

1/10 Scale Model Car Platform Design CAD

Powertrain, Suspension, and
Steering System



Design Criteria

Chassis: Wheelbase = 370 mm, Tread = 176 mm

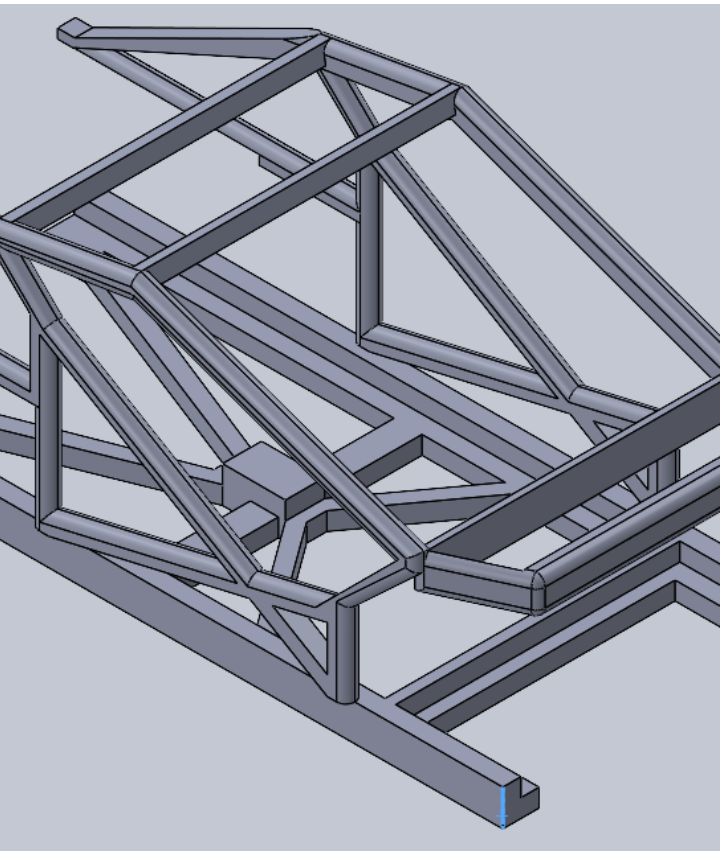
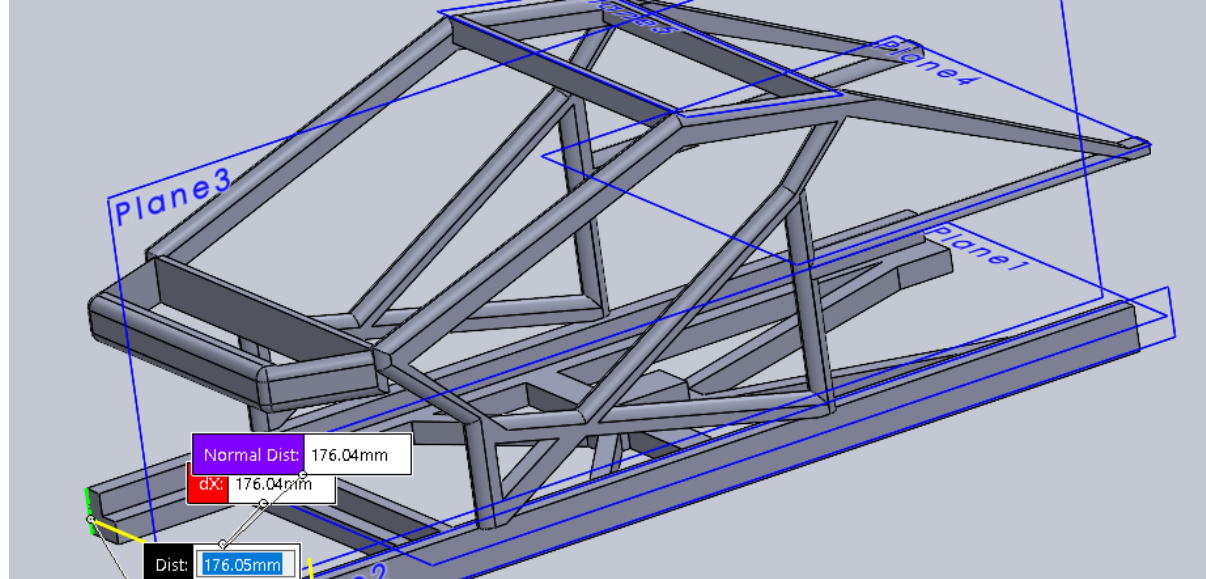
Engine: 50×50×50 mm, Transmission: 25×25×35 mm

Powertrain: RWD, Non-locking differential, Final drive ratio = 3.5

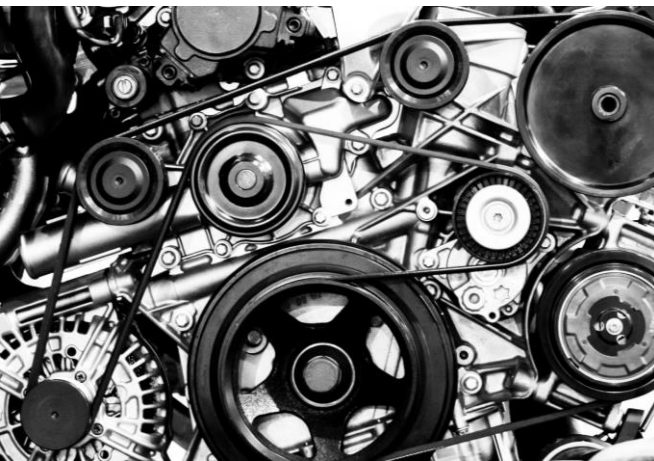
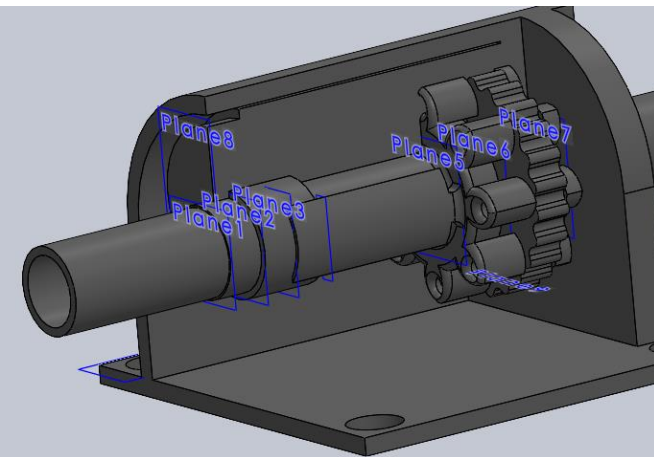
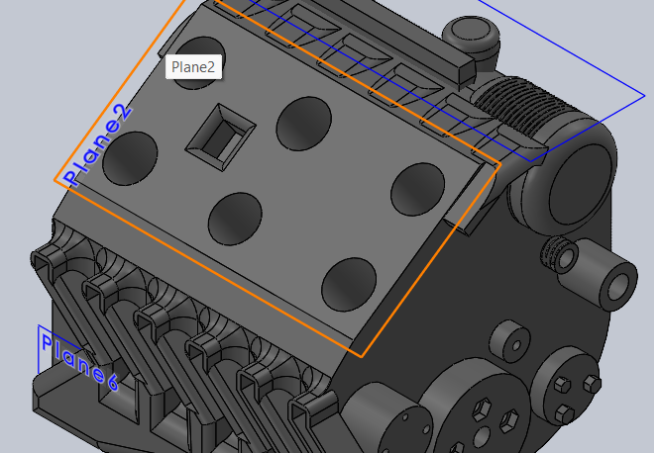
Suspensions: Front SLA, Rear Hotchkiss, 30–60% anti-squat

Steering: Gearbox ratio = 15, Ackerman trapezoidal, Self-centering

Car Shell Design



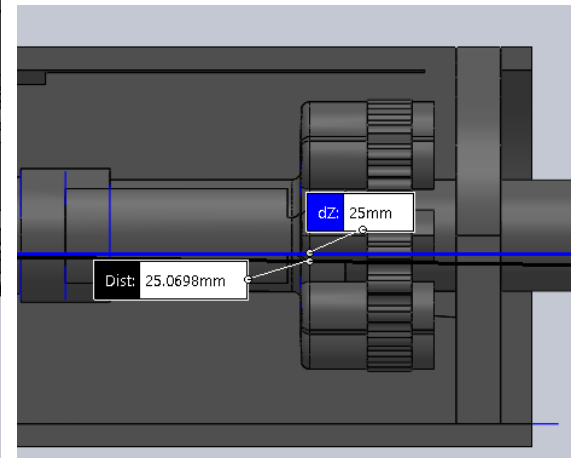
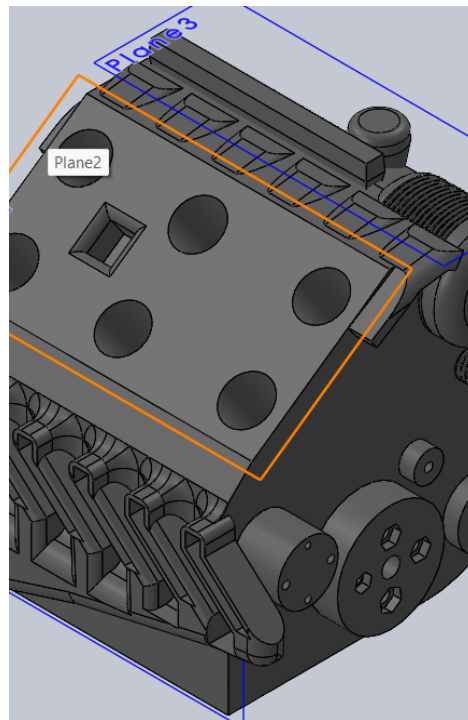
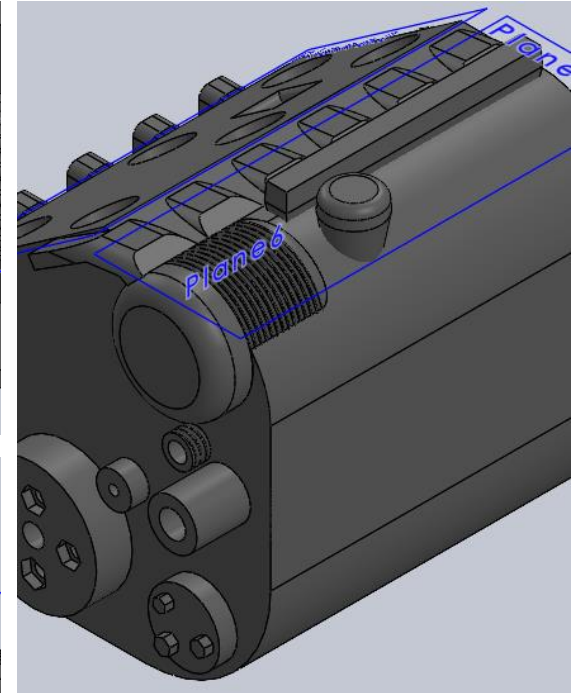
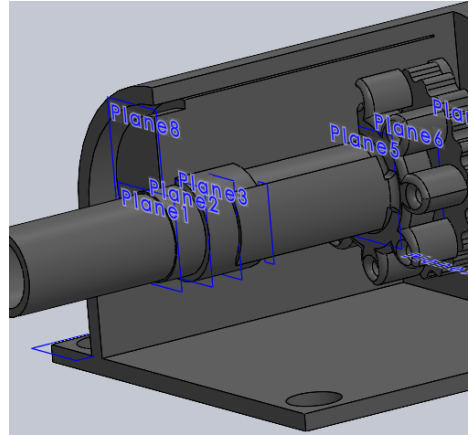
- CAD Sketches of frame rails and mounting points
- Shell is within specification

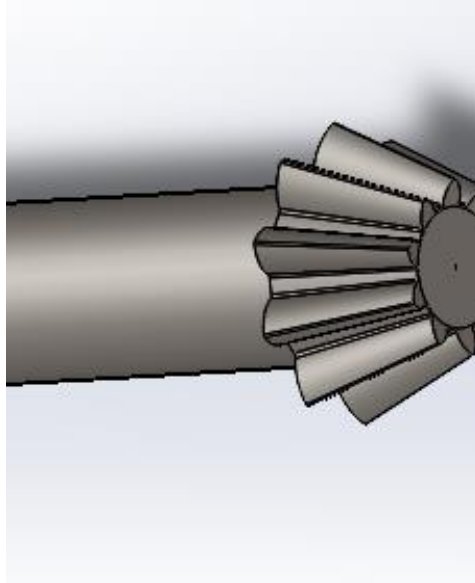
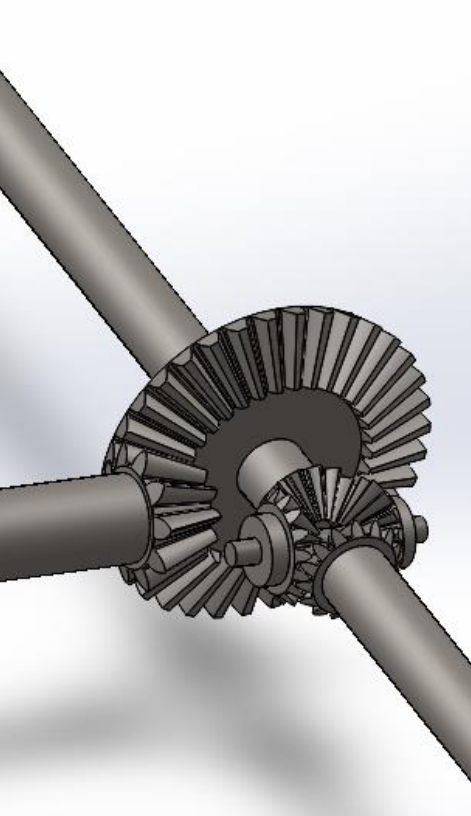


Powertrain

Engine & Transmission

- Transmission Dimensions: 25mm x 15mm x 20mm
- Engine Dimensions: 50mm x 50mm x 50mm

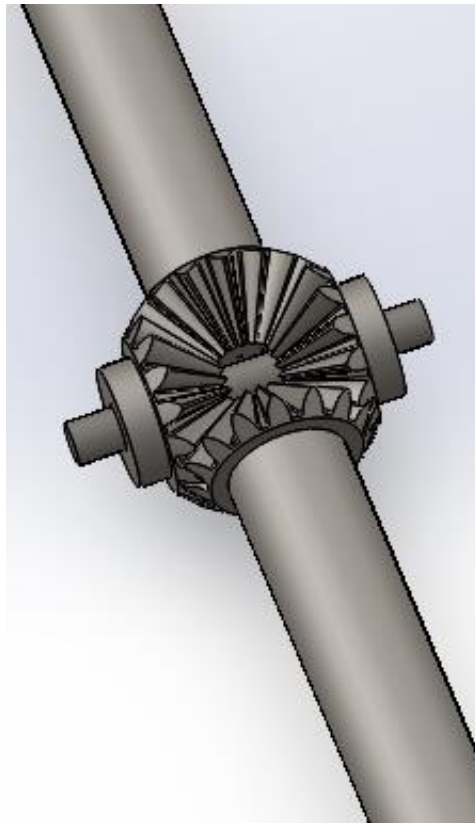
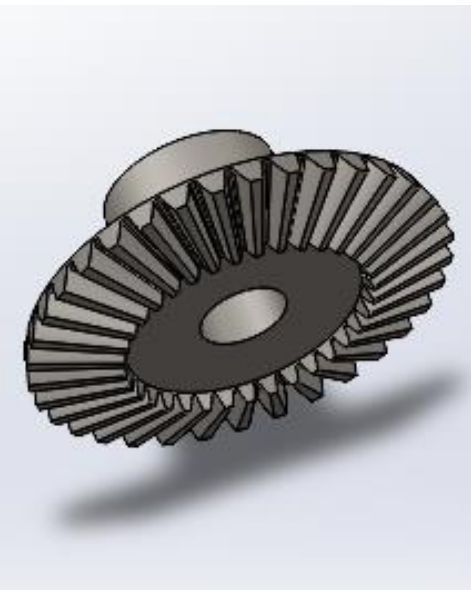




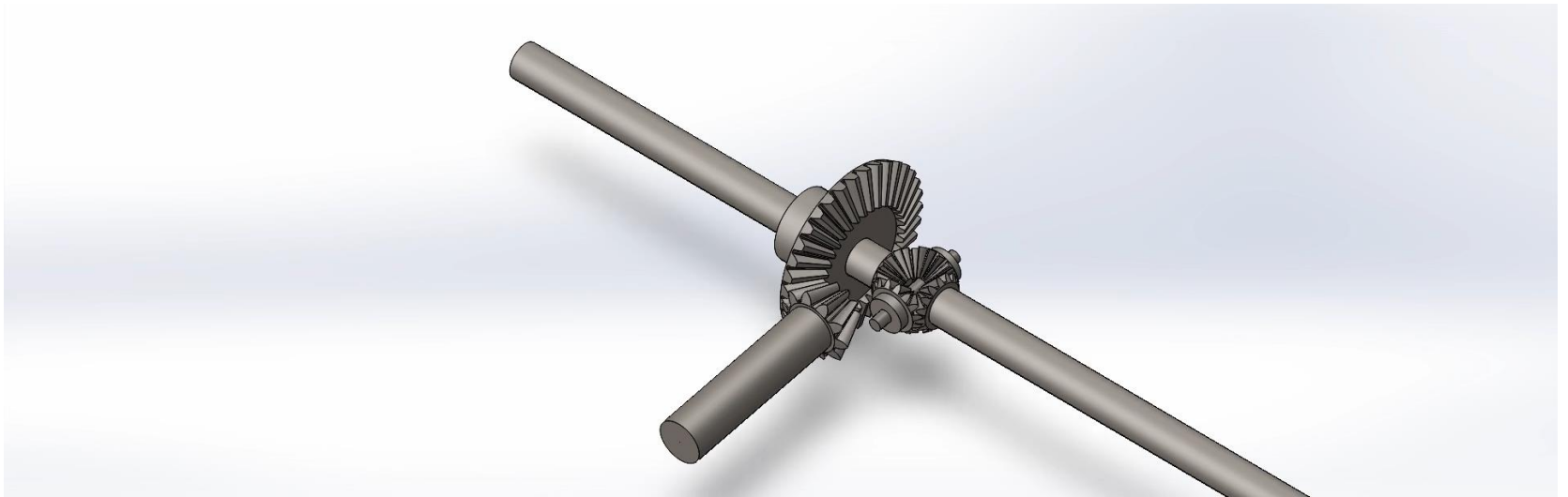
Differential

Final drive ratio: 3.5

- Gear has 35 teeth
- Pinion has 10 teeth



Differential Motion Study



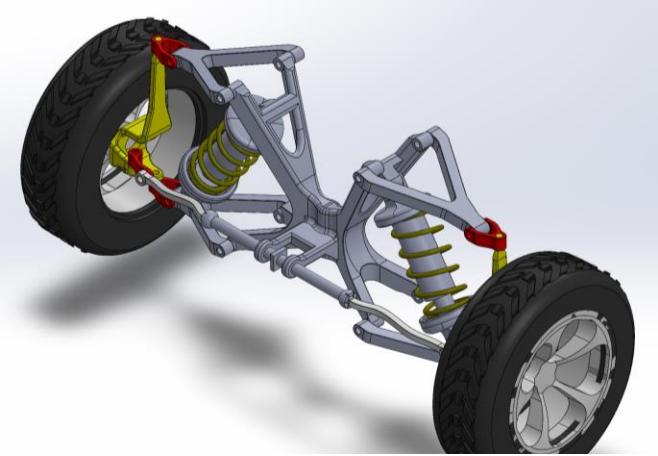
Differential Calculations

Gear Ratio Calculation

- Final drive ratio required = 3.5
- Gear Ratio = (Number of Ring Gear Teeth) $\div$ (Number of Pinion Teeth)
- **Gear Ratio = $35 \div 10 = 3.5$**

Bevel Gear Pitch Angle Calculation

- Pinion pitch angle (θ_p) = arctangent (Pinion Teeth $\div$ Ring Gear Teeth)
- $\theta_p = \text{arctangent}(10 \div 35)$
- **$\theta_p \approx 15.95$ degrees**
- Ring gear pitch angle (θ_r) = 90 degrees - Pinion pitch angle
- $\theta_r = 90 - 15.95$
- **$\theta_r \approx 74.05$ degrees**



Suspension Systems

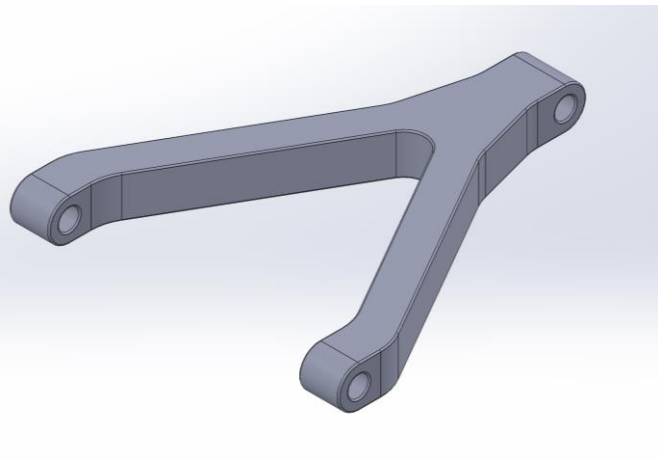


Front SLA:

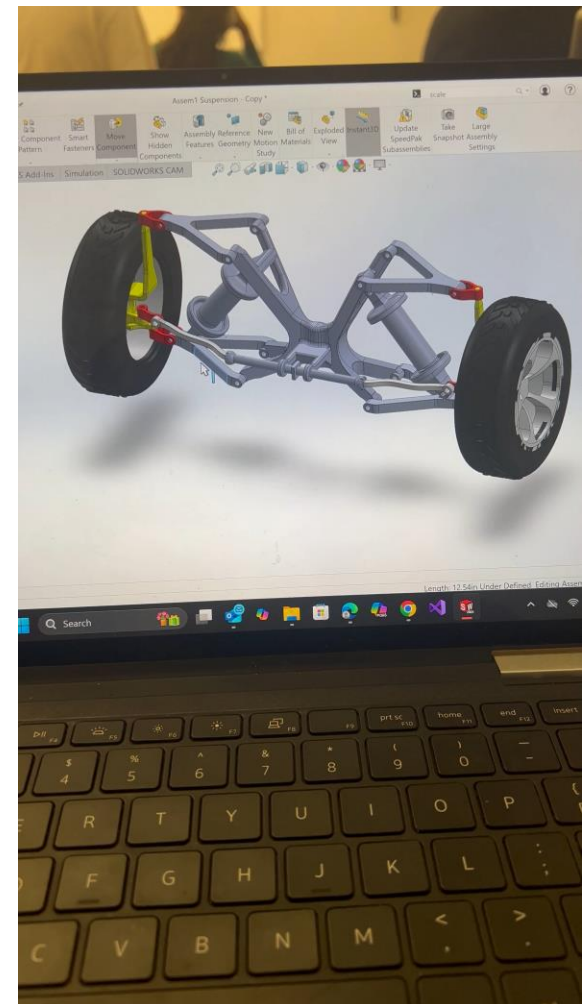
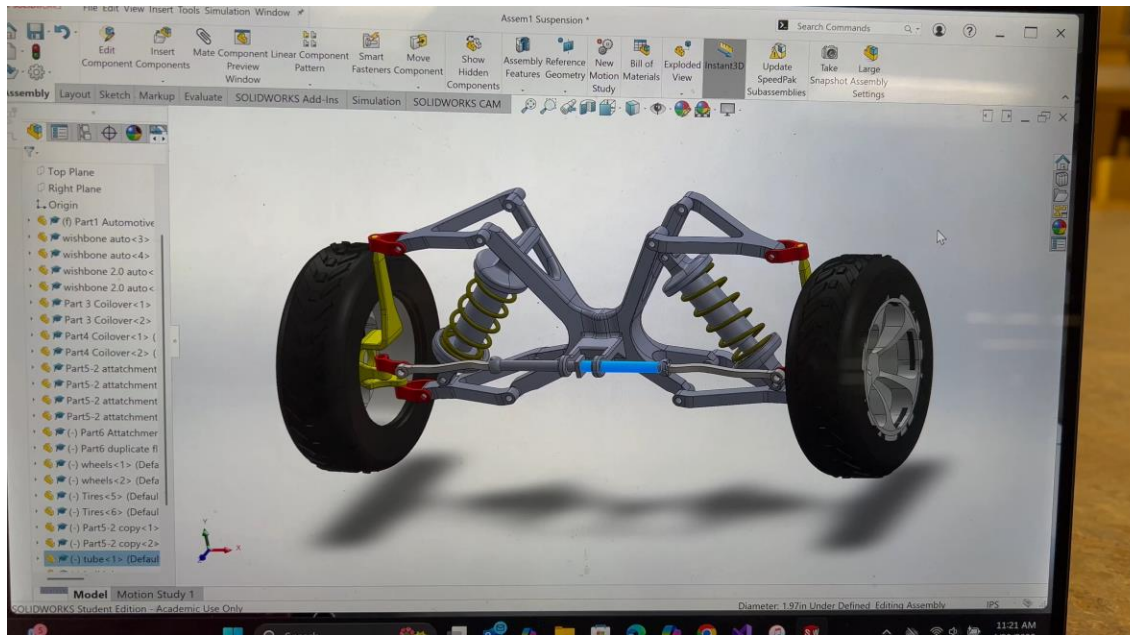
- Upper/lower arms geometry
- Roll center and camber control

Rear Hotchkiss:

- Solid axle layout
- Anti-squat analysis (30–60%)



Suspension/RWD-Video



Suspension Calculations

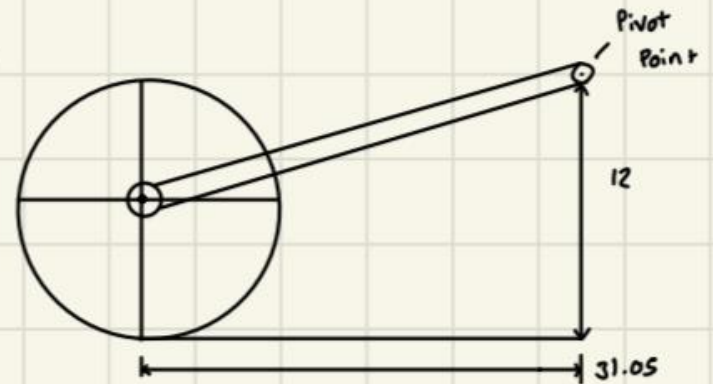
$$e = 12$$

$$d = 31.05$$

60% anti-squat

$$h = \frac{e}{d} \cdot \frac{L}{0.6} = \frac{12}{31.05} \cdot \frac{370}{6}$$

$$h = 23.83 \text{ cm}$$

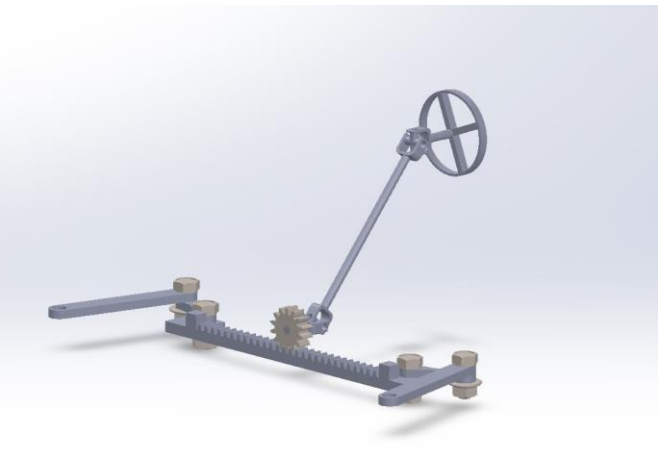
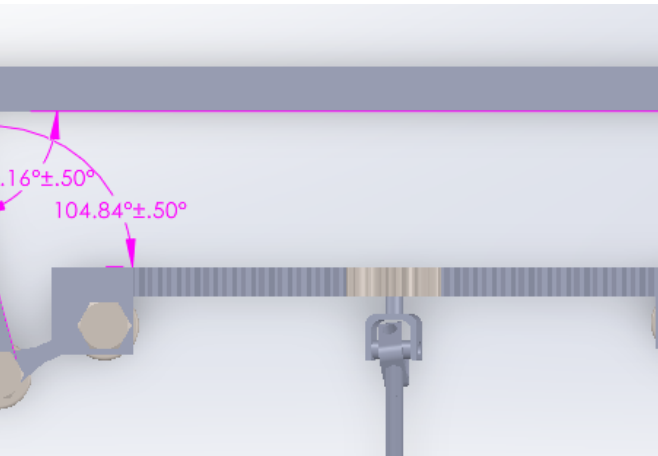




Steering System

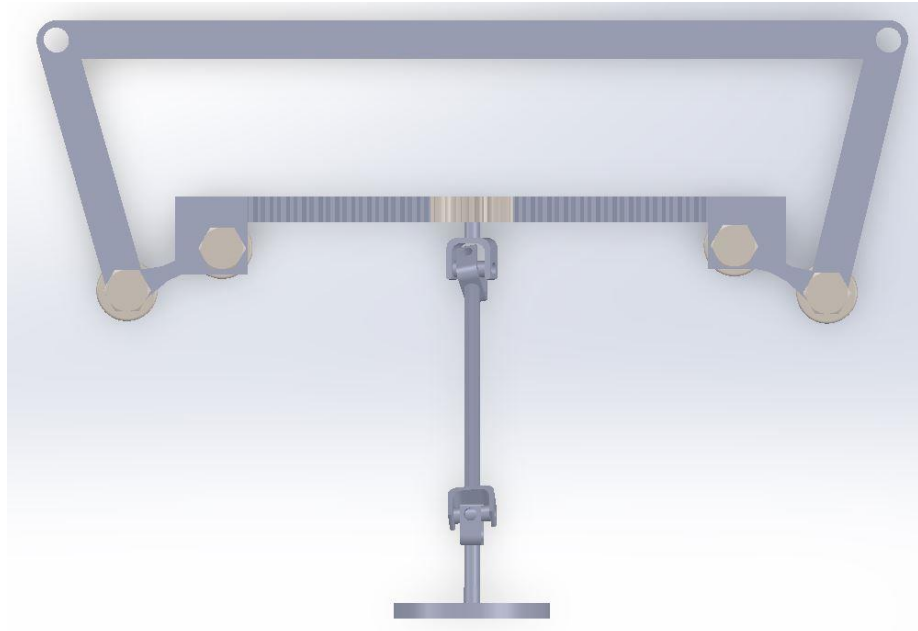
Gearbox ratio: 15:1

- Trapezoidal linkage for Ackerman
- Self-centering torque springs
- Straight-line stability considerations

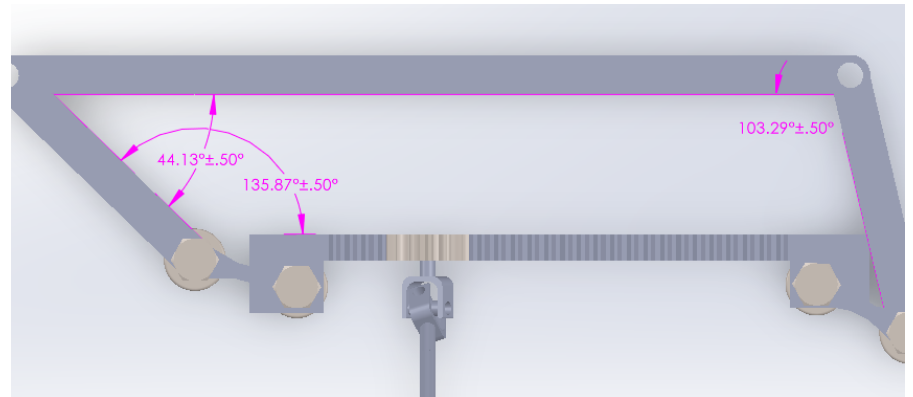
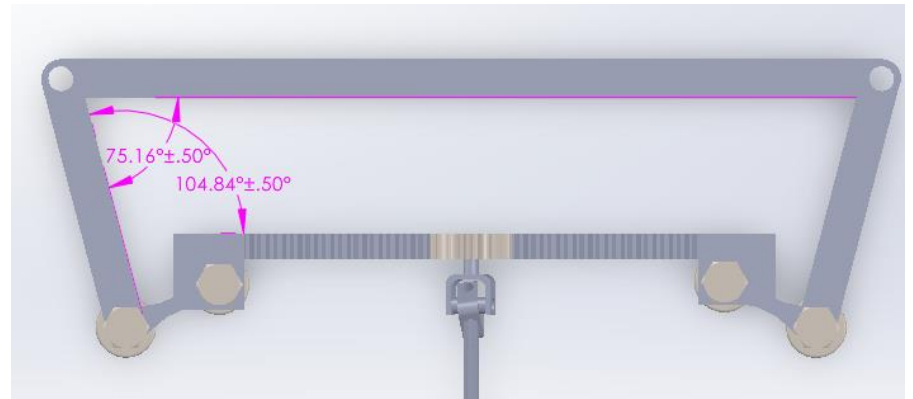


Self-Centering Torque Springs

- The Self Centering Torque Spring on the steering system aids in the movement of the wheels in part with the steering's trapezoid geometry.
- The force generated by the wheels and suspensions caster, slip and camber angles are returned to their initial state with the help of the center spring expanding or compressing when these angles are exercised.

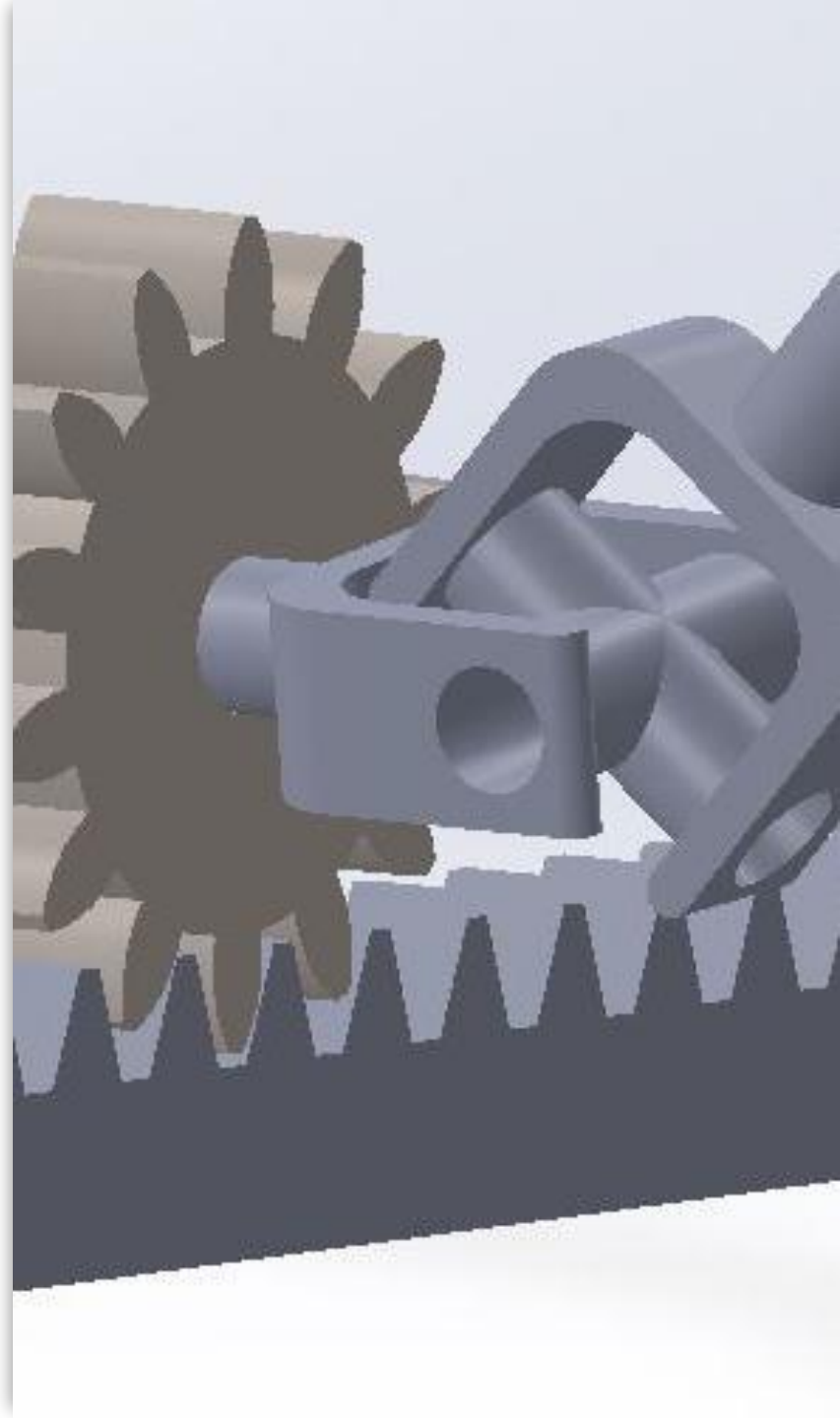


Angle At Neutral and Full Lock w/ Ackerman Angle



15:1 Ratio

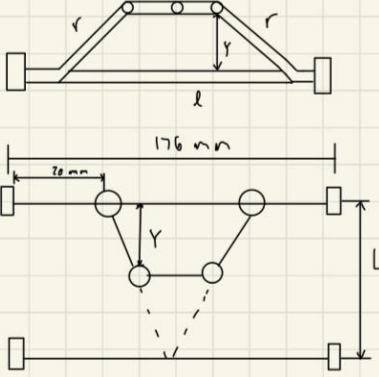
- Module of Pinion and Rack is 1.0
- Circular Pitch = $3.14 \times \text{Module} = 3.14$
- 2 rotation from lock-to- lock
- Linear distance from lock-to-lock is about 88mm
- $88\text{mm}/3.14 = \text{Teeth of Rack Used From Lock to Lock} : 28$
- Steer Arm Length = 50mm
- Approx Steer Angle = $\arcsin(22/50 \text{ mm}) = 26 \text{ deg.}$
- Because of Ackerman Angle we achieve a greated steer angle.



Steering Calculations

Steering calculations

(10 in $\rightarrow$ 254 mm)



$\theta = \tan^{-1} \left(\frac{L}{T/2} \right) = \tan^{-1} \left(\frac{370}{254/2} \right)$
 $\theta = 71.05^\circ$

$\cos(\theta) = \frac{T-l}{2r}$

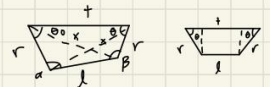
$T-l = 2r \cos(\theta)$
 $l = T - 2r \cos(\theta)$
 $= 254 - 2(53.51) \cos(71.05^\circ)$
 $= 292 \text{ mm}$

$Y = 50 \text{ mm}$
 $\sin \theta = \frac{Y}{r}$
 $r = \frac{Y}{\sin \theta}$
 $= \frac{50 \text{ mm}}{\sin(71.05^\circ)} = 53.51 \text{ mm}$

$176 - (25.6) = 150.4$

$\delta_0 = \frac{L}{R + \frac{T}{2}} \quad \delta_i = \frac{L}{R - \frac{T}{2}}$

$R = \frac{L}{\delta_0} - \frac{T}{2} \quad R = \frac{L}{\delta_i} + \frac{T}{2} = \frac{370}{\delta_0} = \frac{370}{\delta_i} + 176$



$\theta_o = \theta + \delta_o$
 $\theta_i = \theta - \delta_i$
 $\theta = \cos^{-1} \left(\frac{T-l}{2r} \right)$

$x^2 = r^2 + l^2 - 2rl \cos \alpha$
 $x^2 = T^2 + r^2 - 2Tl \cos \theta_i$
 $(176)^2 + l^2 - 2(176)(l) \cos \theta_i = T^2 + r^2 - 2rl \cos \alpha$
 $30976 - 352(l) \cos \theta_i - l^2 = -2rl \cos \alpha$
 $\alpha = \cos^{-1} \left[\frac{30976 - 352(l) \cos \theta_i - l^2}{-2rl} \right]$

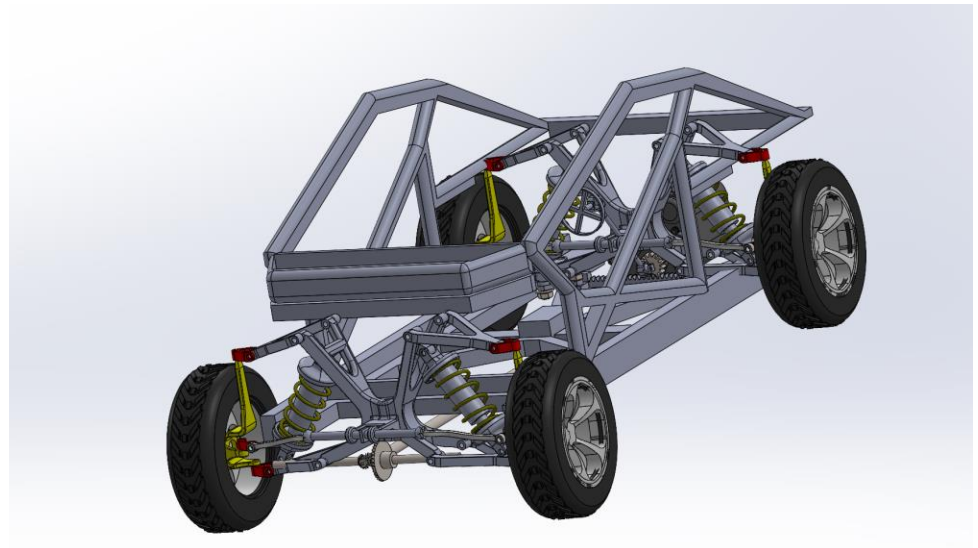
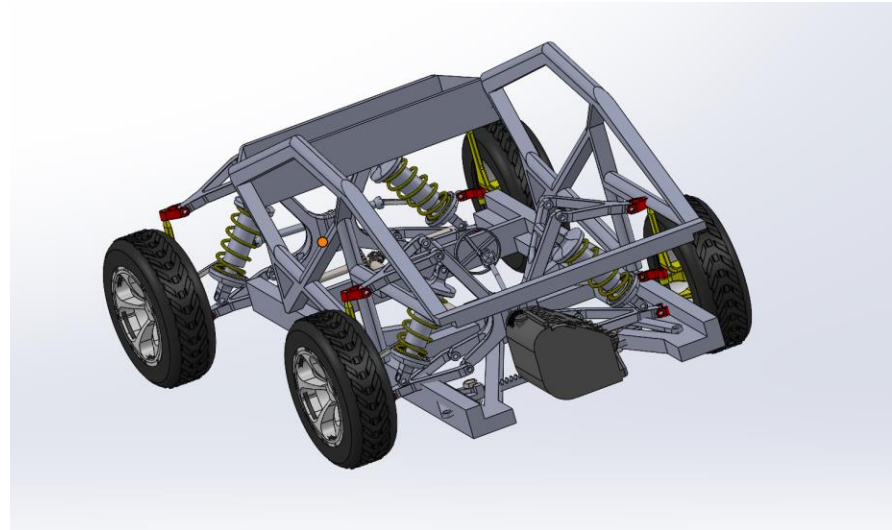
$\theta_o + \theta_i + \alpha + \beta = 2\pi$
 $\theta_o + \left[\cos^{-1} \left(\frac{30976 - 352(r) \cos \theta_o - l^2}{-2rl} \right) \right] = 2\pi - \theta_i - \left[\cos^{-1} \left(\frac{30976 - 352(l) \cos \theta_i - l^2}{-2rl} \right) \right]$
 $\left[\cos^{-1} \left(\frac{176 \cdot l}{2r} \right) \right] + \left[\cos^{-1} \left(\frac{30976 - 352(r) \cos(\cos^{-1}(\frac{176 \cdot l}{2r})) + \delta_o - l^2}{-2rl} \right) \right]$
 $= 2\pi - \left[\cos^{-1} \left(\frac{176 \cdot l}{2r} \right) - \delta_i \right] - \left[\cos^{-1} \left(\frac{30976 - 352(l) \cos(\cos^{-1}(\frac{176 \cdot l}{2r}) - \delta_i) - l^2}{-2rl} \right) \right]$

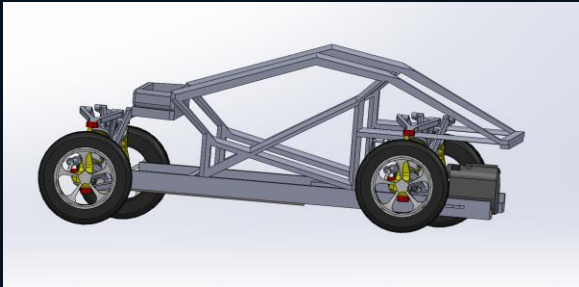
$y^2 = r^2 + l^2 - 2rl \cos \beta$
 $y^2 = r^2 + T^2 - 2rT \cos \theta_o$
 $r^2 + l^2 - 2rl \cos \beta = r^2 + (176)^2 - 2r(176) \cos \theta_o$
 $\alpha \cos \beta = 30976 - l^2 - 352(r) \cos \theta_o$
 $\beta = \cos^{-1} \left[\frac{30976 - 352(r) \cos \theta_o - l^2}{-2rl} \right]$

Assembly Structure

CAD assembly hierarchy

- Sub-assemblies: Chassis, Powertrain, Suspension, Steering
- BOM and part naming conventions





Additional Vehicle Assembly Visuals

- Isometric views and side view of the vehicle components
- Chassis, engine, transmission, differential, front and rear suspensions, as well as steering system shown

Conclusion

- The 1/10 scale model car platform was successfully designed to meet the key project requirements.
- A rear-wheel-drive system with a non-locking differential was created, using a 35-tooth ring gear and a 10-tooth pinion gear to achieve the target 3.5 final drive ratio.
- Bevel gear pitch angles were calculated and applied to ensure proper meshing and smooth torque transfer.
- The suspension systems, including a front SLA setup and rear Hotchkiss layout with 30–60% anti-squat, were designed to provide realistic vehicle dynamics.
- A 15:1 steering gearbox with Ackerman geometry and self-centering torque springs was implemented to enhance stability and maneuverability.
- The CAD models validate the design intent and are ready for future prototyping and testing phases.