R5X: Individual-Wheel-Drive Design & Analysis

Nolan Thomas Hornby

Mechanical Design Engineering

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Supervisor:	A. Harvey Bell

University of Michigan Ann Arbor
College of Engineering Honors Program

Formula SAE

Formula SAE is a collegiate student design competition organized by SAE International, which has been running officially since 1981. The competition was spun off of the SAE Mini-Indy in 1979, and has since developed into an international competition between hundreds of universities, sponsored and organized by thousands of engineering companies across the globe. The basis of the competition is that each university has been contracted to develop a Formula-style race car, which must compete in a series of on- and off-track events to evaluate not only its performance but also its design, cost-competitiveness, presentation, and fuel economy. These teams come together several times each year to compete in-person.

Scoring

On-track, or "Dynamic" scoring, consists of five categories, based on performance in four events. The "Acceleration" event (100 points, or 10% of the overall competition score) is a 0-75 meter sprint, which evaluates the vehicle acceleration in a straight line. The "Skid Pad" event (75 points, or 7.5% overall) evaluates the vehicle's cornering ability while making a constant radius turn. The "Autocross" event (125 points, or 12.5% overall) evaluates vehicle maneuverability and handling quality overall on a tight course no longer than 1.0 km. Finally, the "Endurance" event (275 points, or 27.5% overall), consists of multiple wheel-to-wheel laps over a closed course approximately 22 km in length, evaluating the durability and reliability of the vehicle. Additionally, the fuel economy (or "efficiency") of the vehicle during this event represents an additional 100 points (or 10% overall) of the 1000 point competition.

MRacing

MRacing is the University of Michigan's FSAE team, consisting of approximately 100 dedicated members spread across 7 distinct divisions: Aerodynamics, Business, Chassis, Drivetrain, Powertrain, Suspension, and Vehicle Dynamics/Simulation. A combustion team until 2021, which ranked #1 in the world twice and sat atop the US podium ten times, MRacing has since converted to an electric-only team, which placed 1st at Michigan International Speedway in 2021 and 2nd in 2022. This merger with Michigan Electric Racing has led to significant development in the teams' technical and manufacturing capability, and many design changes have been made in the interest of improving performance in the past three years.

The first all-electric vehicle was a shared vehicle with Michigan Electric Racing, MER20/21, which was based on a 400V, rear-wheel-drive platform using two independent EMRAX 188 motors, each coupled with an epicyclic gearbox. Placing 1st at Michigan International Speedway, it was Michigan Electric Racing/MRacing's first truly successful electric vehicle.

Figure 1: MER20 at Michigan International Speedway

The following year, post-merger, MR22 was developed; though it was based on a similar powertrain platform, it replaced the tube/space-frame chassis with the technology of MRacing's full-tub carbon fiber monocoque, significantly reducing the mass and improving the stiffness of the car. It also adopted a roll-heave suspension system, improving cornering performance.

Figure 2: MR22 at Michigan International Speedway

Finally, for the 2023 competition, Nolan Hornby and Chris Symonds designed a new powertrain system, capable of all-wheel-drive. Using four motors from AMKmotion embedded in the upright of each wheel, this vehicle was the first successful implementation of all-wheel-drive in a Michigan Formula SAE vehicle.

Figure 3: MR23 at Michigan International Speedway

Individual-Wheel-Drive

All-wheel-drive, or more specifically to the nature of our vehicle, individual-wheel-drive, was adopted in order to improve both cornering and acceleration performance; the control and traction capabilities of two extra powered wheels was able to significantly reduce acceleration times, give enhanced performance in cornering due to extra controllability through torque vectoring/traction control, and allow the vehicle to regenerate power in braking at much higher levels.

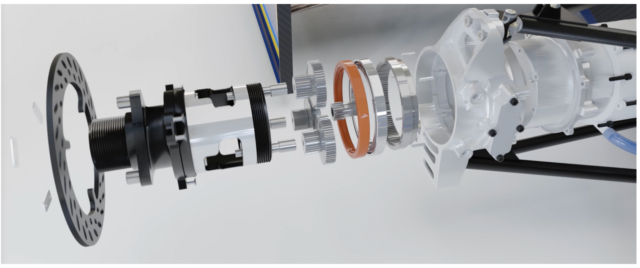

Figure 4: Exploded Render – R4X (MR23) Drive Unit

R4X, or the drive unit first developed for MR23 and now used in MR24, exists in each corner of the vehicle. The system consists of a 600V, 35kW (or 47hp) permanent magnet synchronous motor coupled to a compound epicyclic geartrain at a reduction 12.15:1, yielding an instantaneous output torque of approximately 250 Nm (or 185 ft-lb) to each wheel of the car and a top speed of 104 kmph (or 62 mph), extended to 130 kmph (or 80 mph) with changes to motor control strategy. This system was first designed and manufactured over approximately 10 months, and allowed our vehicle to place 1st at the Pittsburgh Shootout in 2023. It was also remarkably reliable for a first attempt, without a single critical failure across the length of the entire competition season.

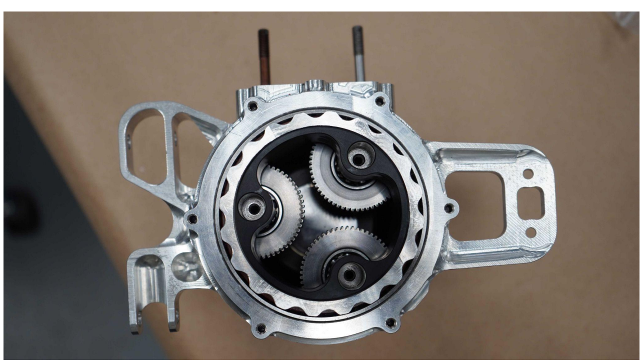

Figure 5: R4X (MR23) Assembly – Axis of the Wheel

System Challenges

This new powertrain system was quickly adopted in response to changes in our competition; with the switch to electric, we found that other teams, especially outside of the U.S., were rapidly becoming more competitive. In particular, many Canadian and German teams began to run individual-wheel-drive systems that rear-wheel-drive platforms could not effectively compete with. Looking at Formula Student Germany (the largest international competition) data for the previous two years, almost all of the teams competing at the top-10 level ran some form of all-wheel-drive system. With the advantages being extremely clear, our team had to develop this system quickly in order to remain competitive. As a result, this system was developed with a lack of significant testing data – though a rear-wheel-drive system can give us many useful data points, it is difficult to validate a vehicle model with a completely new mechanical powertrain system. Furthermore, without the necessary experience for manufacturing these types of drive units, many decisions were made in the pursuit of presumed simplicity, at a cost to mass, vehicle performance, and occasionally unexpected complexity.

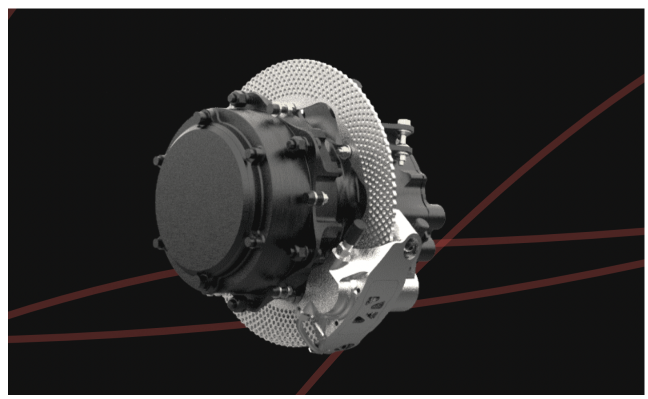

Figure 6: IWD Implementation by Formule ETS[1]

New Drive Unit Development

The decision to begin to develop a revision to this drive unit was made in response to these problems and questions; though the existing platform is successful and will be reused in our 2024 vehicle, MR24, there is further optimization that can be made to this system in the interest of improving performance, efficiency, packaging, assembly time, and manufacturing costs.

Project Scope

The scope of this project lies only in the design of the transmission section of the drive unit – our team seeks to use the same motors in the interest of cost, and their peak power density of ~10 W/g is difficult to improve on with limited off-the-shelf options that are able to fit in our packaging space. Further, suspension design will not be considered in this analysis, as it depends heavily on chassis design and inboard suspension setup, which may change significantly by 2025, when this system is intended to be implemented; suspension design space constraints will be kept at the same specification as our outgoing drive unit system, with a Hoosier 16.0x7.5 as the planned vehicle tire choice. Finally, wheel rim choice will be considered in this analysis, but not ultimately selected, as the development of Carbon Fiber rims is an ongoing research effort and may later influence mounting geometry.

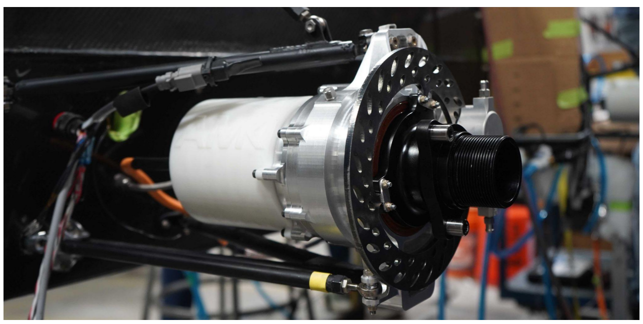

Figure 7: R4X (MR23) On-Car Assembly

Questions Addressed – Reduction Ratio

The first question to address was that of the reduction ratio – the previously chosen 12.15:1 was chosen out of relation to the performance of earlier cars and simulation of the Acceleration event due to simplicity, but it is difficult to confidently validate these decisions for other events without test data from comparable drive units. The planned reduction ratio for this updated drive unit was adjusted through simulation and comparison to data collected from the outgoing vehicle, MR23.

Questions Addressed – Architecture

The second question to address was that of the overall architecture of the system – the previously chosen compound epicyclic (also found in the Ford Mach-E) was chosen because of its similarity to existing epicyclic transmission systems designed for MER 20/21 and MR22 and packaging efficiency, but it has significant flaws with respect to driveline efficiency. This is especially problematic with a proposed change to FSAE efficiency scoring, which has the potential to make this event far more influential in the overall scoring. The planned architecture of this updated drive unit was modified to improve the output efficiency of the vehicle and reduce the length of the system (or the amount that the drive unit "sticks out" into the suspension members).

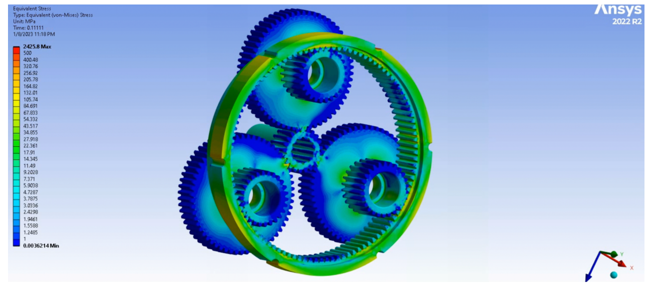

Figure 8: R4X (MR23) Isolated Geartrain FEA

Questions Addressed – Manufacturing

The last question to address was that of manufacturing complexity. In particular, the interface between each of the planet gears. Though there is limited literature on the design of these systems, *Planetary Gear Trains* by *Kiril Arnaudov* and *Dimitar Petkov Karaivanov* details this well, stating the following:

> 1. *Planet manufacturing is considerably more complicated due to both rims and especially because of the need for their exact angular positioning.*[2]
> 2. *The problem of planet load equalization is considerably more complex because of the greater number of negative influence factors that cause the greater risk of failure.*[2]

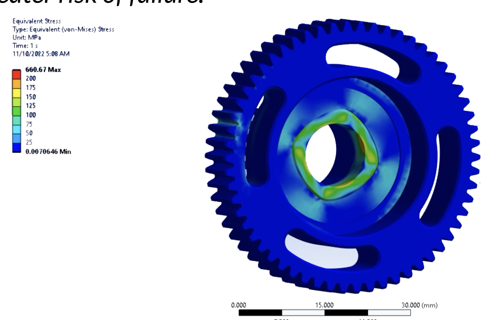

Figure 9: R4X (MR23) Planet Interface FEA

With R4X (MR23), we successfully addressed this problem with a timed polygonal spline design, but it limited our manufacturing processes to shops capable of CNC gear and non-cylindrical grinding, significantly complicating and extending the timeline of the manufacturing of these gears. This interface was adjusted, allowing electrical discharge machining (a process available in-house) to be used for the machining of splines.

Methods – Pre-simulation Reasoning

The AMKmotion motors require some degree of gearing in order to be useful for our application. This has been previously discussed & simulated for earlier generations of this system; however, a proposed efficiency rule change for Formula SAE 2024 has brought about the need for further research. Previously, although the FSAE efficiency "event" was valued at 100 points, the lowest possible score was at least 85 points, making vehicle efficiency near-insignificant against the total of 675 dynamic points. Changes to the calculation of these scores (making better use of the 100-point range of this event) have, however, incentivized higher performance in this event. Simulation bounds for gear ratio fall between 10.0:1 and 18.0:1, representing limits beyond which either top speed increases senselessly or torque increases beyond the usable limits of the competition regulations, respectively. A point-mass model was chosen for model simplicity and solution time, ignoring L/R weight transfer and assuming that each corner of the car has the same amount of traction. To defend this assumption, tires were modeled with a range of performance coefficients to show the sensitivity (and direction) of the optimal gear ratio to our tire performance "guess." The vehicle parameters for simulation can be found below; this analysis was completed assuming that future cars will, generally, be able to achieve similar weight and aerodynamic properties as compared with present-day MRacing cars.

Table 10. Simulation Environmental Parameters

Parameter	Value	Units
a_g (gravity)	9.807	m/s^2
ρ_{air}	1.177	kg/m^3
μ	0.01845	cP
v	15.67	cSt
T_∞	27.00	^oC

Table 11: Simulation Vehicle Parameters

Parameter	Value	Units
$m_{vehicle}$	190 (~420 lb)	kg
m_{driver}	70 (~155 lb)	kg
C_LA	2.1	
C_DA	1.1	
R_{tire}	0.206	m
μ_x, μ_y	$1.0 - 2.0$	
R	$10.0 - 18.0$	

Finally, the motor was modeled using dyno data supplied by AMKmotion — we assume for the design that the motor is not allowed to exceed the mechanical speed limit of 20,000 RPM and we enforce a 80,000 W power limit as per competition regulations.

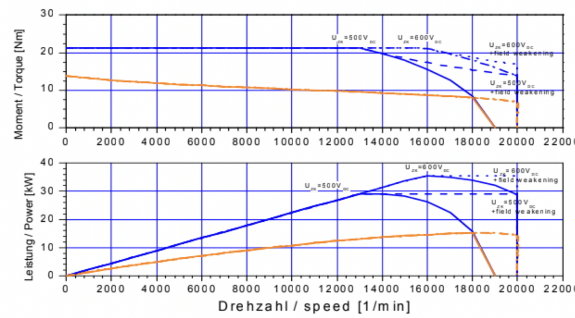

Figure 12: AMKmotion DD5 Torque Curve

This analysis was simulated on a large variety of tracks; to get a holistic model of the competition, this included one acceleration map and a mix of both autocross and endurance maps from various years of competition.

Generally, mechanical gear reductions are most efficient when run at high input torque and low input speed; in other words, lower (taller) ratios. Lower numerical ratios also happen to be much easier to package. Therefore, since we cannot take the efficiency map of a to-be-designed gearbox into account for this simulation, the combined benefit of ratios on the lower numerical end of our "optimal" range for performance will be noted when selecting a configuration.

Methods – Model Validation

In order to confirm that the point-mass model is representative of the true performance of the system, simulated tracks were compared against recorded data from competitions. One set of laps in particular, from Pittsburgh Shootout 2023, was closely compared as this was the competition during which the car was "pushed" (as shown by average power consumption data and competition results) the hardest. This lap is valuable to look at, as a model cannot take into account a slow driver – only a slow car. The most accurate agreement comes from laps where the car is pushed to a point where the effects of driver error and other difficult-to-control variables are as limited as possible.

Simulation-Data Agreement

Pittsburgh 2023 was the race where our outgoing AWD electric race car performed closest to its limits; model agreement was done by comparing simulation results against recorded lap data.

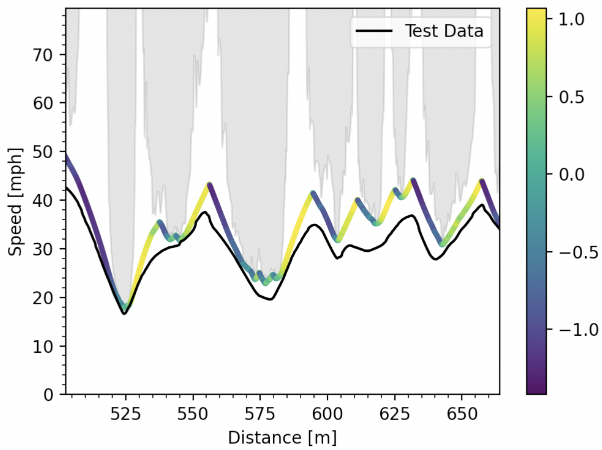

Figure 13: Pittsburgh Shootout 2023 Data Agreement

For visibility purposes, this is a small section of the Pittsburgh lap. As shown above, there is decent agreement between the model and our recorded data, with a few exceptions – the model assumes both a perfect driver (able to perfectly time braking zones and zero reaction delay) and a car that has negligible longitudinal load transfer when transitioning from accelerating to braking. As a result, the model tends to run faster than our testing data – to account for some of these issues, further simulation is run with a sweep of both μ_x and μ_y to determine the extent to which gear ratio is sensitive to aggressiveness of the driver.

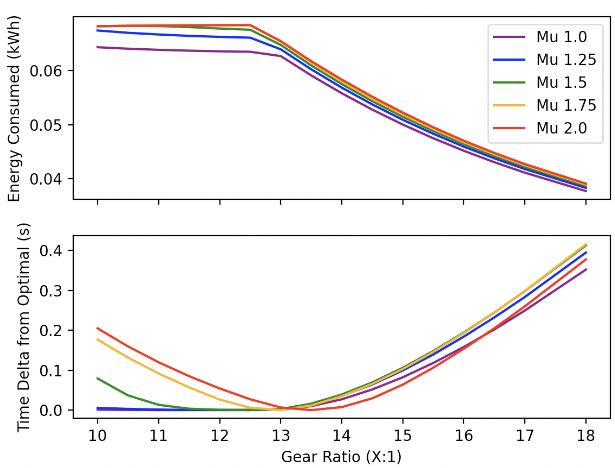

Figure 14: Standardized Event – Acceleration

For a pure acceleration event, the ideal gear ratio is somewhat sensitive to tire performance: where 10:1 to 13:1 is optimal for a tire with a frictional coefficient of ~1.0, 13:1 to 14:1 is optimal for a tire with a frictional coefficient of ~2.0. This makes intuitive sense since a lower-performing tire will perform poorly during the launch, so it will benefit from decreased impact of the mechanical limit of the motor (with a lower/taller ratio). From previous expectations of our tires, ratios of ~12.5:1 appear to perform the best.

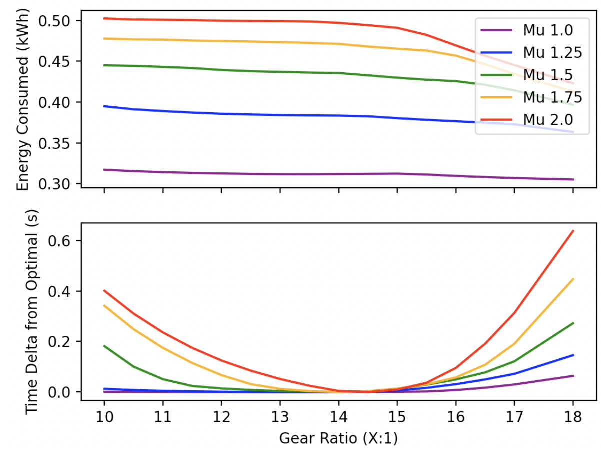

Figure 15: Pittsburgh 2023 – Autocross

For the Pittsburgh autocross lap, the ideal gear ratio is essentially insensitive to tire performance. Approximately 14.2:1 is optimal across the entire sweep of tire frictional values. Notably, high ratios tend to consume far less energy – however, this is not necessarily a quality that should be targeted, as it is a result of the vehicle being limited more by the mechanical limit of the motor.

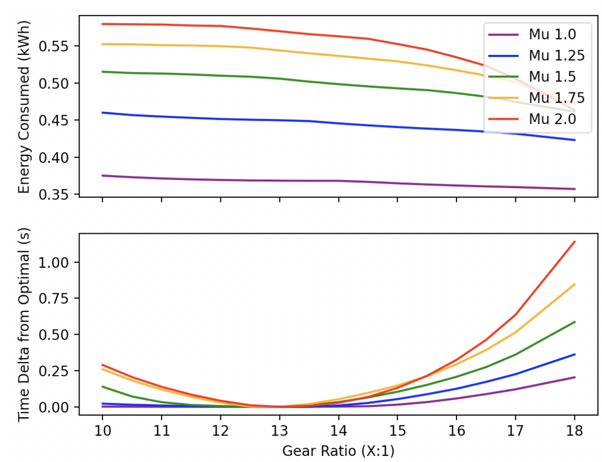

Figure 16: MIS 2021 – Autocross

For the MIS autocross lap (2021), the ideal gear ratio is again, essentially insensitive to tire performance. Approximately 13.0:1 is optimal across the entire sweep of tire frictional values. The energy note from the Pittsburgh lap applies here as well.

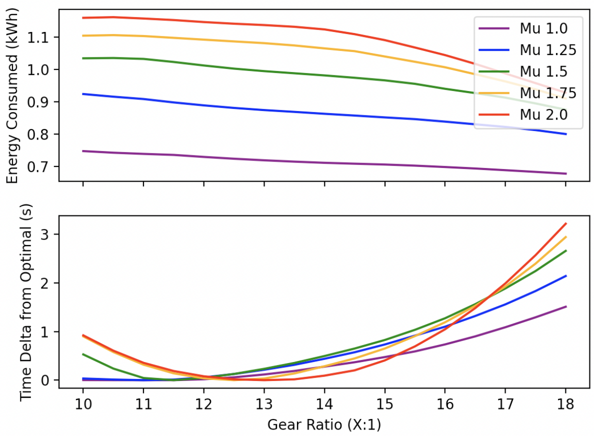

Figure 17: MIS 2019 – Endurance

For the MIS endurance event (2019), the ideal gear ratio appears to be somewhat sensitive to tire performance, with a wide range between 11.5:1 to 13.5:1. This track was particularly fast, but is a good example of the difference between autocross and endurance in terms of ideal setup.

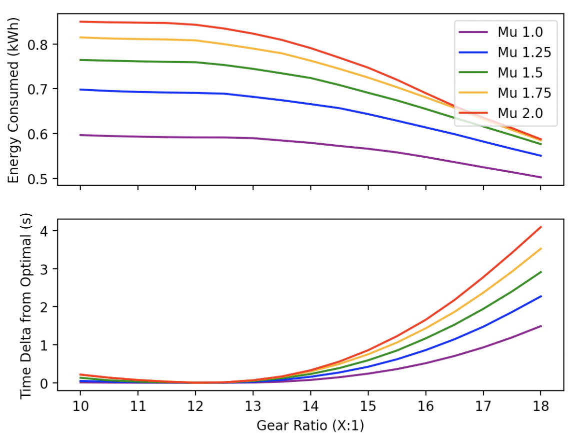

Figure 18: FSG 2012 – Endurance

For the FSG endurance event (2012), the ideal gear ratio is insensitive to tire performance. Approximately 12.0:1 is optimal across the entire sweep of tire frictional values. Notably, it is clear from this plot that, for efficiency purposes, bias towards shorter (higher numerical) gear ratios seems to yield an improvement in motor efficiency in comparison to taller (lower numerical) gear ratios, for similar lap times.

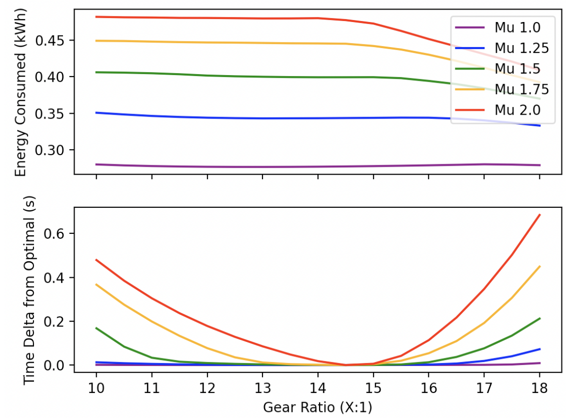

Figure 19: FSG 2010 – Endurance

For the FSG endurance event (2010), the ideal gear ratio is, again, insensitive to tire performance. Approximately 14.5:1 is optimal across the entire sweep of tire frictional values. Once again, bias towards higher ratios is desirable for motor efficiency purposes at similar lap times.

Simulation Discussion

Where the top speed is not reached during the run, an increase in gear reduction generally led to an increase in efficiency. This change, however, is marginal relative to the performance gains associated with selecting reduction solely on lap times. It appears that ideal gear ratio is mostly insensitive to tire performance, but varies more widely depending on the track setup; despite the variance, a range of 12.0:1 to 14.0:1 is generally desirable for most events. For the same increase in lap time, using a higher (numerical) ratio yields a higher motor efficiency than a lower (numerical) ratio. This may not be a useful target, however; one benefit of lower (numerical) ratios is that they generally yield higher efficiency at the earlier transmission stages, significantly reducing the effects of seal/bearing drag in comparison. Because a lower numerical ratio yields smaller differences in tooth count at each stage, these ratios can also generally be packaged far smaller.

Simulation Conclusion

From these results, it is concluded that tire tractive ability has a significant effect on lap time, but not on selecting an ideal ratio. 13.0:1 appears to be somewhat optimal for all events, but gains may be found in packaging and vehicle weight by using a slightly lower numerical ratio. Efficiency gains are inconclusive, and likely depend far more on geartrain efficiency to increase this score without hurting lap times. A range of 12.0 – 13.0:1 remains the target design range.

Design Exploration

Where improving motor efficiency depends on cooling and changing the gear reduction ratio to a range potentially undesirable for performance, an increase to gearbox efficiency is always desirable. Because an 80kW power limit is imposed on all teams and most of this power can be put to the ground with an all-wheel-drive system, efficiency is the only way to gain acceleration performance above 48 kmph (or 30 mph). As a result, new coaxial gearbox architectures were considered for a revision of this system, presented below.

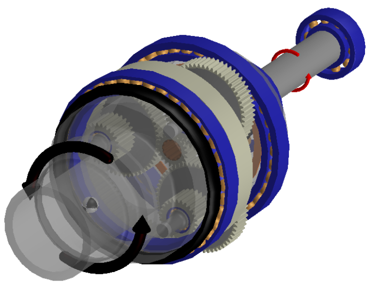

Figure 20: Design of AI-PGT – 3 Planets

(12.14:1 shown) A three-planet AI-PGT (sun gear input, carrier output, with three pairs of linked planets) was used for our first IWD electric vehicle, MR23. This system is incredibly power dense and has low mesh losses (as it has no more gear meshes than a standard planetary). Unfortunately, due to the orbiting nature of multiple clocked planets, it experiences extremely high churning/drag losses and requires tight tolerances on difficult-to-control surfaces. These problems are very difficult to solve with only minor design changes, so other architectures were considered for a system revision.

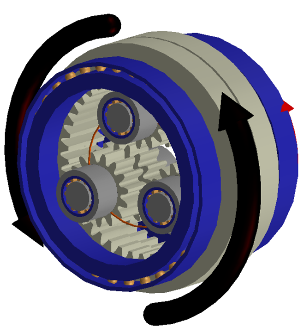

Figure 21: Design of II-PGT – 3 Planets

(13.40:1 shown) A three-planet II-PGT (carrier input, ring gear output, with three pairs of linked planets) was considered as a solution to meet our motor speed reduction target. Despite its high power density and compact size, it is severely limited by balancing the capabilities of bearings under centripetal acceleration, as they will be driven at the full speed of the motor. This design was found to be very inefficient and self-locking in practice, making it unsuitable for our application.

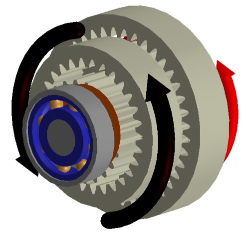

Figure 22: Design of II-PGT – 1 Planet (Wobble)

(13.26:1 shown) A single-planet II-PGT (carrier input, ring gear output, with one pair of linked planets) was also considered as a solution to meet our motor speed reduction target. This design, unfortunately, was found to be very heavy, as it does not benefit from the splitting of torque to multiple planets. Furthermore, as it behaves like a harmonic drive gearbox, it is just as subject to poor efficiency and self-locking (in practice) as the previous design.

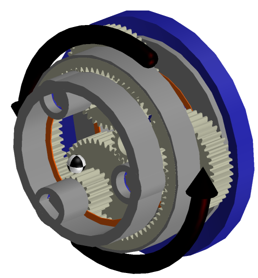

Figure 23: Design of AI-PGT (Pseudo) – 3 Planets

(12.02:1 shown) Another three-planet AI-PGT, similarly to the previous edition shown, has many strong attributes; firstly, it is just as power dense as a standard planetary geartrain, and with only six meshes, is at least as efficient as a standard planetary gearbox. The benefit of this design is the "fixing" of the planets – rather than an orbiting design, which relies on the orbit of planets to achieve the necessary reduction ratio, this design has a pin-fixed carrier, and the rotation of the ring gear is the output of the system. This is occasionally referred to as a "pseudo-planetary," as there is no orbiting involved, but instead resembles a typical spur/gear gearbox but with sharing of torque between multiple gears at each stage. Without the characteristic of orbiting, the churning and drag losses of the oil are significantly reduced, at a cost to diametral packaging, as the lack of orbiting reduces the ratio by one rotation of the output.

Choice of Geartrain Architecture

A pseudo AI-PGT was selected as an ideal architecture change for this system. The practically self-locking nature of II-PGTs was found to be too inefficient in regeneration to improve the overall efficiency of our system, despite the significant weight and packaging gains. A modification of the true AI-PGT style used in R4X (MR23) was found to be too difficult to make more efficient, as the sloshing effects of three gears orbiting at about ⅓ of the motor speed are high, especially considering that a submerged oil bath is the simplest and most lightweight form of lubrication.

The pseudo AI-PGT (or pseudo compound planetary) gearbox solves this problem by switching the fixed "body" in the system to the carrier, rather than the ring gear. This causes the planet gears to spin in place, and the ring gear to spin around the axis of the gearbox.

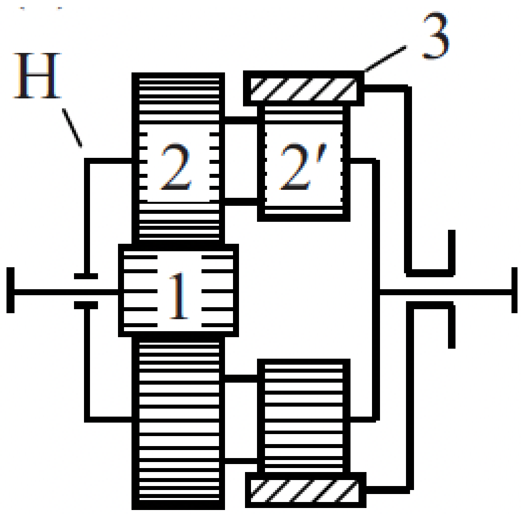

Figure 24: Diagram of pseudo AI-PGT[2]

As shown by the figure above, this style of fixed-ratio gearbox fixes the carrier holding each of the planets represented by *2* and *2'*, with the ring gear *3* as the output to the system. This modification eliminates the orbiting which is characteristic of most planetary gearboxes, where the ring gear *3* is instead held stationary, and the resulting rotation of the carrier is the output of the system.

Another benefit of this design, distinct from efficiency, is the nature of its packaging. If the ring gear is the spinning body, the wheel can be mounted at any point to the external casing of the gearbox, because the entire external gearbox casing is spinning. As a result, it is possible to design this system such that the majority of the drive unit lies much more outboard of the wheel, rather than into the area inboard of the wheel which is threatened by cones, rocks, and control arms under extreme bump suspension conditions.

Figure 25: R4X (MR23) Corner On-Car

A change to this style of wheel mounting could lead to a reduction in approximately 100mm (or 4") of the axial extent of the drive unit beyond the wheel, which is the approximate length between the motor and the rim of the wheel on our outgoing drive unit. This would reduce the current restrictions on aerodynamic and suspension design in this area, as illustrated above in Figure 25.

One detrimental effect of this design is that the rolling nature of true planetary gearboxes increases the number of input revolutions per revolution of the output by 1. As a result, the design used for R4X (MR23), which achieved a ratio of 12.154:1, would only achieve 11.154:1 in a spinning-ring configuration. As a result, this style of this design (for the same ratio) will lead to a drive unit with a slightly larger diameter.

Macrogeometry Constraints

The design of an AI-PGT (pseudo compound epicyclic geartrain) has far more constraints than a standard planetary gearbox, mostly due to the nature of the two stages. The first "constraint" is represented by the reduction ratio, which can be calculated using Eq. 1 below,

$$i_0 = \frac{-z_3}{z_2'} * \frac{z_2}{z_1} \qquad \text{Eq. 1}^{[2]}$$

where z_1, z_2, z_2', and z_3 represent the number of teeth on the input gear, the first stage (large) planet, the second stage (small) planet, and the ring gear, respectively, and i_0 is the basic speed ratio. This value must fall between 12.0:1 and 13.0:1 to meet system requirements. Because each of the two stages must mesh together and lie at the same pitch circle diameter, a assembly condition must be met, as calculated using Eq. 2 below,

$$\frac{(z_1 * z_2') + (z_2 * z_3)}{k * \delta} = \text{an integer} \qquad \text{Eq. 2}^{[2]}$$

where k is the number of planets, and δ is the largest total divisor of the planet teeth number z_2' and z_2. The maximum practical value for k is 3 planets, limited by interference between the planets as they grow to meet reduction ratio requirements.

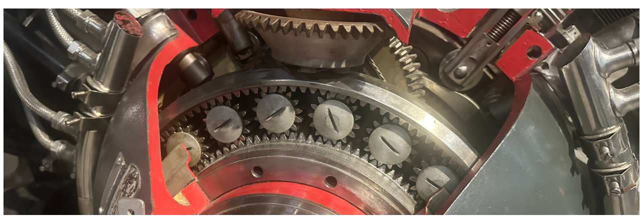

Figure 26: Wright R-1820 Engine Epicyclic Geartrain

Generally speaking, adding more planets increases the load capacity of a planetary transmission at a small cost to weight, but at diminishing returns due to imperfect load sharing. In this case, just *k=2* was found to be unsuitable for almost any configuration, so the geometrical maximum number of planets was used. To avoid the requirement of blind-hole rotary broaching or plunge EDM manufacturing processes, which may be extremely costly, the root diameter of the input gear must also be larger than the motor spline that the gear mounts on – therefore, the following condition must be satisfied,

$$(z_1 - 2.5) * m_{12} > 11.0mm \qquad \text{Eq. 3}$$

where m_{12} is the gear tooth module of the input gear *1*. In order to promote even wear among teeth, a hunting tooth ratio (or at least very close to it) at each stage is also desirable; therefore, the following two conditions must be satisfied.

$$GCD(z_1, z_2) \ll z_1 \qquad \text{Eq. 4}$$
$$GCD(z_2', z_3) \ll z_3$$

Ideally, the greatest common denominator for each of these pairings would be equal to 1, representing pairs of gears where each tooth on one gear meshes with every tooth on the other as the gear rotates, preventing wear from contaminants/overheating/etc. from deteriorating specific teeth more quickly than others. Using a Python script, the resulting set of approximately 100 applicable geartrain combinations was calculated before running further in-depth analysis. A subset of these combinations are presented in Table 27 below, representing the most practical examples that were considered, along with the ratio and estimated diameter $Ø_{du}$ of each potential system.

Table 27: Subset of Macrogeometry Options

i_0	$Ø_{du}$	m_{12}	z_1	z_2	$m_{2'3}$	$z_{2'}$	z_3
x:1	mm	mm	#	#	mm	#	#
12.73	111.0	1.00	17	46	1.00	17	80
12.27	112.0	0.80	20	59	0.80	25	104
12.08	113.0	1.00	17	47	1.00	19	83
12.08	115.0	1.00	19	47	1.00	17	83
12.55	115.2	0.80	20	61	0.80	26	107

Geartrain Materials

Our earlier drive unit, R4X (MR23), was run with nitrided gears (AISI 4140, AISI 4340) to trial using this material and treatment to reduce the manufacturing cost of dealing with the deformation of the carburizing process, but this was unfortunately unsuccessful, as shown by pitting highlighted below in Figure 28.

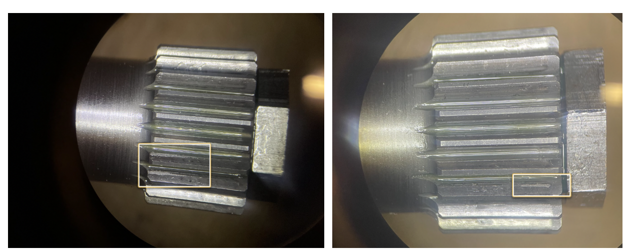

Figure 28: R4X (MR23) Nitrided Input Gear Trial

For gears, carburizing steels perform far better than nitriding steels, in part, due to their deeper case depth, which is able to sustain higher loads and with higher contact resistance. AISI 8620 was selected as an accessible automotive-grade gear material, with superior AISI 9310 as an alternative in the event of limited manufacturing quantities or premature failure.

Modeling – Development

Romax Technology (Hexagon) is a sponsor of our team and supports our development of drive units through the use of their electromechanical design software. In order to evaluate the selection and lifetime of our gears, bearings, seals, and other components, a model was first built up in Romax Enduro.

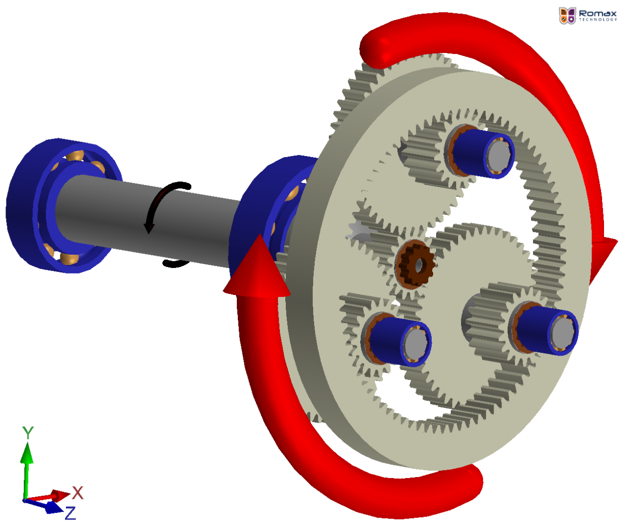

Figure 29: Model of a pseudo AI-PGT transmission

Modeling – Load Cases

In order to effectively gauge the lifetime/safety factor for each design, the development of a representative set of load cases was critical – similarly to the selection of reduction ratio in earlier years, in the design of R4X, a large safety factor was placed on our estimates for load cases, as we designed the system without validation data for the needs of an AWD system. Now, with existing data to review, we were able to design far closer to the true requirements of the system. These load cases were collected by taking hours of data from testing and competitions, and using a K-means cluster to organize this data into specific "load cases" that are analyzed individually. Because our earlier system was not tested significantly enough under regenerative braking, these load cases (shown with negative torque values) were conservatively derived from simulation as shown earlier in this paper. As shown in Table 30, the load cases are with respect to the output shaft of the motor, or the input to the gearbox. The system was modeled for 10 hours of total operation, which has historically represented about half of a season of testing.

Table 30: Transmission Load Cases

#	$n\ [1/min]$	$T\ [Nm]$	$t\ [min]$ (10h)	$t\ [\%]$
1	13,700	-18.78	20.0	3.33%
2	10,800	-17.40	42.5	7.06%
3	8,000	-14.60	38.3	6.34%
4	11,000	-11.99	19.2	3.13%
5	9,100	-9.60	43.3	7.20%
6	6,500	-9.26	40.0	6.68%
7	8,600	-4.70	25.0	4.07%
8	5,800	-4.70	39.2	6.50%
9	8,800	1.79	16.7	2.71%
10	5,600	2.17	26.7	4.46%
11	8,500	6.95	30.0	4.94%
12	6,100	8.47	35.0	5.82%
13	12,800	10.55	13.3	2.11%
14	9,200	12.07	40.8	6.74%
15	6,600	15.49	34.2	5.76%
16	2,200	15.82	7.5	1.20%
17	9,000	16.30	45.0	7.45%
18	11,200	16.91	53.3	8.95%
19	18,100	18.25	7.5	1.18%
20	14,000	19.00	25.8	4.35%

Modeling – Materials

AISI 8620 was modeled as a case hardened steel, rated as AGMA grade 2. Relevant material properties can be found below in Table 31.

Table 31: Geartrain Material Properties

Property	Value	Units
Core hardness	35.0	HRC
Surface Hardness	60.0	HRC
Modulus of Elasticity	2.05e5	MPa
Yield Strength	1.00e3	MPa
Tensile Strength	1.30e3	MPa
Density	7.85	g/cm^3
Poisson's ratio	0.29	
Thermal Conductivity	49.0	W/m C
Specific Heat Capacity	490.0	J/kg C

Modeling – Conclusions

Based on modeling results from the list of applicable geartrain configurations, the following configuration was found as the most suitable for continued design.

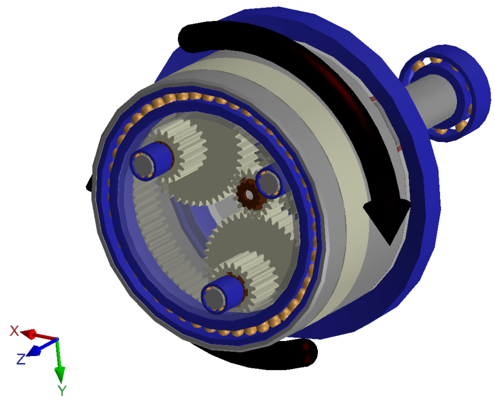

Figure 32: R5X Proposed Model

This system uses two stages of gears, each at a gear tooth size, or module, of 1.0mm (an increase from the 0.8mm size of the previous system), which reduces the impact of manufacturing imperfections, allowing the system to be correctly manufactured with lower tolerances, and therefore at a lower cost. The first and second stages are represented by a 17/47 mesh and a 19/83 mesh, respectively, leading to a reduction ratio of 12.08:1 (3901/323) which is within the specification of the requirements. The existing and validated size of needle roller bearings can be used in this design, with potential to decrease the size further, allowing the space beneath the gears to be freed up for design changes for the planet clocking interface.

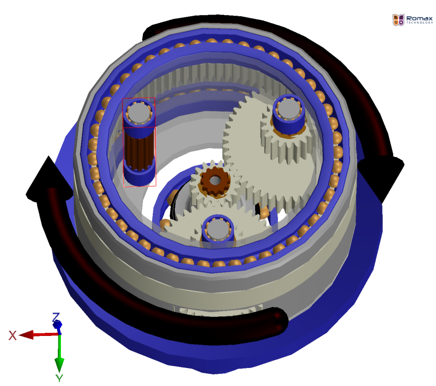

Figure 33: R5X Proposed Model, Hidden Planet

Further, the existing and validated size of AC bearings can be used in this design, though this size may not be changed significantly, because assemblability requires that the inner diameter of this bearing must be less than the tip diameter of the ring gear, but still fit over the carrier containing each of the needle roller bearings.

Reliability modeling yielded strong results for this configuration; at a primary and secondary stage effective face width of 8.0mm/10.0mm respectively, the resulting factors of safety against the season test profile can be found below.

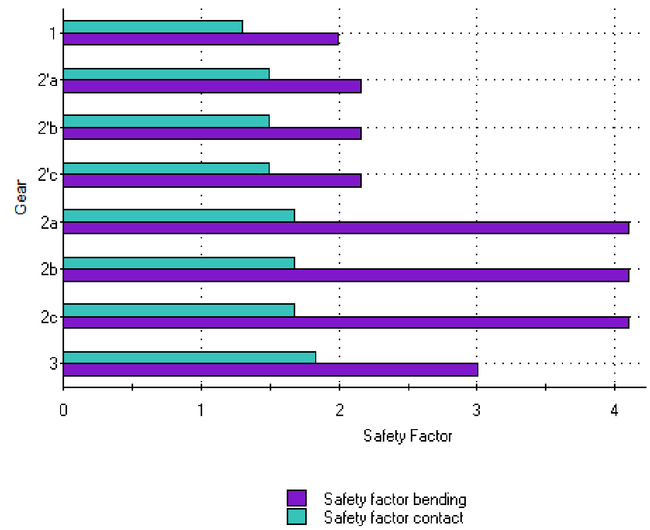

Figure 34: R5X Proposed Model, Safety Factors

Relative to the outgoing geartrain, this represents a 33% reduction in width, and a 1% increase in diameter, for comparable power density. Further, this design combines the upright and planet carrier into just one part, further reducing the mass of the system.

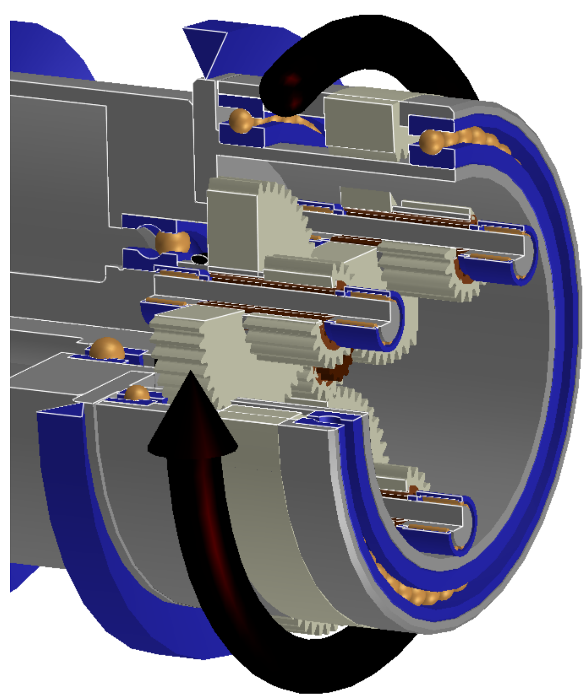

Figure 35: R5X Proposed Model, 90 degree cutaway

Recommendation – Structural Changes

The integration of this system requires an inversion of the current structural package. With the entire housing of the gearbox spinning, the inner carrier body makes up the upright structure used to transmit loads from the tire. This structure should be done using a one-piece carrier, and is supported by the proposed R5X geartrain model. Depending on the needs of assembly, a one piece housing may be possible, but a two piece housing is recommended for this application, since it may be difficult to maintain the same level of structural rigidity in the carrier/upright while leaving space to install gears. Since the housing is now a rotating component, the "revolving door" style of assembly previously found on R4X is no longer applicable.

Recommendation – Packaging Advantages

The most distinctive advantage of this system is the impact on packaging. Pictured below in Figure 36 are a proposed model for R5X and the outgoing model for R4X; the new system is remarkably shorter in length due to the spinning housing, on which the wheel can be mounted at any point.

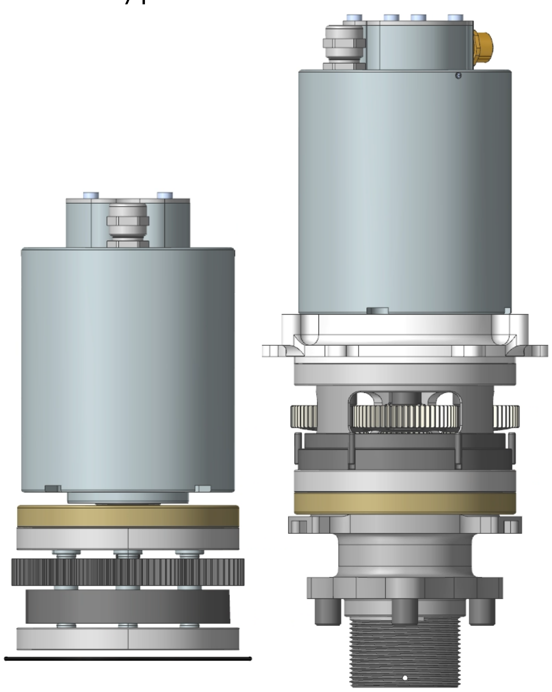

Figure 36: R5X proposed model vs. R4X (MR23)

This change will likely open up a pull-rod suspension system as a possibility, which was not realistic with the previous system due to interference with the motor. This likely leads to an improvement in aerodynamic performance as well as a reduction in the height of the center of gravity of the car.

Recommendation – Wheel Alternatives

The most significant challenge to adopting this system for new MRacing vehicles is the limited availability of off-the-shelf rims that are able to mount to this drive unit. Historically, MRacing has used magnesium alloy rims purchased from OZracing, but these are not manufactured at sizes capable of mounting to the periphery of the drive unit. For OTS options, the best option is to switch to an aluminum 10" 12 bolt shell offered by Keizer racing, which has proven successful for competitors despite concerns over manufacturing quality. The pursuit of in-house carbon fiber wheels is an ongoing research project for MRacing and should be continued in the interest of adopting superior drive unit designs and decreasing vehicle weight.

Figure 37: OZRacing CL 10" vs. Keizer 10" shell

Recommendation – Efficiency Measurement

Based on existing modeling capabilities, this drive unit yields a 1-2% gain in efficiency over the existing design. This, unfortunately, may be not be entirely accurate, as the modeling of the sloshing effects of oil in a true epicyclic geartrain is difficult and not entirely modeled by Romax. As a result, it is essential that this be tested physically after the adoption of this system; based on first principles, it is extremely unlikely that this change would decrease the efficiency of the drive unit, but this still must be tested. MRacing has the tools and equipment necessary to evaluate the performance of each of these gearbox architectures, and the true efficiency gain can be measured once this system has been adopted for the first time.

References

[1] *Formule ETS Homepage*. Formule ETS. (n.d.). https://formule-ets.ca/

[2] Arnaudov, K., & Karaivanov, D. P. (2019). *Planetary Gear Trains*. CRC Press, Taylor and Francis Group.