Analysis of Lower A-Arm in Wishbone Suspension

Finite Element Analysis Project

Andres Uranga

3/10/2014
Updated:
4/11/2016

Contents

Abstract

The automobile consists of many load-bearing components, all of which are extremely critical to the safe operation of the vehicle. Failure of any of these load-bearing components will result in potentially hazardous situations to the operator of the vehicle as well as other motor vehicle operators in the immediate vicinity. This paper explores critical structural considerations and provides a conclusive analysis on the viability of the components examined.

The component analyzed is taken from the SAE Mini-Baja off-road vehicle. This vehicle is designed to withstand extreme conditions and circumstances. The vehicle is put onto a racetrack where it sees a variety of conditions that test the ability of the suspension.

First, the loads and stresses are obtained using an analytical approach for a basis of Finite Element calculation. The A-arm is designed to withstand the loads seen by the vehicle in these dynamic situations. The stresses are analyzed statically at theoretical maximum value in order to analyze the element more thoroughly.

Introduction

A suspension system is used to isolate the components and the passengers of the vehicle from shocks and vibrations due to road surface. Any suspension is designed to meet the following requirements: ride comfort, road-holding and handling. (Guler, 2006)

Suspensions are classified in two groups: solid-axle and independent suspension system. For solid axle suspension systems, the wheels are mounted at the ends of the rigid beam. This allows that any movement of one wheel is transmitted to the opposite wheel. Independent suspension systems allow the wheels to move independently, so any movement that is applied to one wheel will not affect the other one. This suspension is mostly used in passenger's cars and light trucks. The advantages of the independent suspension are that it provides room for the engine and it has better resistance to steering vibrations. Many independent suspensions have been created for example, MacPherson strut, Trailing arm, Swing axle, Multi link and Double wishbone suspension. (Mohammad, 2006)

The double A-arm, or wishbone suspension is a type of vehicle suspension consisting of an upper A-arm and a lower A-arm that connects the wheel to the frame in a dynamic manner allowing vehicle control over a variety of terrain. This system is used on most commercially available vehicles today. In specific the A-arm, or wishbone, is a critical component in the ability of a vehicle to function and maneuver safely and efficiently. It is designed to absorb the impact seen by the tire while maintaining a consistent and maximum frictional force between the road and the contact patch of the tire. Due to its critical nature in the safe and effective operation of a vehicle, the A-arm must meet the demands of the most extreme uses. Material selection and geometric construction are combined with stress and strain analysis of the system to quantify the displacements and forces seen throughout the A-arm. This allows engineers to make qualitative decision on the ability of the system to withstand operating conditions. For this project it will be analyzed the lower arm of a double wishbone suspension from the SAE Mini-Baja off-road vehicle.

The movement of the suspension consists of two directional displacements; jounce and rebound. Jounce occurs when the suspension is deflects in the upward direction, causing the damper to absorb the impact. The rebound occurs when the suspension deflects in the downward motion; here the wheel is attempting to stay in contact with the surface of the road.

The A-Arm portion of the suspension is the critical link between the frame and the wheel, where the shock, or damper is mounted to a fixed point on the A-Arm. Figure 1 displays a schematic of the system under consideration.
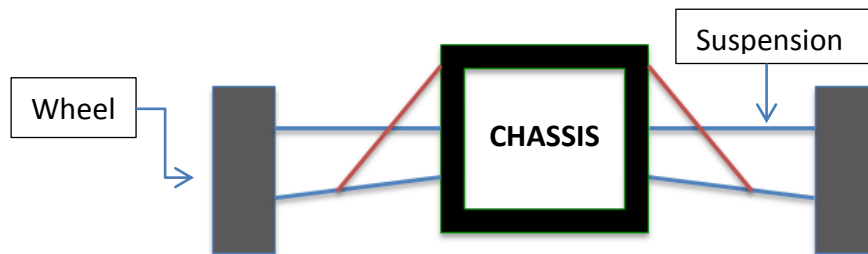

Figure 1. Schematic of the front part of the car.

The A-Arms are colored blue and the damper/shock-absorbers are colored red to display the basic set-up of the system

Analysis of Static Loading on A-Arm Component

The A-Arm is a horizontal member that is attached at one end to the wheel hub assembly and to the frame at the other end. The damper is attached to the lower A-Arm, which provides the resistive motion upon the jounce of the suspension. Figure 2 shows the actual lower arm from the wishbone suspension that is analyzed.

Figure 2. Actual lower arm from a Wishbone Suspension.

Free Body Diagram

The A-Arm is loaded in a dynamic manner, where the force applied to the wishbone suspension by the damper is at a 60° angle. Horizontally, the A-Arm is 17" in length. The shock or damper on the system is mounted at a fixed point, 9.5" from the frame (pin connection, which is assumed to be a fixed point). At maximum jounce, the A-Arm is assumed to be a static, cantilevered beam since it is in equilibrium. Figure 3 shows the body diagram of the wishbone suspension.
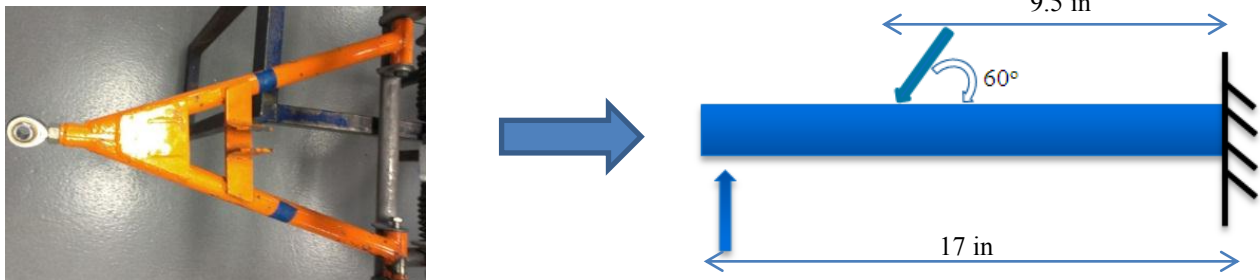

Figure 3. Body diagram to be analyzed.

Pre-processing

Figure 4 shows the beam used and dimension to represent the lower arm of the double wishbone suspension in NX 7.5 to do the FEA analysis. Appendix [1A] shows the blue prints.
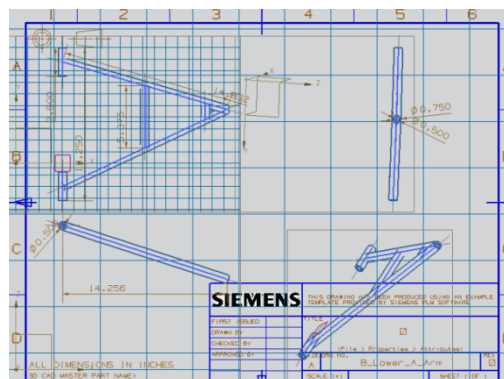

Figure 4. Dimensions of the beam used for the analysis.

Justifications

Justification of Material

Per regulations in the Society of Automotive Engineers (SAE) Mini-Baja rulebook, the suspension components must be constructed of 1" thin-walled, steel tubing. This narrows material selection down to a steel with a high strength properties. Since the components considered will see the most cyclic loading of almost any other component on a vehicle, the material must be selected carefully. The materials considered were AISI 4130 and AISI 1080. These materials are chosen because of their combination of strength characteristics. The material properties are shown in Table 1.

	Ultimate Tensile Strength	Tensile Strength, Yield	Poisson's Ratio	Shear Modulus
AISI 1080	60 ksi	86 ksi	0.29	11600 ksi
AISI 4130	130 ksi	86 ksi	0.27-0.30	11600 ksi

Table 1. Material properties for AISI 1080 and AISI 4130

These steels are commonly found in shaft applications in vehicles. Each steel has its unique properties that make it better than another. AISI 4130 steel also known as "chromoly" for its high content of chromium and molybdenum is very popular in high power applications, framing, sub-structures, pressure-vessels, structural tubing, and roll cages. It has very high ultimate and yield tensile strengths. This steel provides adequate strengths at a fair cost to manufacture and process.

The finish for manufactured components is normally machine finished or polished to try and remove any imperfections from the manufacturing process. This minimizes the distribution of cracks, scratches, or knicks that can cause the part to fail. Case hardening is used to add to the strength of the component. What this means is that the outside of the shaft is much harder and stronger than the interior of the component wall. This makes it a much better choice than the AISI 1080. The major benefit of case hardening is that it creates a very hard surface that results in compressive residual stress. This residual impedes cracks to grow with ease, thus providing better fatigue resistance and prolongs fatigue life of the component. The hardness is about 55 Rockwell, which makes it slightly brittle but is still maintains significant strength for this application

It can be seen that the material properties of AISI 4130 (Chromoly) are more favorable. The higher tensile strength makes it more desirable for the given application. The Ultimate Tensile strength is at 130 ksi and the Yield Strength is 86 ksi. The vehicle will see consistent exposure to water and mud; AISI 4130 is corrosion resistant therefore increasing the life of the component. The last two digits, xx30, in AISI 4130 denote a carbon content of 0.30%, making it a medium carbon steel. Medium carbon steel "balances ductility and strength and has good wear resistance; used for large parts, forging and automotive components." (efunda; low carbon steels) The increased carbon content causes the material property of the metal to be more effectively modified through various heat and surface treatment techniques.

Justification of Load Selection

The mass of the vehicle is estimated to be 18.6 slugs. The force on the system is calculated for a maximum loading situation. Here the vehicle experiences a 3 foot drop and the max force seen by the element is calculated as F=1235.53 lbf. However, the vehicle will rarely see equally distributed loading to each of the 4 tires, therefore ½ the force, $F = 617.765 \, lbf$. Finding only the

force acting vertical $F = 612.765 \sin 60 = 535\ lbf$ is considered to act at 1 tire in a maximum loading situation. A free body diagram of the force is shown in figure 5.

F=535 lbf

9.5 in.

17 in

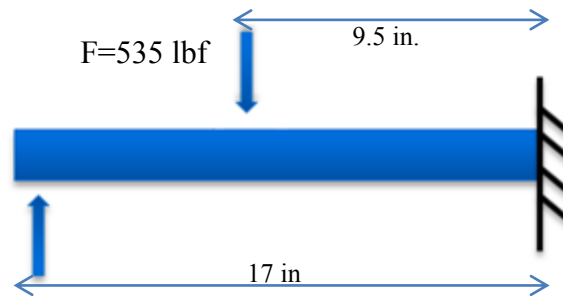Figure 5. Load applied to the beam

Justification of Element

The element is assumed to be a cantilevered beam because of the loading experienced by the A-Arm in normal operation. In this simulation, the loading is strictly in the vertical plane. This allows for a simplified beam element to be used in both the hand calculations and the FEA simulation. The "A" geometry in the A-Arm was further simplified due to its symmetry in geometry and loading. This allows for more simple simulation results in FEA.

Justification of Boundary Conditions

The boundary conditions applied to this element are chosen based on assumptions made in the analysis. The fixed connection at one end of the beam allows for the displacement and phi values to be assumed at zero. The force applied by the shock/damper at full jounce of the suspension is -535lbf. And the force of the ground on the tire is stated at 298.97lbf. The applied moments at the end of the beam and at the sock mount are zero.

FEA Solution

Based on the model of the pre-processing, the following FEA solution was generated.

Mesh Generation

For the mesh generation tetrahedral were used. Tetrahedral elements were chosen because of the accuracy gained in using triangular elements on a curved surface. Square elements are preferable in linear strain because the square elements are more closely approximate to square or rectangular stress/strain elements undergoing linear loading. Tetrahedral elements are better to approximate the dimensions of a curve, in this case, a cylindrical tube or beam. This added accuracy, which is important to the results obtained because the strain seen in the elements on a curve varies with position on that curve. Besides, small dimensions for the mesh are required to have more elements in order to be more accurate. Figure 6 shows the mesh generated in the beam.
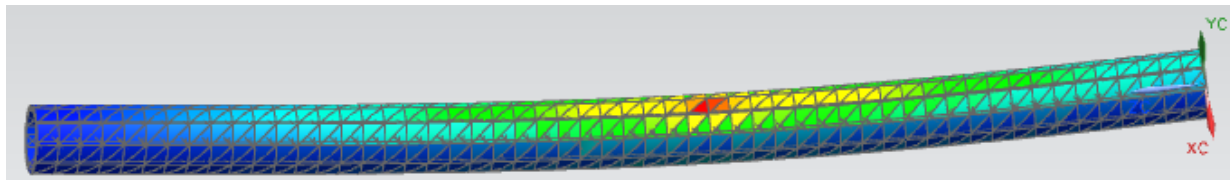
Figure 6. Mesh generated on the beam.

Von Misses

Figure 7 shows the FEA analysis obtained from the Von Misses, as it was expected where the force of 535 lbf was applied, it is red. It means that there is a load concentrate on it that will cause the beam to bend.
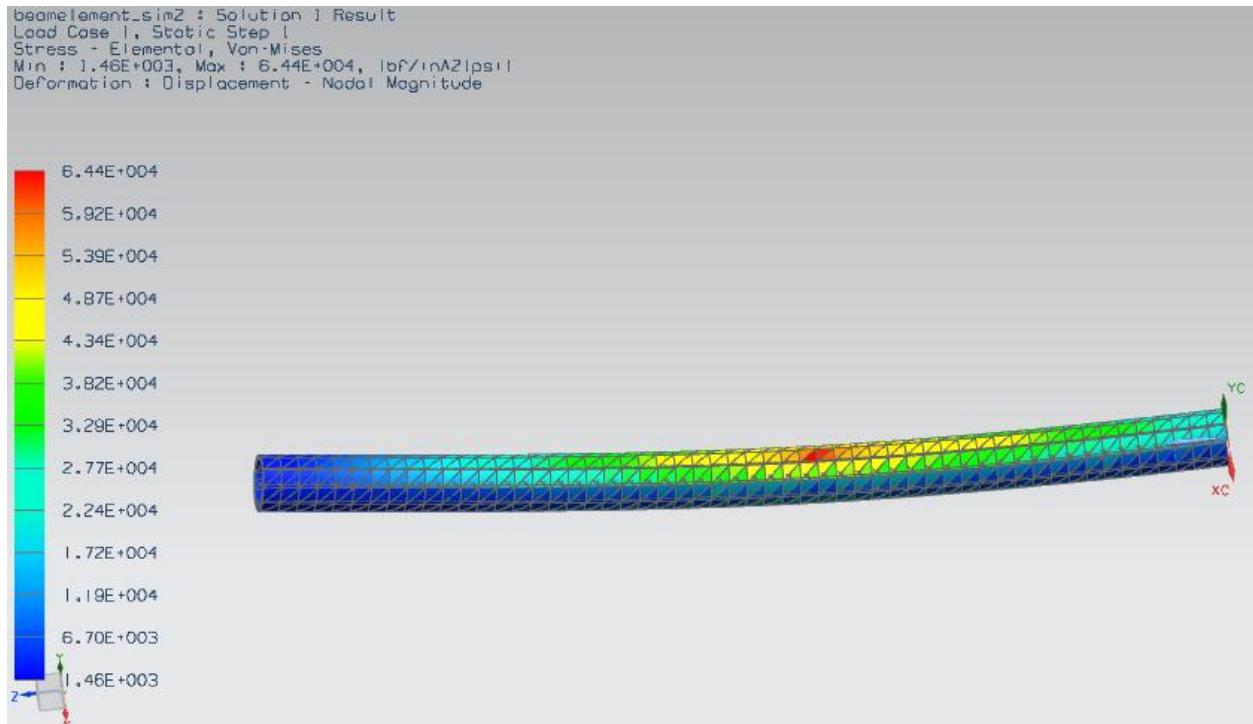

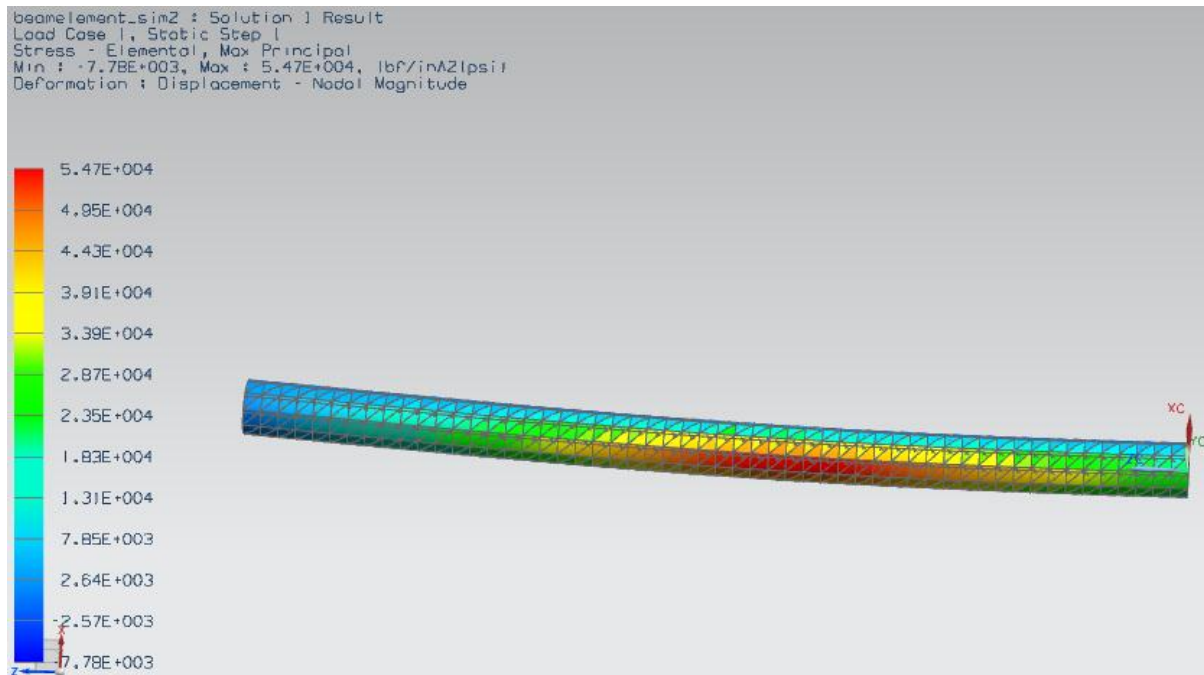Figure 7. Von Misses FEA Solution

Deflection

Figure 8. Deflection

Max Shear stress


```
beamelement_sim2 : Solution 1 Result
Load Case 1, Static Step 1
Stress - Elemental, Max Shear
Min : 7.71E+002, Max : 3.58E+004, lbf/in^2[psi]
Deformation : Displacement - Nodal Magnitude
```
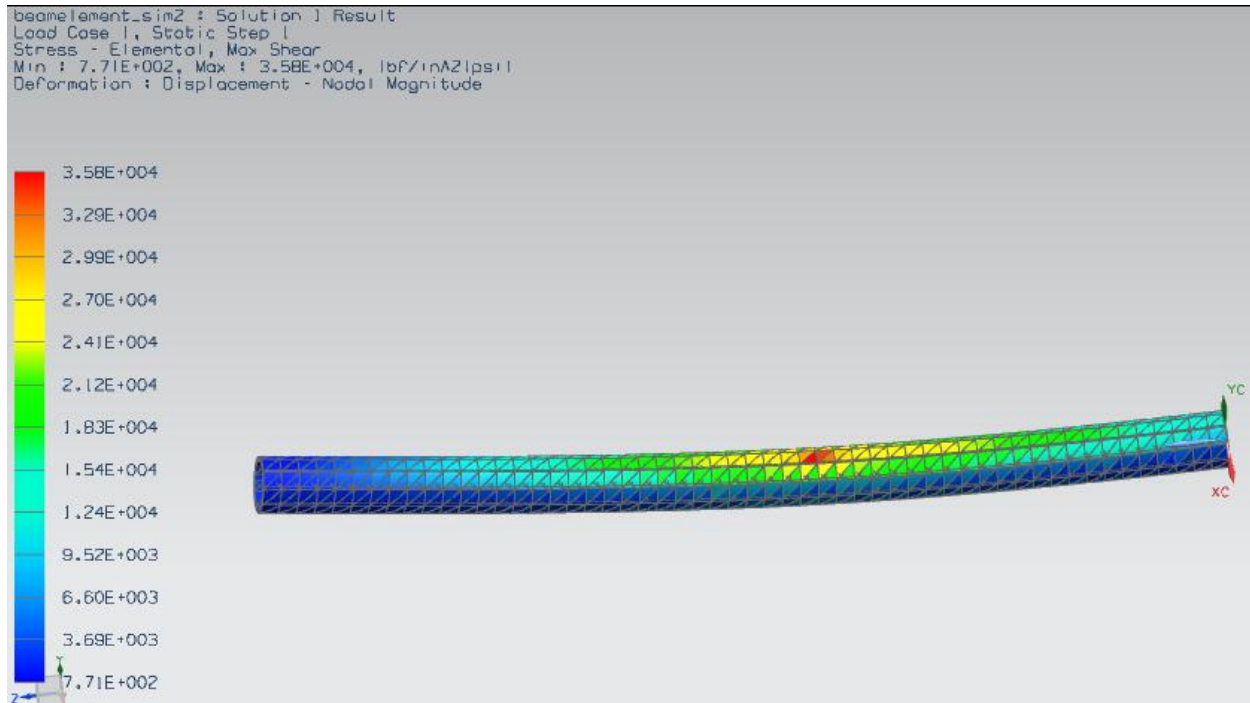
Figure 9 Shear stress analysis

Analytical Analysis

Load Analysis

The load applied to the lower A-arm wishbone suspension is 535 lbf that is located 9.5 in from the fixed point. The pin connection of the lower A-arm of the wishbone suspension was replaced with a fix end. Calculations are shown on Appendix [1b].

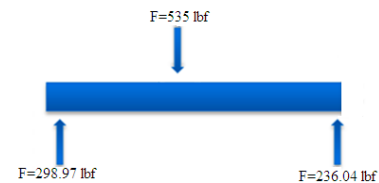$$\sum M_{fix\ end} = 0; \quad (1.B)$$

$$\sum F_y = 0; \quad (2.B)$$
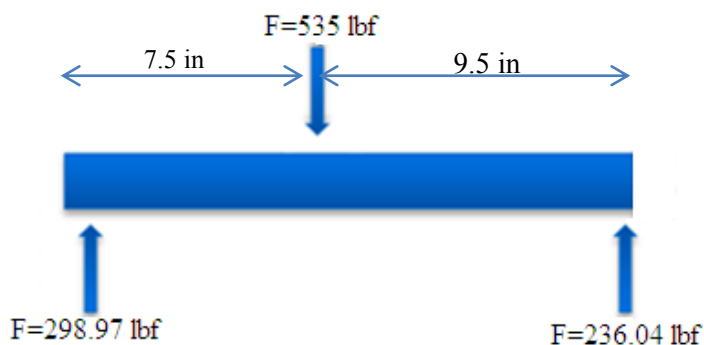

Figure 10. Body diagram with forces

From equation (1.B) and (2.B) the following values are obtained:

$F_1 = 236.03\ lbf \qquad F_R = 298.97\ lbf$

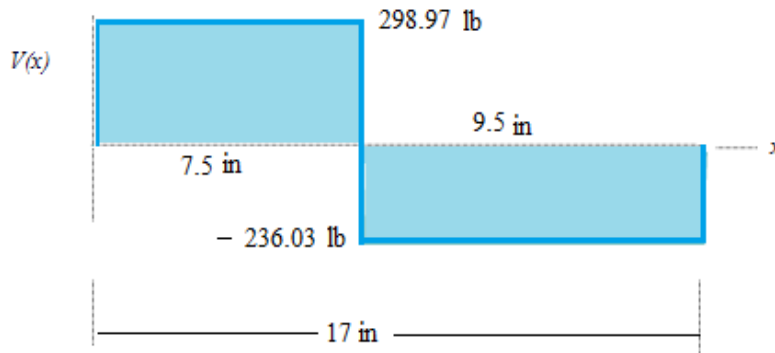From the forces applied to the beam the following shear and moment diagram were obtained.

298.97 lb

9.5 in

x

7.5 in

− 236.03 lb

17 in

Figure 11. Shear Diagram
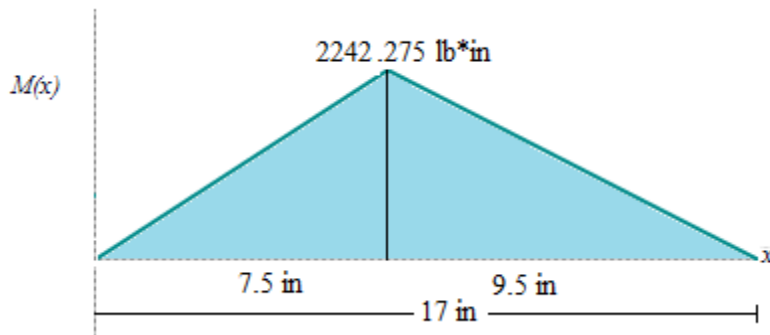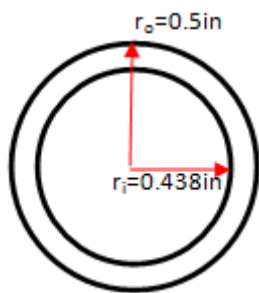

2242.275 lb*in

7.5 in

9.5 in

17 in

x

Figure 12. Moment Diagram

Stress Analysis

The stress analysis applied in the beam is showing below. Calcualtions are shown on Appendix [1C]. Figure 13 shows the cross sectional area of the beam.

$r_o = 0.5$ in

$r_i = 0.438$ in

$$I = \frac{\pi}{4}(r_0^4 - r_i^4) = 0.02018 \ in^4 \qquad (1.C)$$

$M_{max} = 2242.275 \ lbf * in$ (Obtained from the Moment Diagram Figure 11)).

$$A = 0.1827 \ in^2 \qquad (4.C)$$

Figure 13. Cross section area of the beam

Figure 14 shows the points that are analyzed on the beam. We are focusing in the highest stress applied to the beam.
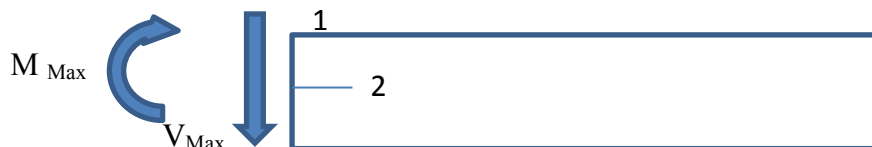

M $_{Max}$

1

2

V_{Max}

Figure 14.Stress distribution on the beam

Point 1

$\sigma_{max} = 69.68\ ksi$ (2.C)

$Principal\ stress\ \sigma_2 = 69.68\ ksi$ (3.C)

$\tau = 0$

Point 2

$\sigma = 0$

$V = 298.97\ lbf$ (Obtained from the shear diagram Figure 10)

$\tau_{max} = 3.272\ ksi$ (5.C)

$Principal\ stress\ \sigma_{1,2} = \pm 3272.8\ psi \quad \sigma_1 = 3.272\ ksi \quad \sigma_2 = -3.272\ ksi$ (5.C)

The max shear stress obtained from FEA was $3.58*10^{-3}$ from figure 9. It is very similar to the values obtained analytically.

The results are shown in Figure 14 for the stresses acting on elements located at point 1 and 2.
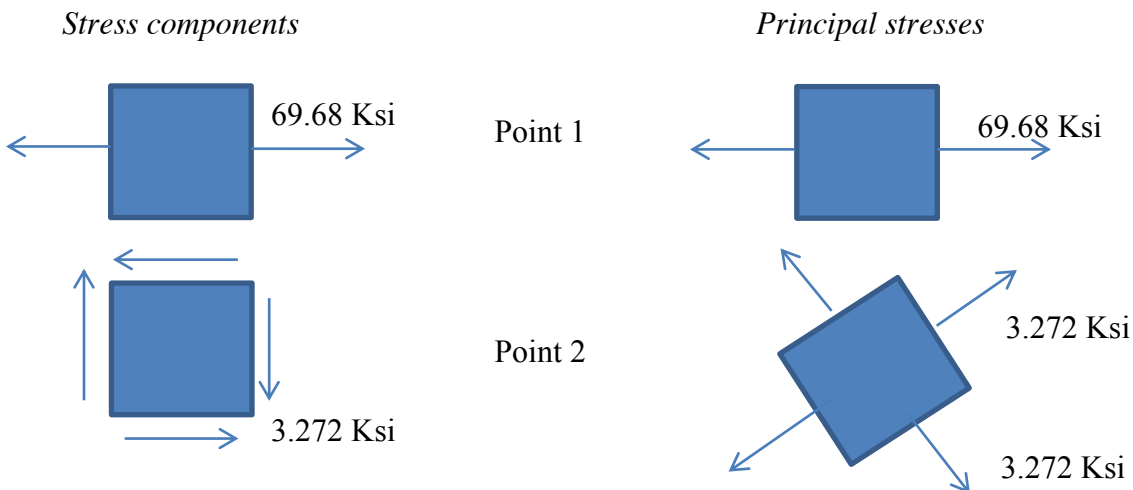
Stress components   *Principal stresses*

69.68 Ksi   Point 1   69.68 Ksi

Point 2   3.272 Ksi

3.272 Ksi   3.272 Ksi

Figure 15. Stress components and Principal Stresses from point 1 and point 2

Deflection Analysis

The A-arm has been taken to be considered as a simple cantilever beam the deflection can be obtained with the force, caused by the weight of the car, acting on it. The A-arm is attached to the frame at one end and will be attached to the wheel at the other end, which creates to reaction forces. This means that the component will deflect in a parabolic way with a downward concave. It is also noted that the force is not being applied directly in the middle of the arm. This causes each end of the arm to have different angle of deflection and slope. R.C. Hibbeler's *Mechanics of Materials* was used for the formulas for the deflection and angle of deflection which are in Appendix [1D].

Results	
δ	8.39 x 10^{-4}in
Θ_1	-1.53 x 10^{-3}°
Θ_2	1.65 x 10^{-3}°

Table 2. Values from the deflection of the beam

Table 2 shows the results that were obtained with the formulas on appendix [1D]. This verifies that there was minimal deflection with the forces acting upon it. The deflection that was

calculated happens at the point where the load brought by the weight of the car is applied. From the FEA solution the value of deflection was $5*10^{-4}$ which was very close to the value that we got analytically.
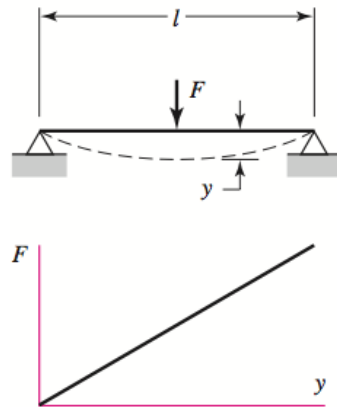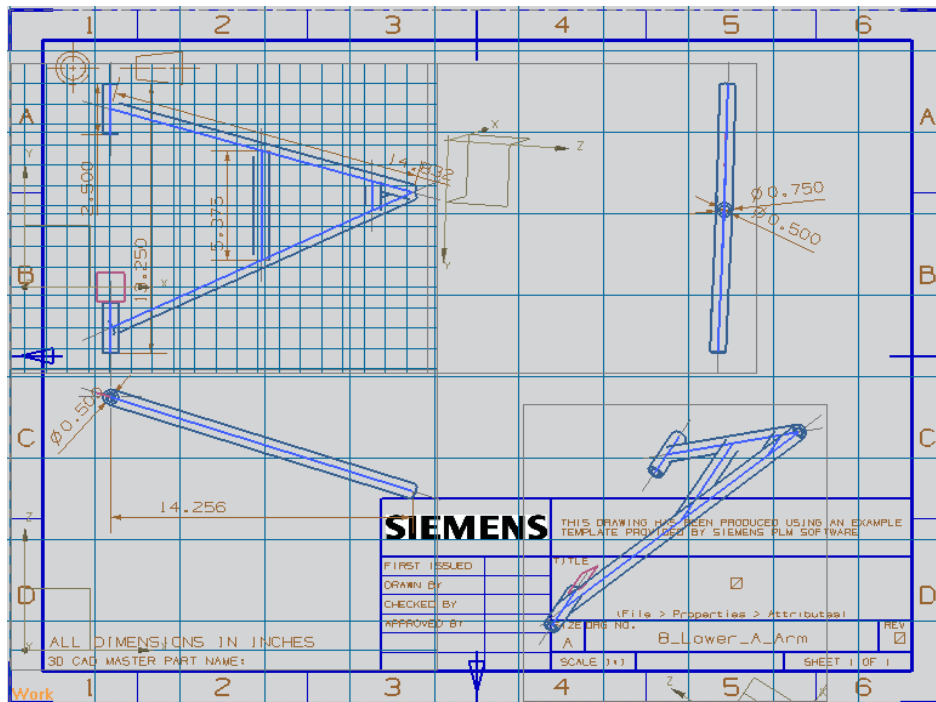

Figure 16. Deflection of the beam.

Failure Analysis

As stated, the beam element must withstand all of the loads seen in its operational life. The beam element must withstand continual loading and unloading. This cyclic loading of the element cannot approach the yield strength of the material so that the life of the element is at a maximum. The max bending stress, σ_{max}=69,987.8 ksi. At the top of the cylindrical element, the shear stress is zero, $\tau = 0$. This makes the Von Mises, σ'= 69,687.8 ksi. After finding this, the Von Mises is applied to the Factor of Safety equation. Here, with a material yield strength of 130ksi, the factor of safety is found to be η=1.86. This factor of safety is sufficient for the element considered and will allow for the vehicle to experience loading without failing

Conclusion

After obtaining the results from Nx it was clear that the stresses calculated by hand were accurate to the readings of the program. The slight differences in numbers might be due to the nodes chosen at which the forces were applied in the program itself. The applied forces in the computer program were placed on the beam at approximate locations because of the complexity of modeling and simulating a complete A-Arm. This complexity resulted in the simplification of the A-arm component to a beam element. This simplification, while broad, can be approximated with certain accuracy due to the simple geometry of an A-arm. The A-arm is used as a link between the vehicle and the tire, this supposed to withstand the impact loading that a tire is subjected to. Without the A-arm a rougher ride in the vehicle is expected and tire traction could be lost with the floor if the A-arm is not working properly. This means that the A-arm is not only important for a comfortable and smooth ride but also can affect the performance of the vehicle and the wear of the tire. The A-arm moves very well with the bumps on the road and helps the tire keep traction, which is why it is a common component when it comes to suspensions of automobiles. It is easy to assemble and doesn't require much maintenance. This is due to the fact that the arm is basically a triangular element thus making it very robust mechanically. This design allows for the A-arm to be very strong but also very light weight, which is important if you are in a competitive vehicle.
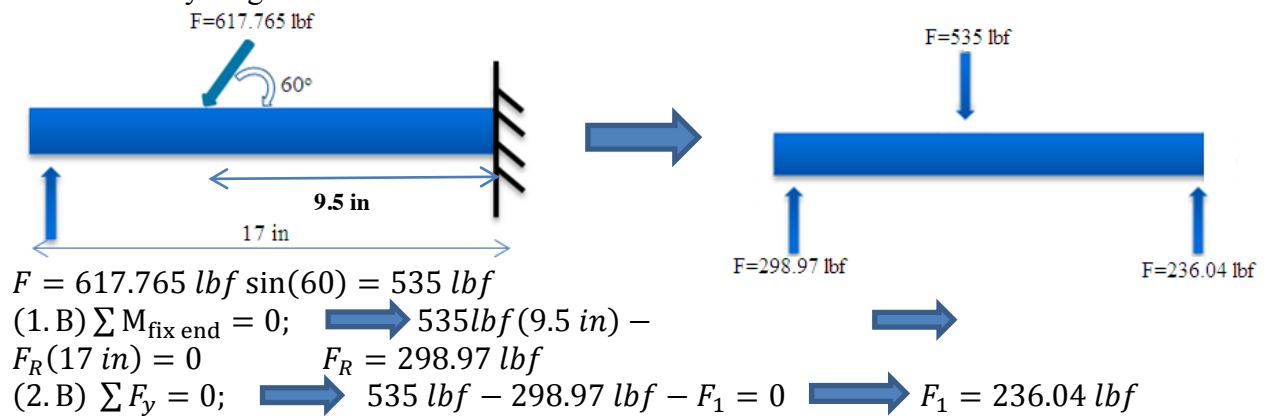
Appendix [1A]

Blue print

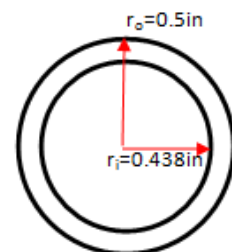# Appendix [1B]

Load Analysis
From the body diagram obtained

F=617.765 lbf

60°

9.5 in

17 in

F=535 lbf

F=298.97 lbf

F=236.04 lbf

$F = 617.765 \ lbf \sin(60) = 535 \ lbf$

$(1.B) \sum M_{\text{fix end}} = 0; \quad 535 lbf (9.5 \ in) - F_R(17 \ in) = 0 \quad F_R = 298.97 \ lbf$

$(2.B) \sum F_y = 0; \quad 535 \ lbf - 298.97 \ lbf - F_1 = 0 \quad F_1 = 236.04 \ lbf$

Appendix [1C]

Stress Analysis

$I = \frac{\pi}{4}(r_0^4 - r_i^4) = \frac{\pi}{4}(0.5^4 - 0.438^4) = 0.02018 \ in^4 \qquad (1.C)$

Point 1

$r_o=0.5in$

$r_i=0.438in$

$$\sigma_{max} = \frac{32M}{\pi d^3} = \frac{2242.275\,lbf\ in\ 32}{\pi(1^3 - 0.876^3)} = 69680.03\ psi = 69.68ksi$$

$$\tau = 0$$

$$Principal\ stress\ \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \sigma_2 = 69.68\ ksi\ (3.C)$$

Point 2

$$A = \pi(r_o^2 - r_i^2) = \pi(0.5^2 - 0.438^2) = 0.1827\ in^2\ (4.C)$$

$$\tau_{max} = \frac{2V}{A} = \frac{2(298.97\ lbf)}{0.1827\ in^2} = 3272.8\ psi = 3.272ksi$$

$$Principal\ stress\ \sigma_{1,2} = \pm 3272.8\ psi$$

$$\sigma_1 = 3272.8\ psi$$

$$\sigma_2 = -3272.8\ psi$$

Appendix [1D]

Deflection
Deflection

$$\delta = \frac{F\ ba}{6E\left(\pi\frac{D_o^4 - D_i^4}{64}\right)L} \qquad (1.D)$$

Angle of Twist

$$\theta_1 = \frac{-Fab(L+b)}{6E\left(\pi\frac{D_o^4 - D_i^4}{64}\right)L} \qquad (2.D)$$

$$\theta_2 = \frac{Fab(L+a)}{6E\left(\pi\frac{D_o^4 - D_i^4}{64}\right)L} \qquad (3.D)$$

Bibliography

Guler, D. (2006). *DYNAMIC ANALYSIS OF DOUBLE WISHBONE SUSPENSION*. İzmir.
     Mohammad, M. (2006). *Analysis of a Double Wishbone Suspension System Components*.
     Coventry.

Terry, Len and Alan Baker. *Racing Car Design and Development*. Cambridge,   Mass. Robert
     Bentley, Inc, 1973. Print

Norby, Jan P. *Chassis Tuning*. New York: Sports Car Press, 1972. Print.

Definition: Low Carbon Steels, Efunda. Retrieved from:
http://www.efunda.com/materials/alloys/carbon_steels/low_carbon.cfm