Final Project Report
MECH 5220 – Virtual Prototyping
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2010 BAJA SAE SUSPENSION

Tripp Schlereth **Stephen Sparks**

ABSTRACT

Suspension design is one of the most complex systems on a Baja SAE vehicle. The terrain that must be covered is extreme and the horsepower is limited. Suspension design for this competition is one of the most varied items seen at race. As many people have not seen a Baja SAE vehicle, below is an image of the 2010 car during testing.

Figure 1

INTRODUCTION

Coming off a bad season, the 2010 Baja SAE vehicle needed some massive redesign. Repetitive rear suspension failures and ultimately a drivetrain failure crippled the 2009 car. Suspension redesign was a massive part of the evolution of the 2010 car. For over 5 years Auburn Baja has run a double wishbone (unequal length, non-parallel) rear suspension with an integrated rear steer point. This rear steer point helped eliminate the understeer produced by the spooled rear drive. The front end of Auburn Baja cars have always been a double wishbone setup utilizing rack and pinion steering. The main choice for this setup is the ability of a SLA double wishbone to control camber. This report is written by Stephen Sparks, drivetrain designer for 2010, and Tripp Schlereth, suspension designer for 2010.

NOMENCLATURE

The common axis system for a road vehicle defines the X as the longitudinal axis, Y as the width or lateral axis, and Z as the vertical axis. SLA, or short long arm, is a double wishbone front suspension setup that controls camber by running unequal length upper and lower arms. It also provides a predictable camber gain in bump travel. Camber is the angle of the tire from in the Y-Z plane. The tires lean top in for negative camber and top out for positive camber.

FRONT SUSPENSION

The only set rule for Baja suspension is that the track width cannot exceed 64 inches. The 2009 car was close to that limit and as it was as wide as the pathway out of the shop, it was hard to maneuver. On a tight track, like the site of the 2009 Alabama SAE competition at the NCAT test facility, the wide track width might hinder the ability to pass other cars. The 2010 car was designed to have a front width of about 52 inches at ride height. Through design refinement, it has been proven that 10 inches of wheel travel is enough to dissipate the energy from the track and smooth out the ride enough to traverse standard track terrains. Due to budget constraints and the fact that the 2009 vehicle destroyed its gearbox, the choice was made to reuse the Custom Axis shocks from that car. Once some of the design specifications were set, the actual design evolution could begin.

The Baja team has as sponsorship from Lotus Engineering Software for their suspension modeling program Shark. Shark is a front for the Adam’s program that is tuned for a suspension designer. The image below in Figure 2 shows a screenshot of the Lotus Shark interface.

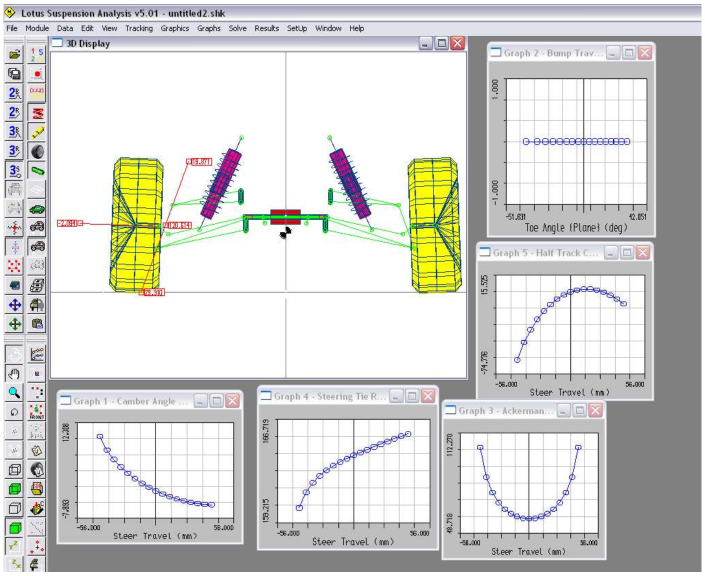

Figure 2

The basic principle the 2010 design is to achieve camber gain in roll. In other words, as the car corners, the goal is to gain a degree of camber for every degree of body roll. That way the car maintains a predictable handling characteristic. From analytical and experimental values, the design standard for this car is 7 degrees of roll. Shown below in Figure 3 is a graph of camber vs. roll angle. As shown, the vehicle gains 9 degrees of negative camber at full roll. Figure 3-b shows the front end at ride height, full droop, full bump, and halfway to each extreme. It is meant to demonstrate the camber gain through bump travel.

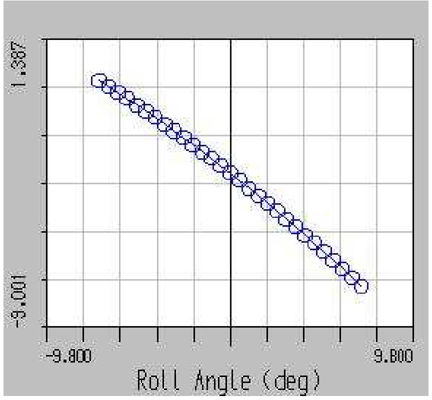

Figure 3- Camber (deg) vs. Roll Angle (deg)

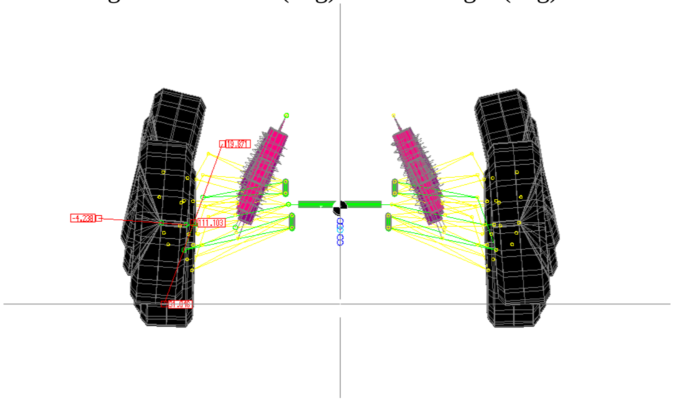

Figure 3-b Front end travel ranges

Baja vehicles, while not required in the rules, are made to jump on the competition tracks. In order to allow a car to jump and maintain control, the toe angle must remain constant through bump. This is commonly referred to as bump steer. If a tire in the system cycles up the toe angle needs to stay neutral to avoid steering the vehicle. Figure 4 shows a plot of suspension travel vs. toe angle. It can be seen on the X axis that the 2010 Baja car achieves this goal by acquiring only 2 degrees of toe change through 10 inches of wheel travel.

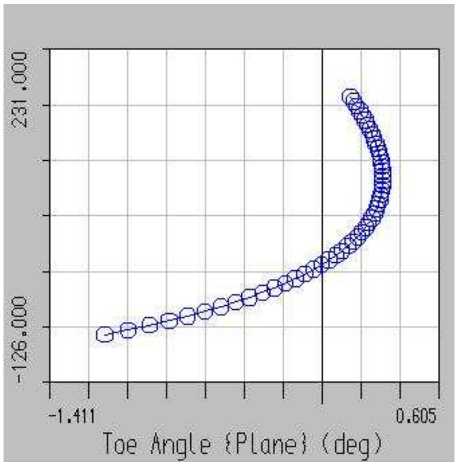

Figure 4- Suspension Travel (mm) vs. Toe Angle (deg)

There are several other key points to this year’s design. First, the scrub radius was reduced. Scrub radius refers to the distance from tire centerline to the centerline of the kingpin at ground height in the Y-Z plane. Scrub radius affects steering effort at low speeds and can cause premature steering component wear and driver fatigue. Figure 5 shows that this design has around 1 inch of scrub radius. (37.38mm)

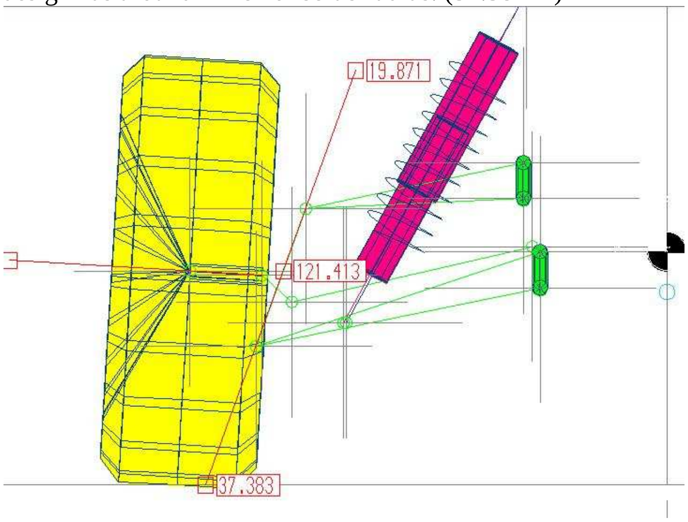

Figure 5- Front Right Corner

The kingpin on this years car was inclined not only in the YZ plane for scrub radius reduction, but in combination with a castor of 4.5 degrees, allows for a camber gain in steer. Figure 6 shows camber gain versus steer rack travel. The graph is meant to show that the inside tire gains almost 8 degrees of positive camber while the outside tire remains negative at 3 degrees camber. If one were to view this car from the front while it was steered, it would appear that the inside tire is laying down more than the outside tire, and both tires would be leaning away from the turn. This provides more force on the tires so that they can better push the car into turning.

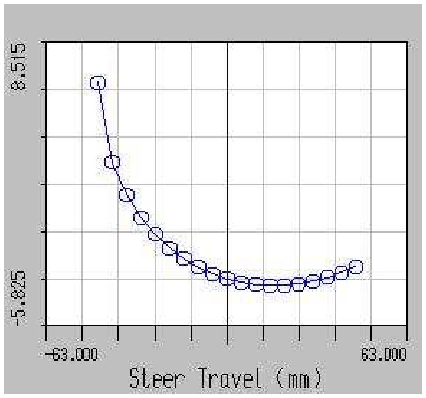

Figure 6- Camber (deg) vs. Steer Travel (mm)

A study of castor effects and kingpin inclination on camber gain in steer travel is attached as appendix 1. Besides castor, the other major component of steering is Ackermann. The Ackermann principle deals with the ratio of toe angles between the inside and outside tires of a car. In true Ackermann steering, the inside tire would turn twice what the outside turns. Vehicles that need a tight turning radius and turn at low speed (i.e. 5-10mph) run true Ackermann steering. One of these vehicles is a London Taxi. Since the Baja car operates slightly above speeds this, past designs have proven that 50% Ackermann steering is acceptable for these cars. Shown in Figure 7 below, is a graph of Ackermann change in steer travel. After calculation this car achieves an 82 percent change in Ackermann. Computing from the equations for Ackermann and using max toe angles, the Ackermann percentage is close to 60%.

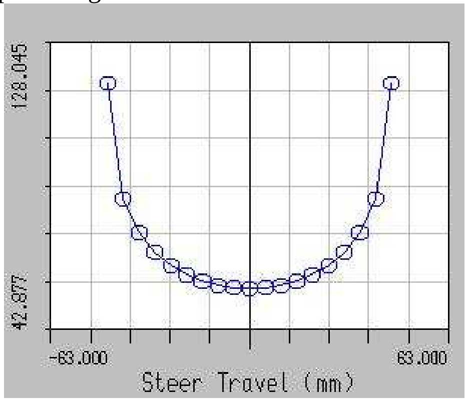

Figure 7- Ackermann Percentage vs. Steer Travel

Ackermann steering is normally achieved by levering the steering point off the front knuckle. Figure 8 shows the location of this point.

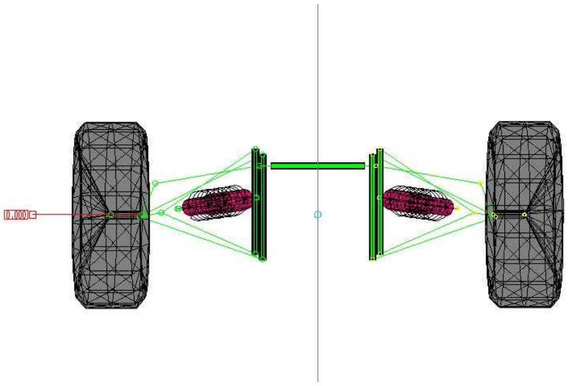

Figure 8- Front suspension in top view

In order to place this steer point where the design dictated, and allow for the kingpin inclination designed into the system, no stock ATV or UTV knuckle would work. Therefore I designed and the team machined a custom front knuckle out of 7075-T651 aluminum. The points of the upper ball joint, lower ball joint, spindle, and steer point were taken into Solid Edge where a design was conceived as shown in Figure 9.

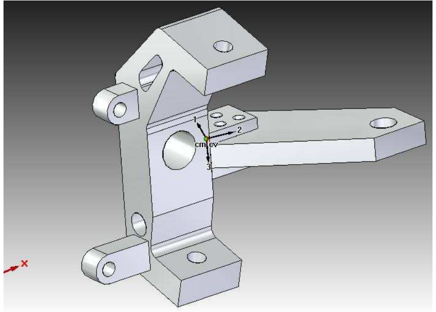

Figure 9- Initial front knuckle design in Solid Edge

The knuckle was then FEA’ed in FemPro (ALGOR) and SolidWorks CosmosXpress for failure in steering loads and failure in spindle shear loads. After two days of testing and a rollover, a knuckle was failed as shown in Figure 10.

Figure 10-failed front left knuckle at steer mounts

The failure was analyzed and determined to be cause by the location of the bolts for the steer arm and the thickness of the mounting tabs capturing the steer arm. Shown below in figure 11 are images of the old and new knuckle designs in ALGOR. The stress concentration wend down from 2.4e2 N/mm^2 to

2.1e2 N/mm^2, and the location of this max stress moved away from the bolts and into the actual knuckle.

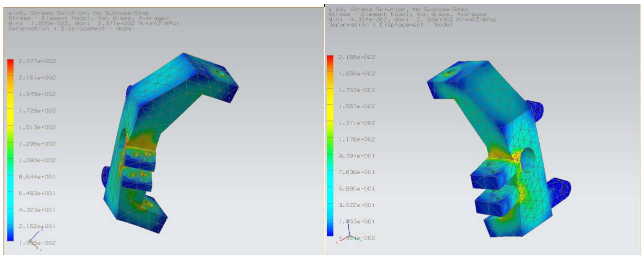

Figure 11- Original (left) and new (right) knuckle design

The same failures were seen in the upper front A-arms. Original Designs shown in Figure 12 below incorporated large radius bends in the arm.

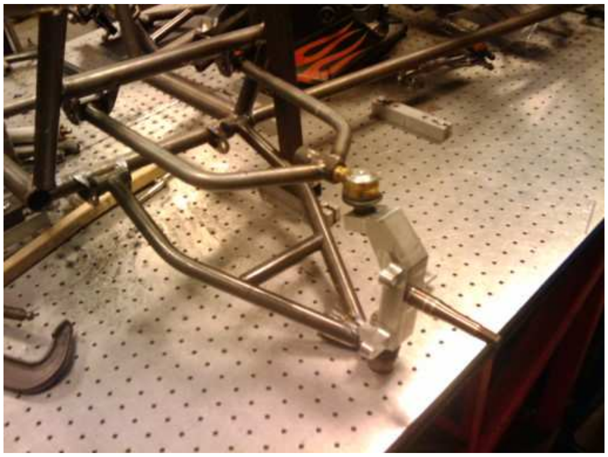

Figure 12- Front Suspension Design (left/driver corner)

The most complicated part to fabricate as it is not cut on a CNC, and often the most commonly broken part, is a suspension control arm. These parts are damaged easily for two reasons. First, they hang off the frame and are put into harm's way in order to get the tires to the optimal location for traction. Second, the arms are designed to be weaker than the supporting frame structure. In race, an A-arm is not incredibly hard to replace—3 bolts and the arm is free from the car. If a frame member were to deform, the frame could be disqualified as being structurally unsound. Therefore, A-arms must tread between failing easily and being overbuilt. FEA analysis is vital to a designer of this system. The design of 2010's a-arms started with analyzing failures of past arms. Shown below in Figure 13 is an image of Tripp holding an upper arm from the 2008 car and comparing it to the loads it takes to deform the arm in that manner in Solid Works.

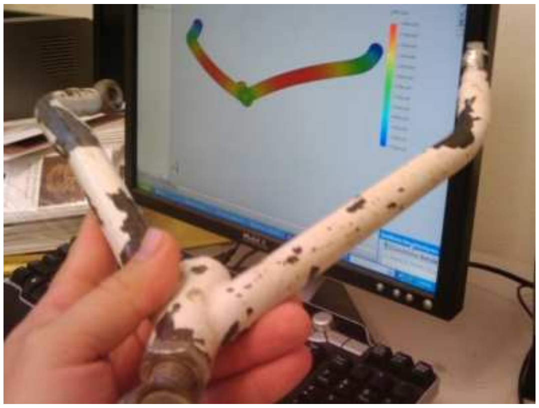

Figure 13- 2008 Upper a arm

It was determined that the load needed to deform that arm was 1200lbf in the x direction. After testing with arms designed for that load, we saw deformation in a bend as shown in Figure 14.

Figure 14- Deformation of upper a-arms

The current arm design, as shown in Figure 15, alleviates the bend that caused the deformation.

Figure 15- Front Passenger corner

The current front suspension design has survived over 5 high speed rollovers.

REAR SUSPENSION

The rear of the 2009 Baja SAE vehicle was the cause of much frustration at race. Due to the location of an upper a-arm mounting location, CVT clearance was minimal and arm strength was compromised. While the double wishbone setup worked well when Auburn Baja ran a chain drive, the implementation of a gearbox has made the rear frame design almost impossible to incorporate a suspension mount into. Because of this, a new design was started. The system needed to match the front in its ability to gain camber in roll. It also must provide adequate clearance for a CVT and gearbox and not hinder the removal of the CVT secondary pulley. For these reasons the team decided to implement a semi-trailing arm design. Normally frowned upon in current automotive design for its lack of overall camber control, it was at one time implemented in the BMW Z3 and Porsche 911 along with many cars running a McPherson Strut configuration in its front end. Shown below are the characteristic curves of the rear suspension.

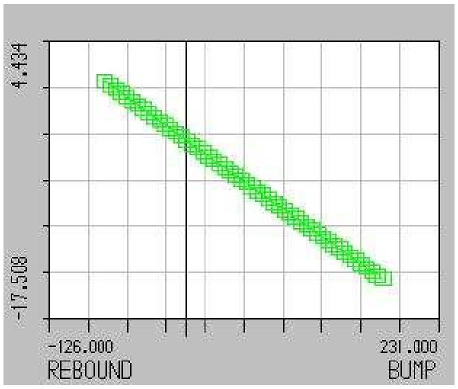

Figure 16 -Camber (deg) in bump (mm)

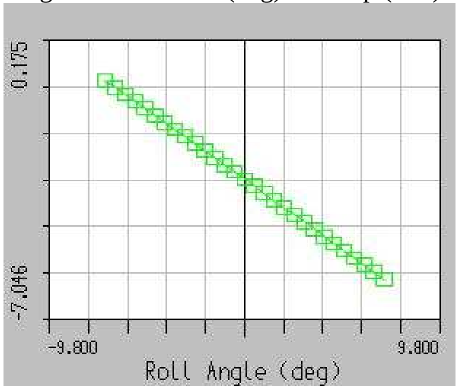

Figure 17 - Camber (mm) in roll (deg)

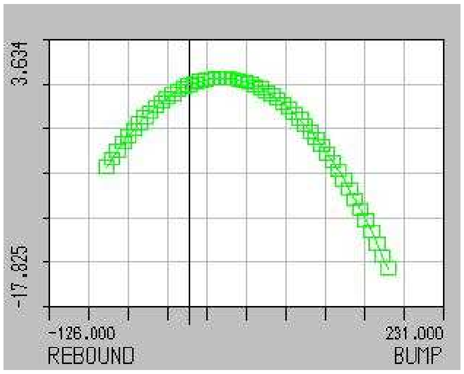

Figure 18- Wheelbase change (mm) in bump travel (mm)

The semi trailing arm design is nice in the way it clears the drivetrain of suspension points all together. A top view of the rear suspension design is shown in figure 19 below.

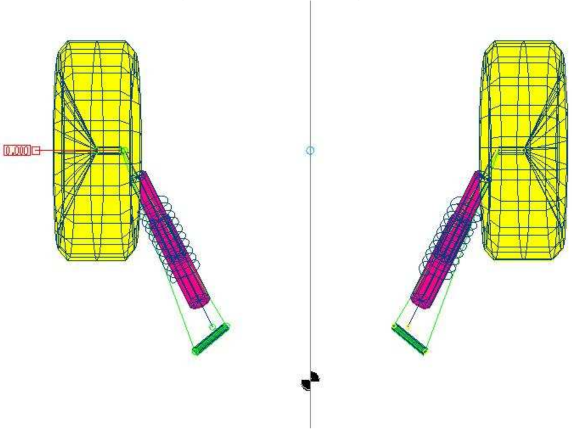

Figure 19 - 2010 Rear suspension top view

A rear view is shown next.

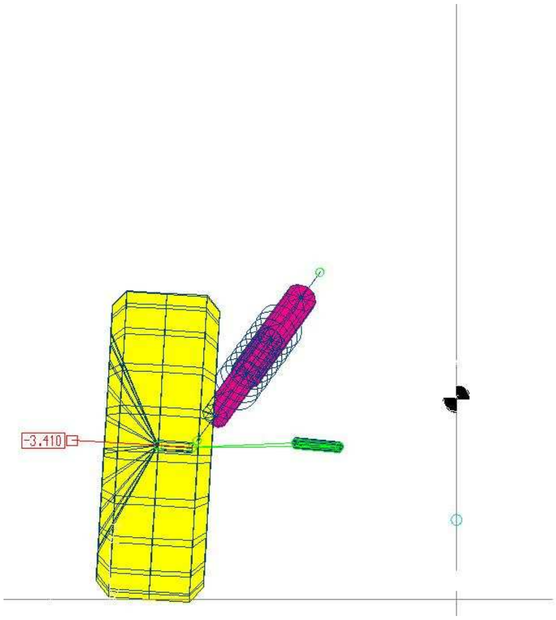

Figure 20 - 2010 Rear suspension rear view

The rear camber is shown via figure 21 where stills are overlaid of full droop, ride height, full bump, and halfway points.

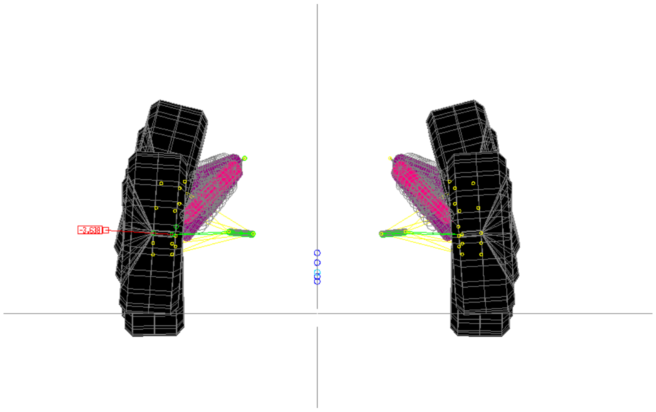

Figure 21 - Rear end throughout travel range

A benefit of a non steering axle, and of a semi trailing arm, is the lack of a toe change in bump travel. As shown below, the toe change is minimal throughout travel.

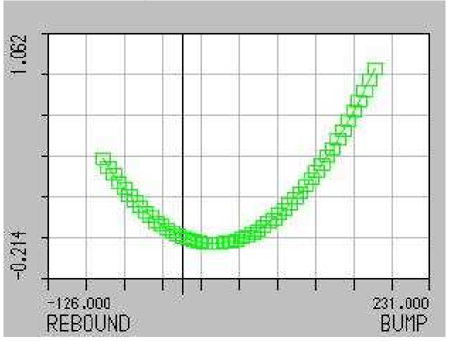

Figure 22- Toe Change (deg) vs. Bump Travel (mm)

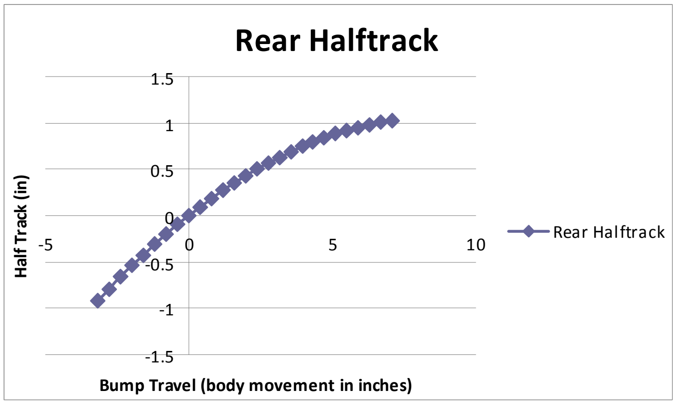

Figure 23 – Rear Halftrack change (inches) vs. Bump Travel (inches)

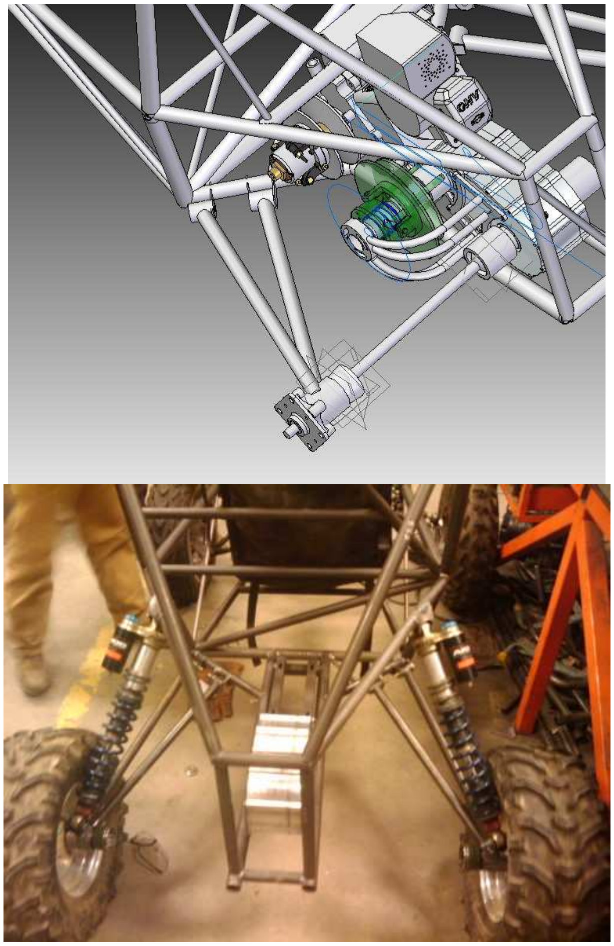

Figure 24-Rear End in Solid Edge and during early assembly

FULL CAR DYNAMICS

Now that the dynamics of each end of the car have been assessed individually using the same design scheme, it is time to view the full car. Shown below in Figure 25 are the camber images from Lotus overlaid on each other. It is clearly visible that neither end of the vehicle is gaining significantly more camber than the other end. This will aid in handling and feel of the vehicle.

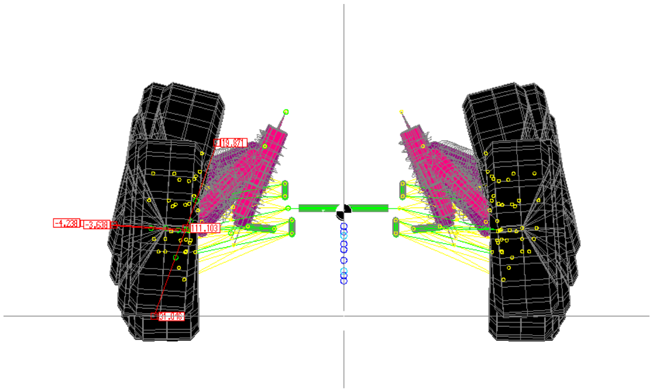

Figure 25- Full Car Camber Gain

This camber characteristic was then further checked by importing a data file from Lotus into Microsoft Excel and graphing the camber curves together. Shown in Figure 26 are both front and rear camber characteristic curves in inches.

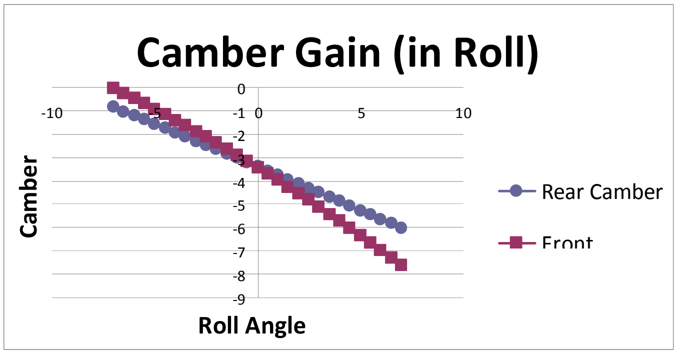

Figure 26- Camber Gain (deg) vs. Roll Angle (deg)

Another benefit of Lotus is the full dynamic model of the vehicle. Shown below is an image of the full car suspension model.

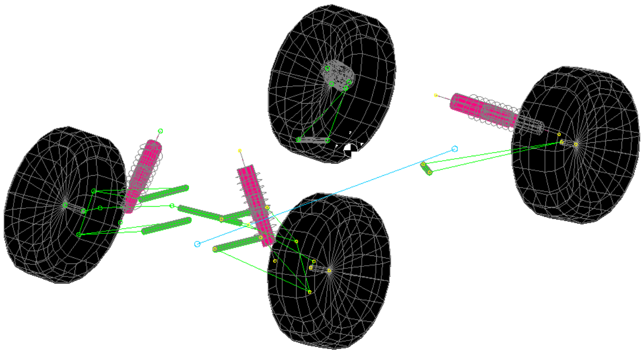

Figure 27- Full Car Shark Model

Also shown in Figure 27 is the roll axis. This is the one downfall of this year's semi-trailing arm. While trying to match camber gain curves, the rear roll axis ended up lower than the front roll axis. This is limiting the amount to load transfer to the front end of the vehicle which affects its handling and cornering characteristics. Figure 28 shows a final Solid Edge assembly of each modeled suspension component, frame, and drivetrain.

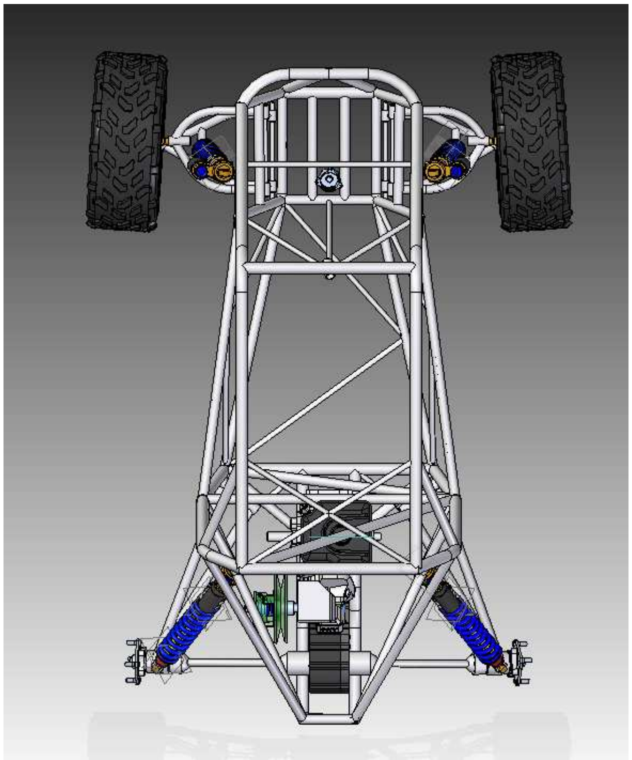

Figure 28- Full Car in Solid Edge

ACKNOWLEDGMENTS

All data contained in this file is collective knowledge of Auburn Baja. While it was all written by two members of the 2010 team, some of the past research into camber characteristics and overall suspension travel must be credited to previous year team members.

REFERENCES

All graphs and data used in this document are property of Auburn Baja. All data provided from Lotus Shark software was created and compiled by Tripp Schlereth. Vehicle models were compiled by Tripp Schlereth, Stephen Sparks, and Blake Sparks.

APPENDIX A

EFFECTS OF CASTOR AND KINGPIN INCLINATION ON CAMBER GAIN IN STEER

This appendix deals directly with the effects of a large kingpin inclination and castor adjustment. The data is tabulated from a .sdf output file generated by Lotus Shark software. Original setup is 4 deg. castor and -3.5 deg camber. For maneuvering, the plan is to adjust castor to 11.75 degrees. As the tables show, the large castor value allows the outside tire to gain camber from static -2 degrees to a value of 5. The inside tire would also plant better due to the 10 degrees of positive camber gain it achieves. The left column in each table is steer travel. The header shows the current settings. The right column shows camber at respective steer travel values.

castor 3.9	camber - .5
40	-0.292573243
35	-0.52868849
30	-0.709188044
25	-0.833821714
20	-0.901830435
15	-0.911852837
10	-0.861790299
5	-0.748609245
0	-0.56804812
-5	-0.314167261
-10	2.14E-02
-15	0.450492471
-20	0.990904391
-25	1.67033505
-30	2.53613043
-35	3.68155694
-40	5.35021305

castor 0	Camber -3.5
40	-1.69821882
35	-2.08571815
30	-2.42696953
25	-2.72145271
20	-2.96825337
15	-3.16598272
10	-3.31266117
5	-3.40555382
0	-3.44092464
-5	-3.41366243
-10	-3.31668615
-15	-3.13995028
-20	-2.86865926
-25	-2.47976041
-30	-1.93410146
-35	-1.1550411
-40	5.53E-02

castor 3.9	camber - 2
40	-1.80145812
35	-2.04063034
30	-2.22392368
25	-2.351089
20	-2.42137051
15	-2.43341064
10	-2.38511729
5	-2.27346563
0	-2.09420681
-5	-1.84142005
-10	-1.50680017
-15	-1.07846344
-20	-0.538804054
-25	0.139720276
-30	1.00401258
-35	2.14604831
-40	3.80359864

castor 2	camber -2
40	-1.19312441
35	-1.49415541
30	-1.74349785
25	-1.94076204
20	-2.0851028
15	-2.17513084
10	-2.2087872
5	-2.18315887
0	-2.09420681
-5	-1.93635178
-10	-1.70181131
-15	-1.37949347
-20	-0.953010261
-25	-0.396774441
-30	0.332723975
-35	1.32069564
-40	2.78704691

castor 0	camber -.5
40	1.16092396
35	0.776453555
30	0.437908232
25	0.145783991
20	-9.90E-02
15	-0.295172662
10	-0.440692306
5	-0.532885134
0	-0.56804812
-5	-0.541105688
-10	-0.445013404
-15	-0.26975885
-20	-5.63E-04
-25	0.385664523
-30	0.928305507
-35	1.70499086
-40	2.91934085

castor 4	camber -3.5
40	-3.1330409
35	-3.37489581
30	-3.56063986
25	-3.69002557
20	-3.76229882
15	-3.77610636
10	-3.7293613
5	-3.61904645
0	-3.44092464
-5	-3.18909192
-10	-2.85526943
-15	-2.42761874
-20	-1.88861561
-25	-1.21088481
-30	-0.347910553
-35	0.791161597
-40	2.43912601

castor 0	camber -2
40	-0.357919872
35	-0.744004428
30	-1.08399129
25	-1.37737191
20	-1.62324417
15	-1.82023132
10	-1.96636784
5	-2.05893326
0	-2.09420681
-5	-2.06709409
-10	-1.97053087
-15	-1.79448688
-20	-1.52417409
-25	-1.13652062
-30	-0.592265606
-35	0.185698181
-40	1.39794219

castor 11.75	camber 2
40	-5.00717258
35	-4.92181492
30	-4.75676298
25	-4.51338625
20	-4.19209337
15	-3.79222536
10	-3.31188226
5	-2.7476542
0	-2.09420681
-5	-1.34362769
-10	-0.484363437
-15	0.500599563
-20	1.63707078
-25	2.96639252
-30	4.5610857
-35	6.5702219
-40	9.42233181

Provided on this page is a compiled graph of the steer travel vs. castor and camber setting curves. The larger the range at each extreme, the more camber is gained through the same steer travel. It can be noted that the pink curve (the table highlighted in blue) would gain the most camber and would make the car more maneuverable than the purple (original design) curve.