Mohamed Khider University of Biskra
Faculty of science and technology
Department of mechanical engineering

MASTER THESIS

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Presented and supported by:

Guettaf Abderahmene and Gheurd Soufyane

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Conception of multi-function machine for sweeping, cleaning roads and mowing lawns

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Jury :

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Academic year : 2023 - 2024

Dedication

*To the one who has always been and continues to be a guiding light in my life and academic journey, who sacrificed, stayed awake, and toiled for me to learn and study and to become who I am today, to my mother **Fatiha Riqt**, my support in this life, whom no collection of languages' dictionaries can adequately describe, and no matter how much I try, I can never repay her kindness, may Allah preserve you and bless your life, my dear mother.*

*To the one who taught me how to stand tall in the face of challenges and obstacles in pursuit of my goals, to my father **LazhariGeuttaf**, who toiled and stood firm in the face of difficult circumstances so that I could study and become who I am today, and no matter how hard I try, I can never fully repay his efforts, may Allah protect you and bless your life, my dear father.*

*To my heroic brothers and champion sisters **Karim, Mohamed, Soumia, Ghania, Amina**, members of my family and my support in this life, who rejoice in my achievements even more than I do, may Allah protect us, keep us united, and bless all of you.*

To my dear uncles who are also my greatest support, may Allah protect and bless you and bless your lives.

*To my professors in the field of Mechanical Engineering, especially Professors 'Yacoub' and **Harzli** who have greatly contributed to guiding me and assisting me in gathering information, may Allah preserve you both, benefit from your knowledge, and bless your lives.*

To my dear friends **'Soufyane, Nour El-Din, Emad, Wassila** may Allah keep you all as sources of goodness and support, protect you, and bless your lives.

Geuttaf Abdrahmen

Dedication

To the one who has always illuminated my life and contributed to my academic journey, to my dear mother ***MessoudaZaboubi*** *,whose beauty and kindness I cannot describe even if I gather all the words of languages, to you, my dear mother, I offer thanks and appreciation. May Allah preserve you and bless your life.*

To my father ***AmmarGheurd****, who taught me resilience in the face of challenges and stood by my side until I became who I am today, I cannot repay your kindness, my dear father. May Allah protect you and bless your life.*

*To my heroic brothers and sisters****, Ismahen, Hadil,Imad,*** *members of my family and pillars of my life, thank you for your support and joy in my success. May Allah preserve our unity and bless all of you.*

To all the dear members of my family who have been a support to me, may Allah protect them and bless them in their lives.

To my professors in the field of Mechanical Engineering, who have greatly contributed to guiding me and assisting me, may Allah preserve them, benefit from their knowledge, and bless their lives.

To my dear friends ***AbdRahmen, Nour El-Din, Emad,*** *may Allah keep you all as sources of goodness and support, protect you, and bless your lives.*

Gheurd soufyane

Remerciement

In the name of Allah, the Most Gracious, the Most Merciful. By the grace and mercy of Allah, we have achieved excellence. All thanks and praise are due to Him in times of ease and hardship, and by His praise, we ascend to greatness and glory, and with His light, dreams are realized in this existence.

First and foremost, we would like to express our deepest gratitude to Professor ***Tahar Masri*** *for accepting to guide us in this project and providing us with all necessary support to complete it. We also extend our sincere thanks to the members of the supervisory committee for reviewing and evaluating our work.*

Furthermore, we cannot forget to thank all the teachers and administrators in the Mechanical Engineering Department for their efforts throughout our academic journey. Special thanks are also due to the professors of economics and accounting for providing us with valuable training courses.

Finally, we thank all those who contributed in one way or another to the success of this project, and to all those whom we did not mention, thank you.

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General Introduction

Design is the creation of synthesized solutions in the form of products or systems that satisfy customer's requirements. When we are given a design problem, we try to make the best use of our knowledge and the available information to understand the problem and generate as many feasible solutions as possible. The design process can be logically divided into three interrelated phases: (1) product specification and planning phase, (2) conceptual design phase, and (3) product design phase.

During the product specification and planning phase, we identify the customer's requirements and translate them into engineering specifications in terms of the functional requirements and the time and money available for the development, and plan the project accordingly.

In the conceptual design phase, we generate as many design alternatives as possible, evaluate them against the functional requirements, and select the most promising concept for design detailing. A rough idea of how the product will function and what it will look like is developed.

In the product design phase, we perform a thorough design analysis, design optimization, and simulation of the selected concept. Function, shape, material, and production methods are considered. Techniques for generation of concepts include literature and patent search, imitation of natural systems, analysis of competitor products, brainstorming, etc.

The design is traditionally accomplished by the designer's and experience using the abstract representation of the kinematic structure. The kinematic structure contains the essential information about which link is connected to which other links by what types of joint. Finally, an engineering documentation is produced and the design goes into the production phase. However, if the concept selected for the product design is shown to be impractical, it may be necessary to go back to the conceptual design phase to select an alternate concept or to generate additional concepts. In this regard, it may be necessary to revaluate the engineering specifications developed in the product specification and planning phase.

Our work aims to conduct a comprehensive analysis of a multifunctional road maintenance device, using advanced tools and techniques for designing and analysing the device's performance. The main objective of this work is to create a machine that performs several tasks at once; such as sweeping and cleaning roads, tracing roads, mowing lawns and equipped with a GPS system which makes the machine programmable. The main objective of this work is to create a machine that performs several tasks at once; such as sweeping and cleaning roads, tracing roads, mowing lawns and equipped with a GPS system which makes the machine programmable. Due to time constraints, the machine studied will be used as a cleaning machine or a lawn mower.

In this study, the analysis will cover multiple aspects such as mechanical design, materials used, device performance under various conditions, and economic efficiency.

To implement this project, we have divided the work into several parts:

The first chapter represent a review on the history of dynamic motion modelling and the foundations of kinematics and dynamics, focusing on mechanisms and linear, rotational, and oscillating motions. It also addresses applied mechanics through the study of joints, gears, and pulleys, and their roles in mechanical systems.

The second part covers the design of the multifunctional device and its importance in urban environments. The chapter also includes an analysis of the device, encompassing mechanisms and control, and a comprehensive analysis of the electric drive system in terms of kinematics, energy analysis, and efficiency.

Chapter three deals with the static analysis of the forces affecting the device and the improvement of shaft dimensions to enhance durability and performance. Moreover, this part includes the selection and verification of keys.

Finally, the last part represents a modelling of the machine with Solidworks, where the parts have been drawn and assembled. In addition, this part includes numerical simulations on certain essential parts of the machine.

Chapter I

Introduction and Overview

I.1 Introduction

In the realm of mechanical engineering, the design and innovation of machinery are profoundly influenced by the principles of kinematic and dynamic modelling. Kinematic modeling focuses on the motion of components within a system, providing a detailed description of their trajectories, velocities, and accelerations without regard to the forces that produce such movements[1]. Dynamic modelling goes a step further to incorporate these forces, offering a comprehensive understanding of how mechanical systems respond under various conditions[2].

The significance of kinematic and dynamic modelling cannot be overstated. They serve as the cornerstone for developing efficient, reliable, and innovative machines. Through these models, engineers can predict the behaviour of mechanical systems, optimize their performance, and identify potential issues before they arise. This predictive capability is crucial in the design process, enabling the creation of sophisticated machinery that meets the evolving demands of industry and technology[3].

Moreover, kinematic and dynamic modelling plays a pivotal role in the field of robotics and automation. They provide the mathematical framework necessary for programming robotic systems to perform complex tasks with precision and accuracy. As we continue to push the boundaries of what machines can do, these models will be instrumental in shaping the future of mechanical design and innovation[4].

In this chapter, we will explore the foundational concepts of kinematic and dynamic modeling, their historical development, and their application in modern engineering practices. This exploration will set the stage for the subsequent chapters, where we will delve into the specifics of designing an innovative machine, complete with kinematic calculations, energy considerations, and dynamic analysis[2, 5].

I.2 History of Kinematic Modelling

Kinematic modelling, the study of motion without considering the forces that cause it, has been a cornerstone in the field of mechanical engineering. Its history is rich and spans several centuries, reflecting the evolution of human understanding of motion and mechanics.

I.2.1 Ancient Origins

The earliest known studies of kinematics can be traced back to ancient Greece. Aristotle's work on the motion of objects laid the groundwork for future exploration in this field. However, it was Archimedes who made significant contributions by applying geometric principles to understand the kinematics of levers and pulleys[6].

I.2.2 Renaissance Innovations

During the Renaissance, kinematic modelling flourished as scholars such as Leonardo da Vinci and Albrecht Durer began to analyse and document mechanical systems with greater precision. Their work led to the development of new mechanisms and machines, which were crucial for the advancement of engineering and technology[7].

I.2.3Industrial Revolution and Beyond

The Industrial Revolution brought about a surge in the development of kinematic models as the need for more sophisticated machinery grew. Engineers like James Watt and Franz Reuleaux made notable advancements. Reuleaux, in particular, is often referred to as the father of kinematics for his systematic study of machine components and the creation of a vast collection of kinematic models[8].

I.2.4 Modern Kinematics

In the 20th century, the field of kinematics expanded rapidly with the advent of computers and robotics. The development of kinematic chains, linkages, and cams transformed the design of automated systems. The introduction of the Denavit-Hartenberg*(DH)* parameters in the 1950s standardized the mathematical description of jointed mechanisms, which is still widely used in robotics today[9].

I.2.5 Impact on Machine

Design: Kinematic modelling has profoundly impacted machine design throughout history. By understanding the motion of individual components, engineers can create more efficient and precise machinery. In modern times, kinematic modelling is integral to the design of robots and automated systems, allowing for complex tasks to be performed with high accuracy[9].

Figure I. 1 : Historical Visit Two of Over 35,000 Visitors Viewing the Collection in 1930.

I.3Foundations of Kinematics and Dynamics

The study of kinematics and dynamics forms the bedrock of mechanical engineering, providing the tools and principles necessary to analyse and design complex mechanical systems. Kinematics deals with the geometry of motion, focusing on how objects move, while dynamics brings in the forces and torques that cause or result from that motion. Together, they offer a comprehensive framework for understanding the behaviour of machines and mechanisms, from the simplest gears to the most intricate robotic systems[10-12].

I.3.1Mechanism

A mechanism is an assembly of moving parts that harnesses the mechanical principles to perform a specific function or task. It often includes elements such as gears, levers, and cams, which work together to convert types of motion (e.g., from rotational to linear) or to change the magnitude and direction of forces[13].

I.3.2 Types of Motion in Mechanisms

Mechanisms include different types of motion, which are:

I.3.2.1 Linear Motion

Linear motion, or translation, occurs when all parts of an object move in the same direction and distance, parallel to a straight line.This type of motion is governed by Newton's laws of motion and can be uniform (with constant velocity) or non-uniform (with varying velocity due to acceleration)[12].

I.3.2.2 Rotational Motion

Rotational motion involves an object spinning about an axis. This axis can be internal (like the Earth rotating on its axis) or external (like a wheel rotating around a fixed pivot). The dynamics of rotational motion are described by equations that are analogous to those for linear motion but expressed in terms of angular quantities[10-12].

I.3.2.3 Oscillating Motion

Oscillating motion, also known as vibratory motion, is characterized by the back-and-forth movement of an object about a central position. This type of motion is typically periodic, such as the swinging of a pendulum or the vibration of a guitar string. The study of oscillations involves understanding the restoring forces and damping that affect the system's behaviour[10]**.**

I.4 Applied Mechanics: Joints, Gears, and Pulleys

In the realm of applied mechanics, understanding the intricacies of joints is crucial as they form the connections that allow structures and mechanisms to function harmoniously, leading us to explore the various types of joints utilized in mechanical engineering.

I.4.1Joints in Mechanical Engineer

Joints are essential elements in mechanical systems that facilitate controlled motion between parts. They are the points at which two or more parts are connected and can be designed to allow various types of movement, including translational, rotational, or a combination of both. The design of joints is critical as it affects the efficiency and functionality of the entire mechanical system.

Translational Joints allow movement in one or more linear directions without rotation. Examples include sliders and linear bearings.Rotational Joints permit rotation around an axis and include common joints such as hinges and pivots. Combination Joints can handle both translational and rotational movements, providing a more complex range of motion.

The design and selection of joints depend on several factors, such as the required range of motion, the loads they will carry, the environment in which they will operate, and the desired lifespan of the joint. Engineers must also consider manufacturing capabilities, material properties, and maintenance requirements[14].

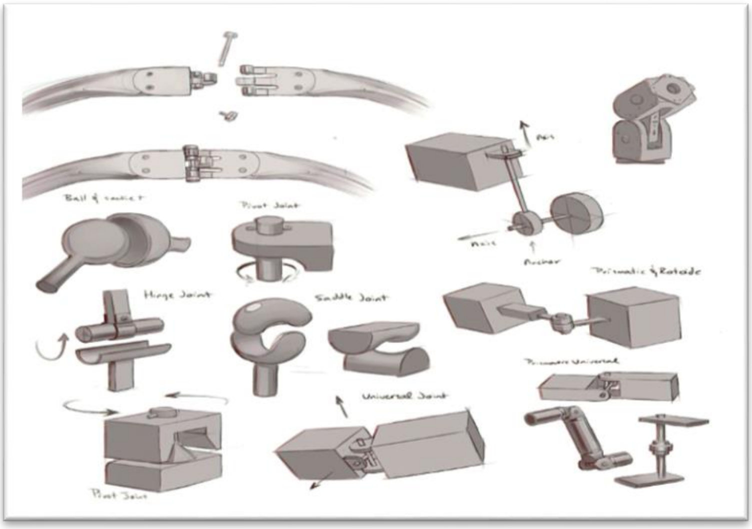


Figure I. 2 : Mechanical Parts - Hinges & Joints.

I.4.2 Types of Joints and Their Functions

Joints are essential structural elements in mechanical designs, providing the necessary connections between different parts of machines and structures. These joints vary to accommodate the required movements and loads they bear, including both fixed and movable types, each performing a specific function that contributes to the efficiency and safety of the overall system[18].

I.4.2.1 Fixed Joints

Fixed joints are integral to mechanical design, providing permanence and stability where needed.

- **Welded joints** are formed by the high-heat fusion of two metal pieces, creating a bond that is not easily broken without cutting. Their strength is essential for structures demanding rigidity, such as frames and bridges.
- **Riveted joints**, consisting of a head and a tail, connect two plates by deforming the tail to clamp them together, offering high strength in scenarios unsuitable for welding, like in boilers and tanks. Lastly.
- **Adhesive joints** employ a bonding agent to unite parts, particularly beneficial for materials challenging to weld or rivet. They ensure stress is distributed evenly across the joint and find widespread use in the automotive and aerospace industries, where reliability and durability are paramount[18].

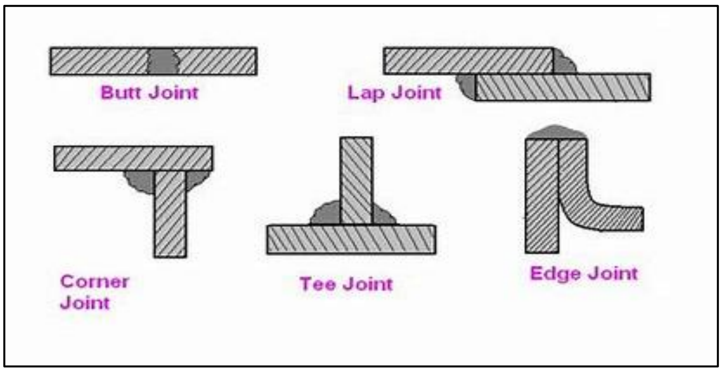


Figure I. 3 : Welded joints in the Design of Machines.

I.4.2.2 Movable Joints

Movable joints are crucial in mechanical systems where flexibility and reconfiguration are necessary.

- **Pivot joint** allows for rotational movement around a single axis. It's typically found at the baseplate's center, enabling the attached coupling to swivel from side to side.
- **Bolted joints** are a prime example, offering a non-permanent solution that can be easily assembled and disassembled, making them ideal for maintenance and machinery adjustments**.**
- **Hinged joints** provide rotational movement along a single axis and are the go-to choice for applications like doors and lids that require a reliable pivot point.
- **Ball joints** stand out for their ability to support movement in multiple axes, which is essential in complex systems such as vehicle suspensions and robotics, where a wide range of motion is required for optimal functionality.

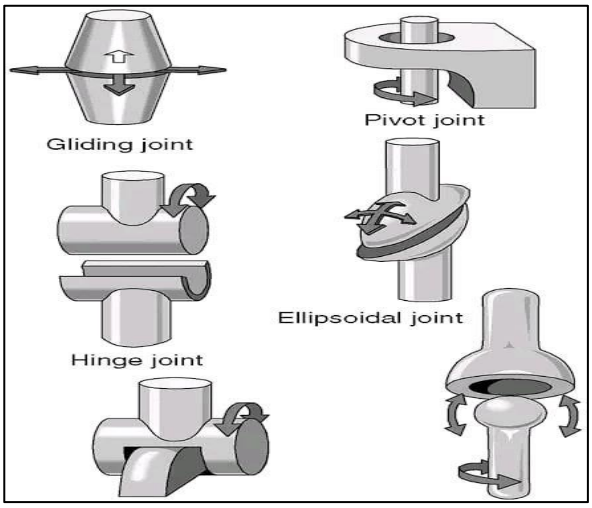
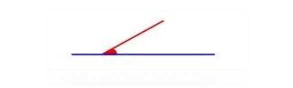
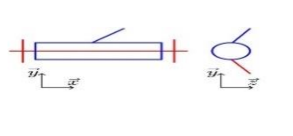
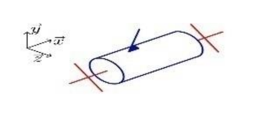
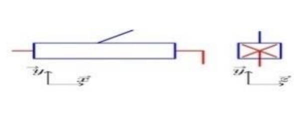
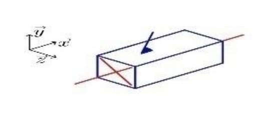
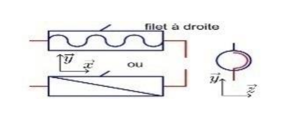
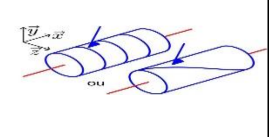
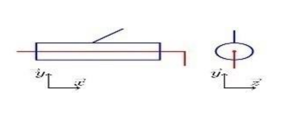
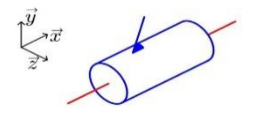
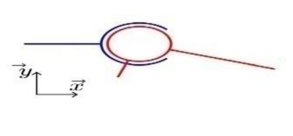
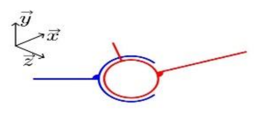
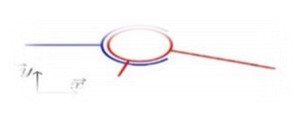
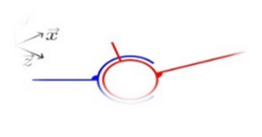
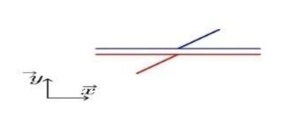
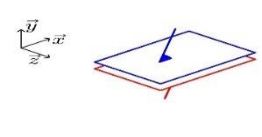
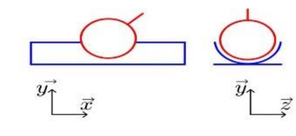
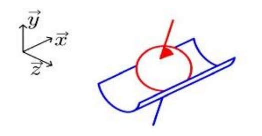
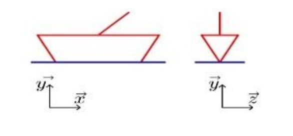
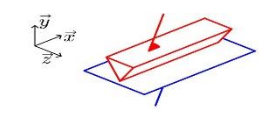
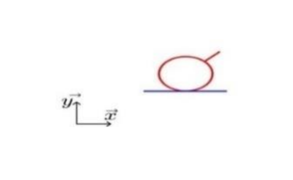
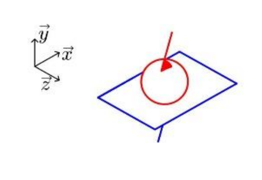


Figure I. 4 : Types of Movable Joints in the Design of Machines.

Table I. 1 : of connections in the case of a plane movement[15].

Name	DOFs	Representationplane	3D representation	Torsorcinematic	AM Torso
Recessed	0			$A\begin{bmatrix}0 & 0\\0 & 0\\0 & 0\end{bmatrix}$	$\begin{pmatrix}x_{01} & l_{01}\\y_{01} & M_{01}\\z_{01} & N_{01}\end{pmatrix}A$
Axle pivot(A,$\vec{x}$)	1			$A\begin{bmatrix}w_x & 0\\0 & 0\\0 & 0\end{bmatrix}R$	$\begin{pmatrix}x_{01} & 0\\y_{01} & M_{01}\\z_{01} & N_{01}\end{pmatrix}A$
Slide direction $\vec{x}$	1			$A\begin{pmatrix}0 & v_x\\0 & 0\\0 & 0\end{pmatrix}R$	$\begin{pmatrix}0 & l_{01}\\y_{01} & M_{01}\\z_{01} & N_{01}\end{pmatrix}A$
Helicalof axis (A, $\vec{x}$)	1			$A\begin{pmatrix}w_x & P.w_x\\0 & 0\\0 & 0\end{pmatrix}R$	$\begin{pmatrix}x_{01} & pX_{01}\\y_{01} & M_{01}\\z_{01} & N_{01}\end{pmatrix}A$
Sliding pivotof axis (A, $\vec{x}$)	2			$A\begin{pmatrix}w_x & v_x\\0 & 0\\0 & 0\end{pmatrix}R$	$\begin{pmatrix}0 & 0\\y_{01} & M_{01}\\z_{01} & N_{01}\end{pmatrix}A$
Finger balljointofcenterAblocked at $\vec{x}$	2			$A\begin{bmatrix}0 & 0\\w_y & 0\\w_z & 0\end{bmatrix}R$	$\begin{pmatrix}x_{01} & l_{01}\\y_{01} & 0\\z_{01} & 0\end{pmatrix}A$
Ball jointcenter A	3			$A\begin{bmatrix}w_x & 0\\w_y & 0\\w_z & 0\end{bmatrix}R$	$\begin{pmatrix}x_{01} & 0\\y_{01} & 0\\z_{01} & 0\end{pmatrix}$A
Plane ofnormal$\vec{y}$	3			$A\begin{pmatrix}0 & v_x\\w_y & 0\\-0 & v_z\end{pmatrix}R$	$\begin{pmatrix}0 & l_{01}\\y_{01} & 0\\0 & N_{01}\end{pmatrix}$A
Ring finger inAxis A(A, $\vec{x}$)	4			$A\begin{vmatrix}w_x & v_x\\w_y & 0\\w_z & 0\end{vmatrix}R$	$\begin{pmatrix}0 & 0\\y_{01} & 0\\z_{01} & 0\end{pmatrix}$A
Rectilinearline (A, $\vec{x}$)and normal$\vec{y}$	4			$A\begin{pmatrix}w_x & x\\w_y & 0\\0 & v_z\end{pmatrix}R$	$\begin{pmatrix}0 & 0\\y_{01} & 0\\0 & N_{01}\end{pmatrix}A$
Punctual inA standard $\vec{y}$	5			$A\begin{pmatrix}w_x & x\\w_y & 0\\0 & v_z\end{pmatrix}R$	$\begin{pmatrix}0 & 0\\y_{01} & 0\\0 & 0\end{pmatrix}A$

I.4.3 GearsConcepts and Main Types

We can introduce a sentence that explains the significance of gears and their fundamental role in transmitting motion and altering speeds in various machines. For example:

I.4.3.1 Gear Basics

In the realm of mechanical engineering, gears are essential components that facilitate the transmission of power and motion between machinery parts. These intricate devices come in various shapes and sizes, each tailored to specific mechanical needs. The design and application of gears are a testament to the ingenuity of engineers and their quest for precision and efficiency.

Gears operate on the principle of rotational motion, where the teeth of one gear interlock with those of another, allowing for the controlled transfer of torque. This interaction can alter the speed, torque, and direction of mechanical components, making gears indispensable in complex machinery[16]**.**

I.4.3.2 Main Types of Gears

Gears are fundamental elements in mechanical machines, playing a vital role in transmitting motion and force between different system components. Gears come in various shapes and sizes to meet the requirements of diverse applications, and the main types of gears include:

- **Spur Gears**

Spur gears are perhaps the most recognizable type of gear. They feature straight teeth that are parallel to the axis of rotation. This design simplicity makes them easy to manufacture and maintain. Spur gears are commonly used in applications where the drive and driven gears are required to be on parallel shafts. Their straightforward tooth design allows for a consistent transfer of power, making them suitable for a wide range of applications, from simple machinery to complex automotive systems[17].

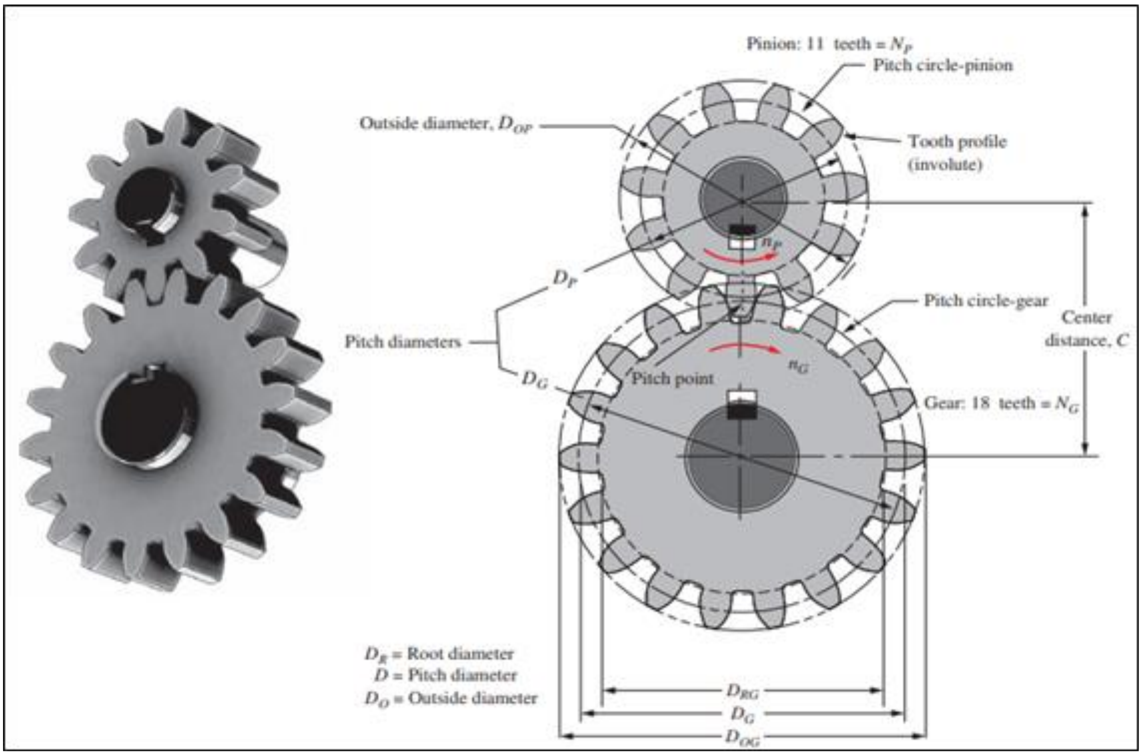


Figure I. 5: *Pair of spur gears. The pinion drives the gear [1].*

The characteristics of a spur gear are represented in the following table:

Table I. 2 : Designations ofa spur gear characteristics [17].

Designation	Symbol	Formula
Module	m	By an RDM calculation
Number of teeth	z	By a speed ratio
Pitch diameter	d	d= m×Z
Head diameter	d_a	da = d+2×m
Foot diameter	d_f	df = d-2,5×m
Projection	h_a	ha = m
Hollow	h_f	hf = 1,25×m
Tooth height	h	h = 2,25×m
Primitive pitch	p	p = π×m
Tooth width	b	b= km (5≤k≤16)
Center distance	a	a= (d1+d2)/2

❖ Helical Gears

Helical gears are an evolution of spur gears with angled teeth. This angled approach allows for a gradual engagement between the teeth of the meshing gears, resulting in a smoother and quieter operation. The helix angle of the teeth also helps distribute the load more evenly across the gear face, which can increase the gear's lifespan. Helical gears are typically used in applications similar to those of spur gears but are particularly advantageous in high-speed or high-load scenarios where noise and vibration reduction are critical[17].

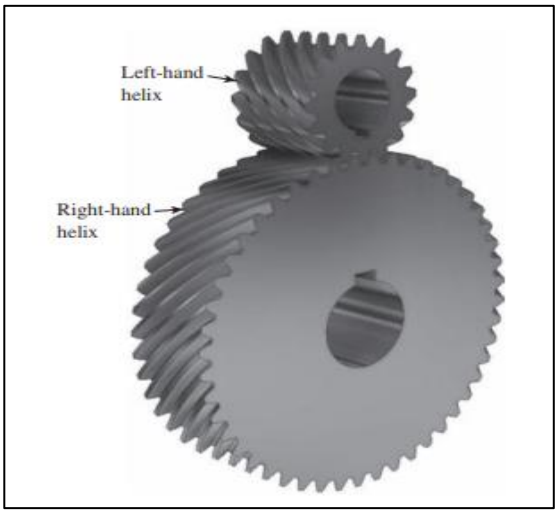


Figure I. 6 : Helical gears in mesh. These gears have a 45° helix angle[1].

❖ The characteristics of a Helical gears are represented in the following table:

Table I. 3 : Characteristics of helical Gear [18].

Designation	**Symbol**	**Value**
Helix angle	β*	Value between 15° and 30°
Propeller direction		If the gear is left-handed, the wheel will have a right-handed propeller
Number of teeth	Z	Positive integer linked toOperating and manufacturing conditions
Actual module	m_n	Determined by an Rdm calculation and chosen from the values standardized
Not real	p_n	Pn=π×mn
Exposed module	m_t	mt=mn/cosß
Not apparent	p_t	Pt=π×mt
Pitch diameter	d_p	df=d–2,5×mn
Center distance	a	a= (d1+d2)/2
Pressure angle	α	Généralement α=20°

❖ Worm Gears

Worm gears consist of a worm (which resembles a screw) and a worm wheel (which resembles a spur gear). This combination allows for high reduction ratios, making worm gears ideal for applications requiring a significant decrease in speed and an increase in torque. The non-parallel and non-intersecting shaft configuration of worm gears makes them versatile for various applications, including lifting mechanisms, conveyors, and tuning instruments. Their design also inherently provides a locking mechanism, preventing back-driving in many cases[17].

Each gear type has its own set of advantages and is chosen based on the specific needs of the application. The selection of the appropriate gear type is crucial for the optimal performance and longevity of the mechanical system.

Figure I. 7 : Worm and worm gear with a single threaded worm[1].

Table I. 4 : Worm gear characteristics [17].

Designation	**Symbol**	**Value**
Number of threads	Z_v	
Helix angle	*ß**	Reversibility functions of the transmission.
Actual module	m_n	Determined by calculation RDM
Axial modulus	m_z	$m_z=m_n/\cos\gamma_{vis}$
Unreal pitch	p_n	$p_n=\pi\times m$
Axial pitch	P_x	$P_x=p_n$
Screw length	*L*	L= 5×px environ

I.4.3.3 The Role of Gears in Mechanical Systems

Gears are mechanical components that are essential for transmitting power and motion in various machines. They come in different shapes and sizes, each designed for specific applications. The role of gears can be summarized as follows:

- **Transmission of Power**: Gears are designed to transmit power from one part of a machine to another. This is achieved by the interlocking of teeth, which allows for the transfer of torque and rotational motion[19].
- **Speed and Torque Conversion**: Gears can increase or decrease the speed of motion or increase the force, depending on their size and the number of teeth. This is known as the gear ratio[19].
- **Directional Change**: Gears can change the direction of motion. For instance, when two gears mesh together, they rotate in opposite directions, which is useful for machines that require a change in the motion's direction[20].
- **Motion Conversion**: Some gear arrangements can convert rotational motion into linear motion, which is essential in applications like rack and pinion systems[20].
- **Precision and Control**: Gears provide precise control over the movement of a machine. This is particularly important in applications where accuracy and control are critical, such as in robotics and aerospace[19].
- **Load Distribution**: Gears distribute the load over multiple teeth, which reduces the stress on individual teeth and extends the life of the gear[20].

I.4.4 Pulleys Concepts and Main Types

I.4.4.1 Pulley Basics

A pulley is a simple machine consisting of a wheel on an axle or shaft that is designed to support movement and change of direction of a taut cable or belt, or transfer of power between the shaft and cable or belt[21]. Pulleys are used to lift heavy loads, apply forces, and to transmit power.

I.4.4.2 Main Types of Pulleys

Pulleys are simple machines used to transmit force, typically in the form of tension, to move or lift heavy objects more efficiently. Understanding the main types of pulleys is essential for grasping their diverse applications in various mechanical systems.

- **Fixed Pulley**

Structure: A fixed pulley has its wheel fixed to a support(Figure I.8). The axle is stationary, meaning it doesn't move with the load.

Function: It changes the direction of the force applied(Figure I.8). When you pull down on the rope, the load is lifted up.

Mechanical Advantage: The mechanical advantage of a fixed pulley is 1, which means the force required to lift the load is equal to the weight of the load itself[22].

- **Movable Pulley**

is a type of pulley that moves with the load being lifted. Unlike a fixed pulley, which only changes the direction of the force applied, a movable pulley also provides a mechanical advantage, making it easier to lift heavy objects.

Advantages: It reduces the effort needed to lift an object, effectively halving the force required if there's one movable pulley.

Mechanical Advantage: The mechanical advantage is typically 2, as the force required is approximately half the weight of the load[22, 23].

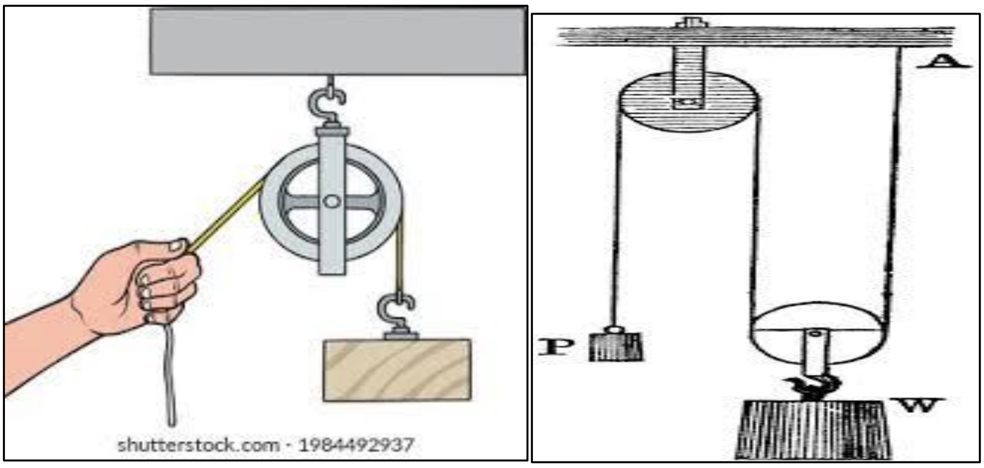


Figure I. 8 : Example of Movable and **Fixed Pulley**

- **Compound Pulley**

A compound pulley consists of multiple pulleys, which can be a combination of fixed and movable pulleys. The system is designed to distribute the load across several ropes and wheels, increasing the mechanical advantage.

Function: It significantly reduces the effort needed to lift heavy loads. By using multiple wheels and ropes, the force applied is multiplied, allowing for the lifting of heavier objects with less input force.

Mechanical Advantage: The mechanical advantage of a compound pulley is calculated by the number of rope segments supporting the load. If a system has four rope segments, the mechanical advantage is 4, meaning the force required to lift the load is a quarter of the load's weight[23, 24].

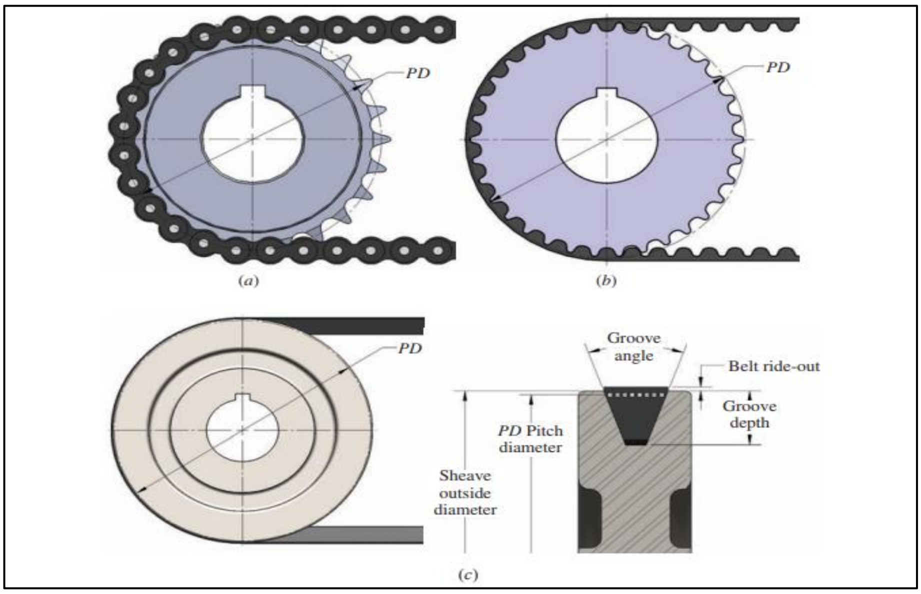


Figure I. 9 : Pitch diameter on a (a) chain sprocket, (b) synchronous belt sprocket, and (c) V-belt sheave with section view [1].

I.4.4.3 The Role of Pulleys in Mechanical Systems

Pulleys are crucial components in mechanical systems, serving multiple functions from lifting loads to transmitting power. They are particularly effective in systems where direct force application is impractical.

- **Transmission of Power**: Pulleys are used to transmit rotational power between different parts of a machine, often over distances and where direct coupling is not possible. This is achieved by the belt or rope wrapped around the pulley, which allows for the transfer of rotational motion.
- **Speed and Torque Conversion**: By using pulleys of different diameters, the speed and torque of the connected belts can be controlled. This allows for the regulation of the operational speed of machinery and the amount of force exerted.
- **Directional Change**: Pulleys can change the direction of an applied force, enabling the lifting

of loads in a direction opposite to the applied force, such as lifting a weight upwards by pulling down on a rope.

- **Mechanical Advantage**: Pulley systems, especially compound pulleys, provide a mechanical advantage that allows for lifting heavier loads with less input force. The mechanical advantage is determined by the number of rope segments supporting the load.
- **Precision and Control**: While pulleys do not offer the same level of precision as gears, they still provide significant control over the movement of loads, especially when used in systems with tensioners and guides.
- **Load Distribution**: Pulleys distribute the load across the belt or rope, reducing the stress on any single point and increasing the system's overall lifting or pulling capacity[23, 24].

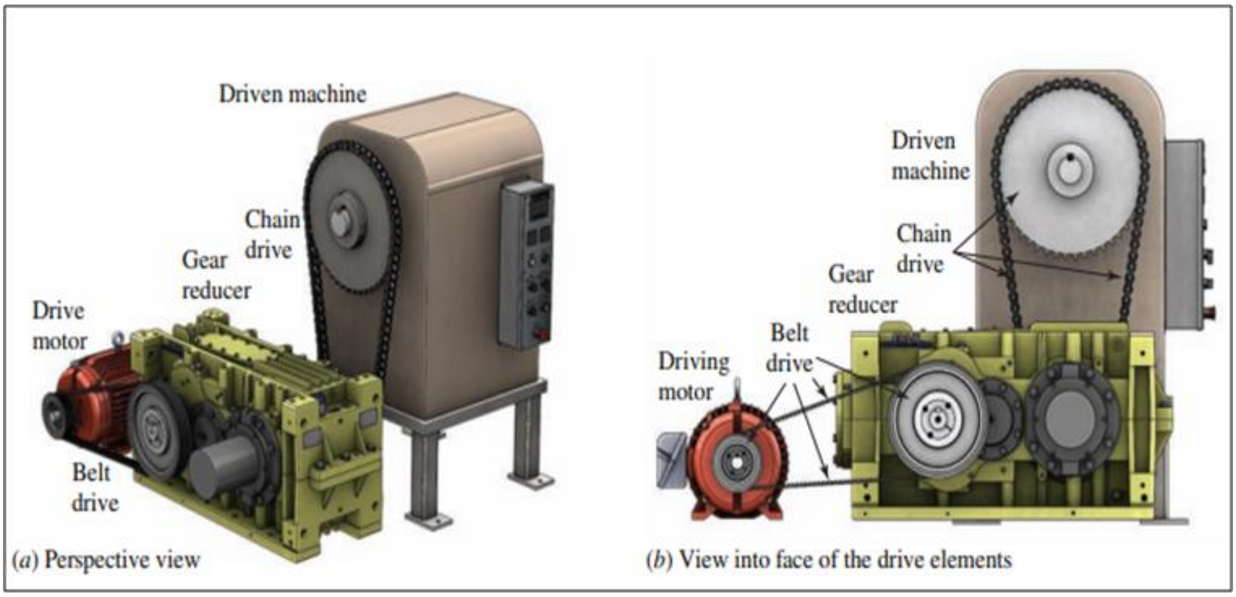


Figure I. 10 : Drive system for an industrial application employing a belt drive, a gear reducer, and a chain drive [1].

I.5 From Kinematic Modelling Basics to Practical Applications in Mechanical Engineering

In the second chapter, we will delve into the development of a multi-functional road maintenance device that combines sweeping, water spraying, and grass cutting. This innovative apparatus is designed to enhance the efficiency of road upkeep tasks. We will provide an extensive overview of the device, detailing its primary components and their roles in facilitating comprehensive road maintenance.

A kinematic diagram will be constructed to depict the interplay between the various parts, illustrating how they work in concert to execute the sweeping, spraying, and cutting functions necessary for thorough road care.

We will undertake a kinematic analysis to ascertain the different speeds and accelerations of the moving elements within the device. An energy analysis will also be conducted to evaluate the device’s operational efficiency and energy demands. Subsequently, a dynamic analysis will be carried out to comprehend the forces exerted and the interactions among the components during the maintenance activities.

This chapter aims to lay the groundwork for the design and examination of intricate mechanical devices and their practical deployment in tasks such as road maintenance. The principles and notions introduced in the initial chapter will be applied in a practical context to realize this objective.

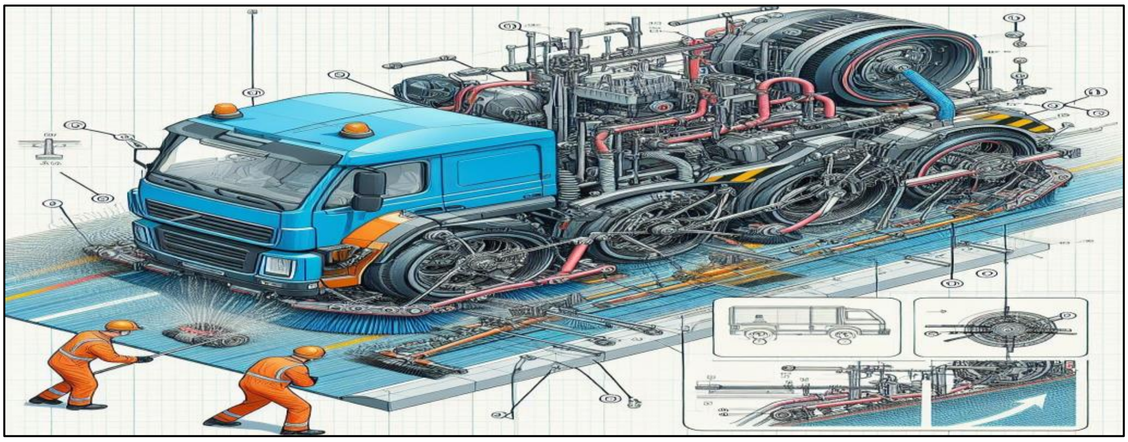


Figure I. 11 : The Future of Road Cleaning: An AI-Inspired Apparatus.

Chapter II

Design and Analysis of a Multi-Functional Road Maintenance Device

II.1Introduction

Roads serve as vital lifelines in modern society, enabling transportation, trade, and connectivity. However, alongside their crucial role, roads also pose challenges related to cleanliness and maintenance. The build-up of dirt, debris, and pollutants not only diminishes their visual appeal but also jeopardizes public health and safety[25]. Neglected maintenance accelerates road infrastructure deterioration, leading to higher repair costs and compromised user experiences[26].

To address these challenges, multi-functional road maintenance devices have been introduced as innovative solutions. These advanced systems not only clean roads but also offer additional services such as water spraying and grass cutting, providing a comprehensive approach to road maintenance. The integration of these functions into a single device streamlines the maintenance process, improving the efficiency of urban infrastructure management and ensuring the thorough upkeep of roadways for the benefit of all users[27].

II.2 Multi-Functional Road Maintenance Device

In our pursuit to develop a superior road maintenance solution, we engage in a rigorous analysis and design phase. This stage is essential to guarantee that each element and engineering aspect is meticulously crafted to produce a device that not only fulfils but surpasses the anticipated standards of efficiency and functionality [28]. The Multi-Functional Road Maintenance Device we envision is a comprehensive system that integrates various services to maintain roads, enhancing the effectiveness of urban infrastructure management.

II.2.1 Conceptualizing the Multi-Functional Road Maintenance Device

In the journey to revolutionize road maintenance, the conceptualization of the Multi-Functional Road Maintenance Device marks a significant milestone [28]. This device is envisioned as a comprehensive solution, adept at performing a multitude of tasks that go beyond mere cleaning. It's designed to adapt to various urban maintenance needs, including but not limited to sweeping, water spraying, grass cutting, and even minor road repairs.

The device's multi-functional nature allows for a more efficient allocation of resources, reducing the need for multiple specialized machines [26]. This not only streamlines the maintenance process but also contributes to a reduction in the carbon footprint associated with road upkeep. By integrating advanced technologies such as automation, sensor-based monitoring, and smart systems, the device ensures precision and effectiveness in maintaining urban infrastructure.

The conceptual phase focuses on creating a blueprint that balances innovative engineering with practical functionality, ensuring that the device can operate in diverse urban landscapes. The goal is to enhance the quality of life for city residents by keeping the roads in optimal condition, thus facilitating smoother transportation and contributing to the aesthetic and environmental health of the urban space [28]. The Multi-Functional Road Maintenance Device is a testament to the ingenuity and forward-thinking approach necessary for sustainable urban development.

II.2.2 Crucial Role of Multi-Functional Road Maintenance Devicein Urban Environments

At the heart of the urban fabric, the Multi-Functional Road Maintenance Device emerges as a fundamental element in enhancing sustainable urban development. Its role extends beyond traditional maintenance and cleaning functions to encompass a wide range of activities that contribute to the overall quality of life. By providing innovative solutions to everyday challenges faced by urban environments, the device improves road infrastructure, thereby facilitating traffic flow and reducing congestion, which positively impacts public health and the environment.

II.2.2.1 Enhancing Infrastructure and Contributing to Public Health

The Multi-Functional Road Maintenance Device is a vital tool in maintaining and improving roads, contributing to the removal of dirt and pollutants that can negatively affect air quality and the health of residents. By reducing levels of visual and environmental pollution, the device helps create a more attractive and cleaner urban environment, enhancing the comfort and well-being of both residents and visitors.

II.2.2.2 Innovation and Sustainability at the Core of Urban Planning

The Multi-Functional Road Maintenance Device reflects the innovative spirit of urban planning, combining advanced technologies and sustainable design to offer comprehensive solutions to environmental challenges. Its ability to adapt to various urban conditions and meet the needs of growing cities makes it an essential component in achieving sustainable development goals and enhancing the quality of urban life.

II.2.2.3 Strengthening Urban Infrastructure with The Multi-Functional Road Maintenance Device

The Multi-Functional Road Maintenance Device is essential for ensuring the longevity and reliability of our urban thoroughfares. As a critical component in the preservation and enhancement of our infrastructure, it stands at the forefront of urban maintenance.

Figure II. 1 : Road Sweeping and Grass Cutting Device

- **Preventing Road and Street Deterioration**

This innovative device combats the adverse effects of debris accumulation, which can accelerate the wear and tear of road surfaces. By proactively cleaning and maintaining the roads, The Multi-Functional Road Maintenance Device extends their longevity, thereby reducing the frequency and cost of repairs—a significant benefit for municipal budgets and infrastructure sustainability.

- **Safeguarding Pedestrians and Drivers**

With its comprehensive cleaning capabilities, The Multi-Functional Road Maintenance Device ensures that pathways, sidewalks, and crosswalks are kept free of hazards, enhancing pedestrian safety. For drivers, the benefits are twofold: cleaner roads lead to smoother surfaces, improving vehicle stability and decreasing the likelihood of accidents.

II.3 Integral Role of the Multi-Functional Road Maintenance Device in Urban Enhancement and Infrastructure Conservation

The Multi-Functional Road Maintenance Device embodies the nexus of innovation and sustainability, standing as a cornerstone in the quest for enhanced urban living and resilient infrastructure.

II.3.1 Urban Quality of Life Enhancement

The device's contribution to cleaner air is a testament to its role in enhancing the quality of life in urban settings. The visual appeal brought about by clean streets has a positive impact on the daily experiences and satisfaction of city dwellers.

II.3.2 Infrastructure Preservation

By preventing the premature deterioration of road surfaces, the device ensures that roads remain in optimal condition, safe for all users. The comfort and safety provided by well-maintained pathways foster a dynamic and engaging urban environment, conducive to an active and healthy lifestyle.

II.4 Schematic Diagram of Multi-Functional Road Maintenance Device: Visualizing Motion and Interactions

The schematic diagram illustrates several crucial aspects of the Multi-Functional Road Maintenance Device, including its various components and multiple functionalities. It visualizes the different motions of parts and the interactions between them, facilitating understanding of how the device operates and executes various maintenance tasks on urban roads.

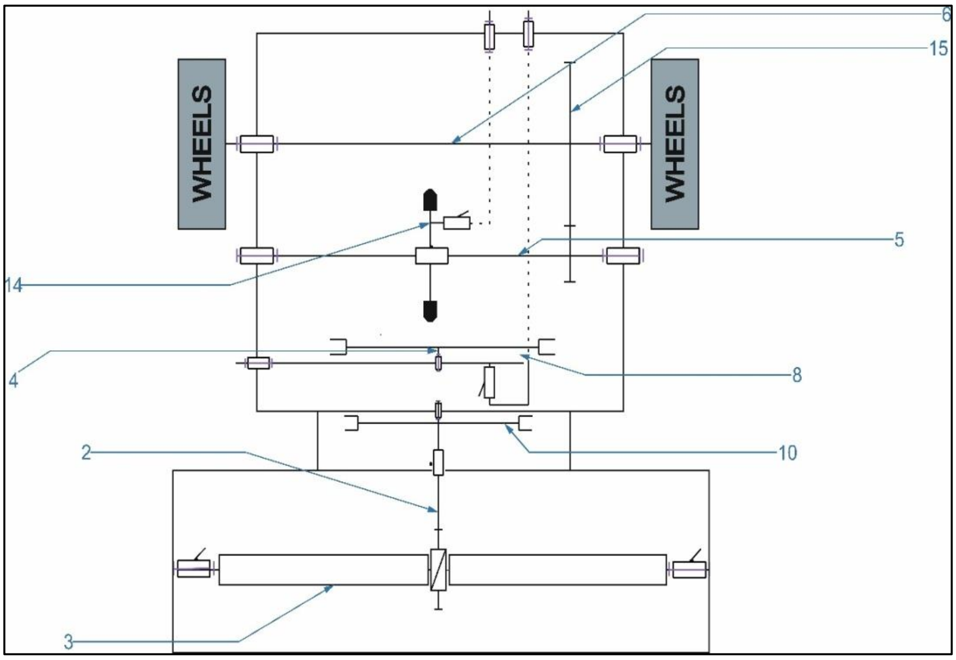


Figure II. 2 : The Top View of the Multi-Functional Road Maintenance Device.

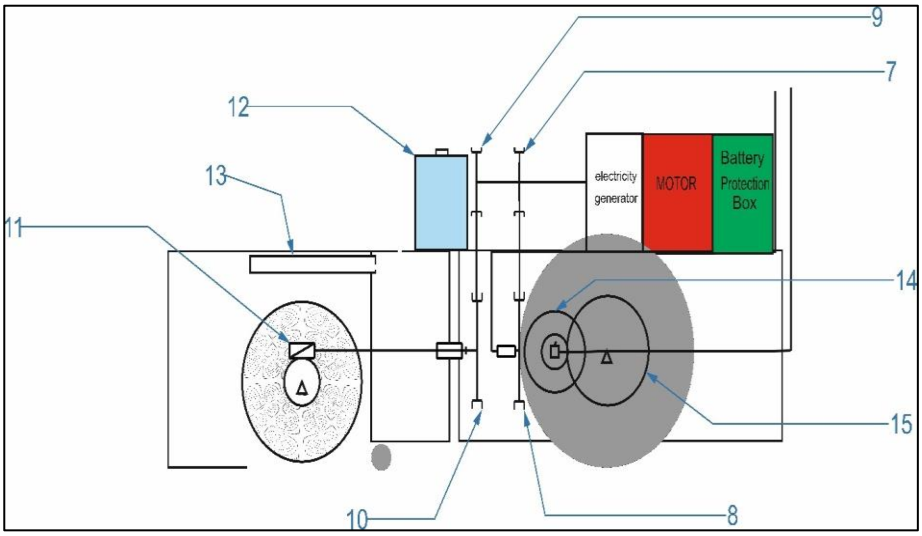


Figure II. 3 : The Side View of the Multi-Functional Road Maintenance Device.

II.5 Dynamic Analysis of Multi-Functional Road Maintenance Device Mechanisms and Control

- **Phase One: Starting the Rotation** Shaft **1** begins to rotate. This shaft contains two pulleys, pulley **7** and pulley **9**. Pulley **7** drives pulley **9**, which is connected to shaft **4**. Pulley **9** has a rough surface that rubs against pulley **14**, and this friction leads to the rotation of shaft **5**, which contains gears at one of its ends. These gears mesh with gears **15**, which turn shaft **6**, and in turn, rotate the wheels.
- **Phase Two: Movement Control and Stopping** The mechanism is characterized by its ability to stop, where a sliding connection in pulley **8** can be activated to distance it from friction pulley **14**, which stops the wheels. The speed of the wheel rotation can also be controlled by adjusting the position of pulley **14** relative to the centre of pulley **8**. To reverse the wheels, the direction of rotation of pulley **14** is reversed.
- **Phase Three: Sweeping Roads and Collecting Debris** When pulley **9** rotates, it turns pulley **10**, which is connected to shaft **2**, which in turn is connected to shaft **3** via a worm gear. Shaft **3** contains a sweeper that sweeps the roads and collects debris and trash in a removable box. There is also a small mechanism, **12** and **13**, for spraying water to avoid dust emitted from sweeping.
- **Phase Four: Soil Breaking and Grass Cutting** The sweeper can be replaced with special blades for cutting grass, and the grass is collected in the same box. The device can also be used for breaking up soil.

- **Phase Five: Recharging the Engine** The engine is rechargeable thanks to the electric generator connected to shaft **1**, ensuring the continuous operation of the device.

We will study the sweeping mechanism and the walking mechanism and analyse their components. We will focus on how the device's components interact to achieve high efficiency in the sweeping process, the sweeping and the integrated operation mechanism, as well as how to improve the walking mechanism to ensure smooth and efficient movement of the device.

II.6 Kinematic and Energetic Calculation

Before starting the calculations of the machine, we must first choose the engine that suits the machine. The following part summarizes the choice and the calculation of the puissance and efficiency which is necessary for the proper functioning of the machine.

II.6.1 Selection of the Electric Motor

Starting with the initial data, we calculate the power and rotational speed required for the drive motor. The geometric data will be derived from the standardized values available in electric motor manufacturers' catalogues[29]. The motor's necessary power, denoted as $\boldsymbol{P_m}$, is determined by taking into account the power at the reducer's output, referred to as $\boldsymbol{P_s}$

$\boldsymbol{P_w}$, along with the energy losses of the transmission components, which are indicated by their efficiency ratings [30].

Table II. 1 : data given

Parameter	Value
Motor Speed $\boldsymbol{N_M}$	1400 Rpm
Wheel Speed $\boldsymbol{N_W}$	10 Rpm
Sweeper Shaft Speed $\boldsymbol{N_s}$	140 Rpm
Sweeper Power $\boldsymbol{P_s}$	950 W
Wheel Power $\boldsymbol{P_w}$	1300 W

When there are two different power outputs for a system, such as a sweeper and a wheel in your case, and we want to ensure safety and reliability, we select the larger of the two powers as the basis for our calculations. To this, we add an additional 10% of that power to account for any unforeseen loads or inefficiencies. This sum is then divided by the total efficiency of the motor to determine the total power required for the motor [31, 32].

To calculate the total power required for the motor $\boldsymbol{P_m}$ when there are two power outputs, one for the sweeper $\boldsymbol{P_s}$ and one for the wheel $\boldsymbol{P_w}$, and to ensure safety, we select the larger power value. We then add an additional 10% to this value to account for safety margin.

The formula to calculate the motor power needed is:

$$P_M = \frac{P_w + (P_w \times 0.1)}{\eta_{tot}}$$

Where:$\boldsymbol{P_M}$ is the power of the motor to be selected.

The Larger Power is the greater value between $\boldsymbol{P_S}$ and $\boldsymbol{P_W}$.

$\boldsymbol{\eta_{tot}}$ represents the total efficiency of the motor.

With:

$$\boldsymbol{\eta_{tot} = \eta_{eng}^{a} . \eta_{r}^{b} . \eta_{c}^{c}}$$

$\boldsymbol{a}$: Number of contacts. $\boldsymbol{b}$: Number of bearings. $\boldsymbol{C}$: Number of grease contacts.

In ***Table II.1***, orders of magnitude for the efficiencies of various friction pairs present in a mechanical transmission are indicated. These efficiencies play a crucial role in determining the overall performance of the transmission system[29].

Table II. 2 :. Efficiency of some friction pairs [29].

Frictional couples	Efficiency
Cylidrical Gear	0,97…..0,99
Bearing Pair	0,99…..0,995
v-belt Transmission	0,94….0,97

To ensure safety, the power required for the drive motor should be calculated using the lower values of the ranges mentioned for each efficiency. Knowing the no-load running speed of the electric motor, the appropriate motor is selected from the manufacturers' catalogue. presents the main functional and d dimensional characteristics of some motors [29].

$$\boldsymbol{\eta_{tot} = \eta_{eng}^{3} . \eta_{r}^{13} . \eta_{c}^{2} \eta_{tot} = 0.707}$$

The larger power is for the sweeperP_S. Therefore, the calculation will be:

$$P_M = \frac{1300W + (1300W \times 0.1)}{0.707}$$

The larger power is for the sweeperP_S. Therefore, the calculation will be:

$$P_M = \frac{1430W}{0.707}$$

Now, you would need to insert the actual value of η_{tot} to find the required motor power P_M.

$$P_M = 2022.6308W$$

According to the specifications detailed in ***appendix N1***, an AmTecs motor of type ***AMAS-SR100L4*** with a power rating of ***2.2kW*** and a speed of ***1400 rpm*** has been selected. This motor complies with ***IEC 60034-1*** standards and is classified under efficiency class ***IE1***, making it ideal for our applications due to its ability to provide the necessary power while maintaining efficient energy consumption. The motor has an efficiency of **76%**, operates at a voltage of ***220V***and a frequency of ***50 Hz***. It is equipped with insulation ***class F*** nd has a sound pressure level of ***86 dB(A)***at 1 meter, and it comes with an ***IP55*** rating for dust and water protection.

II.6.2 Kinematic Calculation

The following flowchart details the kinematic and energy calculation steps of the multifunction maintenance device. Two separated parts of calculations will be carried out on the sweeping and the walking mechanism.

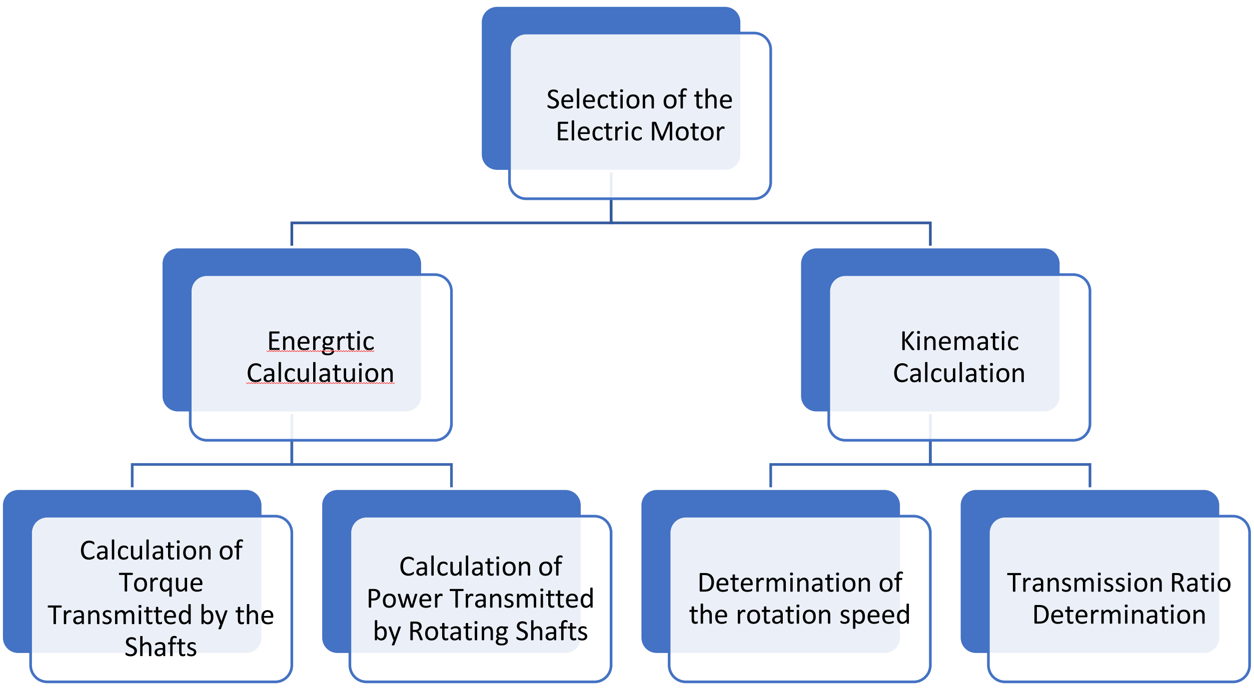


Figure II. 4 : Flowchart of Kinematic and Energetic Calculation steps.

II.6.2.1 Transmission Ratio Determination

To proceed with the calculations, we would typically use these speeds to determine the gear ratios required for the transmission system that connects the motor to the sweeper shaft and wheels [29]. The gear ratio can be calculated using the formula:

$$\text{total Ratio} = \frac{\text{Motor Speed}}{\text{Output Speed}}$$

The nominal values of the transmission ratios are standardized (see tableII.2).

Table II. 3 : Nominal Transmission Ratios[29].

Nominal transmission Ratios	1.00	1.60	1.80	2.00	3.15	3.55	4.00	4.50	6.30	7.1	10.5

Kinematic diagram of the sweeping mechanismwith numbering

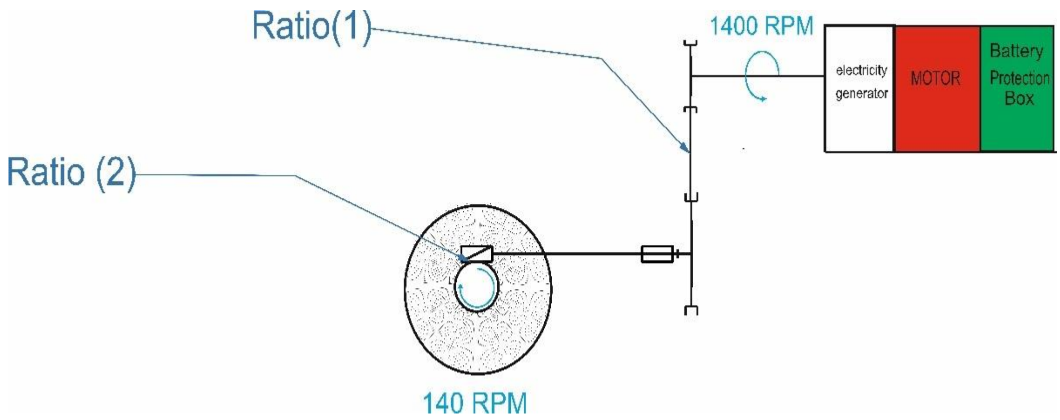


Figure II. 5 : sweeping mechanism.

For the sweeper shaft, the gear ratio would be

$$\text{total Ratio} = \frac{N_m}{N_s}$$

$$\text{total Ratio} = \frac{1400}{140} = 10$$

This means the motor’s speed needs to be reduced by a factor of **10** to drive the sweeper shaft at the desired speed.

After calculating the total ratio of the sweeper mechanism, which has been previously determined, we take into account the assumed ratio for the two pulleys connected by a belt, which we have assumed to be ratio (1)= 2. (***Table II.3***) This ratio is used to determine the worm gear ratio as follows

$$\text{ratio (2)} = \frac{\text{Total Ratio}}{\text{ratio (1)}} \quad , \quad \text{ratio}(2) = \frac{10}{2} = 5$$

This way, we can determine the necessary worm gear ratio to achieve the desired sweeper speed when connected to the motor through the pulley system and the worm gear.

Table II. 4: Nominal Transmission Worm Gear Ratio**[1]**.

Worm Gear Ratio	5	7.5	10	12.5	15	20	25	30	35	40	45	50	60

For our application, I have selected a gear ratio of **5**.

Kinematic diagram of the walking mechanism with numbering

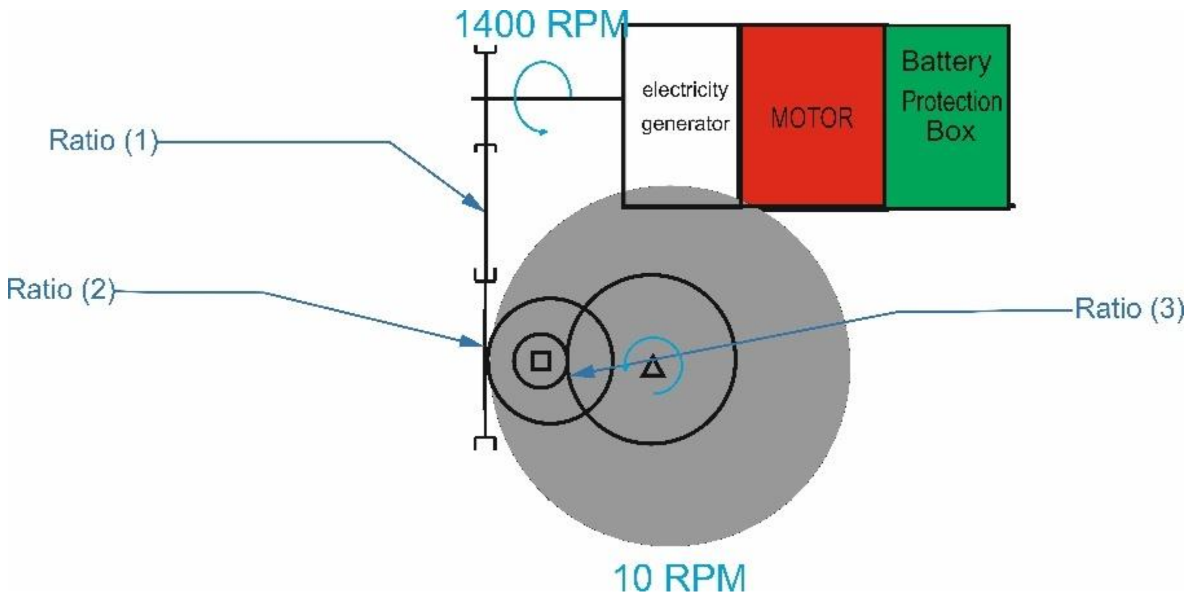


Figure II. 6 : walking mechanism.

Similarly, for the wheels, the gear ratio would be :

$$\text{Total Ratio} = \frac{N_m}{N_w} \quad , \quad \text{total Ratio} = \frac{1400}{10} = 140$$

This indicates a much larger reduction is needed to achieve the slow rotation speed required for the wheels. To calculate the variable gear ratio for the second connection, you can use the actual values from ***Table II.3***for the first and third gear connections, which are ratio (1)= 3.15 and 10.5 respectively, in the following formula:

$$\text{Variable Gear Ratio} = \frac{\text{Total Ratio}}{\text{Ratio (1)} \times \text{Ratio (3)}}$$

$$\text{Variable Gear Ratio} = \frac{140}{3.55 \times 10.5} = 3.75$$

We substitute the value 4 from ***Table II.3*** because it approximates the desired Ratio.

II.6.2.2 Determination of the rotation speed

The sweeping mechanism ratio speed

The equations corresponding to the above descriptions are as follows:

1. $N_1 = N_{motor}$=1400 rpm
2. $N_2 = \frac{N_1}{2}$
3. $N_3 = \frac{N_2}{5}$

The walking mechanism ratio speed

4. $N_4 = \frac{N_{motor}}{3.55}$
5. $N_5 = \frac{N_4}{4}$
6. $N_6 = \frac{N_5}{10.5}$

Table II. 5 : Shaft Speeds Table.

Shaft Number	**N1**	**N2**	**N3**	**N4**	**N5**	**N6**
Speed *rpm*	1400	700	140	394	98.59	9.38

II.6.3 Energetic calculations

II.6.3.1 Calculation of Power Transmitted by Rotating Shafts

In this section, we will focus on calculating the power transmitted by rotating shafts. The power transmitted through mechanical systems is crucial for understanding their efficiency and performance. The equation you've provided represents the relationship between the mechanical power output of a shaft $\boldsymbol{P}_{\mathbf{shaft}}$and the total efficiency $\boldsymbol{\eta}_i$ of a system to calculate the mechanical power input $\boldsymbol{P}$[29]: $P_i = P_m \times \eta_i$

Here's what each term represents:

- $\mathbf{P_m}$: Mechanical power input to the system in Watts.
- $\boldsymbol{P}_{\mathbf{shaft}}$: Mechanical power output from the shaft in Watts.
- $\boldsymbol{\eta}_i$: Total efficiency of the system (dimensionless) to efficiency from the engine shaft

➢ **Analysis of Rotating Shafts Efficiency**

In this section, we will calculate and analyze the efficiency of each rotating shaft in the mechanical system. By evaluating the performance of each component

a) **The sweeping mechanism**

- **Shaft 1**: $\eta_1 = \eta_{eng}^0 \times \eta_r^2 \times \eta_c^0$, $\eta_1 = 0.9801$
- **Shaft 2**: $\eta_2 = \eta_{eng}^0 \times \eta_r^5 \times \eta_c^1$, $\eta_2 = 0.8939$
- **Shaft 3**: $\eta_3 = \eta_{eng}^1 \times \eta_r^7 \times \eta_c^1$, $\eta_3 = 0.8498$

b) **The walking mechanism ratio**

- **Shaft 4**: $\eta_4 = \eta_{eng}^0 \times \eta_r^3 \times \eta_c^1$, $\eta_4 = 0.912$
- **Shaft 5**: $\eta_5 = \eta_{eng}^1 \times \eta_r^5 \times \eta_c^2$, $\eta_5 = 0.815$
- **Shaft 6**: $\eta_6 = \eta_{eng}^2 \times \eta_r^7 \times \eta_c^2$, $\eta_6 = 0.7748$

In the mechanical linkage between shafts 4 and 5, it is prudent to consider a safety factor to account for potential inefficiencies that may not be apparent in the initial calculations. This adjustment is made to mitigate the risk of overloading the system and to ensure reliable operation under varying conditions[33].

c) **The sweeping mechanism**

- $P_1 = P_m \times \eta_1$, $P_1 = 2156.22\ Watts$
- $P_2 = P_m \times \eta_2$, $P_2 = 1966.58$ Watts
- $P_3 = P_m \times \eta_3$, $P_3 = 1869.5$ Watts

d) **The walking mechanism ratio**

- $P_4 = P_m \times \eta_4$, $P_4 = 2006.4\ Watts$

The adjusted efficiency η_{5adj} can be calculated as follows:

η_{5adj}=0.815×0.60 , η_{5adj}=0.489

This adjusted efficiency must then be used in subsequent calculations involving shaft 5 to ensure that the system is designed with an adequate safety margin.

η_{6adj}=0.7748×0.60 , η_{6adj}=0.46488

- $P_5 = P_m \times \eta_{5adj}$, $P_5 = 1075.8\ Wtts$
- $P_6 = P_m \times \eta_{6adj}$, $P_6 = 1022.736\ Watts$

II.6.3.2 Calculation of Torque Transmitted by the Shafts

In this section, we focus on calculating the torque transmitted by rotating shafts. Torque is a measure of the rotational force of an object and is a crucial factor in determining the mechanical system's capacity to perform work [29].

The torque **T** can be calculated using the mechanical power input $\boldsymbol{P_i}$ and the angular velocity $\boldsymbol{\omega_i}$ of the shaft according to the equation:

$$T_i = \frac{P_i}{\omega_{i_i}}$$

Where:

- $\boldsymbol{T_i}$: Torque transmitted by the shaft in *N.m*.
- $\boldsymbol{P_i}$: Mechanical power input to the shaft (i) in *W*.
- $\boldsymbol{\omega_i}$: Angular velocity of the shaft (i) in *Rpm*.

The angular velocityωi can be calculated from the rotation speed $\boldsymbol{N_i}$ using the relationship:

$$\omega_i = \frac{2\pi N_i}{60}$$

e) **The sweeping mechanism**

- $T_1 = \frac{P_1}{\omega_1}$, $T_1 = 14.71\ Nm$
- $T_2 = \frac{P_2}{\omega_2}$, $T_2 = 26.83 Nm$

- $T_3 = \frac{P_3}{\omega_3}$, T_3 =127.52 Nm

f) **The walking mechanism ratio**

- $T_4 = \frac{P_4}{\omega_4}$, T_4 =48.63Nm
- $T_5 = \frac{P_5}{\omega_5}$, T_5 =104.20Nm
- $T_6 = \frac{P_6}{\omega_6}$, T_6 =1041.20Nm

II.7 Calculation of Pulleys Transmission

a) The sweeping mechanism

On the other hand, for pulleys 9 and 10, wrapped construction belts have been selected because they offer greater flexibility in dealing with variable loads and different operating conditions. The outer wrapped layer protects the belt from external factors and increases its service life, while also allowing for some slippage that can be beneficial in certain situations, such as preventing the sudden stoppage of the driven shaft.

By using these two types of belts, a balance between precision and flexibility in the transmission system can be achieved, enhancing the efficiency of the device and ensuring its reliable operation under various conditions. This design takes into account the need for efficient power transfer and handling heavy loads without compromising on flexibility and the ability to adapt to changing conditions.

Figure II. 7 :wrapped construction belts and driven sprockets [1].

The steps for the calculation are as follows:

- **Choice of Belt Type**

The belt type is determined using the monogram based on the motor shaft power and rotational speed

For belt profiles close to the limits between domains on the monogram, it is recommended to choose the belt type that falls below the oblique line.

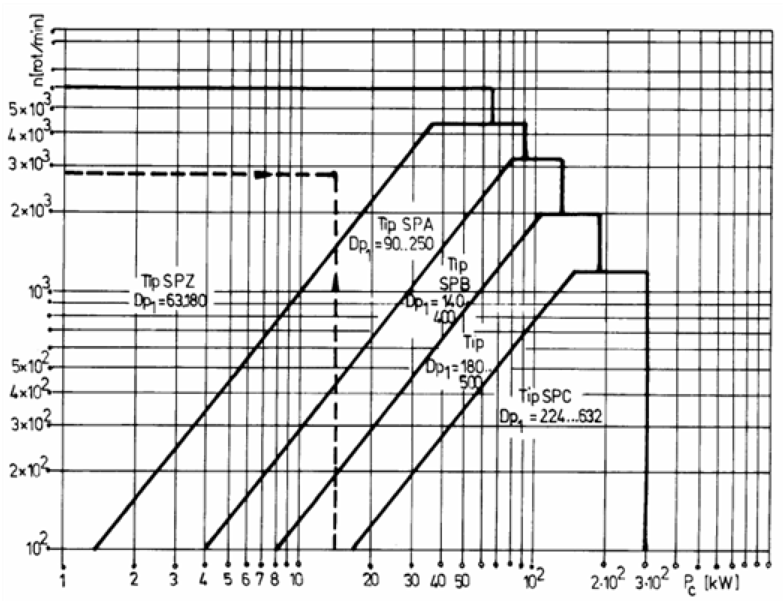


Figure II. 8 : Monogram for the selection of narrow V-belts [29].

Given that the motor power $\boldsymbol{P_m}$ is ***2200W*** and the motor speed $\boldsymbol{N_m}$ is ***1400 rpm***, the appropriate belt type would be ***SP.***

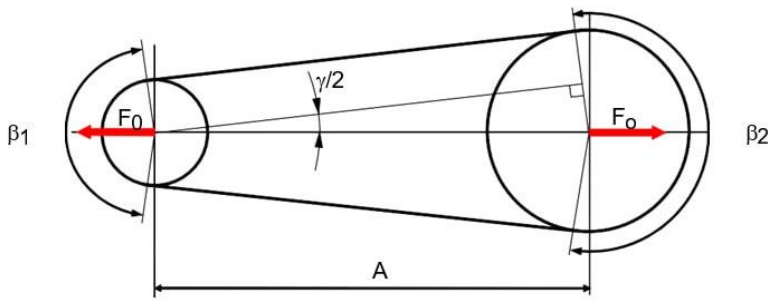


Figure II. 9 : Belt/chain drive configuration[29].

➢ **Choice of Smallest Pulley Diameter**

The diameter of the smallest pulley is chosen according to the belt type, following the standards' recommendations.

Table II. 6 : Nominal Wire Rope Diameter [1, 29].

Nominal wire rope diameter mm	Nominal wire rope diameter mm	Nominal wire rope diameter mm	Nominal wire rope diameter mm
52	87	103	250
58	90	109	280
60	96	115	315
64	103	122	400
67	109	128	450
71	115	140	500
74	122	160	560
77	128	180	630
80	90	200	710

- **Calculation of the Diameter of the Largest Pulley D_{P2}**
  - To calculate the diameter of the largest pulley, use the gear ratio i_T and the diameter of the smaller pulleyD_{P1} [29]:

$$D_{P2} = i_C \times D_{P1}$$

- **Determining the Distance Between the Axes A**
  - The distance between the pulleys' axes, denoted as **A** , should be within a specific range calculated using the diameters of the pulleys[29]:

$$0.7 \times (D_{P1} + D_{P2}) \leq A \leq 2 \times (D_{P1} + D_{P2})$$

- **Calculating the Angle Between the Belt Branches γ**
  - The angle $\boldsymbol{\gamma}$ between the belt branches is determined by the difference in the diameters of the two pulleys and the center distance [29] A :

$$\gamma = 2\arcsin(\frac{D_{P2} - D_{P1}}{2A})$$

- **Winding Angles on the Pulleys $\beta_1$and β_2**
  - The winding angles on the smallest and largest pulleys are calculated as follows:

- For the smallest pulley $\boldsymbol{\beta_1}$[29]:

$$\beta_1 = 180° - \gamma$$

- For the largest pulley$\boldsymbol{\beta_2}$[29]:

$$\beta_2 = 180° + \gamma$$

- **Length of the Belt in Free State** $\boldsymbol{L_p}$

The length of the belt in its free state is approximated using the centre distance A , the diameters of the pulleys, and the winding angles[29]:

$$L_p = 2Acos\frac{\gamma}{2} + \frac{\pi}{360} \times (\beta_1 \times D_{P1} + \beta_2 \times D_{P2}) \approx 2A + \frac{\pi(D_{P1} + D_{P2})}{2} + \frac{(D_{P2} - D_{P1})^2}{4A}$$

- **Peripheral Speed of the Belt** $\boldsymbol{V}$

The peripheral speed of the belt is calculated using the diameter of the smaller pulley

D_{p1} and the motor's rotation speed N_m:

$$v = \frac{\pi D_{P1} \times N_m}{60000}(m/s)$$

It is recommended that the peripheral speed of the V-belt does not exceed 30 *m/s* for narrow V-belts [29].

- **The Frequency of Direction Changes of the Belts is Calculated With the Relation:**

$$f = 10^3 \times x \times \frac{V}{L_p} H)$$

Where :

- ***x***– Number of pulleys $x = 2$.
- ***v*** – Peripheral speed of the belt, in m/s.
- $\boldsymbol{L_p}$- Free-state length of the belt (chosen standardized value), in *mm*.

It is recommended to avoid exceeding a frequency of direction changes of 40 ***Hz*** for woven belts and 80 ***Hz*** for belts with a central wire [29].

- **The Transmitted Peripheral Force:**

$$F = 10^3 \times \frac{P_C}{v} \qquad N$$

For the purposes of the table, the diameter of the largest pulley $\boldsymbol{D_{p2}}$ was taken to be ***103 mm.***

Table II. 7 : The sweeping Chain mechanism.

Parameter	Sweeper Chain
Diameter of the largest pulley $\boldsymbol{D_{p2}}$	103.00***mm***
Distance between the axes $\boldsymbol{A}$	202.50***mm***
Angle between the belt branches$\boldsymbol{\gamma}$	14.18°
Winding angle on the smallest pulley $\boldsymbol{\beta_1}$	165.82°
Winding angle on the largest pulley $\boldsymbol{\beta_2}$	194.18°
Length of the belt in free state $\boldsymbol{L_p}$	646.80 ***mm***
Peripheral speed of the belt $\boldsymbol{V}$	3.93 ***m/s***
Frequency of direction changes of the belts $\boldsymbol{f}$	12.14***Hz***
Transmitted peripheral force $\boldsymbol{F}$	381.97***N***

b) The walking mechanism

To calculate the transmission of motion by pulleys, standardized equations are used that take into account the power on the drive motor shaft in kilowatts, the rotational speed in rpm, and the belt transmission ratio [2]. In designing the belt transmission system for a road maintenance device, choosing the right type of belt is critical to ensure effective and reliable performance. I have chosen synchronous belts for pulleys 7 and 8 (***Figure II.10***) because of their ability to handle heavy loads and their high efficiency. Most importantly, their non-slippage characteristic guarantees precise and synchronized motion transfer. These features make them ideal for applications that require exact synchronization, such as operating the wheels on a road maintenance device.

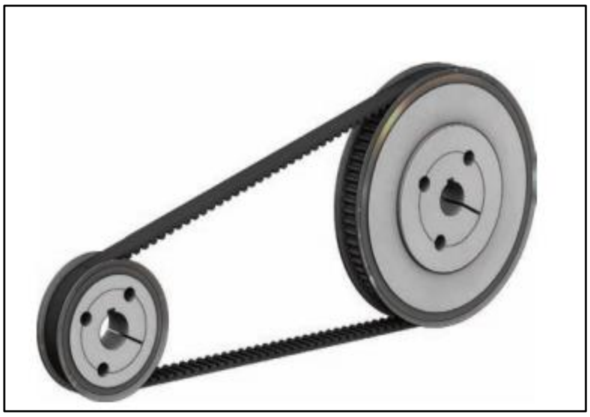


Figure II. 10 : Synchronous belt on driving and driven Timing pulley [1].

- **Calculating Design Power:**

Service Factor (S_F): Refer to a table (Appendix 2) to determine the S_F based on operating conditions.

The Design Power $\boldsymbol{P_{\mathrm{des}}}$ Calculation is obtained by the following equation[1]:

$$P_{\mathrm{des}} = P_{\mathrm{rated}} \times S_f$$

- **Determining Belt Pitch and Size:**

Belt Pitch: The diagram (***Figure II.11***) was used to define the required pitch for the belt by intersecting the design power and motor rpm.

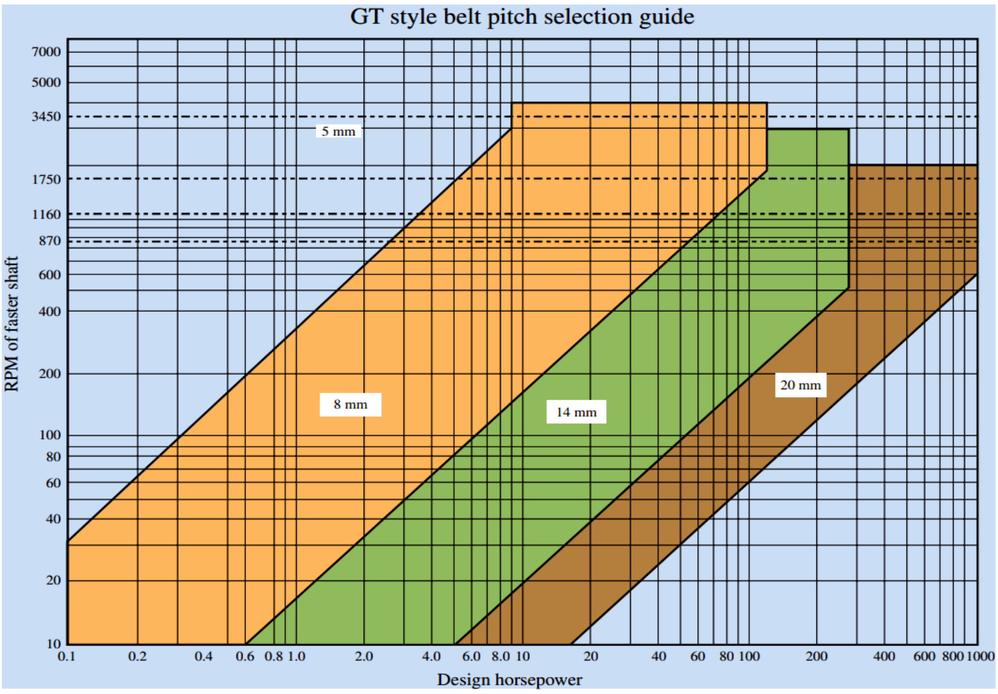


Figure II. 11 : Belt pitch selection guide for GT style belts [1].

- **Selection of Sprockets:**
  - **Candidate Combinations:** List possible combinations of driving and driven sprockets that achieve the desired ***VR***.
  - **Elimination Based on Constraints:** Remove combinations that do not meet space limitations and shaft diameter requirements, referencing tables for bushing style and maximum bore sizes.
- **Belt Length and Centre Distance:**

  Belt Length: Check a table (Appendix 3)
  - To find the belt length that corresponds to the selected sprocket sizes[34].
  - **Centre Distance:** Ensure the centre distance is within the required range.
- **Belt Width Selection:**

  Rated Power for Belt Width: Refer to a table (Appendix 4) to find the rated power for the selected belt width.
- **Belt Length Correction Factor:** Apply a correction factor from another table (Appendix 5) to adjust the rated power.
- **Belt Speed Calculation:**
  - **Belt Speed Formula :**the belt speed is obtained by using the following equation :

$$V_{belt} = \frac{P}{D_1} \times \pi \times \mathrm{v1}$$

To calculate the belt speed and ensure it is within acceptable limits.

- **Final Design Details:**
  - **Sprockets and Bushings:** Specify the final sprockets and bushings, including part numbers and sizes.
  - **Belt Specification:** Provide the final belt type and size.
  - **Given data :**

To specify a synchronous belt drive system for a motor operating under the conditions you've provided, follow these steps:

- **Motor Specifications:**

Speed: 1400 *rpm*

Rated Power: *2.2 kW*, which is equivalent to 2.95 *hp*

- **Service Factor S_F:**

  Based on *Appendix 2*, for the given operating conditions, the S_F is 1.7.

- **Design Power P_{des}:**

  Calculate the design power using the formula:

$$P_{des} = P_{rated} \times SF = 2.95\ hp \times 1.7 = 5.01\ hp$$

- **Belt Pitch Selection:**

  Refer to ***Figure II.11*** to determine the required pitch for the belt. For a motor with 5.01 *hp* and 1400 *rpm*, the 8 *mm*, 8*MGT* belt system is suitable.

- **Velocity Ratio V_R:**

  The V_R is determined to be 3.55.

- **Sprocket Combination Selection:**

  Consult for possible sprocket combinations that achieve a V_R of 3.55. The options are:

  25 teeth driving sprocket and 90 teeth driven sprocket

- **Belt length and centre distance:**

  From *Appendix 3*, the **1040-8*MGT*** belt with a pitch length ***(P.L.)*** of 40.945 inches and 130 teeth will have a centre distance of 10.93 *inches* 277.6 *mm*, which fits within the required range.

Table II. 8 : Timing pulley Synchronous Belt Drive Design Specifications.

Description	**Result**
Motor Speed and Rated Power	1400 rpm, 2.95 hp (2.2 kW)
Pitch Diameter DP1	$2.506\ inches = 63.65\ mm$
Pitch Diameter DP2	9.023 inches = 229.18 mm

Sprocket Combination Selection (Z1)	25 teeth
Sprocket Combination Selection (Z2)	90 teeth
Service Factor S_F	1.7
Design Power P_{des}	$P_{des} = 2.95\text{h}_p \times 1.7 = 5.01\text{h}_p$
Belt Pitch Selection	8 mm, 8MGT belt system
Velocity Ratio V_R	3.55
Belt Length	1040-8MGT belt, P.L. 40.945 inches,
Center Distance	10.93 inches (277.6 mm) (Appendix3)

II.8 Material Selection for Gear Wheels

Cylindrical (straight-toothed or helical) and bevel gears used in reducers are heavily stressed machine components. The primary stresses, for which resistance calculations are performed, include bending at the tooth root unit stress, $\boldsymbol{\sigma_f}$ and Hertzian pressure at the flank contact unit stress, $\boldsymbol{\sigma_H}$. These stresses vary periodically over time. Consequently, for gear design, it is essential to know the general mechanical properties of the materials used (ultimate tensile strength, yield strength, hardness, etc.) as well as the fatigue resistance values for the mentioned stresses $\boldsymbol{\sigma_{f\,lim}}$and $\boldsymbol{\sigma_{H\,\mathrm{lim}}}$Resistance values are determined through tests conducted on specimens (gears) using specialized testing equipment.

Gears used in machine construction can be made from rolled, forged, or cast steels, as well as from cast iron, non-ferrous alloys (brass, copper, aluminium alloys, etc.), and sometimes even plastic. Rolled or forged steels are commonly used in the construction of cylindrical and bevel gears for speed reducers transmitting significant power.

The steels used in gear construction can be divided into two groups based on the heat or thermochemical treatment they undergo:

- Enhanced or normalized steels, where the Brunel hardness of the tooth flank after treatment is less than 3500 N/mm^2.

- Hardened steels subjected to heat treatments (quenching after flame heating) or thermochemical treatments (carburizing, intruding). Their Brunel hardness after treatment exceeds 3500 N/mm^2.

Appendix 6 presents the main types of steels used in the execution of cylindrical and bevel gears for speed reducers, along with the necessary mechanical characteristics for gear design.

In selecting materials for gear wheels, we prioritize a composition that offers a robust core capable of withstanding shocks while also providing a durable surface to minimize wear. Consequently, we opt for ***20CrMo5 steel*** as outlined in *Appendix 4* . This material strikes an optimal balance between resilience and hardness, ensuring the gears can endure the rigors of operation without succumbing to premature wear [29].

II.9 Worm Gear Calculations for sweeping mechanism

The meshing requirement of a worm and worm gear set hinges on the equivalence between the axial pitch of the worm and the circular pitch of the worm gear [35]. Below are the key concepts and calculations related to the pitches, lead, and velocity ratios for worm and worm gear sets, extracted from the provided information:

- Axial Pitch and Circular Pitch
  - Axial Pitch $\boldsymbol{p_x}$: The distance measured axially on the pitch cylinder from a point on one worm thread to the corresponding point on the next adjacent thread [36].
  - Circular Pitch $\boldsymbol{p}$: The distance measured along the pitch circle of the wormgear from a point on one tooth to the corresponding point on the next adjacent tooth[36]. It can be calculated using the equation[1]:

$$P = \frac{\pi \times D_G}{N_G}$$

where D_G is the pitch diameter of the gear, and $N_G$is the number of teeth in the gear.

Diametral Pitch

$$P_d = \frac{N_G}{D_G}$$

The conversion from diametral pitch to circular pitch

- Module $\boldsymbol{m}$: For metric systems, the module is defined as[1]:

$$m = \frac{D}{N}$$

where D is the pitch diameter and N is the number of teeth. The circular pitch in terms of module is[1]:

- Lead $\boldsymbol{L}$

Lead $\boldsymbol{L}$: The axial distance a point on the worm moves in one revolution. For a worm with N_W threads[1]:

$$L = N_W \times P_x$$

- Lead Angle $\boldsymbol{\lambda}$: The angle between the tangent to the worm thread and the line perpendicular to the worm's axis. It can be calculated using[1]:

$$tan\,\lambda = \frac{L}{\pi \times D_W}$$

where $\boldsymbol{D_W}$ is the pitch diameter of the worm.

- Pitch Line Speed $\boldsymbol{V_t}$
  - For the worm [1]:

$$v_{tW} = \frac{\pi \times D_W \times n_W}{60000} m/s$$

where n_W is the rotational speed of the worm.

  - For the worm gear [1]:

$$v_{tG} = \frac{\pi \times D_G \times n_G}{60000} m/s$$

Face width $\boldsymbol{F_G}$ of the gear is crucial for ensuring that the gear can withstand the forces it will encounter during operation without excessive stress or wear. The face width is often recommended to be approximately 2.0 times the circular pitch, which in a diametral pitch system translates to

$$F_G = \frac{6}{P_d}$$

where $P_d$is the diametral pitch

Typical commercially available worm gear drives have ratios such as the following table lists the standard lead angles:

Table II. 9 : Lead Angles for Gear Systems [1].

Lead Angle λ (°)	0.5	1	1.5	2	3	4	5	7	9	11	14	17	21	25	30

The pressure angle

influences the shape of the gear teeth and affects how the worm and worm gear mesh together. It's important for determining the strength and efficiency of the gear set.

Table II. 10 : Pressure Angles for Worm Gear Lead Angles

Lead Angle Range (°)	Pressure Angle (°)
Down to 30	20
30 to 45	25

Given Data

- Velocity ratio V_R = 5
- Speed of shaft 2 worm speed, n_W = 700 *rpm*
- Speed of shaft 3 gear speed, n_G = 140 *rpm*
- Number of teeth on the worm gear N_G = 20
- Pitch diameter of the worm gearD_G = 70 *mm*
- Pitch diameter of the worm D_W = 40 *mm*

➢ **Result**

Table II. 11 : Worm Gear System Parameters.

Parameter	Value
Velocity ratio V_R	5
Speed of worm speed, n_W	700***rpm***
Speed of worm gear speed, n_G	140 ***rpm***

Number of teeth on the worm gear $\boldsymbol{N_G}$	20
Pitch diameter of the worm gear$\boldsymbol{D_G}$	70 $\boldsymbol{mm}$
Pitch diameter of the worm $\boldsymbol{D_W}$	40 $\boldsymbol{mm}$
Circular pitch $\boldsymbol{P}$	11 $\boldsymbol{mm}$
DiametralPitch$\boldsymbol{P_d}$	0.28$\boldsymbol{mm}$
Module $\boldsymbol{m}$	3.5
Lead $\boldsymbol{L}$	44 $\boldsymbol{mm}$
Axial pitch $\boldsymbol{P_x}$	11 $\boldsymbol{mm}$
Number of threads in the worm $\boldsymbol{N_W}$	4.00
Lead angle $\boldsymbol{\lambda}$	19.29 °
Selected Lead Angle*degrees*	21°
The pressure angle	20°
face width$\boldsymbol{F_G}$	21.42 *mm*

II.10 Preliminary Sizing of Gears for the Walking Mechanism

II.10.1 Preliminary Sizing of an External Cylindrical Gear with Straight Teeth for the Walking Mechanism

During the preliminary sizing of an external cylindrical gear with straight teeth, we determine the centre distance a, the module m, the number of teeth of both gears, and the tooth offsets in the case of corrected gear teeth[30].

The steps of the calculation are as follows:

$$a_{12} = (i_w + 1)\sqrt[3]{\frac{K_A.K_V.K_{HB}.C_i}{2.i_{12}.T_a}\left(\frac{Z_M.Z_H.Z_s}{\frac{\sigma_{H\lim}}{S_H}.K_{HN}.Z_R.Z_W}\right)^2}$$

Reduction Ratio i_{12}: The reduction ratio between the small pinion and the large wheel is i_{12}= 10.5.

Factor of External Dynamic Load $\boldsymbol{K_A}$: Given K_A = 1[29].

Internal Dynamic Factor $\boldsymbol{K_V}$: Given K_V = 1[29].

Factor of Longitudinal Load Distribution for Hertzian Stress $\boldsymbol{K_{HB}}$: Given K_{HB} = 1.15[29]

Torsional Torque on the Pinion Shaft Given C_5= 104200 N·mm using C_5 instead.

Width Coefficient $\boldsymbol{T_a}$: Given T_a = 0.5[29].

Material Factor $\boldsymbol{Z_M}$: Given Z_M = 271[29].

Rolling Point Factor $\boldsymbol{Z_H}$: Given Z_H= 1[29].

Contact Length Factor $\boldsymbol{Z_s}$: Given Z_s= 1[29].

Unit Limit Stress for Hertzian Stress $\boldsymbol{\sigma_{H\,\lim}}$: Given $\sigma_{H\,\lim}$= 1606 *N/mm²*.

Safety Factor Against Hertzian Stress $\boldsymbol{S_H}$: Given S_H= 2.

Factor of the Number of Cycles of Stress for Hertzian Stress $\boldsymbol{K_{HN}}$: Given K_{HN} = 1.4[29].

Roughness Factor $\boldsymbol{Z_R}$: GivenZ_R= 1.1[29].

Factor of the Ratio of Flank Hardness $\boldsymbol{Z_W}$: Given Z_W= 1.1[29].

The calculated distance between the axes $\boldsymbol{a_{12}}$ is approximately: $\boldsymbol{a_{12}} = 88.3\ mm$

To determine the module M for the preliminary sizing of gears, you can use the following formula:

$$M_{12} = 1{,}28.\sqrt[3]{\frac{C_1.K_c.K_v}{Z_P.T_m.i_{12}..y_f.\frac{\lfloor \sigma f \rfloor}{S_f}}}$$

Where each symbol represents the following:

- $\boldsymbol{M_{12}}$is the module of the gear pair.
- $\boldsymbol{C_1}$ is the torsional torque on the pinion shaft. Let's assume C_1= 104200 *N·m.*
- $\boldsymbol{K_c}$is the concentration coefficient. We'll assume K_c = 1.3 [29].
- $\boldsymbol{K_v}$ is the dynamic load coefficient. We'll assume K_v = 1.0 [29].
- $\boldsymbol{Z_P}$ is the number of pinion teeth. Let's assume Z_P = 6 .
- $\boldsymbol{T_m}$ is the relative width of the wheels. We'll assume T_m = 8 [29].
- $\boldsymbol{y_f}$ Shape factor $y_f = 0.33$ for preliminary sizing/design[29].

- $\boldsymbol{S_f}$ is the safety factor for fatigue stress at the tooth root. Let's assume $Sf = 2$.
- $[\boldsymbol{\sigma_f}]$ is the allowable material stress limit for 20CrMo5 steel(Appendix 6). We'll assume σ_f = 360 *N/mm²*.

The calculated module $\boldsymbol{M_{12}}$ is approximately: 2.11

The modulus $\boldsymbol{M_{12}}$ is approximately: 2.5

II.10.2 Geometric Elements of straight Cylindrical Gears for the Walking Mechanism

In the study of straight cylindrical gears[17], it is imperative to consider the following geometric elements :

- **Pitch Diameter** d_p

The pitch diameter is determined by the formula[1] :

$$d_p = \pi \times m$$

where:

d_p represents the pitch diameter ,m denotes the module of the gear.

- **Addendum** h_a

The addendum height is equivalent to the module[2]:

$$h_a = m$$

Where : $\boldsymbol{h_a}$ is the addendum height.

- **Dedendum** h_f**:**

The dedendum is calculated as[1]: $h_f = 1.25 \times m$

Where: h_f is the dedendum.

- **Tooth Thickness** ***s*** **:**

Tooth thickness is given by[1]:

$$s = 2 \times \pi \times m$$

where **s** is the tooth thickness.

- **Tooth Width b :**

The tooth width can be expressed as[2]: $b = k \times m$

where:$\boldsymbol{b}$ is the tooth width and k=7 is a thickness factor.

- **Tooth Height h :**

Tooth height is the sum of addendum and dedendum of gear [1]: $h = h_a + h_f$

where:$\boldsymbol{h}$ is the tooth height, $\boldsymbol{h_a}$ is the addendum height, $\boldsymbol{h_f}$ is the fillet height.

- **Pitch Diameter of Gear j d_{pj} :**

The pitch diameter of gear j is calculated by[1]: $\boldsymbol{d_{pj}} = m \times \boldsymbol{z_j}$

with:$\boldsymbol{d_{pj}}$is the pitch diameter of gear j,**m** is the module, $\boldsymbol{z_j}$ is the number of teeth on gear $\boldsymbol{j}$.

- **Head Diameter d_a :**

The head diameter is determined by[1]:

$\boldsymbol{d_a} = \boldsymbol{d_{pj}} + 2m$ with $\boldsymbol{d_a}$ is the head diameter.

- **Root Diameter of Gear j d_{fj} :**

The root diameter of gear $\boldsymbol{j}$ is[1]: $d_{fj} = d_{pj} - 2.5m$

where:$\boldsymbol{d_{fj}}$ is the root diameter of gear j, $\boldsymbol{d_{pj}}$ is the pitch diameter of gear j.

- **Base Diameter d_b:**

The base diameter is calculated using[1]:

$$d_b = d_j \cdot cos\alpha$$

Where : $\boldsymbol{d_b}$ is the base diameter. , $\boldsymbol{d_j}$ is the diameter of gear j. ,$\boldsymbol{\alpha}$ is the pressure angle.

The executed code provided the following results:

Table II. 12 : Gear Dimensional Specifications.

Gear Type	Number of Teeth	Pitch Diameterd_p	Head Diameterd_a	RootDiameter d_f	Addendum h_a	Total HeightTooth h	ToothWidth b	ToothThickness s	Base Diameter d_b
Gear of shaft 5	6	15.00 *mm*	20.00 *mm*	8.75 *mm*	2.50 *mm*	5.63 *mm*	17.50 mm	3.93 mm	14.10 *mm*

Gear of shaft 6	63	157.50 *mm*	162.50 *mm*	151.25 *mm*	2.50 *mm*	5.63 mm	17.50 *mm*	3.93 *mm*	148.00 *mm*

II.11 Conclusion

In conclusion, this chapter has systematically explored the multifaceted aspects of urban environment enhancement, with a particular focus on strengthening urban infrastructure. Through the prevention of road and street deterioration and the safeguarding of pedestrians and drivers, we have underscored the integral role of innovative devices in urban enhancement and infrastructure conservation. The schematic diagrams, dynamic analyses, and kinematic and energetic calculations presented herein not only elucidate the mechanisms and control of the road cleaning device but also highlight the importance of meticulous design and selection of its components. The calculations for V-belt transmission, material selection for gear wheels, and the preliminary sizing of gears with helical teeth are critical to ensuring the efficiency and longevity of such devices. As we move forward, the insights gained from this chapter will serve as a cornerstone for the development of sustainable urban environments that prioritize quality of life and infrastructure preservation. The journey towards urban enhancement is on-going, and the technologies and methodologies discussed here will undoubtedly contribute to the evolution of smarter, cleaner, and safer cities.

Furthermore, the forthcoming Chapter III will delve into the rigorous study of torsional forces applied to shafts and the intricate calculations of power. This analysis is pivotal for understanding the mechanical integrity and operational efficacy of the shafts within the urban infrastructure enhancement devices. By examining the applied torque and its implications on the structural performance, we will gain deeper insights into the design requirements and optimization strategies necessary for developing robust and reliable urban enhancement solutions.

Chapter III

Static Analysis and Optimization of Multi-Functional Road Maintenance Device Components

III.1 Introduction

In this chapter, we delve into the meticulous calculation of various forces acting on the mechanical components of a street sweeping device. Precise force calculations are crucial for ensuring the device operates efficiently and reliably under expected load conditions. This analysis encompasses tangential, radial, and normal forces, providing a comprehensive understanding of the mechanical interactions and stresses within the system. By methodically calculating these forces, we aim to optimize the design for durability and performance, ensuring the street sweeper meets its operational requirements effectively. This chapter is structured into sections that detail the calculation of forces on different components, including the worm gear forces in the sweeping mechanism and the spur gear forces in the walking mechanism, followed by the determination of shaft dimensions and the selection of keys and bearings for enhanced durability and performance.

III.2 Calculation of Forces F

In this section, the detailed calculation of various acting forces has been carried out. The calculations of accurate forces are essential to ensure the device operates efficiently and reliably under the expected load conditions. This analysis will cover tangential, radial, and normal forces, providing a comprehensive understanding of the mechanical interactions and stresses within the system. By methodically calculating these forces, we can optimize the design for durability and performance, ensuring the street sweeper meets its operational requirements effectively [1].

III.2.1 Calculation of Worm gear Forces for the sweeping mechanism

In the context of worm gear sets, the force system acting on the worm and worm gear is usually considered to be made up of three perpendicular components:

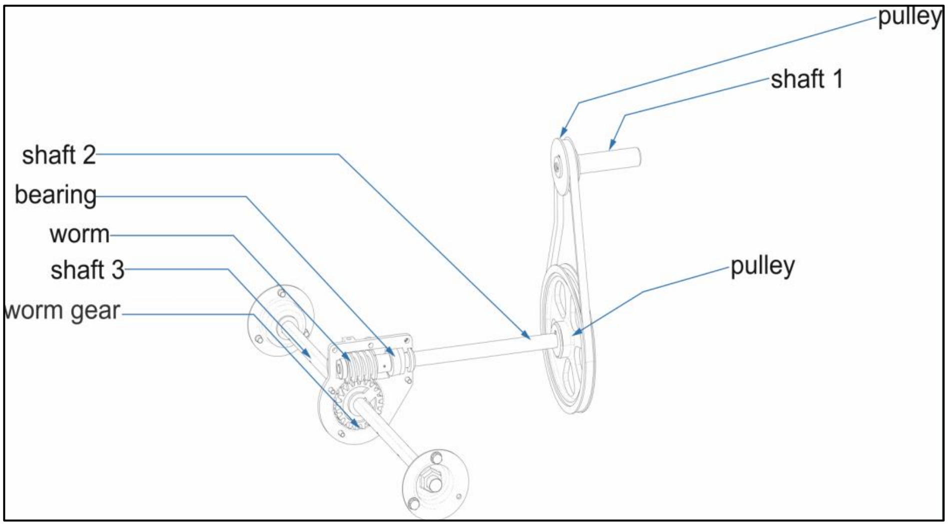


Figure III. 1 : the sweeping mechanism

the tangential force, the radial force, and the axial force. This approach is similar to the analysis of forces in helical and bevel gears. Calculating and understanding these forces is crucial for the design and operational efficiency of the mechanical system of a street cleaning device [37].

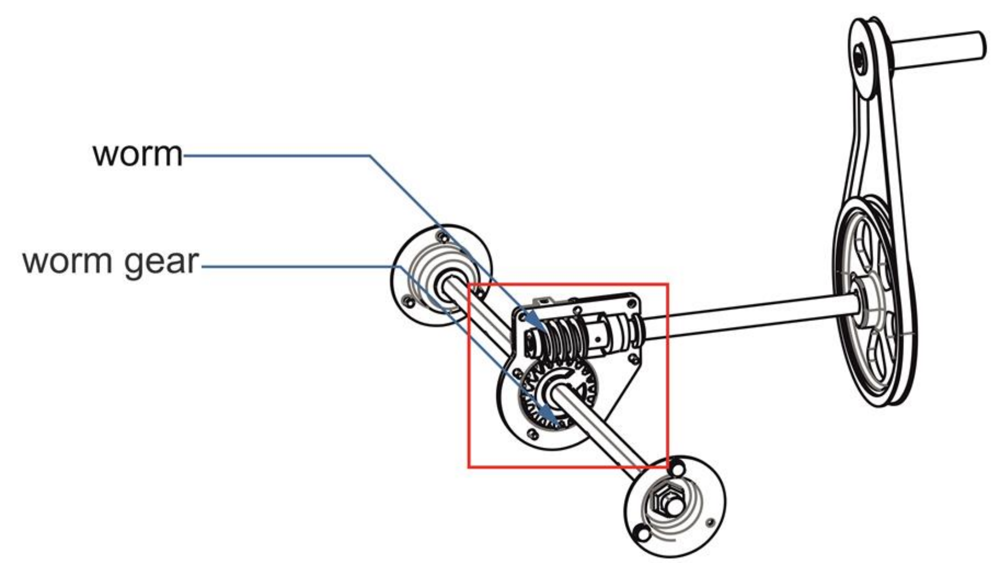


Figure III. 2 : The location of the worm gear in the sweeping mechanism

The relationships between the forces on the worm and worm gear are based on the action/reaction principle, where the forces on the worm are equal in magnitude but opposite in direction to the forces on the worm gear [37].

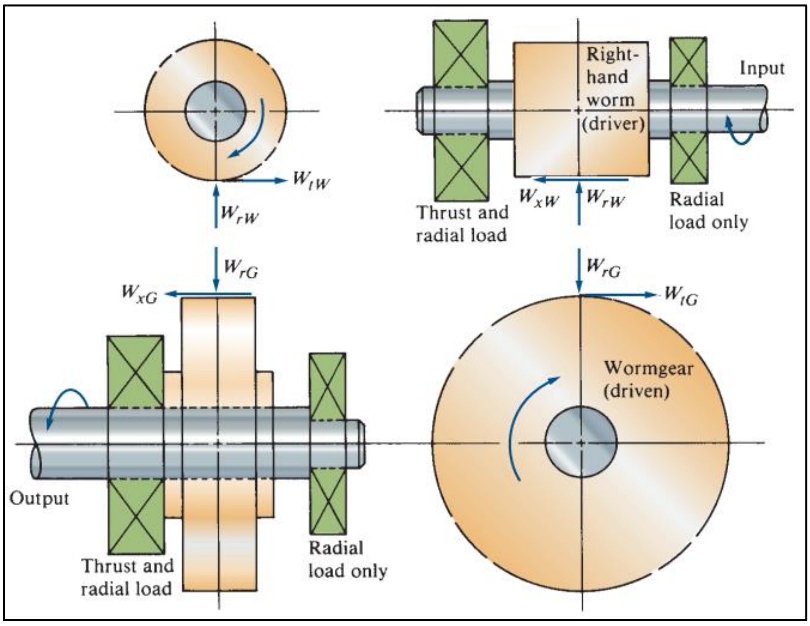


Figure III. 3: Forces on a worm and a worm gear [1].

The worm gear forces can be obtained using the following equations [1]:

- Tangential force on the gear **W_{tG}** [1]**:**

$$W_{tG} = \frac{2 \times T_O}{D_G}$$

- Axial force on the gear **W_{xG}** [1]**:**

$$W_{xG} = W_{tG} \cdot \frac{COS(\phi_n) \times sin(\boldsymbol{\lambda}) + \mu \times cos(\lambda)}{COS(\phi_n) \times COS(\boldsymbol{\lambda}) - \mu \times cos(\lambda)}$$

- Radial force on the gear W_{rG} [1]**:**

$$W_{rG} = \frac{W_{tG} \times \sin(\phi_n)}{COS(\phi_n) \times COS(\boldsymbol{\lambda}) - \mu \times \cos(\lambda)}$$

With: $W_{xW} = W_{tG}$, $W_{tW} = W_{xG}$, $W_{rW} = W_{rG}$

$\boldsymbol{\mu}$: Coefficient of friction

$\boldsymbol{\lambda}$ (°): load angle

$\boldsymbol{\phi_n}$**:** Pressure angle

: is the tangential force on the worm gear.

: is the axial force on the worm.

: is the axial force on the worm gear.

: is the tangential force on the worm.

: is the radial force on the worm gear.

: is the radial force on the worm.

The tangential force on the worm gear is calculated first, based on the operating conditions of torque, power, and speed at the output shaft. This force is critical as it directly affects the efficiency and performance of the worm gear set [1, 37].

- **Coefficient of Friction, μ**

According to the American Gear Manufacturers Association ***(AGMA)*** recommendations, the coefficient of friction for a hardened steel worm which is smoothly ground, polished, rolled, or has an equivalent finish, operating on a bronze worm gear, can be estimated using the following formulas based on the sliding velocity [1]. Note that $\boldsymbol{vs}$must be in m/min in the formulas, where

1.0 m/min = 0.0051 ft/min.

- **Static Condition:** if $v_s == 0$, $\mu = 0.150$
- **Low Speed:** if $v_s \leq 3.05\, m/min$, $\mu = 0.103e^{-0.074v_s^{0.450}}$
- **Higher Speed:** if $v_s > 3.05\, m/min$, $\mu = 0.103e^{-0.110v_s^{0.450}} + 0.012$

These equations are derived from observations regarding the forces acting on the worm gear system. They involve parameters such as the torque applied to the worm T_O, the pitch diameter of the gear $\boldsymbol{D_G}$, the pressure angle of the gear teeth $\boldsymbol{\phi_n}$, the lead angle of the worm$\boldsymbol{\lambda}$, and the coefficient of friction [38].

Friction plays a major role in the operation of a worm gear set due to the inherent sliding contact between the worm threads and the worm gear teeth. The coefficient of friction depends on the materials used, the lubricant, and the sliding velocity. Based on the pitch line speed of the gear, the sliding velocity is calculated as follows:

- **Sliding Velocity for Worm:**

$$v_s = \frac{v_{tW}}{cos(\lambda)}$$

- **Sliding Velocity for Gear:**

$$v_s = \frac{v_{tG}}{sin\lambda}$$

- **Pitch Line Speed, v_t**

The pitch line speed is the linear velocity of a point on the pitch line for the worm or the worm gear. It is calculated using the following formulas[1]:

$$v_{tw} = \frac{\pi \times D_W \times n_W}{60,000}$$

For the worm with a pitch diameter D_W (mm), rotating at $\boldsymbol{n_W}$ RPM:

$$v_{tG} = \frac{\pi \times D_G \times n_G}{60,000} \text{ m/s}$$

For the worm gear with a pitch diameter D_G (mm), rotating at n_GRpm: **m/s**

The following table summarize the obtained forces result:

Table III. 1: Forces applied to the worm gear

Forces worm gear	**W_{xW}= W_{tG}**	**W_{tW}= W_{xG}**	**W_{rW}= W_{rG}**
Results (N)	766.57	405.10	313.32

III.2.2 Calculation spur gear forces of the Walking Mechanism

In the walking mechanism of the device, various forces act on the spur gears, which are crucial for its proper functioning. Understanding and calculating these forces help ensure that the mechanism operates smoothly and efficiently.

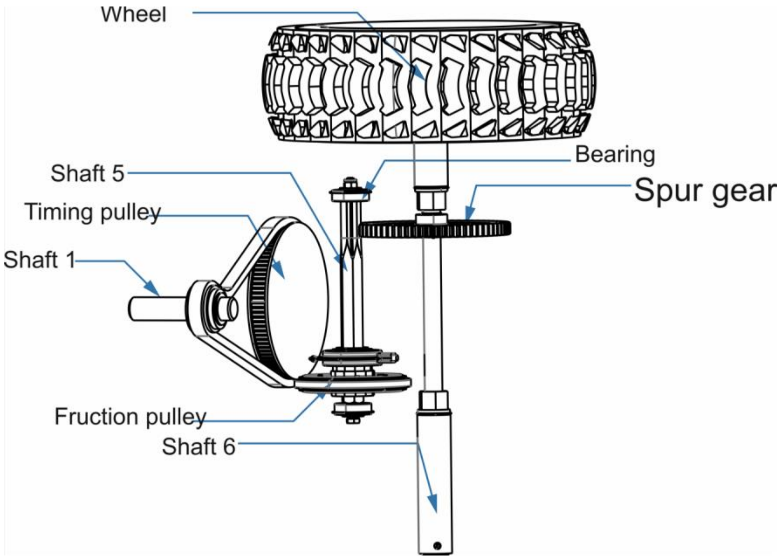


Figure III. 4 : The Walking Mechanism

In this section, we will determine the tangential, radial, and normal forces applied to the spur gears.

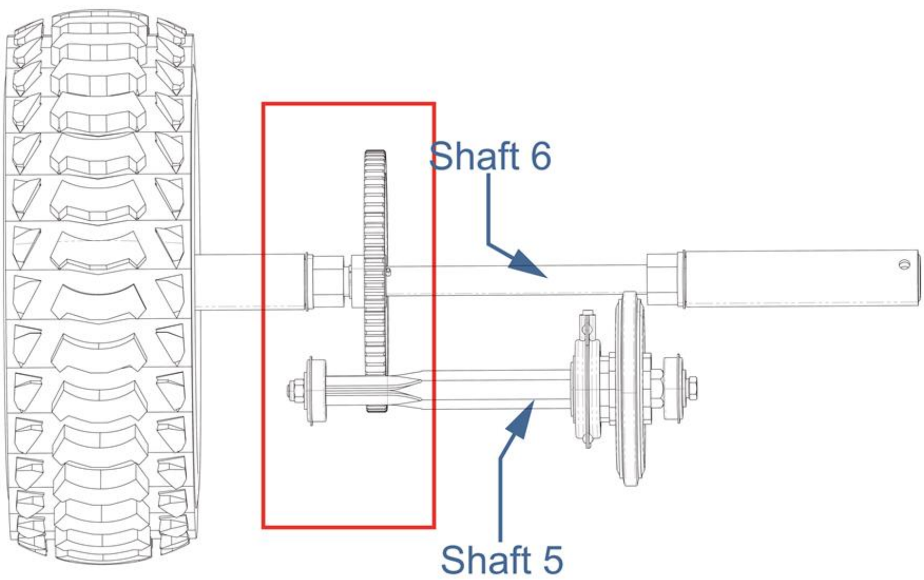


Figure III. 5 : Spur gear in the walking mechanism.

The **figure III.3** represent the forces applied to the spur gear of walking mechanism. In this section, the tangential, radial and normal forces will be calculated.

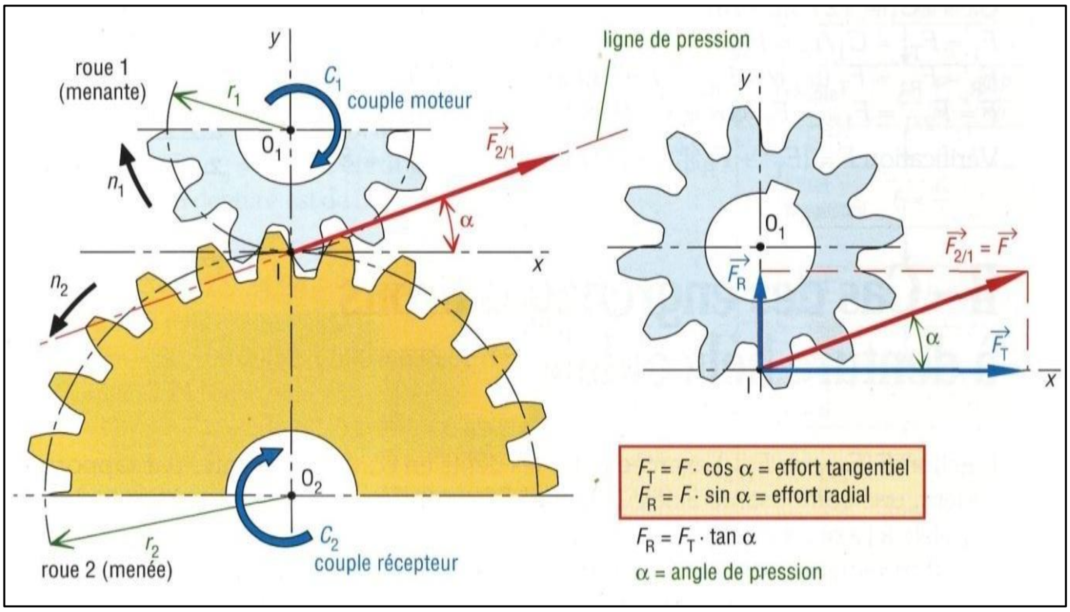


Figure III. 6: (d) Tangential and radial forces exerted by the pinion tooth on the gear tooth.

a) Tangential Forces F_T

In the context of a the walking mechanism, the tangential forces are crucial for understanding the interaction between the cleaning brushes and the surface. These forces are directly related to the efficiency of debris removal. The tangential force F_T can be calculated using the formula:

$$F_T = \frac{T_i}{R}$$

Where:

- F_T is the tangential force.
- T_i is the torque of the shaft selected
- R is the radius at which the force is applied, typically at the point where the brush contacts the surface.

a) Radial Forces F_R

The radial force F_R is influenced by the angle at which the brushes contact the surface and is given by:

$$F_R = F_T \times \tan(\alpha)$$

Where: F_R is the radial force and F_T is the tangential force.

α is the pressure angle, a critical design parameter that influences the efficiency of force transfer from the brushes to the surface. For the purpose of this analysis, the pressure angle α is assumed to be 20 degrees[1].

b) Normal Forces F_N

The normal force F_N is the resultant force that combines the effects of F_T and F_R and is calculated as:

$$F_N = \sqrt{F_T^2 + F_R^2}$$

The table III.2 summarize the obtained results:

Table III. 2: Results of applied forces to the spur gear

Spur Gear	F_T (N)	F_R (N)	F_N (N)
Shaft 5 (pinion)	13893.33	5056.76	14784.98
Shaft 6 (wheel gear)	13221.58	4812.26	14070.11

III.3 Calculation of Shaft Dimensions

To refine the preliminary sizing of the sweeper's shafts, let's clarify the formulas and the material selection process. The shafts are subjected to torsion and bending, but due to the lack of specific information about the forces and support conditions, we'll focus on torsion for the initial sizing. Here's the revised approach [29]. (**Figure III. 3**) represent the global mechanism of machine and the two parts mechanism, walking and sweeping mechanism, of machine.

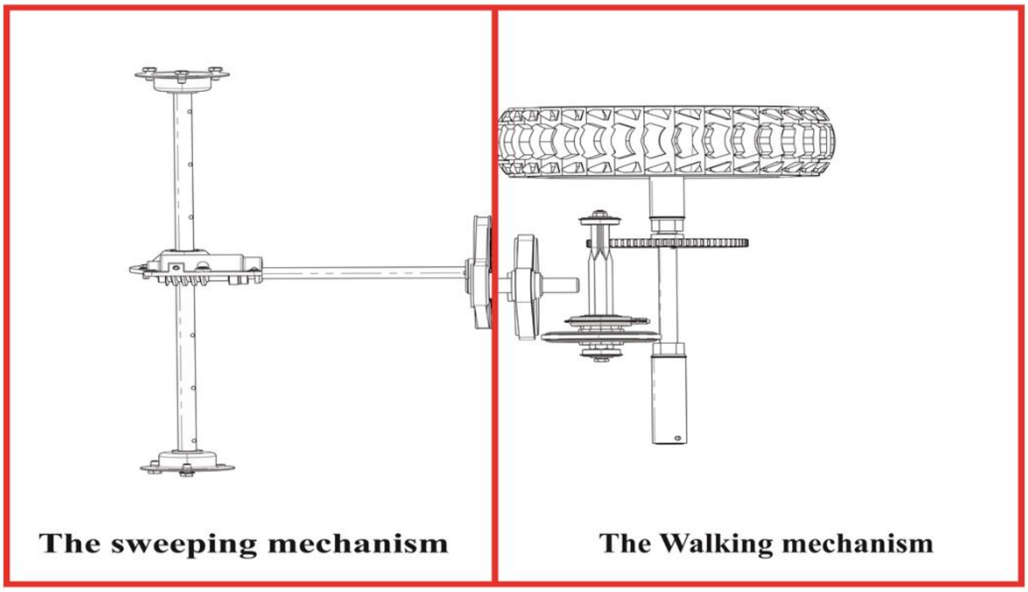


Figure III. 7 : The sweeping and walking mechanism.

The formula to estimate the shaft diameter d based on torsion is:

$$d \geq \sqrt[3]{\frac{T_i}{0{,}2\ \times R_{Pg}}}$$

The polar section modulus R_{Pg} is related to the material's resistance to torsion [1, 39].

It is calculated as half of the yield strength$R_P$which in turn is half of the elastic limit R_{pe}:

$$R_{Pe} = \frac{R_P}{2}$$

$$R_{Pg} = \frac{R_{Pe}}{2}$$

For the material selection, ***C45*** steel has been chosen due to its mechanical properties. ***C45*** steel is a medium carbon steel with yield strength$\boldsymbol{R_{Pe}}$ of 490 N/mm². This makes it a suitable choice for components that require a balance between strength and ductility, which is essential for parts that will undergo torsional stress.

The selection of ***C45*** steel for the device's components that are subject to torsional forces ensures that the device will have the necessary resilience and performance under operational conditions. The polar section modulus, in this case, would be calculated as:

$$R_{Pe} = \frac{490}{2} = 245\ \mathrm{N/m}m^2 \qquad \text{and} \qquad R_{Pg} = \frac{245}{2} = 122.5\ \mathrm{N/m}m^2$$

- **the sweeping mechanism**

Diameter of Shaft 1: 8.49 mm Approximated diameter $\boldsymbol{D_1 = 20\ mm}$

Diameter of Shaft 2: 10.37 mm Approximated diameter $\boldsymbol{D_2 = 20\ mm}$

Diameter of Shaft 3: 17.44 mm Approximated diameter $\boldsymbol{D_3 = 20\ mm}$

- **The walking mechanism**

Diameter for Shaft 4: 12.64 mm Approximated diameter $\boldsymbol{D_4 = 16\ mm}$

Diameter for Shaft 5: 16.30 mm Approximated diameter $\boldsymbol{D_5 = 20\ mm}$

Diameter for Shaft 6: 35.11 mm Approximated diameter $\boldsymbol{D_6 = 60\ mm}$

III.4 Optimizing Shaft Design for Enhanced Durability and Performance in Multi-Functional Road Maintenance Device

Following a comprehensive series of trials and analyses, which included a thorough examination of mechanical and dynamic factors, we have established the optimal shaft lengths for the Multi-Functional Road Maintenance Device. This determination was made after considering a variety of factors such as the anticipated load on the shafts, the spacing between supports, and the even distribution of forces along each shaft. Additionally, a safety factor was incorporated to ensure stability and reliable performance across different operational conditions.

The lengths of the shafts have been meticulously calculated to align with the overall dimensions of the sweeper and to adhere to the design's geometric constraints. This careful balancing act between efficiency and durability guarantees ample room for maintenance and any necessary repairs, thereby reflecting our dedication to delivering quality and high-performance machinery. The sweeper is designed to operate effectively with minimal maintenance over a prolonged period, which is a testament to our commitment to innovation and excellent [46].

The finalized shaft lengths are as follows:

Shaft 1: 119 mm , Shaft 2: 353.5 mm , Shaft 3: 680 mm , Shaft 4: 50 mm , Shaft 5: 256 mm

Shaft 6: 552 mm

These lengths are the result of our rigorous application of mechanical engineering principles and the employment of state-of-the-art analytical techniques. We take pride in our methodical approach and our unwavering commitment to innovation and excellence in all our endeavours.

III.5 Selection and Verification of Keys

Having determined the minimum diameters for our shafts and the corresponding gear diameters. In this section we study the dimension and the selection of keys. These small but crucial components are responsible for connecting the gear to the shaft, ensuring a synchronized rotation between the two. The key must be robust enough to withstand the torsional forces transmitted through the shaft without failing[40].

Keys are typically subjected to shear and compressive forces and their dimensions must be chosen to prevent shearing or crushing under load. The material of the key should also be compatible with that of the shaft and gear to avoid undue wear or corrosion.

To ensure the reliability of the key in operation, we will conduct a verification process. This involves calculating the expected forces and comparing them to the material strength of the key. We'll consider factors such as the torque transmitted by the shaft, the power of the system, and the operational conditions. If the calculated stress on the key is within acceptable limits, we can proceed with confidence that the key will perform its function reliably.

In summary, the selection and verification of keys are vital steps in the design of a mechanical system, requiring careful consideration of the forces involved and the materials used to ensure a durable and efficient connection between the shaft and gear [1, 41].

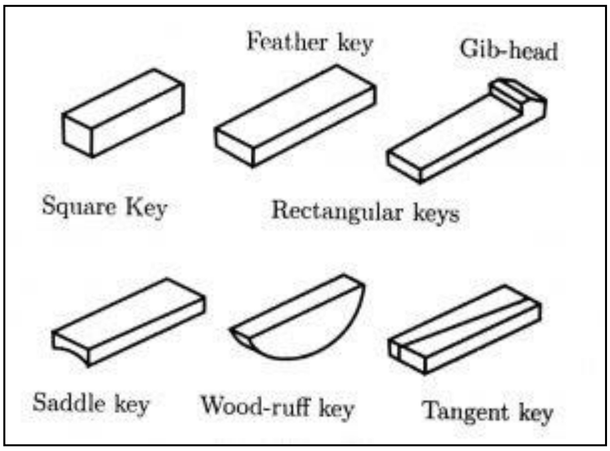


Figure III. 8 : Parallel keys [1].

III.5.1 Stress Analysis for Determining Key Length

To determine the dimensions of the key, two checks are necessary; resistance to shear and matting

a) Shear Stress Analysis

When torque is applied to the shaft, it creates a force F on the key **(Figure III.5)**, causing the key to exert a force on the hub key seat. This interaction generates opposing forces, subjecting the key to direct shear across its cross-sectional area, given by the width W and length L of the key. The key support the force F if the maximum shear stress, created by the force $\boldsymbol{F}$, is lower than the limit shear stress τ_{rpg} of material [1].

$$\tau_{max} < \tau_{rpg}$$

The maximum shear stress τ_{max} is obtained by the following equation:

$$\tau_{max} = \frac{F}{A_s} = \frac{2.T}{D.W.L}$$

Where:

T (N.mm) : Torque , ***R***(mm): Shaft Radius,

L(mm): length of the key (generally equal to the width of the pulley or pinion)

The limit shear stress is obtained from the properties of material by:

$$\tau_{rpg} = \frac{\tau_g}{s}$$

With ***s*** is safety factor (generally taken between 2-5, in this this study s = 2).

$\tau_g = 0.5 \; to \; 0.8 \; \sigma_e$ (In this study $\tau_g = 0.5 \; \sigma_e$)

The final equation of shear strength is:

$$\frac{2.T}{D.W.L} < \frac{\sigma_e}{4}$$

Therefore :

$$W > \frac{8.T}{\sigma_e.D.L}$$

The used material in this study is the steel C45, who has a limit elastic $\sigma_e = 390 \; MPa$ (Table III.3).

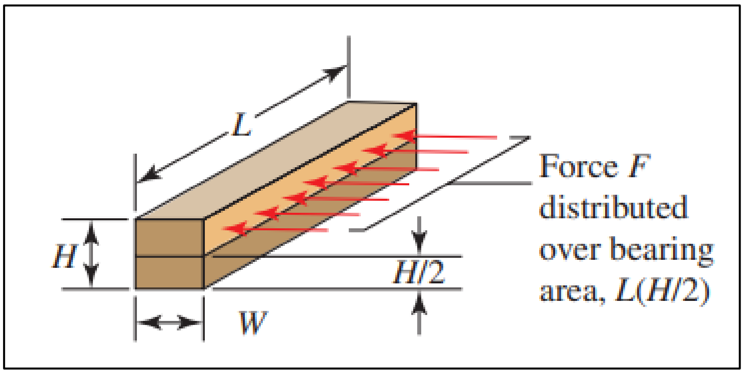


Figure III. 9 : Forces on a key [1].

b) Compression Stress Analysis

Compression stress analysis focuses on the compressive force exerted on the sides of the key, the shaft key seat, or the hub key seat. The area under compression is $\times \frac{H}{2}$, where ***H*** is the **Compression Stress Analysis** height of the key. The key support the maximum pressure created by the force ***F*** if the maximum stress σ_{max}, is lower than the limit pressure of mating $\boldsymbol{P_m}$ of material [1].

$$\sigma_{max} < P_m$$

The maximum stress σ_{max} is obtained by the following equation:

$$\sigma_{max} = \frac{4.T}{D.H.L}$$

Where: H is the thickness of key.

Table III. 3 : Mechanical properties of used material.

Material designation	**Tensile strength S_U**		**Yield strength S_y**	
	(Ksi)	(MPa)	(Ksi)	(MPa)
Carbon steels (SAE)				
1018	64	441	54	372
1035	72	496	39.5	272
1045	91	627	77	531
1095	140	965	83	572
Alloy steels (SAE)				
4140	102	703	90	621
8630	100	690	95	655
Stainless steels (SAE)				
303	90	621	35	241
304	85	586	35	241
316	85	586	35	241
416	75	517	40	276
Aluminium				
6061	18	124	12	83

The limit pressure of mating P_m depends on the type of mounting (sliding, normal, or tight) and the operating conditions (excellent, normal, or poor). Operating conditions are poor in the event of significant shocks or jolts, or in the event of axial "freedom" of the hub on the shaft.

Table III. 4: Assembly and working condition of keys

Assembly	Working conditions		
	Excellent	Normal	Poor
Glissant	10 à 20 Mpa	5 à 15 Mpa	2 à 10 Mpa
Normal	30 à 50 Mpa	20 à 40 Mpa	15 à 30 Mpa
Tight	80 à 150 Mpa	60 à 100 Mpa	40 à 70 Mpa

The final equation of shear strength is:

$$\frac{4.T}{D.H.L} < P_m$$

Therefore :

$$H > \frac{4.T}{P_m.D.L}$$

In this study $P_m = 50\ MPa$ is used to determine the thickness of key with a normal assembly (Shaft 1, 2 and 4). For the shaft 3 and 6, the assembly between elements can be tight. To determine the dimension of keys, the pressure of mating $P_m = 120\ MPa$ is considered. Table III.4 summarize the calculated and the selected dimensions of used keys.

Table III. 5 : Dimensions of calculated and selected keys

		Calculated			**selected**		
		L (mm)	**W (mm)**	**H (mm)**	**L (mm)**	**W (mm)**	**H (mm)**
Shaft 1	Pulley 1	21	0.75	2.8	21	6	6
	Timing pulley	28	0.75	2.8	21	6	6
Shaft 2	/	/	/	/	/	/	/
	Worm	21	1.37	5.11	21	6	6
Shaft 3	Worm gear	21.43	6.39	9.91	21.43	8	10
Shaft 4	Timing pulley	28	2.33	8.68	28	14	9
Shaft 5	Spur gear	/	/	/	/	/	/

Shaft 6	Spur gear	30	12.43	15.42	30	28	16

III.6 Selection and Verification the Right Pin

In the pinning process, an element is positioned on the shaft, and a hole is drilled through both the hub and the shaft. A pin is then inserted into this hole. (**Figure III. 6)** illustrates three examples of this approach.

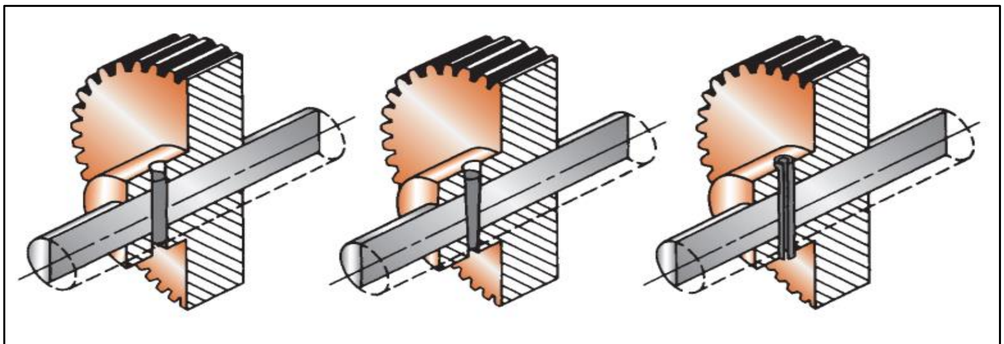


Figure III. 10 : Pinning [1].

The shear stress in the pin is calculated as:

$$\tau_{max} < \tau_{rpg}$$

$$\tau_{max} = \frac{F}{A_s} = \frac{T}{D \times (\pi \times d^2/4)} = \frac{4 \times T}{D \times (\pi \times d^2)}$$

The pin support load if the maximum shear stress τ_{max} can be lower than the shear stress strength τ_{rpg}.

The selected material for pin is carbon steel SAE 1035 With : $\sigma_e = 272\ MPa,\ s = 1.5, \tau_{rpg} = 90\ MPa$. In this mechanism the cylindrical pin is selected.

The diameter of pin is calculated by the following equation :

$$d = \sqrt{\frac{4 \times T}{D \times (\pi) \times (\tau_{rpg})}}$$

Where : T = 26.83 N.m , D (mm) : diameter of shaft 2 (D = 20 mm), d(mm) : Diameter of pin (d=3.71 mm).

According the standard, the selected diameter D = 4mm (Table III.6).

Table III. 6 : Relationship between diameter and length of pins

d	**1.5**	**2**	**2.5**	**3**	**4**	**5**	**6**	**8**	**10**	**12**	**16**	**20**
L	8-20	8-30	8-30	8-40	8-60	8-60	10-80	12-100	14-120	14-120	24-120	26-120

III.7 Overview of Rolling Contact Bearings and Their Mechanical Applications

In the realm of mechanical engineering, rolling contact bearings are pivotal components that facilitate the smooth operation of machinery by reducing friction between moving parts. Here's a concise summary of the various types of bearings and their applications, suitable for inclusion in a graduation thesis[1].

III.7.1 Rolling Contact Bearings Types and Applications

Rolling contact bearings are designed to support radial and thrust loads on shafts, with different types catering to specific load orientations and magnitudes. Radial loads act perpendicularly to the axis of the shaft, while thrust loads align parallel to it. Misalignment, the angular deviation of the shaft's axis at the bearing from the true axis, is also a critical factor in bearing selection.

a. Single-Row, Deep-Groove Ball Bearings

These types of bearings are the most common type, suitable for handling radial loads and moderate thrust loads. They feature an inner race fitted onto the shaft and spherical balls maintained by retainers, rolling in deep grooves of both races.

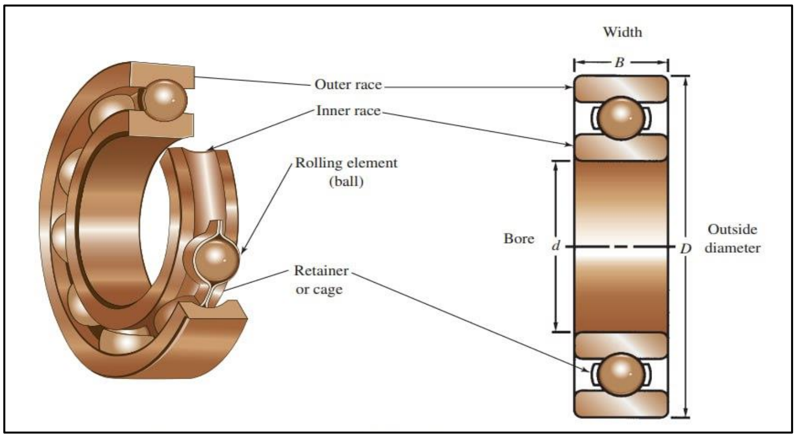


Figure III. 11 : Single-Row, Deep-Groove Ball Bearings [1].

b. Double-Row, Deep-Groove Ball Bearings

The double row bearings increase radial load capacity by sharing the load across more balls, allowing for a greater load in the same space or a given load in a smaller space.

Figure III. 12 : Double-Row, Deep-Groove Ball Bearings [1].

Table III. 7 : Comparison of Bearing Type [1].

Bearing type	**Radial load capacity**	**Thrust load**	**Misalignment**
Single-row, deep-groove ball	Good	Fair	Fair
Double-row, deep-groove bal	Excellent	Good	Fair
Angular contact	Good	Excellent	Poor
Cylindrical roller	Excellent	Poor	Fair
Needle	Excellent	Poor	Poor
Spherical roller	Excellent	Fair/good	Excellent
Tapered roller	Excellent	Excellent	

When selecting bearings, consider the type of loads, expected life, size, installation method, and requirements for lubrication, shields, and seals. This information assists in making informed design decisions for mechanical drives and related components. The Table III.8 summarizes the selected SKF bearings taking account the supported load type [42].

Table III. 8 : bearing selection

<table>
<tr><th></th><th>Elements</th><th>Tangential force (N)</th><th>Radial force (N)</th><th>SKF Reference of bearings [42]</th><th>Static load limit (KN) [42]</th><th>d_{int} (mm)</th><th>D_{ext}(mm)</th><th>b (mm)</th><th>limit rotation speed (rpm)</th></tr>
<tr><td rowspan="2">Shaft 1</td><td>Pulley 1</td><td>576.92</td><td>381.97</td><td rowspan="2">6204</td><td rowspan="2">6.55</td><td rowspan="2">20</td><td rowspan="2">47</td><td rowspan="2">14</td><td rowspan="2">13000</td></tr>
<tr><td>Timing pulley</td><td>471.32 (Normal load)</td><td>/</td></tr>
<tr><td rowspan="2">Shaft 2</td><td>Pulley 2</td><td>520.97</td><td>381.97</td><td>6204</td><td>6.55</td><td>20</td><td>47</td><td>14</td><td>13000</td></tr>
<tr><td>Worm</td><td>766.57 (axial load)</td><td>313.32</td><td>30204</td><td>28</td><td>20</td><td>47</td><td>15.25</td><td>15000</td></tr>
<tr><td>Shaft 3</td><td>Worm gear</td><td>405.1 (Axial load)</td><td>313.32</td><td>30204</td><td>28</td><td>20</td><td>47</td><td>15.25</td><td>15000</td></tr>
<tr><td>Shaft 4</td><td>Timing pulley</td><td>424.38 (Normal load)</td><td>/</td><td>6303</td><td>6.55</td><td>17</td><td>47</td><td>14</td><td>22000</td></tr>
<tr><td>Shaft 5</td><td>Spur gear</td><td>13893</td><td>5056.76</td><td>305</td><td>15.6</td><td>25</td><td>62</td><td>17</td><td>13000</td></tr>
<tr><td>Shaft 6</td><td>Spur gear</td><td>13221.58</td><td>4812.26</td><td>6012</td><td>23.2</td><td>60</td><td>95</td><td>18</td><td>9500</td></tr>
</table>

Chapter IV

Numerical modelling of mechanisms

IV.1 Introduction

In this part a numerical study of some important part of the machine was carried out using Solidworks software. The parts and assemblies were modelled before starting the numerical calculations. Boundary conditions were introduced in order to find deformations and stresses in elements such as shafts and keys. The blocking of an element is considered as an assumption in the calculation. Blocking is considered in the case of machine malfunction due to obstacles. The results obtained will be compared with the resistance conditions to verify the choice of material and the dimensions calculated previously.

IV.2 Machine modelling

For the modelling of the multifunction machine, the parts and elements of the machine (shafts, pinions, bearings, pulley keys, transmission belts, etc.) were modelled separately, then assembled according to each mechanism (sweeping, walking) by introducing adequate constraints for each two elements. Figure IV.1-a and figure IV.1-b shows an overall overview of the machine and an exploded view respectively.

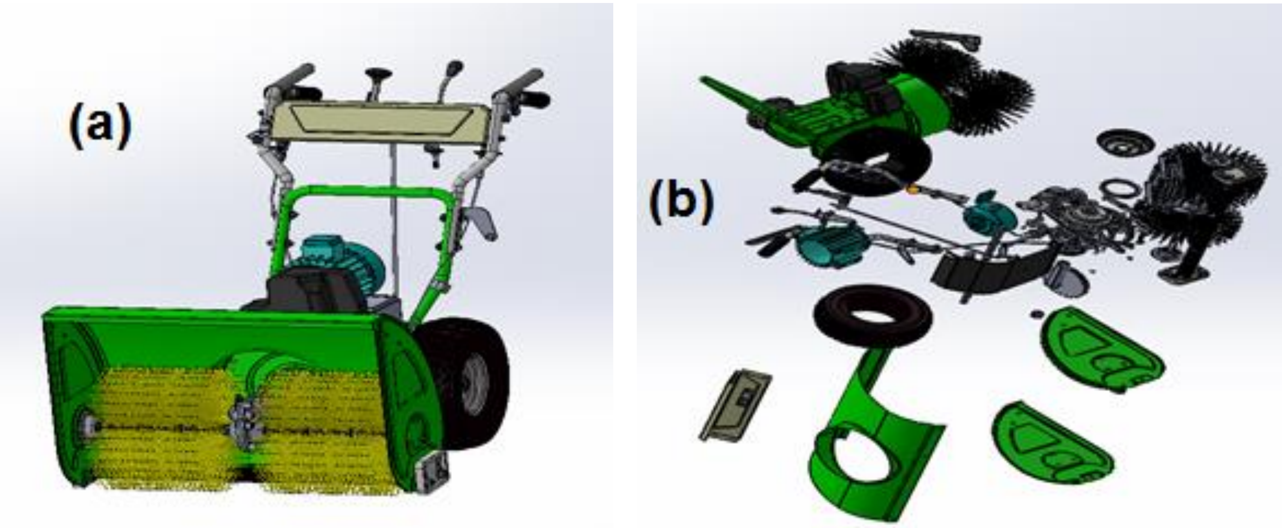


Figure IV. 1: Overall overview of the machine.

The machine mechanism is divided into two parts, the sweeping mechanism and the walking mechanism.

IV.2.1 Elements and modelling of sweeping mechanism

Figure IV.2 , already presented previously, represents the sweeping mechanism. The elements which constitute this mechanism are modelled by Solid works and presented in figure IV.3.

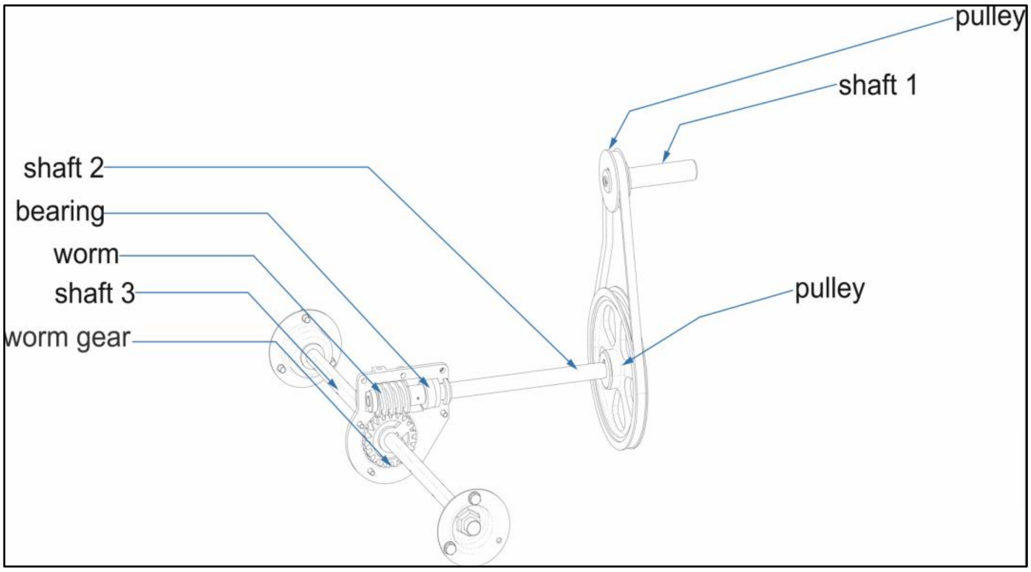
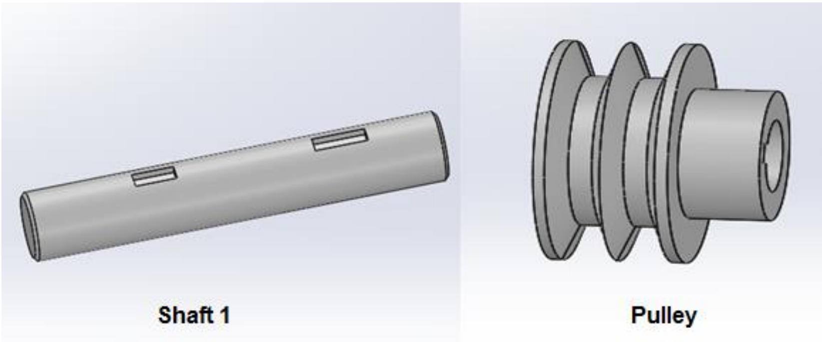
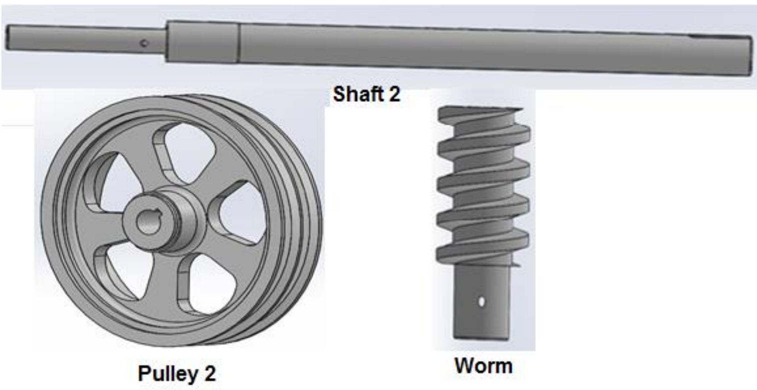


Figure IV. 2 : Sweeping mechanism elements.





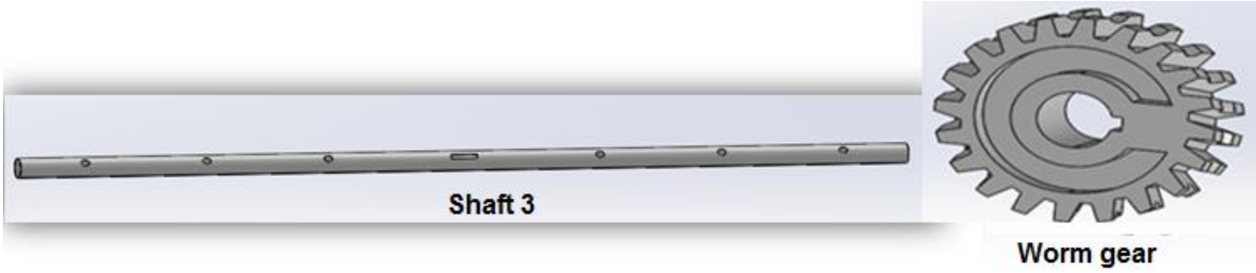


Figure IV. 3: Principal elements of sweeping mechanism

IV.2.2 Elements and modelling of walking mechanism

Figure IV.4 , presents the walking mechanism and theirs elements.

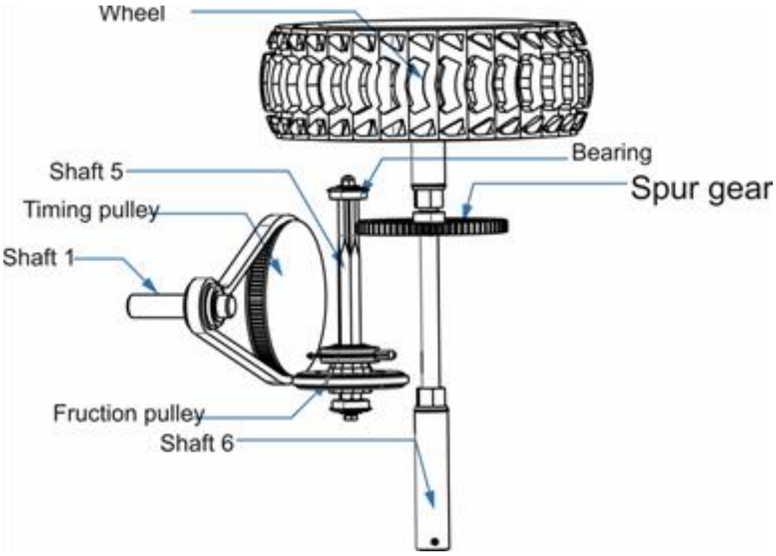
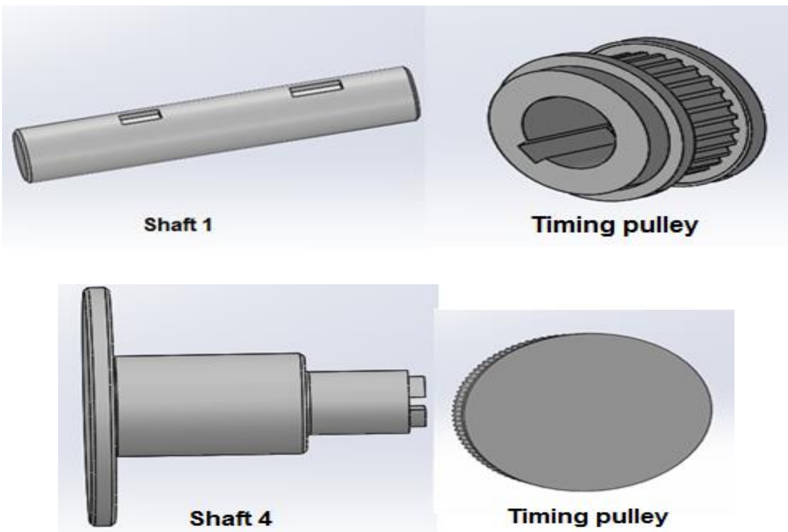


Figure IV. 4 : Walking mechanism elements.



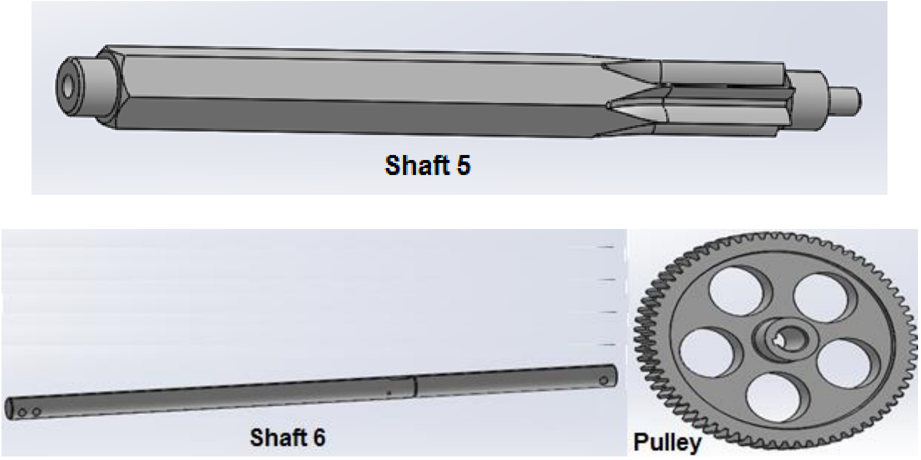


Figure IV. 5 : Essential elements of walking mechanism.

The elements which constitute this mechanism are modelled by Solidworks. Figure IV.5 present the essential elements of the walking mechanism.

VI.3 Secondary elements of the multifunction machine

a) Frame

The frame protects the internal parts and provides the overall shape of the device (Figure IV.6). It can be designed using a combination of tools such as 'Extrude', 'Cut', and 'Fillet' to create complex shapes.

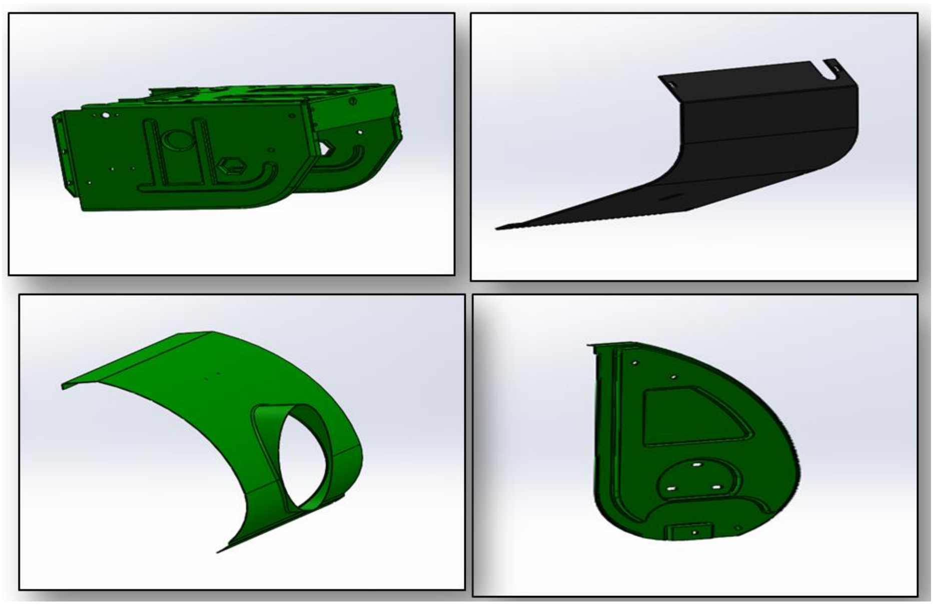


Figure IV. 6 : Frame of multifunction machine.

b) AC Motor

The AC motor is responsible for providing the necessary power to the device (Figure IV.7).

It converts electrical energy into mechanical energy to drive the moving parts. The motor's specifications, such as power, voltage, and efficiency, can be included in the design to ensure it meets the device's requirements.

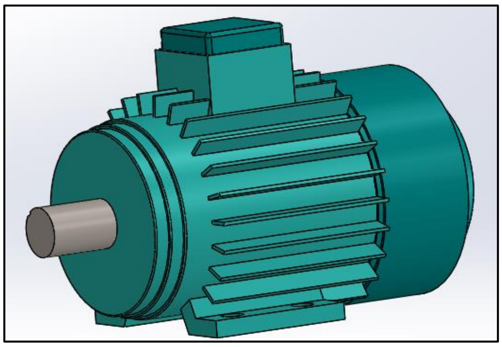


Figure IV. 7: AC motor.

c) Battery

The battery is a critical component that supplies power to the AC motor and other electrical parts of the device (Figure IV.8). Proper placement and size are crucial to ensure it fits well within the device’s frame and provides adequate power capacity.

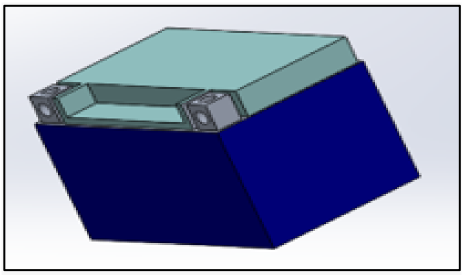


Figure IV. 8: Battery.

d) Wheels

Wheels are essential for the mobility of the device. They can be designed using the ‘Revolve’ tool to create the circular shape and the ‘Pattern’ tool for tread design. Wheels must be designed for durability and traction, considering the operational environment and load they will carry.

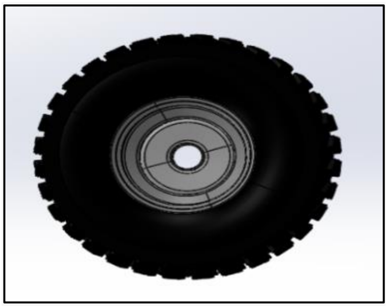


Figure IV. 9: Wheels of the multifunction machine.

IV.4 Numerical analysis of mechanism elements

In this part, we only focus on the numerical study of the sweeping mechanism, specifically the shafts and the keys. Calculation hypotheses were posed such as the blocking of a rotating element of the mechanism, friction is neglected between the mechanical parts and the contact between the elements is simple to facilitate calculation and convergence. The torques values calculated previously were applied in order to see the deformation of the shaft and the maximum stress in each mechanism (Shaft 3 and Shaft 4). The maximum stresses will be compared to the breaking stress limit of each part. Table IV.1 summarize the mechanical properties of steel materials used for each element.

Table IV. 1: Applied torques and Mechanical properties of used materials.

Property	**Shaft 2**	**Shaft 3**	**Worm and worm gear**	**Pine**
Elastic Modulus (N/mm^2)	210000	210000	210000	210000
Poison's Ratio	0.28	0.28	0.28	0.28
Shear Modulus (N/mm^2)	79000	79000	79000	79000
Mass Density (kg/m^3)	7800	7800	7800	7800
Tensile Strength (N/mm^2)	750	750	850	496
Yield Strength (N/mm^2)	490	490	312	272
Applied torque in shaft	26.83	127.52	/	/

IV.4.1 Numerical analysis of the assembly of the shaft 2, worm and pin

In the numerical analysis, the worm is blocked $(x = y = z = 0, \theta_x = \theta_y = \theta_z)$ and a torque of 26.83 N.m is applied to the end of the shaft (Figure IV. . A medium-sized mesh is considered in the calculation.

Figure IV. 10 : Boundary conditions of the assembly elements (Shaft 2, Worm and pin).

Figure IV.11 shows the results of shear stress on the shaft 2, worm and pine. From the resultats, the shear stress concentration is at the pin level with a maximum value 68.83 MPa. This value always remains lower than the yield strength of pin material (272 MPa) and the yield strength of worm material (312 MPa) and the yield strength of shaft material (490 MPa). So the dimensions and choice of material for each component is correct.

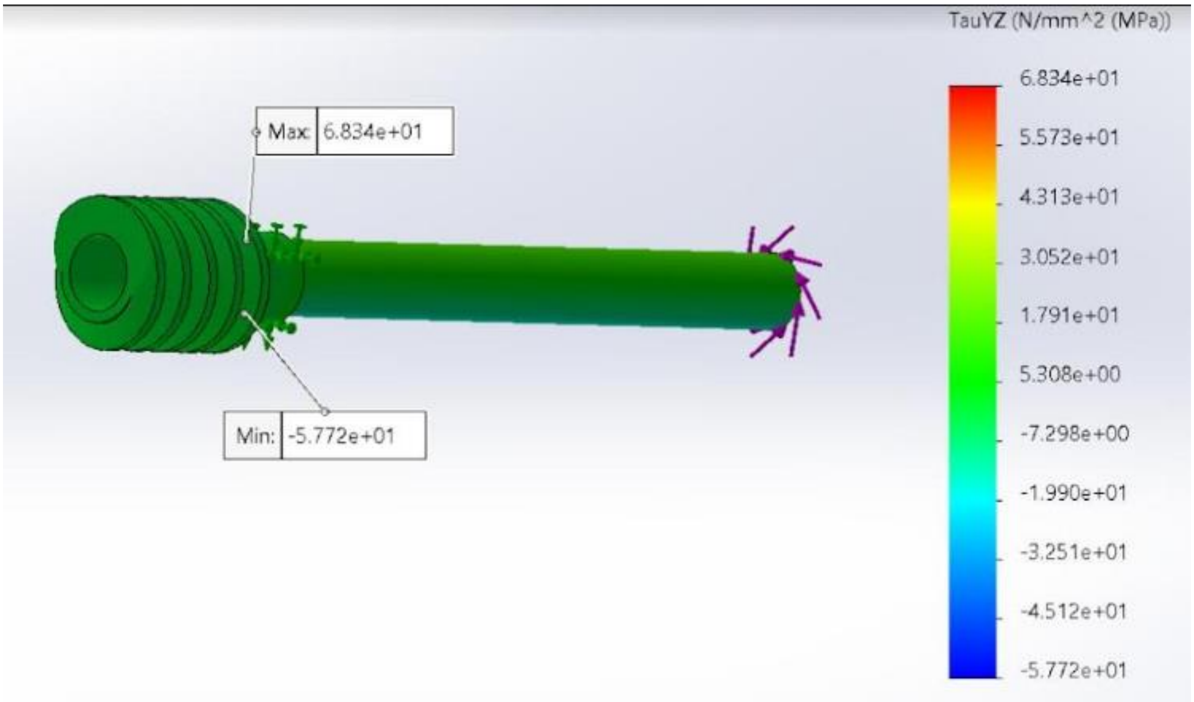


Figure IV. 11 : Shear stress on shaft 2, worm and pin.

The figure IV.12 show the deformation of all elements with maximum deformation equal to 0.028 (mm/mm). this this value is negligible and does not cause the machine to malfunction.

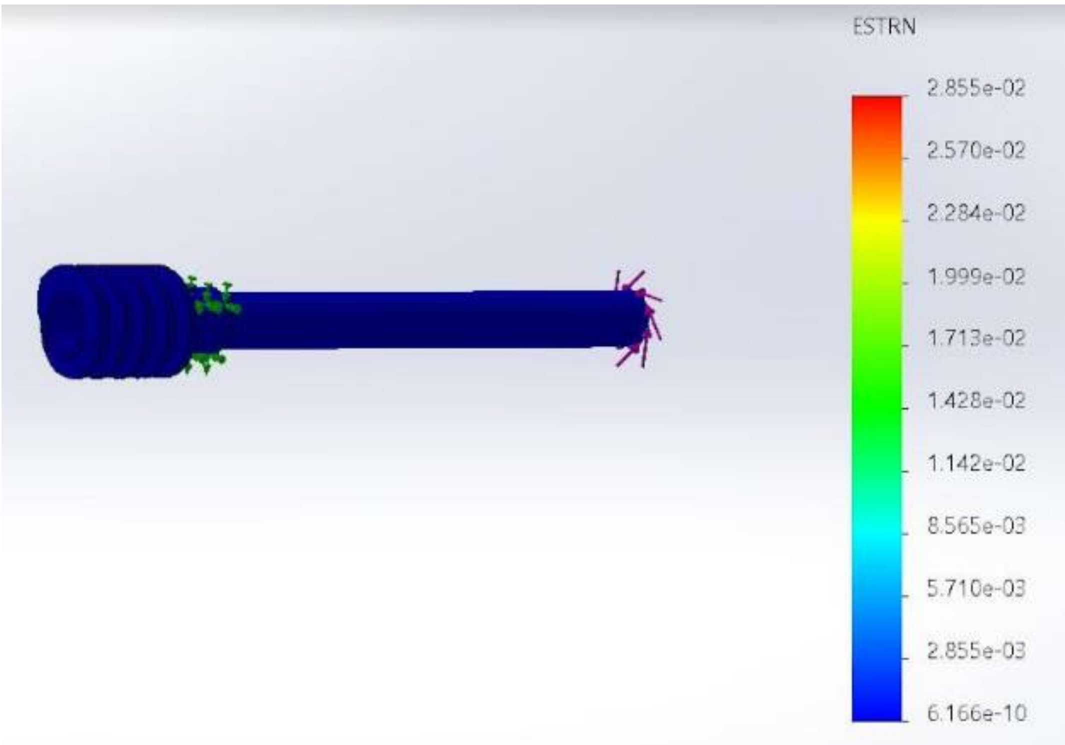


Figure IV. 12 : *Strain on* shaft 2, worm and pin.

IV.4.2 Numerical analysis of the assembly of the shaft 3, pinion and key

In this part, the shaft 3 is blocked in these two extremities and the torque is applied in the worm gear (Figure IV.13).

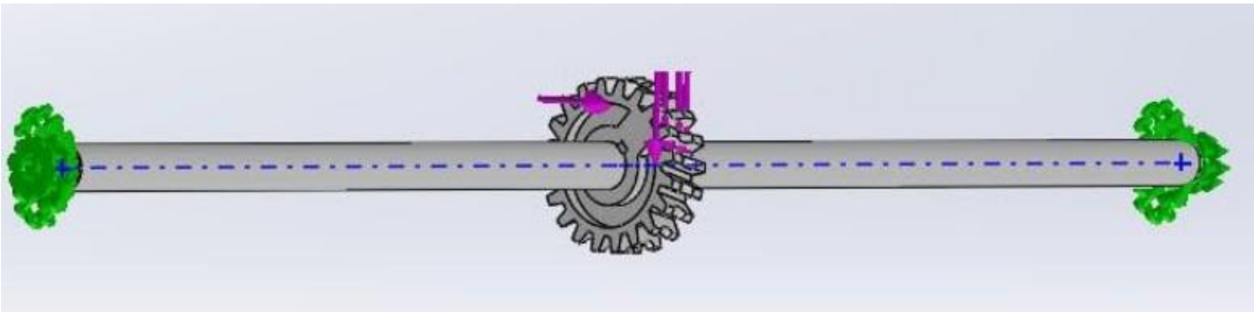


Figure IV. 13 : Boundary condition applied to the shaft 3, worm gear and the key.

Figure IV.14 represent shear stress distribution in the shaft 3, worm gear and key. From the results, the maximum shear stress is equal to 51 MPa and is applied in the end of shaft 3. This value remains lower than the yield shear stress of tha materials of the shaft 3, worm gear and key, cited previously. This shows that the structure resists the applied loading.

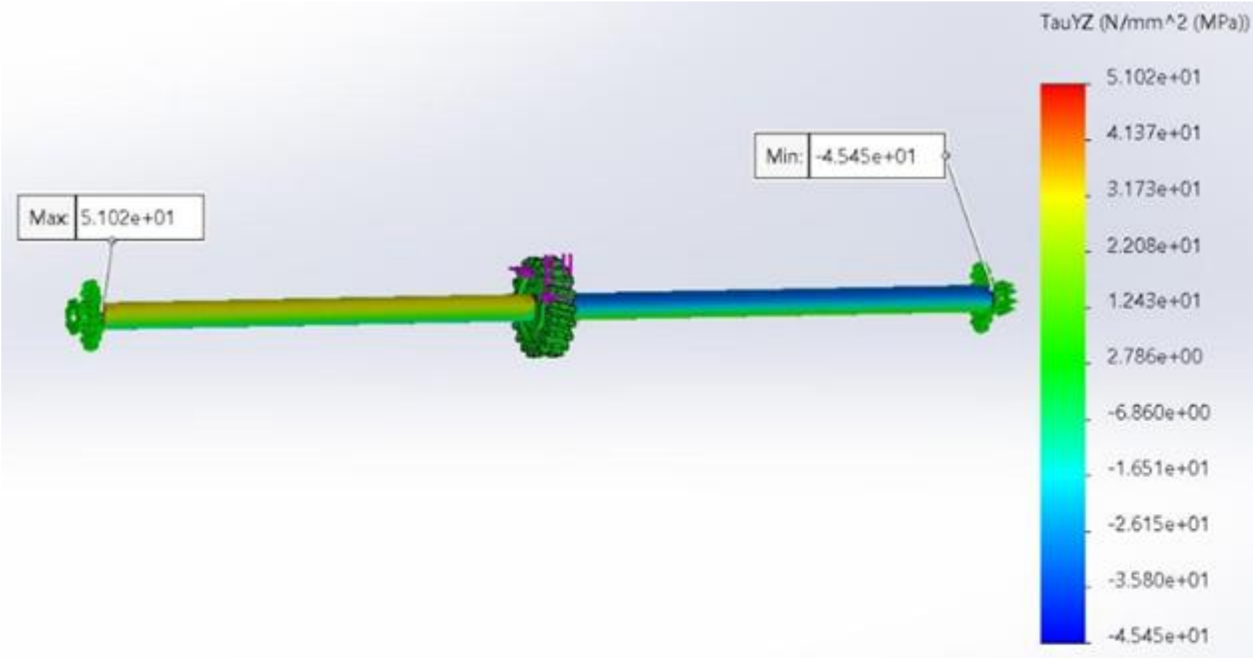


Figure IV. 14: Shear stress distribution in the shaft 3, worm gear and key.

The figure IV.15, show the Z stress distribution. For the key, the Z stress represents the malting pressure. The maximum of this pressure is equal to 80 MPa. This value remains lower than the pressure of malting used in the dimension of key (120 MPa). We conclude that there is not melting deformation in the key.

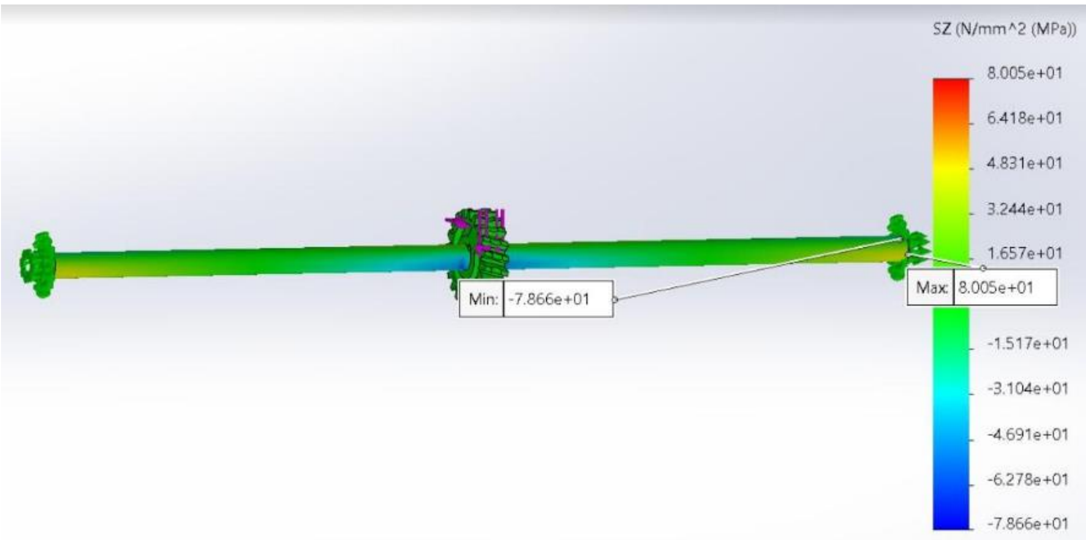


Figure IV. 15 : Z stress distribution on the shaft 3, worm gear and the malting pressure on the key.

The deformation of the shaft 3, worm gear and the key are very small and equal to 0.000035. We conclude that the deformation of elements does not interfere with the operation of the machine.

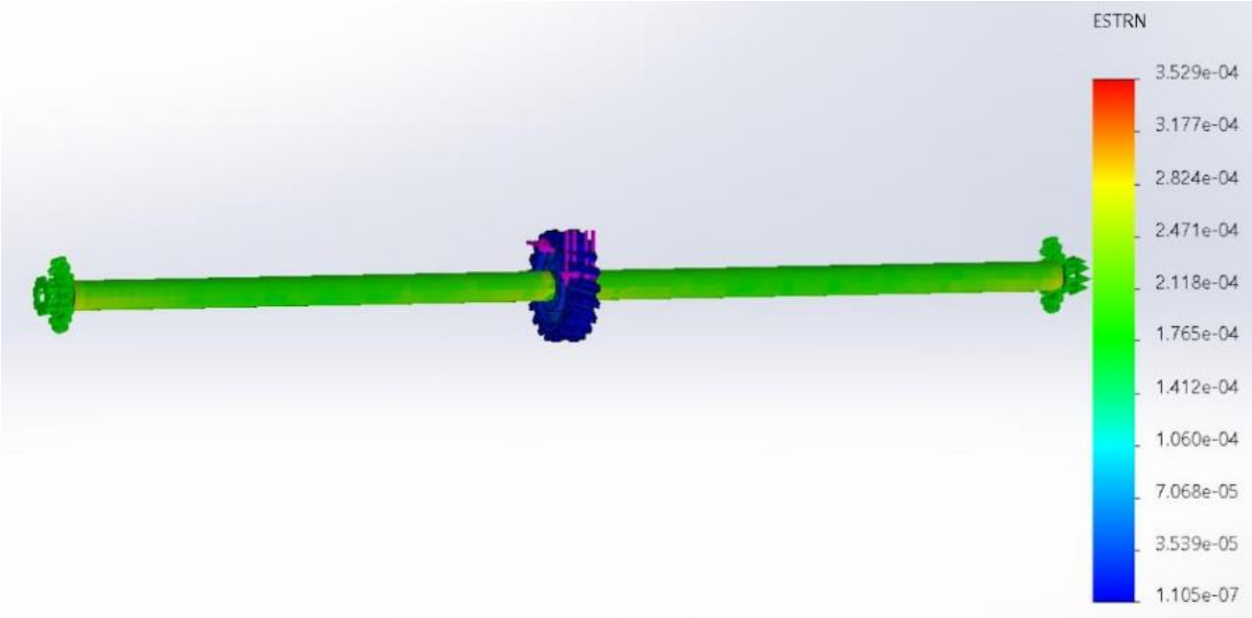


Figure IV. 16 : Deformation of the shaft 3, worm gear and key.

General conclusion

Conclusion general

In this work a design study of a multifunction machine was carried out. The main objective of this work is to create a machine that performs several tasks at once; such as sweeping and cleaning roads, tracing roads, mowing lawns and equipped with a GPS system which makes the machine programmable. Due to time constraints, the machine studied will be used as a cleaning machine or a lawn mower.

Before discussing the main axes of this work a bibliographic research on the history of kinematics and mechanical mechanisms. The proposed multifunction machine is composed by two parts, part named sweeping mechanism and the second part named walking mechnaims. For study this two mechanism, this work is composed by, a kinematic and energy study, a study of sizing and choice of movement transmission elements, resistance study and a digital modelling of certain mechanisms.

In the kinematic and energy study a choice of electric motor was made for the calculation of the transmission ratios, the rotation speeds of each shaft and their powers. The results obtained from the kinematic and energy studies were used in determining the dimensions and the choice of elements (gears, pulleys, belts). In the third part, the dimensions of the shafts, keys and pins and the choice of bearings were made using the fundamental principles of statics and the resistance of materials. Finally, a numerical study on some mechanisms to validate the results obtained theoretically. This design has been improved by introducing other functions mentioned above.

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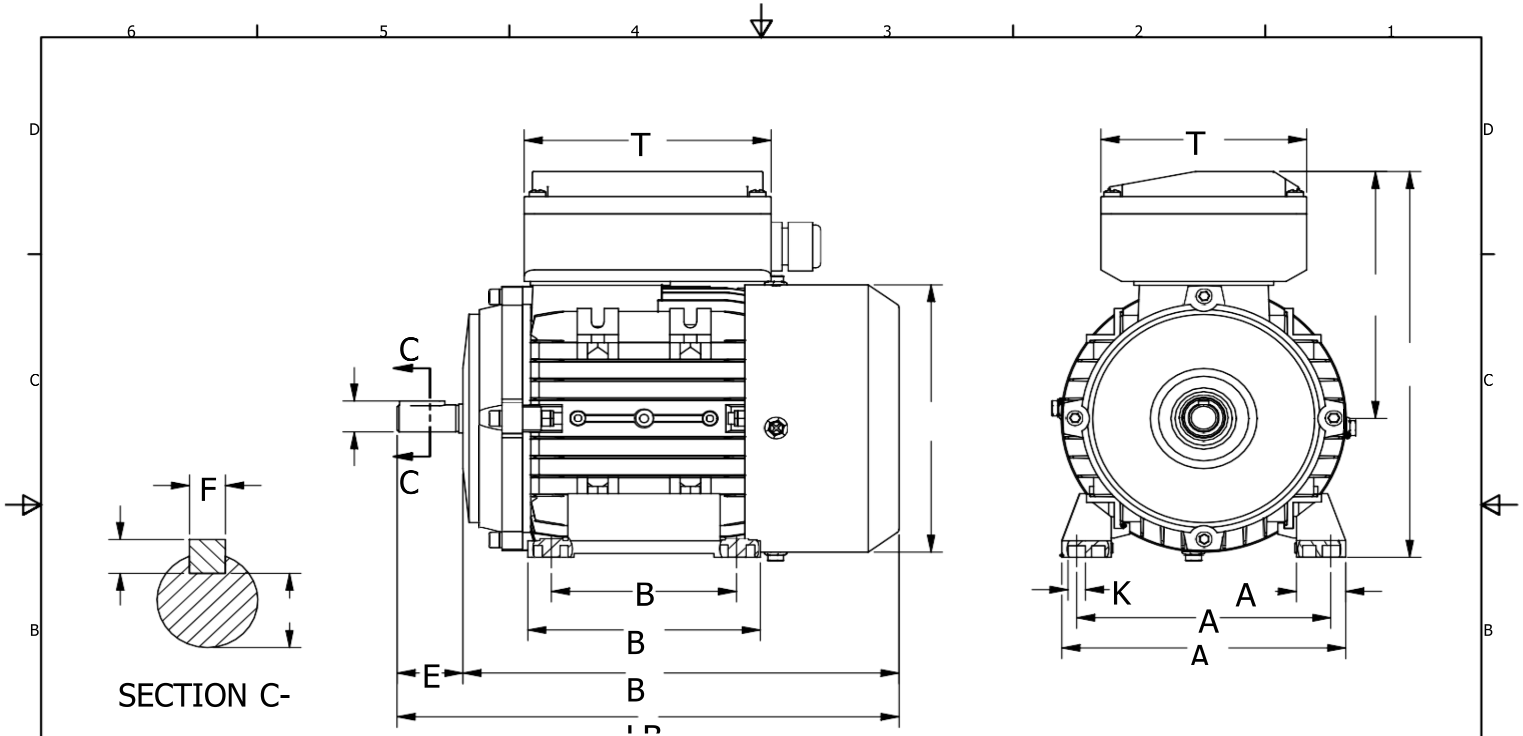
Appendix

Appendix

Appendix 1



A	B	C	D	E	F	G	H	K	AB	BB	AC	AD	HD	L
160	140	63	28	60	8	24	100	12	401	125	175	145	235	308

AMTECS LTD

TITLE AMAS-SR

SIZE | REV

SCALE 1 / 1.75 MY71B3 SHEET 1 OF 1

This drawing serves only as a sketch to show dimensions.
The product delivered may differ. A detailed description of the product can be found below.

Appendix 2

Appendix 2 Service Factor.

Machine Type	**Service**	**Intermittent (Up to**	**Normal (8–16**	**Continuous (16–**
ACMotors:NormalTorque		1.0	1.2	1.4
AC Motors: High Torque		1.2	1.4	1.6
DCMotors: Shunt Wound		1.1	1.3	1.5
Engines:Multiple Cylinder		1.3	1.5	1.7
Display,Dispensing		1.2	1.4	1.6
Appliances, Sweepers		1.3	1.5	1.7
Agitators for Liquids		1.4	1.6	1.8
Compressor: Centrifugal		1.5	1.7	1.9
Brick Machinery		1.6	1.8	2.0
Centrifuges		1.7	1.9	2.1
Blowers:Positive		1.8	2.0	2.2
Compressors: Reciprocating		1.9	2.1	2.3

Appendix 3

Appendix 3.8-mm Pitch GT Drive Selection Table.

Sprocket Combinations					Center Distance, Inches														
DriveR No. of Grooves	DriveR Pitch Diameter (Inches)	DriveN No. of Grooves	DriveN Pitch Diameter (Inches)	Speed Ratio	384-8MGT PL 15.118 48 teeth	480-8MGT PL 18.898 60 teeth	560-8MGT PL 22.047 70 teeth	600-8MGT PL 23.622 75 teeth	640-8MGT PL 25.197 80 teeth	720-8MGT PL 28.346 90 teeth	800-8MGT PL 31.496 100 teeth	840-8MGT PL 33.071 105 teeth	880-8MGT PL 34.646 110 teeth	920-8MGT PL 36.220 115 teeth	960-8MGT PL 37.795 120 teeth	1040-8MGT PL 40.945 130 teeth	1064-8MGT PL 41.890 133 teeth	1120-8MGT PL 44.094 140 teeth	1160-8MGT PL 45.669 145 teeth
26	2.607	64	6.416	2.462					5.16	6.82	8.45	9.25	10.06	10.86	11.66	13.25	13.73	14.84	15.63
29	2.907	72	7.218	2.483						5.82	7.49	8.30	9.12	9.92	10.73	12.33	12.81	13.93	14.72
32	3.208	80	8.020	2.500							6.48	7.32	8.15	8.97	9.78	11.40	11.88	13.00	13.81
36	3.609	90	9.023	2.500									6.86	7.71	8.55	10.19	10.68	11.81	12.62
22	2.206	56	5.614	2.545			4.56	5.40	6.22	7.85	9.45	10.25	11.05	11.85	12.64	14.23	14.70	15.81	16.61
44	4.411	112	11.229	2.545														9.12	9.96
25	2.506	64	6.416	2.560					5.22	6.89	8.52	9.32	10.13	10.93	11.73	13.32	13.80	14.91	15.71
28	2.807	72	7.218	2.571						5.88	7.55	8.37	9.19	9.99	10.80	12.40	12.88	14.00	14.80
35	3.509	90	9.023	2.571									6.93	7.77	8.61	10.25	10.75	11.88	12.69
56	5.614	144	14.437	2.571															
31	3.108	80	8.020	2.581							6.54	7.38	8.22	9.03	9.85	11.47	11.95	13.07	13.88
34	3.409	90	9.023	2.647									6.99	7.84	8.68	10.32	10.81	11.95	12.76
24	2.406	64	6.416	2.667					5.29	6.96	8.59	9.39	10.20	11.00	11.80	13.39	13.87	14.98	15.78
27	2.707	72	7.218	2.667						5.95	7.62	8.44	9.25	10.06	10.87	12.47	12.95	14.07	14.87
30	3.008	80	8.020	2.667							6.61	7.45	8.28	9.10	9.92	11.54	12.02	13.14	13.95
42	4.211	112	11.229	2.667													8.04	9.24	10.09
53	5.314	144	14.437	2.717															
33	3.308	90	9.023	2.727									7.05	7.90	8.74	10.39	10.88	12.02	12.83
29	2.907	80	8.020	2.759							6.67	7.51	8.35	9.17	9.99	11.61	12.09	13.21	14.02
26	2.607	72	7.218	2.769						6.01	7.68	8.50	9.32	10.13	10.94	12.54	13.02	14.14	14.94
40	4.010	112	11.229	2.800													8.16	9.37	10.22
32	3.208	90	9.023	2.813									7.12	7.97	8.81	10.46	10.95	12.09	12.90
28	2.807	80	8.020	2.857							6.74	7.58	8.41	9.24	10.06	11.67	12.16	13.28	14.09
39	3.910	112	11.229	2.872													8.23	9.44	10.29
25	2.506	72	7.218	2.880						6.07	7.75	8.57	9.39	10.20	11.01	12.61	13.10	14.21	15.01
50	5.013	144	14.437	2.880															
31	3.108	90	9.023	2.903									7.18	8.03	8.87	10.52	11.02	12.16	12.97
22	2.206	64	6.416	2.909					5.41	7.09	8.72	9.53	10.34	11.14	11.94	13.53	14.01	15.13	15.92
38	3.810	112	11.229	2.947													8.29	9.50	10.35
27	2.707	80	8.020	2.963							6.80	7.64	8.48	9.30	10.12	11.74	12.23	13.35	14.16
24	2.406	72	7.218	3.000						6.14	7.82	8.64	9.46	10.27	11.08	12.68	13.17	14.28	15.08
30	3.008	90	9.023	3.000								6.36	7.24	8.10	8.94	10.59	11.09	12.22	13.04
48	4.812	144	14.437	3.000															
37	3.709	112	11.229	3.027												7.81	8.35	9.56	10.42
26	2.607	80	8.020	3.077							6.86	7.71	8.55	9.37	10.19	11.81	12.30	13.42	14.23
29	2.907	90	9.023	3.103								6.42	7.31	8.16	9.01	10.66	11.15	12.29	13.11
36	3.609	112	11.229	3.111												7.88	8.41	9.63	10.48
46	4.612	144	14.437	3.130															
25	2.506	80	8.020	3.200							6.93	7.77	8.61	9.44	10.26	11.88	12.37	13.50	14.30
35	3.509	112	11.229	3.200												7.94	8.47	9.69	10.55
28	2.807	90	9.023	3.214								6.48	7.37	8.22	9.07	10.73	11.22	12.36	13.18
22	2.206	72	7.218	3.273						6.27	7.95	8.77	9.59	10.41	11.22	12.82	13.31	14.43	15.23
44	4.411	144	14.437	3.273															
34	3.409	112	11.229	3.294												8.00	8.54	9.75	10.61
112																			
24	2.406	80	8.020	3.333							6.99	7.84	8.68	9.50	10.33	11.95	12.44	13.56	14.37
27	2.707	90	9.023	3.333								6.54	7.43	8.29	9.14	10.79	11.29	12.43	13.24
33	3.308	112	11.229	3.394												8.06	8.60	9.82	10.67
42	4.211	144	14.437	3.429															
26	2.607	90	9.023	3.462								6.61	7.49	8.35	9.20	10.86	11.35	12.50	13.31
32	3.208	112	11.229	3.500												8.12	8.66	9.88	10.74
25	2.506	90	9.023	3.600								6.67	7.56	8.42	9.27	10.93	11.42	12.57	13.38
40	4.010	144	14.437	3.600															
31	3.108	112	11.229	3.613												8.18	8.72	9.94	10.80
22	2.206	80	8.020	3.636							7.12	7.97	8.81	9.64	10.46	12.09	12.58	13.70	14.51
39	3.910	144	14.437	3.692															
30	3.008	112	11.229	3.733												8.24	8.78	10.01	10.87
24	2.406	90	9.023	3.750								6.73	7.62	8.48	9.33	10.99	11.49	12.63	13.45
38	3.810	144	14.437	3.789															
29	2.907	112	11.229	3.862												8.30	8.84	10.07	10.93
37	3.709	144	14.437	3.892															
28	2.807	112	11.229	4.000												8.36	8.91	10.13	10.99
36	3.609	144	14.437	4.000															
22	2.206	90	9.023	4.091							5.92	6.85	7.74	8.61	9.46	11.12	11.62	12.77	13.59
35	3.509	144	14.437	4.114															
27	2.707	112	11.229	4.148												8.42	8.97	10.20	11.06
34	3.409	144	14.437	4.235															
26	2.607	112	11.229	4.308												8.48	9.03	10.26	11.12
33	3.308	144	14.437	4.364															
25	2.506	112	11.229	4.480												8.54	9.09	10.32	11.19
32	3.208	144	14.437	4.500															
31	3.108	144	14.437	4.645															
24	2.406	112	11.229	4.667												8.60	9.15	10.38	11.25
30	3.008	144	14.437	4.800															
29	2.907	144	14.437	4.966															
22	2.206	112	11.229	5.091												8.72	9.27	10.51	11.38
28	2.807	144	14.437	5.143															
27	2.707	144	14.437	5.333															
26	2.607	144	14.437	5.538															
25	2.506	144	14.437	5.760															
24	2.406	144	14.437	6.000															
22	2.206	144	14.437	6.545															
Length Factor*					0.70	0.80	0.80	0.80	0.90	0.90	0.90	0.90	0.90	1.00	1.00	1.00	1.00	1.00	1.00

Note: 25, 27, 29, 31, 33, 35, 37, 39, 42, 46, 50 and 53 groove sprockets are only available as stock products in 20 and 30mm widths.

Appendix 4

Aciers recommandés pour la construction des roues dentées cylindriques et coniques des réducteurs

Matériau	DIN	STAS	Traitement thermique ou thermochimique	Dureté		Résistance à la rupture, σ_r (N/mm^2)	Limite d'élasticité, σ_c (N/mm^2)	Résistance limite à la fatigue au pied de la dent $\sigma_{f\lim}$ (N/mm^2)	Pression hertzienne limite à la fatigue, $\sigma_{H\lim}$ (N/mm^2)
				noyau (HB)	flanc (HRC)				
OL 50	Fe 490-2 (St 50-2)	500/2-80	Normalisation	$HB = 150 \div 170$		$500 \div 620$	$270 \div 300$	$0,4HB + 100$	$1,5HB + 120$
OL 70	Fe 690-2 (St 70-2)	500/2-80	Normalisation	$HB = 200 \div 220$		$700 \div 850$	$340 \div 370$	$0,4HB + 100$	$1,5HB + 120$
OLC 45*	C 45	880-88	Amélioration	$HB = 220 \div 260$		620	360	$0,4HB + 140$	$1,5HB + 200$
			Trempe après chauffage à la flamme ou CIF	$200 \div 260$	$50 \div 57$			$160 \div 170$	$20HRC + 10$
OLC 55	C55	880-88	Amélioration	$HB = 200 \div 300$		720	420	$0,4HB + 140$	$1,5HB + 200$
			Trempe après chauffage à la flamme ou CIF	$200 \div 300$	$50 \div 57$			$180 \div 190$	$20HRC + 20$
41 MoCr 11	42 CrMo 4	791-88	Amélioration	$HB = 270 \div 320$		950	750	$0,4HB + 155$	$1,8HB + 200$
			Trempe après chauffage à la flamme ou CIF	$270 \div 320$	$50 \div 57$			$230 \div 290$	$20HRC + 60$
			Nitruration	$270 \div 320$	$52 \div 60$			$250 \div 350$	$20HRC$
40 Cr 10	41 Cr 4	791-88	Amélioration	$HB = 240 \div 340$		1000	800	$0,4HB + 155$	$1,8HB + 200$
			Trempe après chauffage à la flamme ou CIF	$240 \div 340$	$50 \div 57$			$230 \div 290$	$20HRC + 60$
			Nitruration	$240 \div 340$	$50 \div 57$			$250 \div 350$	$20HRC$
34 MoCrNi 15	34 CrNiMo 6	791-88	Amélioration	$HB = 310 \div 330$		1100	900	$0,4HB + 155$	$1,8HB + 200$
OLC 15*	C 15	880-88	Cémentation	$120 \div 140$	$55 \div 63$	390	280	$140 \div 150$	$24HRC$
21 MoMnCr 12	20 CrMo 5	791-88	Cémentation	$300 \div 350$	$55 \div 63$	1100	850	$390 \div 460$	$25,5HRC$

ملخص المذكرة

تهدف هذه المذكرة إلى دراسة ميكانيزم المشي وميكانيزم الكنس لجهاز صيانة الطرق متعدد الوظائف من الجانب الميكانيكي. قمنا بعمل مخطط سينماتيكي وأجرينا حسابات السينماتيكية لكل من ميكانيزم المشي وميكانيزم الكنس لتحديد الحركات والسرعات والاتجاهات .بالإضافة إلى ذلك، أجرينا حسابات طاقوية لتحديد القدرة والعزم المطلوب لكل ميكانيزم لضمان توفير الطاقة اللازمة لأداء المهام بكفاءة. تم تحليل القوى المؤثرة على الأعمدة والتروس من خلال تحليل ستاتيكي، مما ساعد في تحديد النقاط الحرجة والإجهادات الميكانيكية على المكونات المختلفة.

تم نمذجة ميكانيزم المشي وميكانيزم الكنس باستخدام برنامجSoli Works ، حيث قمنا بإنشاء نماذج ثلاثية الأبعاد لكل ميكانيزم. بعد ذلك، قمنا بإجراء محاكاة للقوى المؤثرة على ميكانيزم الكنس باستخدام طريقة العناصر المحدودة لتحليل الإجهادات والتشوهات. ساعدت هذه المحاكاة في التحقق من متانة التصميم وتحديد المناطق التي تحتاج إلى تعزيز أو تحسين.

تهدف هذه الدراسة إلى تحسين تصميم ميكانيزم المشي وميكانيزم الكنس لضمان الأداء الأمثل والكفاءة في عمليات الصيانة. من خلال تحليل شامل واستخدام أدوات هندسية متقدمة، نسعى إلى تعزيز كفاءة النظام وتحقيق نتائج أفضل في صيانة الطرق، مما يساهم في تمديد عمر الطرق وتحسين السلامة المرورية.

Summary of the Memorandum

This memorandum aims to study the walking mechanism and the sweeping mechanism of a multi-functional road maintenance machine from a mechanical perspective. A kinematic diagram was created and kinematic calculations were conducted for both the walking mechanism and the sweeping mechanism to determine movements, speeds, and directions.

Additionally, energy calculations were performed to determine the power and torque required for each mechanism to ensure the necessary energy is provided to perform the tasks efficiently. The forces acting on the shafts and gears were analysed through static analysis, helping to identify critical points and mechanical stresses on various components. The walking and sweeping mechanisms were modelled using Solid Works software, creating three-dimensional models for each mechanism. Subsequently, a simulation of the forces acting on the sweeping mechanism was conducted using the finite element method to analyse stresses and deformations. This simulation helped verify the robustness of the design and identify areas needing reinforcement or improvement. This study aims to improve the design of the walking and sweeping mechanisms to ensure optimal performance and efficiency in maintenance operations. Through comprehensive analysis and the use of advanced engineering tools, we seek to enhance system efficiency and achieve better results in road maintenance, contributing to longer road lifespans and improved traffic safety.

Résumé du Mémoire

Ce mémoire vise à étudier le mécanisme de marche et le mécanisme de balayage d'une machine de maintenance routière multifonctionnelle du point de vue mécanique. Un schéma cinématique a été créé et des calculs cinématiques ont été effectués pour le mécanisme de marche et le mécanisme de balayage afin de déterminer les mouvements, les vitesses et les directions. De plus, des calculs énergétiques ont été réalisés pour déterminer la puissance et le couple requis pour chaque mécanisme afin de garantir l'apport de l'énergie nécessaire à l'accomplissement des tâches de manière efficace. Les forces agissant sur les arbres et les engrenages ont été analysées par une analyse statique, ce qui a aidé à identifier les points critiques et les contraintes mécaniques sur les différents composants. Les mécanismes de marche et de balayage ont été modélisés en utilisant le logiciel Soli Works, créant des modèles tridimensionnels pour chaque mécanisme. Par la suite, une simulation des forces agissant sur le mécanisme de balayage a été réalisée en utilisant la méthode des éléments finis pour analyser les contraintes et les déformations. Cette simulation a aidé à vérifier la robustesse du design et à identifier les zones nécessitant un renforcement ou une amélioration. Cette étude vise à améliorer la conception des mécanismes de marche et de balayage afin d'assurer une performance optimale et une efficacité dans les opérations de maintenance. Grâce à une analyse complète et à l'utilisation d'outils d'ingénierie avancés, nous cherchons à améliorer l'efficacité du système et à obtenir de meilleurs résultats dans la maintenance des routes, contribuant à une durée de vie prolongée des routes et à une sécurité routière améliorée.