Feasibility Study for the Use of Compliant Structures in Insert Elements to Allow for an Isostatic Mounting of Components

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Bachelor Thesis
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The main goal of the project iBOSS (Intelligent Building Blocks for On-Orbit Satellite Servicing) is to develop a modular concept for spacecraft. The building block concept shall allow for spacecraft maintenance in space by an autonomously operating servicing robot. An interdisciplinary team from various universities and research institutes is performing the study. In this context, the Institute for Structural Mechanics and Lightweight Design (SLA) is developing, analysing and producing structural components and mechanisms.

The challenge created by the current design is the use of different materials for structural elements and payload components. Structural elements need to be built from CFRP materials in order to reduce the thermal expansion of one module. This is necessary as otherwise the occurring temperature gradient over multiple modules of the assembled satellite would cause high deformations, thermal stresses and may hinder the robotic manipulation of modules due to increased tolerances. On the other side mechanisms and payload components tend to be constructed from metallic materials like aluminum or titanium which have a significantly larger coefficient of thermal expansion than the surrounding CFRP structure. To prevent any damage to both structures the design needs to guarantee that the resulting thermal stresses do not exceed the allowed maximum stresses of the materials.

One approach for a design solution regarding this problem is the use of an isostatic mounting which allows to reduce the occurring thermal stresses by a compliant behavior of the mounting elements. The aim of this work is to analyze if it is possible to integrate such a structure into a specially designed insert. Therefore, different design concepts shall be developed and investigated regarding their feasibility. This includes for example the compliance in the desired direction, remaining strength and stiffness in the two load carrying directions and the maximum stresses in the insert structure itself.

To be able to give reliable results regarding strength and stiffness values appropriate FE Models need to be set up representing the relevant structures and boundary conditions. Based on these results at least one design concept shall be chosen to be developed in more detail. The detailed design should improve the previously identified critical areas for example by focusing on reducing stress levels. Also the manufacturing of the mechanical parts shall be looked at by considering possible manufacturing and machining processes.

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Abstract

For almost three decades now, there have been projects on technology associated to On-Orbit Servicint (OOS) and fields such as space robotics and modular design of space infrastructure elements. Many challenges aroused due to the lack of standards and interfaces, but because of technology developments made over recent years, the iBOSS concept now aims to combine modularity, interfaces, multi-mission scenarios and servicing principles all the way to economic assesments.

Within this context, there is a need for a mechanism that enables the thermal expansion of the interface within the modular block systems, without possible damage or deformation to avoid major issues such as the hindering of the On-Orbit Servicing. An adequate design of a compliant insert would meet these needs.

This thesis presents different designs that would potentially behave within the requirements, both for a launch scenario and in resistance to the space environment. A study on the performance of the stresses is carried out by varying parameters such as the thickness and length of the inserts using the Finite Elements Method.

The work concludes with the extraction of a trend that describes the correlation between the variation in the parameters and the resulting values for the stresses. The designs that would best work for the problem solving are outlined and a way to reach the required stress values is suggested.

Keywords: Compliant Mechanisms, Finite Elements Method, Parametric Study, iBOSS

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Chapter 1

Objectives and scope of the project

This thesis is a contribution to the iBOSS Project, in its will to prevent potential problems related to increased tolerances in the conceived spacecraft system. The concerns reside in a possible thermal expansion over the modules that form its structure, causing high deformations, thermal stresses and impediments in their manipulation. In order to prevent any damage to adjacent structures, a design that guarantees that the thermal stresses between components of different materials do not exceed the allowed maximums needs to be developed. A first approach to the problem may be the use of a compliant mechanism, which would enable an isostatic mounting of the parts.

This chapter will briefly point out the main objectives of the project as well as examining the approach to be followed to attain them. Finally, the details of the problem to be solved will be addressed.

1.1 Objectives

The objectives to accomplish in this project will be the following:

- to analyze the possibility of integrating a compliant structure into an already designed part,
- to create different suitable designs,
- to study the feasibility of such designs,
- to develop in more detail the most convenient design,

– to prove the stress levels are reduced with respect to the critical areas,

1.2 Approach

In order to achieve the abovementioned objectives, the fundamental approach to any project will be followed: research, design and testing.

– Research

  As a non-traditional type of mechanism, compliant designs are not yet widely employed in engineering. Also, as a developing framework and due to the complexity of their design, some still feel reluctant to give them a try.

  The research part will consist in proving whether or not these types of elements will be suitable to allow for an isostatic mounting and searching for possibilities of design.

– Design

  A compliant mechanism shall then be designed with the help of a CAD program. More than one design will be produced, including improvement and optimization of the different models. Manuals and previous research will be used as a guide to design such models. The most suitable design shall be chosen to continue to work with.

– Testing

  As it is actually a part destined for the aerospace industry, the testing will take part in a FEM software, in which it will be proved whether or not the objectives aforementioned are reached. The FE Models will be constrained by a set of boundary conditions. If a design fails to work, it shall be corrected or improved.

  In particular, it should be checked whether there is compliance in the desired direction, if there is strength and stiffness remaining in the two load-carrying directions and if the maximum stresses in the insert structure itself are suitable for the overall model.

1.3 Problem Addressed

Within the cited iBOSS Project, the aim of this thesis is to find a way to allow for an isostatic mounting between components in the satellite by the use of a compliant structure in insert elements.

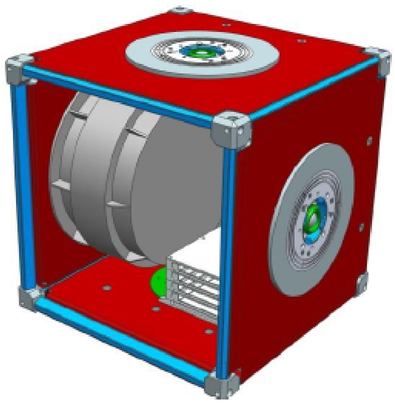
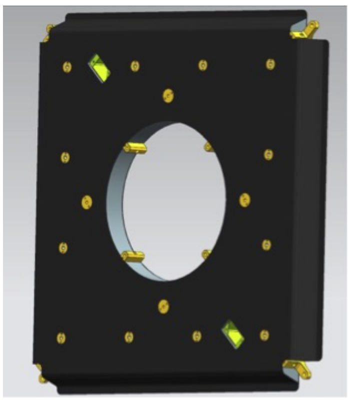
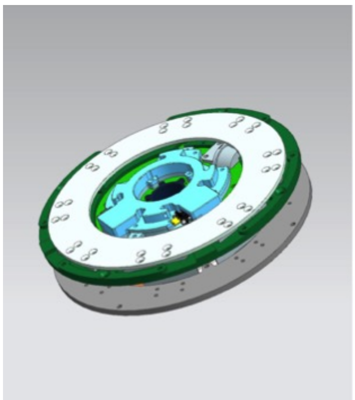


Figure 1.1: Block detail

iBOSS is a modular aerospace system composed by building blocks as the one seen in Figure 1.1. The insert elements will be found connecting the round interface to the walls of the cubic blocks. In detail, it should link the interface seen in Figure 1.2b and the pannel seen in Figure 1.2a, through three points of the structure, instead of the four seen, and should allow the materials to expand without damage or deformation of the overall structure.



(a) Pannel



(b) Interface

Figure 1.2: Components of walls of iBOSS blocks.

To sum up, the goal is for the compliant elements to make the parts remain in a state of equilibrium, in the desired pose relative to the rest, without change due to thermal stresses.

Chapter 2

Introduction and Background

This chapter presents the researched theory the thesis is based on. An introduction to the iBOSS project, as well as an overview to understand the characteristics of compliant mechanisms and isostatic mounting will be described.

2.1 Overview

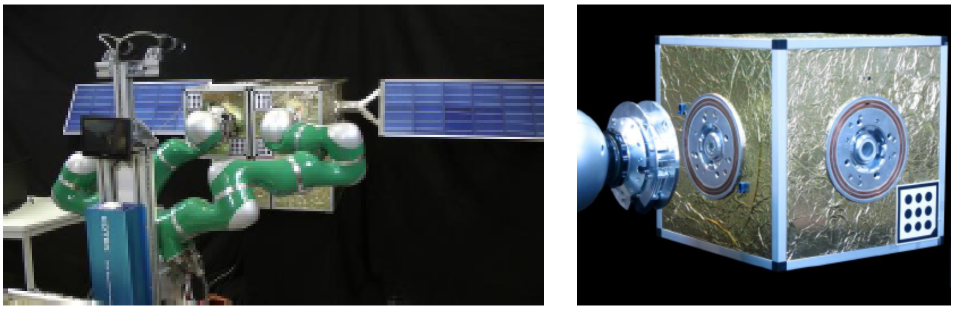
In the need for spacecraft systems with modularity, interfaces, multi-mission scenarios and servicing principles, the German Aerospace Center DLR, took profit of recent technology developments to create iBOSS, short for Intelligent Bulding Blocks for On-Orbit Satellite Servicing. With the cooperation of the Institute of Structural Mechanics and Lightweight Design (SLA) of RWTH Aachen University, among others, the iBOSS concept combines all the above in an integrated approach.

Hereafter, the iBOSS project is briefly presented.

2.1.1 Intelligent Bulding Blocks for On-Orbit Satellite Servicing

The idea behind iBOSS is the belief that the performance of a spacecraft system can be significantly improved by integrating a modular and serviceable design. Modularity represents the idea of building subsystems, independent from one another, by summarizing objects and reducing their complexity. Fragmentation of common satellite systems into standardized building blocks would also enable the idea of serial production, allowing the building of any type of orbital or exploratory space system. All common satellite subsystems could be mounted within iBOSS building blocks as well as payloads. On the other hand, being serviceable would also enable the system to be upgraded and have an autonomous on-orbit maintenance, pictured in Figure 2.1.[1]

Other advantages would include the reduction of design and technical risks, increased flexibility regarding schedules and mission changes as well as economy-of-scale effects at different levels. [1]



(a) Robotic manipulation [1]

(b) Robotic docking mechanical interface [1]

Figure 2.1: On-Orbit Servicing

As seen in Figure 2.2, the system is composed of two parts: a central structure and the building modules. While the former consists of a standardized 4-in-1 interface for docking, power, data and thermal interconnection; the latter mentioned, are blocks with a modular design done both on structural and system levels. The main function of these blocks is to protect the system against the hazardous environment, to control the thermal balance and integrate the docking interface. They also enable a simple and flexible integration of all the components of the system. [1] [2]

The major challenge for the design of the spacecraft system is the great amount of boundary conditions and requirements. The sets of requirements can be classified into two types. [3]

The first type, as in any other spacecraft system, calls for iBOSS to structurally resist the launch period, including the space environment and the withstanding of severe static and dynamic loads. It is also important that the system has a high level of redundancy as per the possible handling that the on-orbit servicing concept imply. [3]

The second group of requirements refers specifically to the desired level of modularity, which must additionally have maximum degree of flexibility in-orbit. Thermal boundary conditions are especially taken into account due to the amount

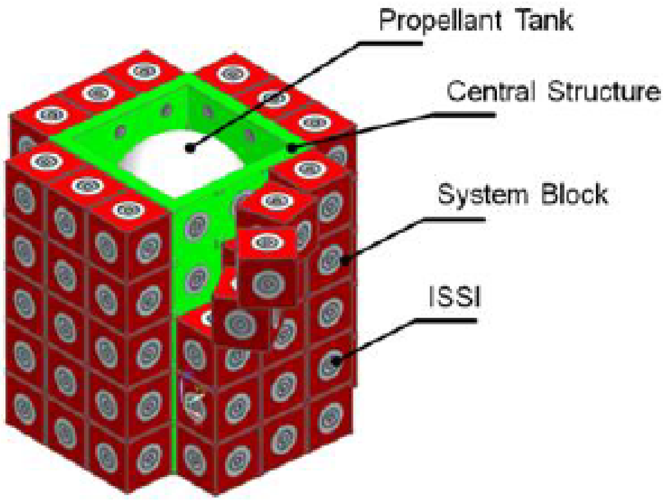


Figure 2.2: Parts of iBoss Aerospace System [1]

of modules and the fact that they will be delt with in a space environment, which demands for all components to stay within calculated tolerances. The cubic building blocks themselves allow high flexibility when it comes to position and orienation of each block. This enables them to be fully retraceable inside the block with no outstanding parts and makes it easy for the servicing robotic system to work with the desired block without interference from other blocks.[3]

All in all, from the structural point of view, the main tasks to achieve this modular design are the development of suitable structural members and a concept for an interface that allows the transfer of loads as well as docking scenarios. In this context, the SLA is mainly in charge of the design of the primary structural components and the required docking interface, as well as the analysis of the structure by FEM. [1]

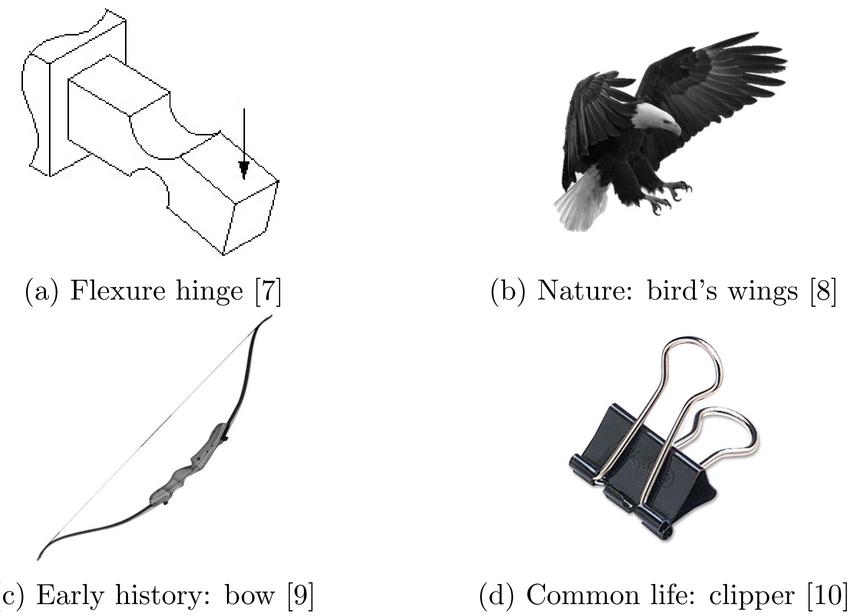
2.2 State of the Art

Interest in compliant mechanisms is arousing in precision engineering. As any other mechanism, they convert or transmit forces or movements from one location to another, but unlike typical mechanisms, they are pliable. The following section will develop this concept and give an insight into their main characteristics and advantages. It should be taken into account that the ultimate purpose of designing a compliant mechanism is to achieve an isostatic mounting of the parts, not only their stress resistance. The concept of isostatic mounting will also be briefly developed.

2.2.1 Introduction to Compliant Mechanisms

A compliant mechanism works as any other mechanism: it transfers or transforms motion, force or energy in order to produce work. The main difference from a traditional mechanism is that, instead of acting as a rigid body, formed by moving parts and joints, compliant mechanisms comply, as their name states. They normally consist of monolithic structures that bend within their elastic range without failing or rupturing. [4]

A common constituent of compliant mechanisms are flexure hinges. These are flexible, slender links that connect two rigid parts that must undergo limited relative rotation in a mechanism.[5]. Other examples of these mechanisms are found in nature, early mechanisms and also in our daily life, exemplified in Figure 2.3. However, because of the complexity of integrating multiple functions into one part, the main design option has traditionally been rigid-body mechanisms. Nowadays, though, thanks to developing technologies, compliant mechanisms may potentially start substituting traditional mechanisms. Their simplicity, among other multiple advantages that will be later on described, emulates a closer approach to nature, which always provides the most optimized solutions. [6]



(a) Flexure hinge [7]

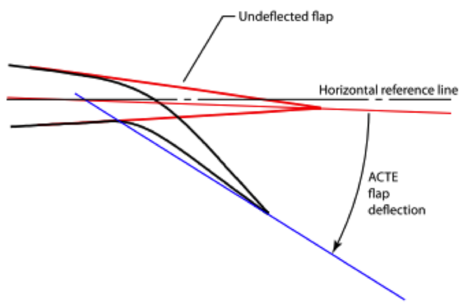
(b) Nature: bird's wings [8]

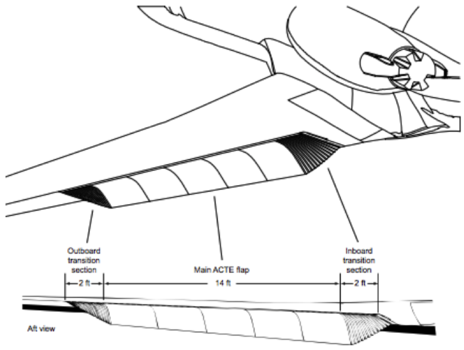
(c) Early history: bow [9]

(d) Common life: clipper [10]

Figure 2.3: Examples of compliant mechanisms.

In this context, new applications in the Aerospace Industry are being tested, such as the Adaptative Compliant Trailing Edge (ACTE), seen in Figure 2.4. This design approaches the behaviour of a plane to that of a bird, imitating the elasticity of their wings with the bending of the flaps instead of them rotating around a hinge. A compliant structure in both the trailing and leading edges enables a hinge-free shape morphing wing and eliminates gaps between the flap and the wing, allowing for advantages such as reduction in noise and weight as well as increase in aerodynamic efficiency, resulting in significant decreases in fuel consumption. [11]





(a) Definition of ACTE flap deflection. [11]

(b) Illustration of the left ACTE flap (deflection of 30 deg). [11]

Figure 2.4: Compliant flaps.

In the same way, patents as the Rotorcraft Leading Edge Technology and Rotorcraft Trailing Edge Technology have been registered. These combine compliant structures into the rotorblades, which enable equal aero performance,less power consumption and delay stall, as Figures 2.5 and 2.6 show. [12]

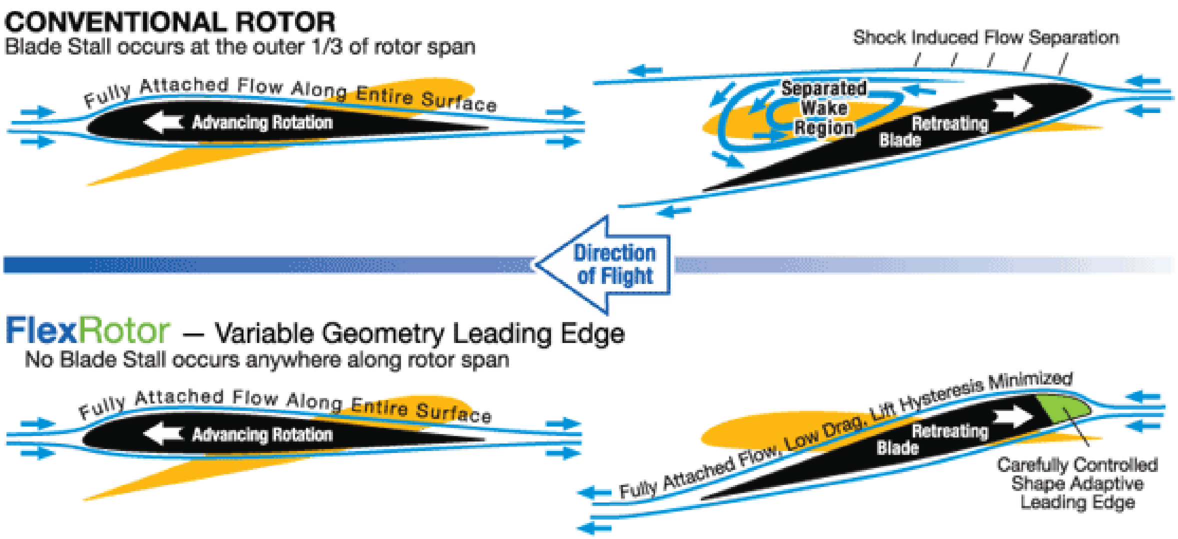


Figure 2.5: [12]

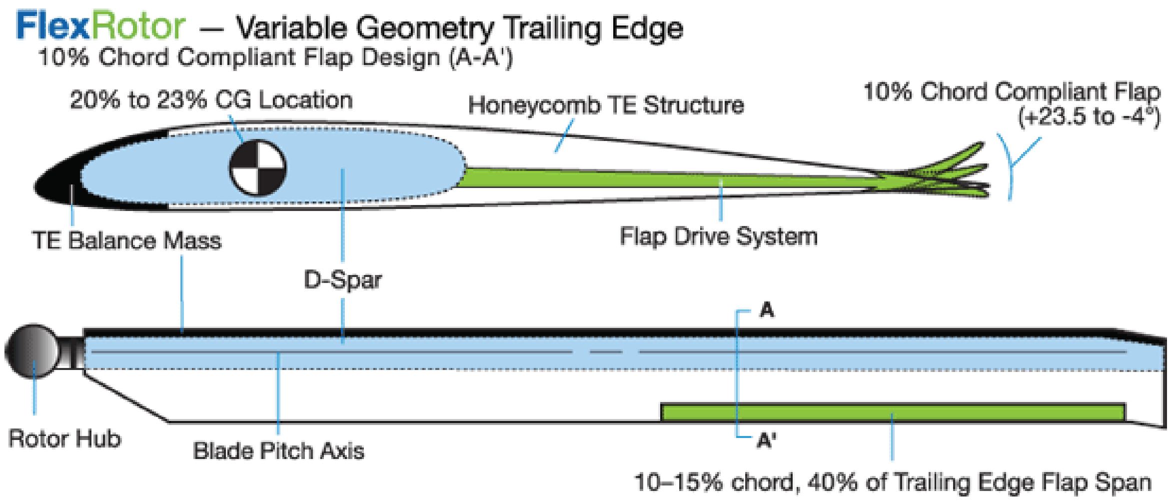


Figure 2.6: [12]

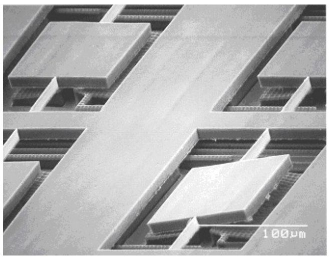
As mentioned, compliant mechanisms are noted for their simplicity. Simplicity, mainly in terms of fewer parts, which is then translated to many other advantages, such as lower costs, due to the need for less components, material, manufacturing and assembly. Also, all this means less weight, which makes the mechanisms suitable in many more applications with a weight requirement, as in the aerospace sector; and facilitates transportation or shipping. [13]

The fact that there is no assembly needed also translates in no need for lubrication between parts and the possibility of scaling the parts, even down to nano-scale applications, as in Microelectromechanical System (MEMS), pictured in Figure 2.7a. No assembly also means no friction losses and, all in all, no need for maintenance. This also eases the integration of the part in an overall mechanism, as it can be set in a place where it is hard to reach, without worrying about maintenance issues, given that the part has a high life expectancy. [14]

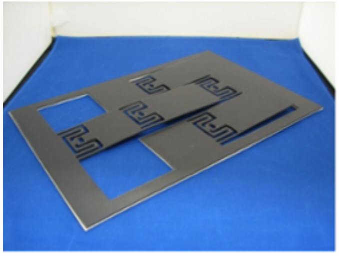
A final advantage may be the possible compactness of the mechanism. The configuration of Lamina Emergent Mechanisms (LEMS) are exemplified in Figures 2.7b and 2.7c. The former shows how they are easily stored as sheets, but in the latter it is seen how they bend to serve their purpose. [14]

Compliant mechanisms may also present some challenges, which should be taken into account before deciding whether their use is suitable or not for the application at issue.

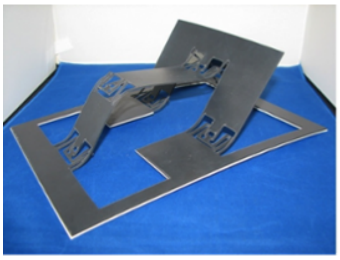
The sometimes-complex design is one of them. It should usually take more time and knowledge to couple into just one piece what a rigid-body mechanism can do; as well as resources, as these sometimes produce non-linear deflections.



(a) Scalable: MEMS [15]



(b) Compactness:LEMS [14]



(c) Compactness:LEMS [14]

Figure 2.7: Advantages of compliant mechanisms.

A proper design will require the prediction of the deflection of flexible parts. Although quations that assume rigidity are no longer valid in compliant mechanisms and new more complicated equations should be employed, nowadays powerful software enables the manipulation of designs much more easily than before. [6]

Most of the times, structures subjected to elastic body deformation are perceived as weak, due to the misconception of stiffness and strength as equivalent concepts. This is not necessarily true, and depending on the materials selected, a mechanism can be both flexible and strong. In order to avoid strength limitations, the materials must be strong enough without comprising too much of the elastic properties. When strength is a major requirement, compliant mechanisms may not be the best option, but there should be a consideration of all factors before a mechanism is chosen or discarded. [6]

Unlike rigid bodies mechanisms, compliant mechanisms are loaded cyclically, so it is important to consider the fatigue life of the parts to ensure that failure is not produced within design parameters and that the purpose of the design is achieved. The concepts of stress relaxation and fatigue life appear and predicting their lifespan requires a lengthy iterative design process, which can only be shortened by the experience in design of these types of mechanisms.[6]

Compliant mechanisms also have the capacity of storing strain energy, by absorbing the energy from an undergoing deformation. This may be either an advantage or a disadvantage depending on the application. It is usually an advantage as it reduces the reliance of spring mechanisms, but when the aim of the mechanism is to transmit the energy from one point to another, the efficiency of the mechanism is reduced as not all of it is transferred. [16] [6]

Also, stiffness or spring rate goes off the scale when high load carrying capacity is needed and it is important to note that they have limited linear and rotatory motion. [16]

2.2.2 Isostatic Mounting

The use of compliant inserts has as an objective to allow for an isostatic mounting of components in the iBOSS aerospace system. An isostatic mounting ensures a highly stable, rigid and vibrationally damped structure which eliminates movement of components relative to each other [17][18]. To successfully achieve isostatic mounting, it is required to understand the coupled space of configurations and controls and avoid singularities that can lead to large constraint forces or internal stresses. Bending and twisting of the surfaces must be minimized to prevent displacement of the components relative to one another to hold structures in the desired pose relative to the rest [19].

Patents have been registered for isostatic mounts mainly in optics applications: allignment of telescopes for astronomy and holding of mirrors on spacecraft for vibration isolation. The main requirements in these cases are good optical performance as well as mechanical and thermal requirements [20][21]. This calls for the structure to support stresses as well as maintain in an equilibrium position with negligible optical distortions. This is important to achieve a proper isostatic mounting while meeting frequency requirements. [22]

Chapter 3

Design Solutions

This chapter will introduce possible design solutions to solve the problem addressed. All of them will have to physically fit the existent design of the components of the iBOSS system, as well as aim to meet the already mentioned specifications in order to become a suitable option for the solution of the problem.

Taking into account all possible downsides, a compliant mechanism can be said to fit the specifications for the insert. Although all the challenges mentioned in the previous chapter should be carefully studied, their advantages make these mechanisms very suitable for the aerospace sector, specially considering future prospects of on-orbit servicing and long endurance geostationary missions. The fact that they store strain energy will particularly result in an advantage. This is because what is indeed looked for in this thesis is that the compliant mechanism absorbs all the stress and deformation, damping possible vibration and avoiding their expansion to the rest of the structure, allowing for an isostatic mounting.

Hence, the final design should be a compliant mechanism with the ability to bend. In order to create flexibility, three main concepts should be taken into account: loading and material properties, geometry and boundary conditions.

3.1 Configuration Of Parts

The loading of the inserts will be determined by the configuration of the parts the inserts are attached to. The simulations will always present the same configuration: a top and bottom plates with three inserts. The inserts will be equally spaced within the circular plates, 120^o from each other and in a radius of 75,5mm from the center of the plates. All of them will be 25,4mm high. The diameter of the plates will be 220mm. The thickness of the top plate will be of 4mm and that of the bottom plate will be 5mm. These dimensions are taken from the CAD models shown in Figure 1.2, as the geometrical boundary conditions. Figure 3.1 shows the dimensions of the plates and the distribution of the inserts with respect to the them.

All elements will have the properties of Aluminium Alloy 7075. Aluminium is widely used in the aerospace industry because of its lightweight and its high elastic limit while still keeping its resistance at low temperatures, as well as its resistance to corrosion and ease to mechanize.

The values for the material's properties are, as seen in Table 3.1, an Elastic Modulus of 72GPa, a Poisson Ratio of 0,33, a density of 2800 kg/m^3 and a Thermal Expansion Coefficient of 24,4·$10^{-6}$$K^{-1}$. Also, it will have to be taken into account that the maximum stress achievable is its Tensile Yield Strength, which adds up to 503MPa. Bearing in mind that there has to be considered a safety factor of 1,5, it should be aimed for the stresses to be actually below 335,33MPa

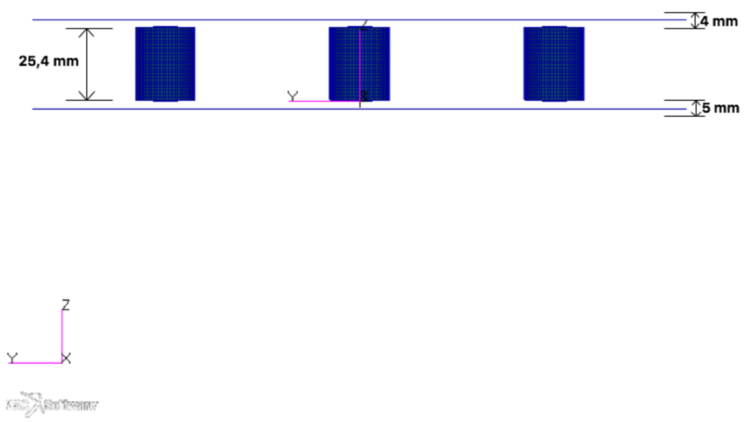
Elastic Modulus	72 Gpa
Poisson Ratio	0,33
Density	2800 kg/m^3
Thermal Expansion Coefficient	24,4·$10^{-6}$$K^{-1}$
Tensile Yield Strength	503MPa

Table 3.1: Properties of Aluminium Alloy 7075

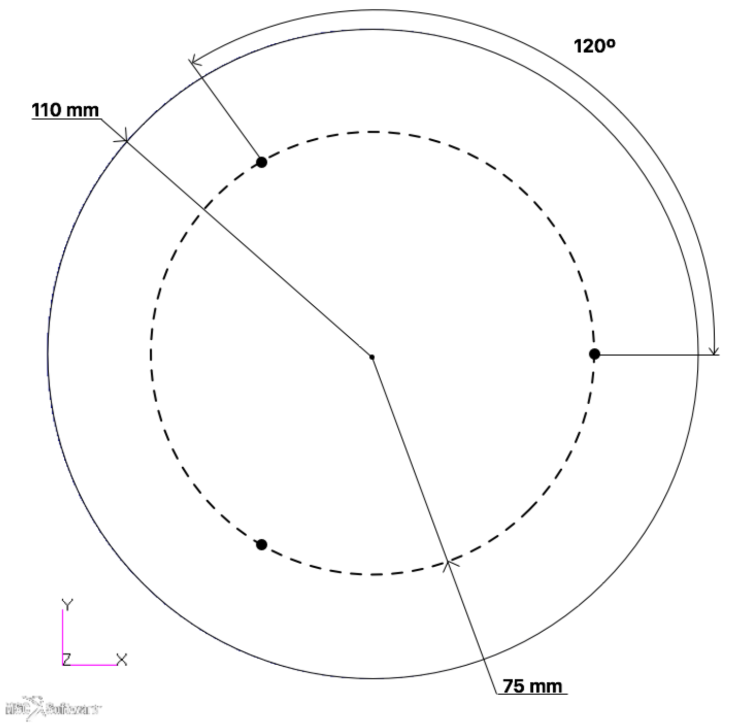
Apart from the mass values of the components that the simulation itself calculates with the size of the present elements and their material properties, an extra mass of 3kg will be added and distributed between the upper and lower plates. This non-structural mass will emulate the mass of the rest of the body which the plates represent. Together, this will make the total structure mass of approximately 4kg. This distribution of the additional mass was chosen, as opposed to the initial thought of including a point mass, because the latter caused an unrealistic distribution of the mass and stresses at a specific zone, instead of equally distributed throughout the whole body.

3.2 Design Solutions

This section shows the final three designed inserts. The decision on whether further developing or not a design idea was mainly made on the geometry itself. The behaviour in reaction to thermal expansion and acceleration in all directions was



(a)

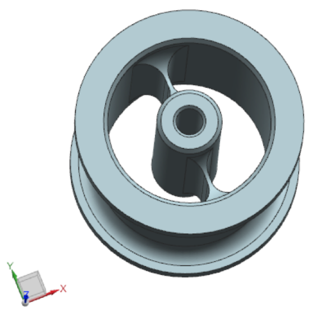


(b)

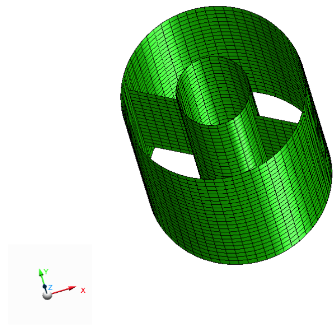
Figure 3.1: Insert distribution within plates.

visualized beforehand. Symmetry played an important role on the decisions, as a symmetrical design would have more probabilities of behaving in a more harmonic way and balancing the stresses out.

The design of the inserts were made in a 3D-CAD model as a solid in NX. For an easier testing of the model, these solid inserts were simplified into surface geometries in MSC Apex, where they were meshed with quadratic elements to a size of $4{\cdot}10^{-4}m$. The testing was carried out using the Finite Elements Method in MSC Patran, where a certain thickness was given to each of the surfaces.



(a) CAD Design 1



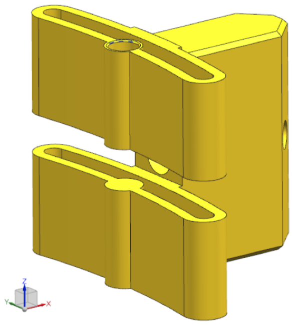
(b) Simplified Geometry Design 1

Figure 3.2

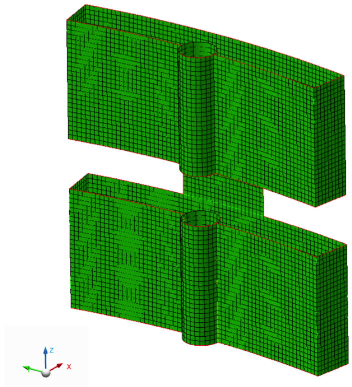
Figure 3.2a shows the geometry of Desgin 1. The coupling for this design does not correspond to the one shown in Figure 1.2, as many design options are still being held for the whole building block. This was the original example design provided as the base model to test and optimize.

All edges and thicknesses are observed to have been simplified in Figure 3.2b into plain surfaces. Carrying out the FEM simulations with surfaces instead of solid bodies will simplify the calculations as well as reduce the time they take to place results. Still, these will show where the structural stresses may be found and give an approximate solution fot their values.

Design 2 is shown in Figure 3.3a. This coupling for this design is completely different to the previous one and does correspond to that shown in Figure 1.2. The idea behind this design comes from a discussion with one of the supervisors. The curved structure of the arms of the insert is proven to work as a stress distributor. It was originally conceived to be a whole piece, instead of being separated into two parts with a gap between the two symmetric arms of the structure. It was at first separated because a screw had to go through the middle of the body and the design did not look harmonic, but it was then thought that it would probably work better this way because the material from the upper plate could be different



(a) CAD Design 2

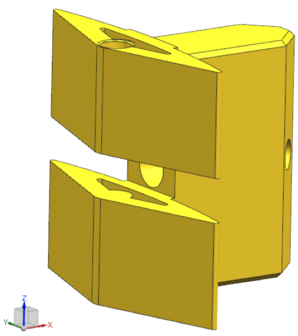


(b) Simplified Geometry Design 2

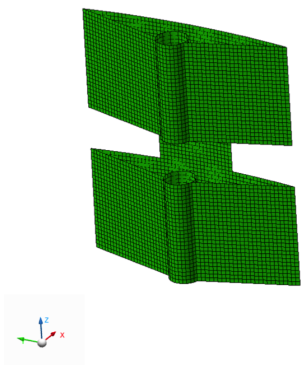
Figure 3.3

from that from the lower plate and thus the thermal expansion could be different and to avoid more stresses. These parts were made long enough in the vertical direction to achieve a more robust structure.

As seen in Figure 3.3b, the back part that is inserted into the interface has been completely simplified as it was to be considered fixed. Also the round corners of the design, which have been completely squared out, as it was thought to work better this way.



(a) CAD Design 3.



(b) Simplified Geometry Design 3

Figure 3.4

Design 3 is shown in Figure 3.4a. Again, this coupling corresponds to the same one as Design 2. The idea was inspired in the guiding examples given in one of the reference manuals, seen in Figure 3.5. Taking as a reference Design 2, this geometry was further developed and the base was widened to obtain better stress absorbtion and the splitted shape from the previous design was again adopted.

It is again seen in Figure 3.4b that the back part that is coupled into the interface has been completely simplified while the rest of surfaces remain as the

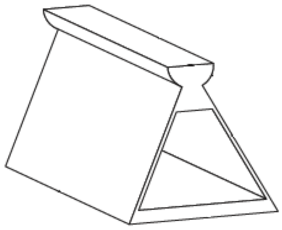


Figure 3.5: Idea from Design 3 [6]

original CAD design.

3.3 Boundary Condtions

The boundary conditions will apply in the same way to all simulations.

In order to attach the plates to the inserts, a contact MPC type RBE2 will be used, connecting the cylinders through which the screws are inserted to the nodes of the plate. In a first trial, the nodes of the cylinders were connected to just one node of the plate, but this would cause an unrealistic high stress distribution on this zone, so this was changed to the four adjacent modes of the plate, which showed a more plausible distribution of stress in the plate.

The zones of the inserts that are directly attached to the pannel will be considered as fixed, this is, no translation or rotation in any of the directions, and will be constrained as such. In the case of the circular coupling of Design 1, this will refer to the outer cylinder; in the other two cases, it will be the part that is assemblied into the pannels, which has been entirely simplified. The orange-coloured nodes in Figure 3.6 exemplify which are the fixed parts for the two types of coupling.

Two different load cases will be put to the test, having previously confirmed that the whole body moves with 6 vibration modes using a Normal Modes Solution Type to make sure that everything is attached to each other. The first case examined will be the thermal load case to simulate conditions in space; in the second one, the body will be subjected to an acceleration in x, y and z to simulate launch conditions. Both will be carried out using a Linear Static Solution Type.

In the thermal case, the whole model will be brought under 1K of temperature and it shall be seen how the compliant mechanism responds to thermal expansion. This value will then have to be scaled to up to 100K, the maximum temperature

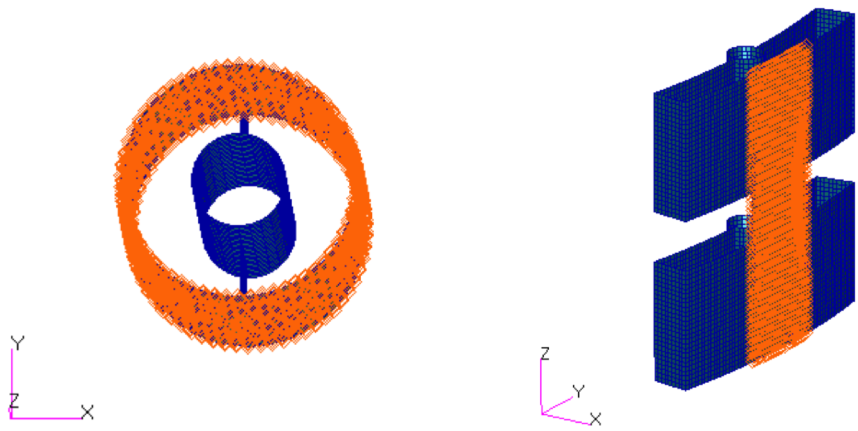


Figure 3.6: Fixed nodes

value that the inserts will have to support.

In the acceleration case, three different subcases are distinguished. An acceleration of 1g (9,81m/s^2) will be applied separately in the x, y and z directions in order to predict stresses in all possible directions, as the inserts will be setted up in many different orientations. The resulting values will also be scaled to up to 30gs to see if the inserts can support this maximum acceleration.

For a quicker processing of the resulting values, Table 3.2 shows the maximum allowable values at 1K and 1g for which the stresses would reach the previously mentioned maximum of 335 MPa for 100K and 30g.

Thermal load case	3,35
Acceleration load case	11,2

Table 3.2: Maximum stresses allowable at 1K and 1g [MPa]

3.4 Simulations

The models will be tested for varying parameters to reach an optimum solution within the same design, in particular for different thicknesses (t) and lengths (l).

The thickness of the non-compliant parts will be 2mm in all cases, while the thickness carried throughout the cross section of the compliant parts, pictured in red in Figure 3.7, will be simulated for 1mm, 2mm and 3mm for all the cases.

For Design 1, it is worth noting that although the body of the insert is cylindrical and could be disposed in any orientation, it was specifically chosen for

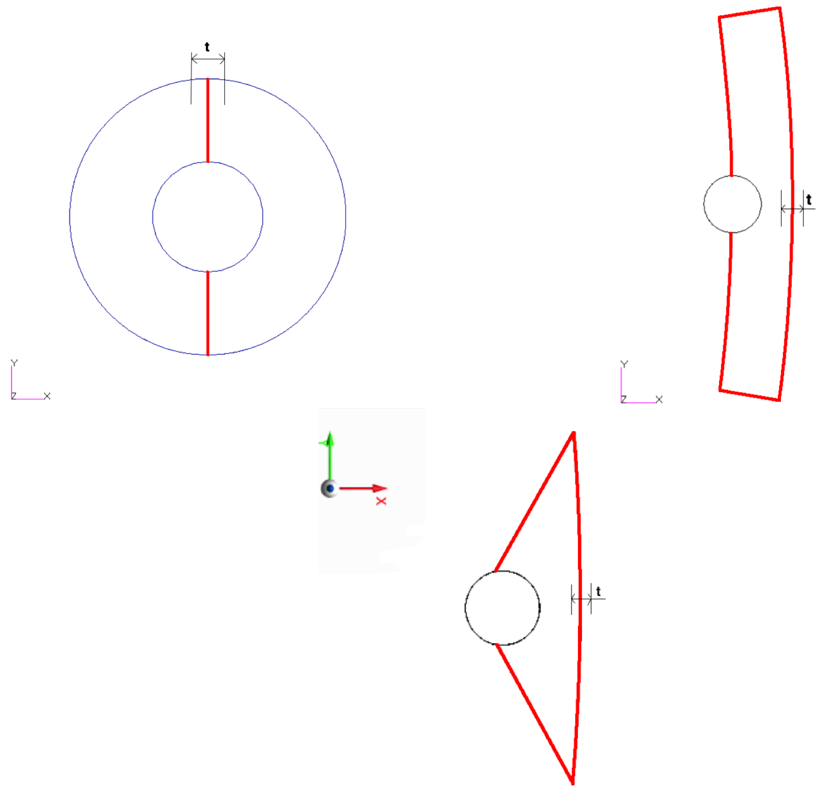


Figure 3.7: Thickness variation

the spring parts to be tangent to the 75mm circle pictured in Figure 3.1b so that compliance was met in the correct direction to support expansion. In a similar way, Designs 2 and 3 were coupled as shown in Figure 1.2a.

The lengths will vary as pictured in Figure 3.8. As detailed further on, only Designs 2 and 3 will be varied lengthwise. Table 3.3 shows the value of lengths for each variation, as they are not consistent with each other and cannot be compared as the designs have completely different geometries.

D1	L1	10,6
	L2	12,6
	L3	14,6
D2	L1	10,0
	L2	15,0
	L3	17,0

Table 3.3: Lengths [mm]

As there are many simulations, an example for each design will show where the maximum stresses are found. Although the parameters are varied these will always be found in the same place, but in a different proportion.

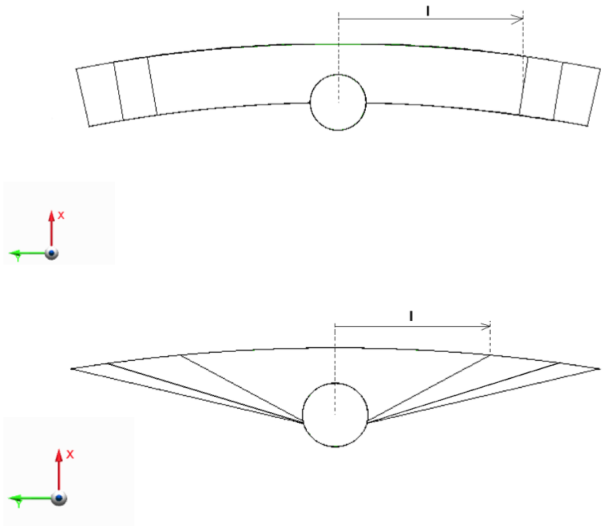


Figure 3.8: Length variation

On the whole, it will be analyzed whether the results are coherent or not, shown where the biggest displacements are found and whether the stresses stay within the requirements.

Facing the results of the simulation, it is worth noting that one of the inserts lies on the X axis, so, specifically for the case of acceleration in X Direction, the other two inserts will behave symmetrically and with higher stresses, as the acceleration will not face them directly. The acceleration in Y direction will be a more significant representation in the acceleration in a random direction. For the thermal load case and the acceleration in Z direction, all the inserts will be affected in the same way, so all the inserts will behave in the same way.

	CASES			
	Thermal	Acc z	Acc x	Acc y
t = 1mm	12,1	0,745	1,05	1,01
t = 2mm	12,7	0,473	0,715	0,648
t = 3mm	13,2	0,470	0,578	0,538

Table 3.4: Maximum Stresses Design 1 [MPa] for 1K (Thermal Load Case) and 1g (Acceleration Load Case).

Table 3.4 shows the results of the simulations for three different thicknesses in Design 1. The first column shows the stress results to an applied temperature of 1K, the remaining three columns show the stress results when an acceleration of 1g is applied in the z, x and y directions respectively.

The values for the stresses in the acceleration load case remain very low and do not reach in any case the maximum value of 11,2MPa. On the other hand, the values for the thermal load case are far higher and exceed the maximum stress permited at 100K, which is 3,55MPa.

These thermal stresses are seen to increase with thickness. This is expected, as the thickness parameter is related to the moment of inertia in the mechanical equations and when thickness increases, so do the stresses. Nevertheless, these increase variations remain quite small as the compliant part of this design is small in proportion to the whole body of the insert.

On the other hand, the stresses for the three acceleration load cases are seen to decrease with thickness. This behaviour is also expected, as more thickness provides more stiffness to a body and therefore makes it more resistant. Again, due to the small proportion that the compliant part represents in the body, the stress variations are quite small.

As the values for the thermal stresses are too high, this design would not work as a solution to the problem addressed. These stress values could only be lowered with a different design of the compliant part, but as the overall design itself does not allow for big modifications, it was thought that the values would not drop enough when modified. For this reason, the parametric study for this design was not extended.

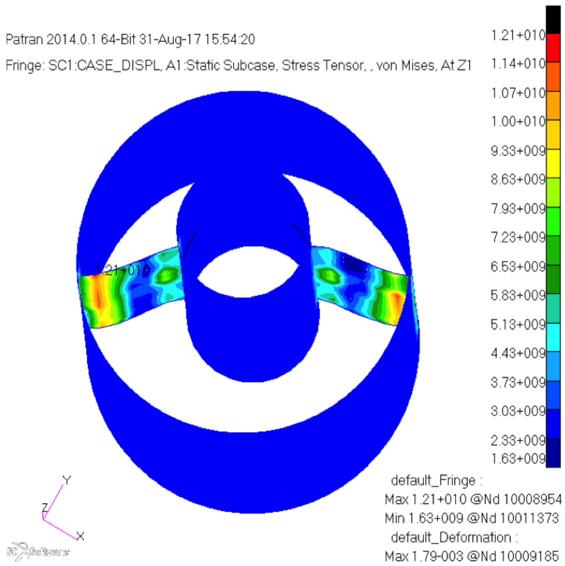


Figure 3.9: Design 1. Thermal stresses for t=1mm and 1000K.

For Design 1, Figures 3.9 and 3.10 show where the maximum stresses would be found, considering a scaling for each thickness case. For the first figure, the highest stresses are held where the maximum deflection is found. This is where the fixed and the compliant part meet. In the second case, a stress peak is found in one of the sides of the spring part, possibly due to the angle in which the acceleration

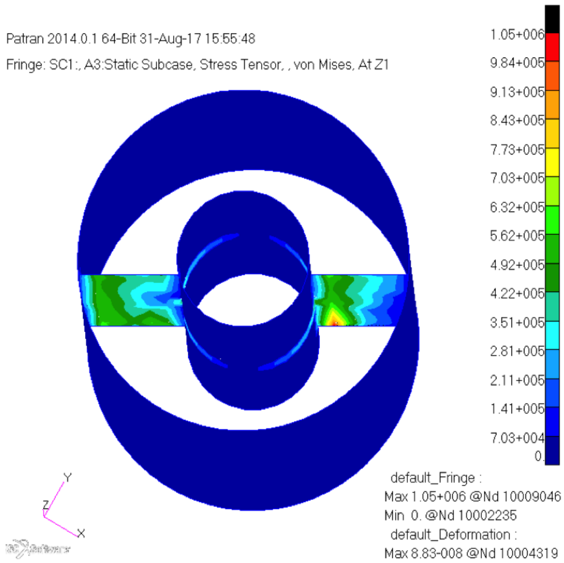


Figure 3.10: Design 1. Acceleration X direction for t=1mm and 1g.

in the x direction approaches the insert.

		CASES			
		Thermal	Acc z	Acc x	Acc y
L1	t = 1mm	4,21	6,45	8,11	7,11
	t = 2mm	5,48	2,40	2,55	2,31
	t = 3mm	5,85	1,34	1,26	1,18
L2	t = 1mm	3,78	6,85	7,56	6,88
	t = 2mm	4,72	2,41	2,47	2,25
	t = 3mm	5,30	1,38	1,27	1,18
L3	t = 1mm	3,62	7,33	7,54	7,21
	t = 2mm	4,14	2,52	2,57	2,37
	t = 3mm	5,65	1,46	1,42	1,35

Table 3.5: Maximum Stresses Design 2 [MPa] for 1K (Thermal Load Case) and 1g (Acceleration Load Case).

Table 3.5 shows the results of the simulations of Design 2 for the three different lengths and three different thicknesses for each length. As with the previous results and at its same conditions, each column applies for a different load case.

Again, the thermal stress increases with thickness for every length, although the variations are seen to be in a bigger proportion than in Design 1 as the compliant section comprises almost the whole body of this design. A trend regarding thermal stress and thickness can definitely be outlined.

As the length of the design grows, it is seen how the value for the thermal stresses for each thickness decreases. This is expected as the loads applied remain the same, while the structure that supports them is wider in length and overcomes less deflection.

The stresses due to acceleration behave in the opposite way, as happens in Design 1. The values of the stresses decrease as stiffness increases with thickness for every length. A trend regarding thickness and acceleration stres can be perceived. This variation is very significant, going from 8,11MPa down to 1,26MPa when the thickness is increased by 2mm.

The stresses for the acceleration load case behave irregularly for the same thickness when the length is increased and do not seem to follow any particular trend. The influence of many variables may be the cause for this, but still the variations are not very significant and is controllable.

For almost every result, the stress values are below the Tensile Yield Strength of 503MPa for 100K and 30g, but do not fit the specifications for the thermal stress when the safety factor of 1,5 is applied. Still, the values for lengths 2 and 3 with thickness of 1mm are close enough to the maximum stress and a value within the requirements could be found if the thickness was slightly lowered or the length increased. It should be taken into account, though, the amount in which the stresses varie with the parameter change, especially with the thickness in the acceleration case, as the limit is set on 11,2MPa and a big decrease in thickness may disable the design from working because of the high stresses in the acceleration case.

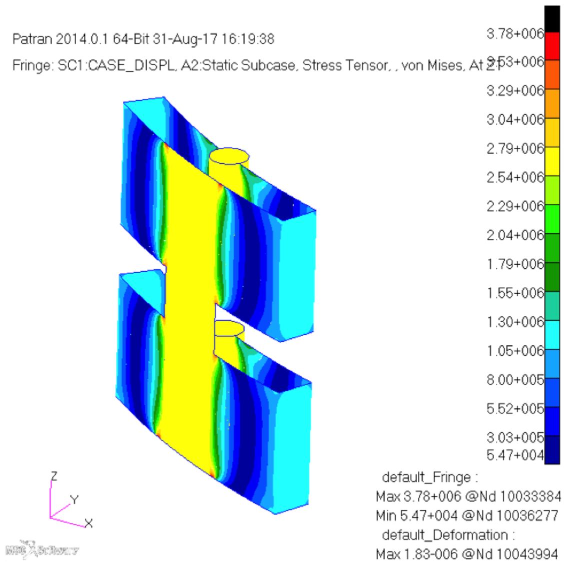


Figure 3.11: Design 2. Thermal stresses for L2, t=1mm and 1K.

The stresses for Design 2 are found in Figure 3.11. For the thermal case, max-

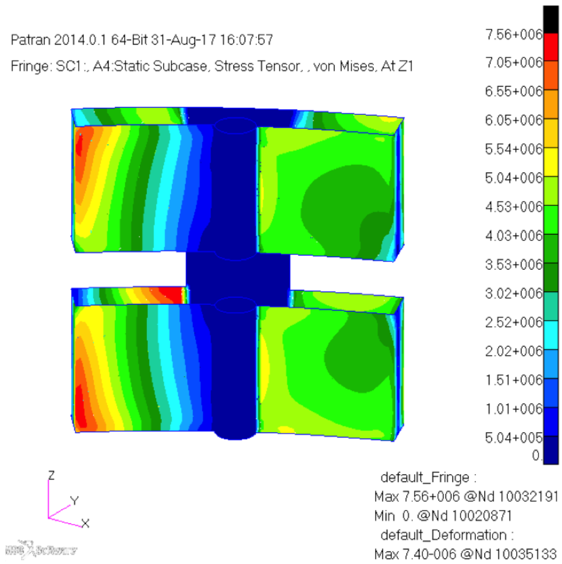


Figure 3.12: Design 2. Acceleration X direction for L2, t=1mm and 1g.

imum stresses can be found between the fixed and the compliant parts of the insert in its outer-most parts zones. In Figure 3.12 it is seen that the maximum stresses are found where the angle of the insert with the x direction is maximum.

		CASES			
		Thermal	Acc z	Acc x	Acc y
L1	t = 1mm	11,5	2,80	5,90	5,28
	t = 2mm	10,5	1,45	2,12	1,85
	t = 3mm	9,04	0,843	1,19	1,13
L2	t = 1mm	4,72	2,25	7,80	7,22
	t = 2mm	6,00	1,63	2,66	2,40
	t = 3mm	5,89	1,10	1,46	1,32
L3	t = 1mm	3,60	2,28	7,91	7,54
	t = 2mm	4,81	1,56	2,76	2,59
	t = 3mm	4,97	1,14	1,53	1,42

Table 3.6: Maximum Stresses Design 3 [MPa] for 1K (Thermal Load Case) and 1g (Acceleration Load Case).

Table 3.6 shows the results of the simulations of Design 2 for the three different lengths and three different thicknesses each in the same way as with the previous design.

For length 1, the results appear to be completely unexpected, as contrary to all the other solutions, for a bigger thicknesses, the values for the stresses in the thermal case decrease. The same happens for length 2, as there is not a consistent trend in the development of the values. For length 3, though, stresses do increase with thickness.

The trend for which the stresses due to temperature decrease with length size is again consistent for this case, but for the acceleration case, the variation is still irregular. It can be confirmed, then, that there is a pattern regarding length and thermal stress.

The values for the stresses in the acceleration load case are seen to be lower in general for Design 3 than for Design 2. However, within the same length, stress decreases with thickness as happened with Design 2, although there is a smaller difference between values of similar gap length to those in Design 2.

For this design, length 3 with t=1mm is the one closest to solving the problem, so by reducing thickness or making the part even wider, the stress requirements could be met.

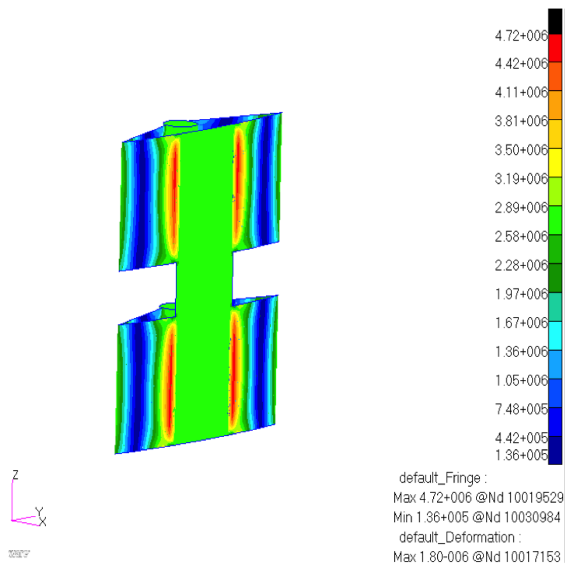


Figure 3.13: Design 1. Thermal stresses for L2, t=1mm and 1K.

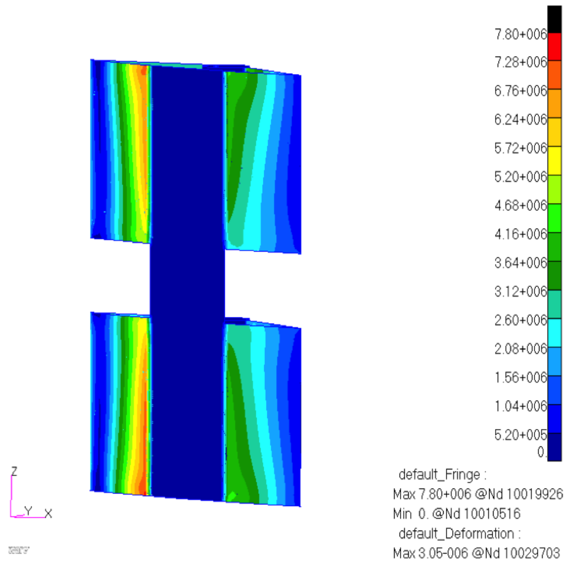


Figure 3.14: Design 1. Acceleration X direction for L2, t=1mm and 1g.

The maximum stresses for Design 3 are seen to be found in a similar locations to Design 2. For the thermal case, stresses are high where the fixed part meets the compliant part although the high stress distribution is distributed throughout all the joint. For the acceleration case the behaviour is again similar to the previous design, where the maximum stresses are found where a higher angle with the x axis is found.

Chapter 4

Results

The data obtained in the previous chapter will be processed down below. According to the trends obtained, the length and thickness will be again modified in order to obtain solutions within the requirements.

4.1 Possible Solutions

From the results in the previous chapter it can be extracted that there are no problems for meeting the requirements for the stress in the acceleration case; the difficulties are found mostly in the thermal case. The solutions that present the most approximate values to 335MPa are Design 2 for lengths 2 and 3 with thickness 1, and Design 3 with length 3 also with thickness 1. These values are collected in Table 4.1, where D stands for Design.

Although the values for the stresses in the thermal load case would exceed the value of the maximum stress with the applied safety factor of 1,5, parameters could be again changed to meet specifications, following the trends obtained from the variation of t and l. Provided that a specific length or thickness is desired, the other parameter can be adjusted to obtain the required the stresses.

Model	Thermal	Acc z	Acc x	Acc y
D2, L2, t=1mm	3,78	6,85	7,56	6,88
D2, L3, t=1mm	3,62	7,33	7,54	7,21
D3, L3, t=1mm	3,60	2,28	7,91	7,54

Table 4.1: Most approximate solutions. Stresses in MPa

4.2 Extracted Trends

There are three different trends extracted from the results. For the three models cited in the previous section, a rough graphic idea of the trends are plotted below. These are not intended to show an exact model of the trends' behaviour or picture as real the stress values for the intermediate thicknesses or lengths.

(1) Stresses due to temperature generally increase with thickness

(2) Stresses due to acceleration decrease with thickness

(3) Stresses due to temperature decrease with length

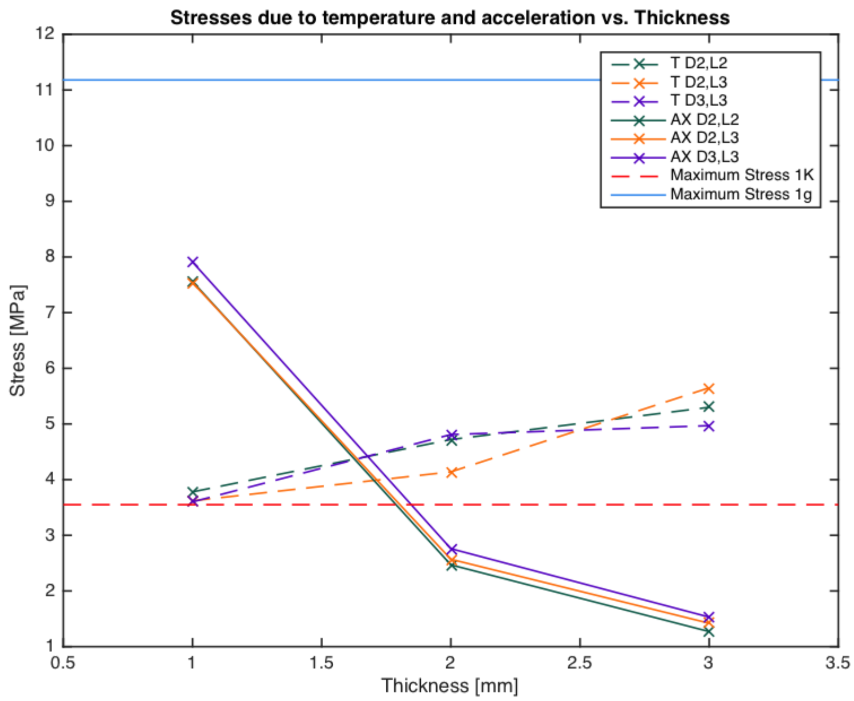


Figure 4.1: Stresses due to temperature vs. Thickness

Figure 4.1 contrasts behaviours (1) and (2). The horizontal lines at 3,35MPa and 11,2MPa show the limits for the thermal case and the acceleration case respectively. The stresses of the solutions that meet the requirements should be under these values.

Behaviour (1) can be explained in the relationship of the thickness with the moment of inertia in the mechanical equations and (2) is due to the increase in stiffness in the parts.

It is seen that with a slight decrease in thickness for the abovementioned design solutions, the designs would completely work. For the acceleration case it may be thought that there is space for big decreases in thickness before reaching the maximum acceletion stress value. It should be taken into account, though, as mentioned in Section 3.4, that the variation in stresses from one thickness to another become more bigger as thickness decreases.

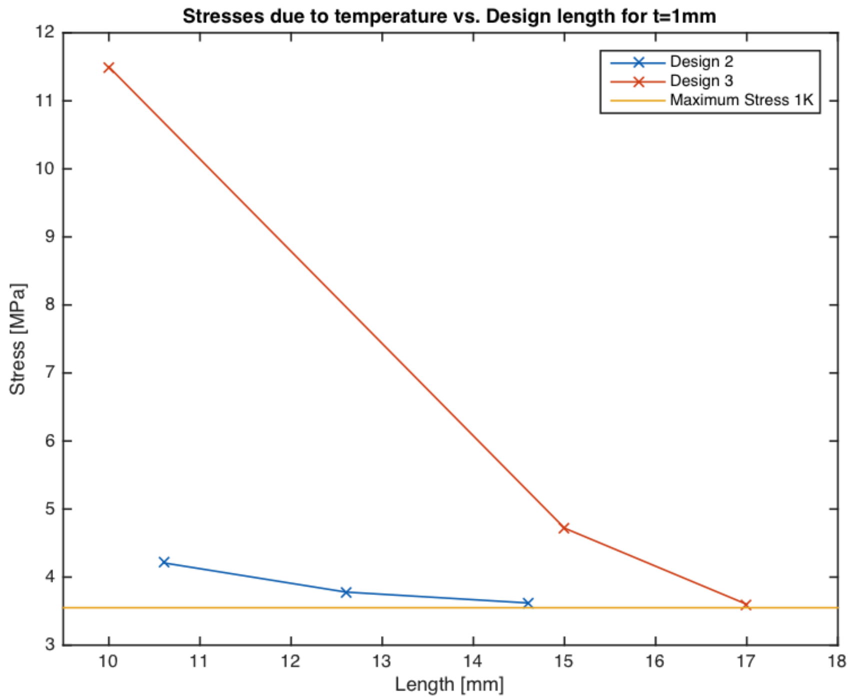


Figure 4.2: Stresses due to acceleration vs. Thickness

The behaviour pictured in Figure (3) can be explained with the increase in length with the same applied load, resulting in less displacement and therefore, less stress. For the acceleration case no results were plotted as these are taken for t=1mm and the stresses due to acceleration were in no case surpassed or were in danger of doing so for this thickness in any of the designs. Plus, no trend was actually detected.

With a desired thickness of t=1mm, a solution within the requirement could be found if the length was increased even more.

4.3 Solutions

Following the trends, a new set of simulations have been done varying the parameters in order to achieve a solution within the specifications. The models that have been put at test again are the ones mentioned in Sections 4.1 and 4.2.

		CASES			
		Thermal	Acc z	Acc x	Acc y
D2 L2	t=0,7mm	3,59	12,8	9,50	13,7
	t=0,8mm	3,66	10,1	7,34	10,6
D2 L3	t=0,5mm	3,41	24,2	29,2	28,2
	t=0,8mm	3,54	10,7	11,5	11,0
D3 L3	t=0,8mm	3,20	2,82	11,2	11,0
	t=0,9mm	3,40	2,51	9,36	8,92

Table 4.2: Thickness variation

Table 4.2, shows the resulting stresses when varying the thickness for the three models. The trends in Section 4.2 still follow. The objective of this extended study was to find a suitable solution for the problem addressed, where all stress values stood below 335MPa.

For each model a different thickness was tested, taking into account the results from Section 4.1 and the trends in Section 4.2. Also, depending on the stress values obtained for the first trial, a second thickness value was tried out.

For the Design 2, L2, it was initially simulated for t=0,8mm. The values of the thermal stress were still too high and the acceleration stresses were seen to increase considerably. The simulation was then repeated for t=0,7mm to try to find a lower thermal stress but the decrease was not enough, but the acceleration stresses grew too much, surpassing 11,2MPa.

Length 3 from Design 2 was initially tested for t=0,8mm. The acceleration stresses for this case were on the limit, while the thermal stress needed to decrease way more for the solution to be suitable. When tried for t=5mm, the thermal stress still did not meet the limits and the acceleration stresses were way above the maximum required.

Finally, length 3 of Desgin 3 for t=0,8mm did stay within the requirements as the maximum stress for acceleration is on the stress limit and the thermal stress is below the limit. Yet, t=0,9mm was tested and although the thermal stress is slightly above the requirements, the stresses for accelerations in x and y directions decreased significantly. An intermediate thickness could achieve for this model an optimum value with minimum acceleration stresses and suitable thermal stress.

To sum up, Design 3 with L3 and t=0,8mm would suit as solution for the problem.

		CASES			
		Thermal	Acc z	Acc x	Acc y
D2	L4, t=1mm	3,52	7,86	7,54	7,46
D3	L4, t=1mm	4,06	3,82	7,05	6,71
	L5, t=1mm	4,45	3,73	7,15	6,98

Table 4.3: Length variation

D2	L4	16,6 mm
D3	L4	18,0 mm
	L5	19,0 mm

Table 4.4: New lengths

For a fixed thickness of 1mm, the length in the designs was increased. The results are shown in Table 4.3 and the new lengths are found in Table 4.4. For Design 2, the trend was again followed, but for Design 3 the solutions were unexpected as the thermal stress increased instead of decrease.

As the acceleration stress values did not increase much, it is assumed that a further increase in the length of the part could prove to work for the problem addressed. Although the increase in length for Design 2 did work as expected and the search of the solution seems to be on track, the study was not continued and no solution was found for a fixed thickness of 1mm.

Chapter 5

Conclusion and Prospects

At the beginning of the thesis, a series of objectives were stated. These goals were to be met, if the project was to be considered successful. In this chapter it will be analyzed whether or not they have been accomplished and if there is any way to further develop this project.

In the first place, it was to be analyzed if it would be suitable to use a compliant structure to solve the problem addressed. This was the first thing to be done, mostly as a personal introduction to the subject in case. After researching on compliant mechanisms, their advantages and possible challenges made them suitable to for the problem solution, as the designs could be adapted to the existing satellite geometry and could potentially allow for an isostatic mounting of the parts.

Chapter 3 collects the work behind the points that follow: the creation of different suitable designs, the study of their feasibility, their development and the presentation of the data of the stresses.

Although various designs were considered, the ones introduced in Chapter 3 had the major probability to be successful. Those were the ones produced and optimized in CAD and processed in the FEM software. A parametric study was carried out by varying the length and thickness of the compliant parts, in order to obtain a trend in the behaviour of the parts.

As pointed out in the last chapter, most of the difficulties were present in lowering the stresses due to temperature. There were designs, such as the first one, that were not worthwile to be developed because any change would not make much difference, as there was no space for such big design changes in the compliant part. The answer for a suitable solution had to be a balance between the change in length and thickness.

The main conclusions extracted from the results were also pointed out in the previous chapter. Three trends were obtained that linked the behaviour of the thermal stresses with thickness and length and acceleration stress with length. It was basically derived that what was needed was a longer or thinner design to lower the stresses due to temperature. The simulations of Design 2 with lengths 2 and 3 and Design 3 with length 3 were close enough to the maximum stress value, so with a slight decrease in thickness or increase in length they would definitely be found within the requirements. Following these trends, a suitable solution was given for Design 3 with length 3 and thickness of 0,8mm and a close one was found for 0,9mm. Also, an insight on how to become closer to a good solution with a fixed thickness of 1mm was given.

This study shall act as a guide on upcoming decisions with respect to the final design of the inserts, also depending on the final design of the overall space system. The main focus was of course to try to achieve a solution within the stress limits when varying the parameters.

An optimization in the values of the stress levels was clearly found. Not only a design that adjusted better for the requirements with respect to the original part in Design 1 was found; but also, within designs, a way to reduce the stresses was found.

Given the great difference that the variation of parameters made, a possible development for the project would be to actually calculate an optimum value for the parameters by deepening the study of the trend and calculating for more thicknesses and lengths or even finding a function that described the trend of these parameters. It would also be useful to find the limits in which length and thickness could be varied to always obtain useful solutions. Another option would be varying other parameters, such as the length of the gap in the inserts from Designs 2 and 3 or the thickness of the non-compliant parts of the bodies. This would definitely make a difference in stiffness but it would probably also decrease the thermal stresses and a balance may be again found. As well as the possibility of finding out new designs, there could be another option of expanding the project by finding any other materials that would work for these designs, which should still be suitable for the aerospace industry and adjusting to a possible budget.

Finally, the prototyping and real-life testing of the designs would also be a possible expansion of the project. This would support the results from the simulations made in the FEM software and show whether it makes a real difference doing the simulations with a simplified version of the solid inserts.

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