Base Load Power Hub (BLPH)

DAVIT CRANE
MECHANICAL CALCULATIONS

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1 INTRODUCTION

1.1 PROJECT OVERVIEW

Crosswind, a Shell and Eneco joint venture, was awarded the tender to develop the offshore Hollandse Kust Noord (HKN) wind farm, located 18.5km off the coast of Netherlands near the town of Egmond aan Zee. The consortium plans to have HKN operational in 2023 with an installed capacity of 759MW generating at least 3.3TWh per year. This is sufficient renewable power to supply more than 1 million Dutch households with green electricity.

Offshore wind farms produce intermittent electricity and balancing these dips and peaks and integrating this electricity into the national energy system requires new technologies. For this reason, the offshore wind farm will include several new technologies that could be implemented at full-scale in the future. The new technologies being explored include a floating solar park, a short-duration battery storage as well as long-term storage via green hydrogen produced by water electrolysis and subsequently consumed in a fuel cell. The combination of these individual measures, herein referred to as the Base Load Power Hub (BLPH) will ensure a continuous power supply regardless of the wind situation.

1.1.1 Project Location

The Hollandse Kust Noord (HKN) wind farm is located 18.5km off the coast of the Netherlands near the town of Egmond aan Zee.

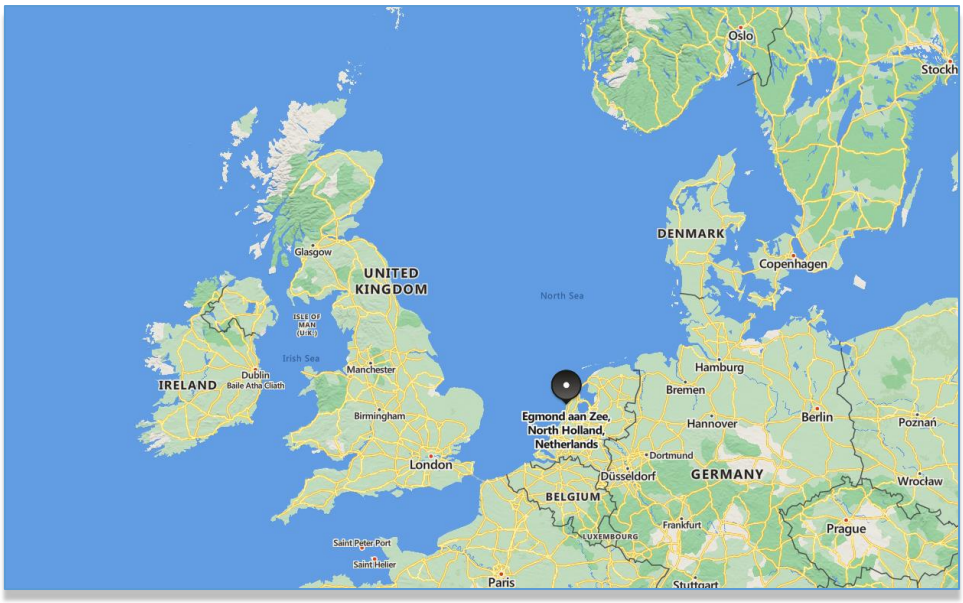


Figure 1 HOLLANDSE KUST NOORD WIND PARK Location Map

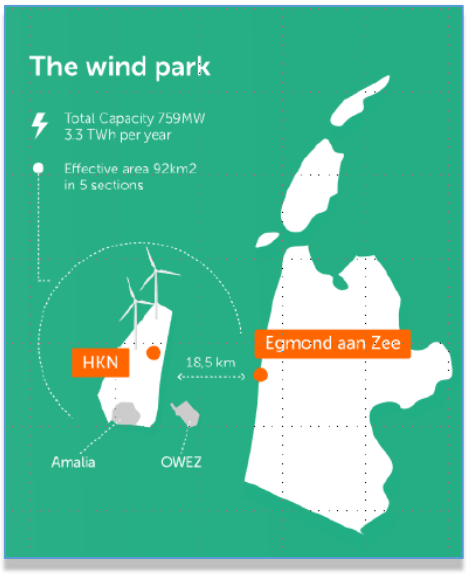


Figure 2 - Wind Park Details

The wind farm will consist of No. 69 x 11.5MW wind turbine generators each to be installed on a dedicated monopile structure. A 70th monopile located on coordinates E 590719.1, N 5841602.3 is reserved for installing the platform accommodating the BLPH. The monopile structure was designed and installed by Van Oord Offshore Wind (VooW) and therefore not part of the BLPH scope.

1.1.2 Project Concept

All components included in the BLPH will be integrated onshore on a topsides and installed in an offshore location within the boundaries of the HKN wind farm. At its simplest, the BLPH will store energy during periods of high electricity generation from renewable sources (a wind turbine generator and the floating solar PV farm) and release it during periods of low generation. In practice. It is expected that the energy stored will supplement generation from renewable sources to ensure that at least 1.0MW of power is exported to the electrical grid regardless of wind/solar availability.

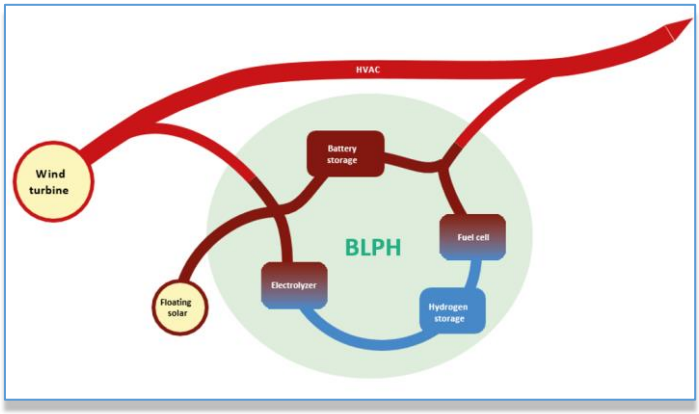


Figure 3 – BLPH Concept

Electricity in the BLPH will be stored in the form of electrons in a battery energy storage system (BESS) as well as converted into green hydrogen via water electrolysis. When the power output from the connecting wind turbine drops to below 1 MWe, the BLPH will return its stored energy to supplement the remaining power output. This will be by discharging the BESS, while stored hydrogen will be converted back into electricity using a fuel cell to extend the discharge duration.

The purpose of the BLPH s to function as a testbed for the integration of energy storage technologies with renewable generation. It is expected that equipment installed on the BLPH will operate as Normally Unmanned Installation for at least two (2) years with no or minimal maintenance and it will have its own independent control room with the ability to communicate with Crosswind's control room.

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2 DOCUMENT SCOPE

The analysis presented in this document is to verify the suitability of the crane structures to support the environmental loads and loads acting on the hook during operation. Calculation premises have been developed based on the following references.

2.1 REFERENCES

Doc. No.	Description
123A90-DDL-MAC-DS-50036	TDS – DAVIT CRANE
ISO 13852-1	Cranes - Offshore cranes
DNV-ST-0378	Standard for offshore and platform lifting Appliances
FEM - 1.001	Rules for design of hoisting appliances

Table 1 - Reference Documents

2.2 DEFINITIONS AND ABBREVIATIONS

COMPANY or Client (CPY):	Crosswind
CONTRACTOR (CTR):	ROSETTI MARINO S.p.A.
PROJECT:	HOLLANDSE KUST NOORD WIND PARK Project

Table 2 - Definitions and Abbreviations

2.3 GENERAL INFORMATION

Application type:	Wind offshore farm platform
Design Life:	2 years
Working temperature	-8°C to +25°C
Wind conditions	Annual avg @ 100m: 9.56 m/s
	Extreme for 1-year recurrence @ 100m: 33.1 m/s
Haz. site area classification	Safe

Table 3: General info

3 GENERAL DESCRIPTION

3.1 MAIN COMPONENTS LAYOUT

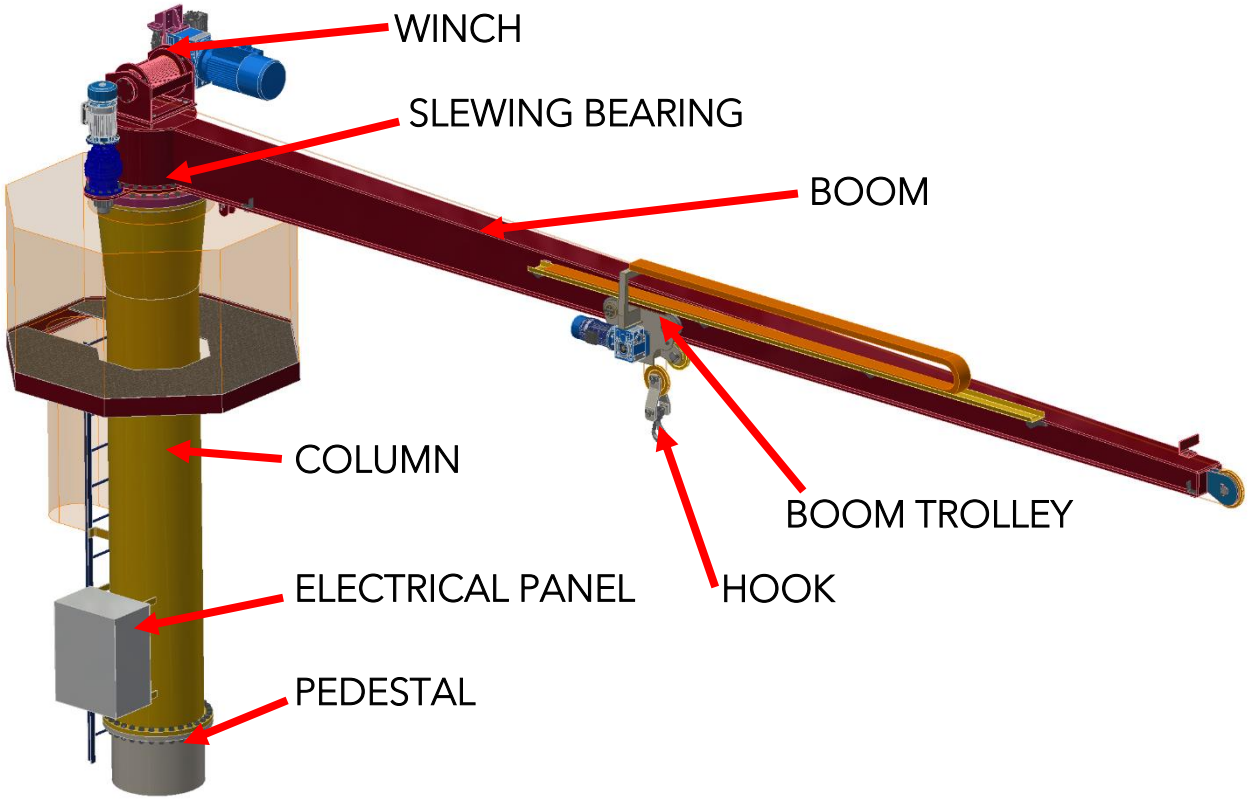


Figure 4: Main components

3.2 TECHNICAL DATA SUMMARY

TECHNICAL DATA		
Type	DAVIT CRANE	
Customer TAG		Y-XXX
General assembly	Drawing	C09-0001-A0002
DESIGN CRITERIA		DNV-ST-0378
Category of crane		Offshore crane
Ambient working temperature (min/max)	°C	-8°C to +25°C
Wind speed at davit crane[1]	m/s	Operative: 24 Out of service: 44
Crane static inclination angle heel/trim	deg	List / Trim = 0° / 0°
MECHANISMS CLASSIFICATION		
According to European Federation of Handling Industries FEM - 1.001 (10.1998)-Rules for design of hoisting appliances.	Equipment	A4
	Mechanism	M4
HOISTING		
General data		
Boom height from deck	m	4.85
Deck height from sea level	m	30.13
Max cargo lift	SWL	3.5 ton @ 5m | 1.0 ton @ 10m
Significant wave height	Hs	1.5
Hoisting 1st speed	m/min	3.75
Hoisting 2nd speed	m/min	15
Minimum Outreach	m	1.5
Maximum Outreach	m	10
Maximum hook stroke	m	40
Personnel handling		NO
AOPS		YES
MOPS		YES
Hook trolley (radial movement)		
1st speed	m/min	2

[1] Wind conditions have been considered as per DNV-ST-0378 Appendix A, Table A-2 to be more conservative.

TECHNICAL DATA		
2nd speed	m/min	8
Slewing		
Slewing 1st speed	rpm	0. 5
Slewing 2nd speed	rpm	1
Slewing range angle (continuous)	deg	+/-90°
Electric motors		
Installed power (N. of motors x unitary power)	kW	1 x 15 for winch 1 x 4 for slewing 2 x1.5 for trolley and auxiliaries

Table 4: Technical specification

3.3 INSTALLATION SITE

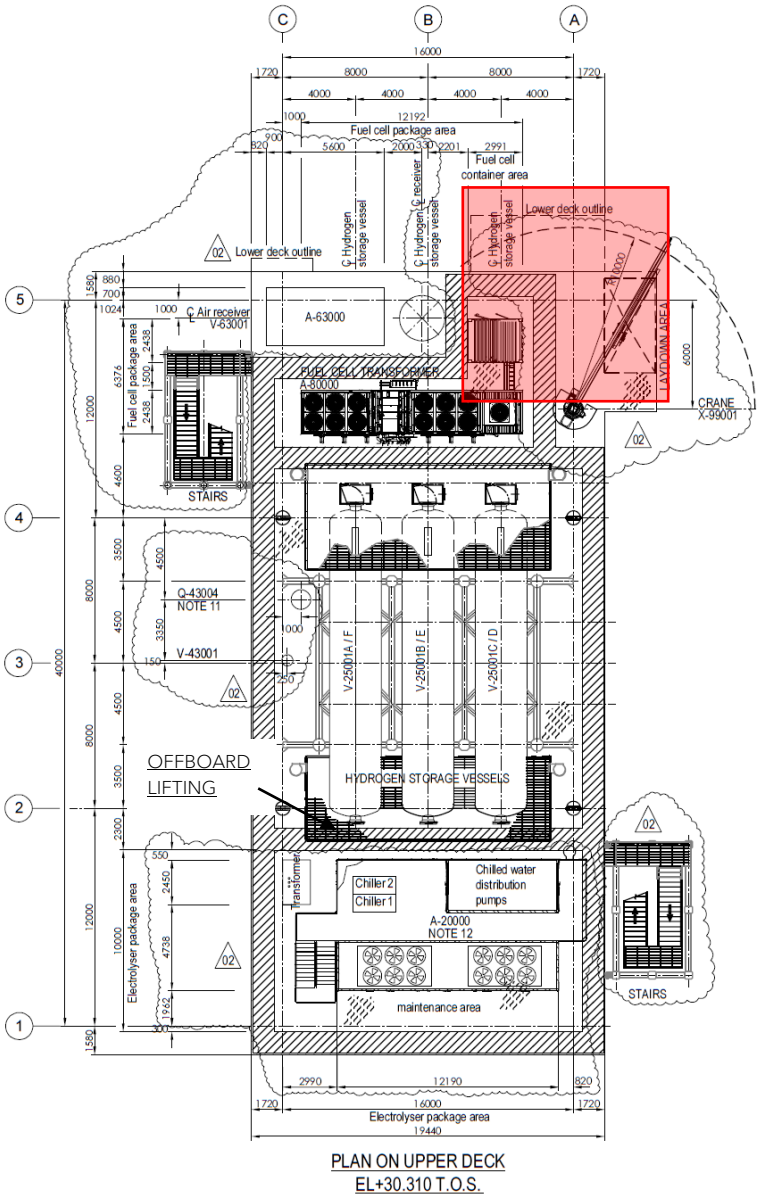


Figure 5: Installation site layout

4 CALCULATION CRITERIA

Structural calculation has been performed through analytical methods and Finite Element Analysis.

The Finite Element Analysis has been performed using a Finite Element Model in accordance with the drawings prepared for the construction.

The FEM software used for the calculation is ANSYS Mechanical from ANSYS, Inc. Canonsburg, PA, USA. ANSYS Mechanical is a general-purpose FEM code for liner/non-linear static and dynamic analysis of structures of any shape with any kind of load.

It's worth pointing out that ANSYS Mechanical code has been certified by NAFEMS (National Agency for Finite Element Methods and Standards in England).

The structural calculation will consider design conditions according to DNV-ST-0378, Sec. 4.

For strength calculation, assuming an elastic analysis (all the values below are taken according to DNV-ST-0378):

- Permissible stresses:

Permissible stress	Load case
$\sigma_y/1.50$	Load case I
$\sigma_y/1.33$	Load case II
$\sigma_y/1.10$	Load case III

Table 5: Safety factors

- Safety factors for buckling analysis:

Safety factor	Load case
1.86	Load case I
1.66	Load case II
1.38	Load case III

Table 6: Safety factors in buckling analysis

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5 DESIGN LOADS

The following load combinations are considered:

- Case I: Crane working without wind. $S_G + \psi * S_L + S_H$
- Case II: Crane working with wind. $S_G + \psi * S_L + S_H + S_W$
- Case IIIb: Crane subjected to exceptional loadings (in this case we will consider vessel's transit condition and out of service wind as exceptional loads). $S_G + S_M + S_{W_\max}$

S_G = loads due to dead weight of the components;

S_L = loads due to working load;

S_H = horizontal loads due to operational motions. In this case, where travel motion takes place only for positioning the crane and is not used for moving loads, the effect of this motion has not be combined with the effect of other motions;

S_W = loads due to working wind;

S_M = loads due to vessel's transit condition;

S_{W_max} = loads due to out of service wind;

ψ = dynamic amplification factor.

5.1 LIFTING CONFIGURATIONS

Four lifting configurations are considered:

	Conf. 1	Conf. 2	Conf. 3	Conf. 4	Unit
SWL	3.5	1.0	3.5	1.0	*ton*
Lifting condition	Offboard	Offboard	Onboard	Onboard	-
Outreach	5.0	10.0	5.0	10.0	*m*

Table 7: Lifting configurations

Each lifting configuration is evaluated according to DNV-ST-0378 by the loading cases I, II.

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5.2 DEAD WEIGHTS

Dead weights considered in the calculations are listed below.

Part	Mass
	kg
Slewing bearing	200
Pedestal + column	1500
Boom	2850
Winch	300
Hook block	100

Table 8: Dead weights

For the calculation, a contingency factor of 1.05 is applied to the masses.

5.3 VERTICAL LOADS DUE TO OPERATIONAL MOTIONS

Influence of operational motions in Z-direction is described by means of the dynamic amplification factor (DAF) Ψ, which depends on:

- Crane stiffness coefficient *C*
- Working load *W*
- Relative velocity between load and hook at the time of pick-up V_R

The amplification factor Ψ for each lifting configuration, calculated according to DNV-ST-0378 [4.1.3.2], is presented in Table 9, considering that:

$V_L = 15\ \frac{m}{min}$

$V_{in} = 0.6 * H_S = 0.9\ \frac{m}{min}$

$V_t = 0 \frac{m}{s}$

$V_H = 0.1 * (H_S + 1) = 0.25 \frac{m}{min}$

	Configuration 1	Configuration 2	Configuration 3	Configuration 4	Unit
Lifting type	Offboard	Offboard	Onboard	Onboard	-
Crane stiffness	96.5	58.8	178	81.6	*kN/m*
Vertical speed V_R	1.15	1.15	0.33	0.33	*m/s*
Ψ	1.6	1.8	1.3	1.3	-

Table 9: Dynamic amplification factor

The crane stiffness coefficient C has been calculated with a Finite Element Analysis. This stiffness is a consequence of the total deformation of the crane and the rope elongation.

Figure 6 presents the vertical deformation when applying the hoist load of 3.5 ton in the first configuration.

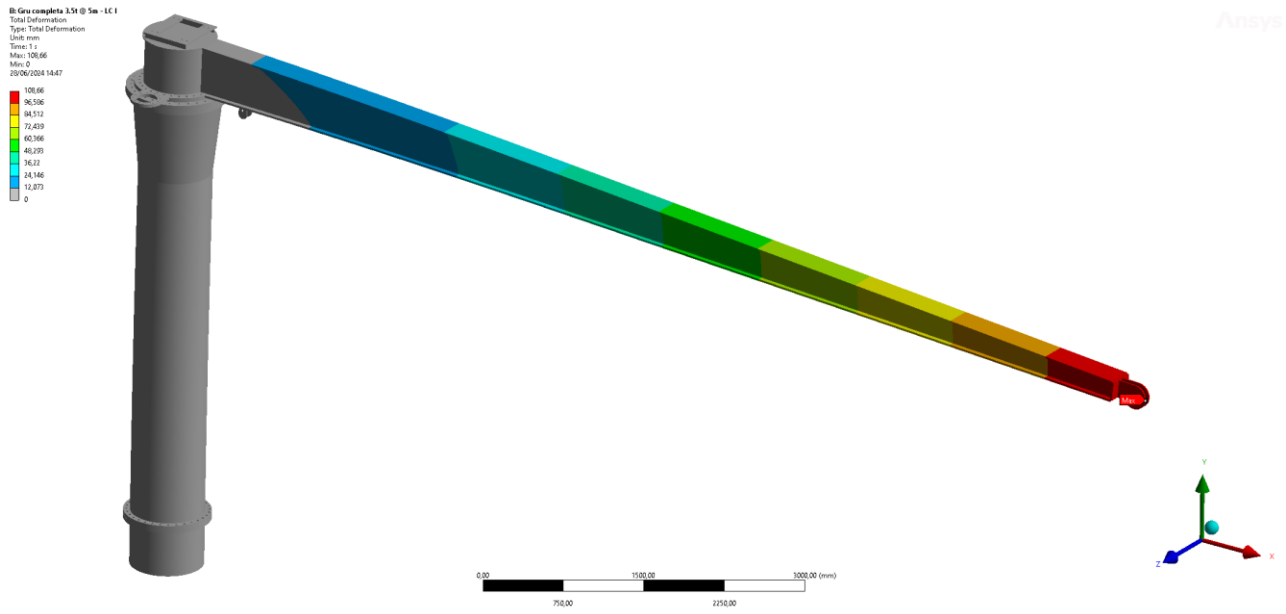


Figure 6: Crane stiffness calculation in Configuration 1

5.4 HORIZONTAL LOADS DUE TO OPERATIONAL MOTIONS

The horizontal loads due to operational motions to be considered are:

- Lateral forces (sidelead)
- Radial forces (offlead)

Horizontal forces are determined considering the following parameters:

	Value		Unit
Lifting configuration	3.5t @ 5m	1.0t @ 10m	
Rotation speed	0.5	0.5	*m/s*
Breaking time	1	1	*s*
Outreach	5	10	*m*
Vertical distance from jib sheave to vessel deck	35	35	*m*
Sidelead	1.5	0.5	*kN*
Offlead	7.6	2.6	*kN*

Table 10: Slewing data

5.5 LOADS DUE TO WIND EFFECTS

The wind load on the structure is calculated considering the following wind condition:

Wind condition	Wind speed V_{10} [*m/s*]	Velocity pressure q_{10} [*Pa*]	Velocity pressure q @ 35m amsl[2] [*Pa*]
Operating	24	353	442
Out of service	44	1186	1484

Table 11: Wind condition

where:

$q_{@\,35m} = q_{10}(0.9 + 0.01 * H)$

$H = 35m$

The wind load is calculated considering the lateral area and assuming drag coefficients C_d of each part (according to DNV-ST-0378, App. A, Tab A-1), as per the following Table 12.

[2] Amsl: Above mean sea level.

Part	Shape factor C_d	Lateral area m^2
Cargo	2.0	2
Pedestal column	0.7	3.5
Boom	2.0	3.9

Table 12: Shape coefficient for wind load

6 STRUCTURAL DESIGN AND STRENGTH

As per DNV-ST-0378 requirements, three load cases have been considered to properly conduct the strength analysis.

- LC I: Crane working without wind;
- LC II: Crane working with wind;
- LC III: Crane not working but subjected to exceptional loadings.

6.1 ANALYTICAL EXTERNAL LOADS

Here below the external loads considered for each configuration.

Lifting configuration 1					
Loads			Load Case I	Load Case II	Load case IIIb
Hoist	F_x	*kN*	7.6	7.6	0
	F_y	*kN*	1.5	2.9	0
	F_z	*kN*	56.5	56.5	0
Inertial	a_x	mm/s^2	0	0	0
	a_y	mm/s^2	0	0	0
	a_z	mm/s^2	9810	9810	9810
Wind	P	*Pa*	0	442	1484
Lifting configuration 2					
Loads			Load Case I	Load Case II	Load case IIIb
Hoist	F_x	*kN*	2.6	2.6	0
	F_y	*kN*	0.6	2	0
	F_z	*kN*	19.4	19.4	0
Inertial	a_x	mm/s^2	0	0	0
	a_y	mm/s^2	0	0	0
	a_z	mm/s^2	9810	9810	9810
Wind	P	*Pa*	0	442	1484

Lifting configuration 3					
Loads			Load Case I	Load Case II	Load case IIIb
Hoist	F_x	*kN*	45	45	0
	F_y	*kN*	1.5	2.9	0
	F_z	*kN*	45.9	45.9	0
Inertial	a_x	mm/s^2	0	0	0
	a_y	mm/s^2	0	0	0
	a_z	mm/s^2	9810	9810	9810
Wind	P	*Pa*	0	442	1484
Lifting configuration 4					
Loads			Load Case I	Load Case II	Load case IIIb
Hoist	F_x	*kN*	13.8	13.7	0
	F_y	*kN*	0.6	2	0
	F_z	*kN*	14	14	0
Inertial	a_x	mm/s^2	0	0	0
	a_y	mm/s^2	0	0	0
	a_z	mm/s^2	9810	9810	9810
Wind	P	*Pa*	0	442	1484

Table 13: External loads in the considered configuration

6.2 FEA MODEL

Note that the 3D model has been simplified from components that do not participate to stress resistance and stress propagation.

6.2.1 Mesh

Crane model is discretized with tetrahedral elements with of 10mm size, leading to the best results–computational duty ratio and a good level of mesh independency outcome.

6.2.2 Materials

Structural steel has been assigned to each component with the main mechanical characteristics as per following:

Material	Elastic modulus [MPa]	Density [kg/m^3]	Yield strength [MPa]	Plates thickness [mm]
Structural steel	210000	7810	355	5-16
Structural steel	210000	7810	345	16-40
Structural steel	210000	7810	335	40-60

Table 14: Material's mechanical characteristics

6.2.3 Connections

Connections between components are listed in the following table.

Part	Connection type
Boom – Slewing bearing	Bonded
Slewing bearing - Column	Bonded
Column – Pedestal	Bonded

Table 15: Connections used in the model

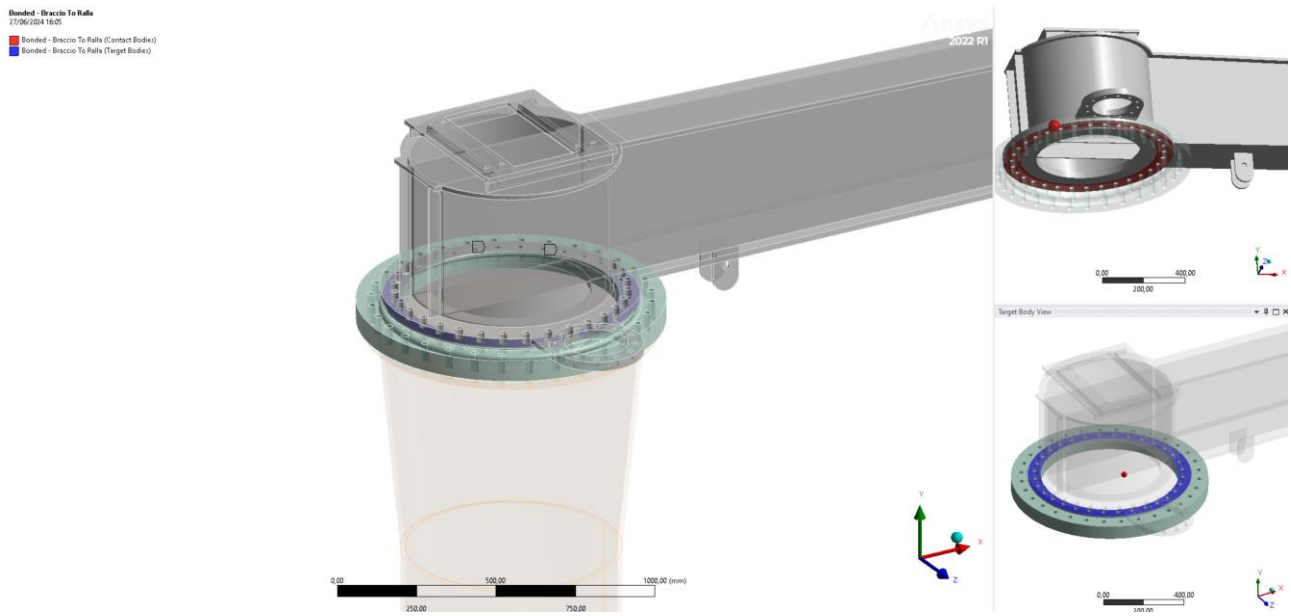


Figure 7: Boom – Slewing bearing joint

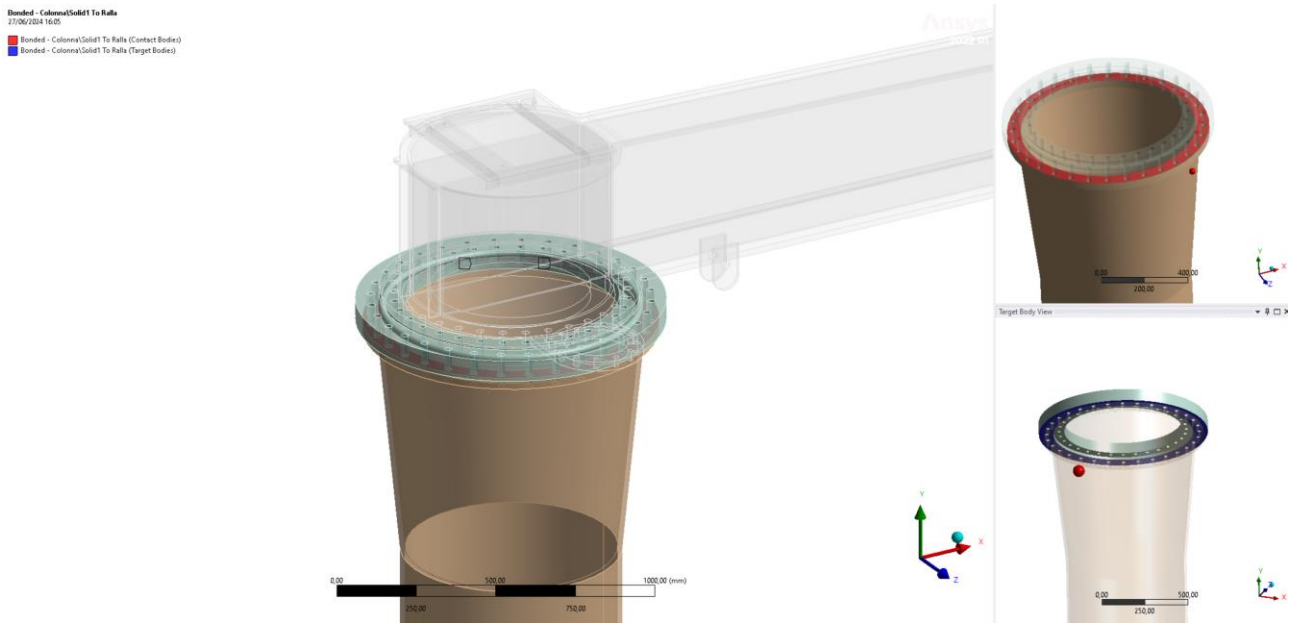


Figure 8: Slewing bearing – column joint

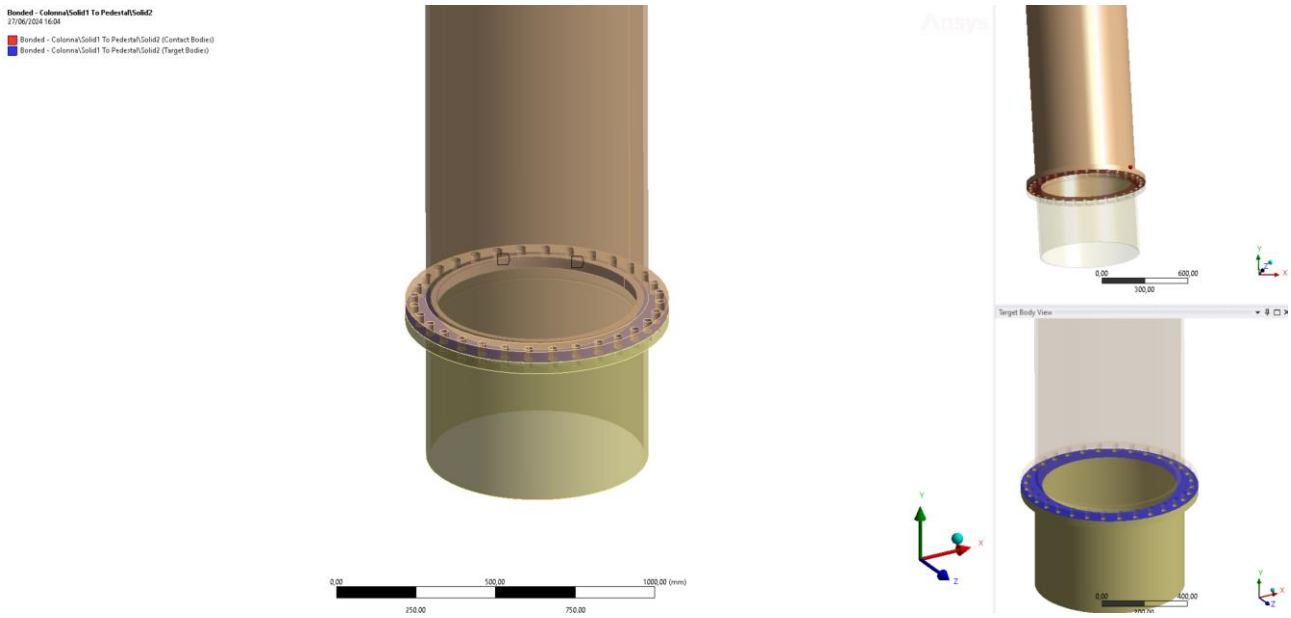


Figure 9: Column – pedestal joint

6.2.4 External loads

External loads considered:

1. Hoisted load ($SWL * \psi$) applied vertically on the 4 prints of boom trolley's wheels;
2. Half of hoisted load applied horizontally on the 2 cylindrical surfaces where the wire rope is fixed;
3. Hoisted load applied horizontally on the 2 cylindrical surfaces where the main pulley is assembled;
4. Half of hoisted load applied horizontally on the 8 holes where the winch is fixed, at 77.4mm from winch's fixing plate;
5. Wind pressure, related to LC II, applied on all lateral surfaces.

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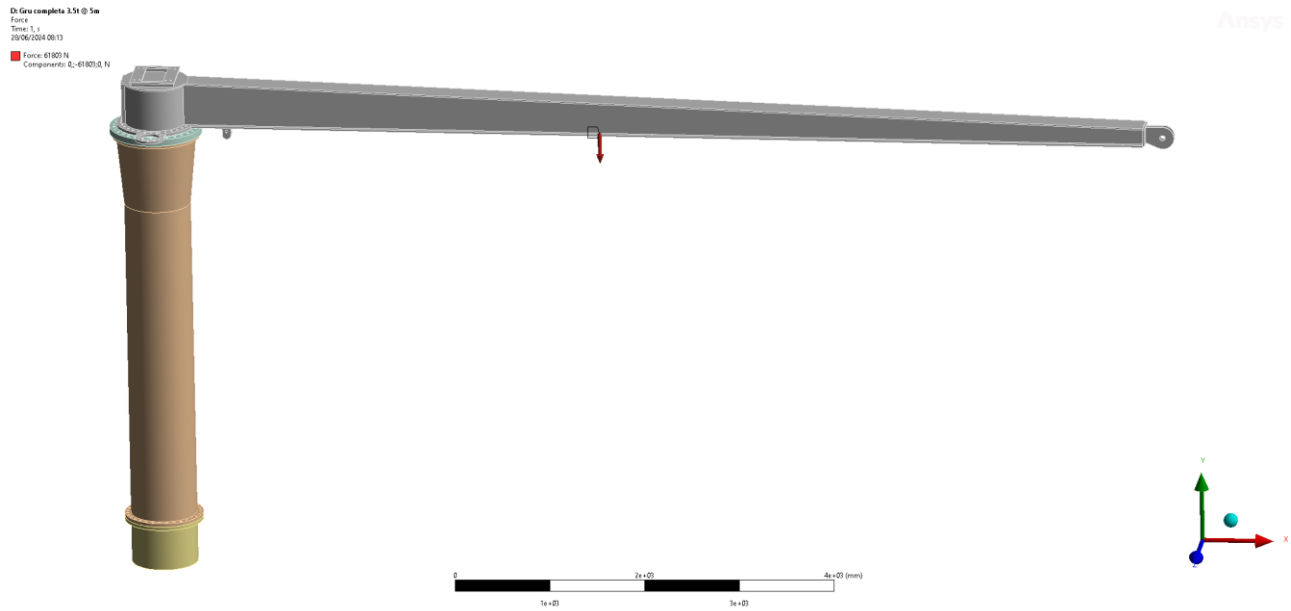


Figure 10: External load #1

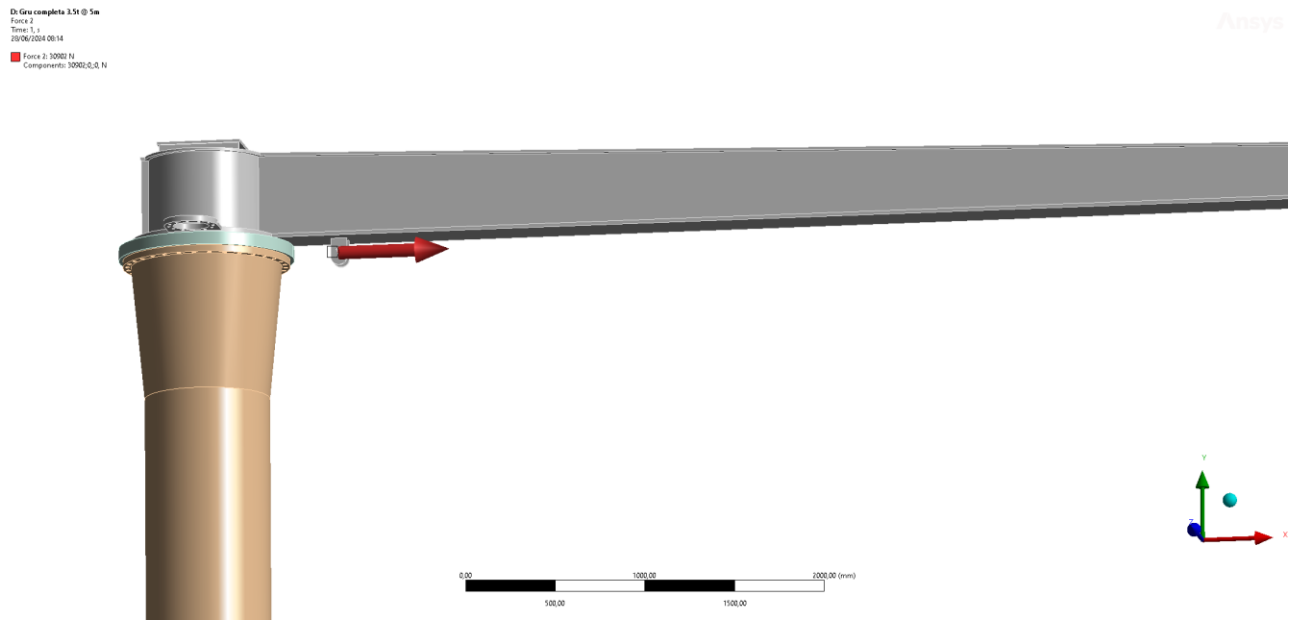


Figure 11: External load #2

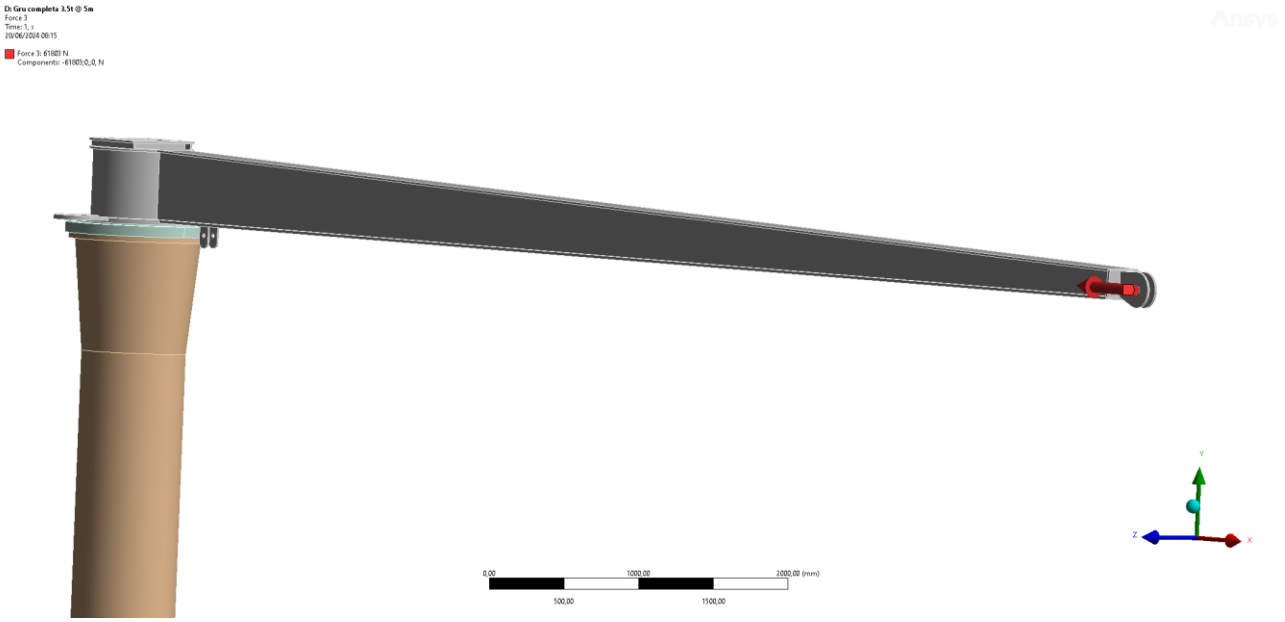


Figure 12: External load #3

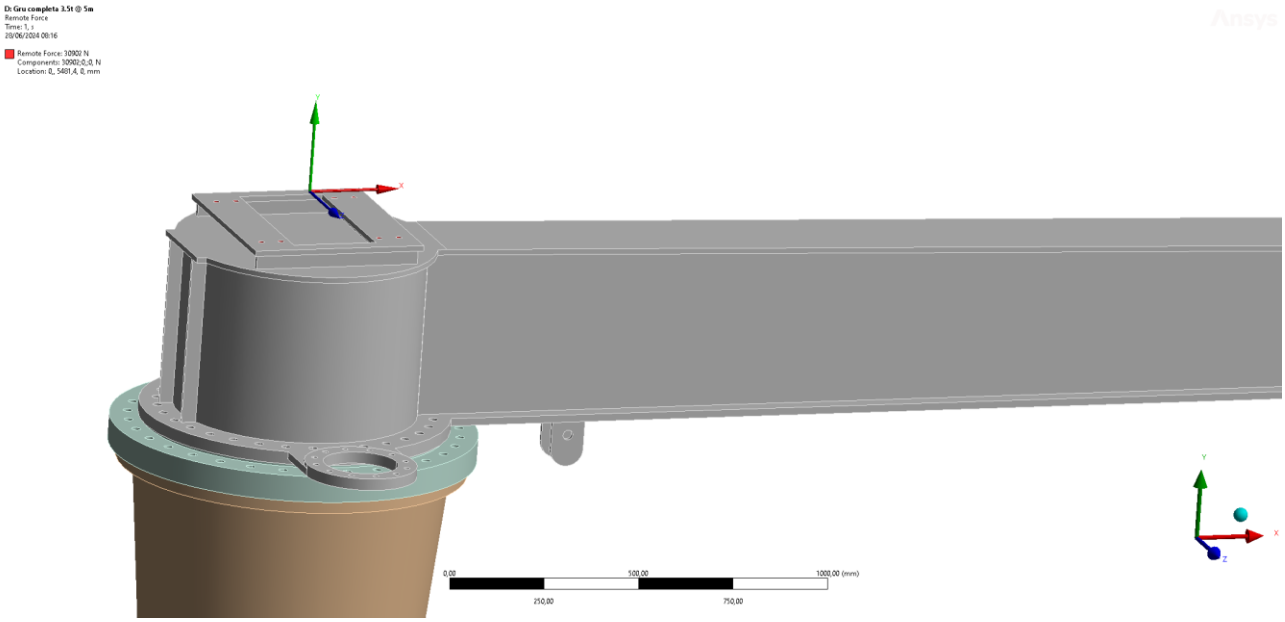


Figure 13: External load #4

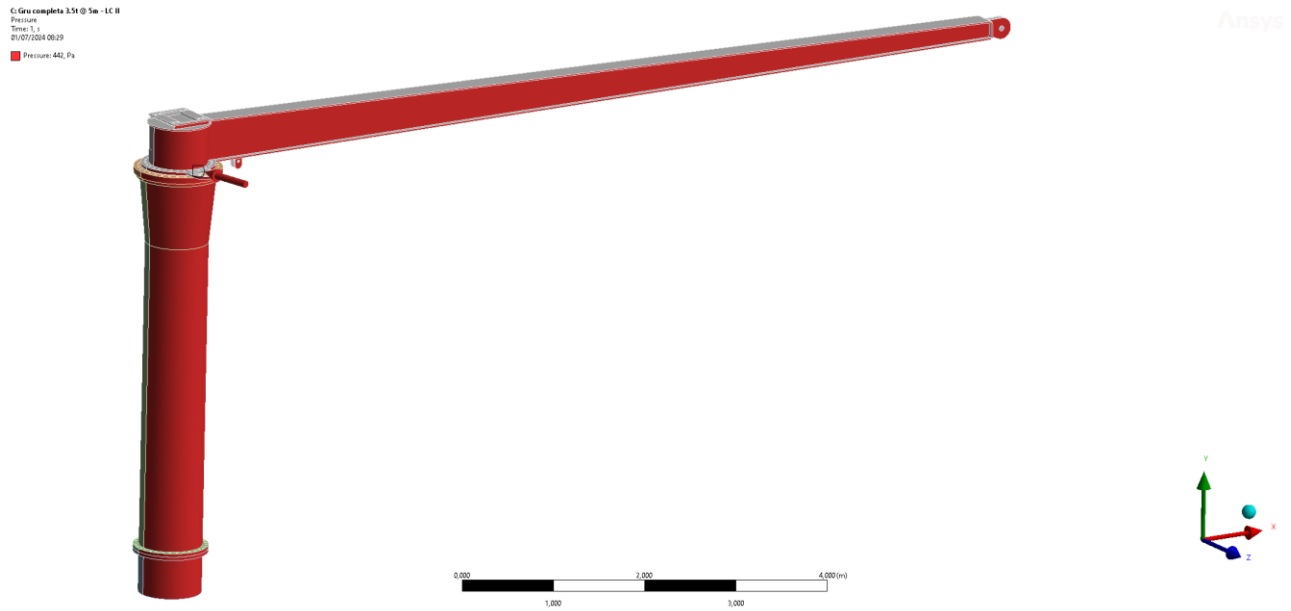


Figure 14: External load #5

6.2.5 Constraints

The crane is constrained with a fixed support applied to the column's flange footprint (translations and rotations locked).

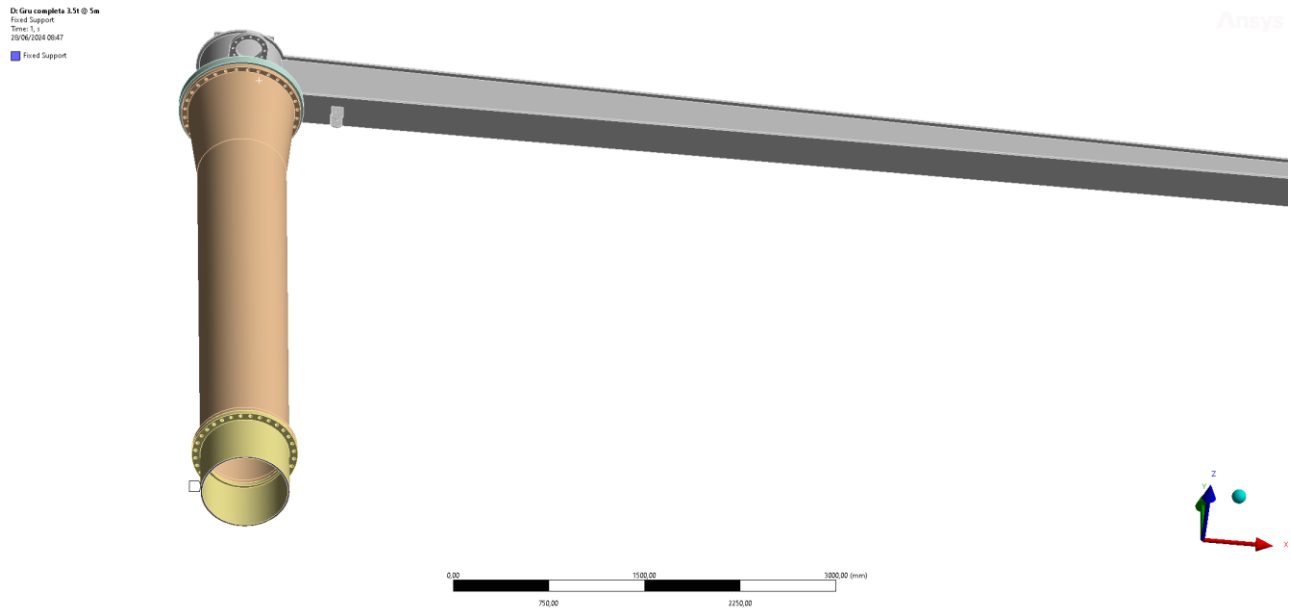


Figure 15: Fixed support

6.2.6 Permissible stresses

Permissible stresses have been assumed as per follows, according to Table 5:

Material	Permissible stress		
	Load Case I *MPa*	Load Case II *MPa*	Load Case IIIb *MPa*
S355 - EN 10025-2	236	267	323

Table 16: Permissible stresses considering safety factors for each load case

6.3 FEA RESULTS

Results of the FEA for LC I and LC II, in the first 2 different configurations, are presented below. The third load case will not be analyzed in detail because the hoist load is equal to 0, as presented in Table 13.

Only first 2 configurations (offboard condition) have been considered, because they are the highest stressing condition.

6.3.1 Stress and displacements

First configuration

Figure 16 and Figure 17 present the Equivalent von Mises stress and Total Deformation of the crane, for the load case I.

Figure 18 and Figure 19 for the load case II.

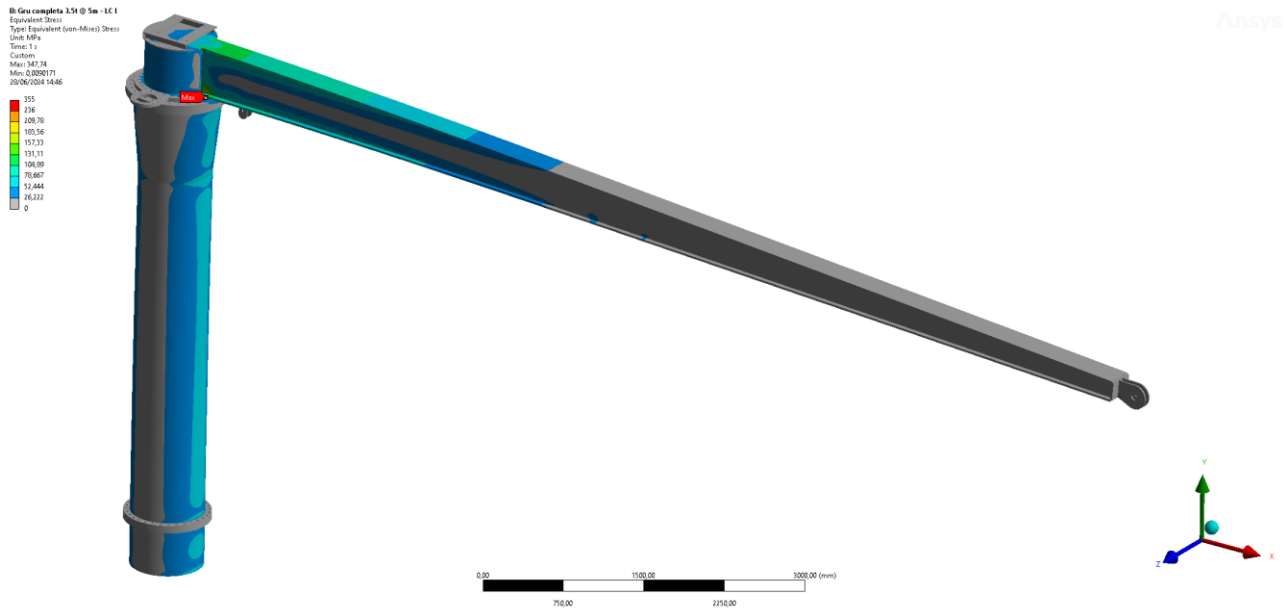


Figure 16: Equivalent Von Mises stress for Load Case I

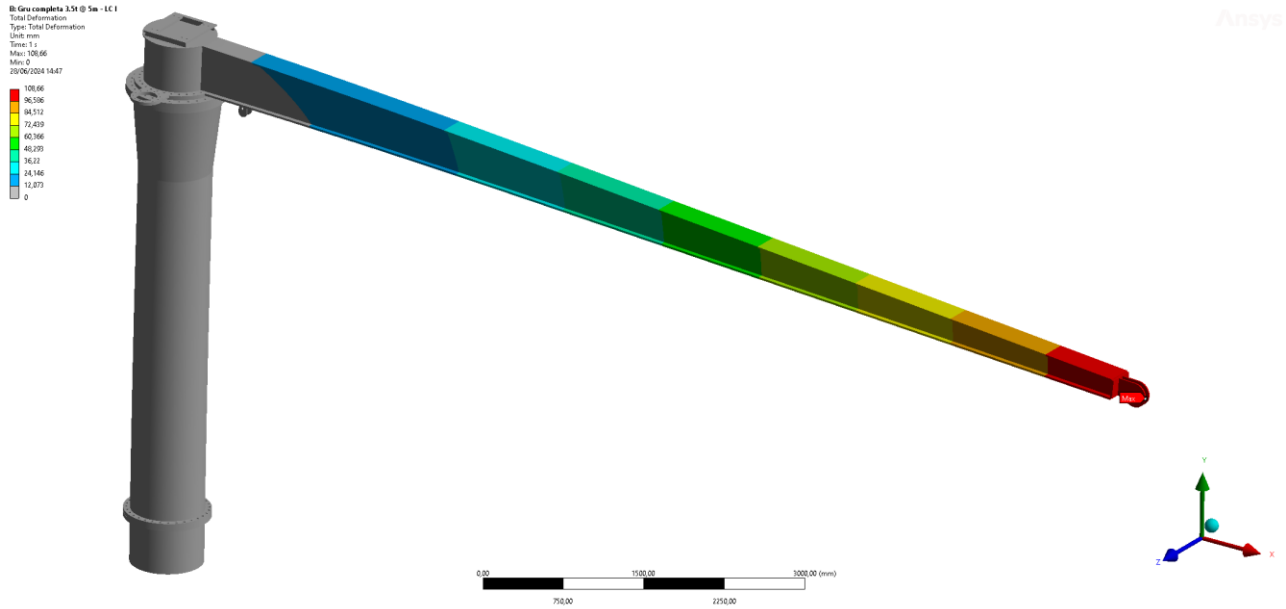


Figure 17: Total deformation for Load Case I

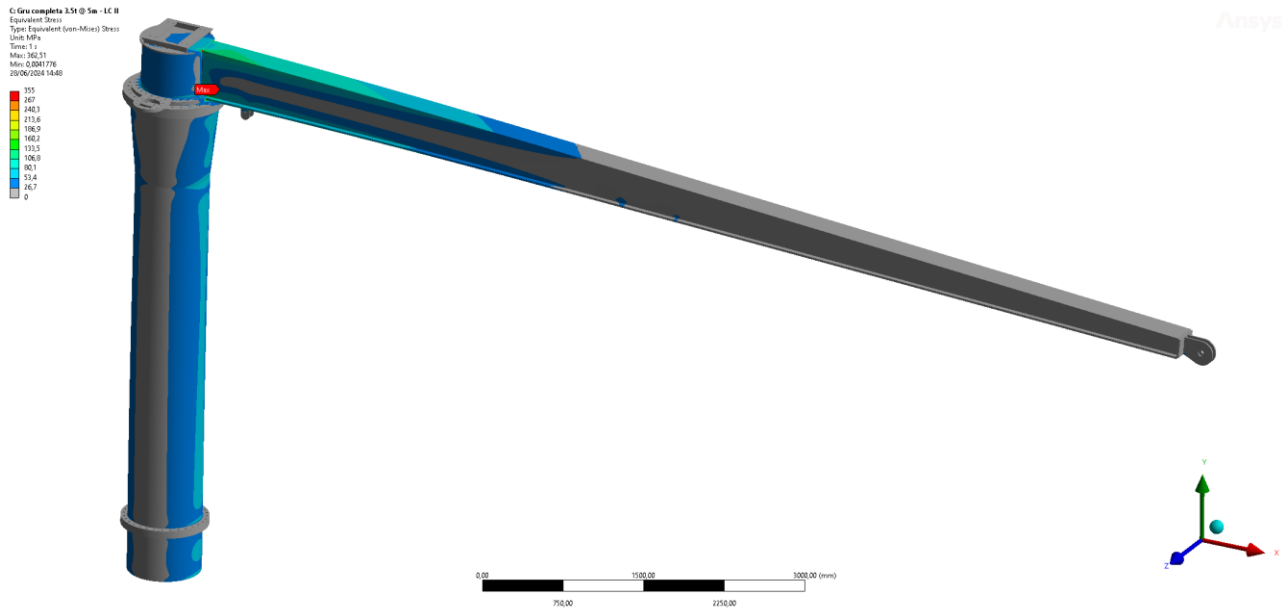


Figure 18: Equivalent Von Mises stress for Load Case II

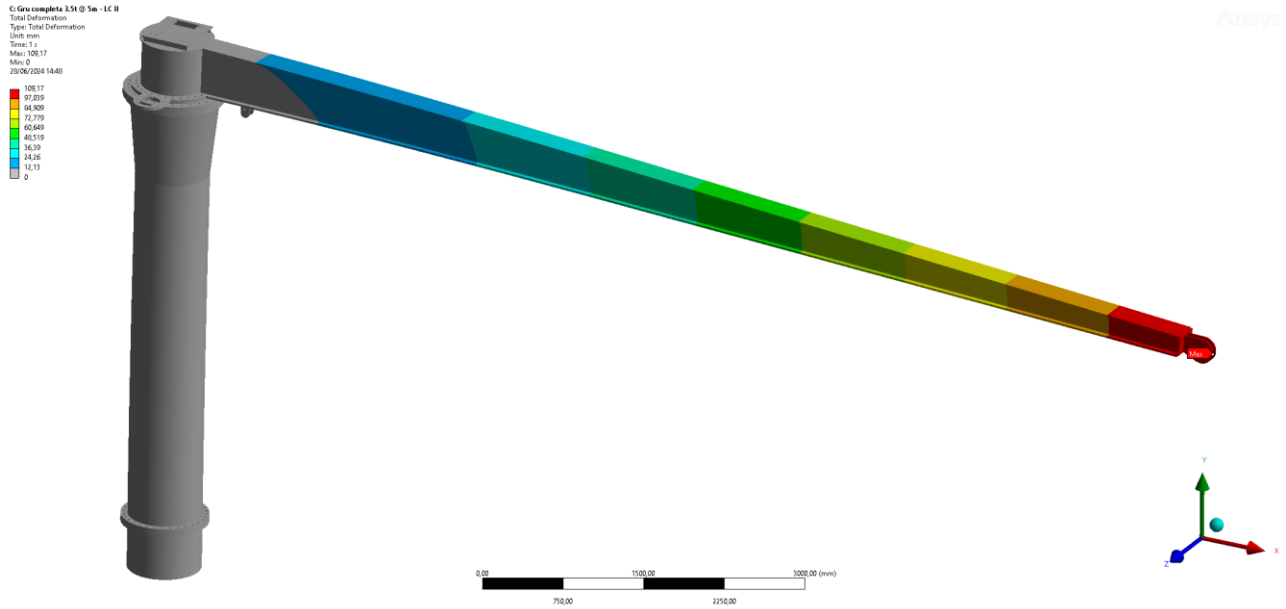


Figure 19: Total deformation for Load Case II

Second configuration

Figure 20 and Figure 21 present the Equivalent von Mises stress and Total Deformation of the crane, for the load case I.

Figure 22 and Figure 23 for the load case II.

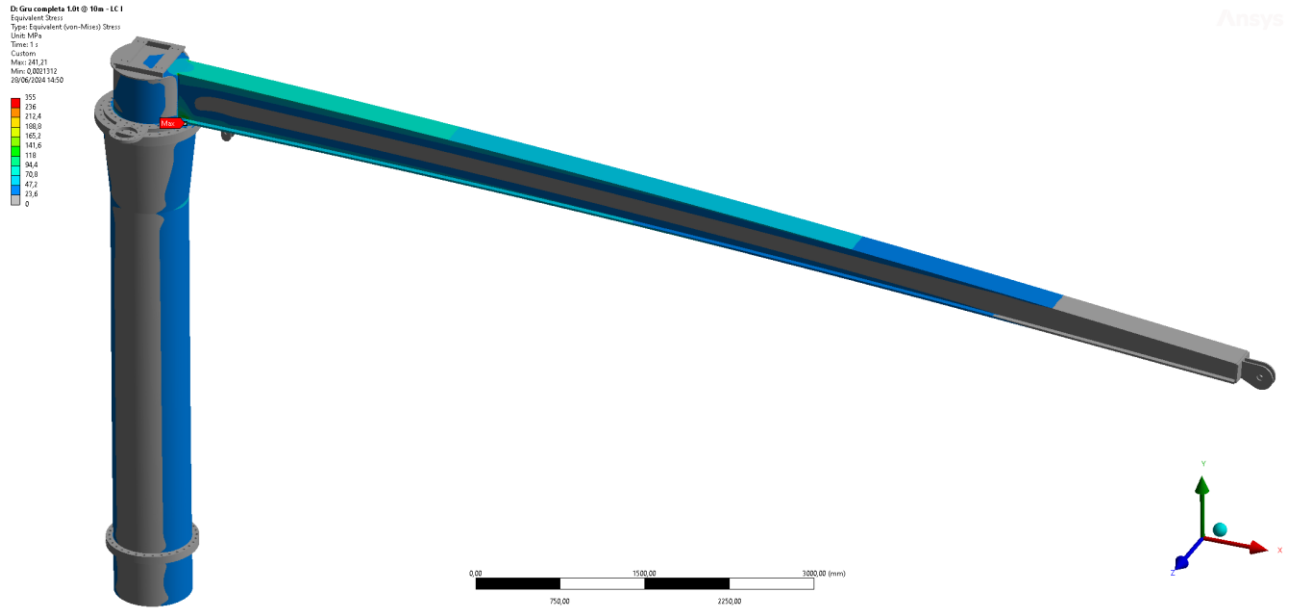


Figure 20: Equivalent Von Mises stress for Load Case I

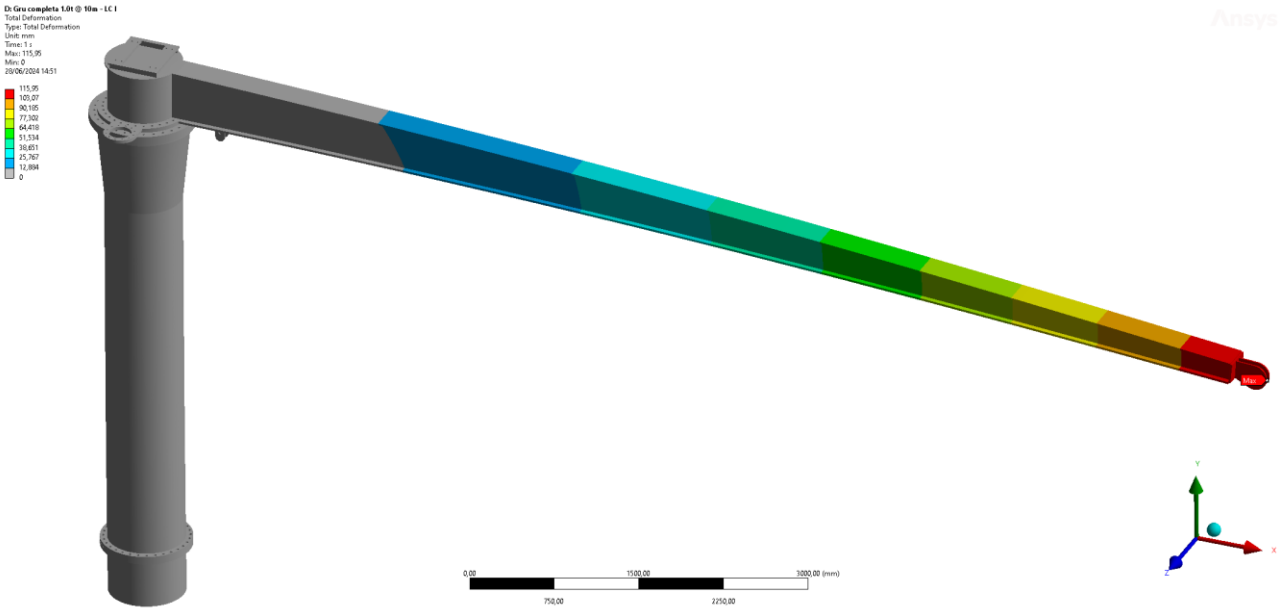


Figure 21: Total Deformation for Load Case I

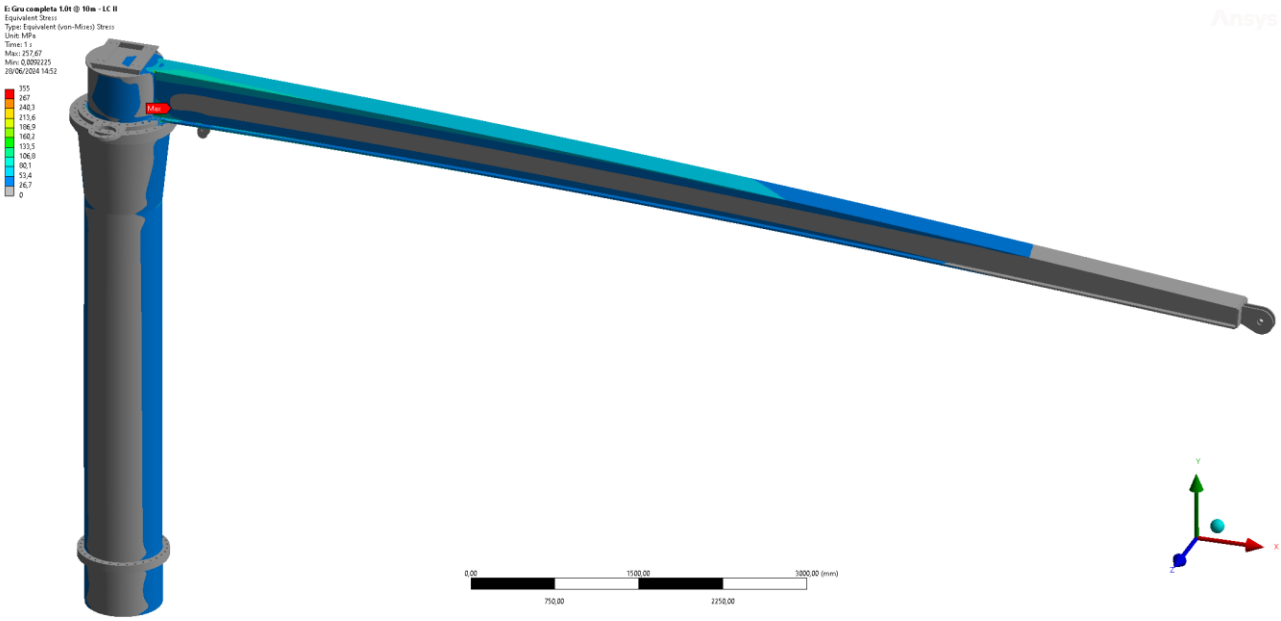


Figure 22: Equivalent Von Mises stress for Load Case II

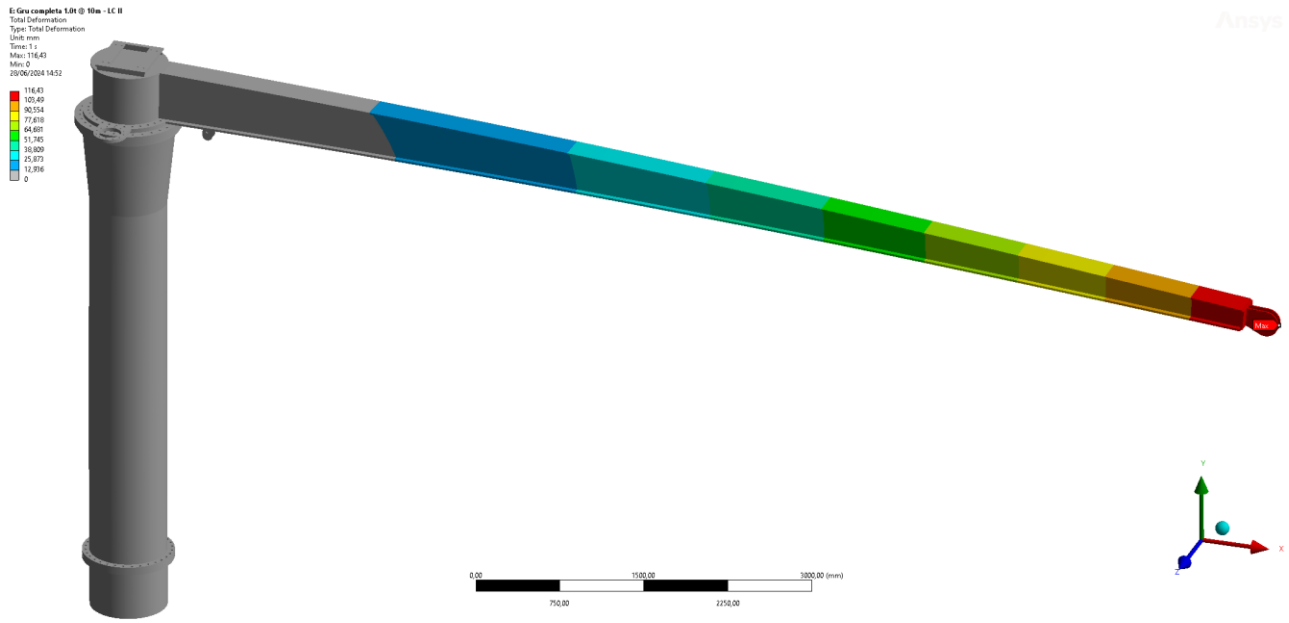


Figure 23: Total Deformation for Load Case II

6.4 BUCKLING ANALYSIS

The following analysis, limited to lifting configurations 1 and 2 in LC II condition, considers different safety factors, as you can see in Table 17.

Type of buckling	SF		
	Load case I	*Load case II*	*Load case III*
Elastic buckling	1.86	1.66	1.38

Table 17: Safety factors in buckling analysis

A linearized approach has been adopted for carrying out the analysis, being guided by the standard *DNV-RP-C208: Determination of structural capacity by non-linear finite element analysis methods*.

All the values are collected in Table 18.

		LC II	
Material		Config 1	Config 2
Yield Strength/Stress [MPa]	fy	355	355
Material factor	γM	1	1
Applied load from linear analysis [kN]	SREP	1	1
First buckling mode	kg	4,1	6,9
Representative stress from linear static analysis [MPa]	σREP	66,5	40,2
Critical buckling stress [MPa]	σki	273	277
Reduced slenderness	λ	1,14	1,13
Coefficient	α	0,15	0,15
Coefficient	ϕ	1,22	1,21
Type of buckling	-	1	1
Buckling factor (buckling curve)	k	0,60	0,61
Buckling resistance [kN]	Rd	3,22	5,39
Safety factor	SF	3,22	5,39

Table 18: Buckling analysis values

γ_M : material factor assumed as 1, if yield strength is considered, or 1.25, if tensile strength is considered.

S_{REP} : normalized applied load from linear analysis so to have a direct comparison to buckling resistance.

$Type\ of\ buckling$: 1, if column and stiffened plate and plate without redistribution possibilities, 2, if plate with redistribution possibilities, 3, if shell buckling. In this case, for a more conservative analysis, type 1 is assumed.

$Buckling\ resistance$ value means that the structure withstands a load of buckling of 3.22 times applied load, for example in configuration 1.

As you can see, the safety factor (assumed equal to the buckling resistance) is clearly higher than the safety factor seen in Table 17.

6.5 INTERFACE LOADS

The reaction forces have been calculated for each load case. Forces have been evaluated at the pedestal footprint. The Table 19 reports the max values among all load cases and all lifting configurations. These data were calculated analytically.

At pedestal footprint	Unit	LC I	LC II	LC III	Max
R_rad	*kN*	8	11	16	*16*
Rz	*kN*	112	112	52	*112*
M_tilt	*kNm*	432	432	113	*432*
Mz	*kNm*	8	29	44	*44*

Table 19: Loads envelope at the slewing bearing

6.6 SLEWING BEARING DATA

Slewing bearing has been chosen with the following characteristics:

Max bearing loads	Unit	Max static verification
Vertical load	*kN*	120
Radial load	*kN*	20
Tilting moment	*kNm*	510
Torque moment	*kNm*	50

Table 20: Max bearing loads

So, in order to be more conservative (on fact the loads at slewing bearing interface will be lower than the ones at pedestal footprint), you can compare these values with the ones shown in Table 19, obtaining the following safety factors:

	SF
Vertical load	1,1
Radial load	1,3
Tilting moment	1,2
Torque moment	1,2

Table 21: SF for bearing loads

6.7 LIFTING

6.7.1 Rope check

The tables presented in this section summarize the technical data of the *Wire Rope* chosen for the crane, the calculated *Rope Pull* and the *Safety Factor*.

Wire rope		
Wire rope type	Sirtef Funi Verope	
Rope diameter	11	mm
Wire rope MBF	115,5	kN
Linear weight	0,59	kg/m
Elastic modulus	91037	Mpa
Rope area	70,13	mm2

Table 22: Wire rope main data

Max weight to be lifted	P	kN	34
Weight to be lifted (including hook weight)	SWL	kN	35,3
Pulley efficiency	h	--	0,985
Number of falls	N1	--	2
Number of pulleys	N2	--	3
Number of ropes on drum	N3	--	1

Rope Pull (RP)	RP	kN	18,6
Requested safety factor	η	--	5
Actual safety factor	η	--	6,2

Table 23: Wire rope verification

6.7.2 Winch gearbox check

The hoisting winch gearbox has the following characteristics:

Hoisting gearbox data		
Drum nominal diameter	253	mm
Drum nominal torque	3,1	kNm
Drum max torque	5,1	kNm
Gearbox nominal torque	4	kNm
Gearbox max torque	6	kNm
SF at nominal torque	1,3	-
SF at max torque	1,2	-

Table 24: Hoisting gearbox data

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6.8 PEDESTAL SCREWS CHECK

Input screws data			
Screw nominal diameter	d_n	24	mm
Minimum bolt area	A_i	353	mm2
Bolts class	B_C	8.8	-
Bolt Yield stress	fy	640	N/mm2

Table 25: Pedestal screws data

Loads data		
Force X	8	kN
Force Y	16	kN
Force Z	96	kN
Momentum X	2	kNm
Momentum Y	413,0	kNm
Momentum Z	44,0	kNm

Table 26: Screws loads data

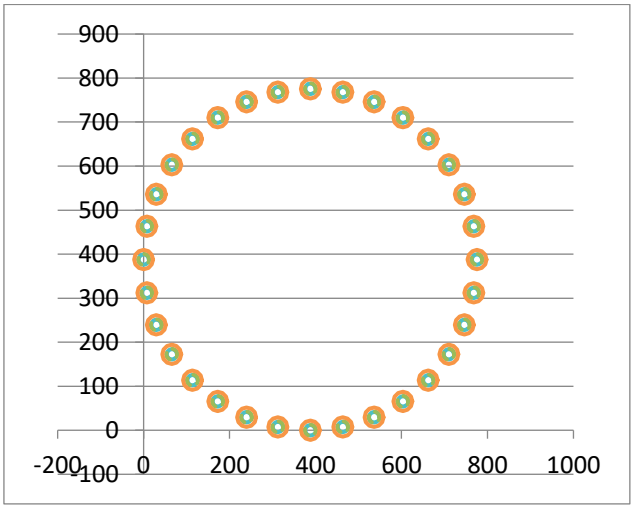


Figure 24: Bolts schematic representation

Results		
Due to preload + external loads		
Max Axial force on bolt - Fa	69,6	N
Preload force - N_s'	121,4	N
SF on Detachment (>0,1)	0,9	OK
VM stress on screw	494	N/mm2
SF on VM stress	1,3	OK
Due to preload + external loads		
Max stress due to Mx	1	N/mm2
Max stress due to My	189	N/mm2
Max stress due to Fz	8	N/mm2
Max total stress	198	N/mm2
Safety factor compared to Sy	3,2	OK

Table 27: Screws results

Screws check can be considered ok, as presented in table above.

6.9 ROTATION GEARBOX CHECK

Table 28 shows the main characterizing torque data of gearbox for slewing movement and the related safety factors in max static condition.

	Unit	
Max crane static torque moment	*kNm*	44
Gearbox – pinion ratio	-	8,5
Max gearbox static torque moment	*kNm*	10
SF at max static torque	-	1,9

Table 28: Slewing gearbox torque data

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6.10 MAIN PIN CHECK

An analysis has been performed in order to check the bending and shear resistance of main pins: boom's and hook's pulley ones.

Both pins are made of AISI 630 structural steel, with the following characteristics:

	Unit	
Yield strength	*MPa*	725
Ultimate strength	*MPa*	860

Table 29: AISI 630 structural steel main mechanical data

6.10.1 BOOM'S PULLEY PIN CHECK

Considering the forces scheme below, the results are collected in Table 30.

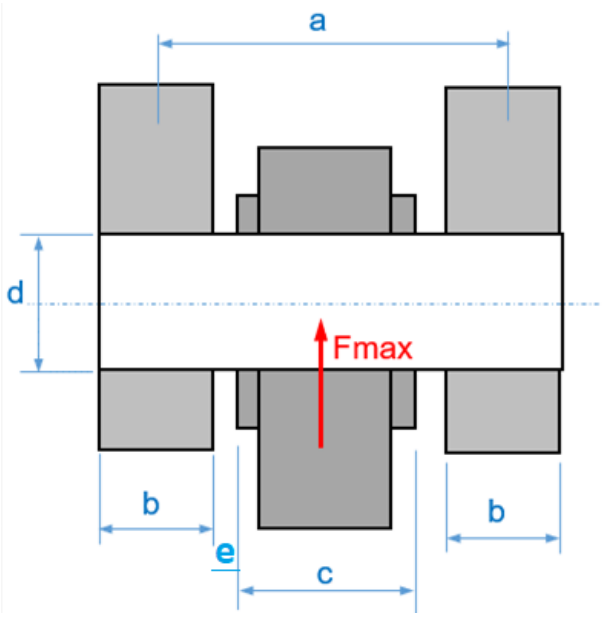


Figure 25: Pin forces scheme

Data	Unit	
a	*mm*	63,5
b	*mm*	15
c	*mm*	46
d	mm	60
e	mm	1,25
Rope pull on single arm	kN	28
Winding angle on pulley	-	180°
Resulting force	kN	56,5
Reaction force on each support	kN	28,3
Maximum bending moment	kNmm	897
Resistance modulus	mm^3	21206
Bending stress	MPa	42
SF at bending stress	-	17,1
Shear force	kN	56,5
Pin's area	mm^2	2827
Shear stress	MPa	20
SF at shear stress	-	36,3

Table 30: Boom's pulley pin check

6.10.2 HOOK'S PULLEY PIN CHECK

Considering the scheme in Figure 25, the results are collected in Table 31.

Data	Unit	
a	*mm*	54,5
b	*mm*	16
c	*mm*	36
d	mm	35
e	mm	1,25
Rope pull on single arm	kN	28
Winding angle on pulley	-	180°
Resulting force	kN	56,5
Reaction force on each support	kN	28,3

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Maximum bending moment	kNmm	770
Resistance modulus	mm^3	4209
Bending stress	MPa	183
SF at bending stress	-	4
Shear force	kN	56,5
Pin's area	mm^2	962
Shear stress	MPa	59
SF at shear stress	-	12,3

Table 31: Hook's pulley pin check

6.11 TRANSPORTATION ANALYSIS

An analysis has been performed to evaluate the values of stress on structures during the transportation phase, that is when the crane is already installed on platform and this is transported to the final installation off-shore site.

The following boundary conditions have been applied:

- 0.7g, as transversal acceleration;
- -0.5g, as vertical acceleration (this has to be added to gravity acceleration);
- Same constraint as previous simulation (see Sec. 6.2.5)

6.11.1 Stress and displacements results

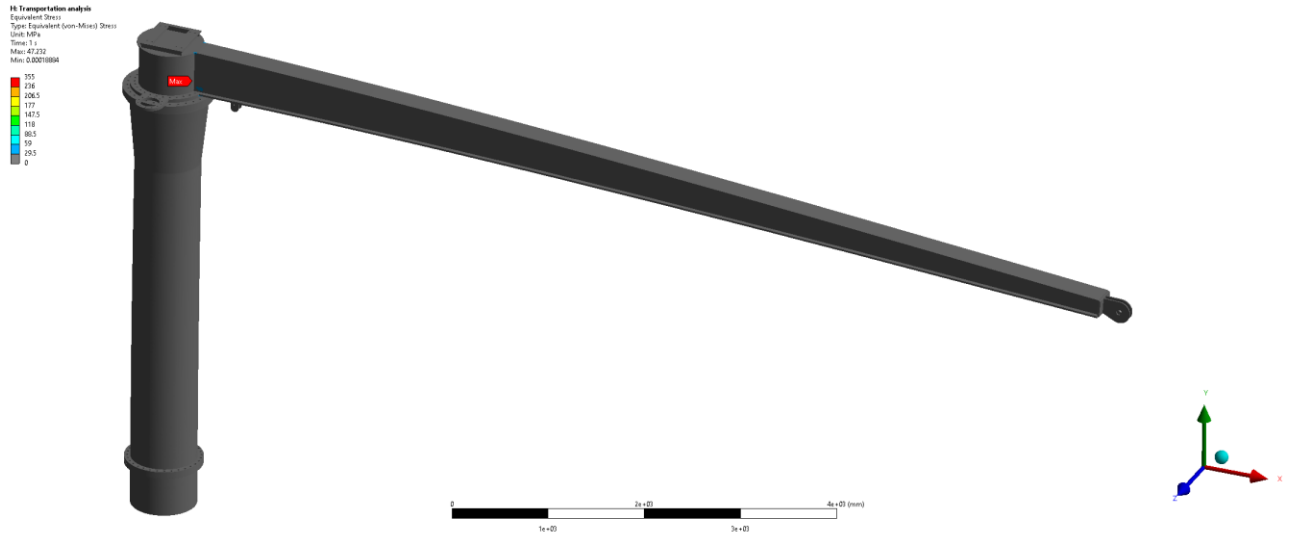


Figure 26: Equivalent Von Mises stress

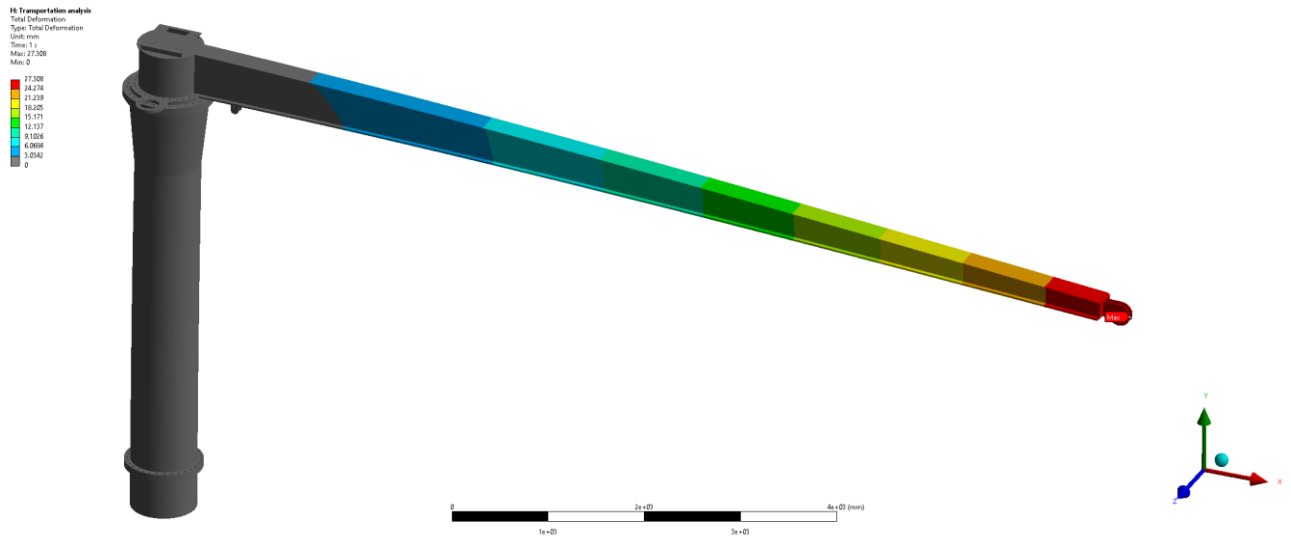


Figure 27: Total deformation

As you can see in pictures above, no relevant stress or deformation occur during transportation phase and no boom rest system is necessary.

Remazel Engineering S.p.A.