Design of a Chain Driven Limited Slip Differential and Rear Driveline Package for Formula SAE Applications

by

Keith Durand

Submitted to the Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of

Bachelor of Science

at the

Massachusetts Institute of Technology

June 2005

Signature of Author ....
Department of Mechanical Engineering
May 6, 2005

Certified by ....
Daniel Frey
Professor of Mechanical Engineering
Thesis Supervisor

Accepted by ....
Ernest Cravalho
Professor of Mechanical Engineering
Chairman, Undergraduate Thesis Committee

Table of Contents

List of Figures

1.0 Concept of the FSAE Competition

Formula SAE is the largest intercollegiate competition in the world. Students must design, fabricate, and race a small formula style car. The Society of Automotive Engineers publishes a rule book outlining what constitutes a formula car. Other than basic safety restrictions and engine restrictions, teams are free to innovate and develop what they feel is a winning package.

The cars are judged on cost, design, marketability, and of course, performance. Every year, up to 140 teams gather in Detroit to determine which university has the best car.

For the purposes of designing a differential, cost, design, and performance are of utmost importance. These events will be described in detail below.

The cost event is comprised of three sections—lowest cost, cost report presentation, and a manufacturing presentation. Well designed cars tend not to be the cheapest, but the other two events allow expensive cars to still be competitive. The cost of the car includes raw stock, machine time, labor, and the retail price of commercially built parts. Thus, the designer should be very concerned with limiting cost

In the design competition, judges from automotive companies and from the racing industry scrutinize the car. Teams also give a design presentation. Top teams are selected to compete in the design semifinals, with more rigorous judging and more time for presentations. Points are awarded based on how well though out the design is and how well the team can justify engineering decisions.

Additionally, points are awarded for performance in four dynamic events--skidpad, acceleration, autocross, and endurance.

The skidpad event focuses on a car's cornering ability. The elapsed time in this event corresponds to the maximum lateral acceleration of the car.

The acceleration event judges the engine package as well as the weight of the car. Light cars with well developed power plants tend to do well in this event. Typical 0-60 times are under 4 seconds.

Autocross focuses on a car's ability to negotiate a tight course at relatively low speed, as well as the ability of the drivers.

Finally, the endurance race is a measure of the reliability of the car, as well as high speed handling. The endurance course is approximately 22 km long, with many tight turns. Roughly 1/3 of the entrants finish the endurance race. Two drivers are needed for this event. During the driver change, any evidence of a fluid leak is grounds for removal from the race. Teams that do not finish are not awarded any points for the event, hurting their final ranking significantly.

Much more detailed information on FSAE events and how points are awarded are contained in the official rules [1].

2.0 Design Philosophy and Methodology

The concept of the competition leads to some obvious functional requirements. These are described in detail in the next section, as these bear directly on the mechanical design.

Additionally, there are some additional pressures placed on the designer. Formula SAE emphasizes quick development of complex designs. As soon as the competition in May is over, teams must reorganize, elect new management to replace senior members, recruit new members, start designing, and have a car ready to race in one more year.

In general, the fall term is used for design. In certain cases, construction may begin on small projects, or ongoing research projects, such as engine development. The spring terms is used largely for construction, car preparation, and driver training. The spring is very busy; thus it is in the best interest of the team to have all design work done and documented by the end of the fall term.

Due to the time constraints, one of the keys to a successful team is the transfer of knowledge from one year to the next. A team that has solid designs to build upon will do well compared to a team that has no solid basis for new designs.

The design presented here is intended to be very detailed, right down to the specifications for fasteners used in the assembly. While this is expected in industrial practice, the time pressures placed on FSAE designs has traditionally limited the amount of detail that goes into the final design. The details are important for several reasons. Time invested in the design pays off during construction. As time becomes more precious closer to the competition, working out the details in advance means a project gets finished sooner. Projects finished on time means more debugging time at the track.

Two items of great importance during the design phase is a complete parts list and a complete tool list. Lists should be very specific, for example, if the designer needs a piece of aluminum stock, the size, alloy, and heat treat should be specified. In the case of items such as bearings, seals, fasteners, or tooling, a part number and supplier is best. Detailed parts and tooling lists are included in the appendix; this is intended to serve as an example to future team members.

Working out these details during the design phase helps with procurement and with preparing the cost report. Team management, who is usually responsible for buying parts and tools, is not always familiar with the designer's specific needs. This causes confusion when the management receives a request for parts with only a vague description.

Placing orders through a few central people allows orders to be consolidated; many aspects of the car draw from the same few suppliers. Consolidating orders reduces shipping costs and promotes productivity by having all products delivered before construction begins. Consolidating orders also consolidates receipts, which is convenient when constructing the cost report. A detailed parts list also makes the task of creating a cost report much easier. As the cost report is a significant portion of a car's score in competition, quality work on the cost report pays off.

Finally, tooling lists help when designing for manufacture. While the team budget is large enough to afford proper cutters, large investments in expensive tooling can often be avoided by choosing materials and operations accordingly.

All parts were designed with SolidWorks for CAD. The MIT FSAE team currently uses this program exclusively, as does the undergraduate curriculum. CosmosWorks, a relatively easy to use finite element analysis package, is integrated into SolidWorks. CosmosWorks was used to verify strength of the components for the expected design loads.

Mechanical design is an iterative process. Concepts are realized in CAD, assemblies are generated from parts, interferences are resolved, parts are tested with FEA, and then strength issues are addressed in the CAD model. At any point, the original CAD model may have to be modified, and the process begins again. The way this document will present the design is that a projected 3D view of the part will be shown near the beginning of the section. This is to help the reader understand the written arguments. Detail views may also be included in the text for clarity of written arguments. FEA results will appear at the end of the section. Though this design used FEA for strength verification, FEA was not relied on heavily for reducing weight or optimizing material placement. Parts were intentionally designed with high safety factors and with little faith in the FEA results, as racing is very abusive. Shock loads and hard driving can easily stress components beyond their design limits. Though optimization with FEA may be a future goal of the team, a robust drivetrain was the goal for the 2005 MIT entry.

Fully dimensioned drawings will not appear in the text, as these are messy and contribute little to the written arguments.. Most of the parts in this assembly were designed with the intent of going directly from CAD model to CAM, eliminating the need for dimensioned drawings. Electronic copies of the CAD models will be maintained by MIT FSAE and will be available to people associated with the MIT team.

2.1 Functional Requirements:

The differential design started with a set of design criteria. First, two broad goals were set at the beginning of the design process.

1. The drive train should be durable. This means that it must survive a season of testing with as little maintenance as possible. Specifically, no expensive parts, especially the main housing, should have to be replaced during the testing season. Additionally, the drive train package must be able to survive an autocross, skidpad, acceleration, and endurance event with no maintenance.

2. The drive train package should be light in weight. A good metric for this would be at least 10% lighter than the drivetrain on MIT's 2004 car.

Next, a list of very specific functional requirements was generated. These were broken into three areas: functionality, manufacturability, and integration with the rest of the car.

Functionality:

1. A single rear brake disk should act on both wheels through the differential.
2. The differential must not show any signs of oil leakage during extended running, such as an endurance event.
3. The half shafts driving the rear wheels should be approximately equal length to prevent torque steer.
4. There must be a suitable method of tensioning the chain that allows for different tooth counts on the sprockets

Manufacturability:

1. The differential should be based on an existing limited slip gear system.
2. The entire assembly must be able to be manufactured with a minimal of specialized tooling. Tooling used should be available for general student use. For example, CNC milling and turning would not be objectionable, but wire EDM would be.

Integration:

1. The differential should be chain driven.
2. The differential should mount directly to the engine for modularity.
3. The rear brake components should share as many components as possible with the front brakes to reduce unique part count.
4. The differential mount should also provide a rear engine mount to the frame.

3.0 Mechanical Design

The logical place to start the design was with the mounting brackets. Of course, it would be foolish to fully design the brackets first and then expect other components to fit. The first step was to take stock of what mounting options were available, figure out the constraints that the brackets placed on other components within the system, and develop the rest of the drivetrain based on those constraints.

The next section describes the design of the brackets in detail. It should be noted that the interdependencies between components required that the brackets go through many revisions; only the final design is presented.

3.1.0 Mounting Brackets:

The MIT FSAE program has invested heavily in the development of the Honda CBR600 F4i engine. The stock motorcycle has swingarm and engine mounts as shown in Figure 1. The width of the swingarm mount, measured from the outside of the swingarm bushings, is very close to 6 inches.

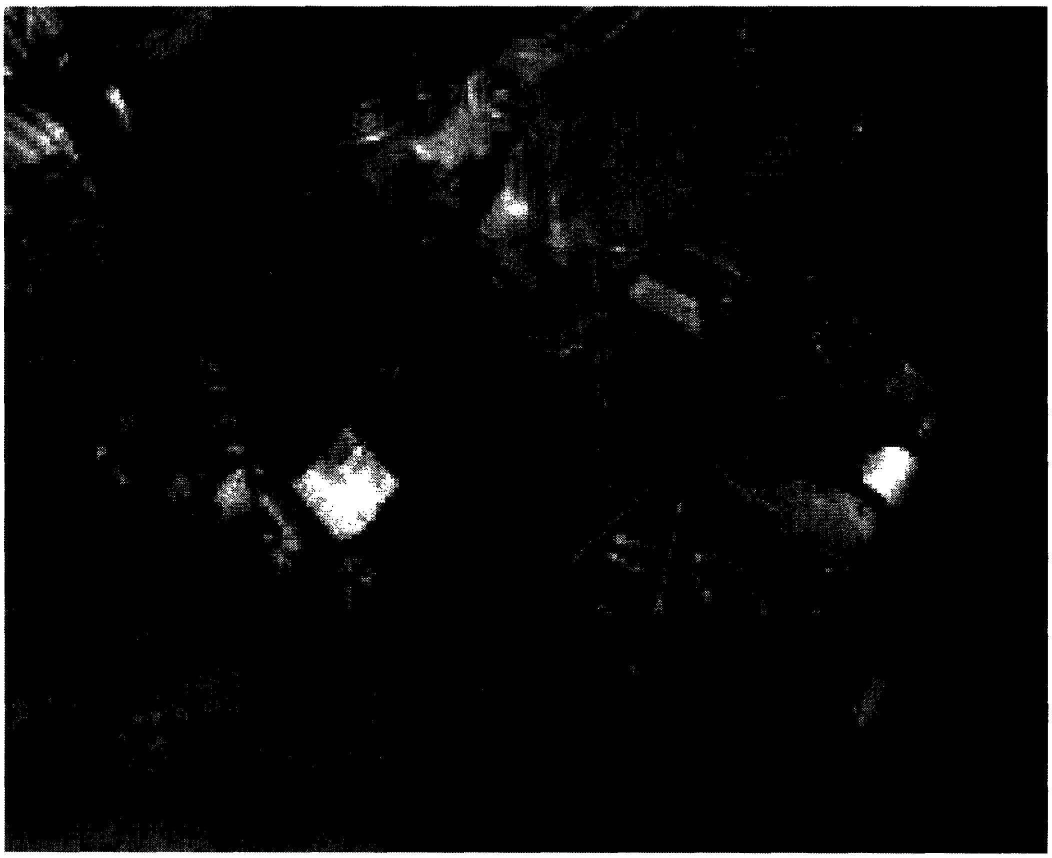


Figure 1: Swingarm mount of the Honda CBR 600 f4i engine.

Using this mount is desirable for strength reasons. Forces transmitted by the chain must be counteracted in equal and opposite forces in the structure connecting the output shaft of the engine to the input shaft of the differential. On a stock bike, tension forces in the chain must act through that pivot. This means that the pivot is designed to withstand these forces.

3.1.1 Benefits of a Direct Load Path

Because the load path ultimately must go through the engine, it makes little sense to attach the differential to the frame of the car. The maximum chain tension can reach up to around 2300 lbs, as will be demonstrated by calculation in section 3.1.7. This force can distort the frame under power; this can affect suspension geometry, create misalignment in the sprockets, and cause binding in differential support bearings. Additionally, maintaining the necessary accuracy to keep the bearings in line is difficult in steel weldments such as the frame. Proper triangulation is also difficult to achieve in the differential area, as so many mechanical elements restrict placement of braces.

Despite the benefits of a direct load path, mounting the differential directly to the engine is uncommon in FSAE. Some other factors that play into this decision are differential type, chain tensioning method, and manufacturing capabilities of the student and school. However, if a direct mounting scheme could prove both robust and lightweight while meeting the functional requirements, then there would be no reason at all to mount the differential to the frame.

3.1.2 Bracket Concept

Brackets meeting the above design criteria were solid modeled in CAD. A rendering is shown in Figure 2. Notable features include aluminum construction, direct engine mounting, a slot for a chain tensioner, and a window for easy access to the brake bleeder.

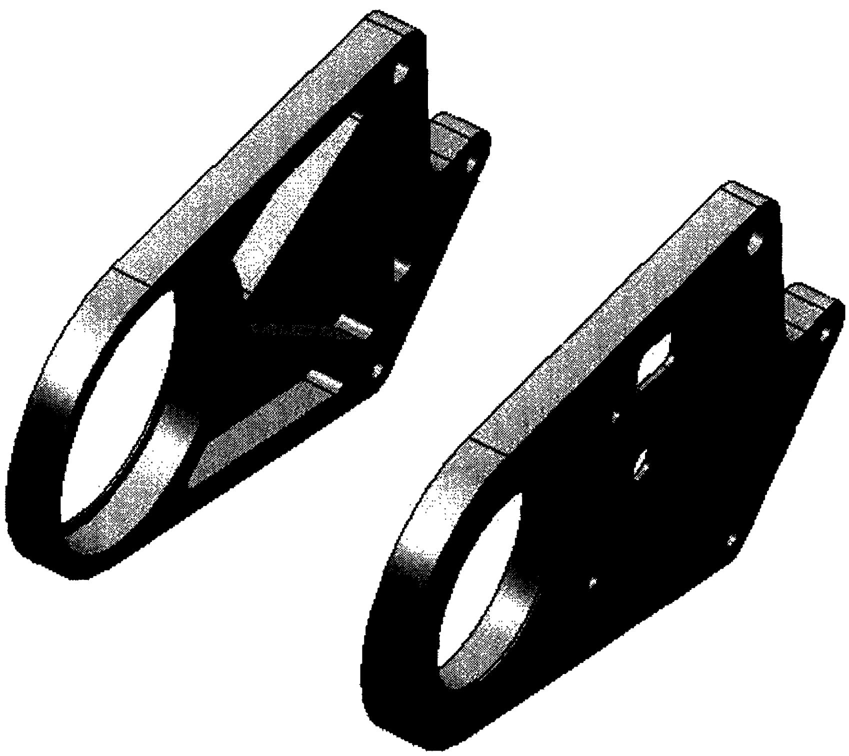


Figure 2: Solid model of differential mounting brackets.

3.1.3 Bearing Selection

Mounting the bracket to the swingarm mount poses one challenge: the plane that the chain operates is not between the brackets. This is solved by driving the differential through the bore of a large bearing.

A 6917LL bearing was used for the left side bracket. This bearing has dimensions of 85 mm ID x 120 mm OD x 18 mm width. The LL suffix denotes that the bearing is equipped with neoprene seals. This is standard nomenclature for metric interchange bearings. The 85 mm bore allows the differential carrier to have a large face for transmitting torque to the differential center section.

A 6012LL bearing was used for the right side bracket. This bearing has dimensions of 60 x 95 x 18 mm. Note that both bearings have the same width. This is a result of the 60xx series of bearing being a heavier class than 69xx. This was exploited to match the widths of the bearings, simplifying mounting. This allowed both bearing pockets to be bored to the same depth.

3.1.4 Mounting and Chain Tensioning

The bearing selection and the location of the rear axle fixed certain points of the brackets. Engine placement was decided by the chassis team early in their design work, so this also fixed more mounting points. The rest of the bracket was determined by the remaining functional requirements-- chain tensioning and engine mounting.

The chassis team found that connecting the back of the engine to a lower frame rail running across the back of the engine added significantly to the torsional rigidity of the chassis. Due to this finding, the bracket was designed to extend to this lower frame rail.

Originally, it was desired that the chain should be tensioned by sliding the brackets forward or back. Half links do not exist in motorcycle chain, so to be able to tension the chain in all circumstances, the sprockets need to be able to be moved by at least 5/8 of an inch. Additionally, because of the small front sprocket, the wrap angle of the large sprocket is well over 180°, meaning that the amount the bracket needs to slide is even larger. 1 inch of movement is a safe design parameter, though the exact amount can be calculated from the wrap angle.

Several methods of securely locking two piece brackets together were investigated, such as clamping the assembly with several large bolts, using eccentric spacers, or using shims. Ultimately, none of the methods could easily allow for 1 inch of movement and not interfere with the engine block, the sprocket, or the differential center section. While admittedly, it should be possible to make a sliding bracket without interfering with other components, the sliding brackets were abandoned in favor of a sliding idler sprocket.

A sliding idler sprocket has been used on the two previous MIT cars with moderate success. In both instances, the bracketry to support the idler had to be made extremely heavy to avoid distortion. Both cars had a plate of 1/4 inch steel welded between an engine mount and the frame; on both cars, however, the idler was added as an afterthought. Despite shoddy design work on the idler of the 2004 car, a plastic idler sprocket survived the endurance race. A properly designed and implemented idler should have no problems.

To accommodate for the idler, a 5/8 inch wide by 3.25 inch tall slot was designed into the left side bracket. The slot was oriented vertically and close to the front sprocket to minimize the chance of interfering with the drive sprocket should a larger sprocket be used.

3.1.5 Material selection

It was desired that the brackets be machined from aluminum plate for strength, weight, and ability to hold tight tolerances. CNC milling facilities are readily available for student use, so odd shapes and arcs do not detract from manufacturability. Originally, 2024-T65 was specified for the brackets, but was only available locally through special order. Due to time constraints, the material was changed to 6061-T6. As all aluminum alloys are treated equally on the cost report, 6061 should generally not be used for any structural billet parts of the car. 2024 and 7075 offer much greater strength and better machinability, without appearing to cost more on the cost report.

The material thickness was determined by the bearings used. Since both had an 18 mm (0.709 in.) width, a logical choice was 3/4 inch thick plate. This allowed the full width of the bearing to be supported while maintaining a 1/16 inch thick lip for the outer race to rest against.

3.1.6 Manufacturing considerations

Both brackets were designed such that all milling operations could be done from one side, including cutting the part free. This saved much machine setup time; the only setup necessary was setting the Z coordinates for each cutter and placing the origin such that none of the cuts went beyond the edge of the blank.

Radii were left as large as possible so that heavy feeds could be maintained. The smallest cutter used was 7/16 inch, which allowed for fast material removal rates. The bulk of the material was removed with a 3/4 inch cutter, which allowed for very fast roughing of the pockets. MasterCAM allows for automatic calculation of areas inaccessible by large cutters and can then machine just those areas with smaller cutters.

3.1.7 Estimation of reaction forces

The strength of the design must be validated, so an estimate of the reaction forces is needed for FEA. The reaction forces can be estimated with some basic assumptions about the traction of the car. The chassis team predicted a maximum of 1.8 g's of linear acceleration and a weight (including driver) of 680 lbs. Equation 1 gives torque T on the input sprocket of the differential in terms of mass M, acceleration a, and tire radius R_{tire}. With a tire diameter of 20 inches, the shaft torque is 1020 ft-lbs.

$$T = M \cdot a \cdot R_{tire} \qquad \text{(Eq. 1)}$$

The chain tension can then be estimated if the sprocket diameter is known. Assuming a 54 tooth 520 pitch sprocket, the pitch diameter is approximately 10.75 inches. Equation 2 gives the chain tension F in terms of torque and the radius of the sprocket $R_{sprocket}$. This results in a chain tension of 2280 lbs.

$$F = T / R_{sprocket} \qquad \text{(Eq. 2)}$$

This force is seen mostly by the left side bracket, as the right side bracket has a considerably longer moment arm restraining the differential housing. Based on the length of the moment arms, the force on the right side bracket from the chain tension is on the order of 1/6 that of the left bracket, or about 380 pounds. However, the right side bracket must withstand the torque exerted by the brake caliper acting on the rotor. This was estimated at 1000 ft-lbs, the limit of traction, though weight transfer to the front limits this torque to below that value.

3.1.8 FEA of Differential Brackets

The left side bracket forces and constraints were selected to mimic 2280 lbs applied to the chain. This force must be counteracted by a equal and opposite force through the center of the bearing; this gives rise to the torque that moves the car forward. This was modeled in FEA by fixing the bearing face and applying half of the chain tension to the top engine mount and the bottom frame mount. The force was applied parallel to the bottom of the bracket, as this angle is close to parallel with the chain. Figure 3 shows the mesh size, restraints, and locations of applied force for the purposes of design verification. The arrows normal to each other indicate a fixed restraint, while the parallel arrows are forces. The lowest safety factor reported by CosmosWorks was 12. This was located in the gussets running forward and back, where it would be expected. While this high a safety factor is possibly excessive, this bracket does often see shock loading from slack in the driveline. In the future, further analysis may show that more material could be safely removed from the bracket. If the idler slot were tilted from vertical, material could be removed in that area as well.

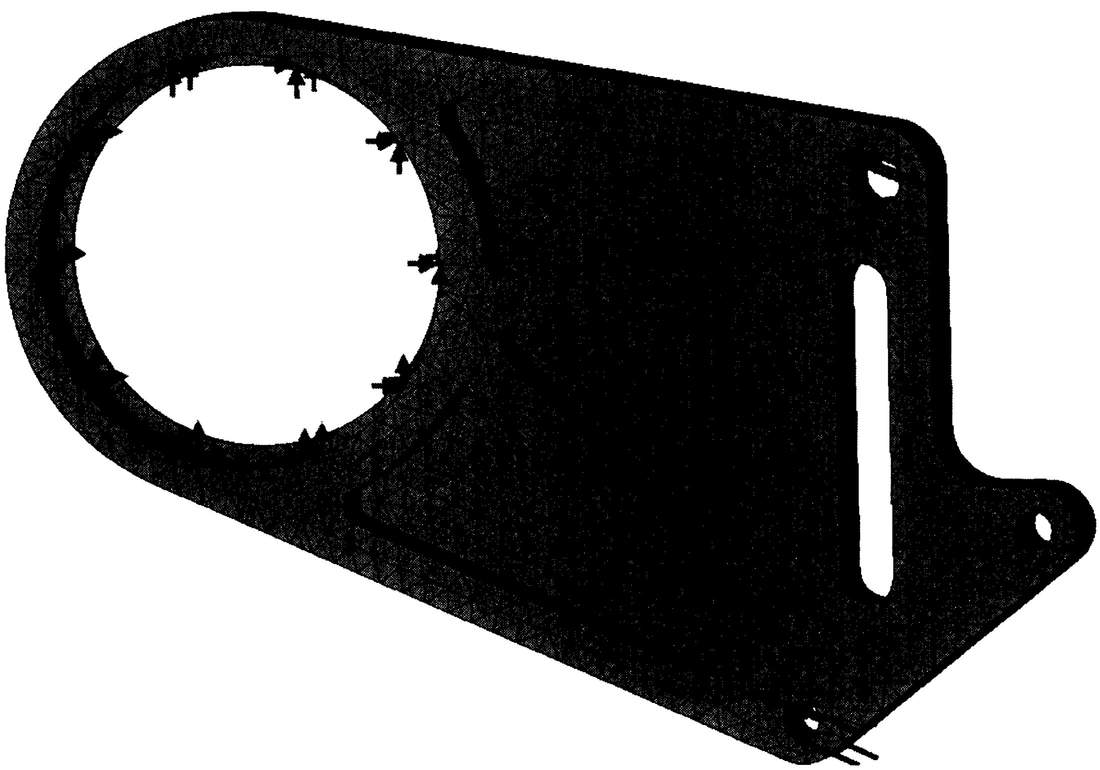


Figure 3: FEA mesh, forces, and restraints on left side bracket.

The right side bracket was expected to see the most loading when full brake was applied. This situation was modeled by fixing both the top engine mounting point, the bottom frame mounting point, and the bearing face. 1000 ft-lbs was applied through the caliper mounting bolts. CosmosWorks will automatically resolve torques into forces if a torque and axis is specified. Figure 4 shows the mesh, forces, and restraints used to analyze the right side bracket. The lowest safety factor reported was 4.9. This analysis did not include the effects of chain tension or engine restraint, as these forces are estimated to be far smaller than the braking forces.

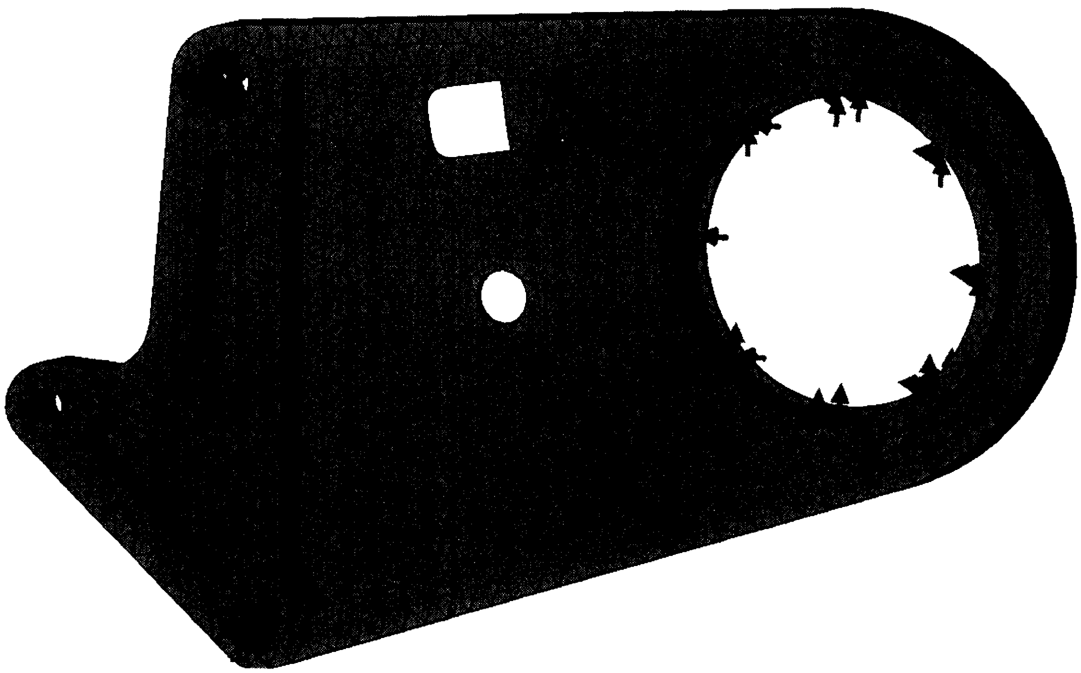


Figure 4: FEA mesh, forces, and restraints for right side bracket.

3.2.0 Differential center section

Both of the previous MIT cars used a Torsen "University Special" differential with good success. The stock cast iron housing, however, is quite heavy and much stronger than necessary. Torsen quotes the strength of the stock differential at about 5200 N-m, or about 3800 ft-lbs [2].

Some teams have had good success making a new housing out of aluminum. Aside from the obvious weight savings, creating a custom case allows better integration of the assembly. Also, the stock housing has little space for drilling and tapping. This makes the task of fitting end caps to the differential difficult and of questionable strength. The Torsen geartrain, however, is excellent for FSAE use and can be used in a custom aluminum housing quite readily. Torsen supplies a drawing of the differential case for those teams interested in making their own housing

The Torsen gear train uses two "side gears" operating on axis with the output shafts. The housing also supports six "element gears" divided among three windows. A picture of the Torsen geartrain is shown in Figure 5. Torque transmitted to the output shafts is transmitted from the rotating case to the side gears by the element gears. Torque biasing for limited slip is handled by the thrust forces generated by the angled tooth contact between element and side gears. A complete description of the operation of this differential is available from Torsen [3]. The Torsen document also provides a mathematical model of the differential, but the coefficients of friction particular to the University Special are not included on any external document. These could be estimated from coefficients of friction for oiled surfaces, but analysis of the geartrain is only necessary if one wishes to improve the performance of the differential by changing friction surfaces. As of date, the MIT team has not pursued this avenue.

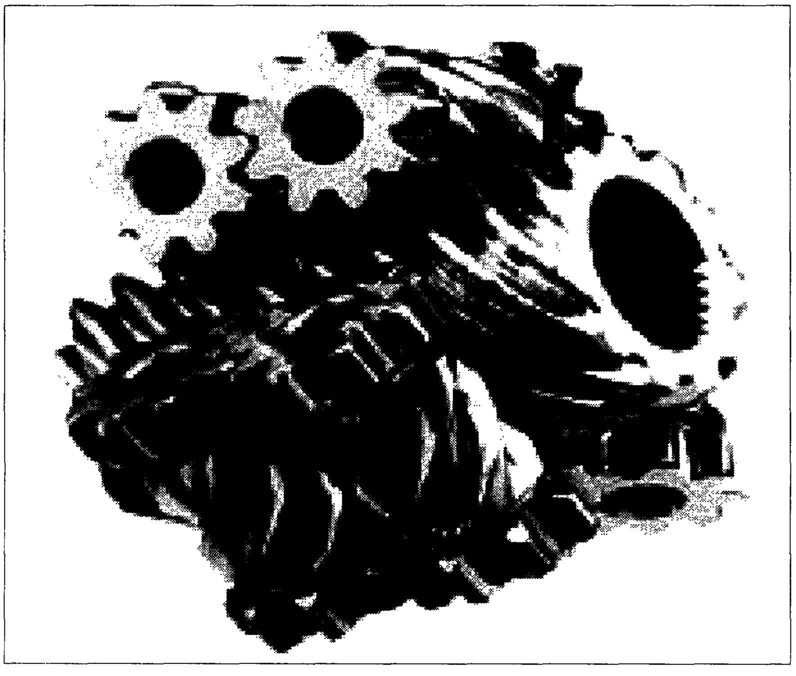


Figure 5:Torsen geartrain, shown without case or journal pins [3].

3.2.1 Differential center section concept

It was decided that the Torsen case should be replaced with a custom aluminum housing to reduce weight and improve integration with drive and brake components of the car. The aluminum housing concept is shown in Figure 6.

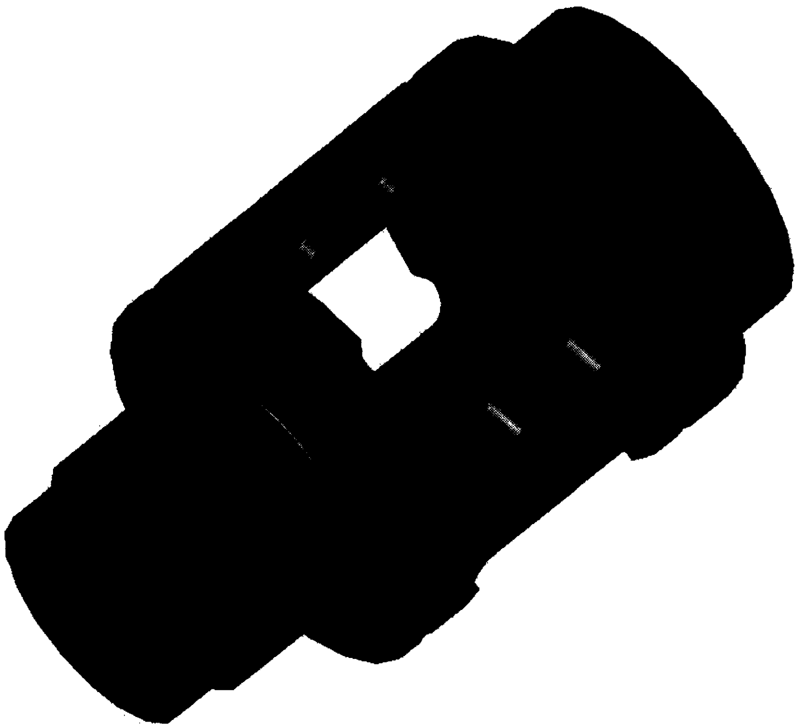


Figure 6: Solid model of custom Torsen housing.

3.2.2 Material Selection

7075-T651 was selected to make the housing out of. This is one of the strongest aluminums available without losing too much ductility. 7075 is also readily available. Stronger aluminums exist, but the yield strength on these alloys is very close to the ultimate strength. This makes them prone to failure in an application that sees cyclical loading, such as this differential.

3.2.3 Torque Transmission

The bearing selection for the brackets fixed the dimensions of the input side of the differential at an OD of 85 mm. Additionally, the output shafts of the differential must be supported on bearings. The final selection of these forced the ID to be 42 mm. Three bolts and three dowel pins were used to transmit the torque to the center section of the differential. 3/8 inch shear pins were used with 5/16"-24 bolts. The bolts merely supply clamping force, so large bolts are not necessary for torque transmission. Fine threads are generally not recommended for aluminum, and even coarse threads are prone to failure when tightened repeatedly in aluminum. For this reason, Helicoils were installed to increase the tear out strength of the bolts.

The final dimensions of the drive flange are shown in Figure 7. Note that dimensions are in inches, though metric bearing dimensions determined the two diameters shown at the top of the diagram.

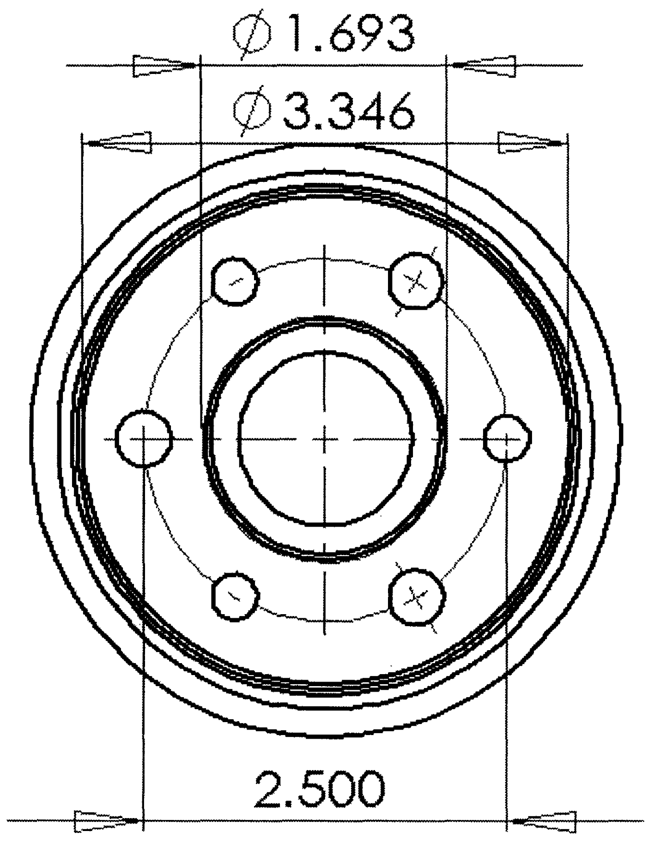


Figure 7: Drive flange dimensions.

The torque that the shear pins can transmit can be estimated by Equation 3, where D_{pin} is the diameter of the shear pin (3/8"), τ_y is the shear stress of the pin, N is the number of pins, and R is the radius of the bolt circle.

$$T = D_{pin}^2 \cdot \frac{\pi}{4} \cdot \tau_y \cdot N \cdot R \qquad \text{(Eq. 3)}$$

Using a shear stress of 130,000 psi, this equation gives a drive side shear strength of 4490 ft-lb, and a brake side shear strength of 5830 ft-lb, as the brake side bolt circle is bigger. A thorough analysis would also include contact stresses, because the aluminum is likely to yield before the shear pin. An indication of what the shear pin could support is given by Equation 4, where $\sigma_{avg,b}$ is the average bearing contact stress in the aluminum and L is the length of engagement of the pin. For 1020 ft-lbs and a pin engagement of ½ inch, $\sigma_{avg,b}$ =17400 psi. With the yield strength of 7075-T651 at about 73,000 psi; this leaves a safety factor of 4.2. Admittedly, the stress distribution will not be flat across the pin length. However, should the yield stress be exceeded locally, residual stresses would limit the maximum stress to the yield value, while areas with stress under the average value would have the stress elevated to keep the pin in static equilibrium. Thus, Equation 4 can be used with reasonable confidence as long as a large safety factor is used.

$$\sigma_{avg,b} = \frac{T}{R \cdot N \cdot L \cdot D_{pin}} \qquad \text{(Eq. 4)}$$

3.2.4 Journal Pin Retention

Another modification to the Torsen design was necessary for manufacturability. Stock Torsen journal pins are retained with Spirol roll pins. This method was not used for the aluminum housing because it presented two major problems. First, the bit necessary to drill these holes would have to extend past the chuck about 6 inches; the likelihood of a small diameter bit this long is high. Second, Spirol pins would provide a leak path that would have to be sealed with silicone or o-rings. Instead, 1/16 dowel pins were used for journal pin retention. These were drilled perpendicular to the axis of the journal pins and located in an area already sealed by an aluminum sleeve. These prevent the journal pins from sliding axially, and the sealing sleeve surrounding the entire differential keeps the retaining pins from sliding out. Figure 8 shows the retention pins as viewed coaxially with the pin.

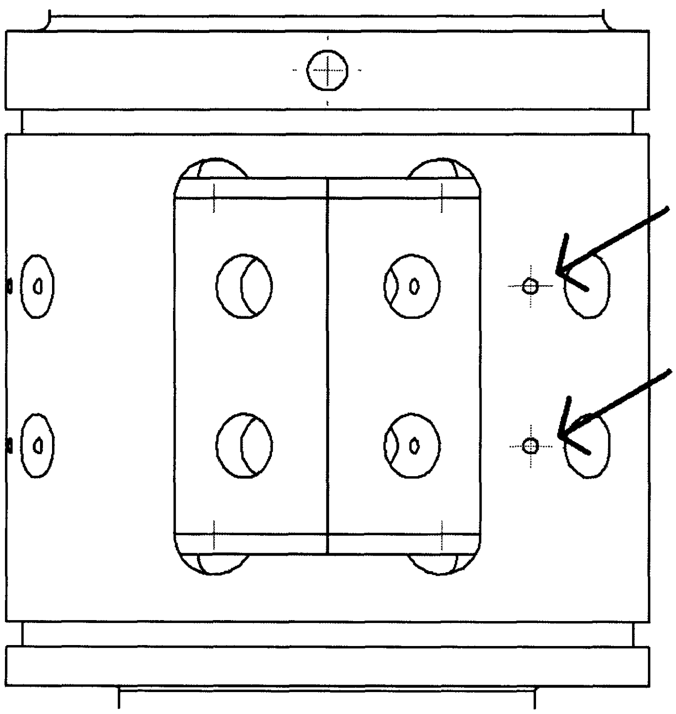


Figure 8: Journal pin retention (see arrows).

3.2.4 FEA of Differential Center Section

Internal forces in the Torsen gearing were calculated by Richard James in his senior thesis [7]. His thesis also goes into much detail of the operation of the Torsen geartrain. In depth analysis of the gear forces will not be included here.

Under acceleration, the side gears are forced to the left, while the reaction force pushes the element gears to the right, as viewed from the back of the car. A worst case scenario for thrust force is when the maximum torque is applied to the input. 1020 ft-lbs causes 76000 N to be exerted on the face where the leftmost washer rides on the housing. The 76000 N is also divided equally among the 12 places where the journal pins contact the housing.

Figure 9 shows the differential housing with the shear pins holes restrained, 1020 ft-lbs applied to the element gear thrust faces, and appropriate forces applied to the left thrust face and journal pin holes such that 76000 N of internal axial thrust force is attained. The mesh is not shown, as this makes the forces and restraints difficult to see.

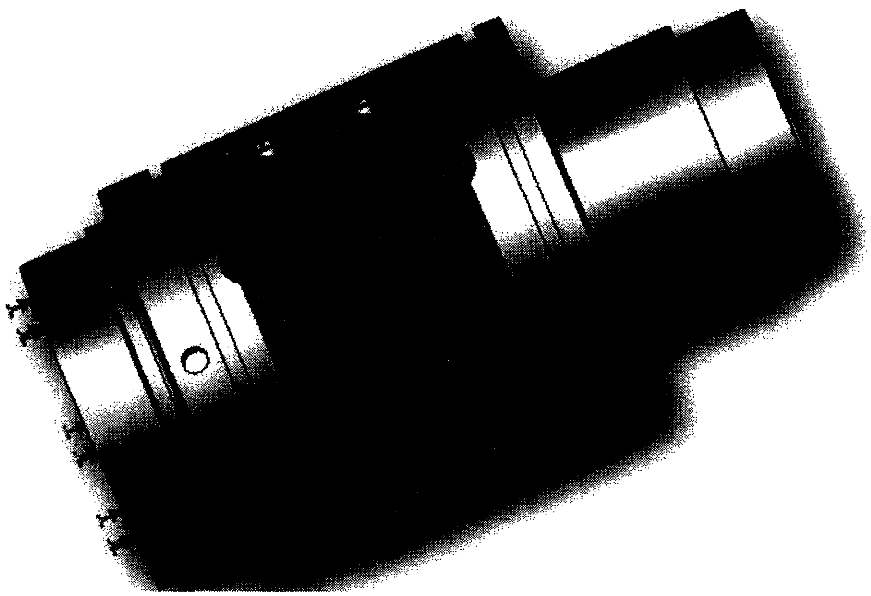


Figure 9: Forces and Restraints on FEA of differential housing.

The lowest factor of safety found was 2.8. This was near a stress concentration; most of the housing had safety factors of between 6 and 8. Should the design stress be exceeded, localized yielding would prevent large scale failure of the part. It is likely that this part is overbuilt; at 3.9 lbs, the weight is roughly the same as the cast iron Torsen piece. However, the integration of the part with the brackets, brake system, drive system, and axle supports is superior to the Torsen housing. Additional material would have to be added to the stock housing to meet these requirements, so there is definitely a net reduction in weight from stock. Weight could be reduced even further in this design, but this particular compromise of weight and functionality was deemed adequate.

3.2.5 Thrust Washers for Element Gears

Other dimensions of the housing were driven by the functional requirements. The internal dimensions were largely taken from the Torsen print [8], though the width of the windows was increased by 0.160 inches to allow for 0.080 inch thick hardened thrust washers to be placed between the element gears and the housing. These thrust washers were shaped like and "8" to prevent rotation when installed. This guarantees that the gear rides on a steel washer, rather than the washer sliding on the housing. This prevents wear on the differential case and makes the case rebuildable by replacing washers.

The washers were machined from prehardended 4142 steel. This required a carbide cutter, high spindle speeds, and low feeds. Using prehardened steel, however, eliminated a heat treating step. This would have required sending the parts out for processing.

The 0.080 thickness was chosen as a compromise between the stiffness of the washer and the engagement of the journal pins. The thicker the washer, the less the pin engages in the housing. McMaster had 4142 prehardened stock available in 0.094 inch thickness, so MIT Central Machine surface ground the washers to final thickness after the milling operation. A view of the washer is provided in figure 10.

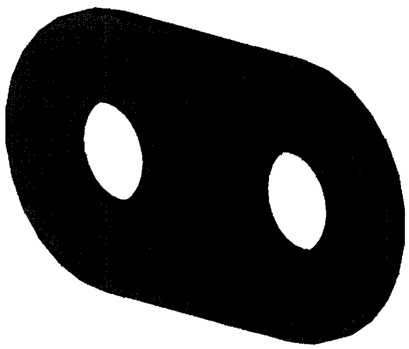


Figure 10: Isometric view of the thrust washer

3.2.6 Case Sealing: O-ring Glands and Sealing Sleeve

The windows in the differential housing must be sealed with a sleeve. O-rings were used to seal between the housing and the sleeve. For the o-ring material, Viton was selected for it's compatibility with gear oil and temperature resistance to 400°F. Buna N was also investigated, as it is cheaper and also compatible with gear oil, but was ruled out due to poor performance beyond 250°F [5].

The o-ring gland on the center section was designed to provide for 25% crush. With the nominal cross section of the selected o-ring at 1/8 inch diameter, the groove was designed to be 0.100" in depth and of 0.156" width. The additional width is to allow the o-ring to compress while not being restrained axially.

The outside diameter of the center section was selected based on manufacturability. The windows on the Torsen housing must be sealed with a sleeve, as it is 4 inch schedule 40 aluminum pipe is readily available. This has a nominal inside diameter of 4.026 inch. To make sure there was enough material to bore the inside of the pipe to final diameter, 4.080 inch was selected for the OD of the center section. 4.080

inch was used instead of a fractional size, such as 4.0625 inch for a simple but subtle reason: 80 is divisible by 2. This helps with manufacturing, because the center of the billet must be found with an edge finder. A 4.080 inch OD also allows a 4 inch o-ring to be used without fear of stretching the o-ring too far.

The sealing sleeve was matched to the cylindrical portion of the differential in size. The sleeve was bored 0.005 inch over to allow enough space between the two parts to prevent galling.

It is important that the sleeve not rotate or slide axially to prevent o-ring failure. A 1/4 inch dowel pin was press fitted in the differential housing to torque to the sealing sleeve and prevent relative rotation. This also provides a stop that prevents the sleeve from being able to move axially toward the drive sprocket. Additionally, the brake flange on the right side of the differential provides a positive stop on the opposite side.

Two other small details were included on the design of the sleeve. A 15° ramp was cut on the left side of the sleeve to facilitate loading of the o-ring. This prevents shearing of the seal when the sleeve is installed. Also, the housing was drilled and tapped for a 1/8" NPT plug. This is the oil drain and fill plug for the differential. It is important that the plug be located over a window to prevent the plug from hitting the case. Two issues with the drain plug went unaddressed in this design, though these could be fixed by purchasing a different plug than the one available from McMaster Carr. One, the drain plug should ideally be magnetic to catch wear particles from the gears. Two, the drain plug should be installed with some kind of locking mechanism, such as safety wire or a setting pipe dope.

A rendering of the sealing sleeve is provided in Figure 9.

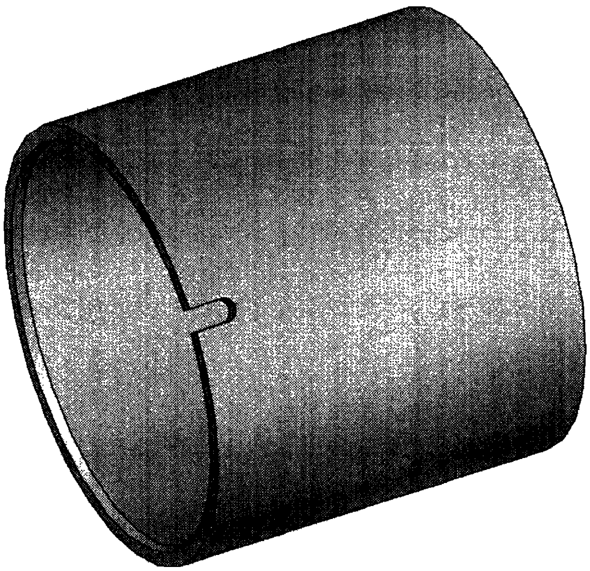


Figure 11: Aluminum sealing sleeve.

3.3.0 Brakes

The functional requirements dictated that the rear brakes share as many components with the front brakes as possible. Ultimately, the brake rotor, the Wilwood Billet Dynalite Single caliper, and the rotor mounting bobbins were shared front and rear. The front brakes employ a 10 inch cast iron floating rotor. A floating rotor is allowed limited radial and axial play, which reduces thermal stress and improves braking response, according to some drivers.

The rotor mounting hat was designed for a 5.875 inch bolt circle with 8 bobbins of 14 mm diameter. A 5.875 inch BC was chosen because the availability of 6 inch round stock. This pattern was selected for manufacturability; a 5.875 inch round can be machined from 6 inch nominal diameter stock, even if the stock is under nominal or is scarred from mishandling of the raw stock, a surprisingly common occurrence.

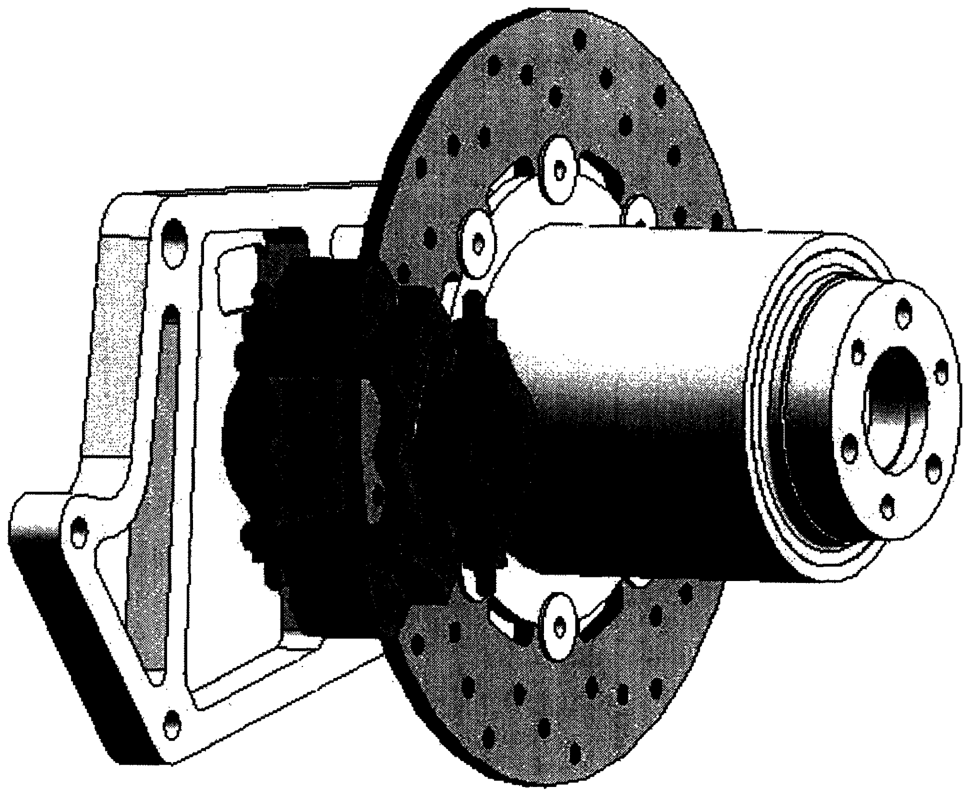


Figure 12: Brake components, shown with left differential bracket hidden.

Late in the design process, it was found that the bobbins could interfere with the swept area of the brake pad. Though caliper mounts had not yet been machined, rotors had already been custom ordered. To accommodate for the interference, the calipers were moved radially outward 1/16 beyond the distance recommended by Wilwood of 3.48 inch [6]. This change was made on both the front and rear of the car. In the future, the bobbin bolt circle should be reduced to 5.75 inch to increase clearance. Alternatively, the rotor diameter could be increased by 1/8 inch and the caliper moved radially outward by the same amount. Interference with the front wheels would have to be checked before choosing this route, as the clearance is already minimal. Wheel deflection must also be considered when checking for interference.

The caliper mount was integrated into the right side differential bracket. The caliper was bolted to it in an unconventional way; the outside face of the mounting ears bolt to the bracket, rather than the inside face. This does increase the bending moment on the caliper by about 60%, but FSAE cars are very light and the caliper is intended for much heavier drag racing applications. The caliper was deemed to be designed robustly enough to endure this kind of abuse. The unconventional mounting allowed the bracket and caliper mount to be milled from a single piece of stock and also kept the caliper inboard of the bracket. This kept cost and complexity down, as the large bearing required for the driven side of the differential was very expensive.

This caliper mounting method required that the brake line be installed through the right side differential bracket. A hole was included in the bracket to allow a fitting to be installed from the outside. A brake bleeding window was also provided at the top bleeder screw. Details such as the bleeder window add much to the finished product, while taking very little time to design and implement.

3.3.1 FEA of Rotor Mounting Hat

CosmosWorks was used to verify the strength of the rotor mounting hat. 1020 ft-lbs was applied to the bobbins and the dowel pin holes in opposite directions. To restrain the system for FEA, the 5/16" bolt holes were chosen for restraint. This allows the material to deflect in both places where forces are applied, a fixed restraint on the shear pin hole would not allow movement, which affects the stress in those locations. Figure 13 shows the mesh size, the restrains, and the loads on the rotor mounting hat. The lowest factor of safety found was 5.3.

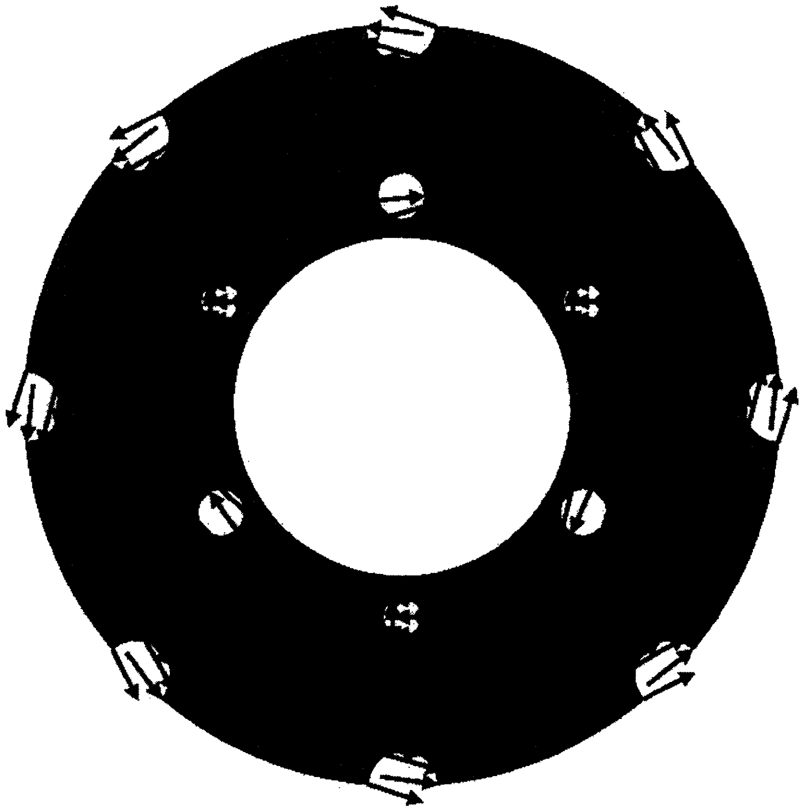


Figure 13: FEA of rotor mounting hat, showing mesh, forces, and restraints.

3.4.0 CV Joints and Axles

Several options exist for lightweight constant velocity joints. Two companies produce axle kits that lend themselves to FSAE application, Taylor Race Engineering and Rockford Acromatic Products. Some teams also make their own or use production parts from ATVs. Due to time constraints, it was decided that a commercial kit should be used.

MIT's 2003 and 2004 entries used the Taylor Race kit, but there are two major drawbacks: cost and completeness. A complete Taylor race kit costs about $2000. Additional machine work is also required, such as shortening axles and cutting grooves for retaining rings. The quality of the Taylor race kit is high; all steel parts are heat treated and polished 4340, steel surfaces are plated or black oxide treated, axles are gun drilled, etc. The fit and finish is excellent, but the price is too high for a cost sensitive application such as an FSAE car.

The cost of the Taylor race kit can be reduced significantly by substituting parts. 1980-84 Volkswagen Rabbit/Jetta/Scirocco hubs are dimensionally compatible, though heavier and of questionable strength. This is also the first year that Taylor race has offered a splined shaft compatible with the Torsen center section. This was previously made by welding a 4340 splined shaft (available from Paradigm Motorsports) to a custom turned flange compatible with the Taylor Race tripod housing. This custom work cost less than the newly offered inner stub shafts, but came at the cost of a significant amount of labor being invested in the parts. Also, the welds on 4340 are difficult to perform and are likely to crack. Preheating and post heat treatment is absolutely necessary.

Last year, Rockford Acromatic Products introduced a lightweight CV joint kit intended for Formula SAE applications. The retail price on this kit is $1000. This includes the inner and outer stub axles, tripods, shafts, and boots. There is only one major drawback with the Rockford performance kit. They are specifically designed for a Polaris Sportsman 750 hub. When approached about custom splines to match the Volkswagen hub used on previous MIT cars, Rockford suggested that we could make our own aluminum hubs or use a late model Volkswagen hub with an adapting sleeve.

The Rockford kit is simplicity at it's finest. Taylor controls axle plunge with a complicated system of springs, plastic buttons, and plastic plungers. Plunge is controlled on the Rockford kit with the boots; the stiff rubber makes an adequate spring. The Rockford kit also weighs considerably less, as it is specifically designed for FSAE use and uses stub shafts with integrated tripod housings. The Taylor kit is designed for D Sports Racer class cars, which weigh more and have more power than the typical FSAE car.

3.4.1 Inner Stub Shaft Bearings and Seals

It was desired that the stub shaft would be supported on sealed ball bearings. The 2004 car used bronze bushings; due to insufficient lubrication, these lasted only a short time. Radial play in the shaft quickly causes seal failure, and zero leakage was one of the main goals of this design. A 61806LL bearing was selected for this application due to the small size necessary to mount the bearing. This bearing is often used in bicycles, so it turned out to be readily available and low cost. This bearing has a 30 mm bore and a 42 mm outside diameter. Rockford was asked to grind the inner stub shaft to 30mm, which was close to their standard size. Figure 14 shows the left side inner stub shaft inserted into the differential housing. The stub shaft is just a placeholder, as the only available drawing of the Rockford part is a sketch of the outside dimensions of their kit.

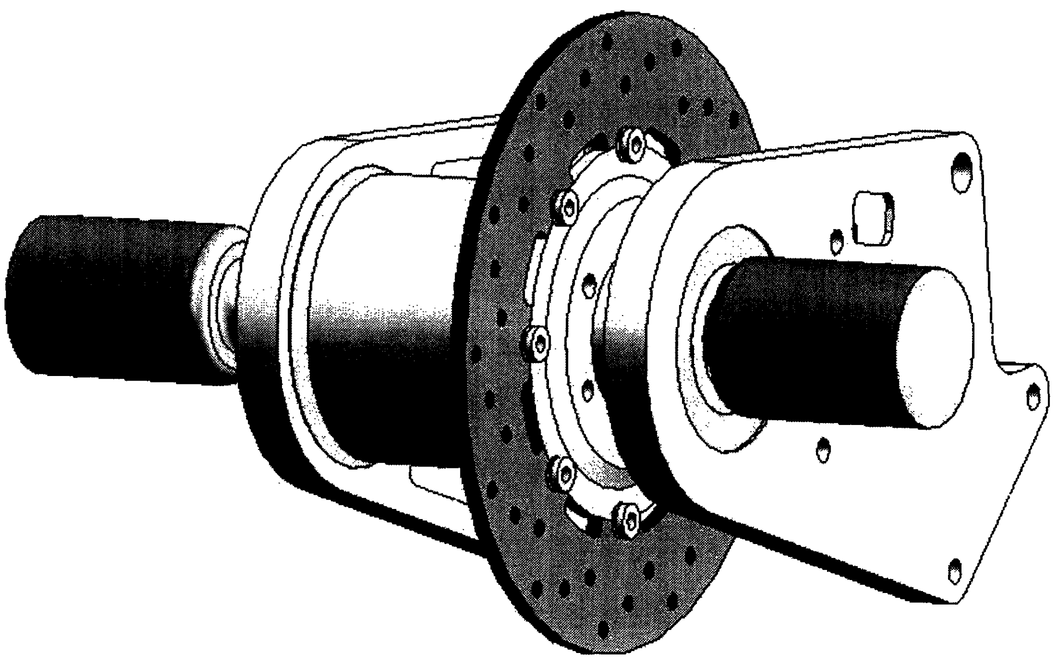


Figure 14: Differential housing with inner stub shafts shown.

The Chicago Rawhide online catalog was consulted for seal selection [8]. A CRW1 cross section was selected, as this represented a common small seal design that would be readily available from distributors. Viton was chosen for the seal material, as CR rates temperature operation to 400 °F. Nitrile seals were only rated to 250 °F. The smallest outer diameter available with a 30 mm bore is 40 mm, which is a part number 11606.

The CRW1 cross section is rated to 10 psi at up to 1000 surface feet per minute shaft speed. 1000 SFPM translates into over 3200 rpm for a 30 mm diameter shaft, so it is a safe assumption that these seals could withstand up to 10 psi in the differential case. Using the ideal gas law, the pressure rise due to an isochoric heating of air inside the differential case can be found by equation 5. Note that both pressures and temperatures must be on an absolute scale. A quick calculation shows that even if the temperature of the gearbox increases from 50° F to 300° F, the pressure rise would be 7.2 psi over atmospheric. 50° F is a reasonable lower bound for ambient conditions in Detroit during the competition, and 300° F is an upper bound of what the gearbox oil can tolerate.

$$\frac{T_{cold}}{T_{hot}} \cdot P_{cold} = P_{hot} \qquad \text{(Eq. 5)}$$

It is actually possible that the differential oil temperature could exceed 300° F in operation; calculations for heat generation were not performed. However, this high a temperature oxidizes the oil as well as risks oil escaping past the seal. Should testing show temperatures this high, the differential case should be finned to increase heat dissipation.

3.4.2 Rear hubs

The rear hubs were designed to use a Volkswagen dual row angular contact ball bearing. This bearing is used on the 1980-1984 Volkswagen Jetta, Rabbit, and Scirocco. This is also the bearing specified for the Taylor race axle kit and the bearing used on the front of the 2005 MIT FSAE car. This allows for a reduced unique part count. This bearing was selected for it's small size, more than adequate strength, built in seals, and easy mounting. Mounting the bearing is much easier than using two tapered rollers because the split inner race is ground such that when clamped together, bearing clearance is already set. The hub model is shown in Figure 15.

The hub was designed around a 4 x 100 mm bolt pattern. This matches the 2004 car, and is also a popular pattern for imports. This allows much flexibility in wheel selection. The hub was drilled to accept 1/2"- 20 studs with a 35/64" inch splined shoulder. These studs are available from Autozone, though the actual application is unknown.

The hubs were machined from 2024-T351 stock. 7075-T651 would have been preferred for higher strength, but 2024 stock in the right size was already available in the shop, while 7075 is a special order from Admiral Metals. The spline was broached by RCV performance to match their axle kit. The cost for broaching was very reasonable at $30. Other options such as wire EDM would not be nearly as cost effective.

For FEA purposes, restraints were applied to the cylindrical and flat faces where the bearing is located. Two cases were tried, a drive torque of 670 ft-lbs, and a 670 ft-lb bending moment applied at the front face by the wheel. The expected values are in fact the same; the tire can generate a maximum moment equal to the maximum force at the contact patch multiplied by the radius of the tire. It doesn't matter if the acceleration of the car is in a forward or lateral direction. According to estimates from the suspension team, each tire can transmit roughly a maximum of 800 pounds with all of the weight of the rear on one tire, a common situation in a tight turn. At a radius of 10 inches, the drive torque and bending moment is limited to about 670 ft-lbs per wheel. The lowest safety factors were 5.8 for forward acceleration and 2.8 for lateral acceleration. It is important to remember, however, that the wheel provides additional support that prevents the hub from deflecting as much under lateral loading, reducing stress on the component from the values estimated by FEA.

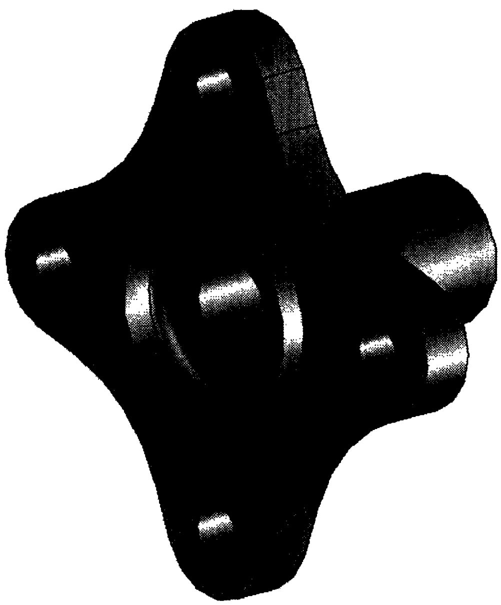


Figure 15: Rear hub CAD model.

A danger with aluminum is that it will fatigue and crack under cyclical loading. Further analysis of the hub should be done to make sure that the hub can last through enough cycles to complete a season of testing and competition. 200,000 cycles would be an adequate design life, as this corresponds to about 200 minutes of full speed operation. In reality, much of the racing conditions are at much lower speed, increasing run time.

3.5.0 Drive Sprockets & Chain

The rear sprocket and chain was sourced from Patriot Sprockets. In the past, the MIT team has used Sprocket Specialists, but were backlogged due to material delivery problems. The front 12 tooth sprocket was bought from another FSAE team, but was ultimately manufactured by Sprocket Specialists. Custom front sprockets require a minimum order of 6 pieces, so placing group orders with other teams is common.

A 54 tooth sprocket was specified for the rear; this gives a final drive ratio of 4.5:1. This was a slight increase in reduction from previous the 2004 design, which used a 4.13:1 final reduction. The custom 12 tooth front sprocket allowed for a smaller rear sprocket than in previous years.

Non o-ring chain was selected for reduced friction. While o-ring chains last much longer and are relatively maintenance free, constant maintenance is normal on FSAE cars. The idler also causes the chain to see much more flexing, which causes even more losses with an o-ring chain than in a normal application. Motorcycle chain is also very expensive-- 70 dollars is typical for non o-ring chain. O-ring chain costs twice as much. The cost and friction benefits of non o-ring chain makes the choice easy. The consequence of this decision is that the chain must be kept clean and lubricated—not a problem on a race car that is serviced regularly.

D.I.D. 520-ERT racing chain was selected as a compromise between strength, price, and availability. This is a light weight chain, but its breaking strength of 8470 pounds compares well to standard 520 pitch o-ring chain with an approximate breaking strength of 9000 lbs. Also, the Patriot Sprockets stocked this chain, so it was chosen over some lighter duty chains that had to be special ordered. D.I.D. publishes an excellent selection chart that lists breaking strength, dimensions, weight, and seal type [].

3.5.1 Chain tensioner

Due to the decision to use one-piece mounting brackets, an idler sprocket was needed to tension the chain. Teams seem to differ on opinion of whether an idler sprocket is a good engineering choice. Anecdotal evidence exists that idlers have failed catastrophically during the endurance race. Some other teams have successfully used idlers for years, and point to poor implementation as a cause of failure. A plastic idler survived the Endurance race on MIT's 2004 competition entry, though with much visible wear. This suggests that a properly designed and built idler should have no problem with longevity.

Industrial 50 pitch idlers are widely available. However, motorcycle 520 pitch chain is narrower, so an industrial idler would have to be cut down in thickness and deburred extensively. Instead, a custom steel idler was designed. This allowed it to be designed for manufacturability and serviceability.

The design included an internal snap ring to retain the bearing. This allows quick bearing changes in case of failure. For the idler sprocket, a 14 tooth steel sprocket was ordered from Patriot Sprocket along with the other sprocket and chain. The application of this sprocket is a Honda CR250.

This sprocket was bored to accept a bearing housing, also made of steel. The 14 tooth sprocket was impossible to hold in a 3 jaw lathe chuck, so an emergency collet was machined to the diameter of the sprocket. This allowed it to be held securely while turning. Optionally, the sprocket could have been set up in a mill and bored.

A 1018 steel ring was TIG welded to the inside of the sprocket. The weld showed signs of cracking, likely due to high carbon content of the sprocket. A better option would be to braze or silver solder the sprocket to the inner ring, or to make an entire sprocket and bearing pocket out of a single piece of aluminum. An aluminum sprocket would save weight and should be durable enough, but there was not enough time to pursue this option further. Also, Patriot Sprockets would not make aluminum sprockets with fewer than 20 teeth and was only set up to machine from plate stock, so the sprocket would have to be machined in house.

The idler sprocket and mounting design is shown in figure 16. A T nut prevents rotation of the nut when tightening; this engages a slot in the right side differential bracket. Note that the hole in the T nut was drilled offset from center. This was intentional; it allows the nut to be flipped to increase the effective travel of the idler. Clamping force is transmitted through the inner race of the bearing to the spacer. This spacer has a wide base to withstand the bending moment applied by the chain tension, as well as prevent deformation of the soft aluminum bracket. This does mean that the inner race of the bearing sees very large clamping forces. This was chosen over transmitting the clamping force through the wall of the spacer, which must be quite thin for the bearing chosen. If a bigger bearing could be fitted, clamping through the wall of the spacer would be preferred. A Belleville or wave washer could then be used to prevent rotation of the inner race on the spacer, as well as limit axial movement of the bearing.

Using a taper to increase the normal force was considered. However, this creates force components in the plane of the bracket which would cause a large bending moment in the bracket in an area. This idea was abandoned for using a single bolt to apply a large clamping force on the bracket. The 7/16" bolt used can develop approximately 7000 lbs of force at half of yield. Assuming a coefficient of friction of 0.2, the tensioner would require 1400 lbs to slip. If the surfaces are clean, the coefficient of friction can be even higher.

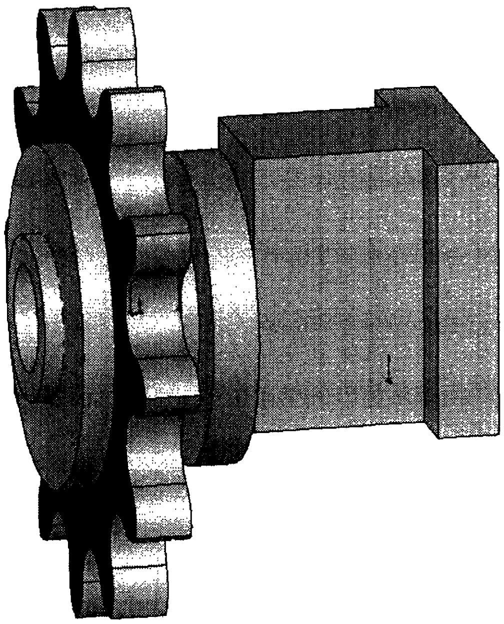


Figure 16: Chain tensioner assembly.

A 6003LL bearing was specified for the tensioner. This bearing is rated at 1200 lbs (dynamic) at 13000 RPM [9]. The idler can be expected to rotate at speeds of up to 10000 RPM, so the speed rating is adequate. While the load capacity would ideally be higher, fitting a bigger bearing into a 14 tooth sprocket is difficult. The maximum load can be calculated from the wrap angle and the chain tension while engine braking. Assuming a maximum wrap angle of 60°, the force on the tensioner simply becomes the chain tension while engine braking. The chassis team estimates the maximum braking force to be about 70% of the acceleration force, or about 1500 lbs. This is the limit of traction; the limit of engine braking may be considerably lower. It is also unlikely that the wrap angle will ever need to be as much as 60 degrees. For the 54T/12T combination used for initial car testing, the wrap angle is nearly flat; no idler would even be needed in this case. Idler loading, should one be used at all, is essentially zero with the flat wrap. Thus, by varying sprocket size in small increments, poor wrap angles can be avoided.

3.6.0 Bolts, nuts, and other hardware

Aircraft AN specification bolts were specified exclusively for several reasons. One, they are available in an assortment of grip lengths. The threads on these bolts are only slightly longer than the nut, which prevents threads from acting as a bearing surface if the proper grip length is specified. Two, they are heat treated and tempered to 125 ksi yield strength. This offers greater strength than SAE grade 5 hardware while keeping more ductility than grade 8 hardware. These bolts also have thinner heads than SAE hardware, making them lighter.

MS21042 nuts were used for critical applications requiring positive locking. These nuts have a smaller hex than the SAE or AN standard, yet develop the full strength of the bolt. These are an all metal prevailing torque locknut; the nut is deformed to prevent loosening. These are ideal for racing due to their high strength, light weight, and locking ability. These were used to fasten the brake caliper to the bracket, the bracket to the engine, and the sprocket to the input bell.

Bolts threaded into the differential case must also be secured to prevent loosening. Drilled head AN fasteners were specified in these locations so that they could be safety wired. This is another advantage of AN bolts; drilled heads are readily available off the shelf.

AN washers were used throughout. These are small, light, and made to tight tolerances. They are also hard enough to prevent yielding of the washer when used with high strength AN bolts. AN washers are just slightly bigger than the bolt head. SAE washers tend to be considerably larger than the bolt head, adding weight and providing little benefit for components with tight tolerances.

Two long studs were needed to fasten the differential to the engine. Threaded rod was not desirable because threads make a poor bearing surface and would damage the aluminum brackets. Commercial heat treat 4140 bar stock was threaded on both ends to manufacture these studs. The ASTM B7 bolt specification calls for 4140 steel, so this material is quite suitable for high strength fasteners.

4.0 Center Section Manufacturing and Assembly

A detailed manufacturing plan was conceived before any machining took place. With the time investment per center section at about 30 hours, a machining mistake can set the project far behind schedule. Also, the cost of each aluminum blank was about $90, so mistakes also become expensive.

The differential housing was machined at Edgerton shop using a manual lathe and an EZ-TRAK CNC mill. No EDM, horizontal mill, CNC lathe, custom broaches, or any other specialized equipment was necessary.

First, the housing was turned to the final 4.080 inch diameter and faced on one end. The faced end was also center drilled. This center is of great importance for setting up accurately in the mill.

A dividing head with a 4 jaw independent chuck was bolted to the mill table. The aluminum billet was then mounted on the dividing head with the faced and center drilled end facing away from the dividing head.

A dial test indicator was mounted in the spindle for checking setup. The concentricity was checked at both ends of the billet. 0.0002 inch TIR was attained relatively easily, and was considered more than adequate. This verifies that the axis of the dividing head matches the central axis of the billet. However, this does not guarantee parallelism.

The dividing head bolts were then loosened. The dial indicator was used to check for variation in the X and Z directions as the X axis was jogged. When these errors were within 0.0002 inch TIR, the bolts were tightened. Concentricity was checked again to make sure nothing had shifted.

An adjustable tailstock was used to support the far end of the billet. The center height of the tailstock was adjusted with shims to the center height of the billet. The tailstock was bolted to the table and used to secure the far end of the billet. Parallelism was checked once again.

Figure 17 shows the aluminum billet on the CNC mill. This picture should help to illustrate the fixturing used.

Figure 17: Machining of differential housing on an EZ-TRAK CNC mill.

After milling, all parts were deburred and with a file and sandpaper. This prevents cuts during assembly and helps protect mating parts from damage. It is very important not to scratch the inside of the sealing sleeve on assembly, as this would cause a leak.

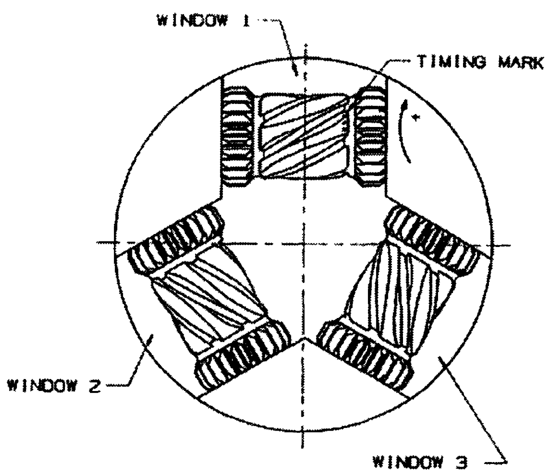
4.1 Notes on Proper Gear Timing

Proper timing of the gears is essential for the longevity of the Torsen gear train. If improperly timed, the journal pins usually cannot be installed due to misalignment. However, it is possible to assemble the unit anyway in certain conditions. MIT's first car suffered from improper timing, which resulted in damaged gears and thrust washers the first time the car was tested on the track. Damage from improper assembly is immediate and catastrophic.

Torsen provides a message about proper timing of the gears on their website. The same note is provided on the print and is reproduced below in Figure 18, but unless the person assembling the differential has previous experience, the note makes little sense.

TIMING

WINDOW	1	2	3
TOP	0	30	120
BOTTOM	0	90	120



ASSEMBLY AND TIMING NOTES:

ROTATE THE HOUSING CLOCKWISE TO PROGRESS FROM WINDOW 1 TO 2 TO 3.

ALWAYS ASSEMBLE THE ELEMENT GEARS SO THAT THE TIMING MARK IS TOWARD THE SAME SIDE OF THE WINDOW. ALSO, THE SIDE GEARS MUST BE KEPT COAXIAL FOR PROPER ASSEMBLY.

THE TIMING MARK IS LOCATED ON THE TOP OF ONE HELICAL TOOTH. TIMING IS GIVEN IN TERMS OF THE DEGREES THAT THE GEAR IS ROTATED OUTWARD TOWARDS YOU. ZERO (0) INDEX REFERS TO THE TIMING MARK BEING STRAIGHT UP.

Figure 18: Timing notes from 012000 print ©Toyoda-Koki Torsen [10].

In the interest of preventing more failures from improper timing, a factory Torsen unit was carefully disassembly and documented. The method below works every time, though there are other variations that should work equally well due to symmetry in the element gears.

1. Place the differential housing on a table with the power input side facing up.
2. Pick a window to serve as window one. Rotate so you can see directly into the window.
3. Coat all gears, journal pins, and washers in gear oil for assembly. Install two element gears such that the timing marks are pointed straight up. The mark should be to the left side of the window.
4. Install side gears and washers. This requires placing the end washers in first and sliding the gears and middle washers in. Note that the washers have a specific order. Torsen provides a list of washers and a diagram on their website. Torsen also suggests that swapping two washers end for end will increase Torque Bias Ratio. The custom aluminum housing was designed such that the entire stack of washers and gears is swapped end for end, so the slotted washer should go towards the brake rotor side.
5. Rotate unit clockwise (viewed from top) to next window. Install element gears with the top gear timing mark rotated 30° outward from vertical and the bottom gear timing mark 90° outward from vertical. Both marks should be installed to the left of window.
6. Repeat step 5 with both top and bottom gears rotated 120° outward from vertical.
7. The differential should roll freely. If not, disassemble and try again.
8. Insert dowel pins to lock journal pins in place.

4.2 Other Notes on Assembly

It is important to make sure that the journal pin retaining pins are all in place when the sealing sleeve is installed. A missing pin would quickly cause failure when the journal pin disengages from the housing.

The seals should be installed with a cylindrical pusher turned to 0.010 inch below the OD of the seal. Seal damage will occur with undersize pushers, especially if a hammer is substitute for the press. A similar device should be machined for the installation of the bearings.

O-rings should be lubricated with either gear oil or a silicone o-ring lubricant. This prevents failure of the o-ring from rotating in the gland on assembly, which can cause damage.

All bolts should be torqued to proper values for consistency. Bolt torques were estimated from recommended torques for grade 6 bolts. A fastener chart published by Engineers' Edge was used as a guideline [11]. Torques were adjusted up or down based on whether the fastener was in tension or shear.

The bolts on the brake and input hats should be torqued and safety wired. 24 ft-lbs was deemed appropriate for the 5/16 inch fine thread AN bolts, as these are in pure tension. Proper safety wiring is an art; techniques can be found in *Tune to Win* [12].

The 3/8 inch bolts attaching the brackets to the engine and frame were torqued to 40 ft-lbs. Prevailing torque locknuts were used because safety wire cannot be used on these fasteners. 3/8 inch bolts on the brake rotor see high shear stresses, so these were torqued to 30 ft-lb.

1/4 inch bolts on the sprocket were torqued to 10 ft-lbs, as these also must see considerable shear forces. Prevailing torque locknuts were also used here.

The splined inner stub shafts should be inserted carefully into the housing so as not to damage the seal. The spline is smaller than the seal diameter, so this is not a difficult step, but it is a particularly important one.

It is important to make sure the snap ring on the inner stub shaft is seated in the chamfer in the Torsen gearing. A mallet may be needed to ensure that the ring locks in place. If the snap ring seats, the stub shaft will not move axially when pushed by hand.

The unit should be filled with 75W 90 GL5 specification oil. A static oil level of 2/3 full is appropriate, according to Torsen. The level can be dropped to below the level of the seals for the endurance race. While this may accelerate wear, not leaking is far more important.

A photograph of the completed differential is shown in Figure 19.

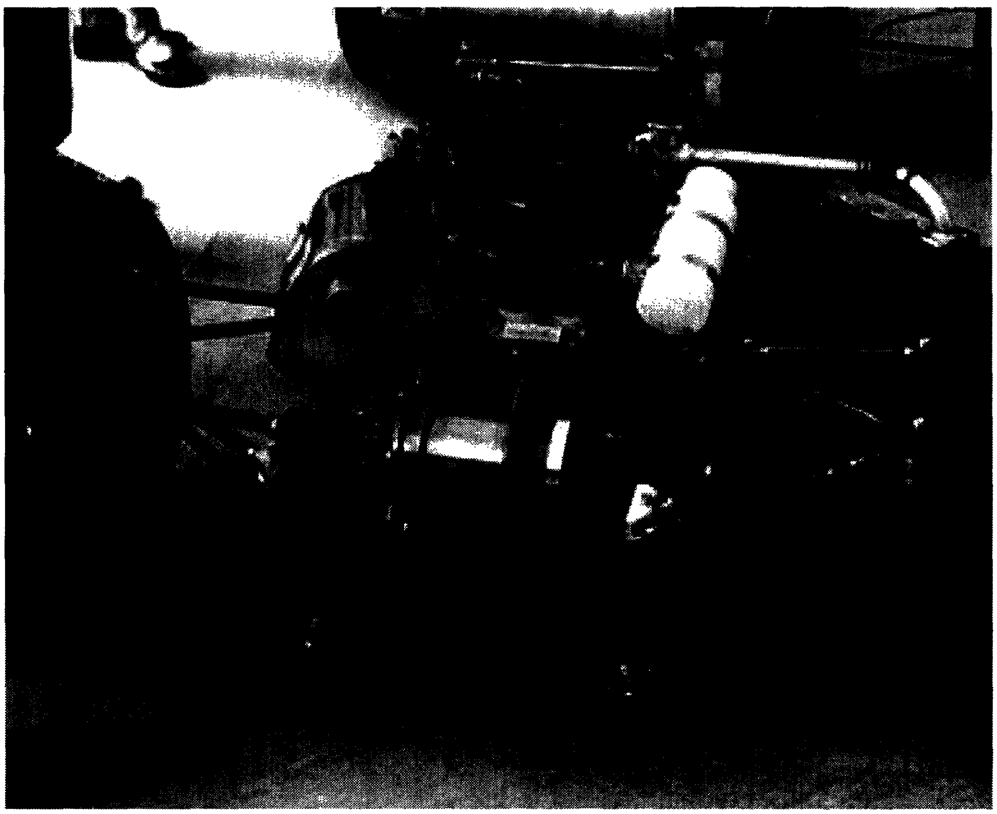


Figure 19: Completed differential at the FSAE Launch.

5.0 Testing

No new mechanical design would be complete without adequate field testing. While at the time of writing, the differential only has a few miles on it, initial shakedown tests were promising.

The differential did leak some oil during shakedown testing. Disassembly showed grooves cut into the seals by the spline on the stub shaft from when the shaft was installed. This happened because the suspension was on the car when the stub shaft was installed, not allowing the shaft to be inserted straight into the seal. It is important that the stub shaft be inserted into the differential with the suspension off. This allows the lip of the seal to load properly on the shaft without getting gouged by the splines. The splined portion of the shaft is smaller than the area where the seal rides, so assembly is not difficult with the suspension removed.

Two recommendations should be followed when preparing the car for the endurance race. One, the oil level in the differential should be lowered to below the halfway point when still. This will allow the oil to sling away from the seal area while running the car. Should high temperatures develop, air should be the only thing that leaks out. Second, Loctite should be applied to the stub shaft where it meets the inner race of the bearing that supports it, as well as to the outer race of the bearing. This seals a potential leak path such that if oil does escape past the main seal, it would have to get through the seals of the bearing as well.

The oil has been changed regularly while the differential is breaking in. Small amounts of tiny wear particles were evident in the used oil. These were mostly magnetic, indicating that the differential case is holding up well. As the gears are covered in a rough steam oxide coating, some wear particles are expected during break in.

6.0 Conclusion

This design of a Formula SAE differential appears to meet the stated goals of reduced weight, better integration with the car, and increased robustness over previous MIT designs. The mechanical design itself was also very detailed, right down to the fasteners used and the intended machining operations. This detail oriented design should serve as an example to some of the other aspects of the car that did not receive this kind attention to detail.

This design required many parts to be designed and manufactured, including mounting brackets, a differential housing, a means of sealing the housing, a method of tensioning the chain, and a means of transmitting power and brake forces to the differential. The differential turned out to be one of the highlights of the 2005 MIT FSAE entry, and it is expected that this will contribute to success in the design part of the competition.

However, there is room for improvement in any project. The FEA was admittedly not very thorough—certain loads were ignored, the load cases were only roughly estimated, and results were not interpreted meticulously. It is hoped that this design will serve as a starting point for further optimization of the differential. An expert with FEA could likely reduce weight without significantly impacting reliability.

In the future, a different type of differential may prove to be lighter, cheaper, or more suitable to formula racing. The Torsen University Special has proven effective by many top teams, but the newer Torsen Type 2 offers easier case manufacturing, easier assembly, and more balanced behavior in right vs. left turns. Other manufacturers also make suitable limited slip differentials, such as Eaton.

Even with significant changes in the differential center section, the direct engine mounting seems to hold continued promise for a lightweight, strong, and modular differential mounting system. Even with this kind of modularity, however, a change in engine would necessitate sweeping changes in the bracket design. A possible change to the Honda 600RR engine in the near future may require substantial rework of the design.

Determining the “best” design solution is always a compromise of strength, functionality, manufacturability, cost, and other factors. While this differential offers much strength in these areas, the old adage still applies—a racecar is never done.

7.0 References (in order of appearance)

[1] "2005 Formula SAE Rules" [Online Document]. Available HTTP: http://www.sae.org/students/fsaerules.pdf

[2] "Torsen FSAE FAQ" [Online Document]. Available HTTP: http://www.torsen.com/fsae/fsaefaq.htm

[3] Chocholek, SE. The Development of a Differential for the Improvement of Traction Control. IMechE 1998, C368/88 p.75-82.

[4] "Torsen 012000 print" [Online Document]. Available HTTP: http://www.torsen.com/files/University%20Special%20012000.pdf
Formerly only available by request.

[5] "Parker O-Ring Handbook" [Online Document]. Available HTTP: http://www.parker.com/o-ring/Literature/00-5700.pdf

[6] "Wilwood Billet Dynalite Single Mouting Dimensions" [Online Document]. Available HTTP: http://www.wilwood.com/Products/001-Calipers/021-DLS/dlsmtg.gif

[7] James, R. (2004) Design of an Aluminum Differential Housing and Driveline Components for High Performance Applications. Unpublished undergraduate thesis, Massachusetts Institute of Technology.

[8] "Chicago Rawhide Handbook of Seals" [Online Document]. Available HTTP: http://www2.chicago-rawhide.com/catalog_pdf.htm

[9] "NTN Radial Ball Bearings" [Online Document]. Available HTTP: http://www.ntnamerica.com/Engineering/PDFs/A1000/A1000_Radial.pdf

[10] "Assembly and Timing Notes" [Online Document]. Available HTTP: http://www.torsen.com/files/012000%20Gear%20Assembly%20&%20Timing.pdf

[11] "Torque Table" [Online Document]. Available HTTP: http://www.engineersedge.com/torque_table_sae_ftlbs.htm

[12] Smith, Carroll. <u>Prepare to Win: the Nuts and Bolts Guide to Professional Race Car Preparation</u>. Washington D.C.: Aero Publishing, 1974

8.0 Suppliers

Admiral Metals
1 Forbes Road
Woburn, MA 01801
(781) 933-8300
www.admiralmetals.com

D.I.D. Racing Chain
check www.didchain.com for US distributors

DXP Enterprises
112 North 12th
LaPorte, TX 77571-3125
(281) 471-6241
www.dxpe.com

JEGS High Performance
101 JEG'S Place
Delaware, OH 43015
(800) 345-4545
www.jegs.com

McMaster Carr
473 Ridge Rd.
Dayton, NJ 08810-0317
(732)329-3200
www.mcmaster.com

Patriot Racing Spockets
PO Box 3132
St. Charles, IL 60174
www.patriotsprockets.com

Rockford Acromatic Products
611 Beacon St.
Loves Park, IL 61111
http://www.rockfordcv.com/

Spectro Oils
993 Federal Rd., Route 7
Brookfield, CT 06804
(800) 243-8645
www.spectro-oils.com

Taylor Race Engineering
Suite 914
2010 Avenue G

Plano, TX 75074
(800) 922-4237
www.taylor-race.com

Toyoda-Koki Automotive Torsen North America
2 Jet View Dr.
Rochester, NY 14624
(585) 464-5000
www.torsen.com

Trident Metals
9501 Baythorne Dr.
Houston, TX 77040
(800) 392-7730
www.trident-metals.com

9.0 Appendices

9.1 Parts List

Item	Description	Supplier
Differential gearing	Torsen University Special	Toyoda-Koki Torsen
FSAE axle kit	custom for FSAE	Rockford Acromatic Products
Hub Splining	service	Rockford Acromatic Products
Driven sprocket	54 tooth, 520 pitch, 7075 aluminum	Patriot Sprockets
Idler Sprocket	14 tooth, steel, 520 pitch, CR250 app.	Patriot Sprockets
Drive sprocket	12 tooth, 520 pitch, forged steel	Sprocket Specialists
Brake Caliper	Billet Dynalite Singe, 3.25" bolt pattern	Wilwood/JEGS
Brake Rotor	10" cast iron, floating, matches front of car	Hoerr Racing Products
Motorcycle chain	D.I.D. 520-ERT	Patriot Sprockets
Master link	for 520-ERT chain	Patriot Sprockets
Bearing, left bracket	NTN 6917LL	DXP Enterprises
Bearing, right bracket	NTN 6012LL	DXP Enterprises
Bearing, stub shaft	NTN 61806LL	DXP Enterprises
Seal, stub shaft	Chicago Rawhide 11606	DXP Enterprises
Oil	75W90 GL5 gear oil	Spectro Oils
Fasteners- Drive & Brake Flange Shear pins	3/8" x 1.25", alloy steel	McMaster Carr
Drain plug	1/8 NPT, brass, hex drive	McMaster Carr
journal pin retainer dowel pins	3/32"x1" alloy steel	McMaster Carr
Fasteners- drive and brake flage helicoils	5/16 NF x ½ inch	McMaster Carr
Fasteners- Diff Bracket to Engine Nuts	AN364-1018A (5/8, half height, elastic)	Aircraft Spruce
Fasteners- Diff Bracket to Engine Washers	AN960-1016LII	Aircraft Spruce
Fasteners- Diff Bracket to Frame	AN6-15A	Aircraft Spruce
Fasteners- Drive Flange Bolts	AN5H-7A	Aircraft Spruce
Fasteners- Drive Flange Washers	AN960-516 (5/16")	Aircraft Spruce
Fasteners- Brake Flange Bolts	AN5H-7A	Aircraft Spruce
Fasteners- Brake Flange Washers	AN960-516 (5/16")	Aircraft Spruce
Fasteners- Driven Spocket Bolts	AN4-7A	Aircraft Spruce
Fasteners- Driven Sprocket Nuts	MS21042-4 (1/4", all metal, half height)	Aircraft Spruce
Fasteners- Driven Sprocket Washers	AN960-416	Aircraft Spruce
Fasteners- Wheel Studs	1/2-20, 35/64 knurl	Autozone

9.2 Stock List

Stock	Purpose	Supplier
5/8 dia. x 1 ft hot rolled 4140, commerical heat treat	Fasteners- Diff Bracket to Engine Stud	McMaster Carr
3/8 dia. x 1 ft hot rolled 4140, commercial heat treat	Fasteners- Diff Bracket to Engine Stud	McMaster Carr
1.5"x18"x.09375 ground prehardened 4142	Thrust washers	McMaster Carr
9"x22"x.75" 6061-T6	Differential brackets	Admiral Metals
5.5" dia. X 7" 2024-T351	Rear hubs	Admiral Metals
4" dia. x 5" sch. 40 6061-T6 pipe	Differential sealing sleeve	Admiral Metals
6" dia. x 2" 2024-T351	Input and Brake flanges	Admiral Metals
4.5" dia x 8 inch 7075-T651	Differential housing	Trident Metals

9.3 Special Tooling List

Tool	Location Used	
Helicoil Prewinder, 5/16" NF	brake and drive flange mounting	McMaster Carr
Helicoil plug taper tap	brake and drive flange mounting	McMaster Carr
Helicoil bottoming tap	brake and drive flange mounting	McMaster Carr
Reamer, 0.402 inch	journal pin reaming	McMaster Carr
3/8 inch spotting drill	general spotting	McMaster Carr
letter X jobber length drill	journal pin roughing	McMaster Carr
7/16" x 2" flute HSS endmill, center cutting	long endmill for diff center section pockets	McMaster Carr

10.0 Acknowledgements

Fred Cote, manager of the Edgerton Student Shop, for donating 200 hours of CNC and manual machine time.

David Dow for his expertise with CNC turning.

Pat McAtamney, for his MasterCAM expertise.

Richard James, for sharing his work on the development of a Torsen based FSAE drivetrain.

Larry Durand, for procuring of bearings, seals, and stock on *really* short notice.

Joe Audette, for running a tight ship and providing a credit card number.

Brad Schiller, for being "The Glue" that holds the FSAE production team together.

Daniel Frey, for being really understanding about the "metal before paperwork" concept.