STUDY ON DESIGN OF FOUR-BAR MECHANISM IN VIBRATORY TILLAGE TOOLS

BY

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PROJECT REPORT

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DECLARATION

We, hereby declare that this report entitles **"STUDY ON DESIGN OF FOUR-BAR MECHANISM IN VIBRATORY TILLAGE TOOLS"** is a bona-fide record of research work done by us during the course of B. Tech and this report has not previously formed the basis for the award to us of any degree, diploma, associateship, fellowship or other similar title, of any other university or society.

Place: Tavanur

Date:

Mohammed Suhail V K (2021-02-025)

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Roshni V S (2021-02-034)

CERTIFICATE

Certified that this report entitled **"STUDY ON DESIGN OF FOUR-BAR MECHANISM IN VIBRATORY TILLAGE TOOLS"** is a record of research work done independently by Mr. Mohammed Suhail V K, Ms. Chaithanya Vijayan, Ms. Roshni V S under my guidance and supervision and that it has not previously formed the basis for the award of any degree, diploma, fellowship or associateship to them.

Place: Tavanur
Date:

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DEDICATED TO OUR PROFESSION

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CHAPTER I

INTRODUCTION

Vibratory tillage tools have revolutionized soil preparation by incorporating oscillatory motion to conventional tillage methods. These tools work by introducing vibrations to the usual soil-cutting action, which significantly reduces the resistance faced by the tool as it moves through the soil. Vibrations make it easier for the tillage tool to penetrate compacted, hard soils, breaking them up more effectively. The result is improved soil structure, finer soil particles, and reduced compaction, all of which contribute to better plant growth. One of the major benefits of vibratory tillage tools is that they require less power to operate compared to traditional tillage tools, as the vibrations reduce friction and drag forces during soil engagement. This leads to energy efficiency and lower operational costs, making them particularly useful for farming in challenging conditions such as compacted, dry, or rocky soils. Vibratory tillage has gained traction in modern farming as it also helps preserve soil structure, minimizes wear and tear on equipment, and is more environmentally friendly by reducing fuel consumption.

In agricultural machinery, various mechanical mechanisms are used to generate the required motion for specific tasks, including tillage. These mechanisms are designed to transform motion from the power source into the desired motion at the tool. Some common mechanisms include:

- Slider-Crank Mechanism: Often used to convert rotary motion into linear motion. It is a simple and effective system used in engines and pumps.
- Cam-Follower Mechanism: This mechanism uses a cam profile to generate oscillatory or reciprocating motion. It is often found in systems requiring precise control over the tool’s movement.
- Four-Bar Mechanism: This is a closed-loop linkage system that can be configured to produce rotary or oscillatory motion. The four-bar mechanism is one of the most widely used because of its simplicity, versatility, and efficiency in converting motion.

- Elliptical Mechanism: Generates an elliptical path, often used for tools that require more complex motion patterns, such as in certain harvesting machinery.
- Worm Gear Mechanism: Used when a large reduction in speed is needed while maintaining a high torque output. It’s commonly used in systems where gear reduction is a priority.

Each of these mechanisms can be selected based on the kind of motion required by the tillage tool. However, for the purpose of generating vibratory motion with precision and simplicity, the four-bar mechanism is often the preferred choice.

Among these mechanisms, the four-bar mechanism stands out as the most versatile and reliable choice for generating the required oscillatory or vibrating motion in tillage tools. It is selected due to its ability to convert rotary motion from a motor or power source into a smooth and controlled oscillation. This oscillation is crucial for the vibratory action in tillage tools. The four-bar linkage is particularly attractive because it can be configured to provide a range of different motion outputs (such as continuous rotation, limited oscillation, or reciprocating motion) depending on the design and needs of the tillage tool. Its relatively simple design, consisting of four rigid links connected in a closed loop, enables efficient and precise control over the tool's movement, making it ideal for a wide variety of agricultural machinery, including vibratory tillage tools. Additionally, the four-bar mechanism allows for easy customization of the link lengths and angles, enabling the mechanism to perform effectively under different field conditions, whether in compacted soil or loose, well-aerated soil.

The four-bar mechanism is a fundamental type of mechanical linkage system that consists of four rigid links connected in a closed loop by four rotating joints (also known as revolute pairs). These four links include:

- Input Link (Crank): The link that is typically driven by an external power source or motor.
- Coupler Link: A link that connects the input and output links and generally performs an intermediate motion.

- Output Link (Rocker): The link that generates the desired output motion (such as oscillation or rotation).
- Fixed Link (Frame): The stationary link that supports the entire mechanism.

The working principle of the four-bar mechanism is simple yet effective, when the input link is rotated by an external force, it causes the other links to move in a predictable manner, either resulting in oscillatory or continuous rotary motion at the output link. This motion is transmitted through the coupler link. The lengths of the links and their arrangement are crucial in determining the type of motion generated. For example, the mechanism can be configured as a crank-rocker mechanism, where one link undergoes continuous rotation while the other rocks back and forth, or as a double-crank mechanism, where both input and output links rotate fully.

This mechanism is especially advantageous for generating vibratory motion because it allows for precise control over the frequency and amplitude of the vibration. By adjusting the link lengths and other parameters, the four-bar mechanism can produce the ideal oscillatory motion needed for tools such as vibratory tillage equipment. It provides an efficient, cost-effective way to generate the dynamic cutting or vibrating actions required to break up soil and prepare fields for planting.

The four-bar mechanism's ability to create controlled oscillation with low energy consumption makes it an ideal choice for vibratory tillage tools. By using a motor or engine to drive the input link (crank), the mechanism can produce oscillations that help the tillage tool penetrate the soil more easily, break up compacted layers, and improve soil aeration. These vibrations reduce the friction between the tool and the soil, leading to lower energy usage, higher efficiency, and reduced wear on the equipment. In the context of tillage, this makes the four-bar mechanism a highly effective, low-maintenance option for creating the dynamic and periodic movement needed to optimize soil preparation for crop growth.

The project has been carried out to develop a four-bar mechanism in vibratory tillage tools

with the following objective.

- To study engineering parameters towards the development of four-bar mechanism in vibratory tillage tools.
- To simulate the four-bar mechanism in vibratory tillage tools using MATLAB.
- To design four-bar mechanism in vibratory tillage tools using solid work software.
- To optimise the design finite element analysis.

Review of literature

CHAPTER II

REVIEW OF LITERATURE

Brief review of the works done relevant to different aspects of this research, which include study of the four-bar mechanism, widely used in agriculture, robotics, and rehabilitation for efficient motion control. Researchers apply MATLAB simulations and optimization algorithms like genetic and particle swarm to enhance design performance. These studies show improved energy efficiency, precision, and sustainability., are explained in this chapter.

2.1 DESIGN OF FOURBAR MECHANISM FOR VIBRATORY TILLAGE TOOL

According to Chaudhary *et al*. (2018), the design and development of a vibratory cultivator using optimization algorithms aim to enhance soil tilling efficiency by optimizing vibration frequency, amplitude, and blade design. By employing advanced techniques such as genetic algorithms and particle swarm optimization, the study focuses on improving tillage performance while significantly reducing energy consumption. This approach ensures better soil penetration and reduced traction requirements, making vibratory tillage more efficient and sustainable.

Butson and McIntyre (1981) conducted soil tank experiments to measure the draught and power requirements of agricultural implements. Their research aimed to simulate field conditions in a controlled environment, enabling the determination of how soil texture, moisture, and implement design affect machinery performance and energy efficiency.

According to Gauripati Rao *et al*. (2018), tillage is one of the most energy-intensive agricultural operations, second only to irrigation. To improve efficiency, they designed a four-bar linkage-based vibratory mechanism for tillage applications. The study utilized MATLAB and MSC Adams to model and validate the tool's motion, ensuring it followed the desired sinusoidal trajectory. The analysis of velocity and acceleration confirmed that the vibratory mechanism reduces traction requirements and enhances soil penetration. Their findings suggest that oscillatory motion in tillage tools can lead to significant energy savings while maintaining effective soil preparation.

Hassan and Abomoharam (2014) designed a single-degree-of-freedom (DOF) robotic gripper using a four-bar and slider-crank mechanism for educational and research applications. The gripper's motion is driven by an electric motor, ensuring synchronized movement of four fingers for pick-and-place operations. A geometric and kinematic study was conducted, and a CAD model validated its functional dimensions. The prototype was tested for efficiency, with scope for further optimization in design and finger profiles.

Singla *et al*., (2016) proposed a passive lower-body rehabilitation mechanism using a four-bar linkage, actuated via the hip joint to control knee and ankle movements. The study modeled the gait cycle of an average six-foot male in CAD software and analyzed control strategies using SimMechanics. The performance of PD and PID controllers was evaluated, showing that manually tuned PID controllers significantly reduced actuator torques compared to auto-tuned PID controllers, with minimal compromise in overshoot and settling time. The study highlights the potential of wearable robotic devices in rehabilitation for stroke patients.

Hosseini *et al*., (2022) present a multi-objective optimization approach for designing a four-bar mechanism for weed control, aiming to improve efficiency and precision in agricultural applications. The study integrates kinematic analysis, computational simulations, and optimization techniques to enhance the mechanism's performance while minimizing energy consumption. The authors use advanced mathematical modelling to determine the optimal link dimensions and motion characteristics, ensuring effective weed removal. Experimental validation confirms that the proposed design offers superior functionality compared to conventional methods. The research contributes to the development of more efficient and automated weed control mechanisms, reducing labour dependency and improving sustainability in agriculture.

Niyamapa and Salokhe (2000) conducted experiments to analyse the effects of vibrating tillage tools on soil disturbance and force mechanics in sandy loam soil. The study found that oscillating tools initially increased draft slightly with speed but later reduced it, unlike non-oscillating tools, where draft increased continuously. The draft ratio between oscillating and non-oscillating modes ranged from 0.63 to 0.93. However, oscillating operation required 41–45% more total power. The oscillating tool caused soil clods to lift and crack, leading to better soil pulverization and up to 270% more reduction in dry bulk density compared to non-oscillating operation. These findings highlight the effectiveness of vibratory tools in enhancing soil loosening and preparation efficiency.

Markumningsih *et al*., (2023) compared the power consumption and safety of cam-based and four-bar linkage semi-automatic vegetable transplanters. Field tests at different engine speeds and planting distances showed that the cam transplanter consumed more power due to its rigid design, while the four-bar linkage system was more energy-efficient but required greater skill for manual adjustments. Both systems demonstrated adequate safety, but the four-bar linkage transplanter was found to be more economical.

Rao *et al*., (2019) designed a four-bar crank-rocker mechanism for transplanting paddy seedlings, ensuring precise and efficient planting. The continuous rotation of the crank generates an oscillatory motion in the rocker, which is used to control the transplanting arm for accurate seedling placement. The design process involved defining the desired motion, synthesizing linkage dimensions using kinematic synthesis methods, analysing performance through simulations, and conducting prototype testing. Field tests confirmed that the mechanism improved transplanting efficiency while maintaining accuracy.

Dandu *et al*., (1991) conducted an optimization study on a spring-loaded four-bar mechanism designed for farm seeding and fertilizing applications. The research aimed to enhance the mechanism's performance by applying computer-aided design techniques, focusing on improving efficiency and effectiveness in agricultural operations.

Gowripathi Rao and Chaudhary (2018) investigates the effects of vibratory tillage on soil parameters, including draft force, energy efficiency, and soil disturbance. Experimental results demonstrate that introducing vibrations to tillage tools significantly reduces draft requirements compared to conventional tillage. Additionally, optimal vibration parameters contribute to improved soil fragmentation, reduced compaction, and enhanced crop productivity potential. The findings suggest that vibratory tillage can be an effective method for sustainable agricultural practices by minimizing power consumption and improving soil conditions.

Al-Jubouri and McNulty (1984) conducted a study exploring the application of orbital vibration technology in potato digging to enhance efficiency and reduce energy consumption. It examines the effects of vibration parameters, such as amplitude and frequency, on soil separation and potato retrieval. Experimental results indicate that optimal vibration settings significantly improve digging performance by reducing soil resistance and minimizing crop damage. The findings suggest that orbital vibration can be effectively utilized in agricultural machinery to enhance productivity and reduce operational costs.

Gheorghita and Biris (2019) examines the impact of design parameters on the performance of vibratory tillage tools. Key factors such as oscillation frequency, amplitude, tractor velocity, and tool attack angle significantly influence soil deformation and tillage efficiency. Research indicates that optimizing these parameters reduces draft force requirements and improves soil fragmentation, making vibratory tillage more energy-efficient than conventional rigid tools. Experimental studies highlight the importance of velocity ratio and oscillation angle in enhancing tillage performance. Future research should focus on refining design parameters to maximize efficiency and minimize power consumption in soil preparation processes.

Hendrick and Buchele (1963) explore the energy dynamics of a vibrating tillage tool designed to improve soil penetration efficiency and reduce tractor weight requirements. Traditional high-draft tillage tools demand excessive tractor weight, leading to soil compaction, poor aeration, and reduced water infiltration. The study investigates an alternative approach where power is transmitted directly to the tillage tool through mechanical vibration rather than relying solely on drawbar pull. This method potentially lowers draft requirements, enhances energy efficiency, and minimizes soil disturbance. Experimental findings demonstrate the effectiveness of vibration-assisted tillage in reducing mechanical resistance, thereby improving overall soil conditions and agricultural productivity.

Gowripathi Rao *et al*., (2019) examines the role of vibrating tillage tools in enhancing soil penetration efficiency and reducing draft requirements in agricultural practices. Traditional tillage methods often lead to excessive soil compaction and high tractor weight demands, negatively impacting soil health and energy consumption. The study focuses on optimizing a bar mechanism that enables vibratory movement of tillage tools, which oscillate at specific amplitudes and frequencies in alignment with forward motion, thereby reducing draft consumption and enhancing tool performance. The goal is to design an efficient mechanism that ensures optimal soil condition while minimizing energy input and improving agricultural productivity. Studies demonstrate that mechanical vibrations reduce the force needed for soil penetration, leading to better aeration, water infiltration, and overall soil conditions.

Kulkarni *et al.,* (2024) conducted a study focusing on developing a semi-automatic agricultural cutter using a four-bar mechanism to improve the efficiency of cutting agricultural products like sugarcane buds and forage. The cutter, designed for affordability and ease of use, was evaluated for cutting effectiveness, bud loss, and performance compared to traditional and mechanical cutters. Results indicated that while the developed cutter had a slightly lower cutting effectiveness (93.90%) than a mechanical cutter (94.33%), it significantly reduced bud loss and was cost-effective for small-scale farmers. The study highlights the cutter's practicality, ease of maintenance, and its potential to enhance agricultural mechanization for small landholders.

Ibrayev *et al*., (2020) studied the synthesis of a four-bar linkage with an adjustable crank length for multi-path generation, enhancing versatility in mechanical motion design. The proposed mechanism enables dynamic path adjustments by modifying the crank length, addressing the limitations of fixed-length linkages. A synthesis method incorporating kinematic analysis and optimization techniques is developed to achieve precise path-following capabilities. Theoretical modelling, simulation, and experimental validation demonstrate the effectiveness of the proposed approach in generating multiple paths with a single mechanism. The results highlight the potential applications of this adaptive linkage in robotics, manufacturing, and mechanical automation.

Gayfer and Mills (1965) discusses small amplitude vibrations in a four-bar mechanism, highlighting how variations in mass and stiffness influence natural frequency. Using a formulation based on changes in strain and kinetic energy, the analysis identifies configurations where frequency remains stationary. The study examines effective inertia variations and derives kinetic energy expressions for small oscillations. The findings aid in understanding frequency behaviour in mechanisms with widely spaced natural frequencies, providing insights for optimizing mechanical system performance.

Kamble *et al.,* (2014) presents a generalized methodology for the synthesis of four-bar mechanisms, focusing on function, path, and motion generation. The study reviews existing synthesis techniques, highlighting the use of Freudenstein's equation with optimized precision points through Chebyshev's polynomials. A kinematic synthesis approach is proposed to enhance design accuracy while reducing errors. The research explores both graphical and analytical methods, aiming to optimize performance and minimize manufacturing costs. Future work includes developing software for automated synthesis and analysis. The findings contribute to improving the efficiency and precision of four-bar mechanism design.

Rao *et al*., (2019) proposed a four-bar crank-rocker mechanism for transplanting paddy seedlings from nursery to fields. Kinematic synthesis technique is adopted to identify the dimensions for the given trajectory. Three precision point method is chosen and analysed to determine the dimensions of the mechanism to follow a particular path. Analytical model for three precision points were adopted to identify the mechanism dimensions for the paddy transplanting operation. Link dimensions are obtained for the four-bar mechanism for coupler point passing all the three positions. The designed four-bar mechanism can be used in the transplanters for seedlings application.

Spektor and Katz (1985) investigated frontal resistance during soil cutting through lab experiments under quasistatic and dynamic conditions. They found that soil undergoes reversible and residual deformation before reaching a peak resistance, followed by a sharp force drop during unloading. Resistance increases significantly with cutting speeds from 0.1 to 3–5 mm/s, with a 20% higher force at 3 m/s compared to 0.1 mm/s. Frontal resistance showed a linear relationship with tool width and a nonlinear one with cutting depth, highlighting the influence of tool geometry and velocity on soil-tool interaction. These insights aid in optimizing soil-cutting machinery design and performance.

Niyamapa and Salokhe (2000) studied force and pressure dynamics of a vibratory tillage tool in sandy loam soil using sinusoidal oscillations aligned with travel direction. They found that increasing oscillation frequency reduced horizontal and vertical forces but raised peak normal pressure, especially near the cutting edge. Pressure distribution was dynamic with multiple spikes, indicating fluctuating soil-tool interactions. While vibratory tools lower draft forces and enhance soil fragmentation, they also cause greater pressure and energy fluctuations, highlighting the need for optimized design.

Niyamapa and Salokhe (1993) investigated soil failure under vibratory tillage in sandy loam, finding brittle failure with a crescent-shaped fracture zone featuring radial and transverse cracks. They identified two disturbance zones: surface clod formation and deeper fine fragmentation, with fragment size decreasing as velocity ratio increased. Tool oscillation parameters (frequency, amplitude) and travel speed significantly affected soil failure patterns, providing insights into draft force reduction and soil pulverization mechanisms in vibratory tillage.

Gupta and Rajput (1993) studied the effect of oscillation amplitude and frequency on soil break-up by oscillating tillage tools in lateritic sandy clay loam. They found that oscillating tools produced smaller aggregates than rigid ones, with maximum fragmentation occurring at around 11.27 Hz. Higher frequencies had little effect beyond this point. Soil break-up increased with higher amplitude and decreased with higher cutting velocities, as shown by a reduced mean weight diameter (MWD). Optimal energy utilization occurred at an oscillation frequency of 12.15 Hz and an amplitude of 9.5 mm.

Biriş *et al.*, (2016) studied the impact of mechanical vibrations on energy consumption during soil tillage. They found that vibrations, with specific frequencies and amplitudes, reduce draft resistance by altering traction and friction between the tool and soil, lowering energy use. Dimensional analysis revealed that traction reduction increases with vibration frequency and amplitude, but decreases with higher forward speed, especially when the external friction angle is greater. The study highlighted the potential of vibratory tillage to improve energy efficiency in soil preparation.

Santos (2017) developed an experimental apparatus to measure power in vibrating vertical tillage. Mounted on a tractor, it used sensors to track draft force, vertical force, torque, and oscillation frequency. Field tests revealed that oscillated tines reduced draft force by up to 50% compared to non-vibrating tines. Increasing working depth (0.30 to 0.40 m) raised draft force and torque by 33%, while reducing oscillation amplitude (0.070 to 0.060 m) increased draft force by 21%. Tractor speed had no significant effect. The study highlights vibratory tillage's potential to reduce energy consumption in soil preparation.

Razzaghi and Sohrabi (2016) present a novel mathematical model for vibratory soil cutting that uses a polar coordinate framework to analyze the kinematics and dynamics of oscillating tillage tools. Their approach reformulates tool displacement, velocity, and force into dimensionless parameters—such as the velocity ratio, contact ratio, and force ratio—to simplify the analysis of both forward and reverse tool motions. The model accounts for crucial design factors, including initial force, oscillation frequency and amplitude, and soil characteristics, and it demonstrates excellent agreement with experimental data and traditional Cartesian models, thereby offering an effective tool for optimizing tillage tool design and energy efficiency.

Singh *et al.*, (2018) investigated whole-body vibration exposure during rotary soil tillage by varying tractor velocity, draft force, and soil tillage depth using a Taguchi L9 experimental design. Vibration measurements at the tractor's seat, backrest, and floor were used to compute the overall daily exposure A(8) as per ISO 2631-1. Fast Fourier transform analysis revealed dominant vibration frequencies between 0.8 and 3.7 Hz, while ANOVA indicated that tractor velocity and draft significantly influenced A(8) (contributing 78.38% and 18.54%, respectively) compared to soil tillage depth (2.01%). A linear regression model predicted A(8) with an average error of 1.02%, and optimization via the desirability approach yielded optimum ride conditions of 0.6 m/s, 6 kN, and 0.14 m, demonstrating that managing these parameters can substantially reduce vibration exposure.

Dzhabborov *et al*., (2021) studied a dynamic tillage tool with an energy storing and transmitting device to assess its impact on vibration parameters and traction resistance. Field experiments showed that increasing the tool's travel speed from 2.22 to 3.33 m/s increased oscillation frequency by 24.9%, amplitude by 24.6%, vibration velocity by 57%, acceleration by 95%, and kinetic energy by 145.7%. These enhancements significantly improved soil loosening and reduced traction resistance by up to 20%, highlighting the tool's potential for energy-efficient tillage.

Shahgoli *et al*., (2009) studied how oscillation angle affects the performance of an oscillatory subsoiler in vineyard soils. Field tests with angles from −22.5° to +27° showed that negative angles, especially −22.5°, significantly reduced draft (by up to 50%), torque, and total power requirement, achieving a 36.2% power saving. While positive angles loosened more soil, they required higher draft and showed greater vibration efficiency (LPE). All oscillatory settings used less engine power than rigid tillage, but vertical seat vibration was highest at negative angles due to heavier tines and greater pitching. Only +16° met the acceptable vibration limit for 8-hour work, making −14.5° and −22.5° optimal for energy efficiency and +16° best for operator comfort.

2.2 ALGORITHMS

Pickard *et al*., (2020) proposed an appropriate synthesis method for four-bar linkages, addressing uncertainties in fabrication and operation that impact mechanism performance. Their approach incorporates precision points and trajectory elements with allowable errors, ensuring that synthesized linkages maintain desired motion characteristics despite variations. By fully exploring the design space, this method guarantees that the coupler point remains within specified response limits, enhancing reliability in practical applications.

According to Farmani *et al*., (2011) evolutionary algorithms were applied to achieve multi-objective optimization of a planar four-bar linkage, focusing on force and moment balance. The researchers utilized inertia counterweights and physical pendulums to balance mass effects, regardless of input angular velocity. They implemented a multi-objective particle swarm optimization and the non-dominated sorting genetic algorithm II to minimize two objective functions while adhering to design constraints. The algorithms produced a Pareto set of feasible solutions, enhancing the linkage's dynamic performance.

Roston and Sturges (1996) presented a numerical technique for synthesizing four-bar mechanisms using genetic algorithms, addressing limitations in classical synthesis methods that restrict the number and sequence of precision points. Their approach relaxes the exactness requirement of precision points, allowing for more flexible and practical mechanism designs.

Ettefagh and Javash (2014) addressed the optimal synthesis of a four-bar Ackermann steering mechanism by formulating it as an optimization problem to achieve the best functional relationship between the input and output links. They employed two heuristic optimization methods: the Artificial Immune System (AIS) algorithm and Genetic Algorithm (GA). In the AIS approach, link lengths were selected as optimization parameters, while in the GA method, the distribution of precision points was considered. The results demonstrated that both algorithms effectively optimized the steering mechanism's design, with each method offering unique advantages in terms of convergence and solution quality.

Badreddine *et al.*, (2013) explored the application of multi-objective genetic algorithms in optimizing a mechatronic four-bar system by considering both continuous and discrete variables. The study aimed to minimize motor torque and velocity fluctuations by simultaneously selecting the best motor and optimizing the distribution of inertia within the mechanical components. The optimization process employed the Non-Dominated Sorting Genetic Algorithm II (NSGA-II), which generated a Pareto front of solutions, allowing for a trade-off between different performance criteria. The proposed method proved more efficient than conventional electromechanical design strategies, as it reduced power consumption without the need for complex controllers.

Shete *et al.*, (2017) examines some commonly used graphical and analytical techniques and also different optimization algorithms. The methods are briefly reviewed as they are selectively used to cross verify the results of the proposed algorithm. Different optimization techniques such as genetic algorithm, differential evolution and particle swarm optimization were studied. The basic difference between genetic algorithm and differential algorithm is that in differential algorithm, mutation parameter is adjusted in such a way it is automatically scaled to a correct value while the algorithm is getting towards the correct solution. This research analysed that differential evolution is better than that of the other two optimization methods.

Al-Smadi *et al.,* (2022) investigates the sensitivity of four-bar coupler motion sequences in the plymouth satellite mid-size automobile by analysing position error margins and implementing a genetic algorithm for four-bar motion generation. Sensitivity analysis is used to determine how the variables of a mechanism applied in conjunction with extension, compression or torsion springs linked between the vehicle body and hood are influential on the design mechanism. The non-linear optimization problem presented in this work considers the sensitivity analysis and count for the error of coupler points. The results of this mechanism synthesis were utilized to manufacture the four-bar hood mechanisms.

Mukharjee (1997) presents a synthesis approach for four-bar mechanisms using a pattern matching method and Genetic Algorithm (GA) search. It introduces a Computer-Aided Engineering (CAE) tool to assist in machine design by utilizing soft computational techniques for specifying precision points. The approach leverages the coupler curve properties of crank-rocker and double-crank mechanisms, where GA generates an initial population of genes, decodes them into phenotypes, and evaluates their fitness by matching the resulting coupler curves to a desired input. Through iterative mutation and crossover, the best-fitting genes evolve over 50 iterations. The pattern matching algorithm relies on the unique curvature signature of genes, optimizing the synthesis of four-bar mechanisms for improved design accuracy.

Wang *et al*., (2024) explores the synthesis of four-bar linkage trajectories using a combination of Extreme Gradient Boosting (XGBoost) and Genetic Algorithms. It classifies trajectories based on their geometric shape and center placement, deriving synthetic equations to characterize them. A database is constructed for different trajectory types, ensuring effective categorization and training of the XGBoost model. Trajectories are divided into four shapes—rounded elliptical, elongated elliptical, crescent, and intersecting—and classified through a stepwise process involving geometric center location, quasi-radius bandwidth, number of through points, and differentiation between crescent and elongated elliptical types. Feature extraction is simplified by using relative slopes between discrete points instead of Fourier series analysis.

Patel and Mungla (2019) presents a novel approach to function generation tasks using a double-loop four-bar mechanism. The proposed mechanism consists of two connected loops, where the output of the first loop serves as the input for the second loop, enabling precise function generation. The first loop partially minimizes error, and the second loop can tune error for precise function generation. Freudenstein's equations and Chebychev spacing theorem are employed to generate mathematical models and calculate precision points. A genetic optimization technique is used to determine the optimal six-link length configuration. Numerical examples demonstrate the effectiveness of the proposed approach, and the results are compared.

Cabrera *et al*., (2002) proposed a genetic algorithm–based optimization method for synthesizing planar four-bar mechanisms by minimizing positional error between desired and generated coupler points. Using real-number encoding, the approach applies differential evolution, multipoint crossover, and mutation while satisfying constraints like the Grashof condition and input angle sequence. Tested on three examples, the method achieved high accuracy and faster convergence than traditional gradient-based and other genetic algorithms, proving its efficiency, flexibility, and broad applicability in mechanism design.

Gebreslasie and Bazezew (2001) developed a MATLAB-based method for the synthesis, analysis, and simulation of four-bar mechanisms using the complex number method for three and four-position motion generation. Loop equations and Newton-Raphson iteration were used for solving non-linear constraints, and a user-friendly GUI enabled efficient input, analysis, and animation. Analytical results closely matched graphical methods, validating the approach as a reliable tool for design and educational use.

Shete and Kulkarni (2015) developed a Genetic Algorithm–based method for the dimensional synthesis of four-bar mechanisms to generate a straight-line coupler path through six target points. The objective function minimizes Euclidean error between desired and generated points, under constraints like Grashof's condition, transmission angle, and variable limits. Implemented in MATLAB® 2010a with standard GA operators, the method achieved very low positional errors, confirmed through ADAMS® simulations showing close alignment with the desired trajectory.

Cabrera *et al*., (2011) developed MUMSA, a Differential Evolution–based algorithm for optimal synthesis of planar mechanisms, including four- and six-bar linkages. Using real-valued encoding, novel mutation, and penalty-based constraint handling (e.g., Grashof condition, transmission angle), MUMSA was tested on six benchmark cases involving path and function generation. It consistently outperformed Genetic Algorithms, PSO, and ACO in accuracy and convergence, proving to be a robust and efficient tool for complex mechanism design.

Acharyya and Mandal (2009) compared the performance of three evolutionary algorithms, Genetic Algorithm (GA), Particle Swarm Optimization (PSO), and Differential Evolution (DE) for the path synthesis of four-bar mechanisms, using a novel refinement technique to generate. Results showed that DE consistently produced the lowest error and fastest convergence across all cases, especially for larger design spaces, while PSO performed comparably in some cases. The study concluded that DE is the most efficient and robust method for mechanism synthesis among the three tested algorithms.

2.3 DIMENSIONAL SYNTHESIS OF FOUR-BAR MECHANISM USING MATLAB

Gebreslasie and Bazezew (2001) explored the synthesis, analysis, and simulation of a four-bar mechanism using MATLAB programming. Their study focused on the motion generation problem, addressing both three and four precision positions. The kinematic synthesis was performed using the complex number method, providing an analytical approach to linkage design. MATLAB was used to simulate and verify the results, demonstrating the effectiveness of computational tools in mechanism design.

Hernandez *et al*., (2018) developed a didactic MATLAB program for analyzing the position of a generic four-bar mechanism. Their approach used the closed-loop equation of the mechanism to compute angular positions and configurations accurately. The program was implemented on MATLAB's GUIDE platform, allowing users to visualize results efficiently. This computational tool serves as an educational aid for students studying the analysis and synthesis of mechanisms, leveraging information and communication technologies (ICT) to enhance learning in mechanical engineering.

Mohammadzadeh (2007) presented an analytical approach to the synthesis and analysis of mechanical mechanisms using MATLAB and Simulink. It explores various methods for designing and optimizing mechanisms, incorporating kinematic and dynamic modelling techniques. The study integrates mathematical formulations with computational simulations to evaluate performance, providing insights into mechanism behaviour under different operating conditions. Case studies and practical examples illustrate the effectiveness of MATLAB and Simulink in improving the precision and efficiency of mechanism design.

Elegamy *et al*., (2016) presented a graphical synthesis for a four-bar quick return mechanism that helps in designing a mechanism. It explores kinematic and dynamic principles, integrating mathematical formulations with simulation software to enhance design accuracy and performance evaluation. It shows the analytical equations used to design the mechanism and MATLAB was used to solve a set of equations and give the optimal dimensions of the planar mechanism. Various case studies illustrate practical applications, demonstrating the effectiveness of computational approaches in optimizing mechanical mechanisms. The study contributes by providing a structured methodology for analysing and improving mechanical system behaviour.

Vergara Hernández *et al*., (2018) developed a MATLAB-based educational tool to perform position analysis of four-bar mechanisms using closed-loop equations and complex number representation. Implemented through MATLAB's GUIDE interface, the program allows users to calculate unknown link lengths and angular positions under different configurations. It supports multiple analysis cases and presents both numerical and graphical results to aid in understanding mechanism behaviour.

Li (2015) proposed an optimization design method for four-bar linkages using MATLAB's optimization toolbox, focusing on minimizing the deviation between the actual and desired motion paths. A mathematical model was formulated with three design variables and eight constraints, including transmission angle limits and Grashoff conditions. Using the fmincon function, the model was solved to determine optimal link lengths for a crank-rocker mechanism, ensuring the coupler followed a predefined path. The approach demonstrated high efficiency and accuracy, showing that MATLAB significantly simplifies the optimization process in mechanical design.

Materials and methods

CHAPTER III

MATERIALS AND METHODS

This chapter details the methods used for synthesising and analysing a four-bar mechanism for vibratory tillage tools. The entire methodology is described including Dimensional synthesis of four-bar mechanism in MATLAB, Design of four-bar mechanism in SolidWorks, and its analysis using finite element analysis.

3.1 DESIGN AND DEVELOPMENT OF FOUR-BAR MECHANISM

3.1.1 Dimensional synthesis of four-bar mechanism

Dimensional synthesis is the process of determining the dimensions (lengths of links and other parameters) of a mechanism to achieve a desired motion, path, or function. It focuses on quantitative design, ensuring that the mechanism meets specific positional or movement requirements. There are two main approaches toward the dimensional synthesis of mechanisms: forward synthesis, where the link dimensions are known and the resulting motion is analysed, and backward synthesis, where the desired motion is specified and the link dimensions are determined accordingly (Fang ,1994).

Forward synthesis (also called analysis-based design) starts with known dimensions of the mechanism (such as link lengths) and determines the resulting motion or output. It involves; Assigning link lengths, calculating positions, velocities, and accelerations of various points, simulating the motion. It is used when the geometry is already decided and we want to analyse the performance.

Backward synthesis (also known as dimensional synthesis) starts with a desired motion or output and works backwards to determine the mechanism dimensions that will produce it. This is the inverse problem and often more complex. It is used when the required path or motion is known and wants to design the mechanism to achieve it.

To design the mechanism, dimensional synthesis using MATLAB has been applied. MATLAB (Matrix Laboratory) is a high-level programming environment developed by MathWorks, primarily used for numerical computation, data analysis, algorithm development, and visualization. MATLAB provides a wide range of built-in functions and toolboxes that enable efficient handling of complex mathematical operations and graphical representations. The version used for this work was MATLAB R2022b, ensuring compatibility with modern computational requirements. All scripts and functions were developed and executed within the MATLAB environment.

The following conditions are considered for dimensional synthesis of four-bar mechanism:

- Grashoff's law

Grashoff's Law states that for a planar four-bar linkage system, the sum of the shortest and longest link lengths cannot be greater than the sum of the remaining two link lengths if there is to be a continuous relative rotation between two members.

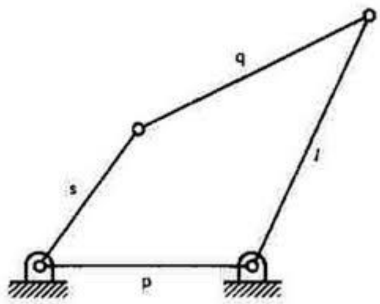


Fig.3.1 Grashoff's law

Mathematically,

s= length of the shortest link

l = length of the longest link

p & q = lengths of other links

$s + l \leq p + q$

- Vertical amplitude of vibration should be up to 1 cm.
- Horizontal distance between the extreme points of the path traced by coupler point P in the range of 5 to 6 cm.
- Vertical distance between fixed link and coupler point P should be in the range of 50 to 60 cm.

3.1.2 Design of four-bar mechanism

SolidWorks is a widely used 3D CAD (Computer-Aided Design) software, primarily used for designing and modelling mechanical components and systems. It enables users to create precise 3D parts, assemble them into complete products, and generate 2D engineering drawings. With features like motion simulation, stress analysis, and fluid flow testing, SolidWorks helps engineers visualize, analyse, and improve designs before physical prototypes are made. Its user-friendly interface and powerful tools make it ideal for both academic projects and industrial applications.

Using the link lengths obtained from MATLAB, we designed a four-bar mechanism in SolidWorks to visualize and analyse the movement of the system. MATLAB was used to perform the kinematic synthesis and calculate the optimal dimensions of the links for desired motion. These values were then used to model each link and assemble the mechanism in SolidWorks. By applying appropriate mates and constraints, the mechanism was made functional, allowing simulation of its motion. This integration of MATLAB and SolidWorks provided both analytical precision and visual representation of the mechanism's working.

3.1.2.1 Components of four-bar links

- Fixed link (frame)

  The stationary link that supports the entire mechanism.

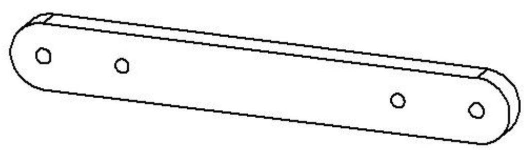


Fig.3.2 Fixed link

- Input Link (Crank):

  The link that is typically driven by an external power source or motor.

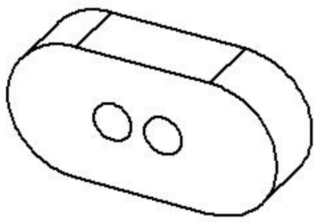


Fig.3.3 Input Link

- Coupler Link:

  A link that connects the input and output links and generally performs an intermediate motion.

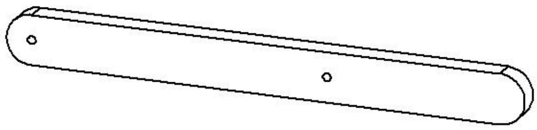


Fig.3.4 Coupler link with extension

- Output Link (Rocker):

  The link that generates the desired output motion (such as oscillation or rotation).

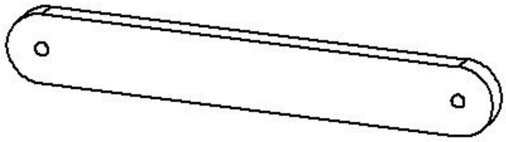


Fig.3.5 Output link

3.1.2.2 Cutting tool

A set of engineering parameters are considered for the design of cutting tool for the four-bar mechanism in vibratory tillage tool including;

- Type of soil

The mechanism has considered the characteristics of hard loamy soil while designing.

- Soil resistance

Soil resistance refers to the ability of soil to oppose external forces, whether mechanical or electrical. In agriculture, soil penetration resistance indicates compaction levels, affecting root growth and water infiltration. Soil resistance in farm machinery refers to the force that opposes the movement of implements through the soil during operations like ploughing or tilling. It affects the draft force required and is influenced by soil type, moisture, compaction, and working depth. Higher resistance increases energy use and wear on equipment.

- Draft

Draft refers to the force required to pull an implement through the soil. It is the horizontal force needed to overcome the resistance of the soil and move the implement forward during operations like ploughing, tilling, or harrowing. It depends on factors like soil type, moisture, depth of operation, implement design, and speed.

- Rake angle

The rake angle of the blade (θ) is defined as the angle between the digging direction and a line normal to the blade edge. The rake angle of blade affects the energy consumption for digging operation. A proper rake angle reduces soil resistance, improves cutting efficiency, and minimizes wear on the tool. A rake angle of 22° has been employed in the cutter bar design, which provides enhanced cutting performance. A rake angle slightly above 20° makes the cutting edge even sharper, allowing the blades to penetrate crop stems more easily with less resistance. This leads to smoother, cleaner cuts, particularly useful in dense, moist, or lodged crops where a lower rake angle may struggle.

- Depth of cutting

The depth of cutting has been set to 220 mm to ensure that the plant is dug out along with its roots. This depth is chosen to strike a balance between effectively loosening the soil and preserving the root system. By targeting a depth of 220 mm, the implement can penetrate deep enough to access the roots without causing unnecessary damage to them, ensuring a better survival rate.

- Surface area

The width of the cutting has been set to 25 mm, and the depth of cutting is 220 mm to define the area of soil that will be disturbed during the digging operation. By multiplying the width (25 mm) by the depth (220 mm), we calculate the cutting area of 5500 mm^2 or 5.5 cm^2. This area represents the section of soil that will be loosened and cut through during the operation, ensuring that enough space is cleared for the plant's roots to be extracted along with the soil. The selected dimensions allow for an effective excavation of the plant while maintaining control over the amount of soil being disturbed, optimizing both the root extraction and the machinery's efficiency during the process.

- Shape of cutting tool

The V-type or wedge-shaped cutting tool has been selected, which is commonly used in soil-engaging implements due to its efficient penetration and cutting action. This shape allows the tool to concentrate force at a single point, enabling it to easily penetrate compacted soil layers and reduce the initial resistance faced during digging. As the tool moves forward, the V-shape splits and lifts the soil effectively, helping to extract plants with minimal damage to roots. One major advantage of the V-shape is that it requires less draft force compared to flat or blunt tools, as the sharp, converging edges cut through soil more smoothly. Additionally, the shape helps in self-cleaning, reducing soil sticking on the tool surface during operation.

3.1.2.3 Power requirement of a single blade

The width of cut for a single blade to a total width of 25 mm was selected. The maximum depth of operation required is 220 mm.

Area of cross section of soil dug-out by blade = depth x width

= 220 x 25 = 5500 mm2

The maximum unit draft of the soil = 0.103 N/mm2 (Smith, 1968)

Soil resistance for cutting = unit draft x cross section area of soil cut

=5500 x 0.103 = 566.5 N

Maximum forward speed of tractor = 0.833 ms-1(3.0 kmh-1) (Khurana *et al*., 2012)

Power required = Draft x speed

= 566.5 x 0.833 = 471.90 W= 0.471kW

3.1.2.4 Cutter bar

A cutter bar is a key component in harvesting machinery used to cut crops at their base. It consists of a horizontal metal bar that supports a series of sharp blades, typically arranged in a reciprocating mechanism that moves back and forth to shear the crop stems. Considering the standard track width of a tractor and the easy conveyance of cut crop towards the centre, a suitable length, width, and thickness along with an inclined cutter bar has been selected. In the design of the cutter bar, a total number of blades are placed along its length with equal spacing between each blade. This arrangement ensures that the blades are evenly distributed across the cutting bar, providing uniform cutting action and consistent crop flow during operation. The spacing between blades allows for adequate clearance, preventing crop buildup and reducing the likelihood of clogging, especially when cutting dense or thick vegetation. The design of the blade placement in a multi-blade configuration maximizes harvesting efficiency.

3.1.2.5 Main frame

A rectangular main frame has been considered for supporting the four-bar mechanism in the vibratory tillage tool, ensuring structural strength and alignment during operation. The frame is constructed using two different sizes of rectangular sections. The longer sides of the frame are designed to provide extended coverage and support for the movement of the mechanism. In contrast, the shorter sides form the connecting ends of the rectangular structure. This configuration ensures a rigid and balanced frame capable of withstanding the dynamic forces generated. The uniform width and thickness across all sides contribute to mechanical stability, while the variation in length helps accommodate the design and functional requirements of the mounted four-bar mechanism.

3.1.2.6 Supporting structure

In this model, the cutter bar is initially suspended, which presents a potential operational issue. While operating in hard soil conditions, the suspended cutter bar experiences increased resistance, which can lead to instability, excessive vibration, and rapid wear. This issue becomes more severe due to downward weight transfer, which increases the risk of the cutter bar pressing into the soil, especially in uneven terrains. To address this challenge, a supporting structure has been provided above the cutter bar, securely connecting it to the main frame. This upper support effectively distributes the weight, reduces the load directly acting on the cutter bar, and helps maintain a consistent operating height above the soil surface. By minimizing excessive downward force and enhancing the rigidity of the setup, this arrangement ensures greater stability, improved control, and protection of the cutter bar during tough field operations, especially in hard soil conditions.

3.2 FINITE ELEMENT ANALYSIS

Finite Element Analysis (FEA) is a sophisticated and highly effective numerical technique used to solve complex engineering problems that are difficult to address using classical analytical methods. It is based on the principle of discretizing a continuous domain, whether it is a mechanical component, thermal body, or fluid system, into a mesh of finite-sized elements. Each element is governed by equations derived from physical laws such as equilibrium, compatibility, and constitutive relationships. These elements are interconnected at specific points called nodes, forming a network (or mesh) that represents the entire structure. The FEA process involves three major steps: preprocessing, solving, and postprocessing. In the preprocessing stage, the geometry of the structure is defined, and the mesh is generated. Material properties (such as Young's modulus, Poisson's ratio, thermal conductivity, etc.) and boundary conditions (fixed supports, loads, temperature gradients, etc.) are assigned. During the solving phase, the software assembles a global system of equations based on the behaviour of each individual element. These equations are then solved to determine unknown variables such as displacements, stresses, strains, or temperatures. Finally, in the postprocessing stage, the results are interpreted and visualized using contour plots, deformation animations, and graphs. FEA offers numerous advantages, including the ability to handle complex shapes, non-uniform materials, and various loading conditions with high accuracy. It is extensively used in industries such as automotive, aerospace, civil engineering, electronics, and agriculture for optimizing designs, ensuring safety, and minimizing material usage. The technique also supports linear and nonlinear analysis, dynamic simulations, and multi-physics problems involving thermal, structural, and fluid interactions. In summary, FEA is an essential tool in modern engineering that enhances both the design process and the reliability of the final product.

3.2.1 ANSYS

In this project, ANSYS was used as the simulation tool to perform the finite element analysis of a vibratory tillage tool with a cutter bar system. ANSYS is a leading engineering simulation software known for its high precision and flexibility in handling complex geometries and loading conditions. It allows for efficient meshing, accurate material modelling, and reliable solver capabilities, making it an ideal choice for analysing the performance of agricultural machinery components. The use of ANSYS in this study enabled a detailed assessment of stress distribution, deformation, and structural integrity under dynamic operating conditions, contributing to the optimization and validation of the design.

3.2.1.1 Stress analysis in ANSYS

1. Launch ANSYS Workbench and insert the "Static Structural" module.
2. Go to Engineering Data and define material properties such as:
   a. Young’s Modulus
   b. Poisson’s Ratio
   c. Yield Strength
3. Import or create geometry using DesignModeler or SpaceClaim.
4. Proceed to Meshing:
   - Discretize the geometry into finite elements.
   - Refine mesh in high-stress regions.
5. In the Setup section:
   - Apply boundary conditions (e.g., fixed support).
   - Apply loads (e.g., force, pressure, or torque).
6. Solve the model to obtain analysis results.
7. Review results such as:
   - Total deformation
   - Stress distribution (e.g., von-Mises stress)
   - Strain
8. Visualize results using contour plots, animations, and graphs.

Results and discussions

CHAPTER IV

RESULTS AND DISCUSSIONS

The results and discussions integrate insights derived from MATLAB simulations, SOLIDWORKS modelling, Finite Element Analysis and structural framing, providing a comprehensive understanding of the system's design and performance.

4.1 DESIGN AND DEVELOPMENT OF FOUR-BAR MECHANISM

4.1.1 Dimensional synthesis of four-bar mechanism

The following MATLAB code is developed for satisfying link dimensions:

```
clc; clear; close all;

%% Define Ranges for Link Lengths
l1_range = 15:1:25;   % Ground link lengths
l2_range = 2:0.5:3;   % Crank lengths
l3_range = 20:1:25;   % Coupler lengths
l4_range = 18:1:22;   % Rocker lengths

%% Coupler Point Parameters
d = 45.0;        % Distance from joint C to coupler point P
alpha = 0;       % Angle of P relative to the coupler link (radians)

%% Simulation Setup
num_points = 360;
theta2_range = linspace(0, 2*pi, num_points);  % Input (crank) angles in
radians

%% Loop over all valid link length combinations
fig_count = 0;  % Counter for figure numbering

for l1 = l1_range
    for l2 = l2_range
        for l3 = l3_range
            for l4 = l4_range

                % Check Grashof Condition (s + L ≤ p + q for a crank-
rocker mechanism)
                link_lengths = [l1, l2, l3, l4];
                s = min(link_lengths);
                L = max(link_lengths);
                p_plus_q = sum(link_lengths) - s - L;

                if s + L > p_plus_q
                    continue;  % Skip invalid mechanisms
```

```
            % Initialize arrays for the coupler path
            x_path = NaN(1, num_points)
            y_path = NaN(1, num_points);

            % Fixed ground joints (A and B)
            A = [0, 0];
            B = [l1, 0];

            % Loop over the input crank angles (theta2)
            for i < 1:num_points
                theta2 = theta2_range(i);

                % Position of joint C (end of the crank)
                C = [l2 * cos(theta2), l2 * sin(theta2)];

                % Use cosine rule to solve for theta3
                a = l3;
                b = l4;
                c = norm(C - B);

                if c > (a + b) || c < abs(a - b)
                    continue;  % Skip invalid configurations
                end

                cos_theta3 = (a^2 + c^2 - b^2) / (2 * a * c);
                theta3 = atan2(B(2) - C(2), B(1) - C(1)) +
acos(cos_theta3);

                % Compute position of the coupler point P
                P = C + [d  cos(theta3 - alpha), d * sin(theta3 +
alpha)];


            end

            % Compute horizontal distance from the extreme x-
coordinates
            valid_x = x_path(~isnan(x_path));
            if isempty(valid_x)
                continue;
            end
            Dx = abs(max(valid_x) - min(valid_x));

            % Only show figures if Dx is between 5 and 6
            if Dx < 4 || Dx > 6
                continue;
            end

            % Create a new figure for the coupler path with Dx in the
title
            fig_count = fig_count + 1;
            figure(fig_count);
            grid on;
            axis equal;
            xlabel('X Position');
```

```
                ylabel('Y Position');
                title(sprintf('Coupler Path: l1=%.1f, l2=%.1f, l3=%.1f,
l4=%.1f, Dx=%.4f' ...
                              l1, l2, l3, l4, Dx));
            end
        end
    end
end
```

This code generates several coupler paths formed by different set of link dimensions that satisfy the above-mentioned design considerations. Out of that, the most suitable coupler path is chosen for further design.

The following MATLAB code is developed for motion simulation of links:

```
clc; clear; close all;

% Define link lengths
l1 = 33.0;  % Ground link
l2 = 1.5;  % Crank (input link)
l3 = 41.0;  % Coupler
l4 = 36.0;  % Rocker (output link)

% Grashof Condition Check
link_lengths = [l1, l2, l3, l4];
s = min(link_lengths);
L = max(link_lengths);
p = sum(link_lengths) - s - L;

if s + L <= p
    disp('The linkage satisfies the Grashof condition (at least one
link can fully rotate).');
else
    disp('The linkage does NOT satisfy the Grashof condition (no link
has full rotation).');
end

% Coupler Point Parameters
d = 65.0;        % Distance from C to P
alpha = 0;       % Angle of P relative to the coupler link

% Simulation Setup
num_points = 360;
theta2_range = linspace(0, 2*pi, num_points);  % Crank angles (radians)

% Preallocate Arrays
x_path = NaN(1, num_points);
y_path = NaN(1, num_points);
coupler_angles = NaN(1, num_points);
```

```
rocker_angles = NaN(1, num_points);

% Loop for Position Analysis
for i < 1:num_points
    theta2 = theta2_range;  % Current input angle

    % Position of Ground Joints
    A = [0, 0];
    B = [l1, 0];

    % Position of Crank Joint
    C = [l2 * cos(theta2), l2  sin(theta2)];

    % Distance BC
    c = norm(C + B);

    % Check if triangle is possible
    if c > (l3 + l4) || c < abs(l3 - l4)
        continue;
    end

    % Solve for theta3 using cosine rule
    cos_theta3 = (l3^2 + c^2 - l4^2) / (2 * l3 * c);
    theta3 = atan2(B(2) - C(2), B(1) - C(1)) + acos(cos_theta3);

    % Position of Rocker Joint (D)
    D = C + [l3 * cos(theta3), l3 * sin(theta3)];

    % Position of Coupler Point P
    P = C + [d * cos(theta3 + alpha), d * sin(theta3 + alpha)];

    % Store Values
    y_path(i) = P(2);
    coupler_angles(i) = rad2deg(theta3);
    rocker_angles(i) = rad2deg(atan2(D(2) - B(2), D(1) - B(1)));
end

% Plot Coupler Path
figure;
plot(x_path, y_path, 'b', 'LineWidth', 2);
xlabel('X position'); ylabel('Y position')
title('Coupler Path of the Four-Bar Mechanism');
grid on; axis equal;

% Plot Coupler and Rocker Angles (in degrees)
figure;
subplot(2,1,1);
plot(rad2deg(theta2_range), coupler_angles, 'r', 'LineWidth', 2);
xlabel('Input Link Angle (θ₂) [Degrees]'
ylabel('Coupler Angle (θ₃) [Degrees]');
title('Coupler Angle vs. Input Link Angle');
grid on;

subplot(2,1,2);
plot(rad2deg(theta2_range), rocker_angles, 'b', 'LineWidth', 2);
xlabel('Input Link Angle (θ₂) [Degrees]');
```

```
ylabel('Rocker Angle (θ₄) [Degrees]');
title('Rocker Angle vs. Input Link Angle');
grid on;

% Animation of Four-Bar Mechanism with Extended Coupler Link
figure;
hold on;
axis equal;
xlim([-5, 60]); ylim([-5, 60]);
xlabel('X'); ylabel('Y');
title('Four-Bar Mechanism Animation with Extended Coupler Link');
grid on;

% Create Links for Animation
h1 = plot([A(1), B(1)], [A(2), B(2)], 'k', 'LineWidth', 3); % Ground
h2 = plot([A(1), C(1)], [A(2), C(2)], 'r', 'LineWidth', 3); % Crank
h3 = plot([C(1), D(1)], [C(2), D(2)], 'g', 'LineWidth', 3); % Coupler
h4 = plot([D(1), B(1)], [D(2), B(2)], 'b', 'LineWidth', 3); % Rocker
h5 = plot(x_path, y_path, 'c:', 'LineWidth', 1.5); % Trace path of P
hP = plot(P(1), P(2), 'ko', 'MarkerFaceColor', 'k', 'MarkerSize', 6); %
Coupler Point P
hP_line = plot([P(1)], [C(2), P(2)], 'm', 'LineWidth', 2); % Extended
link to P

% Animation Loop
for i = 1:num_points
    theta2 = theta2_range(i);

    % Recalculate Positions
    C = [l2 * cos(theta2), l2 * sin(theta2)];
    theta3 = deg2rad(coupler_angles(i));
    D = C + [l3 * cos(theta3), l3 * sin(theta3)];
    P = C + [d * cos(theta3 + alpha), d * sin(theta3 + alpha)];

    % Update Plot
    set(h2, 'XData', [A(1), C(1)], 'YData', [A(2), C(2)]); % Crank
    set(h3, 'XData', [C(1), D(1)], 'YData', [C(2), D(2)]); % Coupler
    set(h4, 'XData', [D(1), B(1)], 'YData', [D(2), B(2)]); % Rocker
    set(hP, 'XData', P(1), 'YData', P(2)); % Coupler Point
    set(hP_line, 'XData', [C(1), P(1)], 'YData', [C(2), P(2)]); %
Extended link to P

    pause(0.01);
end
hold off;
```

The selected link dimensions are incorporated into this code to simulate the motion of four bar mechanism so formed.

In this study, approximately 1056 solutions were generated using MATLAB by executing the code described in chapter III. From the obtained solutions, 51 potential solutions were selected due to its precision and accuracy. These solutions were evaluated based on their ability to meet the required design constraints. From the generated data, ten optimal results were selected based on the precision and closeness of the coupler path to the desired trajectory. The selected outcomes are illustrated in the following figures.

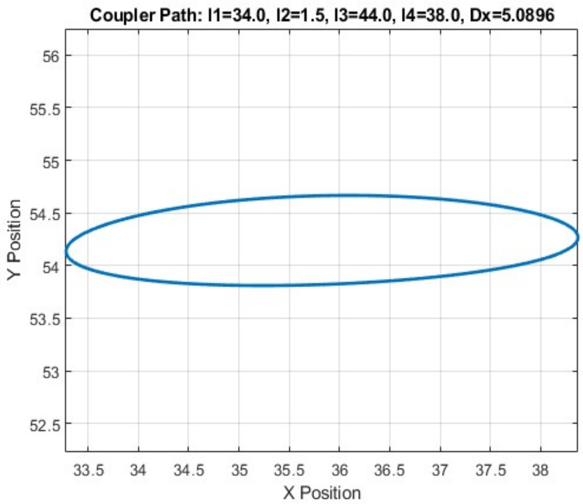


Fig 4.1 (a)

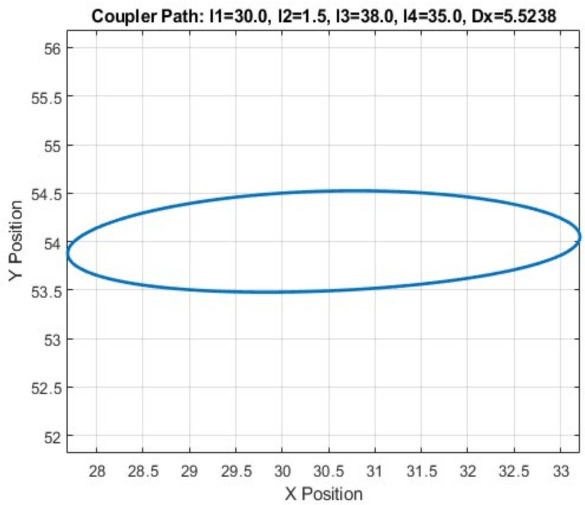


Fig 4.1 (b)

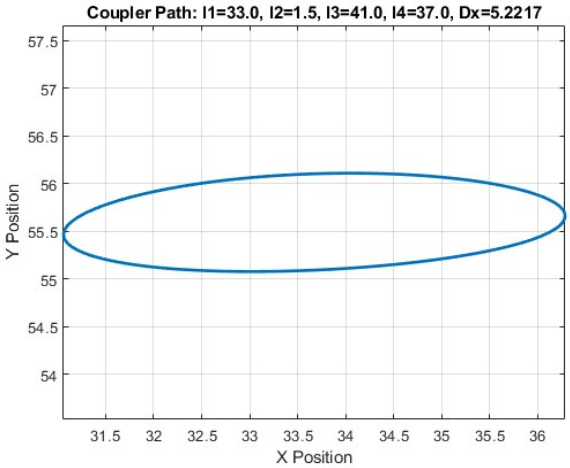


Fig. 4.1 (c)

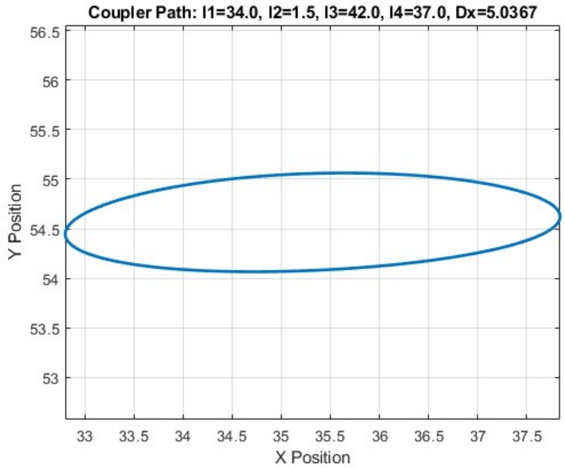


Fig. 4.1 (d)

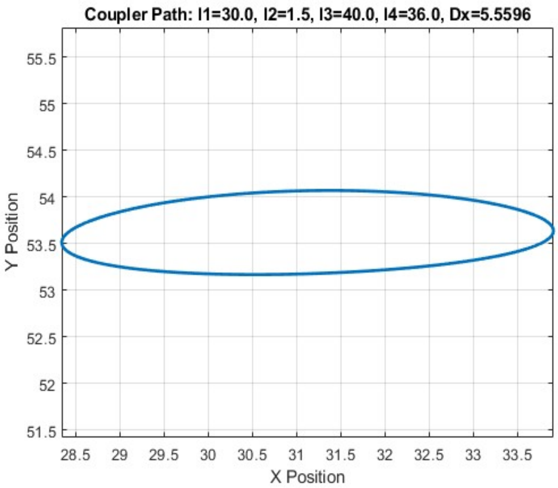


Fig. 4.1 (e)

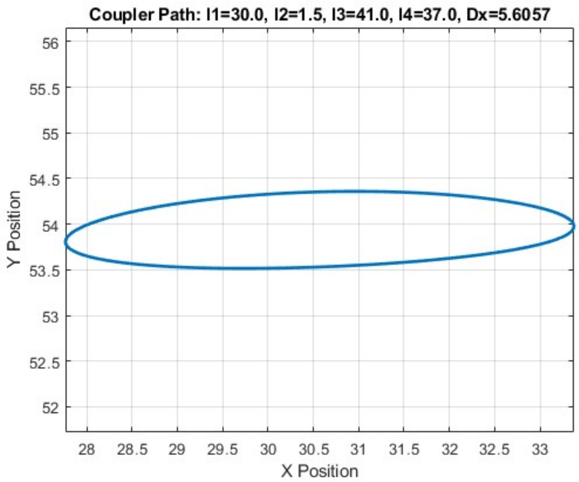


Fig. 4.1 (f)

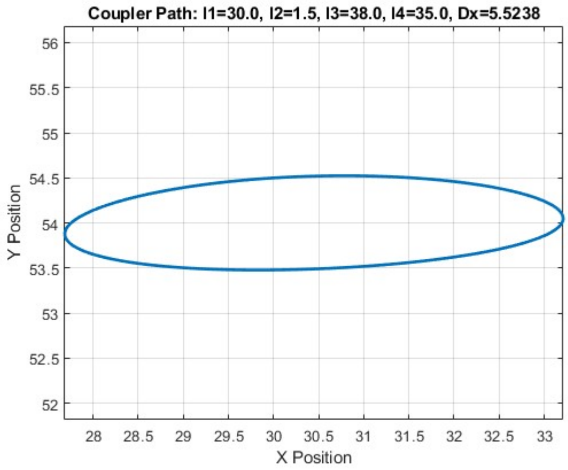


Fig. 4.1 (g)

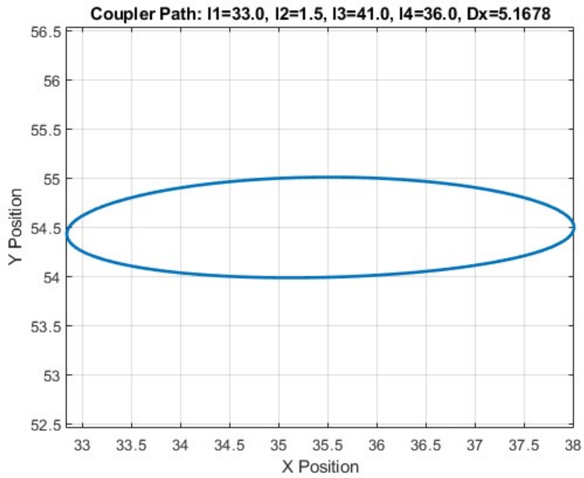
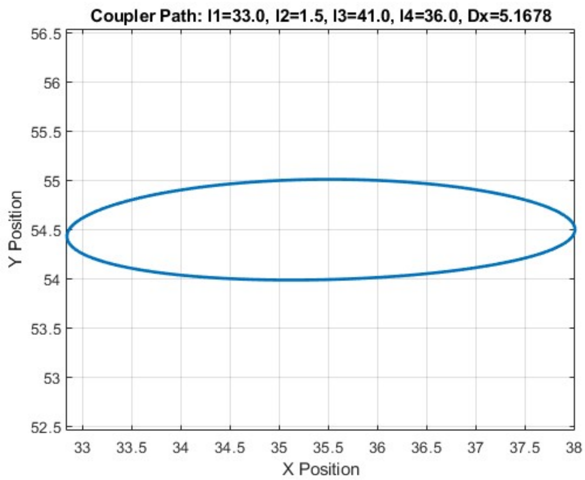


Fig. 4.1 (h)



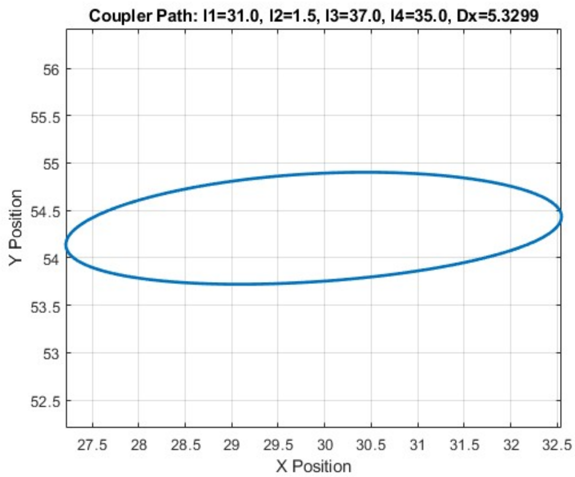


Fig. 4.1 (i)

Fig. 4.1 (j)

Fig. 4.1 Coupler paths traced by different potential solutions

Among the ten results, four-bar linkage having dimensions mentioned below has been selected which exhibit the most refined and reliable coupler path. The selected dimensions are;

Fixed link (l1) – 33.0

Crank (l2) – 1.5

Coupler (l3) – 41.0

Rocker (l4) – 36.0

All dimensions are in cm.

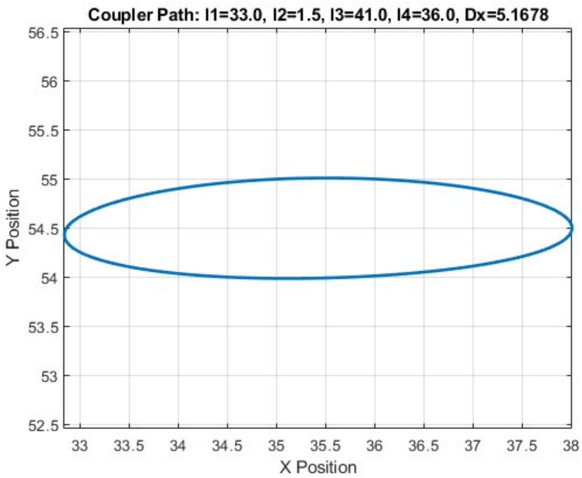


Fig. 4.2 Selected coupler path

Animation of the coupler path is shown below.

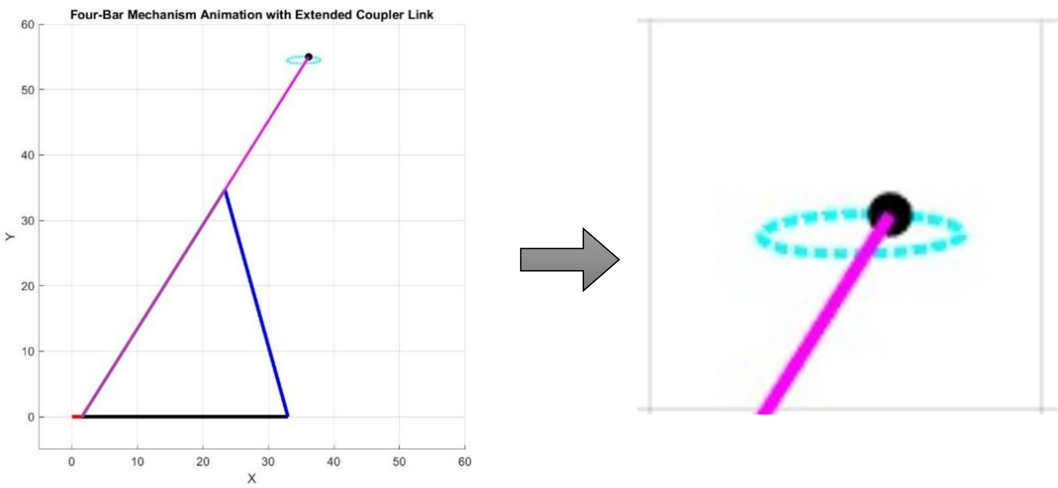


Fig. 4.3(a) 0rigin (-5, -5)

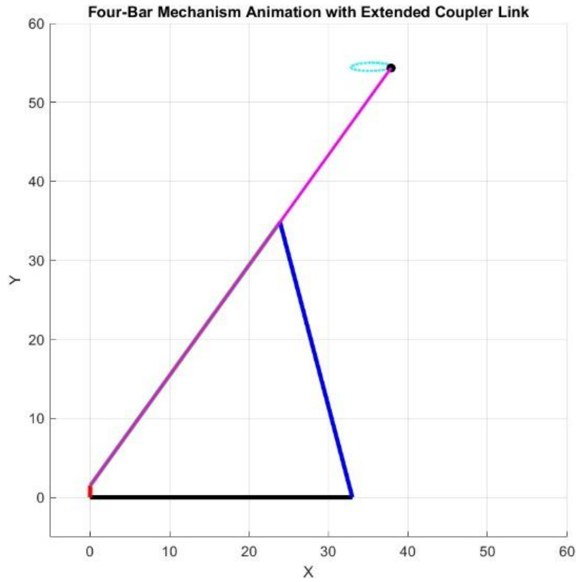


Fig. 4.3(b) 0rigin (-5, -5)

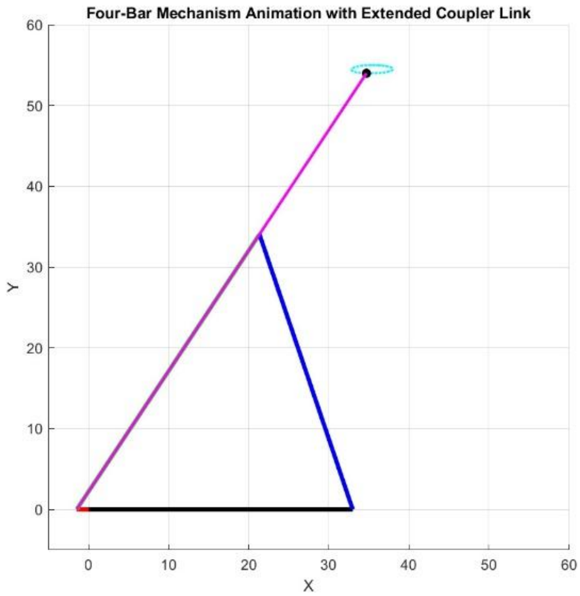


Fig. 4.3(c) 0rigin (-5, -5)

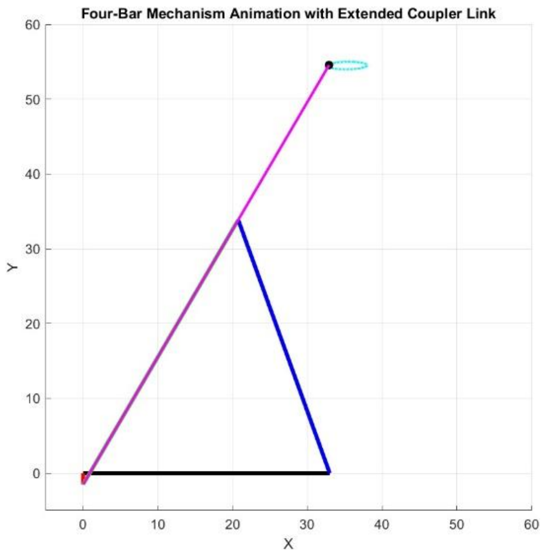


Fig. 4.3(d) 0rigin (-5, -5)

Fig. 4.3 Animated view of the coupler point at different positions on the coupler curve

The four views represent different instances in the crank's rotation cycle, highlighting the continuous transformation of the mechanism's configuration. The blue, pink, and black lines depict the links of the mechanism, while the light blue trace marks the path traced by the extended coupler link. These sequential frames help in understanding how the coupler point traverses along its path. This animation plays a crucial role in verifying the functional feasibility of the selected dimensions and ensuring that the coupler point follows the desired trajectory with minimal deviation.

4.1.2 Design of four-bar mechanism

Using the link lengths obtained from MATLAB, a four-bar mechanism is designed in SolidWorks to visualize and analyse the movement of the system.

4.1.2.1 Components of four-bar mechanism

- Fixed link (frame)

  The stationary link that supports the entire mechanism. The selected link dimensions are 330 mm length, 40 mm width and 10 mm thickness. Holes of diameter 11 mm are provided at both ends to attach to other links and 10 mm diameter are provided to connect it to the main frame.

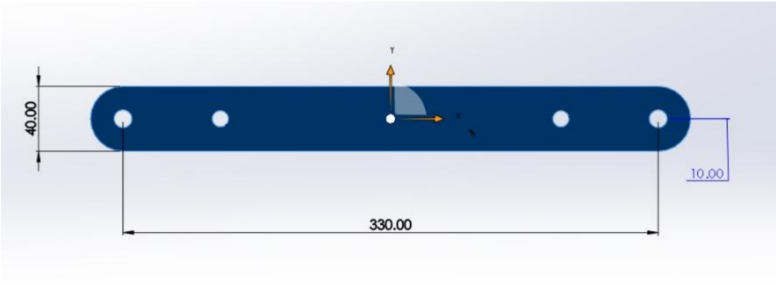


Fig. 4.4 Fixed link

- Input Link (Crank):

  The link that is typically driven by an external power source or motor. The selected link dimension is 15 mm length, 25 mm width and 10 mm thickness. A hole of diameter 11 mm is provided at both ends.

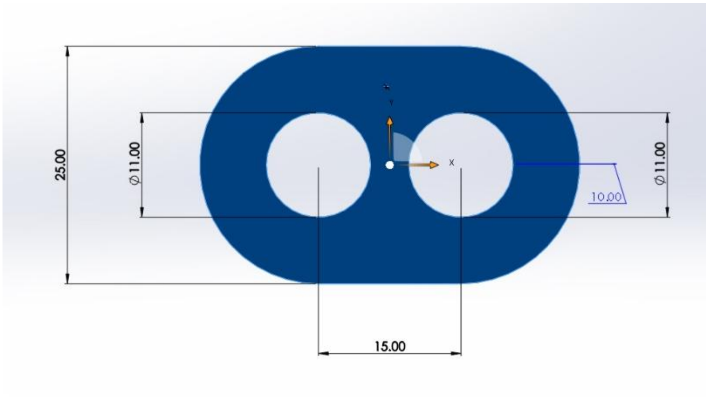


Fig. 4.5 Input Link

- Coupler Link:

  A link that connects the input and output links and generally performs an intermediate motion. The selected link dimensions are 410 mm length, 40 mm width and 10 mm thickness. The coupler is extended up to 650 mm with 240 mm as coupler extension to facilitate the attachment of cutting tool. Holes of 11 mm diameter are provided to attach the coupler to other links.

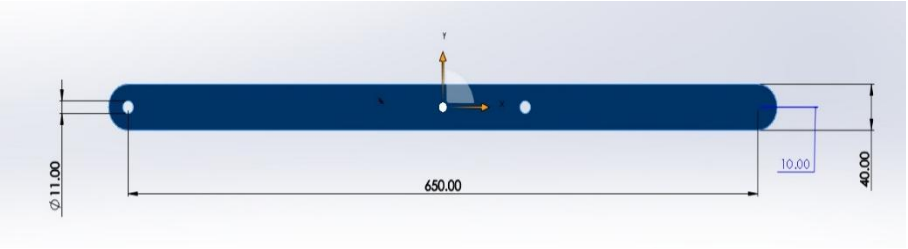


Fig. 4.6 Coupler Link with extension

- Output Link (Rocker):

  The link that generates the desired output motion (such as oscillation or rotation). The selected link dimension is 360 mm, 40 mm width and 10 mm thickness. A hole of diameter 11 mm is provided at both ends.

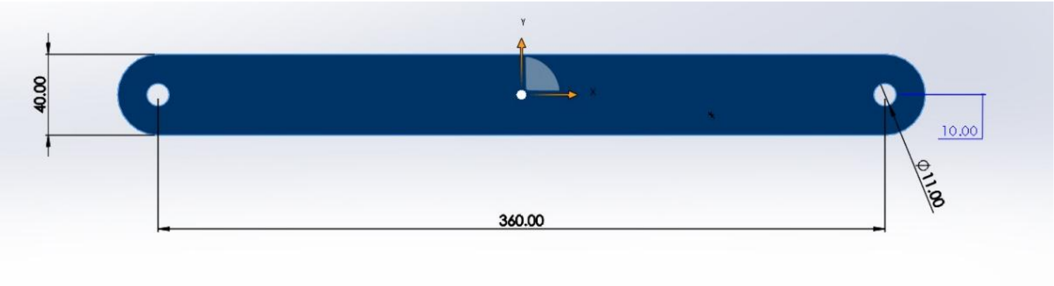


Fig. 4.7 Output Link

These components are assembled to for a four-bar mechanism in vibratory tillage tool as shown in the figure below.

Fig.4.8 Four-bar Assembly

4.1.2.2 Cutting tool

Taking the factors mentioned in chapter III into consideration, the selected dimensions of cutting tool are 120 mm length, 25 mm width and 30 mm thickness and a rake angle of 22°.

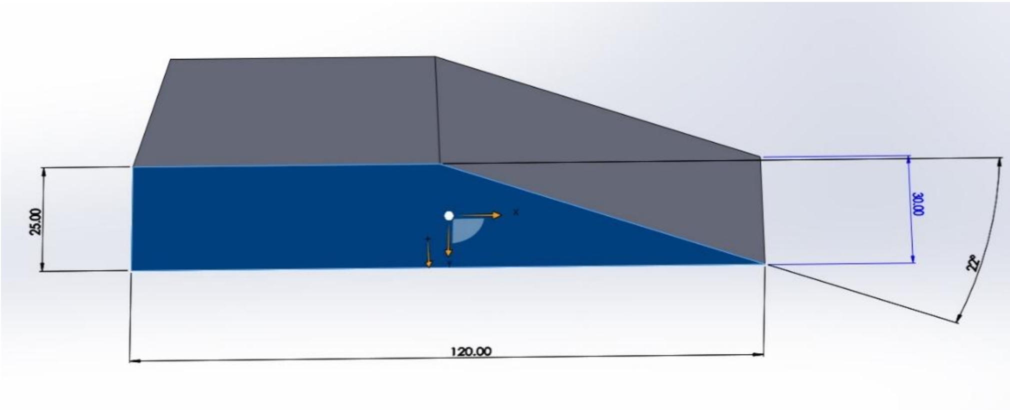


Fig. 4.9 Cutting tool

4.1.2.3 Cutter bar

Considering standard track width of tractor as 42 inches (1320 mm) and easy conveyance of cut crop towards the centre, 600 mm length,60 mm width, 40 mm thickness and an inclination of 15° to the horizontal has been selected for the cutter bar. In the design of the cutter bar, a total of 15 blades are placed along a 600 mm length, with a spacing of 10 mm between each blade. This arrangement ensures that the blades are evenly distributed across the cutting bar, providing uniform cutting action and consistent crop flow during operation. The spacing of 10 mm between blades allows for adequate clearance, preventing crop buildup and reducing the likelihood of clogging, especially when cutting dense or thick vegetation. The design of the blade placement in a 15 blade configuration maximizes the harvesting efficiency and improves the machine's overall performance in various field conditions. Two similar cutter bars are placed symmetrically at both ends.

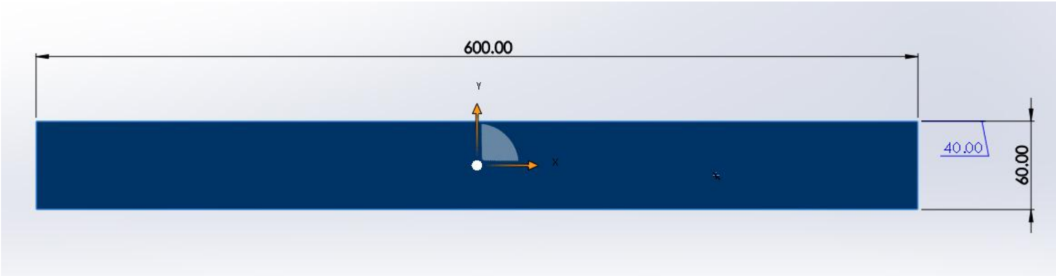


Fig. 4.10 Cutter bar frame

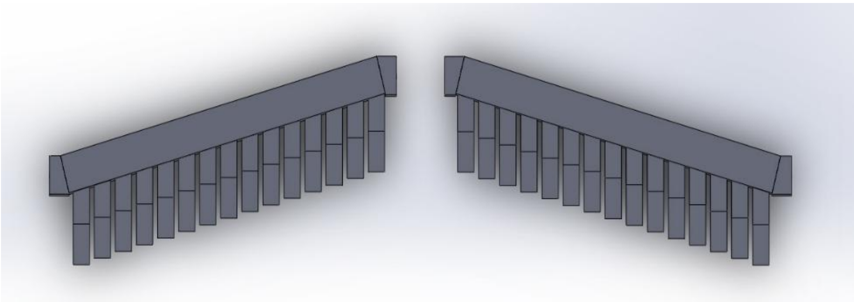


Fig. 4.11 Cutter bar

4.1.2.4 Main frame

A rectangular main frame has been considered for supporting the four-bar mechanism in the vibratory tillage tool, ensuring structural strength and alignment during operation. The frame is constructed using two different sizes of rectangular sections. The longer sides of the frame are designed with dimensions of 1320 mm in length, 60 mm in width, and 20 mm in thickness, providing extended coverage and support for the movement of the mechanism. In contrast, the shorter sides of the frame measure 600 mm in length, 60 mm in width, and 20 mm in thickness, forming the connecting ends of the rectangular structure. This configuration ensures a rigid and balanced frame capable of withstanding the dynamic forces generated. The uniform width and thickness across all sides contribute to mechanical stability, while the variation in length helps accommodate the design and functional requirements of the mounted four-bar mechanism.

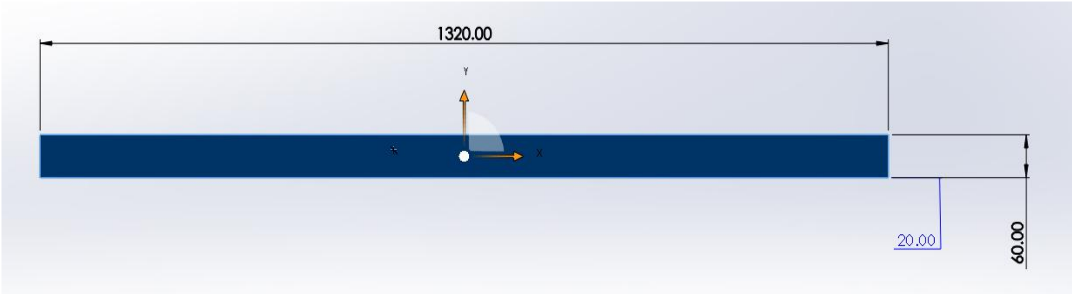


Fig. 4.12 Main frame longer side

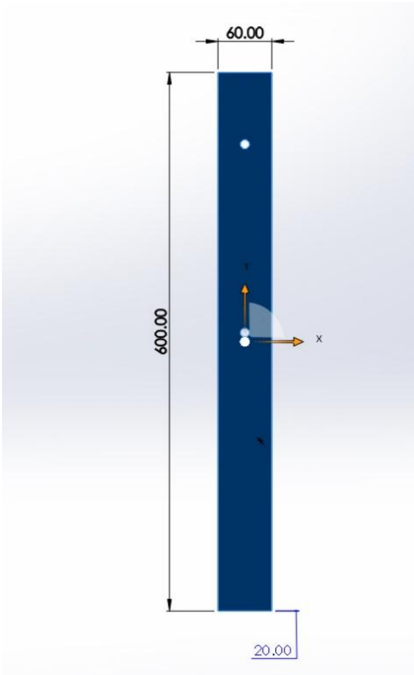


Fig. 4.13 Main frame shorter side

Fig. 4.14 Main frame assembly

4.1.2.5 Supporting structure

A support bar with dimensions of 532 mm in length, 40 mm in width, and 10 mm in thickness has been provided to establish a connection between the cutter bar and the main frame on both sides. This support bar plays a critical role in maintaining the structural integrity and alignment of the cutter bar during operation. A connecting bar with dimensions of 600 mm in length, 60 mm in width, and 20 mm in thickness has been installed to link the longer sides of the rectangular main frame, to which the support bar is attached. To further ensure uninterrupted and smooth motion, a crank measuring 112 mm in length, 40 mm in width, and 10 mm in thickness has been strategically positioned between the support bar and the connecting bar. The crank functions as a mechanical link that prevents the constriction of motion during the reciprocating movement of the cutter bar, allowing for flexibility and proper transfer of motion without causing misalignment or mechanical stress.

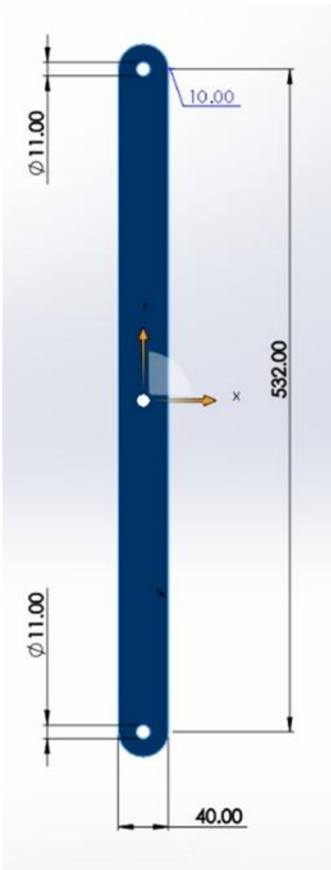


Fig. 4.15 Supporting bar

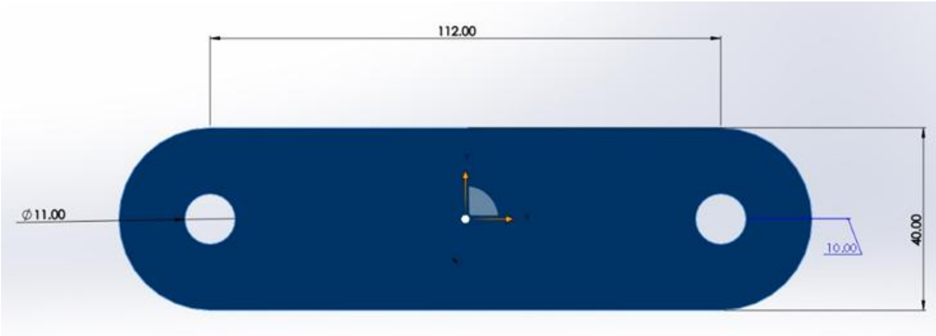


Fig. 4.16 Crank of supporting bar

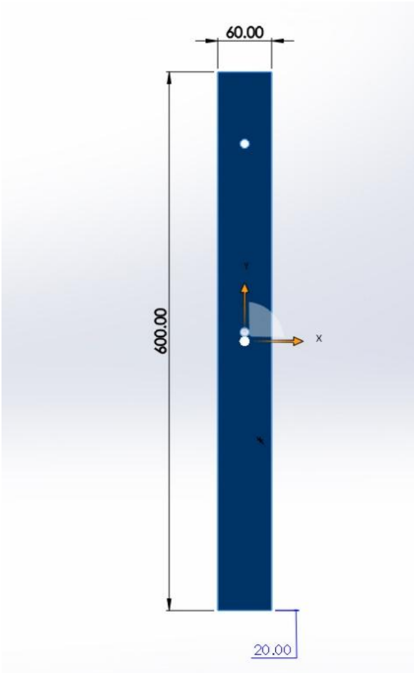
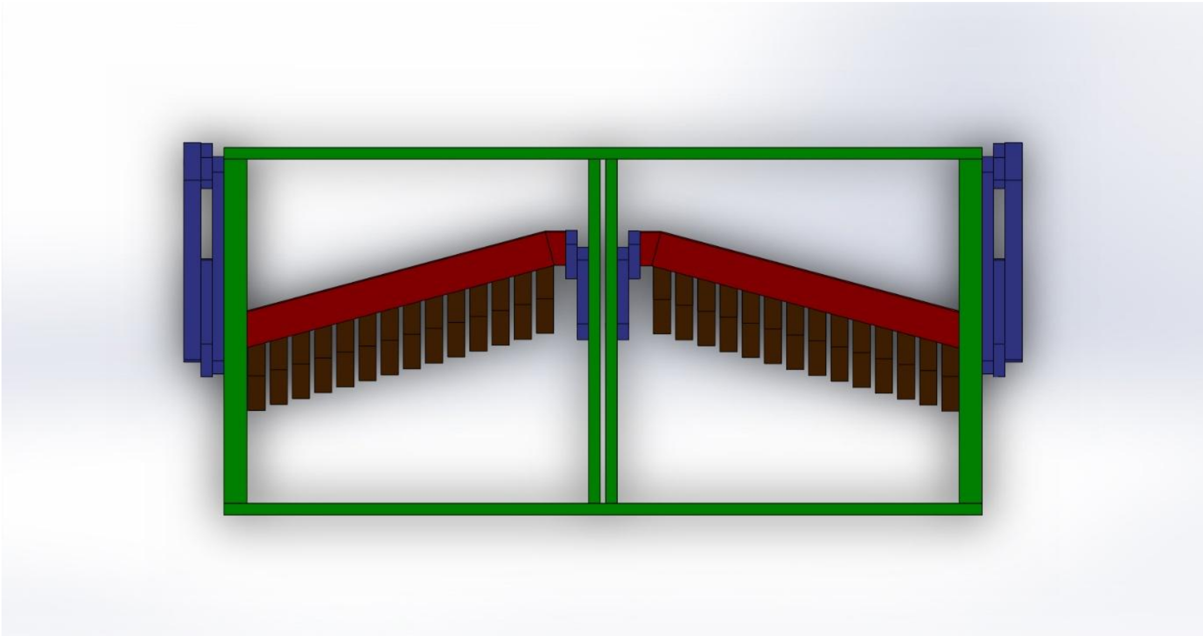
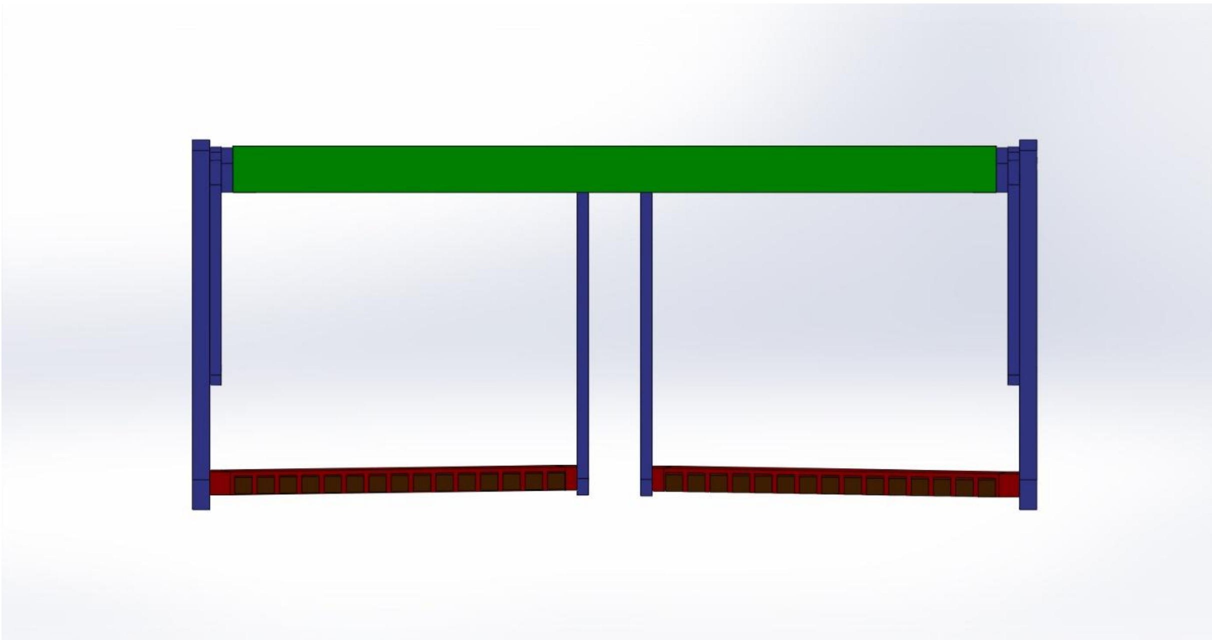


Fig. 4.17 supporting bar connecting main frame

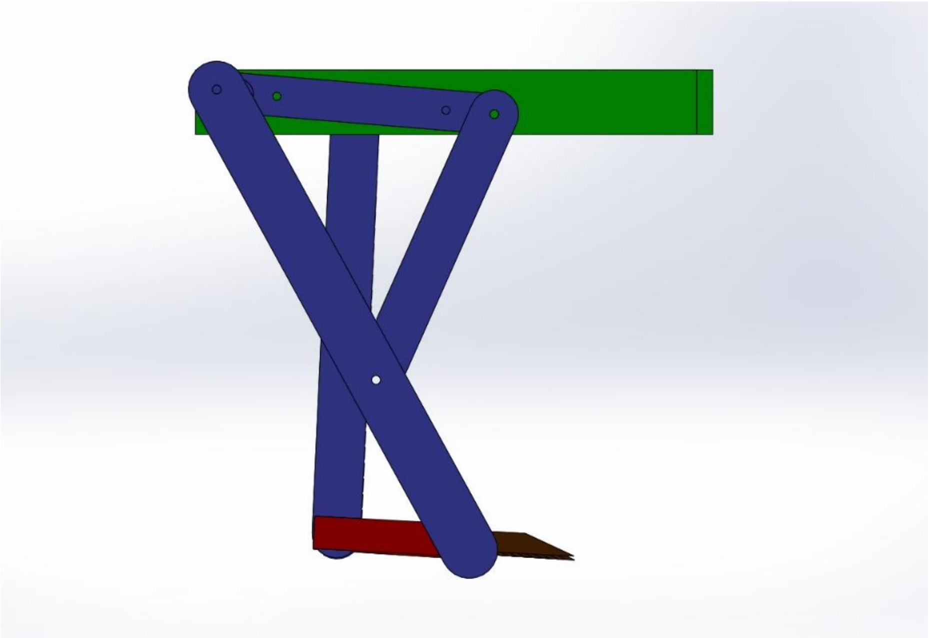
The parts designed in SOLIDWORKS were carefully assembled to create the final model, which represents a four-bar mechanism in a vibrated configuration. Each component was positioned and constrained using appropriate mates to ensure accurate alignment and functional movement within the assembly. The resulting model demonstrates the intended vibratory motion relationship among the links, effectively showcasing the dynamic interaction of the mechanism. This assembled configuration serves as a visual and functional representation of the proposed design, forming the basis for further simulation, analysis, and performance evaluation of the vibratory tillage tool system.



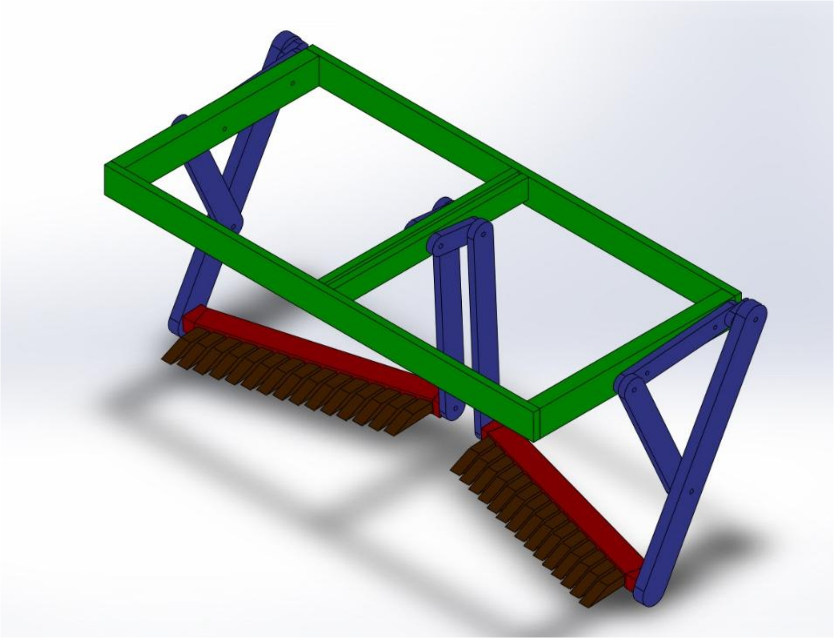
4.18 (a) Top view



4.18 (b) Front view



4.18 (c) side view



4.18 (d) isometric view

4.18 Final assembly of four-bar mechanism in vibratory tillage tool

3.2 FINITE ELEMENT ANALYSIS

The harvester model was initially designed using SOLIDWORKS and subsequently imported into ANSYS Workbench to perform a finite element analysis (FEA) under static loading conditions. Material properties and boundary conditions were carefully defined to simulate realistic working scenarios. Upon solving the simulation, the analysis revealed that the maximum stress developed in the structure exceeded the allowable stress limit of 460 MPa of structural steel. This result indicates that the design, in its current state, does not meet the required safety standards, as the calculated factor of safety falls below acceptable limit of 2. Therefore, design modifications or material optimization are necessary to ensure structural integrity and reliable field performance.

$$\text{Factor of safety} = \frac{\text{Maximum permissible stress}}{\text{Maximum induced stress}}$$

3.2.1 First trial

The results obtained are as follows:

Maximum permissible stress = 460 MPa

Maximum stress obtained = 581.05 MPa

Maximum deformation= 27.534 mm

Factor of safety = 460/ 581.05 = 0.79

The result obtained has the factor of safety 0.79, which is much less than the allowable factor of safety limit of 2. This indicated that the structure cannot withstand the provided load conditions which necessitates the need of structural redesign.

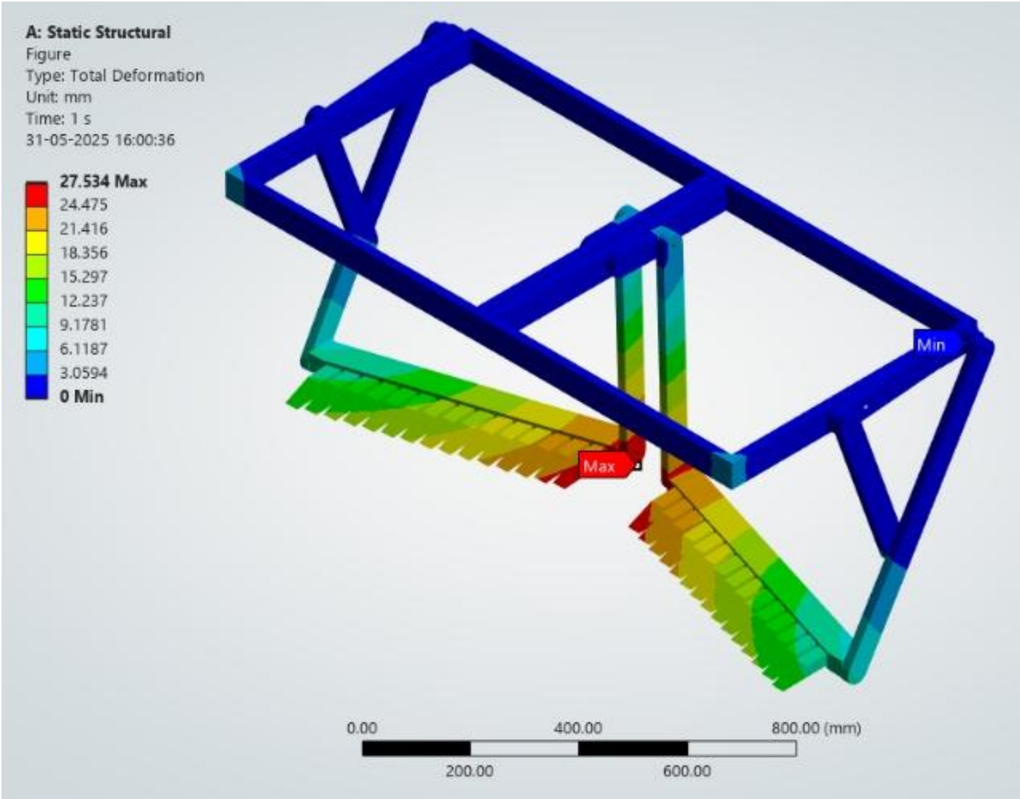


Fig. 4.19 Maximum stress obtained = 581.05 MPa in first trial

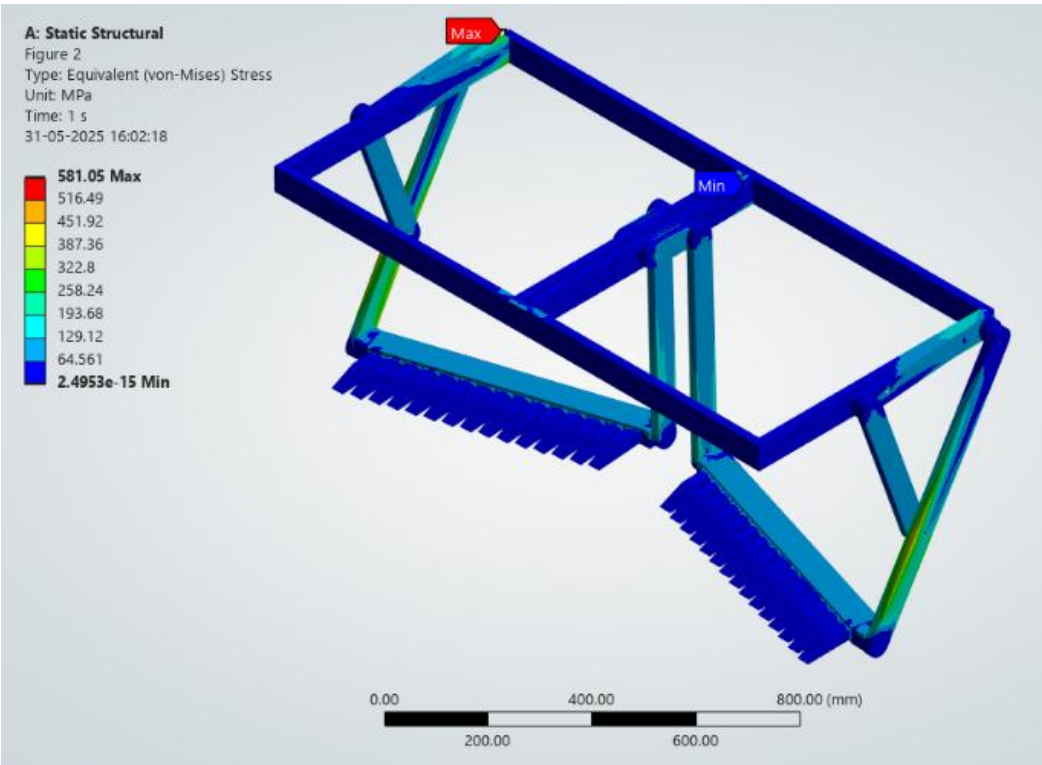


Fig. 4.20 Maximum deformation= 27.534 mm in first trial

3.2.2 Fourth trial

The structure has been redesigned by changing some of the dimensions as follows:

- Blade support:

  Width changed from 40 to 60

  Thickness 10 to 20

- Blade crank:

  Thickness changed from 10 to 20

- Main frame:

  Thickness changed from 60 to 80

- Crank:

  Width changed from 25 to 40

  Thickness- changed from 10 to 20

- Coupler:

  Width changed from 40 to 50

  Thickness changed from 10 to 20

- Rocker:

  Width changed from 40 to 50

  Thickness changed from 10 to s 20

- Frame:

  Width changed from 40 to 50

  Thickness changed from 10 to 20

All dimensions are in mm.

The harvester model has been redesigned using SOLIDWORKS and subsequently imported into ANSYS Workbench to perform a finite element analysis (FEA) under static loading conditions. The results obtained are:

Maximum permissible stress = 460 MPa

Maximum stress obtained = 298.48 MPa

Maximum deformation= 13.089 mm

Factor of safety = 460/ 298.48 = 1.54

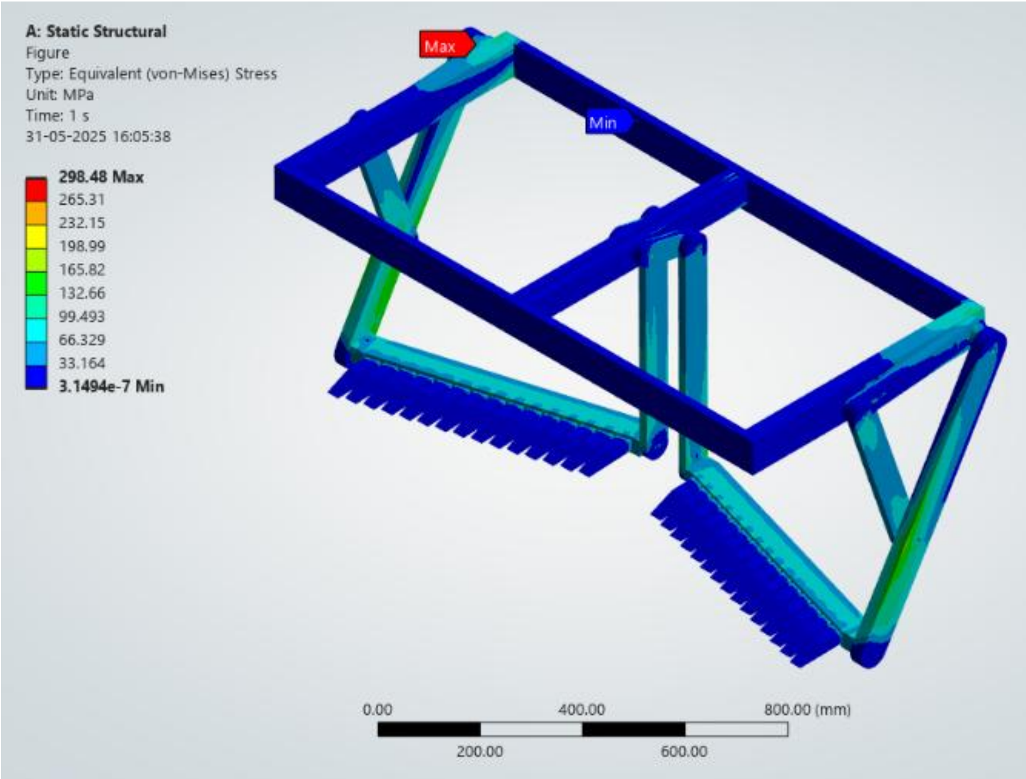


Fig. 4.21 Maximum stress obtained = 298.48 MPa in Fourth trial

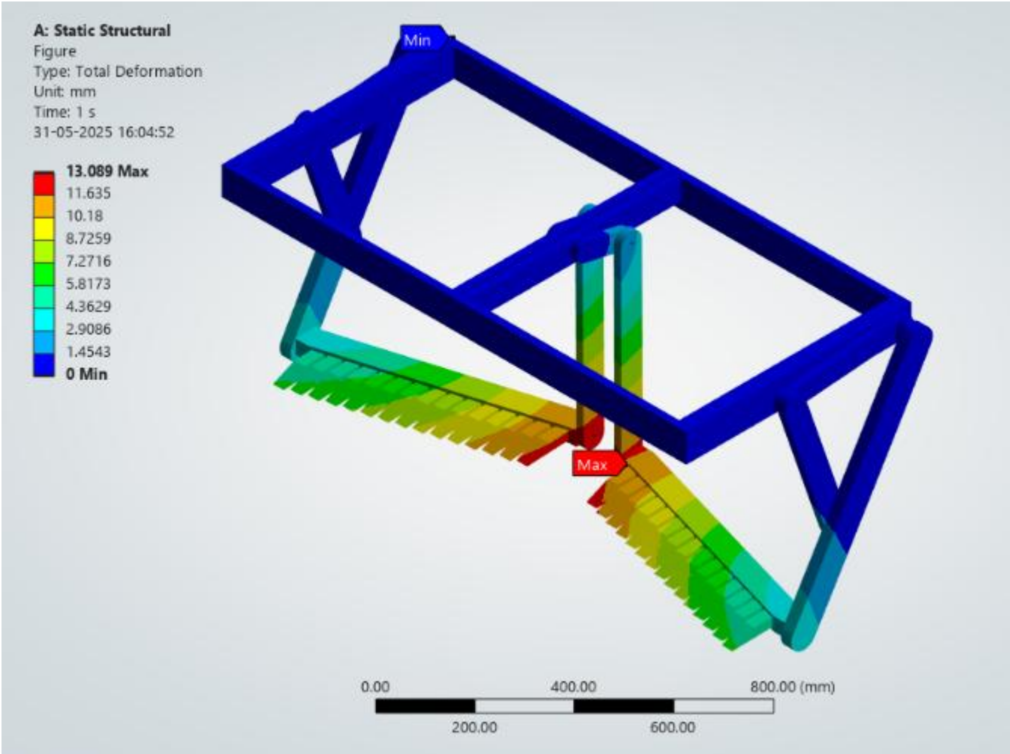


Fig. 4.22 Maximum deformation= 13.089 mm in Fourth trial

The result obtained has the factor of safety 1.54, which is less than the allowable factor of safety limit of 2. This indicated that the structure can withstand the provided load conditions, but it is not safe as the safety factor is less than 2, which necessitates the need of structural redesign.

3.2.3 Eighth trial

The structure has been redesigned by changing some of the dimensions as follows:

- Crank:

  Total length changed to 80, where holes are maintained at a difference of 15

- Coupler:

  Width changed from 50 to 70

  Thickness changed from 20 to 30

- Rocker:

Width changed from 50 to 60

The harvester model has been redesigned using SOLIDWORKS and subsequently imported into ANSYS Workbench to perform a finite element analysis (FEA) under static loading conditions. The results obtained are:

The results obtained are as follows:

Maximum permissible stress = 460 MPa

Maximum stress obtained = 233.96 MPa

Maximum deformation= 12.615 mm

Factor of safety = 460/ 233.96 = 1.97

The result obtained has the factor of safety 1.97, which is much closer to the allowable factor of safety limit of 2. This indicated that the structure can withstand the provided load conditions.

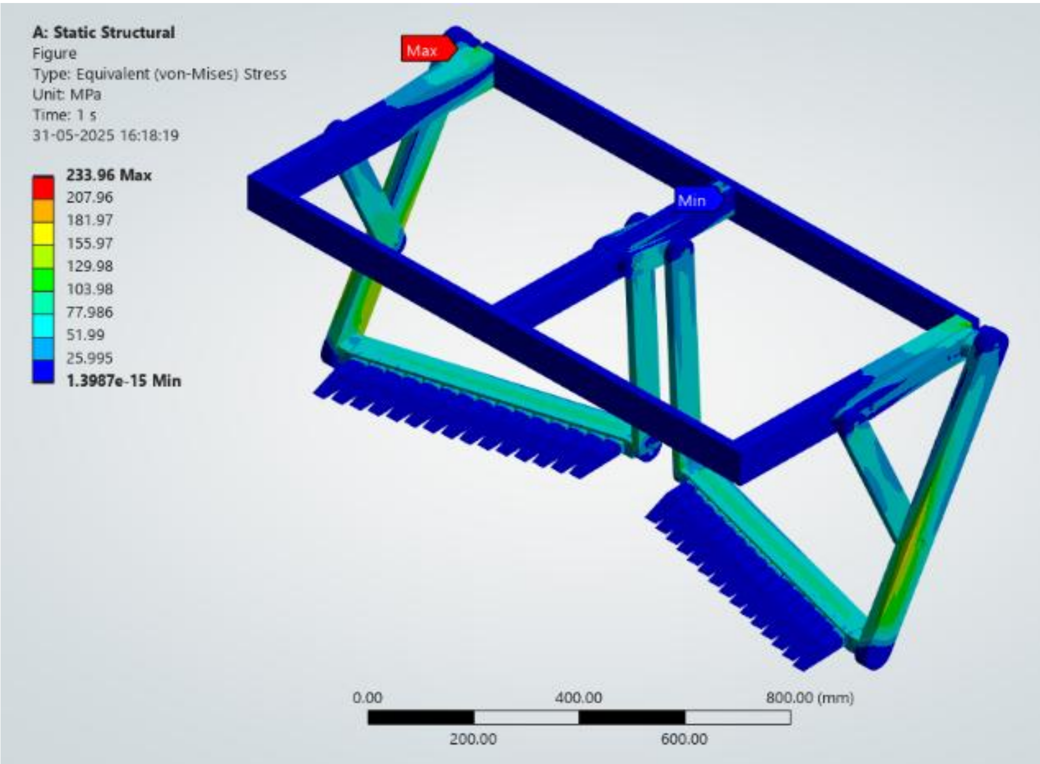


Fig. 4.23 Maximum stress obtained = 233.96 MPa in Eighth trial

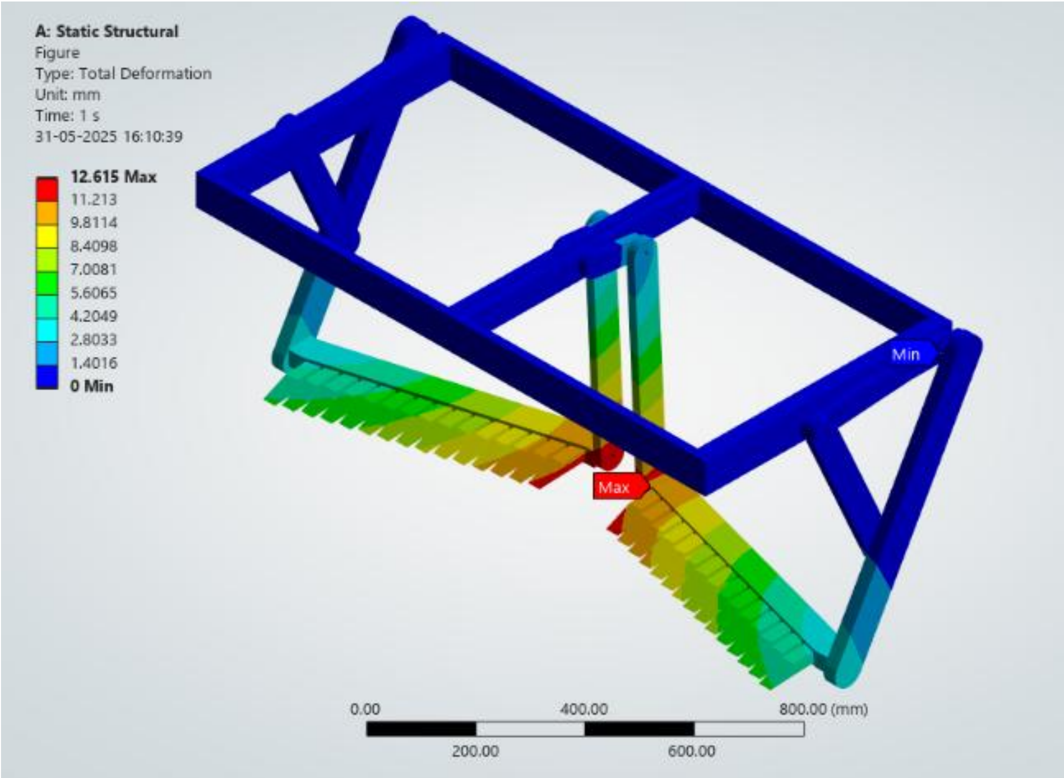


Fig. 4.24 Maximum deformation= 12.615 mm in Eighth trial

3.2.4 Tenth trial

The structure has been redesigned by changing some of the dimensions as follows:

- Cutting tool:

  Thickness changed from 30 to 40

- Blade support:

  Width changed from 60 to 70

- Blade:

  No. of blades on one side changed from 15 to 12

All dimensions are in mm.

The harvester model has been redesigned using SOLIDWORKS and subsequently imported into ANSYS Workbench to perform a finite element analysis (FEA) under static loading conditions. The results obtained are:

The results obtained are as follows:

Maximum permissible stress = 460 MPa

Maximum stress obtained = 188.47 MPa

Maximum deformation= 9.86 mm

Factor of safety = 460/ 188.47 = 2.44

The calculated factor of safety is 2.44, which exceeds the minimum allowable limit of 2. This indicates that the redesigned structure is capable of withstanding the applied load conditions safely.

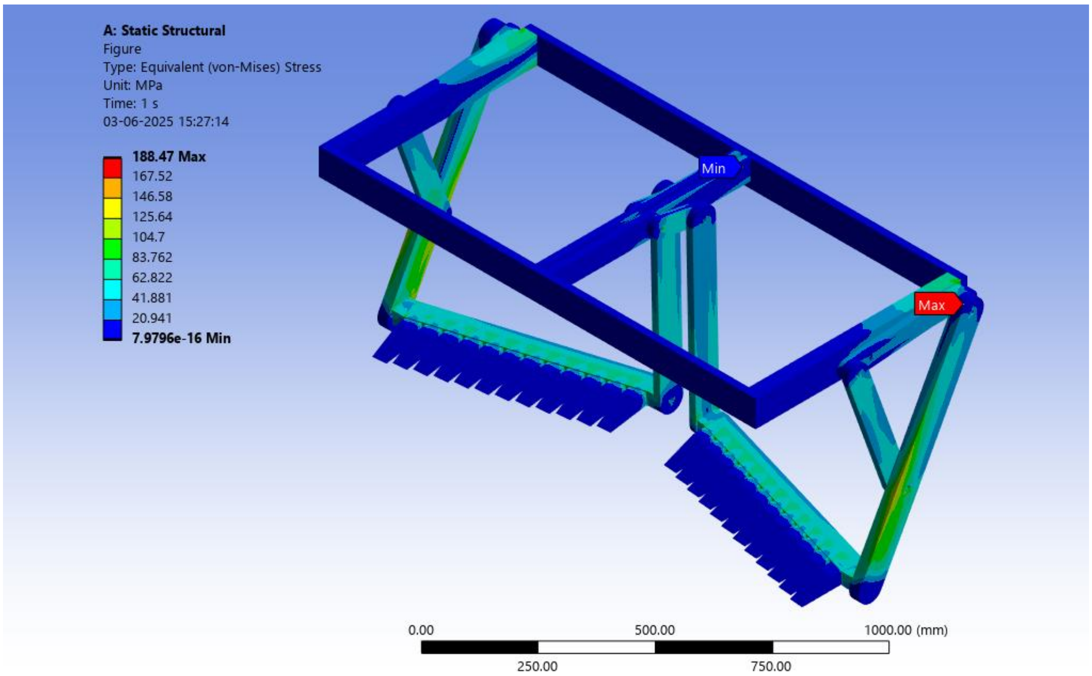


Fig. 4.25 Maximum stress obtained = 188.47 MPa in Tenth trial

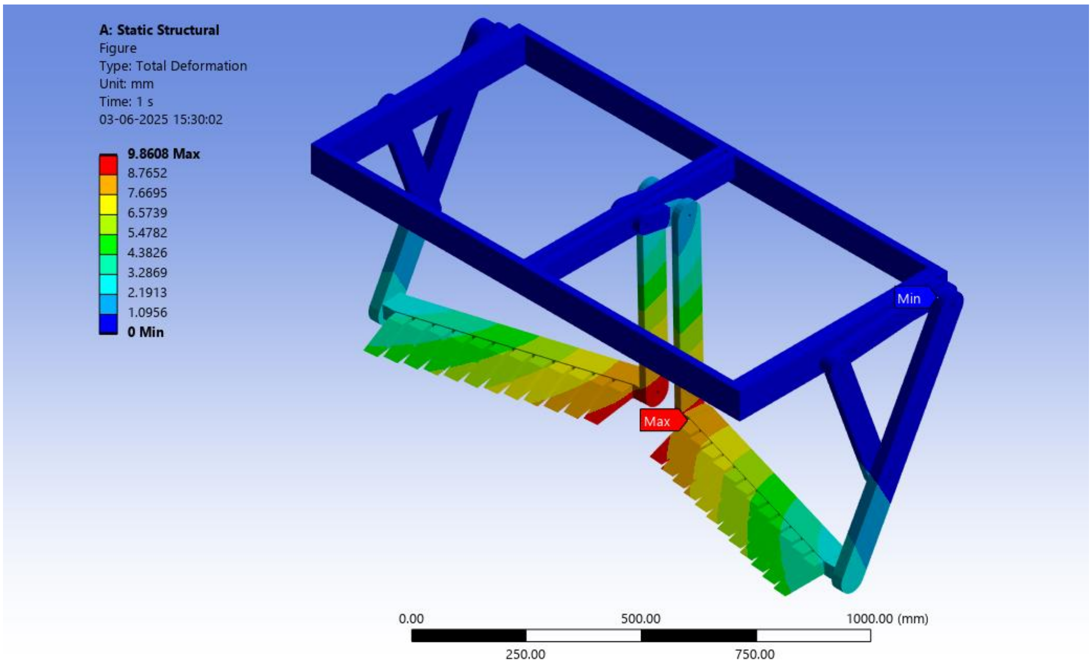


Fig. 4.26 Maximum deformation= 9.86 mm in Tenth trial

Conclusion

CHAPTER V

CONCLUSION

The study successfully demonstrated the feasibility and effectiveness of incorporating a four-bar mechanism in vibratory tillage tools. Through systematic dimensional synthesis using MATLAB, a precise and optimized linkage configuration was selected based on the desired coupler path characteristics. MATLAB-based dimensional synthesis enabled the identification of optimal link dimensions that comply with mechanical constraints, including Grashoff's condition. The design was further validated and visualized through SolidWorks modelling and subsequently subjected to structural evaluation using finite element analysis in ANSYS. Initial trials revealed that design improvements were necessary to meet safety standards; however, through multiple iterations and parameter adjustments, a final structure was achieved with a factor of safety of 2.44, closely aligning with the target safety threshold. The integration of vibration with optimal link dimensions and tool geometry significantly reduces draft requirements and enhances soil penetration, making this approach promising for sustainable and energy-efficient agricultural practices. This research validates the use of the four-bar linkage as a cost-effective and energy-efficient mechanism for vibratory tillage applications, offering significant benefits in modern agricultural operations. This project paves the way for advanced mechanization in tillage operations, especially in challenging soil conditions.

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CHAPTER VI

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STUDY ON DESIGN OF FOUR-BAR MECHANISM IN VIBRATORY TILLAGE TOOLS

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ABSTRACT

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ABSTRACT

The increasing demand for energy-efficient and effective tillage practices has led to the development of innovative machinery such as vibratory tillage tools. This project focused on the design, simulation, and optimization of a four-bar mechanism integrated into a vibratory tillage tool, with the goal of improving soil penetration, reducing draft force, and enhancing energy efficiency in field operations. The work was carried out with four key objectives: to study the engineering parameters for the development of the mechanism, perform dimensional synthesis using MATLAB, design the mechanism in SolidWorks, and validate the structural integrity using Finite Element Analysis (FEA) in ANSYS. Through MATLAB simulations, a total of 1056 design iterations were evaluated, from which the most suitable link dimensions were selected: fixed link (l_1) = 33.0 cm, crank (l_2) = 1.5 cm, coupler (l_3) = 41.0 cm, and rocker (l_4) = 36.0 cm. These dimensions satisfied Grashoff's condition and produced a smooth and controlled oscillatory motion at the coupler point, which was critical for effective vibratory action. The selected mechanism was then modelled in SolidWorks and analysed in ANSYS under simulated working conditions. The FEA revealed a maximum stress of 188.47 MPa and a maximum deformation of 9.86 mm in the best trial. The design exhibited a factor of safety of approximately 2.44, indicating that the structure could withstand the applied loads under vibratory operation without failure. The results confirmed that the optimized four-bar mechanism can efficiently transmit vibratory motion with reduced power requirements and improved soil cutting performance. This study demonstrates the potential of integrating a customized four-bar linkage into tillage tools to enhance functionality, durability, and sustainability in agricultural mechanization. The results confirm the ability of such mechanisms to reduce energy consumption, improve soil breakup, and contribute to more efficient and precise agricultural operations, particularly in compacted or hard soils.