# THE BOOM DESIGN OF THE DE-ORBIT SAIL SATELLITE

**Martin Hillebrandt [(1)], Sebastian Meyer [(1)], Martin Zander [(1)], Marco Straubel [(1)], Christian Hühne [(1)]**

*[(1)] DLR – German Aerospace Center, Institute of Composite Structures and Adaptive Systems, Lilienthalplatz 7, 38108 Braunschweig, Germany, E-Mail: Martin.Hillebrandt@dlr.de*

## ABSTRACT

**DE-ORBIT SAIL is a cubesat based drag sail for the de-orbiting of satellites in a low earth orbit. It is scheduled for launch in late 2014 and will deploy a 25m² sail supported by deployable carbon fiber booms designed and manufactured by DLR. This boom possesses a closed cross-section formed by two omega-shaped half-shells. Due to this cross-sectional design the boom features a high torsional stiffness. Thereby a high bending strength is achieved compared to other boom concepts for similar applications as the boom is less sensitive to flexural torsional buckling. The boom concept selection is based on a detailed analysis of three types of deployable booms which differ in their cross-sectional design. From this analysis the double-omega boom was determined as most suited for DE-ORBIT SAIL. For the manufacturing of the booms a novel method is used where the booms are manufactured in an integral way in one piece.**

## 1. INTRODUCTION

De-orbiting of satellites which are inoperative or have exceeded their operational lifetime is of high importance to limit the growth of space debris which endangers other spaceflight missions.

One strategy for space debris mitigation in the low earth orbit is to use a drag augmentation device which is added to a satellite. For initiation of the de-orbit manoeuvre a large surface which multiplies the actual satellite surface is deployed. This increase in surface area also increases the drag forces resulting from the residual earth atmosphere and causes accelerated decay in altitude until re-entry. One design solution is to deploy a drag sail similar to a solar sail with a large membrane supported by deployable booms. This approach is chosen for the DE-ORBIT SAIL project (see Figure 1-1) which is developed within EU's FP-7 program by a consortium of universities, research facilities and industry. It is based on a 3U cubesat (1U = 100mm x 100mm x 100mm) and will demonstrate the deployment of a 25m² sail on-orbit in late 2014.

Currently there are several similar 3U-cubesat based sail projects under development or – in case of NANOSAIL-D2 [1] – have already flown. The Planetary Society has recently finished development of LIGHTSAIL-1 [2] which has a sail-size of 32m² and is regarding the boom design based on NANOSAIL-D. The University of Surrey – which also holds the project management of DE-ORBIT SAIL – is currently developing CUBESAIL [3], GOSSAMER DEORBITER [4] and INFLATESAIL [5]. All will deploy sails with a size of 25m². While the sails of CUBESAIL and GOSSAMER DEORBITER are supported by bi-stable composite booms, INFLATESAIL uses inflatable booms. NASA's cubesat based solar sail project NANOSAIL-D possesses booms made of metal which are self-deploying. NANOSAIL-D2 was launched in late 2010 and successfully deployed a 10m² sail.

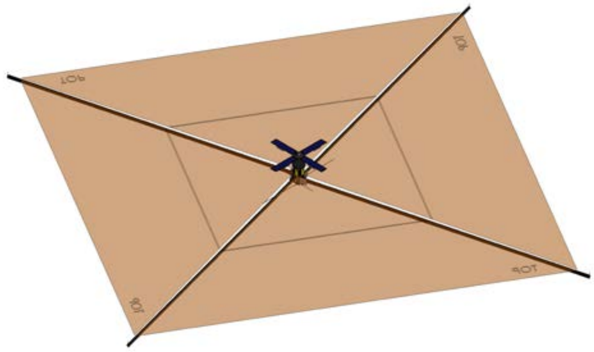

**Figure 1-1: DE-ORBIT SAIL in deployed configuration.**

In the following the design of the DE-ORBIT SAIL booms is presented with a main focus on the concept selection. Firstly, applicable boom concepts are reviewed in section 2. Section 3 provides all necessary data for the concept selection in section 4. Section 5 describes the final design of the DE-ORBIT SAIL booms and is followed by a description of the novel integral manufacturing method in section 6.

## 2. REVIEW OF BOOM CONCEPTS

Boom concepts for small satellite applications need to be highly volume efficient especially in case of a cubesat. The boom technologies selected for LIGHTSAIl-1, NANOSAIL-D, CUBESAIL and GOSSAMER DEORBITER are all shape memory structures based on elastically stored strain energy. These booms are rigid structures which are deformed elastically for stowage and are self-deploying. An alternative are inflatable structures as used for INFLATESAIL. They are based on tubes of thin membranes, can be stowed very compactly and are deployed using an inflation gas. Inflatable booms need to maintain the internal gas pressure to be able to carry loads or require an additional rigidization mechanism.

For DE-ORBIT SAIL a shape memory boom design is selected. The deployment behaviour of such booms is well predictable and controllable. Their rigid structure does not require any additional rigidization mechanisms

and enables verification testing of the flight article prior to integration. As high stiffness materials are applicable and a higher shell thickness is achievable compared to the thin, highly deformed membrane of an inflatable boom, one can also assume better mechanical performance.
In the following applicable boom concepts are presented. They all are stowed by flattening the cross-section first and reeling the boom afterwards which is a very volume efficient type of folding.

**Storable Tubular Extendible Member (STEM) [6]**
The STEM-boom (see Figure 2-1) has been developed by Northrop-Grumman and has extensive flight heritage. It is an open profile boom of circular cross-section and is formed from a single composite or metal sheet. There are several types of STEM booms such as the Bi-STEM consisting of two STEMs where one encloses the other.

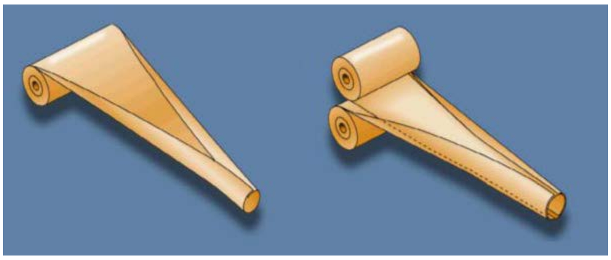

**Figure 2-1: STEM boom (left) and Bi-STEM (right) [6].**

**Triangular Rollable and Collapsible Mast (TRAC) [7]**
The TRAC (see Figure 2-2) boom has a v-shaped cross-section and was developed by the US Air Force Research Laboratory for small satellite applications. It was designed with a main focus on maximizing the moment of inertia achievable from a given stowage volume and has flight heritage on NANOSAIL-D2.

**Figure 2-2: Folding of a TRAC boom sample [7].**

**Collapsible Tube Mast (CTM) and CFRP-Boom [8]**
The CTM boom and DLR’s CFRP-Boom (see Figure 2-3) are tube-like booms composed of two symmetrical omega-shaped half-shells. Its closed cross-section allows for a high boom length without being endangered to flexural torsional buckling. It is deployable by inflation using an additional internal hose or by a separate motorized deployment mechanism.

**Figure 2-3: DLR's CFRP-boom in semi-deployed configuration.**

**C-Shaped, bi-stable boom [4]**
The University of Surrey has developed a deployable composite boom with a cross-section of a semi-circle (see Figure 2-4) which is stable in deployed and stowed configuration. Its bi-stable properties are achieved by utilizing the lateral contraction of a laminate with a high percentage of fibers oriented around ±45deg. The advantage of the design lies in its ability to be stowed with very little constraint forces and to deploy in a controlled way without the need for a complex deployment mechanism.

**Figure 2-4: Four C-shaped bi-stable booms attached to the central hub of the GOSSAMER DEORBITER [4].**

## 3. BOOM CONCEPT ANALYSIS

In this section basic information for the selection of the DE-ORBIT SAIL boom concept in section 4 are provided. Of particular interest is the impact of the stowage volume and thereby boom size on the mechanical properties. Therefore boom versions adapted to different stowage volumes are analysed.

### 3.1. DESIGN OBJECTIVES AND CONSTRAINTS

The objectives for the boom design are given by the purpose of DE-ORBIT SAIL as a de-orbiting device. Added to a satellite, it shall not cause significant increase in mass or volume as both will lead to increased launch costs. Of high importance is also the scalability of the concept to ensure applicability for a wide range of satellites.
Therefore the design objectives for the DE-ORBIT SAIL booms are as follows:


**Figure 3-1: Deployment sequence of the DE-ORBIT SAIL engineering model: (a) fully stowed, (b) prior to in-plane opening of the sails, (c) sail opening and (d) fully deployed.**

(1) Ensure scalability,
(2) Minimize mass,
(3) Minimize volume.

The design constraints are resulting from the mission scenario, the cubesat based architecture, mechanical loading, environmental conditions, agreements among consortium partners and others. These constraints are as follows:

(1) Volume: $V \leq 1U$
(2) Mass: $m \leq 500g$
(3) Mechanical loads: $F_{Sail} \leq 1N$
(4) Thermal loads
(5) Interface requirements
(6) Material availability
(7) Manufacturing limitations
(8) Costs
(9) Research focus of DLR

The values given for the design constraint (1), (2) and (3) result from agreements among consortium partners.

## 3.2. LOAD CASES AND SAFETY FACTORS

The design driving load environment is mechanical loading. There are three phases where mechanical loads are acting on the booms: constraint forces during stowage, static and dynamic loads during deployment and static and dynamic loads during the operational phase. For the structural design of the booms the deployment and the operational phases are important.

Decisive for the load derivation are the sail forces and the geometrical arrangement of booms and sails which is given by

- the sail architecture,
- the type of boom-sail interface,
- the type of sail stowage and
- the type of boom stowage.

Early in the project a sail divided into four sail quadrants with booms running along the sail diagonals was selected (see Figure 1-1). The triangular sail quadrants are attached in the three corner points: each of the two outward corners is connected to one boom-tip while the inner corner is attached to the cubesat structure. A simultaneous deployment of booms and sails is selected whereby the sail quadrants are pulled out of their stowage module when the booms are being deployed (see Figure 3-1). It was agreed among the partners that the forces applied by the sails at each attachment point shall never exceed 1N. The resulting maximum load per boom tip is thereby 2N. For the sail stowage a zigzag folded sail reeled on a central spool is assumed (for the final DE-ORBIT SAIL design the type of sail stowage has changed; described are the conditions during the boom development phase).

Two dimensioning load cases are identified for the boom design:

(1) Bending due to lateral forces with axial compression during the deployment phase,
(2) Axial compression during the deployment and operational phase.

The first load case arises when the booms and sails are being deployed. If one of the two sail quadrants attached to one boom tip becomes slack the boom is loaded asymmetrically and significant bending arises superimposed by axial compression. Decisive for the bending load is the in-plane angle of attack of the sail tension force. In the beginning the angle is close to zero as the sail force is parallel to the booms longitudinal axis. When in-plane opening of the sail quadrant occurs the angle increases to a maximum of 22.5deg at the end of the deployment (see Figure 3-1).
With the geometrical arrangement of booms and sails known, one can calculate the forces and moments acting on the boom during the deployment phase. Figure 3-2 shows the forces when only a single quadrant applies loads to the boom tip plotted over deployed boom length. This leads to a maximum in in-plane lateral forces and moments. The serrated curves result from the zigzag folding pattern of the sail.

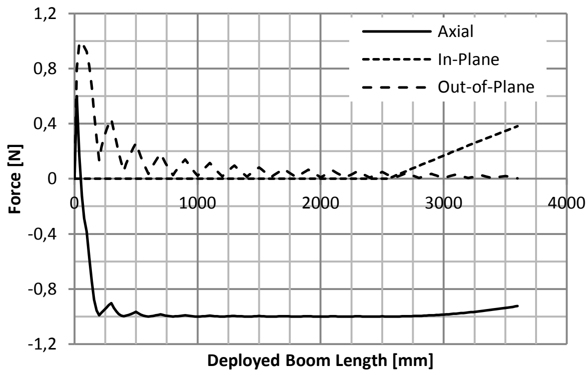


**Figure 3-2: Forces during deployment resulting from load case one.**

The second load case arises when the sail is fully deployed or close to being fully deployed. Both sail quadrants are symmetrically attached to one boom tip causing axial compression on the fully or almost fully deployed boom. Figure 3-3 shows the forces for symmetric loading of the boom plotted over deployed boom length. For load case two the relevant region is at the very end of the deployment with a deployed length of 3600mm.

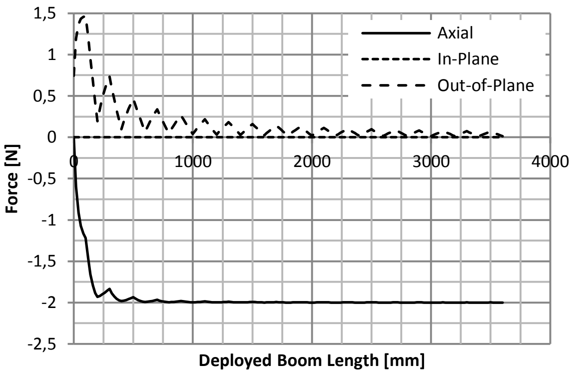


**Figure 3-3: Forces during deployment resulting from load case two.**

For both load cases the out-of-plane forces possess significant values in the very beginning of the deployment but do not cause critical moments due to the small lever arm.

For the final derivation of design loads, safety factors are defined according to [9] and [10]:

- Model factor $K_M = 1.2$;
- Project factor $K_P = 1.0$;
- Qualification factor $K_Q = 1.25$;
- Ultimate safety factor for buckling FoSU = 1.5;
- Yield safety factor FoSY is not considered.

The overall safety factor FoS is thereby 2.25 and is applied to the sail tension force. The resulting sail forces applied at the boom tip are thereby 2.25N for load case one and 4.5N for load case two. Table 3-1 gives the axial, in-plane and out-of-plane forces and moments acting on the booms (the values in brackets denote maximum values at the very beginning of the deployment which decay rapidly and cause no critical loading).

**Table 3-1: Forces and moments resulting from load case one and two.**

| **Load Case** | **F Axial** [N] | **F In-Plane** [N] | **F Out-of-Plane** [N] | **M In-Plane** [Nm] | **M Out-of-Plane** [Nm] |
|---|---|---|---|---|---|
| LC 1 w/o FoS | 1.0 | 0.38 | 0.02 (1.0) | 1.38 | 0.09 |
| LC 1 w FoS | 2.25 | 0.86 | 0.05 (2.25) | 3.11 | 0.20 |
| LC 2 w/o FoS | 2.0 | 0.0 | 0.08 (1.46) | 0.0 | 0.32 |
| LC 2 w FoS | 4.5 | 0.0 | 0.18 (3.29) | 0.0 | 0.72 |

## 3.3. MATERIALS

The boom concepts considered for DE-ORBIT SAIL are shape memory structures based on strain energy. The materials for such structures require a sufficient region of elasticity to avoid plastic deformation when being stowed. The higher this region is the smaller is the achievable minimum curvature radius and the higher is the potential volume efficiency. This aspect is also beneficial for the booms cross-section design as more complex shapes are enabled.

For the booms stability the materials Youngs modulus is crucial. The slenderness ratio ($\lambda = \sqrt{A/I} > 80$, [10]) of the booms is high and the shell thickness is small. Therefore, the tip-loaded boom will fail due to global column or local wall buckling which both are determined by stiffness rather than material strength.

Another important parameter is obviously the material density as it determines the weight of the booms.

Table 3-2 lists potential materials for the DE-ORBIT SAIL booms.

**Table 3-2: Properties of materials suited for small satellite applications.**

| **Material** | **E, $E_{11}/E_{22}$ [GPa]** | **ρ [kg/m³]** | **$\varepsilon_{max}$ [%]** |
|---|---|---|---|
| CFRP-HS UD (Torayca T700S 60% FVC) [11] | 135 | 1540 | 1.7 |
| CFRP-IM UD (Torayca M30S 60% FVC) [12] | 175 | 1540 | 1.6 |
| CFRP-HM UD (Torayca M55J 60% FVC) [13] | 340 | 1540 | 0.6 |
| GFRP UD (E-glas 60% FVC) [14] | 45 | 2000 | 2.2 |
| Titanium Alloy Ti-3Al-8V-6Cr-4Mo-4Zr [15] | 104 | 4820 | 1.07 – 1.37 |
| Copper-Beryllium CuBe2 [16] | 131 | 8360 | 0.56 - 0.95 |
| Steel (1.1231, 1.5024, …) [17] | 210 | 7800 | 0.57 – 0.9 |
| Cobalt Alloy (Elgiloy, Phynox) [18] | 190 - 210 | 8300 | 0.8 – 1.05 |

The material chosen for the DE-ORBIT SAIL booms are intermediate modulus (IM) carbon fibers (see Table 3-2). They possess a similar elastic strain limit as high strength fibers while being in parallel superior regarding Youngs modulus. For small sails steel is of high interest as well due to its even higher modulus. But the much higher density limits the systems scalability to larger sails where booms of increasing volume are required.

To stay within the volume limitations a low shell thickness is required. This results in the need for very thin fiber material and causes a laminate which is composed of only a small number of plies. Therefore intermediate modulus fibers of type Toray M30S are chosen which are available with a thickness of the dry fibers of 0.04mm.

## 3.4. BOOM PROPERTIES

For the derivation of boom properties firstly concepts are selected for further analysis. For the selected concepts boom geometry properties for different stowage volumes are determined and analysed regarding their stiffness and strength properties.

**Boom Concepts**
Three boom concepts which differ in cross-sectional shape and shell thickness are selected for further analysis:

- Open profile boom composed of two shells: **TRAC**-boom;
- Open profile boom composed of one shell: STEM or C-Shape-boom, hereafter denoted as **SEMICIRCLE**-boom;
- Closed profile boom composed of two shells: CTM- or CFRP-boom, hereafter denoted as **LENTICULAR**-boom.

The TRAC boom was selected as it achieves a superior bending stiffness compared to other concepts with same stowage volume[7].
The SEMICIRCLE is chosen because it possesses due to its single shell nature the doubled shell thickness for same stowage volume. It is thereby also not endangered to buckling when being reeled for stowage which is an issue for all concepts composed of two shells as the inner shell is loaded in compression.
The LENTICULAR boom was selected as it possesses a closed cross-section which features a high torsional stiffness and is thereby less vulnerable to flexural-torsional buckling which is critical for highly slender booms.

**Boom Geometry**
The overall volume for the boom deployment module is limited to 1U. As there is some space required for structural parts, mechanisms and harnessing the upper volume boundary for the booms is set to 0.8U. The lower boundary is chosen to be 0.4U. These limitations apply to the height of the stowed boom package. Therefore the width of the flattened boom is limited to vary between 40mm and 80mm.
For the shell thickness of the booms the maximum allowable strain and the number of plies in the laminate are of importance. A minimum of three plies of UD-fibers are required for a symmetric setup (0-90-0). For the fiber material chosen above this leads to a thickness of 0.12mm for the TRAC and the LENTICULAR boom and of 0.24mm for the SEMICIRCLE boom. Using a maximum allowable strain of 0.8% ($t/D$) this leads to a minimum curvature radius of the booms cross-section of 7.5mm for the TRAC and the LENTICULAR boom and of 15mm for the SEMICIRCLE boom.

The cross-sectional shape of the TRAC boom is determined by the flare angle of its curved parts and by the ratio of flange width to curvature radius. For a flare angle of 90deg and a ratio of 0.37 same moments of inertia around both main axes are achieved.
For the SEMICIRCLE the opening angle of its cross-section is the main geometrical parameter and is set to 180deg. To achieve same moments of inertia an angle of 360deg is necessary but curvature radii below 15mm would be required causing strains above the allowable strain.
The LENTICULAR boom has a more complex shape than the TRAC and the SEMICIRCLE. Here the minimum curvature radius of 7.5mm is the limiting factor. This applies especially to the boom versions with lower stowage volumes.

**Bending Stiffness**
The bending stiffness is derived from analysis of the booms cross-sectional shape and the axial material modulus. Using Eulers's equation for column buckling one can determine the critical axial compression loads.
Table 3-3 shows the bending stiffness values and corresponding critical compression loads for the booms of 3.6m length.

**Table 3-3: Bending stiffness and compression strength.**

| **Type** | **Size** | **EI In-Plane** [Nm²] | **EI Out-Of-Plane** [Nm²] | **$P_{crit}$ In-Plane** [N] | **$P_{crit}$ Out-of-Plane** [N] |
|---|---|---|---|---|---|
| TRAC | 0.4U | 81.9 | 81.4 | 15.6 | 15.4 |
| | 0.6U | 278.8 | 275.4 | 53.1 | 52.4 |
| | 0.8U | 656.8 | 653.2 | 125.0 | 124.4 |
| C-Shape | 0.4U | 16.2 | 77.6 | 3.1 | 14.8 |
| | 0.6U | 54.6 | 254.4 | 10.4 | 48.4 |
| | 0.8U | 129.5 | 616.0 | 24.7 | 117.3 |
| LENTICU-LAR | 0.4U | 21.6 | 101.0 | 4.1 | 19.2 |
| | 0.6U | 126.6 | 267.1 | 24.1 | 50.9 |
| | 0.8U | 381.0 | 576.2 | 72.5 | 109.7 |

**Bending Strength**
Bending strength is determined with finite element analysis on boom samples of 1m length with one end fixed. A sensitivity analysis on the element size is performed prior to the strength calculations.
To account for mixed shear and bending loads a lateral force is used instead of pure bending. Imperfections in the load introduction are considered by applying a small torsional moment at the load introduction point with a magnitude of 0.1Nmm per 1N lateral force. Imperfections in the shell are considered by a small disturbance force applied in a distance of 150mm to the clamped boom root which is acting on the compression loaded border area and has a magnitude of 0.1% of the applied lateral force.
The resulting maximum lateral forces are shown in Table 3-4.

**Table 3-4: In-plane and out-of-plane strength values.**

| **Type** | **Size** | **$F_{crit}$ In-Plane** [N] | **$F_{crit}$ Out-of-Plane** [N] |
|---|---|---|---|
| TRAC | 0.4U | 0.66 | 1.03 |
| | 0.6U | 0.94 | 1.62 |
| | 0.8U | 1.20 | 2.28 |
| C-Shape | 0.4U | 0.80 | 2.65 |
| | 0.6U | 2.68 | 7.35 |
| | 0.8U | 3.66 | 10.11 |
| LENTICULAR | 0.4U | 3.32 | 4.02 |
| | 0.6U | 6.95 | 4.95 |
| | 0.8U | 10.55 | 6.58 |

## 4. CONCEPT SELECTION

Based on the results of section 3 the selection of a boom concept for DE-ORBIT SAIL is performed. Figure 4-1, Figure 4-2 and Figure 4-3 display the properties of the boom concepts for 0.4U, 0.6U and 0.8U stowage volume using a radar chart. The upper vertical axis shows the in-plane stiffness and the lower part the in-plane strength. The right horizontal axis shows the out-of-plane stiffness and the left axis the out-of-plane strength. The values shown are normalized to the maximum values reached by any of the boom concepts of same stowage volume.

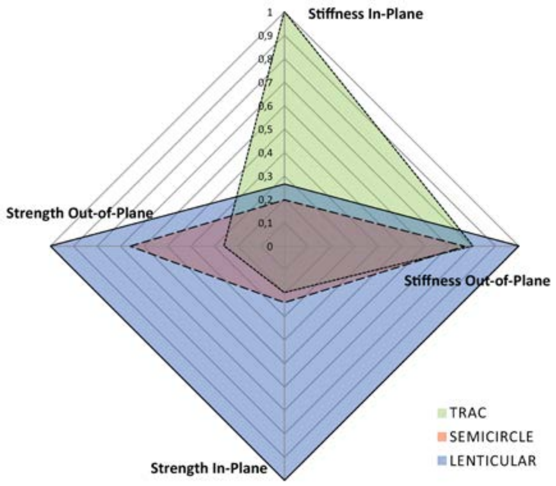


**Figure 4-1: Radar chart of the 0.4U boom properties.**

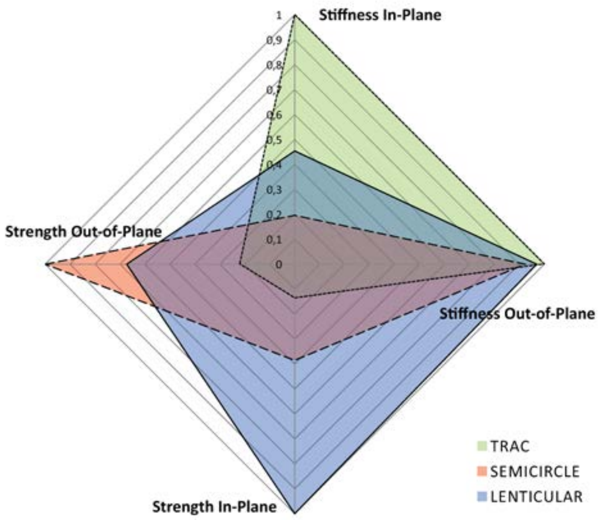


**Figure 4-2: Radar chart of the 0.6U boom properties.**

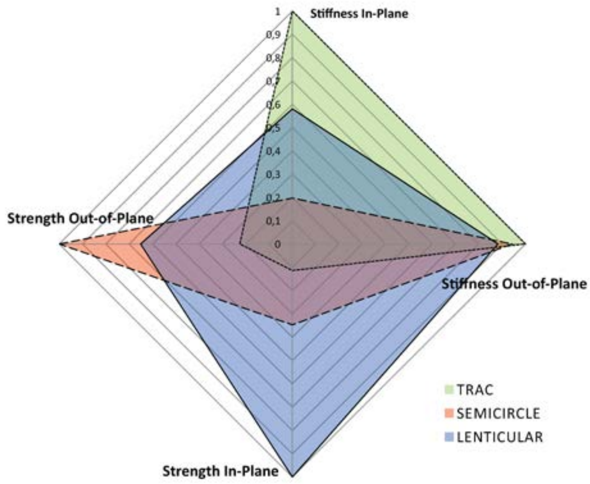


**Figure 4-3: Radar chart of the 0.8U boom properties.**

When evaluating the concepts one can see that the TRAC-boom reaches the highest stiffness values. Table 3-1 shows that the smallest version of 0.4U stowage volume already satisfies the requirements from load case two. But its strength towards lateral forces is small in both axes wherefore load case 1 is critical.

The SEMICIRCLE shows a mixture of good out-of-plane strength and stiffness properties but poor performance in the in-plane direction. For the second load case the minimum stiffness of both main axes is relevant and for load case two the in-plane strength is important. Therefore, the SEMICIRCLE does not profit from its good out-of-plane properties.

The LENTICULAR boom has in the 0.4U version a low out-of-plane stiffness due to a strong cross-sectional asymmetry. This is caused by the minimum curvature radius which limits the cross-sectional in-plane dimensions. For the larger versions the values of in-plane and out-of-plane stiffness start to converge. In total the LENTICULAR boom concept shows very good strength and good stiffness values.

As a result the LENTICULAR design is chosen for the DE-ORBIT SAIL booms over the TRAC and the SEMICIRCLE design. Its good strength properties is a major factor regarding the design objective of scalability as flexural torsional buckling of the open-profile booms becomes more critical with increasing boom length and slenderness. The strength values for this comparison are derived from a rather short boom of one meter length. Therefore, one can assume that this behaviour will become more dominant when going to the full length of 3.6 meter.

## 5. FINAL BOOM DESIGN

For the final DE-ORBIT SAIL boom there was a set-back on the materials side as the originally selected material was not available for the project. The alternative material selected is a plain-weave fabric of 140g/m² weight and 0.14mm thickness which is also used for the booms of DLR's GOSSAMER project (the high thickness is caused by a rather rough fabric). It utilizes Torayca T700 fibers and is applied in a ±45deg

orientation to limit the stresses in the compressed inner half-shell by the expense of axial stiffness.
The selection of the geometrical dimensions of the final boom design is based on FE-analysis with full length booms. The analysis is conducted for the two load cases identified in section 3.2. A boom design with 0.67U flattened height complies with the strength and stiffness requirements of both load cases. Figure 5-1 displays the geometrical dimensions of this design.

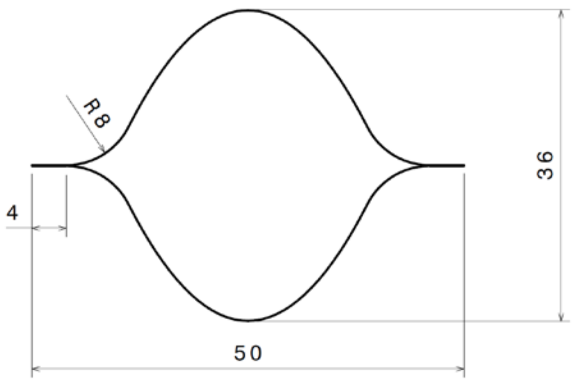


**Figure 5-1: Deployed cross-sectional dimensions fo the DE-ORBIT SAIL boom.**

Figure 5-2 shows the applied tip loads for the two load cases plotted over boom tip displacement. The maximum sail tension force for load case one is thereby 3.3N which is well above the required value of 2.25N. The maximum axial compression load is the limiting factor for the boom design due to the low axial material modulus of the ±45deg fabric. The LENTICULAR boom of 0.67U just satisfies with 4.7N the compression load requirement of 4.5N from load case two.
The specific weight of the final boom design results to 19g/m whereby a total weight for the four 3.6m booms of 274g is achieved.

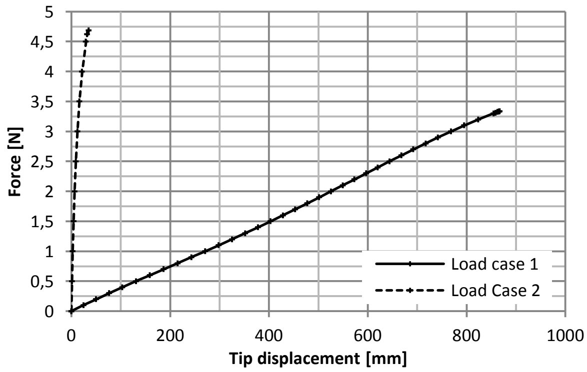


**Figure 5-2: Forces from load case one and two plotted over boom tip displacement.**

## 6. MANUFACTURING

Currently the half-shells of DLR's CFRP-booms are manufactured separately and are bonded in the following step to one boom. The quality of this process depends strongly on the thickness of the applied adhesive, the surface preparation and the applied pressure. To simplify this task and ensure a consistent bonding, a new manufacturing has been developed within the DE-ORBIT SAIL project. The booms (see Figure 6-1) are manufactured in an integral way in one piece by using an outer and an inner vacuum bag: the inner vacuum is running inside the boom while the outer one surrounds the entire manufacturing tool. This manufacturing process is displayed in Figure 6-2.

**Figure 6-1: DE-ORBIT SAIL boom of 3.6m length self-supporting under gravity.**

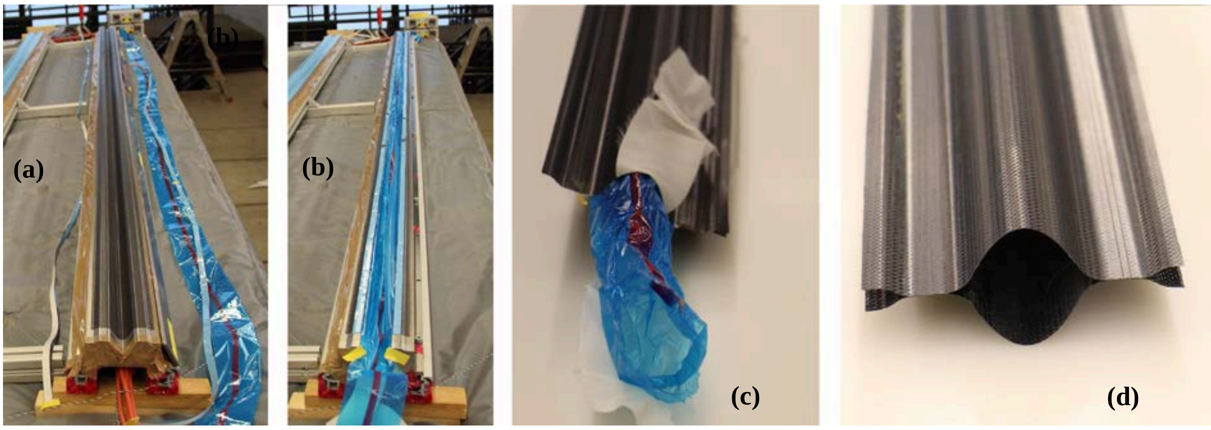


**Figure 6-2: Manufacturing process 1: (a) carbon fiber prepreg laid into one tool-half, (b) inner vacuum bag added, (c) boom after curing still with inner vacuum bag, (d) boom with inner vacuum bag removed before cutting of the edges.**

## 7. CONCLUSION

The design of the DE-ORBIT SAIL booms has been presented. It is based on a survey of applicable boom concepts from which the TRAC, the SEMICIRCLE and the LENTICULAR boom design are selected for further analysis regarding their stiffness and strength properties. Booms with a size of 0.4U, 0.6U and 0.8U are examined and compared. Based on these results and the design objectives and constraints identified for DE-ORBIT SAIL the LENTICULAR boom concept was selected as it features very good strength and good stiffness properties. The final boom geometry with a flattened width of 0.67U was found by FE-analysis and satisfies the load case requirements. Finally the integral manufacturing method which allows manufacturing of the booms in one piece is described.
Testing of the mechanical performance of the booms with the full length of 3.6m is scheduled for the near future.

## 8. REFERENCES


1. L. Johnson, *Nanosail-D: A Solar Sail Demonstration Mission*, 6th IAA Symposium on Realistic Near-Term Advanced Scientific Space Missions, July 6-9, 2009, Aosta, Italy
2. C. Biddy, *LightSail-1 Solar Sail Design and Qualification*, 41st Aerospace Mechanism Symposium, May 16-18, 2012, Pasadena CA, United States
3. V. Lappas et al, *CubeSail: A low cost CubeSat based solar sail demonstration mission*, Advances in Space Research, Volume 48, Issue 11, 1 December 2011, Pages 1890–1901
4. J. Fernandez et al., *Deployment Mechanism of a Gossamer Satellite Deorbiter*, 15th European Space Mechanisms and Tribology Symposium 2013, 25th to 27th September 2013, Noordwijk, The Netherlands
5. http://deploytech.eu, 03.03.2014
6. Northrop Grumman, *STEM Products & Programs*, http://www.northropgrumman.com/BusinessVentures/AstroAerospace/Products/Documents/pageDocs/STEM_Hardware_Programs.pdf, 03.03.2013
7. J. Banik et al, *Performance Validation of the Triangular Rollable and Collapsible Mast*, 24th Annual AIAA/USU Conference on Small Satellites, 09-12.08.2010, Logan UT, United States
8. J. Block et al, *Ultralight Deployable Booms for Solar Sails and other Large Gossamer Structures in Space*, 60th International Astronautical Congress, 2-16 October 2009, Daejeon, Republic of Korea
9. ECSS Secretariat, Space engineering - Structural factors of safety for spaceflight hardware, ECSS-E-32-10 Rev.1, 6. March 2009
10. ECSS Secretariat, Space engineering – Buckling of structures, ECSS-HB-32-24A, 24. March 2010
11. Torayca T700S Data Sheet, Technical Data Sheet No. CFA-005
12. Torayca M30S Data Sheet, Technical Data Sheet No. CFA-010
13. Torayca M55J Data Sheet, Technical Data Sheet No. CFA-017
14. H. Schürmann, Konstruieren mit Faser-Kunststoff-Verbunden, Springer, 2005, ISBN 3-540-40283-7
15. http://www.matweb.com/search/datasheettext.aspx?matguid=fd55749e2f7a4715aeb166dc60c02e3d, 03.03.2014
16. Copper Beryllium CuBe2 Data Sheet, http://www.matthey.ch/fileadmin/user_upload/downloads/Fichiers_PDF/CuBe_comp_sansTM_al.pdf, 09.12.2013
17. Saarstahl Data Sheet, Werkstoff-Datenblatt Saarstahl – C67S, 09.12.2013
18. UGI Phynox Data Sheet 02-02-2010 – REV01, http://www.ugitech.com/uploads/ugitransfer/docs/GED_1857.pdf, 09.12.2013