DESIGN OF STACKER CRANE FOR MINI-LOAD AUTOMATED STORAGE AND RETRIEVAL SYSTEMS

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ABSTRACT

DESING OF STACKER CRANE FOR MINI-LOAD AUTOMATED STORAGE AND RETRIEVAL SYSTEMS

The objection of the thesis is to design and analyze a 3 degree of freedom stacker crane mechanism for Mini Load Automated Storage and Retrieval Systems. Automated Storage and Retrieval System (AS/RS) is an automated system in which the product of a company to be stored is placed in or retrieved from specified location on a rack system by means of computer aided mechanism which is nothing but a stacker crane.

This study first introduces an inclusive literature review on AS/RSs. Several types of AS/RSs are examined and the differences between them are introduced. Additionally, several types of stacker cranes in the literature are also presented. Moreover, the linear motion systems used in similar mechanism are explained briefly in the chapter.

In the design chapter of the thesis, theory of the stacker crane mechanism is introduced. A conceptual design is created and, elements and sub-sections of the desired mechanism are presented with the conceptual design of the mechanism. After the conceptual design, critical design calculations are performed approximately before the detailed design of the mechanism. Afterwards, the detailed design of the mechanism is created and calculations are verified due to detailed design.

The detailed design of the mechanism is analyzed using Finite Element Analysis methods in the analysis chapter of the thesis. Analysis of critical components and critical sub-sections of the stacker crane mechanism are analyzed using Computer Aided Engineering program. Revises are made according to the results of the FEA and design is validated at the end.

At the prototyping chapter of the thesis, the manufacturing studies of the mechanism are presented. Manufacturing stages are introduced and experiments of the sub-section of the mechanism are done.

ÖZET

OTOMATİK DEPOLAMA VE BOŞALTMA SİSTEMİ İSTİFLEYİCİ VİNÇ TASARIMI

Bu tezin amacı, Mini Load Otomatik Depolama ve Geri Alma sistemlerinde kullanılmak üzere 3 serbestlik dereceli istifleyici vinç tasarım ve analizini yapmaktır. Otomatik Depolama ve Geri Alma sistemleri; şirketlerin depolarında bulunan sıralı raf yapılarının sistem tarafından önceden belirlenmiş bir bölümüne, bir ürünün yerleştirilmesinin ya da yerleştirildiği yerden geri alınmasının, bilgisayar destekli bir istifleyici vinç tarafından (ya da benzeri başka bir mekanizma yardımı ile) tam otomatik olarak sağlandığı bir depo otomasyon sistemidir.

Bu tez çalışması öncelikle Otomatik Depolama ve Geri Alma sistemleri hakkında kapsayıcı bir literatür taraması sunar. Bu literatür taraması kapsamında, sektörde kullanılan birçok sistem incelenmiş ve özellikleri sunulmuştur. Bunlara ek olarak, piyasada kullanılan sistemler tarafından kullanılan istifleyici vinç çeşitleri de incelenmiştir.

Tezin tasarım bölümünde ise, istifleyici vinç mekanizmasının teorisi sunulmuştur. Daha sonra, konsept bir tasarım ortaya konmuş ve vinç tasarımında kullanılan elemanlar ve alt kısımlar konsept tasarım kapsamında sunulmuştur. Konsept tasarım sonucu elde edilen veriler kullanılarak, detaylı tasarım yapılması için gerekli hesaplamalar yapılmış ve yaklaşık veriler elde edilmiştir. Devamında ise detaylı tasarım aşamasına geçilmiş ve konsept tasarım ile elde edilen veriler kontrol edilerek doğrulanmıştır.

Tezin analiz bölümünde, detaylı tasarım sonucu ortaya çıkan mekanizma, sonlu elemanlar analizi yöntemi kullanılarak analiz edilmiştir. İstifleyici vinç mekanizmasının kritik elemanlarının ve kritik alt kısımlarının analizleri, bilgisayar destekli mühendislik programı yardımıyla yapılmıştır. Analiz sonucu gerekli revizeler yapılmış ve son tasarım kontrol edilmiştir.

Tezin sonunda ise tezin prototip çalışmalarına yer verilmiştir. Prototipi yapılan alt sistem aşamaları ile incelenmiş ve sistem test sonuçları irdelenmiştir.

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CHAPTER 1

INTRODUCTION

Material Handling can be expressed as "providing the right amount of the right materials, in the right condition, at the right place, at the right time, in the right position, in the right sequence, and for the right cost, by using the right methods." (Tompkins,1996). Automated Storage and Retrieval System (AS/RS) is a computer-aided material handling system (Vasili,2014) which is used to store and retrieve the products of a company in the warehouse. AS/RSs are used widely in the distribution centers and e-commerce companies. The system assists the company to operate the storage and to manage the products in the warehouse by a cost-effective, time and space saving solution.

AS/RSs comprise of one or more parallel aisles with multi-tiered racks, one or more stacker cranes (S/R machine), input/output (I/O) station, conveyor lines and computer systems (Vasili,2014). This thesis studies the design of a stacker crane to be used for AS/RSs. Stacker crane is a 3 degree-of freedom machine that works in aisle to store the products from input station to racks, and to retrieve the products from racks to output station. Stacker crane moves horizontally on a rail which is mounted on the ground and vertically on a rail which is mounted on a rigid mast. The end-effector of the machine is a carriage that also carries the shuttle mechanism and the product on it. The shuttle mechanism is a deployable type of mechanism that moves the product into or from its location.

The contribution of the AS/RSs to company can be listed as follows:

- Quick, effective and easy respond to current or potential market demands.
- Productive usage of the storage area.
- Vertical storage opportunity.
- Decreasing of the labor costs.
- Reducing the human-induced errors
- Decreasing of the industrial accident due to high level security.
- Easier control of the inventory.

Beside the numerous advantages of AS/RSs, there are some disadvantages that should be mentioned at this part. The most important of these is the high cost of installation of the system. There are several mechanical, electronical and computational parts to set up the system. Additionally, AS/RSs have many sub-systems and several engineering branches need to be in coordination with each other to accomplish the project and keep the project working. This increases the installation process time and cost. Moreover, the personnel who will work in the AS/RSs must be well trained.

1.1. Motivation of the Thesis

With the widespread of the internet usage along the world, new business opportunities related to the internet is emerged and widespread. One of the most important of these is the online shopping. In parallel with this development, people have changed their shopping habits and turned into more internet usage. In addition, people have started to give more importance to the delivery time of the products as well as the safety and quality of the product while online shopping.

The delivery time, then, become a criterion for choosing the supplier and led to competition among the suppliers. This competition has increased the use of new systems and machinery in warehouse management and product preparation for the delivery.

This thesis is on the design of a stacker crane for AS/RSs, which are automation systems being used along the warehouses frequently. In addition to the many advantages mentioned in the previous section, the usage of the stacker crane in warehouse also provides important advantages to the user company in competition on delivery time.

1.2. The Aim of the Thesis

The aim of this thesis is to design and analyze a mini-load stacker crane to be used in the Automated Storage and Retrieval Systems. Stacker Crane is a 3-degree-of-freedom machine that carries out the retrieving and storing tasks in the AS/RSs. The working principle of the stacker crane can be explained as that the crane moves horizontally on a rail in the first axis (let’s say it is on the x-axis), the carriage of the crane moves vertically on the mast of the crane in the second axis (let’s say it is on the y-axis) to reach the specified location on the rack/IO station, and the fork mechanism on the carriage deploys on the third axis (let's say it is on the z-axis) to retrieve/store the goods from/to specified rack section (Figure 1). Since all the movements of the mechanism are linear motion, the mechanism can be thought as a cartesian robot basically.

1.3. Outline of the Thesis

This thesis consists of 6 chapters: Introduction; Literature Review; Design of the Mechanism; FEM Analysis; Prototyping and Experiments and Conclusion.

In Chapter 2, literature review of AS/RSs, classification of the systems, types of the stacker cranes, mechanical and structural elements used in the AS/RSs are presented.

In Chapter 3, firstly the theory of the stacker crane mechanism is explained. Later on, a conceptual design is presented and calculations due to conceptual design is performed. Afterwards, the detailed mechanical design of the stacker crane is created. The design is developed by Autodesk Inventor environment.

In Chapter 4, the FEM analysis of the critical elements and sub-sections of the stacker crane design are performed.

In Chapter 5, prototyping of the fork mechanism of the stacker crane is produced. Experiments of the mechanism is performed on prototype and results are listed.

The conclusion of the thesis presented in Chapter 6. Output of the thesis is stated in the Conclusion chapter.

CHAPTER 2

LITERATURE REVİEW

Automated Storage and Retrieval System (AS / RS) or Automatic Storage System (ASS) is the system in which the product / material is placed on the racks in the warehouse by means of a computer-controlled machine, and the operations are carried out by retrieve it from the place where it is stored and lead to the desired location. The AS/RSs consist of I/O stations, conveyor lines, stacker cranes, aisles, racks, computers, and relevant electronic equipment.

The aim of this thesis is to design and prototype a stacker crane to be used in the Automated Storage and Retrieval Systems. Stacker Crane is a 3-degree-of-freedom machine that carries out the retrieving and storing tasks in the AS/RSs. The basic working principle of the machine is that the machine moves horizontally and vertically in the aisle and carries the product/material on its end-effector. The end-effector of the stacker crane is called “carriage”. The carriage carries a single degree-of-freedom telescopic fork mechanism named “shuttle mechanism” which performs the retrieving and storing task when stacker crane’s carriage is on the desired place.

Automated Storage and Retrieval Systems (AS/RSs) have been widely used in warehouse environments since 1950s (Roodbergen and Vis, 2009). Since they were first used in 1950s, AS/RSs has been the essential systems used for distribution centers and warehouses. According to the Groover (2001), AS/RSs can decrease the manual retrieval time by 70%. Increase in the storage and retrieval (S/R) process is not only advantage AS/RSs can provide the users, but also, they have the advantages including saving in labor cost, inventory control, improved throughput level, vertical storage opportunity, increased storage capacity, decreased failure and safety. (0’Shea, 2007)

Figure 1. Automated Storage and Retrieval System

(Source: Vasili,2014)

A company may want to automate the storage operations to achieve the possible objectives below: (Groover,2001)

- To increase storage capacity
- To increase storage density
- To recover factory floor space presently used for storing work-in-process
- To improve security and reduce pilferage
- To improve safety in the storage function
- To reduce labor cost and/or increase labor productivity in storage operations
- To improve control over inventories
- To improve stock rotation
- To improve customer service
- To increase throughput

Typically, AS/RSs consist of aisle(s), racks on the one or both sides of the aisles, stacker crane (S/R Machine), I/O station and conveyor system to complete the line of process. According to the type of the AS/RS, a worker can be also included into the system to operate.

2.1. Classification of Automated Storage and Retrieval Systems

The most known version of the AS/RS is the one with only one crane on each aisle. This crane is only capable of moving on this aisle and carries a palletized load to the racks. Racks are single deep therefore stacker crane can reach any load directly on the rack. This type of AS/RS is called single unit-load aisle-captive AS/RS (Roodbergen and Vis, 2009).

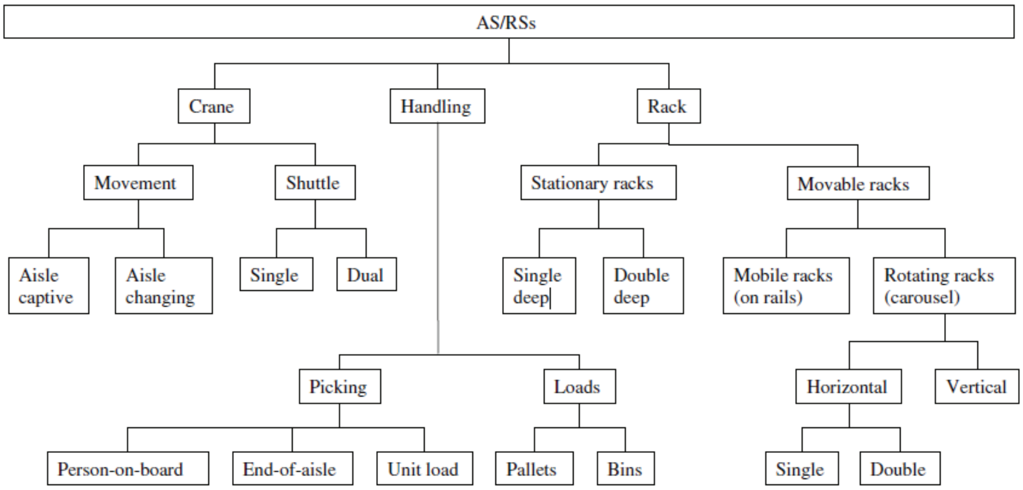

Figure 2. Classification of AS/RS

(Source: Roodbergen and Vis, 2009)

Numerous versions of AS/RS exist are classified by Roodbergen and Vis (2009) in the Figure 1. According to the classification of AS/RS in the Figure 1, the type of the AS/RS this thesis interested in is Single Mini-Load Aisle-Captive AS/RS. In Mini-Load AS/RSs, bins are used to store the products. The main difference between Unit-Load AS/RS and Mini-Load AS/RS is the mass of the product being stored. Generally, Mini-Load AS/RSs deals with the loads about 50-100 kg. These types of loads have smaller volume in comparison with the loads in Unit-Load AS/RS. The load of the product of the Unit-Load AS/RSs are generally above 250kg and stored with pallets. They have larger volume than the loads in Mini-Load AS/RSs. Rack structures of the AS/RS are also made according to the volume of the load to be stored.

There are other types of AS/RS according to their crane types (movement and shuttle capacity), rack structure and handling types. For example, the company would like to prefer to use only one crane on multi aisles to decrease the initial cost of the system. That type of AS/RSs are called the aisle changing AS/RSs. Another example is to control the AS/RS with a person which is called person-on-board type AS/RS. In these types of AS/RSs, there is an operator on the cabin that drives the machine to store and retrieve the loads from the racks.

According to Groover (2001), there are several principal types of AS/RS according to the size and the volume of the product to be stored, storage and retrieval type and the role of the human worker on the system:

2.1.1. Unit-Load AS/RS

The Unit-Load AS/RS is generally used when the product to be handled is large and carried by pallets or other types of standard containers. The system is fully automated with the computer systems and stacker cranes are controlled by the computer systems to store and retrieve the load/product. The aisle width is about 1.5-2 meters, and the height of the racks can be up to 40 meters.

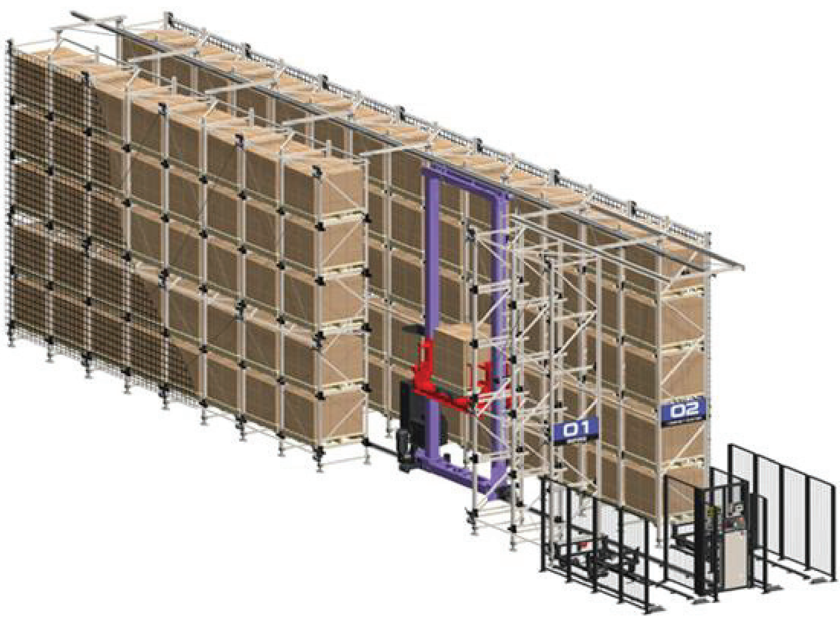

Figure 3. Unit Load AS/RS

(Source: Bastian Solutions, 2010)

2.1.2. Mini-Load AS/RS

The Mini-Load AS/RS is generally used when the product to be handled is light and smaller in size. Products of the system are stored in the carton or plastic boxes/bins and bins are stored on the racks by the crane. The system is fully automated as in Unit-Load AS/RSs. The aisle width is about 1 meter and height of the racks is typically up to 12-15 meters.

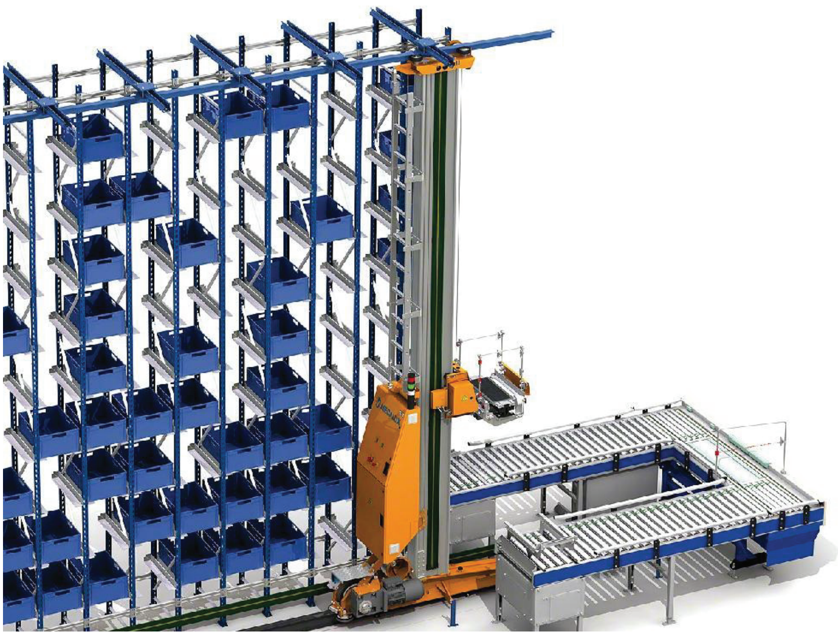

Figure 4. Mini Load AS/RS

(Source: Interlake Mecalux, 2022)

2.1.3. Deep-Lane AS/RS

Deep-Lane AS/RSs are systems with less product variety beside higher storage density. These types of systems are suitable for the warehouses where product quantity is high, but product variety is low. In Deep-Lane systems, rack depth is higher on both sides of the aisle. Generally, the depth is more than two loads deep. Main difference is that they run slower than previous systems.

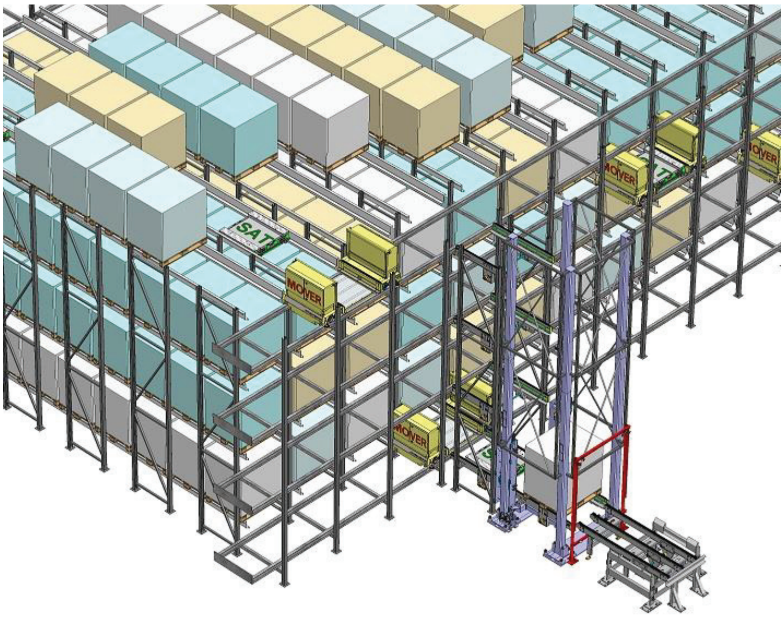

Figure 5. Deep Lane AS/RS

(Source: Soyaslan, 2015)

2.1.4. Man-on-Board AS/RS

Unlike the other systems, Man-on-Board AS/RSs are controlled by a human operator. Operator rides the stacker crane on the crane's carriage. The system is slower than previous systems.

Figure 6. Man on Board AS/RS

(Source: Soyaslan, 2015)

2.1.5. Vertical Lift AS/RS

All previous types of AS/RSs are located along a horizontal aisle. Vertical Lift AS/RSs (VL-AS/RS) has the same working principal but the crane/lift of the systems run on a vertical path. These types of systems are preferred to save available warehouse area.

2.2. Components of Automated Storage and Retrieval Systems

Automated Storage and Retrieval Systems consist of several mechanical and electronical components to accomplish the storing and retrieving task. These components are:

1. Storage Structure
2. Stacker Crane (S/R Machine)
3. Storage Modules
4. Conveyors

5. I/O Station
6. Control System

2.2.1. Storage Structure

The storage structure is a rack construction that is used in the warehouses to carry the loads which are stored on it. Rack structure is made by fabricated steel and is expected to have sufficient strength and rigidity in order not to deflect due to weight of the loads stored (0'Shea, 2007). Guiding safety components of S/R Machine are also mounted to the rack structure along the aisle.

2.2.2. Storage Modules

In AS/RSs, containers are used to store the loads in them. These containers are called storage modules. The storage modules in the AS/RSs are in standard dimensions because the rack structure and the carriage of the stacker crane are generally designed for one type of container. The container types used in AS/RSs are pallets, wire baskets, plastic totes etc.

2.2.3. Conveyors

Conveyors are mechanical carrier systems that allow objects to be moved from one location to another. Conveyors are generally used to facilitate the transportation of heavy or bulky loads and to save time in product transportation. Conveyors can be used wherever these advantages are needed. In automatic storage and retrieval systems, they are used as a complement to the system in transporting the product to the I/O station where the stacker crane machine picks up the product and leaves it back, and then delivering the product to the relevant personnel/station. Conveyor systems are made smart with the addition of various electronic equipment and thus they can be used fully automatically. The plastic boxes used in Mini-Load AS/RS can be guided on the conveyor side while moving on the conveyor lines thanks to the barcodes on them, and station diversity can be provided in the system. Roller conveyors are generally used in AS/RS.

Figure 7. Conveyor System

2.2.4. I/O Stations

The I/O station is the section where the product to be stored or retrieval the racks is picked up or dropped by the stacker crane. It is mechanically designed in accordance with the geometry of the fork mechanism on the stacker crane. It connects the stacker crane with the conveyors. In some AS/Rs, the product can be left directly by the personnel, or the product can be left and picked up by other moving vehicles other than the conveyors.

2.2.5. Control System

In automatic storage and retrieval systems, the task of the control system is to manipulate the stacker crane to store or retrieve the loads to its correct position on the racks within an acceptable tolerance. Horizontal and vertical position information on the racks of each product in the system (plastic bins in Mini-Load systems) is defined. The control system uses this information to control the stacker crane.

2.2.6. Stacker Crane (S/R Machine)

Stacker Crane is a 3-degrees-of-freedom machine that accomplish the storing and retrieving task in the Automated Storage and Retrieval Systems. The Stacker Crane receives the load to be stored from the I/O station, moves horizontally on the guide rail extending along the corridor, and vertically in the direction of the mast it carries on its main chassis, reaching the relevant rack section. It leaves the load on the rack section. In the opposite case, it takes the load on the rack to leave it to the I/O station. Along the corridor where the Stacker Crane moves, there are guide rails on the floor and ceiling. Its movement on the rail on the ground takes place by means of wheels. During this movement, the rails on the ceiling provide guidance. At the same time, vertical movement along the mast is supported by the guide rails on the pole.

Stacker Crane has a shuttle mechanism in its wagon to leave the load it carries on the rack or to take it off the rack. Shuttle mechanism opens and closes along the third axis to take the load in the rack section or to leave the load to the section in the rack (the first two axes on which the machine moves are considered as the horizontal movement along the corridor and the vertical movement along the mast).

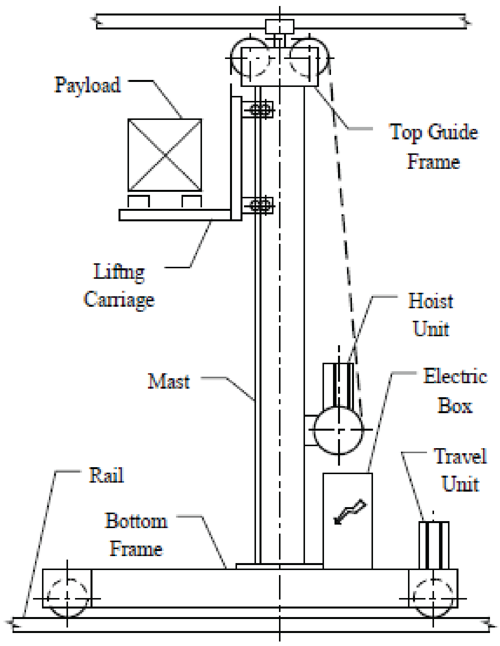

Figure 8. Single Mast Stacker Crane

(Source: Hajdu and Gaspar, 2014)

2.2.7. Stacker Crane Classification

Stacker cranes are classified according to the mast structures and the working type of the shuttle/fork mechanism (opening directions and opening deep).

For the mast structure classification, there are mainly 2 types of stacker cranes in the literature. These are single-mast stacker crane and double-mast stacker crane. In the single mast designed stacker cranes, the carriage of the machine is mounted on the one side of the mast and generally there are other components/equipment (electric box, etc.) on the other side of the mast. On the other hand, in the double-mast designed stacker cranes, the carriage(s) of the machine is mounted between two masts. Double-mast designed stacker cranes are generally preferred in the AS/RSs where the load to be stored is heavier (Unit-Load AS/RS). This thesis deals with a Mini-Load AS/RS and single mast stacker crane are considered to be sufficient due to lighter load.

For the working type classification of the shuttle mechanism, there are mainly 4 types of stacker cranes in the literature. These are double/single side fork mechanisms and double/single deep fork mechanisms. In order to take the advantages of using the both sides of the aisle and using the rack deeper, double side and double deep fork mechanisms are generally preferred to single side and single deep ones in the literature.

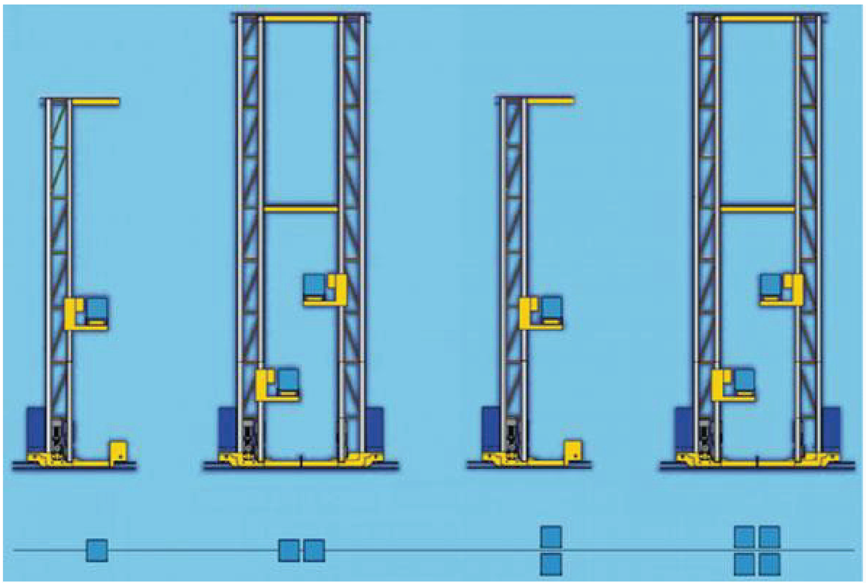

Figure 9. Common Types of Stacker Cranes in AS/RSs

(Source: Vasili,2014)

2.2.8. Conclusion

The main focus of the thesis is to provide an automation solution to customers of the company, Teknokom Electronic AS. The company provides order picking automation solutions to its customers. According to the costumer portfolio, it is the best option to choose the Mini-Load type AS/RS among all the types of AS/RSs.

2.3. Linear Motion Systems Used in AS/RSs

The AS/RSs has 3 degrees of freedom in total. The end-effector's position is calculated with 3 parameters and can be driven by 3 motors. In these systems, the rotational motion produced by the motor is converted into linear motion by the power transmission elements and the load is moved/carried linearly.

The most frequently used power transmission elements for AS/RSs that convert rotational motion to linear motion are timing belt and pulleys, rack and pinions, rope and pulleys and drive wheels. In addition, the ball/lead screw mechanisms can be shown among the linear power transmission elements used in Mini Load systems, which is seen only in the shuttle mechanism. However, ball/lead screw mechanisms are too slow compared to other systems therefore they are not preferred except some examples.

Each power transmission element has advantages and disadvantages, and the designer chooses according to the requirements of the design. Efficiency, performance, cost, applicability and security are the main criteria for deciding the mechanism for linear motion.

In addition to linear motion mechanism; structural support, guides, ease of maintenance, accuracy, precision and repeatability are the critical factors to consider. Since the main focus of the thesis is to design a Stacker Crane for Mini-Load AS/RSs with good accuracy, good precision and easy maintainability; there are two options left to consider for linear motions: “Belt and pulley” and “Rack and Pinion”. Steel rope and pulley mechanism is preferred when the load is relatively heavy (typically for the cranes for Unit-Load AS/RSs), and the positioning accuracy is lower than the opponents, therefore this type of linear mechanism is eliminated. On the other hand, due to probability of slipping while working, wheel drive mechanisms are also eliminated, although it has been used frequently in the industry.

On the other hand, the guidance of the linear motion system is critical since the accuracy of the system is vital. There are mainly three types of linear guidance used in AS/RSs. These are ball guides, wheel guides and prism slides. Wheel guides are generally used in AS/RSs due to cost and ease of maintenance.

2.3.1. Timing Belt and Pulley

One of the most frequently encountered power transmission elements in Mini Load systems is the timing (toothed) belt. Unlike other belts, there is nearly no slippage during power transmission due to toothed geometry. In this way, precise power transmission at constant angular speed is possible.

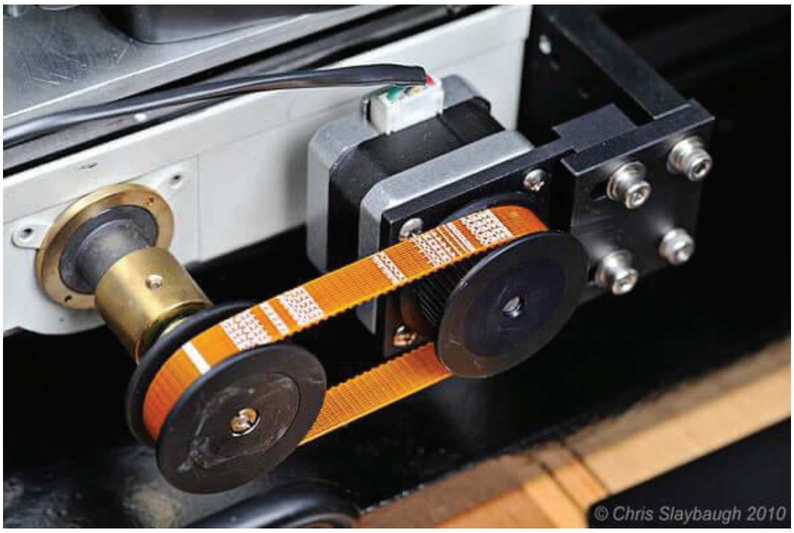

Figure 10. Timing Belt and Pulley Drive

(Source: Layosa, 2014)

The main advantages of the timing belt pulley systems have low-cost, ease of installation, working range at high speeds, no lubrication needed and light weight. Moreover, timing belts have production flexibility that they can be produced in any desired length up to 100 meters.

It's light weight feature provides advantages for the AS/RS cranes because in the AS/RSs, the vertical motion and the fork mechanism elements are carried by the machine during the process.

Since the maintenance is an essential factor for AS/RSs, another important advantage of belt mechanisms is that they do not require lubrication. This allows them to work in the open and continuously. In addition, belt-pulley mechanisms can operate at high speeds.

Besides, timing belts also have disadvantages that should be considered during the design. Timing belts are not suitable for the high temperature environments and they have relatively low resistance to chemical products.

2.3.2. Rack and Pinion

Rack is a linear gear rod with threads on it. A rack and pinion gear mechanism is used to convert the rotational motion to linear motion or visa verse. Usually, the rotational force (torque) on the pinion is transmitted to linear force on the rack. Rack and pinion gear mechanism is encountered in many applications where rotational motion is desired to be converted into linear motion.[6]

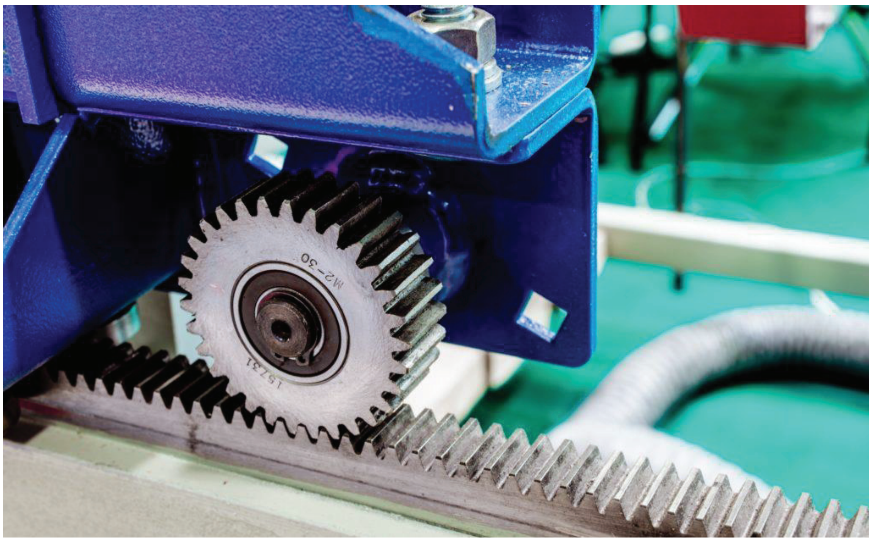

Figure 11. Rack and Pinion Drive

(Source: Eitel, 2021)

The most important advantage of rack and pinion gears is that more than one system can work independently. Besides, it has installation flexibility. They can be installed at very long working distances by joining the rack profile together and work without changing their performance. In addition to their low cost, they are simple in structure and have high strength. In this way, the rigidity of the systems in which these systems are used is high.

Another prominent advantage of rack and pinion gears is their high speed. This feature indicates that they are suitable for Mini Load systems. Although they can be controlled with precision under normal conditions, a correct installation is important. Besides, rack and pinion gears are heavy. With this aspect, they are not used in applications where mass is important.

The biggest disadvantage of rack and pinion gears is that although their cost is low, they require expertise and experience for installation. In addition, the need for regular lubrication is another factor that increases the cost.

As a result of the section; for an accurate, low-cost, easy maintenance, fast and light weight system requirements, belt and pulley mechanism is decided to be used in both horizontal and vertical motion of the stacker crane. Additionally, for the simplicity, rack and pinion mechanism is decided for the linear motion of the fork mechanism.

CHAPTER 3

DESIGN OF THE MECHANISM

3.1. Conceptual Design

3.1.1. Theory of the Mechanism

The aim of this thesis is to design and analyze a mini-load stacker crane to be used in the Mini Load Automated Storage and Retrieval Systems. Stacker Crane is a 3-degree-of-freedom machine that carries out the retrieving and storing tasks in the AS/RSs. The working principle of the stacker crane can be explained as that the crane moves horizontally on a rail in the first axis (let's say it is on the x-axis), the carriage of the crane moves vertically on the mast of the crane in the second axis (let's say it is on the y-axis) to reach the specified location on the rack/IO station, and the fork mechanism on the carriage deploys on the third axis (let's say it is on the z-axis) to retrieve/store the goods from/to specified rack section (Figure 1). Since all the movements of the mechanism are linear motion, the mechanism can be considered as a cartesian robot basically as seen in the Figure 2.

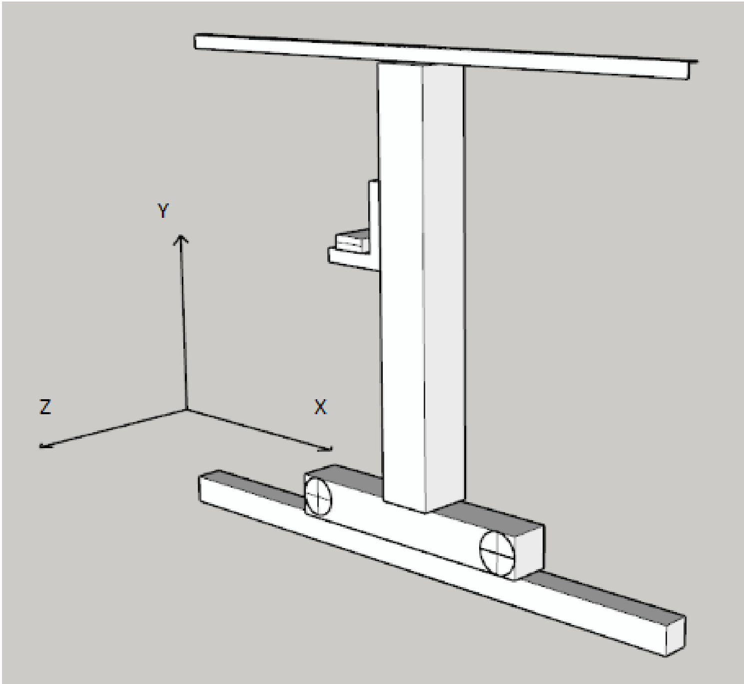

Figure 12. Stacker Crane Configuration

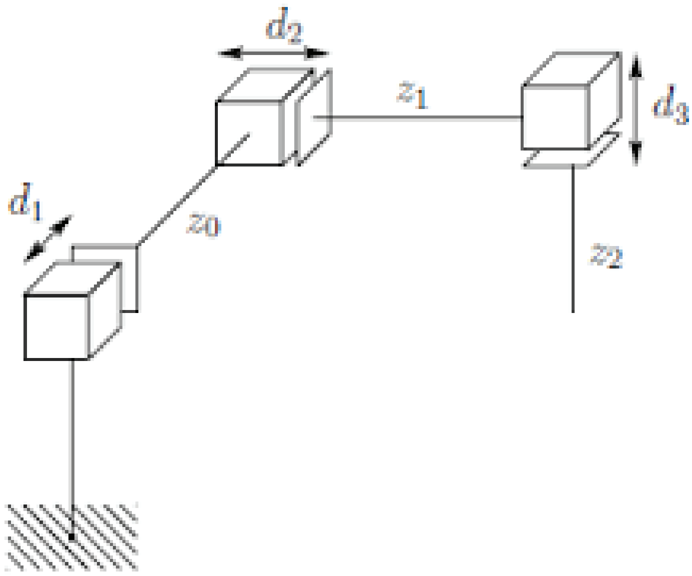

Figure 13. Cartesian Robot Theory

(Source: El Naser, 2018)

3.1.2. Elements of the Mechanism

In the thesis, a conceptual design of the mechanism will be completed firstly. In order to evaluate a detailed design, standard machine elements to be used in the mechanism must be specified beforehand. Therefore, in the conceptual design, the main structure of the mechanism will be designed roughly. According to the custom designed sections, mass information will be obtained approximately. Approximate mass values are critical for the selection of the standard elements and electric motors, and estimated according to the size limitations of the prototyping location which is the warehouse of the company. Based on the limitations of the warehouse, the prototype of the mechanism will be designed and operated for a 3x4 rack system with the height of 2,5 meters. On the other hand, according to the selected type of the AS/RS, which is Mini Load AS/RS, the design specifications have been determined in order to design the stacker crane. These specifications include the velocities of the system (3 dof), accelerations of the system, mass of the load to be moved and power supplying properties. Velocity and acceleration values for the prototype are kept lower than the planned sales product in order to keep the motor specifications and cost due to these specifications lower in prototyping phase. In any case, obtained values will be multiplied with a safety factor to ensure the design. After the determination of the specifications, the conceptual design of the mechanism can be design and prototype.

Conceptual design of the mechanism has been completed using the Sketchup environment. Design includes the rail of the system on which the stacker crane moves, wheel hubs on two sides, main body on which the mast of the stacker crane mounted, mast of the crane, carriage and telescopic fork mechanism. Additionally, other critical components (electric motors, linear guidance elements, belt and pulleys, etc.) are included due to get a real-like design for determining the masses of the mechanism before the detailed design. During the design of the mechanism, if possible, standard parts in the market were preferred instead of custom designed parts in order to reduce the cost, time and manufacturing defects.

In the rail section of the system, IPE standard steel I-beam is used as rail. An IPE is a structural steel beam with parallel internal surface flanges according to the European Standard EN 10365. Tolerances of the beam characterized in EN 10034 (Montanstahl, 2019). Additionally, rail pad and rail clips are used for the setup of the rail section which are not included on the conceptual design.

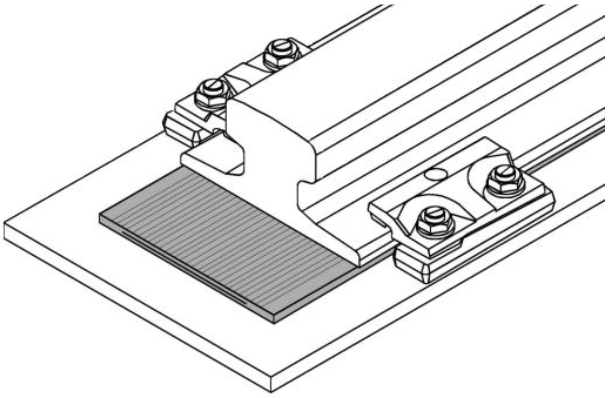

Figure 14. Rail Section Elements

(Source : Gantrex, 2006)

Wheel hub sections of the mechanism consists of grooved wheel, shaft, the wheel hub structure and related elements. Wheel hubs carry the main frame on both sides and guides the mechanism on the rail by the help of the geometry of the grooved wheels. Conceptual design of the wheel hub is showed in the Figure 4.

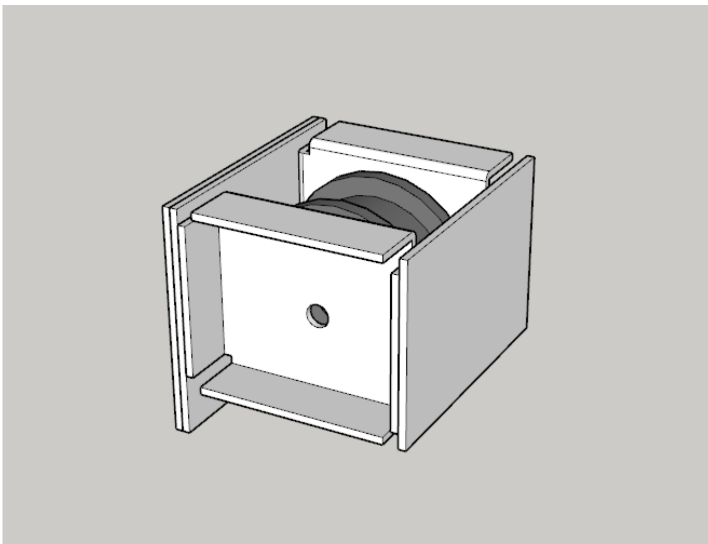

Figure 15. Wheel Hub Section

Main frame of the mechanism carries the mast, carriage, electric motors, fork mechanism, load of the system and linear drive elements. Main frame consists of 2 IPE standard I-beams and connection parts (Figure 5). The design of the main frame kept as simple as possible due to ease of installation.

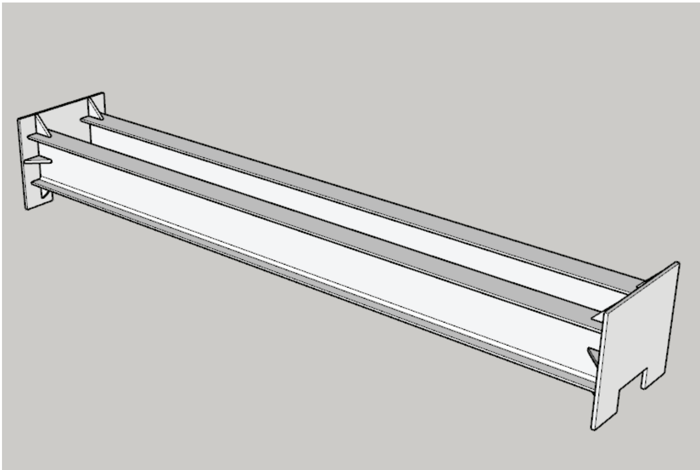

Figure 16. Main Frame

For the mast structure of the crane, a square shape standard profile is used. Additionally, for the guidance of the lifting mechanism, 2 rectangular profiles are added to the mast. On the other hand, carriage of the crane also consists of rectangular profiles in order to carry the fork mechanism and the load. Carriage is driven by a lifting electric motor, a belt and 2 pulleys; and has wheels for linear guidance. A conceptual design of mast and carriage with the parts mentioned are shown in the Figure 6.

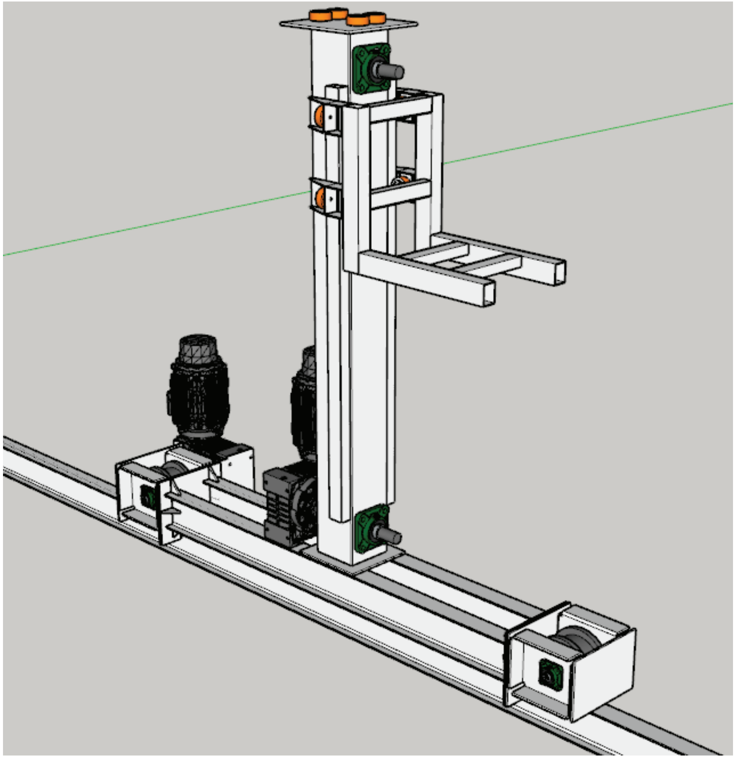

Figure 17. Stacker Crane Conceptual Design

As mentioned before, fork mechanism is a kind of deployable mechanism that stores/retrieves the product from/to rack structure (Figure 18). It will be located on the carriage and the top plate of the fork mechanism is nothing but the end-effector of the mechanism. Fixed plate is designated as bottom plate of the fork mechanism in the thesis. Moving plates of the mechanism are designated as middle and top plates, respectively.

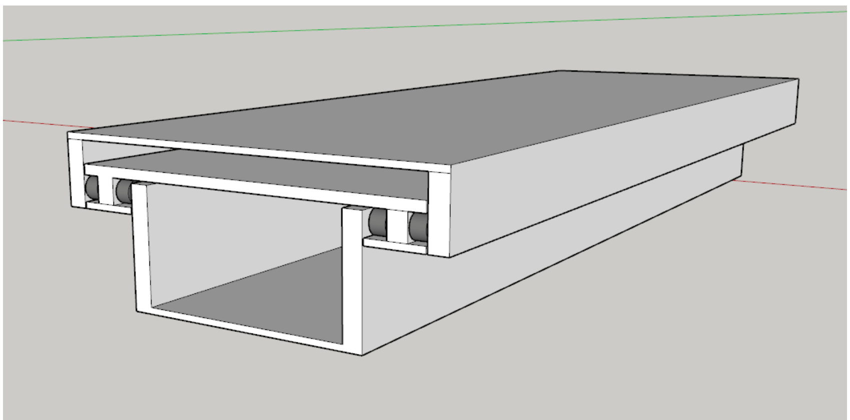

Figure 18. Fork Mechanism Conceptual Design

3.2. Calculations due to Conceptual Design

During the conceptual design of the mechanism, standard structural and machine elements in the market were preferred instead of custom designed parts in order to reduce the cost, time and manufacturing defects. Moreover, standard elements have catalogue information for the engineering calculations therefore it is also time saving for the project.

3.2.1. Mass Calculations

Mass calculation of the mechanism is important for the determination of standard machine elements, electric motor, timing belts and related pulleys, and also important for the verification of the selected standard structural elements. Mass calculation is done approximately and calculations will be checked and verified after the detailed design of the mechanism.

For the mass calculation, each sub-section of the stacker crane has been calculated separately and a table of mass parts has been created afterwards. Sub-sections of the mass calculation are wheel hubs (there are 2), main frame, mast structure, carriage and fork mechanism (load is included). Rail of the system is not included because it does not affect any of the calculations.

- Wheel hubs consist of wheel, shaft of the wheel, bearings, metal plates and fasteners.
- Main frame consists of 2 pieces of I-beam and connection metal plates. The electric motors are also mounted to the main frame.
- Mast of the mechanism consists of 2 types of rectangular/square profiles, bearings, top guide wheel and connections metal plates. Mast structure also carries the belt and pulleys of the lifting motion.
- Carriage consists of rectangular profiles and guidance wheels.
- Fork mechanism consists of 3 plates, rack profiles, pinion gears and related parts.

Table 1. Approximate Mass Values of the Sub-Sections

Sub-Section	Mass (kg)
Wheel Hubs	120 (2 of the hubs)
Main Frame	90
Mast	90
Carriage	50
Fork Mechanism	50
Load	50
Total	**450**

3.2.2. Motor Calculations

Since the mini-load AS/RS stacker crane is a 3 degrees-of-freedom mechanism, there have to be at least 3 motors to drive the mechanism. These motors drive the mechanism on three axes individually. As mentioned before, rotational motion of the motors is converted to linear motion by the help of linear motion systems, therefore linear motion of the mechanism is provided.

Motor calculations include the determination of the motor types and sizing of each motor. There are several types of electric motors in the industry and each type has its own characteristics. Design engineers select the electric motors according to type of application, duty cycle, power requirements, operating lifetime and cost.

Types of electric motors are commonly divided into 2 categories: AC motors and DC motors. As the names suggest, AC motors use alternating current to power while DC motors use direct current.

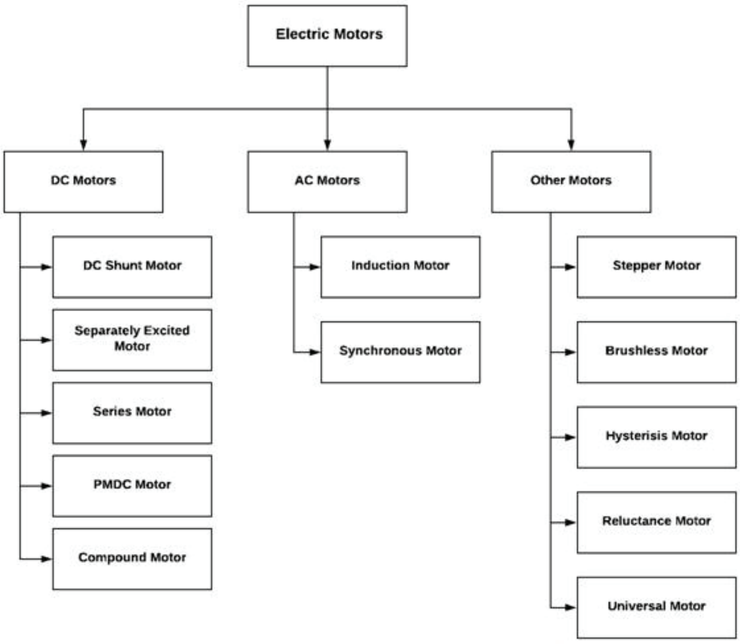

Figure 19. Types of Electric Motors

(Source: Elprocus, 2013)

As indicated in the Figure 7, AC motors are separated commonly into 2 main categories: Synchronous and Asynchronous (Induction) AC motors. Squirrel cage induction motors are the most common type of AC motors in the industry. On the other hand, DC motors are separated commonly into 2 categories: Brushless and Brushed DC motors.

In this project, all the motors are selected as asynchronous motors for the following advantages they have (Elprocus, 2013):

- Low cost: Due to their modest design, asynchronous motors are cheap compared to other types of motors. Moreover, for industrial or domestic applications, AC power supplying can be easily use to run the motor.
- Low maintenance cost: Due to its simple construction, maintenance of asynchronous motors is easy and cheap.
- Ease of operation: Asynchronous motors are self-started motors and do not need extra electrical connector to its rotor.
- Speed variation: Speed variation of asynchronous motors are almost constant.

- High starting torque: For crane-like operations, where the load is applied before the starting of the motor, high starting torque is needed to drive the mechanism. Asynchronous motors have high starting torque values and suitable for the project.
- Durability: Asynchronous motors are also durable. These motors can be used for many years with low maintenance.

Beside all the advantages mentioned, asynchronous motors have also some disadvantages to consider. High initial torque, speed control difficulty and low efficiency are the most commonly known disadvantages of the asynchronous motors.

In order to size the correct motor for any project, there are 3 main criteria to consider: torque, load inertia and speed (Tang, 2020). Torque value has 2 main components to calculate: load torque and acceleration/deceleration torque. Load torque is the required amount of torque which is needed when the machine is in the constant final velocity, and includes frictional and gravitational effects. Calculation of load torque differs according to the application. In general, it is the sum of frictional and gravitational forces.

Acceleration torque is the required amount of torque which is needed when the machine is accelerating or decelerating. In order to calculate the acceleration torque, load inertia and speed must be known beforehand.

Inertia is the resistance of any physical object to change in velocity (Newton, 1846). For any rotational object, the inertia is calculated by its mass and the square of its radius of gyration. For the linear motion systems, inertia of the objects in linear motion is calculated by multiplying the mass of the objects and the square of the radius of the element which converts the rotational motion to linear motion (Equation 1).

$$J = m\,x\,r^2 \qquad [1]$$

where:

J = Inertia of the objects in linear motion (kgm^2),

m = Mass of the objects (kg),

r = Radius of the pulley/gear (m).

For example, for timing belt pulley systems, inertia of the objects in linear motion is calculated by the mass of the objects and square of the radius of the drive pulley.

Additionally, there are rotational objects in cylindrical shape which rotate when the machine moves. Inertia of these objects is calculated by the equation (Equation 2) below:

$$Jr = \frac{1}{8}\ x\ m\ x\ D^2 \qquad [2]$$

where:

J = Inertia of the objects (kgm^2),

m = Mass of the objects (kg),

D = Diameter of the pulley/gear (m).

Therefore, total load inertia (J_L) is simply the sum of all the individual inertias of the objects moved by the motor.

In the project, the linear velocity of each motion is specified beforehand. According to the final linear velocity, angular velocity (rpm) of each motion system is specified using the equation (Equation 3) below:

$$N\ (rpm) = \frac{60\ x\ v(ms^{-1})}{\pi\ x\ D} \qquad [3]$$

where:

N = Angular velocity (rpm),

v = Linear velocity (m/s),

D = Diameter of the pulley/gear (m).

After determining the load inertia and velocity of the system, acceleration torque can be calculated by the equation (Equation 4) below:

$$Ta = \frac{Jo\ x\ i^2 + J_L}{9.55}\ x\ \frac{N}{t} \qquad [4]$$

where:

Ta = Acceleration torque (Nm),

J_o = Rotor inertia (kgm^2),

i = reduction ratio,

J_L = Load inertia (kgm^2),

N = Angular velocity (rpm),

t = Acceleration time (s).

After the calculation of both load torque and acceleration torque, total torque can be easily calculated by summing of 2 torque values (Equation 5):

$$T_T = (T_L + T_a)\ x\ Sf \quad [5]$$

where:

T_T = Total torque (Nm),

T_L = Load torque (Nm),

Ta = Acceleration torque (Nm),

Sf = Safety Factor.

At the end of the sizing procedure, required power of the motor can be calculated by the equation (Equation 6) below:

$$P = \frac{T\ x\ N}{9548} \quad [6]$$

where:

P = Power (kW),

T = Torque (Nm),

N = Angular velocity (rpm).

Sizing of Motor-1

Using the motor sizing procedure explained, all of the motors are selected. The first motor of the mechanism, which is called Motor-1, is used to drive stacker crane over the rail horizontally and pulls all the mechanism except rail. The total mass pulled by Motor-1 is calculated approximately in Table 1. The load torque for Motor-1 is:

$$F = fr \; x \; m \; x \; g = 0.019 \; x \; 450 \; x \; 9.81 = 83.87 \; N$$

$$T_L = F \; x \; r = 83.87 \; x \; 0.1 = 8.387 \; Nm$$

where:

fr = Coefficient of rolling friction (Lippert and Spektor, 2013)

Since all of the load, except wheels and shafts, is moving linearly by the pulley of the belt system, it is considered to use the Equation 1 which is about the inertia calculation of the objects in linear motion:

$$J = m \; x \; r^2$$

$$J_M = 450 \; x \; 0.1^2 = 4.5 \; kgm^2$$

Additionally, the inertia of the rotating parts (pulley, wheels and shafts) is calculated by the Equation 2:

$$Jr = \frac{1}{8} \; x \; m \; x \; D^2$$

$$Jp = \left(\frac{1}{8} \; x \; 1 \; x \; 0.2^2\right) = \; 0.005 \; kgm^2$$

$$Jw = \left(\frac{1}{8} \; x \; 10 \; x \; 0.2^2\right) x \; 2 = \; 0.1 \; kgm^2$$

$$Js = \left(\frac{1}{8} \; x \; 2 \; x \; 0.05^2\right) x \; 2 = \; 0.0006 \; kgm^2$$

Total inertia of the system is calculated by summing all the inertia values which is:

$$J_L = 4.6056 \; kgm^2$$

Final velocity of the mechanism in horizontal axis is considered to be 1 m/s. Time to reach to final velocity is considered to be 1.5 s. Therefore, acceleration of the system is 0.66 m/s^2. Using the specified information, angular velocity of the motor can be calculated by Equation 3 in rpm:

$$N\ (rpm) = \frac{60\ x\ v(ms^{-1})}{\pi\ x\ D}$$

$$N\ (rpm) = \frac{60\ x\ 1(ms^{-1})}{\pi\ x\ 0.2} = 95.49\ rpm$$

Later on, required acceleration torque can be calculated by Equation 4:

$$Ta = \frac{Jo\ x\ i^2 + J_L}{9.55}\ x\ \frac{N}{t}$$

$$Ta = \frac{4.60}{9.55}\ x\ \frac{95.49}{1.5} \cong 30.66\ Nm$$

Where the inertia due to rotor is so small compared to the system's inertia hence it is neglected. Total torque of the system (Equation 5):

$$T_T = (T_L + T_a)\ x\ Sf$$

$$T_T = (8.387 + 30.66)\ x\ Sf \ \cong 39.05 x Sf\ Nm$$

Required motor power of the mechanism (Equation 6):

$$P = \frac{T\ x\ N}{9548}$$

$$P = \frac{39.05\ x\ Sf\ x\ 95.49}{9548} = 0.39\ x\ Sf\ kW$$

According to the calculations, required motor power is found 0.39 kW, except safety factor. Safety factor is considered to be 2 at least (Türk Loydu, 2017), hence the motor power will be 0.78 kW. Nearest electric motor with this power specification in the market is 1.1 kW electric motor, with a safety factor of 2.82.

Sizing of Motor-2

The second motor of the mechanism, which is called Motor-2, is used to drive stacker crane's carriage on vertical axis and carries the load, fork mechanism and carriage structure. The total mass pulled by Motor-2 is calculated approximately in Table 1. Since the Motor-2 works against gravity, the load torque is greater compared to Motor-1. The load torque for Motor-2 is:

$$F = Fa + Fg$$

$$F = m\ x\ g\ x(1 + fr) = 150\ x\ 9.81\ x\ 1.019 = 1499.45\ N$$

$$T_L = F\ x\ r = 1499.45\ x\ 0.05 = 74.97\ Nm$$

where:

Fa = Frictional resistive force

Fg = Gravitational resistive force

fr = Coefficient of rolling friction (Lippert and Spektor, 2013)

Since all of the load, except wheels and shafts, is moving linearly by the pulley of the belt system, it is considered to use the Equation 1 which is about the inertia calculation of the objects in linear motion:

$$J = m\ x\ r^2$$

$$J_M = 150\ x\ 0.05^2 = 0.375\ kgm^2$$

Additionally, the inertia of the rotating parts (pulleys, wheels and shafts) is calculated by the Equation 2:

$$Jr = \frac{1}{8}\ x\ m\ x\ D^2$$

$$Jp = \left(\frac{1}{8}\ x\ 1\ x\ 0.1^2\right) x\ 2 = 0.0025\ kgm^2$$

$$Jw = \left(\frac{1}{8}\ x\ 1\ x\ 0.1^2\right) x\ 6 = 0.0075\ kgm^2$$

$$Js = \left(\frac{1}{8}\ x\ 2\ x\ 0.05^2\right) x\ 2 = 0.0006\ kgm^2$$

Total inertia of the system is calculated by summing all the inertia values which is:

$$J_L = 0.3856\ kgm^2$$

Final velocity of the mechanism in vertical axis is considered to be 0.5 m/s. Time to reach to final velocity is considered to be 1.5 s. Therefore, acceleration of the system is 0.33 m/s^2. Using the specified information, angular velocity of the motor can be calculated by Equation 3 in rpm:

$$N\ (rpm) = \frac{60\ x\ v(ms^{-1})}{\pi\ x\ D}$$

$$N\ (rpm) = \frac{60\ x\ 0.5(ms^{-1})}{\pi\ x\ 0.1} = \ 95.49\ rpm$$

Later on, required acceleration torque can be calculated by Equation 4:

$$Ta = \frac{Jo\ x\ i^2 + J_L}{9.55}\ x\ \frac{N}{t}$$

$$Ta = \frac{0.3831}{9.55}\ x\ \frac{95.49}{1.5} \cong 2.99\ Nm$$

Where the inertia due to rotor is so small compared to the system's inertia hence it is neglected. Total torque of the system (Equation 5):

$$T_T = (T_L + T_a)\ x\ Sf$$

$$T_T = (74.97 + 2.99)\ x\ Sf\ \cong 77.96\ x\ Sf\ Nm$$

Required motor power of the mechanism (Equation 6):

$$P = \frac{T\ x\ N}{9548}$$

$$P = \frac{77.96\ x\ Sf\ x\ 95.49}{9548} = 0.78\ x\ Sf\ kW$$

According to the calculations, required motor power is found 0.78 kW, except safety factor. Safety factor is considered to be 2 at least (Türk Loydu, 2017), hence the motor power will be 1.56 kW. Nearest electric motor with this power specification in the market is 2.2 kW electric motor, with a safety factor of 2.82.

Sizing of Motor-3

The third motor of the mechanism, which is called Motor-3, is used to drive the Fork Mechanism on third axis and carries the fork mechanism moving layers and workload. The total mass pulled by Motor-3 is calculated approximately in Table-1.

$$F = fr\ x\ m\ x\ g = 0.019\ x\ 150\ x\ 9.81 = 27.96\ N$$

$$T_L = F\ x\ r = 18.64\ x\ 0.0375 = 1.05\ Nm$$

where:

fr = Coefficient of rolling friction (Lippert and Spektor, 2013)

Since all of the load, except pulleys and shafts, is moving linearly by the pulley of the rack system, it is considered to use the Equation 1 which is about the inertia calculation of the objects in linear motion:

$$J = m\ x\ r^2$$

$$J_M = 150\ x\ 0.0375^2 = 0.21\ kgm^2$$

Additionally, the inertia of the rotating parts (pulleys and shafts) is calculated by the Equation 2:

$$Jr = \frac{1}{8}\ x\ m\ x\ D^2$$

$$Jp = \left(\frac{1}{8}\ x\ 1\ x\ 0.075^2\right) x\ 2 = 0.0014\ kgm^2$$

$$Js = \left(\frac{1}{8}\ x\ 2\ x\ 0.05^2\right) x\ 2 = 0.0006\ kgm^2$$

Total inertia of the system is calculated by summing all the inertia values which is

$$J_L = 0.212\ kgm^2$$

Final velocity of the fork mechanism in third axis is considered to be 0.75 m/s. Time to reach to final velocity is considered to be 1.5 s. Therefore, acceleration of the system is 0.5 m/s^2. Using the specified information, angular velocity of the motor can be calculated by Equation 3 in rpm:

$$N\ (rpm) = \frac{60\ x\ v(ms^{-1})}{\pi\ x\ D}$$

$$N\ (rpm) = \frac{60\ x\ 0.75(ms^{-1})}{\pi\ x\ 0.075} = \ 190.98\, rpm$$

Later on, required acceleration torque can be calculated by Equation 4:

$$Ta = \frac{Jo\ x\ i^2 + J_L}{9.55}\ x\ \frac{N}{t}$$

$$Ta = \frac{0.212}{9.55}\ x\ \frac{190.98}{1.5} \cong 2.83Nm$$

Where the inertia due to rotor is so small compared to the system's inertia hence it is neglected. Total torque of the system (Equation 5):

$$T_T = (T_L + T_a)\ x\ Sf$$

$$T_T\ = (1.05 + 2.83)\ x\ Sf\ \cong 3.88\ x\ Sf\ Nm$$

Required motor power of the mechanism (Equation 6):

$$P\ = \frac{T\ x\ N}{9548}$$

$$P = \frac{2.59\ x\ Sf\ x\ 190.98}{9548} = 0.078\ x\ Sf\ kW$$

According to the calculations, required motor power is found 0.078 kW, except safety factor. Safety factor is considered to be 2 at least (Türk Loydu, 2017), hence the motor power will be 0.15 kW. Nearest electric motor with this power specification in the market is 0.25 kW electric motor, with a safety factor of 3.2.

3.2.3. Drive Shaft Calculations

Since the mini-load AS/RS stacker crane is a 3 degrees-of-freedom mechanism, there have to be at least 3 motors to drive the mechanism. These motors drive the mechanism on three axes individually. As mentioned before, rotational motion of the motors is converted to linear motion by the help of linear motion systems, therefore linear motion of the mechanism is provided. The rotational motion of the motors is transmitted to linear motion systems by the help of the drive shafts.

In this part of the chapter, minimum required diameter of the drive shafts of each motor are calculated using the motor specifications with safety factor 2 (Türk Loydu, 2017). Material used for the shafts is Medium Carbon Steel (SEA 1040, SEA 1050 (Engineering Edge, 2022)) and average values (S_{ys} = 300 MPa and G = 80 GPa) regarding to the material are being used for the calculations. The calculations are performed considering two phenomena: strength and distortion (Ugural, 2004). Equation 7 is based on the strength specification and Equation 8 is based on the distortion specification:

$$\frac{J}{c} = \frac{T}{\tau_{all}} \quad [7]$$

Where:

T = Transmitted torque (Nm),

J = Polar Moment of Inertia ($\pi c^4/2$ for solid shafts) (m^4),

c = Shaft Diameter (m),

τ_{all} = Allowable Stress (Yield /Safety Factor) (N/m^2).

$$\frac{\emptyset_{all}}{L} = \frac{T}{GJ} \quad [8]$$

Where:

T = Transmitted torque (Nm),

J = Polar Moment of Inertia ($\pi c^4/2$ for solid shafts) (m^4),

L=Length of the Shaft (m)

G=Shear Modulus (N/m^2)

ϕ_{all} =Twisting angle (rad)

Design of Shaft 1 and Shaft-2

The first drive shaft of the mechanism, which is called Shaft-1, is used to transmit the power of the Motor-1 to the pulley of the horizontal motion system. In addition, second shaft of the mechanism is used to transmit the power of the Motor-2 to the pulley of the vertical motion system. Since the two motors' size are equal, minimum shaft diameter calculations are the same. The strength calculation for the Shaft-1 and Shaft-2 is according to Equation 7:

$$\frac{\pi}{2}c^3 = \frac{111\ x\ 2}{300(10^6)}$$

$$c = 7.78\ mm$$

According to the strength calculation, minimum diameter of the Shaft-1 and Shaft-2 should be at least 7.78 mm.

In order to calculate the minimum diameter of Shaft-1 and Shaft-2, the distortion calculation for the shafts is performed for 3 different allowable twisting angles (0.5°, 0.2° and 0.1°) using Equation 8:

$$\frac{0.5°}{0.225} = \frac{111}{80(10^9)x\ \pi c^4/4}$$

$$c = 12.28\ mm$$

$$\frac{0.2°}{0.225} = \frac{111}{80(10^9)x\ \pi c^4/4}$$

$$c = 15.45\ mm$$

$$\frac{0.1°}{0.225} = \frac{111}{80(10^9)x\ \pi c^4/4}$$

$$c = 18.39\ mm$$

According to the distortion calculation, minimum diameter of the Shaft-1 and Shaft-2 should be at least 12.28 mm for a twisting angle 0.5°. In addition, greater diameter results in smaller twisting angle.

Design of Shaft 3

The third drive shaft of the mechanism, which is called Shaft-3, is used to transmit the power of the Motor-3 to the pinion gear of the fork mechanism. The strength calculation for the Shaft-3 is according to Equation 7:

$$\frac{\pi}{2}c^3 = \frac{12.4\ x\ 2}{300(10^6)}$$

$$c = \ 3.75\ mm$$

According to the strength calculation, minimum diameter of the Shaft-1 and Shaft-2 should be at least 3.75 mm.

In order to calculate the minimum diameter of Shaft-3, the distortion calculation for the shafts is performed for 3 different allowable twisting angles (0.5°, 0.2° and 0.1°) using Equation 8:

$$\frac{0.5°}{0.275} = \frac{12.4}{80(10^9)x\ \pi c^4/4}$$

$$c = \ 7.48mm$$

$$\frac{0.2°}{0.275} = \frac{12.4}{80(10^9)x\ \pi c^4/4}$$

$$c = \ 9.38\ mm$$

$$\frac{0.1°}{0.275} = \frac{12.4}{80(10^9)x\ \pi c^4/4}$$

$$c = \ 11.18\ mm$$

According to the distortion calculation, minimum diameter of the Shaft-3 should be at least 7.48 mm for a twisting angle 0.5°. In addition, greater diameter results in smaller twisting angle.

3.3. Detailed Design of the Mechanism

In the previous parts of the chapter, firstly a brief explanation of the theory of the stacker crane mechanism is presented. After the theory of the mechanism, common sections of the mechanism are specified and a conceptual design of the mechanism is also created. According to the conceptual design, some critical information for a detailed design is calculated.

In this part of the chapter, some details about the final design of the mechanism are explained. Since this thesis aims to design and analyze a stacker crane in order to be used or marketed by the company, explanations do not involve critical values about the design due to confidential business information. Detailed design of the mechanism is done using Autodesk Inventor Professional 2021.

First of all, the wheel hub of the mechanism is designed. As explained in the previous parts, it consists of grooved wheel, shaft, the wheel hub structure and related elements. For the structural part of the section, bended sheet metals are used and connection is provided by fasteners. Shaft and related parts on the shaft are designed with standard machine elements in shaft design. At the end of the shaft, 2 pillow block bearing are selected from the manufacturer catalogue.

Main Chassis of the mechanism is designed so similar to the one in the conceptual design part. It mainly consists of two I beam profile and two plates at two ends. Additional with the shown in that part, linear motion systems on horizontal axis (on the x-axis as designated in the theory part of the chapter), motor of both horizontal and vertical motions and connection parts are located on the main chassis. Assembly between the wheel hubs and main chassis is provided by fasteners. Wheel hub and main chassis can be seen in Figure 20.

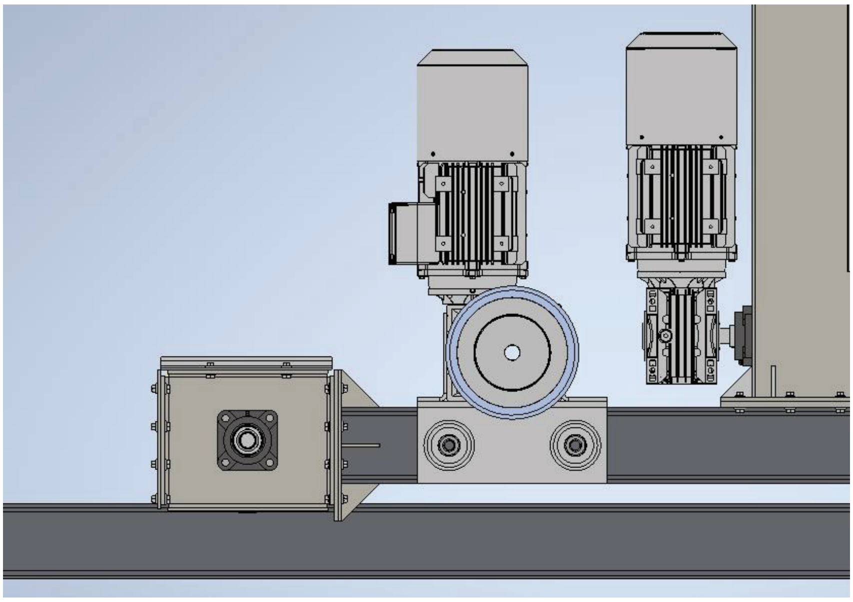

Figure 20. Assembly View of Main Chassis and Wheel Hub

For the mast section of the mechanism, a standard rectangular profile is used with two additional profiles for guidance as mentioned in conceptual design part. Additionally, mast of the mechanism carries the pulleys of the timing belt which is used for the vertical motion (on the y-axis as designated in the theory part of the chapter) and wheeled guidance part for the top guidance of the mechanism. Assembly between main chassis and mast is provided by fasteners (Figure 21).

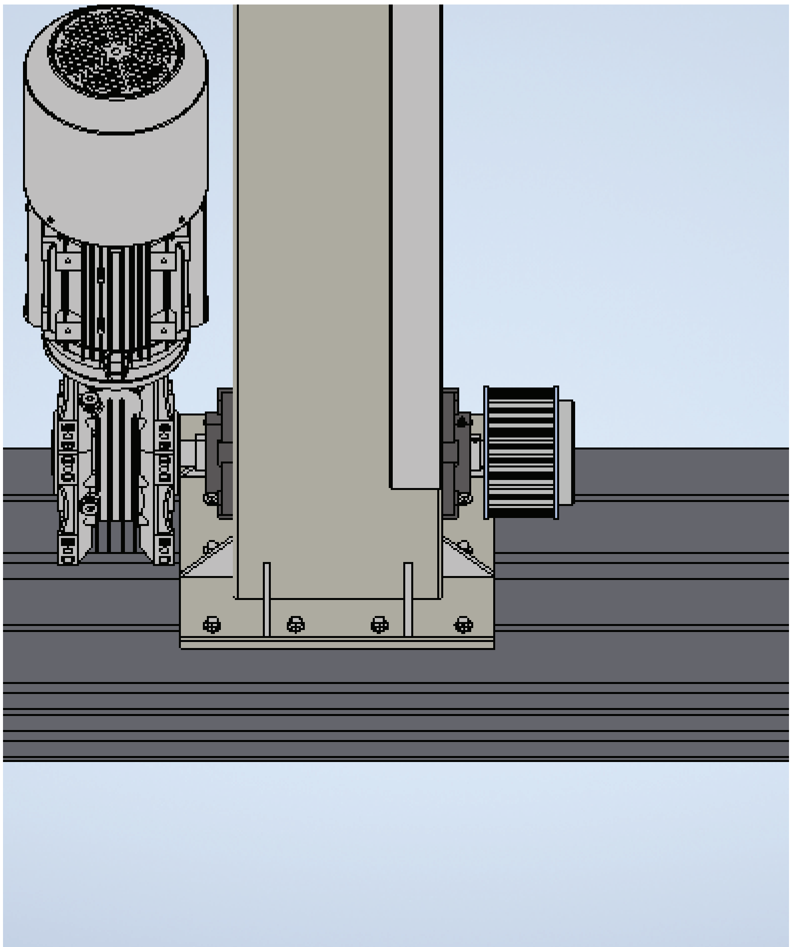

Figure 21. Assembly View of the Mast and Main Chassis

Carriage of the mechanism carries the fork mechanism and is driven by the belt system in vertical axis (y-axis). Connection between carriage and belt is provided by belt clamps (toothed plates) at both ends of the open-loop belt. Moreover, the carriage is guided linearly by the mast of the mechanism. Guidance is provided by wheels which are mounted at the carriage. Connection of the carriage to the mast with wheel can be seen in Figure 22.

At the top of the carriage of the mechanism, a telescopic fork mechanism is mounted in order to store or retrieve the goods of the system. Mechanism consists of three layers. First layer of the mechanism (base layer) is mounted to the carriage and fixed while other two layers are deploying (Figure 24). Due to working principle of the mechanism, top layer deploys two times faster than the middle layer (with respect to base layer).

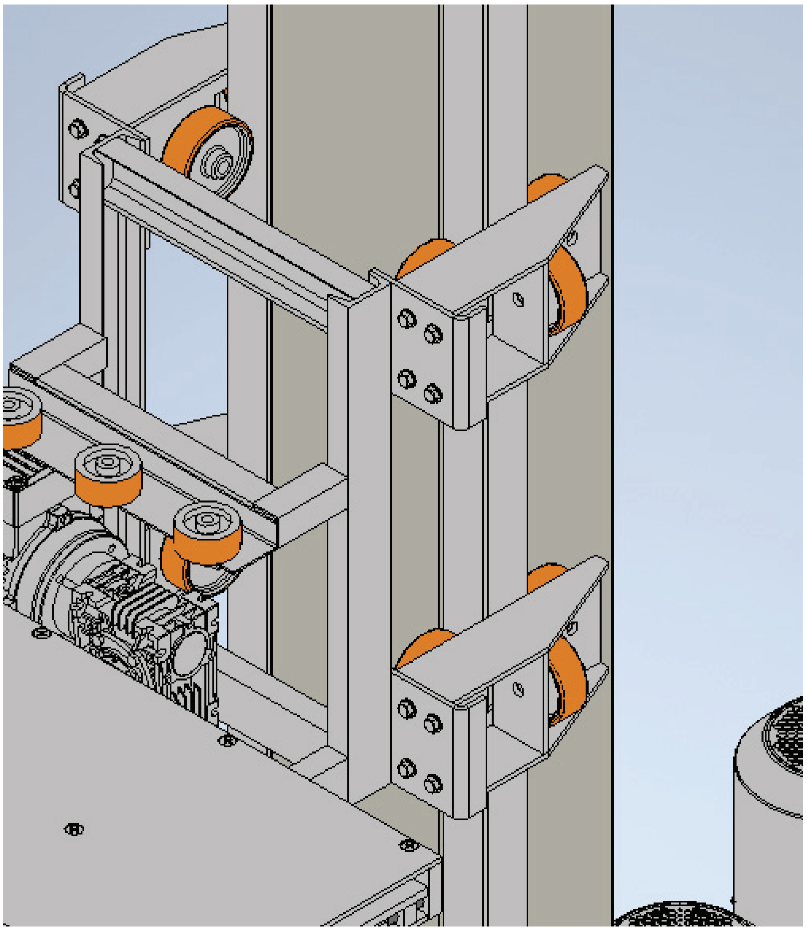

Figure 22. Linear Guidance of the Carriage

Fork mechanism consists of two pinion gears. Middle layer of the mechanism is driven by a pinion gear (Let's say it is gear-1) which is on the shaft of the electric motor (Motor-3) and another pinion gear (Let's say it is gear-2) is mounted on the middle layer and assembled to two parallel rack gears on other two layers. While middle layer moves, gear-2 (blue one in Figure 23) moves with the middle layer while it is assembled to a fixed rack gear on the base layer. This causes gear-2 to turn at the same angular velocity with gear-1. Since gear-2 is also assembled the rack gear on the top layer at the same time, it drives the top layer with the same velocity at which gear-1 drives the middle layer, to the same direction. Although pinion gears turn at the same velocity and drive the rack gears at the same velocity, top layer moves two times faster than the middle layer with respect to fixed layer, this is because top layer is mounted on the middle layer. In other words, top layer is driven by a pinion gear while moving on middle layer.

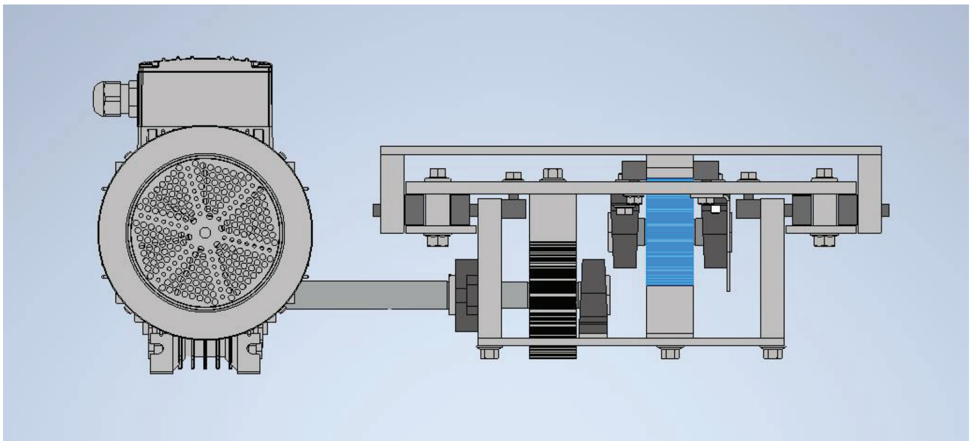

Figure 23. Two Geared Assembly of the Fork Mechanism

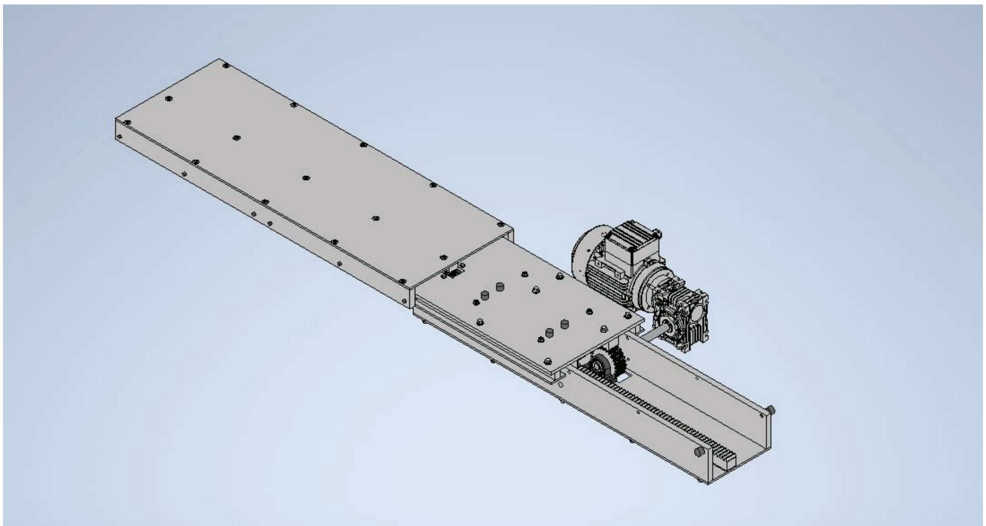

Figure 24. Deployed Form of the Fork Mechanism

CHAPTER 4

FEM ANALYSIS

Finite Element Analysis is a numerical method to calculate the stress and strain in the objects in complicated forms where the traditional methods are not suitable to use. FEA provides the results of materials elastic behavior under external forces. In order to validate the design of the mechanism, Finite Element Analysis method has been used for the critical parts of the mechanism in this section.

There are two most common types of FEM analysis being used in mechanical applications: Static Analysis and Modal Analysis. In this thesis, since the mechanism works slowly (below or equal to 1 m/s), static analysis is performed. Static analysis means that we assumed the mechanism to be analyzed is working time independent. Therefore, the loads and constraints also do not depend on time. (Rusu, 2016)

Since modal analysis is used to specify the vibration characteristics of a part under case of dynamic loading, it is not included and FEM analysis are used for the validation of the designed parts and sections' strength values. Additionally, since the mechanism is guided with additional wheel hub section at the top of the mast of the mechanism, buckling analysis for mast is also not included.

For FEM analysis, Autodesk Inventor Professional 2021 is used. Normally, Autodesk Inventor Professional 2021 is used to design 3D models of the machines. Nevertheless, the program offers stress analysis tool by including FEA software, Autodesk Nastran. (Autodesk, 2020)

First of all, critical parts and sections of the mechanism have been specified which are subjected to maximum load and specified according to the previous failures of the similar mechanisms in the field. After the former FEM analyses performed, designs of each part and section were revised, wherever required. After the revisions, analyses of the revised parts are repeated. In this section of the thesis, the analyses of the latest designs of each part/section are reported.

Sections and parts to be analyzed are specified as follows:

- Rail of the system
- Wheels and Shafts
- Wheel Hub
- Main Chassis
- Carriage
- Fork Mechanism Layers

In a standard FEM analysis process, first the 3D model of the part to be analyzed is created. Later on, the properties of the material of the part are assigned. These properties are the behavior of the material (isotropic or orthotropic), Young's modulus, Poisson's ratio, shear modulus and density. After assigning the material, boundary conditions (loads and constraints) of the part are applied. After all, mesh of the part is created and simulation is done.

During the FEM analysis, 2 critical results of the analysis are important to evaluate: Von Mises Stress Analysis and Displacement Analysis. Von Mises stress is a value which is used for ductile materials' FEM analysis (such as metals) to determine if a material will yield or fracture (Armenakas, 2006). In the FEM analysis, Von Mises stress value is to be compare to the yield limit of the material itself. Maximum Von Mises stress value is expected to be lower than the yield limit of the material in order not to yield. Moreover, exact location of the maximum Von Mises stress value can be seen on the part; therefore, designer can revise the design according to the weakest point. On the other hand, displacement is an important result of the FEM analysis due to effects on geometrical performance on working. Maximum displacement of each point on the part can be seen after the analysis.

In the FEM analysis in the thesis, mesh element used is tetrahedron (3D mesh element). A mesh consists of elements and nodes where elements are the 3D objects and nodes are the points in 3D space which defines the elements. (Autodesk, 2020) Mesh size, node number and element number information of each part is different because of the size of the geometry. Computational power and computational time are the limiting factors for FEM analysis. Therefore, the mesh element size of the analysis kept as smaller as possible in order to get most accurate possible.

The materials used in the project are listed in the Table-2. Addition with the names of the materials, the yield strength values of each one is also listed in MPa. The table is also included the parts in which the materials used. The aim of the analysis is to keep the maximum Von Mises Stress of each part below the yield strength value of related material used. Success criteria of the analysis in this thesis designated as keeping the maximum Von Mises Stress of any part at least 3 times lower than the yield strength of the material (Türk Loydu, 2017), and the maximum displacement of any part below than 1 mm.

Table 2. Material Properties of the Project

Material	Yield Strength	Used Part
Low Carbon Steel (St 37-2, St 37-3)	215-235 MPa (Matmatch, 2022)	• Rail • Wheel Hub • Main Chassis • Carriage • Fork Mechanism Layers
Medium Carbon Steel (SEA 1040, SEA 1050)	290-580 MPa (Engineering Edge, 2022)	• Shaft • Wheel

4.1. Rail

First of all, the rail of the system, on which the whole mechanism moves, has been analyzed in order to validate the selection of the rail beam with right profile and dimensions.

While working, there are only two contact points/lines of the mechanism with rail, which are on the wheels. In order to simulate the analysis of the beam correctly, the load of the mechanism should only be located on these two lines, not on the whole upper surface of the beam. Therefore, two guidance line located on the beam on 3D model to obtain a real-like result.

Load applied on the rail is nothing but the mass of the all system and the rail is constraint at the whole bottom surface because it will be lean on the ground. FEM analysis inputs and outputs results of the rail are given in the Table-3.

Table 3. FEM Analysis Inputs and Results of Rail

Property	Value
Load	6000 N
Element Number	58044
Nodes Number	28743
Von Mises Stress (Maximum)	37.37 MPa
Displacement (Maximum)	0.01891 mm

According to the results of the FEM analysis in Table 3, maximum Von Mises stress that occurs on the rail is about 38 MPa (Figure 19) and maximum displacement (Figure 20) on the rail is about 0.018 mm. According to the properties mentioned in Table-2, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected beam are acceptable.

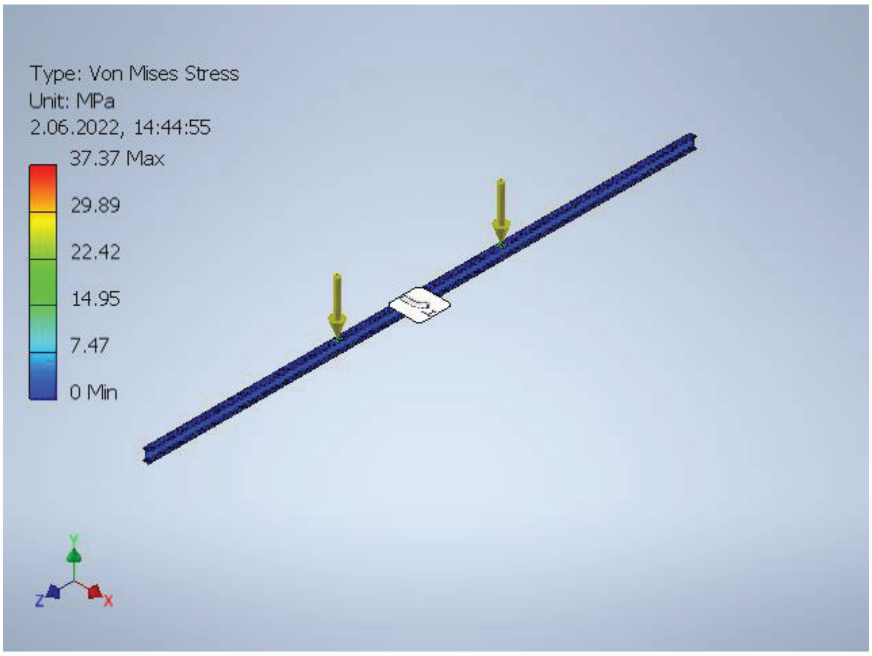

Figure 25. Von Mises Stress Result of the Rail

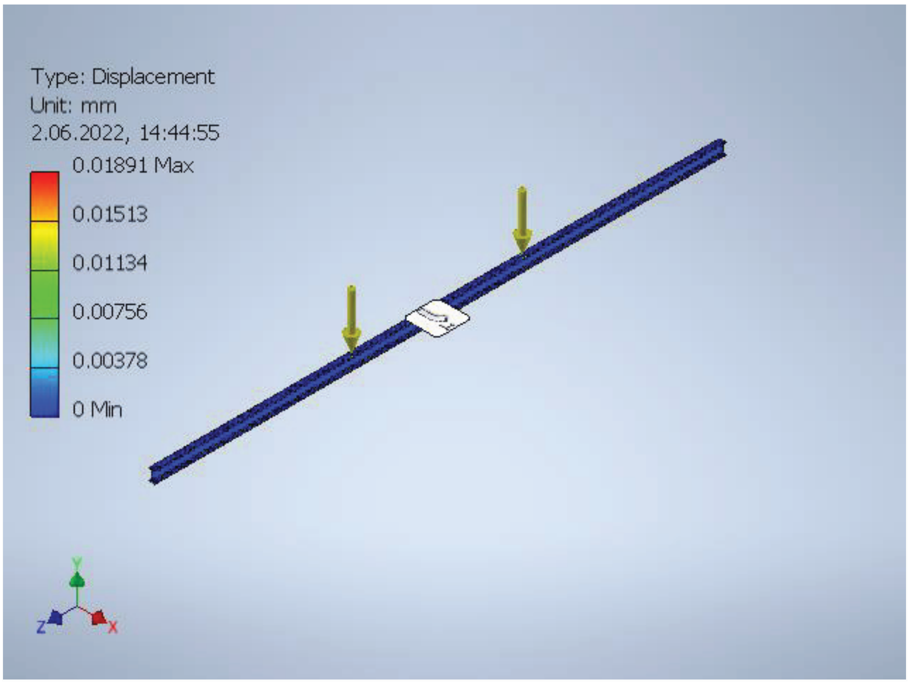

Figure 26. Displacement Result of the Rail

4.2. Wheels and Shafts

The mechanism moves on the rail with its wheels and shafts. Wheels and Shafts on both sides carries the whole mechanism. Load carrying capacity of the shafts and wheels calculated and analyzed using FEM analysis in order to validate the dimensions and material selection.

Since it is an axle shaft and does not transmit torque, only the bending of the shaft is analyzed using FEM analysis. Shaft is fixed on the wheel hub and pillow block bearing from blue surfaces on two sides as shown in Figure 21. The load on the shaft is also shown in the figure. FEM analysis inputs and outputs results of the shaft are given in the Table 4.

Table 4. FEM Analysis Inputs and Results of Shaft

Property	Value
Load	3000 N
Element Number	82172
Nodes Number	120301
Von Mises Stress (Maximum)	7.848 MPa
Displacement (Maximum)	0.0008 mm

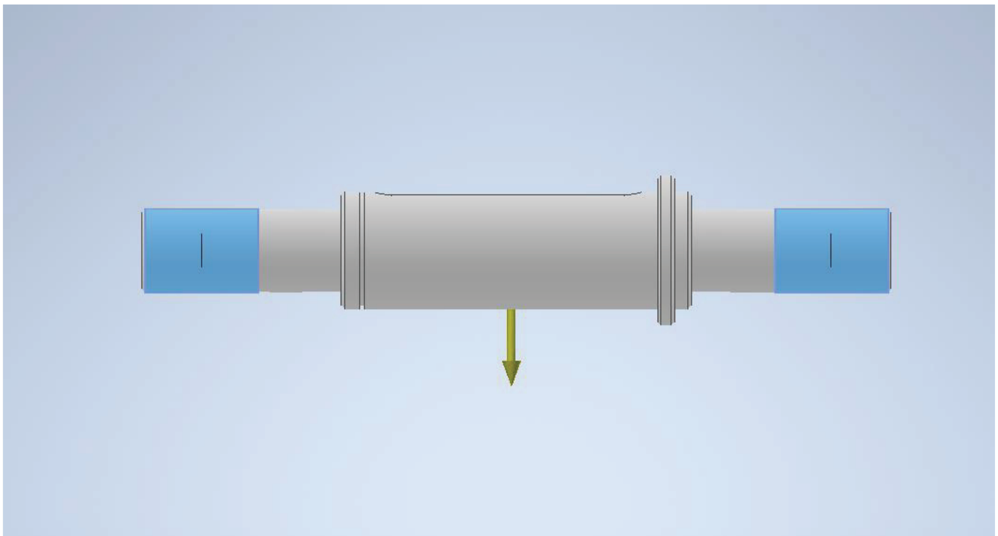

Figure 27. Demonstration of the Boundary Conditions of Shaft

According to the results of the FEM analysis in Table 4, maximum Von Mises stress that occurs on the shaft is about 9 MPa (Figure 22) and maximum displacement (Figure 23) on the shaft is about 0.0008 mm. According to the properties mentioned in Table-2, both maximum stress and maximum displacement are allowable, therefore, the dimensions and selected material are acceptable for shaft.

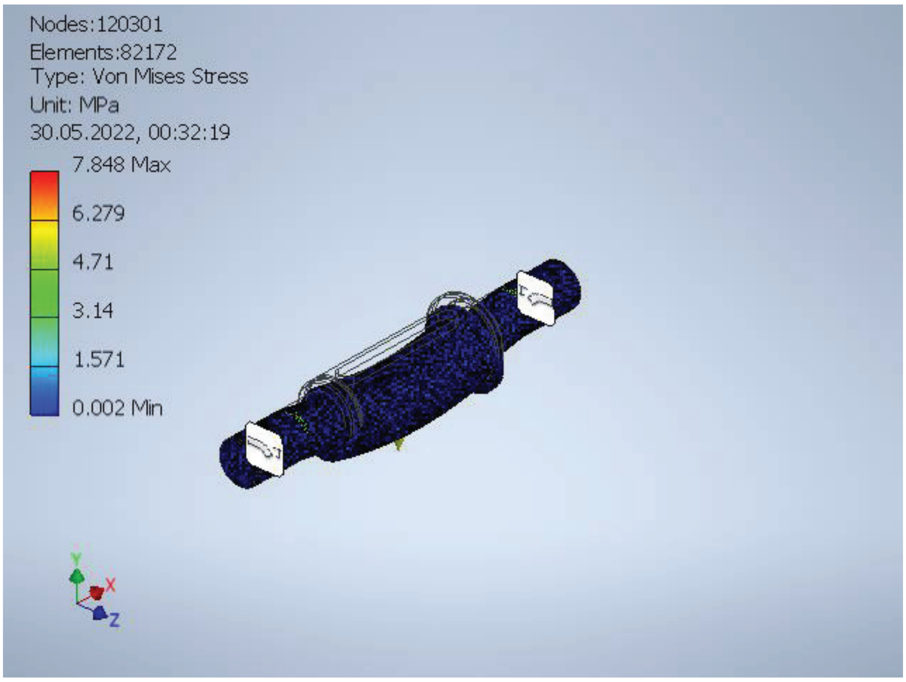

Figure 28. Von Mises Stress Result of the Shaft

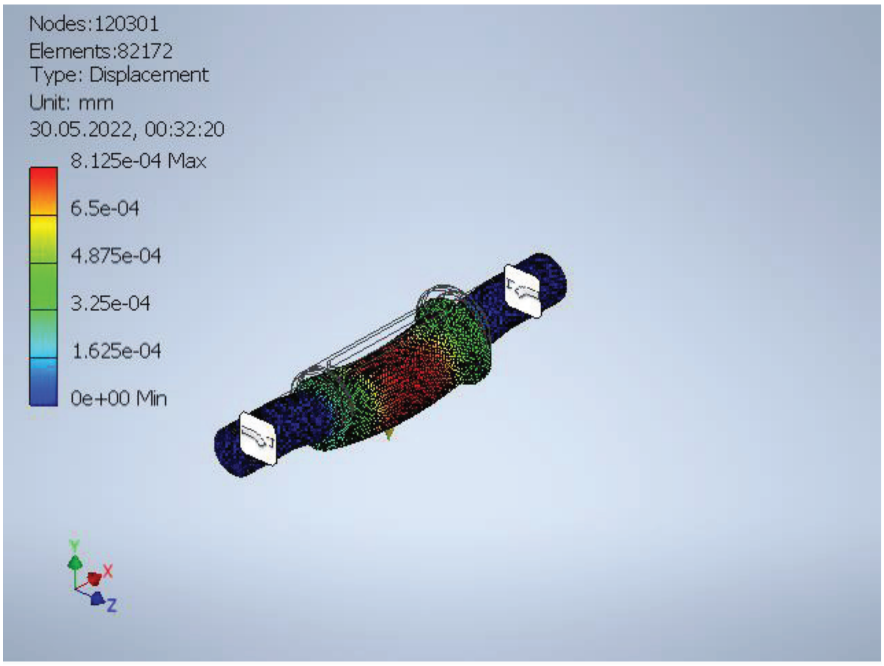

Figure 29. Displacement Result of the Shaft

Afterwards, FEM analysis of the wheel is performed. Wheel is fixed from inner hole and load on the wheel is applied on the outer surface from rail through inner hole as shown in Figure 24. FEM analysis inputs and outputs results of the wheel are given in the Table 5.

Table 5. FEM Analysis Inputs and Results of Wheel

Property	Value
Load	3000 N
Element Number	115719
Nodes Number	173331
Von Mises Stress (Maximum)	0.3985 MPa
Displacement (Maximum)	0.00006 mm

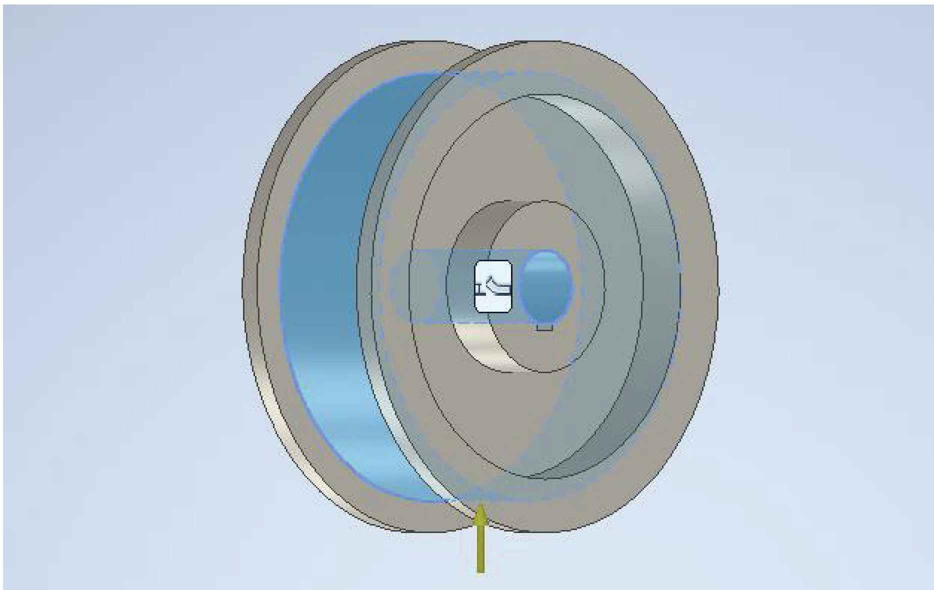

Figure 30. Demonstration of the Boundary Conditions of the Wheel

According to the results of the FEM analysis in Table 5, maximum Von Mises stress that occurs on the wheel is about 0.4 MPa (Figure 25) and maximum displacement (Figure 26) on the wheel is about 0.00006 mm. According to the properties mentioned in Table-2, both maximum stress and maximum displacement are allowable, therefore, the dimensions and selected material are acceptable for wheel.

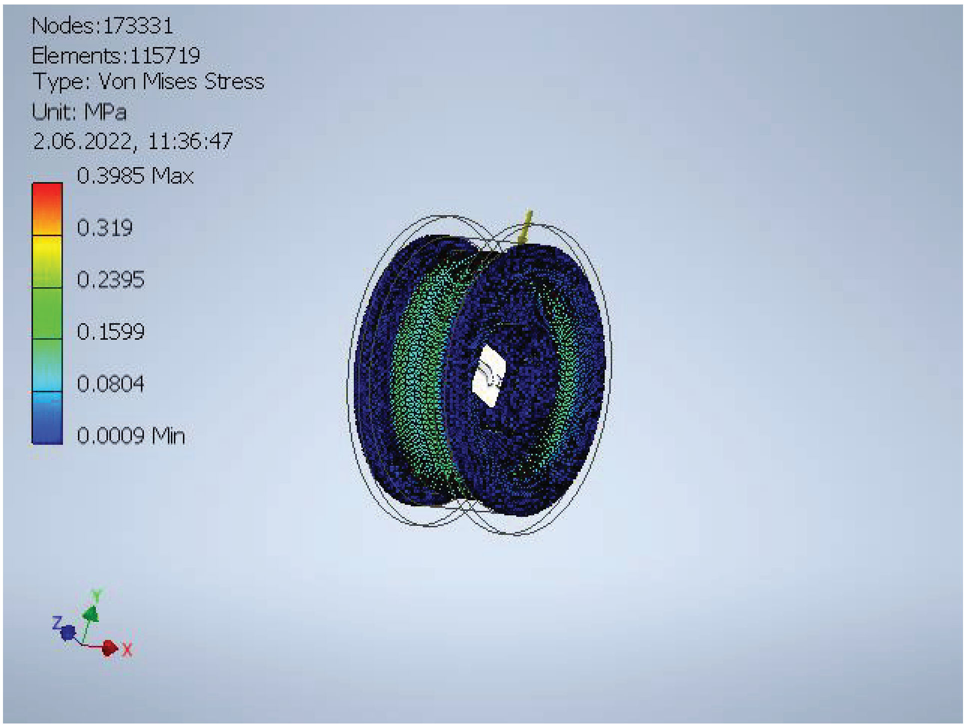

Figure 31. Von Mises Stress Result of the Wheel

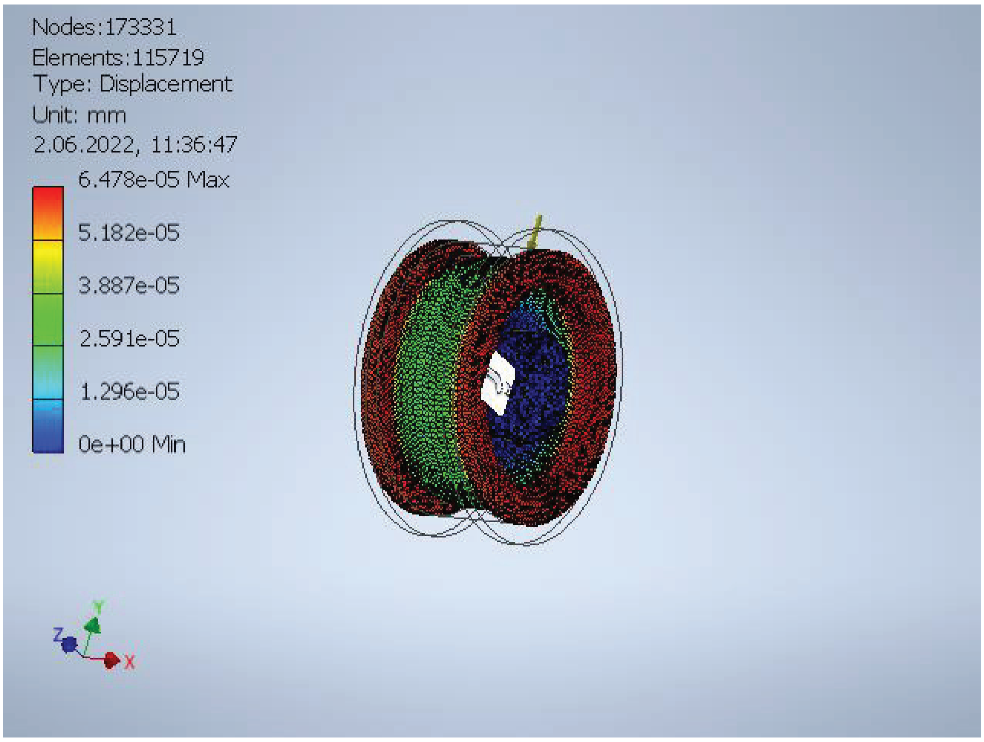

Figure 32. Displacement Result of the Wheel

4.3. Wheel Hub

The mechanism moves on the rail with its wheels. Wheels are mounted on the wheel hubs on both sides via shafts. Wheel hub consists of sheet plates and analyzed in order to validate the selection of the dimensions of sheet plates.

While working, wheels are carried by pillow block bearings mounted on the hubs. Load on the wheel hub applied on the bolt holes of pillow block bearings and wheel hub is fixed from bolt holes to Main Chassis, on the blue surface in Figure 27. FEM analysis inputs and outputs results of the wheel hub are given in the Table 6.

Table 6. FEM Analysis Inputs and Results of Wheel Hub

Property	**Value**
Load	3000 N
Element Number	97104
Nodes Number	195055
Von Mises Stress (Maximum)	49.73 MPa
Displacement (Maximum)	0.05445 mm

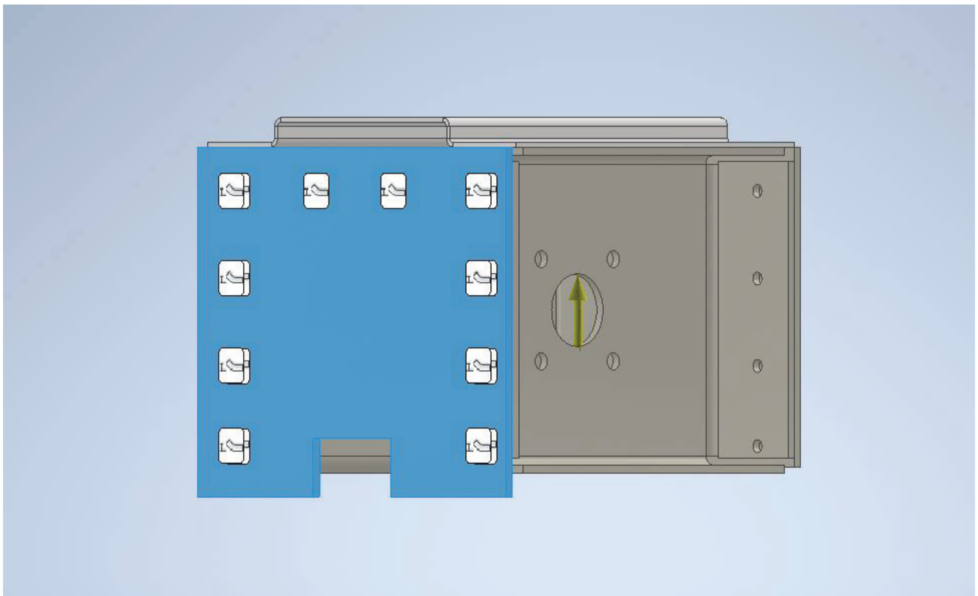

Figure 33. Demonstration of the Boundary Conditions of the Wheel Hub

According to the results of the FEM analysis in Table 6, maximum Von Mises stress that occurs on the hub is about 50 MPa (Figure 28) and maximum displacement (Figure 29) on the hub plate is about 0.054 mm. According to the properties mentioned in Table-2, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates are acceptable.

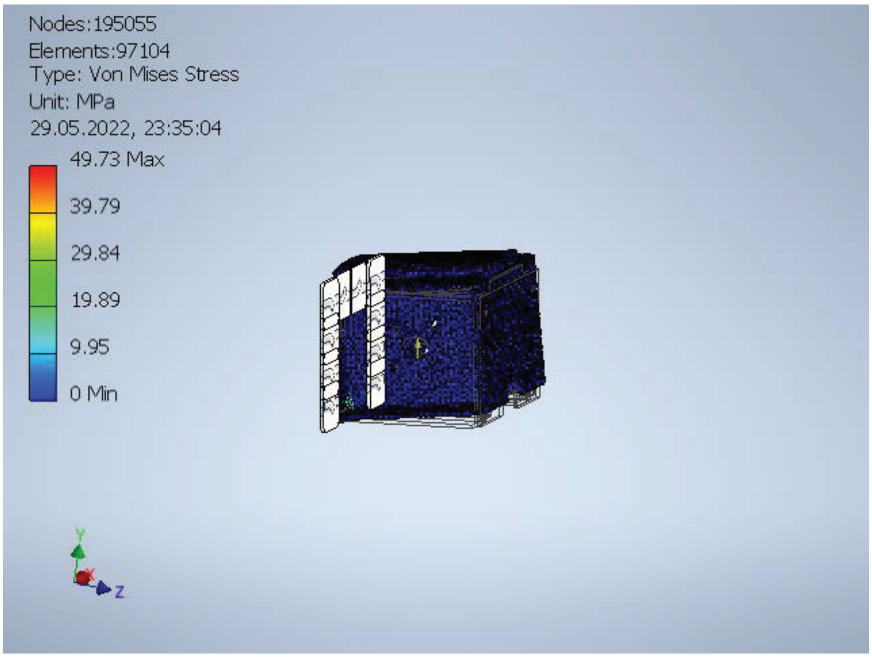

Figure 34. Von Mises Stress Result of the Wheel Hub

Figure 35. Displacement Result of the Wheel Hub

4.4. Main Chassis

The main chassis of the mechanism is mounted between the two wheels hubs on both sides. It carries the mass structure and the sections on the structure. It consists of 2 profiles and 2 plates at both ends, and analyzed in order to validate the design. Main Chassis is fixed to the wheel hubs at both ends via bolt holes and the load on the chassis is applied on the plate which is mounted at the middle of the profiles as shown in the Figure 30.

According to the results of the FEM analysis in Table 7, maximum Von Mises stress that occurs on the main chassis is about 63.5 MPa (Figure 31) and maximum displacement (Figure 32) on the hub plate is about 0.13 mm. According to the properties mentioned in Table-7, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates and beams are acceptable.

Table 7. FEM Analysis Inputs and Results of Main Chassis

Property	**Value**
Load	4000 N
Element Number	220649
Nodes Number	375216
Von Mises Stress (Maximum)	63.51 MPa
Displacement (Maximum)	0.1307 mm

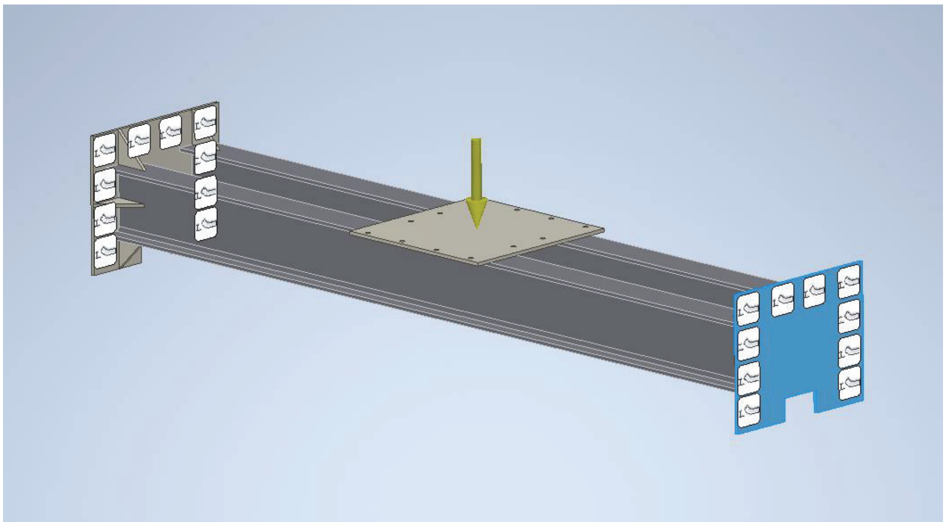

Figure 36. Boundary Conditions of the Main Chassis

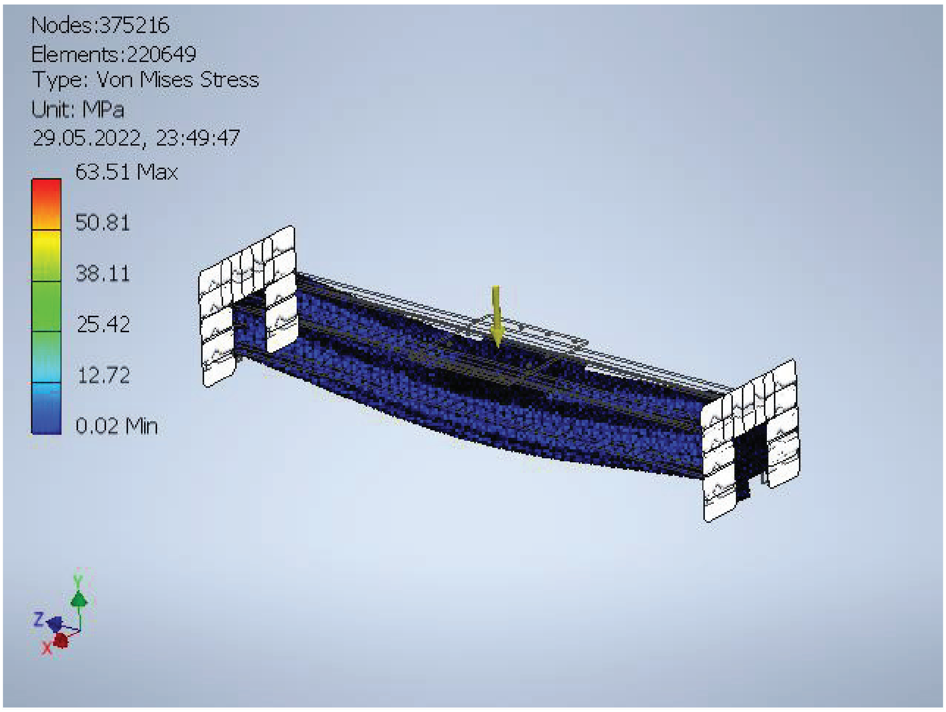

Figure 37. Von Mises Stress Result of the Main Chassis

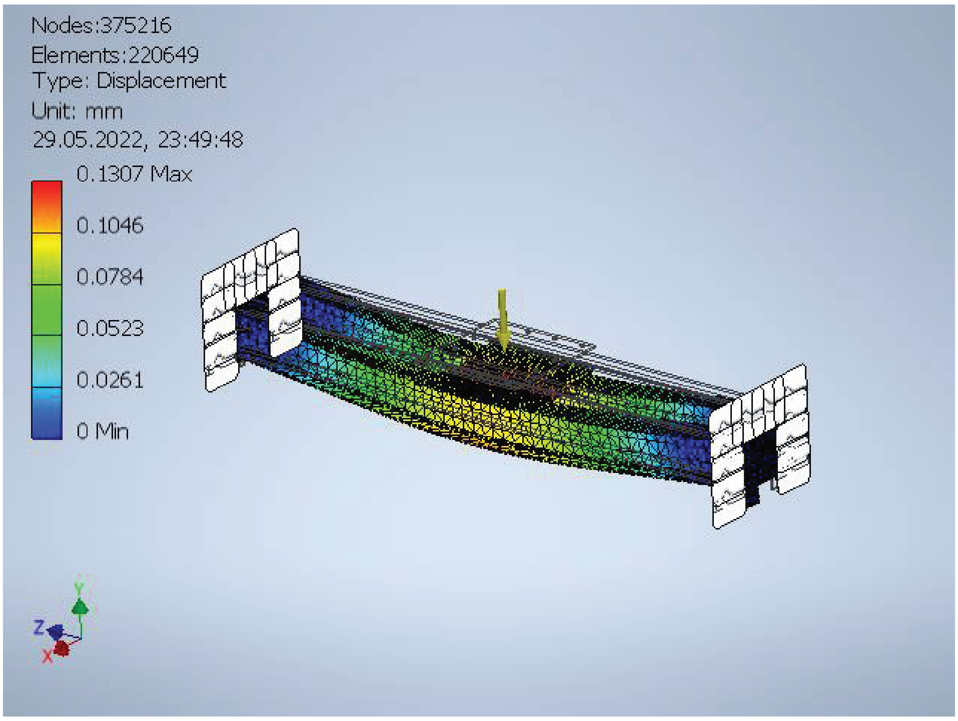

Figure 38. Displacement Result of the Main Chassis

4.5. Carriage

The carriage of the mechanism is mounted on the mass structure and carries the fork mechanism and the workload of the system. It consists of welded profiles and analyzed in order to validate the design. The carriage mounted on the mast structure via wheel sections which provide the linear movement along the mast. Vertical movement of the carriage is provided by a timing belt and pulley system, and connection between the carriage and belt is provided by clamping plates. In this analysis of the mechanism, clamping plates are not included. The assembly between the carriage and the mast is shown only by bolted plates, and the load on the carriage is also shown in Figure 39.

According to the results of the FEM analysis in Table 8, maximum Von Mises stress that occurs on the main chassis is about 71 MPa (Figure 34) and maximum displacement (Figure 35) on the hub plate is about 0.59 mm. According to the properties mentioned in Table-2, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates and beams are acceptable.

Table 8. FEM Analysis Inputs and Results of Carriage

Property	**Value**
Load	1500 N
Element Number	540644
Nodes Number	871882
Von Mises Stress (Maximum)	71.04 MPa
Displacement (Maximum)	0.5959 mm

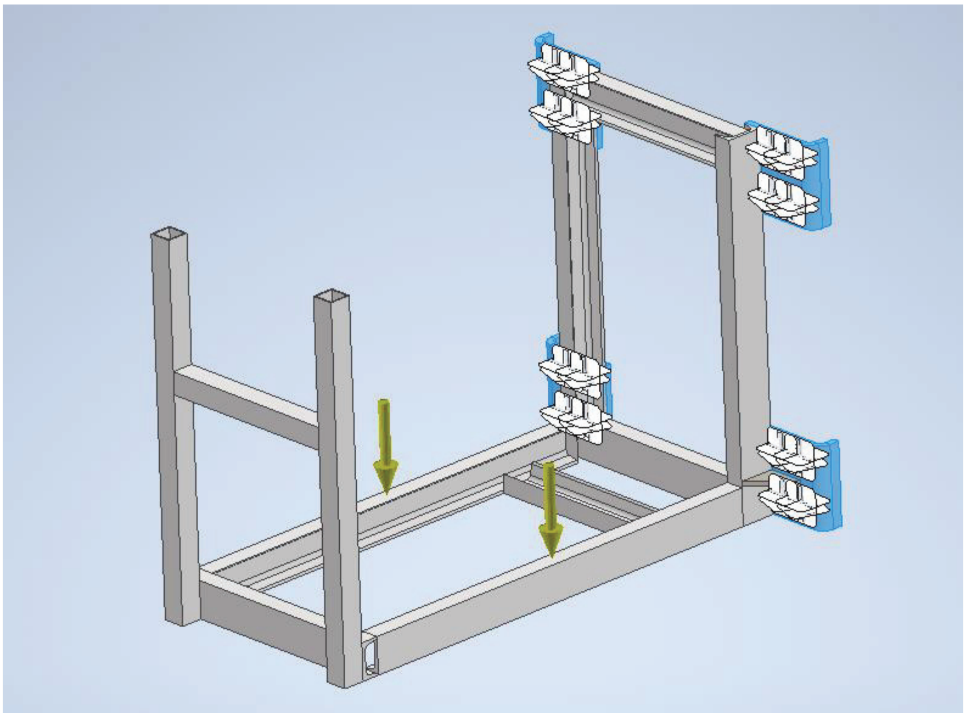

Figure 39. Boundary Conditions of the Carriage

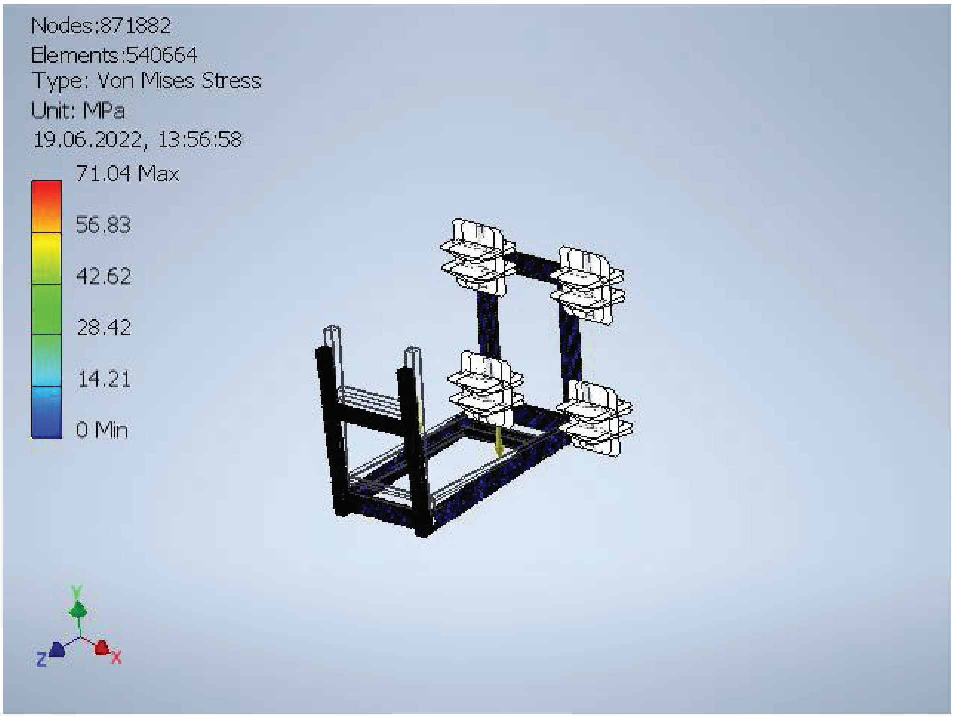

Figure 40. Von Mises Stress Result of the Carriage

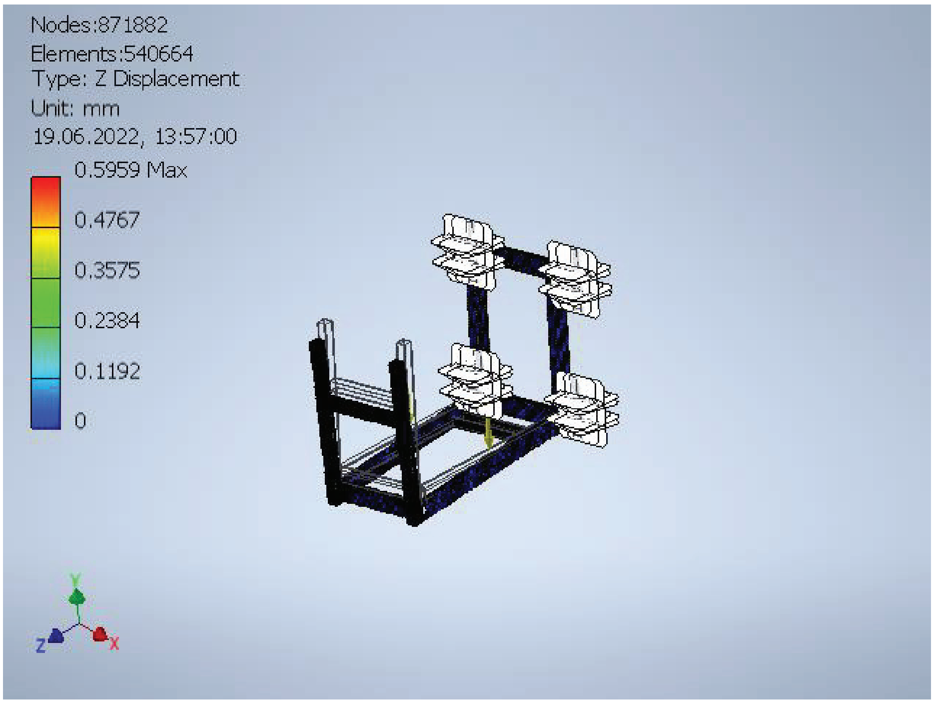

Figure 41. Displacement Result of the Carriage

4.6. Fork Mechanism

Fork mechanism of the system is a one-degree-of-freedom telescopic mechanism consist of 3 layers. Base layer of the mechanism is mounted and fixed on the carriage; other 2 layers are deploying as explained previously in the thesis. The structure of the mechanism consists of several parts: layer plates, rack and pinion systems, drive shaft of the mechanism, cam follower bearings and connection parts. In this part of the section, 3 layers of the mechanism (Base, top layer and middle layer) are analyzed in assembly form.

The layers of the mechanism move on the cam follower bearings mounted at two sides of the base and top layers. Middle layer has two channel-like design at both sides and cam follower bearings move inside the channel-like structures. At the most opened form of the mechanism, at which the most stress occurs, top and base layers of the mechanism subject to load on their bearing holes, where the shaft of the cam follower bearings are mounted (half of the total number of bearings which are at the non-opened side). On the other hand, load is applied to the holes at the opened side for base layer. At the top layer, load is applied on the top surface of the layer. And for the middle layer, load is applied on the surfaces of channel like structure.

In Figure 42, top layer of the fork mechanism is shown with fixed holes (with white marks) and load applied surface. Additionally, in Table 9, FEM analysis inputs and results for top layer are presented.

Table 9. FEM Analysis inputs and Results for Fork Mechanism Top Layer

Property	**Value**
Load	500 N
Element Number	60394
Nodes Number	119390
Von Mises Stress (Maximum)	24.78 MPa
Displacement (Maximum)	0.2179 mm

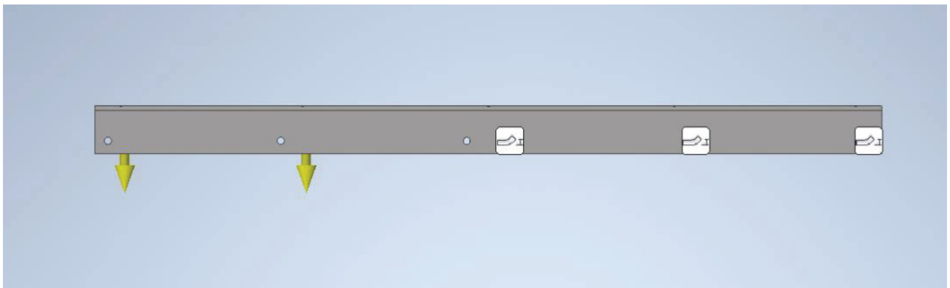

Figure 42. Boundary Conditions of the Fork Mechanism Top Plate

According to the results of the FEM analysis in Table 9, maximum Von Mises stress that occurs on the top layer is about 25 MPa (Figure 43) and maximum displacement is about 0.21 mm (Figure 44). According to the properties mentioned in Table-9, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates and beams are acceptable.

Figure 43. Von Mises Stress Result of the Fork Mechanism-Top Layer

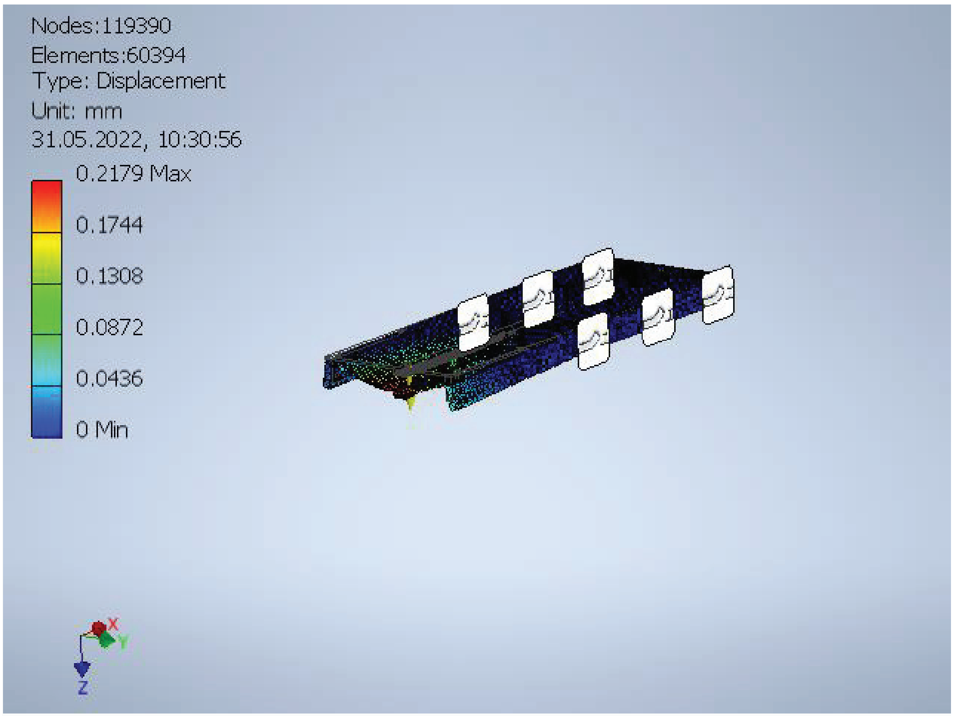

Figure 44. Displacement Result of the Fork Mechanism-Top Layer

In Figure 45, middle layer of the fork mechanism is shown with fixed holes (with white marks) and load applied surface. Additionally, in Table 10, FEM analysis inputs and results for top layer are presented.

Table 10. FEM Analysis Inputs and Results for Fork Mechanism Middle Layer

Property	**Value**
Load	650 N
Element Number	67084
Nodes Number	120369
Von Mises Stress (Maximum)	37.36 MPa
Displacement (Maximum)	0.2134 mm

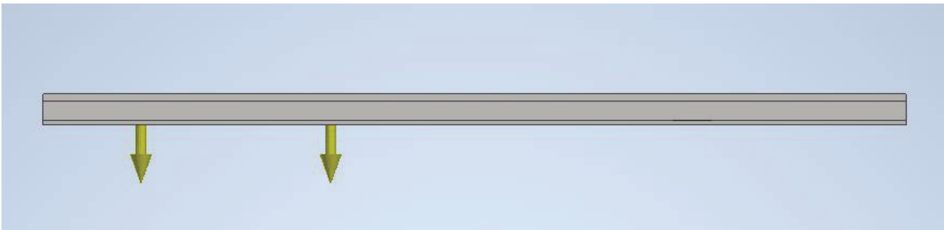

Figure 45. Boundary Conditions of the Fork Mechanism Middle Plate

According to the results of the FEM analysis in Table 10, maximum Von Mises stress that occurs on the middle layer is about 38 MPa (Figure 46) and maximum displacement is about 0.21 mm (Figure 47). According to the properties mentioned in Table-10, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates and beams are acceptable.

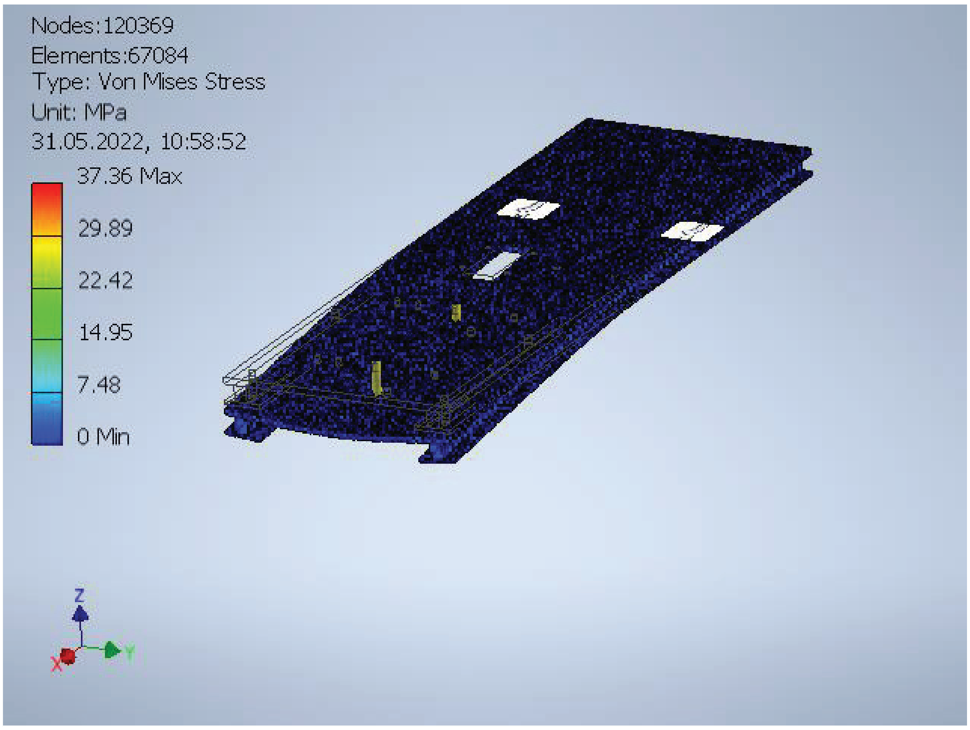

Figure 46. Von Mises Stress Result of the Fork Mechanism-Middle Layer

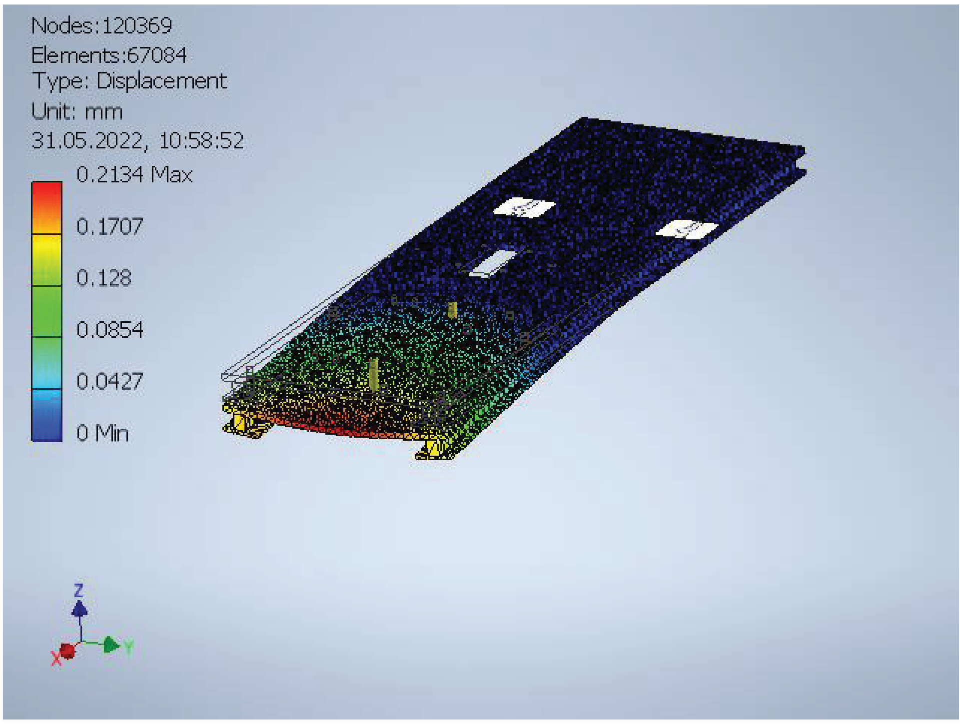

Figure 47. Displacement Result of the Fork Mechanism-Middle Layer

In Figure 48, base layer of the fork mechanism is shown. Base layer is fixed to the carriage with its bottom surface and load is applied to the holes for cam followers. Additionally, in Table 11, FEM analysis inputs and results for top layer are presented.

Table 11. FEM Analysis Inputs and Results for Fork Mechanism Base Layer

Property	Value
Load	750 N
Element Number	37853
Nodes Number	73102
Von Mises Stress (Maximum)	3.169 MPa
Displacement (Maximum)	0.006 mm

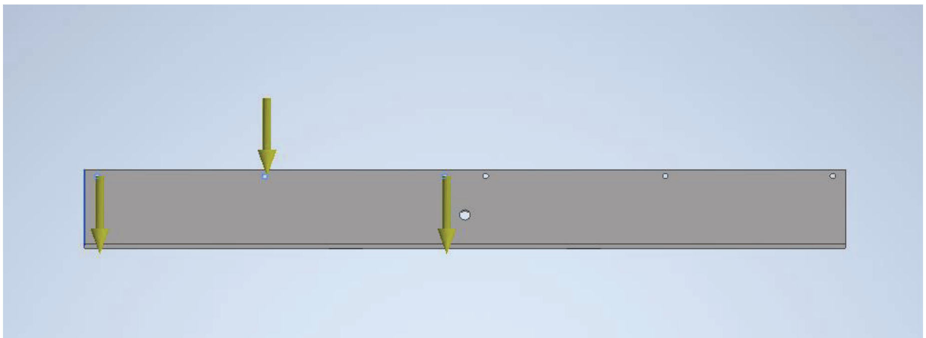

Figure 48. Boundary Conditions of the Fork Mechanism Top Plate

According to the results of the FEM analysis in Table 11, maximum Von Mises stress that occurs on the middle layer is about 3 MPa (Figure 49) and maximum displacement is about 0.006 mm (Figure 50). According to the properties mentioned in Table-11, both maximum stress and maximum displacement are allowable, therefore, the dimensions of the selected plates and beams are acceptable.

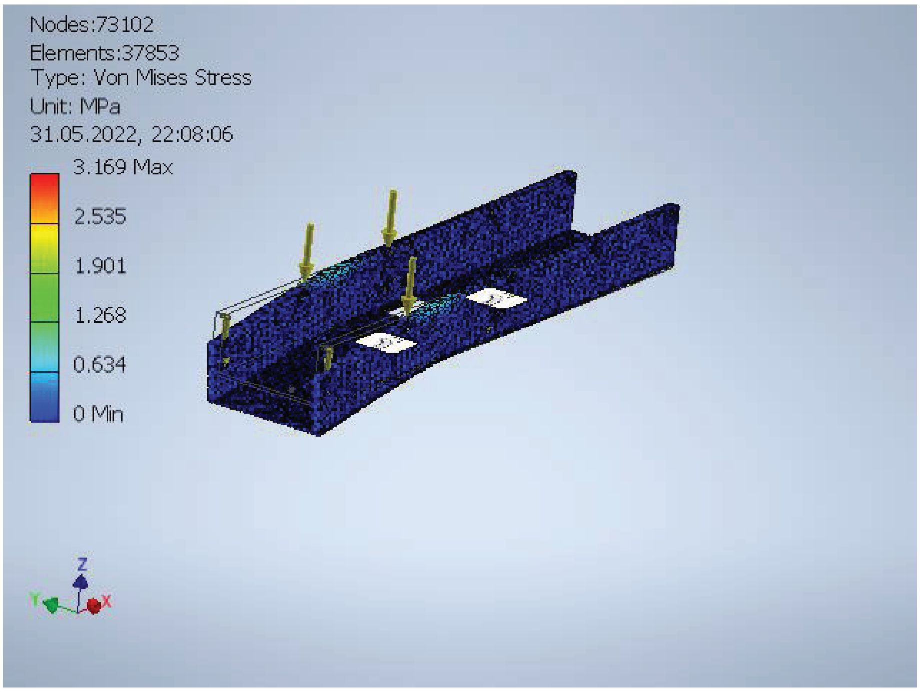

Figure 49. Von Mises Stress Result of the Fork Mechanism-Base

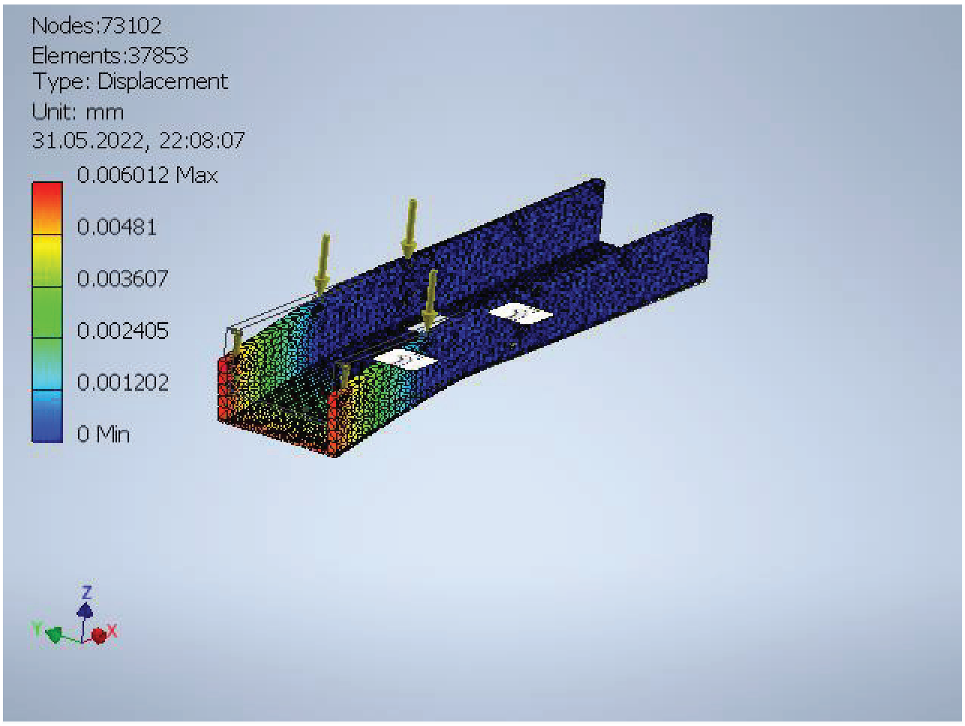

Figure 50. Displacement Result of the Fork Mechanism-Base

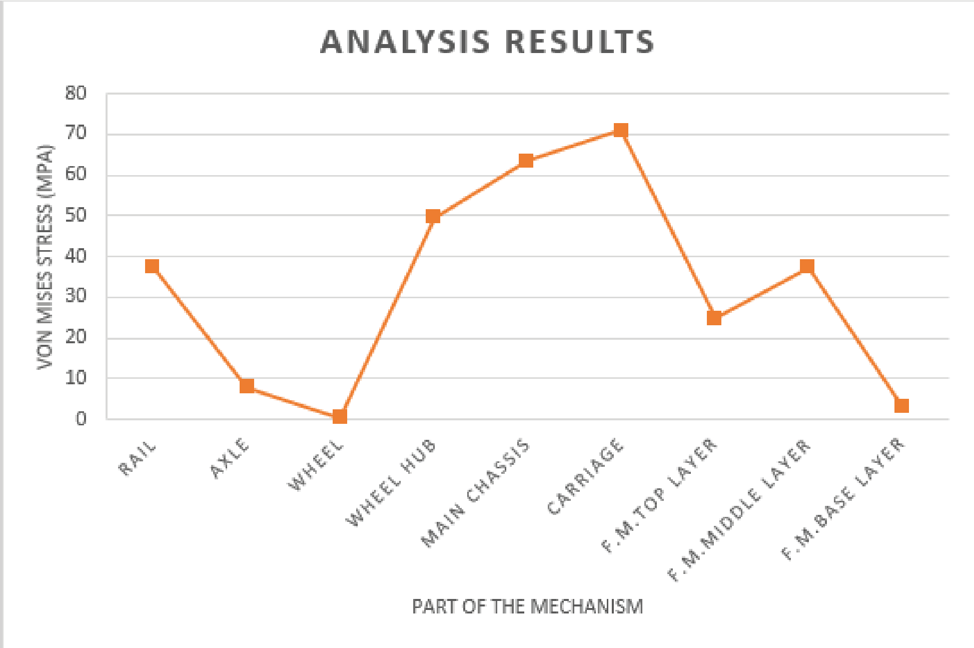

Figure 51. Analysis Results Chart

In Figure 51, all the final Von Mises Stress results of the Finite Element Analysis is presented in a line chart. Due to geometry of the mechanism; wheel hub, main chassis and carriage are subjected to the most stress. Nevertheless, according to the success criteria designated at the beginning of the chapter, all the results are acceptable and safety factor calculations verify the design.

CHAPTER 5

PROTOTYPING AND EXPERIMENTS

Stacker crane mechanism consists of several parts as mentioned in the previous chapters. Parts of the mechanism are considered to manufactured separately and assembly in the designated area. Therefore, an installation procedure is developed for the assembly of the stacker crane mechanism:

1. Installation of the rail of the mechanism.
2. Manufacturing and assembly of the main chassis and wheel hubs of the mechanism.
3. Testing of the mechanism in horizontal axis with the installed parts.
4. Installation of the top guidance rail of the mechanism.
5. Assembly of the mast of the mechanism to the pre-installed parts.
6. Testing of the mechanism in horizontal axis with the addition of the mast to the mechanism.
7. Assembly of the carriage of the mechanism to the mast.
8. Testing of the mechanism in both horizontal and vertical axes with the installed parts.
9. Assembly of the fork mechanism to the carriage.
10. Testing of the mechanism in three axes with the addition of rack structure of the AS/RS.

In this chapter of the thesis, prototyping and experiments of the telescopic fork mechanism is presented. Other parts of the stacker crane mechanism are also prepared for manufacturing, however, according to the annual work schedule of the company, only the telescopic fork mechanism is manufactured.

For the manufacturing of the fork mechanism, first of all, standard machine elements are selected and provided (Figure 52). These are rack and pinion gears (Figure 52(a)), pillow block bearings (Figure 52(b, c)), cam followers (Figure 52(a)) and fasteners. Rack and pinion gears and shafts of the mechanism are machined using lathe and CNC milling machine according to the design (Figure 53). Structural parts of the mechanism produced using laser cut (Figure 54) and machined using CNC milling machine (Figure 55).

Figure 52. Standard Machine Elements Used in Fork Mechanism
(Source: SKF, 2022; Dogus Kalıp, 2022)

Figure 53. Machined Rack Gear

Figure 54. Laser Cut Steel Parts

Figure 55. Machines Parts

Produced parts are assembly according to design and performed in order. Firstly, the base layer of the mechanism is assembled. It consists of structural elements, shaft from the electric motor, pillow block bearings, cam followers and rack and pinion gears (Figure 56 (left)). Later on, the middle layer of the mechanism is assembled. It consists of a pinion gear to be matched the rack gears on the base and top layers, pillow block bearings, rack gear to be driven by the pinion gear of the base layer and structural elements (Figure 56 (Right)).

Figure 56. Base and Middle Layer Assemblies of the Fork Mechanism

Later on, the top layer of the mechanism is assembled. Top layer consists of structural parts, cam followers and rack gear (Figure 57). After the assembly of the layers separately, they are joined together (Figure 58).

Figure 57. Top Layer Assembly of the Fork Mechanism

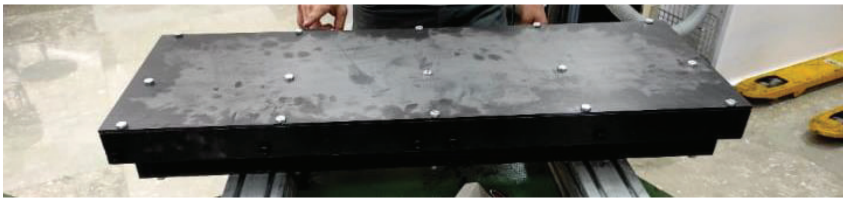

Figure 58. Assembly of the Fork Mechanism

After the assembly of the fork mechanism, experiments regarding the deflection of the most-end point of the mechanism are performed. Purpose of the experiments is to determine the maximum possible deflection of the most-end point of the fork mechanism at the deployed form. For performing the experiments; fork mechanism assembly, workload (50 kg), platform for fork mechanism, and measuring equipment are needed. Methodology of the experiments is listed below:

- Fix the platform to the ground.
- Fix the fork mechanism assembly on the platform.
- Place the workload on the top of the fork mechanism
- Deploy the fork mechanism with workload
- Measure the most-end point of the mechanism from the ground.
- Repeat the process.
- Compare the result

Since cam followers are used as linear guidance on the mechanism, there are channel-like structure on the middle layer where cam followers are placed in as mentioned in previous chapters. These structures are produced with tolerance with respect to outer diameter of cam followers in order not to be stuck. This geometry results in deflection when mechanism is in deployed form as it is expected.

For the experiments of the prototype, fork mechanism is subjected to the prespecified workload of the system. Mechanism is deployed 10 times and deflection at the end point is measured each time as indicated in the experimental protocol given previously on the chapter. (Figure 59, Figure 60).

Figure 59. Experiments of the Fork Mechanism 1

Figure 60. Experiments of the Fork Mechanism 2

The results of the experiments are listed in the Table 12 and also in Figure 61 in a line chart. The channel-like structures of the middle layer are produced 1.5 mm greater than the outer diameter of the cam followers for the prototype. This geometry results in deflection when mechanism is in deployed form and it is calculated with a simple triangle. The result of the theoretical calculation is 9.84. Additionally, according to the results of FEM analysis, the sum of the displacements of the layers is 0.49 mm. Total deflection expected according to the theory is 10.33.

The results of the experiments have a mean value of 11.4 mm and differ from theoretical results by 1.07 mm. Standard deviation of the results is 0.49 and the coefficient of variation is 0.043. It is seen that the data which collected by the experiments are gathered around the mean value. Therefore, the experiment can be considered as reliable due to low variation. The different between theory and experiments may cause by several reasons. These are:

- Manufacturing defects
- Measuring errors
- Possible deflections on platform

Table 12. Experimental Results

Experiments	Deflection (mm)
1	12
2	11
3	11
4	12
5	11
6	12
7	11
8	11
9	12
10	11
Mean	11.4
Standard Deviation	0.49
Coefficient of Variation	0.043

Figure 61. Experimental Results Chart

CHAPTER 6

CONCLUSION

The objection of the thesis was to design and analyze a 3-degrees-of-freedom stacker crane mechanism for Mini Load Automated Storage and Retrieval Systems. This study first introduced an inclusive literature review on AS/RSs. Several types of AS/RSs and stacker cranes were examined, advantages and disadvantages of the systems and mechanisms were studied and the differences between them were introduced. Literature review provided an insight about the current status, deficiencies, development processes and probable future studies of the products available in the market. However, there are limited studies on stacker crane design and analysis. One of the most important outputs of this study is that it will contribute to fill the gap in the literature about design and FEM analysis for a stacker crane for AS/RSs.

In the design chapter of the thesis, theory of the stacker crane mechanism was introduced. A conceptual design was created and, elements and sub-sections of the desired mechanism were presented with the conceptual design of the mechanism. After the conceptual design, detailed motor and carrier axle calculations were made. For a stacker crane mechanism, a detailed motor calculations have not been made in the literature before. Another contribution to the literature with this thesis is to introduce a motor calculation methodology for a stacker crane mechanism.

In the next chapter of the thesis, the detailed design of the mechanism was analyzed using Finite Element Analysis methods by a Computer Aided Engineering software. In this way, safety factor of each part/section was calculated and the reliability of the proposed mechanism was confirmed. As a result of analysis, weak and over-safe sections of the parts and sections of the mechanism were revised. Some over-safe parts of the mechanism cannot be revised due to geometrical reasons.

At the prototyping chapter of the thesis, the manufacturing studies of the fork mechanism were presented. Manufacturing stages were introduced and experiments of the mechanism were performed. The operability of the design confirmed using the experiments.

On the other hand, there are some limitations in this thesis study. This study was performed only for a certain mass value. A parametric design was not performed to be used for different mass values. It is required to repeat the study using the methodology given in the thesis for different values. In the analysis part, the vibration on the mechanism was ignored as the mechanism operates at low speeds. This issue needs to be taken into account in case of use the mechanism at higher speeds. On the other hand, only the fork mechanism of the whole design was prototyped. For other sections of the mechanism, prototyping was not carried out within the scope of the company's annual planning calendar during the thesis period. Additionally, the linear guidance systems of the mechanism were design as wheel guidance in order to reduce the cost. Higher-priced linear systems can be used for a more precise operation which will facilitate the control stage of the mechanism. In addition, with a higher budget, lower mass materials can be used where appropriate. (As far as strength properties allows.)

As a result of the thesis, a new and novel mechanism design and a ready-to-sale product were presented. In addition, a design methodology for a new and different, or similar mechanism designs was revealed and required design stages for a reliable product were demonstrated. Unlike the competing products available on the market, the reliability of the design has been supported by an academic publication. In the future studies, designs for different sizes can be performed by using the methodology proposed in the thesis. For example, this thesis presented a stacker crane design for Mini-Load Automated Storage and Retrieval Systems. With this methodology, a design study for a stacker crane to be used in Unit-Load or other types of Automated Storage and Retrieval Systems can be carried out. In addition to all these, a new work area has been created for the company with this study and the company's product portfolio has expended.

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