



ING. MECATRÓNICA

**Thesis submitted prior to obtaining the degree of
Mechatronics Engineer.**

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**Design and Simulation of an Electro-Hydraulic Mechatronic
Lifting System for Non-Motorized Airport Ground Support
Equipment**

Quito-2026

Aprobación del Tutor del Trabajo de Integración Curricular

Yo Guillermo Mosquera, en calidad de tutor del Trabajo de Integración Curricular denominado: *Design and Simulation of an Electro-Hydraulic Mechatronic Lifting System for Non-Motorized Airport Ground Support Equipment*, realizado por Carlos Paredes, ha sido orientado y revisado en todas sus partes y considero que cumple los requisitos establecidos por la institución. En virtud de ello doy mi aprobación a fin de continuar con el proceso académico correspondiente.



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Dedicatoria

Quiero dedicar este trabajo, principalmente, a la única persona por la cual todo lo que soy es posible. Todo mi progreso, mis valores, mis sueños y mis metas son gracias a mi mamá, Ginna Vargas; eres el motor de mi vida.

Dedico esto a mi hermano José Antonio, por ser quien me impulsa a ser mejor; eres mi motivación principal para todas las ambiciones de mi vida.

Dedico esto a mi padre, a mis hermanos y a mis tíos, por estar siempre para mí cada vez que los necesité.

Dedico esto a mi pareja, Anahy, quien me ha acompañado de la mano en prácticamente todo mi camino universitario.

Agradezco a la universidad, no solo por los conocimientos adquiridos, sino por regalarme a mis amigos: Martín, Ismael, Chamorro y Ochoa. Mis ganas de ser mejor, superarme, no rendirme e intentarlo una vez más, fueron totalmente gracias a ellos.

Por último, y no menos importante, dedico esto a mi abuelo Carlos Abdón. Sé que no me pudo ver como ingeniero, pero él sabía que llegaría lejos. Por él, nunca retrocederé un solo paso ni me rendiré, sea cual sea la circunstancia. Cada logro se lo dedicaré a él también.

Agradecimiento

Agradezco principalmente a mi tutor, Guillermo Mosquera, por su guía constante, su rigor técnico y su paciencia durante el desarrollo de este proyecto. Su dirección fue fundamental para estructurar este trabajo con un enfoque práctico y sólido.

A mis profesores de la facultad, por el nivel de exigencia y por haberme brindado las herramientas y el criterio analítico necesarios a lo largo de toda la carrera.

Finalmente, a Swissport, por abrirme sus puertas. La experiencia directa en campo y las enseñanzas operativas que adquirí en sus instalaciones fueron invaluableles para aterrizar la teoría y consolidar mi perfil profesional.

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Resumen

El mantenimiento de plataformas de carga (dollies) aeroportuarias emplea elevaciones manuales asimétricas con montacargas, generando riesgos ocupacionales. Este proyecto diseña y valida virtualmente un sistema mecatrónico de elevación electrohidráulico de 1500 kg de capacidad para automatizar este proceso. Aplicando la metodología V-Model, se integró el diseño conceptual, dimensionamiento analítico y optimización estructural mediante Análisis de Elementos Finitos (FEA). Posteriormente, se desarrolló un gemelo digital multidominio en Simscape y Stateflow para la validación dinámica. Los resultados demostraron que la optimización de los brazos a un perfil estructural HEB 100 garantiza un factor de seguridad de 3.02 y una deflexión máxima de 1.65 mm. La simulación validó una operación estable a 105 bar, un error de sincronización inferior a 1.0 mm entre actuadores paralelos, y un ciclo automatizado de 34 segundos, comprobando su robustez dinámica ante cargas transitorias. Se concluye que la arquitectura propuesta elimina la dependencia de equipos auxiliares, demostrando la viabilidad técnica para estandarizar el mantenimiento de forma segura y eficiente.

Palabras clave: Equipos de soporte en tierra (GSE), electrohidráulica, simulación virtual, V-Model, FEA.

Abstract

In the maintenance process of non-motorized ground support vehicles (Dollies), an unsafe procedure is currently used, where a forklift and auxiliary tools are required to lift the equipment. In addition, two or more technicians are needed to complete this maintenance task, which entails risks for the operators. This project virtually validates a hydraulic lifting system with automatic support and a capacity of 1500 kg to ensure this process. The V-model methodology was applied and integrated into the conceptual design, while the structural sizing was validated through Finite Element Analysis (FEA). Subsequently, a multi-domain virtual model was developed for dynamic validation. The results demonstrated that the objectives and requirements were fully achieved, obtaining a safety factor of 3.02 in the structural analysis and correctly completing the automatic load cycles in the simulation. It is concluded that the proposed architecture eliminates the dependence on auxiliary equipment, demonstrating the technical feasibility of standardizing this process and improving the continuous maintenance workflow in daily operations.

Keywords: Ground Support Equipment (GSE), electro-hydraulic, multi-domain simulation, V-Model, FEA.

Chapter I

Introduction

The handling and operational flow at Swissport, apart from its technicians and operators, depend largely on the availability of equipment. The company offers an availability of 90% of its equipment, and in order to maintain this availability, several preventive and corrective maintenance activities are scheduled. The most numerous vehicle is the cargo dolly, with more than 300 units of this type of vehicle. As a result, constant maintenance is required, forcing the use of a forklift to lift one side of the platform, manually place mechanical supports under the chassis, and then repeat the process on the opposite side. This practice involves instability risks and depends on external equipment, which could affect the daily flow of operations, significantly increasing the risk of accidents and occupational health issues [2–4]. Continuous optimization of each process is critical to minimizing equipment turnaround times in the hangar [1]. The absence of a lifting mechanism results in daily manual intervention to complete scheduled maintenance tasks. Automation has been shown to streamline workflows and reduce delays or lack of availability [5].

In the state of the art concerning heavy load handling, hydraulic systems are commonly the industry standard due to their reliability [6]. Recent studies have demonstrated the feasibility of modeling prototypes to predict dynamic behavior [7, 8]. To avoid abrupt actuation and ensure reliable automation, the integration of sequential control systems has proven effective in coordinating multi-actuator systems [9, 10].

Existing studies focus on hydraulic actuation and control strategies individually. No specific work was found that combines structural static sizing, multi-domain hydraulic simulation, and a control sequence for an automatic system aimed at improving the maintenance process of airport ground support vehicles.

The main contribution of this work is the integration of a methodology that combines analytical design, modeling, finite element analysis, and multi-domain simulation for the virtual validation of automatic operation. This work followed an approach based on the V-model methodology, allowing the establishment of requirements, sizing, and virtual validation, a contemporary framework widely recognized for optimizing the design of complex industrial systems before their physical manufacturing stage [11].

Regarding the justification of this project, it is approached from both operational and administrative perspectives. Designing this automated system contributes to improving industrial safety by eliminating the need for forklifts and manual supports, while enhancing the workflow of maintenance activities involving these vehicles. From a mechatronic perspective, designing a virtual model allows different operational variables and control logic to be adjusted in a safe and cost-effective environment, validating compliance with requirements before initiating any manufacturing stage.

The objective of this project is to design and validate, through simulation, an electro-hydraulic system capable of automating its operating cycle for the maintenance of non-motorized airport cargo vehicles (Dollies).

To achieve this objective, the following specific objectives will be fulfilled:

1. Determine the hydraulic and mechanical requirements through analytical calculations.
2. Design the conceptual and functional architecture, as well as the modeling required to automate the system.
3. Simulate a multi-domain model to analyze the dynamic behavior of the system.
4. Validate the structural integrity, synchronization capability, and automatic behavior of the proposed system through virtual multi-domain tests.

Chapter II

Methodology and Design

At present, the routine maintenance of non-motorized equipment (dollies) presents some significant operational challenges. Technicians have an unconventional practice in which they use a forklift to lift one side of the Dolly, place supports, and then repeat the same process on the opposite end. In addition to being unconventional, this type of practice introduces serious risks to safety, load stability, and also depends on multiple auxiliary pieces of equipment, sometimes even two or more technicians, just to lift the vehicle. Also, considering that the repetitive movement around the vehicle while it is partially lifted on only one side generates delays and operational inefficiencies.

To mitigate these vulnerabilities and provide a safe and automated solution, the V-model methodology was strictly followed. This type of methodology provides an approach that is suitable for mechatronic systems. The descending branch of the V mainly focuses on the decomposition of the system from its operational needs to the requirements of specific components as well as detailed design. On the other hand, the ascending branch focuses on the integration and virtual validation phases, ensuring that each design choice can satisfy the initial requirements, in this case through multi-domain dynamic simulation and finite element analysis.

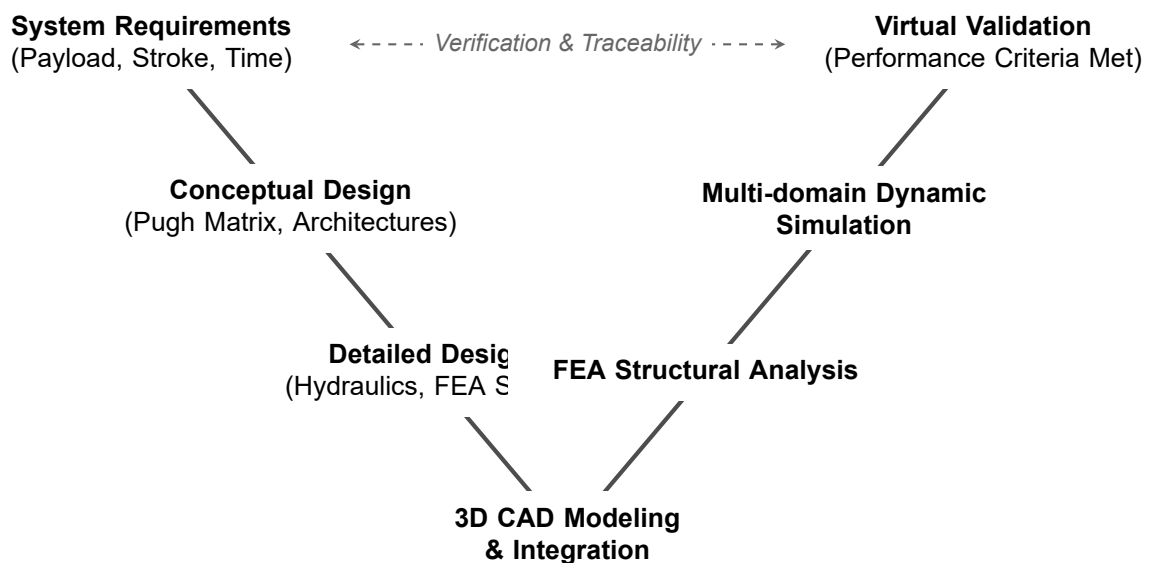


Fig. 1. V-Model systems engineering methodology applied to the lifting system project.

Requirements

The foundation of the V-model starts from the definition of the system requirements, from which the current maintenance limitations described above arise. The general objective is not only to lift the cargo platform but also to streamline the time required to prepare the maintenance of these units, reduce manual intervention, and increase the availability of tools in the hangar by eliminating dependence on them for this type of maintenance.

To ensure strict compliance throughout the design phases and virtual validation, the requirements have been formalized and categorized into Functional Requirements (FR), Non-Functional Requirements (NFR), and Constraints (C), as shown in Table 1.

TABLE 1
System Requirements and Classifications

ID	Requirement Description
Functional Requirements (FR)	
FR-01	The system shall safely elevate a design load of 1500 kg.
FR-02	The system shall provide sufficient vertical displacement to allow axle and wheel maintenance operations.
FR-03	The system shall provide synchronized and symmetrical lifting motion to minimize load imbalance and improve operational safety.
FR-04	The system shall allow automatic operation to ensure repeatability and minimize human intervention.
FR-05	The system shall optimize operational efficiency by eliminating the need for auxiliary lifting vehicles and manual jack stands during the preparation phase.
Non-Functional Requirements (NFR)	
NFR-01	All load-bearing structural components shall be designed with a minimum Safety Factor (FS) of 3.0.
NFR-02	The system shall be compatible with standard industrial power grids.
NFR-03	The control interface shall allow operation from a safe distance.
NFR-04	The proposed system shall be validated through modeling and simulation prior to any manufacturing stage.
System Constraints (C)	
C-01	The lifting base and arms shall fit underneath the standard ground clearance of the dolly without modifications.
C-02	The design shall prioritize commercially available components.

Conceptual Design

This section explains the translation of the requirements into a feasible conceptual framework, including the analysis of alternatives for different designs and also defining the functional and logical architectures that will determine the actions of this project.

To objectively evaluate different solutions for each main function of the lifting system,

they were placed in a morphological chart in Table 2. The selected option is marked in bold.

TABLE 2
Morphological Chart for System Conceptualization

Function	Solution 1	Solution 2	Solution 3
Lifting Mechanism	4-Post Independent Lift	Parallel Posts (Twin Cylinders)	Single Telescopic Lift
Guidance System	Guide Rollers	Linear Guide Rails	Friction Pads
Power Source	Pneumatic Compressor	Electro-Hydraulic (HPU)	Pure Electromechanical
Control Strategy	Manual Control	Electro-Hydraulic Automatic Sequence	PLC-Based Control

Analysis of Alternatives

During this conceptualization phase, two main architectural concepts were evaluated to achieve the simultaneous lifting of the Dolly:

- **Concept A:** a four-post lift with one cylinder and one automatic individual arm on each post.
- **Concept B:** a two-post lift with one cylinder and two arms with symmetrical and automatic movement on each side.

To achieve an objective and feasible selection, the weighted Pugh decision matrix in Table 3 was used. These lifting proposals originated with the objective of mitigating the operational deficiencies in the maintenance procedures of the dollies. Concept A was established as a baseline (datum), meaning that all criteria for Concept B were rated in relation to Concept A as better, worse, or equal.

Selected Concept Description

By applying an objective evaluation, Concept B was identified as the most viable configuration. This system, by having parallel posts, helps with the simultaneous lifting of the entire platform. Thus, it significantly reduces the preparation time required to perform maintenance on the Dolly and also mitigates certain risks associated with unstable loads and occupational health risks for technicians performing these unconventional practices.

Although a system with four posts allows maximizing the support points during lifting, and the cylinders that would have to be used would lift less load than when using only two cylinders, synchronization to achieve safe symmetry, as well as implementation and maintenance

TABLE 3
Weighted Pugh Decision Matrix for Lifting Architecture

Evaluation Criteria	Weight (1-5)	Concept A (4-Post)	Concept B (Parallel)
Operator safety	5	Datum (0)	+1 (+5)
Reduction of prep time	4	Datum (0)	+1 (+4)
Reduction of auxiliary equipment	4	Datum (0)	0 (0)
Simplicity of operation	5	Datum (0)	+1 (+5)
Compactness (Stowed clearance)	3	Datum (0)	+1 (+3)
Cost and manufacturing complexity	4	Datum (0)	+1 (+4)
Net Weighted Score		0	+21
Decision		Discard	Proceed

costs, would increase due to having twice as many components.

Choosing to rely on only two linear actuators significantly simplifies process control, makes it easier to control symmetrical movement in the arms, and requires half the components compared to case A.

System Architecture

In the architecture of this system, the functional energy flow and the operational logic are defined.

The functional architecture describes the energy transfer. The system starts with the extraction of electrical energy from the Swissport hangar power grid, which is converted into fluid power through the hydraulic power unit, where an electric motor drives a hydraulic gear pump, which modulates it through directional control valves. This hydraulic power is subsequently supplied to the double-acting cylinders. Simultaneously, the structural arms transfer the required mechanical lifting force from the cylinders to the platform chassis.

The logical architecture describes the operational sequence of the system. The operating cycles will be executed through a single command, automatically controlling each stage to minimize technician intervention.

1. *Standby*: here the system is powered on but inactive, waiting for a command.

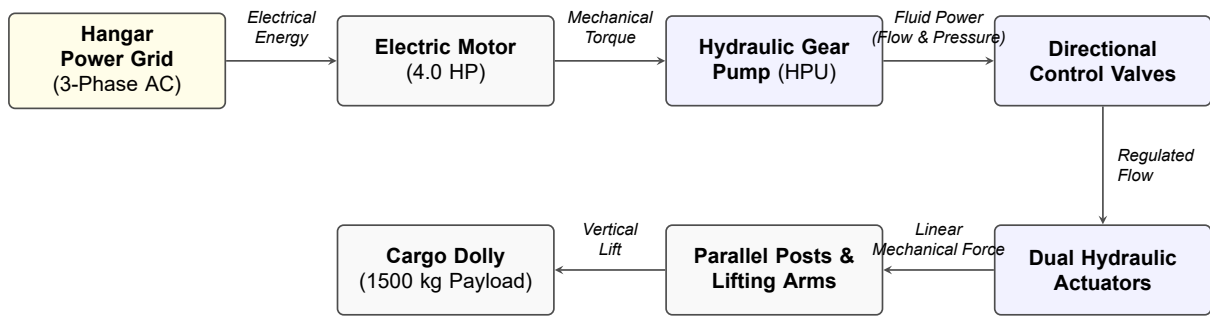


Fig. 2. Functional architecture mapping the flow of electrical, fluid, and mechanical power.

2. *Clamping*: once the start button is pressed, the arms begin to move until they are in the correct position underneath the chassis.
3. *Lifting*: here the system, with the arms coupled underneath the chassis, begins to rise without any required action other than the start button that was pressed at the beginning.
4. *Maintenance*: during this maintenance process, the vehicle reaches the desired height and remains there for as long as necessary until the operator presses the descent button.
5. *Descent*: once vehicle maintenance is completed, the vehicle is lowered by pressing the descent button, which only causes the vehicle to return to the initial position and rest on the ground.
6. *Release*: once the vehicle reaches the floor and the system returns to its initial position, the arms retract to allow free vehicle circulation. Once this process is completed, the system returns to Standby.

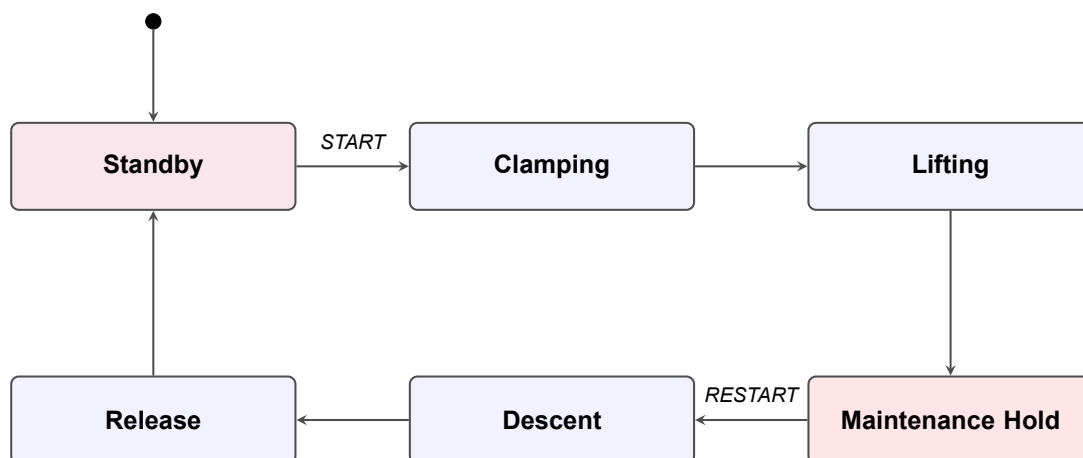


Fig. 3. Logical architecture and operational state machine of the automated lifting sequence.

Detailed Design

In this stage, three-dimensional geometric models were developed, and the analytical equations for the mechanical and hydraulic sizing were formulated following accepted engineering practices. These mathematical models served as the geometric baseline for the validation methodologies.

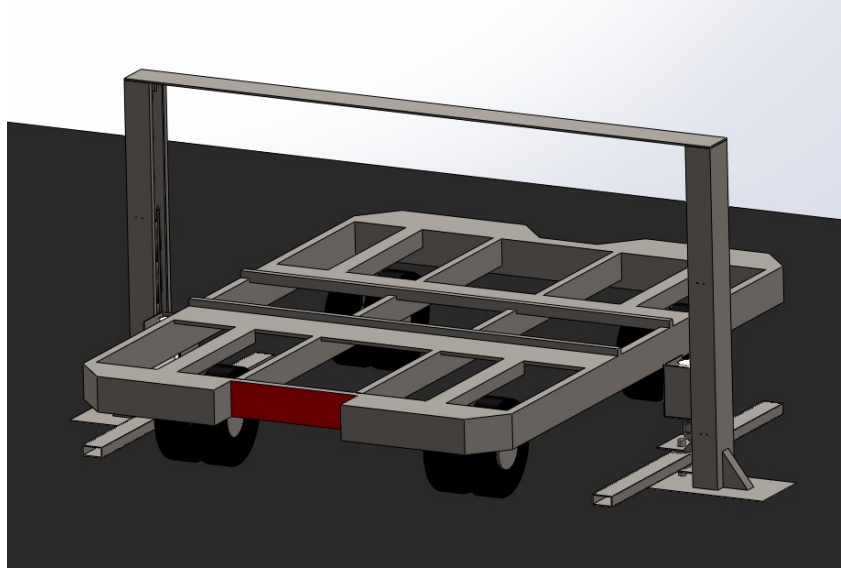


Fig. 4. Isometric view of the proposed electro-hydraulic lifting system.

TABLE 4
Material Selection for Primary Structural Components

Component	Material	Yield Strength	Target FS
Lifting Arms & Columns	ASTM A36 Steel	250 MPa	≥ 3.0
Transmission Shaft	SAE 1045 Steel	530 MPa	≥ 3.0
Base Anchor Bolts	SAE Grade 5	634 MPa	≥ 3.0

Mechanical Subsystem

To guarantee safe operation, the primary structure must withstand dynamic loading conditions without experiencing excessive deformation or yielding. Consequently, the lifting arms were mathematically modeled considering a cantilever beam subjected to a dynamic live load scenario to ensure a minimum safety factor (FS) of 3.0. A dynamic starting impact factor (K_{dyn}) of 1.25 was applied over the nominal vertical load, distributed across four lifting points:

$$W_{dyn} = m \cdot g \cdot K_{dyn} = 1500 \text{ kg} \cdot 9.81 \text{ m/s}^2 \cdot 1.25 = 18393.75 \text{ N} \quad (2.1)$$

$$F_{arm} = \frac{W_{dyn}}{4} = \frac{18393.75 \text{ N}}{4} = 4598.44 \text{ N} \quad (2.2)$$

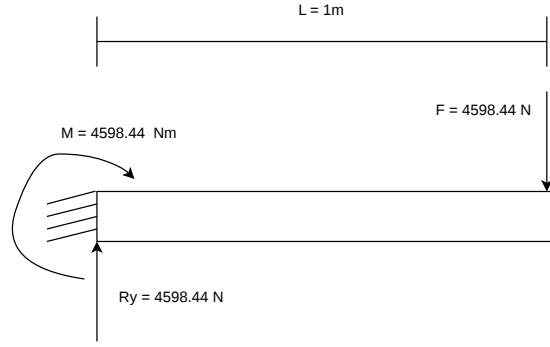


Fig. 5. Free Body Diagram (FBD) of the main lifting arm under vertical load.

For a free arm length of $L = 1.0$ m, the maximum bending moment at the fixed support was calculated as:

$$M_{bending} = F_{arm} \cdot L = 4598.44 \text{ N} \cdot 1.0 \text{ m} = 4598.44 \text{ N} \cdot \text{m} \quad (2.3)$$

Using ASTM A36 structural steel (250 MPa), the required geometric section modulus (S_{req}) was established to guide the initial CAD modeling:

$$\sigma_{allow} = \frac{S_y}{FS} = \frac{250 \times 10^6 \text{ Pa}}{3} = 83.33 \text{ MPa} \quad (2.4)$$

$$S_{req} = \frac{M_{bending}}{\sigma_{allow}} = \frac{4598.44}{83.33 \times 10^6} = 5.518 \times 10^{-5} \text{ m}^3 = 55.18 \text{ cm}^3 \quad (2.5)$$

Structural Iterative Optimization. In this structural design process, an iterative approach was followed to guarantee both safety and operational synchronization in the system. Initially, a rectangular hollow beam was selected due to its simplicity in manufacturing. From the analytical calculations performed, it was concluded that the section could withstand the expected bending loads; however, finite element analysis revealed that the elastic deflections were excessive, leaving a safety factor below that required by the specifications.

Afterward, the arm geometry was redesigned, this time using an HEB 100 structural profile, profiles that are used for heavy loads since they are made to withstand moments. Unlike the previous arm structure, the corresponding finite element analyses demonstrated that the moment of inertia increased significantly and improved bending stiffness. Table 5 compares the mass and design between the two iterations that were carried out, demonstrating the necessity of the added structural weight (4.6 kg per arm) to achieve the required stiffness.

TABLE 5
Iterative Design Comparison for the Primary Lifting Arm

Parameter	Rectangular Hollow Section	HEB 100 Structural Beam
Material	ASTM A36 Steel	ASTM A36 Steel
Net Mass per Arm	12.93 kg	17.54 kg
Maximum Elastic Deflection	≈ 4.8 mm	1.65 mm
Safety Factor (FS)	Marginal (< 3.0)	3.02 (Satisfied)
Final Selection	Discarded	Selected

In Figure 6, the iterative engineering workflow carried out to finalize the selection of structural components can be observed.

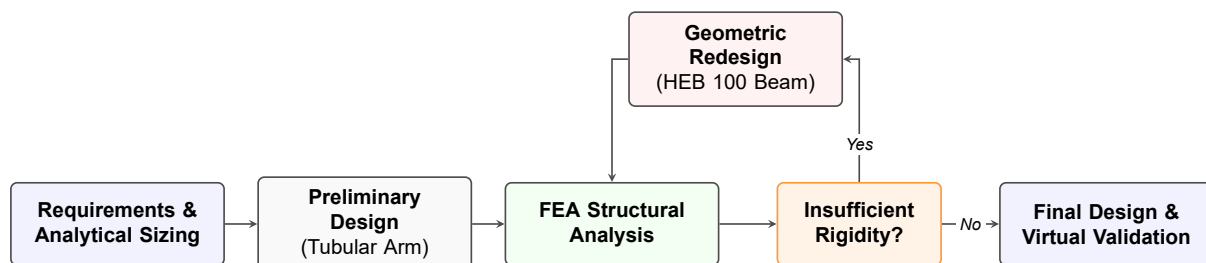


Fig. 6. Flowchart of the iterative structural optimization and virtual validation process.

Transmission Shaft and Columns. This section discusses a critical interface in the mechanism whose main function is to restrict carriage alignment while transmitting the rotational reactions caused by the beam. Consequently, this bending moment is transferred directly to the vertical shaft, thereby inducing combined internal stresses together with the torsional load that the shaft must withstand as a result of the unloaded displacement.

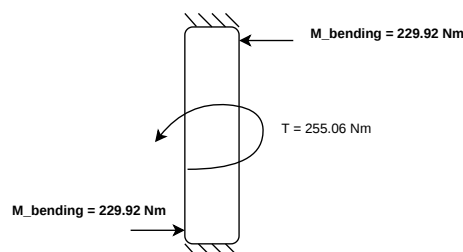


Fig. 7. Free Body Diagram of the transmission shaft subject to combined torsion and bending.

Selecting a nominal shaft diameter of $d = 0.060$ m and SAE 1045 steel (530 MPa), the

principal stresses were analytically defined:

$$\tau = \frac{16 \cdot T}{\pi \cdot d^3} = \frac{16 \cdot 255.06}{\pi \cdot (0.060)^3} = 6.01 \text{ MPa} \quad (2.6)$$

$$\sigma = \frac{32 \cdot M}{\pi \cdot d^3} = \frac{32 \cdot 229.92}{\pi \cdot (0.060)^3} = 10.84 \text{ MPa} \quad (2.7)$$

For fixed-guided ends ($K = 1.0$), a free height of $L_{col} = 0.60$ m, and a compressive force per post of $P = 9196.88$ N:

$$I_{min} = \frac{P \cdot FS \cdot (K \cdot L_{col})^2}{\pi^2 \cdot E} = \frac{9196.88 \cdot 3 \cdot (1.0 \cdot 0.60)^2}{\pi^2 \cdot 200 \times 10^9} = 5.04 \times 10^{-9} \text{ m}^4 = 0.50 \text{ cm}^4 \quad (2.8)$$

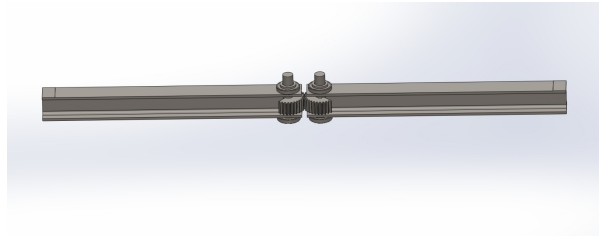


Fig. 8. Auxiliary gear mechanisms integrated for carriage guidance and mechanical locking.

Hydraulic Actuation System

Finally, the system was modeled in a CAD environment and two double-acting cylinders were designated as the main linear actuators.

The lifting force requirement also includes the dead mass of the sliding carriage assembly (28 kg based on the 17.54 kg HEB arm plus 10.46 kg from the remaining components).

$$F_{weight} = (1500 + 28) \cdot 9.81 = 14989.48 \text{ N} \quad (2.9)$$

$$F_{friction} = F_{weight} \cdot 0.15 = 2248.42 \text{ N} \quad (2.10)$$

$$F_{cylinder} = \frac{F_{weight} + F_{friction}}{2 \text{ actuators}} = \frac{17237.90}{2} = 8618.95 \text{ N} \quad (2.11)$$

With the system operating at a safe nominal pressure of 120 bar (12×10^6 Pa), the minimum theoretical piston diameter required to overcome the load was determined analytically:

$$A_{req} = \frac{F_{cylinder}}{P_{work}} = \frac{8618.95}{12 \times 10^6} = 7.18 \times 10^{-4} \text{ m}^2 = 7.18 \text{ cm}^2 \quad (2.12)$$

$$D_{req} = \sqrt{\frac{4 \cdot A_{req}}{\pi}} \cdot 1000 = 30.24 \text{ mm} \quad (2.13)$$

Taking into account the requirements and analyses performed, a standard commercial cylinder with a piston diameter of 40 mm and a rod diameter of 18 mm was selected.

To determine the volumetric flow demand, and based on the standard lifting times for the vehicles, which range from 10 to 15 seconds from the moment the vehicle is already coupled to the system until reaching the desired position, and with the purpose of accelerating this process and following the recommendations of the head of the Swissport maintenance team, a time of 8.0 seconds was adopted as the sizing basis, which translates into:

$$Q_{total} = (v \cdot A_{actual} \cdot 60000) \cdot 2 = (0.075 \cdot 0.001256 \cdot 60000) \cdot 2 = 11.30 \text{ L/min} \quad (2.14)$$

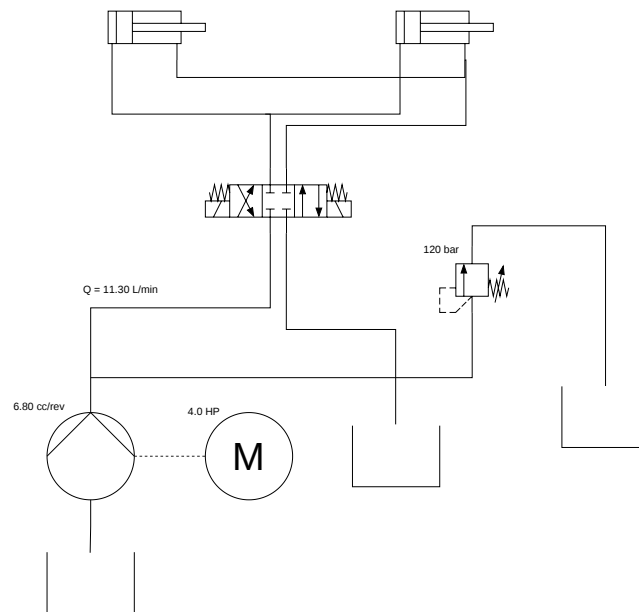


Fig. 9. ISO 1219 standardized hydraulic circuit diagram for dual-cylinder synchronization.

Hydraulic Power Unit (HPU)

This power unit was sized to reliably and safely supply the requirement of 11.3 L/min under 120 bar of continuous pressure, thus selecting a standard four-pole industrial motor operating at 1750 RPM. The design criterion assumed a volumetric efficiency of $\eta_v = 0.95$ for the gear pump:

$$V_g = \frac{Q_{total} \cdot 1000}{\text{RPM} \cdot \eta_v} = \frac{11.30 \cdot 1000}{1750 \cdot 0.95} = 6.80 \text{ cc/rev} \quad (2.15)$$

The nominal power required to drive the pump, considering as a design criterion an overall hydraulic efficiency of $\eta_t = 0.85$, was analytically defined as follows:

$$P_{kW} = \frac{Q_{total} \cdot P_{bar}}{600 \cdot \eta_t} = \frac{11.30 \cdot 120}{600 \cdot 0.85} = 2.65 \text{ kW} \quad (3.55 \text{ HP}) \quad (2.16)$$

A 4.0 HP AC motor was selected and paired with the nominal 8.0 cc/rev gear pump, providing a sufficient operating margin to prevent motor overload during pressure peaks.

The geometric storage volume of the hydraulic fluid was selected or specified as 40 Liters. This capacity satisfies the standard industrial retention requirement of storing approximately three times the desired nominal flow per minute, thereby adding an essential thermal safety margin to mitigate fluid aeration and any type of heat accumulation during multiple cycles.

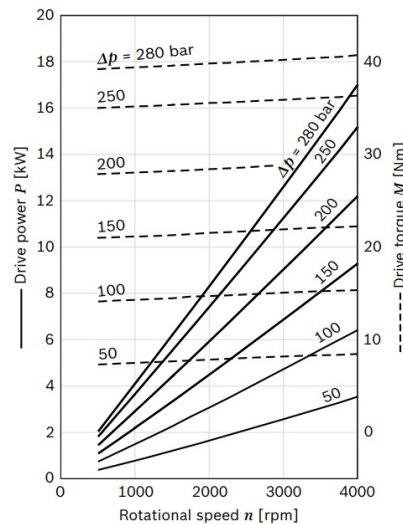


Fig. 10. Flow vs. Rotational Speed performance curve characterizing the selected gear pump (Size 8).

TABLE 6
Technical Specifications of Selected Hydraulic Components

Subsystem / Component	Technical Specification
Double-Acting Cylinders (x2)	Bore: $\varnothing$ 40 mm, Rod: $\varnothing$ 18 mm, Stroke: 600 mm.
Hydraulic Gear Pump	Fixed displacement: 8.00 cc/rev (Nominal Size 8).
Electric Motor	3-Phase AC, 4.0 HP, 1750 RPM.
Directional Control Valve	4/3-way, solenoid actuated, spring-centered, closed center.
Pressure Relief Valve	Adjustable, system safety set at 120 bar.
Reservoir (Tank)	40 Liters capacity (includes thermal margin).

Iterative Virtual Prototyping and Simscape Integration

The transition from the CAD model to a fully functional multi-domain simulation required several prototype iterations. Initially, an attempt was made to export the complete assembly from SolidWorks to MATLAB Simulink. Following simulation example recommendations, only basic mate relationships were applied in the complete assembly. All these mate relationships also considered certain bolt holes and couplings present in the system. This caused Simscape Multibody to generate many computational redundancies, making the block tree representing the simulation excessively extensive and interpreting all mate relationships as spatial movement of every small component in the assembly. This process generated over-constrained equations that caused algebraic loops in the system, preventing the correct use of the prototype movements.

To mitigate this problem and these computational limitations, a rigorous geometric simplification of the assembly was required. The general assembly was decomposed into rigid macro bodies. Each subassembly was saved as a single rigid part to create a simple assembly containing only the motion constraints required by the system.

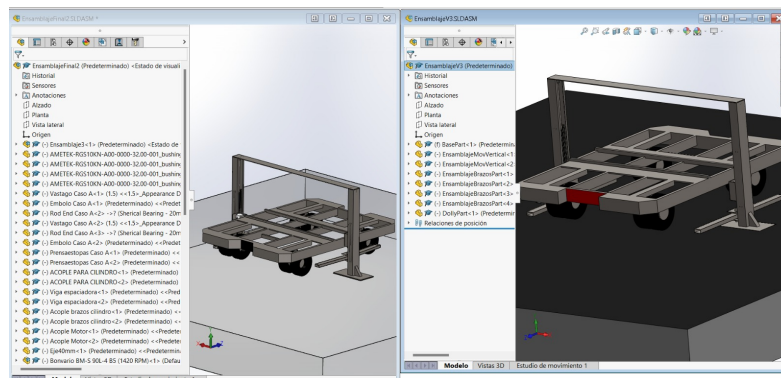


Fig. 11. CAD optimization process mapping the transition from a highly redundant assembly feature tree to serialized rigid bodies.

Once the simulation successfully compiled in its simplified form, showing the correct movement of each system, this dynamic system was developed and structured for validation through three progressive iterations in this virtual prototyping process:

- **Iteration 1: Kinematic Evaluation with Ideal Actuation Sources.** A kinematic evaluation was carried out using purely ideal actuation sources, that is, pulses or constants that moved the system to infinity. This confirmed that the movement with this type of actuation was correct.
- **Iteration 2: Integration of Multi-Domain Physical Networks.** In this second iteration,

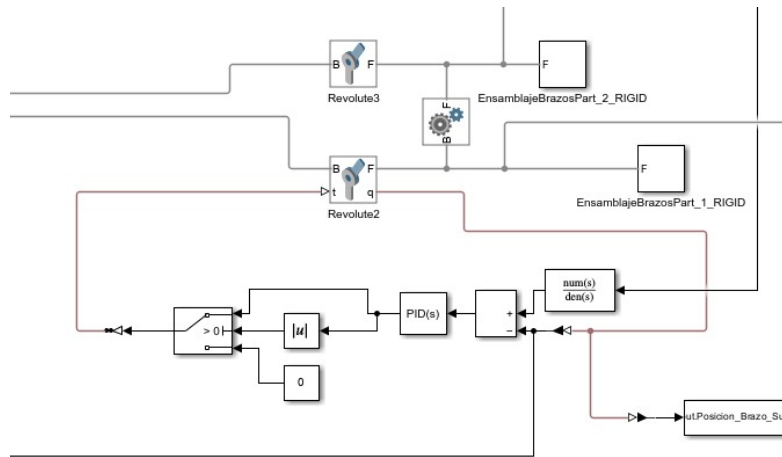


Fig. 12. Simscape Multibody block diagram layout utilizing ideal mathematical motion sources for initial kinematic validation.

after proving that the movement of each subsystem was correct, ideal sources were introduced, for example an ideal hydraulic source and a step input simulating an ideal actuator such as a motor. Here, the hydraulic circuit was integrated, considering its 4/3 directional valve and the hydraulic cylinders, in order to transfer the correct motion to the prismatic joints responsible for the vertical movement, while also configuring the gears between the arms for their symmetrical motion.

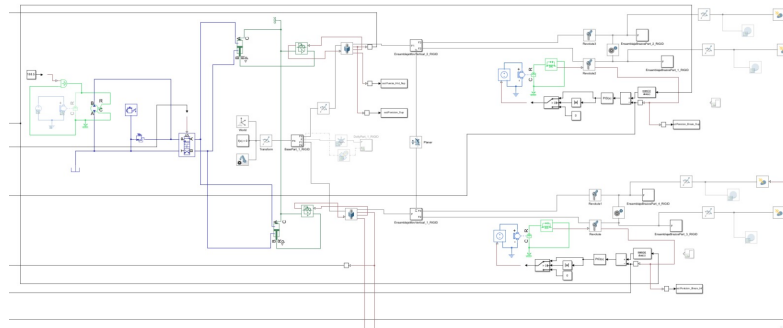


Fig. 13. Coupled multi-domain network architecture integrating Simscape Fluids and Simscape Electrical subsystems.

- **Iteration 3: Mechanical Losses and Control Sequence Layout (Final Prototype).** The final prototyping layer introduced non-ideal effects and digital control logic. Mechanical friction losses along the vertical columns were modeled by introducing a *Translational Friction* block ($\mu = 0.15$). Furthermore, to evaluate the transient dynamic stability of the lifting carriages along the vertical columns, the viscous damping coefficient within the prismatic joints was parameterized. Values of 0, 5000, and 8000 N/(m/s) were established to study the system's susceptibility to vibrations and the effectiveness of the mechanical

guides in dissipating kinetic energy during state transitions. To coordinate state transitions and manage the system's operational phases, a *Stateflow* chart was linked to the solenoids of the directional valve. This controller implemented a strict logical sequence matching the operational states, acting as the final virtual prototype for data extraction.

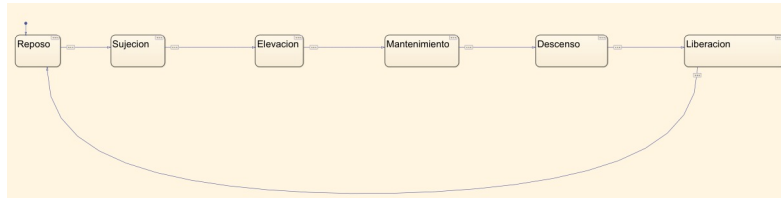


Fig. 14. Stateflow logic implementation coordinating the sequential execution of the multi-state lifting routine.

To establish strict traceability and close the loop defined by the V-Model methodology, the mathematical design and the final virtual prototype developed in this chapter serve as the basis for the dynamic property verification presented in Chapter 3. The subsequent simulation results will specifically address the fulfillment of the primary requirements, including structural integrity (NFR-01), symmetrical synchronization (FR-03), and automated operational efficiency (FR-04, FR-05).

Chapter III

Results and Discussion

In this results section, the structural integrity of the arms, the kinematic response, the dynamic response, and the validation of the control sequence were systematically evaluated, leading to the verification of compliance with the initial requirements of the project.

Structural Validation (FEA)

To ensure that the critical safety requirement was fulfilled, the components that support the load were subjected to static finite element analysis under a maximum payload scenario of 1500 kg plus a dynamic impact factor of 1.25.

A design process with an iterative optimization approach was used in order to guarantee that the final structure or the final prototype would comply with the required safety factor.

Preliminary Structural Configuration

Initially, the arms had a hollow rectangular section structure, which, when subjected to finite element analysis, revealed critical deficiencies in structural stiffness when the full operating load was applied.

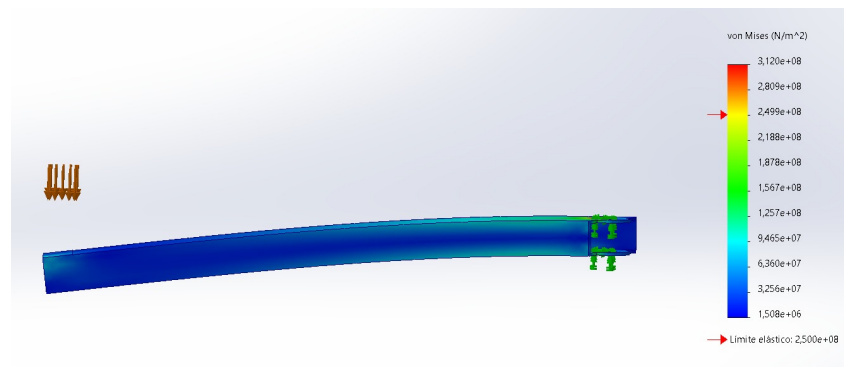


Fig. 15. Von Mises stress distribution on the preliminary rectangular hollow arm.

Since this was a multi-domain simulation, the initial focus was the correct operation of the system, generating a problem that went unnoticed at first sight.

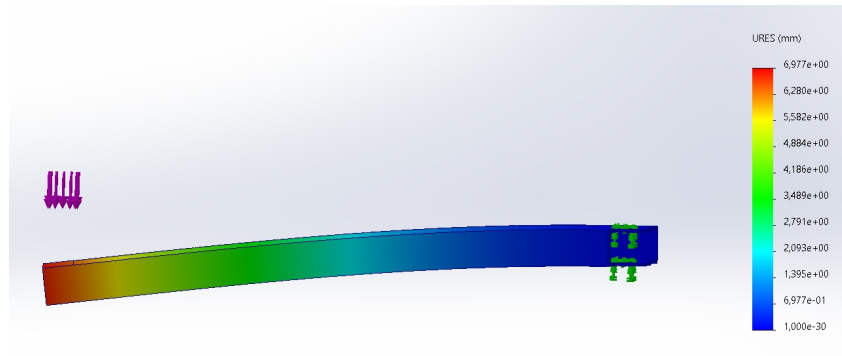


Fig. 16. Vertical displacement analysis of the preliminary design, showing excessive deflection.

Once the finite element analyses were performed, it was concluded that this structure was unfeasible for this type of load. Due to a deflection greater than 6 mm, it was concluded that the structural stiffness was insufficient. This could generate risks for the structure and the safety factor, failing to comply with the initial requirements and critically reducing the service life of the prototype.

Optimized Structural Configuration

To mitigate this lack of stiffness and increase the safety factor, the structures of the main arms were redesigned using an HEB 100 industrial steel beam profile, beams that are designed to withstand greater moments of inertia without an unreasonable weight penalty, achieving a better distribution of bending stresses so that, with dimensions almost identical to the previous structure, it was capable of reaching a safety factor of 3.02.

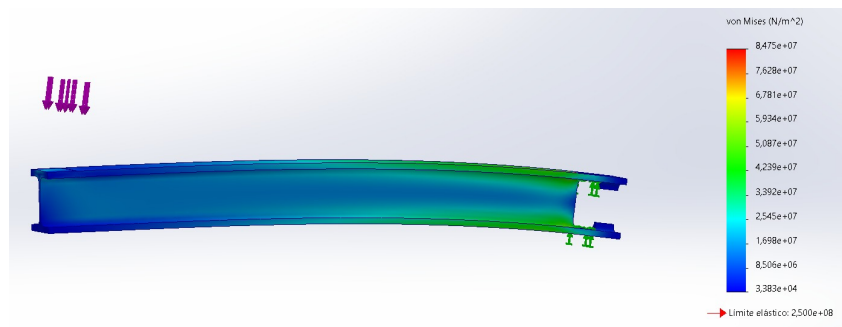


Fig. 17. Von Mises stress distribution on the optimized HEB 100 lifting arm.

The maximum recorded Von Mises stress occurs at the fixed support, reaching a peak value of approximately 82.64 MPa. With the selected material, ASTM A36 steel, the beam remains well below the yield strength limit of 250 MPa.

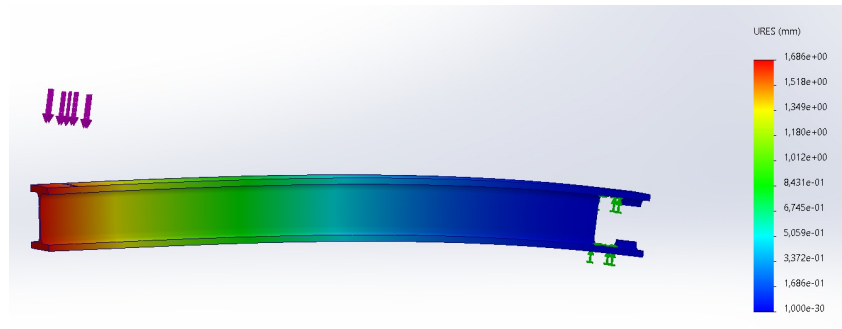


Fig. 18. Maximum vertical displacement map of the optimized HEB 100 arm.

With this new design, the maximum vertical deflection was significantly reduced to 1.65 mm, also remaining well below the allowable deflection limit of 4 mm.

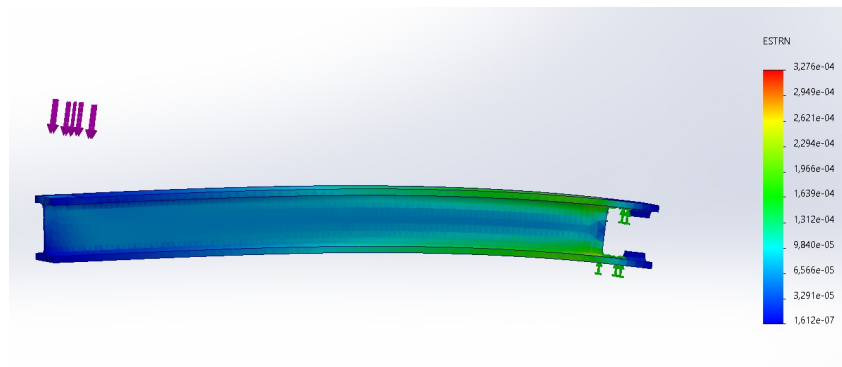


Fig. 19. Elastic strain profile demonstrating material behavior safely within the proportional limit.

These numerical results extracted from the analysis carried out in SolidWorks demonstrate a significant improvement in structural stiffness and also in stress distribution after re-designing the arms with an HEB 100 profile.

Comparison Between Preliminary and Optimized Designs

This iterative redesign process represented one of the most significant stages of this project because completing this requirement fully fulfilled the purpose of obtaining a functional prototype capable of meeting all the initial requirements.

Evaluated Parameter	Preliminary (Hollow Section)	Optimized (HEB 100)
Maximum Equivalent Stress	115.50 MPa	82.64 MPa
Maximum Vertical Deflection	4.80 mm	1.65 mm
Net Mass per Arm	12.93 kg	17.54 kg
Compliance with L/250 Limit	Failed	Satisfied
Minimum Safety Factor	2.16	3.02

TABLE 7
Comparison Between Preliminary and Optimized Lifting Arm Designs

Table 7 summarizes the quantitative differences between these two configurations, demonstrating that increasing the second moment of area through the HEB 100 profile considerably helped increase structural stiffness without compromising the safety factor.

Dynamic Kinematic Response

This integration of a multi-domain simulation makes it possible to evaluate the kinematic behavior during the operating cycles.

Parameter	Value
Maximum Stroke	615 mm
Lifting Time	11.0 s
Average Velocity	0.056 m/s
Design Payload	1500 kg

TABLE 8
Lifting Cycle Kinematic Characteristics

Table 8 summarizes the kinematic characteristics obtained during the steady-state lifting phase.

Position Profile

Fig. 20. Vertical position profile $x(t)$ of the lifting carriages during a full operational cycle.

Here, the vertical trajectory defines the useful stroke of the system. By analyzing the position tests, it can be concluded that the system satisfactorily lifts the load from a zero position point to a maximum operating height of 615 mm. Thus, the requirement is confirmed, demonstrating that the system provides sufficient ground clearance for any technician to comfortably perform the corresponding maintenance.

Velocity Profile

Fig. 21. Transient velocity profile $v(t)$ highlighting the steady-state elevation phase.

Hydraulic Performance

Parameter	Value
Pump Displacement	8.0 cc/rev
Nominal Flow Rate	11.3 L/min
Steady-State Working Pressure	105 bar
Pressure Relief Valve Setting	120 bar
Cylinder Bore Diameter	40 mm
Cylinder Rod Diameter	18 mm
Total Cylinder Stroke	615 mm

TABLE 9
Hydraulic System Performance and Dimensioning Specifications

Table 9 shows the hydraulic variables that represent the central energy transfer mechanism of the lift, where the performance specifications within the multi-domain simulation are explained.

System Pressure

Fig. 22. Hydraulic pressure evolution $P(t)$ recorded at the pump outlet during the lifting cycle.

In the pressure graph, an initial peak can be observed to overcome static friction and load inertia, stabilizing at approximately 105 bar. This value safely remains below the 120 bar relief valve limit, preventing unwanted fluid bypass and demonstrating the functionality of the

system.

Cylinder Force and Kinematic Constraint Analysis

Different cases were presented with a specific restriction in the prismatic motion, with the upper limit being a variable that produced an unusual change in the force recorded in this motion.

Joint Limit	Upper	Left Cylinder	Right Cylinder	Kinematic Observation
0.6 meters		$\approx 16,155 \text{ N}$	$\approx 260 \text{ N}$	Severe load monopolization due to strict rigid-body collision before target stroke.
1.0 meters		$\approx 8,650 \text{ N}$	$\approx 8,580 \text{ N}$	Uniform distribution during stroke; slight offset during hold due to solver tolerance.
No Limit		$\approx 8,619 \text{ N}$	$\approx 8,619 \text{ N}$	Perfect dynamic distribution dominated solely by fluid equilibrium.

TABLE 10
Effect of Mechanical Joint Limits on Simulated Actuator Force

Table 10 summarizes the impact of three distinct boundary conditions on the simulated actuator force distribution.

Condition A: Restrictive Upper Limit (0.6 meters). Condition 1: In this first scenario, the motion limits that were configured were intentionally set at 0.6 m, shorter than the steady-state stroke of 0.615 m defined by the hydraulic volume.

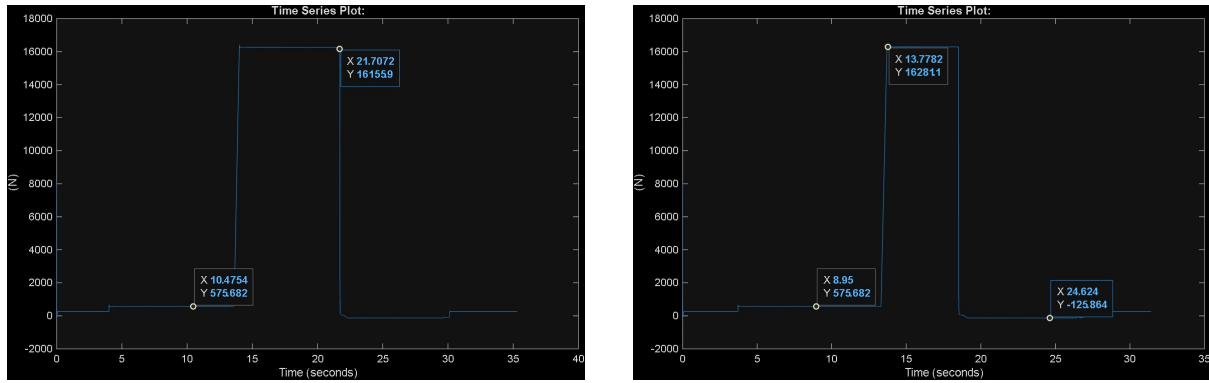


Fig. 23. Transient force profile $F_c(t)$ for left and right cylinders under a restrictive 0.6 m stroke limit.

As can be seen in Figure 23, some type of anomaly occurs when reaching the geometric limit, since the cylinder force approaches 16,155 Newtons, generating an inconsistency and showing approximately the total force exerted by the complete system.

Condition B: Holstered Upper Limit (1.0 meters). Condition 2: To mitigate this hard-stop collision, the upper limit was extended to one meter, allowing the cylinders to freely reach their maximum stroke controlled by the hydraulic actuator.

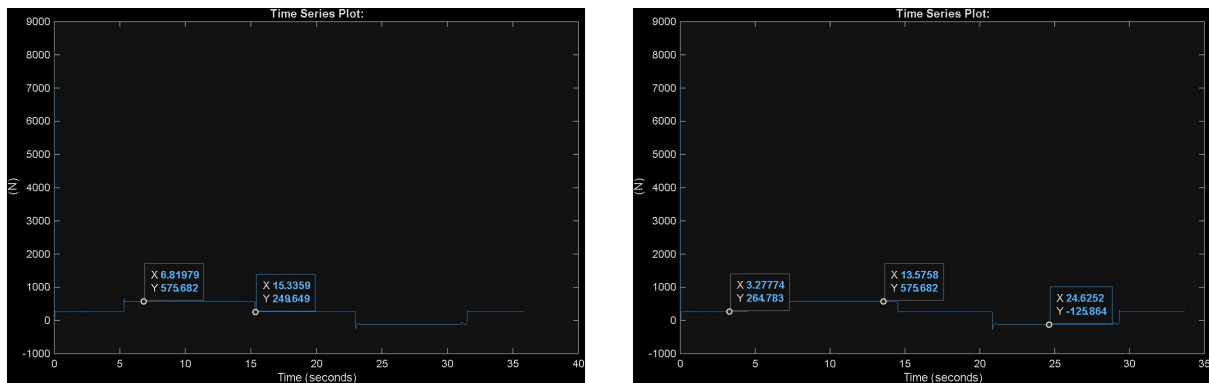


Fig. 24. Transient force profile $F_c(t)$ for left and right cylinders under a 1.0 m expanded limit.

Under this condition, the system apparently behaves normally during the lifting phase, distributing the load uniformly at approximately 8,600 Newtons per side. Small numerical mismatches occur during the holding phase, apparently because the solver balances microscopic positional differences without a mechanical restriction on which it can rely.

Condition C: No Upper Limit. Condition 3: In this final case, the upper limit was completely removed, thereby disabling the restriction of the mathematical model and making it a model purely dependent on the hydraulic actuators.

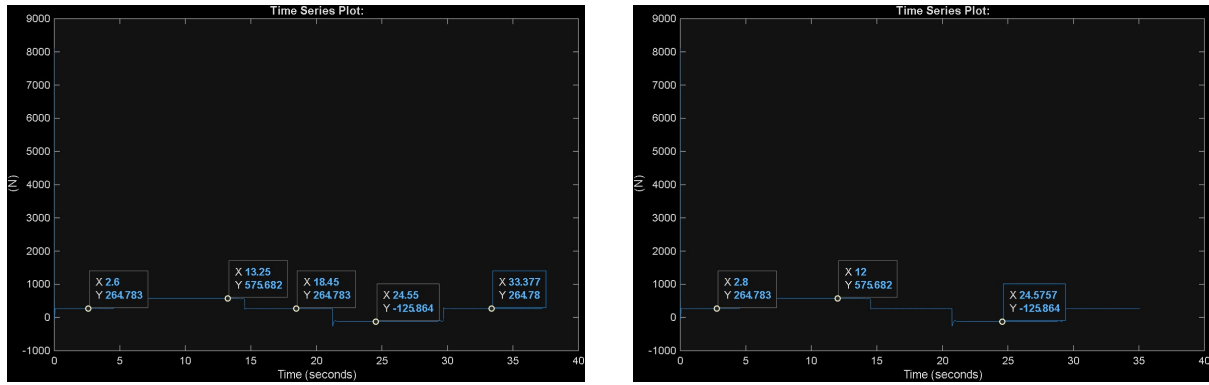


Fig. 25. Transient force profile $F_c(t)$ for left and right cylinders with no mechanical upper limit.

As can be seen in Figure 25, removing the upper limit also eliminates the hyperstatic conflict. This required holding force per cylinder stabilizes at the theoretical value of 8618.95 N. In a real physical implementation, the structural elasticity of the arms and the compressibility of the hydraulic oil would help naturally create a more balanced force distribution between both actuators.

Synchronization Verification

Synchronization, in this synchronization step of the two posts it is important to demonstrate that symmetry exists.

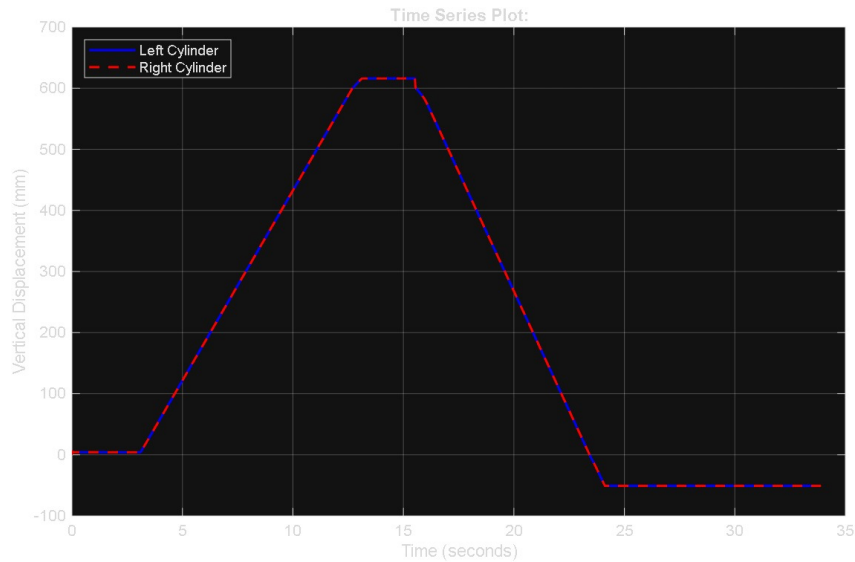


Fig. 26. Superimposed vertical displacement of the left and right lifting cylinders.

In Figure 26 the visual symmetry is confirmed, and the difference calculation demonstrates an absolute synchronization error lower than one millimeter during the lifting phase. This tiny deviation validates compliance with the requirement, confirming that the architecture guarantees symmetry in the load.

Control Sequence Validation

The implementation of the Stateflow logic acts as the electronic supervisor of this process, satisfying the automation requirement.

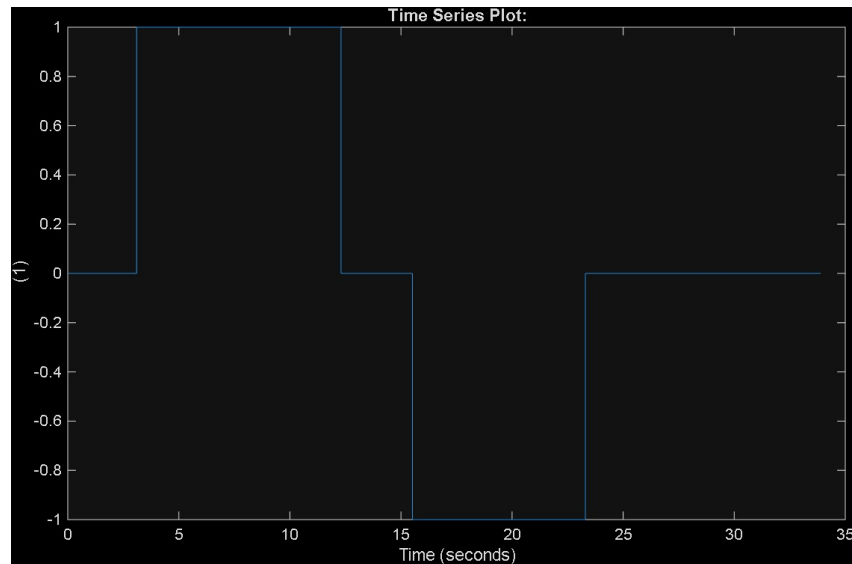


Fig. 27. Digital control signal executing the automated sequence: Standby (0), Elevation (1), and Descent (-1).

Operational Stage	Duration (Seconds)
Initial Standby	3.0
Elevation (Value: +1)	11.0
Maintenance Hold (Value: 0)	10.0
Gravity Descent (Value: -1)	10.0
Total Automated Cycle	34.0

TABLE 11
Cycle Time Analysis of the Automated Lifting Sequence

Table 11 breaks down the temporal duration of the automated states of this machine. Here we can validate that the maintenance cycle is achieved without requiring continuous manipulation by the technician.

Sensitivity Analysis

A sensitivity analysis was also performed on this virtual prototype by varying friction.

Friction Variation

The friction coefficient along the linear guides can degrade over time due to lack of lubrication or wear.

Friction Coefficient (μ)	Lifting Time (s)	Steady-State Pressure (bar)
0.00 (Ideal)	10.85	92.5
0.15 (Nominal)	11.00	105.0
0.25 (Adverse)	11.25	114.2

TABLE 12

System Performance under Varying Guide Rail Friction Coefficients

These results indicate that the hydraulic architecture has a sufficient operating margin. Even under adverse friction conditions, the pressure remains below the 120 bar relief valve setting.

Payload Variation

Applied Payload (kg)	Lifting Time (s)	Steady-State Pressure (bar)
500	10.90	45.3
1000	10.95	78.6
1500 (Design Load)	11.00	105.0

TABLE 13

Hydraulic and Kinematic Response under Payload Variations

Likewise, increasing the payload under the nominal condition of 1500 kg produced an almost linear increase in pressure demand. This behavior demonstrates that the selected dimensions for the pump and the actuator are correct.

Transient Stability and Viscous Damping

To evaluate the dynamic stability of the carriages during the initial acceleration phase, the viscous damping coefficient inside the prismatic guides was varied. Tests were performed under three discrete damping conditions with values of 0, 5000, and 8000 N(m/s).

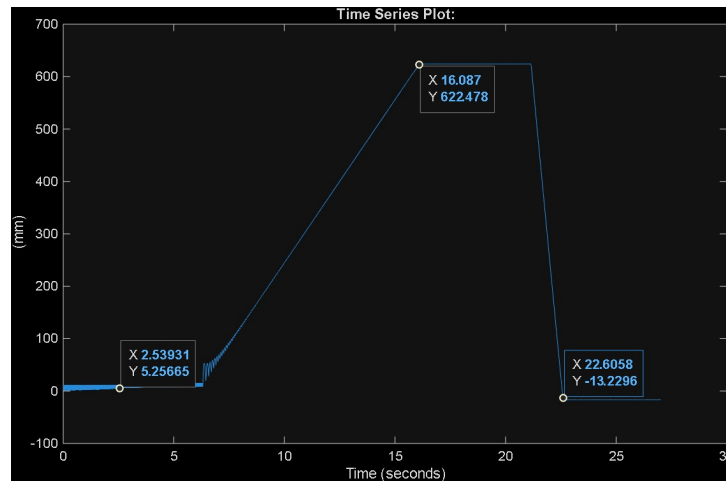


Fig. 28. Transient position profile with 0 N/(m/s) damping, exhibiting severe initial oscillation.

In the kinematic graphs we can see that in the absence of mechanical damping, the system initially experiences position oscillations or rattling during the standby phase, that is, it continues falling and bouncing without friction for a long time as shown in Figure 28.

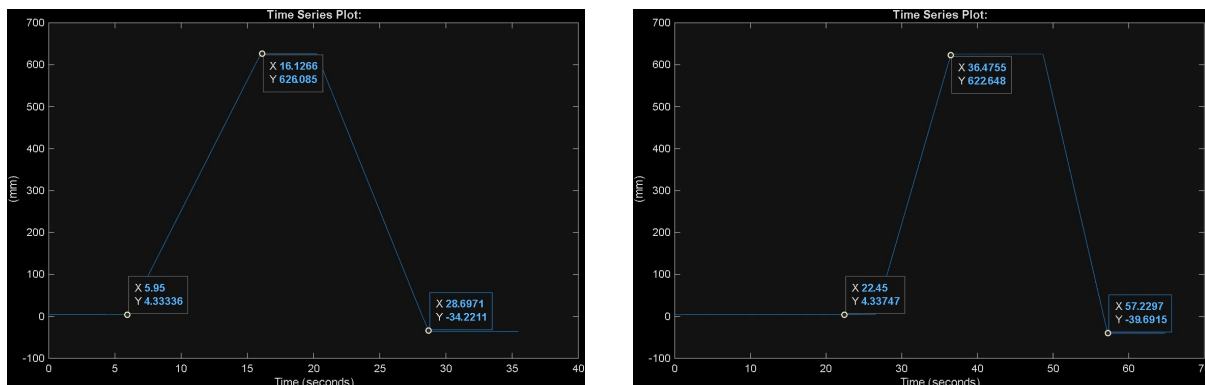


Fig. 29. Transient position profiles with 5000 N/(m/s) (left) and 8000 N/(m/s) (right) viscous damping.

As the damping coefficient was increased to 5000 and subsequently to 8000, as shown in Figure 29, the initial oscillation practically disappeared, leaving the system motionless from the beginning in order to perform an accurate symmetric motion analysis. The condition with a damping coefficient of 8000 provides an almost critically damped response, ensuring a lifting profile that shows no complications. This analysis confirms that integrating linear guides with appropriate friction and damping characteristics can prevent dynamic instability and ensure the unanchored payload during operation.

Requirement Traceability Matrix

To close the V-model development cycle, Table 14 relates the initial project requirements with the obtained validation evidence.

ID	Requirement Description	Verification Status
FR-01	Safely elevate a design load of 1500 kg.	Satisfied (Sections 3.1 and 3.3)
FR-02	Provide sufficient vertical displacement.	Satisfied (Section 3.2.1)
FR-03	Provide synchronized and symmetrical lifting.	Satisfied (Section 3.4)
FR-04	Allow automatic operation.	Satisfied (Section 3.5)
FR-05	Optimize operational efficiency (no auxiliary lifting).	Satisfied (Section 3.8)
NFR-01	Minimum Safety Factor (FS) of 3.0.	Satisfied (Section 3.1.3)
NFR-02	Compatible with standard industrial grids.	Satisfied (Table 3.2 parameters)
NFR-03	Control interface from a safe distance.	Satisfied (Section 3.5)
NFR-04	Validated through modeling and simulation.	Satisfied (Chapter 3 Validation)

TABLE 14
Requirement Traceability and Verification Matrix

Comparison with Conventional Procedure

The fundamental value of this engineering system lies mainly in its impact within maintenance environments.

Operational Aspect	Conventional Method	Proposed System
Preparation Time efficiency	Low (Multi-stage process)	High (Single automated cycle)
Operators Required	Minimum 2	1
Forklift Requirement	Required	Not Required
Manual Jack Stands Required	Required	Not Required
Number of Hoisting Operations	Multiple (Asymmetrical)	Single
Load Symmetry During Lifting	No	Yes
Process Automation	None (Manual)	Fully Automated

TABLE 15
Qualitative Operational Comparison: Conventional vs. Proposed Automated Method

Table 15 represents a comparison between the traditional maintenance approach and the proposed automatic lifting system.

Comparative Performance Summary

Design Variable	Target Specification	Obtained Value	Status
Design Payload Capacity	1500 kg	1500 kg	Satisfied (✓)
Useful Vertical Displacement	≥ 600 mm	615 mm	Satisfied (✓)
Structural Safety Factor (FS)	≥ 3.0	3.02	Satisfied (✓)
Maximum Arm Elastic Deflection	≤ 4.0 mm	1.65 mm	Satisfied (✓)
Peak Synchronization Error	< 5.0 mm	< 1.0 mm	Satisfied (✓)
Steady-State Working Pressure	< 120 bar	105 bar	Satisfied (✓)
Operational Automation Control	Sequential Loop	Achieved	Satisfied (✓)

TABLE 16
Comparative Design Optimization Summary Table

In Table 16 we can explicitly see the target design criteria against the values obtained through simulation.

Simulation limitations

Although the simulation can provide different valuable information points regarding the behavior of the system, simplifying assumptions were also adopted during the validation stage. For example, the components in the solver were modeled as rigid bodies, manufacturing and assembly tolerances were neglected, and parameters such as complex fluids, including the dependence of oil compressibility on temperature and microleakages through the internal seals of the valves, were simplified.

Discussion of Results

This validation process carried out through finite element analysis and multi-domain simulation indicates that the proposed automatic electro-hydraulic lifting system satisfies all objectives and requirements.

Each iterative redesign process proved to be very helpful. Testing and replacing components or structures allowed the system to reach a satisfactory final version, meeting both the safety factor of 3.0 and the strict permissible deflection criteria.

Likewise, the hydraulic pressure levels obtained, approximately 105 bar under nominal conditions, proved to be consistent with the pressure ranges typically measured in medium-capacity electro-hydraulic lifting systems.

The analysis of actuator forces in the simulation also demonstrated that when kinematic limits are reached, the loads in that block can increase sharply.

Similarly, achieving a synchronization error below 1 mm demonstrates that the symmetry between both posts can provide safe motion, which complies with the requirements. Furthermore, the sensitivity analysis revealed that the selected hydraulic power unit sufficiently meets the operating margin. By operating everything under the sequence of the digital state machine, that is, Stateflow, the prototype demonstrates that there is feasibility in safely replacing the dangerous forklift procedures currently used in dolly maintenance operations.

Chapter IV

Conclusions and Recommendations

Conclusions

- It is concluded that it was possible to successfully design and virtually validate an automatic electro-hydraulic system capable of carrying out maintenance cycles for the dollies, eliminating the risks associated with unconventional practices.
- The mechanical and hydraulic requirements were correctly determined through analytical calculations. This made it possible to properly size the structure capable of guaranteeing a safety factor of 3.02, and also to provide the necessary flow and pressure for the system.
- It was concluded that the design of this architecture and modeling managed to eliminate the need for asymmetrical lifting and the use of external tools, providing an optimal spatial arrangement to perform maintenance on the equipment without obstacles.
- It can be concluded that the simulation in this multi-domain model allowed the analysis of the dynamic behavior of the system and also validated that its logical states are capable of managing a flow without manual intervention during maintenance.
- We can also conclude that the simulation successfully validated the structural integrity and synchronization capability of this prototype. This system managed to lift the load with minimal deviation, demonstrating that the proposal ensures automatic and safe behavior.

Recommendations

- For future work, it is recommended to continue to a manufacturing and assembly stage of the full-scale physical prototype. This will make it possible to compare the data obtained in this simulation with the physical behavior of the prototype.
- It is recommended to add physical cylinder displacement sensors and to program a closed-loop control, for example a PID, to allow the system to automatically correct any asymmetry if in the future the vehicle presents an extremely unbalanced load.
- It is recommended for future research to subject the model to thermal fatigue and continuous mechanical wear tests, thus making it possible to simulate years of operation.
- It is recommended to expand the research in order to adapt this machine with a similar architecture and address the different types of heavier vehicles that exist in the hangar.

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