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Lightweighting strategies for Main Support Structures of ELT instrumentation

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ABSTRACT

This paper outlines an overview of the mechanical design of the support structure and optical bench, for ELT (Extremely Large Telescope) class of instrument utilizing standard materials. It also provides a presentation of a possible strategy to lighten this mechanical design. The mass requirements for the new class of astronomical instruments, for the Extremely Large Telescope, put tight constraints for the mechanical design of said instruments. While aluminum or steel are still the main materials that are usually employed for main support structures, carbon fiber reinforced polymer (CFRP) offers a wide range of promising possibilities. The interesting properties of carbon fibers make composite materials reinforced with carbon fibers ideal for numerous applications with high mechanical performance and low weight requirements. Carbon fibers have high strength and low weight (high strength-to-weight ratio). Furthermore, carbon fibers have a high modulus of elasticity and low thermal expansion, they are corrosion resistant and their density is lower than that of aluminum: thus, they are ideal for "light applications". In this paper two possible mechanical designs for a "large optical bench", based on different materials, are presented and compared: structural steel and carbon fiber reinforced polymer (CFRP). The models are validated through FEM Analysis approach, utilizing CAD and CAE software.

Keywords: ESO, ELT, MAORY, AO, Adaptive Optics, FEM, FE, CAD, CAE, CFRP, Mechanical Structure, Bench.

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 is based on a system of five mirrors and incorporates adaptive optics mirrors. The primary mirror consists of 798 segments, each 1.4 meters wide. Figure 1 shows the main optical path of the ELT telescope.

The ELT will be the largest ground based optical telescope of the world, in the next future, and will require, to guarantee its potentialities regarding the angular resolution, the wave front correction of the star images caused by the atmospheric turbulence. Figure 2 presents the preliminary design of the ELT telescope.

MAORY is the multi-adaptive optics module for the ELT. MAORY will be among the "first light" class of instrument that will be installed on the Nasmyth Platform of ELT. It will provide a corrected field-of-view to its client instrument MICADO, a near-infrared camera and spectrograph. MAORY will foresee a second port 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) [1][2][3]. 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.

The MAORY project is designed and developed by a Consortium including the Istituto Nazionale di Astrofisica (INAF), the Institut de Planétologie et d'Astrophysique de Grenoble (IPAG) and the National University of Ireland, Galway (NUI-Galway). The European Southern Observatory is also involved in the development of the whole instrument.

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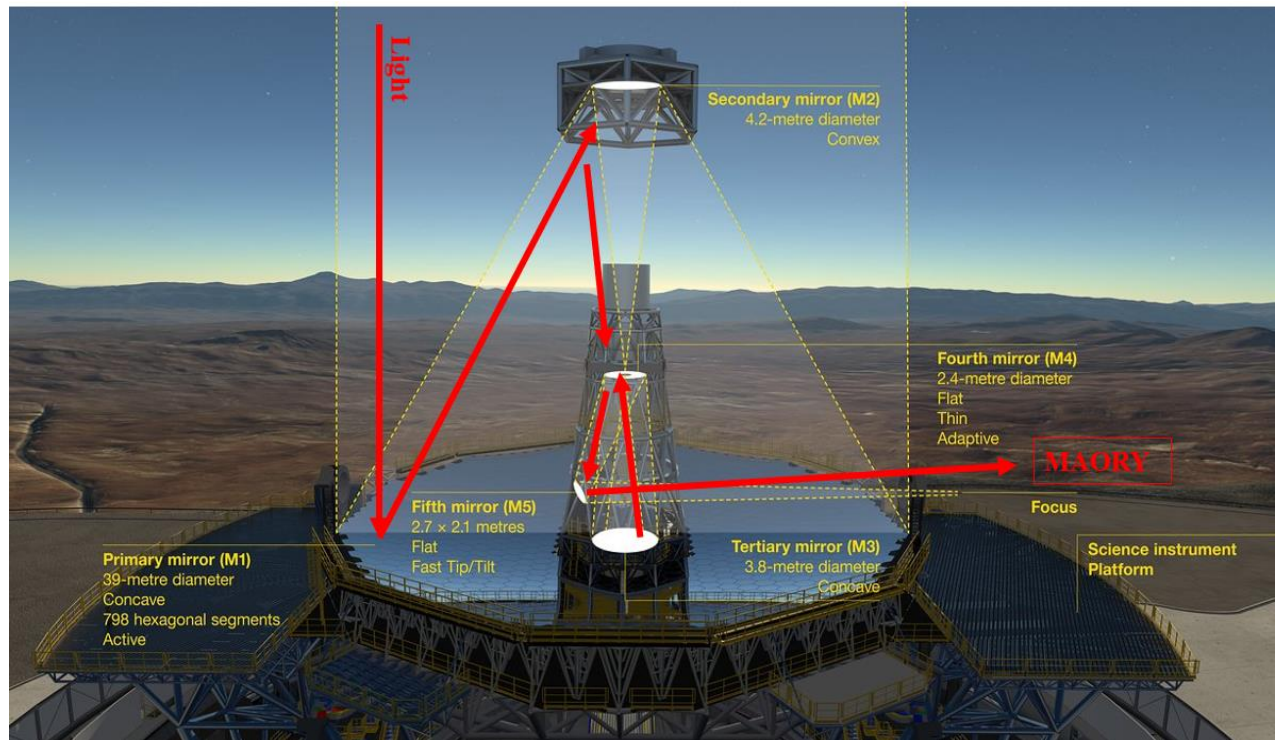


Figure 1. Layout of main optical path of ELT.

The MAORY instrument project entered its phase B in February 2016. Consolidation of the baseline optical design is closed [4] [5] [6] and the mechanical design (shown in the next chapter) refers to the optical design developed until February 2020. Both the mechanical designs shown in this paper, Steel and CFRP design, were developed taking into account scientific performance requirements and interface constraints.

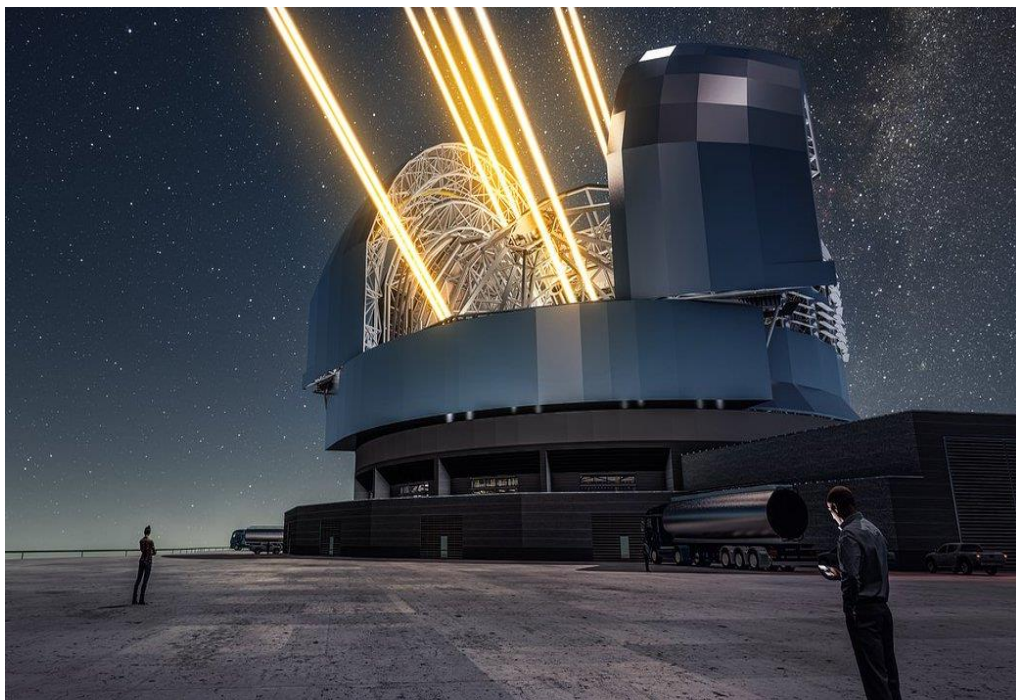


Figure 2. The Extremely Large Telescope.

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

This paper provides a comparison between two different designs for the MAORY bench. The first approach foresees to use the classical materials for this kind of constructions, utilizing the know-how developed in these years and offering the significant advantage, for the construction phase, of allowing to joint parts with bolts or welds. The second typology of mechanical design has maintained the same geometry of the previous one but, in this case, different materials were used and, consequently, different modes to connect the various elements of the optical bench.

The paper provides an analysis of the two solutions and their main differences. It specifically focuses on the structural requirements that impose a maximum overall mass for the optical bench of 8 Tons and a first eigenfrequency mode of 7 Hz; the compliance with said requirements for both solutions was investigated.

2. MAORY OPTICAL BENCH: PRELIMINARY DESIGN DESCRIPTION

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 (this requirement derived from the first mode of the ELT main structure, almost, 2 Hz).

In order to comply with all the requirements, including the best shape for easy accessibility for operators to all opto-mechanical components, the choice was made to design a steel truss spatial structure. This model will be composed of 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, the repetition of a pyramidal structure with a rectangular base has been employed. Figure 3 shows the basic element of the structure utilized for the optical bench.

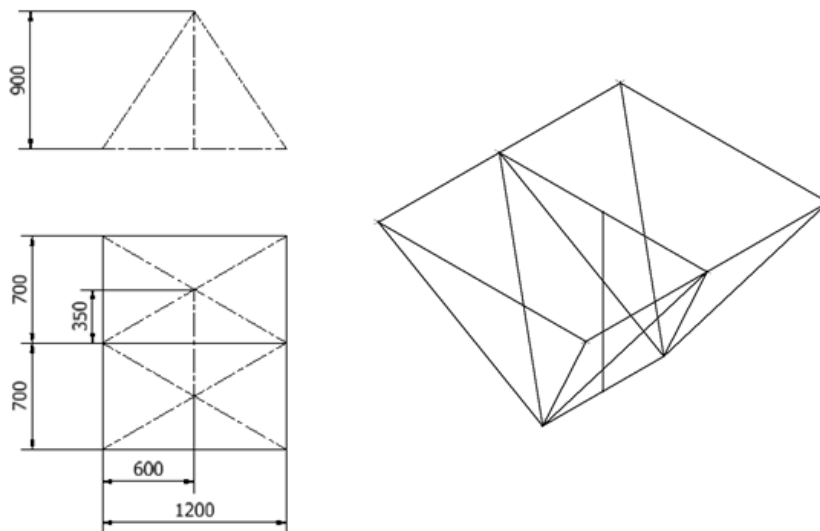


Figure 3. Layout of basic elements of the bench.

The LGS (Laser Guide Star) module has to be positioned at a higher opto-mechanical plane compared to the opto-mechanical plane of the MAORY main optical path, 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 walkable floor) has a distance from the MAORY focal plane of 1200mm. The final layout, considering the different design constraints, will appear as in Figure 4.

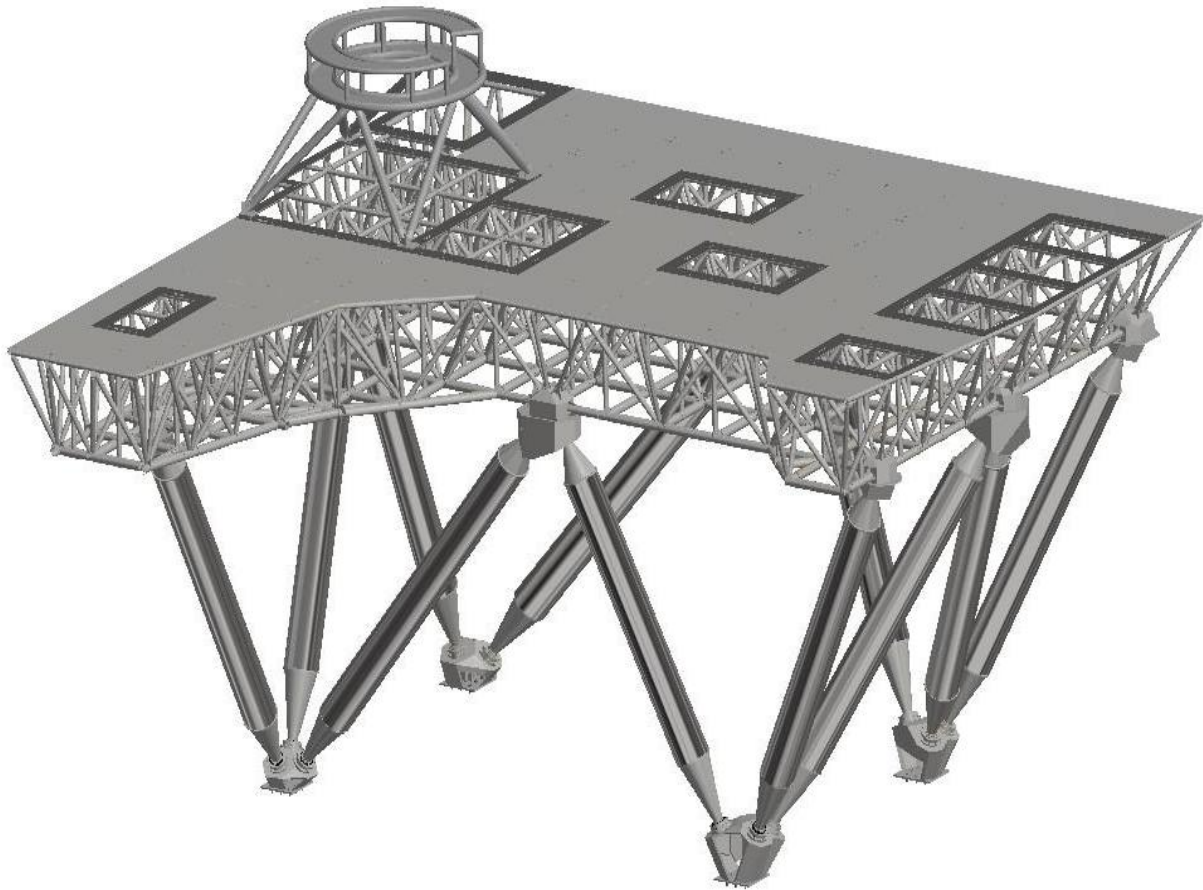


Figure 4. MAORY Optical Bench – Steel design

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

The top of the bench is divided into two parts: the interface plate with the opto-mechanical elements, made of steel with a thickness of 10 mm, necessary 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 during the assembly and disassembly phases. Also the electronic routing implemented fits this mechanic preliminary design; more information on the electronic cabling layout is included in the papers [7][8], presented in this Conference

The MAORY environment is thermalized [9], therefore, 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 with the steel tubulars were insulated with another layer of insulating material: Aerogel. Under each opto-mechanical support, a panel consisting of a 2mm steel sheet and insulating material (PIR) will be inserted.

2.1 Interface system to the Nasmyth Platform

The type of constraint used to attach the bench on 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 5):

- 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.

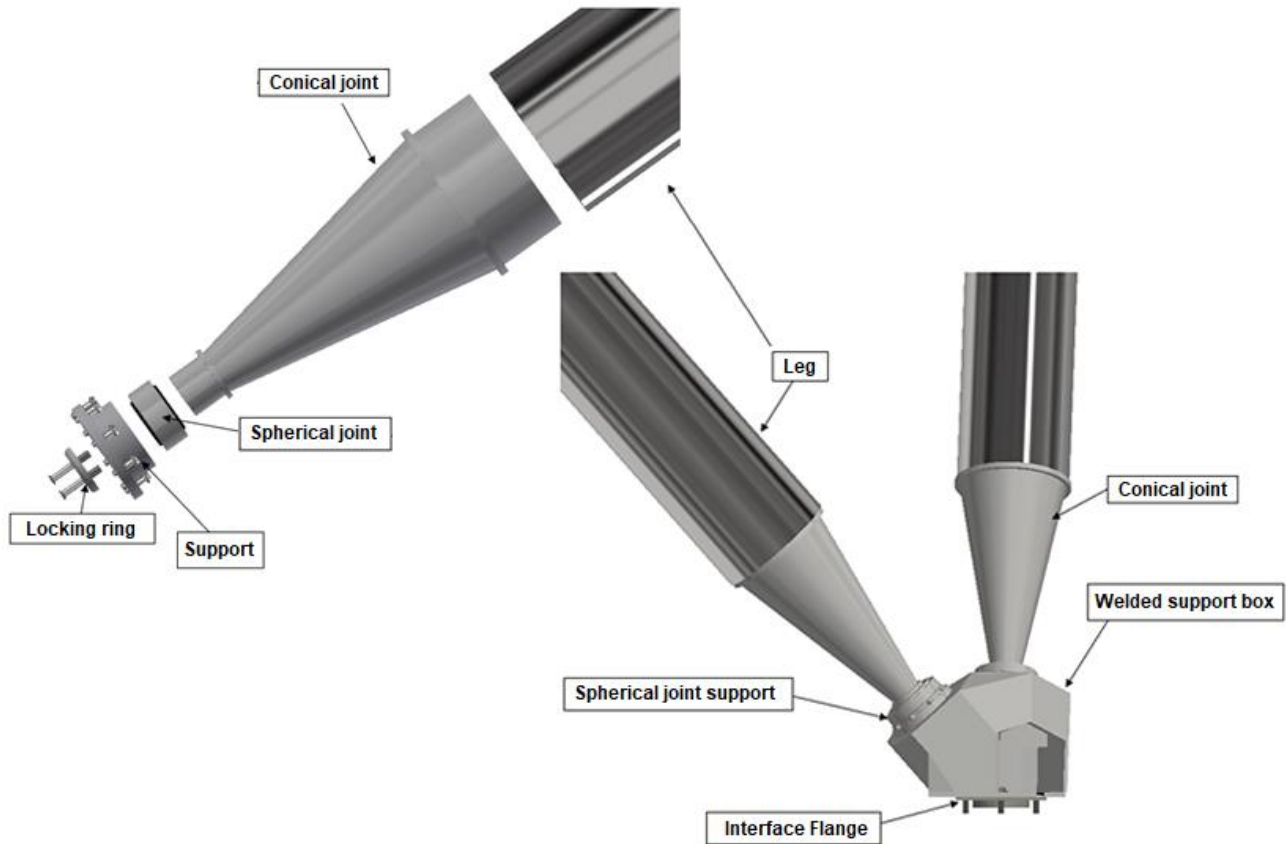


Figure 5. Nasmyth interface assembly (right) – spherical joint sub-assembly (left).

3. MAORY OPTICAL BENCH: STEEL DESIGN

The MAORY bench is made of sheet metal and the pipes of structural steel. The CAD model 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). Figure 6 shows the FE model in Ansys CAE software.

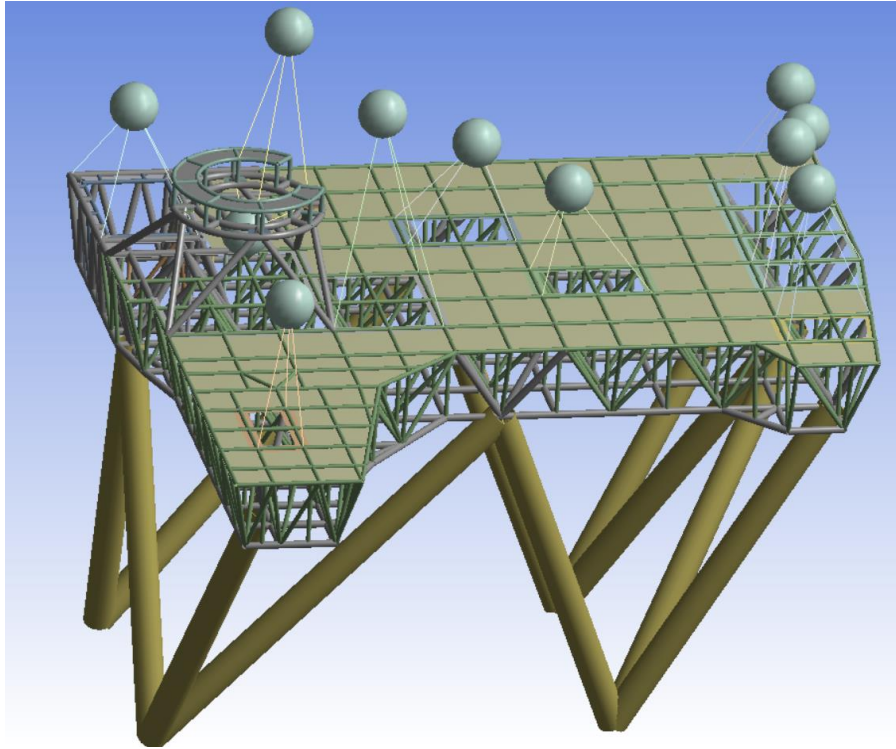


Figure 6. MAORY steel design – FE model.

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

Table 1. Material properties: Steel AISI 4130.

Property	Unit	Value
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. Material properties: Honeycomb Aluminum.

Property	Unit	Value
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 bench, different profiles for the main structure were used. In particular, for the lattice structure of the bench the following profiles were employed:

- bottom profile external diameter and thickness Ø 80 mm – th. 3 mm (Figure 7: n.2);
- base profile external diameter and thickness: Ø 50 mm - th. 3 mm (Figure 7: n.3);

- legs external diameter and thickness: $\varnothing$ 323 mm – th. 10 mm (Figure 7: n.4);
- interface plates with opto-mechanical thickness: th. 10 mm;
- honeycomb panels thickness: th. 25mm.

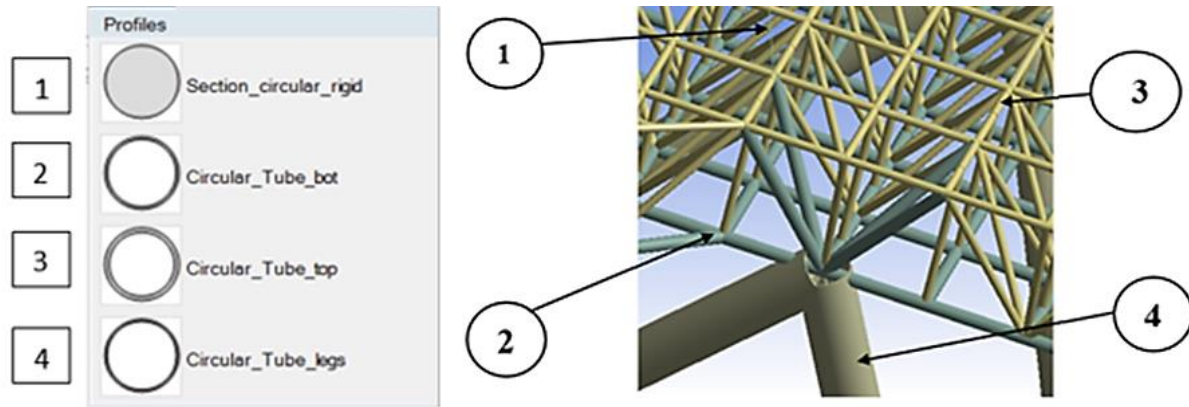
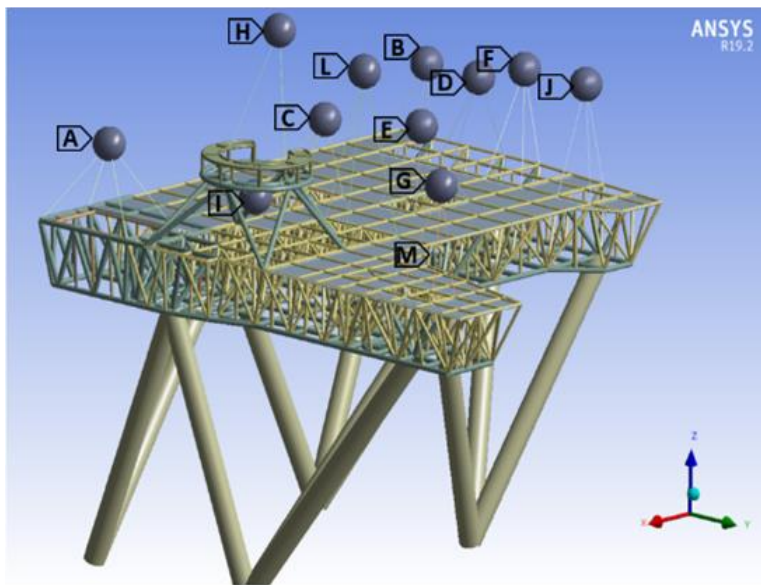


Figure 7. FE model - Geometry sections.

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 barycenter. This point is connected to the top surface of the bench using 3 or 4 rigid elements. The estimated loads (including 20% of contingency) that are shown below in the Figure 8, have been used for the opto-mechanical mountings and sub-assemblies in the FE model.



		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+LGS	860	1032
I	LGS	700	840
J	M12	556	667
L	M13b	606	727
M	Enclosure	2916	3500
	TOT		13185

Figure 8. FE model - Loads applied in FEM analysis.

The interface between the legs of the bench and the Nasmyth platform is modelled as nodal joint which allows only the three rotations (three displacements are 0 mm). The mass obtained from the CAE software is **7.9 tons**.

Finite element analyses were performed on the defined model: 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. The results of the analyses are as follows:

- **Static analysis:** the maximum displacement is **0.69** mm (see Figure 9a);
- **Static analysis:** maximum Von Mises stress is **42.6** MPa (see Figure 9b);
- **Modal analysis:** first natural frequency is **9.18** Hz (see Figure 9c).

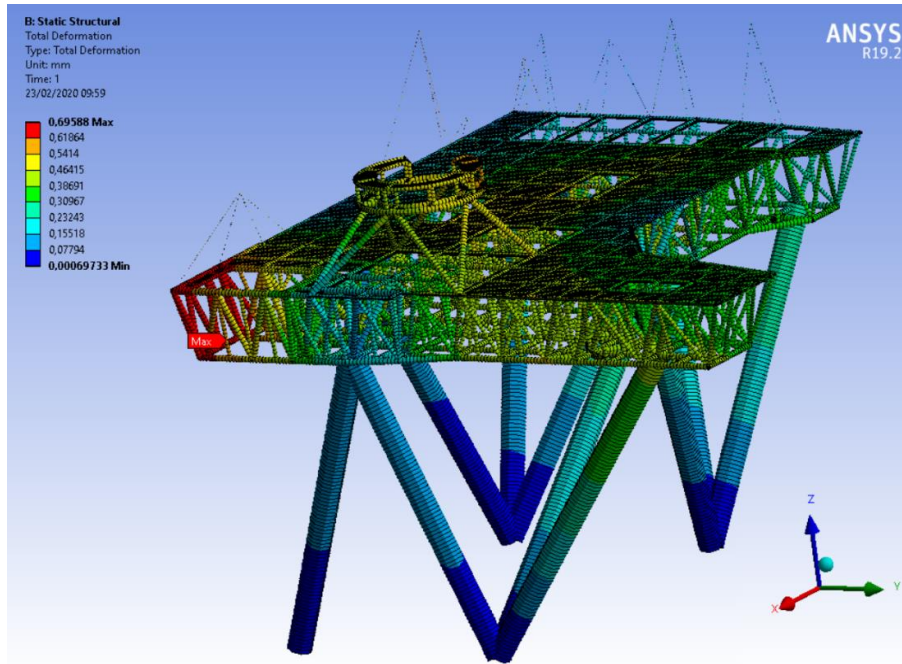


Figure 9a. Steel Design - Static results: maximum displacement.

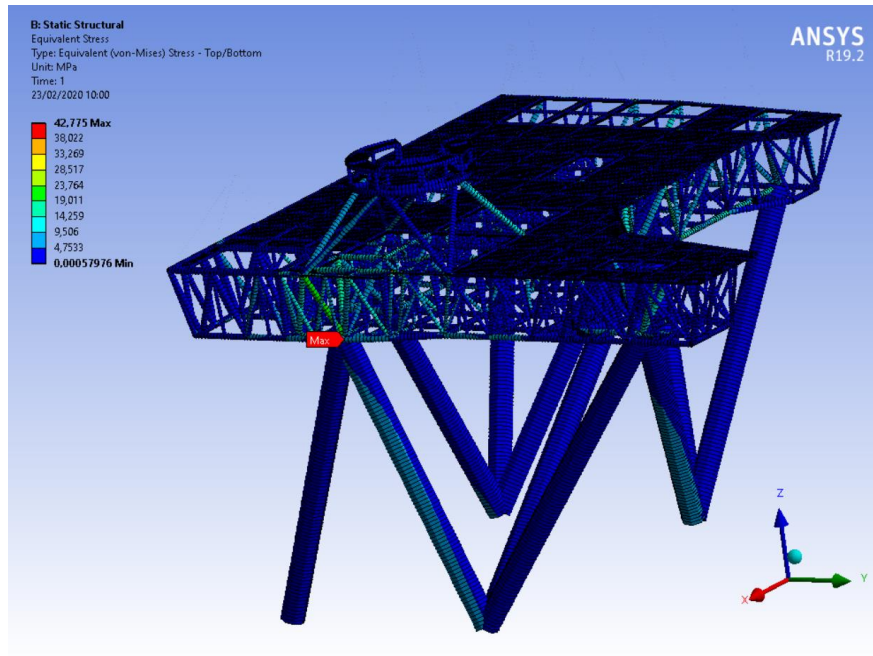


Figure 9b. Steel Design - Static results: Equivalent Von Mises stress.

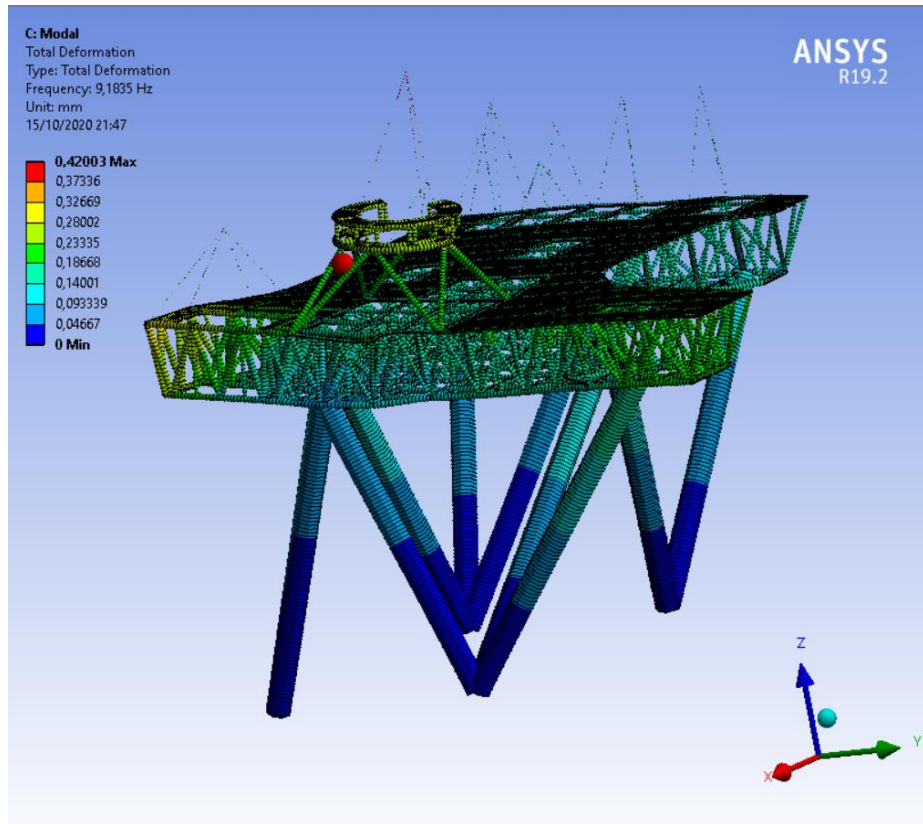


Figure 9c. Steel Design - Modal results: First eigenfrequency mode.

The earthquake analysis was carried out using the Simplified Seismic Analysis Method without response spectrum. 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, for 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**.

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

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

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

	Force Reaction X	Force Reaction Y	Force Reaction Z
MAX [N]	308880.31	231726.66	800889.26
MIN [N]	-304641.19	-233289.32	-947667.89

The model is compliant with the requirements, since the maximum 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 center of gravity of the opto-mechanics under the action of a failure imposed on the structure is evaluated in output. The number of analyses to be carried out is 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 10):

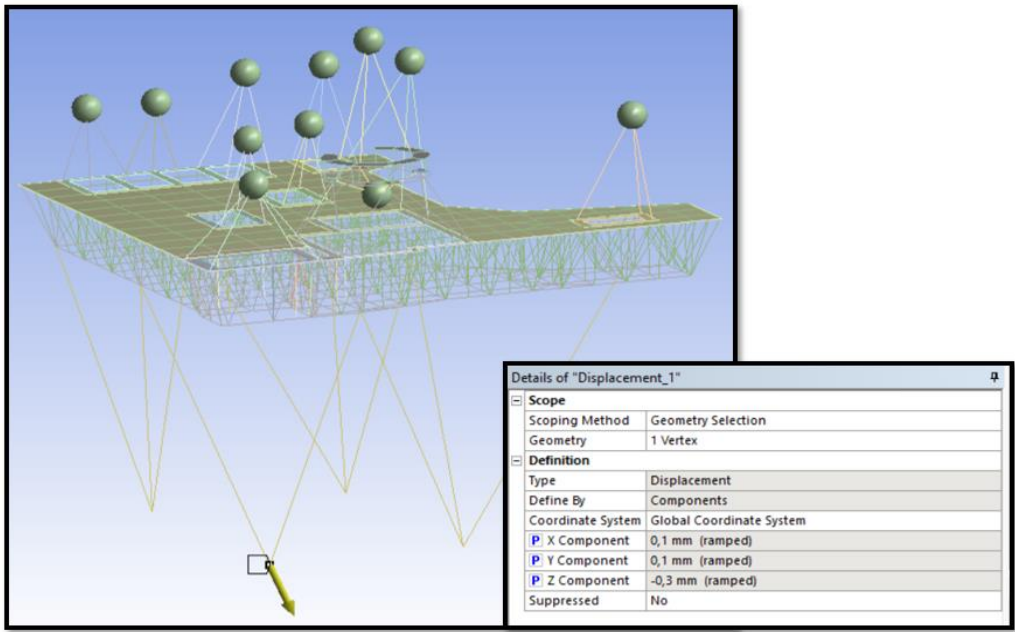


Figure 10. Setting displacement Node_1.

For the remaining three points of constraint to the Nasmyth platform it is required that there are no movements in the three directions and that the rotations are free (Figure 11).

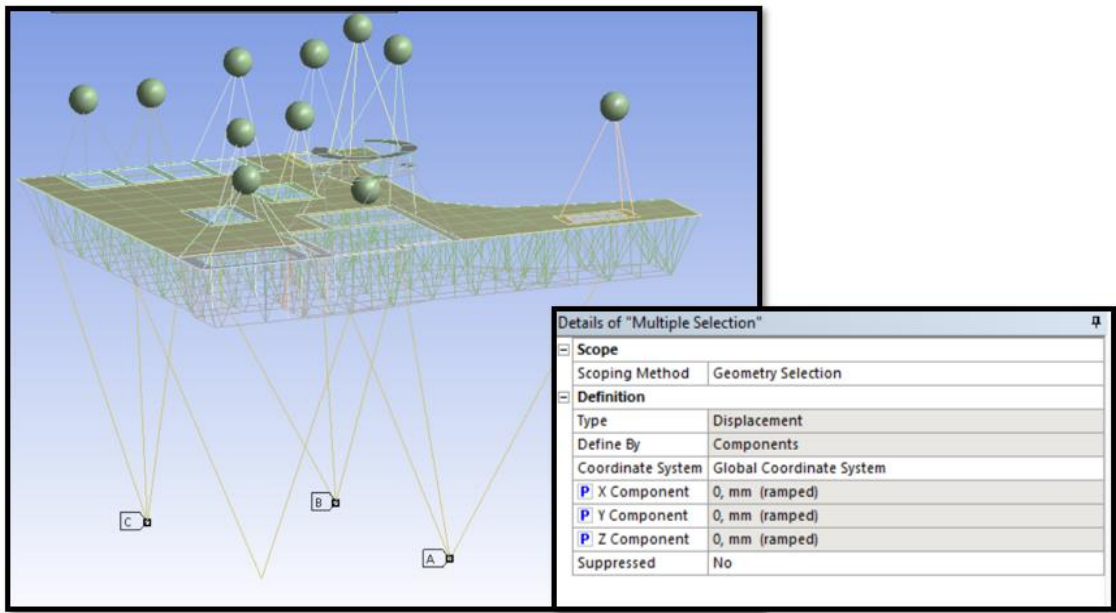


Figure 11. Setting displacement Node_2, Node_3, Node_4.

The output for each center of gravity will include both flexible rotations and deformations in the three directions of the optomechanical center of gravity. As an example, the output for the M6 mirror is shown in Figure 12:

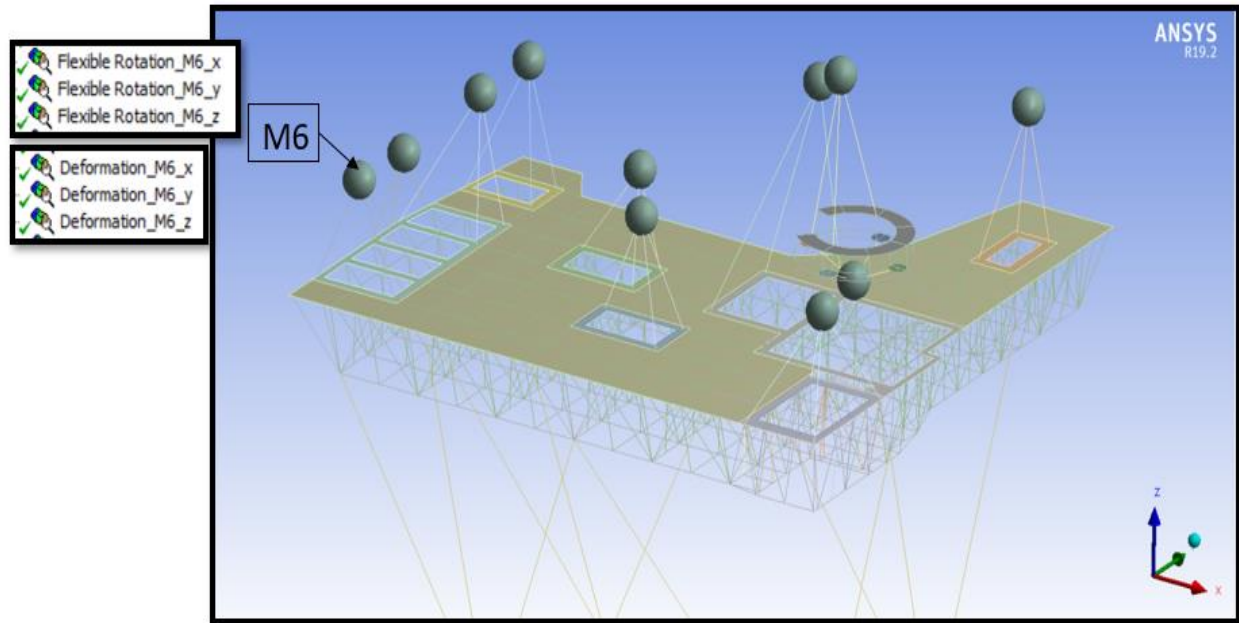


Figure 12. Output Failure analysis M6.

The displacements of the optomechanics barycentre are not all compliant with the requirements, as in the case of M11 (Table 4). In order to achieve full compliance, 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)

	Value	Value	Value	Value
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

4. MAORY OPTICAL BENCH: CFRP DESIGN

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 to increase the margin of compliance with the total mass requirement. The carbon fiber 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 13). 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 can be held together by a high-performance glue.

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.

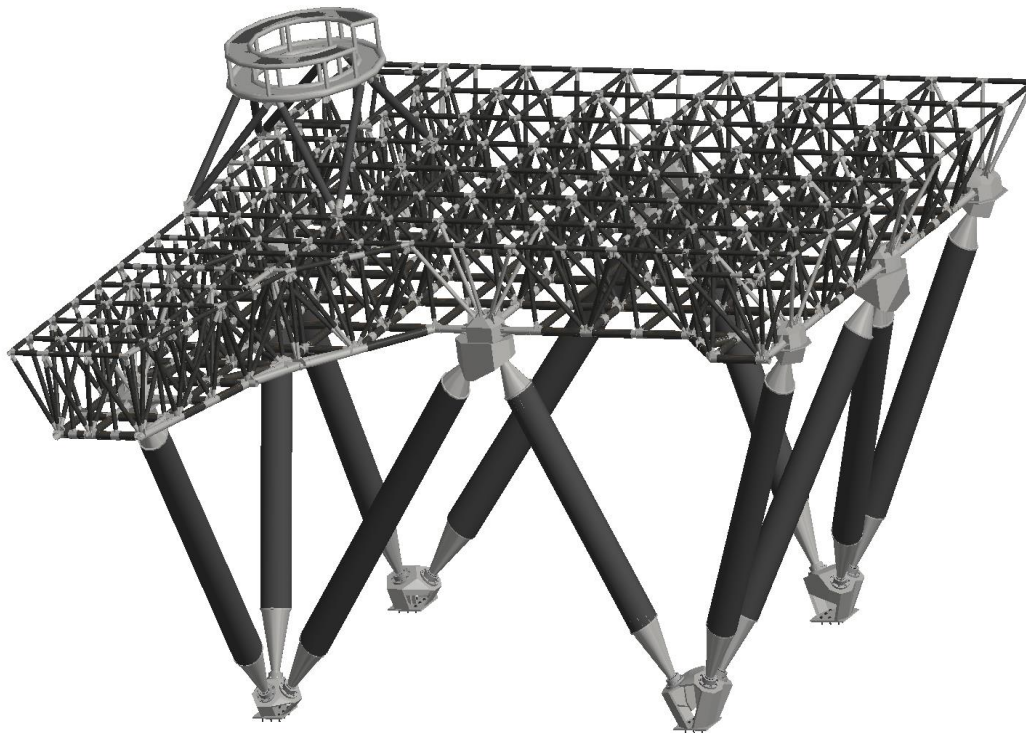


Figure 13. MAORY Optical Bench: CFRP design.

The sections were defined according to these parameters: for the beams representing the steel joint the sections are the same as seen in the previous section; to ensure performance comparable to that of the steel bench, the beams corresponding to the carbon fiber tubulars have the following sections:

- base profile external diameter and thickness: $\varnothing$ 60 mm - th. 8 mm (Figure 14: n.1);
- bottom profile external diameter and thickness: $\varnothing$ 90 mm – th. 8 mm (Figure 14: n.2);
- legs external diameter and thickness: $\varnothing$ 337.8 mm – th. 17.4 mm (Figure 14: n.3).

The carbon fiber bench has been loaded in the same way as the steel bench so that a comparison can be made, also the constrains used for this simulation are the same. The payload setting has been presented before in the Figure 8.

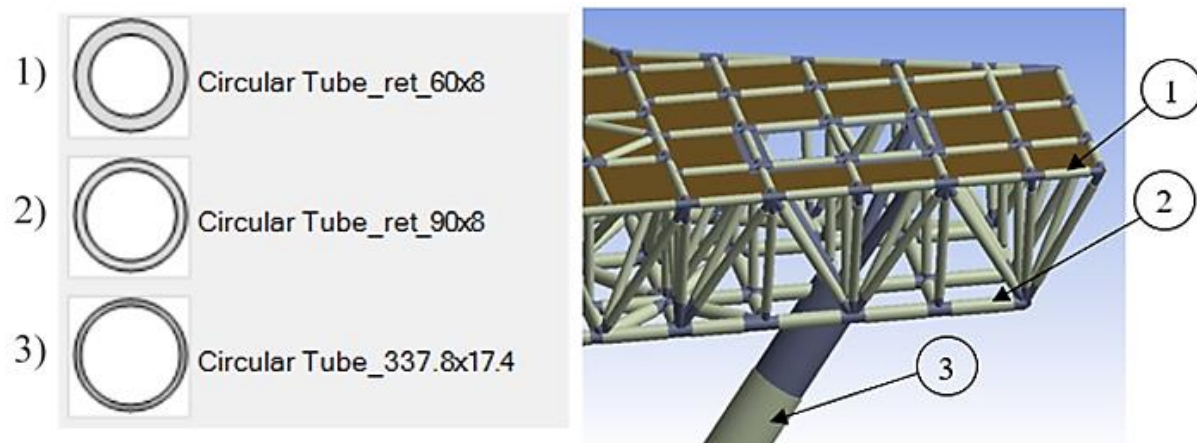


Figure 14. FE model - Geometry sections CFRP design.

The datasheet of CFRP material used is reported in the Table 5.

Table 5. Material properties: Epoxy Carbon Woven Pre-preg

Property	Unit	Value
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

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

- **Static analysis:** the maximum displacement is **0.70** mm (see Figure 15a);
- **Static analysis:** maximum Von Mises stress is **39.6** MPa (see Figure 15b);
- **Modal analysis:** first natural frequency is **8.63** Hz (see Figure 15c), in compliance with the requirement of 7 Hz.

The mass derived from the FE model is **5 tons**.

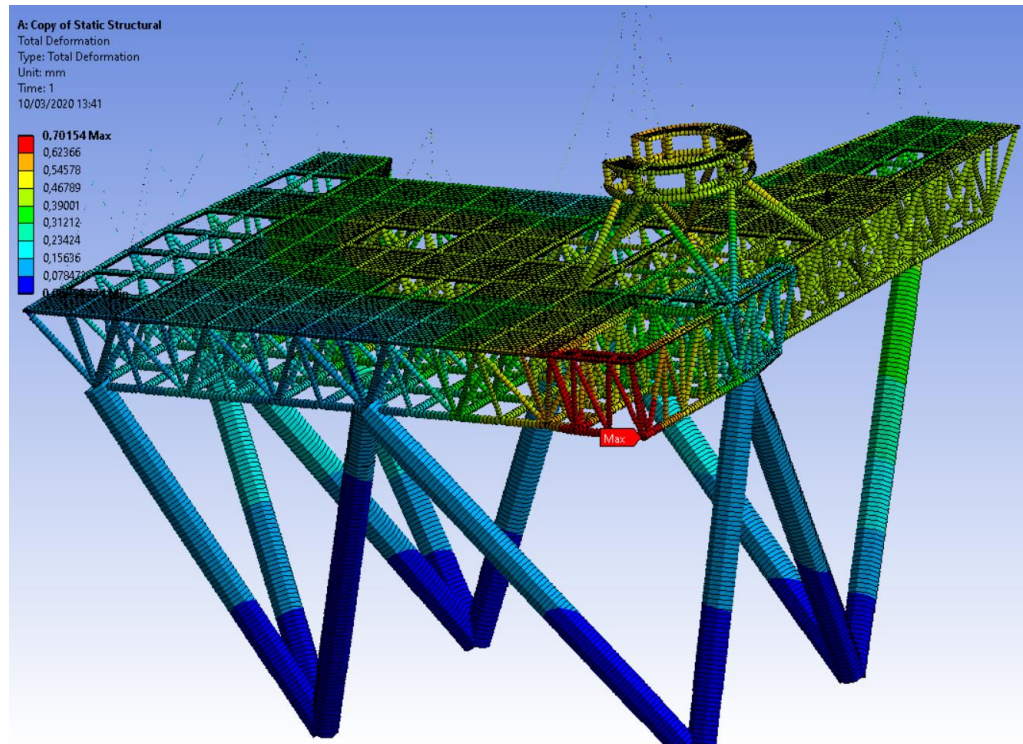


Figure 15a. CFRP Design - Static results: Maximum displacement.

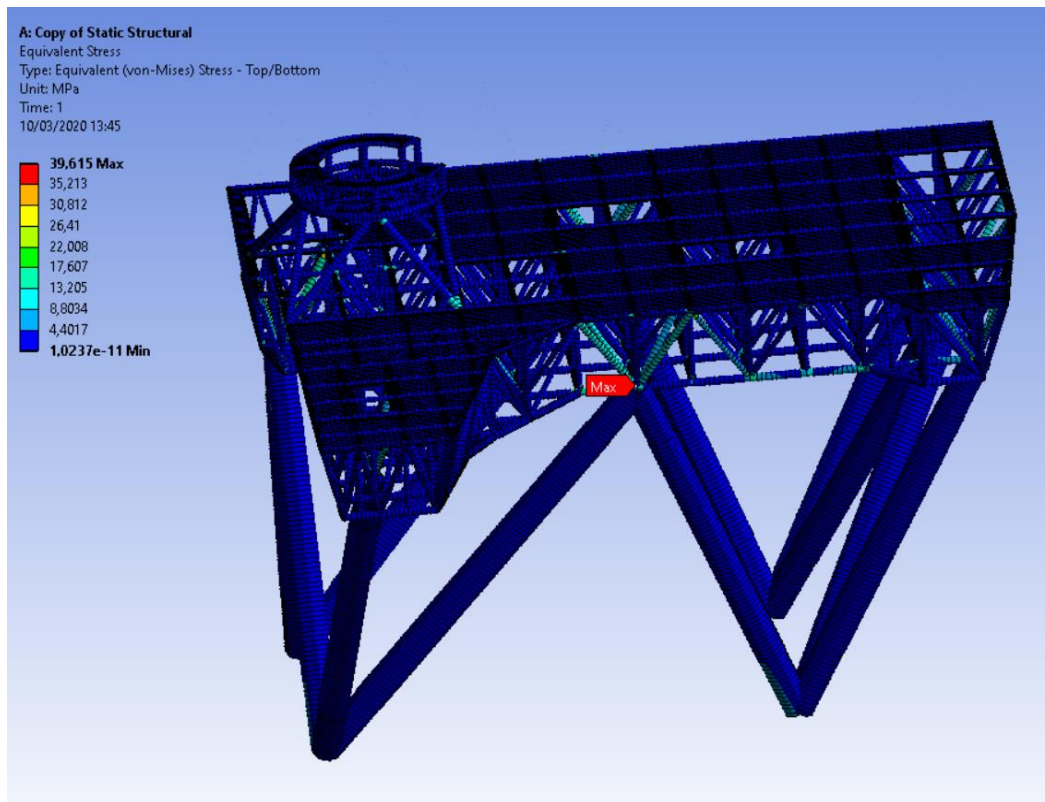


Figure 15b. CFRP Design - Static results: Equivalent Von Mises Stress.

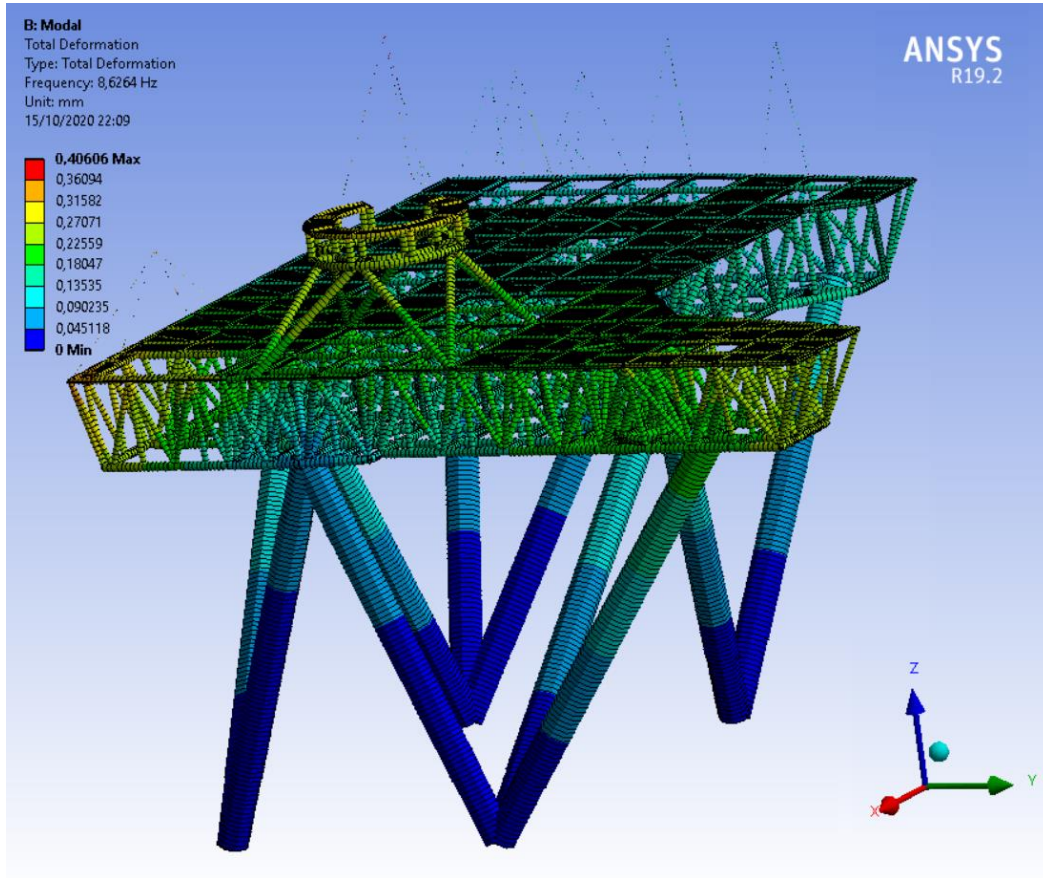


Figure 15c. CFRP Design - Modal results: First eigenfrequency mode.

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 fiber bench were compared, as shown below.

$$\begin{aligned}\% \varepsilon_{def} &= \left(\frac{(0.70 - 0.69)}{0.70} \right) * 100 = 1.4\% \\ \% \varepsilon_{Eq} &= \left(\frac{(42.77 - 39.61)}{42.77} \right) * 100 = 7.38\%\end{aligned}\quad (2)$$

The earthquake and failure analyses were carried out as in the case of the steel bench (Table 3 and Table 4). The earthquake analyses were performed using equations (1).

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

	Force Reaction X	Force Reaction Y	Force Reaction Z
MAX [N]	281433.65	190483.47	806371.77
MIN [N]	-274021.61	-191150.33	-937546.42

Table 7. Results of failure analysis (CFRP design)

	Value	Value	Value	Value
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 fiber solution allows to save **58%** of the overall mass.

5. CONCLUSIONS

Two possible solutions for the main structures of ELT class of instrument for an adaptive optics module were presented. The study of the very stringent constraints and design requirements represented the main factor in the comparison between the two solutions. In fact, the spatial reticular structure 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, while the carbon fiber solution allowed for a lower overall weight. From the analyses reported it is possible to deduce that:

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

This mechanical solution for MAORY will appear as follows, complete of its thermal enclosure (Figure 16).

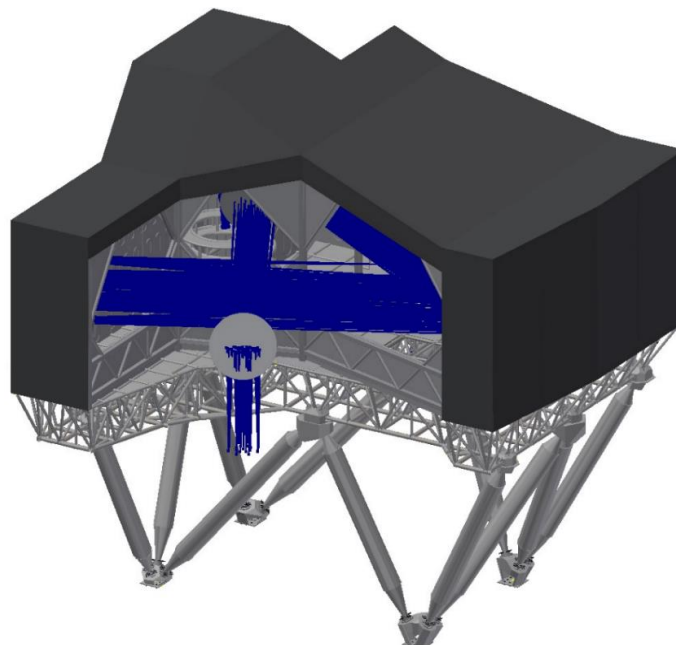


Figure 16. MAORY: the optical bench and the enclosure.

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