Bansilal Ramnath Agarwal Charitable Trust's

Vishwakarma Institute of Technology

(An Autonomous Institute affiliated to Savitribai Phule Pune University

Bibwewadi, Upper Indira Nagar, Pune 411037

A Final Project Report

based on

Design and Analysis of a Tractor-Mounted Pump Lifting Mechanism for Deep Bore Applications

Submitted by

Wani Prajwal Chandrashekhar – 12220254

Thorat Yash Sachin – 12220255

Alekar Pratham Sunil - 12110935

Under the Guidance of

PROF. Dr. M. Nalawade

Mechanical Engineering Department

Academic year 2024-25

BANSILAL RAMNATH AGARWAL CHARITABLE TRUST'S

VISHWAKARMA INSTITUTE OF TECHNOLOGY

PUNE-411 037

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CERTIFICATE

This is to certify that the Project titled "**Design and Analysis of a Tractor-Mounted Pump Lifting Mechanism for Deep Bore Applications**" has been completed in the academic year 2024 – 2025, by **WANI PRAJWAL CHANDRASHEKHAR** (GR No: 12220254, Roll NO: 88), in partial fulfilment of B. Tech in Mechanical Engineering as prescribed by Savitribai Phule Pune University.

Prof. M. Nalawade
Institute Guide,

Examiner

Prof. (Dr.) D. B. Hulwan
Head of Department,

Mechanical

Place: Pune

Date: /5//2025

ACKNOWLEDGMENT

I would like to express my deepest gratitude to **Dr. D. B. Hulwan**, Head of the Department of Mechanical Engineering, for his unwavering support, guidance, and for granting me the permission to undertake this project. His encouragement and leadership have been instrumental in shaping the direction and success of this endeavor.

My heartfelt thanks also go to **Prof. (Dr.) M. Nalawade**, Project Coordinator and College Guide at Vishwakarma Institute of Technology, for his invaluable mentorship throughout the course of this project. His expert advice, constant encouragement, and constructive feedback greatly enhanced the quality of my work. I am truly fortunate to have worked under his guidance.

I am also grateful to all the faculty members and staff of the Mechanical Engineering Department for their support, insights, and for providing a learning environment conducive to innovation and growth.

I would like to extend special thanks to all those who, directly or indirectly, contributed to the successful completion of my final year project. Their assistance, encouragement, and cooperation made a significant difference in overcoming the challenges faced during various stages of the project.

This project has provided me with a unique opportunity to apply theoretical knowledge to practical scenarios. The skills, insights, and hands-on experience I have gained throughout this journey will be cherished and will continue to influence my professional and academic growth in the years to come.

Finally, I am thankful to my peers, friends, and family members whose moral support and motivation helped me stay focused and driven throughout these endeavours.

Article

Design and Analysis of a Tractor-Mounted Pump Lifting Mechanism for Deep Bore Applications

Wani Prajwal [1], Thorat Yash [2], Alekar Pratham [3], Nalawade Mukund [4]

1. Vishwakarma Institute of Technology, Mechanical dept. Pune, India prajwal.wani22@vit.edu
2. Vishwakarma Institute of Technology, Mechanical dept. Pune, India yash.thorat22@vit.edu
3. Vishwakarma Institute of Technology, Mechanical dept. Pune, India pratham.alekar21@vit.edu
4. Vishwakarma Institute of Technology, Mechanical dept. Pune, India nalwade.mukund22@vit.edu

- Correspondence: prajwal.wani22@vit.edu

Featured Application: The current research may serve as a reference to the loading and lifting equipment Design and Analysis of a Tractor-Mounted Pump Lifting Mechanism for Deep Bore Applications

Abstract - Submersible pumps installed for farming irrigation through borewells tend to have to be withdrawn from time to time for upkeep. Conventional manual lifting proves to be an energy-intensive process. This work brings forth the design and testing of a half-automatic tractor- mounted pump lift system specifically adapted to deep bore use. The mechanism uses a PTO-powered rope drum, bevel gear system, and worm and worm wheel setup to hoist loads of 150 kg at a depth of 100 feet. The 6×19 IWRC steel wire rope was chosen for strength, fatigue, and abrasion. The torque calculation, drum size selection, fatigue bending analysis, and rope selection were done in accordance with global standards. The mechanism provides a safe, energy-saving solution for remote or rural locations that maximizes both safety and maintenance ease.

Keywords: Semi-automatic lifting system, tractor PTO, borewell pump retrieval, steel wire rope design, agricultural engineering, rural water systems.

Introduction

The design of effective and dependable lifting devices has been the subject of many studies in areas ranging from heavy construction, rural water supply, to automated material handling. Researchers have attempted structural optimization, dynamic control systems, safety methods, and mechanical advancements to enhance performance, save costs, and increase usability in real-world applications.

Reliability-based structural optimization has been an important research area in heavy lifting systems. A robust design optimization (RBDO) framework wasadopted for crawler cranes' lattice booms for which weight reduction was attained while not compromising on safety by combining probabilistic restrictions like strength, stiffness, overall stability, and compressive-bending reliability. The method also proved to efficiently achieve up to 9.62% reduction in weight by keeping the reliability index greater than 4.364, with still attaining a 2.8% reduction at an even more cautious reliability level, providing huge savings in material as well as costs [1].

In crane operation and motion control, dynamic modeling with wire rope elasticity was brought into play to improve the simulation of crane behavior under sophisticated conditions. This model enabled the creation of high-performance robust sliding mode control (SMC) algorithms such as standard, terminal, and fast terminal SMC that handled multiple degrees of freedom like boom luffing, cargo-lifting, and swing suppression at the same time. [1]. These strategies showed high precision in motion coordination while minimizing oscillations, ensuring system stability in dynamic environments [2].

Crane safety has also been extensively reviewed, highlighting a gap in the attention given to non-technical risk dimensions. While equipment-related hazards were well covered, other factors such as operator behavior, environmental conditions, and regulatory deficiencies received relatively little scholarly focus. Eight significant research gaps were identified, urging a shift toward a more multidisciplinary approach in crane safety research to better prevent accidents and improve operational guidelines [3]. Complementary to this, practical literature has provided detailed engineering insights into components such as wire ropes, drives, and maintenance practices, supporting the implementation of safer and longer-lasting crane systems [4].

Parallel advancements have been made in the field of pump and water lifting mechanisms. A deep-sea motor pump utilizing an increased flowrate approach was designed to expand passage dimensions, enabling the transport of coarse particles and improving anti-blocking performance. A six-stage pump configuration was validated through experiments, confirming its ability to handle suspended particles without motor overload—demonstrating the design's effectiveness in particle-laden environments [5].

Automation and predictability in small-scale lifting systems have also been examined. An electric motor power-driven, mechanical lift automation system with a 1-hp motor and a counterweight scheme was established in industrial material handling. The third-order polynomial character of the lift time vs load relationship and multiple operating zones present in the lift cycle were seen in experiment outcomes. This improves the understanding of load behavior trends and efficiency performance in low-capacity automated systems [6].

Hydraulic lifting circuits widely employed by cranes are strongly affected by the mechanical properties of steel wire ropes. Investigation showed that the pre-tensioning of hoisting ropes has a remarkable effect on minimizing peak dynamic loads and stabilization time—by 92.6% and 49%, respectively—demonstrating the significance of wire rope layout in system dynamics. Comparatively, rope stiffness increase affected the system to a relatively lesser extent, demonstrating that initial tension is a more significant parameter for performance adjustment [7].

Applications of bore well have motivated various innovations with respect to affordability, portability, and simplicity. A low-cost compact bore well pump lifting mechanism was developed for rural and small-scale users with low cost and less labor demand. The system is effective in enabling easy installation and removal of submersible pumps, thereby lessening the reliance on manual processes [8]. In another connected development, a wind-energy-water-lifting-pump mechanism was suggested, employing rotating foils and a piston-linear drive to harness the mechanical energy of the wind as a means to extract usable water—acting as a renewable solution to far-off water acquisition [9].

To surpass the shortcomings of conventional lifting systems, a review of bore well pump installation methods cited disadvantages in mechanisms such as chain pulleys and hydraulic lifters. A recently suggested automated machine overcame such shortcomings with its compact size and tool-free assembly/disassembly, which suited confined or rural settings [10]. The design was further improved upon with the creation of a gear-powered submersible pump lift machine that could lift pumps from 300 ft deep bore wells using only two workers. Working at a rate greater than 3 meters per minute, the machine was ten times more efficient than the conventional system and did not require any special tools—something that made it ideal for use in rural agriculture [11].

The literature reviewed as a whole highlights the increasing interest in improving efficiency, safety, and automation in bore well operations and farm machinery, especially in pipe lifting mechanisms and tractor-based systems.

Bajbalkar and Bhajibhakare et.al [12] propose an inclusive summary of present techniques of installation and hoisting of bore well pipes and submersible pumps, identifying major disadvantages of available methods, which are very manpower-intensive, time-consuming, and inefficient. They stress the imminent need for a compact, affordable mechanism to minimize manpower and cost of operations. Similarly Choure and Kale [17] suggest a motor-operated bore well pipe lifting and transporting machine, having a gear and roller operated lifting mechanism.The mechanism lifts automatically at a speed of more than 3 meters per minute, lifts the load of 100 to 1000 kg, and has the features of safety devices such as adjustable pressure rollers and a worm gearbox. Its simplicity is low-cost and appropriate for rural and limited-space operations, providing a significant improvement over conventional chain pulley system.

Moving to farm machinery, Cong et al. [13] discuss the shortcomings of traditional testing techniques for hydraulic linkages in high- and medium-horsepower tractors. They created a test device using a four-bar mechanism to assess load-lifting performance. The system decreases detection time considerably and does away with manual connection between the tractor and the load, thus enhancing efficiency and safety in hydraulic system testing. Structural optimization and analysis, Tarighi et al. [14] used finite element analysis to find the optimal design for the lower hitch link of a heavy-duty tractor. Two materials, CK45 and 945M38, were compared under the same stress conditions. Although both materials were able to endure one million cycles of loading, 945M38 showed a much greater factor of safety (5.13) and hence is an appropriate material for durability and reliability under tough, demanding applications. Likewise, Seyedabadi [15] performed a finite element analysis of the MF-285 tractor lift arm. The results showed that although the arm is safe for normal plow loads (500 kg), it is not structurally sufficient for the maximum hydraulic capacity of the tractor, which is 2230 kg. The research advises either reducing the lifting capacity to 1430 kg, thus having a minimum safety factor of 1.5, or re-engineering the lift arm for increased strength.

In addition, Bentaher et al. [16] presented a new instrumentation system for the three-point hitch mechanism of tractors to optimize tillage operations. The system uses only three force transducers to measure the three orthogonal components of tillage force—longitudinal, lateral, and vertical. A key advantage of this system is its ability to detect asymmetrical loading caused by implements like mouldboards, through differences in the lift arm reactions. Calibration results confirmed its accuracy, with errors below 2.8% for vertical force measurements and a maximum of 13% for the lowest force, demonstrating the system's practicality and reliability in field applications.

Device Design

The Tractor (model 5050E, JOHN DEERE) widely used in large and medium horsepower tractors is adopted during the present study. The lifting torque and Speed range of the device used in the test is 126Nm and 540rpm.

Test Requirements	Index
Maximum Weight to Lift	150 kg
Lifting Height	100 feet
Maximum Lifting Torque	126Nm
Speed	2100rpm

Principle of Design:

Based on field observations and insights from relevant literature, the process of lifting submersible pumps from deep bore wells involves two primary stages: the connection of the lifting mechanism to the pump assembly and the subsequent lifting operation. The initial stage includes securely attaching the lifting rope or hook to the pump or pipe. The mechanism designed in the present study is mounted on a tractor and focuses on providing a simple, safe, and efficient solution for lifting pumps without the need for manual excavation or trenching at the bore site. This not only enhances safety but also minimizes site disruption during pump installation and retrieval.

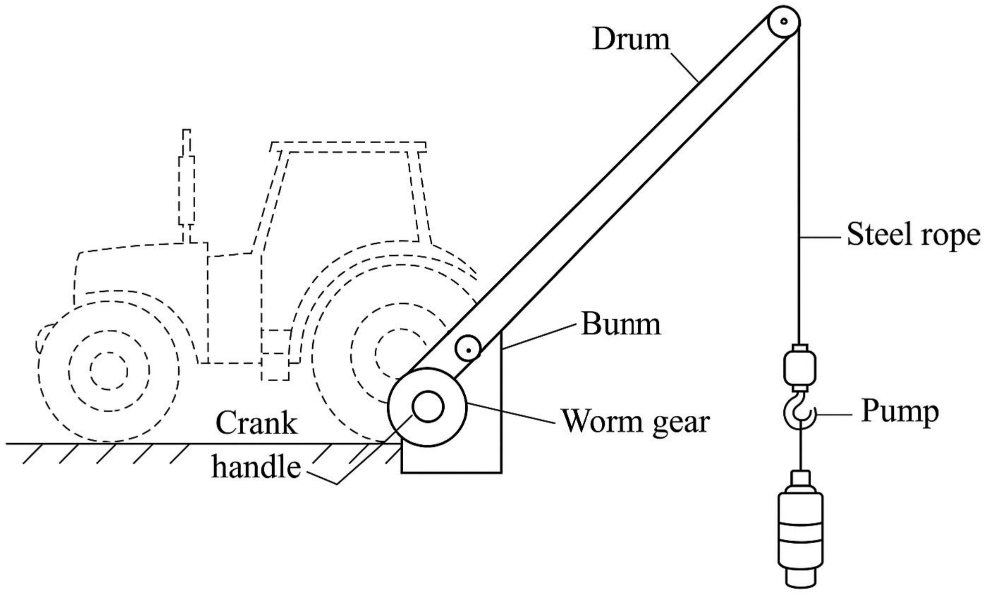


Fig no: 1 Design Assembly of Tractor Mounted Pump Lifting Mechanism

Components to Design: -

PTO Shaft, Worm & Worm Wheel, Bevel Gear, Spline, Hand Lever, Drum, Steel Rope, Pulley, Crane Boom.

Work on Design Data	
Pump to Lift	150Kg (1470.9975N)
Depth	100feet (30.48m)
Tractor Torque	126Nm
Hand Torque	-

Table no 1: Design Data

The weight of the Pump that is transferred through the pulley

Work on the Design Data of Pulley Load	
Coefficient of Friction (μ)	0.3
Tmin	1471N
Contact	135Deg
Tmax	2651.1N

Table no 2: Design Data of Pulley

Rope Material	**Rope Material**	**Rope Material**
Steel rope on steel pulley (dry)	Steel	0.15 to 0.25
Steel rope on steel pulley (oil)	Steel	0.05–0.08
Nylon rope on steel pulley	Steel	0.08 to 0.15
Cotton rope on wood pulley	Wood	0.25-0.4

Table no 3: Steel Rope Friction Factor

$$\frac{T_{max}}{T_{min}} = e^{\mu.\theta}$$

$$\frac{\mathrm{T_{max}}}{1471} = \mathrm{e}^{0.25\times135\times(\frac{3.14}{180})}$$

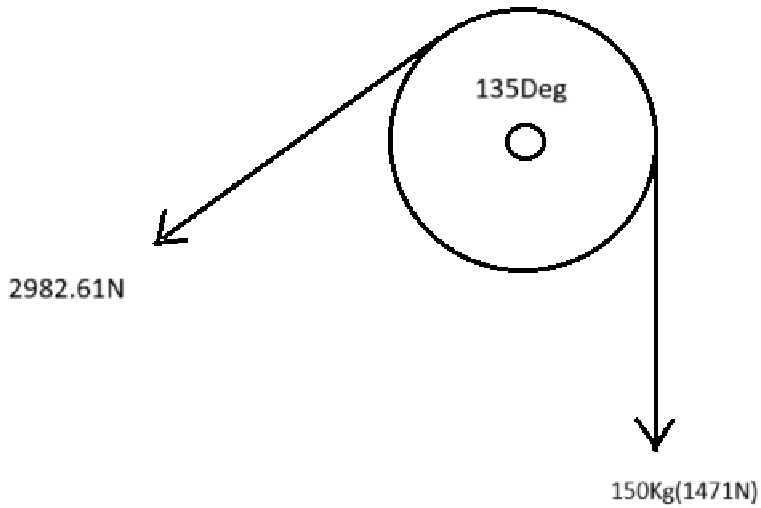


Fig no 2: Friction in Rope and Pulley

1. PTO Shaft: -

Power (KW)	27.6
Speed (rpm)	2100
Torque (Nm)	126

Table no 4: PTO Shaft Design Data

Torque (Nm) = $\frac{9550 \times \text{Power (kW)}}{\text{Speed (rpm)}}$

Torque (Nm) = $\frac{9550 \times 27.6}{2100}$

Torque (Nm) = 126Nm

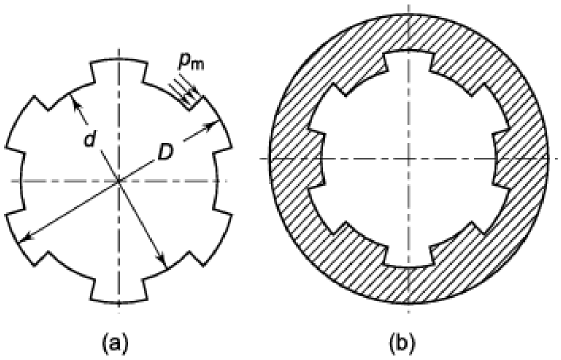
Maximum Length of Shaft:	1447.8 (57")
Minimum Length of Shaft:	889mm (35")

Table no 5: Length of PTO Shaft

Where ,

D = major diameter of splines (mm)	d = minor diameter of splines (mm)
l = length of hub (mm)	n = number of splines
Mt = transmitted torque (N-mm)	pm = permissible pressure on spline (N/mm2)
A = total area of splines (mm2)	Rm = mean radius of splines (mm)

The Permissible pressure on the Splines is Limited to 6.5 N/mm^2



$$M_t = \frac{1}{8} p_m \ln (D^2 - d^2)$$

$$M_t = \frac{1}{8} 6.5 \times 76 \times 6\ (34.87^2 - 27.89^2)$$

M_t = 162303 Nmm << 126000 Nmm

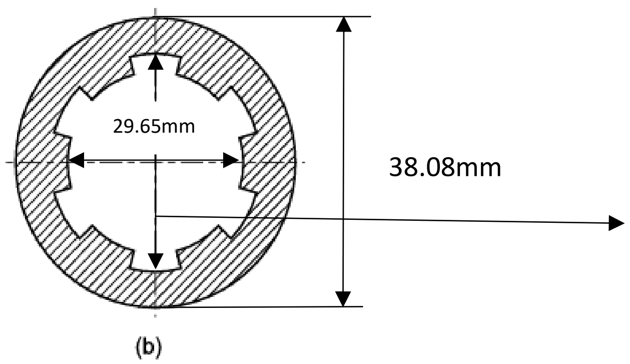
Fig no 3: Spline Dimensions

Assume allowable shear stress τ_{max} = 40 MPa = 40 N/mm2 (typical for mild steel in conservative design)

$$T = \frac{\pi}{16} \times \tau \times \frac{D^4 - d^4}{D}$$

$$126000 = \frac{\pi}{16} \times 40 \times \frac{D^4 - 29.65^4}{D}$$

D = 38.08 mm



D = 34.95mm

Fig no 4: Actual Spline Design for PTO Shaft

Fig no 5: How Actually a PTO Shaft Looks

1 3/8"-Z6

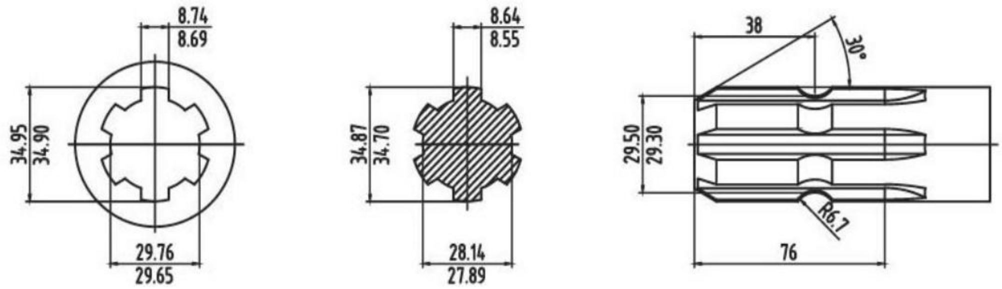


Fig no 6: Tractor Standard PTO Spline Design for 5050E Model

NEAPCO	***Round***	***Rectangle***
	Hexagon	*Square*
WALTERSCHEID	*Lemon Shape*	*Star Shape*
WEASLER	*Hexagon*	*Splined*
	Square	*Rectangular*
BONDIOLI & PAVESI	*4 Lobe*	*2 Lobe*
	Splined	*Free Rotation*

Table no 6: Types of Drive Line

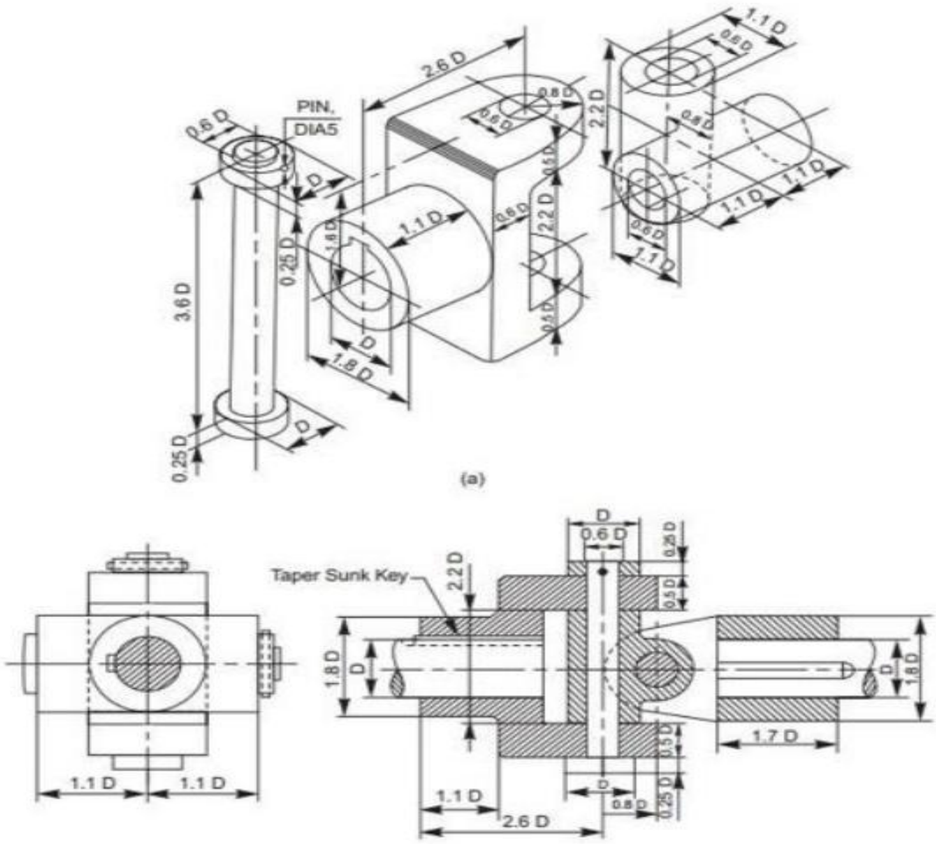


Fig no 7: Dimension Calculation Diagram

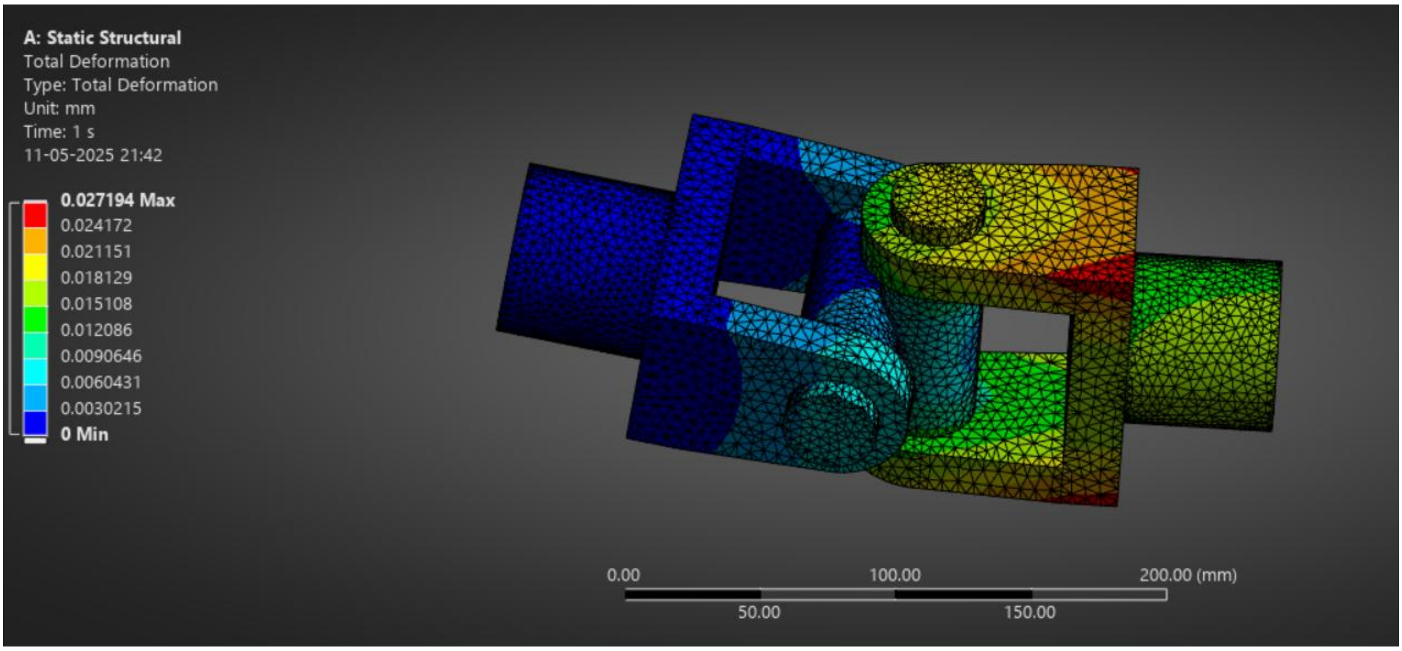


Fig no 8: Simulation Result of Universal Joint

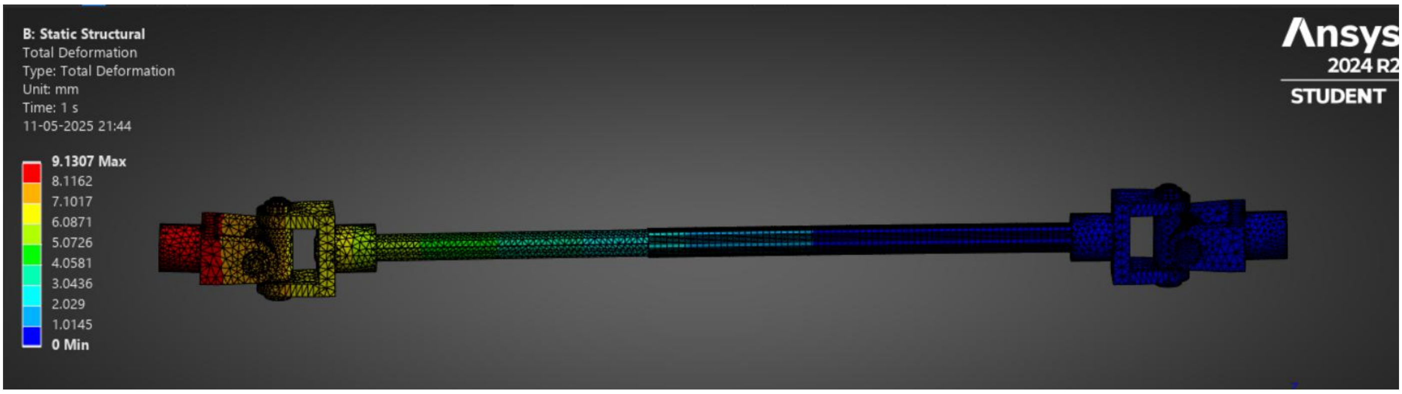


Fig no 9: Simulation Result of Universal Coupling

2. Steel Rope:

Standard Nominal Tensile Strengths of Wire Rope Wires

Grade	Nominal Tensile Strength (N) (MPa)	Common Use
1370	1370 MPa	Light applications, low-load hoisting
1570	1570 MPa	General lifting, moderate stresses
1770	1770 MPa	Heavy lifting, cranes, construction
1960	1960 MPa	Very heavy lifting, elevators, mining
2160	2160 MPa	Special Purpose (e.g. high rise Crane)

Table no 7: Tensile Strength of Steel Wire based on Grade

Steel Wire Rope Design Calculation Process

1. **Determine Required Load Capacity**
   The first step is to calculate the Minimum Breaking Load (MBL) based on the working load.

   MBL = SF × WLL

   Where:

   MBL = Minimum Breaking Load (kN)

   SF = Safety Factor (typically 5 to 8 for lifting applications)

   WLL = Working Load Limit (kN)

   MBL = 2651.1 × 5 = 13.26KN

2. **Select Wire Rope Diameter**
   The approximate breaking strength (BS) of a steel wire rope is calculated using the empirical formula:

   BS = 0.38 × d^2 × N
   24.21 = 0.38 × d^2 × 1570
   d = 6 mm
   BS > MBL

3. **Calculate Rope Weight**
   The weight of the rope per meter can be estimated as
   W = 0.034 × d^2
   Where W is the weight in **kg/m** and is the diameter in **mm**.
   W = 0.034 × 6^2
   W = 1.224kg/m

4. **Calculate Fatigue Life:**

Number of Bends Per Cycle:
1 bend over the pulley
1 bend while winding on the drum
Total bends per cycle = 2

Empirical Fatigue Constants (for wire rope, typical values from ISO 4309 & ASME B30.9):
C = 10^7(for steel rope, empirical constant)
b = 3 (for bending fatigue)

$$N_f = \mathrm{C} \times (\frac{D}{d})$$

For Pulley bending:

$$N_f = 10^7 \times \left(\frac{200}{6}\right)$$

$$N_f = 3 \times 10^8 Cycle$$

For Drum Winding

$$N_f = 10^7 \times \left(\frac{200}{6}\right)^3$$

$$N_f = 3.70 \times 10^{11}\ Cycle$$

Both Combine

$$\frac{1}{N_f} = \frac{1}{N_{fp}} + \frac{1}{N_{fd}}$$

$$\frac{1}{N_f} = \frac{1}{3 \times 10^8} + \frac{1}{3.70 \times 10^{11}}$$

N_f= 2.997 × 10^8 cycles

The expected fatigue life is approximately 300 million cycles.
This is a theoretical value assuming ideal conditions. In real-world applications, abrasion, corrosion, and dynamic loading reduce fatigue life significantly.

Type	Wire per Strand	Total Wires	Flexibility	Abrasion Resistance	Fatigue Resistance	Typical Uses
6×7	7	42	Low (Stiff)	Very High	Low	Guy Wires, Low Bending
6×19	19	118	Medium	High	Medium	Cranes, Pump Lifting
6×25	25	150	Medium-High	Medium	Good	Elevators, Crane, Lifting
6×36	36	210	High	Medium - Low	High	Mobile cranes, Winches, Pulleys
7×19	19	197	Very High	Low	High	Aircraft cables, garage doors
8×19	19	152	Very High	Medium – Low	High	Mine hoists, marine uses
8×36	36	288	Very High	Low	Very High	Slings, heavy cranes, drag lines.

Table no 8: Types of Steel Wire and its Applications

Core Type	Description	Usage
FC (Fiber Core)	Natural/synthetic fiber core; more flexible, but lower strength.	Light-duty applications.
IWRC (Independent Wire Rope Core)	Steel strand core; higher strength and heat resistance.	Heavy-duty lifting, cranes, high-load situations.

Table no 9: Core Type Steel Rope

Core Selection:

- **Fiber Core (FC):** Made from natural or synthetic fibers, offering flexibility but less strength.
- **Steel Core:**
    - **Wire Strand Core (WSC):** Consists of a single steel strand.
    - **Independent Wire Rope Core (IWRC):** Comprises a small wire rope, enhancing strength and resistance to crushing.

Considerations:

- **Fiber Core:** Suitable for applications requiring flexibility and where the rope isn't subjected to high temperatures or crushing forces.
- **Steel Core:** Ideal for heavy-duty applications, providing higher strength and better resistance to compression and high temperatures.

Based on Rope Diameter Proportions

These are rough empirical rules derived from wire rope design standards (e.g., ASTM, ISO):

Component	Formula	For 6 mm Rope
Strand Diameter	0.65 × D	0.65 × 6 = **3.9 mm**
Center Wire	0.20 × D	0.20 × 6 = **1.2 mm**
Outer Wire	0.18 × D	0.18 × 6 = **1.08 mm**
Inner Wire	0.14 × D	0.14 × 6 = **0.84 mm**
IWRC Core Diameter	0.4 × D	0.4 × 6 = **2.4 mm**

Table no 10: Wire Thickness Calculation

This is a common layout for a 19-wire strand:

- 1 center wire
- 6 wires in the first layer
- 12 wires in the outer layer

Total = 1 + 6 + 12 = 19 wires per strand

So for 6 strands, you'll have:

- 6 × (1 + 6 + 12) = 114 wires

7×7 means:

- 7 strands
- Each strand has 7 wires (1 center + 6 around)
- **Total: 49 wires**

Component	Description
Outer Rope	6 strands × 19 wires = 114 wires
IWRC (core)	7×7 = 49 wires
Total wires	114 + 49 = **163 wires**
Core diameter	≈ 2.4 mm

Table no 11: Wire of Rope

Step 1: Estimate strand diameter in 7×7

- Strand count = 7
- Typical rope structure: rope diameter ≈ 3 × wire diameter (for small constructions)
- More accurately, strand diameter ≈ 0.65 × IWRC diameter

Strand diameter ≈ 0.65 × 2.4 ≈ 1.56mm

Step 2: Estimate wire diameter

- Each strand has 7 wires in 1+6 pattern
- Strand diameter ≈ 3 × wire diameter (approx for 1+6)

$$Wire\ Diameter = \frac{1.56}{3} = 0.52mm$$

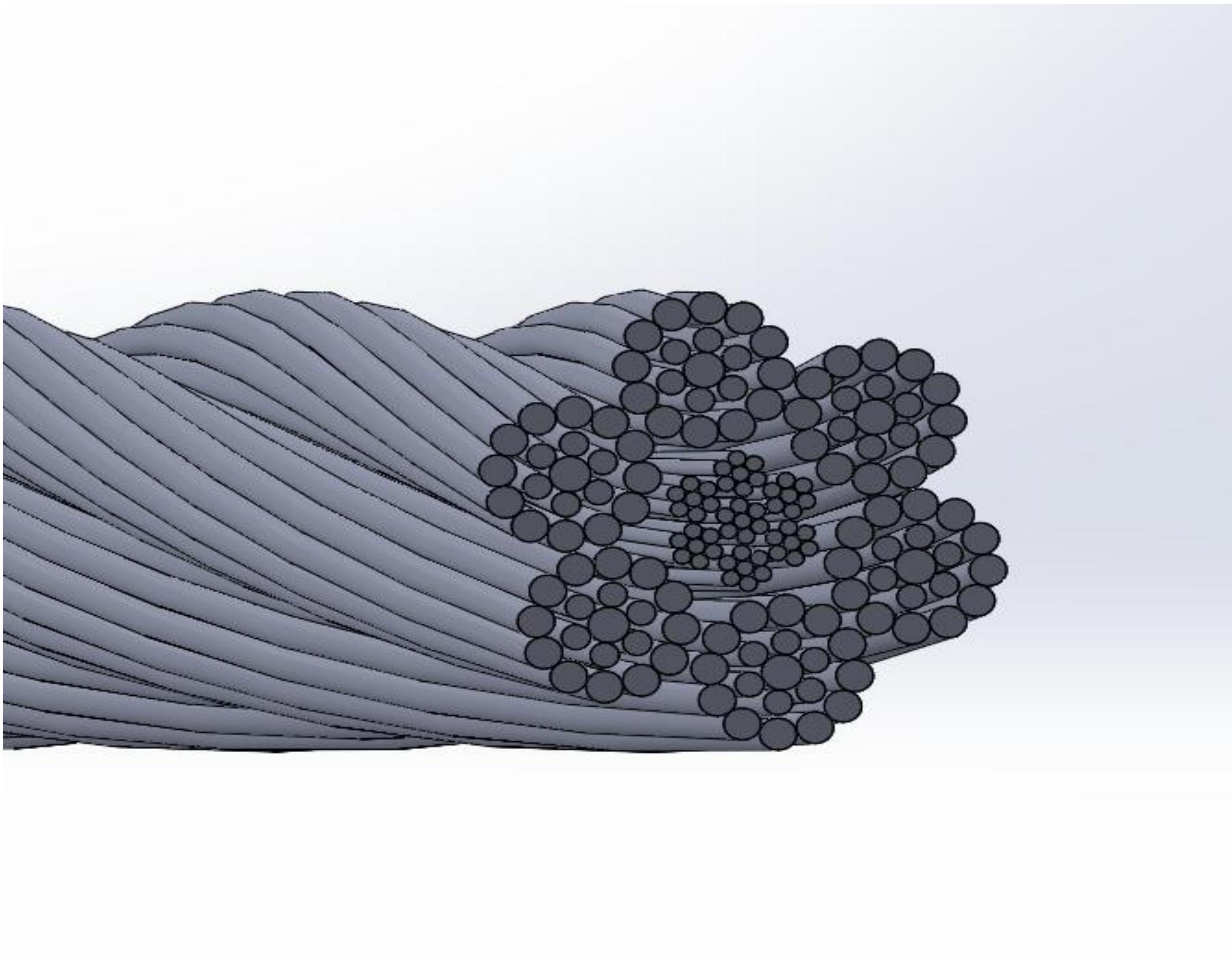


Fig no 10: Design of Steel Rope

3. Bevel Gear Design:

1. Design Requirements

Input Torque = 126 Nm

Output Torque = 126 Nm

Gear Ratio = 1:1

Shaft Angle = 90°

Material: AISI 1045 Steel, Allowable Bending Stress ≈ 100MPa

Service Factor: 1.25 (For Moderate Load)

2. Select Number of Teeth

To avoid undercutting, minimum teeth on pinion (Straight Bevel) = 16

Pinion Teeth: $Z_1 = 20$

Gear Teeth: $Z_2 = 20$

3. Determine Pitch Angles

For Equal Gear sizes and 90° Shaft angle.

$\Delta_1 = \Delta_2 = 45°$

4. Section Modulus

We'll choose a modulus that support 126 Nm Safely

Try module m = 5mm

Then Pitch Diameter:

$\mathrm{d} = \mathrm{m} \times \mathrm{Z} = 5 \times 20 = 100\ \mathrm{mm}$

Tangential radius: $\mathrm{r} = 50\ \mathrm{mm} = 0.05\ \mathrm{m}$

$F_t = \frac{126}{0.05} = 2520\ \mathrm{N}$

5. Face Width (b)

Standard Face Width:

$\mathrm{b} = \frac{1}{3}d = \frac{1}{5} \times 100 = 20\mathrm{mm}$

6. Check Bending Strength

Using

$$\boldsymbol{\sigma} = \frac{F_t}{b \times m \times Y} = \frac{2520}{20 \times 5 \times 0.5} = 50.4 \text{ MPa}$$

7. Addendum (a)

$1 \times \text{m} = 5\text{mm}$

8. Dedendum (da)

$3.25.1 \times \text{m} = 6.25\text{mm}$

9. Outer diameter (da)

$\text{d} + 2\text{a} = 110\text{mm}$

10. Root Diameter (dr)

$\text{dr} = \text{d} - 2\text{da} = 100 - 2 \times 6.25 = 87.5\text{mm}$

11. Cone Distance ®

$$\text{R} = \frac{d}{2sin\Delta} = \frac{100}{2sin45°} = 70.71\text{mm}$$

12. Back Cone angle (β)

. $\beta = 90° - \Delta = 45°$

13. Face Angle

$$\Delta + \tan^{-1}\frac{\text{b}}{\text{R}} = 45° + \tan^{-1}\frac{20}{70.71} = 61°$$

14. Root Angle

$$\Delta - \tan^{-1}\frac{b}{R} = 45° - \tan^{-1}\frac{20}{70.71} = 29°$$

15. Clearance ©

$0.25 \times \text{m} = 0.25 \times 5 = 1.25\text{mm}$

Shaft Hole Design

Based on Torque

$$d_{shaft} = (\frac{16T}{\pi \times \tau_{allow}})^{1/3}$$

Assume allowable shear stress τ = 40 MPa

$$d_{shaft} = (\frac{16 \times 126}{\pi \times 40 \times 10^6})^{1/3} = 21.5\text{mm}$$

Using Shaft hole = 25mm Diameter

Keyway Dimensions (per ISO 773)

For 25 mm Shaft

Width = 6 mm

Depth = 3.3 mm

Key Length = 20-30 mm

Hub Diameter

45 mm or More

Hub Length

Equal to Shaft Diameter

25-30mm

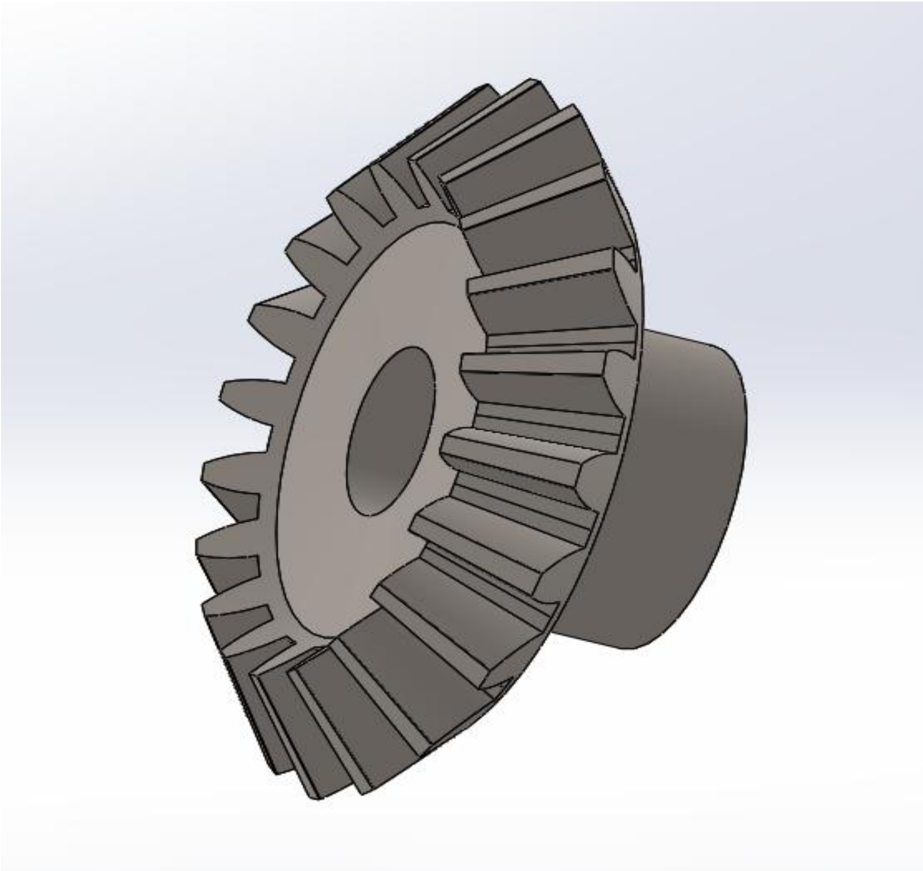


Fig no 11: Design of Bevel Gear

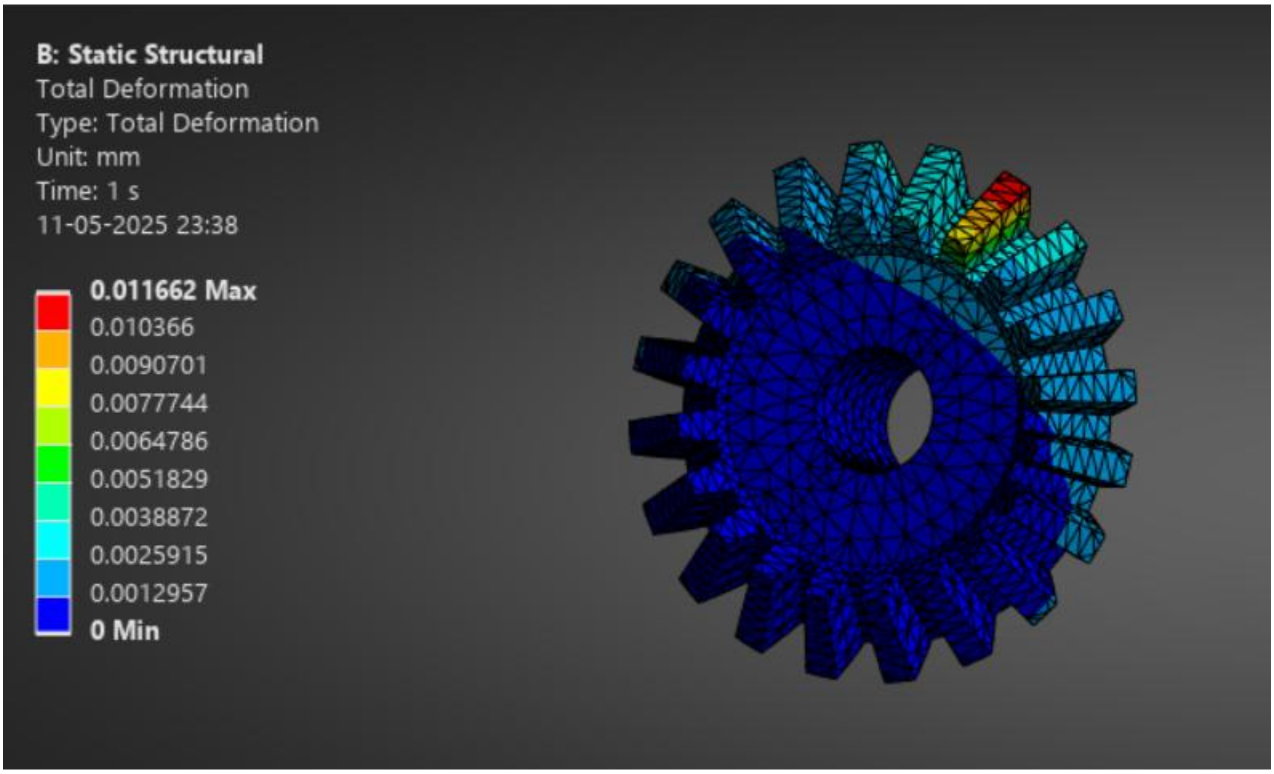


Fig no 12: Simulation Result of Bevel Gear

4. Drum Design:

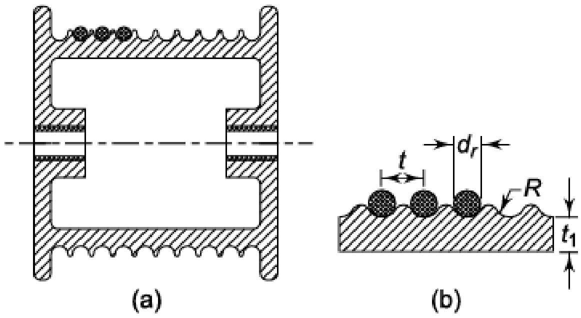


Fig no 13: Drum Design

Rope Diameter of 10 mm (dr)

t = d_r + (2 to 3 mm)

t_1 = 0.02 D + (6 to 10 mm)

Groove radius = 1.05 (nominal rope radius)

D = Drum Diameter (mm)

Data:

Drum Diameter (D) = 200 mm

Drum Width (W) = 300 mm

Rope Diamter ($d_\circ$) $= 10$ mm

t = 10 + 2 = 12 mm

t_1 = 0.02 × 200 + 6 = 10 mm

Torsional Shear Stress in Shaft

F = 2651 N – if we Consider Friction

T = 2651 N × 0.1 m = 265.1 Nm

Groove radius = 1.05 × 10 = 10.5mm

If shaft Diameter is 25 mm

$\tau_{torsional} = \frac{16T}{\pi d^3} = \frac{16 \times 265.1 \times 1000}{\pi \times 25^3}$ = 86.40 Nmm

To use Material as Medium Carbon and Alloy Steel.

For Bending Stress for 25 mm diameter Shaft

$\sigma_{\text{bending}} = \frac{32 \times M}{\pi \times d^3}$

M = 397.65 Nm = 397650 Nmm

d = 25 mm

$\sigma_{\text{bending}} = \frac{32 \times 397650}{\pi \times 25^3}$ = 259.3 MPa

For this Bending Stress Material should be Used is Medium Carbon and Alloy Steel

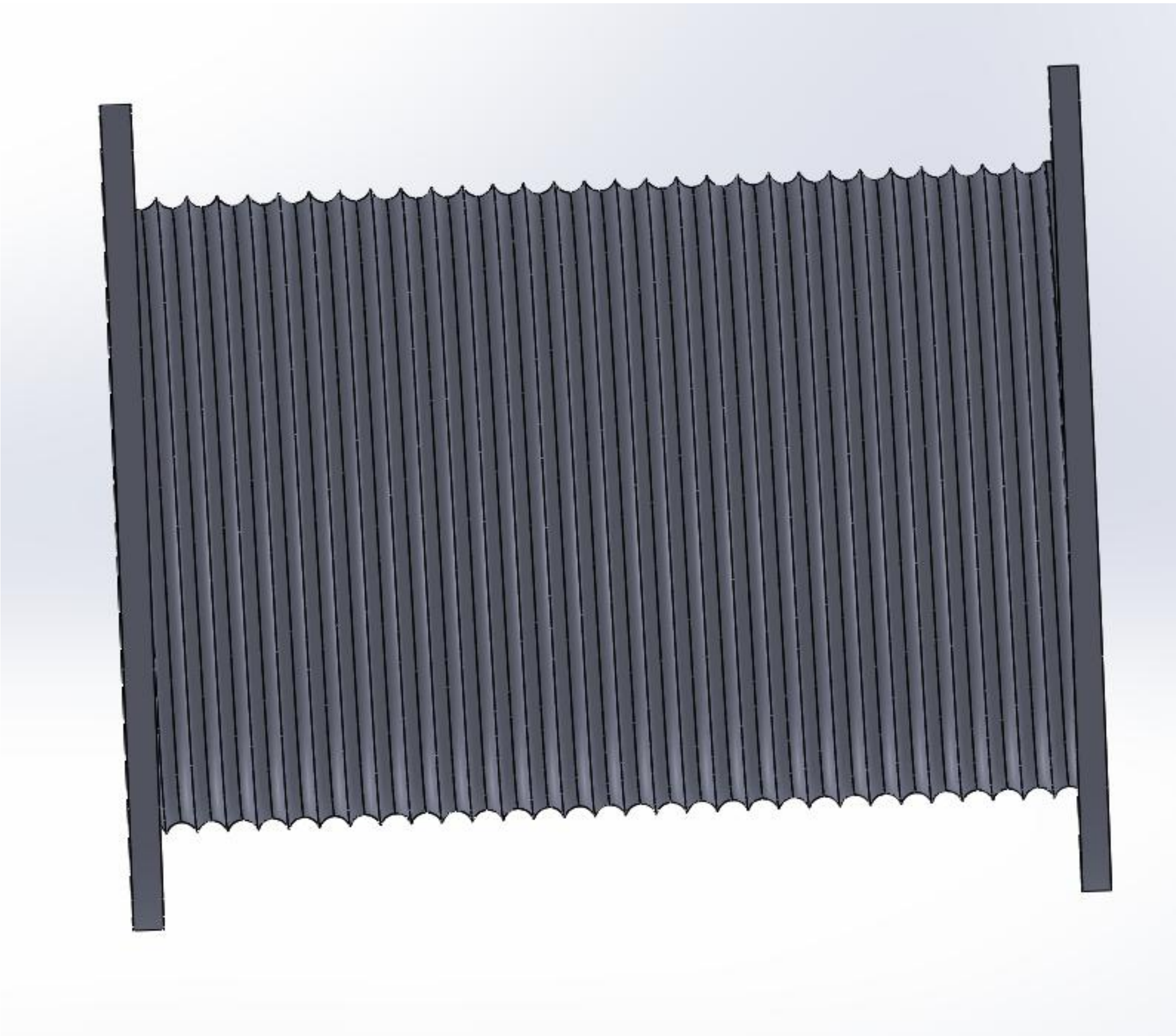


Fig no 14: Design of Drum

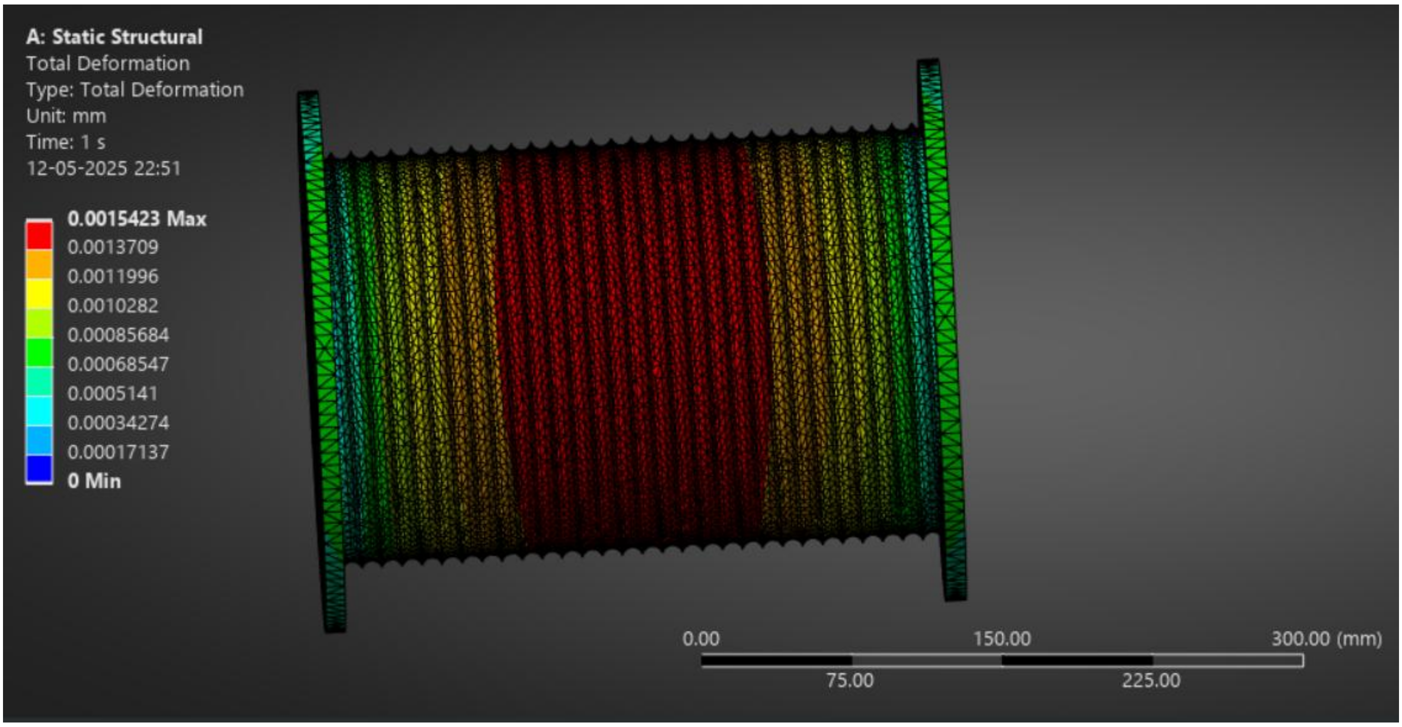


Fig no 15: Simulation Result of Steel Rope Drum

5. Pulley Design: -

Basic Parameters:

Load (W) = 153 × 9.81 = 1501.9 N

Rope Diameter (d_0) = 10 mm (Wire Rope)

Minimum Drum/Pulley Diameter (D_p) >= 20 × d_0

Pulley Diameter (D_p) = 20 × 10 mm = 200 mm

Pulley Groove

Groove Angle = 40° to 45°

Groove Radius R = 0.53 × Rope Diameter = 0.53 × 10 = 5.3 mm

Groove Depth = 1.5 × Rope Diameter = 1.5 × 10 = 15 mm

Pulley Width (b)

b = Rope Diameter + 2 × Flange Thickness + Clearance

Suggestion: b = 35 to 50 mm (Safe for 10 mm rope)

Shaft: - 25mm – Diameter

Material: - Mild Steel or CI

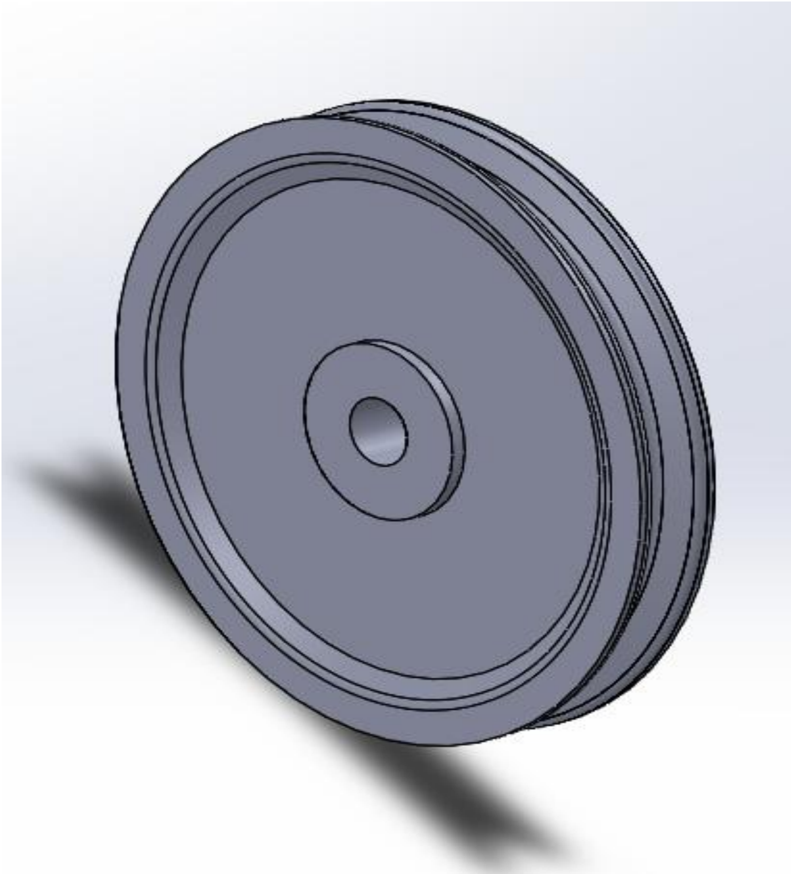


Fig no 16: Design of Pulley

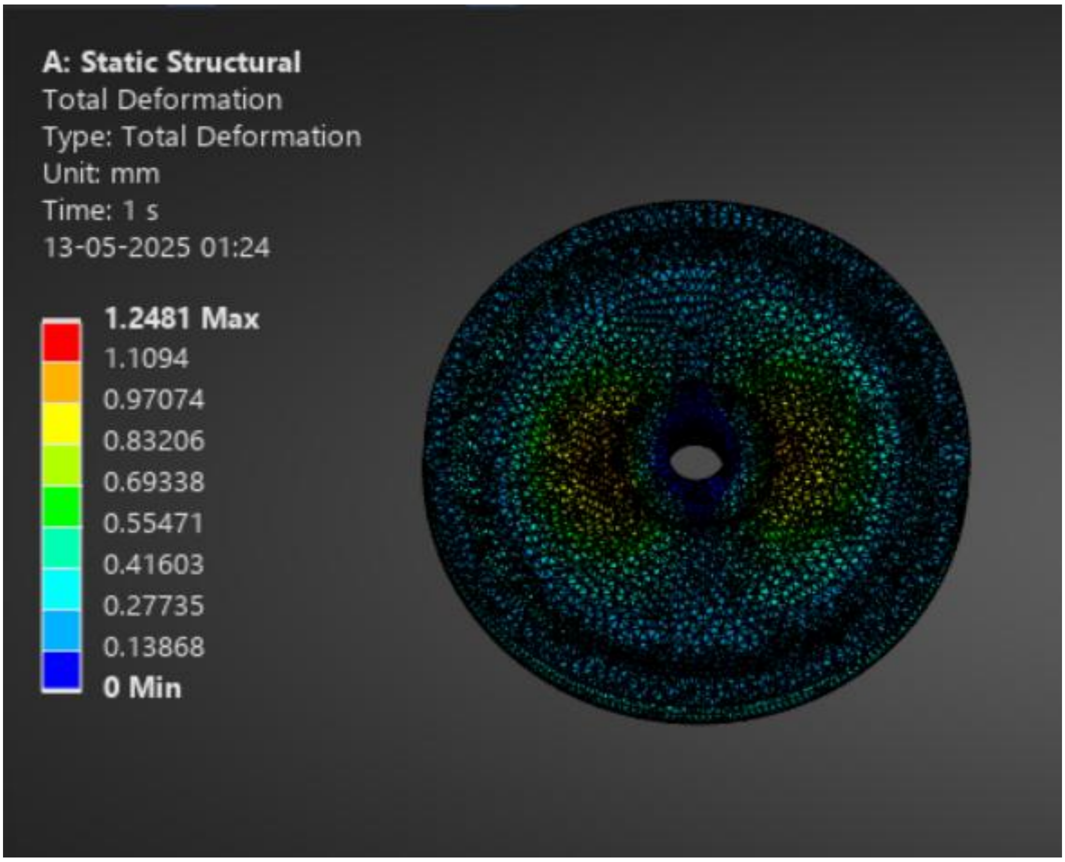


Fig no 17: Simulation Result of Pulley

6. Pulley Shaft:

Requirement: -

Length = 200 mm

Load = 153 Kg = 153 × 9.81 = 1502 N

Load Applied at Centre

Material: - Mild Steel (Yield Strength = 250 MPa)

Safety Factor: 3

To Find Minimum Safe Diameter to resist:

1.Bending Stress

2.Shear Stress

1. **Bending Moment at Centre:**

M = $\frac{F \times L}{A}$ = $\frac{1502 \times 0.2}{4}$ = 75.1 Nm

2. **Section Modulus for a Circular Rod:**

Z = $\frac{\pi d^3}{32}$

3. **Bending Stress:**

$\sigma_b = \frac{M}{Z} = \frac{32\,M}{\pi d^3} = \frac{32 \times 75.1}{\pi d^3} = \frac{2403.2}{\pi d^3}$

4. **Shear Stress:**

$\tau = \frac{4 \times F}{3 \times \pi \times d^2} = \frac{4 \times 1502}{3 \times \pi \times d^2} = \frac{2002.7}{\pi \times d^2}$

5. **Combined Stress Check**

$\sigma_{eq} = \sqrt{\sigma_b^2 + 3\tau^2}$ <= $\frac{\sigma_{yield}}{SF}$ = $\frac{250}{3}$ = 83.3 MPa

Try d = 20 mm

$\sigma_b = \frac{2403.2}{\pi \times 20^3}$ = 9.55 MPa

$\tau = \frac{2002.7}{\pi \times 20^2}$ = 1.59 MPa

$\sigma_{eq} = \sqrt{(9.55)^2 + 3 \times (1.59)^2}$ = $\sqrt{91.2 + 7.6}$ = 9.98 MPa

Well Below 83.3 MPa

Minimum safe Diameter = 20mm

We take as 25 mm

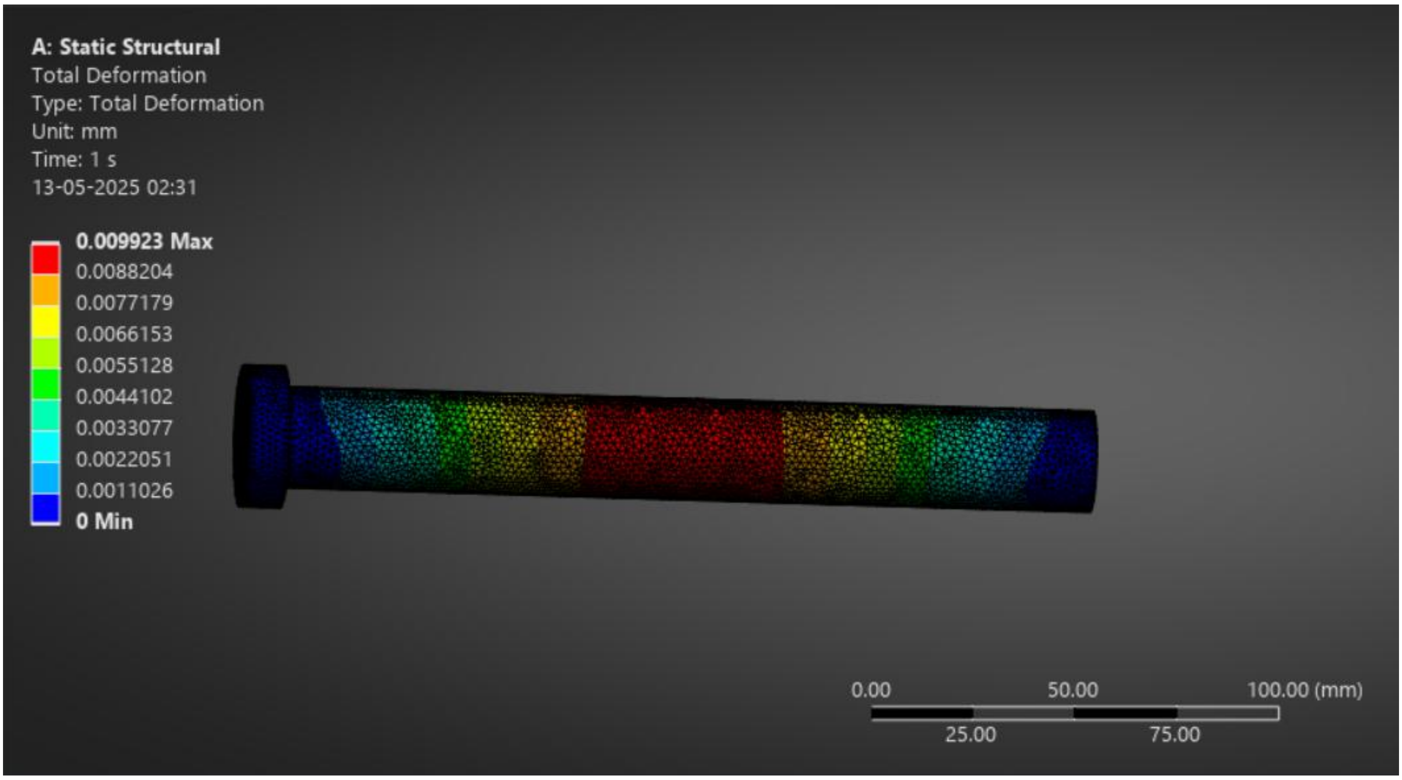


Fig no 18: Simulation Result of Pulley Shaft

7. Worm And Worm Wheel Design: -

To design a worm and worm wheel system where the output torque on the worm wheel is 126 Nm

1. Define Gear Ratio (i)
$$i = \frac{\mathrm{T}}{\mathrm{Z}}$$
For Single Start Worm (Z = 1), i = T
Lets Assume I = 30 (Common for Lifting Applications)

2. Find input Torque on Worm
$$T_{worm} = \frac{T_{wheel}}{i.\eta}$$

$$T_{wheel} = 126Nm$$

I = 30

η = Efficiency (typical value of worm gears 0.3 to 0.5; we will use 0.4 for conservation estimate)

$$T_{worm} = \frac{126}{30.0.4} = \frac{126}{12} = 10.5Nm$$

Material
Worm = Hardened Steel
Worm Wheel = Bronze or Cast Iron

Design Calculation

1. Gear Ratio and Teeth Selection
Worm Starts = 1
Worm Wheel Teeth, T = i* 1 = 30 Teeth

Module (m) Selection
To transmit 126Nm torque
$$m = \sqrt[3]{\frac{2.T_{wheel}}{\varphi.\sigma allow}}$$

Assume,
Face width factor, φ = 10
Assume Bending Stress = 100MPa

$$m = \sqrt[3]{\frac{2.126*10^3}{10.100.10^6}} = 0.63$$

Use Standard module m = 2mm

Parameter	Formula / Assumption	Value
Module (m)	Selected standard	2 mm
No. of Teeth (T)	Worm wheel	30
Starts on Worm (Z)		1
Pitch Diameter of Wheel	$d_w = m \cdot T$	$2 \cdot 30 = 60$
Pitch Diameter of Worm	$d_1 = m \cdot Z \cdot q$	Let $q = 10 \Rightarrow$ $2 \cdot 1 \cdot 10 = 20$
Lead Angle (λ)	$\tan\lambda = Z \cdot m / \pi d_1$	$\tan^{-1}(0.0318) \approx 1.82^\circ$
Center Distance (a)	$a = d_1 + d_w / 2$	(20 + 60)/2 = 40 mm
Face Width of Worm Wheel	$b = (9-10) \cdot m$	18–20 mm
Length of Worm	$l = (Z+2) \cdot m = 3 \cdot 2$	6 mm (min)

Component	Dimension
Module (m)	2 mm
Teeth on Wheel	30
Worm Starts	1
Worm dia	20 mm
Wheel dia	60 mm
Center Distance	40 mm
Face Width	18–20 mm

8. Design of Crane Boom

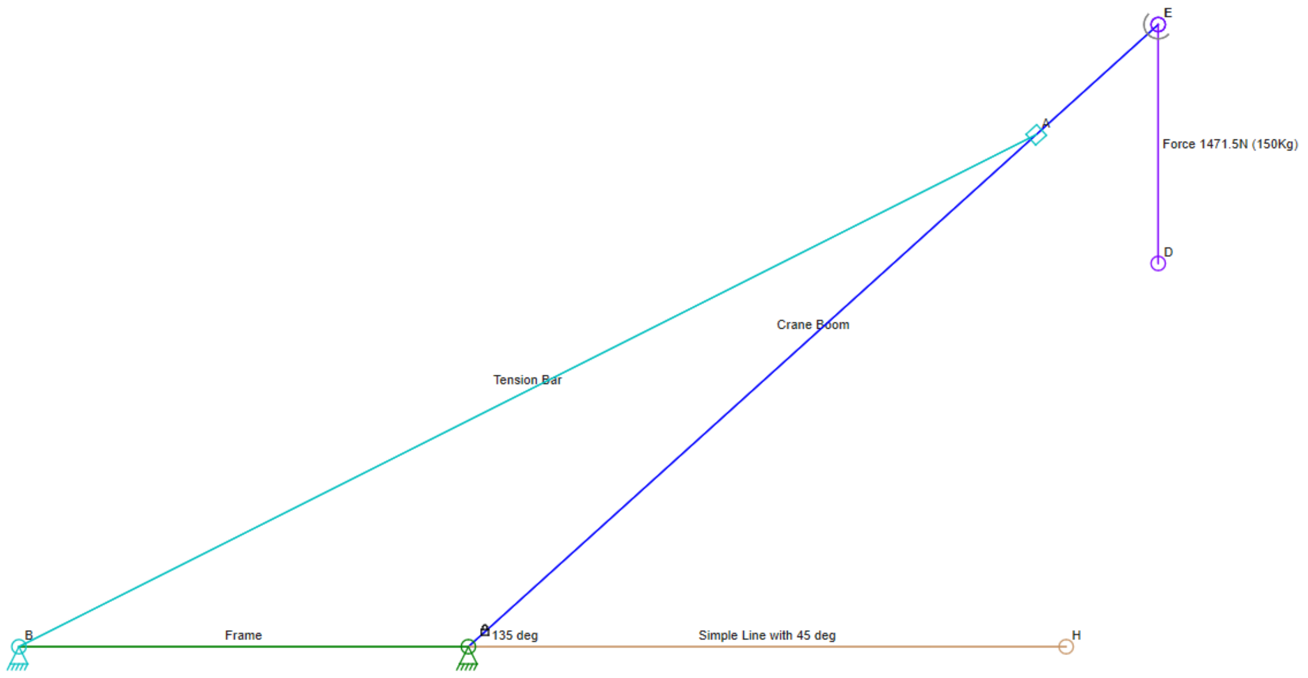


Figure is Rotated 45 degree Clockwise

Fig no 19: Boom Crane Design

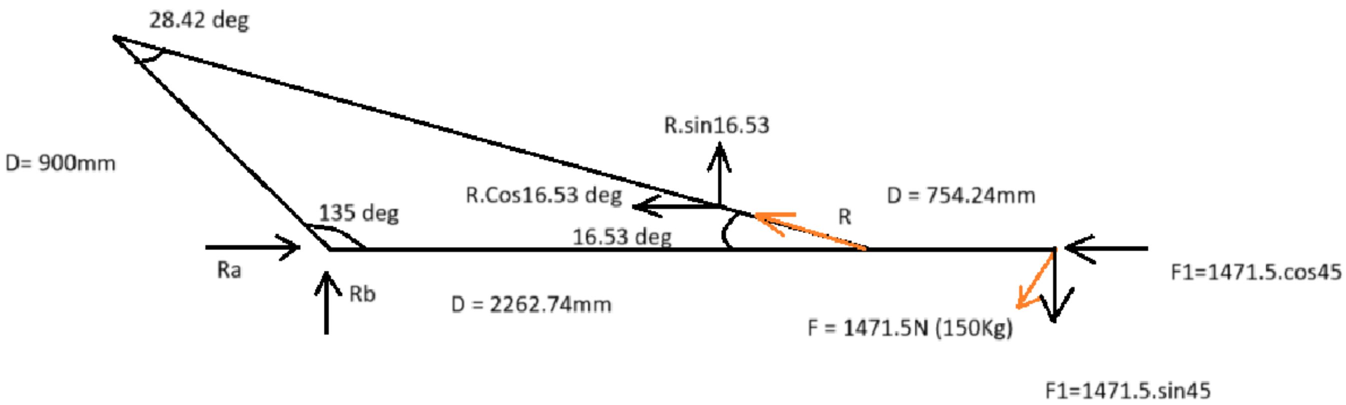


Fig no 20: Resultant Force Calculation

Moment at point C

$\sum M_c = (1471.5 \times \cos 45) \times 2262.74 - (R \times \sin 16.53) \times (2262.74 - 754.24)$

$\sum M_c = 1040.51 \times 2262.74 - (R \times \sin 16.53) \times 1508.5$

$= 2354403.59 - R \times 429.1944$

$R = 5485.67N$

Forces:

R × Sin16.53 = 1560.77N

R × cos16.53 = 5258.95N

Boom Design Calculation:

1. Structural Analysis - Bending Moment & Shear

It is a Cantilever Beam (X = 0)
At this Step (X = 1508.49mm)
Moment due to 1040.51 N vertical load at x = 1508.49 + 754.24 = 2262.73 mm:
M = 1040.51 × (2262.73 – 1508.49) = 1040.51 × 754.24 = 785255.9N/mm

Moment due to 1560.77 N upward force at x = 1508.49 mm:

M = −1560.77 × 0 = 0 (applied at pivot)

Moment due to horizontal 1040.51 N has no vertical moment effect

So, maximum bending moment at fixed support:

M_{max} = 785255.9N/mm = 785.26N/m

2. Material properties (Assume it is Steel)

Parameter	Symbol	Value
Yield Strength	f_y	250Mpa
Elastic Modulus	E	200Gpa
Allowable Stress	σ_{allow}	150Mpa

Table no 12: Material Property for Boom Design

3. Section Modulus Required

$Z_{req} = \frac{M_{max}}{\sigma_{allow}} = \frac{785255.9}{150} = 5235 \text{ mm}^3$

4. Select Rectangular Section

$Z = \frac{bh^2}{6} > 5235$
Assume width b = 50mm

$\frac{50.h^2}{6} = 5235 = h^2 = \frac{6.5235}{50} = 628.2 = h = \sqrt{628.}2 = 25.06mm$

This is very small, so increase depth for stiffness and deflection control.
Let's Choose:
b = 50mm
h = 100mm

$Z = \frac{50.100^2}{6} = 83333mm^3$

Much greater than Required so this Section is Safe in Bending.

5. Shear Stress Check

Shear force max = 5258.95N

Shear area: A = b. h = 50 × 100 = 5000mm^2

$$T_{avg} = \frac{V}{A} = \frac{5258.95}{5000} = 1.0518MPa << 150MPa$$

Safe in Shear

6. Deflection Check

$$\Delta = \frac{PL^3}{3EL}$$

P = 1560.77N
L = 1508.493 mm
$I = \frac{bh^3}{12} = \frac{50.100^3}{12} = 4.17 \times 10^6\ mm^4$

$$\Delta = \frac{1560.77.1508.493^3}{3 \times 200000 \times 4.17 \times 10^6} = 2.14mm$$

Allowable(L/180) = 1508.493/180 = 8.380mm

$$\Delta = \frac{PL^3}{3EL}$$

P = 1040.51N
L = 2262.73 mm
$I = \frac{bh^3}{12} = \frac{50.100^3}{12} = 4.17 \times 10^6\ mm^4$

$$\Delta = \frac{1040.51 \times 2262.73^3}{3 \times 200000 \times 4.17 \times 10^6} = 4.82mm$$

Allowable (L/180) = 2262.73/180 = 12.57mm
Safe in Deflection

7. Bending Stress Check

Max Moment

M_{max} = 785.26N/m = 785260N/mm

Section Modulus:

$$Z = \frac{bh^2}{6} = \frac{50\times100^2}{6} = 83333 \text{ mm}^3$$

Bending Stress:

$$\sigma_b = \frac{M}{Z} = \frac{785260}{83333} = 9.42 \text{ MPa}$$

Compare with allowable

Mild Steel Yield: f_y = 250 Mpa

Allowable: σ_{allow} = 150 Mpa

Bending Stress is Safe

8. Shear Stress Check

Max Shear: V_{max} = 1508.493N

Shear area: A = b × h = 50 × 100 = 5000mm^2

Average Shear Stress:

$$\tau_{avg} = \frac{V}{A} = \frac{1508.493}{5000} = 0.302 \text{ MPa}$$

Shear Stress is Safe

9. Axial Compressive Stress:

N = 1040.507 + 5258.95 ≈ 6300

Area: A = 50 × 100 = 5000 mm^2

$$\sigma_c = \frac{N}{A} = \frac{6300}{5000} = 1.259 \text{ MPa}$$

Compare to yield: $f_y = 250$ Mpa

Compressive stress is very Small – Safe

10.Combined Stress:

$$\frac{\sigma_b}{f_y} + \frac{\sigma_c}{f_y} = \frac{9.42}{250} + \frac{1.259}{250} = 0.0427$$

$$0.0427 < 1.0 = \text{Safe}$$

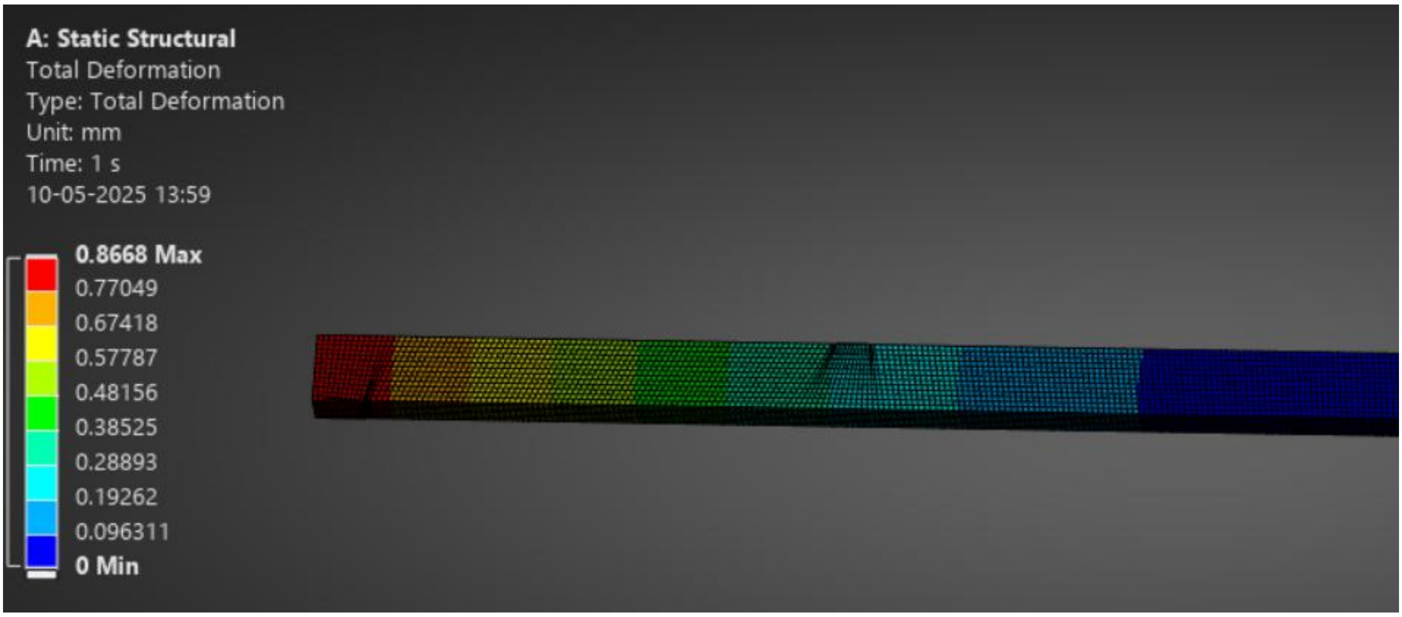


Fig no 21: Simulation Result of Crane Boom

11.Frame Design:

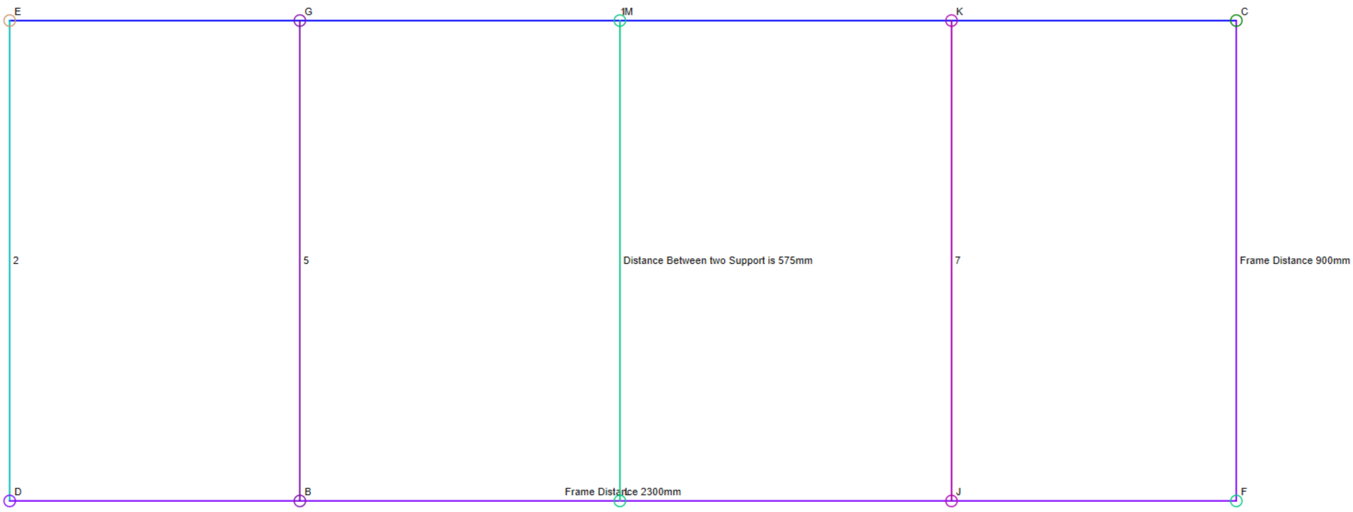


Fig no 22: Frame Design

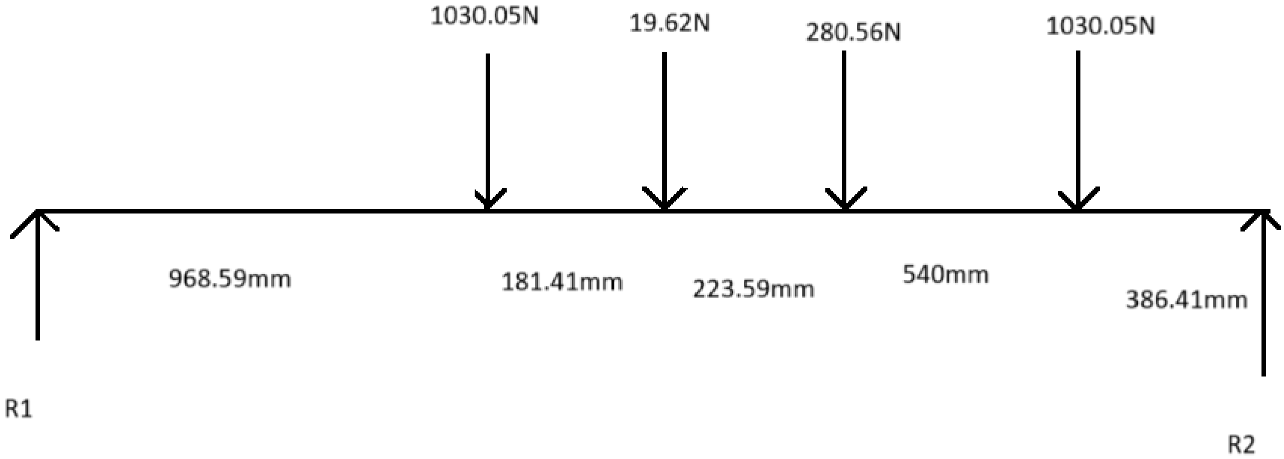


Fig no 23: Resultant Force Calculation

Material	Yield Strength (Fy)	Elastic modulus (E)
Steel	250 MPa	200000 MPa

Table no 13: Material Property for Frame Design

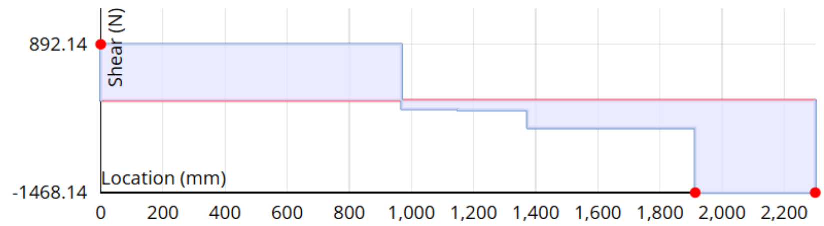
Shear Diagram

(Max +ve)Shear Load (N): 892.138,

Location (mm): 0.000

(Max -ve)Shear Load (N): -1468.142,

Location (mm): 1913.590, 2300.000



Moment Diagram

(Max +ve)Moment Load (N-mm): 864115.811,

Location (mm): 968.590

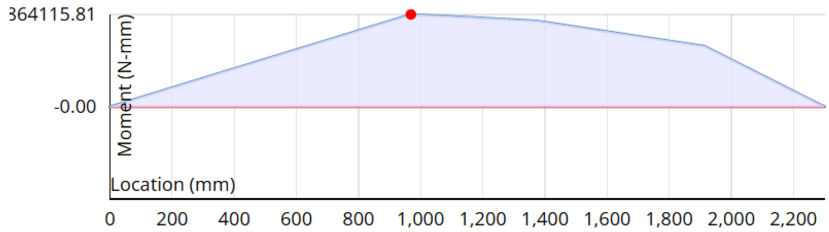


Fig no 24: SFD and BMD

1. Finding Reaction and Moment

$\sum \mathbf{M}_{R1}$ =(1030.05 × 968.59) + (19.62 × 1150) + (280.56 × 1373.59) + (1030.05 × 1913.59) – (R_2×2300)

$R_2 = \frac{3376726.919}{2300} = 1468.14N$

R_1 = (2360.28-1468.14) = 892.14N

Maximum Bending Moment = 864115.811 N.mm

2. Beam Section Design

Assume b = 50mm

Finding required height h:

$\boldsymbol{\sigma}_b = \frac{M}{Z}$, $Z = \frac{bh^2}{6}$, $h = \sqrt{\frac{6M}{b \times \sigma_{allow}}}$

$\boldsymbol{\sigma}_{allow}$ = 150MPa

$h = \sqrt{\frac{6 \times 864115.811}{50 \times 150}} = \sqrt{691.292} = 26.29mm$

This height is surprisingly small due to low loads — we'll increase to 60 mm height for stiffness and safety margin.

3. Check Stress with Final Dimensions

Use:

b = 50mm

h = 60mm

Section Modulus:

Z = $\frac{bh^2}{6}$ = $\frac{50\times60^2}{6}$ = 30000mm^3

Maximum Bending Stress:

σ_b = $\frac{M_{max}}{Z}$ = $\frac{864115.811}{30000}$ = 28.80 MPa << 250 MPa (Safe)

4. Shear Force and Stress

Maximum Shear Force:

V_{max} = 892.14 N

Shear Stress:

τ = $\frac{V}{A}$ = $\frac{892.14}{50\times60}$ = 0.297 MPa << 145 MPa (Safe)

5. Deflection

Maximum Deflection:

Δ_{max} = 7.56mm – (at 2300mm)

Δ_{allow} = $\frac{L}{180}$ = $\frac{2300}{180} = 12.78\text{mm (Safe)}$

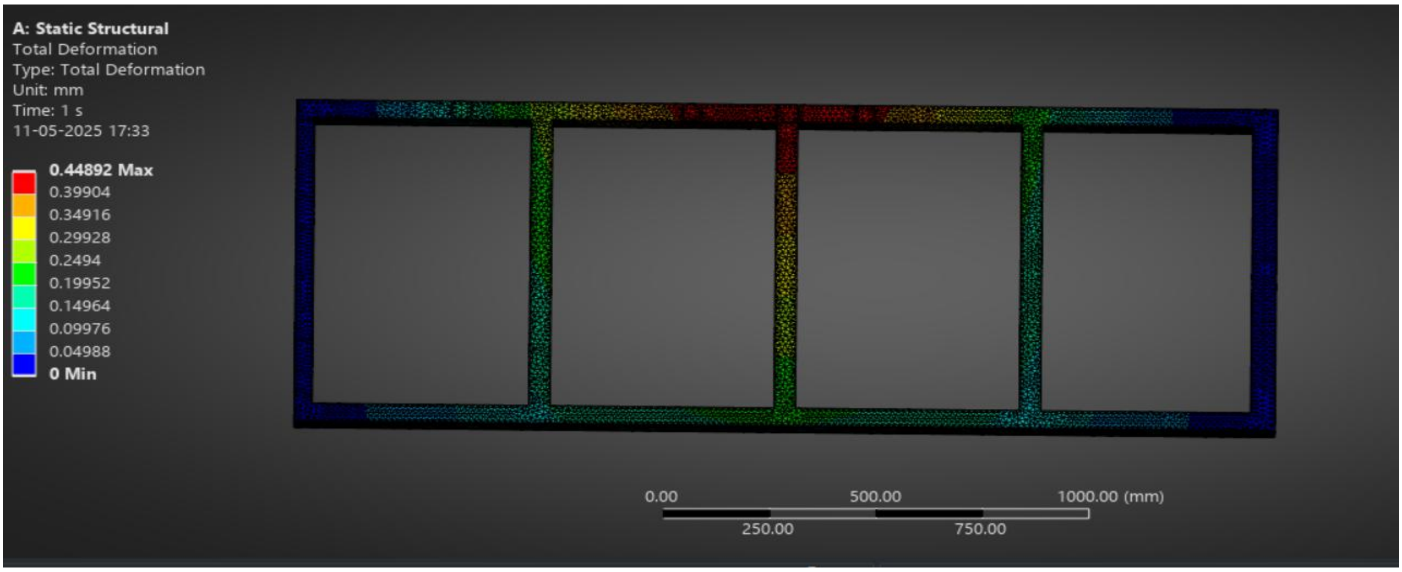


Fig no 25: Simulation Result of Frame

12.Tension Bar

Tensile Force Acting on the Tension Bar

F = 5485.67 N

L = 2237.32mm

Let's assume the Bar is made of Mild Steel with an allowable Tensile Stress σ_{allow} = 150 MPa

$\sigma = \frac{F}{A}$; $A = \frac{F}{\sigma_{allow}}$

$A = \frac{5485.67}{150} = 36.571\ mm^2$

Let's assume Width b = 6mm, then

$h = \frac{A}{b} = \frac{36.571}{6} = 6.10\ mm$

For using Bolt Pin for Attachment, we Use b = h = 15mm.

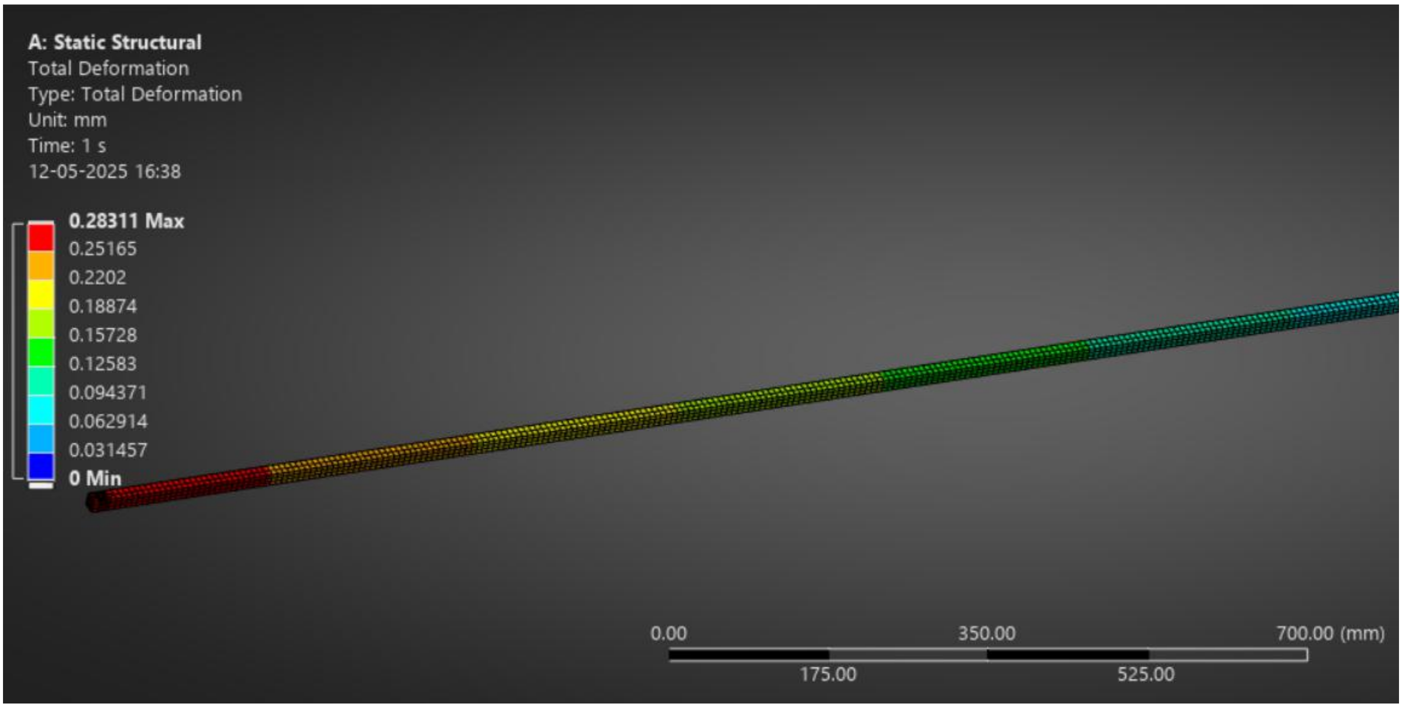


Fig no 26: Simulation Result of Tension Bar

13. Bolt Pin:

Requirement:

Length: 70 mm

Shear Force: 5485.67 N

Material: Mild Steel = 75 MPa

τ_{allow} = 75 MPa

Shear Stress on Circular Cross-Section

$\tau = \frac{F}{A} = \frac{4F}{\pi d^2}$

d = $\sqrt{\frac{4F}{\pi \times \tau_{allow}}}$

d = $\sqrt{\frac{4 \times 5485.67}{\pi \times 75}}$ = 9.65mm

d ≈ 10 mm.

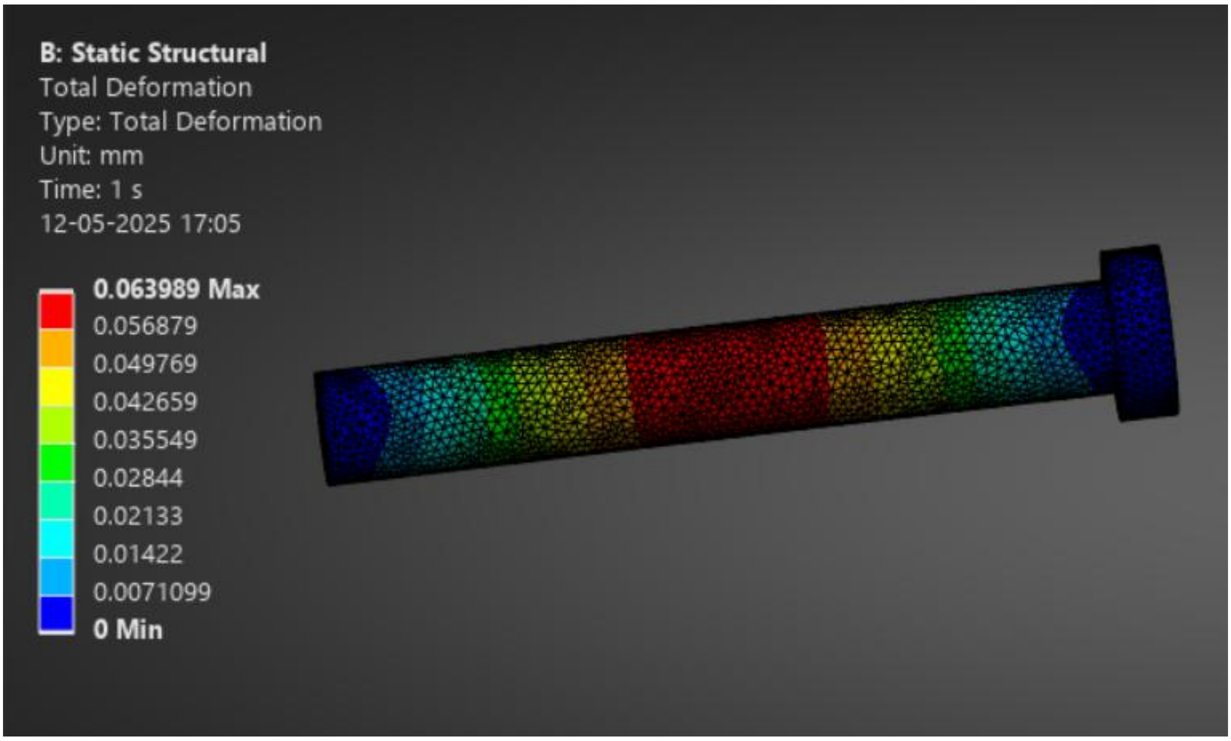


Fig no 27: Simulation Result of Tension Pin

14. Three Hinge Mechanism

Comparison of Different Hitch based on HP

Category	Tractor HP	Approximate Max Weight
Category 0	Up to 20	500 Pounds
Category 1	20-50	2000 Pounds
Category 2	50-125	5000 Pounds
Category 3	100-225	15000 Pounds
Category 4	200-400	20000 Pounds

Table no 14: Comparison of Different Hitch Based on HP

Category	Upper link hole diameter	Upper link pin diameter	Lower link hole diameter	Lower link pin diameter	Lower hitch point span	Mast Height
Category 0	20.32 mm or 51/64 inch	15.87 mm or 5/8 inch	22.86 mm or 9/10 inch	22.22 mm or 7/8 inch	508 mm or 20 inches	304.8 mm or 12 inches
Category 1	25.65 mm or 1.01 inches	19.05 mm or 3/4 inch	28.70 mm or 1.13 inches	27.94 mm or 1.10 inches	660.4 mm or 26 inches	381 mm or 15 inches
Category 2	25.65 mm or 1.01 inches	25.4 mm or 1 inch	28.70 mm or 1.13 inches	27.94 mm or 1.10 inches	825 mm or 32.48 inches	610.11 mm or 24.02 inches
Category 3	32.00 mm or 1.26 inches	31.75 mm or 1.25 inches	37.34 mm or 1.47 inches	36.58 mm or 1.44 inches	964.95 mm or 37.99 inches	685.04 mm or 26.9 inches

Table no 15: Based on Category of Three Point Hitch Design

Category 1	25.65mm	19.05mm	28.70mm	27.94mm	660.4mm	381mm

Table no 16: Category 1 for Frame Design

Shear Force in Hitch Point V = 2000 N

Allowable Shear Stress τ_{allow} = 60 MPa

Section Shape: Rectangular (width b, height h)

Shear Area Required

$\tau = \frac{V}{A}$: $A = \frac{V}{\tau_{allow}}$

$A = \frac{2000}{60 \times 10^6} = 3.33 \times 10^{-5}\ m^2 = 33.33 mm^2$

Choose Dimensions

$b \times h = A > 33.33\ mm^2$

$b = 4\ mm,\ h = 10\ mm:\ A = 40\ mm^2$

Shear Stress = $\frac{2000}{40}$ = 50 MPa < 60 MPa

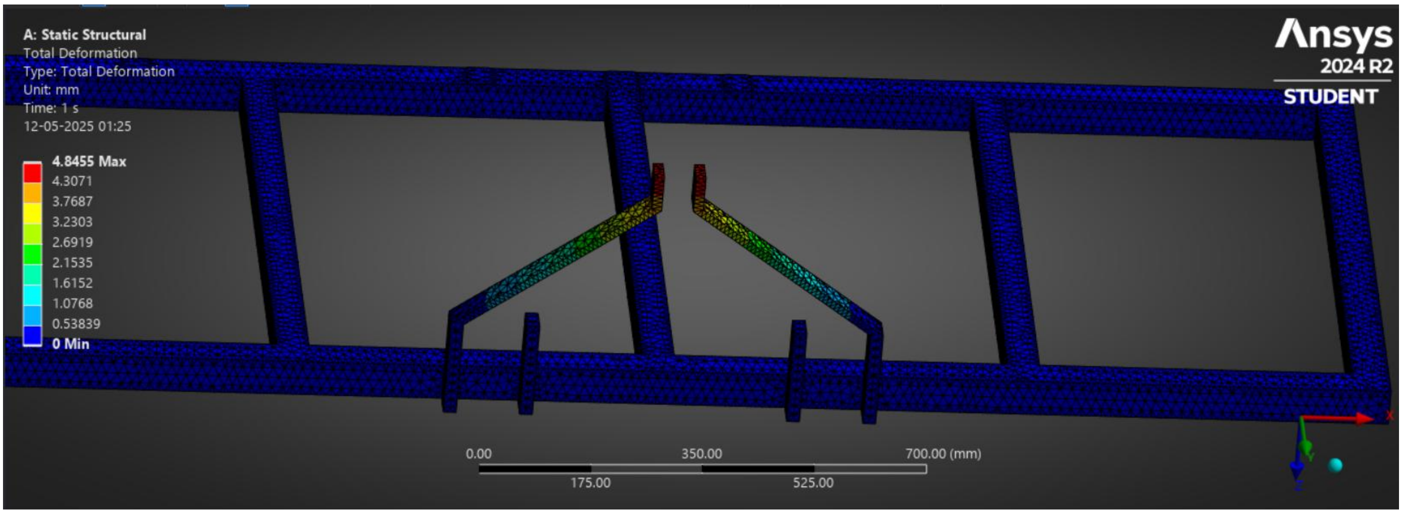


Fig no 29: Simulation Result of Three Point Hinch

15. Design for Hinge Pin

Requirement Data:

Shear Force V = 2000 N

Length= 110mm

Material: Mild Steel (τ_{allow} = 60MPa)

1. Shear Stress in a Circular Cross Section

$$\tau = \frac{V}{A} = \frac{V}{\frac{\pi d^2}{4}} \rightarrow d = \sqrt{\frac{4V}{\pi \tau_{allow}}}$$

2. BY Putting the Values

$$d = \sqrt{\frac{4\times 2000}{\pi \times 60 \times 10^6}} = \sqrt{\frac{8000}{188.5\times 10^6}} = 6.51 \text{ mm}$$

For Safe and Manufacturable we choose Standard Size d = 8 mm

16.Frame Stand:

Width (b) = 40mm
Height (h) = 40mm

Length = 450mm

Bolt Hole Diameter = 16mm

Hollow Inner:
Width (b) = 20mm
Height (h) = 20mm

Load = (150×9.81) = (1471.5/2) = 735.75 N

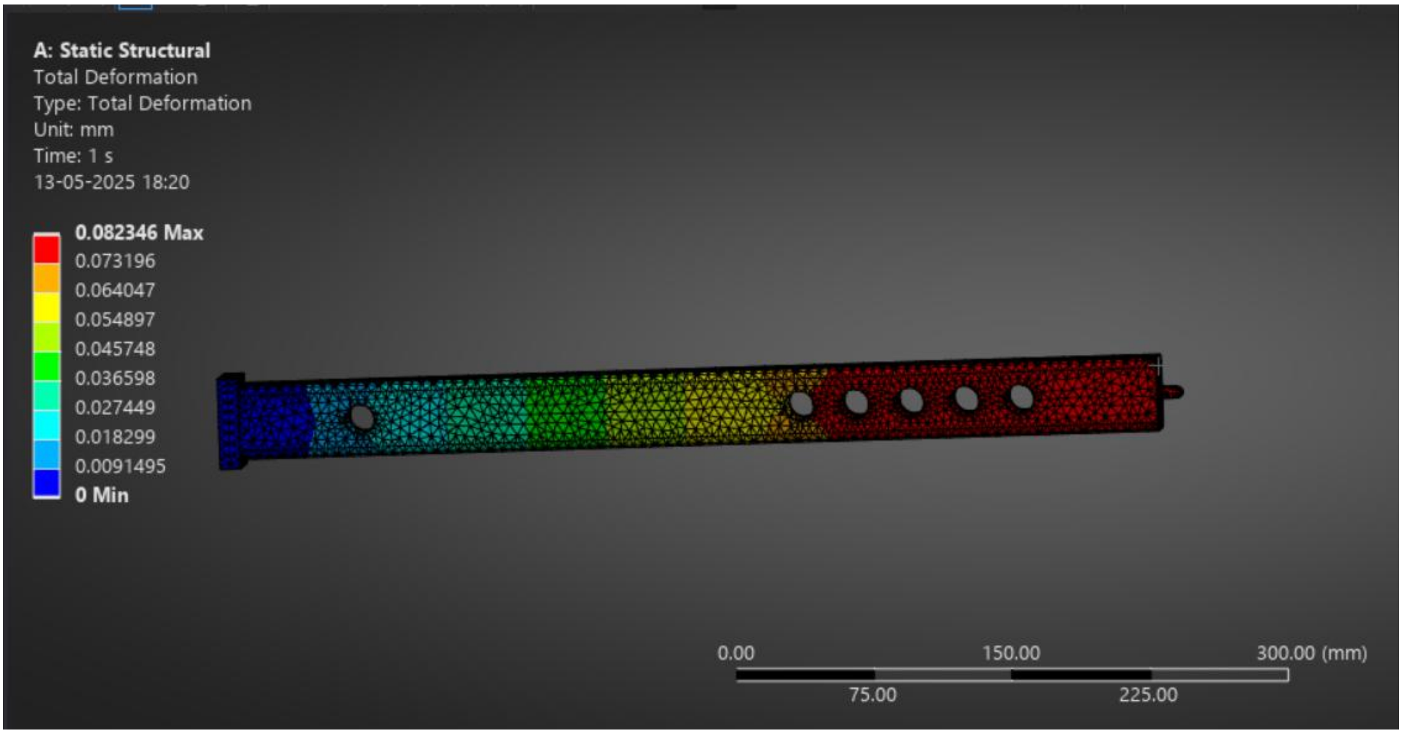


Fig no 30: Frame Stand

17.Wheel Shaft

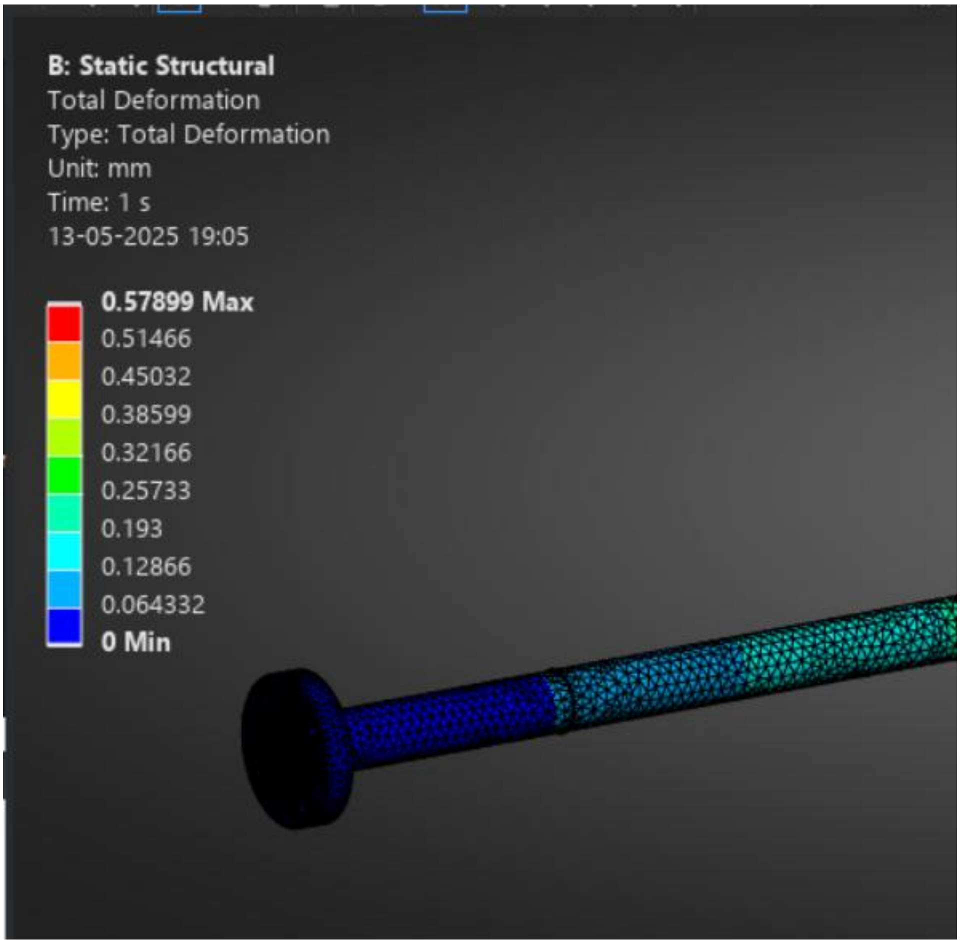


Fig no 31: Wheel Shaft

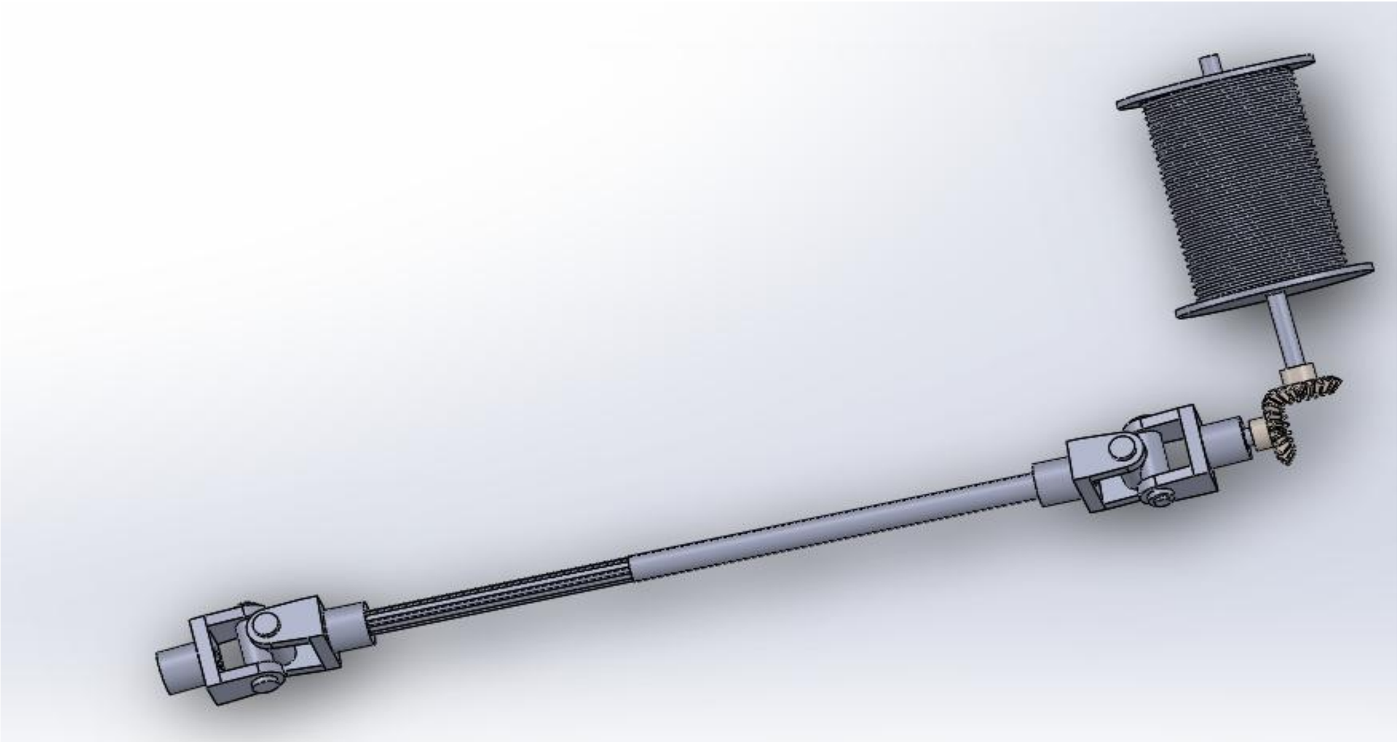


Fig no 32: Bevel Gear Assembly

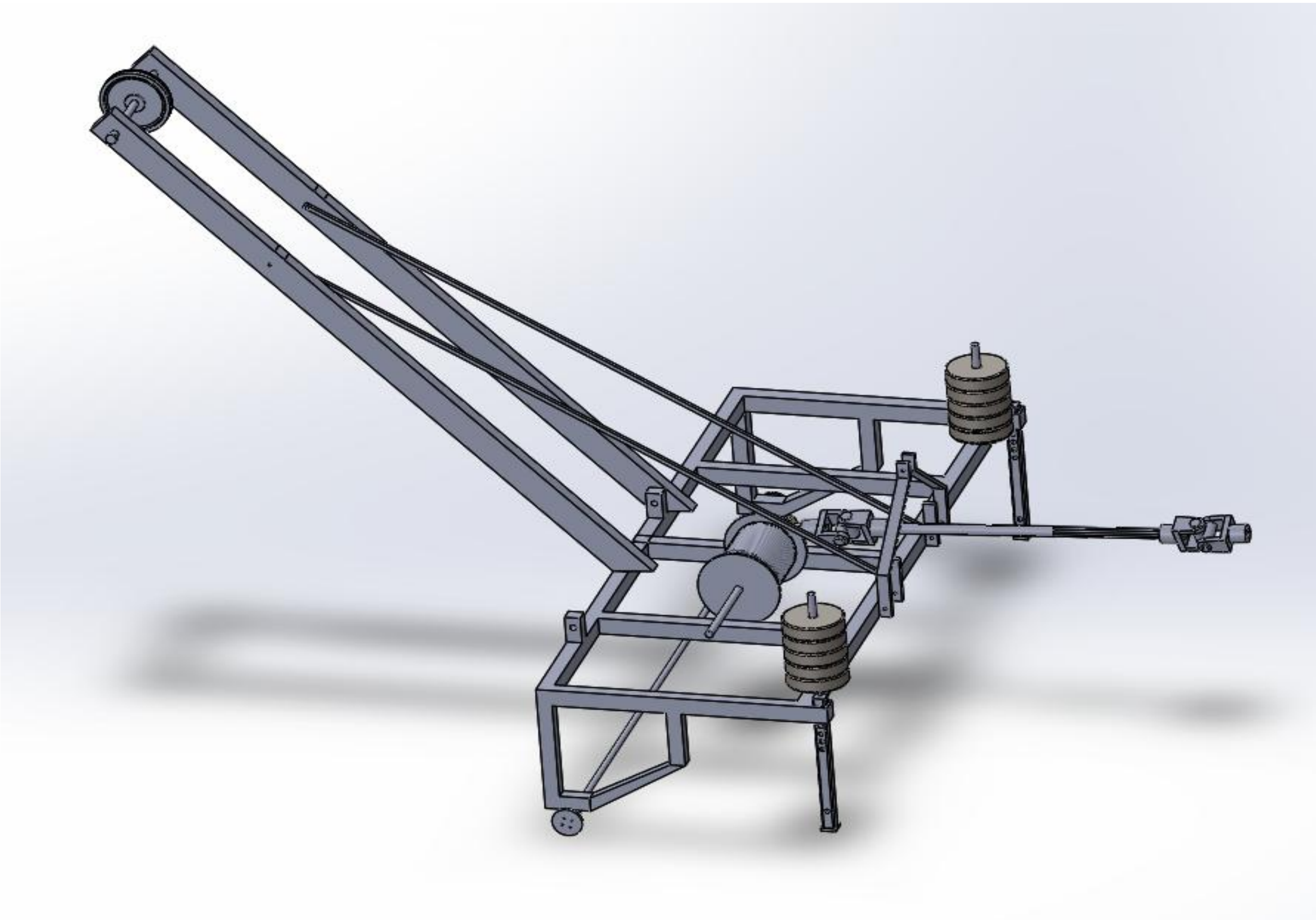


Fig no 31: Assembly

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