

Lightweight Torsen Style Limited Slip Differential
and Rear Driveline Package for Formula SAE

by

Tony Scelfo

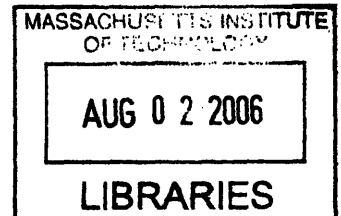
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Table of Contents

ABSTRACT.....	7
1 FSAE Competition.....	9
2 Overall Design Philosophy.....	10
2.1 Functional Requirements.....	10
2.2 Manufacturing Concerns.....	11
2.3 Integration.....	11
2.4 Total Package.....	11
3 Mechanical Design.....	12
3.1 Mounting brackets.....	12
3.1.1 Bracket concept.....	12
3.1.2 Bearing Selection.....	13
3.1.3 Chain Tensioning.....	14
3.1.4 Material Selection.....	14
3.1.5 Manufacturing and Assembly Considerations.....	15
3.1.6 Estimation of Reaction Forces.....	15
3.1.7 FEA of Differential Brackets.....	16
3.2 Differential Center Section.....	18
3.2.1 Differential Center Section Concept.....	18
3.2.2 Material Selection.....	19
3.2.3 Bearing and Seal Selection.....	20
3.2.4 Manufacturing and Assembly Considerations.....	20
3.2.4.1 Picture of Assembly.....	20
3.2.4.2 Journal pin retention.....	21
3.2.4.3 Thrust washers for element gears.....	21
3.2.4.4 Case sealing: o-rings and sealing sleeve.....	22
3.2.5 Estimation of Reaction Forces.....	23
3.2.6 FEA of Differential Center Section.....	23
3.3 CV Joints and Axles.....	24
3.4 Rear Hubs.....	25
3.4.1 Hub Concept.....	25
3.4.2 Material Selection.....	26
3.4.3 Bearing Selection.....	26
3.4.4 Manufacturing and Assembly Considerations.....	26
3.5 Drive Components.....	27
3.5.1 Component Concept.....	27
3.5.1.1 Sprocket.....	27
3.5.1.2 Input Bell.....	27
3.5.1.3 Chain.....	28
3.5.2 Manufacturing and Assembly Considerations.....	28
3.6 Bolts, Nuts and other Hardware.....	29
4 Manufacturing.....	29
4.1 Mounting Brackets.....	29

4.2 Differential Center Section.....	30
4.3 Rear Hubs.....	36
5 Testing.....	38
6 Conclusion.....	39
7 References.....	40
8 Suppliers.....	41
9 Acknowledgments.....	42

Illustration Index

Illustration 1: Differential Total Package.....	11
Illustration 2: Bracket Concept.....	12
Illustration 3: Left Bracket FEA Forces and Restraints.....	16
Illustration 4: Left Bracket FEA Results.....	16
Illustration 5: Right Bracket FEA Forces and Restraints.....	17
Illustration 6: Right Bracket FEA Results.....	17
Illustration 7: Completed Differential Center Section.....	18
Illustration 8: Assembly of Differential Center Section.....	21
Illustration 9: Differential Center Section FEA Forces and Restraints.....	23
Illustration 10: Differential Center Section FEA Results.....	24
Illustration 11: Rear Hub Assembly.....	25
Illustration 12: Differential Bracket EZ-Trak Beginning.....	29
Illustration 13: Differential Bracket EZ-Trak Bearing Pocket.....	30
Illustration 14: Humble Beginning.....	31
Illustration 15: Drilling a Center Hole.....	31
Illustration 16: Checking Alignment in EZ-Track.....	32
Illustration 17: Cutting Element Gear Windows.....	32
Illustration 18: Cleaning Inside Face.....	33
Illustration 19: Installing Helicoils on Input Face.....	33
Illustration 20: Completed Center Section with Torsen Internals Shown Next to Stock Housing.....	34
Illustration 21: Inserting Friction Plate and Side Gears.....	35
Illustration 22: Timing Marks on Element Gears.....	35
Illustration 23: Installing Element Gears and Thrust Washers.....	36
Illustration 24: Setup for Drilling and Reaming Bolt/Pin Pattern.....	37
Illustration 25: Cutting Bolt Head Pattern.....	37

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Submitted to the Department of Mechanical Engineering on
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Degree of Bachelor of Science in Mechanical Engineering

ABSTRACT

This document describes the design and fabrication of a complete driveline package for the MIT Formula SAE race car. The driveline is centered around a custom aluminum housing for a Torsen® T1 gearing. This gearing provides a torque biases limited slip differential which has had good success in the Formula SAE series. The design includes all of the components needed to deliver power from a Honda CBR600 F4i engine to the wheels of a custom race car. This document is intended to document the design process that went into the 2006 MIT FSAE car. Furthermore, this document describes many of the steps required to manufacture each component. FSAE competitors often face the challenge of fabricating the parts that they design. Manufacturing setups are shown and described in order to help designers best prepare for fabrication.

Thesis Supervisor: Daniel Frey

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1 FSAE Competition

Formula SAE® is a competition for university undergraduate and graduate students. Teams that compete must design, fabricate and compete with small, formula style auto cross cars. Teams typically spend between eight and twelve months designing, building and testing the vehicles for competition. Cars are judged on cost, design, marketability and performance. MIT has brought a car to the Formula SAE® competition three times. This year will mark the team's fourth entry into the competition.

The cost event is comprised of three sections. The entries are judged based on the lowest cost, the presentation of a cost report, and a manufacturing presentation. The cost of a car is determined based on the value of raw stock, machine time, labor, and commercial parts purchased at retail value. Designers must focus on minimizing cost throughout the design process.

The design portion of the competition consists both of an evaluation of the car's design and an evaluation of how well the team can justify design decisions. Teams give presentations to a panel of judges made up of engineers from automotive companies and the racing industry. Top teams are selected from the initial round of presentations and those teams advance to a final round of competition where the designs are further scrutinized.

In addition to the cost and design portions of the competition, teams can earn points for their performances in four dynamic events designed to thoroughly test each vehicle. Approximately two thirds of all potential points come from the dynamic events.

The first dynamic event is run on a skid pad. The skid pad tests the car's cornering ability by measuring elapsed time around a figure eight. The time directly corresponds to the maximum lateral acceleration of the car.

The second dynamic event is an acceleration run. The performance of a car in this event is directly related to the power plant, the vehicle weight, suspension geometry and traction. Typically, cars achieve 0-60 times in the mid to upper 3 second range.

The third dynamic event is an autocross. Cones are laid out to form a low speed course with tight corners. Fast times are a function of vehicle agility and driver skill.

Most college age drivers are not used to handling vehicles with the power to weight ratio achieved by FSAE cars and for this reason, extensive testing, tuning and driver training are essential.

The fourth and final dynamic event is an endurance race. The course is approximately 22km long comprised of numerous laps on a track laid out in an enormous parking lot. The course is made up of many tight turns and several long straightaways. Two drivers must complete the event with a driver change half way through the endurance run. During the driver change, the car must be turned off and then restarted by the second driver. Judges scrutinize the car, looking for any fluid leaks evidence of which is grounds for removal from the race. Teams that finish all 22km are awarded points based on their overall time and its ratio of the fastest recorded time. Due to the length of the race and the scrutiny of the judges, only approximately one third of the entrants finish the race. Entrants who do not finish the race earn no points for the event. For the overall competition, the endurance race is worth enough points to make or break a team's overall finish.

2 Overall Design Philosophy

In order to design the differential and driveline package for the 2006 MIT FSAE car, I first needed to establish a set of requirements. The following sections outline the main factors that influenced the design process.

2.1 *Functional Requirements*

- The differential should show no signs of oil leakage during the long endurance event.
- The half shafts should be close to equal lengths to prevent torque steer.
- The differential housing should be lighter than last year's design since no braking loads need to be transmitted through the housing.

2.2 Manufacturing Concerns

- The entire assembly must be manufactured with a minimal set of specialized tooling.
- All tooling must be readily available for general student use. I am a student and I am manufacturing the entire assembly.

2.3 Integration

- The differential should be chain driven off a standard sprocket configuration for a Honda CBR F4i engine.
- The differential should mount only to the engine to achieve a modular design.

2.4 Total Package

The design requirements resulted in the following overall package. I will break out and describe each section of the overall assembly.

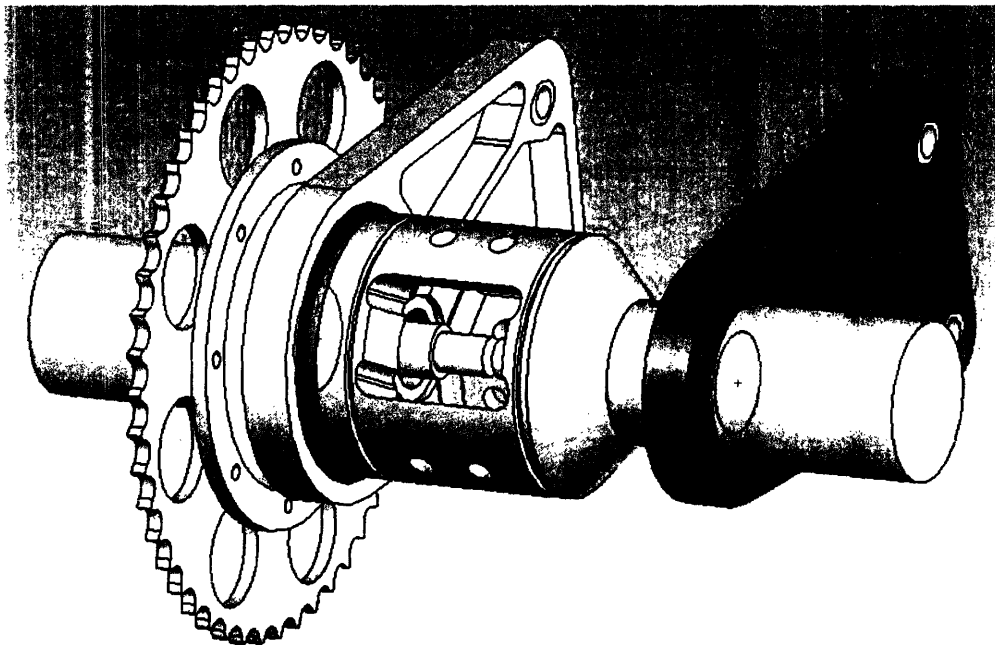


Illustration 1: Differential Total Package

3 Mechanical Design

3.1 Mounting brackets

3.1.1 Bracket concept

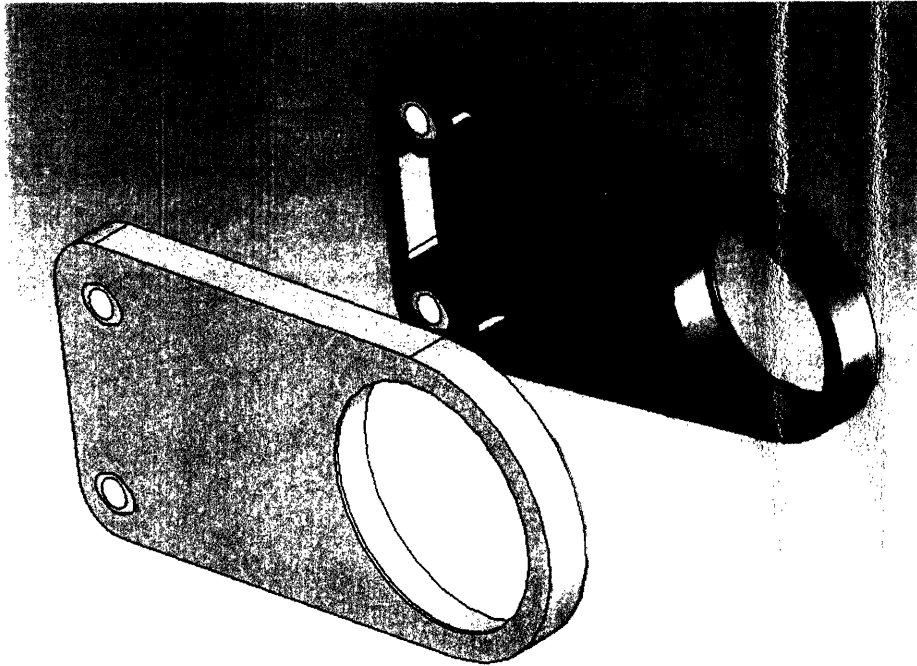


Illustration 2: Bracket Concept

The design of this year's mounting brackets are an iteration of last year's design. Several key factors changed due to changes in the requirements. Last year's differential made use of a central brake rotor for the rear brakes. This requirement put constraints on the design because a rear caliper needed to be integrated into the bracket. Another design requirement that was lost this year is the need to supply an engine mounting point on the differential brackets. The team has gone with a dry sump oil pan this year and as a result, the engine has been lowered relative to the frame. Due to the lower placement of the engine, and a redesign of the rear end of the frame, it would not be feasible to mount the engine to any point on the differential brackets.

Bracket placement was well chosen last year. The CBR F4i engine swing arm mounting points serve as a perfect place to mount parallel brackets to support a

differential housing. The swing arm mount points are approximately 6 inches apart which is wide enough to hold all the differential internals without being excessively wide. On a stock bike, tension forces on the chain are transmitted through a pivot in the swing arm mounting points so it is known that this point can support the loads that we will want to apply through our driveline.

By mounting the differential brackets directly to the engine, we take advantage of the many benefits provided by a direct load path. The entire FSAE car is designed around the engine. This allows us to take advantage of the strength of the engine while minimizing structural components in the frame. The maximum chain tension is approximately 2300 lbs, as we will see in later calculations. This amount of force is fully capable of distorting thin wall tubular steel frame members. If we mount the differential brackets to the frame, a small amount of distortion could cause the differential housing to bind or introduce misalignments that could cause damage. Additionally, tight tolerances can be held when machining aluminum brackets to mount directly off precision mounting points on the engine. It would be very difficult to maintain tight tolerances for bearing alignment in pieces made of welded sheet steel.

Despite the obvious benefits of direct engine mounting, this scheme is uncommon for FSAE teams. There are many factors outside of the load path that contribute to the mounting position. The choice of a different style differential, the chain tensioning method, interferences caused by the sprocket and chain and the manufacturing capabilities of the students and the school all play a role in the final decision. We are confident that we can properly execute a modular design to mount directly off the engine.

3.1.2 Bearing Selection

The choice of mounting points requires us to mount the sprocket outside of the differential brackets. For this reason, we need a large bearing on the drive side of the differential housing to allow torque to be transmitted through the inside of the bearing. The opposite side of the differential housing doesn't see much load at all, so we are able to choose a much smaller bearing. We decided on a 6916 bearing for the drive side and a 6010 bearing for the opposite side. The 6916 bearing is 16mm wide with an OD of

110mm and an ID of 80mm. The 6010 bearing is 16mm wide with an OD of 80mm and an ID of 50mm. Both bearings were chosen to have a rubber seal to prevent road grime from interfering with the performance of the bearings.

3.1.3 Chain Tensioning

Chain tensioning is accomplished this year though a set of football shaped eccentrics in the mounting holes of the brackets. The footballs half a half inch hole drilled off center in the football. Depending on which ways the footballs are installed in the brackets, the sprocket can be moved up to one quarter of an inch. By manufacturing multiple footballs, we can easily dial in perfect chain tension. In our installation, we needed a set of footballs with a hole offset of 0.020". This movement is sufficient for tensioning the chain. Although we will not be able to accommodate single tooth increments in the rear sprocket, we will be able to accommodate three tooth increments in the sprocket. This is because the tangent points on the sprocket do not change when three teeth are added. This solution is an elegant approach to solving the problem of chain tensioning. Without a separate chain idler, we are able to simplify the design and tighten our overall packaging.

3.1.4 Material Selection

The brackets are made of 7075-T6 aluminum. This is one of the strongest alloys available allowing us to create the lightest bracket possible to handle the loads we expect to see in the brackets. CNC machining facilities are readily available for student use, so we are able to create complicated geometries to within high tolerances. While always thinking about the cost report, we must notice that all aluminum is treated the same. This means that more expensive 7075 and 2024 should always be used instead of 6061 to allow us to create smaller and lighter parts for the same price. The thickness of the brackets is determined by the choice of the bearings. As we saw in the previous section, the bearings are both 16mm wide (~0.63") so a reasonable choice will be ¾ inch thick plate. This will give enough room to fully support the bearing while keeping a thin lip

for the outer race to rest against to ensure the bearing is properly aligned when it is pressed in.

3.1.5 Manufacturing and Assembly Considerations

The support brackets are by far the easiest pieces on the car to manufacture. The design lends itself nicely to single setup machining in a CNC mill. All the milling operations are done from one side of the aluminum plate, so the part only needs to be clamped and located in the machine once. After locating the part in the machine, a CNC operation is run to cut out the weight saving pockets, the bearing pockets and the through holes. All of the radii on the part are made to be 0.25 inches so that the entire part can be made with a half inch end mill.

3.1.6 Estimation of Reaction Forces

The suspension group of the team predict a maximum lateral acceleration of 1.8g. The weight of the car including a driver is estimated to be 620 pounds. The tire diameter is known to be 20 inches.

$$T = M \cdot a \cdot R = (620)(1.8)\left(\frac{10}{12}\right) = 930 \text{ ft}\cdot\text{lbs}$$

Chain tension can then be estimated based on the size of the sprocket. We choose to run a 48 tooth 520 pitch sprocket. The pitch diameter is 9.556 inches.

$$F = \frac{T}{R} = \frac{(930)}{\left(\frac{9.556 \cdot 12}{2}\right)} = 2336 \text{ lbs}$$

The chain load must pass through the left side bracket into the engine. The right side bracket sees a much smaller load because there is a significantly longer lever arm between the sprocket and the support bearing. As a result, the right side bracket sees about one sixth of the forces that the left side bracket sees, which is small enough to be insignificant.

3.1.7 FEA of Differential Brackets

For the left side bracket, I fixed the bearing face and applied half of the chain tension to each mounting point in the directions of the chain.

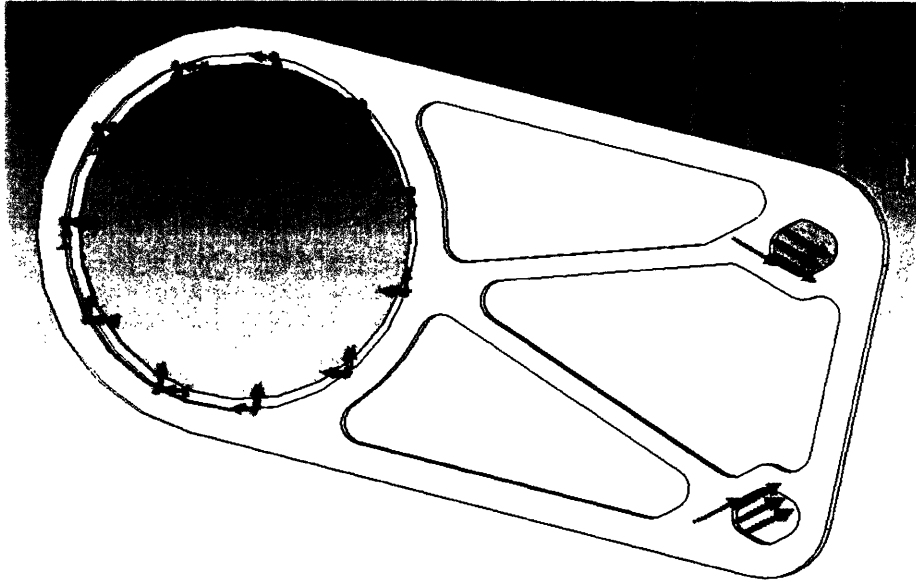


Illustration 3: Left Bracket FEA Forces and Restraints

The values used for the forces were 1200 lbs each.

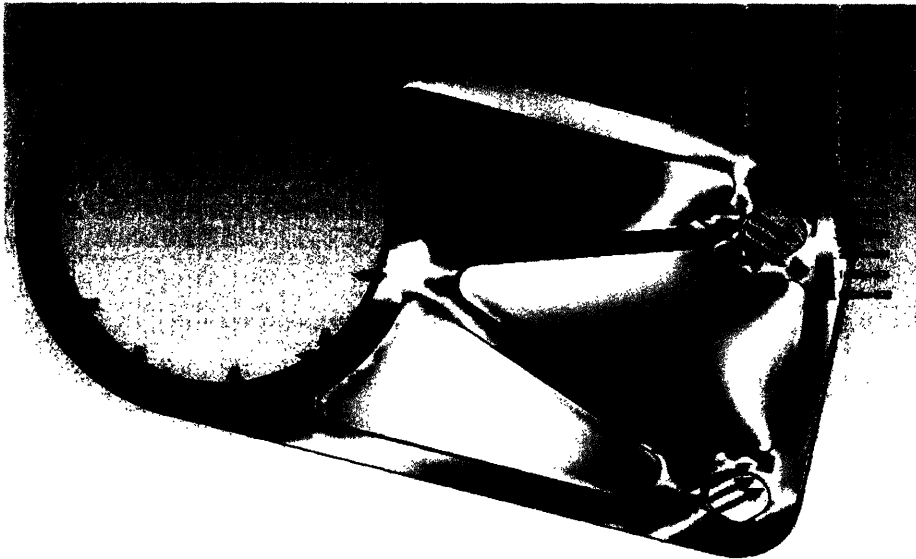


Illustration 4: Left Bracket FEA Results

The minimum factor of safety for the left bracket was found to be 8.1 in 7075-T6 aluminum.

For the right side bracket, I fixed the bearing face and applied one quarter of the load that the left side sees.

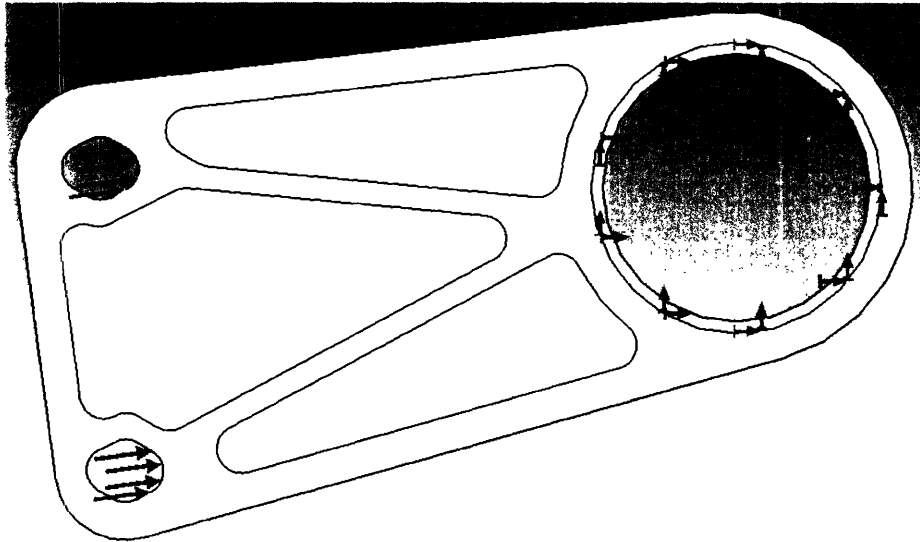


Illustration 5: Right Bracket FEA Forces and Restraints

The values used for the forces were 300 lbs. This is a conservative overestimate on the force that the bracket will see.

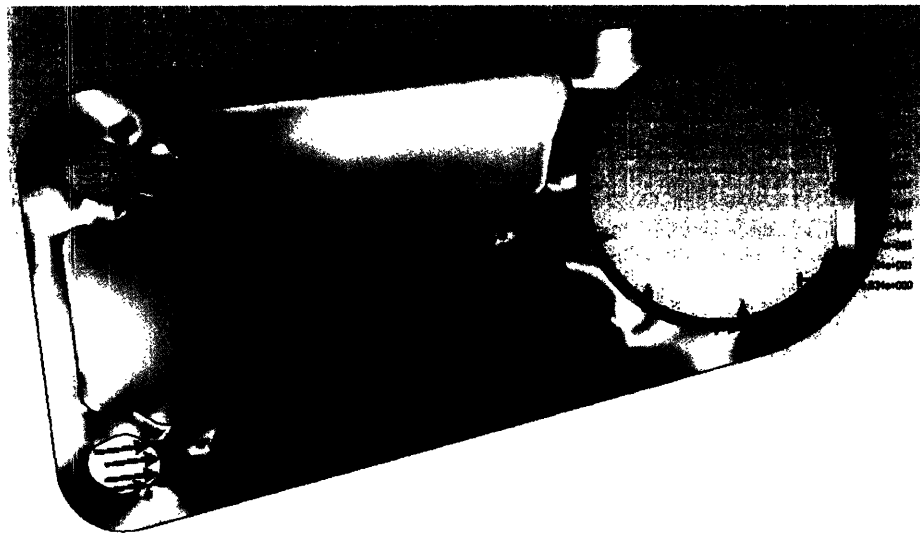


Illustration 6: Right Bracket FEA Results

The minimum factor of safety for the right bracket was found to be 3.9 in 6061 aluminum. Testing this bracket with 6061 aluminum allows us to remake the bracket in the weaker aluminum if we are forced to do so due to time constraints and material availability.

3.2 *Differential Center Section*

3.2.1 Differential Center Section Concept

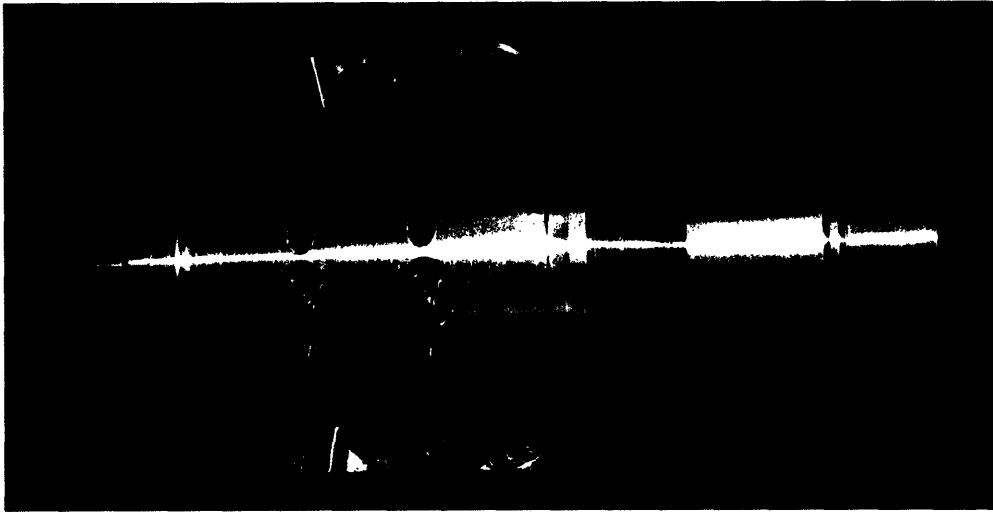


Illustration 7: Completed Differential Center Section

Much like the design of the differential brackets, the design of the center section is an iteration of last year's design. Last year's center section held a brake rotor in addition to the Torsen gearing. By eliminating the need for a rear rotor in this year's design, we are able to simplify the design of the center section and remove some weight. The standard Torsen housing is made of cast iron and its design makes integration into our car challenging. Last year was the first year that the MIT team created a custom aluminum housing for the Torsen internals. The custom aluminum housing served two main purposes. The first purpose of the custom housing is weight savings. The second is improved integration with the rest of the vehicle.

The stock housing is both heavy and over built. Torsen quotes the strength of the housing at 5200 Nm, or 3800 ft-lbs. As we saw in the equations for a force estimation on the support brackets, the stock housing is nearly twice as strong as it needs to be. In a race application, parts are often subjected to shock loadings and it is often desirable to have high safety factors, but we believe that we can produce our own housing that is better suited for our application.

As far as integration goes, the stock Torsen housing is difficult to interface with because there is not much room for drilling and tapping. In order to interface with the stock housing, end caps need to be fabricated to interface with either a large spline on the housing or the housing itself. We would need to make the end caps and weld them onto the housing. The MIT team's capability to deliver a precise and durable interface is questionable. Additionally, the width of the housing is such that it can't be simply mounted to the engine with brackets made from plate aluminum. In order to integrate the stock housing, we would need to come up with a more complicated and heavier mounting scheme. Additionally, FSAE teams have had a lot of success creating aluminum housings for the Torsen internals based on drawings that Torsen provides.

In order to design a custom aluminum housing, I first needed to understand how the Torsen differential works. The Torsen differential uses two “side gears” that operate on the axis of the drive axles. The side gears interface with six “element gears” that operate in three pairs around the side gears. The rotating case transmits torque to the half shafts through the element gears and then the side gears. The helical grooves cut into the gears allow the case to transmit torque to the drive shafts proportional to a torque biasing between the two side gears. As the two side gears turn at different rates, they move axially and compress thrust washers between the side gears and the housing itself. The friction between thrust washers determines the torque biasing. The MIT team has not decided to experiment with different biasing and we use the standard washer setup that is provided in the Torsen University Special.

3.2.2 Material Selection

The center section of the differential is made of 7075-T6 aluminum. As mentioned during the selection of material for the brackets, this is a very strong aluminum alloy and it is not vulnerable to cyclical loadings like some harder aluminum alloys. A massive bar of 7075 aluminum is fairly expensive. It is therefore very important to choose the right size to ensure that there is enough material to turn down to get a nice finish without wasting too much material. This year's design deviated from last year's design because the center section was made from a 4 inch diameter piece of stock.

The differential from last year had a larger diameter. The 4 inch diameter is more common than 4.25 diameter, so this is why a 4 inch diameter stock was chosen. This change in diameter means that the sealing sleeve must have a smaller inner diameter. Standard schedule 40 aluminum pipe, which was used last year, does not have a thick enough wall to properly seal a center section that is just under 4 inches in diameter. We were fortunate to find that McMaster Carr carries schedule 80 pipe. Schedule 80 pipe has a larger wall thickness and therefore it is capable of sealing a center section that is just under 4 inches in diameter.

3.2.3 Bearing and Seal Selection

The bearings and seal design for this year's differential changed somewhat from what was used last year. Last year, seals were used to isolate the Torsen internals as close to the gears as possible. This resulted in the half shafts passing through a bearing in the differential and then the seals. This year, we choose to use wider needle bearings to provide more support for the half shafts. These wider needle bearings have an open design that lends themselves to a lubricated environment. For this reason, the order of the seals and the bearings were reversed to put the seals on the outside of the differential center section and the bearings on the inside. The needle bearings have a width of 20mm which provides a large surface to support the 30mm diameter drive shafts. An additional benefit is that the OD of the needle bearings is the same as the OD of the seals used for the 30mm drive shafts. This allows the bearing pocket and the seal pocket to be made in one boring operation on the lathe.

3.2.4 Manufacturing and Assembly Considerations

3.2.4.1 *Picture of Assembly*

This following picture is provided to help clarify points that are made in the next sections.

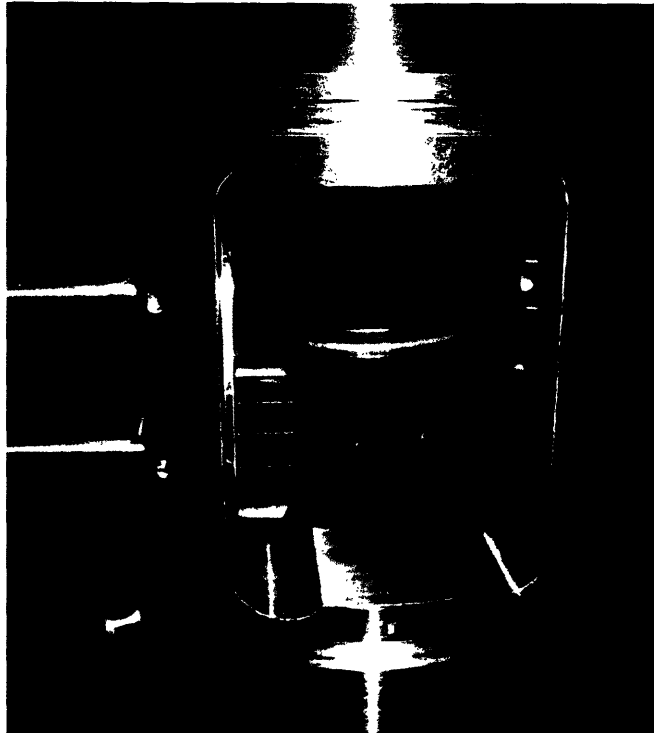


Illustration 8: Assembly of Differential Center Section

3.2.4.2 Journal pin retention

In the original Torsen case, the journal pins that support the element gears are held in place with spirol pins. In the aluminum case, it isn't ideal to use a spirol pin because of long term wear on the housing. Instead, we simply use a 3/32" piece of welding rod installed in a hole that is perpendicular to the axis of the journal pin. Once the sleeve is in place, the welding rod is constrained and in turn it constrains the journal pins.

3.2.4.3 Thrust washers for element gears

The element gears in the Torsen see a tremendous amount of torque. Due to the helical grooves in the side gears, the element gears are pressed up against the face at each end of the journal pin under load. Depending on which way the side gear is being turned, the element gear will apply a large load to one of the two faces. In the aluminum housing, the repetitive abuse of a hardened steel gear riding on an aluminum face is less

than desirable. In order to prevent wear on the aluminum housing, thrust washers were made to fit on each side of the element gear between the gear and the aluminum face. The gears were made of hardened steel and were cut out of a jig on the EZ-Trak. Once they were cut, they were surface ground to 0.0625" at MIT Central Machining. In the above photograph, one thrust washer can be seen on the left hand side of the window.

3.2.4.4 Case sealing: o-rings and sealing sleeve

The differential center section must hold oil to prevent the Torsen gears from ripping into each other and seizing. The FSAE competition is designed in such a way as to severely penalize teams that have any kind of fluid leak. For this reason, it is essential that the differential sleeve is properly sealed. The sleeve seals are Viton o-rings that are set into grooves in the center section on each side of the three element windows, as can be seen in the above picture. Viton is an excellent choice for a race application because it can withstand temperatures up to 400°F and it does not react with gear oil. The groove in the center section is cut based on a sizing chart for a 1/8" o-ring with a 4" diameter. The groove is about 20% wider than the o-ring and about 80% as deep as the o-ring. This allows for proper o-ring crush when the sleeve is installed. With a clearance of 0.005" between the differential center section and the sealing sleeve, the sleeve requires a large amount of force to slip over the o-rings but the resulting seal is solid. Even though a large amount of force is required to press the sleeve over the o-rings, it is possible that the sleeve can move and either damage the o-rings or cause a leak. In order to prevent rotation between the sleeve and the center section, a 3/8" pin is pressed into the center section perpendicular to the surface. A corresponding groove is cut into the sleeve providing both a positive stop and a means to prevent rotation. The sleeve is drilled and tapped to accommodate a 1/8" NPT drain plug. This allows us to both fill and empty the housing because we are able to rotate the whole housing. The drain plug must be located over a window, but this is easy to ensure because of the notch in the sleeve that mates with a pin in the housing.

3.2.5 Estimation of Reaction Forces

Calculations of the internal forces for a Torsen differential were done by Rich James in his undergraduate thesis two years ago.⁴ The internal face that sees the maximum load from the side gears under acceleration can see 24,000 lbs. Under maximum load, the element gears can put 8000 lbs of force on the thrust washers that they ride against.

3.2.6 FEA of Differential Center Section

The loads were applied in CosmosWorks by fixing the face that is in contact with the input bell and modeling each estimated force as normal forces.

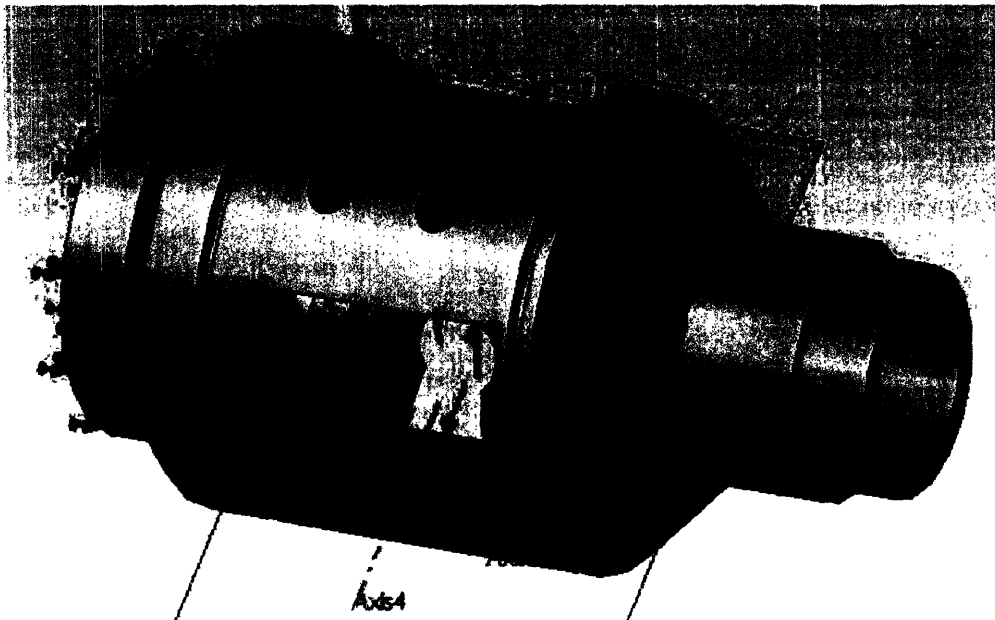


Illustration 9: Differential Center Section FEA Forces and Restraints

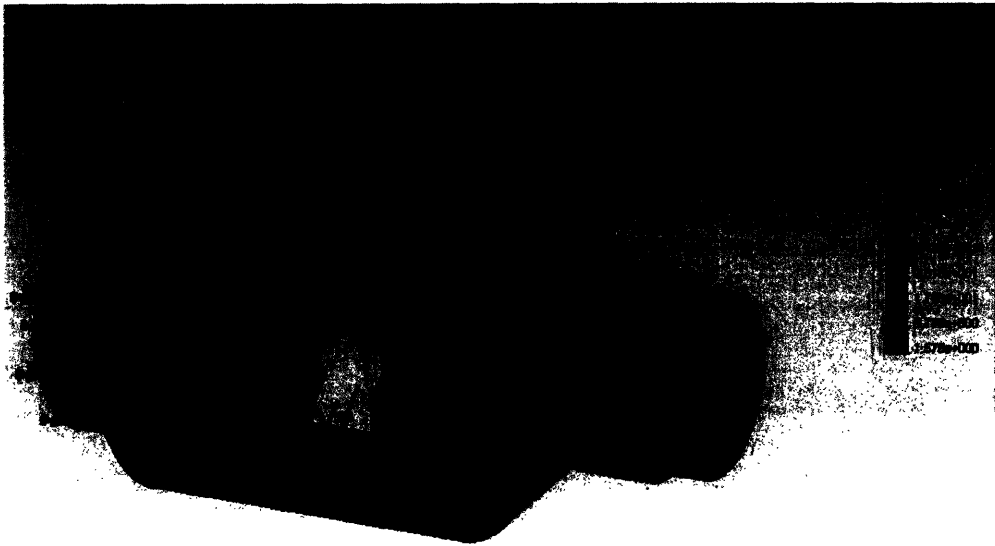


Illustration 10: Differential Center Section FEA Results

The minimum factor of safety was found to be 1.6 at the face where the maximum thrust load is applied. Although we would likely want to see a higher factor of safety, this number is adequate for our application.

3.3 CV Joints and Axles

The CV Joints and Axles are purchased as a kit from Rockford Performance. Last year, the MIT team began using the Rockford axle kits because of their low cost and overall complete package. The splines on the inner sides of the drive shafts are designed to mate with the splines on the inside of the Torsen University Special. The splines on the outside of the shaft are design to mate with a Polaris hub. Rockford also offers a broaching service, so we had the hubs broached to match with the outer splines provided in the Rockford axle kits. This proves to be an exceptionally easy and cost effective way to get power from our differential to our rear hubs.

3.4 Rear Hubs

3.4.1 Hub Concept

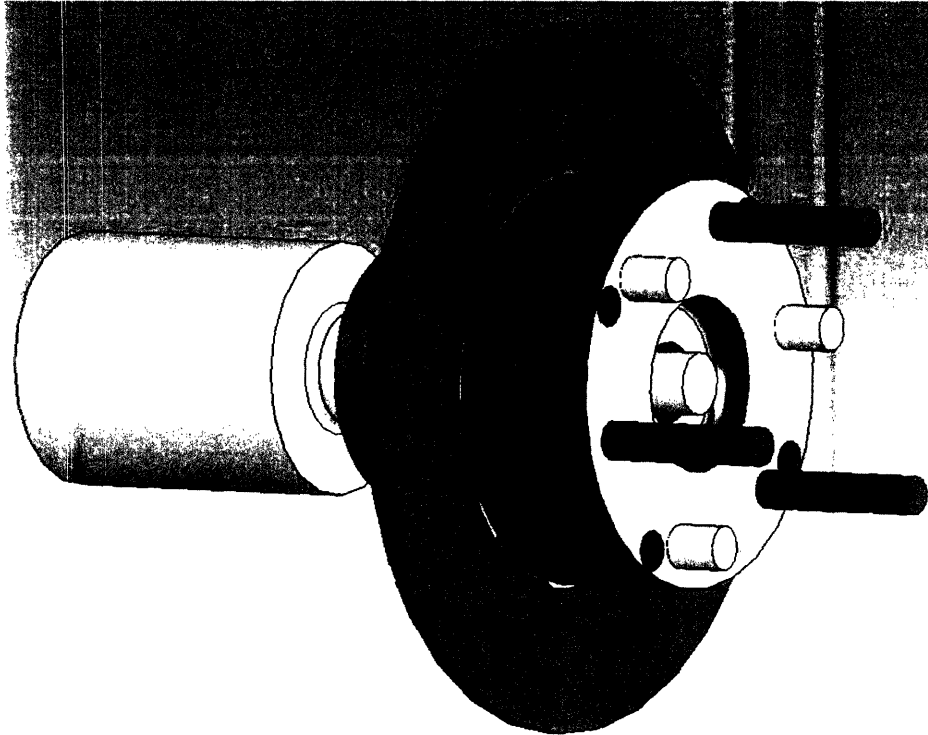


Illustration 11: Rear Hub Assembly

This year's FSAE car has a radically different hub design than last year's car. In an attempt to save weight, we have decided to create our own custom wheel centers to mate with ultra light weight aluminum wheels. Because we designed the wheel centers ourselves, we have decided to go with a custom bolt pattern which allows us to use light weight AN hardware in a three bolt pattern. In order to bring power through the drive shafts and into the wheels, we needed to come up with a design for rear hubs that could properly mate with our custom bolt pattern.

In addition to mating with the rear wheels, the rear hubs needed to hold brake rotors for the outboard brakes in this year's FSAE car. As can be seen in the design for the hubs and brakes, the hubs use a three bolt pattern and the rear brakes use a four bolt pattern. This was due to the fact that the brakes were designed and manufactured early in the build schedule before the custom bolt pattern was fully worked out between the

members of the suspension design team. Aside from looking somewhat strange, clearance issues were not a problem and we were able to accommodate both the four bolt and three bolt pattern in the rear hubs. The wheel mounting bolts are retained with a small custom ring that is cut on the water jet. This custom ring is held in place by the brake rotor and in turn prevents the wheel bolts from backing out of the hubs.

3.4.2 Material Selection

The rear hubs were made of 7075-T6 aluminum. The decision was based on the same factors that governed the design for the differential center section. By using a strong aluminum, we are able to remove large amounts of material while maintaining reasonable safety factors. As with the differential center section, the cyclical loading properties of 7075-T6 aluminum are also extremely relevant for the rear hubs.

3.4.3 Bearing Selection

The bearings for the rear hubs were chosen by the suspension group and installed in the vehicle's uprights. The bearings are standard Volkswagen wheel bearings and they are the same bearings that were used last year.

3.4.4 Manufacturing and Assembly Considerations

The most important task faced while manufacturing the rear hubs was maintaining a concentric relationship between the bearing face and the inner bore. Because we were having the splines cut by a third party, it was important to adhere to the tight tolerances requested by Rockford Performance. For the inner bore, they requested that we provide a hole between 0.812" and 0.816". We were able to provide them with two hubs with 0.814" center bores within a tolerance of a few ten thousandths of an inch. This ensured that all machine work could be done before the hubs were broached without fear of having a hub with a misaligned spline.

3.5 Drive Components

3.5.1 Component Concept

3.5.1.1 Sprocket

The sprocket was chosen to be a 48 tooth 520 pitch sprocket. When combined with a 12 tooth front sprocket, we achieve a much more aggressive gear ratio than the 12-54 that we ran last year. This is good for two reasons. The first is that our car is much lighter than last year. With a lighter car, it will be easier to run a taller gear. The second reason is that drivers noticed that we were not spending much time in first gear and they were often questioning whether or not it would be appropriate to shift during an auto cross event. It is much better to make use of the entire available power band than it is to run out of RPMs and force a shift just prior to reaching a corner. In the ideal situation, the gear ratio would be chosen based on the exact track layout and conditions, but a slightly taller ratio than last year is a good start. The sprocket was purchased from Sprocket Specialists and is made of 7075-T6 aluminum. We came up with our own custom weight savings pattern to cut into the sprocket. As this is both a design and a performance competition, it is important to not overlook small details. Highly visible part, such as the sprocket, should be made to look as good as possible.

3.5.1.2 Input Bell

The input bell connects the sprocket to the center differential section. The center section has three pins and three bolt holes. The input bell slides over the three pins and is bolted into place. In order to use fine thread AN bolts, the differential center section is tapped with Helicoils. The Helicoils provide a hard steel surface for the fine threads of the AN hardware. The input bell is required due to the location of the drive sprocket. There is no way to get power to the differential center section other than through an input bell that transmits torque through the large bearing in the drive side differential bracket. Although the input bell sees all of the torque that the engine provides, it does not need to be a massive piece to withstand the forces. The input bell is made of 2024 aluminum.

3.5.1.3 Chain

The chain was chosen from Sprocket Specialists. It is a 520 pitch o-ring-less chain that is designed for race applications. The 520 pitch chain is significantly heavier than a 480 pitch chain and it is also a bit wider. The extra width translates into extra weight in a chain guard since more thick steel is needed to cover a 520 chain than is needed to cover a 480 chain. I called Sprocket Specialists and they assured me that there is no way to get a 480 pitch front sprocket for a CBR F4i engine. We know that this is not true because some teams run a 480 chain pitch with a F4i engine. We have read posts on the FSAE forum that say teams have taken large 520 pitch gears that are splined for a F4i engine and had them custom cut with 480 pitch teeth. This is something that the MIT team can consider in the future, but at this time we have not been able to consider a 480 pitch chain.

3.5.2 Manufacturing and Assembly Considerations

The largest challenge when making the drive components was to ensure that the input bell can be removed from the differential center section without allowing slop that can cause extra wear on the parts. In order to create a tight fit, the drive pin holes in the input bell were reamed to 0.376" in order to fit nicely on the differential center section's 0.375" drive pins.

Assembly of the drive components is very straight forward because of the module design of the differential. The mounting brackets slide onto the engine and are bolted on. The sprocket is bolted to the input bell which is then bolted to the differential center section. When we assembled the differential, we mounted the input bell to the differential center section before mounting the differential brackets to the engine. This was done so that the bolts on the input bell could be safety wired away from the car in an open area. This point is minor, however, since it would be possible to safety wire the bolts after the differential is installed on the vehicle.

3.6 Bolts, Nuts and other Hardware

The sprocket uses 1/4" AN bolts and jet nuts to attach it to the input bell. The input bell uses three 3/8" AN bolts to attach it to the differential center section. The three AN bolts have cross drilled heads so that they can be safety wired together. The differential brackets bolt into place with 1/2" AN bolts at both the top mount and the lower mount. The differential mounting bolts also have cross drilled heads so that they can be safety wired together as well.

4 Manufacturing

This section contains pictures of the manufacturing process and some tips that I learned along the way.

4.1 Mounting Brackets

Manufacturing the brackets was straight forward. Stock is held in the EZ-Trak and the differential pattern is cut from one side.



Illustration 12: Differential Bracket EZ-Trak Beginning



Illustration 13: Differential Bracket EZ-Trak Bearing Pocket

4.2 Differential Center Section

The differential center section was extremely difficult to manufacture and it took many hours to complete. Anyone considering this project in the future should realize how much of a time commitment it is before starting. Due to the high tolerances required in this piece, concentric alignment must be verified every time the part is placed into a new setup.

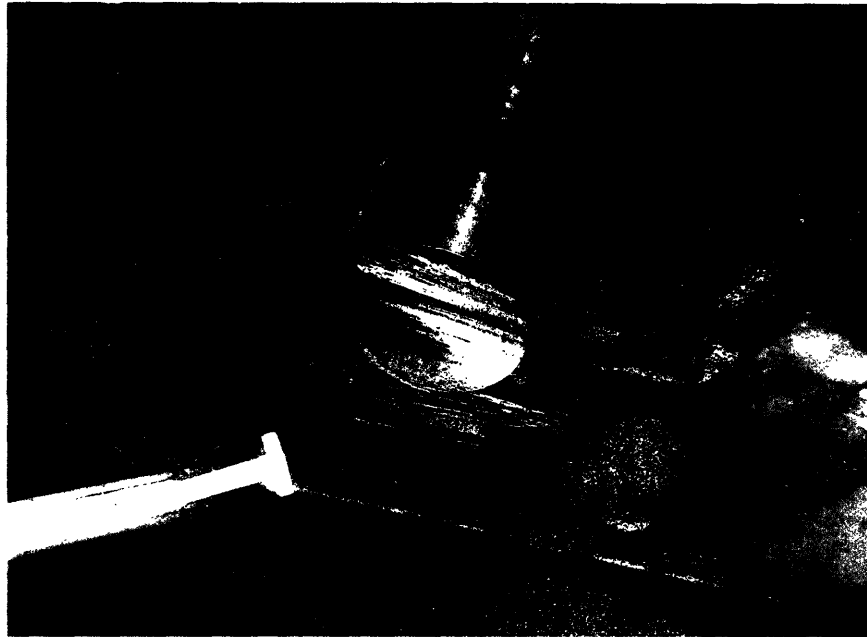


Illustration 14: Humble Beginning

I had no idea what I was getting myself into when I first cut the stock to length.

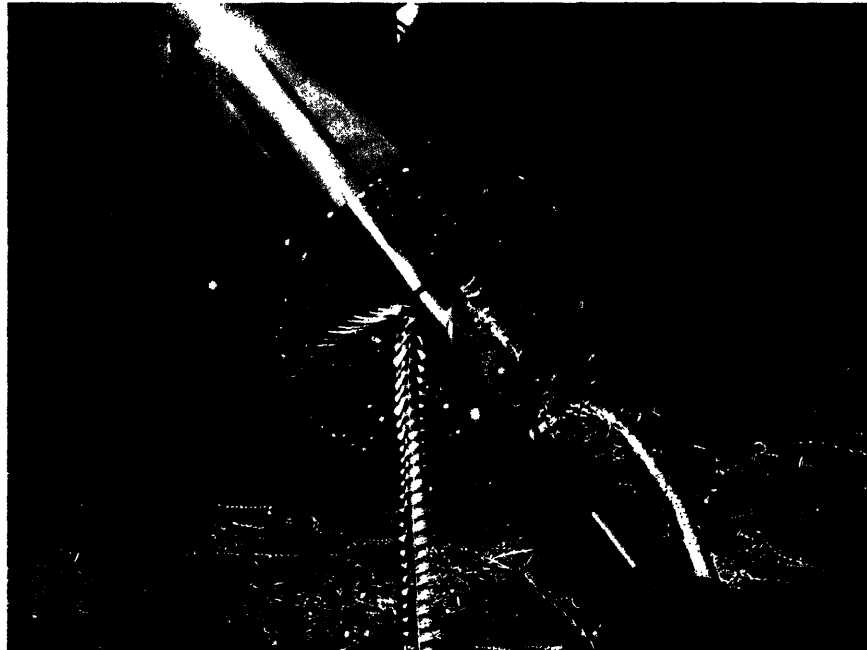


Illustration 15: Drilling a Center Hole

After cutting the stock to length, the outside faces must be cleaned up and made perpendicular to each other. A center hole is drilled in the same setup.



Illustration 16: Checking Alignment in EZ-Track

Alignment must be checked at every step. This setup shows a rotary vice on the EZ-Track. A dial indicator is used to ensure that the setup is in line with the axis of the mill.

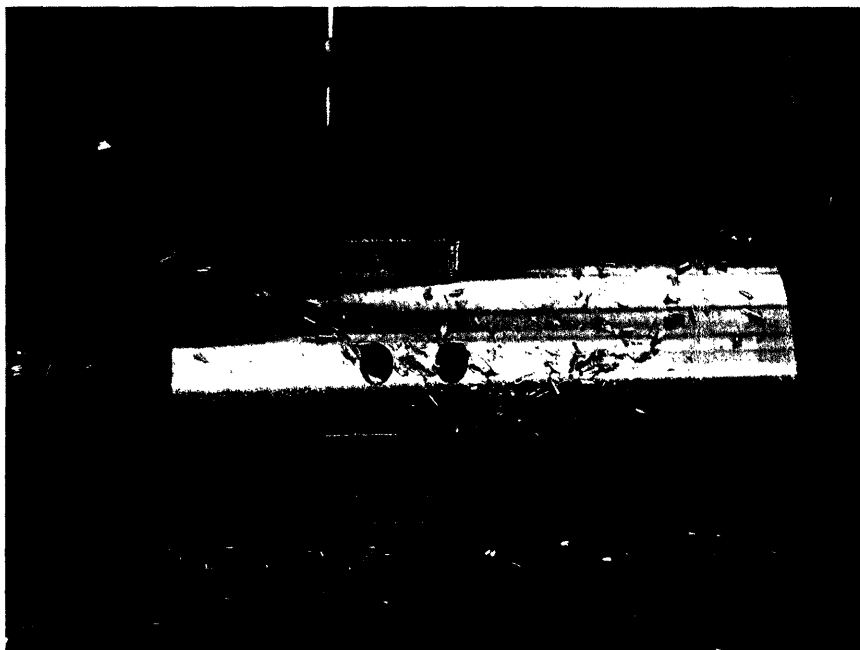


Illustration 17: Cutting Element Gear Windows

A window is cut, the rotary vice is rotated and then the next window is cut.



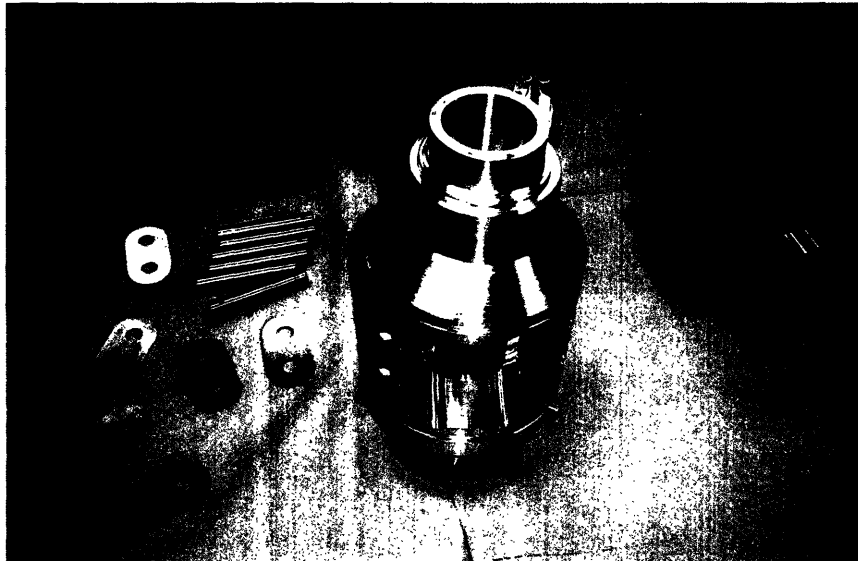
Illustration 18: Cleaning Inside Face

After boring the center hole, the inside faces are cleaned up with a boring bar



Illustration 19: Installing Helicoils on Input Face

The input face is then drilled and tapped in the mill. This operation is particularly interesting because to ensure concentric alignment with the sprocket and input bell, the bolts holes are drilled and tapped first. Once the bolt holes are tapped, the input bell is bolted to the center section. The alignment is checked with a dial indicator and then holes are drilled for the drive pins. This ensures that the drive pin holes are properly lined up in both the center section and the input bell. Unfortunately, I don't have a picture of that step.



*Illustration 20: Completed Center Section with Torsen Internals
Shown Next to Stock Housing*

The completed differential center section is shown next to all of the parts that must fit inside of it. Next to the aluminum center section is the stock cast iron center section.



Illustration 21: Inserting Friction Plate and Side Gears

Assembly of the center section begins with the friction washers and the side gears.



Illustration 22: Timing Marks on Element Gears

The timing mark for the element gears is shown here. Installation of the element gears is tricky because the gears must be installed with the proper orientation to prevent binding.

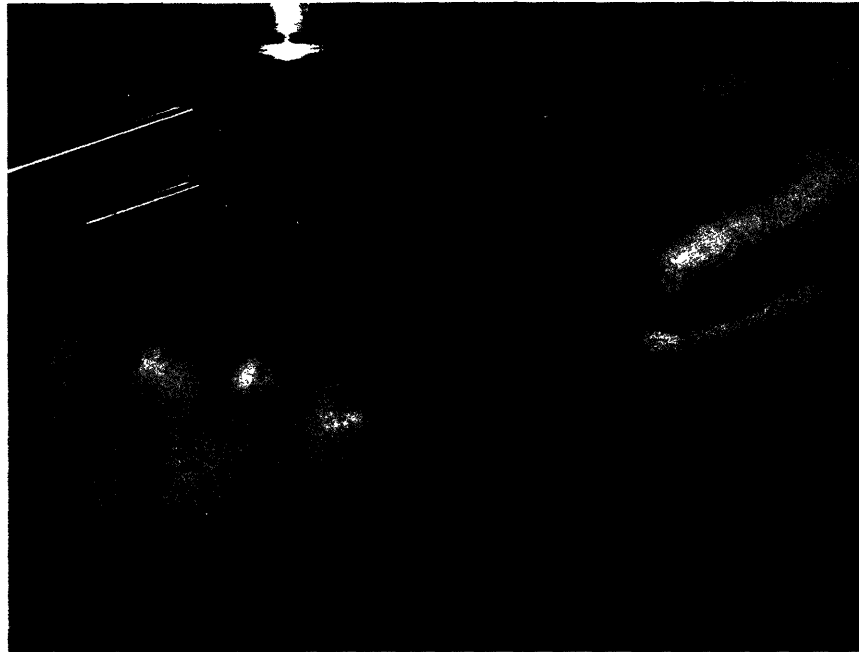


Illustration 23: Installing Element Gears and Thrust Washers

The element gears and thrust washers are being installed. A timing chart is provided by the manufacturer and it is used to ensure that the element gears are rotated to the proper orientation. If they are installed incorrectly, binding will quickly occur and can be checked by installing half shafts and turning them by hand. If the half shafts rotate freely when the element gears are lubricated, the element gears are installed correctly.

4.3 Rear Hubs

Unfortunately, I didn't take pictures of the lathe processes required to make the rear hubs. While making the rear hubs, it is essential that the inner bore and the face that mates with the bearing are perfectly concentric. It is then important that the face that mates with the wheel is perpendicular to the inner bore. After the rear hub is machined on the lathe, we sent it out to be broached so that it would have a spline to match the drive shafts.

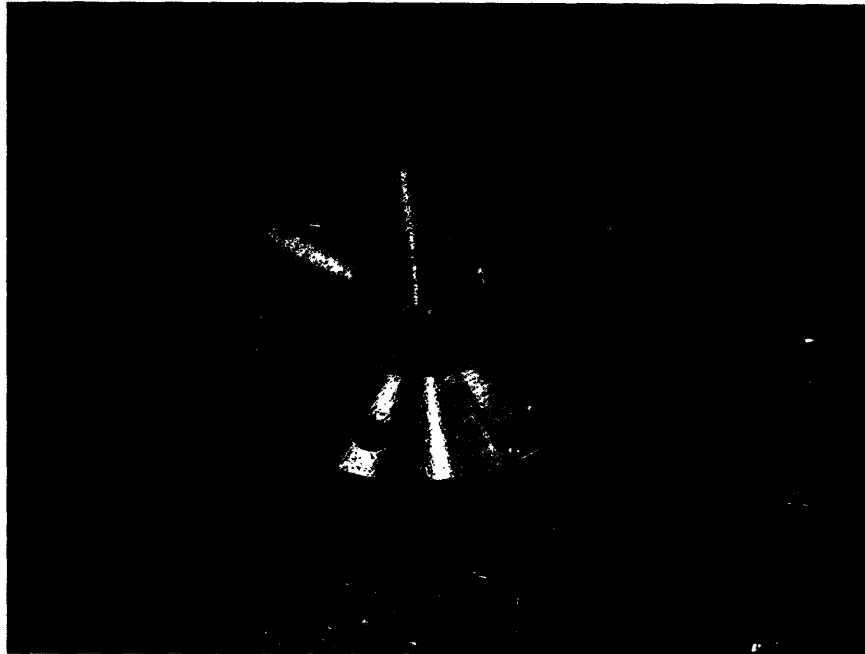


Illustration 24: Setup for Drilling and Reaming Bolt/Pin Pattern

This picture shows the rear hub clamped into the EZ-Trak mill. The drive pin configuration requires tight tolerances. For this reason, the drive pins should be drilled and reamed from the side of the hub that mates with the wheels.

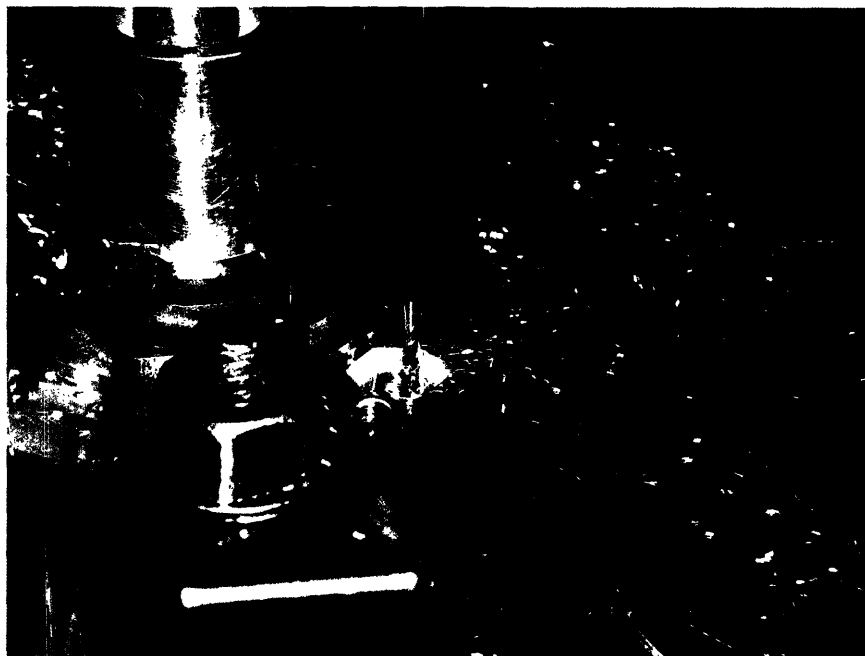


Illustration 25: Cutting Bolt Head Pattern

5 Testing

As of yet, the 2006 MIT FSAE car has been tested for one weekend. The differential has held up well under several hours of hard testing. There was a very minor leak through the right hand seal which was caused by damage to the seal during half shaft installation. If the splines of the half shafts slide along the seals during installation, they cut small grooves that allow oil to force its way past the seals. The seals are easily replaced, but it is important to pay close attention to clearance during installation. After replacing the seal, the oil leak has stopped completely.

The oil in the differential center section showed trace amounts of metal speckles. This was expected during initial break in due to the contact between the element gears and the thrust washers. There were very few metal speckles, which shows that the differential is holding together well and is expected to last well through the endurance event.

I was extremely please with the performance of the entire driveline package. Everything performed as expected. The only slight leakage problem is easily corrected and we have put in enough testing to make me confident that everything will hold up well for the actual FSAE competition. The next step is to get our driver's comfortable enough with the car so that they can turn in competitive times in the dynamic events.

6 Conclusion

The design presented in this document for a Formula SAE differential and drive train package meets the intended goals of weight savings and modular integration with the car. The design benefited greatly from previous work done on the MIT team. Many design decisions were extensions of previous ideas. Refinements were made to save weight and simplify the design and packaging.

Fabrication required the differential team, me, to manufacture a lot of parts including mounting brackets, a differential housing, a sealing sleeve and rear hubs. This experience has taught me many lessons about manufacturing and design. I have learned more while completing this project than I ever thought I could in a year of design and manufacturing. Although I was able to complete all of the driveline components, I strongly recommend that this project is divided up among several team members in the future.

Despite the improvements made from the 2005 design to the 2006 design, there is, of course, still room for improvement. The rear frame of the 2006 car is drastically different than the rear frame of the 2005 car due to the use of a dry sump to lower the engine. It should be possible to further integrate the differential mounting plates into a bulkhead for the rear end of the car. This would eliminate the need for some or all of the rear frame members which maintaining a high torsional rigidity.

During every phase of design and manufacture, potential improvements are realized. Some improvements can be applied immediately while some need to sit on the back burner until next year's design is finalized. This year's differential maintains a solid balance of strength, manufacturability, cost and reliability. It is exciting to think about where the MIT FSAE team will be in a few years. Looking at the leaps and bounds that the team makes from one year to the next, the iterative design process promises to deliver some incredible innovations in the next few years.

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8 Suppliers

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