2013 Bearcats Baja Braking System

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by:

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ABSTRACT

The purpose of this project is to design a braking system for a Baja SAE vehicle that can produce adequate braking force to meet competition regulations while being as light weight as possible. The system should also limit un-sprung weight to help improve maneuverability. Similar products and design of components will be researched for this project. A budget, timeline, proof of design and testing will also be looked at in this report.

INTRODUCTION

The main objective for the Baja SAE competition is to design and fabricate a one-man all-terrain vehicle having four wheels that could compete with manufactured versions from the viewpoint of safety, appearance, design, performance, and ultimately cost. The University of Cincinnati Bearcats Baja team will be competing in the 2013 Tennessee and New York Baja SAE competitions.

The main objective for this year’s team was to design a small lightweight vehicle to out-perform the competition while being as safe as possible. To meet a vehicle weight goal of 380 pounds the team is focusing on a new lightweight frame, simple lightweight brake system, lightweight front and rear suspension system, low center of gravity, and a robust drivetrain configuration.

BRAKE SYSTEM

OBJECTIVE - Design a braking system that can produce more than adequate braking force to meet Baja SAE competition regulations while being as light weight as possible. The system should also limit un-sprung weight to help improve maneuverability. The addition of a cutting brake will also help improve maneuverability.

BRAKING SYSTEM DESIGN – The brake system was designed to the rules, restrictions, and requirements provided by the SAE to ensure the vehicle can decelerate and stop within a reasonable distance (See Appendix E). The brake system was designed with a budget of $2000 and 20 lbs.

MASTER CYLINDER – The master cylinders play a large part in the design phase. In order to satisfy the deceleration goal of 0.9 g the master cylinders would have to be able to transfer the correct amount of pressure to the brake caliper pistons. The master cylinders chosen to do so are the rear master cylinders for a 2005 Polaris Predator 500. With a 0.5 in. bore, they would provide more than enough pressure given an average 100 lbf driver input force and a 5.3:1 pedal ratio (lever arm ratio for master cylinder input force). The average 100 lbf driver input force was determined experimentally by the team and the 5.3:1 pedal ratio was decided based off calculations and pedal configuration and verified by the average driver foot size.

Figure 1: Master Cylinder

BRAKE ROTORS – The brake rotors are a part of the system that can be optimized to gain performance as well as limit weight. The front rotors chosen for our application are Galfer performance wave rotors designed for the Polaris Predator 500. Because we are using the spindles and calipers from the Polaris Predator 500 the Galfer performance rotors are more effective than the stock rotors while also limiting un-sprung weight. For the rear rotors we are using stock rotors from a Polaris Sportsman 300 to match the hubs and bearing carriers selected by the rear suspension design lead.
BRAKE PEDAL – The brake pedal was designed to accommodate the required pedal ratio calculated in Table 1. The minimum ratio required to generate required force to stop the vehicle is 3.4:1 with a driver input force of 100 lbf. In order to make the brake pedal the correct length for the average foot size of our drivers the ratio was increased to 5.3:1 (8 inch pedal). This will allow us to require even less driver input force than the minimum experimentally measured. With the pedal ratio of 5.3:1 the new minimum required driver input force to stop the vehicle becomes 55 lbf. Not only will this help us stop quicker but this will also help prevent driver fatigue during an endurance race. The brake pedal will be designed out of 6061-T6 aluminum to keep weight down while still having more than enough strength. Solidworks FEA shows that under a maximum load of 250 pounds on the pedal (250 pound max load was determined experimentally by pressing against a scale with the left foot of the drivers) it will see 24.9 ksi of stress, where the yield strength of 6061-T6 aluminum is 40 ksi (See Figure 4). This gives us a factor of safety for the pedal of 1.6 which is acceptable for a racing application.

CALCULATIONS – Since the calipers, rotors, and master cylinders were purchased parts with known dimensions, the calculations would be leading to the required pedal ratio to match the required force needed to be applied to the master cylinders. For these calculations the height of the center of gravity was assumed to be 24 inches based on a ride height of 12 inches and the relative heights of the driver and engine. This was verified using accurate weight values in Solidworks. The overall weight of 550 pounds was determined from a target vehicle weight of 380 lbs and an average driver weight of 170 lbs. The weight balance was assumed to be 40:60 (40% front, 60% rear). This was also based off of the SolidWorks model with accurate weight values for the components.

The first item to be determined would be the dynamic weights on axle. This is to determine how much weight will shift from the rear to the front under braking. Using a calculated maximum deceleration rate of 0.9 g from 35 mph to 0 mph it was calculated that 247 lbs would shift from the rear to the front. Using the individual weights on each axle (220 pounds on front axle, 330...
pounds on rear axle) and a coefficient of friction for rubber and asphalt of 0.7 [2], the braking force required to lock up the wheels was calculated. This force was then divided by 2 to give the force required at each wheel (assuming side to side braking is equal), and then multiplied by a factor of safety of 1.2 [1]. From there the torque required to lock up each individual wheel was calculated using the known radius of the rotors. Knowing the torque required to lock up each wheel the required clamping force of the calipers could be calculated by dividing the torque by the number of friction surfaces times the rotor radius and coefficient of friction between the brake pads and rotor material. The brake pad to rotor coefficient of friction was assumed to be 0.35 [3].

With the required clamping force calculated the required system pressure could now be calculated by dividing the clamping force by the known caliper piston area. From the required system pressure needed for each caliper, the minimum pedal ratio could be calculated. The minimum pedal ratio was calculated by taking the known area of the master cylinder piston times the required pressure, then dividing that by the minimum experimental driver input force of 100 lbs. This gave us a minimum pedal ratio required of 3.4:1 for the front system and 0.28:1 for the rear system. This can be evened out with the adjustment of the brake bias bar attached to the brake pedal and the master cylinders. Calculations are shown in Table 1 below.

Table 1: Calculations [4]

<table>
<thead>
<tr>
<th>TargetConfiguration</th>
<th>TargetWeight</th>
<th>40/35ActualWeightDistribution</th>
<th>Height of CG (h)</th>
<th>Where Used (t)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake Caliper (m)</td>
<td>0.300</td>
<td>335.000 g</td>
<td>0.006 m</td>
<td>0.042 m</td>
</tr>
<tr>
<td>Master Cylinder (m)</td>
<td>0.300</td>
<td>172.000 g</td>
<td>0.006 m</td>
<td>0.042 m</td>
</tr>
<tr>
<td>Master Cylinder (m)</td>
<td>0.300</td>
<td>172.000 g</td>
<td>0.006 m</td>
<td>0.042 m</td>
</tr>
</tbody>
</table>

PEDAL CONFIGURATION - The pedals will be mounted to a pivot welded to the baseplate. The base plate and pedal supports were originally going to be made from 7075-T6 plate aluminum but after running and comparing FEA numbers and fatigue studies of both materials, 6061-T6 proved to be more than sufficient (See Table 2). Using 6061-T6 would also reduce the overall weight and cost of the components. The mounting configuration for the brake pedal will allow for the 5.3:1 ratio. The throttle pedal was designed for a 1:1, 2:1, or 3:1 motion ratio allowing for a tunable throttle response using 3 different drilled holes at different distances from pivot point. The pedals will be spaced 4.5 inches apart to package well in the designed frame (See Figure 7). FEA done on the brake pedal mounting support using worst case force on the brake pedal of 250 pounds, which translates into 2000 lbf on the tabs creates a maximum stress of 16.2 ksi when the yield stress is 40 ksi (See Figure 8). This gives us a factor of safety of 2.47 for the mounts.
Table 2: 7075-T6 versus 6061-T6 Analysis

<table>
<thead>
<tr>
<th>Material</th>
<th>6061-T6</th>
<th>7075-T6 Plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength</td>
<td>45</td>
<td>83</td>
</tr>
<tr>
<td>Yield Strength</td>
<td>40</td>
<td>73</td>
</tr>
<tr>
<td>Max Torque Study</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stress</td>
<td>16</td>
<td>16</td>
</tr>
<tr>
<td>Displacement</td>
<td>0.000085</td>
<td>0.000081</td>
</tr>
<tr>
<td>Min. Factor of Safety</td>
<td>2.47</td>
<td>4.518518519</td>
</tr>
<tr>
<td>Fatigue Study</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Damage</td>
<td>1.807</td>
<td>1.279</td>
</tr>
<tr>
<td>Life</td>
<td>55330</td>
<td>78180</td>
</tr>
</tbody>
</table>

Figure 7: Pedal Configuration

Figure 8: Pedal Mount FEA

CUTTING BRAKE – A cutting brake was going to be utilized to improve maneuverability but during testing the cut brake malfunctioned which ultimately led to its deletion from the system. Also during maneuverability testing we found the vehicle to be sufficiently agile without it. With its removal we also saved around 4 pounds.

Figure 9: Cutting Brake

MANUFACTURING – For quick manufacturing of the pedals and base plate water jet cutting was utilized. This allowed us to manufacture very precise features and holes sizes as well. After the pedal bodies and faces were water jet cut the pedal faces were bent using a dead blow hammer to conform to the shape dictated by the pedal body. The pedal bodies and faces were joined together using TIG welding. This is shown in Figures 10 and 11.

Figure 10: Pedal Body and Face Welded Together

Figure 11: Mounting Tab Welded to Base Plate

BUDGET – For this project the brake system was given $2000 and 20 pounds for budgets. Throughout the project only $1484 was spent on the brake system leaving $516 unspent. As for the weight budget the braking system with the deletion of the cutting brake only came in at 15.7 pounds. This means the braking system was under its weight budget by 4.3 pounds. This was accomplished due to the light weight design of the entire system.
TESTING AND PROOF OF DESIGN – To validate the design and functionality of the braking system, a test was devised where we would accelerate and then hit the brakes firmly to lock them up. As you can see from Figure 12, the skid marks show that the brakes are able to lock up all four corners without hesitation. This would allow us to pass the brake test at the Baja SAE competitions. The braking system was also validated at the Baja SAE competition held at Tennessee Tech University where we earned the decal for completing the brakes test according to the SAE rules governing the event. This decal can be seen as the lower third section of the picture in Figure 13 and reads “Baja SAE Tennessee Tech”.

Figure 12: Brake Lock-up Test

Figure 13: Technical Inspection Passing Sticker

CONCLUSION

The brake system for the 2013 University of Cincinnati Baja SAE vehicle was designed with performance and weight in mind. To hit our overall weight goal of 380 pounds, the brake system was allotted 20 pounds and ended up weighing in at 15.7 pounds. Overall, the car performed as well if not better than we had expected. We ended up placing 12th place overall out of the one hundred teams that competed at the Baja SAE Tennessee Tech Competition. We completed the required brake test at the competition on the first attempt and did not break a single component throughout the entire competition. Not breaking any components is a great feat considering that the winningest team to compete (Cornell University) broke many parts in the four hour endurance race. In the different events of the competition we ended up placing 20th in design, 46th in cost, 31st in acceleration, 13th in maneuverability, 63rd in the sled pull, 17th in suspension and traction, and 13th in the four hour endurance race.

RECOMMENDATIONS

The recommendations I have for next year’s team mostly involve weight. Even though we were able to reduce weight by over 130 pounds this year, we were still in the range of 100 pounds heavier than most of the top 10 finishing teams. I believe that if next year’s team can reduce the overall weight of the vehicle by at least 50 pounds and keep the structural integrity of the vehicle, they have the potential, with better than average driving, to easily finish in the top 10 overall.

REFERENCES


CONTACT

Jason Taxis, Brake System Design Lead
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ACRONYMS AND ABBREVIATIONS

‘: foot
“: inch
FEA: finite element analysis
ft: feet
hp: horse power
in: inches
kgs: kilograms
ksi: kilopound per square inch
lbf: pound force
MPH: miles per hour
N: newton
psi: Pounds per square inch
RPM: revolutions per minute

APPENDIX A

RESEARCH

Disc Brakes vs. Drum Brakes

"Disc brakes are generally considered superior to drum brakes for several reasons. First, they dissipate heat better (brakes work by converting motion energy to heat energy). Under severe usage, such as repeated hard stops or riding the brakes down a long incline, disc brakes take longer to lose effectiveness (a condition known as brake fade). Disc brakes also perform better in wet weather, because centrifugal force tends to fling water off the brake disc and keep it dry, whereas drum brakes will collect some water on the inside surface where the brake shoes contact the drums."

(cars.about.com)

For our application I believe disc brakes are going to work much better than drum brakes ever could. We are going to endure a long, rough, and wet endurance course that will put our braking system through a lot of challenges. From the research it is easy to see that disc brakes will work much better for this application for many reasons. The first reason being that disc brakes offer superior braking force to drum brakes. For our competition the braking system must lock up all four wheels simultaneously on dry tarmac. The second reason disc brakes are going to work better for our application is that disc brakes can dissipate heat better than drum brakes can. This is going to be a key factor for the endurance race because we will have to brake hard and often going through, around and over obstacles on the course as well as all of the corners we will have to turn on. Another reason disc brakes will be better for our application is that they are proven to work better in wet situations. If the course is anything like past years there will be a lot of mud and a lot of water. Disc brakes will fling the water and mud (if wet enough) off of itself and continue to give us good braking force. The final reason is the weight and complexity factor. Disc brake systems are much simpler and weigh less overall which is a big factor when the power of the motor is set and cannot be modified.

EXISTING PRODUCTS

Wilwood Master Cylinder for Racing Applications

8/24/12

Suzuki Master Cylinder (Rear from Dirt Bike)

http://www.powersportsplus.com/parts/detail/suz/100-69600-37/000.html
8/24/12

Polaris 2005 Predator 500 Rotor (Stock)

http://www.rockymountainatv.com/omap- schematic/11
8/20/12

Galfer Wave Rotors for Polaris Predator 500 (Aftermarket)

http://www.rockymountainatv.com/omap-schematic/14/832/193132/1860750/-/-/193132/805062/
8/20/12

Polaris 2005 Predator 500 Caliper (Stock)

http://www.rockymountainatv.com/omap-schematic/18
8/20/12

Compact Design
Robust
Lightweight
Inexpensive

Master Cylinder used on the 2012 car. Although it is a great master cylinder we want to lighten the vehicle as much as possible.

Very Lightweight
Robust
Relocatable Reservoir

This design eliminates aluminum reservoir and replaces it with a smaller plastic reservoir

Inexpensive
Designed to work with hub and calipers being used

Proven Design

Very Lightweight
Better heat dissipation

Designed to work with spindle and rotors being used

Proven Design

Designed to work with spindle and rotors being used

Proven Design
APPENDIX B

CUSTOMER REQUIREMENTS

The system will be:
- Safe
- Affordable
- Easy to manufacture
- Easy to service
- Easy to use
- Easy to access components
- Reliable
- Robust
- Durable

CUSTOMER SURVEY

Baja SAE Brake System Customer Survey

The purpose of this survey is to understand the features that are most important to the customers who would be purchasing our Baja SAE vehicle.

How important is each individual feature for the design of a Baja SAE brake system?

Please circle your answer. 1 = low importance 5 = high importance
Safety 1 2 3 4 5 N/A
Durability 1 2 3 4 5 N/A
Stopping 1 2 3 4 5 N/A
Ease of Braking 1 2 3 4 5 N/A
Noise 1 2 3 4 5 N/A
Ease of operation 1 2 3 4 5 N/A
Cost 1 2 3 4 5 N/A
Appearance 1 2 3 4 5 N/A
Longevity 1 2 3 4 5 N/A

APPENDIX C

SCHEDULE

APPENDIX D

BUDGET

MONEY

BRAKE SYSTEM: $2000
2013 Car - Budget Distribution

WEIGHT

BRAKE SYSTEM: 20 LBS
APPENDIX E

SAE RULES AND REGULATIONS [5]

ARTICLE 11: BRAKING SYSTEM

B11.1 FOOT BRAKE

The vehicle must have hydraulic braking system that acts on all wheels and is operated by a single foot pedal. The pedal must directly actuate the master cylinder through a rigid link (i.e., cables are not allowed). The brake system must be capable of locking ALL FOUR wheels, both in a static condition as well as from speed on pavement AND on unpaved surfaces.

B11.2 INDEPENDENT BRAKE CIRCUITS

The braking system must be segregated into at least two (2) independent hydraulic circuits such that in case of a leak or failure at any point in the system, effective braking power shall be maintained on at least two wheels. Each hydraulic circuit must have its own fluid reserve either through separate reservoirs or by the use of a dammed, OEM-style reservoir.

Note: Plastic brake lines are not allowed

B11.3 BRAKE(S) LOCATION

The brake(s) on the driven axle must operate through the final drive. Inboard braking through universal joints is permitted. Braking on a jackshaft through an intermediate reduction stage is prohibited

B11.4 CUTTING BRAKES

Hand or feet operated “cutting brakes” are permitted provided the section (B11.1) on “foot brakes” is also satisfied. A primary brake must be able to lock all four wheels with a single foot. If using two separate pedals to lock 2 wheels apiece; the pedals must be close enough to use one foot to lock all four wheels. No brake, including cutting brakes, may operate without lighting the brake light.
Throttle Pedal Body

Gas Pedal Body

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES. EXCEPT AS ANNOTATED.

ANGLES ARE 90 DEGREES EXCEPT AS ANNOTATED.

TOLERANCES: THREE PLACE DECIMAL.

MATERIAL:

FINISH:

APPLICATION:

DO NOT SCALE DRAWING

PROPRIETARY AND CONFIDENTIAL

THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF [COMPANY NAME]. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF [COMPANY NAME] IS PROHIBITED.
Pedal Face