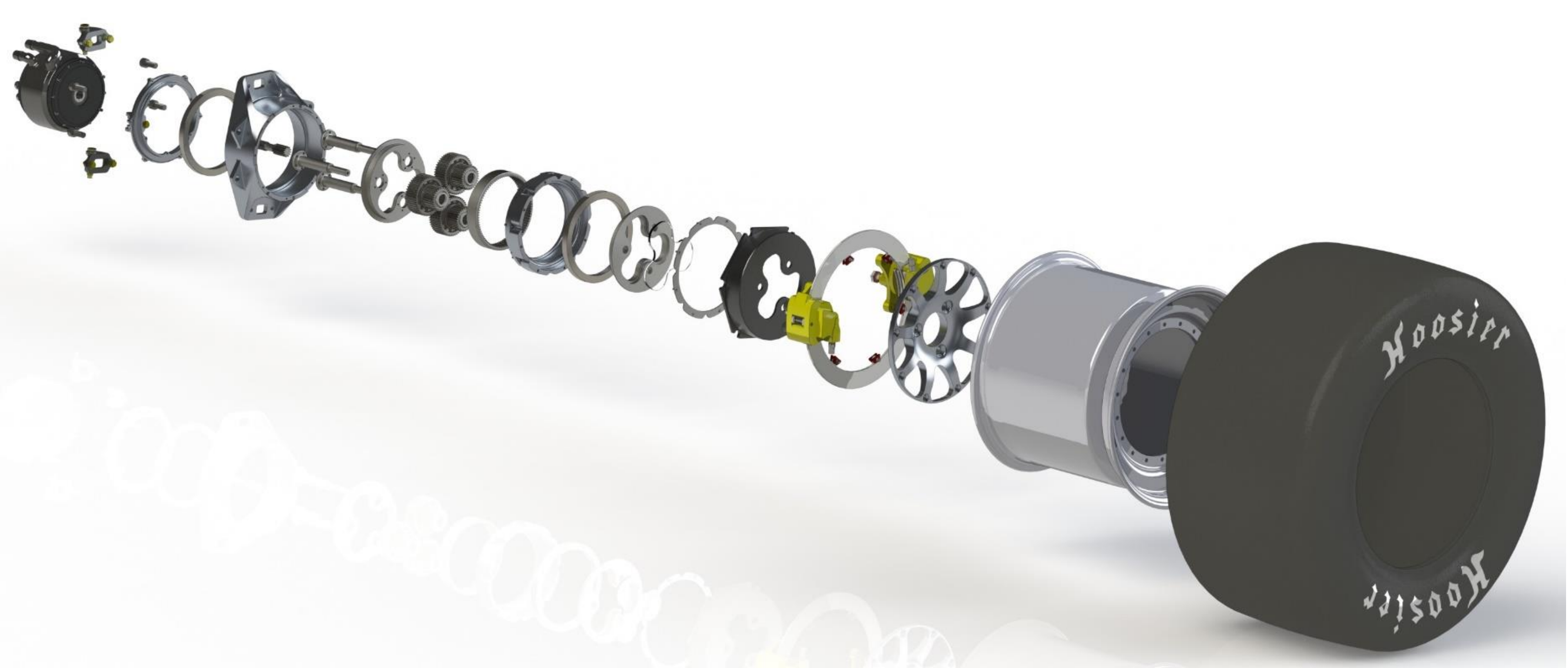
Wisconsin Racing 218e In-Hub Corner Assembly

Lead Engineer: Kevin Byrne



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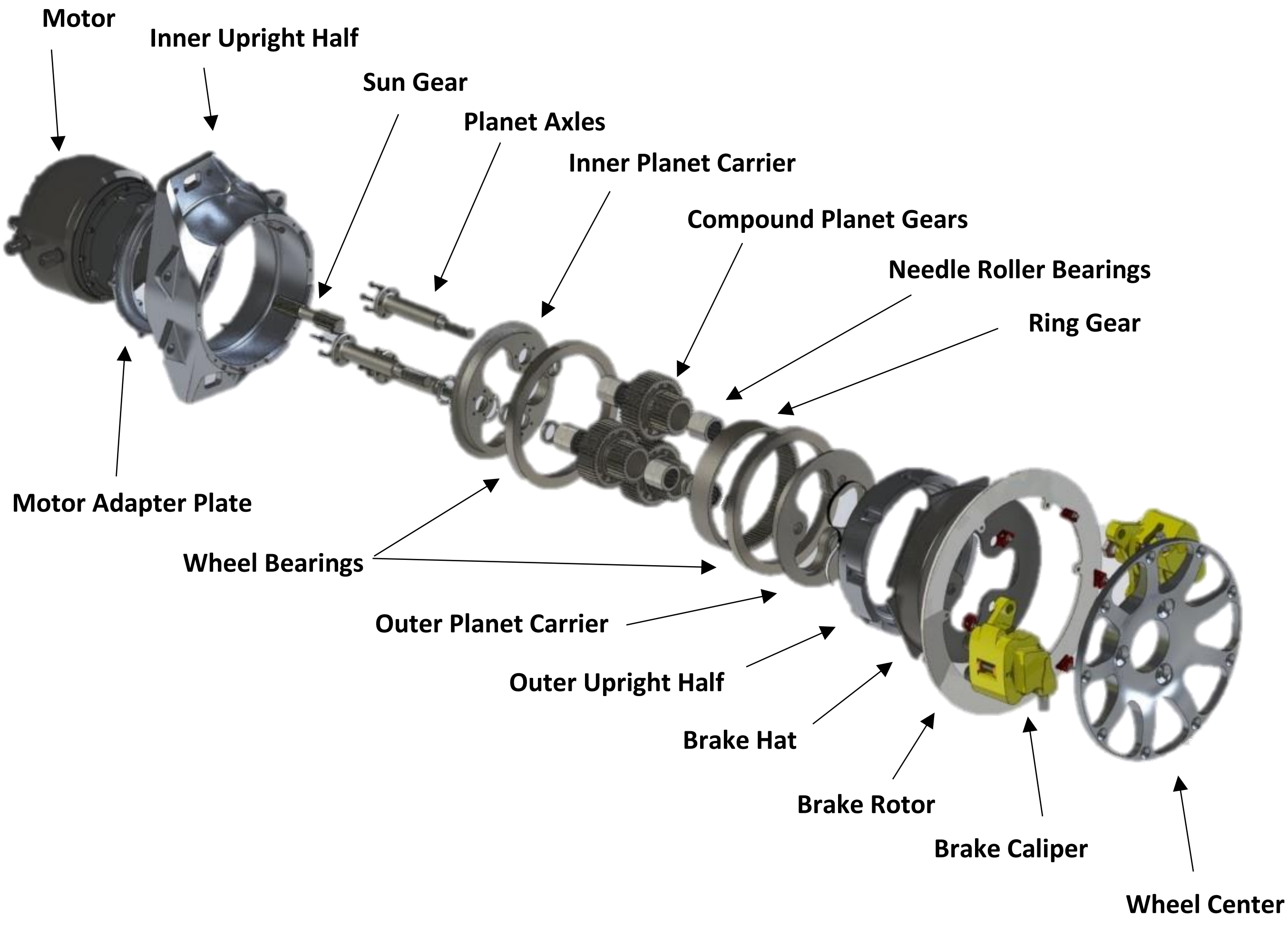

Project Background

With the WR-217e, the Wisconsin Racing Formula Electric team brought to life one of the most ambitious and powerful drivetrains for a first-year team. We combined two different drivetrain packages within one car in hopes of maximizing the feasibility, reliability, and performance all simultaneously. We did this by taking an aggressive approach on a in-hub design in the front and pairing it with a traditional and reliable gearbox in the rear. To give a quick overview, the WR-217e showcased a rear drivetrain package featuring two Plettenberg Nova 30 motors paired with a two-stage gear box that housed a 4.5:1 gear reduction on each side. The front two corners contained in-hub 6:1 single stage planetary gearsets with a rotating ring gear coupled to a Nova 15 motor. These front corners also required custom inverted brake calipers due to packaging constraints. Extensive documentation of this system was published following the 2017 FSAE Electric season and can be found on our website (wisconsinracing.org). This paper will mention these older systems occasionally in justifying certain design changes that better accomplish the team's goals. The primary purpose of this paper is to provide a high-level overview of the drivetrain system designed for the WR-218e. While it touches on some of the custom parts and selection of the stock items also featured within the assembly, there will be no focus on in-depth Finite Element Analysis of most of the structures. Some prior knowledge of a planetary transmission is also assumed throughout this paper, and it is recommended to look back at the paper written on the 217e front corner drivetrain system for more insight into the basic understandings assumed in this paper. While Wisconsin Racing tries to cover as much as possible within our documentation, we may not explain everything to the point where it can be universally understood but we encourage you to utilize any and all resources available to you; most of what our team has learned was through self-lead teaching from reading books, papers, and countless Google searches. Also feel free to reach out to our team if you have specific clarifying questions at eformula@go.uwracing.com.

Drivetrain Architecture Selection

After our first year designing the 217e, we took a critical look at our car and used what we had learned to identify the areas where the most sizable improvements could be made. The Plettenberg motors and inverters that the team selected for the 217e did not meet the performance targets throughout our testing; this in turn heavily impeded the overall performance of the car. Put simply, the inverters completely lacked any safety features (or even current control, which we had to implement externally) and we were unable to reproduce the performance claimed on the datasheet for the motors.

After thoroughly discussing what we wanted the future of our team to be and looking at what some of the most prestigious Formula E teams were doing around the world, we decided on an architecture that could be utilized for years to come: four outboard or 'in-hub' drivetrain assemblies. Given the talent on the team, we set out to design our own motors and inverters to pull this off. A gearset was to be designed to transmit the power from a custom 30 kW high-speed permanent magnet motor inside each wheel. The motors and inverters will each have their own dedicated document on our website, so we will not be diving any further into those components here. This all in-hub configuration decreases the overall weight of the drivetrain by eliminating many hefty parts from the previous rear drivetrain package, such as half-shafts, CV joints, and the gearbox housing itself (in this architecture, the upright doubles as the gearbox housing). Additionally, by moving the rear drivetrain out of the rear space frame of the car, the new design significantly reduced the complexity of the packaging within the rear of the car, vastly improving electronic component accessibility and maintainability.

However, this configuration does result in additional unsprung mass within each wheel. This can make it more challenging for the suspension system to maintain consistent load on the tires during transient road inputs and may require stiffer springs and additional damping. That being said, with intelligent suspension design combined with the increased control of each of the wheels through torque vectoring and traction control, we have so far seen very little negative net effect on the car. Torque vectoring, in short, is the ability to request a torque at each wheel independently of the other three. This added advantage allows the car to increase its yaw acceleration during cornering events by moving torque from the inside track to the outside track during turn-in, and vice versa during turn-out. There is a supplementary document from 217e that further elaborates on the advantages of this control strategy and how we utilize it on our car.

Lastly, by moving to four near-identical in-hub assemblies, the overall complexity of the vehicle is decreased, mostly due to the reduction in the number of unique custom parts.

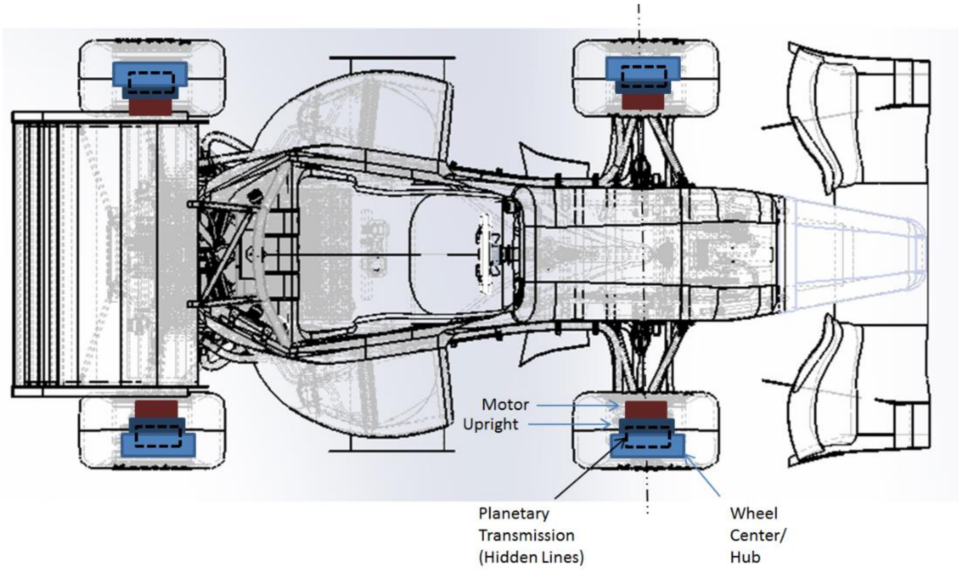

Figure 1: 218e Drivetrain Architecture

Determining Performance Goals

It is important to identify and quantify performance goals prior to designing specific parts within a system. By quantifying our goals prior to in-depth design, we ensure that we are making all design decisions in a way that is justified by these high-level goals. To set our performance goals, we used a quasi-steady lap simulator that is updated and improved upon each year. The quasi-steady lap simulator is elaborated on in more depth in another paper. Based on an acceleration event simulation, we were able to determine the maximum wheel torque that the tires could transmit until the rules-imposed 80 kW battery power limit comes into effect, as well as the power at each wheel when operating at this limit. Ultimately, by designing the motors to be able to produce this torque and power, we would never be limited by the motors at any point during the competition; we would only be limited by the tires, the rules-imposed power limit, or driver ability. With a target tire torque curve set, we needed to determine the ideal gear ratio necessary to meet that requirement. Here, it is important to understand the behavior at the extremes of gear ratios in order to understand how to select the optimum. If a direct drive motor were implemented, all of the wheel torque would be supplied directly by the motor. For a motor to produce this much torque, it would be massive- an absolute unit. This is because motor mass scales directly with torque, not power. At the other extreme, having an extremely high gear reduction would require very little torque from the motor, (and very high motor speed to achieve the target power) but the resulting transmission would have to be massive to achieve such a high reduction. Therefore, when one has the ability to design the power source characteristics as well as the gear set (as is the situation here), the optimal gear ratio will be the one that achieves the minimum total system mass. This optimum can be determined either iteratively or through some idealized scaling laws for electric motors and gear sets. With this in mind, we selected a target gear ratio of 13.8:1. This ratio minimized the predicted overall in-hub assembly volume while still meeting all other requirements set previously.

Assembly volume is extremely important in this application, since the small space available inside of our small 10” wheel shells must house the electric motor, transmission assembly, and brake rotors and calipers, all while maintaining the necessary clearances between the moving components. Using previous models of wheel shells, brake calipers, and estimating the thickness of the housing to be at least 1/8th of an inch, we determined the maximum outer diameter (OD) of the entire planetary transmission to be 4.25 inch. With a target gear ratio set and maximum packaging volume determined for the planetary, we could then move onto designing the individual components. It should be mentioned here that the design is built to satisfy the requirements of the rear tires. By using identical corner assemblies in the front and rear, we have “overbuilt” the front. This design choice was driven by decreasing the complexity and increasing consistency throughout the vehicle.

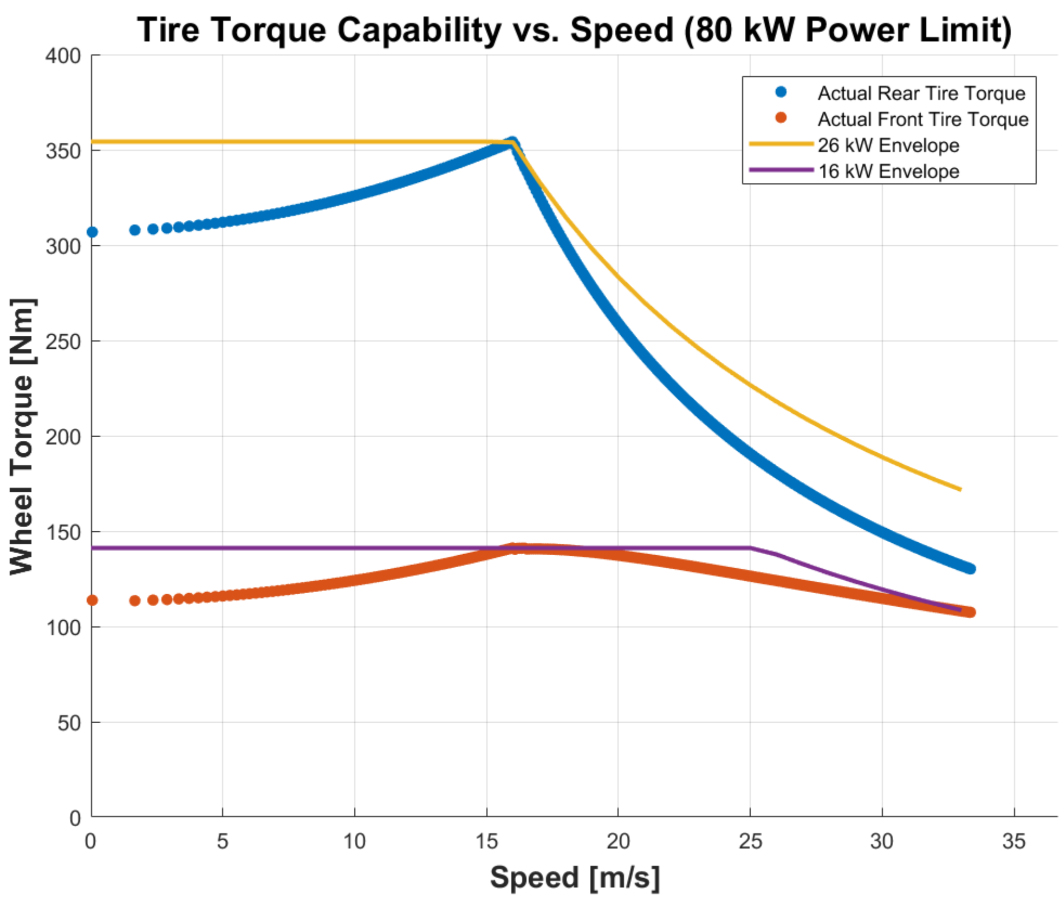

Figure 2: Wheel torque vs recommended power envelopes

Compound Planetary Transmission

A gear ratio of 13.8:1 is not easily attainable within a small volume. A planetary gearset is the best way to package the transmission within the wheel, since any other configuration would be too large to achieve such a high gear reduction. Since this new design required over twice the gear ratio compared to the 217e front corners and a decreased maximum OD, it was clear that a single stage planetary reduction would not be sufficient due to the large diameter that would result. The smaller maximum OD for the 218e transmission is due in part to the decision to use AP calipers in a more traditional brake location. Regarding the brake system, we decided to switch to AP Calipers for the 218e; these calipers have been used in the rear of the car on the 217e as well as tested for many years on the combustion car. Although building our own brakes was a fantastic learning experience and an elegant solution to the packaging problems, we had with the 217e corners, we ran into difficulties getting the brake properly bled and the calipers tended to conduct too much heat into our corner assembly. At one point, this even caused the press fit on the ring gear to slip due to the differing thermal coefficients of expansion in the aluminum housing and the steel gears. That being said, the axial space available tripled as a result of the changed brake system, resulting in an overall 50% increase in available volume for the transmission. The only way to achieve over twice the gear ratio of the 217e without increasing the OD was to add a compound planetary stage.

Figure 3: 217e Planetary

Figure 4: 218e Compound Planetary

Single Stage Planetary Gear Reduction Calculation (217e -> Ring Driven):

$$\text{Gear Ratio: } \frac{N_{ring}}{N_{sun}}$$

Compound Planetary Gear Reduction Calculation (218e -> Planet Driven):

$$\text{Gear Ratio: } \left(\frac{N_{big\ planet}}{N_{sun}} * \frac{N_{ring}}{N_{small\ planet}}\right) + 1$$

*** N_{gear} - number of teeth of that gear*

The largest available gear ratio increases considerably with a compound stage added. Planetary transmissions can either be “ring-driven” or “planet-driven”. When a system is ring-driven, the planets of the system are fixed in place while the ring gear rotates, providing the output torque. The ring gear is generally fixed to a housing that also rotates along with it; this was the set-up we had in the 217e corner assembly. The opposite is true in the case of the planet-driven system; the ring gear and the housing it is pressed into both do not rotate, and instead the planet gears walk along the ring gear. In a ring driven system, the sun torque is mechanically transmitted through the planet gears that then turn the ring gear. In the planet-driven system, sun torque is transmitted to the planets and then to the ring. However, since the ring is fixed, the planet carrier rotates as a result. Additionally, the input torque from the sun gear also has a contribution to the total torque on the planet carrier. Therefore, the input torque from the sun gear in a planet driven system is added to the reaction torque of the ring and the sum of the two is applied to the planet carrier (hence the “+1” term in the equation). Thus, since we designed a planet drive system, the plus one is included in the compound planetary transmission calculation above, but this is not the case for all planetary gear boxes.

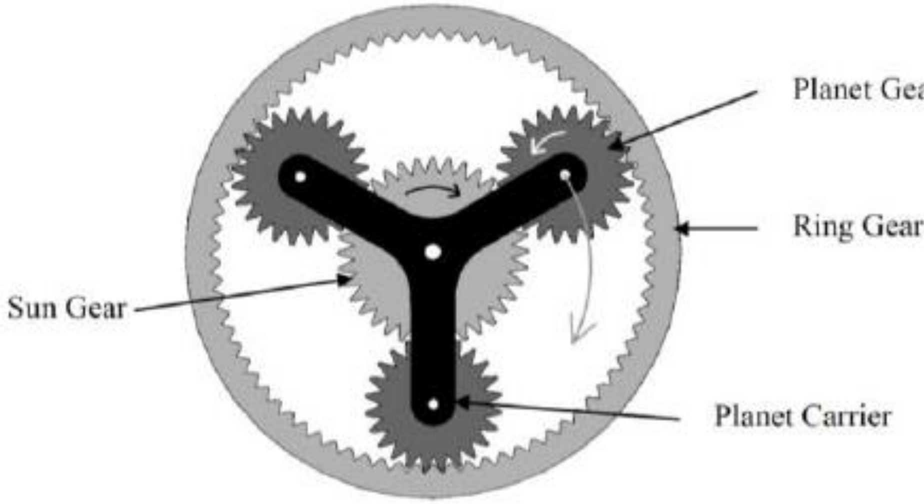

Figure 5: Dynamics of a planet driven system. The sun gear is driving, ring gear is fixed, and the planet carrier is the output that rotates [1].

Our motivation to have a planet driven system was based on a couple of reasons. First, the gain in gear reduction within the same volume made it easier for us to achieve the same gear ratio while maintaining a smaller overall system OD. Second, this allowed us to use a wheel center that has been used by our combustion team for many years with only small modifications to the diameter of the bolt hole pattern. The wheel center was designed using a mechanical optimizer a few years prior and has been tested and proven. This adds to the simplification and confidence we have in the new system and is one less part to design specifically for this application.

Gear Calculations

Gear calculations were presented in depth within the 217e corner assembly document and the same governing equations were used in designing the gears for the 218e transmission. Because of this, the equations and gear factors will not be covered again in this document, but we encourage the 217e document to be used as reference for any questions.

There are many software options available that are dedicated to calculating the stresses present during operation in each gear tooth, with one of the most well recognized options being KissSoft. While these tools are very powerful and can generate many graphs and other visuals displaying how the stresses are propagating through the gears, for the 218e we decided to focus on creating our own gear stress calculator. Our goal with this was to incorporate important geometric constraints of a compound planetary transmission with the first principle equations from our 217e document. By designing this tool in-house, it would force us to better understand the underlying principles in gear design.

The tool we created is a factorial design of experiments (DOE) in MATLAB, which we affectionately call “transmissionMagic”.

transmissionMagic

The 217e gear calculations were implemented in a parametrized Excel spreadsheet that updated gear safety factors as a function of input parameters such as number of gear teeth, diametral pitch, pressure angle, input torque & speed, and many more. This was and still is a powerful visual tool, allowing us to see how changes affected the strength of the gears and furthermore draw concise correlations. Along with that, many diagrams and other visual aids are attached to the spreadsheet for ease of referencing. This spreadsheet, however, was not built for optimization of a transmission and the gears; it is most useful to input the final parameters once a design has been chosen so that the user can clearly show others the final safety factors and stresses without getting too lost in calculations. On the other hand, with MATLAB we were focused on optimization. With the knowledge acquired from the 217e, we

built a script that allowed us to consider 616,853,160 possible planetary transmission configurations in mere seconds. Furthermore, it was built to allow the user to dictate a wide variety of other constraints (such as geometry) on the system. This tool narrowed down the millions of combinations to about 30 designs that fit within our constraints, which we could then analyze in-depth and hand-select the final configuration.

The approach strategy for our script was to generate a factorial DOE or a series of nested 'for' loops that cycle through every possible combination of gears and eliminate the infeasible and erroneous designs. Inputs to transmissionMagic were:

1) Minimum safety factors for yielding, bending fatigue, and wear fatigue
2) Desired gear ratio and a tolerance band
3) Number of planet gears
4) Desired lifetime of the gears based off number of endurances ran
5) Maximum Ring OD
6) Maximum Big Planet OD (the diameter of a circle tangent to the big planet gears)
7) Maximum width of combined gear faces
8) Minimum bore size for the planet gears (allow space for planet axle and bearing)
9) Ranges of gear teeth for each gear
10) List of standard diametral pitches
11) List of standard pressure angles
12) Gear material properties
13) AGMA quality number from our manufacturer

Before beginning the DOE, the script generates a collection of look-up tables that it will utilize to calculate factors such as the bending and pitting resistance factor, the size factor, and the Lewis form factor. As the script cycles through each configuration, after each calculation it does systematic checks against governing bounds on the assembly and throws out infeasible combinations of gears. The systematic checks decrease the run time of the code by stopping the current iteration before it does the work to calculate everything else specified if the configuration fails a previous check. As the script cycles through, it makes crucial checks to ensure proper strength and correctly meshing gears. The script mathematically calculates if there is any chance of involute interference in the gear mesh and eliminates the configuration if there is a need for undercutting the gear teeth, which would decrease the strength. There are also 3 geometric constraint related to the number of gear teeth to check that the compound planetary will mesh properly, and that the planets will not interfere:

$$1)\quad PD_{ring} = PD_{sun} + PD_{big\ planet} + PD_{small\ planet}$$

PD – Pitch Diameter; if diametral pitch is the same for the first and second mesh, the equation can be simplified from pitch diameter to the number of teeth on each gear.

$$2)\quad \frac{N_{ring}+N_{sun}}{No.\ of\ Planets} = Integer$$

Constraint two should be utilized if there is the ability to uniquely change the angle for each compound planet in the assembly. Otherwise, our team uses a modified version of the second constraint that ensures both the sun gear teeth and the ring gear teeth are each individually multiples of the number of planets. This yields three identical compound planets and significantly decreases the difficulty of manufacturing.

$$3)\quad N_{big\ planet} + 2 < (N_{sun} + N_{big\ planet}) * \sin\left(\frac{180}{No.\ of\ Planets}\right)$$

Additionally, it checks that each gear has 1.2x the gear tooth height of material support underneath the teeth to ensure full strength in the gear tooth. And finally, we also have the script do a check for a hunting ratio. The hunting ratio in a transmission improves the life of the gears by providing ideal wear patterns and preventing a galled surface on one of the gears to cause a cascading failure. A hunting ratio occurs when the greatest common denominator between the number of teeth on two mating gears is equal to one. This check means that each tooth on one gear with mesh with every tooth on the mating gear before returning to the first.

When we first ran the script, we were getting very low safety factors on all configurations, many below one. The highest concerns were with the sun gear and small planet gear, in both bending and wear stress. After consulting with industry professionals and professors, it became clear that the calculations were being implemented correctly, but the gear box input parameters for speed and torque for our application were incorrect. By inputting only one speed and torque, the gears were being simulated as running at that load for the entire lifetime. This, however, is drastically different than the reality of the life of the gears due to the widely varying duty cycle in traction applications like this one. Initially, we had input the peak speed and torque of our motor, an operating point that in reality is unachievable simultaneously. While using maximum speed and torque was the most conservative approach, we decided to go in a different direction in the spirit of truly optimizing the transmission. Our solution was to feed a histogram of torques and speeds during an autocross lap into the script to more accurately calculate the stresses due to the cyclic loading of the gears over their lifetimes. Using Miner's Law, we were able to calculate the percentage of life consumed for each of the gears at each different operating point in the histogram.

Miner's Law involves first calculating the maximum number of cycles a gear could withstand at a given stress level (torque/speed point), then using the number of cycles the gear will actually see at that stress level from our histogram to calculate the percentage of total life of the gear that is consumed at that stress level. This process is done for each stress level the gear encounters throughout its life (each point on the histogram) and the final percentage of the remaining gear life can be calculated, which is then used to find the safety factor for each gear.

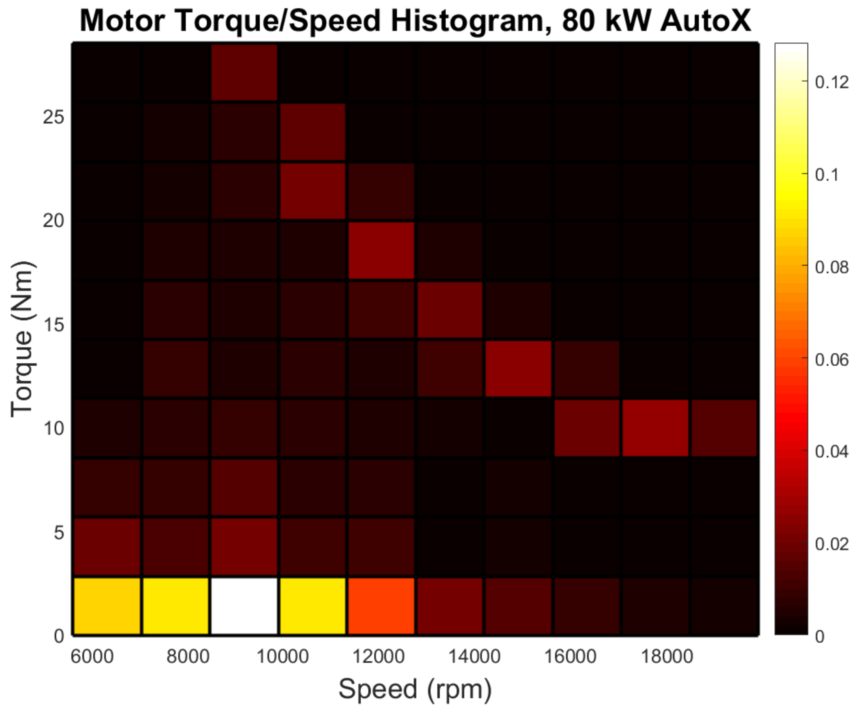

Figure 6: Torque vs Speed frequency during autocross lap

In the end, the script is set to output the yielding, bending, and wear stresses of each gear as well as their corresponding percent life consumed for bending and wear fatigue. It also outputs the high-level system parameters such as the number of teeth on each gear, the pressure angle and diametral pitch of each stage, and the gear ratio of the combination. From there we can dissect each configuration and find the highest average safety factors or look for combinations that have specific safety factors that are more desirable than others.

The final output of our compound planetary system is a 13.67:1 gear ratio. From this gear reduction, we can take the max speed out of the motor of 20,000 rpm and reduce that to a top speed of ~1500 rpm (75 miles per hour) at the wheel. This planetary also allows us to have 300 ft-lbs of torque at each wheel, allowing us to have the capacity to slip the wheel at any point up to the power limit of the car. With four of the assemblies on the car, the drivetrain is capable of 160 HP (when unlimited).

Our final configuration*

$N_{sun} = 12$; $N_{big\ planet} = 49$; $N_{small\ planet} = 29$; $N_{ring} = 90$

$$GR = \left(\frac{49}{12} * \frac{90}{29}\right) + 1 = 13.67$$

** - notice that the greatest common denominator of every mesh = 1 (Hunting ratio)*

The assembly features a sun gear that is as small as possible (N = 12 at 25 deg pressure angle) before undercutting of the gear tooth is needed to properly mesh. The big planet gears are as large as possible before they would interfere with each other. This combination maximizes the gear reduction in the first stage and allows for larger small planet gears in the second stage. This is important because it maximizes the space available to fit the planet axles and needle bearings inside the planet gears.

$$N_{min} = \frac{2}{\sin(Pressure\ Angle)^2} = \frac{2}{\sin(25)^2} = 11.1978$$

The planet axle stiffness is proportional to its diameter to the fourth power, so leaving a lot of room inside the gear goes a long way in keeping the assembly rigid. The gears are made from 4140 prehard steel which goes through a gas nitriding process to harden the surface to 55 Rc at a 0.015” case depth. Finally, the gears are sent through an acid tumble drip to eliminate the most brittle surface impurities which cause stress concentrations on the gear tooth.

There are nearly unlimited combinations of gears that could provide the same gear ratio from motor to wheel. The real difficulty with this application is the packaging and weight optimization. With the MATLAB code constrained heavily along different degrees of freedom, we can create this gear reduction in one of the most ‘torque dense’ drivetrain packages. The full assembly, including the electric motor, upright, and gears weighs only slightly more than 12lbs. This eliminated 30+ lbs. from our total drivetrain weight featured on the 217e and is 6 lbs. lighter than the 217e corner assembly.

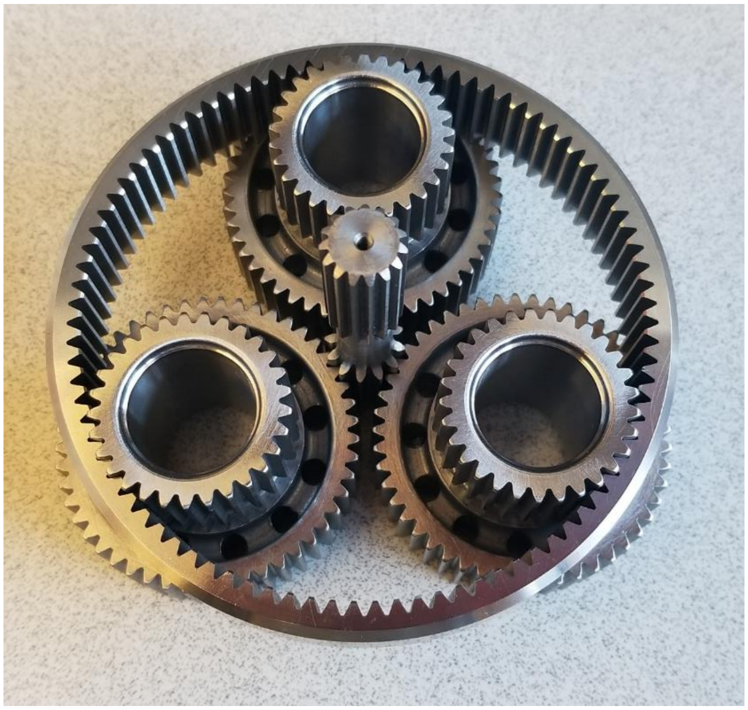

Figure 7: 218e planetary final assembly

Bearings

Thin Section Wheel Bearings

Similar to the 217e corner assembly, we package two thin section angular contact bearings into each of the corner assemblies. The angular contact bearings serve a dual purpose, first, to allow for continuous smooth rotation of the assembly, and second, to react the wheel forces experienced during cornering. The corner wheel forces are calculated during each point along an autocross lap in our student-developed lap simulator, and the forces (and resulting moments) are used to set the requirements for the wheel bearings. The selection of the wheel bearings for us are thin section roller bearings from Silverthin. Angular contact bearings can react axial force in only one direction; we feature two within each assembly to react the bidirectional wheel forces during the different cornering events we see. The wheel bearings are spaced approximately 2.5 inches apart (centerline to centerline), a major improvement from our 217e angular contact bearings which were spaced only a 0.5 inch apart. The advantage of separating the wheel bearings is to allow for better reaction of the moments (specifically the overturning moment and aligning moments from the tire). The increased spacing of the bearings reduces the overall load each bearing must react. We also chose a back-to-back pairing configuration for our angular contact bearings because of the increased resistance moment arm in comparison to a face-to-face configuration. We were able to send a time history of Fx, Fy, and Fz forces to Silverthin (simulated over an autocross lap in our lap simulator) and they used those forces to predict the lifetime under the specified operating conditions.

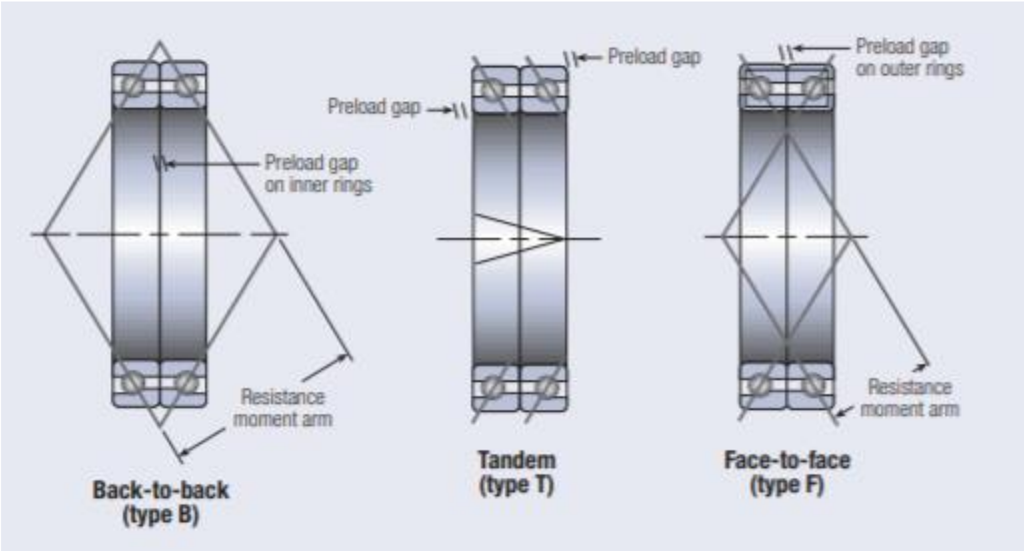

Figure 10: Paired angular contact bearing configurations

Needle Roller Bearings

The second type of bearing selected for the assembly is a needle roller bearing. These bearings are pressed into the inner bore of the compound planet gears and ride along the planet axles. The selection of needle roller bearings over the numerous other types was due to their ability to react high radial loads in a very small package and roll with very minimal resistance at a high speed; they also allow for easy assembly within the gearbox in case a planet gear had to be replaced or removed to service another part of the corner assembly. We were able to calculate the maximum amount of radial force on the needle bearings due to the big planet and small planet meshes to ensure the bearings could handle the radial loads in both dynamic and static situations. Our planet gears use two of these bearings because there was not a face width large enough to satisfy the entire compound planet gear.

As mentioned, we factored a constraint on the planet gears into our transmissionMagic code such that the gears could be machined with the proper bore diameter to fit needle roller bearings inside while still maintaining proper material backing of the gear tooth. Our final selection was the NK14/20 needle roller bearing. Also, our gears utilize a larger pressure angle of 25-degrees because of the increased tooth strength, with the downside of having higher radial loads. We ensured that our selection of bearings could react these high radial loads. For example, increasing the pressure angle from 20-degrees to 25-degrees results in ~23.5% higher radial forces.

SKF Needle Roller Bearing NK14/20	
Inner Diameter	14 mm
Outer Diameter	22 mm
Face Width	20 mm
Dynamic Load Rating	12.8 kN
Static Load Rating	16.6 kN
Maximum Speed	28,000 RPM

Figure 11: SKF Needle Roller Bearing

Figure 12: Needle roller bearings pressed into planet gear

Planet Axles

The planet gears rotate in two different axes; they rotate both locally around a planet axle, and globally with the planet carrier as they walk along the fixed ring gear. The planet axles many not be the most complex part in the assembly, but they are heavily constrained in shape, size, and must meet difficult requirements in strength, stiffness, hardness, and precision. In that way, they are the components that are the most sensitive to manufacturing and treatment processes. The responsibilities of the planet axle within the assembly include:

1) Locating the planet gears relative to the sun and ring gears
2) Allowing for smooth travel of each of the planet gears incorporating a needle bearing raceway surface
3) Serving as the wheel studs
4) Transferring the entire torque from the gears to the planet carriers, and ultimately to the wheel center

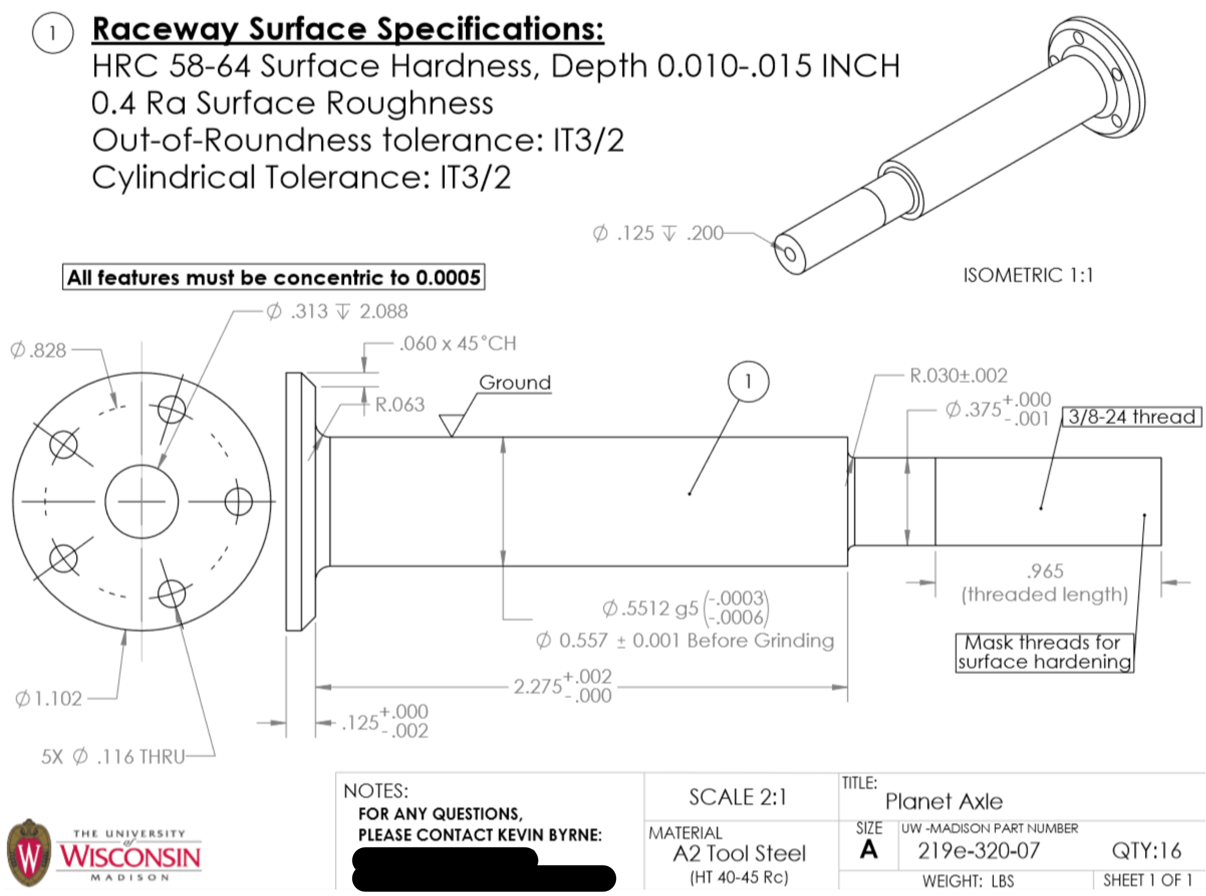

Figure 13: Planet axle final drawing

With these requirements, it is not difficult to see that they are a crucial part within the assembly. Due to the multiple responsibilities of the planet axles, they have very specific material and manufacturing requirements. To transmit the gear forces, the planet axle needs to have a very high core strength, while also requiring a very high surface hardness and straightness on the needle bearing raceway surface. Additionally, it cannot be brittle due to the threads on the wheel stud portion of the part. To meet all these requirements, we started with annealed A2 tool steel round stock, which can be heat treated to a very high strength. We started by through-hardening the stock to 40-45 Rc. Then, using carbide tooling, the initial outer profile and the external threads are cut on a lathe. Following that, the planet axle is gas-nitrided to bring the surface to 58-64 Rc at a case depth of 0.015”. We masked the threads for this process to ensure that they did not become too brittle. Finally, we had the raceway surface of the planet axle ground to within three ten thousandths of an inch with a surface finish of 0.4 Ra (basically a mirror) to meet the proper spec for the needle roller bearing. In the end, after through-hardening, turning, gas nitride surface hardening, and grinding, the planet axles bring the entire system together and hold an unparalleled duty to the assembly operating properly.

These parts would not have been feasible without our generous and skilled sponsors. The parts were machined and finished to perfection and our team could not have been happier with the finished product. A huge thank you to Revolutionary Machine Design for the machining (at 45 Rc, no less!) and T&L Grinding for hitting our crucial tolerance.

Figure 14: Planet Axle

Figure 15: Compound planet and planet axle

Teflon Spacers

The optimization done in the transmissionMagic code is useless if we do not ensure that the gears mesh along the full-face width of the gear tooth. Our final assembly design has a face width of 0.75” for each mesh. The big and small planet gear faces both have this face width, but our sun gear and ring gear are slightly oversized. This strategy is one of the ways we ensure full face width mesh. The other way is through use of Teflon spacers. These spacers are placed on the planet axles after the axles are inserted into the planet carrier. There is one on each side of the compound planet gear. These correctly space out the compound planet gears along the axle so that they mesh with their respective mating gear face. The Teflon spacers also serve as an axial retention tool in the assembly because the needle roller bearings do not prevent axial movement.

The Teflon material was chosen for this application due to its self-lubricating qualities, low coefficient of friction, and its high melting point. The spacers are pressed against the rotating compound planet gears; therefore, they must not provide additional resistance or be susceptible to heat due to friction.

During a teardown of one of the assemblies we were able to inspect the gears. Much to our delight we were able to see proof that the stages were meshing as we had hoped, utilizing the entire face width. The oil patterns on the sun gear clearly show proper meshing between the gears.

Figure 16: Gear oil pattern on sun gear

Assembly

A key focus in the design of the 218e assembly was to decrease the amount of time it takes to service each corner. Although we never intended to have to service either of the assemblies on the 217e, the in-hub assemblies had to be opened multiple times due to a poor oiling method and to diagnose resistance issues. The 217e in-hub assembly took approximately an hour to remove the brakes and sealed cover to look at the planetary transmission. Additionally, installing the oil ring again to properly seal the assembly could add up to another hour and we would often damage the oil ring in trying to reinstall it. Beyond that, due to the method of mounting the gears onto the upright, the gears could not be easily removed or replaced. The overall assembly was essentially permanent after initial assembly, and the only removeable parts were the brake calipers, brake rotors, and brake caliper mounting plate. The remaining disassembly would involve removing press fit bearings from the mating surfaces. These problems resulted in many hours of tedious time disassembling and reassembling the corner with very little opportunity to adjust or interchange damaged parts if problems did occur. All of these factors added up to a combined effort to make the 218e corner assemblies more easily accessible and serviceable.

Designing for Manufacturability and Serviceability

As we designed the 218e in-hub assembly, we addressed each of the problems that the 217e in hub assemblies had directly. The main problems we wanted to address were the lack of interchangeable parts, difficulty taking the wheel off each corner, and a lack of access to view the assembly to fully inspect each part.

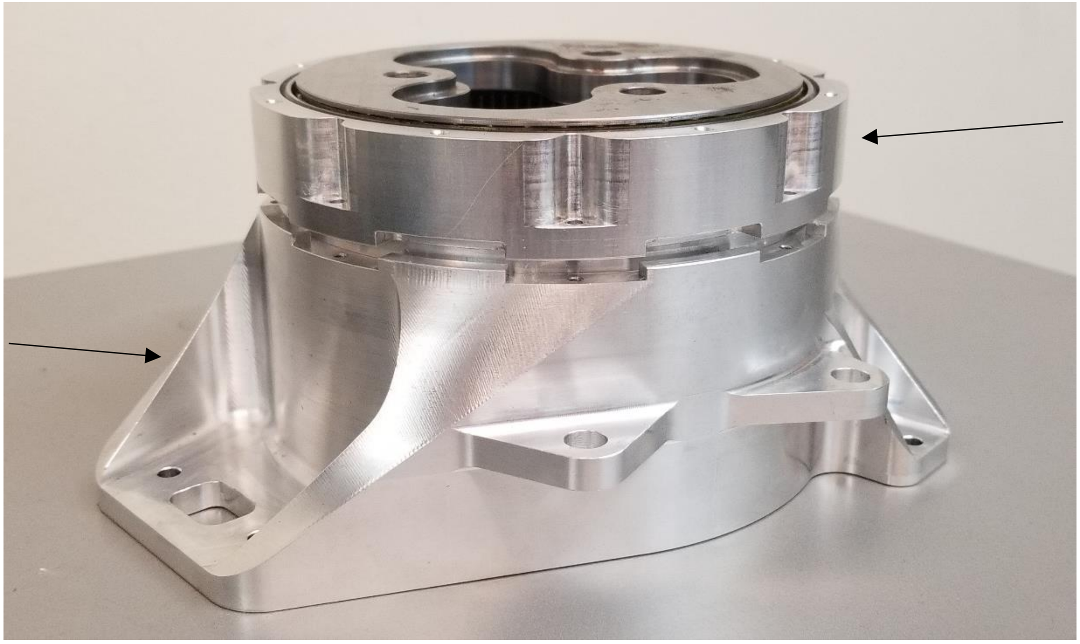

Figure 17: Upright halves mating interface (castle features mesh upon full assembly)

The 218e corner assembly features a split upright, with four unique inner halves to match the unique handedness of the suspension in the corners of the vehicle and a universal outer half. Each of the upright halves has one of the wheel bearings and a planet carrier. Each wheel bearing reacts axial forces in a singular direction, thus by facing them away from each other we ensure that the assembly can react the forces experienced in all cornering situations. By dividing the upright into two pieces and placing a wheel bearing and planet carrier in each, it allows us to easily split the two at the interface and have more opportunity to service and inspect the entire assembly thoroughly.

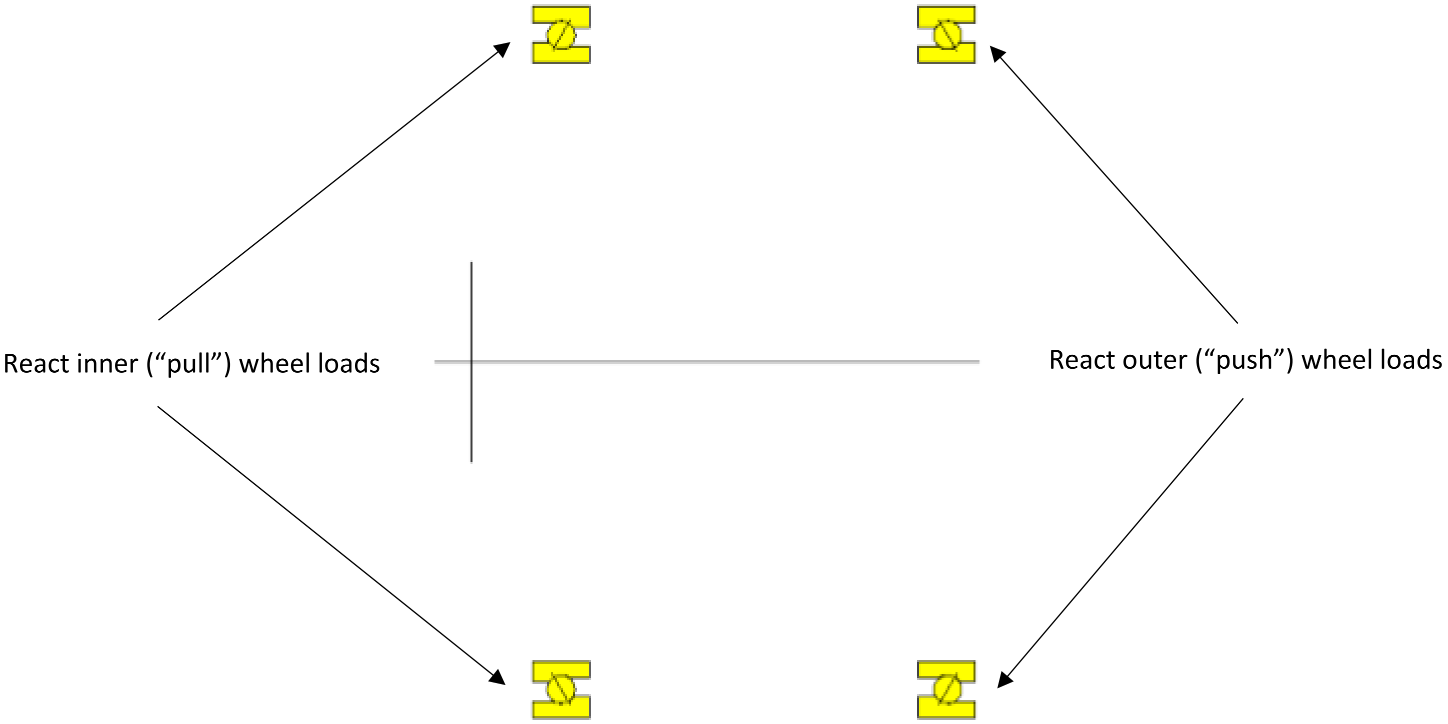

Figure 18: Back-to-back wheel bearing configuration (right side of car)

By being able to split the upright in half, it provides a better opportunity to interchange parts including bearings, gears, planet axles, and spacers. This change presented the opportunity to replace damaged parts and provides the team the ability to continue testing or competing even after a part is damaged. While ideally the design never should have to be disassembled it is advantageous to be able to easily inspect gears, add new oil, and remove the motor.

Figure 19: 217e corner assembly (9 attachment points)

Figure 20: 218e corner assembly (3 attachment points)

To address the difficulty in taking off the wheel, we changed the set-up from nine attachment points to the standard wheel center we have used on our combustion car. The wheel center has three attachment points, with each lug concentric with an individual planet gear. With a standard wheel center, we were able to decrease the time to remove and attach the wheel from at least five minutes to under a minute. This small change saves a minimal amount of time in the overall scope of assembly, but the ease is felt significantly throughout the life of the assembly. In this way, the assembly was designed so that the wheel studs were integrated into the same part as the axle on which each planet rotates.

Figure 21: Assembly of the inner half of the upright

Finally, we worked to increase the access of view to the assembly and inspect each part. As was mentioned previously, the division of the upright provides a significant improvement to the overall assembly and allows for more accessibility to the gears, bearings, and uprights. The division creates an opportunity to inspect them individually with precision. To further add to the ability to inspect the assembly, we added a small plexiglass window. We had not planned to include the window in the assembly from the beginning; it was a bonus from trying to eliminate weight and inertia from the rotating assembly. The plexiglass window provides a unique view into the assembly and helps many new team members and other viewers to get a better insight into the motion of the assembly. For the team, it provides the opportunity to inspect the gears and oil accumulation at the bottom of the assembly, which helps us in identifying the proper amount to oil the assembly. The aesthetic factor that it adds cannot be overlooked, providing a unique look for any viewer.

Figure 22: Side view of final corner assembly (without motor) mounted on the car

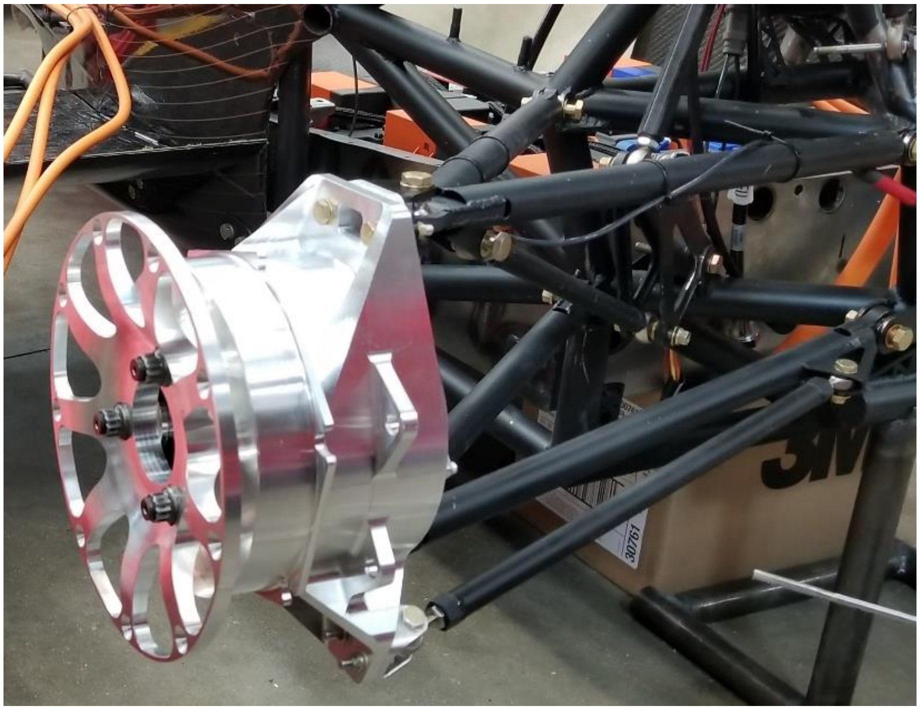

Figure 23: Rear left final corner assembly (without motor) mounted on the car

Setting Bearing Preload

Setting bearing preload in the wheel bearings was one of the more complex problems that we explored in this assembly. Wheel bearing preload is set by the axial distance between the two angular contact bearings, and the assembly contains 12 custom parts that each contribute a machining tolerance to that crucial distance. This tolerance stack-up could vary as much as .016 across the assembly based on the tolerances we set. Preloading the angular contact bearings is an important factor in determining the total life of the bearings. In angular contact bearings, preload refers to the axial clearance or interference (as opposed to radial clearance/interference, as is the case in deep groove radial ball bearings). Our wheel bearing sponsor, Silverthin, ran simulations at different bearing preloads allowing us to look at the life of the bearing at different clearance and interference fits. There are three different parameters that determine the final fit of the bearing: the ID and OD fit (which will effectively change the contact angle of the bearing), and the axial fit. The OD and ID fit of the bearings were addressed in the machining of our uprights and planet carriers, with a tolerance band specified through Silverthin. The bore diameter mating with the OD of the bearing had a tolerance band of +0.000” - +0.0006”, while the shaft diameter mating with the ID had a tolerance band of +0.000” - +0.0008”. After our bearing sponsor ran simulations on the axial fit, we determined that the ideal for our system was about 0.001” [0.025mm] of axial clearance. This was selected based on the practical limit of axial resolution using shims (0.001” shims break if you look at them wrong!). Although bearing life is maximized with slight preload (as seen in Figure 26), it is safer to end up on the clearance side of the optimum due to the shallower slope of lifetime with respect to clearance.

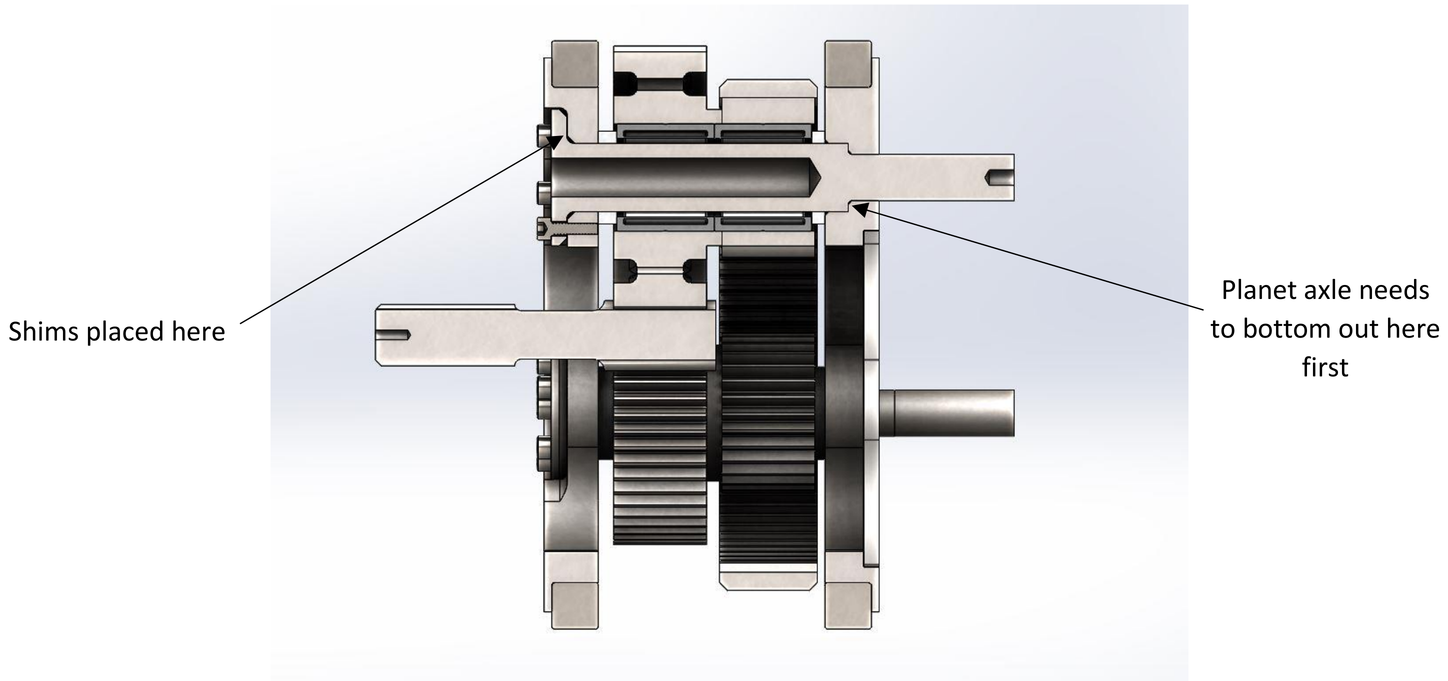

Figure 24: Section view of transmission assembly

After debating many solutions on how to properly set the axial preload, we landed upon an efficient and simple way to set the preload, using the wheel lug torque. Although the solution may seem obvious, it wasn't easy to carry out. Due to tolerance stack up across the entire assembly, we could not be certain all three of the planet axles would bottom out on the outer planet carrier at the same time and allow for the ideal preload across the system. We addressed this by appropriately tolerancing each part in the system so that the stack up only could add space between the inner planet carrier and the base of the planet axle. This space would then be filled with custom made .002' shims to adjust with fairly high resolution the final preload that the system would have. In the end, this method decreases the overall assembly time and allows for quicker serviceability to the system by eliminating other clamping devices that may have been needed to set the preload for the assembly. The key point here is that torqueing on the wheel lugs also sets the axial clearance for the wheel bearings- a clever and convenient way to maximize the utility of each part in the system, especially since the wheel lugs already have a very specific torque spec that is always checked upon assembly anyway. For the sake of clarity: each planet axle is shimmed such that when the wheel lug is tightened onto the planet axle at the specified torque spec, the correct axial clearance remains between the pair of angular contact bearings.

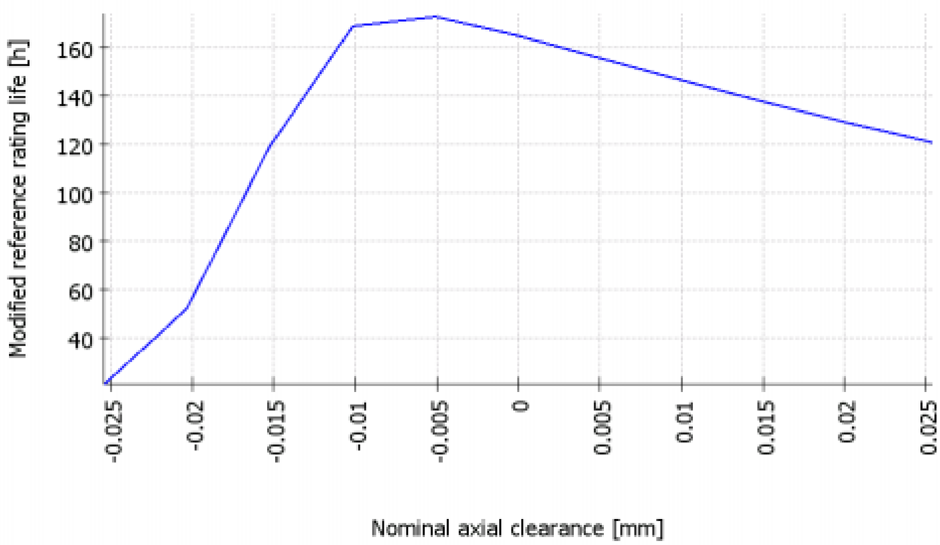

Figure 25: Bearing life vs axial preload

Final Assembly Procedure

Decreasing final assembly time was a major goal for the design. Although ideally assembly only should have to happen once, reality never follows that rule. Making an easy to tear down and rebuild part is often a second thought in the design process but after a large inability to service or change most of the 217e assembly after its initial assembly, it ranked high on our priority list. The final assembly steps consist of:

1) Shrink fitting the ring gear into the outer upright half (add dowel pins to the interface to prevent ring slippage)

Figure 26: Outer upright half with ring gear after shrink fitting

2) Shrink fitting the wheel bearings onto the inner and outer planet carriers
3) Shrink fitting the assembled wheel bearing & planet carriers into the inner and outer halves of the upright respectively

Inner Half Assembly - https://youtu.be/v0b-k1F97t8
Outer Half Assembly - https://youtu.be/ZJclfmtn1iQ

4) Inserting the axles and shimming the planet axles as needed with measurement to ensure all three axles were at same height (*see note)
5) Pressing the needle roller bearings into the compound planets
6) Sliding the Teflon spacers and compound planets onto the planet axles (ensuring each compound planet gear has a small bit of axial play)

Figure 27: Aligned planet axles with compound planets and Teflon spacers

Inner Half Assembly with Planet Axles - https://youtu.be/k00o1WbHKRI

7) Mating the inner and outer halves of the upright & meshing the compound planets with the ring gear teeth
8) Attaching the motor to the motor adapter plate and mounting that entire assembly to the upright
9) Inserting the sun gear, meshing the spline with the motor shaft and the sun gear with the big planet gears
10) Adding the plexiglass window, brake hat & rotor
11) Mounting the brake caliper onto the upright
12) Mounting the wheel & torqueing down the bolts

* - in reality the shimming process is very difficult to correctly measure and preload the wheel bearings the first time, it is an iterative process to check the number of shims and align the planet axles with checks each time after torqueing down the bolts until there is little to no resistance felt through the rotating assembly.

Final Assembly (without motor) - https://youtu.be/Xez7RXNWcZs

Conclusion

The design process for the 218e assembly began during the summer before the 217e had even competed. All parts were finalized by January of the following year, a 7-month endeavor from start to finish, with over half of the time during the school year when class consumes a large portion of time as well. The 218e corner assembly took everything that we learned from the 217e and improved upon it, meeting our design goals and bringing to life an elegant powerhouse of a transmission. The 218e corner assemblies paired with our custom motors helped propel our team to a second straight 1st place in design at the 2018 Lincoln Formula SAE Electric competition. As with the 217e corner assemblies we learned many new things along the way and plan to make small iterative changes to further improve upon the design. The most major point to address is containing the oil within the assembly; the unsealed thin section wheel bearings allowed for oil to escape through the bearing and sometimes leak onto the brake rotor. We have plans to work with Silverthin this year to create a custom sealed version of the same bearing to use for the 219e. Beyond that, with a brand-new design for the 218e there was not ample time to weight-optimize the assembly. There will be iterative steps taken, looking at several parts within the assembly to eliminate weight and inertia from the system. Please feel free to reach out to byrne@go.uwracing.com with questions, clarifications, or concerns about any of the information shared in this document. I hope that this document will help other teams like Wisconsin Racing develop their own in-hub assemblies and provide useful insight into the design process. On Wisconsin!

Motor Driven Final Assembly - https://youtu.be/jKpWo-oAt4c

Sources

[1] Smith, Wade & Deshpande, L & Randall, R.B. & Li, Huaizhong. (2013). Gear diagnostics in a planetary gearbox: A study using internal and external vibration signals. International Journal of Condition Monitoring,. 3. 36-41. 10.1784/204764213808146617.

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