

SAE Mini Baja: Suspension and Steering

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Team 19

Engineering Analysis

Document

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Table of Contents:

1. Abstract.....	3
2. Introduction.....	3
3. Front Suspension.....	3
a. Control Arm Geometry	3
b. Shock Placement.....	5
4. Rear Suspension	
a. Table 1.....	7
b. 3- link Trailing Arm Description and Selection.....	7
c. Drop Test.....	8
d. Table 2.....	9
e. Drop Test Deflection Analysis.....	10
f. Table 3.....	10
g. Rear Suspension Conclusion and Future Analysis.....	10
5. Steering.....	11
a. Analysis of Steering Components (Tie Rod).....	11
b. Analysis of Steering Components (Rack and Pinion Gear Stresses).....	14
c. Table 4.....	14
6. Conclusion.....	16
7. References.....	16
8. Appendices.....	17
a. Appendix Figure 1.....	17
b. Appendix Figure 2.....	17
c. Appendix Figure 3.....	18
d. Appendix Figure 4.....	18
e. Appendix Figure 5.....	19
f. Appendix Figure 6.....	19
g. Appendix Figure 7.....	19
h. Appendix Figure 8.....	20
i. Appendix Figure 9.....	20
j. Appendix Matlab code 1.....	21
k. Appendix Matlab code 2.....	24
l. Appendix Matlab code 3.....	25

1. Abstract

The suspension and steering subsystems of an off-road racing vehicle are the most important components. With bad suspension geometry, the vehicle may not be able to traverse the rugged environment associated with off-road racing and recreation due to lack of clearance or inadequate suspension travel. Correct material selection must be made in order to assure that the suspension members will survive fast and slow compression impacts, large drops, static weight, and withstand multiple cycles of fatigue loading. The rack and pinion steering system must be able to maneuver around any type of obstacle width and turn radius as well as handle repeated input from the driver throughout its lifetime. These subsystems were analyzed in a variety of geometries, materials, and situations. The analysis results will show that the materials and designs chosen will survive the abuse of off-road racing. Further testing and analysis will be completed to refine the team's selected designs for both front and rear suspension as well as steering.

2. Introduction

In engineering the ability to analyze a design before production a very powerful skill. These processes include, but are not limited to, theoretical analysis of static and dynamic systems given specific material properties and geometries and computer aided design. For the steering and suspension subsystems of the mini Baja vehicle, the theoretical approach will be conducted on the desired materials and geometries until a final design is chosen by the entire SAE Mini Baja team. Included in this report is the analysis of a spur gear rack and pinion steering system and two independent suspension designs.

3. Front Suspension

a. Control Arm Geometry

To design a front suspension the first thing to determine is track width. From the frame team the width of the chassis is 20" with a desired track width of about 60" with two 8" tires. This gives us an A-arm width of about 12". In order to have minimal steering effort and steering effects from bumps and cornering forces a scrub radius of zero is desired. To prevent the tires from flopping excessively on fast turns a kingpin angle of between 5-10 degrees is generally used. Balancing these dimensions and using a tire size of 205-80-12 we designed the following design.

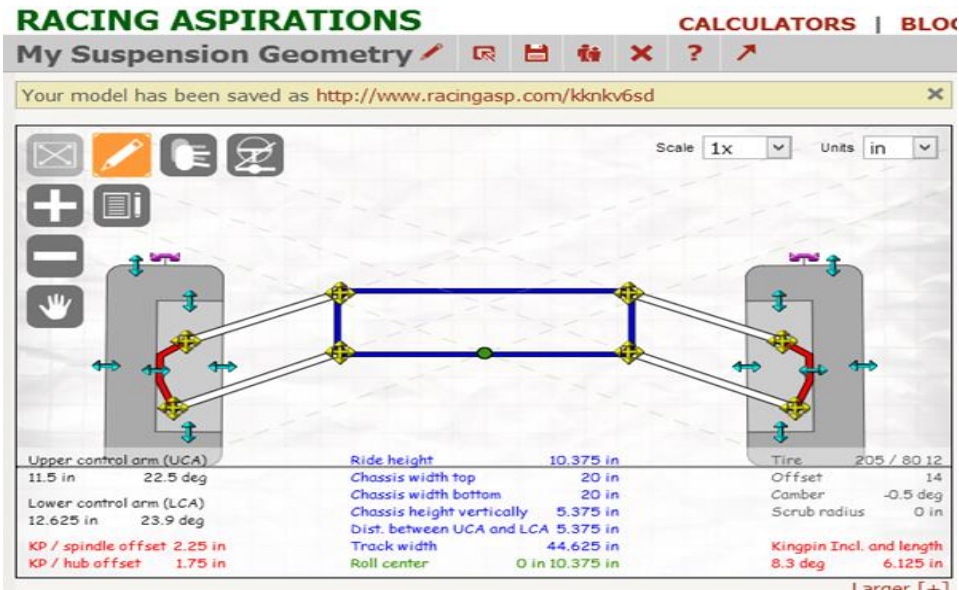


Figure 1: Front SLA A-arm geometry

This gives us a scrub radius of 0, a kingpin angle of 8.3 and a track width of 44.625. The control arms are a short long arm arrangement with the lower arm measuring 12.625” and the upper arm measuring 11.5”. This gives us a static camber of -.5. We designed the suspension to have a traditional 60% 40% compression to drop ratio. Below is the simulation at full droop and compression.

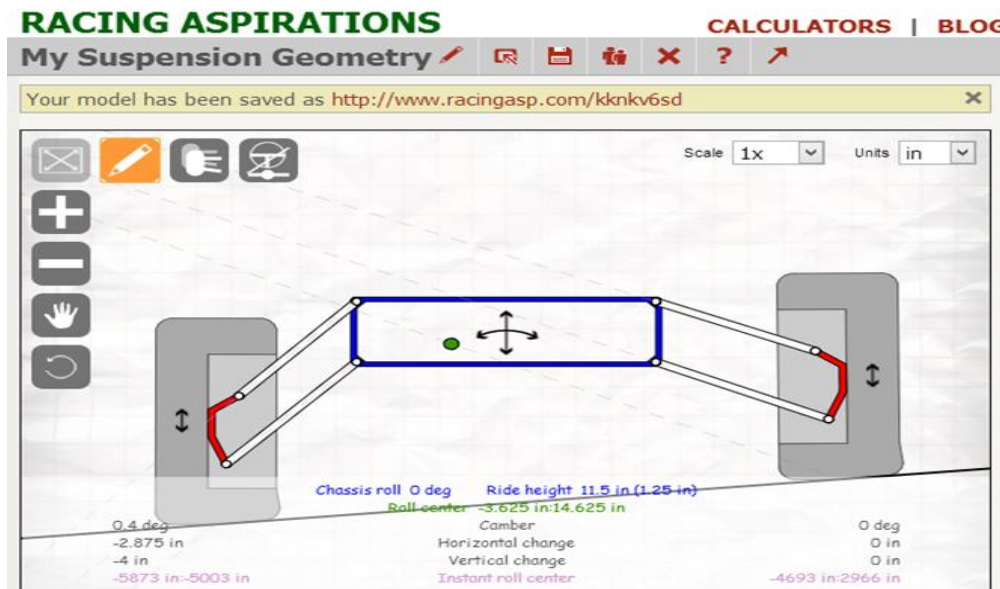


Figure 2: Full Droop

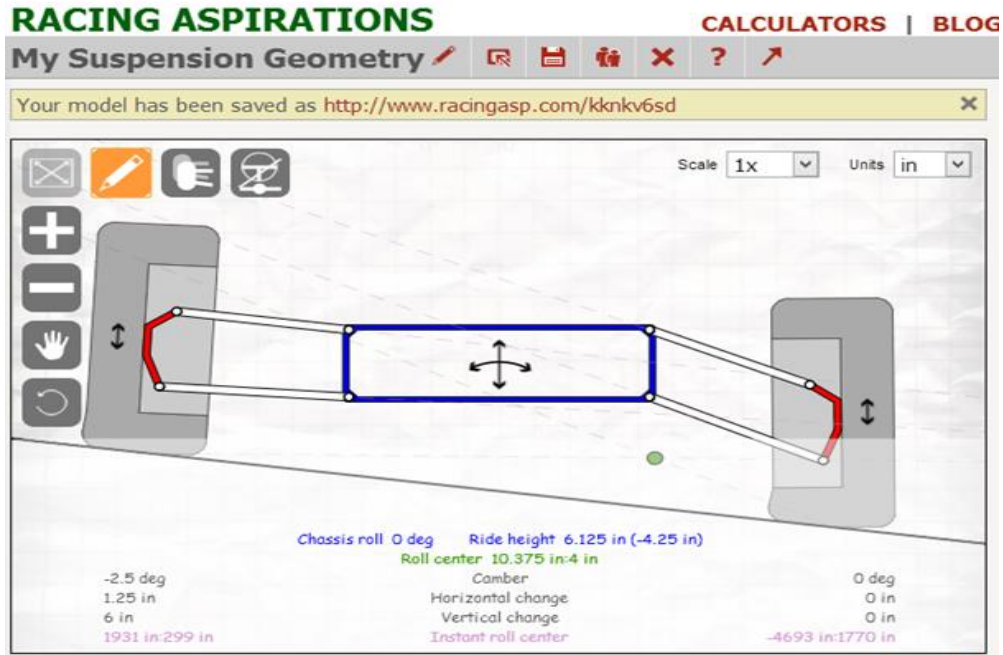


Figure 3: Full Compression

At full droop of 4” we have a camber gain of .4 degrees and at full compression of 6” we have a camber loss of 2.5 degrees

b. Shock Placement

Shock placement heavily influences how strong a shock has to be to absorb the impact of the vehicle with the ground, and given the geometry of the front suspension, it would be very difficult to get a shock mounted completely vertically. The shock mounting angle off vertical is critical for determining how strong the shock must be, for example, if a shock is mounted at 45* off vertical the shock must be twice as strong to compensate.

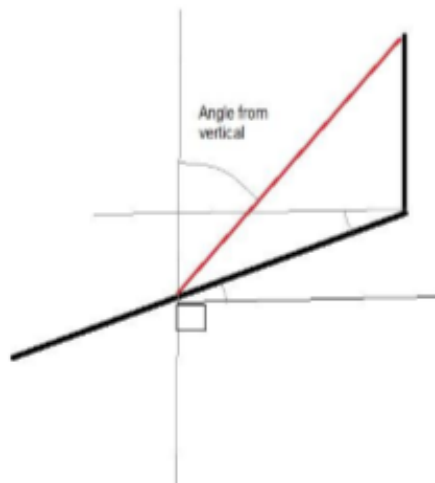


Figure 4: Simple Shock Angle Drawing

The lower control arm sits at ride height at 24° down from the horizontal axis, meaning that the shock starts slightly closer to vertical. To determine the angle of the resting shock, we used some simple triangle geometry (law of cosines) to find a shock length for a given position on the control arm (between 8 inches and 10 inches from the control arm pivot), the shock lengths were found to be between 10.7 inches and 12.5 inches. Then, once shock length is determined, we used more triangle geometry (law of sines) to find the shock angle with respect to the lower control arm. At this point we know the angle between the shock and the vertical axis to be between 36° and 41° depending on shock length.

In order to keep the shock from having to be over built and to keep costs and weight down, we picked a good compromise between shock length and angle. We picked a shock length of 11.5 inches which gives an angle of 39°, meaning that for the desired shock strength we need to have a shock that is about 1.5 times stronger than the desired strength.

3. Rear Suspension

The rear suspension of the mini Baja vehicle must be able to bear the weight of the frame, engine, transmission, driver, and other carried loads during a race. The objects listed above will be referred to as sprung weight from this point on. The following independent rear suspension designs were chosen and analyzed for the mini Baja vehicle:

1. Double a-arm
2. 3-link trailing arms

The suspension designs were analyzed for proper geometry, shock placement, material strength, and survivability from fatigue/impact. Double a-arms were chosen for the front suspension design, as well as one option for the rear suspension; the second rear suspension design is a form of a trailing arm suspension called a three link. Analysis of the suspension systems began with material selection and static analysis. The SAE Baja rulebook specifies AISI 1018 steel with a 1in outside diameter and a 0.120in wall thickness or a material with equivalent bending strength, bending stiffness, and a minimum wall thickness of 0.062in for frame and component construction. Using these requirements multiple AISI steel configurations were evaluated from the material property tables [1]. Table 1 shows the materials evaluated for the suspension members.

Table 1: Suspension Material

Material	Properties					
	Sy (ksi)	Ts (ksi)	E (ksi)	ρ (lb/in ³)	G (ksi)	ν
AISI 1018 (CD)	54	64	29000	0.284	11600	0.292
AISI 4130 (normalized)	63	97	29700	0.284	11600	0.292

The team decided to use normalized AISI 4130 for the several reasons. AISI 4130 chromoly steel was chosen for its equivalent strength and material properties that were required by the SAE rules and regulation as well as the materials ability to maintain its physical properties after being welded, which is key to maintaining the needed strength during the demands of off-road racing. AISI 4130 steel is a very strong and rigid material that is widely used in the off-road industry and through research seems to be the norm for the SAE Baja competition.

a. 3-link Trailing Arm Description and Selection

This design was chosen based on the customer and team's need for a competitive off-road vehicle. The 3-link trailing arm design offers maximum suspension travel in a light weight and simple package. Compared to the double a-arm design, trailing arms typically require fewer components for proper operation and are much easier to manufacture. The analyzed design is shown in Figure 1.

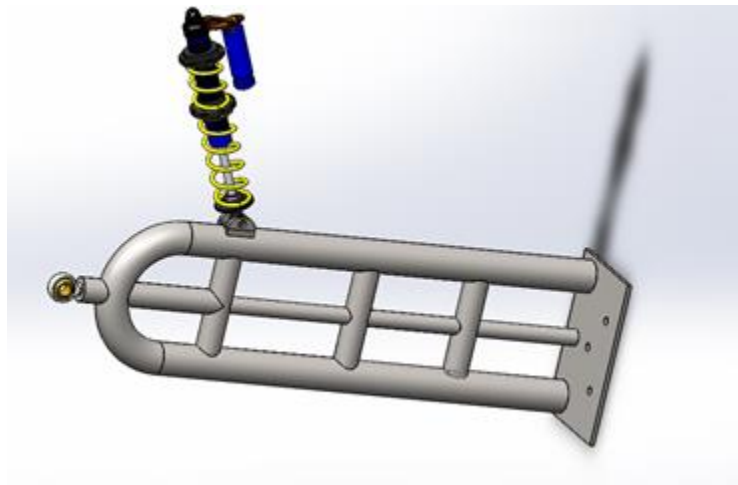


Figure 5: Non-finalized Trailing Arm Design

b. Drop Test

The following empirical analysis was performed on the 3-link member and individual tube sizes of for 1.5 in OD 0.065, AISI 4130. For this analysis the following assumptions were made:

1. Vehicle weight of 500 lb
2. Drop height of 6 feet
3. Acceleration of gravity is equal to 32.2 ft/s²
4. Impulse time was estimated at 1.5 s

First the forces on the member were calculated by determining the suspension geometry, i.e. shock placement, lever arm distance, effort arm distance, leverage ratio, displacement ratio, and others. These values were calculated using the Equations 1-7 [2] listed below. The values obtained from these equations were used to calculate stresses in the trailing arm member at full compression from a drop height of 6 ft.

Equation 1:

$$DR = d1/d2$$

Equation 2:

$$ACF = \cos(\alpha)$$

Equation 3:

$$IR = ACF*DR$$

Equation 4:

$$WT = ST/IR$$

Equation 5:

$$FW = IR*FS$$

Equation 6:

$$FS = k*ST$$

Equation 7:

$$WR = k*IR^2$$

Where

- DR=displacement ratio
- d1=distance between the lower shock mount and the suspension pivot, [in.]
- d2 = distance between the wheel center and the suspension pivot, [in]
- ACF = angle correction factor
- α = angle of shock, [degrees]
- IR = installation ratio or leverage ratio
- k = spring rate, [lb/in]
- ST = shock travel, [in]
- FS = shock force, [lb_f]
- WR = wheel rate, [lb/in]
- WT = wheel travel, [in]
- FW = wheel force, [lb_f]

Table 2 shows the calculation results for Equations 1-7. Ideally, the displacement ratio should be as close to 1 as possible meaning the wheel rate and spring rate will be equivalent. The calculated value of DR is shown to be 0.86207:1 at a distance of 25 inches, or 3 inches away from the wheel center. These values can be changed to obtain the Baja team's desired rear suspension geometry.

Table 2: Calculated Suspension Factors

Factors	Chosen/Calculated Values	Units
α	25	(degrees)
Spring Travel	0-12	(in)
k	550	(lb/in)
d1	25	(in)
d2	28	(in)
IR	44.765	(ratio)
DR	0.86207	(ratio)
ACF	0.991	(degrees)
WR	408.740	(lb/in)

Using Microsoft Excel the shock forces and wheel forces were calculated throughout the desired 12 inches of shock travel, resulting in a range of increasing values. The impact force experienced by the Baja vehicle at a drop of 6 feet was calculated using Equation 8 [3] and compared to the shock force, wheel force, and shock travel to obtain the correct values for a 6 foot drop.

The results showed that the impact force experienced per-wheel for this particular geometry caused by six foot drop is approximately equivalent to the force required to compress the shock between 8 and 9 inches.

c. Drop Test Deflection Analysis

The deflection analysis of the trailing arm member used the information obtained from the drop test analysis and material properties of AISI 4130 steel. A few assumptions were made in order to simplify the calculation process for the forces acting on this member. Due to the shape of the trailing arm member and its added supports in the center, the member was assumed to act like a beam with a rectangular cross-sectional area. Using Case 12 from the shear, moment, and deflection of beams, Table A-9 [1], the deflection of the member was calculated for any distance x along the length of the beam due to a load input from the wheel and shock.

Table 3: Deflection and Factor of Safety

	Maximum Deflection (in)	Yield Factor of Safety
yAB	-7.978048453E-04	1.54
yBC	2.72	1.54

d. Rear Suspension Conclusion and Future Analysis

The results obtained from this analysis leave room for improvement in many areas such as weight reduction. Once the final trailing arm design is chosen by the entire Mini Baja team further fatigue, bending, and torsion analysis, as well as improving the factor of safety, will be done on the member. The geometry will be tweaked to mesh together with the frame and the geometry will be updated to reduce weight and improve performance. Furthermore FEA analysis will be done on the suspension member to refine the design before production sometime next semester.

4. Steering

a. Analysis of Steering Components (Tie Rod)

When analyzing the steering components of the SAE Baja vehicle, the most important factors are the yield strength of the tie rod and the steering column. Other factors that are dependent on these specific parts of the system are how the gearing and the torque from either the driver or the environment affect the system intermediately. By analyzing the steering system with different situations in MATLAB, it could be seen that the force put on the tie rod by the driver and the vehicles environment affects the need for a larger cross sectional area of the tie rod. The placement of the tie rod on the back of the hubs will also affect the steering system because the force exerted on the tie rod will be greater if the tie rod is connected farther from the point of rotation on the hubs of the vehicle.

Assumptions made are that the material that will be used for the tie rods will be Chromoly (AISI 4130 Steel) and that the steering column be rigid because the torsional yield

(shear stress on outer radius) of the steering column would be negligible compared to the yield stress in the tie rod. The reason for using Chromoly as a base material is because most pre-manufactured tie rods are made of this common steel so during post analysis, choosing a tie rod for the vehicle will come with ease. The properties of the chromoly used for the simulated analysis is given in Figure 4 and the results for the analysis are given in Figures 5,6, & 7. The results are provided by the MATLAB code SAE Baja Engineering Analysis (Rack and Pinion) and can be referred to in the Appendix. The results generally show that as the forces exerted by the tie rod and the environment increase so does the need of a greater radius on the tie rod.

A more detailed factor to take into account in analyzing the steering system is the gearing of the rack and pinion and it is directly correlated to the forces that are exerted on the tie rod and the steering column. The results and testing of this simulation will be explained and calculated in the following section.

Properties	Metric	Imperial
Tensile strength, ultimate	560 MPa	81200 psi
Tensile strength, yield	460 MPa	66700 psi
Modulus of elasticity	190-210 GPa	27557-30458 ksi
Bulk modulus (Typical for steel)	140 GPa	20300 ksi
Shear modulus (Typical for steel)	80 GPa	11600 ksi
Poissons ratio	0.27-0.30	0.27-0.30
Elongation at break (in 50 mm)	21.50%	21.50%
Reduction of area	59.6	59.60%
Hardness, Brinell	217	217
Hardness, Knoop (Converted from Brinell hardness)	240	240
Hardness, Rockwell B (Converted from Brinell hardness)	95	95
Hardness, Rockwell C (Converted from Brinell hardness, value below normal HRC range, for comparison purposes only.)	17	17
Hardness, Vickers (Converted from Brinell hardness)	228	228
Machinability (Annealed and cold drawn. Based on 100% machinability for AISI 1212 steel.)	70	70

Figure 6: Material Properties of Chromoly (Normalized AISI 4130 Steel) AZOM “The A to Z of Materials”

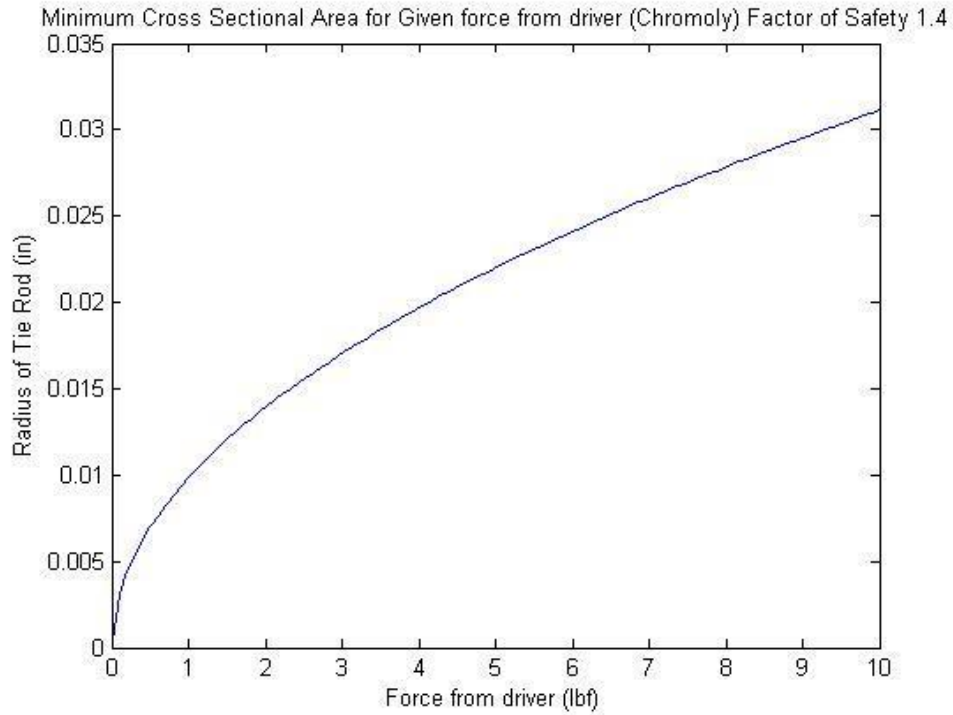


Figure 7: Driver Force vs. Tie Rod Radius

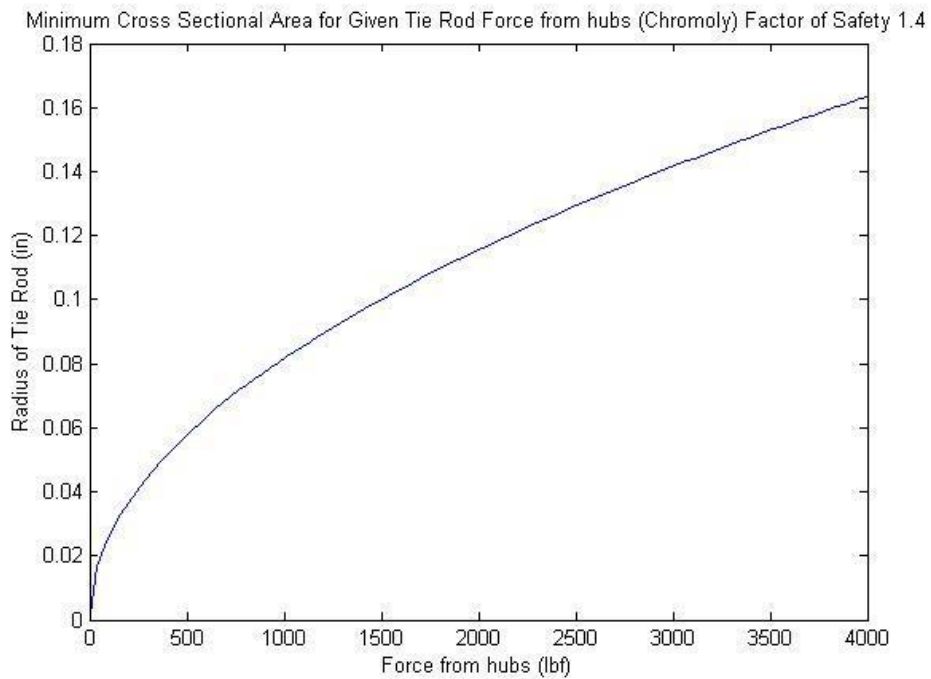


Figure 8: Hub (Environment) Force vs. Tie Rod Radius

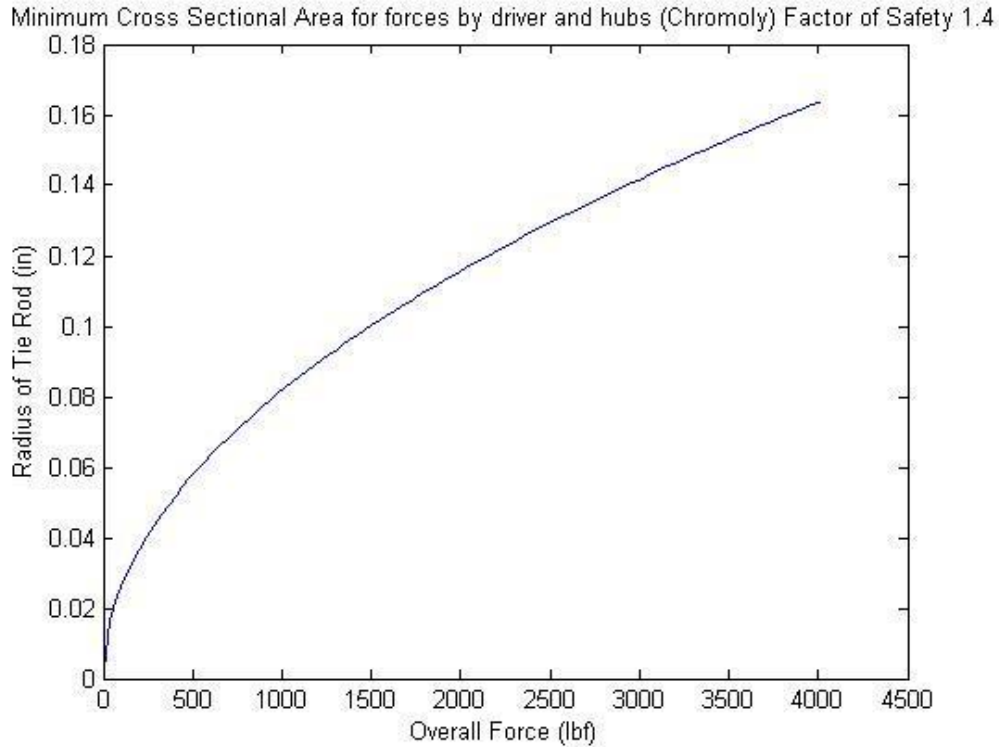


Figure 9: Total Force vs Tie Rod Radius

b. Analysis of Steering Components (Rack and Pinion Gear Stresses)

For the Steering System the team will have to calculate the forces acting on the individual components in order to determine material selection and “off shelf” parts. The Steering System will be a rack and pinion set up. For the pinion, the team was able to calculate the appropriate dimensions of the pinion and rack in order to choose an “off the shelf” rack and pinion steering system. Appendix A shows the Matlab code used to calculate the following dimensions. Table 1 and Figure 8 will show the dimensions and geometry of pinion/rack.

Table 4: Dimensions of Pinion and Rack

	teeth #	face width (in)	bending stress (kpsi)	radii for pitch circle (in)	radii for base circle (in)	Adden (in)	Deden (in)
pinion	20	0.74	0.04 - 3.9	0.787	.739	0.078	0.098
rack	20	0.74	-	inf	inf	0.078	0.098

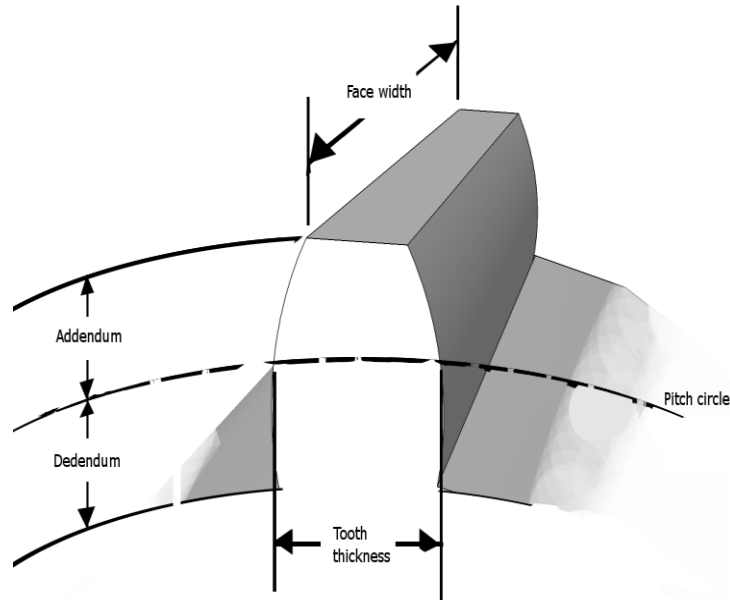


Figure 10: Pinion/Rack Geometry

The calculations for the pinion were made by assuming that the steering column (the shaft connected to the pinion) was a rigid member. This would then allow the team to set a range for the force acting on the steering wheel, which would allow the team to calculate a transmitted load on the pinion to the rack for calculating the bending stress on the pinion. For the rack, most of the dimensions will be the same with the exception of the bending stress and teeth on the rack. These parameters depend on availability of “off shelf” rack/pinion systems. Although the team was able to calculate a teeth number of 20 for the rack, given that the pinion will turn a maximum of 180 degrees. As for the face width, Addendum and Dedendum, will be the same for the rack as well. As for the rack’s pitch and base radii, they will be infinite since rack is non-circular. As for the bending stress on the rack, that depends on “off shelf” availability given that different manufacturers will make thicker (or thinner) racks or add rims to the side of the racks to reduce material use.

5. Conclusion:

By analyzing steering and suspension systems through the use of Statics/Dynamics, Mechanics of Materials and Machine design, the ideal geometries and materials of the steering and suspension components could be narrowed down.

With the suspension A-arms many situations were analyzed with the use of the online racing simulator, “Racing Aspirations”. It was concluded that the ideal mount angle for the front shocks are 45 degrees and the A arm will be at an angle 24 degrees from the horizontal. For the rear suspension, the dimensions and geometry are listed for many different situations in the Figures listed in the rear suspension section.

For the steering it is concluded that the system that will be used is a straight tooth rack and pinion and not a helical tooth rack and pinion. The pinion gear will have a 20 teeth in the simulation and will provide enough torque to move the wheels in the right direction and put ease on the driver. The tie rod was analyzed in terms of stresses put on the component by the wheels and by the driver. The correlation of the minimal area needed to sustain the forces are linearly related to the increasing forces put on the tie rods, which can be seen in Figures 5,6 & 7.

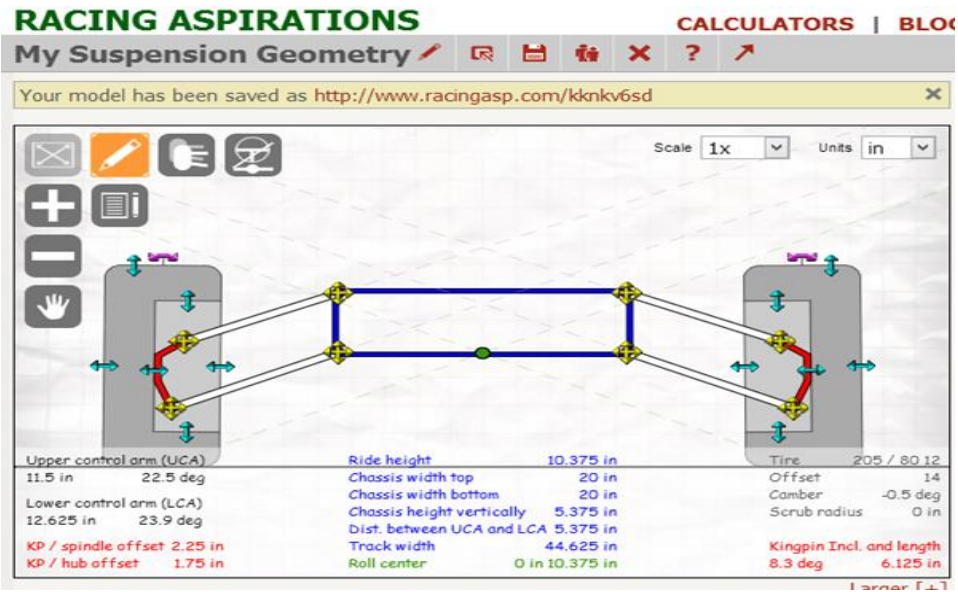
For both components, the material that is going to be used for analysis and fabrication is Chromoly (AISI 4130 Steel). This steel is widely available and is used in many off-road applications. It is light, strong and many off-road components are readily manufactured out of this material.

6. References:

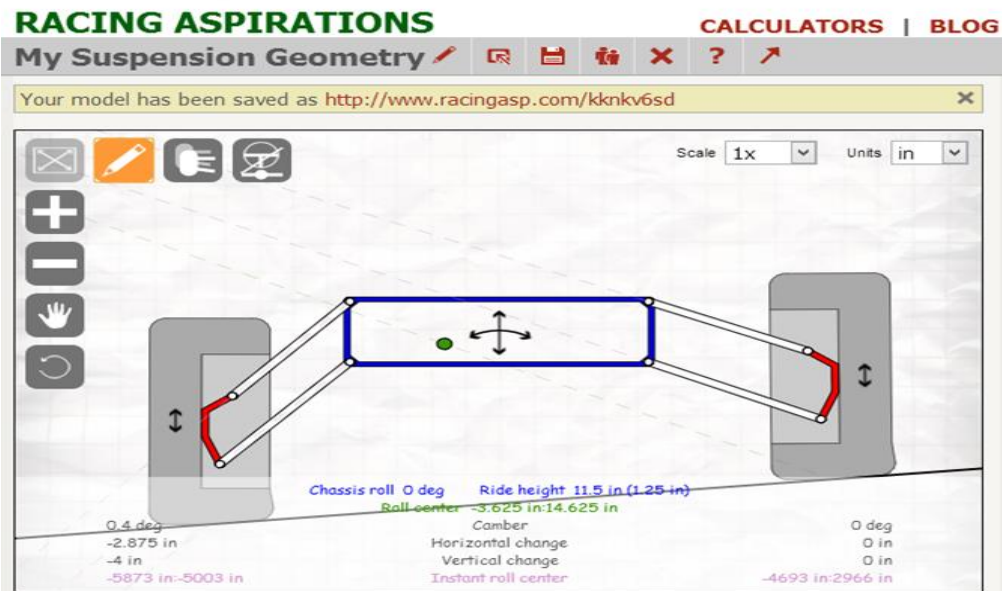
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7. Appendices:

Appendix Figure 1

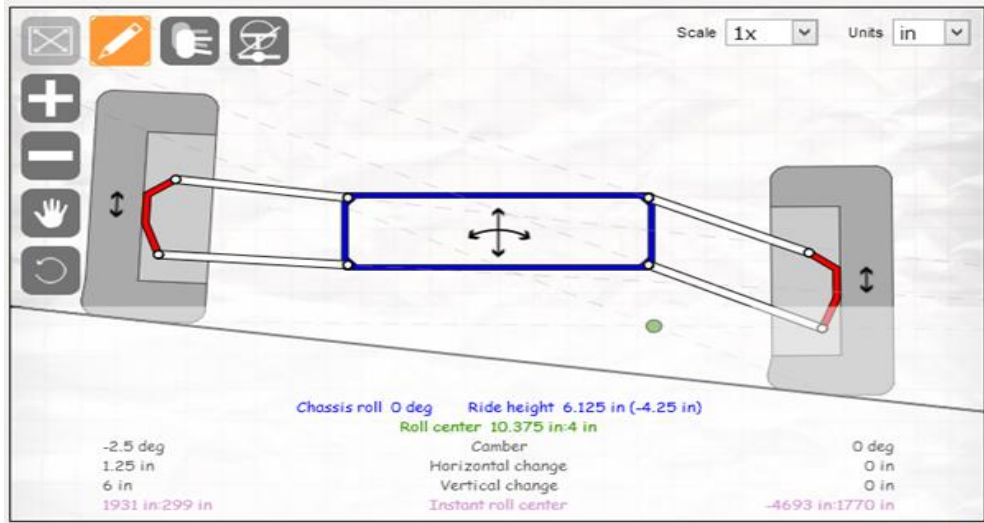


Appendix Figure 2

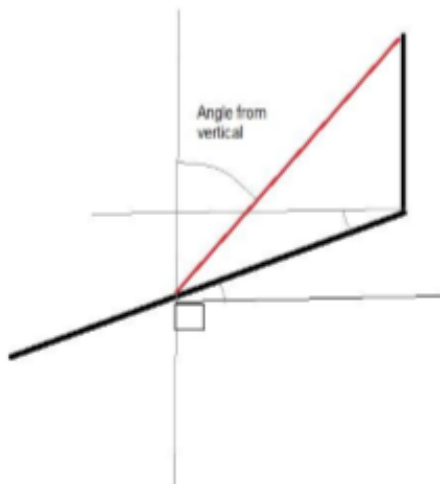


Appendix Figure 3

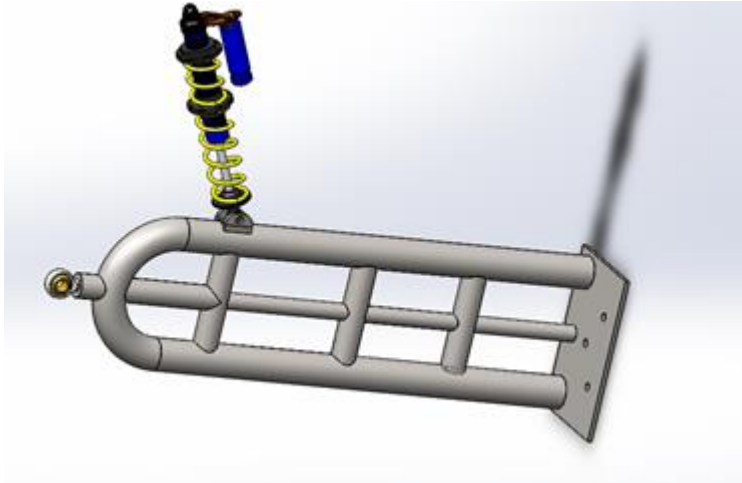
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Appendix Figure 4



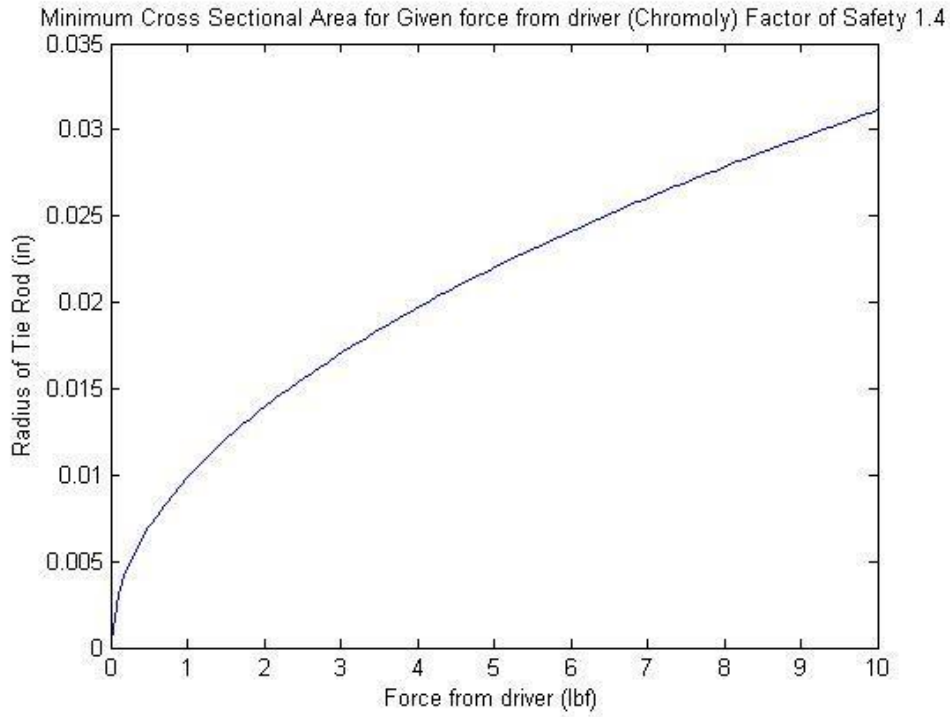
Appendix Figure 5



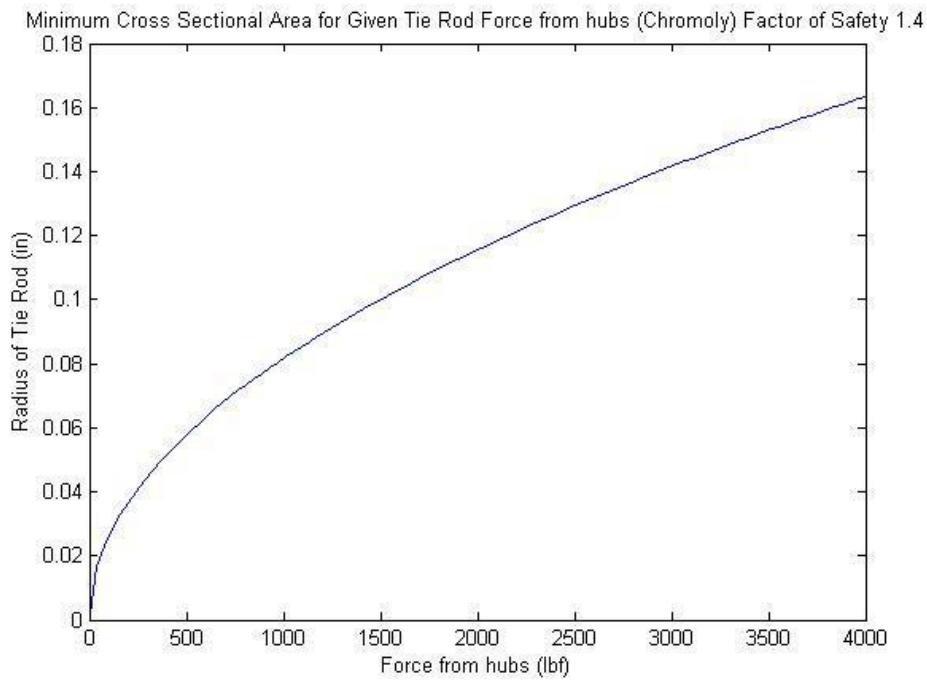
Appendix Figure 6

Properties	Metric	Imperial
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Poissons ratio	0.27- 0.30	0.27-0.30
Elongation at break (in 50 mm)	21.50%	21.50%
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Hardness, Knoop (Converted from Brinell hardness)	240	240
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Hardness, Vickers (Converted from Brinell hardness)	228	228
Machinability (Annealed and cold drawn. Based on 100% machinability for AISI 1212 steel.)	70	70

Appendix Figure 7

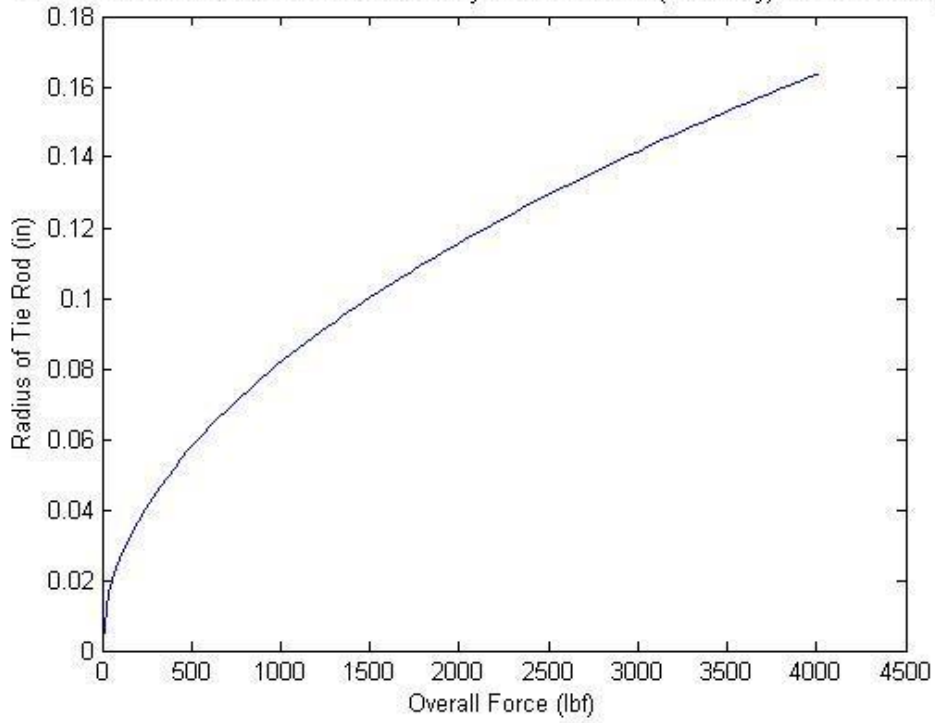


Appendix Figure 8

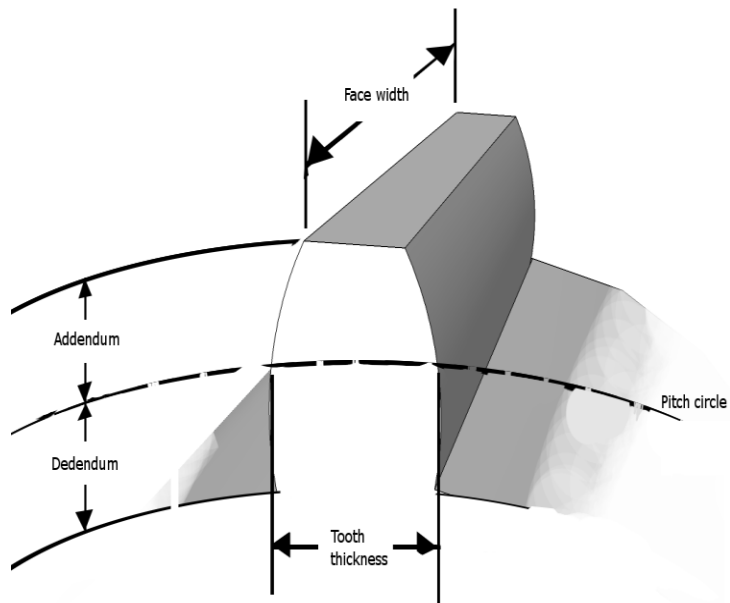


Appendix Figure 9

Minimum Cross Sectional Area for forces by driver and hubs (Chromoly) Factor of Safety 1.4



Appendix Figure 10



Appendix Matlab Code 1

Matlab code used to calculate Pinion dimensions and Stresses

```
%=====
%
% By: Benjamin Bastidos
% File Name: Spur_Gear_Design
% Description:
%   This code will be used to perform several calculations in order to
%   properly design a pinion and rack
%
%=====
clc
% For full-depth tooth
k=1;

% Pressure Angle
pres_ang=20;

% Rotation of pinion
n_p=100

% Rotation of rack
n_rack=100;

% Force by Driver pound force
F_D=(0.1:0.1:10);
% force by Driver in Newtons
W_t=F_D*4.44822162;

%Converting dia from feet to meters
%Dia_or_wheel=7.25/12;
%Dia_of_wheel=Dia_or_wheel*0.3048;

%Torque in newtons/meter
%Torque_by_Driver=F_D.*(7.25/2);

%Power Transmitted (Assuming torque will be applied over .5min)
%H=Torque_by_Driver/2;

% Here we are setting our preferred values for the module into a
```

```

% vector
m=2

% Here we are calculating the amount of teeth for our pinion
N_p=((2*k)/((1+2.*m)*(sind(pres_ang))^2)).*(m+sqrt((m.^2)+(1+2.*m).*(sind(pres_ang))^2))
N_p = input('Number of teeth on pinion (can only be multiples of 10 above N_p: \n');

% Now we will solve for the pitch diameters of the pinion of mm
d_p=N_p.*m

% Now we can solve pitch line velocity
V=(pi.*d_p.*n_p)/60;

% Here we are calculating the Transmitted Load in units of kN
% W_t=(60000*H)/(pi.*d_p.*n_p)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
Stress on pinion

% Now we will begin solving for bending stress
% Assuming that our power source will be uniform we will choose K_o=1
K_o=1;

% Here we will calculate our dynamic factor, K_v, while assuming a quality
% number of 6 given that the range between 3-7 will include most
% commercial-quality gears
Q_v=6;

B=0.25*(12-Q_v)^(2/3);
A=50+56*(1-B);

K_v=((A+sqrt(200*V))/(A)).^B;

% Here we will set our size factor, K_s, for both pinion and
% gear equal to 1 for simplification reasons
K_s_p=1;
K_s_g=1;

% Here we will calculate our face width
F = 3*pi*m

```

```

% Now we will calculate our load distribution factor, K_H. We will assume
% our gear and pinion to be uncrowned, with both having a face width under
% 1in, our pinion will be straddle-mounted with S_1/S be less than 0.175, having
% open gearing installation, and gear was not adjusted at assembly.
C_mc=1;
C_pf_pin=F./(10.*d_p);
C_pm=1;

A=0.247;
B=0.0167;
C=-0.0000765;

C_ma=A+B.*F+C.*F.^2;
C_e=1;

K_H_pin=1+C_mc.*(C_pf_pin.*C_pm+C_ma*C_e);

% Our gear will not have a rim so rim-thickness factor, K_B, will be 1
K_B=1;

% From Fig 14-6 and using N_p, N_g, and m=2 we get Y_J
Y_J_pin=.23;

% Now we calculate bending stress in units of MPa
sigma_b_pin=W_t.*K_o.*K_v.*K_s_p.*(1./(F.*m)).*((K_H_pin.*K_B)./Y_J_pin);
fprintf('The bending stress for the pinion is:%g \n',sigma_b_pin)

%%%%%%%%%%%%%%
%%%%%%%%%%%%%%

fprintf('The teeth number for the pinion is:%g \n',N_p)
fprintf('The pitch diameters for the pinion is:%g \n',d_p)

r_b_pin=(d_p./2).*cosd(20);
fprintf('The radii for the base circle on the pinion is:%g \n',r_b_pin)

fprintf('The face width of the pinion and rack is:%g \n',F)

Adden=1.*m;
Deden=1.25*m;

```

```
fprintf('The Addens for the pinion and rack is:%g \n',Adden)
fprintf('The Dedens for the pinion and rack is:%g \n',Deden)
```

Appendix Matlab code 2

MATLAB Code used for Tie Rod Analysis

% Author: Victor Cabilan

% Algorithm Name: SAE Baja Engineering Analysis (Rack and Pinion)

% Assignment Name: ME476C

```
F = linspace(0,4000,100); %Axial Force on Tie Rod (lbf)
sigma = 66700; %Yield Stength of Chromoly Steel(AISI4130)(psi)
A = F/sigma; %Cross Sectional Area of Tie Rod(in^2)
N = 1.4; %Factor of Safety
A_s = A*N; %Cross Sectional with Saftey Factor(in^2)
r_rod = sqrt(A_s/pi); %Radius of tie rod
```

```
F_d = linspace(0,10,100); %Force appied by driver (lbf)
r = 0.5; %Pinion Radius(in)
tau_shaft = (14.5/2)*F_d; %torque on shaft from driver
F_r = tau_shaft/r; %force on tie rod by driver
A_d = F_r/sigma; %minimum cross sectional area
r_rod = sqrt((N*A_d)/pi); %Radius of tie rod
```

```
Fall = F + F_d; %Total Force on tie rod
A_a = Fall/sigma; %cross sectional area for all forces
r_rodall = sqrt((N*A_a)/pi); %radius of tie rod for all forces
```

```
figure(1);
plot(F,r_rod);
title('Minimum Cross Sectional Area for Given Tie Rod Force from hubs (Chromoly) Factor of
Safety 1.4');
xlabel('Force from hubs (lbf)');
ylabel('Radius of Tie Rod (in)');
```

```
figure(2);
plot(F_d,r_rod);
title('Minimum Cross Sectional Area for Given force from driver (Chromoly) Factor of Safety
1.4');
xlabel('Force from driver (lbf)');
ylabel('Radius of Tie Rod (in)');
```



```

figure(3);
plot(Fall,r_rodall);
title('Minimum Cross Sectional Area for forces by driver and hubs (Chromoly) Factor of Safety
1.4');
xlabel('Overall Force (lbf)');
ylabel('Radius of Tie Rod (in)');

```

```
G = 27557e3; %Modulus of Rigidity (AISI 4130)
```

```
%Use Yield Strength for Chromally
```

Appendix Matlab Code 3

Matlab Code used for Shock Length and Shock Angle

```
%Program for shock angle/shock length calc
```

```
%By: William Mitchell
```

```
clear;
```

```
clc;
```

```
x=5.4; %height between lower and upper control arms (in)
```

```
LCA= 12.625; %length of lower control arm (in)
```

```
SC= (8:.5:10); %distance from shock mount to control arm pivot (in)
```

```
SL= sqrt( (x^2)+(SC.^2)-(2*x*SC*cosd(104))) %shock length for given mounting point SC (in)
```

```
A=asind(((sind(104))./SL)*x) %shock angle with respect to LCA (deg)
```

```
Ar=90-(A+24) %shock angle with respect to vertical
```

```
%%%%%%%%%%
%%%%%%%%%%
```