Modal Analysis of Gantry Crane and Superstrucutre Made by Steel and Alluminium Alloy

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The aims of this research are twofold. The first one is to design an innovative machine consist of a gantry crane and superstructure as a loader crane, and the second one is to implement an innovative material for this machine, using aluminium alloy instead of steel for the loader crane arms. Another objective concerns the study of the dynamic behaviour of the crane with two different material configurations.
Due to this machine's composition, the lifting equipment exhibits a high degree of versatility. The design process was developed using an analytical method to estimate the loads of each component of the machine, followed by solid modelling and numerical analyses. The main results show that the weight of the loader crane arms made of aluminium alloy is approximately 50% lighter than those made of steel, with dynamic behaviour very similar for both solutions

Keywords: *Innovative crane, gantry crane, loader crane, structural lightening, modal behaviour.*

1. INTRODUCTION

In the current global context, cargo handling is beco–ming increasingly significant. A clear example of this is the recent incident in the Suez Canal, where the groun–ding of a container ship caused a massive traffic jam, leading to significant port congestion for an extended period [1]. Moreover, the growing trend of outsourcing production to companies, particularly in Eastern Europe, has further emphasized the importance of transportation and material handling, especially in port operations.

This research focuses on the study and design of an original crane that offers exceptional versatility for handling both bulk materials and containers.

The crane consists of a gantry crane with an additional loader crane mounted on it, capable of adapting to various geometric configurations and handling diverse load conditions.

The primary aim of this research is the design of the entire machine, with particular attention to its dynamic behaviour. For certain components, such as the loader arms, aluminium alloy is used in place of structural steel to take advantage of its lightweight properties.

The aluminium alloy began to be used for crane structures in the mid of 20th century, due to two important advantages: the first one its lighter weight compared to steel and the second one the high corrosion resistant capability. Some preliminary research are reported in the [2,3,4].

This advanced solution provides the added benefit of enabling the crane to operate not only on a port quay for efficient cargo handling, but also on ships for loading, unloading, or transferring cargo between vessels.

This flexibility is especially valuable in situations where a large ship cannot access a port directly, thus requiring adaptable handling capabilities.

2. PROCESSING

The maximum payload capacity is directly related to the position of the loader crane arms. Specifically, when the arm is fully extended, reaching its maximum distance of 30 meters from the axis of rotation, the payload capacity is 100 kN. On the other hand, at shorter distances, the payload increases, reaching up to 350 kN. Figure 1 illustrates the main geometric parameters and the payload chart for this machine.

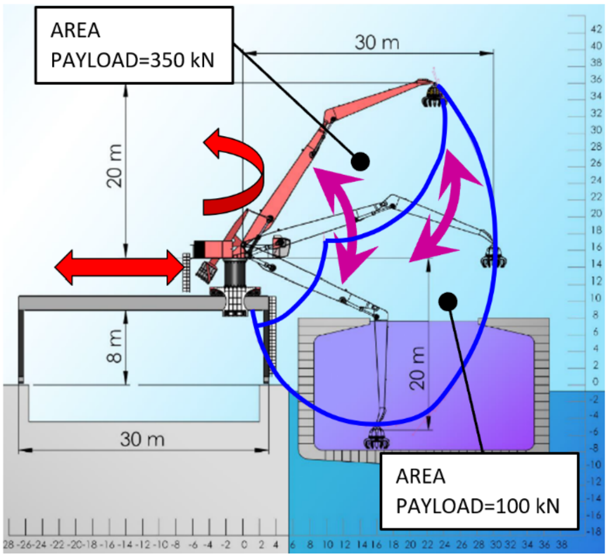

Figure 1. Crane specifications: main movements and load charter.

The grab is a classical component with the inner vo–lume when the valves are close equal to 0.5 m^3 . Its

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weight is 9 kN. This research begins with a comprehensive study of the specifications required for the entire machine, considering a wide range of factors and load conditions, including the payload, dead load, and other variables. In particular, these types of cranes are subjected to various external forces, such as wind [5,6], seismic activity [7,8] (which the response is very influenced by the damping magnitude coefficient [9]), inertia forces resulting from the movement of the load [10,11], or the motion of machine components (e.g., arms, trolley) [12-15]. Additionally, when the crane is placed on a ship, the forces induced by the ship's motion (waves) must also be considered [16-18].

The machine offers high versatility due to the position of the loader crane on the gantry crane, its orientation, and the positioning of the loader crane arms.

Figure 2 shows the loader crane positioned at the end of the Gantry crane, oriented parallel to the main beam, while Figure 3 illustrates two of the most critic geometric configurations for the boom and stick. In synthesis, the configuration a) corresponds to the maximum bending action on the stick, while the configuration b) corresponds to the maximum bending action on the boom.

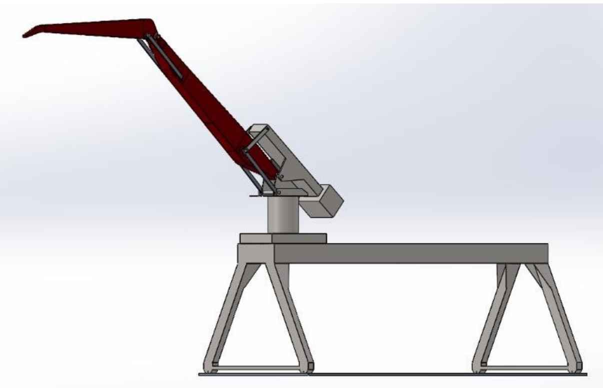

Figure 2. Example of the position of the loader crane (and its arms) on the Gantry crane.

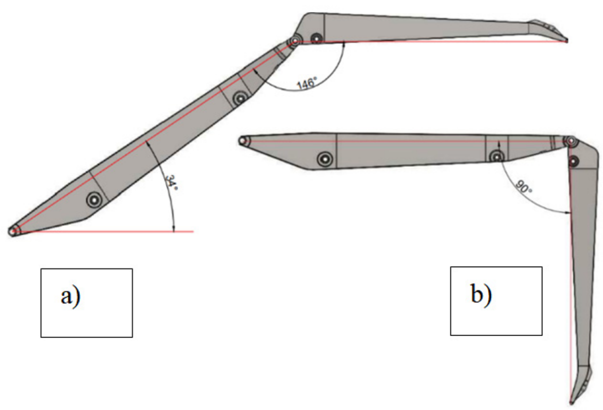

Figure 3. Two critical geometric configurations.

By a specifically developed program in Mathcad® software, the forces, moments, and other actions operating on each component of the machine (such as the loader crane arms, hydraulic cylinders, gantry crane beams, etc.) were estimated. This step was followed by the sizing and verification of each element of the machine.

2.1 Steel material

Table 1 shows the main mechanical properties of the steel used in the design of this machine, specifically a classical structural steel (S355 JR EN 10027-1). This material was chosen due to its good mechanical properties, commercial availability, ease of welding, and cost-effectiveness.

Table 1. Main mechanical properties for the steel adopted.

Thickness [mm]		Mechanical properties		
>	≤	R [N/mm^2]	Reh [N/mm^2]	A%
0	3	510-680	355	22
3	16	470-630	355	22
16	40	470-630	345	21
40	63	470-630	325	21

2.2 Criteria

The main criteria adopted for the design of this machine are as follows: a safety factor of $\eta = 2.5$ was applied with respect to yield strength, fatigue limit ($n = 2\times10$□ cycles, with pulsating loads from zero), and buckling phenomena (both local and global).

The dynamic effect, which increases the action applied to the machine, in absence of deeper information is assumed equal to 1.2 i.e. the payload was increased by 20%.

This force acts in the vertical plane that is defined by the stick and boom arms.

Displacement, and consequently stiffness, is a critical parameter in the design. It was assumed that the maximum displacement (δ) must be lower than a specific value in relation to the length of the elements, as shown in equation (1).

This value was assumed to be good practice as a compromise between the values of the machine's own first frequencies and the size of the machine

$$\delta \le \frac{L}{250} \tag{1}$$

2.3 Sizing process for the boom arms

The procedure described in the previous chapter leads to the sizing of the main crane components (boom and stick of loader crane).

Figure 4 shows the main geometric size of the boom and the plates thickness that compose the element.

Figure 5 shows the same results for the stick (main geometrical dimensions and the plates thickness).

Figure 6 reports the results from numerical analyses carried out by finite element method through Simulation of SolidWorks® software.

The maximum displacement for each elements (stick and boom) is about 30 mm which is lower that the one defined in the equation (1); in fact for the stick the maximum displacement may be (2):

$$\delta \le \frac{14000}{250} = 56\,\text{mm} \tag{2}$$

while for the boom the maximum displacement may be (3):

$$\delta \le \frac{17500}{250} = 70\,\text{mm} \tag{3}$$

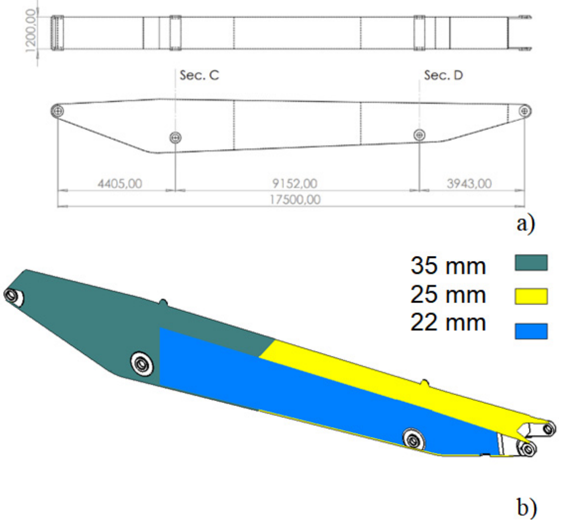

Figure 4. Boom arm: a) main geometrical dimensions [mm]; b) plates thickness that compose the boom

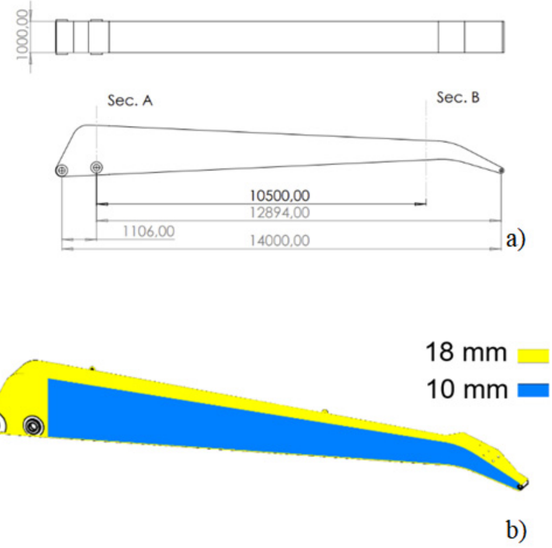

Figure 5. Stick arm: a) main geometrical dimensions [mm]; b) plates thickness that compose the stick.

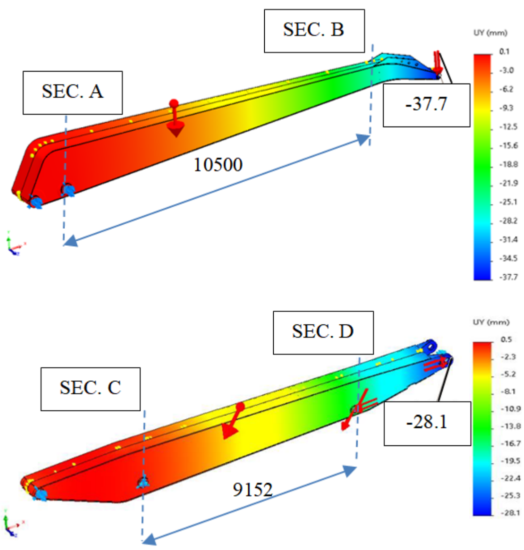

Figure 6. Displacement of the Stick and Boom arms in the Figure 3 a) geometrical configuration [mm].

2.4 Sizing of the main elements that compose the loader crane

As shown in Figures 1 and 2, a counterweight arm is placed above the boom arm (its weight is about 200 kN). The length of this arm is 9 meters, and the counterweight is 500 kN, designed to balance 80% of the maximum overturning moment at the base of the tower. Following the process used to design the arms, Figure 7 presents the solid model and the main dimensions for the counterweight arm.

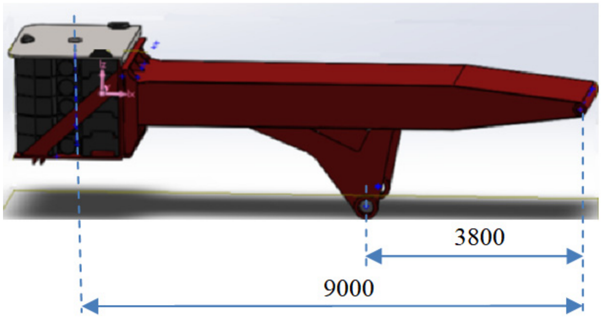

Figure 7. Counterweight [mm].

Figure 8 shows the chassis, where the boom and counterweight are attached, as well as the cabin and the machine room for the loader crane.

Below this assembly, a fifth wheel is positioned on a specific tower.

The dimensions of the tower, which is a cylindrical tube, are as follows: height = 3000 mm, inner diameter = 2400 mm, and thickness = 30 mm.

Figure 8 also illustrates the stress state at the chassis structure due to the arms attached to it. The maximum stress value observed in the chassis is approximately 100 MPa.

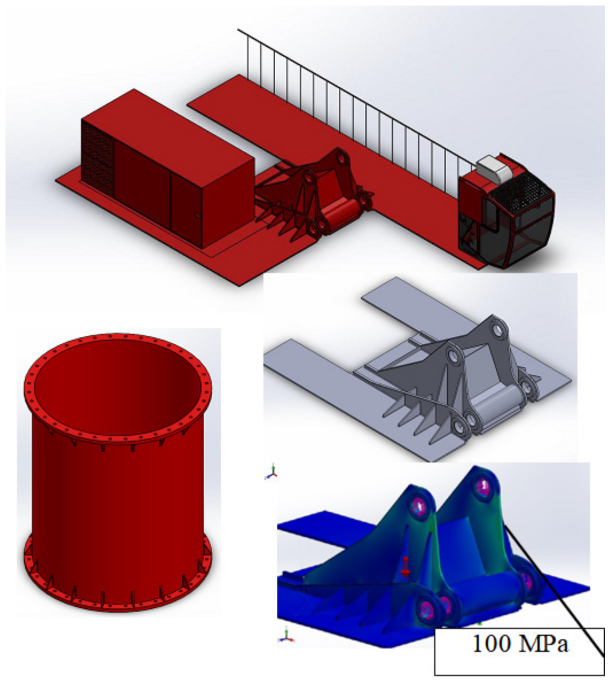

Figure 8. Chassis and the tower of the loader crane.

The tower described above is fixed on the trolley (figure 9) which move on the two beams of the Gantry crane.

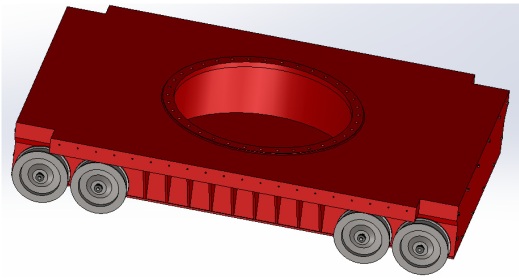

Figure 9. Trolley placed under the tower.

The primary forces acting on the trolley include vertical forces and the overturning moment generated by the loader crane.

To secure the trolley to the two beams of the gantry crane, four rails were installed: two on the upper plate and two on the lower plate of the gantry crane beams, preventing the trolley from overturning.

Different wheels were placed on these rails to allow the loader crane's translational movement.

Additionally, to ensure that the loader crane does not oscillate and moves in a straight line, four more rails were installed inside the two beams of the gantry crane.

This solution is illustrated in Figure 10.

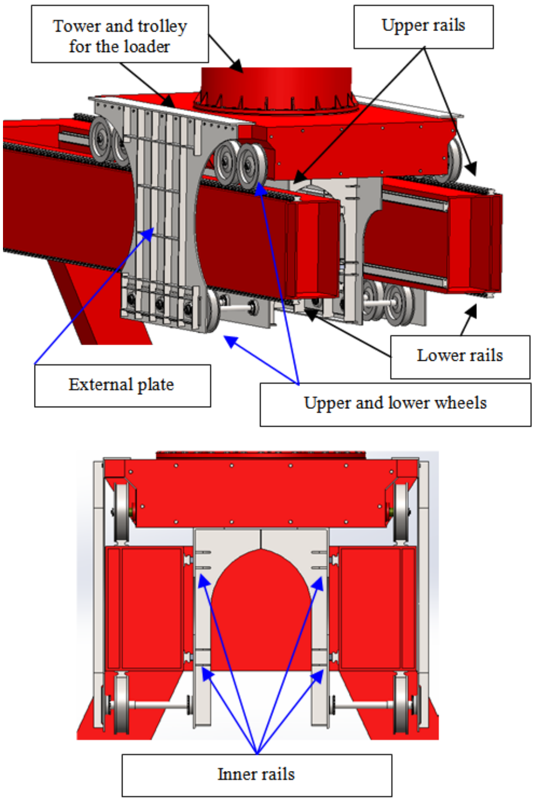

Figure 10. System to boundary the loader crane to the beams of the gantry crane.

2.5 Gantry crane

Figure 11 shows the loader crane positioned on the gantry crane in two specific geometric configurations. The gantry crane is a traditional structure consisting of two beams. Each beam has the following dimensions: height = 1600 mm, width = 1000 mm, plate thickness = 30 mm, and length = 30 m. Inside each beam, additional plates or beams are included to prevent local buckling phenomena.

Figure 11. Whole machine with loader crane in two specific geometrical configuration.

Numerical analyses were performed by applying a specific “block” to the gantry crane to simulate the forces induced by the loader crane, including both dead and payload forces. These analyses were conducted using SolidWorks Simulation through the finite element method. Figure 12 presents some of the results.

The stiffness of the beam is a critical parameter. As shown in the figure 12, the maximum displacement for the first load condition (figure 12 a)) is approximately 20 mm, while for the second load condition (figure 12 b)) is about 50 mm. In general, for all geometric configurations, the total displacement varies from 5 mm to 50 mm on the gantry crane.

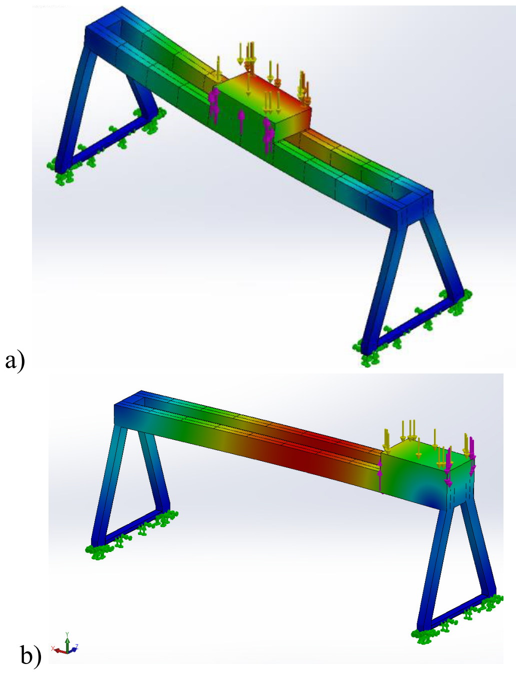

Figure 12. Gantry crane results (displacement) from different load / geometrical configurations [mm]: a) displacement for figure 12 a) geometrical configuration; b) displacement for figure 12 b) geometrical configuration.

3. MODAL ANALYSIS OF THE STEEL CRANE

After sizing the entire machine, various numerical analyses were conducted to estimate the dynamic behaviour of the crane, focusing specifically on the first natural frequencies.

Figures 13 and 14 illustrate the first deformation mode of the machine for each natural frequency.

These figures depict the displacement trends of the crane corresponding to different vibration modes and do not represent the structure's response to a specific time-varying action.

Therefore, the figures do not provide quantitative values but highlight which parts of the crane experience the most significant vibrations at specific frequencies.

Tables 2 and 3 show the vibration frequencies and mass coefficients.

These values represent the the contribution of each mode to the dynamic response of a structure; these factors are more important to assess which modes are more significant for dynamic behaviour of the crane.

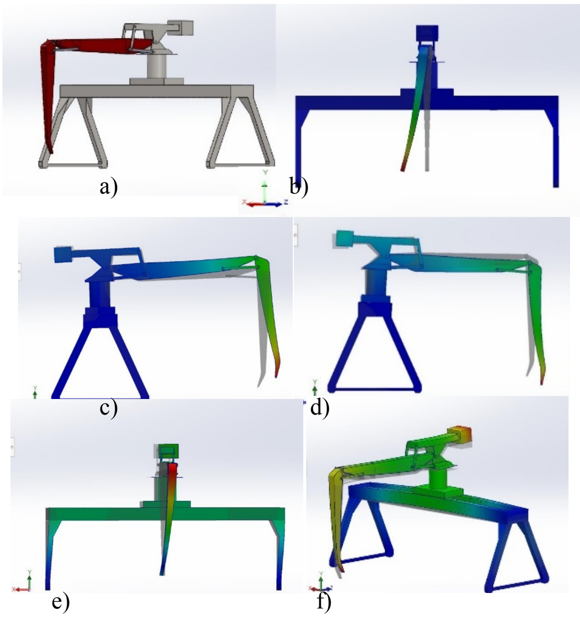

Figure 13. Fist five vibration mode of the crane made by steel material in specific geometrical position.

Table 2. Firth values of natural frequencies and mass coefficients participation factors for the vibration modes reported in the figure 11.

Num ber	Freq. [Hz]	X_Dir.	Y_Dir.	Z_Dir.
1_b	1.14	0.062	2.11E-06	6.65E-07
2_c	1.73	2.10E-06	0.026	3.34E-03
3_d	2.44	4.64E-07	3.37E-03	0.305
4_e	2.90	0.840	3.11E-05	5.65E-07
5_f	3.46	3.06E-05	0.681	0.020

Table 3. Firth values of natural frequencies and mass coefficients participation factors for the vibration modes reported in the figure 12.

Num ber	Freq. [Hz]	X_Dir.	Y_Dir.	Z_Dir.
1_b	1.00	0.067	1.70E-04	3.11E-04
2_c	1.25	1.63E-04	0.042	0.046
3_d	2.85	5.06E-03	0.064	0.338
4_e	2.96	0.846	5.87E-04	2.37E-03
5_f	3.74	4.22E-05	0.619	0.064

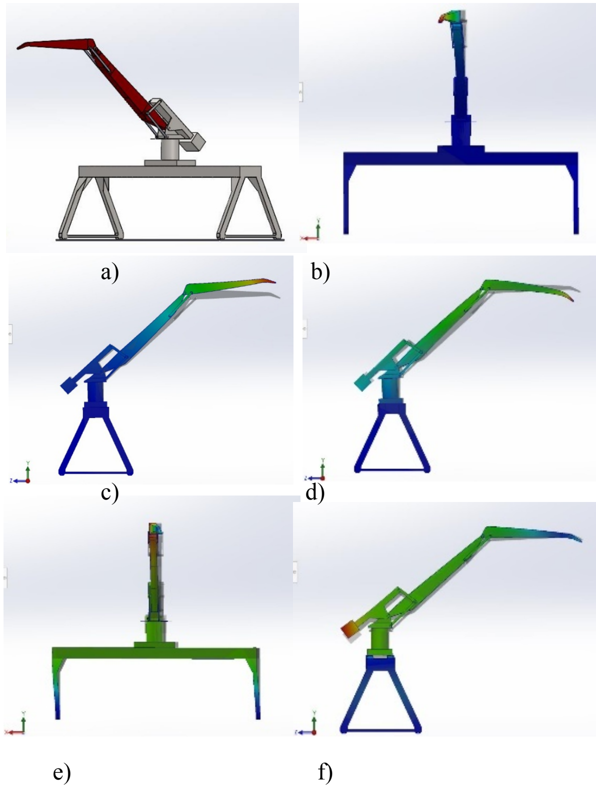

Figure 14 The first five vibration mode of the crane, made by steel material, in specific geometrical position.

The column number in the tables corresponds to the vibration mode number, and the displacement of the structures is shown in the figures.

For example 1_b of table 2 corresponds to the first vibration mode and the deformation is reported in the figure 13 b).

4. DESIGN THE LOADER CRANE ARMS WITH ALLUMINIUM ALLOY AND THE DYNAMICAL BEHAVIOUR OF THE WHOLE MACHINE

Due to the high inertia forces generated by the movement of the loader crane arms (specifically the boom and stick) resulting from their weight, this chapter addresses the implementation of an innovative material for these structures.

The selected material is an aluminium alloy, in particular EN AW6061 T6, chosen instead of the traditional structural steel. The reasons for selecting this material include its availability, cost-effectiveness, weldability, and structural properties that are quite comparable to those of steel. The main mechanical properties are detailed in Table 4. It is also possible to use composite materials to further reduce the weight of the structures, for example, by applying them to the hydraulic cylinders which will be considered in the development of this research. [19,20,21,22].

In designing the loader crane arms with aluminium alloy, the same criteria used for the steel crane design were applied. It is important to note that stiffness and buckling load in the structure are generally related to the product of Young's modulus E and the moment of inertia J of the section. For example, equations (4) describe the deformation of a beam subjected to a concentrated load at the free edge and the critical buckling load for a compressed column as estimated by Euler's equation

$$f = \frac{1}{3} \cdot \frac{F * L^3}{E * J} \quad P_{cr} = \frac{\pi^2 * E * J}{L^2} \tag{4}$$

In sizing the arms, to maintain the similar mechanical behaviour of the structure made by steel, the equations (5) were used to determine the different sections of the boom and stick.

$$E_{steel} * J_{steel} \cong E_{alluminium} * J_{alluminium} \tag{5}$$

Table 4. Main mechanical properties for the aluminium adopted.

Thickness [mm]		Mechanical properties		
>	≤	R [N/mm^2]	Reh [N/mm^2]	A%
1.5	6	290	240	10
6	12.5	290	240	9
12.5	40	290	240	8
40	80	290	240	6

Table 5 reports the dimensions of the different arms sections showed in the figures 4 and 5. Figure 15 show the rectangular section.

As the solution performed by steel, after designing the section of each element, a solid model was created followed by finite element method (FEM) analyses.

Table 5. Dimensions of the different arm sections. The U_B Plates column shows the thickness of the up and bottom plates; while the L Plates column reports the thickness of the side plates.

Section	Size [mm] HxB	Thick_U_B Plates[mm]	Thick _L Plates mm]
Stick			
A_Steel	1600x1000	18	18
B_Steel	600x375	18	10
A_All	2300x1220	24	24
B_All	900x400	24	14
Boom			
C_Steel	1900x1000	35	35
D_Steel	1500x1000	25	22
C_All	2600x1150	40	20
D_All	2100x1150	35	40

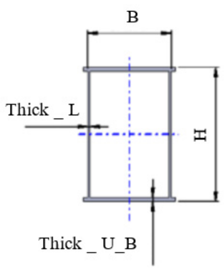

Figure 15 Main variable of the sections of the arms.

The buckling load factors are 10.54 for the stick and 13.69 and 18.78 for the boom arms; these values exceed the minimum requirements. Deformations mainly affect the side plates, particularly in the areas between the inner plates used to increase the stiffness of the section. This effect can be avoided, further increasing the load buckling factor, by specific stiffened plates or beams placed within the box section of the arms.

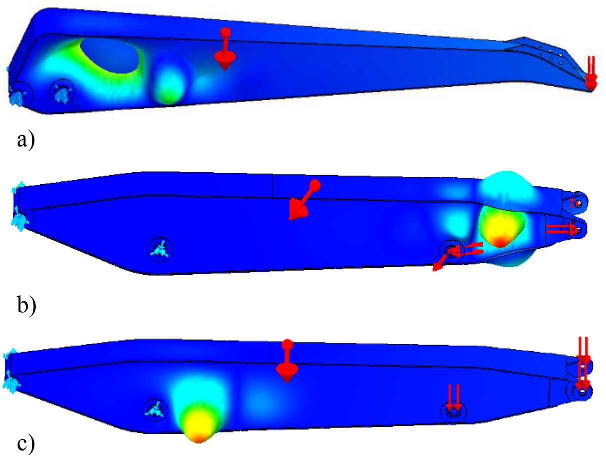

Figure 16. Stick a) and boom b) and c) deformation due to the minimum buckling load.

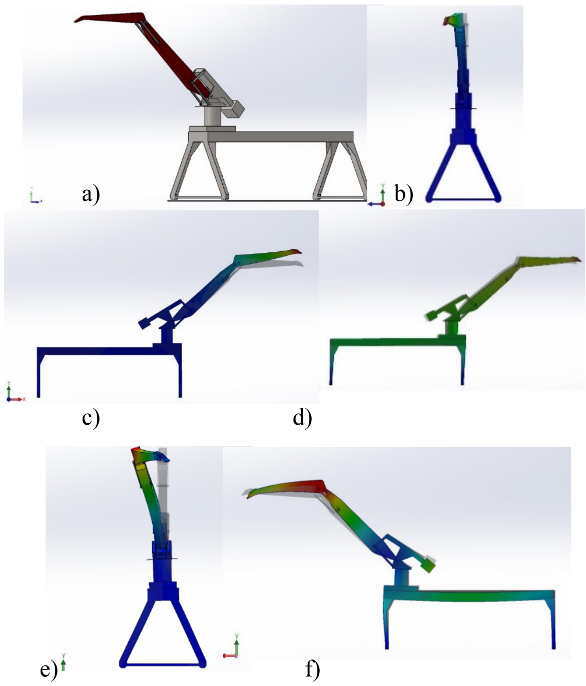

Figure 17. Fist five vibration mode of the loader crane arms made by aluminium alloy material in specific geometrical position.

The solution, which involves using aluminium alloy for the loader crane arms, i.e. stick and boom, was analyzed with the same procedure to investigated the steel solution.

Figure 17 shows the first vibration mode for the entire machine in a specific geometrical configuration specifically for Figure 11 b configuration.

Table 6. Firth values of natural frequencies and mass coefficients participation factors for the vibration modes reported in the figure 15.

Num ber	Freq. [Hz]	X_Dir.	Y_Dir.	Z_Dir.
1_b	0.87	4.45e-07	7.30e-09	0.05
2_c	1.47	0.02	0.03	6.32e-07
3_d	2.74	0.84	0.02	1.20e-08
4_e	3.86	3.32e-06	6.60e-06	0.03
5_f	4.20	1.76e-05	2.58e-05	0.44

5. COMPARISON OF THE STEEL AND ALUMINUM SOLUTION

The lightweight design process brings to a significant reduction in the dead weight of the crane, particularly for loader cranes that are primarily subjected to frequent movements.

As shown in Table 7, the dead weight of the loader crane arms, when made by aluminium alloy, is reduced to approximately 50% of the original weight that was achieved with steel.

The precise reduction value is about 54%.

Table 7. Weight of the stick and boom and comparison for the two-solution developed.

	Steel	Aluminium	Variation W_{All} / W_{Steel} %
Stick [N]	60410	33710	55.8
Boom [N]	209450	112290	53.6
Total [N]	269860	146000	54.1

The reduction in weight brings several significant advantages.

For example, in the original steel configuration, the ratio of the total weight of the arms to the crane's maximum payload capacity (350 kN) is approximately 78%. With the implementation of aluminium alloy, this ratio decreases to 42%.

This reduction opens up two potential development paths for the machine:

1) increase Payload Capacity: With the same power installed on the machine, the increased efficiency could allow for a higher payload capacity while still handling both the payload and the arms.
2) decrease Required Lifting Power: The significant reduction in total weight could lower the lifting power required, leading to potential energy savings.

Preliminary evaluations suggest that adopting aluminium alloy for the components could reduce the required power for the machine by up to 50% compared to the original steel solution, resulting in substantial energy savings.

Additionally, the hydraulic system could be further optimized for weight reduction, potentially by incorporating composite materials.

With the steel solution, preliminary sizing of the hydraulic cylinders indicated barrel diameters ranging from 250 mm to 320 mm, lengths from 4.8 m to 5.3 m, and weights between 13 kN and 23 kN, depending on their position (whether connecting the boom to the stick or the boom to the chassis). According to [19, 20, 21, 22], the final weight of hydraulic cylinders made from composite materials could be reduced to approximately 20% of the weight of the original steel versions.

Moreover, the use of aluminium alloy for the crane arms yields a dynamic behaviour very similar to that of the steel design. The first natural frequencies for both the steel and aluminium solutions are in the range of 1-2 Hz, which closely aligns with the frequencies observed in other lifting machines, such as cranes [5, 6, 7, 15].

This aspect is very important and fundamental in the study of machines especially when it is subject to actions that change over time.

6. CONCLUSIONS

This research presents the design process for an innovative crane composed of two machines: a Gantry crane and a loader crane mounted on it. The crane, which is of medium size, was initially designed using traditional structural steel.

The design was validated through numerical analyses performed on the solid model of the whole machine.

The implementation of aluminium alloy has proven to be a significant improvement for reducing the machine's weight. Specifically, the weight of the loader crane has been reduced by approximately 50% compared to the original steel version.

This reduction has a substantial impact on the overall machine, particularly in terms of the power required to move the machine or its components.

The dynamic behaviour, investigated using the finite element method, is crucial for understanding the machine's response to time-varying actions such as wind, seismic activity, payload variations, machine movement, and ship movement.

The analysis shows that the dynamic response of both the steel and aluminium solutions is very similar, with the first natural frequencies around 1-5 Hz, primarily involving the loader crane and its arms.

The Gantry crane shows greater stiffness.

Further weight reduction could be achieved by using composite materials for hydraulic cylinders, for example. The research is ongoing with two main objectives: first, to study the machine's response to variable actions, and second, to implement composite materials for the loader crane arms.

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МОДАЛНА АНАЛИЗА ПОРТАЛНЕ ДИЗАЛИЦЕ И НАДГРАДЊЕ ОД ЧЕЛИКА И ЛЕГУРЕ АЛУМИНИЈУМА

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Циљеви овог истраживања су двоструки. Први је пројектовање иновативне машине која се састоји од порталне дизалице и надградње као утоваривача, а други је имплементација иновативног материјала за ову машину, коришћењем легуре алуминијума уме–сто челика за руке крана утоваривача. Други циљ се односи на проучавање динамичког понашања дизалице са две различите конфигурације матери–јала.

Због састава ове машине, опрема за дизање показује висок степен свестраности. Процес пројектовања је развијен коришћењем аналитичке методе за процену оптерећења сваке компоненте машине, праћен соли–дним моделирањем и нумеричком анализом. Главни резултати показују да је тежина кракова крана утоваривача од легуре алуминијума приближно 50% лакша од оних од челика, са веома сличним динамичким понашањем за оба решења.