SAE Mini Baja: Suspension and Steering

Project Proposal

Zane Cross, Kyle Egan, Nick Garry, Trevor Hochhaus

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Overview

- Problem Definition and Project Plan
- Concept Generation
- Design Selection and Analysis
- Bill of Materials
- Conclusion
Problem Definition and Project Plan
Introduction

Design, build, and test suspension and steering components to compete in Mini Baja SAE competition.

Client: Dr. John Tester

SAE Mini Baja Competition Environment:
- Portland, Oregon
- Rough, natural terrain and wet weather
Customer Needs

The previous car was too large, heavy, and lacked maneuverability.
Project Goals

- **Need:** Approach angle is too shallow reducing maneuverability
- **Need:** Current suspension mounts were an after thought for the frame increasing weight
- **Need:** Existing turning radius is too large limiting maneuverability
Project Goals cont.

- **Need:** Current car is too bulky in overall dimensions
- **Need:** There were too many bought components with no analytical justification
- **Need:** Most components are over engineered and too heavy
## Objectives

<table>
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<tr>
<th>Objective</th>
<th>Measurement Basis</th>
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<td>Increase Width</td>
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<tr>
<td>Increase Reliability</td>
<td>Repetition of suspension and steering components</td>
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Constraints

- Track width must be 59” or less
- 2 lane width turning radius
- All suspension mounts integrated into frame
- Entire Mini Baja must weigh less than 450 lbs
  - Suspension and Steering components need to weigh no more than 150 lbs
- Conform to SAE Baja competition rules
# QFD

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House of Quality

![House of Quality Diagram]

- Durability
- Weight
- Cost Sensitive
- Safety
- Manueverability
Gathering Information

- State-of-the-art Research
  - Previous NAU SAE Baja Projects
  - Previous SAE Baja Projects from other schools
  - Suspension Systems
  - Steering Systems

- Resources
  - SAE 2015 Rules and Regulations
  - Technical Suspension Books
  - Technical Steering Books
  - Previous SAE Baja Projects
# Project Planning

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Concept Generation
Front Suspension Concepts

- Double A Arms
- MacPherson
- Torsion Bars
- Extended A Arms
Double A Arms

● **Advantages:**
  ○ High strength
  ○ Highly adjustable
  ○ Good ground clearance

● **Disadvantages:**
  ○ Can be heavy
  ○ Can be difficult to analyze

www.lostjeeps.com
MacPherson

- **Advantages:**
  - Lighter weight
  - Less design and machining

- **Disadvantages:**
  - Higher stresses
  - Requires wheel hub modification

www.multibody.net
Torsion Bars

● **Advantages:**
  ○ Very high strength
  ○ Only one member
  ○ Large travel

● **Disadvantages:**
  ○ Less ground clearance
  ○ Heavier design

www.eurobricks.com
Extended A Arms

● **Advantages:**
  ○ More travel
  ○ More ground clearance

● **Disadvantages:**
  ○ Heavier
  ○ Less impact resistance
# Front Suspension Decision Matrix

<table>
<thead>
<tr>
<th>Front Suspension</th>
<th>Weight</th>
<th>McPherson</th>
<th>Double A-arms</th>
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</table>

Trevor Hochhaus
Rear Suspension Concepts

- Double A Arms
- 2 link
- 3 link
Double A Arms

- **Advantages:**
  - Easy to analyze design
  - High strength
  - Adequate ground clearance
  - Proven design

- **Disadvantages:**
  - Difficult to machine
  - Space constraint (shock and driveshaft)
2 Link

- **Advantages:**
  - Light weight
  - High strength
  - Low cost

- **Disadvantages:**
  - Difficult to design
  - Ground clearance
3 Link

- **Advantages:**
  - High strength
  - Durable

- **Disadvantages:**
  - Difficult to analyze
  - High weight
  - High cost
Rear Suspension Decision Matrix

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Steering Concepts

- Back mounted
- Front mounted
- Power assist
Back Mounted

- **Advantages:**
  - Less likely to break on impact
  - More footwell room

- **Disadvantages:**
  - Less room for driver’s legs
  - Possible for u-joint to bind
Front Mounted

- **Advantages:**
  - More room for driver’s legs
  - Easier to adjust

- **Disadvantages:**
  - More weight
  - More likely to break on impact
Power Assist

- **Advantages:**
  - Easier for driver
  - Adjustable

- **Disadvantages:**
  - Much heavier compared to non power assist
  - Uses much needed engine power
# Steering Decision Matrix

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Trevor Hochhaus
Design Selection and Analysis
Final Assembly

Front Isometric View

Back Isometric View
Suspension Analysis-Final Design

A-Shaped Members:
● Designed for weight reduction
● Simplistic

Pros:
● Strong against front impact
● Easy to manufacture
● Lightweight

Cons:
● Weaker in loading
Hand Calculations

Bending stress in A-arms
- Analysed half of one A-arm
- Hinge joint connecting to frame, force from vehicle weight, force of shock

Force from vehicle weight: 200lbs
Moment of hinge joint
Force of shock
Hand Calculations Cont.

Moment around hinge:
Force of shock= 325.83lbf

Sum of forces in Y direction:
Force of hinge Y dir=51.85lbf

Sum of forces in X direction:
Force of hinge X dir=124.66lbf

Half forces due to symmetric geometry:

Force of shock= 162.92lbf
Force of hinge Y dir= 25.93lbf
Force of hinge X dir= 62.33lbf

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Hand Calculations Cont.

Max Moment = 337.04 lb-in
Hand Calculations Cont.

\[ \sigma = \frac{Mc}{I} \]

\[ I = \frac{\pi}{64} (D^4 - d^4) \]

\[ c = \frac{D}{2} \]

M = 337.04lb-in
\[ \sigma = 36\text{ksi for Structural A36} \]
d=0.8D

Output:
\[ D = .78\text{in}, t=.156\text{in} \]

FOS of 2:
\[ D = .98\text{in}, t=.20\text{in} \]
Shock Placement

- After performing dynamic analysis on the suspension design, the forces experienced by the members can be greatly mitigated by mounting the shock as close as possible to the wheel hub.
- There are physical limitations on how close to the hub the shocks can get, such as: extended length of the shocks, desired ride height, and potential interference.
Stress Analysis

- Axial loading of the members is going to be ignored, because stresses caused by bending will far exceed them.
- The member the shock mounts to will experience the largest force in the system, so it will fail first.
- The bolts for the members will also be a mode of failure when the suspension receives a front or side impact. The shear stress on the bolts will be calculated using the axial loading on the members.
Stress Analysis- Front Impact

Situation:

- 25 mph impact on one tire
- Simulates the car suddenly hitting an object, like a tree or rock, and coming to a full stop
- The chosen material is 4130 steel with a OD of 1.25in and an ID of 1.15in
- The load was applied to the end of one A arm, with the arm fixed to where it would mount to the chassis
• The lowest factor of safety for this loading 2.1
• The force applied in this loading is an extreme case
Stress Analysis-Vertical Loading

Situation:

- The car jumps off a surface from 3 ft and lands crooked on only one wheel
- Same material and dimensions as the previous design scenario
- The load is applied vertically to the end of the A arm as well as axially
- The A arm is pinned at the shock mount and the chassis mount
The minimum factor of safety for this situation is 3.2
This loading is an extreme case due to the suspension bottoming out before it reaches stress levels this high
Steering Design Selection

- **Back Mounted Steering**
  - Minor modification to existing hub
  - Front of frame constraints
  - Room for tie rod mounting to hub (Brake Caliper)
Steering Analysis - Akerman Angles

- Ackerman Steering Angles
  - Inside wheel turns at greater angle than outside wheel
  - Determine max angle of both tires such that a U-turn can be achieved within a width of 2 lanes (144 in)

### Inside Tire Maximum Angle

\[
\tan(\delta_i) = \frac{L}{R_1 - \frac{W}{2}}
\]

### Outside Tire Maximum Angle

\[
\tan(\delta_o) = \frac{L}{R_1 - \frac{W}{2}}
\]
Ackerman Angles Calculations

- Track Width = $W = 49$ in
- Wheelbase = $L = 65$ in
- Mid-Radius = $R(1) = 155.5$ in

After inputting variables into formula and solving……

Max Turning Angles

- Inside Tire = 35.54 Degrees
- Outside Tire = 24.90 Degrees

Turning Radius

- 9.63 ft
Steering Analysis - Tie Rod Mount

- Determine where the tie rods are mounted to the hub of the vehicle to achieve an Akerman Angle with zero toe on turn in

Zane Cross
Tie Rod Mount Calculations

- Through the use of similar triangles:

\[
\frac{24.5}{65} = \frac{y}{3.45}
\]

\[y = 1.43 \text{ in}\]
Steering Analysis - Steering Ratios

- Determine the ratio of the rack to pinion to give the right amount of assist in the steering system
- 1:2 Steering Quickener currently installed
- Remove steering quickener to regain 12:1 steering ratio of original steering rack
Steering Analysis - Tie Rod Force

- Determine axial force tie rod encounters when hitting an obstacle and coming to a complete stop

**Situation**
- Velocity of Vehicle is 20 mph
- After hitting obstacle, vehicle comes to a complete stop in 0.5 sec

**Solution**
- \( F_{Tie\ Rod} = 1,269.86 \text{ lbf} \)

\[
F_{Rock} = M_V \times A
\]

\[
M_{Kingpin} = F_{Rock} \times d_1
\]

\[
F_{Tie\ Rod} = \frac{M_{Kingpin}}{d_2}
\]
Steering Analysis - Tie Rod Buckling

\[ F_{Tie\ Rod} = \frac{\pi^2 EI}{(KL)^2} \]

\[ I_x = \frac{\pi}{64} D^4 \]

- \[ F_{Tie\ Rod} = 1,269.86 \text{ lbf} \]
- \[ E = (29.0 \times 10^6) \text{ psi (A36 Steel)} \]
- \[ K = 1.0 \text{ (Pinned Support at both ends)} \]
- \[ L = 15 \text{ in} \]

After solving for \(I\), then solving for diameter.....

**Tie Rod Dimension**

- Safety Factor of 2
- \[ D = .4491 \text{ in} \]
Hollow vs. Solid Tie Rod

- **Hollow**
  - Outer Diameter = .534 in
  - Wall Thickness = .159 in
  - Weight = .4845 lb

- **Solid**
  - Outer Diameter = .500 in
  - Weight = .8364 lb

- **Weight Decrease**
  - Old Tie Rod = 2 lb
  - New Tie Rod = .8364 lb
  - **58 percent decrease in weight**
Bolt Shear Analysis

- Double Shear Situation
- Assume Grade 8 Bolts
- Max Shear Force on Bolt = 5,000 lbf
- After comparing different size bolts……

- Bolt Size Diameter = 5/16 in
- Bolt Dimensions = ¼ - 20
- Safety Factor = 3.68

\[ \tau = \frac{F}{2A} \]
## Bill of Materials

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Conclusion

- Problem Definition and Project Plan
- Concept Generation
- Design Selection and Analysis
- Bill of Materials
References

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