SAE TECHNICAL PAPER SERIES

2002-01-3310

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Reprinted From: Proceedings of the 2002 SAE Motorsports Engineering Conference and Exhibition (P-382)

Motorsports Engineering Conference & Exhibition
Indianapolis, Indiana
December 2-5, 2002

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ISSN 0148-7191

Printed in USA

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Design of Formula SAE Suspension

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ABSTRACT

Formula SAE is a Student project that involves a complete design and fabrication of an open wheel formula-style racecar. This paper will cover the suspension geometry and its components, which include the control arm, uprights, spindles, hubs, and pullrods. The 2002 Lawrence Technological Universities Formula SAE car will be used as an example throughout this paper.

INTRODUCTION

The suspension system is one of the most important systems to consider when designing a FSAE car. All accelerations, either lateral or longitudinal, must be put to the ground through the tires, which are held in contact with the ground by the suspension system. The suspension system must therefore keep the largest contact tire patch at all times. If the suspension does a poor job of this, the car will not perform up to its full potential. A good suspension must therefore incorporate a good kinematics design to keep the tire as perpendicular to the pavement as possible, optimal damping and spring rates to keep the tire on the ground at all times, and strong components that do not deflect under the loads induced upon them.

During the design phase of the 2002 FSAE vehicle safety, durability, and maintainability were placed as top priorities. With use many computer aided design programs, and Finite Element Analysis software the max stress and deflection of each part was determined at various load conditions. The use of the software served as a valuable tool in the selection of the proper metal alloy, and the geometry of each part.

SUSPENSION DESIGN

DETERMINE LOADS

To properly design the suspension components the loads experienced by each part had to be determined. The FSAE vehicle will see many different loading conditions, it was decided that a spreadsheet should be set up to calculate the loads in each part under all possible inputs of both lateral and longitudinal accelerations. A free body diagram of one corner of the vehicle was used to solve equations for the reaction forces in each component (Figure 1). These resulting equations were then put into a spreadsheet that relates all suspension components and vehicle parameter to numerous acceleration conditions and calculates the loads in each component for those conditions (Figure 1). The max loading conditions observed were then used for a 'worst-case' analysis of every suspension component.

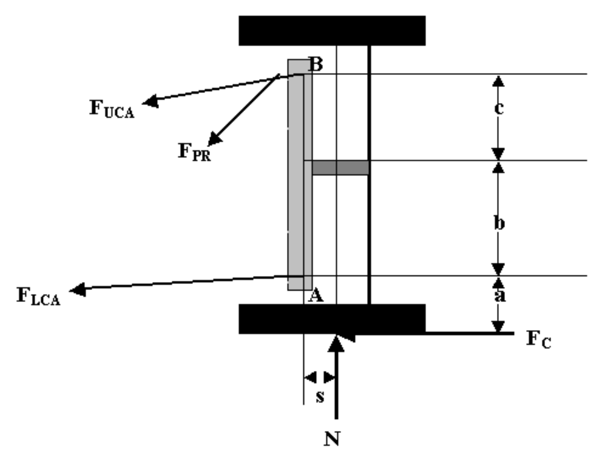

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Figure 1: Free Body Diagram

CONTROL ARMS

Control arms have the important task of connecting the uprights to the chassis. The control arm also plays a key roll in determining the camber and roll stability. The 2002 Formula Car utilizes rear control arms constructed of 19mm O.D. x .7mm wall thickness round 4130 chrome-moly tubing. This material was used on the 2001 FSAE and proved to be lightweight, strong, and easy to manufacture.

UPPER CONTROL ARM (UCA)

The upper control arms (Figure 2) use two spherical bearings at the frame mounts with an adjustable rod end at the upper ball joint to allow for camber adjustment. In order to make camber adjustments more efficiently while tuning the suspension, Quick Camber Adjustment (QCA) was designed and incorporated in all upper control arms (Figure 4). Length of the upper control arm has been determined to be 356mm, with a spread of 368mm

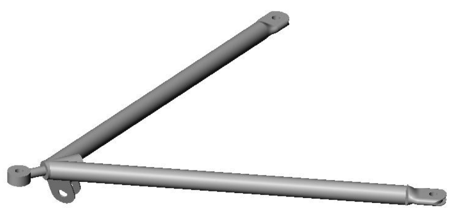

Figure 2: Upper Control Arm

LOWER CONTROL ARM (LCA)

The lower control arms (Figure 3) utilize two spherical bearings at the frame mounts, a spherical bearing at the lower ball joint, and an adjustable rod end at the toe link to allow for rear tow adjustment. The lower control arm will also incorporate the toe link on the front of the control arm. This will allow for rear toe adjustment and will also transmit most of the rear suspension load into the central part of the chassis, where it is strongest. Length of the lower control arm has been determined to be 460mm, with a spread of 355mm.

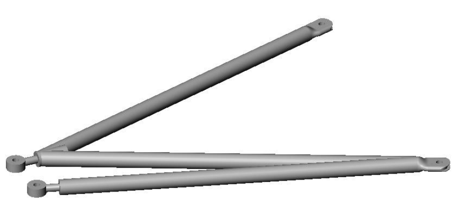

Figure 3: Lower Control Arm

CAMBER ADJUSTMENT

Camber is the angle between the wheel plane and the vertical, perpendicular to the pavement. Negative camber is when the top of the tire leans inward. If the tire leans outward that is considered to be positive camber. Camber is used to maintain the maximum tire contact patch The QCA system will allow the driver to adjust the camber quickly for varying driving conditions.

Figure 4: Quick Camber Adjustment

Analyzing the Control Arms

The control arms were analyzed at the maximum load, and buckling analysis was performed using Euler's buckling equation.

	Upper Control Arm		Lower Control Arm		
	Tube 1	Tube 2	Toe Link	Tube 1	Tube 2
Length (mm)	403.979	398.148	481.650	504.546	481.650
Le (mm)	807.959	796.295	963.300	1009.091	963.300
Max Force (n)	2053.541	2165.743	2791.429	1166.745	1620.092
Tube O.D. (mm)	19.050	19.050	19.050	19.050	19.050
Wall Thickness (mm	0.711	0.711	0.711	0.711	0.711
E (Pa)	2.06E+11	2.06E+11	2.06E+11	2.06E+11	2.06E+11
I (mm4)	1725.111	1725.111	1725.111	1725.111	1725.111
Pcr (kg)	550.000	566.000	387.000	352.000	386.000
Safety Factor	2.625	2.560	1.357	2.959	2.338

Table 1: Control Arm Buckling Analysis

UPRIGHTS

The uprights provide a link between the upper and lower ball joints. The weight of the uprights must be minimized. During a bump the weight of the upright are controlled by the shocks.

FRONT UPRIGHTS

The Front uprights connect the upper and lower ball joints of the control arm, also provides a mounting point for the break calipers. The weight of the front uprights was minimized and the strength was maximized by the use of Finite Element Analysis. The Front Uprights were machined out of one solid aluminum block of 7075 T6, which provided excellent strength to weight ratio.

The Front Uprights were analyzed by using the center of the spindle mount fixed. Forces were applied at the upper and lower ball joints and applied at the caliper mounting holes. The forces applied simulated a cornering force of 1.3g, 895N was applied to the upper ball joint and -2304.7N was applied to the lower ball joint, also a force of 1121.1N was applied to the caliper mounting holes.

The Final design used triangular pockets to lighten the Upright. Finite Element Analysis showed that triangular pockets provided the greatest strength with the least amount of deflection. After several iterations the result of the final upright was 1.16 kg. The Front Upright had a Safety Factor of 2.5 while a combined cornering and braking force was applied.

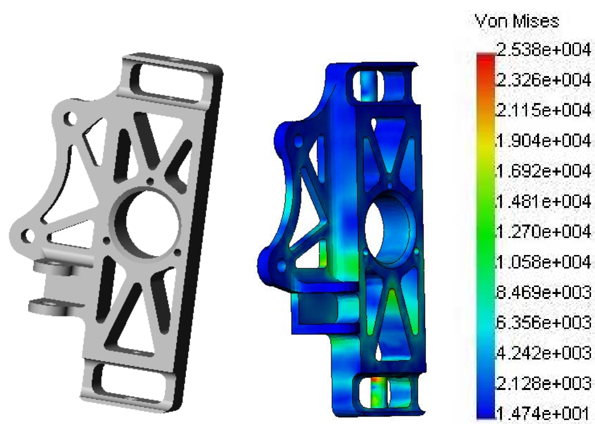

Figure 5: Front Uprights

REAR UPRIGHTS

The rear uprights were analyzed using Finite Element Analysis. This technique was used to maximize performance while minimizing weight. The rear uprights are analyzed at braking conditions, and combined braking and cornering.

To analyze cornering force the center of the upright is held fixed and a load of 781.3N was applied to the upper ball joint and a force of -2227.0N applied to the lower ball joint.

Simulating braking force 224.1 N was applied at the lower ball joint. The center of the Upright is held fixed. This force is different that the force used to analyze the front uprights because the use of a three rotor braking system. With the use of a three rotor braking system there is only one rotor located in the rear. The rear rotor does not apply a torque on the rear upright, it is absorbed by the chassis, therefore it only reacts to acceleration and deceleration forces.

Through the help of Finite Element Analysis and bench marking the uprights used years prior, we were able to provide maximum strength and minimize weight by eliminating small triangular pockets. Weight of the 2002 rear upright is .932 kg.. Max deflection and stress were analyzed using the loads calculated in the previously mentioned spreadsheet and FEA software. Results of the analysis show .5mm max. deflection with a max. stress of 174Mpa. The Safety Factor determined was 3.2 at worst case conditions, which was cornering combined with braking.

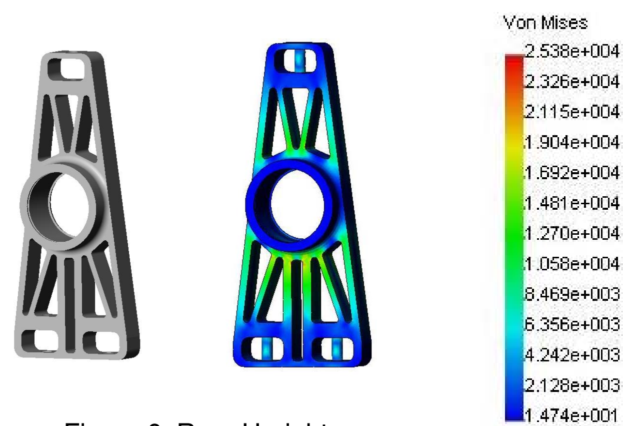

Figure 6: Rear Uprights

SPINDLES

The purpose of the spindle is to provide a base about which the wheel assembly rotates. The spindle is a part that has caused prior teams problems with excessive wear. Due to the many load the spindle experiences with cross car weight transfers, and many changes in road conditions this part must be analyzed extensively. When designing and fabricating the spindle the outer diameter where the bearing mounts must be fabricated with little tolerance (+0. 100mm / -0. 000mm).

The spindles were manufactured out of 4340 steel due to excellent wear resistance properties. While manufacturing mating parts dimensions had to be taken into account. The mating parts include the bearing and the upright. Similar to years before a step in the spindle was used the same thickness of the bearing so that impact loads would be distributed though the entire bearing. Another step was used to free the end of the spindle so that a washer and castle nut could be tightened down on the bearing and the hubs.

Finite Element Analysis was used to analyze the spindles at maximum braking force and vertical force was used. The force exerted during a braking maneuver is 1121 N and 1445 N of force applied at the tires during cornering. The results revealed 25.8 safety factor during breaking, and 2.8 during cornering. When combining cornering force and braking forces a safety factor of 2.6 was maintained.

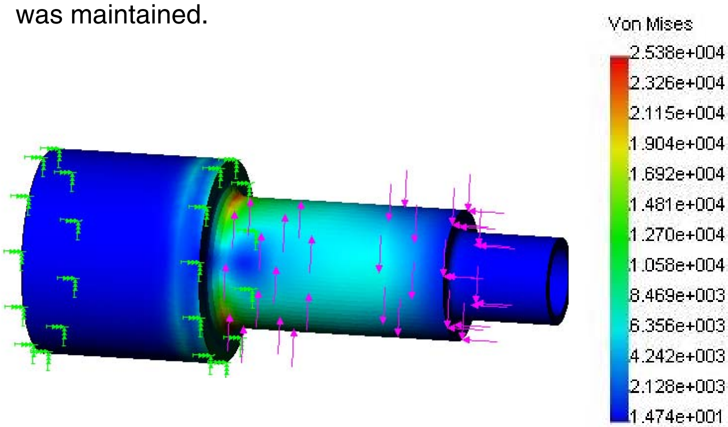

Figure 7: Spindle (FEA results)

FRONT HUBS

The Front hubs were manufactured out of 70785-T6 aluminum. The hubs are designed at a 4-leaf 45-degree offset. The front hubs provided mounting holes for the front brake rotors. The brake rotors have a mounting tab that is applied to the hub, which allows the brake rotor to float. The floating of the brake rotor allows for perfect caliper alignment and evenly wears the brake pads.

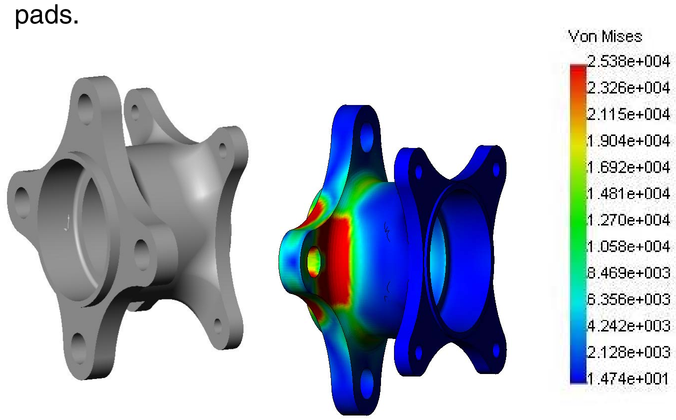

Figure 8: Front Hub

REAR HUBS

The main function of the rear hub is to connect the driveline to the wheels through the half shafts and the constant velocity joints. The 2002 rear hubs were designed similar to the 2001 rear hubs. The major changes include the use of threaded studs instead of press fit studs for a more accurate wheel alignment and the use of a larger radius on the back face to help with stresses and deflection. The material chosen for the rear hubs was 4340 steel due in large part to its high yield strength of 910Mpa. The rear hubs showed a maximum deflection of .101mm and a maximum stress of 543.7Mpa when analyzed with a 2-g cornering load using FEA

With the use of Finite Element Analysis a 4-leaf clover design with a 45-degree offset was developed. A combination loading of cornering and breaking was applied to the hub. To simulate a cornering force a load of 1445 N a load of 4469 N was placed on the bottom lug while at the same time a load of 3024 N was placed on the top lug. The hub held a safety factor of 4.2 during these loading conditions.

Similar to years before a bearing was used to reduce friction between the upright and the shaft on the hub. As a safety measure a spindle lock nut was used at the end of the CV joint, this made sure that the hub would not disengage from the driveline.

DAMPING MECHANISM

The damping systems consist of the following components; pull rods, suspension mechanism or bell crank, and the shocks. The purpose of this system is to transfer load from the wheel to the inboard shock. The geometry of this system is critical because it determines the motion ratio, which is the amount of movement at the wheel compared to the amount of movement at the shock. This ratio is used to determine the cars natural frequency. It is critical that these parts have a very low friction factor, therefor transferring the load directly to the shock without putting excess stress on the individual parts.

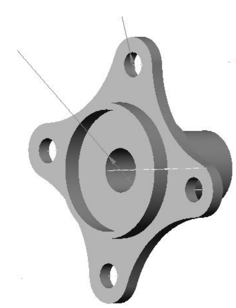

Figure 9: Rear Hub

SUSPENSION MECHANISM

The suspension mechanism is commonly called the bellcranks. The bellcrank aides packaging, it allows the pullrod and the shock displacement to be aligned in different directions. Unlike the 2001 FSAE car, this year the front shocks were positioned vertically in the same direction as the force, to stress. The motion ration in the front was a 1 to 1 ratio. This means the shock moves the same distance at the wheel.

The bellcranks were manufactured out of 7075-T6 aluminum due to superior strength to weight ratio. Similar to the 2001 car roller bearings were used in between two bellcranks and thrust bearings were used at the pivot points.

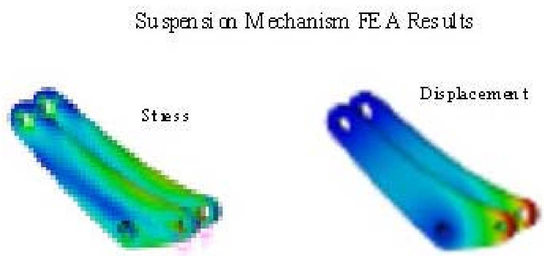

Figure10: Bellcrank FEA results

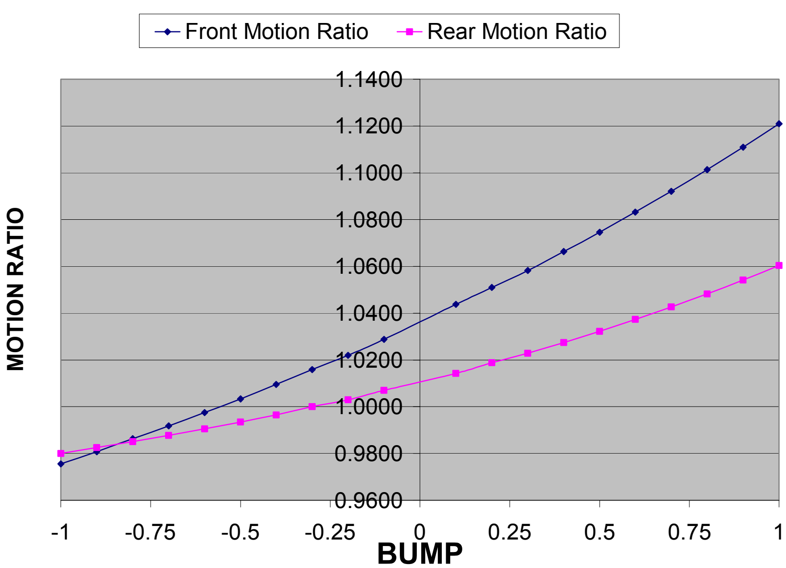

Figure 11: Motion Ratio

PULLRODS

The Pullrods oppose the pushrods, during a bump the pullrod pulls the bellcrank and compresses the shock. The force experienced by the pullrod is mainly tensile, however during jounce the pullrod experiences minimal buckling which consists of the weight of the wheel. Because the buckling force is minimal the buckling analysis can be ignored. The Pullrod allows the shock to be positioned in any geometry needed for packing issues. The pullrod has allowed the shocks to be positioned at a low center of gravity.

Finite element analysis was used to analyze the pullrod using a tensile force of 1445.7 N resulted in a maximum stress of 118.89Mpa A safety factor of 2.6 was determined.

CONCLUSION

This suspension analysis is to be used as a guideline for future FSAE suspension teams. This suspension has been put through rigorous testing and has yet to fail on the 2002 Lawrence Technological University FSAE car. Each component has been improved through the years either made lighter or stronger with the use of computers, and finite element analysis (FEA).

ACKNOWLEDGEMENTS

The authors would like to thank the 2002 Lawrence Technological University Formula SAE Car team for all their effort put into this project.

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