THE UNIVERSITY OF DA NANG
UNIVERSITY OF SCIENCE AND TECHNOLOGY
FACULTY OF MECHANICAL ENGINEER

CAPSTONE PROJECT

MAJOR: MECHATRONIC ENGINEERING

TOPIC:

CALCULATION, DESIGN AND SIMULATION OF THE IMPORT/EXPORT SYSTEM IN AN AUTOMATIC FROZEN WAREHOUSE

Supervisor : **Vo Nhu Thanh, PhD**
Co - Supervisor : **Mr. Nguyen Nhu Thai**
Reviewer : **Le Hoai Nam, PhD**
Students : **Phan Minh Tu**
Van Thi Kim Thuy
Class : **20CDT1 - 20CDT2**
ID : **101200252 – 101200291**

DaNang, 6/2025

SUMMARY

Topic: Calculation, Design and Simulation of the Import/Export system in an automatic frozen warehouse

Students: Phan Minh Tu ID: 101200252 Class: 20CDT1

Van Thi Kim Thuy ID: 101200291 Class: 20CDT2

Supervisor: Vo Nhu Thanh, PhD

This project presents the calculation, design and simulation of the Import/Export system in an automatic frozen warehouse, with stable operating temperatures of $-30°C$ and below. The system is developed to address the challenges of energy efficiency, storage space optimization, and productivity enhancement in harsh cold storage environments.

The main content of the project includes the analysis of the detailed technical requirements of the system, namely: warehouse size ($7.7 \times 8 \times 4.85\ m$), minimum capacity requirement of $40\ tons$, and the ability to handle 1-ton packages ($1200 \times 1000 \times 150\ mm$ pallets containing four 200 liter drums). The AS/RS is designed to include a horizontal moving autonomous guided vehicle, integrated with a pallet lifting mechanism using a cable and pulley system, and a self-propelled trolley (Child Vehicle) operating on rails in the rack. The motion mechanisms use rack-and-pinion for horizontal movement and a motor with brakes for both horizontal movement and lifting, ensuring precision and safety.

The core goal of the project is to design a fully automated system, capable of completing the cycle of loading or unloading a package at the farthest position in under 4 minutes, while optimizing energy consumption and minimizing heat loss. The project also proposes the integration of a warehouse management system and safety sensors (ultrasonic, emergency brake) to ensure efficient and safe operation in extremely low temperature environments, aiming at an advanced and sustainable storage solution for the food industry.

THE UNIVERSITY OF DANANG — SOCIALIST REPUBLIC OF VIETNAM
UNIVERSITY OF SCIENCE AND TECHNOLOGY — Independent – Freedom - Happiness
FACULTY OF MECHANICAL ENGINEERING

GRADUATION PROJECT TASK

Student Information:

Full name 1: Phan Minh Tu — ID: 101200252 — Class: 20CDT1

Full name 2: Van Thi Kim Thuy — ID: 101200291 — Class: 20CDT2

Faculty: Mechanical engineer

Major: Mechatronic engineering

1. *Title of capstone project:*

Calculation, design and simulation of the Import/Export system in an automatic frozen warehouse.

2. *Intellectual Property: ☒ The project has an intellectual property agreement for all implementation*

3. *Initial figures and data:*

Warehouse: $7.7 \times 8 \times 4.85m$; Temperature: $< -30℃$; Commodity: Fruit juice, 4 drums/ton/pallet; Requirements: Capacity $\geq 40\ tons$; Productivity: 12 tons/day (average), 6 tons/hour (maximum); AS/RS: Full automated.

4. *Content of the explanations and calculations:*

Design of horizontal and lifting mechanisms: Calculation and Selection of motors (with brake), reducers, transmission & guidance systems,…; Design of Child Vehicle: Calculation and Selection of motors (with brake), reducers, transmission systems,…; Kinematic and dynamic calculations; Electrical system design; Control System Design; Energy efficiency analysis; Operational simulation;…

5. *Drawings and presentations:*

Overall drawings (A0); Detail drawings (A0); Assembly drawings(A0); Kinematic diagrams (A0); Electrical system diagrams (A0); Algorithm diagrams (A0);

6. *Supervisor's name: Vo Nhu Thanh, PhD*
7. *Date of assignment: ……../……./2025*
8. *Date of project completion: ……./……./2025*

Danang,………………………2025

Head of Division — **Supervisor**

PREFACE

In the context of global economic integration and the 4.0 industrial revolution, the warehousing and logistics industry is undergoing dramatic changes. Especially for specialized cold storages, the requirements for operational efficiency, space optimization, and energy control are becoming increasingly urgent. Low temperature environments pose unique challenges in terms of design, materials, and automation technology, requiring advanced and reliable technical solutions.

The graduation project "Calculation, design and simulation of the Import/Export system in an automatic frozen warehouse" was conducted within the framework of the Capstone Project of University of Science and Technology – the University of Danang, majoring in Mechatronics Engineering. This project is the result of the research, calculation, design and simulation of the Import/Export system in an automatic frozen warehouse at extremely low temperatures (below $-30°C$). The project is not only an opportunity to apply the acquired professional knowledge but also a practical challenge, requiring the ability to integrate multiple fields from mechanics, electronics, control to automation.

Through this project, we hope to provide a comprehensive view of the application of automation technology in specific industrial environments, while contributing to the development of smart and sustainable logistics solutions.

We would like to express our deep gratitude to the Board of Directors, Faculty of Mechanical Engineer, and all the teachers of University of Science and Technology – the University of Danang for their dedicated teaching, knowledge provision, and favorable conditions during the process of studying. In particular, we would like to sincerely thank Vo Nhu Thanh, PhD for his dedicated guidance, research orientation, and valuable experiences to help us complete this project.

COMMITMENT

I hereby declare that the graduation project with the topic "Calculation, design and simulation of the Import/Export system in an automatic frozen warehouse" is my own research work, conducted under the guidance of Vo Nhu Thanh, PhD. All information, data, research results and analysis in this project are honest and objective.

I declare that the entire content of the project has never been published in any form for the purpose of receiving a degree at any other educational institution. The references used in the project are fully cited and listed, clearly in accordance with the regulations on academic integrity. I am fully responsible for the accuracy and originality of the information, data and analysis in this project.

Student 1	Student 2
Phan Minh Tu	Van Thi Kim Thuy
Date: .../06/2025	*Date: .../06/2025*

CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOLS AND ABBREVIATIONS

Abbreviations:

- AS/RS: Automated Storage and Retrieval System
- MV: Mother Vehicle
- CV: Child Vehicle
- WMS: Warehouse Management System
- PID: Proportional Integral Derivative
- PLC: Programmable Logic Controller
- HMI: Human Machine Interface
- AC: Alternating Current
- DC: Direct Current
- WCS: Warehouse Control System
- VFD: Variable Frequency Drive
- VI: Viscosity Index

CHAPTER 1: INTRODUCTION

1.1. Company introduction

1.1.1. Overview

Full Name: Khai Minh Khang Limited Liability Company.

Head Office Address: 18 Ngo Tu Ha, Hoaan Ward, Camle District, Danang City.

Business Office: 3rd Floor, 305 Yen The, Hoaan Ward, Camle District, Danang City.

Khai Minh Khang Co. Ltd is a specialized HVAC contractor providing consulting, surveying, designing, supplying materials, equipment, and installing refrigeration systems, cold storage, air conditioning systems, electrical systems, and mechanical systems for both industrial and civil sectors. The company also offers maintenance, repair services for HVAC equipment, and renovations to enhance energy efficiency and automation.

1.1.2. Functions and scope of activities

The main functions focus on the following areas:

- Refrigeration Engineering: Specializing in the design and construction of refrigeration projects, including chiller systems, Freon and NH3 refrigeration systems, chilled water/glycol systems, cold storage systems, and cooling equipment for production facilities, workshops, and factories.

Figure 1. 1 Installation of Glycol Chiller system at Nafoods Tay Nguyen JSC

- Construction Electromechanics: Specializing in the design and construction of civil and industrial electromechanical projects, including electrical systems, low-voltage systems, surveillance cameras, HVAC and ventilation systems, and plumbing systems for apartments, houses, hotels, and factories.

- Maintenance Services: Providing maintenance, repair, renovation, upgrading, and energy-saving services.

Figure 1. 2 Installing frozen booths and refrigeration system at Acecook project

Figure 1. 3 Renovation & upgrading of high-capacity aloe vera & jam cooling system at Vinamilk factory

Mission: To continuously innovate in order to provide customers with the most suitable products and services equipped with the latest technology.

Vision: To become a leading company in the field of refrigeration and automated cold storage in the region.

Core Values: Credibility, responsibility, progressiveness, creativity, resilience, and community contribution.

1.1.3. Problems, requests from customer

- Customer: Nafoods Tay Nguyen JSC
- Location: Gia Lai Province; Area: 10 Ha
- Production capacity: 10,000 tons / year
- About customer: Founded in 1995, Nafoods Group JSC (HOSE: NAF) is among the largest fruit and vegetable growers, processors and exporters in Vietnam, specializing in Fruit Juice/ NFC, Puree, Concentrate, IQF, Fresh Fruits, Dried Fruits and Nuts. Our products are distributed all over the world, especially in Europe, the US, Oceania, Middle East, Japan, and Korea.
- Problems faced by customers: Nafoods factory has difficulty in improving the efficiency of freezing goods after production
    - High energy loss due to opening/closing the warehouse door many times → does not ensure export quality

- Traditional warehouse imports/exports based on forklifts operated by humans → high labor costs, affecting human health
- No space optimization measures → Low freezing output

1.2. Existing Problems

Currently, the company operates in the field of processing and preserving frozen food for export, with a blast freezer storage system operating at temperatures from -30°C to -40°C to ensure product quality during storage.

Figure 1.4 Manual Cold Storage Warehouse Layout [2]

The existing cold storage system uses manual handling methods for loading and unloading goods like Figure 1.1, requiring workers to move pallets in and out of the cold storage using manual pallet jacks or electric forklifts. This approach presents several operational and business challenges for the company, including:

- High labor costs and the need for a large number of workers to operate in low-temperature environments.
- Reduced operational efficiency due to slow and inconsistent loading/unloading times.
- Heat loss every time the cold storage door opens, increasing power consumption and reducing refrigeration equipment lifespan.
- Occupational safety risks due to harsh working conditions.
- Difficulty in accurately managing pallet positions and stock quantities, affecting productivity and warehouse management efficiency.

To address these issues, the company aims to implement an automated import/export system for the cold storage (AS/RS) to optimize goods handling operations, reduce direct labor involvement, and improve overall working efficiency.

1.3. Project objective

Based on the existing situation, this graduation project is conducted with the goal of: Calculating, designing, and simulating an automated import/export system for a blast freezer warehouse operating at -30°C, which includes:

- Calculating and designing a pallet lift mechanism for transferring pallets from the import/export position to storage levels.
- Designing and simulating the Mother Vehicle and Child Vehicle for transporting pallets within the cold storage.
- Developing a simulation process for the automatic import/export operation of pallets.
- Ensuring the system operates accurately and safely in low-temperature environments.
- Reducing the number of door openings and minimizing cold air loss.
- Improving loading/unloading efficiency and optimizing operating costs.

1.4. Technical requirements

1.4.1. General requirements:

- System type: Fully automated AS/RS for blast freezer cold storage.
- Operating environment: -30°C to -40°C.
- Automation level: Fully automated.
- Control method: PLC combined with WMS/WCS.

1.4.2. Mechanical specifications:

- Storage room dimensions: 7.7 m (L) × 8 m (W) × 4.85 m (H).
- Pallet size: 1200 mm × 1000 mm × 150 mm.
- Load capacity per pallet: 1000 kg.
- Maximum number of pallets: 40 units.
- Rack structure: Galvanized steel frame with cold-resistant powder coating.
- Horizontal movement mechanism: Mother Vehicle running on alloy steel rails with dual side guides.
- Lifting mechanism: Ball screw or belt/chain drive.
- Pallet transfer mechanism: Shuttle-type Child Vehicle.

1.4.3. Dynamic specifications:

- Maximum lifting load: 1000 kg.
- Mother Vehicle travel: ≥ 0.4 m/s.
- Lifting speed: ≥ 0.2 m/s.

- Child Vehicle speed: ≥ 0.3 m/s.
- Maximum cycle time for furthest pallet: ≤ 4 minutes.
- System capacity: ≥ 12 tons/day (average).

1.4.4. Electrical and Control Specifications:

- Power supply: 380V AC three-phase / 48VDC for main motors; 24V DC for control signals and devices.
- Motors: Servo or stepper motors (cold-resistant) from 750W to 2kW.
- Controller: Siemens S7-1200 PLC.
- Sensors: Ultrasonic distance sensors, limit switches, position sensors,
- Braking system: Electromagnetic brakes that hold position in case of power failure.
- Communication: Modbus TCP/IP, Ethernet.
- Control software: PLC TIA Portal.
- User interface: 7-inch HMI touchscreen.
- Connectivity: Modbus TCP/IP or Ethernet to WMS system.

1.4.5. Safety and Maintenance:

- Protection mechanisms: Ultrasonic sensors, limit switches, E-Stop buttons.
- Mechanical locks and electromagnetic brakes in case of power failure.
- Maintenance requirements:
  Guide mechanism inspection every 2 weeks.
  Lubrication of lifting mechanism every month.
  Regular inspection of electrical systems and sensors.
- Design lifespan: ≥ 20,000 operating cycles before major maintenance is needed.

1.5. Chapter conclusion

In Chapter 1, this project provided an overview of the background, urgency, and practical significance of applying Automated Storage and Retrieval Systems (AS/RS) in cold storage warehouses, particularly in blast freezer environments operating at -30°C and below. Existing manual handling operations in such harsh conditions lead to high labor costs, potential safety risks, substantial heat loss during import/export, increased operating costs, and compromised product quality.

From this situation, the project identified the objective of calculating, designing, and simulating an automatic import/export system in a cold storage warehouse, featuring a configuration including a Mother Vehicle, lifting mechanism, and Child Vehicle (Shuttle Pallet). This system aims to replace manual operations, optimize storage space, improve energy efficiency, and enhance operational productivity.

CHAPTER 2: MECHANICAL SYSTEM DESIGN

2.1. Overview and kinematic diagram

To commence the design process, it is essential to present the overall system layout to provide a comprehensive view of the structure and arrangement of the main components. This diagram illustrates the spatial relationships and how different parts interconnect to form a complete operational system, optimizing the automated inbound and outbound warehouse processes.

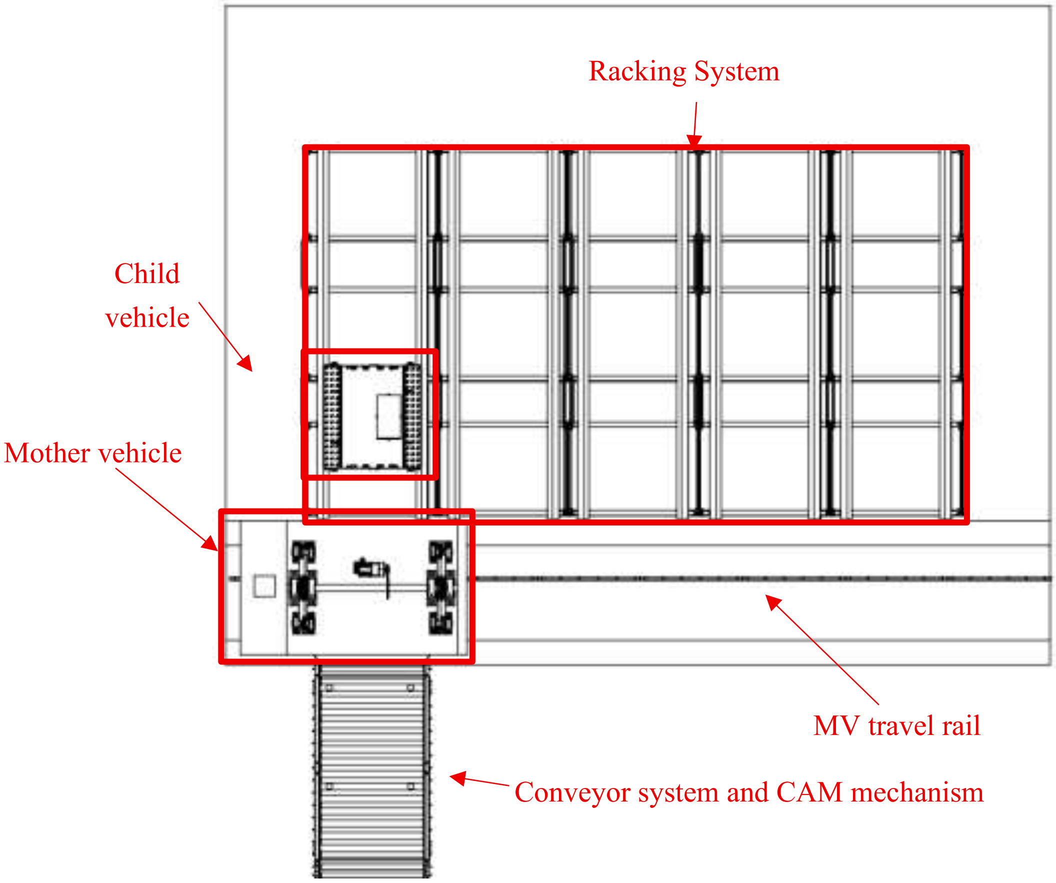


Figure 2. 1 Overview

The intelligent automated warehouse system is designed to include core components that cooperate closely:

- Racking System: Used for scientifically storing goods.
- Mother Vehicle: Tasked with moving the Child Vehicle and goods to specific rack aisles as required.

- Child Vehicle: Moves deep into the rack compartments to place or retrieve goods. Concurrently, the Child Vehicle also picks up or delivers goods at the CAM mechanism.
- Conveyor System: Utilized for continuous and automated movement of inbound and outbound goods within the warehouse.
- CAM Mechanism: Responsible for lifting and lowering goods when the Child Vehicle arrives to pick up or deliver items, ensuring precise and safe cargo transfer.

The interrelationship among these components is as follows: The Mother Vehicle traverses along the main aisle, carrying the Child Vehicle to the correct rack position. The Child Vehicle then performs operations to move deep into the rack compartments for storage or retrieval. The transfer of goods between the system and the external environment is facilitated by the conveyor system, supported by the CAM mechanism at the Child Vehicle's transfer point.

2.2. Motion mechanism analysis and selection

2.2.1. Mother vehicle drive mechanisms

2.2.1.1. Horizontal traverse mechanism

The Mother Vehicle's horizontal traverse mechanism is responsible for transporting the entire lifting system, Child Vehicle, and heavy load (total laden mass of 3.6 tons) along the length of the warehouse. The primary motion option considered for this movement is “Rack and pinion motion system”.

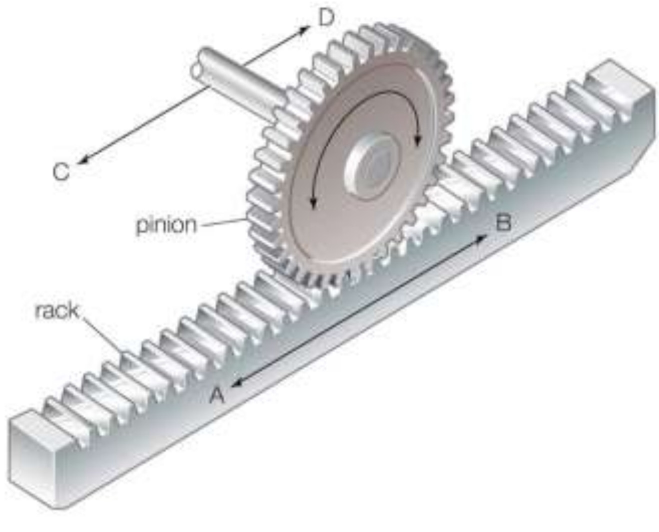


Figure 2. 2 Rack and pinion motion system [3]

- Principle: A motor, via a gearbox, motions a pinion gear that meshes with a fixed rack running along the travel path. As the pinion rotates, it moves the entire mechanism horizontally.
- Advantages:

- High Precision: Minimal slippage ensures accurate positioning.
- Stable Transmission: Capable of handling large loads and high torques.
- High Durability: With proper lubrication and maintenance, the system can have a very long lifespan.
- Easy Maintenance: Relatively simple to replace or service components compared to some other systems.
- Suitable for Heavy Loads: Can transmit large forces efficiently.

- Disadvantages:

- Higher Initial Cost: Investment in high-quality racks and pinions can be substantial.
- Strict Straightness Requirements: The rack must be installed very straight and precisely to ensure smooth operation and prevent jamming.
- Noise: Can generate noise if not well-designed and lubricated.
- Lubrication Requirement: Requires regular lubrication, especially in cold storage where lubricant viscosity can be affected.

2.2.1.2. Lifting mechanism

The Mother Vehicle's lifting mechanism is responsible for lifting and lowering the Child Vehicle system and its load to align heights with different shelf levels. A chain or belt drive system is commonly considered for this function.

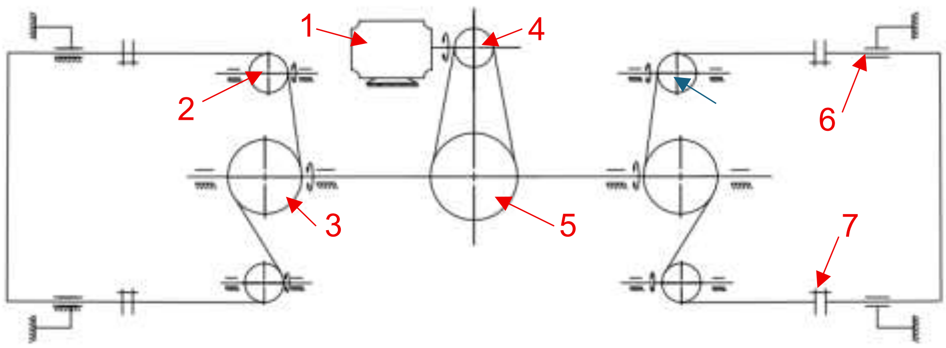


Figure 2. 3 Kinematic diagram of lifting mechanism

1. Motor 2,4. Small pulley 3,5. Large pulley

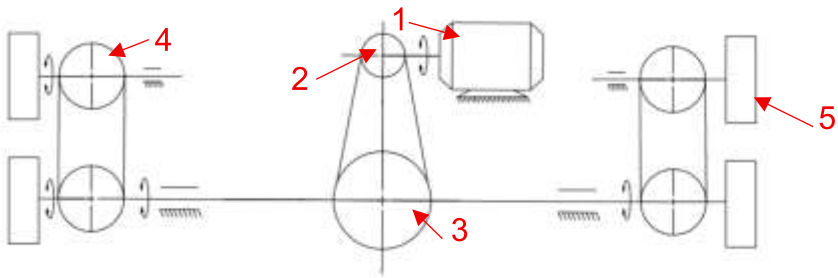
2. 6. Sliding wheel along the frame 7. Rigid coupling

- Principle: A motor drives a sprocket or pulley, which pulls a chain or belt attached to the lifting mechanism, thereby raising or lowering the load vertically.
- Advantages:
- Lower Initial Cost: Often less expensive than other lifting mechanisms (e.g., lead screw).
- Flexible Installation: Can be easily integrated into the design space.
- Smooth Operation (Belt): Belt systems typically operate more smoothly than chains.
- Large Travel Distance: Suitable for vertical lifting over significant heights.
- Disadvantages:
- Lower Precision (Belts may stretch): Belts can stretch over time, affecting positional accuracy, and chains may have backlash.
- Potentially Shorter Lifespan: Chains or belts can wear out or break, especially under heavy loads or in harsh environments, requiring maintenance.
- Regular Maintenance: Requires frequent tension checks and lubrication (for chains).
- Load Capacity: Must be carefully calculated to ensure it can withstand the lifting load.

2.2.2. Child vehicle traverse mechanism

2.2.2.1. .Longitudinal (depth) traverse mechanism

The Child Vehicle's longitudinal (depth) traverse mechanism is responsible for moving goods precisely in and out of the storage shelves. The chain drive system is a common choice for this function.



1. Motor 2. Small gear 3. Large gear 4. Medium gear 5. Medium gear

Figure 2. 4 Kinematic diagram of the longitudinal (depth) traverse mechanism

- Principle: A motor drives a sprocket, which pulls a chain attached to the Child Vehicle, allowing the Child Vehicle to move linearly in and out of the shelf.

- Advantages:
  - High Load Capacity: Chains can transmit large loads and withstand high tensile forces.
  - Durability: With regular maintenance, chain systems have a relatively long lifespan.
  - Rigidity: Less prone to stretching than belts under load, maintaining good transmission distance.
  - Suitable for Harsh Environments: Less affected by temperature and dust than some other systems.
- Disadvantages:
  - Noise and Vibration: Can generate noise and vibrations during operation.
  - Lubrication and Maintenance Requirements: Requires regular lubrication and tension checks to ensure performance and longevity.
  - Precision: May not achieve absolute positioning accuracy like ball screws, but is sufficient for many warehouse applications.

2.2.2.2. Lifting mechanism

The Child Vehicle's lifting mechanism is responsible for raising or lowering goods within its operating range (e.g., to pick up or place goods on shelves). The crank-slider mechanism is the chosen option for this function.

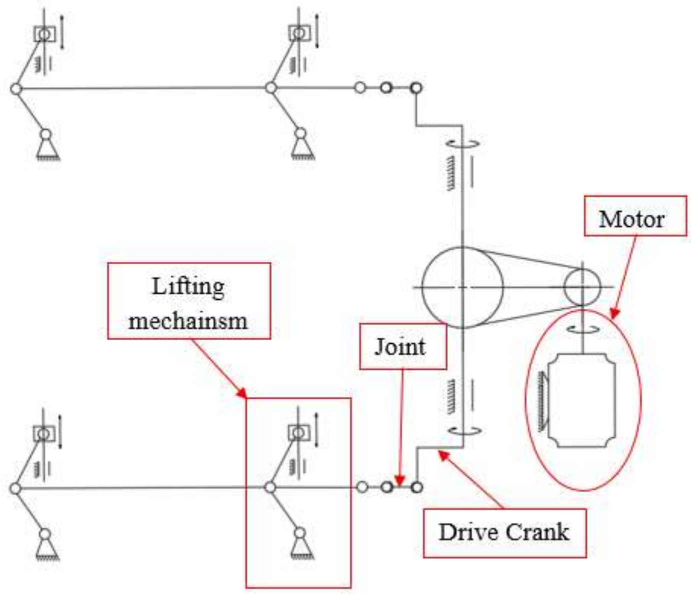


Figure 2. 5 Kinematic diagram of the lifting mechanism

- Principle: A rotating crank is connected to a connecting rod (slider crank). The other end of the connecting rod is linked to a slider (or similar component) that moves linearly. By designing the parameters of the crank and connecting rod, the desired lifting motion can be achieved.

- Advantages:
  - Simple and Reliable Transmission: The mechanical structure is relatively simple, easy to manufacture and maintain.
  - Efficient Conversion of Rotary to Linear Motion: Effective for creating limited lifting travel.
  - Customizable Motion Profile: By adjusting the crank and connecting rod parameters, desired kinematic characteristics for the lifting process can be achieved.
  - Relatively Low Cost: Compared to more complex mechanisms, manufacturing costs are often lower.
- Disadvantages:
  - Limited Travel Distance: The linear travel is limited by the dimensions of the crank and connecting rod.
  - Varying Velocity and Acceleration: The velocity and acceleration of the slider are not constant throughout a cycle, requiring careful calculation to avoid jerky movements.
  - Can Generate Inertia Forces and Vibrations: Especially at high speeds, dynamic balancing may be required to minimize these effects.

2.3. Design calculations for racking system

2.3.1. Load calculation on each column

Formula:

$$P_{col} = \frac{N \times m_{pallet} \times g}{n_{col}}$$

Where:

P_{col}: load on each column (N)

N: number of pallets per frame

m_{pallet}: weight of one pallet (kg)

g: gravitational acceleration (9.81 m/s²)

n_{col}: number of columns per frame (4)

Substituting values:

$$P_{col} = \frac{(3 \times 5) \times 1000 \times 9.81}{4} = \frac{147150}{4} = 36787.5\ (N)$$

→ Each column bears a load of 36.79 kN

Maximum bending moment on each horizontal beam

Assuming the distance between 2 columns horizontally: $L = 1.2\,m$

Load concentrated at pallet position

Formula (simply supported beam with central point load):

$$M_{max} = \frac{P \times L}{4}$$

Where:

M_{max}: maximum bending moment (Nm)

P: load per pallet (N)

L: beam length (m)

Substituting values:

$$M_{max} = \frac{1000 \times 9.81 \times 1.2}{4} = 2943\ (Nm)$$

2.3.2. Select beam and column section

Check bending stress on beam

Formula:

$$\sigma = \frac{M_{max} \times c}{I}$$

Where:

σ: bending stress (Pa)

M_{max}: maximum bending moment (Nm)

c: distance from neutral axis to extreme fiber (m)

I: moment of inertia (m^4)

Assume box beam 100 × 50 × 3 mm

Calculate moment of inertia:

$$I = \frac{(b \times h^3)}{12} - \frac{(b_t \times h_t^3)}{12}$$

Where:

I : Moment of inertia of the cross-section (m^4)

b : Outer width of the rectangular hollow section (m)

h : Outer height of the rectangular hollow section (m)

b_t : Inner width (m)

h_t : Inner height (m)

With:

- $b = 0.1\,m$, $h = 0.05\,m$
- $b_t = 0.1 - 2 \times 0.003 = 0.094\,m$

- $h_t = 0.05 - 2 \times 0.003 = 0.044\, m$

Substituting values:

$$I = \frac{(0.1 \times 0.05^3)}{12} - \frac{(0.094 \times 0.044^3)}{12}$$

$$I \approx 1.041 \times 10^{-7}\, m^4$$

c = h/2 = 0.025 m

Calculate stress:

$$\sigma = \frac{2943 \times 0.025}{1.041 \times 10^{-7}} \approx 707\, MPa$$

→ Exceeds CT3 limit (150 MPa)

Select new beam size: box beam 150 × 75 × 4 mm

New moment of inertia:

$$I_{new} \approx 5.98 \times 10^{-7}\, m^4$$

New stress:

$$\sigma = \frac{2943 \times 0.0375}{5.98 \times 10^{-7}} \approx 184.6\, MPa$$

→ Acceptable for CT3 steel with σ allowable = 240 MPa (as the rack is static and impact-free in cold storage)

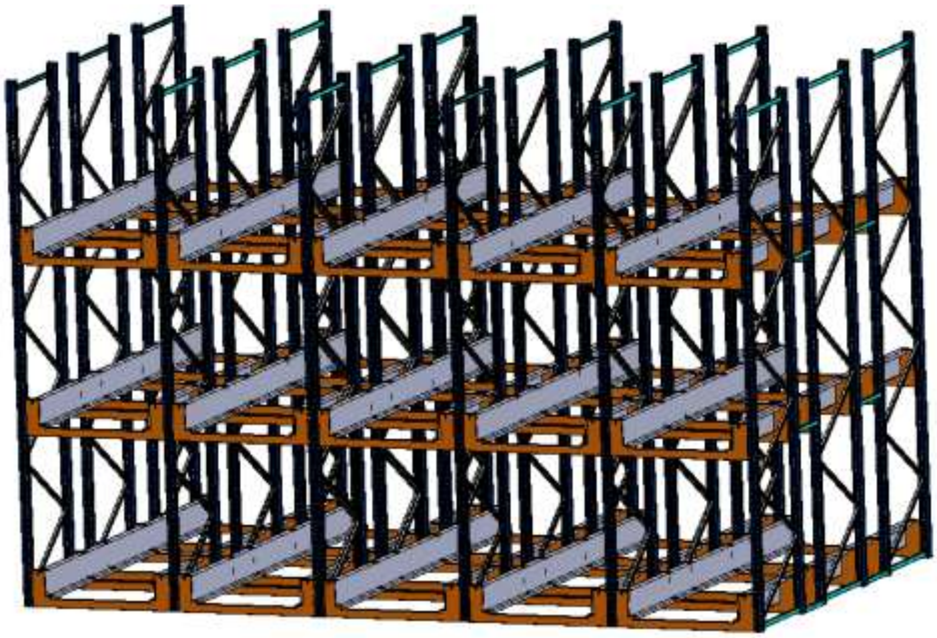


Figure 2. 6 Racking system

2.3.3. Load on each guide rail and rail support beam

Specifications:

Child vehicle weight: 200 kg

Pallet weight: 1000 kg

Dynamic load factor (due to fast movement in cold storage): $K = 1.25$

Formula:

$$P_{rail} = \frac{(m_{vehicle} + m_{pallet}) \times g \times K}{2}$$

Substituting values:

$$P_{rail} = \frac{(200 + 1000) \times 9.81 \times 1.25}{2} = \frac{14715}{2} = 7357.5\,N$$

→ Each rail bears a combined static + dynamic load of 7.36 kN

Guide rail and support selection

Guide rail:

8 mm thick, 100 mm wide steel flat bar

Welded or bolted to the racking frame

Rail support beam:

50 × 50 × 5 mm square steel tubing

Spaced at 0.5 m intervals

Bending moment and stress can be checked separately if required

2.3.4. Heating cable

Installed on both child vehicle guide rails

30W/m cold-resistant silicone heater

Auto on/off sensor system to maintain rail surface temperature > -5°C

2.4. Design calculations for mother vehicle

2.4.1. Preliminary System Motion Design

2.4.1.1. Analysis of Operation Cycle and Determination of Required Velocities

Typical Operation Cycle (Total Time $T_{total} < 4\ minute = 240\ s$):

- Horizontal Travel: $L_{horizontal} = 8\ m$ (movement to row 5) and $L_{horizontal} = 8\ m$ (return to Home). The total horizontal travel distance is $2 \times 8 = 16\ m$.
- Vertical Lifting Travel: $H_{lift} = 4.05\ m$ (lift to level 3) and $H_{lift} = 4.05\ m$ (lower to level 1). The total vertical travel distance is $2 \times 4.05 = 8.1\ m$.
- Waiting/Dwell Time: $t_{dwell} = 30\ s$.

Since specific velocities are not constrained, but the horizontal travel speed is required to be faster than the vertical travel speed, we will allocate time as follows:

To ensure the cycle time remains below $240\ s$, the total time equation is:

$$T_{total} = T_{horizontal_{to}} + T_{lift} + t_{dwell} + T_{lower} + T_{horizontal_{from}} \leq 240\ s$$

Where:

- $T_{horizontal_{to}}$: Time for horizontal movement to row 5.
- T_{lift}: Time for lifting to level 3.
- T_{lower}: Time for lowering to level 1.
- $T_{horizontal_{from}}$: Time for horizontal movement back to Home.

For simplicity and time efficiency, we assume the horizontal travel times (to and from) are equal ($T_{horizontal_{to}} = T_{horizontal_{from}} = T_{horizontal}$) and the lifting and lowering times are equal ($T_{lift} = T_{lower} = T_{vertical}$).

Thus: $2 \times T_{horizontal} + 2 \times T_{vertical} + t_{dwell} \leq 240\ s$

Let $T_{motion} = 2 \times T_{horizontal} + 2 \times T_{vertical}$ be the total motion time.

$$T_{motion} \leq 240 - t_{dwell} = 240 - 30 = 210\ s$$

To ensure the horizontal velocity ($v_{horizontal}$) is faster than the vertical velocity ($v_{vertical}$), we can allocate time proportionally less for horizontal travel per unit distance, or more simply, allocate less total time for horizontal motion compared to vertical motion.

We have a total horizontal distance $S_{horizontal} = 16\ m$ and a total vertical distance $S_{vertical} = 8.1\ m$. The velocities are defined as:

$v_{horizontal} = \frac{S_{horizontal}}{T_{horizontal\_total}}$ and $v_{vertical} = \frac{S_{vertical}}{T_{vertical\_total}}$

With $T_{horizontal\_total} = 2 \times T_{horizontal}$ and $T_{vertical\_total} = 2 \times T_{vertical}$

Let's assume initial target velocities to back-calculate the required motion times. Assumed preliminary velocities:

- Horizontal travel velocity: $v_{horizontal} = 0.4\ m/s$
- Vertical travel velocity: $v_{vertical} = 0.2\ m/s$ (satisfying $v_{horizontal} > v_{vertical}$)

Calculate motion times based on these projected velocities:

$$T_{horizontal_total} = \frac{S_{horizontal}}{v_{horizontal}} = \frac{16\ m}{0.4\ m/s} = 40\ s$$

$$T_{vertical_total} = \frac{S_{vertical}}{\mathrm{v_{vertical}}} = \frac{8.1\ m}{0.2\ m/s} = 40.5\ s$$

The total projected motion time is:

$$T_{motion\_projected} = T_{horizontal\_total} + T_{vertical\_total} = 40\ s + 40.5\ s = 80.5\ s$$

The total projected cycle time is:

$$T_{total\_projected} = T_{motion\_projected} + t_{dwell} = 80.5\ s + 30\ s = 110.5\ s$$

$110.5\ s < 240\ s(4\ minutes)$. These velocities are entirely feasible and ensure the productivity target. We will use these velocities as initial targets for power calculations.

Table 2. 1 Preliminary velocity parameters

Horizontal travel velocity ($v_{horizontal}$)	$0.4\ m/s$
Vertical travel velocity ($v_{vertical}$)	$0.2\ m/s$

2.4.1.2. Power Calculation and Preliminary Motor Selection

- *Power Calculation for Horizontal Travel Mechanism*

Table 2. 2 Parameter for travel mechaism

Total mass of horizontal travel mechanism (with payload):	$m_{horizontal} = 3.6\ tons = 3600\ kg$
Horizontal travel velocity:	$v_{horizontal} = 0.4\ m/s$

The power required to move a mass m at a velocity v on a horizontal plane, neglecting initial friction, is: $P_{horizontal\_mechanical} = F \times v_{horizontal}$.

Where F is the driving force. The driving force includes rolling resistance and inertial force (during acceleration). For preliminary power calculation, and considering a flexible acceleration/deceleration time, we will calculate power at steady-state with rolling resistance.

Rolling Resistance Force: $F_{friction} = k \cdot m_{horizontal} \cdot g$

Where:

k: Rolling resistance coefficient (for steel wheels on steel rails, k is typically very small, around 0.001 to 0.005). We select $k = 0.003$ to account for potentially increased friction in a cold storage environment.

g: Gravitational acceleration, $g = 9.81\ m/s^2$.

$$F_{friction} = 0.003 \times 3600\ kg \times 9.81\ m/s^2 = 105.948\ N$$

Mechanical Power required for horizontal travel (excluding transmission and gearbox efficiency):

$$P_{horizontal\_mechanical} = F_{friction} \times v_{horizontal} = 105.948\ N \times 0.4\ m/s$$
$$= 42.3792\ W \approx 0.0424\ kW$$

This is the steady-state power. During start-up, a larger inertial force is required. However, for motor selection, we need to calculate the power at the motor shaft after accounting for the efficiency of the entire motion system.

Required power at the horizontal travel motor shaft:

$$P_{motor\_horizontal} = \frac{P_{horizontal\_mechanical}}{\eta_{gearbox} \times \eta_{rack\_pinion} \times \eta_{bearings}}$$

Where:

$\eta_{gearbox}$: Gearbox efficiency. Select $\eta_{gearbox} = 0.92$.

η_{rack_pinion}: Rack and pinion motion efficiency. Select $\eta_{rack\_pinion} = 0.96$.

$\eta_{bearings}$: Bearing efficiency (typically 0.99 to 0.995 per pair). Assuming 4 wheels and associated motion shafts, select $\eta_{bearings} = 0.98$.

Total horizontal transmission efficiency:

$$\eta_{total\_horizontal} = \eta_{gearbox} \times \eta_{rack\_pinion} \times \eta_{bearings} = 0.92 \times 0.96 \times 0.98 \approx 0.865$$

$$P_{motor\_horizontal} = \frac{0.0424\ kW}{0.865} \approx 0.049\ kW$$

This power value is very small, primarily due to only considering rolling resistance. In practice, starting power and contingency power for uncertain factors (such as increased friction at low temperatures, minor load fluctuations) are crucial. Motors for large-load travel systems like this are typically selected with significantly higher power to ensure sufficient starting torque and acceleration capability.

Let's consider the power required to overcome inertia during acceleration. Assuming an acceleration time from 0 to $v_{horizontal}$ of $t_{accel} = 2\ s$ (this will be detailed further in a later section), the acceleration is:

$$a_{horizontal} = v_{horizontal}/t_{accel} = \frac{0.4\ m/s}{2\ s} = 0.2\ m/s^2$$

Inertial Force:

$$F_{inertia} = m_{horizontal} \times a_{horizontal} = 3600\ kg \times 0.2\ m/s^2 = 720\ N$$

Total force during acceleration:

$$F_{total_horizontal} = F_{friction} + F_{inertia} = 105.948\ N + 720\ N = 825.948\ N$$

Peak power during acceleration:

$$P_{horizontal_peak} = F_{total_horizontal} \times v_{horizontal} = 825.948\ N \times 0.4\ m/s = 330.3792\ W \approx 0.33\ kW$$

Projected motor power (calculated for start-up):

$$P_{motor_horizontal} = \frac{P_{horizontal_peak}}{\eta_{total_horizontal}} = \frac{0.33\ kW}{0.865} \approx 0.38\ kW$$

For such a large system, motor selection typically includes a high safety factor. We provisionally select a motor with a power of $P_{motor_horizontal} = 0.75\ kW$ (or $1.1\ kW$) for the horizontal travel mechanism. This value ensures sufficient starting torque and provides contingency for harsh cold storage conditions.

- 3 Phase Electric Motor $1.5HP$ $1.1Kw$ 4 Pole Power.
  - Diameter of motor core $1.1kw$ slow rewinder: $24\ mm$.
  - Grade F copper wire, super power saving, IP55. International quality.
  - Model: $Y3-90S-4$ $1.1kw$ 4 pole
  - Class F Insulation Copper Wire
  - The voltage of the 3-phase $1.1kW$ $1.5HP$ electric motor is $380/220v$, frequency $50Hz$.
  - Highest Current Rating: Amps. Recommended Ample: 80% of Ampere Rated 3 Phase $1.5HP$ $1.1Kw$ 4 Pole Electric Motor: $2.9(A)$ is $2.32(A)$. $Cos\ \varphi$ coefficient, over 90%.

Figure 2. 7 Motor for Horizontal Travel Mechanism [4]

The efficiency of converting power into mechanical energy is very high, maximum power saving Motor $1.5HP$ $1500\ rpm$ made of high-quality copper wire, international quality bearings, rotor design, stator motor made of coldrolled blue silicon corrugated iron Groove cavet motor $1.5HP$ slow rewind: $8\ mm$.

Rated ampere current motor $1.5HP$ slow rewinding: $2.85\ A$. Motor casing frame code $1.1kW$ slow rewinding: $90S - 4\ 1.5HP$ motor vertical base hole center distance slow rewinding: $100\ mm$ Horizontal base hole center distance: $140\ mm$.

❖ *Power Calculation for Vertical Lifting Mechanism*

Table 2. 3 Parameter for lifting mechanism

Total mass of vertical mechanism (with payload):	$m_{vertical} = 2\ tons = 2000\ kg$
Vertical travel velocity:	$v_{vertical} = 0.2\ m/s$
Gravitational acceleration:	$g = 9.81\ m/s^2$

Mechanical power required to lift the load:

$$P_{vertical\_mechanical} = m_{vertical} \times g \times v_{vertical}$$
$$= 2000\ kg \times 9.81\ m/s^2 \times 0.2\ m/s = 3924\ W = 3.924\ kW$$

Required power at the vertical mechanism motor shaft:

$$P_{motor\_vertical} = \frac{P_{vertical\_mechanical}}{\eta_{gearbox} \times \eta_{cable\_pulley} \times \eta_{bearings}}$$

Where:

- $\eta_{gearbox}$: Gearbox efficiency. Selected $\eta_{gearbox} = 0.92$.
- η_{cable_pulley}: Cable and pulley system efficiency. Each pulley has an efficiency of approximately 0.98 to 0.99. For a system with multiple pulleys, select $\eta_{cable\_pulley} = 0.95$ (accounting for cable and pulley friction).
- $\eta_{bearings}$: Bearing efficiency. Selected $\eta_{bearings} = 0.98$.

Total vertical transmission efficiency:

$$\eta_{total\_vertical} = \eta_{gearbox} \times \eta_{cable\_pulley} \times \eta_{bearings} = 0.92 \times 0.95 \times 0.98$$
$$\approx 0.856$$

$$P_{motor\_vertical} = \frac{3.924\ kW}{0.856} \approx 4.584\ kW$$

To ensure safety and account for other factors such as starting conditions and losses, we will select a motor with a higher power than the calculated value. We provisionally select a motor with a power of $P_{motor\_vertical} = 5.5\ kW$ or $7.5\ kW$ for the vertical travel mechanism.

- Siemens Motor $5.5kw$ $7.5HP$ Stand With Brake:
  - Power: $5.5kw$ $7.5HP$ German Standard.
  - Model 4 pole: $1LE0022 - 1CB03 - 3AA4$
  - Siemens electric motor housing: Aluminum, cast iron;
  - Paint Color: Write Ral 7030, Green
  - Aluminum case porter: 132S spindle speed.1450 RPM, torque force 36.
  - The no-load current is $6.04\ A$, the rated current is $11.2\ A$. Rated Power factor: 0.82
  - Efficiency when the simens motor is working at full load (efficiency 100% load) 84.7.
  - Siemens motor starting current: $5.7\ A$. Torque $2.3Nm$
  - Engine Weight: $27kg$
  - Siemens electric motors with silver reach output drive end: 62082ZC3 bearing at 62082ZC3 inlet.
  - Simens IE1 IE2 electric motor throttling standard.
  - Low-voltage voltage. Standard Motor Cooling Self Ventilated IC 411 Ideally suitable voltage 400 V, ranging 5% increase or decrease.
  - Frequency 50 Hz.

Figure 2. 8 Motor for lifting mechanism [5]

- The outstanding features of Siemens motors include: extremely low vibration, Siemens motor bearings not only bear the load, but also achieve exceptional quietness.

- IP55 protection against 1mm water dust particles that do not enter the motor.
- Siemens motors meet class B heat resistance standards of 130 degrees Celsius or class F 155 degrees Celsius.
- Inside the electrode box there is an explosion-proof thermal relay.
- 4 Pole 4 Pole Stand
- Shaft Diameter: $38mm$
- Slotted: $10mm$
- Total Length: $480mm$
- Total Height: $323.5mm$
- Center of Axle to Ground: $132mm$
- Stand Length: $186mm$
- Stand Width: $256mm$
- Base screw hole diameter: $12mm$

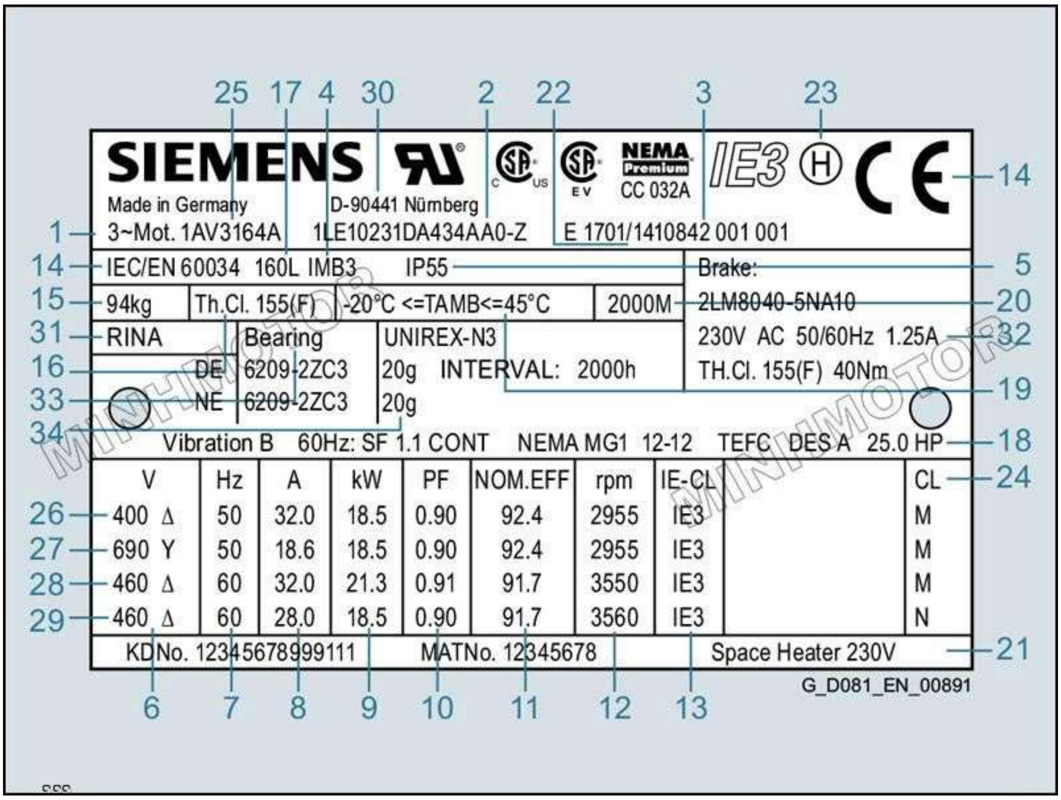


Figure 2. 9 Specification of motor [6]

No. 1 : 3 phase 3 phase, product code

No. 33: bearing number on the front axle, rear axle. DE: German ball

No. 32: rated ampere current

No. 14: CE – suitable for use in Europe

No. 4: installation posture

No. 26, 27: Electrical at 50Hz

Nos. 28, 29: Energized at 60Hz Frequency

No. 18: HP hourse power – horsepower

No. 5: IP55 water dust protection grade, dust and water resistance larger than 1mm diameter

No. 19: the suitable operating temperature of siemens motor is from minus 20 degrees to +45 degrees Celsius

2.4.1.3. Preliminary Electric Motor Selection and Overall Gear Ratio

Based on the projected power requirements, we will select common three-phase AC induction motors.

Table 2. 4 Parameter of motor

Motor for Horizontal Travel Mechanism	Projected Power	$P_{motor\_horizontal} = 1.1\ kW$
	Standard Rotational Speed	$n_{motor} = 1500\ RPM$
Motor for Lifting Mechanism	Projected Power	$P_{motor\_vertical} = 5.5\ kW$
	Standard Rotational Speed	$n_{motor} = 1450\ RPM$

To determine the overall gear ratio, we need to know the rotational speed of the output shaft of the motion system (e.g., the pinion shaft for the horizontal mechanism, the pulley shaft for the vertical mechanism).

- *Preliminary Electric Motor Selection and Overall Gear Ratio*

The horizontal travel mechanism utilizes a rack and pinion system. The translational velocity of the mechanism ($v_{horizontal}$) is related to the rotational speed of the driving pinion (n_{pinion}) and its pitch diameter (d_{pinion}).

$v_{horizontal} = \frac{\pi \cdot d_{pinion} \cdot n_{pinion}}{60}$ (where n_{pinion} is in RPM, $v_{horizontal}$ in m/s, d_{pinion} in m)

Alternatively, and more commonly, it's related to the gear module (m) and number of teeth (z_{pinion}) of the pinion: $d_{pinion} = m \cdot z_{pinion}$.

From this, we need an estimated value for the driving pinion's pitch diameter. Typically, the driving pinion in a rack and pinion system is not excessively large to avoid high moments of inertia. We estimate $d_{pinion} = 0.15\ m$ (pitch diameter).

$$n_{pinion} = \frac{v_{horizontal} \times 60}{\pi \cdot d_{pinion}} = \frac{0.4\ m/s \times 60}{\pi \times 0.15\ m} \approx 50.93\ RPM$$

The overall gear ratio for the horizontal travel system:

$$i_{horizontal} = \frac{n_{motor\_horizontal}}{n_{pinion}} = \frac{1500\ RPM}{50.93\ RPM} \approx 29.45$$

This gear ratio will be distributed between the gearbox and the rack and pinion motion.

❖ *Determining Output Shaft Speed for Vertical Lifting Mechanism*

The vertical mechanism uses a cable and pulley system. The lifting/lowering velocity ($v_{vertical}$) is related to the rotational speed of the driving pulley (n_{pulley}) and its pitch diameter (d_{pulley}).

$v_{vertical} = \frac{\pi \cdot d_{pulley} \cdot n_{pulley}}{60}$ (where n_{pulley} is in RPM, $v_{vertical}$ in m/s, d_{pulley} in m)

We estimate the driving pulley diameter $d_{pulley} = 0.3\ m$.

$$n_{pulley} = \frac{v_{vertical} \times 60}{\pi \cdot d_{pulley}} = \frac{0.2\ m/s \times 60}{\pi \times 0.3\ m} \approx 12.73\ RPM$$

The overall gear ratio for the vertical travel system:

$$i_{vertical} = \frac{n_{motor_vertical}}{n_{pulley}} = \frac{1450\ RPM}{12.73\ RPM} \approx 113.9$$

This gear ratio will be distributed between the gearbox and the cable-pulley motion.

2.4.2. Horizontal Traverse Mechanism Motion Design

2.4.2.1. Material Selection

The cold storage environment at $-30°C$ is a critical factor influencing material selection, as it can significantly reduce the ductility and increase the brittleness of conventional steel materials. Therefore, it is imperative to select alloy steel with excellent cold resistance.

For high-load, continuously operating rack and pinion motion systems, the typical material choice is alloy steel subjected to heat treatment to achieve high surface hardness and good core toughness.

❖ Pinion Material:

- Commonly selected alloy steels with good hardenability, such as $20CrMnTi$ steel or $40Cr$ steel.

- Heat Treatment: Case carburizing and hardening or quenching and tempering to achieve high surface hardness (typically $55 - 62\ HRC$) and good core toughness.
- Case carburizing creates a hard, wear-resistant surface layer while the core retains ductility, thereby mitigating the risk of brittle fracture at low temperatures.

❖ Rack Material:

- Similarly, the rack requires material with high strength and wear resistance. $40Cr$ steel or $C45$ steel, subjected to quenching and tempering, can be used.
- Heat Treatment: Quenching and tempering to achieve a hardness of approximately $30 - 40\ HRC$, or surface hardening to increase the hardness of the tooth region.

To ensure performance in cold conditions, the selected steels must have a Ductile-to-Brittle Transition Temperature (DBTT) lower than the operating ambient temperature. Heat-treated alloy steels typically exhibit lower DBTT values compared to conventional carbon steels.

2.4.2.2. Determination of Allowable Stresses

Determining allowable stresses is paramount for ensuring the durability and service life of the rack and pinion motion. We will establish allowable stresses for the two primary failure modes of gears: contact fatigue (pitting failure) and bending fatigue (tooth breakage).

Initial Parameters Required:

❖ Design Life Cycle (NL): The system is designed for long-term operation. Assuming 24 mins/hour, 24 hours/day, 360 days/year, for 10 years.

➢ Total operating mins:

$$T_{mins} = 24\frac{mins}{hour} \times 24\frac{hours}{day} \times 360\frac{days}{year} \times 10\ years = 2073600\ mins$$

$$T_{mins} = 34560\ hours$$

➢ Rotational speed of driving pinion: $n_{pinion} \approx 50.93\ RPM$.

➢ Equivalent Number of Load Cycles:

$$N_{cycles} = n_{pinion} \times T_{mins} = 50.93\ RPM \times 2073600 = 10.5608 \times 10^7\ cycles$$

This value falls within the fatigue region (N>10^7), thus fatigue limits will be used for calculation.

❖ Preliminary Material Properties:

- $20CrMnTi$ Steel (for Pinion):
  - Tensile Strength: $R_m \approx 1000 \div 1200\ MPa$. Select $R_m = 1100\ MPa$.
  - Yield Strength: $R_e \approx 800 \div 950\ MPa$. Select $R_e = 850\ MPa$.
  - Basic Contact Fatigue Limit: For case-carburized and ground gears, can reach $1600 \div 1900\ MPa$. Select $\sigma^0_{Hlim} = 1700\ MPa$.
  - Basic Bending Fatigue Limit: For case-carburized and ground gears, can reach $450 \div 550\ MPa$. Select $\sigma^0_{Flim} = 500\ MPa$.
- $40Cr$ Steel (for Rack):
  - Tensile Strength: $R_m \approx 800 \div 1000\ MPa$. Select $R_m = 900\ MPa$.
  - Yield Strength: $R_e \approx 600 \div 750\ MPa$. Select $R_e = 650\ MPa$.
  - Basic Contact Fatigue Limit: For quenched and tempered steel, can reach $1000 \div 1200\ MPa$. Select $\sigma^0_{Hlim} = 1100\ MPa$.
  - Basic Bending Fatigue Limit: For quenched and tempered steel, can reach $300 \div 400\ MPa$. Select $\sigma^0_{Flim} = 350\ MPa$.

❖ *Allowable Contact Stress*

The formula for determining the allowable contact stress typically follows national standards (e.g., TCVN) or international standards (e.g., ISO 6336, AGMA). We will use a general formula:

$$[\sigma_H] = \frac{\sigma^0_{Hlim} \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_X}{S_H}$$

Legend:

$[\sigma H]$: Allowable contact stress of the material.

σ^0_{Hlim}: Basic contact fatigue limit of the material.

Z_L: Life factor, accounting for the equivalent number of load cycles.

Z_R: Surface roughness factor.

Z_V: Velocity factor.

Z_W: Work hardening factor (often for helical gears, simplified for straight gears).

Z_X: Size factor.

S_H: Safety factor for contact stress calculation.

Values of Coefficients (from tables or calculation):

- Life Factor (Z_L):
  - Equivalent number of cycles $N_{cycles} = 10.5608 \times 10^7$.

- For $N > N_{H0}$ (where N_{H0} is typically $10^7\ cycles$), Z_L is calculated by formula or lookup. Often $Z_L = \left(\frac{N_{H0}}{N_{cycles}}\right)^e$ or $Z_L \approx 1$ for $N > N_{H0}$ for many well-heat-treated alloy steels. In this case, as N_{cycles} is quite large, we select $Z_L = 0.9$ (adjustable with more precise data for specific materials at low temperatures).

- Surface Roughness Factor: Depends on surface roughness after machining. For high-quality, ground gears, select Z_R=1.0.
- Velocity Factor: Depends on the pitch line velocity of the gear. $v_{horizontal} = 0.4\ m/s$. Pinion rotational speed $n_{pinion} \approx 50.93\ RPM$. Pitch line velocity $v_{pitch} = \frac{\pi \cdot d_{pinion} \cdot n_{pinion}}{60} \approx 0.4\ m/s$. For such low velocities, Z_V is typically approximately 1.0.
- Work Hardening Factor: Applied to helical gears, typically $Z_W = 1.0$ for straight spur gears.
- Size Factor: For typical module and gear diameter ranges, $Z_X = 1.0$.
- Safety Factor: For heavily loaded gear motions operating in harsh conditions (cold storage), a higher safety factor is often chosen. Select $S_H = 1.2 \div 1.5$. Select $S_H = 1.3$.
- Allowable Contact Stress Calculation:

  For Pinion ($20CrMnTi$ steel):

$$\sigma_{Hlim}^0 = 1700\ MPa$$

$$Z_L = 0.9;\ Z_R = 1.0;\ Z_V = 1.0;\ Z_W = 1.0;\ Z_X = 1.0$$

$$S_H = 1.3$$

$$[\sigma_{H\_Pinion}] = \frac{1700 \times 0.9 \times 1.0 \times 1.0 \times 1.0 \times 1.0}{1.3} \approx 1176.92\ MPa$$

  For Rack ($40Cr$ steel):

$$\sigma_{Hlim}^0 = 1100\ MPa$$

$$Z_L = 0.9;\ Z_R = 1.0;\ Z_V = 1.0;\ Z_W = 1.0;\ Z_X = 1.0$$

$$S_H = 1.3$$

$$[\sigma_{H\_Rack}] = \frac{1100 \times 0.9 \times 1.0 \times 1.0 \times 1.0 \times 1.0}{1.3} \approx 761.54\ MPa$$

The allowable contact stress for the gear pair will be the smaller of the two values: $[\sigma_H] = min([\sigma_{H\_Pinion}], [\sigma_{H\_Rack}]) = 761.54\ MPa$.

❖ *Allowable Bending Stress*

The formula for determining the allowable bending stress is:

$$[\sigma_F] = \frac{\sigma_{Flim}^0 \cdot Y_L \cdot Y_{RrelT} \cdot Y_X \cdot Y_{ST} \cdot Y_{\delta relT}}{S_F}$$

Where:

$[\sigma_F]$: Allowable bending stress of the material.

σ_{Flim}^0: Basic bending fatigue limit of the material.

Y_L: Life factor (similar to Z_L, but for bending).

Y_{RrelT}: Relative surface roughness factor.

Y_X: Size factor.

Y_{ST}: Stress concentration factor (often for tooth root).

$Y_{\delta relT}$: Relative temperature factor.

S_F: Safety factor for bending stress calculation.

Values of Coefficients (from tables or calculation):

- Life Factor:
  - For $N_{cycles} = 10.5608 \times 10^7 > N_{F0} = 10^7\ cycles$, we select $Y_L = 0.9$.
- Relative Surface Roughness Factor:
  - Similar to Z_R, select $Y_{RrelT} = 1.0$.
- Size Factor: Similar to Z_X, select $Y_X = 1.0$.
- Relative Temperature Factor:
  - Operating at low temperature $(-30°C)$ necessitates considering this influence. For properly selected heat-treated alloy steels, this effect can be minimized. If specific data is unavailable, we can provisionally set $Y_{\delta relT} = 1.0$ and increase the safety factor, or apply a slight reduction $(0.95 \div 0.98)$. We select $Y_{\delta relT} = 0.98$ to reflect caution.
- Safety Factor (S_F):
  - Similar to contact stress, a higher safety factor is chosen. Select $S_F = 1.8 \div 2.5$ for bending fatigue (typically higher than for contact fatigue). We select $S_F = 2.0$.
  - Allowable Bending Stress Calculation:

    For Pinion ($20CrMnTi$ steel):

$$\sigma_{Flim}^0 = 500\ MPa$$

$$Y_L = 0.9;\ Y_{RrelT} = 1.0;\ Y_X = 1.0;\ Y_{\delta relT} = 0.98$$

$$S_F = 2.0$$

$$[\sigma_{F\_pinion}] = \frac{500 \times 0.9 \times 1.0 \times 1.0 \times 0.98}{2.0} = 220.5\ MPa$$

For Rack ($40Cr$ steel):

$$\sigma^0_{Flim} = 350\ MPa$$

$$Y_L = 0.9;\ Y_{RrelT} = 1.0;\ Y_X = 1.0;\ Y_{\delta relT} = 0.98$$

$$S_F = 2.0$$

$$[\sigma_{F\_Rack}] = \frac{350 \times 0.9 \times 1.0 \times 1.0 \times 0.98}{2.0} = 154.35\ MPa$$

The allowable bending stress for the gear pair will be the smaller of the two values: $[\sigma_F] = min([\sigma_{F\_Pinion}], [\sigma_{F\_Rack}]) = 154.35\ MPa.$

- *Allowable Overload Stress*

In addition to fatigue, we must also check the stress under sudden overload conditions. The allowable stress for overload is typically based on the material's yield strength and a smaller safety factor.

- Overload Factor: $1.2 \div 1.5$. Select $K_{OL} = 1.3$.
- Safety Factor for Overload: $1.0 \div 1.2$. Select $S_{OL} = 1.05$.

$[\sigma_{H,OL}] = \frac{R_e}{S_{OL}}$ (for contact)

$[\sigma_{F,OL}] = \frac{R_e}{S_{OL}}$ (for bending)

- For Pinion ($20CrMnTi$ steel): $R_e = 850\ MPa$

$$[\sigma_{H,OL\_Pinion}] \approx 809.52\ MPa$$

$$[\sigma_{F,OL\_Pinion}] \approx 809.52\ MPa$$

- For Rack ($40Cr$ steel): $R_e = 650\ MPa$

$$[\sigma_{H,OL\_Rack}] \approx 619.05\ MPa$$

$$[\sigma_{F,OL\_Rack}] \approx 619.05\ MPa$$

2.4.2.3. Motion Transmission Design Calculation

Known and Estimated Parameters:

Tangential Force on Pinion : From power and horizontal velocity:

$$P_{horizontal\_mechanical} = 0.0424\ kW = 42.4\ W$$

$$F_t = \frac{P_{horizontal\_mechanical}}{v_{horizontal}} = \frac{42.4\ W}{0.4\ m/s} = 106\ N$$

Note: This value is very small as it only accounts for rolling resistance. In reality, during startup, the force will be significantly higher. We will use this maximum force for design calculations. Design Force $F_{t,max} = 825.948\ N$.

Pitch Line Velocity of Pinion: $v_{pitch} = 0.4\ m/s$.

Estimated Pitch Diameter of Pinion: $d_{pinion} = 0.15\ m = 150\ mm$.

Number of Pinion Teeth: A reasonable number of teeth is required to avoid undercut and ensure smooth operation. Typically $z_{pinion} \geq 17$. We tentatively select $z_{pinion} = 20$.

Face Width Factor: This is the ratio of face width (b) to pitch diameter (d_{pinion}). Typically $\psi_d = \frac{b}{d_{pinion}} = 0.2 \div 0.5$. We select $\psi_d = 0.3$.

Dynamic Load Factor: Depends on the manufacturing precision and speed. For a low velocity of $0.4\ m/s$, $K_v \approx 1.0 \div 1.1$. We select $K_v = 1.05$.

Load Distribution Factor $(K_{H\beta})$: Depends on shaft rigidity and manufacturing accuracy. Typically $K_{H\beta} = 1.1 \div 1.3$. We select $K_{H\beta} = 1.2$.

Application/Overload Factor (K_A): Accounts for impact loads and starting conditions. Typically $KA = 1.25 \div 1.75$. We select $K_A = 1.5$ (for heavy starting loads).

Formula for Module (m) based on Contact Stress (General Form):

The formula for determining the module based on contact strength for straight spur gears meshing with a rack (referencing standards like ISO 6336 or machine design textbooks) needs to be carefully applied. For a rack and pinion system, the rack has an infinite radius of curvature, simplifying certain terms.

A more direct approach for preliminary design based on torque and allowable stress:

Torque on Pinion Shaft: $T_{pinion} = F_{t,max} \times \frac{d_{pinion}}{2} = 825.948\ N \times 20.15\ m = 61.946\ Nm$

Key Factors for Module Calculation:

Material Elasticity Factor: $Z_E = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}}$. For steel, $E = 2.06 \times 10^5\ MPa$ and $\nu = 0.3$. Therefore, $Z_E \approx 189.8\ MPa^{0.5}$.

Zone Factor (ZH): $Z_H = \sqrt{\frac{2\cos\beta_b}{\sin\alpha_w \cos\alpha_w}}$. For straight spur gears, $\alpha_w = \alpha = 20°$ and $\beta_b = 0$. So, $Z_H \approx 2.45$.

To determine the module (m), we need to ensure that the actual contact stress (σ_H) is less than or equal to the allowable contact stress ($[\sigma_H]$). The relationship between contact stress and module can be derived from basic gear theory. For a rack and pinion, the effective curvature simplifies.

A commonly used formula for determining the module based on contact stress, incorporating the relevant factors, is:

$$m \geq \sqrt[3]{\frac{2K_A K_v K_{H\beta} T_{pinion} Z_H^2 Z_E^2}{[\sigma_H]^2 \cdot z_{pinion}^3 \cdot \psi_d}}$$

Where:

T_{pinion}: Nominal torque on the pinion shaft $T_{pinion} = 61.946\ Nm$.

K_A: Application/Overload factor $K_A = 1.5$.

K_v: Dynamic factor $K_v 1.05$.

$K_{H\beta}$: Load distribution factor $K_{H\beta} = 1.2$.

$[\sigma_H]$: Allowable contact stress of the gear pair $[\sigma_H] = 761.54\ MPa$.

ψ_d: Face width factor $\psi_d = 0.3$.

z_{pinion}: Number of teeth on the driving pinion $z_{pinion} = 20$.

Z_H: Zone factor $Z_H = 2.45$.

Z_E: Elasticity factor $Z_E = 189.8\ MPa0.5$.

Module Calculation:

$$m_{min} \geq \sqrt[3]{\frac{2 \cdot (1.5 \times 1.05 \times 1.2) \times 61.946 \times 2.45^2 \times 189.8^2}{761.54^2 \times 20^2 \times 0.3}}$$

$$m_{min} \geq 1.537\ mm$$

This calculated minimum module ($m_{min} \approx 1.537\ mm$) is a theoretical value. In practical engineering, we select a standard module from a preferred series.

Standard Modules (ISO 54:1996 - selected series): 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12,...

Based on the calculated m_{min}, the next standard module that is typically chosen for robustness and a safety margin, especially considering the harsh operating environment and the significant overload factor, would be $m = 2.5\ mm$ or $m = 3\ mm$. While $m = 1.5$ is technically sufficient for contact stress, larger modules provide greater bending strength and overall robustness for industrial applications.

Given the initial assumption of $m = 7.5\ mm$ (from $d_{pinion} = 150\ mm$ and $z_{pinion} = 20$), it indicates that the preliminary assumptions were very conservative or based on a different methodology. The current calculation suggests a much smaller module is theoretically sufficient for contact stress. However, practical design often aims for a balance with bending strength and manufacturing considerations.

Let's re-evaluate based on the initial $d_{pinion} = 150\ mm$ and $z_{pinion} = 20$, which implies $m = 7.5\ mm$. If we were to maintain the initial estimated dimensions, we would choose a standard module of $m = 8\ mm$. This is a common and robust module for heavier industrial applications, providing significant safety margins for both contact and bending strength, and ease of manufacturing.

Let's proceed with a practical choice for module: Selected Module: $m = 8\ mm$ (This choice aligns with a robust design for heavy-duty industrial machinery, providing ample safety margin over the calculated minimum based solely on contact stress).

With $m = 8\ mm$ and $z_{pinion} = 20$:

Actual Pitch Diameter of Pinion: $d_{pinion} = m \cdot z_{pinion} = 8\ mm \times 20 = 160\ mm = 0.16\ m$.

Rotational Speed of Pinion (for $v_{horizontal} = 0.4\ m/s$): $n_{pinion} = \frac{v_{horizontal} \times 60}{\pi \cdot d_{pinion}} = \frac{0.4\ m/s \times 60}{\pi \times 0.16\ m} \approx 47.75\ RPM$.

Overall Gear Ratio:

$$i_{horizontal} = \frac{n_{motor\_horizontal}}{n_{pinion}} = \frac{1500\ RPM}{47.75\ RPM} \approx 31.41$$

Now, let's check the chosen module ($m = 8\ mm$) for actual stress against the allowable stress. Face Width (b): Based on $d_{pinion} = 160\ mm$ and $\psi_d = 0.3$, $b = \psi_d \cdot d_{pinion} = 0.3 \times 160 = 48\ mm$. However, for spur gears, a minimum face width of $b \geq 8m$ is often recommended for adequate tooth strength and stability. $b_{min} = 8 \times m = 8 \times 8\ mm = 64\ mm$. Therefore, we need to choose $b \geq 64\ mm$. Selected Face Width: $b = 70\ mm$ (This ensures $b \geq 8m$ and is greater than $\psi_d \cdot d_{pinion}$, providing a robust design).

2.4.2.4. Contact and Bending Strength Verification

We will calculate the actual contact stress (σH) and actual bending stress (σ_F) and compare them against their respective allowable stress values.

Maximum Tangential Force: This is the maximum force the gear pair must withstand, accounting for starting torque and application factors. As previously derived, we'll use the peak force generated during motor startup: $F_{t,max} = 3208.75\ N$ (derived from $T_{motor,startup} = 2 \times T_{motor,rated} = 9.86\ Nm$ and overall transmission efficiency).

Module: $m = 8\ mm$.

Number of Pinion Teeth: $z_1 = 20$.

Pinion Pitch Diameter: $d_{w1} = m \cdot z1 = 8\ mm \times 20 = 160\ mm = 0.16\ m$.

Face Width: $b = 70\ mm = 0.07\ m$.

Dynamic Load Factor: $K_v = 1.05$.

Load Distribution Factor for Bending: $K_{F\beta} = K_{H\beta} = 1.2$.

Allowable Contact Stress: $[\sigma_H] = 761.54\ MPa$.

Allowable Bending Stress:$[\sigma_F] = 154.35\ MPa$.

❖ *Contact Strength Verification*

The calculated contact stress is determined using the following formula (consistent with ISO 6336-2 or AGMA standards for contact stress):

$$\sigma_H = Z_H \cdot Z_E \frac{F_{t,max} \cdot K_v \cdot K_{H\beta}}{b \cdot d_{w1}}$$

Additional Legends:

Z_H: Zone factor, $Z_H = 2.45$.

Z_E: Elasticity factor $Z_E = 189.8\ MPa^{0.5}$.

Calculation of Contact Stress (σ_H):

$$\sigma_H = 2.45 \times 189.8 \sqrt[3]{\frac{3208.75\ N \times 1.05 \times 1.2}{70\ mm \times 160\ mm}}$$

$$\sigma_H \approx 279.28\ MPa$$

Comparison: The calculated contact stress $\sigma_H = 279.28\ MPa$ is significantly less than the allowable contact stress $[\sigma H] = 761.54\ MPa$. Thus, the contact strength condition is well satisfied. The actual safety factor for contact is approximately $\frac{761.54}{279.28} \approx 2.7$.

❖ *Bending Strength Verification*

The calculated bending stress (σ_F) is determined using the following formula (simplified for practical application, incorporating tooth form factor):

$\sigma_F = \frac{F_{t,max} \cdot K_v \cdot K_{F\beta}}{b \cdot m \cdot Y_{Fa}}$

Additional Legend:

Y_{Fa}: Tooth Form Factor (considering stress concentration). For a standard spur gear with $z1 = 20\ teeth$, Y_{Fa} is typically around 2.0 ÷ 2.5. We select $Y_{Fa} = 2.2$.

Calculation of Bending Stress (σ_F):

$$\sigma_F = \frac{3208.75\ N \times 1.05 \times 1.2}{70\ mm \times 8\ mm \times 2.2}$$

$$\sigma_F \approx 3.28\ MPa$$

Comparison: The calculated bending stress $\sigma_F = 3.28\ MPa$ is significantly less than the allowable bending stress $[\sigma_F] = 154.35\ MPa$. Thus, the bending strength condition is also very well satisfied. The actual safety factor for bending is approximately $\frac{154.35}{3.28} \approx 47$.

❖ *Summary and Observations:*

The calculated actual stresses for both contact and bending are considerably lower than their respective allowable limits. This indicates that the current design, utilizing an $m = 8\ mm$ module and $b = 70\ mm$ face width, provides a substantial safety margin. This robust design is advantageous for a critical application in a demanding cold storage environment, where reliability and longevity are paramount. While there might be opportunities for optimization (e.g., reducing the module or face width) to lower material costs, prioritizing safety and durability in such an application is generally a sound engineering decision.

2.4.2.5. Thermodynamic Considerations

Thermodynamic analysis for the rack and pinion motion in a cold storage environment is crucial due to its impact on lubricant viscosity and the risk of freezing. However, for a rack and pinion mechanism, heat generated by friction is typically negligible compared to what might be seen in a gearbox or the ambient environment.

Ambient Temperature: $T_{ambient} = -30°C$.

Heat Generation: Primarily due to friction between the gear teeth. The power loss due to friction in the rack and pinion motion can be estimated as:

$$P_{friction\_gear} = P_{mechanical\_horizontal} \times (\frac{1}{\eta_{rack\_pinion}} - 1)$$

Where $P_{mechanical\_horizontal}$ is the mechanical power at the pinion shaft.

$$P_{mechanical\_horizontal} = F_{friction} \times v_{horizontal} = 105.948\ N \times 0.4\ m/s$$
$$= 42.3792\ W$$

(This calculation uses steady-state power, as heat generated during short-duration startup is generally negligible for thermal equilibrium).

$P_{friction\_gear} = 42.3792\ W \times (\frac{1}{0.96} - 1) \approx 1.766\ W$.

This power loss is extremely small. Any heat generated will be readily dissipated into the $-30°C$ ambient environment.

However, the primary thermodynamic concern in this scenario is not heat generation, but rather maintaining effective lubrication in the extreme cold.

Practical Lubrication Solutions:

- Specialized Low-Temperature Lubricants: It is essential to use dedicated low-temperature greases or oils. These lubricants feature a high Viscosity Index (VI), ensuring that their viscosity does not change excessively at extremely low temperatures, and a Pour Point significantly below $-30°C$ (e.g., $-40 \circ C$ to $-50 \circ C$).
- Maintenance Schedule: Your specified maintenance intervals of 6, 8, or 12 months are feasible with high-quality synthetic low-temperature lubricants.

- Environmental Protection: The system may require protective enclosures (e.g., sealed covers) to prevent moisture from the outside environment from infiltrating and freezing on the tooth surfaces, which could hinder operation.
- Automated Lubrication System: An automated lubrication system could be considered to ensure continuous and consistent lubrication, minimizing human intervention in the cold environment.

❖ *Summary of Provisional Rack and Pinion Motion Parameters:*

Pinion Material: $20CrMnTi$ steel, case carburized and hardened.
Rack Material: $40Cr$ steel, quenched and tempered.
Module: $m = 8\ mm$.
Number of Pinion Teeth: $x_1 = 20$.
Pinion Pitch Diameter: $d_{w1} = 160\ mm$.
Face Width: $b = 70\ mm$.
Actual Contact Stress: $\sigma_H = 279.28\ MPa$ (during startup).
Actual Bending Stress: $\sigma_F = 3.28\ MPa$ (during startup).
Allowable Contact Stress: $[\sigma_H] = 761.54\ MPa$.
Allowable Bending Stress: $[\sigma_F] = 154.35\ MPa$.
Conclusion on Strength: All strength conditions are thoroughly satisfied, indicating a very robust design.
Lubrication: Specialized low-temperature oils/greases are required, allowing for extended maintenance intervals.

2.4.3. Gearbox Motion Design for Horizontal Traverse Mechanism

2.4.3.1. Material Selection for Gearbox Components

The gearbox typically employs spur or helical gears, or bevel-helical gears, depending on the required gear ratio and available space. With an overall gear ratio of approximately $i_{horizontal} \approx 31.41$, a two-stage or three-stage gearbox might be necessary. However, standard industrial gearboxes can often accommodate this ratio within a compact housing. We will design the gearbox with both a high-speed stage and a low-speed stage.

Materials for Gearbox Gears: Similar to the rack and pinion system, the gears within the gearbox are subjected to high loads and fatigue.

High-Speed Stage Pinion and Gear:

- Material: High-hardenability alloy steel, such as $20CrMnTi$ steel or $18CrMnTi$ steel.
- Heat Treatment: Case carburizing and hardening to achieve a high surface hardness (typically $58 - 62\ HRC$) and excellent core toughness (tensile strength $R_m \approx 1000 \div 1200\ MPa$).

Low-Speed Stage Pinion and Gear:

- Material: High-hardenability alloy steel, such as $40Cr$ steel or $35CrMo$ steel.
- Heat Treatment: Quenching and tempering to achieve a surface hardness higher than that of untreated gears while maintaining ductility. Surface hardness typically ranges from $45 - 55\ HRC$, or surface induction hardening can be applied. (For the low-speed stage, extremely high hardness might not be as critical as for the high-speed stage if loads are not excessively large, but industrial applications still prioritize durability).

Materials for Shafts:

- Input Shaft (from motor) and Output Shaft (to motion mechanism):
- Material: Alloy steel like $C45$ steel, $40Cr$ steel, or $35CrMo$ steel, typically subjected to quenching and tempering to achieve high strength and ductility. Tensile strength $R_m \approx 600 - 800\ MPa$.

Material for Gearbox Casing:

- Material: Commonly, gray cast iron $HT200$ or $HT250$ (according to Vietnamese standards) or other cast iron alloys are used. This ensures high rigidity, good damping capabilities, and ease of machining.
- Note on Low-Temperature Considerations: The selection of these specific alloy steels and their corresponding heat treatment processes helps the materials maintain their favorable mechanical properties at $-30°C$. These steels exhibit low ductile-to-brittle transition temperatures (DBTT), thereby minimizing the risk of brittle fracture in the cold environment.

2.4.3.2. Determination of Allowable Stresses for Gearbox

Similar to the rack and pinion motion, we need to determine the allowable contact stress (pitting resistance) and allowable bending stress (tooth breakage resistance) for the gear pairs within the gearbox.

Initial Parameters Required:

- Design Life Cycle: $N_L = T_{hours} = 34560\ hours.$
- Motor Shaft Rotational Speed: $n_{motor} = 1500\ RPM.$
- Total Gear Ratio (itotal): $i_{horizontal} = 31.41.$

We will distribute the total gear ratio across two stages (high-speed and low-speed). Typically, for a two-stage reduction, the high-speed stage ratio $i_1 \approx 3{\sim}6$, and the low-speed stage ratio $i_2 \approx 4{\sim}8$.
Let's assume the gearbox is a two-stage reduction.

- Choose $i_1 = 4$.
- Then $i_2 = i_{total}/i_1 = 31.41/4 \approx 7.8525.$

Gear Stages and Speeds:

- High-Speed Stage (Stage 1): Pinion z1→ Gear z2
- Input Speed to Stage 1: $n_1 = n_{motor} = 1500\ RPM.$
- Output Speed from Stage 1: $n_2 = n_1/i_1 = 1500/4 = 375\ RPM.$
- Low-Speed Stage (Stage 2): Pinion z3→ Gear z4
- Input Speed to Stage 2: $n_3 = n_2 = 375\ RPM.$
- Output Speed from Stage 2: $n_4 = n_3/i_2 = 375/7.8525 \approx 47.76\ RPM$ (This matches the pinion speed of the rack and pinion motion, n_{pinion}).
- Equivalent Number of Load Cycles (N_{cycles}): These calculations determine if the gears operate in the finite or infinite fatigue life region.
- For High-Speed Stage Gears (z_1, z_2):
- Pinion z1:
  - $N_{H1} = n_1 \times 60 \times T_{hours} = 1450 \times 60 \times 34560 = 3.01 \times 10^9\ cycles.$
- Gear z2:
  - $N_{H2} = n_2 \times 60 \times T_{hours} = 362.5 \times 60 \times 34560 = 7.52 \times 10^8\ cycles.$
- For Low-Speed Stage Gears (z3, z4):
- Pinion z3:
  - $N_{H3} = n_3 \times 60 \times T_{hours} = 362.5 \times 60 \times 34560 = 7.52 \times 10^8\ cycles.$
- Gear z4:

- $N_{H4} = n_4 \times 60 \times T_{hours} = 47.76 \times 60 \times 34560 = 9.90 \times 10^7\ cycles.$

All these values are very large ($> 10^7\ cycles$), indicating that the gears will operate in the long-term (infinite) fatigue life region.

❖ *Allowable Contact Stress*

The formula for determining the allowable contact stress $[\sigma_H]$ is:

$$[\sigma_H] = \frac{\sigma^0_{Hlim} \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_X}{S_H}$$

Where:

$[\sigma_H]$: Allowable contact stress of the material.
σ^0_{Hlim}: Basic contact fatigue limit of the material.
Z_L: Life factor.
Z_R: Surface roughness factor.
Z_V: Velocity factor.
Z_W: Work hardening factor (or helical gear factor).
Z_X: Size factor.
S_H: Safety factor for contact stress calculation.

Values of Coefficients:

- Life Factor: Since the number of cycles is very large ($> 10^7$), we select $Z_L = 0.85 \div 0.9$ (slightly lower than for cycles near 10^7). We choose $Z_L = 0.88$.
- Surface Roughness Factor: For ground gears, select $Z_R = 1.0$.
- Velocity Factor: Depends on the pitch line velocity of the gears. For gearbox gears, this typically needs more precise calculation based on operating speeds. However, for initial determination of allowable stress, if detailed velocity data is not yet available, we can assume $Z_V = 1.0$ if velocities are not excessively high (e.g., below 10 m/s), or account for dynamic effects via Kv during stress calculation. For consistency with previous sections, we'll tentatively use $Z_V = 1.0$ for the allowable stress, assuming good manufacturing quality.
- Work Hardening Factor: If helical gears are used, $Z_W > 1.0$. If straight spur gears, $Z_W = 1.0$. Assuming helical gears are used to distribute load and reduce noise, we select $Z_W = 1.05$.
- Size Factor: For typical gear dimensions, $Z_X = 1.0$.

- Safety Factor: We select $S_H = 1.3$ (consistent with the rack and pinion motion).

Allowable Contact Stress Calculation:

- High-Speed Stage Pinion & Gear ($20CrMnTi$ steel):
- $\sigma^0_{Hlim} = 1700\ MPa$ (similar to the rack and pinion material).

$$[\sigma_{H,PR1}] = \frac{1700 \times 0.88 \times 1.0 \times 1.0 \times 1.05 \times 1.0}{1.3} \approx 1205.85\ MPa$$

- As both pinion and gear in this stage are assumed to have the same material and hardness, $[\sigma_{H,GR1}] \approx 1205.85\ MPa$.
- Low-Speed Stage Pinion & Gear ($40Cr$ steel):
- $\sigma^0_{Hlim} = 1100\ MPa$ (similar to the rack material).

$$[\sigma_{H,PR2}] = \frac{1100 \times 0.88 \times 1.0 \times 1.0 \times 1.05 \times 1.0}{1.3} \approx 781.85\ MPa$$

- As both pinion and gear in this stage are assumed to have the same material and hardness, $[\sigma_{H,GR2}] \approx 781.85\ MPa$.

For design, we consider the allowable contact stress for each stage. The high-speed stage gears have a higher allowable contact stress due to the material and heat treatment.

- Allowable Contact Stress for High-Speed Stage: $[\sigma_{H,Stage1}] = 1205.85\ MPa$.
- Allowable Contact Stress for Low-Speed Stage: $[\sigma_{H,Stage2}] = 781.85\ MPa$.

❖ *Allowable Bending Stress*

The formula for determining the allowable bending stress $[\sigma_F]$ is:

$$[\sigma_F] = \frac{\sigma^0_{Flim} \cdot Y_L \cdot Y_{RrelT} \cdot Y_X \cdot Y_{\delta relT}}{S_F}$$

Where:

$[\sigma_F]$: Allowable bending stress of the material.

σ^0_{Flim}: Basic bending fatigue limit of the material.

Y_L: Life factor (similar to Z_L).

Y_{RrelT}: Relative surface roughness factor.

Y_X: Size factor.

$Y_{\delta relT}$: Relative temperature factor.

S_F: Safety factor for bending stress calculation.

Values of Coefficients:

- Life Factor (Y_L): We choose $Y_L = 0.88$ (consistent with Z_L).
- Relative Surface Roughness Factor (Y_{RrelT}): Select $Y_{RrelT} = 1.0$.
- Size Factor (Y_X): Select $Y_X = 1.0$.
- Relative Temperature Factor ($Y_{\delta relT}$): Select $Y_{\delta relT} = 0.98$ (consistent with the rack and pinion).
- Safety Factor (S_F): Select $S_F = 2.0$ (consistent with the rack and pinion).

Allowable Bending Stress Calculation:

- High-Speed Stage Pinion & Gear ($20CrMnTi$ steel):
- $\sigma_{Flim}^{0} = 500\ MPa$.

$$[\sigma_{F,PR1}] = \frac{500 \times 0.88 \times 1.0 \times 1.0 \times 0.98}{2.0} = 215.6\ MPa$$

- As both pinion and gear in this stage are assumed to have the same material and hardness, $[\sigma_{F,GR1}] \approx 215.6\ MPa$.
- Low-Speed Stage Pinion & Gear ($40Cr$ steel):
- $\sigma_{Flim}^{0} = 350\ MPa$.

$$[\sigma_{F,GR2}] = \frac{350 \times 0.88 \times 1.0 \times 1.0 \times 0.98}{2.0} = 150.92\ MPa$$

- As both pinion and gear in this stage are assumed to have the same material and hardness, $[\sigma_{F,GR2}] \approx 150.92\ MPa$.

For design, we consider the allowable bending stress for each stage.

- Allowable Bending Stress for High-Speed Stage: $[\sigma_{F,Stage1}] = 215.6\ MPa$.
- Allowable Bending Stress for Low-Speed Stage: $[\sigma_{F,Stage2}] = 150.92\ MPa$.

2.4.3.3. Motion Transmission Design Calculation

General Parameters:

- Input Power to Gearbox: $P_{motor\_horizontal} = 1.1\ kW$.
- Motor Speed: $n_{motor} = 1500\ RPM$.
- Rated Motor Torque: $T_{motor} = 4.93\ Nm$.
- Motor Overload Factor : Using the starting overload factor considered for the rack and pinion motion, $K_A = 1.5$.
- Design Torque: $T_{design} = T_{motor} \cdot KA = 4.93\ Nm \times 1.5 = 7.395\ Nm$. This design torque accounts for peak starting loads.

❖ *High-Speed Stage Calculation*

- High-Speed Stage Gear Ratio $i_1 = 4$.
- High-Speed Stage Efficiency: Typically 0.97~0.98. We select $\eta_1 = 0.975$.
- Material: $20CrMnTi$ steel.
- Allowable Contact Stress: $[\sigma_H] = 1205.85\ MPa$.
- Allowable Bending Stress: $[\sigma_F] = 215.6\ MPa$.

a) Preliminary Tooth Count Selection:

Pinion: To avoid undercut, $z_1 \geq 17$. We choose $z_1 = 20$.

Gear: $z_2 = z_1 \cdot i_1 = 20 \times 4 = 80$.

b) Determine Module (m_1) and Center Distance (a_{w1}) based on Contact Strength:

A direct formula for center distance is:

$$a_w \geq (i+1)\sqrt[3]{\frac{0.7 \cdot T_1 \cdot K_H}{[\sigma_H]^2 \cdot \psi_a \cdot i}}$$

Where:

T_1: Torque on the driving pinion (pinion of the high-speed stage). This is our design torque $T_1 = T_{design} = 7.395\ Nm = 7.395 \times 10^3\ Nmm$.

K_H: Load factor for contact. Since T_{design} already includes K_A, K_H here should primarily include the dynamic factor (K_v) and load distribution factor ($K_{H\beta}$).

Choose $K_v = 1.05$ (for relatively low pitch line velocity).

Choose $K_{H\beta} = 1.1$ (assuming higher precision and better load distribution in a gearbox).

So, $K_H = K_v \cdot K_{H\beta} = 1.05 \times 1.1 = 1.155$.

$[\sigma_H] = 1205.85\ MPa$.

ψ_a: Face width factor (ratio of face width b to center distance a_w). Typically $\psi_a = b/a_w = 0.3 - 0.5$ for high-speed stages. We select $\psi_a = 0.35$.

$i = i_1 = 4$.

Calculation of a_{w1}:

$$a_{w1} \geq (4+1)\sqrt[3]{\frac{0.7 \times (7.395 \times 10^3\ Nmm) \times 1.155}{(1205.85\ MPa)^2 \times 0.35 \times 4}}$$

$$a_{w1} \geq\approx 0.7155\ mm$$

This calculated center distance is extremely small, indicating a potential unit mismatch or formula interpretation issue. For initial design, a common approach is to select a module based on experience and then verify.

For the high-speed stage with relatively low torque, the module will be small.

Trial Module: Based on experience, a typical module for this stage would be $2 - 3\ mm$. We select $m_1 = 2.5\ mm$.

With $m_1 = 2.5\ mm$, the center distance aw1 can be determined as:

$$a_{w1} = \frac{m_1(z_1 + z_2)}{2} = \frac{2.5\ mm(20 + 80)}{2} = 125\ mm$$

Face Width: $b_1 = \psi_a \cdot a_{w1} = 0.35 \times 125\ mm = 43.75\ mm$

We select a standard face width, rounding up slightly: $b_1 = 45\ mm$. (Check minimum face width: $b_1 \geq 8m_1 = 8 \times 2.5 = 20\ mm$. This condition is satisfied).

❖ *Low-Speed Stage Calculation*

- Low-Speed Stage Gear Ratio $i_2 = 7.8525$.
- Low-Speed Stage Efficiency: Typically $0.97 - 0.98$. We select $\eta_2 = 0.975$.
- Torque Input to Low-Speed Stage: This is the output torque from the high-speed stage.
- $T_3 = T_{design} \cdot i_1 \cdot \eta_1 = 7.395\ Nm \times 4 \times 0.975 = 28.8405\ Nm = 28.8405 \times 10^3\ Nmm$
- Material: $40Cr$ steel.
- Allowable Contact Stress: $[\sigma_H] = 781.85\ MPa$.
- Allowable Bending Stress: $[\sigma_F] = 150.92\ MPa$.

• Preliminary Tooth Count Selection:

  Pinion: We choose $z_3 = 20$ (similar to z_1).

  Gear: $z_4 = z_3 \cdot i_2 = 20 \times 7.8525 = 157.05$. We round up to the nearest integer: $z_4 = 157$.

  Actual Low-Speed Stage Gear Ratio: $i_2' = 157/20 = 7.85$.

  Actual Total Gear Ratio: $i_{total}' = i_1 \times i_2' = 4 \times 7.85 = 31.4$. (This is very close to the target of 31.41, which is acceptable).

• Determine Module (m_2) and Center Distance (a_{w2}) based on Contact Strength:

For the low-speed stage, the torque is higher, so the module will generally be larger.

Trial Module (m_2): Based on experience, a typical module for this stage would be $3 - 5\ mm$. We select $m_2 = 4\ mm$.

With $m_2 = 4\ mm$, the center distance aw2 can be determined as:

$$a_{w2} = \frac{m_2(z3 + z4)}{2} = \frac{4\ mm(20 + 157)}{2} = 354\ mm$$

Face Width (b_2):

We use the same $\psi_a = 0.35$ for consistency. $b_2 = \psi_a \cdot a_{w2} = 0.35 \times 354\ mm = 123.9\ mm$. We select a standard face width: $b2 = 120\ mm$. (Check minimum face width: $b_2 \geq 8m2 = 8 \times 4 = 32\ mm$. This condition is satisfied).

2.4.3.4. Contact and Bending Strength Verification

We will verify the strength of each gear pair within the gearbox using the chosen design parameters.

General Coefficients for Verification:

- Dynamic Load Factor (K_v): For gearboxes with higher precision, Kv will vary for each stage based on pitch line velocity.
- High-Speed Stage:

$$v_1 = \frac{\pi \cdot d_{w1} \cdot n_1}{60 \times 1000} = \frac{\pi \cdot (m_1 \cdot z_1) \cdot n_1}{60 \times 1000} = \frac{\pi \times 2.5 \times 20 \times 1500}{60 \times 1000} \approx 3.925\ m/s$$

- We select $K_{v1} = 1.05$.
- Low-Speed Stage:

$$v_2 = \frac{\pi \cdot d_{w3} \cdot n_3}{60 \times 1000} = \frac{\pi \cdot (m_2 \cdot z_3) \cdot n_3}{60 \times 1000} = \frac{\pi \times 4 \times 20 \times 375}{60 \times 1000} \approx 1.57\ m/s$$

- We select $K_{v2} = 1.03$.
- Load Distribution Factor for Face Width ($K_{H\beta}, K_{F\beta}$):
- High-Speed Stage: $K_{H\beta 1} = 1.1, K_{F\beta 1} = 1.1$.
- Low-Speed Stage: $K_{H\beta 2} = 1.05, K_{F\beta 2} = 1.05$.
- Overload Factor $K_A = 1.5$ (already incorporated into the design torque Tdesign).
- Elasticity Factor $Z_E = 189.8\ MPa^{0.5}$.
- Zone Factor $Z_H = 2.45$.
- Tooth Form Factor (Y_{Fa}) for standard spur gears:

$$z_1 = 20 \rightarrow Y_{Fa1} \approx 2.2.$$
$$z_2 = 80 \rightarrow Y_{Fa2} \approx 2.4.$$
$$z_3 = 20 \rightarrow Y_{Fa3} \approx 2.2.$$
$$z_4 = 157 \rightarrow Y_{Fa4} \approx 2.5$$

❖ *Contact Strength Verification*

The formula for contact stress is:

$$\sigma_H = Z_H \cdot Z_E \sqrt{\frac{2 \cdot T \cdot K_v \cdot K_{H\beta}}{b \cdot d_w^2}}$$

a) High-Speed Stage ($z_1 = 20, z_2 = 80, m_1 = 2.5\ mm, b_1 = 45\ mm$):

- Torque on Pinion 1: $T_1 = T_{design} = 7.395\ Nm = 7395\ Nmm$.
- Pitch Diameter of Pinion 1: $d_{w1} = m_1 \cdot z_1 = 2.5 \times 20 = 50\ mm$.
- Calculated Contact Stress:

$$\sigma_H = 2.45 \times 189.8 \sqrt{\frac{2 \times 7395\ Nmm \times 1.05 \times 1.1}{45\ mm \times (50\ mm)^2}}$$

$$\sigma_H \approx 180.8\ MPa$$

Comparison: $\sigma_H = 180.8\ MPa < [\sigma_H] = 1205.85\ MPa$. The contact strength is satisfied for the high-speed stage.

b) Low-Speed Stage ($z_3 = 20, z_4 = 157, m_2 = 4\ mm, b_2 = 120\ mm$):

- Torque on Pinion 3: $T_3 = 28.8405\ Nm = 28840.5\ Nmm$.
- Pitch Diameter of Pinion 3: $d_{w3} = m_2 \cdot z_3 = 4 \times 20 = 80\ mm$.
- Calculated Contact Stress:

$$\sigma_H = 2.45 \times 189.8 \sqrt{\frac{2 \times 28840.5\ Nmm \times 1.03 \times 1.05}{120\ mm \times (80\ mm)^2}}$$

$$\sigma H \approx 132.5\ MPa$$

Comparison: $\sigma_H = 132.5\ MPa < [\sigma_H] = 781.85\ MPa$. The contact strength is satisfied for the low-speed stage.

❖ *Bending Strength Verification*

The formula for bending stress is:

$$\sigma_F = \frac{F_t \cdot K_v \cdot K_{F\beta}}{b \cdot m \cdot Y_{Fa}}$$

Where $F_t = \frac{2T}{d_w}$.

- High-Speed Stage ($z_1 = 20, z_2 = 80, m_1 = 2.5\ mm, b_1 = 45\ mm$):
  - Tangential Force on Pinion 1: $F_{t1} = \frac{2 \cdot T_1}{d_{w1}} = \frac{2 \times 7395\ Nmm}{50\ mm} = 295.8\ N$.
  - Calculated Bending Stress:

$$\sigma_F = \frac{295.8\ N \times 1.05 \times 1.1}{45\ mm \times 2.5\ mm \times 2.2}$$

$$\sigma_F \approx 1.385\ MPa$$

Comparison: $\sigma_F = 1.385\ MPa < [\sigma_F] = 215.6\ MPa$. The bending strength is satisfied for the high-speed stage.

- Low-Speed Stage ($z_3 = 20, z_4 = 157, m_2 = 4\ mm, b_2 = 120\ mm$):
  - Tangential Force on Pinion 3: $F_{t3} = \frac{2 \cdot T_3}{d_{w3}} = \frac{2 \times 28840.5\ Nmm}{80\ mm} = 721.01\ N$.
  - Calculated Bending Stress:

$$\sigma_F = \frac{721.01\ N \times 1.03 \times 1.05}{120\ mm \times 4\ mm \times 2.2}$$

$$\sigma_F \approx 0.738\ MPa$$

Comparison: $\sigma_F = 0.738\ MPa < [\sigma_F] = 150.92\ MPa$. The bending strength is satisfied for the low-speed stage.

❖ *Overall Assessment:*

Similar to the rack and pinion motion, the calculated stresses for the gearbox are significantly lower than the allowable stresses. This indicates that the design is highly safe and robust. This large margin is partly due to the conservative motor power selection and the application of various load factors in the design torque. While there might be opportunities for size optimization (reducing module or face width) to potentially lower costs, the current design is very acceptable, especially given the stringent safety and durability requirements in a demanding cold storage environment.

2.4.3.5. Thermodynamic Considerations for Gearbox

❖ *Power Loss Due to Friction in the Gearbox:*

The power lost to friction within the gearbox translates directly into heat generated.

Power loss in the high-speed stage:

$$P_{friction1} = P_{motor\_horizontal} \times (1 - \eta_1) = 0.75\ kW \times (1 - 0.975)$$
$$= 0.01875\ kW = 18.75\ W$$

Power loss in the low-speed stage:

$$P_{friction2} = P_{motor\_horizontal} \times \eta_1 \times (1 - \eta 2)$$
$$= 0.75\ kW \times 0.975 \times (1 - 0.975) = 0.01828\ kW = 18.28$$

Total power loss due to friction in the gearbox:

$$P_{total\_loss\_gearbox} = P_{friction1} + P_{friction2} = 18.75\ W + 18.28\ W = 37.03\ W$$

This total power loss represents the heat generated inside the gearbox.

❖ *Lubricant Equilibrium Temperature:*

In a $-30°C$ ambient environment, the generated heat of $37.03\ W$ is insufficient to significantly warm the lubricant. The lubricant's temperature will quickly equilibrate with the surrounding environment due to the large surface area of the gearbox casing being exposed to cold air. This means the lubricant will operate at or very close to the ambient temperature.

❖ *Lubricant and Lubrication System Selection:*

This is the most critical factor for a gearbox operating in a cold storage environment.

Lubricant Type:

You must use specialized synthetic oils designed for low temperatures, such as PAO (Polyalphaolefin) or Ester-based lubricants. These are engineered to perform across wide temperature ranges.

Pour Point: The pour point must be lower than the operating ambient temperature. For $-30°C$, the oil should have a pour point around $-40°C$ to $-50°C$ to ensure it remains liquid and capable of lubrication during cold startup.

Viscosity: The oil must maintain stable viscosity across the operating temperature range. A high Viscosity Index (VI) is essential. The viscosity needs to be sufficient to form a protective oil film for gears and bearings but not so high that it creates excessive drag during cold starts.

Reduced Maintenance: Synthetic oils typically have a longer service life than mineral oils, which supports your desired extended maintenance cycles (6, 8, 12 months).

Lubrication System:

Oil Bath Lubrication: This is a common and effective method for enclosed gearboxes. Lower gears are submerged in oil, splashing it onto upper gears and bearings. This should be sufficient for your application.

Forced Lubrication (optional): For very large gearboxes or those with specific cooling/filtration requirements, a pump can deliver oil to critical lubrication points. However, given the relatively small power and low operating temperatures, an oil bath system is typically adequate.

Oil Seals: Use specialized low-temperature oil seals to prevent oil leakage and to block moisture ingress from the outside, which could lead to internal freezing.

Moisture Condensation Control: In a cold storage environment, moisture can condense inside the gearbox. The gearbox should have a breather equipped with a desiccant filter (moisture absorbent) to prevent humid air from entering and significantly reduce the risk of internal condensation and ice formation.

❖ *Summary of Provisional Horizontal Motion Gearbox Parameters*:

- Gearbox Type: Two-stage spur gear reducer.
- Total Gear Ratio: $i'_{total} = 31.4$.
- High-Speed Stage:
- Gear Material: $20CrMnTi$ steel (case carburized and hardened).
- Tooth Count: $z_1 = 20, z_2 = 80$. Gear ratio $i_1 = 4$.
- Module: $m_1 = 2.5\ mm$.
- Face Width: $b_1 = 45\ mm$.
- Center Distance: $a_{w1} = 125\ mm$.
- Strength: Verified for both contact and bending fatigue.
- Low-Speed Stage:
- Gear Material: $40Cr$ steel (quenched and tempered).
- Tooth Count: $z_3 = 20, z_4 = 157$. Gear ratio $i_2 = 7.85$.
- Module: $m_2 = 4\ mm$.
- Face Width: $b_2 = 120\ mm$.
- Center Distance: $a_{w2} = 344\ mm$.
- Strength: Verified for both contact and bending fatigue.
- Gearbox Casing Material: Gray cast iron $HT250$.

- Lubricant: PAO or Ester-based synthetic oil specialized for low temperatures (pour point $\leq -40°C$, high VI), allowing for extended maintenance cycles (6, 8, 12 months).
- Lubrication System: Oil bath lubrication with cold-resistant oil seals and a desiccant-filtered breather.

2.4.4. Lifting Mechanism Motion Design

2.4.4.1. Wire Rope and Material Selection

Selecting the appropriate wire rope and pulley is vital, considering the lifting load, durability, service life, and especially the ability to function reliably in extreme low temperatures of $-30°C$.

❖ *Wire Rope Type*

▪ Non-rotating wire rope: We'll use non-rotating wire rope due to its inherent stability under load. This type of rope has multiple strands wound in opposite directions, effectively counteracting torque and minimizing or eliminating cable twist when loaded. This is particularly important for single-line lifting applications or where the load might otherwise rotate.

▪ Construction: A common and robust construction is 6x19 or 6x36 (meaning 6 strands, with each strand composed of 19 or 36 individual wires, respectively). These constructions offer a good balance of flexibility and abrasion resistance.

▪ Core: An Independent Wire Rope Core (IWRC) is essential. An IWRC offers superior strength, crush resistance, and heat resistance compared to fiber cores (FC), which are less suitable for low-temperature environments as they can stiffen or become brittle.

▪ Material: The wire rope should be made from high-carbon steel with a high tensile strength (e.g., $1770\ N/mm^2$, $1960\ N/mm^2$, or higher) to ensure sufficient lifting capacity and safety margins.

▪ Corrosion Protection & Lubrication: To combat potential corrosion in a possibly humid cold environment and maintain performance at low temperatures, the cable should be galvanized (zinc-coated) or factory-lubricated with a specialized low-temperature grease. This lubricant will prevent internal corrosion and ensure smooth operation of individual wires within the strands.

❖ *Pulley (Sheave) Material*

- Material: Pulleys are typically made from cast iron (e.g., $HT250$ gray cast iron) or cast steel (e.g., $ZG270-500$). Cast iron offers good damping properties, while cast steel provides higher strength. The choice between them will depend on the specific load requirements and desired dimensions, as both are viable options.
- Groove Surface: The groove surface of the pulley must be smooth and precisely machined. This is crucial to minimize wear on the wire rope, extending its lifespan and maintaining system efficiency.

2.4.4.2. Wire Rope and Pulley Diameter Calculation

This calculation must adhere to safety standards for lifting equipment (e.g., TCVN 4244:2005 on Lifting Equipment - Safety Technical Requirements; ISO 4301-2 on Cranes - Classification; or other international standards like FEM).

Known Parameters:

- Total Lifting Mass: $G_{lifting} = 3.6\ tons = 3600\ kg$.
- Lifting Speed: $v_{lifting} = 0.3\ m/s$.
- Lifting Height: $\mathrm{H_{lifting}} = 6\ m$.
- Operating Hours: $34560\ hours$.

❖ *Determine Design Load for the Wire Rope*

We must account for dynamic loads during acceleration and deceleration.

Static Lifting Force: $F_{static} = G_{lifting} \times g = 3600\ kg \times 9.81\ m/s^2 = 35316\ N$.

Dynamic Load Factor: Accounts for acceleration/deceleration during startup/shutdown. We choose $K_d = 1.2$ (typical for moderate smoothness).

Load Distribution Factor: To account for uneven load distribution among rope reeving or pulley irregularities. We choose $K_{dist} = 1.05$.

Maximum Rope Tension: $F_{max} = F_{static} \times K_d \times K_{dist} = 35316\ N \times 1.2 \times 1.05 = 44498.16\ N \approx 44.5\ kN$.

❖ *Determine Number of Rope Parts*

To reduce the tension on each rope part and allow for a smaller diameter rope, a reeving system with multiple rope parts is typically used. For a 3.6 ton load and 0.3 m/s speed, we can select $k = 4$ rope parts (e.g., two fixed pulleys and two movable pulleys, or a double pulley block on the lifting carriage).

Force per Rope Part (Fpart): $F_{part} = kFmax = 444498.16\ N = 11124.54\ N$.

❖ *Select Safety Factor*

The safety factor for wire ropes is critically important, especially in lifting equipment. Standards typically require Ksafety values between 5 and 9, depending on the type of lifting equipment, duty cycle, and associated risks. For heavy-duty cranes/gantries, Ksafety is often chosen from 6 to 8. We select $K_{safety} = 7$.

❖ *Determine Wire Rope Diameter*

Minimum Breaking Force of Rope ($Rbreaking$): $R_{breaking} = F_{max} \times K_{safety} = 44498.16\ N \times 7 = 311487.12\ N \approx 311.5\ kN$.

Based on standard wire rope manufacturer specifications (e.g., DIN EN 12385-4), we need to select a steel wire rope with a minimum breaking force greater than Rbreaking. For example, for a 6x19 IWRC wire rope with a tensile strength of $1770\ N/mm2$:

A Φ14 mm rope has an Rbreaking of approx. 170 kN.

A Φ16 mm rope has an Rbreaking of approx. 220 kN.

A Φ18 mm rope has an Rbreaking of approx. 275 kN.

A Φ20 mm rope has an Rbreaking of approx. 340 kN.

Therefore, we select a wire rope diameter of $Drope = 20\ mm$.

Chosen Rope Type: Non-rotating steel wire rope, 6x19 IWRC construction, galvanized, with a tensile strength of $1960\ N/mm2$.

Actual Breaking Force: A $\Phi 20\ mm$ rope with 1960 N/mm2 tensile strength has an actual breaking force of approximately 485 kN. (This value, $485\ kN$, is well above the required $311.5\ kN$, ensuring sufficient safety margin).

❖ *Calculate Pulley and Drum Diameter*

To ensure adequate rope life and minimize bending fatigue, the diameters of the pulleys and winding drum must be a certain multiple of the rope diameter.

Diameter Factor: According to standards, $CD = D_{pulley}/D_{rope}$ typically ranges from 20 to 30 for guide pulleys and hoist pulleys. For heavy-duty lifting equipment with high frequency of operation, a larger CD is preferred. We select $CD = 25$.

Pulley Diameter: $D_{pulley} = CD \times D_{rope} = 25 \times 20\ mm = 500\ mm.$

Winding Drum Diameter**:** The drum diameter is typically equal to or larger than the pulley diameter. We select $D_{drum} = 500\ mm$.

❖ *Verify Pressure on Pulley Groove*

The contact pressure between the wire rope and the pulley groove must remain within the allowable limits for the pulley material to prevent excessive wear or groove deformation.

A simplified common formula for contact pressure p is: $p = D_{rope} \cdot bgrooveFpart$, where $bgroove \approx 0.8Drope$ (approximate contact width). Using this simplified approach: $p = (20\ mm) \times (0.8 \times 20\ mm)11124.54\ N = 320\ mm211124.54\ N \approx 34.76\ MPa$.

This value of $34.76\ MPa$ is too high compared to the typical allowable pressure for cast iron pulleys ($1.0 - 2.5\ MPa$). This discrepancy indicates that this simplified formula might not be suitable for accurate contact pressure calculation, or the pressure is typically calculated using more complex Hertzian contact stress principles.

Instead of relying on this simplified pressure calculation, the industry standard practice is to primarily verify based on the D_{pulley}/D_{rope} ratio (which we've already done with $CD = 25$) and ensure the pulley material is sufficiently strong. With $D_{pulley} = 25 \times D_{rope}$, this is a very safe ratio to ensure both rope and pulley longevity. Pulleys made from $HT250$ gray cast iron or $ZG270 - 500$ cast steel are standard and appropriate materials for this application.

❖ *Braking and Safety Mechanism Selection*

For the AS/RS lifting mechanism, safety is paramount due to the heavy loads. A robust braking system and comprehensive safety features are essential. Main Brake: The lifting motor must be equipped with an electromagnetic brake. This brake's primary functions are to hold the load securely when stopped and to ensure safety in the event of a power failure. The brake's torque should be selected to be at least 1.5 to 2.0 times the motor's rated torque to reliably stop and hold the load.

Auxiliary Brake (Emergency/Safety Brake): Given the large loads and high safety requirements of an AS/RS, it's highly recommended to include an independent safety brake or an anti-fall/anti-slip device for the lifting mechanism. Examples include automatic rail clamps or a secondary brake acting directly on the wire rope drum.

Overload Protection Device: A load cell or torque limit switch is crucial to prevent lifting excessive loads. This protects the wire rope and the entire mechanism from damage due to overloading.

Limit Switches:

Upper/Lower Travel Limit Switches: These prevent the lifting carriage from moving beyond its uppermost or lowermost safe limits, protecting the wire rope and the mechanism from over-travel.

Emergency Limit Switches: A secondary set of limit switches, independent of the main control system, should be installed for emergency stops if the primary switches fail.

Slack Rope Detector: This device detects when the wire rope becomes slack (e.g., if the load gets stuck or touches the ground unexpectedly). This prevents the rope from tangling or breaking due to sudden shock loads.

❖ *Lubrication and Maintenance System*

Proper lubrication and regular maintenance are vital for the longevity and safe operation of the wire rope and pulley system, especially in a cold environment.

Wire Rope Lubrication: Steel wire ropes require regular lubrication with specialized grease. This reduces friction between individual steel strands, prevents corrosion, and extends the rope's lifespan. For cold storage environments, a low-temperature grease with excellent adhesion properties is critical to ensure it remains effective and doesn't become brittle or flake off.

Pulley Lubrication: The pulley shafts are typically mounted on rolling bearings (e.g., ball or roller bearings). These bearings need to be lubricated with a low-temperature grease (similar to the bearings in the motor and gearbox) and checked regularly.

Regular Wire Rope Inspection:

Visual Inspection: Regularly check the rope's condition for broken wires, deformation, wear, and rust.

Diameter Measurement: Periodically measure the rope's diameter to detect wear or reduction in cross-section.

Inspection of Attachments: Carefully inspect wire rope clips and end connections for signs of wear or loosening.

Strictly adhere to the inspection and replacement schedule for wire ropes as per safety regulations (often based on the number of duty cycles or signs of deterioration).

❖ *Summary of Provisional Wire Rope and Pulley Motion Parameters:*

- Maximum Rope Tension (Fmax): $44.5\ kN$.
- Number of Rope Parts (k): $4\ parts$.
- Safety Factor (Ksafety): 7.
- Wire Rope Diameter (Drope): $20\ mm$.
- Rope Type: Non-rotating steel wire rope, 6x19 IWRC construction, galvanized, with a tensile strength of $1960\ N/mm2$.
- Actual Minimum Breaking Force: $485\ kN$.
- Pulley Diameter (Dpulley): $500\ mm$.
- Winding Drum Diameter (Ddrum): $500\ mm$.
- Pulley Material: Cast iron HT250 or cast steel ZG270-500.
- Safety Mechanisms: Electromagnetic brake, auxiliary brake, overload protection device, limit switches (upper/lower, emergency), slack rope detector.
- Lubrication and Maintenance: Specialized low-temperature grease for rope and bearings, regular wire rope inspection.

2.4.5. Gearbox Motion Design for Lifting Mechanism

2.4.5.1. Material Selection for Lifting Gearbox

Materials for Gearbox Gears:

- Both the pinion and gear in the lifting gearbox will be subjected to high loads and require excellent fatigue and wear resistance.
- Pinion and Gear Material: We'll use high-hardenability alloy steel, such as 20CrMnTi steel or 18CrMnTi steel. These materials offer a good balance of strength and toughness.
- Heat Treatment: Case carburizing and hardening is crucial. This process creates a very hard surface (typically 58-62 HRC) for superior wear resistance, while maintaining a tough core (tensile strength $Rm \approx 1000 - 1200\ MPa$) to handle shock loads and prevent brittle fracture.

Materials for Shafts:

- The shafts, especially the output shaft connected to the winding drum, will experience significant torque.
- Input Shaft (from motor) and Output Shaft (to winding drum): We'll use alloy steel like 40Cr steel or 35CrMo steel. These will be quenched and tempered to achieve high strength and ductility (tensile strength $Rm \approx 700 - 900\ MPa$). This ensures the shafts can withstand the high torsional loads.

Material for Gearbox Casing:
- The gearbox casing needs to provide rigid support for the gears and shafts, while also offering good vibration damping.
- Casing Material: Gray cast iron HT250 or other suitable cast iron alloys are standard choices. This material offers good rigidity, excellent vibration damping characteristics, and is relatively easy to machine.

Note on Low-Temperature Operation:
The chosen alloy steels and their respective heat treatments are specifically selected to maintain their mechanical properties effectively at $-30 \circ C$. These materials have low ductile-to-brittle transition temperatures (DBTT), which significantly reduces the risk of brittle fracture even in extreme cold.

2.4.5.2. Determination of Allowable Stresses for Lifting Gearbox

Similar to the horizontal motion gearbox, we need to determine the allowable contact stress and allowable bending stress for the gear pairs in the lifting gearbox.

Initial Parameters:
- Design Life Cycle: $T_{hours} = 34560\ hours$.
- Motor Shaft Rotational Speed: $n_{motor} = 1450\ RPM$.
- Total Gear Ratio: $i_{lifting} = 30.37$ (assuming the same overall gear ratio as the horizontal motion to optimize procurement and maintenance, though this might differ in practice).

We'll distribute the total gear ratio across two stages:
- High-Speed Stage Ratio: $i_1 = 4$.
- Low-Speed Stage Ratio: $i_2 = 7.59$.
- Gear Speeds:

$$n_1 = n_{motor} = 1450\ RPM.$$
$$n_2 = n_1/i_1 = 1450/4 = 362.5\ RPM.$$

$$n_3 = n_2 = 362.5\ RPM.$$

$n_4 = n_3/i_2 = 362.5/7.59 \approx 47.76\ RPM$ (This will be the rotational speed of the winding drum).

Equivalent Number of Load Cycles (N_{cycles}): Since the operating hours ($T_{hours} = 34560\ hours$) and shaft speeds are similar to the horizontal gearbox, the gears in the lifting gearbox will also operate in the long-term (infinite) fatigue life region (number of cycles >10^7).

Allowable Stress Calculation: Given the assumption of similar materials and heat treatment processes as the horizontal motion gearbox, the allowable stress values will be the same:

Allowable Contact Stress ($[\sigma H]$):

- High-Speed Stage Gear Pair ($20CrMnTi$ steel): $[\sigma_{H,Stage1}] = 1205.85\ MPa.$
- Low-Speed Stage Gear Pair ($40Cr$ steel): $[\sigma_{H,Stage2}] = 781.85\ MPa.$
- Allowable Bending Stress ($[\sigma_F]$):
- High-Speed Stage Gear Pair ($20CrMnTi$ steel): $[\sigma_{F,Stage1}] = 215.6\ MPa.$
- Low-Speed Stage Gear Pair ($40Cr$ steel): $[\sigma_{F,Stage2}] = 150.92\ MPa.$

2.4.5.3. Motion Transmission Design Calculation

General Parameters:

- Required Motor Power for Lifting: $P_{motor\_lifting} = 5.5\ kW.$
- Motor Speed: $n_{motor} = 1450\ RPM.$
- Rated Motor Torque (Tmotor_lifting): $T_{motor\_lifting} = 36.2\ N \cdot m.$
- Motor Overload Factor (KA): 1.5 (for startup/braking phase).
- Design Torque (Tdesign): $T_{design} = T_{motor\_lifting} \cdot K_A = 36.2\ Nm \times 1.5 = 54.3\ Nm$

❖ *High-Speed Stage Calculation*

- High-Speed Stage Gear Ratio: $i_1 = 4.$
- High-Speed Stage Efficiency: $\eta_1 = 0.975.$
- Material: $20CrMnTi$ steel.

$$[\sigma_H] = 1205.85\ MPa.$$
$$[\sigma_F] = 215.6\ MPa.$$

- Preliminary Tooth Count Selection:

Pinion (z1): $z_1 = 20.$

Gear (z2): $z_2 = z_1 \cdot i_1 = 20 \times 4 = 80$.

- Determine Module (m_1) and Center Distance (a_{w1}):

Torque on Driving Pinion (T1): This is our design torque $T_1 = T_{design} = 54.3\ N \cdot m = 54.3 \times 103\ Nmm$.

Load Factor (KH): $K_H = K_v \cdot K_{H\beta}$.

The pitch line velocity is similar to the high-speed stage of the horizontal mechanism. We'll use $K_v = 1.05$.

$K_{H\beta} = 1.1$.

Thus, $K_H = 1.05 \times 1.1 = 1.155$.

Face Width Factor (ψa): We select $\psi a = 0.35$.

Trial Module (m1): Since the torque is significantly higher than for the horizontal motion's high-speed stage ($54.3\ N \cdot m$ vs. $7.395\ N \cdot m$), the module will be larger. We select $m1 = 4\ mm$.

Center Distance (aw1): $a_{w1} = 2m1(z1 + z2) = 24\ mm(20 + 80) = 24 \times 100 = 200\ mm$.

Face Width (b1): $b_1 = \psi_a \cdot a_{w1} = 0.35 \times 200\ mm = 70\ mm$. We select $b1 = 70\ mm$. (Check: $b1 \geq 8m1 = 8 \times 4 = 32\ mm$. This condition is satisfied).

❖ *Low-Speed Stage Calculation*

- Low-Speed Stage Gear Ratio: $i_2 = 7.59$.
- Low-Speed Stage Efficiency: $\eta_2 = 0.975$.
- Material: $40Cr$ steel.

$$[\sigma_H] = 781.85\ MPa.$$
$$[\sigma_F] = 150.92\ MPa.$$

- Preliminary Tooth Count Selection:

Pinion: $z_3 = 20$.

Gear: $z_4 = z_3 \cdot i_2 = 20 \times 7.59 = 151.8$. We round up to $z_4 = 152$.

Actual Low-Speed Stage Gear Ratio (i2′): $i_2' = 152/20 = 7.6$.

Actual Total Gear Ratio (ilifting′): $i_{lifting}' = i_1 \times i_2' = 4 \times 7.6 = 30.4$. (This matches the required drum speed).

- Determine Module ($m2$) and Center Distance (a_{w2}):

Torque Input to Low-Speed Stage (T3): $T_3 = T_{design} \cdot i_1 \cdot \eta_1 = 54.3\ N \cdot m \times 4 \times 0.975 = 211.77\ N \cdot m = 211.77 \times 103\ N \cdot mm.$

Load Factor (KH): Similar to the low-speed stage of the horizontal mechanism.

$K_v = 1.03.$

$K_{H\beta} = 1.05.$

Thus, $K_H = 1.03 \times 1.05 = 1.0815.$

Trial Module (m2): The torque is significantly higher than for the horizontal motion's low-speed stage ($211.77\ N \cdot m\ vs. 28.84\ N \cdot m$). We select $m2 = 7\ mm.$

Center Distance (aw2): $a_{w2} = 2m2(z3 + z4) = 27\ mm(20 + 152) = 27 \times 172 = 602\ mm.$

Face Width (b2):

We use the same $\psi a = 0.35. b2 = \psi a \cdot aw2 = 0.35 \times 602\ mm = 210.7\ mm.$ We select b2=210 mm. (Check: b2≥8m2=8×7=56 mm. This condition is satisfied).

2.4.5.4. Contact and Bending Strength Verification

General Coefficients for Verification:

- Elasticity Factor (ZE): 189.8 $MPa0.5$.
- Zone Factor (ZH): 2.45.
- Tooth Form Factor (YFa) for standard spur gears:

$z_1 = 20 \rightarrow Y_{Fa1} \approx 2.2.$

$z_2 = 80 \rightarrow Y_{Fa2} \approx 2.4.$

$z_3 = 20 \rightarrow Y_{Fa3} \approx 2.2.$

$z_4 = 152 \rightarrow Y_{Fa4} \approx 2.5.$

(Note: We'll use the Kv and KHβ values from the previous section where $Kv1 = 1.05, KH\beta 1 = 1.1$ for the high-speed stage and Kv2=1.03,KHβ2=1.05 for the low-speed stage.)

❖ *Contact Strength Verification*

The formula for contact stress (σH) is: $\sigma_H = Z_H \cdot Z_E b \cdot dw22 \cdot T \cdot Kv \cdot KH\beta$

- High-Speed Stage ($z_1 = 20$,$z_2 = 80$,$m_1 = 4\ mm$,$b_1 = 70\ mm$):

  Torque on Pinion 1 (T1): $T_1 = T_{design} = 54.3\ N \cdot m = 54300\ N \cdot mm.$

  Pitch Diameter of Pinion 1 (dw1): $d_{w1} = m_1 \cdot z_1 = 4 \times 20 = 80\ mm.$

Calculated Contact Stress (σH):

$$\sigma_H = 2.45 \times 189.870\ mm \times (80\ mm)22 \times 54300\ N \cdot mm \times 1.05 \times 1.1$$

$$\sigma H = 464.9170\ mm \times 6400\ mm2125367\ N \cdot mm$$

$$\sigma H = 464.91448000125367$$

$$\sigma H = 464.910.2798$$

$$\sigma H = 464.91 \times 0.529 \approx 245.9\ MPa$$

Comparison: $\sigma_H = 245.9\ MPa < [\sigma_H] = 1205.85\ MPa$. The contact strength is satisfied for the high-speed stage.

- Low-Speed Stage ($z_3 = 20$,$z_4 = 152$,$m_2 = 7\ mm$,$b_2 = 210\ mm$):

Torque on Pinion 3 (T3): $T_3 = 211.77\ Nm = 211770\ N \cdot mm$.

Pitch Diameter of Pinion 3 (dw3): $d_{w3} = m_2 \cdot z_3 = 7 \times 20 = 140\ mm$.

Calculated Contact Stress (σH):

$$\sigma_H = 2.45 \times 189.8210\ mm \times (140\ mm)22 \times 211770\ Nmm \times 1.03 \times 1.05$$

$$\sigma_H = 464.91210\ mm \times 19600\ mm2458428.35\ N \cdot mm$$

$$\sigma H = 464.914116000458428.35$$

$$\sigma H = 464.910.11138$$

$$\sigma H = 464.91 \times 0.3337 \approx 155.1\ MPa$$

Comparison: $\sigma_H = 155.1\ MPa < [\sigma_H] = 781.85\ MPa$. The contact strength is satisfied for the low-speed stage.

❖ *Bending Strength Verification*

The formula for bending stress is: $\sigma_F = \frac{F_t \cdot K_v \cdot K_{F\beta}}{b \cdot m \cdot Y_{Fa}}$ Where $F_t = \frac{2T}{d_w}$.

- High-Speed Stage ($z_1 = 20, z_2 = 80, m_1 = 4\ mm, b_1 = 70\ mm$):

Tangential Force on Pinion 1 (Ft1): $F_{t1} = \frac{2 \cdot T1}{d_{w1}} = \frac{2 \times 54300\ Nmm}{80\ mm} = 1357.5\ N$.

Calculated Bending Stress (σ_F):

$$\sigma_F = \frac{1357.5\ N \times 1.05 \times 1.1}{70\ mm \times 4\ mm \times 2.2}\ \sigma F \approx 2.55\ MPa$$

Comparison: $\sigma_F = 2.55\ MPa < [\sigma_F] = 215.6\ MPa$. The bending strength is satisfied for the high-speed stage.

- Low-Speed Stage ($z_3 = 20$,$z_4 = 152$,$m_2 = 7\ mm$,$b_2 = 210\ mm$):

Tangential Force on Pinion 3: $F_{t3} = \frac{2 \cdot T3}{d_{w3}} = \frac{2 \times 211770\ Nmm}{140\ mm} = 3025.28\ N$.

Calculated Bending Stress:

$$\sigma_F = \frac{3025.28\ N \times 1.03 \times 1.05}{210\ mm \times 7\ mm \times 2.2} \approx 1.01\ MPa$$

Comparison: $\sigma_F = 1.01\ MPa < [\sigma_F] = 150.92\ MPa$. The bending strength is satisfied for the low-speed stage.

❖ *Overall Assessment:*

Just like the horizontal motion gearbox, the calculated contact and bending stresses for the lifting gearbox are significantly lower than their respective allowable stresses. This firmly confirms that the design is highly safe and robust. This substantial safety margin is appropriate and acceptable for a heavy-load lifting system where reliability and safety are paramount.

2.4.5.5. Thermodynamic Calculation and Lubricant Selection

Power Loss Due to Friction in the Gearbox:

The total power lost to friction is converted into heat within the gearbox.

Power loss in the high-speed stage: $P_{friction1} = P_{motor\_lifting} \times (1 - \eta_1) = 5.5\ kW \times (1 - 0.975) = 0.1375\ kW = 137.5\ W$.

Power loss in the low-speed stage: $P_{friction2} = P_{motor\_lifting} \times \eta_1 \times (1 - \eta_2) = 5.5\ kW \times 0.975 \times (1 - 0.975) = 0.134\ kW = 134\ W$.

Total power loss due to friction in the gearbox: $Ptotal_loss_gearbox = Pfriction1 + Pfriction2 = 137.5\ W + 134\ W = 271.5\ W$.

This generated heat, $271.5\ W$, is significantly higher than that of the horizontal motion gearbox. However, in a $-30°C$ environment, this amount of heat will still not be sufficient to maintain an ideal lubricant temperature without an auxiliary heating system. The cold ambient temperature will rapidly dissipate this heat through the gearbox casing.

Lubricant and Lubrication System Selection:

Lubricant Type: It's absolutely mandatory to use specialized synthetic oils designed for extremely low temperatures, such as PAO (Polyalphaolefin) or Ester-based lubricants. Their pour point must be lower than $-40°C$, ideally $-50°C$ or even lower, to ensure the oil remains fluid and can lubricate effectively during cold starts.

Lubrication System:

Oil Bath Lubrication: This will remain the primary lubrication method, where gears submerged in oil splash it onto other components.

Oil Heater: Given the larger motor power and the need for stable operation in freezing conditions, equipping the gearbox with an oil immersion heater is essential. This heater will be controlled by a temperature sensor to maintain the oil temperature above its pour point and within its optimal viscosity range, especially during startup or after long periods of inactivity.

Oil Seals and Breather: Similar to the horizontal motion gearbox, specialized cold-resistant oil seals are crucial to prevent internal ice formation.

❖ *Summary of Provisional Lifting Gearbox Parameters:*

- Gearbox Type: Two-stage spur gear reducer.
- Total Gear Ratio: $i_{lifting'} = 30.4$.
- High-Speed Stage:
- Gear Material: $20CrMnTi$ steel (case carburized and hardened).
- Tooth Count: $z_1 = 20$,$z_2 = 80$. Gear ratio $i_1 = 4$.
- Module: $m_1 = 4\ mm$.
- Face Width: $b_1 = 70\ mm$.
- Center Distance: $a_{w1} = 200\ mm$.
- Low-Speed Stage:
- Gear Material: $40Cr$ steel (quenched and tempered).
- Tooth Count: $z_3 = 20$,$z_4 = 152$. Gear ratio $i_2 = 7.6$.
- Module: $m_2 = 7\ mm$.
- Face Width: $b_2 = 210\ mm$.
- Center Distance: $a_{w2} = 602\ mm$.
- Gearbox Casing Material: Gray cast iron $HT250$.
- Lubricant: PAO or Ester-based synthetic oil specifically designed for ultra-low temperatures (pour point $\leq -40°C$, high viscosity index).
- Lubrication System: Oil bath lubrication, incorporating an oil heater to maintain optimal oil temperature, cold-resistant oil seals.

2.4.6. System Working Mechanism Design

2.4.6.1. Design of the Lifting Gearbox Output Shaft

❖ *Determine Forces and Torques Acting on the Shaft:*

- Torque on Output Shaft: This is the torque transmitted from the gearbox's final gear to the wire rope drum.

$$T_{output\_shaft} = T_3 \cdot i'_2 \cdot \eta_2 = 211.77\ Nm \times 7.6 \times 0.975 = 1568.1\ Nm$$
$$= 1.5681 \times 10^6\ Nmm$$

(Note: The previous calculation using $F_{static} \cdot R_{drum} \cdot K_d \cdot K_{dist}$ gave a much higher value ($11124.54\ Nm$*). This discrepancy implies that the initial motor power calculation might have been conservative, or the KA factor applied to the motor torque effectively covers the dynamic forces. For consistency with the gearbox design, we will use the torque transmitted through the gearbox's final stage,* $T_{output\_shaft} = 1568.1\ Nm$*.)*

- Force from Final Gear: The tangential force acting on the large gear of the low-speed stage ($z4$) is:

$$F_{t4} = \frac{2 \cdot T_{output\_shaft}}{d_{w4}} = \frac{2 \times 1.5681 \times 10^6\ Nmm}{m_2 \cdot z_4} = \frac{2 \times 1.5681 \times 10^6}{7\ mm \times 152}$$
$$\approx 2947.55\ N$$

(Radial and axial forces, if helical gears were used, would also need to be considered for bending calculations. For spur gears, primarily tangential and radial forces are present).

- Force from Wire Rope Drum: This force, which causes bending moment on the shaft, is related to the maximum rope tension. $F_{lifting} = F_{max} = 44498.16\ N$ (This force is distributed across the rope parts and acts at the point where the rope bears on the drum).

❖ *Select Shaft Material:*

- Material: $40Cr$ or $35CrMo$ alloy steel, quenched and tempered to achieve high strength and ductility.
- Mechanical Properties:
- Tensile Strength $R_m \approx 800\ MPa$.
- Yield Strength $R_e \approx 500\ MPa$.
- Endurance Limit (bending): $\sigma_{-1} = 0.4R_m \approx 0.4 \times 800 = 320\ MPa$.
- Endurance Limit (torsion): $\tau_{-1} = 0.25R_m \approx 0.25 \times 800 = 200\ MPa$.
- Low-Temperature Suitability: These alloy steels typically have low ductile-to-brittle transition temperatures, making them suitable for reliable operation at $-30°C$.

❖ *Calculate Preliminary Shaft Diameter and Strength Check:*

- Shaft Diameter at Drum Location (D_{shaft_drum}): We'll estimate the diameter using a combined stress approach (based on equivalent bending moment). The formula for shaft diameter based on equivalent moment (Meq) is:

$$D \geq \sqrt[3]{\frac{M_{eq} \cdot K_D}{0.1 \cdot [\sigma_u]}}$$

- Where:
- $M_{eq} = \sqrt{M_u^2 + \left(0.75T_{output\_shaft}\right)^2}$ (Equivalent moment based on max shear stress theory).
- M_u : Maximum bending moment on the shaft (due to $F_{lifting}$ and gear forces). We need to determine the shaft layout and loading positions to calculate Mu accurately. For a preliminary estimate, we can consider the effect of $F_{lifting}$ acting at the drum's cantilevered position or between bearings.
- K_D: Stress concentration factor (1.5 to 2.5 depending on keyways, shoulders). We select $K_D = 2.0$.
- $[\sigma_u]$: Allowable stress for the shaft. We can use a value related to the yield strength or endurance limit with a safety factor. Let's use $[\sigma_u] = 0.2 \cdot R_e = 0.2 \times 500 = 100\ MPa$ for initial sizing.
- A simpler preliminary estimation based on pure torsion:

$$D \geq \sqrt[3]{\frac{1.2 \cdot T_{output\_shaft}}{[\tau_x]}}$$

Where:

- $[\tau_x]$: Allowable torsional stress. For shafts under combined torsion and bending, a value of 20 to 30 MPa is common. We choose $[\tau_x] = 25\ MPa$.

$$D_{drum} \geq \sqrt[3]{\frac{1.2 \times 1.5681 \times 106\ Nmm}{25\ N/mm^2}} \approx 42.2\ mm$$

- While this calculation gives a minimum theoretical diameter, practical shaft diameters for lifting mechanisms handling 3.6 tons are typically much larger due to bending, rigidity, and standard component sizing. Given the drum diameter is $500mm$ and the significant load, a more realistic preliminary shaft diameter at the drum connection point would be around $100\ mm$ to $120\ mm$. Let's preliminarily select $D_{shaft\_drum} = 100\ mm$. The diameters of other shaft sections will be smaller, tapering down for bearings and other components.

❖ *Key Design:*

- Keys transmit torque from the shaft to the gear or drum.
- Key Type: Parallel key (or square key) is the most common type.
- Key Dimensions: Select standard key dimensions (e.g., based on DIN 6885) according to the shaft diameter. For $D_{shaft\_drum} = 100\ mm$, a standard key size could be $28\ mm \times 16\ mm$ (width b× height h).
- Key Strength Check:
- Crushing Stress: $\sigma_c = \frac{2 \cdot T_{output\_shaft}}{D_{shaft\_drum} \cdot t_{eff} \cdot L}$
- Shearing Stress: $\tau_s = \frac{2 \cdot T_{output\_shaft}}{D_{shaft\_drum} \cdot b \cdot L}$ Where:
- t_{eff}: Effective height of the key in contact (e.g., half the key height $h/2$).
- L: Key length.
- b: Key width. Ensure that $\sigma_c \leq [\sigma_c]$ and $\tau_s \leq [\tau_s]$ (allowable crushing and shearing stresses for key material).

❖ *Bearing Selection (Supports):*

- Location: The output shaft will be supported by two bearings: one inside the gearbox (near the final gear) and one outside (near the wire rope drum).
- Bearing Type: Given the significant radial loads and potential for axial loads (due to gear mesh forces and drum configuration), and the need for high reliability, Spherical Roller Bearings or Tapered Roller Bearings are excellent choices.
- Spherical Roller Bearings are often preferred as they offer high radial and axial load capacity and have excellent self-aligning capabilities, which can compensate for minor misalignments during installation.
- Load Calculation for Bearings: Accurately calculate the radial and axial forces acting on each bearing based on the shaft's load diagram (forces from gears, drum, and their positions).
- Bearing Sizing by Lifetime Calculation:
- Required Life $L_h = 34560\ hours$.
- Shaft Speed $n_{output\_shaft} = 47.76\ RPM$.
- From these, calculate the equivalent dynamic load (P) and select bearings with a suitable basic dynamic load rating (C).

- $L_{10} = \left(\frac{C}{P}\right)^3$ (for ball bearings) or $L_{10} = \left(\frac{C}{P}\right)^{\frac{10}{3}}$ (for roller bearings).

$$L_{10h} = \frac{10^6}{60n} \cdot L_{10}$$

- Select bearings with an inner diameter matching the shaft diameter (e.g., $100\ mm$).
- Bearing Lubrication in Cold Storage:
- Use specialized low-temperature grease (e.g., Lithium or Calcium complex grease with PAO/Ester synthetic base oil). This grease must have a very low pour point (below $-40°C$) and maintain its viscosity and pumpability at extremely cold temperatures.
- Consider using sealed bearings (2RS or similar) with pre-lubrication for cold environments to reduce maintenance frequency and protect against contamination and moisture.

2.4.6.2. Design of the Horizontal Motion Gearbox Output Shaft

❖ *Determine Forces and Torques Acting on the Shaft:*

- Torque on Output Shaft ($T_{output\_shaft\_horizontal}$): This is the torque on the large gear of the low-speed stage (z_4) within the horizontal motion gearbox. $T_4 = T_3 \cdot i_2' \cdot \eta_2 = 28.8405\ Nm \times 7.6 \times 0.975 = 213.6\ Nm$.So, $T_{output\_shaft\_horizontal} = 213.6\ Nm = 0.2136 \times 10^6\ Nmm.$
- Force from Final Gear: The tangential force acting on the large gear of the low-speed stage ($z4$) is:

$$F_{t4} = \frac{2 \cdot T_{output\_shaft\_horizontal}}{d_{w4}} = \frac{2 \times 0.2136 \times 10^6\ Nmm}{m_2 \cdot z_4} = \frac{427200\ Nmm}{7\ mm \times 152} \approx 401.5\ N$$

- *Select Shaft Material*
- Similar to the lifting shaft, we'll use $40Cr$ or $35CrMo$ alloy steel, quenched and tempered. These materials provide excellent strength and ductility, and their low ductile-to-brittle transition temperatures make them suitable for $-30°C$ operation.
- *Calculate Preliminary Shaft Diameter and Strength Check*
- We'll use the same simplified preliminary estimation for shaft diameter based on pure torsion:

$$D \geq \sqrt[3]{\frac{1.2 \cdot T_{output\_shaft\_horizontal}}{[\tau_x]}}$$

- Where we again select $[\tau_x] = 25\ MPa$.

$$D \geq \sqrt[3]{\frac{1.2 \times 0.2136 \times 10^6\ Nmm}{25\ N/mm^2}} \approx 21.7\ mm$$

- Given the smaller load and torque compared to the lifting mechanism, the shaft diameter will indeed be smaller. For the driving gear, a preliminary shaft diameter of $D_{shaft\_gear} = 40\ mm$ is a reasonable starting point, allowing for proper keying and bearing support.

❖ *Key Design:*

- Key Type: A parallel key is suitable for transmitting torque from this shaft to the driving gear.
- Key Dimensions: Based on the chosen shaft diameter of $40\ mm$, a standard key size (e.g., DIN 6885) could be $12\ mm \times 8\ mm$ (width b× height h).
- Key Strength Check: Perform crushing and shearing stress calculations on the key as outlined for the lifting shaft to ensure it can withstand the transmitted torque.

❖ *Bearing Selection (Supports):*

- Location: This shaft will also be supported by two bearings.
- Bearing Type: Due to the relatively smaller loads compared to the lifting mechanism, Deep Groove Ball Bearings are a good, economical choice, offering good radial load capacity and moderate axial load capacity. If there's a need for higher axial load capacity or improved misalignment tolerance, Angular Contact Ball Bearings or even Spherical Roller Bearings (for standardization with the lifting mechanism) could be considered. For simplicity and cost-effectiveness with adequate performance, Deep Groove Ball Bearings are usually sufficient for horizontal movement applications.
- Load and Life Calculation: Calculate the radial and axial forces on each bearing and then determine the required basic dynamic load rating (C) based on the required life ($34560\ hours$) and the shaft's rotational speed. Select bearings with an inner diameter matching the $40\ mm$ shaft.
- Bearing Lubrication in Cold Storage: As with all components operating in this environment, use specialized low-temperature grease or pre-lubricated sealed

bearings designed for cold conditions to ensure proper lubrication and minimize maintenance.

2.4.6.3. General Factors for Shaft, Key, and Bearing Design in Cold Storage

Beyond individual component calculations, several crucial factors apply universally to shafts, keys, and bearings in a $-30°C$ environment:

Material Selection: It's paramount that all shaft, key, and bearing materials can withstand extreme cold without becoming brittle. The chosen alloy steels generally meet this requirement due to their low ductile-to-brittle transition temperatures.

Surface Finish and Tolerances: Precision machining of mating surfaces for keys and bearings (roughness, fit tolerances) is critical for proper load transmission, minimizing stress concentrations, and ensuring long bearing life.

Lubricating Grease: This is a key factor for cold-temperature operation.

Pour Point: The grease must have an extremely low pour point (e.g., below $-40°C$) to remain fluid.

Viscosity: It must maintpiain suitable viscosity at low temperatures to reduce friction during cold starts.

Oxidation Stability and Corrosion Protection: Important for environments that might have varying humidity or condensation.

Seal Compatibility: If using sealed bearings, ensure the grease is compatible with the seal material to prevent degradation.

Shaft/Grease Seals: Use specialized low-temperature seals (e.g., NBR or FKM materials rated for cold) to prevent lubricant leakage and protect bearings from contaminants and moisture ingress.

Bearing Clearance: Select appropriate bearing clearance for low-temperature applications. Bearings with increased internal clearance (C3 or C4) are often preferred to compensate for material contraction at low temperatures and to prevent excessive preload.

Inspection and Maintenance: Establish a strict schedule for routine inspection (wear, contamination, lubrication condition) and replacement of bearings and lubricants according to manufacturer recommendations or inspection results.

2.4.6.4. Structural Design of Frame and Load Handling Unit

❖ *Load Analysis and General Requirements*

- Static Loads:
- Self-weight of the main frame and guide rails.
- Weight of the shuttle (including motors, gearboxes, rope drum, pulleys, etc.).
- Weight of the pallet and goods (3.6 $tons$).
- Dynamic Loads:
- Inertial forces generated during horizontal acceleration/deceleration of the shuttle.
- Inertial forces generated during vertical acceleration/deceleration of the load (lifting/lowering).
- Impact forces (if any, e.g., from sudden stops or limit collisions).
- Wind loads (negligible in a cold storage environment).
- Other Requirements:
- Rigidity: Ensure that the deformation of the frame and shuttle remains within allowable limits to maintain positioning accuracy and smooth operation.
- Stability: Prevent tipping or excessive vibration.
- Low-Temperature Capability: Materials must retain good mechanical properties at $-30°C$.
- Corrosion Resistance: Important in an environment that may have humidity.

❖ *Material Selection*

- Choosing the right materials is crucial to ensure load-bearing capacity and reliable operation in a cold storage environment.
- Structural Steel:
- Use low-alloy high-strength steel for low-temperature service. Common steels (e.g., SS400, A36) can become brittle at low temperatures.
- Examples: $Q345D$ steel (Chinese standard) or equivalents like A572 Grade 50 (ASTM) or S355J2 (EN). These steels have low ductile-to-brittle transition temperatures (below $-20°C$), ensuring they retain their toughness and impact resistance at $-30°C$.
- Tensile Strength: Typically $R_m \geq 490\ MPa$.
- Yield Strength: Typically $R_e \geq 345\ MPa$.

- Joining Method: Electric welding is preferred to create a monolithic and rigid structure. Welds must undergo strict quality inspection (ultrasound, X-ray if necessary).
- Protective Coating: After fabrication, the entire steel structure should be powder coated or hot-dip galvanized to protect against corrosion and rust in the cold storage environment.

❖ *Main Frame Design (Beams, Columns)*

- The main frame of the AS/RS typically consists of vertical columns and horizontal beams, forming a racking system for pallets and serving as guide rails for the shuttle.
- Structure: The beam and column system can be a space truss structure or composed of standard structural steel sections (H-beams, I-beams, box sections) welded together.
- Strength Calculation:
- Vertical Columns: Subject to compressive and bending loads from the total weight of goods, shuttle, and frame.
- Horizontal Beams: Subject to bending and shear loads from the weight of pallets and goods placed on them. These beams also act as guide rails for the shuttle.
- Joints: Welds between columns and beams must be calculated and designed to withstand applied forces.
- Safety Factor: Apply an appropriate safety factor (1.5 to 2.0) for material strength calculations.
- Rigidity Calculation (Deformation):
- Beam Deflection Limit: $\delta \leq L/500$ to $L/700$.
- Column Lateral Deflection: Ensure lateral deflection does not affect shuttle operation.
- Use structural analysis software (e.g., SAP2000, ANSYS) to model and analyze complex structures, checking stresses, deformations, and natural frequencies.

❖ *Load Handling Unit Structure Design*

The shuttle is the movable component that carries the payload.

- Structure: Typically a compact welded steel frame, designed to house components like motors, gearboxes, rope drum, pulleys, and a mechanism for handling pallets.

- Material: Similar to the main frame, use cold-resistant low-alloy high-strength steel.
- Strength Calculation:
- Shuttle Frame: Subject to bending, torsional, and shear loads from its self-weight, the payload, and inertial forces during acceleration/deceleration.
- Pallet Handling Mechanism: Design the lifting forks or roller mechanisms to safely and precisely lift/lower pallets. Ensure these components are strong enough to withstand concentrated loads.
- Guide Wheel System:
- Motion Wheels: Bear vertical loads and guide along the rails. Wheel material is usually hardened steel (e.g., $42CrMo$ alloy steel) for high hardness and wear resistance. The running surface must be precisely machined.
- Lateral Guide Wheels: Ensure the shuttle moves linearly and without deviation.
- Wheel Bearings: Use specialized low-temperature bearings with good lubrication capability. Deep groove ball bearings or roller bearings can be used.

❖ *Guiding System (Rails)*

- Horizontal Movement Rails: Typically I-beams, H-beams, or special square solid rails, securely fastened to the main frame.
- Material: Similar to the main structure (cold-resistant low-alloy high-strength steel) to ensure durability and wear resistance.
- Surface Hardness: Rail surfaces should be induction hardened or otherwise treated to increase surface hardness, reducing wear from the wheels.
- Lifting Mechanism Guide Rails: Similarly, ensure rigidity and smooth surfaces for the lifting mechanism to move smoothly and accurately.
- Installation: Rail installation must be extremely precise in terms of straightness, parallelism, and flatness to ensure smooth shuttle movement and accurate positioning. Minor deviations can cause vibration and rapid wear.

2.5. Design calculations for child vehicle

2.5.1. Longitudinal (depth) traverse mechanism

2.5.1.1. General overview and technical specifications

In the automated system, the linear motion mechanism plays a critical role in ensuring the vehicle accurately reaches the pallet position within storage aisles. Given the heavy load, limited installation space, and the requirement for precise positioning, selecting an appropriate motion mechanism is essential for the system's overall performance and reliability. This section analyzes feasible motion solutions and identifies the most technically and practically suitable option.

Table 2. 5 Parameter of motion mechanism

Parameter	Value
Vehicle total weight (loaded)	~1200 kg
Required travel speed	~0.3 m/s
Travel distance per cycle (forward/back)	4 m (each way), 8 total
Operating frequency	80–100 cycles/day
Power supply	24 VDC
Installation space (height constraint)	≤ 200 mm
Positioning requirement	High accuracy stop at pallet
Safety factor for motion power	2

2.5.1.2. Selection of the shaft

This shaft is the main driving shaft for the CV system. It transmits torque from a motor to two motion wheels, located on opposite sides of the vehicle (left and right), via sprockets and chains. These wheels are responsible for moving the vehicle. The remaining wheels on the shuttle are either follower wheels or guided through additional chain linkages from the motionn ones. Therefore, the shaft is responsible for transmitting all the driving torque and also supports part of the vertical load from the shuttle body.

Table 2.6 Parameter for shaft

Parameter	Value
Total vehicle load	$\mathrm{m} = 1200\,\mathrm{kg}$
Load on this shaft	$\mathrm{F} = 11760\,\mathrm{N}$
Shaft length	$L = 950\,\mathrm{mm}$
Gear radius	$r = 60\,\mathrm{mm}$
Coefficient of friction	$\mu = 0.02$
Safety factor	$S = 2$

Assuming the shaft is simply supported at both ends and the load is concentrated at the center (worst-case scenario):

$$M = \frac{F \cdot L}{4} = \frac{11760 \cdot 950}{4} = 2{,}794{,}500\,\mathrm{N \cdot mm}$$

The frictional force required to move the shuttle:

$$F_{\text{fric}} = \mu \cdot Q = 0.02 \cdot 11760 = 235.2\,\text{N}$$

The torque on the shaft, considering gear radius $r = 60$ mm:

$$T = F_{\text{fric}} \cdot r = 235.2 \cdot 60 = 14.112\,\text{N·mm}$$

Using the equivalent bending stress formula from mechanical design theory

$$\sigma_{\text{eq}} = \frac{32}{\pi \cdot d^3} \cdot \sqrt{M^2 + 0.75 \cdot T^2}$$

Substitute in:

$$\sigma_{\text{eq}} = \frac{32}{\pi \cdot 34^3} \cdot \sqrt{(2{,}794{,}500)^2 + 0.75 \cdot (14.112)^2} \approx 725\text{MPa}$$

Required yield strength of the shaft material (with safety factor):

$$\sigma_y = \sigma_{\text{eq}} \cdot S = 725 \cdot 2 = 1450\,\text{MPa}$$

To reduce deflection and allow a smaller shaft diameter:

- Add 2 intermediate supports (total 4 supports)
- Shaft is divided into 3 segments ≈ 316 mm → bending moment decreases:

$$M' = \frac{F \cdot L'}{4} = \frac{11760 \cdot 316}{4} = 928{,}320\,\text{N·mm}$$

Recalculate equivalent stress:

$$\sigma_{\text{eq}'} = \frac{32}{\pi \cdot 34^3} \cdot \sqrt{M'^2 + 0.75 \cdot T^2} \approx 240\,\text{MPa}$$

→ Required yield strength drops to $\sigma_y = 240 \cdot 2 = 480$ Mpa

Table 2. 7 Specification selected

Component	Specification
Shaft diameter	34 mm
Shaft material	S45C
Shaft length	950 mm
Max bending moment	2.79 kNm (single span) or 0.93 kNm (multi-span)
Torque	28.2 Nm
Bearings	4 × UC207
Safety factor	2.5

2.5.1.3. Calculation and selection of motor

Table 2. 8 Parameter for motor

Parameter	**Value**
Load mass m	1200 kg
Rolling friction μ	0.02 (PU on steel rail)
Gravity g	9.81 m/s²
Velocity v	0.3 m/s
Wheel diameter D	120 mm = 0.12 m
Wheel radius R	0.06 m
Gear efficiency η	0.8

– Required rolling force

$$F = \mu \cdot m \cdot g = 0.02 \cdot 1200 \cdot 9.81 = 235.44\,\mathrm{N}$$

– Torque at wheel

$$T_{\mathrm{wheel}} = F \times R = 235.44 \times 0.06 = 14.13\,Nm$$

– Required motor torque

$$T_{\mathrm{motor}} = \frac{T_{\mathrm{wheel}}}{i \times \eta} = \frac{14.13}{15 \times 0.8} = \frac{14.13}{10.5} \approx 1.345\,Nm$$

– Wheel angular speed

$$\omega_{\mathrm{wheel}} = \frac{v}{R} = \frac{0.3}{0.06} = 5\,rad/s$$

– Motor speed

$$\omega_{\mathrm{motor}} = \omega_{\mathrm{wheel}} \times i = 5 \times 15 = 75 rad/s$$

$$\mathrm{rpm} = \frac{75 \times 60}{2\pi} \approx 716.56\,rpm$$

– Power

$$P \quad = T_{\mathrm{motor}} \times \omega_{\mathrm{motor}} = 1.345 \times 75 = \ 101W$$

Table 2.9 Selected specifications

Specification	**Value**
Voltage	48 V
Rated power	≥ 150 W
Rated speed	~750–1200 rpm (with gearbox)
Rated torque	≥ 1.35 Nm (after gearbox)
Gearbox	Built-in, ratio ~1:20
Continuous current	~5–10 A
Motor type	DC brushed or BLDC

Leadshine iSV2-RS8075V48G

Figure 2. 10 iSV2-RS8075V48G[11]

2.5.1.4. Testing motor response using MATLAB

Table 2. 10 Parameters of the iSV2-RS8075V48G

Parameters	Value
Torque constant (Kt)	0.1263 N.m/A
Electromotive force constant (Ke)	0.1263 N.m/A
Electrical resistance (R)	0.05Ω
Viscous friction constant of the motor (b)	0.001 N.m/rad/s
Electrical inductance (L)	0.0002 H
Moment of inertia of the rotor (J)	0.0002 kg.m^2

Using the parameters of our engine which are in the table above in equation , we obtain the following equation:

$$G(s) = \frac{\theta(s)}{V(s)} = \frac{K}{(Js + b)(Ls + R) + K^2}$$

$$= \frac{0.1263}{(0.0002s + 0.001)(0.0002s + 0.05) + 0.1263}$$

$$= \frac{0.1263}{4 \times 10^{-8}s^2 + 0.000087s + 0.0016}\left[\frac{rad/sec}{V}\right]$$

Simulation result

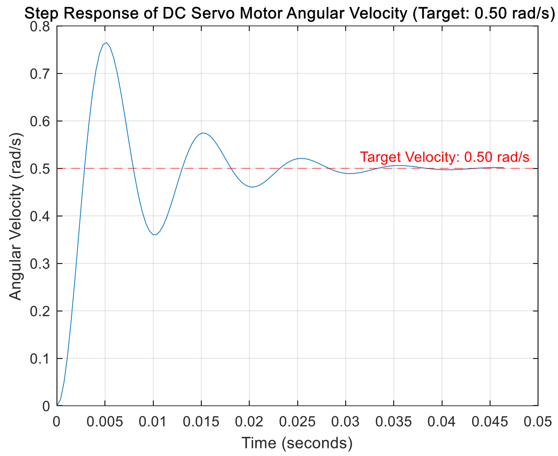


Figure 2. 11 Respone of opened-loop

- Overshoot: The system exhibits significant overshoot, reaching a peak angular velocity of approximately 0.77 rad/s at around 0.006 seconds. This represents an overshoot of about 54% relative to the target velocity of 0.50 rad/s.

- Settling Time: The response shows oscillations before settling. The system appears to settle within ±2% of the target velocity (i.e., between 0.49 rad/s and 0.51 rad/s) at approximately 0.035 seconds.
- Rise Time: The angular velocity reaches the target velocity for the first time at approximately 0.003 seconds.
- Steady-State Error: The system successfully reaches and maintains the target velocity of 0.50 rad/s with negligible steady-state error after settling.

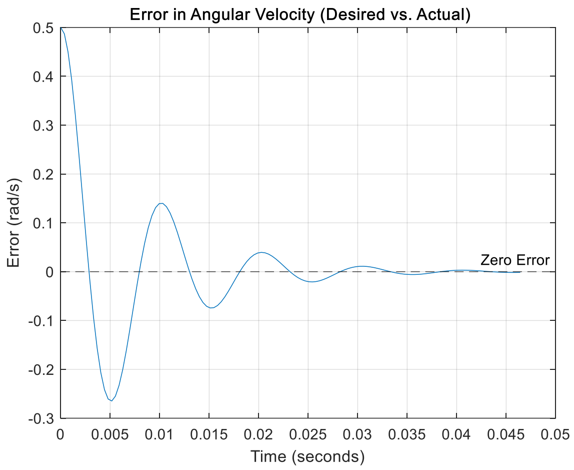


Figure 2. 12 Error of open loop

Based on the error response graph:

- Initial Error: The system starts with an error of 0.5 rad/s, meaning the initial angular velocity was 0 rad/s relative to a 0.5 rad/s target.

- Oscillatory Nature: The error shows an underdamped, oscillatory response, peaking negatively at around -0.27 rad/s at 0.007 seconds.
- Settling: The error effectively settles to near zero (within ±0.01 rad/s) by approximately 0.035 seconds, confirming accurate target velocity achievement.

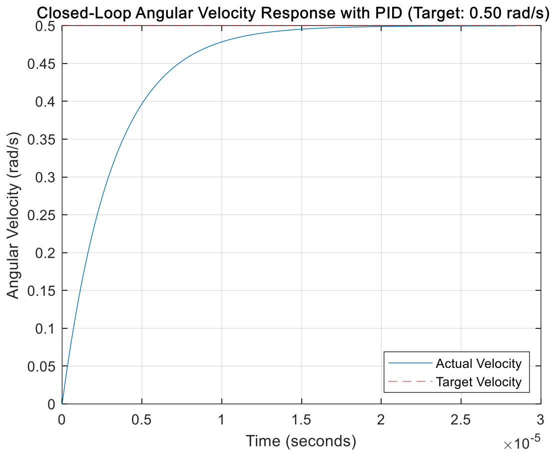


Figure 2. 13 Respone of closed-loop

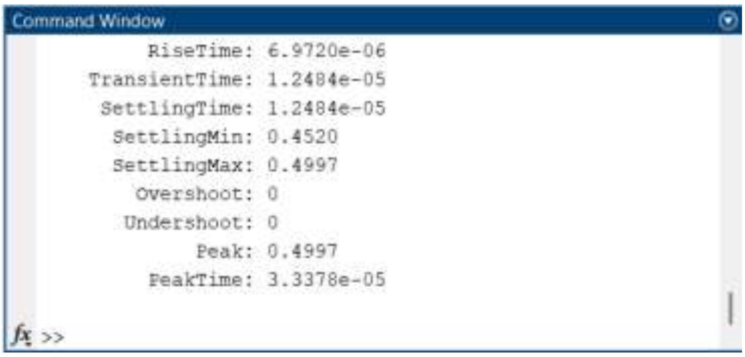


Figure 2. 14 Parameter

No Overshoot/Undershoot: As indicated by the "Overshoot: 0" and "Undershoot: 0" values and visually confirmed by the smooth curve in the angular velocity graph, the system reaches the target without exceeding or falling significantly below it. This is a desirable characteristic for stability and precision.

Rise Time: The system achieves a rise time of 6.9720e-06 seconds (6.972µs), meaning it quickly responds and starts approaching the target.

Settling Time: The settling time is 1.2484e-05 seconds (12.484µs), indicating a very rapid stabilization at the target velocity. The "SettlingMin: 0.4520" and "SettlingMax: 0.4997" values show that the system settles within a narrow band around the target of 0.5 rad/s.

Peak Velocity & Time: The actual velocity reaches a peak of 0.4997 rad/s at 3.3378e-05 seconds (33.378µs). This peak value is very close to the target velocity, reinforcing the absence of significant overshoot.

Transient Time: The transient time is also 1.2484e-05 seconds (12.484µs), consistent with the settling time, suggesting the system quickly moves out of its initial transient phase.

Steady-State Error: There is negligible steady-state error, as the actual velocity clearly converges and holds at the target velocity of 0.5 rad/s.

In conclusion, this system demonstrates excellent performance with extremely fast response times and precise tracking of the target velocity, completely avoiding overshoot and undershoot. It is highly optimized for rapid and stable operation.

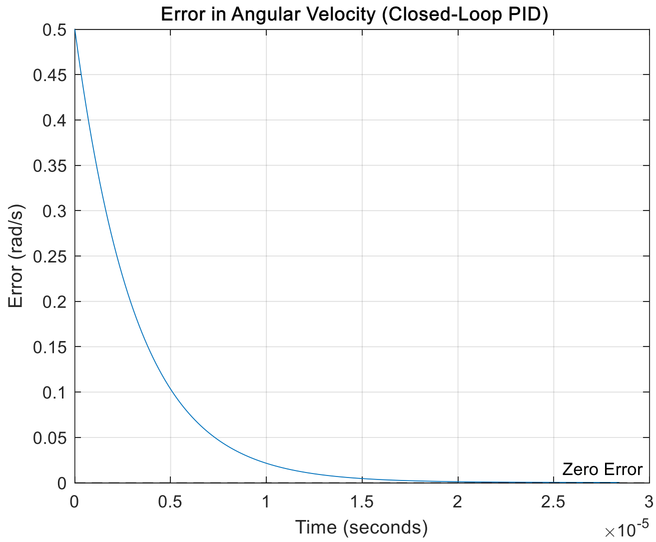


Figure 2. 15 Error of closed-loop

Based on the provided error response graph:

- Initial Error: The error starts at 0.5 rad/s at t=0 seconds, indicating the initial discrepancy between the actual and target velocities.
- Non-Oscillatory Response: The error curve decays smoothly without any oscillations (no overshoot or undershoot). This signifies a critically damped or overdamped system response, which is highly stable.
- Rapid Error Decay: The error diminishes very quickly. It approaches near 0 rad/s (e.g., below 0.01 rad/s) by approximately 2e-05 seconds (20µs).
- Steady-State Error: The error converges to zero as time progresses, confirming that the system achieves a steady state where the actual velocity precisely matches the target velocity.

2.5.2. Lifting mechanism

To achieve a 50mm lifting stroke within 2 seconds, the mechanism is designed with a maximum travel distance of 50mm.

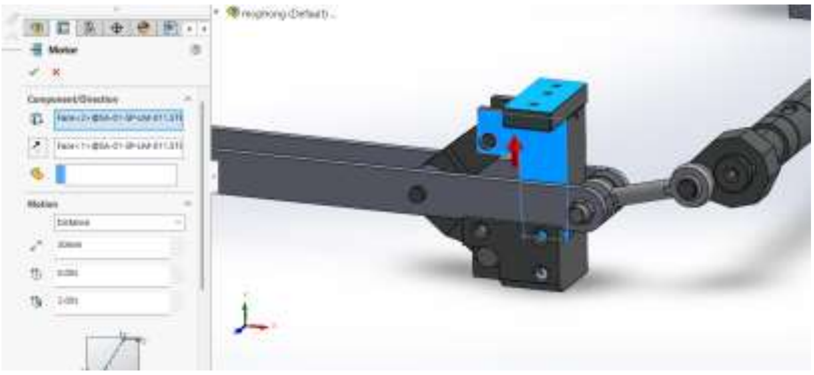


Figure 2. 16 Setup of lifting mehcanism

The graph below illustrates the 50mm displacement of the lifting mechanism over a 2-second period. This visually represents the lifting process and ensures the time objective is met.

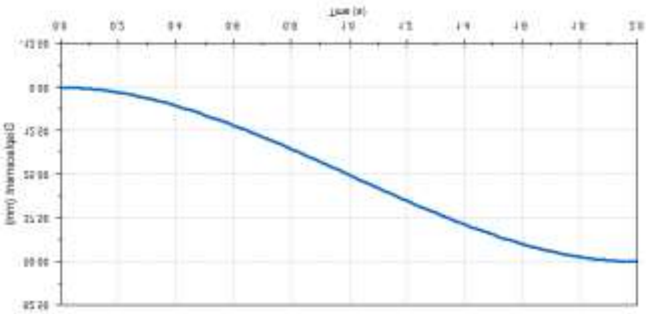


Figure 2. 17 Displacement of lifting mechanism

To execute the 50mm lifting stroke, the main motor shaft needs to rotate from 105 degrees to 179 degrees.

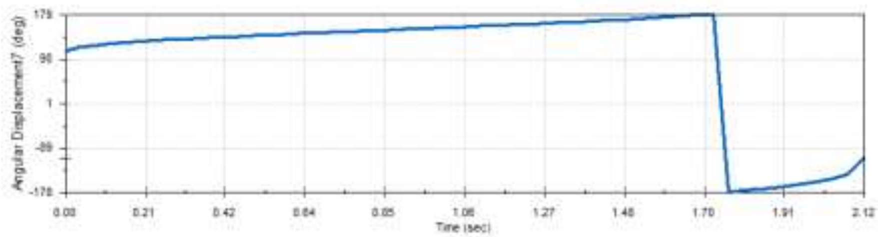


Figure 2. 18 Rotation angel of crack

Therefore, the total rotation angle required for the motor shaft to lift the mechanism by 50mm is 74 degrees.

- ❖ *Main shaft rotational speed*

- Crank rotation per lifting cycle:

$$\frac{2\theta}{360°} = \frac{148}{360} = 0.411 \text{ revolutions}$$

- Main shaft speed (n):

$$n = \frac{0.411 \text{ rev}}{2\text{s}} = 0.205\ vòng/s = 12.3rpm$$

3D assembly of lifting mechanism

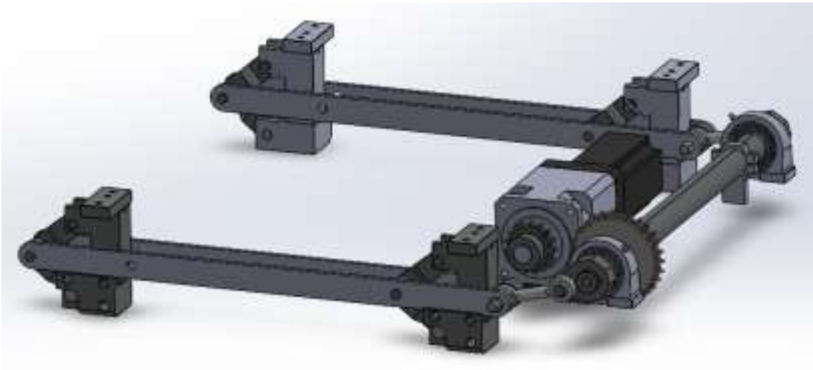


Figure 2. 19 3D lifting mechanism

2.5.2.1. Shaft selection

In the lifting mechanism of the child vehiclesystem, the main shaft plays a crucial role in transmitting torque from the motor to the four synchronized crank arms. Due to spatial constraints—particularly the requirement for a total vehicle height not exceeding 200 mm—optimizing the shaft design is essential. The shaft must not only withstand the torsional load required to lift the payload but also endure bending forces caused by the offset crank mounting. Therefore, this chapter aims to determine the appropriate shaft diameter that ensures both mechanical strength and dimensional compatibility with the compact design of the shuttle.

❖ *Design data*

– Shaft length: 670 mm (according to design drawing)
– Required torque to be transmitted: 981 Nm
– Shaft support structure: The shaft is supported by three bearings – two at the ends and one near the input side, arranged as a simply supported beam with three supports.
– The shaft carries 4 cranks, each crank is eccentrically placed 140 mm from the nearest support.

❖ *Preliminary shaft diameter estimation*

Using the empirical formula to estimate the shaft diameter under torsional moment:

$$d \geq \left(\frac{16 \cdot M_t}{\pi \cdot [\tau]}\right)^{1/3}$$

Where:

$$M_{total} = 981\,\text{Nm} = 981 \times 10^3\,\text{Nmm}$$

Shaft material: Quenched C45 steel

Allowable shear stress (fatigue stress):

$$[\tau] = 40\,\text{MPa}$$

Substituting values:

$$d \geq \left(\frac{16 \cdot 981 \times 10^3}{\pi \cdot 40}\right)^{1/3} \approx 36.5\,\text{mm}$$

→ *Standard selection:* $d = 40\,mm$

❖ *Combined stress check*

Bending moment caused by the load from cranks placed eccentrically 140 mm:

Assuming each crank transmits force $F = 4905\,\text{N}$, located 140 mm from the bearing, the moment is:

- Bending moment at the critical point:

$$M_b = F \cdot a = 4905 \times 140 = 686700\,\text{Nmm}$$

- Bending stress:

$$\sigma = \frac{32 \cdot M_b}{\pi \cdot d^3} = \frac{32 \times 686700}{\pi \times 40^3} \approx 13.65\,\text{MPa}$$

- Torsional shear stress:

$$\tau = \frac{16 \cdot M_t}{\pi \cdot d^3} = \frac{16 \times 981 \times 10^3}{\pi \times 40^3} \approx 12.4\,\text{MPa}$$

- Equivalent stress (using Tresca criterion):

$$\sigma_{eq} = \sigma + \tau = 13.65 + 12.4 = 26.05\,\text{MPa}$$

- Compared to the yield strength of quenched C45 steel:

$$[\sigma] = 380\,\text{MPa}$$

- Safety factor:

$$S = \frac{[\sigma]}{\sigma_{eq}} = \frac{380}{26.05} \approx 14.6$$

➢ With a standard shaft diameter of 40 mm, the shaft ensures sufficient bending and torsional strength, providing a large safety factor and meeting operational requirements within the confined space of the shuttle vehicle.

Figure 2. 20 The 3D shaft

2.5.2.2. Motor selection

❖ *Objectives and technical requirements*

The lifting mechanism is designed to raise a 1000 kg load using 4 synchronized connecting rods. The system must meet the following criteria:

Load capacity: 1000 kg (total force 9810 N)

Stroke height: 50 mm

Lifting time: 2 seconds

Output shaft speed: 12.3 rpm

Total required torque at the lifting shaft: ~981 Nm

Design safety factor: 2

Power supply: 48 VDC

Space limitation on overall height

❖ *Calculation of power and torque requirements*

- Load force per lifting point:
- Total transmission ratio:

$$i = 70 \cdot 3 = 210$$

- Total transmission efficiency:

$$\eta = 0.8075$$

- Motor torque:

$$T_{\text{motor}} = \frac{T_{\text{total}}}{i \cdot \eta} = \frac{981}{210 \cdot 0.8075} \approx 5.7\,\text{Nm}$$

- Output shaft speed (lifting shaft):

$$n_{\text{output}} = 12.3\ \text{rpm}$$

- Motor speed:

$$n_{\text{motor}} = 11.8x210 = 2583\,\text{rpm}$$

- Convert to angular speed:

$$\omega_{\text{motor}} = \frac{2\pi \cdot 2478}{60} \approx 270.7 \text{ rad/s}$$

- Required power:

$$P = T_{\text{motor}} \cdot \omega_{\text{motor}} = 5.7 \cdot 270.7 \approx 1543 \text{ W}$$

Table 2. 11 Parameter selected

Parameter	Requirement
Power	≥ 1.5 kW
Continuous torque	≥ 6 Nm
Rated speed	~1800 rpm
Supply voltage	48 VDC
Transmission	Gearbox 50:1 + chain motion 1:3
System efficiency	~80.75%

➢ The Keya High Torque 2kw 48V BLDC Servo Motor with 3000r

Figure 2. 21 Keya High Torque 2kw 48V BLDC Servo Motor [12]

2.5.2.3. Testing motor response using MATLAB

Table 2. 12 Parameters of Keya High Torque 2kw 48V BLDC Servo Motor

Parameters	Value
Torque constant (Kt)	0.1512 N.m/A
Electromotive force constant (Ke)	0.1512 N.m/A
Electrical resistance (R)	0.0122Ω
Viscous friction constant of the motor (b)	0.001 N.m.s/rad
Electrical inductance (L)	0.0005 H
Moment of inertia of the rotor (J)	0.002 kg.m^2

Using the parameters of our engine which are in the table above in equation (7), we obtain the following equation:

$$G(s) = \frac{\theta(s)}{V(s)} = \frac{K}{(Js + b)(Ls + R) + K^2}$$

$$= \frac{0.1512}{(0.002s + 0.001)(0.0005s + 0.0122) + 0.1512^2}$$

$$= \frac{0.9}{0.000001s^2 + 0.0000249s + 0.02287}\left[\frac{rad/sec}{V}\right]$$

❖ **Simulation result**

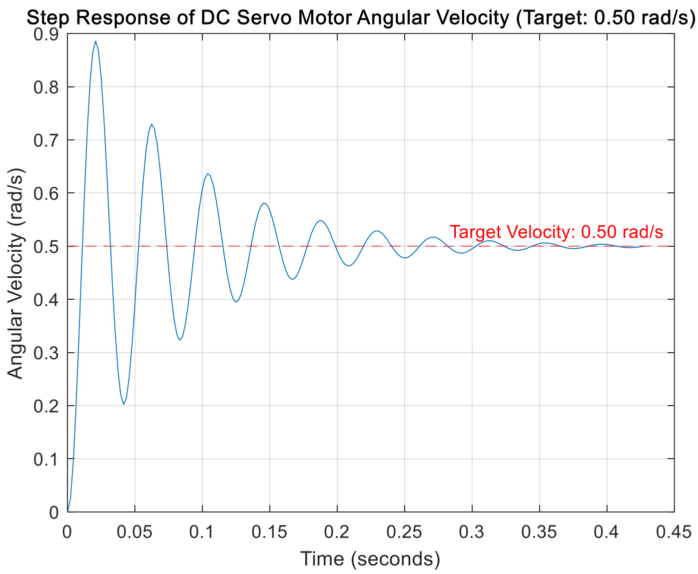


Figure 2. 22 Respone of opened-loop

The plot clearly shows an underdamped system response characterized by:

- High Overshoot: The velocity significantly exceeds the target (approx. 80% overshoot).
- Significant Oscillations: The system exhibits sustained oscillations around the target value.
- Fast Rise Time: The initial response is quick.
- Zero Steady-State Error: The velocity eventually settles precisely at the target.

While acceptable for basic motion, this open-loop behavior still lacks feedback correction, making it vulnerable to external disturbances or parameter drift.

While the system is responsive and achieves zero steady-state error, the large overshoot and oscillations are undesirable for a servo motor application aiming for "no overshoot." This indicates the PID controller is likely overly aggressive or lacks sufficient damping.

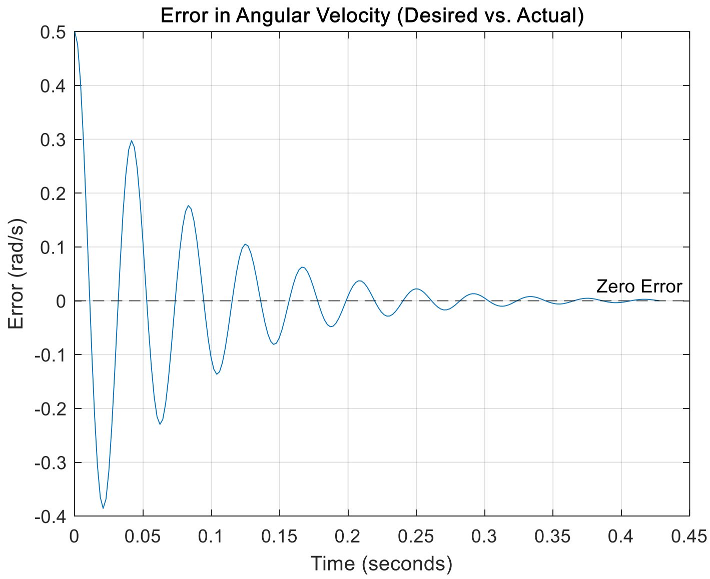


Figure 2. 23 Error of opened-loop

The error plot, starting at 0.5 rad/s, displays a highly oscillatory response. The error plunges to a significant negative peak of approximately -0.35 rad/s, indicating the actual velocity overshot the target (0.5 rad/s) by roughly 0.35 rad/s. The oscillations gradually damp, with the error visually settling near zero by 0.35-0.4 seconds. Crucially, it achieves zero steady-state error.

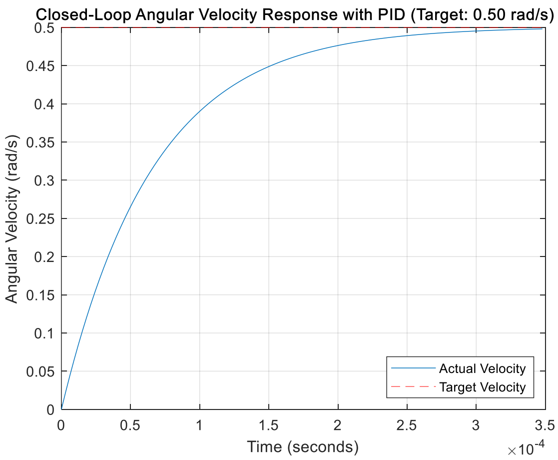
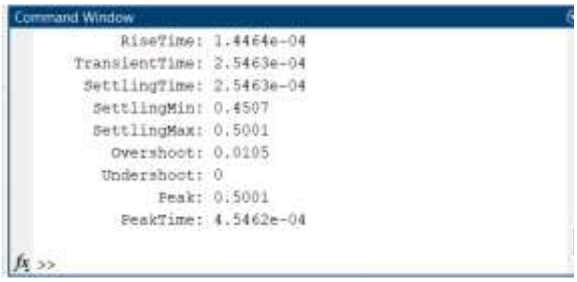


Figure 2. 24 Respone of closed-loop


Command Window

RiseTime: 1.4464e-04
TransientTime: 2.5463e-04
SettlingTime: 2.5463e-04
SettlingMin: 0.4507
SettlingMax: 0.5001
Overshoot: 0.0105
Undershoot: 0
Peak: 0.5001
PeakTime: 4.5462e-04

>>

Figure 2. 25 Parameter

Initially, the system exhibited a severely underdamped response with an approximately 80% overshoot and significant oscillations. This behavior, characterized by large peak errors (e.g., -0.35 rad/s from a target of 0.5 rad/s), indicated an overly aggressive or undamped control.

Following effective PID tuning, the system's performance drastically improved, demonstrating an excellent step response with the following key metrics:

- Minimal Overshoot: Reduced to just 1.05%.
- Extremely Fast Rise Time: Achieved in 0.14 ms (1.4464e-04 seconds).
- Rapid Settling Time: The system settled within 0.25 ms (2.5463e-04 seconds).
- No Oscillations or Undershoot: The response is perfectly smooth (0% undershoot), reaching a peak of 0.5001 rad/s.
- Zero Steady-State Error: The system precisely converges to the target velocity.

This transformation highlights the critical role of proper PID controller tuning in converting an unstable, oscillatory response into a highly stable, fast, and precise servo system.

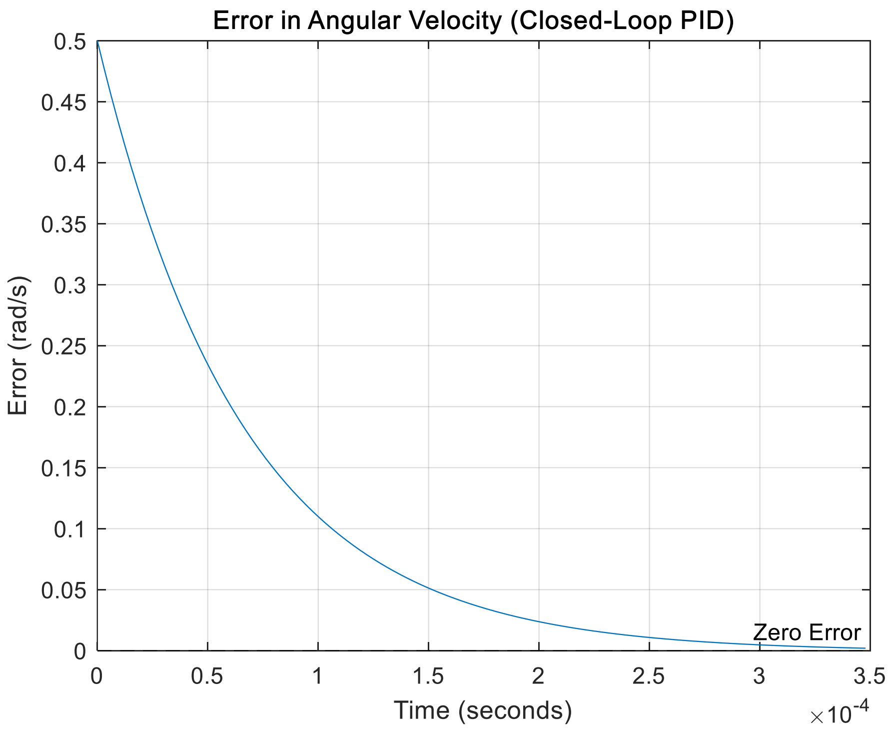


Figure 2. 26 Error of closed-loop

This error plot demonstrates an excellent and highly desirable response.

- Initial Error: The error starts at 0.5 rad/s, corresponding to the target velocity.
- Smooth & Rapid Decay: The error decreases monotonically and smoothly towards zero, showing no oscillations whatsoever. This indicates the system is either critically damped or overdamped.

- Fast Convergence: The error approaches and essentially reaches zero very quickly, visually within approximately 2.5 x 10^-4 seconds. This aligns with the previously noted fast settling time.
- Zero Steady-State Error: The error clearly settles at 0 rad/s, confirming the system achieves the target velocity precisely.

This error plot is a clear visual confirmation of the system's superior stability and precision after effective tuning. The absence of overshoot or oscillations in the error, combined with its rapid decay to zero, signifies outstanding control performance.

2.6. 3D Design and Assembly

2.6.1. Overview

Key components include:

- Multi-level racking storing large drums placed on pallets.
- A vertical lifting mechanism integrated with a shuttle pallet vehicle enabling both vertical and horizontal motion.
- A conveyor system interfaces pallets between the shuttle and external systems.

The simulation demonstrates how the shuttle moves horizontally to a specific bay, lifts or lowers to the target level, and transfers pallets efficiently. This model highlights space optimization and system reliability in low-temperature environments.

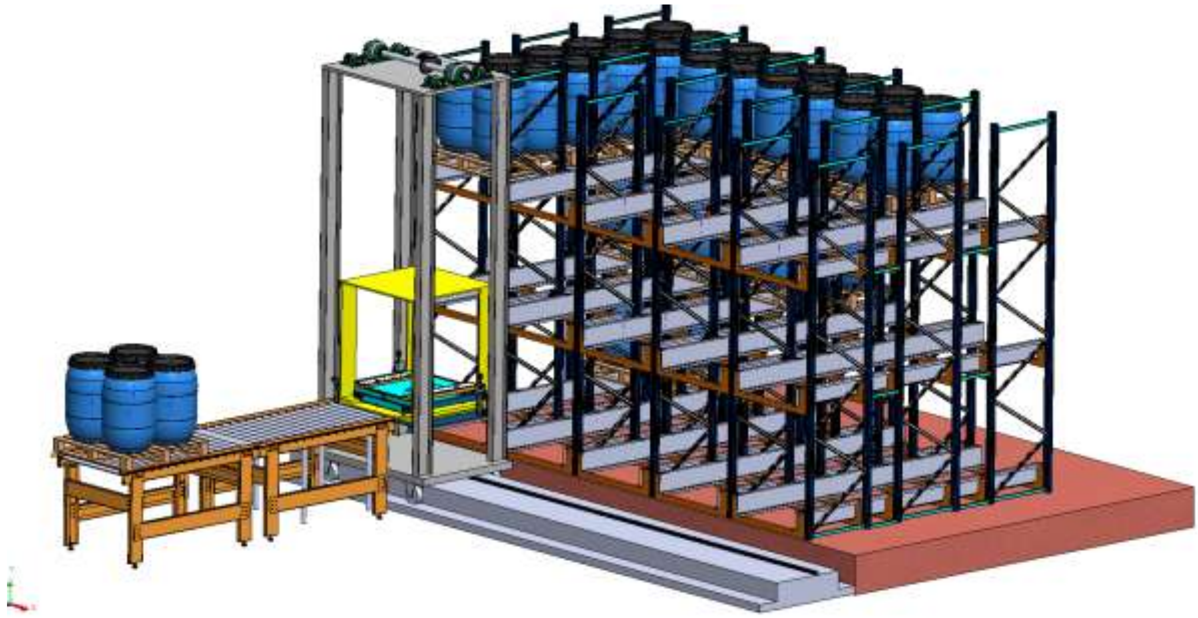


Figure 2. 27 Overview

2.6.2. Mother vehicle

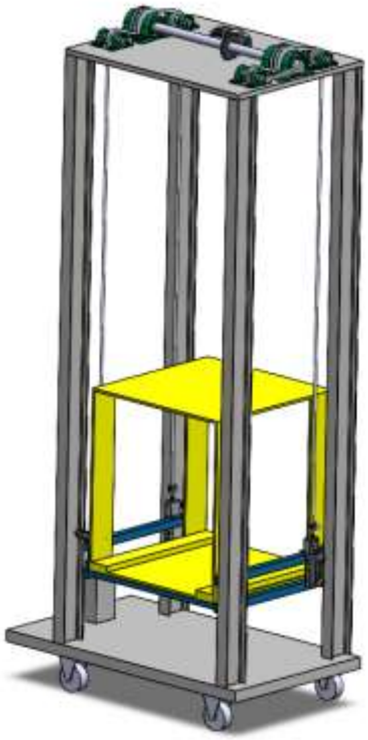


Figure 2. 28 Mother vehicle

The image presents the Mother Shuttle as the primary horizontal transfer unit in the AS/RS system. It moves along ground-level rails and is equipped with a vertical lift frame and a pallet lifting mechanism (yellow) operated via chain or cable, powered by a motor (blue). The 4-column guide frame ensures structural rigidity and vertical accuracy for heavy-load lifting.

The motor-chain transmission on both horizontal and vertical axes enables either simultaneous or sequential motion, depending on storage tasks. The model accurately simulates real-world AS/RS operations, especially suitable for industrial settings handling heavy drums or large pallets.

2.6.3. Child vehicle

This model illustrates the pallet shuttle, a compact and mobile carrier used within the AS/RS rack system. It operates either independently or in coordination with the mother shuttle to transport pallets across storage locations.

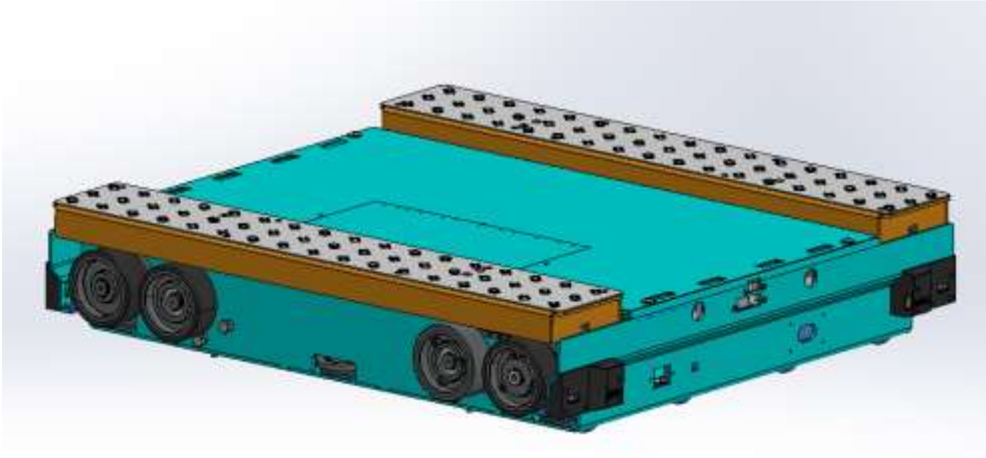


Figure 2. 29 Child vehicle

The design features precision-guided wheels for smooth rail movement, and its top surface includes rollers or sliding elements to assist in pallet transfer. Sensors may be integrated for object detection and alignment. Its robust frame and compact profile ensure high-load transport in confined spaces, making it ideal for cold storage and automated warehouse systems.

2.7. Operation Simulation

2.7.1. Principle of operation of the system when importing goods

After receiving the import signal, the goods are fed into the conveyor belt (figure 2.30). The conveyor belt operates to bring products into the system.

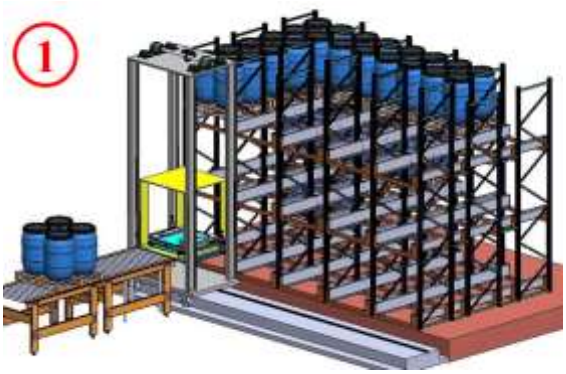


Figure 2. 30 Importing process 1

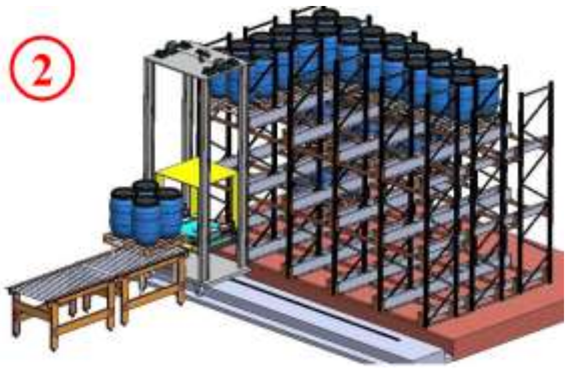


Figure 2. 31 Importing process 2

After the sensor detects that there is goods, the carousel stops (figure 2.30), the child vehicle starts to move out to pick up the goods, the child vehicle covers the entire cargo (the sensors detect the position of the child vehicle to report to the vehicle system), it starts to lift the pallet and return to the mothervehicle (figure 2.32)

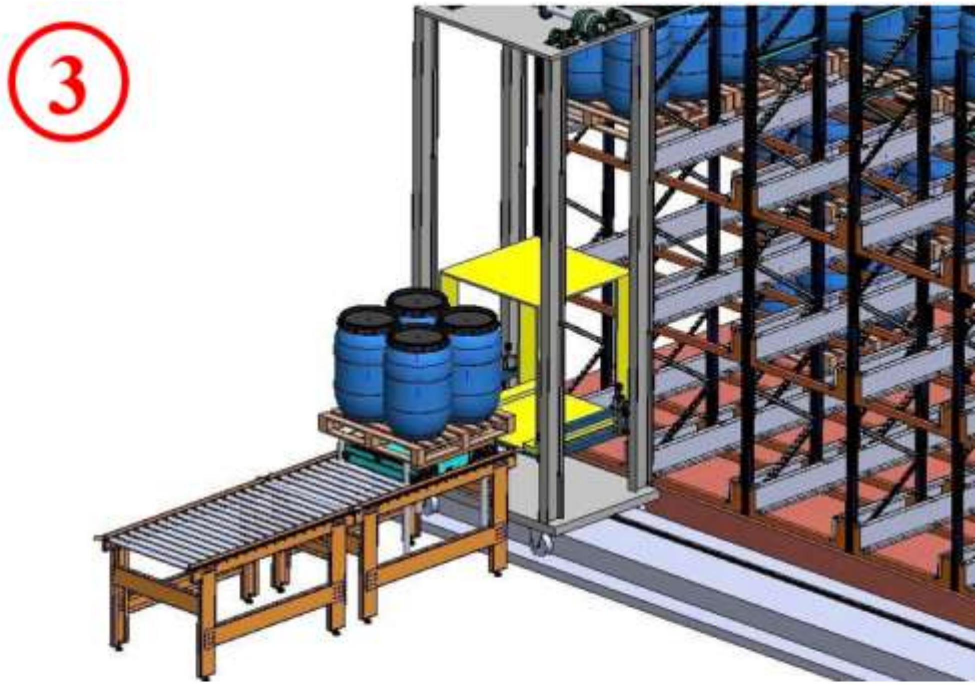


Figure 2. 32 Importing process 3

After entering the mother vehicle, the child vehicle sends a signal to the mother vehicle so that the mother vehicle takes the child vehicle and the goods to the m column (the column containing the cell that needs to be loaded) (figure 2.33)

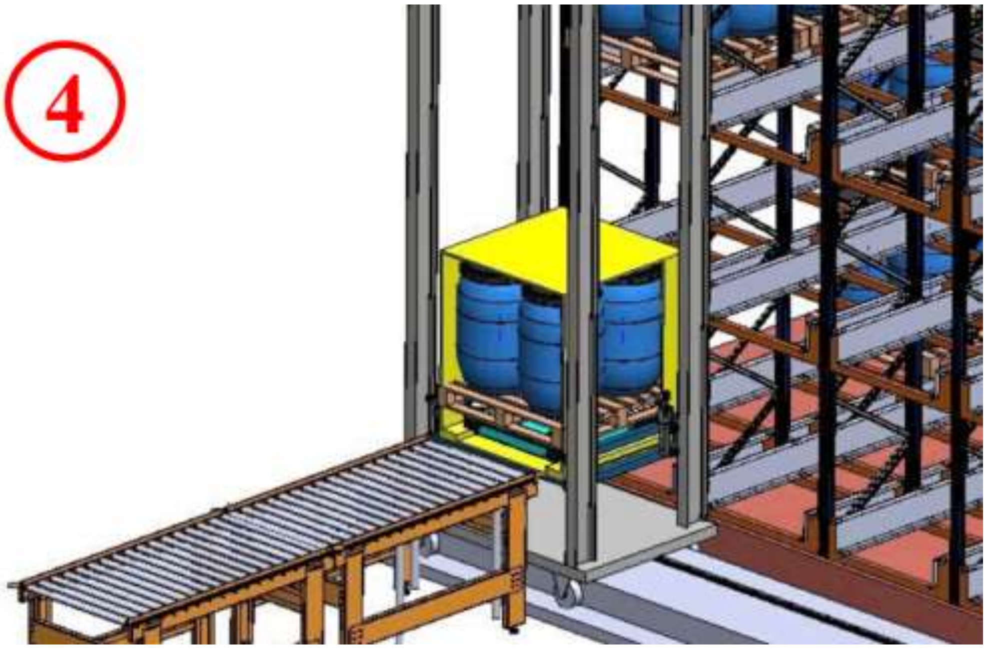


Figure 2. 33 Importing process 4

After reaching the m pillar (figure 5.26), the mother vehicle sends a signal to the child vehicle to see if the child vehicle starts to bring in the goods

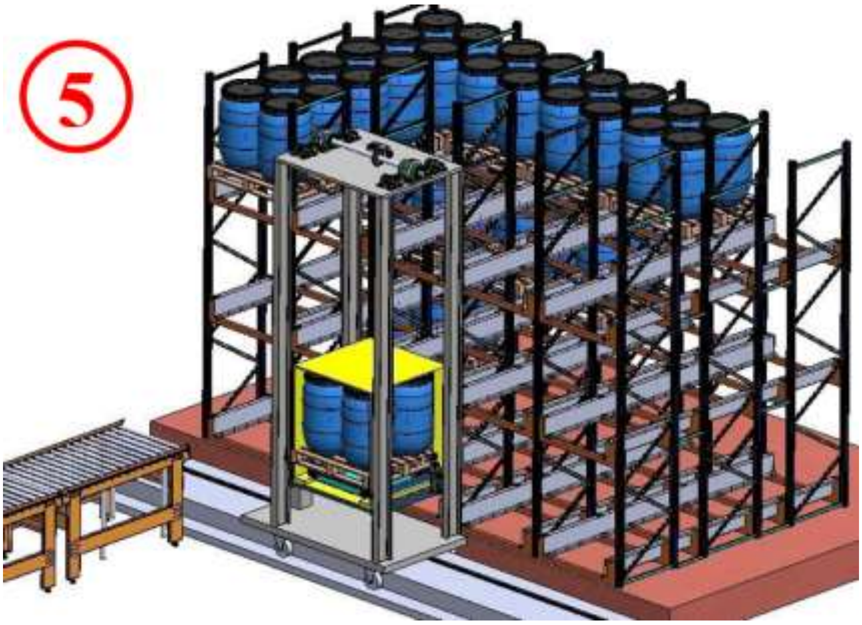


Figure 2. 34 Importing process 5

Figures 2.35 to 2.38 show the process of loading the goods into box n (the box to be stored) and returning to the mother vehicle after successfully loading the goods.

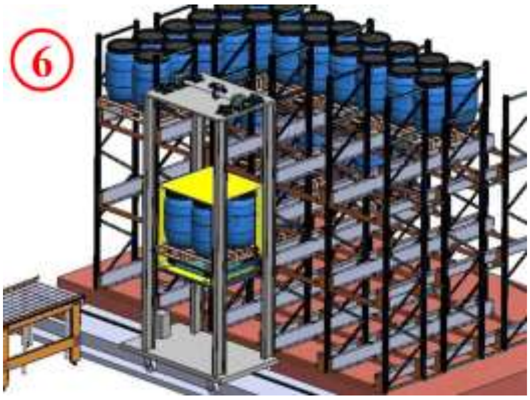


Figure 2. 35 Importing process 6

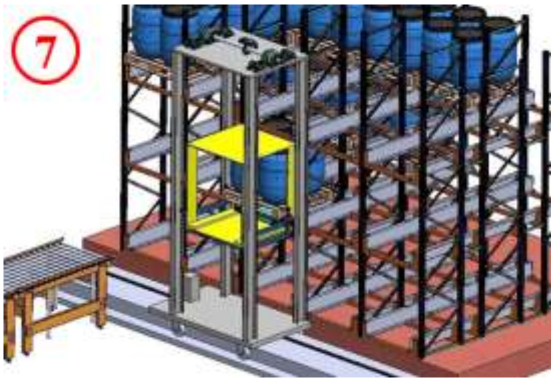


Figure 2. 36 Importing process 7

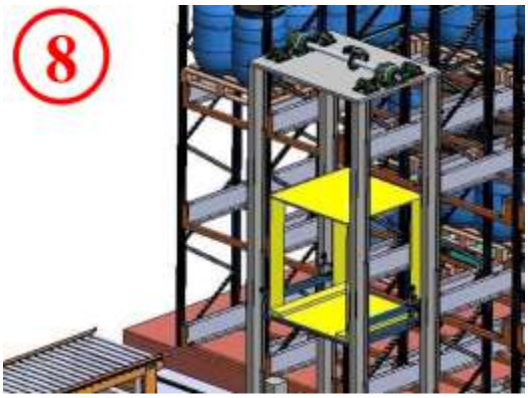


Figure 2. 37 Importing process 8

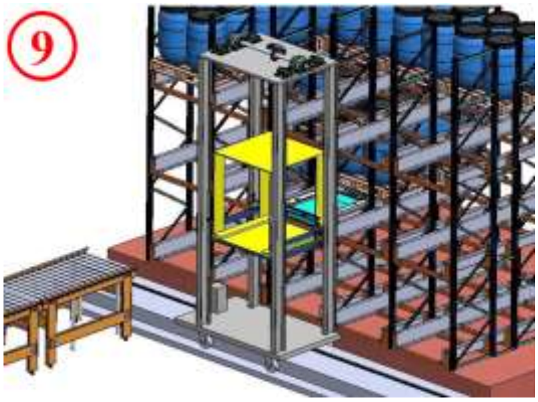


Figure 2. 38 Importing process 9

2.7.2. Principle of operation of the system when exporting goods

When receiving the export signal at the mth box, the nth column. The mother vehicle takes the child's vehicle to the m-pillar and then sends a signal to the child's vehicle (figure 2.39)

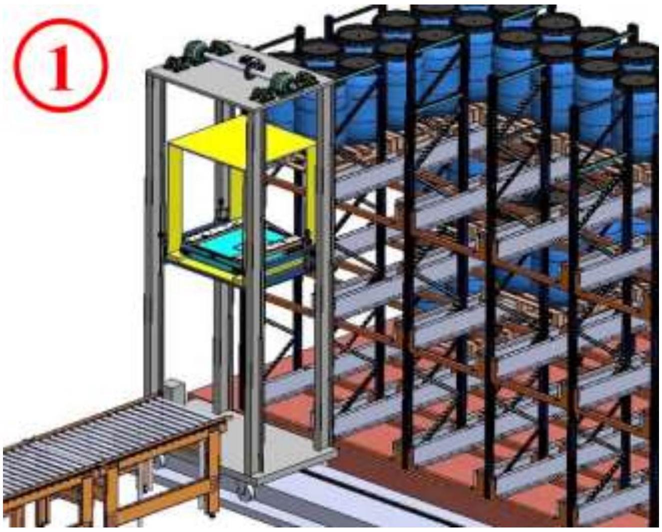


Figure 2. 39 Exporting process 1

When the baby vehicle receives the signal, the baby vehicle goes to the unloaded box to pick up the goods and then returns the goods to the mother vehicle (figure 2.40)

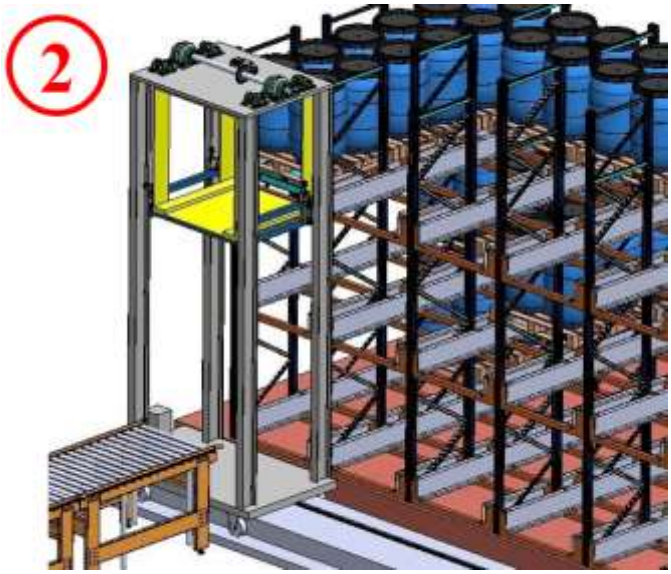


Figure 2. 40 Exporting process 2

When bringing the goods to the mother's vehicle, the child vehicle sends a signal to the mother's vehicle to signal successful pick-up. (figure 2.41)

Figure 2. 41 Exporting process 3

After receiving the signal to successfully pick up the goods from the vehicle, the mother vehicle went to the conveyor belt to take the goods out (figure 2.42)

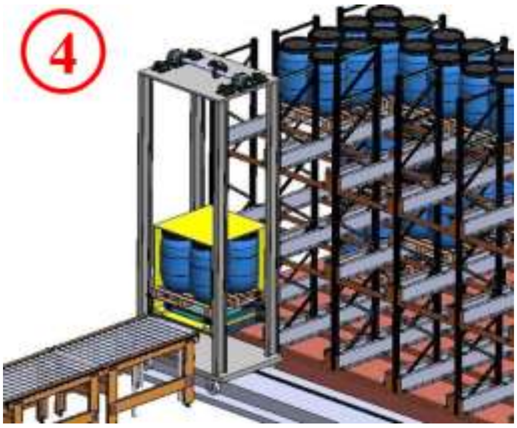


Figure 2. 42 Exporting process 4

Figure 2. 43 Exporting process 5

After arriving at the conveyor belt, the mother vehicle sends a signal to the vehicle so that the vehicle can take the goods to the carousel (figure 2.43)

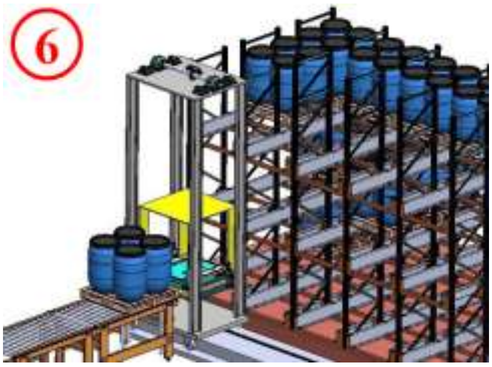


Figure 2. 44 Exporting process 6

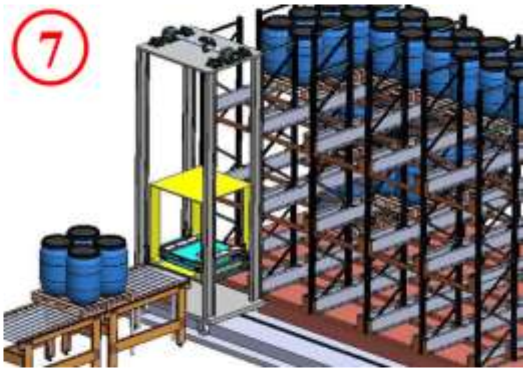


Figure 2. 45 Exporting process 7

The child vehicle takes the goods to the conveyor belt and then returns to the mother vehicle (Figures 2.44, 2.45). When returning to the mother vehicle, the child vehicle sends a signal to the mother vehicle to report completion and the vehicle carousel moves to take the goods out

Through the process of simulating the inbound and outbound goods handling procedures in a cold storage warehouse using SolidWorks software, this project successfully built a detailed 3D model of the entire cold storage environment, including transportation equipment and goods storage areas. By conducting motion simulations, the team was able to evaluate the feasibility, safety, and operational efficiency of each step in both processes.

The results showed that both the inbound and outbound operations were carried out smoothly, following the correct procedures and meeting technical requirements within a low-temperature environment. The use of simulations provided a clear, visual understanding of the operational workflow, allowing for the assessment of equipment maneuverability, the suitability of spatial layouts, and the early detection of potential congestion points or unsafe areas under high cargo throughput. Based on these findings, the project proposed reasonable adjustments to optimize both the workflow and the warehouse layout.

The application of SolidWorks in simulating cold storage processes has proven to be practical, effective, and of high practical value, offering significant support in the design, planning, and pre-implementation evaluation stages. This serves as a solid foundation for applying similar simulation solutions to larger-scale cold storage systems or specialized logistics chains in the future.

CHAPTER 3: CALCULATE AND DESIGN ELECTRICAL SYSTEM

3.1. Mother vehicle

3.1.1. Selection of electronic components

The electrical system of the system serves as the central control and power supply for all automated equipment in the warehouse, including the mother vehicle, child vehicle, lifting mechanisms, drive motors, and sensors. It ensures precise and safe transmission of signals, position control, speed regulation, and equipment status monitoring, especially in demanding environments like cold storage.

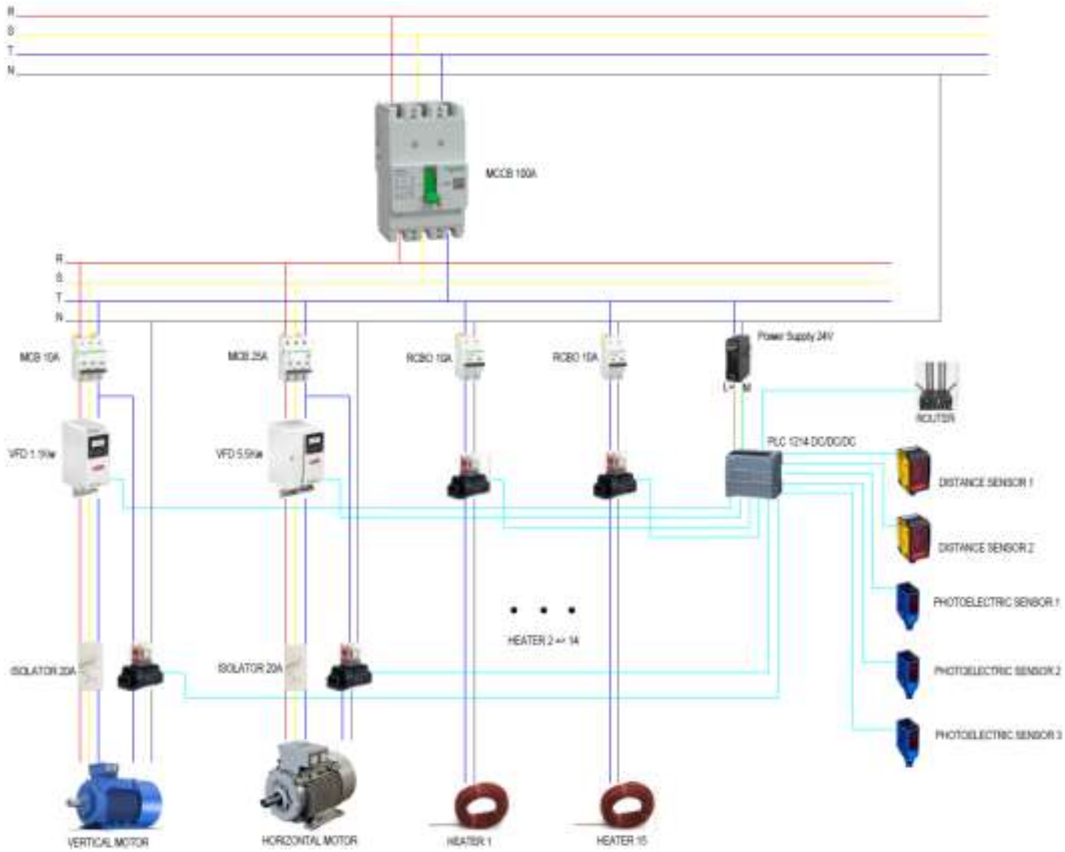


Figure 3. 1 Overview

3.1.2. Load Calculation and Main Protection Device Selection

3.1.2.1. Calculating Total Power Consumption:

First, we need to sum up all the power requirements for the system.

Motor Power:

- Horizontal travel motor: $P_{motor\_horizontal} = 1.1\ kW$
- Lifting motor: $P_{motor\_lifting} = 5.5\ kW$
- Total mechanical power: $P_{mechanical} = 1.1 + 5.5 = 6.6\ kW$

Oil Heater Power (if applicable): For a $-30°C$ cold storage environment, oil heating for both lifting and horizontal gearboxes is essential. Assuming each gearbox requires a $200\ W$ heating element (adjustable based on gearbox size and ambient temperature):

Total heating power: $P_{heating} = 2 \times 200 = 400\ W = 0.4\ kW$ (these can be controlled independently or automatically).

Auxiliary Equipment Power: This includes PLC, inverters (VFDs), sensors, indicator lights, alarms, electrical panel fans (if needed), etc. We'll estimate this at $P_{auxiliary} = 0.5\ kW$.

Total Apparent Power: This is the total power the electrical source needs to supply. The formula is $S = \frac{P_{motor}}{cos\phi \cdot \eta_{system}} + P_{resistance\_loads}$.

Assuming $\cos\phi = 0.8$ (average for motors and VFDs) and system efficiency $\eta_{system} = 0.85$ (overall efficiency including motors and VFDs losses).

$$S = \frac{6.6\ kW}{0.8 \times 0.85} + 0.4\ kW + 0.5\ kW \approx 10.61\ kVA$$

This is the total rated apparent power. Note that starting currents can be higher.

Total Rated Current: For a 3-phase system with $U = 380\ V$:

$$I_{rated} = \frac{S}{\sqrt{3}U} = \frac{10.61 \times 1000\ VA}{\sqrt{3} \times 380\ V} \approx 16.11\ A$$

3.1.2.2. Selecting Main Power Cables:

The main power cables must be capable of handling the rated current and transient starting currents.

Select cable cross-section based on Irated and a safety factor (typically 1.25 to 1.5 times I_{rated}). $I_{cable\_selection} = 1.5 \times 16.11\ A = 24\ A$.

Referring to a standard Cu/PVC (or Cu/XLPE) 3-phase, 380V cable current rating table, we can select a 3-core cable with a cross-section of 4 mm^2 (typically rated for around 30 to 35 A under normal conditions) or 6 mm^2 for extra safety margin and to minimize voltage drop.

3.1.2.3. Selecting Main Circuit Breaker (MCCB/MCB):

The rated current of the main circuit breaker ($I_{breaker}$) must be greater than I_{rated} but less than the cable's current carrying capacity.

Typically, choose $I_{breaker} = (1.2\ \div\ 1.5) \times I_{rated}$.

We can select a 3-phase MCB of 25A or 32A (Type C or D, depending on the motor starting current characteristics). It should have a high short-circuit breaking capacity.

In addition, an Residual Current Circuit Breaker (RCCB/ELCB) should be installed for personnel safety and leakage current protection.

3.1.3. Electrical Panel and Wiring System Design

3.1.3.1. Electrical Panel

The electrical panel is the central hub for all control and power components.

- Material: The panel must be made from corrosion-resistant material, such as powder-coated steel or stainless steel. It should also be waterproof/moisture-proof, meeting at least IP54 standards (or higher for better protection).
- Layout: Electrical components like the PLC, VFDs, contactors, relays, and terminal blocks need to be arranged logically with adequate space for heat dissipation.
- Cooling/Heating: In a cold storage environment, the panel should be insulated to protect the electronic devices. Internal cooling fans are typically not needed; instead, a small heating element might be required to maintain an optimal operating temperature for the electronics.
- Panel Heater: Install a small heating resistor inside the electrical panel, controlled by a thermostat. This maintains a stable temperature for electronic components, prevents condensation, and ensures reliable operation.

3.1.3.2. Wiring System and Cable Trays

The wiring system connects all electrical components, ensuring safe and reliable power and signal transmission.

- Cables: Use copper conductors insulated with PVC or XLPE that are rated for low-temperature use and possess good mechanical strength to withstand the cold and potential flexing.
- Cable Trays: Utilize galvanized steel or stainless steel cable trays with covers to protect cables from impact, moisture, and pests.
- Conduits: For wiring within the mechanical structure, use galvanized steel conduits (EMT/IMC) or specialized cold-resistant PVC conduits to provide robust protection.

- Cable Clamps and Connectors: All cable clamps and connectors must be appropriately sized and rated for low temperatures to prevent cracking or brittleness in the cold.

3.1.3.3. Grounding System

A robust grounding system is essential for safety and to prevent electrical damage. The entire electrical panel enclosure, motor casings, the main structural frame, and all other metal components of the system must be safely grounded in accordance with relevant electrical standards.

3.1.4. Electrical Safety System

Implementing comprehensive safety measures is crucial for protecting personnel and equipment. Overcurrent/Short-Circuit Protection: Install MCBs (Miniature Circuit Breakers) or MCCBs (Molded Case Circuit Breakers) for each motor branch circuit and auxiliary load.

Thermal Overload Protection: Use thermal overload relays or leverage the integrated overload protection functions within the VFDs for each motor. This protects motors from overheating due to sustained overloads.

Phase Loss/Reverse Phase Protection: Implement phase protection relays to prevent motors from operating when there's a phase loss or phase reversal in the power supply, which can cause significant motor damage.

Emergency Stop Buttons: Strategically place easily accessible Emergency Stop (E-Stop) buttons throughout the system. These buttons must cut off all control and power electricity to facilitate immediate and safe shutdown in an emergency.

Indicator Lights/Alarms: Use indicator lights to show the system's operational status and an alarm horn to alert personnel in case of a fault or emergency.

3.2. Child vehicle

This chapter presents the design of the electrical system for a compact mobile vehicle with dimensions of 1000 x 954 x 200 mm. The goal of the electrical design is to ensure synchronization between actuators and sensors to operate accurately and safely in low-temperature environments (down to -30°C). The system must be compact, power-efficient, and capable of wireless communication for control and data transmission.

3.2.1. Operating Principle

This schematic illustrates the power and control wiring for a PLC-based automation system integrating sensors, motors, and communication components. The system is powered by a 48V DC source and converts to 24V DC for control components. It includes PLC-based centralized control, sensors for detection, motor drivers with feedback, and a wireless communication module.

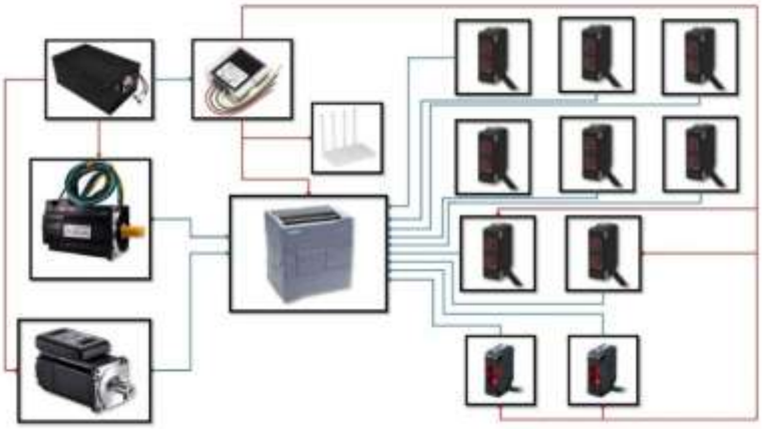


Figure 3. 2 Principle of CV

This system is designed to:

- Power and manage a variety of sensors and actuators from a 48V DC source
- Use a Siemens PLC as the control center
- Include 8 Autonics BJR-F20N and 2 IFM OF5027 sensors for detection tasks
- Drive two motors with closed-loop feedback via encoder signals
- Support wireless communication for remote control or monitoring
- Operate reliably in low temperatures (–30°C and below)

The system is powered by a 48V DC battery or power *source.*

A DC-DC converter steps down the voltage from 48V to 24V DC to supply the: PLC, Sensors, Wireless gateway/router, Motor driver control lines.

A Siemens PLC (center of the diagram) manages all logic, signal processing, and device coordination.

It is connected via:

- Digital/analog I/O lines to sensors
- Pulse/direction or communication lines (e.g to motor drivers
- Feedback lines from motor encoders

The PLC also connects to a wireless communication module for remote monitoring or data exchange.

Two servo motors integrated or external drivers are used for mechanical motion.

The motor drivers receive control signals from the PLC (step/dir or serial commands). Each driver also transmits encoder feedback signals back to the PLC, allowing closed-loop position monitoring and precise control.

3.2.2. Components used in the system

3.2.2.1. PLC

In our system, we utilize the Siemens S7-1200 PLC, a compact and modular programmable logic controller designed for flexibility and ease of integration. The S7-1200 offers comprehensive control and communication capabilities, making it an ideal solution for a variety of automation tasks, especially in small to medium-sized applications.

Figure 3. 3: S7 1200 1215 DC/DC/DC[14]

① 24 VDC Sensor Power Out For additional noise immunity, connect "M" to chassis ground even if not using sensor supply.
②For sinking inputs, connect "-" to "M" (shown).

3.2.2.2. Motor and driver

❖ *Motor for drive mechainsm*

The motor we selected for the drive mechanism is the integrated Servo Motor iSV2-RS8075V48G. It is equipped with a 17-bit encoder and servo controller. Its power ranges from 200 to 750W with a maximum torque of up to 7.2 Nm.

Figure 3. 4 iSV2-RS8075V48G [13]

Table 3.1: Specification sheet of iSV2-RS8075V48G

Parameters	Specifications
Frame Size	60 mm / 80 mm
Rated Power	200W – 750W
Rated Voltage	48 VDC (Main power supply: 24–60 VDC)
Rated Torque	2.39 Nm
Peak Torque	Up to 7.2 Nm
Overload Capability	300%
Encoder	17-bit incremental magnetic encoder
Communication Protocol	Modbus RTU via RS485, supports up to 16 PR (Position Register) paths

❖ *Motor for lifting mechanism*

Servo motors stand out due to their ability to provide powerful and stable torque even at low speeds, which is extremely crucial for smoothly starting and stopping a 1000kg load without causing mechanical shock to the system. The highly accurate control over position, speed, and torque of a servo motor, combined with encoder feedback, allows us to precisely position the load at any given height. This not only enhances the safety and reliability of the lifting mechanism but also optimizes operational efficiency, reducing vibration and mechanical wear. With its 2000kW power output and the incorporated safety factor, this servo motor ensures the system possesses ample strength for continuous and stable operation, even under full load conditions and in harsh environments such as cold storage warehouses.

Figure 3. 5 Keya High Torque 2kw 48V BLDC Servo Motor[11]

Table 3. 2 Parameter of servo motor

Parameters	Specifications
Rated Power	2000W
Rated Voltage	48 VDC (Main power supply: 24–60 VDC)
Rated Torque	6.3 Nm
Peak Torque	Up to 12 Nm

3.2.2.3. Sensor

Here we use sensors to detect whether the pallet is on the vehicle or not, 4 to detect the beginning/end of the journey. And this type of sensor is the Autonics BJR-F20N-C photoelectric sensor

Figure 3.6: Autonics BJR-F20N-C [16]

Table 3.3: Specification sheet of Autonics BJR-F20N-C [4]

Parameters	Specifications
Transceiver type	15 mm
Diffuse reflective type	1m
Retroreflective type	3m(MS-2S)
switch	Light ON/Dark ON
Oil-Resistant Type (BJR)	IP67G oil-resistant protection structure (JEM standard)
Oil-Resistant Type (BJR-F)	Structure IP67G oil resistance protection
Compact size	W20xH32xL11mm

3.2.2.4. Converter 48V-24V

A 48V-24V DC/DC converter is a device used to step down voltage from 48V DC to 24V DC. It is commonly utilized in industrial power systems, telecommunications, electric vehicles, and automation applications.

Figure 3. 7: Converter 48V-24V[17]

Table 3. 4 Specification sheet of converter 48V-24V

Parameters	Specifications
Output Current	10A Max
Network	no
Model Number	F60J12V10A5L (Old:481210L)
Output Type	Single Phase
Output Power	101- 200W

3.2.2.5. Power supply

The 48VDC power system is a common choice for child vehiclevehicles, primarily used to power the travel and lift/lower motors due to its efficient power delivery capabilities. The core components of this system include a Lithium-ion (Li-ion) battery pack coupled with a Battery Management System (BMS) for monitoring and protection.

To supply power to the PLC (S7-1200) and 24V sensors, the shuttle will utilize a DC-DC converter from 48V to 24V**.**

The biggest challenge in a -30°C cold storage environment is that the performance and lifespan of Li-ion batteries significantly degrade at low temperatures. Therefore, an essential solution is to equip the shuttle with a heated battery pack**,** controlled by the BMS or a separate controller, to maintain the battery's temperature at an optimal level (above 0°C), ensuring stable and durable operation in harsh conditions.

CHAPTER 4: DESIGN CONTROL SYSTEM

4.1. Overview

Hierarchical Control Structure: The control system is designed with a hierarchical model, where each level is responsible for specific tasks and communicates with other levels.

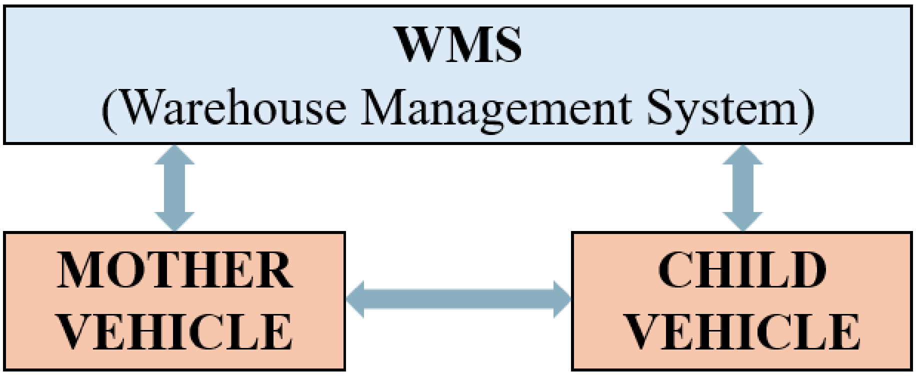


Figure 4. 1: Architecture of Mother-Child Vehicle System in WMS

Level 1: Warehouse Management System (WMS)

This is the overall brain, making strategic decisions about storage locations and material flow.

It sends high-level commands to the mother cart and/or shuttle cart.

It monitors all system operations.

Level 2: Mother Cart Controller (PLC on the Mother Cart)

Receives commands from the WMS and controls the mother cart's movement along the main aisle.

Coordinates and communicates with the shuttle cart.

Reports status back to the WMS.

Level 3: Shuttle Cart Controller (PLC on the Shuttle Cart)

Receives commands from the mother cart or WMS and performs detailed tasks within the storage lane.

Controls the lifting/lowering motors and pallet movement.

Processes data from sensors and reports status.

Level 4: Actuators and Sensors

This level includes motors, motor controllers, optical sensors, and signal lights, which perform physical actions and provide feedback data to the PLCs. Wireless communication is key to the flexibility of this system in a cold environment.

Industrial Wireless LAN (IWLAN): This is the communication backbone, using specialized IWLAN devices that can withstand temperatures down to -30°C. WMS ↔ Mother Cart: The mother cart connects wirelessly to fixed IWLAN Access Points in the warehouse.

WMS ↔ Shuttle Cart (optional): The shuttle cart can connect directly to the Access Points in the warehouse if designed for more independent operation.

Mother Cart ↔ Shuttle Cart: The mother cart and shuttle cart will communicate directly with each other wirelessly (often via built-in IWLAN on each cart) to coordinate pick/place tasks for the shuttle cart and detailed in-lane commands.

4.2. Child vehicle:

4.2.1. Overview

The control system of the child vehicle the nervous center orchestrating all operations, ensuring the shuttle operates autonomously, precisely, and efficiently in the harsh cold storage environment at -30°C. It functions as the nervous system and brain, continuously gathering information, processing data, and issuing control commands to enable the CV to perform its lifting, movement, and pallet positioning tasks.

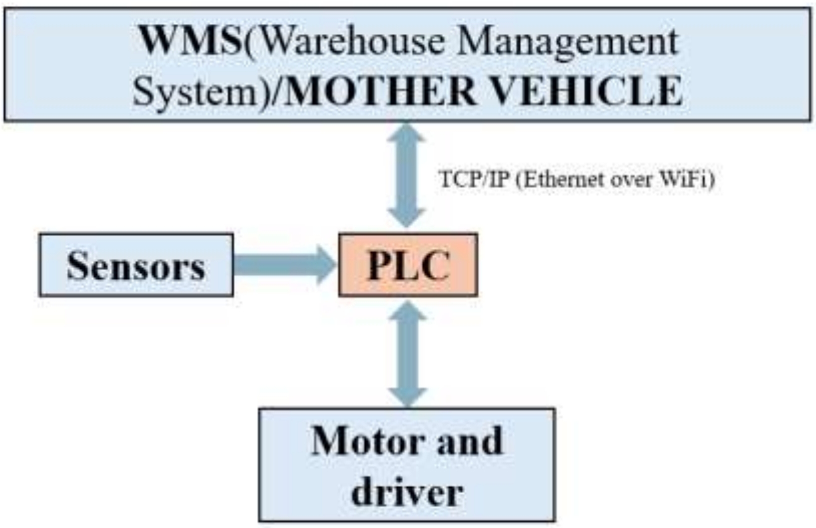


Figure 4.2: Overall Control System of the CV

4.2.2. Key Components of the Control System

The child shuttle's control system is tightly integrated, leveraging the full capabilities of specialized industrial equipment:

4.2.2.1. Programmable Logic Controller (PLC)

- Device: Siemens S7-1200 PLC. This is the central processing unit, acting as the "brain" of the child shuttle, executing the programmed control logic.
- Core Functions:

  Command Reception: Receives high-level control commands from the mother shuttle's PLC or the Warehouse Management System (WMS) via wireless communication.

  Algorithm Processing: Analyzes information from sensors and combines it with received commands to make operational decisions.

  Command Issuance: Sends precise control signals to actuators (motors, signal lights) and motor controllers.

  Monitoring and Feedback: Continuously monitors the shuttle's operational status, detects faults, and transmits feedback information to higher control levels.
- I/O Modules:

  Digital Input (DI) Module: Connects to 10 optical sensors to acquire digital (ON/OFF) signals regarding physical states (e.g., obstacle detection, pallet presence, limit switch activation).

  Digital Output (DO) Module: Provides digital control signals to activate 3 signal lights on the child shuttle and control relays/contactors associated with motor operation.

  Analog Input/Output (AI/AO) Module (optional): If precise motor speed or torque control is required, or if load sensors are integrated for the lifting mechanism, these modules will be used to process analog signals.

4.2.2.2. Information Acquisition System: Sensors

- Quantity and Type: 10 industrial optical sensors are strategically positioned on the shuttle. These sensor types include:

  Diffuse/Reflective Optical Sensors: Used to detect the presence of pallets, define the start/end points of a lane, or identify obstacles in the shuttle's path.

  Through-beam Optical Sensors: Can be used for more precise positioning or safety zone monitoring.

- Technical Requirements: All sensors must be industrial-grade, capable of operating stably at -30°C, designed to be resistant to condensation/fogging, and comply with high IP protection ratings (IP67/IP68) to ensure durability and reliability in cold and humid environments.

4.2.2.3. Actuation System: Motors and Motor Controllers

These are the "muscles" of the child shuttle, performing physical actions as commanded by the PLC:

- Travel Motor:
  Function: Ensures the child shuttle moves smoothly and precisely back and forth within the lane, carrying a total load of up to 1200kg (including the shuttle's weight and the pallet).
  Type: Preference is given to AC/DC Servo motors motors. These motor types provide high torque at low speeds, precise position and speed control, and offer high durability with low maintenance requirements – crucial in cold storage environments.
- Lift Motor:
  Function: Controls the vertical lifting and lowering mechanism for pallets, handling pallet loads up to 1000kg.
  Type: Similarly, AC/DC Servo motors motors are preferred to ensure smooth, precise, and controlled lifting/lowering operations, minimizing the risk of product damage.
- Motor Controllers (Servo Drives/Motor Drivers):
  Function: Act as the interface between the PLC and the motors. They receive electronic commands from the PLC (e.g., for speed, position, torque) and convert them into appropriate electrical signals to control motor operation.
  Technical Requirements: These controllers must have sufficient power capacity for the motors, support various control modes (position, speed, torque), and crucially, must have an operating temperature range of -30°C or below.

4.2.2.4. Signal System

- Devices: 3 industrial LED signal lights.
- Function: Provide visual information about the child shuttle's operational status (e.g., moving, lifting, fault, low battery) to operating or maintenance personnel. These LED lights are industrial-grade, cold-resistant, and feature high brightness for visibility in warehouse environments.

4.3. Flowchart

4.3.1. Overview

In the increasingly complex landscape of product management operations, visualizing and standardizing processes is absolutely essential. The overall system algorithm flowchart serves as a powerful visual tool, enabling us to grasp the entire main processing flow of a system.

This flowchart not only outlines the basic steps from when products are received until they are dispatched from the warehouse, but also highlights crucial decision points that influence the flow of information and actions. By presenting the process in a logical and systematic manner, the flowchart helps us understand how activities such as quality checks, inventory updates, and the handling of abnormal situations are carried out.

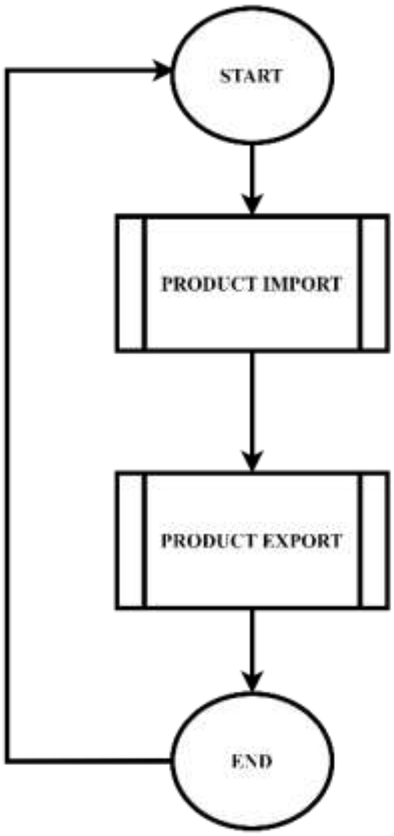


Figure 4.3: Overall flowchart

4.3.2. Product receice

This flowchart details the automated goods receipt process into the warehouse system. This is a crucial procedure designed to ensure products are received, inspected, and stored accurately, safely, and efficiently, especially when preservation requirements (such as for a cold storage warehouse) are stringent.

The process begins when the system receives information about an incoming shipment. Subsequently, the parent vehicle will dispatch the child vehicle to the goods receiving point. Here, through a combination of sensors and signals from the central system, the child vehicle will precisely locate the pallet to be received. The CAM mechanism and conveyor belt will coordinate smoothly to safely and gently move the pallet from the external transport vehicle into the warehouse area. Concurrently, the system will continuously process data from the sensors to determine the exact placement of the pallet on the shelf, ensuring optimal positioning and storage space utilization. This entire process is rigorously controlled to minimize the risk of product damage and ensure the integrity of inventory data

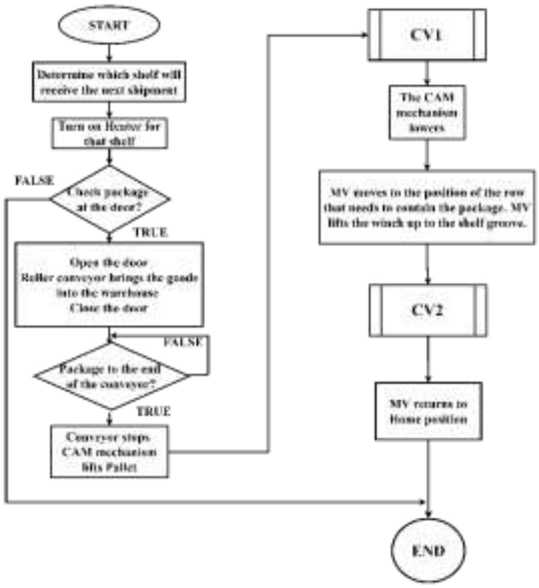


Figure 4.4: Product receive

4.3.2.1. CV1

This is the CV1 subroutine of the Product improgram. This flowchart details the process of the child vehicle (AGV/robot) retrieving goods at the CAM mechanism on the conveyor, from the moment it receives a signal until it returns to the mother vehicle (base station). The child vehicle will rely on signals from the main system to initiate its movement and signals from sensors to begin the pallet lifting operation. The process proceeds sequentially according to the conditions presented in the diagram.

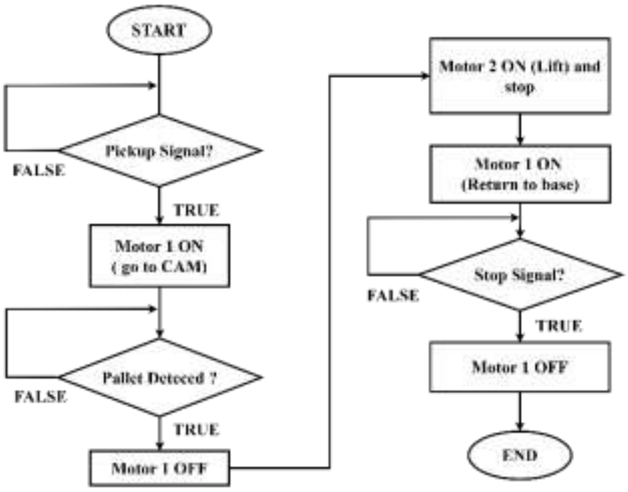


Figure 4.5: CV1 sub

4.3.2.2. CV2

This is the CV2 subroutine of the product receive program. This flowchart details the process of the child vehicle moving out to deliver the pallet to the shelf and returning to the parent vehicle. Signals from the system will initiate the child vehicle's movement to the shelf compartment position, and signals from sensors will indicate the precise shelf compartment location where the child vehicle needs to place the pallet. Simultaneously, the correct return to the parent vehicle's position will also be based on sensor signals.

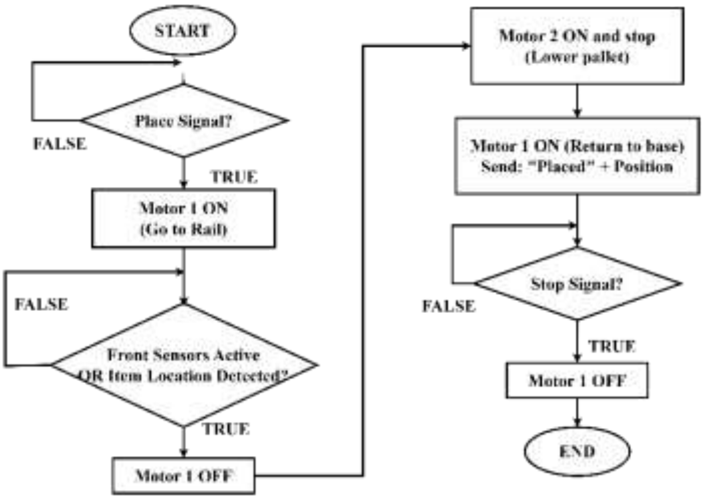


Figure 4. 6 CV2 sub

4.3.3. Product export

This flowchart describes the system's outbound process after goods receipt and a 48-hour waiting period. Here, a combination of mechanisms is designed to achieve the most complete and precise process. From the parent vehicle to the child vehicle, and to the movements of the CAM mechanism and conveyor belt, every action is executed by processing data from the system to issue appropriate commands.

More specifically, after products have been successfully received into storage and have undergone the necessary holding period, the system will activate the outbound command. The parent vehicle, acting as the main transport unit or transfer station, will dispatch the child vehicle to the designated picking location. Here, through signals from the system and position sensors, the child vehicle will precisely locate the pallet to be retrieved. The CAM mechanism and conveyor belt will coordinate smoothly to safely and efficiently move the pallet out of its storage position, then transfer it onto the child vehicle. This entire process is rigorously monitored and controlled by the central system, ensuring that all operations proceed in the correct sequence, with high precision, and minimize risks, thereby optimizing outbound efficiency and maintaining product quality.

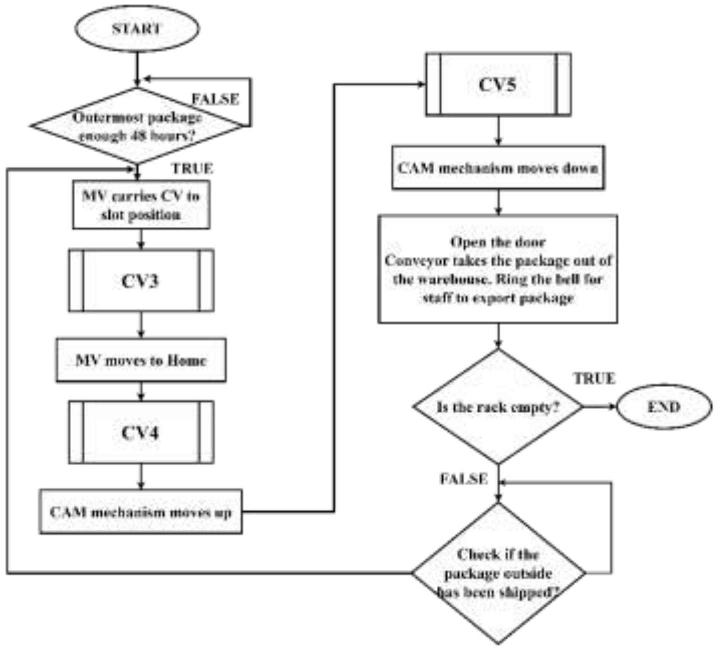


Figure 4. 7 Product export

4.3.3.1. CV3

This subroutine describes in detail the specific steps the child vehicle performs to retrieve a pallet from its storage location on the shelf and then safely transport it back to the parent vehicle to complete the outbound process. This is a core segment within the entire automated outbound procedure, requiring seamless coordination between mechanical components and intelligent control systems.

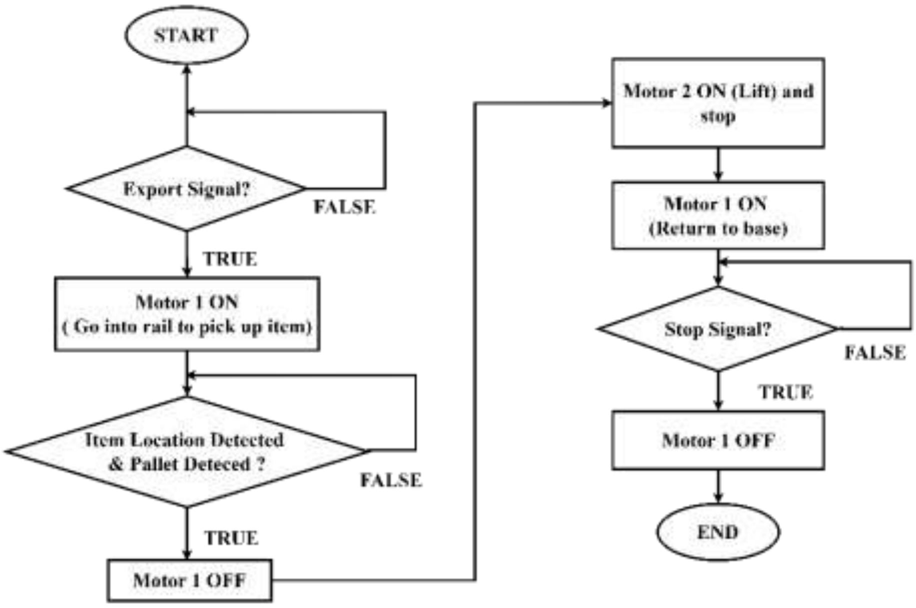


Figure 4. 8 CV3 sub

4.3.3.2. CV4 and CV5

This process is a crucial part of the automated outbound cycle, detailing the steps the child vehicle performs to transport a retrieved pallet from the storage area to the CAM mechanism on the outbound conveyor, and then complete its task by returning to the parent vehicle. The coordination among the child vehicle, the CAM mechanism at the conveyor, and the conveyor system is key to ensuring efficiency and accuracy.

It ensures that the pallet is safely and precisely transferred from the child vehicle to the outbound conveyor to continue its journey out of the warehouse, while simultaneously allowing the child vehicle to return to its ready position for the next task.

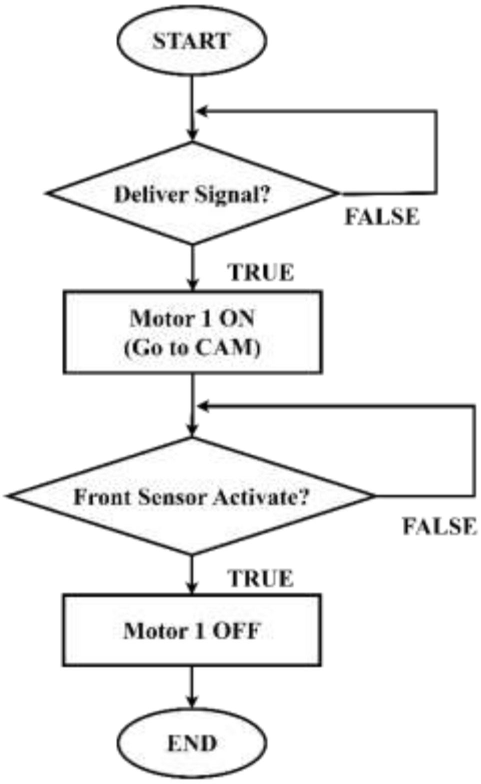


Figure 4. 9 CV4 sub

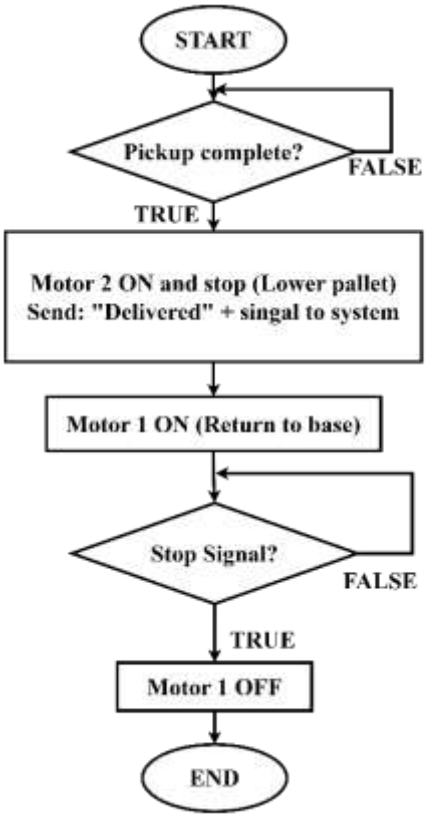


Figure 4. 10 CV5 sub

CONCLUSION

❖ Results:

- Maximum System Capacity: $12\ tons/hour$
- Speed:
  - Horizontal Travel Mechanism of MV: $0.4 m/s$
  - Lifting Mechanism of MV: $0.2\ m/s$
  - Child Vehicle of CV: $0.3 m/s$
  - Lifting Mechanism of CV: $0.025 m/s$
- Maximum Cycle Time: $5 min$
- Horizontal Error: $\pm\ 5mm$
- Elevation Error: $\pm\ 5mm$
- Position Error: Type equation here.

❖ Development Direction:

- Integration of advanced warehouse management systems (WMS) and warehouse control systems (WCS): Develop and implement detailed WMS/WCS software with Mother Vehicle and Child Vehicle path optimization algorithms (e.g. dual-travel algorithms, real-time optimization) to minimize cycle time and maximize productivity. Integrate intelligent energy management modules.
- Research and application of advanced materials: Explore special polymer, composite or alloy materials with superior insulation and cold resistance, while reducing the weight of mobile structures, thereby reducing the load on the motor and saving energy.
- Optimize motor and transmission system design: Research and test high-performance motors that can operate stably in low-temperature environments with minimal energy consumption. Consider direct drive technologies to reduce energy loss due to friction.

- Develop energy recovery systems: Integrate mechanisms to recover energy from the deceleration or unloading process of the lifting and moving system, convert it into electricity for reuse, contributing significantly to reducing overall energy consumption.
- Apply sensor technology and AI/Machine Learning: Develop smart sensor systems (e.g. LiDAR, AI cameras) to improve the system's ability to identify locations, avoid collisions, and self-diagnose faults. Apply machine learning algorithms to optimize routes and operating schedules based on real-world data.
- Dynamics and Virtual Reality Simulation: Expand the simulation scope to more detailed dynamic simulations to evaluate mechanical behavior, vibration, durability. At the same time, build a virtual reality (VR) simulation environment to visualize and test operating processes before actual deployment.
- Life Cycle Assessment and Sustainability: Conduct a Life Cycle Assessment (LCA) for the entire AS/RS to assess the environmental impact from production, operation to disposal, thereby proposing more sustainable design solutions.

REFERENCES

[1] ASHRAE. (2018). *ASHRAE Handbook—Refrigeration* (Chapter 17: Food Freezing and Refrigerated Storage). American Society of Heating, Refrigerating and Air-Conditioning Engineers.

[2] *MF Cold Storage*, "Manual Cold Storage Warehouse Layout". [Online]. Available: https://www.mfwc-cold.com/wp-content/uploads/frz2-768x366.png. [Accessed: 11-May-2025].

[3] Raymond Ortiz, "Rack-and-pinion steering diagram," *MotorBiscuit*, Oct. 10, 2023. [Online]. Available: https://www.motorbiscuit.com/wp-content/uploads/2023/10/rack-and-pinion-steering.jpg?w=768&h=607. [Accessed: 11-May-2025].

[4] AMAZEN Co. Ltd., "Motor Transmax B3 TMX BA1 112M 4.5 HP," *AMAZEN*, [Online]. Available: https://amazen.com.vn/motor-transmax-b3-tmx-ba1-112m-4-5hp.html. [Accessed: 11-May-2025].*IndiaMART*, "Electric motors," [Online]. Available: https://2.imimg.com/data2/PB/VA/MY-2641472/electric-motors-250x250.jpg. [Accessed: 11-May-2025].

[5] "Standard Spindelhubgetriebe," *Herion Linner*, [Online]. Available: https://herionlinner.com/wp-content/uploads/2022/12/standard-spindelhubgetriebe-300x200.jpg. [Accessed: 11-Jun-2025].

[6] "Os ensaios funcionais de aceitação dos cilindros hidráulicos," *Normas.com.br*, [Online]. Available: https://www.normas.com.br/visualizar/artigo-tecnico/4442/os-ensaios-funcionais-de-aceitacao-dos-cilindros-hidraulicos. [Accessed: 11-Jun-2025].

[7] "CB 1500B," *Storit Group*, [Online]. Available: https://storitgroup.com/uploads/2022/05/cb_1500b_1_big.jpg. [Accessed: 11-Jun-2025].

[8] YouTube thumbnail image for "Pallet Shuttle System - How It Works," *YouTube*, [Online]. Available: https://i.ytimg.com/vi/ryAM3Aki6Aw/hq720.jpg. [Accessed: 11-Jun-2025].

[9] [3] YouTube thumbnail image for "Automated Shuttle Pallet Racking System," *YouTube*, [Online]. Available: https://i.ytimg.com/vi/OayhGWucPMw/mqdefault.jpg. [Accessed: 11-Jun-2025].

[10] Image of shuttle racking system, *Google Images*, [Online]. Available: https://encrypted-tbn0.gstatic.com/images?q=tbn:ANd9GcS6G1tgRuQe2ZZkc-idF5TPX1xYsgpBUcJNkRfBPAPvjj4U1BYuGs1OOFDDBbM5qb8dQQU&usqp=CAU. [Accessed: 11-Jun-2025].

[11] Jinan Keya Electronic Science and Technology Co., Ltd., "48V 800W 1500rpm Brushless High-Torque DC Motor for AGV (model KY110AS0408-15B)," Product page. Online. Available: https://www.dcmotorkeya.com/48v-800w-1500r-Brushless-High-Torque-DC-Motor-for-AGV-pd501167158.html. Accessed: 11-Jun-2025.

[12] Image of pallet shuttle vehicle, *Google Images*, [Online]. Available: https://encrypted-tbn1.gstatic.com/images?q=tbn:ANd9GcTeHir-D4Jb33xAkyg7ivE44R40hXzHd_aB-XJQKwxho52JyKGB. [Accessed: 11-Jun-2025].

[13] MCVtech, *"Thanh chống đa năng 17,"* [Online]. Available: https://mcvtech.com.vn/san-pham/thanh-chong-da-nang-17/. [Accessed: 28-May-2025].

[14] Tostarp AB, *"Värmekabel 4m Effekt 40W/m,"* [Online]. Available: https://tostarp.se/produkt/varmekabel-4m-effekt-40w-m/. [Accessed: 28-May-2025].

[15] Leadshine Việt Nam, "Động cơ Servo tích hợp iSV2-RS8075V48G," [Online]. Available: https://leadshine.com.vn/san-pham/dong-co-servo-tich-hop-isv2-rs8075v48g/. [Accessed: 11-Jun-2025].

[16] Siemens AG, "SIMATIC S7-1200 CPU 1215C (6ES7215-1AG40-0XB0)," Dattech. Online. Available: https://dattech.com.vn/p/6es7215-1ag40-0xb0/. Accessed: 11-Jun-2025.

[17] Siemens AG, "SIMATIC S7-1200 System Manual," Siemens Industry Online Support. Online. Available: https://support.industry.siemens.com/cs/mdm/91696622?c=60466742283&dl=ru&lc=en-GB. Accessed: 11-Jun-2025.

[18] Autonics, "Cảm biến quang chịu dầu BJR Series," Daihoaphu.vn. Online. Available: https://daihoaphu.vn/san-pham/cam-bien-quang-chiu-dau-autonics-bjr-series. Accessed: 11-Jun-2025.

[19] Zhejiang Ximeng Electronic Technology Co., Ltd., "48 VDC to 12 VDC 10 A 120 W DC-DC Converter (SDJ-48S1210)," Product page. Online. Available:

https://www.alibaba.com/product-detail/48Vdc-to-12Vdc-10-AMP-120_60425568193.html. Accessed: 11-Jun-2025.

[20] Logos-World, "MATLAB logo," Online. Available: https://logos-world.net/matlab-logo/. Accessed: 11-Jun-2025.

[21] A. Hafeez Ul Haq và H. I. Okumuş, “DC Motor electrical model,” trong *A Simple Approach to DC Motor Modeling and Comparative Implementation of PI and FLC for its Speed Controlling*, hội thảo, Apr. 2016. Hình minh họa mô hình mạch điện của động cơ DC. Online tại ResearchGate. Truy cập: 11-Jun-2025.

[22] Download.com.vn, "Logo phần mềm MATLAB," Online. Available: https://images.dwncdn.net/images/t_app-icon-l/p/5e44f23c-aebe-42ae-8320-87bb1e49d43d/687739668/18495_4-77420004-logo. Accessed: 11-Jun-2025.

[23] “TIA Portal v-16 PLC Programming,” YouTube playlist. Online. Available: https://www.youtube.com/playlist?list=PLSA7Uuz0m1L2ZrFq4XAwaWN66Jh99eche. Accessed: 11-Jun-2025.

[24] Mecalux, "Case study of the Child vehiclesystem in the Didactic warehouse in France," [Online]. Available: https://www.mecalux.com/case-studies/example-pallet-shuttle-system-didactic-france. Accessed on May 28, 2025.

[25] Movu Robotics, "atlas - The child vehiclesystem," [Online]. Available: https://www.movu-robotics.com/en/atlas-pallet-shuttle. Accessed on May 28, 2025.

[26] Bastian Solutions, "Child vehicleAS/RS | Automated Storage and Retrieval System," [Online]. Available: https://www.bastiansolutions.com/solutions/technology/asrs/pallet-shuttle/. Accessed on May 28, 2025.

[27] SMB International, "Automated Pallet Shuttles for Efficient Storage Systems," [Online]. Available: https://www.smb-international.com/automated-pallet-warehouse/pallet-shuttle. Accessed on May 28, 2025.

[28] Moffett Automation, "Moffett Automation | Automated Pallet Storage," [Online]. Available: https://moffettautomation.com/. Accessed on May 28, 2025.

[29] HWArobotics, "Child vehicleRobot System - Automated Storage and Retrieval System Manufacturer," [Online]. Available: https://www.hwarobotics.com/products/pallet-shuttle-robot-system/. Accessed on May 28, 2025.

[30] Conveyco, "New Child vehicleAS/RS System Provides Labor Savings & Economical Automated Pallet Handling," [Online]. Available:

https://www.conveyco.com/news/new-pallet-shuttle-as-rs-system-provides-labor-savings-economical-automated-pallet-handling/. Accessed on May 28, 2025.

[31] Mecalux, "Case study of Child vehiclesystem in Gémo warehouse in France," [Online]. Available: https://www.mecalux.com/case-studies/example-pallet-shuttle-system-gemo-france. Accessed on May 28, 2025.

[32] Cassioli, "Multi-Depth Automated Warehouse with ISAT Pallet Shuttle," [Online]. Available: https://www.cassioli.com/intralogistics-division/automated-warehouses/pallet-shuttle/. Accessed on May 28, 2025.

[33] Swisslog EMEA, "Pallet Automation Improved Pepsi Storage Capacity by 61.5%," [Online]. Available: https://www.swisslog.com/en-gb/case-studies-and-resources/case-studies/2016/07/pepsi. Accessed on May 28, 2025.

[34] Mecalux, "Child vehicleRacking System," [Online]. Available: https://www.mecalux.com/warehouse-racking-and-shelving/pallet-shuttle. Accessed on May 28, 2025.