Indian Institute of Technology Kanpur
Department of Mechanical Engineering

ME351: 2024-25 - II
Course Project Report

Electric Foldable Scooter

August 26, 2025

Instructor:

Dr. Pankaj Wahi

Contribution Page

Name	Topic	Signature
Antarya Mondal [220177]	Suspension/Spring Analysis	
Arpit Yadav [220206]	Chassis Analysis	
Ashutosh Tripathi [220242]	Motor/Battery	
Ayush Agarwal [220260]	CAD Modelling, Assembly Drawings, Chassis design	
Ayush Baghel [220261]	Brake System Analysis	
Bhagat Gaurang [220287]	Handle/Joint Design	
Vishakha Goyal [221199]	Wheel Analysis	
Yerukonda Soumya Venkanna [221230]	Transmission and Gear Design	

Note: Although the individual contribution is stated above, every participant was involved in discussion, ideation and problem solving of different sections. Also everyone was involved in making report in LATEX format.

Contents

Section 1

Chassis Design

1.1 Design Constraints

- Weight of Passenger, P: 981 N (100×9.81 N)
- Dimensions of Deck: Length $L = 60$ cm, Width $w = 20$ cm
- Goal: To design scooter deck capable of supporting passenger for infinite life

1.2 Model

- The deck of the scooter can be estimated as a simply supported rectangular beam
- Two designs are considered: Hollow Rectangular Beam and Solid Rectangular Beam
- The load on the deck is due to the weight of the passenger, considered as point load at centre.

1.3 Material Properties

- Material: Aluminum
- Density: 2.7 g/cm^3
- Yield Strength, $\sigma_{\text{yield}} = 276$ MPa
- Ultimate Tensile Strength, $\sigma_u \approx 304$ MPa

1.4 Static Analysis Calculations

- The maximum moment in a simply supported beam:

$$M = \frac{PL}{4}$$

Substituting the values:

$$M = \frac{981 \times 0.60}{4} = 147.15 \text{ Nm}$$

- The bending stress is given by:
$$\sigma = \frac{My}{I}$$

1.4.1 Solid Beam

Assuming a factor of safety (FoS) of 1.5:

$$\sigma_a = \frac{\sigma_{\text{yield}}}{FoS}$$

For a solid rectangular beam, the moment of inertia:

$$I = \frac{wh^3}{12}$$

Using the bending stress formula:

$$\sigma = \frac{6M}{wh^2} \leq \sigma_a$$

Solving for minimum height:

$$h \geq \left(\frac{6 \times 147.15}{0.2 \times 184 \times 10^6}\right)^{0.5} = 4.9 \text{ mm}$$

Using standard size, chosen height $h = 5$ mm, weight:

$$\text{Weight} = 20 \times 60 \times 0.5 \text{ cm}^3 \times 2.7 \text{ g/cc} = 1620 \text{ g}$$

1.4.2 Hollow Beam

- Overall height $h_o = 10$ mm
- Wall thickness $t = 1.5$ mm

Moment of Inertia

$$I_{\text{hollow}} = \frac{wh_o^3 - (w - 2t)(h_o - 2t)^3}{12}$$

Inner dimensions:

$$w_i = 0.197 \text{ m}, \ h_i = 0.007 \text{ m}$$

Calculated:

$$I_{\text{hollow}} = 1.103575 \times 10^{-8} \text{ m}^4$$

Bending Stress

$$\sigma_b = \frac{147.15 \times 0.005}{1.103575 \times 10^{-8}} = 66.67 \text{ MPa}$$

$$\text{FoS} = \frac{276}{66.67} \approx 4.14$$

Weight: 1.006 kg

1.5 Fatigue Analysis

1.5.1 Theoretical Background

Fatigue strength:

$$S'_e \approx 0.5 \times 304 = 152 \text{ MPa}$$

Endurance limit:

$$S_e = S'_e \times k_a \times k_b \times k_c \times k_d \times k_e \times k_f$$

1.5.2 Size Factors

Size Factor for Solid Beam

For non-circular cross-sections under bending loads, the size factor k_b is calculated using an equivalent diameter d_e, which for a rectangular section is given by:

$$d_e = 0.808\sqrt{h \times w}$$

For our solid beam:

$$d_e = 0.808\sqrt{5 \times 20} = 0.808\sqrt{100} = 8.08 \text{ mm}$$

The size factor is then calculated using:

$$k_b = \begin{cases} 1.24 d_e^{-0.107} & \text{for } 2.79 \text{ mm} < d_e \leq 51 \text{ mm} \\ 1.51 d_e^{-0.157} & \text{for } 51 \text{ mm} < d_e \leq 254 \text{ mm} \end{cases}$$

Since $d_e = 8.08$ mm falls in the first range:

$$k_b = 1.24 \times (8.08)^{-0.107} = 1.24 \times 0.799 = 0.99$$

Size Factor for Hollow Beam

For hollow rectangular beams, we can calculate the equivalent diameter using two methods:

Based on equal cross-sectional area:

$$A_{hollow} = w \times h_o - (w - 2t) \times (h_o - 2t) = 20 \times 10 - 17 \times 7 = 200 - 119 = 81 \text{ mm}^2$$

$$d_{e,area} = \sqrt{\frac{4A_{hollow}}{\pi}} = \sqrt{\frac{4 \times 81}{\pi}} = 10.16 \text{ mm}$$

$$k_b = 1.24 \times (10.16)^{-0.107} = 1.24 \times 0.78 = 0.96$$

1.5.3 Corrected Endurance Limits

- Solid beam: 110.4 MPa
- Hollow beam: 107.05 MPa

1.5.4 Stress Analysis

Assuming the load varies from zero to full passenger weight (981 N), we have:

For solid beam (5 mm):

- Maximum stress: $\sigma_{max} = 184$ MPa (based on static FoS = 1.5)
- Alternating stress: $\sigma_a = \sigma_{max}/2 = 92$ MPa
- Mean stress: $\sigma_m = \sigma_{max}/2 = 92$ MPa

For hollow beam (10 mm, 1.5 mm wall):

- Maximum stress: $\sigma_{max} = 66.67$ MPa
- Alternating stress: $\sigma_a = \sigma_{max}/2 = 33.34$ MPa
- Mean stress: $\sigma_m = \sigma_{max}/2 = 33.34$ MPa

1.5.5 Fatigue Factor of Safety

Goodman criterion:

$$\text{FS} = \frac{1}{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_u}}$$

Results:

- Solid beam: FS ≈ 0.88
- Hollow beam: FS ≈ 2.36

1.6 Conclusion

- Solid beam: Inadequate fatigue safety (FS < 1)
- Hollow beam: Acceptable fatigue safety (FS > 2)
- Hollow beam offers better static safety (FoS = 4.14) and lower weight

Inadequacies

- Simplified beam model is assumed instead of actual support conditions due to analytical complexities

1.7 Comparison with FEM analysis

- Since the scooter deck geometry differs due to wheel assembly and battery/motor holding, computational stress analysis has been conducted to validate model.
- In the computational model, passenger weight is modeled as a line load, on the deck.

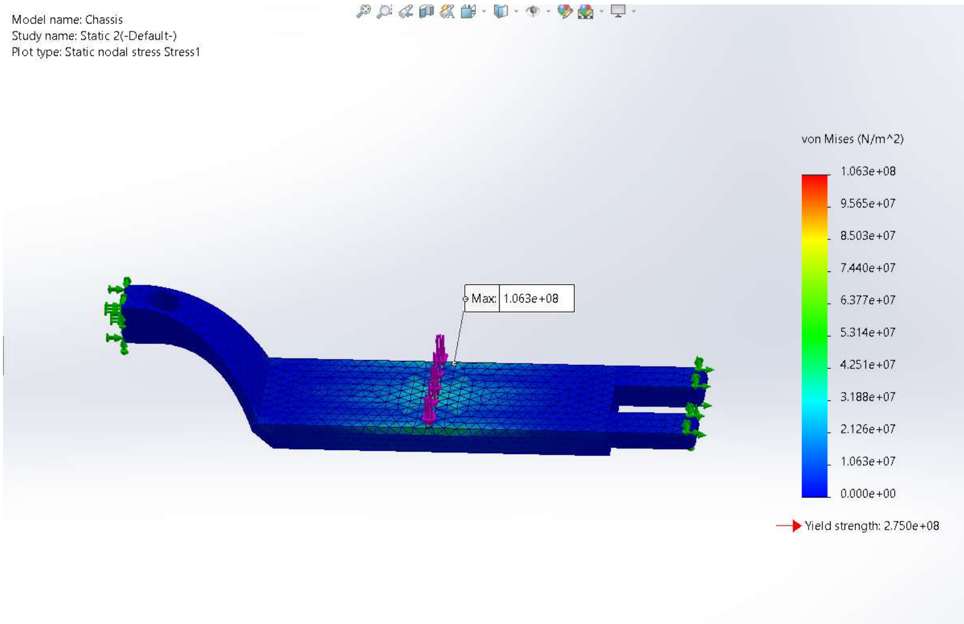

Figure 1.1: Stress distribution across chassis.

- In the static stress analysis, the stresses are maximum at the mid-point of deck edge, about 106.3 MPa, due to the nature of support at front and rear wheels.

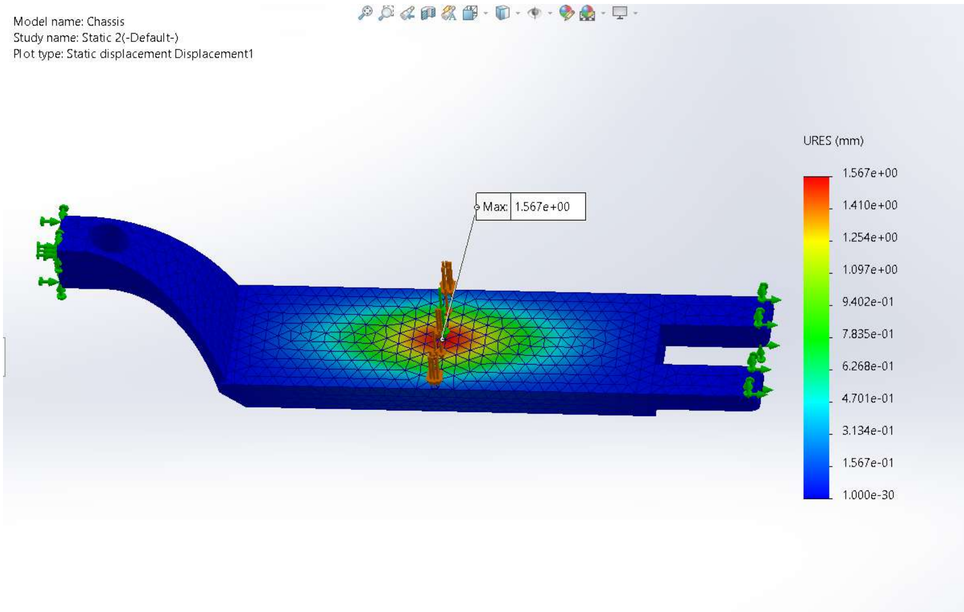

Figure 1.2: Displacement distribution across chassis.

- The maximum displacement is about 1.567 mm at the centre, which is insignificant considering span of deck.

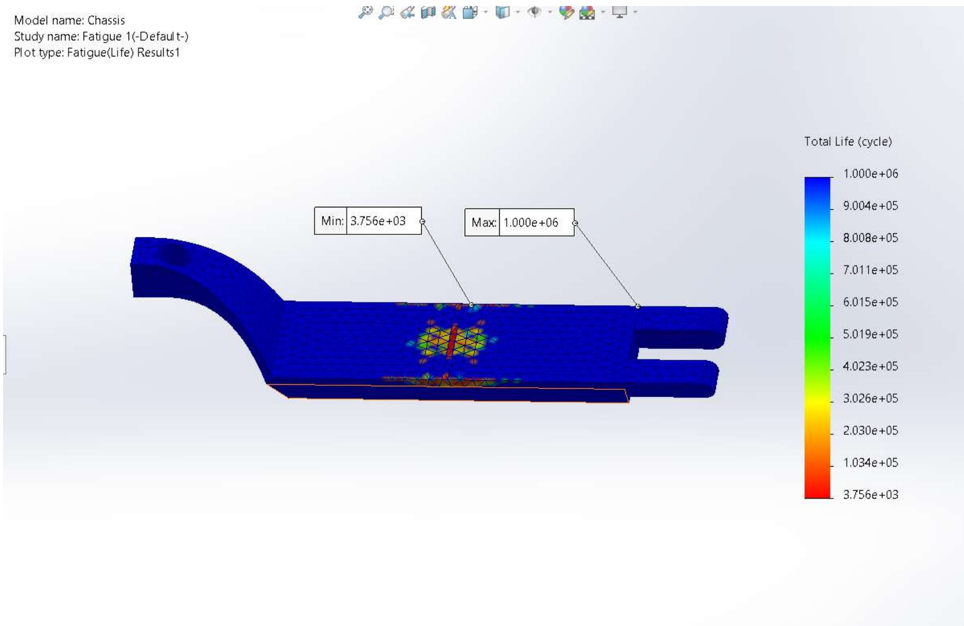

Figure 1.3: Fatigue life across chassis.

- The minimum fatigue life is about $3,756$ cycles at the midpoint of edges.

Section 2

Foldable Hinge Joint with Locking Mechanism

2.1 Design of Joint

2.1.1 Problem Statement

A mechanical joint is required at the handle assembly of the scooter to enable controlled folding motion. This joint must allow the grip handle to rotate and align with the wheel handle axis during the folding process. It must also ensure stable retention in both operational (unfolded) and folded positions, providing secure locking in each configuration to maintain safety and usability.

2.1.2 Methodology

First, I decided what type of joint should be used. And after some searching and thinking, I decided to build a hinge joint for folding of scooter. Now according to our design, the shaft connected to wheel should be held at the fixed position as shown in figure.

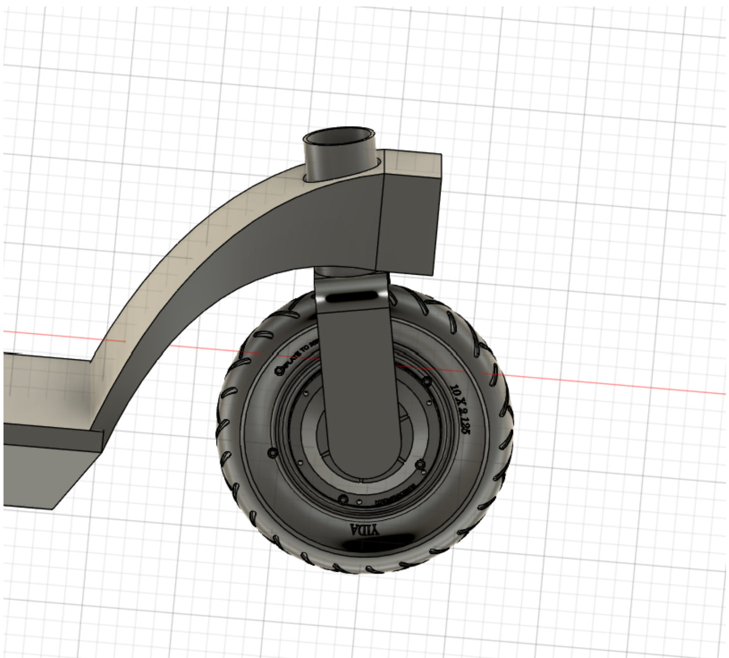

Figure 2.1: Fixed position of wheel shaft (placeholder)

The handle part is attached via the joint. First, we will make a fix part of hinge joint which will be fixed on wheel shaft by a bolt and nut joint through the hole marked as shown in figure.

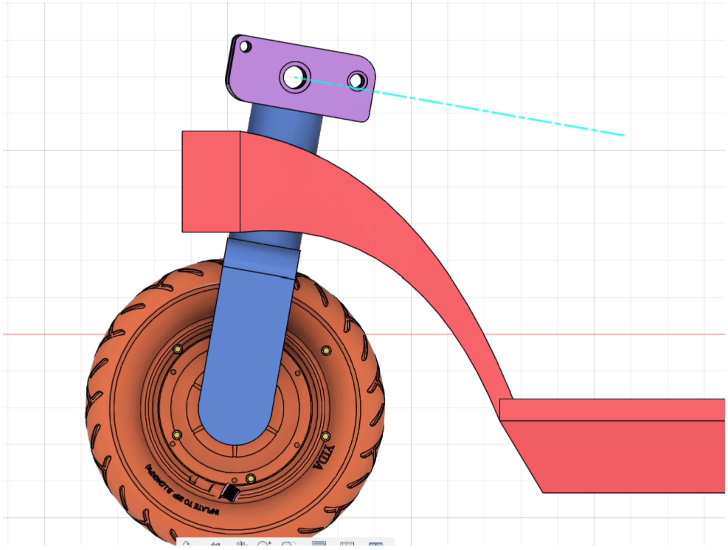

Figure 2.2: Hinge base fixed to wheel shaft (placeholder)

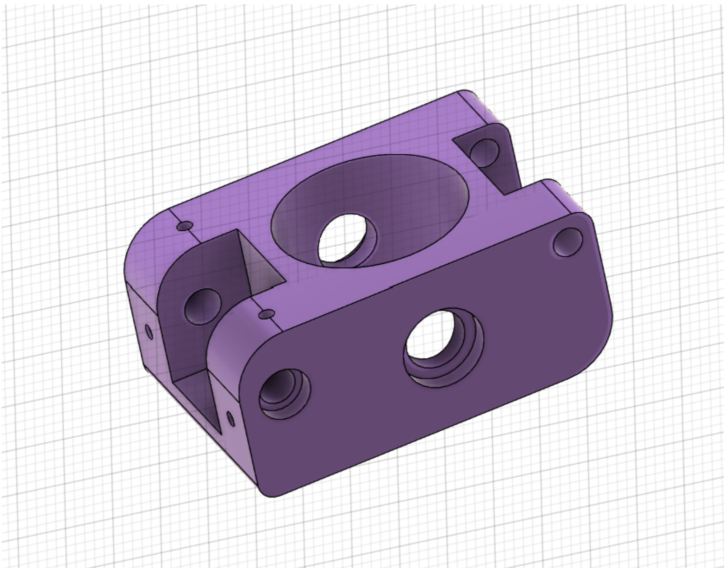

Figure 2.3: Hinge Base

Next, a part is designed that connects to the handle and facilitates the folding:

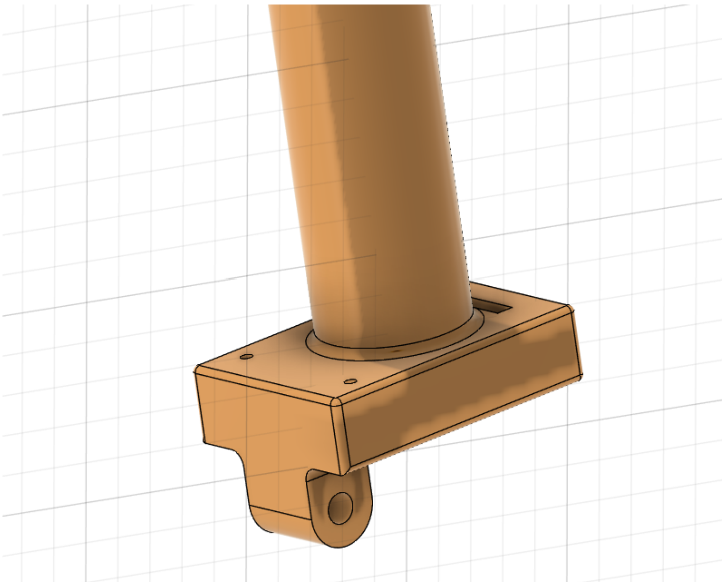

Figure 2.4: Handle-side hinge component (placeholder)

A hinge pin is used to connect both joint parts:

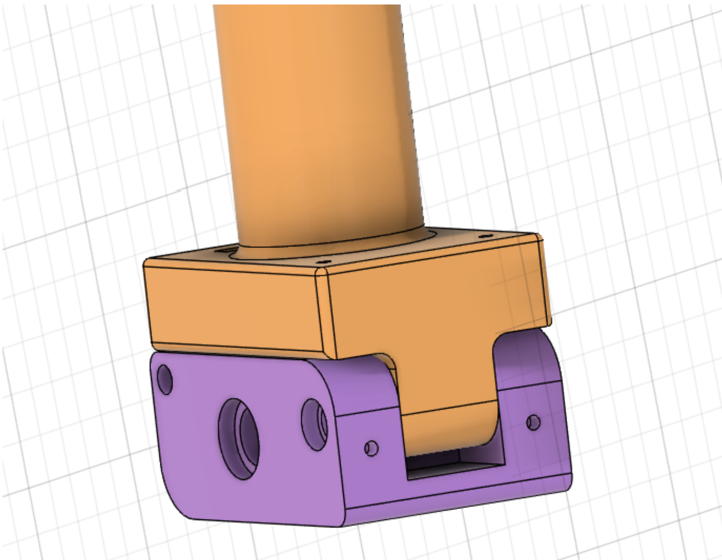

Figure 2.5: Hinge pin connecting both parts (placeholder)

Now there is the need of a pin, which fixes this unfold position which ensures no relative motion between the handle shaft and wheel shaft. So, I will make a groove in joint in handle part which gets connected by a pin from wheel joint component. See the figure for the idea.

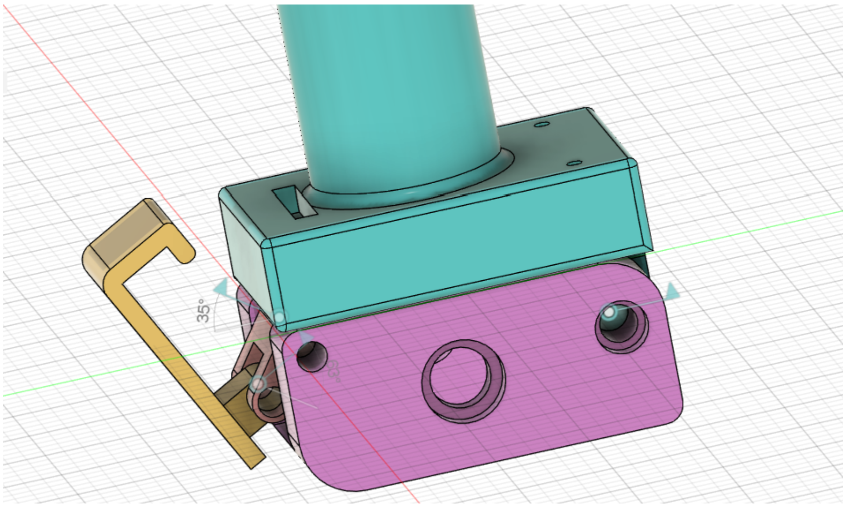

Figure 2.6: Unlocked Position

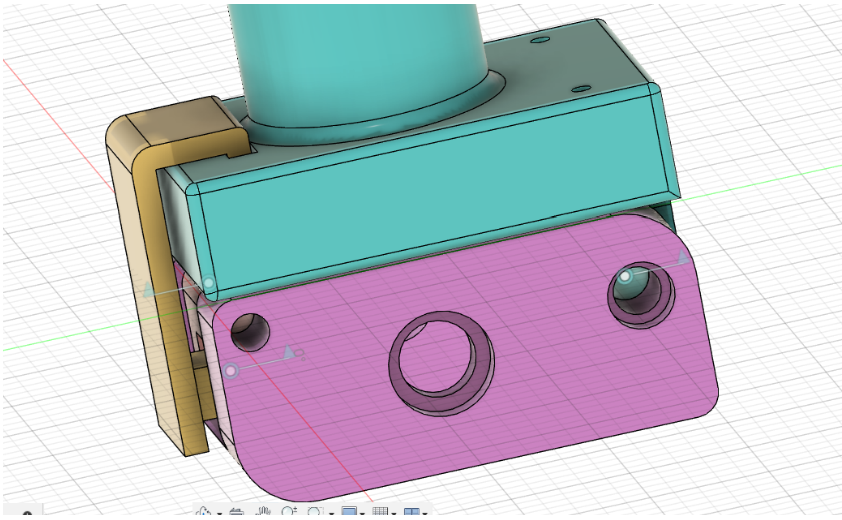

Figure 2.7: Locked Position

There is an appropriate clearance in pin fixing groove which helps to lock and unlock the position. To unlock it first one need to press pin at bottom and then try to lift the pin upward and then it can easily be unlocked.

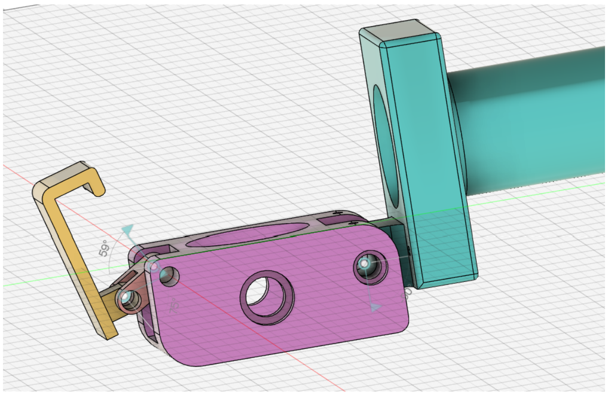

Figure 2.8: Folded position with finger pin lock (placeholder)

In the folded position, a finger pin is used to lock the configuration securely:

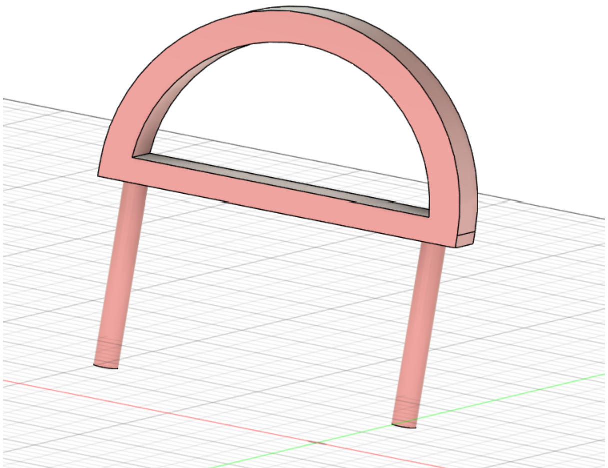

Figure 2.9: Finger pin used for locking in folded config

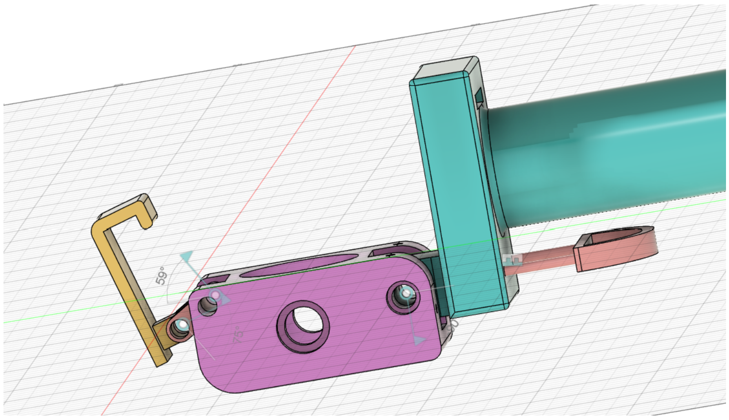

Figure 2.10: Locked with Finger Pin

Finally, in folded position, the scooter is shown in figure.

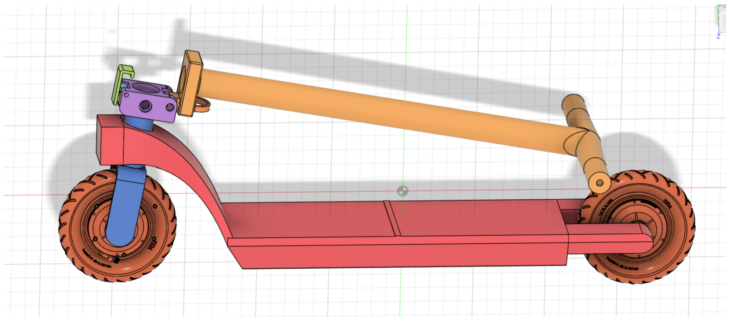

Figure 2.11: Final locked position in folded mode)

2.2 Analysis of Weak Part of Joint: Pivot Bracket

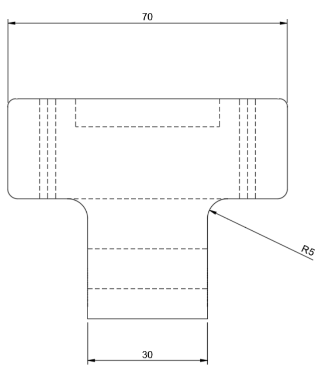

Figure 2.12: Dimensions of Pivot Bracket (in mm)

Considering 15 kgf as the maximum force acting on the handle in the given direction, accounting for human yawing and vertical fall impact.

2.2.1 Free Body Diagram

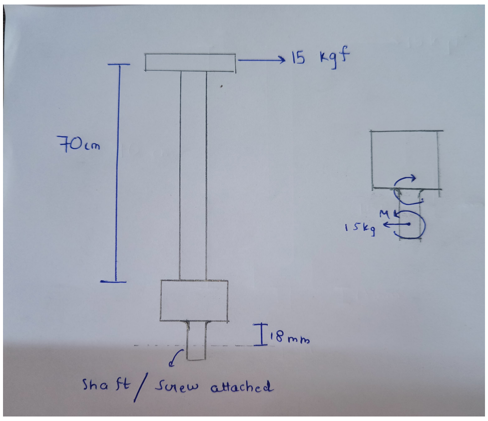

Figure 2.13: FBD

2.2.2 Net Bending Moment at Stress Concentration Point

$$
\begin{aligned}
M &= 15 \times 9.81 \times 0.7 + 15 \times 9.81 \times 0.018 \\
&= 105.65\,\text{Nm}
\end{aligned}
$$

2.2.3 Areal Moment of Inertia for Bending

$$I_{zz} = \frac{1}{12} \times (0.04)^3 \times (0.025)\text{m}^4$$

$$I_{zz} = 133.33 \times 10^{-9}\,\text{m}^4$$

2.2.4 Stress Concentration Factor

$$\frac{D}{d} = 2.33, \quad \frac{r}{d} = 0.166$$

From Figure A-15-6:

$$K_t = 1.6$$

2.2.5 Equivalent Bending Stress on Fillet

$y = 15 \times 10^{-3}$ m:

$$\begin{aligned}\sigma &= \frac{K_t \cdot M \cdot y}{I_{zz}} \\ &= \frac{1.6 \cdot 105.65 \cdot 0.015}{133.33 \times 10^{-9}} \\ &= 19.07 \times 10^6\,\text{Pa} = 19.07\,\text{MPa}\end{aligned}$$

2.2.6 Material Selection

Here as we are building a small part we will choose 2024 aluminium alloy:(source :internet)

$$S_y = 76\,\text{MPa}$$

2.2.7 Factor of Safety for Yielding

$$\eta_f = \frac{S_y}{\sigma} = \frac{76}{19.07} \approx 3.98$$

2.2.8 Fatigue Strength

Since aluminium alloy does not have a defined endurance limit,
So for 10^7 cycles:

$$S_f = 90\,\text{MPa}$$

As $S_f > \sigma$, the component is considered to have infinite life.

2.2.9 Factor of Safety for Fatigue

$$\text{FoS}_{\text{Fatigue}} = \frac{S_f}{\sigma} = \frac{90}{19.07} \approx 4.72$$

Section 3

Brake System Analysis : Drum Brakes

Drum brakes are a type of braking system commonly used in vehicles, where brake shoes press outward against a rotating drum attached to the wheel to create friction and slow down the vehicle. When the brake pedal is applied, hydraulic pressure forces the brake shoes to expand and contact the inner surface of the drum, converting kinetic energy into heat

3.1 Braking Torque Requirement Calculation

- Total mass of scooter and rider: $m = 100\,\mathrm{kg}$
- Radius of wheel: $R = 10\,\mathrm{inches} = 0.254\,\mathrm{m}$
- Maximum rotational speed: $N = 250\,\mathrm{RPM}$
- Brake drum radius: $r = 3\,\mathrm{inches} = 0.0762\,\mathrm{m}$
- Stopping time: $t = 2.5\,\mathrm{s}$

Assuming a level ground, the angular velocity of the wheel is:

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 250}{60} = 26.18\,\mathrm{rad/s}$$

Linear velocity of the scooter:

$$v = \omega R = 26.18 \times 0.254 = 6.65\,\mathrm{m/s}$$

Required deceleration:

$$a = \frac{v}{t} = \frac{6.65}{2.5} = 2.66\,\mathrm{m/s^2}$$

Braking force:

$$F = ma = 100 \times 2.66 = 266.0\,\mathrm{N}$$

As the brake shoe will act against the brake disc. Perpendicular distance for acting force $'F'$ is 2.5 inches = 0.0635 m
Braking torque:

$$T = Fr = 266.0 \times 0.0635 = 16.891\,\mathrm{Nm}$$

Therefore, the required braking torque is approximately $\boxed{16.981\,\text{Nm}}$.

3.2 Torque and Force Analysis

Let T_1 and T_2 be the torques from the primary and secondary brake shoes respectively. The total braking torque is:

$$T_1 + T_2 = 16.981\,\text{Nm}$$

From Shigley's Formulation:

$$T_1 = \int f r\, dN = \frac{f P_{a1} b r^2}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin\theta\, d\theta$$

$$T_2 = \frac{f P_{a2} b r^2}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin\theta\, d\theta$$

dividing above two equations we get the following ratio:

$$\frac{T_1}{T_2} = \frac{P_{a1}}{P_{a2}} \tag{a}$$

Moment due to friction force:

$$M_f = \frac{f P_a b r}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin\theta (r - a\cos\theta)\, d\theta$$

Moment due to normal force:

$$M_N = \frac{P_a b r a}{\sin\theta_a} \int_{\theta_1}^{\theta_2} \sin^2\theta\, d\theta$$

Force required for brake operation:

- Self-energizing condition:

$$F = \frac{M_N - M_f}{C} \tag{b}$$

- Non-self-energizing condition:

$$F = \frac{M_N + M_f}{C} \tag{c}$$

Here, C = 2 · r', where r' is half of the chord length of the brake shoe end points.
By equating forces on the two shoes, we get the following relation:

$$M_{N1} - M_{f1} = M_{N2} + M_{f2}$$

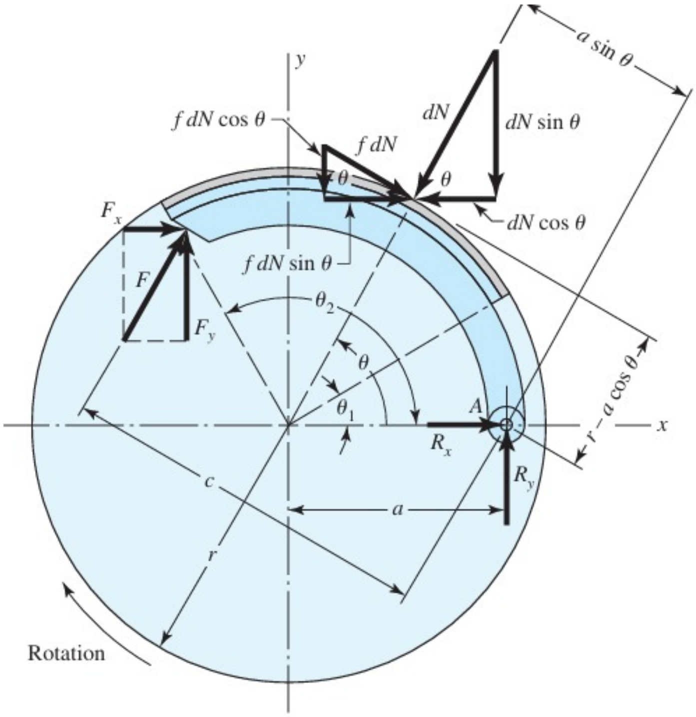

Figure 3.1: Drum-brake Schematic

Figure 3.2: Drum-brake CAD

Given Data

$$
\begin{aligned}
f &= 0.4 \quad (Shigley - table: 16-3) \\
h &= 0.75\,\text{in} = 0.019\,05\,\text{m} \\
r' &= 2.5\,\text{in} = 0.0635\,\text{m} \\
r &= 3\,\text{in} = 0.0762\,\text{m} \\
b &= 1\,\text{in} = 0.0254\,\text{m} \\
\theta_a &= 90^\circ \Rightarrow \sin\theta_a = 1 \\
a &= \sqrt{r^2 + h^2} = \sqrt{0.0635^2 + 0.01905^2} \approx 0.0663\,\text{m}
\end{aligned}
$$

Where:

P_a = largest pressure on shoe
f = friction factor between brake lining and drum (rigid molded asbestos, dry)
h = half the minimum distance between the brake shoes at pin joints
r' = half the chord length between shoe end points
r = drum brake radius
b = brake shoe width
θ_a = maximum pressure angle (for long brake shoe)
a = radius of brake shoe

Integral Evaluations

Angles of brake shoe:

$$\theta_1 = \frac{35\pi}{180}, \quad \theta_2 = \frac{135\pi}{180}$$

$$
\begin{aligned}
\int_{\theta_1}^{\theta_2} \sin^2\theta\, d\theta &= \frac{1}{2}(\theta_2 - \theta_1) - \frac{1}{4}(\sin 2\theta_2 - \sin 2\theta_1) \\
&\approx 1.35758
\end{aligned}
$$

$$\int_{\theta_1}^{\theta_2} \sin\theta(r - a\cos\theta)\, d\theta \approx 0.11063$$

Moments in Terms of P_a

$$\begin{aligned}
M_{Ni} &= \frac{P_{ai}bra}{\sin\theta_a}\cdot\int_{\theta_1}^{\theta_2}\sin^2\theta\,d\theta \\
&= P_a\cdot(0.0254)(0.0762)(0.0663)\cdot 1.3758 \\
&\approx P_a\cdot 0.0001765\,\mathrm{Nm} \\
M_{fi} &= fP_{ai}br\cdot\int_{\theta_1}^{\theta_2}\sin\theta(r-a\cos\theta)\,d\theta \\
&= 0.4\cdot P_a\cdot(0.0254)(0.0762)\cdot 0.11063 \\
&\approx P_a\cdot 0.00008564\,\mathrm{Nm}
\end{aligned}$$

Torque Ratio and Values

Using:

$$M_{N1}-M_{f1}=M_{N2}+M_{f2}$$

$$P_{a1}(0.0001765-0.00008564)=P_{a2}(0.0001765+0.00008564)$$

$$\frac{P_{a1}}{P_{a2}}=\frac{M_{N1}+M_{f1}}{M_{N2}-M_{f2}}=\frac{0.0001765+0.00008564}{0.0001765-0.00008564}\approx 2.885$$

$$T_1+T_2=16.981\quad\text{and}\quad\frac{T_1}{T_2}=2.885$$

$$T_1=2.885\,T_2\Rightarrow 2.885\,T_2+T_2=16.981\Rightarrow\boxed{T_2\approx 4.371\,\mathrm{N\,m},\quad T_1\approx 12.610\,\mathrm{N\,m}}$$

Moment and P_a Calculations

Putting the valuse in the below relation stated above:

$$T_1=\int fr\,dN=\frac{fP_{a1}br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta\,d\theta$$

$$12.610=\frac{0.4P_{a1}0.0254(0.0762^2)}{\sin(\pi/2)}\int_{35^\circ}^{135^\circ}\sin\theta\,d\theta$$

$$\boxed{P_{a1}=140.050kPa}$$

similarly we can find for P_{a2}:

$$T_2=\frac{fP_{a2}br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta\,d\theta$$

$$4.371 = \frac{0.4P_{a2}0.0254(0.0762^2)}{\sin(\pi/2)} \int_{35^\circ}^{135^\circ} \sin\theta\, d\theta$$

$$\boxed{P_{a2} = 48.544kPa}$$

With the help of above values, we can find moment values due to friction and reaction:

$$M_{N1} = P_{a1} \cdot 0.0001765 = 24.72Nm$$

$$M_{N2} = P_{a2} \cdot 0.0001765 = 8.568Nm$$

$$M_{f1} = P_{a1} \cdot 0.00008564 = 11.99Nm$$

$$M_{f2} = P_{a2} \cdot 0.00008564 = 4.16Nm$$

3.3 Final Results

$$\begin{aligned} T_1 &= 12.610\,\mathrm{Nm} & T_2 &= 4.371\,\mathrm{Nm} \\ M_{N1} &= 24.72\,\mathrm{Nm} & M_{f1} &= 11.99\,\mathrm{Nm} \\ M_{N2} &= 8.568\,\mathrm{Nm} & M_{f2} &= 4.16\,\mathrm{Nm} \end{aligned}$$

Section 4

Powertrain: Motor and Battery

4.1 Battery Specification

Parameter	Specification
Type:	Lithium-Ion Battery Pack
Voltage:	36 V
Capacity:	7.8 Ah
Energy:	280.8 Wh (36 × 7.8)
Weight:	Approximately 1 kg
Dimensions:	31 cm × 7 cm × 4.1 cm

4.2 Justification of Battery Selection

The battery system for our electric foldable scooter is carefully designed to optimize range, portability, and overall performance for urban mobility. With a **target range of 25–30 km** and an **average energy consumption of approximately 10 Wh/km**, the estimated real-world range is:

$$\text{Range} = \frac{280.8 \text{ Wh}}{10 \text{ Wh/km}} \approx 28 \text{ km}$$

A standard **2A charger** is used to strike a balance between fast charging and battery longevity. The full charging time is:

$$\frac{7.8 \text{ Ah}}{2 \text{ A}} \approx 4 \text{ hours}$$

The battery uses **Lithium-Ion chemistry**, offering high energy density and low weight—critical features for a foldable scooter. At just **1 kg**, the battery helps maintain the scooter's total weight around **15 kg**, making it easy to fold and carry. The battery is capable of **500–1000 charge cycles**, supporting long-term daily use.

To accommodate performance demands such as slopes and quick acceleration, the battery supports a **maximum current draw of 10A**. The corresponding C-rate is:

$$\frac{10A}{7.8Ah} \approx 1.28C$$

which remains within the typical safe range of 1–2C for Lithium-Ion cells.

A direct **36V compatibility** between the battery and the BLDC motor ensures efficient power delivery without the need for converters, minimizing energy losses and maximizing system efficiency.

Summary Table

Parameter	**Battery Spec**	**Logic**
Range	280.8 Wh	Meets 25–30 km target at 10 Wh/km
Charging Time	2A charger	$\approx$ 4 hours; balanced for performance
Chemistry	Lithium-Ion	High energy density and low weight
Discharge Rate	$\approx$1.28C	Supports 10 A peak draw safely
Weight	$\sim$1 kg	Ensures portability for folding
Voltage Match	36 V	Directly compatible with the motor
Cost-Benefit	High value	Favorable balance of cost, performance, and longevity

4.3 Placement Consideration

The battery is positioned under the standing space in the middle of the scooter frame. One of the desired criteria for the battery pack is desirable weight on top of the center of rotation and low center of gravity. Having the battery pack at this location creates a stable body while also allowing for a functional folding mechanism. The battery pack will be secured in a waterproof enclosure to protect it from moisture and debris in the environment but again maintainability is key to design. This enclosure will need to be designed for access to connections and wires to allow for replacement, servicing or repairs. Altogether the compact methodology provides a functional and balanced scooter.

4.4 Motor Selection, Torque & Power Analysis

4.4.1 Drive Type Selection

A direct connection is implemented between the BLDC motor and the planetary gearbox mounted on the rear wheel. By avoiding belt or chain drives, the design simplifies the drivetrain, reduces maintenance, and maximizes power transmission efficiency.

4.4.2 Motor Specifications

Parameter	Motor Spec
Type of Motor	BLDC Motor
Rated Voltage	36 V
Rated Power	250 W
Rated Current	14 A
Rated Torque	6 Nm
No-load Current	1 A
Weight	3.03 kg
Speed	300 RPM
Compatible Wheel Size	8–10 inch
Shaft Diameter	11.5 mm (ROUND); 10 mm (FLAT)
Shaft Length	55 mm

4.4.3 Required Motor Power Calculation

- **Target Speed:** 20 km/h = 5.56 m/s
- **Wheel Diameter:** 8 inch = 0.2032 m
- **Area (A):** $A = b \times h = 20\,\text{cm} \times 100\,\text{cm} = 0.2\,\text{m}^2$
- **Efficiency of Motor:** $\eta = 0.8$ [Ref]
- **Rolling Resistance Coefficient (C_{rr}):** $C_{rr} = 0.015$ [Ref]
- **Drag Coefficient:** $C_d = 1.0$ [Ref]
- **Drag Force:**

$$F_{\text{drag}} = \frac{1}{2} C_d A \rho V^2 = 0.5 \times 1.0 \times 0.2 \times 1.225 \times (5.56)^2 \approx 3.8\,\text{N}$$

- **Rolling Resistance:**

$$F_{\text{roll}} = C_{rr} \cdot mg = 0.015 \cdot 87 \cdot 9.81 \approx 12.8\,\text{N}$$

- **Total Power:**

$$P = (F_{\text{drag}} + F_{\text{roll}}) \cdot V = (3.8 + 12.8) \cdot 5.56 \approx 93\,\text{W}$$

- **Required Motor Power (with margin):**

$$P_{\text{required}} = \frac{P}{\eta} = \frac{93}{0.8} \approx 117\,\text{W}$$

- Chosen 250W motor has sufficient margin.

4.4.4 Stall Torque Analysis

Stall torque is the torque at zero RPM (starting torque). For a BLDC motor:

$$\tau_{\text{stall}} = K_t \cdot \frac{V_{\text{battery}}}{R}$$

Where:

- $K_t = \frac{\text{Rated Torque}}{\text{Rated Current}} = \frac{6}{14} = 0.43$ Nm/A
- $R = \frac{V^2}{P} = \frac{36^2}{250} = 5.18\ \Omega$ (ideal), assumed actual: 1.28 Ω

Thus,

$$\tau_{\text{stall}} = \frac{32 \cdot 0.43}{1.28} \approx 10.75 \text{ Nm}$$

The calculated stall torque (10.75 Nm) is well above the required 6 Nm, ensuring sufficient starting torque. This overhead enables the motor to handle:

- Static friction and inertia at startup
- Sudden loads such as climbing inclines
- Smooth acceleration without stalling

4.4.5 BLDC Motor Operation and Features

Function: The BLDC motor converts electrical energy from the battery into mechanical motion via electronic commutation (using Hall sensors), thereby eliminating losses associated with brushes.

Advantages:

- **Precision:** Fine speed control through electronic commutation.
- **Efficiency:** Reduced friction leads to higher efficiency.
- **Durability:** Extended operational life with minimal maintenance.
- **Cooling:** Fixed stator design enhances heat dissipation.
- **Performance:** Capable of high RPM and rapid acceleration.

4.4.6 Integration and Mounting

The final drivetrain design for the electric foldable scooter utilized a compact and efficient transmission using a 36V 250W BLDC motor and external planetary gearbox. The motor drives the sun gear of the planetary gear set with no belt or chain drive between the motor and the rear tire. The BLDC motor is mounted to the chassis and drives the sun gear of the planetary gear set. The ring gear is rigidly held against the scooter frame while the planet carrier is rigidly attached to the rear wheel axle. A 4:1 gear reduction at the wheel greatly increases the peak torque at the wheel with helps with acceleration and low-speed manoeuvrability without taking away from the vehicle size. In addition to the 4:1 gear reduction we achieved a direct coupling of the motor output shaft to the axle which avoided energy losses of any kind synonymous with belt- and chain-drives. The reduction in losses achieves together a gain the efficiency and reliability of the scooter without cluttering the mechanical layout of the scooter. With the motor, gearbox and direct coupling into the wheel we achieve the right balance of performance and simplicity for the available power of the foldable scooter.

4.4.7 System Design Summary

The drivetrain includes a 36V, 250W BLDC hub motor (6 Nm, 300 RPM) paired with a 0.9 kg 4:1 planetary gearbox, delivering 24 Nm output torque. Power is provided by a 36V, 7.8 Ah Li-ion battery (280.8 Wh), offering efficient performance for urban use.

Section 5

Transmission and Gears

5.1 Selection of Transmission

Table 5.1: Comparison of Transmission Options

	Chain Sprocket	**Single Gear**	**Multi Gear**	**Belt Drive**	**Planetary Gear**
Design	Uses a chain to transfer power from motor to rear wheel via sprockets	Uses a single fixed gear ratio connecting motor to wheel	Employs multiple gear ratios for performance optimization	Uses toothed belts and pulleys for smoother, quieter drive	Consists of a central sun gear, planet gears, and a ring gear; compact and often integrated into hub motors
Efficiency	High, with some frictional losses	Very high due to direct transmission	High, depending on gear selection	Moderately high, slight belt slippage losses	High; compact design reduces power loss
Performance	Delivers strong torque; good for acceleration	Smooth but with limited torque at higher speeds	Allows a wide range of speed-torque combinations	Smooth ride with moderate torque	Offers good torque density; supports compact high-power drivetrains

Continued on next page...

Table 5.1 – Continued

	Chain Sprocket	**Single Gear**	**Multi Gear**	**Belt Drive**	**Planetary Gear**
Simplicity	Mechanically simple but requires alignment	Very simple and easy to implement	Mechanically complex with shifting mechanism	Simple and lightweight	Mechanically intricate, but space-efficient; no external shifting mechanism
Maintenance	Needs regular lubrication and tensioning	Minimal maintenance	Requires frequent servicing and lubrication	Low maintenance; no lubrication needed	Low maintenance when sealed; internal components may wear over time
Life Expectancy	Moderate; chain wears over time	High; minimal wear parts	High if maintained well	Moderate; belt may stretch or wear	High; durable if not overloaded and properly sealed

Considering the above, planetary gears are a great fit for electric scooters because of their small size and high torque density, so they are the best for applications where space is limited. With their design, efficient power transfer is achieved and the overall drivetrain remains small and enclosed, ideal for use with hub motors. Planetary gear systems need little maintenance compared to chain or multi-gear systems and give consistent performance for extended periods of time. They also offer a well-balanced compromise between simplicity and performance without requiring external shifting units. All in all, planetary gears integrate efficiency, ruggedness, and minimal maintenance, offering a dependable and convenient option for contemporary electric scooters.

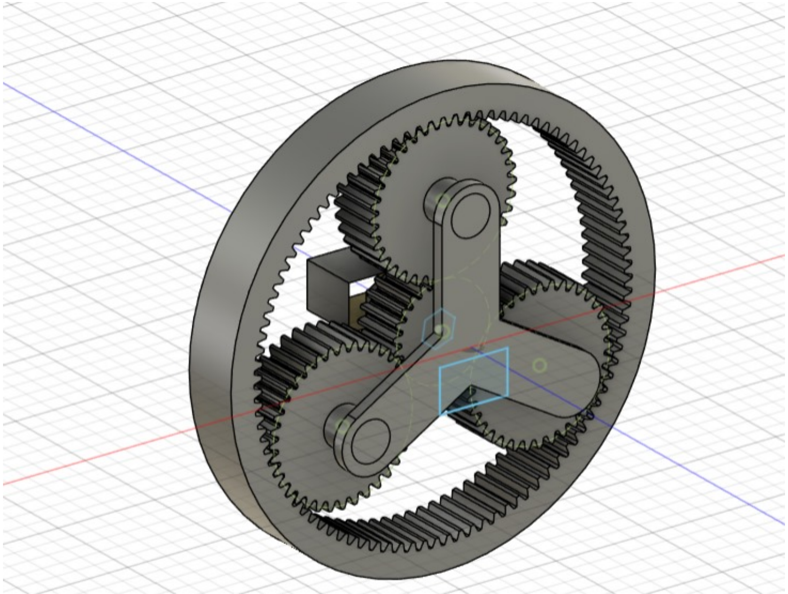

Figure 5.1: Planetary Gear cad

5.2 Selection of type of Gear reduction ratio

Willis equation for planetary gears, the following fundamental equation was derived describing the motion of sun gear (s), ring gear (r) and carrier (c) of a planetary gear:

$$n_r \cdot z_r = n_c \cdot (z_r + z_s) - z_s \cdot n_s$$

Fixed sun gear: If the sun gear is fixed (ns=0) and the gearbox input is carried out by the ring gear and the output by the carrier, the following transmission ratio is=nr/nc results according to equation:

$$i_s = 1 + \frac{z_s}{z_r}$$

Fixed ring gear: when the ring gear is fixed (nr=0) and the gearbox input is carried out by the sun gear and the output by the carrier. This results in the following transmission ratio ir=ns/nc:

$$i_s = 1 + \frac{z_r}{z_s}$$

Fixed carrier; A last possibility for the transmission ratio is obtained when the carrier ist fixed and the gearbox input is carried out by the sun gear and the output by the ring gear. In this case the following transmission ratio i0=ns/nr results:

$$i_o = -\frac{z_r}{z_s}$$

The highest gear reduction is obtained by keeping ring gear stationary and considering sun gear as input and planet gear as output .Also, the ring gear is generally easy to be kept stationary as compared to planet carrier.

5.3 Gear reduction ratio calculation

Given:

- Radius of wheel, $r = 4\,\text{in} = 101.6\,\text{mm} = 0.1016\,\text{m}$
- Acceleration, $a = 2.22\,\text{m}\,\text{s}^{-2}$
- Mass of the vehicle $(m) = 100\,\text{kg}$
- Maximum torque provided by the motor, $(T_{\text{motor}}) = 6\,\text{N}\,\text{m}$

Torque required at the differential:

$$T_{\text{diff}} = m \cdot a \cdot r = 100\,\text{kg} \cdot 2.22\,\text{m}\,\text{s}^{-2} \cdot 0.1016\,\text{m} = 22.55\,\text{N}\,\text{m}$$

Final Drive Ratio:

$$\text{Final Drive Ratio} = \frac{T_{\text{diff}}}{T_{\text{motor}}} = \frac{22.55\,\text{N}\,\text{m}}{6\,\text{N}\,\text{m}} \approx 4$$

5.4 Planetary Gear Calculations

Given:

- Module $(m) = 2$
- Number of teeth on ring gear $(T_r) = 68$
- Gear reduction ratio$(GR) = 4$

From the gear reduction ratio formula for a planetary gear:

$$GR = 1+\frac{T_r}{T_s} \Rightarrow 4 = 1+\frac{68}{T_s} \Rightarrow \frac{68}{T_s} = 3 \Rightarrow T_s = \frac{68}{3} = 22.67 \approx 22 \text{ (rounded for practical design)}$$

Number of teeth on sun gear:

$$T_s = 22$$

Using the planetary gear relation:

$$T_r = T_s + 2 \cdot T_p \Rightarrow 68 = 22 + 2 \cdot T_p \Rightarrow T_p = \frac{68-22}{2} = 23$$

Therefore,

- Number of teeth on planet gear, $T_p = 23$

Ring Gear Outer Diameter:

$$D_r = T_r \cdot m = 68 \cdot 2 + 25 = 161\,\text{mm}$$

(Note: The no. of ring gear is decided beforehand based on the commercial models available. Also, number of teeth between 60 and 120 is considered favorable for ring gears.)

5.5 Input shaft fatigue analysis

Given:

- Torque$(T) = 6000\,\text{N mm}$
- Radius $(r) = 80.5\,\text{mm}$
- Pressure angle $(\phi) = 20°$
- Diameter of shaft$d$ (used in stress formulas)

Tangential Load:

$$F_t = \frac{T}{r} = \frac{6000}{80.5} = 74.5\,\text{N}$$

Normal Load:

$$F_n = \frac{F_t}{\cos(\phi)} = \frac{74.5}{\cos(20°)} = 79.26\,\text{N}$$

Bending Moment on Gear:

$$M = F_t \cdot r = 74.5 \cdot 80.5 = 5997\,\text{N mm}$$

Bending Stress:

$$\sigma = \frac{32M}{\pi d^3} \Rightarrow \sigma = 40.2\,\text{MPa} \quad \text{(for given } d\text{)}$$

Torsional Stress:

$$\tau = \frac{16T}{\pi d^3} \Rightarrow \tau = 20.1\,\text{MPa} \quad \text{(for same } d\text{)}$$

Von Mises Equivalent Stress:

$$\sigma_{\text{eq}} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{(40.2)^2 + 3(20.1)^2} = 53.2\,\text{MPa}$$

Material: Al 6065
Yield Strength: $\sigma_{\text{yield}} = 290\,\text{MPa}$
Conclusion: Since

$$\sigma_{\text{eq}} = 53.2\,\text{MPa} < \sigma_{\text{yield}} = 290\,\text{MPa}$$

The shaft will not yield under the given loading conditions.

5.6 Gear fatigue analysis

Given:

- Face width, $F = 16\,\text{mm}$
- Module, $m = 2\,\text{mm}$
- Number of teeth on sun gear (input), $N_i = 22$
- Number of teeth on planet gear (output), $N_o = 23$
- Pitch diameter, $d_p = m \cdot N = 2 \cdot 22 = 44\,\text{mm} = 0.044\,\text{m}$
- Torque, $T = 9.3\,\text{N}\,\text{m}$
- Rotational speed, $N = 250\,\text{rpm}$
- Pressure angle, $\phi = 20°$
- Quality number, $Q_v = 8$
- Geometry factor for bending strength,
  - For inpur gear : $Y_j = 0.328$
  - For outpput gear: $Y_j = 0.335$

Pitch Line Velocity

$$V = \frac{\pi d_p N}{60} = \frac{\pi \cdot 0.16 \cdot 250}{60} = 2.14\,\text{m}\,\text{s}^{-1}$$

Tangential Load

$$W_t = \frac{2T}{d_p} = \frac{2 \cdot 9.3}{0.16} = 116.25\,\text{N}$$

Dynamic Factor K_v

$$B = \frac{0.25(12 - Q_v)^2}{3} = \frac{0.25(4)^2}{3} = 0.62$$

$$A = 50 + 56(1 - B) = 50 + 56(1 - 0.62) = 71.28$$

$$K_v = \left(\frac{A + \sqrt{V}}{A}\right)^B = \left(\frac{71.28 + \sqrt{2.14}}{71.28}\right)^{0.62} \approx 1.17$$

Other Load Factors

- Overload Factor, $K_o = 1$
- Rim Thickness Factor, $K_b = 1$
- Load Distribution Factor, $K_h = 1$ (due to good mounting and small face width)
- Size Factor, $K_s = 1$ (since $m < 5\,\text{mm}$)

Bending Stress

- For input gear:

$$\sigma = \frac{K_o \cdot K_v \cdot W_t \cdot K_s \cdot K_h \cdot K_b}{F \cdot m \cdot Y_j} = \frac{1 \cdot 1.17 \cdot 116.25 \cdot 1 \cdot 1 \cdot 1}{16 \cdot 2 \cdot 0.328} = 16.8\,\text{MPa}$$

- For output gear:

$$\sigma = \frac{K_o \cdot K_v \cdot W_t \cdot K_s \cdot K_h \cdot K_b}{F \cdot m \cdot Y_j} = \frac{1 \cdot 1.17 \cdot 116.25 \cdot 1 \cdot 1 \cdot 1}{16 \cdot 2 \cdot 0.335} = 12.68\,\text{MPa}$$

Given:

- Bending Strength, $S_t = 221.55\,\text{MPa}$
- Temperature Factor, $Y_\theta = 1$
- Reliability Factor, $Y_z = 1.25$ (for reliability $= 0.999$)
- Load Cycles, $N = 10^8$
- Life Factor for Bending,
  - For input gear : $Y_N = 4.9404 \cdot N^{-0.1045}$
  - For output gear : $Y_N = 4.9404 \cdot (N/4)^{-0.1045}$
- Calculated Bending Stress

- For input gear: $\sigma = 16.8\,\text{MPa}$
- For output gear: $\sigma = 12.68\,\text{MPa}$

- Life Factor
  - Input Gear:
$$Y_N = 4.9404 \cdot (10^8)^{-0.1045} = 0.72$$
  - Output Gear:
$$Y_N = 4.9404 \cdot (10^8/4)^{-0.1045} = 0.75$$

Allowable Bending Stress:

- $$\sigma_{\text{all}} = \frac{S_t \cdot Y_N}{Y_\theta \cdot Y_z} = \frac{221.55 \cdot 0.72}{1 \cdot 1.25} = 127.29\,\text{MPa}$$
- $$\sigma_{\text{all}} = \frac{S_t \cdot Y_N}{Y_\theta \cdot Y_z} = \frac{221.55 \cdot 0.75}{1 \cdot 1.25} = 132.93\,\text{MPa}$$

Factor of Safety:

- For input gear:
$$S_{\text{fg}} = \frac{\sigma_{\text{all}}}{\sigma} = \frac{127.29}{16.8} \approx \boxed{7.59}$$
- For output gear :
$$S_{\text{fg}} = \frac{\sigma_{\text{all}}}{\sigma} = \frac{127.29}{16.8} \approx \boxed{10.4}$$

Conclusion: The input gear operates with a high factor of safety of **7.59** and the putput gear operates with a high factor of safety of **10.4**, indicating that it is well within safe design limits for bending failure.

Results

- The planetary gear system is chosen for its high torque density, compact size, and efficient power transfer, making it ideal for electric scooters.
- Among various drivetrain options, planetary gears offer the best space-efficiency, smooth ride, and minimal maintenance.
- Required torque at wheels calculated as $22.55\,\text{Nm}$ to achieve an acceleration of $2.22\,\text{m/s}^2$ for a $100\,\text{kg}$ scooter.

- Motor torque is 6 Nm, resulting in a required gear reduction ratio of approximately 4.
- Using the gear reduction formula:
  - Ring gear teeth (T_r) = 68
  - Sun gear teeth (T_s) = 22 (rounded from 22.67)
  - Planet gear teeth (T_p) = 23
- Outer diameter of ring gear calculated as 161 mm, based on module = 2 and standard size constraints.
- Input shaft analysis shows:
  - Von Mises stress = 53.2 MPa
  - Yield strength of Al 6065 = 290 MPa
  - $\Rightarrow$ Shaft will not yield under applied torque.
- Gear fatigue analysis:
  - Tangential load (W_t) = 116.25 N
  - Pitch line velocity (V) = 2.14 m/s
  - Dynamic factor (K_v) $\approx$ 1.17
- Bending stress:
  - Input gear: 16.8 MPa
  - Output gear: 12.68 MPa
- Allowable bending stress:
  - Input gear: 127.29 MPa
  - Output gear: 132.93 MPa
- Factors of safety:
  - Input gear: 7.59
  - Output gear: 10.4
- The design provides a high safety margin, ensuring reliability and long life under operational conditions.

Section 6

Wheel Design

6.1 Wheel Selection and Material Analysis

The wheel affects ride comfort, performance, and mobility. There are two main wheel dimensions: **diameter** (the perpendicular distance to the plane of rotation, around the wheel) and **width** or tire thickness (the distance from one edge of the rim to the other). The 8–10 inch diameter range is commonly used in the market.

Large wheels (8.5–13 in) are excellent shock absorbers and have slower pickup due to inertia, higher grip, and high stability at higher speeds. Small wheels (<8.5 in) provide quick acceleration, low stability at higher speeds, lower grip, and high control. While larger wheels offer better ride quality, their mass can hinder the foldability and portability of the scooter. Smaller wheels deliver speed but compromise stability and obstacle handling.

Selected wheel size: Diameter 10 inches, width 2.0–2.5 inches. Rim diameter is 8 inches and brake disc diameter is 6 inches. This size ensures adequate ground clearance for urban obstacles, stable handling at moderate speeds, and portability-friendly dimensions. It is compatible with standard decks and motors, and ensures cost efficiency with available tires and rims.

6.1.1 Wheel Rim Material Choice

Aluminium alloys are optimal materials for wheel rims in electric vehicles, due to their strength and low weight. Detailed analysis by Korkut et al. (2020) focused on Al6063 T6 and Al5083 alloys. Key conclusions:

- **Strength-to-Weight Ratio:** Both alloys have high tensile strengths and low densities, making wheel rims both strong and lightweight.

- **Fatigue Resistance:** The wheels can resist heavy cyclic loads, with equivalent stresses below critical levels, ensuring durability over prolonged periods.

- **Modal Characteristics:** Modal analysis shows good resistance to vibrational stresses due to favorable natural frequencies, ensuring ride comfort and structural integrity under all operating conditions.

6.1.2 Tire Material and Types

The tire material is styrene-butadiene rubber (SBR), comprising 75% butadiene and 25% styrene, both petroleum-derived. Butadiene is a by-product of ethylene production from steam crackers. Styrene is produced by dehydrogenating ethylbenzene, itself derived from ethylene and benzene.

There are two main tire types:

- **Pneumatic (air-filled) tires:** Provide excellent comfort as air cushioning reduces vibrations and impacts from road irregularities. They offer better traction due to flexible construction and tread design, especially on wet and uneven surfaces, and adaptive structure ensures greater control. However, they require frequent maintenance (pressure checks), are prone to punctures, and have higher maintenance overhead.

- **Solid (airless) tires:** Puncture-proof and require minimal maintenance, as there's no need for air pressure management. They are more durable, especially in consistent urban environments. However, they offer less comfort (due to lack of shock absorption), limited grip in wet conditions, and more complex installation.

For urban commuting, where long-term durability and minimal maintenance are key, **solid tires are more suitable**.

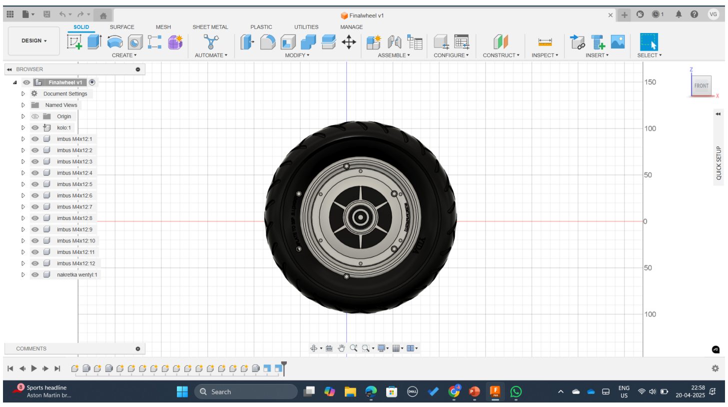

Figure 6.1: CAD Design of Wheel

6.1.3 Wheel & Deck Specifications

- **Wheel Diameter:** 10 inches = 0.254 m
- **Tire Width:** 2.5 inches = 0.0635 m
- **Rim Diameter:** 8 inches = 0.203 m
- **Deck Dimensions:** 60 cm (L) × 20 cm (W) = 0.6 m × 0.2 m

Load Distribution

Total Mass and Weight

$$\begin{aligned}\text{Vehicle Mass} &= 20\,\text{kg}\\ \text{Passenger Mass} &= 80\,\text{kg}\\ \text{Total Mass} &= 100\,\text{kg}\\ \text{Total Weight (W)} &= 100 \times 9.81 = 981\,\text{N}\end{aligned}$$

Static Load Distribution (60-40)

$$\begin{aligned}\text{Rear Wheel Load (Static)} &= 0.6 \times 981 = 588.6\,\text{N}\\ \text{Front Wheel Load (Static)} &= 0.4 \times 981 = 392.4\,\text{N}\end{aligned}$$

Dynamic Load Distribution (Acceleration = $2.22\,\text{m/s}^2$)

$$\begin{aligned}\text{CG Height (h)} &= 0.4\,\text{m}\\ \text{Wheelbase (L)} &= 1.2\,\text{m}\\ \Delta F_z &= \frac{100 \times 2.22 \times 0.4}{1.2} = 74\,\text{N}\\ \text{Total Rear Load (Dynamic)} &= 588.6 + 74 = 662.6\,\text{N}\end{aligned}$$

Rim Stress Analysis (Material: Al6063-T6)

Material Properties

- **Yield Strength:** $214\,\text{MPa}$
- **Fatigue Strength:** $68.9\,\text{MPa}$ (at 5×10^8 cycles)
- **Shear Strength:** $152\,\text{MPa}$

Spoke Load from Motor Torque

$$T = 24\,\text{Nm}, \quad R = 0.127\,\text{m}$$

$$F_t = \frac{T}{R} = \frac{24}{0.127} = 188.97\,\text{N}$$

Shear Stress from Torque

For a 3 mm diameter spoke: $r = 1.5\,\text{mm} = 0.0015\,\text{m}$

$$A_{\text{spoke}} = \pi r^2 = \pi(0.0015)^2 = 7.07 \times 10^{-6}\,\text{m}^2$$

$$\tau_{\text{torque}} = \frac{F_t}{A_{\text{spoke}}} = \frac{188.97}{7.07 \times 10^{-6}} = 26.72\,\text{MPa}$$

Vertical Load per Spoke

$$F_{\text{weight, spoke}} = \frac{F_{z,\text{dynamic}}}{4} = \frac{662.6}{4} = 165.65\,\text{N}$$

Shear Stress from Weight

$$\tau_{\text{weight}} = \frac{F_{\text{weight, spoke}}}{A_{\text{spoke}}} = \frac{165.65}{7.07 \times 10^{-6}} = 23.42\,\text{MPa}$$

Combined Shear Stress

$$\tau_{\text{total}} = \sqrt{\tau_{\text{torque}}^2 + \tau_{\text{weight}}^2} = \sqrt{26.72^2 + 23.42^2} = 35.53\,\text{MPa}$$

Shear Safety Factor

Using the shear yield strength of Al6063-T6: $\tau_{\text{yield}} = 152\,\text{MPa}$

$$SF = \frac{\tau_{\text{yield}}}{\tau_{\text{total}}} = \frac{152}{35.53} = 4.28(Safe)$$

Tire Contact Patch & Ground Pressure

Tire Deflection and Contact Area

$$\text{Tire Radius} = 0.127\,\text{m}$$
$$\delta = 0.1 \times 0.127 = 0.0127\,\text{m}$$
$$L = 2\sqrt{2 \times 0.127 \times 0.0127} = 0.113\,\text{m}$$
$$A_c = L \times \text{Width} = 0.113 \times 0.0635 = 0.0072\,\text{m}^2$$

Peak Ground Pressure

$$\text{Total Rear Load (Dynamic)} = 588.6 + 74 = 662.6\,\text{N}$$

$$P_{\text{peak}} = \frac{3}{2} \times P_{\text{avg}} = 1.5 \times P_{\text{avg}}$$

$$P_{\text{peak}} = \frac{1.5 \times 662.6}{0.0072} = 13.8\,\text{kPa}$$

Section 7

Suspension and Springs

7.1 Suspension Analysis

The spring used in the front suspension of a bicycle is part of the suspension fork, which helps absorb shocks and smooth the ride on rough terrain. There are two main types of suspension systems used in bicycle forks:

1. Coil Spring Suspension

The coil spring suspension fork contains a metal coil spring within the fork legs that compresses when the wheel encounters an obstruction, absorbing the shock and then returning to its original shape for stability and comfort. This system delivers excellent durability and reliable performance in absorbing large impacts, although it may provide a slightly rougher ride compared to air suspension and offers limited adjustability.

2. Air Spring Suspension

This system uses compressed air within a sealed chamber rather than a metal coil, featuring a progressive compression rate where the force increases more rapidly with displacement. Air suspension is lighter and highly adjustable by modifying the air pressure, making it popular on high-performance bicycles. However, it generally requires more maintenance and is prone to gradual air loss over time.

7.2 Coil Spring Suspension Details

A coil spring suspension fork uses a metal spring (typically steel or titanium for reduced weight) in tandem with a damping system that controls compression and rebound. A preload adjuster allows fine-tuning of stiffness based on rider weight and terrain conditions, while the stanchions and lowers guide the suspension's movement.

7.2.1 Design Considerations

$$\begin{aligned}
\delta &= 1\,\text{cm} \\
L_0 &= 20\,\text{cm} \\
F_{\max} &= 400\,\text{N} \\
F_{\min} &= 0\,\text{N} \\
F_{\max} &= K\delta \quad \Rightarrow \quad K = \frac{400}{1} = 40\,\text{kN/m} \\
\tau &= 2\pi\sqrt{\frac{m}{K}} = 2\pi\sqrt{\frac{80}{40000}} = 0.281\,\text{sec}
\end{aligned}$$

For static loading:

$$t_a > 3\tau \quad \Rightarrow \quad t_a > 0.843\,\text{sec}$$

Assuming a spring index $C = 6$ and choosing a mean coil diameter $D = 4\,\text{cm}$, we have:

$$C = \frac{D}{d} \quad \Rightarrow \quad d = \frac{4}{6}\,\text{cm} \approx 0.67\,\text{cm} \quad (\text{choose } d = 6.5\,\text{mm})$$

$$C = \frac{4}{0.65} \approx 6.15$$

7.2.2 Material Properties

Material chosen: Chrome Silicon

$$E = 203.4\,\text{GPa}, \quad G = 77.2\,\text{GPa}$$

$$F_s = (1+\xi)F_{\max}, \quad \xi = 0.2 \quad \Rightarrow \quad F_s = 1.2 \times 400 = 480\,\text{N}$$

$$\begin{aligned}
A &= 1974\,\text{MPa}\cdot\text{mm}^m, \quad m = 0.108 \\
S_{ut} &= \frac{A}{d^m} = \frac{1974}{(6.5)^{0.108}} \approx 1612.69\,\text{MPa} \\
S_{yt} &= 0.75\,S_{ut} \approx 1209.52\,\text{MPa} \\
S_{sy} &= 0.577\,S_{yt} \approx 697.89\,\text{MPa} \\
S_{su} &= 0.67\,S_{ut} \approx 1080.50\,\text{MPa}
\end{aligned}$$

7.2.3 Stress Calculations

$$K_B = \frac{4C+2}{4C-3} \approx 1.231$$

$$\tau_s = \frac{8K_B F_s D}{\pi d^3} \approx 219.16\,\text{MPa}$$

$$\eta_s = \frac{S_{sy}}{\tau_s} \approx 3.18$$

$$\tau_{\max} = \frac{8K_B F_{\max} D}{\pi d^3} \approx 182.63\,\text{MPa}$$

$$\eta_{\max} = \frac{S_{sy}}{\tau_{\max}} \approx 3.82$$

7.2.4 Fatigue Design

$$F_a = \frac{F_{\max} - F_{\min}}{2} = 200\,\text{N}, \quad F_m = \frac{F_{\max} + F_{\min}}{2} = 200\,\text{N}$$

$$\tau_a = \tau_m = \frac{8K_B F_a D}{\pi d^3} \approx 91.32\,\text{MPa}$$

For an unpeened spring:

$$S_{sm}^z = 241\,\text{MPa}, \quad S_{sa}^z = 379\,\text{MPa}$$

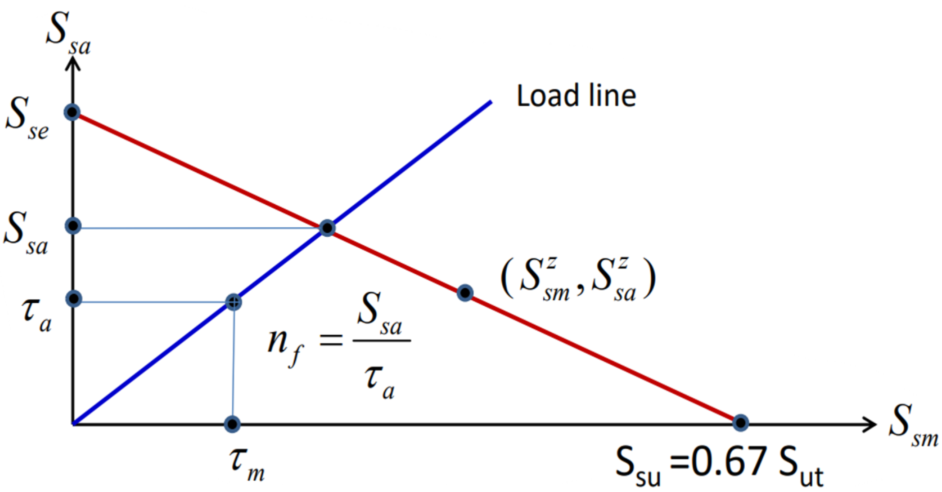

Figure 7.1: Modified Goodman Diagram for fatigue design [1]

7.2.5 Equivalent Strength Calculation

Using the linear relation $y = mx + c$:

$$0 = (-0.3435)(1080.50) + c \quad \Rightarrow \quad c \approx 371.21$$

Thus, the equivalent strength is:

$$S_{se} = 371.21\,\text{MPa}$$

And by solving $y = x$:

$$y = (-0.3435)x + 371.21 \quad \Rightarrow \quad y \approx 276.30\,\text{MPa}$$

$$\eta_f = \frac{S_{sa}}{\tau_a} \approx 3.03$$

7.2.6 Buckling Analysis

$$L_{cr} = \left(\frac{\pi D}{\alpha}\right)\sqrt{\frac{2(E-G)}{2G+E}}, \quad \alpha = 0.5$$

$$L_{cr} \approx 211.09\,\text{mm} \quad \text{and} \quad L_0 = 200\,\text{mm} \quad \Rightarrow \quad \text{No buckling issues.}$$

7.2.7 Coil Specifications

$$K = \frac{Gd^4}{8D^3N_a} \quad \Rightarrow \quad N_a \approx 6.73 \quad \text{(round to 7)}$$

If the number of inactive coils $N_e = 2$, then the total number of coils $N_t = N_a + N_e = 9$. The pitch is given by:

$$P = \frac{L_0 - 2d}{N_a} \approx 26.71\,\text{mm}$$

7.2.8 Final Design Parameters

- **Free Length:** $L_0 = 20\,\text{cm}$
- **Maximum Deflection:** $\delta_{\max} = 1\,\text{cm}$
- **Mean Coil Diameter:** $D = 4\,\text{cm}$
- **Wire Diameter:** $d = 6.5\,\text{mm}$
- **Time Period:** $\tau = 0.281\,\text{sec}$
- **Spring Constant:** $K = 40\,\text{kN/m}$
- **Material:** Chrome Silicon
- **Static Safety Factor:** $\eta_s \approx 3.18$
- **Maximum Load Safety Factor:** $\eta_{\max} \approx 3.82$
- **Fatigue Safety Factor:** $\eta_f \approx 3.03$
- **Number of Active Coils:** $N_a = 7$
- **Pitch:** $P \approx 26.71\,\text{mm}$

Bibliography

[1] J. E. Shigley, L. D. Mitchell, and H. Saunders, *Mechanical Engineering Design*, 1985, pp. 145.

[2] ASM International, "6063-T6 Aluminum :: MakeItFrom.com Material Data Sheet," Available at: https://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA6063T6.

[3] GrabCAD Library, "CAD Models and Engineering Resources," Available at: https://grabcad.com/library.

[4] Electric Scooter Guide, "Electric Scooters and Wheel Sizes – Why Wheel Size is Important," Available at: https://scooter.guide/electric-scooters-and-wheel-sizes-why-wheel-size-is-important/.

[5] "Design and Comparative Strength Analysis of Wheel Rims," *International Journal of Engineering Research*.

[6] Design Life Cycle, "Electric Scooter: Materials and Sustainability," Available at: http://www.designlife-cycle.com/electric-scooter.

[7] Aventura-X, "Pneumatic vs Solid Tires," Available at: https://aventura-x.com/pneumatic-vs-solid-tires.

[8] Rider Guide, "Electric Scooter Tires – The Ultimate Guide," Available at: https://riderguide.com/guides/electric-scooter-tires/.

[9] Efficiency of Motor: https://powerdevicecorp.com/resources/FileManager/power/whitepapers/BLDC_Motor_Controllers_Perf_Efficiency_wp.pdf

[10] Rolling Resistance Coefficient: https://www.engineeringtoolbox.com/rolling-friction-resistance-d_1303.html

[11] Drag Coefficient: https://scholarsbank.uoregon.edu/server/api/core/bitstreams/f8af1331-c689-4e7c-baa2-8ac9d077417f/content