CORNELL BAJA RACING

FALL 2025

Senior Design Report

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Projects:
Front Differential Casing
Rear Differential Driveline Retrofit

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Contents

1 Senior Design Executive Summary

- **What are the desired function(s) of your design?**
  The casing shall effectively constrain the front differential gears within the front gearbox and allow for efficient mounting of the differential carrier gear. The casing shall not yield or deform under loading conditions. The casing should allow for weight reduction over the OEM hardened steel casing, reducing rotational inertia and making the front driveline lighter and more efficient.

- **What constraints related to the main function(s) must your design satisfy?**
  The casing must survive sustained thrust loads from the differential gears while driving. The casing must not interfere with the front gears. The casing must cost a reasonable amount ($<$ \$2000) to produce and assemble.

- **What are the performance objectives of your design?**
  The new casing should reduce front gearbox total weight by at least 15% by being made out of 7075-T6 aluminum. The new casing should not increase the difficulty of front gearbox assembly, and should be manufacturable on a normal CNC mill without special considerations.

- **What alternative design concepts were considered?**
  Keeping the OEM casing would be the simplest alternative design. However, the front reduction is currently extremely heavy and one of our vehicle-level design goals this year is to reduce rotational inertia and mass of driveline components, so keeping the OEM casing conflicts with those goals. Another alternative design could be a custom casing made of steel. This would not solve the problem of weight reduction, however a successful attempt at a custom steel casing would still allow us more design freedom in the coming years for the design of the front gearbox, rather than packaging around an OEM casing. Another alternative design concept would be to eliminate the front differential entirely and return to a sprag carrier front gearbox. However, this conflicts with our vehicle-level goal of 4WD steering effort and maneuverability, as the sprag front gears cause undriveable levels of understeer and high steering effort.

- **What analyses were used to select among these alternative design concepts?**
  A decision matrix was developed to help select the best option, taking into account total mass, manufacturability, and vehicle dynamics:

Option	Mass (1-3)	Manufact. (1-3)	Vehicle Dyn. (1-5)	Total
OEM	1	3	4	8
Aluminum	2	2	5	9
Steel	1	1	4	6
No Diff	3	3	2	8

- **What industry or society standards were used to help evaluate your design?**
  We used AGMA gear standards to help back-calculate thrust loads on the casing from the differential gears. We used MMPDS standards for material choice.

- **Which concepts or skills learned in your coursework were applied to the design? Projects are expected to make substantial use of MAE and related ENGRD classes. Please provide a list with each entry providing the department and number of the course, plus a brief description of the particular concept or skill used.**
  MAE 3270 (Mechanics of Materials) was used for material selection as well as the analysis of the casing, and to design against specific failure modes. MAE 2020 (Statics) helped with the structural analysis of the casing. MAE 2030 (Dynamics) was used to help create a dynamic model of the front reduction to assist in back-calculation of thrust loads along with CS 1112 (Matlab). MAE 2250 (Mechanical Synthesis) helped us better understand tolerancing standards and how things will get machined.

- **Evaluate your design, relative to its function(s) and constraints. How well did your design meet each of the performance objectives? How well does your design compare to other, existing solutions to the problem?**
  The design fully constrains the gears in the same way as the original OEM casing, and provides similar stiffness and hardness values to the original casing, while also cutting nearly two pounds of rotational mass off of the driveline. The elimination of 8 additional bolted connections by directly integrating the ring carrier into the differential casing further increases the installability of the system and reduces points of failure. On paper, compared to the original casing, it will work better for the purposes of Baja SAE. Since this report is written prior to the assembly and testing of the casing, it is impossible to make an assessment on the real-world design efficacy.

- **What impact do you see your design, if implemented, having upon public health, safety, and human welfare, as well as upon current global, cultural, social, environmental, and economic concerns?**
  The design will likely not have an impact on any of the above stated items. It is a differential casing made specifically for a Baja SAE car. However, conducting this in-depth design will allow us to learn a lot, and then apply the developed skills to make a positive impact on society in our future jobs.

- **What format did your design take? For example, is it a complete set of CAD drawings, a working prototype, a full finished product, a system configuration, a process map, or something**

else?
The design takes the form of a complete set of CAD drawings and FEA analysis. Since it has been determined as both feasible and cost-effective, a set of manufacturing drawings have also been created, which will allow assembly and testing of the design on the TG22 Baja car during the Spring semester and potential use at competition.

- **Describe each student's role in the design project if it was a group project.**
  Joey Nadol - 3D scan of current OEM casing, CAD of custom differential casing, derivation of gear loads and model building, FEA of custom differential casing, creation of manufacturing drawings.
  Zoe Szymanski - Implementation plan for rear differential, CAD of rear differential, FEA of rear differential components, modified rear box packaging, and improved casing design.

2 Front Differential Casing

2.1 Abstract

The Baja SAE series has continued to get more and more competitive as teams adjust to new regulations; namely the encouragement (essential requirement) of 4WD in all cars, the subsequent challenging track design by competition organizers in order to test the robustness of 4WD systems, and the implementation of a new required engine (Kohler Command Pro CH440) which puts much more peak torque down than the prior Briggs & Stratton Engine. Last year, Cornell Baja implemented an aftermarket Porsche 996 limited-slip differential into the front reduction (Savage, Swedek and Hunt 2024) to minimize torque steer effects in 4WD and allow for optimal corner exit in slippery conditions. The differential succeeded in heightening yaw moment during 4WD driving, allowing the car to have similar steer characteristics as RWD while maximizing grip during cornering scenarios. However, due to its original use in a Porsche roadster, the differential is quite overbuilt for Baja SAE usage. The differential itself weighs in at 7.12 lbs, 4.93 lbs of which is just the casing, and is made from billet 8620 case-hardened steel. The rotational weight increase in the front gearbox last year led to decreased efficiency due to frictional effects, as well as increased shock-loading on all upstream components (the repeated breakage of the TG21 propshaft during testing could be attributed to this effect). By engineering a custom casing for the front differential while leaving the differential internals untouched, we can ensure driving performance stays consistent while taking weight off the car in-line with full vehicle goals for the TG22 season. The custom differential casing will also "delete" a part in the front gearbox by eliminating the ring gear carrier and directly integrating it into one side of the casing, to increase installability and decrease needless complexity in-line with TG22 full vehicle goals. Overall, over 1.8 lbs of rotational mass will be removed

from the front reduction while retaining robust factors of safety and preserving the vehicle dynamics that made TG21 an Iron Team-winning car.

2.2 Design Requirements

The design must:

- Fully constrain the differential internals within the housing
- Achieve a factor of safety of 1.25 against yield
- Not fatigue over an operation period of 40 hours
- Be backwards-compatible to the OEM casing, for risk management

The design should:

- Minimize mass wherever possible, especially far from the rotation axis
- Decrease mechanical complexity and points of failure
- Strive for simple machinability and not require excess components

2.3 Initial Research

2.3.1 Understanding Casing Loads

The driving mechanism behind the loading of the casing is not easy to understand, but is imperative for understanding how a Torsen B-type or Quaife differential (such as our Wavetrac differential) works. The key operating principle, as in any front differential, is to allow the front wheels to spin at different speeds while torque is transferred through the system. The specific way in which torque is transferred through a parallel-axis helical-gear differential is relatively hard to visualize; essentially, the gears themselves are not mounted on bearings or shafts but instead are only constrained by other gears and the casing itself. They sit within pockets bored out to be tight clearance fits, such that when the casing is powered by the ring gear, the casing applies a force tangential to the outer gears (which I will call planet gears). These gears then transfer force to an interlocked central gear on each side (which I will call the sun gear), which powers the plunging cups, and in turn, the halfshafts. When one side hits its tractive limit, any additional torque transferred through to that side of the casing will force the planet gears against the side of the casing, generating resistance and causing torque to flow to the tractive side at the Torque Biasing Ratio, or TBR. The Wavetrac mechanism, meant to prevent a zero-torque scenario on an axle, also adds its own force to the above list: when one wheel is stuck and the other is completely free-spinning, the difference in wheelspeed engages a center wave-shaped cam. This cam causes the washer spring stack to deform, adding a ton of axial load to each sun gear, allowing some torque to be sent to both axles in an effort to power the car out of the zero-torque scenario.

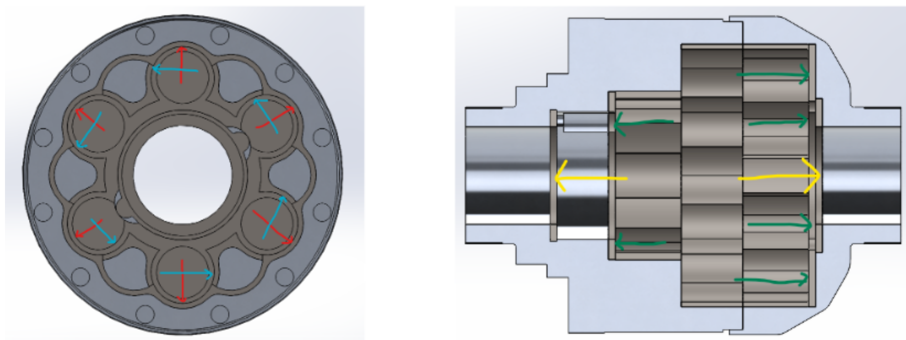

Figure 1: Sketched image of forces on the internal casing. The **blue** arrows represent the tangential contact of the planet gears on the casing as the casing spins, the **red** arrows represent the radial force reacted by the casing due to the gear mesh between the sun gear and planet gears, the **green** arrows represent the axial/thrust force of the planet gears reacted by the casing due to the gear mesh between the sun gear and planet gears, and the yellow arrows represent the axial/thrust force of the sun gear reacted by the casing due to the static preload of the bolts compressing the Belleville washer stackup, and additionally the dynamic load of the Wavetrac compressing the washer stackup further during a zero-torque scenario.

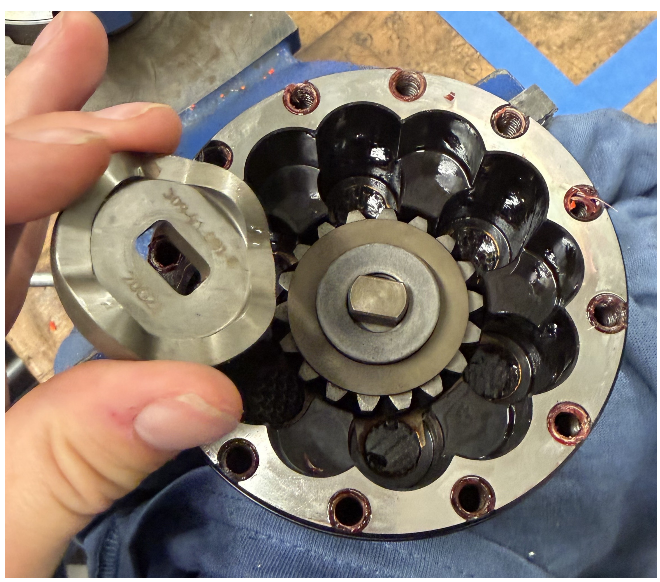

Figure 2: The Wavetrac center cam and Belleville spring stackup, as seen on the disassembled OEM differential.

2.3.2 Determining Front Gearbox Operating Temperatures

In order to determine what material to use for the custom differential casing, it is necessary to have an understanding of operating temperatures. To determine this, we did around an hour of driving in 4WD with the legacy vehicle, TG21. After driving, we then examined the temperature on the outside of the casing using a thermal camera. We saw that while the rear gearbox had an elevated value around 82 degrees Celsius, the front gearbox was still approximately the same temperature as the surrounding air. This is likely due to the overrunning sprag architecture used in TG21's 4WD system; since the front is driven at a 15% higher torque than the rear, and therefore runs 15% slower, once the rears have traction and the whole car runs at the speed of the rear wheels, the front sprag simply slips and no torque is transferred through the propshaft to the front gearbox. This means for straightline operation, and in fact for the majority of the time the car is driving, the front gearbox is not under power and sees relatively little rotation resistance, and thus very little heat from friction. Thus we can conclude that temperature is not a design consideration for the differential casing.

Figure 3: An example image from the thermal camera.

2.4 Design

2.4.1 Selected Solutions

The first important design choice to make is material selection for the casing. Since the gears (AISI 9310 steel) interact directly with the casing, the inside edge of the casing cannot be aluminum or it would wear out extremely quick. In order to distribute the load effectively, the inner edge of the casing will be "sleeved" with a thin hardened steel piece, which will be through-hardened AISI 4340 steel. The piece is required to be manufactured using EDM (electrical discharge machining), since on a traditional machine there would be a ton of chatter and likely the piece would warp in its workholding. On the bottom of the casing, where the planet gears will thrust axially into the casing, there will be laser-cut AR500 steel sheet pieces (the hardest steel SendCutSend offers). This provides a sort of inner "core" which will react out the contact forces with the gears without wearing down. Outside of the inner core, the casing will be made out of two pieces of 7075-T651 aluminum alloy, renowned for its high strength-to-weight ratio.

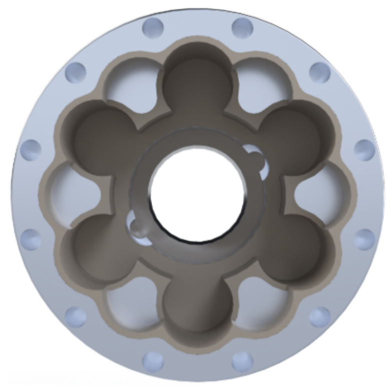

Figure 4: Inner sleeving of the aluminum differential casing.

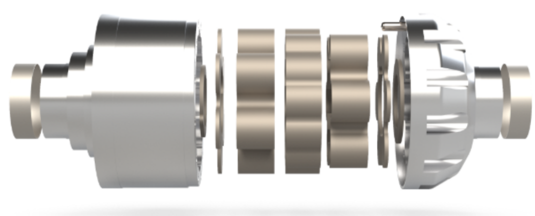

Figure 5: Exploded view of the differential casing internals, showing how it makes a complete "core" around the gears. This version of the casing did not have an integrated ring gear carrier, which was a later addition.

The next design consideration to address was other wear surfaces, specifically, the bearing seats and the threads that would bolt the two halves together. Since these were previously steel but now the body would become machined aluminum, we needed to find a way to retain the original hardness on these contact surfaces. For the bearing seats, the selected solution was to use SKF SpeediSleeves, originally meant for repair of worn surfaces. The sleeve is a thin stainless steel layer that presses over a specific shaft diameter in order for the bearing to sit on a higher-flatness and harder surface, which ensures longer wear life of the assembly. Anodizing the aluminum was also considered but rejected due to two main concerns, the first being the relative uncertainty of the anodized layer's thickness, which could vary by .0008" (1.6 thousandths diametrically), meaning we could not control it to a good enough extent to get a good bearing fit every time without having to potentially go in and sand down the layer afterwards. The other reason was the effect of anodizing on fatigue life. As this part will be highly cyclically loaded, avoiding a knockdown on the fatigue life (especially in a material like 7075 alloy which already has relatively poor fatigue performance) is necessary.

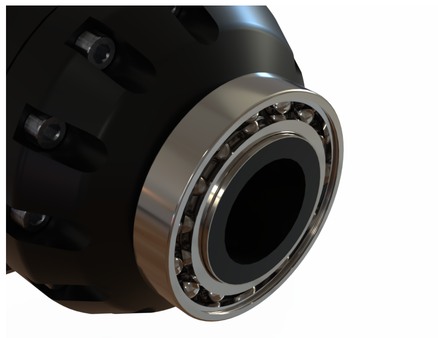

Figure 6: Closeup of the bearing sitting on the SpeediSleeved surface.

For the issue of the pretensioned bolts having to thread into an aluminum body, we selected helicoils to protect the aluminum threads. These stainless steel inserts thread into a slightly larger aluminum thread, and then the bolt threads into the helicoils. This helps protect the aluminum thread against wear.

2.4.2 Left-Hand Side Casing Half

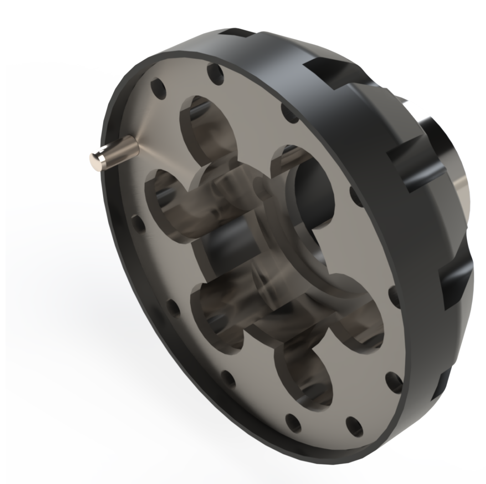

Figure 7: Left-hand side of the casing. Note the locating pin and contact plate stackup.

The differential casing halves require a locating feature in order to align the planet gear bores, as for one side of the planet gears, the bore is split across the two halves. As such, a precision-ground pin is pushed into the left-hand side to interact with a close-fit hole on the other side. Combined with the sizing of the outer lip to also be a close-fit with the right-hand side, this ensures concentricity between the two sides. All EDM pieces (”sleeves”) are designed to be very light interference fits with the casing, while the laser-cut pieces are slip-fits to account for the looser tolerance on laser parts as well as for easy installation.

2.4.3 Right-Hand Side Casing Half

Figure 8: Right-hand side of the casing. Note the integrated ring gear carrier, and multiple sleeves stacked on top of each other.

In order to eliminate as many points of failure as possible as well as decrease overall mass and installation time, the decision was made to directly integrate the ring gear carrier into the differential casing. This removed 8 bolted connections from the car, and saved around a tenth of a pound just on its own. The other notable feature of this casing half is the need for multiple stacked sleeves, since this side of the casing houses one set of the planet gears, as well as half of the other side. The outside surface has an area with decreased diameter to take up the concentricity feature from the other side to ensure both halves locate together effectively.

2.4.4 Bolted Connection

Figure 9: Close-up of the bolted connection between the halves. In order to keep compact packaging while maximizing internal wall thickness, low-profile internal-hex-drive M5 bolts were chosen, with a counterbore clearance of about 10 thou radially.

2.5 Calculations and Derivations

2.5.1 Gear Load Derivation

In order to analyze the differential casing, the first step is to understand how the gear loads are applied, and based off known information about the driveline, calculate the load magnitude. To this end, a spreadsheet model was developed.

The easiest place to start for this model is at the output - based off of Wavetrac's specs, we know that the TBR for the differential is approximately 2.6, meaning that at maximum, the torque on the tractive side can be 2.6 times the torque on the non-tractive side. We know the input ring gear torque from our ratio calculations, putting the nominal low-end (maximum) value at 574 lb-ft. From this, we know:
(1) $T_{high} = T_{nom} \cdot \frac{TBR}{1+TBR} = 574 \cdot \frac{2.6}{3.6} = 414.5$ ft-lb.
We can also easily derive:
(2) $T_{low} = T_{nom} \cdot \frac{1}{1+TBR} = 574 \cdot \frac{1}{3.6} = 159.4$ ft-lb.
From this, we can get an order-of-magnitude estimate for the loads that the

planet gears exert on the casing. We measure the pitch radius of the sun gear: $r_{p,sun} = 0.8$ in. Other important measurements are the helix angle of the gearset ($\beta = 35°$), as well as the pressure angle ($\alpha = 20°$). From this, we can calculate the planet gear loads on the high side:
(3) $F_{t,planet} = \frac{T_{high}}{r_{p,sun} \cdot n_{gears}} = \frac{414.5 \cdot 12}{0.8 \cdot 6} = 1036.4$ lbf. From the tangential force, we can now calculate the axial and radial loads which must be absorbed by the casing:
(4) $F_{r,planet} = F_{t,planet} \cdot \tan\alpha = 1036.4 \cdot 0.364 = 377.2$ lbf.
(5) $F_{a,planet} = \frac{F_{t,planet} \cdot \tan\beta}{\cos\alpha} = \frac{1036.4 \cdot 0.700}{0.9396} = 772.3$ lbf.
This is for each individual planet gear, which are assumed to equally share the load. The low-torque-side casing loads can be calculated in the same way.

2.5.2 Belleville Washer Load Derivation

Another key component of the loads on the casing is the Belleville washer stackup between the two center gears. This stackup gets compressed during the installation process as the two halves get bolted together, leading to a passive pre-loading of the sun gears on the casing. This preload increases drastically when the Wavetrac cam actuates, which compresses the washer stackup even further. For the derivation of loads on the casing, we assume "worst-case loading" where the Wavetrac is fully actuated, leading to maximum preload.
For a given Belleville washer at deflection s, the spring force can be modeled as:
(6) $F_{spring} = \frac{4E \cdot t^4}{(1-\mu^2) \cdot \alpha \cdot D^2} \cdot \frac{s}{t} \cdot [(\frac{h}{t} - \frac{s}{t}) \cdot (\frac{h}{t} - \frac{s}{2t}) + 1]$
Where E is the modulus of elasticity, t is the thickness of the spring material, D is the outside diameter of the washer, μ is the Poisson's ratio, h is the unloaded height of the washer, and α is a calculation coefficient given as:
(7) $\alpha = \frac{1}{\pi} \cdot \frac{(\frac{\delta-1}{\delta})^2}{\frac{\delta+1}{\delta-1} - \frac{2}{\ln\delta}}$
Where δ is the D/d ratio of the washer (outer to inner diameter).
For the Belleville stackup inside the differential, it is nominally assembled fully in-series. For Belleville washers, this means the top of the washer is assembled to the top of the next, and then the bottom of that washer is assembled to the bottom of the next. They are never stacked cone-in-cone. For equal springs in series, to calculate the spring force, they can be treated as one singular spring with the deformation divided by the total number of springs in series. There are six total washers, and the total stackup compression was measured to be 0.235 in. Therefore:
$s = 0.0392$ in, $F_{spring} = 1007.9$ lbf.
For purposes of being able to replicate results, the other variables used were:

E	2.90×10^7 psi
t	0.04 in
μ	0.3
D	0.9825 in
d	0.485 in
h	0.073 in

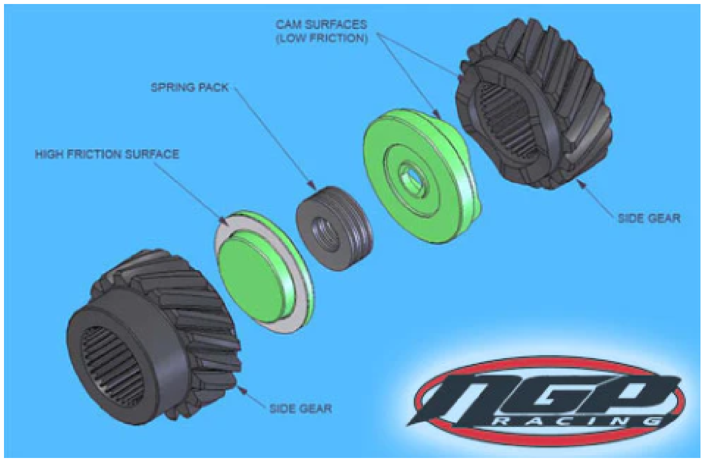

Figure 10: The cam stackup, including the Belleville washers. When the Wavetrac cam actuates, it pushes the washers against each other more, inducing preload into the sun gears.

2.5.3 Casing Tangential Loading

The tangential reaction loading of the casing from having to carry the planet gears is quite simple. First, the contact radius of the gears on the casing was found: $r_{t,casing} = 1.094$ in. Then, the total reaction force was calculated:
(8) $F_{t,casing} = \frac{T_{nom}}{r_{t,casing}} = \frac{574 \cdot 12}{1.094} = 6296.2$ lbf.
Due to the floating nature of the gears in the casing, it is extremely likely that all twelve are contacting the side of the casing during torque transfer. Therefore, the force on each gear is 524.7 lbf, oriented tangential to the direction of rotation.

2.6 Finite Element Analysis

2.6.1 Initial Small-Scope Modeling

The initial modeling of the system in FEA included all internal sleeving, the two casing halves, and the ring gear mounted to the differential casing. This was an intuitive setup and allowed for easy minimal constrainment:

- 2 cylindrical supports on each bearing seat (radially fixed, otherwise free) - elimination of 4 DOF
- 2 compression only supports, one on each side of the casing - elimination of 1 DOF
- frictionless support at ring gear tooth - elimination of last DOF

From here, all that had to be done was to apply the internal loads previously derived. For this FEA, we considered combined-loading of each worst-case scenario derived above.

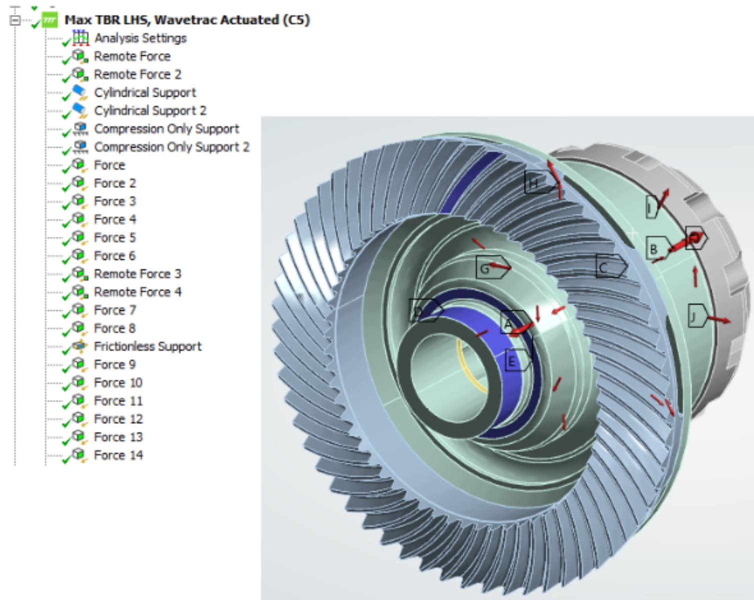

Figure 11: The initial setup for FEA.

This led to the following results:

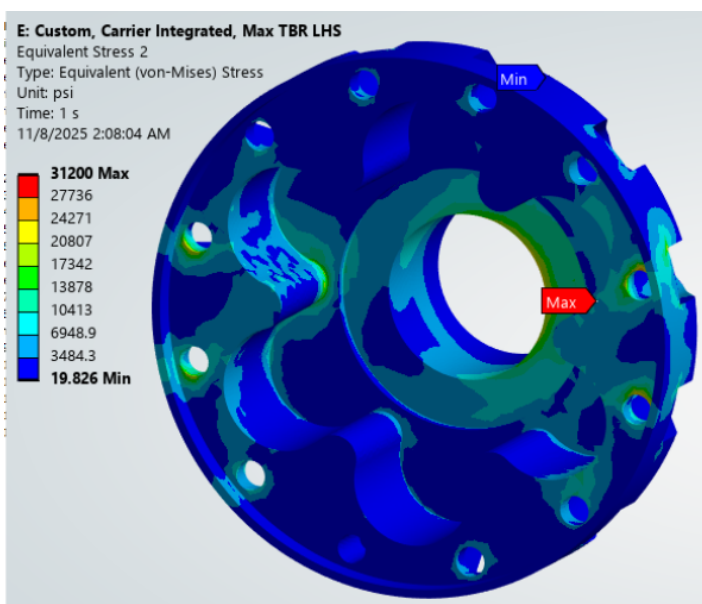

Figure 12: Von-Mises Equivalent Stress results for the left-hand side.

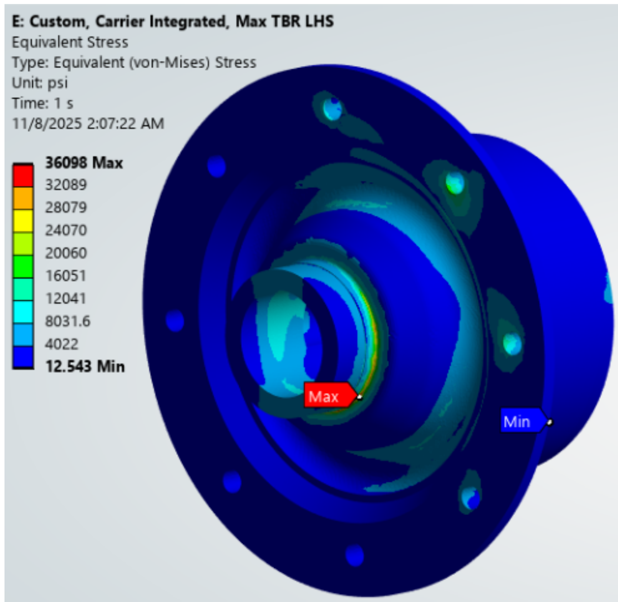

Figure 13: Von-Mises Equivalent Stress results for the right-hand side.

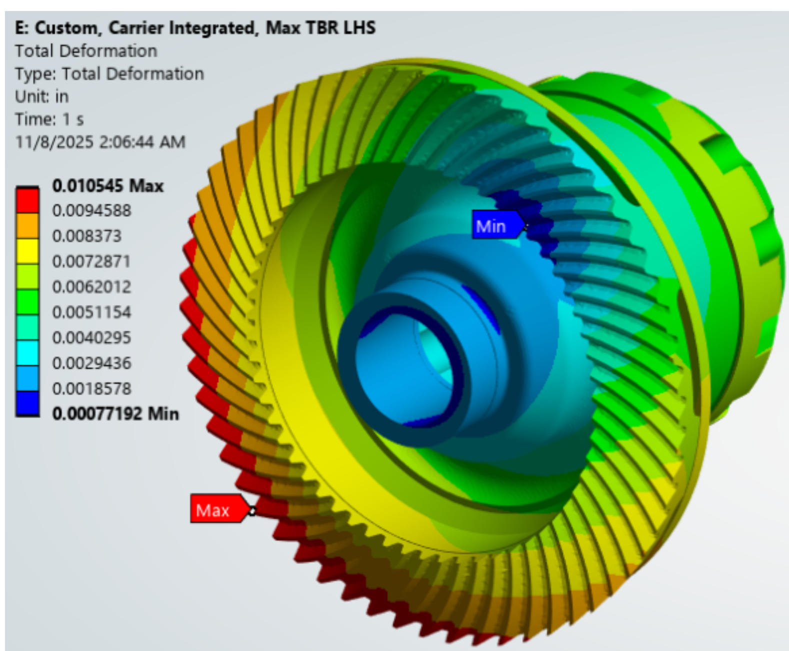

Figure 14: Von-Mises Equivalent Stress results for total deformation.

These results seemed promising for the project, but one thing that stood out was the stress concentration at the compression-only boundary condition. In my head I was having trouble equating that with an actual stress that would occur; the bearing it was butting up against was extremely stiff, but the larger front gearbox casing that the bearing was pressed into was extremely deformable, due to its low thickness and fully aluminum construction. I had a hypothesis that if I were to put the front gearbox into the model, it would take up a lot of the deformation from the previously infinitely-stiff boundary condition, and that stresses would be completely redistributed on the differential casing itself (due to worrying less about the region of the casing interacting with the bearing, and more about the casing being pulled apart at the bolted connections since it now had expanded freedom to move in that direction.

2.6.2 Expanded-Scope Modeling

For the expanded-scope modeling, I modeled in dummy bearings that supported the differential, and then added in the front gearbox casing. I then pushed the boundary conditions further outwards, such that the gearbox tabs to the frame were now the infinitely stiff points, and the region we want to study (the differential casing) was far from this influenced region.

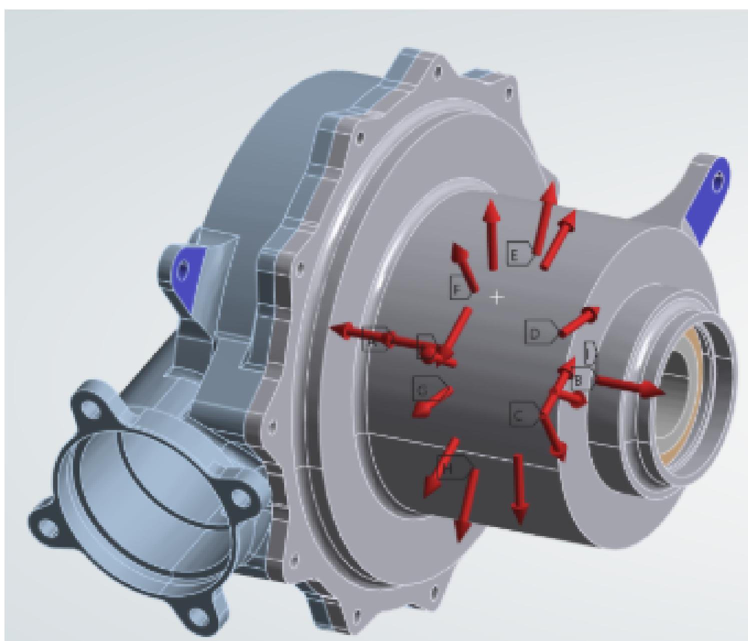

Figure 15: Final FEA setup for the differential casing. Note that 6DOF of boundary conditions have been pushed outward to the front gearbox casing. This constrains the differential in 5DOF, with the gear tooth support still needed to fully constrain it within the housing.

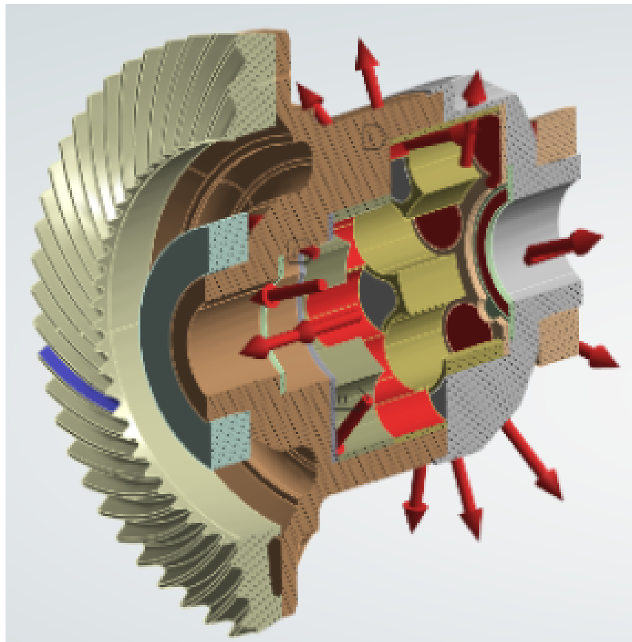

Figure 16: Cut-section view of loads applied on the internal of the casing.

The mesh was carefully optimized, with a coarser mesh being left on the outer gearbox and dummy bearings, but refined in the areas of study.

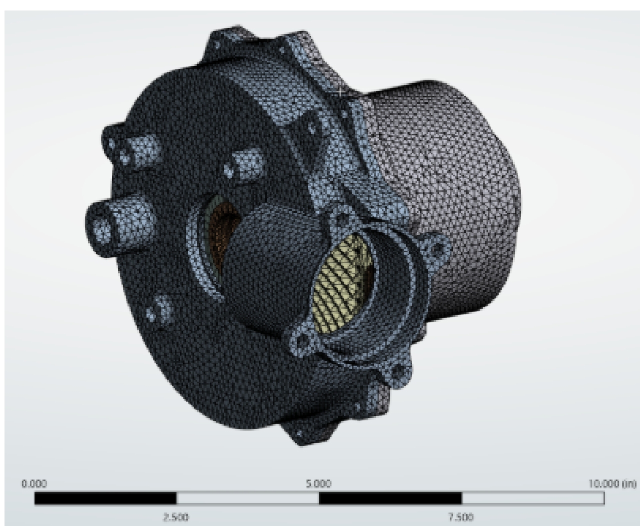

Figure 17: Coarser mesh on the outer gearbox casing.

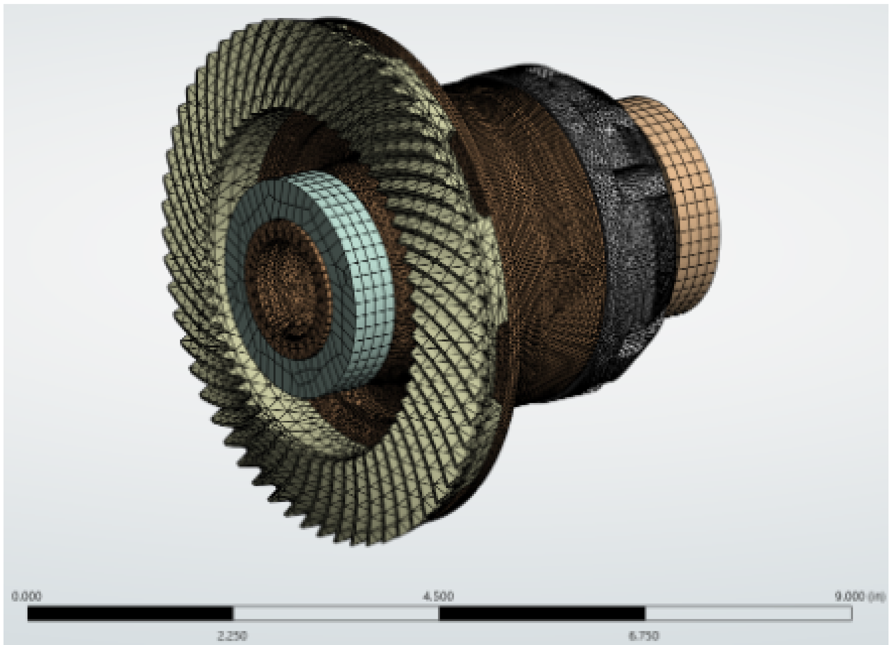

Figure 18: Finer mesh on the differential casing itself.

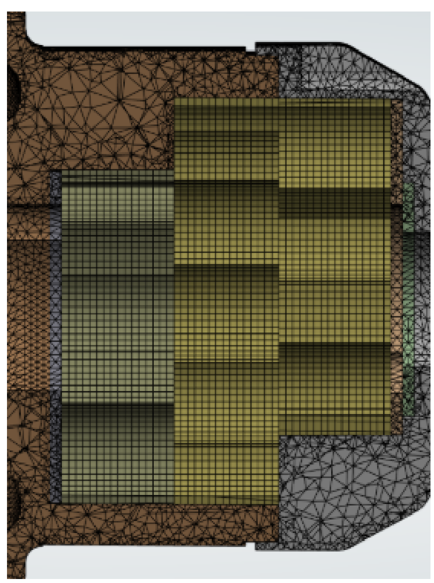

Figure 19: Even finer mesh on the contact sleeves inside the differential.

The stress and deformation tells an interesting story:

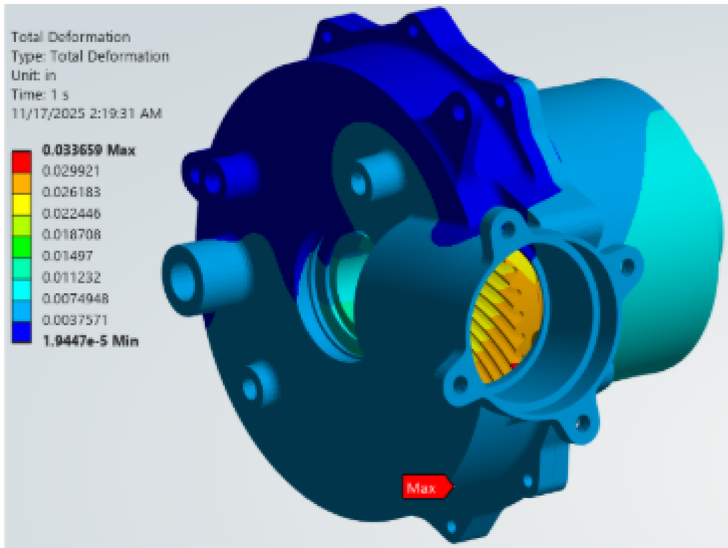

Figure 20: For the total deformation result, it can be seen that the internals deflect torsionally much more than the outer casing deflects axially. This makes sense given the large magnitude of torque that the ring gear applies as well as its relatively un-stiff load path.

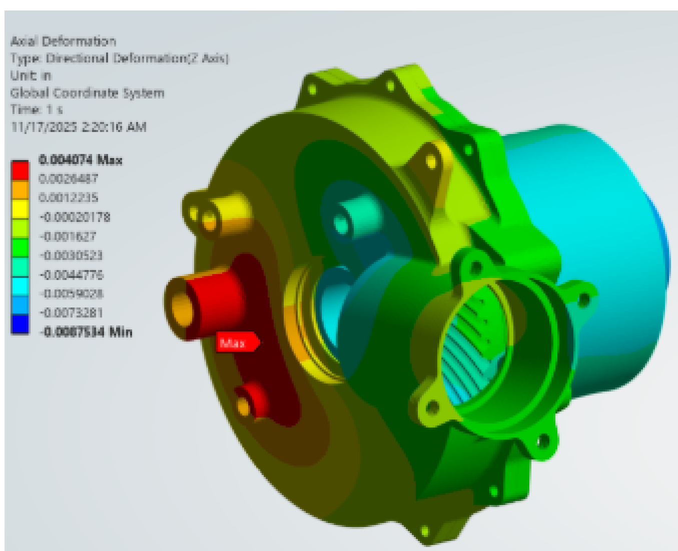

Figure 21: For the axial deformation result (e.g. along the plunging cup axis), it can be seen that the casing is deflecting outwards at a high rate, showing that some of the force from the diff internals is reacted away through this load path.

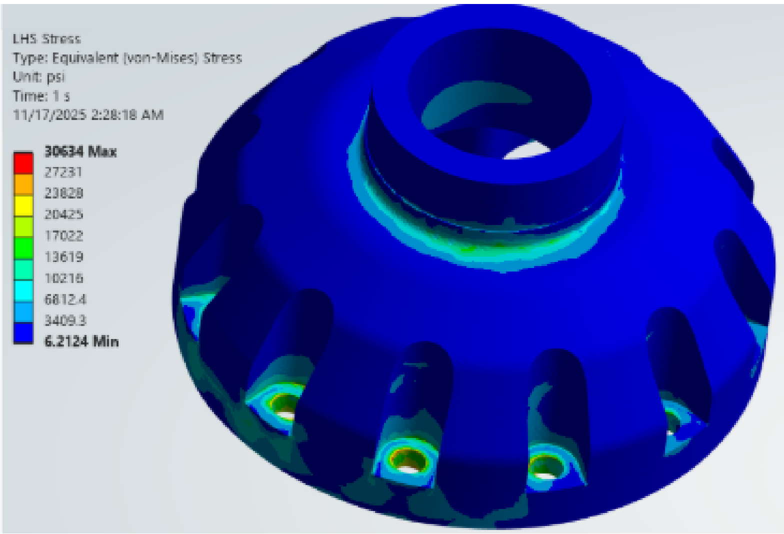

Figure 22: On the left-hand side, we still see high stress near the bearing abutment. This shows the casing is somewhat stiffer than anticipated, as I had hypothesized there would be near-zero stress in that region. There is a redistribution of some of the stress to around the bolt holes, which was expected, as now some of the load has to path through those joints.

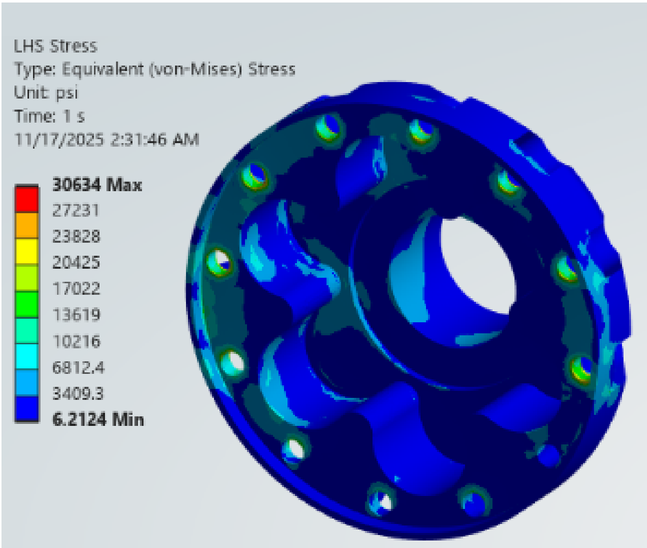

Figure 23: On the inner view, it is even more apparent that a lot of stress has been redirected away from the contact surfaces and is instead reacted around the bolt holes.

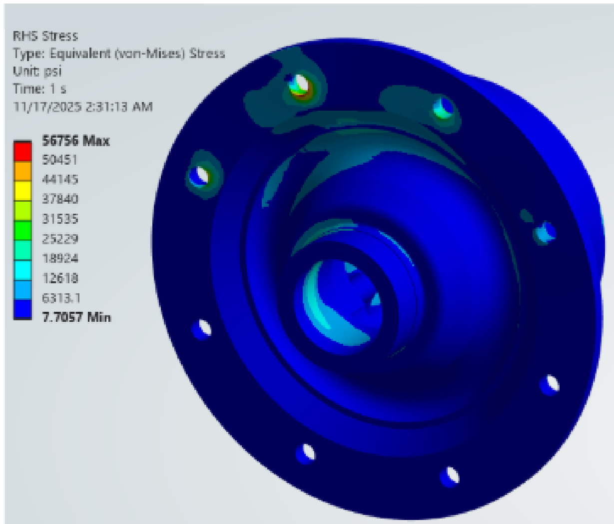

Figure 24: For the right-hand side, we see a bearing-stress region around the bolt hole closest to the frictionless support on the ring gear. This makes sense as it is close to the infinitely stiff boundary condition. The model also does not take into account the bolt pretension, which would substantially lessen this bearing stress as some of the force would be reacted away through the clamped face. Overall this is not a region of concern.

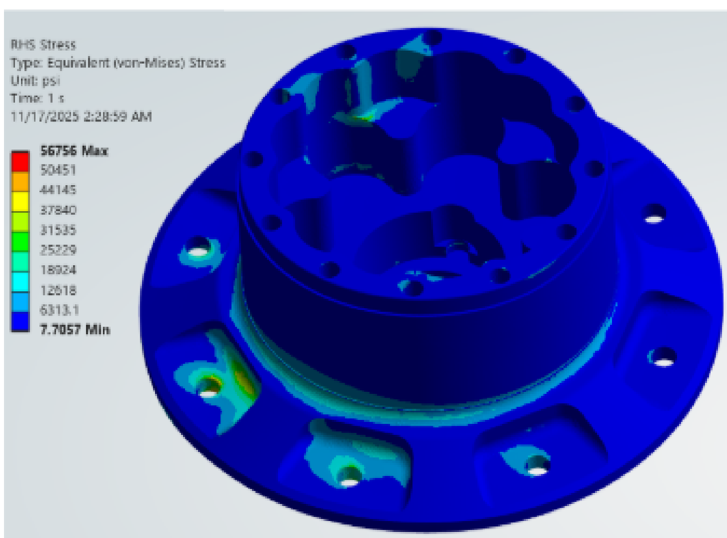

Figure 25: On the inner view, it is very apparent that this was modeled as the low-torque side, as there is essentially no stress on the internals.

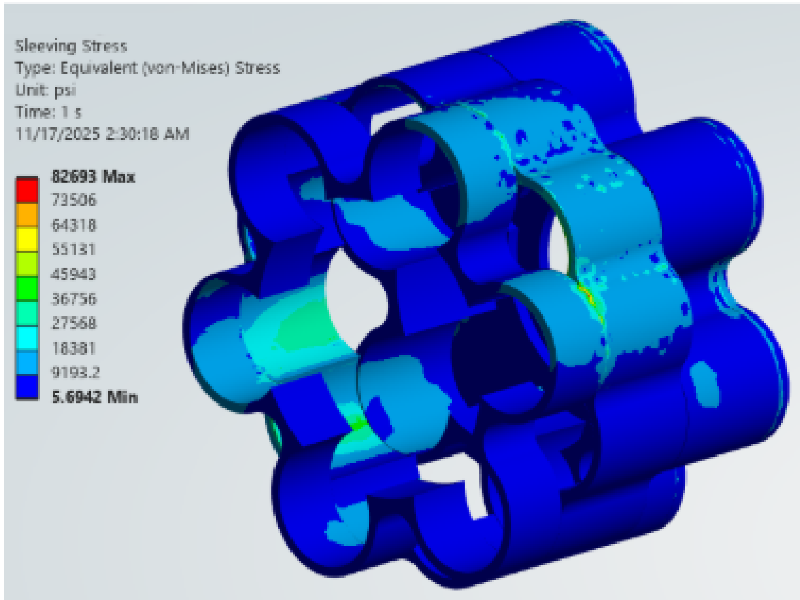

Figure 26: Finally, we also take a look at the stresses on the contact sleeving. The stress appears to be well-distributed and the sleeving is doing its job of reacting out most of the force.

Overall Factors of Safety:

LHS	2.38
RHS	1.29 (artificially low)
Sleeving	1.63

2.7 Manufacturing Drawings

Since the design was assessed to be worthy of production, manufacturing drawings were created for the two casing halves.

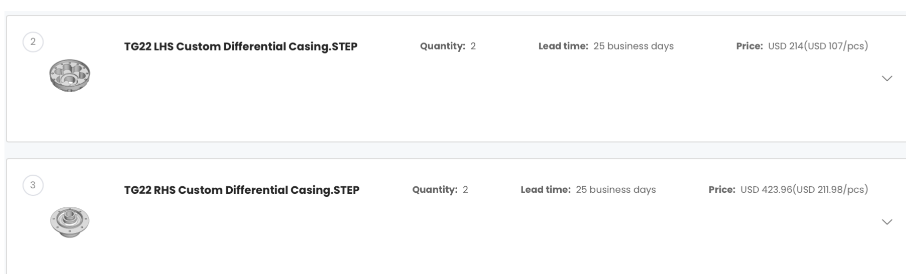

Figure 27: Quote for the two casing halves. The total comes out to around $319 for one full casing set, leaving ample budget for hardware and laser-cut pieces.

2.7.1 Left-Hand Side Casing Half

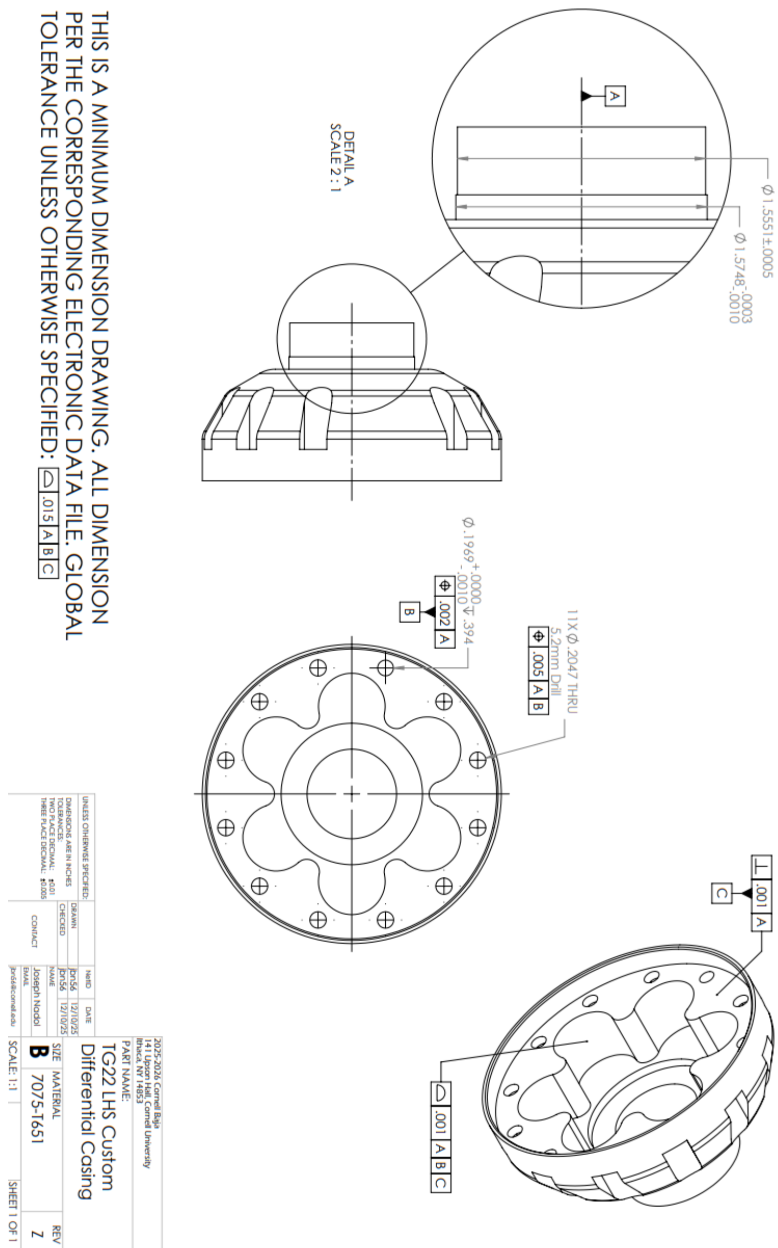

Figure 28: GD&T Drawing for the left-hand casing side. Datums defined as part centerline, locating pin, and mating surface. Important features controlled: bolt pattern, surface profile of the sleeving press, SpeediSleeve press.

2.7.2 Right-Hand Side Casing Half

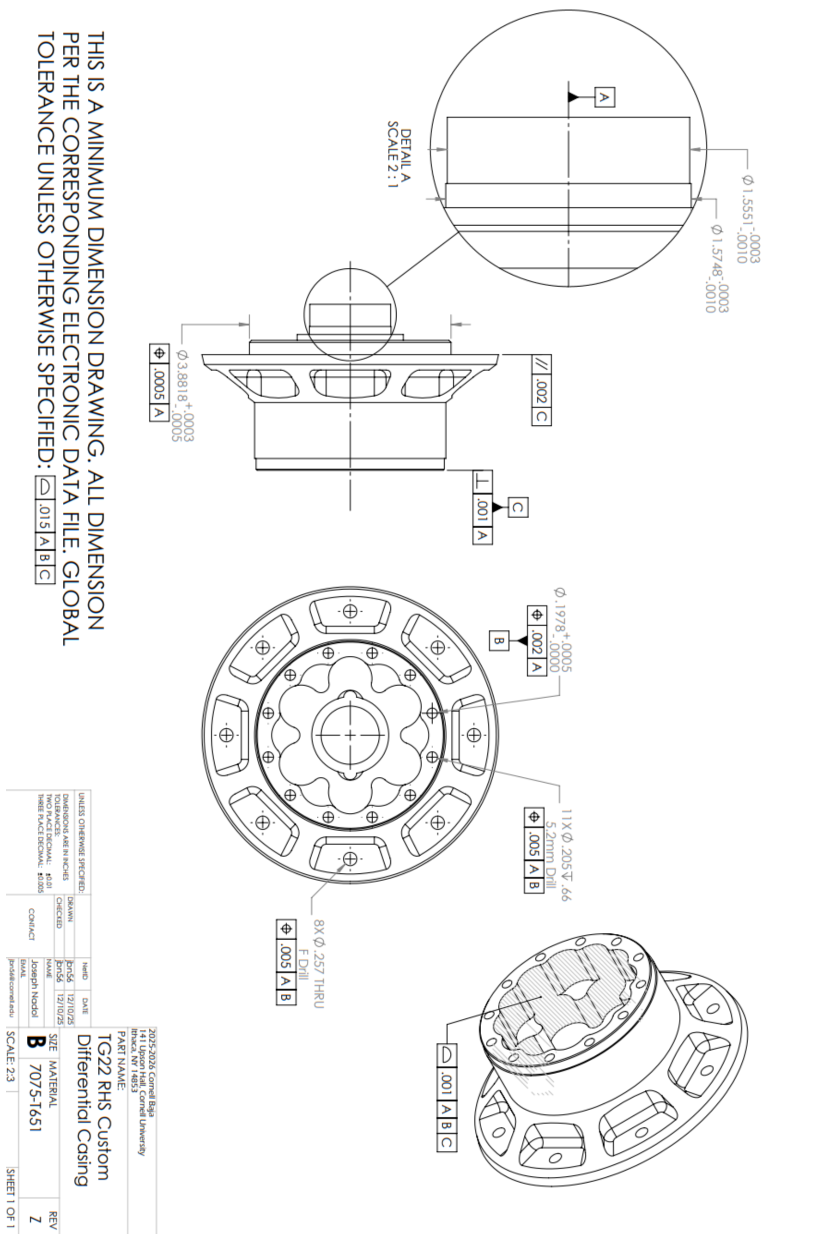

Figure 29: GD&T Drawing for the right-hand casing side. Datums defined as part centerline, locating pin, and mating surface. Important features controlled: bolt pattern, surface profile of the sleeving press, SpeediSleeve press, parallelism of ring gear carrier, concentricity feature for ring gear location.

2.8 Full Assembly Renders

Figure 30: Isometric assembly view.

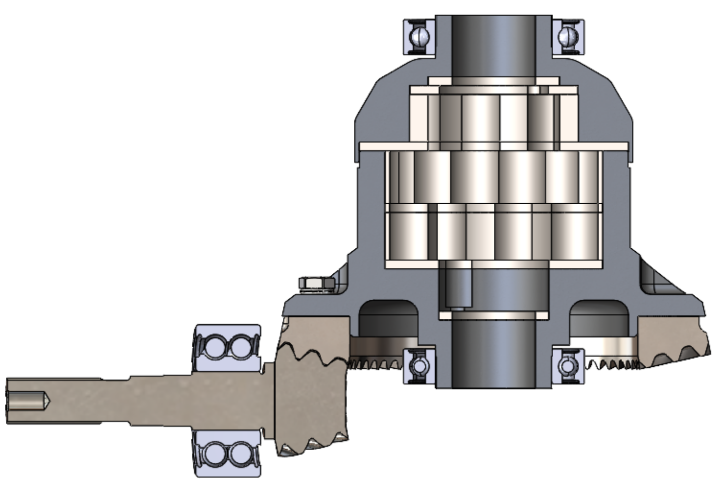

Figure 31: Cross-section assembly view.

2.9 Conclusion

2.9.1 Project Outcome

The aluminum custom differential casing weighs 3.12 lbs in CAD, as opposed to the OEM's 4.93 lbs. The weight reduction in this system allows us to buy back structural integrity in other systems at net zero weight gain. As such, we have upsized our front and rear reductions to be able to deliver more torque to the ground, with weight gain in both components fully covered by the weight loss in the differential. The place in which the most weight was removed from the casing was the outer edge, which led to a reduction in rotational inertia. This will have the effect of increasing efficiency in 4WD, as well as decreasing shock-load on upstream components. Finally, the decreased amount of bolted connections decreases the points of failure and helps further contribute to weight reduction.

2.9.2 Next Steps

The casing is currently being manufactured by an external machine shop, and the EDM and laser cut sleeving will similarly be outsourced soon. During January we will assemble the casing and install it into the TG22 front gearbox in time for first drive. During the spring we will have to conduct rigorous testing on the casing to ensure that it will hold up during an endurance race and not critically fail. Overall, this project has been on the right track so far but it will require a lot of work to realize success on the car.

3 Rear Differential Driveline Retrofit

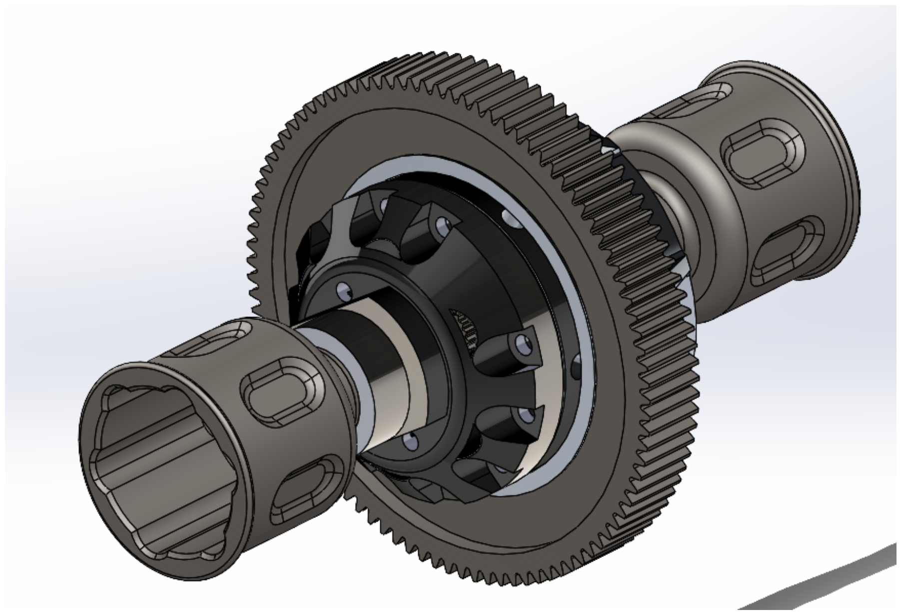

Figure 32: The Wavetrac integrated into G2 Gear with plunging cups and ring gear carrier attached.

3.1 Abstract

The Baja SAE series has continued to increase the necessary capabilities to maintain a competitive edge. One of the largest ways this has been accomplished is through the increase in 4WD, not just needing a working system, but a system that offers significant vehicle benefits. In the past four years, the need for 4WD has grown significantly with the inclusion of rollers to start races and challenging obstacles. For the TG21 car, major improvements were made to the usability of 4WD with the integration of the Porsche 996 limited-slip differential into the front gearbox. This integration helped minimize the torque steer effects in 4WD, allowing for improved long-term steering effort. The differential integration also successfully heightened the yaw moment during 4WD driving, which allows the car to be comparable to the RWD characteristics. As previously mentioned, the design tested and completed last year allowed the team to test the integration of the differential and determine if the improvements were significant. In a similar fashion, this year we will design an updated TG21 driveline that will integrate the Porsche 996 differential into the rear gearbox. The second differential will continue to the handling of the vehicle and will improve the present understeer in cornering situations and improve lap consistency in low-traction scenarios like the endurance course over a four-hour period. The integration of the rear

differential will be tested against the current TG21 performance to gauge the effect and will be used to drive the TG23 design decisions and possible packaging.

3.2 Design Requirements

The design must:

- Must be fully integrable into a legacy vehicle
- Must utilize the OEM casing and require no further modification than the legacy front integration
- Achieve a factor of safety of 1.15 against yield

The design should:

- Minimize the weight gain of the rear box
- Minimize the number of components that need to be re-machined
- Require simple machinability
- Maintain maximum possible function of the legacy vehicle

3.3 Initial Research

3.3.1 Previous Integration of Differential

As pictured below, for the TG21 car, a Porsche 996 limited-slip differential was integrated into the front gearbox. For this process, they used a modified OEM casing. The only major modification was that the bearing surfaces were turned down to accommodate a SKF bearing. The system used a part called the ring gear carrier, which is an aluminum structure that is rigidly attached to both the differential casing and the ring gear. Due to the splined output of the differential, splined adapters were designed to attach the legacy plunging cup design to the system with limited modifications. It is also important to note that the project was designed around being able to switch out the differential for a large sprag bearing in the case that the tested performance was significantly worse. Other large considerations of this integration are that the ring gear is subjected to large reaction loads, and the deflection of the carrier must be minimal to allow for proper meshing of the gear teeth. This system utilized two sets of bearings, one located on the differential and the other set pressed into the front gearbox casing that reactions the load of the plunging cups. The plunging cup bearings what is used to shim the front reduction.

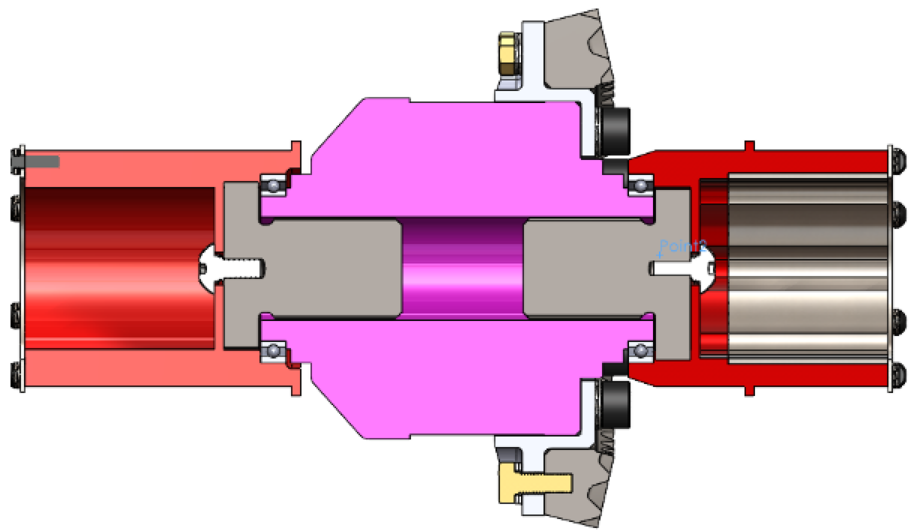

Figure 33: The Wavetrac integrated into the TG21 front box with plunging cups and ring gear carrier attached.

3.4 Needed Modification of Existing Front Integration

One of the key issues that was identified in the competition cycle last year was that the bolt axially constrained the plunging cups, partially backed out of the adapter. While this was not an issue during driving due to the bolt not fully disengaging and the plunging cups also being pressed on, it is an area of improvement for the rear design.

Another key aspect is that the front integration is purposely offset to accommodate the propshaft routing and achieve the best possible angle for the efficiency benefits. The system is also offset to allow for clearance of the steering rack and meet driver foot clearances. This had the added benefit of the ring gear directly reacting the load onto the casing as it is not held away from the casing. However, in the rear, there is not enough room to have the differential off-center. The CVT guard and the bevel box would contact the casing if it were significantly offset, as well as the rear driveshafts would have to be significantly different lengths, which has negative suspension geometry effects and increased machining time.

3.5 Design

3.5.1 Design Constraints

The largest design constraint to solve was the goal of this project is to fully integrate the OEM casing without any major modifications. In order to maintain the correct amount of driveshaft plunge, the width of the rear gearbox has to increase by over 2 inches if the differential were completely centered in the car. Due to the already significant increase, it was determined that centering is vital because if not, the suspension would have to undergo significant changes. Still with the centering of the differential, the clearance to the CVT guard needs to be heavily considered. As visible in the image below of the legacy clearance, there is only an inch and a half of clearance. The final design results in just over half of an inch of clearance. Other design constraints came from keeping the existing driveline geometry. Since the relative location of the gear could not change, any modifications to the casing could not interfere with the mounting of the dog clutch or the meshing of the first reduction.

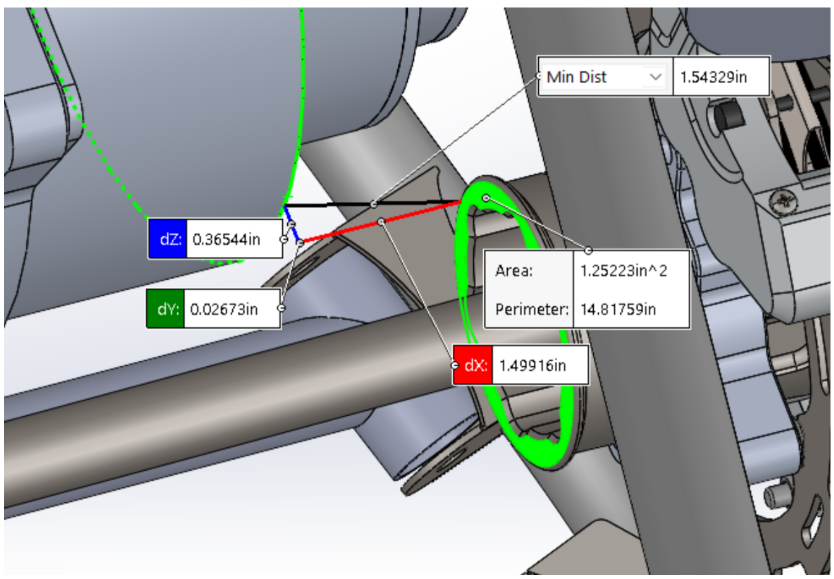

Figure 34: CAD Image of the Clearance to the CVT Guard From the Rear Gearbox

3.6 Selected Solution

The selected Solution was to use a carrier to constrain the G2 gear in the center of the differential and car. This carrier would translate the torque from the gear to the differential. Additionally, to solve the legacy issue of the plunging cup axial constraint, the TG22 plunging cup design would be used instead of a modification of the legacy system. This was because the new design was created for the front box, meaning that the spline profile would match the new rear design and would allow for spares to be used for testing, which would save on machining times. The selected solution modifies the driveshaft lengths, making the TG21 front driveshafts now usable in the rear. As well as the casing and G2 design. G2 is the large gear in the second reduction that will wrap around the differential. G2 was previously keyed to the plunging cups but will now have a solid design with weight-saving pockets that is thread-milled to a 10-24 bolt size. G2 will then be rigidly attached to the carrier. The selected solution also modifies the differential casing slightly by turning down the bearing surface to allow for a different selected bearing to be used.

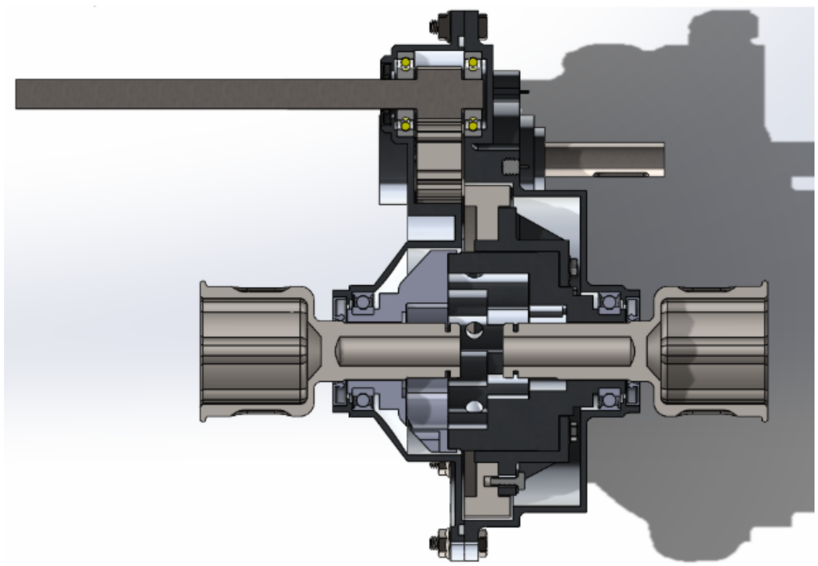

Figure 35: CAD Image of the Cross-section of the Full Assembly

3.6.1 G2 Carrier

The carrier is an Al 6061 support that rigidly connects G2 to the differential. It also positions G2 to be centered in the car. Due to this connection, G2 will be shimmed using the bearing pressed into the casing. The carrier is designed to provide structural support to G2 and ensure that it is constrained axially. The carrier was vital in adapting G2 to fit on the OEM casing without having to make any major differential casing modifications. Below is the completed carrier CAD featuring the bolt patterns that attach it to the webbing of G2 and to the back of the OEM casing.

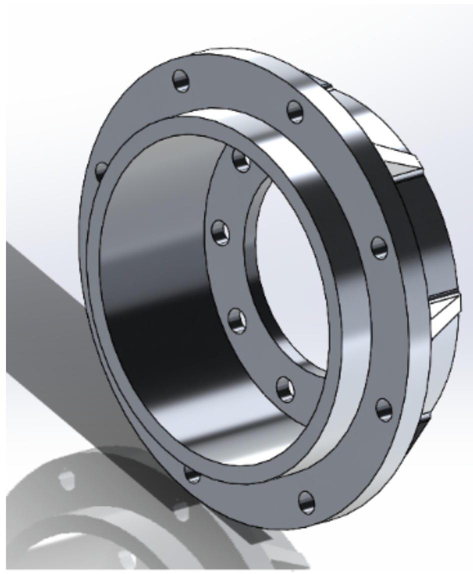

Figure 36: CAD Image of Rear Carrier

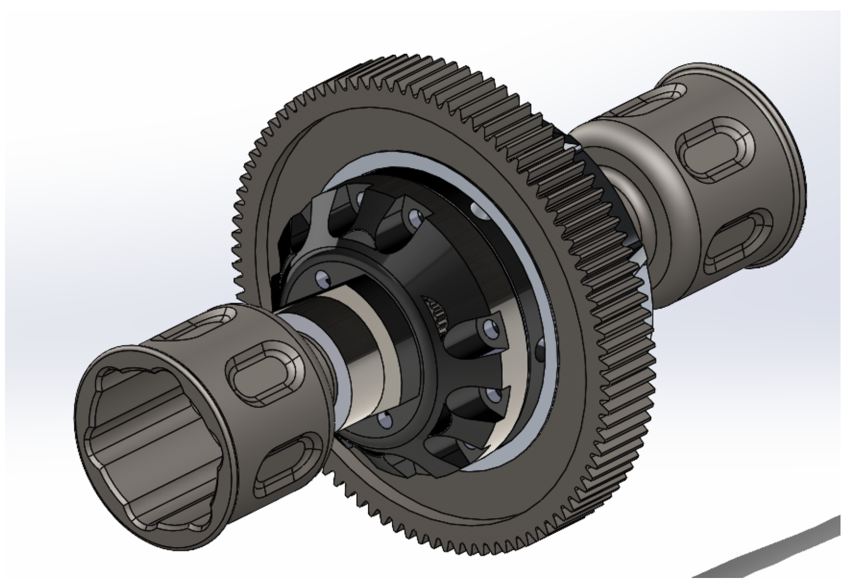

Figure 37: CAD Image of the Rear Carrier Assembly

3.6.2 Casing

The TG21 casing was heavily modified to fit the integration of the rear differential. In order to maintain the Baja SAE rules, the side walls of the casing are to extend 1.5 inches on side A and 1.25 inches on side B. Additionally, side A was tapered to ensure that there was clearance for the rotor and the first rear reduction. While the taper of the casing does significantly increase the machining effort and possible costs, it is optimal to ensure that the necessary components fit. However, the side B casing was not tapered to fit the exact outline of the carrier. This is largely due to concerns about machinability and unnecessarily driving the costs of machining up. Additionally, in a brief comparative analysis of the casing weight, the form-fitted side B was heavier than the presented boxy solution. It is important to note that this only compared the CAD weight and did not include the projected AFT weight that would be added to the casing.

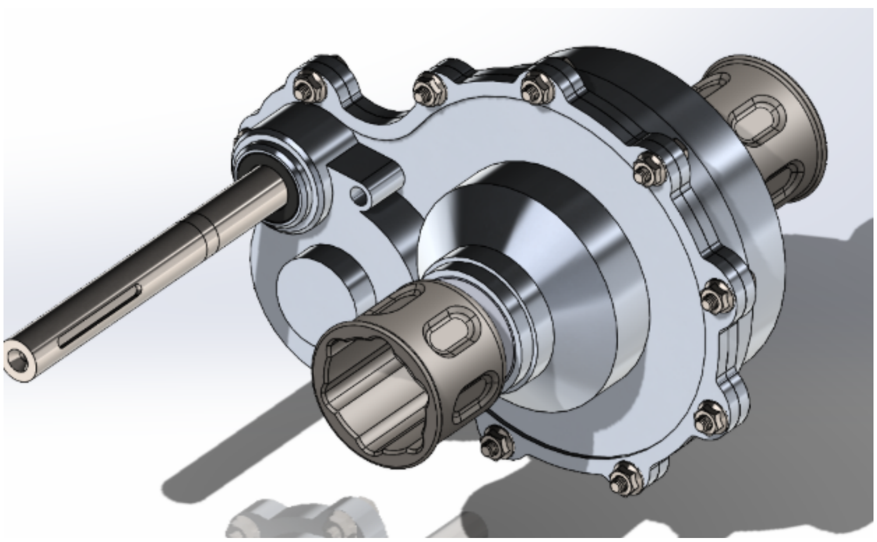

Figure 38: CAD Image of the Casing Side A

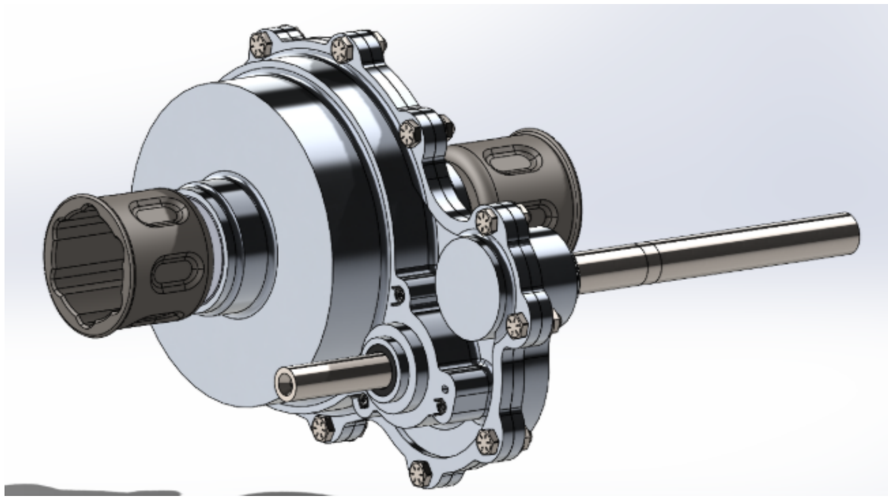

Figure 39: CAD Image of the Casing Side B

3.7 Finite Element Analysis

3.7.1 Initial Load Derivation

The major loads interacting with the rear gearbox are the reaction of the dog clutch in the mounting surface, the braking force acting against the top two sealing tabs, the gear loads, and the force of the CVT pulling the P1 input shaft.

Rear Reduction Loads: All rear reduction loads are calculated using KISS-soft AG which is a Gleason software. This program takes the designed gear parameters and calculates the reaction load onto the gears.

CVT Loading: The load used to reflect the CVT is the low-end force reacting across the span of the shaft that the secondary rests on. This load is applied in the direction of the CVT in the car space using split lines projected onto the shaft.

Braking Force: The braking force is applied as a remote for acting along the reaction line of the caliper. This load is determined by the braking force calculations.

Dog Clutch Loading: The load of the 4WD system is measured from a force probe of the dog clutch and bevel box assembly ANSYS, and is applied in on the surface the dog clutch is constrained by.

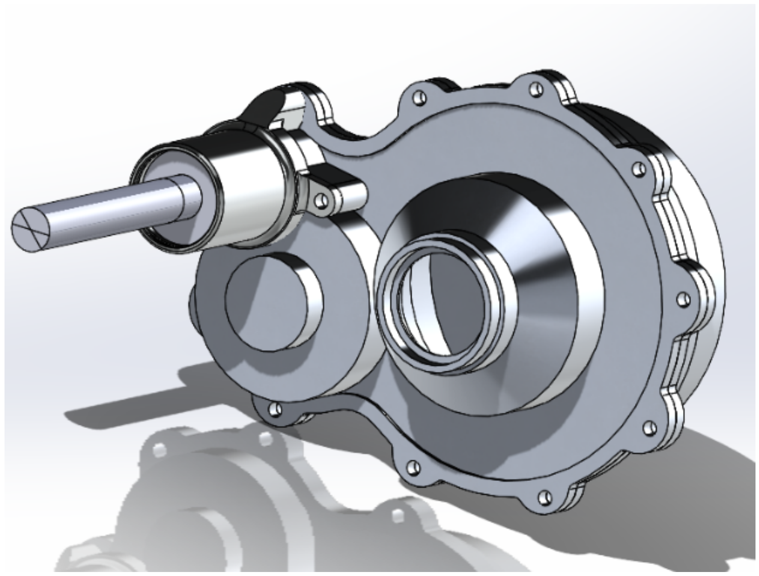

Figure 40: The Rear Casing ANSYS CAD with Reaction Shafts and Split Lines

The figure above shows how the split lines are applied to the modified shafts in order to apply the load of the gear reaction in the correct direction vector. These shafts have additional split lines that are the face widths of the gears to approximate the full effect of the reaction onto the casing.

3.7.2 Initial Casing ANSYS

Due to the large changes to the rear casing design, an initial ANSYS run was run to validate that the casing was not significantly structurally affected by the changes. The loads and locations for this case match the ones outlined in the previous section. The results of this ANSYS setup show an improved casing FOS of 1.74 for the 4WD load case with the material being Al 6061 T6. This is significantly higher than the TG21 initial FOS of 1.30 to the same load case. The major contributing factor for this change was that the casing was forced to take on a boxier and less fitted design to accommodate the differential, which added a large amount of structure to the design.

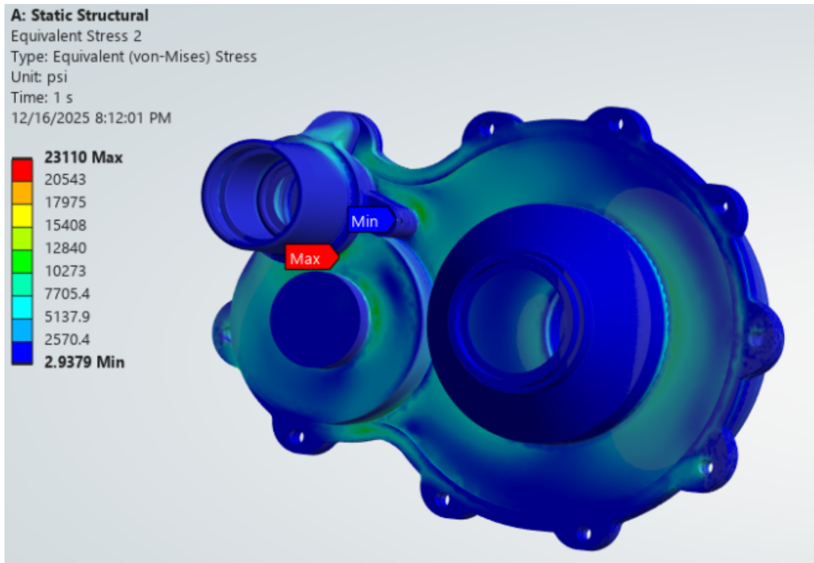

Figure 41: The Rear Casing ANSYS Experiencing the Standard Load Cases (No Diff)

3.7.3 Carrier and G2 ANSYS

The carrier and G2 were analyzed in a combined ANSYS setup. This was done to determine the deflection of the carrier as well as to calculate the loading on the internal casing bearing. The first step in this model was to set up the internal differential load as outlined in Joey Nadol's section of the report. This setup was then validated by comparing the results to his OEM ANSYS files. Next, the carrier and G2 assembly were added to a separate solve file to initially test that the constraints and supports of the system were producing the expected results, and the file was able to converge. Finally, the two setups were combined, resulting in the setup image below.

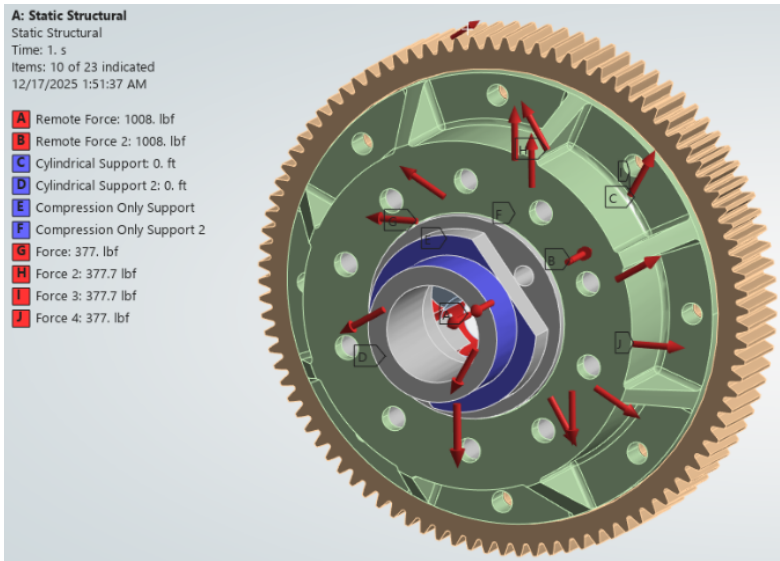

Figure 42: The Rear Differential With Carrier and G2 ANSYS Setup

For this setup, the load of the gear teeth to the face of three teeth is better replicated to better replicate how spur gears mesh. While this is still not entirely accurate for the desired outcome of calculating a conservative maximum stress and deformation, this setup worked. Additionally, the same setup was run with the load only being applied to one tooth to gain an additional validation of confidence.

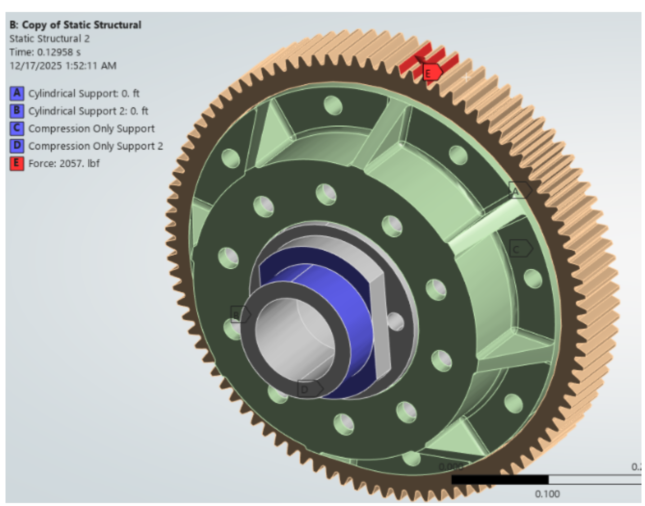

Figure 43: The Rear Differential With Carrier and G2 ANSYS Setup in Initial Validation Process

The results of this setup demonstrate that the carrier does not experience a large amount of deformation. This result makes sense due to spur gears not having significant axial reaction forces. Meaning that when compared to the front carrier to rear see much lower loading. These results point to the conclusion that the carrier can be decreased in the material thickness; however, the rear-box experiences a high amount of thermal effects as well as a large deflection, so it was decided not to decrease the thickness but to instead increase the cutouts on the carrier. Additionally, further reduction of the carrier would break the Baja SAE rule set on HROE guarding.

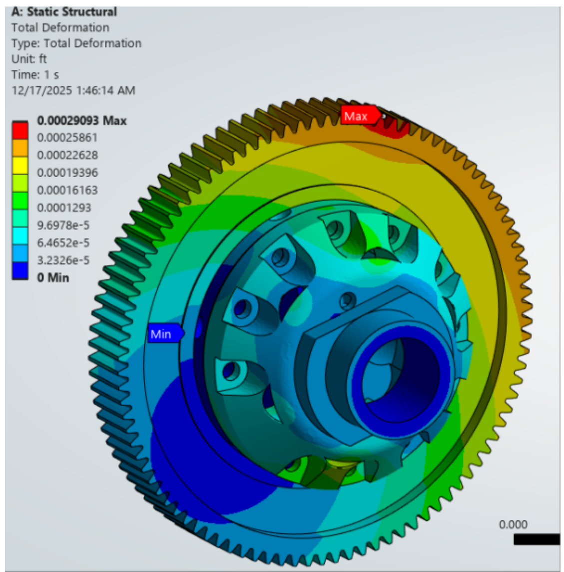

Figure 44: The Rear Casing ANSYS Results Deformation

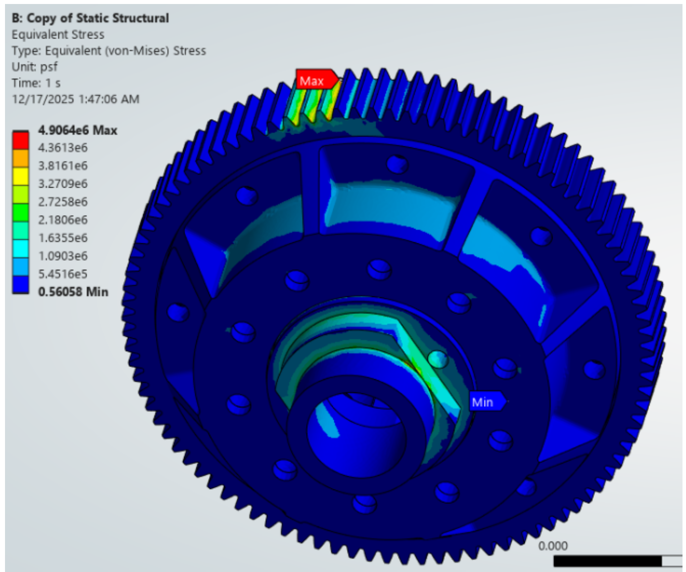

Figure 45: The Rear Casing ANSYS Results Stress

Further conclusions of the results demonstrated that the factor of safety on the carrier is 1.75. These values were determined by probing the maximum stress on the carrier and comparing it to the yield strength of AL 6061. Due to the same conclusion as above, this was deemed an acceptable and optimal FOS for testing and ensuring that the system is rigid.

3.7.4 Differential Load Measurement

In order to properly apply the reaction load of the full differential system onto the bearing surfaces of the gearbox, the load was measured from the carrier ANSYS with a force probe. Then, this load was able to be applied in the full casing setup with bearing force. This force was also used to determine the bearing selection.

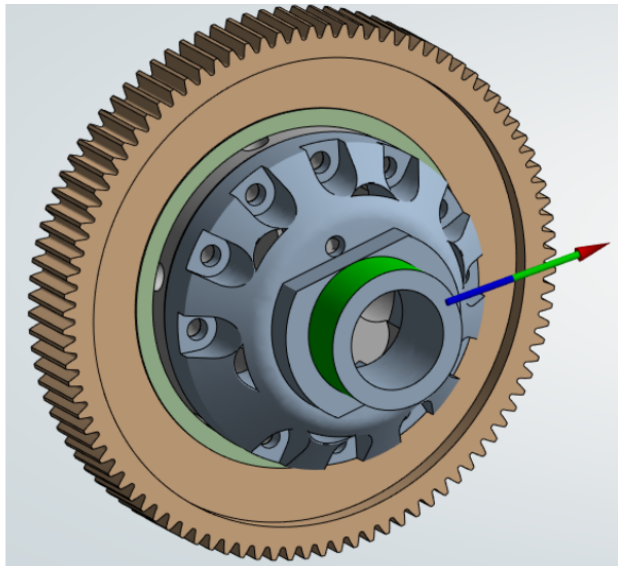

Figure 46: The Carrier ANSYS Setup with Force Probe Visible

3.7.5 Full Casing ANSYS

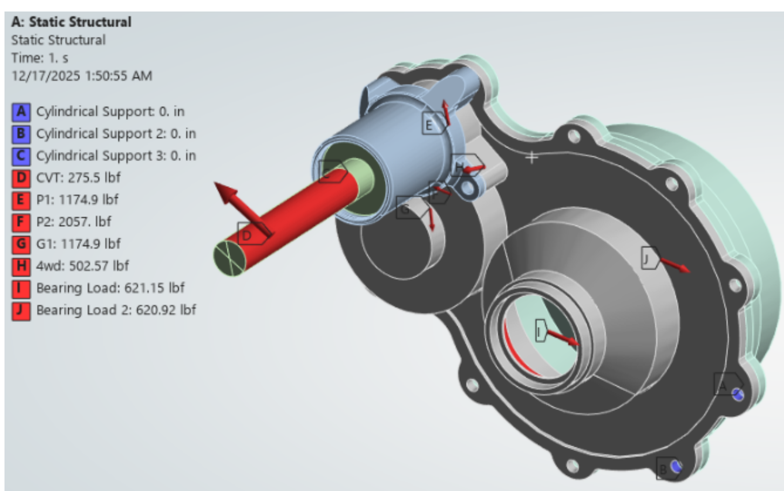

Figure 47: The Rear Casing ANSYS Setup

The full casing ANSYS was setup in the procedure outlined above with the addition of the derived differential loading. Due to the nature of the casing loads the most rigorous loading is 4WD + Braking + Differential with is the results outlines below.

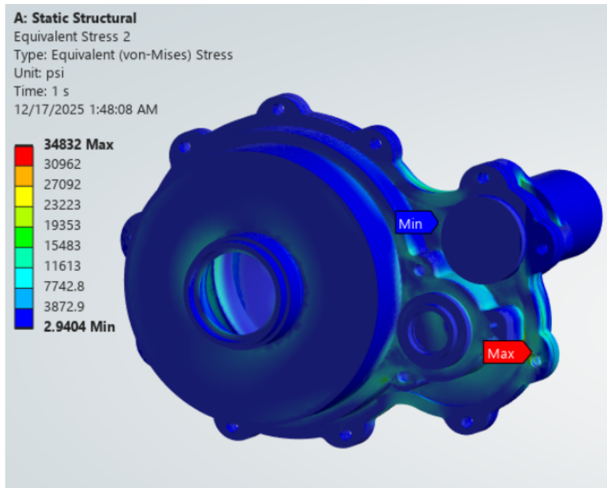

Figure 48: The Rear Casing ANSYS Results Stress

The results of the equivalent stress demonstrate that the location of the highest stress is at the joint of the dog clutch and the frame mounted bolt. This result is consistent to that of a standard casing ANSYS. Additionally, this case calculates the FOS to be 1.2, to yield which achieved the objective outlined in the design constraints.

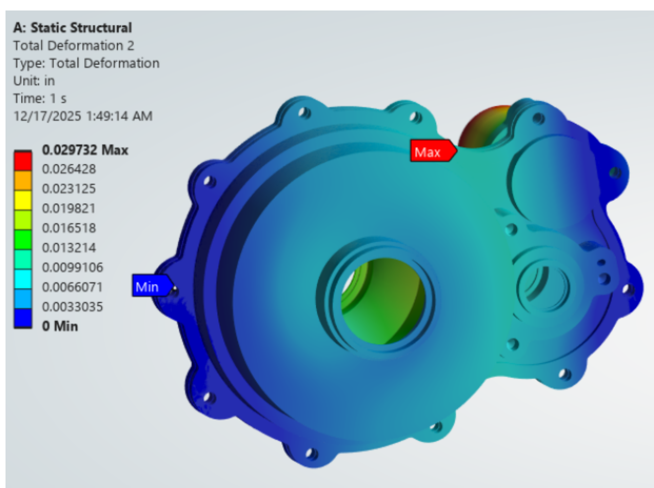

Figure 49: The Rear Casing ANSYS Results Deflection

The results of the deformation demonstrate the effects of the CVT force. This large deflection is also consistent with the casing results; however, the internal deformation was significantly higher than average. As seen in this image, the inside face of side A is experiencing high deformation. To assist in this ribbing was added to the outer surface of the casing. The ribbing had the added benefit of improving the machinability of the casing.

3.8 Hardware Selection

Due to the gears not changing in terms of their relative sizing and reaction loads, the same bearings could be used on the Input and Intermediate shafts. However, the bearing that will be pressed into the casing side and will react to the loads of the differential and carrier was changed. The selected bearing was the SKF 61907 due to the high maximum load and the minimal width. The FOS of this bearing is 1.07. Which, per industry standards, is considered concerning; however, in our use case of this bearing, the FOS is satisfactory and is comparable to the FOS of the other casing bearings. This is due to the understanding that a SKF maximum load is designed with a set FOS and the high loads will not cause significant damage or failure. Additionally, the seal was changed to accommodate the geometry changes of the rear system. The new seal is a SKF 11544 single lip wave seal. Additionally, all bolts were validated with the Baja Bolt to Joint calculator. These values were then used to model the bolts as bushings in ANSYS and further probed to ensure performance.

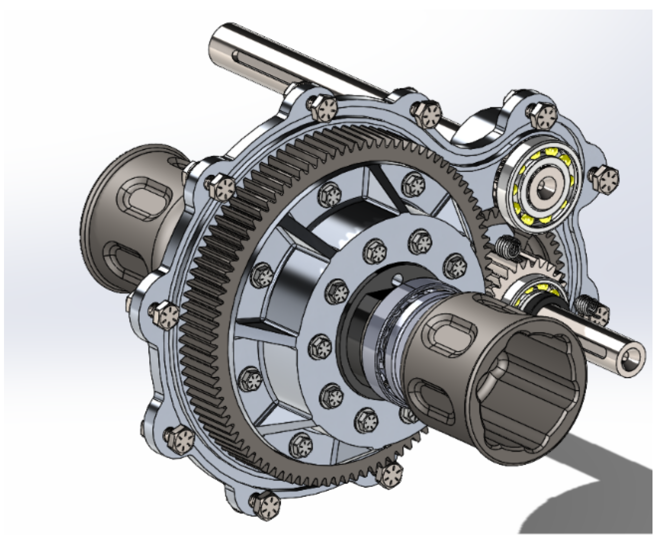

Figure 50: Full Rear Hardware Assembly Partial Cross Section

3.9 Full Assembly CAD

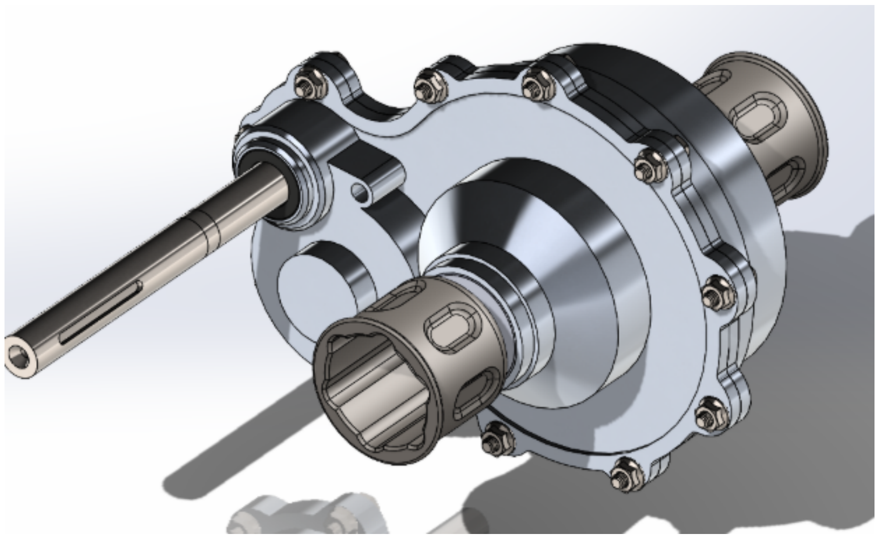

Figure 51: Full Rear Assembly Including Hardware Side A

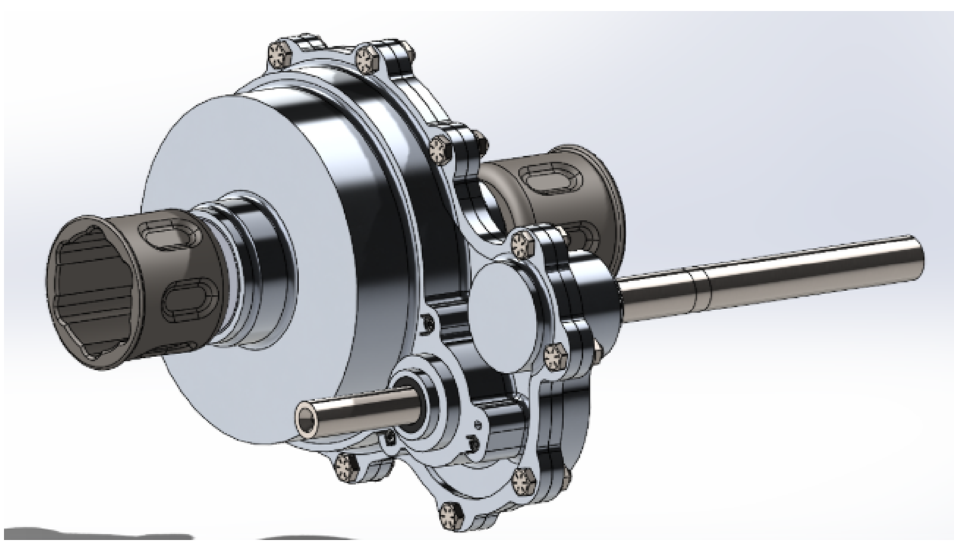

Figure 52: Full Rear Assembly Including Hardware Side B

3.10 Conclusion

3.10.1 Project Outcome

The outcome of this project is a fully integrated rear differential that can be installed and tested on the TG21 competition vehicle. This testing will inform the team about the performance characteristics of a competition car with both a front and rear differential. The testing will include comparing the performance TG21 in four different configurations. This first is TG21 with no differential, then with the front differential, followed by just the rear differential, then with both differentials. This project also provides a preliminary integration that can be modified to include a custom casing on the differential to improve the weight allocation that would be required of the system.

3.10.2 Next Steps

The next steps for this project are to determine a testing timeline based on manufacturing lead times and finalize drive shaft lengths. While the lengths of the driveshafts were not greatly impacted by the integration of the rear differential, the driveshafts will need to be shortened by an inch. This reduction is still compatible with the geometry and should not greatly impact the performance; however, a final clearance check will need to be performed with the suspension geometry designer before part send-out. Additionally, since this is largely a research and development project, the manufacturing process can not interfere with the development of this season's competition vehicle. This means that a large number of parts will be sent out for machining after we have received the current parts for the TG22 car.