# Michigan Baja Racing 3-Speed Transmission

*Design Report*

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Section 003-Luntz, Team 21
December 16, 2024

## Baja Transmission Executive Summary

The Michigan Baja Racing team has requested that our team reevaluate their current gearbox setup. In order to help keep the MBR team competitive with other Baja SAE teams, our project team compared the benefits of the current MBR gearbox design to that of alternative designs, and ultimately created a prototype of our recommended design.

The main motivation of our project is to help the MBR team maximize the amount of points won in each competition. To determine where to focus our energies, we analyzed the cause of points lost over the past 6 competitions. During the 2023 & 2024 seasons, over 20% of points lost were due to lack of torque out of the transmission, and due to implementing reverse in 2024, the loss of points due to halted forward progress dropped from 25% in 2023 to 3% in 2024. Based on the points breakdown, the team decided to fix the issue of lack of torque outputted by the gearbox while maintaining reverse capability.

The requirements and specifications place an emphasis on backwards compatibility with past vehicles. The backwards compatibility requirement implies that our project team must work within the current systems implemented by the MBR team, and minimally impact surrounding subsystems without sacrificing performance.

The current benchmarks for this project include past MBR gearboxes as well as commercially available gearboxes. Using these benchmarks as inspiration, we deconstructed our design space into subsystems and created a morphological chart and a concept tree. We also used design heuristics to further increase our concept pool. By sketching out each design, we eliminated impractical designs while also gathering data on each design: (1) number of gear ratios, (2) part count, (3) manufacturability, (4) mass, and (5) efficiency loss. Gear analysis was conducted on each design in order to ensure compliance with the requirements and specifications and to provide a mass & volume estimate: to determine the face widths of the gears, we used Lewis Bending Stress, contact stress, and bending and pitting fatigue equations. The five data points were each put into an analytical hierarchy process which in turn informed a Kepner-Tregoe decision matrix which guided our decision in selecting the final design.

The selected design is an evolution of the 2024 transmission with an extra gear train added in parallel. Updated ratios provide more torque and higher speed across the two forward gear trains, and the reverse gear train preserves the ability to back out of obstacles and continue forward progress.

With gear sizes established from our initial sketches, the rest of the gearbox was designed around the gears with a focus on packaging within a previous vehicle's architecture. Shifting was achieved through the use of male and female dog gears that select between the high, low, and reverse gears via a shifting barrel. Bending and torsion calculations were performed on the shafts and bearing calculations ensured they were sized correctly. Finite element analysis was also run on the gear webbing and case to determine the maximum stress each would see. These types of analysis verified all components had a 1.2 safety factor on their worst case loading scenarios ensuring we meet our durability and load capability specifications.

Manufacturing plans that made use of MBR sponsors and a bill of materials were outlined to organize all components needed for the gearbox, their manufacturing, and their assembly, meeting our first deliverable for this project: a comprehensive plan for the construction and integration of the final design into the rest of the vehicle. The other deliverable was a manufactured prototype of our design. This was a 3D printed demonstration prototype meant to test shifting functionality between the 3 speeds, thus requiring a separate design that omits spacers and bearings is more 3D printable. The initial design required a few iterations. However, after a couple of design and material changes, we were able to prove shiftability of our prototype, thereby validating the barrel drum shifter design.

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# 1 Abstract

The goal of this project is to develop a transmission for an off-road vehicle being built by the Michigan Baja Racing Team. Michigan Baja Racing races at 3 competitions annually, with the goal of winning the most cumulative points across all 3 points compared to other competing teams. During the 2023 & 2024 season, over 20% of points lost were due to lack of torque out of the transmission, and due to a transmission design improvement in 2024, the loss of points due to halted forward progress dropped from 25% in 2023 to 3% in 2024. Michigan Baja Racing has come to our team with the request to redesign the current transmission in order to decrease the amount of points lost during their competition season. Our ME450 team will compare the benefits of the current design to that of alternative designs and ultimately create a prototype of a recommended design. The final prototype will ideally include accurate gear sizing, integration plans with the existing CVT, 4WD system, rear axle, braking system, and chassis, and a reliable method for shifting between various gears. The scope of this report encompasses the context of the design problem, a high level overview of the design process employed to solve this problem, stakeholder analysis, the research and derivation of requirements and specifications, benchmarking analysis, design generation, design selection, and a timeline illustrating project milestones, tasks, and status. [1]

# 2 Introduction, Background, and Information Sources

## 2.1 The Competition

The Society of Automotive Engineers (SAE), gives college students around the world the opportunity to design, build, and test solutions for a variety of vehicle engineering competitions through several Collegiate Design Series (CDS). In the Baja SAE competition, students are tasked with designing and building a single-seat, four wheel drive, all-terrain vehicle that is to be a prototype for a reliable, maintainable, ergonomic, and economically viable production vehicle. The students must work together as a team to design, engineer, build, and test a vehicle that will compete on off-road courses developed by the competition organizers. Each competition runs for four days. The race courses are designed for several distinct dynamic events to test the performance of the vehicle's suspension, acceleration, torque, durability, and maneuverability. The dynamic events and take up the majority of days three and four at a competition [2]. Days one and two are filled with a few static events such as a design competition where the design process to build the vehicle is explained. There is a cost competition in which the cheapest car wins, and a business presentation where a pitch for a fictional business scenario is presented. All these static and dynamic events are scored to sum to 1000 points; the team who accumulates the most points wins the competition [1] and the team that accumulates the most points across all 3 North American competitions within a season wins the Mike Schmidt Memorial Iron Team Award.

## 2.2 The Team

Michigan Baja Racing (MBR) is a student team housed in Wilson Center at the University of Michigan - Ann Arbor that competes to win the Baja SAE competition National Championship. MBR is made up of a small group of dedicated undergraduate students who design, build, and race a new off-road vehicle from scratch every year. The team is led by administrative members who are responsible for choosing the directors of each vehicle subteam (Suspension-Steering-Brakes, Chassis/Ergonomics, and Drivetrain) as well as the subsystem leads for the vehicle subsystems within each subteam. Subsystem leads are then responsible for designing, testing, and in some cases manufacturing their subsystems. The team aims to manufacture as much of the vehicle at their facility as possible, with certain parts or operations that cannot be done in house outsourced to specialty manufacturers. The team has access to a range of CNC and manual machine tools and metal fabrication tools available in university machine shops and makerspaces, but lacks some which may become relevant to this project such as a wire EDM, a CNC mill with more than 4 axes, and any precision grinding or hobbing tooling for the manufacturing of gears. The team's primary goal is to win the overall competition, which drives vehicle level requirements that are then broken down into subteam and subsystem level requirements. These help push the car towards the best competition performance possible. MBR has won the Mike Schmidt Memorial Iron Team Award in seven of the past ten competition seasons [1].

## 2.3 Project Description

Section 2.3.1 gives an overview of the whole drivetrain system and the way that power is transferred from the engine all the way to the wheels. Section 2.3.2 will give an overview of MBR's engine. The following sections are a description of the current vehicle transmission set up with a CVT or continuously variable transmission (2.3.3), gearbox (2.3.4), and rear drive shafts (2.3.5), as well as the brake system (2.3.6).

**2.3.1 Drivetrain Power Overview.** The drivetrain of a vehicle transfers power from the engine all the way through the wheels through a collection of systems so that the vehicle can move. In a standard 2 wheel drive (2WD) drivetrain of an MBR vehicle, power generated at the engine is transmitted through a mechanical continuously variable transmission (CVT), then through a fixed-ratio rear gearbox, and lastly, through the rear driveshafts. A 4 wheel drive (4WD) drivetrain additionally has a transfer case that splits power from the gearbox towards the front of the car, a belt or propshaft system to carry that power to the front, a front differential or gearbox to split power between the front wheels, and front driveshafts to carry power to the front wheels. In this project we will focus on the gearbox. A CAD model of the drivetrain for MBR 34 can be seen in Figure 2.3.1.

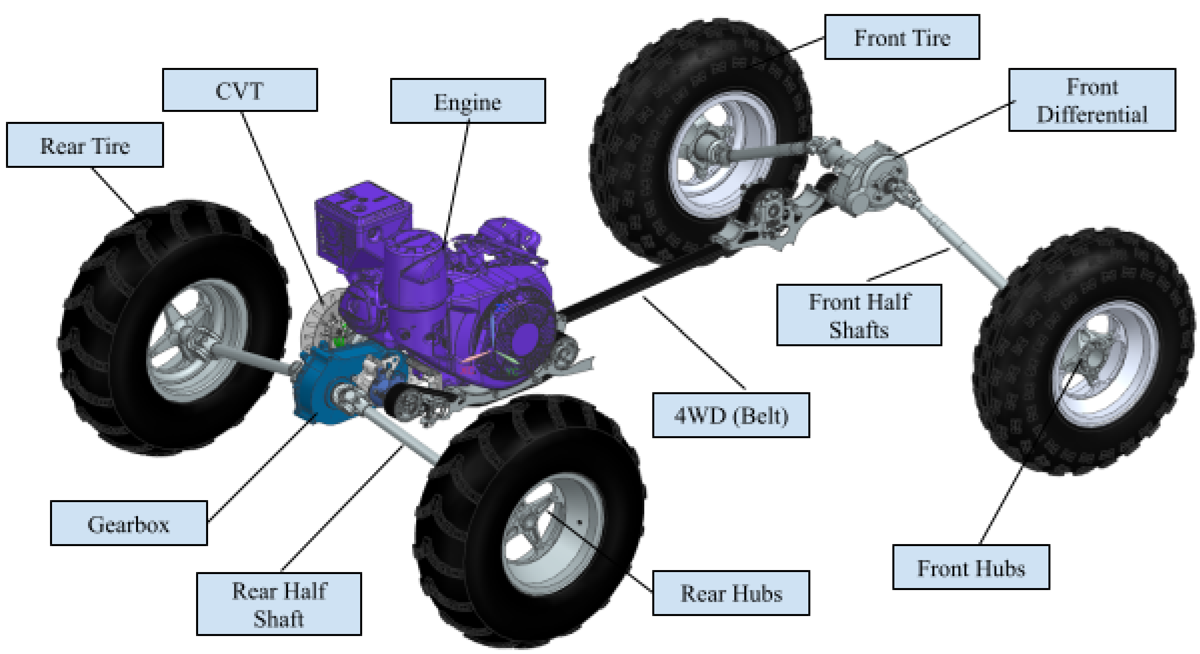



**Figure 2.3.1.** MBR 34 drivetrain CAD. Our area of concern for this project is the gearbox seen on the left side of the image.

The CVT, gearbox, and rear half shafts are all shown in figure 2.3.1 above. This project will focus mainly on the gearbox while keeping in mind its interface with the CVT, rear half-shafts, and rear brakes in an effort to improve MBR's transmission system.

**2.3.2 Engine Overview.** SAE dictates that each Baja Racing Team must use a Kohler Command Pro CH440 engine to power their cars. This engine is shown in Figure 2.3.2. below [3].


**Figure 2.3.2.** A Kohler gasoline, single cylinder engine Command Pro Small Horizontal series CH440 Model. This is the engine that SAE specifies all Baja teams must use.

The power and torque output from this Kohler CH440 engine is transferred to the CVT via its horizontal output shaft, then the gearbox, and finally the wheels. This engine must be the sole method of power generation for the car and cannot be modified. The engine can provide up to 14 horsepower, however, SAE limits it to 10 and has a peak power of 3.5 kW [4].

**2.3.3 Continuous Variable Transmission (CVT) Overview.** The continuously variable transmission (CVT) is the transmission used to route power from the engine crankshaft to the input of the gearbox, through an infinite number of gear ratios. A CAD model for the MBR CVT from the previous year's car is shown below in Figure 2.3.3.

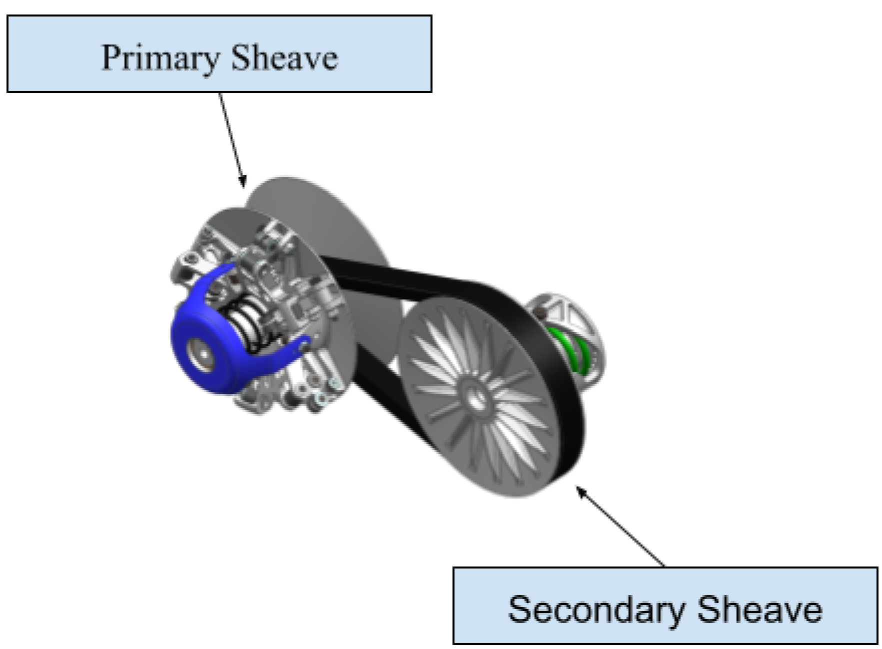


**Figure 2.3.3.** Custom CVT developed by the MBR Team. This CVT consists of two pulleys, called the primary and secondary sheaves, joined with a rubber v-belt.

The MBR CVT secondary sheave sits on the input shaft of the gearbox.

**2.3.4 Gearbox Overview.** The gearbox receives input power from the CVT and its output power is routed to the rear axle and 4WD system.. A CAD model for the MBR gearbox from the previous year's car is shown below in Figure 2.3.4.

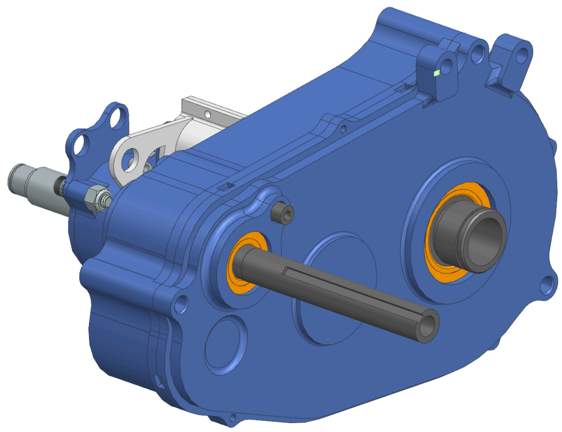


**Figure 2.3.4.** Custom gearbox developed by the MBR Team. The gearbox consists of a double reduction gear set with a forward gear train and a reverse gear train.

The gearbox's double reduction gear set is designed to act as final drive for the CVT. It contains oil and is sealed using a precision o-ring. The gearbox also has a split of power at the intermediate stage to a friction clutch and belt system that can transfer power to the front wheels. The friction clutch and main reduction box are joined through the use of dog clutches and a driver-actuated shifter fork.

A previous iteration of the car, MBR 33, split power at the intermediate stage to a transfer case that connected to a propshaft to transfer power to the front. This is likely what the team will be reverting to instead of a belt system.

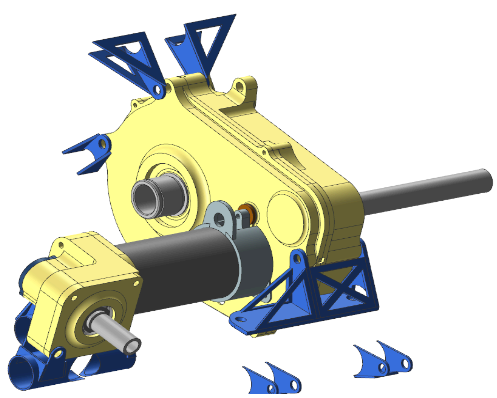


**Figure 2.3.4.** The gearbox consists of a single-speed, double reduction gear set that can split power to a transfer case.

**2.3.5 Rear Drive Shafts Overview.** The drive shafts send power from the gearbox to the rear wheels of the car. A CAD model for the MBR drive shafts from the previous year's car is shown below in Figure 2.3.5.

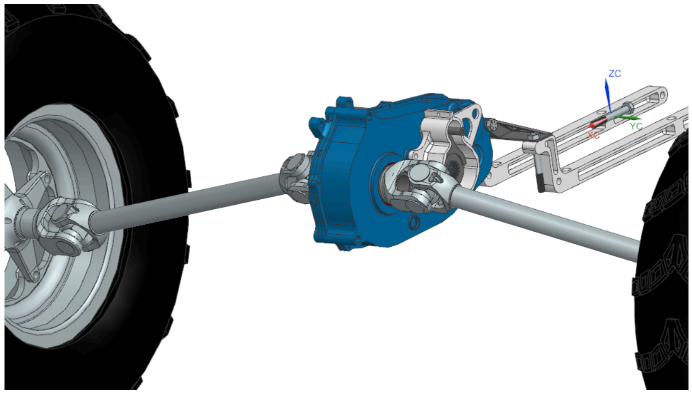


**Figure 2.3.5.** Custom drive shafts developed by the MBR Team. The drive shafts consist of an outboard u-joint that splines to the center of the wheel and an inboard u-joint.

The inboard u-joints of the drive shafts spline onto the output shaft of the gearbox and mount the rear brake rotor.

**2.3.6 Brakes Overview.** The brakes are the primary method of decelerating the car. A CAD model for the MBR rear brakes assembly from the previous year's car is shown below in Figure 2.3.6.

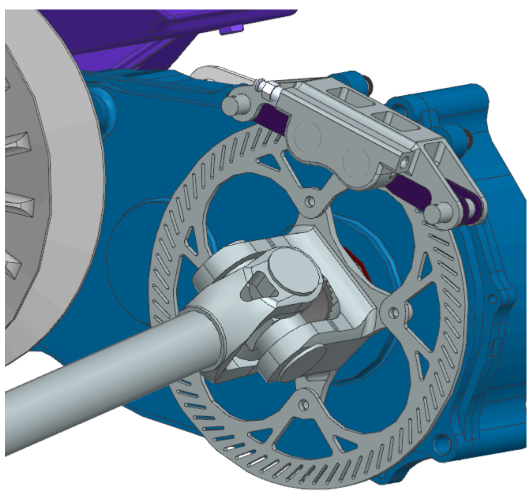


**Figure 2.3.6.** Custom rear brakes developed by the MBR Team. The brakes include floating calipers that hold brake pads in place and pistons that force the brake pads around a disc rotor attached to the wheels to decelerate the car through friction.

The rear caliper is mounted to the driver left side of the gearbox case.

**2.3.7 Project Motivation**

To determine where the most improvement could be attained, the team conducted a points breakdown of the past two competition seasons (2023 and 2024) which totals to 6 competitions examined. For each competition, the difference between the maximum number of points possible in each dynamic and endurance event and the score MBR achieved was calculated to determine where points were lost. Each of these differences was assigned a numerical causation code corresponding to the cause behind the loss of points including a large turn radius, lack of torque, CVT belt slip, and halted forward progress.

| Ohio Competition | Overall | Overall Dynamic | Overall Static | Adjustments | Cost Event | Design Event | Sales | Acceleration | Maneuverability | Sled Pull | Suspension | Endurance |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Points Possible** | **1037** | **280** | **357** | | **100** | **180** | **77** | **70** | **70** | **70** | **70** | **400** |
| | | | | | | | | | | | | |
| **Michigan Baja Racing** | 853.19 | 211.73 | 298.6 | 0 | 95.6 | 126 | 77 | 68.8 | 65.83 | 25.7 | 51.4 | 342.86 |
| Points Lost Quantity | 183.81 | 68.27 | 58.4 | 0 | 4.4 | 54 | 0 | 1.2 | 4.17 | 44.3 | 18.6 | 57.14 |
| Points Lost Percentage | | 37.14 | 31.77 | 0 | 2.39 | 29.38 | 0 | 0.65 | 2.27 | 24.1 | 10.12 | 31.09 |
| Cause | | | | | | | | 3 | 1 | 11 | 6 | 8 |
| Secondary Cause | | | | | | | | 7 | 2 | 4 | 1 | |

| Code | Reason | Points Lost | Percentage of Total Points Lost |
|---|---|---|---|
| | | | |
| 1 | High CG | 11.385 | 9.08 |
| 2 | Large turn radius | 2.085 | 1.66 |
| 3 | Low traction | 0.6 | 0.48 |
| 4 | CVT belt slip | 22.15 | 17.66 |
| 5 | Halted forward p | 0 | 0 |
| 6 | Other (Driver err | 9.3 | 7.42 |
| 7 | Speed | 0.6 | 0.48 |
| 8 | Durability | 28.57 | 22.78 |
| 9 | 4WD Engageme | 0 | 0 |
| 10 | 4WD Efficiency | 0 | 0 |
| 11 | Lack of Torque | 22.15 | 17.66 |

**Figures 2.3.7 (top) and 2.3.8 (bottom).** The points breakdown from the Ohio competition of the 2023 season illustrates a lack of torque in the sled pull event to be one of the most costly reasons for points lost.

Through this analysis, the team found that across both seasons, an average of 20% of points were lost due to lack of torque, especially in the sled pull event. Furthermore, during the 2023 season, 25% of points lost were from halted forward process, which guided the implementation of a reverse gear in the 2024 vehicle. With this new addition, only 3% of points lost in the 2024 season were due to halted forward progress, encouraging the team to keep reverse in future iterations of the vehicle. Based on this points breakdown, the team has decided it stands to gain the most points from resolving the issue of lack of torque outputted by the gearbox.

**2.3.8 Problem Statement.** The Michigan Baja Racing Team's transmission architecture has largely been stagnant over the past few seasons. In order to stay competitive with other teams, an investigation into more mass efficient and capable gearboxes is necessary. Through thorough engineering analysis, the team will conceptualize, verify, and validate a design solution that increases the packaging performance & mass efficiency of the transmission, while maintaining current capabilities (gear ratio, reverse gear, 4WD integration, etc.). A more capable gearbox will allow the team to yield a higher performing car in competition events via more customized gear ratios for each event such as sled pull, which requires a higher gear ratio to increase torque output.

## 2.4 Project Goals

This project exists in order to assist our sponsor, the Michigan Baja Racing team, build a more competitive race car. The motivation for this project is to decrease the number of points lost per competition from inadequate torque delivered to the wheels. Our team will help MBR improve the performance of their car by re-evaluating its gearbox design. MBR has also seen recent improvements in design that have decreased overall points losses, and these improvements must be retained to avoid increasing point losses in other areas.

The major objectives of this project are to design a gearbox for the MBR car and prototype it. A successful outcome for this project will be determined by a completed design for the gearbox, a manufactured prototype of the design, and a plan in place for the construction of the final design.

## 2.5 Information Sources

We gathered most of our information by interviewing relevant stakeholders, which include interfacing substem leads and manufacturers (see section 3.2). The Baja team also has a repository of previous projects which were referenced as research done to improve the gearbox and other drivetrain subsystems. In addition to reviewing previous work done by our team, we conducted research on commercially available gearbox solutions for passenger vehicles, race cars, and go-carts. This market research is outlined in detail in the benchmarking information found in Section 3. The Baja SAE rulebook is also an information source, providing a source of requirements that must be adhered to by all competing teams in order to be eligible to compete at competitions. The most relevant section of the rules pertaining to the gearbox are the hazardous release of energy (HROE) guarding requirements that outline the minimum strength and thickness allowable for safety guarding that shields the driver and spectators from rotating components. We plan to source technical content from textbooks such as Shigley's Mechanical Engineering Design and Dudley's Handbook of Practical Gear Design and Manufacture. The ME 450 course content has also inspired our design process framework, as well as the process to define stakeholders and determine requirements and specifications. As the project progresses, we may incorporate standards, such as a 1.2 safety factor for stresses, depending on the relevant application. The sources for such standards used in the future will be identified as we determine which are applicable.

## 2.6 Design Process

At this stage in the design process, we have followed the ME Capstone design process framework [5] laid out by the ME 450 instructional team. The framework diagram is shown below in Figure 2.6.1:

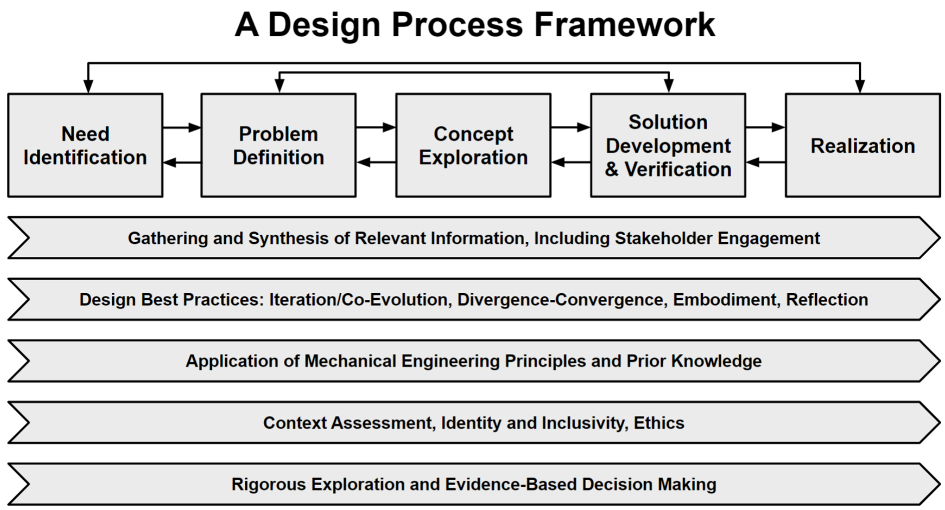



**Figure 2.6.1.** A diagram showing the ME Capstone Design Process Framework outlined in the learning blocks for ME 450 [6]. This solution-based and iterative system closely matches MBR's design process.

Our team has found this framework is compatible with our project due to its emphasis on solution-based design and cyclic nature of being able to repeatedly go back and rework the design. Several gearbox projects have been conducted by MBR that used a variety of different types of gearboxes making it possible to iterate on those designs and take key components of each that we found worked well. With these resources available and a solution-oriented approach, the proposed design of the gearbox can constantly be modified throughout the design process as requirements are further evaluated and updated [7]. Before settling on this framework, the team also considered several other design frameworks shared in the ME 450 course resources such as the five-stage Dym and Little design process model. This is a more linear and chronologically sequenced method from need identification to the final design [8]. However, these steps left little room for design iteration possible with more of a problem-based approach.

From the considerable amount of resources available to us, we will use the ME Capstone Design Process Framework with an emphasis on iteration between concept exploration and solution development. This ties in well for our project because we already have previous MBR gearbox designs to act as a starting point, so we will mostly be iterating on the previous MBR gearbox design while keeping elements that worked well in them while incorporating new elements to enhance its capability. To ensure the gearbox stays competitive with those of our peer teams, we expect to repeatedly refine our requirements and specifications throughout the design process. Considering this, we find that the ME Capstone Design Process Framework will best enable us to reflect these changes in our design, making it a good fit for our team and our project goals. [1]

# 3 Research and Benchmarking

## 3.1 Current Gearbox Benchmarking Standards

### 3.1.1. Gearboxes on the Market

The Baja SAE series is a collegiate design series where the vehicles built by competing teams are typically built with many, if not all, custom components. Michigan Baja Racing has utilized custom transmissions for over a decade, designing and machining much of the drivetrain themselves. However, there are many ways that their transmissions and other teams' custom transmissions are influenced by commercially available components & assemblies.

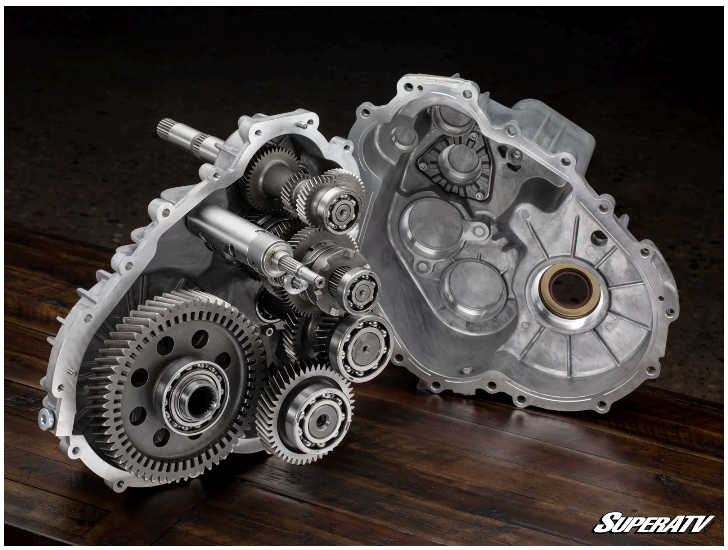



**Figure 3.1.1** Polaris RZR XP 1000 Transmission

Many teams utilize Polaris ATV style transfer cases, shown in Figure 3.1.1, to distribute power to their rear wheels & to 4WD. These transfer cases cost around $3,000-$5,000 and are around 50 lbs fully assembled [9]. They utilize a cast aluminum or steel housing, steel shafts to transfer torque from input to output, and steel gears. They also have multiple forward gears, assembled side to side, to give multiple forward ratios and one reverse ratio. Other similar transmissions are produced by ATV and side-by-side companies such as Honda, Can-Am, and Yamaha, utilizing similar architecture.

### 3.1.2. Existing Baja Gearbox

Since 2013, Michigan Baja Racing has run a transmission consisting of a CVT transmitting engine torque to a 2-stage spur-gear gearbox. This architecture was first designed by a 450 project team in 2012 [10]. Prior to that, the team had run both a purely belt driven and/or chain transmissions that were great for packaging and vehicle architecture, but often had failures due to shock loading, and were not very efficient. Since 2012 the transmission has gone through some slight iterations in areas such as gear ratios, gear pocketing, and gearbox case geometry. However, no big changes were introduced to the fundamental design until the 2024 transmission, where an extra gear train was introduced to obtain reverse capabilities.

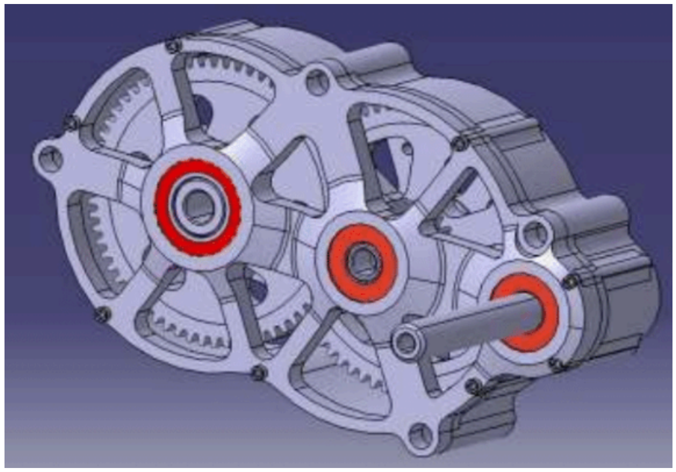


**Figure 3.1.2** MBR's first ever gearbox from 2012 [10]

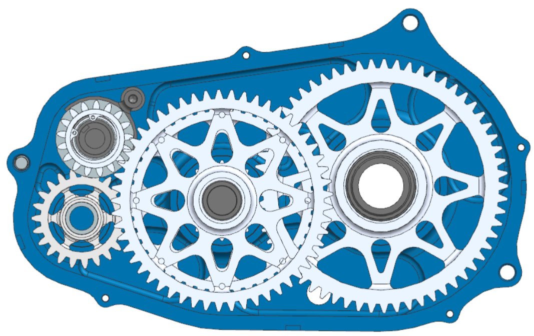


**Figure 3.1.3** 2024 gearbox with reverse capabilities

### 3.1.3. Existing Transmission Designs

In order to gain inspiration for design solutions, we looked at existing transmissions in different non-Baja SAE vehicles, which see similar styles of power transmission as MBR's, including side-by-side ATV's, motorcycles, go-karts, and sports cars.

A majority of side-by-side vehicles follow the same main architecture that MBR has been utilizing: a CVT to a final reduction gearbox (See figure 3.1.1). This setup allows the vehicles to automatically and smoothly continuously shift into whatever overall reduction is needed to keep the vehicle moving at its maximum engine power output. This helps eliminate driver error in shifting, keeps the engine at its peak output power, and maintains high efficiency, which is why MBR has used a similar system for the past 12 years.

Motorcycles such as the Honda CBR1000 [11] don't use a CVT to transfer power from the engine to the drivetrain and perform shifting, but rather use a multi-plate friction clutch and a manual transmission with between 4 and 8 gears to transfer power at different speeds and torques. Manual transmissions with clutches are harder for the driver to use consistently, requiring practice and familiarity with the system, and have predetermined ratios for torque and speed. This means the driver will not always be driving at peak power, and errors in shifting can cost time when driving at the limit. However, these manual transmissions do utilize a very unique method of shifting: dog gears. As seen in Figure 3.1.4, dog gears allow power transmission between the faces of two bodies rotating along the same axis. They are used in manual transmissions, sequential transmissions, and some automatic transmissions in cars, motorcycles, and other vehicles. This way of transferring power in a binary on/off state has been used in MBR's 4WD system and more recently has been introduced to switch between the forward and reverse gear trains in 2024.

**Figure 3.1.4** Honda CBR1000 Dog Gears

Sequential Gearboxes utilize a component called a "selector drum", shown in Figure 3.1.5 [23]. This selector drum allows for any number of shifting forks internally to be controlled by one external input. The driver inputs a certain angle of twist on the selector drum via a ratchet gear knob, and as the drum twists, it moves certain shifter forks along its length to engage or disengage them in a predetermined order, which is the sequential selection of the gears.


**Figure 3.1.5** Honda Recon 2x4 Transmission Selector Drum

Finally, spur gear trains are used in many different configurations, including planetary style gearboxes in automotive & go-kart transmissions. Planetary transmissions provide a gear reduction in a smaller overall package at the cost of more complex parts. They are also useful when different components are locked against each other, seen in the GY6 Go-Kart Planetary gearbox (Figure 3.1.6) [12]. This gearbox allows for the same components to be used to transmit torque in the forward direction and in reverse by locking the carrier to the ring gear for forward, and to the case for reverse.

**Figure 3.1.6** GY6 Go-Kart Reverse Gearbox

## 3.2 Stakeholder Analysis and Interviews

Our stakeholder map includes primary stakeholders who would be most affected by our design, secondary stakeholders who may be indirectly affected by our solution due to being part of the problem context, and tertiary stakeholders who are outside of the problem context but may still have some influence on the problem. Our stakeholder map also has another categorization system that includes Resource Providers, Supporters & Beneficiaries of the Status Quo, Complementary Organizations and Allies, Beneficiaries and Customers, Opponents & Problem Makers, and Affected or Influential Bystanders. Resource Providers may provide financial aid or other services for the project, knowledge or expertise, or technological resources. Supporters & Beneficiaries of the Status Quo are those who benefit from there being no change or new solution. Complementary Organizations and Allies may have some influence on the process and also may be working to solve the same problem. Opponents & Problem Makers play a part in the problem or directly or indirectly work against any solution being designed. Affected or Influential Bystanders currently do not have influence over the solution, but may have influence or be affected in the future [13] [1].

Our stakeholders are largely in support of our design, all of them are actively in support of our design or ambivalent to it unless otherwise stated. Our primary stakeholders are the MBR team and MBR endurance driver as Beneficiaries & Customers because they will be receiving our end product to integrate within their vehicle and the driver will be actively using the gearbox and shifting between its different gears while driving in each event. Current MBR subsystem leads will make up our Supporters and Beneficiaries of the Status Quo because no change in the gearbox means they do not need to change their subsystems at all to integrate with the new design. Our secondary stakeholders include Chardam Gear as a Resource Provider since they grind our gears for us and other manufacturers for the team (both machinists within the team and manufacturing sponsors) who would benefit from no change in the design as parts from the previous year's gearbox could be recycled for the new car reducing their workloads. Interfacing Subsystem Leads such as for the CVT, Brakes, and Half Shafts may additionally act as Opponents and Troublemakers because any changes they make to their designs could affect how their subsystems interface with the gearbox and require a corresponding change within the gearbox mounting design. Finally, Complementary Allies and Organizations include Baja SAE and other baja teams because they are not directly affected by the gearbox, but inspiration could be taken from the types of gearboxes they use. For tertiary stakeholders, other Wilson Center Teams and Chris Gordon are Affected or Influential Bystanders because MBR must work with those other teams and Chris to gain time on in-house machines that would be used to manufacture the gearbox such as the 3-axis mill. UM COE is a Resource Provider from the funding they provide MBR and SAE technical inspectors could serve as Opponents and Problem Makers based on the rules they set for transmission designs and any differences in interpretation of those rules that may arise. Our stakeholder map can be seen below in Figure 3.2.1.

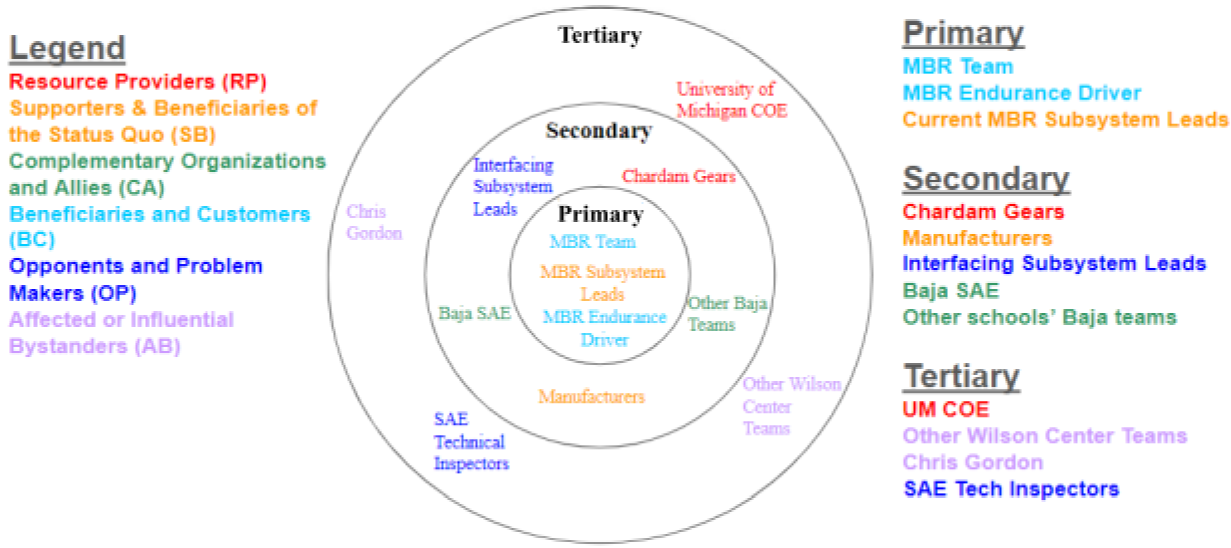



**Figure 3.2.1.** Our stakeholder map in which we analyzed all of our stakeholders and thought about their relevance and influence on this gearbox solution and on MBR performance at competitions. We only interviewed the primary stakeholders as they are the only ones who will be directly affected by this solution and will be either using it or designing their components to integrate with it.

**3.2.1. Gearbox Subsystem Lead.** According to the MBR Gearbox lead, the past transmissions have been efficient at delivering enough power to the rear wheels and contributing to the overall vehicle speed. However for maximizing points in a competition setting a transmission or gearbox that focuses on adding capability to the gearbox and therefore the vehicle as a whole. Changing the capabilities might have to mean to try and experiment with potentially a completely different transmission concept. Through a model of the drivetrain, we have looked into different possible ratios and compared their acceleration times across 100ft, 200ft, and 1000ft. A ratio of about 9:1 gives the quickest 1000 ft acceleration time as the highest top speed, while a ratio of around 13:1 gives the quickest acceleration time across 100ft. Having a ratio lower than 9:1 does not give any more marginal gains to 1000 ft acceleration time, as the air drag term exponentially drowns out the increase in ratio. While having a higher reduction would provide more torque we have calculated that any ratio higher than 13:1 would cause packaging issues from either making the gearbox too long or too tall. This leads to the main difficulty with experimenting drastically is being able to fit within the already existing integrating subsystem leads such as brakes, CVT, and Half Shafts. Since that leaves the subsystem with very limiting freedom dimension wise. If we do not fit within these constraints then the vehicle as a whole would most probably need a complete redesign which in that case makes experimenting with the transmission not worth the trouble and potential issues. Point gains were seen through the use of reverse in suspension & traction and endurance events, where the driver was able to extricate themselves from an obstacle and continue forward. Eliminating reverse will cause excessive point losses, as seen in previous seasons, and is not an option for any future design. [14]

**3.2.2. Endurance and S&T Driver.** Even though our current transmission has great overall performance, creating a more adaptable system could increase vehicle performance according to our S&T and Endurance driver. Since no one from the team knows what the course is going to look like until we get there, having more options for the driver could make us more prepared no matter the terrain, geography, or unique obstacles created. There have been some competitions that involve a lot of uphills or difficult obstacles where we would benefit from having the transmission deliver more torque. On the other hand, there have been some other competitions where the courses are relatively flat and have easy obstacles where speed is more important than torque. If we had an adaptable system, we can be more prepared and not have to worry about the inconsistencies between competition courses throughout the season. A problem to think about if we do end up developing a more adaptable transmission that can shift in and out of the gear ratios would be the ease of shifting. In the previous vehicle the gearbox was able to shift between forward and reverse and due to the design mechanism it was sometimes a lot of effort to shift while driving. With the potential that a more capable transmission might introduce different speeds and/or reverse, shifting it should not require more force than about 10 lbs., anything greater than that might make it difficult to shift and take away the driver's focus from what is in front of him. [15]

**3.2.3. CVT Lead.** According to the CVT (continuous variable transmission) lead, while experimenting with different types of transmissions to provide more capability to the vehicle, any new concept that gets rid of the CVT would not be overall beneficial to the vehicle performance. A CVT allows the vehicle to react to any new reacting forces through backshifting and is shifted all the way from reduction to overdrive within seconds creating an extremely efficient system. For a vehicle our size and with its performing parameters it would make a lot of sense to utilize a CVT. So it would be highly recommended that any new concept or iteration of our transmission mostly involve the gearbox and not the CVT. The CVT, however, does need to be taken into consideration when designing the gearbox because it controls how far from the engine crankshaft the gearbox will be located. This distance is controlled by the size of the CVT and its belt and must be accommodated to allow packaging room for the CVT. Furthermore, we learned from the CVT lead that there is a maximum input torque the gearbox can receive from the CVT that all gearbox components must be able to withstand. These constraints will be taken into account for our final design. [16]

**3.2.4. Brakes Lead.** The current rear brake assembly is mounted inboard on the car, with the rotor mounting on the inboard axle yoke and caliper mounted to the gearbox case. Packaging the assembly around drivetrain and chassis components is challenging, and according to the current brakes lead, many issues have been caused with the caliper & rotor hitting frame tubing or gearbox tab mounting. This can be caused by excess width in the gearbox forcing the caliper to move further outboard, therefore putting in contact with the frame tubing surrounding the rear of the car. To remedy this, we will impose a limit on the width of the final design to maintain our current caliper & rotor placement, and guarantee their functionality. [17]

**3.2.5. CNC Machinists.** The current gearbox we make is probably one of the most complex and time consuming subsystems we have to machine even when considering the sponsors we have at hand. From pocketing the numerous gears, making the gearbox case, to machining all the shafts and spacers for the gears uses the most capable set of machines that the Wilson Center has to offer. With a new redesign of the transmission the designer has to take into account the capabilities that our shop has to offer. With a more complicated design that would have to involve a 5 axis CNC Machine the team would have to outsource which is not a viable option due to the budget and timeline constraints. [18]

**3.2.6. Rear Half Shafts Lead.** The current rear half shaft also acts as the long arm for our suspension geometry. Because of this, it not only sees the torsional loads provided by the gearbox, but also axial loads from the suspension. We have calculated from a mathematical model that the theoretical maximum axial load case would be 17 kN. Since the half shaft is directly mounted to the gearbox, the output shaft and gearbox case will also have a maximum axial load case of 17 kN. [19]

## 3.3 Broader Design Context

Most of our stakeholders are ones who are directly affected by the outcome of our design (see Figure 3.2.1). However, we acknowledge that our design may have unintended effects on other groups not currently listed in our stakeholder list. In our list of stakeholders, most will be affected positively by our design. Notable exceptions are the potential opponents and problem makers, namely the SAE technical inspectors and interfacing subsystem leads. SAE technical inspectors are often resistant to changed designs, and sometimes impose amendments to the rules based on teams' innovations. Individual SAE technical inspectors are perhaps the hardest (practically impossible) stakeholders to communicate with, since they are randomly sourced from volunteer Baja SAE alumni. To circumvent this issue, SAE provides a rules clarification service in which teams can submit questions to the lead technical inspector about the rules to preempt any sort of tech inspector resistance. Our team will use the rules clarification service in the event that we need a clarification on the rules to ensure that an aspect of our design is rules compliant. Interfacing subsystem leads are also potentially negatively affected by our design since they will need to change legacy team designs to accommodate a new gearbox design. In contrast to the individual SAE technical inspectors, interfacing subsystem leads are easy to communicate with via Slack or in person meetings. We will work closely with interfacing subsystem leads to ensure that our project has minimal negative impact on their work.

Broadly speaking, there is no reasonable social or societal aspect of the problem that is driving this work to be done beyond the interests of our sponsor. Our sponsor is a collegiate racing team and the Baja SAE competition affects a very small subset of society: students involved with the competition and supporting industries and sponsors. While social impact is very important to the members of our ME450 team, in communicating with our sponsor it is clear that social impact ranks very low on their list of priorities, and is not usually considered in design. Team performance is the sponsor's highest priority which is reflected in the prioritization of functionality, quality of design, and speed of delivery for our project. Despite the lack of prioritization of social impact, our team will try its best to minimize social and environmental impacts of our project. For the duration of this semester, the project will be manufactured by the members of our ME450 team and should have minimal impact on the jobs of salaried or hourly workers either at the University of Michigan or at any external sponsors/suppliers. In addition to complying with the Baja SAE HROE requirements, during the design phase we will be conscious of the safety of machinists at each manufacturing operation, and will also be conscious of the safety of mechanics servicing the gearbox. At the end of its life, our gearbox will likely be recyclable, increasing the sustainability of our project. If not recyclable, the end of life cost of our gearbox will be minimal due to its small size. Gearboxes almost invariably use oil or grease for lubrication–our team is committed to protecting the environment and will do so by disposing of any petroleum products such as used oil, or gasoline through the proper Environmental Health and Safety channels. We will also avoid wasting raw materials in order to conserve the sponsor's financial resources as well as the environment. Our sponsor will require limited production runs of any prototype we produce. As with any sort of hardware production, manufacturing processes will take some amount of energy. Our team will minimize our energy consumption with our limited production run. The main environmental impact of our project is likely the use of metallic materials which have a high environmental footprint, from mining to processing, all the way to machining and heat treatment. In choosing more sustainable materials such as wood, our project requirements would not be met due to the stringent packaging constraints. While using non-metallic materials would increase project sustainability, it is not feasible for this project.

The intellectual property produced by this project will belong to the members of this ME450 team. As such, we do not have an intellectual property agreement with our sponsor, the MBR team. However, there is an expectation of discretion, meaning that MBR's direct competitors should not have access to any designs that may give MBR an advantage over other teams. If this project yields a novel product, it would be reasonable to apply for US and international patent protection.

We hope to not face many ethical dilemmas in the design of our project. However, a potential ethical dilemma we may face is enlisting MBR team members to assist in the manufacturing of our project. We will deal with this ethical dilemma by ensuring that the majority of the manufacturing work is done by ourselves, unless specialized expertise is required in the manufacturing of our product. We expect this dilemma to be kept to a minimum due to the depth of manufacturing experience present on our team. Our team ethics align well with the University of Michigan's ethics as well as future employers' ethics–there will be absolutely no forced labor during any phase of our project, and precautions will be taken to protect the work environment.

Our project team is made up of the current technical director of the MBR team, the current captain, and two former leads on the MBR team. This creates an obvious power dynamic between our project team and the rest of the MBR team, especially in terms of perceived technical authority and seniority. Our project team is aware of this power dynamic and will make an effort to not create a power imbalance. Our efforts will include listening to stakeholders on the MBR team and receiving feedback from interfacing subsystem leads, ensuring that the sponsor has as much influence as appropriate. Within our project team, we are not worried about power imbalances due to the entire team having worked with each other on the Baja team since freshman year. In terms of inclusivity, when interacting with our sponsor we will take care to identify any unconscious biases we may have, and to include as many voices as possible when seeking feedback.

# 4 Requirements and Engineering Specifications

## 4.1 Requirements and Specifications

After speaking with the sponsor and stakeholders of our project during the interview process described in section 3.3, our group developed a list of high level requirements. These high level requirements are listed in table 4.1 below. The high level requirements were each expanded into multiple sub-requirements, each with a corresponding specification and justification. Each specification was driven by either external research relating to the system being analyzed, or internal research relating to one of the subsystems that makes up the MBR race car. The sub-requirement breakdowns may be found in tables 4.1.1-4.1.8.

**Table 4.1.** Stakeholder requests and high level requirements with sub-requirements.

| *High level requirements with sub-requirements* |
|---|
| The MBR Gearbox must have:<br>● High Efficiency<br>    ○ Limited power loss<br>● Backwards compatible packaging<br>    ○ Packages with CVT<br>    ○ Output aligns with inboard half shaft points<br>    ○ Packages within past vehicle frames<br>● High durability<br>    ○ Forward driveline components survive the duration of a competition<br>    ○ Reverse driveline components survive the duration of a competition<br>    ○ All bearings survive duration of a competition<br>● Load Capability<br>    ○ All components must have a safety factor<br>    ○ Withstand loading from rear half shafts<br>    ○ All components withstand maximum input load from CVT<br>● Gear Ratio Functionality<br>    ○ 3 speed ratios (high, low, reverse)<br>● Manufacturable given current capabilities<br>    ○ All milled components manufacturable by machinery in Wilson Center<br>    ○ Out of house manufacturing can be done by existing team sponsors<br>    ○ All fillets machinable with current tooling<br>● Ease of Use<br>    ○ Driver can shift between gears easily<br>● Light-weight |

**4.1.1. The gearbox must have high efficiency.** The engine only outputs 10hp so we need to maintain as much power as we can to have optimal speed and torque output at the wheels.

**Table 4.1.1.** Requirements and specifications for the efficiency requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Limited Power Loss | Input to output power loss < 15% | Assuming a gear efficiency of 98% and ball bearing efficiency of 99%, the current gearbox has a ~95% efficiency. If the new gearbox has an efficiency <85%, it will not be considered for being used on the vehicle. | Strain gauge the rear half shaft and get the engine torque at that rpm to calculate efficiency<br><br>Test on fully built gearbox once assembled into the vehicle |

**4.1.2. The gearbox must have backwards compatible packaging.** To make full-scale testing of the gearbox once it is built simpler, it needs to be able to package within the architectures of previous vehicles builts by MBR.

**Table 4.1.2.** Requirements and specifications for the backwards compatible packaging requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Packages with CVT | 8.5in distance from input to engine crankshaft | CVT needs 8.5" of space from the engine crankshaft to gearbox input shaft based on its belt size | Integrate into previous vehicles' CAD<br><br>Check when designing gearbox and verify by installing within the vehicle once the gearbox is fully built |
| Output aligns with inboard half shafts points | Between 5.75in & 6.5in distance from input to output | Rear half shafts (set by suspension kinematics) are placed roughly that far behind the input shaft of the gearbox | |
| Packages within past vehicle frames | Fastens to 3 existing fastener holes<br><br>< 2.5in wide (laterally in car) | Can be installed with the same mounting as older vehicles and serviced easily<br>Fits within frames of older vehicles with space for the rear brake caliper | |

**4.1.3. The gearbox must have high durability.** The gearbox must be durable for the length of the competition otherwise it will cost the team driving time during endurance to fix it and points.

**Table 4.1.3.** Requirements and specifications for the durability requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Forward driveline components survive the duration of a competition | Survive at least 10^6 wheel revolutions | Assume 6 hours (length of a competition) of forward driving time at 25 mph with a 11.5" radius wheel | Perform a mock competition (endurance, accel, S&T, etc.) and examine all components afterwards to search for any signs of failure<br><br>Test on fully built gearbox once assembled into the |
| Reverse driveline components survive the duration of a competition | Survive 10^4 wheel revolutions | Assume 30 minutes of driving time in reverse at a competition at 5 mph with a 11.5" radius wheel | |

| | | | |
|---|---|---|---|
| All bearings survive duration of a competition | Bearings rated to C10 lifetime of 10^6 revolutions at maximum loads | Assume 6 hours of driving time at 25 mph with a 11.5" radius wheel | vehicle |

**4.1.4. The gearbox must have load capability.** The gearbox must be able to survive its worst case loadings during competition as having to fix it takes time away from actively competing and earning points.

**Table 4.1.4.** Requirements and specifications for the load capability requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| All components must have a safety factor | SF of 1.2 on maximum load cases on all components | Team standard safety factor for drivetrain components | Verify each component has 1.2 SF analytically and conduct drop testing (worst case loads) |
| Withstand loading from rear half shafts | 17kN axial loading | Maximum loading from axially-stressed drive shafts | Drop test with fully built gearbox once assembled into the vehicle and conduct SF verification during analysis |
| All components withstand maximum input load from CVT | 54 ft-lbs of torque | Input torque from CVT at max reduction from engine | |

**4.1.5. The gearbox must have gear ratio functionality.** Having different gear ratios the driver can switch between will allow him to select the best ratio for each event.

**Table 4.1.5.** Requirements and specifications for the gear ratio functionality requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| 3 speed ratios (high, low, reverse) | Low gear reduction of ~9:1<br><br>High gear reduction of ~13:1<br><br>Reverse gear reduction greater than 2.8 | Based on points breakdown: high gear to increase torque (sled pull, etc), low gear to retain points in higher speed events, reverse to maintain reverse capabilities | Analytically confirm gear ratios (based on tooth count) during design<br><br>Confirm gearbox outputs at 3 speeds with the prototype |

**4.1.6 The gearbox must be manufacturable given the team's current capabilities.** This will significantly reduce the costs of manufacturing the gearbox.

**Table 4.1.6.** Requirements and specifications for the manufacturability requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Milled components manufacturable by machinery in Wilson Center | Manufacturable by <= 4 axis machinery | The team will not have to pay extra for manufacturing of gearbox components or search for additional sponsors | Confirm with team machinists throughout design process and after design is completed |
| Out of house manufacturing can be done by existing team sponsors | 0 new manufacturing sponsors needed | | |
| All fillets machinable with current tooling | All fillets of radius >=0.125in | | |

**4.1.7 The gearbox must be easy to use.** The driver will have to shift with one hand and drive simultaneously.

**Table 4.1.7.** Requirements and specifications for the ease of use requirement.

| *Sub-requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Driver can shift between gears easily | < 10lbs of shifting force | Driver ergonomics considerations | Put a force gauge on linkage after gearbox and vehicle are fully built and test how much force it requires to shift |
| | 0.25in separation between dog gear engagements | Ideal dog gear separation derived over past years of use & packaging constraints | |

**4.1.8. The gearbox must be light-weight.** The vehicle is already rear heavy and adding too much additional weight to the gearbox will impact its overall speed.

**Table 4.1.8.** Specifications for the light-weight requirement.

| *Requirement* | *Specification* | *Justification* | *Testing/Validation Method* |
|---|---|---|---|
| Light-Weight | Weighs <11 lbs | No more than 2 lbs greater | Check CAD mass estimation |

| | | |
|---|---|---|
| | than current gearbox | during design and confirm on scale once full scale gearbox is built |

# 5 Concept Generation

To begin the concept generation process, the team met together and decomposed the overall design problem into several smaller sub problems. Starting with the engine output and choosing the type of clutch that is used to transfer that power, we separated each distinct subsystem from the overall system. Moving from the engine and into the gearbox, we designated the internal reduction method, which sets how speed is stepped down and torque is stepped up from the clutch's output, as its own subsystem. Following the internal reduction method, the way to transfer torque within the components in the selected reduction was another subsystem. Once those are determined the implementation of the multiple ratios within the reduction method and which shifting mechanisms are going to be used to select and power the desired ratio need to be figured out.

Once the transmission was divided into these five smaller problems to solve, we organized them into a morphological chart specifying a few potential solutions for each sub function displayed in Table 5.1.1.

**Table 5.1.1** Our morphological chart in which we brainstormed the different types of clutches, internal reduction methods, internal component torque transfer methods, shifting mechanisms, and methods of implementing multiple ratios that could be implemented in our design.

| *Sub- Functions* | *Solutions* | | | |
|---|---|---|---|---|
| Clutch Method | CVT | Tension Belt | Centrifugal Clutch | Hydraulic Torque Converter |
| Internal Reduction Method | Gears | Belt | Chain | Magnetic Gears |
| Internal Component Torque Transfer Method | Keyed Connection | Splines | Polygon | Welded |
| Shifting Mechanism | Dog Clutch | Friction Clutch | EM Clutch | Synchromesh |
| Method of Implementing Multiple Ratios | Stacked Trains | Parallel Trains | Continuously Variable Shifting | |

We aimed to come up with three to four solutions for each sub task that could be combined in different permutations to generate complete designs.

After the morphological chart was completed, we split up and individually generated 20 concepts each, with a focus on not rejecting any idea no matter how extreme or improbable. Along with deferring judgment, we emphasized quality over quantity of our ideas, using more visual means such as sketches to convey our solutions, and building on previous ideas, and combining previous ideas. These tools enabled us to explore the solution space more thoroughly and break away from conventional solutions. Team members also made use of their own morphological charts and design heuristics like using one component for multiple functions, twisting one idea to create another, and applying existing mechanisms in a new way. All of these ideas are listed out in Appendix B.

Once we spent some time generating solutions separately, we came back together during a brainstorming meeting to go over them as well as come up with a few more together. We started by reviewing our ideas and then used a few more design heuristics to build on them. A few examples of these are illustrated below in Figures 5.1.1 a-c where we made use of the heuristics stack, rotate and nest.

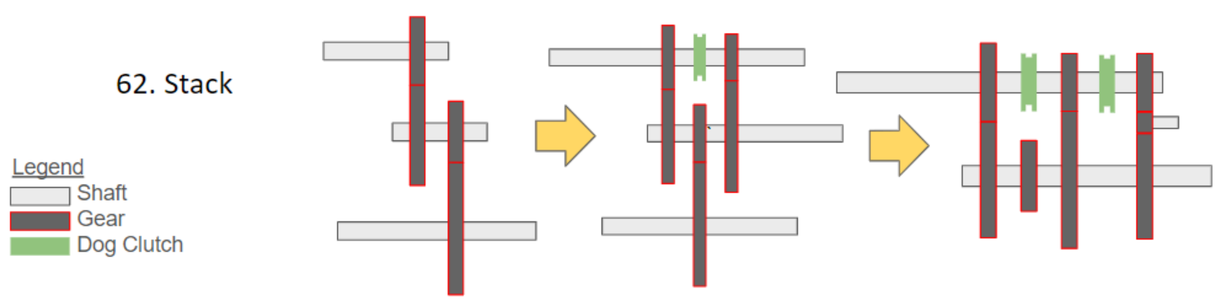



**Figure 5.1.1**a MBR has previously used the stacking heuristic in its designs when creating its 2024 gearbox (middle) from its 2023 gearbox (left) by stacking an additional gear train to introduce a reverse gear. One idea the 450 team had was to employ this again by stacking another gear train onto the 2024 gearbox to introduce the high torque gear (right).

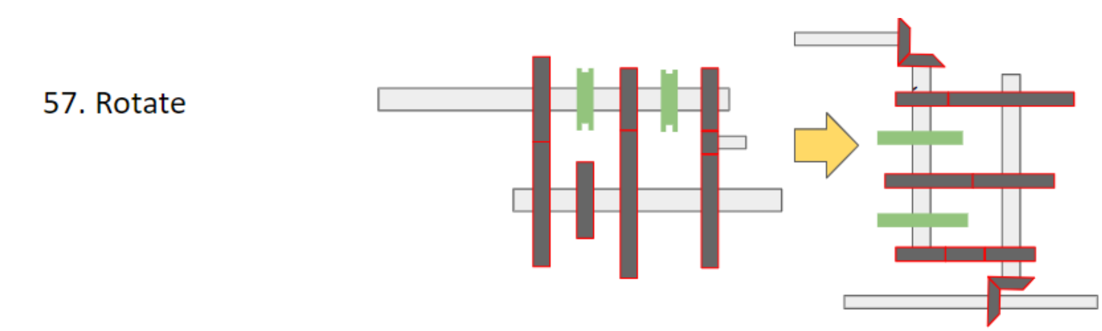



**Figure 5.1.1**b The rotation heuristic was used on the previous concept by rotating it 90 degrees to create more of a sequential-style gearbox that uses bevel gears and has intermediate shafts sitting perpendicular to the input and outputs, rather than parallel to them.

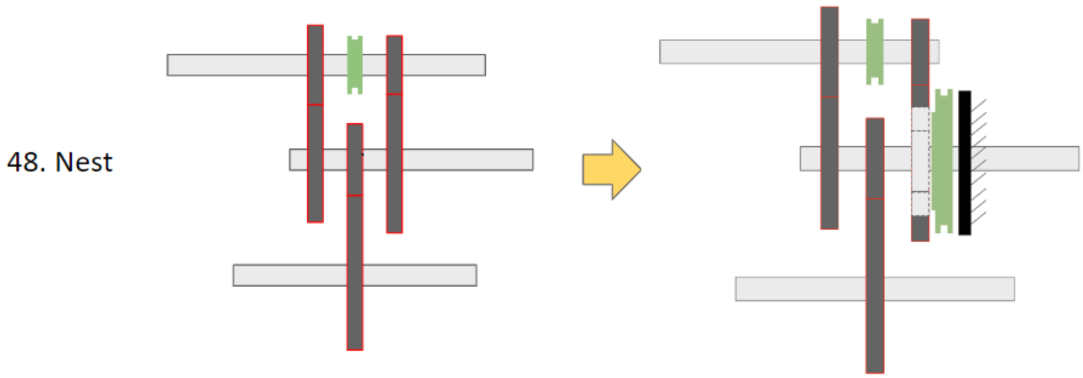



**Figure 5.1.1**c Another idea was generated by nesting a planetary gearbox within the 2024 gearbox on the intermediate shaft that would allow shifting between the reverse gear and high torque gear.

# 6 Concept Selection

After all of the group's concepts were generated, they were organized and narrowed down by utilizing a concept tree that visibly displays all the different types of methods that can potentially be used to solve our design problem and how they branch from each other to set them apart. This can be seen in Figure 6.1.1 below.

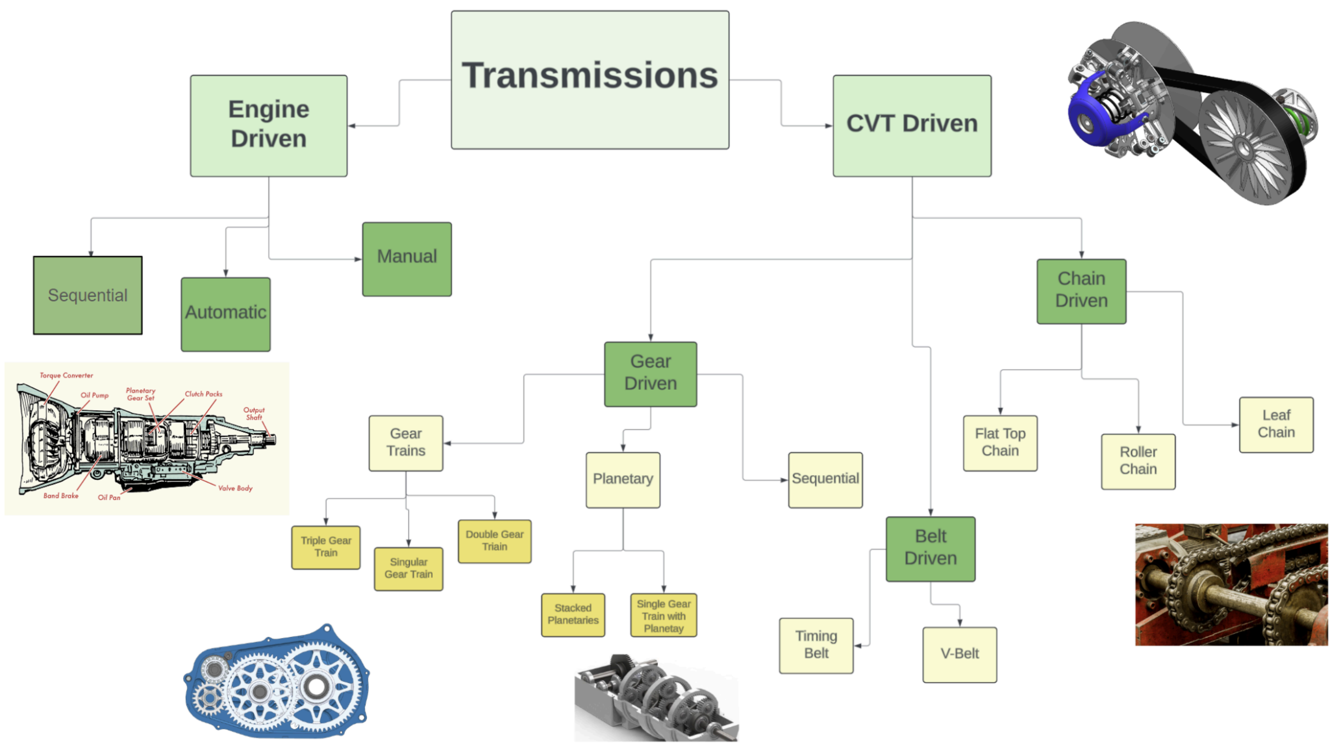
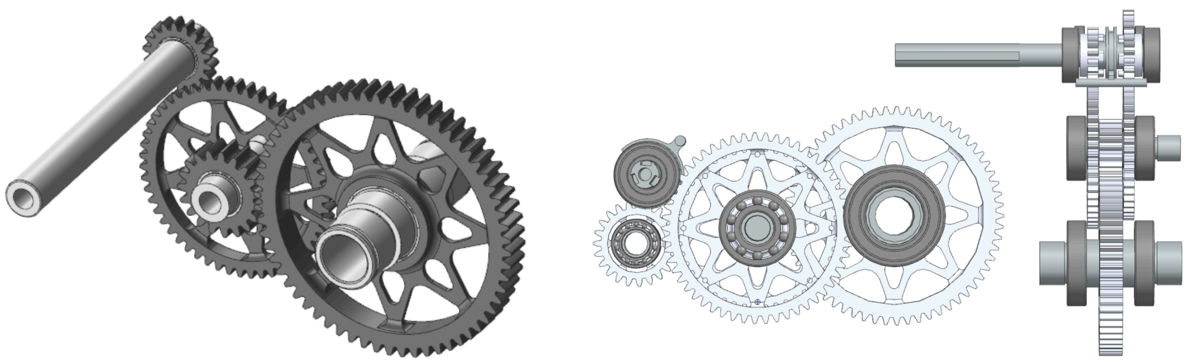
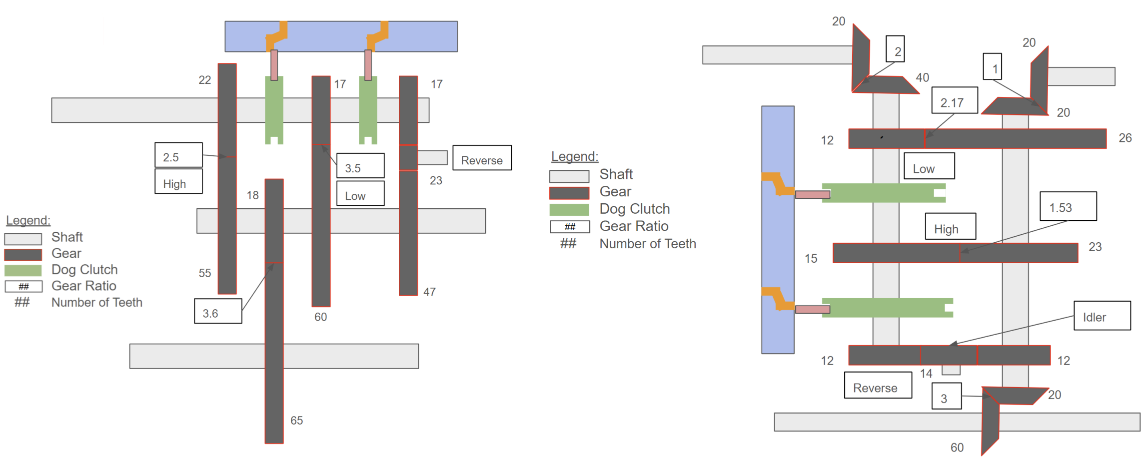



**Figure 6.1.1:** The concept tree shows all the high level branching that each solution fits into while most of the specific solutions and lower branches, such as on the engine driven side of the tree, have been concealed for readability.

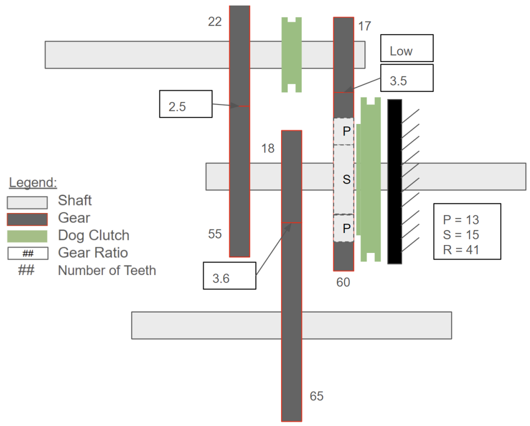
In the concept tree, the branches visibly lay out all the different paths the design could take. Branches can then be eliminated by doing some rough calculations and preliminary analysis to see if they clearly would not meet our requirements and specifications. For starters, anything engine driven was eliminated almost immediately due to the high weight increase along with packaging and manufacturing concerns. Engine driven solutions would require significantly bulkier and heavier transmissions in order for us to achieve our specific reductions than a CVT would be. This would make both packaging within our already limited space a challenge as well as manufacturing since the components would be more complex and have a higher part count. An engine driven solution would require a torque coupling device from the engine such as a fluid torque converter or a friction clutch in order to mitigate engine stalling, increasing mass. Such devices also have their own manufacturing concerns such as impeller, stator, and turbine manufacturing, and clutch friction material sourcing and manufacturing. From here, we were able to focus solely on the CVT driven branch of the tree. The chain driven and timing belt transmissions were eliminated due to how they handle shock loading causing durability issues based on the MBR's previous experience with them (the 2020-2021 vehicle implemented a chain drive system which consistently sheared its sprocket teeth in max loading conditions). V-belt driven transmission was also dismissed due to significant efficiency losses, especially from belt slip in the acceleration event. Within the gear driven branch, the stacked planetary transmission was eliminated due to the highly complicated shifting mechanisms that would be needed along with packaging and weight concerns. From there, five final design concepts were left, shown in Figures **6.1.2 a-e.**.



**Figures 6.1.2 a-b:** (a) 2023 Single Reduction Gearbox, (b) 2024 Double Reduction Gearbox with reverse capabilities



**Figures 6.1.2 c-d:** (c) 2024 Gearbox with Additional Gear Train Model (d) Sequential Model, both with 3 reductions: low, high, and reverse




**Figures 6.1.2e:** Planetary Model with 3 reductions: low, high, and reverse

## 6.1 System Analysis

In order to quantitatively compare each potential concept, five comparison metrics were determined. Concepts were evaluated on (1) the number of gear ratios, (2) complexity as measured by the number of parts, (3) manufacturability as measured by the required number of machining setups, (4) mass in lbs, and (5) efficiency loss as a percentage. In order to assign values to each of these metrics, detailed designs were required and preliminary calculations were performed in order to determine the necessary gear sizing. Each design was sketched out to ensure requirements and specifications would be met, and that each design could be manufactured and assembled. In sketching each design, the number of gear teeth to accomplish each required gear ratio was determined while maintaining equidistant center distances on gears that shared shafts. To fulfill this requirement, the sum of the teeth of each meshing set sharing shafts needed to be equal. Each sketch showed the number of gear ratios and the number of parts required. The sketches allowed machinists on the team to determine the number of manufacturing setups required to produce each design. Additionally, preliminary gear sizing (face width and pitch) was determined after conducting Lewis bending analysis and further analysis according to AGMA 20010-D04 to analyze contact stress and wear/fatigue (see appendix C) [21]. Gear sizing calculations were used to determine the material required to achieve the specified 1.2 safety factor over the expected loads, and to then estimate the mass of each gear train. The preliminary gear analysis determined 9310 steel hardened to 36 HRC to be a candidate material for which mass analysis was conducted [22]. Sketching each design also determined the number of bearings and gear meshes for each design. Due to the fact that MBR does not have a benchtop dynamometer to test the efficiency of drivetrain components, an estimation based on assumed 99% efficient bearings and 99% efficient gear ratios was used to determine the overall efficiency losses of each gearbox design [21]. An example of this system analysis for the three speed planetary gearbox is below, assuming a diametral pitch of 12 teeth/inch, a density of 9310 steel of 0.28 lb/in$^3$, and 3 planets.

| Parts list | Part Count | Number of setups |
|---|---|---|
| High Input | 1 | 2 |
| Low/Rev Input | 1 | 2 |
| High Intermediate | 1 | 2 |
| Small Intermediate | 1 | 2 |
| Output | 1 | 2 |
| Planetary Input | 1 | 2 |
| Sun | 1 | 2 |
| 3 Planets | 3 | 6 |
| Ring | 1 | 2 |
| Carrier | 1 | 2 |
| Input shaft | 1 | 2 |
| Intermediate shaft | 1 | 2 |
| Output shaft | 1 | 2 |
| Locker | 1 | 3 |
| Walking dog | 1 | 3 |
| Fork | 2 | 2 |
| Case | 2 | 4 |
| Bearings | 8 | 0 |
| **Total** | **29** | **42** |

| Forward Efficiency | |
|---|---|
| Number of Gearmeshes | 2 |
| Number of Bearings | 7 |
| **Efficiency Loss [%]** | **8.65** |

**Figure 6.1.1** Determining Part Count, Number of Manufacturing Steps, and Percent Efficiency Lost of the Three Speed Planetary Gearbox Design

| Gear | Gear teeth | Radius [in] | Facewidth [in] | Volume [in^3] | Mass [lbs] |
|---|---|---|---|---|---|
| High Input | 22 | 0.92 | 0.30 | 0.79 | 0.22 |
| Low/Rev Input | 17 | 0.71 | 0.30 | 0.47 | 0.13 |
| High Intermediate | 55 | 2.29 | 0.30 | 4.95 | 1.41 |
| Small Intermediate | 18 | 0.75 | 0.30 | 0.53 | 0.15 |
| Output | 65 | 2.71 | 0.30 | 6.91 | 1.96 |
| Planetary Input | 60 | 2.50 | 0.30 | 5.89 | 1.67 |
| Sun teeth | 15 | 0.63 | 0.30 | 0.37 | 0.10 |
| Planet teeth | 13 | 0.54 | 0.30 | 0.83 | 0.24 |
| Ring teeth | 41 | 1.71 | 0.30 | 2.75 | 0.78 |
| | | | | **Total Gear Mass:** | **6.67** |

| | | |
|---|---|---|
| **Shafts estimate** | | |
| MBR34 | 1.31 | lbs |
| Add 20% | 1.57 | lbs |
| **Bearings estimates** | | |
| MBR34 | 1.53 | lbs |
| Add 20% | 1.84 | lbs |
| **Other Structures** | | |
| Fork and Locker | 0.3 | lbs |
| Case/Carrier | 3 | lbs |
| **Total estimated Mass:** | **13.08** | **Lbs** |

**Figure 6.1.2** Estimating Mass of the Three Speed Planetary Gearbox Design. This mass analysis assumes a diametral pitch of 12 teeth/inch, a density of 9310 steel of 0.28 lb/in$^3$, and 3 planets.

This process was repeated for each of the other potential designs. The results of the system analysis for each system is presented in the table below.

**Table 6.1.1.** System Properties

| *Criteria* | *2023 Gearbox* | *2024 Gearbox* | *2024 Gearbox With* | *Planetary Model* | *Sequential Model* |
|---|---|---|---|---|---|

| | | | *Additional Gear Train* | | |
|---|---|---|---|---|---|
| # Gear Ratios | 1 | 2 | 3 | 3 | 3 |
| # Parts | 15 | 25 | 30 | 29 | 36 |
| # Manufacturing Setups | 18 | 29 | 38 | 42 | 48 |
| Mass (lbs) | 7.5 | 9 | 9.6 | 13 | 13 |
| Efficiency Loss (%) | 7.7 | 7.7 | 9.6 | 8.6 | 10 |

## 6.2 Analytical System Selection

To systematically determine which of the 5 concepts the team would continue with, we used a combination of an analytical hierarchy process (AHP) in Figure 6.2.1 and Kepner-Tregoe decision matrix in Figure 6.2.2. This method of down-selection provides a more objective and quantitative approach to choosing a solution by calculating scores solely from the numerical information provided, keeping it isolated from personal bias towards a specific solution.

To begin, all five categories that these final concepts were evaluated on were entered into the AHP (top chart), which uses pairwise comparisons of how important one category is relative to another to determine the weights of each. The AHP allows concrete pairwise comparisons, as opposed to arbitrary "vibes" based weight assignment in a classical pugh chart. We decided that the number of gear ratios a solution can provide was 9 times as important as efficiency loss because the project goal was to introduce another gear ratio for higher torque, which we knew would increase efficiency loss due to the increased number of rotating and meshing parts. Complexity and manufacturability were determined to be equally important, or have a 1:1 ratio of importance relative to each other – because fewer parts and fewer machining setups both simply save time for our members who have to conduct the operations – but 3 times as important as mass. Mass affects the overall speed of the vehicle, however, a couple of additional pounds in the gearbox, which is already allowed for by our specifications, will not change this drastically which is why mass was ranked to be less important than number of gear ratios, complexity, and manufacturability. Mass and efficiency both affect vehicle speed, but mass was determined to be 3 times more important than efficiency because we expect to see efficiency loss from the new designs and are thus okay with some efficiency loss as long as it is less than 15% as specified by our specifications beyond which we would not even consider the solution in the decision matrix. For similar reasons, complexity and manufacturability were determined to be 5 times as important as mass because we valued being able to not add too much extra work onto our members' and sponsor's already heavy workloads over marginal savings in weight. The rest of the rankings for number of gear ratios were calculated according to the relative weights of complexity, manufacturability, and mass against efficiency loss. Based on this input, the AHP totals each column's relative weights to get the sums in the bottom row and sums the quotients of the row of the category being looked at by the row of sums to get the criteria's absolute weight.

| Criteria | # Gear Ratios | Complexity | Manufacturability | Mass (lbs) | Efficiency Loss (%) | Criteria Weights |
|---|---|---|---|---|---|---|
| # Gear Ratios | 1 | 5 | 5 | 7 | 9 | 2.81 |
| Complexity (# parts) | 0.20 | 1 | 1 | 3 | 5 | 0.81 |
| Manufacturability (# setups) | 0.20 | 1 | 1 | 3 | 5 | 0.81 |
| Mass (lbs) | 0.14 | 0.33 | 0.33 | 1 | 3 | 0.38 |
| Efficiency Loss (%) | 0.11 | 0.20 | 0.20 | 0.33 | 1 | 0.19 |
| Sum | 1.65 | 7.53 | 7.53 | 14.33 | 23 | |

| | | | |
|---|---|---|---|
| 1 | Equal importance | 1 | Equal importance |
| 3 | Moderate importance | 1/3 | Moderate unimportance |
| 5 | Strong importance | 1/5 | Strong unimportance |
| 7 | Very strong importance | 1/7 | Very strong unimportance |
| 9 | Extreme importance | 1/9 | Extreme unimportance |
| | ONLY FILL IN ORANGE BOXES | | |

**Figure 6.2.1** Analytical Hierarchy Process

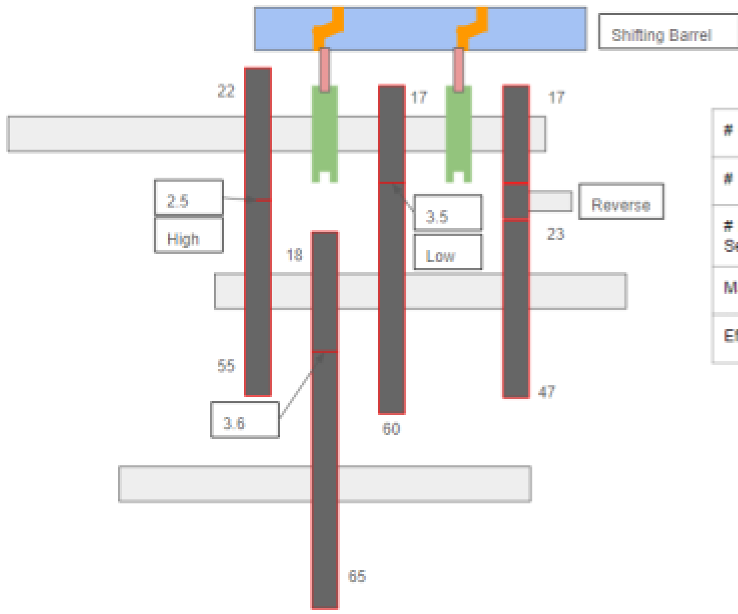
The criteria weights determined from the AHP were fed into the Kepner-Tregoe decision matrix, along with an information number for each category and solution. The information was the results of the engineering analysis conducted in System Analysis where we determined exactly how many gear ratios, how many parts, how many manufacturing setups, how much mass, and how much efficiency loss each concept would have. The matrix then compares how well or how much one solution meets the criteria relative to how much the others do to determine its value. Finally, the values are all multiplied by criteria weight to get scores which are summed up to get the total score of each solution.

| | | Gearbox Type | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 2023 Gearbox | | | 2024 Gearbox | | | 2024 Gearbox with added Gear Train | | | Planetary Gearbox | | | Sequential Gearbox | | |
| Decision Criteria | Weights | Info | Value | Score | Info | Value | Score | Info | Value | Score | Info | Value | Score | Info | Value | Score |
| # Gear Ratios | 2.81 | 1 | 0.18 | 0.50 | 2 | 0.35 | 0.99 | 3 | 0.53 | 1.49 | 3 | 0.53 | 1.49 | 3 | 0.53 | 1.49 |
| Complexity (# parts) | 0.81 | 15 | 0.64 | 0.52 | 25 | 0.41 | 0.33 | 30 | 0.29 | 0.23 | 29 | 0.31 | 0.25 | 36 | 0.14 | 0.12 |
| Manufacturability (# setups) | 0.81 | 18 | 0.74 | 0.60 | 29 | 0.58 | 0.47 | 38 | 0.45 | 0.36 | 42 | 0.39 | 0.32 | 48 | 0.30 | 0.25 |
| Mass (lbs) | 0.38 | 7.5 | 0.89 | 0.33 | 9 | 0.86 | 0.32 | 9.6 | 0.85 | 0.32 | 13 | 0.80 | 0.30 | 13 | 0.80 | 0.30 |
| Efficiency Loss (%) | 0.19 | 7.7 | 0.67 | 0.13 | 7.7 | 0.67 | 0.13 | 9.6 | 0.59 | 0.11 | 8.6 | 0.63 | 0.12 | 10 | 0.57 | 0.11 |
| **Final Score** | | 2.08 | | | 2.24 | | | 2.52 | | | 2.48 | | | 2.26 | | |

**Figure 6.2.2** Kepner-Tregoe Decision Matrix

According to the AHP and Kepner-Tregoe matrix, the winning solution was the 2024 gearbox with an added gear train for the high torque gear. Though its score was close to that of the planetary gearbox, we are confident the analysis run on each was accurate enough to validate the difference between the two.

# 7 Selected Concept




| | |
|---|---|
| # Gear Ratios | 3 |
| # Parts | 30 |
| # Manufacturing Setups | 38 |
| Mass (lb) | 9.6 |
| Efficiency Loss (%) | 9.6 |

**Figure 7.1.1** Selected Concept: 2024 Gearbox with Added Geartrain. Torque is input through the top shaft and output through the bottom-most shaft. Dog clutches select between high, low, and reverse gears through an actuated shifting barrel.

The selected concept was determined to be the 2024 Gearbox with an added gear train (Figure 7.1.1). It has several main advantages, as it achieves all three gear ratios that were specified, and compared to the other 3-speed concepts, it has a lower part count, it has less manufacturing setups, it is lighter, and it is more efficient.

The concept is an extension of the 2024 Gearbox, utilizing free-spinning gears on the input shaft that are selected one at a time via a shifting barrel controlled by the driver. The driver pulls a lever in the cockpit, rotating the shifting barrel, which pushes a selector fork and selector dog gear against the dog gear teeth on the free-spinning input gear the driver wishes to select, transferring torque through it and thus through the rest of the gearbox to the output shaft. The three ratios include a high gear ratio of 9:1 for high top speed, a low gear ratio of 12.6:1 for higher torque, and a reverse ratio of -9.95:1. Four-wheel drive will be outputted to the driver right side of the car from the intermediate shaft. The gears and shafts will be enclosed in a 7075 Aluminum casing, machined on a 3-axis CNC mill, and submerged in gear oil for cooling and efficiency benefits.

Shafts will be designed by looking at gear contact forces and bearing supports modeled as simply supported beams. The overall OD and ID will be selected through geometrical constraints and maintaining a 1.2 Safety Factor in bending & torsion through the shaft. Shaft splines will be designed looking at shear stress at the root of the spline and bending stress on the tooth at maximum torque seen in an overload situation, and designed to have a 1.2 SF for both conditions. Gear tooth count, diametral pitch, pressure angle, and face widths have already been predetermined after looking into the packaging constraints of our project and capabilities of manufacturing sponsors, and have been selected based on the Lewis Bending equations and Hertzian contact stress on the teeth considering the same overload scenario as the shafts, with a 1.5 SF placed on both. Bending and pitting fatigue have also been considered, and have at least a 1.2 SF for all gears. Gears with large distances radially between the root of the gear teeth and root of their inner splines will be pocketed as to be lightweight, and a non-linear contact finite element analysis will be performed on the pocketing to verify it's structural integrity under the aforementioned overload scenario to a 1.5 SF. The gearbox casing will be designed to contain all bearings and gears with clearance for rotation and slight deformation. The case will be pocketed similar to the gears, and a non-linear contact finite element analysis will be performed on the maximum bearing reaction loads and the axial loading through the rear axle into the gearbox to a 1.3 SF. Bearings for shafts and free-spinning gears will be selected by looking at the C10 life of the bearings and the maximum revolutions of the component at its maximum dynamic load.

The shafts will be machined on a CNC Lathe in house, and made of either Grade 5 Titanium or 300M Steel, and the 300M shafts will be heat-treated to 36 HRC to increase their yield strength. They will then be sent to a sponsor to have spline profiles shaped onto them. Gear blanks will be manufactured on a 3-axis CNC mill in-house, and made out of 9310 Steel. The gear teeth will be sent to a sponsor for rough shaping or hobbing, heat treated and case carburized, and then finally ground to the specified involute profile with +/- 0.0003in of tolerance. The following parts will all be manufactured in house on a 4-axis CNC Mill: the shifter barrel out of 300M, the shifter forks out of 9310, and the dog gear selectors out of 9310.

# 8 Design Solution

## 8.1 Concept Overview

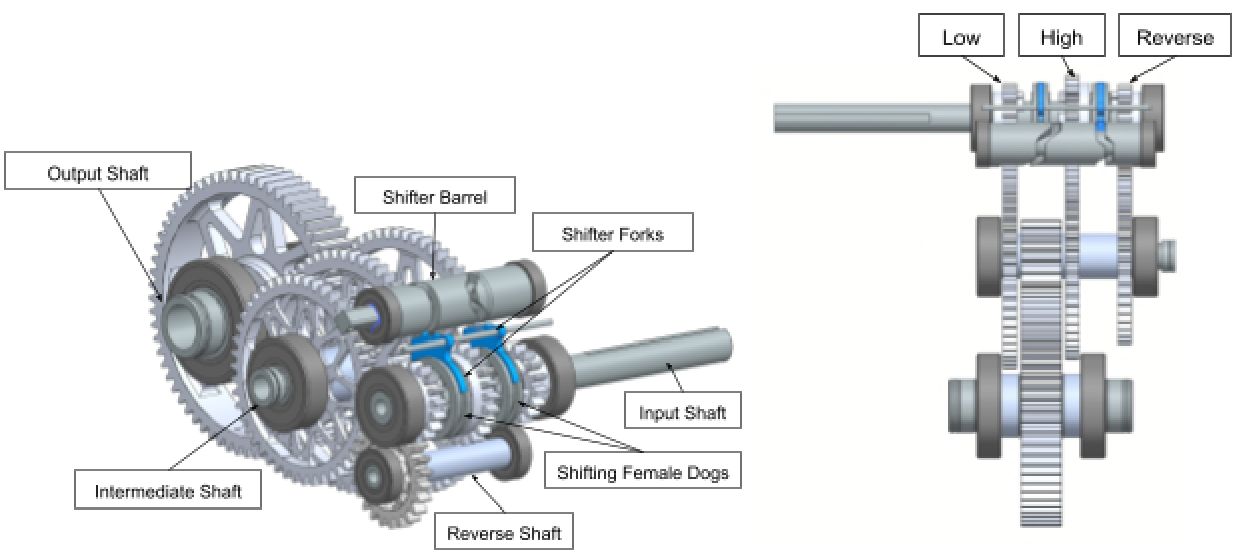



**Figure 8.1.1 a-b:** (a) Isometric view of Gearbox CAD Model (b) Top view of Gearbox CAD Model. Note the low gear and high gear positions have changed with respect to the sketched design for ease of assembly

From our selected design generated for inputting parameters into the Kepner-Tregoe matrix, we translated our sketch into a CAD model. The sketch dictated gear face width, gear tooth count, and overall gearbox architecture. While the overall design changed very little between the sketch and the CAD model, there was one minor change in gear train arrangement: the high gear and low gear positions were switched to accommodate the free wheeling action of the high input gear. In order for the input gears to freewheel on the input shaft, the input gears must ride on bearings on the input shaft. The input gears must freewheel on the input shaft in order to facilitate shifting. Shifting is achieved by moving the female shifting dogs onto the male dogs on the appropriate input gear, thus selecting high, low, or reverse gears. Once the male and female dogs are meshed, the selected input gear rotates with the same angular velocity as the input shaft, providing torque to the selected gear train. The female shifting dogs are splined to the input shaft. This creates the issue of installing an input gear over the shifting dog splines of the input shaft. Since the middle input gear must slip over the outer splines during installation, the diametrically largest input gear, the high gear input gear, was selected to fit in the middle of the input shaft. In order for the high input gear to freewheel on the input shaft, a "spline plug" was designed to slide over the splines, and to provide a bearing surface on which the high input gear bearing will interface. To achieve axial constraint of the high input gear, the spline plug is axially constrained to the input shaft via snap rings. The high gear input bearing abuts onto a lip on the spline plug on one end and a snap ring on the other. Finally, the high input gear is slipped over the entire assembly, with the inner face of the surface supporting the male dogs constraining it axially against the bearing. The high input gear is further constrained on the other side by a snap ring against the bearing. The low and reverse input gears are also mounted on bearings along the non-splined segments of the input shaft. The low and reverse inputs use the same mounting technique, and use the same tooth count, and physical part.

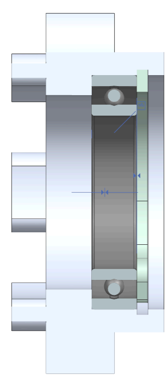


**Figure 8.1.2** Free-spinning input gear cross section highlighting the inner bearing it rides on.

The only difference is their orientation, with the low gear male dogs pointing to the right side of the car while the male dogs of the reverse input point towards the left side of the car such that the male dogs of the reverse input and low gear input face each other but are separated by the female shifting dog and fork assembly. The low/reverse bearing is constrained axially to the shaft via snap rings. The gear is constrained to the bearing with the inner face of the surface supporting the male dogs, constraining it axially against the bearing, and constrained on the other side by a snap ring against the bearing.

- Throughout the design process, the gearbox design was informed by our requirements. While producing the CAD, we confirmed that the design meets the requirement of 8.5 in center to center distance from crankshaft to input shaft, which was specified by MBR's CVT lead. The design also fulfills the requirement of a distance between 5.75in and 6.5 from output to input shaft, as specified by the gearbox and rear half shafts leads, with a distance of 6.5in from output to input shaft. Furthermore, the estimated weight of the design is about 10.9 lbs dry, meeting our requirement of a less than 11 lbs gearbox, a benchmark set by our sponsor. Each component is manufacturable with 4 axis machinery, as confirmed by our team's machinists, meeting our manufacturability requirement also specified by our sponsor. For ease of bearing selection, bearings were selected from the Michigan Baja Racing team's bearing sponsor's catalog, the NSK standard catalog. While designing the shifting barrel, a 45º shift angle was chosen to ensure that the X and Y force components are equal on the shifting fork shafts, allowing for a symmetric shifting fork shaft design. Manufacturing concerns were addressed in the design of the shifting forks and shifting barrel. In order for the shifting forks to be manufactured in house, a two piece design was selected. The rounded interface of the fork which is in contact with the female shifting dogs was designed to be laser-cut and subsequently welded to a machined shaft via an interfacing connection. The shifting barrel was designed to be machined on the fourth axis of a CNC mill. Additionally a shaft on which the shifting forks run on was added to ensure no misalignment and to constrain the shifter forks to an axis parallel to the input shaft. Each of the gears, low, high, and reverse, are separated by 120º on the shifting barrel, and the shifting barrel interfaces with the shifting lever, the design of which is outside the scope of this project, via a polygon (square) torque transfer inspired by square drive ratchets.

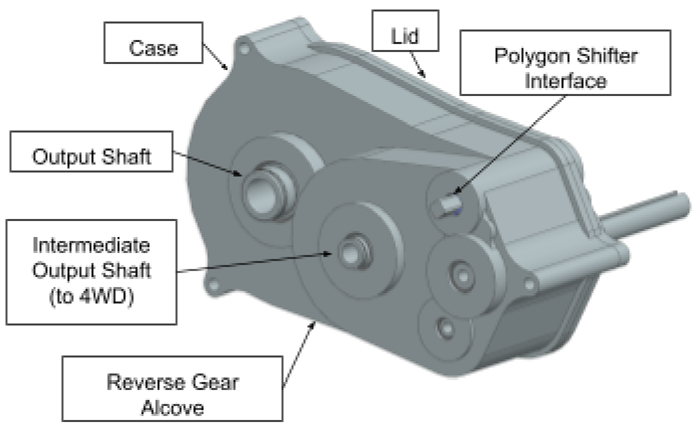



**Figure 8.1.3** Gearbox Design With Case. Gearbox design has necessary output, intermediate output, and input shafts to be back-compatible with previous vehicles, fulfilling our back compatibility requirement.

In order to meet our lightweight requirement of less than 11 lbs, gear design included gear pocketing to decrease the mass of the center of the gears. The shape of the gear pocketing was determined by running topology optimization on the gears, optimizing for the mass moment of inertia of the gears about their axes. Additionally, the intermediate output was designed to integrate with the 4WD system of MBR35, while the output shafts were designed to integrate with the legacy driveshafts common to previous Michigan Baja Racing vehicles. The center plane of the output gear is centered on the car centerline in order to accommodate drive shafts, and to ensure equal torque through each shaft in the limiting condition of both wheels acting as fixed supports (CVT belt slip limited load case). Centering the output gear on the centerline of the car results in necessitating an alcove housing for the reverse gear. In order to ensure proper lubrication of the reverse gear train and to mitigate oil distribution concerns, the gearbox requires enough oil to partially submerge the reverse idler gear. The alcove design took into account manufacturing concerns, and is manufacturable with a 4in ball-nose end-mill already in the possession of the Michigan Baja Racing Team, and therefore satisfies the requirement of being manufacturable in house.

## 8.2 Analysis

Loads were gathered for gearbox components using a combination of first principle analysis, free-body diagrams, hand calculations, and empirical data derived from annual testing of each iteration of MBR's vehicle. The maximum load cases are the torque of the CVT while it is locked in it's maximum torque output ratio, which also causes bending on the input shaft, the axial loading of 17kN from a worst-case scenario of landing on one wheel and the force being transmitted through the rear half-shaft into the gearbox, and all the associated internal shaft reaction forces on supporting bearings. This section details an example of each type of analysis, however thorough analysis and iterative design was completed on all components under their maximum loading and subsequent fatigue to ensure all components meet a standard 1.2 SF, with potential coherence modifiers depending on confidence in either loading parameters or model accuracy. Below is a flow chart describing the analysis process for individual repeated gearbox components.

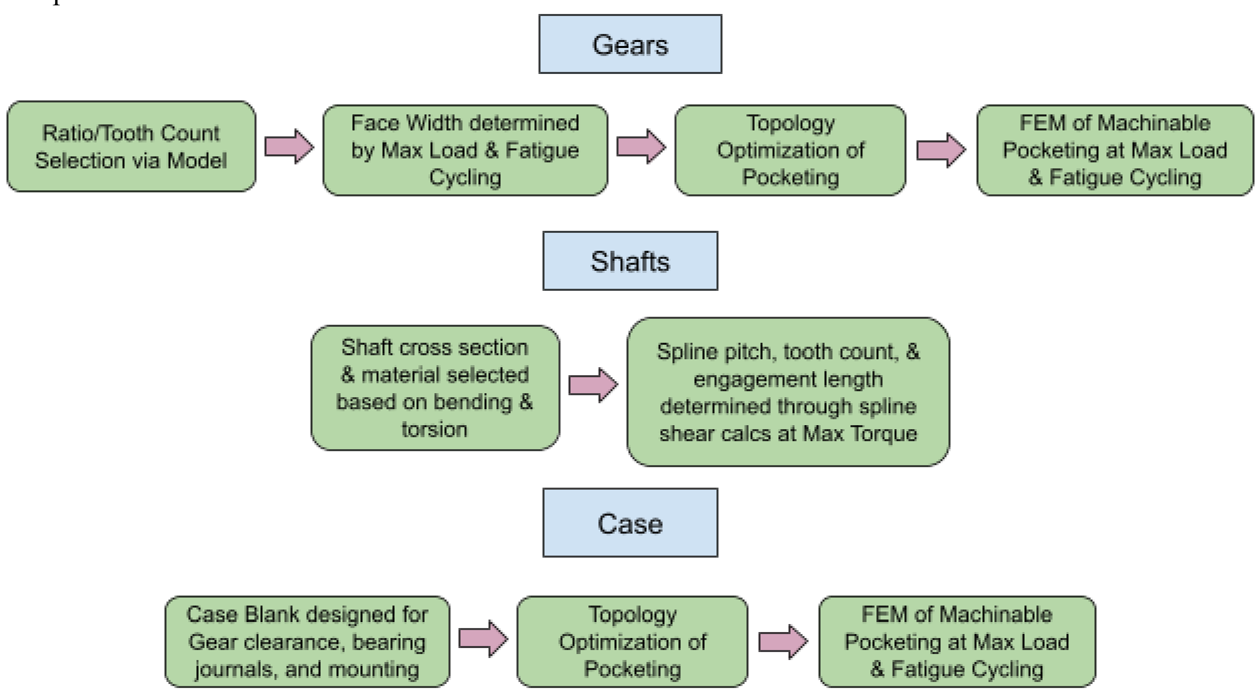



**Figure 8.2:** Analysis flow of gearbox components

### 8.2.1 Gears: Lewis Bending, Hertzian Contact, Fatigue

The American Gear Manufacturing Association (AGMA) sets standards for gear geometry and manufacturing in the United States. Before they defined equations for involute profile gear tooth strength, Wilfred Lewis completed empirical testing for different pitches and manufacturing methods and derived equations to calculate the strength of a gear tooth approximated as a beam in bending, while including factors derived from his empirical testing. This equation, along with the modified Hertzian Contact stress equation of two cylinders in contact, and AGMA equations for bending and pitting fatigue safety factors, was applied to all gear geometry to ensure gears have an acceptable safety factor for their maximum load case.

**Table 8.2.1.1:** Equations for Lewis Bending, Hertzian Contact Stress

$$\sigma_{Lewis\ Bending} = \frac{K_v W^t P}{FY}$$

$$\sigma_{Contact} = C_P \sqrt{W^t K_o K_v K_s \frac{K_m C_f}{d_p F I}}$$

$$\sigma_{Fatigue\ Bending} = \frac{S_t}{S_F} \frac{Y_N}{K_T K_R}$$

$$\sigma_{Fatigue\ Pitting} = \frac{S_c Z_N C_H}{S_H K_T K_R}$$

| Syb | Name | Syb | Name |
|---|---|---|---|
| $C_f$ | Surface condition factor | $K_s$ | Size factor |
| $C_H$ | Hardness-ratio factor | $K_T$ | Temp factor |
| $C_p$ | Elastic Coeff | $K_v$ | Dynamic factor |
| $d_p$ | Pinion PD | P | Diametral pitch |
| F | Face width | $S_c$ | AGMA surf strength |
| I | Geom factor | $S_t$ | AGMA bending strength |
| $K_m$ | Load dist factor | $S_F$ | SF bending |
| $K_o$ | Overload factor | $S_H$ | SF pitting |
| $K_R$ | Reliability Factor | $W^t$ | Load |
| $Y_N$ | Stress cycle factor bending | $Z_N$ | Stress cycle factor for pitting |

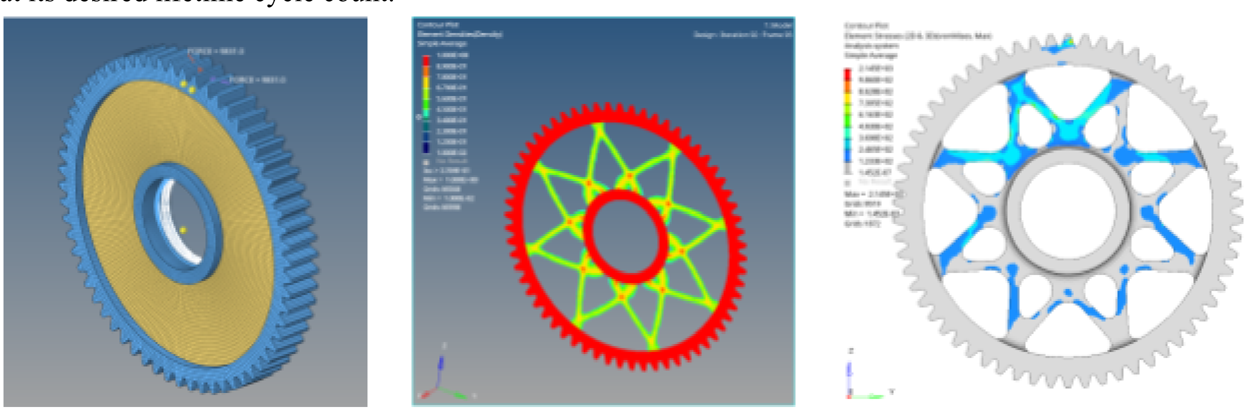
**Table 8.2.1.2:** Calculation of Lewis Bending Stress, Contact Stress, Bending and Pitting Fatigue using standard AGMA equations on the Low Input Gear & Low Intermediate Gear

| | | | | | | |
|---|---|---|---|---|---|---|
| Fatigue limit for bending | 535 | MPa | | Fatigue limit for bending | 535 | MPa |
| Fatigue limit for contact stress | 930.7924767 | MPa | | Fatigue limit for contact stress | 930.7924767 | MPa |
| Temperature Factor | 1 | | | Temperature Factor | 1 | |
| Reliability Factor | 1 | | | Reliability Factor | 1 | |
| Hardness Ratio Factor for Pitting Resistance | 1 | | | Hardness Ratio Factor for Pitting Resistance | 1 | |
| Bending Stress | 513.1576414 | MPa | | Bending Stress | 351.217316 | MPa |
| Contact Stress | 1207.000686 | MPa | | Contact Stress | 690.2852358 | MPa |
| Elastic Coefficient | 1 | | | Elastic Coefficient | 1 | |
| Transmitted Tangential Load | 3754.464 | N | | Transmitted Tangential Load | 4960.368 | N |
| Overload Factor | 1 | | | Overload Factor | 1 | |
| Dynamic Factor | 1 | | | Dynamic Factor | 1 | |
| Load Distribution Factor | 1 | | | Load Distribution Factor | 1 | |
| Size Factor | 1 | | | Size Factor | 1 | |
| Surface Condition factor | 1 | | | Surface Condition factor | 1 | |
| Rim thickness factor | 1 | | | Rim thickness factor | 1 | |
| Face width | 7.62 | mm | | Face width | 0 | mm |
| Pitch Diameter | 35.98333333 | mm | | Pitch Diameter | 99.48333333 | mm |
| Geometry Factor for pitting | 1 | | | Geometry Factor for pitting | 1 | |
| Geometry factor for bending | 1 | | | Geometry factor for bending | 1 | |
| | | | | | | |
| Stress cycle factor for bending (Zn) | 1.1 | Can also get from graphs below | | Stress cycle factor for bending (Zn) | 1.08 | Can also g |
| Stress cycle factor for pitting resistance (Yn) | 1.25 | Can also get from graphs below | | Stress cycle factor for pitting resistance (Yn) | 1.19 | Can also g |
| | | | | | | |
| Permissible Bending Stress (sigmaF) | 588.5 | | | Permissible Bending Stress (sigmaF) | 577.8 | |
| Permissible Contact Stress (sigmaH) | 1163.490596 | | | Permissible Contact Stress (sigmaH) | 1107.643047 | |
| | | | | | | |
| **Safety Factor on bending** | 1.146821079 | 2.282575932 | | **Safety Factor on bending** | 1.645135287 | -7.77981 |
| **Safety Factor on pitting** | 1.263951893 | -28.25405424 | | **Safety Factor on pitting** | 1.604616454 | 0.328001 |

| Lewis Bending Stress Calculations (Full Underdrive, Max Torque), Power Based Equation | | | | | | |
|---|---|---|---|---|---|---|
| Max Tooth Load @ Pitch | 845.6 | lbs | | Max Tooth Load @ Pitch | 1117.2 | lbs |
| Tooth Lewis Form Factor | 0.302 | | | Tooth Lewis Form Factor | 0.399 | |
| Pitch Line Velocity | 304.8349036 | ft/min | | Pitch Line Velocity | 337.1115405 | ft/min |
| Max. Torque (in-lbs) | 598.9666667 | | | Max. Torque (in-lbs) | 2187.85 | |
| Max Torque (ft. lbs) | 49.91388889 | | | Max Torque (ft. lbs) | 182.3208333 | |
| Max. Horsepower | 7.811163469 | | | Max. Horsepower | 11.41275797 | |
| Max Torque w/contact ratio (ft-lbs) | 81.32342597 | | | Max Torque w/contact ratio (ft-lbs) | 297.0506831 | |
| Face Width to Circular Pitch | 1.14591559 | | | Face Width to Circular Pitch | 1.14591559 | |
| | | | | | | |
| Power | 8.457142857 | hp | | Power | 8.457142857 | hp |
| Tangential Force on Teeth | 561.9253905 | lbs | | Tangential Force on Teeth | 508.1240234 | lbs |
| Torque | 37.07146674 | ft-lb | | Torque | 86.45165675 | ft-lb |
| Max Bending Stress | 74427.20404 | psi | | Max Bending Stress | 50939.75172 | psi |
| **Safety Factor from Safe Stress** | 1.504826111 | 2.335894498 | | **Safety Factor from Safe Stress** | 2.198675813 | -7.21825 |
| | 1.539977261 | | | | 2.039969878 | |
| **Contact Stress Analysis (Hertz Equations)** | | | | | | |
| Poisson's Ratio (nu) | 0.3 | | | Poisson's Ratio (nu) | 0.3 | |
| Young's Modulus (E) | 29000000 | psi | | Young's Modulus (E) | 29000000 | psi |
| Elastic Coefficient | 2252.10362 | | | Elastic Coefficient | 2252.10362 | |
| Form Factor | 0.2188213139 | | | Form Factor | 0.2188213139 | |
| Contact Stress | 175.0606033 | ksi | | Contact Stress | 100.1173829 | ksi |
| Allowable Contact Stress (Table 9-3) | 225 | ksi | | Allowable Contact Stress (Table 9-3) | 225 | ksi |
| **Safety Factor** | 1.285269191 | | | **Safety Factor** | 2.24736198 | |

### 8.2.3 Gear Pocketing

MBR's vehicle sees a much shorter lifetime than that of a typical vehicle, meaning the cycles that each component sees are much lower, and fatigue is not the main cause of failure in components. Therefore, in pursuit of maximum performance, many of the larger, heavier gears are pocketed to decrease rotational inertia and overall mass of the system. Topology optimization is performed on the center section of the gear, allowing for acceptable gear & spline tooth root thickness at the maximum torque load case, with inertial optimization parameters. From the output of this analysis, 3-axis machinable pocketing is designed to mimic the resultant geometry, and a finite element analysis is performed at the maximum and routine loads to determine the safety factors of the pocketing in both maximum load cases and fatigue. An S-N curve for 9310 steel is used to validate that the routine loading does not cause the component to fail at its desired lifetime cycle count.

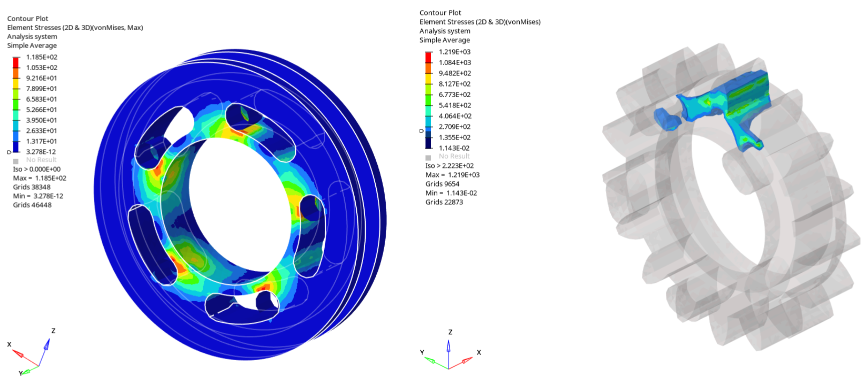


(a) (b) (c)

**Figure 8.2.3(a-c):** Gear Pocketing design example. Here, the output gear has the geometry to be optimized selected in yellow and loading is applied to the tooth while the splined center is constrained (**a**), then the output of this topology optimization (**b**) is used to design machinable pocketing that is then reanalyzed with the same forces and constraints (**c**).

### 8.2.4 HLNR Dog Teeth

One of the critical pieces of each gear train is the dog teeth transferring torque from the input shaft to the input gears of each gear train. If these teeth were to fail, the gearbox would not be able to function at all, and would cause highly detrimental points losses during a competition. These teeth are first analyzed using shear calculations, looking at the torque through the input shaft to derive the force at the centerline of the teeth, how many teeth are used to distribute forces, and the cross sectional area of each tooth to determine shear stress. Because the teeth are wider than they are long, a beam-in-bending approximation does not hold like it does for the shafts. Therefore, a nonlinear finite element analysis is performed with contact surfaces to mimic how the teeth will put stress on the pocketing in the female splined selectors. The maximum input torque derived from drop-testing is used to validate the teeth's performance, as shown below, and geometry is iterated until a desirable safety factor is achieved.




**(a)** **(b)**

**Figure 8.2.4(a-c):** Input female (**a**) and male (**b**) dog teeth are analyzed using finite element models to determine safety factors at the maximum load case.

### 8.2.5 Shaft Bending, Torsion, and Splines

To verify our gearbox shafts can survive the loads they see from the gears, bearings, and other inputs, we model them as beams in bending. We calculated how much force they see from the gears radial loading and used the sum of forces and sum of moments to estimate how much force the bearings apply. We were then able to create shear force diagrams to determine the maximum bending moments seen by the shafts and thus, calculate their bending stresses using the left equation below.

Bending Stress: $\sigma_{bending} = \frac{M\,y}{I} = \sigma_{zz}$ Shear Stress: $\boldsymbol{\tau} = \frac{T\,r}{J}$

The torsional loads experienced by the shafts come from the torques they are transferring that are determined based on the gear ratios before each shaft and the input torque from the CVT. This got us the shear stress in the shafts from the right, above equation which in combination with the axial stress and principal stress equations, gave us the total estimated stress the shaft sees under von Mises criterion. The principal stress and von Mises equations are described in Figure 8.2.5.1.

$$\sigma_{1,2} = \left(\frac{\sigma_x + \sigma_y}{2}\right) \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \qquad \sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2)}{2}}$$

**Figure 8.2.5.2:** Principal Stress Equations (Left) and von Mises Stress Equation (Right).

By comparing the von Mises stress of the shafts to the yield stress of the materials we considered making the shafts out of, we calculated the safety factor for each design. An example of this is shown for the input shaft in Figure 8.2.5.2.

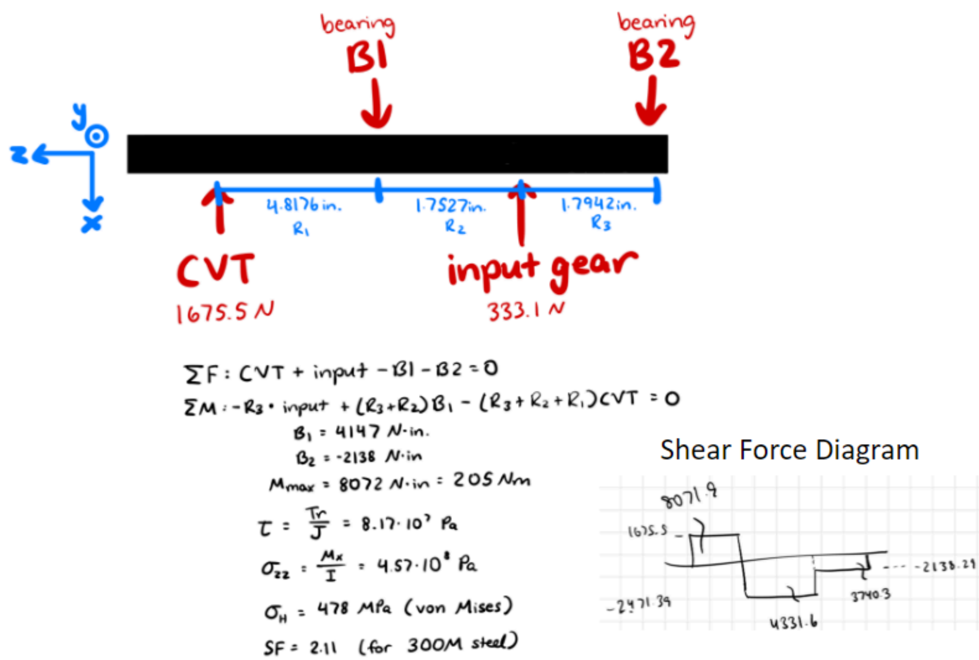



**Figure 8.2.5.2:** Input Shaft Bending Calculations. The input shaft sees loading from the CVT, input gears, and bearings. We chose to use the low gear input gear for this calculation because the gearbox sees the most torque in this setting so therefore, the input shaft would see the most load from this input gear compared to the rest. A 300M steel input shaft would have a safety factor of 2.11, while a titanium input shaft's SF would be 1.7 which is well above our goal of a 1.2 SF, so we chose to make the input shaft out of titanium.

We additionally ran calculations of the shafts in torsion to ensure the spline teeth do not fail under the torques they see. Specifically, we looked at shear stress under the root and at the pitch of the splines using the equations in table 8.2.5.1 (24).

**Table 8.2.5.1:** Equations for Spline Torsion calculations looking at shear stress under the root and at the pitch.

$$\text{Shear Stress Under the Root: } \boldsymbol{\tau} = \frac{16\,T\,D_{re}\,K_a}{\pi(D_{re}^{\ 4} - ID^4)\,K_f}$$

$$\text{Shear Stress at Pitch Diameter: } \boldsymbol{\tau} = \frac{4\,T\,K_m\,K_a}{D\,N\,L_e\,t\,K_f}$$

| Syb | Name |
|---|---|
| T | Torque |
| $D_{re}$ | Minor Diameter |
| ID | Interior Diameter |
| D | Pitch Diameter |
| N | Number of Spline Teeth |

| | |
|---|---|
| $L_e$ | Length of Engagement |
| t | Maximum Effective Tooth Thickness |
| $K_a$ | Application Factor |
| $K_f$ | Fatigue Life Factor |
| $K_m$ | Load Distribution Factor |

An example of the spline torsion calculations for the input shaft is shown in Figure 8.2.5.3.

| | | | Stress on the Teeth | |
|---|---|---|---|---|
| | | | Notes: | |
| Interior Diameter of Shaft | 0.4134 | in | | |
| Outer Diameter of the Shaft | 0.6692 | in | | |
| Wall thickness | 0.1279070866 | in | | |
| Hardness | 36 | HRC | Material is Grade 5 Ti | |
| Max allowable shear stress | 79800 | psi | matweb | |
| Max allowable compressive stress | | psi | | |
| application factor (Ka) | 2 | | p 2172 table 7* | |
| Load Distribution Factor (Km) | 1 | | p 2172 table 8* | |
| Fatigue Life Factor (Kf) | 1 | | p 2173 table 9*, ~10,000 cycles | |
| Max Torque | 648.24 | lb-in | | |
| Length of Engagement (Le) | 0.3 | | | |
| Pressure Angle (phiD) | 30 | degrees | | |
| Number of Teeth (N) | 15 | | | |
| Pitch (P) | 20 | | | |
| Pitch Diameter (D) | 0.75 | in | eq. on p 2162* | |
| Major Diameter (Do) | 0.8 | in | eq. on p 2162* | |
| Minor Diameter (Dre) | 0.6825 | in | eq. on p 2162* | |
| Circular Pitch (p) | 0.1570796327 | in | see p 2169 table 6* | |
| Maximum Effective Tooth | 0.0785398163 | in | see p 2169 table 6* | |
| Maximum Tolerance for Tooth | 0.00125 | in | p 2164 table 4* | |
| Wall Thickness at root of | 0.1345570866 | in | | |
| Tooth Depth | 0.1175 | | | |
| Backup Ratio | 1.145166695 | | | Safety Factors |
| Shear Stress Under Roots of | 23999.70405 | psi | | 3.325 |
| Shear Stress at the pitch | 19564.20273 | psi | | 4.079 |
| Compressive Stresses on | 17072.98765 | psi | p 2174 eq 4* | |

**Figure 8.2.5.3:** Input Shaft Spline Torsion Calculations. These are conducted to ensure the spline teeth do not shear. According to the calculations, the input shaft's spline teeth have a large safety factor on them, well over MBR's standard of 1.2, when compared to the yield strength of titanium which the shaft will be made out of. This higher factor of safety is due to the spline minor diameter needing to be larger than the outer diameter of the shaft.

### 8.2.6 Bearing L10 Life

All bearings are ordered from the manufacturer NSK, who is a sponsor of the team. These bearings are chosen based on the Dynamic Equivalent Load applied by either the shaft or the gear they are placed on at maximum torque out of the engine. Any axial load is derived from hand calculations or from empirical data from external driving forces. Bearing choices are narrowed down based on size constraints of the shaft and journal they ride on. Then, the standard $L_{10}$ life formula is used to calculate safety factors for bearing options and the lightest weight passing bearing is chosen.

**Dynamic Equivalent Load**

$P = XF_r + YF_a$

| $\frac{f_0 F_a}{C_{0r}}$ | $e$ | $\frac{F_a}{F_r} \le e$ X | $\frac{F_a}{F_r} \le e$ Y | $\frac{F_a}{F_r} > e$ X | $\frac{F_a}{F_r} > e$ Y |
|---|---|---|---|---|---|
| 0.172 | 0.19 | 1 | 0 | 0.56 | 2.30 |
| 0.345 | 0.22 | 1 | 0 | 0.56 | 1.99 |
| 0.689 | 0.26 | 1 | 0 | 0.56 | 1.71 |
| 1.03 | 0.28 | 1 | 0 | 0.56 | 1.55 |
| 1.38 | 0.30 | 1 | 0 | 0.56 | 1.45 |
| 2.07 | 0.34 | 1 | 0 | 0.56 | 1.31 |
| 3.45 | 0.38 | 1 | 0 | 0.56 | 1.15 |
| 5.17 | 0.42 | 1 | 0 | 0.56 | 1.04 |
| 6.89 | 0.44 | 1 | 0 | 0.56 | 1.00 |

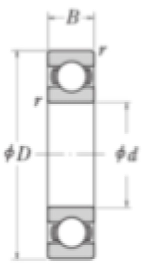
**Static Equivalent Load**

$\frac{F_a}{F_r} > 0.8,\ P_0 = 0.6F_r + 0.5F_a$

$\frac{F_a}{F_r} \le 0.8,\ P_0 = F_r$

NSK Single-Row Deep-Groove Ball Bearing Catalog

| Boundary Dimensions (mm) d | D | B | r min. | Basic Load Ratings (N) $C_r$ | $C_{0r}$ | Factor $f_0$ | Limiting Speeds (min⁻¹) Grease Open Z · ZZ V · VV | Grease DU DDU | Oil Open Z |
|---|---|---|---|---|---|---|---|---|---|
| 17 | 26 | 5 | 0.3 | 2 630 | 1 570 | 15.7 | 26 000 | 15 000 | 30 000 |




$$L_{10} = \left(\frac{C}{F}\right)^P \left(\frac{10^6}{60 \times N}\right)$$

where:

C = dynamic load rating (numbered of rolling elements, roller length/diameter and contact angle

F = applied load

P = 3 (ball bearings) or 10/3 (roller bearings)

N = RPM

| | Input Gear |
|---|---|
| Revolutions in Life | 8527720.722 |
| Load (N) | 343.9333828 |
| Dynamic Load Rating (N) | 1570 |
| L10 Life | 95120902.78 |
| Safety Factor | 11.1543173 |

**Figure 8.2.6:** Example calculations for the Input Gear bearing that sees the highest cycle count of any drivetrain component.

### 8.2.7 Case Analysis

The housing for the 3 gear trains is a 3-axis machined Aluminum 7075 casing with bearing journals for shafts. The input, intermediate, and output shafts extrude out of the case to mate with their respective parts. The case is first extruded based on the outer profile of the gears, allowing 0.050in of radial clearance to the teeth, and 0.035in of lateral clearance between the case and any rotating component. Then, pocketing or ribs are added to support the output bearing journal to the mounting bolt holes, where the stress will be distributed. All features must be manufacturable by in house tooling, meaning all edge blends must be greater than 0.125in in diameter and less than 3 inches deep into the case (for tool length). The final case is analyzed using a non-linear finite element model that can replicate the purely compressive contact stresses of bearings on the housing.

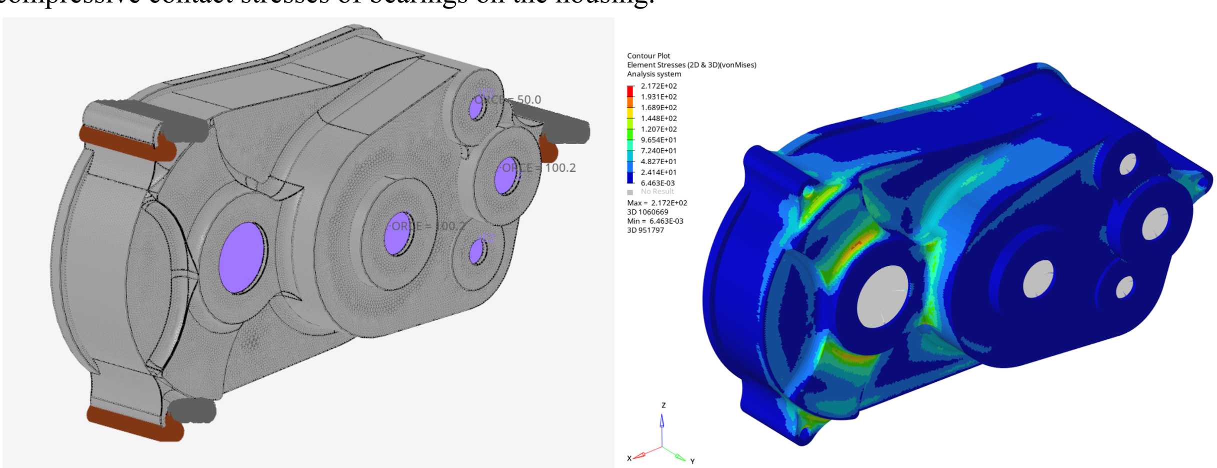



**Figure 8.2.7:** Driver Right Gearbox Case FEM and analysis results.

## 8.3 Bill of Materials

The team generated the bill of materials in Table 8.3.1. to outline and organize all the components needed for the gearbox.

**Table 8.3.1** Bill of Materials displaying all the components of the designed gearbox, their part numbers, suppliers, quantity, material, unit price, and total price.

| Part Name | Part Number | Supplier | Quantity | Material | Price per Unit | Total Price |
|---|---|---|---|---|---|---|
| High Input Gear | 51751 | TW Metals | 1 | 9310 Steel | $50/ft | $3 |
| Low/Rev Input Gear | 51750 | TW Metals | 2 | 9310 Steel | $50/ft | $6 |
| Output Gear | 51743 | TW Metals | 1 | 9310 Steel | $100/ft | $10 |
| Small Intermediate Gear | 51742 | TW Metals | 2 | 9310 Steel | $50/ft | $8.40 |
| Reverse Idler Gear | 51741 | TW Metals | 1 | 9310 Steel | $50/ft | $3 |
| Reverse Intermediate Gear | 51740 | TW Metals | 1 | 9310 Steel | $100/ft | $8.00 |
| Low Intermediate Gear | 51739 | TW Metals | 1 | 9310 Steel | $100/ft | $8.00 |
| High Intermediate Gear | 51738 | TW Metals | 1 | 9310 Steel | $100/ft | $8.00 |
| FNR Dog | 956101-009 | TW Metals | 2 | 9310 Steel | $50/ft | $5.60 |
| Input Shaft | 52170 | McMaster | 1 | Grade 5 Titanium | $80/ft | $53 |
| Intermediate Shaft | 52172 | McMaster | 1 | 300M Steel | $72/ft | $19.50 |
| Output Shaft | 52223 | Performance Titanium Group | 1 | Grade 5 Titanium | $80/ft | $27.50 |
| Reverse Idler Shaft | 52221 | Performance Titanium Group | 1 | Grade 5 Titanium | $80/ft | $15.30 |
| Gearbox Case DR | | Alro | 1 | 7075 Aluminum | $0 | $0 |
| Gearbox Case DL | | Alro | 1 | 7075 Aluminum | $0 | $0 |
| Shifting Barrel | 52165 | OnlineMetals | 1 | 4130 Steel | $118/ft | $40 |
| Selector Forks | 52213 | OnlineMetals | 2 | 4130 Steel | $118/ft | $5 |
| Fork Rail | | OnlineMetals | 1 | 4130 Steel | $118/ft | $5 |
| Spacers | | Alro | 15 | 7075 Aluminum | $0 | $0 |
| Bearings | | NSK | 13 | | $0 | $0 |
| Snap Rings | 15694 | McMaster | 6 | Spring Steel | $8.20/pkg | $8.20 |
| | | | | | **Total** | **$226** |

The total cost to MBR to manufacture the gearbox is under $250, largely in part due to how much of the materials and components we can get from our sponsors free of cost. Though the monetary cost appears pretty cheap, the more significant cost of the gearbox comes from the time it takes to manufacture it: around 6 months. This is a combination of how time-intensive making these parts is for both the team that gets a limited amount of time on Wilson Center's CNC machines and for sponsors who require large lead times as they are fitting our parts in along with their normal work load from other paying customers. MBR is well aware of this constraint and willing to accept it.

## 8.4 Manufacturing

MBR has a well established manufacturing process for the gearbox and all of its individual components that it has used for several years. These processes include a combination of in-house machining at Wilson Center by team machinists and operations performed by MBR Sponsors. The process for the major components of the gearbox – the gears, shafts, and case – are summarized below.

### 8.4.1. Gears

Our gears will be made out of 9310 Steel that MBR will need to purchase. Gears begin with in-house operations where the gear blank, essentially a puck with the dimensions that the gear will ultimately have, is made on the TL-1, a CNC lathe. A part drawing is needed for this operation and an example of the one used for the input gear is illustrated in Figure 8.4.1.1.

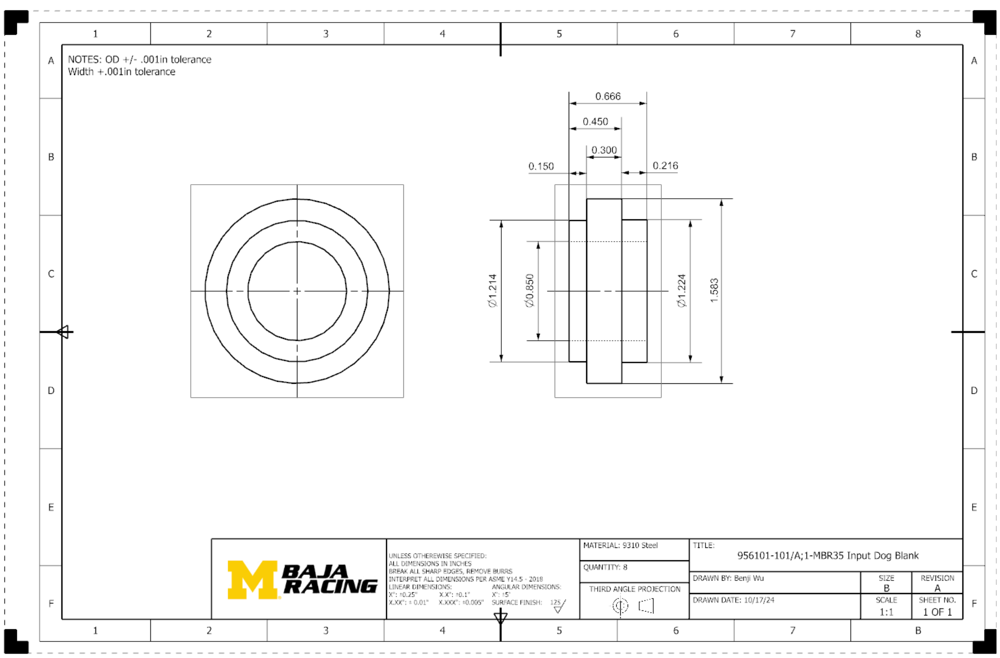



**Figure 8.4.1.1.** Part drawing of the input gear blank to be machined of the TL-1 CNC lathe with all critical dimensions and tolerances called out.

If the gear has webbing, pocketing, or any additional features such as dog teeth, these will be machined on a HAAS VF-2, the CNC 3-axis mill in Wilson Center. No part drawing is required for this operation because the team programs the VF-2 through Fusion 360 CAM (computer-aided machining) that uses the CAD geometry directly, rather than a drawing. Next, the gears go to KA-Wood Gear Manufacturing who creates the gear teeth through shaping or hobbing, leaving all dimensions slightly oversized so that the gears can be heat-treated. The part drawing sent to KA-Wood for the input gear is shown in Figure 8.4.1.2.

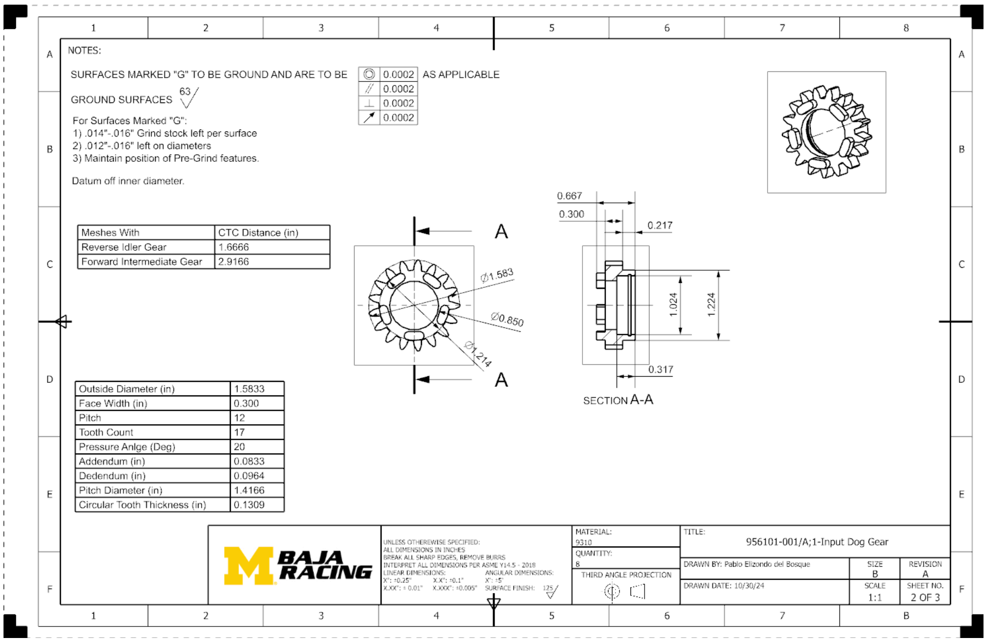



**Figure 8.4.1.2.** Part drawing of the input gear for KA-Wood to reference with important callouts of the gear teeth profile such as diameter, pressure angle, pitch, tooth count, and face width.

The gears then go to Temprite for through hardening to 36 HRC to increase the yield strength of the part and for a 0.020in 56 HRC case carburization to make the gear teeth more resistant to wear and pitting. Once this is complete, they go back to KA-Wood for a final grind to bring the gear perfectly to size before finally getting their internal splines cut at Accurate Wire EDM if the design requires it.

### 8.4.2. Shafts

The shafts will be made out of 300M Steel that MBR must purchase and grade 5 titanium that MBR gets discounted from a sponsor. Shafts begin similarly to gears with their blanks being made in-house on the TL-1. An example of this is the input shaft whose part drawing given to our lathe machinists is depicted in Figure 8.4.2.1.

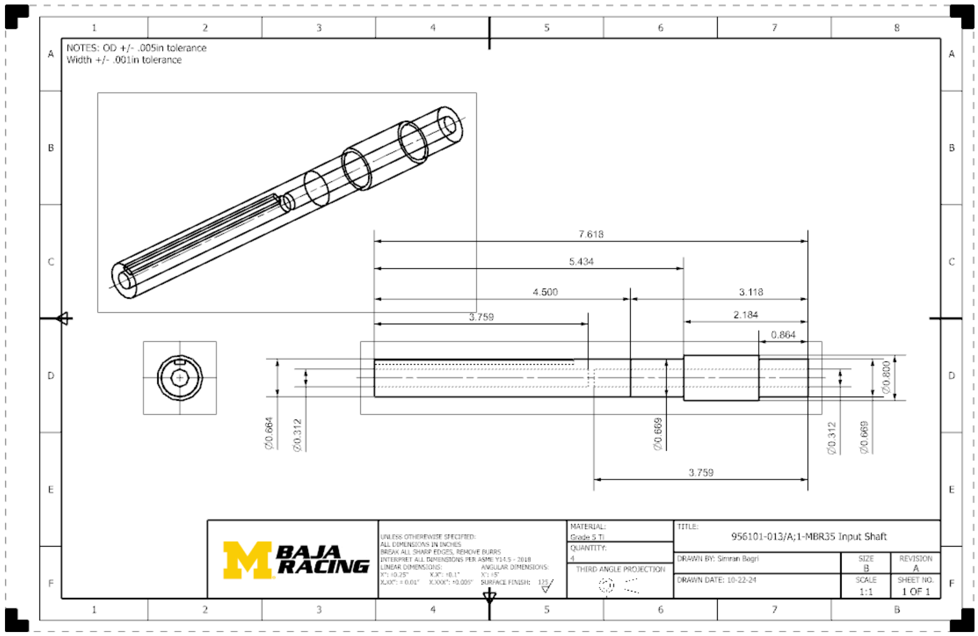



**Figure 8.4.2.1.** Part drawing of the input shaft for MBR machinists.

Once the shaft blank is complete, it is sent to Temprite for hardening to 36 HRC if it is made out of 300M steel and post machined once it comes back to fix any warping that occurred during the heat treatment, especially on bearing surfaces which are critical. Titanium shafts can skip this step. Finally, the external splines are machined onto the shaft by Modified Gear Inc. with the part drawing that is sent with it for the input shaft pictured in Figure 8.4.2.2.

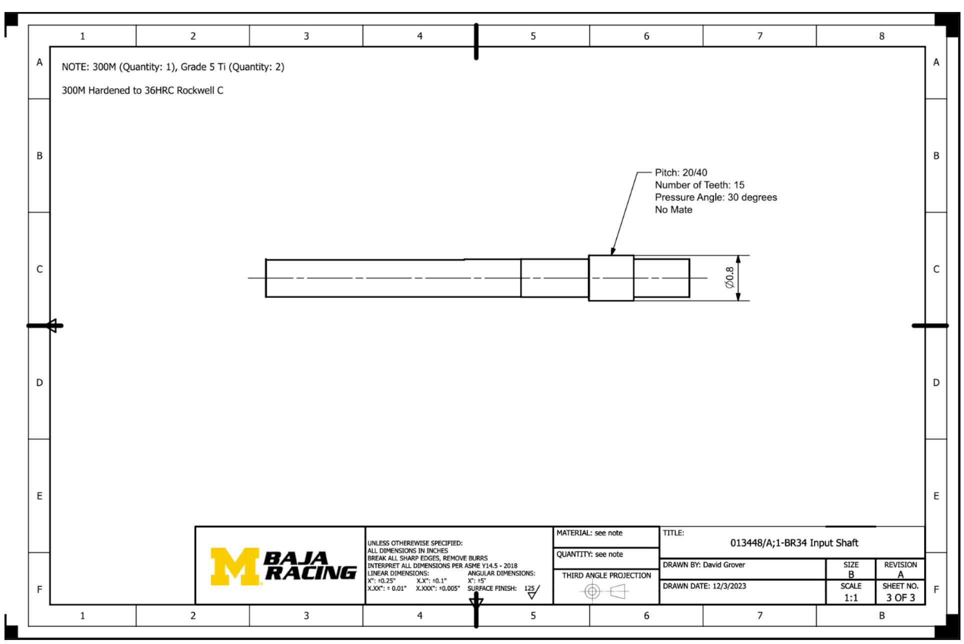



**Figure 8.4.2.2.** Part drawing of the input shaft for Modified to reference with important callouts of the pitch, number of teeth, and pressure angle.

### 8.4.3. Gearbox Case

The gearbox case is fully machined in house as two separate parts on the VF-2 with 7075 aluminum stock that is supplied free of charge from Alro, a team sponsor. For something like the case that has several important bearing surfaces with a tighter tolerance than the other features, the machinists need a part drawing like in Figure 8.4.3.1.

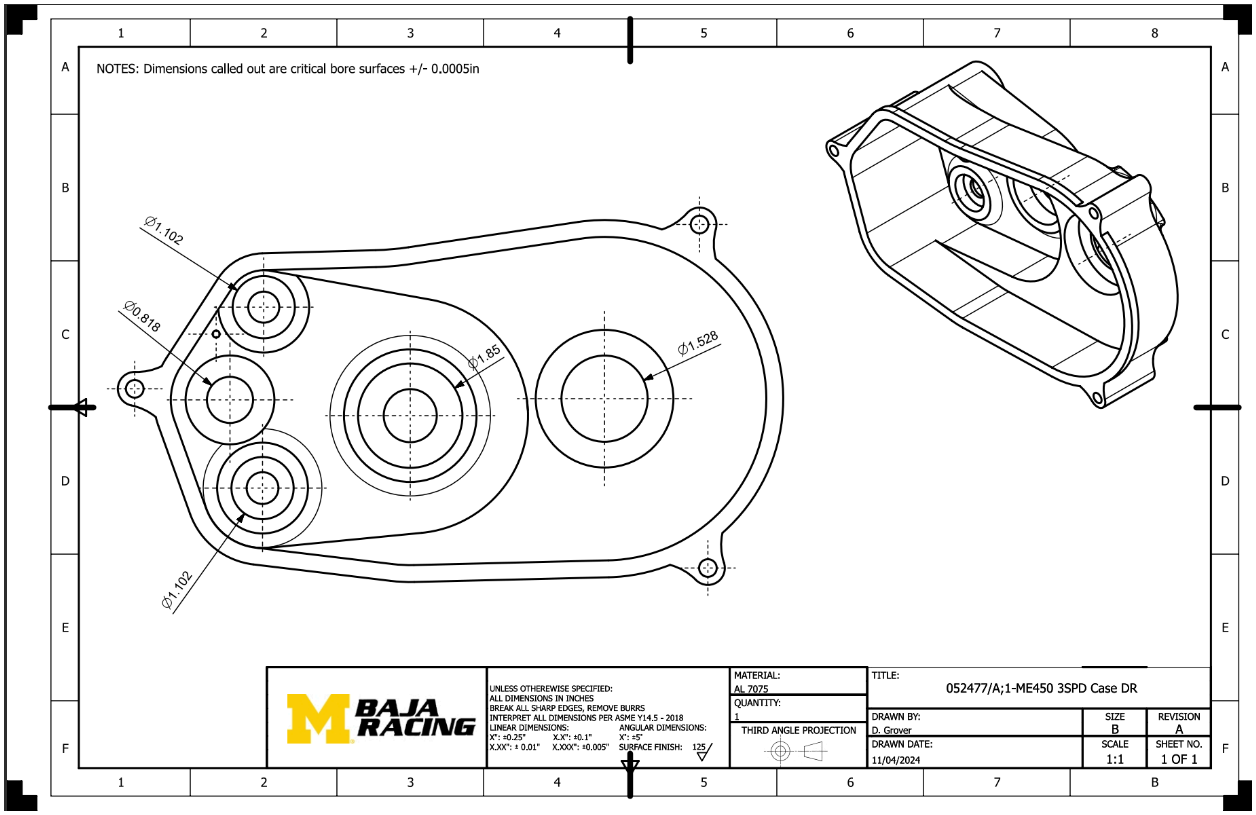



**Figure 8.4.3.1.** Part drawing of the gearbox case DR with callouts for bearing journals' tighter tolerances.

Once the gearbox case is machined, its last step is anodize at CMF to make the outer surface more corrosion and wear resistant.

## 8.5 Prototype

The team will be creating a prototype of the concept of our gearbox to present at the Design Expo. Since the prototype will likely be 3D printed out of plastic, resin, and nylon rather than machined out of metal and will not include smaller components such as spacers and bearings, it will require a separate design that accounts for these differences and is 3D printing friendly. The case will also be modified to only consist of two sidewalls with the shafts extended to go through them such that people can see the inside of the gearbox with the gears turning when they crank the input shaft. Rather than printing splines, shafts will transfer torque to gears either via polygons or by printing the shaft and gear together as one piece. The purpose of the prototype will be to test functionality of the gear shifting mechanism to make sure the shifting barrel works as intended as MBR has no previous experience with these to draw from. A successful prototype will be determined based solely on if a user can shift between all 3 gears through the shifting barrel and observe that the different gear ratios generate 3 different speeds – a high speed, low speed, and reverse – at the output shaft. Since this prototype is purely to test the functionality of gear shifts, it will not be tested for the loading requirements and loading functionality because it will be made out of weaker materials. Therefore, analysis for failure of the 3D printed components under loads generated from use by a person turning the shafts does not need to be conducted as it is not analysis significant to this project and a break in any 3D printed component will not be considered a failure in design. Since the prototype is completely 3D printed, no manufacturing plans were needed for it. A bill of materials was also not required because the only materials used in the prototype were nylon, resin, and PLA which the team received at no cost.

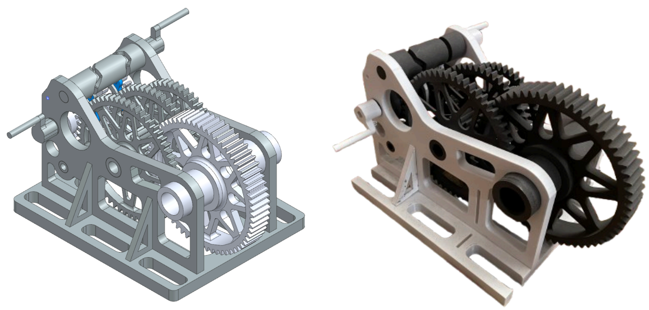


**Figure 8.5.1.** CAD (left) and actual Prototype (right) of the 3 speed gear train

The 3D printed prototype worked well when validating requirements such as having 3 different ratios and shifting between them but it also brought some issues into light. The shaft used to support the shifter fork deflected significantly due to the bending. This was mostly due to the material properties of the 3D printed PLA being more flexible than the aluminum or steel used in the final design. This supporting shaft diameter will be increased, and in order to create less bending force, the slope in the selector drum troughs of less than 45 degrees between positions.

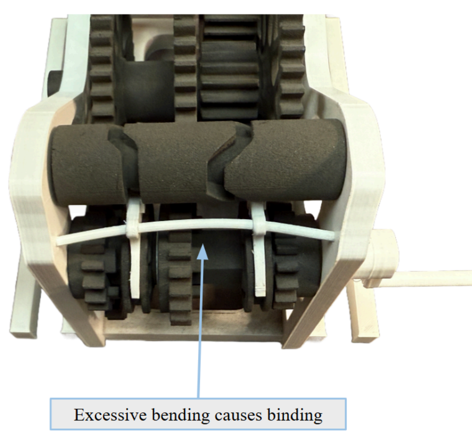



**Figure 8.5.2.** Display of the bending in the supporting shaft

# 9 Verification and Validation

While the scope of this project outlasts the duration of the class, there are still multiple specifications that could be verified and/or validated in some cases through the 3D printed prototype, CAD, and other softwares. A couple of examples are listed below:

**Backwards Compatible Packaging:** by designing this whole gearbox CAD in the frame/chassis of the most previous vehicle it can be verified that the existing subsystems for that vehicle that integrate with the gearbox can all fit without clearance issues or limiting their previous capabilities by making an assembly with said components. The main limitation with this method would be that components sometimes do not translate well from CAD to the physical product due to machining and welding tolerance build up may create some flaws that could not replicate in CAD with accuracy. However if this is kept in mind when designing, this method is the best to check for backwards compatible packaging. In the past, the gearbox these tolerance stack up issues have not been grave enough to make it non compatible to the other respective subsystems.

**Load Capability:** while adding another gear train for three gear ratio has never been done before, most of the components themselves are nothing new to Michigan Baja Racing. Due to this we are highly confident with the loads and the type of analysis done on most components (gears, shafts, cases, bearings, etc.) to optimize their designs and verify this specification because we would use the same methods that have been used for years and have created successful gearboxes with zero component failures. These methods would include FEA for gear pocketing and case, shaft bending and torsion calculations for shafts, bearing calculations, etc. A fallback with this method would be that the loads used for any type of analysis are usually calculated from theoretical situations (landing on throttle, landing on the side of the car, etc.) based on obstacles we have encountered before. However, since the obstacles we see each season are not determined until the day off, there is always the possibility of seeing an obstacle with some type of cross loading that we have not predicted or seen before. To make up for this a team standard of a 1.2 safety factor is set to across all of the components. Plus when doing hand calculations we try to round up or take conservative assumptions to make up for this uncertainty. With these methods, there have not been any component failures in the past.

**Light-weight:** By adding material properties to all of the components that were custom designed in CAD and getting the weight from all the components that were not designed by us from their supplier such as bearings, snap rings, and bolts, we were able to verify our light-weight specification of having a dry gearbox of approximately 10.53 lbs which is under the required 11 lbs. Main fallback from this method is any machining inaccuracies but the amount it usually takes is extremely minimal there should not be any issues.

**Gear Ratio Functionality:** The specification of having a high, low, and reverse gear ratio in our transmission can be validated confirming that the three different gear trains in the CAD meet those requirements by making sure one gear train has an extra idler than the other two (reverse requirement) and that the other two have different number of gear teeth in the gears (high and low reduction requirement). This was able to be completely validated with the 3D prototype that was made. By powering the different input gears, the ratio between the input and output can physically be seen between high, low, and reverse gear.

**Manufacturable Given Current Capabilities:** This can be verified through talking thoroughly to the team machinists about the manufacturing process. Most components are very similar to previous years such as gears, shafts, and the case so we would just need to double check that we have filets and hole sizes that can be done given the current tools. With the newer components like the barrel shifter, we made sure that it was still machinable through our 4th axis capabilities by confirming with our machinists and since they had not done a part like that for the gearbox in the past, CAM was written to verify that it is indeed manufacturable in house. As for our external sponsors, they are not seeing anything different conceptually from this new design in comparison to previous years so they will not have any problems on their part.

**Gear Shifting Functionality:** Through our 3D printed prototype, we would be able to shift between the three different gear ratios with the barrel shifter. While this was able to be met/validated through the 3D model, the material properties of dry, 3D printed plastic that would have a higher coefficient of friction that oiled metal alloys and some design choices such as the angles of the fork paths in the barrel shifter created a lot issues like binding when shifting. While the shifting requirement was still met either way, the product still needs some minor design changes to minimize or fix the current issues.

Other validation/verification would have to be with either the whole complete product integrated into a previous vehicle. The requirements, specifications, and how we would validate/verify them can be seen in the table below:

**Table 9.1** Requirements, Specifications, and Testing/Validation

| Requirements | Specifications | Testing/Validation |
|---|---|---|
| High efficiency | Input to output power loss < 15% | Strain gauge the rear half shaft and get the engine torque at that rpm to calculate efficiency |
| High durability | Forward driveline components survive at least 10^6 wheel revolutions<br>Reverse driveline components survive 10^4 wheel revolutions<br>All bearings rated to C10 lifetime of 10^6 revolutions at maximum loads | Perform a mock competition (endurance, accel, S&T, etc.) and examine all components afterwards to search for any signs of failure |
| Load Capability | All components designed to 1.2 SF on maximum load case<br>17kn axial loading<br>All components withstand input torque of 54 ft-lbs | Conduct drop testing (worst case loads) |
| Light-Weight | Weighs <11lbs | Check weight on scale |
| Ease of Use | < 10lbs of shifting force<br>0.25in separation between dog gear engagements | Put a force gauge on linkage and see how much force it requires to shift |

The requirements of high efficiency, high durability, load capability, and ease of use can only really be validated with a real prototype due to how the material properties highly influence the results of tests that can only be achieved by going through the long process of proper machining, heat treatment and case carburized, and precision gear teeth grinding/shaft splines which is way beyond the scope of this class. Any type of validation without any of these attributes would simply be inaccurate hence not useful. For the light weight requirement, while it can be verified through adding material properties to all custom design components and getting the weight for everything else as it was stated before, we would have to physically weigh the gearbox in a scale to fully validate the requirement. The main fallback for most of these would be the error within the sensors during data collection. At the end of the day, these are the most accurate ways to validate these requirements and the error from the sensors could be minimized through proper calibration.

# 10 Challenges/Solutions

One of the biggest challenges that we faced is being backwards compatible to a previous MBR vehicle while meeting capability and efficiency requirements. In this project's main design problem we state that we want to create a solution that increases gearbox capabilities while meeting current benchmarks that our previous transmissions meet. In order to do that we needed to potentially develop a new transmission style that is drastically different from our previous iterations while simultaneously having similar dimensions to those iterations. If we failed at doing that we would not have a reliable way to test our prototype in the real life vehicle conditions that we see. A potentially significantly different gearbox style additionally means that meeting our weight requirement was a challenge because we cannot be sure how much the gearbox will weigh until we pick a concept for it.

Another challenge was coming up with a system capable of three speed shifting with a single lever. From our requirements and specifications we have deduced that we needed at least three speeds: low gear, high gear, and reverse. In the previous year's gearbox we had only had to shift back and forth between the forward and reverse gear trains and while that had its own complications in design the concept was fairly straightforward. However, shifting between three gear trains with the movement of one lever has never been done before in our team's history, and while it is not our direct responsibility to design the linkage, we have set up the transmission in such a way that makes it possible for the driver controls lead to achieve it.

Due to the nature of the transmission components (gears, heat treated splined-shafts, complex gearbox case) and how long it takes to make them properly, timeline has been the major anticipated challenge of that project. From our previous experience in making a transmission, the whole design and manufacturing process takes about 6-7 months to complete and an additional week or two for testing. Because of the limited time we have in this project we will have to construct a simpler model, most likely 3D printed or simplified water jet/laser cut gears, to display the overall concept of our design. This will require some slight design modifications between the final concept and the manufactured prototype to adjust to whatever different manufacturing mediums we have available.

Lastly, while designing the gearbox model, two different unforeseen roadblocks were encountered. First it was discovered that the installation of the three input gears was potentially going to alter the shaft/gear design drastically. They all have to free spinning in bearings in order to not power their respective gear train when not selected and because of this they all have to be on a smooth or non-splined part of the shaft since it is good engineering practice to have bearings be in a smooth surface. However, there has to be a splined surface between each of the inputs. This entails that the shaft potentially had to go from smooth surface, splines, smooth surface, splines, and then back to smooth surface. This causes installation issues since normally the inner diameter of the input bearings have to be smaller than the outer diameter than the outer diameter of the splines. One approach that was considered was having different diameters for each surface and having them slightly increase the further along the shaft as if they were layered, but that created problems with having splines diameters that were too small to theoretically survive their respective max load case. So instead the approach that is being taken is to have the shaft be smooth surface, splines, to smooth surface as it has been in previous models and make a spline plug that has internal splines in the inner diameter surface and a smooth surface in the outer diameter as a bearing journal so the 2nd input gear's bearing can utilized properly. This meant that a bearing with a bigger inner diameter had to be chosen so to help not drastically change the initial designs of the input gears, the low reduction (high speed) input gear was moved from its initial position in the left to the middle since it is bigger than the other two inputs making it easier to select a bigger bearing without sacrificing any component strength or adding too much complexity.

The other design roadblock was that the shaft/bearing location caused the case to exceed 2.5 inches width due to the nature of the design choice of adding another gear train. However, due to the fact that we set the 2.5 inches specification to make sure that the brake's rear caliper can still fit in the frame this specification is not necessary to meet along the whole gearbox. Consequently, the gearbox case was only made wider in the front exactly where it needs to be due to the three intermediates being stacked laterally there while in the rear there is only the output gear. Now the rear of the gearbox is well within the 2.5 inches of width requirement creating more than enough space to fit the brakes which was again the main reason that specification was created initially.

Lastly, while building the prototype, we initially printed a few components such as the shifter fork and shifting barrel out of PLA. However, upon receiving the printed parts, we realized their print quality and tolerances were not up to our standards. Therefore, we chose to reprint them out of resin instead, and found that they functioned in our prototype better. This did not have any impact on our budget or timeline because reprinting did not take up much time and the materials to print was free of cost for us.

# 11 Project Status

Our key milestones and tasks are outlined in our project plan in the form of a Gantt chart [20], shown in Figure 6.1 below. This includes detailed steps in addition to the course deadlines that are specific to our project.

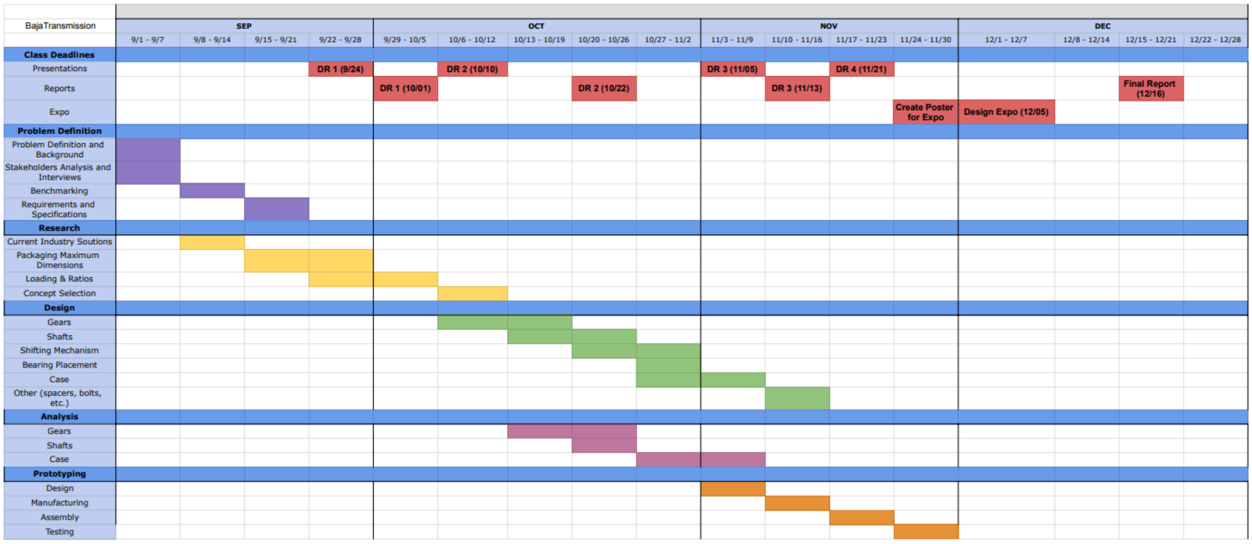



**Figure 6.1.** A high-level timeline of our project plan.

As shown above, our timeline consists of the ME 450 deliverables with hard deadlines below and further detailed steps above. The more team-specific tasks are both a breakdown of the requirements for the deliverables for ME 450 and the details for our project in particular. At this point in time, we have clearly defined the problem and completed the research, interviews, requirements, and performance specifications. We have also determined our maximum packaging dimension, maximum loadings, and gear ratios for the gearbox from resources existing within the team. For example, a Python script generated by a now alumnus of the team that models the entire drivetrain system was used to determine the gearbox ratios we need for optimal speed and torque while a free-body diagram was sketched to calculate maximum loads on the gearbox in a worst case scenario. Additionally, we have completed our design generation and selection process as well as finalized our CAD design. Manufacturing plans are in place for each of these components based on MBR's manufacturing sponsors and previous experience making them. Furthermore, analysis of each component has been conducted such as FEA on the gears and case, beam bending and spline shear calculations for the shafts, and bearing load calculations to ensure they can all survive their worst case loading scenarios. Lastly, the team has completed the design of our prototype, 3D printed it, and assembled it. The poster to present along with the prototype at the ME Design Expo on December 5th was also created so ultimately, all project deliverables were accomplished resulting in a successful project.

Although the team does not have a set budget provided by our sponsor, our team is committed to keeping our costs as low as possible while delivering a viable product. As members of the MBR team as well as our ME450 team, we have access to all MBR's resources which range from 3D printers, CNC machines, and tooling to their manufacturing sponsors and material stock. These covered any expenses that the 450 team had to incur from cost of materials, such as resin and nylon, and manufacturing of the prototype. However, if the 450 team decided they needed to make a purchase vital to the development of the gearbox, we had MBR's support to do so.

# 12 Discussion

If given more time and resources, our team would look more into traction as an aspect of this project alongside torque. Being a team that builds an all terrain vehicle as light as possible to optimize speed usually has a negative impact on our traction. While it is not enough of a drawback to reconsider making the vehicle heavier and our competitors also struggle with the lack of traction, hence why it does not affect our points breakdown greatly, it is still an aspect of our transmission that could use improvement that would ultimately give us the upper hand. This would mean adding a traction capability to our Kepner-Tregoe Decision Matrix and most of our top design choices would stay somewhat similar while being integrated with a LSD (limited slip differential) in the output that distributes the power through the two rear wheels. Of course going deeper into the traction route in this project would create some packaging and complexity concerns but in terms of the direction this project would take if given the time and resources this is definitely the next step.

In our designs the main strengths root from the fact that we met all of the requirements that could be met within this semester alongside the requirement that directly was rooted straight from our initial problem definition: increasing the torque capability of the transmission while not hindering any of the current capabilities. Our completed design has low, high, and reverse gear ratio, fits in a previous vehicle, and under the weight requirement that was set all while limiting complexity in manufacturing and design. With that, the biggest weakness of this project would have to be the shifting mechanism that selects the gear ratio. The one that we designed tends to bind often which creates issues in the shifting process. Some of these issues were fixed like increasing the contact area of the shafts from the forks. Some issues will be highly minimized with the real product like the friction between all moving components in metal alloys covered in gearbox oil will be much lower than the dry, 3D printed plastic. If we kept going with this project we would ultimately have to redesign the shifting barrel. The angles between ratios that drive the shifting of the forks will have to be less steep specifically to transfer more force laterally (along the axis of movement of the fork) rather than radially in comparison to the 50/50 split we had with the initial 45 degrees. This would most likely result in an increase in diameter of the barrel since there would be more travel in the forks between ratios. This would affect the fork design as well alongside the overall packaging of the gearbox which leads to a gearbox case redesign as well.

One of the biggest challenges that were encountered in our design process would have to be packaging. While adding a third gear train was the least complex design in manufacturing and design it did make the gearbox wider. This highly affects the brakes and rear half shafts subsystem leads. What was done to minimize this we made the gearbox case extrude out more in the front, where there were the three gear trains, and decrease in width in the rear, where only the output sits, since that is where the rear brakes and rear half shafts interact with the transmission. This would create additional complexity into manufacturing the case but it would still be possible with our current machining capabilities while allowing our gearbox to fit in a previous vehicle which was one of our main requirements for testing in this project.

# 13 Reflection

As a part of the Mechanical Engineering Department, our team is committed to making the world work better. Throughout our project, each design decision was made with the goal of improving the Michigan

Baja Racing team's performance at their competitions in mind. While this was the primary goal of the project, we also considered the project's impact on broader society when it was appropriate. Throughout our project, we adhered to the SAE hazardous release of energy guarding rules to protect the safety and welfare of mechanics and bystanders while the gearbox is in use. Our gearbox is designed for a very specific use case in a very specific competition, so our design is not terribly applicable in a global marketplace. However, this type of gearbox could be applicable in recreational vehicles such as those manufactured by companies such as Polaris. Our prototype was made from 3D printed materials, notably nylon and PLA and have few societal impacts associated with them. The prototype will also be saved for furthering team knowledge transfer and will not be disposed of in the near future. In the event that it is disposed of, it will likely be landfilled or recycled. The projected societal and economic impacts of the actual prototype (that was not manufactured within the scope of this project) can be found in Section 3.3. These impacts have not been changed but were not realized within the scope of ME450 due to only the 3D printed prototype being manufactured. We used the stakeholder map in Section 3.2 to characterize the potential societal impacts of our design. Throughout the project, cultural, privilege, identity, and stylistic similarities and differences between each team member did not significantly impact the approaches our team took throughout the project. However, stylistic differences between team members were evident in presentation and report writing skills as well as in how each member wanted to approach the engineering problem and other course requirements. These discrepancies were minor such as for reports, a couple members wanting to start working on it immediately whereas others did not have time until later, so the report was divided into sections for each member to work on when they had time. Since the team was sponsored by the Michigan Baja Racing team, there were almost no cultural differences between the team and the sponsor due to the team members all being members of the Michigan Baja Racing team since freshman year. Similarly, there were no privilege or identity influences between the team and the sponsor, and few stylistic differences between the team and the Michigan Baja Racing team. We ensured we were including diverse viewpoints of stakeholders by interviewing them when determining our specifications and requirements. We spoke extensively with leads of subsystems that interface directly with our gearbox as well as the driver to ensure their ideas were incorporated into our final design. There were not any noticeable competing ideas between our stakeholders so we were able to adequately include them. The power dynamics were as predicted in Section 3.3, with the team taking extra care to not overpower the sponsor due to any perceived seniority or technical authority. Idea selection was left up to the Kepner-Tregoe decision matrix eliminating subjectivity and cultural biases. To ensure inclusion and equity, all team members agreed with the Analytical Hierarchy Process which set weights for the Kepner-Tregoe matrix. While our team members come from a diverse set of backgrounds, we were able to communicate well despite those differences to produce a final engineering solution to our design problem, and do not think those cultural differences played any significant role in the design processes. There were minimal ethical dilemmas involved with this project due to its small scale. But, as mentioned in section 3.3, a potential concern that may be encountered later in the life of this project, specifically during its manufacturing which is out of the scope of this class, is enlisting MBR team members to manufacture the gearbox. We will resolve this dilemma by ensuring that the majority of the manufacturing work is done by ourselves, unless specialized expertise is required in the manufacturing of our product. We expect this dilemma to be kept to a minimum due to the depth of manufacturing experience present on our team. This would also minimize any manufacturing related ethical concerns that might arise if our product was to be released on the market. Our team ethics align well with the University of Michigan's ethics as well as future employers' ethics–there will be absolutely no forced labor during any phase of our project, and precautions will be taken to protect the work environment. Moreover, we did encounter an ethical dilemma during concept selection when narrowing down which of our generated concepts to use for our final design. Through the Kepner-Tregoe decision matrix, we came down to two designs we could use – the 2024 gearbox with an added gear-train and the planetary gearbox – that were within 0.04 points of each other. Though the planetary gearbox lost, our 450 team preferred to design that because it was something new for us and for Michigan Baja Racing whereas the 2024 gearbox with an added gear-train was simply an iteration of our previous year's gearbox. However, we trusted our decision matrix that had labeled the iterative gearbox as the winner and understood why that would be better for the MBR Team. Thus, we ultimately decided to follow the decision matrix with the 2024 gearbox with an added gear-train and produce a design that would be more beneficial for the team, even if it was not our preference.

# 14 Recommendations

Our main recommendation is a redesign of the selector drum and shifter forks. The prototype gave good insight into the downfalls of the current design, including the sharp angle of the selector drum paths, and the thin supporting shaft of the shifter forks. For a fully working final design, the supporting shaft should be larger, possibly around ¼" in diameter, and the selector drum should utilize less steep angles. Testing on angles should be completed to confirm the steepest angle possible, as decreasing the angle increases the overall size of the selector drum and there is limited space between the drum and the engine heat shield above. The introduction of neutral between each selection would also be helpful, but further increases the overall size of the selector drum. The shifter forks should continue to have longer support surfaces, as it helps decrease the overall force on the support shaft and decreases friction. The implementation of Oilite bushings could help with reduction in friction as well.

Besides a redesign of the current concept, we also recommend fully fleshing out a planetary concept with the gear dimensions derived earlier in the report. We believe it could work similarly well to the concept developed and would have better overall packaging, which was the main downfall of the new design. The Kepner-Tregoe design matrix also shows that they are very close with the given parameters and weights on the categories, and it would be good to have two fleshed out designs to determine what works better in its final form.

# 15 Conclusion

The Michigan Baja Racing team has requested that our team reevaluate their current gearbox setup. In order to help keep the MBR team competitive with other Baja SAE teams, our project team will compare the benefits of the current MBR gearbox design to that of alternative designs, and ultimately create a prototype of our recommended design.

The main motivation of our project is to help the MBR team maximize the amount of points gained in each dynamic and endurance event across the three competitions in a season. To determine where to focus our energies, we analyzed the cause of points lost over the past 6 competitions: across two racing seasons, the MBR team lost an average of 20% of points due to lack of torque. Furthermore, during the 2023 season (run without a reverse gear), 25% of points lost were from halted forward progress, which decreased to only 3% of points lost in the 2024 season after the implementation of a reverse gear. This decrease in points lost due to halted forward progress has prompted the team to keep reverse in future iterations of the vehicle. Based on this points breakdown, the team has decided it stands to gain the most points from resolving the issue of lack of torque outputted by the gearbox.

The requirements and specifications place an emphasis on backwards compatibility with past vehicles. These requirements and specifications were determined through interviews with past and current subsystems leads on the MBR, and by consulting past MBR design materials. The backwards compatibility requirement implies that our project team must work within the current systems implemented by the MBR team, and minimally impact surrounding subsystems without sacrificing performance.

The current benchmarks for this project include past MBR gearboxes as well as commercially available gearboxes for similar vehicles such as side-by-sides and go-carts. Using these benchmarks as inspiration, we deconstructed our design space into distinct subsystems within the project. From these subsystems we created a morphological chart to help generate ideas, and also created a concept tree which helped not only generate numerous ideas but also eliminate the ones that clearly would not fit under our requirements and specifications. In addition to these methods, we also used design heuristics to further increase our concept pool. By sketching out each design, we eliminated impractical designs while also gathering data on each design: (1) number of gear ratios, (2) part count, (3) manufacturability, (4) mass, and (5) efficiency loss. Preliminary gear analysis was conducted on each design in order to ensure compliance with the requirements and specifications and to provide a mass & volume estimate. The five data points were each inputs to an analytical hierarchy process which in turn informed a Kepner-Tregoe decision matrix through pairwise comparison. The Kepner-Tregoe decision matrix guided our decision in selecting the 2024 transmission with an extra gear train as our selected design.

The selected design is an evolution of the 2024 transmission with an extra gear train added in parallel. Updated ratios to provide more torque and higher speed across the two forward gear trains, and the reverse gear train preserves the ability to back out of obstacles and continue forward progress. It has several main advantages, as it achieves all three gear ratios that were specified, and compared to the other 3-speed concepts, it has a lower part count, less manufacturing setups, less overall mass, and higher efficiency.

To determine the face widths of the gears, we used Lewis Bending Stress, Hertzian Contact, and fatigue through pitting and bending equations. Once gear sizes and placement were established, the rest of the gearbox was designed around them with a focus on packaging it within a previous vehicle's architecture. Shifting was achieved through the use of male and female dog gears that select between the high, low, and reverse gears via a shifting barrel.

Several types of engineering analysis was conducted on the various gearbox components to ensure they met our design requirements. Bending and torsion calculations were performed on the shafts to ensure they would not yield and bearing calculations for C10 lifetimes ensured they were sized correctly for the loads they would see. Finite element analysis was also run on the gear webbing and case to determine the maximum stress each would see. These types of analysis verified all components had a 1.2 safety factor on their worst case loading scenarios to help us meet our durability and load capability specifications.

Manufacturing plans that made use of MBR sponsors and a bill of materials were outlined to organize all components needed for the gearbox, their manufacturing, and their assembly, meeting our first deliverable for this project: a comprehensive plan for the construction and integration of the final design into the rest of the vehicle. The other deliverable that determined the success of our project was a manufactured prototype of our design. This was 3D printed out of PLA and nylon, meant only to test functionality shifting between the 3 different gear speeds. Thus, this required a separate design that omits spacers and bearings is more 3D printable. The shaft of the initial 3D print design that the shifter forks were mounted to was too weak to support the forks which would bind easily against it making it difficult to actually shift between gears. However, after a couple of design changes involving replacing that PLA shaft with a steel one and increasing the width of the fork mount onto the shaft, we were able to mitigate these affects and prove shiftability of our prototype, thereby validating the barrel drum design.

# 16 Acknowledgments

We would like to acknowledge continued long-time support of the Wilson Student Team Project Center, the Ford Robotics Makerspace, and Michigan Engineering, specifically Christopher Gordon, Casey Dixon, Jared Roy, Blake Desrosiers, Alyssa Emigh, and Jon Luntz. We would also like to thank Michigan Baja Racing and their sponsors: Temprite, K-A Wood Gear, Modified Gear and Spline, Accurate Wire EDM, and Distinctive Manufacturing. Finally, we would like to thank Jon Estrada and Jeffery Kohler.

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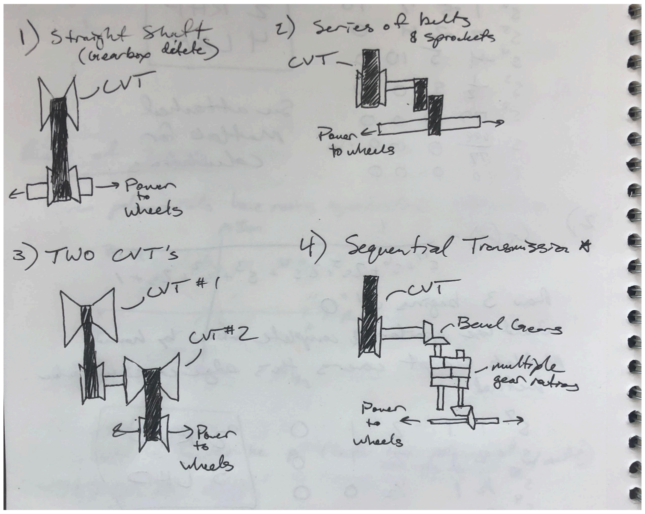
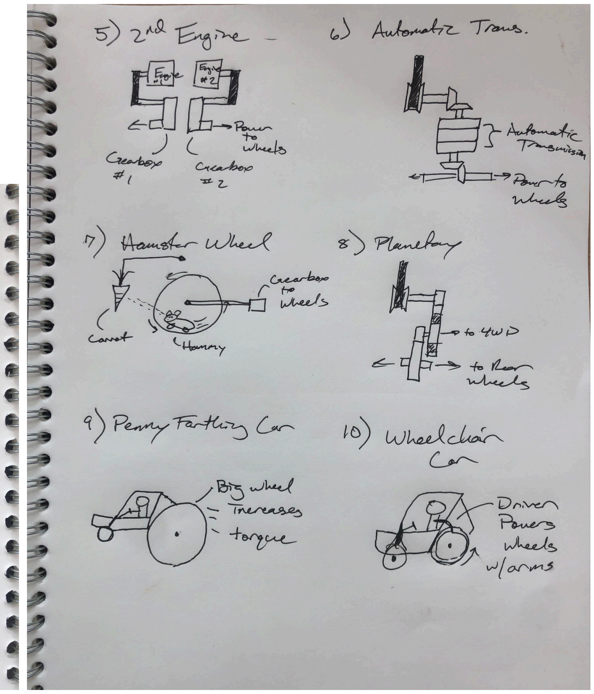
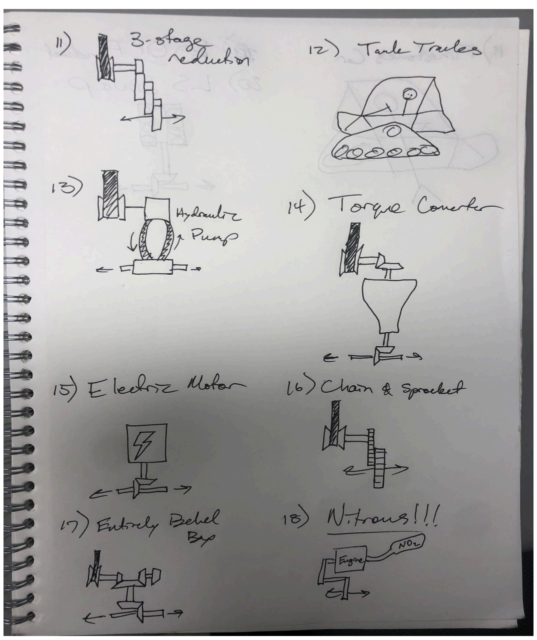
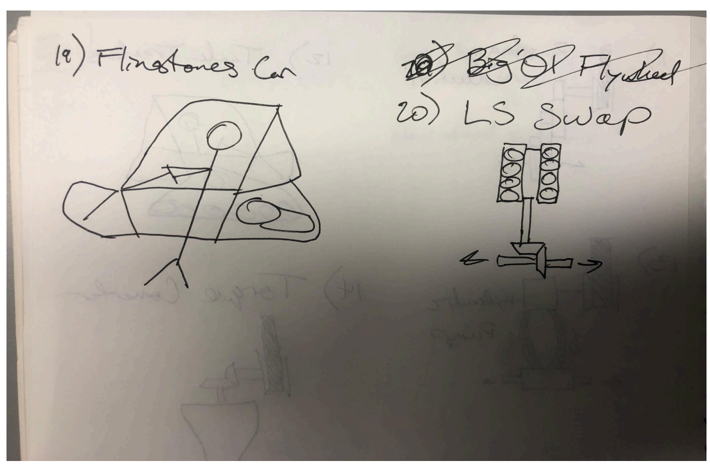
# Appendix B - All Generated Concepts

David’s Ideas

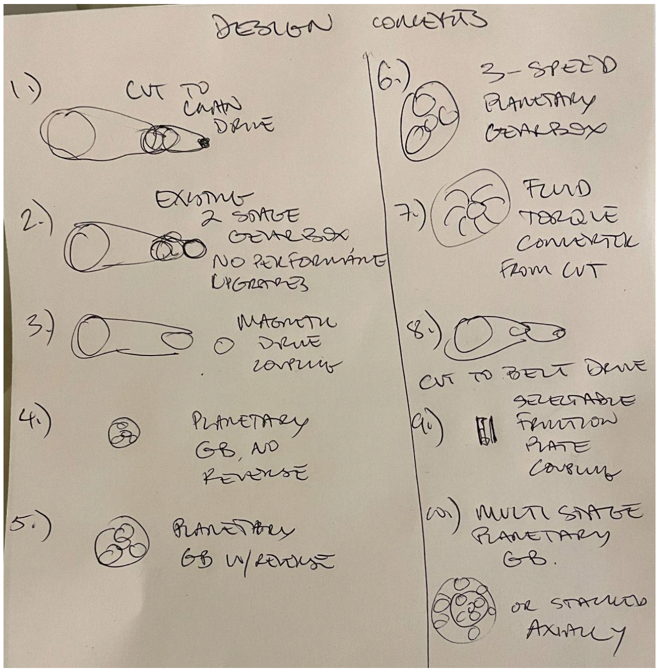


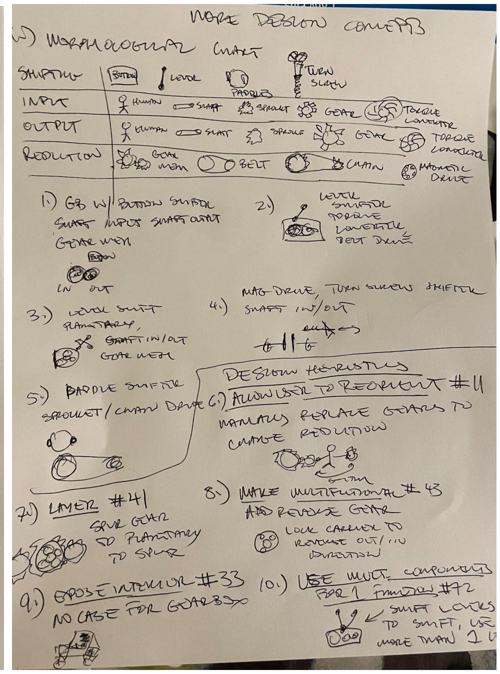



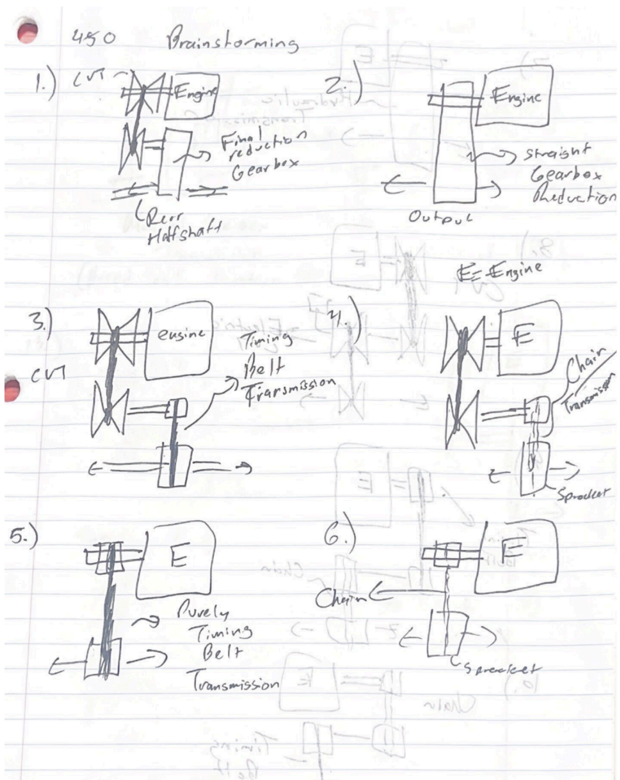



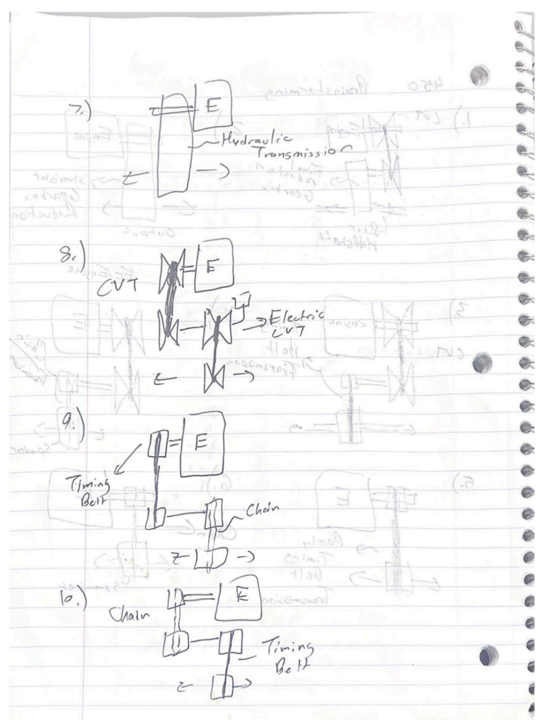

Benji’s Ideas

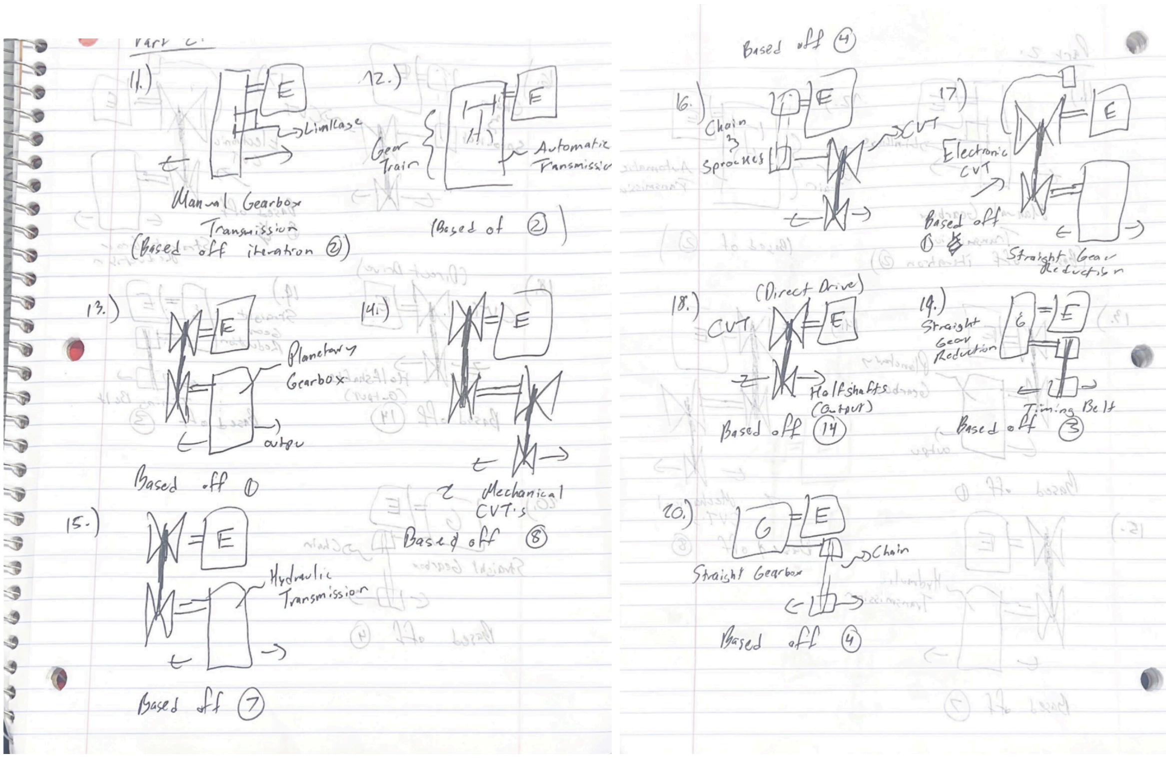



Pablo’s Ideas



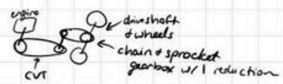





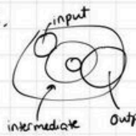

Simran's Ideas

Concept Exploration

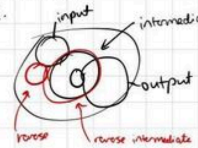
1. Direct Drive from CVT to rear drive shafts (no reduction in the gearbox)
2.



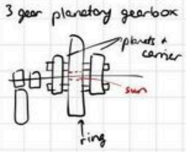

3. 2 stage gear reduction



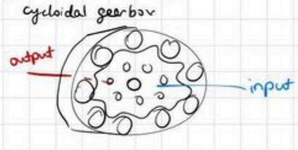

4. 2 stage gear reduction with a reverse gear




5. planetary gearbox with reverse
6. Direct drive from engine to rear drive shafts (no reduction in gearbox + no CVT)
7. 3 gear planetary gearbox (high, low, reverse), each gear comes from locking a different component

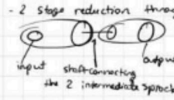



8. gearbox with helical gears instead of spur gears
9. cycloidal gearbox

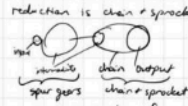



10. harmonic drive gearbox

11. Chain + Sprocket Transmission
    - builds off idea 2
    - 2 stage reduction through 2 sets of sprocket / chains




12. 1 Spur gear reduction + 1 chain + sprocket
    - combines ideas 2 + 3 where 1 gear reduction is gears and the second reduction is chain + sprocket

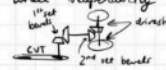



13. worm gears in gearbox for more torque transfer

Morph Chart

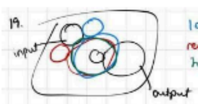
| Sub Function | Solutions | | | |
|---|---|---|---|---|
| 2 reductions | planetary | spool | chain + sprocket | differential |
| switch between 3 gear stages | dogs | friction clutch | EM clutch | electronic controls |
| Types of Gears | spur | helical | worm | bevel |

14. planetary gear box with dogs to control what is locked
    high gear: sun locked to carrier  low: sun locked to case
    reverse: carrier locked to case
15. spool gearbox with friction clutch to adjust how much torque sent from gearbox to drive shafts
16. differential gearbox to adjust how much power sent to each wheel independently (contains helical gears to reduce noise)
17. spool gearbox w/ bevel gears




18. electronic controls in planetary gearbox to change between gears in differential gearbox

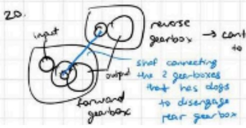
19.




low gear train
reverse gear train
high gear train

- builds on idea 4 by adding another gear train for low gear
- uses spur gears

20.




reverse gearbox → contains idler gear to reverse direction

shaf connecting the 2 gearboxes that has dogs to disengage rear gearbox

# Appendix C - Lewis Bending and AGMA 2001-D04 Equations [21]

$$\sigma_{Lewis\ Bending} = \frac{K_v W^t P}{FY}$$

$$\sigma_{Contact} = C_P \sqrt{W^t K_o K_v K_s \frac{K_m C_f}{d_p F I}}$$

$$\sigma_{Fatigue\ Bending} = \frac{S_t}{S_F} \frac{Y_N}{K_T K_R}$$

$$\sigma_{Fatigue\ Pitting} = \frac{S_c Z_N C_H}{S_H K_T K_R}$$

| Syb | Name | Syb | Name |
|---|---|---|---|
| $C_f$ | Surface condition factor | $K_s$ | Size factor |
| $C_H$ | Hardness-ratio factor | $K_T$ | Temp factor |
| $C_p$ | Elastic Coeff | $K_v$ | Dynamic factor |
| $d_p$ | Pinion PD | P | Diametral pitch |
| F | Face width | $S_c$ | AGMA surf strength |
| I | Geom factor | $S_t$ | AGMA bending strength |
| $K_m$ | Load dist factor | $S_F$ | SF bending |
| $K_o$ | Overload factor | $S_H$ | SF pitting |
| $K_R$ | Reliability Factor | $W^t$ | Load |
| $Y_N$ | Stress cycle factor bending | $Z_N$ | Stress cycle factor for pitting |

Example: Reverse Input Gear (Left) and Intermediate Gear (Right)

## Lewis Bending Stress and Contact Stress (Hertz Equations) Analysis

| **Lewis Bending Stress Calculations (Full Underdrive, Max Torque), Power Based Equation** | | | | | |
|---|---|---|---|---|---|
| Max Tooth Load @ Pitch | 845.6 | lbs | Max Tooth Load @ Pitch | 1117.2 | lbs |
| Tooth Lewis Form Factor | 0.302 | | Tooth Lewis Form Factor | 0.399 | |
| Pitch Line Velocity | 304.8349036 | ft/min | Pitch Line Velocity | 337.1115405 | ft/min |
| Max. Torque (in-lbs) | 598.9666667 | | Max. Torque (in-lbs) | 2187.85 | |
| Max Torque (ft. lbs) | 49.91388889 | | Max Torque (ft. lbs) | 182.3208333 | |
| Max. Horsepower | 7.811163469 | | Max. Horsepower | 11.41275797 | |
| Max Torque w/contact ratio (ft-lbs) | 81.32342597 | | Max Torque w/contact ratio (ft-lbs) | 297.0506831 | |
| Face Width to Circular Pitch | 1.14591559 | | Face Width to Circular Pitch | 1.14591559 | |
| | | | | | |
| Power | 8.457142857 | hp | Power | 8.457142857 | hp |
| Tangential Force on Teeth | 561.9253905 | lbs | Tangential Force on Teeth | 508.1240234 | lbs |
| Torque | 37.07146674 | ft-lb | Torque | 86.45165675 | ft-lb |
| Max Bending Stress | 74427.20404 | psi | Max Bending Stress | 50939.75172 | psi |
| **Safety Factor from Safe Stress** | 1.504826111 | 2.335894498 | **Safety Factor from Safe Stress** | 2.198675813 | -7.21825 |
| | 1.539977261 | | | 2.039969878 | |
| **Contact Stress Analysis (Hertz Equations)** | | | | | |
| Poisson's Ratio (nu) | 0.3 | | Poisson's Ratio (nu) | 0.3 | |
| Young's Modulus (E) | 29000000 | psi | Young's Modulus (E) | 29000000 | psi |
| Elastic Coefficient | 2252.10362 | | Elastic Coefficient | 2252.10362 | |
| Form Factor | 0.2188213139 | | Form Factor | 0.2188213139 | |
| Contact Stress | 175.0606033 | ksi | Contact Stress | 100.1173829 | ksi |
| Allowable Contact Stress (Table 9-3) | 225 | ksi | Allowable Contact Stress (Table 9-3) | 225 | ksi |
| **Safety Factor** | 1.285269191 | | **Safety Factor** | 2.24736198 | |

## Fatigue Analysis

| | | | | | |
|---|---|---|---|---|---|
| Fatigue limit for bending | 535 | MPa | Fatigue limit for bending | 535 | MPa |
| Fatigue limit for contact stress | 930.7924767 | MPa | Fatigue limit for contact stress | 930.7924767 | MPa |
| Temperature Factor | 1 | | Temperature Factor | 1 | |
| Reliability Factor | 1 | | Reliability Factor | 1 | |
| Hardness Ratio Factor for Pitting Resistance | 1 | | Hardness Ratio Factor for Pitting Resistance | 1 | |
| Bending Stress | 513.1576414 | MPa | Bending Stress | 351.217316 | MPa |
| Contact Stress | 1207.000686 | MPa | Contact Stress | 690.2852358 | MPa |
| Elastic Coefficient | 1 | | Elastic Coefficient | 1 | |
| Transmitted Tangential Load | 3754.464 | N | Transmitted Tangential Load | 4960.368 | N |
| Overload Factor | 1 | | Overload Factor | 1 | |
| Dynamic Factor | 1 | | Dynamic Factor | 1 | |
| Load Distribution Factor | 1 | | Load Distribution Factor | 1 | |
| Size Factor | 1 | | Size Factor | 1 | |
| Surface Condition factor | 1 | | Surface Condition factor | 1 | |
| Rim thickness factor | 1 | | Rim thickness factor | 1 | |
| Face width | 7.62 | mm | Face width | 0 | mm |
| Pitch Diameter | 35.98333333 | mm | Pitch Diameter | 99.48333333 | mm |
| Geometry Factor for pitting | 1 | | Geometry Factor for pitting | 1 | |
| Geometry factor for bending | 1 | | Geometry factor for bending | 1 | |
| | | | | | |
| Stress cycle factor for bending (Zn) | 1.1 | Can also get from graphs below | Stress cycle factor for bending (Zn) | 1.08 | Can also |
| Stress cycle factor for pitting resistance (Yn) | 1.25 | Can also get from graphs below | Stress cycle factor for pitting resistance (Yn) | 1.19 | Can also |
| | | | | | |
| Permissible Bending Stress (sigmaF) | 588.5 | | Permissible Bending Stress (sigmaF) | 577.8 | |
| Permissible Contact Stress (sigmaH) | 1163.490596 | | Permissible Contact Stress (sigmaH) | 1107.643047 | |
| | | | | | |
| **Safety Factor on bending** | **1.146821079** | 2.282575932 | **Safety Factor on bending** | **1.645135287** | -7.77981 |
| **Safety Factor on pitting** | **1.263951893** | -28.25405424 | **Safety Factor on pitting** | **1.604616454** | 0.328001 |

# Appendix D - Shaft Bending and Spline Shear Calculations

Bending Stress: $\sigma_{bending} = \frac{M\,y}{I} = \sigma_{zz}$

Shear Stress: $\boldsymbol{\tau} = \frac{T\,r}{J}$

Principal Stress Equations:

$$\sigma_{1,2} = \left(\frac{\sigma_x + \sigma_y}{2}\right) \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

Von Mises Stress Equation

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2)}{2}}$$

# Appendix E - Bearing Calculations

NSK Single-Row Deep-Groove Ball Bearing Catalog

| Boundary Dimensions (mm) | | | | Basic Load Ratings (N) | | Factor | Limiting Speeds ($min^{-1}$) | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Grease | | Oil |
| $d$ | $D$ | $B$ | $r$ min. | $C_r$ | $C_{0r}$ | $f_0$ | Open Z · ZZ V · VV | DU DDU | Open Z |
| **17** | 26 | 5 | 0.3 | 2 630 | 1 570 | 15.7 | 26 000 | 15 000 | 30 000 |

**Dynamic Equivalent Load**

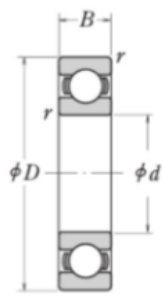
$P = XF_r + YF_a$

| $\frac{f_0 F_a}{C_{0r}}$ | $e$ | $\frac{F_a}{F_r} \leq e$ | | $\frac{F_a}{F_r} > e$ | |
|---|---|---|---|---|---|
| | | $X$ | $Y$ | $X$ | $Y$ |
| 0.172 | 0.19 | 1 | 0 | 0.56 | 2.30 |
| 0.345 | 0.22 | 1 | 0 | 0.56 | 1.99 |
| 0.689 | 0.26 | 1 | 0 | 0.56 | 1.71 |
| 1.03 | 0.28 | 1 | 0 | 0.56 | 1.55 |
| 1.38 | 0.30 | 1 | 0 | 0.56 | 1.45 |
| 2.07 | 0.34 | 1 | 0 | 0.56 | 1.31 |
| 3.45 | 0.38 | 1 | 0 | 0.56 | 1.15 |
| 5.17 | 0.42 | 1 | 0 | 0.56 | 1.04 |
| 6.89 | 0.44 | 1 | 0 | 0.56 | 1.00 |

**Static Equivalent Load**

$\frac{F_a}{F_r} > 0.8,\ P_0 = 0.6F_r + 0.5F_a$

$\frac{F_a}{F_r} \leq 0.8,\ P_0 = F_r$




$$L_{10} = \left(\frac{C}{F}\right)^P \left(\frac{10^6}{60\ \times N}\right)$$

where:

C = dynamic load rating (numbered of rolling elements, roller length/diameter and contact angle

F = applied load

P = 3 (ball bearings) or 10/3 (roller bearings)

N = RPM

Example: Input Gear Bearing

| | **Input Gear** |
|---|---|
| **Revolutions in Life** | 8527720.722 |
| **Load (N)** | 343.9333828 |
| **Dynamic Load Rating (N)** | 1570 |
| **L10 Life** | 95120902.78 |
| **Safety Factor** | 11.1543173 |

# Appendix F - Build Design Bill of Materials

| Part Name | Quantity | Supplier | Material | Price per Unit | Total Price |
|---|---|---|---|---|---|
| Intermediate Assembly | 1 | FRB Makerspace | Nylon | $0 | $0 |
| High Input Gear | 1 | FRB Makerspace | Nylon | $0 | $0 |
| Low Input Gear | 1 | FRB Makerspace | Nylon | $0 | $0 |
| Reverse Input Gear | 1 | FRB Makerspace | Nylon | $0 | $0 |
| FNR Dog | 2 | FRB Makerspace | Nylon | $0 | $0 |
| Output Assembly | 1 | FRB Makerspace | Nylon | $0 | $0 |
| Barrel Drum | 1 | FRB Makerspace | Nylon | $0 | $0 |
| Crank Handles | 2 | FRB Makerspace | PLA | $0 | $0 |
| Side Walls | 2 | FRB Makerspace | PLA | $0 | $0 |
| Base | 1 | FRB Makerspace | PLA | $0 | $0 |
| Gussets | 4 | FRB Makerspace | PLA | $0 | $0 |
| Shifter Forks | 2 | FRB Makerspace | PLA | $0 | $0 |
| Input Shaft | 1 | FRB Makerspace | Nylon | $0 | $0 |
| Fork Shaft | 1 | Wilson Center | Steel | $0 | $0 |
| Reverse Idler | 1 | FRB Makerspace | Nylon | $0 | $0 |
| | | | | **Total** | **$0** |

# Appendix G - Author Bios


**Simran Bagri**
*B.S.E Mechanical Engineering Winter '25*

Simran is a senior in mechanical engineering with minors in computer science and multidisciplinary design. She grew up in San Jose, California, and spent her time after school playing on the varsity basketball team. Simran decided to become a mechanical engineer after loving the STEM classes she took in high school. On the Baja Racing team, the notable roles she held were the front brakes lead for MBR33 and the in-kind manufacturing director and Suspension-Steering-Brakes subteam director for MBR34. Her current interests involve robotic prostheses and exoskeletons. After graduating, Simran will be pursuing her Masters' Degree.


**Benji Wu**
*B.S.E Mechanical Engineering Winter '25*

Benji is a senior in mechanical engineering with a concentration in robotics and a minor in computer science. He grew up in Marin County, California and established his high school engineering club. Benji has always loved taking things apart and working on home improvement projects with his father–a large factor in his choice to become a mechanical engineer. On the Baja Racing team, Benji was the front frame lead for the 2022-2023. For the 2023-2024 season, Benji was the front gearbox lead and lead welder for the team. Benji's work on the front gearbox produced a novel locking differential design and he is currently seeking patent protection for his gearbox design. His current interests include wave energy harvesters and optimized path planning. After graduation, Benji will be pursuing his masters degree in mechanical engineering.

**David Grover**
*B.S.E Mechanical Engineering Winter '25*

David is a senior majoring in Mechanical Engineering with a concentration in Robotics. David grew up in Cincinnati, Ohio, which he asserts is the only good part of Ohio. As a young lad, he played soccer for his high school and on a club team, eventually finding a love for other sports including volleyball and rock climbing. In high school, he and his brother took on restoring a 2005 Audi A4 that had a blown head gasket (or so they thought). This led them down a 2-year-long rabbit hole of problem analysis and greasy hands, culminating in David getting to drive the car to and from school during his senior year of high school (check engine light flickering on and off). In college, David fell in love with the Baja team dynamic and competitive spirit, becoming Chassis/Ergonomics Director as a sophomore, Gearbox Lead as a Junior, and Team Captain his senior year. He will graduate in April 2025, and will work at SpaceX as a Launch Engineer full time after graduation.

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**Pablo Elizondo Del Bosque**
*B.S.E Mechanical Engineering Winter '25*

Pablo is a senior in mechanical engineering and a minor in mathematics. He grew up in Laredo, Texas which is right on the border between the US and Mexico. As a kid, Pablo always took an interest in mathematics and science. As time went on, Pablo decided to go into mechanical engineering due to its heavy relationship with mathematics while also being able to work with his hands which was an added benefit. In Pablo's freshman year of college he decided to join Michigan Baja Racing as it provided him with the opportunity to apply what he learned in real life. Pablo was the CVT lead his sophomore and junior year, Drivetrain Director his junior year, and is currently this year's Technical Director. Pablo is also one of the team's CNC machinists in which he contributes to the various machining of the car parts. He will graduate in April 2025 and plans to work in a Research and Development role post-graduation.