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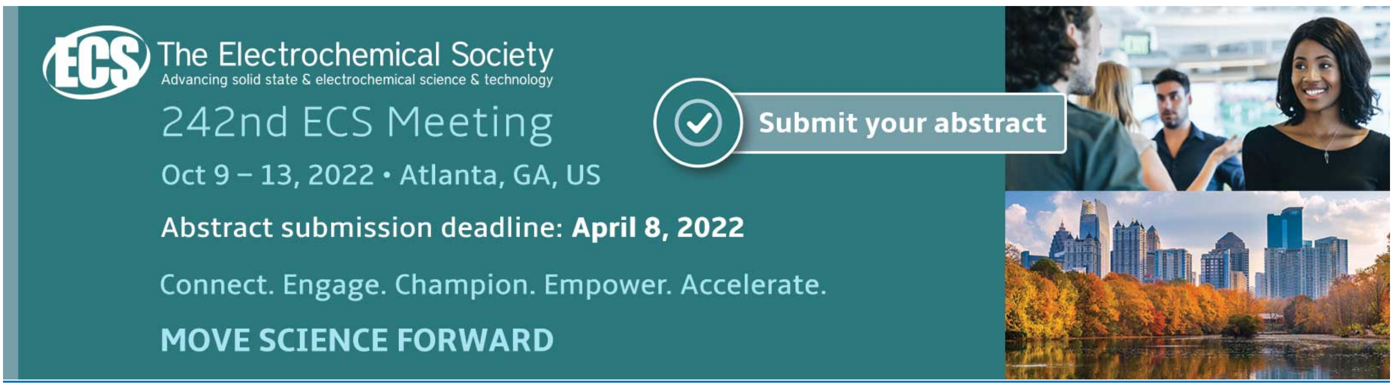

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Structural design of the optical bench and enclosure for MAORY, adaptive optics module for the ELT

A Andolfo[1], A Belperio[2], V Cianniello[1,2], C Eredia[2], V De Caprio[2] and E Armentani[1]

[1] University of Naples Federico II, Naples, Italy
[2] Istituto Nazionale di Astrofisica (INAF) – Osservatorio Astronomico di Capodimonte, Naples, Italy

E-mail: enrico.armentani@unina.it

Abstract. This paper outlines an overview of the mechanical design of the optical bench and the enclosure for MAORY (Multi-conjugate Adaptive Optics RelaY) for the Extremely Large Telescope. MAORY will enable high-angular resolution observations in the near infrared by employing real-time compensation of the wave-front distortions due to atmospheric turbulence and other disturbances on the telescope. The main purpose of the optical bench is to provide support to the opto-mechanical mountings and subsystems that will be integrated on it. The design philosophy behind the proposed architecture is a truss spatial structure with the aim of optimizing the mass of the Main Structure. The enclosure has to protect the optomechanical elements and to achieve a uniform temperature distribution in its internal environment. The mechanical design of the bench and the enclosure was supported by a set of structural FE analyses, to verify the design compliance with ESO (European Southern Observatory) requirements.

1. Introduction

The Extremely Large Telescope (ELT) is an optical/near-infrared telescope with a 39-meter primary mirror under construction at an altitude of about 3000m in the Chilean Atacama Desert. The optical design (figure 1) is based on a five-mirror scheme and incorporates adaptive optics mirrors. The primary mirror consists of 798 segments, each 1.4 meters wide [1]. The ELT will be the largest ground based optical telescope of the world and will require, to exploit its potentialities regarding the angular resolution, the wave front correction of the star images caused by the atmospheric turbulence [2].

MAORY is the multi-adaptive optics module for the ELT first light. It will provide a corrected field-of-view to its client instrument MICADO, a near-infrared camera and spectrograph. A second port will be available for a second instrument still to be determined.

MAORY is an adaptive optics (AO) module offering two AO modes: multi-conjugate adaptive optics (MCAO) and single-conjugate adaptive optics (SCAO). The MCAO mode is required to achieve uniform adaptive optics compensation over the full MICADO field of view; the SCAO mode is required for peak performance over a smaller field of view [3], [4].

The MAORY instrument is designed and built by a Consortium including the Istituto Nazionale di Astrofisica (INAF), the Institut de Planétologie et d'Astrophysique de Grenoble (IPAG) and the

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National University of Ireland, Galway (NUI Galway). The European Southern Observatory is also involved in the development of the instrument.

The research activity described is the result of the collaboration between the University of Naples Federico II and INAF – Osservatorio Astronomico di Capodimonte (Naples) [5].

Figure 1. The Extremely Large Telescope.

The MAORY instrument project entered its phase B in February 2016. Consolidation of the baseline design is underway, taking into account scientific performance requirements and interface constraints.

In the following paragraphs the structural design of the optical bench and the enclosure will be presented, showing the design constraints and demonstrating compliance with the requirements provided by ESO. The analyses conducted are static and modal, followed by analyses of earthquakes, wind and failure. In addition, an alternative solution to the design of the optical bench has been proposed using an innovative material for astronomical applications.

2. The optical bench design: model description and overall dimensions

The main purpose of the bench is to provide a very stable opto-mechanical reference and a support for all optical and service components weighing on it. It has, therefore, to be very precise in flatness and very stiff in order to minimize load deformation as well. Of course, it must also have natural frequencies decoupled from the telescope; the lowest own frequency must be above 7 Hz.

In order to comply with all the requirements, including the best shape for good accessibility to all opto-mechanical components by operators, the choice was made to design a steel truss spatial structure. This model will be machined in three separate parts and jointed with bolts and reference pins to increase the accuracy of the connection. In order to cover the available three-dimensional space, it has been decided to use the repetition of a pyramidal structure with a rectangular base. The LGS (Laser Guide Star) module has to be positioned at a higher opto-mechanical plane compared to the opto-mechanical plane of main optical path of MAORY, for this reason it has been provided with a support system, which consists of a top interface flange for mounting the LGS module and a hexapod support structure up to the bench.

The height of the bench is 900 mm and the top plane of the bench (the floor) has a distance from the focal plane of 1200mm.

The structure is made of truss-beams: to optimize the ratio between the global stiffness and the global mass of the MAORY bench some circle-shaped pipe profiles were chosen. The circular pipe profiles optimize the inertia and minimize the mass compared to other types of common profiles. To contain the global mass of the system profiles of different overall dimension and thickness were used for the bench structure.

The final layout, considering the different design constraints, will appear as in figure 2.

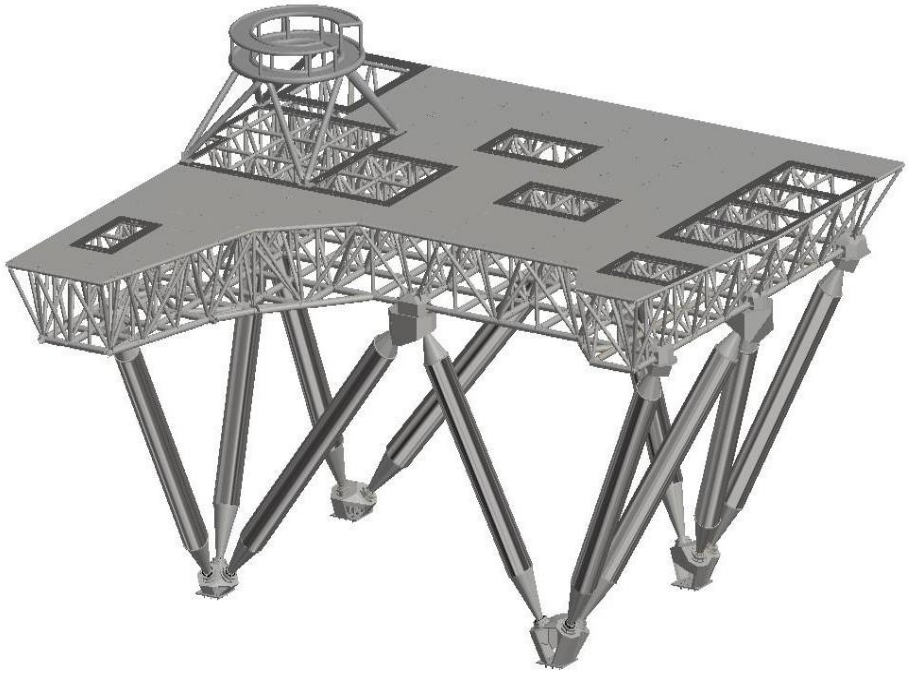

Figure 2. MAORY Optical Bench.

The top of the bench is divided into two parts: the interface surface with the opto-mechanical elements, made of steel with a thickness of 10 mm, necessary to facilitate the assembly of the support structures and to provide appropriate stability to optical elements; honeycomb panels to be used as a walkable floor, with a thickness of 25 mm. This type of panel was chosen considering that access on the bench must be guaranteed to the operators for the assembly and disassembly.

The MAORY environment is thermalized, so to avoid unacceptable temperature variations the floor was passively insulated as well. The lower part of the bench will be covered with a layer of rigid polyiso foam (PIR); the areas in contact to the steel tubulars were insulated with another layer of insulator material: Aerogel. Under each opto-mechanical support, a panel consisting of a 2mm steel sheet and insulating material (PIR) will be inserted.

2.1. MAORY Optical Bench Steel design: FEM analyses

The bench is made of sheet metal and the legs are pipes of structural steel. The CAD model (figure 3) is made of surfaces, 3D lines and points. This model is directly exported from CAD software (Autodesk Inventor®) to the CAE software (Ansys Workbench® release 19.2) [6-9].

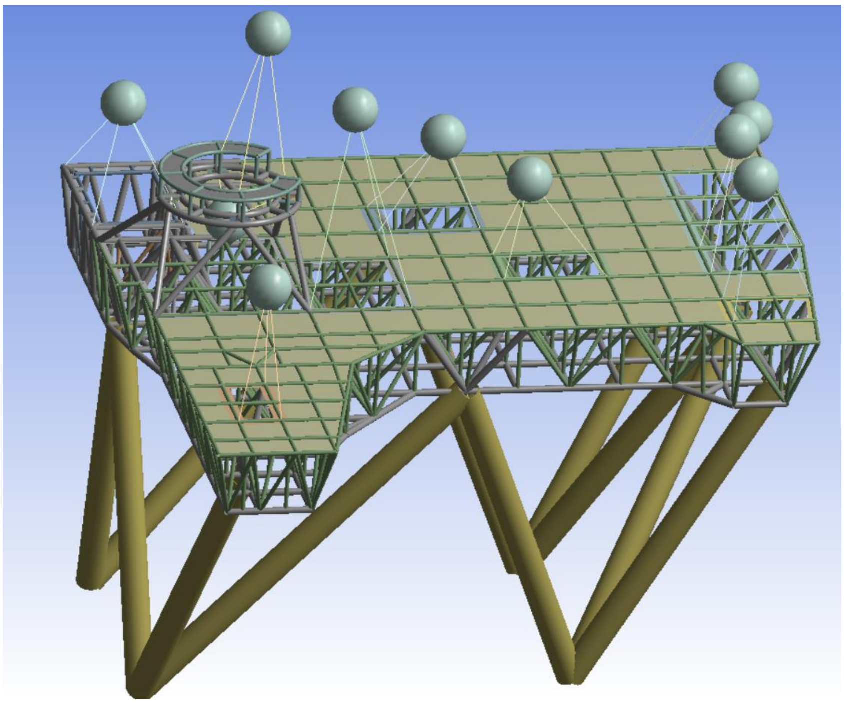

Figure 3. CAD model of MAORY: Steel Design.

Tables 1 and 2 summarize the properties of the materials in object:

Table 1. Steel AISI 4130

Density [kg/mm^3]	7.85*10^{-6}
Young's modulus [MPa]	210000
Poisson's ratio	0.3
Yield strength [MPa]	400
Tensile strength [MPa]	700

Table 2. Honeycomb Aluminum

Equivalent density [kg/mm^3]	0.304
Young's modulus [MPa]	70000
Poisson's ratio	0.3

In order to minimize the overall mass of the system, different profiles for the bench were used. In particular, the bench has the following characteristics:

- bottom profile external diameter and thickness: Ø 80 mm – th. 3 mm (figure 4: n.2);
- base profile external diameter and thickness: Ø 50 mm- th. 3 mm (figure 4: n.3);
- legs external diameter and thickness: Ø 323 mm – th. 10 mm (figure 4: n.4);
- interface plates with opto-mechanical thickness: th. 10 mm;
- honeycomb panels thickness: th. 25mm.

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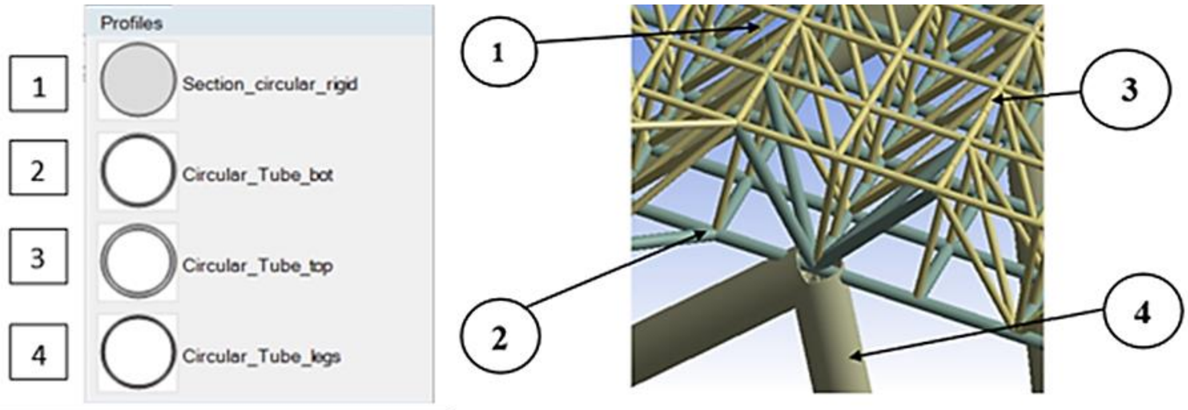
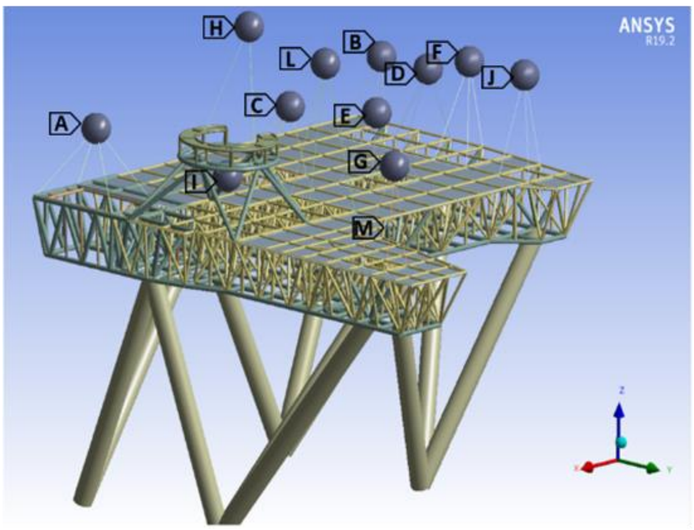


Figure 4. Geometry section.

On the top surface of the bench there are the interface points between the bench and the opto-mechanical mountings. The total mass of each opto-mechanical mounting and sub-assembly has been applied, as nodal weight, in its barycentre. This point is connected to the top surface of the bench using rigid elements. The estimated loads (including contingency) shown in the next figure 5, have been used for the opto-mechanical mountings and sub-assembly in the FE model.

The interface between the legs of the bench and the Nasmyth platform are modelled as nodal joint which allows only the three rotations (three displacements are 0 mm).



		Massa[kg]	Massa con 20% [kg]
A	Elevator1	1250	1500
B	M6	511	613
C	M7	624	749
D	M8	340	408
E	DM9	1000	1200
F	DM10	1000	1200
G	M11	623	749
H	DC+LGSo	860	1032
I	LGS	700	840
J	M12	556	667
L	M13b	606	727
M	Enclosure	2916	3500
	TOT		13185

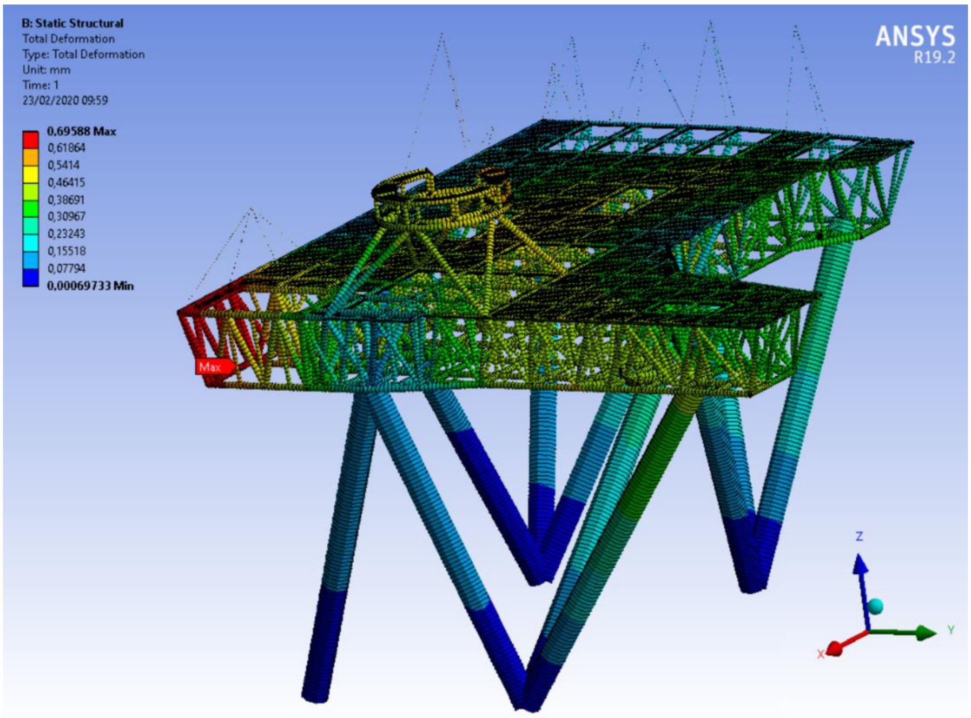
(a) (b)

Figure 5. MAORY Optical Bench with opto-mechanical mountings and sub-assembly units (a) - Loads applied in FEM analysis (b).

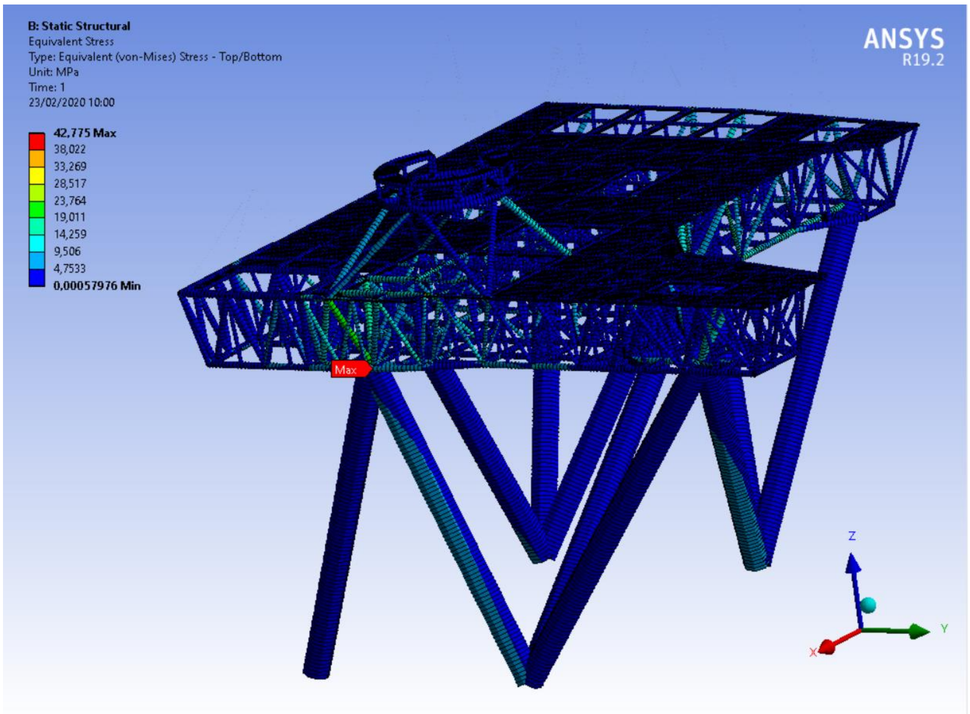
Finite element analyses were carried out once the model was defined: static and modal analysis, earthquake analysis and failure analysis.

Static stress analysis was carried out on the model of the bench with all loads described in the previous paragraph and the results are shown below:

- ***Static analysis***: the maximum displacement is **0.69** mm (see figure 6a);
- ***Static analysis***: maximum Von Mises stress is **42.6** MPa (see figure 6b);
- ***Modal analysis***: first natural frequency is **9.18** Hz (see figure 6c) rather than allowable one (7 Hz).



(a)



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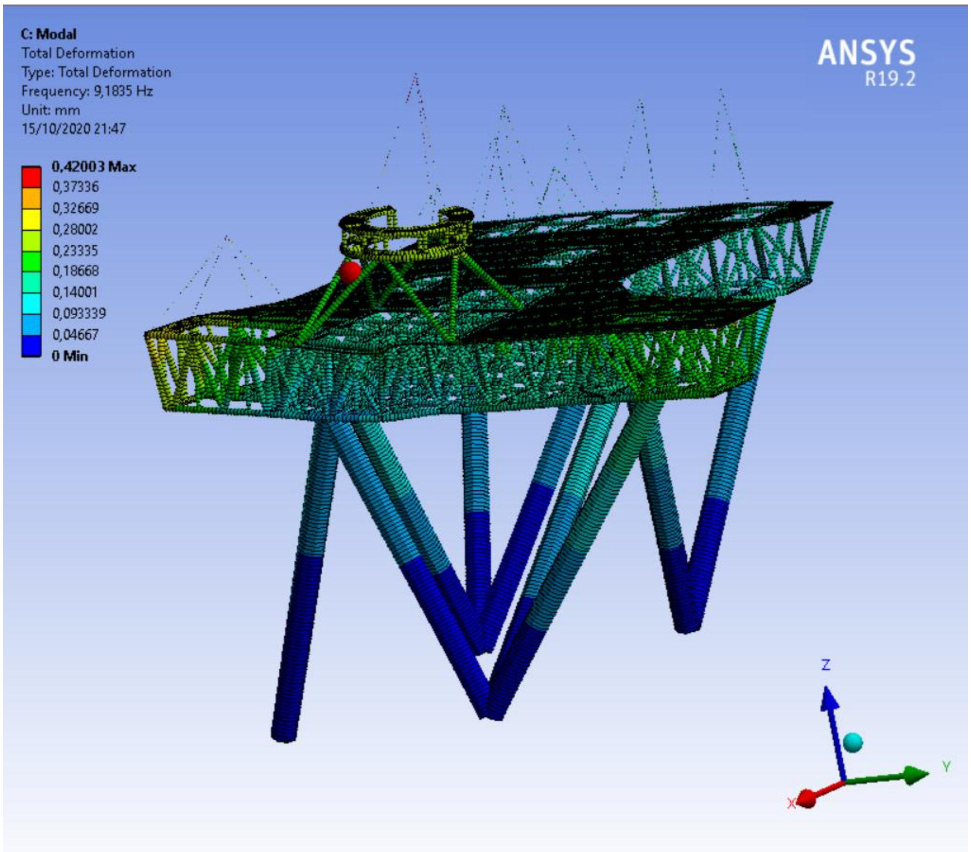


(c)

Figure 6. Static results (a-b) and modal results - first mode (c).

The mass derived from the FE model is **7.9 tons**.

For the ***earthquake analysis*** the case of the Simplified Seismic Analysis Method without response spectrum was considered. The quasi-static seismic acceleration to be applied to the structure in the three directions is obtained by performing a linear combination of the three equations below, obtaining a total of 24 load cases. The following equations can be found in **EN 1998-1 (2004): Eurocode 8**: Design of structures for earthquake resistance – Part 1.

$$\begin{aligned} &\textbf{a)}\ A_{Ed1}=\pm E_{dx}\pm 0.3E_{dy}\pm 0.3E_{dz}\\ &\textbf{b)}\ A_{Ed2}=\pm 0.3E_{dx}\pm E_{dy}\pm 0.3E_{dz}\\ &\textbf{c)}\ A_{Ed3}=\pm 0.3E_{dx}\pm 0.3E_{dy}\pm E_{dz} \end{aligned} \tag{1}$$

The maximum reactions on the Nasmyth flanges are shown in table 3.

Table 3. Reaction forces of earthquake analyses (Steel design)

	FORCE REACTION X [N]	FORCE REACTION Y [N]	FORCE REACTION Z [N]
MAX	308880.31	231726.66	800889.26
MIN	-304641.19	-233289.32	-947667.89

The specifications are met considering that the values imposed by ESO are:

- **500000 N** in directions of X_{AZ} and Y_{AZ};
- **1000000 N** of traction and **1250000 N** of compression in the Z_{AZ} direction.

The ***failure analysis*** is a static analysis, in which the displacement of the centre of gravity of the opto-mechanics under the action of a failure imposed on the structure is evaluated in output. The analyses carried out are four and for each of them a setting of imposed displacements is associated to a single point of constraint to the Nasmyth.

For the first analysis (at Node_1) the following displacement setting is imposed (figure 7):

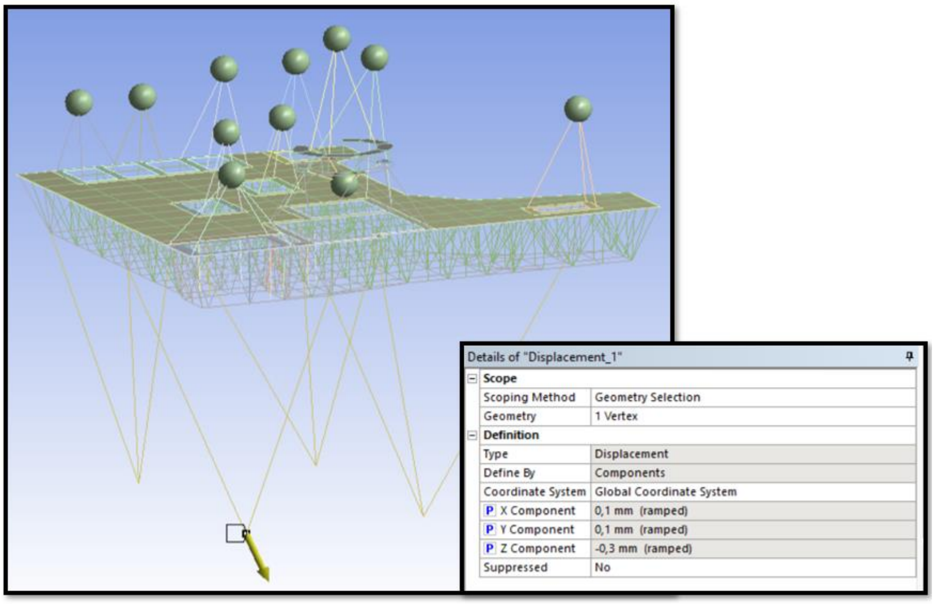


Figure 7. Setting displacement Node_1

While for the remaining three points of constraint to the Nasmyth platform it is required that the movements in the three directions are zero and the rotations are free (figure 8).

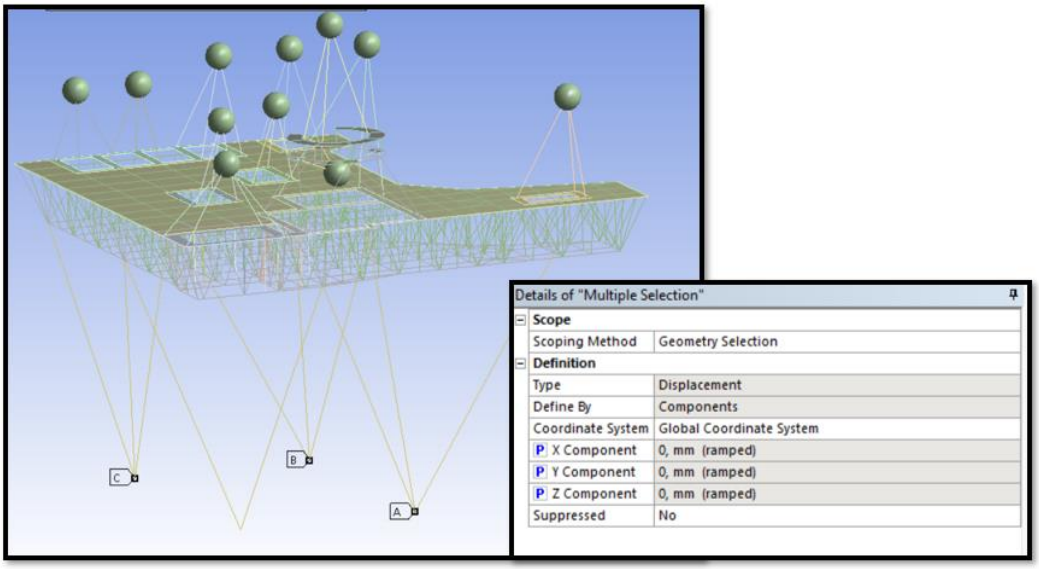


Figure 8. Setting displacement Node_2, Node_3, Node_4

The output for each centre of gravity will include both flexible rotation and deformations in the three directions of the optomechanical centre of gravity. For example, the output for the M6 mirror is as follows (figure 9):

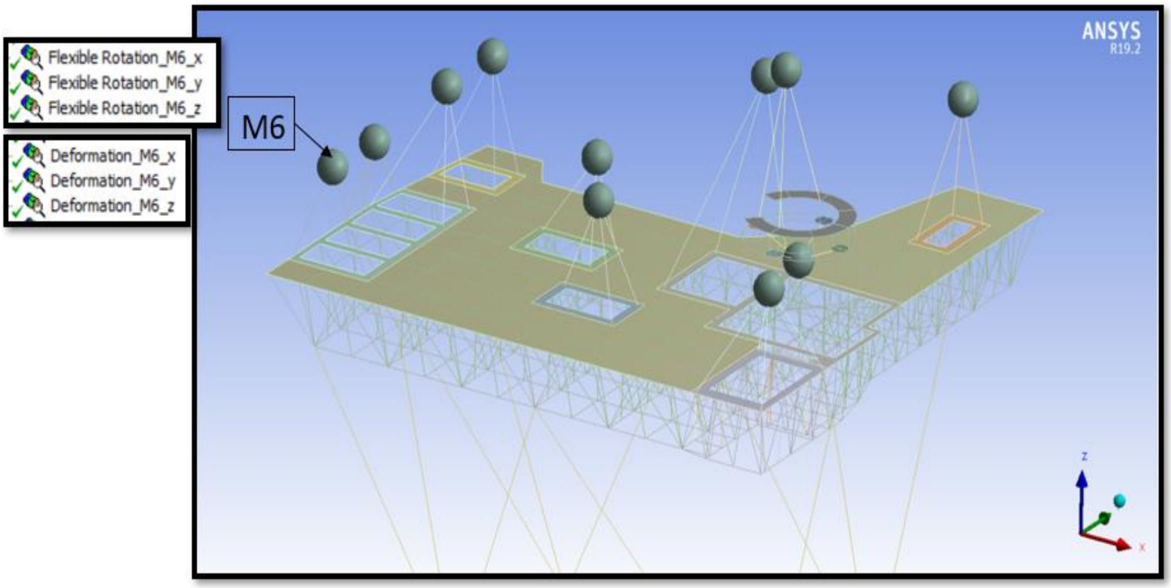


Figure 9. Output Failure analysis M6

The displacements of the optomechanics barycentre are not all compliant with the specifications, as in the case of M11 (table 4). To overcome this problem, the optical elements will be equipped with an electromechanical active control system that will allow them to return to the nominal position.

Table 4. Results of failure analysis (Steel design)

Flexible Rotation_M11_x (arcsec)	13.90	-13.24	-4.93	4.27
Flexible Rotation_M11_y (arcsec)	2.29	**8.26**	**-9.22**	-1.33
Flexible Rotation_M11_z (arcsec)	-1.56	**3.92**	**-6.90**	**4.54**
Deformation_M11_x (mm)	0.20	**0.20**	-0.08	-0.21
Deformation_M11_y (mm)	-0.32	**0.40**	**0.27**	-0.24
Deformation_M11_z (mm)	0.13	-0.39	-0.17	0.14

2.2. MAORY Optical Bench CFPR design: FEM analysis

The steel bench has a total weight of 7.9 tons. Considering that the total mass cannot exceed 8 tons, an alternative solution has been proposed. The carbon fibre reinforced polymer high strength-to-weight ratio and good stiffness make it the perfect candidate for a lightweight structure, although it can be relatively expensive (figure 10). CFRP tubes are not weldable and drilling causes damage, so they need metal inserts to be spliced. For this reason, joints have been designed to bond carbon tubing with steel. The two parts will be held together by a high-performance glue.

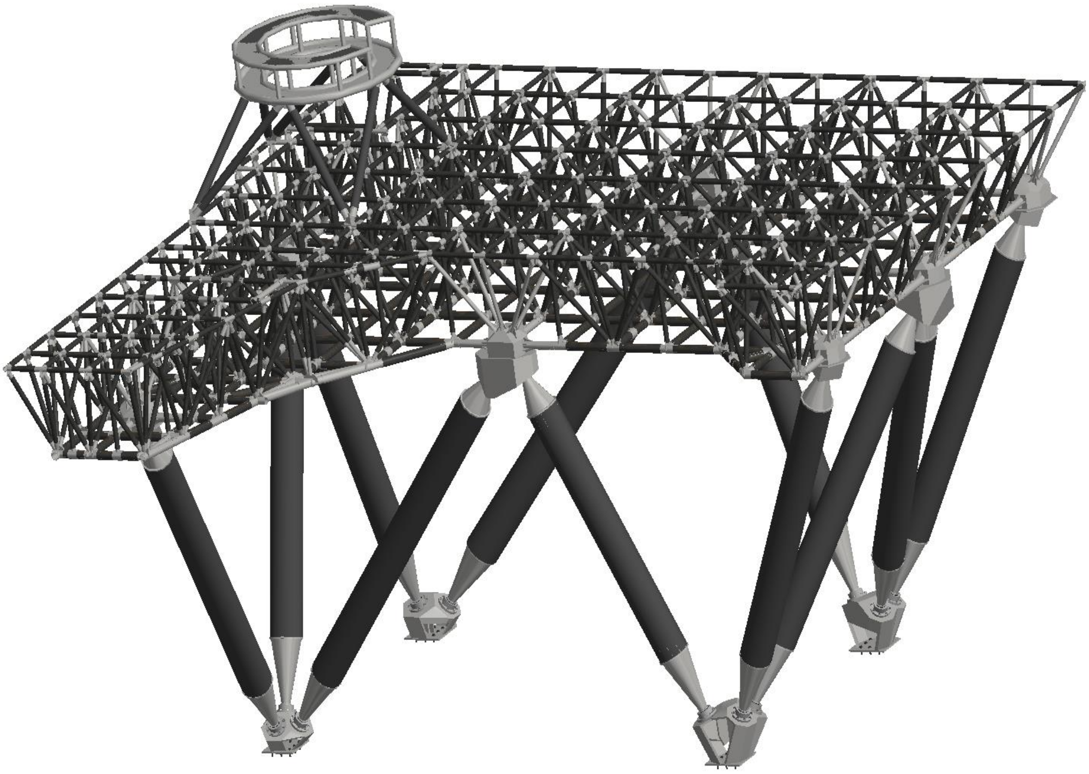


Figure 10. MAORY Optical Bench: CFRP design.

Table 5 summarizes the properties of the materials in object:

Table 5. Epoxy Carbon Woven Pre-preg

Density [kg/m^3]	1480
Coefficient of Thermal Expansion X direction [1/°C]	$2.5*10^{-6}$
Coefficient of Thermal Expansion Y direction [1/°C]	$2.5*10^{-6}$
Coefficient of Thermal Expansion Z direction [1/°C]	$1*10^{-5}$
Young’s modulus X direction [MPa]	91820
Young’s modulus Y direction [MPa]	91820
Young’s modulus Z direction [MPa]	9000
Poisson’s ratio XY	0.05
Poisson’s ratio YZ	0.3
Poisson’s ratio XZ	0.3
Shear Modulus XY [MPa]	19500
Shear Modulus YZ [MPa]	3000
Shear Modulus XZ [MPa]	3000
Tensile stress X direction [MPa]	829
Tensile stress Y direction [MPa]	829
Tensile stress Z direction [MPa]	50

Compressive stress X direction [MPa]	-439
Compressive stress Y direction [MPa]	-439
Compressive stress Z direction [MPa]	-140
Shear stress XY [MPa]	120
Shear stress YZ [MPa]	50
Shear stress XZ [MPa]	50

Finite element analyses were conducted using a model consisting of beams. In order to correctly associate the material, the beams have been divided in correspondence with the joints. The definition of the sections is as follows: for the beams representing the steel joint the sections are the same as seen in paragraph 2.1; to ensure performance comparable to that of the steel bench, the beams corresponding to the carbon fibre tubulars have the following sections:

- base profile external diameter and thickness: Ø $60\ mm$- th. $8\ mm$ (figure 11: n.1);
- bottom profile external diameter and thickness: Ø $90\ mm$ – th. $8\ mm$ (figure 11: n.2);
- legs external diameter and thickness: Ø $337.8\ mm$ – th. $17{,}4\ mm$ (figure 11: n.3).

The carbon fibre bench has been loaded in the same way as the steel bench so that a comparison can be made. The payload setting has been presented in paragraph 2.1.

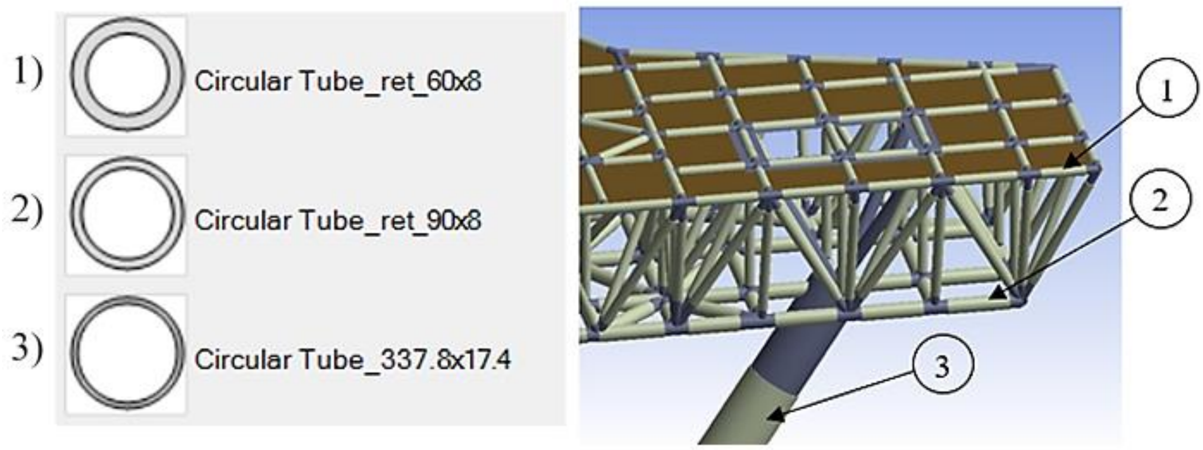
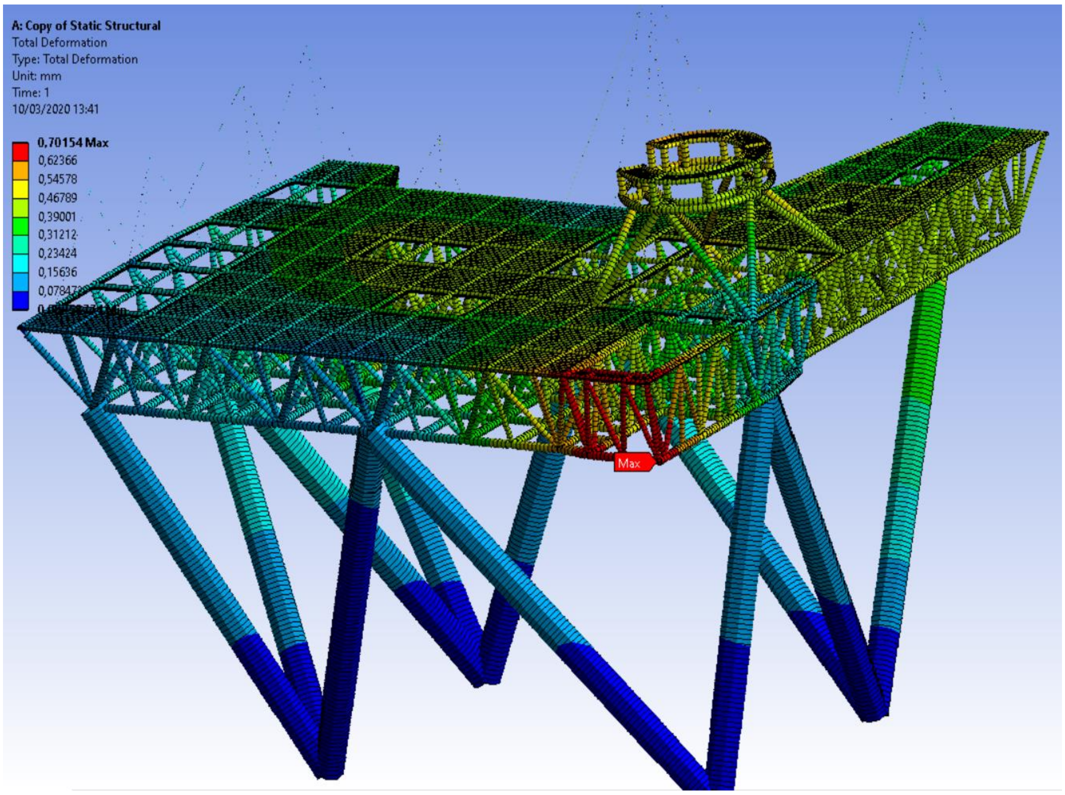


Figure 11. Geometry section (CFRP design).

Static stress analysis was carried out on the model of the bench with all loads described in the previous paragraph and the results are shown below:

- ***Static analysis***: the maximum displacement is **0.70** mm (see figure 12a);
- ***Static analysis***: maximum Von Mises stress is **39.6** MPa (see figure 12b);
- ***Modal analysis***: first natural frequency is **8.63** Hz (see figure 12c) rather than allowable one (7 Hz).

The mass derived from the FE model is **5 tons**.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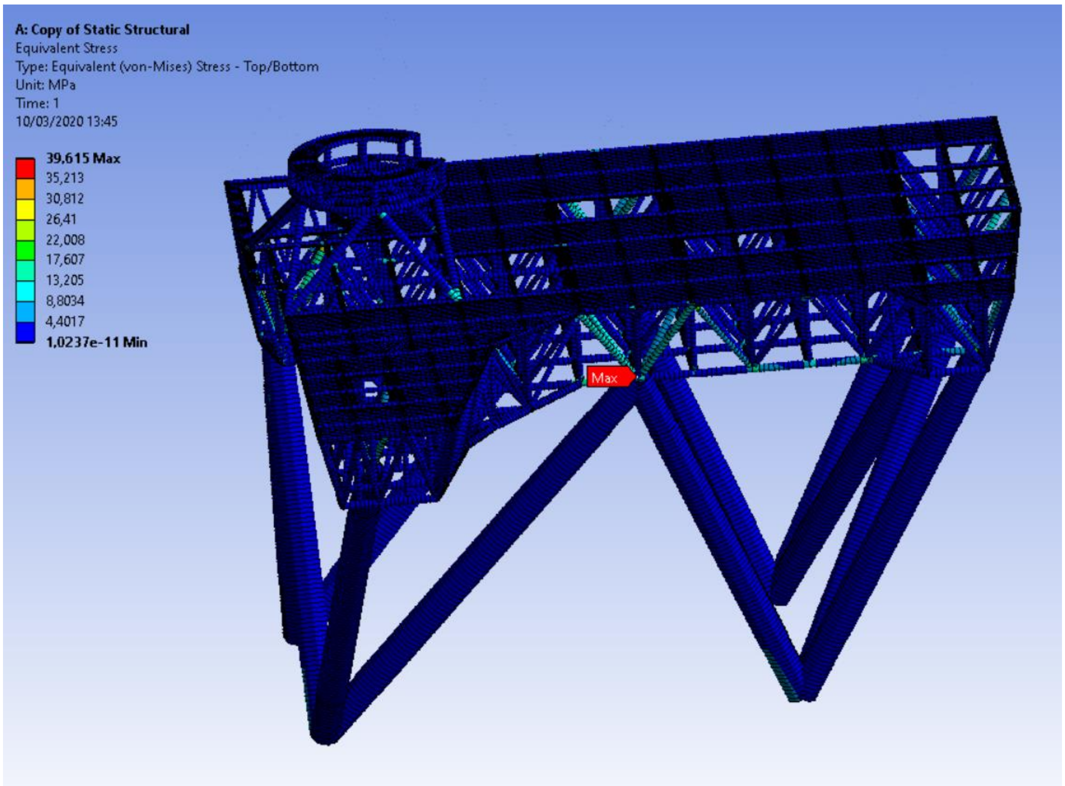
(a)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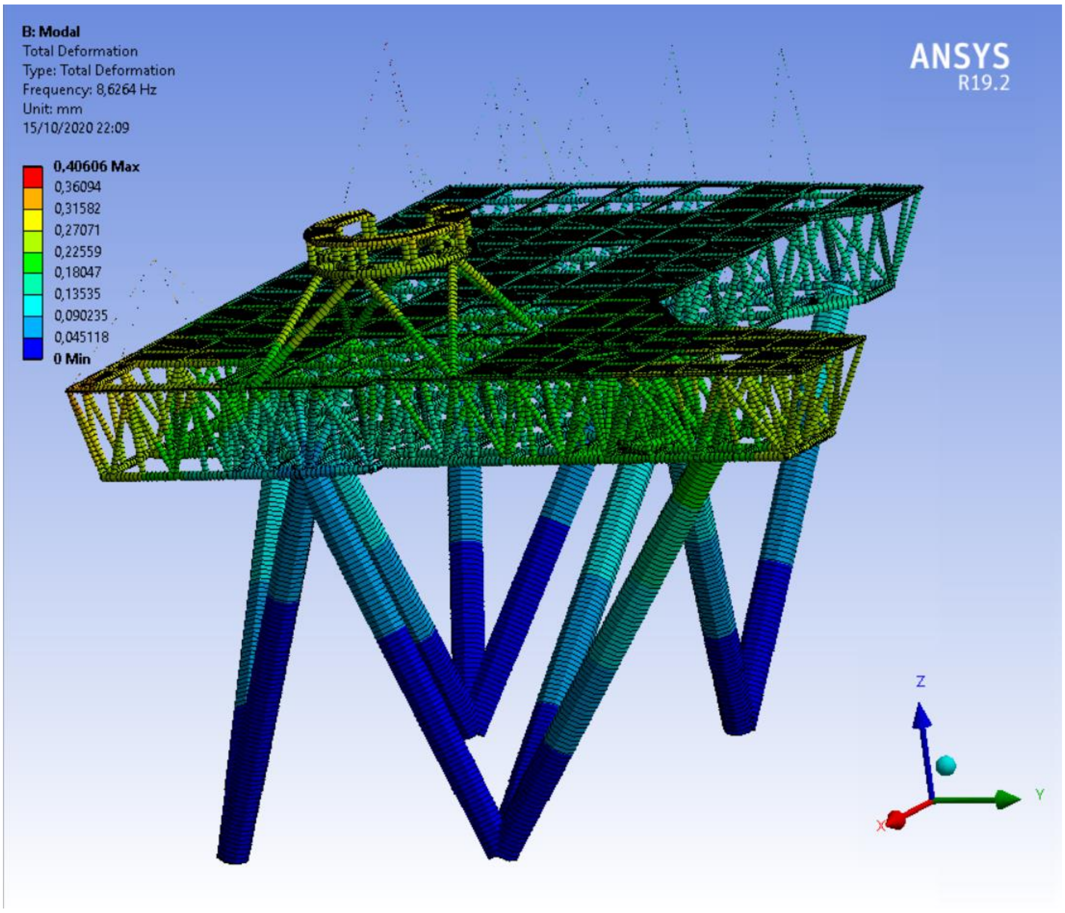
Max

(b)



(c)

Figure 12. Static results (a-b) and modal results - first mode (c) (CFRP design).

The results (total displacement of the structure and maximum equivalent stress of Von Mises) of the static analysis in the case of the steel and carbon fibre bench were compared, as shown below.

$$\%\varepsilon_{def} = \left(\frac{0.70 - 0.69}{0.70}\right) * 100 = 1.4\%$$

$$\%\varepsilon_{Eq} = \left(\frac{42.77 - 39.61}{42.77}\right) * 100 = 7.38\% \quad (2)$$

The earthquake and failure analyses were carried out as in the case of the steel bench (table 6 and table 7). The earthquake analyses were performed using equations (1).

Table 6. Reaction forces of earthquake analyses (CFRP design)

	FORCE REACTION X [N]	**FORCE REACTION Y** [N]	**FORCE REACTION Z** [N]
MAX	281433.65	190483.47	806371.77
MIN	-274021.61	-191150.33	-937546.42

Table 7. Results of failure (CFRP design)

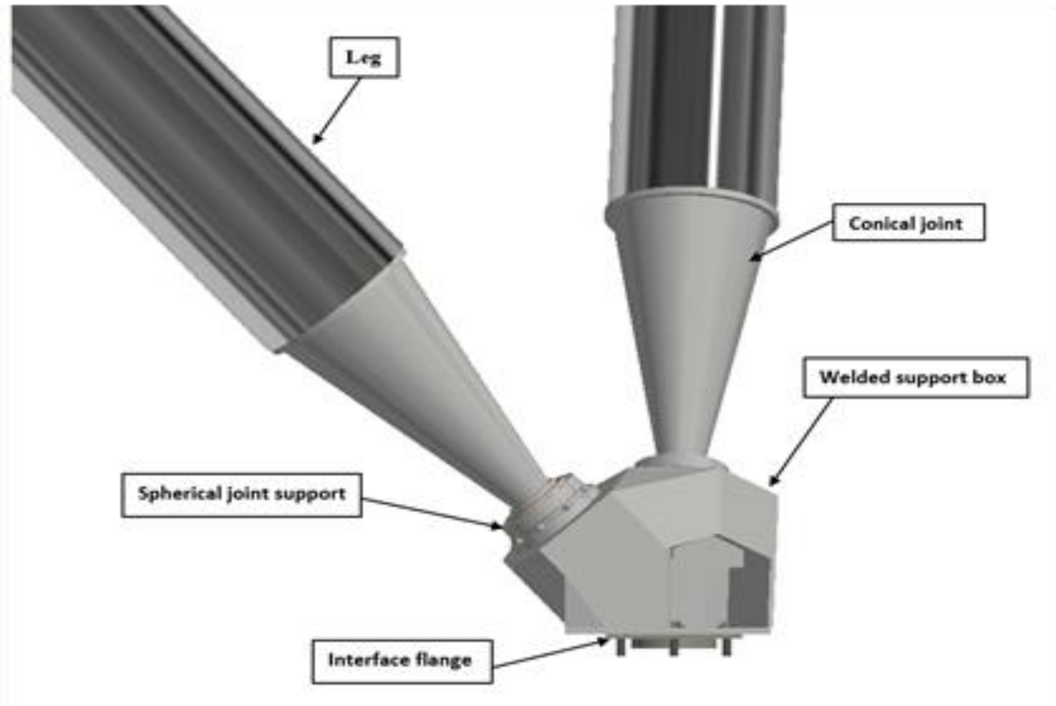
Flexible Rotation_M11_x (arcsec)	12.22	-11.47	-6.69	5.94
Flexible Rotation_M11_y (arcsec)	3.43	**7.43**	**-8.41**	**-2.44**
Flexible Rotation_M11_z (arcsec)	-0.48	**2.76**	**-5.76**	**3.48**
Deformation_M11_x (mm)	0.19	**0.21**	-0.09	-0.20
Deformation_M11_y (mm)	-0.26	**0.33**	**0.32**	-0.30
Deformation_M11_z (mm)	0.10	-0.36	-0.20	0.16

Both structural solutions are valid, in particular the carbon fibre solution allows to save 58% of the overall mass.

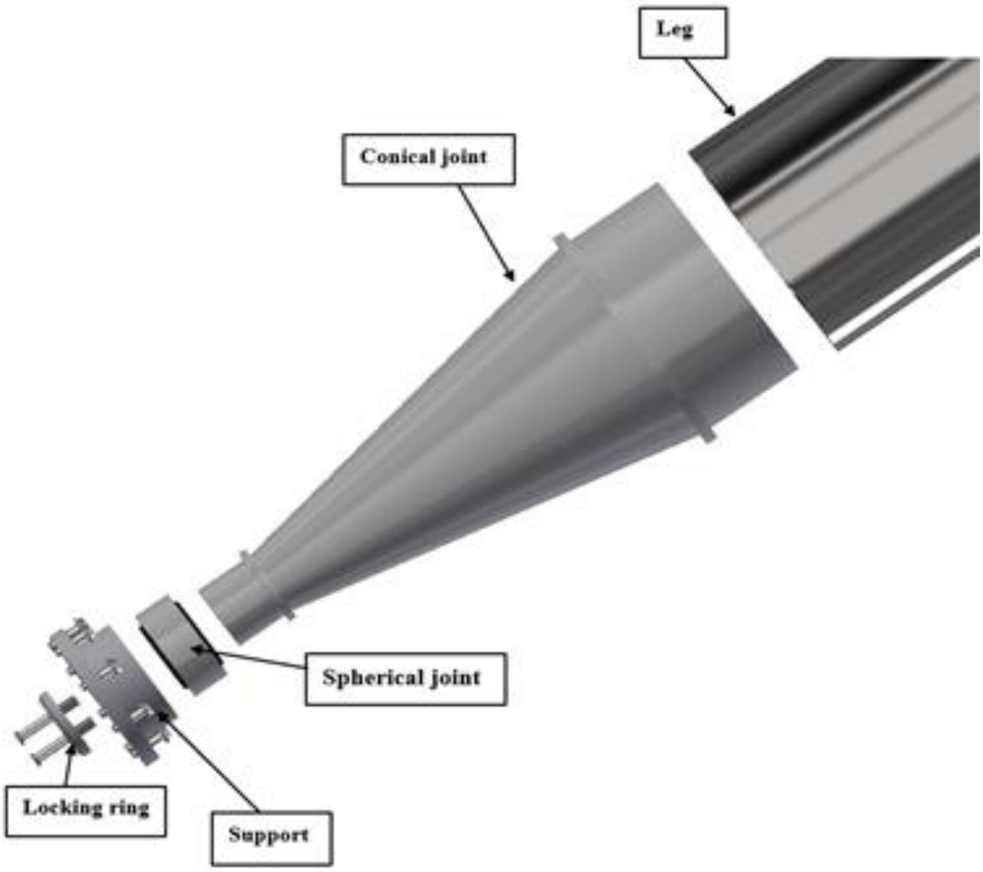
2.3. Interface system to the Nasmyth platform

The type of constraint used to attach the bench to the Nasmyth platform is a spherical knot, a nodal joint that allows only the three rotations. The interface point is actually made up of multiple parts (figure 13):

- an interface flange with the Nasmyth;
- a welded support box between the legs and the interface flange to the Nasmyth, rigidly fixed with M16 hexagonal head screws;
- a conical joint that allows the legs to be connected to the support;
- a spherical joint, which has been selected according to the loads;
- a connection support, shaped in order to leave the rotations free and to guarantee the upper fixing of the spherical joint;
- a locking ring nut for the lower fixing of the spherical joint.



(a)



(b)

Figure 13. Nasmyth interface assembly (a) – spherical joint sub-assembly (b).

3. The enclosure design: model description and overall dimensions

The main purpose of the enclosure is to protect the optomechanical elements from all external agents and to achieve a uniform temperature distribution in its internal environment. This is possible through a system of active panels cooled by refrigerant, which is a mixture of water with 30% ethylene glycol.

The enclosure shall have natural frequencies decoupled from the optical bench and the lowest eigenfrequency shall be higher than 21 Hz. The enclosure structure must support the thermal panels, resist the action of the wind and, in case of earthquake, must not suffer damage that could compromise its functionality. MAORY will be installed on the Nasmyth Platform A, so it has a mass limit to comply with. This aspect affects the design of the enclosure, such as the choice of materials employed and the types of profiles used. The enclosure has been designed with a specific safety distance between the inner wall and the back of the optomechanical elements (figure 14).

The overall dimensions of the enclosure are:

- max height of the enclosure = **4.4** m
- max internal height of the enclosure = **4.1** m
- min height of the enclosure = **3.0** m
- min internal height of the enclosure = **2.7** m
- max length = **8.0** m

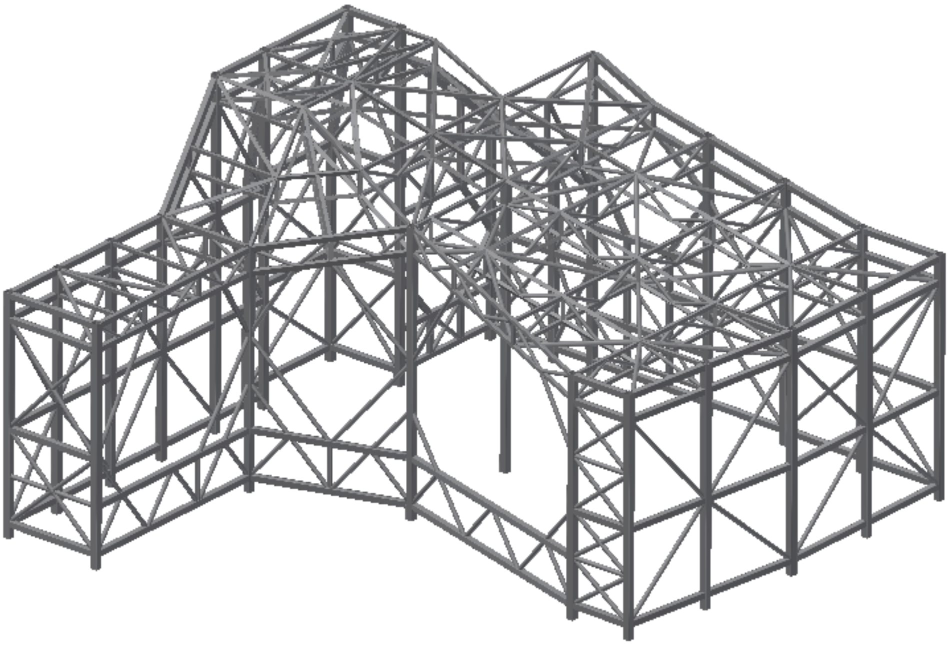


Figure 14. The structure of the enclosure.

The structure will be spread over several levels, considering that the optical elements are not all at the same height compared to the bench. In detail, the optical path that points towards the LGS module, in the central area of the bench, follows an inclined direction that influences the shape of the enclosure.

Optical beams influence the conformation of the structure:

- in the interface area with MICADO, the structure will not be completely closed, so as not to obstruct the path of the beams, and a cover will be inserted to ensure constant and uniform temperature in the MAORY-MICADO environment (figure 15);
- the optical beam will enter MAORY thanks to an aperture that will accommodate a meniscus lens (figure 16a);
- furthermore, a shutter will be inserted in the second port area until the second tool is installed on the Nasmyth platform (figure 16b).

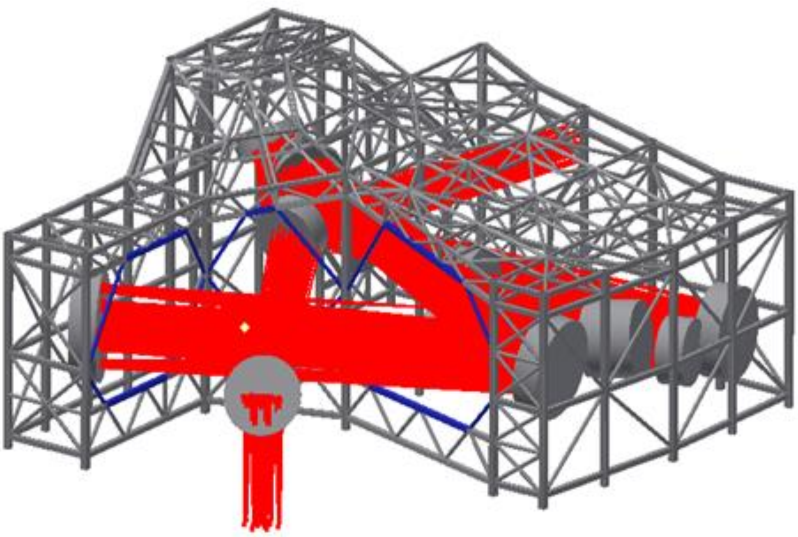
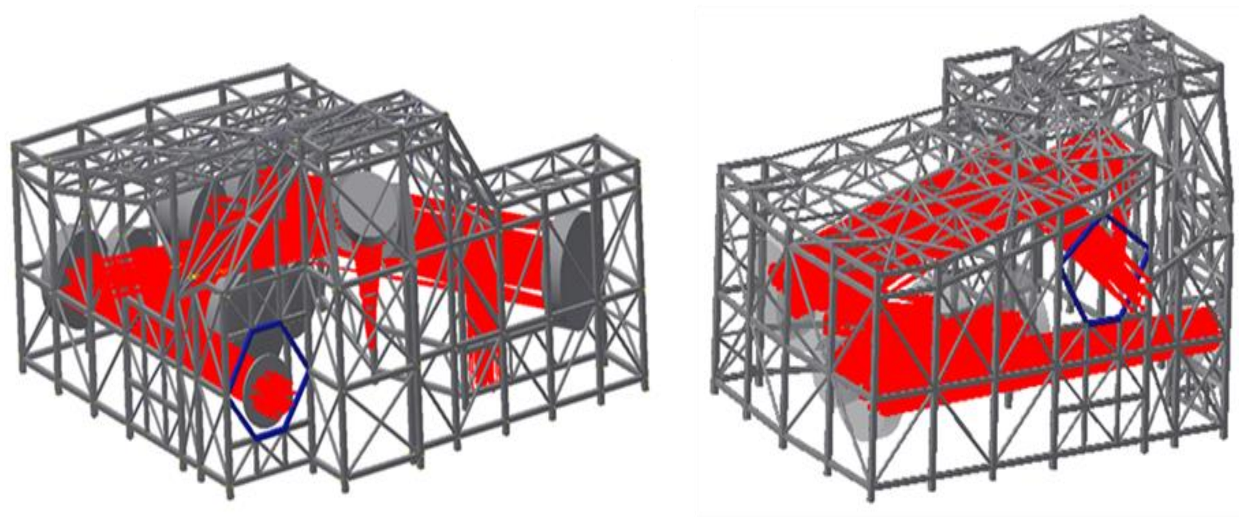


Figure 15. Opening interface zone with MICADO.



(a) (b)

Figure 16. Optical beam input opening (a) - Opening second port (b).

The structure will be provided with lateral safety protection for the operators and will not follow the perimeter of the bench perfectly. A reticular structure has been used to provide greater rigidity and lightness to the enclosure. The main structure will be made with standard square and circular hollow sections, that will be mounted on the optical bench which therefore bears their weight. The support structure has been designed entirely in aluminum, employing aluminum alloy 2024-T4 (figure 17).

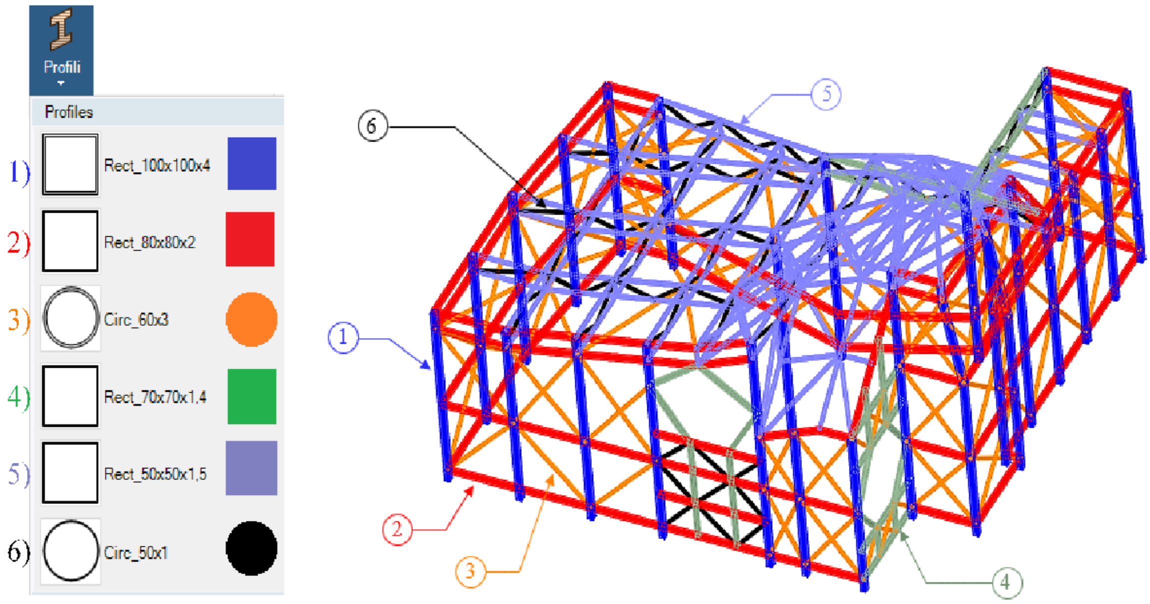


Figure 17. Geometry section of the enclosure.

The panels that cover the enclosure can be classified into two categories:

- carbon fibre coating panels (1 mm thickness), stiffened with stringers to prevent inflection of the panels;
- panels actively refrigerated by the use of cooling fluid.

The thermal panels consist of a 50 mm thick layer of insulating material (PIR, rigid polyiso foam) coated with a thin layer of aluminum, carbon fibre stiffener stringers and two aluminum plates, welded together to form a serpentine partially embedded into the PIR.

3.1. The enclosure: FEM analysis

A preliminary FEM analysis has been carried out to check and improve the structural behavior of the enclosure. The model of the enclosure was prepared using the 3D CAD software Inventor® and the finite element analyses were performed using ANSYS Workbench®.

The carbon fibre panels have been simulated as a mass applied to the structure, so to underestimate the rigidity of the enclosure. The thermal panels were simulated as surfaces and a distributed mass was applied to take into consideration the weight of the refrigerant, the insulating material and the corresponding aluminum coating. The tables 8, 9 and 10 below summarize the properties of the materials in object:

Table 8. 2024-T4 Aluminum Alloy

Density [kg/m^3]	2700
Young's modulus [MPa]	71000
Poisson's ratio	0.33
Coefficient of Thermal Expansion [1/°C]	2.3
Yield strength [MPa]	325
Tensile strength [MPa]	465

Table 9. Polyisocyanurate foam (PIR)

Density [kg/m^3]	40
Thermal Conductivity [W/(mK)]	0.022
Specific Heat [J/(kg K)]	1370

Table 10. Epoxy Carbon Woven Pre-preg

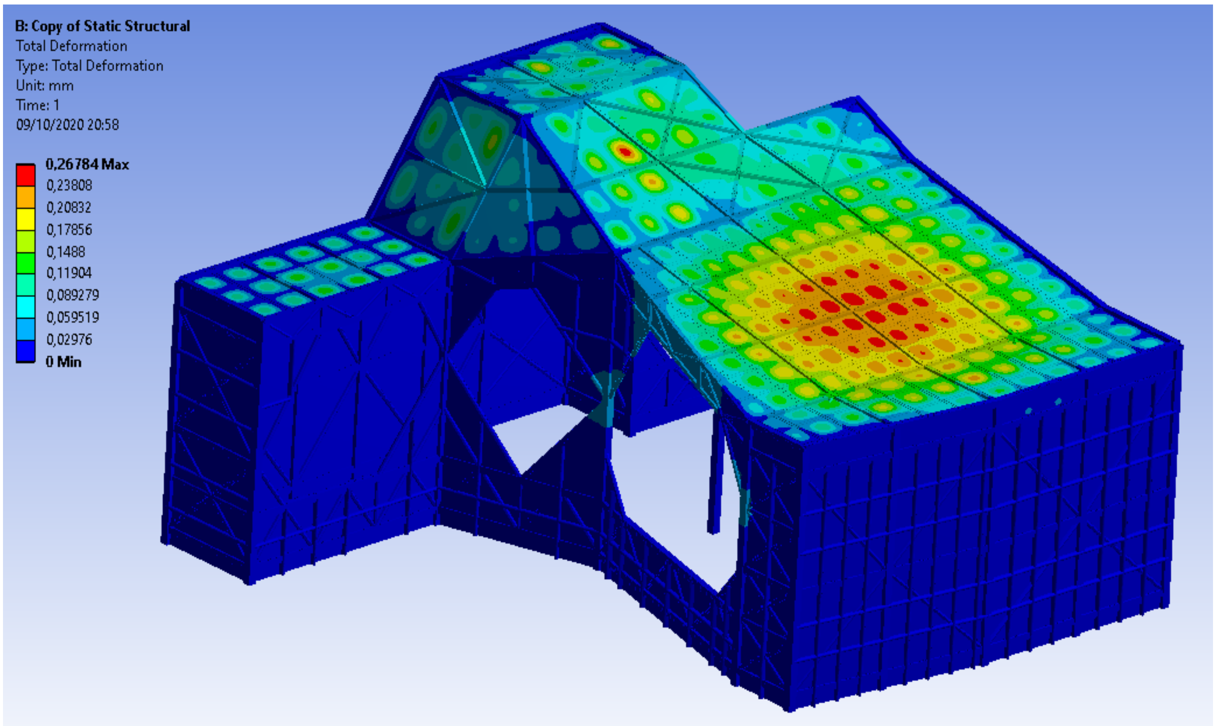
Density [kg/m^3]	1480
Coefficient of Thermal Expansion X direction [1/°C]	$2.5*10^{-6}$
Coefficient of Thermal Expansion Y direction [1/°C]	$2.5*10^{-6}$
Coefficient of Thermal Expansion Z direction [1/°C]	$1*10^{-5}$
Young's modulus X direction [MPa]	91820
Young's modulus Y direction [MPa]	91820
Young's modulus Z direction [MPa]	9000
Poisson's ratio XY	0.05
Poisson's ratio YZ	0.3
Poisson's ratio XZ	0,3
Shear Modulus XY [MPa]	19500
Shear Modulus YZ [MPa]	3000

Shear Modulus XZ [MPa]	3000
Tensile stress X direction [MPa]	829
Tensile stress Y direction [MPa]	829
Tensile stress Z direction [MPa]	50
Compressive stress X direction [MPa]	-439
Compressive stress Y direction [MPa]	-439
Compressive stress Z direction [MPa]	-140
Shear stress XY [MPa]	120
Shear stress YZ [MPa]	50
Shear stress XZ [MPa]	50

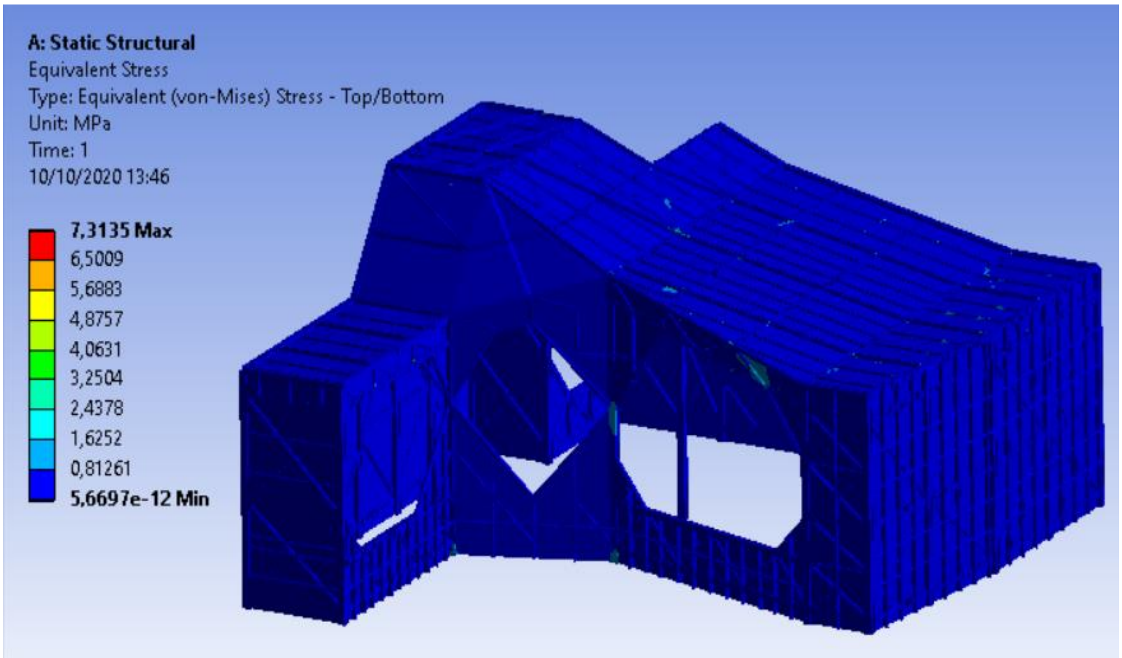
The mass of the enclosure support structure is 1013 kg; the distributed mass applied to the structure is 607 kg. Ultimately the enclosure has a mass of 2475 kg, meeting the requirement imposed by ESO (3.5 tons).

Static stress analysis has been carried out on the model of the enclosure with all the loads previously described and the results of the static and modal analysis are shown below:

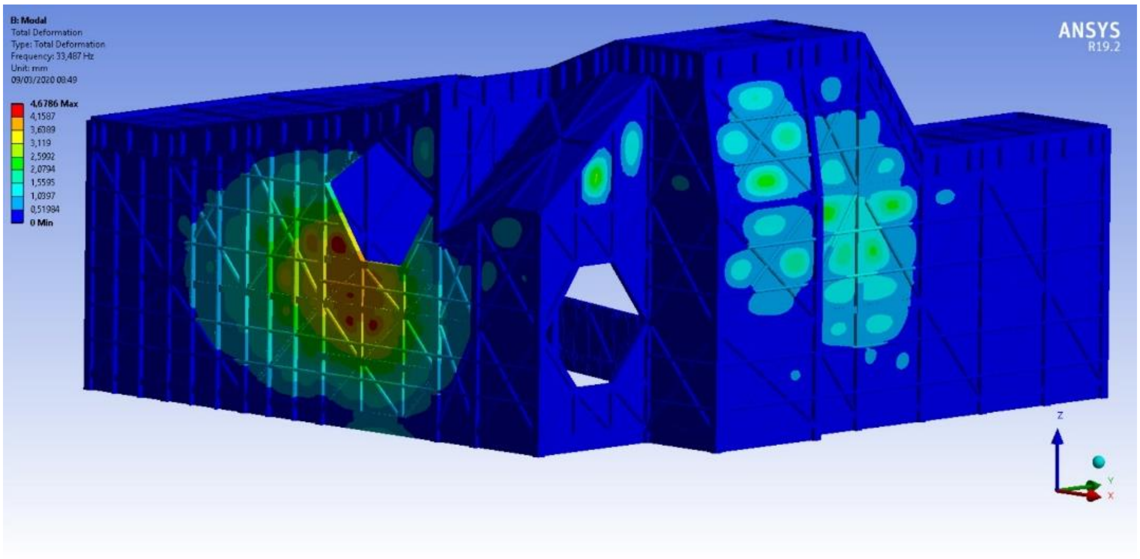
- ***Static analysis***: the maximum deflection is about **0.27** mm in the central part of the main structure (figure 18a); the Von Mises Stress is about **7.3** MPa (figure 18b);
- ***Modal analysis***: the natural frequency of the first mode is **33.5** Hz (>21 Hz).



(a)



(b)



(c)

Figure 18. Total displacement [mm] (a) - Von Mises stress [MPa] (b) – First mode [Hz] (c).

For earthquake analysis the case of the Simplified Seismic Analysis Method without response spectrum has been considered. The maximum displacement is about 3 mm and is an acceptable value since the structure has been designed with a specific safety distance from the optomechanical elements. The maximum Von Mises Stress is about 33 MPa, not exceeding the maximum allowable yield strength of the material.

During the operating hours of the telescope, the dome of the ELT will open and therefore it is necessary to perform a wind analysis. The wind was simulated as a distributed pressure on the surfaces and applied to the individual surfaces involved and then various load combinations were considered.

The maximum displacement is about **0.60** mm and is shown below (figure 19).

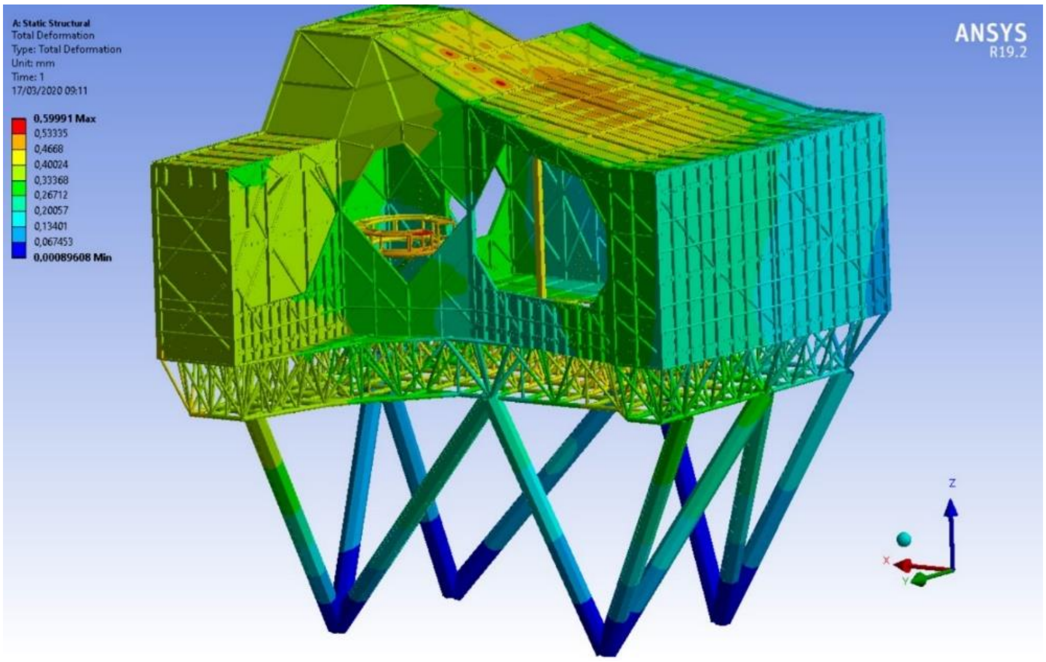


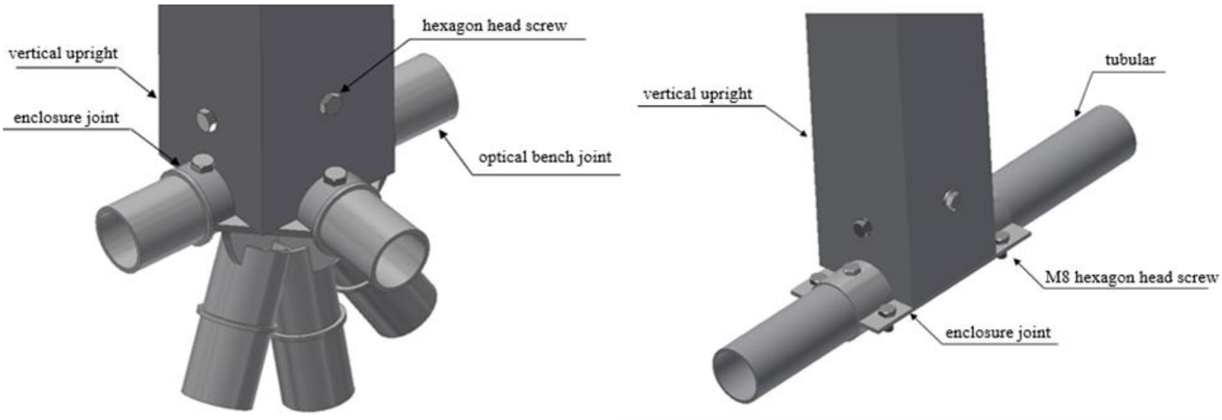
Figure 19. Total displacement of main structure [mm]– Wind analysis.

3.2. Fixing systems

The support structure of the enclosure has been designed to fit the bench perfectly, symmetrically shaped like the interface surface of the bench joint. The interface points between the enclosure and the bench are twenty-five: twenty-two of them are located at the joints of the reticular structure (along the perimeter of the bench), and are called "Type 1" (figure 20a); while the remaining three are located in areas distant from the joints and close to the tubulars, and are called "Type 2" (figure 20b). Consequently, for the latter ones, ad hoc connecting elements were made to join the tubulars. Both types of joints are made entirely of steel.

The fixing system of the enclosure consists of:

- an attachment joint (Type 1 or 2), which is mounted on the joints of the reticular structure of the bench (Type 1) or on the tubular (Type 2), using M8 hexagonal head screws;
- the vertical upright of the enclosure is fixed to the coupling by means of hexagonal head screws joined to self-locking nuts.



(a) (b)

Figure 20. Enclosure joint: Type 1 (a), Type 2 (b)

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A structural analysis of the most stressed joint was carried out, verifying that the maximum permissible yield strength of the material was not exceeded.

3.3. *Thermal panels*

The panels actively cooled by refrigerant must ensure that the temperature inside the enclosure is constant, uniform and equal to 5°C. This is due to the fact that significant temperature gradients generate convective motions near the optical elements and this would negatively affect image quality. In addition, an air conditioner will be used to achieve the required temperature, but it will be turned off during the observation hours of the telescope, as it would cause a turbulent motion regime. To ensure greater continuity between the panels, these will be shaped to ensure a male-female joint (figure 21). In addition, the panels will be equipped with perforated aluminum inserts to allow fixing to the uprights of the enclosure.

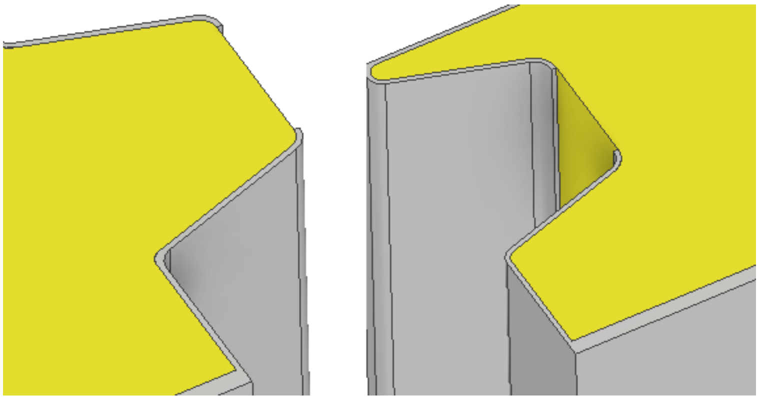

Figure 21. Custom-shaped panels with male-female interlocking.

4. Conclusions

The aim of this paper was to provide a fairly detailed picture of the mechanical-structural design of MAORY's optical bench and enclosure. The study of the very stringent constraints and design requirements was decisive for the choice of the optical bench design. In fact, the spatial reticular with circular profiles made it possible to obtain a lightweight frame that would adapt to all the constraints imposed. Although the structure of the bench was made of high-performance steel, the overall mass was at the limit imposed, so a carbon fibre solution was implemented.

From the analyses reported it is possible to deduce that:

1) The static, modal, earthquake and failure analyses are comparable in the two cases (steel and carbon fibre)
2) The carbon fibre structure allows to obtain a weight saving of 58%.
3) Carbon fibre profiles are extremely expensive compared to steel
4) Carbon fibre causes problems in manufacturing due to its non-weldability.

The enclosure has been designed in aluminum, choosing suitable profiles to guarantee the project requirements. In addition, the structure is not perfectly closed but has openings to allow the passage of optical beams. The structure has been conceived on several levels, to respect the position of the optical elements and to minimize the overall dimensions and its consequent weight. In order to respect the thermal constraints, a system of active panels cooled by refrigerant has been implemented. The entire enclosure was finally coated with carbon fibre coating panels, stiffened with stringers to prevent inflection of the panels.

The optical bench will provide support for all optical elements and the enclosure will be fixed on the bench to protect the optomechanical elements. The environment will be thermalized by panels and actively cooled by refrigerant, and the required temperature will be reached by an air conditioner.

MAORY will appear as follows (figure 22).

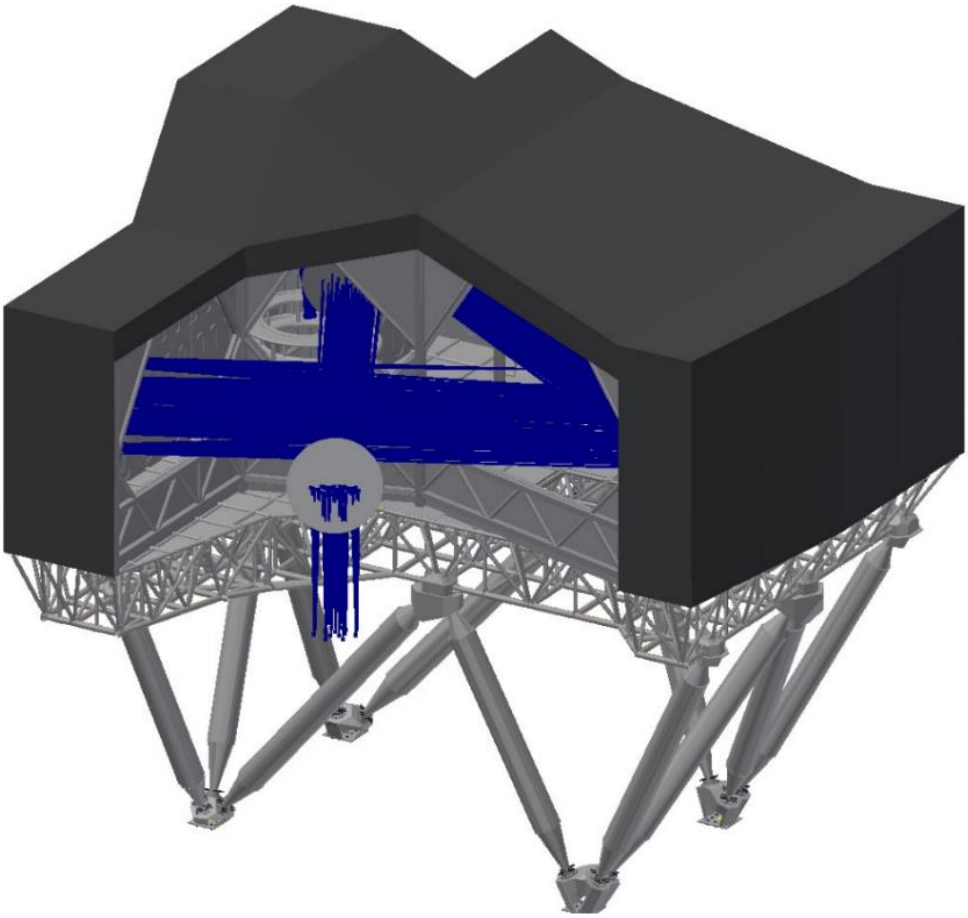

Figure 22. MAORY: the optical bench and the enclosure.

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