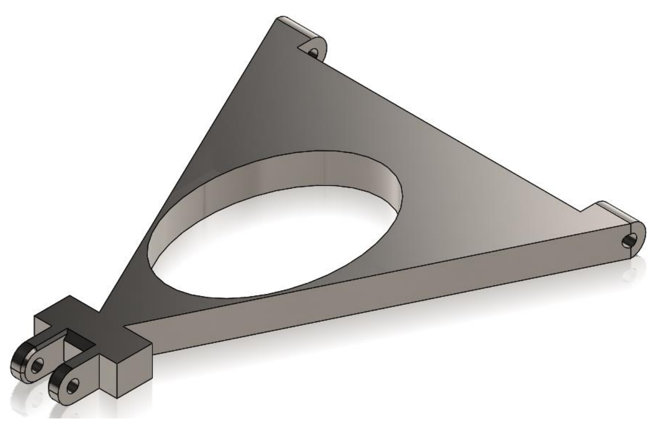
MecE 468

Assignment 6

Midterm Mini Project: Design of an A-arm for a Vehicle Suspension



Toya Okeke

October 24, 2017

Abstract

Vehicle suspensions are used to control the relative motion between a vehicle and its wheels. The shock absorber of the vehicle is what prevent large amplitudes of motions when the vehicle experiences some sort of impulsive load. If the spring and damper of the shock absorber are predetermined, the forces experienced by the connecting arms (A-arms) during the motion must be minimized so the suspension stays intact. Therefore, the purpose of this assignment is to design a new A-arm made of alloy steel for a 2017 Dodge Charger SE that maximizes the stress without yielding and keeps the deflections in the A-arm less than 1 mm. The new design maximizes the stress maximizes the stress 193.82 MPa and reduces the reaction force at the pin connections from 64.9 kN to 51.6 kN. However, the displacements in the A-arm increase, and larger displacements now occur around the hole discontinuity. To build 8 A-arms, the cost of each would be 242.02 USD. 28% of this cost goes to materials and 72% to manufacturing. Although the design goals were met, different designs should be explored because of the displacements that occur around the hole.

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1 Introduction

Vehicle suspensions are used to control the relative motion between a vehicle and its wheels. The shock absorber of the vehicle is what prevent large amplitudes of motions when the vehicle experiences some sort of impulsive load. If the spring and damper of the shock absorber are predetermined, the forces experienced by the connecting arms (A-arms) during the motion must be minimized so the suspension stays intact. Software that can do numerical simulation, like Solidworks, will run a motion and design study to optimize the dimensions of critical areas in the design. Therefore, the purpose of this experiment is to design a new A-arm made of alloy steel for a 2017 Dodge Charger SE. the design should maximize the stress that can be applied to the A-arm without yielding. Also, the deflection in the design during its motion should not exceed 1 mm. To analyze this, a simulation of one tire going over a speed bump at 30 km/h will be done. The initial design of the A-arm is shown in *Figure 1* below.

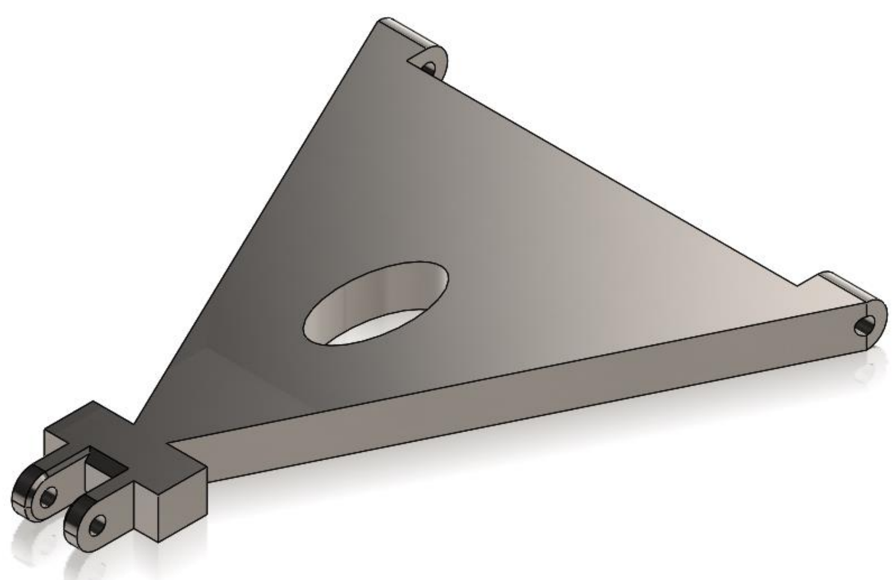


Figure 1 Initial design of the A-arm with an elliptic hole of 40 mm vertical radius and 20 mm horizontal radius

2 Description

Table B 1 contains the vehicle specifications for a 2017 Dodge Charger SE. a speed bump with a 10 cm radius was placed along the path of the wheel. Speed bumps are normally found in areas where the maximum vehicle speed is normally less than 30 km/h, such as parking lots. Given the radius of the wheel is 26 cm, the angular speed of one wheel is equivalent to approximately 306 RPM based on the speed of the vehicle. The motion study done on the wheel keeps the wheel in contact with the ground. Using the vehicle design specifications, the damping coefficient for the shock absorber was determined from the MATLAB code in **Appendix A**, yielding a value of 11.33 Ns/mm. The motion study was assumed to be under-damped with a damping ratio of 0.7. With these parameters, the preliminary motion study of the suspension was altered to analyze the forces applied for Dodge Charger.

3 Analysis

3.1 Motion Analysis of Assembly using the Initial A-arm Design

From the initial motion study of the A-arm design, the plots of the reaction force plot at the pin connections of the A-arm were generated. These can be seen in *Figure 2*, *Figure 3*, *Figure 4* and *Figure 5*. The maximum reaction force is 64.9 kN and occurs at 0.68 s.

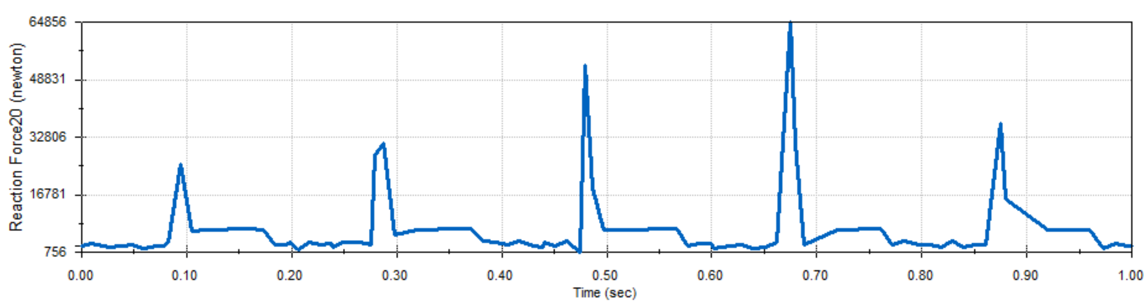


Figure 2 Reaction force magnitude for the pin connections of the initial A-arm design

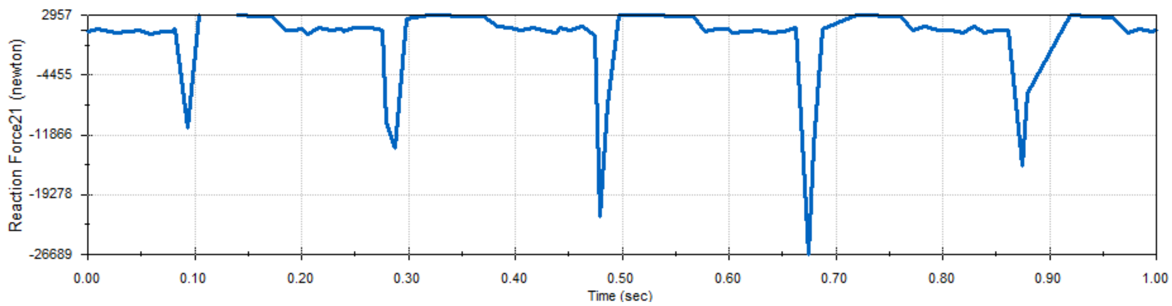


Figure 3 Reaction force x component for the pin connections of the initial A-arm design

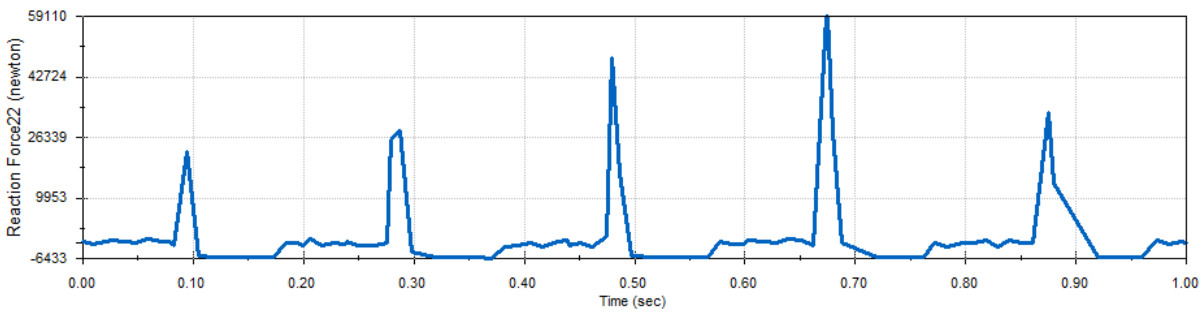


Figure 4 Reaction force y component for the pin connections of the initial A-arm design

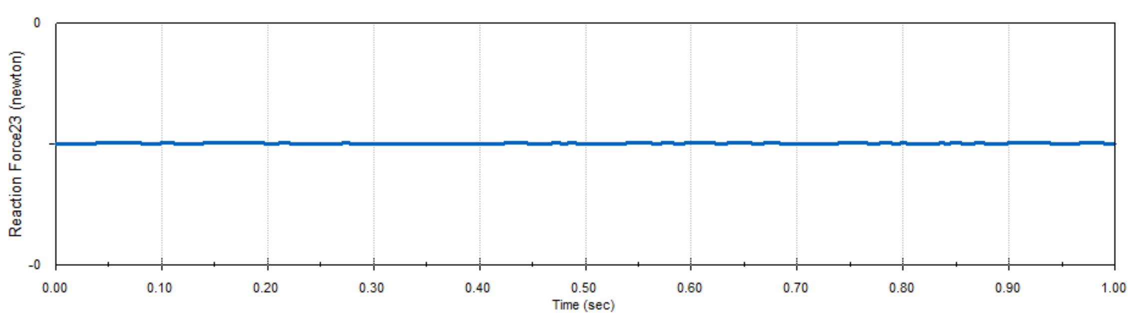


Figure 5 Reaction force z component for the pin connections of the initial A-arm design

3.2 Initial Mesh, Stress Distribution and FOS of A-arm Design

Using manual convergence, the maximum von Mises stress converges to approximately 118 MPa, as shown in *Figure 6*. The percent difference between a 2.5 mm and 2 mm element size, shown in Table 1, suggests the same thing.

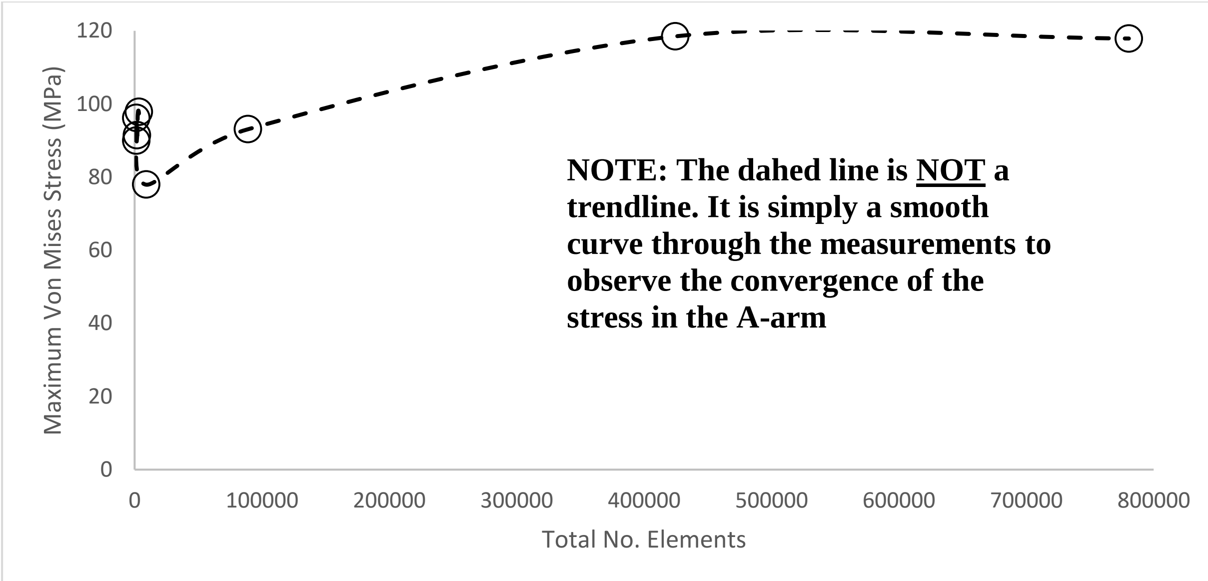
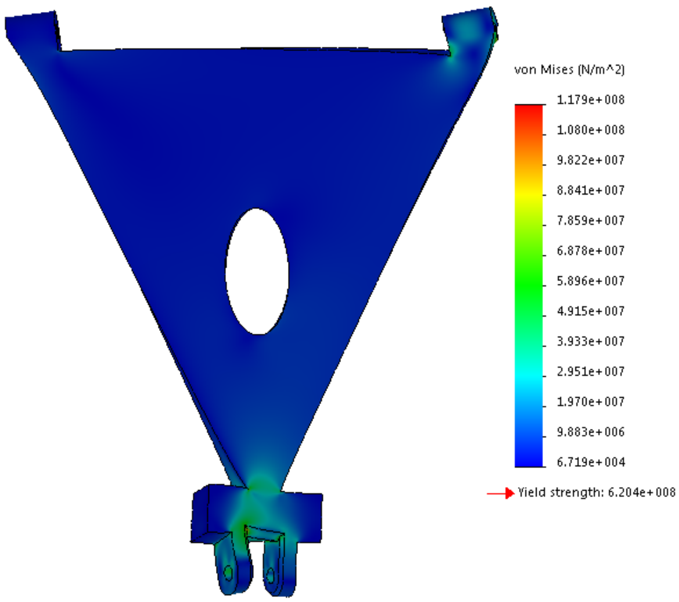


Figure 6 Manual convergence plot of the initial A-arm design

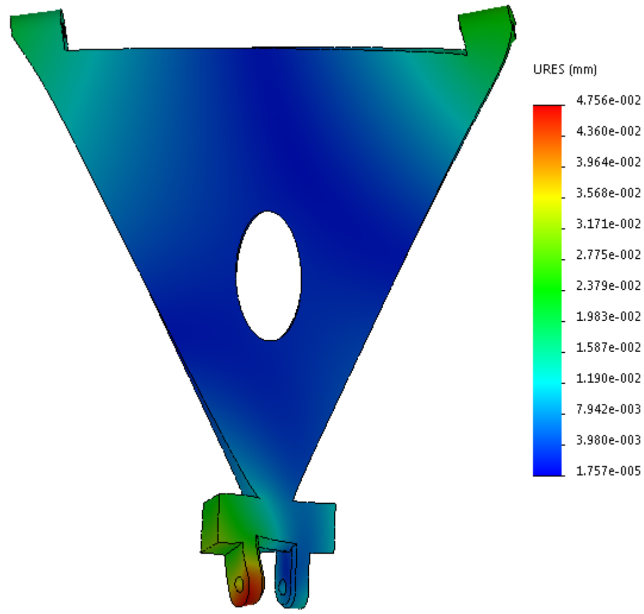
Table 1 Convergence analysis of the initial A-arm design

Element size (mm)	Total No. Elements	Mex. VM Stress (Mpa)	Percent Difference
30	1264	89.95	
25	1462	96.14	6.65%
20	1981	91.41	5.04%
15	3280	97.82	6.77%
10	9038	77.93	22.63%
5	89010	93.08	17.72%
2.5	424605	118.5	24.03%
2	780562	117.9	0.51%

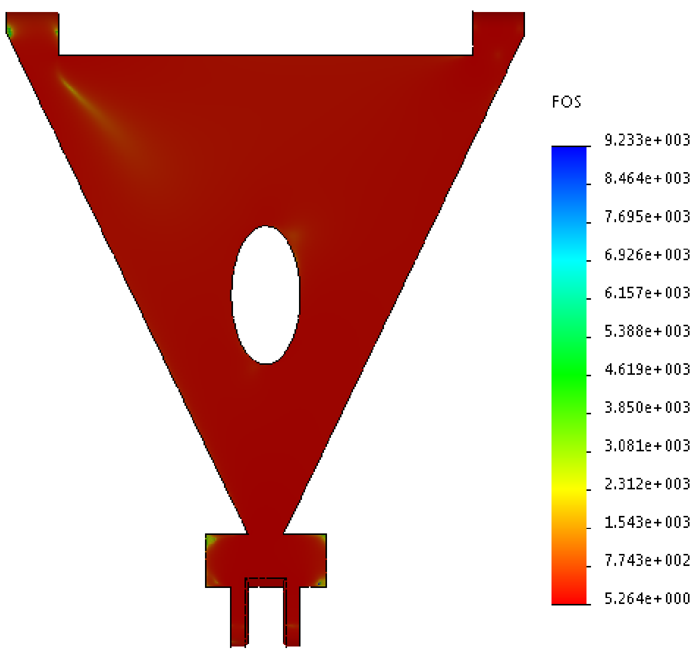
The stress converges at a 2 mm element size, which is a very accurate mesh. Therefore, this will be used throughout the design study. The stress, displacement and FOS color maps for the initial design and in *Figure 7*. The mesh of the A-arm can be seen in *Figure 8*. Notice that the deformations in the design have been exaggerated in the stress and displacement color maps.



(a)



(b)



(c)

Figure 7 (a) stress, (b) displacement and (c) FOS color maps for the initial A-arm design

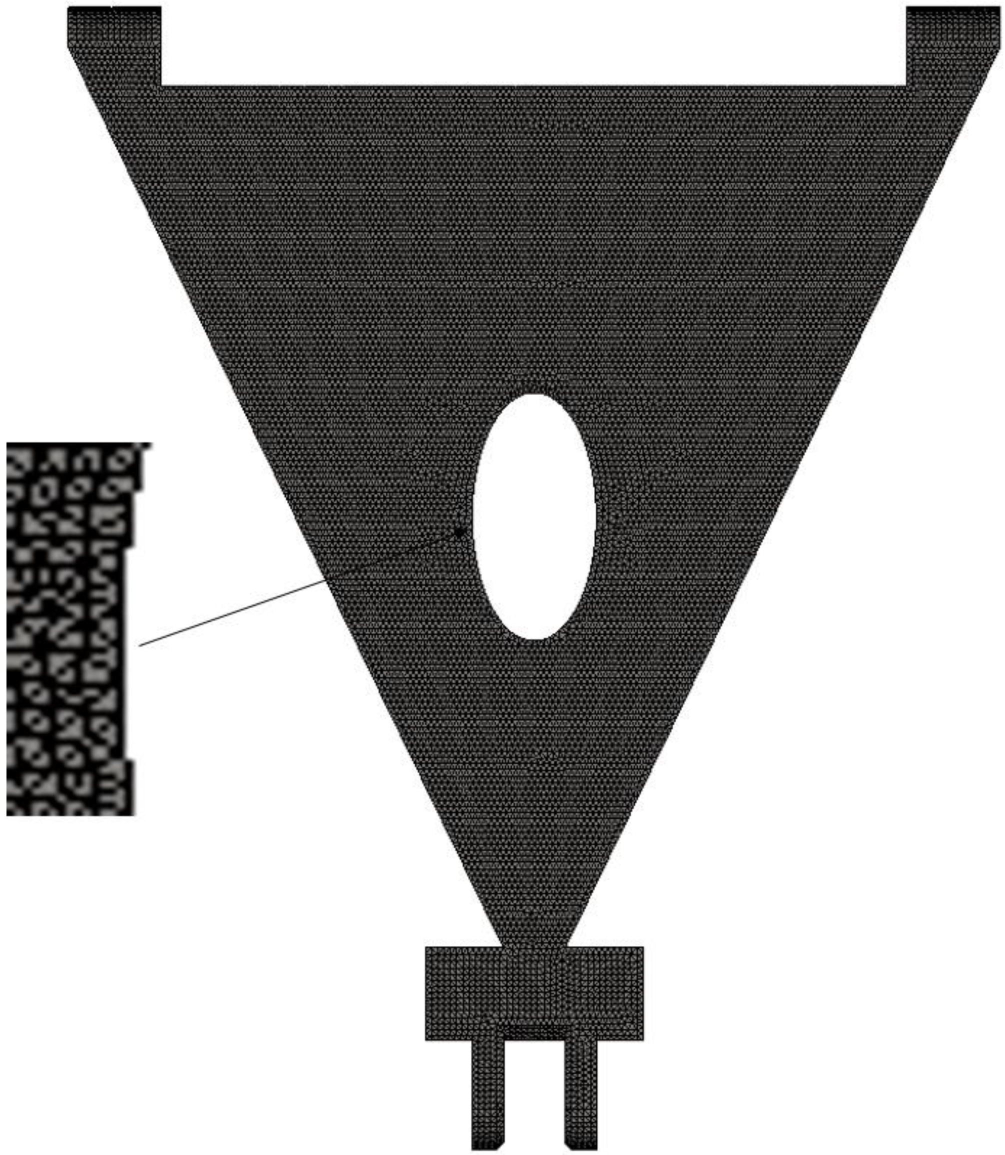


Figure 8 Mesh of 2 mm element size for the intial A-arm design

3.3 Design Study of Initial A-arm

Initially, the hole in the A-arm had a 40 mm vertical radius and 20 horizontal radius, as shown in *Figure 9*. The geometry of the hole discontinuity was altered to the dimensions shown in Table 2. After running the design study, scenario 7 was deemed optimal, as shown in Table 3. The maximum stress occurred by the optimal design was 193.82 MPa, while maintaining a deflection less than 1mm.

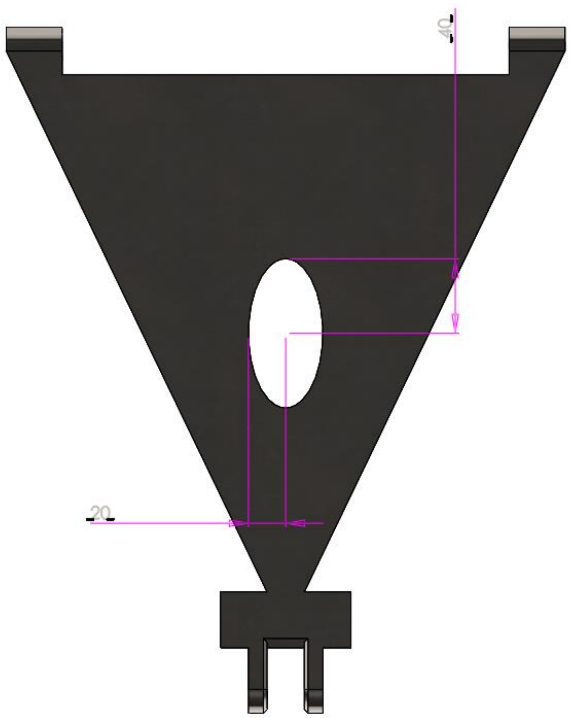


Figure 9 Initial design with hole dimensions

Table 2 Different configurations of the hole in the A-arm design

	vert_rad @Sketch 14	hor_rad @Sketch 14
Default	40	20
hole2	20	10
hole3	50	30
hole4	70	40
hole5	70	60
hole6	80	60
hole7	90	60

Table 3 Define study table of different hole configurations in the A-arm

Hole Optimization											
Scenarios/Iterations:	7										
Parameter Constraint or Goal	Format	Unit	Initial Value	Optimal Value	Scenario 1	Scenario 2	Scenario 3	Scenario 4	Scenario 5	Scenario 6	Scenario 7
					Calculated	Calculated	Calculated	Calculated	Calculated	Calculated	Calculated
vert_rad		mm	40	90	40	20	50	70	70	80	90
hor_rad		mm	20	60	20	10	30	40	60	60	60
Minimum Factor of Safety1	Monitor Only		5.264232	3.201104	5.264232	5.300045	5.295958	5.285632	5.286413	5.312378	3.201104
Stress1	Maximize	N/mm^2	117.86	193.82	117.86	117.06	117.15	117.38	117.36	116.79	193.82

The stress, displacement and FOS color maps for the new A-arm design can be seen in___. Again, the deformations in the color maps have been exaggerated by Solidworks.

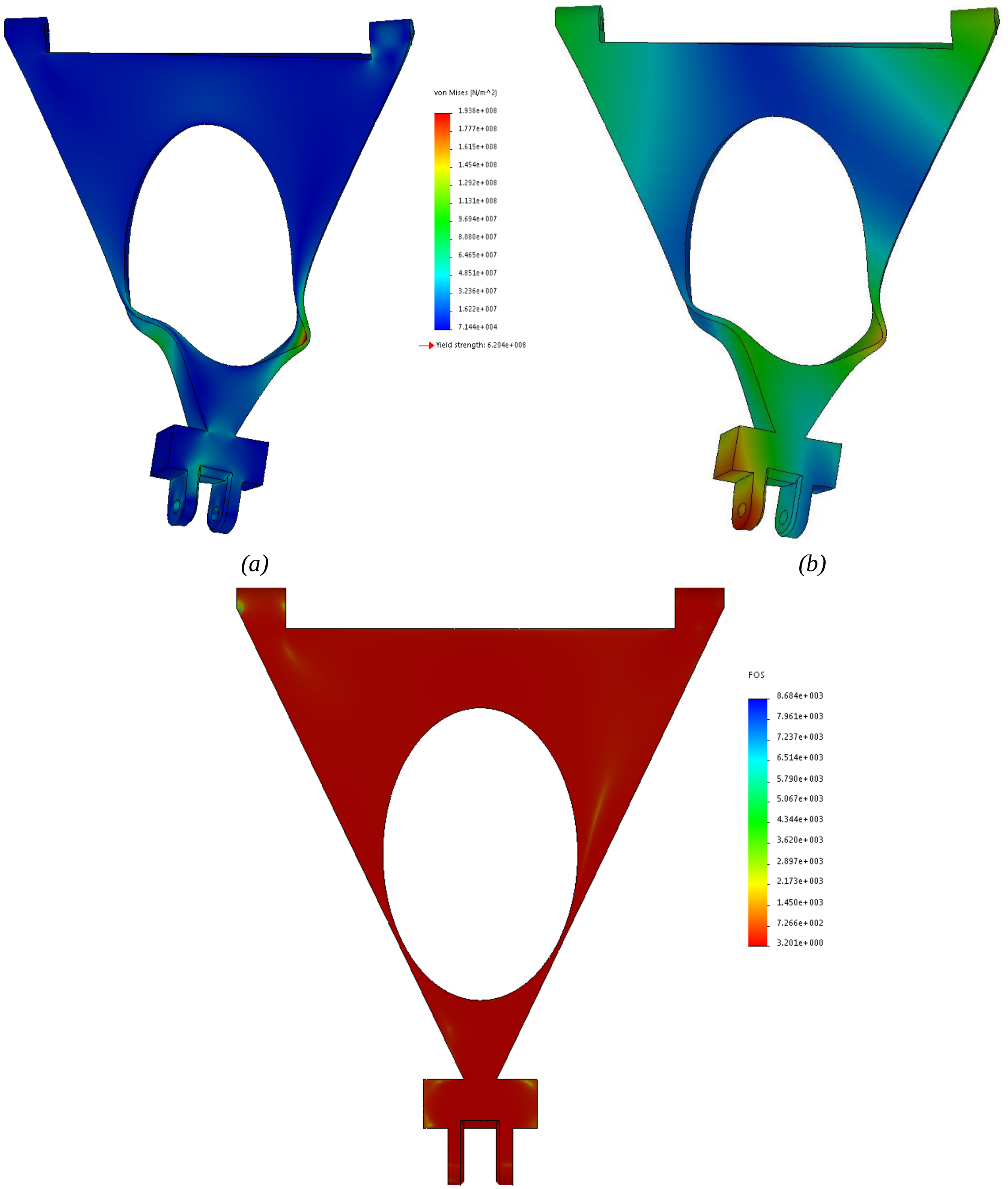


Figure 10 (a) stress, (b) displacement and (c) FOS color maps for the new A-arm design

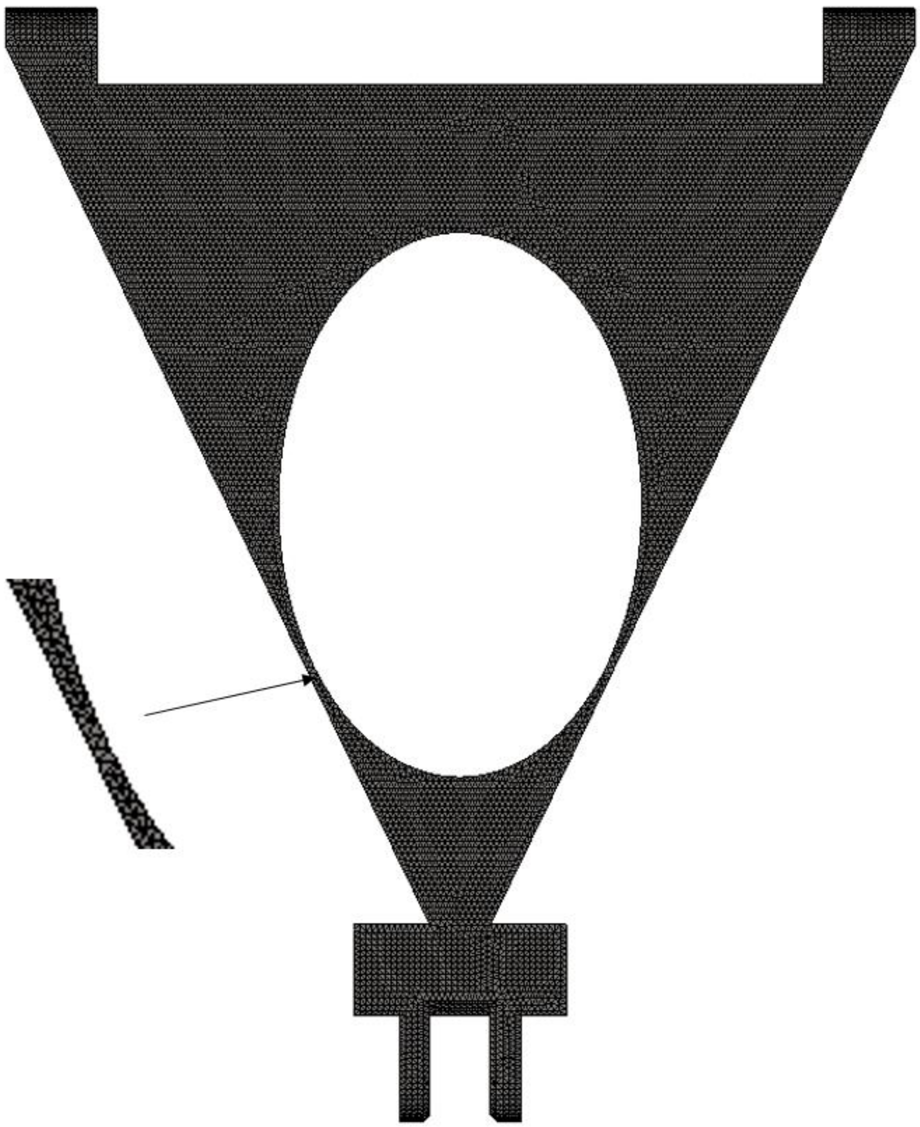


Figure 11 Mesh of 2 mm element size for the new A-arm design

3.4 Motion Study of New A-arm Design

Re-running the motion study resulted in a change in the reaction forces at the pin connections, as shown in *Figure 12*, *Figure 13*, *Figure 14* and *Figure 15* below. The maximum force is now 51.6 kN and occurs at 0.48 s.

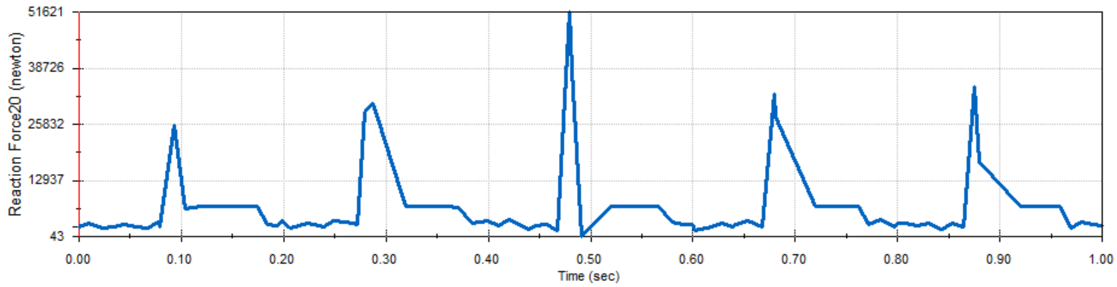


Figure 12 Reaction force magnitude for the pin connections of the new A-arm design

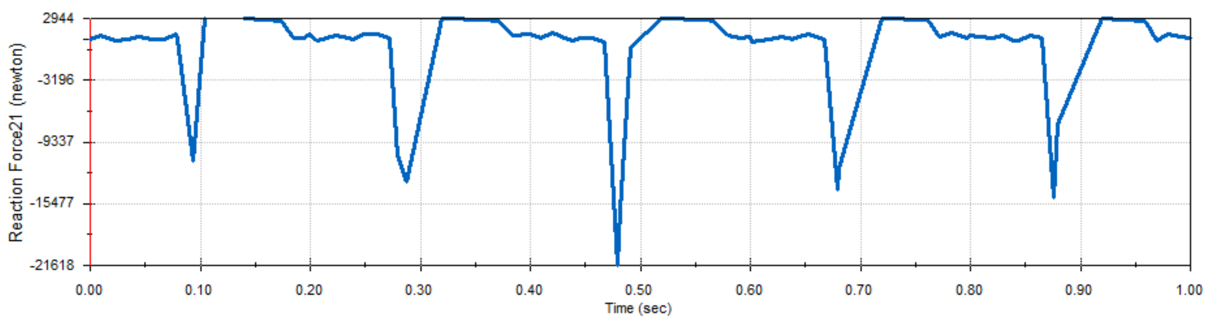


Figure 13 Reaction force x component for the pin connections of the new A-arm design

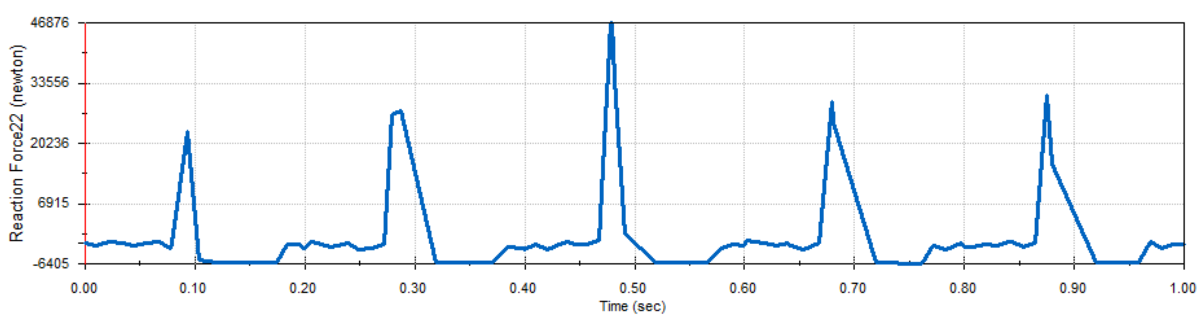


Figure 14 Reaction force y component for the pin connections of the new A-arm design

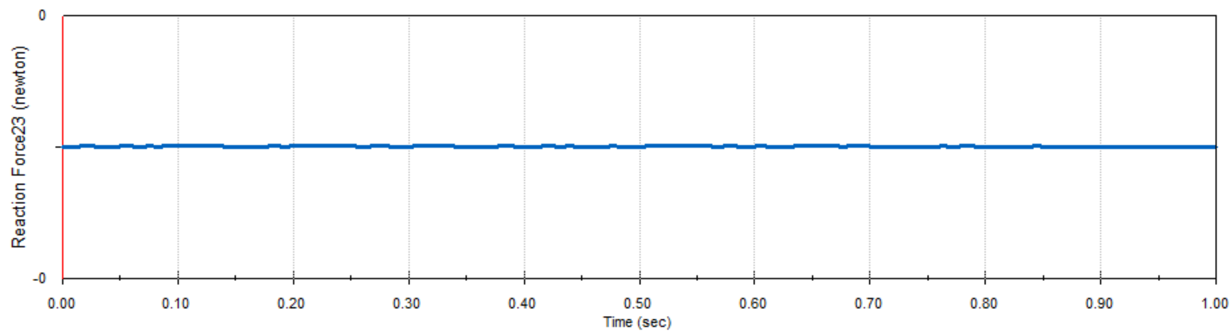


Figure 15 Reaction force z component for the pin connections of the new A-arm design

3.5 Cost Analysis of New A-arm Design

The part is to be built from the 300 mm x 25.4 mm x 367.7 mm steel block. 8 A-arms are required for a vehicle suspension. Typically, alloy steels can cost 3-3.5 USD/kg. therefore, the standard price of plain carbon steel in Solidworks was used, which 3.11 USD/kg. The chamfers in the design were not accessible for milling, so no cost was assigned to it. The machining setup cost 18.75 USD. The total cost to drill the holes was 0.50 USD and the milling operations cost 154.80 USD. Overall, to machine 8 A-arms would cost 242.02 USD per A-arm, with 28% going to the material cost and 72% going to the cost of manufacturing. No custom operations were required for this design.

4 Discussion

From the initial motion study, there is no unusual motion in the z component, which should be expected. The reaction force plot shows an increasing trend in reaction force and then decreasing. This is likely due to the to the damper and spring in the shock absorber after each impulse from the wheel hitting the bump. The maximum force occurs at ___

From the initial stress, displacement and FOS maps show that the requirements for yielding and deflection are met. The maximum stress occurs near the pin connections of the A-arm. Though no yielding occurs during the motion, the geometry of the design can still be changed to increase the maximum stress that occurs before yielding. The only geometry that can change is the central hole in the plate, since the geometry of the pin connections must remain the same to fit on the suspension.

The design table in ___ shows the different configurations of the hole in the A-arm. After running the design study, the effects changing the dimensions of the hole show that increasing the vertical and horizontal radii were small until the thickness near the edge of the elliptic hole was ___. Also, in the optimal design, notice that the FOS was also minimized because of maximizing the stress. Also, from the design study table, the FOS also reduces when the stress increases between subsequent scenarios. Additionally, the new design reduces the reaction force at the pin connections. Therefore, this design optimizes the maximum stress and reduces the pin reaction forces at the expense of increasing the displacement in the A-arm, particularly around the hole discontinuity.

The stress, displacement and FOS color maps show that the stress and displacement around the hole increased. Although the geometry of the design has been optimized for stress, this does not necessarily make this a good design since the displacements in the A-arm have now increased. Therefore, different geometries and configurations would need to be considered to reduce the displacement around the hole.

The cost analysis shows that the cost of manufacturing is greater than the cost of the raw materials. To save money, the raw material can be sourced by a local store like McMaster Carr to reduce the total cost by 28%. Adding smooth surface finishes to the pin connection areas also raised the price from initially being 218.46 USD to 242.02 USD. This is because changing the finishing increases the machining time. The total machining time to make one A-arm is 5 hours 48 minutes, costing about1936 USD. Since the A-arm optimizes the stress and reduces the reaction forces at the pin connections, the displacement in the A-arm rises. Therefore, different designs would need to be explored first before deciding to machine this design.

During the simulation of the A-arm, Solidworks ran into issues that may have affected the analysis of the design. the motion loads were causing large displacements to Solidworks. To get the simulation to run, the displacements were approximated as small. This may have affected the obtained stresses and displacements in the design and resulted in the exaggerated deformations shown in the color map.

5 Conclusion

The purpose of this assignment was to design a new A-arm made of alloy steel for a 2017 Dodge Charger SE. The goals of the design were to maximize the stress and keep the displacements in the design less than 1 mm. The new design proposed in ___ maximizes the stress and reduces the reaction force at the pin connections. However, the displacements in the A-arm increase, and larger displacements now occur around the hole discontinuity. To build 8 A-arms, the cost of each would be 242.02 USD. 28% of this cost goes to materials and 72% to manufacturing. Note that the chamfers in the design were unable to be machined because the obstruction caused by their geometries. Although the design goals were met, different designs should be explored because of the displacements that occur around the hole.

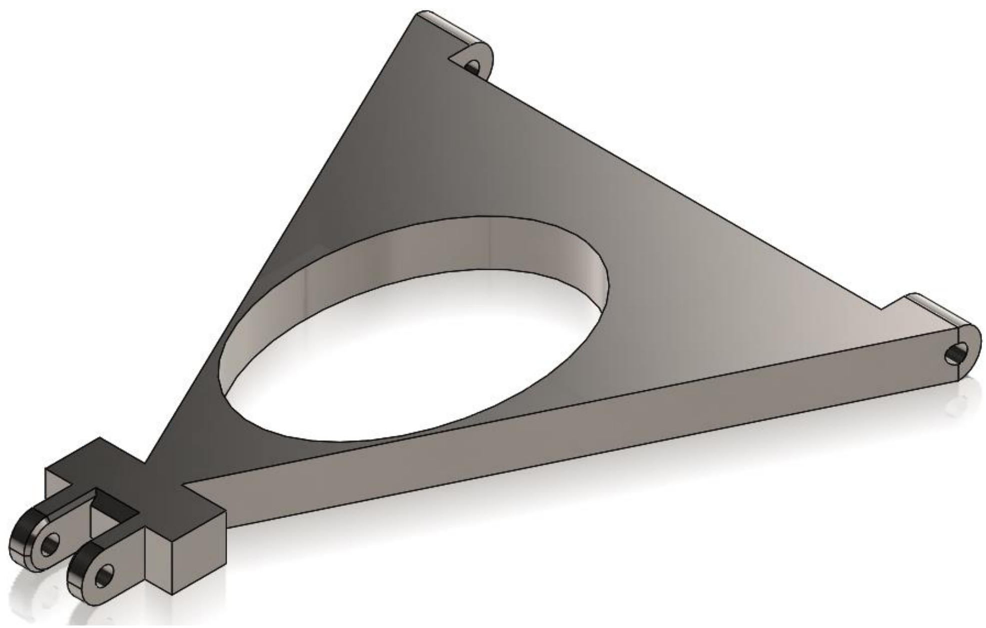


Figure 16 Final A-arm design with a 90 mm vertical radius and 60 mm horizontal radius

6 Appendix

6.1 A: MATLAB Code for Determining the Effective Damping Coefficient

```
%% Determination of Car Suspensions System Damping Coefficient
% CLASS: MecE 468 Numerical Simulation in Mechanical Engineering Design
% Author: Garrett Melenka
%
% Date: 2015-10-05
%
% Source material: Dr. Bob Koch car_susp_tf.m
%
% Textbook1: Feedback Control of Dynamic Systems (2006) Example 2.2
%
% Textbook2: Modeling and Analysis of Dynamic Systems (2010) Example 5.5

%% Clear the workspace and figures
clear
close all
%% Size and position figures
figure(1), set(gcf,'color','w');
figure(2), set(gcf,'color','w');
sz_x = 600; sz_y = 600; % size of the figure
y_pos = 200; x_pos =75;  % position on the screen
set(1,'pos', [x_pos, y_pos, sz_x, sz_y]);
set(2,'pos', [x_pos+sz_x, y_pos, sz_x, sz_y]);
%% Free Body Diagram of a Two Degree of Freedom Car Model
figure(1);
img=imread('TwoDOFCarModel.jpg');
img2 = imresize(img,0.5);
imshow(img2);
title('Quarter Car Model Free-body Diagram');
%% Setup Constants
m1=4.85; %Mass of wheel (kg)
m2=500; %mass of 1/4 car (kg)
kw=500000; %stiffness of wheel (N/m)
ks=131000; %stiffness of suspension (N/m)

%% Critical Damping
% (a) Critical Damping $$b =2m\omega$
%
% (b) Over damped $${\frac{b}{2m}}^2>\omega^2$
%
% (c) Under damped $${\frac{b}{2m}}^2<\omega^2$
%
% Defining the damping Ratio
% Damping Ratio
% $$\zeta = \frac{b}{2*\sqrt{m_2*k_s}}$$

figure(2);
% get initial b to give damping - uncomment next line once
% try three cases for damping ratio: under damped, slight under damping,
% critically damped
zeta = [0.5 0.7 1]
```

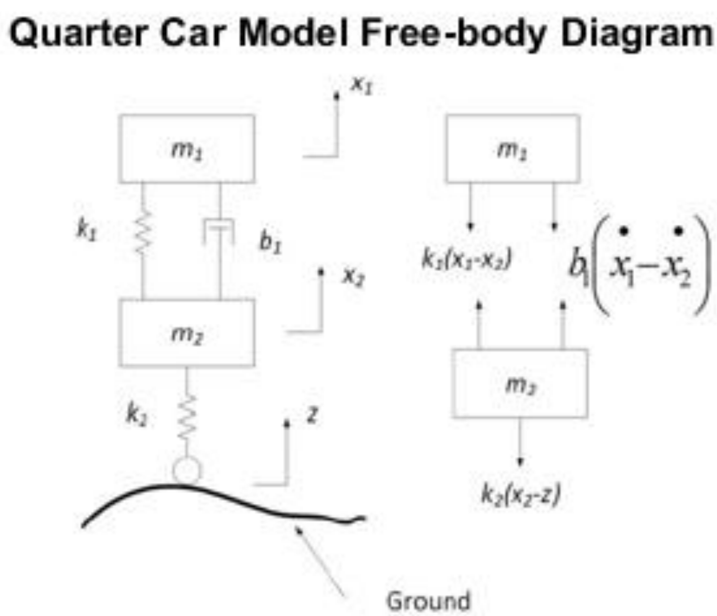
```
%% Transfer Function for Two DOF System
%The transfer function is determined by first solving the equations of
%motion for the Two DOF system.  Equations of motion are found based on the
%Free-body diagram.  After equations of motion have been determined Laplace
%transforms of the differential equation are used.  Finally, the equations
%are rearranged in order to yield the transfer function (tf)
for i = 1:3
    b=2*zeta(i)*sqrt(ks*m2);

m12=m1*m2;
c1=kw*b/m12;
b1=c1*ks/b;
a1=b/m1+b/m2;
a2=ks/m1+ks/m2+kw/m1;
a3=c1;
a4=kw*ks/m12;
% transfer fuction
num=c1*[1 b1];
den = [1 a1 a2 a3 a4];
sys1=tf(num,den);
hold on;
step(sys1) %apply a steup inpulse to simulate going over a bump
str1=num2str(b);
%title(['Damping Coeffecient b (Ns/m) = ' str1])
end
legend('Underdamped \zeta = 0.5', 'Slightly Underdamped \zeta = 0.7',
'Critically Damped \zeta = 1.0', 'location', 'SouthEast')


%% Determine Damping Coefficient

b=2*zeta*sqrt(ks*m2);
%Underdamped
b(1)
%Slight Underdamping
b(2)
%Cricitally damped
b(3)
```

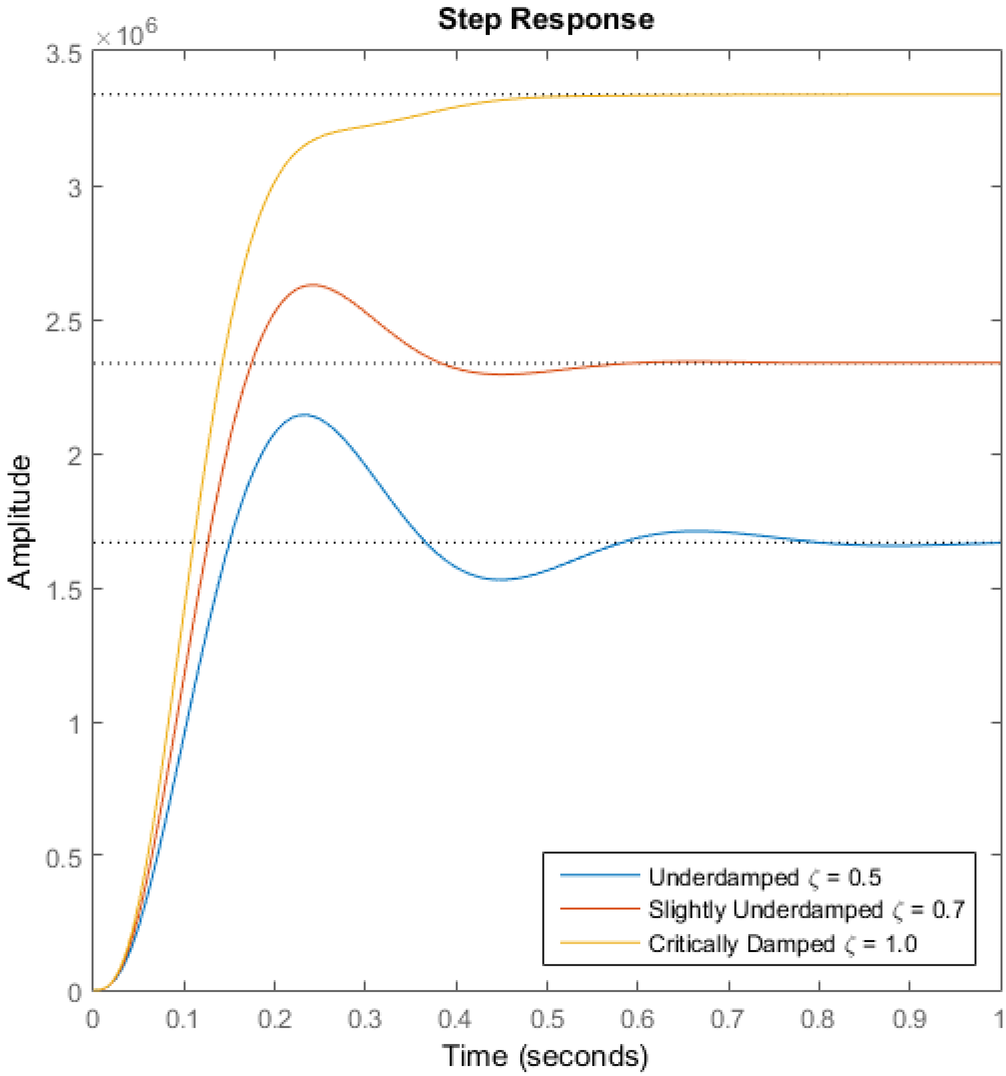
6.2 B: Vehicle Specs and Schematic for Vehicle Suspension System

Below is the vehicle specs for a Dodge Charger, schematic of the Dodge Charger travelling on a bumpy road its damping step response plot.

Table B 1 2017 Dodge Charger SE Vehicle Specifications

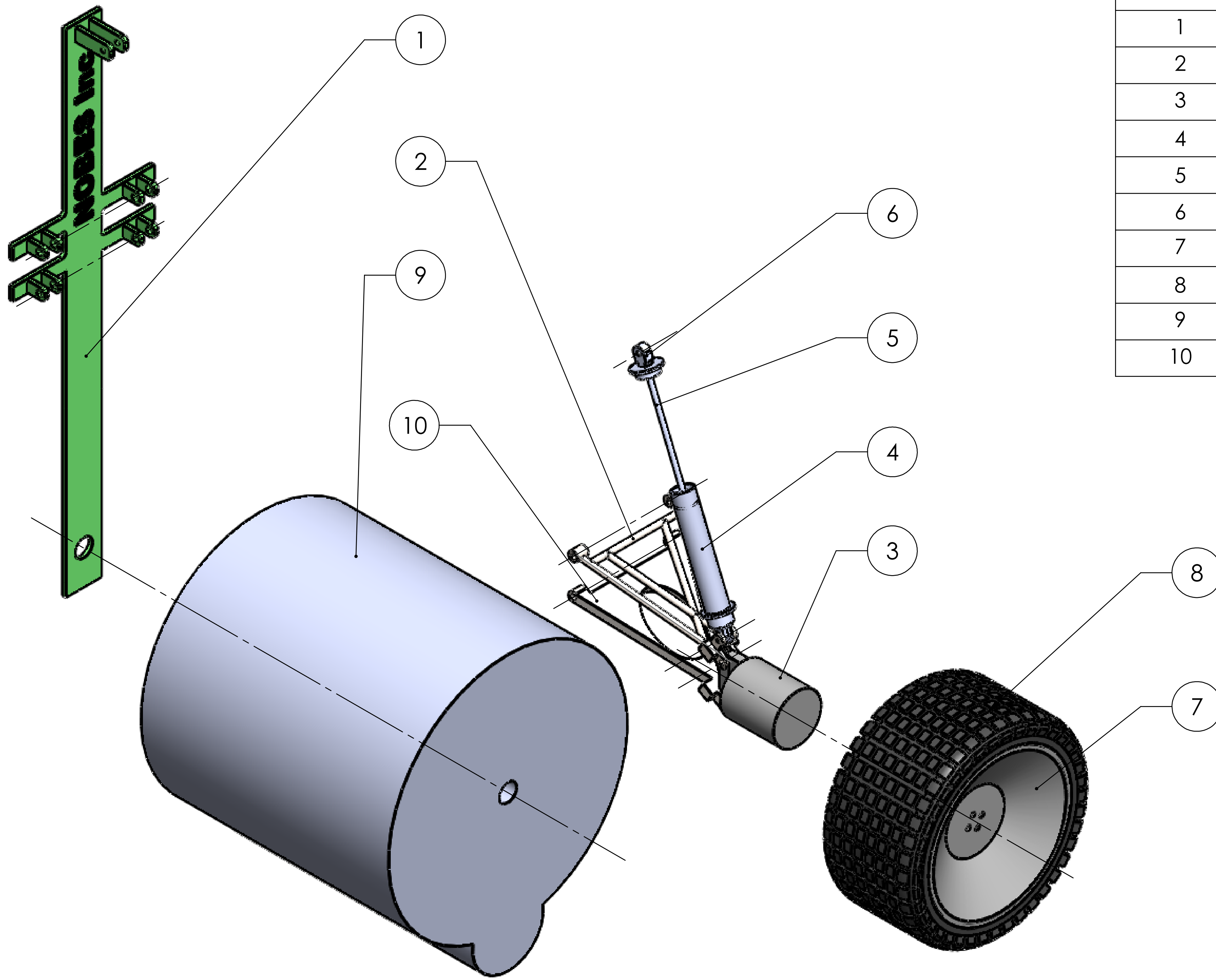
Wheel Mass	4.85 kg
Vehicle Weight	4400 lbf
Maximum Torque	260 lb-ft
Maximum Horsepower	292 hp
Engine Type	3.6 L Pentastar VVT V6 Engine
Spring Stiffness (Part # 2000.375.0750S from Eibach Motorsport Catalog)	131 N/mm




Step Response
$\times 10^6$
3.5
3
2.5
2
1.5
1
0.5
0
Amplitude
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
Time (seconds)
Underdamped ζ = 0.5
Slightly Underdamped ζ = 0.7
Critically Damped ζ = 1.0

6.3 C: Engineering Drawings

Following are the engineering drawings for Assignment #6



ITEM NO.	PART NUMBER	Material	Default/ QTY.
1	Suspension Mount	Pure Gold	1
2	Top_A_Arm		1
3	Simple_Hub		1
4	Shocker_Tube		1
5	Shocker_Ram		1
6	Shocker_End		1
7	Simple_Rim		1
8	Fat_Tire		1
9	Simple_Cam		1
10	tokeke_A_arm	Alloy Steel	1

MecE 468

Instructor:
Dr DS Nobes
Fall 2017

Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

MATERIAL:
309149.64

FILE NAME:
MecE468_Suspension_New2

DRAWN BY:
Toya J. Okeke

Student # 1432145
Lab Day Friday
SM By T.J. Okeke
TA Initials J.M.

October 24, 2017 4:03:41 AM
October 3, 2014 11:10:59 AM

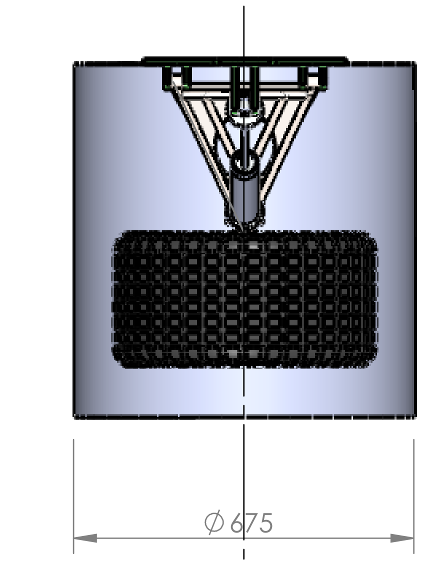
The Department of Mechanical Engineering
UNIVERSITY OF ALBERTA

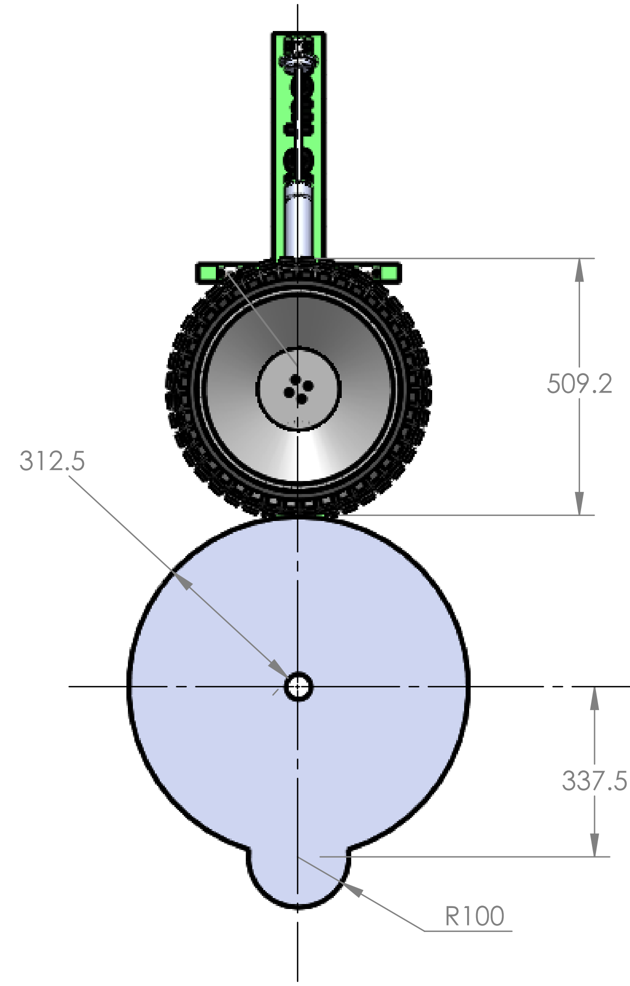
TITLE:
Exploded View of Vehicle Suspension

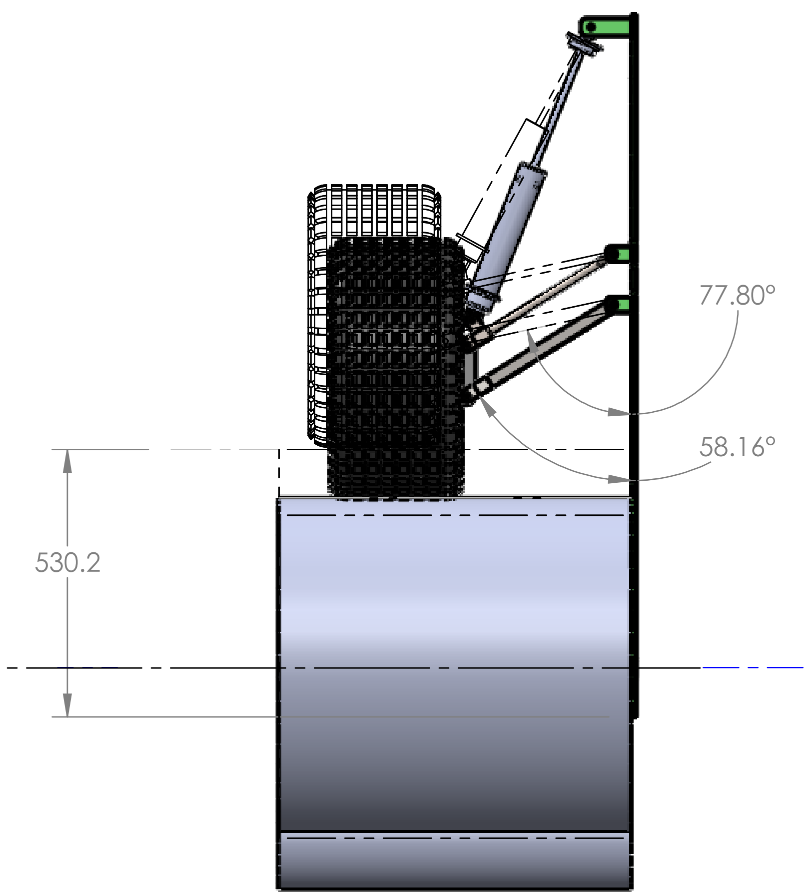
SIZE **B** | Assignment Number 6 | REV B

SCALE: 1:10 | Mass: 309149.64 | SHEET 1 OF 3

ITEM NO.	PART NUMBER	Material	Default/ QTY.
1	Suspension Mount	Pure Gold	1
2	Top_A_Arm		1
3	Simple_Hub		1
4	Shocker_Tube		1
5	Shocker_Ram		1
6	Shocker_End		1
7	Simple_Rim		1
8	Fat_Tire		1
9	Simple_Cam		1
10	tokeke_A_arm	Alloy Steel	1







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Fall 2017

Comments:

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
X.X = ± 0.1
X.XX = ± 0.025

SURFACE FINISH µm 0.6

DO NOT SCALE DRAWING

DRAWN BY:
Toya J. Okeke

Student # 1432145
Lab Day Friday
SM By T.J. Okeke
TA Initials J.M.

October 24, 2017 4:03:41 AM
October 3, 2014 11:10:59 AM

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UNIVERSITY OF ALBERTA

TITLE:
Orthogonal Views of Vehicle Suspension

MATERIAL:
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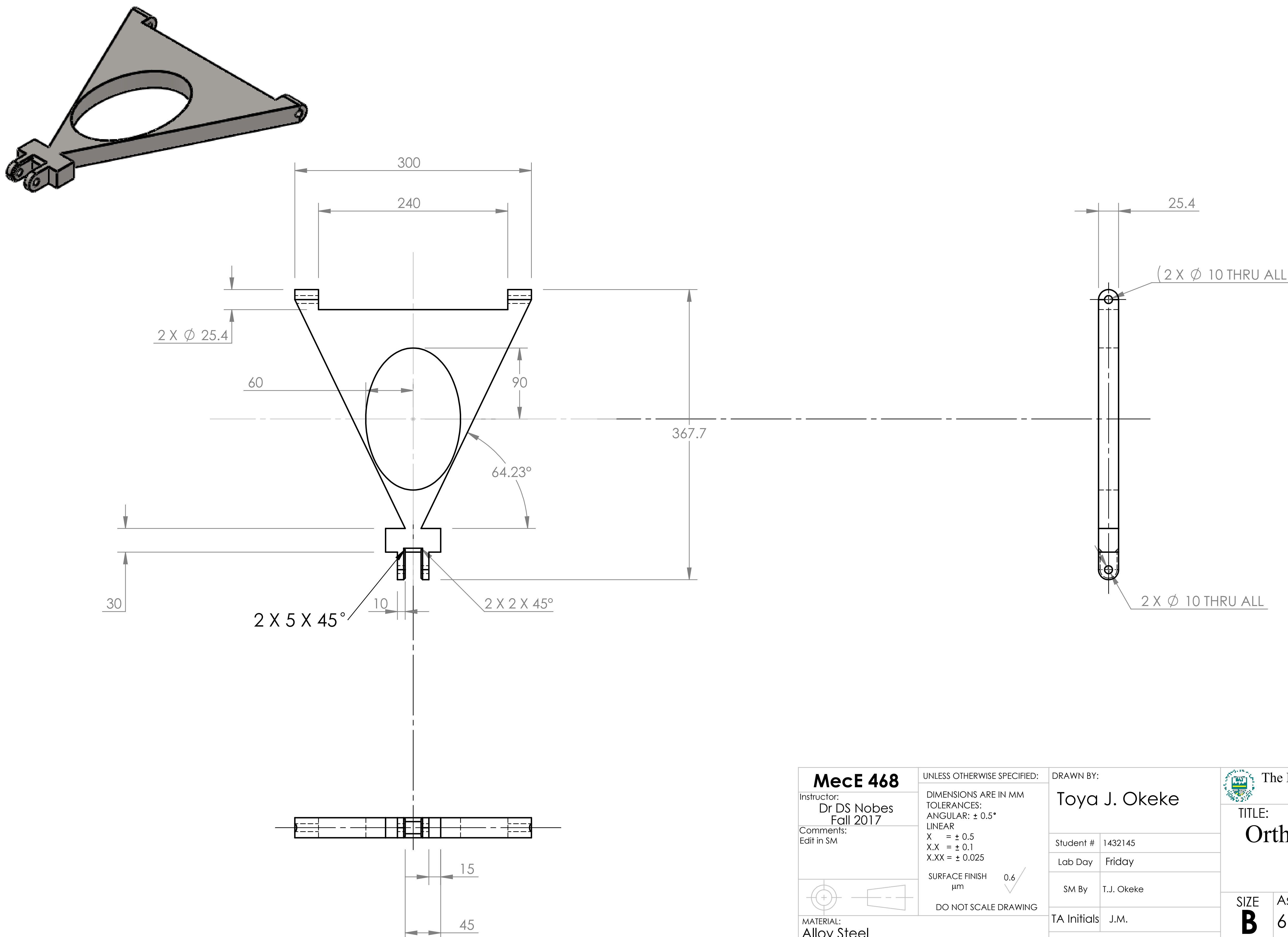

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Fall 2017
Comments:
Edit in SM

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MM
TOLERANCES:
ANGULAR: ± 0.5°
LINEAR
X = ± 0.5
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X.XX = ± 0.025
SURFACE FINISH µm 0.6
DO NOT SCALE DRAWING

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FILE NAME:
tokeke_A_arm

DRAWN BY:
Toya J. Okeke

Student #	1432145
Lab Day	Friday
SM By	T.J. Okeke
TA Initials	J.M.

October 24, 2017 4:03:47 AM
October 20, 2017 4:40:53 PM

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TITLE:
Orthogonal Views of A-arm Design

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