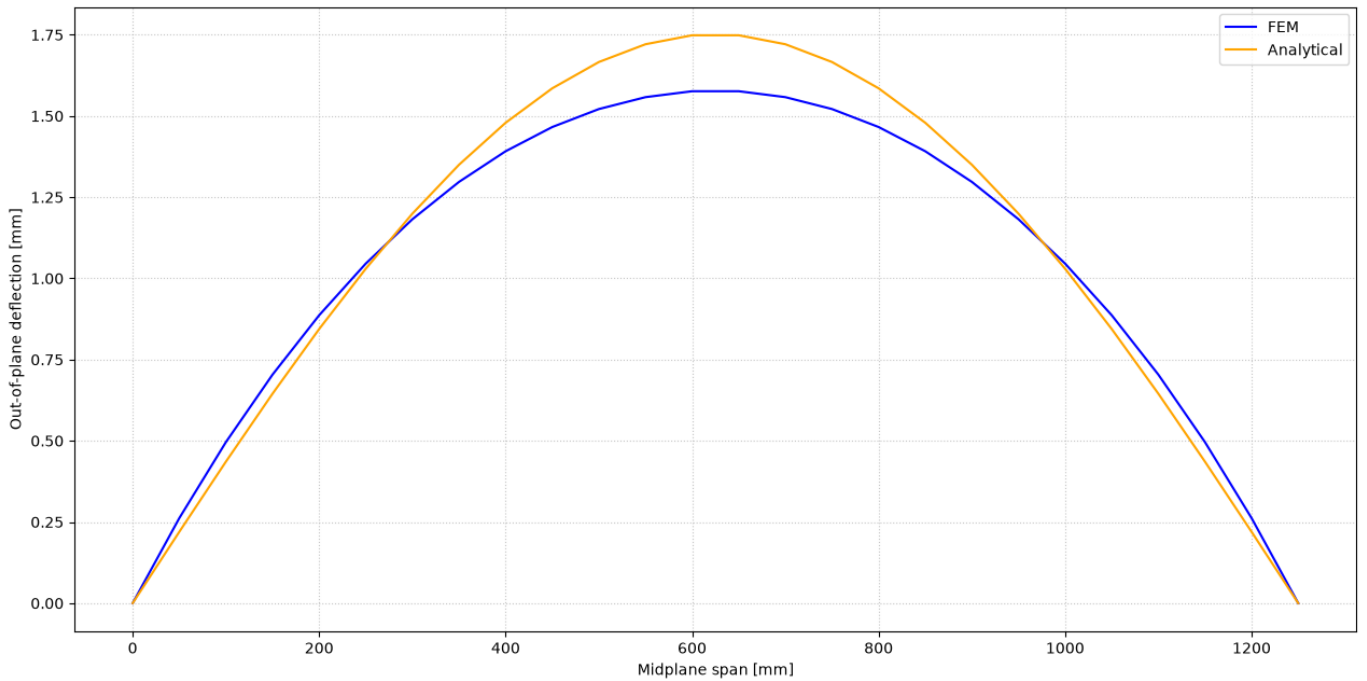


Figure 4.15: The FEM predicts localized membrane pillowing near the constrained edges, resulting in steeper boundary gradients and a flatter central region than the analytical sine-wave approximation.

a fundamental architectural constraint. Specifically, utilizing membrane pretension to mitigate out-of-plane launch deflections is only viable if the surrounding frame elements are engineered with sufficient in-plane bending stiffness to prevent slack.

4.6. Architectural Implications and Mitigation Strategy

The numerical findings from this preliminary baseline study highlight a fundamental structural trade-off in semi-flexible solar array design:

1. **The Pretension Paradox:** While increasing baseline pretension theoretically reduces transverse out-of-plane membrane displacement, the resulting compressive line loads induce substantial in-plane bending in slender composite frames. This frame compliance relaxes the membrane tension at the mid-span—precisely where stiffness is needed most.
2. **Mass Penalty of Purely Structural Solutions:** Overcoming this relaxation purely through structural stiffening would require significantly bulkier longerons and cross-members with larger cross-sectional moments of inertia (I_{yy}). This would incur a severe structural mass penalty, negating the specific power advantages of a flexible-substrate architecture.
3. **Long-Term Mechanical Risks:** Maintaining high sustained static pre-stress increases susceptibility to long-term viscoelastic creep in polymer substrates and elevated stress concentrations at discrete mechanical attachment points [7].

Although treating the CFRP profiles as isotropic slightly overestimates frame compliance, the primary conclusion holds: relying solely on high static pretension to suppress launch-induced dynamic displacements is structurally inefficient for minimal-frame architectures. To maintain a lightweight primary structure while strictly limiting dynamic membrane oscillations, dynamic response mitigation must instead rely on energy dissipation and structural damping. This mitigation strategy is seen in both the ROSA (Figure 2.2) and ELSA (Figure E.1a) designs, where strips of polyimide foam are used to suppress oscillation amplitudes.

In a deployed or isolated state, the structural response is governed by inherent material properties, local boundary conditions, and the complex tension-stiffness coupling between the frame and the membrane. Since the primary objective of this research is to prove that the real-world dynamic behavior of this sensitive architecture can be modeled accurately, the subsequent experimental campaign focuses entirely on precise Finite Element Model (FEM) correlation. By building a physical prototype and extracting its fundamental resonant frequencies and dynamic mode shapes, the numerical modeling methodology can be validated. Consequently, investigating the highly non-linear contact mechanics and damping interactions of a fully stowed, multi-panel stack falls outside the scope of this baseline validation and is recommended for future work.

5

Prototype Design

This chapter details the design of a nested, low-profile membrane-frame solar array prototype. This hardware serves as a small-scale version of the baseline panel investigated in Chapter 4, specifically scaled to interface with an aluminum test rig for the subsequent experimental modal analysis.

First, Section 5.1 establishes the prototype's primary role, functions, and physical requirements. Next, Section 3.1 evaluates the core design trade-offs regarding the membrane tensioning mechanisms and global architectural configuration. Following this, Section 5.3 details the sizing calculations for the CFRP cross-sections and material properties, culminating in an iterated, as-built finite element model. Finally, Section 5.4 presents the completed CAD assembly alongside the physical, manufactured prototype.

5.1. Prototype Role, Functions, and Requirements

In Section 5.1, the purpose, functions, and requirements of the prototype design are presented. Experimental validation was restricted to a small-scale prototype to properly bound the scope of this thesis and accommodate facility limitations. This scaled article faithfully captures the critical structural features of the selected design concept.

5.1.1. Prototype Role in the Project

The baseline design of the membrane-frame Photovoltaic Assembly (PVA) is currently in a preliminary conceptual phase. Consequently, this physical prototype provides empirical data on manufacturing and integration while evaluating the experimental modal analysis feasibility of the Polytec PSV-500 and LDS V550 hardware.

The foremost role of the prototype is to directly support the main research question: validating the dynamic behavior obtained from the finite element model against a physical system. Although a full-scale panel would provide direct 1:1 observations, a scaled-down test subject was deliberately chosen to effectively mitigate costs. A small-scale Membrane-Frame Panel (MFP) also allows for the use of the in-house V550 series vibrators at GTM Advanced Structures, significantly improving the practicality of this experiment.

Additionally, the prototype provides vital physical insights into key design features. It allows for the practical evaluation of leading geometric constraints, such as the maximum allowable fastener heights and the lateral stiffener clearances required to minimize the stacking height of the nested frame. It also physically reveals the structural load paths of a uni-directionally tensioned panel, highlighting the most stress-critical areas.

Finally, empirical data is obtained regarding the actual assembly process of a tensioned PVA system. Overcoming practical challenges like ensuring uniform tension distribution and securely clamping ultra-thin interfaces is essential. Doing so highlights which manufacturing variables have the most significant influence on the final structural dynamics.

5.1.2. Functional Requirements

To ensure the prototype functions as intended and yields scientifically valid data, a formal requirements list has been developed for the detailed design process. The MFP requirements are presented in Table 5.1, complete with a verification strategy and engineering rationale. The following justifications dictate the specific values for the driving requirements:

- **MFP-REQ-001** dictates a pretension value of 500 N/m. This was determined based on the previous calculations during the initial design conceptualization in Section 4.2. During this phase, 500 N/m was chosen as an optimal middle-tension baseline, which for a 50 μm thick membrane translates to a nominal planar stress of approximately 10 MPa.
- **MFP-REQ-005** will be tested by applying a high-contrast reference line across the membrane-frame boundary. Over the testing period, this interface can be visually and optically inspected to verify whether localized relaxation or slippage of the membrane occurs.
- **MFP-REQ-009** is based on the assumption that an mPower DragonScales test coupon, covering a maximum of 80% of the membrane surface, will be applied if the initial baseline measurements prove successful. The areal density of these specific coupons has been measured and is documented in Section F.1.

Table 5.1: Prototype Functional and Performance Requirements, where the verification type is divided as A = Analysis, I = inspection, D = demonstration & T = test.

PROTOTYPE REQUIREMENTS				Verification Type				Rationale / Comment
Req. ID	Req. Statement	Value	Type	A	I	D	T	
MFP-REQ-001	The test rig shall apply and maintain a distributed tension across the membrane.	$500 \pm 10\%$ N/m	Key	✓	✗	✗	✓	Target operational tension for the experimental setup.
MFP-REQ-002	The required membrane tension shall be applied exclusively by displacing the edge hat profile.	N/A	Key	✗	✓	✓	✗	Determines the mechanical loading and assembly strategy.
MFP-REQ-003	The maximum principal stress in the Kapton membrane shall not exceed 50% of its yield stress.	$\leq 0.5\sigma_y$	Killer	✓	✗	✗	✗	Provides a Factor of Safety ≥ 2.0 to prevent material yielding.
MFP-REQ-004	The structural frame elements shall exhibit no global or local buckling under maximum test loads.	N/A	Killer	✓	✗	✗	✓	Buckling compromises membrane tension and structural stability.
MFP-REQ-005	The membrane-to-frame interface shall permit less than 0.5 mm of relative slippage at the target tension of 500 N/m.	≤ 0.5 mm	Driver	✗	✓	✗	✓	Essential for the functionality of the isolated tension concept.
MFP-REQ-006	The adhesive joint shall withstand a proof load of 1.5 times the nominal tension without localized peeling, shear failure, or adhesive failure.	≥ 750 N/m	Killer	✓	✗	✗	✓	Ensures structural integrity of the membrane-frame interface.
MFP-REQ-007	The structural frame elements shall be manufactured exclusively from CFRP.	CFRP	Constr.	✗	✓	✗	✗	Pre-determined material choice for required stiffness-to-mass ratio.
MFP-REQ-008	The height of the lateral profile shall be strictly less than the height of the hold-down blocks.	$< H_{\text{block}}$	Key	✓	✓	✗	✗	Required to prevent interference during mounting and testing.
MFP-REQ-009	The tensioned membrane structure shall be capable of supporting a mass of 30 g distributed over a 15.2×15.8 cm area (simulating an mPower module) without failure.	30 g	Driver	✓	✗	✓	✗	Validates the practical structural application of the array and allows performing populated test.
MFP-REQ-010	The tensioned assembly shall exhibit no more than a 2.5% reduction in its fundamental natural frequency over a 72-hour period, verifying a sustained membrane tension within 5% of the initial load.	$\Delta f_n \leq 2.5\%$	Driver	✗	✗	✗	✓	Required for consistent dynamic testing and modal analysis.

5.2. Concept Development and Design Rationale

5.2.1. Structural Membrane Concept Trade-off

Before selecting a specific material used for the prototype structural membrane, the macroscopic architecture was evaluated. Three different architectures are considered: the continuous sheet, the strip mesh and the wire mesh (presented in Figure 5.1a, Figure 5.1b and Figure 5.1c respectively). The continuous sheet is defined as a thin film that consists of one piece. While the strip mesh consists of many strips inter-weaved creating effectively a net that can carry the SCAs. Similarly, the wire mesh does the same but uses very thin, potentially fiber wires to support the SCAs. While spaceflight solar arrays occasionally utilize discrete tensioned networks or laminates to manage in-orbit thermal loads, the requirements for this experimental modal prototype prioritize different behaviors. To determine the optimal architecture for the test campaign, a trade-off was performed between a continuous sheet, a strip mesh, and a wire mesh.

The evaluation criteria for this trade-off are defined as follows:

- **Manufacturability:** The ease of procurement and integration into the CFRP frame without requir-

ing specialized weaving or alignment equipment.

- **Modal predictability:** The ability to accurately model the baseline dynamics in a Finite Element Model (FEM) and measure them cleanly using optical equipment (e.g., a PSV-500 scanning vibrometer).
- **Tension uniformity:** The ability to achieve a perfectly distributed load across the structure without inducing localized wrinkles or sagging.
- **Flight damage tolerance:** The resistance to tear propagation and catastrophic failure if a localized puncture occurs.

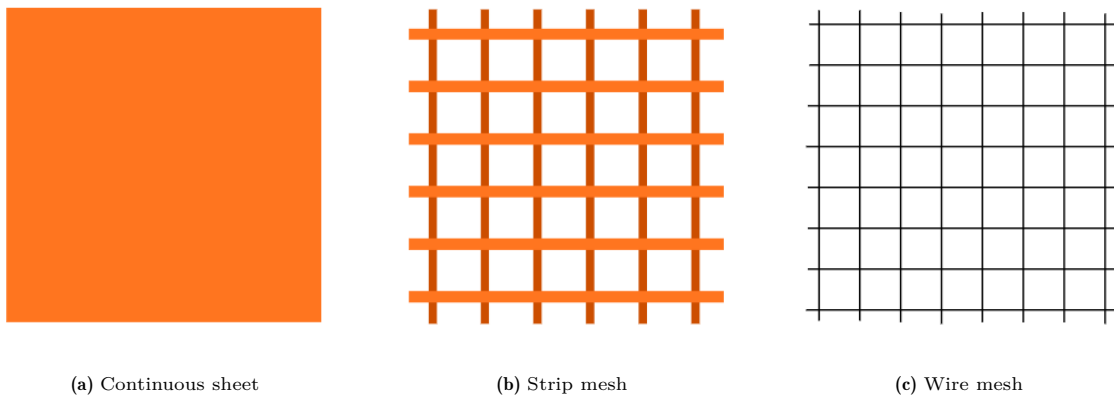


Figure 5.1: Overview of membrane architecture options for the prototype.

The final scores of the membrane architecture are presented in Table 5.2. As presented, the final chosen membrane architecture for the prototype is the continuous sheet. Where the continuous sheet excels compared to mesh architectures manufacturability. The sheet only requires cutting to size and edge preparation for clamping. In contrast, both strip and wire meshes demand extensive preprocessing and a specialized alignment tool to ensure correct distribution and configuration. Furthermore, the modal predictability of the membrane is more achievable both numerically and experimentally, because complex real-world effects such as micro-damping or inter wire/strip friction can be neglected. The disadvantage of a continuous sheet is that it is harder to prevent wrinkling of the membrane when applying tension. Also if a continuous membrane gets damaged because the tension is carried over the full membrane, it will have little resistance to tear propagation while the mesh architectures will have some level of redundancy incorporated.

Table 5.2: Concept selection matrix for the membrane architecture, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	Continuous sheet	Strip mesh	Wire mesh
Manufacturability (0.30)	5	3	2
Modal predictability (0.35)	5	3	2
Tension uniformity (0.15)	3	4	5
Flight damage tolerance (0.20)	2	4	5
Final score (weighted total)	4.10	3.35	3.05

Membrane material selection

Having established the continuous sheet as the optimal architecture, the specific material for the membrane must be selected. Because this prototype serves as both a structural testbed and a baseline for Finite Element Model (FEM) correlation, the material must balance physical prototyping constraints with flight-representative mechanical properties.

The candidate materials evaluated for the membrane include Polyimide (Kapton HN), a custom Kapton-GF laminate, Versiv CT205 (PTFE-GF), PVF (White Tedlar), and FEP (Teflon). The selection of the first three materials is driven by a prior experimental study conducted at GTM Advanced Structures [42]. The Kapton-GF-966 is a custom laminate utilizing two layers of 3M EY966 acrylic adhesive to bond the Kapton film to the glass-fiber reinforcement, created to gain insight into the substrate behavior of existing flexible array architectures. Furthermore, PVF and FEP are included in the trade space due to their exceptional chemical stability as fluoropolymers and high resistance to atomic oxygen [43].

To determine the optimal material, a trade-off was performed using the following five criteria:

- **Isotropic behavior:** The degree to which the material behaves homogeneously in all directions. High isotropy is critical for baseline experimental modal analysis, as orthotropic or woven materials introduce complex, non-linear dynamic modes that are difficult to correlate in a baseline FEM.
- **Specific properties (E/ρ , σ_y/ρ):** The specific stiffness and specific yield strength of the material. High specific properties ensure the membrane can carry the 500 N/m pretension without yielding, while keeping parasitic mass low.
- **Environmental properties:** The material's resilience to the space environment (e.g., Atomic Oxygen (ATOX) resistance, UV degradation) and its dimensional stability under structural loads (low Coefficient of Thermal Expansion (CTE) and high creep resistance).
- **Manufacturing effort:** The amount of labor required to integrate the material into the prototype. This includes whether the material is a ready-to-use standard product versus a custom fabrication, and the difficulty of surface treatment required to achieve a reliable adhesive bond at the clamping interface.
- **Availability and cost:** The logistical feasibility of procuring the material within the budgetary and temporal constraints of the prototyping phase.

As presented in Table 5.4, Kapton HN emerges as the optimal membrane material. The properties used to determine the score for specific and environmental properties are presented in Table 5.3. While pure polymer films are susceptible to visco-elastic creep under sustained tension, a major issue that typically requires glass-fiber (GF) reinforcement for long-term flight arrays, this limitation is accepted for the scope of the prototype. As observed during initial fabrication trials [42], producing a reliable Kapton-GF laminate in-house introduces significant manufacturing inconsistencies. More importantly, the addition of a woven GF reinforcement severely penalizes the material's isotropic behavior. Because the primary objective of this prototype is to establish a clean, predictable structural baseline for optical modal measurements and FEM correlation, the orthotropic complexities of a composite laminate would obscure the dynamic results. Therefore, pure Kapton HN is selected as the optimal compromise, providing exceptional specific properties and sufficient dimensional stability for the duration of the experimental campaign.

Table 5.3: Typical mechanical and environmental properties of candidate membrane materials.

Material	Density, ρ (g/cm ³)	Yield Str., σ_y (MPa)	Modulus, E (GPa)	CTE (ppm/°C)	Max Temp. (°C)	Erosion Yield, E_y ^c (cm ³ /atom)
Polyimide (Kapton HN) Dupont2008, [42]	1.42	66	2.5	20	(-269, 260)	2.81×10^{-24}
Kapton-GF-966 [42]	1.44 ^a	39.6 ^a	3.25 ^a	ND	(-40, 232) ^b	ND
Versiv PTFE-GF [42], [44]	1.97	35.4 ^a	3.6 ^a	ND	(-150, 260)	ND
PVF (White Tedlar) [45]	1.37-1.39	ND	2.1-2.6	50-97	(-70, 150)	1.01×10^{-25}
FEP (Teflon) [46]	2.15	12	0.483	58 (MD)	(-200, 204)	2.00×10^{-25}

a: These values are obtained experimentally during a prior internship at GTM-AS and are not cross-checked [42].

b: The Kapton-GF-966 is a self-made laminate in which the temperature range is assumed to be based on 3M EY966 [42], [47].

c: Erosion yield (E_y) data sourced from the MISSE 2 PEACE Polymers experiment [43].

Table 5.4: Concept selection matrix for the membrane material, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	Kapton	Kapton-GF	PTFE-GF	PVF	FEP
Isotropic behavior (0.25)	5	1	1	5	5
Specific properties (0.25)	5	4	3	3	3
Environmental properties (0.20)	4	3	4	4	5
Manufacturing effort (0.15)	5	1	5	5	5
Availability & cost (0.15)	5	5	5	3	3
Final score (weighted total)	4.80	2.75	3.30	4.00	4.20

5.2.2. Tensioning and clamping concepts

The tensioning method of the membrane is closely related to the interface between the membrane and longitudinal profiles. Following an initial brainstorming phase, Figure 5.2 presents the interface and tensioning methods that are considered for the prototype. The concepts can be roughly divided into two categories: Integrated tensioning systems (Figure 5.2a to Figure 5.2e) and passive clamped interfaces that require an external tensioning tool for assembly (Figure 5.2f to Figure 5.2k). The advantage of having a tensioning system integrated into the system is that it allows for fast and standalone assembly of the prototype. However, such integrated tensioning systems do introduce two major issues: 1. It adds significant parasitic mass to the panel and 2. it requires significant volume which affects the stacking height of the nested frame design. The latter is formulated in MFP-REQ-008 which is deemed not feasible for the method shown in Figure 5.2a to Figure 5.2e. Similarly, whilst not being an integrated tensioning system, the modular rod (presented in Figure 5.2f) is also eliminated due to the height requirement MPF-REQ-008.

The remaining passive clamping interfaces that require an external tensioning tool for assembly, are presented in Figure 5.2g to Figure 5.2k. To select the optimal interface, a trade-off was performed and is presented in Table 5.5. The trade-off consists of four criteria which are scored a "5" for the highest and "1" for the lowest score. The trade-off criteria are defined as the following:

- **Manufacturability:** The expected effort required to implement the passive interface without damaging the thin film and apply the required pretension in a controlled manner.
- **Slippage resistance:** Derived from MPF-REQ-005, this defines how well the membrane is expected to retain its tension over time.
- **Mass and footprint:** The amount of parasitic mass and physical volume is added.
- **Mechanical redundancy:** The extent to which the interface prevents instantaneous, catastrophic tension loss in the event of a single-point failure.

As seen in Table 5.5, the combined bonded and clamped concept emerged as the optimal solution. Although a purely bonded interface offers advantages in mass and footprint, it suffers from a lack of mechanical redundancy and high vulnerability to peel stresses and viscoelastic creep. Consequently, it is unsuited for the primary load-bearing bond along the direction of pretension. As a joining interface along the lateral profile direction however, it does seem to be a reasonable joining method because the tension perpendicular to it is significantly lower. The runner-up is the bonded and riveted solution which ultimately scores very poor on the manufacturability. The big problem here is that drilling and punching holes through a tensioned 50-micron Kapton film creates massive, localized stress concentrations. Local reinforcement along the edge must be required to make this work or a thicker membrane needs to be used in general.

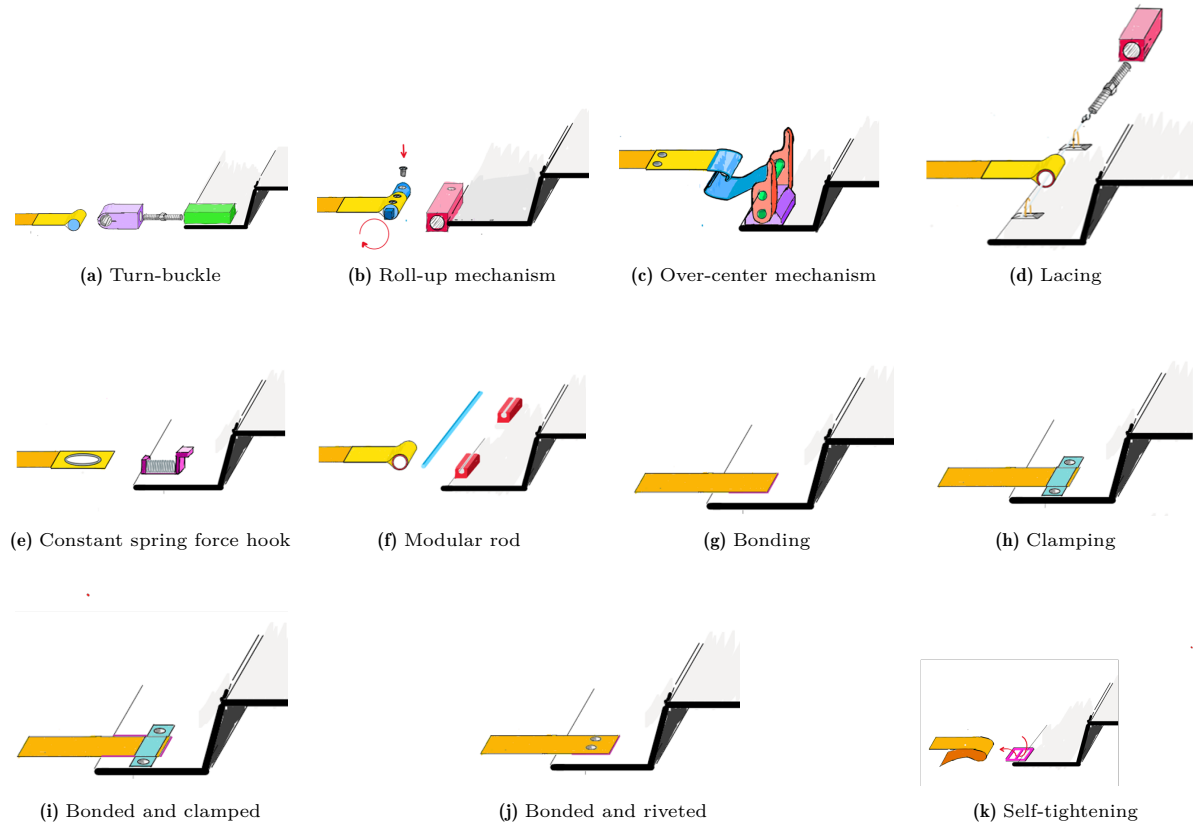


Figure 5.2: Overview of initial tensioning and clamping concepts.

Table 5.5: Concept selection matrix for the clamping interface, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	Bonding	Clamping	Bond & clamp	Self-tighten	Bond & rivet
Manufacturability (0.25)	3	4	2	2	1
Slippage resistance (0.30)	3	2	5	4	3
Mass and footprint (0.20)	5	3	3	2	4
Mechanical redundancy (0.25)	1	1	5	2	4
Final score (weighted total)	2.90	2.45	3.85	2.60	2.95

5.2.3. Prototype Profile Cross-Section Selection

During the initial design as presented in Chapter 4, the selected strawman concept utilized a hollow rectangular cross-section for the lateral profile. However, the issue arises that the allowed height of the lateral profile should be smaller than the hold-down block, as stated in MFP-REQ-008. To ensure an optimal stacking height for the prototype, a hollow rectangular cross-section profile cannot exceed 6mm, which is not an off-the-shelf dimension. Therefore the choice has been made that for the lateral profiles, rectangular strips will be used. To ensure these cross-sections provide enough inertia calculations have been performed, which are presented in Section 5.3.3.

5.3. Sizing calculations

5.3.1. Profile Second Moment of Area Calculations

In Section 5.3.1 the second moment of inertia calculations are provided for all carbon fiber profiles. Firstly, the longitudinal profiles are designed to both resist bending induced by the tensioned mem-

brane and the global bending induced when the full solar array is deployed. Because manufacturing constraints dictate an inner radius of at least 5 mm to ensure proper demolding, calculating the exact analytical second moment of inertia is complex. Therefore, a thin-walled approximation is utilized and subsequently verified against computational values obtained from 3DEXPERIENCE.

The thin walled approximation of the longitudinal stiffener is depicted in Figure 5.3. The origin is located in the y-symmetry line at a height of $9.51mm$. Calculating the second moment of inertia of area is done using the following formula's:

$$I_{xx} = \frac{b \cdot h^3}{12} + A \cdot \bar{y}^2 \quad \& \quad I_{yy} = \frac{h \cdot b^3}{12} + A \cdot \bar{x}^2 \quad (5.1)$$

For segment 23 and 45, the following equations hold for the second moment of inertia:

$$I_{xx} = \frac{h \cdot b^3 \cdot \cos^2(\theta)}{12} + A \cdot \bar{y}^2 \quad \& \quad I_{yy} = \frac{h \cdot b^3 \cdot \sin^2(\theta)}{12} + A \cdot \bar{x}^2 \quad (5.2)$$

Using these equations alongside the thin-walled approximation, the approximate I_{xx} and I_{yy} values are calculated in Table 5.6. From 3DEXperience the inertia's obtained are $I_{xx} = 8323.451mm^4$ and $I_{yy} = 54192.47mm^4$. Comparing the thin-walled approximation with the 3DEXperience I_{xx} is 6.03% lower and I_{yy} 3.17% higher. This relatively small deviation confirms that the values provided by 3DEXperience are accurate and will be used in further calculations.

Table 5.6: Dimensions and calculations corresponding to the second moment of inertia calculations assuming a thin-walled structure as depicted in Figure 5.3.

Section	b[mm]	h[mm]	x_{cg} [mm]	y_{cg} [mm]	$\bar{x}$ [mm]	$\bar{y}$ [mm]	I_{xx} [mm ⁴]	I_{yy} [mm ⁴]
12	13.7	1.6	-27.7	0.8	-27.7	8.7	1668.3	17142
23	20.0	1.6	-16.6	9.9	-16.6	0.3	452.2	9798.7
34	24.8	1.6	0	12.4	0	-9.4	3505.7	2031.3
45	20.0	1.6	16.6	9.9	16.6	0.3	452.2	9798.7
56	13.7	1.6	27.7	0.8	27.7	8.7	1668.3	17141.7
Total							7747	55912

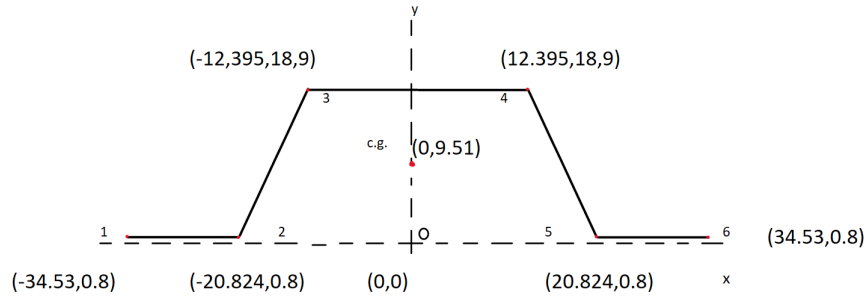


Figure 5.3: Schematic sketch of the thin-walled approximation of the longitudinal hat stiffeners.

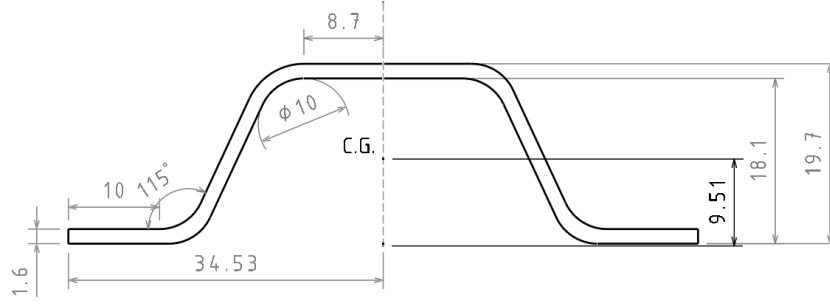


Figure 5.4: True final designed dimensions of the longitudinal hat stiffeners.

For the lateral frame elements also Equation 5.1 can be applied. However, because the lateral profiles consist of only of one solid rectangular section, the center of gravity is always located in the middle of the cross-section. This results in the $\bar{x}$ and $\bar{y}$ to be 0 mm. The lateral frame element consists of three strips bonded to each other and the Kapton membrane. In turn, the calculations for a different number of strips is found in Table 5.7.

Table 5.7: Second moment of Inertia calculations for lateral profile varying over the number of strips

number of strips	b[mm]	h[mm]	$I_{xx}[mm^4]$	$I_{yy}[mm^4]$
1	24	1.6	8.2	1843.2
2	24	3.2	65.5	3686.4
3	24	4.8	221.2	5529.6

5.3.2. Laminate Tensile Properties Calculations

In Section 5.3.2, the macroscopic mechanical properties of the carbon fiber components are estimated. The prepreg utilized for manufacturing the frame elements of the membrane-frame solar array panel is a G200T (Toray FT300B)-DT120-42 fabric laminate. The specific mechanical properties for a single ply are presented in Table 5.8.

Table 5.8: Indicative averaged mechanical characteristics of the GG200T (Toray FT300B)-DT120-42 carbon fiber laminate [48].

Mechanical property	Value
Tensile Strength (0°) (MPa)	693
Tensile Modulus (0°) (GPa)	55.8
Tensile Strength (90°) (MPa)	610
Tensile Modulus (90°) (GPa)	53.7
In-Plane Shear Strength (MPa)	109.1
Shear Modulus (GPa)	3.12

The lay-up pattern chosen for the structural profiles is $[0/90^\circ, \pm 45^\circ, 0/90^\circ, \pm 45^\circ]_s$, which produces a quasi-orthotropic laminate. A quasi-orthotropic lay-up is symmetric and balanced, with principal material axes that align directly with the laminate axes. Consequently, no bending-extension or shear-extension coupling occurs. This lay-up ensures highly predictable stiffness characteristics using Classical Laminate Theory (CLT), simplifies manufacturing, and allows for clean mechanical drilling at attachment points without inducing asymmetric delamination.

To predict the final in-plane stiffness of the laminate, CLT is utilized [49]. First, assuming a major Poisson's ratio of $\nu_{12} = 0.06$, the minor Poisson's ratio is calculated:

$$\nu_{21} = \nu_{12} \cdot \frac{E_2}{E_1} = 0.06 \cdot \frac{53.7 \times 10^9}{55.8 \times 10^9} \approx 0.058 \quad (5.3)$$

The denominator appearing in the reduced stiffness terms is:

$$\Delta = 1 - \nu_{12} \cdot \nu_{21} \approx 0.997 \quad (5.4)$$

The reduced stiffness matrix ($\mathbf{Q}$) of the lamina is defined as:

$$\mathbf{Q} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \quad \text{with} \quad Q_{11} = \frac{E_1}{\Delta}, \quad Q_{22} = \frac{E_2}{\Delta}, \quad Q_{12} = \frac{\nu_{12}E_2}{\Delta}, \quad Q_{66} = G_{12}. \quad (5.5)$$

Filling in the specific material values yields the reduced stiffness matrix:

$$\mathbf{Q} = \begin{bmatrix} 55.99 & 3.233 & 0 \\ 3.233 & 53.89 & 0 \\ 0 & 0 & 3.12 \end{bmatrix} \times 10^9 \text{ Pa} \quad (5.6)$$

To incorporate the contribution of the $\pm 45^\circ$ plies, the $\mathbf{Q}$ matrix for these specific laminates must be transformed with respect to the global x-y coordinate system to obtain $\bar{\mathbf{Q}} = \mathbf{T}^T \cdot \mathbf{Q} \cdot \mathbf{T}$. Assuming $m = \cos(\theta)$ and $n = \sin(\theta)$, the transformation matrix is defined as:

$$\mathbf{T} = \begin{bmatrix} m^2 & n^2 & 2mn \\ n^2 & m^2 & -2mn \\ -mn & mn & m^2 - n^2 \end{bmatrix} \quad (5.7)$$

To obtain the final theoretical Young's moduli, the ABD stiffness matrices must be assembled [49]. Since each individual ply has a thickness of $t_{ply} = 0.195$ mm and the full symmetric laminate consists of 8 layers, the final height becomes $h = 1.56$ mm. Assuming the geometric mid-plane of the layup is $z = 0$ mm, the bottom of the laminate is defined as $z = -\frac{h}{2} = -0.78$ mm. By summing the transformed lamina stiffnesses for each angle, the extensional stiffness matrix ($\mathbf{A}$) is obtained:

$$\mathbf{A} = \sum_{i=1}^8 \bar{\mathbf{Q}}_i \cdot (z_{top} - z_{bot}) = \begin{bmatrix} 67.0 & 24.6 & 0.8 \\ 24.6 & 65.3 & 0.8 \\ 0.8 & 0.8 & 83.1 \end{bmatrix} \text{ GPa} \cdot \text{mm} \quad (5.8)$$

The theoretical macroscopic in-plane stiffnesses are then derived using the following formulas:

$$E_x = \frac{A_{11} \cdot A_{22} - A_{12}^2}{A_{22} \cdot h} = \frac{67.0 \cdot 65.3 - 24.6^2}{65.3 \cdot 1.56} \approx 37.0 \text{ GPa} \quad (5.9)$$

$$E_y = \frac{A_{11} \cdot A_{22} - A_{12}^2}{A_{11} \cdot h} = \frac{67.0 \cdot 65.3 - 24.6^2}{67.0 \cdot 1.56} \approx 36.1 \text{ GPa} \quad (5.10)$$

To verify these manual calculations, Kolibri composite analysis software was utilized. This software also relies on CLT to predict holistic laminate behavior. The utilized lay-up is presented in Figure 5.5, including the program's calculated mechanical properties. Notably, the computed stiffnesses in both the x- and y-directions match the described analytical calculations closely, confirming the correct application of CLT. Factors that could slightly influence the real-world accuracy of these theoretical moduli include:

- Angular alignment inaccuracies introduced during the physical lay-up of each ply.
- Micro-voids or defects introduced during the curing cycle.
- A slight over- or under-estimation of the baseline ν_{12} property.

Layer	Angle (°)	Thickness (mm)	Material
8	0	0.195	GG200T(T300)-DT120-42
7	45	0.195	GG200T(T300)-DT120-42
6	0	0.195	GG200T(T300)-DT120-42
5	45	0.195	GG200T(T300)-DT120-42
4	45	0.195	GG200T(T300)-DT120-42
3	0	0.195	GG200T(T300)-DT120-42
2	45	0.195	GG200T(T300)-DT120-42
1	0	0.195	GG200T(T300)-DT120-42

n : 8	E_x : 39146 MPa
h : 1,56 mm	E_y : 38211 MPa
ρ : 1025,6 kg/m ³	G_{xy} : 14484 MPa
m_A : 1,6 kg/m ²	ν_{xy} : 0.33909
symmetric;	ν_{yx} : 0.33099

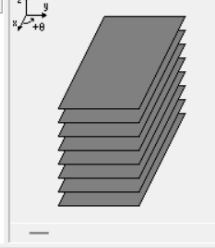


Figure 5.5: CFRP laminate properties obtained and verified via Kolibri composite software.

5.3.3. Lateral Profile Buckling Limit Calculation

A primary structural risk for the prototype is that the lateral CFRP strips may buckle under the membrane's pretension load the moment the external tensioning rig is released. To prevent this catastrophic failure mode, analytical buckling limits must be established. The classic Euler column buckling equation is presented in Equation 5.11. Here, E_{xx} represents the Young's modulus in the x-direction, I_{xx} is the second moment of inertia about the x-axis, K is the effective length factor based on the boundary conditions, and L_{eff} is the physical length of the beam (225 mm). The column effective length factor is designated as $K = 2$, representing the worst-case fixed-free boundary condition, ensuring a highly conservative safety margin.

$$P_{crit} = \frac{\pi^2 \cdot E_{xx} \cdot I_{xx}}{K^2 \cdot L_{eff}^2} \quad (5.11)$$

Inputting the geometric I_{xx} obtained in Section 5.3.1 for the varying number of stacked strips and the elastic E_{xx} from Section 5.3.2, Table 5.9 details the different buckling load thresholds. Because the target membrane line load requires a total compressive force in the magnitude of 200 N, each of the two lateral profiles must safely carry a compressive force of at least 100 N. When enforcing a standard aerospace structural safety factor of ≥ 2.0 , the required number of stacked CFRP strips per lateral profile is determined to be at least 3.

Table 5.9: Euler buckling limit calculations for the lateral profile varying by the number of stacked CFRP strips ($L_{eff} = 225$ mm).

Number of strips	b [mm]	h [mm]	I_{xx} [mm ⁴]	I_{yy} [mm ⁴]	P_{crit} [N]
1	24	1.6	8.192	1843.2	15.6
2	24	3.2	65.536	3686.4	125.2
3	24	4.8	221.184	5529.6	422.5
4	24	6.4	524.288	7372.8	1001.4

5.3.4. Adhesive Joint Shear-Stress Distribution

To safely transfer the 167 N longitudinal pretension load into the lateral CFRP frame, the bonded interfaces must be dimensioned to prevent adhesive failure. For the present calculation, the full 167 N load is conservatively assumed to be transferred through the analysed adhesive interface. In a lap-joint configuration, differential longitudinal deformation of the adherends results in a non-uniform shear stress distribution across the bondline. This phenomenon, commonly referred to as shear lag, causes the shear stress to concentrate near the ends of the overlap rather than being uniformly distributed across the bonded area.

To quantify this stress concentration, Volkersen's analytical shear-lag model for a single-lap joint is employed [50]. The shear-lag parameter λ is defined as:

$$\lambda^2 = \frac{G_a}{t_a} \left(\frac{1}{E_1 t_1} + \frac{1}{E_2 t_2} \right) \quad (5.12)$$

Since the two bonded CFRP adherends are assumed to have identical material properties and thicknesses ($E_1 = E_2 = E$ and $t_1 = t_2 = t$), this expression simplifies to:

$$\lambda = \sqrt{\frac{2G}{E t t_a}} \quad (5.13)$$

For the adopted Volkersen formulation and coordinate convention, evaluation of the shear stress distribution at the overlap boundaries gives the maximum shear-stress concentration factor [50]:

$$k = \frac{\tau_{\max}}{\tau_{\text{ave}}} = \frac{\lambda l}{2 \sinh(\lambda l)} [\cosh(\lambda l) + 1] \quad (5.14)$$

The calculation uses the material and geometric properties of the selected 3M DP490 structural epoxy and CFRP hat-stiffeners:

- G : shear modulus of the DP490 adhesive (669.2 MPa).
- E : longitudinal Young's modulus of the CFRP (37.0 GPa = 37,000 MPa).
- t : thickness of the CFRP adherend (1.6 mm).
- t_a : adhesive bondline thickness (0.1 mm).
- l : designed overlap length (10 mm).

The shear modulus of the 3M DP490 adhesive is estimated from its Young's modulus and Poisson's ratio using the standard isotropic relationship [51]:

$$G = \frac{E_a}{2(1+\nu)} = \frac{1.847 \times 10^9}{2(1+0.38)} \approx 669.2 \text{ MPa} \quad (5.15)$$

Substitution of the material and geometric properties gives:

$$\lambda = \sqrt{\frac{2(669.2)}{37000(1.6)(0.1)}} = \sqrt{\frac{1338.4}{5920}} \approx 0.4755 \text{ mm}^{-1} \quad (5.16)$$

The corresponding shear-stress concentration factor is:

$$k = \frac{0.4755(10)}{2 \sinh(0.4755(10))} [\cosh(0.4755(10)) + 1] \approx 2.42 \quad (5.17)$$

Thus, under the assumptions of the Volkersen model, the maximum shear stress at the overlap boundaries is approximately 2.42 times the average shear stress. The average shear stress is calculated from the applied load, bond width, and overlap length as:

$$\tau_{\text{ave}} = \frac{F_{\max}}{bl} = \frac{167}{24(10)} \approx 0.696 \text{ MPa} \quad (5.18)$$

Applying the calculated stress concentration factor gives the peak adhesive shear stress:

$$\tau_{\text{peak}} = k \tau_{\text{ave}} = 2.42(0.696) \approx 1.68 \text{ MPa} \quad (5.19)$$

The calculated peak shear stress of 1.68 MPa is substantially below the quoted ultimate lap-shear strength of 36.1 MPa for DP490 [52]. The corresponding static strength ratio is:

$$SF = \frac{\tau_{\text{ultimate}}}{\tau_{\text{peak}}} = \frac{36.1}{1.68} \approx 21.5 \quad (5.20)$$

Equivalently, using the conventional definition of margin of safety:

$$MS = SF - 1 = \frac{36.1}{1.68} - 1 \approx 20.5 \quad (5.21)$$

Therefore, under the assumptions of the analytical shear-lag model, the 10 mm overlap provides substantial static shear strength against the calculated peak adhesive stress. This result demonstrates that the proposed overlap length has sufficient analytical capacity against localized adhesive shear failure for the considered 167 N pretension load. It does not, however, account for other potential failure mechanisms such as peel stresses, adhesive defects, environmental degradation, fatigue, or manufacturing-induced variations in bondline thickness.

The effective bonded area is also marginally reduced by the presence of slotted assembly holes within the overlap region. Although these voids are expected to fill with adhesive during manufacturing and may consequently contribute to mechanical load transfer, their detailed load-transfer contribution is not quantified in the present analytical model. The effect of the slots is therefore conservatively neglected. Given the calculated static strength ratio of approximately 21.5, the resulting peak shear stress remains substantially below the quoted ultimate lap-shear strength even without crediting any additional load transfer through the filled slots.

5.3.5. Finite Element Analysis of UD Tensioning

To determine the required mechanical force for achieving a target membrane pretension of 500 N/m, a Finite Element Model was developed in Abaqus. Rather than artificially inducing membrane tension via a predefined temperature field (as described in Section 4.1), this model iteration physically simulates the exact mechanical tensioning procedure used during manufacturing.

Geometry simplifications and element selection

To ensure computational efficiency while capturing the general force-tension relationship of the frame, several geometrical simplifications are implemented in the Finite Element Model. The model consists of a 245 × 275 mm continuous surface, of which the edges are assigned as stringers. In reality the hold-down coordinates are slightly inward at each corner at 225 × 275 mm. Physically, the membrane only spans from flange to flange of the opposing hat profiles, giving a final dimension of 225 × 245 mm. This initial geometric misalignment is deemed acceptable for the first global model because the main goal remains to obtain insight into the macroscopic force required to achieve a pretension of approximately 500 N/m in the x-direction.

Similar to Section 4.1, the element selection for the membrane is M3D4R (4-node quadrilateral membrane element) and for the ribs, B21 (2-node linear beam element). The choice for the M3D4R elements is made because 50 μm Kapton has zero bending stiffness. If shell elements (S4R) were chosen, Abaqus could try to calculate bending moments across the cross-section, which often could lead to numerical instabilities and in turn convergence issues. For the stiffeners 2-node linear beams are chosen because it effectively allows for assigning section properties to the edges. The reason for not choosing B22 is to reduce the number of nodes and elements required assuring for little computational time of the problem below the 1000-node Abaqus Student Edition limitation.

Material and section assignments

The material and section chosen for the profiles is as described in Section 5.3.1 and Section 5.3.2. For the longitudinal hat profile, a custom cross-section is generated (utilizing the Abaqus *Arbitrary* profile type) based on the thin walled coordinate points presented in Table 5.6 and Figure 5.3. It should be noted that the corner radii are not considered in the formulation of the section. The section of the lateral profile consists of three integrated strips, which together create a rectangular cross-section. The

dimensions and inertia properties of the lateral profiles are presented in Table 5.7. The final material properties utilized by the profiles are presented in Figure 5.5.

Finally, The Polyimide (Kapton HN) film is assigned a membrane section with a thickness of 0.0508 mm. The corresponding material properties are presented in Table 5.3 with a Poisson's Ratio of 0.34.

Boundary conditions and sequential loading protocol

To capture the final stress state of the prototype, the Finite Element Model needs to be modeled adhering to the manufacturing process. Because thin films undergo large deformations and rely on stress-stiffening to carry out-of-plane loads, nonlinear geometric effects (NLGEOM) are enabled throughout this simulation. Also, the stiffness required to retain the membrane tension, is only added after the tension is applied externally. To ensure this is also captured, the lateral profile elements need to be deactivated during the loading step of the membrane. To capture this manufacturing sequence accurately, the Abaqus *Model Change* capability is utilized. The following steps are performed to obtain the final model:

1. **Initial State & Element Deactivation:** The left edge of the frame is initially fully clamped (*encastre*) to anchor the global structure. Before any loads are applied, the lateral profiles (top and bottom edges) are deactivated (as depicted in Figure 5.6a). This temporarily removes their structural stiffness from the global matrix to ensure the membrane can deform independently during tensioning.
2. **Step 1: Pretension Application:** With the lateral ribs removed, the tension load is applied on the right longitudinal profile (as depicted in Figure 5.6b). The force is applied at two points, each located at the 25% and 75% span locations of the right longeron. This is done to limit the right longeron to bend during the loading process. These loads are applied using an inward-pointing vector with components $(50\sqrt{3}, \pm 50)$, actively guiding the profile. To ensure numerical stability, the following temporary boundary conditions are applied: out-of-plane translation is locked along the lateral edges ($U_3 = 0$), and the right profile is constrained against twisting and transverse sliding ($U_2, U_3, UR_1, UR_2 = 0$).
3. **Step 2: Element Reactivation (The "Birth" Phase):** Once the membrane reaches its target pretension, the lateral profile elements are reactivated in a strain-free state. This analytically locks the deformed geometry into its final form as seen in Figure 5.6c.
4. **Step 3: Self-Equilibrating Settling:** With the frame fully assembled, the temporary pulling loads and kinematic guide constraints are removed. A fully fixed boundary condition (*encastre*) is applied at the right geometric corners. The model is allowed to settle into a new static equilibrium, representing the true, self-equilibrated state of the fully assembled physical prototype under internal pretension. This step is depicted in Figure 5.6d.
5. **Step 4: Modal Extraction (Linear Perturbation):** To extract the fundamental structural dynamics, a linear perturbation frequency step is initiated. The previous settling boundary conditions are deleted, and all four geometric corners of the frame are fully clamped (*encastre*). This specific boundary configuration perfectly replicates the physical test setup where the prototype will be mounted for the experimental laser vibrometer testing.

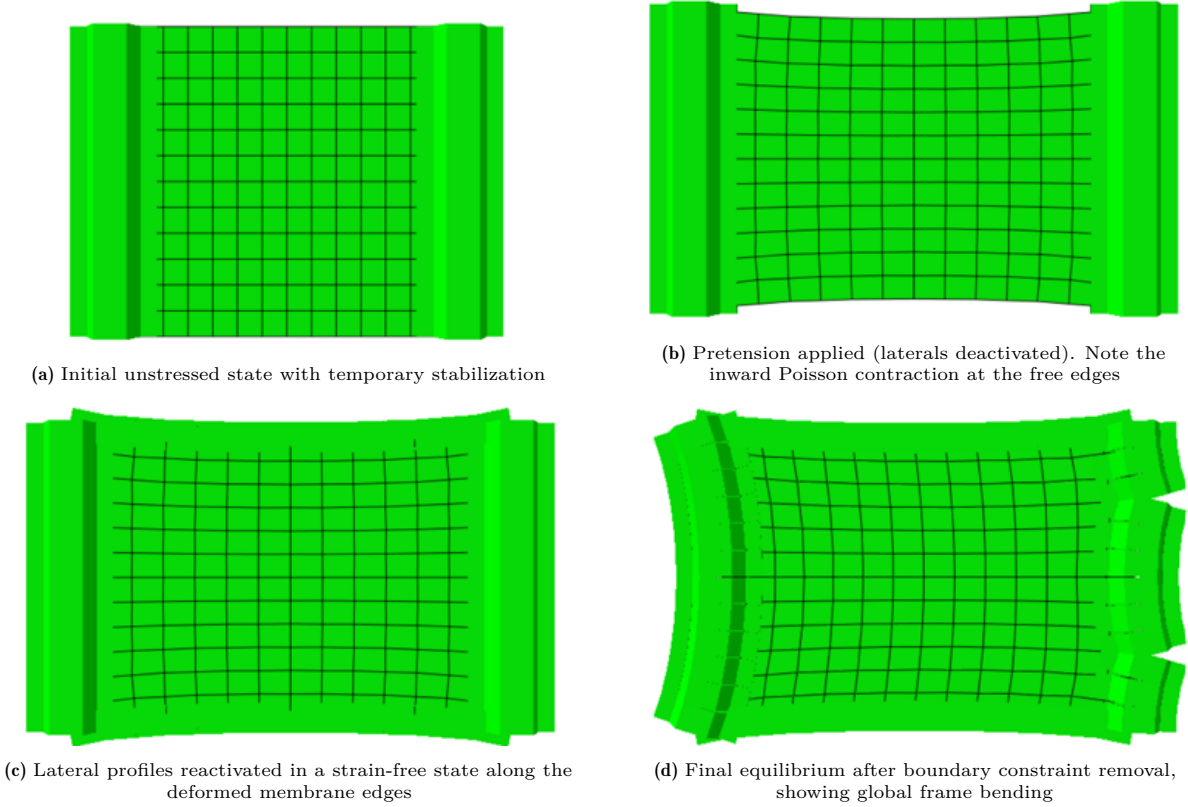


Figure 5.6: Abaqus simulation sequence illustrating the birth-and-death loading protocol used to replicate the physical prototype assembly and tensioning process. Deformations are scaled by a factor of 50 for clarity.

Mesh generation and convergence

To obtain the final mesh, a balance between the macroscopic kinematic accuracy and numerical robustness was required. In general, by minimizing the element size, the finite element accuracy improves. However, a highly flexible membrane structure undergoing large geometric deformations presents additional numerical challenges.

After performing a mesh sensitivity study, the global mesh density of 20×20 elements was found to provide the best compromise. Smaller meshes resulted in the numerical model to not run while coarser meshes exhibited artificially stiff behavior, underestimating the non-linear behavior.

During longitudinal pretension, Poisson contraction generates transverse compressive stresses within the membrane. Because the M3D4R elements do not possess a bending stiffness, fine meshes are susceptible to local instabilities, resulting in local out-of-plane buckling. As a results, the stiffness is lost in these elements which could result in the global stiffness matrix to not further solve the problem and in turn aborts. Furthermore, the concentrated loads applied through the longeron could also create stress singularities at the load introduction, as seen in Figure 5.8. As the mesh is refined, these singular regions become more pronounced, which could lead to convergence issues.

Pretension results and required force

The primary objective of the finite element model is to obtain the exact external loading conditions to obtain a pretension of 500 N/m . Capturing both the stress-stiffening of the Kapton film and the bending behavior of the frame requires accounting for geometric non-linearities. Consequently, external forces are applied iteratively to resolve these deformations before establishing the final equilibrium state.

As illustrated in Figure 5.7, three distinct load iterations (160 N, 180 N, and 200 N total pulling force) were simulated. For each iteration, the longitudinal line tension (T_x) was extracted along a transverse path spanning the mid-plane of the membrane. The resulting distributions exhibit a parabolic "U-shape". This non-uniformity of the mid-plane tension is caused by the inward bending of the longerons,

at the moment the external force is removed and the corresponding boundary conditions during the "Settling" step. The maximums near the edges are resulted by the presence of the lateral profiles which constraint the movement and thus retains the applied stress the best.

To account for this structural flexibility, the true macroscopic line load must be represented by the cross-sectional average (indicated by the dashed lines in Figure 5.7). By interpolating between the bounded averages of the 160 N and 180 N iterations, the exact required pulling force to achieve the 500 N/m equilibrium state was mathematically isolated to 167 N total (applied as 83.5 N per point).

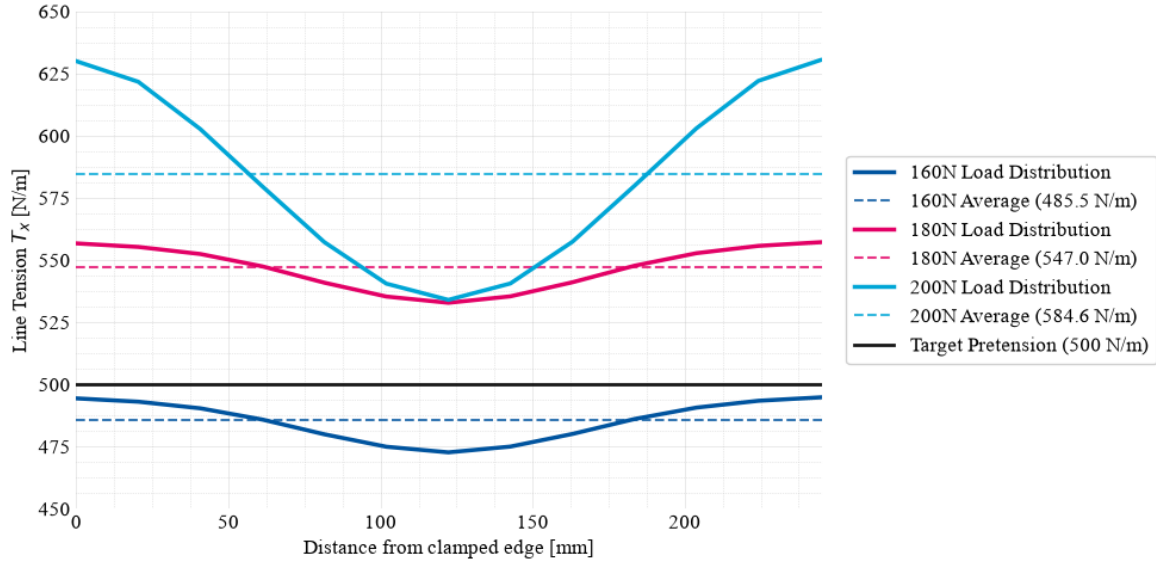


Figure 5.7: Mid-plane longitudinal line tension (T_x) distribution for varying applied loads. The dashed lines indicate the cross-sectional averages used to interpolate the final required force for the 500 N/m target.

Applying this converged 167 N load yields the final simulated stress state depicted in Figure 5.8. What is notable is that local stress concentrations are located at the loading points. However, around the mid-plane, the σ_{11} stress field becomes much more uniform. At a thickness of $t = 0.0508$ mm, this central stress field aligns perfectly with the target 9.84 MPa, successfully confirming the 500 N/m baseline pretension requirement.

It must be noted that the severe stress concentrations along the right edge are an artifact of applying concentrated nodal loads within the finite element framework. Physically, the tension is introduced via steel wires that act along identical loading vectors; however, the load is transferred into the hat stiffener through mounting clamps. This physical interface ensures the macroscopic pulling force is distributed across a broader contact area, preventing the localized singularities observed in the numerical model.

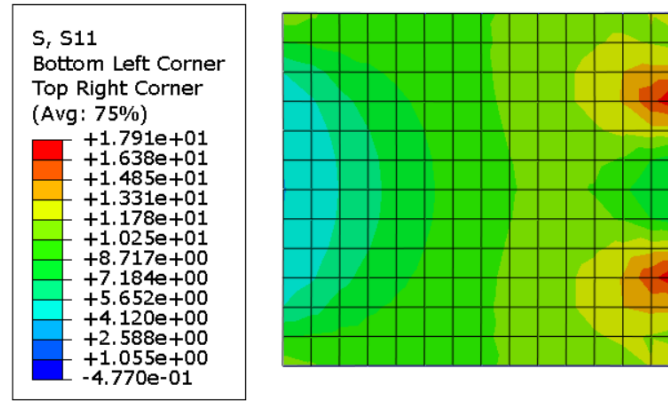


Figure 5.8: Final longitudinal stress distribution (S_{11}) under the converged 167 N total load. The uniform central region demonstrates a steady-state stress field of approximately 9.84 MPa, confirming the target pretension.

5.4. Final prototype design

5.4.1. As-designed configuration

Based on the sizing calculations presented in Section 5.3, the theoretical "as-designed" configuration of the prototype is established. A distinction is made between this theoretical baseline and the physical "as-built" configuration, as manufacturing tolerances and constraints necessitated minor physical adaptations during assembly. A comprehensive assembly plan to realize this configuration is provided in Appendix C.

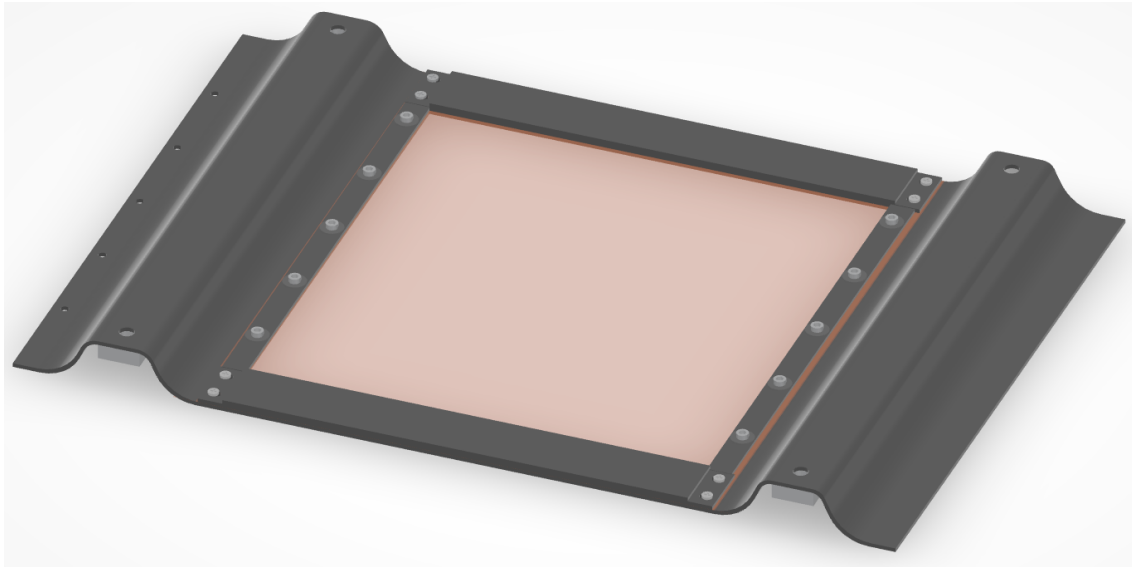


Figure 5.9: Complete as-designed CAD assembly of the membrane-frame prototype.

To physically replicate the sequential loading protocol described in Figure 5.6, the lateral profiles accommodate slotted holes with a length of 1.5 mm each. These slotted holes (presented in Figure 5.10) allow the lateral profiles to be attached during the loading process without carrying any longitudinal pretensioning loads. Although the maximum theoretical membrane deformation is calculated to be approximately 1.6 mm, the slotted holes provide a 1.5×2.0 mm tolerance to accommodate potential manufacturing variations. Similarly, the Kapton membrane is designed such that, upon reaching its target elongation, the holes align precisely with the testing rig.

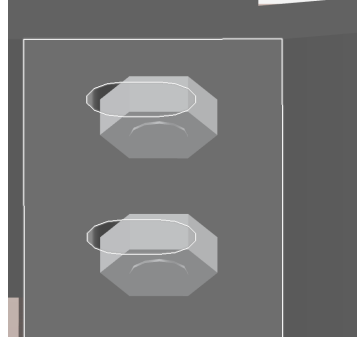


Figure 5.10: Detail view of the slotted-hole feature along the lateral profile, designed to provide kinematic clearance during tensioning.

A comprehensive synthesis of the finalized geometric, material, and structural parameters for this as-designed state is provided in Table 5.10.

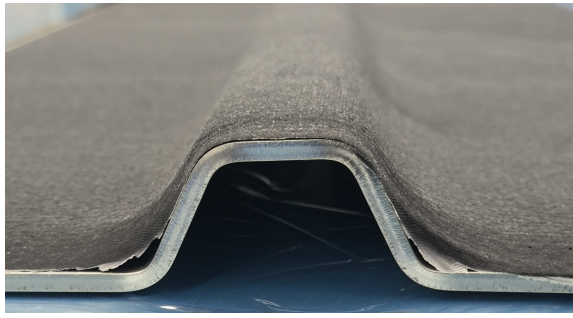
Table 5.10: Synthesized specifications of the as-designed prototype architecture.

Subsystem	Parameter	Design Value
Membrane	Material	Kapton HN
	Thickness (t)	0.0508 mm
	Target Pretension (T_x)	500 N/m
Frame Structure	Material	CFRP
	Longeron Profile Geometry	Hat-Stiffener
	Lateral Profile Geometry	3 Rectangular strip stiffeners
Integration	Nominal required load	167 N
	Fastening Method	Friction-clamped and bonded (DP490)
	Transverse Interface	Slotted clearance holes
	Kinematic mechanism	Under-dimensioned elastic strain

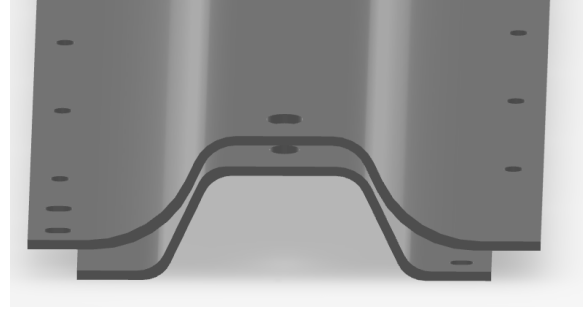
5.4.2. As-built Configuration and Manufacturing deviations

While the designed parameters provide a theoretical baseline, the physical fabrication of the prototype components required specific adaptations. This section highlights these discrepancies, as each geometric or material variance directly influences the subsequent dynamic correlation.

The first notable deviation occurred during the fabrication of the longitudinal CFRP hat-stiffeners. As illustrated in Figure 5.11, the physical manufacturing process yielded a larger effective corner radius than the idealized CAD geometry (an initial 5 mm versus a true 18 mm). This larger radius was necessitated by the tooling setup; a male mold (or tube) was required to assist the vacuum bag in exerting sufficient compaction pressure to conform the prepreg to the lower mold. Consequently, the profile width increased by 16.5 mm (from 69 mm to 85.5 mm). To maintain the final global hold-down coordinates of 275×225 mm, the widths of the lateral profiles, reinforcement strips, and the membrane were correspondingly reduced by 16.5 mm.



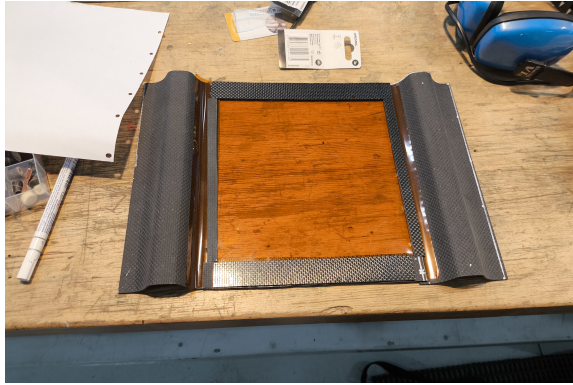
(a) Physical manufacturing radius of the CFRP longeron



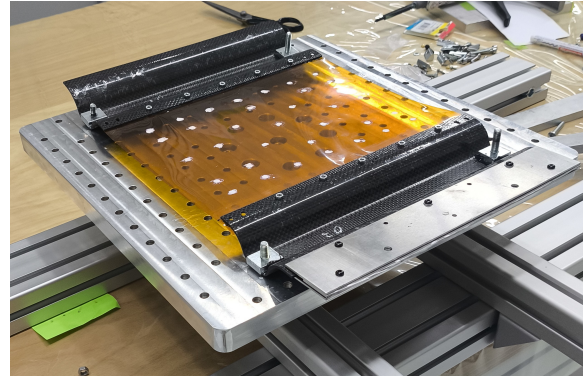
(b) Comparison of the designed CAD radius versus the physical as-built profile

Figure 5.11: Geometric deviations in the CFRP profile radii resulting from the physical lay-up process.

Following the frame assembly (Figure 5.12), secondary integration deviations occurred at the fastener interfaces. In the idealized numerical model, the structural adhesive layers possess negligible volume. However, the physical bond-line of the DP490 epoxy created an unexpected increase in joint thickness, preventing the standard washers from fitting. Additionally, while 2.5 mm diameter bolts were initially specified for the slotted holes, these were substituted with the 3.0 mm bolts used at the membrane-frame interface to ensure hardware commonality.



(a) Initial prototype frame prior to hole integration



(b) Semi-finished prototype assembly

Figure 5.12: Progression of the physical prototype manufacturing and frame assembly.

Finally, due to machining constraints, the critical lateral slotted holes were fabricated using manual rotary tooling rather than precision CNC milling. This manual process introduced localized micro-misalignments and geometrical imperfections along the hole perimeters. While these geometric variances do not compromise the macroscopic 500 N/m tension field, they introduce localized mass and stiffness asymmetries into the final structure.

Assembly Rig Development and Prototype Assembly

6.1. Assembly Rig Purpose, Functions, and Requirements

6.1.1. Role of the Assembly Rig

The primary function of the assembly rig is to impose strict kinematic boundary conditions on the semi-finished prototype during the physical tensioning phase. To accurately replicate the idealized longitudinal displacement (U_1) modeled in Section 5.3.5, the assembly rig provides a rigid anchor for one hat stiffener while the opposing stiffener is displaced using the MTS Exceed E43 tensile tester. Beyond this, the rig architecture was driven by three critical risks related to the tensioning process: parasitic bending moments, lack of kinematic isolation, and lack of tooling rigidity.

The first risk arises when the force applied to the membrane is not purely uniaxial due to the presence of a small vertical offset (δ_f), as depicted in Figure 6.1. If this offset height becomes too large, unwanted bending moments are introduced. This would result in the twisting of the CFRP hat stiffeners, which could hinder achieving the 500 N/m target pretension field (see MFP-REQ-001).

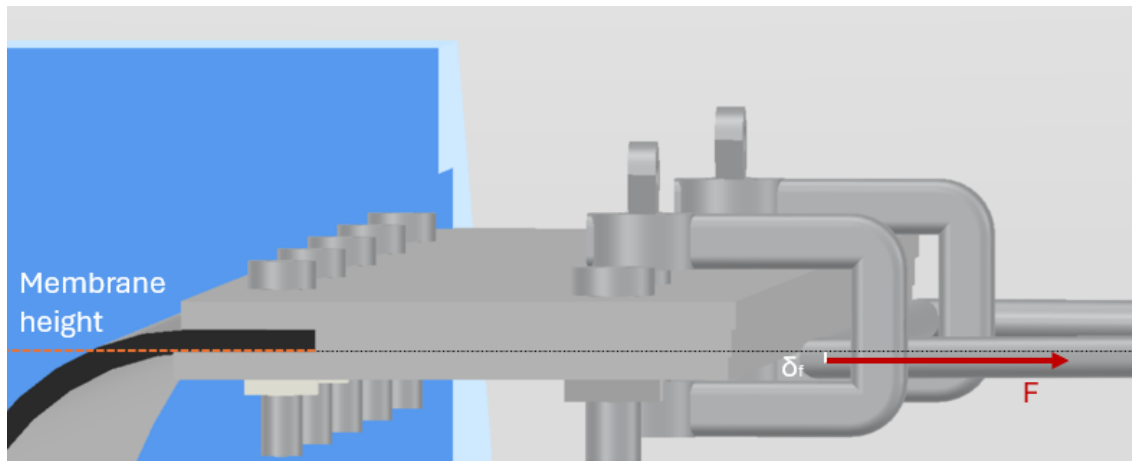


Figure 6.1: Misalignment of the tensile force vector with the membrane mid-plane introduces parasitic bending moments.

The assembly rig must also ensure the kinematic isolation of all directions except for the primary loading direction (U_1). If the prototype is not constrained in the other transverse directions (U_2 & U_3), as well as in all its rotational degrees of freedom, the applied force will not exclusively affect the longitudinal axis. This lack of isolation would severely complicate the correlation with the finite element model, requiring excessive iterations to reach the target 500 N/m pretension (MFP-REQ-001).

The final risk relates to force tracking. To precisely monitor the force-displacement curve, the structural stiffness of the rig components must be significantly higher than that of the Kapton film. Preliminary

tests (described in Section 6.4) revealed that the Kapton exhibits stress relaxation after reaching its target pretension. By ensuring the absolute rigidity of the assembly rig, this relaxation phenomenon can be accurately isolated and factored into the initial force overshoot required to achieve the final equilibrium state.

6.1.2. Functional Requirements

After determining the major risks and functions of the assembly rig, the specific functional and kinematic requirements were established. These requirements are presented in Table 6.1. The following considerations governed the formulation of this matrix:

- **Regarding ASR-REQ-003:** While a perfect theoretical offset of $\delta_f = 0$ mm is preferred, anything within a ± 1 mm tolerance is deemed acceptable. This alignment will be physically inspected prior to applying the final system load.
- **Regarding ASR-REQ-004:** To ensure the longest structural members of the assembly rig do not elastically deform under load, the theoretical axial stiffness ($K = \frac{EA}{L}$) of candidate materials will be analytically verified.
- **Regarding ASR-REQ-006:** The corresponding dimensions onto which the assembly rig will be mounted are presented in Figure 6.2a. Additionally, the bottom mounting support, depicted in Figure 6.2b, must be accounted for, as it physically obstructs the area enclosed by the four M10 mounting holes.

Table 6.1: Assembly Rig Functional and Kinematic Requirements

ASSEMBLY RIG REQUIREMENTS				Verification Type				Rationale/
Req. ID	Req. Statement	Value	Type	A	I	D	T	Comment
ASR-REQ-001	The assembly rig shall provide a fully rigid, hard-mounted anchor for the static CFRP hat stiffener.	$U_{1,2,3} = 0$ $UR_{1,2,3} = 0$	Key	✓	✗	✓	✗	Replicates the static boundary condition from the FEA model.
ASR-REQ-002	The rig shall constrain the moving hat stiffener in all transverse and rotational degrees of freedom, permitting only longitudinal translation.	$U_1 = \text{Free}$ $U_{2,3} = 0$ $UR_{1,2,3} = 0$	Key	✓	✗	✓	✗	Ensures kinematic isolation and pure uniaxial membrane tension.
ASR-REQ-003	The tensile load vector applied by the rig shall be strictly co-planar with the membrane mid-plane to prevent eccentric loading.	$\delta_f \approx 0 \pm 1$ mm	Key	✓	✓	✗	✗	Prevents parasitic bending moments and torsion in the CFRP profiles.
ASR-REQ-004	The structural stiffness of the rig in the loading direction shall be significantly greater than the longitudinal stiffness of the Kapton membrane.	$K_{\text{rig}} \geq 100 \cdot K_{\text{mem}}$	Driver	✓	✗	✗	✓	Eliminates tooling compliance to accurately measure polymer relaxation.
ASR-REQ-005	The rig architecture shall safely withstand the target operational tension force plus an initial force overshoot without yielding.	$\geq 1.5 \times 200$ N	Killer	✓	✗	✗	✓	Accommodates the ~50% overshoot required to offset relaxation.
ASR-REQ-006	The rig shall provide a direct mechanical interface for attachment to the MTS Exceed E43 universal testing machine.	see Figure 6.2	Constr.	✗	✓	✓	✗	Utilizes available laboratory equipment for controlled longitudinal displacement.
ASR-REQ-007	The rig shall be supported by a table with a height of 900 mm.	900 mm	Constr.	✗	✗	✓	✗	Aligns with standard laboratory workbench height for ergonomic assembly and operation.
ASR-REQ-008	The rig shall ensure a mounting interface for 6 mm diameter bolts spaced 225 mm apart.	225 mm pitch	Constr.	✗	✗	✓	✗	Ensures physical compatibility with the baseplate and MTS machine hardpoints.

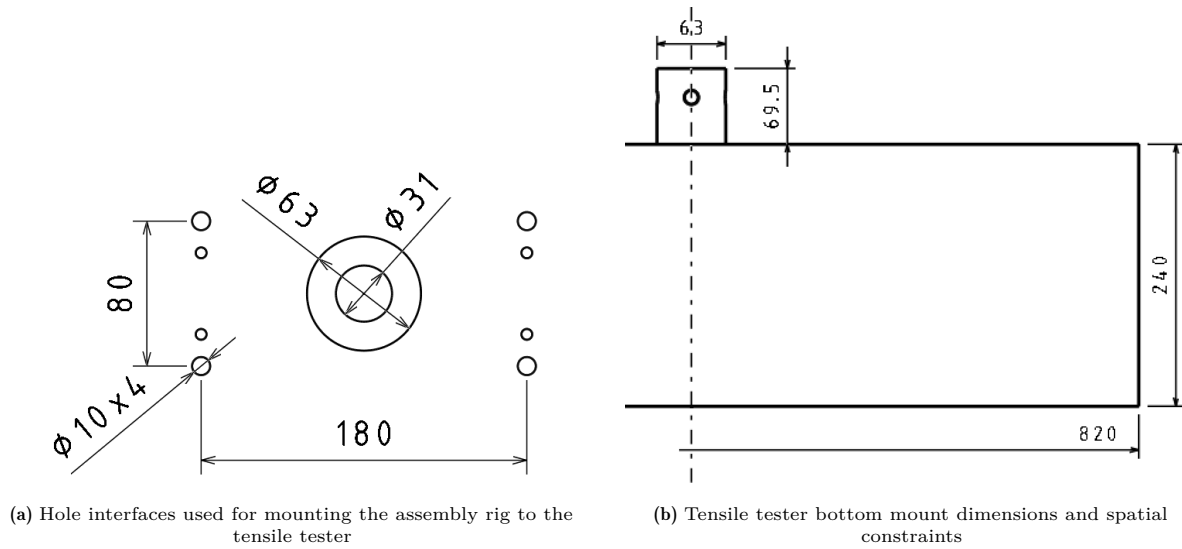


Figure 6.2: Mechanical interfaces and spatial constraints of the MTS Exceed E43 tensile tester used for rig integration.

6.2. Rig Architecture and Material Selection

6.2.1. Assembly rig frame

To mechanically interface the assembly rig with the MTS Exceed E43 tensile tester, which operates at a standard workbench height of 900 mm (satisfying ASR-REQ-006 and ASR-REQ-007), a vertical clearance offset of 40 mm must be maintained. Furthermore, because the bottom mount of the tensile tester obstructs the central area between the four M10 mounting holes, a perimeter beam structure was determined to be the most viable architectural solution for the rig frame.

To select an appropriate material for these structural beams, the axial stiffness (K_{rig}) of three candidate materials was evaluated against requirement ASR-REQ-004. The candidates considered were: Spruce pine wood (44×44 mm solid cross-section), Aluminum 6063-T6 Item Profile 8 (40×40 mm extrusion), and 304 Stainless Steel hollow tubing (40×40 mm with a 3 mm wall thickness).

The calculated axial stiffness parameters for the three candidates are compared in Table 6.2. The analytical verification shows that all three options successfully fulfill ASR-REQ-004 ($K_{\text{rig}} \geq 100 \cdot K_{\text{mem}}$). However, while the spruce pine wood only marginally clears the requirement threshold (ratio of 121), both the stainless steel tubing and the aluminum profiles surpass the stiffness requirement by an additional factor of 4. Notably, the overall structural stiffness of the aluminum and steel candidates is virtually identical. Although 304 stainless steel possesses a significantly higher Young's Modulus, its thin-walled hollow geometry yields a lower cross-sectional area compared to the complex webbing of the aluminum extrusion, equalizing their macroscopic rigidity.

Table 6.2: Axial stiffness verification of candidate assembly rig materials compared to the Kapton HN membrane to satisfy requirement ASR-REQ-004.

Material	E (GPa)	A (m ²)	L (m)	K (N/m)	Ratio (K/K_{mem})
Kapton HN (Membrane) Dupont2008	2.50	1.24×10^{-5}	0.197	1.58×10^5	1.0
Pine wood (Spruce) [53]	6.90	1.94×10^{-3}	0.700	1.91×10^7	121
Item Aluminum 6063-T6 [54][55]	68.9	6.46×10^{-4}	0.700	5.06×10^7	403
304 Stainless Steel [56]	193.0	2.31×10^{-4}	0.700	6.60×10^7	403

Knowing that the stiffness ratio of all inspected options is acceptable, a formal trade-off was conducted to determine the most practical solution. The selection criteria were defined as follows:

- **Stiffness ratio (Weight 0.40):** The magnitude by which the structural stiffness of the rig exceeds the stiffness of the Kapton membrane. A higher factor correlates directly to minimized tooling deformation and higher fidelity during force-displacement measurements.
- **Manufacturing & Assembly effort (Weight 0.30):** The complexity and number of supplementary steps required to cut elements to size, machine mounting interfaces, and integrate the final rig assembly.
- **Availability and cost (Weight 0.30):** The logistical ease of attaining the components (including existing in-house inventory) and the relative financial expenditure required.

The results of the trade-off are presented in Table 6.3. The aluminum Item profiles emerged as the optimal candidate for the assembly rig architecture. These extrusions provide an excellent balance of extreme rigidity and modularity, easily integrating with standard corner brackets and sliding T-nuts. Furthermore, the majority of the required profiles were already available within the laboratory inventory, requiring only the procurement of specific corner connectors.

While pine wood presented a highly cost-effective and accessible alternative, its stiffness margin was minimal, and every beam would necessitate custom cutting, drilling, and precision alignment. Similarly, stainless steel hollow tubes matched the rigidity of the aluminum option. However, they carry a higher material cost and introduce severe manufacturing challenges regarding custom cutting and drilling compared to the modular T-slot system.

Table 6.3: Material selection matrix for the assembly rig, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	Pine wood (Spruce)	aluminum 6063-T6	304 Stainless Steel
Stiffness ratio K/K_{mem} (0.40)	3	5	5
Manufacturing & Assembly effort (0.30)	3	5	3
Availability and cost (0.30)	4	4	2
Final score (weighted total)	3.30	4.70	3.50

6.2.2. Tensioning interface and mechanism design

With the static boundary conditions secured by the main assembly rig frame, a dedicated tensioning system was required to safely transfer the longitudinal load (U_1) from the MTS machine to the moving CFRP hat stiffener. The tensioning system encompasses all interacting components that ensure the Kapton membrane obtains and maintains its final uniform pretension. To achieve this kinematic load transfer, the system is decomposed into four functional parts:

- **Stiffener-to-wire connector:** A modular clamp that connects the unconstrained CFRP hat stiffener to the tensioning wires.
- **Tensioning wire:** A flexible steel cable connecting the hat stiffener to the tensile tester to exert the required force.
- **Pulley bridge:** A mechanism to translate the vertical displacement of the tensile tester head horizontally, strictly in-line with the Kapton thin-film.
- **Support bracket:** A structural mount that ensures the correct dimensional placement and stability of the pulley mechanism.

Stiffener-to-wire connector design

For the stiffener connector, it is essential that the attachment to the hat stiffener is fully reversible. The prototype must not have any unnecessary mass or non-flight representative hardware permanently bonded to it during the subsequent dynamic modal analysis. Therefore, a friction-clamped mechanical fastener approach on the outer flange was deemed the only viable method. Two candidate clamping mechanisms were considered that align with the co-planarity requirement (ASR-REQ-003): an aluminum plate clamping mechanism using lever clasps (Figure 6.3a), and a 3D-printed PLA housing reinforced with an internal aluminum tube (Figure 6.3b).

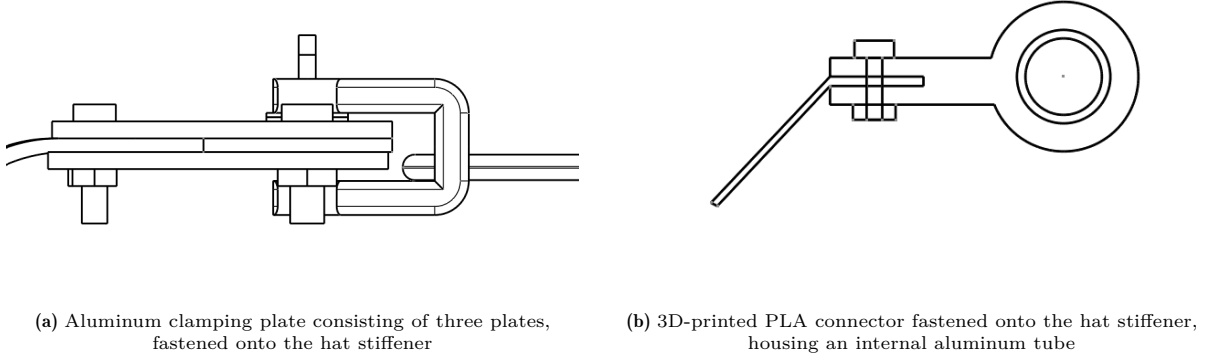


Figure 6.3: Candidate stiffener-to-wire connector architectures.

The philosophy behind the aluminum candidate is to create a double lap joint over the stiffener using flat aluminum plates to build up the required geometric thickness. By adding two lever clasps at 25% and 75% of the span, the tensioning wire can be routed such that the load transfer vector operates within a ± 1 mm vertical offset of the Kapton membrane, successfully fulfilling ASR-REQ-003. Analytically, the structural stiffness (K) of this three-plate aluminum assembly heavily exceeds the membrane stiffness:

$$K_{\text{al,clamp}} = \frac{68.9 \times 10^9 \cdot (3 \cdot 0.002 \cdot 0.245)}{0.04} = 2.53 \times 10^9 \text{ N/m} \gg K_{\text{memb}} \quad (6.1)$$

Alternatively, the 3D-printed PLA candidate minimizes manufacturing assembly steps while theoretically achieving a high level of precision to align the steel wire. To prevent the steel wire from slicing through the softer PLA polymer under load, an aluminum tube is housed inside the print to distribute the bearing stress. Because arbitrary thickness can be added to a 3D print with minimal effort, the axial stiffness can be scaled to match the aluminum counterpart:

$$K_{\text{pla,clamp}} = \frac{2.75 \times 10^9 \cdot (0.245 \cdot 0.02)}{0.04} = 3.37 \times 10^9 \text{ N/m} \gg K_{\text{memb}} \quad (6.2)$$

While both options satisfy the structural stiffness requirement (ASR-REQ-004), a critical discrepancy exists regarding their shear strength at the fastener interfaces. Standard PLA filament exhibits a shear strength of approximately 25 MPa [57], whereas Aluminum 6063-T6 yields at approximately 207 MPa [54]. Under the 167 N operational load, the nominal shear stresses remain well below the failure limits for both materials. However, due to the connector's flat geometry and the highly concentrated bearing loads at the fasteners, aluminum provides roughly eight times greater resistance to localized yielding. A concise trade-off (Table 6.4) confirms that the aluminum clamping plates provide the most reliable, high-stiffness solution for this specific component. In this case, availability and cost were not selected as trade-off criteria because both PLA filament and aluminum plates were readily available in-house.

Table 6.4: Concept selection matrix for the stiffener-to-wire connector.

Criterion (Weight)	Aluminum Plates	3D-Printed PLA + Tube
Shear & bearing strength (0.40)	5	2
Manufacturability & assembly (0.30)	3	5
Co-planar precision (0.30)	4	4
Final score (weighted total)	4.10	3.50

Pulley Bridge

The pulley bridge interfaces directly with the tensile tester, ensuring the tensioning wires align precisely with both the moving crosshead and the membrane mid-plane. Because the tensile tester's bottom mount obstructs the area between the M10 mounting holes, this component requires a spanning architecture (a bridge) to bypass the physical interference.

For this component, two manufacturing paradigms were considered: a 3D-printed custom polymer bridge and a custom CNC-machined aluminum block. Similar to the logic in Section 6.2.2, the 3D-printed component has the advantage of highly precise geometric tunability (with a layer-height accuracy of ± 0.3 mm). Additionally, when the print orientation is optimized, limited support material is required, resulting in a rapid manufacturing process.

The primary structural disadvantage of an FDM 3D-printed component is its susceptibility to bending moments, as depicted in the free-body diagram in Figure 6.4. As the tensioning wire routes through a 90-degree turn over the pulley, the mechanism is subjected to a horizontal force from the membrane (F_{Tx}) and a vertical force from the MTS machine (F_{Ty}), where nominally $F_{Tx} \approx F_{Ty} = 167$ N. To maintain static equilibrium at the base fasteners (Point C), the following conditions must be satisfied:

$$\sum F_x = R_x - F_{Tx} = 0 \quad (6.3)$$

$$\sum F_y = F_{Ty} - R_y = 0 \quad (6.4)$$

$$\sum M_C = M_R + (F_{Tx} \cdot \delta_x) + (F_{Ty} \cdot \delta_y) = 0 \quad (6.5)$$

Where R_x and R_y are the required reaction forces at the mounting interface, δ_x is the vertical moment arm of the horizontal membrane load, and δ_y is the horizontal moment arm of the vertical MTS load.

Because both the horizontal and vertical forces induce a counter-clockwise moment around the base, their effects are additive. The structural flanges of the bridge must therefore provide a substantial opposing reaction moment (M_R). Because this resulting bending stress acts perpendicularly to the print layer direction (the structurally anisotropic Z-axis), it poses a significant risk of inter-layer delamination that must be experimentally assessed prior to full-load application.

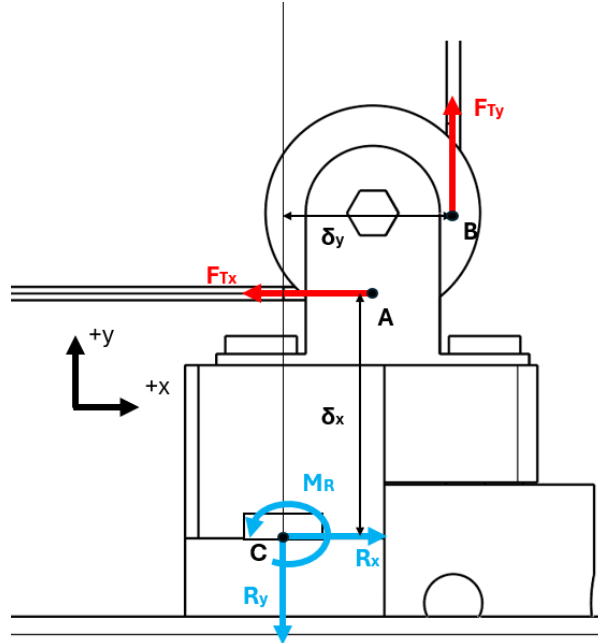


Figure 6.4: Free-body diagram of the pulley bridge under tensile loading, illustrating the induced bending moments acting on the 3D-printed layers.

Conversely, an aluminum machined part would easily withstand the forces and moments introduced during the tensioning phase without risk of delamination. However, machining this specific geometry requires both a large solid aluminum billet and extensive CNC milling time, incurring significant cost and lead time. While this cost could optionally be mitigated by assembling multiple smaller aluminum blocks, it cannot match the rapid prototyping speed, low cost, and geometric customizability of a single 3D-printed part.

A formal trade-off is presented in Table 6.5. While the scores are closely matched, the 3D-printed pulley bridge design was ultimately selected. To mitigate the primary structural concern regarding bending resistance and flange delamination, preliminary static load tests will be performed to empirically validate its structural rigidity.

Table 6.5: Concept selection matrix for the pulley bridge manufacturing paradigm.

Criterion (Weight)	3D-Printed Pulley Bridge	Aluminum
Bending resistance (0.40)	3	5
Manufacturability and assembly (0.30)	5	4
Availability and cost (0.30)	5	2
Final score (weighted total)	4.20	3.80

6.3. Final Assembly Rig Design

6.3.1. Final detailed CAD

Following the architectural trade-offs discussed in the preceding sections, numerous design iterations were conducted to consolidate the subsystems into the final assembly rig. The resulting detailed CAD model is presented in Figure 6.5, depicting the rig in its fully integrated configuration mounted onto the tensile tester and supported by the workbench.

The membrane-frame prototype is shown interfaced with the rig, where the static CFRP hat stiffener is rigidly clamped by the hold-down blocks to satisfy the fixed boundary condition (ASR-REQ-001). Furthermore, two 3D-printed linear guiding rails were introduced into the design. These rails strictly constrain the moving stiffener, granting it exactly one degree of freedom (longitudinal translation, U_1), thereby fulfilling the kinematic isolation requirement (ASR-REQ-002). Figure 6.5b details the side profile of the aluminum clamping plate. This design routes the stainless steel tensioning cable coplanar with the membrane mid-plane, successfully satisfying the geometric offset requirement of ASR-REQ-003.

To optimize the manufacturing effort and minimize costs, the overarching dimensions of the aluminum frame were dictated by the stock lengths of the available item profiles at the GTM-AS laboratory. Consequently, no transverse cuts were required to alter the length. The only mechanical machining required for the primary frame involved drilling localized clearance holes to accommodate M10 fasteners at the tensile tester interface.

A comprehensive, step-by-step manufacturing plan and the detailed integration procedure for the semi-finished prototype can be found in Appendix C.

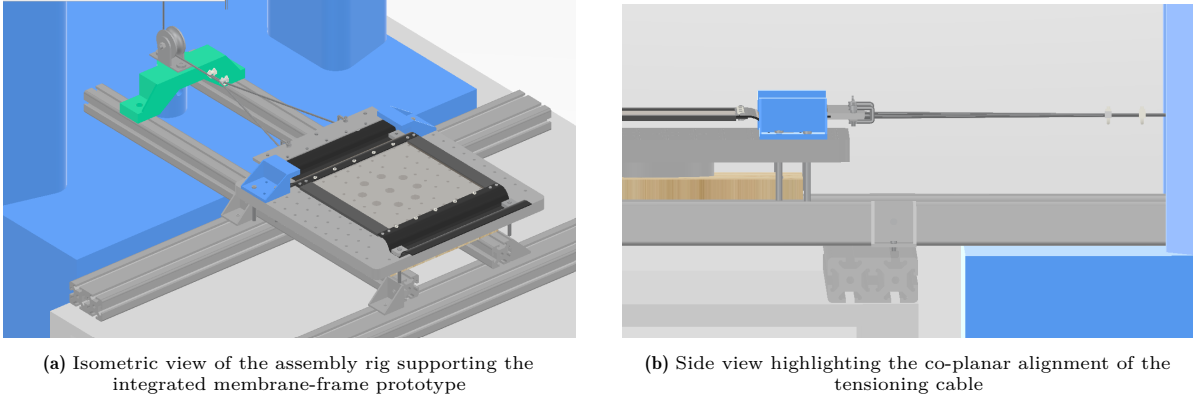


Figure 6.5: Final CAD renders of the tensioning and assembly rig.

6.3.2. Physical hardware integration and CAD deviations

Following the systematic procedure outlined in Appendix C, a physical assembly rig was constructed and integrated. The finalized hardware, presented in Figure 6.6, largely corresponds to the nominal CAD architecture detailed in Figure 6.5. However, during the physical integration phase, two minor structural adjustments were implemented to optimize performance and mitigate tolerance stack-ups:

- The Item profile corner brackets were relocated to the bottom face of the assembly rather than the side face, as depicted in Figure 6.6b. This reorientation was determined to provide a more robust load path against the anticipated longitudinal tensioning vector.
- A 3 mm thick layer of Tarrox anti-slip rubber was adhered to the 3D-printed guiding rails. This field adjustment was necessitated by a dimensional discrepancy in the manufactured profile height, which initially resulted in excessive play in the vertical (U_3) direction. The rubber layer effectively shimmed this gap, ensuring a strict kinematic constraint without inducing localized stress concentrations.

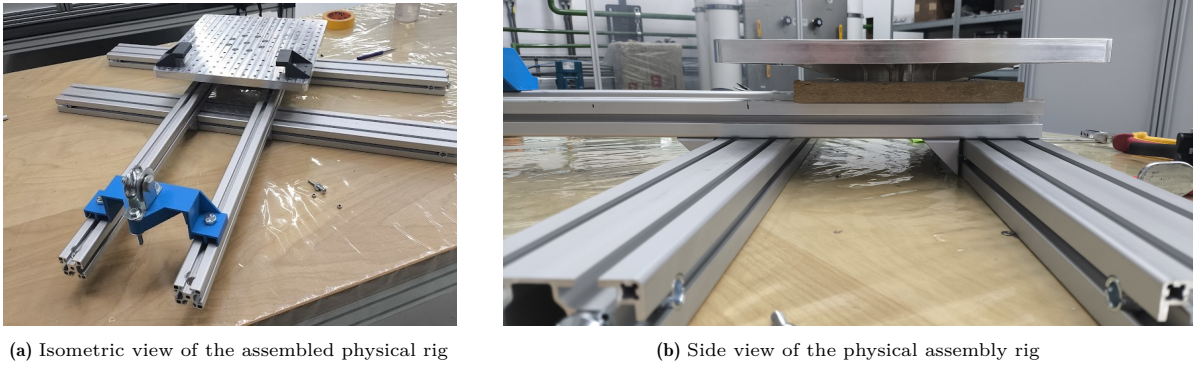


Figure 6.6: The physical assembly rig hardware following final integration.

6.4. Preliminary Tests

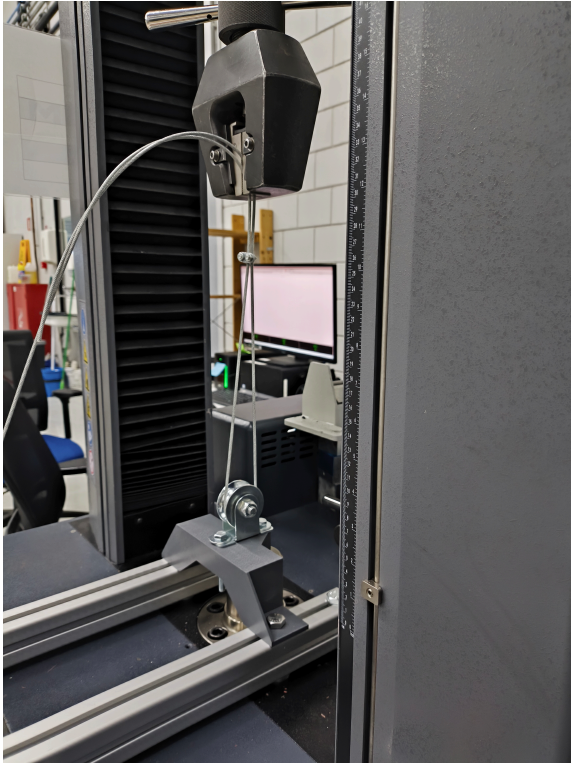
6.4.1. Pulley bridge test

As discussed in Table 6.2.2, the bending resistance of the 3D-printed pulley bridge is a major structural concern. Given the rapid iteration capabilities of FDM printing, an empirical test campaign was conducted to verify its rigidity under a proof load of 300 N (approximately 1.5 to 2 times the anticipated operational load).

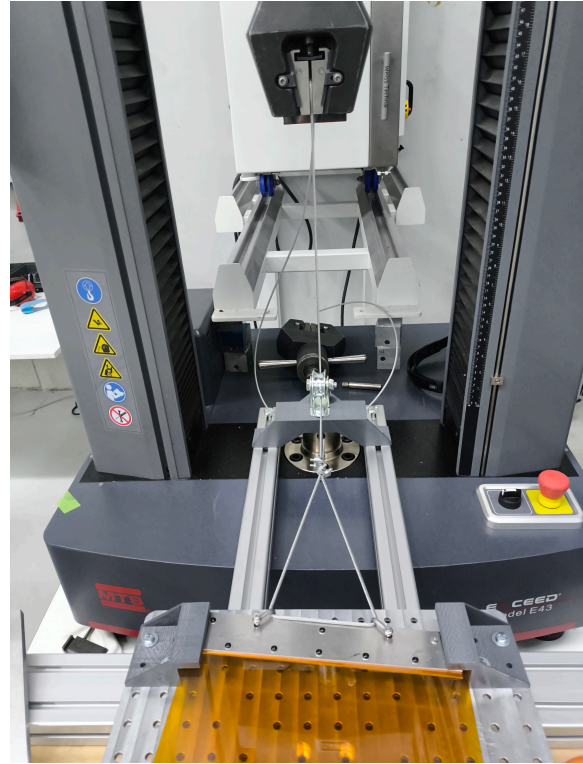
The preliminary verification consisted of a vertical loading test, presented in Figure 6.7a. The tensioned cable was routed over the pulley and loaded vertically. The component successfully sustained the 300 N proof load without any observable yielding or damage. However, because this vertical loading condition

induces a significantly lower bending moment at the mounting flanges, a subsequent in-line loading test was required to accurately represent the operational kinematics.

The in-line loading test was executed using a Kapton sheet clamped between the aluminum clamping plates on one side, and anchored between solid aluminum blocks on the other, as presented in Figure G.1b. During this test, the pulley bridge failed at a load of 255 N, falling short of the 300 N target. Inspection of the fracture interface (Figure 6.8a) revealed that the flange exhibited plastic deformation, followed by inter-layer delamination (cracking along the anisotropic print lines). The primary failure mechanism was identified as an excessive stress concentration induced by the counter-clockwise bending moment pulling against the fastener heads.



(a) Pulley bridge vertical loading test configuration



(b) Pulley bridge representative in-line loading test

Figure 6.7: Preliminary load testing configurations for the initial pulley bridge design.

Consequently, to mitigate this failure mode, the following design modifications were implemented in the second iteration:

- **Load Distribution:** Steel washers were integrated under the bolt heads to distribute the bearing load over a larger surface area on the polymer flange.
- **Geometric Reinforcement:** Structural ribbing was added to the flanges, and the overall flange height was increased to maximize the area moment of inertia against the applied bending moment.
- **Print Density:** The internal gyroid infill percentage was increased from 20% to 30%.
- **Wall Thickness:** The outer shell perimeter (wall line count) was doubled from 2 to 4 layers.

These structural improvements resulted in the iterated pulley bridge design presented in Figure 6.8b. During the slicing and manufacturing process of this iteration, the top solid layers were inadvertently omitted, exposing the internal gyroid infill pattern. Despite this manufacturing anomaly, a final in-line proof test confirmed that the iterated pulley bridge successfully withstood the 300 N target load without exhibiting any signs of damage or structural yielding.

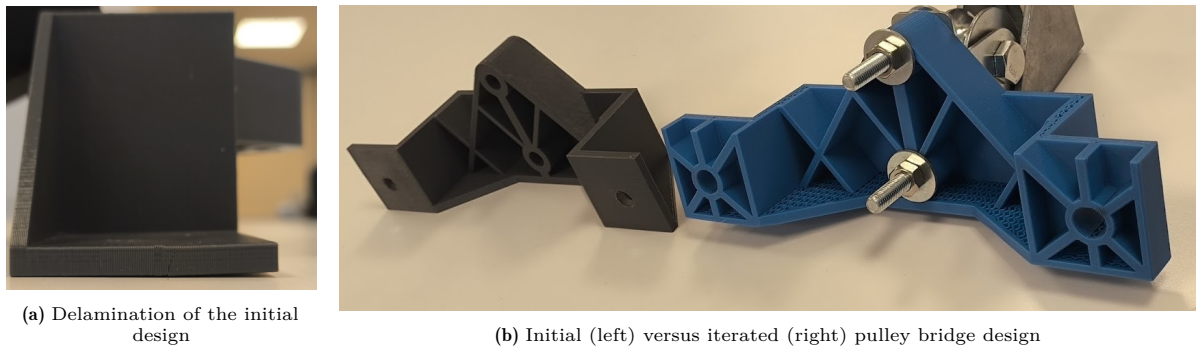


Figure 6.8: Pulley bridge failure analysis and subsequent structural iteration.

6.4.2. Clamping test

The clamping test was conducted to ensure that during operational loading, the Kapton HN membrane would not experience critical stress concentrations around the integration holes. To test this, the same setup was utilized as in Figure G.1b, where the Kapton membrane was connected to the aluminum clamping plates using M3 fasteners. While this component is primarily intended to pull the CFRP hat stiffener, the pitch of the bolts perfectly matches the Kapton membrane holes intended for the final clamping strip. The governing hypothesis was that if the Kapton membrane could sustain a 200 N load via mechanical clamping alone, the final prototype would remain safe from catastrophic stress concentrations. Furthermore, introducing DP490 structural adhesive would provide an additional margin of safety during operational loading.

After completing the test, the clamps were removed for inspection. Notably, the holes retained their circularity with negligible bearing deformation. While the second, fourth, and fifth holes exhibited minor witness marks indicating fastener contact, no critical stress concentrations or localized tearing occurred. This confirmed that the mechanical clamping method, supplemented by adhesive bonding, provides a sufficient structural margin to sustain the connection between the Kapton membrane and the hat stiffeners under the operational load.

6.4.3. Loading and relaxation test

To determine the exact loading sequence settings for the MTS Exceed E43 tensile tester, preliminary experiments were conducted using isolated Kapton HN strips. These test specimens were cut to unclamped dimensions of 20 x 210 mm. After iterating through several testing profiles, the following standardized loading sequence was established:

1. Secure the Kapton sample into the pneumatic grips of the tensile tester, utilizing thick paper adhered to the Kapton strips to prevent gripping damage [42].
2. Initiate the loading step to the target force using a controlled displacement rate of 1 mm/min.
3. Freeze the crosshead displacement the exact moment the target force is reached, and immediately begin logging the force-time decay.

After executing this sequence on four different strips loaded to varying initial force levels, the resulting force-time histories were plotted, as presented in Figure 6.9. A notable observation from these datasets is that the initial load is not maintained; instead, it rapidly decays before slowly asymptoting to a lower steady-state force value.

This phenomenon is known as viscoelastic relaxation, a well-documented mechanical behavior inherent to Kapton HN films [58]. The moment the film is loaded, the intermolecular bonds between the polymer chains stretch, representing the immediate elastic portion of the response. However, because the polymers within the thin film lack a fully crystalline lattice, localized stresses supply the activation energy needed for individual polymer chains to slowly slide past one another, untangling and reorienting over time. This micro-structural mechanism manifests macroscopically as a continuous decrease in the internal stress of the constrained membrane, resulting in the observed drop in reaction force.

Consequently, accounting for this viscoelastic effect during the final operational loading phase is essential to accurately achieve the true 167 N steady-state target load. Based on this phenomenon, applying an initial force overshoot is deemed absolutely critical for the final prototype integration.

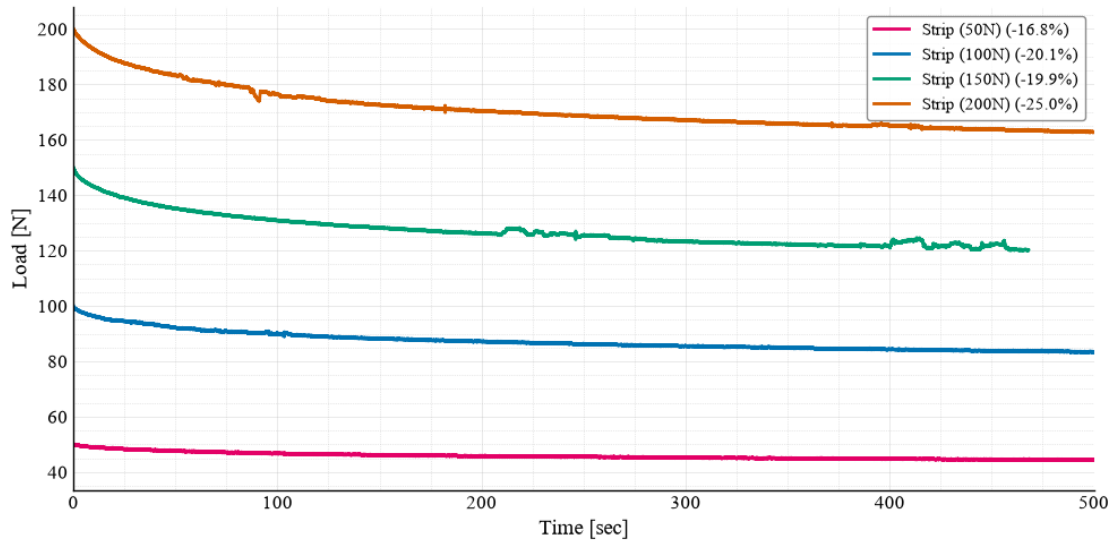


Figure 6.9: Experimental force relaxation comparison of Kapton HN strips under varying initial loads.

6.5. Assembly Process

6.5.1. Prototype integration and alignment

Following the preparation of individual components as detailed in Appendix C, the physical integration of the prototype was executed. The integration process proceeded in two primary phases. First, the untensioned membrane was bonded to the hat stiffeners. Second, controlled tensioning was applied while the membrane was guided by slotted lateral stiffeners and constrained by additional reinforcement strips.

In the initial phase, depicted in Figure 5.12b, the Kapton membrane was adhesively bonded to the clamping strip, which was subsequently secured using M3 fasteners. Optimal adhesion requires thorough surface preparation. Both the Kapton membrane and CFRP interfaces were mechanically abraded with Scotch-Brite pads and degreased using isopropanol to remove volatiles before applying the DP490 structural epoxy. Concurrently, the hold-down blocks were fastened to the rig. To guarantee strict geometric alignment, a dummy fastener was temporarily routed through all the blocks at the correct spacing on the test rig during the curing phase.

After the DP490 adhesive had fully cured, the lateral stiffeners were loosely attached via the slotted holes. A notable observation during this integration step was the presence of a macroscopic wrinkle on the membrane. This was attributed to a slight dimensional asymmetry (likely caused by a minor angular misalignment during the membrane lay-up and/or non-uniform adhesive thickness along the bond line). Upon application of the operational tensioning load, this wrinkle diminished significantly and the membrane visibly flattened. However, the initial presence of this geometric imperfection is a strong indication that a mildly non-uniform stress field was introduced into the membrane boundaries.

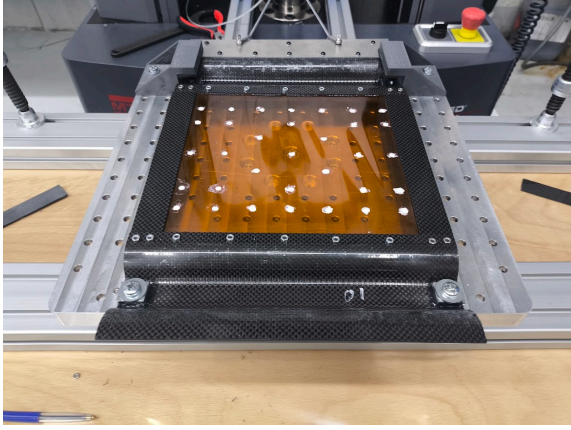
6.5.2. Tensioning operation and in-situ relaxation

As anticipated by the viscoelastic polymer behavior characterized in Section 6.4.3, an initial force overshoot is required to compensate for the in-situ relaxation of the Kapton membrane. Preliminary tests on the 200 N isolated strip exhibited a relaxation magnitude of approximately 25% over a 30-minute period (as presented in Figure 6.9). Because the full prototype integration requires a significantly longer epoxy curing cycle (spanning several hours), the total relaxation drop was projected to be higher. Consequently, an estimated overshoot of 35% was calculated. An initial tensile force of 225 N was thus applied to the system to ensure the membrane would asymptote towards the 167 N steady-state target

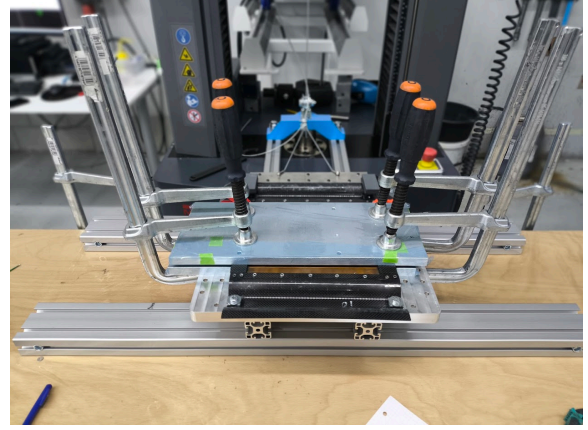
limit.

Once the pretension was established, a dimensional check was performed to verify the physical fit of the lateral reinforcement strips (Figure 6.10a). Upon confirmation, the slotted holes and the surfaces of the reinforcement strips were bonded to each other using DP490 structural adhesive. To ensure a uniform bond-line thickness and equal pressure distribution, two rigid aluminum plates were positioned around the assembly and tightly clamped, as depicted in Figure 6.10b. To prevent excess epoxy from permanently adhering to the tooling, these aluminum plates were wrapped in a non-stick release film.

During this curing phase, force-time data was continuously logged to monitor the in-situ relaxation of the actual prototype. Due to an automated software timeout (stand-by mode) in the MTS data acquisition system, the continuous logging ceased at $T \approx 5.43$ hours (19549 s). The final verified data point prior to the interruption recorded a force of 158.6 N.



(a) Reinforcement strip dimensional check



(b) Clamped setup ensuring uniform pressure distribution

Figure 6.10: Clamped setup and dimensional verification during the prototype preloading phase.

Despite this hardware limitation, the recorded transient phase sufficiently captured the exponential decay profile of the Kapton film. Using this data, the long-term relaxation curve was mathematically extrapolated to estimate the final steady-state tension of the fully cured prototype, as presented in Figure 6.11. To ensure accuracy, the logarithmic regression was strictly anchored to the final recorded measurement, yielding the following relationship ($R^2 = 0.9954$):

$$F(t) = -10.6223 \cdot \ln(t + 1) + 263.6206 \quad (6.6)$$

where F is the pretension force in Newtons and t is the elapsed time in seconds. Based on this extrapolation, the final predicted load at the time of removal from the test rig ($t \approx 71760$ s, or 20 hours) is 144.85 N. This indicates that the estimated 35% initial overshoot was ultimately insufficient to maintain the target steady-state pretension of 167 N.

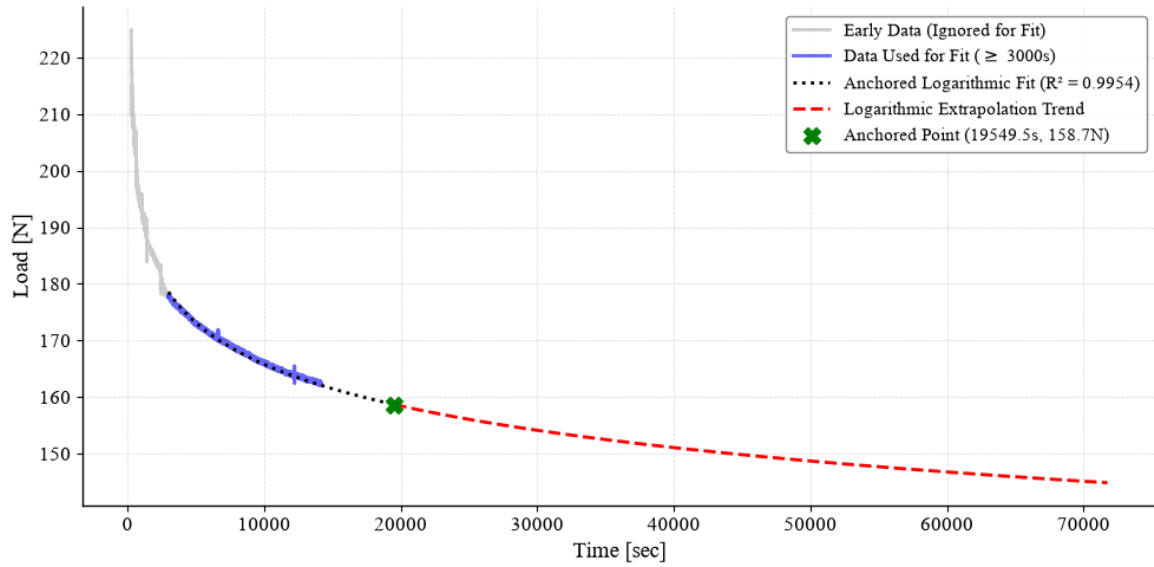


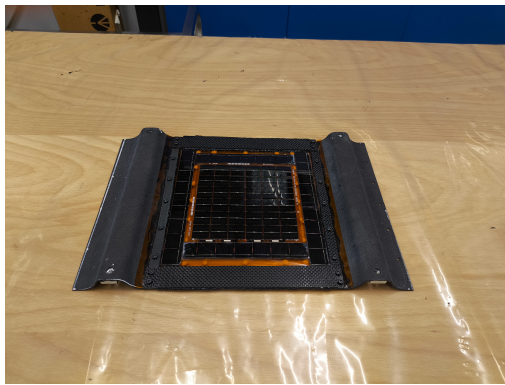
Figure 6.11: In-situ viscoelastic force relaxation curve of the Kapton membrane. The anchored logarithmic extrapolation ($R^2 = 0.9954$) projects the final steady-state tension achieved prior to removal from the assembly rig.

6.5.3. SCA Adhesion to the Unpopulated Structure

Following the successful extraction of experimental modal data from the bare prototype, Solar Cell Assemblies (SCAs) were bonded to the structure to investigate the dynamic effects of added mass and localized bending stiffness. The layout comprised a functional 6×6 mPower DragonScales SCA positioned in the center, flanked by two 9×1 non-functional strings along the longerons and two 6.5×1 non-functional strings parallel to the lateral profiles. The non-functional cells were integrated to increase the overall packing factor and more accurately represent the operational mass distribution of the array.

The solar cell assemblies were bonded to the Kapton membrane using a space-qualified two-component room-temperature-vulcanizing (RTV) silicone adhesive. The populated prototype was subsequently cured under uniform pressure prior to experimental testing.

The fully integrated populated prototype is shown in Figure 6.12.



(a) Front view (active surface) of the populated array



(b) Rear view showing adhesive lines

Figure 6.12: Final prototype assembly following the SCA integration and curing process.

Experimental Setup and Dynamic Analysis

7.1. Experimental Setup Preparation

7.1.1. Test Objectives and Requirements

To extract high-fidelity modal data from the composite lattice frame and tensioned membrane prototype, the experimental design must be fully constrained. A test setup that introduces parasitic mass, alters boundary conditions, or dynamically couples with the structure would fundamentally invalidate subsequent correlation with the finite element models. As a result, a specific set of experimental requirements was established, as outlined in Table 7.1. These requirements govern the physical mounting, excitation methodology, and data quality thresholds, serving as the foundational criteria for the equipment trade-offs.

Table 7.1: Experimental Modal Analysis Test Setup Requirements

TEST SETUP REQUIREMENTS				Verification Type				Rationale/
Req. ID	Req. Statement	Value	Type	A	I	D	T	Comment
TS-REQ-001	The setup shall measure out-of-plane membrane displacement without significantly altering physical mass or local stiffness.	$m_{\text{added}} < 1 \text{ g}$	Killer	✗	✓	✓	✗	Prevents mass-loading from altering thin-film eigenfrequencies.
TS-REQ-002	The rig shall enforce pinned boundary conditions at all four corners of the CFRP frame profile.	$U_{1,2,3} = 0$ $UR_{1,2} \neq 0$ $UR_3 = 0$	Key	✓	✓	✓	✗	Matches FEA boundaries to prevent rigid-body coupling.
TS-REQ-003	The first natural frequency of the aluminum test rig shall be strictly higher than the frequency range of interest.	$f_{n,\text{rig}} \geq 400 \text{ Hz}$	Driver	✓	✗	✗	✓	Prevents rig resonances from corrupting prototype modal data.
TS-REQ-004	The excitation system shall induce structural responses strictly within the linear elastic range of the prototype, preventing composite delamination or membrane tearing.	$F_{\text{exc}} \leq F_{\text{max, safe}}$	Constr.	✓	✗	✗	✓	Prevents structural degradation to guarantee repeatable measurements.
TS-REQ-005	The DAQ shall record a continuous reference signal perfectly synchronized with the scanning laser measurements.	$\Delta t \approx 0$	Key	✗	✗	✓	✓	Required by DAQ to calculate operational deflection shapes.
TS-REQ-006	The primary excitation force vector shall be applied perpendicularly to the membrane plane (out-of-plane).	Z-Axis	Constr.	✗	✓	✓	✗	Optimizes excitation of transverse and localized drumhead modes.
TS-REQ-007	The measured Frequency Response Functions (FRFs) shall maintain a signal coherence strictly greater than 0.85 at all identified global resonances.	$\gamma^2 > 0.85$	Key	✗	✗	✗	✓	Ensures valid, linear data unpolluted by ambient noise.

As presented in Table 7.1, the physical characteristics of the prototype significantly dictate the experimental approach. The most restrictive constraint is the mass-loading limit (**TS-REQ-001**). Because the

flexible membrane relies on tension for its stiffness, any physical sensor attached to its surface would artificially alter its local eigenfrequencies. This strictly mandates a non-contact measurement methodology. Furthermore, to ensure the modal data reflects only the prototype and not the supporting infrastructure, the test rig must approximate the analytical pinned boundary conditions (**TS-REQ-002**) while maintaining a fundamental frequency well above the testing bandwidth (**TS-REQ-003**).

Dynamically, the excitation strategy must balance energy input with structural safety (**TS-REQ-004** and **TS-REQ-006**). The excitation signal must be strictly synchronized with the data acquisition system (**TS-REQ-005**) to enable the scanning laser vibrometer to compute valid phase relationships across the measurement grid. Ultimately, the success of the setup is judged by the coherence of the recorded signals (**TS-REQ-007**), which ensures the captured frequency response functions are highly linear and suitable for FEA correlation. These constraints directly informed the hardware selection discussed in the subsequent trade-off analyses in Section 7.1.2 and Section 7.1.3.

7.1.2. Modal Excitation Method Selection

To accurately perform Experimental Modal Analysis (EMA), a generator is required to provide the input energy necessary to excite the prototype structure at its natural frequencies. The chosen method must be capable of exciting both the localized modes of the Kapton membrane and the global structural modes of the CFRP frame. Four candidate excitation setups were considered: a large V850 electrodynamic shaker, an in-house V550 electrodynamic shaker, a multi-shaker setup utilizing four SmartShaker™ K2007E01 units, and an acoustic loudspeaker setup.

A formal trade-off was conducted to evaluate these candidates. The criteria chosen are defined as follows:

- **Global frame excitation capacity:** The ability of the exciter to deliver a clean, distortion-free energy input capable of sufficiently exciting both the localized membrane modes and the high-stiffness global modes of the CFRP frame, without signal degradation caused by the structure's mechanical hindrance.
- **Boundary condition integrity:** The degree to which the excitation setup preserves the idealized dynamic boundary conditions of the prototype, avoiding the introduction of artificial kinematic constraints, unrepresentative rigid-body modes, or complex multi-point dynamic coupling.
- **Logistics, cost & availability:** The practical feasibility of the excitation method, evaluated by the requirement for external equipment rental, operational financial overhead, and the accessibility of in-house facilities to allow for early data acquisition (DAQ) integration testing.

The results of the trade-off are presented in Table 7.2.

Table 7.2: Concept selection matrix for the modal excitation method, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	V850 Shaker	V550 Shaker	4x SmartShaker	Loudspeaker
Global frame excitation capacity (0.35)	5	4	3	1
Boundary condition integrity (0.30)	4	4	1	5
Logistics, cost & availability (0.35)	1	5	3	4
Final score (weighted total)	3.30	4.35	2.40	3.25

The acoustic loudspeaker is highly effective for non-contact excitation of thin films and has proven to be an effective method for membranes [33]; however, it lacks the mechanical impedance required to adequately excite the stiff CFRP hat-stiffeners, rendering it unsuitable for measuring the global frame dynamics. Additionally, a custom test setup would need to be developed to perform the experiments.

Conversely, the multi-shaker setup (using four small SmartShakers) presented a logistically attractive option, as it would allow the testing campaign to remain at the TU Delft facilities. However, this approach introduces a severe technical flaw: supporting the prototype on four independent shaker

stingers inherently alters the dynamic boundary conditions. This multi-point constraint introduces unrepresentative rigid-body modes and complex dynamic coupling, invalidating the correlation with the idealized Finite Element Model. Furthermore, two additional shakers and a custom suspension frame would need to be sourced or rented.

While the heavy-duty V850 shaker, available at Brookx Company located next to GTM-AS, would effortlessly accommodate the full-scale test setup without boundary condition compromises, the operational costs and logistical overhead associated with its use are prohibitively high for this testing campaign.

Ultimately, the **V550 Electrodynamic Shaker** was selected. As in-house equipment, it drastically reduces testing costs and logistical risks. Having direct access to the V550 permits early de-risking of the Data Acquisition (DAQ) architecture. Although the existing adapter plate for the V550 exhibits a limited high-frequency bandwidth (further elaborated in Appendix G), it is deemed sufficient to capture the primary structural modes of interest. If synchronization hardware limitations arise, the force-input data can be acquired independently and combined with the vibrometer laser data during post-processing to formulate the final H_1 Frequency Response Functions.

7.1.3. Data Acquisition Selection

As previously established in Section 2.4.1, contact-based sensors introduce severe mass-loading effects on ultra-lightweight membranes. Consequently, non-contact methodologies were evaluated [33], [59]. To determine which non-contact measurement architecture is most suitable, three options were evaluated. The first option is utilizing a high-speed camera to perform Digital Image Correlation (DIC). The principle behind DIC is that two imaging sensors looking at different positions on an object provide enough information to perceive the object as three-dimensional. In turn, if a speckle pattern is applied onto the membrane, the position of each object point can be calculated and the dynamic response can be correlated [60].

The second option is a Laser Doppler Vibrometer (LDV), an instrument used for non-contact measurements capable of determining the surface velocity of a vibrating object in the direction of the laser's line of sight. This is achieved by analyzing the Doppler shift between the incoming light and the light scattered back to the device [61]. By installing a scanner in front of the LDV, consisting of a pair of orthogonal scan mirrors, the LDV can be directed to any visible target point on a structure. This can be done with a single scanner (SLDV), limiting the measurements strictly to the out-of-plane vibration of a structure's surface. When three laser beams are used concurrently, it is possible to obtain a full three-dimensional measurement of the test object.

A formal trade-off was conducted based on the following evaluation criteria:

- **Dynamic fidelity & Non-intrusiveness:** The capability of the system to accurately resolve high-frequency, low-amplitude vibrations without imposing parasitic mass loading (via coatings/targets) that could alter the membrane's state of tension.
- **Spatial acquisition efficiency:** The ability to rapidly and accurately acquire displacement data over a dense spatial grid to reconstruct global mode shapes, minimizing the total duration of the testing campaign.
- **Logistical feasibility & Setup complexity:** The practical effort required to integrate, calibrate, and align the measurement hardware, considering the availability of existing in-house assets.

Table 7.3: Concept selection matrix for the non-contact data acquisition method, where red = 1 (lowest), orange = 2, yellow = 3, light-green = 4 and dark-green = 5 (highest).

Criterion (Weight)	DIC Cameras	3D LDV	1D LDV (PSV-500)
Dynamic fidelity & Non-intrusiveness (0.40)	2	5	5
Spatial acquisition efficiency (0.35)	5	4	4
Logistics & Setup complexity (0.25)	3	3	4
Final score (weighted total)	3.30	4.15	4.40

While Digital Image Correlation (DIC) excels at capturing instantaneous, full-field spatial data, its inherent intrusiveness presents a major drawback for this specific application. The translucency of the Kapton film necessitates surface treatment for both DIC and LDV techniques to ensure optical reflectivity. DIC, however, demands a dense, high-contrast paint speckle pattern, often involving a double-layer application. This thick paint layer on a $50\text{ }\mu\text{m}$ thin film adds a measurable parasitic mass and serves as a physical constraint, artificially enhancing the membrane's bending stiffness. Conversely, LDV requires only a few dots of white primer, which imposes minimal mass and stiffness effects. Furthermore, the use of high-speed cameras demands intense external lighting, and the thermal radiation from these sources can cause rapid thermal expansion in the Kapton film.

Both Polytec LDV configurations offer exceptional dynamic fidelity, utilizing low-power scanning lasers that inflict zero mass or thermal loading. While the 3D LDV system provides comprehensive tri-axial displacement data, calibrating and optically aligning three independent scanning heads around the test rig is a much more complex procedure than using only 1D LDV. Because the primary dynamic concern for a flat pre-tensioned membrane lies in its transverse, out-of-plane modes (U_3), capturing the in-plane vibrations offers marginal analytical value at the expense of severe setup complexity.

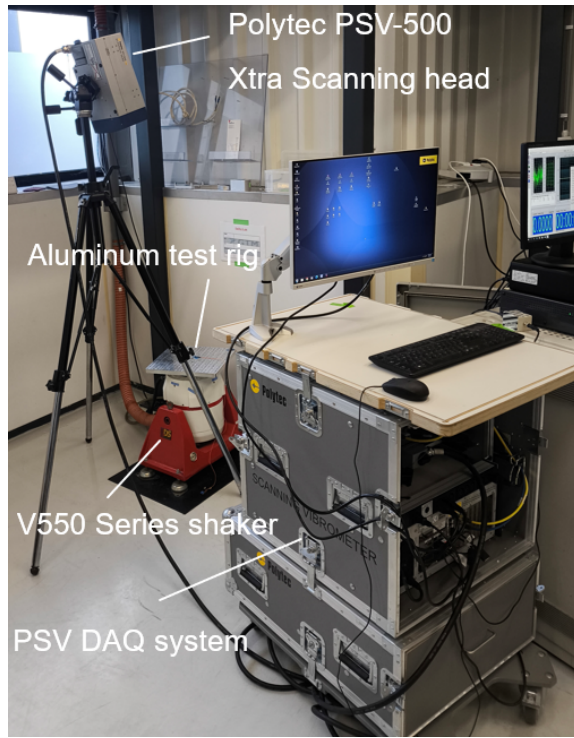
Ultimately, the **1D Polytec PSV-500 Scanning Vibrometer** was selected as the optimal DAQ system. It utilizes internal motorized steering mirrors to automatically and rapidly scan a pre-defined geometric grid. It provides very detailed velocity and displacement resolution ($\leq 0.1\text{ }\mu\text{m/s}$ [62]) in the critical out-of-plane direction while requiring the alignment of only a single optical head, perfectly balancing spatial efficiency with logistical practicality.

7.1.4. Final Test Setup

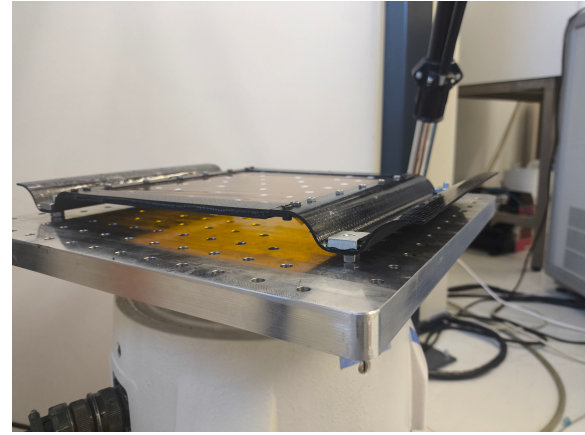
Following the trade-off analysis of the modal excitation methods and data acquisition systems, the final integrated test setup was configured as presented in Figure 7.1. The V550 shaker was successfully integrated with the PSV-500 front-end via the LDS LASER USB vibration controller using a standard BNC connection. This direct coupling eliminated the need for an external trigger system, ensuring precise phase synchronization. Additionally, the Polytec PSV-500 Xtra scanning head was mounted on a tripod and oriented perpendicularly to the aluminum test rig to maximize the laser signal return.

The physical interface between the prototype and the aluminum test rig is detailed in Figure 7.1b. To enforce accurate boundary conditions, a spacer nut was inserted at each of the four hold-down points between the composite hat profiles and the test rig. This offset restricts translational displacement while permitting local rotation, effectively realizing a pinned boundary condition at each corner of the structural frame.

Prior to executing the final experimental modal analysis, preliminary tests were conducted to validate the PSV-500 acquisition settings and identify the fundamental limitations of the V550 shaker test rig. The detailed procedures and results of these calibration tests are documented in Appendix G.



(a) Experimental modal analysis setup



(b) Prototype attached to the aluminum test rig showing distinct pinned boundary conditions due to the addition of spacer nuts

Figure 7.1: Experimental modal analysis setup.

7.2. Experimental Modal Analysis and Finite Element Validation

This section presents the 1D LDV experimental data alongside the updated numerical finite element model, followed by a direct comparative analysis. For the spatial LDV scans, the same acquisition settings were utilized for both the unpopulated and populated prototype configurations, as detailed in Table 7.4. These parameters were systematically established during the iterative preliminary testing campaign.

Table 7.4: General data acquisition and signal processing parameters configured for the 1D LDV spatial measurements.

Parameter	Value
Frequency Span	0 – 500 Hz
FFT Lines	3200
Frequency Resolution (Δf)	0.156 Hz
Excitation Signal	Pseudo-random
Averaging Type	Magnitude
Averaging Count	10
Windowing Function	Rectangular (Uniform)
Reference Channels	Center and Corner Accelerometers
Velocity Range	500 mm/s
Tracking Filter	Fast
Generator Voltage	0.03 V
Measurement Nodes (Bare Membrane)	147
Measurement Nodes (With Solar Cells)	151

7.2.1. Experimental Modal Analysis of the Bare Structure

The frequency response function (FRF) obtained from the 1D LDV measurements on the unpopulated prototype is presented in Figure 7.2, alongside the corresponding coherence function. The figure plots

the H_1 estimators for both the center flexible base (Vib & Ref 1, in grey) and the corner rigid interface (Vib & Ref 2, in blue). The corner measurement serves as the primary reference because this accelerometer is located at the direct contact interface, allowing it to capture the true base excitation more accurately. Regarding the measured FRF, the following observations can be made:

- The frequency response below 20 Hz is invalid, as the coherence drops significantly below the 0.85 validity threshold. This low data fidelity is driven by the physical stroke limitations of the electrodynamic shaker, which actively constrain low-frequency displacements. Furthermore, the broadband energy distribution of the pseudorandom input signal yields an inadequate signal-to-noise ratio in these lower frequency bins.
- The sharp frequency spike and corresponding coherence drop at 50 Hz are artifacts of electrical line noise interference, as established in Section G.2. Therefore, this peak does not represent a structural eigenmode.
- At approximately 460 Hz, a localized resonance of the aluminum test rig occurs. This is evidenced by a massive magnitude spike in the flexible base response (H_1 [Vib & Ref 1]) coupled with a severe drop in coherence, identifying it as a facility-level resonance artifact. Subsequently, at 479.84 Hz, the rigid interface (H_1 [Vib & Ref 2]) detects a high-coherence resonance peak. This indicates the formation of a coupled system-level mode between the test rig plate and the prototype. Because the FEM boundary conditions do not include the mass and stiffness matrices of the test facility, numerical correlation for these high-frequency, rig-coupled modes is inherently limited.

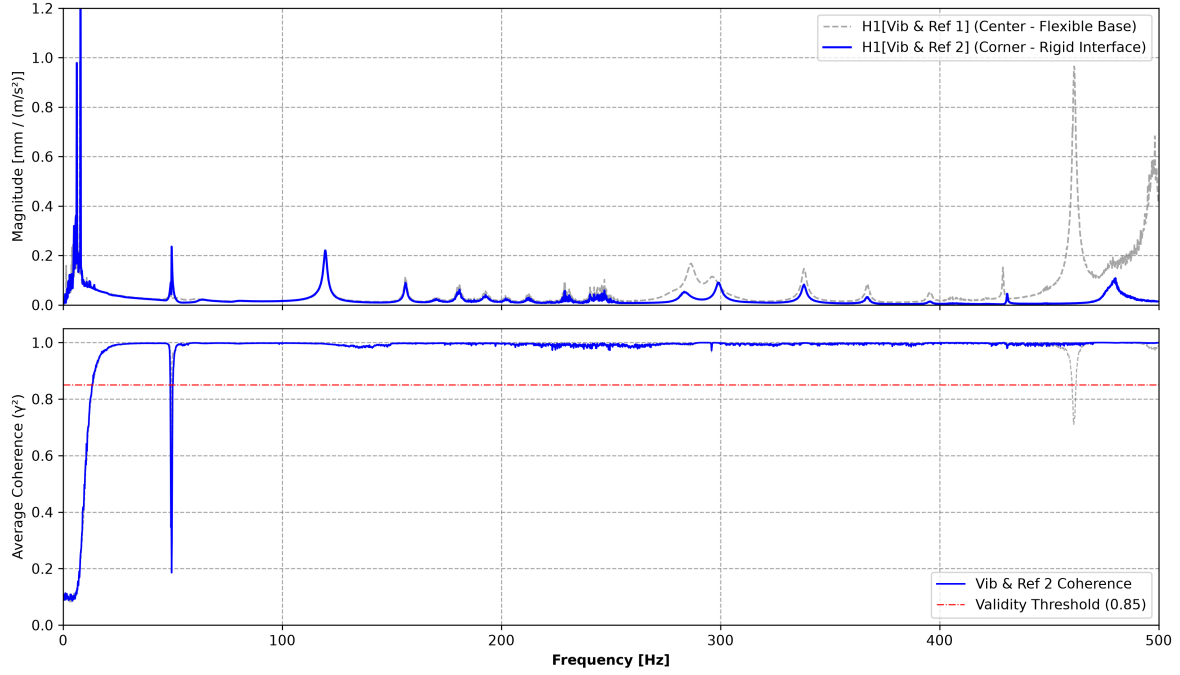


Figure 7.2: Averaged H_1 FRF response (top) and corresponding coherence γ^2 function (bottom) from 0 to 500 Hz of the unpopulated membrane-frame prototype.

Consequently, all FRF peaks between 100 Hz and 430 Hz are confirmed as genuine eigenfrequencies of the unpopulated prototype. The corresponding experimental mode shapes are presented in Figure 7.3.

The fundamental mode, characterized by an out-of-plane central deflection, is clearly visible in Figure 7.3(a). Notably, this mode exhibited the largest measured displacement amplitude ($3.909 \mu\text{m}$). Interestingly, the maximum displacement is not perfectly centered but is slightly offset toward the left corner of the membrane. This spatial offset correlates directly with the slight angular misalignment introduced during the bonding of the Kapton membrane to the hat profiles, as discussed in Section 6.5.1. The resulting differential tension field manifests across the higher-order mode shapes, which all exhibit slight asymmetries compared to idealized theoretical modes.

Between 200 Hz and 270 Hz, several minor, closely spaced resonance peaks are observed. While the coherence remains above the validity threshold, confirming these as genuine structural responses, the peaks exhibit reduced signal-to-noise ratios and slight coherence degradation. This ‘noisy’ localized response is characteristic of high modal density and localized non-linearities (such as micro-wrinkling or boundary micro-slip) commonly seen in higher-order tensioned membrane modes. Because these modes lack significant global mass participation, they are tabulated for completeness but excluded from primary numerical correlation.

At higher frequencies, the dynamic behavior of the prototype shifts. The resonance at 283.59 Hz represents the first frame-dominated mode, characterized by out-of-plane oscillation of the lateral profiles. Shortly thereafter, the 299.06 Hz peak exhibits a coupled mode where both the membrane and the lateral profiles participate simultaneously. From this frequency onward, the structural response is dominated by complex, coupled membrane-frame interactions. Finally, at 430.94 Hz, the mode shape reveals significant torsional twisting of the hat profiles.

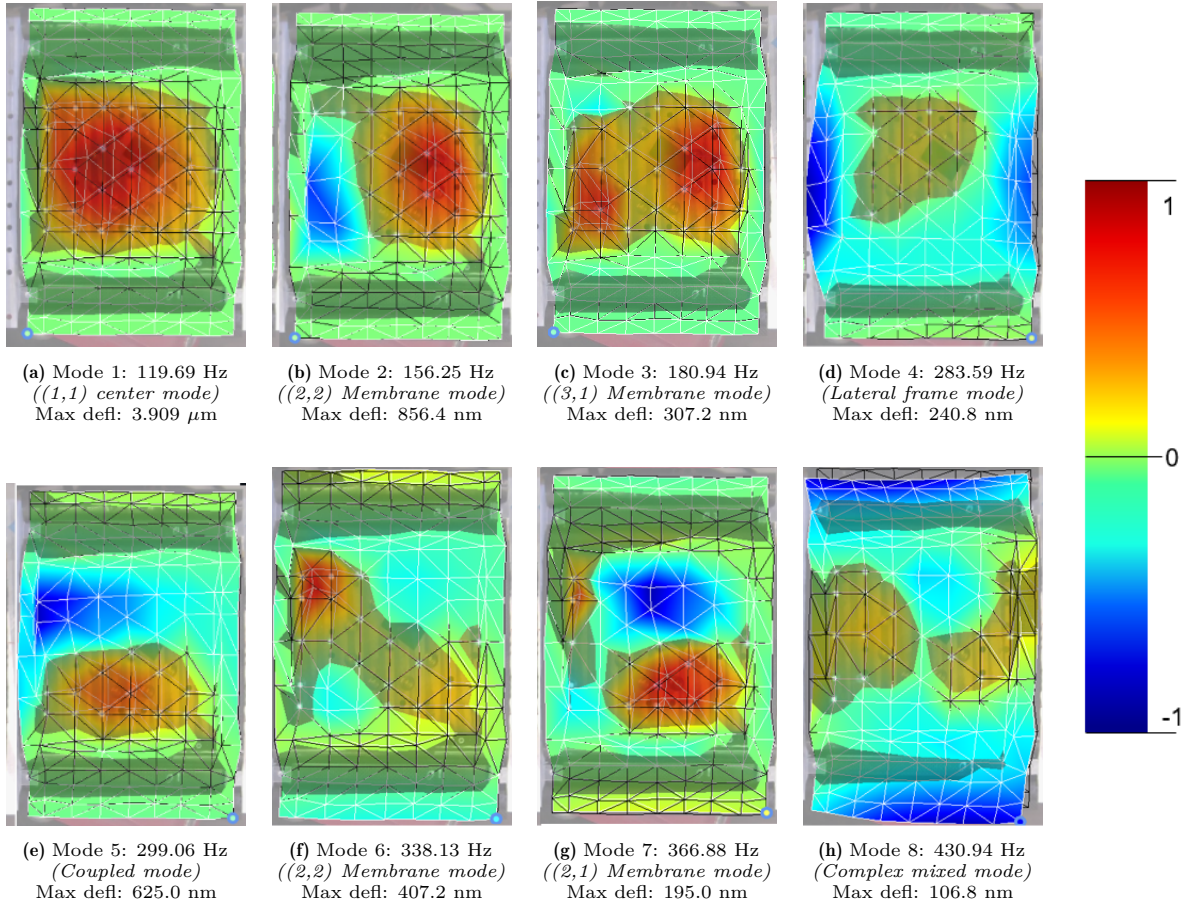


Figure 7.3: Experimental mode shapes of the unpopulated membrane-frame prototype obtained via 1D scanning laser Doppler vibrometry. The color map represents the auto-scaled, normalized out-of-plane deflection to illustrate modal phase and spatial continuity. The maximum absolute deflections provided in the sub-captions correspond to the forced structural response under a 0.0047 G_{rms} pseudo-random base excitation.

7.2.2. Updated As-Built Finite Element Model of the Unpopulated Prototype

Following the experimental modal analysis, it became evident that the initial finite element modeling technique (described in Section 5.3.5) did not sufficiently predict the mode frequencies. The comparison of predicted modes of the initial technique is presented in Appendix I. To improve this dynamic correlation, an updated model was developed to strictly mirror the as-built dimensions and physical boundary conditions of the experimental prototype. The deviations in the preliminary model were traced back to the following idealized assumptions:

- The unconstrained tensioned membrane dimensions in the preliminary model (225×245 mm) deviated from the true as-built geometry (188×245 mm). This discrepancy artificially inflated the estimated force required to achieve the target stress state in the membrane.
- The previous model idealized the boundary conditions by applying them directly at the corner nodes connecting the beam profiles. In reality, the clamping interfaces are located further inboard from the membrane edges, and the lateral profiles are physically attached to the hat profile flanges rather than a singular corner node.
- The original cross-section modeled for the hat profiles matched the nominal design rather than the manufactured reality. As previously established, a slight angular offset occurred during bonding, which inherently altered the effective cross-sectional dimensions and stiffness of both the hat and lateral profiles.
- Consequent to the previously assumed larger membrane dimensions, the lateral profiles in the initial model were excessively long. This artificially lowered the bending stiffness of the frame, resulting in a significant underestimation of the lateral profile eigenfrequencies.
- While the steel cable physically pulls at the 1/4 and 3/4 points on the moving hat profile, the force is more distributed over the entire flange. Therefore, a line load or shell edge load is more representative of this load path.
- The CFRP profiles were initially assumed to behave isotropically, which underestimates the true stiffness of the composite laminates.

Consequently, numerous modifications were introduced to the updated model to accurately capture the dynamic response of the unpopulated prototype. First, all carbon fiber reinforced polymer (CFRP) structures were upgraded from 1D beam elements (B31) to 2D fully integrated shell elements (S4). This allowed each individual composite component, including the clamping plates, to be explicitly modeled, resulting in the exploded assembly presented in Figure 7.4. The fully integrated S4 elements were selected over reduced integration (S4R) elements because they generate a more mathematically robust stiffness matrix, eliminating artificial hourglassing during complex twisting modes. While S4 elements inherently require longer computation times, the computational cost remains entirely acceptable for a model of this geometric scale.

The central Kapton membrane is modeled as a 245×208 mm plane. To replicate the physical bonded assembly, the membrane domain overlaps the longeron flanges by 10 mm and the lateral profiles by 24 mm. The unconstrained middle region of the membrane is meshed with reduced integration membrane elements (M3D4R), while the overlapping bonded edges are modeled with S4 shell elements. This element transition is necessary because the overlapping edges are constrained to the underlying CFRP profiles using surface-to-surface tie constraints. Within this tied formulation, the stiffer CFRP components act as the master surfaces and the thinner membrane edges as the slave surfaces, accurately reflecting the frame's physical stiffness dominance during dynamic loading. Furthermore, the mesh was seeded to ensure perfectly coincident nodes between the assembled parts, guaranteeing successful constraint enforcement.

Finally, rather than assuming isotropic behavior, the CFRP parts are modeled using composite layups. The material properties for the T300-DT120 lamina, detailed in Table 5.8, were applied. Due to limited manufacturer data, the minor Poisson's ratio (ν_{12}) is assumed to be 0.05, and the out-of-plane shear moduli are equated to the in-plane shear modulus ($G_{12} = G_{23} = G_{13}$). In turn, the layup $[0/90, \pm 45, 0/90, \pm 45]_s$ was assigned to the profile clamps and hat profiles, while the lateral profiles were assigned $[0/90, \pm 45, 0/90, \pm 45, \pm 45, 0/90]_s$, with each lamina having a thickness of 0.2 mm. The Kapton membrane retains its baseline isotropic material assignment.

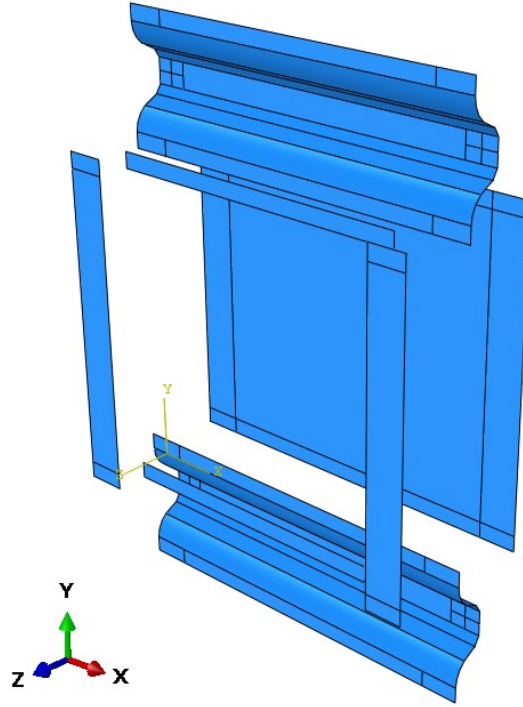


Figure 7.4: Exploded numerical view of the final FEM assembly.

The sequential non-linear loading protocol, utilizing the Abaqus element birth-and-death sequence detailed in Section 5.3.5, was preserved to accurately lock in the initial stress state. However, to fully capture the physical as-built configuration, the following specific adjustments were implemented in the updated model:

- The global coordinate system of the numerical model was translated such that the origin is located at the root of the left lateral profile. This aligns the numerical coordinate system with the physical reference frame utilized during the 1D LDV measurements, facilitating the spatial node matching required for the MAC correlation.
- Consequent to the transition to fully integrated shell elements, the 1D line load was replaced with a shell edge load applied to the unconstrained hat profile. Furthermore, an iterative tuning process determined that reducing the applied distributed load magnitude from 0.5 N/mm to 0.2 N/mm yielded the most representative correlation for the membrane-dominated dynamic modes.
- The boundary conditions were updated from pinned corners to fully clamped (‘encastre’: $U_1 = U_2 = U_3 = UR_1 = UR_2 = UR_3 = 0$) constraints applied directly to the four shell-element clamping planes. This highly constrained boundary condition accurately reflects the kinematics of the physical steel hold-down blocks in the test rig. Because the hat profiles are now modeled with 2D shell elements instead of 1D beams, applying this localized clamped condition at the root accurately permits the unhindered out-of-plane bending of the lateral profiles.

The final mode shapes are presented in Figure 7.5. Notably, the model initially exhibits the expected analytical (m,n) membrane modes, characterized by virtually zero displacement of the CFRP frame, followed sequentially by higher-frequency frame modes. These membrane modes emerge at lower frequencies because the polyimide film possesses minimal mass, and its out-of-plane stiffness is derived almost entirely from the applied geometric pre-tension. Consequently, the membrane requires significantly less excitation energy compared to the CFRP profiles, which possess high intrinsic bending and torsional stiffness. Furthermore, the lateral frame modes consistently precede the hat stiffener modes due to the significantly lower inertia of the lateral profiles.

As the frequency increases beyond 250 Hz, the excitation energy becomes sufficient to activate the intrinsic bending and torsional stiffness of the CFRP laminates, transitioning the dynamic response

from local membrane flutter to global structural participation. This transition is marked by a significant increase in modal density, as seen by the lateral frame modes (Modes 14 and 15) occurring within 0.06 Hz of one another. Ultimately, at higher frequencies (such as Mode 88 at 489.86 Hz), the dynamics evolve into complex, coupled modes where the membrane, frame, and boundary conditions interact heavily. Understanding this transition from isolated membrane behavior to dense, coupled frame participation is critical for contextualizing the subsequent experimental correlation.

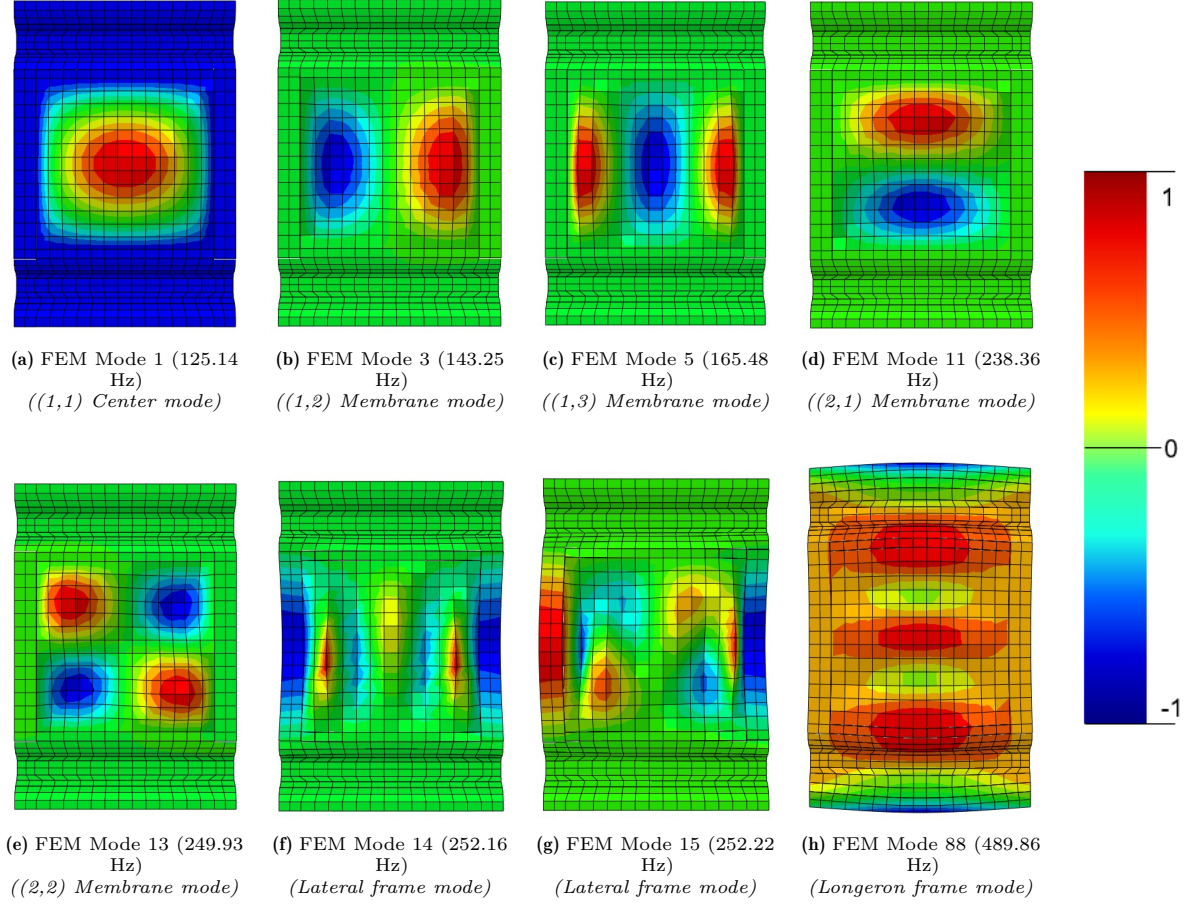


Figure 7.5: Numerical mode shapes of the unpopulated membrane-frame prototype extracted from the updated finite element model. The mode indices reflect the raw eigenvalue extraction numbers from Abaqus, allowing for direct tracking in the subsequent MAC correlation matrix. The color map represents the relative, mass-normalized out-of-plane displacement (eigenvectors); absolute displacement magnitudes are excluded as they hold no physical meaning in a linear perturbation step.

The tuned numerical edge load of 0.2 N/mm corresponds to an effective residual membrane tension of only 200 N/m (45 N total across the 225 mm width). This is significantly lower than the 144.85 N macroscopic pulling force logged during physical assembly. While this discrepancy is large, it represents the hypothesized effective residual tension in the membrane at the time of testing rather than a modeling error. As predicted during the conceptual analysis in Section 4.4, transferring the tension from the rigid assembly rig to the compliant lateral CFRP profiles causes inward bowing and elastic compression of the frame. An analytical estimation indicates that because the 50 μm Kapton film requires less than 1.0 mm of total elongation to reach its target 500 N/m pretension, a combined structural compliance and potential micro-slip (during the curing phase) of merely ~ 0.6 mm would mathematically account for this drop in residual force. This lost displacement was likely exacerbated by the physical tensioning setup itself; the localized compression of the polymer-coated steel cable and the observed mechanical compliance of the polymer pulley bridge inherently introduce parasitic slip into the load path. Therefore, it is highly probable that this cumulative frame compliance, unmeasured rig and joint slip, and long-term viscoelastic relaxation of the Kapton drove the dissipation of the initial overshoot energy. Consequently, the 0.2 N/mm load serves as a representative, albeit deduced, equilibrium state of the as-built structure.

7.2.3. Finite Element Validation of the Bare Prototype

The obtained frequencies from both the FEM and EMA are presented in Table 7.5. Notably, the first three fundamental membrane modes are predicted with high accuracy (error < 10%), and the first two global frame modes show acceptable agreement (error < 15%). This correlation confirms that the baseline geometric stiffness assumption for the pre-tensioned membrane is fundamentally sound.

However, a significant deviation occurs for the (2,1) and (2,2) membrane modes, where the experimental eigenfrequencies are substantially higher than the numerical predictions. This discrepancy arises because, in the finite element model, the CFRP frame acts effectively as a rigid boundary during these mid-frequency localized modes; thus, the modeled modal stiffness is almost exclusively derived from the ideal 0.2 N/mm pre-tension. Conversely, in the physical prototype, these specific modes likely induce early structural coupling. This engages the intrinsic bending and torsional stiffness of the carbon fiber frame and thereby raising the observed frequency. Furthermore, minor non-uniformities in the physical pre-tensioning of the membrane can create localized load paths concentrated through the mid and right sections of the sheet. This load localization introduces additional geometric stiffening that the mathematically ideal, uniformly distributed load case in the FEM does not capture.

Table 7.5: Comparison of experimental (EMA) and numerical (FEM) natural frequencies. The numerical model demonstrates strong agreement for fundamental membrane and global frame modes, with expected deviations in higher-order coupled modes.

Mode Shape	EMA Frequency (Hz)	FEM Frequency (Hz)	Absolute Error (%)
(1,1) Membrane (Center)	119.69	125.16	4.57
(1,2) Membrane	156.25	143.25	8.32
(1,3) Membrane	180.80	165.48	8.47
(2,1) Membrane	299.06	238.36	25.47
(2,2) Membrane	338.13	249.93	26.08
Lateral Frame	283.59	252.16	11.08
Hat Stiffener	430.94	489.86	13.67

To further validate the structural dynamic response, the Modal Assurance Criterion (MAC) was evaluated. The MAC serves as a statistical indicator that quantifies the spatial consistency between numerical and experimental mode shapes, yielding a scalar value between 0 and 1 [63]. The MAC is calculated as the normalized squared inner product of the two sets of mode shape vectors, resulting in the following mathematical definition:

$$\text{MAC}(r, q) = \frac{|\{\varphi_A\}_r^T \{\varphi_X\}_q|^2}{(\{\varphi_A\}_r^T \{\varphi_A\}_r) (\{\varphi_X\}_q^T \{\varphi_X\}_q)} \quad (7.1)$$

where $\{\varphi_A\}_r$ is the mode shape vector (eigenvector) for the r -th mode in Dataset A (FEM), and $\{\varphi_X\}_q$ is the corresponding mode shape vector for the q -th mode of Dataset X (1D LDV scan). In this application, the relevant eigenvector component is the out-of-plane nodal displacement in the z -direction (U3 in the FEM).

Because the experimental LDV scanning grid did not perfectly align with the numerical finite element mesh, the datasets first required spatial correlation. This was achieved utilizing the `scipy.spatial.cKDTree` class, an algorithm that identifies the nearest spatial neighbors to ensure correct nodal pairings [64]. As the LDV data points do not form a perfectly rectangular grid and exhibit slight geometrical slanting, manual nodal matching was essential to provide the algorithm with an accurate baseline. The four boundary condition mounting points and adjacent nodes on the outer hat profile flanges were utilized as these manual control points.

Figure 7.6 illustrates the resulting spatial grid alignment. For the selected control points, a mean residual error of 2.41 mm was achieved, which is deemed acceptable. The final overall mean mapping distance for the structure is 11.17 mm, ensuring that the physical data points were successfully mapped within approximately one element width of their corresponding numerical nodes.

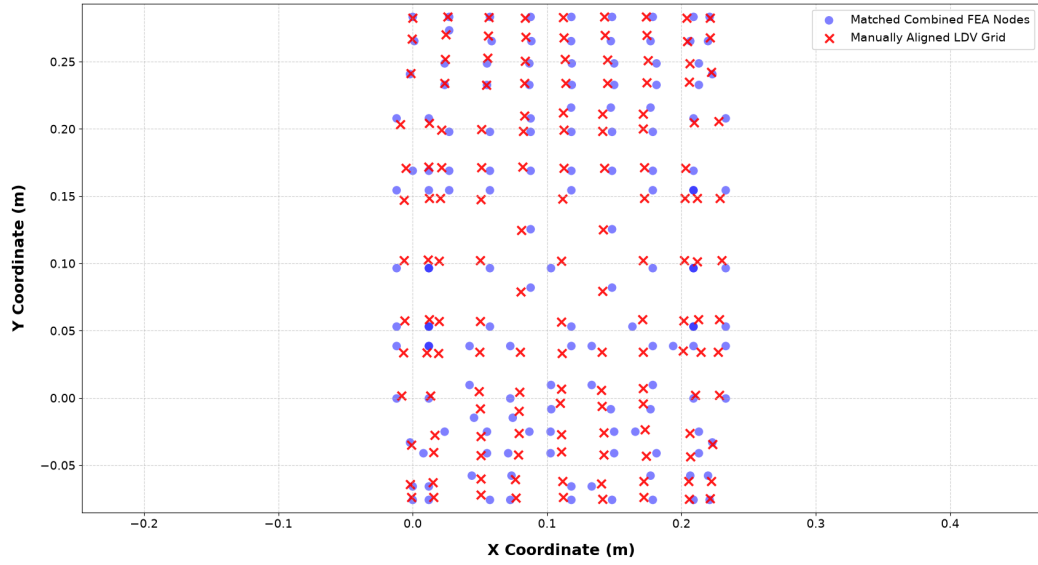


Figure 7.6: Spatial mapping alignment between the non-uniform experimental LDV grid and the numerical finite element mesh.

The final Modal Assurance Criterion (MAC) matrix is presented in Figure 7.7. The highest spatial correlation is observed in the fundamental (1,1) membrane mode, yielding a MAC value of 0.71. Furthermore, the matrix demonstrates strong orthogonality, with the vast majority of off-diagonal terms falling below the 0.15 threshold, confirming that the experimentally captured modes are mathematically distinct without significant spatial bleed.

The deviations from an ideal MAC of 1.0 along the diagonal are primarily attributed to unequal tension distribution. An angular offset during the membrane adhesion process causes the higher-order physical modes to become spatially asymmetrical. For instance, in the (1,2) mode, this physical asymmetry causes the displacement of the experimental right side to misalign with the numerical predictions. This misalignment mathematically penalizes the inner product, resulting in a reduced MAC score of 0.28.

This spatial penalty is most severe in the (1,3) membrane mode, where the correlation drops below the 0.15 reporting threshold. A visual assessment of the LDV measurements for this mode reveals that while the outer edges excite similarly to the FEM, the center of the membrane exhibits near-zero out-of-plane displacement. This physical behavior strongly supports the presence of a localized stiff band resulting from non-uniform tension. This stiff band physically suppresses the central anti-node, consequently degrading the mathematical spatial correlation.

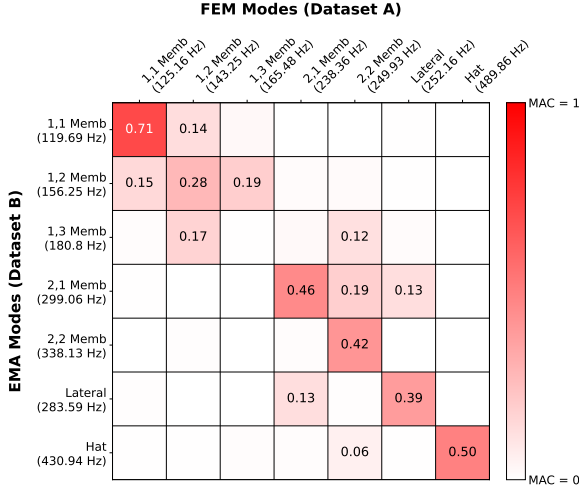


Figure 7.7: Modal Assurance Criterion (MAC) matrix correlating numerical FEM predictions based on a uniformly distributed loading representation with the experimental LDV spatial measurements of the unpopulated membrane-frame prototype.

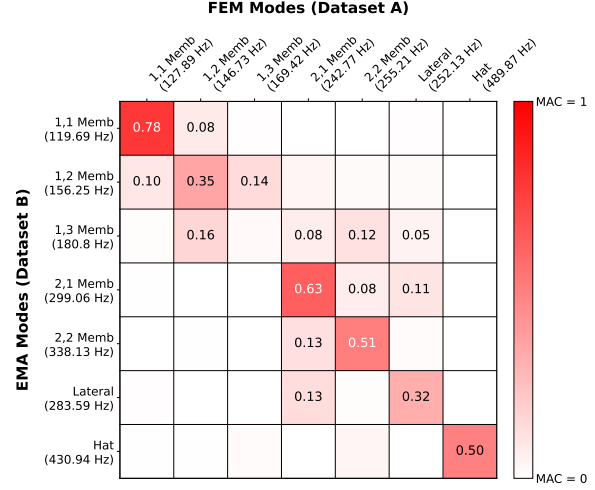


Figure 7.8: Modal Assurance Criterion (MAC) matrix correlating FEM predictions utilizing a triangular distributed load with experimental LDV spatial measurements.

To prove that manufacturing misalignment is a primary driver of the relatively low MAC correlation, the model was subjected to a non-uniform triangular load. Following an iterative tuning process, the load distribution was defined as $w(x) = -0.48 \cdot \frac{x-12}{225}$. In this formulation, -0.48 represents the peak force distribution (N/mm), 12 is the x -coordinate (in mm) of the starting point on the hat stiffener flange relative to the model's origin, and 225 mm is the total length over which the load is applied. As a result, Table 7.6 presents the updated frequency comparison. The fundamental membrane modes demonstrate strong agreement, with absolute errors remaining below 7%. The global frame modes also remained largely unaffected compared to the uniform load case, maintaining errors below 13%.

Table 7.6: Comparison of experimental (EMA) and numerical (FEM) natural frequencies under the slanted load distribution. The numerical model demonstrates strong agreement for fundamental membrane and global frame modes, with expected deviations in higher-order coupled modes.

Mode Shape	EMA Frequency (Hz)	FEM Frequency (Hz)	Absolute Error (%)
(1,1) Membrane (Center)	119.69	127.89	6.4
(1,2) Membrane	156.25	146.73	6.5
(1,3) Membrane	180.80	169.42	6.7
(2,1) Membrane	299.06	242.73	23.2
(2,2) Membrane	338.13	255.21	32.5
Lateral Frame	283.59	252.18	12.5
Hat Stiffener	430.94	489.87	12.0

As shown in Figure 7.9, the experimentally captured 1D LDV mode shapes are compared against numerical predictions subjected to the slanted load distribution. Notably, the simulated displacement fields now exhibit a spatial skew that better mirrors the physical prototype. Re-evaluating the Modal Assurance Criterion (MAC) with this updated stress field reveals a correlation improvement across all membrane modes.

For instance, the fundamental (1,1) membrane mode MAC increased from 0.71 to 0.78 strictly by transitioning from a uniform to a linear load gradient. While this proves that the macroscopic modal distortion is dominated by non-uniform pretension, the physical prototype inherently contains small-scale imperfections not captured by an idealized mathematical model. The remaining discrepancy between the numerical and experimental mode shapes can be attributed to the following physical phenomena:

- **Localized boundary compliance:** The finite element model assumes a perfectly rigid, continu-

ous bonding condition between the membrane and the stiffeners via a tie constraint. In reality, variations in adhesive thickness, or localized regions lacking adhesive entirely, introduce micro-compliance. These small-scale variations locally reduce membrane stiffness, fundamentally altering the physical mode shapes captured by the LDV.

- **Air mass and damping:** While the physical prototype is tested in ambient atmospheric conditions—where the surrounding air acts as an added mass—the finite element model operates in a perfect vacuum. Because the membrane lacks inherent bending stiffness, this inertial aerodynamic effect heavily influences the measured out-of-plane dynamics.
- **Micro-wrinkling:** A linear slanted load assumes the membrane remains perfectly flat under varying tension. In reality, unequal pretension inherently introduces microscopic wrinkles near the boundaries, drastically altering the local geometric stiffness in ways a linear gradient cannot replicate.

Despite these macroscopic improvements, the (1,3) mode did not see an improved MAC score. This lack of geometric correlation stems from the severe physical distortion captured by the 1D LDV. Because higher-order modes have shorter spatial wavelengths, they are exponentially more sensitive to the localized micro-wrinkling mentioned above. In the measurement, the two sides of the membrane showcase significant asymmetry, and the center fails to form a clear anti-node. This results in a large localized area of the physical membrane behaving significantly differently than the mathematically perfect numerical model.

Furthermore, while the (2,1) and (2,2) modes improved with the addition of the triangular force distribution, their MAC values remain relatively low. The primary driver for this discrepancy is unmodeled frame-membrane coupling. In the LDV scans, the physical frame exhibited measurable participation during these higher-order modes, whereas the numerical model predicted purely isolated membrane movement. This undetected dynamic coupling in the experimental setup, combined with spatial asymmetry, heavily penalizes the final MAC correlation for these coupled modes.

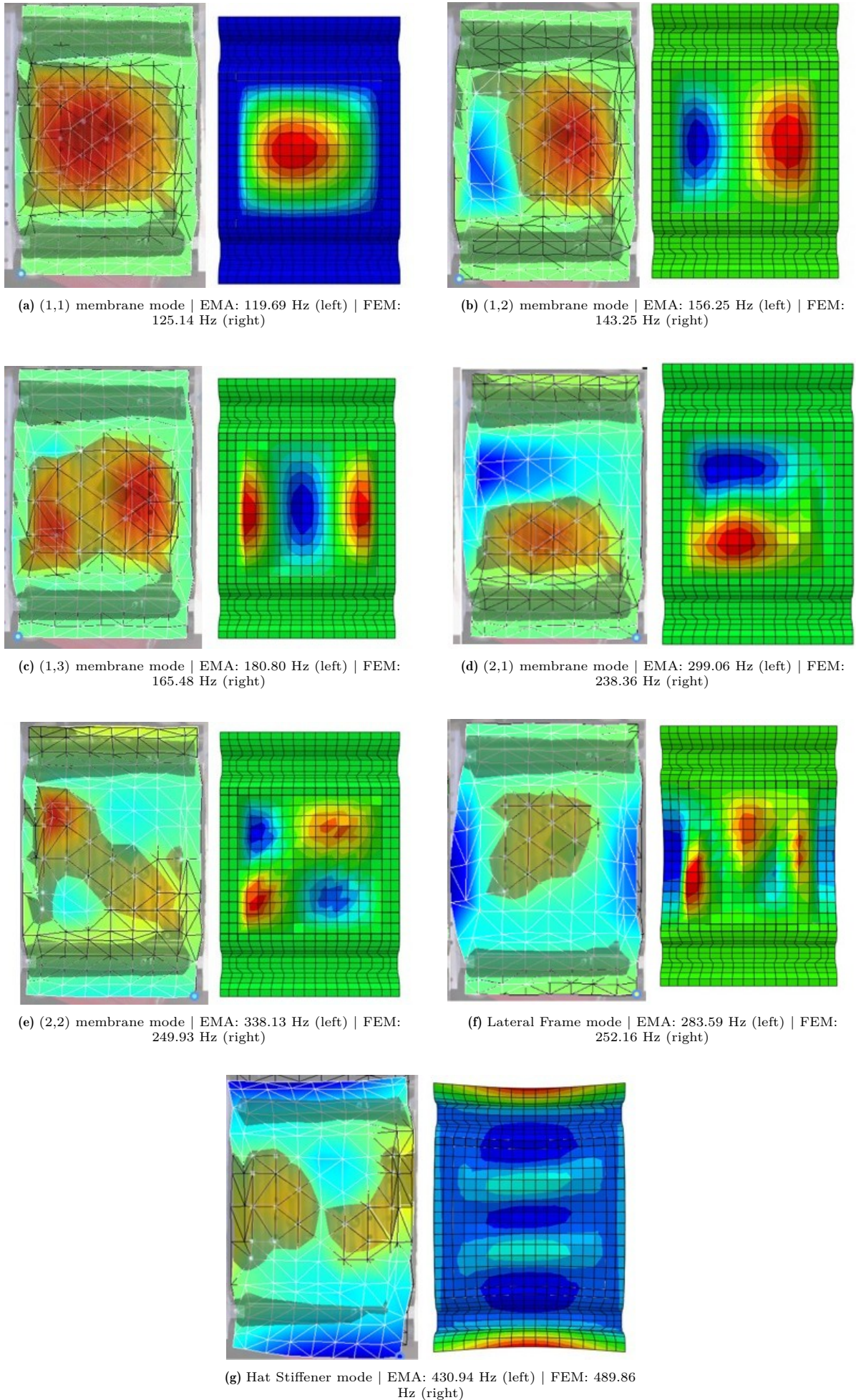


Figure 7.9: Comparison of experimental (EMA) and numerical (FEM) natural frequencies and mode shapes. Each subfigure presents the spatial LDV measurements on the left against the corresponding numerical FEM predictions on the right.

7.2.4. Experimental Modal Analysis of the Populated Structure

Following the baseline structural characterization, the dynamic response of the populated prototype was evaluated. The integrated mPower DragonScales arrays, comprising an active 6×6 central module flanked by 9×1 and 6.5×1 non-functional dummy strings, were arranged to simulate the distributed mass of a high-density operational solar array. Consequently, the total mass added to the solar array module was approximately 30 ± 5 g, bringing the total prototype mass to 324 g. To ensure a direct comparative analysis with the bare structure, these measurements were conducted utilizing the identical LDV scanning parameters previously established in Table 7.4.

The resulting Frequency Response Function (FRF) and corresponding coherence are presented in Figure 7.10. Similar to the bare membrane configuration, coherence below 20 Hz remains unreliable due to the low-frequency mechanical limitations of the shaker's pseudo-random excitation profile. A distinct drop in coherence is also observed precisely at 50 Hz. This is a well-documented instrumentation artifact caused by standard European mains electrical interference coupling into the data acquisition cables, rather than a physical structural anomaly. Furthermore, a sharp magnitude peak in the H1[Vib & Ref 1] response at roughly 460 Hz corresponds directly to the rigid-body resonance of the underlying aluminum test rig.

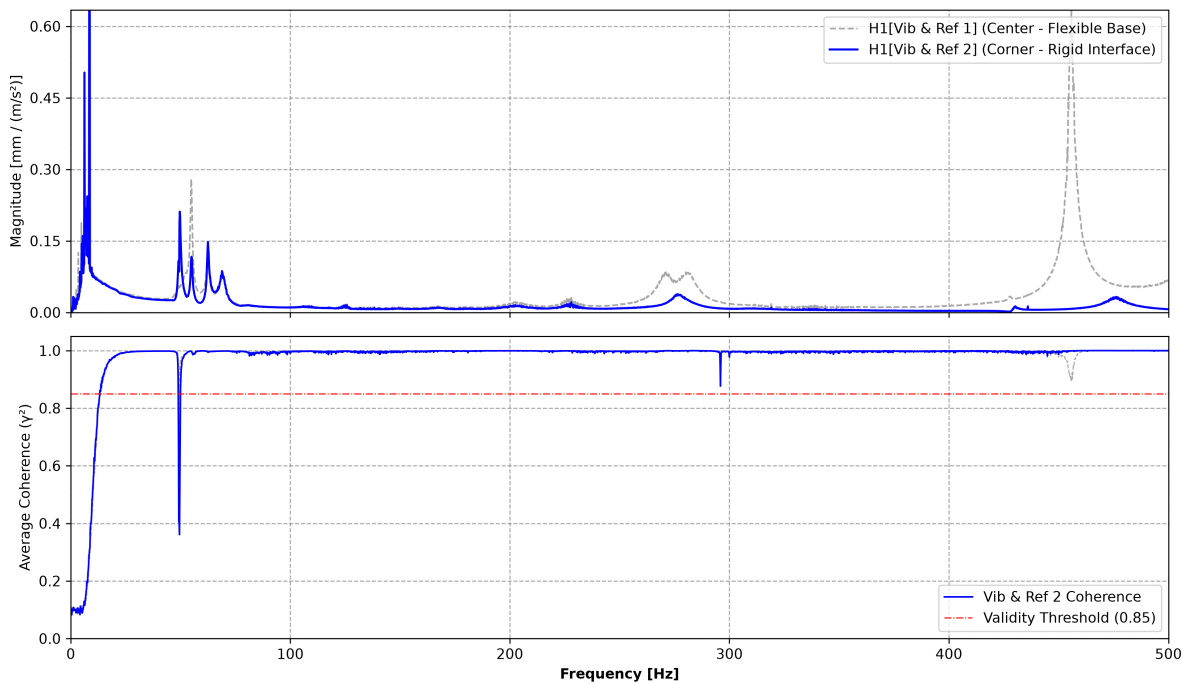


Figure 7.10: Displacement frequency response function (H1) and coherence of the fully populated prototype structure under pseudo-random base excitation.

The extracted operational deflection shapes for the populated structure are detailed in Figure 7.11. While the fundamental macroscopic topologies align well with the bare structure, the localized spatial continuity is noticeably improved. This is remarkably evident in the third mode (68.91 Hz). In the unpopulated prototype, this specific membrane mode was heavily distorted by asymmetrical pretension. However, bonding the rigid solar cells to the Kapton film introduces substantial localized bending stiffness. The footprints of the cell strings effectively bridge the high and low-tension zones, homogenizing the stress distribution and forcing the membrane to deform with a highly pronounced, mathematically symmetric central anti-node.

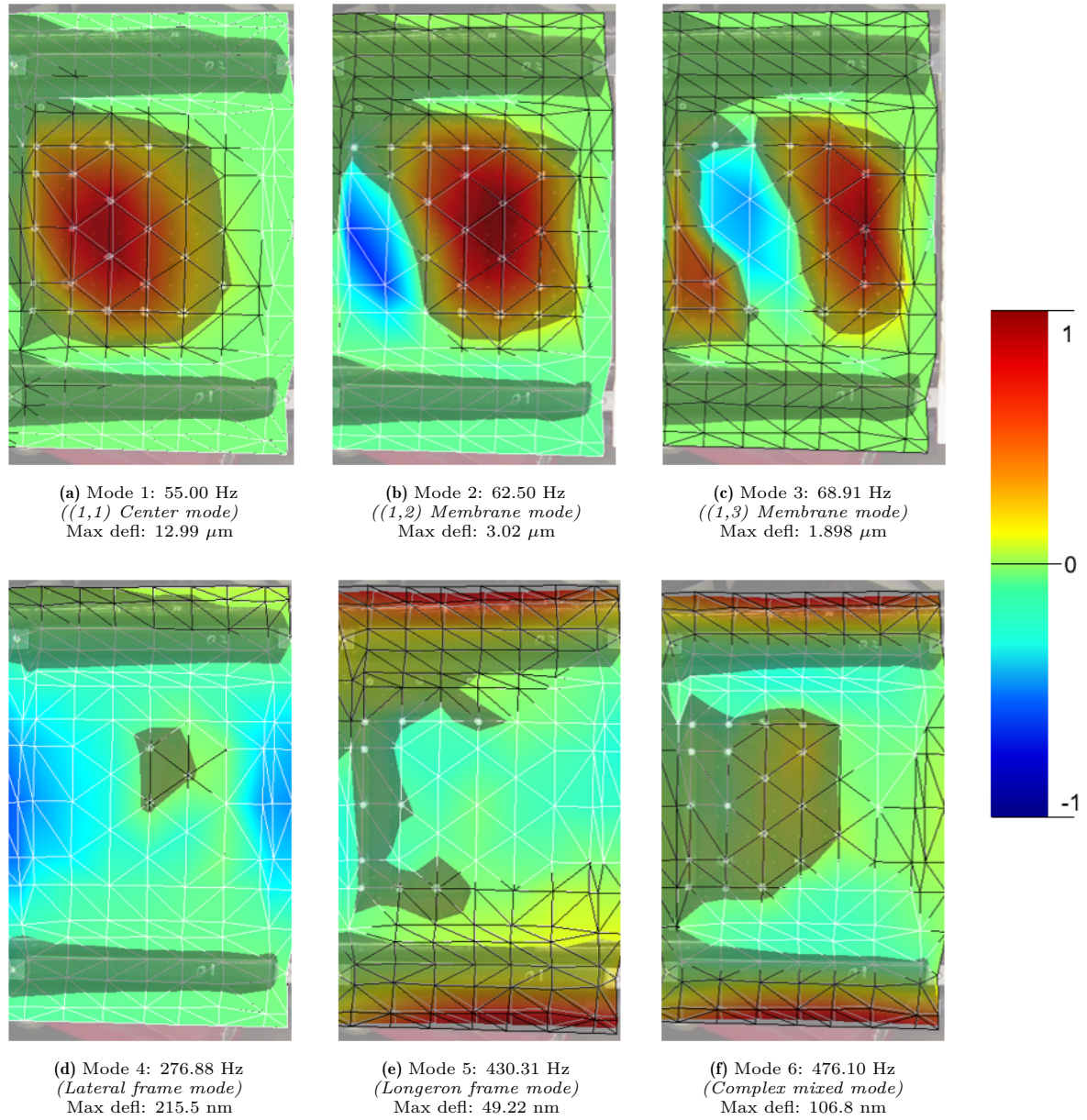


Figure 7.11: Experimental mode shapes of the populated membrane-frame prototype obtained via 1D scanning laser Doppler vibrometry. The color map represents the auto-scaled, normalized out-of-plane deflection to illustrate modal phase and spatial continuity. The maximum absolute deflections provided in the sub-captions correspond to the forced structural response under a $0.0047 G_{rms}$ pseudo-random base excitation.

To quantify the dynamic influence of the solar cells, a stacked comparison of the spatial FRFs is presented in Figure 7.12, with the corresponding frequency shifts tabulated in Table 7.7. The fundamental membrane modes undergo a drastic 54.0% to 61.9% reduction in natural frequency. Consistent with the fundamental mass-stiffness relationship defined in Equation 2.2, this severe drop is physically expected; the addition of the silicon cells and space-grade adhesive constitutes a massive localized mass penalty relative to the negligible baseline mass of the thin-film polyimide. Conversely, the lateral frame and hat stiffener modes exhibit negligible frequency shifts of 2.4% and 0.1%, respectively, demonstrating that their inherent structural mass and stiffness easily dominate the added mass of the arrays.

Finally, the stacked FRF reveals a significant response attenuation for higher-order modes in the populated configuration. Subjected to the same low-energy base excitation, the heavier populated membrane simply cannot accelerate to the same displacement magnitudes, effectively damping the higher-frequency modal participation. While an increased excitation force would likely amplify these modes, they cur-

rently fall outside the primary low-frequency dynamic envelope of interest.

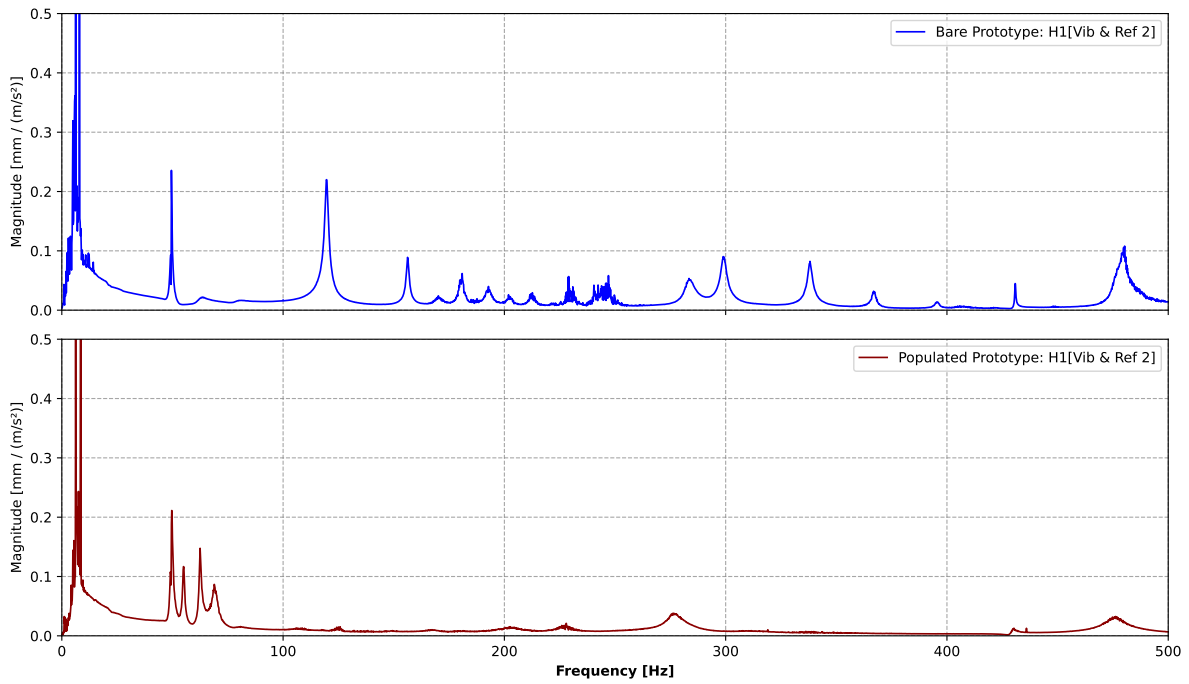


Figure 7.12: Stacked frequency response comparison (H_1 [Vib & Ref 2]) highlighting the inertial frequency shift between the bare structural membrane (top) and the fully populated solar array (bottom).

Table 7.7: Comparison of experimental natural frequencies between the bare membrane prototype (f_{bare}) and the populated structure with solar cells (f_{cells}).

Mode Shape	f_{bare} (Hz)	f_{cells} (Hz)	Frequency Drop (%)
(1,1) Membrane	119.69	55.00	54.0
(1,2) Membrane	156.25	62.50	60.0
(1,3) Membrane	180.80	68.91	61.9
Lateral Frame	283.59	276.88	2.4
Hat Stiffener	430.94	430.31	0.1

7.2.5. Finite Element Validation of the Populated Prototype

To create an accurate Finite Element (FE) model for the populated prototype, numerous challenges must be addressed. The primary difficulty lies in accurately representing the structural interaction between the highly compliant Kapton membrane and the discrete, stiff solar cell assemblies. This complexity is further compounded by the need to model the underlying adhesive layers and electrical routing. Furthermore, the correlation was constrained by strict computational mesh limits (restricted to 1000 nodes due to student software licensing), necessitating highly efficient modeling strategies.

The first approach assumed the solar cell modules acted purely as non-structural mass on the membrane. By distributing the 30 g mass across the membrane using the same bare model baseline established in Section 7.2.3, it was possible to capture the primary mass loading effect of the cells. However, this approach inherently loses the localized stiffening effects provided by the discrete cells. Consequently, this method yielded the first three membrane eigenmodes at 35.8, 41.0, and 47.3 Hz respectively, which are approximately 35% lower than the measured modes.

While the mode order remained consistent and the global shift to a lower frequency was captured, the lack of local stiffness severely penalized the natural frequencies. Dictated by the governing dynamic proportionality (Equation 2.2), it is possible to mathematically compensate for this missing structural

stiffness by artificially reducing the distributed non-structural mass. When the distributed mass was lowered to 12 g (40% of the actual added cell mass), the FE model closely aligned with the experimental data, as presented in Table 7.8. Through this adjustment, a very small error ($< 5\%$) was obtained for the first three membrane modes, while the frame-dominated lateral and longeron modes remained fairly stable. Although it yields minimal error for global dynamic predictions, this method remains a macroscopic approximation. Consequently, it lacks the localized physical fidelity required for detailed stress analyses of populated thin-film membranes.

Table 7.8: Comparison of LDV modal frequencies against the FE model (assuming 40% cell mass as distributed non-structural mass).

	(1,1) mode	(1,2) mode	(1,3) mode	Lateral mode	Longeron mode
FEM	54.6	62.6	72.3	250.9	489.3
LDV	55.0	62.5	68.91	276.88	430.31
Error	0.7%	0.2%	4.7%	10.4%	12.1%

Before adopting this simplified distributed mass approach, alternative modeling techniques were evaluated to capture the localized stiffness of the solar cell modules. One such method involved partitioning the membrane into numerous surfaces assigned with skin elements to represent the SPMs. To execute this effectively, the membrane behavior was approximated using a General Shell Stiffness section. The ABD matrix was configured to mimic a pure membrane by assigning near-zero values (10^{-8} and 10^{-9}) to the bending stiffness matrix (D) to maintain numerical stability. This approach enabled the assignment of a composite section to the skins, encompassing the specific material layers of the modules (RTV, Kapton, silicon, and cover glass).

However, modeling the area as a continuous composite section rather than discrete solar cell compartments significantly overestimates the global stiffness. This overestimation artificially widens the frequency gaps between the predicted eigenmodes. Attempts to properly discretize the cells by defining connector elements between the skin mesh elements proved numerically unstable and quickly exceeded the 1000-node limit of the software license. Without this node restriction, a significantly finer mesh (e.g., 1×1 mm elements) could be implemented. This resolution would allow the physical hinge lines to remain unpopulated by cells, while the precise composite layup could be assigned strictly to the active cell regions.

Although the 40% effective-mass approximation successfully predicts the global eigenvalues, its continuous formulation is fundamentally limited. Specifically, it lacks the localized stiffness discontinuities required to accurately capture the spatial eigenvectors. A detailed spatial correlation of this populated configuration, including an evaluation of the Modal Assurance Criterion (MAC) and the physical limitations of the smeared-mass assumption, is documented in Appendix H.

Conclusions and Recommendations

8.1. Summary of research findings

This thesis investigated to what extent the dynamic characteristics, specifically natural frequencies and mode shapes, of a semi-flexible membrane-frame solar array module can be predicted using an as-built finite element model (FEM), and how closely these predictions correspond to physical responses measured through experimental modal analysis (EMA). Numerical simulations developed in Abaqus were correlated with Laser Doppler Vibrometry (LDV) measurements acquired using a PSV-500 scanning head and V550 shaker.

The results demonstrate that an as-built FEM reproduces the fundamental dynamic behavior of the investigated membrane-frame solar array architecture. For the unpopulated prototype, the first three membrane-dominated natural frequencies were predicted within 9%, while frame-dominated modes were predicted within 14%. For the populated prototype, a simplified distributed-mass representation reproduced the first three membrane-dominated modes within 5% when an effective mass corresponding to 40% of the physical solar cell mass was assigned. Although correlation deteriorated for higher-order modes and populated mode shapes, the model successfully captured the fundamental dynamic characteristics of both configurations, making it suitable for dynamic assessment and early-stage design studies.

The findings associated with each sub-research question are summarized below:

- SQ-1 Primary mechanical design driver** – The primary mechanical driver identified for the investigated architecture is the susceptibility of the low-stiffness membrane to out-of-plane deformation under dynamic launch loading. In particular, membrane-frame interfaces represent critical regions because anti-phase membrane modes can produce significant relative motion and therefore increase the potential for contact between neighboring solar cell modules. The results demonstrate that the dynamic behavior of the flexible membrane must therefore be considered alongside the structural response of the supporting frame.
- SQ-2 Pretension, load factor, and non-linear response** – The maximum out-of-plane membrane deflection increases nonlinearly with load factor, while increasing pretension improves geometric stiffness and reduces deflection. However, excessive pretension causes inward deformation of the CFRP frame, reducing the effective membrane tension and limiting the achievable stiffness. Consequently, membrane pretension cannot be optimized independently of frame stiffness. An effective design therefore requires a balance between membrane stability and frame compliance.
- SQ-3 Numerical modeling strategy** – A hybrid FEM consisting of S4 shell elements for the CFRP frame and M3D4R membrane elements for the Kapton substrate successfully represented the fundamental dynamic behavior of the structure. Incorporating the physical pretensioning sequence and manufacturing-induced stress asymmetry improved correlation with experimental measurements, demonstrating the importance of modeling as-built effects.
- SQ-4 Non-contact acquisition** – Scanning Laser Doppler Vibrometry was shown to be well suited to the dynamic characterization of the ultra-lightweight polyimide membrane. The non-contact measurement approach avoided the additional mass and stiffness associated with conventional contact

sensors. A 0.03 V pseudo-random broadband excitation with 10 spectral averages produced highly coherent measurements over the reliable frequency range of the test while maintaining the prototype within its linear elastic regime.

SQ-5 Experimental limitations and boundary inaccuracies – Several facility-related limitations affected the experimental campaign. Frequencies below approximately 20 Hz were difficult to excite due to the limited shaker stroke, while electrical interference introduced a coherence reduction around 50 Hz. Furthermore, a resonance of the aluminum test rig near 460 Hz introduced facility-level dynamic coupling at higher frequencies. The most significant discrepancy between the numerical and experimental configurations originated from the physical boundary conditions. Unlike the idealized FEM supports, the aluminum test rig possessed finite stiffness and supported the prototype through an overhanging mounting configuration. Consequently, the transmitted excitation was influenced by local structural compliance and dynamic amplification effects that were not explicitly represented in the numerical model. This highlights the sensitivity of membrane-frame structures to boundary-condition representation and demonstrates the importance of accurately characterizing the supporting test hardware during experimental validation.

SQ-6 Inertial impact of solar cells – Populating the array resulted in a substantial reduction in the fundamental membrane natural frequencies, with reductions of approximately 54.0–61.9%. The addition of the solar cells also altered the modal deformation patterns, including the formation of a distinct symmetric anti-node in the (1,3) mode. In contrast to the frame-dominated response, the membrane dynamics were found to be strongly mass-dominated. A distributed effective-mass representation corresponding to 40% of the physical solar cell mass reproduced the first three populated-array natural frequencies within 5% of the experimental results. Furthermore, the dynamic behavior of the global frame was considerably less affected by the addition of the solar cells than that of the membrane. The lateral profile and hat-stiffener modes shifted by only 2.4% and 0.1%, respectively. This indicates that the solar cell mass primarily influences the membrane-dominated response while having a comparatively limited effect on the stiffness-dominated frame response.

Overall, the results demonstrate that an as-built FEM can provide a representation of the fundamental dynamic behavior of the investigated semi-flexible membrane-frame solar array when geometric imperfections, pretensioning, boundary conditions, and the distinction between mass-dominated membrane behavior and stiffness-dominated frame behavior are appropriately considered. The model demonstrated good agreement with the experimental measurements for the fundamental and low-order modes, while correlation deteriorated for higher-order modes. Consequently, the developed modeling approach is considered suitable for the dynamic assessment, validation, and early-stage design of membrane-frame solar-array structures. While the effective-mass representation accurately reproduced the fundamental populated frequencies, it did not fully capture the modified modal deformation patterns introduced by photovoltaic cell stiffness distributions.

8.2. Contribution to the research field

This research contributes to the understanding and modeling of ultra-lightweight membrane-frame space structures by combining experimental testing, finite element modeling, and consideration of the as-built manufacturing configuration. The main contributions of the thesis are:

- **State-of-the-art synthesis:** Provided a consolidated assessment of existing semi-flexible solar-array architectures and identified the principal structural and dynamic challenges associated with membrane-frame systems.
- **Experimental methodology for membrane-frame structures:** Established and validated a non-contact experimental modal analysis methodology using a PSV-500 scanning laser Doppler vibrometer and V550 electrodynamic shaker. Unlike many existing studies that focus on isolated membranes, the method enabled characterization of the coupled dynamic interaction between the both the membrane and supporting frame, without introducing sensor mass-loading effects.
- **As-built finite element modeling framework:** Developed and experimentally validated an as-built FEM methodology incorporating pretensioning,
- **Photovoltaic integration modeling:** Quantified the influence of photovoltaic-cell integration on

the dynamic response of the membrane-frame structure and demonstrated that an effective-mass approximation corresponding to approximately 40% of the physical solar cell mass reproduced the fundamental populated-array frequencies within 5% of experimental measurements.

- **Characterization of thin-film integration mechanics:** Quantified the viscoelastic relaxation behavior of the Kapton membrane and documented practical considerations associated with the assembly and tensioning of thin-film structures. These observations provide additional insight into the sensitivity of membrane pretension and dynamic behavior to the physical integration process.

8.3. Limitations

The experimental and numerical validation of the prototype was subject to several practical limitations associated with the prototype geometry, test facility, material representation, and available computational resources. These limitations define the conditions under which the conclusions of this thesis can be applied:

- **Prototype representativeness:** The experimental article represents a sub-scale, single-panel membrane-frame solar-array module. Consequently, the measured modal characteristics cannot be directly interpreted as representative of a full-scale flight system. Additional effects such as panel coupling, hinge dynamics, stowed-configuration interactions, and flight-representative boundary conditions may significantly alter the dynamic response.
- **Material representation:** Bare Kapton polyimide was used to isolate the fundamental structural behavior of the membrane. Flight-representative substrates may contain embedded electrical routing, stiffening meshes, protective layers, and additional hardware that will influence both mass and stiffness distributions.
- **Experimental facility limitations:** The experimental campaign was affected by reduced excitation authority below approximately 20 Hz, electrical interference near 50 Hz, and structural coupling of the aluminum support rig around 460 Hz. Furthermore, air damping and the finite stiffness of the test fixture and overhanging mounting configuration introduced compliance and dynamic amplification effects that were not explicitly represented in the finite element model.
- **Numerical-model limitations:** The finite element model demonstrated good predictive accuracy for the fundamental and low-order modes but showed reduced correlation for higher-order modal behavior. In addition, software-imposed mesh restrictions prevented explicit modeling of the solar cell assemblies, cover glass, adhesive layers, and local hinge stiffness. Consequently, the populated configuration relied on an iteratively calibrated 40% effective-mass approximation and should primarily be considered validated for fundamental dynamic behavior.

8.4. Recommendations for future work

The findings and limitations of this study identify several opportunities for improving both the numerical model and the experimental representation of a flight-representative semi-flexible solar array.

Improvements for Finite Element Modeling

- **High-fidelity solar cell discretization:** Future numerical models should explicitly represent the composite layers and localized stiffness characteristics of the individual solar cells. Removing the current computational restrictions would allow the cell geometry, hinge regions, adhesive interfaces, and localized stiffness distributions to be modeled directly, eliminating the need for the effective-mass approximation.
- **Multi-panel coupled dynamics:** The modeling approach should be extended from the isolated single-panel configuration to a multi-panel solar array. In particular, the dynamic behavior of stacked panels in the stowed configuration and coupled panels in the deployed configuration should be investigated, including inter-panel contact and hinge behavior.
- **Full-scale numerical scaling:** The modeling methodology should be applied to a full-scale configuration to investigate the resulting shifts in natural frequencies and mode shapes. This would provide an opportunity to assess whether the dynamic characteristics observed in the sub-scale

prototype can be reliably extrapolated to a flight-representative geometry.

Prototype Design Iterations

- **Flight-representative substrates:** Future prototypes should incorporate a representative substrate including a stiffening mesh, embedded electrical routing, and other relevant flight hardware. The effect of these additions on membrane mass, bending stiffness, pretension relaxation, and dynamic response should then be experimentally assessed.
- **Structural frame optimization:** The observed inward deformation of the lateral CFRP profiles under membrane pretension indicates an opportunity for further structural optimization. Future iterations should investigate alternative CFRP cross-sections and laminate layups to increase the in-plane bending stiffness of the frame and reduce pretension-induced deformation.

Enhancements to Experimental Testing

- **Vacuum testing:** Experimental modal analysis under vacuum should be performed to provide a more representative approximation of the on-orbit dynamic environment by removing aerodynamic damping effects. Comparison between ambient-pressure and vacuum measurements would also allow the influence of aerodynamic damping on the membrane response to be quantified.
- **Full-scale experimental validation:** A future test campaign should investigate a full-scale panel using a higher-capacity electrodynamic shaker and a more rigid test fixture. Such testing would provide the experimental validation required to assess the scalability of the present sub-scale modeling approach.

Broader Research Directions

The large surface area and high compliance of membrane-frame structures make acoustic-structural coupling an important consideration during launch. Future work should therefore investigate the response of the architecture under representative acoustic loading conditions. In addition, the long-term stability of membrane pretension should be evaluated under thermal cycling and sustained loading. Quantifying the effects of viscoelastic relaxation and temperature-dependent material behavior would improve understanding of the long-term dynamic stability of semi-flexible solar-array systems.

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Literature landscape mapping and theoretical framework

This appendix details the landscape search conducted to identify existing membrane-frame architectures for space applications. While Chapter 2 presents the primary background information relevant to this thesis, this section outlines the search methodology and the resulting theoretical framework tables, which form the basis for the higher-level architectural trade-off discussed in Chapter 3.

This search was necessary because, during the initial phase of the research, the prevalence and concurrent technological status of semi-flexible solar arrays were largely unknown. Furthermore, mapping the literature regarding solar cell technology, structurally analogous systems, and the space environment provided a vital foundation for the subsequent design choices of the membrane-frame architecture. To achieve this, the following boolean search query was formulated based on the preliminary main research question:

((Solar OR Photovoltaic OR Photo-voltaic OR PV) AND (Architecture OR Array OR Modular OR Panel OR panels)) AND (Flexible OR Semi-rigid OR Semi-flexible OR hybrid OR Tensioned OR Blanket OR Sheet OR Tensile) AND (Composite OR Lattice* OR Frame* OR Grid OR Structure OR Truss) AND Vibration**

This query was executed in Dimensions, a powerful abstracting and indexing database capable of mapping the interconnected nature of research outputs [65]. Restricting the search parameters to titles and abstracts yielded exactly 254 publications and 193 patents, representing a highly targeted and manageable dataset for scanning. The results were subsequently exported to VOSviewer to visualize the interconnections between the various terms captured by the query, as presented in Figure A.1. To ensure visual clarity, the map was configured to display only the 40 most occurring terms utilizing binary counting. VOSviewer organized these terms into three major thematic clusters:

- **Red cluster:** Focus on macroscopic performance, power generation, efficiency, and environmental loads.
- **Green cluster:** Focus on spacecraft dynamics, attitude control, reaction wheels, and actuators.
- **Blue/Yellow cluster:** Focus on analytical methods, including dynamic modeling, dynamic analysis, and flexible solar arrays.

What is notable is that the specific term 'flexible solar array' is treated significantly less in the literature compared to the macroscopic term 'array'. This visualization serves as empirical proof that semi-flexible membrane-frame solar arrays are not a well-established topic compared to traditional rigid architectures. Consequently, it confirms that this thesis operates in a distinct interdisciplinary gap between structural composite layout, membrane mechanics, and modal/vibrational analysis.

In turn, for the most relevant found sources from this search query (but also other databases), below are the made theoretical framework tables that supported familiarization with the relevant domains.

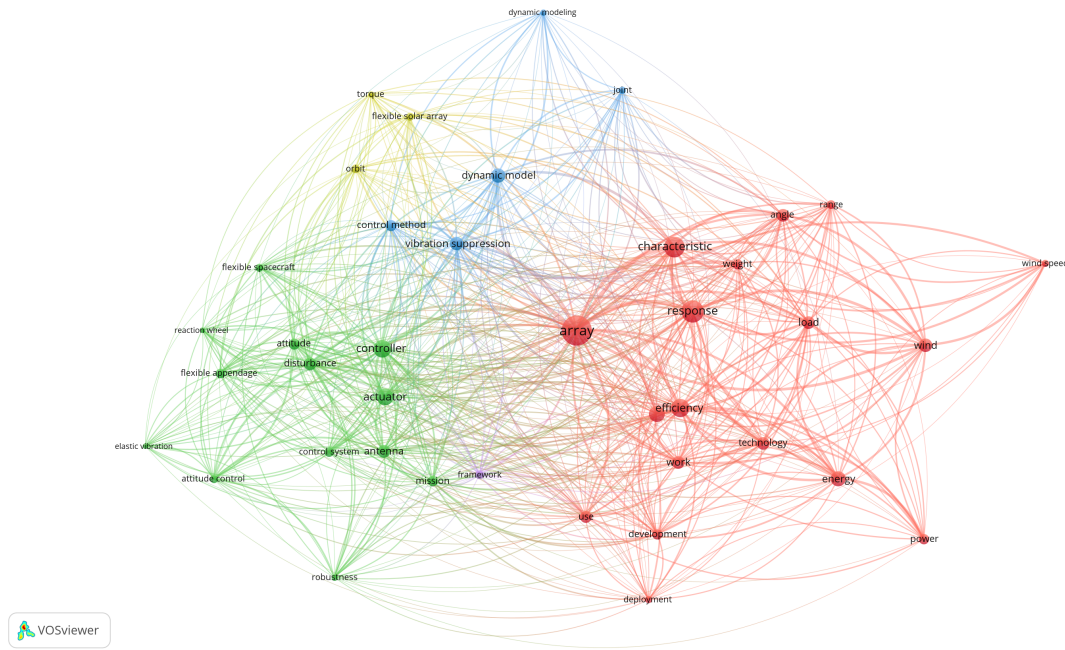


Figure A.1: Search query landscape of interconnected terms within the titles and abstracts.

Table A.1: Theoretical framework and methodology synthesis for selected literature.

Reference & Discipline	Theory / Hypothesis	Key Concepts	Methodology & Assumptions	Problem Insights
Zimmermann et al. (2019) <i>Eurostar Neo Solar Array</i> <i>Spacecraft Power Systems & PV Design</i>	A hybrid architecture (rigid backbone with semi-rigid lateral panels) can deliver approx. 50% more power than conventional designs within identical volume and mass constraints.	Semi-rigid panels: Corrugated sheets for deployed stiffness. Substrate: Charge-dissipative Aramid fiber (carbon black). C-Springs: Passive deployment hinges acting as electrical conductors. Damping Interface: Specialized grips to prevent in-plane sliding during stowed vacuum testing.	Vacuum Sine Testing: Assumes air damping masks launch loads; requires vacuum shaker enclosure. 1/3 Scale: Representative testing justified by identical boundary conditions and acceleration nodes. Thermal Fatigue: Coupons subjected to 15-year GEO thermal cycling (-190/+130°C).	Vulnerability: Semi-rigid sheets lack honeycomb stiffness, risking high-amplitude vibrations in vacuum. Stowage Optimization: Thin profile allows two lateral panels in the volume of one rigid panel. Validation: Measured resonance frequencies successfully validated the linear FEM.

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Table A.1: Theoretical framework and methodology synthesis for selected literature (continued).

Reference & Discipline	Theory / Hypothesis	Key Concepts	Methodology & Assumptions	Problem Insights
Mejía Escobar & Algora (2025) <i>Potential of flexible PV assemblies</i> <i>Flexible Substrates & Structural Integration</i>	Flexible, lightweight solar arrays enable extreme specific power (W/kg) and compact stowage (kW/m^3), making them essential for mass/volume-constrained next-generation space missions.	PVA Families: Classification by integration (deployable vs. body-mounted) and substrate (rigid vs. flexible/hybrid). Solar Cell Assembly (SCA): The fundamental block comprising the bare cell, cover glass, and interconnectors.	Longitudinal Assessment: Comparative analysis of BOL/EOL performance across historical missions under standard AM0 radiation. Mass Offset Assumption: High cell-level specific power is significantly reduced at the system level by the mass of passive structural components (mechanisms, booms).	Interconnector Fatigue: 20–30 μm metal components degrade due to vibrations, bending, and thermal cycling, causing open strings (typically mitigated via plated Kovar). Substrate Criteria: Requires high tensile strength, minimal elongation, and extreme thermal resistance. Deployment Limits: Z-folded mechanism stiffness limits potential for miniaturization. Patent Overview: Provides a survey of public patents related to flexible modules.
Morozov & Lopatin (2012) <i>Flexible composite membrane analysis</i> <i>Structural Dynamics</i>	Rigid-Flexible Combination Theory: A semi-rigid array can be accurately modeled as a pretensioned membrane (fiber net/cells) supported by flexible beam columns (perimeter frame).	Membrane Model: 2D system where bending stiffness is neglected; restoring force is purely tension-based. Flexible Beam Columns: Perimeter frame is modeled as beams compressed by the membrane's pretension.	Galerkin Method: Used to approximate the continuous vibrating system. Membrane Approximation: Modeled as an orthotropic plate with neglected bending stiffness coefficients. Transverse Pressure Loading: Assumes load $p = B_p \cdot n_z \cdot g$. Validation: Analytical solution verified against FEM (max deviation -6.75%).	Deflection Equations: Provides formulas for max deflection based on geometry and pretension, compatible with STAGE similitude. Limitation: Final pretension selection requires full-array buckling and strength analysis, not just transverse pressure. Representative Data: Supplies baseline dimensions, tension, and material properties.
Kellermeier et al. (1984) <i>Retractable ULP Solar Array</i> <i>PV Design & Deployment Mechanisms</i>	Reversible deployment capability can be integrated into ultra-lightweight panels with only a 4–6% performance penalty, providing full mechanism redundancy for retrievable platforms.	ULP Concept: Third-generation hybrid combining rigid and flexible array advantages. Closed Cable Loop (CCL): Mechanism ensuring synchronized, simultaneous deployment. Central Drive: Dual electric motor unit powering wing deployment and retraction. Hinge Units: Integrated main-beam interlinks.	No 'Dead Masses': Assumes tight stowing eliminates the need for protective inter-panel padding materials. Aspect Ratio Validation: Establishes feasibility of large aspect ratios ($W/L = 3$) using inter-panel stiffening elements. Hold-Down Architecture: Two-stage release sequence (90° primary turn followed by stack deployment). Eureca Heritage: Based on the 3 kW four-panel MBB wing design (1976–1978).	Reversibility Trade-off: Proves mechanical retraction is achievable for membrane-frame configurations without excessive mass penalties. Stowage Reliability: Highlights the importance of sequential hold-down release architectures in preventing jamming during deployment.

B

Analytical expression of the out-of-plane deformation of a membrane

```
import numpy as np
import matplotlib.pyplot as plt
from mpl_toolkits.mplot3d import Axes3D
from matplotlib.ticker import AutoMinorLocator

# 1. MATERIAL & GEOMETRY DEFINITIONS

def get_kapton_properties(thickness):
    """Returns isotropic material properties for standard Kapton film."""
    return {
        'h': thickness,
        'rho': 1420.0,      # [kg/m^3]
        'E': 2.5e9,        # [Pa]
        'nu': 0.34,        # [-]
        's_y': 66e6,       # [Pa] Yield Stress
        'CTE': 20e-6       # [/K] Coefficient of Thermal Expansion
    }

def compute_plate_stiffness_isotropic(props):
    """Computes effective membrane stiffness coefficients for an isotropic film."""
    E = props['E']
    nu = props['nu']
    h = props['h']
    B = E * h / (1 - nu**2)
    return {
        'B': B,
        'B11': B,
        'B22': B,
        'B12': nu * B,
        'B33': ((1 - nu) / 2) * B # Shear stiffness G * h
    }

def compute_pressure(props, nz=1.0, cell_mass_area=1.8, g=9.81):
    """Computes transverse pressure due to acceleration (launch loads).
    Includes the self-weight of the membrane and the attached solar cells."""
    h = props['h']
    rho = props['rho']
    membrane_mass_area = rho * h
```

```

total_mass_area = membrane_mass_area + cell_mass_area
p = total_mass_area * nz * g
return p

```

2. STRUCTURAL MECHANICS SOLVER

```

def compute_coefficients(a, b, B11, B22, B12, B33, Tx, Ty):
    """Computes the Galerkin non-linear large deflection coefficients."""
    pi2 = np.pi**2
    pi4 = np.pi**4

    a11 = 9 * (4 * B11 * (pi2 / a**2) + B33 * (pi2 / b**2))
    a12 = (64 / (a * b)) * (B12 + B33)
    a21 = a12
    a22 = 9 * (4 * B22 * (pi2 / b**2) + B33 * (pi2 / a**2))

    b1 = (6 / a) * (2 * B11 * (pi2 / a**2) - (B12 - B33) * (pi2 / b**2))
    b2 = (6 / b) * (2 * B22 * (pi2 / b**2) - (B12 - B33) * (pi2 / a**2))

    c11 = 96 * (Tx * (pi2 / a**2) + Ty * (pi2 / b**2))
    c12 = (128 / a) * (2 * B11 * (pi2 / a**2) + (2 * B12 + B33) * (pi2 / b**2))
    c21 = (128 / b) * (2 * B22 * (pi2 / b**2) + (2 * B12 + B33) * (pi2 / a**2))
    c22 = 3 * (3 * B11 * (pi4 / a**4) + 2 * (3 * B12 - 2 * B33) * (pi2 / a**2) * (pi2 / b**2)
    + 3 * B22 * (pi4 / b**4))

    c33 = c22 + (b1 * (a22 * c12 - a12 * c21) + b2 * (a11 * c21 - a12 * c12)) /
    (a11 * a22 - a12**2)

    return {
        'a11': a11, 'a12': a12, 'a21': a21, 'a22': a22,
        'b1': b1, 'b2': b2,
        'c11': c11, 'c12': c12, 'c21': c21, 'c22': c22, 'c33': c33
    }

def compute_deflection_and_stresses(p, coeffs, stiffness, Tx, Ty, a, b):
    """Computes the real central deflection (W) and updated membrane forces (Nx, Ny)."""
    c11, c33 = coeffs['c11'], coeffs['c33']
    a11, a12, a22 = coeffs['a11'], coeffs['a12'], coeffs['a22']
    b1, b2 = coeffs['b1'], coeffs['b2']
    B11, B12, B22 = stiffness['B11'], stiffness['B12'], stiffness['B22']

    S = c11 / c33
    R = 1536 * p / (np.pi**2 * c33)
    Q = (S / 3)**3 + (R / 2)**2

    # Cardano solution using np.cbrt to safely handle negative roots
    W = np.cbrt(R / 2 + np.sqrt(Q)) - np.cbrt(np.sqrt(Q) - R / 2)

    # Update in-plane stress resultants [N/m]
    denominator = (a11 * a22 - a12**2)
    term_x = B11 * (np.pi / a) * (b1 * a22 - b2 * a12) + B12 * (np.pi / b) * (b2 * a11 - b1 * a12)
    term_y = B12 * (np.pi / a) * (b1 * a22 - b2 * a12) + B22 * (np.pi / b) * (b2 * a11 - b1 * a12)

    Nx = Tx + 2 * (term_x / denominator) * W**2

```

```

Ny = Ty + 2 * (term_y / denominator) * W**2
return W, Nx, Ny

```

3. VISUALIZATION & INTERFACE

```

def plot_deflection_vs_tension(Tx_values, Ty_values, p, stiffness_dict, a, b,
                               max_deflection_limit_mm=3.0):
    """Creates a 3D surface plot mapping central deflection to biaxial pre-tension."""
    Tx_grid, Ty_grid = np.meshgrid(Tx_values, Ty_values)
    W_grid = np.zeros_like(Tx_grid, dtype=float)
    # Populate the deflection grid
    for i in range(Tx_grid.shape[0]):
        for j in range(Ty_grid.shape[1]):
            coeffs = compute_coefficients(
                a, b,
                stiffness_dict['B11'], stiffness_dict['B22'],
                stiffness_dict['B12'], stiffness_dict['B33'],
                Tx_grid[i,j], Ty_grid[i,j]
            )
            W, _, _ = compute_deflection_and_stresses(
                p, coeffs, stiffness_dict, Tx_grid[i,j], Ty_grid[i,j], a, b
            )
            W_grid[i,j] = W * 1000 # Convert to mm
    # Plot Setup
    fig = plt.figure(figsize=(12, 9))
    ax = fig.add_subplot(111, projection='3d')
    surf = ax.plot_surface(
        Tx_grid, Ty_grid, W_grid,
        cmap='viridis', edgecolor='none', alpha=0.9,
        rstride=1, cstride=1
    )
    # Draw the maximum allowable deflection plane
    if max_deflection_limit_mm is not None:
        z_limit = np.full_like(W_grid, max_deflection_limit_mm)
        ax.plot_surface(Tx_grid, Ty_grid, z_limit, color='red', alpha=0.13, edgecolor='none',
                        , zorder=0)
        ax.plot_wireframe(Tx_grid, Ty_grid, z_limit, color='darkred', alpha=0.35, linewidth=0.8)
        ax.text(
            Tx_values[-1]*0.55, Ty_values[-1]*0.55, max_deflection_limit_mm + 0.4,
            f"Max allowed = {max_deflection_limit_mm:.1f} mm",
            color='darkred', fontsize=10.5, fontweight='bold'
        )
    # Plot contour line on the surface
    cs = ax.contour(
        Tx_grid, Ty_grid, W_grid, levels=[max_deflection_limit_mm],
        colors='darkred', linewidths=3.5, linestyles='-', zorder=5
    )
    ax.clabel(cs, inline=True, fontsize=10, fmt=f'{max_deflection_limit_mm:.1f} mm')

    # Formatting
    ax.set_xlabel('Pre-tension Tx [N/m]', fontsize=11, labelpad=10)
    ax.set_ylabel('Pre-tension Ty [N/m]', fontsize=11, labelpad=10)
    ax.set_zlabel('Central deflection W [mm]', fontsize=11, labelpad=10)
    ax.set_title(f'Membrane Deflection vs Biaxial Pre-Tension\n(Transverse Pressure\np = {p:.3f} Pa)', fontsize=13)

```

```

fig.colorbar(surf, ax=ax, shrink=0.6, aspect=10, label='W [mm]')
ax.set_zlim(0, max(W_grid.max(), max_deflection_limit_mm * 1.15))
ax.view_init(elev=30, azim=-45)

plt.tight_layout()
plt.show()

def run_membrane_analysis(a=1.5, b=1.25, thickness=50e-6, cell_mass_area=1.8,
                        nz=6.0, max_tension=1000, grid_resolution=50, limit_mm=15.0):
    """ Main interface to execute the non-linear membrane deformation analysis.

    Parameters:
    a, b          : Panel dimensions [m]
    thickness      : Membrane thickness [m]
    cell_mass_area : Mass per unit area of the solar cells [kg/m^2]
    nz            : Transverse load factor (g-force) [-]
    max_tension    : Upper limit for Tx and Ty in the parameter sweep [N/m]
    grid_resolution : Number of evaluation points per axis
    limit_mm       : Allowable out-of-plane deflection [mm] """
    # 1. Initialize Materials
    props = get_kapton_properties(thickness)
    stiffness = compute_plate_stiffness_isotropic(props)

    # 2. Compute Launch Loads
    p = compute_pressure(props, nz=nz, cell_mass_area=cell_mass_area)

    print("--- SOLAR ARRAY MEMBRANE ANALYSIS ---")
    print(f"Dimensions:      {a} m x {b} m")
    print(f"Film Thickness:    {thickness*1e6:.1f} μm")
    print(f"Cell Mass/Area:    {cell_mass_area:.2f} kg/m^2")
    print(f"Load Factor:       {nz} g")
    print(f"Calculated Load:   {p:.2f} Pa")
    print("-----")

    # 3. Parameter Sweep Setup
    Tx_vals = np.linspace(0, max_tension, grid_resolution)
    Ty_vals = np.linspace(0, max_tension, grid_resolution)

    # 4. Execute Simulation & Plot
    plot_deflection_vs_tension(Tx_vals, Ty_vals, p, stiffness, a, b,
                              max_deflection_limit_mm=limit_mm)

# EXECUTION BLOCK

if __name__ == "__main__":
    # Execute the analysis with typical prototype parameters
    run_membrane_analysis(
        a=1.5,
        b=1.25,
        thickness=50e-6,      # 50 micron Kapton film
        cell_mass_area=1.8,   # 0 for bare film, ~1.8 for populated cells
        nz=1.0,               # 1g static, increase for launch loads
        max_tension=1000,
        limit_mm=3.0)

```

Manufacturing and Assembly Plan

C.1. System Breakdown

This section provides an overview of the membrane-frame prototype and the supporting assembly rig used throughout the experimental campaign. Figure C.1 presents the system architecture, Figure C.2 illustrates the exploded assembly of the prototype alongside its component legend (Figure C.1), and Figure C.3 depicts the assembly support rig used during manufacturing and tensioning.

System Breakdown Overview	
Membrane-frame prototype (MFP): 53 Parts	Assembly Support Rig (ASR): 28 Parts
• Longeron Hat Stiffeners (MFP-HS): 2× CFRP [0/90, ±45]s • Lateral Cross-Members (MFP-LS): 2× CFRP [0/90, ±45]s (Slotted) • Reinforcement Strips (MFP-LSR): 4× CFRP [0/90, ±45]s • Clamp Strips (MFP-CS): 2× CFRP [0/90, ±45]s • Structural Substrate (MFP-MB): 1× Kapton HN Film (50.8 µm)	• Support Frame: Item Aluminium Profiles (40 × 40 & 40 × 80) • Shaker Test Rig & Base: Aluminium Interface & 18mm MDF Support • Kinematic Guidance: 3D-Printed Rails (PLA + Anti-Slip Shims) • Tensioning System: 3mm Steel Cable, Pulley Bridge & Clamp Plates • Tensile Test Machine: MTS Exceed E43 Tensile Tester

Figure C.1: System breakdown overview of the membrane-frame prototype and assembly support rig.

C.2. Process Flow

The fabrication and integration of the prototype follow a phased approach, ensuring the composite flexures and membrane substrate are assembled with high precision. Figure C.4 outlines Phase A, detailing the parallel tracks for component preprocessing and subsystem fabrication. Subsequently, Figure C.5 defines Phases B and C, covering the rig integration, in-situ pre-tensioning protocol, and final quality verification matrix.

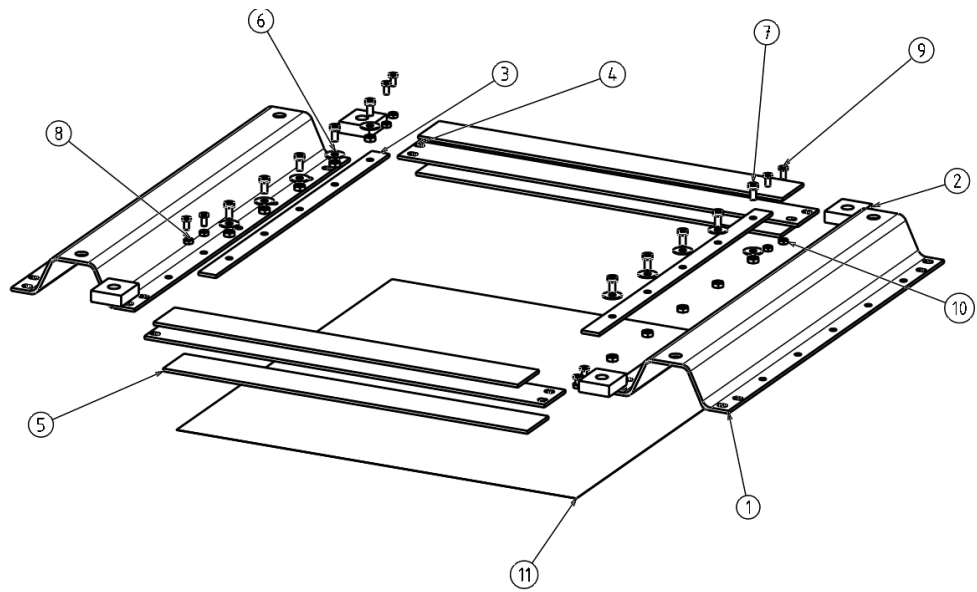


Figure C.2: Exploded view of the membrane-frame prototype.

Table C.1: Prototype assembly components legend.

Item	Component Name	Qty.	Item	Component Name	Qty.
1	Longitudinal Hat Stiffener	2	7	M3 Bolt (6 mm)	10
2	M6 Hold Down Block	4	8	M3 Nut	10
3	Clamp Strip	2	9	M2.5 Bolt (10 mm)	4
4	Lateral Stiffener	2	10	M2.5 Nut	4
5	Lateral Stiffener Reinf.	4	11	Kapton HN Membrane	1
6	M3 Washer	10			

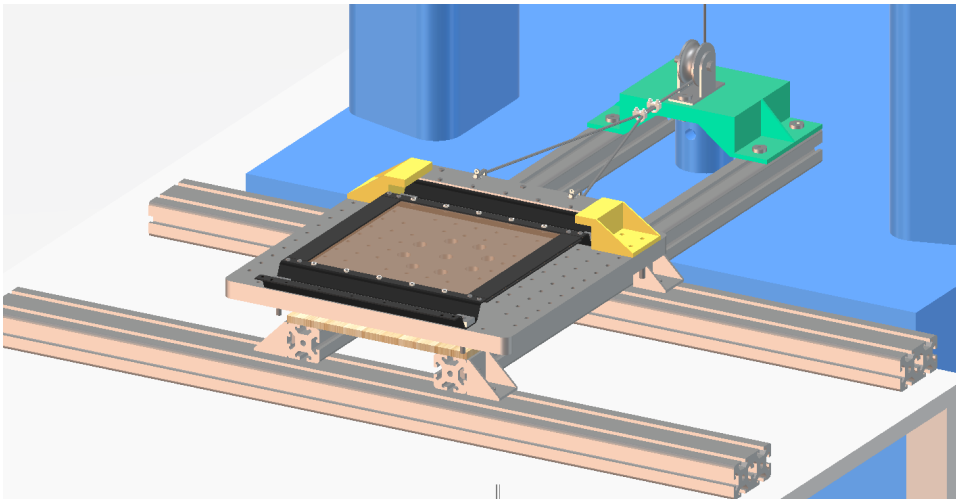


Figure C.3: Kinematic assembly support rig and tensioning set-up.

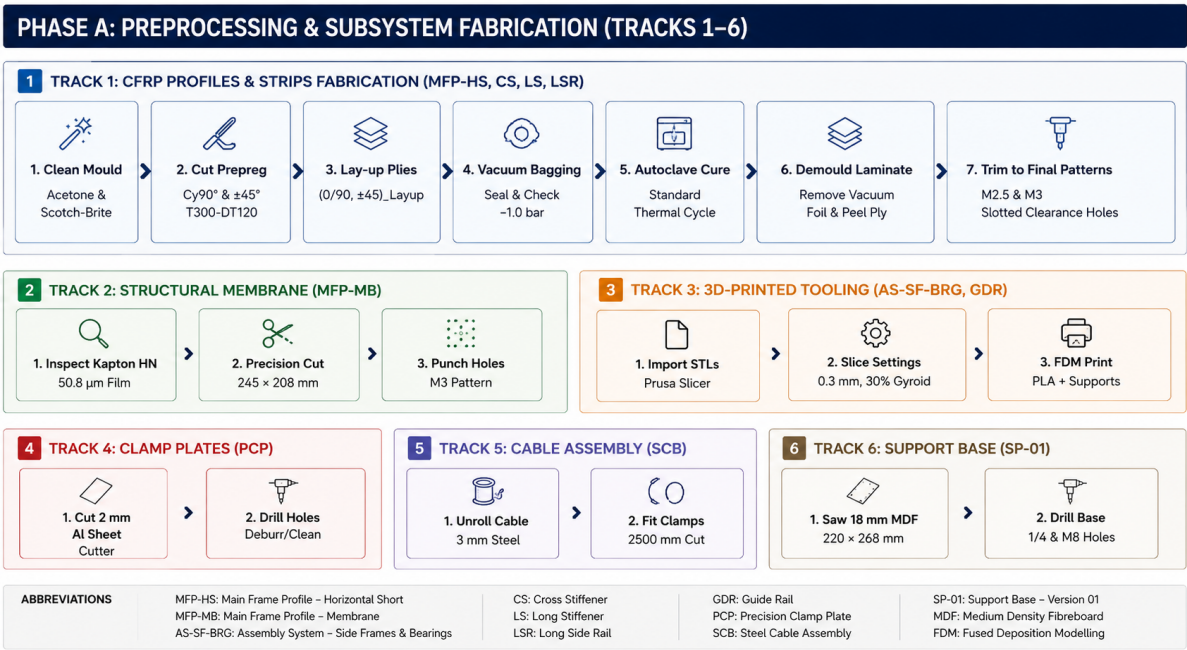


Figure C.4: Phase A: Preprocessing and subsystem fabrication workflow across parallel tracks.



Preliminary Requirements list

Table D.1: Consolidated Preliminary Requirements List (Version 1)

Req. ID	Req. Statement	Value	Type	A	I	D	T	Rationale/Comment
FUNCTIONAL REQUIREMENTS								
FNC-REQ-001	The solar array shall convert incident solar radiation into electrical power	-	Key	✗	✓	✗	✗	Primary system objective that dictates the required surface area for scaling.
FNC-REQ-002	The solar array system shall provide a standard interface for the transfer of electrical power to the spacecraft bus.	-	Key	✗	✓	✗	✗	Defines the mechanical root boundary for the scaling study.
FNC-REQ-003	The solar array shall remain in a locked, stowed configuration during the launch phase.	-	Killer	✗	✓	✗	✗	Essential for survival under launcher vibration loads.
FNC-REQ-004	The array shall deploy into its operational configuration upon receipt of a command from the spacecraft.	-	Driver	✗	✓	✗	✗	Successful deployment essential for reaching primary system objective.
FNC-REQ-005	The deployment mechanism shall lock the panels into their final rigid state to prevent unintended movement during maneuvers.	-	Driver	✗	✗	✓	✗	Drives deployed panel stiffness targets.
PERFORMANCE REQUIREMENTS								
PER-REQ-001	Total Power Output of the solar array at EOL shall be within the range of [250 – 1000 W]	250 – 1000 W	Key	✓	✗	✗	✗	EOL total power will be product of mech. design
PER-REQ-002	The specific power of the solar array shall be larger than 100 W/kg	> 100 W/kg	Key	✓	✗	✗	✗	Derived from ELSA [11]
PER-REQ-003	The first natural frequency of the solar array in stowed configuration shall be larger than 40 Hz	> 40 Hz	Killer	✓	✗	✗	✓	Derived from from ELSA[11]
PER-REQ-004	The deployed first natural frequency of the solar array shall be larger than 0.25 Hz	> 0.25 Hz	Driver	✓	✗	✗	✓	Derived from from ELSA [11]
PER-REQ-005	The deployed solar array surface shall maintain a planarity (flatness) within $\pm 2^\circ$ across the active area in all axes.	$\pm 2^\circ$	Driver	✓	✗	✓	✗	Industry Practice & Essential for optimal power output
PER-REQ-006	The solar array shall demonstrate at least 10 successful deployments	≥ 10 deployments	Driver	✗	✗	✓	✗	Industry practice & rules out one-time deployment systems
PER-REQ-007	The hinge interface shall provide a rotational stiffness of at least [TBD] after deployment	TBD	Driver	✓	✗	✗	✓	TBD after panel concept has been selected
INTERFACE REQUIREMENTS								
INT-REQ-001	The assembly shall interface with the spacecraft via a bolt pattern of [TBD] points.	[TBD]	Driver	✓	✓	✗	✗	TBD, depends on attachment method and panel design.

Continued on next page

Table D.1 – continued from previous page

Req. ID	Req. Statement	Value	Type	A	I	D	T	Rationale/Comment
INT-REQ-002	The solar array shall support a mechanical interface at the root panel for direct side-wall or yoke attachment.	-	Driver	✓	✓	✗	✗	Follows from trade-off analysis & Yoke does add length which lowers deployed natural frequency.
INT-REQ-003	The root interface shall transfer all launch-induced loads to the spacecraft bus structure in deployed configuration.	-	Killer	✓	✗	✗	✓	Interface expected to experience highest bending moments and shear forces during launch.
INT-REQ-004	The stowed array shall be secured by a minimum of [TBD] hold-down and release points.	$\geq [TBD]$	Killer	✓	✓	✗	✓	Industry practice & Follows from final panel concept.
INT-REQ-005	The stowed stack shall interface with snubbers providing a minimum compressive preload.	$\geq [TBD]$ N	Killer	✓	✗	✗	✓	Follows from final panel concept
INT-REQ-006	The deployment hinges shall maintain a kinematic alignment tolerance post-launch.	$< [TBD]$ mm	Driver	✓	✗	✓	✗	Follows from final panel concept.
CONSTRAINTS								
CON-REQ-001	The total mass per panel shall not exceed 2 kg	< 2 kg	Key	✓	✓	✗	✓	Industry practice
CON-REQ-002	Stowed thickness shall be less than 50 mm	< 50 mm	Key	✗	✓	✓	✗	Industry practice
CON-REQ-003	The solar array shall utilize mPower Dragonscales Silicon solar cells	-	Driver	✗	✓	✗	✗	Stakeholder constraint from GTM-AS
CON-REQ-004	Total panel dimensions shall be equal or smaller than $1.5 \text{ m} \times 1.5 \text{ m}$	$\leq 1.5 \text{ m} \times 1.5 \text{ m}$	Driver	✗	✓	✓	✗	Manufacturing constraint from GTM-AS
CON-REQ-005	The solar array shall interface with satellite bus via a hinge mechanism either or to a yoke attachment or directly to the root panel	-	Driver	✗	✓	✓	✗	Constraints deployment method to spacecraft
CON-REQ-006	Shear load per bolt at satellite-array interface shall be < 900 N	< 900 N	Killer	✓	✗	✗	✓	Derived from SSD [66]& far below M4 Bolt shear yield strength of 3000-4000 N
CON-REQ-007	The Margin of safety shall be positive ($\text{MoS} > 0$) for all loads	$\text{MoS} > 0$	Killer	✓	✗	✗	✗	Derived from SSD [66]
CON-REQ-008	Design shall accommodate SSO orbit (600-800 km)	600-800 km	Driver	✓	✗	✗	✗	Chosen based upon identified space trends
CON-REQ-009	The design of the solar array structure shall allow accessibility to all elements for inspection, installation, or removal	-	Driver	✗	✓	✓	✗	Derived from SSD
CON-REQ-009-001	The system shall provide tool clearances for tool installation and actuation for all tool interfaces during maintenance	-	Driver	✗	✓	✓	✗	Derived from NASA-std-3001-V2 V29050 nasa-std-3001-c
CON-REQ-010	The solar cell array shall be assembled and disassembled without damage	-	Driver	✗	✗	✓	✗	Industry practice and SSD [66]
CON-REQ-011	Designed for launch on a Falcon 9 from the Western Range to support insertion into an SSO.	-	Driver	✓	✗	✗	✗	Chosen due to lower acoustic loading & conventional launch site for SSO LEO orbits
CON-REQ-012	The solar array shall be designed for launch on a Falcon 9 which utilizes acoustics blankets inside its fairing.	-	Driver	✓	✗	✗	✗	Lowers acoustic loads, which could be critical for membrane-frame structures
MECHANICAL REQUIREMENTS								
MECH-REQ-001	The solar array in stowed configuration shall withstand the quasi-static load imposed by the SpaceX Falcon 9 launcher without structural degradation	See MECH-REQ-001-001 to 003	Killer	✓	✗	✗	✓	Derived from sine environment of Rideshare [6] by assuming $\zeta = 0.25 \rightarrow Q = 20$ and multiplying with Q
MECH-REQ-001-001	The stowed solar array configuration shall withstand the following quasi-static load in the XPL-direction without structural degradation	± 28 g	Killer	✓	✗	✗	✓	

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Table D.1 – continued from previous page

Req. ID	Req. Statement	Value	Type	A	I	D	T	Rationale/Comment																				
MECH-REQ-001-002	The stowed solar array configuration shall withstand the following quasi-static load in the YPL-direction without structural degradation	± 30 g	Killer	✓	✗	✗	✓																					
MECH-REQ-001-003	The stowed solar array configuration shall withstand the following quasi-static load in the ZPL-direction without structural degradation	± 30 g	Killer	✓	✗	✗	✓																					
MECH-REQ-002	The solar array in stowed configuration shall withstand the specified qualification sine environment imposed by the SpaceX Falcon 9 launcher without structural degradation	See PER-009-001 to 003	Killer	✓	✗	✗	✓	SpaceX Falcon 9 Rideshare User's guide [6]																				
MECH-REQ-002-001	The stowed solar array configuration shall withstand the following sine environment load in the XPL-direction without structural degradation	5-100 Hz 1.4g at 2oct/min	Killer	✓	✗	✗	✓																					
MECH-REQ-002-002	The stowed solar array configuration shall withstand the following sine environment load in the Y-direction without structural degradation	5-100 Hz 1.5g at 2oct/min	Killer	✓	✗	✗	✓																					
MECH-REQ-002-003	The stowed solar array configuration shall withstand the following sine environment load in the Z-direction without structural degradation	5-100 Hz 1.5g at 2oct/min	Killer	✓	✗	✗	✓																					
MECH-REQ-002-004	The qualification safety factor for the sine environment loading in all directions shall be equal or larger than 1.25	≥ 1.25	Key	✓	✗	✗	✓																					
MECH-REQ-003	The solar array assembly in stowed configuration shall maintain structural integrity when subjected to a random vibration environment imposed by the SpaceX Falcon 9 launcher	See ENV-003-001 to 002	Killer	✓	✗	✗	✓	Per SpaceX Falcon 9 Rideshare User's guide [6]																				
MECH-REQ-003-001	The solar array assembly in stowed configuration shall maintain structural integrity when subjected to a random vibration environment with an overall intensity of 5.57 GRMS for at least 60 seconds per axis.	5.57 GRMS, 60 s	Killer	✓	✗	✗	✓																					
MECH-REQ-003-002	The random vibration environment shall follow the Power Spectral Density (PSD) mask defined by the Falcon 9 Maximum Predicted Environment (MPE).	<table><tr><td>f[Hz]</td><td>PSD[g²/Hz]</td></tr><tr><td>20</td><td>0.01</td></tr><tr><td>50</td><td>0.015</td></tr><tr><td>700</td><td>0.015</td></tr><tr><td>800</td><td>0.03</td></tr><tr><td>925</td><td>0.03</td></tr><tr><td>2000</td><td>0.00644</td></tr></table>	f[Hz]	PSD[g ² /Hz]	20	0.01	50	0.015	700	0.015	800	0.03	925	0.03	2000	0.00644	Killer	✓	✗	✗	✓	Per SpaceX Falcon 9 Rideshare payload user guide [6]						
f[Hz]	PSD[g ² /Hz]																											
20	0.01																											
50	0.015																											
700	0.015																											
800	0.03																											
925	0.03																											
2000	0.00644																											
MECH-REQ-003-003	The qualification safety factor for the random environment loading in all directions shall be equal or larger than MPE + 3dB for at least 2 minutes in all axes	≥ MPE+3dB, 2 min	Key	✓	✗	✗	✓	Rideshare qualification margin [6]																				
MECH-REQ-004	The solar array assembly in stowed configuration shall maintain structural integrity when subjected to a vibro-acoustic environment imposed by the SpaceX Falcon 9 launcher	-	Driver	✓	✓	✗	✓	Per SpaceX Falcon 9 Rideshare User's guide [6]																				
MECH-REQ-004-001	The solar array shall maintain structural and functional integrity after exposure to an overall sound pressure level (OASPL) of 139.3 dB.	139.3 dB	Driver	✓	✗	✗	✓	Critical for low-mass membrane structures																				
MECH-REQ-004-002	The acoustic test environment shall follow the Full Octave MPE spectral mask for the Falcon 9	<table><tr><td>f[Hz]</td><td>SPL [dB]</td></tr><tr><td>31.5</td><td>126.1</td></tr><tr><td>63</td><td>128.6</td></tr><tr><td>125</td><td>132.6</td></tr><tr><td>250</td><td>134.5</td></tr><tr><td>500</td><td>132.3</td></tr><tr><td>1000</td><td>127.4</td></tr><tr><td>2000</td><td>123.8</td></tr><tr><td>4000</td><td>120.8</td></tr><tr><td>8000</td><td>117.8</td></tr></table>	f[Hz]	SPL [dB]	31.5	126.1	63	128.6	125	132.6	250	134.5	500	132.3	1000	127.4	2000	123.8	4000	120.8	8000	117.8	Driver	✓	✗	✗	✓	
f[Hz]	SPL [dB]																											
31.5	126.1																											
63	128.6																											
125	132.6																											
250	134.5																											
500	132.3																											
1000	127.4																											
2000	123.8																											
4000	120.8																											
8000	117.8																											

Continued on next page

Table D.1 – continued from previous page

Req. ID	Req. Statement	Value	Type	A	I	D	T	Rationale/Comment								
MECH-REQ-004-003	The qualification safety factor for the vibro-acoustic environment loading in all directions shall be equal or larger than the MPE + 3dB for at least 2 minutes	$\geq$ MPE+3dB, 2 min	Key	✓	✗	✗	✓									
MECH-REQ-005	The solar array assembly in stowed configuration shall maintain structural integrity when subjected to the shock imposed by the SpaceX Falcon 9 launcher	-	Killer	✓	✗	✗	✓	Per SpaceX Falcon 9 Rideshare User's guide [6]								
MECH-REQ-005-001	The solar array assembly in stowed configuration shall maintain structural integrity when being subjected by the specified LV-induced MPE	<table><tr><th>f[Hz]</th><th>SRS [g]</th></tr><tr><td>100</td><td>30</td></tr><tr><td>1950</td><td>2850</td></tr><tr><td>10000</td><td>2850</td></tr></table>	f[Hz]	SRS [g]	100	30	1950	2850	10000	2850	Killer	✓	✗	✗	✓	Shock Response Spectrum (SRS) at payload interface
f[Hz]	SRS [g]															
100	30															
1950	2850															
10000	2850															
MECH-REQ-005-002	The solar array assembly in stowed configuration shall maintain structural integrity when being subjected by the specified separation system-induced shock	<table><tr><th>f[Hz]</th><th>SRS [g]</th></tr><tr><td>100</td><td>30</td></tr><tr><td>1000</td><td>1000</td></tr><tr><td>10000</td><td>1000</td></tr></table>	f[Hz]	SRS [g]	100	30	1000	1000	10000	1000	Killer	✓	✗	✗	✓	Average separation plane shock response
f[Hz]	SRS [g]															
100	30															
1000	1000															
10000	1000															
ENVIRONMENTAL REQUIREMENTS																
ENV-REQ-001	The solar array shall be able to operate within the temperature range of -80°C to 100°C	-80°C to 100°C	Key	✓	✗	✗	✓	Typical LEO temperature fluctuation								
ENV-REQ-002	The solar array shall be able to withstand > 28,000 thermal cycles for a polar SSO orbit at 600 km	> 28,000 TC	Killer	✓	✗	✗	✓	Fatigue driver for material interfaces and cell interconnects over mission life.								
ENV-REQ-003	The solar array shall adhere to the minimum outgassing requirements: TML < 1.00%, RML < 1.00%, CVCM < 0.10%	-	Killer	✓	✓	✗	✗	Industry practice; mandatory for vacuum compatibility and cell health.								
ENV-REQ-004	The solar panel shall be able to be stored in stowed configuration for 30 days and deployed without degradation	Vacuum [$< 10^{-8}$ Torr], Temp [-50°C, 50°C]	Key	✓	✗	✗	✓	Industry practice; ensures no cold-welding or adhesive set in vacuum.								
ENV-REQ-005	The solar panel shall be able to be stored in stowed configuration for 2 years in a clean room without degradation	Ambient P, Temp [+10°C, +30°C], Hum < 55%	Driver	✗	✓	✗	✓	Industry practice; addresses shelf-life and pre-launch storage.								

Membrane-frame solar array concept dissection

Appendix E provides a comprehensive technical expansion of the existing membrane-frame solar array architectures introduced in Chapter 3. To ensure a rigorous comparative analysis, the mechanical and structural features of each concept are systematically categorized using the hierarchical component-level framework established in Table 3.1. The review begins with an analysis of Redwire’s ELSA concept (Section E.1) and MBB’s historical ULP architecture (Section E.2). Subsequently, the hybrid semi-rigid configuration of the Airbus Eurostar Neo is evaluated (Section E.3), concluding with a structural dissection of JAXA’s ultra-lightweight solar panel prototype (Section E.4).

E.1. Redwire's ELSA concept dissection

Redwire’s ELSA (Extendible low-profile solar array) is a patent-pending membrane-frame solar array, presented in Figure E.2 [11]. While not all design-features can be deduced due to the limited information about this design, in Section E.1 a best-effort approximation is made on the systematic dissection. Figure E.2 presents the dissection of ELSA’s individual components. Table E.1 presents a short description of the identified components.

The photovoltaic blanket used in the ELSA design is mentioned to be the similar to the Roll-Out Solar Array (ROSA)[11]. The ROSA utilized the integrated modular photovoltaic blanket assembly, developed by Deployable Space Systems Inc, which has been acquired by Redwire in 2021 [67]. This PV blanket’s layout in exploded view is presented in Figure 2.2a. The philosophy behind its design is to create repetitive modules (consisting of the solar cells, interconnects carried by a thermal insulating layer), which are attached to a mechanical open weave mesh using low profile fasteners [7]. The open weave fiber mesh is made of a low CTE fibrous material and acts as a dimensionally stable planar surface by being the only layer onto which the tension is applied. Like this it is possible to mechanically decouple the fragile solar cells and interconnects from differential tensile forces which occur during the in-orbit temperature gradient.

In April 2026, Redwire also published a promo video of the ELSA, where some more features become apparent [68]. Similar to Figure 2.2, in Figure E.1a it is visible that polyimide foam strips are utilized at the back of ELSA to effectively prevent cell-to-cell contact. What is also notable is that to both the longerons and the lateral profiles, the membrane is attached using rivets (as seen in Figure E.1b). Furthermore in Figure E.1c, it is also seen that the rivets are slightly offset of the adjacent panels, which is done to prevent the rivets touching each other when folded, even further reducing the stacking height.

Information on the floating center flex hinges utilized by ELSA is not easily obtainable. From the illustrations provided by Redwire, this hinge appears to be of a metallic material. Most likely ‘floating center’ refers to the fact that the hinge utilized does not have 1 single rigid axis, but rather a shifting virtual axis that shifts during the deployment. The ‘flex’ part refers most likely to flexure, which is a



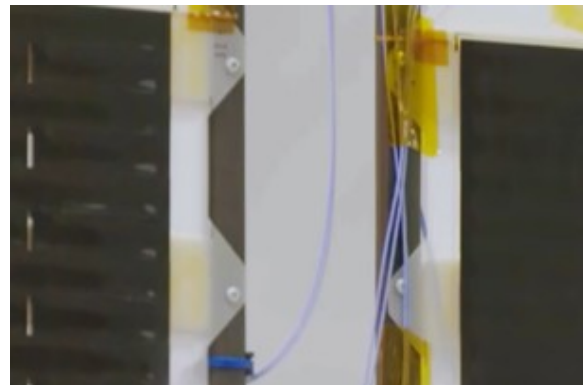
(a) Unfolding ELSA array showcasing polyimide foam strips [68]



(b) Front view of the ELSA during deployment, visualizing the membranes riveted connection onto the CFRP hat stiffener and hollow rectangular stiffeners.



(c) Longeron profile showing mechanical rivet fasteners.



(d) Membrane interface with lateral profiles, showing slight off-center placement to lower the nesting height.

Figure E.1: Visual analysis of Redwire ELSA design features extracted from deployment testing footage, illustrating key structural joints, stowage protections, and integration methods [68].

part that is a compliant mechanism that allows movement into a specific design direction and is rigid in others. Similar to tape springs, which are typically made of CFRP, this hinge might consist of a metallic part flattened and under high strain energy, which when deployment occurs snaps into a curve shape providing rigidity required in the panel's deployed state. To synchronize the deployment, a pantograph is added which also acts as a damping against the deployment torque of the hinges.

To obtain the required stiffness in deployed state, Wide longitudinal stiffening profiles are used by ELSA, as seen in Figure E.2. The longerons consist of Omega/hat profiles which are made of CFRP. This profile is used to ensure the height of the whole folded stack is minimized by nesting the frames into each other. Also this profile allows for adding attachment points for the blanket at its flanges. This design indicates that (if there is any pretension), the lateral pretension of the blanket is higher than the longitudinal pretension.

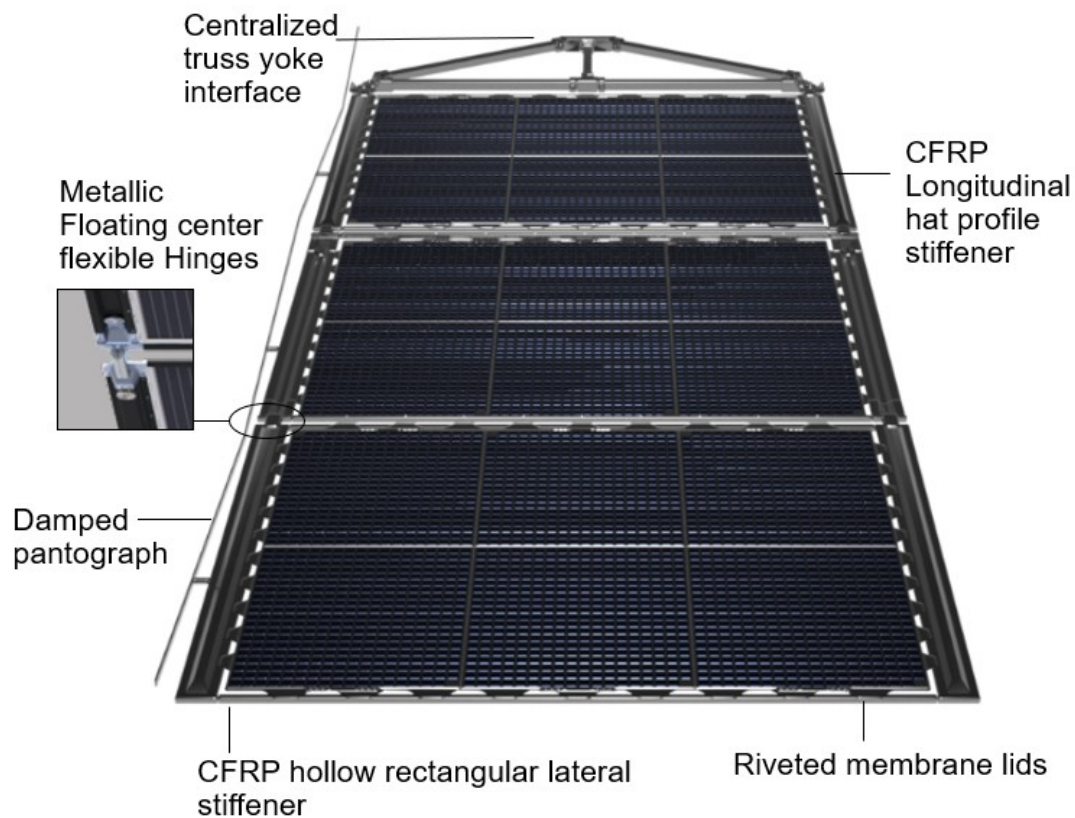


Figure E.2: Redwire's Extensible Low-profile solar array (ELSA), including a description of its most notable design features [11].

Table E.1: ELSA Systematic Dissection: Component Design Approach

Tier	Component	ELSA Technical Approach
1. Primary Structure	Frame Elements	Low-profile, wide CFRP longerons providing longitudinal stiffening (approx. 1.0m) flip each panel to optimize stowage efficiency; Thin rectangular cross-members for lateral support (approx. 1.5m).
	Frame Junction	Metallic (Aluminum) hinges attached to low-profile beams, providing the primary coupling between longitudinal and lateral elements.
	Yoke	Centralized truss-yoke interface transitioning structural loads from the array frame to the spacecraft bus.
2. External Interfaces	HDRM (Hold-downs)	4 Hold-down points located close to the hinges
	Hinge / Deployment	Floating Center Flex Hinges and a dampening pantographic mechanism for synchronized deployment.
	Bus Attachment	Centralized interface acting as both the structural anchor; precise sun-pointing mechanism is unknown.
3. Membrane-frame Interface	Membrane Attachments	Discrete interface utilizing rivets
	Structural Membrane	Low-CTE pre-tensioned grid membrane (most likely Vectran or Kevlar) acting as a separate backplane from the cell-supporting substrate.
4. Power system	SPM	Photovoltaic modules integrated via low-profile fasteners to the underlying structural membrane.

E.2. ULP concept dissection

The Ultra-Lightweight Panel (ULP) is a solar array developed for the EURECA satellite by MBB GmbH between 1973 and 1978 [69]. The solar array, presented in Figure E.4, utilizes a rigid carbon frame (consisting of rectangular cross-sectional members) paired with a pre-tensioned $0.25\mu\text{m}$ Kapton membrane reinforced with a fiberglass substrate [10], [69]. The 25 mm height of the CFRP structural members ensures high dynamic stiffness and thermal stability. Additionally, the design features a retractable closed-cable loop (CCL) mechanism connected to a root motor, satisfying EURECA’s mission requirement for in-orbit retrieval by the Space Transportation System (STS).

Table E.2 presents the systematic dissection of the ULP solar array’s components, with the corresponding deployment kinematics illustrated in Figure E.5a. A patent filed by MBB in 1983 is assumed to be representative of the photovoltaic blanket technology used in this architecture [70].

Figure E.3 illustrates the dimensions of the ULP panels [10]. Notably, each panel features a width of 3.22 m and a length of 1.145 m, yielding a total area of 3.687 m^2 (an aspect ratio of 2.81). This is significantly larger than the $1.5 \times 1.5\text{ m}$ panel area constraint established for this project. While this wide design minimizes the total array length required for a given surface area, it introduces the penalty of requiring multiple internal stiffeners per panel and demands a wide stowage footprint on the spacecraft bus.

The solar cell technology implemented by MBB closely mirrors their 1983 patent titled “Thin Film Solar Array” [70]. This patent outlines a mass-manufacturable module design where silicon solar cells and interconnects are bonded directly to a transparent fluoropolymer superstrate, eliminating the mass and fragility of individual cell cover glasses. The heat-resistant substrate allows direct application of the front cell electrodes [70]. The fluoropolymer is reinforced with threads or fabrics integrated into the spaces between cells to preserve optical transmissivity, and the entire assembly is backed by a $25\mu\text{m}$ glass-fiber reinforced Kapton membrane with uniaxial tensioning applied across its full width [10].

The rectangular panel frame is subdivided by seven carbon fiber composite channel rods spaced every 36

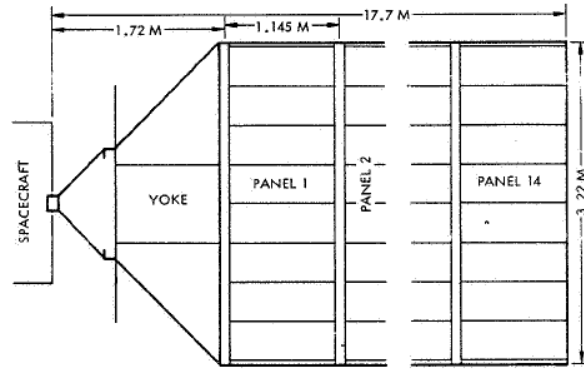


Figure E.3: ULP wing dimensions [10]

cm, which clamp the substrate via grooved friction joints [10]. Due to the near-zero CTE of the CFRP frame and these repetitive local clamping points, thermal cycling effects are effectively mitigated; any minor tension loss in the Kapton-glass fiber layer during hot-case conditions results in negligible overall blanket sagging. Although these additional frame elements add mass, they provide crucial localized bending stiffness to the slender panel sections.

To ensure launch survivability, the ULP frame and yoke are preloaded directly against the spacecraft sidewall, eliminating the need for a separate stowage container and targeting a stowed natural frequency roughly 25% above the spacecraft bus (20 to 30 Hz) [10]. The hold-down architecture utilizes a two-phase deployment sequence (Figure E.5b): a primary hold-down on the first inward channel rods allows the outboard panel to deploy 90 degrees, followed by a secondary release for the remaining panels. This complex multi-stage deployment was specifically driven by the EURECA requirement for full array retractability.

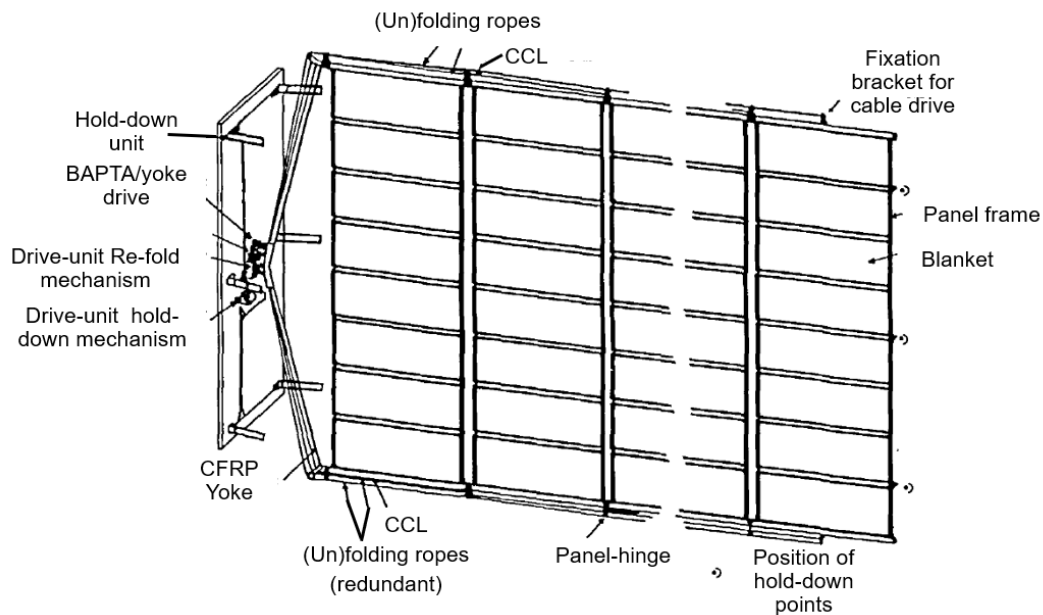
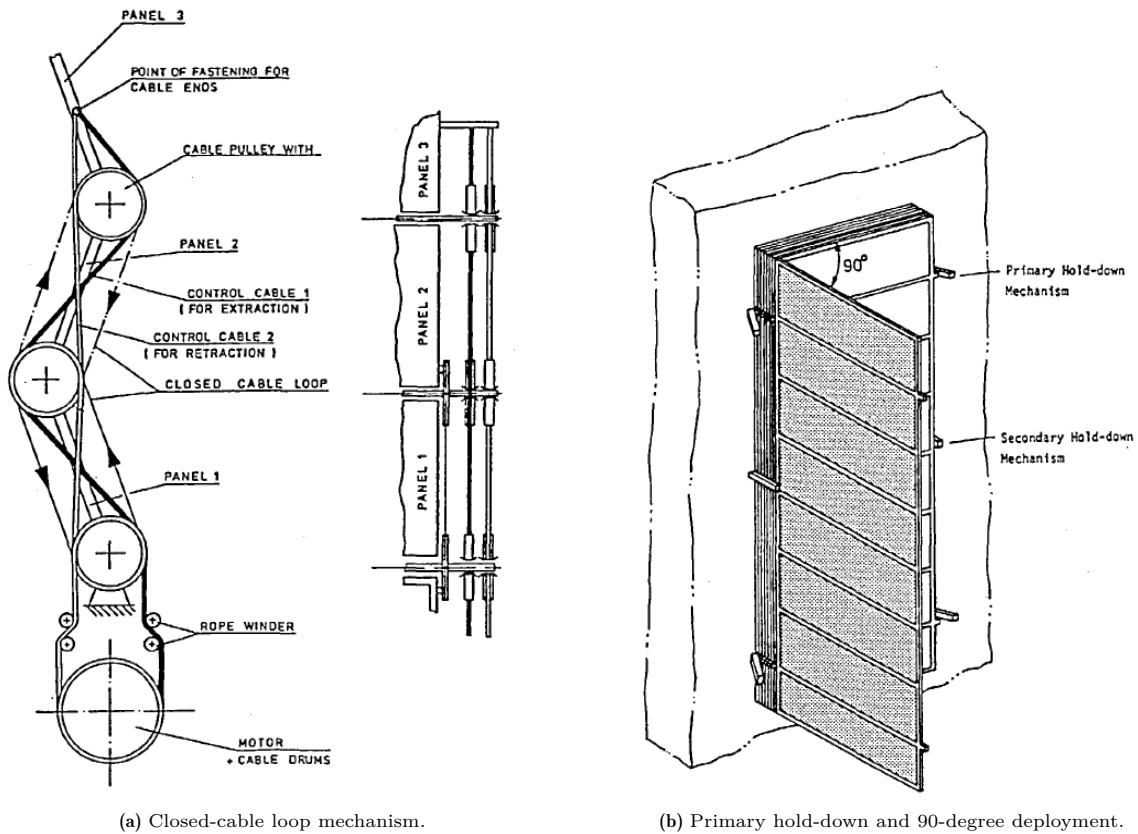


Figure E.4: Configuration of the EURECA solar array wing (ULP) [69]

Table E.2: ULP MBB 1984 Systematic Dissection: Component Design Approach

Tier	Component	ULP Technical Approach
1. Primary Structure	Frame Elements	Narrow, thin CFRP frame utilizing multiple longitudinal inter-panel channel-profile rods (approx. 1.145m) and long uniform cross-section lateral beams (approx. 3.22m), all with a height of 25mm.
	Frame Junction	Partially integrated hinge units housed within the main beam of the panel frame.
	Yoke	Triangular CFRP truss structure positioned between the inboard panel and the sun-tracking motor (BAPTA).
2. External Interfaces	Hinge / Deployment	Root-mounted motor connected to a closed cable loop; bidirectional motor rotation drives synchronized array deployment or retraction.
	Bus Attachment	Interface to the spacecraft via a dedicated sun-tracking motor (BAPTA).
	HDRM (Hold-Down)	Specialized two-phase mechanism allowing for 90-degree deployment of the top panel while secondary mechanisms retain the stack.
3. Membrane Interface	Membrane Attachments	Mechanical substrate clamped onto inter-panel channel cross-section rods using presumably a friction 'clipping' joint.
	Structural Membrane	0.25 μm Kapton membrane reinforced with fiberglass, uniaxially tensioned along the panel's width.
4. Power system	SPM	Integrated solar modules, utilizing a transparent fluoropolymer superstrate onto which silicon solar cells and interconnects are attached.

**Figure E.5:** Deployment and retraction kinematics of the ULP architecture [69].

E.3. Airbus Eurostar Neo concept dissection

The Eurostar Neo is an Airbus initiative that integrates supplementary semi-rigid solar panels into the existing Eurostar 3000 rigid solar array configuration. Without increasing the required stowage volume, these semi-rigid panels allow the existing array to deliver up to 50% more power within a similar mass envelope [71]. Because the solar array is a hybrid system rather than a fully semi-rigid architecture, a direct one-to-one comparison with other concepts is complex; nonetheless, the membrane-frame structure is evaluated in this section to extract relevant design principles. Table E.3 presents a component-level dissection, detailing the deduced design approach utilized by Airbus. A schematic illustration of the semi-rigid panel and a thermal cycling (TC) test coupon are presented in Figure E.6.

To endure the dynamic launch environment, the semi-rigid panels are folded directly between the rigid panels. By applying a compressive preload across the entire hybrid stack, combined with the stiffness of the internal corrugation lattices, the dynamic launch load requirements are effectively met. This indicates that the semi-rigid panels rely heavily on the boundary conditions of the rigid stack for launch survivability, rather than possessing sufficient standalone dynamic stiffness. The term “corrugation lattice” likely refers to the Omega-type cross-section profiles visible in Figure E.6. To fit within the rigid panel stack, it is estimated that the maximum height of these profiles cannot exceed 1.5 cm. During deployment, however, the double C-spring hinges only need to support the mass of a single semi-rigid panel. This localized load path allows the CFRP tubes utilized in the Neo to be significantly less stiff than those required for a fully semi-rigid solar array.

The Coefficient of Thermal Expansion (CTE) of the aramid-reinforced fabric is approximately $5 \times 10^{-6} \text{ m/m}^\circ\text{C}$, which is relatively close to the CTE of the CFRP tubes (assumed to be $\approx 0 \text{ m/m}^\circ\text{C}$). In a hot-case scenario, this slight mismatch will result in a loss of pretension and limited sagging of the fabric. Notably, Airbus bonds the corrugation lattices directly to the blanket without utilizing discrete hooks or spring attachments. This bonded interface presents a significant manufacturing challenge: if the blanket requires baseline pretension, it must be actively tensioned while the frame is permanently bonded to it. This structural constraint makes the design highly complex for scaled, high-volume manufacturing.

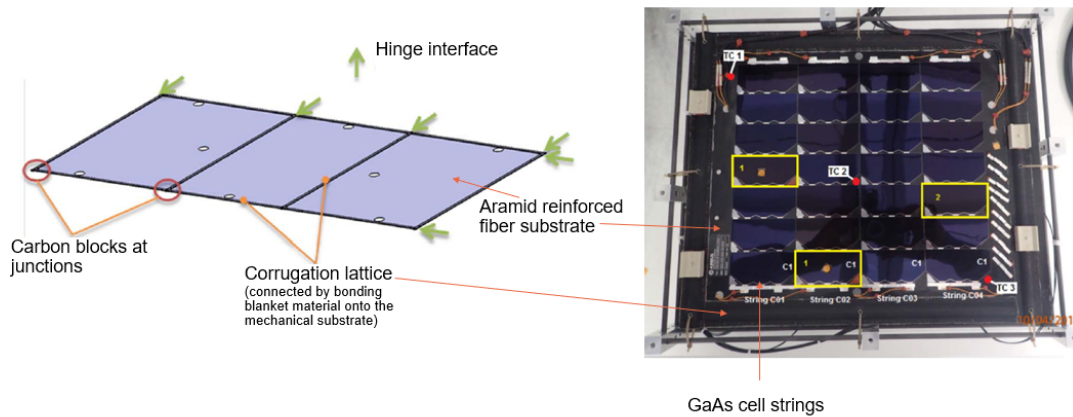


Figure E.6: Airbus Neo membrane-frame solar array design features (left) alongside a thermal cycling test coupon (right) [71].

Table E.3: Airbus Eurostar Neo Systematic Dissection: Component Design Approach

Tier	Component	Eurostar Neo Technical Approach
1. Primary Structure	Frame Elements	Longitudinal stiffening via a single uniform CFRP tube (corrugation lattice) of 3.24 m; lateral stiffening using four uniform CFRP tubes (2.28 m length) spaced every 0.8 m.
	Frame Junction	Carbon block lattices at junctions acting as adapter pieces to connect the frame beams.
	Yoke	'A'-configuration yoke designed to transition structural loads from the array frame to the spacecraft bus.
2. External Interfaces	HDRM (Hold-Downs)	High-preload clamping system positioned between rigid panels to secure the hybrid stack and mitigate launch vibrations.
	Hinge / Deployment	Deployment of semi-rigid panels utilizes a double C-spring hinge mechanism actuated after the rigid panels are deployed.
	Bus Attachment	Centralized mechanical interface via the yoke.
3. Membrane Interface	Membrane Attachments	Continuous bonded interface; CFRP tubes are likely attached by bonding strips of aramid fiber fabric directly onto the main substrate.
	Structural Membrane	Aramid fiber-reinforced sheets impregnated with carbon black resin for enhanced protection against UV degradation.
4. Power system	SPM	GaAs photovoltaic strings integrated onto the aramid substrate; specific cell-to-membrane fastening details remain proprietary.

E.4. JAXA's ultra-lightweight solar panel dissection

The ultra-lightweight solar panel concept is a JAXA initiative aimed at utilizing their proprietary space solar sheets [72]. The philosophy behind this design is to create a modular solar array competitive with rigid array specific power for the LEO environment by severely minimizing the required structural mass. The primary structure consists of highly slender square 1×1 cm CFRP tubes, measuring 55 cm in length and 70 cm in width, which are connected by aluminum hinge units at the corners. The panel utilizes solar sheets comprising InGaP/GaAs dual-junction solar cells bonded between two 75 μ m transparent films (likely FEP or transparent polyimide) [73]. A breadboard model of a 5-panel configuration has been fabricated and successfully maintained structural integrity in both acoustic and sinusoidal vibration environments.

The deployed 5-panel breadboard model is presented in Figure E.7. To establish out-of-plane geometric stiffness, the solar sheet is held taut under a defined mechanical pretension; however, the exact interface method securing the membrane to the frame is not fully documented. Furthermore, in Figure E.7, a yellowish layer is visible between the solar sheets, suggesting the use of Kapton tape in the breadboard model. Because adhesive tape is highly susceptible to long-term viscoelastic creep, it is highly unlikely that this acts as the primary load-bearing interface in a final flight design. Instead, this layer most likely serves as a low-cost secondary securing mechanism for the test setup, attaching the cells to a transparent backplane to accurately approximate the dynamic mass distribution of the panel. While this specific interface deduction remains an engineering estimation, it highlights the severe integration challenges of fastening discrete cells to a highly tensioned membrane.

Table E.4: JAXA Ultra-Lightweight Solar Panel Systematic Dissection: Component Design Approach

Tier	Component	JAXA Technical Approach
1. Primary Structure	Frame Elements	Grid of square 1×1 cm CFRP tubes; longitudinal members are 70 cm and lateral members are 55 cm.
	Frame Junction	Integrated aluminum hinges located at the corners to connect the frame beams.
	Yoke	A 'V'-shaped yoke configuration designed to transition structural loads from the panel to the spacecraft bus.
2. External Interfaces	HDRM (Hold-Downs)	Corner-based hold-down mechanisms (likely mechanical clamps or latches at the four hinge points) to secure the folded panels during launch vibration.
	Hinge / Deployment	Utilization of aluminum spring-loaded hinges for the deployment sequence; includes 4 specific hold-down points located at the corner hinges.
	Bus Attachment	Secured via the V-yoke.
3. Membrane Interface	Membrane Attachments	Exact attachment method unknown, though Kapton tape is utilized in breadboard models to adhere sheets to the backplane; structural membrane is tensioned.
	Structural Membrane	75 μ m thick transparent film (likely FEP, ETFE, or polyimide) encapsulating the cells.
4. Power system	SPM	Double-junction (DJ) GaAs cells with integrated interconnects and bypass diodes encapsulated within the film layers.

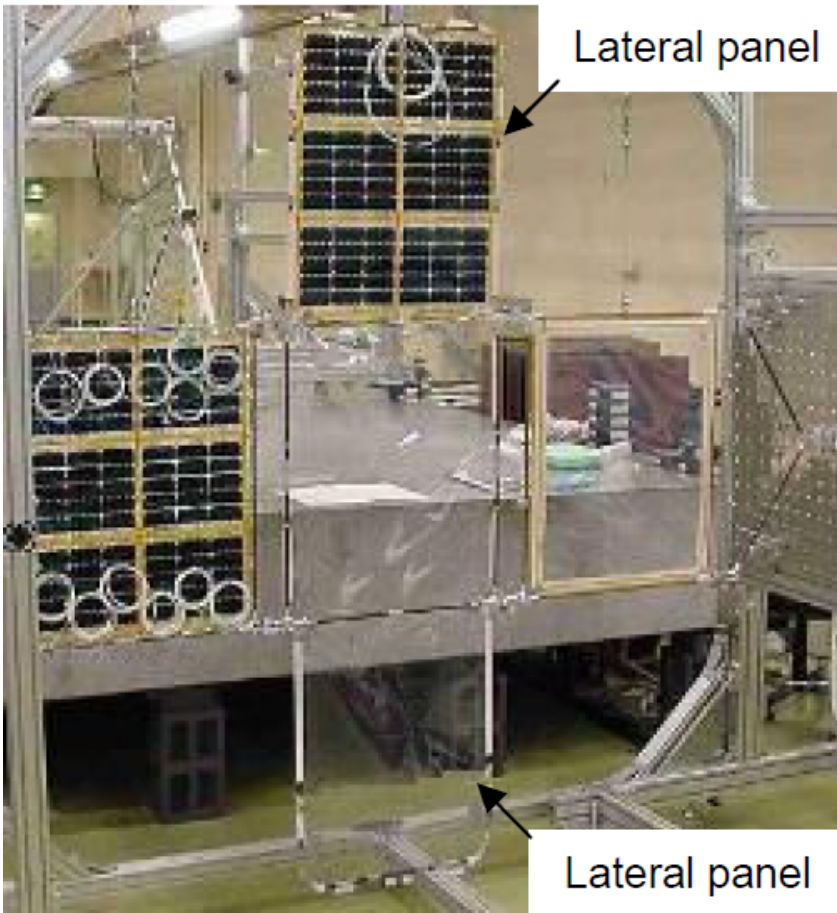


Figure E.7: JAXA ultra-lightweight solar panel breadboard model in deployed configuration [72].

Trade-off Scoring Rationales

F.1. GaAs vs. DragonScales areal density approximation

This section details the areal density approximation used to evaluate payload mass differences. Initially, the trade-off assumed that because Gallium Arsenide (GaAs) cells possess a much higher volumetric mass density, their resulting areal density would naturally exceed that of Silicon. To validate this assumption, a comparative mass assessment was conducted.

First, two distinct mPower silicon modules were measured for mass and dimensions at GTM-AS. A previously tested module yielded an areal density of 0.0904 g/cm^2 . A pristine, smaller module yielded 0.0932 g/cm^2 . The average areal density is therefore 0.0918 g/cm^2 . To account for manufacturing tolerances and measurement inaccuracies, a conservative margin of $\pm 0.002 \text{ g/cm}^2$ is applied, granting a final approximation of 0.092 g/cm^2 for the silicon modules.

For the GaAs baseline, the Azurspace 3G30C-Advanced Triple Junction (TJ) solar cell is evaluated [3]. To ensure a direct comparison, a module comprising 4×3 cells is modeled (as presented in Figure F.1). Because the mPower DragonScales contain integrated interconnects, the mass of the busbars and interconnects must be explicitly added to the GaAs module. Assuming a total of 6 busbars ($N_{\text{busbar}} = N_{\text{col}} \cdot 2$) and 45 interconnects ($N_{\text{int,con}} = N_{\text{cells}} \cdot 3 + N_{\text{col}} \cdot 3$), all manufactured from Kovar ($\rho_{\text{kovar}} = 8.36 \text{ g/cm}^3$), the component mass equals 1.72 g. Adding the mass of the solar cells ($A_{\text{cell}} = 30.18 \text{ cm}^2$, $m''_{\text{cell}} = 0.086 \text{ g/cm}^2$, $t = 0.015 \text{ cm}$) and the CMX coverglass ($\rho_{\text{glass}} = 2.6 \text{ g/cm}^3$, $t_{\text{glass}} = 100 \mu\text{m}$), the total module mass is 42.46 g. This results in an areal density of 0.0956 g/cm^2 for the GaAs strings.

Based on these calculations, the difference in areal density between the two technologies is negligible, as both cell types fall within the same $0.09 - 0.10 \text{ g/cm}^2$ range. Consequently, the initial mass penalty assumption is disproven, and solar cell compatibility has been removed as a weighted criterion from the main architectural trade-off.

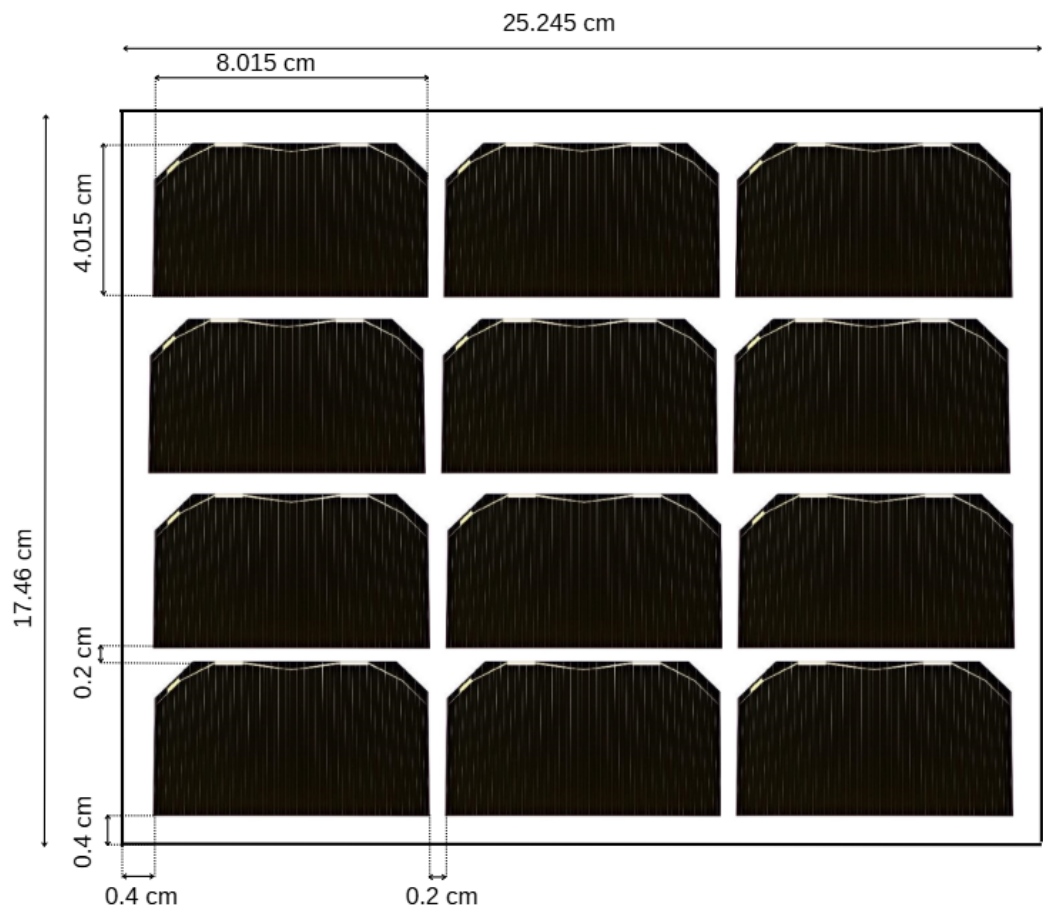


Figure F.1: Gallium Arsenide (GaAs) hypothetical model utilized for areal density calculations.

F.2. Trade-off table with comments

This section presents the fully annotated trade-off matrix (Table F.1), providing the specific engineering justifications for each assigned score. The evaluation matrix is the result of three iterative review cycles, which systematically refined the criteria, weighting distributions, and straw-man baselines.

To mitigate internal bias, this final matrix incorporates independent scoring assessments conducted by engineering colleagues K. Honselaar and H. Cruijssen, who blindly completed the standardized trade-off template prior to the final score consolidation. The scoring methodology strictly follows the quantitative and qualitative framework defined in Section 3.2.3.

Table F.1: Detailed concept comparison with full scoring justifications.

Criterion	ELSA	ULP	Airbus Neo	JAXA
Design flexibility / adaptability				
Mission profile compatibility	LEO compatible	LEO compatible, flight heritage	If designed for GEO, it can withstand LEO environment; flight heritage	No heritage; mission profile unspecified
Accommodation penalty	Dimensions remain in the same range when transferred to template	Requires 50% width downscale; internal sliders must be removed	Requires proportional scaling down in width and length; no structural issues arise	Upscaling required; negatively impacts frame stiffness under pretension
Scalability	High; stacking height delta is minimal due to the nested frame design	Stacking delta is larger, increasing the offset bending moment on the bus	Stacking delta is larger, increasing the offset bending moment on the bus	Stacking delta is larger, increasing the offset bending moment on the bus
Stowage efficiency				
Panel height	Nested design allows for highly compact stowage	High U-profile (2.5 cm); no nesting possible	Low profile, but architecture does not accommodate nesting	Low profile (3 cm), but architecture does not accommodate nesting
Active area packing density	Significant loss due to wide hat-profile longerons utilized	Minor loss due to outer longeron attachments; potential internal stiffener obstruction	Corrugation lattices bonded on the cell side significantly reduces flat effective area	Optimal effective area
Hold-down compatibility	Highly restricted; clamping only possible near the hinge assemblies	Moderate; hold-downs restricted to outer borders, not mid-span	Possible across full outer edge, though corrugation lattices are not completely flat	Highly restricted; clamping only possible near the hinge assemblies
Panel stiffness	Moderate to High; wide profiles provide stiffness, but unconstrained center is vulnerable	Very high; numerous internal cross-members prevent global deformation	Very high; numerous internal cross-members prevent global deformation	Low; minimal mass approach yields questionable dynamic stiffness
Manufacturability / Modularity				
Directional dependency	High; wide profiles dictate longitudinal orientation	Moderate to Low; uniform profiles across outer frame, slight bias for original attachment	Moderate to Low; uniform profiles across outer frame, slight bias for original attachment	Fully modular; no inherent directional dependency
Assembly / disassembly ease	Minimal interface count (mix of bolted and snap-fit); flip-assembly required	High interface count due to internal cross-members (bolted, some snap-fit)	Requires specialized tooling for consistent bond quality; moderate interface count	Minimal interface count; highly repeatable
Final score	4.20	3.80	4.03	3.72

G

Preliminary Experimental Modal Analysis Tests

The preliminary tests documented in this appendix were conducted to validate the feasibility of the experimental setup. Specifically, the feasibility regarding the acquisition of modal data from transparent thin-films and the dynamic behavior of the aluminum test rig was investigated. Furthermore, these initial tests served to mitigate operational risks associated with the experimental modal analysis equipment prior to the final prototype testing.

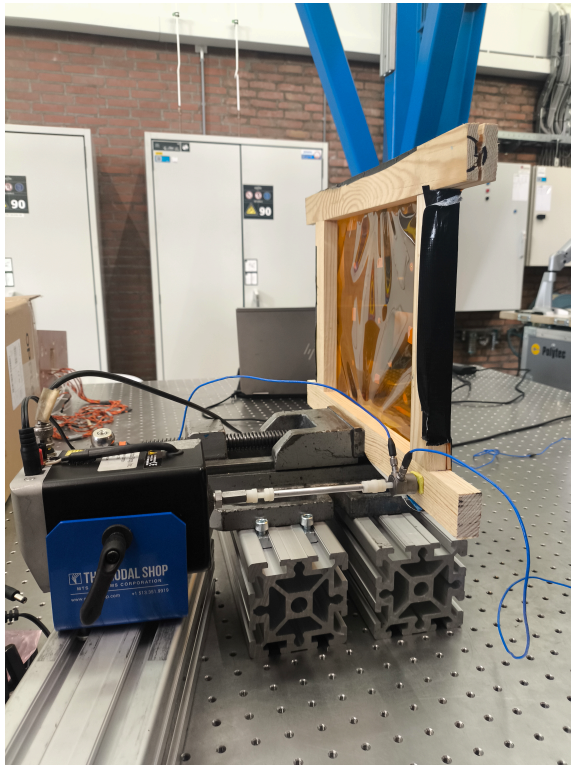
Section G.1 details the initial validation of the Polytec PSV-500 data acquisition system using a simplified proof-of-concept prototype. Subsequently, Section G.2 outlines the frequency response characterization of the V550 electrodynamic shaker and the aluminum test rig, including the contingency evaluation of the Brüel & Kjær vibration control software.

G.1. PSV-500 acquisition tests

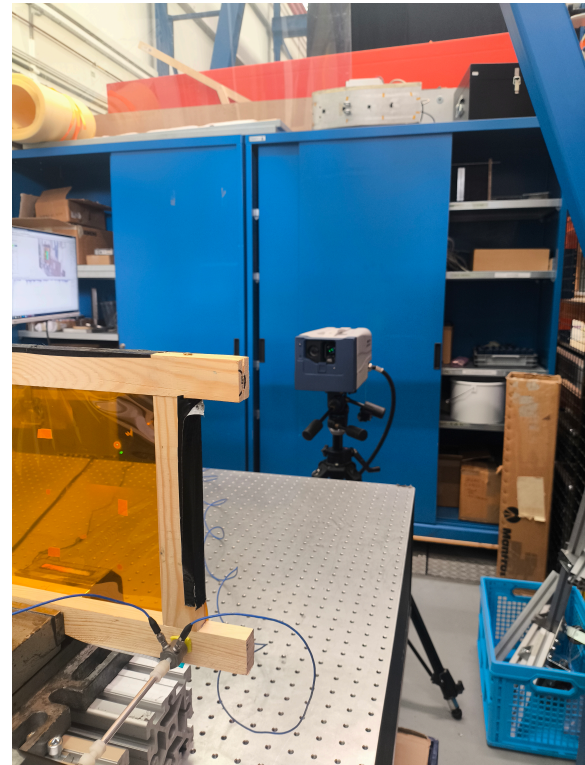
To validate the measurement procedures and data acquisition settings associated with the Polytec PSV-500, a preliminary proof-of-concept test was conducted at the TU Delft Aerospace Engineering Aircraft Hall. For this test, a simplified prototype was constructed by attaching a Kapton membrane to a rigid wooden frame, as depicted in Figure G.1. The membrane was manually affixed using a combination of double-sided tacky tape and duct tape. Due to this manual integration procedure, the membrane exhibited a slightly non-uniform pretension field.

The test setup consisted of clamping the wooden frame centrally to a rigid support structure, while a SmartShaker was attached to the bottom corner of the frame to provide dynamic excitation. The PSV-500 scanning laser head was positioned perpendicular to the membrane at a standoff distance of approximately 1.2 meters.

During the initial testing phase, multiple excitation signal types were evaluated. It was empirically established that a pseudorandom force input yielded the highest coherence (γ^2) across the desired frequency band. To achieve this, it was essential to input a high enough signal to rise above the electrical noise floor, but low enough to ensure the structure remained perfectly linear. The moment the generator input is too high, the maximum range of the impedance head is exceeded (set at 500 mm/s), which corrupts the calculated FRFs. Furthermore, if the generator input is too high, the thin-film polyimide can exhibit non-linear behavior; excessive transverse displacements cyclically stretch the film, temporarily increasing its tension and dynamically shifting the natural frequency during the vibration cycle. Finally, it was critical to ensure fully rigid boundary conditions, such that no micro-slips occurred during the excitation, which would otherwise introduce mechanical noise into the signal.



(a) Front view of the hand-tensioned Kapton membrane on the wooden frame.



(b) Side view detailing the SmartShaker attachment and boundary constraints.

Figure G.1: Simplified proof-of-concept test setup for validating the PSV-500 scanning laser vibrometer.

Utilizing the pseudorandom input, a spatial grid scan was executed on five points on the Kapton membrane. The resulting spatial average Frequency Response Function (H_1) for the proof-of-concept prototype is presented in Figure G.2. By evaluating the established criteria for global eigenmodes (distinct magnitude peaks, corresponding phase shifts, and high coherence), the first three major natural frequencies were successfully identified at 26.6 Hz, 54.0 Hz, and 72.8 Hz.

Despite the imperfections inherent to the hand-tensioned wooden frame, the results clearly demonstrated that the PSV-500 is highly capable of extracting clean, highly coherent modal data from the translucent Kapton film. This successfully de-risked the data acquisition architecture in preparation for the final tensioned CFRP prototype testing.

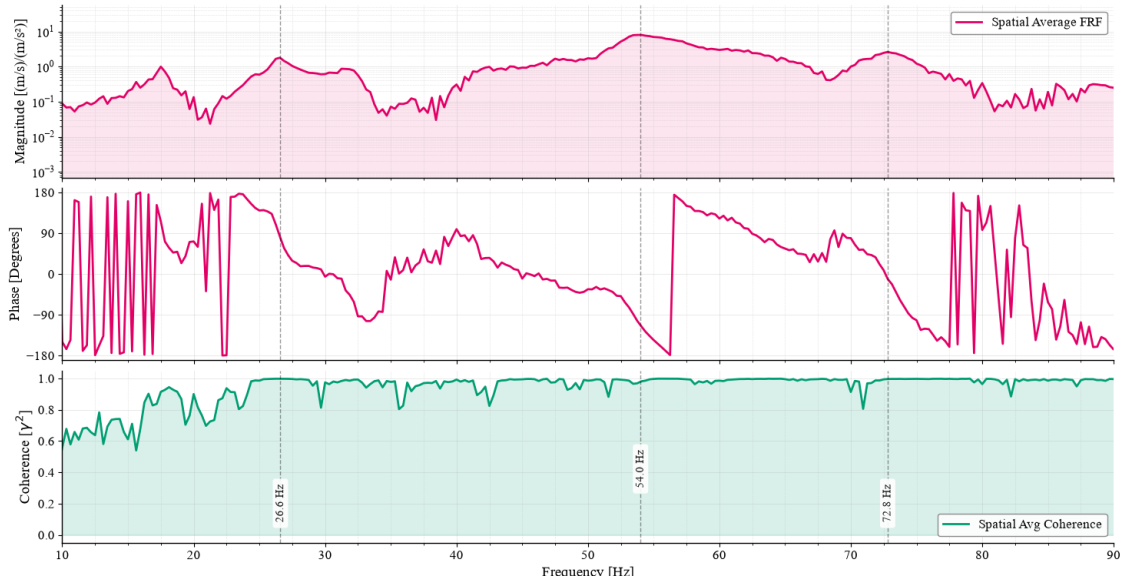


Figure G.2: Frequency response function (H1) of the proof-of-concept prototype.

G.2. V550 shaker acquisition

Similar to the preliminary test of the PSV-500, a series of initial tests were performed to mitigate operational risks prior to the final experimental campaign. Because the compatibility of the V550 shaker system with the PSV-500 DAQ front-end was not initially certain, the corresponding Brüel & Kjær vibration control software was also evaluated as a contingency. During these preliminary tests, the primary objective was to determine the fundamental eigenmode of the aluminum test rig, thereby establishing an upper-frequency limit for identifying purely prototype-dominated modes.

The experimental setup used to perform this calibration is presented in Figure G.3. The aluminum test rig was mounted to the shaker fixture using nine M6 bolts. To measure the dynamic response of the system, two accelerometers were attached to the test rig: one positioned in the center of the plate to measure the direct input translation, and a second placed at the outer corner of the test rig. The accelerometers were affixed by applying a base layer of polyimide tape (briefly abraded with Scotch-Brite) followed by cyanoacrylate adhesive.

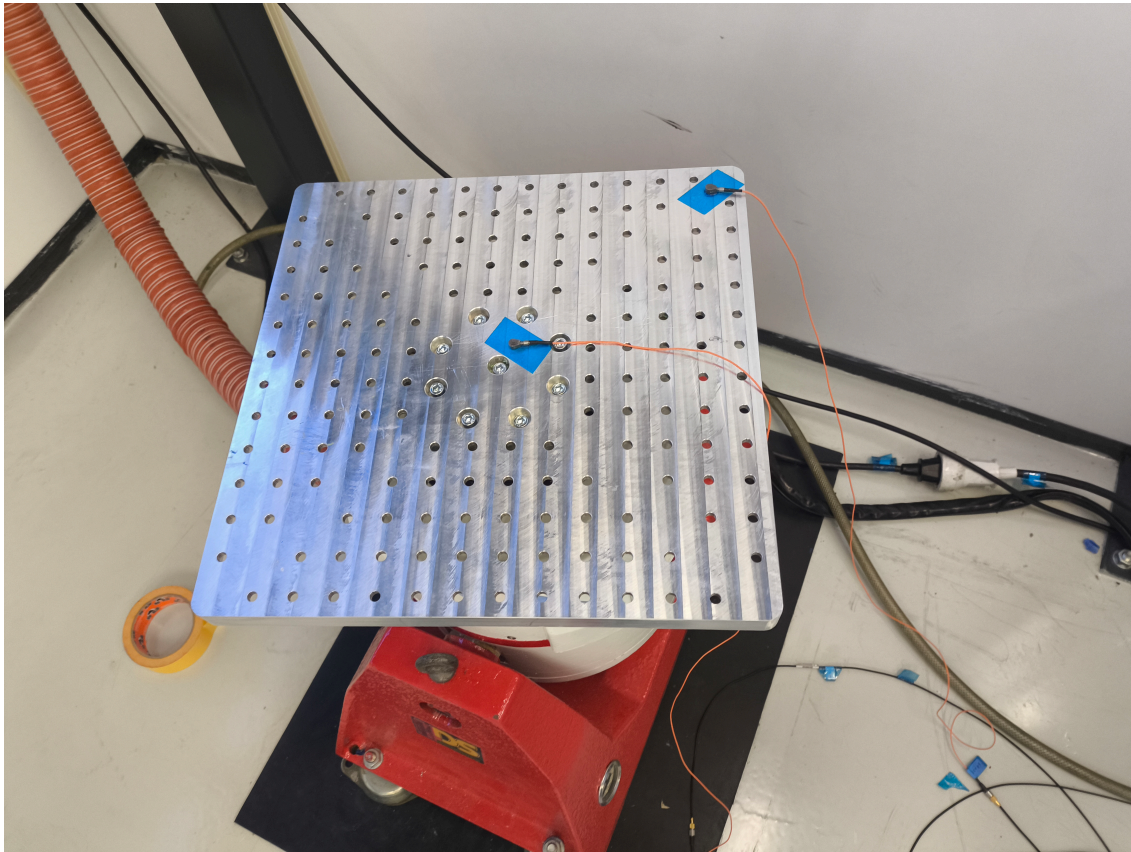


Figure G.3: Aluminum test rig mounted onto the V550 Shaker, instrumented with two reference accelerometers.

Following a sine sweep with an acceleration input of 0.5g, the measured responses of the accelerometers were plotted, as presented in Figure G.4. The fundamental eigenfrequency of the test rig was identified at 460 Hz, exhibiting an extreme acceleration amplification of 19g. This frequency represents a critical upper boundary for the purely prototype-dominated eigenfrequencies; above this limit, complex coupled responses between the prototype and the test facility are highly likely to occur. Furthermore, due to the geometric overhang of the test rig relative to the shaker armature, the response measured at the corner was 38 times higher than the baseline input signal. To prevent structural damage to the delicate prototype during final testing, this demonstrated the necessity of utilizing highly conservative initial input loads.

Finally, a distinct drop in measured acceleration was detected near 50 Hz. This artificial drop occurs due to electrical line noise interference. Electromagnetic noise from the standard 50 Hz European power grid bleeds into the sensor cables, tricking the closed-loop controller into registering a violent vibration at that specific frequency. To compensate for this artificial spike, the controller aggressively cuts the drive power at exactly 50 Hz, causing the actual physical acceleration to drop to near-zero.

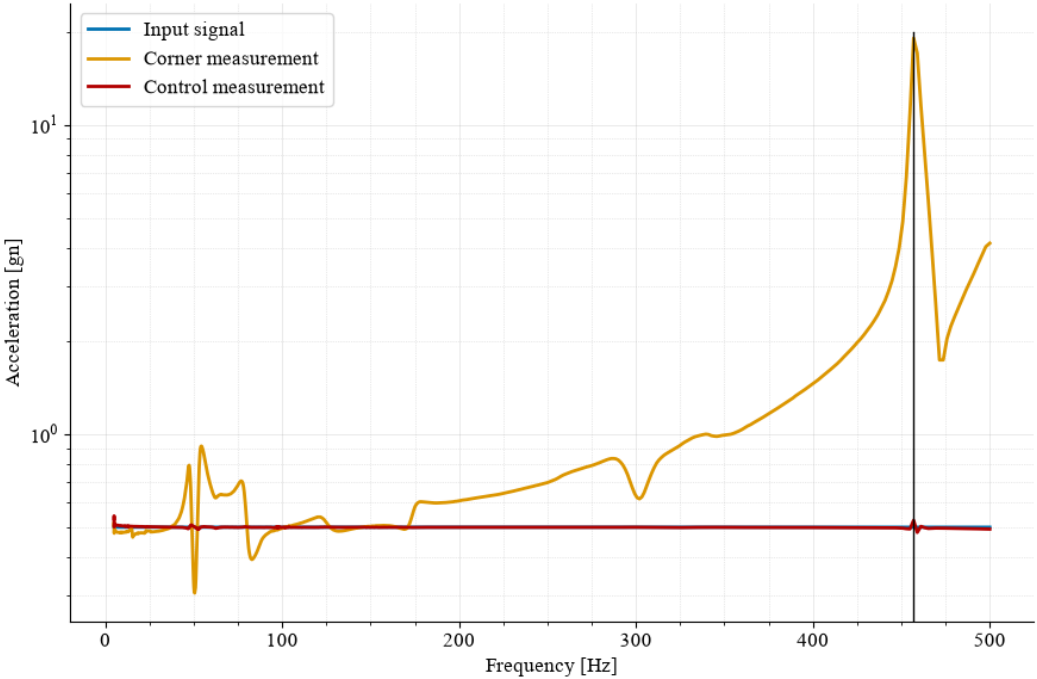


Figure G.4: Measured response of the aluminum test rig during a 0.5g sine sweep (5-500 Hz) utilizing a sweep rate of 2 dB/min.



Spatial Correlation of the Populated Prototype

Following the successful frequency correlation of the populated membrane-frame prototype, a spatial correlation was conducted. The compared EMA and FEM modes are presented in Figure H.2. The objective was to evaluate the Modal Assurance Criterion (MAC) between the physical operational deflection shapes captured by the 1D LDV and the idealized numerical eigenvectors predicted by the 40% effective-mass finite element model.

As established in Section 7.2.4, the numerical model was heavily constrained by a 1000-node computational limit. This restriction necessitated the use of a continuous, smeared distributed mass to represent the discrete solar cell assemblies. While this simplified approach proved highly accurate for predicting global natural frequencies (eigenvalues) within a 5% error margin, the preliminary MAC analysis demonstrates its inherent limitations in predicting localized spatial deformation (eigenvectors).

The resulting MAC matrix for the populated configuration is presented in Figure H.1.

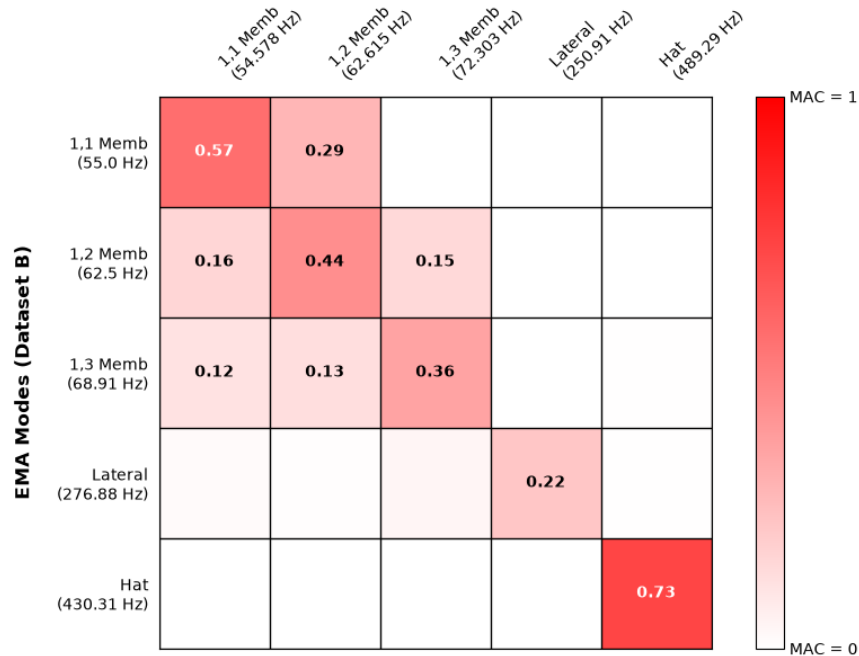


Figure H.1: Preliminary Modal Assurance Criterion (MAC) matrix correlating the 40% smeared-mass numerical predictions with the physical LDV spatial measurements of the populated solar array.

The highest spatial correlation is observed in the fundamental (1,1) membrane mode, yielding a MAC

value of 0.57. While this confirms that the macroscopic topology of the mode (a central out-of-plane displacement) aligns between the datasets, the correlation is heavily penalized by the geometric mismatch introduced by the smeared-mass assumption. This discrepancy is driven by two fundamental physical phenomena:

- **Discrete stiffness vs. continuous compliance:** In the physical prototype, the bonded silicon solar cells and RTV-S 691 adhesive act as highly rigid, localized structural sheets. These discrete modules resist bending, effectively acting as flat plateaus during dynamic excitation and forcing the majority of the bending strain into the narrow strips of bare Kapton between the cell strings.
- **Vector curvature penalty:** Conversely, the numerical model distributes the mass and stiffness evenly across the entire membrane plane (similar to the unpopulated membrane however with a larger deflection). Consequently, the simulated membrane deforms as a perfectly smooth, continuous dome. Because the MAC formula calculates the normalized squared inner product of the mode shape vectors, it is highly sensitive to localized gradient and curvature mismatches. The mathematical comparison between the physical "flat-topped pyramid" deformation and the numerical "smooth dome" deformation naturally results in a degraded inner product.

Despite the mathematical penalties associated with the smeared-mass assumption, the populated MAC matrix also reveals critical physical improvements in structural symmetry. Most notably, the (1,3) membrane mode (seen in Figure H.2c) demonstrates a clear spatial correlation ($\text{MAC} = 0.44$) that was entirely absent in the unpopulated baseline. As observed during the experimental modal analysis, bonding the rigid solar cells to the membrane effectively bridged the non-uniform tension zones, forcing the formation of a distinct, symmetric central anti-node that successfully matched the idealized FEM prediction.

Conversely, the correlation for the lateral frame mode experienced a noticeable reduction ($\text{MAC} = 0.36$) (seen in Figure H.2d). This degradation is primarily driven by dynamic asymmetry within the simplified numerical model rather than the physical membrane itself. The FEM predicts a highly localized response where only one lateral CFRP profile oscillates significantly, whereas the physical prototype exhibited a more uniform, coupled movement across the structural frame.

Ultimately, this preliminary correlation explicitly defines the validity envelope of the numerical modeling strategy utilized in this study. The 40% effective-mass approximation serves as an efficient and accurate approach for predicting global frequency shifts during early-phase structural sizing. However, to achieve high-fidelity spatial correlation and localized stress validation, future numerical models must bypass node limitations to explicitly discretize the geometry, mass, and localized stiffness of the individual solar cell assemblies.

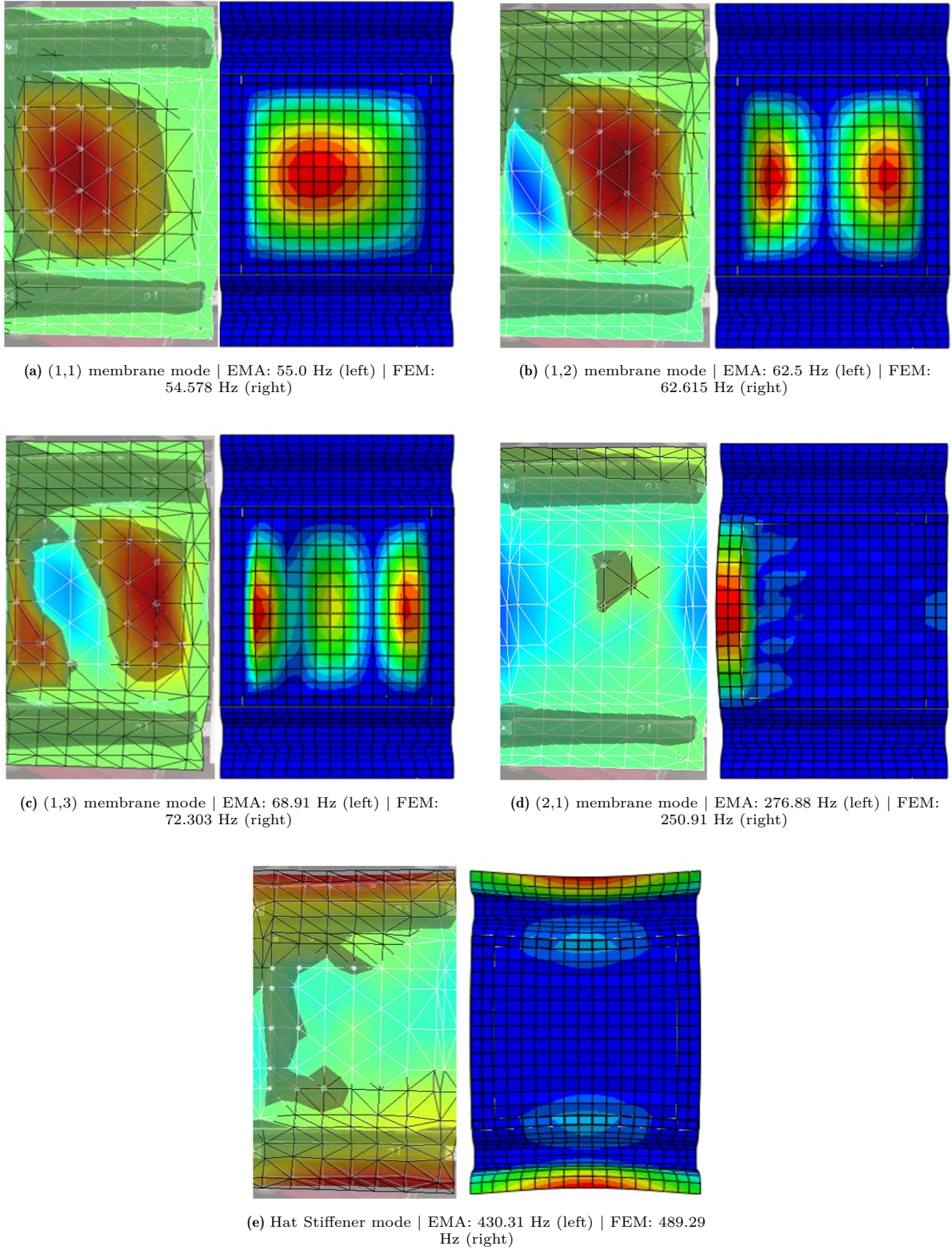


Figure H.2: Comparison of experimental (EMA) and numerical (FEM) natural frequencies and mode shapes of the populated prototype. Each subfigure presents the spatial LDV measurements on the left against the corresponding numerical FEM predictions on the right.

Dynamic Evaluation of the Preliminary Finite Element Model

Table I.1: Comparison of experimental natural frequencies (EMA) against the preliminary finite element model (first iteration with technique used in Section 5.3.5. While the fundamental mode exhibited artificial agreement, the severe deviations in higher-order membrane and frame modes necessitated the transition to the as-built numerical modeling technique.

Mode Shape	EMA Frequency (Hz)	Initial FEM (Hz)	Absolute Error (%)
(1,1) Membrane	119.69	120.12	0.36
(1,2) Membrane	156.25	131.19	16.04
(1,3) Membrane	180.94	143.29	20.81
(2,1) Membrane	299.06	234.48	21.59
(2,2) Membrane	338.13	254.72	24.67
Lateral Frame	283.59	124.78	56.00
Longeron Frame	430.94	—	—