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2023 - 2024
GRADUATION PROJECT

NATIONAL ENGINEERING DEGREE

SPECIALTY : Electromechanics

Design and Development of an Electric Industrial Heavy-Duty Transfer Cart

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شركة فسفاط قفصة
Compagnie des Phosphates de Gafsa

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Dedication

This graduation project is dedicated to my beloved parents, whose unwavering support, encouragement, and love have been my guiding light throughout my educational journey.

To my siblings, thank you for your endless patience, understanding, and support. Your encouragement has been invaluable, and this accomplishment is as much yours as it is mine.

To my friends and mentors, who have been there through the highs and lows, offering advice, support, and companionship. Your contributions have been essential to my success.

And finally, to you, dear readers

Acknowledgment

I thank God for giving me the patience, confidence, and strength to accomplish and complete this work.

I take this opportunity to express my deep gratitude and sincere thanks to all those who have shown support, availability, and collaboration in realizing this work.

I especially thank:

*Mr. Salem Debaoui, Mr. Yassine Boussaa , Mme. Yossra Messaoudi and Mr. Talel ******

For their professional rigor, availability, attentive listening, and wise advice, which have been greatly appreciated. Thanks to their support and the quality of their supervision, this research was conducted in a pleasant and, above all, highly educational context.

Finally, My great thanks also go to all the teachers and school principals who took part of their precious time to respond to our questionnaire.

Abstract

This work represents my graduation project at the Private Higher School of Engineering and Technology « ESPRIT » for the attainment of my National Engineering degree in electromechanics .

With the assistance of the Gafsa Phosphate Company, the project aims to design and develop an Electric Industrial Heavy-Duty Transfer cart with the mission of facilitating the transportation process of heavy equipment inside the sub-assemblies division.

Consequently, I have employed a numerical approach to model and Analyze the mechanism with respect for the predefined needs and constraints to ensure that it performs the desired mission. Additionally, I have conducted the necessary calculations to validate the choices of the selected solutions.

Résumé

Ce travail représente mon projet de fin d'études à l'École Supérieure Privée d'Ingénierie et de Technologies « ESPRIT » pour l'obtention de mon diplôme national d'ingénieur en électromécanique.

Avec l'assistance de la Compagnie des Phosphates de Gafsa, le projet vise à concevoir et développer un véhicule de traction électrique industriel pour service lourd, avec pour mission de faciliter le processus de transport des équipements lourds au sein de la division des sous-ensembles.

Par conséquent, j'ai utilisé une approche numérique pour modéliser et analyser le mécanisme en respectant les besoins et contraintes prédéfinis afin de garantir qu'il remplisse la mission souhaitée. De plus, j'ai effectué les calculs nécessaires pour valider les choix des solutions sélectionnées.

تلخيص

تمثل هذه الاطروحة مشروع تخرجي من المدرسة العليا الخاصة للهندسة والتكنولوجيا «ESPRIT» بهدف نيل شهادة الهندسة الوطنية في اختصاص الهندسة الكهروميكانيكية.

يهدف المشروع إلى تصميم وتطوير مركبة كهربائية ذات استعمال صناعي وقدرة جر كبيرة بهدف تسهيل نقل المعدات الثقيلة داخل قسم التجميعات الفرعية التابع لدائرة الصيانة في شركة فوسفاط قفصة.

بالتالي، تبنيت منهجًا عدديًا في عملية النمذجة والتحليل مع احترام الاحتياجات والقيود المحددة مسبقًا لضمان تنفيذ المهمة المطلوبة. بالإضافة إلى ذلك، قمت بإجراء الحسابات الهندسية اللازمة للتحقق من صحة الخيارات التي تم طرحها.

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Design and Development of an Electric Industrial Heavy-Duty Electric Transfer Cart

QR code

2023/2024, Mohamed Belhassan
Department of Electromechanical Engineering
Private Higher School of Engineering and Technology
Graduation Thesis : National Engineering Degree

SUMMARY

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General Introduction

Heavy material handling has been a necessity since the early days of industrial history. In the past, moving heavy equipment within a factory or workshop often involved a significant amount of labor, time, and work risks. which leads to inefficiencies and increases the logistical and injury-related issues

Consequently, The world has been collectively moving towards electric-controlled mechanization , aiming to utilize machines to decrease manual labor while enhancing productivity and quality.

As a result of these developments, the Gafsa phosphate company « CPG » , specialized in mining and heavy machinery, is strategically moving towards adopting this approach across its divisions, starting with the critical sub-assemblies division.

Thus, my thesis presents a summary of the Development & Design of a proposed solution undertaken to meet the company's goal.

I have divided the work into five steps. The first phase was placing the project within its framework by introducing the host company and the organizational chart of the department . Followed by a literature review aiming to gather general information relevant to the subject.

The functional analysis stage, which was the third step, involved discussing and proposing different ideas and solutions to satisfy the project's requirements. This stage was the foundation for the next two phases, which were the mechanical research and design and the electric study and design, respectively.

CHAPTER-I

- Introduction of the Host Company
- Contextualization & problematic
- Methodology

I.1.Presentation of the Host Company

Gafsa Phosphate Company is a Tunisian company involved in phosphate mining and exploitation based in the city of Gafsa, Tunisia. CPG is among the largest phosphate producers globally, ranking fifth worldwide. Its activities are defined in four main groups: land preparation, extraction, production, and the marketing of phosphates.

Due to the harsh working conditions in the mining quarries, CPG was facing a decline in the efficiency of its machinery. Consequently, it became necessary to establish a maintenance department to maintain the defective machines.

Figure 1:Host Company Logo

I.2. DMM Divisions and Organigram

The DMM is divided into several workshops and Divisions based on the type of assigned tasks. The organigram of DMM illustrates a structure with clear functional Divisions.

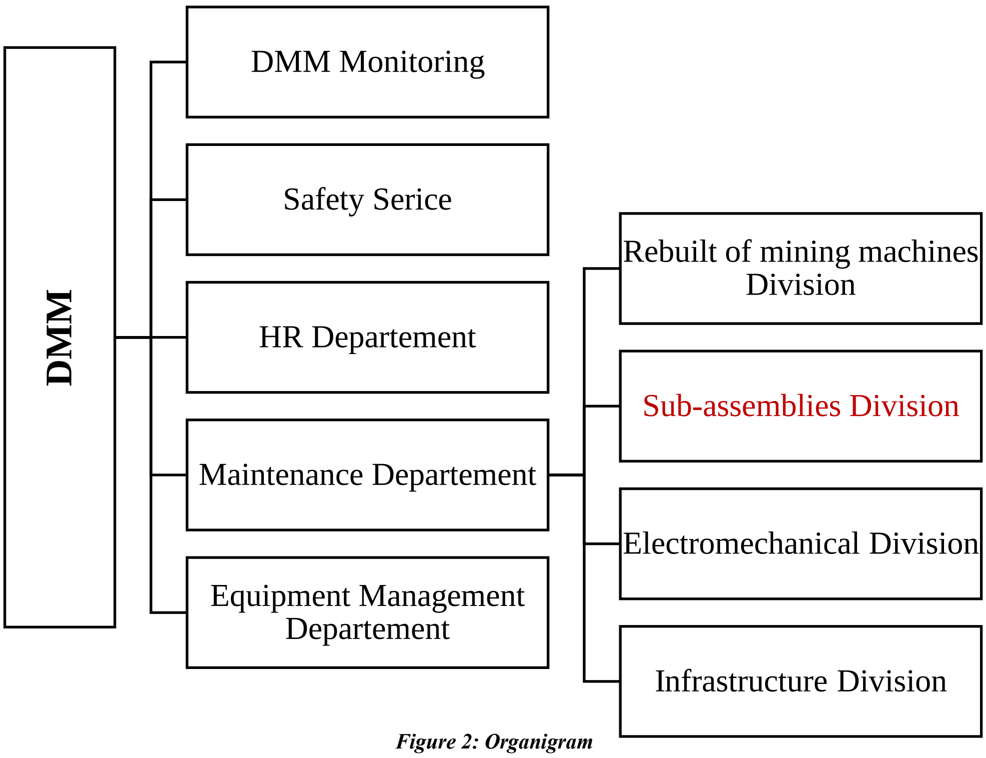


Figure 2: Organigram

I.2.1.Sub-assemblies Division

My graduation project takes place at the Maintenance and Equipment Department « DMM », particularly at the sub-assembly's division which plays an important role in maintaining the workflow of the department. Its main duties of the sub-assembly's division involves rebuilding and repairing powertrains, Gear boxes and engine parts . In addition to conducting interventions on sites.

Figure 3:Sub-Assembly Division Overview

I.3. Contextualization and problematic

Since the Subassemblies Division handles a large range of heavy machinery that needs regular maintenance , the « DMM » places a significant importance on this division and understands the importance of providing an ideal working environment for the division's workers as they need a vast range of equipment and machinery in order to accomplish the required operations.

However, Currently, the disassembly of gearboxes and engines takes place in an unequipped metal platform ***[Figure 4]*** , leading to various inconveniences such as dirt and oil stains on the floor, increasing the risk of slips and injuries for the workers.

Simultaneously, workers utilize a lower-quality cart that depends on human strength to transport heavy parts from the workshop to the disassembly area ***[Figure 5].*** This approach has limitations in terms of efficiency and reduces working time, potentially imposing physical strains on the personnel.

Figure 6: Gear Boxes Disassembly

Figure 5:Manual Transfer Cart

In this context, I was tasked with the responsibility of proposing a Solution to those issues. The solution should guaranty the facilitating the Heavy equipment transfer process in order to relocate the disassembly and maintenance operation to a dedicated room equipped with the necessary tools while keeping a low budget in comparison to the available solutions in the market

To implement this proposal, it is crucial to opt for a means of transportation specifically designed for heavy equipment. consequently, I have suggested the design of an Electric

Industrial Heavy-Duty Transfer Cart with the mission of ensuring an easy transportation of the components.

This solution offers the following advantages:

- ✓ ***Increase Efficiency:*** Industrial transfer carts provide a faster and more efficient means of transporting equipment which increases productivity.
- ✓ ***Increase Load Capacity:*** Industrial carts have higher load capacities compared to manually operated carts, which allows the workers to transport larger and heavier equipment
- ✓ ***Time and Cost Savings:*** The increased speed and efficiency of industrial carts results in time savings which impact the operational costs
- ✓ ***Division Segregation:*** moving the maintenance process to a designated area Reduce the dirt and odors in the workshop arising from lubricants.
- ✓ ***Reduce Physical Strain:*** using an industrial cart to transport heavy equipment, reduce the physical strain and minimize the risk of injuries .

Since this project takes place at the company level, some specific methods and constraints will be adopted, including the work process as well as some design and manufacturing elements such as the choice and availability of materials and parts. Additionally, Obtaining the standard elements will also be consistent with local DMM suppliers.

I.5. Methodology

At the level of this project, two approaches are possible:

- ***The first method*** consists of retaining the structure of an existing system and adjusting the main parameters to ensure the desired mission. The technological solutions are then developed, and the criteria of ergonomics and resistance are verified later by very simplified models.
- ***The second method*** consists of conducting a complete new analytical study which allows to characterize an innovative mechanism from the point of view of kinematics as well as to identify and quantify the forces ap-plied on each component and bond

Considering the context of this project, which aims to establish a new functional and efficient system and following discussion with the supervisor, the decision has been made to adopt this second methodology for designing an industrial heavy-duty Electric cart.

I.5.1.Methodology Diagram

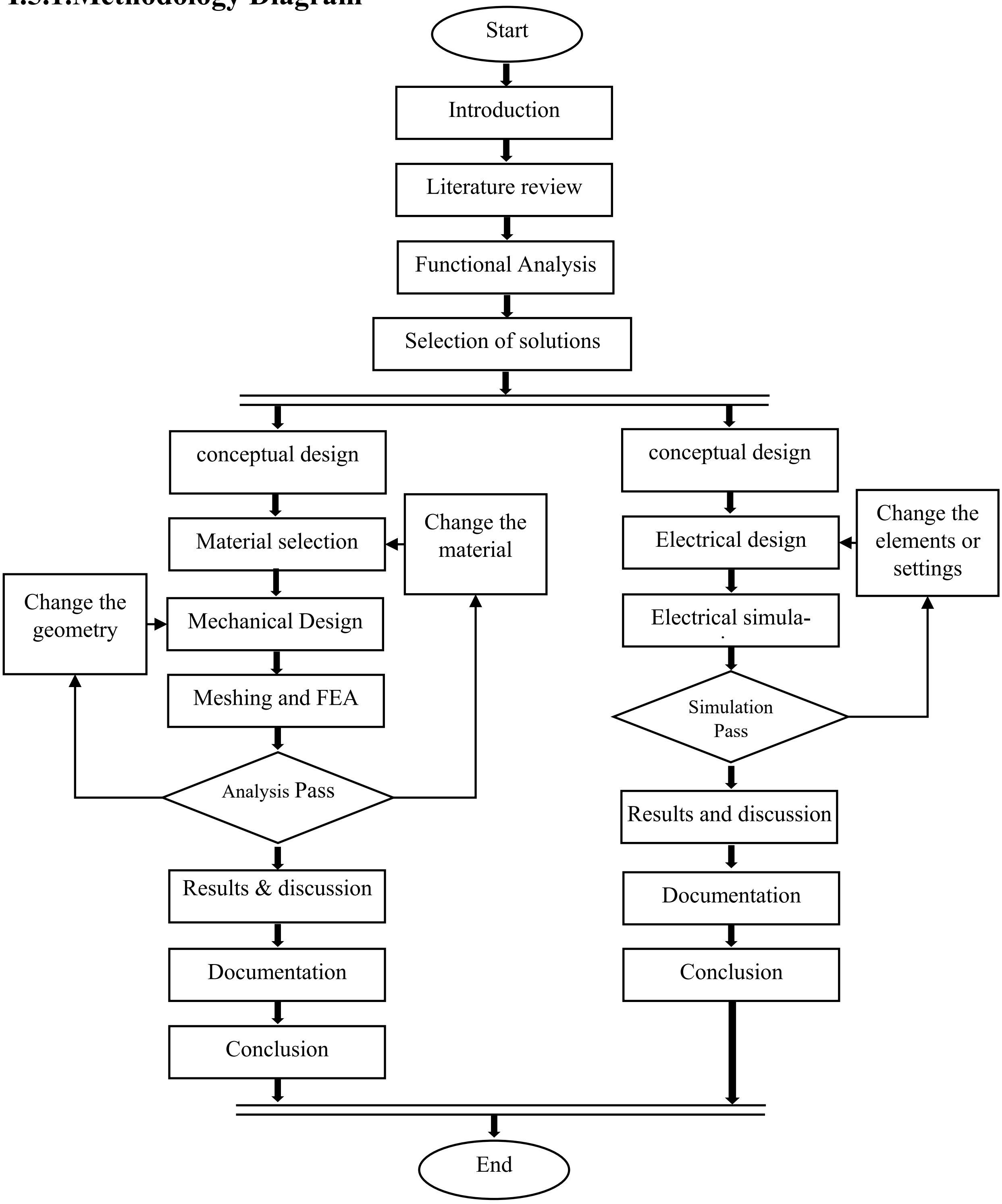


Figure 7:Methodology Diagram

I.6. Conclusion

This initial chapter served as an introduction to the project's environment. It elucidates the context and articulates the problem to be solved, paving the way for the development of a functional methodology designed to tackle the identified issue. This methodology is subdivided into stages that will be explored gradually at the level of the next chapters.

CHAPTER-II

- Literature Review

II.1. The Common types and shapes of Heavy-duty transfer carts

Heavy-duty industrial transfer carts are available in a variety of styles and designs, each ideal for a particular need. These carts are either manually or automatically operated

- ***Rail Transfer Carts:*** This type of transfer cart is designed to operate on railways and is commonly used in long distance industrial environments. Their main disadvantage is their limited maneuverability but on the other hand, the rail transfer carts can handle extremely heavy loads reaching 300 Tons.

Figure 8:Rail Transfer Cart

- ***Trackless transfer carts:*** It is generally an automated vehicle that offers more movement flexibility and is commonly used in warehouses and production facilities where installing rails is not possible. Their main disadvantage is their generally limited load capacity and high cost.

Figure 9:Trackless Transfer Cart

- ***Traction carts:*** it is designed to pull a loaded trailer while offering efficient transfer processes. This type is manually operated which means a cheaper price in comparison to the trackless transfer carts while maintaining the same flexibility advantages

Figure 10:Traction Cart

II.2. Industries and Applications

- ***Manufacturing:*** Within manufacturing plants, the flow of products and raw materials is a nonstop process. Therefore, the Industrial transfer carts play an important role in optimizing these processes by ensuring a continues and efficient flow within production facilities.

- ***Warehousing and Logistics:*** In warehouses and logistics centers, the fast and precise transit of goods is imperative, that's why the transfer carts play a vital role in enhancing storage and distribution processes.

- ***Mining:*** The mining industry manages substantial volumes of raw materials and minerals. Consequently, the transfer carts are employed to transport heavy loads of materials to increase the efficiency of the mining process.

II.3. Emerging Technologies of transfer carts

- ***Emergence of Electric Transfer Carts:*** The electric powered carts showcase the capability of handling heavier loads, making them preferred choices. Equipped with electric motors, they have managed to improve control and maneuverability resulting in smoother material transportation.
- ***Integration of Automation:*** With the addition of advanced technology like sensors, PLCs, and computerized systems, automated transfer carts have revolutionized the transfer cart market by enhancing accuracy, dependability, and efficiency.

- ***The evolution of AI and IOT:*** With the help of cutting-edge technology like artificial intelligence, machine learning, and Internet of Things connectivity, an increasing number of transfer carts are becoming more intelligent and efficient. These carts have the ability to gather and analyze data, communicate with other pieces of equipment, and make intelligent decisions in real time.

II.5. Conclusion

At the end of this literature review, the aim was to outline the various types of heavy-duty industrial carts and their applications, while also highlighting the Emerging Technologies circulating the domain. The functional analysis and the selection of the solution that best suits our needs will be detailed in the third chapter.

CHAPTER-III

- Functional Analysis
- Proposal & selection of Solutions

III.1. Functional Analysis of the Needs

The functional analysis of needs is an approach aiming to develop an explicit specification document that ensures the satisfaction of the needs expressed by the future user. This analysis establishes the relationship between the product to be designed and its context of use.

III.1.1. Expressing the need

The expression of the need is conducted via a graphical tool employed to identify the intended purpose of the product. This tool is the Horned beast Diagram

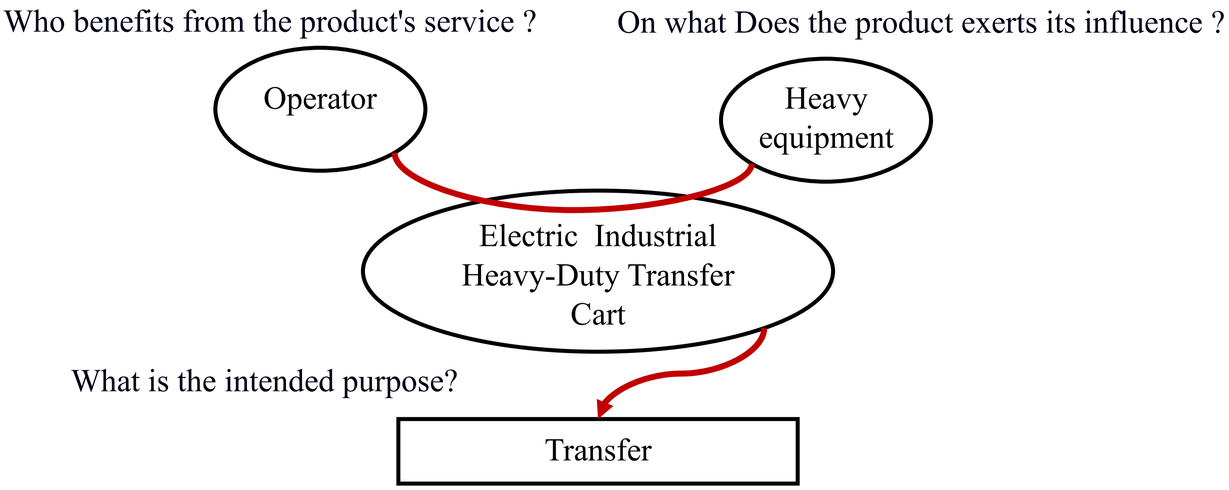


Figure 11:Horned Beast Diagram

Responding to the previous three questions of the Horned beast Diagram, the following statement can be made:

- the Heavy-Duty transfer cart allows the Operator to Transfer Heavy equipment.

III.1.2. Identification of the service Functions

The Octopus diagram aims to determine the relationships between the system and its environment.

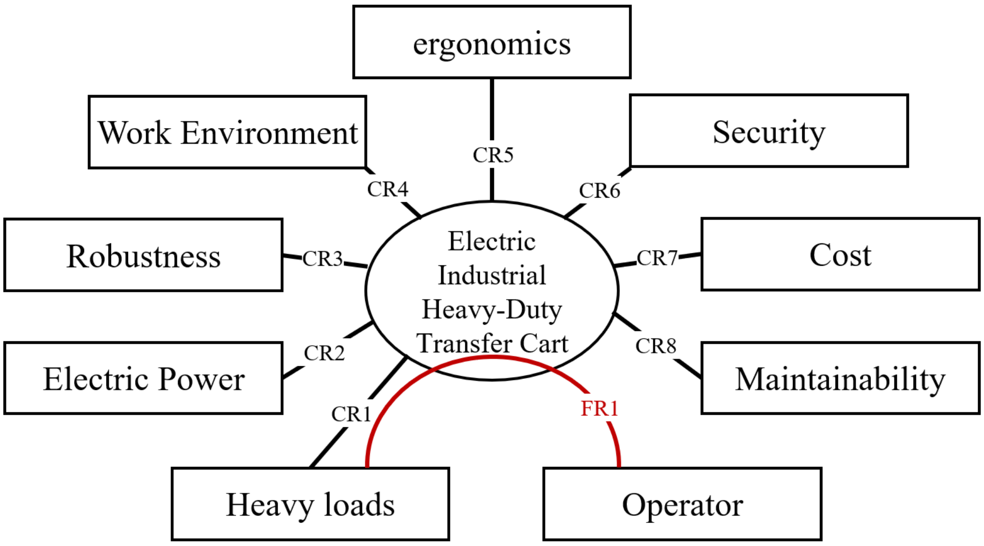


Figure 12: Octopus Diagram

The service Functions are:

- **FR1:** Enables the Operator to transfer Heavy equipment.
- **CR1:** Pull heavy loads.
- **CR2:** Uses Electric Power to operate.
- **CR3:** Must be robust.
- **CR4:** Resist the effects of the work environment.
- **CR5:** Must comply with the ergonomic standards.
- **CR6:** Provides protection for the operator.
- **CR7:** Must be within the budget ceiling.
- **CR8:** Must be easy to maintain.

III.1.3. Validation of the service Functions

Verifying that each function by posing the following two questions:

- Question 1: What is the purpose of the function?
- Question 2: For what reasons does the function exist?

FR1: Enables the Operator to Transfer Heavy equipment.
Q1: What is the purpose of the function?
facilitate the process of Heavy equipment movement.
Q2: For what reasons does the function exist?
The operation is difficult to accomplish manually.
⇨ FR1 is validated.

CR1: Pull heavy loads.
Q1: What is the purpose of the function?
Ensure that the cart has the capability to pull heavy loads.
Q2: For what reasons does the function exist?
The cart's abilities should be suitable for tasks that involve pulling significant weights.
⇨ CR1 is validated.

CR2: Uses Electric Power to operate.
Q1: What is the purpose of the function?
The cart utilizes Electric power for the operation of the service.
Q2: For what reasons does the function exist?
to provide the necessary Electric energy for the service to function efficiently.
⇨ CR2 is validated.

CR3: Must be robust.
Q1: What is the purpose of the function?
The cart should be durable and capable of withstanding various mechanical Factors.
Q2: For what reasons does the function exist?
To enhance the longevity, reliability, and performance
⇨ CR3 is validated.

CR4: Resist the effects of the external work environment.
Q1: What is the purpose of the function?
The cart should be capable of withstanding various work environmental Factors.

Q2: For what reasons does the function exist?

to ensure the system remains operational and effective in all environmental conditions.

⇨ CR4 is validated.

CR5: Must comply with the ergonomic standards.

Q1: What is the purpose of the function?

The cart design should consider human factors to provide comfortable working.

Q2: For what reasons does the function exist?

To reduce the risk of operator fatigue or injury

⇨ CR5 is validated.

CR6: Provides protection for the operator.

Q1: What is the purpose of the function?

to enhance the safety and well-being of the user

Q2: For what reasons does the function exist?

to minimize risks, reduce accidents, and create a secure working environment.

⇨ CR6 is validated.

CR7: Must be within the budget ceiling.

Q1: What is the purpose of the function?

to ensure that the service adheres to the budget constraints.

Q2: For what reasons does the function exist?

to manage costs effectively and ensure that it aligns with the allocated budget.

⇨ CR7 is validated.

CR8: Must be easy to maintain.

Q1: What is the purpose of the function?

The design of the cart should facilitate straightforward maintenance.

Q2: For what reasons does the function exist?

To reduce downtime and lower maintenance costs.

⇨ CR8 is validated.

III.1.4. Characterization of service functions

Express the expected performance from the user by assigning to each service function evaluation criteria, and a degree of flexibility.

The classes and levels of flexibility are detailed in the table below.

Flexibility	Level of Flexibility
F0	Imperative
F1	Not very negotiable
F2	Negotiable
F3	Very negotiable

Table 1: Service Functions Flexibility

The service Functions	Assessment criteria	Level	Flexibility	
			Class	Limits
FR1: Enables the Operator to Transfer Heavy equipment.	Control	Manual	F0	
	working posture	Sitting posture	F0	
	Maneuverability		F0	
CR1: Drag heavy loads.	Load Capacity(kg)	Load $\leq$ 4000	F1	± 500
	Torque (Nm)	1604	F1	±100
	Speed (km/h)	10 $\leq$ S $\leq$ 15	F1	±5
CR2: Uses electric Power to operate.	Power Capacity (Kw)	4 Kw	F0	
	Autonomy (Ah)	800	F1	±100
	Recharging Time (h)	3		±30 min
CR3: Must be robust.	Mechanical Robustness	True		
	Stability	True		
CR4: Resist the effects of the external work environment.	Dust & Water Resistance	IP54	F1	
	Temperature (C°)	5 $\leq$ T $\leq$ 40	F1	±5°C
	Corrosion Resistance		F0	
CR5: Must comply with the ergonomic standards.	Dimensions (m)	L $\leq$ 2.5 W $\leq$ 1.3 H $\leq$ 2	F1	±100 mm ±100 mm ±100 mm
	NF x35-106 standards	True	F0	
	modern Design	True	F1	
CR6: protection for the operator.	ISO 16090-1:2017 Norms	True	F0	
CR7: Must be within the budget ceiling.	Cost (Tunisian Dinar)	Cost $\leq$ 10.000	F0	±500
CR8: Must be easy to maintain.	Modular Design	True	F1	
	Availability of diagnostic tools	True	F0	
	Assemblable	True	F0	

Table 2:Characterization of service functions

III.2. Selection of Solutions

During this phase , various solutions and ideas will be presented to meet the requirements of the functional specifications .

III.2.1.Technical Functional Analysis

The Technical functional analysis is an approach that aims to materialize the service functions stated in the functional analysis of the need by affecting technological solutions that comply with the technological solutions implemented by the DMM.

III.2.1.1.FAST Diagram

In order to accomplish this task, I will use the FAST (Function Analysis System Technic) diagram which describes the technological proposed solutions.

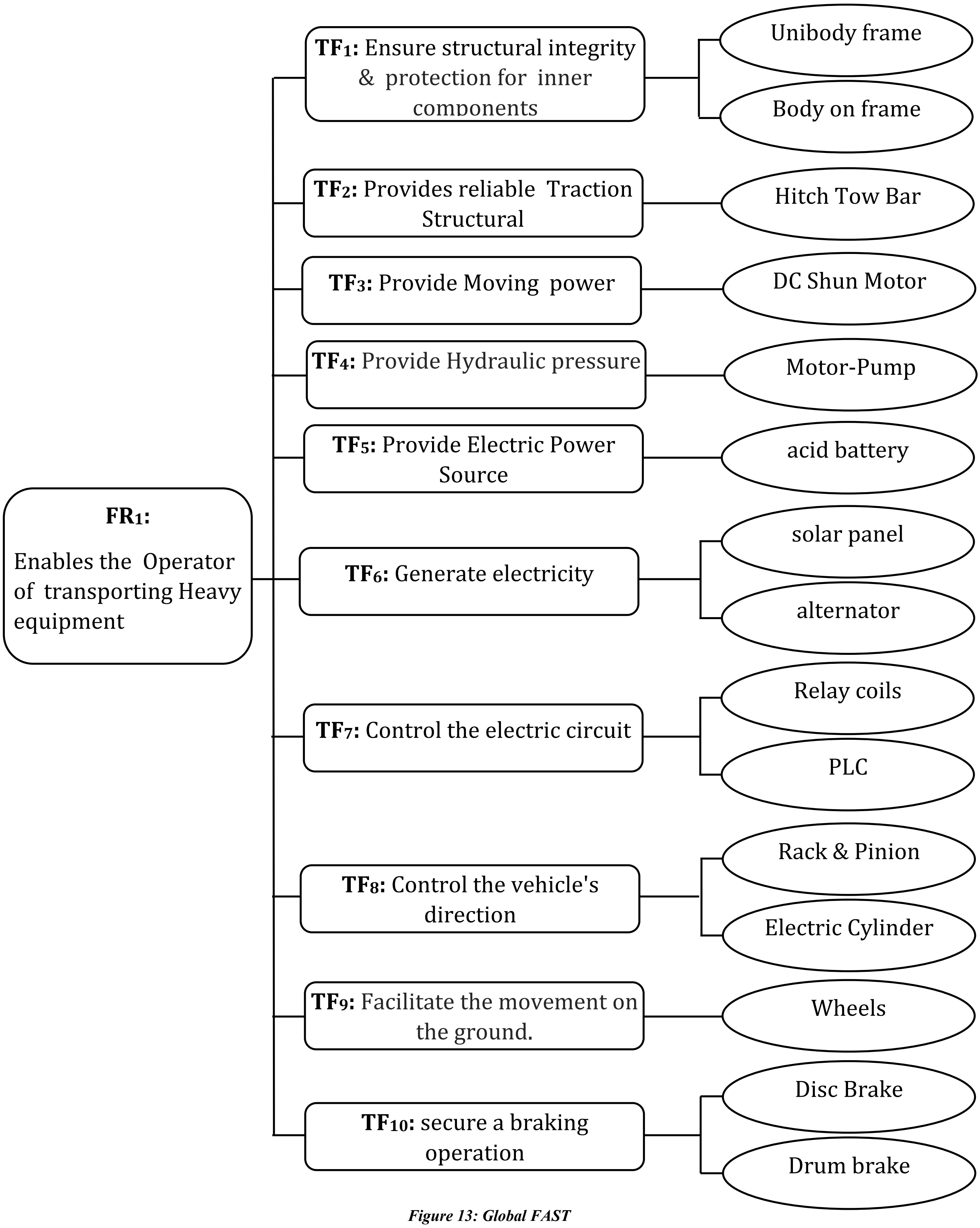


Figure 13: Global FAST

III.2.1. TF1: Ensure structural integrity & protection for inner components

The Frame is the mechanical part that ensures the physical integrity of the cart , as it provides the structural support and rigidity necessary to withstand the stresses encountered during normal operation, including impacts, vibrations, and torsion Forces. While the rest of the body structure ensure the enclosure and the protection for both the driver and the inner components of the cart

There are several types of frames and body structure compositions, each with its own characteristics and suitability for different applications, such as:

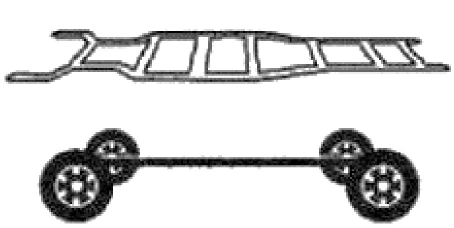
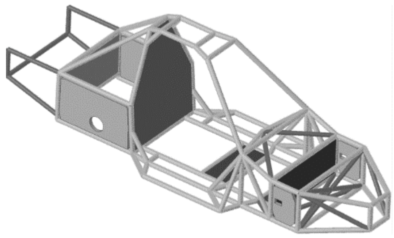
1. **Body on frame**	2. **Unibody frame.**
	

Figure 14: Chassis Types

In order to select the suitable frame type, the following criteria are to be considered with an assigned weighting coefficient as follows:

- Strength and Durability : $C_1 = 3$
- Weight: $C_2 = 2$
- ease of manufacturability: $C_3 = 3$

III.2.1.1. Strength and Durability C_1

The frame must be strong enough to withstand the stresses encountered during normal operation, including impacts, vibrations, and torsion Forces.

The Body on frame design consists of an independent frame onto which the cart's body is mounted afterwords. It offers excellent strength and durability, making it ideal for heavy-duty vehicles because it provides good load handling capacity and robustness.

In the other hand, the unibody frame design consists of a combined frame and body in a single inseparable structure. This structure enhance the steering control of the cart , but it significantly impacts the traction capacities.

III.2.1.2. Weight C_2

Balancing strength with weight is essential for achieving efficiency, handling, and payload capacity.

The body on frame structure represents a modular assembly with parts of different materials which can facilitate the control over the overall weight of the vehicle.

On the contrary, the unibody frame is a one single structure , which limits the material diversification and therefore bounds the ability of controlling the overall weight.

III.2.1.3. Ease of manufacturability C_3

The manufacturability and assembly techniques are critical considerations in the selection of frame type. Therefore, it should be aligned with the manufacturing machinery available in the DMM.

III.2.1.4. Selecting the suitable Frame solution.

The table below classifies the proposed solution based on the objective criteria and the previously assigning weighted scores ranging from 1 to 3 for each criterion.

Solution / Criteria	Body on frame	Unibody
Strength and durability: ($C_1 = 3$)	3	1
Weight ($C_2 =2$)	1	3
Cost and manufacturability: ($C_3 = 3$)	3	1
Score	20/24	12/24

Figure 15: Criteria of solution Selection

The Body on frame solution realizes a total score of 20 compared to a potential score of 24, while the Unibody solution only reaches a score of 12.

Consequently , the Body on frame solution will be chosen for the rest of the project.

IV.2.2. TF2: Provides reliable Traction Structural

Since Pulling trailers loaded with heavy equipment is the main function of the cart, installing a suitable towing mechanism is essential. In this context the DMM have suggested to use the ball hitch Traction mechanism solution since it is Available and simple to use.

IV.2.2.1. Ball Hitch characteristics

- ***Simplicity:*** Easy to use and widely available, making it a popular choice for a variety of towing needs.

- ***Versatility:*** Compatible with a wide range of trailers and loads

- ***Adjustability:*** Can be easily adjusted for height and swapped out for different ball sizes to match different trailer couplers.

Figure 16:Ball Hitch

III.2.3. TF3: Provide Moving power

[Annex 11.1]

The "power Unit" is the source of the necessary mechanical force needed to make the cart move. The DMM have provided **two** 24V DC shunt Motors recovered from a Broken Linde E12 Linde Forklift.

Each of the motors contain an integrated driver gear ratio of (i=20.07) used to modify the Torque and speed . Hence there is no need for an additional Gearbox.

- ***The characteristics of the motor are :***

- Current absorbed : 104 A
-The Reduction Ratio: i= 20.07
- RPM Reduction : 2150 / 20.07 = 107 tr/min
- Torque augmentation : 40×20.07 = 802.8 Nm
- Total Power: 4 kw

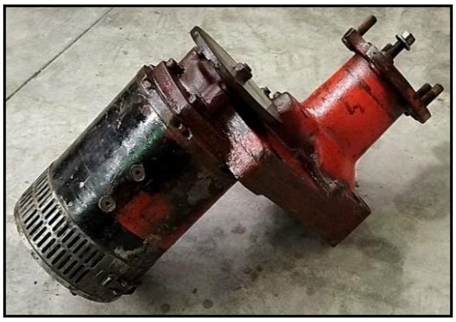


Figure 17:DC Shunt Motor

Since the motors are already selected by the DMM . the cart that I am going to design should be compatible with these motors in terms of weight and size.

IV.2.4. TF4: Provide Hydraulic pressure

[Annex 11.2]

The hydraulic system is an optional feature installed on the transfer cart to enable additional tasks, such as lifting a towed trailer with hydraulic pressure using external hydraulic actuators. Consequently, the DMM has provided a hydraulic moto-pump recuperated from a broken forklift to be employed for generating hydraulic pressure.

- ***The characteristics of the moto-pump are:***
- Power: 5 kW
- Voltage: 24 V
- current absorbed: 260 A
- speed: 2500 RPM

Figure 18:Motor Pump

IV.2.5.TF5: Provide Power Source

[Annex]

The power Source refers to the battery that provides the necessary electric energy to energize the motors and the motor pump

The DMM offers an “8 PzV 1120" lead-acid Heavy-duty rechargeable battery capable of delivering constant voltage of 24V and a capacity of 1120Ah over a 5-hour discharge period. It consists of 12 elements joined together to give the required 24 V voltage at the terminals

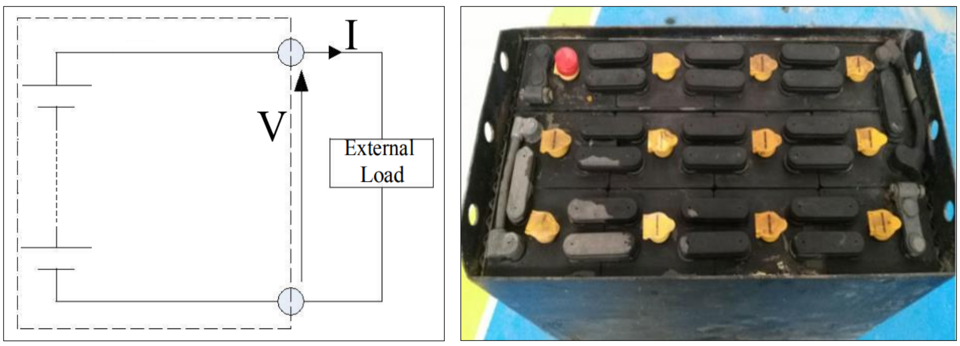


Figure 19: Lead Acid Battery

Given that the motors absorb 208 A both, while the motor pump absorbs 256 A. The motors and the pump will not be able to function simultaneously . Consequently, I will be taking the average current consumption to calculate the efficiency of the battery

$$I_{Average} = \frac{(256 + 208)}{2} = 232 \text{ A.}$$

the duration until the battery is fully depleted :

$$\text{Time} = \frac{1120}{232} = 5 \text{ h}$$

Consequently, this battery can afford 5h of continues usage, therefore it is suitable for the required task.

IV.2.6.TF6: Generate electricity

Since the cart uses electricity to operate, it is a plus to a integrate a mechanism that provides additional electric power in order to extend the operation time of the battery and empower the other electric components such as lights and indicators. Consequently, I thought of two solutions:

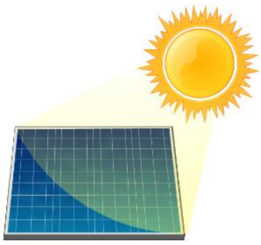
1.Solar Panel	2.Alternator
	

Figure 20:Electricity Generation Methods

In order to select the suitable solution , the following criteria are to be considered with an assigned weighting coefficient as follows:

- Effectiveness: $C_1 = 3$
- Complexity : $C_2 = 3$

IV.2.6.1.Effectiveness C_1

Since the alternator utilizes the engine's rotational movement to produce electrical power, it’s reasonable to expect that it will operate and generate power throughout the cart’s operational hours providing stable voltage output.

On the other hand, solar panels convert sunlight into electricity. However, given the cart operates primarily in an enclosed environment it poses some limitations and constraints . Nevertheless, since workers have a lunch break from 12:00 PM to 2:00 PM, it's feasible to position the cart in an open area during this time to utilize sunlight at its maximum intensity.

IV.2.6.2.Compexity C_2

The use of an alternator necessitates a number of subordinate parts, including a cooling system to avoid overheating, a drive belt, and a voltage regulator that controls the output voltage, all of which lead to additional maintenance interventions.

On the other hand, because solar panels are generally made of photovoltaic cells, which directly convert sunlight into electricity without the need for moving parts, they offer a simpler alternative to alternators in terms of components and maintenance. This reduces complexity and maintenance requirements. Solar panels normally require very little upkeep and only the occasional cleaning.

IV.2.6.3. Selecting the suitable solution

The table below classifies the proposed solution based on the objective criteria and the previously assigning weighted scores ranging from 1 to 3 for each criterion.

Criteria \ Solution	Solar panel	Alternator
Effectiveness : ($C_1 = 3$)	2	3
Complexity ($C_2 = 3$)	3	1
Score	20/24	12/24

Table 3: Criteria of Solution Selection

The solar panel solution realizes a total score of 15 compared to a potential score of 18, while the Unibody solution only reaches a score of 10.

Consequently , the solar panel solution will be chosen for the rest of the project.

IV.2.6.4.Perspectives

In the future, it is preferable to integrate an alternator into the system. Which would complement the power provided by the solar panels and enhance the overall energy reliability and efficiency of the cart.

IV.2.7. TF7: Control the Electric Circuit

The control of the electric circuit refers to the control of the different electric actuators of the cart such as the motors and motor pump. It is essential for ensuring the proper operation of the mechanism. Consequently, two control devices are to be considered :

1.Relay Coils	**2.PLC**
	

Figure 21: Electric Control Methods

In order to select the suitable electric control method , the following criteria are to be considered each with an assigned weighting coefficient as follows:

- Adaptability: $C_1 = 3$
- Flexibility: $C_2 = 3$
- Cost : $C_3 = 3$

IV.2.7.1.Adaptability C_1 :

Relays Coils are simple components and easy to use and setup for controlling electrical circuits. they, are basic devices that use an electromagnetic coil to open or close contacts . Meaning there is no need for programming .

On the other hand, The PLCs Requires important Ladder programming and PID control knowledge as well as the required industrial and network infrastructure which is not available in our cart's working environment

IV.2.7.2.Flexibility C_2 :

After being wired, the relays typically maintain a fixed configuration. Meaning that any changes or modifications to the control logic require physical rewiring, which affects the flexibility

However , PLCs can be easily modified and updated without rewiring using only programs , offering flexibility in adapting to changing operational requirements or system enhancements.

IV.2.7.3.Cost C_3:

While the project budget limit is 10.000 TD. The initial cost of the Relay coils is generally low as they involve basic components. And each relay operates independently, meaning the failure of one relay does not directly affect others, which reduce the maintenance costs .

In contrast, the initial cost of PLC is very high, as it involves purchasing the controller hardware, I/O modules, programming software, and potentially additional accessories like communication modules.

IV.2.7.4.Selection of the suitable control Circuit

The table below classifies the proposed solution based on the objective criteria previously assigned .

Criteria \ Solution	Relay Coils	PLC
Adaptability: ($C_1 = 3$)	3	1
Flexibility ($C_2 = 3$)	1	3
Cost : ($C_3 = 3$)	3	1
Score	21/27	15/27

Table 4: Criteria of Solution selection

The Relay coils solution realizes a total score of 21 compared to a potential score of 27, while the drum Brake solution only reaches a score of 15. Consequently, the relays solution will be employed

IV.2.8. TF8: Control the cart direction

The steering mechanism allows the operator to control the direction of the cart in order to ensure safe movement across the division. The mechanism includes several key components that work together to translate the driver's input into the wheels.

consequently, it is important to select the suitable direction control mechanism ., in this context, Two potential solutions merit consideration

1.Manual steering mechanism	2.Electric steering mechanism
	

Figure 22: Steering Mechanism Types

In order to select the suitable power source, the following criteria are to be considered each with an assigned weighting coefficient as follows:

- Operational Efficiency: $C_1 = 3$
- Energy conception: $C_2 = 3$

IV.2.8.1.Operational efficiency C_1

The operational efficiency is represented in the compatibility with the terrain of action and the maintenance requirements. the manual steering mechanism is commonly used due to its fewer components and ease of usage in the complex environments, in addition to its simple mechanism that that doesn't need a complicated Maintenance interference in case of damage

In the other hand , the electrical steering mechanism offers a smooth control condition with less physical effort however this Electric system involves electronic components such as sensors, motors, and control units, adding complexity in terms of wiring, programming, and troubleshooting.

IV.2.8.2. Energy conception C_2

The Manual steering mechanism do not require any electrical input since it relies on the physical strength of the operator, translating to zero energy consumption for steering.

On the contrary, the electric steering mechanism consume electric energy from the vehicle's battery to power the actuators responsible for steering.

IV.2.8.3. Selecting the suitable steering mechanism

The table below classifies the proposed solution based on the objective criteria and previously assigned

Criteria \ Solution	Manual steering mechanism	Electric steering mechanism
Operational Efficiency: ($C_1 = 3$)	3	2
Energy conception ($C_2 = 3$)	3	1
Score	18/18	9/18

Table 5: Criteria of Solution Selection

The Manual system realizes a total score of 18 compared to a potential score of 18, while the Electric system only reaches a score of 9.

Therefore, the manual steering mechanism solution will be selected for the rest of the application.

IV.2.8.4.Perspectives

It is possible to add an assisting electric motor to the steering columns in order to provides additional torque to help the driver steer the vehicle with less effort if there was difficulties .

IV.2.9. TF9: Facilitate the movement on the ground.

[Annex 11.3]

The wheels are a crucial part of a cart's function, providing support, stability, and motion. Given that I won't use a separate suspension system in the cart since it is built to operate on flat surfaces, selecting wheels that can provide the needed little suspension becomes important.

the DMM has suggested using the wheels of a FIAT UNO car due to their compact size and suitability for the tractor's intended use. Moreover, these wheels, being made of rubber, provide excellent suspension properties.

IV.2.9.1.Wheel Dimensions

The wheel to be used has the following indication:
- Tire size :*135/80R13*
- Rim size: RD= 330.2x114.3

Consequently :
- Section Width: SW= 135 mm
- Aspect Ratio: AR= 80 %
- Section Hight: SH=AR ×SW, SH= 108 mm

-The wheel Diameter:

$$D = 2\times SH + WS$$
$$= 2 \times 108 + 330.2$$
$$= 546.2 \text{ mm}$$

Figure 23: Wheels

-The wheel Circumference:

$$Ci = 2 \times R \times \pi$$
$$= 2 \times (546.2/2) \times \pi$$
$$= 1{,}715 \text{ mm}$$

IV.2.10. TF10: secure a braking operation

Braking System is one of the most important systems for the tractor, its main objective is to decelerate and stop the cat. The plan is to install the braking mechanism on the front wheels, as the rear wheels will utilize engine braking. Meaning that When the front brakes are applied, they will cut power to the motors, effectively assisting in the braking

Consequently, two front wheel brake types are to be considered :

1.Drum Brake	2.Disc Brake

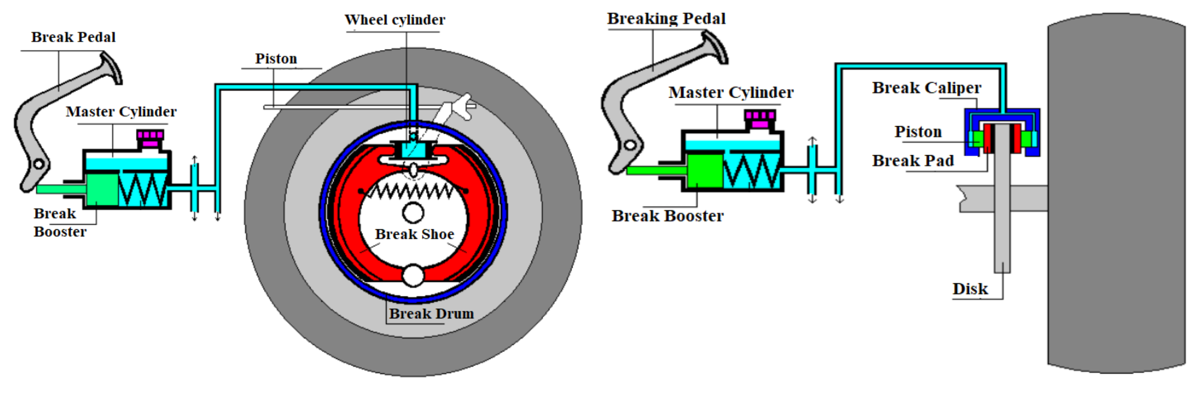


Figure 24: Brake Types

IV.2.10.1. Drum Brakes

The drum brakes comprise a rotating drum mounted along the wheel. When the driver steps on the brake pedal, the power is amplified by the brake booster and changed into hydraulic pressure by the master cylinder, making the brake drum. So, the action of brake shoe stops the rotation of drum which ultimately slows down the rotation of wheel.

- ***Drum brakes advantages.***
  - last long because drum has a large friction contact area.
- ***Drum brakes disadvantages.***
  - Drum brakes are generally less effective at dissipating heat.
  - Braking efficiency is lowered in wet conditions.

IV.2.10.2. Disc Brake

When the driver steps on the brake pedal, the power is amplified by the brake booster and changed into a hydraulic pressure by the master cylinder. The delivered pressure pushes the pistons and presses the brake pads against the brake rotors. The pads clamp on the rotors from both sides and decelerate the wheels, thereby slowing down and stopping the vehicle.

- ***Disc brakes advantages.***
  - Drums have slightly lower frequency of maintenance due to better corrosion resistance.
  - Disc brake systems are typically lighter than drum brakes.
  - The open design of disc brakes makes them less water dust resisting
- ***Disc brakes disadvantages.***
  - Disc brake systems are generally more expensive to manufacture and install compared to drum brakes.
  - Disc brake pads tend to wear out faster than drum brake shoes.

IV.2.10.3. Selection of the suitable Braking system

The table below classifies the proposed solution based on the objective criteria previously assigned

Solution / Criteria	Drum Brake	Disc Brake
Performance: ($C_1 = 3$)	2	3
Parking brake Capability ($C_2 = 3$)	0	2
Durability and Reliability: ($C_3 = 3$)	2	2
Score	12/27	21/27

Table 6: Criteria of Solution Selection

The disc Brake solution realizes a total score of 21 compared to a potential score of 27, while the drum Brake solution only reaches a score of 12.

Consequently, the disc Brake solution will be employed

IV.3.Conclusion

During this chapter, I performed a functional analysis to identify the relationships between the cart and its environment. Additionally, I proposed several solutions for each function and selected the optimal one based on established selection criteria.

CHAPTER-V

- Mechanical Design
- Finite element Analysis
- Hydraulic system Design & Simulation

V.1. Software

I will use SolidWorks for both the mechanical design and FEA simulation, because it offers powerful tools for 3D modeling and assemblies, in addition to its integrated FEA feature allows for thorough testing and validation of designs

V.1. Material selection

The aim of the material selection is to select the suitable material to use for both the frame and the rest of the body structure of the cart . The material efficiency coefficient is a function that serves to enhance the performance of the mechanism. “P”

$$P = \left(\left(\begin{array}{c} \text{Functional} \\ \text{requirements, F} \end{array} \right), \left(\begin{array}{c} \text{Geometric} \\ \text{parameters, G} \end{array} \right), \left(\begin{array}{c} \text{Material} \\ \text{index, M} \end{array} \right) \right)$$

V.1.1. Translation

The first step in the process of material selection is to translate the functional requirements determined in the service functions to mechanical properties.

The performance of the frame depends on a combination of two or more properties called material index. Therefore, it is essential to determine the ratio that connects these two properties to choose the most appropriate material.

In this case I have chosen the combination of (stiffness-Density) and (Strength-Cost) ratios

Function	Strong (Yield strength)
	Stiff (young’s Modulus)
Constraints	Weight
	Cost
Objectives	minimize the weight
	minimize the cost
Free Variables	Area of cross section
	material
Material index	Material index 1: Stiffness-Density (ρ / E $^{(1/2)}$)
	Material index 2: Strength-Cost ($C_{m\rho}$ / $\sigma^{(2/3)}$)

Table 7: Material selection Index

V.1.2. Screening

The screening phase is conducted using the Ashby charts, which are graphical tools applied to compare and select materials based on their properties.

V.1.2.1. Stiffness - Density Ratio

This chart guides the selection of materials for light and stiff components. Each data point on the chart represents a specific material, and its position indicates its Young's Modulus and Density values.

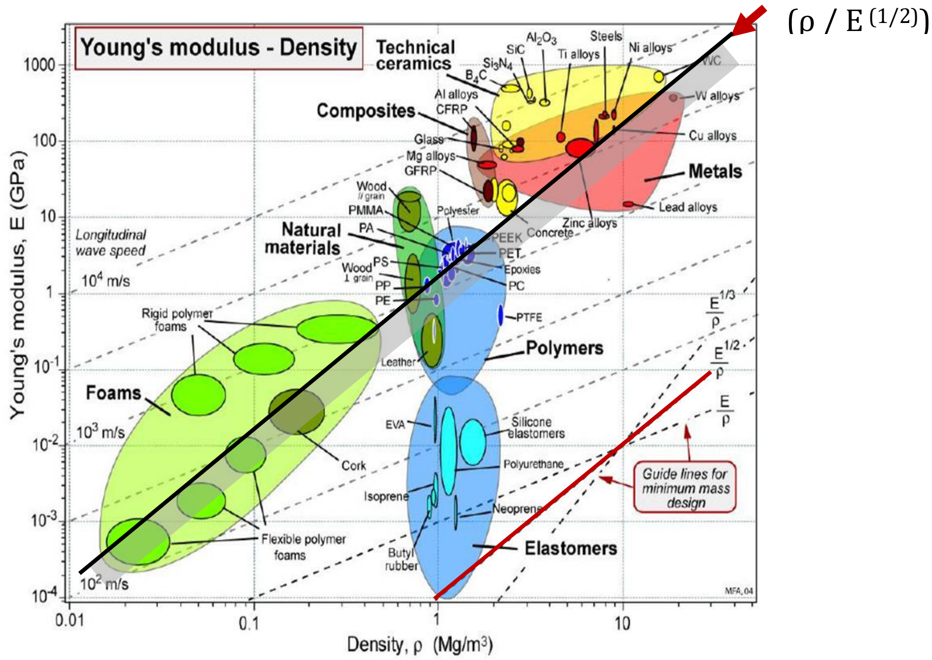


Figure 25: Ashby chart (Stiffness – Density)

Material	Tensile S N/mm^2	Yield S N/mm^2	Young's N/mm^2	Poisson's ratio	Density Kg/m^3
Steel Alloys	723.8	620.4	210000	0.28	7700
Al Alloys	68.9	27.5	69000	0.33	2700
TI Alloys	344	370	105000	0.37	4510

Table 8: Material suggestions

V.1.2.2. Strength - relative cost per volume Ratio

This chart guides the selection of Cheap and strong materials. It shows strength, plotted against relative cost per unit volume.

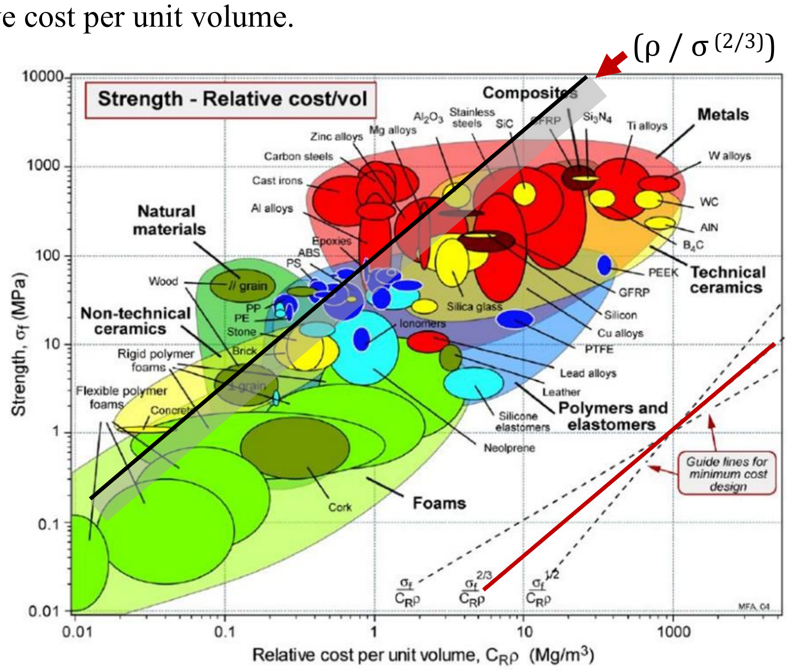


Figure 26: Ashby Chart (cost -Strength)

Material	Tensile S N/mm^2	Yield S N/mm^2	Young's N/mm^2	Poisson's ratio	Density Kg/m^3
Cast iron	151	210	66178	0.27	7200
Carbon steel	475	292.6	204999	0.23	7858
AL alloys	68.9	27.5	69000	0.33	2700

Table 9: Material Suggestions

V.1.3. Interpretation

According to the previous Ashby charts the common candidate that responds to the criteria is the Alloy Steel represented in the Carbon steel which is a type of alloy steel that consists of iron and (0.05% to 2.0%) carbon by weight. It is one of the most widely used materials in engineering and manufacturing applications due to its affordable cost, and favorable mechanical properties.

V.1.4. Conclusion

[Annex 11.5]

Following extensive research into available carbon steel producers in Tunisia, I have selected carbon steel S355J2G3, produced by SOQUIBAT Group as the suitable material.

Material	Tensile S N/mm^2	Yield S N/mm^2	Young's N/mm^2	Poisson's ratio	Density Kg/m^3
Steel S355J2G3	490	315	210000	0.28	7800

Table 10: Chassis Material Selection

V.2.Chassis

The chassis is the cart and is the structural foundation on which the cart is built. It is composed of (Frame, frontal and read body structures and canopy structure)

V.2.1.Fast Diagram

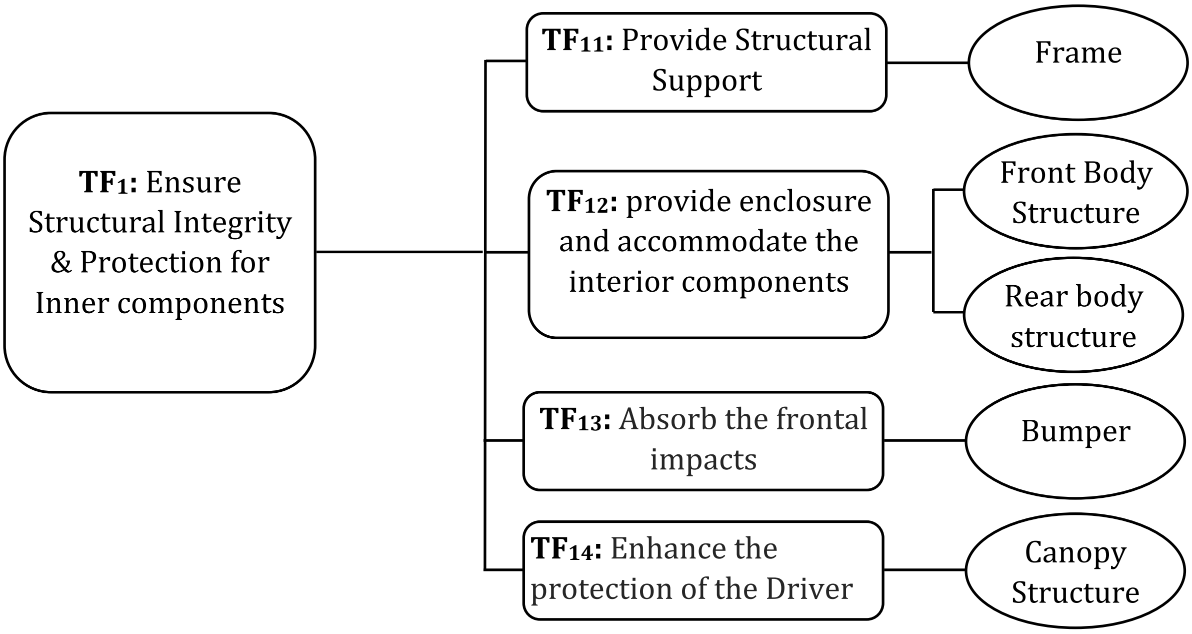


Figure 27: FAST Diagram Of Chassis

V.2.2.The Frame

Having a well-designed frame is important to ensure the Structural Integrity, performance, and Durability of the chassis. Hence, the first step of the designing process of the cart is designing the Frame.

V.2.2.1. Conceptual Design

The design of the Frame requires prior understanding of the kind of conditions the Frame is likely to face on the road because it generally experiences two major loading situations, that include:

- ***Static load***: It refers to the weight or force exerted on the frame while it is stationary, and it is generally translated to vertical bending.
- ***Horizontal Lozenging:*** It occurs when one side of the vehicle has better traction than the other. The unequal horizontal force distorts the chassis into the shape of a parallelogram.

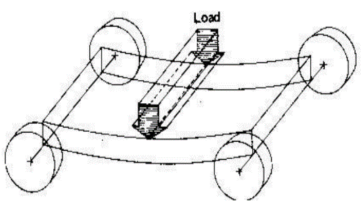
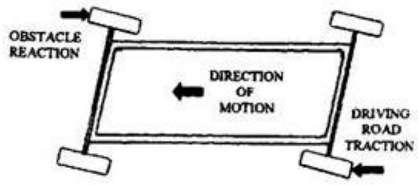
Static load	Horizontal Lozenging
	

Figure 28: Types of Loads

Under those conditions, the moment of inertia "I" and the polar moment of inertia "J", play a crucial role in determining the bending and torsion resistance of the beams that form the frame.

Using the bending stress formula and the shearing torsion stress formula it is possible to say that The greater I and J, the lesser are the bending and torsion impacts on the beam.

bending stress: $\boldsymbol{\sigma}_b = \frac{M.Y}{I}$ shearing torsion stress: $\boldsymbol{\tau} = \frac{T.d}{J}$

- ***Bending resistance:*** The" open channel" and the Hollow chapes provide the highest moment of inertia, meaning that they are the most suitable for the bending resistance and there for the other chapes are to be eliminated.
- ***Torsion resistance:*** The open-channel sections exhibit excellent resistance to bending but have limited torsion resistance properties. Therefore, the frame's geometry must be designed to resist twisting distortion.

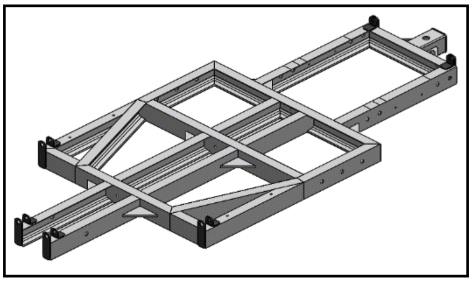
Taking into consideration the previous information, the UPN80 profiles are to be used for the frame design

V.2.2.2. Mechanical Design

[Annex 1.1 - Annex 1.2]

The chassis is composed of the frame (1) which is made up of UPN80 beams that forms the side members and cross members which are placed back-to-back to form a rigid load-supporting base in order to prevent horizontal lozenging and limit side bending. Additionally, gussets (3) strengthen triangular joints essential for distributing the load and stabilizing the frame.

Figure 29: Frame



The mounting brackets (4) are sheet metal parts bolted onto the frame, serving as a mounting platform for the rest of the body structures, the brackets have two different dimensions depending on their position on the frame.

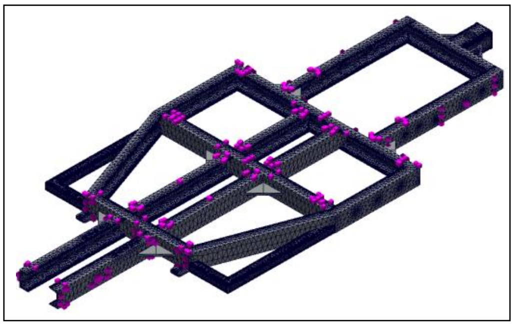
Lastly, the hitch receiver (5) is an 80x80 hollow beam forming an extension welded at the rear of the frame in order to mount the towing hitch and provide a stable and reliable connection between the cart and the trailer.

V.2.2.3. FEA Simulation

- ***Assumptions***
  - *criterion:* Von Mises criterion.
  - *Material* : S355J2G3. Consequently, σ =315 Mpa
  - *Factor of safety:* FOS=3. Consequently, $\sigma_{allowable}$ = 105 Mpa
- ***Meshing***

I will use mesh control to increase the elements number only in the areas where I suspect it will endure the highest stress and I will delete the elements that will not affect the results such as the brackets.

Figure 30: Frame Meshing



A. Static Load Analysis

To perform the static load Simulation analysis, it is essential to determine the estimations of pression values exerted on the Frame.

Component	Estimated Mass (kg)	Force value (N)	Surface (mm^2)	Pressure (Mpa)
Frame	≈ 95	931.95		
Rear Body structures & mounted parts	≈ 170	1667	6899.2	0.24

Front Body structure & mounted parts	≈ 150	1471.5	1174	1.25
Battery , Battery housing & Driver	≈ 600	5886	148120	0.039
Moter + Motors mount	≈ 200	1962	22500	0.072
Motor Pump	≈ 40	392	46350	0.008
Total Weight	≈1215			

Table 11: Loads Distribution

- ***FEA Results***

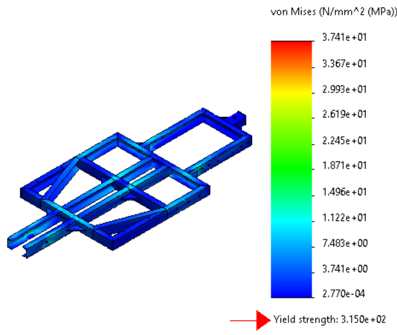
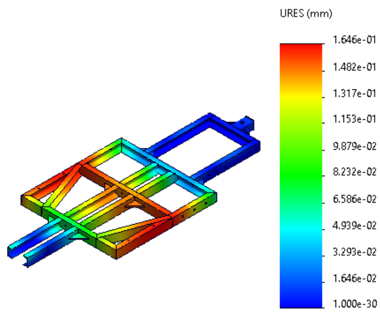
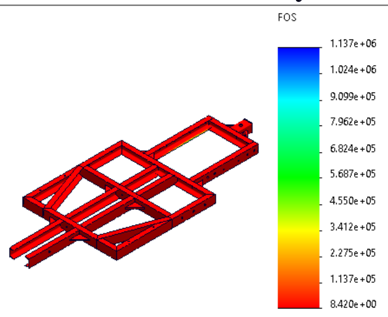
Stress analysis	**Displacement analysis**	**Factor of Safety**
		
The maximum Yield strength value is 37 Mpa < 105 Mpa ► The result is verified.	The maximum displacement value is :0.16 mm ► The result is acceptable.	The minimum FOS is 8.4 > 3 ► The structure is safe

Table 12: Static Loads Simulation Results

B. Horizontal Lozenging Analysis

To simulate the Horizontal Lozenging loads I am going to apply the driving force on one side and boundary on the other side. The driving force value on one side is 2937N and it is determined in a subsequent phase

- ***FEA Results***

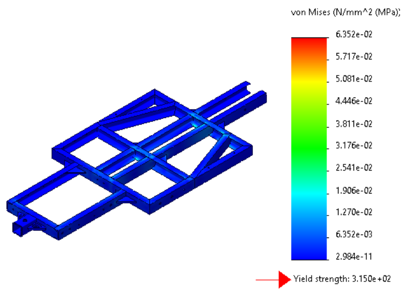
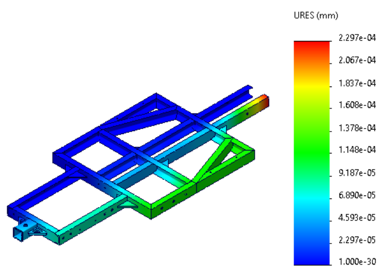
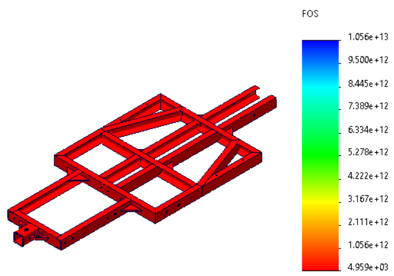
Stress analysis	**Displacement analysis**	**Factor of Safety**
		
The maximum stress value is $1.7.10^{-2}$ Mpa < 105 Mpa ► The analysis is verified.	The maximum displacement values is $5.4.10^{-4}$ ► The structure is safe.	The minimum FOS is: 4959 >> 3 ► The structure is totally safe

Table 13: Horizontal Lozenging Simulation Results

V.2.3.Front Body Structure

The Front body structure represents the frontal part of the cart's body. It is composed of several key elements that impact the cart's appearance, functionality, and safety.

V.2.3.1. Conceptual Design

I will use 40 ×40 hallow beams made of steel "S355J2G3" to reduce the total weight of the cart because the front body structure is subjected to lighter mounting loads than the frame.

while the bumper should be made of a material with important elasticity in order to enhance its resilience and toughness characteristics to absorb the impacts effectively .

Consequently, the DMM proposed to use 6161 aluminum Aloy, because the Aluminum's combination of light weight, energy absorption through deformation, makes it an ideal choice for the bumper which is designed to mitigate impact forces

Material	Tensile S N/mm^2	Yield S N/mm^2	Young's N/mm^2	Poisson's ratio	Density Kg/m^3
Al 6161	310	280	69000	0.33	2700

Table 14: Bumper Material Selection

V.2.3.2. Mechanical Design

[Annex 1.3 - Annex 1.7]

The Front body structure is composed of welded 40×40 hollow steel beams forming the structure (1) which is bolted to the frame of the cart (2) using mounting brackets (3). The structure offers an empty space in the middle used to accommodate the inner components such as the hydraulic system .

In the other hand The bumper (4) is a structural component designed to absorb the impacts, thus reducing damage to the cart's body , components and enhancing the driver's safety.

The bumper is made of 40×80 hollow aluminum beams attached to the front body structure using bolts(5), to ensure the ability to be easily replaced if damaged. Additionally, a metallic sheet cover (6) is wrapped around the bumper to enhance the vehicle's appearance, improving its ergonomic standards.

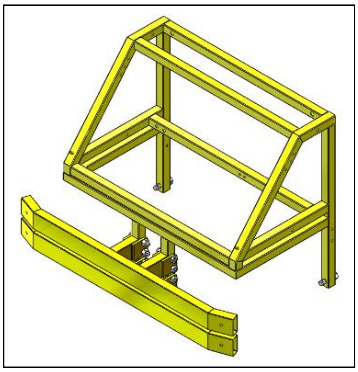


Figure 31: Front Body Structure

V.2.3.3. Frontal impact analysis

The FEA simulation for the front body structure will be about analyzing an impact effect on the front bumper of the cart with the aim of minimizing it as much as possible.

A. Determining the impact Force value

When moving, the cart has a kinetic energy: $E_{Kinetic} = \frac{1}{2} \times m \times V^2$

In the case of impact, the kinetic energy is converted to impact energy: $E_{Kinetic} = E_{Impact}$.

the impact energy is : $E_{Impact} = S \times F$

Consequently: $S \times F = \frac{1}{2} \times m \times V^2$

- m: Total mass the cart (kg): m=1215 kg
- F: Impact Force (N)
- V: velocity at impact (m/s) : proven to be (10km/h ≈ 3 m/s) lately
- t: impact time (s)
- S : Displacement (deformation) : $S = V \times t = 3 \times 0.5 = 1.5$ m

There for, applying the previous formula: $S \times F = \frac{1}{2} \times m \times V^2$

$$1.5 \times F = 0.5 \times 1215 \times 3^2$$

$$F = \frac{0.5 \times 1215 \times 3\hat{}2}{1.5} = 3645 \text{ N}$$

So, The impact force is: F = 3645 N

B. *FEA Assumptions*

- *Material:* Aluminum Consequently, σ =275Mpa
- *Factor of safety:* FOS=2. Consequently, $\sigma_{allowable} = 140$ Mpa

C. *Boundaries, applied forces & Meshing*

The bumper is a welded structure and bolted to the main Front body structure while the impact forces are set to be perpondicular on the frontal surface .

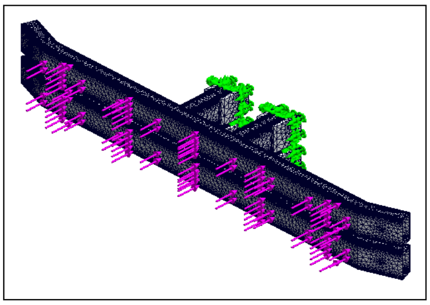


Figure 32: Bumper Meshing

D. *FEA Results*

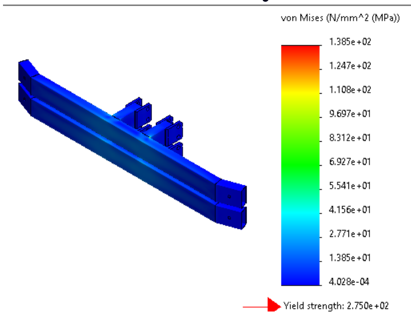
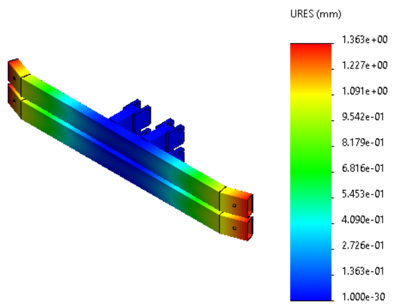
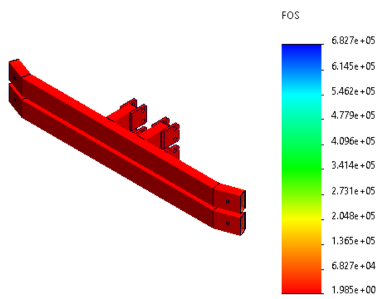
Stress analysis	**Displacement analysis**	**Factor of Safety**
		
The maximum stress value is 138.5 Mpa < 140 Mpa ► The analysis is verified.	The maximum displacement values is 1.3 mm ► The result is acceptable.	The Minimum FOS is: 1.9 ≈ 2 ►The Bumper Structure is relatively safe

Table 15: Bumper Simulation Results

V.2.4. Rear body structure.

As important as the front structure , The rear body structure contributes to the cart functionality, safety, and aesthetics. As it serves as an enclosure to protect the motors and the electric circuit, it represents a platform for additional weight.

V.2.4.1. Conceptual design

The rear body structure bears a significant load compared to the front body structure. Therefore, the beams used must be appropriately designed to handle this load. Consequently, I will use UPN60 for this purpose

V.2.4.5. Mechanical design

[Annex 1.8 – *Annex* 1.10]

The beams composing the rear body structure (1) are welded together to form the skeleton and directly bolted into the frame of the cart . This structure is designed to accommodate and protect the Motors and the electric components fixed on the support (2) . the inner electric components are reached using a hinged cover (3), the hinges (4) and the handle (5). Furthermore, the design creates a flatter platform (6) on the top that can be employed to carry additional loads

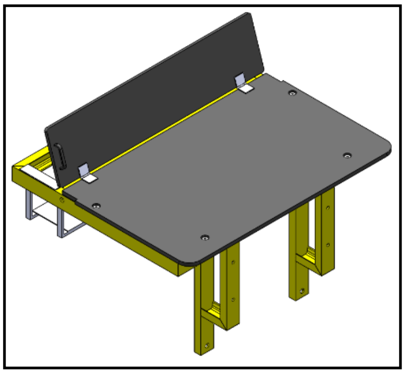


Figure 33: Rear Body Structure

V.2.4.6.FEA Simulation

The analysis in this phase will include both the Front and Rear body structures, each with proper loading conditions and simulation types.

A. Assumptions

- *criterion:* Von Mises criterion.
- *Material* : S355J2G3. Consequently, σ =315 Mpa
- *Factor of safety:* FOS=3. Consequently, $\sigma_{allowable}$ = 105 Mpa

B. Boundaries, Applied Loads & Meshing

The structure is Fixed from the bottom to the frame and from the side to the battery housing , while a 200 kg placed on the platform

$W_p = m_p \times g = 100 \times 9.81 = 1962$ N

the surface of the upper faces of the beams is : $A = 179247\ mm^2$. Consequently , the pressure applied on the rear Body structure is $P = \frac{W_P}{A} = 0.01$ Mpa

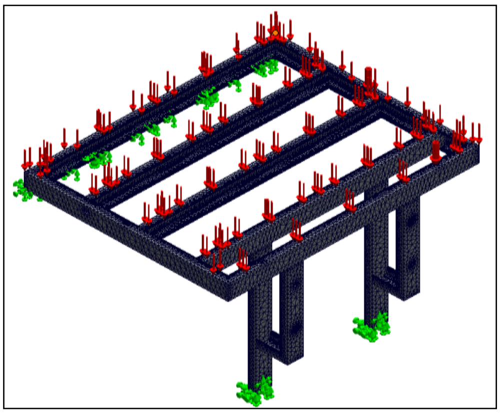


Figure 34: Meshing & Load Distribution of the rear Body Structure

C. *FEA Results*

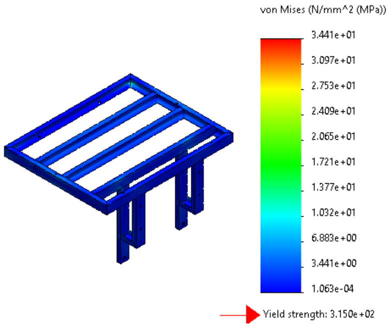
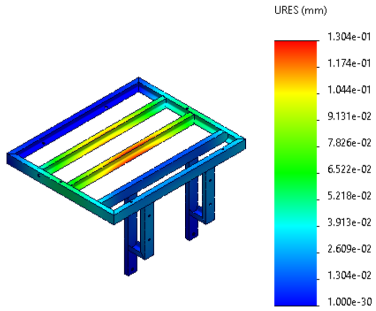
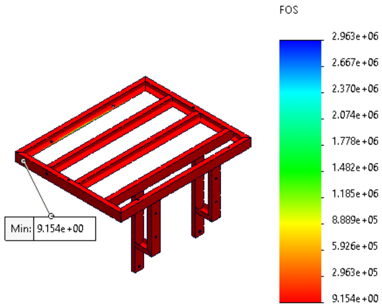
Stress Analysis	Displacement Analysis	Factor of safety
		
The maximum stresses value is 34 Mpa < 105 Mpa ► analysis is verified.	The highest displacement values is 0.1 mm ►The result is acceptable.	The Minimum FOS is 9 >3 ►The Structure is safe

Table 16: Rear Body Simulation Results

V.2.5. Canopy structure.

The canopy is an enclosure roof designed to provide a protective barrier against falling objects in the industrial environment and from the exterior weather like sunlight or rain.

V.2.5.1. Conceptual design

The canopy structure is not subjugated to any external loads. Consequently, light hollow beams are suitable for the structure in order to minimize the overall weight. I will use 40x40 hollow beams

V.2.5.2. Mechanical Design

[Annex 1.11 – Annex 1.12]

The canopy body structure is a modular element that consists of welded vertical and horizontal beams and equipped with frontal mounting point to fix it on the front body structure and rear mounting points to fix the canopy on the rear body structure using bolts and nuts . Additionally, the roof is designed with a grooved shape with the purpose of facilitating the mounting and dismounting of the battery using a crane**.**

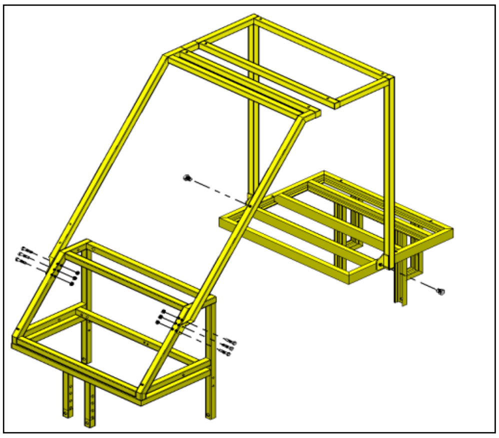


Figure 35: Canopy Structure

V.3. Traction Mechanism

The Traction mechanism is responsible for the pulling process and represents the connection point between the cart and the loaded trailer to be moved.

V.3.1.Fast Diagram

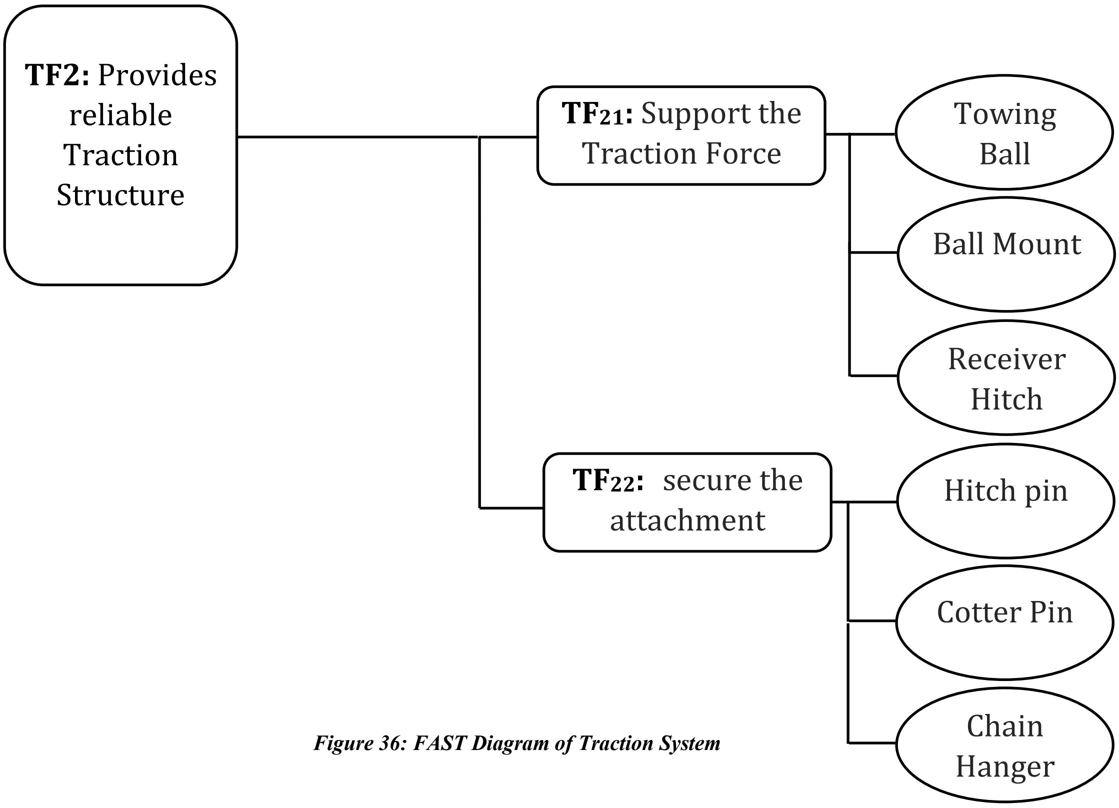


Figure 36: FAST Diagram of Traction System

V.3.3. Conceptual design

V.3.3.1. Load estimation.

To Determine the maximum traction force applied on the Hitch, I need to calculate the Horizontal traction force created by the Trailer.

Given that the mass of a fully loaded trailer is 4000 kg. and that the cart will mainly function on concrete, then The coefficient of friction is $\mu = 0.7$

Traction Force is: $F_T = m \times g \times \mu$

Consequently: $F_T = 4000 \times 9.81 \times 0.7 = 27468$ N

V.3.4. Mechanical Design

[Annex 2]

The Hitch consists of several elements attached to the Towing Mount (1) , an extension of the frame which accommodate The ball mount (2) that contains the High-Strength Grade steel ball (3) responsible of bearing the towing load while hitch pin (4) and the cotter pin (5) act as a security lock passing through the ball mount and the Towing Mount .

In the other hand The chain hanger (6) is also a security feature in the case of hitch loose, so the chain keeps the trailer mounted to the tractor.

V.3.5. FEA Simulation

A. Assumptions

- criterion: Von Mises criterion
- Material : S355J2G3, σ =315 Mpa
- Factor of safety: FOS=3. Consequently, $\sigma_{allowable}$ = 105 Mpa

B. Boundaries, External Loads & Meshing

This Analysis examines the impact of the force created by the pulled load on the traction mechanism . meaning that the simulations are accomplished by placing the Towing Force, F_T = 82404 N on the mounting location of the ball and hitch pin. While fixture locations is placed on the welded side of the towing mechanism.

I have applied curvature based mesh for better results in the rounded areas and mesh control on the places exposed to the stress

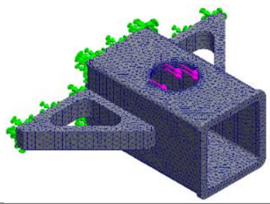
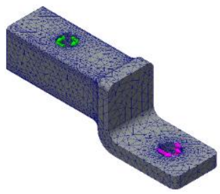
Towing Mount	Ball Mount
	

Figure 37: Traction System Meshing

C. FEA Results

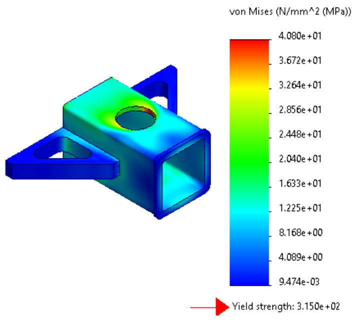
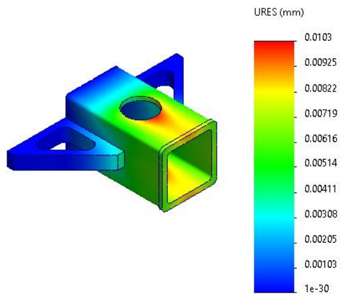
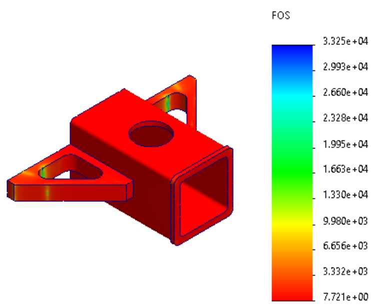
<table>
<tr><th colspan="3">Towing Mount</th></tr>
<tr><th>Stress analysis</th><th>Displacement analysis</th><th>Factor of Safety</th></tr>
<tr><td>
</td><td>
</td><td>
</td></tr>
<tr><td>the max stress is :
40 Mpa < $\sigma_{allowable}$
► The result is acceptable.</td><td>The maximum displacement value is 0.03 mm.
► The structure is safe.</td><td>The minimum FOS is 7.7 > 3
► The structure is safe</td></tr>
</table>

Table 17: Towing Mount Simulation

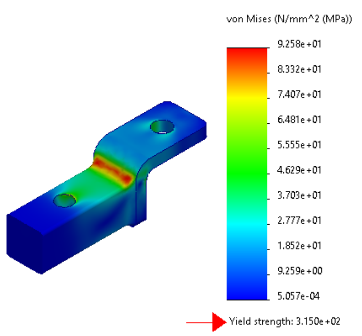
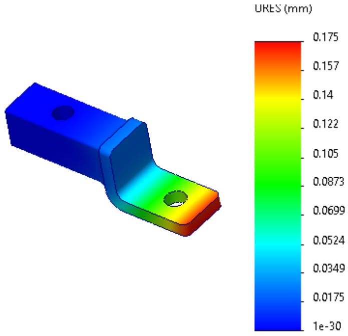
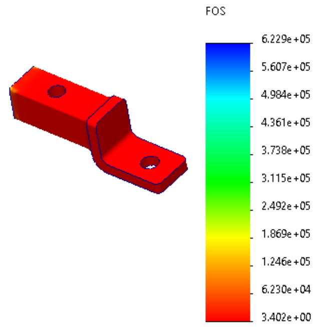
Ball Mount		
Stress analysis	Displacement analysis	Factor of Safety
		
the max stress is: 92 Mpa < $\sigma_{allowable.}$ ► The result is acceptable.	The maximum displacement 0.1 mm ► The structure is safe.	The minimum FOS is 3.4 > 3 ► The structure is totally safe

Table 18: Ball Mount Simulation

V.4. Powertrain

The Powertrain is the mechanism responsible for optimizing the motors functionalities and transmitting their rotational movement into the wheels.

V.4.1. FAST Diagram

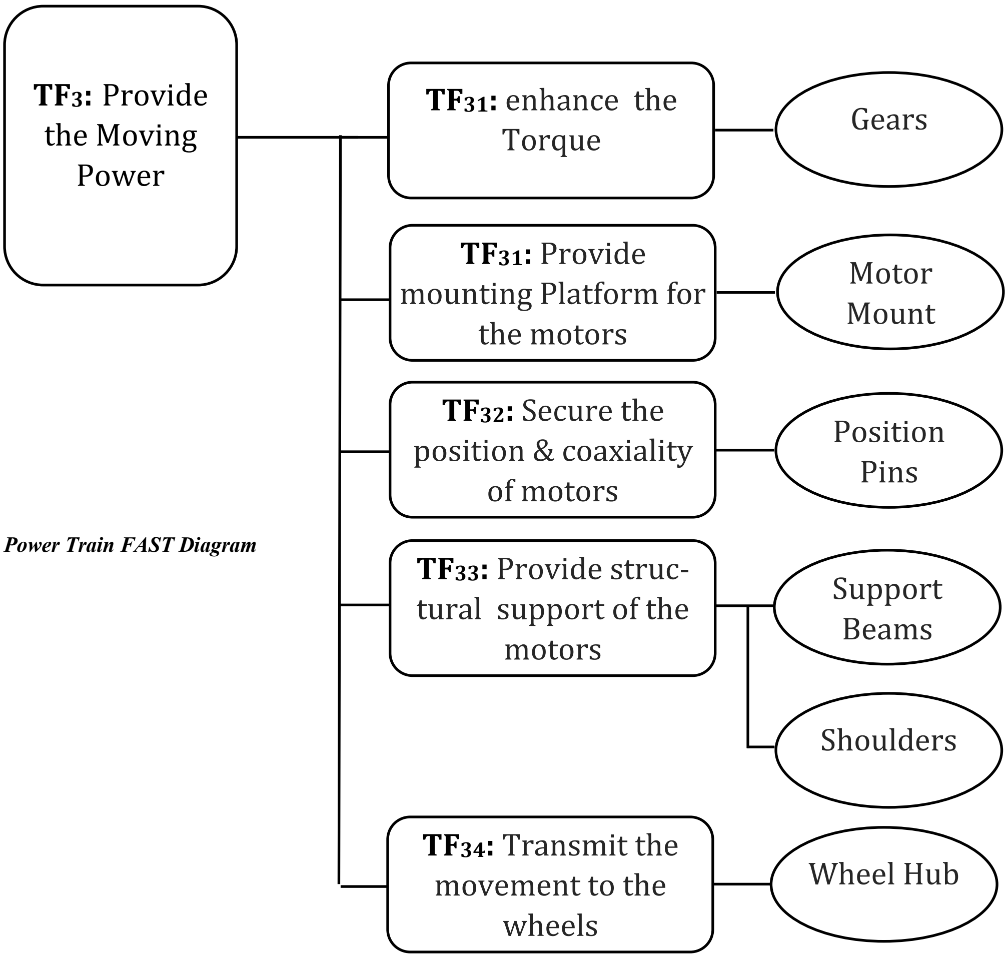


Figure 38: Power Train FAST Diagram

V.4.2. The Reduction Gears

The Mission of the reduction gears is to optimize the capabilities of the motor for heavy-duty operations. The first step to verify the suitability of the motors to this operation, I need to confirm their ability to withstand the resisting forces .

V.4.2.1. The Resistance forces.

The Cart is subjected to a number of exterior forces that affect its movement

A. Aerodynamic Resistance

Given the cart's low speed and compact size, the aerodynamic resistance is negligeable.

B. Rolling Resistance

It is the force that opposes the motion of a tire as it rolls on a surface. The cart will mostly function on concrete. Meaning that the rolling resistance Coefficient C = 0.02

Consequently, the Rolling resistance (F_{rr}) value is:

$$F_{rr} = W \times C$$

- W_1: weight of cart : $W_1 = m_1 \times g = 1215 \times 9.81 = 11056$ N
- W_2: weight of Trailer $= m_2 \times g = 4000 \times 9.81 = 39240$ N

Consequently:

$$F_{rr} = (11056+39240) \times 0.02 = 1006 \text{ N}$$

V.4.2.2. Angular Velocity

The angular velocity of the motors is:

$$\omega = \frac{2\times\pi\times RPM}{60} = \frac{2\times\pi\times 107}{60} = 11.2 \text{ Rad.s}^{-1}$$

V.4.2.3. Linear Velocity

Taking into consideration the radius of the used wheel, $R_{wheel} = 0.273$ m

$$V = \omega \times R_{wheel} = 11.2 \times 0.273$$

$$= 3 \text{ m. s}^{-1} = 10.8 \text{ Km.h}^{-1}$$

V.8.2.4.Driving Force

The driving force is the force generated by the motors in order to move forward or backward.

- Total Torque : $T_T = 2 \times 802$ Nm= 1604 Nm
- Driving Force : $F_D = T_T \times \omega = 17964$ N

V.4.2.5.The Required Power

The required power to move the cart that weights in addition to the Trailer i

$$P_{required} = F_{rr} \times V$$

$$P_{required} = 1006 \text{ N} \times 3 = 3018 \text{ W}$$

V.4.2.6.Conclusion

Upon analyzing the results : $F_D > F_r$ and P_{motors} (4kw) $> P_{required}$

Consequently, the motors are suitable for the required operation.

V.4.3.Mechanical Design

[Annex 3.1]

The Powertrain consists of several mechanical components, including the electric motors and their integrated gears (1) in addition to the motors mounts (2) that attach them to the frame of the cart . The wheels (3) are connected to the powertrain via wheel hubs (4), which act as intermediaries, while the support beams (5) serve as structural support for the mechanism.

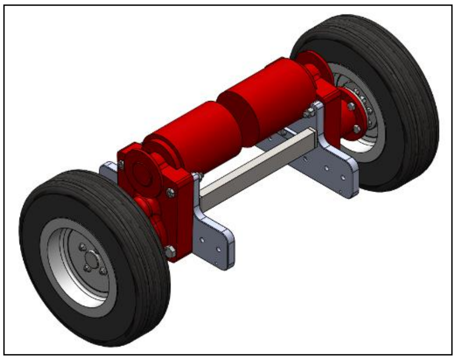


Figure 39: Power Train

V.4.3.1. Motor

[Annex 3.2]

Using the necessary measurement tools (caliper, Roll,) I took the necessary dimensions and fixation points positions . After that I translated the dimensions to a numerical 3D Model. In order to design the motor mount and the rest of the power train components.

V.4.3.2. Motor Mount

[Annex 3.3]

The motor mount is a symmetrical metallic structure, designed to accommodate the Motors in a static position to provide the best stability and performance.

It is customized to be adaptable to the motor shape and dimensions. it has two groups of fixing bolts, one group for fixing the motor mount on the cart's frame and the other group is to fix the motor on its mount.

Despite the bolt fixation, The motor mounts are exposed to several fixture deformation forms Such as translation over $\vec{y}$, and rotation over $\vec{x}$ & $\vec{z}$ that affects the coaxially, symmetrizing and the overall performance of the traction system.

To secure the position of the motor mount and prevent its fixture deformation, I have used Position pins, shoulders, and support beams.

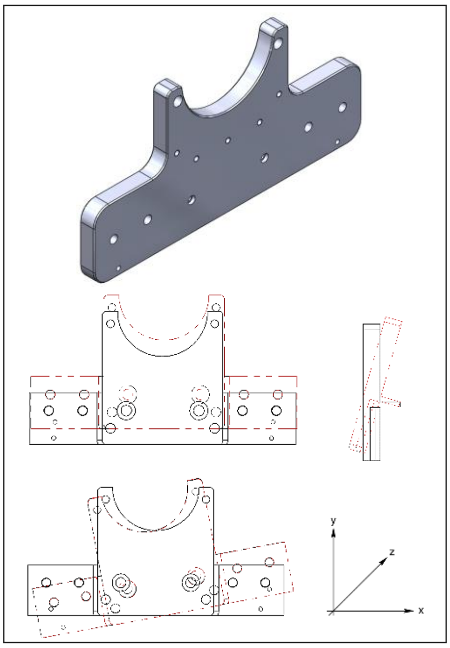


Figure 40: Motor Mount

V.4.3.3. Position pins

[Annex 3.4]

the Position pins are cylindrical components with a pointed tip used to secure an accurate alignment of the motor mount on the frame.

The pins (1) have threaded ends that allow them to be screwed into threaded hole (2) in the frame (3) and fixed with a nut (4) . This method provides a strong and adjustable connection, as the pin can be easily dismounted if needed. additionally, the pin's diameter is slightly bigger then the recipient hole (5) located on the motor mount (6) which creates an interference fit .

To achieve this mechanical property, the tolerance value of the pin diameter and the recipient hole is: ***H7m6.*** Since The nominal diameter of the pin is d= 10 mm. with an m6 tolerance therefore, the max and min tolerance values are: $10^{+0.015}$ and $10^{+0.006}$ with concentricity value of 0.01 mm to insure the coincidence of the axes between the shoulder of the pin and the rest of its body

The pin contains a small pitched thread because they are more suitable for the operations with important vibration and movement.

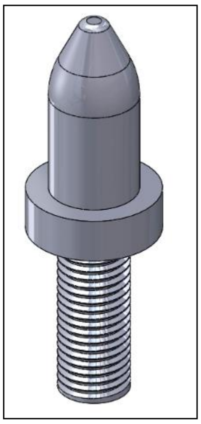


Figure 41: Position Pins

- ***The Recipient Hole***

The nominal diameter of the pin is d= 10 mm. with an H7 tolerance therefore, the Max and min tolerances values are: $10^{+0.015}$ and $10^{+0.006}$ while the big hole is used to accommodate the shoulder of the pin. The accurate positioning of the pin holes on the moto mount is vital. Consequently, a symmetric tolerance of 0.02mm is included to accomplish the accuracy.

V.4.3.4. Shoulders

[Annex 3.5]

The shoulder aims to help in aligning and positioning the motor mount on the frame, distributing the load, and limiting the unwanted rotational deformation.

The shoulder (1) is fixed on the Motor mount with fine pitch screws (2) . I have chosen this solution for a simpler machining for the motor mount.

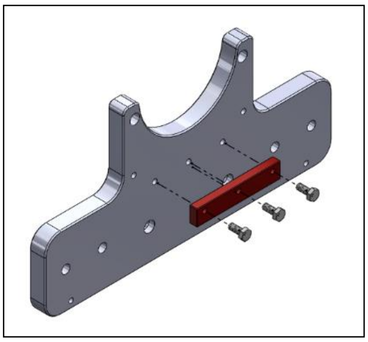


Figure 42: Shoulders

V.3.3.5. Support beams.

[Annex 3.6]

The ground reaction on the wheel creates a moment of force exerted on the motor mount, which causes the bending of both the motor mount and the beams of the frame. For this reason, a structure made of two beams is used to prevent bending.

Consequently, The support beam is a simple structure composed of a ***UPN60*** beam welded to rectangular metallic sheet on both sides and fixed to the motor mount using bolts on both sides.

Figure 43: Support Beam

A. *External Loads*

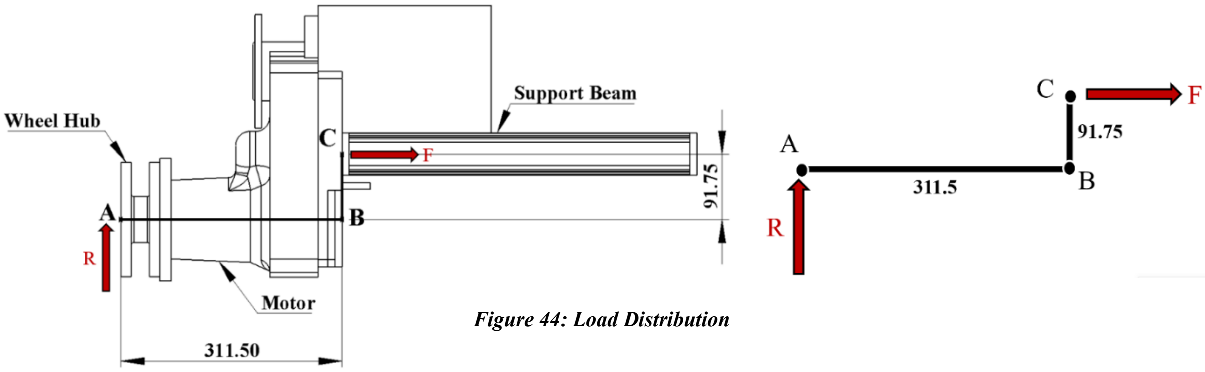


Figure 44: Load Distribution

- R: it represents the reaction force created by the ground on the wheel

$$R= \frac{1}{4} \times m_T \times g \; = 0.25 \times 694.96 \times 9.81 \;\; = 1704 \text{ N}$$

- F: it represents the resultant force applied on the support beams. To determine its value, I will represent it in the form of 2 two perpendicular beams fixed to the sides.

The moment created by the reaction R on the point B is $M_B = R \times 311.5 \; = 530.8$ Nm

Since the beams are considered rigidly fixed, the moment M_B is also the moment experienced in point C. there for $M_B = Mc$

$$M_B = 0.09175 \times F$$

$$F = \frac{MB}{0.09175} = 5785.28 \text{ N}$$

Because I will use two parallel beams, every beam will experience a force of: F/2 = 2892.6 N on each side .

A. *Assumptions*

- criterion: Von Mises criterion.
- Material : S355J2G3. Consequently, σ =315 Mpa
- Factor of safety: FOS=3. Consequently, $\sigma_{allowable} = 105$ Mpa

B. *Meshing & Boundaries*

To simplify the analysis, I will split the beam into two and perform symmetrical analysis, fixing the half beam from one end and apply the forces on the other end to perform a compression test.

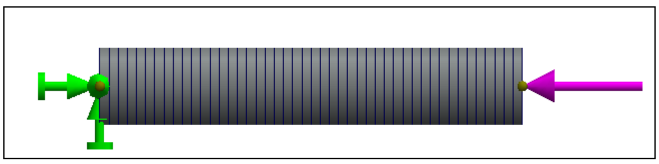


Figure 45: Support Beam Meshing

C. *FEA Results*

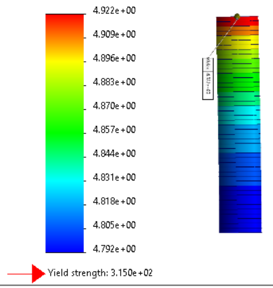
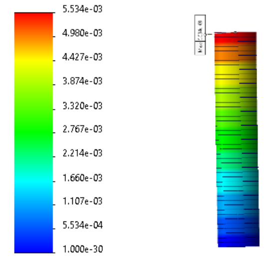
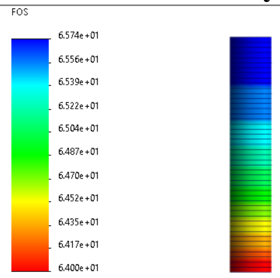
Stress Analysis	Displacement Analysis	Factor of Safety
		
The maximum obtained Yield strength value is 5 Mpa < 105 Mpa ►The analysis is verified.	The Maximum displacement value is 0.005mm ► The result is comfortably acceptable.	The minimum FOS is 64> 3 ►The structure is totally safe

Table 19: Support Beam Simulation

V.3.3.6. Wheel Hub

[Annex 3.7]

In order to transfer the rotational movement produced by the motor to the wheels, an intermediary hub is to be used. The wheel hub holds the wheel to the mechanism, and it supports the torque force.

The wheel hub is composed of three connected discs, a wheel mounting disc (1), a Motor mounting Disc (2) each with a specific number and dimension of fixation holes . in addition to an intermediary disc (4) . while an external shoulder (5) aims to centralize the wheel.

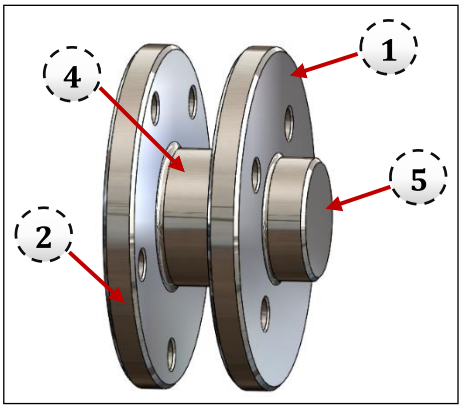


Figure 46: Wheel Mount

A. *Material Selection*

The wheel hub is subjected to important external forces, consequently its material should be strong, durable and with a good machinability for easy shaping operations.

Taking the previous requirements into consideration**,** the XC48 steel makes the preferable choice. It is made of 0.5 % carbon which Improves the yield strength.

B. *Improving Material Properties*

After machining the Wheel hub, further mechanical properties improvement is preferable. To assure that, the XC48 is treated to its crystallization temperature to change the crystal structure.

Because the aim is to increase the toughness and hardness, the heat treatment that I am going to do is called "Hardening."

The hardening process is composed of two steps.

- ✓ Austenitizing: This is the first step where the metal is heated to a temperature where its structure becomes austenite.
- ✓ Quenching: Immediately cooling the part into the water

This mechanical process guarantees a more reliable properties that can be anticipated as follow:

Material	Tensile S N/mm^2	Yield S N/mm^2	Young's N/mm^2	Poisson's ratio	Density Kg/m^3
XC48 Steel	800	570	210000	0.28	7850

Table 20: Wheel Hub Material Selection

C. Assumptions

- criterion: Von Mises criterion.
- Material : XC48 steel. Consequently, σ =570 Mpa
- Factor of safety: FOS=3. Consequently, $\sigma_{allowable}$ = 190 Mpa

D. Boundaries. External Loads & Meshing

the boundaries and The reaction forces are applied on the wheels mounting bolt holes, meanwhile the torque is applied on the motor side of the hub.

I have applied mesh control to optimize the mesh quality and increase elements number in the bolt holes and the intermediary disc where I suspect they well bear the most load.

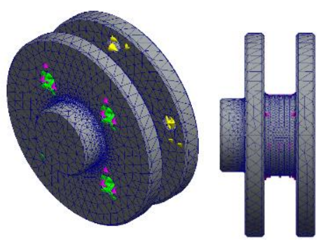
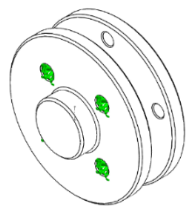
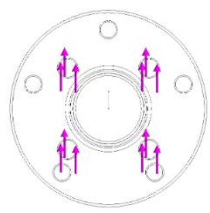
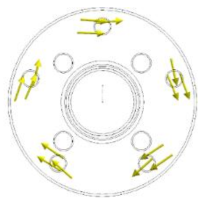
Meshing	Bounderies	Reaction forces	Torque
			

Table 21: Wheel Hub Meshing and Loads Distribution

E. FEA Results.

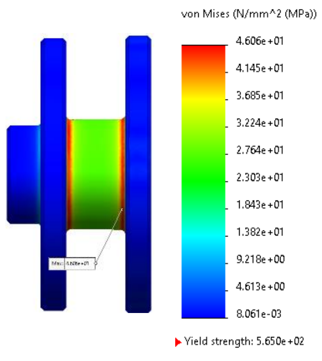
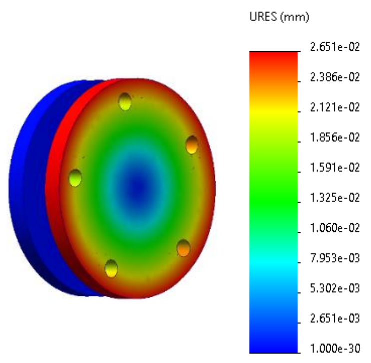
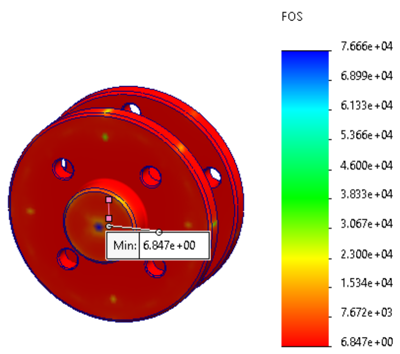
Stress Analysis	Displacement Analysis	Factor of safety
		
The maximum stresses 46 Mpa < 190 Mpa ►The stress analysis is verified.	The highest displacement value is 0.026 mm. ►The result is acceptable.	The minimum FOS is 6.8 > 3 ► The structure is totally safe

Table 22: Wheel Hub Simulation

V.4. Steering mechanism

The steering mechanism plays a crucial role in controlling the direction and maneuverability of the Cart. In this setup, the front wheels are designed as the steering wheels, capable of rotation around vertical axes known as pivots.

V.4.1 FAST Diagram

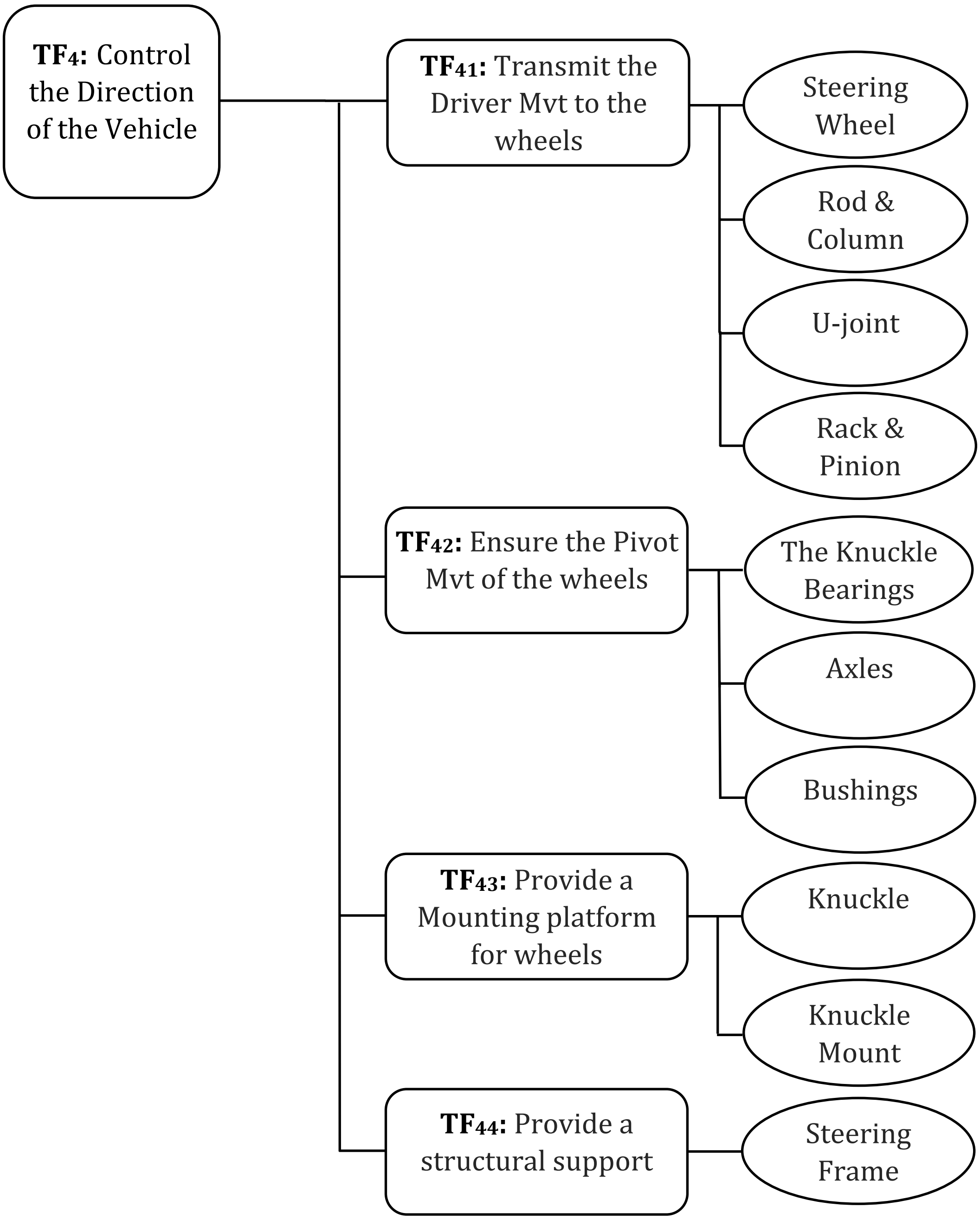


Figure 47:FAST Diagram of Steering Mechanism

V.4.2.Conceptual Design

V.4.2.1. Simplified kinematic diagram.

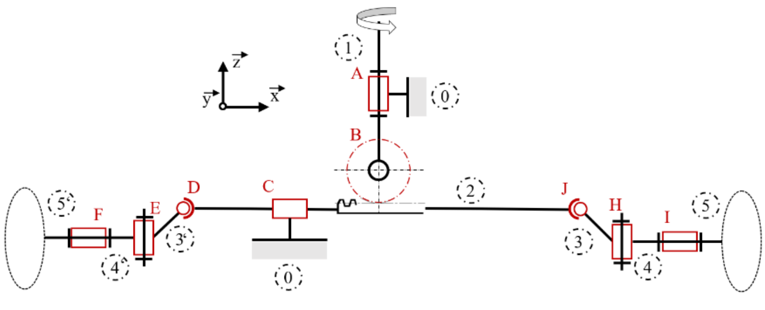
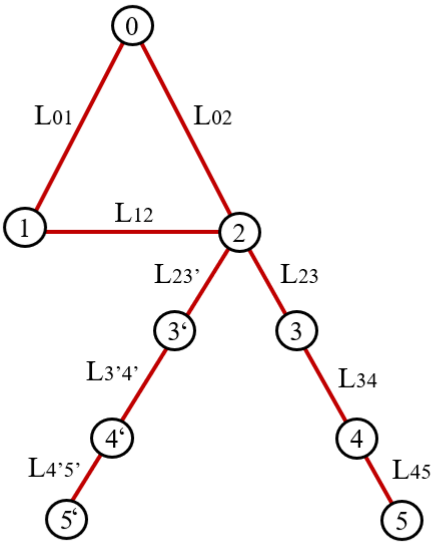
<table>
<tr><td>Equivalence Classes</td><td></td><td>-{0}: Cart Body
-{1}: Steering Column
-{2}: Rack and pinon
-{3}: Rod 1
-{3'}: Rod 2
-{4}: knuckle Mount1
-{4'}: knuckle Mount 2
-{5}: Steering Knuckle 1
-{5'}:Steering Knuckle 2</td></tr>
<tr><td>Connections graph</td><td></td><td>-L_{01}: pivot link, Center A, axial $\vec{y}$
-L_{02}: pivot link, Center C, axial $\vec{y}$
-L_{12}: Rack and Pinion link, Center B
-L_{23}: ball joint, Center J
-L_{34}: pivot link, axial $\vec{z}$
-L_{45}: Pivot Link, axial $\vec{x}$
-$L_{23'}$: ball joint, Center D
-$L_{3'4'}$: Pivot link, axial $\vec{z}$
-$L_{4'5'}$: pivot link, axial $\vec{x}$</td></tr>
</table>

Table 23: Kinematic Diagram

V.4.2.2. Ackerman steering mechanism.

The problem that I am trying to solve is the lack of harmony between the 4 wheels of the cart while taking a turn, Because each wheel has its own turning radius and distance to cover which causes the slipping of the cart during the turning process.

The outer wheel 1 travels the longest distance, while the inner wheel 4 travels the shortest distance, which creates problems in terms of velocity, making the front wheels go faster than the rear wheels.

To fix this issue, Ackerman steering mechanism provides a geometric arrangement of linkages designed to make the turning angle of the inner wheel greater than the turning angle of the outer wheels and thus preventing the slipping of the wheels when making a turn.

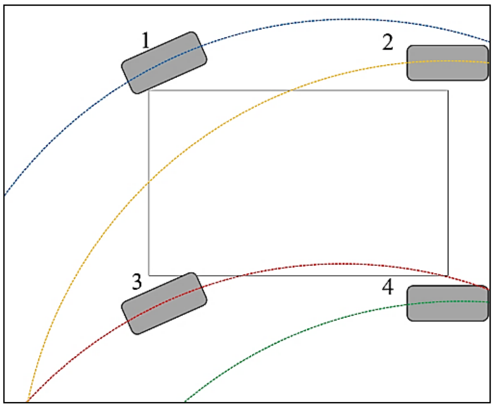


Figure 48: Wheels Turning Radius

According to The Ackerman steering mechanism, The 4 wheels must have the same rotational center point “**I**”.

- Track: Represents the side distance between the 2 wheels.
- L:Wheelbase: Represents the front and rear distance between the wheel’s centers.
- α:The angle of rotation of the inner wheel
- θ:The angle of rotation of the outer wheel

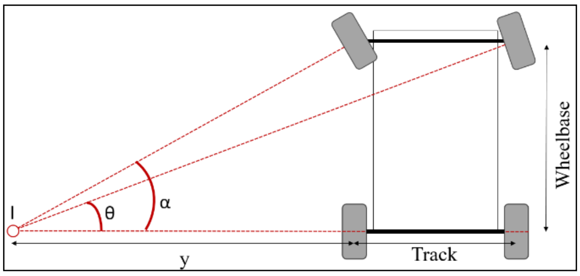


Figure 49: Wheels Turning Angle

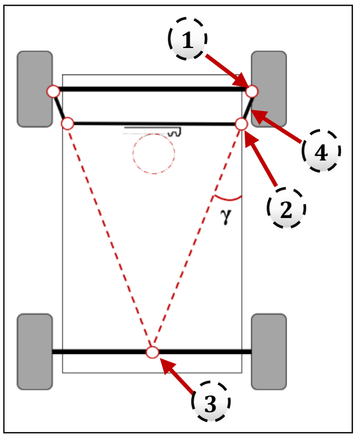


Figure 51: Steering Knuckle Angle

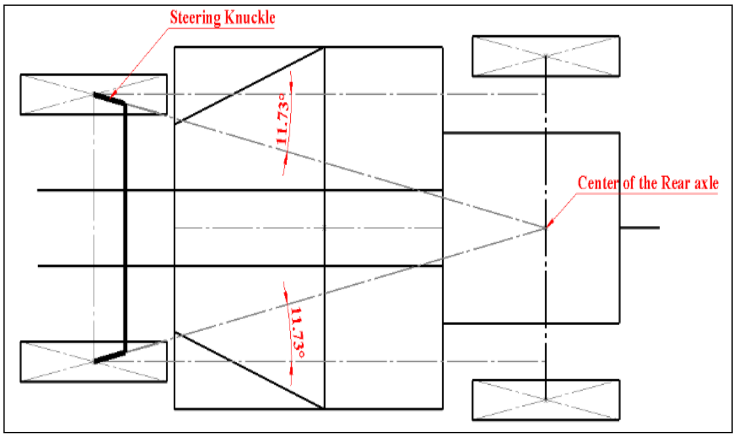


Figure 51: Layout of the cart Frame

In order to achieve the needed harmony, Ackerman states that aligning the pivot point of the wheel (1) with the pivot point of the steering rod (2) and the center point of the rear axle (3), will provide a perfectly functioning steering mechanism. thus, I will utilize the steering Knuckle mount (4) with an angle **γ** which will function as the pivotal base for both the wheel and the steering rod. ***[Figure 51]***

In order to determine the angle **γ** mentioned previously, I have used the layout of the frame of the tractor, After fixing the distance between the front and rear axles. ***[Figure 50]***

V.4.2.3.Components of the steering mechanism

[Annex 4.1 & Annex 4.2]

The steering system is composed of various components that aims to convert the driver's input into controlled movement of the front wheels.

When the driver turns the steering wheel (1), the rotational motion is transmitted through the steering column (2) and u-joint (3) to the rack and pinion mechanism (4). This rotation is then transformed into lateral movement directed to the steering knuckle (8) and its mount (7), causing them to pivot around the axle (5) with the guidance of the bushing (6). This setup allows the front wheels to turn smoothly in the desired direction.


Figure 52: Steering Mechanism

Simultaneously, the wheel (12) is bolted to the steering knuckle (8), which is fitted with bearings (9) to reduce friction. While The braking disc (10) is designed to accommodate the brake caliper (11). The entire steering system is mounted on the steering frame (13), which is securely fixed to the chassis of the cart .

V.4.3. Steering knuckle support

V.4.3.1.Mechanical Design

[Annex 4.3 & 4.4]

The function of the Steering knuckle support is to transfer the linear motion of the Rack & pinon to an angular motion of the wheel while accommodating the steering knuckle.

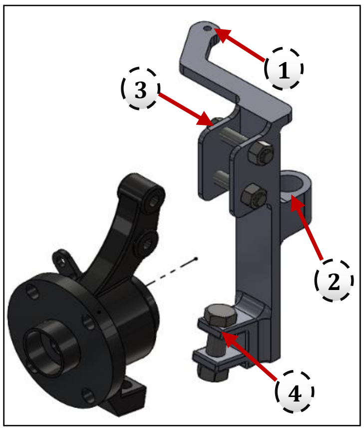


Figure 53: Steering Knuckle Mount

The angle of the steering knuckle mount is set to be ***11.73°*** with a tolerance of 0.05mm due to the importance of accuracy. While The shape and size of this mount depends on the supported weights , the design of the steering knucke and the location of its fixture points.

The hole located in the arm (1) accommodates the rack and pinion rod which is responsible for generating the linear force required to pivot the support . the shape and dimensions of the arm shape are designed to ensure a precise angle of "11.73" degrees between the hole (1) and the axle housing (2) which represents the linkage and pivoting point of the steering knuckle mount to the main structure, while. The housing points (3) and (4) are dedicated to secure and provide structural support for the steering knuckle .

The hole (2) is the pivot point of the axles, it will accommodate the bushings responsible for facilitating the rotation of the steering knuckle support around the axles.

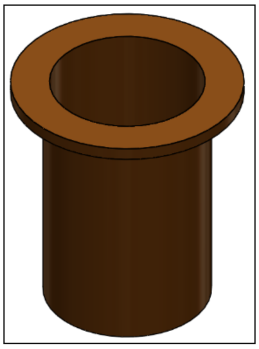


Figure 54: Bushing

Consequently, The tolerance between the bushing and its housing should be H7m6 to secure an Interference fit of the bushing inside its housing . At the same time, the interior diameter of the bushing should secure a clearance fit , therefore the tolerance between the bushing and the axle is H8f7.

- **Rotation Angle**

In order to determine the maximum angle of rotation of the wheels, I need to calculate the maximum linear course made by the rack and pinon system.

Upon reviewing its data sheets []; the rack and pinion system that I am going to use have a course of C= 250 mm, with a minimum length of 270mm and a maximum length of 420 mm on each side.

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$$C = 250\ \text{mm} = \alpha \times 122.57$$

$$\alpha = \frac{C}{122.58} = 2.03\ \text{rad}$$

To convert the angle from radians to degrees:

$$\alpha = 2.03 \times \frac{180}{\pi} = 116.3\ °$$

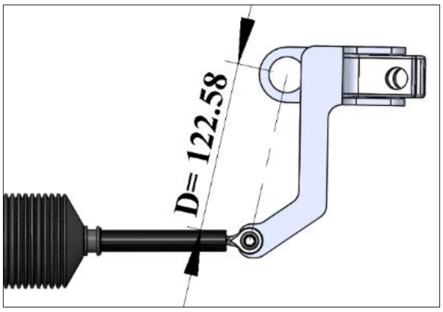


Figure 55:Rotation Angle

- **Turning Radius**

$$R = \frac{L}{Tan(\alpha)} = \frac{1.630}{Tan(2.03)} = 2.45\ \text{m}$$

V.4.3.2.FEA Simulation

A. Assumptions

- criterion: Von Mises criterion.
- Material : XC48 steel. Consequently, σ =570 Mpa
- Factor of safety: FOS=3. Consequently, $\sigma_{allowable}$ = 190 Mpa

B. Boundaries, External loads & Meshing

the support of the steering knuckle should be able to withstand the vertical ground reaction applied by each of the front wheel's R =2979 N which creates a pressure P.

$$P = R / A = 2979 / 1175 = 2.5\ \text{Mpa}$$

I have employed the mesh control on the locations where the forces are applied to increase the number of elements and enhance the analysis result.

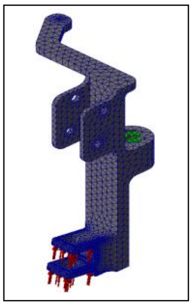


Figure 56: Meshing of the Steering Knuckle Mount

C. FEA Results

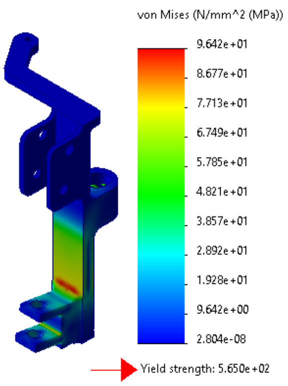
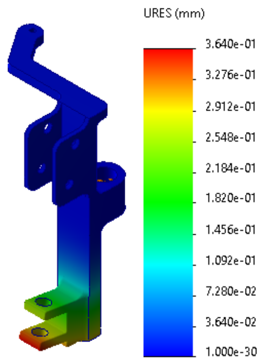
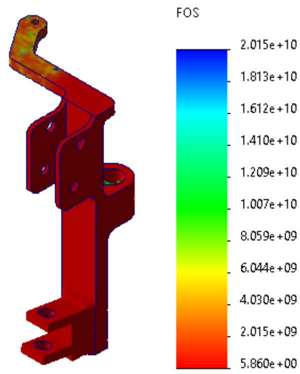
Stress Analysis	Displacement Analysis	Factor of safety
		
The maximum stress is 96 Mpa < 190 Mpa ► The analysis is verified.	The highest displacement values is negligible 0.3mm ►The result is acceptable.	The minimum FOS is 5.8 > 3 ► The structure is totally safe

Table 24: Steering Knuckle Mount Simulation

V.4.4. Steering knuckle

V.4.4.1.Mechanical Design

[Annex 4.5]

The steering knuckle is the crucial part of the steering system as it accommodates the bearings responsible for the rotation of the wheels and it contains the mounting ports of both the wheel the braking disc.

Due to its manufacturing complexity and the unavailability of suitable machinery, I will be using a pre-made steering knuckles of a FIAT UNO car for both the left and right sides . nevertheless, I'll need to make some modifications by removing the appendages of these steering knuckles since I won't be using them. ***[Figure 57]***

Using the necessary measurement tools, I took the overall dimensions of the steering knuckle, excluding the appendages and properly determining the locations of the housing holes . it included (1) and (2) for fixing the steering knuckle mount, (3) for securing the wheel and braking disk, (4) for attaching the braking calipers and (5) for bearing accommodation. ***[Figure 58]***

Figure 58: Steering Knuckle

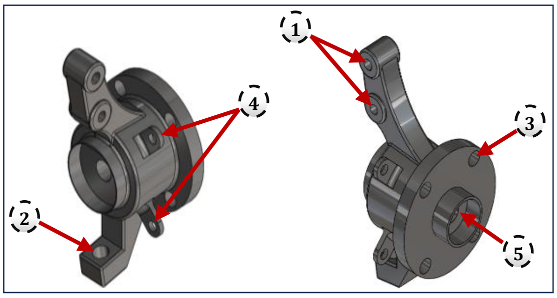


Figure 57: Steering Knuckle Design

V.4.4.2.Bearings

[Annex 4.6]

The steering knuckle accommodates 2 Deep groove bearings already selected and installed . Consequently, there is no need to conduct a bearing selection study. The bearings model is R600-5 , this type of bearings support both radial and axial loads .

V.4.5. Steering frame.

V.4.5.1.Mechanical Design

[Annex 4.7 - Annex 4.8]

To ensure the coordination of the steering mechanism components, I designed a steering frame in order to house and integrate the different parts of the steering mechanism.

The steering frame (1) is bolted to the front of the cart chassis . It is mainly made of welded UPN80 beams to ensure a good bending resistance as it is submitted to the pression created by the weight of the cart , while the welded shoulders (2) prevent lateral sliding of the structure and also providing accommodation for the fixation points. The steering frame also serves as a pivot base for the steering knuckle mount through the axle housings (3) on its sides

The axle housing (3) has a diameter of D=26mm, and since it accommodate the axles in a static position, the tolerances must ensure interference fit with the axles. Therefore, the recommended tolerance is ***H7.*** Using the tolerances Table [] , $D=26^{+0.021}$

V.4.5.2.FEA Simulation

A. Assumptions

- criterion: Von Mises criterion.
- Material : XC48 steel. Consequently, σ =570 Mpa
- Factor of safety: FOS=3. $\sigma_{allowable}$ = 190 Mpa

B. External forces and Meshing

the Structure should be able to withstand half of the weight of the cart applied on the upper face of the middle part of the structure

$$W = m \times g = 5516 \text{ N} .$$

Consequently: $$P = \frac{W}{A} = \frac{5516}{4050} = 1.36 \text{ Mpa}$$

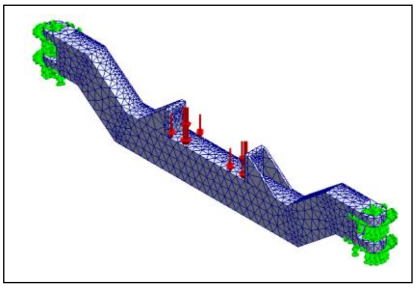


Figure 59: Steering Frame Meshing

C. FEA Results

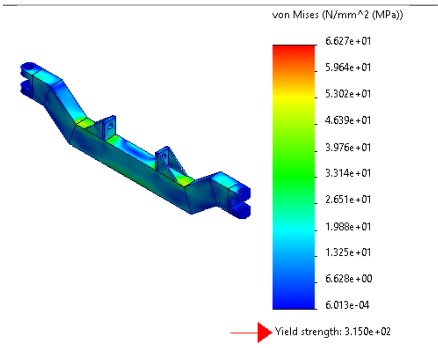
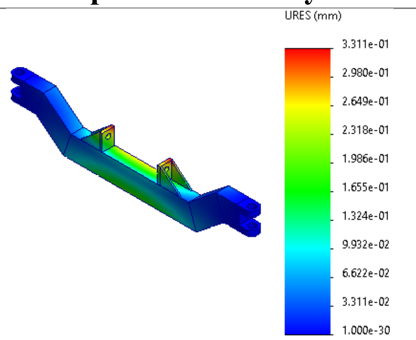
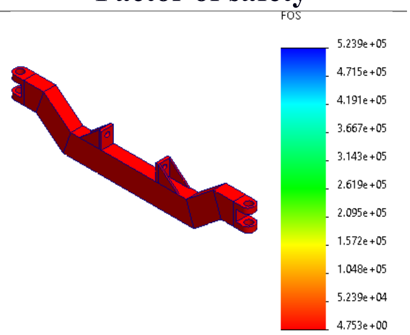
Stress Analysis	Displacement Analysis	Factor of safety
		
The highest stress value is 66 Mpa < 105 Mpa ► The analysis is verified.	The Maximum displacement value is acceptable (0.3 mm) ►The result is acceptable.	The minimum FOS is 4.7 > 3 ► The structure is totally safe

Table 25: Steering Frame Simulation

V.5.Braking System

The perfect functioning of brakes in all conditions is a necessity for the safety of the driver and the transporting operation, it works on the principle of conservation the Kinetic Energy to heat which is then dissipated to the surrounding.

V.5.1.FAST

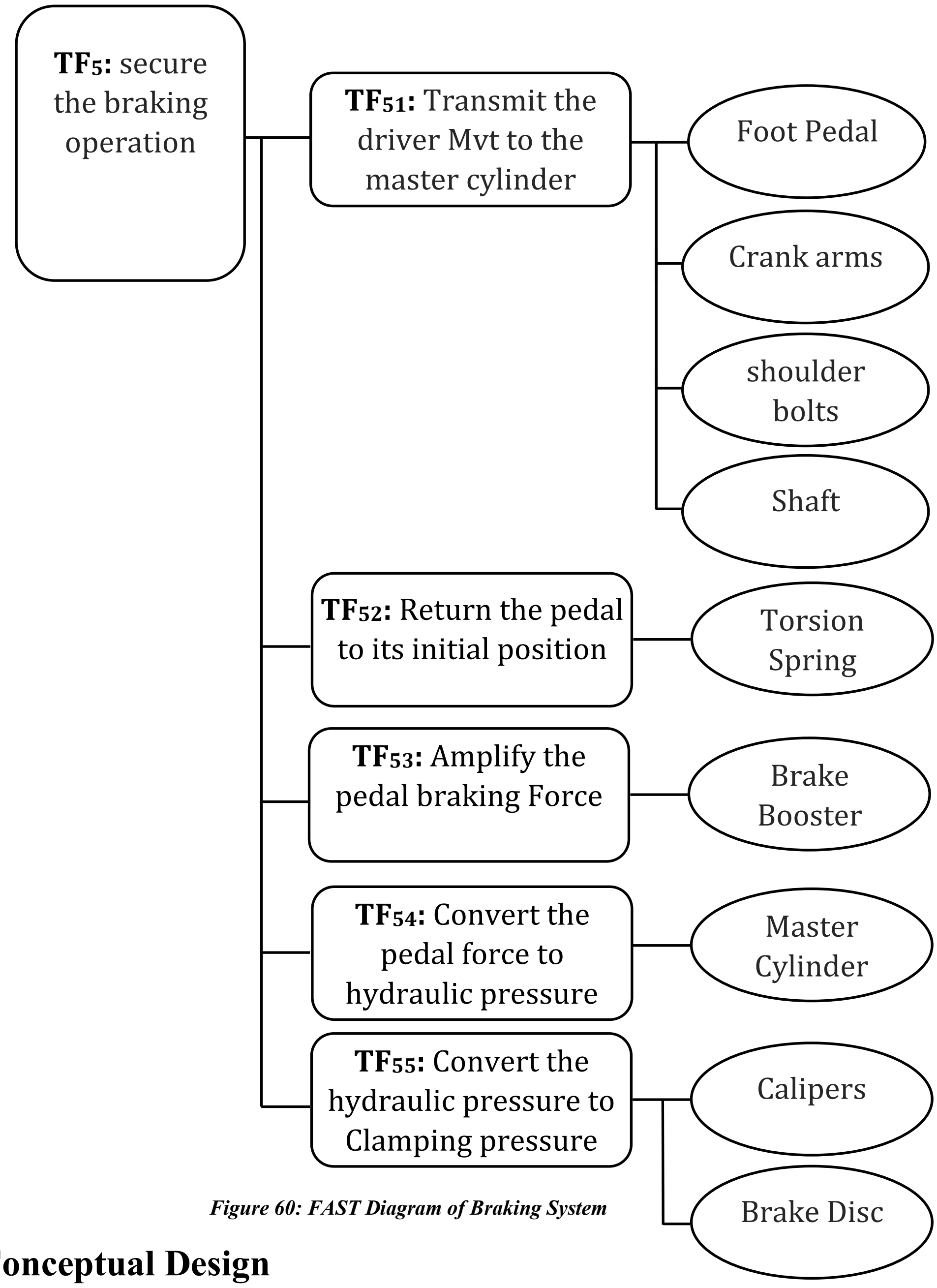


Figure 60: FAST Diagram of Braking System

V.5.2.Conceptual Design

Several brake components are impossible to manufacture in the Division , consequently, I have chosen to use some pre-made components of a FIAT UNO car. And my mission is to design a structure that ensures the compatibility of these recovered components with the traction vehicle.

The Recuperated pre-made components are [Annex] :

- Calipers : The piston diameter of calipers is 48mm
- Master cylinder: piston diameter is19.05mm , Stroke length is: 30.48 mm
- Brake Disc: The disc rotor is mounted on the wheel, and it is bolted with the wheel and rotates along with it , it has a diameter of 227mm and thickness of 10.7 mm
- Brake Booster : the diameter of the vacuum brake booster servo : 158.5 mm.

V.5.3.Components and Functioning

[Annex 5.1]

The design of the braking system structure should be feasible to ensure its easy and proper functioning and assembled with the chassis in such a way that is easy to access.

The braking system involves various components that translates the driver's input into braking force of the front wheels.

As the driver apply force on the pedal (1), this linear motion is converted into angular motion to the shaft (2) through the pivot points connecting the crank arms (3,4 and 5). The rotation of the shaft causes the rotation of the crank arm (6) as it is bolted to the shaft . This angular movement of the crank arm is then transformed into linear motion that presses the brake booster (7) and the master cylinder (8) which are fixed on their mount (9), causing the braking force to be amplified and then converted to hydraulic pressure that flows through the hose (10) to the caliper (11) Causing The caliper to apply a compressive force to the brake disc (12), resulting in the disc coming to a stop. The mechanism is then returned to its initial position using the tension spring (13) when the pedal is released .While the whole braking mechanism is mounted on the front Body structure on the fixation points (14,15)

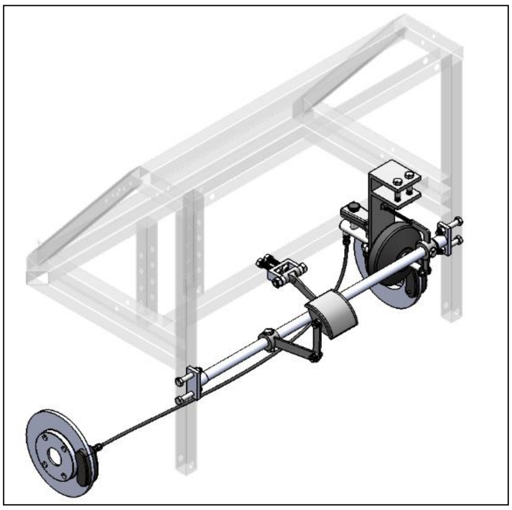


Figure 61: Steering System

V.5.2. Crank arms, Shaft & Foot Pedal

[Annex 5.2 - Annex 5.4]

The crank arms are part of the mechanical linkage that translates pedal action into a rotational movement required to engage the booster and master cylinder of the brake. The braking system that I have designed contains four crank arms each with specific length, shape, and pivoting center in order to deliver the appropriate output.

The Rotation Movements of the Crank arms(1) and (2) are secured with shoulder screws (3) that forms the pivot axis while whashers (4) and Nuts (5) secure the linked components in place.

In the other hand The angular movement of the pedal (6) and crank arm (7) is ensured with a pivot system composed of the axle (8) passing through both the crank arm and the mount (9), while to the bolts (10) and nuts (11) in order to Preventing the Loosening.

Additionally, the shoulder bolts (12) and (12') with their fixation bolts, nuts and washers secure the pivoting of the Foot pedal. The remaining crank (13) is a dual-arm crank attached to the shaft (2), which is secured to the front structure of the cart at two mounting points (14).

V.5.3. Master Cylinder & Brake Booster

[Annex]

V.5.3.1. Master Cylinder

The master cylinder mission is to convert the mechanical force delivered by the driver on the brake pedal to a hydraulic pressure on the brake disc .

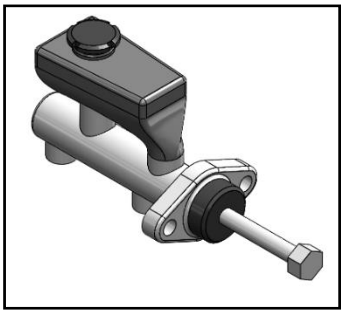


Figure 62: Master Cylinder

The master cylinder consists of an oil reservoir and two separate cylinders equipped with pistons. once the brake pedal is pushed, each cylinder deliver the oil pression to one of the brake calipers. When the pedal is released, two return springs inside the cylinders push the pistons back to their initial position.

V.5.3.2. Brake boosters

The brake booster aims to enhance the braking operation by amplifying the torque applied on the pedal which leads to less required physical effort of the driver.

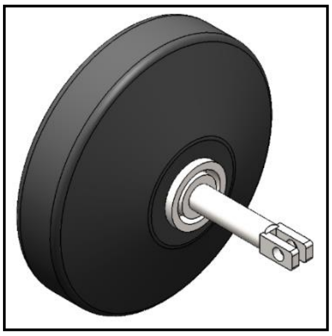
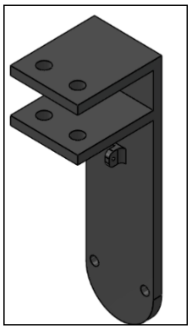


Figure 63: Brake Booster

The brake booster is composed of two separate chambers, when the pedal is pressed, a pressure difference is created between both chambers of the brake booster causing an ambient air to flow into the rear chamber, creating higher pressure transmitted to the master cylinder.

V.5.4. Master Cylinder Mount

Figure 64: Master Cylinder Mount



[Annex 5.5]

The mount is bolted to the frame of the cart with the mission of securing the master cylinder and the Brake booster in position . additionally, it functions as a Spring Retainer to keep the spring in place, preventing it from moving out of position.

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V.5.5.Brake Performance Analysis

V.5.5.1.The Leverage Pedal Ratio

The pedal ration refers to the effort exerted by the driver on the Foot pedals while the aim is to facilitate the operation by decreasing the effort and increasing the output force.

To calculate the Leverage ratio, I need two values :

- L_1 (160 mm): distance from the pedal pivot point to the point where the driver's foot applies force.
- L_2 (70 mm): The distance from the pedal pivot point to the point where the pushrod connects to the master cylinder.

The leverage ratio: $L_R = \frac{L1}{L2} = \frac{160}{70} = 2.3$

This means that the force exerted by the driver's foot is amplified by a factor of 2.3. Given that The average Force applied by the Driver on a brake pedal is around 500N. this force is not perpendicular to the crank bolted on the shaft. Consequently, I will be calculating the projection F_y of that force . $\theta = 42°$, $F_y = \sin(\theta)\times F$

$$F_y = \sin(42)\times 500 = 334.5 \text{ N}$$

taking this value into consideration , it is possible to calculate the amount of force generated by the driver that will reach the shaft . $F_D = 334.5 \times 2.3 = 769.35$ N

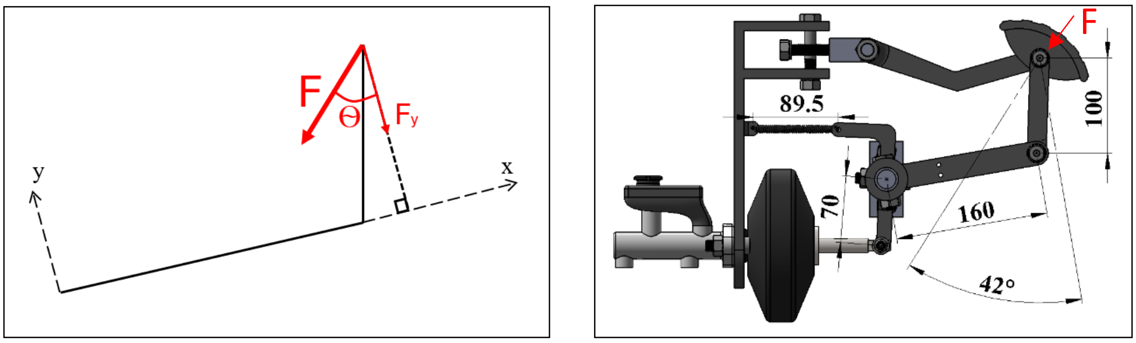


Figure 65: Foot Pedal Force

V.5.5.2. The Force Amplified by the Brake Booster

The diameter of the vacuum brake booster servo is 158.5 mm. which makes the area :

$$A = D^2 \times \frac{\pi}{4} = 0.0197 \text{ m}^2$$

Assuming that the vacuum inside the brake booster is perfect (0 Pa) while the atmospheric pressure is 101 Pa. Consequently, The pressure difference is : $\Delta P = 101$ Pa

Having the pressure and the area values, it is possible to determine the Force value generated by the brake booster :

$$F_{BB} = \Delta P \times 0.0197 \text{ m}^2 = 1996 \text{ N}$$

Consequently, the Total Amplified force is:

$$F_{Amplified} = 1996 \text{ N} + 769.35 \text{ N} = 2765.35 \text{ N}$$

V.5.5.3.The pression Generated by the Master Cylinder

The diameter of the master cylinder piston is 0.019 m , which makes the area of the piston:

$$A_{MC} = (\pi/4)\times 0.019^2 = 2.8 \,.\, 10^{-4} \text{ m}^2 .$$

There for, the pressure applied by the Master cylinder and the caliper is $P_{MC} = \frac{F_{Amlified}}{A_{MC}}$

$$P_{MC} = \frac{2765.35}{0.0028} = 9.87 \text{ Mpa}$$

V.5.5.4.The Clamping Force Generated by the calipers

The diameter of the Caliper piston is 0.048 m, which makes the area of the piston of the caliper $A_{Caliper} = \frac{\pi}{4}\times 0.048^2 = 1.8 \,.\, 10^{-3} \text{ m}^2$.

Consequently, the Mechanical force generated by the caliper is:

$$F_{Caliper} = P_{MC} \times A_{Caliper}$$

$$F_{Caliper} = 9.87.10^6 \times 1.8 \,.\, 10^{-3} = 17766 \text{ N}$$

The clamping Force : $F_{Calamping} = F_{Caliper} \times 2 = 17766\times 2 = 35532 \text{ N}$

V.5.5.6.The Frictional Force

The Brake pads of the Fiat Uno braking system are made of ceramics . Consequently, the coefficient of friction between the Rotor disc and the Brake Pads is 0.4

$$F_{Friction} = F_{Calamping} \times 0.4$$

$$F_{Friction} = 35532 \times 0.4 = 14212.8 \text{ N}$$

V.5.5.7.The Slowing Torque

the frictional force applied by the brake pads on the rotor Disc creates a torque that slows down the wheel as shown in the equation.

$$\text{Slowing Torque} = \text{Frictional Force} \times \text{Effective Radius}$$

While the Effective Radius is the distance from the center of the Disc rotor to the point where the frictional force is applied by the brake pads. I have manually calculated this distance to be 90 mm. Consequently:

$$\text{Slowing Torque} = 14212.8 \times 0.09 = 1279.15 \text{ Nm}$$

V.5.5.8.The Total Braking Force

To calculate the total braking Force, I have first to calculate the braking force applied on one wheel knowing that Both the front wheels of the vehicle are of the same size with a Radius of 0.546m.

Consequently, The braking force exerted on one wheel:

$$F_B = \frac{\text{Slowing Torque}}{\text{Radius of wheel}} = \frac{1279.15}{0.546} = 2220.75 \text{ N}$$

The Total Braking Force

$$F_T = 2220.75 \times 4 = 8883 \text{ N}$$

V.5.5.9.The Resulting deceleration

The resulting deceleration is calculated using Newton's second law of motion: the force applied to an object is equal to the mass of the object multiplied by its acceleration.

Consequently :

$$D = \frac{\text{Total Braking Force}}{\text{mass of the vehicle}}$$

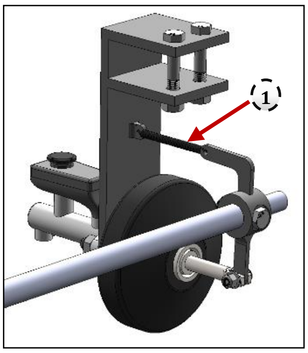
$$D = \frac{8883}{1150} = 7.72 \text{ m.s}^{-2}$$

V.5.5.10.The Braking Distance

The cart speed is previously determined : $V = 10.8 \text{ Km.h}^{-1} = 3 \text{ m. s}^{-1}$
Consequently, The Breaking Distance :

$$B_d = \frac{\text{Velocity}^2}{2 \times \text{Deceleration}} = \frac{3^2}{2 \times 7.72} = 0.58 \text{ m}$$

V.6.The Tension Spring selection



To choose the suitable extension spring (1) for the braking operation, I need to perform several calculations to ensure the spring meets your requirements of the braking system.

V.5.6.1.Spring Rate

The spring rate K is the force per length the spring exerts: $K = \frac{F}{\Delta L}$

F: is the same force transmitted from the pedals to the shaft, $F = F_D = 769.35$ N

ΔL : represents the deflection, which is the same as the master cylinder course, $\Delta L = 30.48$ mm

Consequently:

$$K = \frac{769.35}{30.48} = 25.24 \text{ N.mm}^{-1}$$

V.5.6.2.Deflection Ratio

It refers to the ratio of the change in length of the spring: $D_R = \frac{\Delta L}{L0} \times 100$

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L_0 : is the free length of the spring when it is not under any load or tension , L_0 = 89.5mm

Consequently: $D_R = \frac{30.48}{89.5} \times 100 = 34\ \%$

V.6.Battery Housing

Since the battery is an indispensable and delicate component for the cart, Without proper housing, it is vulnerable to damage and present a danger to the surroundings as well. Therefore, it requires housing primarily for protection and to provides a barrier against the exterior elements. Additionally, since the cart is a compact vehicle, I am going to exploit the battery housing to integrate the driver seat and a hydraulic Tank to its structure.

V.6.1.FAST diagram

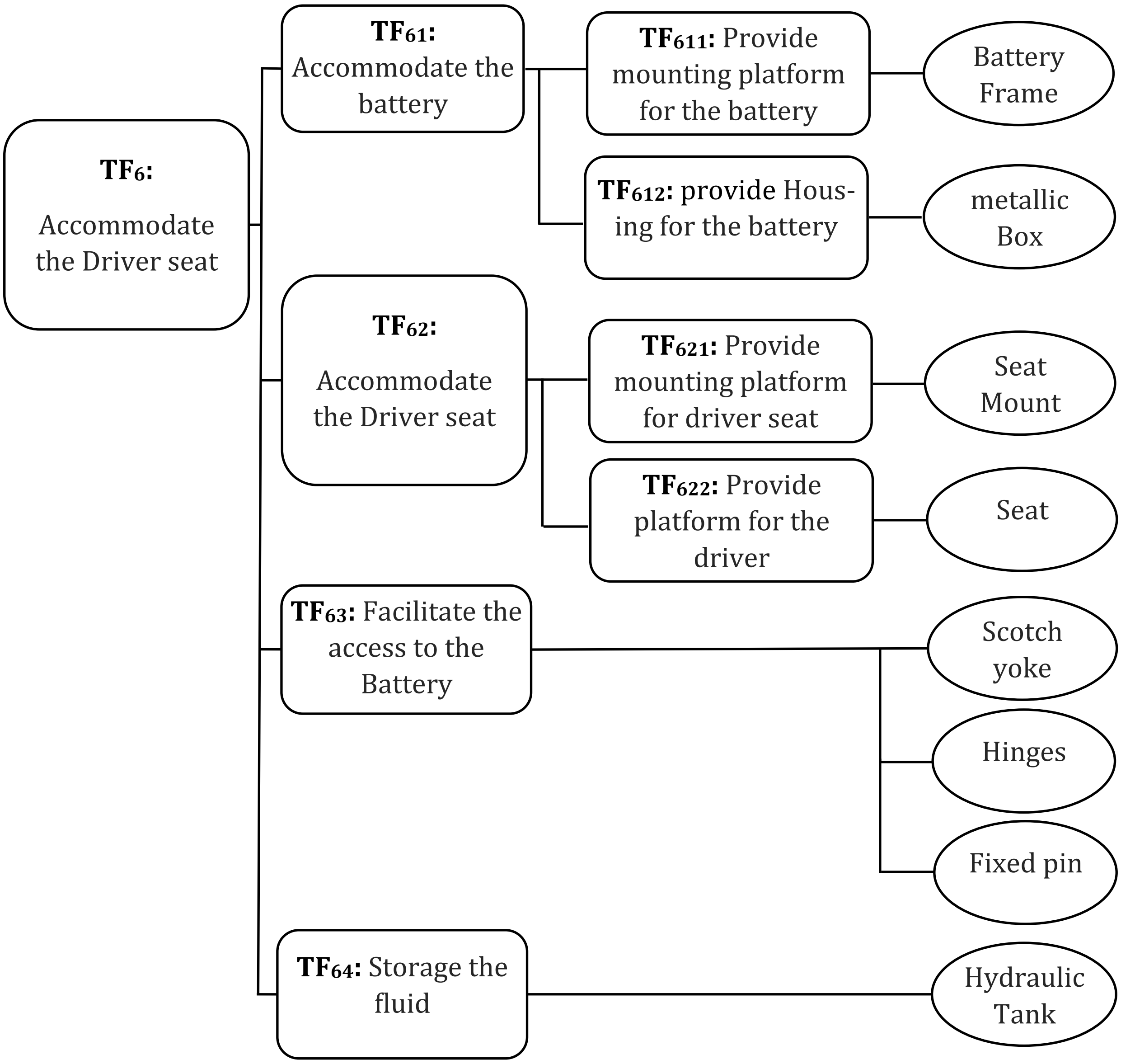

Figure 66: FAST Diagram of Battery Housing

V.6.2. Conceptual Design

A. Material : Since the cart is operating in an industrial condition, the battery housing should be strong to resist the impacts. Consequently, the exterior layer of the battery housing consists of steel sheet metal .

B. comfort: The main objective of this project is to provide assistance and comfort for the workers while displacing heavy loads, consequently the cart should offer comfortable seating.

V.6.3. Mechanical Design

V.6.3.1. Battery housing

[Annex 6.1]

The Battery is mounted on its support composed of corner beams and lodged in the battery housing made of welded sheet metals forming the exterior enclosure for the battery and also accommodating a hydraulic Tank on the front.

the placement of the driver's seat above the battery is intended to optimize space utilization and facilitate the driver control over the cart

Additionally, in order to enhance accessibility for the battery, the seat and the housing cover are capable to pivot to the side via hinges and scotch yoke mechanism and during closure it is locked in position using Latches

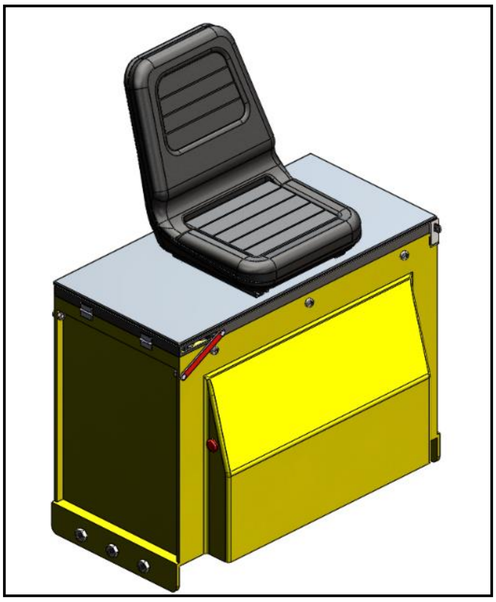

Figure 67: Battery Housing

V.6.3.2. Pivot mechanism

[Annex 6.2]

The pivot operation is assured with a scotch yoke mechanism that consists of Hinges, Fixed pin (1) , sliding pins (2), cranks(3), retaining rings(4), nuts(5), and washers(6).

Opening and closing the battery cover requires manual action. The fixed and sliding pins function as the pivot points for the crank. while the retaining ring, nuts, and washers secure the crank in place, preventing it from sliding during operation and allowing the cover to move along a 137 mm path resulting in a 167.61° angle .

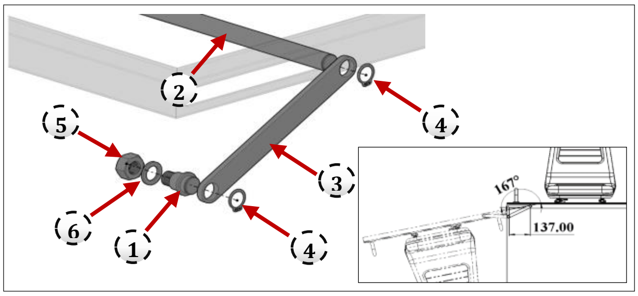

Figure 68: Pivot Mechanism

- ***Tolerance***

The fixed pins and the sliding pin represent the pivot axis of the crank, which means that the adjustment should guarantee rotational movement. Consequently, the tolerance is 15 H8 f7.

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V.6.3.3. The remaining ingredients

[Annex 6.3 - Annex 6.8]
The mechanical design of the remaining parts is detailed in the drafts included in the annex.

V.6.4. Seating comfort

The measurements are taken with a human body of an average Height of 1.8m which is 5% higher than the average height in Tunisia for men 1.71m []

V.6.4.1. reference points.

The reference points are used to define the driver's position while inside the tractor.

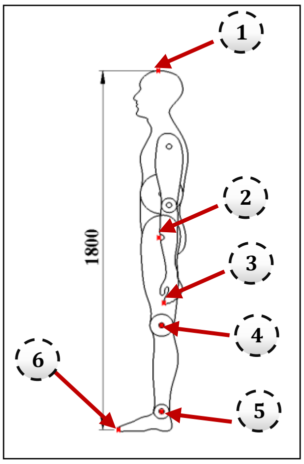


Figure 69:Reference Points

1. Top head position 4. Knees pivot
2. Hip pivot 5. Ball joint Foot
3. Arm reach 6. Foot phalanges

After defining the reference points of the driver on the seat, all other interior components and instruments inside the cart need to be arranged accordingly and to provide sufficient clearance dimensions around the operator.

V.6.4.2.Arrangement.

The arrangement should provide:
- Legroom: A sufficient space for keeping legs of the driver in a comfortable position
- Headroom: The vertical clearance space above the head should be minimum 50 mm
- Hand reach: The control elements should be easy to reach for the driver.

V.6.4.3.The result

The Designed cart offers sufficient head room of 88.36m, a 502.5mm legroom and a maximum horizontal arm reach of 750.41mm. The provided dimensions are satisfactory. Consequently, the seat comfort is verified.

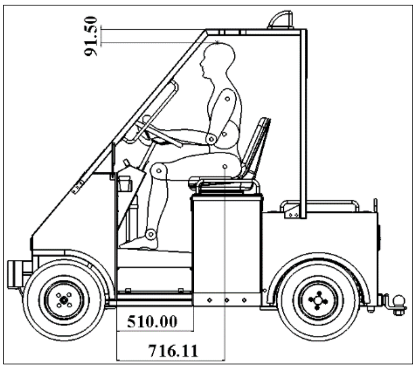


Figure 70: Seating position

V.7. Protective Cover

The cover of the cart serve as an enclosure and protection for the interior parts and structures in order to enhance its utility, comfort, and longevity and improve the cart's agronomics standards.

V.7.1. Conceptual design

V.7.1.1. Material selection

Choosing the right material for the cart's cover and enclosure involves considering factors like weight, manufacturability, and environmental resistance. After evaluating these aspects, it becomes evident that aluminum sheets are the optimal choice.

- ***Weight:*** Aluminum is a lightweight metal with density value 2.7 $g.cm^{-3}$ making it a suitable choice for our application since the weight reduction is crucial
- ***Manufacturability:*** Aluminum is highly manufacturable due to its excellent formability; given that the bending feature is a key aspect for our application
- ***environmental resistance:*** aluminum is suitable for harsh environmental conditions due to its oxide layer that has a high corrosion resistance.

V. 7.2. Mechanical design

V. 7.2.1. Front Cover

[Annex 7.1 - Annex 7.8]

The cover of the front structure is divided into four parts, the largest (1) covers the primary front structure, the second (2) wraps the bumper for ergonomic purposes, the third (3) creates an enclosure, and the fourth (4) serves as a platform for installing the control buttons and the steering wheel. The four segments are made of folded aluminum sheets and fixed with the proper bolts and nuts.

In order to exploit the empty space in the middle of the front body structure, I have decided to add a hood (5) which use the hinges (6) to open, allowing it to be easily lifted to access the interior and any installed components within.

Additionally, the hood incorporate safety features represented in some Attachments such as a Prop Rod holder used to secure its open position. The holder is a simple metal rod (7) that have a Holding shoulder (8) to hold the hood in place and a support (9) fixed on the main structure with the mission of assuring the rotation movement of the rod. Additionally, a rotational locking mechanism (10) is installed to secure the closure of the hood

Moreover, to prevent road debris, such as rocks, mud, and water, from being thrown into the air by the rotating tire I have added wheel fenders (11) above the wheel's fixation points

V.7.2.2. Rear Cover

[Annex 7.9 -Annex 7.11]

The rear cover (1) is made of folded aluminum sheets fixed on the rear structure with bolts. Additionally, wheel fenders (2) are also employed for the same reason previously mentioned.

V.7.2.3. Cover of the Canopy structure

[Annex 7.12]

To improve operator safety, I have included a metal canopy cover (1) to create a protective enclosure. As usual the cover is made of folded aluminum sheet and fixed on the Canopy structure with bolts .

V.7.2.4. floor pan

[Annex 7.13 – Annex 7.15]

The floor pan forms the bottom surface of the vehicle. It is a protective barrier between the operator and the road surface and as a Step board that facilitate the driving operation

The floor plan is made of steel sheets for better strength, and composed of two parts, The first part is the base (1) which serves as a step board and directly bolted to the vehicle's frame. The second part is the Foot platform (2) on which the control pedals are installed, ensuring they are within easy reach of the operator's feet and accommodates.

V.7.2.4. Under-Body Sield

[Annex 7.16]

The under-body shield is a folded sheet metal that extends on the whole lower part of the chassis in order to protect the inner components from dust , water and from road debris.

V.8. Hydraulic system

V.8.1. Software

The diagram is designed, configured, and simulated by FluidSIM software

V.8.1. Diagram

[Annex 8.1]

The actuators, such as cylinders and hydraulic motors, are installed on an external system other than on the cart itself as the cart's role is only to supply hydraulic flow and pressure to any other external hydraulic systems.

To simplify, I will add two cylinders to the diagram as an example of diverse types of actuators that can be employed . I will not specify the dimensions of any actuator (stroke, diameter.) since I am uncertain about the types that will be used in the future. The system will be designed to be upgradable and adaptable to accommodate the requirements of any external system.

The motor-pump unit (1) supplies a constant flow from the hydraulic reservoir. To protect the motor-pump unit, I have installed the check valve (2), whose main mission is to allow the flow to pass only in one direction.

In order to use the hydraulic system, it is essential to manually open the valve (3) while manually controlling The 4/3 control valve (4) with a lever to control the direction of the fluid flow in the circuit.

At the same time, The push button (11) is used to energize the motor pump and can be used as an emergency switch controlling a 4/2 control valve (5). while The pressure relief valve (7) and the check valve (8) represent a security counterbalance valve responsible for creating holding pressure in the cylinders to prevent the falling of the load while descending or ascending (depending on the usage), thereby preventing an over-running load.

Additionally, The 2-way flow control valve (6) is responsible for maintaining a constant speed in the actuators by controlling the fluid flow. Both of these mechanisms affect the loading and unloading speed of the actuators.

V.8.2. Simulation

[Annex 8.2]

I have installed two manometers in series with the actuators in order to visualize the pressure values on the actuators

The diagram analysis shows that the actuators takes more time in turning in the descendance compared to the ascending which is the result of the counterbalance mechanism assured by the pressure relief valve (7) and the check valve (8).

V.7.2. Mechanical Design

[Annex 8.3]

The mechanical design of the hydraulic system translates the schematic diagram into a physical configuration, illustrating how it should be installed within the chassis

The hydraulic components are installed in the interior empty space of the front structure while a metallic sheet serves as a mounting platform for the different hydraulic components while the fixation is secured with the proper bolts and nuts.

At the same time , the pump is bolted in the empty space under the foot platform and close to the oil reservoir integrated in the batter enclosure. While the Pipelines interconnect the different hydraulic components then it extends to the vehicle's rear, creating access points for any subsequent operation.

V.8. Conclusion

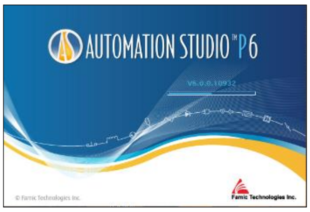
During this chapter, I have conducted a comprehensive mechanical design of the different cart components and mechanisms, building upon the constraints and needs established in the previous chapter. Additionally, I used Finite Element Analysis (FEA), to simulate the mechanical behavior and ensure structural integrity and performance. While performing a se-ries of calculations to validate the design parameters and optimize the overall functionality of the cart.

Furthermore, I have designed and simulated the hydraulical circuit employed in the cart.

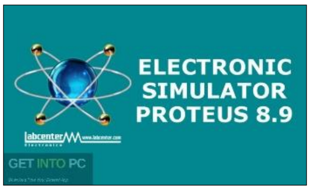
CHAPTER-VI

- Power Control Circuit
- Auxiliary Circuit

VI.1. Design software.



The Cart power Circuit is considered part of Power electronics, there for I will use Automation Studio 6, which is a design and simulation software containing a large library suitable for power electronics that allows the designing of electrical circuits, diagrams, and drawings



In the other hand I will be using Proteus software for the Design of the auxiliary circuit responsible on controlling the lightings and indicators of the cart

Additionally, will use Pvsyst for the dimensioning of the solar panel to provide additional electric power

VI.2.The power Circuit

the electric circuit plays a crucial role in powering and controlling the vehicle's electrical systems, ensuring efficient and reliable operation.

The initial step on the conceptual design is to outlines the necessary tools and equipment needed for our electric circuit

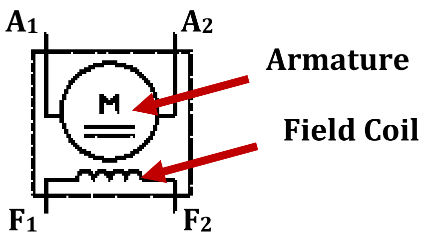
VI.2.1.The Required Instruments

[Annex : Electric Components]

VI.2.1.1.DC shunt Motor

Voltage	Current	Power	Rpm
24 V	104 A	2.0 Kw	2000 min^{-1}

Table 26: Motor Characteristics



The construction of a DC shunt motor is similar to other types of DC motor, as it contains the fundamental parts such as the stator Rotor and brushes. With the field coil as an additional element. There for it contains 4 terminals (A_1,A_2) for the armature and (F_1,F_2) for the field coil.

The purpose of the shunt field coil is to create a magnetic field in the motor. So when current flows through the field winding, it generates a magnetic field that interacts with the magnetic field produced by the armature. This interaction creates the electromagnetic force required to drive the motor's rotation.

The Brushes conduct electricity from the stationary parts of the motor to the rotating parts. This is essential for the motor to function because it enables the armature to receive power to generate motion. While The main role of the Commutator is switching the direction of the current inside the armature wiring.

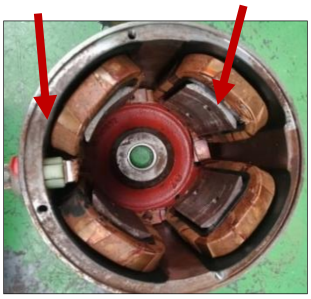


Figure 71: Motor inner Components

VI.2.1.2.Relay - coil switches

The primary function of Coil relays is to control the power circuit through a lower intensity circuit called the control circuit so When a current flows through the coil in the control circuit, it activates the relay switch, which then controls the flow of a larger current in the power circuit.

VI.2.1.3. Pedal Potentiometer

the primary goal of a potentiometer is to function as a variable resistor in order to control the current flow in the circuit which effects angular velocity of the motors In our case, I will be using a pedal potentiometer that can be controlled using the feet pressure

VI.2.1.4.Other Electric components

Several switch, passive and active types of electric components are to be employed in the circuit to ensure its proper functioning , a detailed list is provided in Annex

VI.2.2. Power Circuit Design

[Annex 9.1]

The control of the power circuit relies on relay coils (1) , as previously determined, along with various electrical components such as fuses (2) and Resistors (3) to ensure the circuit functions correctly. The circuit is divided into sub-circuits to efficiently manage different electrical functions and systems

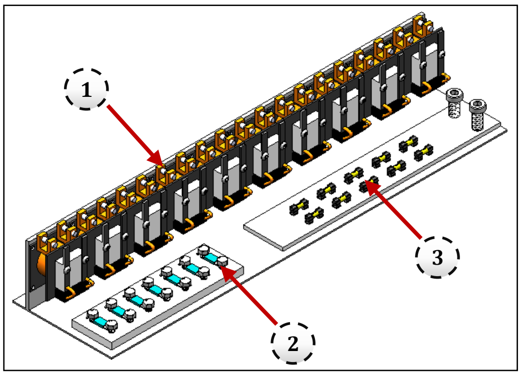


Figure 72: Power Circuit Installation

IV.2.12.1.ON-OFF supply Circuit

[Annex 9.2]

This circuit is used to turn ON and OFF the electricity in the circuit . When the key switch is pressed, the current flows from the battery which causes the Relay coil to energize , creating a magnetic field that changes the state of the contactors.

The Fuses are added to the positive pole for protection against overcurrent conditions. While a Diode and a resistor are installed parallel to the relay coil. The diode aims to protect

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against voltage spikes during the relay's deactivation while the resistor helps to dissipate the stored energy in the coil Upon deactivation.

A. Fuses

The circuit has a maximum current of 104A and operates at 24V, that means that the fuse should be capable of handling higher current than 104 A and higher voltage value than 24 V to bear the potential transient voltages

B. Diodes

The diode should be suitable for handling the back electromotive force generated by the relay coil. Consequently, a Schottky diode should be used for this purpose due to its quick response time.

The maximum voltage and current ratings that the diode will experience during the relay's deactivation are 24V and 104 A

C. Resistor

- resistance : R = $\frac{V_{Coil}}{I_{Coil}}$. consequently, R = $\frac{24}{104}$ = 0.23 Ω
- Power Dissipation: $P = I^2_{\ Coil} \times R$. Consequently, $P = 104^2 \times 0.23 = 2487.6$ W

The power rating of the resistor should be equal to or greater than the calculated power dissipation.

IV.2.2.2.Motors Direction Control circuit

To change the direction of rotation of the shunt motors, I have to change the polarity in either the armature or the field coil and keep the other in the same state ,Because the magnetic field produced by the field coil interacts with the one produced by the armature, and reversing its polarity changes the direction of this interaction.

In order to accomplish the direction switching I have designed a circuit based on (relay coils, contactors, and Rotary switch) and divided into two sub-circuits.

A. Field coils Alimentation:

The first sub-circuit is dedicated for supplying power to the motor's field coils. Pressing the push button energize the relay coil which closes the contactors, thereby providing the field coils with positive and negative terminals from the battery.

Couple of fuses with maximum current of 104A and maximum voltage of 24V ; are placed before the relay and contactors in order to provide protection potential transient voltages

B. Armature Alimentation:

The second sub-circuit is designated for supplying power to the motor's armature and it is responsible for switching the rotation Movement of the motors by changing the polarities of the armature.

C. Direction Selection :

[Annex 9.3- Annex 9.4]

Setting the rotary switch to position 1 or Position 3 closes the circuit for the selected relay coils and activate their corresponding 3-line contactors. This, in turn, energizes the relay coils and along with their associated 1-line contactors.

When the contactors are closed , the power circuit of the motors is closed causing the motors to turn in Direction 1 or direction 2 depending on the switch position.

IV.2.2.3. Speed Control circuit

[Annex 11.6]

In order to control the speed of the DC Shunt motors I have applied a variable resistance on the field coils of the motor using a potentiometer because the resistance in the field coil affects the motor's angular speed. In addition to an automatic return switch in order to cut the current off the circuit when applying the braking

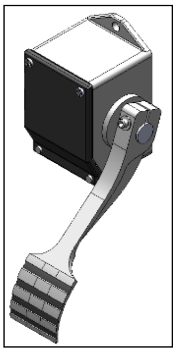

Figure 73: Potentiometer

A. Assumptions:

V: supply voltage	V_f: field resistance
Φ: Flux	ω: Angular Velocity
E_b: back electromotive force	k: Constant
R_f: Field Coil Resistance	R_p: resistance of potentiometer
R_a : Armature Resistance	I_a : Armature Current

B. Determining the relation between the potentiometer resistance and the angular velocity

We have : $$R_{Total} = R_f + R_p$$

And $$I_f = \frac{V_f}{R_{Total}}$$

And $$\Phi = K \times I_f = K \times \frac{V_f}{R_{Total}}$$

And $$E_b = K \times \omega \times \Phi$$

While $$V = E_b + (I_a \times R_a) = (K \times \omega \times \Phi) + (I_a \times R_a)$$

consequently: $$\omega = \frac{V-(I_a \times R_a)}{K \times \Phi}$$

Consequently, the Velocity ***ω*** of the motors decreases when the value of the potentiometer decreases and vice versa.

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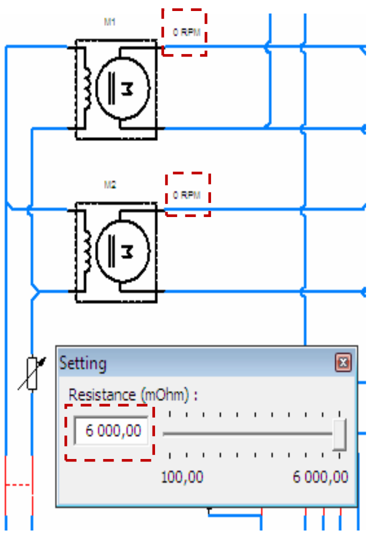
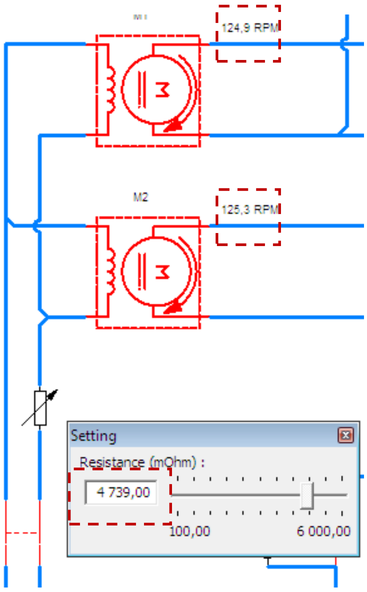
C. *Simulation Results*

R_p = Max R	Min R < R_p < Max R	R_p = Min R





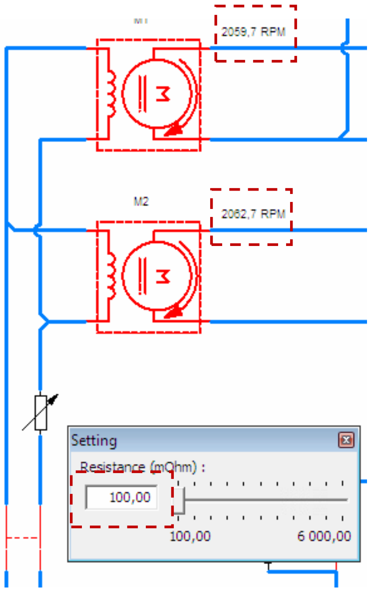


Table 27: Motors direction Simulation

IV.2.2.4. Braking

[Annex 9.5]

Inonder to enhance the mechanical braking operation , the motors should halt receiving current when the braking operation applied . Consequently, I have designed a customized contactor (1) to break the circuit when the barking pedal is pressed

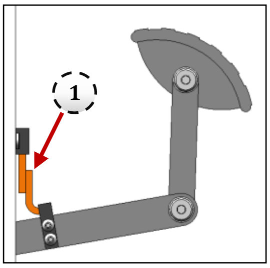


Figure 74: Braking Contactor

IV.2.2.5. Motor Pump circuit

[Annex 9.6]

Similarly, the traction motors alimentation circuit , the motor pump circuit is controlled using a pushbutton. Pressing the pushbutton energizes the relay coils, which in turn closes the contactors and supplies power to the motor pump. The diode and resistor are placed in parallel to the relay coil for the same reasons outlined on page 4.

IV.3. Auxiliary Circuit

The Auxiliary circuit is an independent circuit utilizing a separate power source from that of the control circuit in order to prevent potential damage to the electrical components within the auxiliary circuit.

It refers to the supplementary electrical systems and accessories responsible for safety and Compliance of the driving operation. Which includes: (Road lights, brake lights, emergency light, blinker signals) in addition to indicators on the dashboard to provide Status Information of the vehicle.

IV.3.1. Solar Energy Exploitation

I will be using solar panel in order to generate additional electricity in order to optimize and enhance the energy consumption of the auxiliary circuit.

IV.3.1.1.Geographical parameters.

The First step involves choosing the geographic location in a standalone project. I utilized Pvsyst software to select Metlaoui city, the site of DMM, for capturing and storing weather and sunlight data specific to that area.

The Average global horizontal irradiation in the southwest of Tunisia is between 5 kWh per m² per day and 5.8 kWh per m² per day . Consequently, I will be considering the Average daily irradiation as : I = 5.4 kWh per m² per day

IV.3.1.2.Orientation and position parameters.

Since the cart is mobile and can change orientation frequently, a fixed angle like the ones used for stationary solar installations aren't applicable. Therefore, I will apply flat mounting of the solar panel (1) on the top of the cart with an orientation angle of (0°) in order to exploit the maximum amount of the yearly irradiation yield.

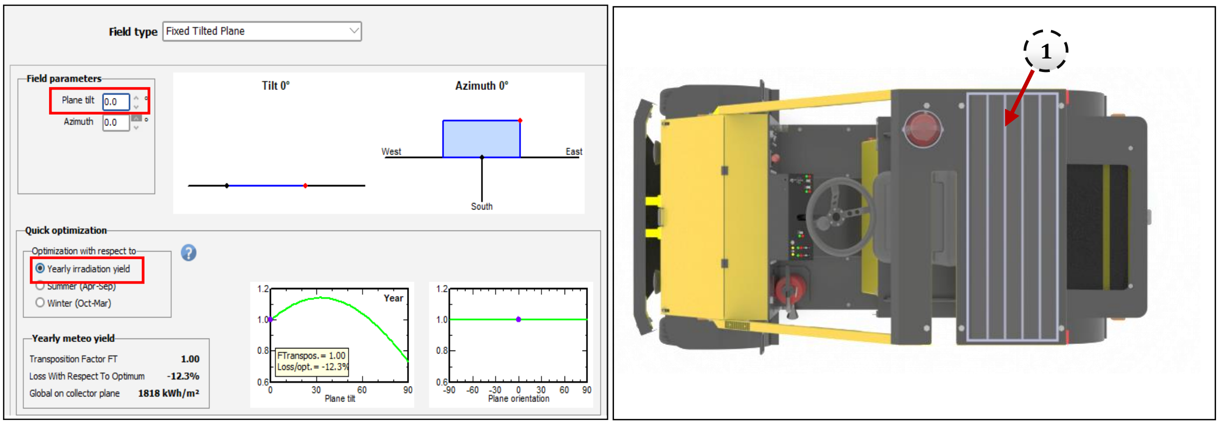


Figure 75: Solar Panel Orientation & Position on the cart

IV.3.1.3.Evaluating the Need

To determine the amount of Energy needed for the auxiliary circuit to function properly I inserted the quantity , power consumption and period of usage for each electric component (13 lamps and 7 indicators) .The indicators are diode led with a negligeable power $P_{Indicators} = 0.02$ W

Daily consumptions

Number	Appliance	Power		Daily use		Hourly distrib.	Daily energy
4	Road Lights	21	W/lamp	4.0	h/day	OK	336 Wh
4	Blinkers	21	W/app	1.0	h/day	OK	84 Wh
2	Brake Light	21	W/app	1.0	h/day	OK	42 Wh
0			kWh/day	24.0			0 Wh
1	Horn	21.0	W aver.	0.5	h/day	OK	11 Wh
1	Emergency Light	21	W/app	6.0	h/day	OK	126 Wh
0	Other uses	0	W/app	0.0	h/day		0 Wh
	Stand-by consumers	1	W tot	24 h/day			24 Wh
	Appliances info				Total daily energy		623 Wh/day
					Monthly energy		18.7 kWh/mth

Figure 76: Auxiliary Circuit components

Given that the daily usage of lights varies, I have estimated the usage time for each component. Consequently, The energy consumed per day is : E_c = 623 Wh/ Day

IV.3.1.4.Selecting the solar panel

- ***The maximal Peak power of the PV field***

It is the highest amount of electrical power the system can produce under optimal conditions.

$$P = \frac{E_c}{K \times I}$$

- K: the efficiency of the photovoltaic system [I will be value of K=0.20]

$$P = \frac{668}{0.65 \times 5.4} = 190 \text{ Wc}$$

- ***Number of panels***

Since the area dedicated to the installation of the panels is limited, I have chosen a panel with a maximal Peak power of : P_P =190 W

Consequently, the number of panels to install is $N = \frac{P}{P_p} = \frac{190}{190} = 1$ panel

Additionally, I have selected a Monocrystalline solar panel due to their superior efficiency compared to the Polycrystalline panels.

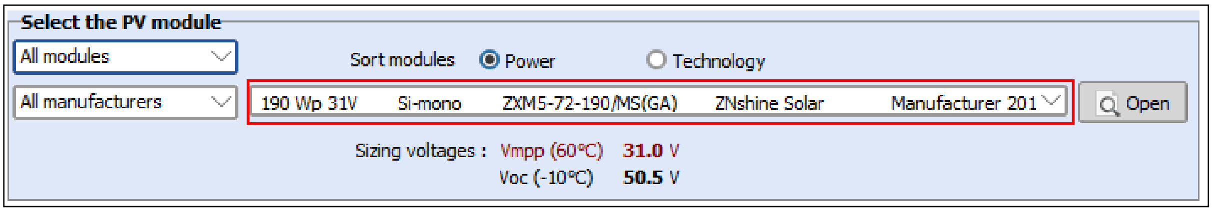


Figure 77: Solar panel Selection

IV.3.1.5.Selecting the battery type

The battery is designed to store the energy produced by the solar panels for use when the vehicle is not under sunlight. Hence, it is important to determine the required capacity of the battery, knowing that :

- V: voltage of the system: V=12V
- N: number of autonomy days, it is required to be at least N= 2
- D: maximum battery discharge , supposedly D= 0.8

$$C = \frac{N \times E_c}{D \times V}$$

$$C = \frac{623 \times 2}{0.8 \times 12} = 129.7 \text{ Ah}$$

Using Pvsyst , I chose the compatible battery with the closest capacity value

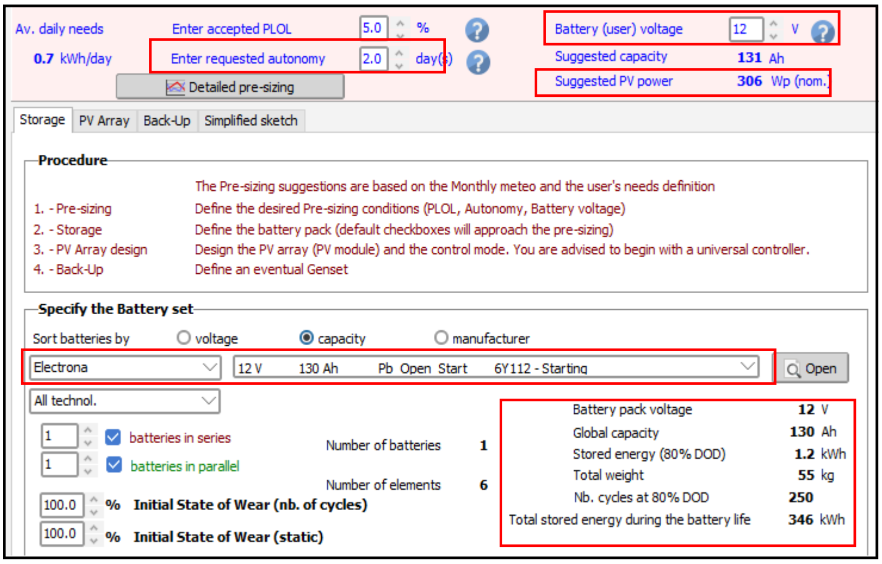


Figure 78: Auxiliary Circuit Battery Selection

IV.3.1.6.Selecting the regulator

The regulator is controller that regulates the voltage and current from the solar panels to the battery and prevent overcharging

the nominal power of the regulator P_r should be compatible with the maximal Peak power of the PV field " $P = 190\ W_C$"

$$0.8\ P < P_r < 1.2\ P$$

$$152\ W < P_r < 228\ W$$

Consequently, I have selected a MPPT regulator with $P_r = 160W$, 12 V and 10A

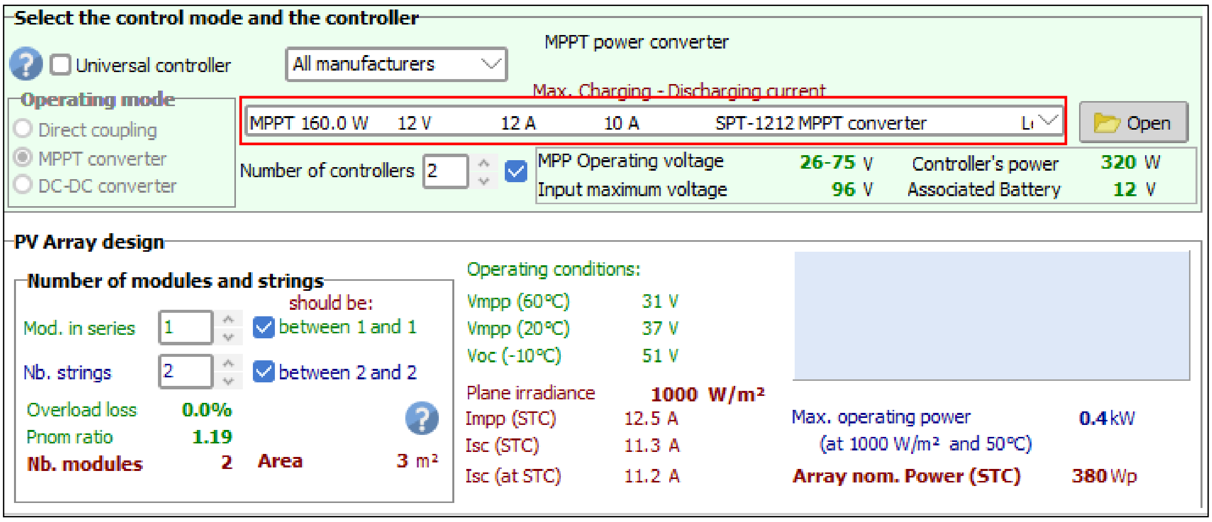


Figure 77: Regulator Selection

IV.4.8. Pvsyst Results

The obtained results are reported in the ***[Annex 10.1]***

IV.3.2.Auxiliary Circuit Design

[Annex 10.2]

The auxiliary circuit is responsible for controlling the lights and indicators of the cart . Using Proteus Software, I translated the selections of the battery, solar panel, and regulator types to design the auxiliary circuit. I also incorporated the necessary electrical components and employed a microcontroller to control the circuit.

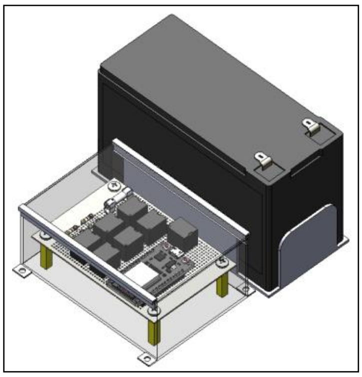


Figure 78: Auxiliar Circuit Installation

IV.4.8.1.Solar System Circuit

The solar System Circuit is the sub-circuit responsible for converting the solar energy to electric energy that will be stored in the battery . the process of conversion passes through the regulator to assure the compatibility between the voltage produced by the solar panel and the battery.

- ***backup AC source***

Additionally, a backup AC source is available to charge the battery when necessary. This AC source requires a bridge rectifier to convert the AC plug current to DC current, making it compatible with the battery. Switching between solar power and AC power is controlled by a switch.

- ***Indicator***

an indicator is energized when the solar panel is charging the battery. Since the indicator is a LED that functions with 2.2 V and 10mA in a circuit of 12V. it needs a resistance.

$$R_{\text{Indicator}} = \frac{12 - 2.2}{10.10^{-3}} = 950\ \Omega$$

- ***Protection Components.***

To protect the circuit from over current, I have added fuses. While to protect the solar panel from reverse polarity I installed a diode on series .

IV.4.8.2.Micro Controller

The control circuit uses an ESP32 microcontroller to convert the operator's actions on the control switches into commands that control the status of the lights and indicators.

In order to energize the microcontroller, an electric signal is given to the relay coil RL1 from the power circuit in order to close the contactors and supply the microcontroller

- ***Micro controller Protection***

The ESP32 utilizes 5.5 V and 150mA to operate, while the circuit provides a 12V.

$$R_{ESP32} = \frac{12 - 5}{150.10^{-3}} = 50\ \Omega$$

- ***Micro controller Coding***

[Annex 10.3]

I have programmed the ESP32 microcontroller Using Arduino IDE platform

IV.4.8.3.Road Lights

The road lights are controlled with a 2-position switch.

- ***ON statute***

when the switch is in the ON position it sends an input signal to the dedicated pin of the Microcontroller. This signal is converted into two Output signals. One is used to energize the coil of the RL3 Relay. When energized, the coil closes the relay contactor, leading to the alimentation of the 4 lamps of the road light. The other Output signal is dedicated to energizing the “Road Lights ON “ indicator LED

- ***OFF statute***

If the switch is in the OFF position, it energizes the “ Road Lights OFF” indicator LED .Due to the difference in voltage , the indicator needs a resistor.

$$R_{\text{Road Light OFF}} = \frac{5 - 2.2}{10.10^{-3}} = 2800\ \Omega$$

IV.4.8.4.Emergency Lights

Similar to The road lights, the Emergency light is controlled with a 2-position switch.

- ***ON statute***

Once the switch is in the ON position it sends an input signal to the dedicated pin of the Microcontroller. This signal is converted into two Output signals. One is used to energize the coil of the RL2 Relay. When energized, the coil closes the relay contactor, leading to the alimentation of the Rotating Roof emergency lamp. The other Output signal is dedicated to energizing the “Emergency Light ON “ indicator LED

- ***OFF statute***

If the switch is in the OFF position, it energizes the “ Emergency Light OFF” indicator LED . Due to the difference in voltage , the indicator needs a resistor.

$$R_{\text{Emergency Light OFF}} = \frac{5 - 2.2}{10.10^{-3}} = 2800\ \Omega$$

IV.4.8.4.Blinkers

The turning direction is selected using a rotary switch composed of 3 positions (Right , Middle and Left).

- ***Right position***

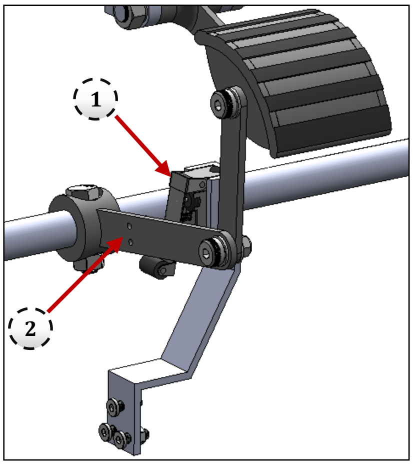
When in the Right position, it sends an Input signal to the proper pins in the ESP32. This signal is converted into two Output signals. One is used to energize the coil of the RL4 Relay. When energized, the coil closes the relay contactor, leading to the alimentation of the right blinker signals. The other Output signal is dedicated to energizing the “Right Blinker ON “ indicator LED

- ***Left position***

When in the Left position, it sends an Input signal to the proper pins in the ESP32. This signal is converted into two Output signals. One is used to energize the coil of the RL5 Relay. When energized, the coil closes the relay contactor, leading to the alimentation of the right blinker signals. The other Output signal is dedicated to energizing the “Left Blinker ON “ indicator LED

IV.4.8.5.Brake Lights

A limit switch (1) is incorporated into the braking system connected to the foot braking pedal (2) .



When braking occurs, the limit switch is activated by being pressed. As a result, it sends an input signal to the microcontroller, which converts it into an output signal. This output signal then powers the coil of the RL6 relay. Once energized, the coil closes the relay contactor, supplying power to the brake lamps.

IV.4.8.6. Horn

The horn is activated through a push button with automatic return. When pushed , the button sends an input signal to the microcontroller pin resulting in an output that energizes the relay coil RL7 resulting in the alimentation of the horn

IV.4.Conclusion

During this chapter, I have designed and simulated both the power electric circuit and the auxiliary circuit of the cart. The design of the power circuit incorporates DC shunt motors for propulsion, relay coils for switching, and a potentiometer to control the speed. While the auxiliary circuits exploits the solar energy to energize the lights and indicators.

Through detailed simulation, analysis and calculations, I have ensured that the electrical system meets the operational requirements and integrates seamlessly with the mechanical design of the cart.

General Conclusion

This graduation project, carried out at CPG, involved designing and developing an industrial heavy-duty electric traction cart. To achieve this goal, I first conducted a literature review that addresses the different types of transfer carts as well as their applications. In the second phase, I utilized functional analysis tools to identify the key characteristics for the developed and selection of the appropriate solutions based predefined criteria.

Once a solution was identified, I proceeded with its development through a 3D modeling using the CAD software, adapting it to the desired kinematics and sizing. I then conducted FEA simulations to verify structural integrity and applied the same process to the hydraulic system

Upon completing the mechanical and hydraulic phases , I designed and simulated the electric circuits responsible for powering and controlling the cart with exploitation to the solar energy to optimize the auxiliary circuit power consumption.

In conclusion , undertaking such a project, from the initial idea to the final design, required enormous effort. consequently, I recommend that CPG invest in the manufacturing of this cart given its great usefulness and utility . This investment will undoubtedly be rewarded with increased rentability and enhancement for the work efficiency.

Figure 79: Full 3D model of the cart at the DMM

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