INTRODUCTION

The MiniBaja SAE program is an intercollegiate competition focused on designing and building an off-road vehicle that will be submitted to the extremely rough terrain. Such vehicle is intended to be designed by the engineering students of each university with the main purpose of providing the students a real world engineering designing experience. The purpose of this design project is to optimize the suspension design used on the MiniBaja vehicle which has already been developed by the students of SAE in the University of Texas at El Paso.

Within the next pages of this report we will explain new ideas in the optimization of the suspension used on the already existing vehicle. The analysis of forces and stresses acting on the suspension will also be examined and discussed, and if possible a solution will be provided.

Hard work has been put into this design where the outcome has been significant improved design over the previous suspension.
• DEFINITION

Suspension is the term given to the system of shock absorbers and linkages that connects a vehicle to its wheels. The job of a suspension system is to contribute to the car’s handling and braking for a better safety driving, and to keep the driver as isolated possible from bumps, vibrations, etc. It is important for the suspension to keep the wheel in contact with the road surface as much as possible, since all the forces acting on the vehicle do so through the tires. The suspension also has the important task to protect the vehicle itself and any cargo from damage. In this report the design of the front suspension for the miniBaja vehicle will be improved.

• PROBLEM IDENTIFICATION

Analyzing the current suspension there are several points that could be improved in order to optimize the functionality of the suspension. The first issue with the current suspension being used on the UTEP MiniBaja vehicle is that there is no recorded analysis for stresses nor forces; it was fabricated without previous examination nor calculations of the components used. The lack of study of these components results in the failure to provide the required information to whether the components will fail under the circumstances to which the suspension will be submitted. That means there is no reference point to know at what point they could fail. It may be noted too that the material used in the current suspension is unknown.

The two A arms that constitute the suspension are another point to discuss. Even when this design is frequently used in different types of suspension the use of space and appearance could be improved. The upper A arm is not adjustable therefore the vehicle cannot take advantage of the different tracks that it might experience.

• SOLUTION IDEAS

After taking into account all the missing data and realizing the points that could be improved, we started the optimization of the components of the suspension. We analyzed each component separately with their respective forces and stresses and calculated the required information to perform a stress analysis to see if they would fail and if so at which point.
Another optimization point that we worked on was the design of the two A arms. Even when the current design of the arms is good and broadly used in suspensions, we realized that it could be improved by changing the design of the upper and lower arms giving a better use of the space and making the performance more efficient. The upper arm was altered to be a single adjustable rod to take advantage of the different scenarios of tracks. This upper arm will also work as a support for the lower arm which will be a machined plate that will be supporting the shock absorber. This lower arm was adjusted to support more strength since we will take material from the upper arm and also to accommodate the pieces better.

As mentioned before the material used in the current suspension is unknown. In order to fix this issue we researched for different materials frequently used in suspension and looked for the one that will meet our criteria. The material that will be used for our design is AISI steel 4130 for which the properties are known as well as the methods to use it. The reasons to choose this material will be explained more broadly in the “Design Introduction” section.

- **DESIGN TOOLS**

Various tools were used in the development of our design. The NX6 software is used for tasks such as the three dimensional solid modeling of each component. The use of solid modeling techniques allows us the computerization of many difficult engineering calculations that are carried out as a part of the design process. Simulation, planning, and verification of processes such as machining and assembly of the components of the suspension are some of the tasks that will be performed in NX6. Also this software will be used to perform an engineering analysis of the stresses acting on each component to reveal the state of stress and failure on the device.

The use of NX6 is very important, but the use of other tools is also necessary. These will be the knowledge acquired in engineering mechanisms and design from our current and prior classes such as mechanical design, mechanics of material, dynamics, etc.

2. **DESIGN CRITERIA**

- Weight and loads:
Weight is one of the main constrain that engineers must consider in mechanical design of any component. First of all, when you design the suspension components of a minibaja, all possible load scenarios must be taken into account. This is highly important since the sum of the forces of each different scenario will be distributed not evenly. After that, depending in the geometry of the components, one or several spots will experience a max stress. Loads can be presented considering two types. One is the static load where the weight has that important roll, and the second one is the dynamic load that will be the one exerted by several race circuit conditions. The weight of the components will be added up to find the resultant static load, and the calculations will be focus at the critical points in the material. We will start the calculations with the static load followed by the dynamic load in order to find these critical spots.

- **Average weight:**

<table>
<thead>
<tr>
<th>components</th>
<th>range</th>
<th>Average (actual)</th>
<th>calculated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frame</td>
<td>(50-80)</td>
<td>65</td>
<td>97</td>
</tr>
<tr>
<td>Driver</td>
<td>(75-115)</td>
<td>95</td>
<td>100</td>
</tr>
<tr>
<td>Engine and Transmition</td>
<td>(40-65)</td>
<td>52.5</td>
<td>75</td>
</tr>
<tr>
<td>Suspension and Accessories</td>
<td>(40-65)</td>
<td>52.5</td>
<td>90</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td>265kg</td>
<td>362kg</td>
</tr>
</tbody>
</table>

- **Materials**

Materials selection is another important point to cover. There is a huge range of different properties that can be achieved with the use of engineering materials, from composites to many different alloys. We focused in three major goals when we try to find the best material that meet our needs, such goals are: the material deformation with several impacts, the cost of the material, and manufacturability. In the second place but not less important, we consider
the weight of components and the operating temperature since this project is design to operate at a temperature of 23.9 degrees Celsius. The material which best match these requisites was the alloy steel 4130 which is also known as chromoly. Steel 4130 properties are explained in detail in the following paragraphs.

- Properties

![Stress-strain diagram of a medium-carbon structural steel](image)

<table>
<thead>
<tr>
<th>Properties</th>
<th>T (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density ((\times 1000 \text{ kg/m}^3))</td>
<td>7.7-8.03 25</td>
</tr>
<tr>
<td>Poisson's Ratio</td>
<td>0.27-0.30 25</td>
</tr>
<tr>
<td>Elastic Modulus (GPa)</td>
<td>190-210 25</td>
</tr>
<tr>
<td>Tensile Strength (Mpa)</td>
<td>560.5 25</td>
</tr>
<tr>
<td>Yield Strength (Mpa)</td>
<td>360.6 25</td>
</tr>
<tr>
<td>Elongation (%)</td>
<td>28.2 25</td>
</tr>
<tr>
<td>Reduction in Area (%)</td>
<td>55.6 25</td>
</tr>
<tr>
<td>Hardness (HB)</td>
<td>156 25</td>
</tr>
<tr>
<td>Impact Strength (J) ((Izod))</td>
<td>61.7 25</td>
</tr>
</tbody>
</table>
- **Properties chart:**

<table>
<thead>
<tr>
<th>Materials</th>
<th>Modulus of elasticity (GPa)</th>
<th>Yield strength (MPa)</th>
<th>Tensile Strength (MPa)</th>
<th>Percent Elongation</th>
<th>Density (g/cm³E3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alloy ti-6AL-4V</td>
<td>114</td>
<td>830</td>
<td>900</td>
<td>14</td>
<td>4.43</td>
</tr>
<tr>
<td>Tungsten (commercial)</td>
<td>400</td>
<td>760</td>
<td>960</td>
<td>2</td>
<td>19.3</td>
</tr>
<tr>
<td>Steel alloy A36</td>
<td>207</td>
<td>220-250</td>
<td>400-500</td>
<td>23</td>
<td>7.85</td>
</tr>
<tr>
<td>AISI 4130 CHROMOLY</td>
<td>210</td>
<td>360.6</td>
<td>560.6</td>
<td>28.2</td>
<td>8</td>
</tr>
<tr>
<td>Stainless alloy 304</td>
<td>193</td>
<td>205</td>
<td>515</td>
<td>40</td>
<td>8</td>
</tr>
</tbody>
</table>

(TABLE 2-2)

- AISI 4130 chromoly has Average properties that match a wide range of operating scenarios, unlike it fails in extreme loaded conditions, those extraordinary cases were calculated assuming top speed impacts, neglecting any driver maneuvers and without applying braking.
- Since the elongation range is considerably wide, we have found out that chromoly can be an excellent option. A big range in the plastic region will give us a warning when the components must be replaced, and it will also help us to avoid a sudden fracture. Also, the yield strength is high enough to withstand several scenarios with considerable load range without being deformed.
LOADS

- Horizontal impact with a magnitude of 80325 N

- Static load

  Vertical load due to the weight of the vehicle with a magnitude of 1252 N
3. DESIGN INTRODUCTION

- OVERLOOK

The suspension designed for the MiniBaja cart consists of two single control arms that are attached to the upright and the hub that is mounted on the spindle, as shown in Figure 3.1. The frame of the existing cart will have to be modified in order to hold the top arm, the bottom arm, and the shock absorber. The following paragraphs will go more in detail in the description of each individual part.

Figure 3.1: Front view of the complete assembled suspension.

Load due to falling

Highest suspension load generated due to a fall of .5 Meters which has a magnitude of 11,014.4 N
PARTS

The top arm will be connected by a ball joint to the upright, and will be pinned to the frame, and was designed to have an adjustable length. The top arm will consist of one solid shaft that will have a nut fixed in the center and it will be threaded on both sides. The solid shaft will fit into the two shafts that will have a hole at the ends. Both shafts will have an inner thread and the ends will be secured by two additional nuts, one on each side. The top arm was designed adjustable in length to take advantage of the track and the environment in which the cart will be placed. For example, if the track only requires the cart to take right turns the length of the arm can be adjusted to change the angle of the tire which will minimize over steer. The range of the length of the upper arm can be adjusted between the lengths of 420 millimeters and 480 millimeters. The ball joint that will have to be placed at one end of the arm will have to be placed into the arm by force. The following image shows the finished arm.

Figure 3.2

The bottom control arm is considered a reinforced single arm. The bottom arm, like the top arm, will be pinned to the frame and will be attached to the upright with a ball joint. The bottom arm will also hold the shock absorber in place at approximately two thirds of the whole length away from the frame. The bottom arm will have two connections that were designed so that the shock will not have touch the top face of the arm. The dimensions of the shock that we used will be given further on. If a shock with different dimensions is
used, then the extrusions should be modified accordingly. In order to reduce the weight of the bottom arm, material will be removed between the connection at the pin and the shock absorber connection. In order to reduce the stress concentration we added two solid shafts that form an “X”. The lower arm also contains a reduction in cross-sectional area. This was done so that the bottom arm does not limit the tire’s turning angle. The ball joint that will connect the arm to the upright will be put in by force, and the appropriate dimensions for the lower control arm will be given further on.

Figure 3.3

The upright will hold the upper control arm, the lower control arm, the spindle, the tie rod and the brake caliper. The control arm connections will be aligned vertically with the spindle. The hole in which both the upper arm ball joint and the bottom arm ball joint was given a diameter of 11 millimeters which will give a clearance of 1 millimeter. The hole in which the spindle will be placed will be approximately in the center of the spindle. The hole will perforate the upright. The upright has a small extrude of a thickness of 5 millimeters. This will not allow the hub to come into contact with the upright and cause friction. In order to avoid a direct impact to the tie rod, the upright was designed to hold the tie rod between the control arms and at the position farthest away from the front of the cart. Additional extrudes were placed on the upright toward the front of the cart in order to mount the brake caliper.
The spindle will be placed by force into the upright. The spindle has three different diameters. The largest diameter will be inserted into the upright. Then the following diameters hold the hub in place. There is a gradual decrease of diameter in the two smaller diameters in order to avoid the concentration of stress in the shoulder. The spindle will also have a thread at the smallest diameter with a length of approximately 25 millimeters. This will allow the tire rim to also be placed here and reinforced with a nut.

The hub will be mounted on the spindle and will also hold the disk. The main shape of the hub consists of a large “X” which will hold the tire rim in place. The hub also has an area in which the disk will be mounted. The hub was also designed with holes which allow the disk to be tightly secured. The inside of the hub will also include the gradual decrease of diameter between the two smaller diameters in order to match the design of the spindle.
Figure 3.5

The disk that will be used in this suspension will be mounted on the hub in order to reduce the length of the spindle. The disk will be close to the upright so that the brake caliper can be mounted on the upright and perform its action.

Figure 3.6

- **MATERIAL**
  
The material in which the parts where analyzed was using AISI 4130. Its ultimate tensile strength was 560.5 MPa and its yield strength is 360 MPa. After some research we came across AISI 4330 which has the same yield strength as the material before. However it has a higher ultimate strength than the one stated above. We suggest using AISI 4330 in the bottom arm which will be subjected to a higher stress under normal conditions.
• **PROS**

In comparison with the original design this new design has many pros and cons. One of the advantages that our new design has is that the whole design has calculations behind it. Another advantage is that our design has an adjustable upper arm that will allow the driver to take advantage of the track and the environment in which the cart will be placed. Also our design has a completely different lower control arm. The lower control arm is a single arm that has a larger cross-sectional area that will reduce the normal stress of the part. Also when taking into account the spindle, we added a chamfer in order to reduce the stress that a shoulder will cause between the hub and spindle. Another advantage to this design is that we placed the tie rod in the back of the assembly. The reason for this is to protect the tie rod from a frontal impact. However there are some details that will make our design less ideal.

• **CONS**

One of the disadvantages to our design is that the lower arm compared to the current design will weight a significantly more. Also another disadvantage to this design is that if the upper arm were to be modified to a smaller diameter, it may fail even in static load. Also another disadvantage to our design is that all of the parts must be machined.

4. **3D MODEL DRAWING**

The 3D modeling was created using the software NX6 which has been a very helpful and handy tool in the design. Within the next pages we can appreciate a series of pictures where every component of the suspension is described. Also there are a couple is images where every single component has been assembled in order to appreciate the final look of this suspension prototype.
Frontal left side view

Down side view
- Upside view

- True shading
The following pictures are drafts where every component of the suspension shows its physical dimensions.

- Lower control arm and dimensions
• Upper arm and dimensions
Upper arm nut (holds the uparn to the upright)
- Disk and dimensions
Hub and dimensions
- Shock screw and nut and dimensions
5. ANALYSIS

Our mass of the car + person will be converted into a force and then divided into four parts, one for each suspension (two frontal, two in the back). This was calculated to be 887.8 N, from the maximum considerations in the mini-baja rules.

The weight \( W = 887.8 \) N was then magnified for each part of the suspension, due to the fact that its equivalent moment needed to be taken into account. It’s assumed that the center of mass of the cart is at the middle, .634 m away from the tire. These are the calculations which show the weight equivalence for each bar:

\[
F_i d_i = F_f d_f
\]

Top arm: \( 887.8 \times .634 = W_{\text{top}}(.4 \cos 30), W_{\text{top}} = 1624.85 \) N

Bottom arm: \( 887.8 \times .634 = W_{\text{bottom}}(.3897), W_{\text{bottom}} = 1444.36 \) N

These equivalent forces are according to the moment equivalence, due to the fact that the top arm is shorter, and farther away from the center of mass. Since both arms are parallel, but at an angle (30) according to the applied force, we need to establish a new set of axes, which will show us the normal and bending forces applied, according to each situation. \( x' \) will represent the axial forces, parallel to the bar; \( y' \) will characterize those that are perpendicular.

In the picture, the parallel axes are shown along with the direction of the applied force. The normal \( x-y \) plane is shown in the upper right corner. Once these have been established, one realizes that the applied force can be broken up into \( x' \) and \( y' \) components. The formula for these will be as such:

\[
F_{x'} = F\cos 60
\]
After these particularities have been exposed, different cases were ready to be analyzed. In the following scenarios, a rough sketch of setup is provided.

First Case: Static

For this case, a simple static analysis will be made. A maximum weight \( W \) will be established for each, and its components will be declared for both upper and lower arms. Having established that the weight for the upper bar will be 1625 N, and the bottom weight will correspond to 1445 N, the forces were broken up as such:

*Upper Bar*

This bar shouldn’t, in theory, absorb the whole impact of the weight or any other forces if the shock absorber fulfills its duty. If it doesn’t, however, it will be assumed that 25% of the weight will be maintained for the bar (this will be for all the cases). The reduction of the weight has already been applied in the resultants.

Having a schematic as such, and realizing that the \( N-x' \) and \( N-y' \) forces are calculated with the equations presented above for the conversion to the parallel axis, the forces found in this case are:

\[ N-y' = R-y' = 351.25 \text{ N} \]
\[ N-x' = R-x' = 203.25 \text{ N} \]
One must also realize that the summation of forces must overall equal to zero, and such the
net forces will be nonexistent. The shear and moment diagrams will be presented next:

With these, one can realize that the maximum moment will exist at the edge of the bar. The
moment calculated using $F \cdot d$ equals 140.7 N*m. (The length of the bar is .4m). This
moment will later be used in the calculation of bending stress.

Checking for yielding

$A = 0.00196$

$C = 0.0125$

$$I = \frac{\pi}{4} r^4 = \frac{\pi}{4}(0.0125)^4 = 1.917e-8 m^4$$

$$\sigma = \frac{Mc}{I} = \frac{(281.4 Nm)(0.0125 m)}{(1.917e-8)m^4} = 1.83e8 Pa$$

$$\sigma_{Axial} = \frac{F}{A} = \frac{406.25}{0.00196} = 207270 Pa$$

$$\sigma_x = \sigma - \sigma_A = (1.83e8 Pa) - (207270 Pa) = 1.828e8 Pa$$

$$J = \frac{\pi}{2} r^4 = \frac{\pi}{2}(0.0125)^4 = 3.83e-8 m^4$$
\[
\tau = \frac{(1184Nm)(.0125m)}{(3.83e-8)m^4} = 3.86e8Pa
\]

The Static Case I is a very extreme case in which the upper arm will receive the 50% of the force. Normally the upper arm will sustain approximately 25% of the load.

Case I (Static) at 25\%

\[
\begin{align*}
\sigma &= 9.15e7Pa \\
\sigma_{Axial} &= 103635Pa \\
\tau &= 1.93e8Pa
\end{align*}
\]

\[
\sigma_x = \sigma - \sigma_A = 9.13e7Pa
\]

NX6 analysis:

Structural performance

Displacements
*Lower Bar

Since the shock absorber will not fulfill its duty when the suspension is at a static position, the force diagram for the lower bar will be equal to the one in the upper bar. For all cases, it will be assumed that the impact is absorbed 100% by this component.

The value of the calculated forces is as such:

\[ N-y' = R-y' = 1252 \text{ N} \]

\[ N-x' = R-x' = 722.5 \text{ N} \]

The shear and moment diagrams are, then, the same as the last case.
The calculation of the maximum moment is obtained by simply multiplying the N-y' force by the .45 m (length of the bar). This value is equal to 563.4 N*m.

Checking for yielding

\[ C = 0.02 \text{m} \]
\[ A = 0.00216 \text{m} \]

\[ I = \frac{1}{12}bh^3 = \frac{1}{12}(0.054)(0.04)^3 = 2.88e^{-7} \text{m}^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(563.4 \text{Nm})(0.02 \text{m})}{(2.88e^{-7}) \text{m}^4} = 3.91e7 \text{Pa} \]

\[ \sigma_{Axial} = \frac{F}{A} = \frac{722.5}{0.00216} = 334491 \text{Pa} \]

\[ \tau = \frac{4.81(T)}{V} = \frac{4.81(1183 \text{Nm})}{(0.004648)^3} = 5.67e7 \text{Pa} \]

*Note: The shear stress will be that same in each case scenario since there is no Polar moment of Inertia, and just one Torque is acting on the element. The formula used to obtain the Shear stress was attained from a book for Mechanics of Materials, that formula was intended for a square cross-sectional area therefore we used our dimension to obtain an area and from there we obtained an equivalent dimension.
NX6 analysis

Structural performance

Displacement
Stresses

**Spindle**

Check for yielding

\[
F = 1252 N
\]

\[
I = \frac{\pi}{64} d^4 = \frac{\pi}{64} (0.025)^4 = 1.917 e - 8 m^4
\]

\[
\sigma = \frac{Mc}{I} = \frac{(1252N)(0.06 - m)(0.0125m)}{(1.917e - 8)m^4} = 2.5e7 Pa = 25 MPa
\]

\[
n = \frac{S_y}{\sigma} = \frac{360.6 MPa}{25 MPa} = 14.4
\]

NX6 analysis

**STRUCTURAL PERFORMANCE**
Upright

\[ F = 1252N \times 0.70 = 876.4N \]

\[ I = \frac{1}{2}bh^3 = \frac{1}{2} \times (0.025)(0.02)^3 = 1.66e - 8m^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(876.4N)(0.0175m)(0.01m)}{(1.66e - 8)m^4} = 9.24MPa \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6MPa}{9.24MPa} = 38.96 \approx 39 \]

NX6 analysis

**Structural performance**

![Structural performance diagram](image)

**Displacements**

![Displacements diagram](image)
Second Case: Horizontal Crash

The second case presents us with an impact in the leftover axis, which would correspond to $z'$. (Frontal impact to the car). In this case, the force was calculated by generating an impact force, described by

$$F_{impact} = \frac{\Delta mv}{\Delta t}$$

Since the average velocity of the car is calculated to be 40 mph, this would correspond to 17.85 m/s; the time assumed for the impact is .04 seconds, derived from different examples present in physics books. The final velocity is 0m/s. If the mass is calculated to be 90 kg (a fourth of the combined masses), the impact force is calculated to be 80325 N. Since this new force is presented in the $z'$ axis, the remaining $x'$ and $y'$ forces will continue to have the same values. The pictures present here are assumed to be of the $x'$-$z'$ plane.

*Upper bar*
As stated, the total force will only be applied by 25% of its magnitude. The schematic will remain as such:

It is to be noted however that the $y'$ axis forces are replaced by $z'$ axis ones (frontal impact). The values for these are as such:

$R_{x'} = N_{x'} = 203.25 \text{ N}$

$R_{z'} = N_{z'} = 10040.8 \text{ N}$

The shear and moment diagrams are omitted as their inclusion will be redundant; one can appreciate that for the upper bar, the maximum moment will be generated at the edge. Once again, multiplying the bending force times the length of the bar (.4m), the resultant moment will be: 4015 N*m.

- Check for yielding

$A =$ cross-sectional area

$C =$ distance from neutral axis to edge

$A = .00196$

$C = .0125$

Case II at 25% $\left\{ \begin{align*}
\sigma_x &= 9.13e7 \text{ Pa} \\
\tau_{xy} &= 1.93e8 \text{ Pa}
\end{align*} \right.$

$$\sigma_z = \frac{Mc}{I} = \frac{(16065.2N \text{m})(.0125m)}{(4)(1.917e-8) m^4} = 2.62e9 \text{ Pa}$$

We now substitute our values in the von Mises stress equation

$$\sigma' = \frac{1}{\sqrt{2}} \left[ (\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2) \right]^{1/2}$$

We get $\sigma' = 2.6e9 \text{ Pa}$

Now we substitute this value in the equation: $n = \frac{S_y}{\sigma'} = \frac{360.6e6 \text{ Pa}}{2.6e9 \text{ Pa}} = 0.14$
*Note: For this given scenario the stress will be greater than the Ultimate strength of the material, therefore it will fail.

**NX6 analysis**

**Answer quality**

**Displacement**
Stresses

*Lower bar

In the lower bar, the shock absorber has not been included in the forces yet, due to its parallel configuration in this setup. The force schematic will be repeated.

\[
N - x' = R - x' = 722.5 \text{ N}
\]

\[
N - z' = R - z' = 40163 \text{ N}
\]

The shear and moment diagrams will have the same schematic as before, for which the multiplication of the bending force times the length of the bar (of .45 m), the maximum moment calculated is to be 18073.15 N*m.

- Check for yielding

For Case II \[
\begin{aligned}
\sigma_x &= 3.88e7 \text{ Pa} \\
\tau_{xy} &= 5.67e7 \text{ Pa}
\end{aligned}
\]
\[ \sigma_z = \frac{Mc}{I} = \frac{(36146.3Nm)(0.02m)}{(2)(2.88e-7)m^4} = 1.26e9 \text{ Pa} \]

We now substitute our values in the von Mises stress equation

\[ \sigma' = \frac{1}{\sqrt{2}} \left[ (\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{1/2} \]

We get \( \sigma' = 1.24e9 \text{ Pa} \)

Now we substitute this value in the equation: 

\[ n = \frac{S_y}{\sigma'} = \frac{360.6e6 \text{ Pa}}{1.24e9 \text{ Pa}} = 0.289 \]

Lower arm structural performance

Displacement
Stresses

AISI 4130
\[ \begin{align*}
S_{ut} &= 560.6 \text{ MPa} \\
S_{y} &= 360.6 \text{ MPa}
\end{align*} \]

Spindle

\( F = 80325 \text{ N} \)

\[ I = \frac{\pi}{64} d^4 = \frac{\pi}{64} (0.25)^4 = 1.917 \times 10^{-8} \text{ m}^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(80325N) \left( \frac{0.06}{2} \text{ m} \right) (0.0125 \text{ m})}{(1.917 \times 10^{-8} \text{ m}^4)} = 1.56 \times 10^9 \]

\[ n = \frac{S_y}{\sigma} = \frac{(360.6 \times 10^6) \text{ Pa}}{(1.56 \times 10^9) \text{ Pa}} = 0.22 \]

*Note: spindle in this case scenario will fracture.*

NX6 analysis

Structural performance
Displacements
Stresses

F=80325N

\[ I = \frac{1}{2}bh^3 = \frac{1}{2}(.025)(.02)^3 = 1.66e - 8m^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(80325N)(.0175m)(.01m)}{(1.66e - 8)m^4} = 4.84e10 \]

\[ n = \frac{S_y}{\sigma} = \frac{(360.6e6)Pa}{(4.84e10)Pa} = 0.0075 \]

*Note: upright in this case scenario will fracture.

- NX6 analysis
- Upright Structural performance
Displacements
Stresses

Third Case: Falling

For this case a basic use of Newtonian physics had to be properly applied. Using one of the formulas for constant acceleration, our purpose was to find the final vertical velocity.

\[ V_f^2 = V_0^2 + 2a\Delta s \]

Assuming a fall of .5 m and an initial vertical velocity of zero, the final vertical velocity was found to have a value of 3.13 m/s. Once again the formula for impact force was used, where the assumption was made that there will be no rebound velocity. The contact time delta \( \Delta t \) was assumed to be of .02 seconds, to magnify the force of the crash.

\[ F_{impact} = \frac{\Delta mv}{\Delta t} = \frac{(90)(3.13)}{.02} = 14164N \]
Since the contact force and the weight are in different directions, the normal force (that will ultimately be applied to the bars) was calculated by taking the difference between them.

\[ F_N = F_{impact} - w \]

*Upper Bar

Basically, the schematic diagram for the force analysis will remain constant for this bar. Once again, only 25% of the whole force was used, for idealization purposes.

\[ R - y' = N - y' = 2715 \text{ N} \]

\[ R - x' = N - x' = 1567.5 \text{ N} \]

The shear and moment diagrams remain the same

Finally, the maximum moment is calculated to be the bending force times the length (.4 m). The moment magnitude is 1086 N*m.
Check for yielding

\[ A = 0.00196 \]
\[ C = 0.0125 \]

\[ l = \frac{\pi}{4} r^4 = \frac{\pi}{4} (0.0125)^4 = 1.917e - 8 m^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(2172 Nm)(0.0125 m)}{(2)(1.917e - 8 m^4)} = 7.08e8 Pa \]

\[ \sigma_{Axial} = \frac{F}{A} = \frac{3135 N}{0.00196m^2} = 1.59e6 Pa \]

\[ \sigma_x = \sigma - \sigma_A = 7.06e8 Pa \]

\[ J = \frac{\pi}{2} r^4 = \frac{\pi}{2} (0.0125)^4 = 3.83e - 8 m^4 \]

\[ \tau = \frac{(592 Nm)(0.0125 m)}{(3.83e - 8 m^4)} = 1.93e8 Pa \]

*Note: For this given scenario the stress will be greater than the Ultimate strength of the material, therefore it will fail.

NX6 analysis

Structural performance
Displacement

Stresses
*Lower Bar*

For the lower bar, the shock absorber will finally play a role, stabilizing the system and allowing the energy to be dispersed. Since the summation of forces always needs to be zero, one end of the bar will sustain an enormous normal force, while the other maintains its original values. The schematic of forces will be as such, along with the magnitude of the reactions:

\[
\begin{align*}
N-x' &= R-x' = 6359 \text{ N} \\
R-y' &= 1252 \text{ N} \\
N-y' &= 11014.4 \text{ N} \\
\text{Shock Absorber Reaction} &= 9762.4 \text{ N}
\end{align*}
\]

The shear and moment diagrams finally change, but still show us that the maximum moment will be present at the end of the bar, where the piece joins the spindle.

The calculation of the maximum moment is as such:

\[
1252(.45) + (9762.4)(.174) = 2264.06 \text{ N m} = M_{\text{end}}
\]

This takes into account the distance from the shock absorber to the edge of the bar.

- Check for yielding
\[ I = \frac{1}{12}bh^3 = \frac{1}{12}(0.054)(0.04)^3 = 2.88 \times 10^{-7} \text{m}^4 \]
\[ \sigma = \frac{Mc}{I} = \frac{(2262 \text{Nm})(0.02 \text{m})}{(2.88 \times 10^{-7} \text{m}^4)} = 1.57 \times 10^6 \text{Pa} \]
\[ \sigma_{Axial} = \frac{F}{A} = \frac{6359.5 \text{N}}{0.00216 \text{m}^2} = 2.94 \times 10^6 \text{Pa} \]
\[ \sigma_x = \sigma - \sigma_A = 1.54 \times 10^6 \text{Pa} \]
\[ \tau = \frac{4.81(T)}{V} = \frac{4.81(1183 \text{Nm})}{(0.004648)^3} = 5.67 \times 10^7 \text{Pa} \]

NX6 analysis

Lower arm structural performance
Displacement

Stresses
Spindle

\[ F = 11014.4 \text{N} \]

\[ l = \frac{\pi}{64} d^4 = \frac{\pi}{64} (0.025)^4 = 1.917e^{-8} \text{m}^4 \]

\[ \sigma = \frac{M_c}{l} = \frac{(11014.4 \text{N}) \left( \frac{0.06}{2} \text{m} \right) (0.0125 \text{m})}{(1.917e^{-8}) \text{m}^4} = 217.4 \text{MPa} \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6 \text{MPa}}{217.4 \text{MPa}} = 1.66 \]

Spindle structural performance

Displacement
Stresses

**Upright**

$F = 11014.4\text{N}$

\[ I = \frac{1}{2} bh^3 = \frac{1}{2} (0.025)(0.02)^3 = 1.66 \times 10^{-8} \text{m}^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(11014.4\text{N})(0.0175\text{m})(0.01\text{m})}{(1.66 \times 10^{-8})\text{m}^4} = 116\text{MPa} \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6\text{MPa}}{116\text{MPa}} = 3.11 \]

**Upright structural performance**
Fourth Case: Bump

In this case, we will assume that the vehicle cruises by at 40mph=17 m/s and sees a semi-circular bump of .5 m radius. Passing through this bump generates a resultant centripetal force, which then magnifies our normal force. The equation for this schematic is:

\[ F_{cen} = F_n - W \]

The weight (maximum) was given by the first case, where the setup is static with a maximum mass. The centripetal force is given by this formula:

\[ F_{cen} = \frac{mv^2}{r} = \frac{17.85^2 \times 90}{.5} = 57670.7 \, N \]

The normal force, which will affect our bars, was taken then by subtracting the individual components of weight from the centripetal force schematic in axial and shear.

*Upper Bar*

The upper bar will receive this greatly magnified force by only 25%, as stated. The moment created will, however still be large and might compromise the integrity of the design.

The force schematic remains as:

Where,

\[ R \cdot x' = N \cdot x' = 7412.1 \, N \]
R-y' = N-y' = 12837.8 N.

The shear and moment diagrams show us that the maximum moment is still concentrated on the end of the bar. Since there are no sharp diameter changes to be considered, it is to be noted that this will be used to calculate the critical location of the bar. A stress analysis of the joints of a machine such as this one show us that this assumption is correct, as failure occurs more than often in these locations.

The calculated maximum moment is then 5135.2 N·m.

*Lower Bar

For the lower bar, the shock absorber will once again have a chance to act and balance the system. The force schematic is then:

Where:

R-x' = N-x' = 29557.5 N

R-y' = 1252 N

Shock Absorber Reaction = 49944.3 N

N-y' = 51196.3 N

The shear and moment diagrams repeat themselves for this case.
The calculation for the maximum moment will then be as such:

\[ 0.45 \times 1252 + 0.174 \times 49944.3 = M = 9253.71 \text{ N} \times \text{m} \]

Conclusion:

After having calculated the force schematic for each of the four cases (static, horizontal dynamic, two vertical dynamics), the problem is to proceed to calculate the stresses based on the critical locations, and then compare these to the presented idealized situation.

**Stress Analysis**

The material properties (AISI 4130) are given as:

\[ S_y = 360.6 \text{ Mpa} \]
\[ S_{ut} = 560.5 \text{ Mpa} \]

These will be used to determine if our cases will be viable, and if so, what life will be expected from them.

For both cases, the torque, which will cause the shear force, was calculated by assuming the horizontal dimensions of the car. The center of mass of the car was assumed to be 4/3m away from both bars. Since the weight is known, the torque was calculated simply by multiplying this by the moment arm.

\[ T = 1183 \text{ N} \times \text{m} \]

This torsional force will not be reduced in any setup, in an attempt to create an extreme situation for each analyzed case. The plane in which it will act will be considered to be x'\text{y}'.

*Lower Bar*

The geometry of the lower bar is presented as an ideal rectangle, and our measurements were taken from the point where the area was at its minimum and moment was given as maximum. The determination of this critical point was based only upon the known critical locations and the fact that there is no sudden change in geometry for this particular element.
The area moment of inertia for this is described with the formula

\[ I = \frac{1}{12} bh^3 = \frac{1}{12} (0.054)(0.04)^3 = 2.88e - 7m^4 \]

The area of this cross section will have a value of .00216 m²

The value of c (distance from the neutral axis to the upper surface) will correspond to .01 m

In the case of the calculation of the torsional shear stress, since the polar moment of inertia of the rectangle is unknown, and the usual formula cannot be applied, it was found that the maximum torsional shear stress of a square is given by

\[ \tau = \frac{4.81 T}{a^3} \]

Where “a” is the value of one side of the square. In order to find a, the area of the rectangle was calculated, and an equivalent square imagined. “a” would be the value of the square root of the area, giving the length.

The normal stress of the loads was given with the formula

\[ \sigma = P/A \]

Where P is the axial load present in each situation. This was calculated in the force setup. Since the value for all Ps is in compression, the normal stress will be subtracted from the bending stress.

Finally, the bending stress is given by the formula

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

The values needed for this equation have already been found. The results of all the stresses and torsional shear stress is presented below.

\[ \sigma = \sigma_m - \sigma_a \]

Case I

\[ \sigma = 3.88E7 Pa \]
\[ \tau = 5.67E7 Pa \]
Case II

\[ \sigma = 3.88E7 \text{Pa}, \quad \tau = 5.67E7 \text{Pa} \]

Case III

\[ \sigma = 1.54E8 \text{Pa}, \quad \tau = 5.67E7 \text{Pa} \]

Case IV

\[ \sigma = 6.29E8 \text{Pa}, \quad \tau = 5.67E7 \text{Pa} \]

*Upper Bar*

The geometry of the upper bar is a perfect, solid circle with a diameter of .25 mm. The determination of the critical point was given as the place where the moment is greatest, which is at the joints. The diameter will not decrease suddenly, which also helps us assume a specific critical location.

The area moment of inertia for this is described with the formula

\[ I = \frac{\pi}{4} r^4 = \frac{\pi}{4} (.0125)^4 = 1.917e - 8m^4 \]

The polar moment of inertia for the cross section is given with

\[ J = \frac{\pi}{2} r^4 = \frac{\pi}{2} (.0125)^4 = 3.83e - 8m^4 \]

The area of this cross section will have a value of .00196 m^2
The value of \( r \) (distance from the neutral axis to the upper surface) will correspond to .0125 m.

Using the moments calculated in the force analysis the maximum stresses and torsional shear forces were able to be calculated. The values for normal and bending stress are given by the formulas presented in the lower bar analysis. The only formula that will differ is that one for torsional shear stress. The formula, corresponding to a more traditional (and much more used) setup is then

\[
\tau = \frac{T r}{J}
\]

The individual results for the upper bar are presented as such:

\[
\sigma = \sigma_m - \sigma_a
\]

Case I

\[
\sigma = 9.13E7 \text{ Pa}
\]

\[
\tau = 3.86 E8 \text{ Pa}
\]

Case II

\[
\sigma = 9.13E7 \text{ Pa}, \text{ in z } \sigma = 2.62E9 \text{ Pa}
\]

\[
\tau = 3.86 E8 \text{ Pa}
\]

Case III

\[
\sigma = 7.072 E8 \text{ Pa}
\]

\[
\tau = 3.86 E8 \text{ Pa}
\]

Case IV

\[
\sigma = 3.346 E9 \text{ Pa}
\]

\[
\tau = 3.86 E8 \text{ Pa}
\]

**Failure Analysis**

Even though the last three cases given correspond to a dynamic setup, one must also analyze what effect would these forces, if given as static, have on the suspension. For this,
there are several manners in which the calculations can be done, the first one corresponding to the formula

\[ F.S. = \frac{3.6 \times 10^8}{\sigma} \]

For this case, however, it would be best to use one of the conservative, ductile theories, in this case distortion energy. This theory uses the main formula

\[ F.S. = \frac{S_y}{\sigma'} = \frac{3.6E8}{\sigma'} \]

For which \( \sigma' \) corresponds to the von misses stress. The formula to obtain the factor of safety is based upon the yield strength of the material and this calculated, modified stress.

\[ \sigma' = \frac{1}{\sqrt{2}} \left[ (\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{1/2} \]

Using the solutions found in the stress analysis, the factor of safety for a static configuration is able to be obtained.

*Upper Bar*

Case 1

\[ \sigma' = 3.47E8 \text{ Pa} \]

\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{3.47E8 \text{ Pa}} = 0.104 \]

Case 2

\[ \sigma' = 2.6E9 \text{ Pa} \]

\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{2.6E9 \text{ Pa}} = 0.14 \]

Case 3

\[ \sigma' = 7.82E8 \text{ Pa} \]

\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{7.82E8 \text{ Pa}} = 0.46 \]

Case 4

\[ \sigma' = 3.36E9 \text{ Pa} \]
Conclusion: From the point where the stresses were calculated, it was apparent that some exceeded the ultimate tensile strength, corresponding to 560.5 MPa. The calculation of the factor of safety was just a formality, as one knows that after this limit is reached, the material will break. Thus, only case one will hold the required load. This shows us that if our shock absorber were to fail, and the upper bar need to take ¼ of the load, it would hold for a period of time; if the mini-baja was driven and set up in extreme situations (both horizontal and vertical dynamic) the components would break and the setup fail. This, however, does not mean that the design is flawed. It simply implies that the shock absorber is a very important component of every suspension design.

*Lower Bar*

Case 1

\[ \sigma' = 1.6E8 \text{ Pa} \]
\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{1.6E8 \text{ Pa}} = 0.34 \]

Case 2

\[ \sigma' = 1.24E9 \text{ Pa} \]
\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{1.24E9 \text{ Pa}} = 0.289 \]

Case 3

\[ \sigma' = 1.83E8 \text{ Pa} \]
\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{1.83E8 \text{ Pa}} = 1.97 \]

Case 4

\[ \sigma' = 6.37E8 \text{ Pa} \]
\[ n = \frac{S_y}{\sigma'} = \frac{360.6E6 \text{ Pa}}{6.37E8 \text{ Pa}} = 0.565 \]

Conclusion: Out of this setup, two factors of safety resulted as more than one, which is the limit that allows us to conclude that the material will hold. Cases two and four represent
extreme situations in which the lower arm would immediately break. When the setup of these was calculated, this was unknown; the calculation of these was informative and shows us that an optimization, or several designs would be needed to achieve maximum reliability and strength.

**Calculation of Se (endurance limit)**

Factors $K_a$ through $K_f$ have to be calculated for both bars. The material in question is AISI 4130, assumed to be machined, with these properties:

$$S_y = 360.6 \text{ Mpa}$$
$$S_{ut} = 560.5 \text{ Mpa}$$

**Upper Bar**

*Surface Factor $K_a$*

$$k_a = aS_{ut}^{-b} = .843 \text{ Where } a=4.51, b=-.266$$

*Size Factor $K_b$*

Since the bar is not rotating, an equivalent diameter needs to be found.

$$d_e = .37(d) = .37(25) = 9.25$$

Since $d=9.25 \text{ mm}$,

$$k_b = .879d^{-1.107} = .977 = 1$$

*Loading Factor $K_c$*

The loading factor simply indicates which variable load is to be analyzed in the setup.

$$k_c = 1 \text{ (bending)}$$

The next factors ($K_d$, $K_e$, $K_f$) will be assumed to be one, since no further information is given.

$Se'$

$$Se' = .5S_{ut} = .5(560.5) = 280.25 \text{ Mpa}$$
\[ Se = Se' \times Ka \times Kb \times Kc \times Kd \times Ke \times Kf \]
\[ Se = 280.25(0.843)(1)(1) = 235.97 \text{ Mpa} \]

*Lower Bar*

**Surface Factor Ka**

\[ k_a = 0.843 \]

**Size Factor Kb**

Since the cross section of the bar is not circular, and it’s not rotating, an equivalent diameter needs to be found.

\[ d_e = 0.808(hb)^{1/2} = 26.55 \text{ mm, where } h = 20 \text{ mm and } b = 54 \text{ mm} \]

Since \( d = 26.5 \text{ mm} \),

\[ k_b = 0.879d^{-0.107} = 0.873 \]

**Loading Factor Kc**

The loading factor simply indicates which variable load is to be analyzed in the setup.

\[ k_c = 1 \text{ (bending)} \]

The next factors (Kd, Ke, Kf) will be assumed to be one, since no further information is given.

**Se’**

\[ Se' = 0.5S_{ut} = 0.5(560.5) = 280.25 \text{ Mpa} \]

**Se**

\[ Se = Se' \times Kb \times Kc \times Kd \times Ke \times Kf \]
\[ Se = 280.25(0.87)(0.843)(1) = 250.54 \text{ Mpa} \]

**Operating Range**

**Dynamic**
The operating range for the dynamic cases is able to be found using the modified Goodman equation, if one were to consider the stress as having fluctuating characteristics. In all cases, our idealized factor of safety will correspond to 2.

Modified Goodman

\[
\frac{\sigma_a}{S_y} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}
\]

Stress Amplitude

\[
\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}
\]

Mean Stress

\[
\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}
\]

*Horizontal*

For this case, our minimum stress would obviously correspond to a value of zero, as it’s not expected for the vehicle to be continuously hit horizontally by other than the force of air resistance or small pebbles. If one were to determine our maximum stress using the modified Goodman equation and assuming this, the setup would look as such:

\[
\frac{\sigma_{max}}{205.54E6} + \frac{\sigma_{max}}{560.5E6} = \frac{1}{2}
\]

The value for maximum stress will then be 7.52 E7 Pa, or 75.2 Mpa.

The operating range for horizontal, continuous impact will thus be from 0 to 75.2 Mpa.

*Vertical*

In this case, our minimum stress will be given by our maximum total load-calculated stress, which is already in the past calculations. Two situations are presented: where the shock absorber fails and the upper bar needs to carry 25% of the load, and that where the lower bar takes 100% of the shock. Modified Goodman is once again applied for a safety factor of two, and our operating range is found.

-Upper bar

\[
\sigma_{min} = 9.13 E7 Pa
\]
Operating Range: 91.3 MPa-192 MPa

-Lower bar

\[
\frac{\sigma_{\text{max}} - 9.13 \times 10^7}{2 \times 205.54 \times 10^6} + \frac{\sigma_{\text{max}} + 9.13 \times 10^7}{2 \times 560.5 \times 10^6} = \frac{1}{2}
\]

\[
\sigma_{\text{max}} = 1.92 \times 10^8 \text{ Pa}
\]

Operating Range: 38.8 MPa-168 Mpa

Static

The analysis of the static maximum setup will only describe a non-moving setup, where a load is applied on the vehicle and the bars should not yield. This will correspond as well to a safety factor of 2. The formula that will be used in the setup will simply be described by:

\[
F.S. = \frac{3.6 \times 10^8}{\sigma}
\]

Where stress is to be considered as maximum. In turn, to be able to find the maximum applied force, the calculation for the stress is given by this formula:

\[
\sigma = \frac{F \times .02}{2.88 \times 10^{-7}}
\]

It’s noticeable that the axial-caused stress will be neglected in this setup for added simplification; only the lower bar will be considered in this case, where it receives 100% of the forces.

The joined equations are then given by

\[
\frac{F \times .02}{2.88 \times 10^{-7}} = \frac{3.6 \times 10^8}{2}
\]

The force is then calculated to have a value of 2592 N. It is to be noted that this is only the force that causes bending, found by the equation

\[
F = F_{\text{total}} \sin(60)
\]
For which the total force is found to be 6651 N. If this total force is then assumed to be corresponding to a weight, and multiplied by 4 suspensions, the maximum amount of mass that would be accepted in this setup would be of 2712 kg. (For both the frame and other added weights).

The operating range is then from the original mass of the frame to 2712 kg.

**Life Cycle**

It is to be noted that the modification factor $k_f$ is going to be neglected for simplification, and also due to the fact that in these bars there are no sharp changes in geometry, but rather happen gradually.

Given that the factors of safety of three cases obtained by the distortion energy theory are more than one, the lives for these situations would be assumed to be infinite. For instructive purposes, however, the cycles to failure will be calculated. Note: Infinite life is assumed where the cycles to failure exceed E6. The factor $f$, to be used in $a$ and $b$, is found to be approximately .875, according to a specialized table. For simplification purposes, the reversible stress will be assumed to be that which was found in the initial assessment. The equations that will be used in this case are as such:

$$N = \left(\frac{\sigma_{\text{rev}}}{a}\right)^{1/b}$$

$$a = (fS_{ut})^2/S_e$$

$$b = -\frac{1}{3\log\left(fS_{ut}/S_e\right)}$$

*Upper Bar*

$$a = \frac{(fS_{ut})^2}{S_e} = \frac{.875 \times 560.6^2}{236} = 1019.55$$

$$b = -\frac{1}{3\log\left(fS_{ut}/S_e\right)} = -.106$$

Case I

$$\sigma = 91.3 \text{ MPa}$$

$$N = \left(\frac{91.3}{1019.55}\right)^{-1/.106} = 7.69E9 \text{ Cycles}$$
Case I

\[ \sigma = 38.8 \text{ MPa} \]

\[ N = \left( \frac{38.8}{1170.65} \right)^{-1/1.26} = 5.53E11 \text{ Cycles} \]

Case III

\[ \sigma = 154 \text{ MPa} \]

\[ N = \left( \frac{154}{1170.65} \right)^{-1/1.26} = 9.8E6 \text{ Cycles} \]

\[ \begin{align*}
\text{AISI 4130} & \quad S_{ut} = 560.6 \text{ MPa} \\
\text{ } & \quad S_y = 360.6 \text{ MPa}
\end{align*} \]

**Upright**

*Note: In static scenario, the static load is 1000N. From here we assumed that for a dynamic scenario the load will be of 2000N.*

\[ F(\text{Dynamic}) = 2000 \text{N} \quad \Rightarrow \quad F(70\%) = 1400 \text{N} \quad \Rightarrow \quad M = (1400 \text{N})(0.0175m) = 24.5 \text{ Nm} \]

\[ I = \frac{1}{2}bh^3 = \frac{1}{2}(0.025)(0.02)^3 = 1.66e-8 \text{m}^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(24.5 \text{Nm})(0.01m)}{(1.66e-8)m^4} = 14.75 \text{MPa} = \sigma_{rev} \]

\[ S'_e = 0.5S_{ut} = 0.5(560.6 \text{MPa}) = 280.3 \text{MPa} \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6 \text{MPa}}{14.76 \text{MPa}} = 24.4 \]
*Note: All the factors and necessary data were obtained from figures and tables in Shigley’s Mechanical Engineering Design book.

These are factors of temperature, surface, size, loading, reliability and miscellaneous-effects taking into account to calculate the endurance limit:

\[ Ka=0.84 \]
\[ Kb=0.91 \]
\[ Kc=1 \]
\[ Kd=1 \]

\[ Se = k_a k_b k_c k_d S_e' = (0.84)(0.91)(280.3) = 214.3 \text{ MPa} \]

Using Figure A-15-6
\[ \frac{d}{d} = \frac{25}{20} = 1.25 \]
\[ \frac{r}{d} = \frac{5}{20} = 0.25 \]
\[ \Rightarrow k_t = 1.4 \]

From Figure 6-20 we assume \( q=0.8 \)

\[ k_f = 1 + q(k_t - 1) = 1 + 0.8(1.4 - 1) = 1.32 \]

“\( f \)” is a material property that varies with \( S_{ut} \). From fig. 6-18 we get that \( f=0.87 \)

\[ a = \frac{(fS_{ut})^2}{Se} = \frac{(0.87 \times 560.6)^2}{214.3} = 1110 \text{ MPa} \]

\[ b = \frac{1}{3} \log \left( \frac{0.87 \times 560.6}{214.3} \right) = -0.119 \]

\[ N = \left( \frac{k_f \sigma_{rev}}{a} \right)^{1/b} = \left( \frac{1.32(14.75)}{1110} \right)^{1/-0.119} = 5.7 \times 14 \text{ cycles} \]

**Spindle**

\[ I = \frac{\pi}{64} d^4 = \frac{\pi}{64} (.025)^4 = 1.917e - 8 \text{ m}^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(2000Nm)(.06m)(.0125m)}{(1.917e - 8)m^4} = 78 \text{ MPa} = \sigma_{rev} \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6 \text{ MPa}}{78 \text{ MPa}} = 4.7 \]

**Life Cycle**

*Note: All the factors and necessary data were obtained from figures and tables in Shigley’s Mechanical Engineering Design book.
These are factors of temperature, surface, size, loading, reliability and miscellaneous-effects taking into account to calculate the endurance limit:

- \( K_a = 0.843 \)
- \( K_b = 0.879 \)
- \( K_c = 1 \)
- \( K_d = 1 \)

\[
S_e' = 0.5S_{ut} = 0.5(560.6\text{MPa}) = 280.3\text{MPa}
\]

\[
Se = k_ak_bk_ck_dS_e' = (0.843)(0.879)(280.3) = 207.3\text{MPa}
\]

Using Figure A-15-6

\[
\begin{align*}
\frac{d}{d} &= \frac{30}{25} = 1.2 \\
\frac{r}{d} &= \frac{1}{25} = 0.04 \\
\Rightarrow 
\end{align*}
\]

\[ k_t = 2.1 \]

From Figure 6-20 we assume \( q = 0.7 \)

\[ k_f = 1 + q(k_t - 1) = 1 + 0.7(2.1 - 1) = 1.77 \]

“\( f \)” is a material property that varies with \( S_{ut} \). From fig. 6-18 we get that \( f = 0.87 \)

\[
a = \frac{(fS_{ut})^2}{Se} = \frac{(0.87 \times 560.6)^2}{214.3} = 1145.3\text{MPa}
\]

\[ b = -\frac{1}{3} \log \left( \frac{0.87 \times 560.6}{207.7} \right) = -0.1236 \]

\[
N = \left( \frac{k_f\sigma_{rev}}{a} \right)^{1/b} = \left( \frac{1.77(78)}{1145.3} \right)^{1/-0.119} = 27e6 \text{ cycles}
\]

**Bottom Arm** (under normal driving conditions)

*Note: Because the static load of the cart is approximately 1000N, we assume that in normal driving conditions the force applied will be 2000N

\( F = 2000N \)

\[
I = \frac{1}{12}bh^3 = \frac{1}{12}(0.054)(0.04)^3 = 2.88e-7m^4
\]

\[
\sigma = \frac{Mc}{I} = \frac{(2000N)(0.45)(0.02m)(0.7m)}{(2.88e-7m^4)} = 43.75\text{MPa}
\]

\[
\sigma_{Axial} = \frac{F}{A} = \frac{(2000N)(0.7)}{0.00216m^2} = 648 \text{ MPa}
\]

\[ \sigma_x = \sigma - \sigma_A = 43.102 \text{ MPa} \]
\[
\tau = \frac{4.81(T)}{V} = \frac{4.81(1183Nm)}{(0.004648)^3} = 56.7 \text{ MPa}
\]

\[
\sigma' = \sqrt{\sigma_x^2 + 3\tau^2} = \sqrt{43.102^2 + 3(56.7)^2} = 107.23 \text{ MPa}
\]

\[
n = \frac{S_y}{\sigma} = \frac{360.6}{107.23} = 3.36
\]

Life Cycle

*Note: All the factors and necessary data were obtained from figures and tables in Shigley’s Mechanical Engineering Design book.

These are factors of temperature, surface, size, loading, reliability and miscellaneous-effects taking into account to calculate the endurance limit:

Ka=0.84
Kb=0.88
Kc=1
Kd=1

\[
S_e' = 0.5S_{ut} = 0.5(560.6\text{MPa}) = 280.3\text{MPa}
\]

\[
S_e = k_ak_bk_ck_dS_e' = (0.843)(0.879)(280.3) = 207.2\text{MPa}
\]

Using Figure A-15-6 we assumed an approximation for a notch radius of 1mm, because the geometry we used for our component doesn’t match the tables at the back of the book. \(k_t = 2.1\)

From Figure 6-20 we assume q=0.7

\[
k_f = 1 + q(k_t - 1) = 1 + 0.7(1.9 - 1) = 1.63
\]

“f” is a material property that varies with Sut. From fig. 6-18 we get that \(f=0.87\)

\[
a = \frac{(fS_{ut})^2}{S_e} = \frac{(0.87\times560.6)^2}{207.2} = 1148\text{MPa}
\]

\[
b = -\frac{1}{3}\log\left(\frac{0.87\times560.6}{207.2}\right) = -0.124
\]
\[ N = \left( \frac{k_f \sigma_{rev}}{a} \right)^{1/b} = \left( \frac{1.63(107.23)}{1148} \right)^{1/-0.119} = 3.91e6 \text{ cycles} \]

**Upper Arm**

\[ I = \frac{\pi}{4} r^4 = \frac{\pi}{4} (0.0125)^4 = 1.917e - 8 m^4 \]

\[ \sigma = \frac{Mc}{I} = \frac{(2000N)(.3m)(.4)(.0125m)}{(2)(1.917e - 8)m^4} = 156.5 \text{ MPa} \]

\[ \sigma_{Axial} = \frac{F}{A} = \frac{240N}{.00196m^2} = .122 \text{ MPa} \]

\[ \sigma_x = \sigma - \sigma_A = 156.4 \text{ MPa} \]

\[ J = \frac{\pi}{2} r^4 = \frac{\pi}{2} (0.0125)^4 = 3.83e - 8 m^4 \]

\[ \tau = \frac{(592Nm)(.0125m)}{(3.83e - 8)m^4} = 1.93e8Pa \approx 193 \text{ MPa} \]

\[ \sigma' = \sqrt{\sigma_x^2 + 3\tau^2} = \sqrt{156.4^2 + 3(193)^2} = 370 \text{ MPa} \]

\[ n = \frac{S_y}{\sigma} = \frac{360.6}{370} = 0.97 \]

*Note: Thus the 30% of the force will be the limit of the load for the upper arm.

**Optimization**

The optimization of the setup would obviously include a completely new design in which the optimal geometry will be calculated, where a lighter alloy could be used. Although our suspension would work very well in a mini-baja design, in a professional environment further analysis would be required. Some of the ways that the design could be improved include a selection of another material, increasing the cross-sectional areas and/or reducing the length of the arms to further decrease the moment created by the forces. Some examples where such modifications were applied where in the cases where the thickness of the lower arm had to be increased from 20mm to 40mm in order to withstand the loads applied. Another modifications needed was to convert the upper arm from hollow to solid in order to increase the safety factor and prolong the life of this part.
6. DRAWBACKS (LIMITATIONS OF DESIGN)

Just like any other design, this prototype suspension has some drawbacks, in other words there are some limitations in its design. The following points do explain and develop such limitations:

- MANUFACTURABILITY/MACHINE

It is very important for engineers to design for manufacturability. In other words the components, parts or the entire assembly being designed need to be relatively easy to manufacture. If a component can be easily manufactured, it means the productions costs can be taken to a minimum and production speed can be improved. For this project it was taken into consideration to design the components of the suspension to be ease to manufacture. In other words most of the parts can be fabricated with basic tools.

The most challenging part to manufacture is the lower arm of the suspension. The frame of this arm needs to be machined in order to obtain the best results. This lower arms was designed to save weight, therefore mass was removed in designated points of the arm without scarifying the integrity of the structure while some reinforcement were added where the mass was removed. Such reinforcement is tubular in shape and need to be welded in place in an X shape which adds some degree and difficulty to manufacture. The end link which attaches to the upright is another point of the lower arm which is a little complex to fabricate. This is because it is necessary to place a ball joint in this place which will hold the lower arm and upright together. This ball joint base needs to be manufactured with very strict standards of measurement in order to provide the best performance.

The upright is one of the most important parts of this system since it connects all parts of the suspension together. Thus it was important for our team to come up with an efficient, strong, light weight model. The model proposed is a very exceptional piece which its benefits are low mass, light weight, improved use of space and durability. The drawbacks of this part are the awkward shape of this part which increases the difficulty in manufacturability. The upright needs to possess several connection links that need to be strong enough to hold the critical
loads which the cart is going to be put into. Such links need to be welded in place, which means change in geometry is going to be present leading to creating stress concentration factors.

Materials used as described before are going to be AISI 4340 and AISI 4130 (most commonly used as chromoly). Both steels are widely used in the industry which facilitated the ease of access to them. Machinability of both alloys is considered to be done by conventional methods, and best with the alloys in normalized and tempered conditions. Respecting to welding both alloys is noted for their weld ability by all commercial methods. Materials are not really considered as drawbacks.

7. DISCUSSION

- The design project is based on a suspension unit that has a combination of functionality for the purpose of (SAE Mini-Baja cart) and a concept of saving space.
- Actually the cart has a front suspension unit that consists of:
  - Two control arms these are made of two tubes at an angle with link connection to the upright (both upper and lower arm have triangle shape and equal length, but lower arm has a shock absorber connection).
  - The upright has a bolt in connection for the control arms and brake caliper, tie rod and spindle mounting area respectively.
  - The spindle is a solid shaft with diameter reduction along its length and tread at the end to use a safety nut to maintain the hub on its place.
  - The hub with a 4 bolt pattern for the wheel and brake disk respectively and inside bearing.
- *Our design has basically the same number of parts but, we decided to save space and material (as we mentioned in the introduction) by doing some modifications on the control arms and their connections. We also modify the upright but we kept the same design format for the spindle and hub in order to obtain better results.*
- The design we are offering is a front suspension unit with modifications on parts previously mentioned consists of:
  - Two control arms:
Upper arm is a single hollow tube that is adjustable using a treaded solid tube in the middle of the part which has a fixed nut on the center of it and safety nuts on each side at the end of the tread. We used a ball joint to connect it with the upright. Lower arm is a single part (rectangular shape but it has a diagonal cut at the end where is connected with the upright.) we also use a ball joint as connection with the upright.

- The upright:

We decided to change the width of this part and considered the same height in order to use the same wheel diameter. We relocated the position of brake caliper, tie rod and control arms connections to obtain good braking, to use the tie bar in the back of the whole suspension unit avoiding breakage under hard driving conditions and arm connections in the back of the upright to obtain good results with the ball joints that we have on the control arms. We use different dimensions on the mountings to balance the changes we made on the upright.

- We considered that the spindle and the hub will be able to produce good results using them same as the actual model of the cart.

After we complete some analysis we defined the following:

- The upper arm that we modify fails in certain scenarios, so we notice that failure was produce on the hollow tube then we decide to make it solid under the same format. Then we find better results on the analysis.

- We notice certain weaknesses during the analysis of the bottom arm. Then we decided to double its height and we find good results on different scenarios where we apply load.

- During the analysis of the modified upright we figure out certain load then after applying it on our critical points, we found that it was having a good reaction with the changes previously mentioned.

Since our design is completely different to the actual model we compare and discuss its functionality, material usage, expenses, and other different criteria. Then we decide to design on the format shown in this report.
8. **CONCLUSION**

After an in depth analysis of each individual part of our suspension unit design we have come to the conclusion that if we optimize our design it will be a good choice for SAE Mini –Baja requirements. Some of the optimizations we have in mind include further analyzing the upper control arm and possibly increasing its diameter; and also considering finding a more appropriate ball joint for this part. In the optimization section of this report it was mentioned that because of the increase in height of the bottom arm, we will need to find a more adequate ball joint for this component that will adapt more to the modified dimensions and increase its performance. After further review of our analysis we took notice that our spindle and upright did not fail in any of the presented scenarios. This implies that the two parts that we designed do not need additional modifications to improve their performance. Putting aside the analysis of our design, we can say that each of our group members have become more familiar with the useful tools that were applied during the design process. This includes the NX6 software, knowledge acquired during lectures, as well as applying basic concepts of design. Furthermore we had to use knowledge acquired from prior classes not just our current ones.