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Driveline for the USU 2006 Mini Baja

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DRIVELINE FOR THE USU 2006 MINI BAJA

by

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DRIVELINE FOR THE USU 2006 MINI BAJA

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Abstract

The Mini Baja project is part of an intercollegiate engineering design competition sponsored by SAE. For this competition senior engineering students at USU design and build a small off-road racecar. The design of the car is broken up into several different subassemblies. The driveline subassembly is responsible for transmitting the power from the transmission to the rear wheels. This is accomplished using several different components that have all been analyzed and designed. The completed driveline is able to accommodate sixteen inches of suspension travel, is lightweight, and was inexpensive to build. The driveline along with the rest of the car will be thoroughly tested at the final race in Portland, Oregon, during the second week of May.

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1 Introduction

The Mini Baja project at Utah State University is a part of an intercollegiate engineering design competition for undergraduate mechanical engineering students sponsored by SAE, the Society of Automotive Engineers. The competition is intended to simulate a real-world engineering design project and its related challenges. Teams from around the country compete to have their design accepted for manufacture by a fictitious firm. Each team must work together to design, build, and test a vehicle capable of surviving the extreme conditions of racing over rough terrain in all types of weather conditions.

Each year all of the participating schools meet to compete against each another and to present their designs to the panel of judges. This year the competition will be held at Portland, Oregon, during the second week of May. All of the competing teams will be scored in several categories. These categories can be broken up into two sections: static and dynamic. The static section includes all of the paperwork generated during the design process. As part of the competition, the students must keep detailed records of the design process. Over the course of the design several reports must be submitted to the judges. These reports include, cost, preliminary design, and final design. In addition at the final competition the teams must present their design to a panel of judges and defend the design decisions made. The dynamic section consists of five different racing events: acceleration, hill climb, maneuverability, rock crawl, and endurance. These events are intended to test all aspects of the design.

In order to increase the fairness and safety of the competition each team must adhere to a detailed set of rules. These rules give the size and weight constraints of the vehicle as well as the acceptable materials. They also state that all vehicles must be powered by a ten-horsepower Intek Model 20 engine supplied by the Briggs and Stratton Corporation. The same engine is used in all of the competing vehicles in hopes of making the competition a more challenging and even engineering design test.

Over the years, the teams at Utah State University have consisted of several engineering students working to fulfil their senior design requirement. Since most of the students who work on the project are seniors, a completely new team is created each year. This year the team consists of nine seniors majoring in mechanical engineering. Each student has been given the responsibility for a different subsystem of the vehicle. In order to have a working vehicle, each subsystem is equally important. These subsystems include frame, transmission, brakes, front suspension, rear suspension, and driveline. This report discusses in detail the designing and building of the driveline subsystem.

This subsystem is responsible for transmitting the power from the transmission to the rear wheels.

2 Driveline Requirements

The first task in designing each subsystem was to define a set of requirements that the designs had to meet. The requirements were intended to ensure that each component of the vehicle would be able to survive the abuse it would receive during the race. In addition the requirements ensured that the different parts would fit together correctly. In order to increase the creativity of the designers, the requirements were kept to a minimum. The driveline subsystem was given five main design requirements.

The first requirement was that the driveline must be strong enough to withstand the extreme forces experienced during the testing and racing of the vehicle. During an off-road race, the vehicle will be subjected to a constant beating. If part of the driveline is broken during the race, the vehicle will not be able to finish the race. It is impossible to design a vehicle that is completely indestructible, but with careful thought and planning a reliable vehicle can be designed. Drawing from lessons learned from the students who have previously participated in the mini baja competition and by tests performed on the car from last year, all of the expected loads and forces were determined. Using these loads and forces, all of the parts for the driveline were designed with a factor of safety of at least two. This means that it would require a force twice as large as the largest expected force to cause a part to fail.

The second requirement was that the driveline must be extremely light. Whenever a car is designed for racing, one of the main factors in determining how fast the car can go is its weight. The lighter the vehicle, the faster it will accelerate and the higher the top speed it will achieve. The mini baja built by USU in 2005 had a driveline subsystem that was bigger and bulkier than needed. The goal for 2006 was to reduce the weight of the subsystem by 30%. Great care had to be taken when deciding how to reduce the weight of the subsystem, because as the weight goes down so does the strength. The numbers of two for the factor of safety and 30% for the weight reduction are the optimized result of the two requirements.

The third requirement was that the driveline must be able to accommodate sixteen inches of travel by the rear suspension. This was important because the suspension would only be able to have as much travel as the driveline would permit. From past experience it was decided that if the baja could achieve a suspension travel of sixteen inches it would be able to clear all obstacles on the course without much trouble. This means that speed could be maintained as these obstacles were past.

The fourth requirement defined the size of the driveline. This requirment stated that the total width of the driveline including the wheels could be no more than 60 inches. An eight inch diameter sprocket must also be used to transfer the power from the transmission. The center of this sprocket had to be located at the center of the vehicle, five inches in front of the rear of the frame, and three inches above the bottom of the frame. These restraints ensured that the driveline would fit properly with the other subsystems.

The final requirement was that the driveline be very inexpensive. As with most student projects the mini baja team had a small budget to work with. The budget to build the entire vehicle and register the team for the competition was only $5000. The goal of building the driveline subsystem for under $100 dollars was set. In order to reach this goal, the parts used had to be donated, taken from a junk yard, or machined by the team.

After considering the requirements, it was decided to design the driveline subsystem similar to the driveline used on many of the newer ATVs. Figure 1 shows a cad model of the design. In the middle of the driveline is a sprocket which transmits the power from the transmission to the driveline using a chain. This sprocket is mounted to the center shaft of the driveline. The shaft is supported on each side of the sprocket by deep groove ball bearings that are held by adjustable mounts. These mounts are adjustable in order to place the proper tension on the chain. On the outside of the mounts are the upper CV joints which are connected to the lower CV joints by the CV joint shafts. The lower CV joints have external splines that mate with internal splines on the wheel hubs. These splines are necessary in order to transmit the rotation of the driveline to the wheel hubs. On the outside of each wheel hub are two Timkin cup and cone bearings. The outer races of these bearings will be supported by the knuckle of the rear suspension.

3 Testing

During normal driving conditions, a car does not experience extreme forces. Extreme forces only occur during an impact with another object. Since the mini baja must be designed to survive an off road race it was important to know the magnitude of the extreme forces experience by the car during an impact. Two different tests were performed to measure the force experienced by the car during an impact.

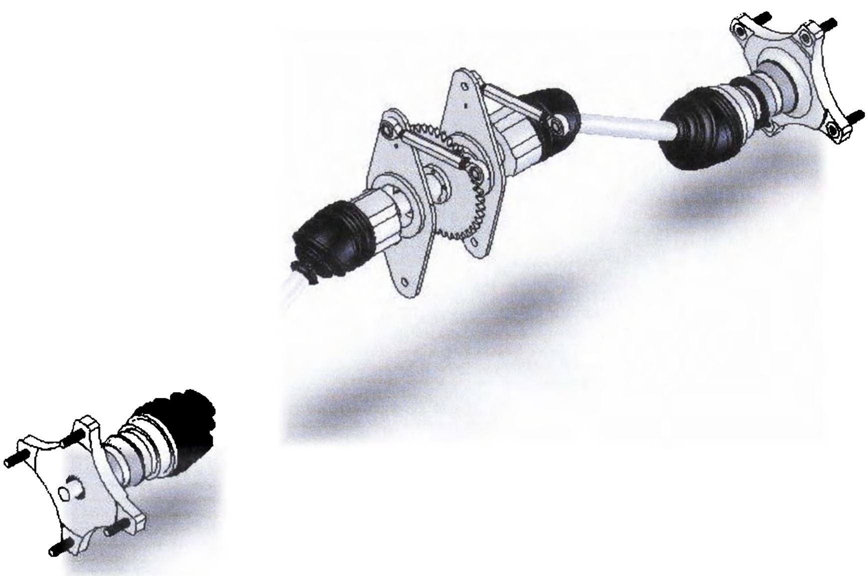

Figure 1: Cad model of the driveline subsystem.

3.1 Strain Gages

The first test attempted to used strain gages to measure the strain experienced in the A-arms of the suspension when the car was driven into a wall, while traveling at a speed of five miles per hour. Strain is defined as the amount of deformation per unit length of an object when a load is applied. With knowledge of the strain experienced by the A-arms, the force applied to the A-arms could then be calculated. Strain gages are designed to convert mechanical motion into an electronic signal. A strain gage is shown in Figure 2. A computer is used to record the electronic signals produced by a strain gages. After attempting to measure the strain on the A-arms during an impact, it was found that the computer used to record the signal from the strain gages was not fast enough to record the sudden spike in strain experienced at the instant of impact.

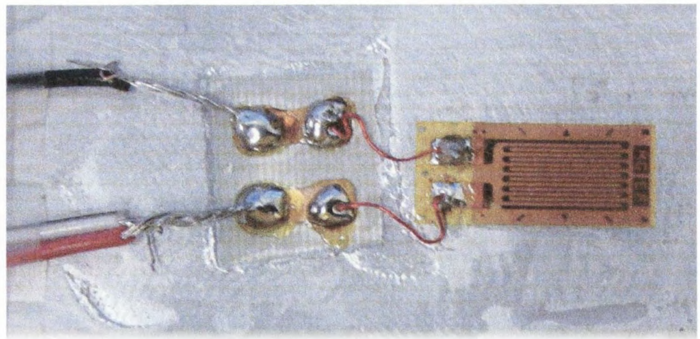

Figure 2: Strain gage used during the impact test of the 2005 mini baja.

3.2 Accelerometer

Since the team did not have access to a faster computer, a different test had to be used to calculate the force on the car during an impact. Because attempting to measure the force when the car impacted an object failed, it was decided to measure the force the car experienced when an object impacted it. A heavy object had to be used to be able to simulate a meaningful impact. A large steel billet was the object chosen to impact the car. The billet was hung form a hoist using a chain so that it could be swung like a pendulum into the car. A picture of the impact test setup is shown in Figure 3.

The force of the impact was measured using an accelerometer and an oscilloscope. An accelerometer is a device used to measure the acceleration of an object. The accelerometer sends a electronic signal to an oscilloscope which is capable of measuring the sharp spike in acceleration experienced during an impact. An accelerometer and the reading of the oscilloscope after an impact are shown in Figure 4.

The heights the billet needed to be raised to simulate a two mph and a five mph impact and were calculated by relating the potential energy of the billet to the kinetic energy of the car using equation 1.

$$\frac{1}{2} m_{car} v_{car}^2 = m_b g h_b \tag{1}$$

where m_{car} is the mass of the car, v_{car} is the velocity of the car, m_b is the mass of the billet, g is the acceleration of the billet due to gravity, and h_b is the height the billet is raised. The complete analysis used to calculate the heights the billet needed to be raised is included in appendix A. From the recorded acceleration, the forces experienced by the car during two mph and five mph impacts were calculated. These forces were used as the maximum expected forces.

Figure 3: Large steel billet used for the impact test.

4 CV Joints

The driveline needed some type of flexible coupling to accommodate the up and down motions of the suspension. The drivelien also needs to be able to change length because the distance between the wheels changes as the operating angle of the driveline increases. In the automotive industry two common couplings are used on drivelines, universal joints and constant velocity or CV joints. CV joints are better for situations that require a large working angle. Most universal joints can only handle a working angle of 15 degrees before fluctuations in drive shaft speed or vibrations occur. Since one of the design requirements was to accommodate 16 inches of suspension travel, CV joints

Figure 4: On the left is the accelerometer used to measure the impact on the suspension of the baja, the signal produced by the impact is shown on the oscilloscope on the right.

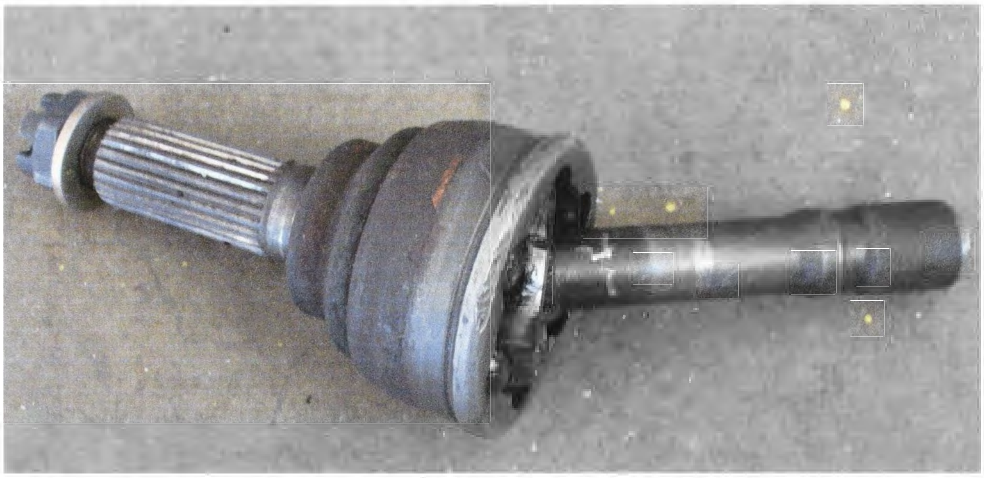

Figure 5: CV joint taken from a Geo Metro.

were the obvious choice for this design.

The CV joints used on the baja were taken from a 1990 Geo Metro. One of the CV joints used is shown in Figure 5 . These particular joints were chosen for several reasons. First, since the Geo is bigger and heavier than the mini baja, the CV joints have already proven that they will easily be able to withstand the expected forces. Next, these CV joints are smaller and lighter than other CV joints used in the past. The reduction in size also allowed other parts of the vehicle to be smaller too, namely the suspension knuckles and the wheel bearings. Another great benefit of these CV joints is that they were obtained free of cost from a local salvage yard.

Each CV joint assembly consists of two CV joints joined by a shaft. Two joints are required to transmit the rotation of the driveline in a consistent manner to the rear wheels. Each joint is protected by a flexible rubber boot. These boots ensure that the joints remain clean and that the proper lubrication is maintained. On the Geo a solid steel shaft was used to connect the two joints. To incorparate the CV joints to the baja this shaft was replaced with a hollow Chrome Moly tube. This modification reduced the weight of the shaft by more than 50% while still keeping a factor of safety of 2.9.

4.1 Torsional Analysis

To ensure that the new connecting shaft would be strong enough, a torsional analysis was performed. The first step in this analysis was to determine the maximum torque that the driveline would experience. There are two sources of torque for the driveline, the brakes and the transmission. By calculation it was determined that the transmission could deliver as much as 600 ft-lbs of torque whereas the brakes could only deliver 460 ft-lbs of torque. Since the amount of torque from the transmission was higher, it was used for the torsional analysis.

After the torque was determined the polar moment of inertia for the solid shaft and the hollow shaft were calculated using the the equations 2 and 3.

$$J_{solid} = \frac{\pi d^4}{32} \tag{2}$$

$$J_{tube} = \frac{\pi \left(d_o^4 - d_i^4\right)}{32} \tag{3}$$

where d is the diameter of the solid shaft, d_o is the outer diameter of the tube, and d_i is the inner diameter of the tube. Next, the maximum shear stress in the shafts due to the applied torque were calculated using equation

$$\tau_{max} = \frac{Td}{2J} \tag{4}$$

where T is the applied torque, d is the outer diameter of the shaft or tube, and J is the polar moment of inertia of the shaft or tube. These shear stresses were then compared to the yield strengths of the tube and the shaft to determine the factor of safety. Based on this analysis, a hollow Chrome Moly tube with a 7/8" outer diameter and a 1/8" wall thickness was chosen. Using this tube would reduce the weight of the shaft by about 50%, while maintaining a factor of safety of 2.9. The complete analysis is included in appendix B of this document.

4.2 Assembly

To incorporate the new hollow shafts to the CV joints, the old shafts had to be cut off. This was done using a chop saw. After removing the old shafts, new shafts were welded to the CV joints. A welding analysis was done to ensure that the welded joints would be strong enough. This analysis was similar to the torsional analysis done to determine the tube size. For this analysis the unit polar moment of inertia was found using equation 5.

$$J_u = 2\pi r^3 \tag{5}$$

where r is the radius of the tube. The polar moment of inertia was then multiply by a weld size factor, h, to obtain the polar moment of inertia. Using equation 4 the maximum shear stress experienced

Figure 6: Custom made rear wheel hub.

by the weld was calculated. This value was compared to a weld strength value taken from a table and a factor of safety was computed. The factor of safety was only 1.3, but if a larger weld is applied the factor of safety can easily be increased to the desired value of 2. The complete weld analysis is included in Appendix C. During the welding process, the lubrication in the joints was heated to very high temperatures. To ensure that the lubrication would not fail, all of the old lubrication was removed from the joints and new lubrication was applied.

5 Wheel Hubs

The most challenging parts of the driveline design were the wheel hubs. Since the CV joints used on the baja were smaller in size, custom wheel hubs with smaller splines had to be manufactured. One of these hubs is shown in Figure 6. The wheel hubs were designed to be as light as possible, using aluminum instead of steel to reduce the weight. Unlike other hubs, in addition to providing a way to mount the wheels, the hubs also provided a bearing surface. This was needed because the smaller CV joints do not have enough surface area for the bearings. An aluminum block large enough to machine the hubs would be very expensive, so the hubs were cast instead. The casting process produced the rough shape of the hubs. The hubs then had to be machined to size. The

Figure 7: On the left is the bottom view of the pattern used to make the hubs, the top view is on the right.

entire process is discussed in detail below.

5.1 Casting

Wikipedia defines casting as "A process by which a material is introduced into a mold while it is liquid, allowed to solidify in the shape inside the mold, and then removed producing a fabricated object, part or casing"[1]. Casting is used extensively in the automotive industry for making parts such as engine blocks and cylinder heads. The shape of the object is determined by the shape of the mold. There are many different methods of casting, these include die, sand, and investment. The method of sand casting was used by the mini baja team. Sand casting is the oldest and best understood casting method. In fact, it is one of the oldest and simplest metal working processes known to man. Biblical records document the use of sand casting as early as 5500 B.C. and Egyptian tomb paintings from about 1500 B.C. show that the Egyptians also utilized this technology. Due to the simplicity and the inexpensiveness of sand casting, many components of the car were made by sand casting. These components include the suspension knuckles, transmission case, and rear wheel hubs.

The first step of sand casting is to make a pattern. The pattern is the physical model of the part, which is used to make the cavity in the sand mold. The pattern used to make the wheel hubs was made out of wood. A picture of this pattern is shown in Figure 7. Since the hubs required an internal spline, they had to be able to withstand large amounts of stress during the machining process. For this reason the patterns for the hubs were much larger than the final hubs needed to be. To facilitate the removal of the pattern from the weak and brittle molding sand, draft was applied to the edges of the pattern. The long column coming off the top of the pattern was used to allow for

Figure 8: Sand cavity made during the casting of the transmission case.

a little bit of overflow when the metal was poured. This is necessary because as the metal starts to solidify, shrinkage will occur. The round part on the right of the patten is a filling reservoir, where the metal is poured into the mold.

The pattern is placed in a multi-part molding box referred to as a flask. Sand is packed in the flask on both sides of the pattern to create the mold. Sand casting molds use fine silica based sands that have round grains that can be closely packed to form a smooth mold surface. Before sand can be packed around the pattern, diamond dust is applied to the pattern. This diamond dust helps to keep the sand from sticking to the pattern. The sand is packed closely and tightly around the pattern through a vibratory process called ramming. Once the sand is completely packed around the pattern the flask is opened and the pattern is removed leaving a hollow cavity or mold in the sand. Figure 8 shows the mold used to cast one of the transmission cases.

Next, the mold is inspected and touched up if needed. Then, a channel running from the filling reservoir to the part cavity is cut in the sand. This provides a way for the liquid metal to reach the part cavity. In addition, a hole is cut from the top of the flask to the filling reservoir. This is to allow a place for the liquid metal to enter the mold. Better results are obtained if a filling reservoir is used. If the liquid metal is poured directly into the part mold, large amounts of shrinkage will occur during the cooling process. When a filling reservoir is used most of the shrinkage will occur in the reservoir and not in the part.

With the mold prepared, the aluminum is heated in the foundry. The aluminum needs to reach a temperature between 1200 and 1500 degrees F. Before the metal can be poured into the mold, a degassing agent must be added. This allows the gases present in the metal to escape before it hardens. If the metal is not degassed, the cast part will have air pockets that will significantly weaken the part. After the metal has been degassed it is poured into the mold and allowed to cool. Once cooled the sand is removed from around the part and the finish machining can be done.

5.2 Splines

Splines are used to transmit large amounts of torque. They are able to transmit much more torque than press fits or keys. Splines are formed by contouring the outside of the shaft and the inside of the hub with tooth-like forms. Since the CV joints came with a spline already cut on the outside of its shaft, matching splines had to be machined into the wheel hubs. The first step of machining the splines into the hub was to make a special cutting tool called a broach. Wikipedia defines a broach "As a series of progressively taller chisel points mounted on a sing piece of steel, typically used to enlarge a circular hole into a larger non-circular shape"[2]. The broach matched the shape of the desired splines. This particular spline has 23 teeth.

Once the broach was prepared, a hole the diameter of the base of the teeth was bored into the wheel hubs. This was done on a standard metal lathe. Next, the broach was pressed through the bored hole several times using a hydraulic press. The first pass required about 20,000 pounds of force. As the broach was passed through, material was removed forming the desired shape. With each pass a little more material was removed and a little less force was required. Several passes were made in order to achieve the desired shape.

5.3 Machining

To be able to hold the hub during the machining process a mandrel had to be made. The mandrel fits the splines on the inside of the hub and provide a surface for the chuck of the lathe to grip. The splines on the mandrel were made in the same way as the broach. Using an indexing jig, the splines were cut on an end mill using a carbide cutting tool. Cutting the splines is a slow process, since 23 had splines had to be cut, and each spline took several passes to cut. The mandrel used is shown in Figure 9. Using the mandrel the hub was held in the lathe and material was removed until the necessary bearing surface was achieved.

Bearing surfaces have to be extremely smooth and accurate. To get the proper fit, the shaft that

Figure 9: Mandrel made for the machining of the wheel hubs.

the bearing fits on must be 1/2 a thousand of an inch bigger than the inner diameter of the bearing. After the bearing surface was made, holes for the wheel studs were tapped and drilled using the end mill. These holes were tapped to a 3/8" - 16 UNF thread and were counter sunk on the top. The counter sink makes it so that the top surface of the wheel studs is flush with the face of the wheel hub.

6 Bearings

Bearings are used to reduce friction between moving parts. The driveline uses six bearings, two on each hub and two on the center shaft. They are needed everywhere a moving part is attached to a stationary part. The bearings on the hubs are Timkin cup and cone bearings and the bearings on the center shaft are standard deep groove ball bearings.

6.1 Hub Bearings

The bearings used on the wheel hubs have to be able to withstand both axial and radial forces. A standard deep groove ball bearing is designed only to withstand radial forces. Because of this, a different type of bearings had to be selected. The Timkin cup and cone bearings, also referred to as tapered roller bearings, are designed to withstand both axial and radial loads. They are used in pairs opposing each other so that axial loads in either direction can be accommodated. Since these bearings are to be used on an off-road vehicle, they must be sealed. If the bearings were not sealed they would quickly become dirty and would not be able to function properly. A picture of one of the Timkin cup and cone bearings used is shown in Figure 10.

Figure 10: Timkin cup and cone bearings used on the wheel hubs.

6.2 Center Shaft Bearings

The center shaft is designed so that it will only provide a radial force on the bearings which support it. For this application, the best bearings are sealed deep groove ball bearings. Some sealed deep groove ball bearings are shown in Figure 11. To ensure that the bearings would survive the race, a bearing analysis was performed. The life of a bearing can be calculated using equation

$$L = \left(\frac{2C}{P}\right)^3 \tag{6}$$

Where P is the applied load and C is the basic dynamic load rating of the bearing given by the manufacturer. The analysis done showed that the bearings would last for over 135 hours of driving at the car's maximum velocity. The complete bearing analysis is in appendix D.

7 Center Shaft and Sprocket

The center shaft was machined of 4140 steel. The shaft is 6 inches long and has a 1.375" outer diameter and a 1" inner diameter. The reason the shaft is hollow is to allow for the ends of the upper CV joints to fit inside. The upper CV joints are bolted to the center shaft using 1/4" bolts. A mounting flange for the sprocket is welded to the center of the shaft. The sprocket is attached to this flange using three indexing pins and three bolts. A shear analysis was performed to ensure that the bolts and pins would be sufficient to hold the sprocket in place. This analysis assumed that six 1/4" diameter bolts or pins would be used to attach the sprocket to the mounting flange. The first step of the analysis was to determine the surface area of the bolts that would be supporting the sprocket, this area is referred to as the bearing area. This was done using equation 7.

Figure 11: Sealed deep groove ball bearing used to support the center shaft.

$$A_b = d * t \tag{7}$$

Where d is the diameter of the bolt, and t is the thickness of the sprocket. Next the force experienced by each bolt was calculated using equation 8.

$$F = \frac{T}{r * n} \tag{8}$$

Where T is the maximum torque applied to the sprocket, r is the distance from the center of the bolt to the center of the shaft, and n is the number of bolts or pins used to attach the sprocket. With the bearing area and the applied force, the shear stress on each bolt was calculated using equation 9.

$$\sigma = \frac{F}{A_b} \tag{9}$$

By comparing this stress value to the yield strength of the bolt, a factor of safety was calculated. In this case a factor of safety of over 7 was calculated. The complete shear stress analysis is included

Figure 12: Adjustable hexagonal rod with rod ends used to tension the chain.

in appendix E.

8 Tensioning System

The tensioning system is critical to delivering the maximum amount of power to the rear wheels. If too little tension is applied to the chain, the chain could fall off of the sprocket during the race. If too much tension is applied to the chain, additional friction and forces will be applied to the driveline and transmission which could lead to part failures. The tensioning system consists of two brackets that are attached to the frame on one end and to adjustable hexagonal rods at the other. The outer races of the center shaft bearings are pressed into these two mounts.

The tensioning of the chain is achieved through the adjustable hexagonal rods. These rods have a rod end with right-handed threads at one end and a rod end with left-handed threads at the other end. The different direction threads allows the rod to change length. If the rod is turned clockwise the length of the rod is increased and the tension on the chain is reduced. If the rod is turned counter-clockwise the length of the rod is decreased and the tension on the chain is increased. The reason for the hexagonal shape of the rod is to allow for a wrench to be used to turn it. One of the hexagonal rods used in the tensioning system is shown in Figure 12.

9 Conclusion

The mini baja project has allowed students to get some real world engineering experience. Being able to take a project from the design stages all the way through the building and testing stages taught the students how to overcome many of the challenges that arise during the process. By using CV joints from a Geo Metro with custom made wheel hubs a driveline was built that would overcome the challenge of allowing for sixteen inches of travel while being lightweight and inexpensive. The completed driveline is shown in Figure 13. At the final race in Portland, it will be seen if the lessons

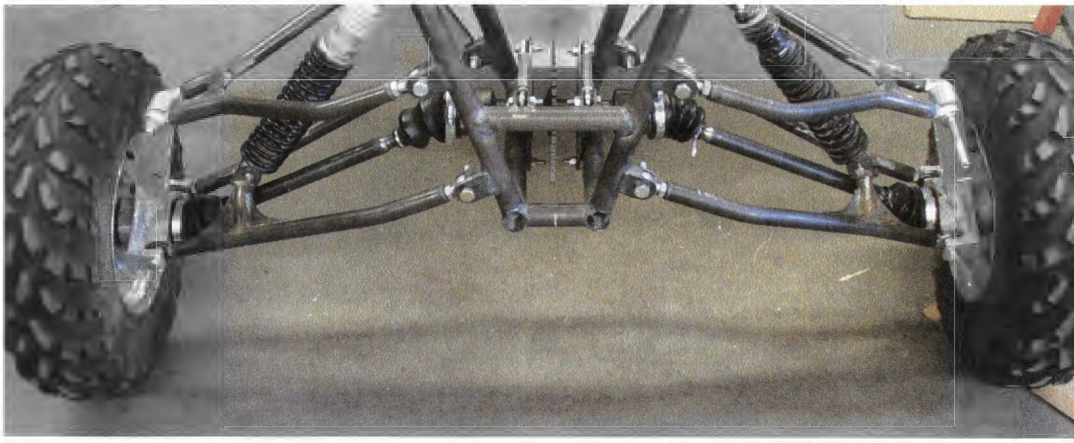

Figure 13: Completed driveline assembled on the mini baja.

learned during this project were enough to produce a winning design.

A Impact Test Calculations

IMPACT TESTING CALCULATIONS

BILET PROPERTIES

$r := 4in$		radius of the steel billet
$L := 23.75in$		length of the steel billet
$\rho_{steel} := 0.3\frac{lb}{in^3}$		density of steel
$V := \pi \cdot r^2 \cdot L$	$V = 1.194 \times 10^3 in^3$	volume of the billet
$W := \rho_{steel} \cdot V$	$W = 358.142 lb$	weight of the billet
$m := \frac{W}{32.2}$	$m = 11.122 lb$	mass of the billet

CAR PROPERTIES

$W_{car} := 500 lb$		weight of the car
$m_{car} := \frac{W_{car}}{32.2}$	$m_{car} = 15.528 lb$	mass of the car

CALCULATIONS

$\frac{1}{2} m \cdot v^2 = m \cdot g \cdot h$		Governing equation equation the speed of the vehicle to the height of a 358 lb steel billet
$h_{2.5mph} = \frac{\frac{1}{2} \cdot m_{car} \cdot (2.5 \cdot mph)^2}{W}$	$h_{2.5mph} := 3.5in$	Height the steel billet has to be raised to simulate a 2.5 mph impact with the baja
$h_{5mph} = \frac{\frac{1}{2} \cdot m_{car} \cdot (5 \cdot mph)^2}{W}$	$h_{5mph} := 13.9in$	Height the steel billet has to be raised to simulate a 5 mph impact with the baja

B CV Shaft Analysis

Torsional Analysis of the Baja CV Shaft

MATERIAL PROPETIES OF 4130 STEEL TUBING

$G := 11600 \cdot 10^3 psi$	Shear modulus	$RPM := \frac{1}{min}$	
$S_{yt} := 138000psi$	Yield strength	$S_{ut} := 161000psi$	Ultimate stregth
$\rho := .284 \frac{lb}{in^3}$	Density		
$l := 18in$	length of the shaft	$d := .875in$	shaft diameter
$d_o := 1in$	outer diameter of tube	$d_i := .75in$	inner diameter of tube

CALCULATIONS

$D_{tire} := 20.5in$		Diameter of the tire
$dist := \pi \cdot D_{tire}$	$dist = 5.367\,ft$	Distance traveled in 1 revolution
$V_{max} := 35 \frac{mi}{hr}$	$V_{max} = 51.333 \frac{ft}{s}$	Maximum velocity of the Baja
$V_{dl} := \frac{V_{max}}{dist}$	$V_{dl} = 573.9\,RPM$	Velocity of the driveline
$F_b := 2000lbf$		Brake force applied
$\mu_b := .27$		Brake coefficient of friction
$T_b := F_b \cdot \mu_b \cdot \frac{D_{tire}}{2}$	$T_b = ■\, ft \cdot lbf$	Maximum torque from the brakes
$T := 600ft \cdot lbf$		Maximum torque from transmission

Maximum torque from the transmission is higher the the maximum braking torque, so it will be used in the shear calculations

$J := \frac{\pi \cdot d^4}{32}$	$J = ■\, in^4$	Polar area moment of inertia
$J_{tube} := \frac{\pi \cdot \left(d_o{}^4 - d_i{}^4\right)}{32}$	$J_{tube} = ■\, in^4$	Polar area moment of inertia of hollow tube

$\tau_{max} := \frac{T \cdot d}{2 \cdot J}$	$\tau_{max} = 54737\,psi$	maxium shear stress
$\tau_{tmax} := \frac{T \cdot d}{2 \cdot J_{tube}}$	$\tau_{tmax} = 46937\,psi$	maxium shear stress in hollow tube
$\theta := \frac{T \cdot l}{J \cdot G}$	$\theta = 11.123\,deg$	maximum angular deflection
$\theta_t := \frac{T \cdot l}{J_{tube} \cdot G}$	$\theta_t = 9.538\,deg$	maximum angular deflection in hollow tube
$SF_{solid} := \frac{S_{yt}}{\tau_{max}}$	$SF_{solid} = 2.521$	Factor of safety with the solid shaft
$SF_{tube} := \frac{S_{yt}}{\tau_{tmax}}$	$SF_{tube} = 2.94$	Factor of safety with the hollow tube
$V := \frac{\pi \cdot d^2}{4} \cdot l$	$V = 10.824\,in^3$	Volume of the solid shaft
$V_{tube} := \frac{\pi \cdot \left(d_o^{\,2} - d_i^{\,2}\right)}{4} \cdot l$	$V_{tube} = 6.185\,in^3$	Volume of the hollow tube
$m := \rho \cdot V$	$m = 3.074\,lb$	Mass of the solid shaft
$m_{tube} := \rho \cdot V_{tube}$	$m_{tube} = 1.757\,lb$	Mass of the hollow tube

The .875 in diameter solid shaft should be replace by a tube of outer diameter 1 inch and inner diameter .75 inch in order to increase the strength and reduce the weight.

This analysis was done using the methods outline in Mechanics of Materials by Gere. The materials properties are of 4130 steel.

C Weld Strength Analysis

Weld strength analysis

Done by Dan Swenson
Jan. 10, 2006

Material properties

$D := 1\text{in}$	Diameter of the shaft	$r := \frac{D}{2}$	Radius of the shaft
$h := .25\text{in}$	size of weld	$S_y := 57000\text{psi}$	Yield strength

Calculations

$M := 600\text{ft}\cdot\text{lbf}$		Worst case moment
$J_u := 2\cdot\pi\cdot r^3$	$J_u = 0.785\,\text{in}^3$	Unit polar moment of inertia
$J := .707\cdot h\cdot J_u$	$J = 0.139\,\text{in}^4$	Polar moment of inertia
$\tau := \frac{M\cdot r}{J}$	$\tau = 25933\,\text{psi}$	Maximum shear stress in the weld
$S_p := .6\cdot S_y$	$S_p = 34200\,\text{psi}$	Permissible stress in the weld
$SF := \frac{S_p}{\tau}$	$SF = 1.319$	Factor of Safety of the weld

Note: All of the Equations and material properties were taken from Mechanical Engineering Design by Joseph Edward Shigley, 1977

D Bearing Analysis

Bearing Analysis

$d_i := 1.25in$ Shaft diameter

$d_o := 2in$ Outer diameter

$F_r := 125lbf$ Radial load

$F_a := 0lbf$ Axial load

$L_{10} := 5 \cdot 10^8$ Desired bearing life

$RPM_{max} := 600$

Calculations

SKF 071100S Deep Groove Ball Bearing

$C := 10600lbf$

$C_o := 12800lbf$

$P := F_r$

$L := \left(\frac{2C}{P}\right)^3$ $L = 4.878 \times 10^6$ C is multiplied by 2 since two bearings are used

$V_{max} := 600 \cdot \frac{1}{min}$ Driveline revolutions at 35 mph

$time := \frac{L}{V_{max}}$ $time = 135.511\,hr$

These bearings would survive more than five and a half days of continuous drive at the maximum velocity of the vehicle.

This analysis followed the methods outline in Machine Design by Norton. The C values used were from a SKF LM 48548 A/510/Q tapered roller bearing

E Shear Analysis of Sprocket Bolts

Shear Analysis of the Sprocket Bolts

$d_{bolt} := .25in$		bolt diameter
$t_{spr} := .25in$		Sprocket thickness
$A_b := d_{bolt} \cdot t_{spr}$	$A_b = 0.063\,in^2$	Bearing area
$T := 600ft \cdot lbf$		Worst case torque
$n := 6$		Number of bolts
$r := 1in$		Distance from the bolts to the shaft center
$F_{bolt} := \frac{T}{r \cdot n}$	$F_{bolt} = 5.338 \times 10^3\,N$	Force experienced in each bolt
$\sigma_b := \frac{F_{bolt}}{A_b}$	$\sigma_b = 19200\,psi$	Bearing stress in each bolt
$S_{yt} := 138000psi$		Yeild strength of the bolt
$SF := \frac{S_{yt}}{\sigma_b}$	$SF = 7.187$	Saftey factor of the bolted joint

This analysis was done by following the methods outlined in Mechanics of Materials by Gere. The material properties were taken from Matweb.

References

[1] *Wikipedia*. 2006. 8 April 2006 <http://en.wikipedia.org/wiki/Castinghttp://en.wikipedia.org/wiki/Casting>.

[2] *Wikipedia*. 2006. 10 April 2006 <http://en.wikipedia.org/wiki/Broach_%28metalwork%29>.