



Master Dissertation in Mechanical Engineering

Research and optimization of a Rail Guided Vehicle

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Abstract

Rail Guided Vehicles (RGV) are fast and flexible rail systems that make it possible to transport large quantities of heavy goods over long distances at low cost with high efficiency.

This allows key benefits when compared with other solutions such as conveyors: avoidance of bottlenecks, a modular and flexible design, and reduction of downtime.

RGV systems are vital for storage, handling, and order fulfillment in a wide range of industries. Due to their widespread application, they are also seen as mission-critical to daily operations for many large industrial plants.

Using this technology, Körber aims to reduce the reliance on overly long and slower-paced conveyor lines and increase speed for greater productivity gains when it comes to transporting goods across the warehouse. Ultimately, this automated solution will help to significantly increase operational efficiency, and reduce costs.

However, nowadays, the industry is ever-changing and evolving. New solutions and designs are required. This dissertation follows that objective: to improve the current Körber's RGV solution by reducing the RGV's components mass while maintaining the structural integrity and quality, allowing to lower the production cost without affecting the efficiency.

In order to accomplish the aforementioned objectives, firstly a study involving the heavier parts that Körber currently uses was developed. Using ANSYS software, a simulation of those parts was performed.

Following that, a study was carried out in order to choose the best optimization method for the frame of the structure: topological optimization. Once again, taking advantage of ANSYS algorithms and processing power, several iterations were calculated until a proper solution was achieved. Through an analytical method, the same optimization was matured for the wheels' bar.

At last, the new best suitable solution was obtained, compared with the current one, and drawn with the assistance of Inventor software.

Resumo

Os Veículos Guiados por Trilhos (RGV) são sistemas que recorrem a rails, rápidos e flexíveis, que possibilitam o transporte de grandes quantidades de mercadorias pesadas com baixo custo e alta eficiência.

Revelam-se benéficos em comparação com outras soluções, como tapetes rolantes, reduzindo o tempo de inatividade e tendo um design modular e flexível.

Os sistemas RGV são vitais para armazenamento, manuseamento e cumprimento de pedidos nas mais variadas indústrias. Devido à sua ampla aplicação, são vistos como peças fundamentais para as operações diárias de muitas fábricas.

Usando esta tecnologia, a Körber visa reduzir a dependência de linhas de transporte excessivamente longas e aumentar a velocidade para maiores ganhos de produtividade, quando se trata de transportar mercadorias no armazém. Em última análise, esta solução automática ajudará a aumentar significativamente a eficiência operacional e a reduzir os custos.

No entanto, hoje em dia, a indústria está em constante mudança e evolução. Novas soluções e designs são necessários. Esta dissertação segue esse objetivo: melhorar a solução atual da estrutura dos RGVs da Körber, reduzindo a massa dos seus componentes, mantendo a integridade e qualidade estrutural, permitindo, assim, reduzir o custo de produção sem afetar a eficiência.

Para isso, foi desenvolvido um estudo envolvendo os componentes atuais mais pesados que a Körber utiliza nos seus RGVs. Utilizando o software ANSYS, foi realizada a simulação desses componentes.

Em seguida, foi realizado um estudo para a escolha do melhor método de otimização para o quadro da estrutura: a otimização topológica. Mais uma vez, aproveitando os algoritmos do ANSYS e o seu poder de processamento, várias iterações foram calculadas até que uma solução adequada fosse alcançada. Através de um método analítico, a mesma otimização foi desenvolvida para a barra das rodas.

Por fim, obteve-se a nova e mais adequada solução, tendo-se comparado esta solução com a atual e desenhando-se a nova solução com o auxílio do software Inventor.

“To dare is to lose one’s footing momentarily. Not to dare is to lose oneself.”

Søren Kierkegaard

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Chapter 1

Introduction

1.1 Background and Main Objectives

Warehouses are one of the essential components of logistics and supply chains, it is easily understood that the efficiency of the whole chain is affected by the performance of warehouse operations and the storage and retrieval of goods [1]. Automated warehouses are stock and product management solutions, through automated storage and retrieval systems, robotic mobile fulfillment systems, or mobile rack warehouses, as it allows companies to save on manual work and space utilization many have started to switch to fully automated warehousing systems [2].

Körber Supply Chain S.A. is one of the top companies regarding Warehouse Management Systems where logistics and intralogistics play a major role. Körber Supply Chain S.A. can provide multiple solutions: supply chain software, software consulting and implementation, SAP software consulting and implementation, robotics, system integration and automation, material handling equipment, manufacturing logistics [3].

Since guided vehicles are very common in automated warehouses, any improvement in them translates into an improvement in the efficiency of the supply chain and/or in the company finances. With this in mind, Körber Supply Chain proposed optimizing the Rail Guided Vehicle (RGV) structure regarding its mass, stresses and deformations.

1.2 Körber Supply Chain S.A. Presentation

Körber Supply Chain S.A. is a leading supplier of fully integrated digitalized processes as it develops, produces, and installs turnkey solutions for intra-logistics applications as well as the integration of the various systems.

In 1984 Efacec created the department of Automation and Robotics. In order to address the growing business, seven years later, the department was set up as an independent company: Efacec Automação e Robótica, S.A. In 1997, the increasing demand in Southeast Asia created the need to found Efacec Singapore Pte Ltd. as a subsidiary. To further establish its presence

in Asia, in 1998, a joint venture with Godrej was created: Godrej-Efacec Automation and Robotics, S.A. By merging Efacec Automation and Robotics, S.A. and several companies within the group, Efacec Engenharia e Sistemas, S.A. was established in 2010 which allowed addressing relevant public projects in Europe, the USA and South America.

To enhance positioning as a worldwide leading supplier of automated material handling and storage systems, in 2014, Efacec Handling Solutions, S.A. emerged from Efacec Engenharia e Sistemas, S.A. In the next year, the newly emerged company was acquired by the international technology group Körber which lead to the internationalization of its Business Area Logistics Systems. In 2017, Efacec Handling Solutions, S.A. changed its name to Consoveyo, S.A.

Finally, in 2020, in order to conquer the supply chain industry, Körber groups all 12 companies and Consoveyo becomes Körber Supply Chain Automation [4].

Körber Supply Chain S.A. belongs to the international technology group Körber. Aberle, Aberle Software, Cirrus Logistics, Cohesio Group, Consoveyo, DMLogic, HighJump, Inconso, Otimis, Langhammer, Riantics and Voiteq also belong to this group. Körber employs more than 2300 professionals and has more than 100 business partners worldwide.

Currently, Körber Supply Chain PT, S.A products range from simple conveyors to fully automated vehicles that allow the transportation and storage of pallets and their respective load. Stacker cranes, as shown in Figure 1.1, are the best selling product as it allows vertical storage meaning that stock can be stored taking less floor area. This solution enables more than 100 000 storage units per warehouse.

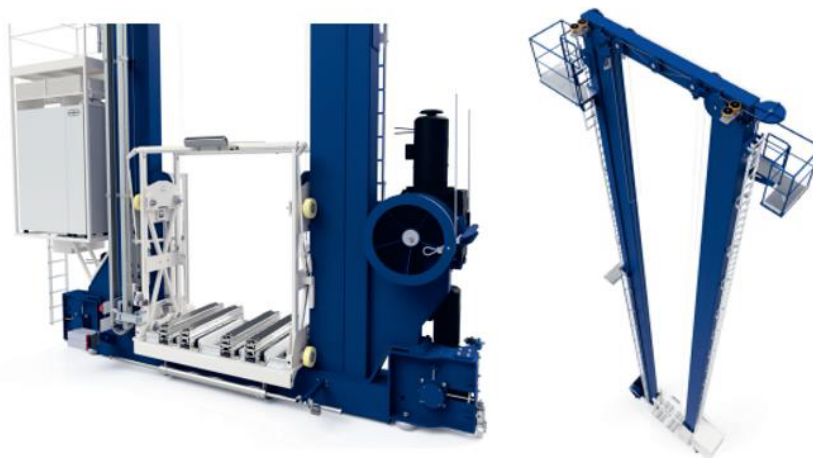


Figure 1.1: Stacker crane.

1.3 Dissertation Outline

The first chapter contains a brief introduction to the project, describes the structure of this dissertation and the objectives this dissertation aims for.

The second chapter covers a literature review about logistics, automated warehouses, rail guided vehicles and their solutions, EU laws regarding machinery design standards, the correct modeling of metal structures through the finite element method, and topological optimization.

The third chapter addresses the RGV characterization and it is divided in two parts, each covering a development step for the RGV: the project restrictions and the structural analysis.

The fourth chapter explores the methodology for optimization of the RGV design focusing on the frame and the wheels' bar.

The fifth chapter covers the design solutions, the requirements fulfillment, the solutions' comparison.

The sixth chapter contains the most relevant conclusions of the dissertation and suggests further development and optimization of the RGV components.

Chapter 2

Literature Review

This chapter contains a brief literature review that addresses logistics and intralogistics, automated warehouses, rail guided vehicles and their solution, EU laws regarding machinery design standards and European Materials Handling Federation rules. It also explains the correct modeling of metal structures through the finite element method and the two most known topological optimization methods.

2.1 Logistics

Nowadays, Logistics is seen as providing cost and time effectiveness services, mainly commercial activities. Obviously, logistics contributes vastly to the business success of any organization. *In loco*, the concept translates in the transport of goods from one point to another, warehousing them in a suitable place, inventory, packaging, and other administrative activities such as order processing [5].

2.1.1 Intralogistics

Intralogistics can be defined as "the organization, control, execution and optimization of in-plant material and information flows, and of goods transshipment in industry, distribution and public sector facilities" [6]. The rising need for higher productivity in the automated warehouses motivated by the consumers' increment makes intralogistics the perfect solution to keep a business competitive, allowing it to be more efficient and resourceful.

2.2 Automated Warehouses

Being warehouses one of the essential components of logistics and supply chains, it is easily understood that the efficiency of the whole chain is affected by the performance of warehouse operations and the storage and retrieval of goods [1].

Automated warehouses (AS/RS - automated storage and retrieval system) are stock and product management solutions, through automated storage and retrieval systems, robotic mobile fulfillment systems or mobile rack warehouses, as it allows companies to save on manual work and space utilization. Many have started to switch to fully automated warehousing systems [2]. These systems range from small automated warehouses and big computer controlled storage systems fully integrated from the manufacturing to the distribution of the products.

Usually, these systems are composed by storage racks that can have many configurations and a stacker crane that operates alongside them that retrieves and delivers the products from/to the storage units. Since the operation is fully automated there is no need to human labour in those tasks.

In order to bridge with the manufacturing and expedition zones there are peripheral equipment, namely, powered or gravity roller conveyors, chain conveyors, slat conveyors, elevators and automated vehicles (Rail Guided Vehicles-RGVs, Automated Guided Vehicles-AGVs) [3]. To simplify the transportation process the stock is handled via standardized pallets.

The flexibility of these systems allows for a variety of uses according to the clients' needs, automated warehouses can fulfill the following functions:

- Order picking: retrieval and deliver of products ordered by the operator;
- Storage: Long-term storage of stock;
- Consolidation: waiting area to store items until the order is completed;
- Assembly and Manufacturing: storage of components for assembly and manufacturing;
- Retail: supply of a large amount of items to costumer service operations.

Therefore automated warehouses can lower labour costs, as the need to employ people is reduced; are faster and more efficient than people-based systems; more accurate since the human error is minimized; safer; space can be optimised and they allow for round-the-clock efficiency.

2.3 Rail Guided Vehicles

Rail Guided Vehicle, commonly known as RGV, is a flexible transportation vehicle. It has a wide range of applications in logistic systems and station production lines. The RGV makes it possible to transport a large number of heavy goods over long distances, providing transportation cost reduction as well as improving transportation efficiency, depending on the plan and instructions for material transportation. It can also operate in an environment where there is not much room and movement is restricted. RGVs are the perfect match for high-density storage three-dimensional warehouses. The RGV's

structure is relatively simple: frame, driving wheels, follower wheels, front and rear bumpers, load handling device, communication system, electrical system, and cover plates. Such simplicity allows it to be very robust and versatile, making it able to meet a wide set of requirements during operation. Along with the relatively low maintenance cost, the RGV is extremely reliable.

Although it has many advantages, the fact that it runs on tracks makes it very costly if there's the need to redefine its route.

Since guided vehicles are very common in automated warehouses, any improvement in them translates into an improvement in the efficiency of the supply chain and/or in the company finances. With this in mind, Körber Supply Chain proposed optimizing a Rail Guided Vehicle regarding its mass and its cost.

2.3.1 RGV's Solutions

The RGVs are customizable for the use according to the clients' needs, and various specifications can differ in order to match the clients' demand: number of rails that the RGV runs on, the maximum payload that it is able to transport, maximum straight traveling speed, the type of load handling device are probably the most important ones.

The number of rails in a RGV project can vary from 1 to 2 rails, although 2 rails allow a higher traveling speed it also makes the project more expensive.

Both the maximum payload and the maximum straight traveling speed concede a greater efficiency to the warehouse, as they allow the RGV to transport more cargo in less time. Even though it's easy to assume that to have maximum efficiency with these two parameters it's to maximize both, usually, the increase of one of them comes at the expense of the decrease in the other.

Regarding the load handling devices, the most common are the roller conveyor (RCV), the chain conveyor (CCV), and the telescopic forks. The RCV handles rigid surface loads and supports them with at least three rollers. Usually, there are two main chain driving systems: transmission between the rollers with different chains or one chain transmission to all the rollers. The first one supports much larger loads but the second one is simpler and cheaper [7]. The CCV utilizes chains to pass on binary as well as to transport the loads. The transport is made on a closed-loop chain between two driving wheels aligned on the same axis imposing a synchronized movement to the chains. The minimum number of chain rails is two, but they may have as much as necessary according to the cargo dimensions and weight [7]. The telescopic forks consist of two telescopic forks operating in parallel that expand and retract allowing to move the pallets according to the need.

In order to have better knowledge of the different RGV's solutions, a market survey was made and the details can be seen in Table 2.1.

Table 2.1: RGV's solutions.

Competitors	Number of Rails	Max. Payload / kg	Straight Traveling Speed / m/s	Load Handling Device Type	Power Supply
Körber Supply Chain	1	1500	3,5	CCV, RCV, Telescopic Forks	Bus bar
Daesung SE	2	2x5000	1,34	Conveyor, Telescopic Forks, Lifter	Bus bar
Cassioli	2	1500	5	CCV	Bus bar
Authoma	2	1500	2	CCV, RCV	Bus bar
Mecalux	1	1500	1,67	CCV, RCV	Bus bar
Eisenmann	2	1500/2x1500	1,34	CCV, RCV	Bus bar
Rofa	2	2400	3	CCV, others	Bus bar
Dambach	2	1500	2,5	CCV, RCV, others	Inductive energy
LogSystems	2	2500	2	CCV, RCV, others	Bus bar

2.4 EU laws

Before the creation of the European Union (EU), product trade between the different European countries was conditioned by the different technical specifications of each country, making the European market hard to commercial trade. For that reason, after the genesis of the EU, it was in the best interest to standardize the technical specifications in order to ease the European commercial trade. Taking that into account, legal guidelines were developed regarding the project, manufacturing, and commercialization of the products to ensure their safety.

Nowadays, machine safety is ensured by a set of legal guidelines with both economic and social scopes. From an economic standpoint, these rules establish a set of requirements for the equipment to enter the market. From a social standpoint, these rules focus on preventing accidents and upgrading the safety and health of the operators in the workplace [8]. The major guideline that regulates machine and equipment safety is Machine Guideline (Guideline 2006/42/EC). This guideline emphasizes the integration of safety into the design and relies on recognized technical specifications such as the standards

[9]. If the machines are used in the workplace, there is also the need to ensure the operators' safety, for that it should be taking into account Guideline 2009/104/EC that promotes better work conditions as well as high-level safety rules that protect the operators.

Regarding load handling equipment the *Fédération Européenne de la Manutention* (FEM) delivers a set of specifications to help the manufacturers ensure the equipment are safe to use [10].

2.4.1 Machine Guideline 2006/42/EC

The Machine Guideline (Guideline 2006/42/EC) is applied to every machine that is on the market or at work since the date it was made official, implying that every machine circulating in the European market must obey all the health and safety criteria in this guideline. The requirements from this guideline must be fulfilled in every phase of the project and manufacturing process, otherwise, the machine cannot be commercialized.

The main requirements from Machine Guideline are:

- Ensuring the machine fulfills every safety, health, and environmental requirement;
- Writing of the machine's technical papers with all the important documents;
- Emission of the EU declaration of conformity and incorporation of the CE stamp on the machine.

The safety, health, and environmental requirements must be studied and adopted during the project phase, implementing the necessary safety measures and/or warning the operator of the machine for possible dangers as well as imposing the need for individual safety equipment [9].

These Machine Guidelines comes from the technical specifications constitute a hierarchy of European Standards (ISO, EN, and IEC) aggregated in three levels:

- Type A – Basic safety guidelines. Fundamental safety guidelines applied to every piece of machinery.
- Type B – Guidelines applied to a large range of machines and it's divided into two categories:
 - Type B1: Specific safety aspects such as safety distances, surface temperature, and noise.
 - Type B2: Safety devices such as two-hand activation buttons, stoppage devices, pressure sensors, and safety barriers.
- Type C: Detailed guidelines applied to a specific machine or a particular group of machines.

Another way to ensure that the machine is safe is to do a Risk Evaluation where a risk level is defined, limiting the machine usage and implementing safety mechanisms [9]. This method allows the implementation of the technical solutions that ensure a safe operation of the machine, having an important role during the project phase of the machine. Some guidelines for risk evaluation are ISO EN 12100-1:2003 and ISO EN 12100-2:2003.

The next step regarding the Machine Guideline is the writing of the technical paper that should contain assemblage drawings, certificates, machine designing notes, machine's technical description, among others. This should work as proof that the machine was designed and manufactured according to the legislation.

The CE declaration is the document that certifies that the manufacturer takes full responsibility for the requirements from the Machine Guideline being fulfilled. After the declaration has been written, the CE must be stamped in the machine providing visual proof that the machine is following the European legislation.

2.4.2 Guideline 2009/104/CE

The Guideline 2009/104/CE ensures the minimum safety and health measures for the operator's usage of the work machine [9]. Although the safety and health requirements must be ensured during the design and manufacturing phases of the project, periodic audits must be done granting the proper machine functioning and preventing accidents [9]. Load handling devices are usually operated by workers, for this reason, the Guideline 2009/104/CE it's applied to equipment used in automated warehouses.

2.4.3 European Materials Handling Federation

The European Materials Handling Federation was founded in 1953. It's a non-profit commercial organization that is based in Brussels and represents the European makers of load handling and storage devices. Its main focus is to support the manufacturers by publishing standards with technical recommendations directed to the design engineers [10]. These standards are technical guides with specific rules to design equipment for automated warehouses, helping the design engineers with the structural analysis, safety factor, audit frequency, among others aspects. They are a complement to the existing legislation [10].

2.5 Finite Element Method

The finite element method, also known as FEM, is a process that allows to numerically solve differential equations that translate the phenomena from the engineering modeling [11].

In this method, each finite element can be seen as an independent sub-domain, subdivision of the whole domain which is the geometric space where the algebraic equations that model the problem are valid. Every element is interconnected to another by a shape function [12].

Each finite element has coordinate locations in space where the degrees of freedom (DOF) are defined, those locations form element boundaries called nodes. The collection of discretized elements and nodes is called the mesh and the process of its construction is called meshing.

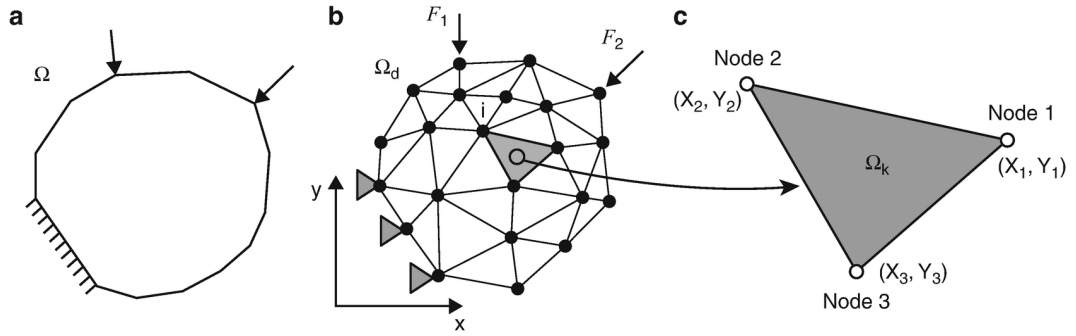


Figure 2.1: a) Boundary conditions; b) Mesh; c) Finite element [13].

Problems regarding boundary- and initial-value are formulated in a weak form, since it demands weaker continuity conditions. This enables the sum of the contributions of each finite element to result in an algebraic equation that characterizes the problem over the whole domain. The equation can be solved using computational help in the form of:

$$[K]\{u\} = \{F\} \quad (2.1)$$

Where $[K]$ represents the stiffness matrix, $\{u\}$ the displacement vector and $\{F\}$ the force vector.

In case of nonlinearities, it is not possible to isolate the vector u , because it is implicit in the stiffness matrix:

$$[K(\{u\})]\{u\} = \{F\} \quad (2.2)$$

To solve them it is common to use iterative schemes [13].

The use of any element depends on the type of the structure, geometry, applied loads and boundary conditions, computational time, and data required from the analysis [14].

The FEM allows for two main approaches that differentiate themselves for the output given: force or flexibility method and displacement or stiffness method. The first one deals with the internal forces as the unknowns of the analysis and the second one considers the nodal displacement as the unknowns of the analysis [14]. The latter method has been shown to have simpler finite element formulations in general consequently it is more desirable for computational purposes [15], as well as a smaller percentage of error for deflection when compared to Von Mises stress' error [16].

The discretization of the problem defines the number of nodes and, consequently, the number of unknowns. As the discretization is refined it may result in a large set of algebraic equations that need to be solved, so it is of extreme importance to find the right numerical method as well as the right solution strategy in order to mitigate the computational power used.

To summarize the analysis of a problem via Finite Element Method it is common to follow some typical steps:

- Discretize the problem domain into finite elements and nodes;
- Formulate element equations by local approximation;
- Assemble element equations to arrive structural equations;
- Modify and solve emerging equations with boundary conditions;
- Interpret the results.

2.5.1 Finite Element Modeling for Metal Structures

As seen in the RGV section, the vehicle has a metal structure, in order to analyze the stresses correctly, it is mandatory to have great criteria regarding the finite elements and mesh used. Those parameters depend on the geometry, cross-section classification, loading, and boundary conditions of the structural member [14].

Metal structures that are composed of compact sections can be modeled using solid elements. Solid or continuum elements are volume elements that do not include structural elements such as beams, shells, and trusses [14] as it is the case in the RGV's structure.

2.5.1.1 Choice of Element Type

The finite element method begins by dividing the structure into finite elements defined by nodes located at the element edge. The nodes' location must account for positions of changes in the structure, such as variation in geometry, material, loadings, and boundary conditions. Moreover, the choice of the finite element to represent the structure also ensures a good quality simulation. From a structural standpoint, the element types can be classified as truss, beam, frame, and shell elements.

2.5.1.1.1 Beam Elements Beam elements transfer lateral forces and bending moments, so the deformations associated with them are transverse displacement and rotation. The beam's dimensions of the cross-section are small compared to the axial dimension., such as distance between supports or distance between changes in the cross-section.

These elements are geometrically simple and have few degrees of freedom, as it's assumed that the member's deformation can be estimated from variables that are only dependent on the position along the beam axis.

Similarly, thin-walled, curved pipes exhibit much softer bending behavior than the one predicted by beam theory, since the pipe wall readily bends in its own section. Although for relatively small axial dimensions the choice of beam elements for these structures is valid [14].

For beam elements, ANSYS uses the BEAM188, its geometry can be seen in Figure 2.2. BEAM188 is a finite element based on Timoshenko beam theory, including shear-deformation effects. It also provides the options for unrestrained warping and restrained warping of cross-sections. It is suitable for analyzing slender to moderately thick beams. The element is a linear, quadratic, or cubic two-node beam element in 3-D. It has six or seven degrees of freedom: translations and rotations in the x,y and z directions, and the optional warping magnitude. BEAM188 is well-suited for linear, large rotation, and/or large strain nonlinear applications [17].

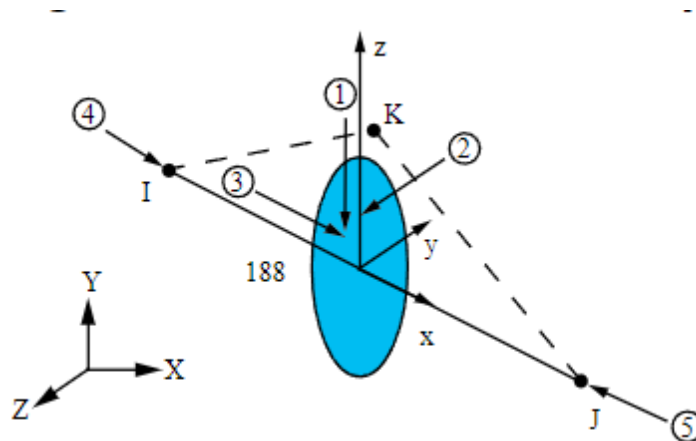


Figure 2.2: BEAM188 geometry [17].

2.5.1.1.2 Frame Elements Frame-like structures composed of straight and slender members can be efficiently modeled for design calculation with two-noded frame elements. These elements provide insight into the axial and transverse displacement as well as rotation, since they operate directly in terms of axial force, transverse force and bending moments at the elements' nodes.

The beam elements' fundamental assumption is that the elements' section cannot deform in its own plane is also valid for frame elements. This assumption should be carefully considered, especially for cases involving large amounts of bending or axial tension/compression of non-solid cross-sections.

Similar to beam elements, frame elements are commonly presented in 2D. However, some of the general-purpose programs have the ability to analyze beams and frames in 3D by including additional degrees of freedom for the elements in the plane perpendicular to their plane [14].

2.5.1.1.3 Solid Elements Solid elements are often used to model metal structures that are loaded by forces in 3D, usually called three-dimensional stress analysis. Tetrahedral and hexahedral elements are the main elements used, usually, to model structures that have a complicated 3D geometry. Linear 3D elements are the simplest 3D elements having only nodes at corners. Quadratic and cubic 3D elements, higher-order elements, are curved surface elements and have one or two nodes between corners, respectively.

Solid elements provide accurate results if not distorted, particularly for quadrilaterals and hexahedra. The triangular and tetrahedral elements are less sensitive to distortion.

Regarding solid elements, ANSYS uses SOLID186 and SOLID187. SOLID186 is a higher-order 3-D solid element (Figure 2.3) that exhibits quadratic displacement behavior. It has 20 nodes each node having three degrees of freedom, translations in the x, y, and z directions. SOLID187 is a higher-order 3-D solid element (Figure 2.4) that exhibits quadratic displacement behavior. It has 10 nodes each node having three degrees of freedom, translations in the x, y, and z directions CAD/CAM systems use both these elements since they are well suited for modeling irregular meshes, such as the ones created by those systems [17].

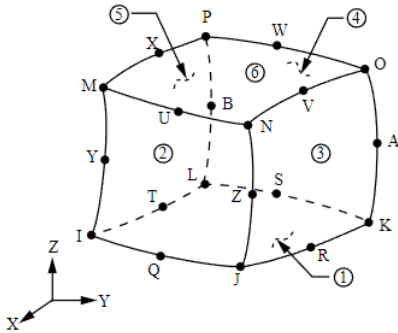


Figure 2.3 SOLID186 geometry [17].

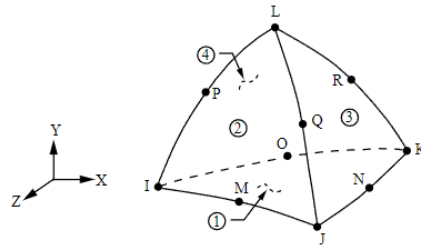


Figure 2.4 SOLID187 geometry [17].

2.5.1.2 Choice of Finite Element Mesh

With the best finite element chosen, there is the need to look into the geometry of the metal structural member in order to decide the most appropriate finite element mesh. To find the best mesh, a convergence study must be conducted. Convergence studies, in term of Finite Element Analysis, it's the iterative process of increasing the mesh density, until the difference between two consecutive solutions is small enough [18], as it can be seen in Figure 2.5. It is of major importance to conduct this study since it allows accurate results when plotting a FE simulation with less computational time.

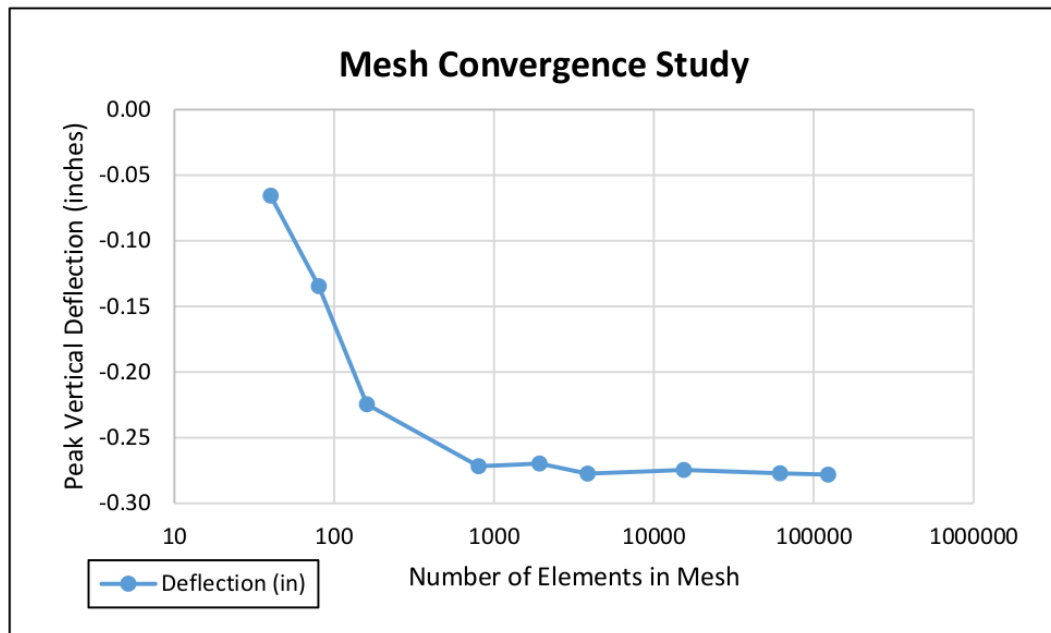


Figure 2.5: Convergence study example [19].

Another way to reduce computational time is to take advantage of symmetry. Through symmetry, it is possible to reduce the size of the FE mesh. However, it must be done very carefully, since the symmetry regarding FEA may be defined as correspondence in size, shape, the position of loads, material properties, boundary conditions, residual stresses due to processing, initial local, and overall geometric imperfections that are on opposite sides of a dividing line or plane, otherwise the results cannot be trusted. Nowadays, finite element software have the ability to perform meshing of the metal structures automatically, although, in many cases, the resulting fine mesh may take huge time in the process analysis. Due to this, the most efficient way to create a mesh is to use the fundamentals in a guided meshing that may be offered in the software. This way, automatic meshing may be of great benefit [14].

2.5.1.3 Load Application

In order to get accurate results, loads applied on metallic structures must be simulated as precisely as possible relative to the real loads, since any deviation from the actual loads affects the accuracy of the results. Usually, there are two types of loads: concentrated loads and distributed loads. The first ones are incorporated in the finite element models by specifying nodes, associated degrees of freedom, magnitude, and direction of the applied concentrated forces and moments. The latter ones are applied on the elements' faces to simulate surfaces distributed loads, a careful application is needed in these types of loads since it must be suitable for each element type [14].

2.5.1.4 Boundary Conditions

Boundary conditions allow to specify the values of all basic solutions variables at nodes, they can be used as model input data to define zero-valued boundary conditions and as history data to add, modify or remove zero-value or nonzero boundary conditions. Similarly to the applied loads modeling, boundary conditions application must be identical to the actual application, otherwise the results will not be accurate. Furthermore, careful application must be extended to guarantee that the model is not overconstrained. In case the modeler chooses to apply symmetry to reduce the computational time, there is the need to apply boundary conditions in the symmetry surface to replicate the entire structure. The nodes in the symmetry plane must be either totally constrained (no rotation or displacement) or partially constrained using a spring with constraint values depending on the stiffness related to the degrees of freedom [14].

2.6 Topological Optimization

Topology optimization has a wide range of applications in the engineering fields, it has been used by mechanical and civil engineers for many years allowing them to minimize the amount of material and the strain energy in structures while maintaining their mechanical strength[Bendsoe et al. 2003]. Nowadays, topology optimization is mostly used at the concept level of a design process, however, these results need to be adapted often, due to the free forms solutions that are very difficult to manufacture. For this reason, increasing the manufacturability in the topology optimization solutions is an active field of research [20]. Topology optimization is an algorithmic process that spatially optimizes the distribution of material within a defined domain, by fulfilling a given set of loads, boundary conditions, constraints and minimizing a pre-defined cost function [20, 21]. Typically, the plane or volumetric structures topology optimization approach is based on the finite element method. The problem domain is discretized in finite elements and then, to each element, is assigned full material, partial material, or lack of material, in an iterative scheme that converges to the optimal material distribution inside the domain.

The two most established methods for topology optimization are the SIMP (Solid Isotropic Microstructure with Penalization) method and the ESO (Evolutionary Structural Optimization) method [22].

2.6.1 Solid Isotropic Microstructure with Penalization - SIMP

This method is the most studied, implemented in commercial software, as well as, mathematically well-defined of all topology optimization methods. It is based on the idea of the homogenization method that aims for a functional

relationship between the material's density and the effective material properties. To find this relationship it is necessary to determine the homogenized elasticity tensor for a given microstructure that can define an appropriate material distribution, ensuring that the solution can be a real structure [23]. This technique minimizes the strain energy for an initial structure volume by rearranging the material density distribution inside the design domain. The design volume is divided into a grid of N elements and to each element e is attributed a fractional material density ρ_e . So, the objective function is the strain energy SE under a constraint of target volume V^* that can be mathematically translated to:

$$SE(P) = \sum_{e=1}^N (\rho_e)^p [u_e]^T [k_e] [u_e] \quad (2.3)$$

subjected to the constraints:

$$V^* - \sum_{e=1}^N V_e \rho_e = 0 \quad (2.4)$$

$$0 < \rho_{min} \leq \rho_e \leq 1 \quad (2.5)$$

Where $[u_e]$ represents the nodal displacement vector and $[k_e]$ the stiffness matrix of element e .

The vector P gathers the densities of the microstructures and represents the optimization parameters. The densities of the microstructures are gathered in the vector P and represent the optimization parameters.

ρ_{min} is the minimum allowable density used to characterize the empty elements, however, must be defined as greater than zero to ensure the stability of the finite element analysis. The penalty factor p is the main parameter of the method, its value being crucial to the algorithm behavior and success. p invigorates the development of elements which are either black ($\rho=1$) or white ($\rho=\rho_{min}$), by decreasing is needed the participation of fractional density elements to the total structural stiffness [22].

The iterative process can be visualized in Figure 2.6 to give a better understanding of this method.

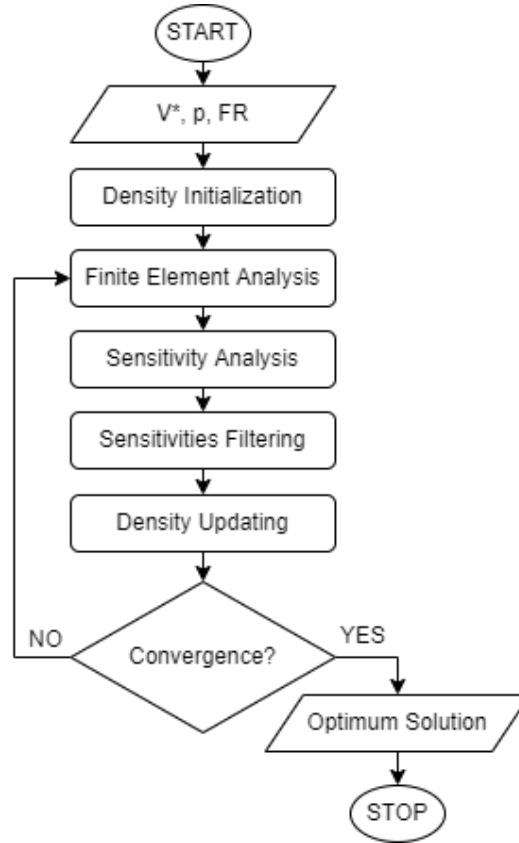


Figure 2.6: Flow chart for the SIMP algorithm [22].

2.6.1.1 Optimality Criteria - OC

The optimality criteria method proved to be an efficient process to solve topology optimization problems. Thus, it is a classic approach to a discretized structural optimization problem for the numerical solution [24]. Compared to mathematical programming methods, OC methods consider the knowledge of the physics and mechanics of the respective problem. Derived from the response behavior of modifications of the physical model that is implemented into the algorithm, OC has redesigned rules and a convergence behavior, which is not possible to obtain with mathematical optimizers [25].

These criteria are well proven for shape and topology, in which a wide number of design variables are required, being an effective method for solving large-scale problems. Not only that but also each design variable is updated independently of other design variables, except the resealing for satisfying the volume constrain, making the algorithm successful. Additionally, OC requires basic programming to solve the compliance topology design problem, with minimal lines of extra code required in the FEM code [25].

However, OC presents limitations in problems where volume is not the only constraint and in application areas where the physical and mechanical knowledge is not well documented. It is also important to note that the specific strain energy remains constant along with all areas of intermediate density

while being lower for regions of minimum density and higher for regions where the density is equal to 1 [25].

2.6.2 Bi-directional Evolutionary Structural Optimization - BESO

This method has been applied in various optimization research areas since it has been developed by Xie and Steven in 1993. Besides being easily understandable, this method is also easy to program and link with existing computer-aided engineering software like ANSYS CFX®, ANSYS Fluent®, or Nastran®. In the ESO method inefficient material is continuously removed from the structure via the rejection criterion which identifies the unneeded material. This is translated by assigning to each element in the discrete design domain a density of 0 or 1, depending on if the material was removed or not, respectively. Afterward, a model called Addictive Evolutionary Structural Optimization (AESO) was proposed based on the premise of adding material from a structure that connects the seed points (loads and supports) with a minimum number of elements. The material is added around points with high sensitivity. Later, the BESO (Bi-directional Evolutionary Structural Optimization) method was introduced, backed by the idea of surpassing the deficiencies of both ESO and AESO methods. In this method, the material is added or eliminated based on the need and allows the consideration of 3D domains and multiple load cases. The formulation of BESO's optimization problem minimizes the mean compliance C , under a target volume constraint V^* :

$$C(X) = \frac{1}{2} \{F\}^T \{u\} \quad (2.6)$$

subjected to the constraints:

$$V^* - \sum_{e=1}^N V_e X_e = 0, X_e = \{0, 1\} \quad (2.7)$$

Where $\{F\}$ represents the vector of external forces vector and $\{u\}$ structure nodal displacement vector, V_e the volume of element, X_e the binary state of element e and X the vector containing all finite elements.

When an element is added or removed the total strain energy in the structure changes that variation is called sensitivity. This quantity allows the algorithm to make a choice since an element with high sensitivity means that the element must be kept or added to the structure. Sensitivity has a great influence on the objective function. Then a filtering scheme is required, the element's sensitivities are distributed to the nodes, averaging for each element the sensitivities of the nodes inside a sphere of preset radius (FR), weighted with the distance from each node to the element's center of gravity. The final new sensitivities are averaged with the ones from the previous iteration, to ensure a better convergence and stability of the algorithm. Following the sensitivities been computed and filtered the current volume is increased if smaller

than V^* , decreased if bigger, and maintained constant if equal with a prescribed Evolution Ratio (ER). The ER is a crucial parameter of the algorithm that assures convergence and stability. It represents the ratio of the volume towards the target. After the volume reaches V^* , if the convergence criterion is yet to be satisfied, the algorithm continues without further modifying the volume meaning that for each element added in an iteration another one is removed. The sensitivities are now sorted in ascendent order. The sensitivities α_- and α_+ are computed, in order to assign the binary state of the element: if $\alpha_e \leq \alpha_-$, $X_e = 0$; $\alpha_e \geq \alpha_+$, $X_e = 1$. The algorithm stops when the objective function finds a stationary value over several iterations [22].

The flow chart of the algorithm can be seen in Figure 2.7.

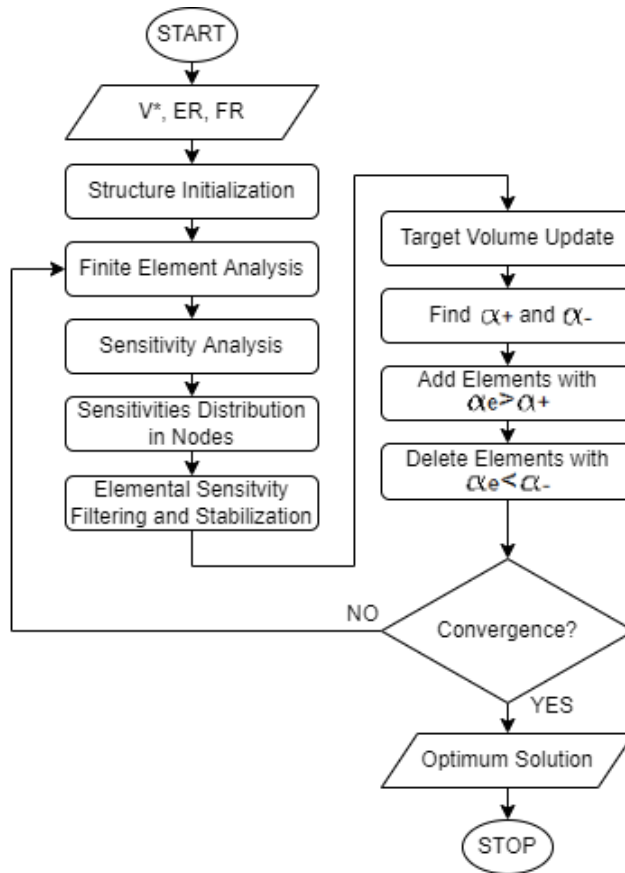


Figure 2.7: Flow chart for the BESO algorithm [22].

Chapter 3

RGV Characterization

This chapter will address the RGV Characterization. It is divided into two sections: the project restrictions and structural analysis for both the frame and the wheels' bar.

3.1 Project Restrictions

This project intends for a reduction of mass in the actual RGV, while maintaining structural integrity, this allows to reduce production costs, as well as, improve its energy efficiency. Since Körber Supply Chain already offers RGVs to their clients some requirements must be fulfilled when optimizing the RGV as can be seen in Table 3.1.

Table 3.1: Körber's RGV's specifications.

Technical Characteristics	Value
Maximum Payload / kg	1500
Maximum Handling Unit Dimensions / mm	1200x1200
Maximum Unit Load Dimensions / mm	1350x1350
Minimum Conveying Height / mm	750
Maximum Length / mm	1800
Maximum Width / mm	1654
Distance between Wheels / mm	1074
Maximum Straight Traveling Speed / ms^{-1}	3,5
Maximum Curve Travelling Speed / ms^{-1}	1
Maximum Acceleration / ms^{-2}	1

In order to optimize the RGV's mass, the steps followed were:

- Study of the RGV's components' weight;
- Identify the component with greater contribution to total weight;

- Analyse the stress and deformation in the component;
- Optimization of the component's mass, analysing the proposed component in terms of stress and deformation to obtain the benchmark values;
- Repeat the process for other components.

The current design of the RGV's frame is accomplished by welded metallic tubular connections and plates. The metallic material is S235JR its chemical composition and its mechanical properties are listed in Table 3.2 and Table 3.3, respectively.

Table 3.2: S235JR steel chemical composition [26]

Chemical Element	C	Mn	P	S	N	Cu
Chemical Composition / %	$\leq 0,17$	$\leq 1,40$	$\leq 0,035$	$\leq 0,035$	$\leq 0,12$	$\leq 0,55$

Table 3.3: S235JR steel mechanical properties [26].

Steel	Density / kg/m ³	Young's Modulus / GPa	Yield Strength / MPa	Ultimate Strength / MPa
S235JR	7850	210	235	360

The RGV is able to transport 1500 kg of goods which are carried in a conveyor with approximately 250 kg. Since the optimized RGV is intended to replace the one used nowadays, the width and length, as well as the distance between wheels (Table 3.4) are mandatory to stay the same, since it guarantees that the new version of RGV can perform the same routes the previous one did. It is also mandatory that the RGV's maximum deflection is 1 mm to allow the cargo to be transferred to the storage units.

Table 3.4: Körber's RGV's dimensions.

Geometrical Restrictions	RGV's Dimensions
Maximum Length / mm	1800
Maximum Width / mm	1654
Distance between the wheels and the rail / mm	1074

The first step was to choose which element had the most impact on the overall weight of the vehicle. In order to do it, the RGV's components were analyzed using Inventor, as can be seen in the Figure 3.1 through Figure 3.4, and listed in Table 3.5. The heaviest components were chosen to be redesigned.

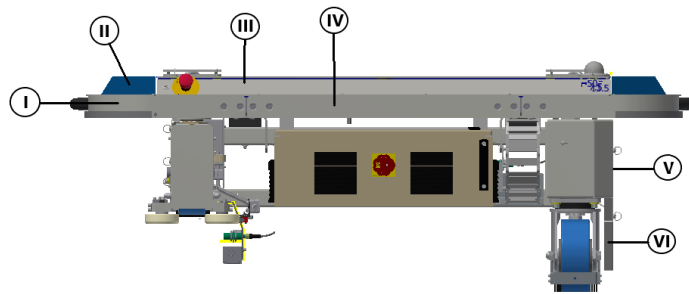


Figure 3.1: RGV's front view.

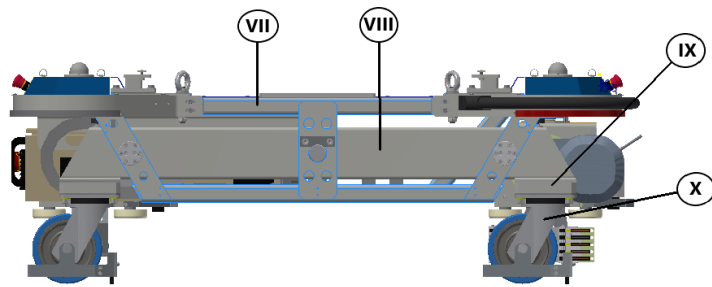


Figure 3.2: RGV's right view.

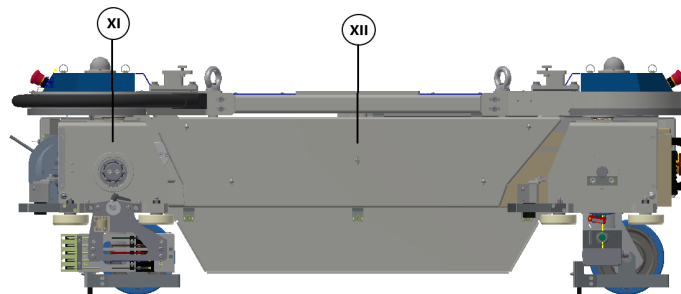


Figure 3.3: RGV's left view.

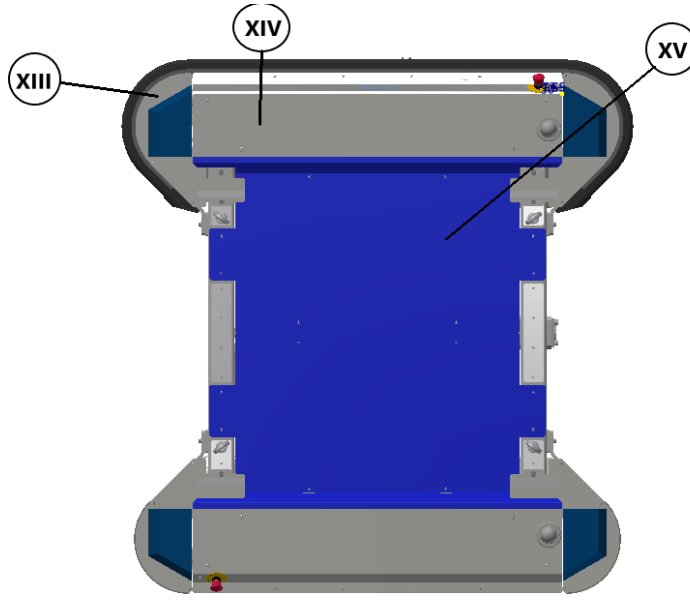


Figure 3.4: RGV's top view.

As it can be seen in Table 3.5, the RGV's frame (VII) is clearly the heaviest component contributing to 34% of the overall weight of the vehicle, followed by the wheels (X), the rail's wheels (XI) the wheels' bar (VIII). Components I through VI and XII through XV are mainly steel plates with no structural relevance. They are needed due to safety standards. The sum of all components' masses adds to 298,28 kg.

From the four heaviest components, the frame and wheels' bar will be studied, due to its contribution to the RGV total weight and its structural importance. The wheels will not be optimized since their designed is predominantly influenced by the cargo's weight, which a variable that cannot be changed.

3.1.1 Frame

The RGV's frame that can be seen in Figure 3.5 has also some requirements that need to be taken into account when optimizing its design.

Table 3.5: Components' mass of Körber's RGV.

Component Code	Quantity	Component's Mass / kg	Total Mass of the Components / kg	% of Components' Total Mass in the RGV's Weight
I	4	5,38	21,54	7
II	4	2,04	8,18	3
III	2	1,25	2,50	1
IV	2	4,18	8,36	3
V	1	6,16	6,16	2
VI	2	1,76	3,51	1
VII	1	102,68	102,68	34
VIII	1	29,77	29,77	10
IX	2	1,65	3,30	1
X	2	18,35	36,70	12
XI	2	18,38	36,77	12
XII	1	2,22	2,22	1
XIII	4	1,51	6,04	2
XIV	2	-	-	-
XV	2	15,28	30,55	10

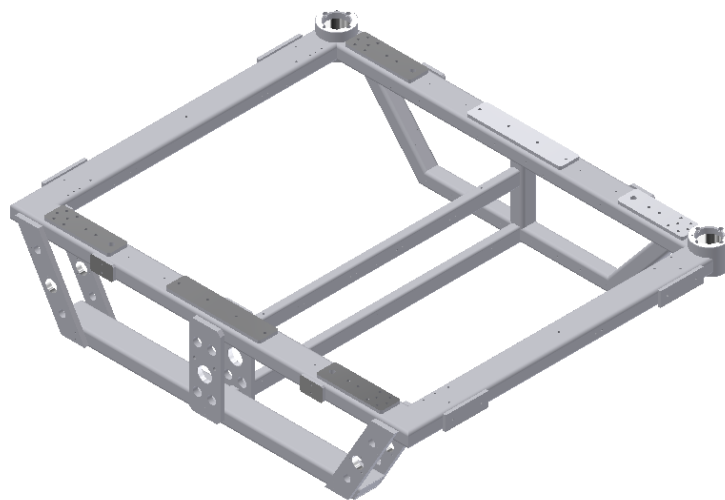


Figure 3.5: RGV's frame.

To do so, a characterization of these components is needed, in Figure 3.6

and in Table 3.6, it will be shown the components' cross-sections, as well as, geometrical and mechanical properties.

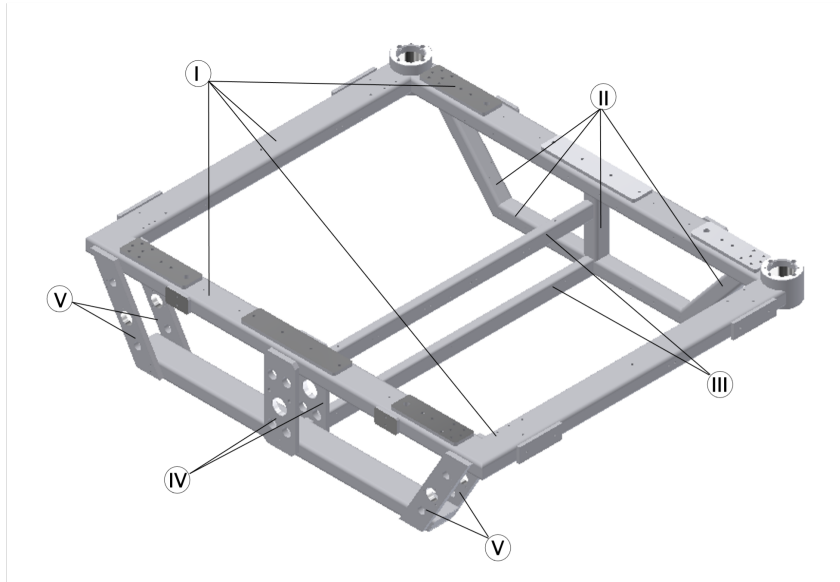


Figure 3.6: RGV's bars' layout.

Table 3.6: Mass of the frame's bars [27].

Component Code	Quantity	Cross-section / mm	Mass per meter / kgm^{-1}
I	4	90x50x5	14,7
II	4	50x50x5	6,85
III	2	50x30x3,2	3,41
IV	2	110x16	13,82
V	4	70x16	8,79

The tubular bars with horizontal plates from set I are where the conveyor sits, so their function is to withstand the cargo's weight. The bars from set III and the parallel bars from set I unite the frame and give structural integrity when the RGV is moving. The plates from set IV connect the frame to the wheels' bar and reduce the deflection on the upper bars, as well as, the vertical bar from set II. The plates from set V and the incline bars from set II help with the force distribution and also reduce deflection in the upper bars.

The conveyor that sits on the top of the frame, Figure 3.7, is attached in a 130x70 [mm] plate, requiring the bars that support it to be able to fit the plate, meaning that the bars should have a width of at least 70 mm.

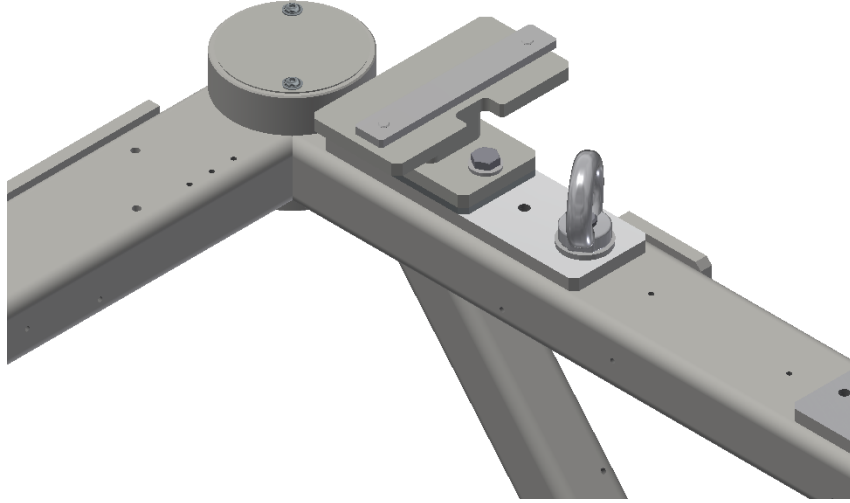


Figure 3.7: Conveyor's support.

The frame bars' layout should allow the inclusion of stoppers that prevent the bar where the wheels are attached to rotate in the plane parallel to the floor, as can be seen in Figure 3.8, as well as, allow the inclusion of the bar that connects the wheels.

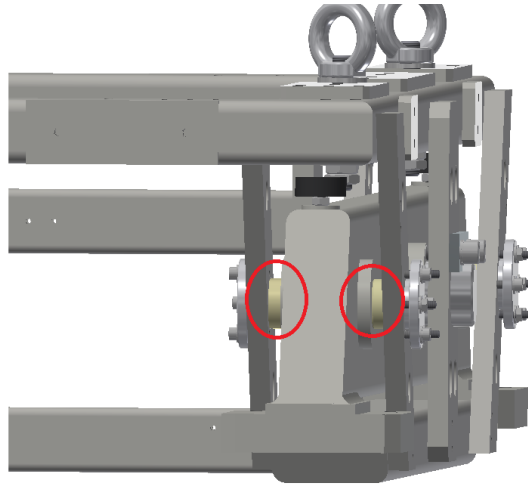


Figure 3.8: Stoppers.

3.1.2 Wheels' Bar

The wheels' bar (Figure 3.9) is a 1675mm long tubular bar, with a rectangular cross-section of 150x50x6 [mm] and its properties are shown in Table 3.7.



Figure 3.9: Wheels' bar.

Table 3.7: Bar's properties [27].

Cross-section / mm	Mass per meter / kgm^{-1}	Area / cm^2	I_y / cm^4
150x50x6	17,4	22,17	551,1

This bar is required to withstand what the frame unloads in it, keep the original wheelbase, and have the wheels attached to it. Also, the bar's width must be smaller than the upper frame bar's width, otherwise, it won't fit.

3.2 Structural Analysis

The cargo on the conveyor is transported in its middle, and the conveyor is supported by the RGV's frame. So, the force is applied on the RGV's center (black arrow) and supported by the two wheels and the bearing housing of the rail wheels (red arrows), as it can be seen in Figure 3.10.

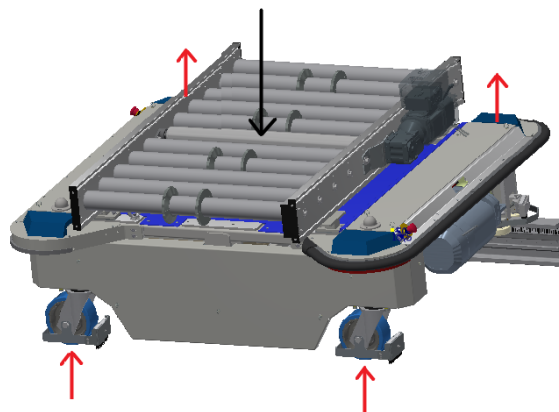


Figure 3.10: Forces and reactions applied in the RGV.

Those are the forces and reactions applied to the overall structure. Next in this chapter, it will be studied the forces in the components that will be optimized, as well as, their free body diagrams.

3.2.1 Frame

The forces from the cargo and the conveyor are transferred to the RGV's frame via the four conveyors' support and applied in the four plates of the frame, as can be seen in Figure 3.11. The force applied adds up to 17500 N (weight of the maximum cargo allowed plus the weight of the conveyor, assuming the acceleration of gravity to be 10 m/s^2).

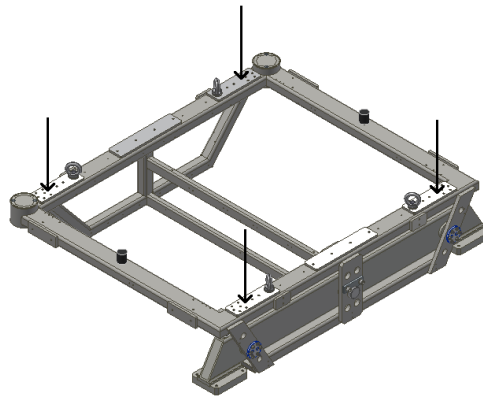


Figure 3.11: Forces applied to the RGV's frame.

The loads on the frame are supported by the bearing in the middle of the bar of the wheels and by the two bearing housings of the rail wheels, as can be seen in Figure 3.12.

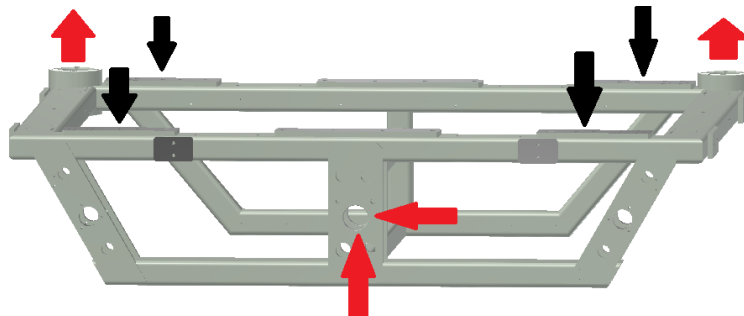


Figure 3.12: Forces and reactions applied to the RGV's frame.

As it can be noticed on the force diagram, it can be expected that the

loads will cause deflection in the bars where they are unloaded, which may be a structural layout improvement that could be done.

3.2.1.1 Initial Finite Element Analysis

In order to decide, how the RGV's frame could be optimized, an initial structural test was made via ANSYS Mechanical®. The FEM calculation showed the maximum and minimum stress and deflection locations, as well as, collected the values for those stresses and deflections. The information obtained by this analysis allowed to determine the areas that needed to be reinforced and gave a baseline for the overall stresses and deformations that the optimized frame should match.

The first step is the meshing of the frame where the structure was meshed via the function *Body Sizing* with 62712 elements of SOLID186 and SOLID 187 with 15 mm each. The plates were meshed via *Sweep Method*, an ANSYS' function suitable for sweepable bodies as is the case with the plates, and it was given a minimum number of 2 elements for the thickness as it is good practice for this type of bodies.

For the boundary conditions inputs, it was used the *Cylindrical Support* function in the bearing housings, represented by **B** in Figure 3.13. This function allows the user to choose if the axial, radial and tangential directions are free or fixed of moving. For the bearing housing the axial direction was fixed and the other two free, stopping the bearing housing to move up or down. The wheels' bar support (represented by **A** in Figure 3.13) was simulated using the *Cylindrical Support* function, allowing to stop displacement in the axial and radial direction and still allowing it to rotate. Finally, it was applied the 17500 N force in the middle of the structure being supported by the four plates via the *Remote Force* function.

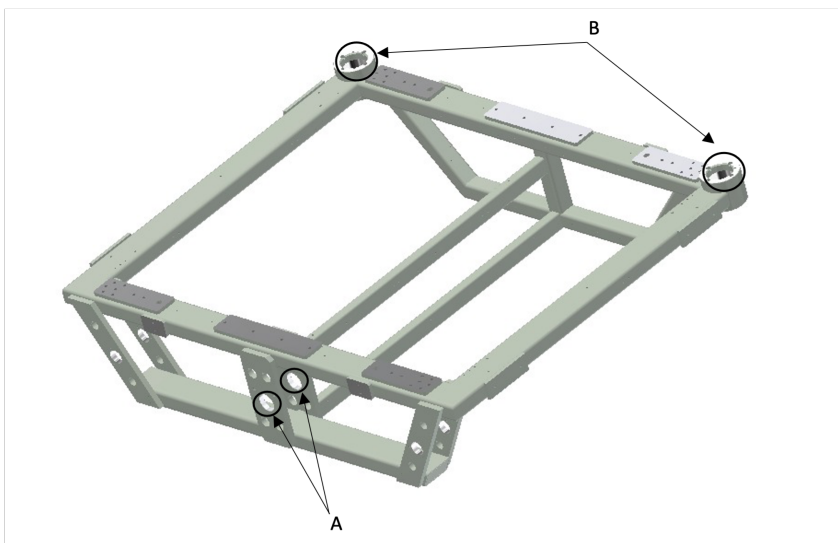


Figure 3.13: Support locations of the RGV's frame.

As for the results outputs, it was programmed to give *Total Deformation*, *Equivalent Stress*, as well as, the reaction in the *Cylindrical Support* via the function *Force Reaction*.

The distributions of the stresses and deformations can be seen the Figure 3.14 and Figure 3.15, respectively.

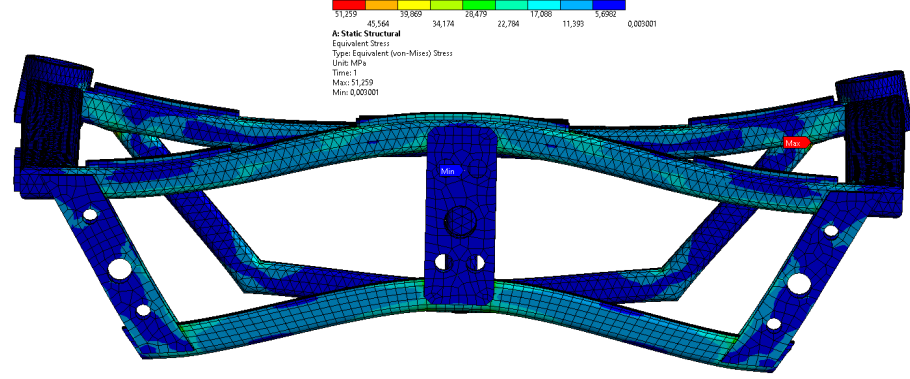


Figure 3.14: Initial stress analysis.

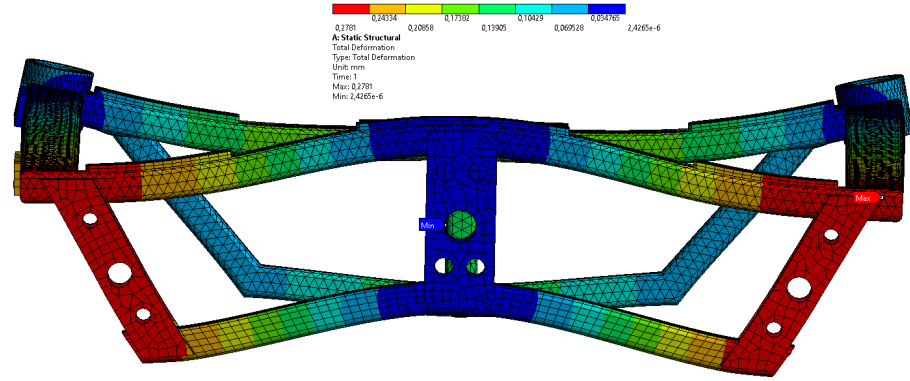


Figure 3.15: Initial deformation analysis.

The results for the maximum stress and deflection are 51,259 MPa and 0,2781 mm, respectively, and the reaction is 8622 N in the upwards direction of the z-axis, as expected.

With the result for maximum stress obtained, is now possible to calculate analytically the Safety Factor, as it is demonstrated in equation 3.1.

$$SF_{Frame} = \frac{\sigma_{Rd}}{\sigma_{Frame_{max}}} \Leftrightarrow SF_{Frame} = \frac{360}{51,259} = 7,0232 \quad (3.1)$$

Where σ_{Rd} is the ultimate strength of the material, $\sigma_{Frame_{max}}$ is the maximum equivalent stress on the frame, and SF_{Frame} is the safety factor.

This value leads to the conclusion that material can be removed, since, according to the European Materials Handling Federation in [28], a safety factor of 1,5 is acceptable.

3.2.2 Wheels' Bar

The wheels' bar is a 1675 mm tubular bar with a rectangular cross-section of 150x50x5 [mm]. It is connected to the frame, which means that some forces will be applied in the middle of the bar and will be supported by both wheels, Figure 3.16.

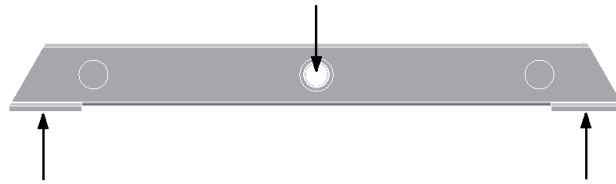


Figure 3.16: Forces and reactions applied to the RGV's wheel bar.

The arrangement of forces indicates that this bar will be subjected to a maximum bending moment and consequent maximum deflection in the middle of the bar, which is a key location since it could enable the RGV's cargo transportation.

3.2.2.1 Initial Analysis

To calculate the maximum stress in the bar, an analytical approach was taken. Firstly, a free body diagram was drawn (Figure 3.17) taking advantage of the symmetry of the bar.

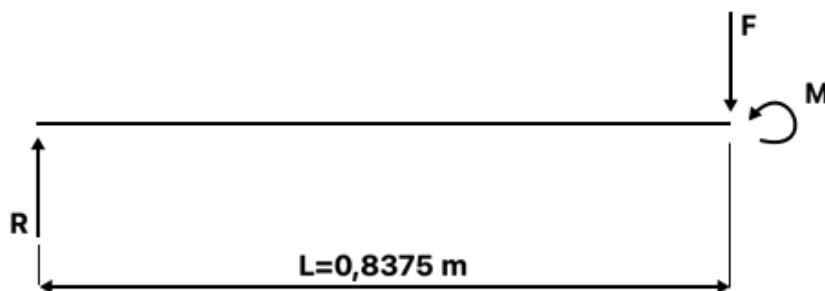


Figure 3.17: Free body diagram of the wheel's bar.

The force applied in the middle of the bar is the sum of the reaction on the frame plus half its weight, totaling 9135,4 N. Since the calculations are being made taking symmetry into account, the force applied in the free body diagram should be half of the total force unloaded in the bar: $F=4567,7$ N. The

maximum bending moment is where the force is applied and can be obtained by the equation 3.2:

$$M_{max} = F \times L \Leftrightarrow M_{max} = 4567,7 \times 0,8375 = 3825,4\text{Nm} \quad (3.2)$$

After obtaining the maximum bending moment is now possible to calculate the maximum stress, using the second moment of area values from Table 3.7:

$$\sigma_{Bar_{max}} = \frac{M_{max} \times y}{I} \Leftrightarrow \sigma_{Bar_{max}} = \frac{3825,4 \times 0,075}{5,511 \times 10^{-6}} = 52,06\text{MPa} \quad (3.3)$$

With the same procedure of equation 3.1 and the stress obtained in equation 3.3, the safety factor of the wheels' bar is:

$$SF_{Bar} = \frac{\sigma_{Rd}}{\sigma_{Bar_{max}}} \Leftrightarrow SF_{Bar} = \frac{360}{52,06} = 6,9151 \quad (3.4)$$

Similar to the frame's safety factor, the value for the bar's safety factor from the equation 3.4 is much higher than the minimum recommended by the European Materials Handling Federation in [28] which allows for material removal.

Chapter 4

Methodology for Optimization

This chapter will address the Methodology for Optimization. It is divided into two sections: the frame and wheels' bar optimization.

4.1 Frame

After the initial analysis, it was created in SpaceClaim® a prototype of the RGV's frame using only beams with the same cross-sections of the real RGV, Figure 4.1, that allowed for a reduction of computation cost modeling the frame. The approximation from shell elements to beam elements is valid since the distance in the axial direction is short enough so that the bending of the beams' cross-section won't make a difference. Also, as the beam model needs significantly less computational power and it's easier to change the cross-section of the beams, the small error from the approximation is worth the trade-off.

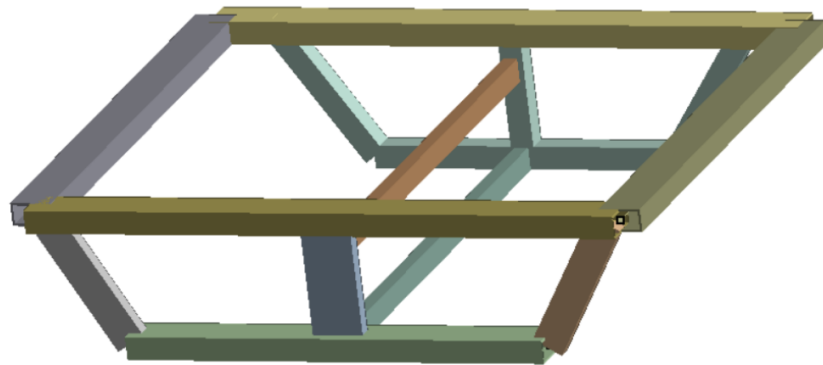


Figure 4.1: Beam FEM model.

This beam model needs to be validated through a convergence study where the number of elements are increased until the deformation in a specific point won't change. In Table 4.1, can be seen the iterations done in the convergence study of the beam model.

The prototype was meshed via the function *Body Sizing* with an element size of 10mm, equaling a mesh of 1061 BEAM188 elements. A decrease in

element size was redundant since it did not get better results only longer computational time.

For the boundary conditions inputs, it was used *Fixed Support* for all the supports, instead of the *Cylindrical Support* used on the initial simulation. Then, a *Remote Force* function was applied with the same inputs of the initial simulation.

As for the results outputs, it was programmed to give *Total Deformation*, *Axial Force* and with the help of the *Beam Tool* it also gave the *Maximum Combined Stress*.

Table 4.1: Beam model's convergence study.

Size of the Elements of the Mesh / mm	Deformation on a Single Point Beam Model / mm	Difference from the Previous Iteration / %
15	0,27655	-
10	0,27655	0
5	0,27655	0

As the values for deformation remained constant with finer mesh, convergence in the beam model was reached (Figure 4.2).

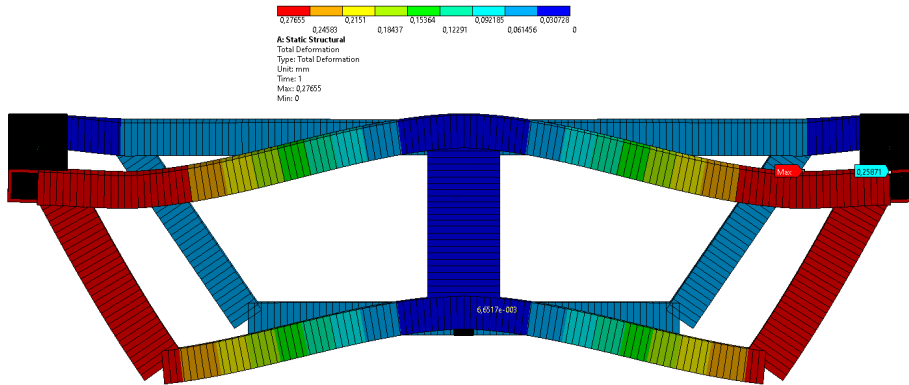


Figure 4.2: Beam model simulation.

4.1.1 Topology Optimization

With the initial parameters determined, a topology optimization through the optimality criteria method using ANSYS Mechanical® was made. The idea is to have a solid block of metal that resembles the frame structure, Figure 4.3, and use the topology optimization tool to remove as much material as possible, but at the same time ensure that the structural integrity is viable. Then, transform the topology optimized structure into a beam-like structure using tubular connections. After the transformation, there is the need to ensure that the new structure supports the initial loads.

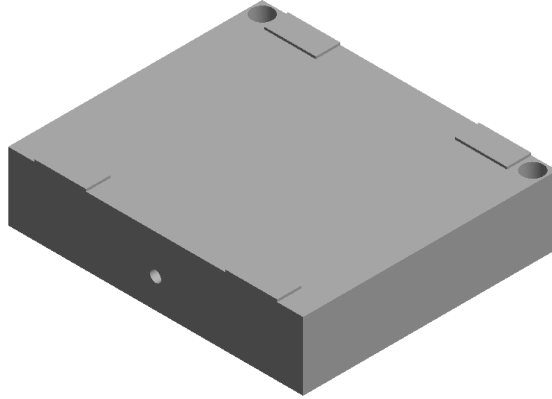


Figure 4.3: Solid block used for topology optimization.

The simulation parameters of the solid block topology optimization were very similar to the ones used in the initial simulation of the frame to make the optimization as viable and correct as possible.

The solid block was meshed via the function *Body Sizing* with a SOLID187 element of 10mm (totaling 138079 elements), which generates a finer mesh than the one used in the initial simulation of the frame, allowing the topology optimization to have great precision when removing material. Since the plates only have function receiving the load and their optimization was not the objective, they were also meshed with the *Body Sizing* used on the block, instead of the *Sweep Method* used on the frame.

For the boundary conditions inputs, the same boundary conditions were programmed: *Cylindrical Support* function with fixed axial displacement in the bearing housings, *Cylindrical Support* function with fixed axial and radial displacement and a *Remote Force* of 17500 N applied in the middle of the block unloaded on the four plates.

As for the results outputs, it was programmed to exclude from material removal the front edge of the wheels' bar support through the *Exclusion Region* function, the mass removal was maximized by imposing a mass retain percentage of 1% in the *Response Constraint* function. Those inputs allowed to study the disposition of material needed to support the force, if there was no frame-like structure.

The result from the topology optimization for a retain threshold of 10% can be seen in Figure 4.4.



Figure 4.4: Topology optimization result.

As it can be seen, this structure doesn't resemble a frame, so another iteration was made to achieve a frame-like structure. In the new iteration, the only parameter that changed was the *Exclusion Region* where were added the edges of the solid block that are on a similar location relative to the bars on the frame. The results from this iteration can be seen in Figure 4.5.

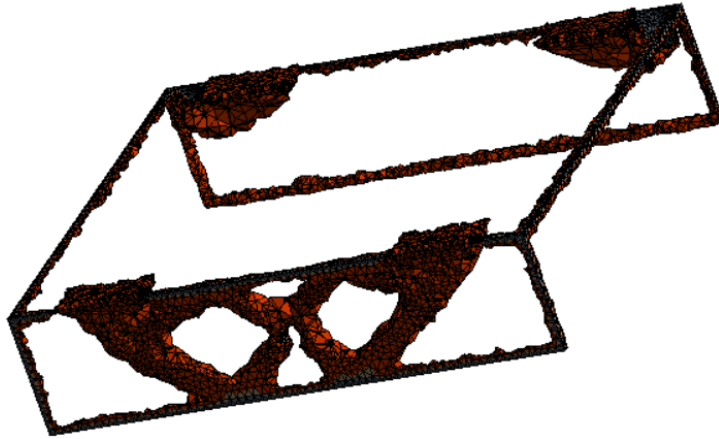


Figure 4.5: Topology optimization iteration result.

Now that both results are shown, some analysis of the results can be made.

Since there isn't any material connecting the points of loading in the first optimization, and in the second the only existing ones were restrictions imposed, it's evident that their contribution in terms of deflection reduction is of diminished importance. This point is also supported by the concentration of material where the force is applied, meaning that the only bars with significant stress are the two bars where the conveyor sits.

4.1.2 New Frame Layout

After the topology optimization study is done, it's now possible to develop a new frame. This new structure is intended to have at least the same stiffness as the old one, to be lighter and less expensive.

Analyzing the layouts obtained from topology optimization, there are mainly two points that differ from the original structure. There are material connecting from the points where the force is applied directly to the lower part of the structure. It is also notorious that the bar layout that seems to better support the loads, is an upside-down triangle layout, where the loads are applied in its base's vertices. This disposition differs from the original layout, as there is no bar connecting the unloading plates directly to the lower part of the frame and the lateral structure of the frame has a trapezoidal shape.

With the layout analysis done, the first layout iteration was done, as can be seen in Figure 4.6, the only difference from the original layout is that the bars from the group set V (Figure 3.6) are directly supporting the unloading plates, as in the topology optimization in Figure 4.5.

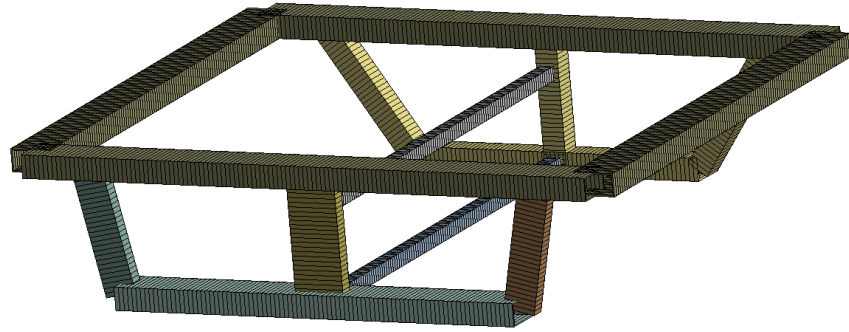


Figure 4.6: First layout iteration.

As Table 4.2, this layout caused a bigger deflection than the original, a new iteration was taken: changing the bars where the maximum stress occurred to an upside-down triangle (Figure 4.7), reminiscent of the topology optimization result in Figure 4.4.

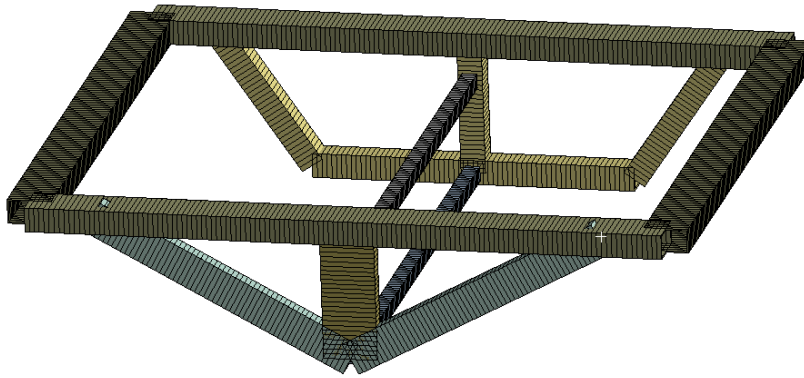


Figure 4.7: Second layout iteration.

The change improved the deflection so greatly that caused the other side to have maximum deflection, and because of that, the same approach was taken to the side of maximum deflection. The bar layout of the new structure can be seen in Figure 4.8.

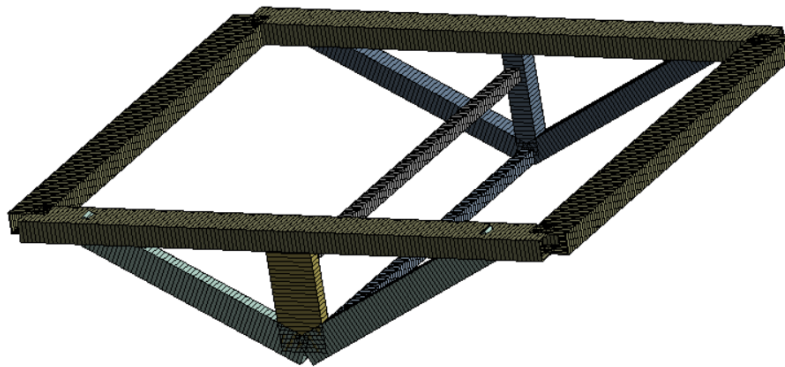


Figure 4.8: New structure bar layout.

The results of the iterative process will be shown in the table below, as well as, the finite element mesh characteristics, since the boundary conditions remain the same.

Table 4.2: Layout iterative process.

Iteration	Element Type	Number of Elements	Total Mass / kg	Maximum Deflection / mm
1	BEAM188	1053	71,07	0,3833
2	BEAM188	1023	75,98	0,04898
3	BEAM188	1009	74,90	0,04888

The third iteration has the least deflection and it is lighter than the original frame, so this is the layout that will be analyzed and worked on.

To be able to compare the new layout with the old one, the cross-sections of the bars are exactly the same, the differences are that the two horizontal lower bars are non-existing in the new model, the incline bars are extended to the bottom of the vertical bars and begin where the force is unloaded. The analysis of the new structure (Figure 4.9, Figure 4.10, Figure 4.11) was made with 1009 BEAM188, 10 mm each element, and the same input parameters for boundary conditions as the beam model simulation.

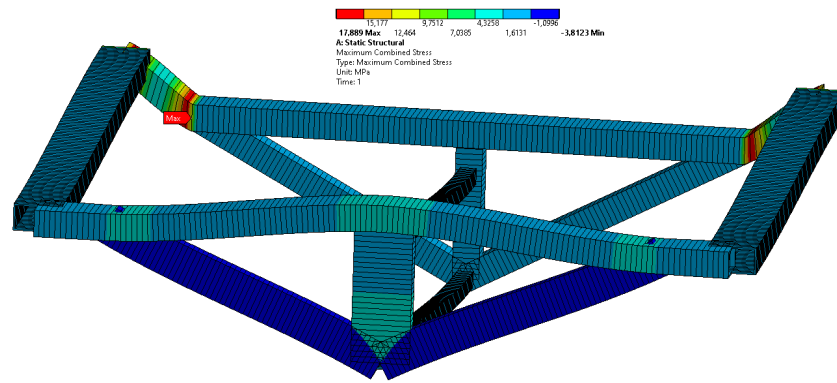


Figure 4.9: New structure stress analysis.

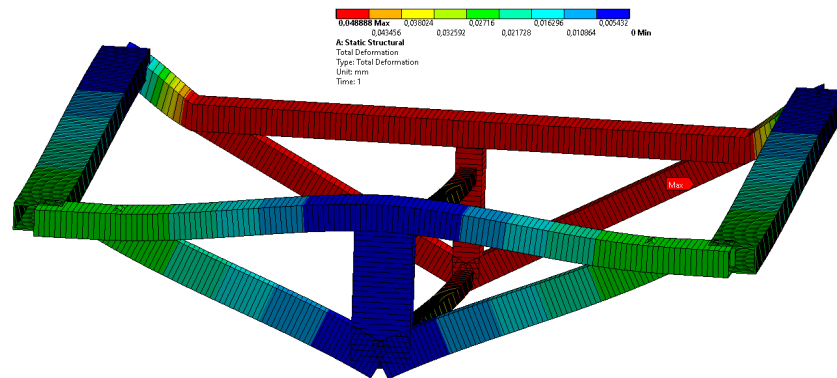


Figure 4.10: New structure deformation analysis.

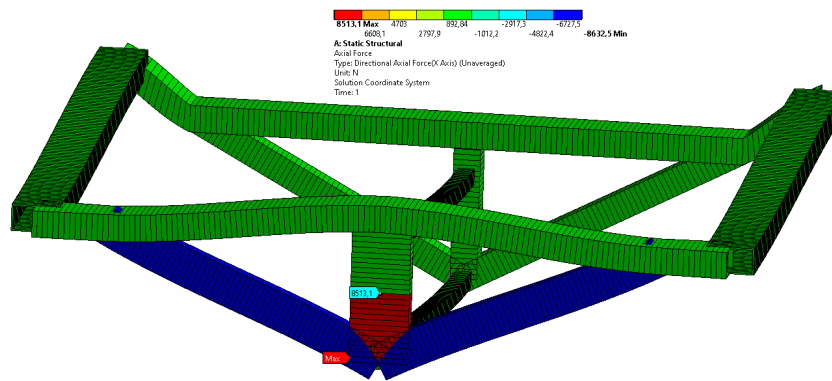


Figure 4.11: New structure axial force analysis.

The results for the maximum stress and deformation are 21,16 MPa and 0,048931 mm, respectively, and the reaction is 8521 N in the upwards direction. The results comparison between the initial simulation of the frame and the new frame layout will be in Table 4.3.

Table 4.3: Structural layout comparison.

Mechanical Results	Initial Layout	New Layout	Relative Variation
Maximum Stress / MPa	51,259	17,889	-65%
Maximum Deformation / mm	0,2781	0,04889	-82%
Reaction / N	8622	8513,1	-1%
Mass / kg	102,68	74,897	-27%

The new layout optimizes the frame in terms of maximum stress, maximum deflection, and mass. It also brings a big improvement the difference in deflection from the plates in different bars is 0,037635 mm which is unnoticeable. Since the deflection is yet very far from the maximum deflection allowed, the mass of the frame can still be reduced. To do so, the cross-sections from the bars were changed iteratively, based on their mass per meter as well as the other bars, in order to legitimize connections between bars. In the image below, it's attributed a number to a set of bars to ease the understanding of the iterations. It will also be shown below a table with all the iterations made.

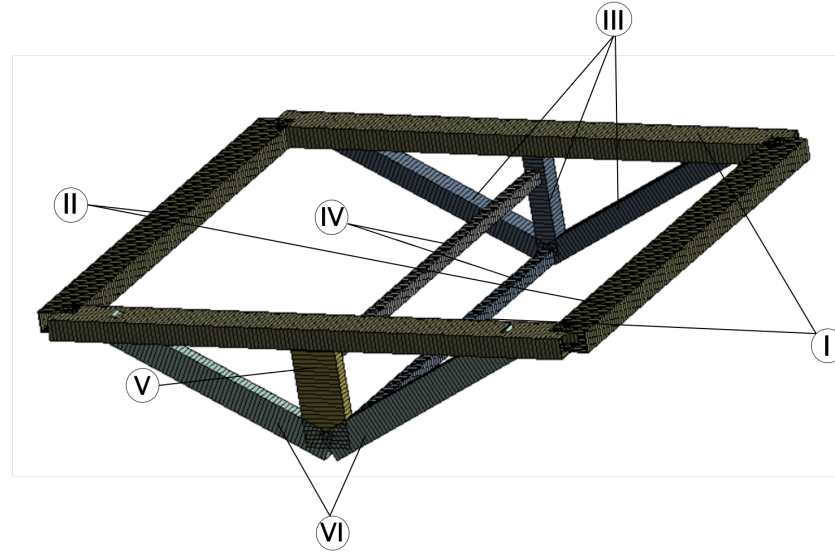


Figure 4.12: Bars group sets.

Table 4.4: Structural layout comparison of the iterations.

Iterations	I	II	III	IV	V	VI	Mass / kg	Max. Deformation / mm
Base	90x50x3	90x50x3	50x50x5	30x30x3	110x32	70x32	74,897	0,04889
1	90x50x3	50x25x3	30x30x3	30x30x3	110x32	70x32	60,271	0,048956
2	80x40x3	40x40x2	25x25x3	25x25x3	110x32	70x32	52,95	0,071053
3	80x40x3	40x40x2	25x25x3	25x25x3	110x24	70x24	45,89	0,07113
4	80x40x3	40x40x2	25x25x2	25x25x2	80x16	50x16	33,808	0,074041
5	90x50x3	50x25x3	25x25x2	25x25x2	80x16	50x16	39,249	0,069054

Although it may seem counter-intuitive, in the last iteration it was tested a bigger cross-section in the bar's set 1 to allow for flexibility when choosing the cross-section of the wheels' bar.

The frame from iteration 4 is shown in Figure 4.13, as well as, its stress, deformation and axial force analysis, in Figure 4.14, Figure 4.15 and Figure 4.16.

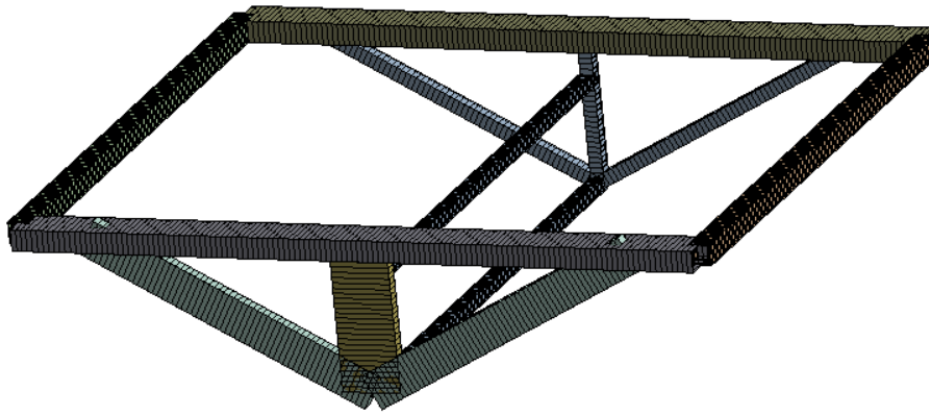


Figure 4.13: Frame model from iteration 4.

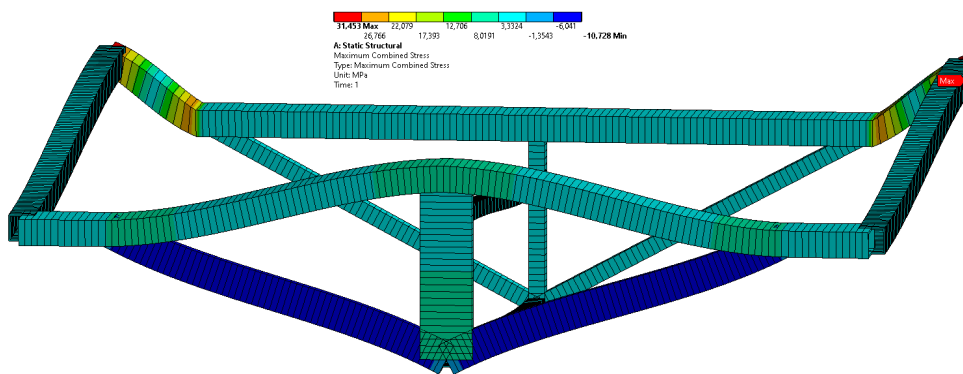


Figure 4.14: Stress analysis of the frame model from iteration 4.

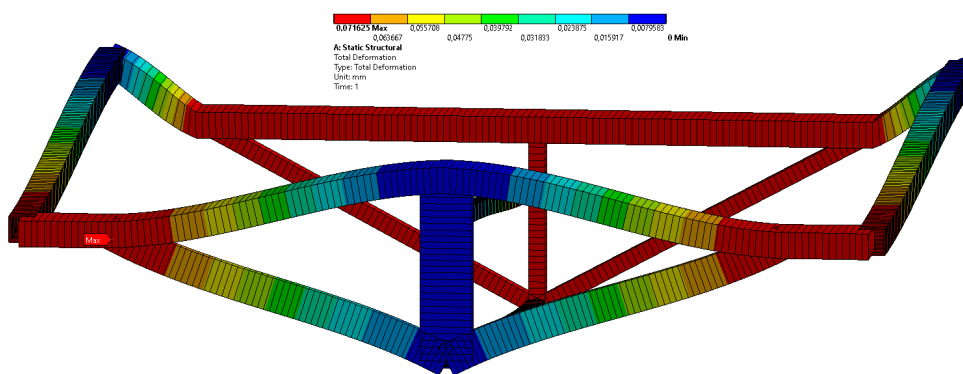


Figure 4.15: Deformation analysis of the frame model from iteration 4.

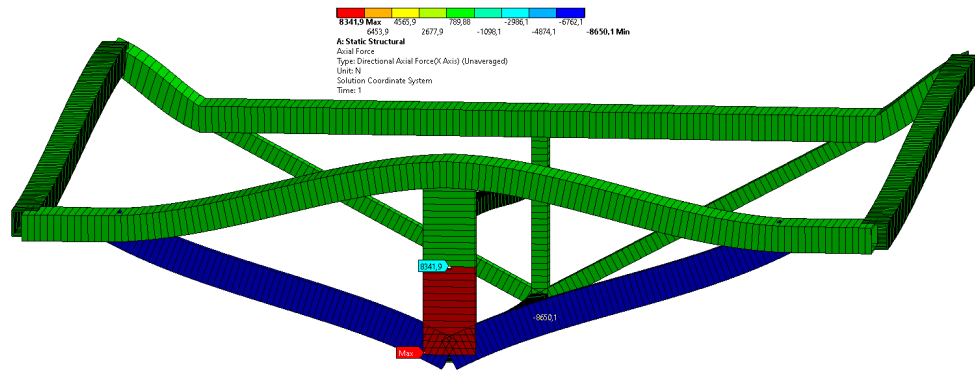


Figure 4.16: Axial force analysis of the frame model from iteration 4.

After the stress, deformation, and axial force analysis are done, a comparison of the values with the ones of the initial frame was done in Table 4.5.

Table 4.5: Structural comparison of the frame from iteration 4.

Mechanical Results	Initial Frame	Iteration 4 Frame	Relative Variation
Maximum Stress / MPa	51,259	31,453	-39%
Maximum Deformation / mm	0,2781	0,07163	-74%
Reaction / N	8622	8574,7	-1%
Mass / kg	102,68	34,874	-66%

This solution allows for a reduction of 66% of the frame's mass, while also reducing maximum stress and deformation for 39% and 74%, respectively. In addition, the difference in deformation from one loading bar to the other is 0,03676 mm ensuring that the cargo won't get unbalanced.

4.2 Wheels' Bar

The approach for the bar optimization will be based on maximum allowed stress and maximum deflection and as a deciding factor the weight of the cross-section.

As a starting point, the allowed stress and the maximum deflection need to be determined. As it was shown in Chapter 3, the σ_{Rd} is 360 MPa and the recommended safety factor is 1,5. So the allowed stress is given by:

$$\sigma_{Sd} = \frac{\sigma_{Rd}}{SF} \Leftrightarrow \sigma_{Sd} = \frac{360}{1,5} = 240\text{MPa} \quad (4.1)$$

Since the wheels' bar holds the frame, its maximum deflection is determined by subtracting the frame's maximum deformation to the RGV maximum deformation:

$$\delta_{Bar_{max}} = \delta_{RGV_{max}} - \delta_{Frame_{max}} \Leftrightarrow \delta_{Bar_{max}} = 1 - 0,07113 = 0,92837\text{mm} \quad (4.2)$$

Now, that the maximum design values are determined, the structural optimization can begin.

Remembering the Chapter 3, the free body diagram from Figure 3.17, it's still valid. Although the force needs to be recalculated, the bar's length remains the same as it is required. The force applied to the bar is the reaction on the wheels' bar support (8574 N) plus half the frame's weight ($\frac{34}{2} \times g$), totaling 8744,7 N. Since the study is made in a half bar due to symmetry, the force in the free body diagram is 4372,35 N. The maximum bending moment, as previously seen, is given by:

$$M_{max} = F \times L \Leftrightarrow M_{max} = 4372,35 \times 0,8375 = 3661,8\text{Nm} \quad (4.3)$$

It's now possible to calculate the section modulus:

$$\sigma_{Sd} \geq \frac{M_{max}}{W} \Leftrightarrow W \geq \frac{M_{max}}{\sigma_{Sd}} = \frac{3661,8}{240 \times 10^6} \Leftrightarrow W \geq 15,258\text{cm}^3 \quad (4.4)$$

With the minimum value of section modulus defined, the only criteria missing is the maximum deformation. For a more flexible choice, it will be imposed multiple values of maximum deformation in order to get multiple values of the second moment of area, as can be seen in Table 4.6. The values on the table are obtained by solving the equation 4.5 [29].

$$\delta_{max} = \frac{F \times L^3}{3 \times E \times I} \Leftrightarrow I \geq \frac{F \times L^3}{3 \times E \times \delta_{max}} \quad (4.5)$$

This table allows having a reference of in which interval of deflection the cross-section is, having the value of the previous bar's deflection, as well as, the limit value for deflection: 0,74 and 0,92, respectively.

As the maximum deflection criterion is more restricted than the allowable stress, the collection of cross-sections began by finding the ones with $I \geq 443,14 \text{ cm}^4$ and mass per meter lower than the original cross-section ($17,4 \text{ kgm}^{-1}$). The potential cross-sections are on Table 4.7.

Since the values of section modulus are all above the minimum value, to proceed to the choice of the cross-section this variable will not be taken into account. The graphic in Figure 4.17 was constructed to study the best cross-section for the wheels' bar: the selected cross-sections from the table were inserted in a graphic where the Y-axis represents the deflection of the beam for the structural case of the bar and in the X-axis are the respective masses per meter, since the objective is to reduce mass in the RGV. It was also drawn an orange line for the 0,74 mm, the value of the deflection in the original bar. As it's possible to strengthen the structural integrity of the RGV, or at least

Table 4.6: Minimum second moment of area for an imposed deflection.

δ_{max} / mm	I / cm ⁴
0,2	2038,44
0,3	1358,96
0,4	1019,22
0,5	815,38
0,6	679,48
0,7	582,41
0,74	550,93
0,8	509,61
0,9	452,99
0,92	443,14

Table 4.7: Cross-section's collection [27] [30].

Cross-section / mm	Mass per meter / kgm ⁻¹	I / cm ⁴	W / cm ³	Deformation Interval / mm
150x50x6	17,4	551,1	73,48	0,7-0,74
150x50x5	14,7	475,8	63,44	0,8-0,9
180x60x3	11,021	545,39	60,60	0,74-0,8
180x60x4	14,57	711,01	79,01	0,5-0,6

maintain it, it was imposed the criterion that only cross-section below the line or very close to it will be chosen.



Figure 4.17: Cross-section study for iteration 4.

As it can be seen in the figure above, there are only two cross-sections that respect the imposed criterion are: 180x60x3 and 180x60x4. The first cross-section will be chosen for the frame from iteration 4 because the main objective of this dissertation is the reduction of the mass of the RGV.

As it was mention earlier, the same analysis was done for iteration 5, to allow more flexibility in the cross-section chosen for the bar, since this layout allows for wider cross-sections. For that reason, the cross-sections added to the cross-section's collection for iteration 5 can be seen in Table 4.8.

Table 4.8: Cross-sections added to the cross-section's collection [27] [30].

Cross-section / mm	Mass per meter / kgm^{-1}	I / cm^4	W / cm^3	Deformation Interval / mm
150x50x6	17,4	551,1	73,48	0,7-0,74
180x80x3	11,9	631,4	70,15	0,6-0,7
160x80x3	10,9	472,1	59,01	0,8-0,9
140x80x5	16,3	534	76,28	0,74-0,8

As the criteria remained the same, less mass as possible and to strengthen the structural integrity if possible, the cross-section study for iteration 5 can be seen in Figure 4.18.

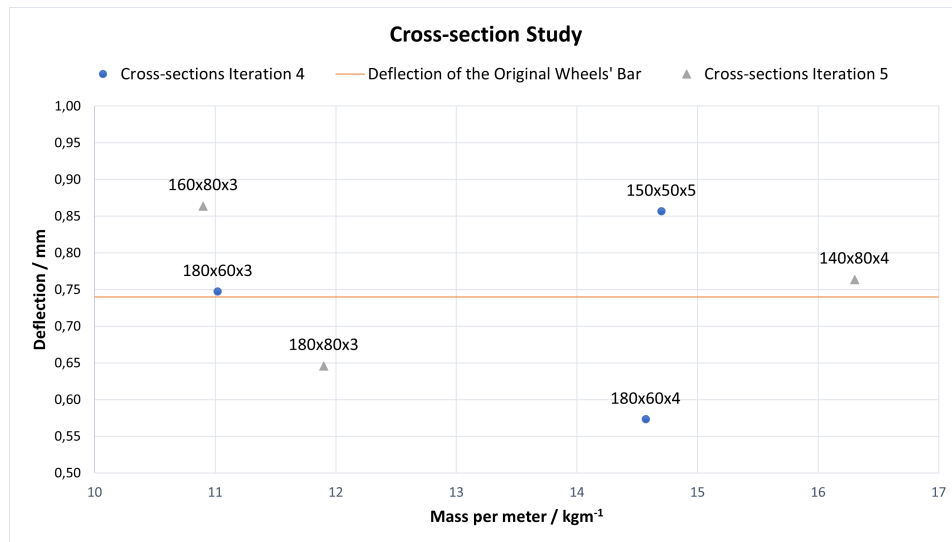


Figure 4.18: Cross-section study for iteration 5.

As it can be seen in Figure 4.18, there are only one cross-section lighter than the one chosen for the frame from iteration 4. Since the study was made to find a lighter cross-section and knowing that increasing deflection from the original one won't put structural integrity at risk, the mass study will be made with cross-section 160x80x3. The mass study will compare the mass of the

frame from iteration 4 plus the mass of the wheels' bar with the cross-section 180x60x3 with the mass of the frame from iteration 5 plus the mass of the wheels' bar with the cross-section 160x80x3.

Table 4.9: Mass study of the lighter frame and wheels' bar combination.

Combination	Mass of the Frame / kg	Mass of the Wheels' Bar / kg	Total Mass / kg
Iteration 4	33,81	18,46	52,27
Iteration 5	39,25	18,26	57,51

As Table 4.9 shows, the lighter combination is the frame from iteration 4 with the wheels' bar whose cross-section in 180x60x3.

Chapter 5

Final Solution

This chapter addresses the final solution regarding the RGV structure. The design solution is shown, displaying the layout of the structure. Also, it covers the examination of the solution respecting the restrictions proposed in Chapter 3. At last, a comparison between solutions is made and conclusions about the properties of both solutions are drawn.

5.1 Design Solutions

Similar to the older version, the new design of the RGV's frame (Figure 5.1) is also accomplished by welded metallic tubular bars and plates of the S235JR steel.

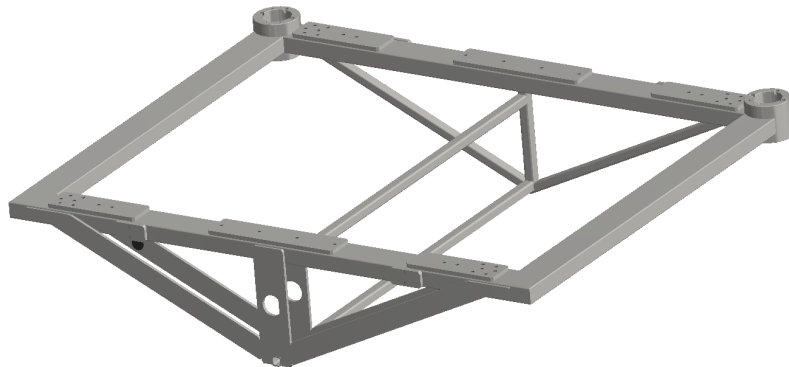


Figure 5.1: Frame final solution.

All tubular bars have commercial cross-sections, so they are easily bought, and are cheaper relative to a non-generic cross-section. The layout of the cross-sections on the frame and its properties can be found in Figure 5.2 and in Table 5.1.

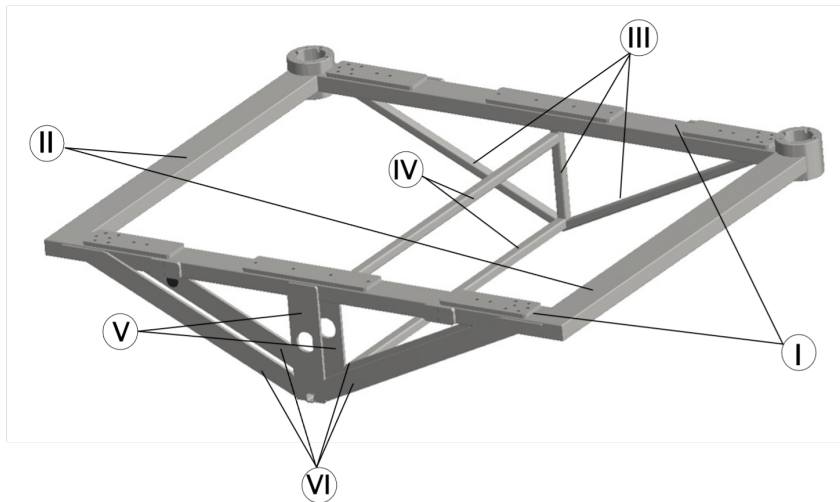


Figure 5.2: Bars' group sets in the new frame.

Table 5.1: Bars cross-section's layout and properties [27].

Bars' set	Cross-section / mm	Quantity	Mass per meter / kgm^{-1}	I / cm^4	W / cm^3
1	80x40x3	2	5,29	54,23	13,56
2	40x40x2	2	2,35	9,775	3,577
3	25x25x2	3	1,41	1,561	1,248
4	25x25x2	2	1,41	1,561	1,248
5	80x8	2	5,02	34,13	8,53
6	50x8	4	3,14	8,33	3,33

As the layout changed, so did the connections, Figure 5.3. These are the only two different connections in the final solution of the frame, the three-bar connection from the rail side and the plates' connection, respectively. Both the new connections are welded.

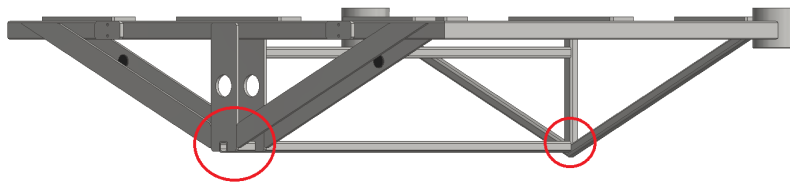


Figure 5.3: Location of the different welding connections.

In the three-bar connection, Figure 5.4, the two incline bars are welded, followed by the welding of the vertical bar. In the plates' connection, Figure 5.5, the incline plates are welded to the middle one and the middle plate is welded to the bar.

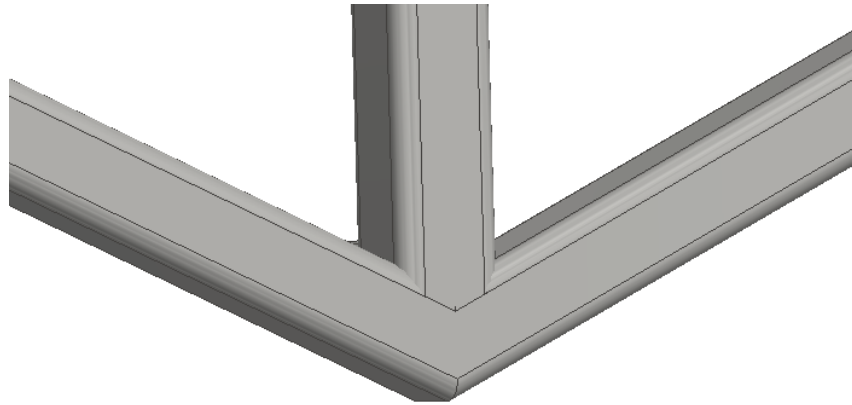


Figure 5.4: Detail of the three-bar connection.

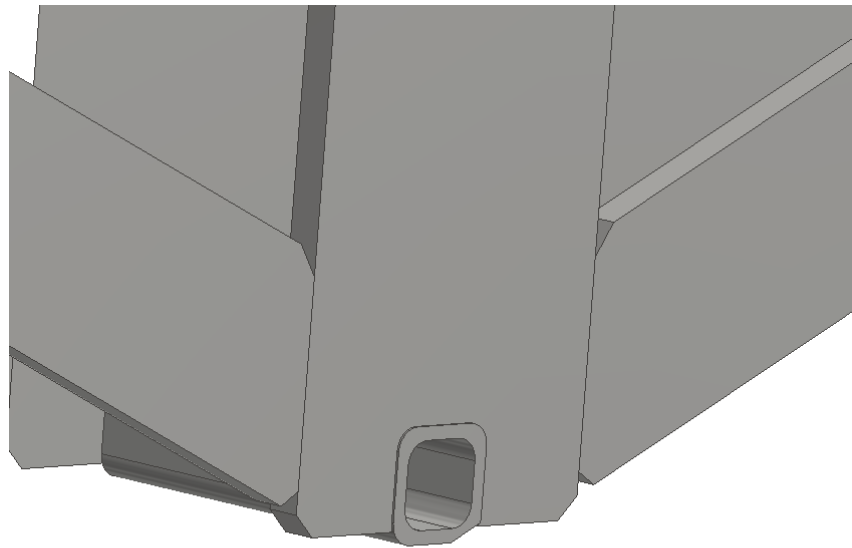


Figure 5.5: Detail of the plates and bar connection.

The wheels' bar final solution, Figure 5.6, was also designed from a commercial cross-section and is made of the S235JR steel, the cross-section's properties are in Table 5.2.

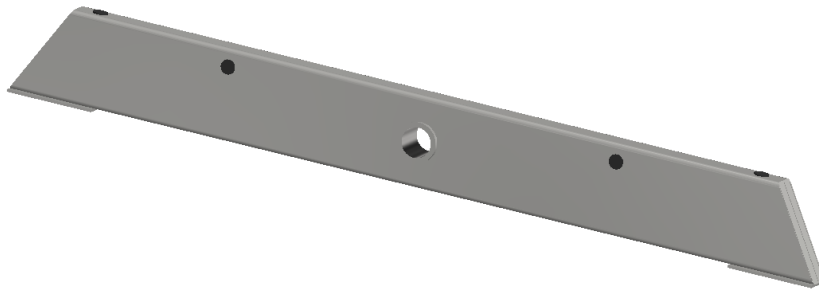


Figure 5.6: Wheels' bar final solution.

Table 5.2: Bar cross-section's properties [30].

Cross-section / mm	Mass per meter / kgm^{-1}	I / cm^4	W / cm^3
180x60x3	11,021	545,39	60,60

Since the frame and the wheels' bar are connected and have relative motion between them, stoppers were put both in the frame and the bar, to prevent any damage to the components, as can be seen in detail in Figure 5.7. The side stoppers also prevent the wheels' bar to twist and consequently, stop the bar to move away the intended trajectory.

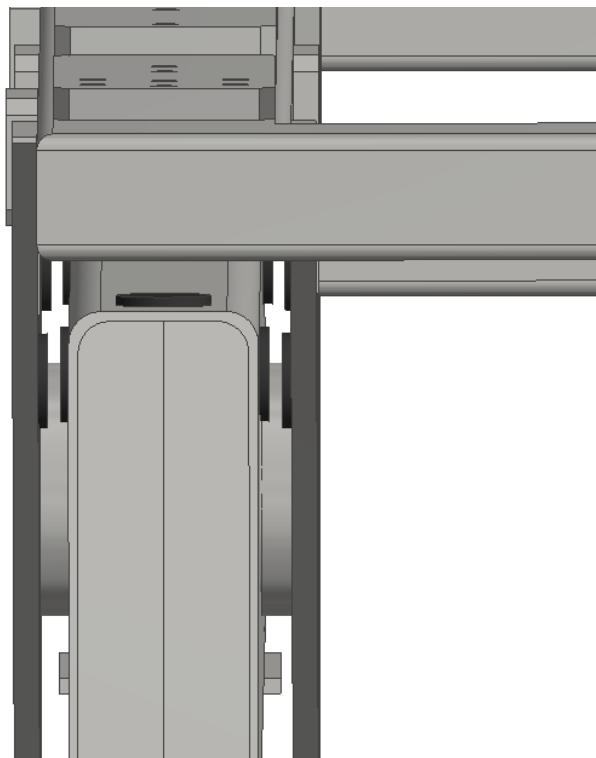


Figure 5.7: Detail of the stoppers in the frame and in the wheels' bar.

Finally, it can be seen the assembly of the new frame solution and the new wheels' bar solution:

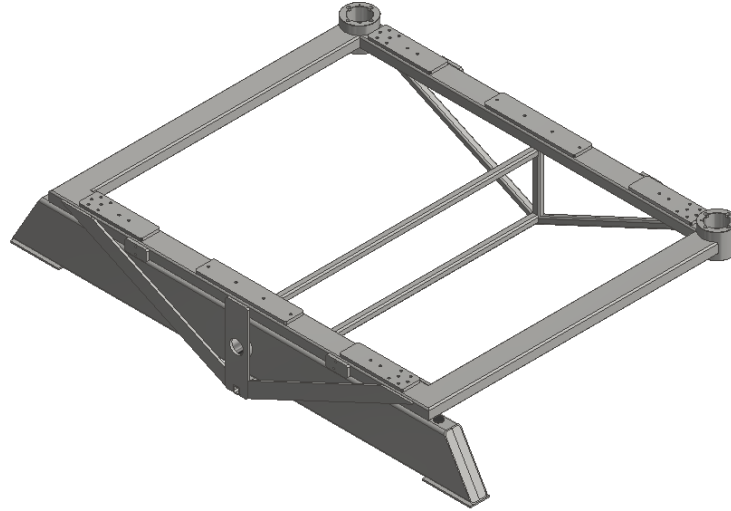


Figure 5.8: Assembly of the new solutions.

5.2 Requirements Fulfillment

As this project intends to be implemented in an existing product, the restrictions from Chapter 3 must be fulfilled.

Starting with the overall dimensions, the maximum length and width are respected if the new solution has at least the same length and width as the older version. Those dimensions are compared in Table 5.3.

Table 5.3: Frames' length and width comparison.

Dimensions	Original Frame	New Frame
Maximum Length / mm	1390	1380
Maximum Width / mm	1140	1140

As the axis lines on the rectangular upper part of the frame are the same, the distance between the rail and the wheels, as well as, the distance between the rail's wheels remained the same.

The minimum conveying height was ensured by lowering the frame's connection to the wheels' bar 10 mm, the difference in heights between the frames.

The new frame was developed so that the inclusion of the mandatory stoppers for the wheels' bar and the plates for the conveyor's support was possible, these details can be seen in Figure 5.9 and Figure 5.10.

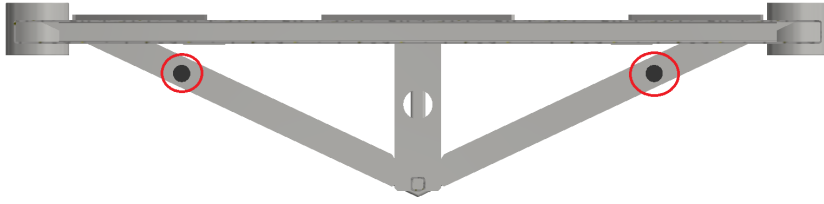


Figure 5.9: Cut-view of the stoppers in the new frame.

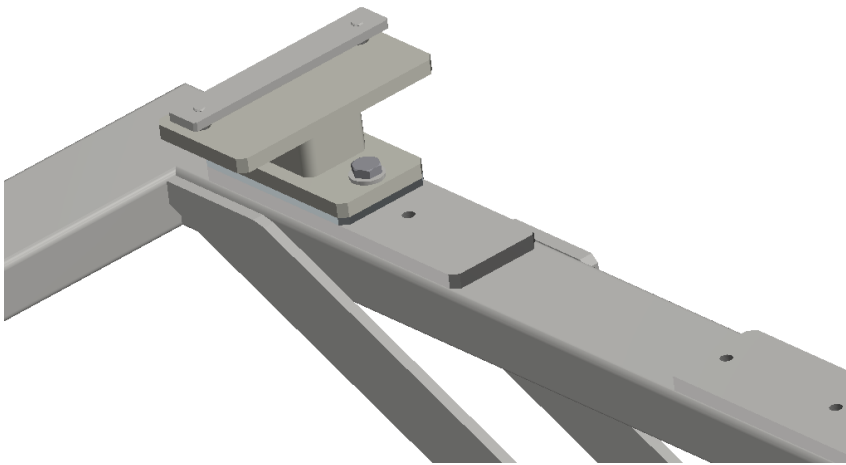


Figure 5.10: Conveyor's support in the new frame.

The wheels' bar kept the original wheelbase and its cross-section was chosen so that it could fit the frame.

5.3 Validation

Now, that the final solution is presented and the restrictions are proven to be fulfilled, it's mandatory to validate them. The validation requirements are a safety factor equal or bigger than 1,5 for both the frame and the wheels' bar and the total deflection needs to be smaller than 1 mm.

Both the safety factor calculations are presented in equation 5.1 and equation 5.2:

$$SF_{NewFrame} = \frac{\sigma_{Rd}}{\sigma_{NewFrame_{max}}} \Leftrightarrow SF_{NewFrame} = \frac{360}{17,889} = 20,124 \quad (5.1)$$

$$SF_{NewBar} = \frac{\sigma_{Rd}}{\sigma_{NewBar_{max}}} \Leftrightarrow SF_{NewBar} = \frac{360}{60,43} = 5,9573 \quad (5.2)$$

In both cases the solutions are validated in terms of the safety factor.

The maximum total deflection given by the sum of both components maximum deflection:

$$\delta_{Total_{max}} = \delta_{Frame_{max}} + \delta_{Bar_{max}} \quad (5.3)$$

From the simulations results in Chapter 4, the frame's maximum deflection is 0,074041 mm. From equation 4.5, the maximum deflection of the wheels' bar is 0,747518 mm. So, according to equation 5.3, the total maximum deformation is 0,82156 mm, which verifies the total maximum deformation criterion.

Since all the criteria is verified, the solution is validated.

5.4 Solutions' Comparison

The new frame solution was achieved by a change in the structural layout, as well as, a change in the bars' cross-sections. A visual comparison of the layouts can be observed in Figure 5.11 and Figure 5.12, as well as, a comparison of their mechanical properties in Table 5.4.

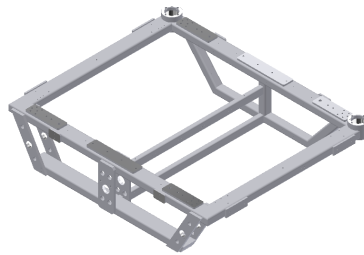


Figure 5.11 Original frame layout.

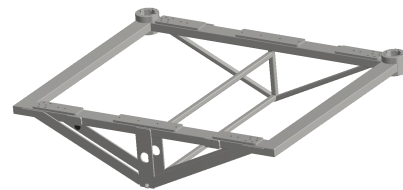


Figure 5.12 New frame layout.

Table 5.4: Frame solutions comparison.

Mechanical Results	Initial Layout	New Layout	Relative Variation
Maximum Stress / MPa	51,259	17,889	-65%
Maximum Deformation / mm	0,2781	0,04889	-82%
Reaction / N	8622	8513,1	-1%
Mass / kg	102,68	74,897	-27%

The change in the layout was the main reason that allowed for a reduction of 82% in deflection. It also reduces the difference in deflections from the supporting bars from 0,27043 mm to 0,03764 mm, which better the stability of the cargo being transported.

The new layout also allowed for a reduction of 65% in the maximum equivalent stress, consequently improving the safety factor.

The change in the bar's cross-sections in concert with the new layout, where bars were removed, made the frame lighter by 67%, ending up with a mass of 33,808 kg.

A comparison between the wheels' bar solutions can also be seen in Table 5.5.

Table 5.5: Wheels' bar solutions comparison.

Mechanical Results	Original Bar	New Bar	Relative Variation
Maximum Stress / MPa	52,06	60,43	16%
Maximum Deflection / mm	0,74	0,75	1%
Reaction / N	8622	8513,1	-1%
Mass / kg	24,14	18,46	-23%

The new wheel's bar solution increased the maximum stress by 16% and the deflection in an almost unnoticeable 1%. The increase in stress did not present a problem, since the bar was designed with a safety factor of 6,9151. The new safety factor, calculated in equation 5.2, is safely distant from the standard 1,5.

The loss in mechanical properties was a necessary trade-off to reduce the bar's mass in 23% as mass reduction is the objective for this optimization.

An analysis of the frame and bar combined can be seen in Table 5.6.

Table 5.6: Frame and wheels' bar solutions comparison.

Mechanical Results	Original Solution	New Solution	Relative Variation
Maximum Deflection / mm	1,0181	0,8215	-19%
Mass / kg	132,44	52,2	-60%

The two new solutions combine for a mass of 52,27 kg, opposing the initial 132,45 kg, resulting in a 60% decrease in mass. In terms of deformation, the sum of both deflections went from 1,0181 mm to 0,8215 mm, which is a 19% reduction.

The structural solution can be seen in Figure 5.13, where can be seen the structural elements combined: Both the original wheel sets, as well as, the new frame and wheels bar solution. The structural solution of the RGV is now 125,67 kg, opposed to the previous 205,91 kg, which is a 38% mass reduction.

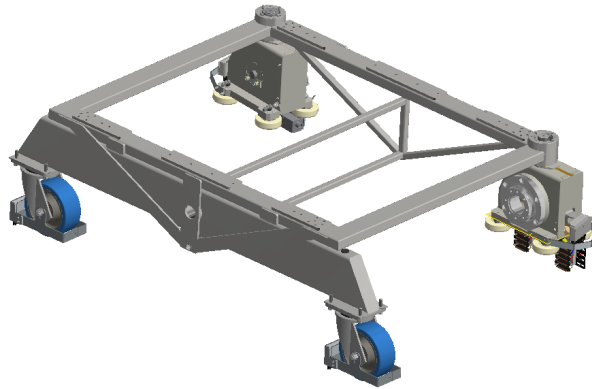


Figure 5.13: New structural solution.

Finally, the optimized RGV's mass is 218,1 kg, 27% less than the original.

5.5 Non-modified Components

As was mentioned in Chapter 3, most of the components analyzed (I-VI and XII-XV, from Table 3.5) do not have a structural purpose, so the study of their stresses, deflections, and forces do not present much relevance. The optimization of these components can be done in terms of mass reduction. It could be done by searching for a material lighter than the original S235 steel that fulfills the safety standards.

Concerning the wheels, their optimization in terms of mass and stress depend heavily on the amount of weight they support. As they should support the RGV, the conveyor and maximum cargo of 1500 kg, the reduction of mass of 27% in the RGV only accounts for a reduction of less than 4% in the overall weight that the wheels support. So their optimization was not viable.

Chapter 6

Conclusion

This chapter summarizes the main conclusion of this dissertation, reporting the objectives achieved and addressing the possible future development that allows the further improvement of the RGV structure.

6.1 Conclusion

The project restrictions were assessed for the RGV and a mass evaluation was made to all the RGV components, resulting in a decision to optimize the RGV's frame and wheels' bar. Each of these components was studied in terms of their materials, cross-sections, function in the RGV, and their accommodation in relation to other components. A structural analysis was made to both components, in order to evaluate the stresses and deflections that they were subjected to. From this characterization was chosen how to optimize each component.

A topology optimization was made to improve the frame's layout. A beam model of that layout was modeled, via ANSYS, approving the tubular bars and plates configuration. Iteratively, different cross-sections for the tubular bars and plates were tested, until any further reduction in mass was not possible due to the restrictions.

An analytical optimization process, imposing a maximum deflection, was used to reduce the wheels' bar mass. From the maximum deflection criterion, the mass reduction objective and the frame restriction allowed three commercial cross-sections were selected. From these three, it was selected the one that had the least mass per meter.

The solutions were then drawn in Inventor and validated according to the safety factor's standards.

The mass reduction results from the optimization were deemed good, as they allowed for a mass reduction of 67% in the frame, 38% in the wheels' bar, and 27% on the overall RGV. In terms of maximum equivalent stress, it was reduced in the frame by 65% and increased by 16% in the wheels' bar. The deflection was reduced by 82% in the frame and remained practically the same in the wheels' bar as it increased 1%.

6.2 Future Work

As the new frame's layout allows it, a further reduction in the RGV's frame is possible if the overall components accommodation is reorganized. An optimization of the conveyors that sit on the frame in terms of maximum stress allowable can also be done since it's possible to increase the maximum cargo weight for transportation in the RGVs.

A study regarding a new cross-section for the wheels' bar may improve it in terms of mass, maximum equivalent stress, and deflection. Since most of the deflection is due to the wheels' bar, the study of an RGV configuration that disregards it may allow for a great reduction of mass and costs.

A material change in the safety plates of the RGV could be studied since they do not have any structural value and may also allow reducing its mass. This study should be extended to the material of the components optimized since their safety values are excessive.

At last, the topology optimization tool could be used in other machinery, since it produced such great results in the RGV's frame.

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