



**Addis Ababa University**

**Addis Ababa Institute of Technology**

**School of Mechanical & Industrial Engineering**

**Mechanism Design for Simultaneous Removal and Installation of  
All Wheels of Passenger Car: Synthesis, Analysis and Simulation**

A Thesis Submitted to the School of Graduate Studies of Addis Ababa  
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Degree of Master of Science in Mechanical Engineering (Mechanical Design)

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## **Declaration**

I hereby declare that the work which is being presented in this thesis entitled “Mechanism Design for Simultaneous Removal and Installation of All Wheels of Passenger Car: Synthesis, Analysis and Simulation” is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis have been duly acknowledged.

Natenael Fantaye

\_\_\_\_\_  
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This is to certify that the above declaration made by the candidate is correct to the best of my Knowledge.

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## Abstract

This thesis is about the design of a novel mechanism that will be used in tire repair shops for installing/removing all wheels of a passenger car at the same time. Numerous important repair activities in such shops require all wheel removal. Such activities include all-tire replacement, tire rotation, wheel balancing, wheel alignment and associated suspension repair and brake inspection and service. The work technicians perform to accomplish such activities, although assisted by modern tools and equipment still requires manual work as there is lack of machine or equipment for simultaneous all wheel removal/installation. Currently, technicians that work in tire repair shops are exposed to various musculoskeletal injuries such as strains, sprains and overuse injuries from manual activities such as lifting and maneuvering of wheels, repetitive motion such during work having to work and dirty and noisy environment. The methodology followed includes identifying the functions of the device to be, decomposing them to sub functions, generating alternative concepts to meet the decomposed functions and embodiment design of components assemblies and modules. Part of the resulting overall mechanism is a symmetric 16-bar mechanism that performs the wheel-positioning task. This mechanism required a special attention because its motion is complex. Therefore, as a graduate-level work, this thesis rigorously analyzes the motion of every link in this particular mechanism and every joint that connects the links by using specialized kinematic and dynamic multibody simulation process. The results of the simulation part are position, velocity, acceleration and joint force values – data critically important for further design decisions. Results of the synthesis part is geometric shapes and dimensions of 31 major components and 5 major modules that create the overall mechanism. Geometry and dimension of every component, how they assemble, and their relative motion with respect to one another is thoroughly worked out to the point that further steps of the design process (i.e. detail design) have been reduced into a routine engineering task of sizing cross sections of components and selecting materials to prevent component failure during service. In conclusion, the resulting design solution enables simultaneous all-wheel removal/installation without a need for manual work in the process thereby removing the problems on technicians mentioned above. Also, the complex 16-bar mechanism's motion is fully characterized by determining all motion parameters that describe its motion. More importantly, significant dynamic forces that result from accelerating masses of the links are determined exhaustively through multibody simulation. Having these force data is crucial to further steps in the design process as these forces must be taken into consideration in subsequent stress and deflection analyses.

**Keywords:** mechanism design, linkage synthesis, kinematic and dynamic simulation of mechanisms

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## List of Acronyms

<b>L</b>	Number of Links in a Mechanism
<b>J<sub>1</sub></b>	Number of One Degree of Freedom (Full) Joints in a Mechanism
<b>J<sub>2</sub></b>	Number of Two Degree of Freedom (Half) Joints in a Mechanism
<b>Ψ<sub>n</sub></b>	Relative Coordinate Angle
<b>AB</b>	Position Vector Between Points A and B with Tail at A and Head at B
<b> AB </b>	Magnitude of Position Vector <b>AB</b>
<b>L<sub>n</sub></b>	Length of the n <sup>th</sup> Link in a mechanism
<b>T<sub>O</sub></b>	Torque About an Axis Through Point O
<b>F</b>	Force
<b>W</b>	Weight
<b>d</b>	Perpendicular Distance Between a Vector and a Point
<b>P</b>	Power
<b>ω</b>	Angular Velocity
<b>V</b>	Linear Velocity

<b>t</b>	Time
<b><math>\alpha</math></b>	Angular Acceleration
<b>A</b>	Linear Acceleration
<b>DOF</b>	Degrees of Freedom
<b>CAD</b>	Computer Aided Design
<b>SUV</b>	Sport Utility Vehicle

# Chapter One

## Introduction

### 1.1. Background

The design of mechanisms and machines is one of the most important and fundamental tasks in mechanical engineering. Fundamentally, machines and mechanisms are designed in order to perform activities that are otherwise carried out by manual labor or animal power. In general, the latter have drawbacks of being excessively tiresome, inefficient, unproductive, unreliable, and time taking. Therefore, mechanization is the result of the key engineering activity – engineering design – that solves such problems. [1], [2]

Although a wide variety of activities have been mechanized in the current world, much remains to be done. Still, there are countless day – to – day tasks manually performed by millions of people in various sectors throughout the world. [5] This is especially true in less economically developed countries. One of the wide range of laborious tasks people carry out is maintenance and repair work. [2] Particularly, in the maintenance of heavy machinery such as motor vehicles, a lot of physically demanding activities need to be completed by mechanics and technicians. In automotive workshops, one task that must be repeatedly done is that of installing and removing the wheels of the vehicle to and from the hub of the axle. Since all ground vehicles have at least two wheels, and since the most numerous ground vehicles are four wheeled vehicles, it is easy to imagine the frequency with which wheel removal and installation needs to be executed. [3]

There are many cases in which the removal and installation of the wheels of a vehicle need to be made at the same time. To name a few, wheel removal and installation during wheel balancing, complete suspension repair, complete brake repair and tire rotation are cases that require the removal and installation of all wheels of the vehicle. In addition, such task is also repeatedly carried out in automotive assembly plants. Therefore, the design of a mechanism to perform task simultaneously has a potential to be of great assistance toward mechanizing the activity. [3]

To the knowledge of the submitter of this document, there is no mechanism currently being used for the simultaneous removal and installation of all wheels of a vehicle. This statement is based on an extensive internet search done up to the date of submission of this document. Therefore, this study is based on the design of such mechanism to solve the above – mentioned problem. This study focuses on the design of a mechanism for installing and removing all wheels of a passenger car at the same time. The mechanism design covers the required steps starting from the understanding of the problem, through the synthesis of appropriate mechanism, to the detailed engineering analysis of the synthesized mechanism using modern simulation tools, and suggestion of the completed mechanism design through standard documentation.

## **1.2. Statement of the Problem**

Mechanization is the process of changing work that is previously done manually to work that is assisted or completely performed by machines or mechanisms. It is one of the primary purposes of engineering design. Throughout human history, countless tasks that were performed by human or animal effort have been transformed to being done by machinery. When previously hand – performed tasks are replaced by mechanization, the advantage is not only the reduction of human burden but also the considerable increase in efficiency of the process itself. [4]

In today’s world, among the most labor-intensive tasks man has to encounter is the job of a mechanic or technician – especially those who are working with heavy machinery. For technicians in the automotive maintenance sector, despite the invention and development of many modern tools, equipment and machinery to make their jobs easier, many tasks in their job still remain to be very physically demanding [46]. There are still many challenging aspects of tasks such as lifting and maneuvering heavy part [47], repetitive motions that cause strain and fatigue [48], physical fatigue and discomfort from working in confined spaces [49], standing for extended long periods of time [49] and working in hot, dirty noisy environments [50].

One category of automotive maintenance is tire repair. Throughout the world, professional tire repair and maintenance of vehicles is carried out in tire repair shops. A tire repair shop is a maintenance workshop that specializes in sale, installation, maintenance and repair of tires for various types of vehicles including cars, trucks, SUVs, etc. The operations of a typical tire repair shop include tire sales, tire installation, tire repair, wheel alignment and wheel balancing. These activities are manually performed on vehicles by tire shop technicians with the help of various tools and equipment. [51], [52], [53]

In tire repair shop operations, there is a need to frequently remove/install all wheels from/to a vehicle. This includes in the case of tire replacement – when customers purchase a full set of new tires [54], in the case of tire rotation [54], [55], in the case of wheel balancing [54], [55], in the case of wheel alignment – particularly in case of significant alignment issues or if other suspension components need to be inspected [55], and full brake service [56]. In all of these cases, tire repair technicians need to remove/ install all wheels of a vehicle.

The tasks technicians have to perform in tire repair shops are often very physically demanding and fatiguing especially considering the volume of vehicles that come into tire repair shops [57]. Lifting and maneuvering wheels of even smaller vehicles into and out of position is a tiresome task for technicians [57]. Repetitive motions that are required during tire installation/removal such as kneeling, squatting, reaching often create risks of muscle strain, and musculoskeletal injuries such as strains, sprains and overuse injuries [58]. Technicians in busy tire shops are often required to work quickly to keep up with the workload especially in periods of high demand; this creates physical fatigue and exhaustion [57], [58]. In addition, tire shop technicians often work in dirty and noisy environments that cause physical discomfort and aggravates fatigue [50], [58], [59]. They also face risks of injuries such as bruises and cuts [49], [59].

Although there are tools and equipment in tire shops to assist the tasks performed by technicians such as impact wrenches, lifts and jacks tire changers, wheel balancers, alignment machines and tire carts and trolleys [40], [41], [54], [55]; there is no single machine or equipment to simultaneously remove/install all four wheels of a vehicle. If such a machine is developed, the

numerous disadvantages of technicians mentioned above would be eliminated or reduced. This is a technology gap identified by the literature review of this study.

Therefore, a solution has to be given to the above – mentioned problem. This solution encompasses the design of a mechanism for installing/removing all wheels of a passenger car at the same time, starting from the understanding of the problem, through the synthesis of appropriate mechanism, to the detailed engineering analysis of the synthesized mechanism.

### **1.3. Objectives**

#### **1.3.1. General Objective**

The general objective of this study is to design a mechanism to be used for the removal and installation of all wheels to and from a vehicle simultaneously.

#### **1.3.2. Specific Objective**

The specific objectives of this thesis are:

- To synthesize new mechanism for simultaneous removal/installation of all wheels of a four-wheeled vehicle
- To make position, velocity and acceleration analysis for the mechanism through kinematic simulation
- To make force analysis of the mechanism through dynamic simulation

### **1.4. Scope of the Study**

The mechanism to be designed is for application on passenger cars. Since passenger cars vary in size, particularly in their wheelbase and track width measurements, the mechanism should be

adjustable in order to be able to serve all types between the largest and the smallest wheelbase and track width dimensions. The smallest and largest dimensions of wheelbase the design must cover are 1686 mm and 3112 mm respectively. Similarly, the smallest and largest dimensions of track width the design must cover are 1234 mm and 2029 mm respectively. This is based on literature review done.

In addition to this, the design of geared or other similar mechanism to remove all lug nuts or bolts from a wheel simultaneously is a basic machine design exercise, which is not included in this study due to time limitation. Moreover, the detailed design of vehicle lifting jacks is not dealt with here for the reason that, as per reviewed literature indicated, vehicle lifting is a relatively mature technology the design details of which are clearly known to engineering designers and scientific community.

As a graduate-level thesis in mechanical engineering, this work emphasized on presenting a solution to the identified problem using the engineering design process. In doing so, much focus is given on thoroughly understanding the problem, studying in detail the function of the solution-to-be, and, synthesizing the design solution accordingly.

This emphasis is due to the fact that, since this is an original design problem, once a proper solution is synthesized, the rest of the engineering tasks required to make this product ready for manufacturing such as material selection and stress analysis to decide on final cross sections of load-bearing components, detail analysis of machine elements such as shafts, gears or fasteners, sizing of components to withstand deflections etc., are straightforward and routine mechanical design tasks that can be done based on the results of this work.

An exception to this emphasis, however, is analysis of the synthesized symmetric 16-bar mechanism. The motion of this mechanism's links is complex and the determination of its characteristics requires distinct attention and specialized multibody kinematic and dynamic analysis. Pertaining to this fact, a thorough analysis of the motion of each of the members in this mechanism and determination of critical parameters needed for engineering design purposes is done in detail in this work.

## **1.5. Limitations of the Study**

The engineering design solution presented in this work – the simultaneous removal and installation of all wheel removal/installation mechanism – is to be used for four wheeled passenger vehicles only. Wheeled vehicles classified outside of the category of passenger cars cannot use this specific design solution although the design concept invented in this thesis can be scaled up or down and slightly modified to suit these vehicles through an easier synthesis work derived from the results presented here. The limitations apply for vehicles whose number of wheels is greater or lesser than four wheels.

## **1.6. Organization of the Paper**

In this thesis, the first chapter discusses the introduction to the main concepts and background of the topic. This introduction chapter includes statement of the problem, general and specific objectives of the study, and scope and limitations of the study. Next, review of relevant literature is the second chapter. The third chapter is on research methods and conditions. It includes the synthesis of the mechanism and the analysis of the mechanism including mathematical modeling and simulation. After this, results and discussion is the chapter that follows. This chapter presents and discusses the results of the motion simulation. In the fifth chapter, conclusions, recommendations and future work are included. Finally, a list of references that are used throughout this thesis followed by appendix is presented at the end of this document.

## **Chapter Two**

### **Literature Review**

As it is stated in the previous chapter, there is no mechanism currently existing to perform simultaneous wheel removal and installation of wheels from vehicles. Therefore, no literature exists to which this work can be compared directly. In this work, literature review is mainly used as a tool to gather necessary information for the design of this new mechanism. One of the most important information that needs to be known before this mechanism can be successfully synthesized is to know the range of the sizes of the vehicles this mechanism can be used for. Clearly, it can not be used for all vehicles in existence. However, it can also not be used for only one particular vehicle type only. It must be able to be used for a defined range of vehicles. Assessing the different vehicle types and identifying how they are sorted, as well as which vehicle dimensions are the important parameters for the design task ahead is the central objective of the first part of this literature review. In addition, the steps in the process of wheel removal and installation are looked at. Section 2.1 of this chapter presents these reviews.

The second reason for performing this literature review is to assess what the current practice in mechanism synthesis as well as analysis looks like. This is important because the majority of the tasks of this study are involved, in great detail, in the synthesis and analysis of the intended mechanism. The second part of this literature review, section 2.2 presents this review. Finally, a statement of the originality of this project is presented in section 2.3.

#### **2.1. Passenger Car Categorization with Respect to Wheel Removal and Installation**

An automobile is defined as a motor vehicle with wheels that is used for the transportation of people and goods on roads. According to the majority of definitions, an automobile or a car is a

four – wheeled vehicle. [6], [7] There are various systems of classification of automobiles put in place by different governments and related governing bodies. Two of the most widely used systems are the European car classification system and the United States car classification system.

The first one is used in countries of the European Union while the latter is implemented in North America. In both systems, the major criteria used for the classification of vehicles are the parameters vehicle dimensions – namely wheelbase and track width, vehicle weight and engine size. [8] Since the parameters directly related to this study are the dimensions of the vehicle, the first part of this literature review will be more focused on it. In addition to wheelbase and track width, vehicle weight as a classification parameter is also assessed because lifting the vehicle is a necessary step in the process of changing a vehicle's wheels and the subsequent design of the mechanism to accomplish the task must consider the vehicle weight. Table – 1 below shows the different schemes of classifications of automobiles using equivalent US and European standards with the typical dimensions of wheelbase and width plus the parameter of weight.

The removal and installation of a wheel from an automobile requires several steps. When the wheel is to be removed in a conventional way, a series of tools are required. These include an automotive jack to raise the automobile from the ground, a lug wrench or an impact wrench to remove the lug nuts or bolts, wheel-blocking devices and a pry bar to remove the wheel caps if present on the specific model. The conventional steps of removing a wheel are untightening the lug bolts or nuts, raising the vehicle from the ground, completely removing the lug bolts or nuts and removing the wheel from the vehicle. [8]

**Table – 1: European and US Classification Standards for Automobiles (Adopted from [9-14])**

US Size Classification	European Size Classification	Typical Dimensions and Parameters		
		Wheelbase (mm)	Width (mm)	Weight (Kg)
—	Quadricycle	1,686	1,234	880
Minicompact	A-segment mini cars	2,385	1,595	1,019
Subcompact	B-segment small cars	2,493	1,734	1,113
Compact	C-segment medium cars	2,736	1,801	1,305
Mid-size	D-segment large cars	2,945	1,875	1,976
Large	E-segment executive cars	2,840	1,900	1,669
Large	F-segment luxury cars	2,992	1,948	1,977
Two-seater	S-segment sports coupés	2,470	1,864	1,405
Minivan	M-segment multipurpose cars	3,089	2,022	2,262
SUV	J-segment sport utility cars	3,112	2,029	2,747

During wheel removal, one common challenge is difficulty in untightening the lug bolts or nuts due to sticking with the studs or threaded holes in the hub. When this happens, a breaker bar is used to overcome the sticking. When using an impact wrench, repeated blows or changing the direction between tightening and untightening helps to break the sticking. The procedure of installation is the reverse of the removal process. After installing the wheel on the vehicle, automobile manufacturers suggest to tighten the lug bolts or nuts using a torque wrench to the specified torque levels. In order to ensure that the tightening force is uniformly distributed among the lug nuts or bolts, these fasteners are tightened in an alternating pattern – a pattern in which opposite fasteners are tightened in sequence. In the conventional procedure of wheel removal and installation, the process of putting the wheel on and off is performed by manual labor. [3], [8]

Key features in the process of automobile wheel removal and installation are lug nuts or bolts. A lug nut or bolt is a threaded fastener that is used for the purpose of securing a wheel on a vehicle. Some vehicle manufacturers use lug nuts while others use lug bolts. In the first case, which is more common, the nuts secure the wheel to wheel - studs that have threads. The studs are connected to the axis of the vehicle. In the latter case, lug bolts are used to attach the wheels by being screwed into threaded holes in the hub, which is part of the vehicle's axis. [15], [16]

Lug bolts and nuts have conical shaped tapers on their faces contacting the rim of the wheel. The purpose of this taper is to assist in the accurate centering of the wheel with respect to the axle. In addition, it also prevents loosening of the fastener as the tire rotates. The common taper angle for lug nuts and bolts is 60 degrees. A spherical shape of the lug bolt or nut end face, in some designs, is used instead of the conical taper. [15] Usually, the specific king of face geometry must match the type of hole in the rim of the wheel to which the lug fastener is attached. [16] The most common hexagonal metric wrench sized to install or remove lug bolts and nuts are socket sizes of 17mm, 19mm and 21mm. Other less common sizes include 22mm and 23 mm socket sizes. [17]

There are a number of problems associated with the conventional method of vehicle wheel removal and installation. The first challenge is related to the manual force required to move the rim – tire assembly. The person performing the task must essentially carry the assembly to or from the vehicle, which is a highly demanding laborious activity. In addition, with the exception of the case

where an impact wrench is used, tightening or untightening the lug bolts or nuts also requires a considerable manual force input. The second problem is in relation to the time it requires to perform the task. In the conventional method of work, the process of wheel removal and installation must be done on each wheel at a time. This multiplies the effort required by four when removing or installing all wheels of a vehicle is needed. In addition, removing and installing all the lug bolts or nuts manually or even with the help of an impact wrench is a time consuming process. Therefore the design of the simultaneous all - wheel removal and installation mechanism discussed in this study has logical justification.

## **2.2. Review on Mechanism Synthesis and Analysis**

Often, the terms mechanism and machine are used interchangeably and, although there is a distinction between them, the distinction can sometimes be partially subjective and not always clear-cut. Their distinction is more based on intensity and less on type. A mechanism is defined as a device that converts a given input motion into an output motion that has some sort of desired pattern while creating or transmitting low magnitudes of forces and power. On the other hand, a machine is a mechanism or a set of mechanisms that create or transmit considerably high magnitudes of forces and power. If a certain mechanism operates at low speed and under small magnitudes of externally applied loads, it can be analyzed as a kinematically without regarding the forces, whereas, if its speed is high and the applied loads are large, it must be analyzed kinetically taking into consideration the forces involved. [19], [20], [21]

The engineering design of systems in motion must involve dynamic analysis and synthesis. Dynamic analysis consist of kinematic analysis, also called kinematics, and kinetic analysis, also known as kinetics. Kinematics is referred to as the study of motion of bodies without considering the forces involved in the bodies' motion. On the other hand, kinetics is the study of the motion of bodies taking into account the forces that either cause motion, are the result of motion, or both. Although both kinematics and kinetics cannot separately exist during the motion of a given system,

they are nonetheless considered individually during design for analysis purposes. Kinematics is performed prior to kinetic analysis. [21], [23]

The central objective of performing kinematic analysis in engineering design is first to generate the desired motions of the mechanical elements, and then calculate the positions, velocities and accelerations that will be imposed on the mechanical elements due to the generated motions. Once accelerations are determined, they are used to obtain forces that act on the elements due to the motion of the elements' mass applying Newton's second law of motion – the product of mass and acceleration is equal to force. These forces are called inertial forces of the elements. Inertial forces, combined with externally applied forces, are responsible for inducing stresses within the material out of which the mechanical elements are made. Therefore, the obtained forces will be used for stress analysis and material selection phase of the design process in order to select materials and adjust dimensions of the elements so that failure of the materials during their intended service life is prevented. [24], [25], [26]

The degree of freedom of a mechanical system is defined as the number of independent coordinates needed to explicitly define the systems' instantaneous position. An object in space has six degrees of freedom. It can rotate about each of the three perpendicular axes and it can translate about three of the axes, hence six degrees of freedom. Joints between objects reduce the degrees of freedom depending on the type of joint. A joint is a mechanical connection between two individual mechanical elements that allows only a certain type of relative motion between the elements. The types of joint commonly used in mechanical systems and their corresponding degrees of freedom are shown in table – 2. [18], [27], [28]

*Table – 2: Joint Types, Degrees of Freedom and Motion Types they Allow*

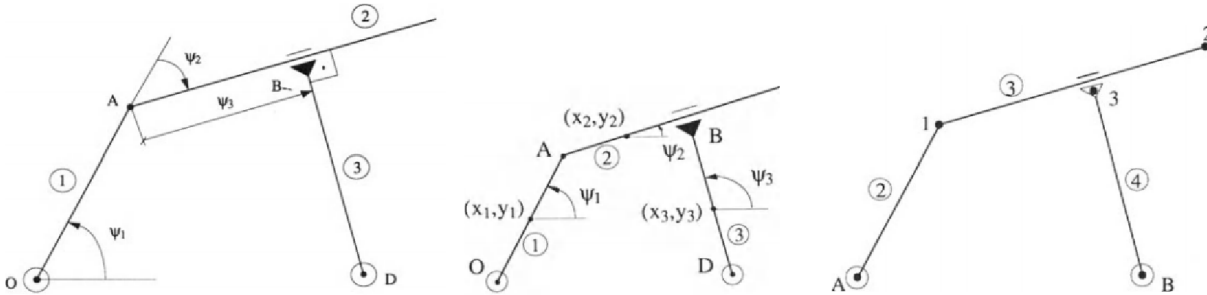
<b>Joint Type</b>	<b>Degrees of Freedom</b>	<b>Type of Motion Allowed by the Joint</b>
Revolute	1	1 Rotation
Prismatic	1	1 Translation
Helical	1	1 Translation/Rotation
Cylindrical	2	1 Rotation and 1 Translation
Spherical	3	3 Rotations
Planar	3	3 Translations

In kinematic and kinetic analysis, coordinate systems are the basis on which the dynamic parameters of motion are defined. Coordinates unambiguously define the position, velocity and acceleration of a mechanism at any given time during its motion. There are two types of coordinates namely dependent and independent coordinates. Independent coordinates are those coordinates whose number is equal to the number of degrees of freedom of the mechanism. If they are chosen to be used to represent the mechanism's motion in building the mathematical model, they offer the advantage of being minimal. [29], [30]

On the other hand, dependent coordinates are increased set of coordinates whose number is larger than that of the degrees of freedom of the mechanism. The choice of using dependent coordinates in the mathematical modeling of a mechanism is much better suited for analysis since the coordinates are not independent but rather interrelated through equations called constraint equations. The number of constraint equations is the difference between the number of dependent coordinates and the number of degrees of freedom. These constraint equations form the basis upon which the kinematic analysis of the mechanism is built. First and second order time derivatives of the constraint equations give the velocity and the acceleration equations respectively. [31], [32]

The predominant types of independent coordinates developed over the years to represent the motion of a given mechanism are relative coordinates, reference point coordinates and natural coordinates. A combination of these coordinates is also possible. Each The graphical

representation of the three coordinate system types using a planar four-bar mechanism as an example are represented in figure – 1. [18]



**Figure – 1:** Planar Four-Bar Crank-Slider Mechanism Represented in Relative (Left), Reference Point (Middle) and Natural (Right) Coordinates [18]

Once constraint equations are obtained for a given mechanism, the next step is to differentiate the equations with respect to time once to obtain velocity equations and twice to get acceleration equations. By using the initial conditions – in most cases the position, velocity and acceleration values of a driving link, the set of simultaneous equations can be solved to obtain the accelerations of the rest of the links in the mechanism. These accelerations are in turn used, from Newton’s second law, to obtain forces caused by the accelerating masses. [18], [31], [32]

In general, there are three types of solving such problems. The first one is analytical solution. This solution is accurate in determining the motion parameters described above. However, it suffers from the complexity of solving a set of simultaneous differential equations. In addition, it can only solve for the position, velocity and acceleration of the links in a mechanism for only one instant in time. The second is graphical approach. This is a trivial but tedious exercise. In the past, before the advent of computers, it was used to analyze mechanisms especially that are difficult to solve analytically. The graphical method, like its analytical counterpart, is limited to determining motion parameters for a single instant of time. The third and modern method is the use of computers to perform dynamic simulations. This simulation approach is a powerful method that is effective and efficient provided the model is built appropriately. Its advantage over the previous methods is not

only speed and ease but also that it can produce motion parameters for any desired length of time during the mechanism's motion – not just a single instantaneous point in time. [30], [31], [32]

## **2.3. Literature Review on Vehicle Lifting Technology**

### **2.3.1. Overview of Vehicle Lifting Technology**

Vehicle lifting refers to raising a vehicle from the ground by various means for a number of reasons. The technology of raising vehicles and the specific processes and techniques within it have various important applications that are distinctly known and that has developed for a long number of years. The first application is for automotive repair and maintenance purposes. The second application is in vehicle manufacturing. A third reason a vehicle is lifted from the ground is for tire removal and installation. Last but not least, vehicle lifting is performed for the application of transporting vehicles to and from different places.

Throughout the many years of its development, significant advancements and innovations have been made to improve the efficiency, safety, and flexibility of vehicle lifting systems. This section of the literature review of this thesis has made an attempt to explore the historical evolution of vehicle lifting technology, current innovations, challenges, and future trends that are being seen in vehicle lifting technology.

### **2.3.2. Historical Evolution of Vehicle Lifting Technology**

Historically, vehicle lifting technology has evolved from simple manual jacks to sophisticated hydraulic and pneumatic systems. The first vehicle lifting methods and equipment were designed to primarily use manual labor. One major problem on this approach is safety problems. After hydraulic lifts have been invented in the early 20th century, vehicle lifting has been significantly changed and the reliability of the methods have been enhanced. In addition to reliability, the use of hydraulic lifting systems created good benefits such as lifting capacity increase and better stability. [38]

### **2.3.3. Mechanical Scissor Mechanism Lifting Systems**

Mechanical scissor lifts are distinguished by the distinctive feature of their scissor-like mechanism, which provides vertical elevation or lifting action. In practice, the most common applications of these mechanical scissor lifting systems includes wheel/tire servicing, wheel alignment, and

vehicle repair and maintenance tasks. Although these type of lifts have been in use for a number of years, more recent advancements in scissor lift technology include the integration of wireless remote control systems and adjustable platforms to accommodate different vehicle sizes. Advancements in the field of mechatronics and processing power of embedded systems is helping this progress in their control system development. [39]

#### **2.3.4. Hydraulic Lifting Systems**

Hydraulic lifting systems are widely used in automotive workshops and garages due to their versatility and efficiency. These lifts utilize hydraulic fluid to generate the lifting force, allowing for smooth and controlled elevation of vehicles. Recent innovations in hydraulic lifting technology have focused on enhancing safety features, such as automatic safety locks and overload protection mechanisms. [40]

#### **2.3.5. Mobile Vehicle Lifting Systems**

Mobile vehicle lifting systems offer flexibility and mobility, allowing technicians to perform maintenance and repairs in various locations. These lifts often feature lightweight construction and quick setup capabilities, making them suitable for roadside assistance and emergency repairs. Innovations in portable lifting technology focus on improving stability and load-bearing capacity while minimizing weight and footprint. [41]

#### **2.3.6. Robotic Vehicle Lifting Systems**

Robotic vehicle lifting systems utilize robotics and automation technology to streamline vehicle lifting processes in manufacturing facilities and assembly lines. These lifts offer high precision and repeatability, reducing cycle times and improving productivity. Recent advancements in automated lifting technology include the integration of artificial intelligence for predictive maintenance and optimization of lifting sequences. [42]

#### **2.3.7. Vehicle Lifting Technology Challenges and Future Trends**

Despite the advancements in vehicle lifting technology, several challenges persist, including safety concerns, environmental impact, and cost constraints. Ensuring operator safety remains a top priority for manufacturers, leading to the development of advanced safety features and training programs. Additionally, the growing emphasis on sustainability has prompted the adoption of eco-friendly materials and energy-efficient lifting systems. [43]

Future trends in vehicle lifting technology are likely to focus on enhancing automation, connectivity, and adaptability. Integrated IoT (Internet of Things) sensors and cloud-based monitoring platforms will enable real-time performance tracking and predictive maintenance, minimizing downtime and optimizing operational efficiency [44]. Furthermore, the emergence of electrification and autonomous vehicles will drive the demand for specialized lifting solutions tailored to the unique requirements of electric drivetrains and self-driving technologies [45].

### **2.3.8. Conclusion on Vehicle Lifting Technology**

In conclusion, vehicle lifting technology has undergone significant advancements and innovations, ranging from hydraulic lifts to automated robotic systems. These advancements have improved efficiency, safety, and flexibility in automotive repair, maintenance, and manufacturing operations. However, challenges such as safety concerns and environmental impact remain, necessitating ongoing research and development efforts. Future trends in vehicle lifting technology are expected to focus on automation, connectivity, and sustainability, driven by the demand for smarter and more efficient lifting solutions in the era of electrification and autonomous vehicles.

## **2.4. Vehicle Wheel Installation and Removal Activities in Tire Repair Shops**

### **2.4.1. Physically-Demanding Nature of Vehicle Repair Activities by Technicians and Mechanics**

Even with technological advancements in the automotive industry, the job of automotive technicians can still be physically demanding, although the nature of the physical demands may have shifted somewhat. While some tasks may now involve less manual labor due to diagnostic tools and computerized systems, there are still many aspects of the job that require physical exertion. [46]

Specific aspects of a technician's job in vehicle repair and maintenance include but are not limited to:

- i. **Lifting and maneuvering heavy parts:** Automotive technicians often need to lift and move heavy components such as engines, transmissions, and suspension parts. Even with the assistance of lifts and other equipment, handling these components can still be physically demanding. [47]

- ii. **Repetitive motions:** Some tasks, such as loosening and tightening bolts, often require repetitive motions that can lead to strain or fatigue over time. [48]
- iii. **Working in confined spaces:** Automotive technicians frequently need to work in tight spaces within the engine bay or underneath the vehicle. This can require contorting their bodies into awkward positions, which can be physically taxing. [49]
- iv. **Standing for long periods:** Automotive technicians typically spend much of their day on their feet, whether they're performing inspections, repairs, or maintenance tasks. [49]
- v. **Environmental factors:** Technicians often need to work in environments that are hot, cold, dirty, or noisy, which can add to the physical demands of the job. This potentially causes adverse health and safety on them. [50]

From the above review, it is clearly visible that, while technological advancements have certainly changed the landscape of automotive repair and maintenance, the job of automotive technicians still involves a significant amount of physical work. More research and development work should be performed in order to develop tools, equipment and machinery that assist the job of technicians in their work.

#### **2.4.2. Tasks of Technicians in Tire Repair Shops**

A tire repair shop is a shop that specializes in the sale, installation, maintenance, and repair of tires for various types of vehicles, including cars, trucks, SUVs, motorcycles, and sometimes even bicycles. [51] The services provided by a tire shop typically include:

- i. **Tire Sales:** Tire shops offer a wide range of tires from different brands, sizes, and types to fit the needs of their customers. This includes tires for different weather conditions (e.g., all-season, winter, and summer tires), performance tires, off-road tires, and more. [51]
- ii. **Tire Installation:** When customers purchase new tires, the tire shop will mount and balance them on the vehicle's wheels. This involves removing the old tires, inspecting the wheels for damage, mounting the new tires, and balancing them to ensure smooth operation and minimize vibrations. [51], [52]

- iii. **Tire Maintenance:** Tire shops may offer services such as tire rotation, which involves moving tires from one position on the vehicle to another to ensure even wear. They may also perform tire inspections to check for signs of damage, uneven wear, or other issues that could affect tire performance and safety. [52]
- iv. **Tire Repair:** If a tire becomes punctured or damaged, tire shops can often repair it rather than replacing it entirely. Common tire repairs include patching punctures, fixing small tears or cuts, and resealing leaks. [52]
- v. **Wheel Alignment:** Many tire shops also offer wheel alignment services to ensure that the vehicle's wheels are properly aligned according to manufacturer specifications. Proper wheel alignment helps improve handling, tire wear, and fuel efficiency. [53]

### **2.4.3. Technicians in Tire Repair Shops**

A technician in a tire shop, often referred to as a tire technician or tire service technician, is responsible for performing these various tasks. Their duties may include assisting customers in selecting the right tires for their vehicles based on factors such as vehicle type, driving habits, and budget, mounting, balancing, and installing tires on vehicles using specialized equipment such as tire changers and wheel balancers, performing tire rotations, inspections, and repairs as needed to ensure optimal tire performance and safety, conducting wheel alignments using alignment equipment to adjust the angles of the wheels according to vehicle specifications, providing recommendations to customers regarding tire maintenance, tire care, and other related services to prolong tire life and enhance vehicle performance, overall, tire technicians play a crucial role in ensuring that vehicles are equipped with properly maintained and serviced tires, contributing to road safety and vehicle performance. [51], [52]

### **2.4.4. The need for Installing and Removing all Wheels of Passenger Vehicles in Tire Repair Shops**

Removing and installing all wheels from passenger cars may be necessary in tire repair shops for several reasons:

- i. **Tire Replacement:** When customers purchase a full set of new tires, all wheels need to be removed to install the new tires properly. This task is performed in order to ensure that the tires are mounted correctly and balanced to maintain vehicle safety and performance. [54]

- ii. **Tire Rotation:** Regular tire rotation is essential for even tire wear and prolonging the life of the tires. During a tire rotation service, all four wheels are typically removed and reinstalled in different positions on the vehicle. This helps ensure that each tire wears evenly over time, which can improve traction, handling, and fuel efficiency. [54], [55]
- iii. **Wheel Balancing:** If a vehicle experiences vibrations or uneven tire wear, it may indicate that the wheels are out of balance. In such cases, all wheels need to be removed and balanced using specialized equipment to distribute weight evenly around the wheel assembly. This helps prevent vibrations and ensures a smooth ride. [54], [55]
- iv. **Wheel Alignment:** While wheel alignment typically involves adjusting the angles of the wheels without removing them, in some cases, particularly if there are significant alignment issues or if other suspension components need to be inspected, all wheels may need to be removed to perform a comprehensive alignment service. [55]
- v. **Tire Inspection and Repair:** When performing a thorough tire inspection or repair, technicians may need to remove the wheels to access both the inside and outside of the tires. This allows them to properly assess the condition of the tires, identify any punctures, tears, or other damage, and perform necessary repairs. [54], [55]
- vi. **Brake Service:** In some cases, brake service may require removing the wheels to access brake components such as brake pads, rotors, calipers, and brake lines. This allows technicians to inspect, repair, or replace brake components as needed to ensure optimal braking performance and safety. [56]

As seen in the above review, the process of removing and reinstalling all wheels from passenger cars is necessary in tire repair shops to perform various maintenance and repair tasks, including tire replacement, rotation, balancing, alignment, inspection, and brake service.

#### **2.4.5. All Wheel Removal in View of Operator Labor-Intensive Tasks and Related Fatigue**

Performing the task of removing and installing all wheels from a vehicle in a tire repair shop can be physically demanding and fatiguing for technicians, especially considering the volume of vehicles that may come into the shop. [57] This can be more explained considering the various aspects of lifting and maneuvering, repetitive motion, continuous workload, working conditions and risk of injuries as follows.

With regard to lifting and maneuvering, wheels and tires can be heavy, even for smaller passenger vehicles considering not all technicians have equal physical strength – especially female and older age technicians. There are several instances during their work in which technicians may need to lift these heavy components multiple times throughout the day, which can strain the muscles of the arms, shoulders, and back. Maneuvering wheels and tires in and out of position also requires physical effort, particularly in tight spaces or when accessing wheels on the underside of the vehicle. [57]

With regard to repetitive motion, the process of removing and installing wheels involves repetitive motions such as bending, kneeling, squatting, and reaching. These repetitive movements can lead to muscle fatigue and strain over time, increasing the risk of musculoskeletal injuries such as strains, sprains, and overuse injuries. [58]

With regard to continuous workload, tire repair shops often have a steady flow of vehicles coming in for service, especially during peak times such as seasonal tire changes or inclement weather. Technicians may need to work quickly and efficiently to keep up with the workload, which can contribute to physical fatigue and exhaustion, particularly during long shifts or periods of high demand. [57], [58]

With regard to working conditions, tire repair shops can be hot, noisy, and dirty environments, which can further add to the physical strain and discomfort experienced by technicians. Working in these conditions for extended periods can aggravate fatigue and make it more challenging to perform tasks safely and effectively. [58], [59], [50]

With regard to risk of injury, In addition to muscle fatigue and strain, the physical demands of removing and installing wheels also increase the risk of acute injuries such as cuts, bruises, and crush injuries, particularly when working with heavy equipment or in fast-paced environments. [59], [49]

It is understandable from the above review that, while the task of removing and installing all wheels from a vehicle is essential for tire repair and maintenance, it can place significant physical demands on technicians, particularly in tire repair shops with a high volume of vehicles. It's essential for technicians to prioritize proper lifting techniques, take breaks as needed, and utilize ergonomic tools and equipment to minimize the risk of injury and fatigue. Additionally, tire shop

management should implement policies and procedures to support the health and safety of their technicians, including providing adequate staffing levels and scheduling sufficient rest periods.

#### **2.4.6. Tools, Machinery, and Equipment used by Tire Shops for all Wheel Removal and Installation**

Tire repair shops often utilize various equipment, mechanisms, and machinery to mechanize and streamline the tasks performed by technicians for the removal and installation of all wheels. These tools help to reduce the physical strain on technicians. [54], [55] Some of the equipment commonly used in tire repair shops includes:

- i. **Tire Changers:** Tire changers are machines designed to remove and install tires onto wheels. They use hydraulic power to assist with bead breaking, tire removal, and tire mounting. Automatic tire changers can significantly reduce the physical effort required to mount and demount tires compared to manual methods. Note that tire changers are for removing/installing the tire from the rim and not the entire wheel to/from the vehicle – a task which has to be done manually. [54], [55]
- ii. **Wheel Balancers:** Wheel balancers are used to ensure that the weight distribution around the wheel assembly is even, minimizing vibrations and improving ride quality. Modern wheel balancers feature automatic data input and electronic sensors, making the balancing process more efficient and less physically demanding for technicians. [55]
- iii. **Lifts and Jacks:** Hydraulic lifts and jacks are essential for raising vehicles off the ground to access the wheels safely. These lifts come in various configurations, including two-post lifts, four-post lifts, scissor lifts, and mobile column lifts, providing flexibility depending on the size and weight of the vehicles being serviced. [40], [41] [54]
- iv. **Impact Wrenches:** Impact wrenches, also known as pneumatic or electric wrenches, are used to quickly and efficiently loosen and tighten lug nuts. These powerful tools reduce the amount of manual effort required to remove and install wheels, especially when working with stubborn or tightly secured lug nuts. [54], [55]
- v. **Alignment Machines:** Computerized wheel alignment machines are used to measure and adjust the angles of the wheels according to manufacturer specifications. These

- machines streamline the alignment process, allowing technicians to perform precise adjustments with minimal physical effort. [54], [55]
- vi. **Tire Carts and Dollies:** Tire carts and dollies are wheeled platforms designed to transport tires and wheels around the shop. They help technicians move heavy components more easily and reduce the risk of strain or injury associated with manual lifting and carrying. [54], [55]

## **2.5. Technology Gap: Lack of Equipment, Machinery or Mechanism for Simultaneous Removal/Installation of all Wheels of passenger Vehicles**

There is absence of single machine or equipment designed specifically for the simultaneous removal and installation of all four wheels of a vehicle. Tire shops often utilize a combination of tools and equipment to accelerate the process. They depend on more labor intensive measures such as teamwork, repetitive single-task equipment, and efficient manual workflow design. Therefore, working on the research and development of a machine for simultaneous all-wheel removal and installation helps to reduce this technological gap.

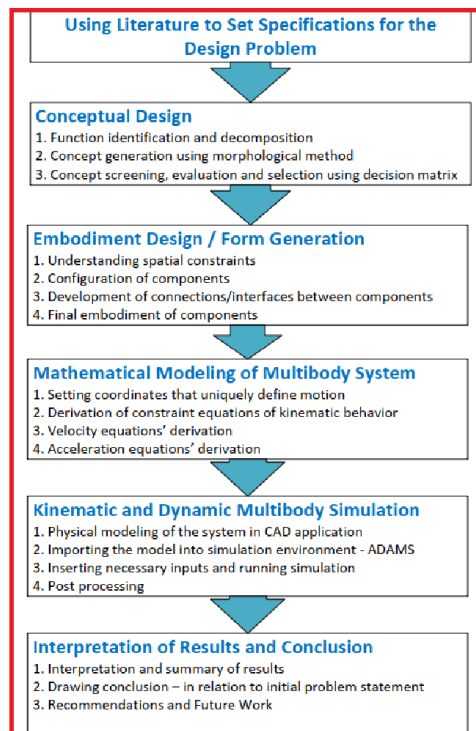
## **2.6. Originality of the Project**

Up to the date of submission of this document, there are no similar mechanisms in use to the knowledge of the submitter. The submitter tried his best to confirm this statement by performing extensive internet search.

## Chapter 3

### Materials, Methods and Conditions

This research followed a systematic approach to find solution to the stated problem by meeting the objectives of the study. First, literature is used to set the design problem, then conceptual design tasks, namely, function identification and decomposition, concept generation using morphological method, concept screening, evaluation and selection using decision matrix are performed. After this, embodiment design tasks, specifically, understanding spatial constraints, configuration of components, development of connections/interfaces between components and final embodiment of components is accomplished. Then, mathematical modeling of the multibody system is done by setting the coordinates that uniquely define the motion, deriving constraint, velocity and acceleration equations. Kinematic and dynamic multibody simulation is used to solve for the motion parameters since it is not possible to solve them analytically. Finally interpretation of the results, drawing conclusions and recommendations of future work is done. This systematic research process is shown by the flow chart in figure 1A below.



*Figure – 1A: Flow Chart Depicting the Systematic Research Process Followed in this Thesis*

In this chapter, first, the synthesis part of the mechanism is presented. This includes the conceptual design of the mechanism followed by the embodiment design used to transform the generated and screened concepts into form that supports the functions. Following the synthesis, multibody dynamic analysis of the mechanism is presented. In order to achieve this, mathematical model is constructed to represent the dynamic behavior followed by numerical simulation to get results that are representative of the multibody dynamic characteristics.

### **3.1. Conceptual Design**

The conceptual design of the simultaneous vehicle wheel removal mechanism is done based on the design principle that states “form follows function”. This means the function (or what the product does) of the device to be synthesized must be thoroughly understood before the form (how the product does the function) can be synthesized [33], [34] and [35].

Moreover, a convergent – divergent design philosophy is followed. This means many alternatives that support the function are first generated and then screened and selected using a systematic decision making process. This approach is advantageous because it creates a situation in which many possible alternatives are suggested and evaluated and, better ideas get the chance to be developed into a product [33], [34], [35] and [36].

#### **3.1.1. Functional Decomposition**

In this phase of the design process, the function of the simultaneous vehicle wheel removal mechanism is thoroughly developed. Function is represented by the logical flow of energy, materials and information between objects and the transformation of these three flows [33]. The approach used is as follows: First, the overall function of the product-to-be is stated. Then, the overall function is decomposed into sub functions. After this, the sub functions are arranged in sequential order. Finally, the sub functions are used to build what is known as morphological chart in which concepts are generated for each sub function. The morphological method is a powerful way that enable the generation of multiple concepts [33], [34], [35], [36] and [37].

### 3.1.1.1. Determination of the Overall Function

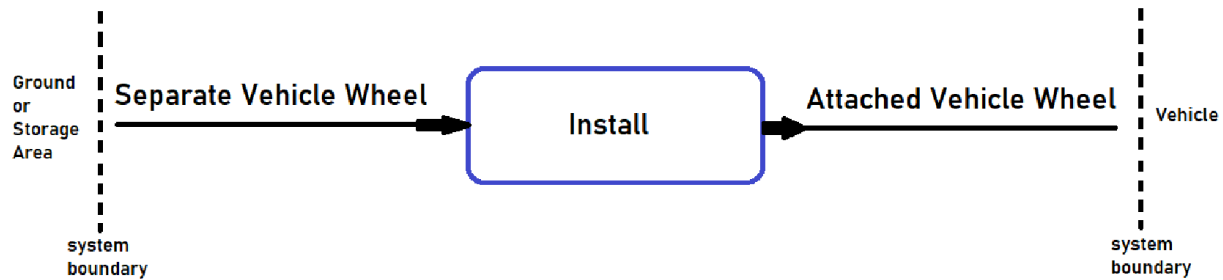
For the simultaneous vehicle wheel removal device being developed, the most important (top level) function is stated as follows:

“Simultaneously install/remove all wheels of a vehicle automatically”

This overall function can be stated in more detail for the installation and removal modes as follows:

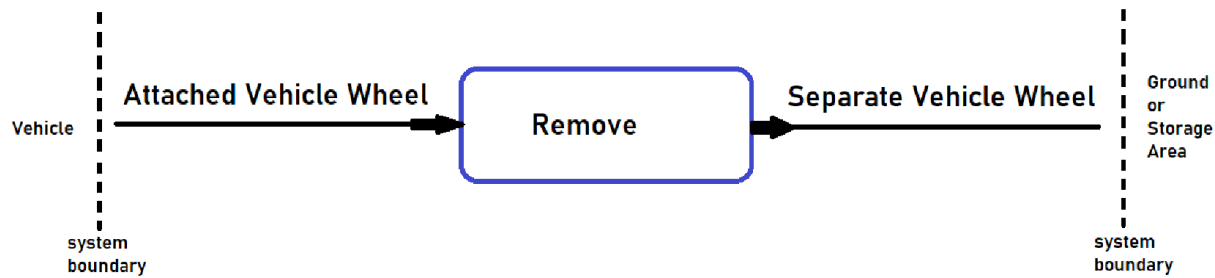
“Transform user order (information) and energy of an external source to a wheel installed on a vehicle to remove it and to a wheel separate from a vehicle to install it.”

The install mode overall function is modelled as shown in the figure below. Note the system boundary, the objects with which the device interacts and the material state transformation done by the function represented in the figure.



**Figure – 1B:** Representation of the Overall Function of the Device in its Wheel Installation Mode

The wheel removal mode overall function is represented in the figure below:



*Figure – 1C: Representation of the Overall Function of the Device in its Wheel Installation Mode*

In stating the overall function, various important guidelines are followed. One essential guideline is that materials should be conserved with respect to the system boundary; here, notice how the vehicle wheels that go into the system are conserved as they go out of the system. Another guideline is that all interfacing objects and known fixed parts of the system are identified namely the vehicle, the ground or storage area and, the vehicle wheel. Finally, action verbs are used to describe the function namely “Install” and “remove”.

### 3.1.1.2. Dividing the Overall Function into Sub Functions

The overall function “Install Wheel on Vehicle” is divided into the following sub functions:

“Hold wheel, position wheel with respect to vehicle, hold the lug nuts/bolts in wrench, position wrench with respect to vehicle, insert lug nuts/bolts, lower vehicle”

The overall function “Remove Wheel from Vehicle” is divided into the following sub functions:

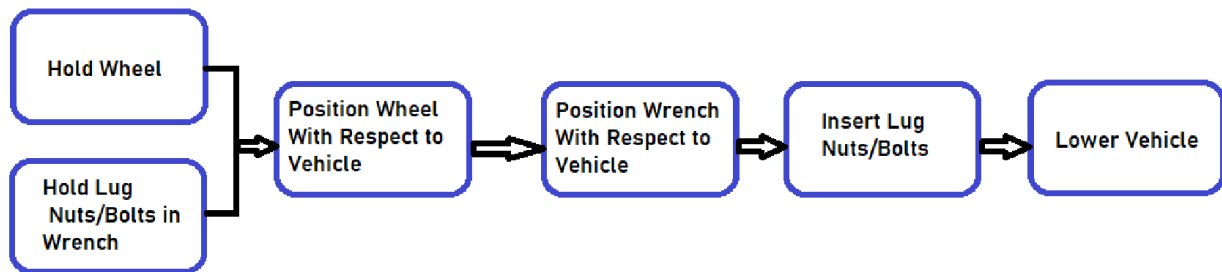
“Lift vehicle, position wrench with respect to lug nuts/bolts, remove lug nuts/bolts, place lug nuts/bolts in storage, hold wheel, remove wheel from vehicle, place wheel on storage location (and release it)”

Here, similar to developing the overall function, guidelines are followed in decomposing the sub functions. First, what should be done by the device (the function) is stated, not how it will be done

(the form). Secondly, only objects stated in the overall function statement (i.e. vehicle and wheel) are stated. There are other intentionally added objects (i.e. wrench and lug nuts/bolts). These objects are added because they are constraints that are inherent to the environment of the device being developed. Addition of such objects is supported by the guidelines [33] [34]. The sub functions are stated in action verbs to represent “what” the product must do.

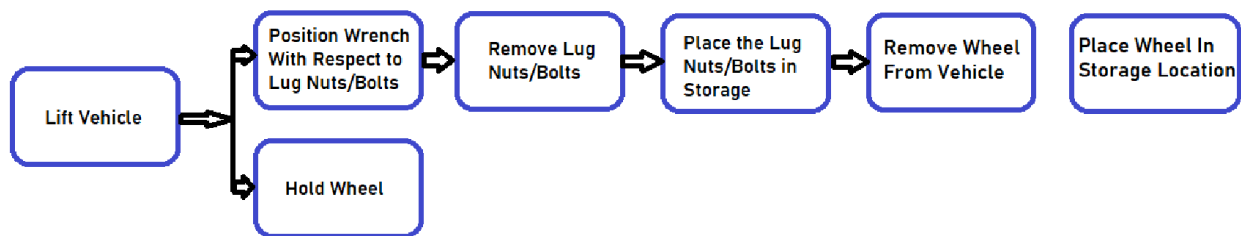
### 3.1.1.3. Ordering the Sub Functions

The sub functions stated in the previous step are sequentially arranged for the wheel installation mode as follows:



*Figure – 1D: Sub Functions for the Wheel Installation Mode Arranged in Logical Order*

The sub functions for the wheel removal mode are also arranged in logical order as shown in the figure below:



*Figure – 1E: Sub Functions for the Wheel Removal Mode Arranged in Logical Order*

### 3.1.1.4. Refinement of the Sub Functions to Their Finest – Possible State

In this step, the sub functions are examined to see if they can be further divided into sub- sub functions. The results shown in the table below were obtained after such examination.

*Table – 3: Further Refinement of the Sub Functions*

<b>Sub Function</b>	<b>Sub – Sub Function</b>
Position Wrench With Respect to Lug Nuts/ Bolts	Search for Lug Nuts’/Bolts’ Location
	Send Signal About the Location
	Engage Wrench to the Lug Nuts/Bolts
Remove Lug Nuts/Bolts	Rotate Counterclockwise to Fully Disengage Lug Nuts/Bolts
	Disengage Wrench
Remove Wheel From Vehicle	Move Wheel Away From Vehicle
	Move to Specific Storage Location
	Release Wheel in Storage Location

### 3.1.2. Generating Concepts Using the Morphological Chart Technique

As mentioned above, the developed sub functions are inputs to this step of building a morphological chart. Here, possible alternative concepts are generated for each of the sub functions using brainstorming and analogy techniques. The resulting table is known as the morphological chart. It is presented in the table below.

**Table – 4: Morphological Chart Showing the Generated Alternative Concepts for each Sub Function**

Sub Function	Concept - 1	Concept - 2	Concept - 3	Concept - 4	Concept - 5
Hold Wheel / Release Wheel	Radially expanding jaws holding outwards	Jaws holding inwards from wheel's outer circumference	U – shaped holder operating at wheel's bottom	Magnetically holding the rim of the wheel (for steel rims)	Vacuum suction to hold rubber of tire
Engage/Disengage Wrench to/from the Lug Nuts/Bolts	Rack and pinion	Linear – acting solenoid	Hydraulic cylinder	Pneumatic cylinder	Power screw
Rotate Counterclockwise to Fully Disengage Lug Nuts/Bolts	Electric Motor	Hydraulic Motor	Pneumatic Motor		
Move Wheel To/Away From Vehicle	Symmetric Double four bar Planar mechanism	Symmetric Double Eight bar Planar mechanism	Robot arm Space mechanism		
Move to Specific Storage Location					
Search for Lug Nuts'/Bolts' Location	Operated by User from Screen and Controllers	Machine Vision using Camera	Encoders tracing spatial motion and predefined locations		
Send Signal About the Location					

By combining the concepts corresponding to each sub function, the total number of overall concepts generated using the morphological chart is:  $5*5*3*3*3 = 675$  concepts. This is obviously

too large a number to consider. In fact, despite being a powerful method to generate concepts, the morphological chart has one weakness - it generates too many concepts than actually possible by assuming that each sub function is independent [33]. Therefore, a selective choice of potential concepts from the morphological chart is necessary. Based on this, the following overall concepts are selected from the morphological chart.

**Table – 5: Selective Generation of Alternative Overall Concepts Based on the Morphology Chart**

	<b>Combination of Alternatives from the Morphological Chart</b>
Overall Concept – 1	Radially expanding jaws holding outwards + Rack and pinion + Electric Motor + Symmetric Double four bar Planar mechanism + Operated by User from Screen and Controllers
Overall Concept – 2	U – shaped holder operating at wheel’s bottom + Hydraulic cylinder + Hydraulic Motor + Symmetric Double Eight bar Planar mechanism + Operated by User from Screen and Controllers
Overall Concept – 3	Vacuum suction to hold rubber of tire + Pneumatic cylinder + Pneumatic Motor + Robot arm Space mechanism + Encoders tracing spatial motion and predefined locations
Overall Concept – 4	Jaws holding inwards from wheel’s outer circumference + Linear – acting solenoid + Electric Motor + Symmetric Double Eight bar Planar mechanism + Machine Vision using Camera
Overall Concept – 5	Magnetically holding the rim of the wheel (for steel rims) + Power screw + Electric Motor + Robot arm Space mechanism + Encoders tracing spatial motion and predefined locations

### 3.1.3. Concept Evaluation and Selection Using Decision Matrix

As it can be clearly seen, the five alternative overall concepts present various strengths and weaknesses that are not easy for decision making when they are subjected to comparative evaluation. Such decision making is made by using a decision matrix. The evaluation criteria, their corresponding weighting factors, and the ranks each overall concept gets is shown in the decision matrix below.

*Table – 6: Decision Matrix Used to Evaluate and Select Overall Concept*

	<b>Cost</b>	<b>Reliability</b>	<b>Manufactu rability</b>	<b>Durability</b>	<b>Maintaina bility</b>	<b>Space Efficiency</b>	<b>Rank</b>
Weighting Factor	0.2	0.2	0.15	0.2	0.1	0.15	1
Overall Concept – 1	8 1.6	7 1.4	7 1.05	5 1.0	6 0.6	1 0.15	5.8
Overall Concept – 2	2 0.4	4 0.8	6 0.9	6 1.2	3 0.3	7 1.05	4.65
Overall Concept – 3	3 0.6	5 1.0	6 0.9	4 0.6	2 0.2	2 0.3	3.6
Overall Concept – 4	8 1.6	5 1.0	8 1.2	5 1.0	7 0.7	7 1.05	6.55
Overall Concept – 5	7 1.4	2 0.4	3 0.45	2 0.4	9 0.9	8 1.2	4.75

The decision matrix was constructed by giving a weighting factor between valued 0 and 1 for each evaluation criteria the sum of which equals 1.0. Then, each overall concept is valued with a number 1 – 10 against each evaluation criteria 1 being the lowest and 10 being the highest. These value is multiplied by its corresponding weighting factor to give the number in the lower diagonal of the cell. These lower numbers across the row were summed up to give the ranks at the rightmost column. As clearly seen, overall concept – 4 has the highest rank. Therefore, it is selected for further evaluation.

### **3.2. Embodiment Design**

Now that the conceptual design phase is completed and a concept is selected from a list of alternatives, the next task is to further develop the selected concept into a product. This is

accomplished by embodiment design. It involves giving form to support the function. To achieve this, first, spatial constraints of the system are understood. Then, components are configured based on the functions developed earlier. Then components and their interfaces are developed.

A modular architecture approach is followed in the embodiment design of the simultaneous wheel removal and installation mechanism under development. There are two basic reasons for choosing this approach. One reason is that it is more convenient to allocate one major sub function to one module (later developed into subassembly). Secondly, since there are four wheels of a vehicle to be dealt with by the device, there must be four modules (or subassemblies) dispersed at four separate locations connected by components.

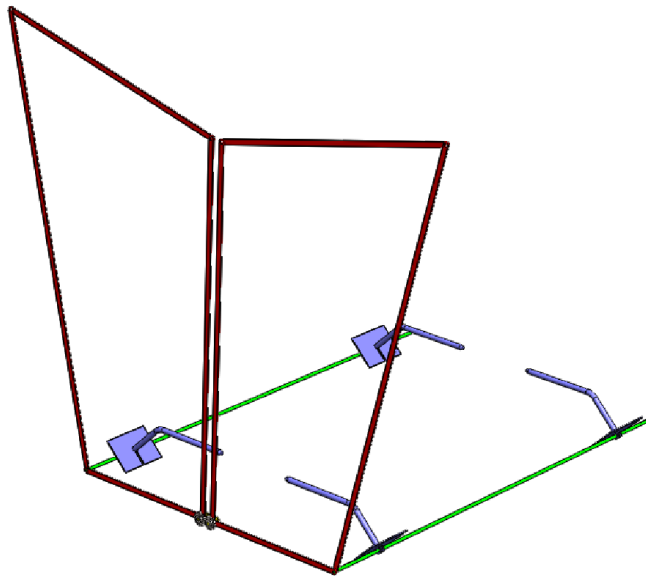
Based on this, four modules are identified and laid out to support the sub functions. These are the wheel positioning module, wheel holding module, lug nut/bolt removal/installation module and the vehicle support module. Out of these four, the vehicle support module or vehicle lift is already developed technology and there are a great deal of existing choices of vehicle lift. Therefore, it is intentionally left out of the subsequent discussion. The other three modules however do not exist previously. Their development is an original design problem. Their embodiment solution is essentially a new invention. Therefore, their individual development as well as how they are integrated together is discussed in detail.

### **3.2.1. Embodiment of Wheel Positioning Module**

This module is part of the mechanism responsible for positioning the four wheels of the vehicle to/away from the vehicle in the wheels' installation/removal. During the course of this synthesis, two alternative solutions were generated. These solutions are given the name synthesis option – I and synthesis option – II. Each are presented in the following subsections below.

### 3.1.1.1. Synthesis Option – I

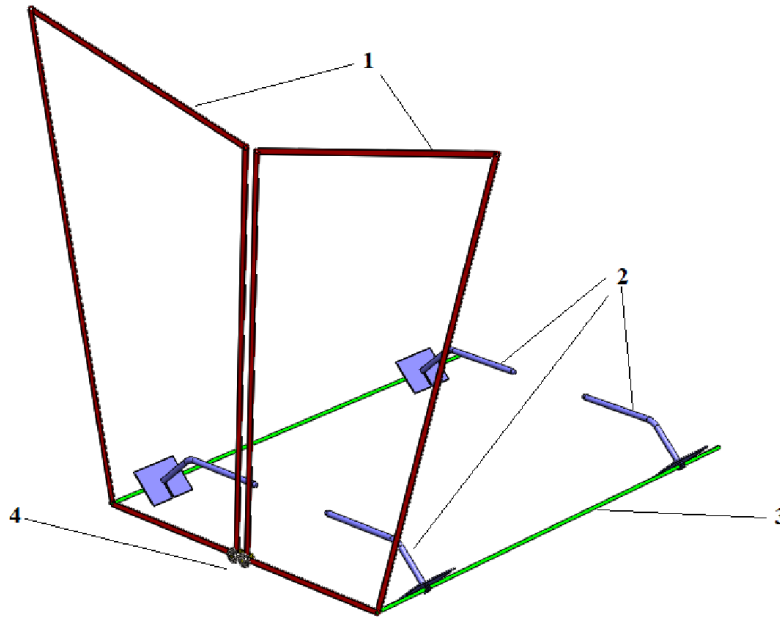
Many linkage synthesis techniques have been attempted to synthesize a mechanism to accomplish the task and inappropriate ones have been discarded. The feasible concept that satisfies all of the above-mentioned requirements is the mechanism shown in figure – 3.



*Figure – 3: CAD Assembly Showing the Synthesized Option – I Mechanism*

#### **Description of the Mechanism's Operation**

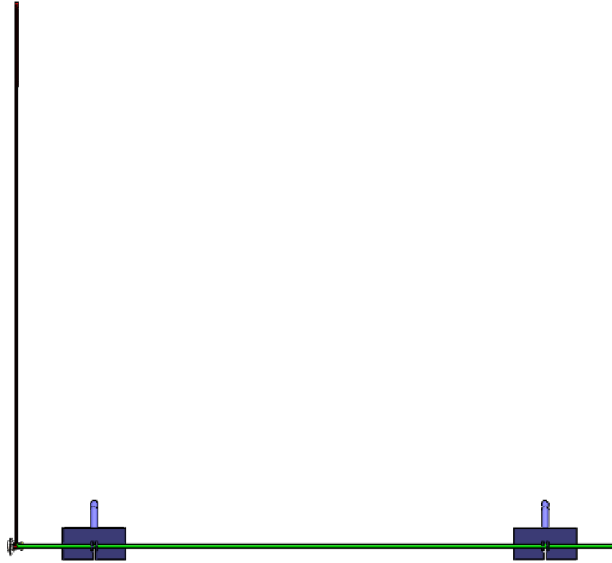
Refer to figure – 4 for the discussion of the description of the operation of the mechanism. The needed motion is to rotate the four tire carriers from the position in which the straight end is horizontal by 225 degrees to position the tip outwards. Each of the double-crank four bar linkages conveniently provide this almost full – circle rotational motion of the tire carrier. However, since the tire carrier on the left and right sides of the vehicle must rotate in opposite directions, a gear drive consisting of two spur gears is provided at the input (middle) links of the two double crank linkages. Since external gear sets reverse the direction of rotation between input and output, the two double crank linkages on each side rotate in opposite direction as desired.



**Figure – 4:** *Numbered Illustration of the Synthesized Option – I Mechanism (1 – Links, 2 – Tire Carrier, 3 – Connecting Shafts, 4 - Gears)*

The incorporation of the gear drive at the center provides a great opportunity to drive the mechanism by means of only one motor attached to one of the gears. This creates a simple solution that is cost – effective and convenient to control.

Another strong point of the synthesized mechanism is that the linkages can be compactly enclosure in a box of small thickness that can be aligned with one of the wall of a workshop. This can be seen on the side view of the assembly shown in figure – 5.



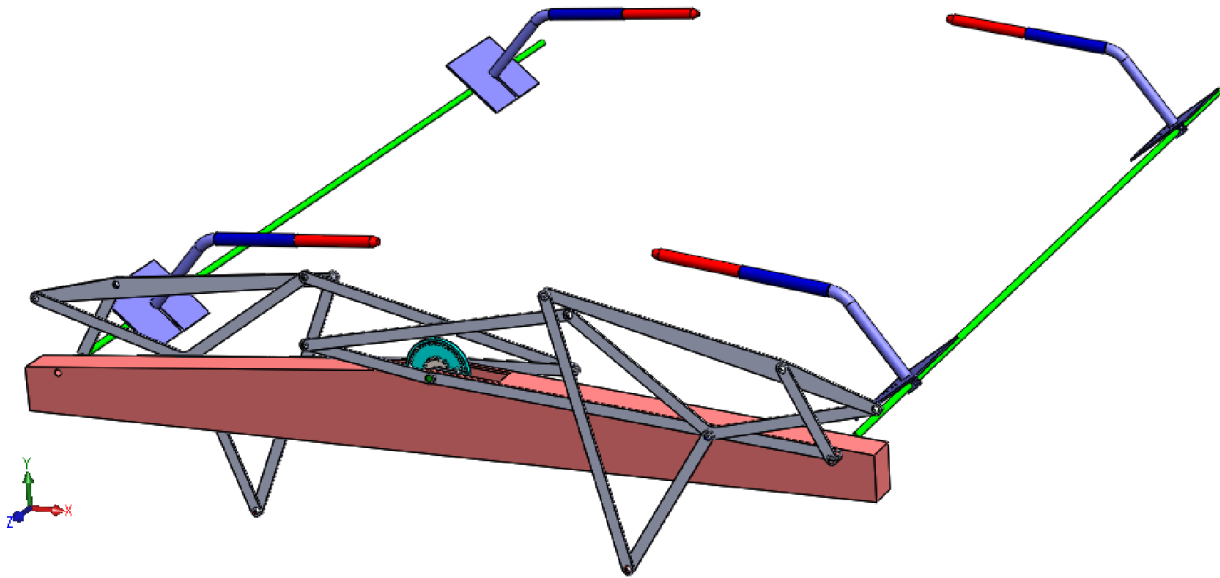
*Figure - 5: Side View of the Option – I Assembly Indicating the Ease of Enclosure of the Linkage Assembly from the Left Side*

### **3.2.1.2. Synthesis Option – II**

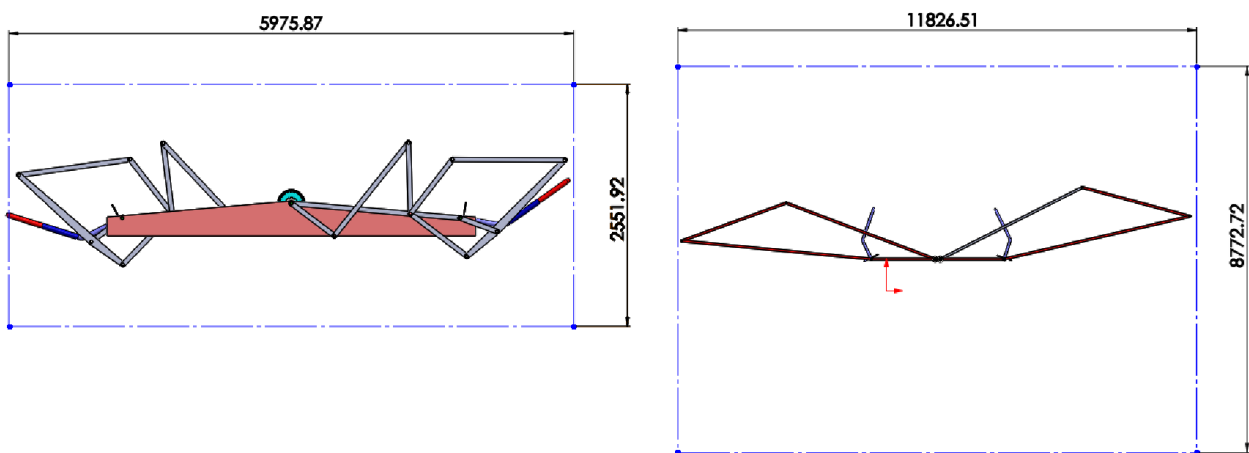
While synthesis option – I presented above has many advantages as a potential solution, it has one major drawback – the area covered by the planar mechanism during its operation is quite large due to the length of the links. Therefore, another solution is synthesized which has a much smaller area covered by the planar mechanism. This solution is named synthesis option – II. It is a 16 – bar mechanism comprising a double, eight bar connected by a gear set. The solid model of this mechanism assembled with a CAD application is shown in the figure – 6.

This mechanism operates exactly like the synthesis option – I mechanism. The only difference is that the planar linkage that generates the desired motion has higher number of links while being much more compact. Therefore, synthesis option – II retains all of the good characteristics of synthesis option – I and eliminates the previous options the major drawback. The difference in

the area covered by the two mechanisms is compared in figure – 7. Because the planar area of synthesis option – II is significantly smaller, it is selected to be further analyzed and developed.



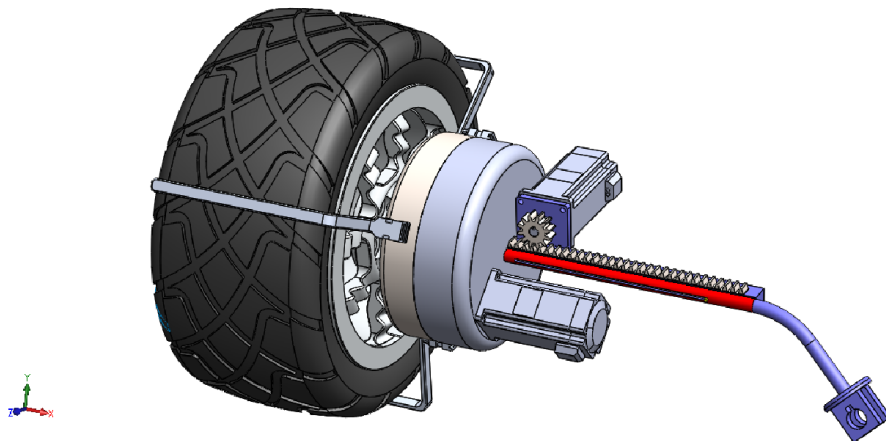
*Figure – 6: CAD Assembly Showing Mechanism Synthesis Option - II*



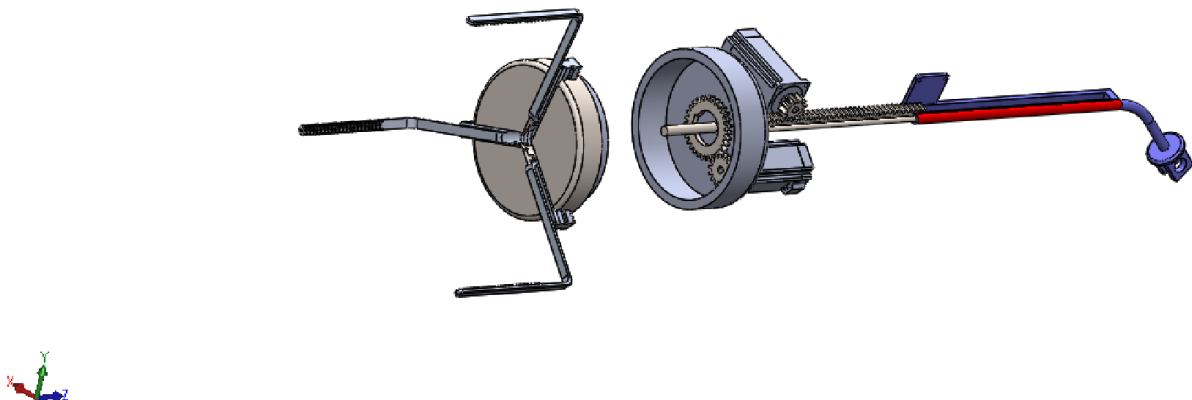
*Figure – 7: Comparison of Area Covered by the Planar Mechanisms during their Operation. Right: Synthesis Option – I, Left: Synthesis Option – II. The Shown Dimensions are in Millimeters.*

### 3.2.2. Embodiment of Wheel Holding Module

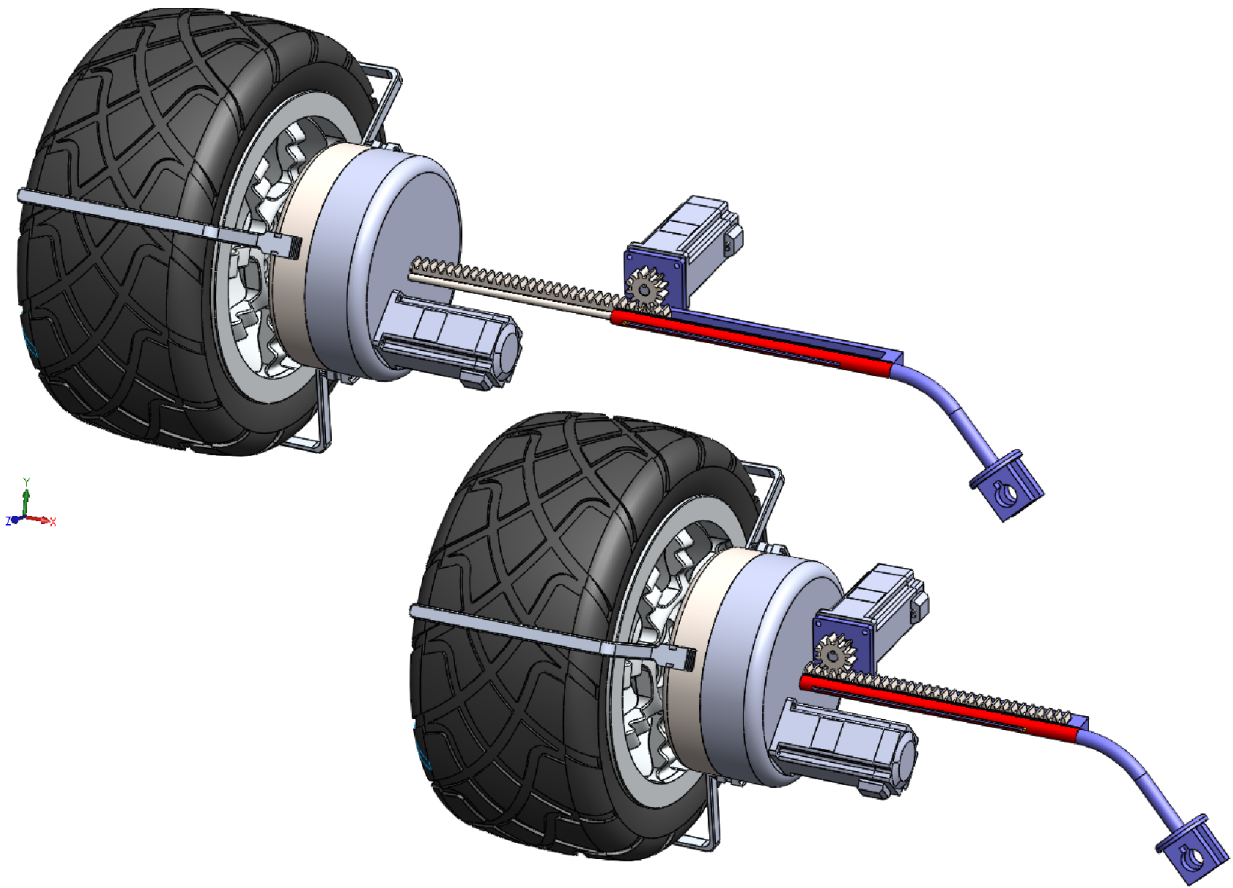
The wheel holding module is responsible for the function of holding the wheel of the vehicle. Since there are four wheels this module is needed at each wheel. The selected concept from the conceptual design phase incorporates jaws holding inwards from wheel's outer circumference. This is developed into form as shown in the figure below.



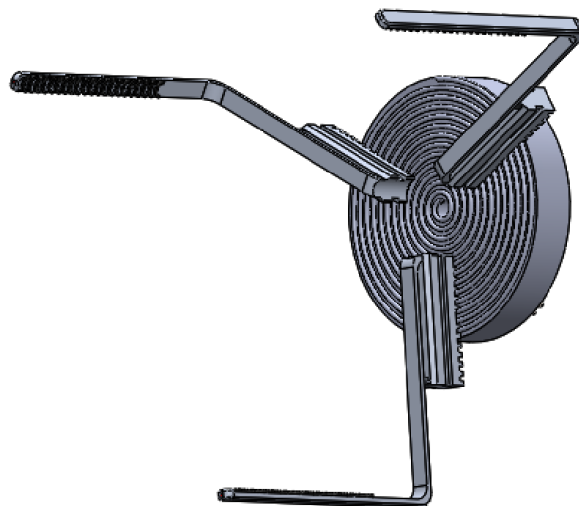
*Figure – 8: The Wheel Holding Module of the Mechanism Modeled with CAD Application*



*Figure – 9: Exploded View of The Wheel Holding Module of the Mechanism Modeled with CAD Application*



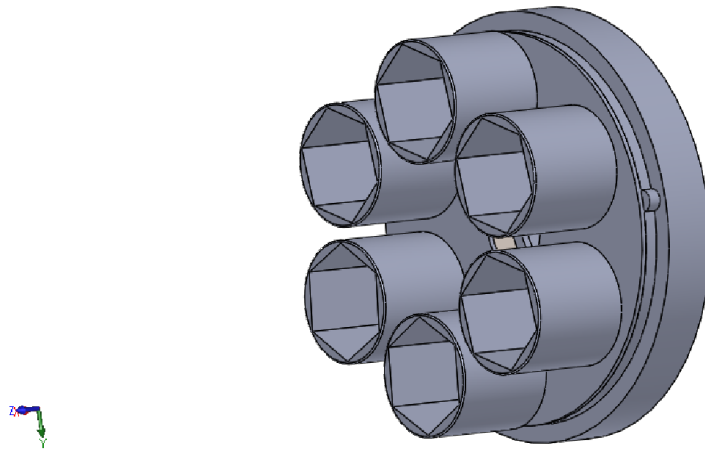
*Figure – 10: Extended and Retracted Modes of the Wheel Holding Assembly*



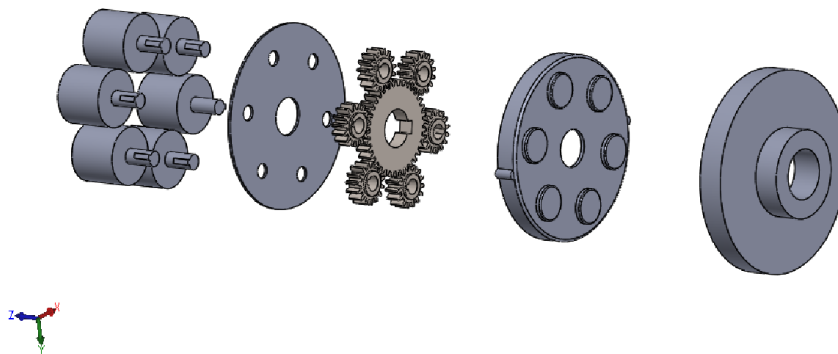
*Figure – 11: Spiral Mechanism of the Wheel Holding Assembly to Adjust for Different Wheel Diameters*

### 3.2.3. Embodiment of Lug Nut/Bolt Removal/Installation Module

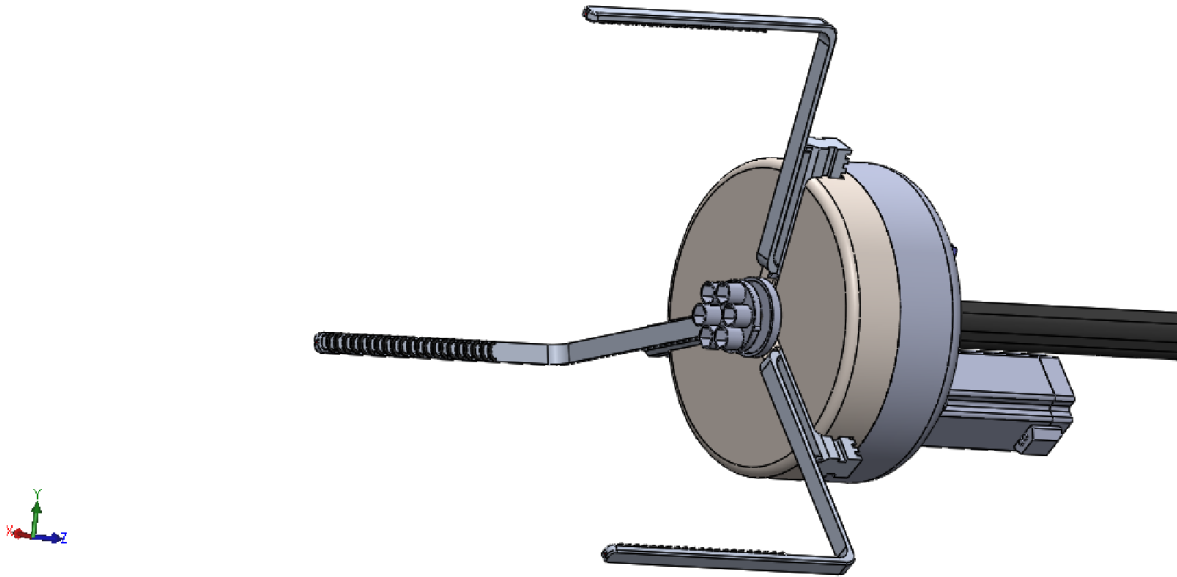
This module is responsible for removing the lug nuts/bolts that fasten the wheel to the vehicle simultaneously. It works by means of gear drive and forward and backward actuation by linearly actuating solenoid. The details of this module were modeled by the CAD system and are presented in the following figures.



*Figure – 12: The Lug Nut/Bolt Removal/Installation Module*



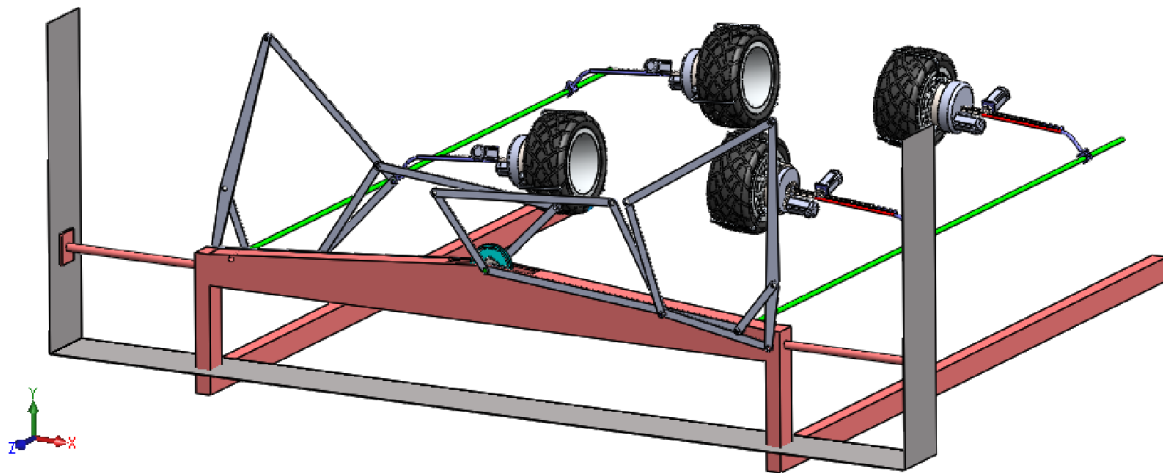
*Figure – 13: Exploded View Showing Internal Components in the Lug Nut/Bolt Removal/Installation Module*



*Figure – 14: Interrelationship between the Lug Nut/Bolt Removal/Installation Module and the Wheel Holding Module*

### **3.2.3. Overall Assembly**

The overall assembly is comprised of all of the modules in the system that support the overall function of the system. This assembly is shown in the figure below.

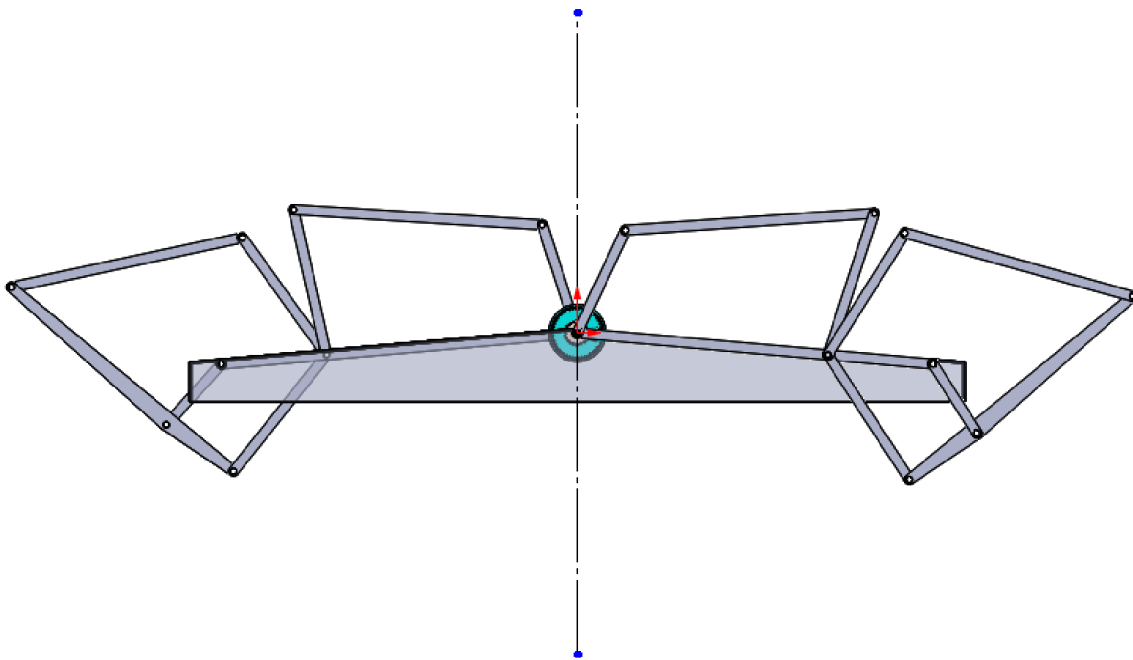


*Figure – 15: The Overall Assembly of the Mechanism*

### 3.3. Mathematical Modeling

In the previous section, two mechanisms were synthesized and proposed as a solution. Of these two, the more compact synthesis option – II was selected for further analysis. As shown, physical modeling of the synthesized mechanism has been constructed. In this chapter, a mathematical model that describes the motion of the mechanism will be constructed. This mathematical model is the basis for the position, velocity, acceleration and dynamic force analyses of the links in the mechanism.

Consider the links within the assembly as shown in figure – 9.



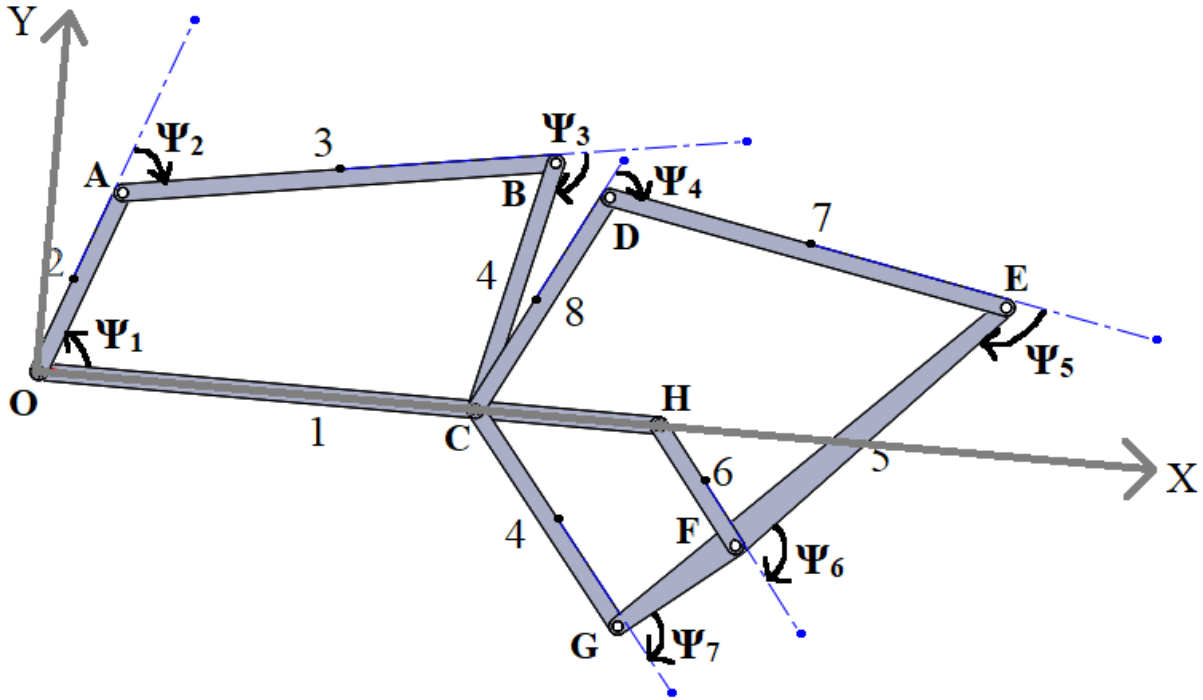
*Figure – 8: The Symmetry of Synthesized Mechanism*

Note the following important points:

- The linkage is symmetric. The motion of each link on one-half is the same as the motion on the other half but in the opposite direction. This is shown in figure – 8.
- The function of the gears at the center is to reverse the direction of the motion between the input links of the two opposite linkage halves. The gear ratio is 1:1.

- One-half will be used to derive the constraint equations in the process of building the mathematical model.

Consider the left half of the mechanism with the labels in figure – 9 below. Note that relative coordinates are used to construct the mathematical model.



*Figure – 9: One-half of the Mechanism Modeled with Relative Coordinates*

### 3.2.1. Constraint Equations

In order to derive the constraint equations, the degree of freedom of the mechanism should be known first. For the above selected half, the degree of freedom is obtained as follows:

$$\text{DOF} = 3(L - 1) - 2(J_1) - (J_2)$$

$$\text{DOF} = 3(8 - 1) - 2(10)$$

$$\text{DOF} = 1$$

Where *DOF* is degrees of freedom of the linkage, *L* is the number of links, *J<sub>1</sub>* is the number of one *DOF* joints (full joints) and *J<sub>2</sub>* is the number of two *DOF* joints (half joints). Note that, for this linkage, *L* = 8 links, *J<sub>1</sub>* = 10 joints and *J<sub>2</sub>* = 0.

The relative coordinates used to explicitly describe the position of the linkage at any given instant of time are  $\Psi_1, \Psi_2, \Psi_3, \Psi_4, \Psi_5, \Psi_6$  and  $\Psi_7$ . There are seven coordinates in this system.

The number of constraint equations is determined as follows:

$$\text{Number of constraint Equations} = \text{Number of Coordinates} - \text{DOF}$$

$$\text{Number of constraint Equations} = 7 - 1 = 6$$

Therefore, six constraint equations will be derived from the information developed so far. Taking note that there are three different loops are formed by the linkage, the following three vector loop equations (also known as loop closure equations) can be written.

$$\mathbf{OA} + \mathbf{AB} + \mathbf{BC} - \mathbf{OC} = \mathbf{0} \dots\dots\dots \text{I}$$

$$\mathbf{CD} + \mathbf{DE} + \mathbf{EF} + \mathbf{HF} - \mathbf{CH} = \mathbf{0} \dots\dots\dots \text{II}$$

$$\mathbf{CG} + \mathbf{GF} + \mathbf{HF} - \mathbf{CH} = \mathbf{0} \dots\dots\dots \text{III}$$

The X and Y components of the vector loop equation I are:

$$L_2 \cos \Psi_1 + L_3 \cos (\Psi_1 + \Psi_2) + L_4 \cos (\Psi_1 + \Psi_2 + \Psi_3) - |\mathbf{OC}| = 0$$

$$L_2 \sin \Psi_1 + L_3 \sin (\Psi_1 + \Psi_2) + L_4 \sin (\Psi_1 + \Psi_2 + \Psi_3) = 0$$

The X and Y components of the vector loop equation II are:

$$L_8 \cos \Psi_4 + L_7 \cos (\Psi_4 + \Psi_5) + |\mathbf{EF}| \cos (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \cos (\Psi_4 + \Psi_5 + \Psi_6) - |\mathbf{CH}| = 0$$

$$L_8 \sin \Psi_4 + L_7 \sin (\Psi_4 + \Psi_5) + |\mathbf{EF}| \sin (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \sin (\Psi_4 + \Psi_5 + \Psi_6) = 0$$

The X and Y components of the vector loop equation III are:

$$L_4 \cos \Psi_3 + |\mathbf{GF}| \cos (\Psi_3 + \Psi_7) + L_6 \cos (\Psi_3 + \Psi_6 + \Psi_7) - |\mathbf{CH}| = 0$$

$$L_4 \sin \Psi_3 + |\mathbf{GF}| \sin (\Psi_3 + \Psi_7) + L_6 \sin (\Psi_3 + \Psi_6 + \Psi_7) = 0$$

The above six equations are the constraint equations representing the mathematically modeled mechanism.

### 3.2.2. Velocity Equations

The velocity equations are obtained by differentiating the above constraint equations with respect to time. The equations are:

$$\frac{d}{dt} [L_2 \cos \Psi_1 + L_3 \cos (\Psi_1 + \Psi_2) + L_4 \cos (\Psi_1 + \Psi_2 + \Psi_3) - |\mathbf{OC}| = 0]$$

$$\frac{d}{dt} [L_2 \sin \Psi_1 + L_3 \sin (\Psi_1 + \Psi_2) + L_4 \sin (\Psi_1 + \Psi_2 + \Psi_3) = 0]$$

$$\frac{d}{dt} [L_8 \cos \Psi_4 + L_7 \cos (\Psi_4 + \Psi_5) + |\mathbf{EF}| \cos (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \cos (\Psi_4 + \Psi_5 + \Psi_6) - |\mathbf{CH}| = 0]$$

$$\frac{d}{dt} [L_8 \sin \Psi_4 + L_7 \sin (\Psi_4 + \Psi_5) + |\mathbf{EF}| \sin (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \sin (\Psi_4 + \Psi_5 + \Psi_6) = 0]$$

$$\frac{d}{dt} [L_4 \cos \Psi_3 + |\mathbf{GF}| \cos (\Psi_3 + \Psi_7) + L_6 \cos (\Psi_3 + \Psi_6 + \Psi_7) - |\mathbf{CH}| = 0]$$

$$\frac{d}{dt} [L_4 \sin \Psi_3 + |\mathbf{GF}| \sin (\Psi_3 + \Psi_7) + L_6 \sin (\Psi_3 + \Psi_6 + \Psi_7) = 0]$$

### 3.2.3. Acceleration Equations

The acceleration equations are obtained from the first time derivatives of the velocity equations, which also are the second time derivatives of the constraint equations. These are:

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_2 \cos \Psi_1 + L_3 \cos (\Psi_1 + \Psi_2) + L_4 \cos (\Psi_1 + \Psi_2 + \Psi_3) - |\mathbf{OC}| = 0] \right\}$$

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_2 \sin \Psi_1 + L_3 \sin (\Psi_1 + \Psi_2) + L_4 \sin (\Psi_1 + \Psi_2 + \Psi_3) = 0] \right\}$$

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_8 \cos \Psi_4 + L_7 \cos (\Psi_4 + \Psi_5) + |\mathbf{EF}| \cos (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \cos (\Psi_4 + \Psi_5 + \Psi_6) - |\mathbf{CH}| = 0] \right\}$$

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_8 \sin \Psi_4 + L_7 \sin (\Psi_4 + \Psi_5) + |\mathbf{EF}| \sin (\Psi_4 + \Psi_5 + \Psi_6) + L_6 \sin (\Psi_4 + \Psi_5 + \Psi_6) = 0] \right\}$$

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_4 \cos \Psi_3 + |\mathbf{GF}| \cos (\Psi_3 + \Psi_7) + L_6 \cos (\Psi_3 + \Psi_6 + \Psi_7) - |\mathbf{CH}| = 0] \right\}$$

$$\frac{d}{dt} \left\{ \frac{d}{dt} [L_4 \sin \Psi_3 + |\mathbf{GF}| \sin (\Psi_3 + \Psi_7) + L_6 \sin (\Psi_3 + \Psi_6 + \Psi_7) = 0] \right\}$$

This completes the mathematical modelling of the mechanism. Note that in these equations all of the link lengths  $L_1, L_2, L_3, L_4, L_5, L_6, L_7$  and  $L_8$  are known from the geometry of the linkage. In addition, the lengths  $|\mathbf{OC}|, |\mathbf{CH}|$  and  $|\mathbf{GF}|$  are also known. All of the positions, velocities and acceleration variables are unknown except the initial values of one link – most often the crank (link – 2).

At this stage, it is important to note that we have six, second order, ordinary differential equations and six unknown variables. It is clear that it is not feasible to solve these equations for the unknown variables. For this reason, a different approach is used to solve for the desired parameters. This approach is the implementation of computer simulation to solve the dynamic system at hand.

### **3.3. Kinematic and Dynamic Simulation**

At this stage, the synthesis of the mechanism is complete and all the link lengths, connection types between links, and locations of the connections with respect to the links are determined. The next step along the design process of this linkage is kinematic and kinetic analysis. Kinematic analysis is the analysis of the mechanism without regard to the forces that cause the motion or that result from the motion. It leads to the determination of all links' linear and angular positions, velocities and accelerations. Kinetic analysis, also known as dynamic force analysis, on the other hand, is performed by taking into consideration the forces associated with the motion of the links. It can be used to determine applied forces that cause motion or inertial forces that are the result of the motion.

Dynamic force analysis, when used to determine the accelerations, velocities and respective positions that are caused by known forces, is referred to as forward dynamics. Its reverse process, namely the determination of forces after the determination of positions, velocities and accelerations from a prior kinematic analysis is known as inverse dynamics. The method used in this chapter for dynamic force analysis is inverse dynamics.

The Simulation of the mechanism for which the mathematical modeling is complete in the previous section is carried out using the well – known multibody kinematic and dynamic simulation computer program called MSC ADAMS. This program is the widely accepted for such simulations in both industry and academics. The version of the software used is MSC ADAMS 2020, build: 2020.0.0-CL711253.

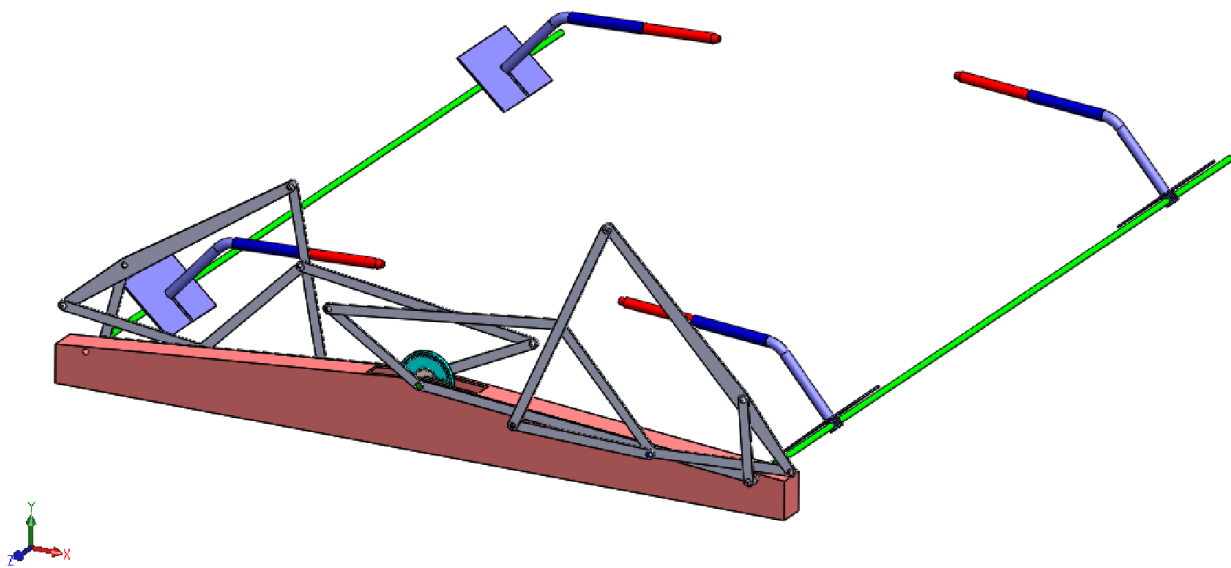
While this program is well – suited for the kinematic and dynamic analysis stage of the simulation process, it is not as suited for the creation of the geometry of the multibody systems. For this reason, the powerful CAD program called SolidWorks is used for the modelling of the geometry of the linkages. Once the geometry of all individual links in the mechanism is modelled using

SolidWorks, the model is exported to ADAMS for the simulation. The particular version of SolidWorks used for this task is SolidWorks Premium 2022 SP1.0.

The following explanation clearly shows the step – by – step procedure followed during the course of the kinematic and dynamic simulation of the mechanism. At each step, the specific conditions set up for the simulation are explicitly outlined.

### **3.2.1. Step – 1: Creation of Geometry of the Links and Their Assembly in SolidWorks**

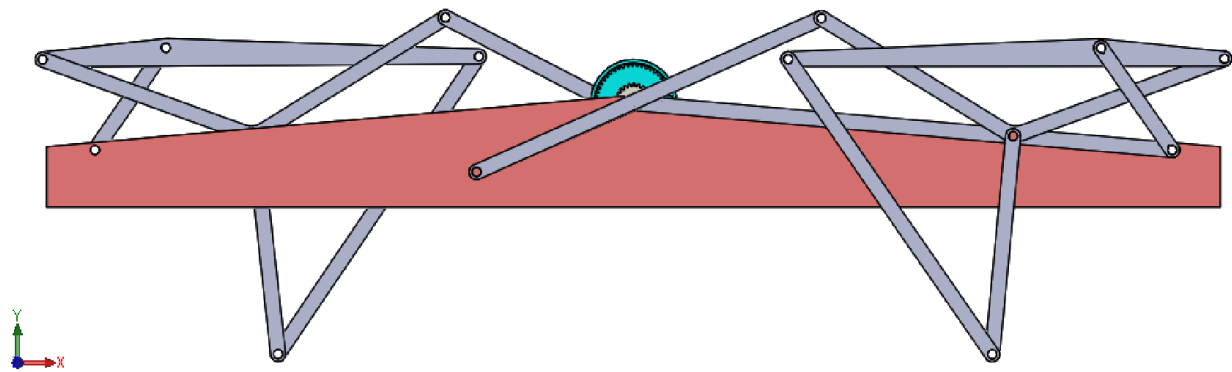
As shown in the mechanism synthesis chapter, the full – scale three – dimensional model of the entire mechanism including all components is modelled using SolidWorks. Here, all the links geometries and lengths, and how they connect to each other is determined. This assembly is shown in the figure – 10.



***Figure – 10: Full – Scale Complete Three – Dimensional Model of the Mechanism***

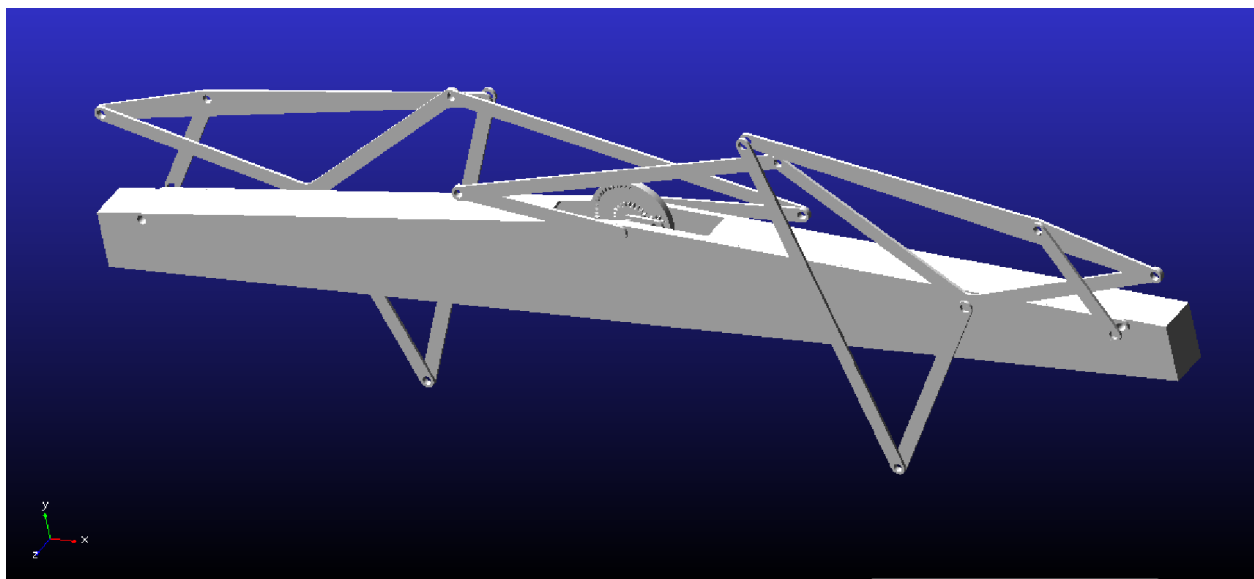
### 3.2.2. Step – 2: Importing the Linkage Portion of the Assembly to MSC ADAMS

The linkage portion of the assembly directly associated with generating the desired motion is shown in figure – 11. This linkage part is exported from SolidWorks to MSC ADAMS for simulation.



**Figure – 11:** *The Linkage Portion of the Assembly exported from SolidWorks to MSC ADAMS*

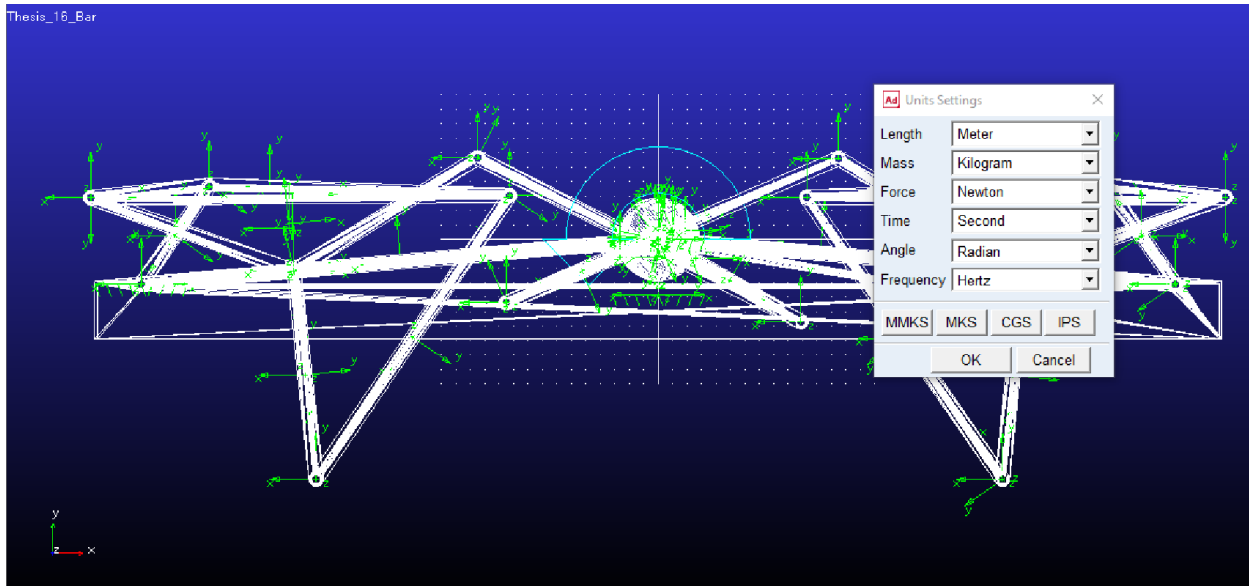
The linkage with all its connections is successfully imported into the MSC ADAMS program. Figure – 12 shows the imported model in MSC ADAMS.



**Figure – 12:** *The Linkage Portion of the Assembly Imported into MSC ADAMS*

### 3.2.3. Step – 3: Setting Up the Units for the Simulation

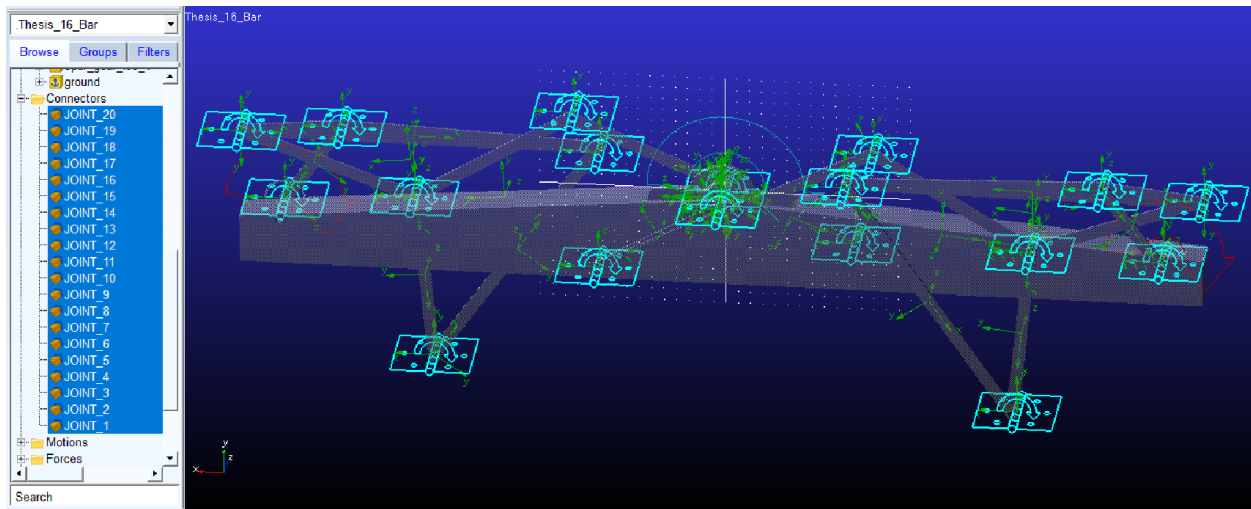
Units are set for the simulation. The selected units are the standard units in the SI system. They are: meter for length, kilogram for mass, Newton for force, second for time, radians for degree, and Hertz for frequency. Unless particularly noted, these units are uniformly used throughout this study. Figure – 13 shows these units selected in the program. Note also that the origin of the global coordinate system of the mechanism is exactly located at the center of the driving gears, which coincides with the driving pins of both left and right cranks. The simulation program automatically assigns material properties of structural steel for the links. Therefore, it automatically computes the masses and mass moments of inertia for each link based on these properties.



*Figure – 13: Indication of the Origin Location and Selection of Units*

### 3.2.4. Step – 4: Connecting the Links with Joints

The next step in the simulation process is connecting the links in the mechanism with one another in the appropriate manner. All of the joints in this mechanism are revolute joints that only allow relative rotation between the links they connect. The joints and the locations where they are applied in the linkage is shown in the figure below. Note that there are 20 revolute joints in this mechanism. These are shown in figure – 14.



*Figure – 14: The Twenty Revolute Joints in the Mechanism and the Locations Where They are Inserted in the Model*

In order to make which joint connects which of the links, the following link – joint grouping for the mechanism whose joint is simulated is presented in the table – 3.

**Table – 8:** List of Joints in the Mechanism and their Corresponding Links they Connect

Joint Name	Connected Links		Type of Joint
	First Link	Second Link	
Joint_1	ground	Link_2_1	Revolute
Joint_2	Link_2_1	Link_3_1	Revolute
Joint_3	Link_3_1	Link_4_1	Revolute
Joint_4	Link_5_1	Link_7_1	Revolute
Joint_5	Link_4_1	ground	Revolute
Joint_6	Link_8_1	Link_7_1	Revolute
Joint_7	Link_8_1	ground	Revolute
Joint_8	Link_5_1	Link_6_1	Revolute
Joint_9	Link_6_1	ground	Revolute
Joint_10	Link_4_1	Link_5_1	Revolute
Joint_11	Link_2_2	ground	Revolute
Joint_12	Link_2_2	Link_3_2	Revolute
Joint_13	Link_4_2	Link_3_2	Revolute
Joint_14	Link_5_2	Link_7_2	Revolute
Joint_15	Link_8_2	Link_7_2	Revolute
Joint_16	Link_4_2	ground	Revolute
Joint_17	Link_8_2	ground	Revolute
Joint_18	Link_5_2	Link_6_2	Revolute
Joint_19	Link_5_2	Link_4_2	Revolute
Joint_20	Link_6_2	ground	Revolute

### 3.2.5. Calculation of Externally Applied Torque and Required Input Power/Speed

The next steps in the simulation of the mechanism require input motion power/speed and externally applied torque. These values are calculated in the following subsections. The calculated values are in turn used in the next steps of the simulation, namely step – 5 and step – 6.

#### 3.2.5.1. Taking Weight of Components at the End of Tire Carrier into Consideration

In order to determine the force acting at the tip of the tire carrier of the mechanism, the weight of the tire as well as the total weight of the components in the assembly at the tip of the tire carrier must be taken into consideration. The components at the tip of the tire carrier assembly and their corresponding weights are listed in table-9 below. Note that these components were designed in the synthesis stage of the design process. Although the purpose of this work focuses on kinematic analysis and material selection is out of the scope here, materials were tentatively assigned for these components based on practical engineering judgement – the purpose being determining their mass properties. Figures 15 through 23 in the following pages present the mass properties determination of each component from the solid-modeling environment with the assigned material type.

*Table – 9: Components in the Assembly at the tip of the Tire Carrier and their Masses*

No.	Item	Material	Qty.	Component Mass (Kg)	Total Mass (Kg)
1	Cover	Gray Cast Iron	1	11.60571	11.60571
2	Wheel Holding Jaw	Gray Cast Iron	3	1.76887	5.30661
3	Spiral	Gray Cast Iron	1	11.76587	11.76587
4	Housing	Gray Cast Iron	1	14.84309	14.84309
5	Spur Gear – 1	AISI 1045	1	0.668936	0.668936
6	Spur Gear – 2	AISI 1045	1	0.237741	0.237741
7	Spur Gear – 3	AISI 1045	1	0.239969	0.239969
8	Rack	AISI 1045	1	2.83774	2.83774
9	Motor	---	2	3.17947	6.35894
<b>Total</b>					<b>53.864606</b>

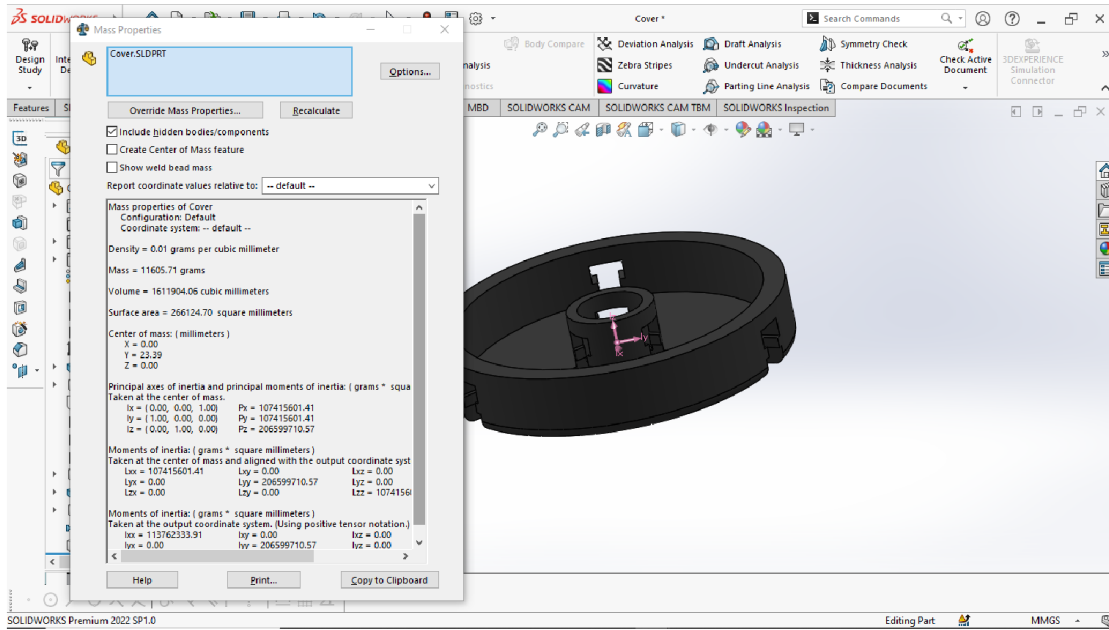


Figure 15: Mass Properties of the Cover Obtained from the Modeling Environment

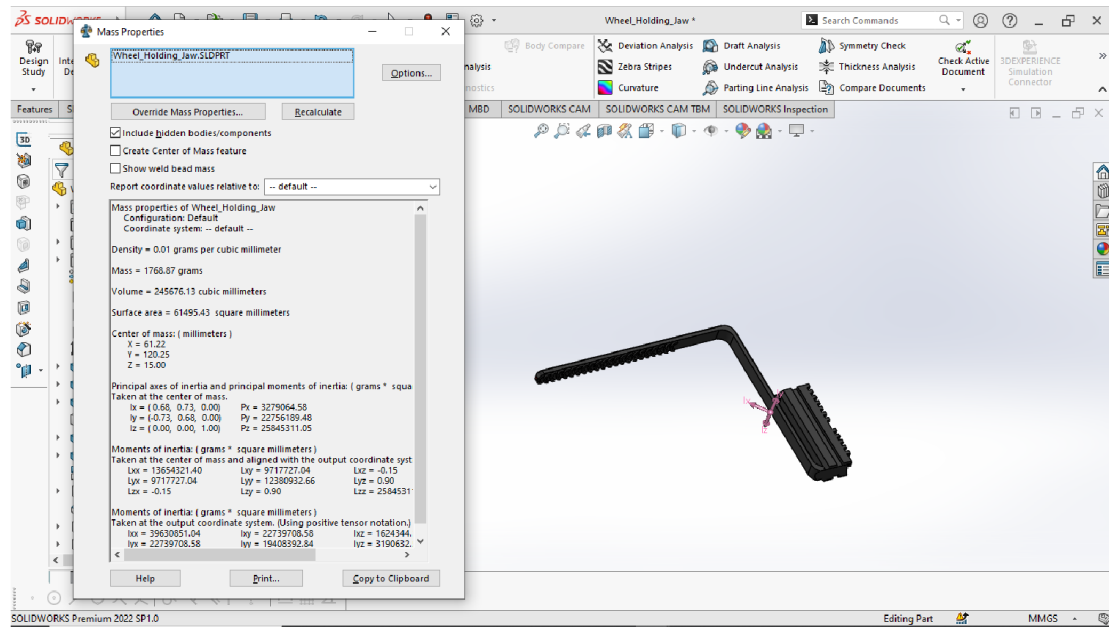


Figure 16: Mass Properties of the Wheel Holding Jaw Obtained from the Modeling Environment

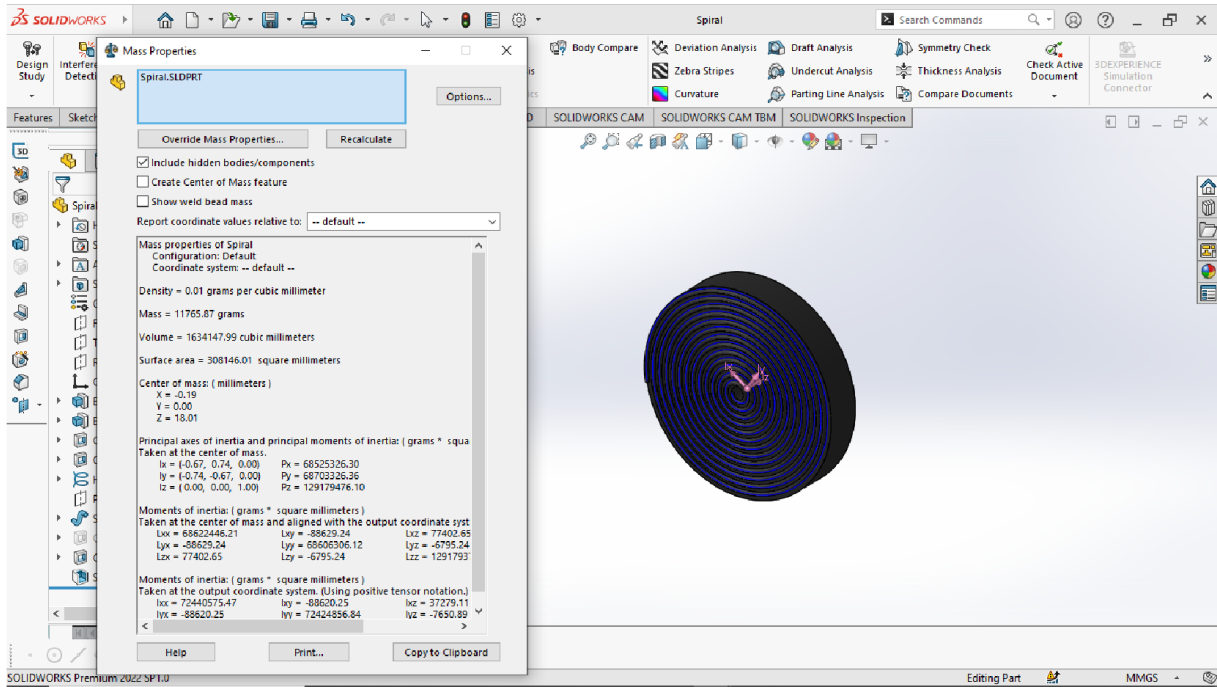


Figure 17: Mass Properties of the Spiral Obtained from the Modeling Environment

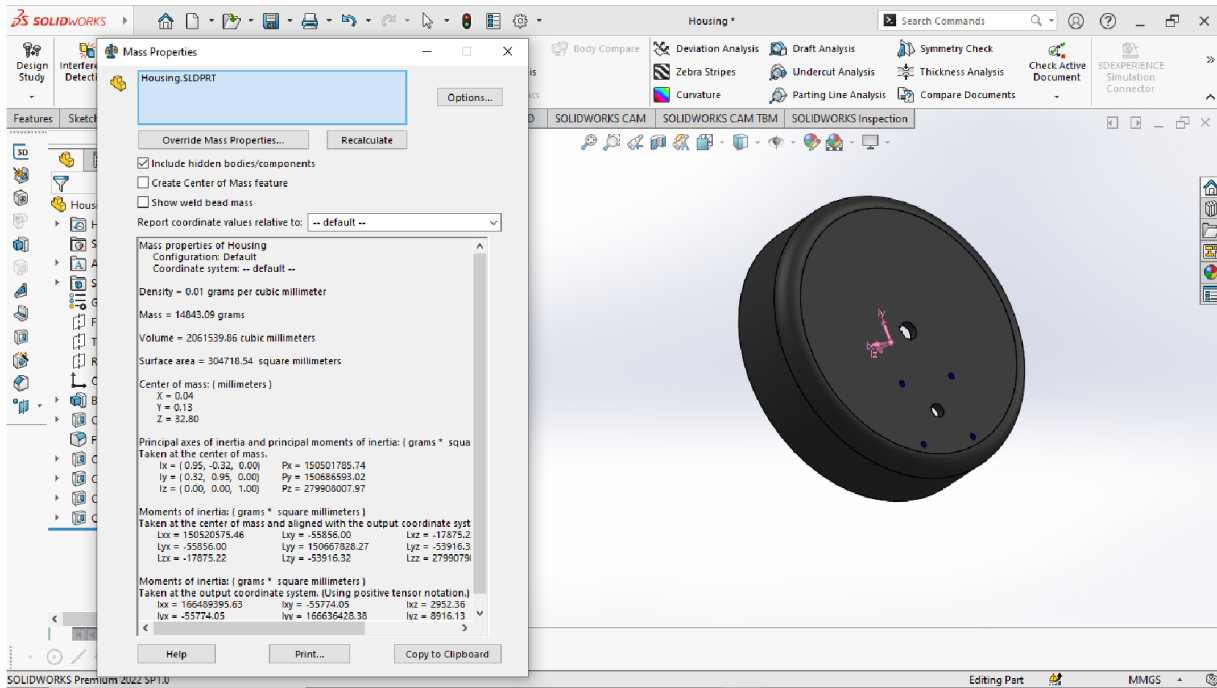
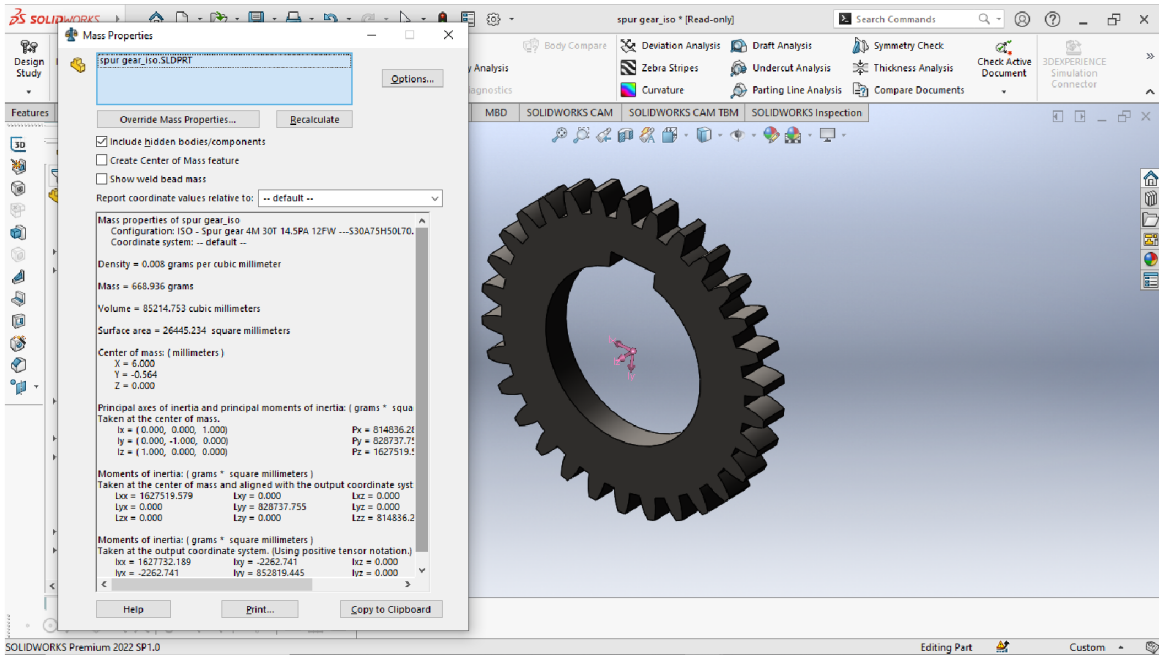
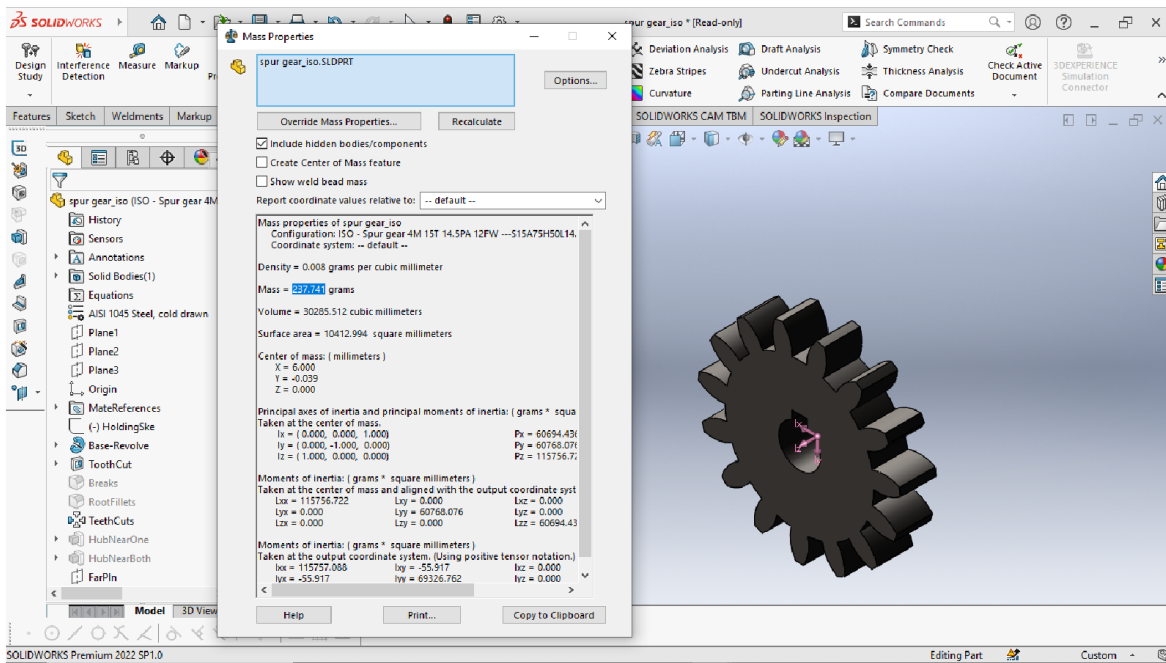


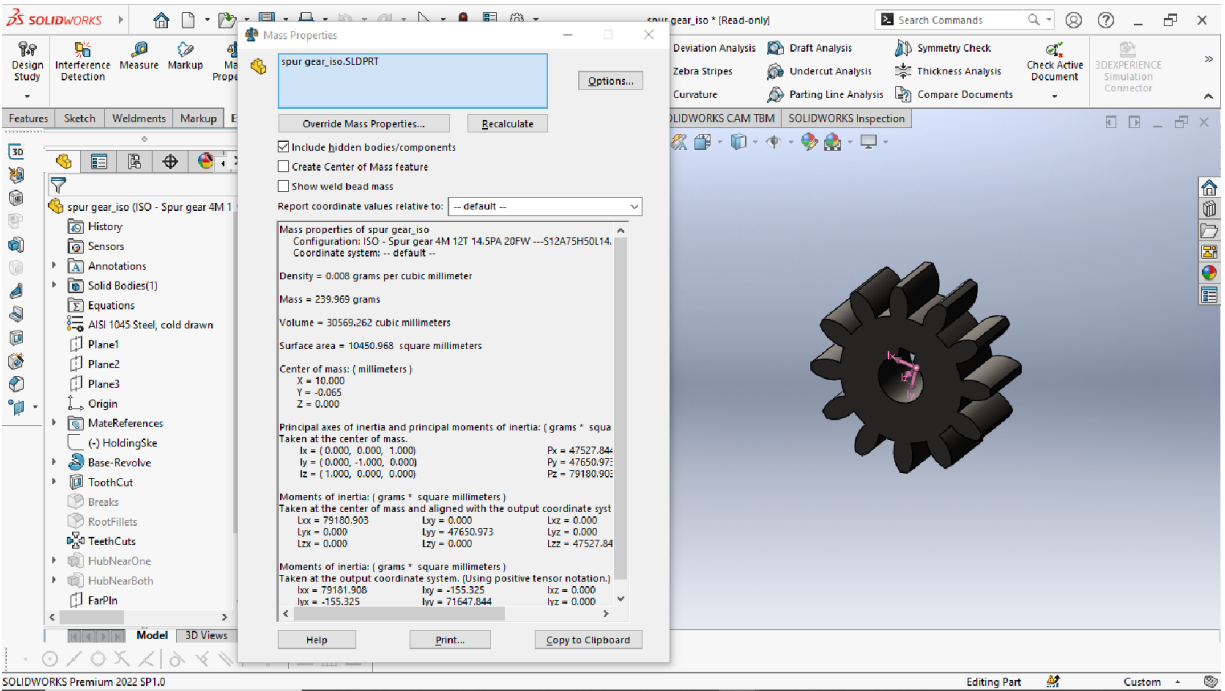
Figure 18: Mass Properties of the Housing Obtained from the Modeling Environment



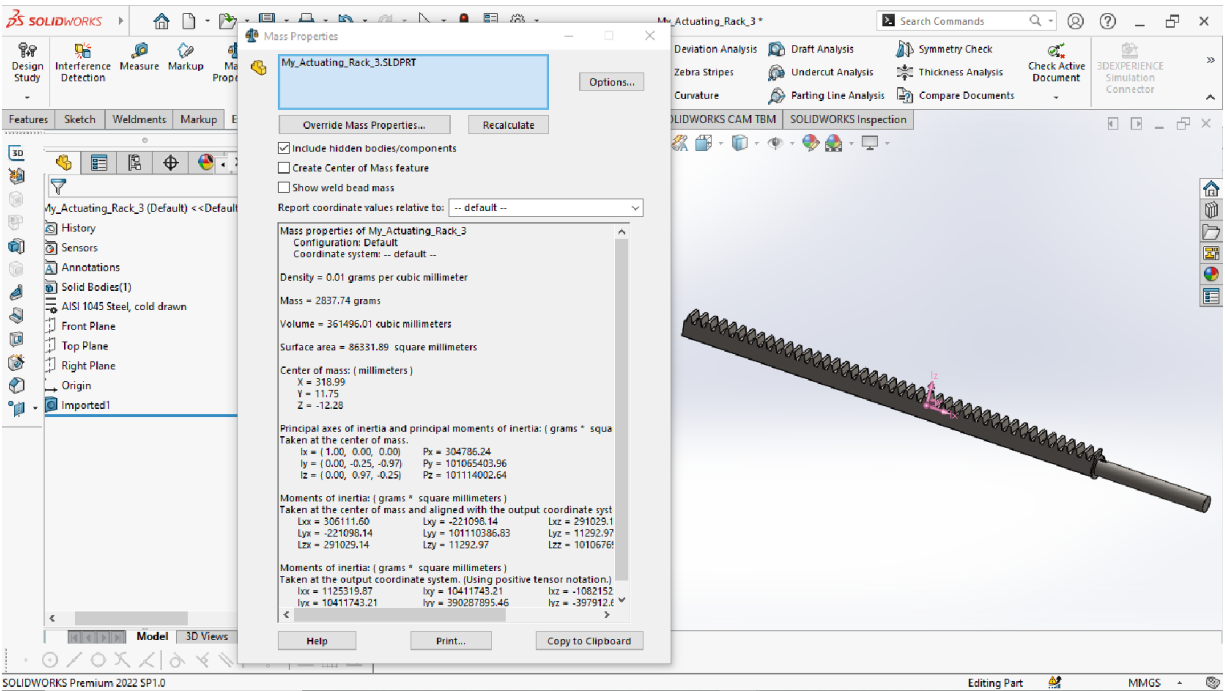
**Figure 19: Mass Properties of Spur Gear – 1 Obtained from the Modeling Environment**



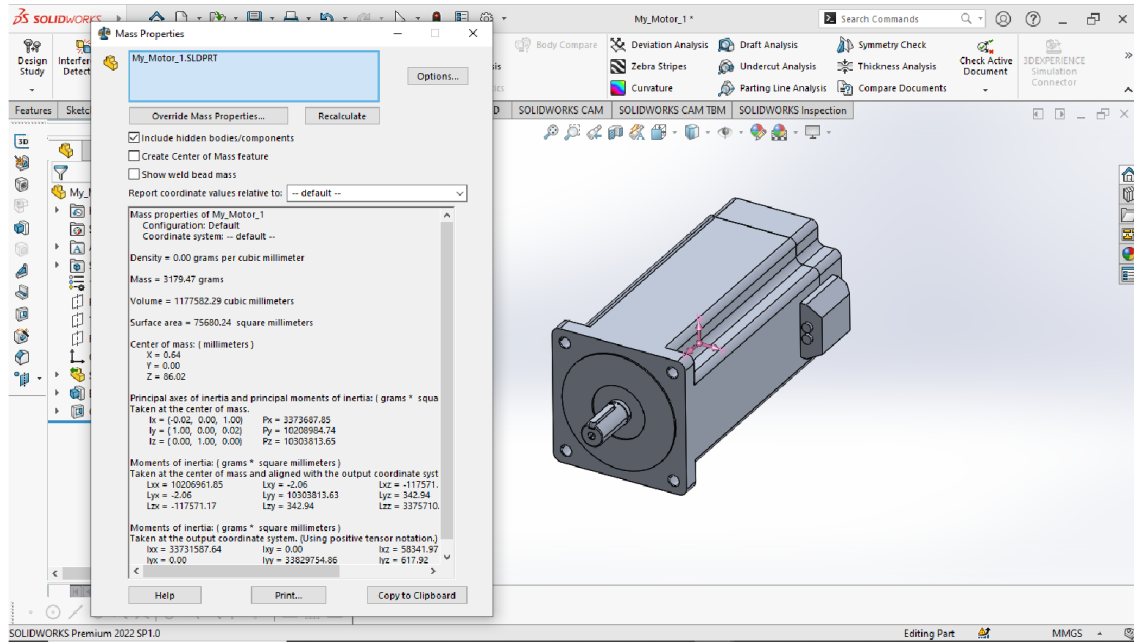
**Figure 20: Mass Properties of Spur Gear – 2 Obtained from the Modeling Environment**



*Figure 21: Mass Properties of Spur Gear – 3 Obtained from the Modeling Environment*



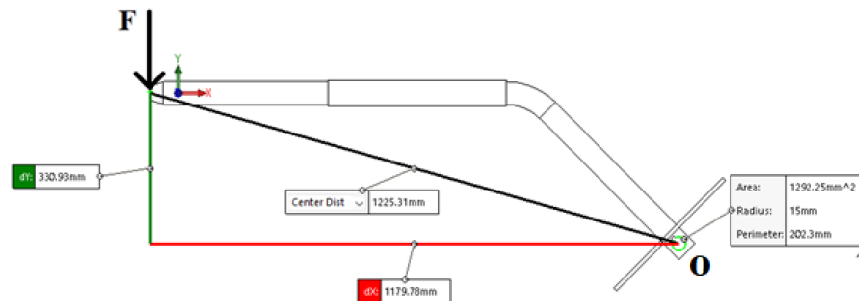
*Figure 22: Mass Properties of Rack Obtained from the Modeling Environment*



**Figure 23:** Mass Properties of Motor Obtained from the Modeling Environment

### 3.2.5.2. Calculation of Externally Applied Torque Due to Tire + Components

Consider the free body diagram of a single tire carrier carrying a tire at its tip shown in figure – 15. The weight of the tire acts downward at the tip of the tire carrier. The tip is selected to take into account the maximum magnitude of the torque that can be imposed on the component during its application. Note also that the weight of the largest tire is considered to get the maximum torque value.



**Figure – 24:** FBD of the Tire Carrier for the Calculation of Torque Resulting from Tire Weight plus Weight of components in the Assembly

The torque about the center of rotation, O, of the tire carrier is given by:

$$T_O = F * d$$

Where:  $F$  is the weight of the tire and  $d$  is the perpendicular distance between the line of action of the force and the torque center O. The distance is:  $d = 1179.78 \text{ mm}$

$$F = W = (m_{\text{tire}} + m_{\text{components}}) * g$$

$$F = (21.364 \text{ Kg} + 53.86 \text{ Kg}) * 9.81(\text{m}/\text{sec}^2)$$

$$F = 737.94 \text{ N}$$

Note that  $m_{\text{tire}} = 21.364 \text{ Kg}$  is the mass of HT275/55R20 113H tire which is the largest tire for the type of passenger vehicle considered for the design of this mechanism.

The torque  $T_O$  then becomes:

$$T_O = F * d$$

$$T_O = 737.94 \text{ N} * 1.17978\text{m}$$

$$T_O = 870.6 \text{ N.m}$$

Note that  $T_O = 870.6 \text{ N.m}$  is torque due to the weight of only one tire and tire holding components. There are two tires (and holding components) on each side of the mechanism and hence, the torque doubles. i.e.

$$\text{Torque on one side of the mechanism} = 2T_O = 1741.2 \text{ N.m}$$

### 3.2.5.3. Calculation of Required Power/Speed

Now that the externally applied torque due to the loaded wheels on the mechanism is obtained, how much input power/speed is required to overcome the externally applied torque and accomplish the desired motion should be determined. In a rotating system, the product of torque ( $T$ ) and angular velocity ( $\omega$ ) gives power ( $P$ ). That is:

$$P = T * \omega$$

Considering one half of the symmetric mechanism, the externally applied torque to be overcome by the motor at the input link of the mechanism is 870.6 N.m. as determined in the previous subsection. Using this torque magnitude, we can iterate using the two unknown variables of power (**P**) and angular velocity ( $\omega$ ) as follows.

**Table – 10: Torque – Speed Iteration to Decide on Initial Crank Speed of the Mechanism**

$\omega$ (rpm)	$\omega$ (rad/sec)	P (W)
50	5.236	4558.46
100	10.472	9116.92
150	15.708	13675.38
200	20.944	18233.85
250	26.179	22791.44

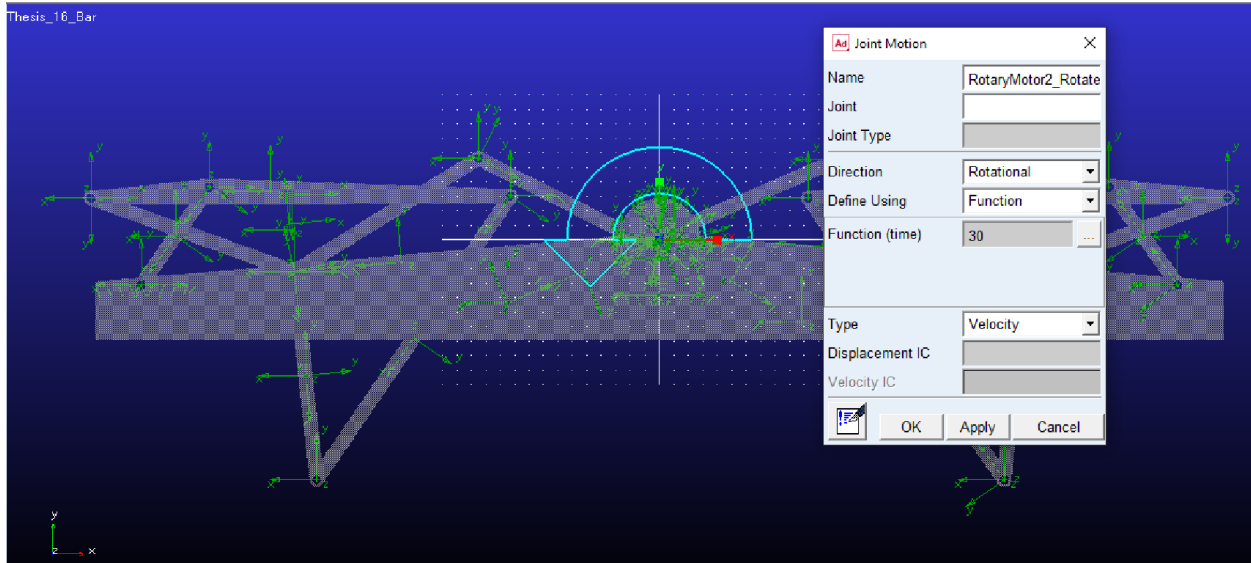
Based on the above first iteration, a design decision is made to make the input speed of the crank 100 rpm (10.472 rad/sec). The justification for this design decision is to use reasonably low power, hence less expensive motor, while obtaining a reasonable input speed. In addition, avoiding too high speeds is important for two reasons – reducing the high power requirements and avoiding excessive accelerations of the links that consequently cause excessive dynamic forces.

It is important to note that the only *input motion parameter* that is needed by the MSC ADAMS program in the subsequent simulation steps is *constant angular velocity* of the crank. The corresponding power is computed by the program taking into consideration not only the externally applied torques but also the inertial effects induced due to the masses of the links.

### **3.2.6. Step – 5: Inserting Driving Motion**

The mechanism is driven by one motor at one location. This is represented in the simulation by inserting a rotary joint motion at the center of the mechanism providing the motion to the shaft of the gears, which, in turn, rotate the two cranks of the mechanism. Note that the gear ratio is 1:1

and there is no speed change by the gears. The only purpose of the gear set is to transmit the motor's rotary motion to the two cranks in the opposite direction. The driving motion is assigned an initial angular velocity of 100 rev/min (10.472 rad/s) in the counter clockwise direction. This is shown in the figure – 16.

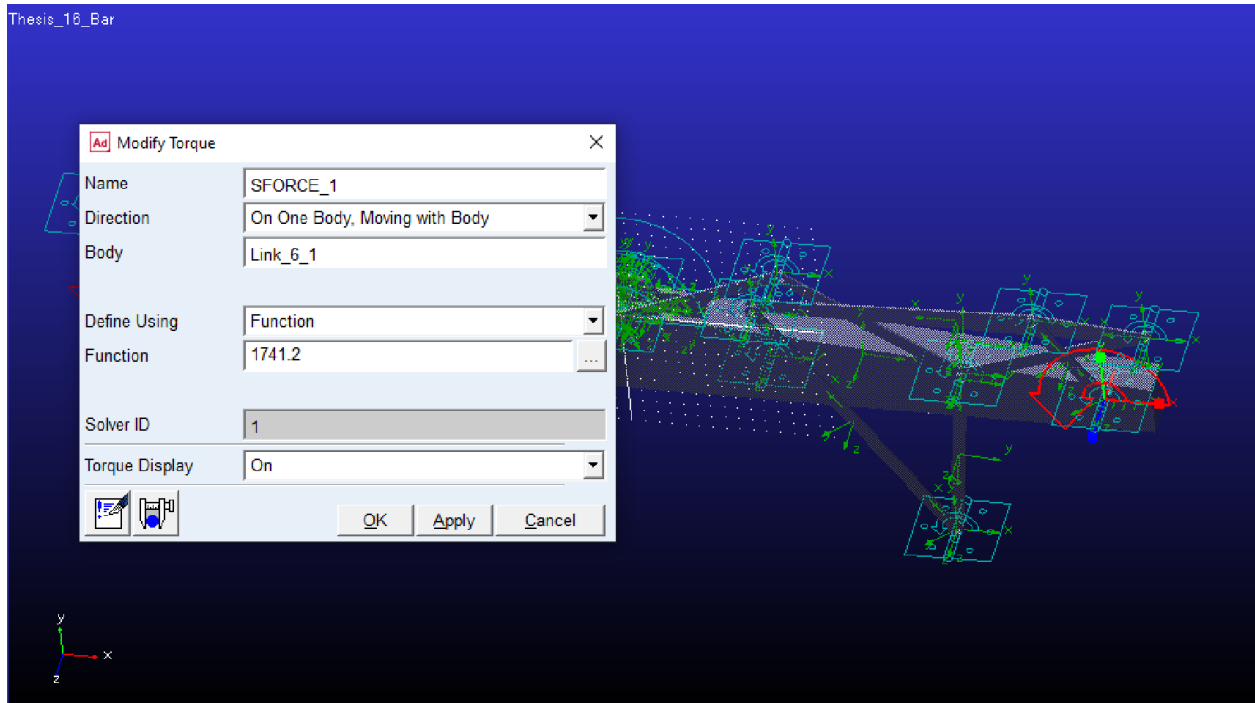


*Figure – 25: Indication of the Origin Location and Selection of Units*

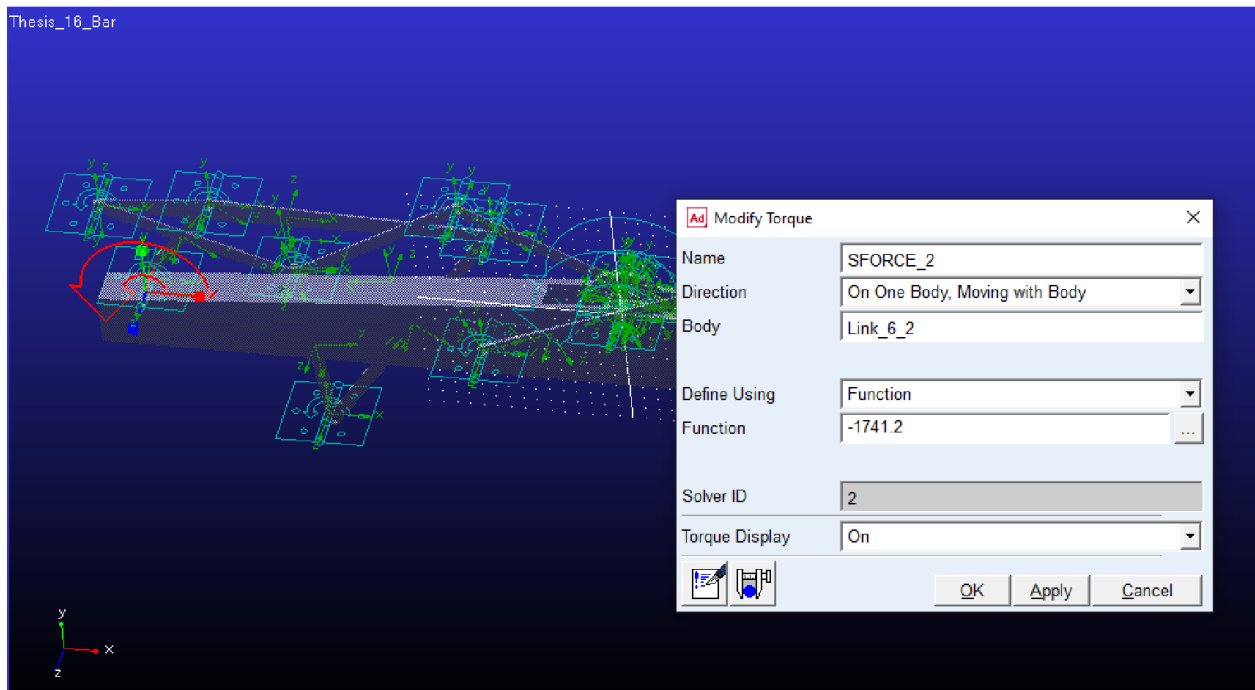
At this step, all externally applied forces and torques that are imposed on the mechanism are inserted into the simulation as inputs. At a time there are four forces that act on the mechanism due to the weights of the wheels. These forces cause torques about the center of rotation of the tire carrier. This torque should be calculated first.

### **3.2.7. Step – 7: Inserting Externally Applied Torques**

The externally applied torque values obtained in section 3.2.5 are inserted in the simulation. This torque acts on the bottom pin of link 6. It acts in the counterclockwise direction for the right part of the mechanism and in the clockwise direction for the left part. It is inserted at the appropriate location as shown in the figures – 17 and 18.



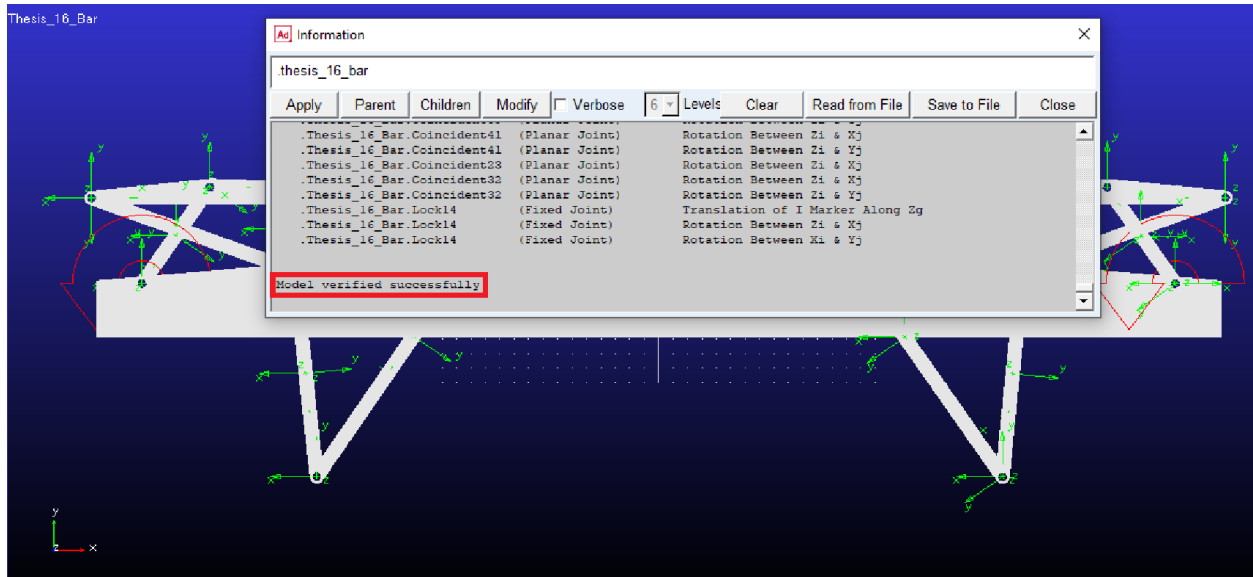
*Figure – 26: Application of Positive (Counterclockwise) Torque at the Right Location of the Mechanism*



*Figure – 27: Application of Negative (Clockwise) Torque at the Left Location of the Mechanism*

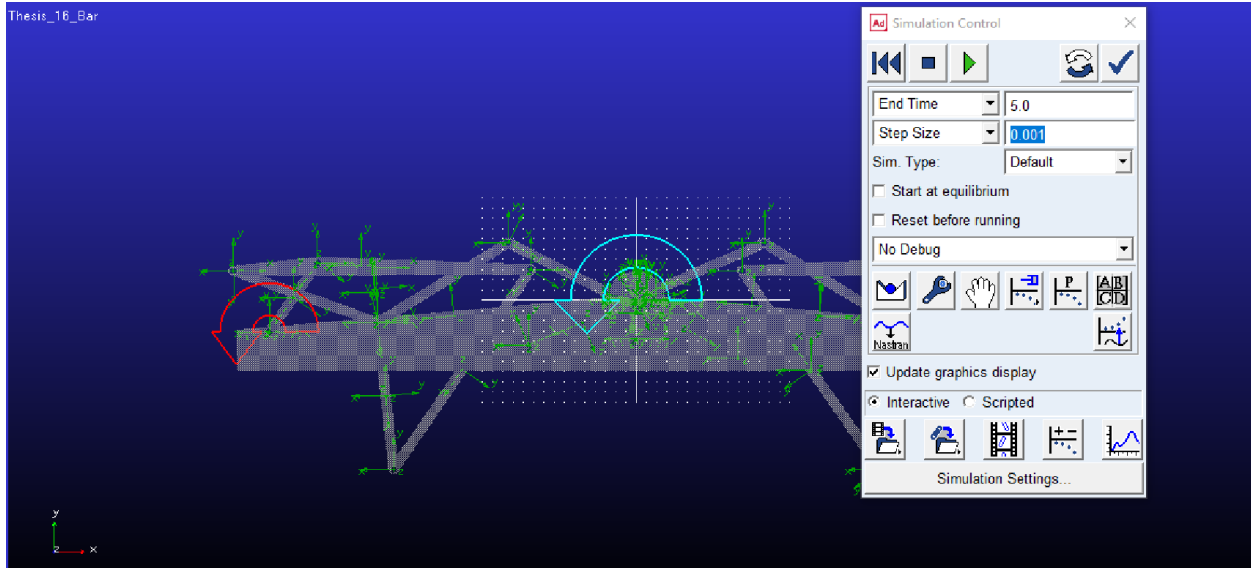
### 3.2.8. Step – 6: Verification of the Model and Running the Simulation

Prior to running the simulation, the validity of the model is checked by model verification. The verification of the model is successful as indicated in the figure – 19.



*Figure – 28: Successful Verification of the Model*

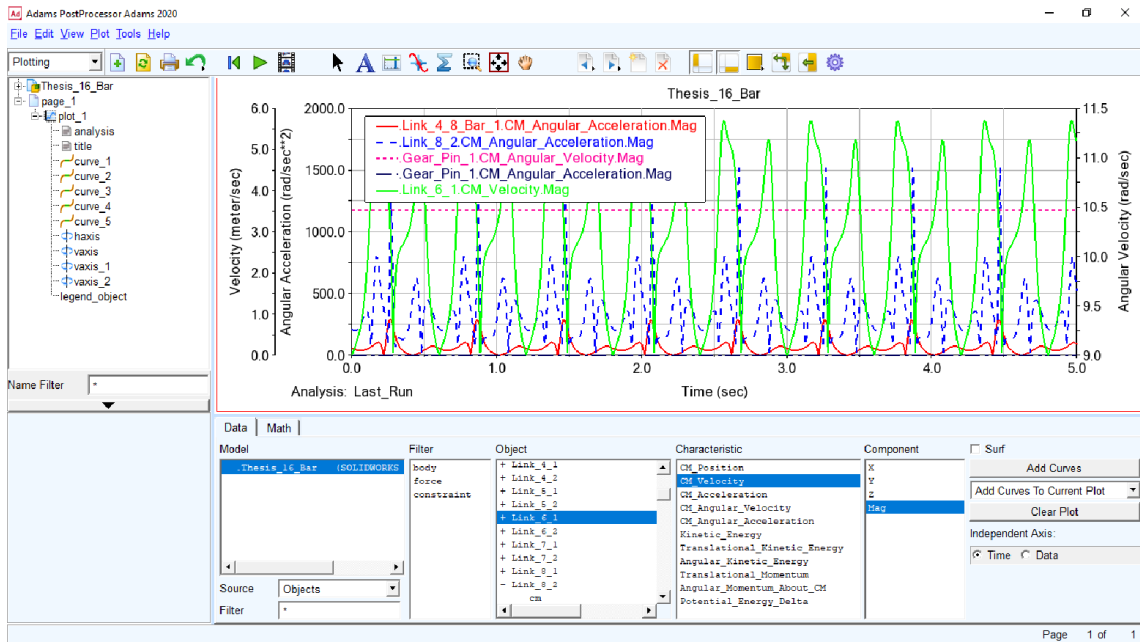
After the successful verification of the model, the next step is to run the simulation. The simulation is run for a simulation time of 5 seconds and with a step size of 0.001 seconds. This step size ensures a very fine simulation which results in high quality smooth curves with finely generated data. The simulation control setup is shown in the figure – 20.



*Figure – 29: Simulation Setup*

### 3.2.9. Step – 7: Post Processing

Following the completion of the simulation, the results are viewed in the post processing stage. As it is shown in the figure – 21, various simulation outputs are obtained as a function of time. These include linear and angular positions, velocities and accelerations. The detailed presentation and explanation of these results of simulation is covered in the next separate chapter on results.



*Figure – 30: Post Processing Stage of the Simulation*



## **Chapter 4**

### **Results and Discussion**

In this chapter, the results obtained by the methodology followed in the previous chapter are presented and discussed. The chapter has two major sections. The first section includes the presentation and discussion of the results of synthesis performed in this thesis. The second section is dedicated for presenting the results of kinematic and dynamic simulation.

#### **4.1. Results of Synthesis**

The general objective of the study was to design a mechanism to be used for the removal and installation of all wheels to/from a vehicle simultaneously. The specific objectives were to synthesize a new mechanism for the task, to make kinematic simulation and finally to make dynamic simulation for the synthesized mechanism. This section presents the results of the synthesis work.

The systematic methodology followed to synthesize the mechanism were presented in chapter-3. This methodology led to thoroughly understanding the overall function and the sub-functions of the device. Based on this function, form is synthesized. This synthesized form, which itself is composed of different mechanical components, their configuration and the way they are connected to each other is a new, unique and novel result of this work. This synthesis enables the function intended in the objective, namely simultaneous wheel removal and installation of all wheels of a vehicle.

##### **4.1.1. Synthesis Results – Overall Assembly**

The assembly of the newly synthesized overall system that enables simultaneous removal and installation of all wheels to/from a passenger vehicle is shown in figure – 31 below. It is composed of the ground support, links 1-8, two spur gears, a planet coupler, a hub for internal gear, a connecting shaft, and a tire carrier. Each of these components, their configuration and the way they are connected and constrained is a unique output of this study. The geometric arrangement of all of these components is critical for the functioning of the overall system. It is this synthesized

configuration that enables the wheel positioning function, how it is driven and how functional components are geometrically arranged.

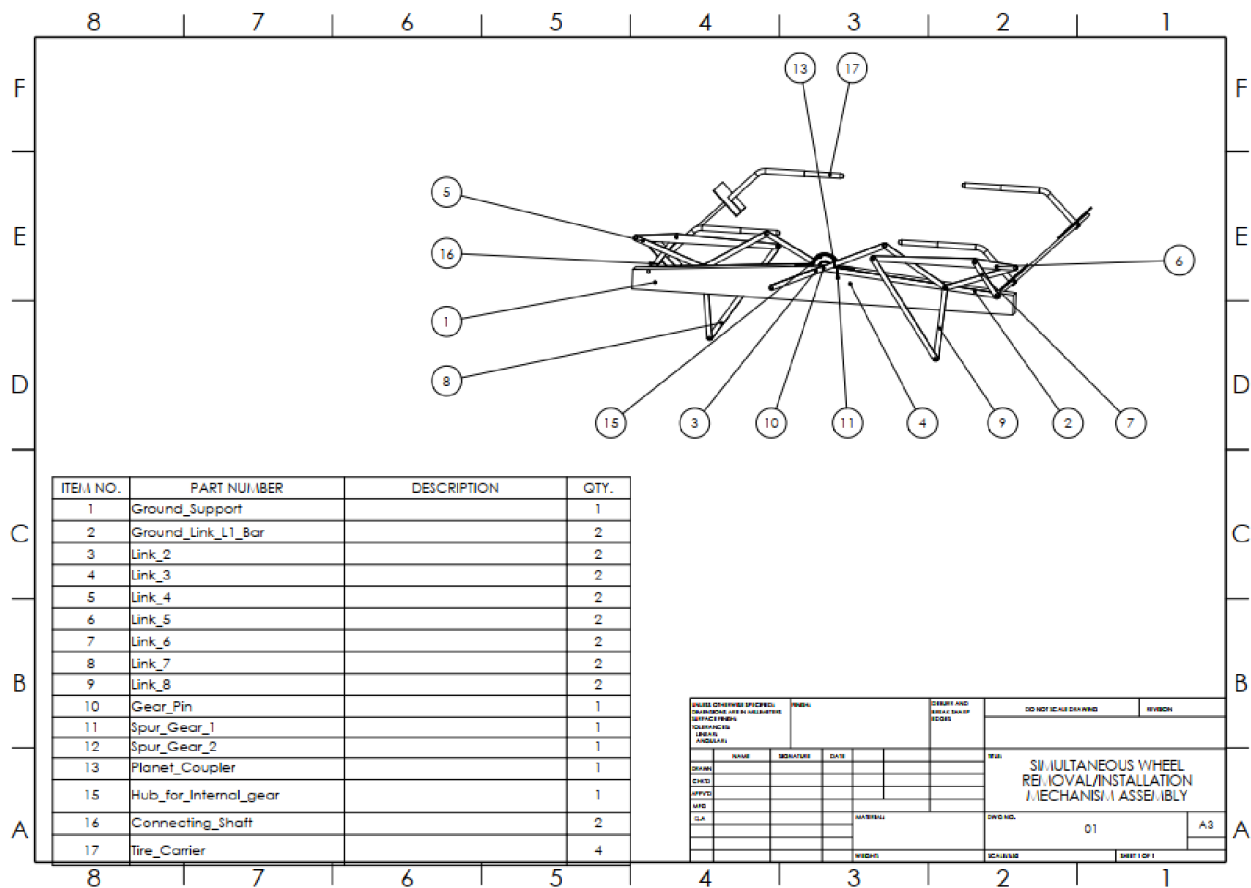


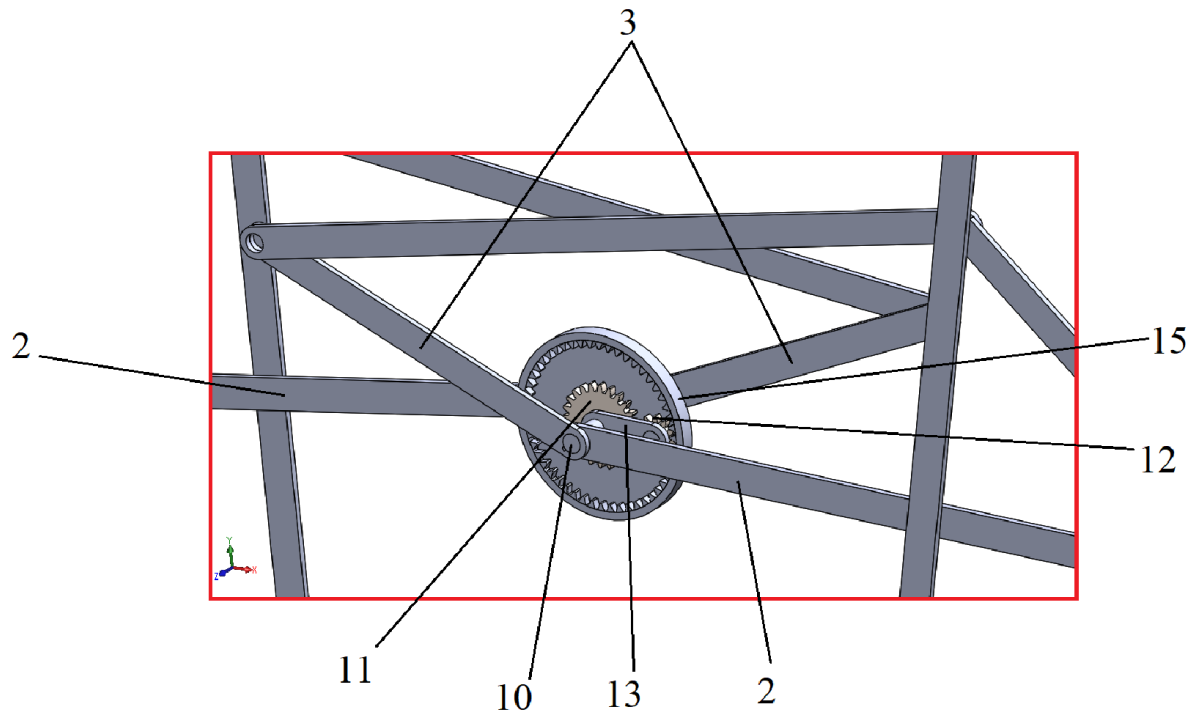
Figure – 31: Overall Synthesis Result: Components and Assembly

#### 4.1.2. Central Driving Mechanism Synthesis Results

The driving mechanism is the central part of the overall system that is responsible to transmit rotational motion from an input motor to the four tire carrier units. This mechanism was synthesized following the systematic design procedure, function understanding, multiple concept generation and iterative mechanism geometric synthesis and selection processes explained in detail in chapter 3. The final solution is the geometric arrangement of the components shown in figure-31 above.

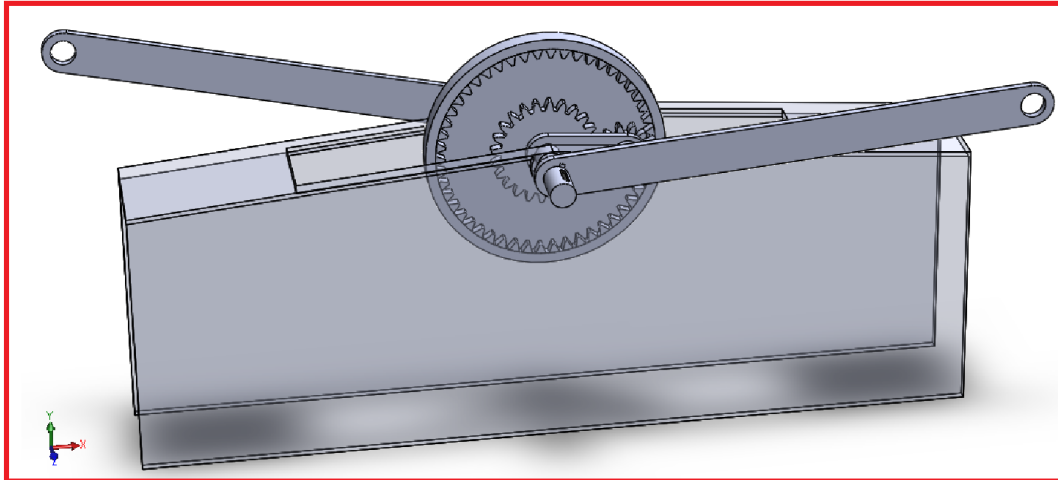
The central driving mechanism is composed of the components: gear pin, link-2 of the right and left half 8-bar linkage, link-1 of the right and left half linkage, planet coupler, spur gear-1, spur

gear-2, and hub of internal gear. The particular arrangement of the above-mentioned synthesized components in the assembly is shown in figure – 32 below.



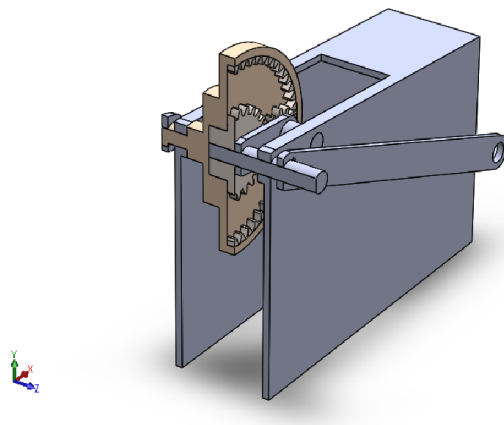
**Figure – 32:** Synthesized Central Driving Arrangement of Components. Numbers Indicate, 2: Link-1, 3: Link-2, 10: Gear Pin, 11: Spur Gear-1, 12: Spur Gear-2, and, 15: Hub of Internal Gear. Note that the Static Ground Support is removed to Show the Internal Components

The mechanical arrangement shown in figure-32 above enables to drive the entire wheel positioning system by just one motor (or any other type of prime mover that produces rotational motion) – the other alternative to this arrangement would be driving the each of the left and right wheel positioning subassemblies by one motor for each which would be twice the number of motors as compared to the current design. The advantage of this is the cost effectiveness of being able to drive the wheel positioning system by one motor only. The details of this synthesized configuration are shown in figure-34 below.



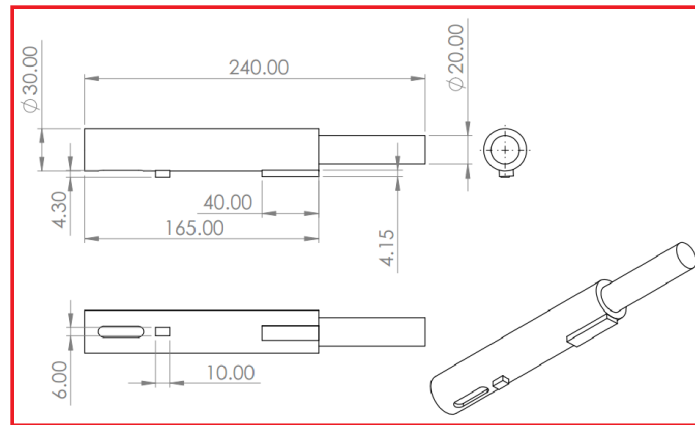
**Figure – 34:** *Assembly Configuration of the Synthesized Drive for the Wheel positioning Module. The Ground Support Housing is Made Transparent for Visibility. Note the Single Motor Input Coupling Location Provided at the Center*

The gear cross section of the assembly is presented in figure-35 below. This configuration is comprised of the above – mentioned components. The cross section of the assembly is shown in the figure for clarity. It is important to note that the shaping, geometric sizing and configuration of the nine components in this manner is a unique result of this work.



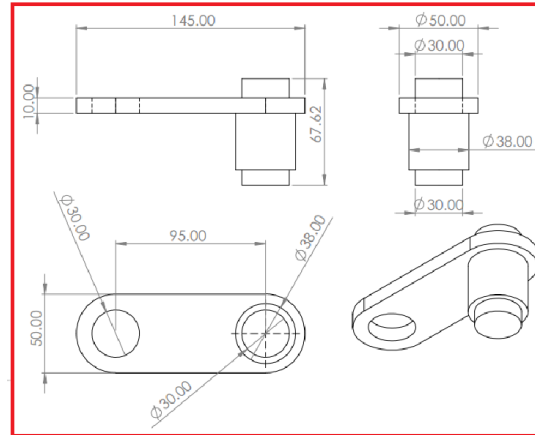
**Figure – 35:** *Cutaway View Showing Assembly Configuration of the Synthesized Drive for the Wheel positioning Module*

The synthesized component called, the gear pin, is designed to transmit rotational motion from an input motor to spur gear-1 and link-2 of one half of the 16-bar mechanism. It also serves as a shaft transmit the entire load on the drive train to the housing on one side. The other side load supporting is done by the hub of the internal spur gear as will be discussed below. In addition to this, the gear pin supports one side of the planet coupler which in turn supports spur gear-2. The geometric dimensions of the gear pin that enable all of these functions, which are results of the synthesis work, are presented in figure-36 below.



**Figure – 36: Gear\_Pin Geometry and Dimensions (mm)**

The planet coupler is another component the geometric features and dimensions of which are specifically synthesized to enable a compact drivetrain for the wheel positioning module. The planet coupler provides the function of providing a revolute joint support for spur gear-2. It itself is supported by a revolute joint formed with the gear pin on one side and with the ground support housing on the other. It is so shaped and geometrically sized in such a manner that enables these functions. The results of this, that is the geometry and dimensions of the planet coupler, are shown in figure-37 below.

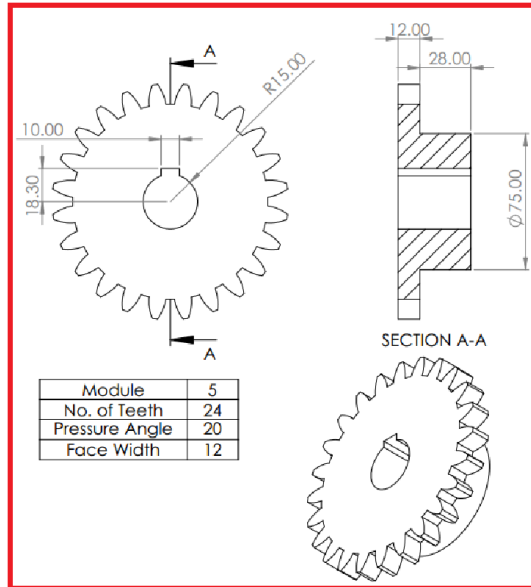


**Figure – 37: Planet\_Coupler Geometry and Dimensions (mm)**

The function of the gear train composed of spur gear-1, spur gear-2 and internal spur gear with hub is to take input motion and convert the direction of motion. Rotational motion enters the gear train through the gear pin to spur gear-1 which drives spur gear-2 making it rotate in the opposite direction (because they are external gear set). Spur gear-2 drives the internal spur gear making it rotate in the same direction to itself (because they are external-internal gear set).

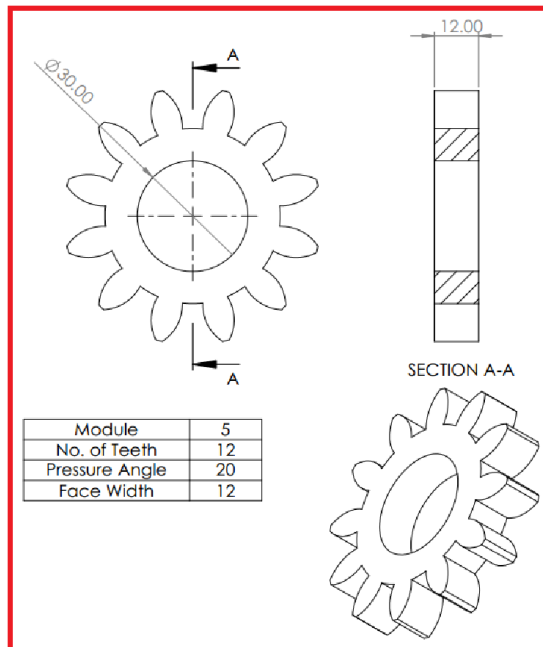
This rotational direction conversion enables driving one half of the 16-bar mechanism in the clockwise direction and the other half in the counterclockwise direction. This gear train is designed so that the motion input at the gear pin – and thus the right half of the 16-bar mechanism, viewed facing the gear pin, - always rotates clockwise. Therefore, only a clockwise-rotating motor must be used as a prime mover of this mechanism.

Note that spur gear-1 is designed to be driven by the gear pin by means of a key at its center. It meshes and drives with spur gear-2 which in turn meshes and drives the internal spur gear. The hub of the internal spur gear drives the other half of the 16-bar mechanism. The hub of spur gear-1 also serves the function of creating a revolute joint support for the hub of the internal spur gear. Figure-38 shows the details and dimensions of spur-gear 1 of this drive train.



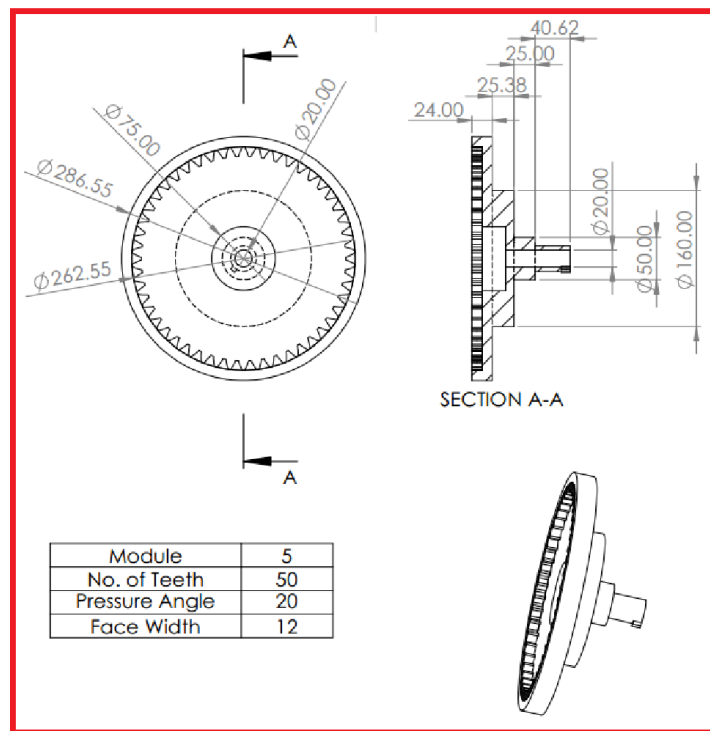
**Figure – 38: Spur\_Gear\_1 Geometry and Dimensions (mm)**

Spur gear-2 has the function of transmitting rotational motion from spur gear-1 to the internal spur gear by changing the direction of motion. Because of this sole function of the gear, it does not transmit motion to or from any shaft thereby having a keyless center. This gear is supported by the planet coupler explained above. Its dimensions and geometry are shown in figure-39 below.



**Figure – 39: Spur\_Gear\_2 Geometry and Dimensions (mm)**

Another synthesized component of the drive train is the internal spur gear with hub. The detailed features and geometric dimensions of this gear, particularly its hub, are created in a manner that allows space-efficient configuration. The internal circular recess in the hub provides for a revolute joint support of spur gear-1. The outside extrusion of the hub serves two functions: transferring the load of the drive train assembly to the ground and driving link-2 of the second half of the 16-bar mechanism by means of a key. The details of the geometric features and dimensions of this component that enable the functions stated are the results of synthesis work of this thesis. They are shown in figure-40 below.

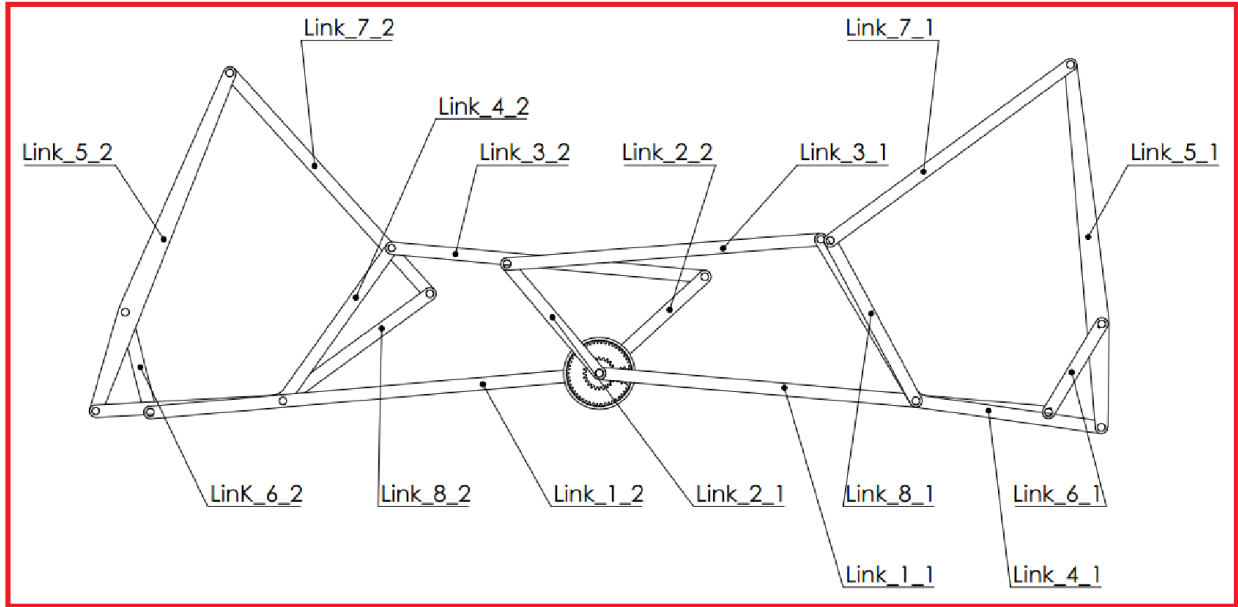


**Figure – 40:** *Internal\_Spur\_Gear\_with\_Hub Geometry and Dimensions (mm)*

#### 4.1.3. Symmetric 16-Bar Mechanism Synthesis Result

In the assembly of synthesized components in this original design work, to accomplish the identified and decomposed function of the wheel positioning module, two alternative design concepts were generated and the symmetric 16-bar mechanism was selected. This systematic process was presented in the previous chapter. In this subsection, the focus is on presenting the results of the synthesis work – which is the geometric configuration of the members in this

assembly and their critical dimensions that create the desired motions that fulfill the function. Figure-41 below shows the double 16-bar mechanism. The detailed geometric dimensions of each link is given in the drawings shown in Appendix A.2 to A10.



**Figure – 41:** *The link Arrangement of the Synthesized Symmetric 16-Bar Mechanism. See the Geometry and Dimensions of Each link in the Drawings Included in Appendix A.2 to A10.*

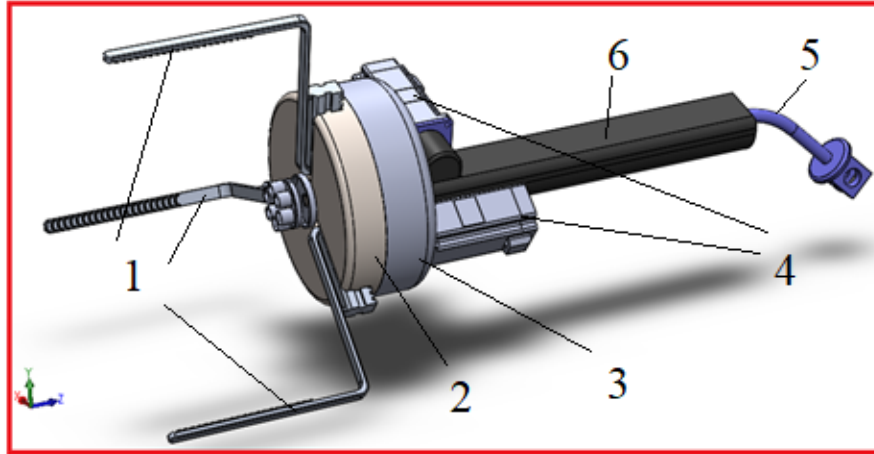
The function of this symmetric 16-bar linkage is to transmit motion to the wheel positioning tire carriers through the connecting components. It is important to note the type of motion each of the links in the mechanism undergo. There are three types of motion namely pure rotation, pure translation and complex motion. All links in the mechanism and their corresponding motion types are presented in table – 11.

**Table – 11: Links in the Symmetric 16-bar Mechanism and their Corresponding Motion Types**

	<b>Motion Type</b>			
	<b>No Motion (Stationary)</b>	<b>Pure Rotation</b>	<b>Pure Translation</b>	<b>Complex Motion</b>
<b>Link Name</b>	<b>Link_1 (Ground)</b>	Link_2_1 (Right Crank)	-	Link_3_1
	-	Link_2_2 (Left Crank)	-	Link_3_2
	-	Link_4_1	-	Link_5_1
	-	Link_4_2	-	Link_5_2
	-	Link_6_1	-	Link_7_1
	-	Link_6_2	-	Link_7_2
	-	Link_8_1	-	-
	-	Link_8_2	-	-

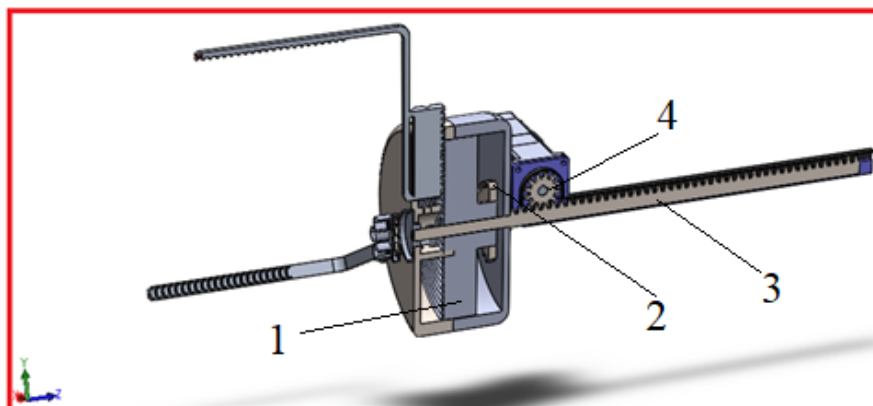
#### **4.1.4. Wheel Holding Module Synthesis Result**

In this original design work, the synthesis of the wheel holding module resulted from the systematic functional decomposition methodology presented in chapter 3. The function of this module is to hold and support the wheel being removed/installed. It is composed of six components, namely, wheel holding jaws, holder cover, jaw locating spiral, holder housing, spur gear-3 and spur gear-4. These components are geometrically sized and arranged in such a manner that module can create the needed function to hold different wheel sizes from the smallest to the largest size for that of passenger cars as established by the literature review. Figure-42 below shows the synthesized wheel holding module (and the attached width adjustment sub-module which is discussed in the next subsection).



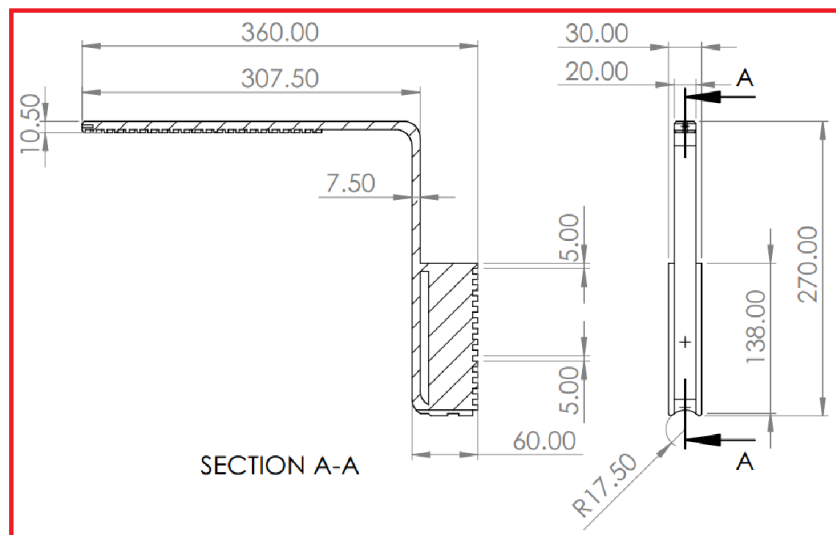
**Figure – 42:** *The Wheel Holding Module Synthesized in this Work. Components Shown in this Figure are: 1- Wheel Holding Jaws, 2- Holder Cover, 3- Holder Housing, 4- Driving Motors, 5- Tire Carrier and 6- Rack Cover*

The cutaway solid model shown in figure-43 below shows the internal components of the wheel adjustment module. Notice that behind this module is an attached sub-module the dual function of which is providing wheel removal/installation motion and adjusting for track widths of different passenger cars within the range of maximum and minimum track width values set by literature as discussed in chapter-2. This configuration, and the geometric shaping and sizing of the critical components that produce the motion are unique results of this work.



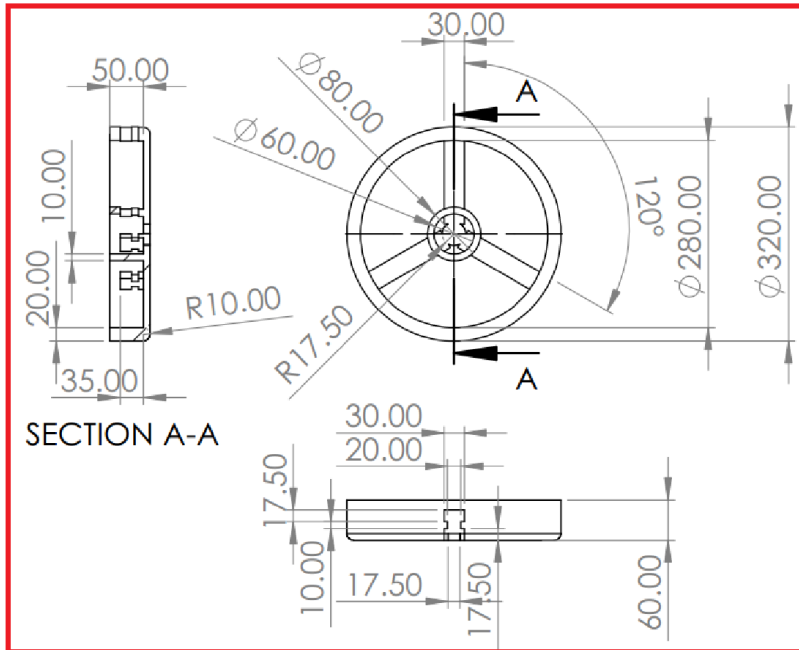
**Figure – 43:** *Cutaway View of the Wheel Holding Module and the Width Adjustment Sub-Module. Components Shown are: 1- Jaw Locating Spiral, 2- Spur Gear-4, 3- Extending Rack, 4- Spur Gear-5*

In the wheel holding module, a critical component is the wheel holding jaw. There are three wheel holding jaws per module. In this work, the embodiment of the jaw is specifically synthesized and sized to be able to accommodate both the largest and the smallest wheels in the passenger car category. The three wheel holding jaws has teeth in their base structure that mates with the wheel locating spiral, which locates their radial motion. In addition, the side of the base of the wheel holding jaw forms a sliding joint with the three slots in the holder cover. Figure-44 below shows the geometric and dimensional details of this synthesized component.



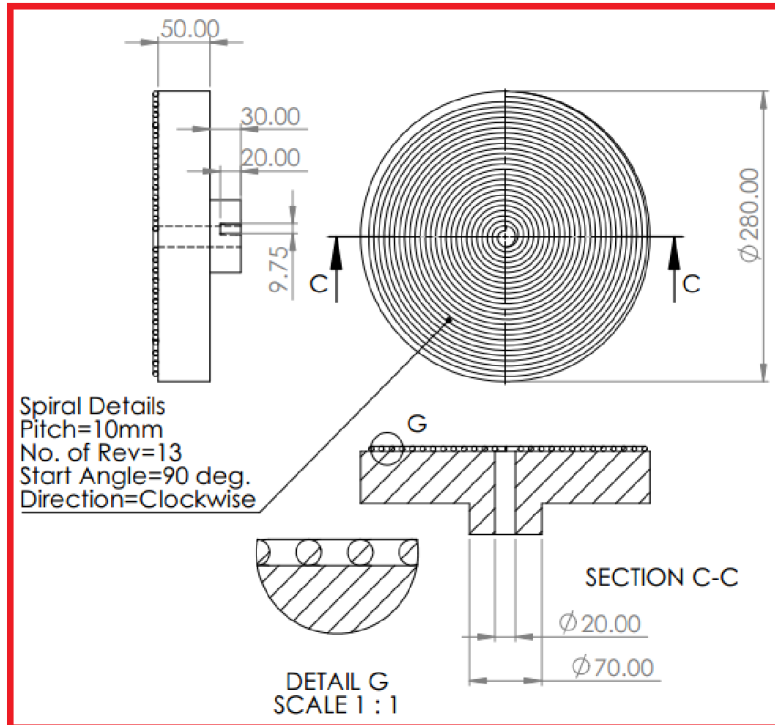
**Figure – 44:** Wheel Holding Jaw Geometry and Dimensions (mm)

Another component in the wheel holding module synthesized in this work to provide the function of the module is the holder cover. This component has three straight slots that guide the radial motion of the wheel holding jaws. The holder cover is itself attached to the holder housing and rotates with respect to the jaw-locating spiral. The holder cover's geometric details and dimensions synthesized in this work are shown in figure-45 below.



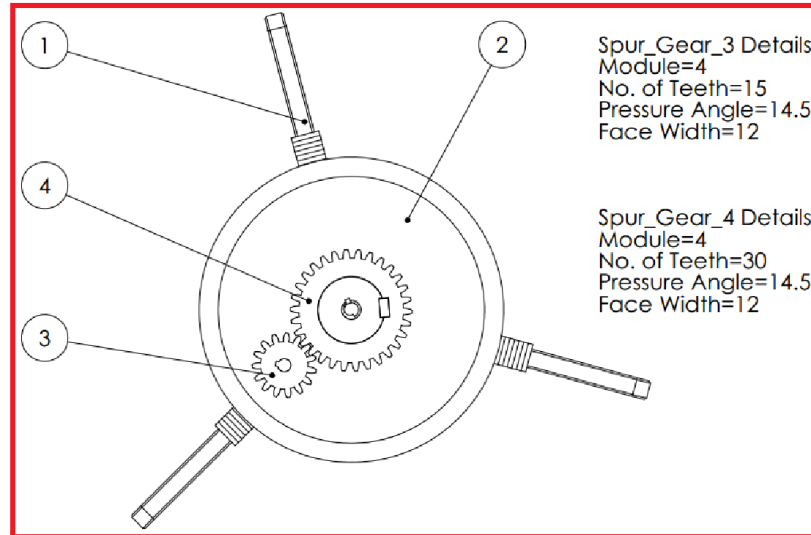
**Figure – 45: Holder Cover Geometry and Dimensions (mm)**

A component responsible for the radial expansion and contraction motion of the wheel holding jaws is the jaw locating spiral. This component is driven by spur gear-4 by which it is concentric and rotational motion is transmitted by a key. The spiral path on the interior face of this component is designed to mesh with the teeth on the base of each wheel holding jaw. The rotational motion of the jaw locating spiral is transformed through this mesh to the radial translation of the three wheel holding jaws. The slot in the holder cover plays a vital role of constraining the motion of the jaws in a straight radial direction. The geometric features and dimensions that enable the described operation of the wheel holding jaws to adjust for and hold the wheels of different size passenger vehicles within the range determined by literature review is a unique output of this study and is shown in figure-46 below.



**Figure – 46: Jaw Locating Spiral Geometry and Dimensions (mm)**

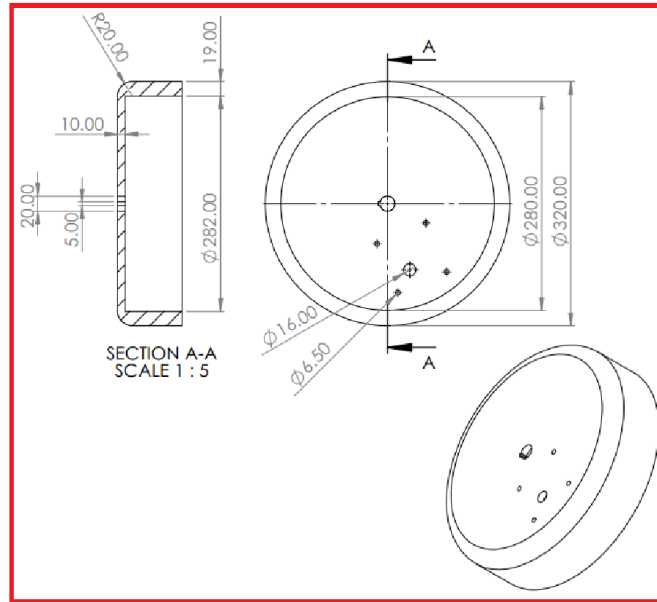
The jaw locating spiral is driven by spur gear-4 by means of a key. Spur gear-4 is itself driven by its pinion – spur gear-3. This pinion is driven by bidirectional stepper motor the direction of rotation of which determines whether wheel is being gripped or released. Geometric and dimensional details of this gear drive system used in the module is shown in figure-47 below.



**Figure – 47: Gear Drive of the Jaw Locating Spiral- Geometry and Dimensions (mm)**

*Numbers indicate: 1-Wheel Holding Jaw, 2-Jaw Locating Spiral, 3-Spur Gear-3, and 4-Spur Gear-4.*

The final component in the discussion of this subsection is the holder housing. This component is sized and configured with neighboring components in the module in such a way that it supports all the above components on it. In addition, this component provides a mounting support for the motor that drives spur gear-3 shown above. This component is itself supported by the extending rack of the width adjustment system. Figure – 48 indicates the synthesized holder housing, its geometry and dimensions.



**Figure – 48:** Holder Housing Geometry and Dimensions (mm)

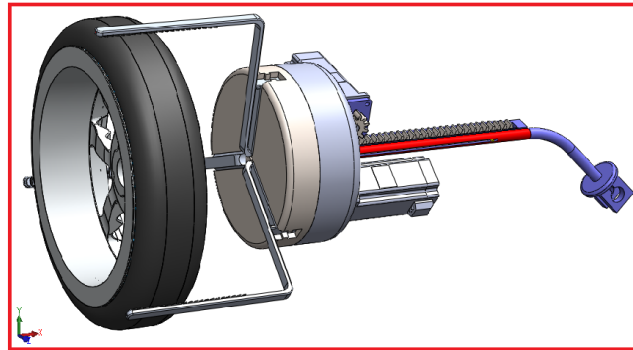
#### 4.1.5. Achieved Maximum and Minimum Wheel Holding Diameters and Wheel Widths

The synthesized wheel holding module has enabled the function of adjusting the diameter the jaws can hold to include both the maximum and minimum wheel diameters. These maximum and minimum wheel diameters for the largest and smallest passenger cars were obtained by literature review. Accordingly, the maximum wheel diameter for the largest passenger car is 810.5 mm and the minimum wheel diameter for the smallest passenger car is 530.2 mm. In addition to the diameters, the jaws of the wheels are designed to hold the largest (and hence the smallest) wheel widths which are 275 mm and 125 mm respectively (See tables A.11.1 and A.11.2 in the Appendix for full tire and rim specifications).

**Table – 12:** Results of Synthesizing the Wheel Holding Module: Achieved Maximum and Minimum Wheel holding Diameters and Wheel Widths

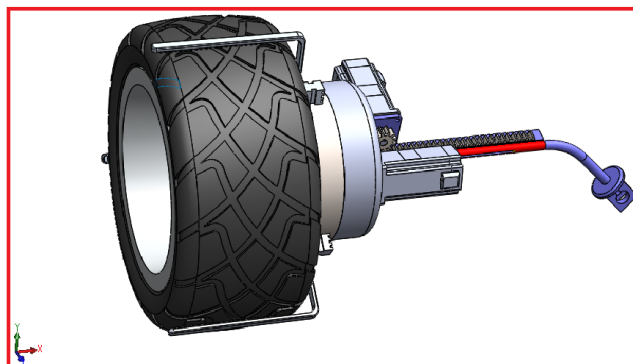
Operation characteristics of the synthesized wheel holding module	Dimensional Parameter (mm)
Largest wheel diameter held	810.5
Smallest wheel diameter held	530.2
Width of largest wheel held	275.00
Width of smallest wheel held	125.00

The wheel holding module synthesized in this original design work has components that are geometrically sized, shaped and assembled to be able to hold these maximum dimensions of wheels. This achieved result is summarized and presented in table 12 above. The wheel holding module while holding the smallest wheel size is shown in figure 49 below. Also note that the depth of the jaws is more than sufficient to hold the width of the wheel which is 125 mm.



**Figure – 49:** *The wheel holding module while holding the smallest size wheel. At this position, the jaws are opening to circumscribe a diameter of 530.2 mm.*

Figure-50 shows the wheel holding module while holding the largest passenger car wheel. This is the case where the three wheel holding jaws are to be adjusted to hold the wheel of the largest vehicle the simultaneous wheel removal/installation mechanism is designed for. Also note that the depth of the jaws is sufficient to hold the width of the largest wheel which is 275 mm.

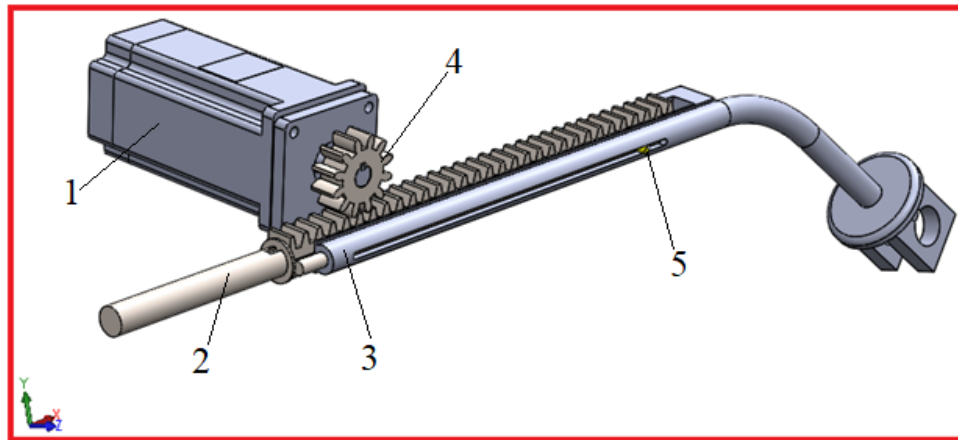


**Figure – 50:** *The wheel holding module while holding the largest size wheel. At this position, the jaws are opening to circumscribe a diameter of 810.5 mm.*

#### 4.1.6. Width Adjustment Assembly Synthesis Result

In this work, the width adjusting assembly is synthesized to fulfill the function of adjusting for the largest and smallest width of vehicles in the passenger car category. Note that this maximum and minimum sizes were determined by literature review (*see chapter-2, section 2.1*). The width adjustment assembly, in addition to its namesake function- adjusting the wheel holder module for different vehicle widths, also serves the function of horizontally positioning the held wheel to and from the vehicle during installation and removal respectively.

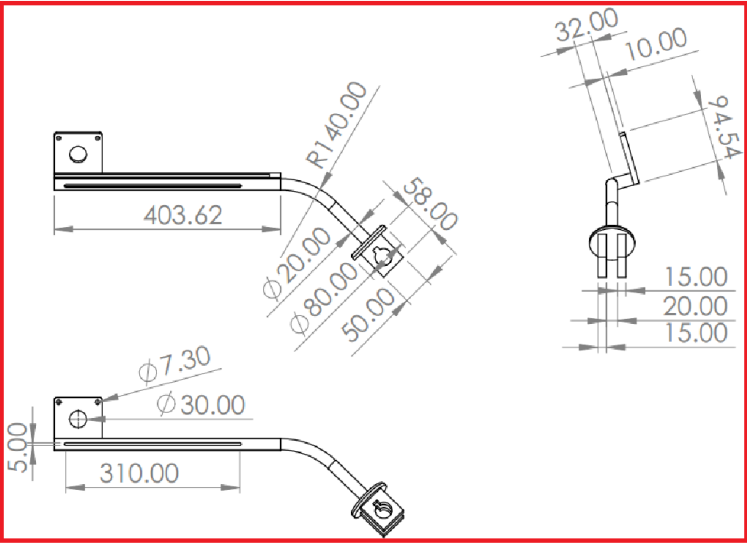
From modularized design perspective, the wheel adjustment mechanism is a submodule within the wheel holding module. The submodule is composed of the components: tire carrier, extending rack, spur gear-5 and rack cover. Rack and pinion system is utilized to provide for the linear extension and retraction motion of the wheel holding module. The configuration and components of this submodule are shown in figure-51 below.



**Figure – 51:** Components in the Width Adjustment Submodule. Numbers Indicate: 1-Driver Motor, 2-Extending Rack, 3-Tire Carrier, 4-Spur\_Gear\_5, 5-Motion Limiting Pin

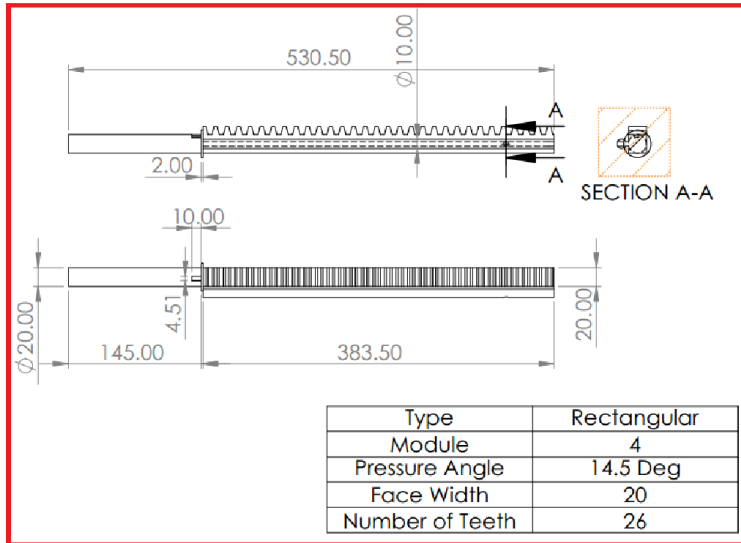
The tire carrier is designed in such a manner that it provides a translational joint for the extending rack while preventing it from rotation about any axis. The extension of the rack is controlled by the rotation of the motor that drives spur gear-5. The length of the extending rack is designed such that the entire length need not be extended in order to achieve width adjustment for the smallest vehicle. The motion of the rack is limited at by the motion limiting pin shown in the figure.

The first component synthesized in this submodule is the tire carrier. The function of this component is supporting the driver motor, supporting the extending rack while allowing it to slide and constraining the linear motion of the extending rack by means of the slot provided at the side in which the motion limiting pin plays. The tire carrier itself is supported at its bottom to the two connecting shafts. The geometric details and dimensions that allow the desired motion is shown in figure-52 below.



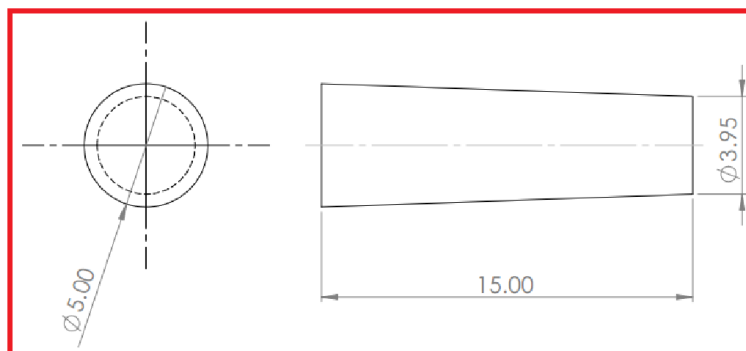
**Figure – 52:** Tire Carrier Geometry and Dimensions (mm)

The second component in the width adjustment submodule is the extending rack the geometric details and dimensions of which are shown in figure-53 below. The function of this component is to transfer the rotational motion from the driving pinion – spur gear-5 to – to the wheel holding assembly. The protruded rode that extends from the rectangular rack supports the housing of the wheel holding assembly.



**Figure – 53: Extending Rack Geometry and Dimensions (mm)**

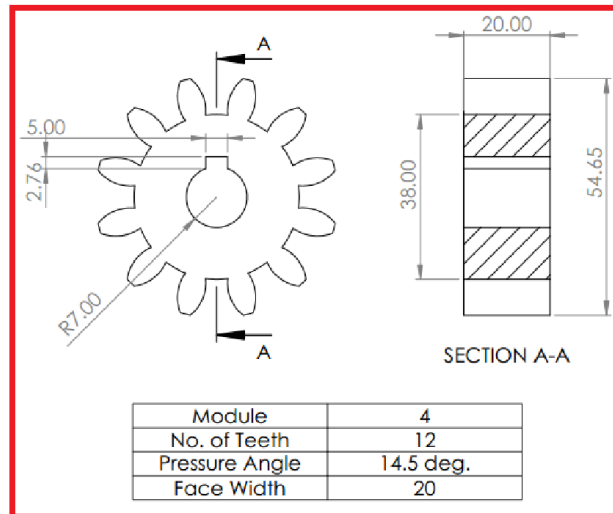
The extending rack moves back and forth in a linear motion. This linear motion is constrained at the two extremes by a pin that is installed on the side of the rack. This pin moves with the rack but since it projects out to the side through a straight slot on the tire carrier, which is stationary relative to the rack and the pin, it provides for motion constraining. This motion limiting pin is made to be a separate component from the extending rack that fits through a hole on the later for assembly purposes. Its dimensions and geometric details are shown in figure-54 below.



**Figure – 54: Motion Limiting Pin Geometry and Dimensions (mm)**

The extending rack is driven by spur gear 5. This gear is in turn driven by the driving motor. The gear is supported on the spindle of the driving motor while the motor itself is supported by the

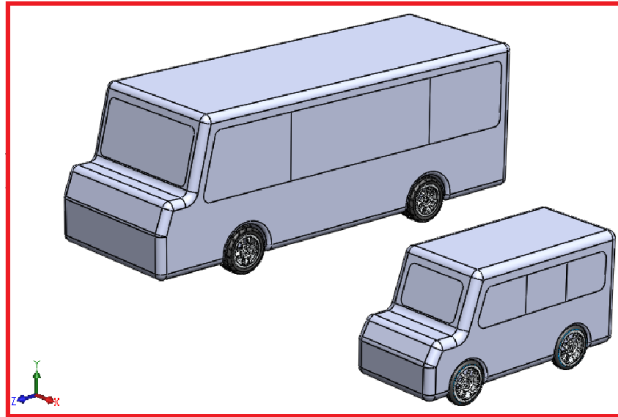
support structure provided on the tire carrier. The geometric details and dimensions of spur gear-5 are shown in figure-55 below.



*Figure – 55: Spur\_Gear\_5 Geometry and Dimensions (mm)*

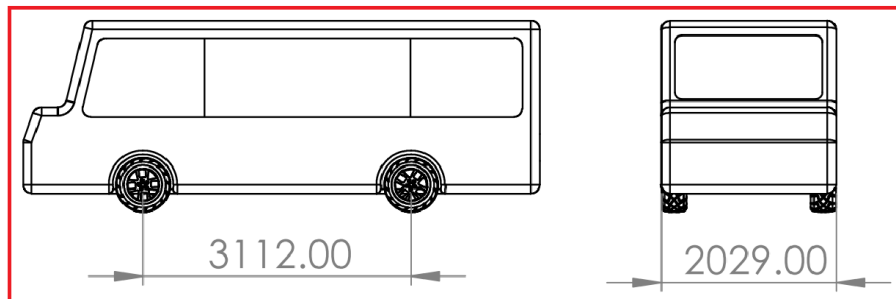
#### **4.1.7. Achieved Design Results of the Width Adjusting Assembly**

The purpose of synthesizing the width adjustment assembly was to enable the simultaneous wheel removal/installation mechanism developed in this work to accommodate a range of passenger car widths from the smallest to the highest vehicle width in the passenger car category as predetermined by literature review. Based on this, the simultaneous wheel removal and installation mechanism works for vehicles having width in the range of 1234.00 mm to 2029.00 mm. This is the width range of passenger cars (see table-1 in chapter 2).

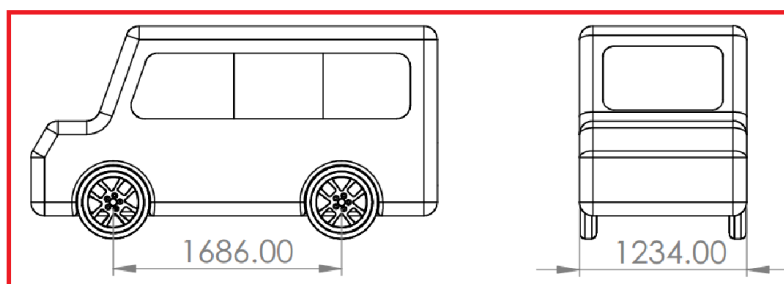


**Figure – 56:** Size Comparison of the Smallest and Largest Vehicles in the Passenger Car Category. Solid Model Prepared in This Work are Based on Dimensions from Literature Review.

Figures 57 and 58 below indicate views with wheelbase and width of the largest and smallest passenger cars respectively.



**Figure – 57:** Wheelbase and Width of the Largest Passenger Car Accommodated by the Simultaneous Wheel Removal/Installation Mechanism Developed in this Work.



**Figure – 58:** Wheelbase and Width of the Smallest Passenger Car Accommodated by the Simultaneous Wheel Removal/Installation Mechanism Developed in this Work.

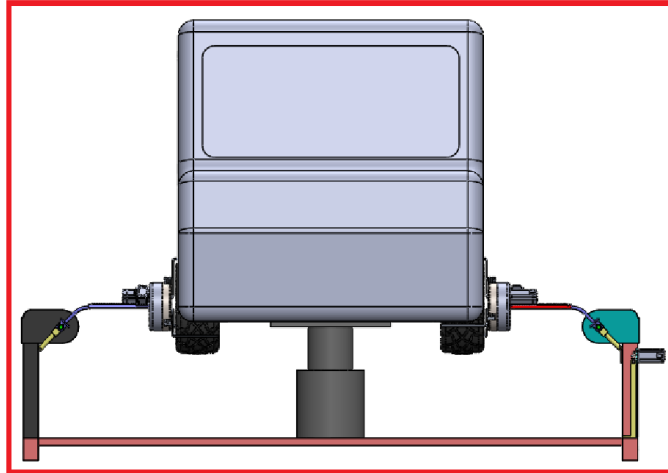
For design purposes, the largest and smallest vehicles were modeled in this work. The solid models are shown in figure-56 above. From table-1 in the literature review, not only the width range of passenger vehicles, but also the wheelbase range has been determined. The wheelbase range for passenger vehicles, according to literature review, is from 1686.00 mm to 3112.00 mm (see table-1 in chapter 2) The wheelbase is a critical vehicle dimension with respect to simultaneous wheel removal/installation and a separate wheelbase adjustment module is developed for it in this work (refer to the next subsection for this module).

The synthesized solution is adjustable for both the largest and smallest vehicle widths. The extension/retraction of the four extending racks on each width adjustment submodule allows for the desired adjustment. Table-13 summarizes the results of the achieved width adjustment submodule. The operating characteristics of this assembly in terms of the largest and smallest vehicle width it can handle is shown in the table.

*Table – 13: Results of Synthesizing the Width Adjustment Assembly: Achieved Maximum and Minimum Vehicle Width Values*

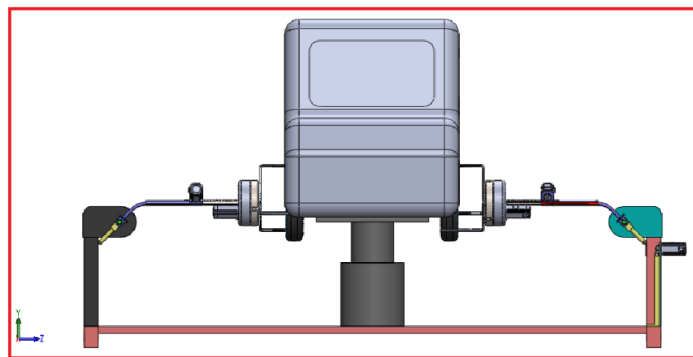
<b>Operation characteristics of the synthesized width adjustment module</b>	Dimensional Parameter (mm)
Largest vehicle width accommodated	2029.00
Smallest vehicle width accommodated	1234.00

Both of the cases in table 13 are shown in figures 59 and 60 below. In the former, assembly model of the largest passenger car width being accommodated by the simultaneous wheel removal/installation mechanism is shown.



**Figure – 59:** Assembly model indicating the width of the largest passenger car (width=2029.00 mm) being accommodated by the width adjustment module.

Figure-60 shows the width adjustment submodule adjusted for the smallest passenger car width. Notice that, since the width of the smallest vehicle is considerably smaller than that of the largest, there is noticeable extension of the both the left and right extruding racks with respect to their respective tire carriers. These shown assembly models are for the two extreme cases of vehicle width. As it logically follows, any passenger car of width in the middle between the maximum and minimum passenger car width values is accordingly accommodated by the width adjustment submodule developed in this original design work.

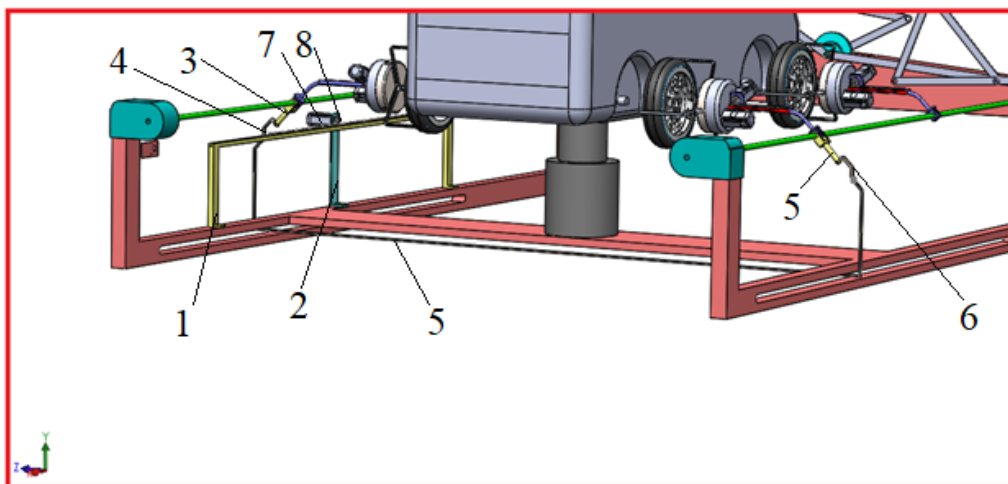


**Figure – 60:** Assembly model indicating the width of the smallest passenger car (width=1234.00 mm) being accommodated by the width adjustment module.

#### 4.1.8. Wheelbase Adjustment Module Synthesis Result

As shown in the previous subsection for the vehicle width adjustment system, the system must also adjust for wheelbases of different size passenger cars. In a similar manner to the width, the maximum and minimum wheelbase of vehicles in the passenger car category has been identified to be 3112.00 mm and 1686.00 mm respectively (*see previous subsection and literature review for details*). In order to achieve adjustment for these extreme dimensions (and values within the range), solution has been synthesized. Based on the design concept selected that uses rack-and-pinion drive presented in the methodology chapter, the dimensions and geometric features of the module has been synthesized. The results of this synthesis work for the wheelbase adjustment module are presented in this subsection.

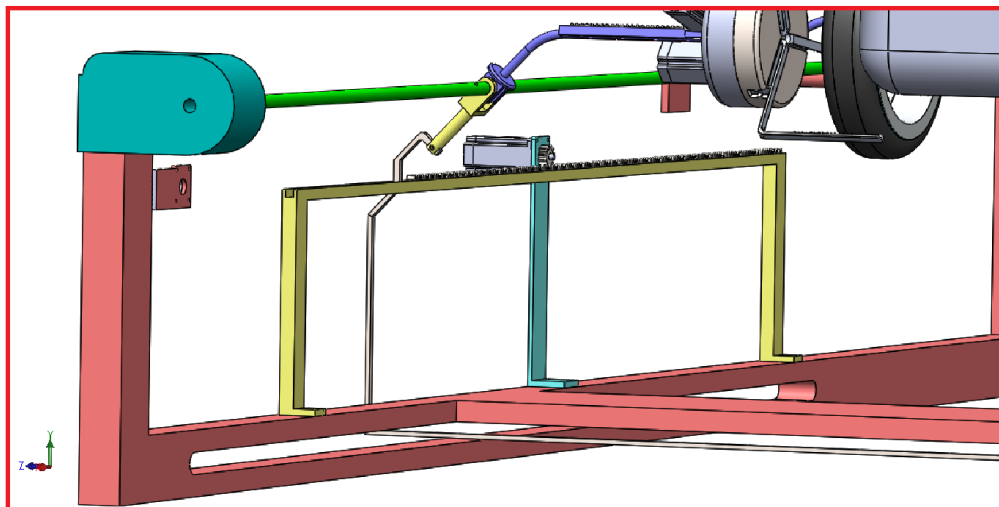
The wheelbase adjustment module has been synthesized in this work to have eight different components. These components are rack guide, motor support, tire carrier locator, adjusting rack with near side connector, middle connector, far side connector, spur gear-6, and driving motor. All of these components and how they are assembled with respect to one another as well as with respect to the larger mechanism assembly are shown in figure-61 below.



**Figure – 61:** *The Synthesized Wheelbase Adjustment Module. Numbers Indicate: 1-Rack Guide, 2-Motor Support, 3-Tire Carrier Locator, 4-Wheelbase Adjusting Rack with Nearside Connector, 5-Middle Connector, 6-Far Side Connector, 7-Spur Gear-6 and 8-Driving Motor.*

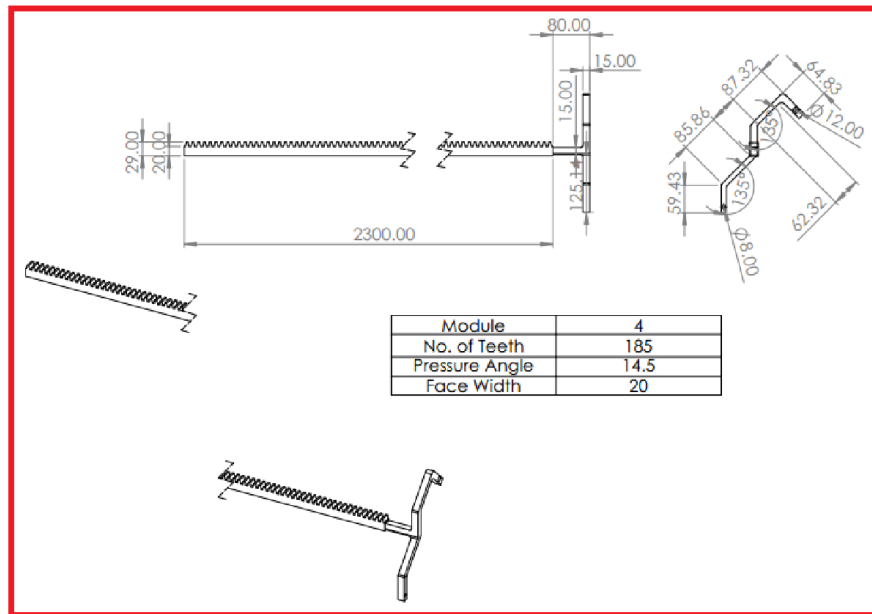
The operation of the wheelbase adjustment module is created to be straightforward. In order to adjust for different wheelbases of different vehicles, only the two rear tire carriers (and respective wheel holding modules installed on them) of the four need to move in a straight line. The front two tire carriers have no motion along an axis parallel to the wheelbase direction (longitudinal direction of the vehicle). While the front two tire carriers are stationary, the rear two will move back and forth thereby increasing and decreasing the wheelbase size the mechanism aligns to.

This motion is attained by means of a bidirectional drive motor that rotates a pinion (spur gear-6 in the module). The pinion's clockwise rotation moves the rack in the direction of wheelbase increment while its counterclockwise rotation results in reduction of wheelbase reduction. The rack is has a bracketing end that serves as a nearside connection (nearside being the side of the vehicle where the driving motor is located) which in turn connects to the tire carrier locator. The tire carrier locator pushes or pulls the base of the tire carrier along the axis of translation. The other part of the near side connector connects to the middle connector which transfers the linear motion to the far side connector, thereby enabling the adjustment of the far side tire carrier synchronously with the nearside one. This design solution enables driving adjusting wheelbase on both sides of the vehicle by using just one driving motor instead of one for each side. Therefore it is cost effective design. A closer look at the assembly described is shown in figure-62 below.



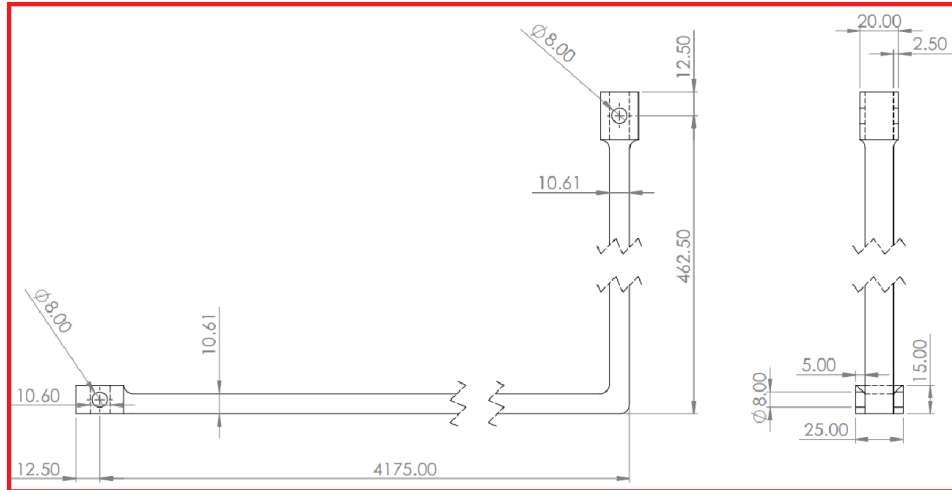
**Figure – 62:** *A Closer View of the Near Side Portion of the Wheelbase Adjustment Module*

The first component in the wheel base adjustment module is the wheelbase adjusting rack with nearside connector. This component is developed to transmit the translational motion generated by the rack-and-pinion gear mate to the tire carrier locator of the nearside rear tire carrier and the connected wheel holding module. The rack is supported by the rack guide. The bracketing end of this component is specifically designed to attach to the tire carrier locator. The geometric features and dimensions of this component, which are results of synthesis are shown in figure-63 below.



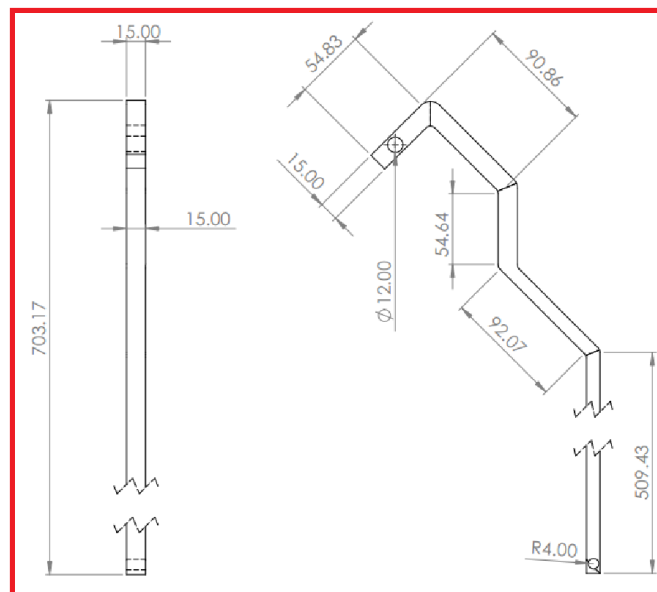
**Figure – 63:** *Wheelbase Adjusting Rack with Nearside Connector: Geometry and Dimensions (mm)*

In the wheelbase adjustment assembly, the component responsible for transmitting the motion from the near side (side of the vehicle where the wheelbase adjustment driving motor is installed) to the far side (opposite side of the vehicle) is the middle connector. This connector is an L-shaped bar with specific features at both of its ends for connecting with the near side and far side connectors. The component transmits the motion over the horizontal distance of 4175 millimeters. As mentioned, this configuration eliminates the need for a second driving motor on the far side of the vehicle. The geometric features and dimensions of this synthesized component are presented on figure-64 below.



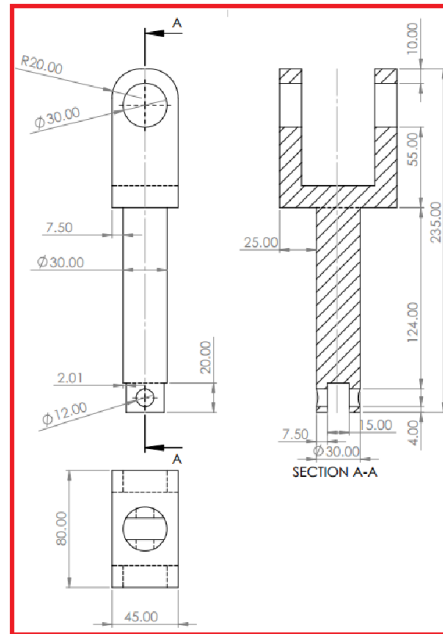
**Figure – 64: Middle Connector Geometry and Dimensions (mm)**

The far side connector is another synthesized component that performs the function of moving the far side tire carrier locator in a motion that results in wheelbase adjustment. It does what the near side connector end of the rack does without the need for an additional rack and driving motor for the other side. This component connects through a hole-and-socket arrangement to the middle connector at its bottom and to the tire carrier locator at its top. The geometric details and dimensions of the far side connector is shown in figure-65 below.



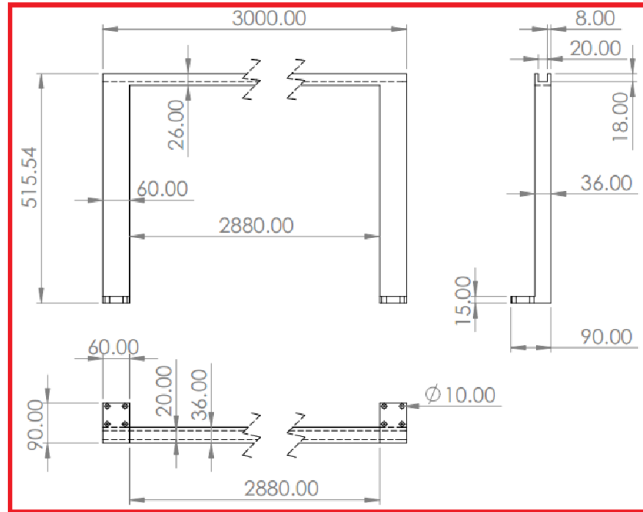
**Figure – 65: Far Side Connector Geometry and Dimensions (mm)**

In addition to the above mentioned components, there are two tire carrier locators in the synthesized wheelbase adjustment assembly. The function of these component is, as its name indicates, locating the tire carrier (and the wheel holding module assembled on that tire carrier). The two tire carrier locators are connected at their bottoms to their respective connectors (far side and near side). The geometric details and dimensions of this component is shown in figure-66 below.



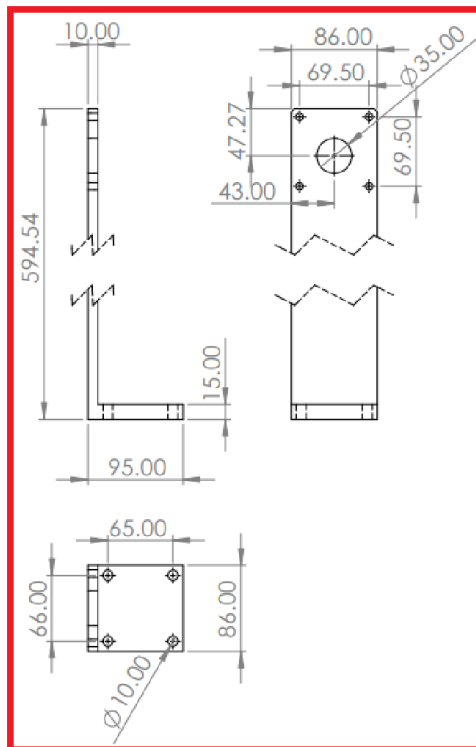
**Figure – 66:** Tire Carrier Locator Geometry and Dimensions (mm)

The synthesized components in the wheelbase adjustment assembly explained so far are moving components. In order for these parts to function as intended, two structural components are also designed. These are the rack guide and the motor support. The rack guide, as its name reveals, provides the function of guiding the rack. It has a channel like feature at its top that forms a translational joint with the rack. Its bottom protrudes to the housing of the overall assembly transmitting all loads to the latter. The rack guide's geometry and dimensional details are shown in figure-67 below.



**Figure – 67: Rack Guide Geometry and Dimensions (mm)**

The final component of the wheelbase adjustment module in this discussion is the motor support. It is a stationary component. Its function is supporting the driving motor of the wheelbase adjustment module. It is supported by the housing structure of the overall assembly. The geometric features and dimensional details of this component are shown in figure-68 below.



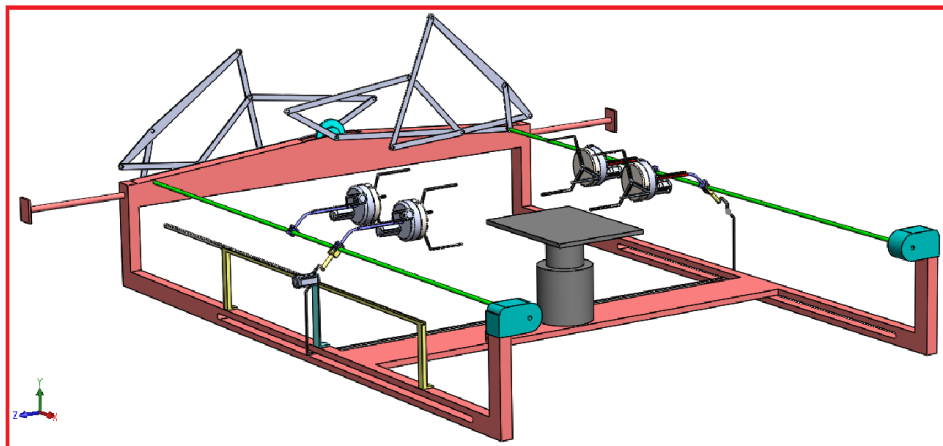
**Figure – 68: Motor Support Geometry and Dimensions (mm)**

#### 4.1.9. Achieved Design Results of the Width Adjusting Assembly

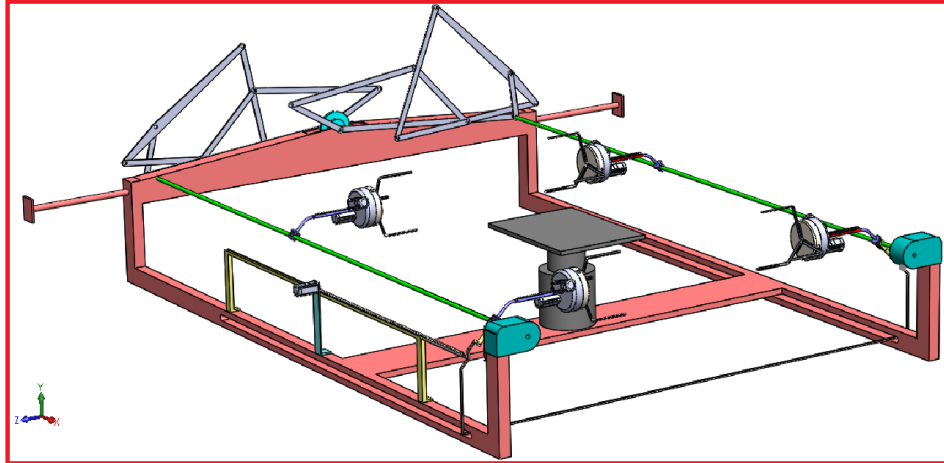
The goal of designing the wheelbase adjustment assembly and its individual components is to create a simultaneous wheel removal and installation mechanism capable of working with passenger cars of different wheelbase dimensions. According to literature, the largest and smallest passenger car wheelbase dimensions were identified to be 3112.00 mm and 1686.00 mm respectively. Adjusting for values within these wheelbase dimensions including these extreme values was achieved by the design solution synthesized. Table-14 summarizes these results.

*Table – 14: Results of Synthesizing the Wheelbase Adjustment Assembly: Achieved Maximum and Minimum Vehicle Width Values*

Operation characteristics of the synthesized wheelbase adjustment module	Dimensional Parameter (mm)
Largest vehicle wheelbase accommodated	3112.00
Smallest vehicle wheelbase accommodated	1686.00



*Figure – 69: Assembly model indicating the wheelbase adjustment for the smallest passenger car (wheelbase=1686.00 mm) being accommodated by the wheelbase adjustment module.*



**Figure – 70:** *Assembly model indicating the wheelbase adjustment for the largest passenger car (wheelbase=3112.00 mm) being accommodated by the wheelbase adjustment module.*

Figures 69 and 70 above show assembly models of the wheelbase adjustment module for the minimum passenger vehicle wheelbase and the maximum passenger vehicle wheelbase respectively. In this figures, notice that the two wheel holding modules near to the viewer are the ones that move along the axis on which their respective tire carriers are mounted. The other pair of wheel holding modules (the pair away from the viewer) are stationary. The relative motion between the two pairs is created by the wheelbase adjustment module.

#### **4.1.10. Summary of Synthesis Results**

One of the objectives of this thesis was to synthesize a mechanism for simultaneous wheel removal/installation of wheels of passenger cars. The iterative engineering design process followed in chapter 3 resulted in synthesizing a design solution consisting of 5 major assemblies and 31 major components. The geometric shapes and dimensions of each of these components, their assembly configuration, and the relative motions between them have been thoroughly worked out. The deeply understood functions of the designed mechanism are achieved by this synthesis work. The results of this synthesis work, the components and assemblies synthesized, are discussed in detail in the preceding subsections. They are presented in a summarized form in table-15 below.

**Table – 15: Summary of Synthesis Results for Synthesized Major Assemblies and Components**

<b>Synthesized Module/ Major Assembly</b>	<b>No.</b>	<b>Synthesized Component</b>	<b>Qty.</b>	<b>Geometry and Dimensional Details of Synthesis (Results of Synthesis)</b>
Central Driving Assembly		Gear Pin	1	<i>Figure – 36</i>
		Planet Coupler	1	<i>Figure – 37</i>
		Hub-Internal Spur Gear	1	<i>Figure – 40</i>
		Spur Gear_1	1	<i>Figure – 38</i>
		Spur Gear_2	1	<i>Figure – 39</i>
Symmetric 16-Bar Mechanism		Link_1_1	2	<i>Figure – A.2</i>
		Link_2_1	2	<i>Figure – A.3</i>
		Link_3_1	2	<i>Figure – A.4</i>
		Link_4_1	2	<i>Figure – A.5</i>
		Link_5_1	2	<i>Figure – A.6</i>
		Link_6_1	2	<i>Figure – A.7</i>
		Link_7_1	2	<i>Figure – A.8</i>
		Link_8_1	2	<i>Figure – A.9</i>
Wheel Holding Module		Wheel Holding Jaw	12	<i>Figure – 44</i>
		Holder Cover	4	<i>Figure – 45</i>
		Holder Housing	4	<i>Figure – 48</i>
		Jaw Locating Spiral	4	<i>Figure – 46</i>
		Spur Gear_3	4	<i>Figure – 47</i>
		Spur Gear_4	4	<i>Figure – 47</i>
Width Adjustment Submodule		Tire Carrier	4	<i>Figure – 52</i>
		Extending Rack	4	<i>Figure – 53</i>
		Rack Cover	4	<i>Figure – 42</i>
		Motion Limiting Pin	4	<i>Figure – 54</i>
		Spur Gear_5	4	<i>Figure – 55</i>
		Rack Guide	1	<i>Figure – 67</i>
		Motor Support	1	<i>Figure – 68</i>

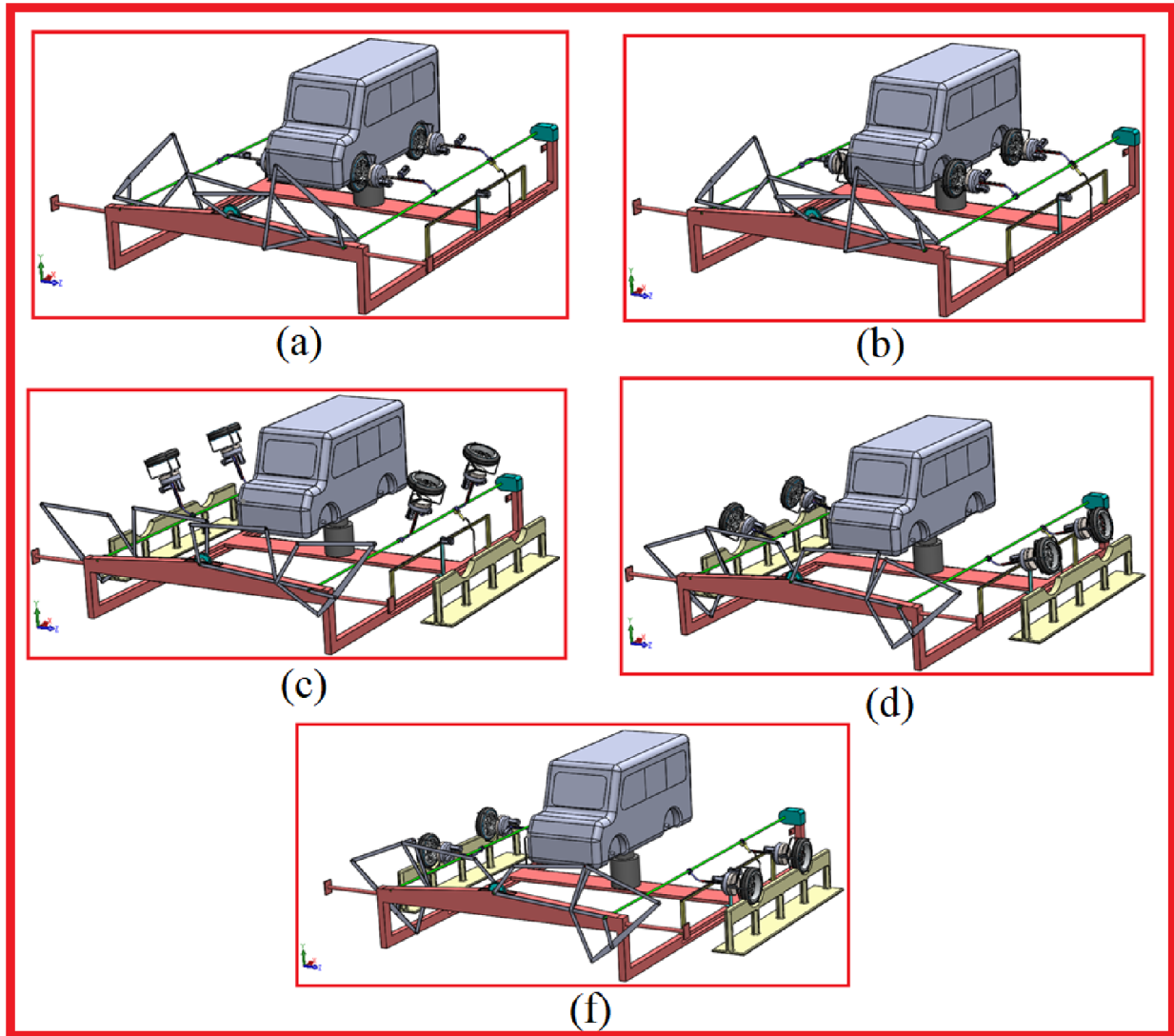
Wheelbase Adjustment Module		Tire Carrier Locator	2	<i>Figure – 66</i>
		Wheelbase Adjusting Rack with Nearside Connector	1	<i>Figure – 63</i>
		Middle Connector	1	<i>Figure – 64</i>
		Far-Side Connector	1	<i>Figure – 65</i>
		Spur Gear-6	1	<i>Figure – 55</i>
Structural Components		Ground Support	1	<i>Figure – A.1</i>
		Connecting Shaft	2	<i>Figure – A.1</i>

In summarizing the results of synthesis, it is important to recognize that the geometric details and dimensions of the synthesized components and their assembly configurations that support the design functions are not the only results of this work by themselves. Also critically important results of this work are the geometric parameters of the assemblies in motion. The achieved parameters are representative of the developed simultaneous wheel removal/installation mechanism's ability to serve for vehicle sizes within the range between the maximum and minimum passenger car as determined by literature review. These results were discussed at depth in the preceding subsections of this chapter. Table-16 summarizes these parameters.

**Table – 16:** Results of Synthesizing the Wheel Holding Module: Achieved Maximum and Minimum Wheel holding Diameters and Wheel Widths

Major Assembly/ Module Synthesized	Function Performed	Range of Dimensional Parameters Achieved by the Design	
		Min. (mm)	Max. (mm)
Wheel Holding Module	Adjusting for wheel diameters ranging from smallest to largest wheel	530.2	810.5
	Holding wheel widths ranging from smallest to largest wheel	125.00	275.00
Width Adjustment Module	Adjusting for vehicle widths ranging between smallest and largest passenger car	1234.00	2029.00
Wheelbase Adjustment Module	Adjusting to accommodate for wheelbase of a passenger car within the range of maximum and minimum wheelbase	1686.00	3113.00

Finally, the summarized synthesis result of this work will be explained by discussing the operation sequence of the designed mechanism. The operational sequence of the wheel removal action of the mechanism is shown in figure- 71 below. Wheel installation is the reverse of removal. As indicated in the figure, after the vehicle is raised from the ground, the wheel holding module is adjusted for the width, wheel base and wheel diameter by the respective subassemblies, the four wheels of the vehicle are held simultaneously by the wheel holding jaws and lug nuts are removed (shown in part(a) in figure). Then, the width adjustment module acts to laterally move the four held wheels at the same time by extracting the extending racks on the respective tire carriers in pairs – the right pair moves to the right and the left pair move to the left (see part(b) in figure). Following this, the central driving system enables the symmetric 16-bar mechanism to simultaneously position the four wheels of the vehicle from the sides of the vehicle to the destination placement areas shown on both sides (see parts (c) though (f) to see this sequence).



*Figure – 71: Sequence of Operations Achieved by the Synthesized Mechanism*

## 4.2. Results of Simulation

The objectives of this thesis were to synthesize a mechanism to be used for the removal and installation of all wheels to and from a vehicle simultaneously and then to perform kinematic and dynamic simulation to determine position, velocity, acceleration and dynamic forces of the individual links. The design methodology rigorously followed to meet these objectives is presented in chapter 3. The synthesis results were presented and thoroughly discussed in section 4.1 of this chapter. This section is dedicated to presenting and discussing the motion analysis results obtained

by simulation. The results are of two types: kinematic results, namely, position, velocity and acceleration, and kinetic results, namely dynamic force exerted on the links' joints as a result of the motion.

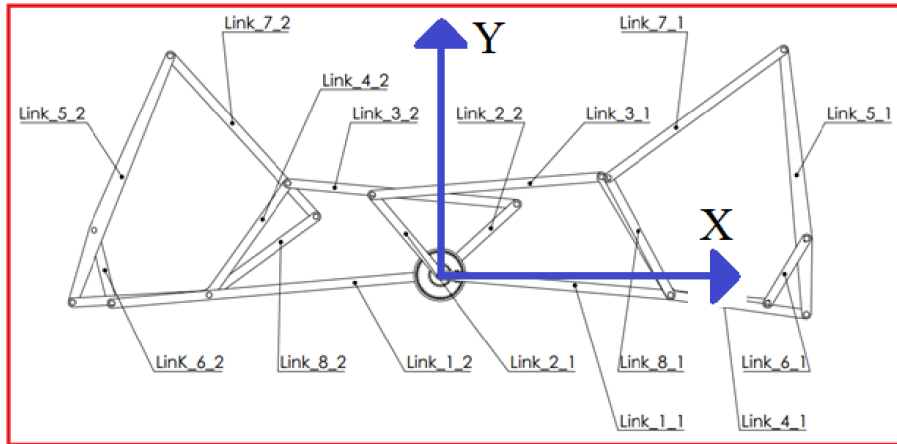
#### 4.2.1. Kinematic Simulation Results: Position Velocity and Acceleration Analysis

There are 16 links in the symmetric 16-bar mechanism. These links undergo different types of motion. Of the three known types of motions in planar mechanisms, namely pure translation, pure rotation and complex motion, two of these types are present in this mechanism – pure rotation and complex motion. A link's motion is described as pure rotation if the link does not undergo any translational motion along any axis, and one rotates about one of its revolute joints. A link's motion is said to be complex motion if the link rotates about an axis and the axis itself is not fixed at a point, i.e. the link also undergoes translational motion while rotating. In the symmetric 16-bar mechanism being dealt with, five links have a motion type of pure rotation, six links have complex motion type and two ground links are stationary. Table-17 shows lists the links according to their motion type.

**Table – 17:** Links in the Mechanism and their Corresponding Motion Types

	Motion Type			
	No Motion (Stationary)	Pure Rotation	Pure Translation	Complex Motion
Link Name	Link_1 (Ground)	Link_2_1 (Right Crank)	-	Link_3_1
	-	Link_2_2 (Left Crank)	-	Link_3_2
	-	Link_4_1	-	Link_5_1
	-	Link_4_2	-	Link_5_2
	-	Link_6_1	-	Link_7_1
	-	Link_6_2	-	Link_7_2
	-	Link_8_1	-	-
	-	Link_8_2	-	-

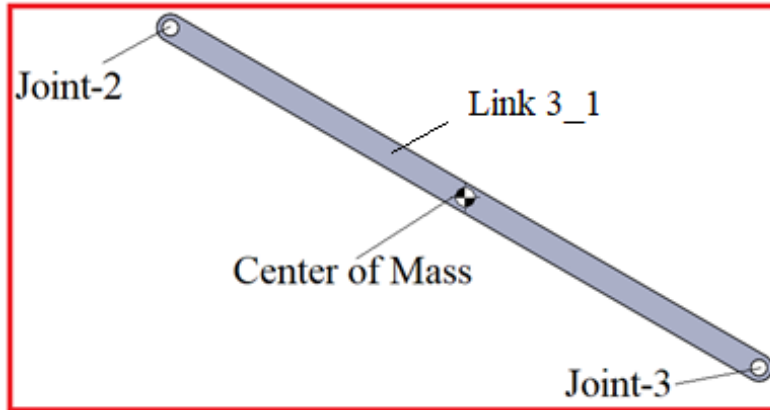
One very important consideration while discussing the parameters which are results of motion analysis is coordinate system. As it is shown in the simulation setup stages in chapter 3, the center of the symmetric 16-bar mechanism is used as an origin of the global rectangular coordinate system as shown in figure-72 below. The motion of all individual links as well as the motion of the mechanism as an assembly is described in reference to this global coordinate system.



**Figure – 72:** Global Coordinate System used in the Motion Analysis. In this Discussion, unless and Otherwise Specified in Particular, all Motion Parameters of Individual Links are Presented with Respect to these Global Coordinates.

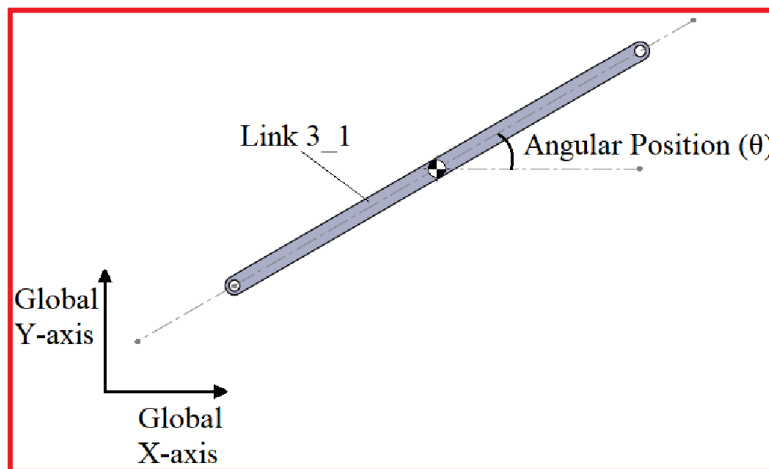
#### 4.2.1.1. Position Results

The first set of results obtained from the kinematic simulation done in this work is the determination of angular position of all of links in the symmetric 16-bar mechanism. As presented in the synthesis part, there are 16 moving links in this mechanism. A plot of the instantaneous angular position each link attains as a function of time is presented for each of the links. For detail discussion purposes, link 3\_1 of the mechanism is used throughout this and the subsequent subsections to illustrate all results of parameters describing the motion of the mechanism. Link 3\_1 is shown in figure-73 below.



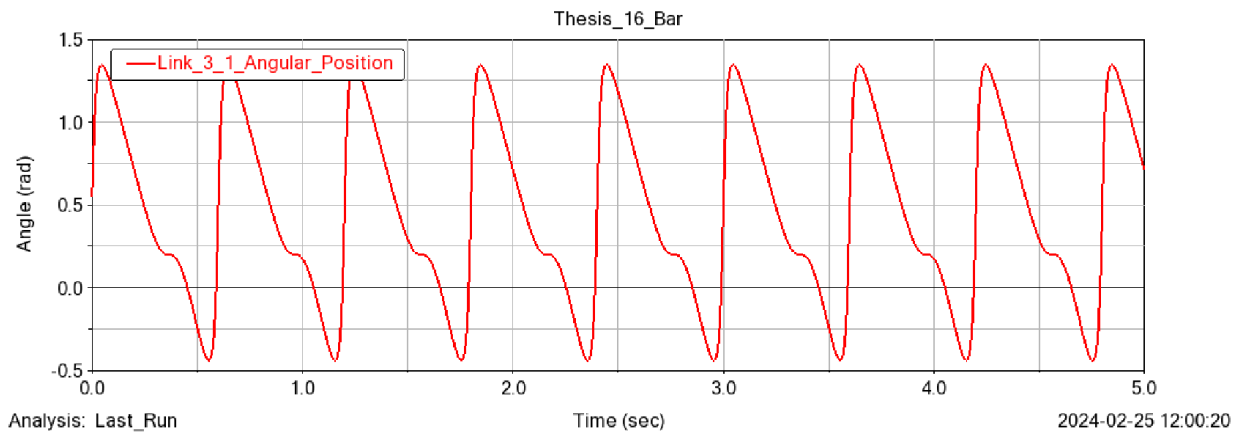
**Figure – 73:** Link 3\_1 of the Mechanism, its two Joints and its Center of Mass. See Appendix-A for dimensions of this link and other links that Assemble with it.

Figure-74 below shows a diagram of link 3\_1 in an instantaneous angular position,  $\theta$ , with respect to the global coordinates. Specifically, angular position is measured from the positive x-axis of the global coordinates to center line of the link drawn along its longitudinal axis as indicated. A counterclockwise angular measurement is taken as positive. Angular parameters are measured in this manner throughout this and the following sections.



**Figure – 74:** Diagram of Link\_3 Showing the Angular position Measured with Respect to the Positive X-Axis of the Global Coordinate System

The angular position of link 3\_1 of the mechanism in radians is plotted versus time in figure-75 below. The time run of simulation is five seconds. The link undergoes complex motion which means it rotates about an axis through its center of mass as the axis moves in the plane. In planar mechanisms mechanical designers are interested in the angular position of links at a given time. The plot presents this value for the entire motion the link makes in its life of operation. The motion the link makes is cyclic; that means, for a continuous rotation of the crank, all other links in the mechanism assume a series of instantaneous positions repeatedly in a cyclic fashion.



**Figure – 75:** Angular Position versus Time of Link\_3\_1 for Five Seconds of Simulation

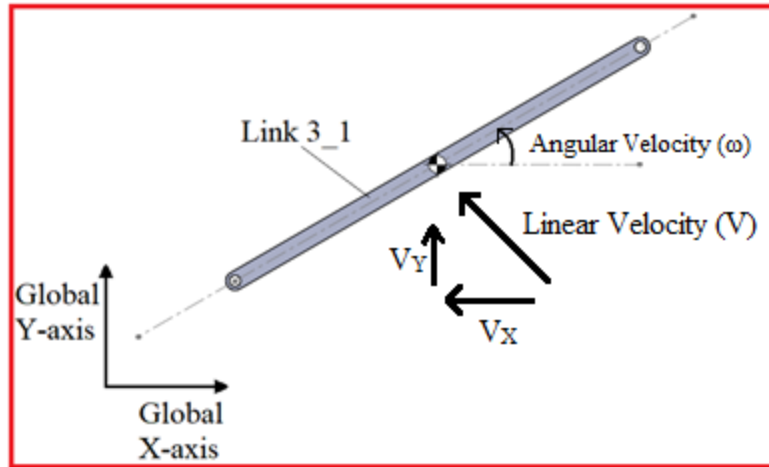
The repeated pattern of the plot in the figure is representative of this phenomena. For engineering design purposes, interest is often on maximum and minimum values. From the plot, the maximum positive value of angular position link 3\_1 repeatedly attains is 1.3409 radians (76.83 degrees) and the negative value is -0.4384 radians (-25.12 degrees). From this, it can be understood that link 3\_1 moves recurrently between about 77 and -25 degrees in repeated cyclic motion. The maximum and minimum angular position values of all moving links in the symmetric 16-bar mechanism is presented in table-18. For the plots of angular position versus time for all links, refer to appendix-B.

**Table – 18:** Numerical Values of Extreme Angular Position of Links in the Mechanism

Link Name	Extreme Angular Position Attained During Motion (Radians)	
	Min.	Max.
Link 2_1	-3.1334	3.1393
Link 3_1	-0.4384	1.3409
Link 4_1	-0.1045	0.0424
Link 5_1	-0.1752	0.2863
Link 6_1	-0.2015	0.0279
Link 7_1	-0.3826	0.6124
Link 8_1	-0.3767	0.2031

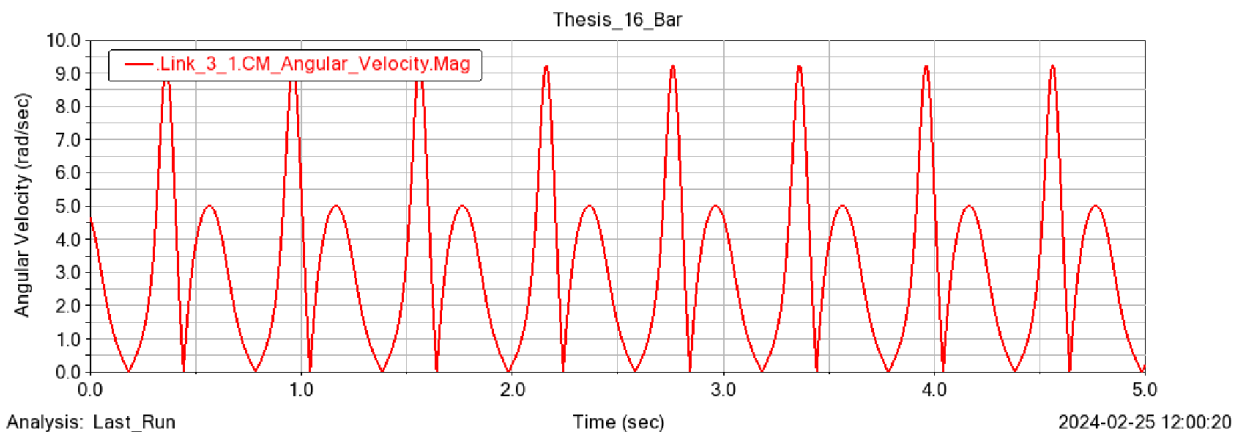
#### 4.2.1.2. Velocity Results

The next set of results obtained from kinematic simulation are angular velocity values. For illustration and discussion purposes, the angular velocity results of link 3\_1. Results of all other links is presented in appendix-B. The angular velocity of link 3\_1 is shown with respect to global coordinates in figure-76. Note that, in a similar manner to angular position, counterclockwise angular velocity is taken to be positive. In addition, the linear velocity of the link with respect to the global coordinates is indicated in the figure along with the horizontal and vertical components of the velocity vector.



**Figure – 76:** Diagram Showing Angular and Linear Velocity of Link 3\_1

A plot of the angular velocity of link 3\_1 in radians per second versus time is presented in figure-77 below. In this plot, the repetitive nature of the motion of the link. The angular velocity, which is one of the parameters that describe the motion of the link is obtained for the entire motion. From this plot, it was found that link 3\_1 attains a maximum angular velocity of 9.2253 radians per second and a minimum value of 0.011 radians per second. This implies that the link repeatedly accelerates and decelerates between from about coming to almost a complete stop to moving at 9.22 radians per second. This abrupt acceleration and deceleration will cause dynamic force fluctuations in the links joints which should be a design consideration. The acceleration and deceleration as well as the forces in the joints are discussed in subsequent subsections.



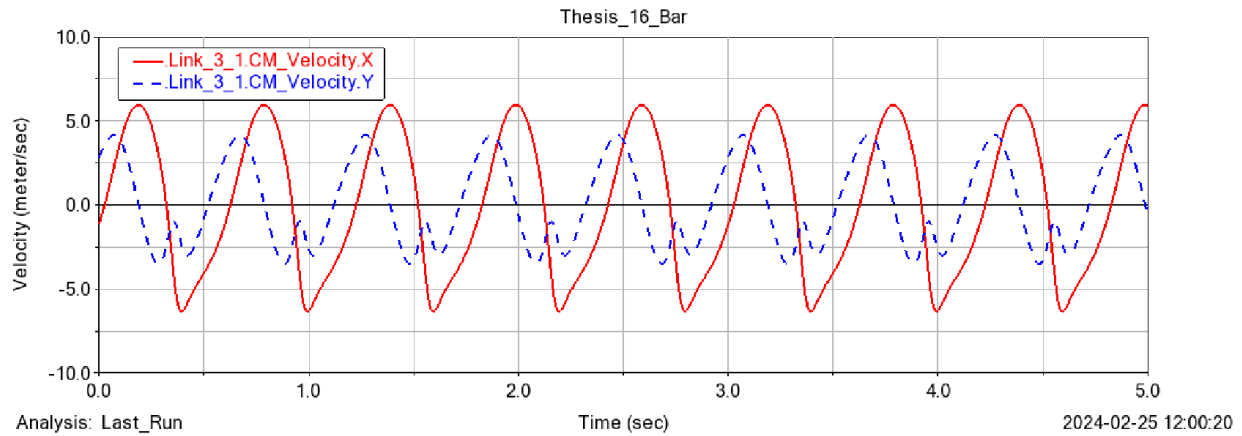
**Figure – 77:** Angular Velocity versus Time of Link\_3\_1 for Five Seconds of Simulation

The maximum and minimum angular velocity values for all links is presented in table-19 below. The corresponding plots of angular velocity versus time for all links, out of which these values are extracted are found, in appendix-B.

**Table – 19: Numerical Values of Extreme Angular Velocity of Links in the Mechanism**

Link Name	Extreme Angular Velocity Attained During Motion (Radians per Second)	
	Min.	Max.
Link 2_1	10.472	10.742
Link 3_1	0.011	9.2253
Link 4_1	0.0041	12.9345
Link 5_1	0.0094	24.0752
Link 6_1	0.1516	28.0037
Link 7_1	0.0139	19.2032
Link 8_1	0.1438	36.5220

In addition to the angular velocities, linear velocity results are also obtained for the links in the mechanism. These results are very important for the reason that while representing the links' rotation effectively, angular velocities are not fully adequate to represent the links' velocities as the translation effect of the motion is not described by them. Since the solutions are being presented in the global rectangular coordinates shown above, the linear velocities in the x- and y- direction were determined by the simulation. Figure-78 presents these velocities for link 3\_1. Note that the red-colored continuous curve indicates y-direction components of the velocity of the link's center of mass while the blue colored dashed curve x components of velocity of link 3\_1's center of mass.



**Figure – 78:** X- and Y- Direction Linear Velocity Components versus Time of Link\_3\_1 for Five Seconds of Simulation

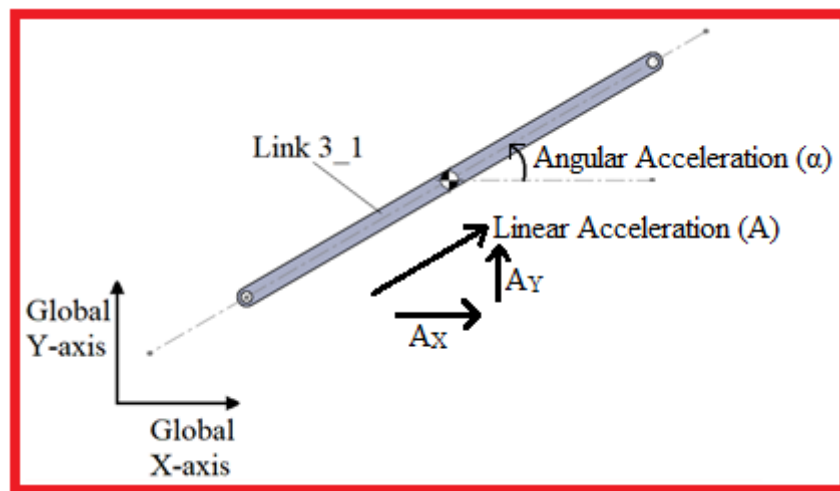
For all links in the mechanism, x- and y- component linear velocity results were obtained from the motion simulation. These are included in appendix-B of this document. The extreme values of these linear velocities, which are of interest for design purposes are listed in table-20 below. The positive and negative signs of these magnitudes of the linear velocities shown in the table are indicative of the directions to which the velocity vectors are facing. The coordinate system in reference to which these directions are denoted is the global rectangular coordinate system shown previously.

**Table – 20:** Numerical Values of Extreme Linear Velocity of Links in the Mechanism

Link Name	Extreme Linear Velocity Attained During Motion (Meters per Second)			
	X- Component		Y-Component	
	Min.	Max.	Min.	Max.
Link 2_1	-2.9859	2.9851	-2.9859	2.9851
Link 3_1	-6.3210	5.9647	-3.4833	4.1559
Link 4_1	-1.1116	0.9647	-1.1952	2.0647
Link 5_1	-15.9625	15.4685	-9.7502	12.9877
Link 6_1	-4.5980	4.5204	-4.1769	5.4368
Link 7_1	-28.9881	27.1686	-16.9778	19.7403
Link 8_1	-12.2414	11.2093	-11.3329	12.4847

### 4.2.1.3. Acceleration Results

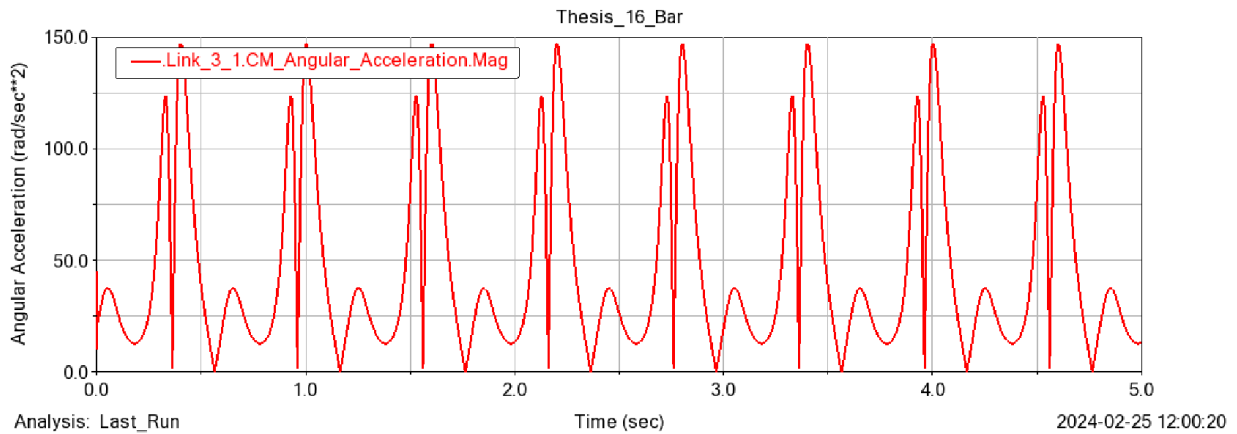
The kinematic simulation results presented next are the acceleration results. First the angular accelerations of the links are presented. Then, x- and y- component linear accelerations of the center of mass of the links are dealt with. In a similar manner to what is done to discuss the position and velocity results in the preceding subsections, acceleration results of link 3\_1 are used for illustration. Plots showing both angular and linear acceleration of all the rest of the links in the mechanism are included in appendix-B. Figure-79 below shows a diagram of link 3\_1 with linear and angular acceleration directions with respect to global coordinates.



**Figure – 79:** Diagram Showing Angular and Linear Acceleration of Link 3\_1

Figure-80 below shows the angular acceleration of link 3\_1 in radians per square seconds versus time for five seconds of simulation. The plot clearly shows that the motion of the link is not only constant-velocity but also it is characterized by changing acceleration. Because the motion of the link is repetitive, the curve in the graph follows a pattern that reflects this nature of motion. Link 3\_1 attains a maximum angular acceleration of 147.1534 radians per square seconds and a minimum angular acceleration of 0.0123 radians per square seconds according to this plot. This means that the links acceleration becomes as high as the indicated maximum value at certain instances during the motion while the motion almost becomes a constant velocity motion at other instances. This high fluctuation in acceleration will result in a corresponding fluctuation in the dynamic force that results from the motion. This dynamic force that acts on the link itself, and on

the neighboring links that are assembled to this link by means of the revolute joints, must be taken into consideration during analyzing the stresses that develop on the respective components and appropriate material selection and cross section decisions should be taken to prevent failure of the links' and connecting accessories during service.



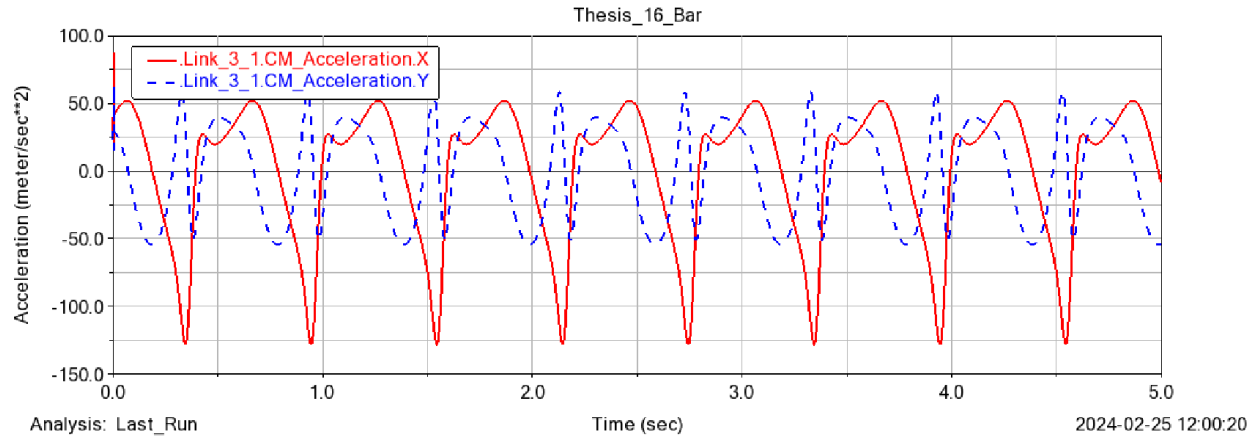
**Figure – 80:** Angular Acceleration versus Time of Link\_3\_1 for Five Seconds of Simulation

Table-21 lists the maximum and minimum values of angular acceleration for all links in the mechanism. The full plots from which these values were collected are included in appendix-B of this document. From this table, it is understood that, while there are instances where the angular acceleration of some of the links becomes near to zero (as in the case of link 3\_1, and link 4\_1), the acceleration of all links never becomes negative. This means, although the motion of the links nearly becomes a constant-velocity motion, it never becomes a decelerating motion. It can also be noticed that the link 2\_1 does not accelerate. This is a reflection of the fact that the link is a crank that is driven at a constant-velocity by the driving motor.

**Table – 21: Numerical Values of Extreme Angular Acceleration of Links in the Mechanism**

Link Name	Extreme Angular Acceleration Attained During Motion (Radians per Square Seconds)	
	Min.	Max.
Link 2_1	0	0
Link 3_1	0.0123	147.1534
Link 4_1	0.9972	276.2509
Link 5_1	1.2603	426.0892
Link 6_1	1.2998	589.0073
Link 7_1	1.2627	319.7826
Link 8_1	0.2695	841.6516

Next are presented the x- and y- components of linear acceleration of the center of mass of link 3\_1. As shown in figure-81, plots of two values are shown in two different colors. The continuous red curve indicated the x- component of linear acceleration of the center of mass of link 3 in meters per second versus time. The blue dashed curve represents the y- component of the linear acceleration center of mass of the link. Both components of the linear acceleration vary continuously and repeatedly throughout the motion of the link. It can be inferred from these plots that the link is in complex motion – the link translates as well as the path along which its center of mass travels changes its direction continuously. This means this resultant acceleration vector not only changes its magnitude as time goes on but also acts in a changing direction. This variation also results in correspondingly varying dynamic forces that result from the motion as forces are a function of acceleration. The varying nature of these forces has very important design implications, the cross sections of links must be designed and materials should be selected such that the links withstand the dynamic forces during their service.



**Figure – 81:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_3\_1 for Five Seconds of Simulation

Extreme values of the x- and y- components of linear acceleration of all links in the mechanism are listed in table-22 below. For the full plots that show all acceleration values versus time, consult appendix-B. The signs of the values in this table indicate directions along which the acceleration vectors act.

**Table – 22:** Numerical Values of Extreme Linear Acceleration of Links in the Mechanism

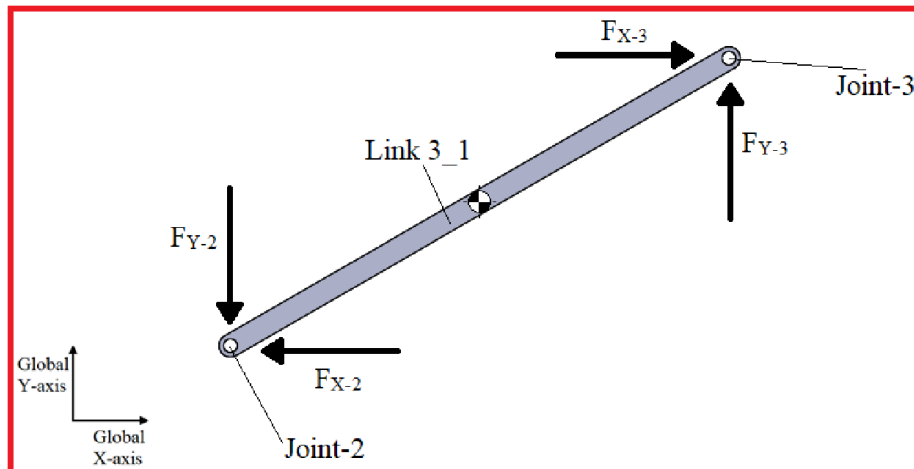
Link Name	Extreme Linear Acceleration Attained During Motion (Meters per Square Seconds)			
	X- Component		Y-Component	
	Min.	Max.	Min.	Max.
Link 2_1	-31.2632	31.2602	-31.2442	31.2632
Link 3_1	-127.7338	51.8624	-54.4020	58.3606
Link 4_1	-27.8826	10.6765	-25.0178	45.7464
Link 5_1	-375.6047	299.2476	-459.9341	256.6204
Link 6_1	-153.9005	120.1330	-147.2441	105.6005
Link 7_1	-749.5500	777.5504	-821.4996	336.0147
Link 8_1	-335.5366	466.8215	-462.2728	248.5364

#### 4.2.2. Dynamic Simulation Results: Dynamic Force Analysis

In this subsection, dynamic simulation results are presented. Kinematic results, namely, position, velocity and acceleration, that describe the motion of each link in the mechanism have been presented and discussed in previous subsections. Here, results dynamic forces (kinetic results) are discussed. These dynamic forces occur because the links are accelerating masses, and the two parameters are proportional to each other according to Newton's second law of motion.

From engineering design perspective, obtaining the value of these dynamic forces acting on the links that result from their motion is very important because these values must be taken into consideration to make design decisions on the cross sectional dimensions and materials from which this links should be manufactured. Failure to take force values generated in this work into consideration in the stress and deflection analysis stages of the design process may lead to disastrous product failure during operation of this mechanism. Therefore the data generated from the simulation is critically important for effectively making use of the synthesized solution.

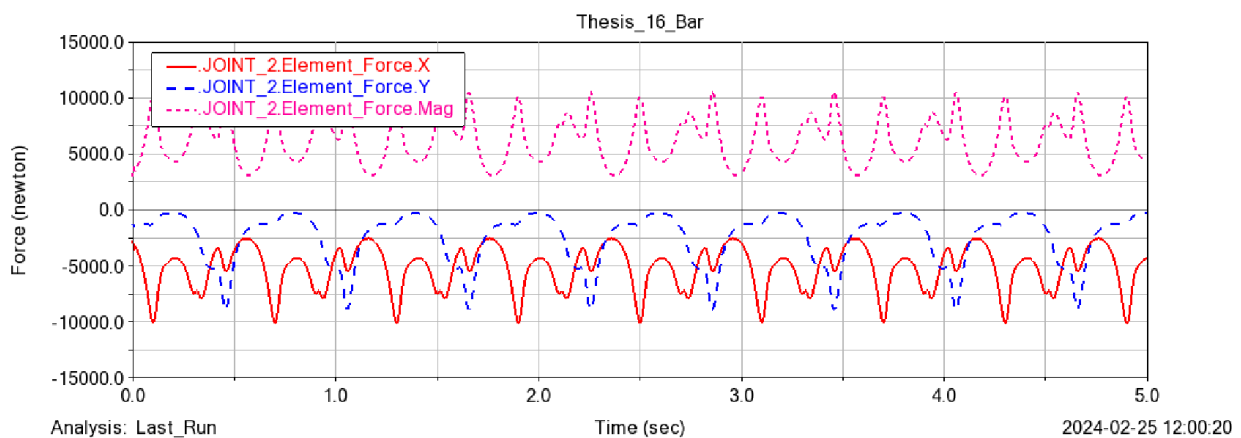
In discussing the dynamic force results, the same global rectangular coordinates mentioned while presenting the kinematic results are used. Similarly, link 3\_1 is used to illustrate the force results and their meanings. For full dynamic force versus time plots of all other links in the mechanism, the reader can refer to appendix-B of this document. Figure-82 below shows a forces acting on the two joints of link 3\_1.



**Figure – 82:** Forces Acting on the Joints of Link 3\_1: X- and Y- components of Forces are shown

Figure 83 and 84 below shows three plots of dynamic force results obtained from the simulation. All curves represent values of force in Newton versus time in seconds. Figure-83 is for joint-2 of link\_3 and figure-84 is for joint-3 of the link. Both joints are revolute joints that only allow relative rotation of the links they connect. In both figures, the continuous red curve shows the x-component of the joint force and the blue dashed curve indicates the y component of the joint force. Magnitude of the resultant force is represented by the purple dashed curve shown in the plots.

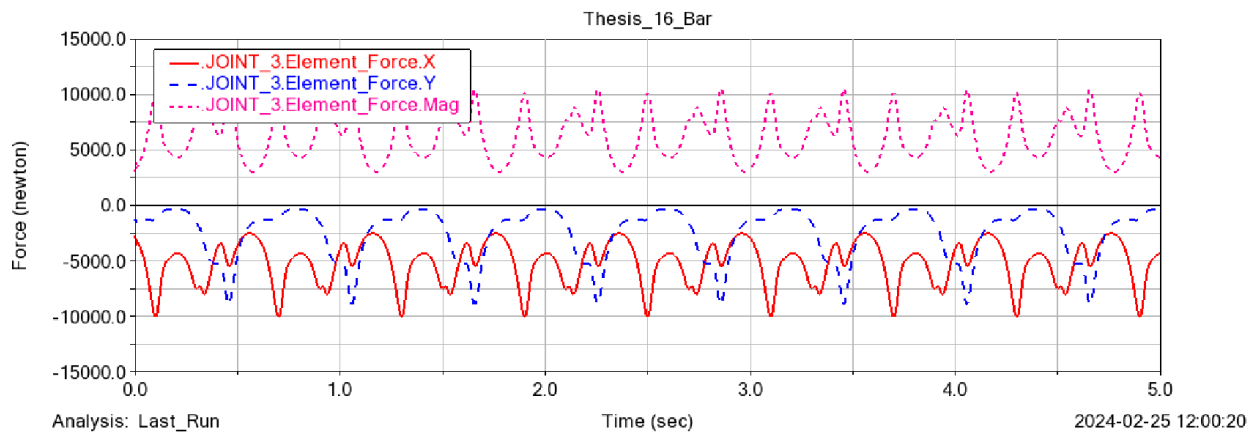
Owing to the repetitive nature of motion of the links in the mechanism for continuous rotation of the crank, the dynamic force values follow a repetitive pattern. For engineering design purposes, designers are interested in the maximum values of this forces. The reason behind is, if the links are designed to withstand the maximum force values, they will not fail during service when they are subject to other load values less than the maximum ones. In the case of link 3\_1 under consideration here, in joint-2 that connects the link with link\_2, the x-component of the dynamic joint force ranges from between -2543.5155 Newton and -10088.5354 Newton. The y-component of the force on this joint is in the range -318.7905 Newton and -8951.175 Newton. The magnitude of the force on this joint is in the range -318.7905 Newton and -8951.175 Newton. The magnitude of the resultant force acting on joint-2 ranges from 3057.1107 Newton to 10,494.1104 Newton. Since force is a vector quantity, a negative sign indicates that the force acts opposite to the positive direction of its respective axis shown in the global coordinate system. These figures are extracted from the force versus time plots of figure-83.



**Figure – 83:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_2 for Five Seconds of Simulation

Similarly, for joint-3 that connects link 3\_1 to link 4\_1, the x-component of the joint force ranges between -2526.0271 Newton and -10049.2181 Newton while the y-component of this force is in the range from -341.5627 Newton to -8931.6540 Newton. Magnitude of the resultant force that acts on joint-3 varies from 3031.4851 Newton to 10467.4743 Newton. The joint forces of all other joints in the mechanism are presented in plots shown in appendix-B.

The importance of having both x- and y- component of joint force values instead of just the magnitude of resultant force, which is obtained by vectorially adding the two components, is to have information of the direction along which this resultant force vector acts. In this manner, the reader interested in using joint force data generated in this work can have full information about the joint forces as vector quantities.



**Figure – 84:** *X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_3 for Five Seconds of Simulation*

Table-23 shows the maximum and minimum values of joint forces on all joints of the mechanism. For the full plots out of which these values were extracted, the reader can refer to appendix – B. These extreme values are summarized and presented in tabulated form because they are of the most interest for engineering design purposes.

**Table – 23: Numerical Values of Extreme Magnitudes of Resultant Forces Acting on the Joints in the Mechanism**

<b>Link Name</b>	<b>Resultant Joint Forces (Newton)</b>	
	Min.	Max.
Joint-1	3066.6722	10499.9400
Joint-2	3057.1107	10494.1104
Joint-3	3031.4851	10467.4743
Joint-4	36.0715	4992.5951
Joint-5	3539.4569	19138.0052
Joint-6	15.7104	5017.4190
Joint-7	18.6771	4975.9106
Joint-8	4233.6780	9982.5236
Joint-9	4234.7891	9962.1334
Joint-10	4245.1519	10409.8755

From the above table, forces that act on every link in the symmetric 16-bar mechanism are determined. These forces are dynamic forces that result from the motion of the mechanism. Since every link in the mechanism is a moving mass the weight of which is concentrated at its center of mass, the acceleration of each individual link induces a dynamic force. These dynamic force of individual links has components that act upon neighboring links through the joint that connects the links. This shows the nature of complexity of the load in the mechanism. In addition, since the acceleration in the links is not constant, the resulting dynamic force is also not constant and varies with time. Therefore, as shown in the force – versus – time plots, the dynamic forces vary between a maximum and minimum value. Thus, it is helpful to sort out the extreme ranges of these values to be used for further engineering design purposes.

## Chapter Five

### Conclusion, Recommendations and Future Work

#### 5.1. Conclusion

In tire repair shops – workshops that specialize in the sale, installation, maintenance and repair of tires and related parts for various types of vehicles, key tasks are manually done by technicians [51], [52] and [53]. All over the world, in places where passenger vehicles are found in large numbers, there is a high and frequent demand in tire repair shops for removal and installation of all of a vehicle's wheels at a time, often in such activities as tire replacement, tire rotation, wheel balancing, wheel alignment and all-suspension inspection [54] and [55]. Despite advancement in technology in the area to develop equipment and tools that make the tasks tire repair technicians perform such as vehicle lifting jacks, tire changers (equipment to separate/install tires on rims of individual wheels), impact wrenches etc. equipment to remove/install all wheels of the vehicle have not been yet developed. As a result, manual work of technicians in tire repair shops is widely existent.

There are a number of problems tire repair technicians encounter related to the physically – demanding nature of their work. Especially, when taking into consideration the volume of vehicles tire repair shops usually serve, physical fatigue of technicians working in such shops is more prevalent [57]. In their job, technicians in tire repair shops are required to do laborious tasks as lifting and maneuvering of wheels, repetitive motions such as kneeling, squatting and reaching. Such tasks often cause musculoskeletal injuries on technicians such as strains, sprains, and overuse injuries [58]. The risk of these injuries increases during periods of high demand in tire repair shops [57], [58]. Moreover, technicians in tire repair shops often work in dirty and noisy environments that cause stress, discomfort and health problems[50], [58], [59]. They also face the risk of external injuries bruises and cuts when physically dealing with the objects they work with.

The solution provided for this problem is the original design of a mechanism to deal with the simultaneous removal and installation of all wheels of passenger car which is ideal for tire repair

shops that serve such vehicles. The design solution synthesized enables the simultaneous removal/installations of wheels without the need for a technician to be manually involved in the process. The design allows for all laborious and injury-causing tasks such as lifting and maneuvering of wheels and repetitive motions, are done by the mechanism. This solution eliminates the risk of physical wellbeing of technicians altogether so that technicians will no longer be injured doing the task of wheel removal and installation. Moreover, technician need not be working in dirty and noisy environments – instead, they can work from control rooms and operate the mechanism from a distance.

The original synthesis work done in this thesis to provide an engineering solution to the problem resulted in a design solution consisting of 5 major assemblies and 31 major components within the assemblies. The geometric features and dimensional details of components that produce the all desired motions have been thoroughly worked out. The interrelationship between components have been developed in detail. The functional requirements of all tasks in the simultaneous removal and installation have been fully met by the design. The mechanism's wheel holding module has the capacity to handle wheel diameter of a maximum of 810.5 mm and a minimum of 530.2 mm, and, a wheel width of a maximum of 275 mm and a minimum of 125 mm. The mechanism can adjust for vehicle width of range between 1234 mm and 2029 mm. In addition, the mechanism has been designed to adjust for vehicle wheelbase of range 1686 mm and 3112 mm. This design achievement enables the mechanism to effectively handle the task of simultaneous wheel removal and installation for all vehicles in the category of passenger cars.

Within the design solution, the symmetric 16-bar mechanism. Since the motion of the links in this mechanism is very complex, special effort had to be placed to determine parameters that characterize this motion. As a graduate-level work in mechanical engineering, this thesis placed specialized attention to thoroughly analyze the kinematic and dynamic characteristics of this complex mechanism through multibody simulation. Position, velocity, acceleration and dynamic force data have been rigorously generated for each individual link and joint. The data is too many to mention here due to the number of links and joint. Each parameter is summarized and discussed in chapter-4 section 4.2. The dynamic force results form the basis for future design analysis of stresses and deflections of links and to size their cross sections and select materials in order to prevent failure of components in service.

## **5.2. Recommendations**

The work presented in this thesis is mainly focused on synthesis and multibody dynamics of the simultaneous vehicle wheel removal/installation system. For the purposes of embodiment of the concept, the dimensional constraints of passenger car sizes were selected. However, the working principles of the generated design are not limited to passenger cars only – they can be applied to any wheeled vehicles. In fact, the application of scale-up versions of the mechanism are much very attractive applications for wheeled vehicle that are heavier since the manual removal and installation of their wheels is more challenging than that of passenger cars. Therefore, such applications are highly recommended.

## **5.3. Future Work**

The novel simultaneous tire removal and installation machine is to evolve into a mechatronics system. Any mechatronics system has mechanical, electrical and software components. In this work, a great deal of effort has been exerted to synthesize the mechanical subsystem and analyze its dynamic behavior in great detail. The electrical circuitry and microprocessor programming are future works that can be conducted based on the foundational work laid out in this thesis.

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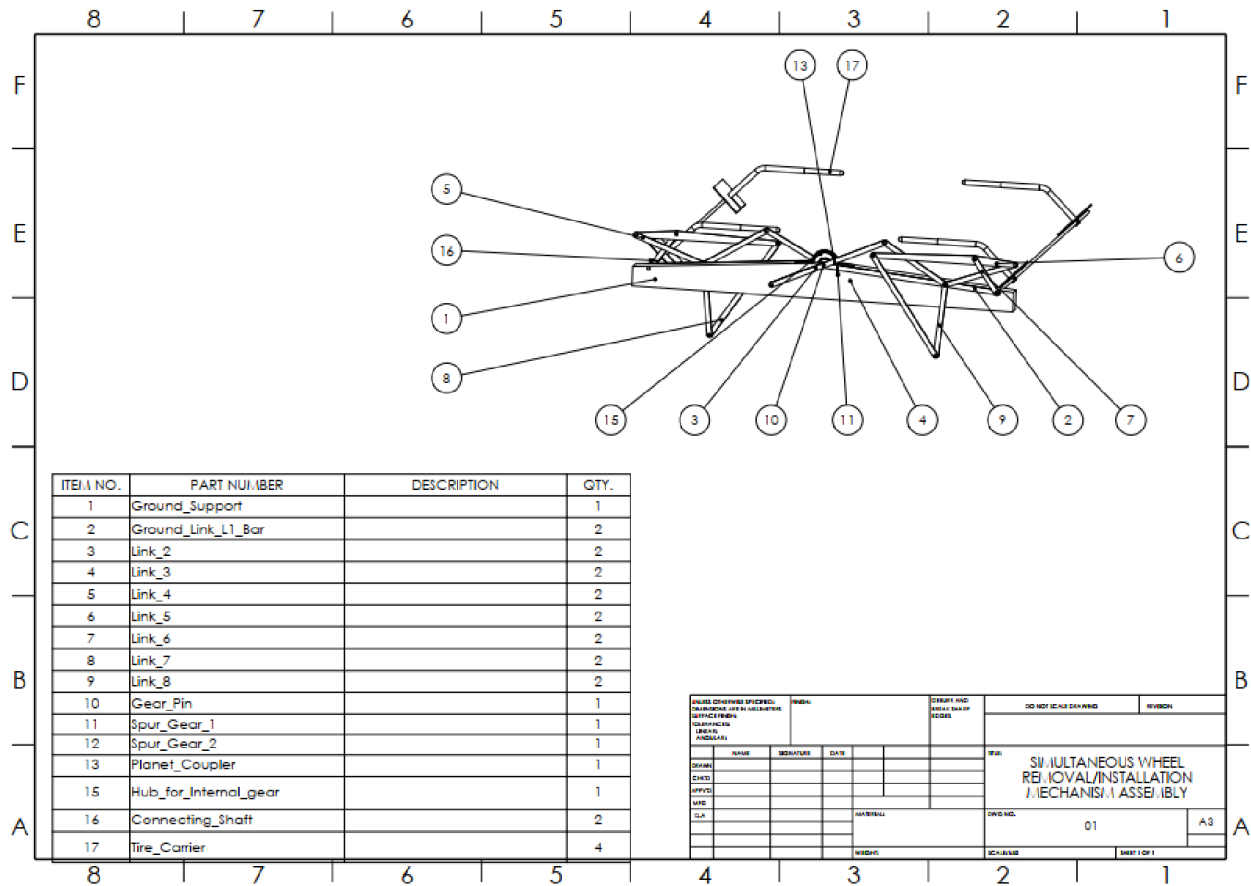
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## Appendix-A

### A.1. Mechanism Assembly, Links and Related Components



*Figure – A.1: Assembly of the Simultaneous Wheel Removal/Installation Mechanism*

**Note:** The drawings of the mechanism’s assembly and the individual links presented from Appendix A.1 to A.9 are intended for communicating the geometry and dimensions of the synthesized mechanism. They are not final design drawings to be used for manufacturing. Final design drawings should be prepared after stress analysis and material selection of each component. This process is not the objective of this study. However, the geometries and dimensions of the mechanism shown here are results of kinematic synthesis, which in turn serve as inputs for further analysis.

## A.2. Geometry of Link\_1\_1

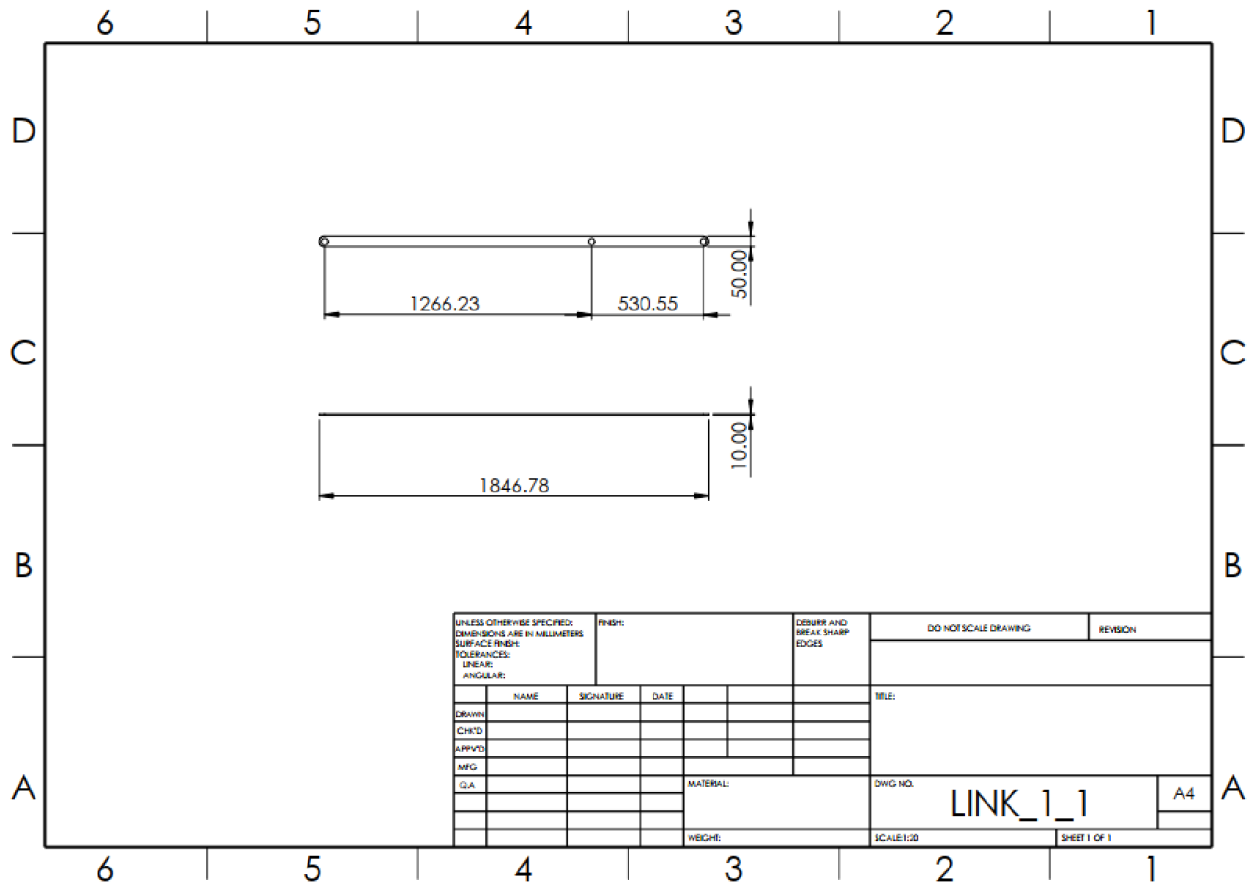
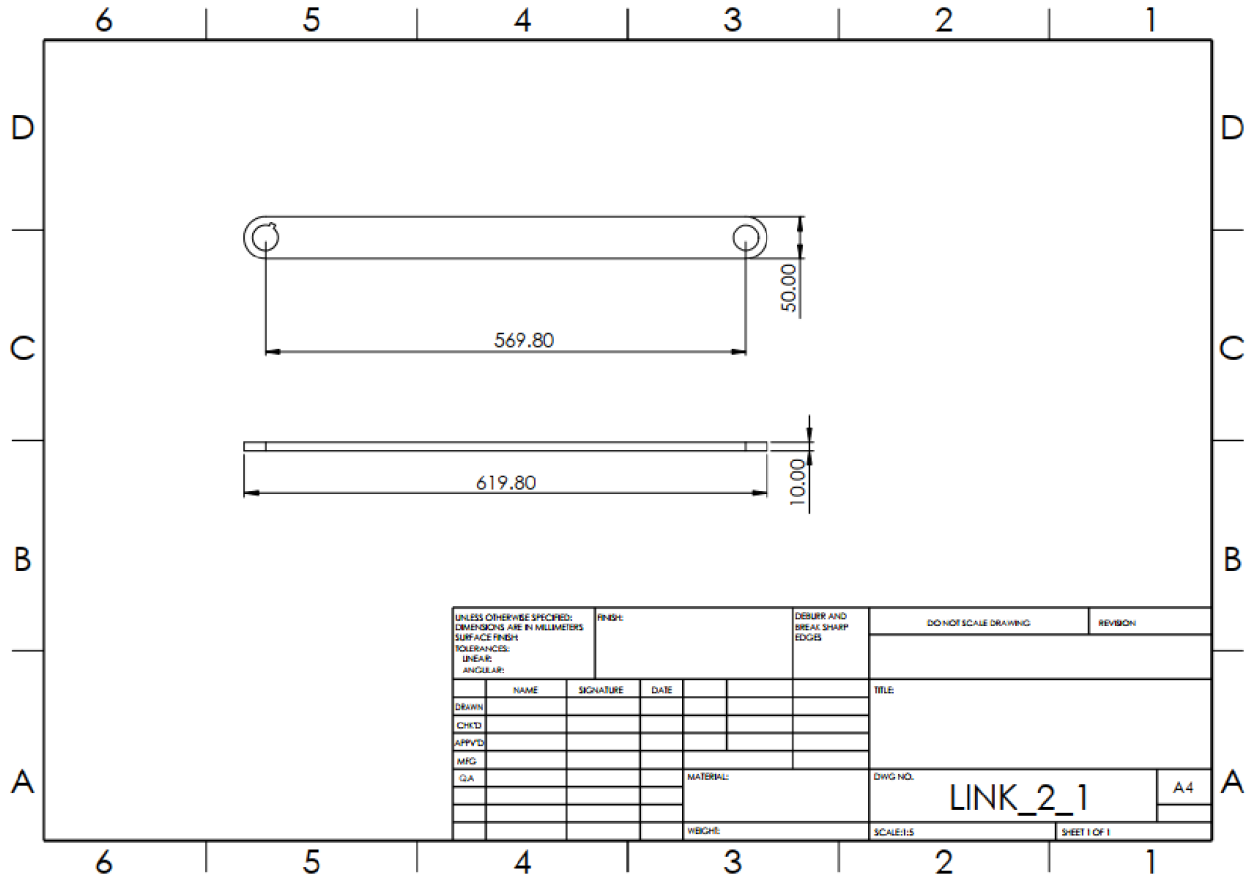


Figure – A.2: Geometry and Dimensions of Link\_1\_1 and Link\_1\_2

### A.3. Geometry of Link\_2\_1



**Figure – A.3: Geometry and Dimensions of Link\_2\_1 and Link\_2\_2**

### A.4. Geometry of Link\_3\_1

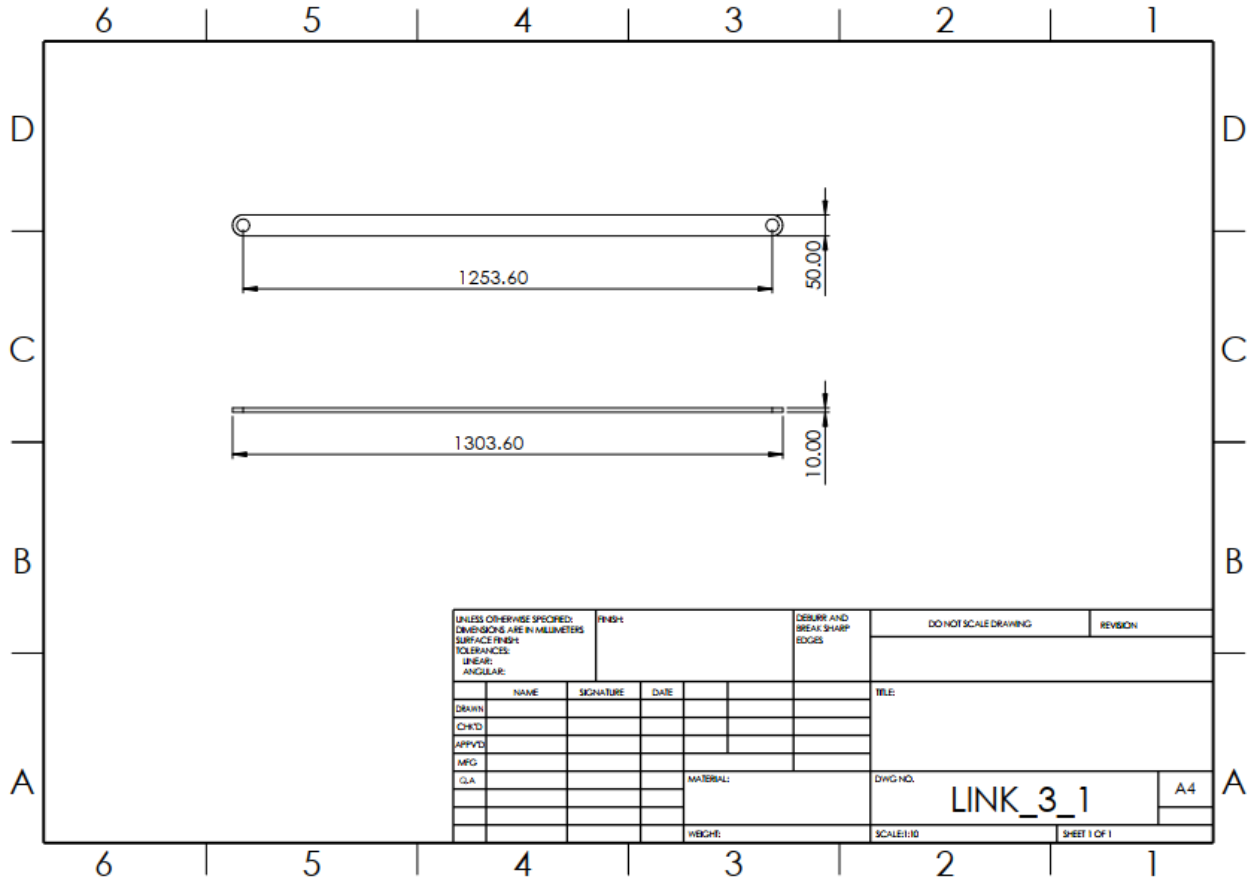


Figure – A.4: Geometry and Dimensions of Link\_3\_1 and Link\_3\_2

### A.5. Geometry of Link\_4\_1

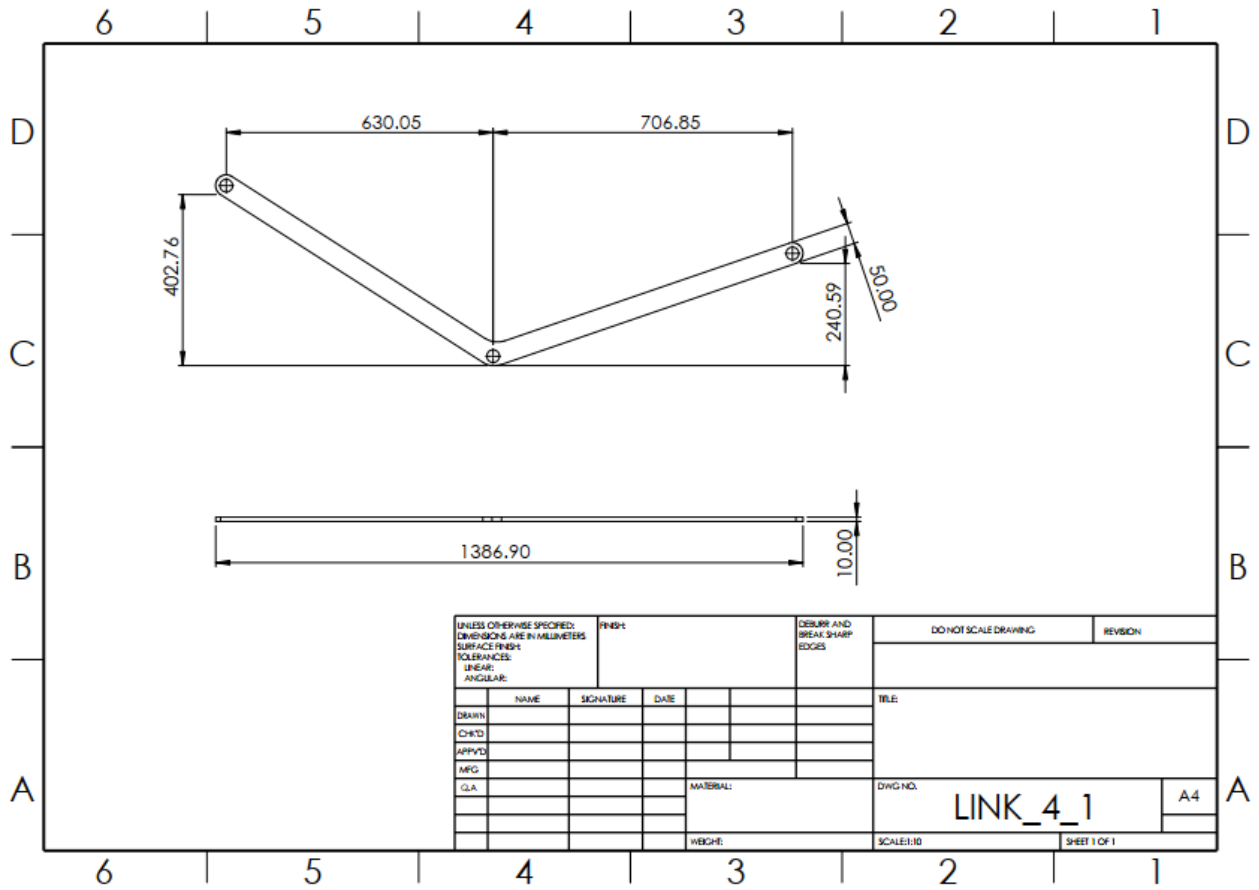


Figure – A.5: Geometry and Dimensions of Link\_4\_1 and Link\_4\_2

## A.6. Geometry of Link\_5\_1

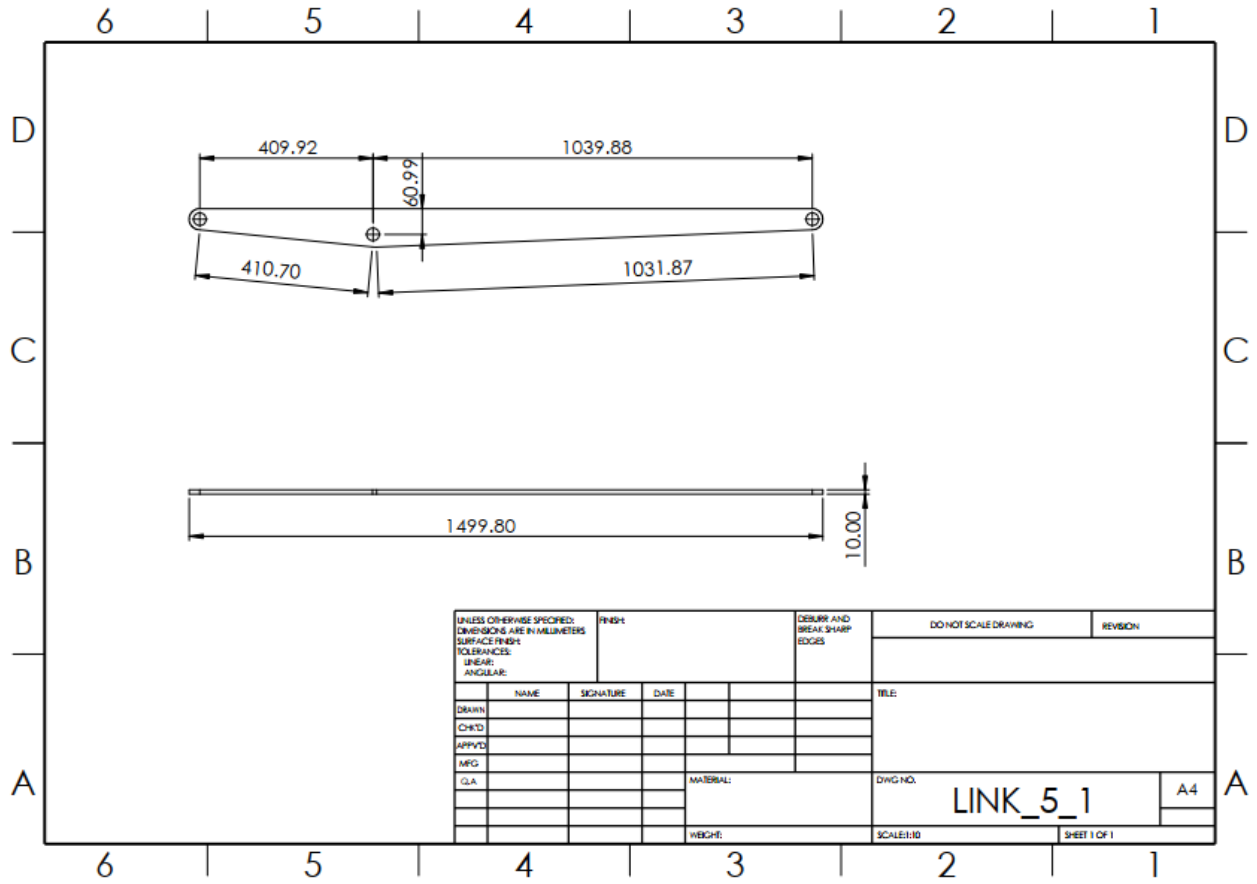


Figure – A.6: Geometry and Dimensions of Link\_5\_1 and Link\_5\_2

### A.7. Geometry of Link\_6\_1

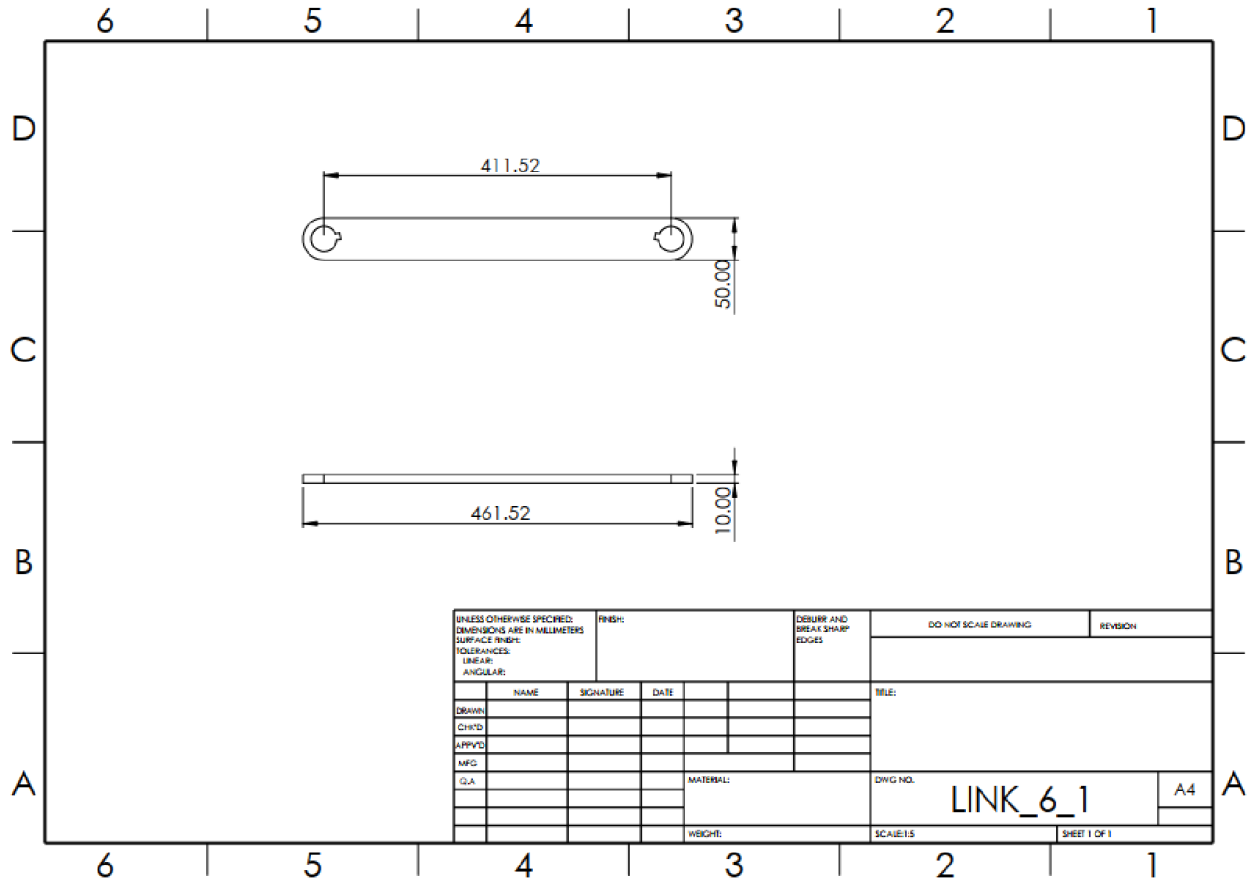


Figure – A.7: Geometry and Dimensions of Link\_6\_1 and Link\_6\_2

### A.8. Geometry of Link\_7\_1

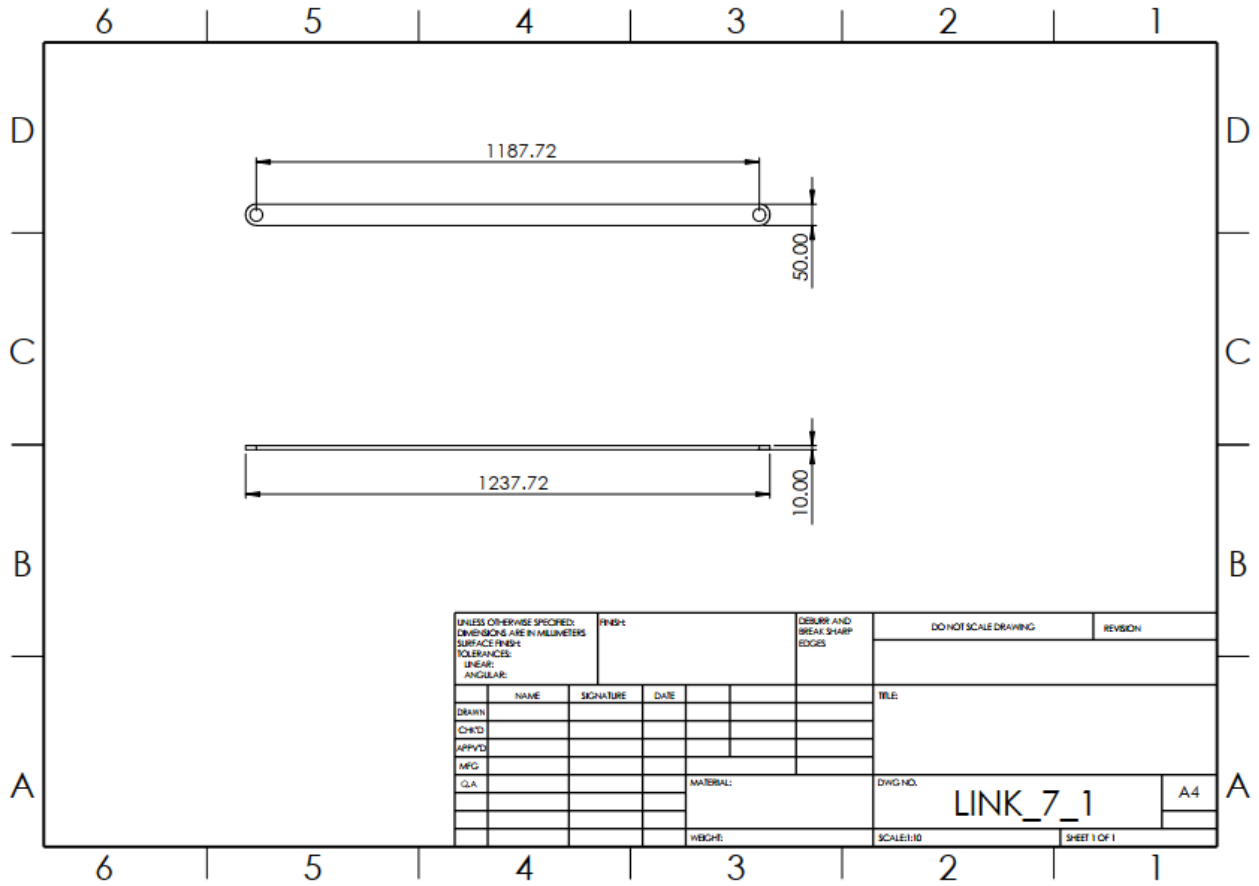


Figure – A.8: Geometry and Dimensions of Link\_7\_1 and Link\_7\_2

### A.9. Geometry of Link\_8\_1

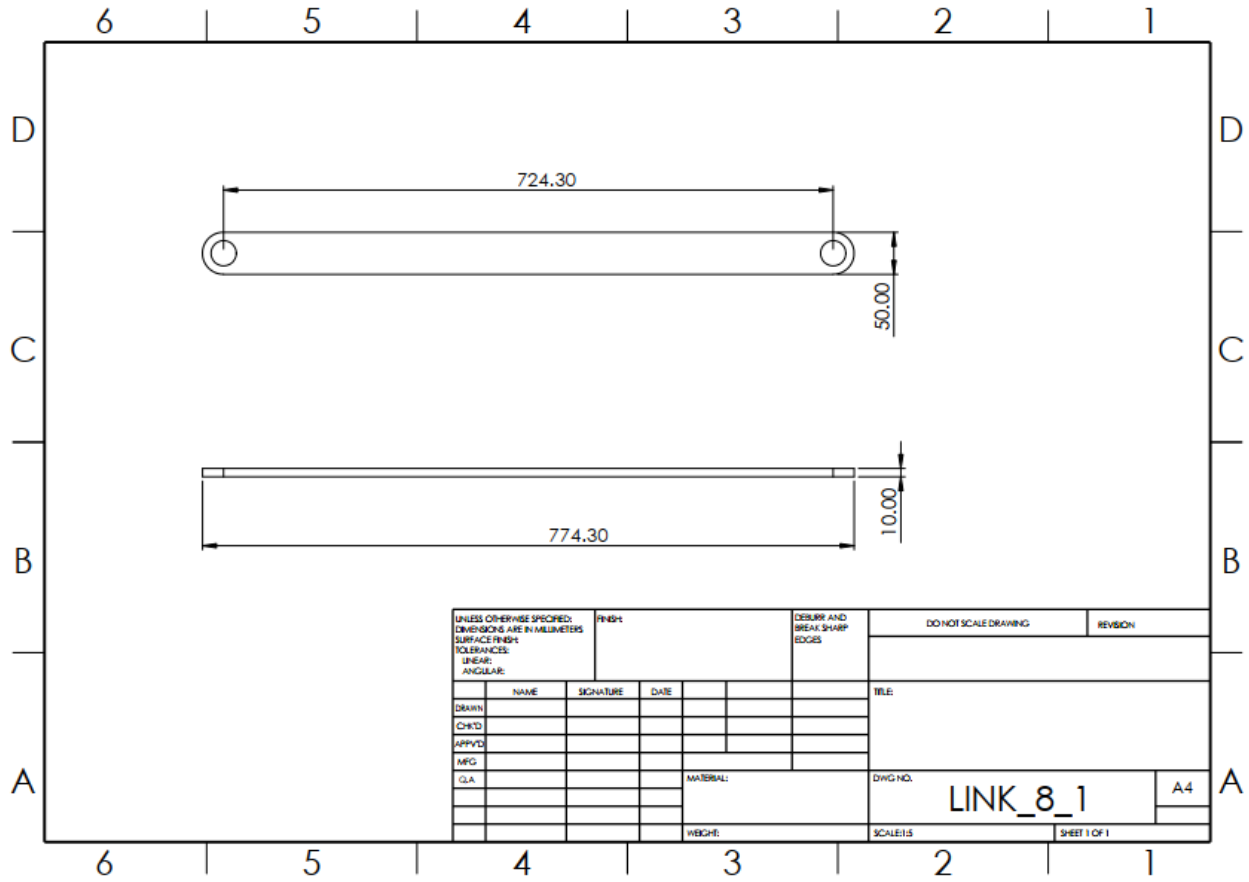


Figure – A.9: Geometry and Dimensions of Link\_8\_1 and Link\_8\_2

### A.10. Geometry and Dimensions of the Tire Carrier

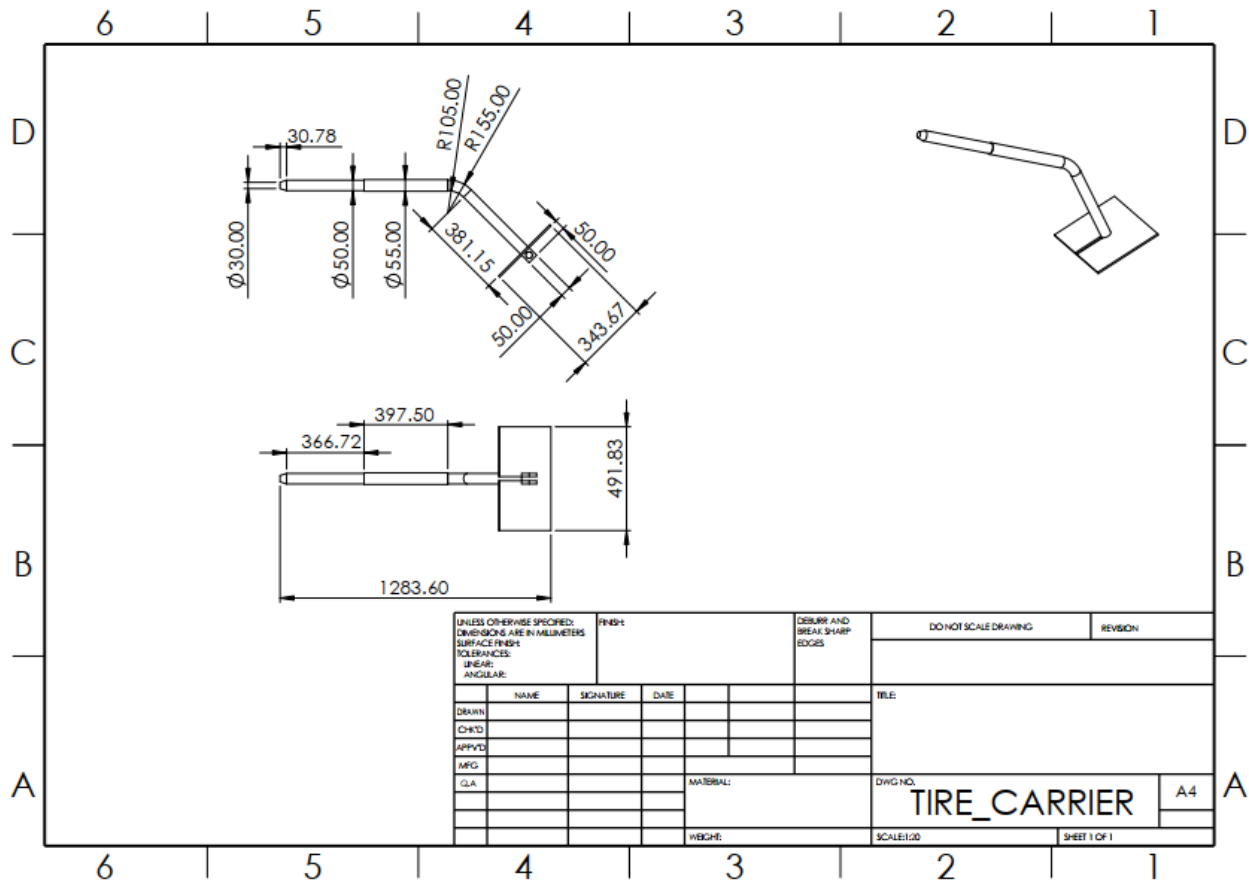


Figure – A.10: Geometry and Dimensions of the Tire Carrier

## A.11. Largest and Smallest Wheel Specifications Used in the Design of the Mechanism

### A.11.1. Largest Wheel Specifications

*Table – A.11.1: Specifications of the Largest Wheel Used in the Mechanism’s Design [60]*

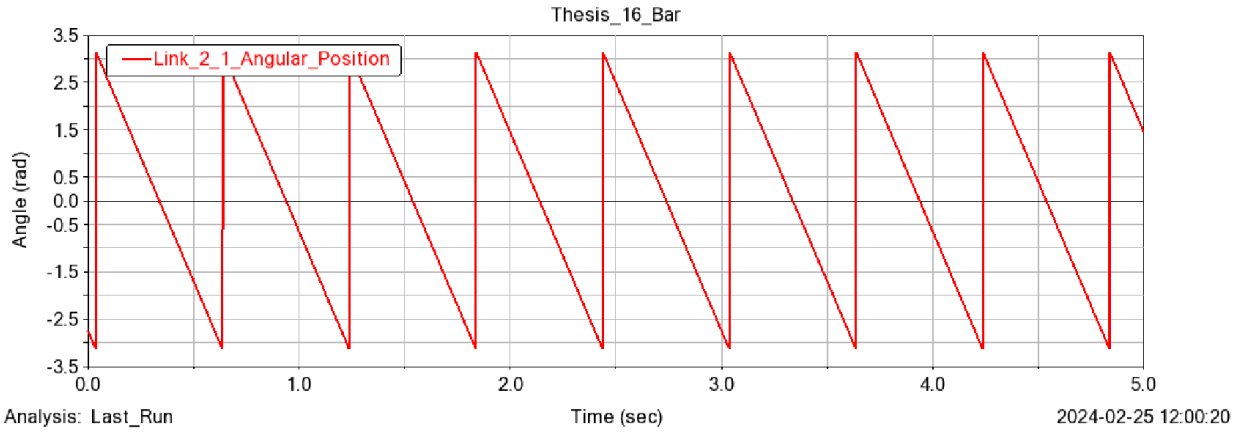
<b>Tire Code</b>	275/55R20113H
<b>Rim</b>	8.5Jx20ET44
<b>Rim Diameter</b>	508 mm
<b>Center Bore</b>	87.1 mm
<b>Tire Width (W)</b>	275 mm
<b>Aspect Ratio (AR)</b>	55%
<b>Height (H)</b>	151.25 mm
<b>Total Diameter (TD)</b>	810.5 mm

### A.11.2. Smallest Wheel Specifications

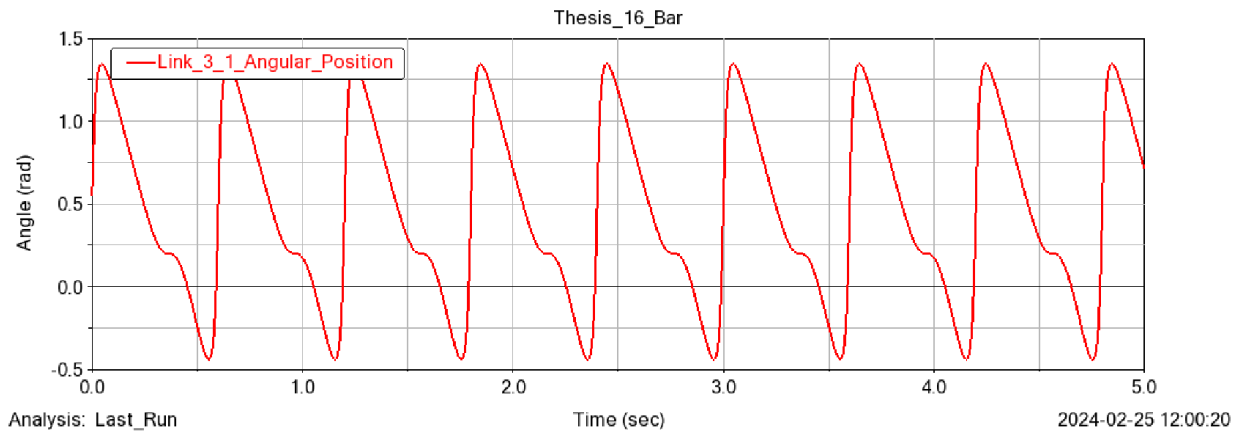
*Table – A.11.2: Specifications of the Smallest Wheel Used in the Mechanism’s Design [60]*

<b>Tire Code</b>	125/80R1365M
<b>Rim</b>	4Bx13ET23.5
<b>Rim Diameter</b>	330.2 mm
<b>Center Bore</b>	60.1 mm
<b>Tire Width (W)</b>	125 mm
<b>Aspect Ratio (AR)</b>	80%
<b>Height (H)</b>	100 mm
<b>Total Diameter (TD)</b>	530.2 mm

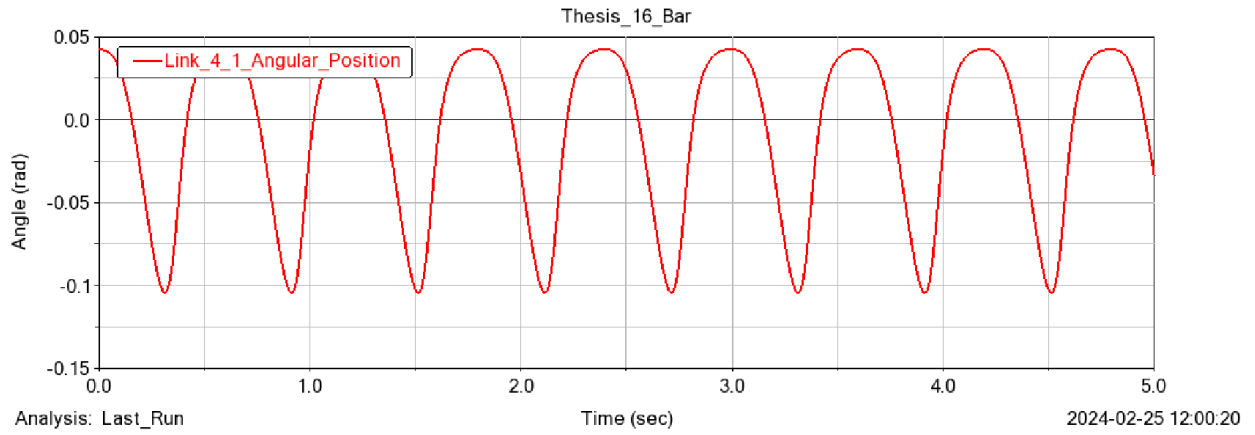
## Appendix-B: Results of Simulation



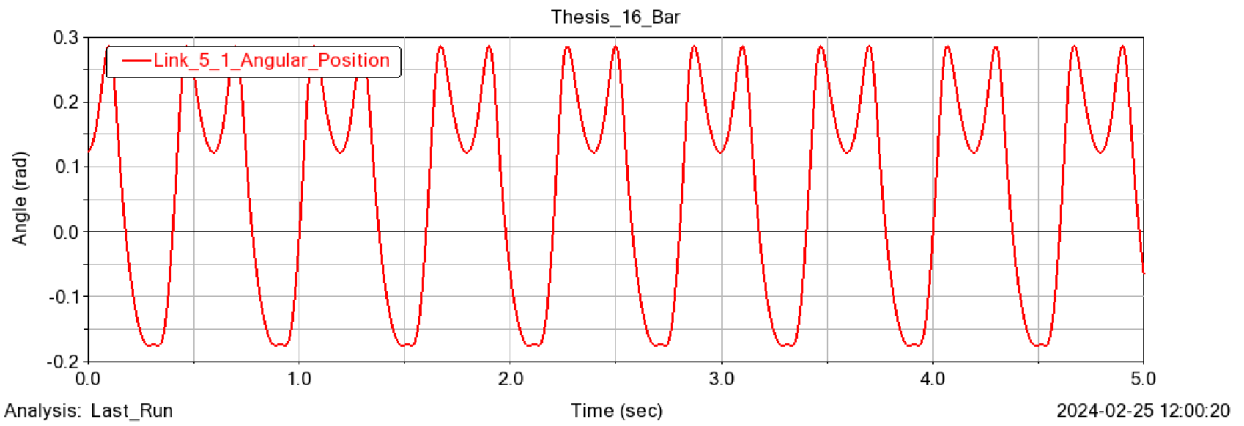
**Figure – B.1:** Angular Position versus Time of Link\_2\_1 for Five Seconds of Simulation



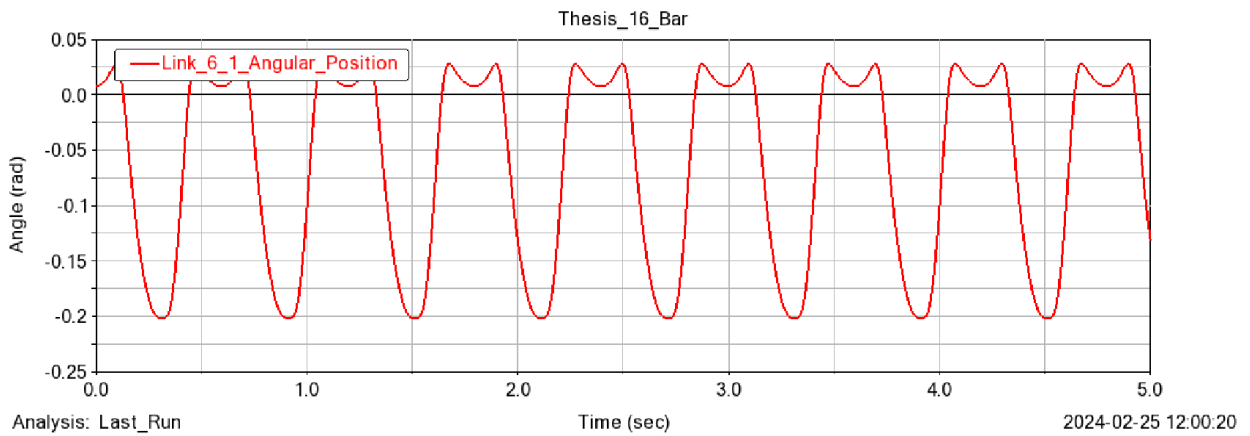
**Figure – B.2:** Angular Position versus Time of Link\_3\_1 for Five Seconds of Simulation



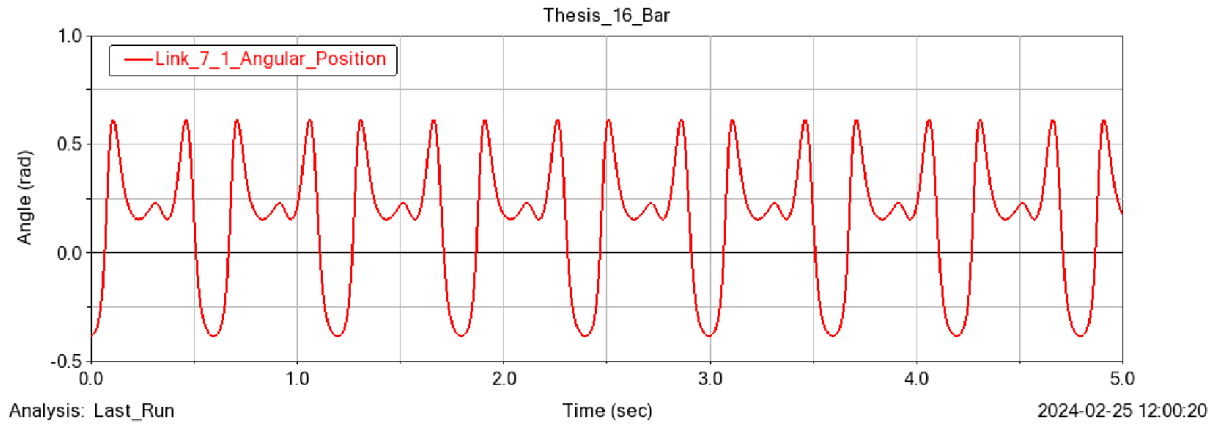
**Figure – B.3:** Angular Position versus Time of Link\_4\_1 for Five Seconds of Simulation



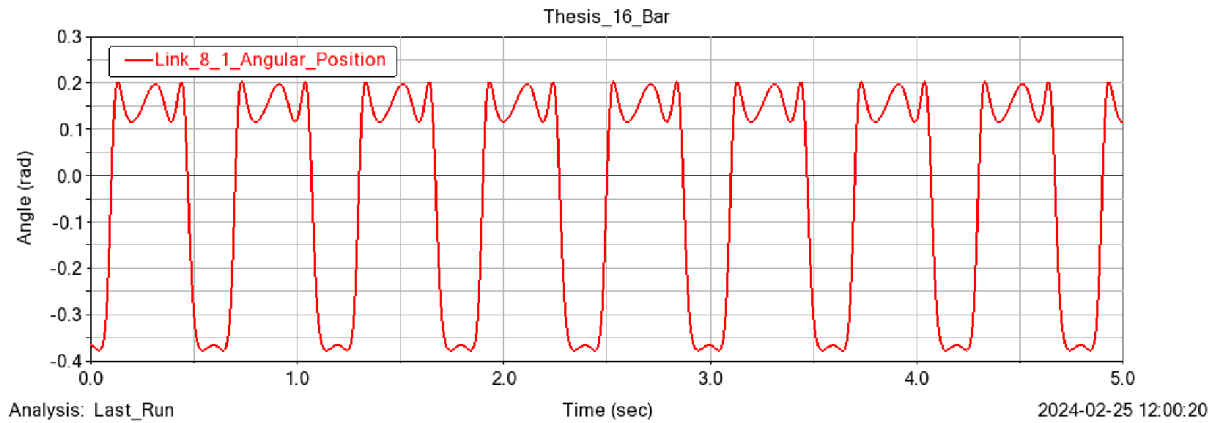
**Figure – B.4:** Angular Position versus Time of Link\_5\_1 for Five Seconds of Simulation



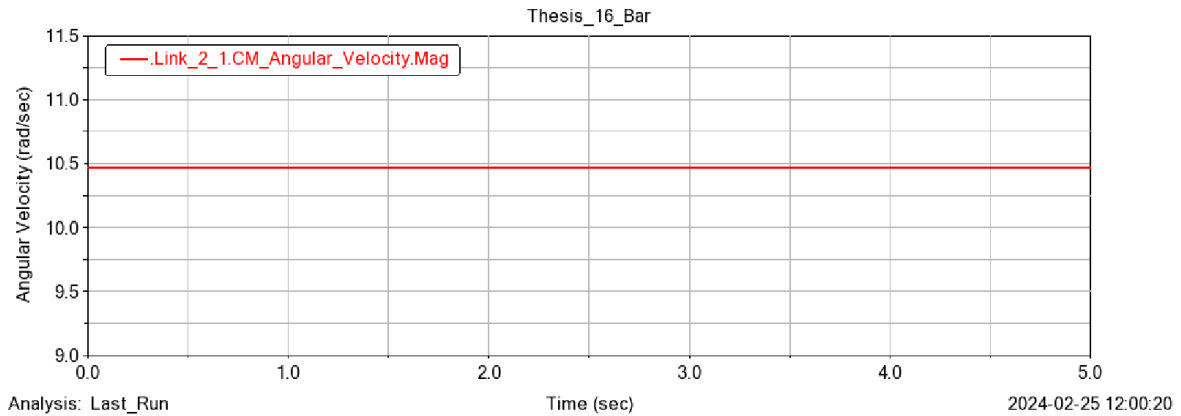
**Figure – B.5:** Angular Position versus Time of Link\_6\_1 for Five Seconds of Simulation



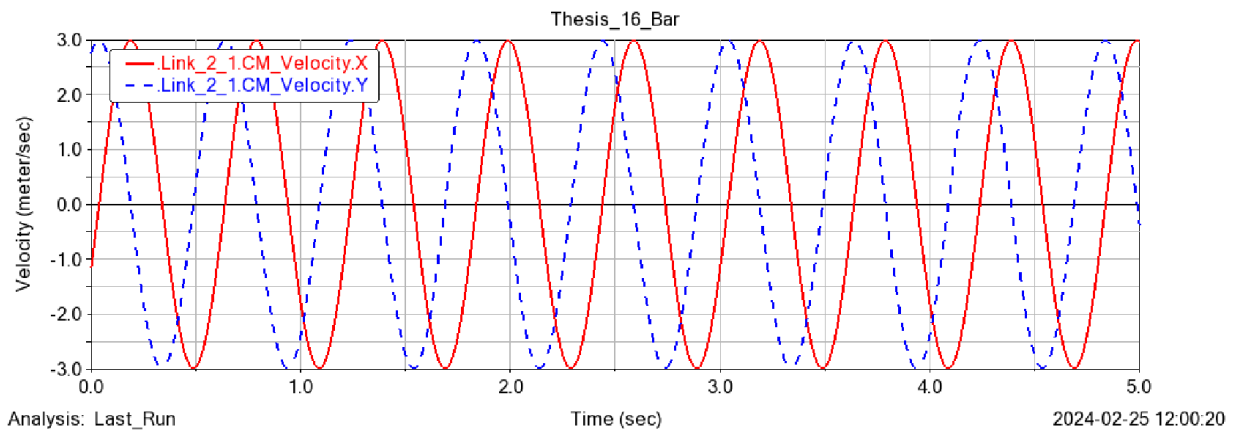
**Figure – B.6:** Angular Position versus Time of Link\_7\_1 for Five Seconds of Simulation



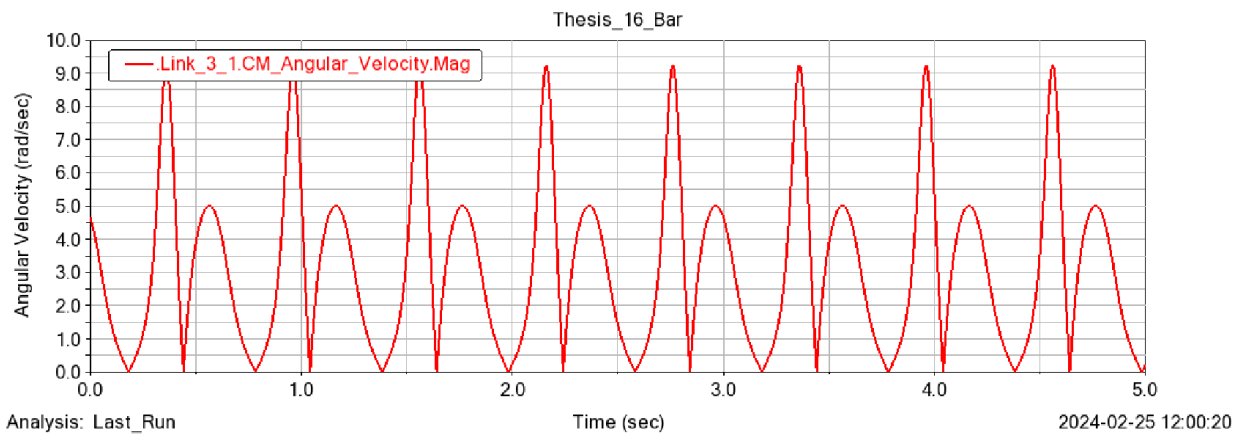
**Figure – B.7:** Angular Position versus Time of Link\_8\_1 for Five Seconds of Simulation



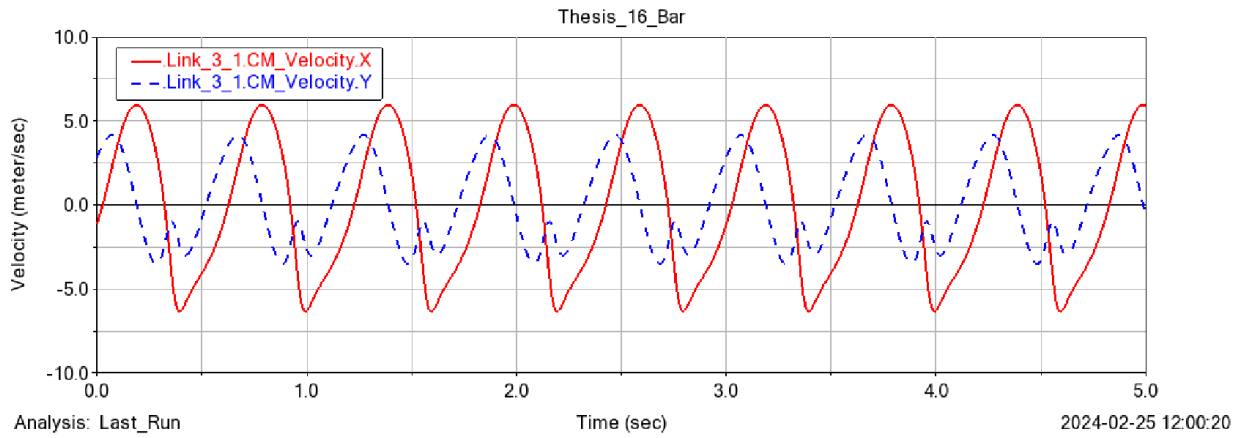
**Figure – B.8:** Angular Velocity versus Time of Link\_2\_1 for Five Seconds of Simulation



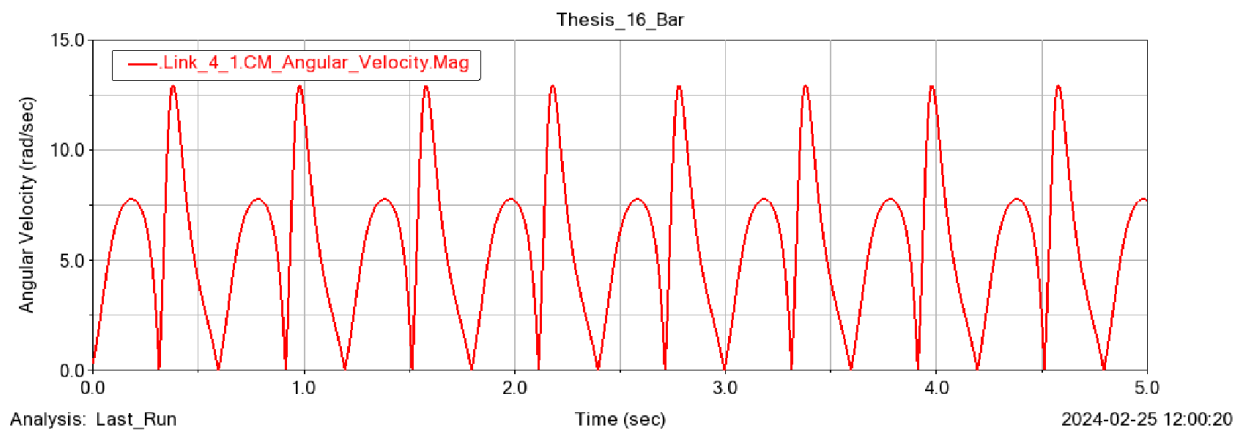
**Figure – B.9:** X- and Y- Direction Linear Velocity Components versus Time of Link\_2\_1 for Five Seconds of Simulation



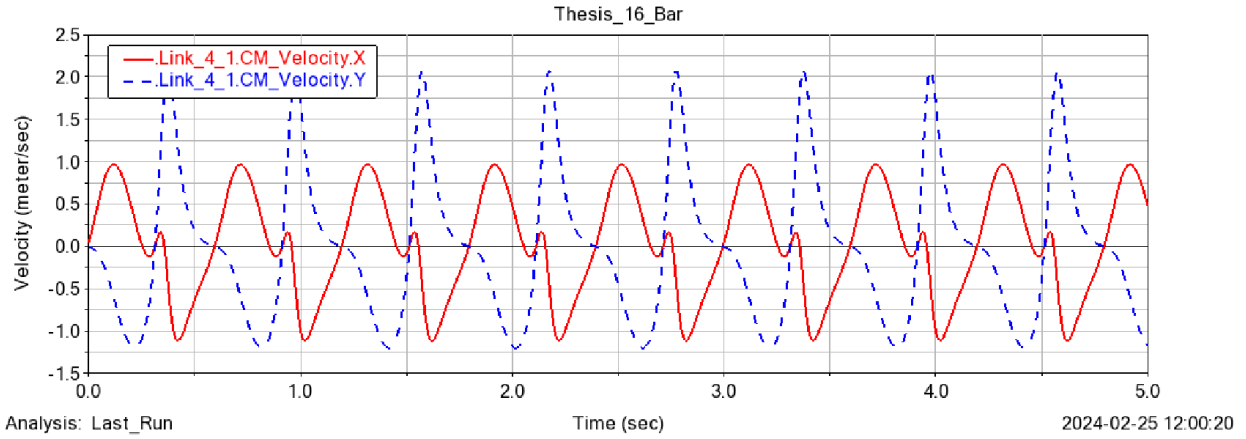
**Figure – B.10:** Angular Velocity versus Time of Link\_3\_1 for Five Seconds of Simulation



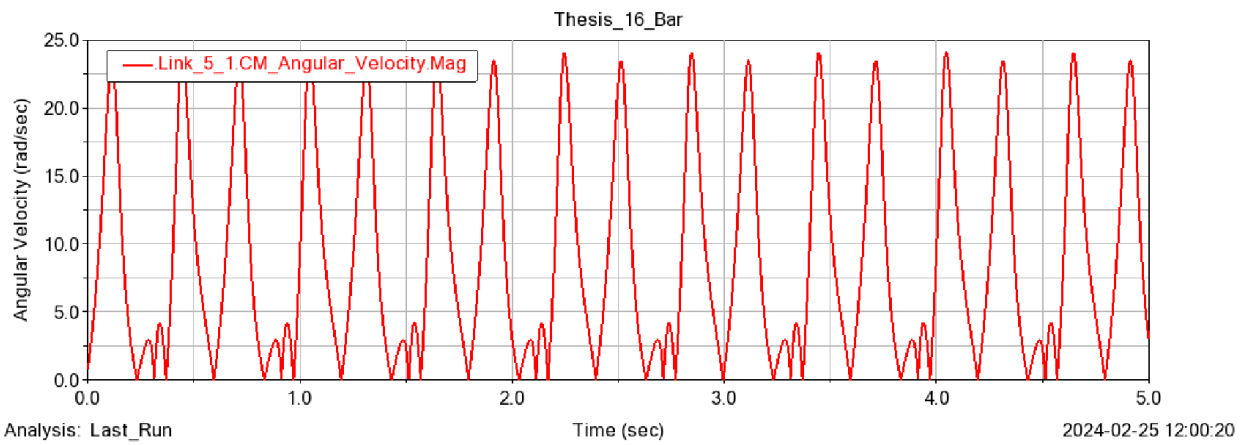
**Figure – B.11:** X- and Y- Direction Linear Velocity Components versus Time of Link\_3\_1 for Five Seconds of Simulation



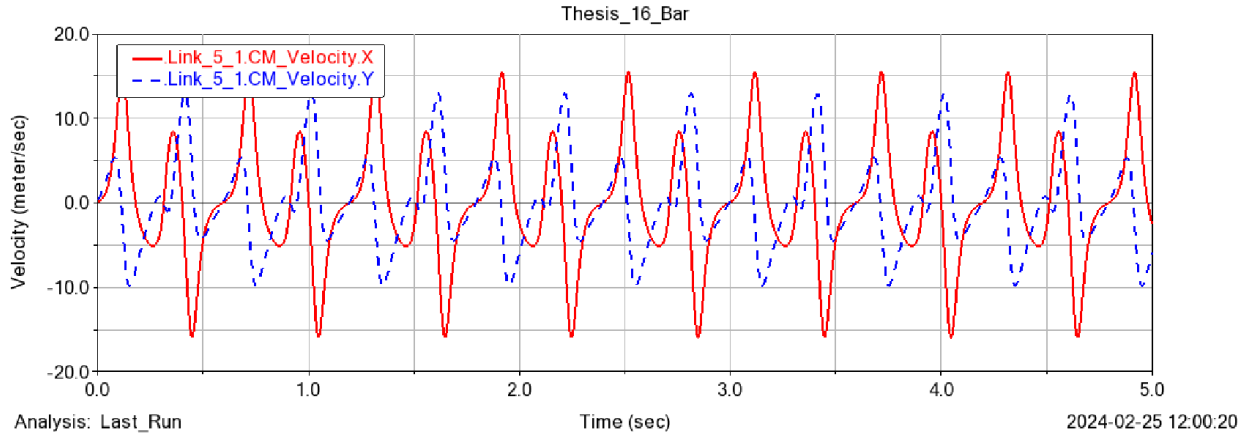
**Figure – B.12:** Angular Velocity versus Time of Link\_4\_1 for Five Seconds of Simulation



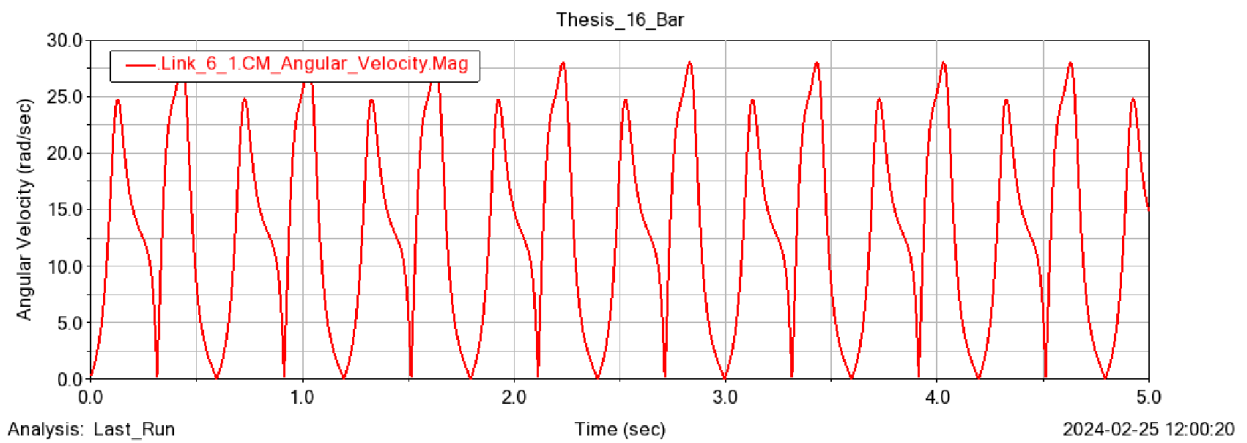
**Figure – B.13:** *X- and Y- Direction Linear Velocity Components versus Time of Link\_4\_1 for Five Seconds of Simulation*



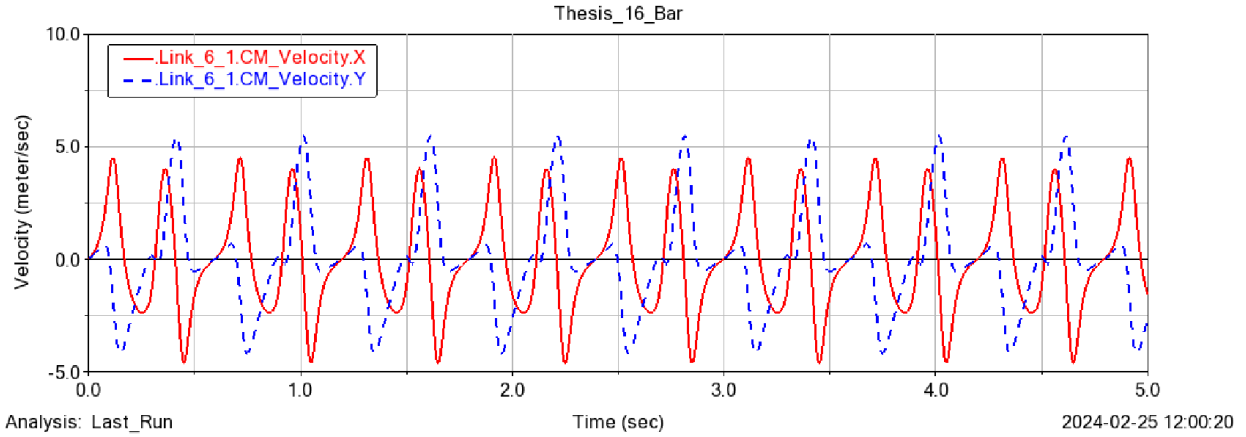
**Figure – B.14:** *Angular Velocity versus Time of Link\_5\_1 for Five Seconds of Simulation*



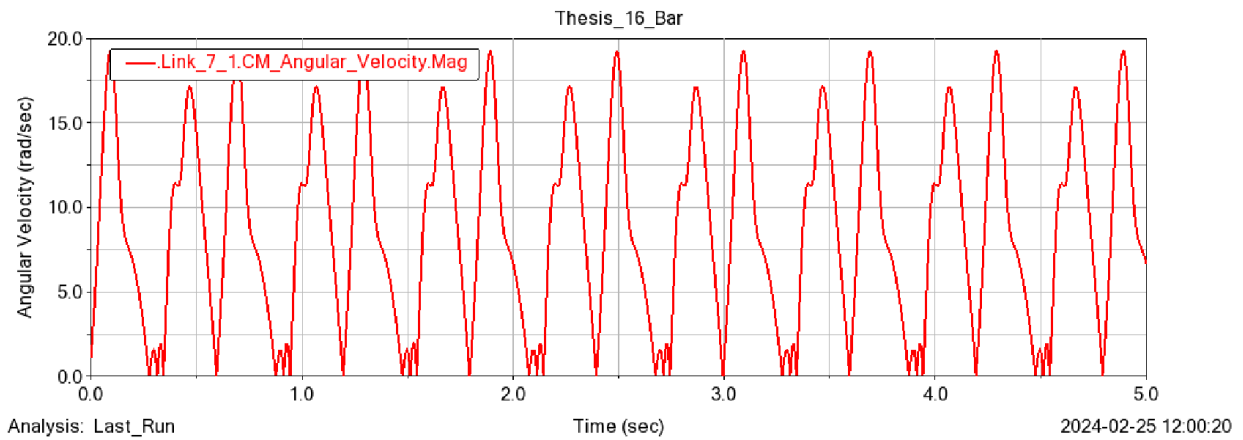
**Figure – B.15:** X- and Y- Direction Linear Velocity Components versus Time of Link\_5\_1 for Five Seconds of Simulation



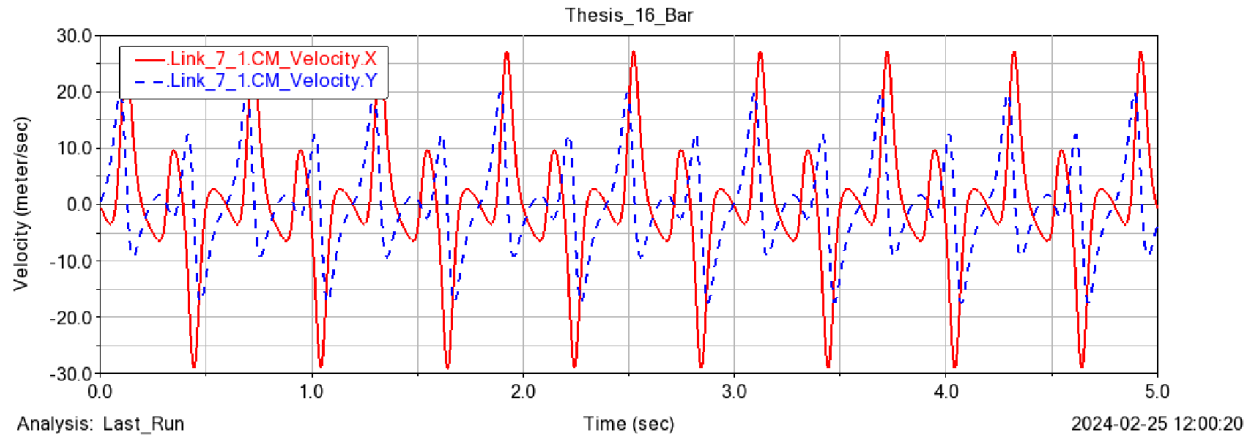
**Figure – B.16:** Angular Velocity versus Time of Link\_6\_1 for Five Seconds of Simulation



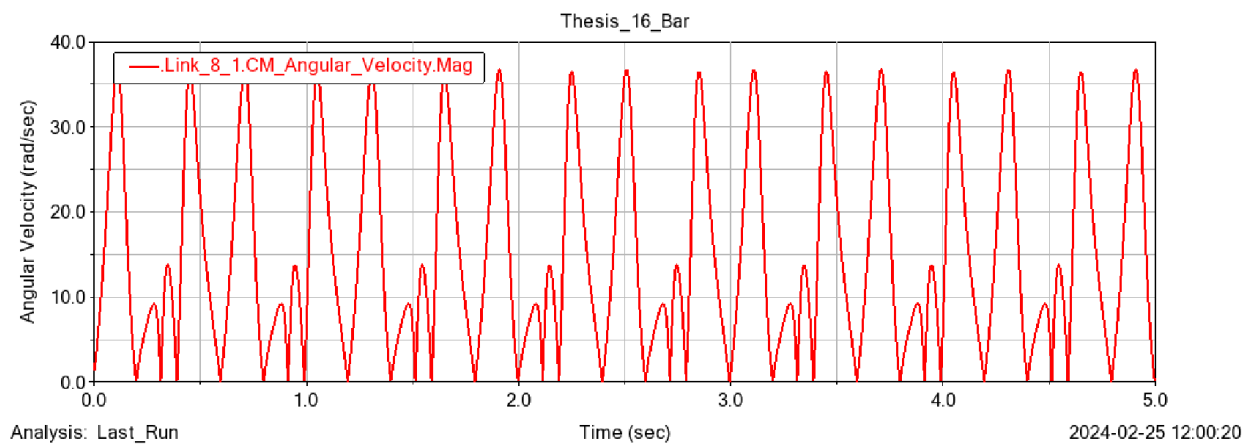
**Figure – B.17:** X- and Y- Direction Linear Velocity Components versus Time of Link\_6\_1 for Five Seconds of Simulation



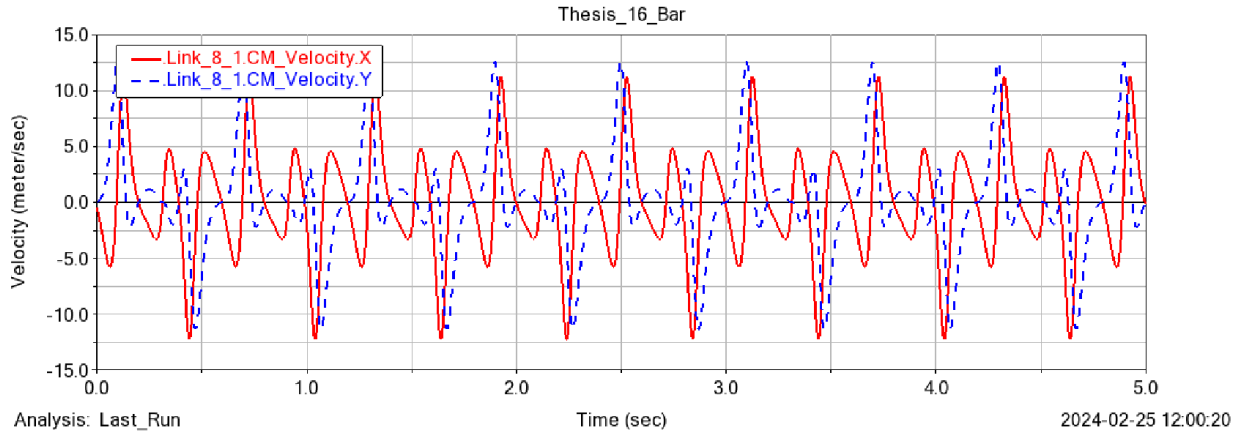
**Figure – B.18:** Angular Velocity versus Time of Link\_7\_1 for Five Seconds of Simulation



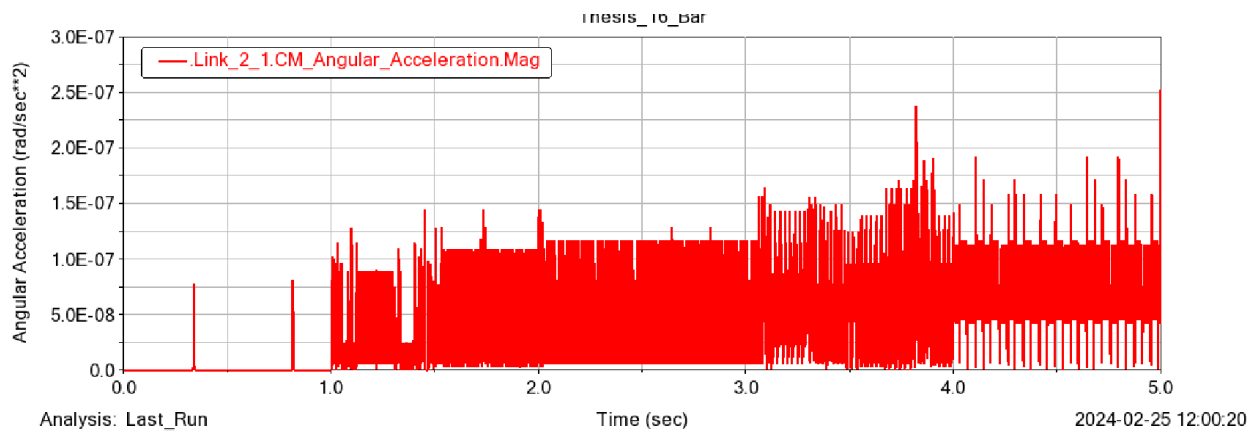
**Figure – B.19:** *X- and Y- Direction Linear Velocity Components versus Time of Link\_7\_1 for Five Seconds of Simulation*



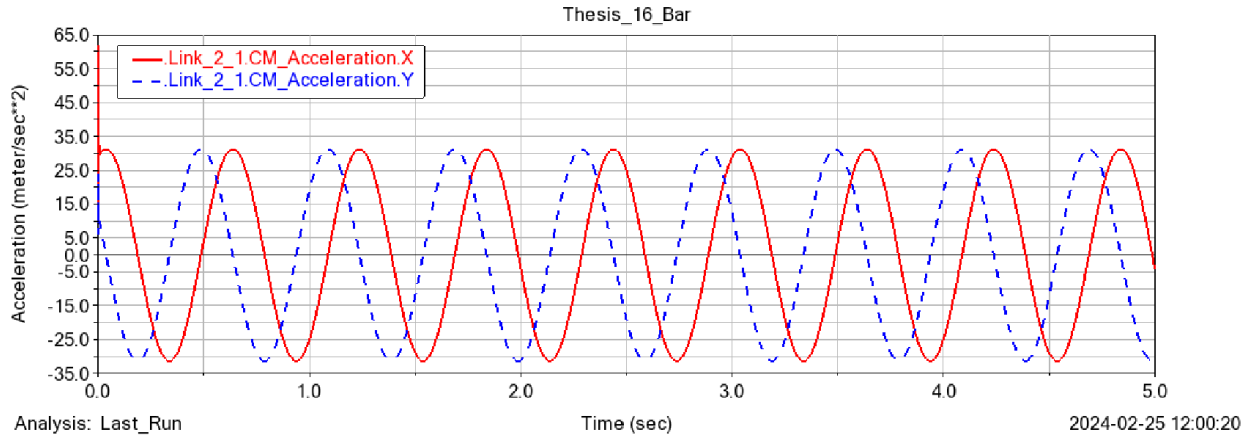
**Figure – B.20:** *Angular Velocity versus Time of Link\_8\_1 for Five Seconds of Simulation*



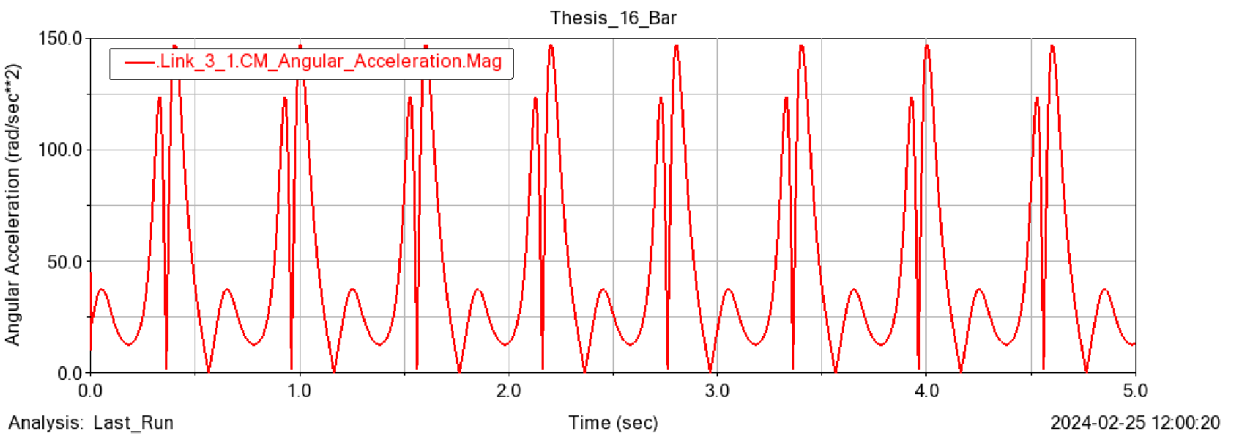
**Figure – B.21:** *X- and Y- Direction Linear Velocity Components versus Time of Link\_8\_1 for Five Seconds of Simulation*



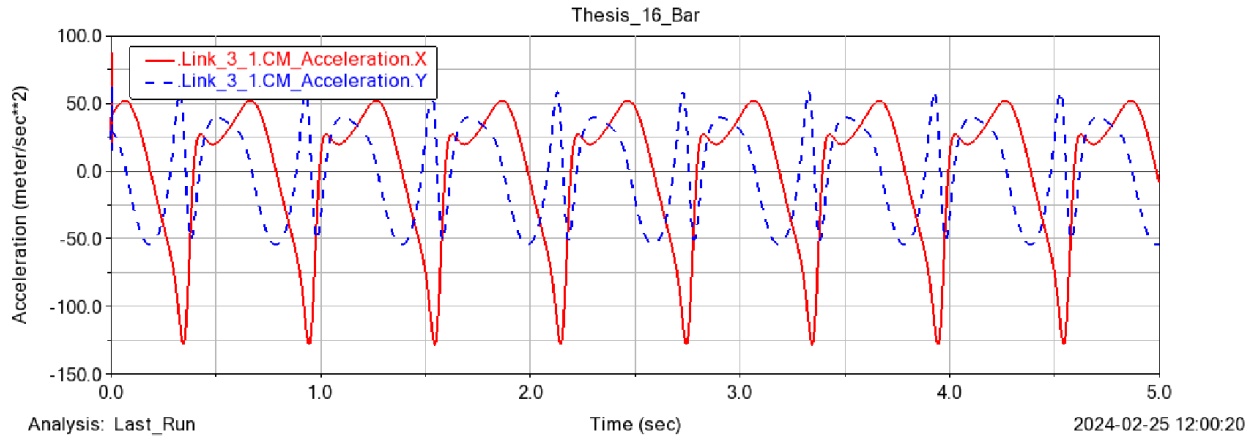
**Figure – B.22:** *Angular Acceleration versus Time of Link\_2\_1 for Five Seconds of Simulation*



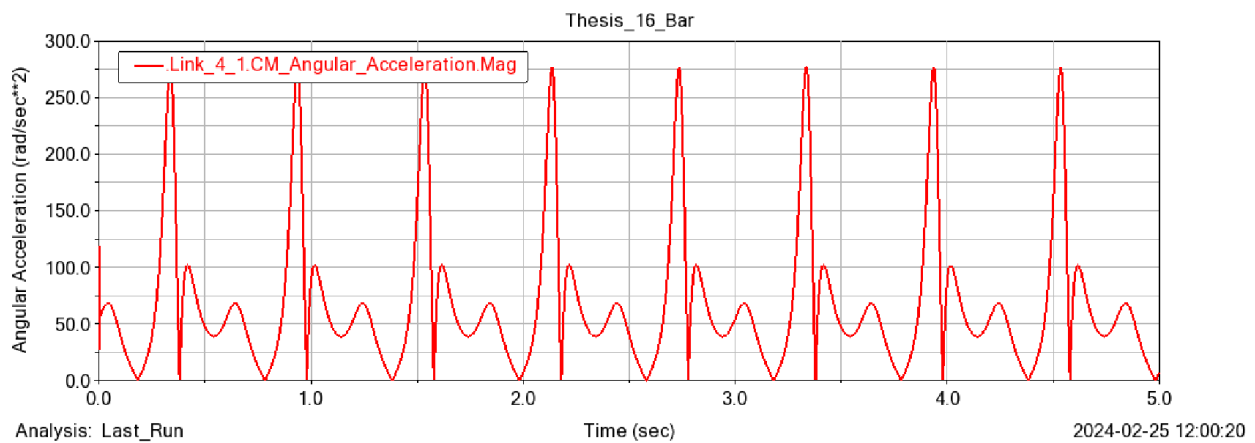
**Figure – B.23:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_2\_1 for Five Seconds of Simulation



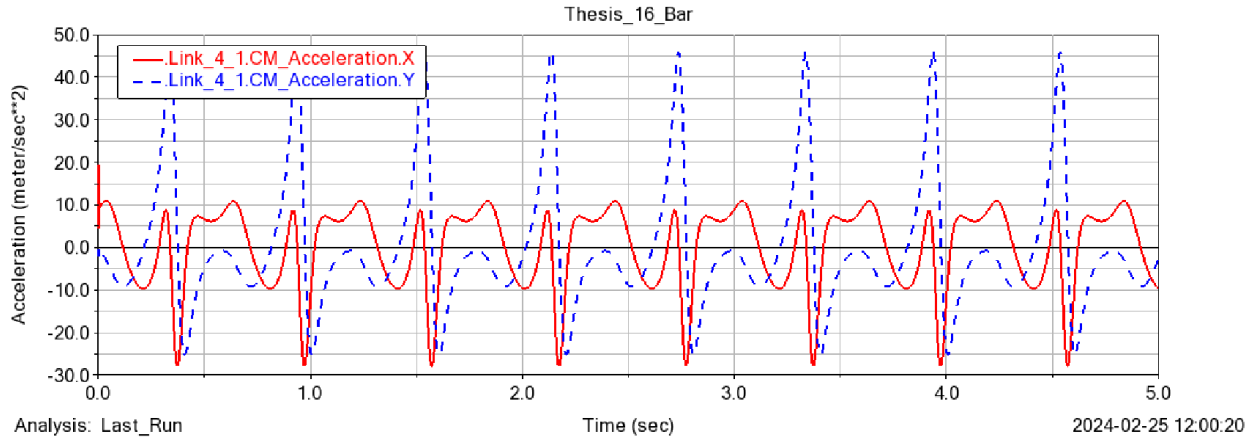
**Figure – B.24:** Angular Acceleration versus Time of Link\_3\_1 for Five Seconds of Simulation



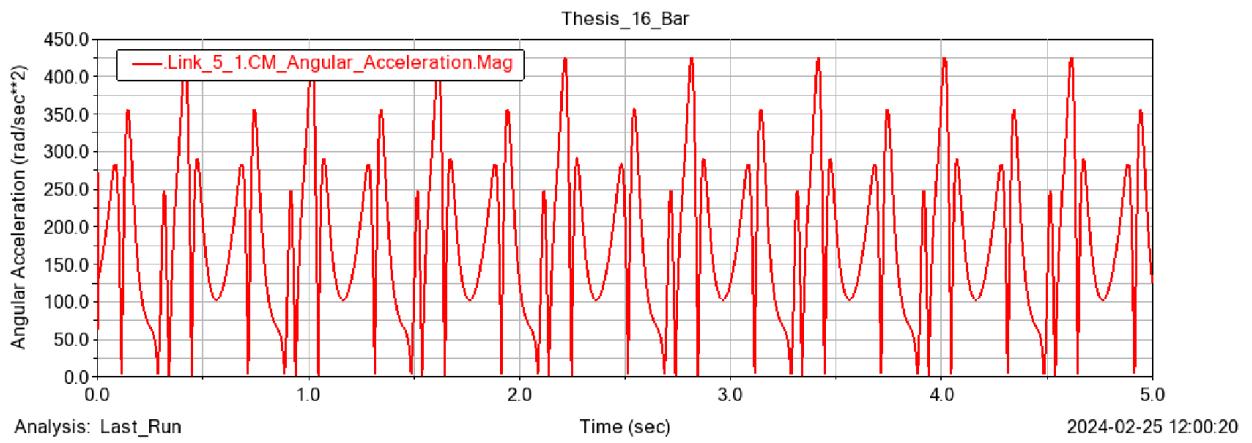
**Figure – B.25:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_3\_1 for Five Seconds of Simulation



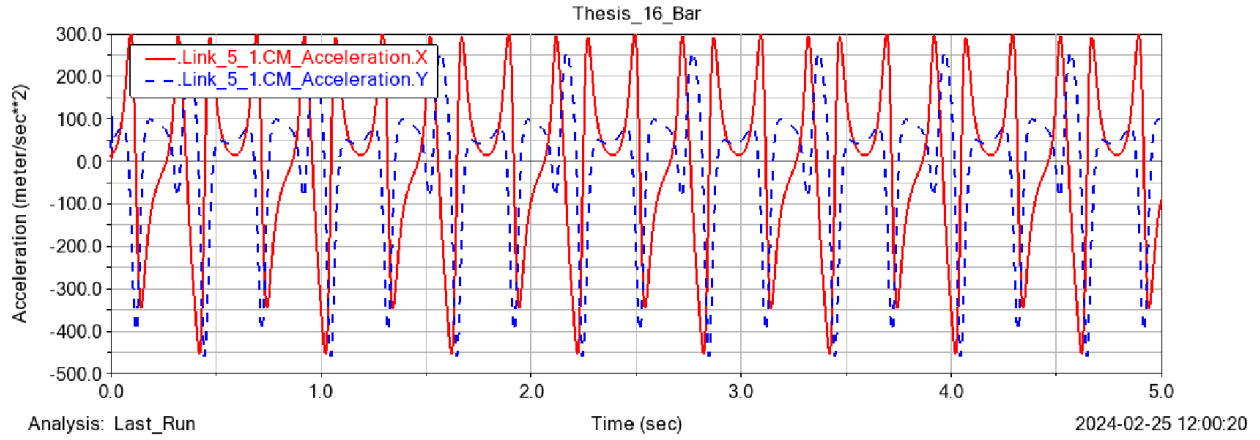
**Figure – B.26:** Angular Acceleration versus Time of Link\_4\_1 for Five Seconds of Simulation



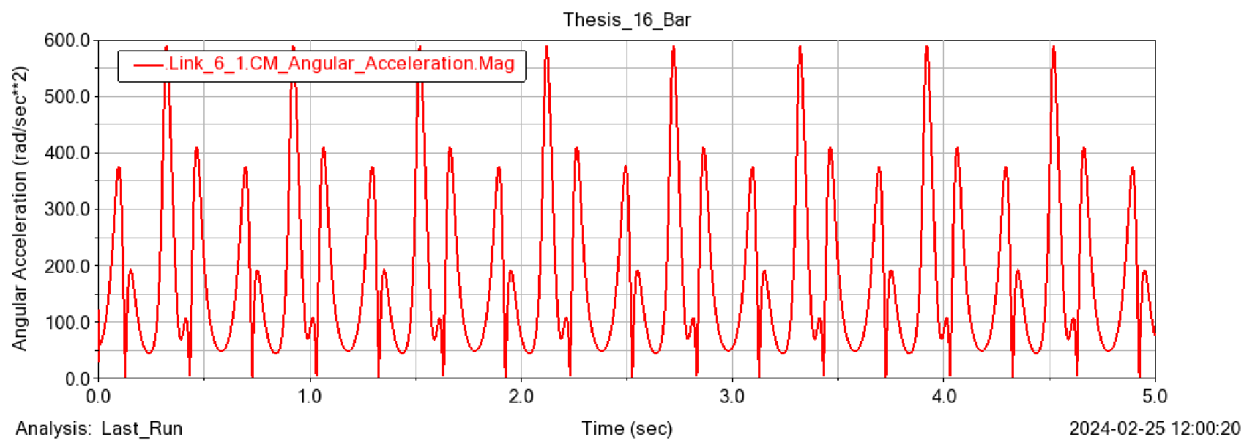
**Figure – B.27:** *X- and Y- Direction Linear Acceleration Components versus Time of Link\_4\_1 for Five Seconds of Simulation*



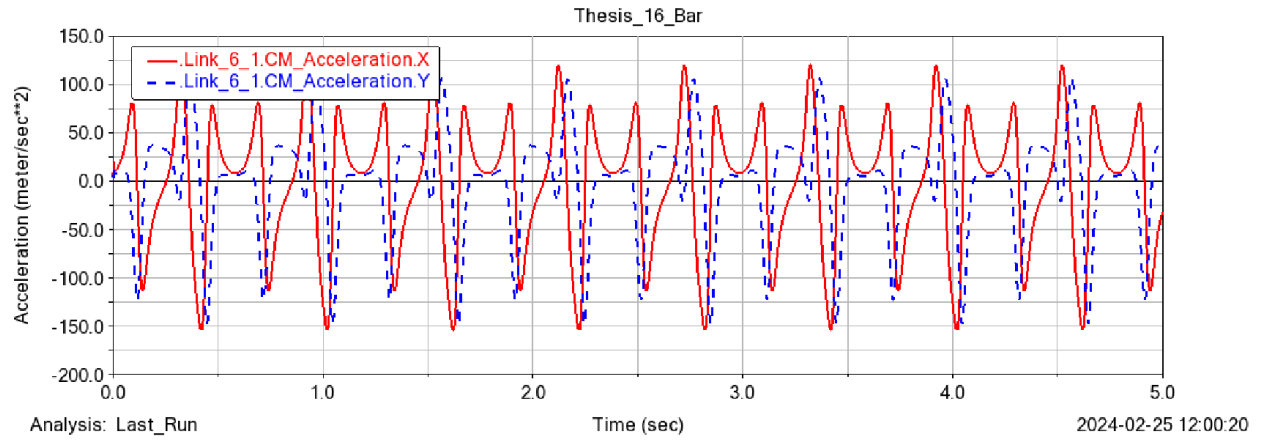
**Figure – B.28:** *Angular Acceleration versus Time of Link\_5\_1 for Five Seconds of Simulation*



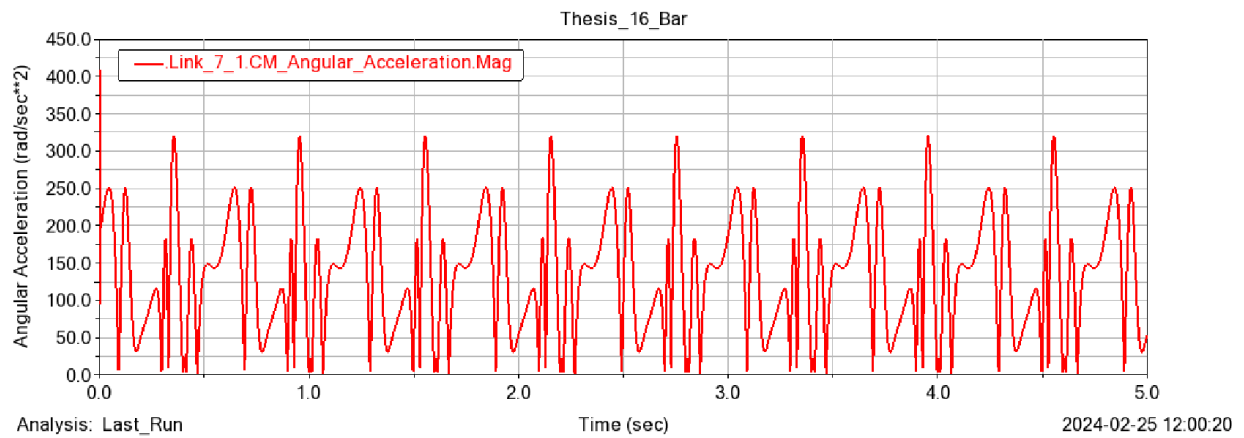
**Figure – B.29:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_5\_1 for Five Seconds of Simulation



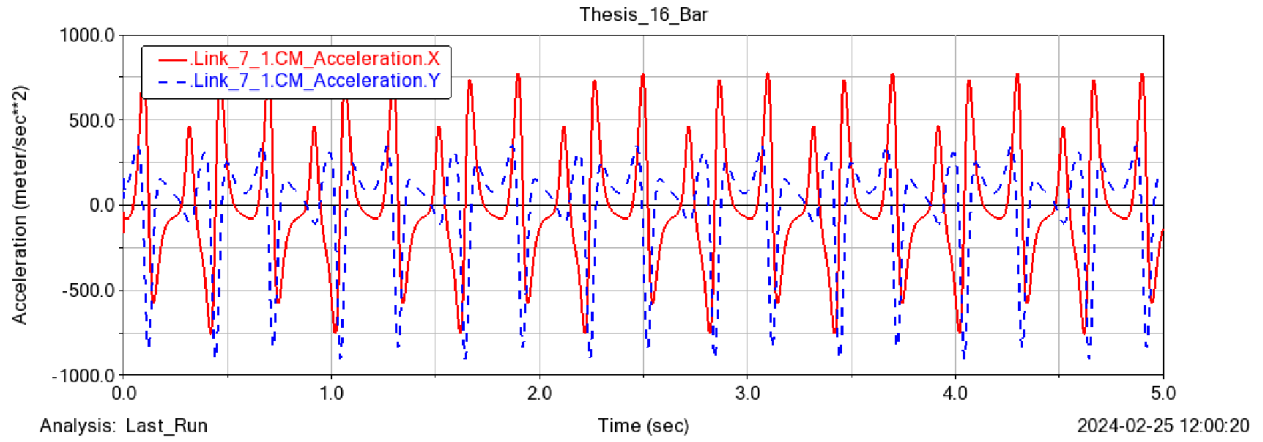
**Figure – B.30:** Angular Acceleration versus Time of Link\_6\_1 for Five Seconds of Simulation



**Figure – B.31:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_6\_1 for Five Seconds of Simulation



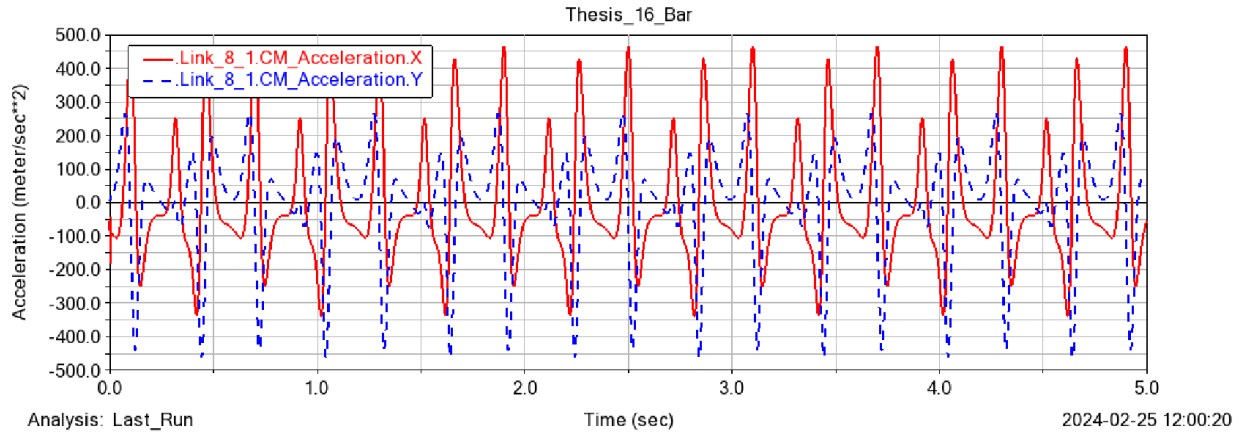
**Figure – B.32:** Angular Acceleration versus Time of Link\_7\_1 for Five Seconds of Simulation



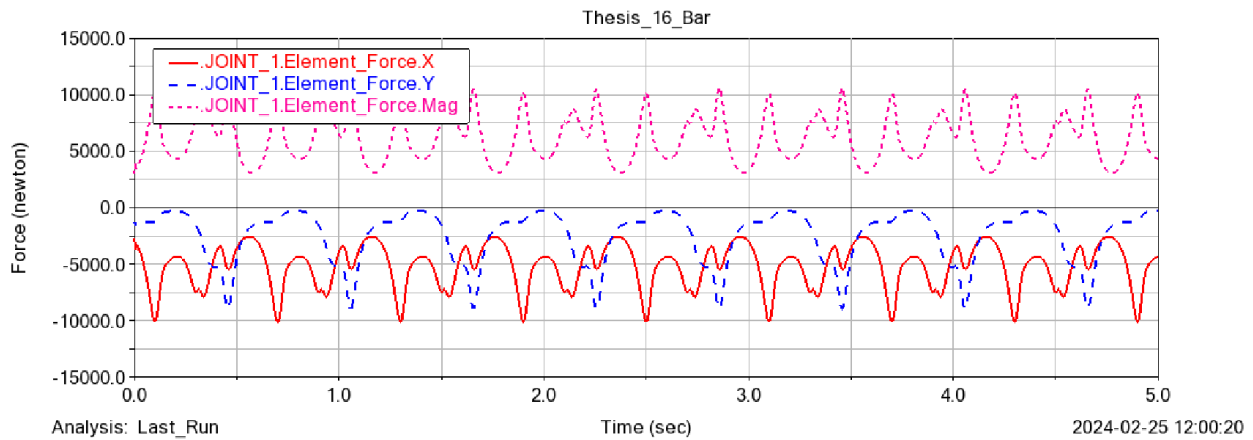
**Figure – B.33:** *X- and Y- Direction Linear Acceleration Components versus Time of Link\_7\_1 for Five Seconds of Simulation*



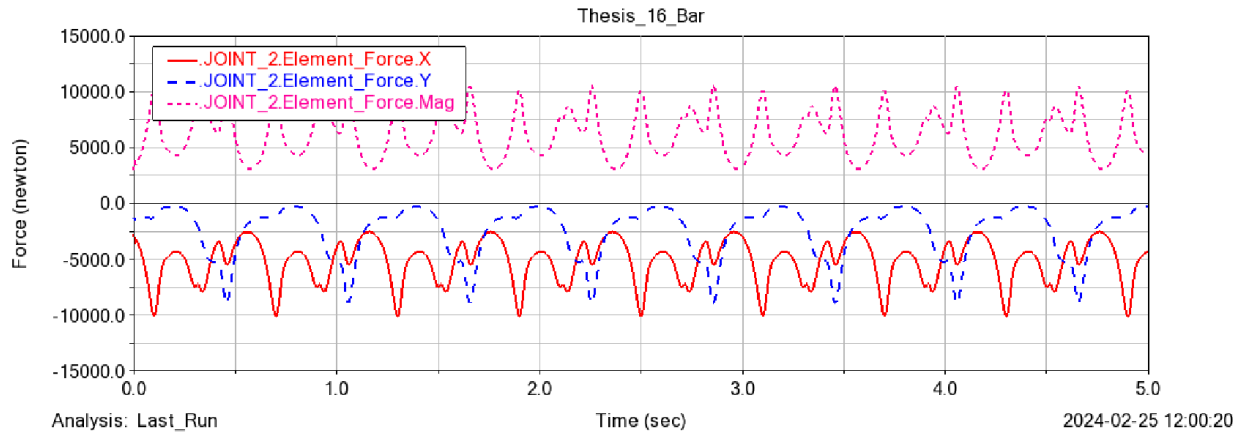
**Figure – B.34:** *Angular Acceleration versus Time of Link\_8\_1 for Five Seconds of Simulation*



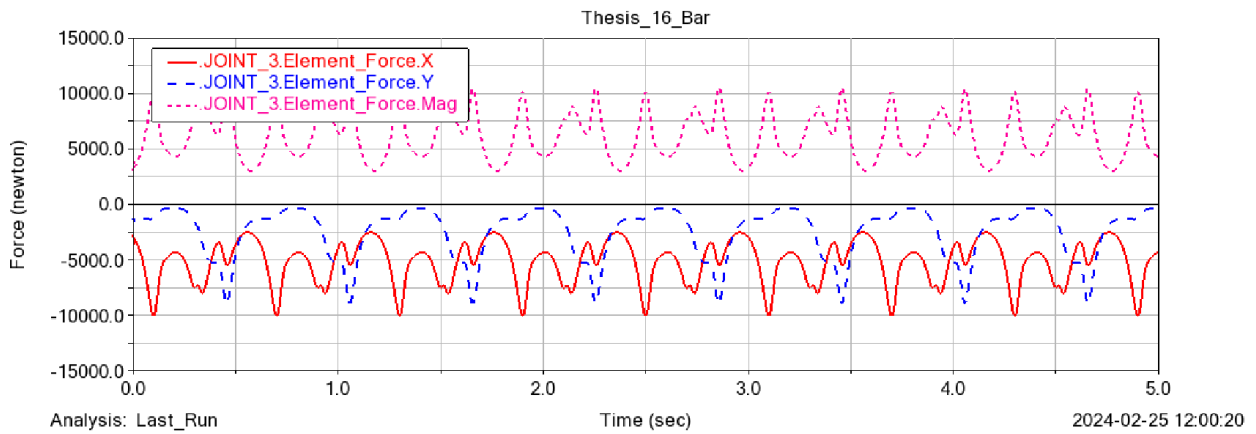
**Figure – B.35:** X- and Y- Direction Linear Acceleration Components versus Time of Link\_8\_1 for Five Seconds of Simulation



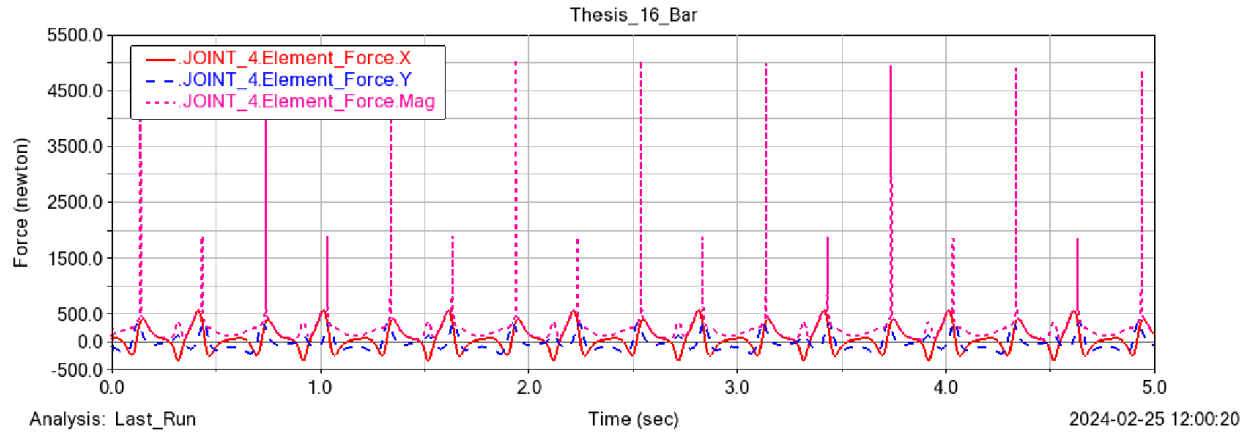
**Figure – B.36:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_1 for Five Seconds of Simulation



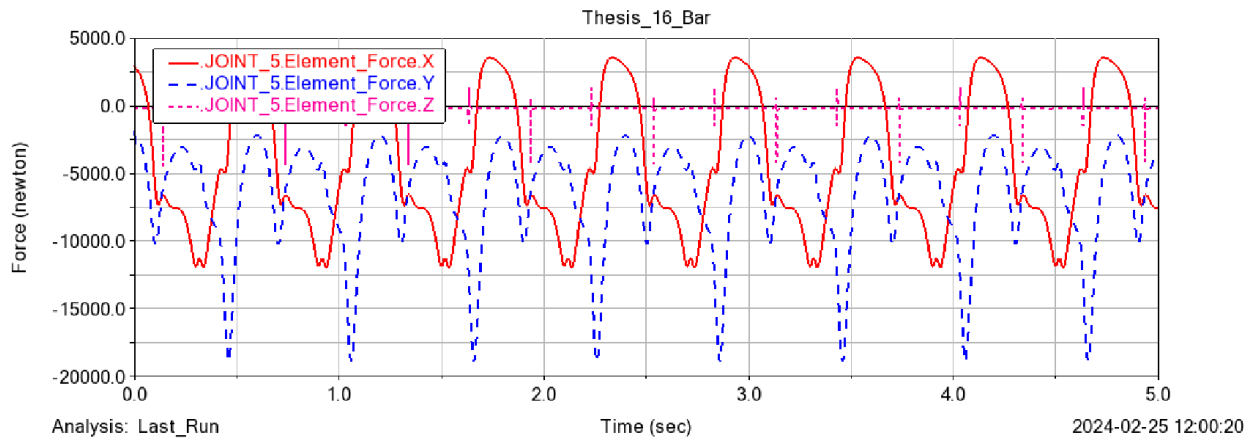
**Figure – B.37:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_2 for Five Seconds of Simulation



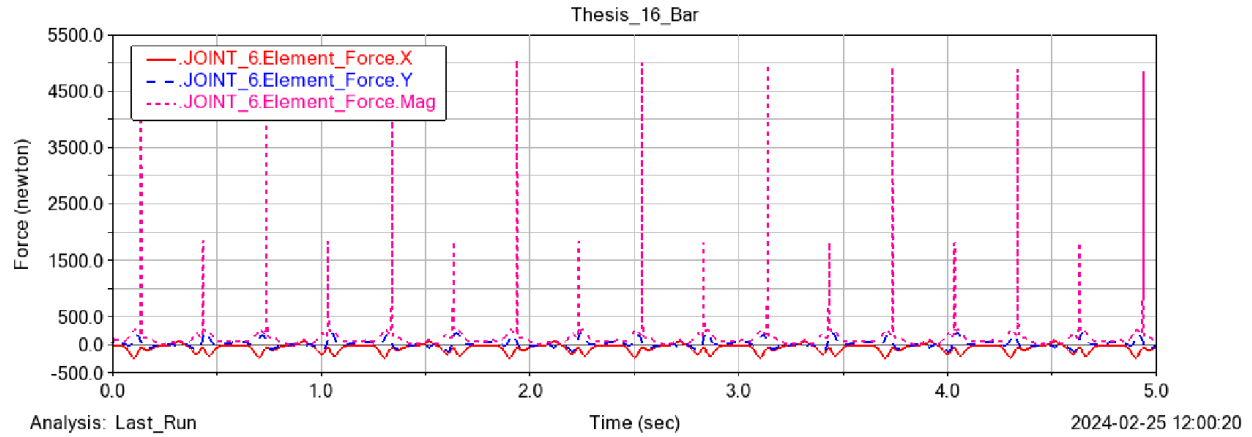
**Figure – B.38:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_3 for Five Seconds of Simulation



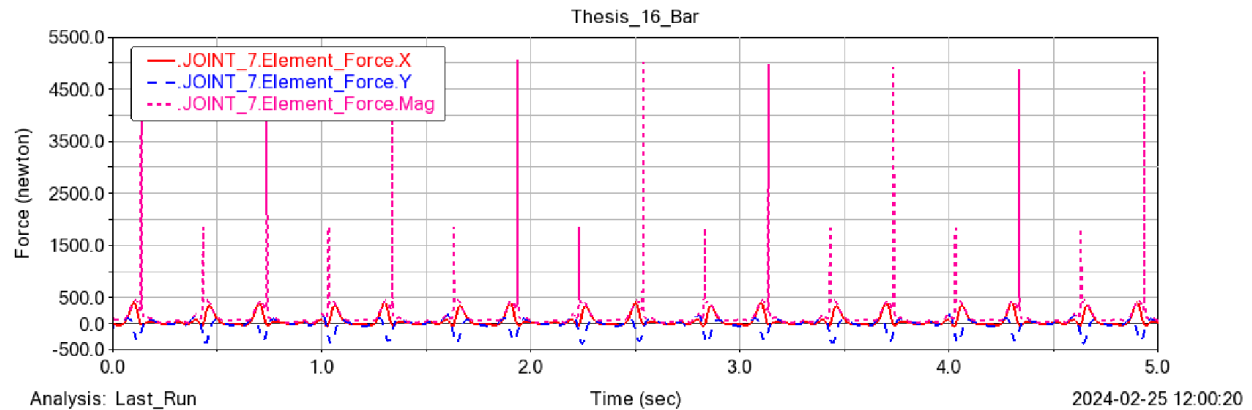
**Figure – B.39:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_4 for Five Seconds of Simulation



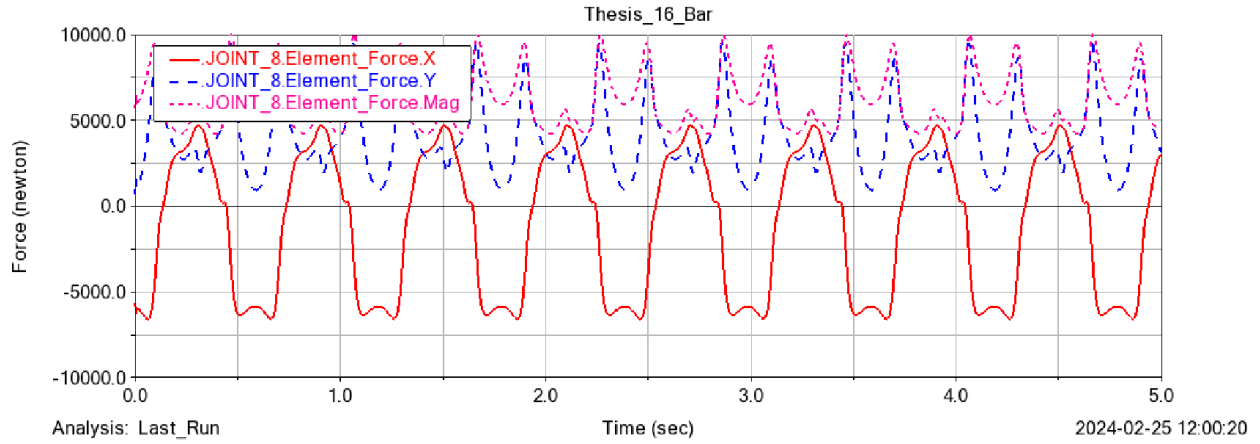
**Figure – B.40:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_5 for Five Seconds of Simulation



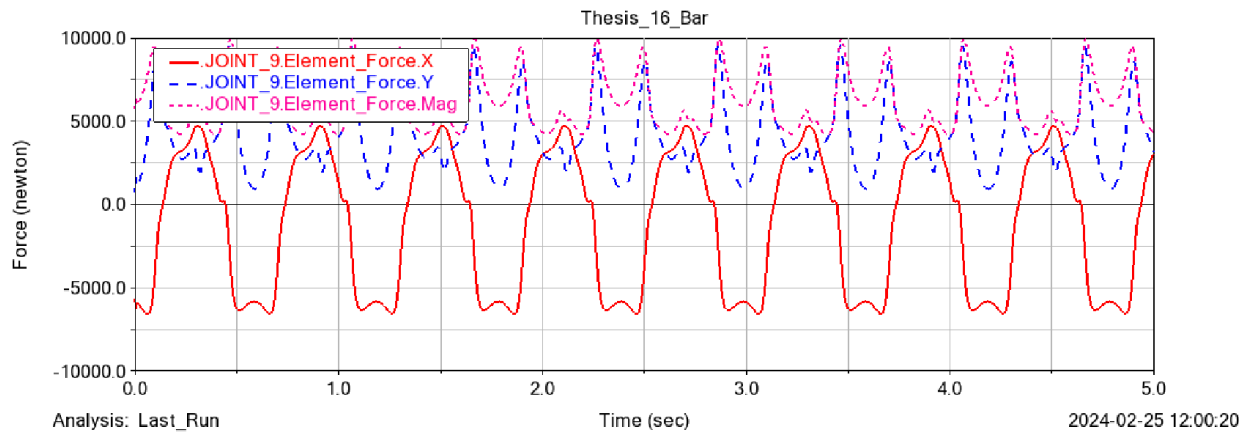
**Figure – B.41:** *X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_6 for Five Seconds of Simulation*



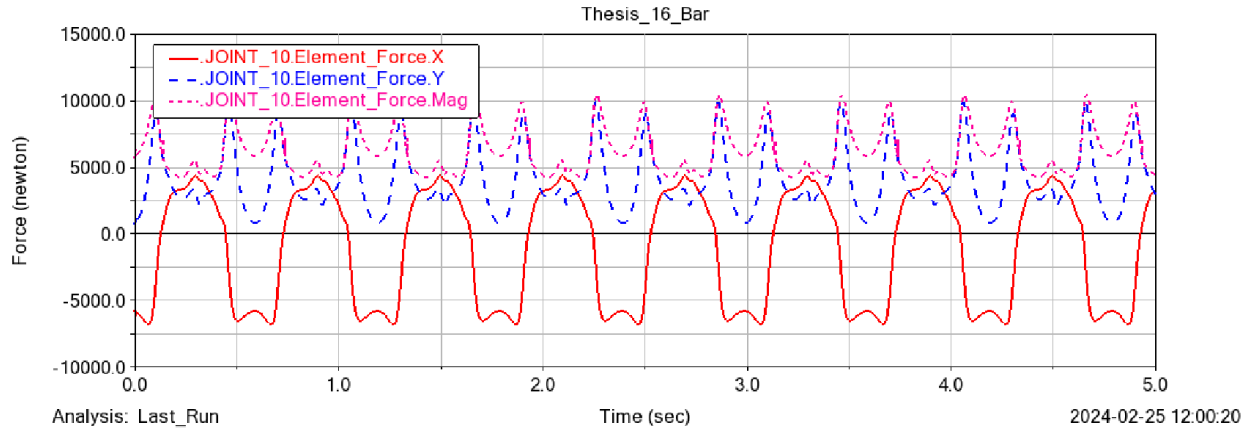
**Figure – B.42:** *X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_7 for Five Seconds of Simulation*



**Figure – B.43:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_8 for Five Seconds of Simulation



**Figure – B.44:** X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_9 for Five Seconds of Simulation



*Figure – B.45: X- and Y- Direction Joint Force Components and Magnitude of Resultant Force versus Time of Joint\_10 for Five Seconds of Simulation*