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ABSTRACT

This paper is an introduction to the design of suspension components for a Formula SAE car. Formula SAE is a student competition where college students conceive, design, fabricate, and compete with a small formula-style open wheel racing car. The suspension components covered in this paper include control arms, uprights, spindles, hubs, pullrods, and rockers. Key parameters in the design of these suspension components are safety, durability and weight. The 2001 Lawrence Technological University Formula SAE car will be used as an example throughout this paper.

OVERVIEW

In designing suspension components for the 2001 Lawrence Technological University Formula SAE vehicle, safety and durability were the top priorities. In order to ensure the safety of the suspension system, the loads acting on every component were extensively studied by utilizing strength of material calculations and Finite Element Analysis. Another measure used to ensure the safety of the suspension system was that every suspension fastener was put into double shear and was either safety wired or secured with a locknut.

Weight was another important consideration while designing and manufacturing the suspension system on the 2001 Lawrence Technological University Formula SAE car. In order to achieve a weight reduction over previous Lawrence Technological University Formula SAE vehicles, Finite Element Analysis was utilized to remove the maximum amount of material from every suspension component while maintaining a critical safety factor of two. Also, chrome moly or 7075-T6 aluminum was used for every suspension component depending on which material was more practical considering any compatibility and dimensional restrictions. These materials were chosen due to their superior strength to weight ratios. The suspension load paths were also extensively studied to ensure that they were fed into the suspension system and frame in a robust manner.

SUSPENSION DESIGN

CONTROL ARMS

The purpose of the control arms is to secure the wheel assembly to the chassis. Coupled with the suspension geometry, the control arms play a key role in determining the kinematics of the car such as camber rejection and roll stability. [1] The control arms also provide a means for tuning the suspension for specific courses.

Figure 1: Assembled Rear Control Arms

In order to reduce weight, spherical bearings are staked into the pivot points and ball joints of the lower control arms, along with the rear upper control arm pivot points and front upper control arm ball joints instead of using traditional heavier rod ends at these locations. In order to stake the spherical bearings into the 4130 steel tubing, a hollow aluminum insert is pressed into the tubing. Non-tempered aluminum was used for the insert due to the fact that tempered aluminum is too hard to flatten, which causes the 4130 steel tubing to crack. Next, the end of the tube is crushed flat and a hole is milled through the flat end of the tubing. The bearing is then firmly staked into the control arm.

Figure 2: Inserted Spherical Bearing

Rod ends with 4130 steel threaded inserts are used where suspension adjustment is necessary for tuning. Rod ends are used on the front upper control arm pivot points to allow for camber and caster adjustments in the front suspension. Also, rod ends are used on the rear upper control arm ball joints to allow for camber adjustment in the rear.

The appropriate size of 4130 steel tubing was determined to be 19.05 mm OD x 0.71 mm wall thickness from performing buckling and bending stress calculations. (See Appendix) This size of tubing is equivalent to 0.75 inch x 0.028 inch wall thickness, which is a common size of tubing in the United States. Using this size of tubing with a cornering load of 1.3 g and a braking force of 1.4 g, which were determined from equipping previous Lawrence Technological University Formula SAE vehicles with data acquisition, the minimum safety factor in any of the control arms subjected to simultaneous braking and cornering was calculated to be 1.7. [2] The cornering and braking forces were calculated using a vehicle and driver weight of 2935.8 N. (See Appendix) These calculations were also verified by performing Finite Element Analysis using COSMOS software.

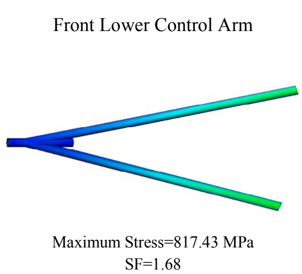


Figure 3: Control Arm FEA Results

Also, an extensive load study performed on the control arms resulted in the arms of the control arms being directly in line with chassis nodes, minimizing any bending moments acting on the frame. All of the control arm fasteners are put into double shear and meet AN specifications. Therefore, the control arms were designed to be lightweight, reliable, and safe.

UPRIGHTS

The main function of the uprights is to provide an interface to connect the upper and lower ball joints with the spindle. It is crucial to minimize the weight of the uprights because they are unsprung mass, and the shocks have to control this weight in bump. [3]

The uprights were manufactured from 7075-T6 aluminum instead of 4130 steel, which is common on many Formula SAE vehicles, as a weight reduction. Also, the 2001 Lawrence Technological University Formula SAE uprights were manufactured using CNC processes instead of cutting and welding tubing in order to increase their strength.

For safety purposes, the ball joint fasteners were put into double shear by passing them through the top web of the ball joint pocket and threading them into the lower web of the ball joint pocket.

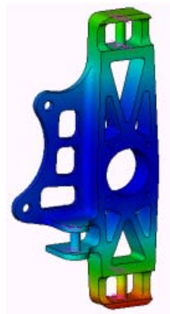
Front Uprights

To maximize the strength and minimize the deflection of the front uprights, many design iterations consisting of various shapes were performed. Finite Element Analysis was executed on each iteration in cornering, braking, and a combination of cornering and braking situations as a worse case scenario. These situations are seen while performing typical maneuvers on a Formula SAE course. To simulate a cornering force of 1.3 g, the spindle hole was rigidly constrained while a load of 895.0 N was applied to the upper ball joint and a load of 2304.7 N was applied to the lower ball joint in the opposite direction. (See Appendix) Braking was simulated in Finite Element Analysis by rigidly constraining the spindle hole and applying a force of 1121.1 N at each of the brake caliper mounting locations.

Preliminary Finite Element Analysis results showed that a wide upright with pockets in it was stronger than a traditional narrow upright without any pockets. It was also discovered through Finite Element Analysis that triangular pockets provided the greatest strength with the least amount of deflection. Further Finite Element Analysis iterations resulted in an upright with optimized triangular pockets with the majority of the mass centered around the spindle hole. The safety factor of the final front upright was 4.0 in cornering, 2.8 in braking, and 2.5 for combined cornering and braking.

Front Upright FEA Results

Loading Case	*Max Stress (MPa)*	*Deflection (mm)*	*Safety Factor*
Cornering	123.35	.363	4
Braking	179.66	.044	2.8
Cornering and Braking	205.10	.409	2.45



Combined Cornering and Braking Stress

Figure 4: Final Front Upright FEA Results

Rear Uprights

Many design iterations were performed on the rear uprights to maximize their strength while minimizing their weight. Finite Element Analysis was performed on the rear uprights in cornering, braking, and a combination of both of these situations. To simulate cornering on the rear uprights, the center hole was rigidly constrained, while forces of 781.3 N were applied to the upper ball joint and toe bar mounting location and a force of 2227.0 N was applied to the lower ball joint in the opposite direction of the upper ball joint force. (See Appendix) To simulate braking, a force of 2224.1 N was applied to the upper and lower ball joints in the longitudinal direction, while the center hole was fixed.

Finite Element Analysis results showed that triangular pockets proved to be the most effective means to reduce the weight of the rear uprights without compromising strength. The resulting safety factors of the rear upright were 4.3 in cornering, 3.3 in braking, and 3.1 in combined cornering and braking.

Rear Upright FEA Results

Loading Case	*Max Stress (MPa)*	*Deflection (mm)*	*Safety Factor*
Cornering	122.80	0.27	4.3
Braking	152.68	0.34	3.3
Cornering and Braking	162.36	0.36	3.1

Cornering Stress

Figure 5: Final Rear Upright FEA Results

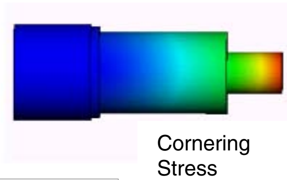
SPINDLES

The spindles provide a base at which the wheel assembly rotates about. Its outer diameter is critical to prevent slop in the bearings, resulting in premature wear. Also, as learned from previous Lawrence Technological University Formula SAE cars, this suspension component is greatly prone to fatigue loads that originate from abnormalities in the road surface and from weight transfer due to lateral and longitudinal accelerations. [4]

The spindles were manufactured from 4340 steel due to its excellent fatigue-resistant properties and strength. Key dimensional considerations for the spindles included bearing and upright dimensions because the spindle is pressed into the upright. In order to prevent the hub from sliding into the upright, a step was designed into the spindle 5.08 mm from the upright. The step is the same thickness as the bearings' width in the hub so that the slight impact load caused by movements of the hub and bearings would be distributed through the entire bearing. Another step was added to the free end of the spindle so that a washer and castle nut could be used to secure the hub to the spindle.

Figure 6: Final Spindle Design

An analysis of the spindle was performed using maximum braking and vertical forces. The maximum braking force experienced by the 2001 Lawrence Technological University Formula SAE car was found to be 1.4 g, and the maximum cornering force on a tire was found to be 1.3 g. Using these forces and Finite Element Analysis, the spindle's safety factor was revealed to be 25.8 in braking and 2.8 during cornering. The spindle's safety factor was determined to be 2.6 when braking and cornering loads were applied to the spindle simultaneously.



Cornering Stress

Spindle FEA Results

Loading Case	*Max Stress (psi)*	*Deflection (in)*	*Safety Factor*
Cornering	324.05	.150	2.8
Braking	35.21	4.32×10^{-4}	25.8
Cornering and Braking	343.90	.155	2.63

Figure 7: Spindle FEA Results

For safety purposes, a backplate was welded to the back of the spindle so that it could be bolted onto the back of the front uprights after it was pressed into the upright.

HUBS

The purpose of the hubs is to rotate the wheels and tires. It is critical to minimize the weight of the hubs because they are rotating and unsprung mass. [5] Also, to reduce friction due to the rotational motion of the hubs, the 2001 Lawrence Technological University Formula SAE car utilizes bearings in the hubs.

<u>Front Hubs</u>

The front hubs are manufactured from 7075-T6 aluminum due to its superior strength to weight ratio. Also, in order to minimize weight without compromising strength, the front hub utilizes a four-leaf clover pattern to hold the lugs and brake rotor fasteners. The brake rotor and lug four-leaf clover patterns are offset 45^0 from each other to allow for ease of pressing lugs into the hub and installation of brake rotors. Also, a 10.16 mm high step that is 19.69 mm wide is designed into the inside center of the front hub to keep the bearings apart from each other and to dissipate heat from the bearings.

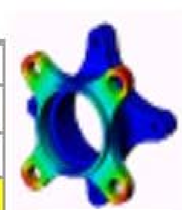
Another key function of the front hubs is to allow the brake rotors to float freely. This is accommodated through a slot in the brake rotor fingers. Floating rotors provide maximum stopping force by allowing both brake pads on the caliper to grip the rotor evenly. Also, the slot in the brake rotor fingers allow the brake rotor fasteners to be put into double shear as a safety feature. [6]

Figure 8: Final Front Hub

Finite Element Analysis was performed on the front hubs to validate their design. A braking force of 1.4 g applied to the brake rotor fastener holes resulted in a safety factor of 5.0. Also, a cornering force of 1.3 g was simulated by applying a force of 4469.9 N on the bottom lug hole and 3024.2 N on the top lug hole in opposite directions while constraining the brake rotor fingers from translating. (See Appendix) This loading resulted in a safety factor of 2.8. A safety factor of 2.7 was obtained when combining braking and cornering.

Front Hub FEA Results

Loading Case	*Max Stress (MPa)*	*Deflection (mm)*	*Safety Factor*
Cornering	178.44	0.13	2.82
Braking	126.66	0.09	5
Cornering and Braking	274.05	0.18	2.66



Cornering Stress

Figure 9: Front Hub FEA Results

<u>Rear Hubs</u>

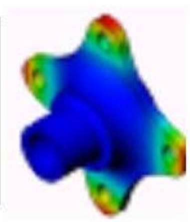
The main function of the rear hubs is to connect the driveline to the wheels through the halfshafts and constant velocity (CV) joints. The rear hubs are manufactured from 4340 steel because they must be compatible with the outer CV joints that are splined to accept a hub.

The rear hubs also utilize a four-leaf clover design to minimize their weight. To reduce friction from the rotating hub, bearings are placed in between the upright and the shaft on the hub. As a safety feature, a spindle lock nut is applied on the end of the CV joint to prevent the hub from becoming disengaged from the driveline.

Finite Element Analysis proved that the rear hub's design was adequate for its application. A safety factor of 4.9 was obtained when a torque of 697.0 N-m originating from the driveline was applied to the hub shaft while restraining the hub from rotating. When a cornering force of 1.3 g was simulated on the hub by placing a load of 3024.2 N on the top lug hole and 4469.9 N on the bottom lug hole in the opposite direction while preventing the shaft from rotating, the resultant safety factor was 4.2. (See Appendix) When combining a cornering situation with the torque acting on the hub, the resultant safety factor was 3.8.

Rear Hub FEA Results

Loading Case	Max Stress (MPa)	Deflection (mm)	Safety Factor
Cornering	246.70	0.09	4.22
Braking	126.66	0.01	4.9
Cornering and Braking	274.05	0.10	3.8



Cornering Stress

Figure 10: Rear Hub FEA Results

ENERGY MANAGEMENT MECHANISMS

The energy management system consists of the pullrods, rockers, and shocks. The system's purpose is to activate the inboard shocks, which greatly improves the handling qualities of the car. The geometry of the suspension mechanism is critical because it determines the motion ratio, which is the ratio of vertical wheel movement to shock displacement. This ratio is used to determine the car's natural frequency, which significantly affects the car's handling qualities. Friction is kept to an absolute minimum in the dampening system to allow for maximum efficiency of the shocks. [7] It is also critical that the pullrod, suspension mechanism, and shock are located in the same plane to eliminate bending moments on these suspension components.

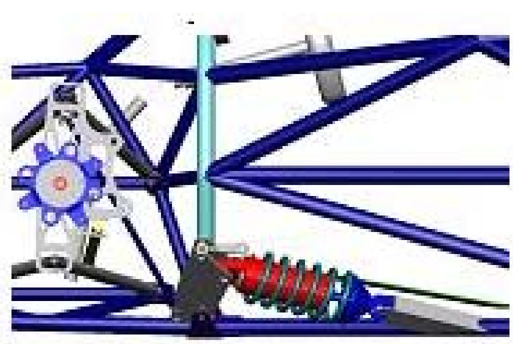


Figure 11: Placement of Pullrod and Suspension Mechanism

Pullrods

Pullrods are utilized on the 2001 Lawrence Technological University Formula SAE car instead of pushrods, which are typically used on Formula SAE vehicles, for a weight reduction. Smaller diameter tubing can be used for pullrods because when the car hits a bump, the pullrod pulls the suspension mechanism, which puts it in tension. During rebound the pullrods are subjected to an insignificant buckling load equivalent to the weight of the wheel assembly. Therefore, buckling does not have to be considered in the design of pullrods for a Formula SAE vehicle. However, a pushrod is subjected to a buckling load whenever the car goes into rebound, so buckling must be considered in the design of pushrods. Also, pullrods allow the rockers and shocks to be packaged towards the bottom of the car resulting in a lower center of gravity.

A stress analysis of the pullrods using a 1.4 g tensile force resulted in a maximum stress of 118.89 MPa. A safety factor of 2.6 is obtained from this stress. Finite Element Analysis was also performed on the pullrods to verify the stress calculations and resulted in a safety factor of 2.5.

Rockers

Rockers, commonly called bellcranks, were placed on the 2001 Lawrence Technological University Formula SAE car so that the shocks could be packaged inboard. This packaging scheme significantly reduces unsprung mass. Also, rockers allow the pullrod and shock displacements to be in different directions, which aids tremendously in the packaging of suspension components. The rockers were designed to have a one to one motion ratio, which means that the shocks travels the same amount that the wheel moves in the vertical direction. A one to one motion ratio was selected because it allows the total range of shock displacement to be used, which improves the sensitivity of the suspension system.

The rockers were manufactured from 7075-T6 aluminum due to its superior weight to strength ratio. To reduce friction in the energy management system, roller bearings were placed in between the two rocker plates and thrust bearings were incorporated into the bellcrank plates at the pivot points.

Extensive Finite Element Analysis was performed on the rockers to minimize their weight. A safety factor of 4.9 was obtained for the final design of the front rockers when a force of 1.3 g was applied to them at the pullrod attachment point while restraining it from translating. A safety factor of 4.5 was obtained for the rear rocker when it was loaded in a similar manner.

Rocker FEA Results

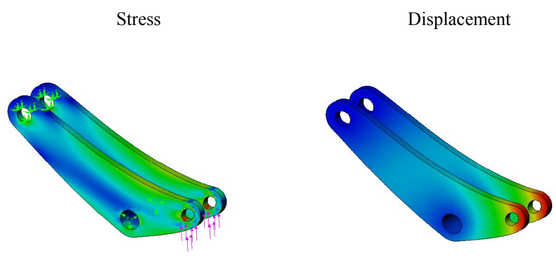


Figure 12: Rocker FEA Results

CONCLUSION

This paper addressed essential design considerations for a small formula-style open wheel racing car. These considerations were addressed properly because none of the suspension components on the 2001 Lawrence Technological University Formula SAE car failed during intensive driver's training or at the 2001 Formula SAE competition.

ACKNOWLEDGMENTS

The authors would like to thank the 2001 Lawrence Technological University Formula SAE team for all of their hard work and effort put into the project.

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APPENDIX

SAMPLE CALCULATIONS

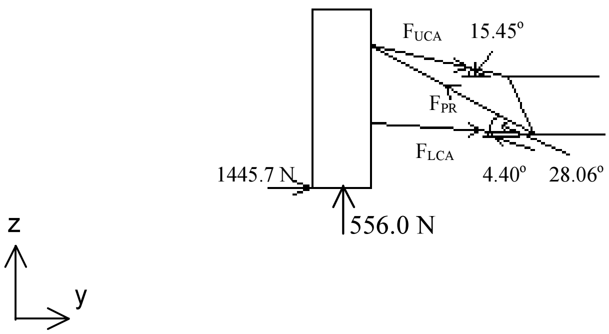
TOTAL WEIGHT

Target Vehicle Weight = 2224.11 N
Average Driver Weight = 711.72 N

Total Weight = 2224.11 + 711.72 = 2935.83 N

CONTROL ARMS

FRONT CONTROL ARM AXIAL FORCES



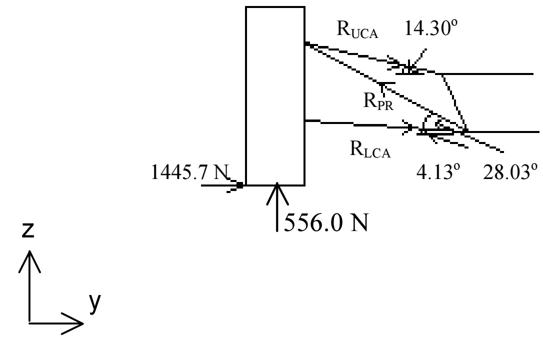
$$+\uparrow\sum F_z = 0 = 556.0 - F_{UCA}\sin 15.45^o$$
$$+ F_{PR}\sin 28.06^o - F_{LCA}\sin 4.40^o$$

$$+\rightarrow\sum F_y = 0 = 1445.7 + F_{UCA}\cos 15.45^o$$
$$- F_{PR}\cos 28.06^o + F_{LCA}\cos 4.40^o$$

$$\hat{+}\sum M = 0 = (1445.7)(381) + (F_{LCA}\cos 4.40^o)(238.8)$$
$$-(556.0)(21.8) + (F_{LCA}\sin 4.40^o)(4.1)$$

$$\underline{F_{LCA} = -2{,}259.7N}$$
$$\underline{F_{UCA} = -1{,}208.6N}$$
$$\underline{F_{PR} = -2{,}233.0N}$$

REAR CONTROL ARM AXIAL FORCES



$$+\uparrow\sum F_z = 0 = 556.0 - R_{UCA}\sin 14.30^o$$
$$+ R_{PR}\sin 28.03^o - R_{LCA}\sin 4.13^o$$

$$+\rightarrow\sum F_y = 0 = 1445.7 + R_{UCA}\cos 14.30^o$$
$$- R_{PR}\cos 28.03^o + R_{LCA}\cos 4.13^o$$

$$\hat{+}\sum M = 0 = (1445.7)(381) + (R_{LCA}\cos 4.13^o)(241.3)$$
$$-(556.0)(8.64) + (R_{LCA}\sin 4.13^o)(2.29)$$

$$\underline{R_{LCA} = -2{,}267.3N}$$
$$\underline{R_{UCA} = -1{,}060.5N}$$
$$\underline{R_{PR} = -2{,}088.0N}$$

BUCKLING CALCULATIONS

$$P_{cr} = \frac{\Pi^2 EI}{L^2}$$

$$E = 206.84GPa$$
$$I = 1723.20\times10^{-12}m^4$$

$$P_{MAX} = \left(1206.58\times10^6\right)\left[\left(\frac{\Pi}{4}\right)\left(0.0191^2 - 0.0176^2\right)\right]$$
$$P_{MAX} = 52{,}167.90N$$

Front Upper Control Arm

- Front Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.423)^2}$$

$$P_{cr} = 19{,}660.26N$$

$$SF = \frac{52{,}167.90}{19{,}660.26} = 2.65$$

- Rear Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.442)^2}$$

$$P_{cr} = 18{,}006.34N$$

$$SF = \frac{52{,}167.90}{18{,}006.34} = 2.90$$

Front Lower Control Arm

- Front Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.520)^2}$$

$$P_{cr} = 13{,}009.58N$$

$$SF = \frac{52{,}167.90}{13{,}009.58} = 4.01$$

- Rear Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.550)^2}$$

$$P_{cr} = 11{,}629.06N$$

$$SF = \frac{52{,}167.90}{11{,}629.06} = 4.49$$

Rear Upper Control Arm

- Front Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.461)^2}$$

$$P_{cr} = 16{,}552.67N$$

$$SF = \frac{52{,}167.90}{16{,}552.67} = 3.15$$

- Rear Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.497)^2}$$

$$P_{cr} = 14{,}241.55N$$

$$SF = \frac{52{,}167.90}{14{,}241.55} = 3.66$$

Rear Lower Control Arm

- Front Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.533)^2}$$

$$P_{cr} = 12{,}382.71N$$

$$SF = \frac{52{,}167.90}{12{,}382.71} = 4.21$$

- Rear Arm

$$P_{cr} = \frac{\Pi^2\left(206.84\times10^9\right)\left(1723.20\times10^{-12}\right)}{(0.543)^2}$$

$$P_{cr} = 11{,}930.82N$$

$$SF = \frac{52{,}167.90}{11{,}930.82} = 4.37$$

MOMENT CALCULATIONS

M=Fd

Bending Moment due to Vertical Force

- Front Upper Control Arm

$$M = (VF)(d) = \left(278.0\cos 25.45^0\right)\left(0.413\cos 25.45^0\right)$$

$$M = 93.61N-m$$

- Front Lower Control Arm

$$M = (VF)(d) = \left(278.0\cos 14.40^0\right)\left(0.504\cos 14.40^0\right)$$

$$M = 131.45N-m$$

- Rear Upper Control Arm

$$M = (VF)(d) = \left(278.0\cos 24.30^0\right)\left(0.438\cos 24.30^0\right)$$

$$M = 101.14N-m$$

- Rear Lower Control Arm

$$M = (VF)(d) = \left(278.0\cos 14.13^0\right)\left(0.413\cos 14.13^0\right)$$

$$M = 107.97N-m$$

Bending Moment due to Cornering Force

- Front Upper Control Arm

$$M = (CF)(d) = \left(841.16\cos 15.45^0\right)(0.117) = 94.86N-m$$

- Front Lower Control Arm

$$M = (CF)(d) = \left(2259.70\cos 4.40^0\right)(0.041) = 92.37N-m$$

- Rear Upper Control Arm

$$M = (CF)(d) = \left(1061.79\cos 14.30^0\right)(0.114) = 117.29N-m$$

- Rear Lower Control Arm

$$M = (CF)(d) = \left(2266.81\cos 4.13^0\right)(0.038) = 85.92N-m$$

Shearing Moment due to Braking Force

- Front Upper Control Arm
  $M = (BF)(d) = (711.72)(0.127) = 90.39 N-m$
- Front Lower Control Arm
  $M = (BF)(d) = (711.72)(0.127) = 90.39 N-m$
- Rear Upper Control Arm
  $M = (BF)(d) = (711.72)(0.127) = 90.39 N-m$
- Rear Lower Control Arm
  $M = (BF)(d) = (711.72)(0.127) = 90.39 N-m$

STRESS CALCULATIONS

$$S = \frac{My}{I}$$

$$I = 1723.20 \times 10^{-12} m^4$$

$$y = 9.525 \times 10^{-3} m$$

Bending Stress due to Vertical Force

- Front Upper Control Arm
  $$S = \frac{(93.61)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 517.43 MPa$$
  $$SF = \frac{1206.58}{517.43} = 2.33$$
- Front Lower Control Arm
  $$S = \frac{(131.45)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 726.59 MPa$$
  $$SF = \frac{1206.58}{726.59} = 1.66$$
- Rear Upper Control Arm
  $$S = \frac{(101.14)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 559.05 MPa$$
  $$SF = \frac{1206.58}{559.05} = 2.16$$
- Rear Lower Control Arm
  $$S = \frac{(107.97)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 596.80 MPa$$
  $$SF = \frac{1206.58}{596.80} = 2.02$$

Bending Stress due to Cornering Force

- Front Upper Control Arm
  $$S = \frac{(94.86)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 524.34 MPa$$
  $$SF = \frac{1206.58}{524.34} = 2.30$$
- Front Lower Control Arm
  $$S = \frac{(92.37)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 510.58 MPa$$
  $$SF = \frac{1206.58}{510.58} = 2.36$$
- Rear Upper Control Arm
  $$S = \frac{(117.21)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 648.32 MPa$$
  $$SF = \frac{1206.58}{648.32} = 1.86$$
- Rear Lower Control Arm
  $$S = \frac{(85.92)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 474.92 MPa$$
  $$SF = \frac{1206.58}{474.92} = 2.54$$

Shearing Stress due to Braking Force

- Front Upper Control Arm
  $$S = \frac{(90.39)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 499.63 MPa$$
  $$SF = \frac{1206.58}{499.63} = 2.42$$
- Front Lower Control Arm
  $$S = \frac{(90.39)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 499.63 MPa$$
  $$SF = \frac{1206.58}{499.63} = 2.42$$
- Rear Upper Control Arm
  $$S = \frac{(90.39)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 499.63 MPa$$
  $$SF = \frac{1206.58}{499.63} = 2.42$$
- Rear Lower Control Arm
  $$S = \frac{(90.39)(9.525 \times 10^{-3})}{1723.20 \times 10^{-12}} = 499.63 MPa$$
  $$SF = \frac{1206.58}{499.63} = 2.42$$

SPINDLES

FATIGUE ANALYSIS

Maximum Vertical Force = 2891.2 N
Minimum Vertical Force = 0 N

Maximum Bending Moment = (2891.2)(0.082)
= 237.08 N-m
Minimum Bending Moment = (0)(0.082)=0 N-m

$$I = \frac{\Pi}{4}\left(0.019^4 - 0.014^4\right) = 7.22\times10^{-8}\, m^4$$

Maximum Stress

$$S_{\max} = \frac{My}{I} = \frac{(237.08)(0.019)}{7.22\times10^{-8}} = 62.39\,MPa$$

Minimum Stress

$$S_{\min} = \frac{My}{I} = \frac{(0)(0.019)}{7.22\times10^{-8}} = 0\,MPa$$

Mean Stress

$$S_m = \frac{S_{\max} + S_{\min}}{2} = \frac{62.39+0}{2} = 31.20\,MPa$$

Average Stress

$$S_a = \frac{S_{\max} - S_{\min}}{2} = \frac{62.39-0}{2} = 31.20\,MPa$$

$$S_{UT} = 825\,MPa$$
$$S_{YT} = 701.25\,MPa$$
$$S_{1000} = 742.5\,MPa$$
$$S_e = 412.5\,MPa$$

Soderberg Approach

$$\frac{S_m}{S_{YT}} + \frac{S_a}{S_e} = 1$$
$$\frac{31.20}{701.25} + \frac{31.20}{S_e} = 1$$
$$S_e = 32.65\,MPa$$

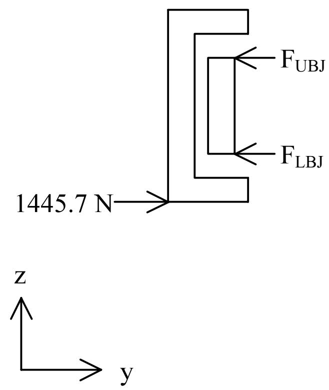
$$N = 10^{\left(\frac{-c}{b}\right)}(S_e)^{1/b}$$

$$b = \frac{-1}{3}\log\left(\frac{S_{1000}}{S_e}\right) = \frac{-1}{3}\log\left(\frac{742.5}{412.5}\right)$$
$$b = -0.085$$
$$c = \log\frac{(S_{1000})^2}{S_e} = \log\frac{(742.5)^2}{412.5}$$
$$c = 3.126$$
$$N = 10^{\left(\frac{-3.126}{-0.085}\right)}(32.65)^{1/-0.085}$$

N=1.39X10^{19} cycles

UPRIGHTS

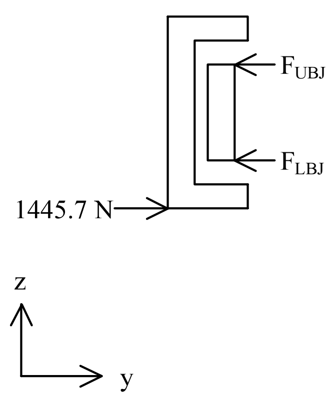
FRONT UPRIGHT CORNERING FORCE



$$\vec{+}\sum F_y = 0 = 1445.7 - F_{UBJ} - F_{LBJ}$$
$$\hat{+}\sum M = 0 = (1445.7)(0.142) + F_{UBJ}(0.239) = 0$$

$$\underline{F_{UBJ} = -859.95N}$$
$$\underline{F_{LBJ} = 2304.65N}$$

REAR UPRIGHT CORNERING FORCE

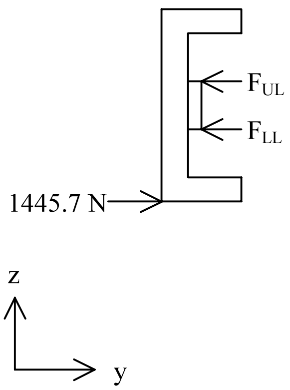


$$\vec{+}\sum F_y = 0 = 1445.7 - F_{UBJ} - F_{LBJ}$$
$$\hat{+}\sum M = 0 = (1445.7)(0.147) + F_{UBJ}(0.272) = 0$$

$$\underline{F_{UBJ} = -781.32N}$$
$$\underline{F_{LBJ} = 2227.02N}$$

HUBS

FRONT HUB CORNERING FORCE



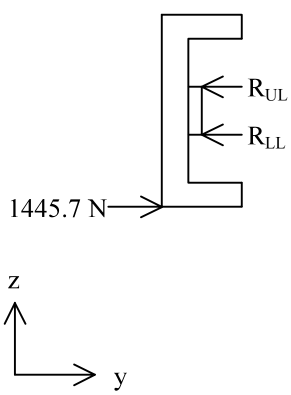
$$\vec{+}\sum F_y = 0 = 1445.7 - F_{LL} - F_{UL}$$

$$\overset{\frown}{+}\sum M = 0 = (1445.7)(205.0) + F_{UL}(98.0)$$

$$\underline{F_{UL} = -3024.17N}$$

$$\underline{F_{LL} = 4469.87N}$$

REAR HUB CORNERING FORCE



$$\vec{+}\sum F_y = 0 = 1445.7 - R_{LL} - R_{UL}$$

$$\overset{\frown}{+}\sum M = 0 = (1445.7)(205.0) + R_{UL}(98.0)$$

$$\underline{R_{UL} = -3024.17N}$$

$$\underline{R_{LL} = 4469.87N}$$

PULLRODS

TENSILE STRESS

$$\sigma = \frac{P}{A}$$

$$A = \frac{\Pi}{4}\left(0.0095^2 - 0.0077^2\right) = 2.432\times10^{-5}\,m^2$$

$$\sigma = \frac{2(1445.7)}{2.432\times10^{-5}} = 118.89MPa$$

$$SF = \frac{310.26}{118.89} = 2.61$$