SAE Baja – Drive Train

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Team 11

Project Proposal

Document

Submitted towards partial fulfillment of the requirements for

Mechanical Engineering Design I – Fall 2014

Department of Mechanical Engineering

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1.0 Introduction

The Northern Arizona University Chapter of the Society of Automotive Engineers (SAE) has instructed our team to design and build a vehicle for the Baja SAE Series. This entails designing a single seat off-road vehicle capable of performing in the top 10 in the collegiate competition at Portland, Oregon from May 27-30, 2015. This competition is a challenge for colleges to design an off-road vehicle capable of exceptional performance and customer appeal. The performance aspect of the vehicle is measured from the accomplishments of the vehicle in 5 dynamic events: the Acceleration, Hill Climb, Maneuverability, Rock Crawl, and Endurance challenges.

The Drive Train Team is responsible for the design of the engine through to the wheels. This will include the engine, transmission, differential, and any power transmitting shafts. The engine is a constraint in our design, as SAE requires the use of a Briggs & Stratton Model 20 engine. This specific model of engine proposes a challenge due to its maximum power output being 10 horsepower. Our particular engine is only 8.5 horsepower, however, which was discovered through the use of a dynamometer.

This year's Drive Train Team has set a goal of placing in the top ten in two specific events: Acceleration, and Hill Climb. These events were chosen because the overall performance in these events depend greatly on the design and execution of the transmission design. The set performance goals to achieve this objective are to complete a 100 foot distance in 4 seconds from a complete stop, and for the vehicle to be able to drive up a hill of about 60 degrees. The contents below describe how the Drive Train Team chose between six transmission concepts, to then analyze the sequential transmission, to implement into the NAU Baja for the May 2015 competition.

2.0 Customer Needs

Dr. John Tester is the current customer and advisor that the Mini Baja transmission team must consult before making any design or manufacturing decisions.

Dr. Tester expressed a need for an improved drive train this year, stating that last year's drive train was too heavy and must be more lightweight in order to improve vehicle performance. Also last year's reverse gear could have been better designed and engaged in a cleaner, more effective manner, so this year's reverse must be better designed and more integrated within this year's gearbox. Finally, the drive train needs to be safe and more reliable than last year's, as a broken engine/drive train mount prevented the team from achieving a better scoring position in competition.

2.1 Project Goal

In last year's competition the NAU Mini Baja placed 58th in the acceleration test and 64th in the hill climb test. The goal the team has for this project is to design a drive train that will be competitive and place in the top 10 for the acceleration and hill climb tests against other competing universities.

2.2 Objectives

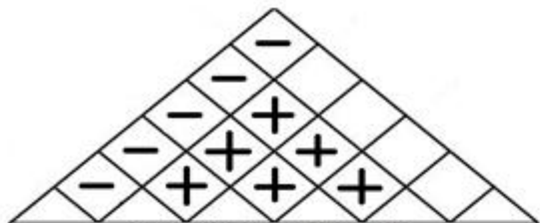
- The Drive Train Team has chosen six objectives for the 2014-2015 Mini Baja project: size, weight, cost, the acceleration and hill climb tests, and safety. For size, the team defined it as the volume the gear box occupies, in inches cubed (in^3). Weight was decided to mean the weight of the entire gearbox, in units of pounds. Cost is the cost to produce the drive train, in dollars. The acceleration test was decided to be the number of seconds it takes for the Mini Baja to travel 100 feet. The hill climb challenge is how far the Mini Baja can make it up an incline, in units of feet. Safety is defined to be a factor of safety of at least 2 in every component of the drive train.

2.3 Constraints

For this year's Baja transmission there were a few constraints the team had to consider before proceeding with a QFD and HOF evaluation.

- Use provided/donated Briggs and Stratton 10hp OHV model 20 engine. The team must use the provided engine and cannot alter the engine in any way to allow for a performance advantage against other competing teams.
- The drive train must be designed within the SAE Baja rules. The only rules that pertain to the drive train portion of the Mini Baja include covering any moving components.
- The Baja must complete a 0-100 foot acceleration trial in 4 seconds on dry level pavement. The team has found that top placing competing schools have completed the 0-100 foot acceleration challenge in around 4 to 4.5 seconds. Since the acceleration challenge relies so heavily on gear ratios and weight, it will be the teams responsibility to choose the right ratios in order to achieve an acceleration time of 4 seconds.
- The Baja must be able to climb an incline of 60 degrees. From research of recent competitions the greatest incline at a given competition was around 60 degrees. The team must choose gear ratios that will allow the Baja to complete the hill climb, as well as complete the hill climb in a competitive time against other Universities.
- The team needs to design components for the drive train that will be practical and have the ability to be machined, preferably in the NAU Machine Shop. Though the team is not limited to only producing parts in the Machine Shop, it is ideal because it will cost less money and allow the team to spend its resources in other needed areas.

3.0 QFD/HOQ



Engineering Requirements

High -9
Medium - 5
Low - 1
None - 0

Customer Needs	Weighted %	Cost	Weight	Volume	Acceleration	Hill Climb Angle	Safety	NAU Baja 2013-14
		↓	↓	↓	↑	↑	↑	
Lightweight	30%	9	9	9	9	9		
Reverse Gear	20%	5	9	9				X
Reliable	30%	9	9	1			9	X
Manufacturing Limitations	10%	9	9	9	9	9		X
Inexpensive	10%	9	9	5	9	9	9	
	Units	$	lbs	in^3	sec	Degrees	F.O.S.	
	Score	8.2	9	6.2	4.5	4.5	3.6	
	Weighted Percentages	22.78%	25.00%	17.22%	12.50%	12.50%	10.00%	
	Targets	2500	100	1000	4	65	>2	
	NAU Baja 2013-14	3000	160	1809.5	5.879	40	10	

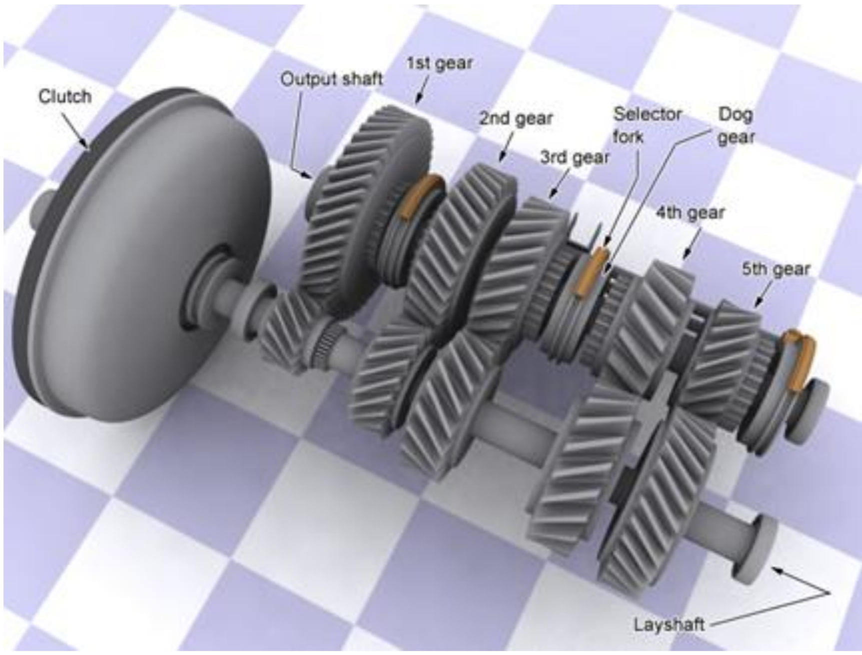
(Figure 3.1, QFD/HOQ)

As you can see from Figure 3.1, we have listed out customer requirements and out engineering requirements with their weighting. We have determined that our largest impacts on desired outcomes depends on our cost and weight. This makes sense since we are trying to cut weight this year and money is always a challenge. As you can also see from our HOQ that these two parameters are negatively correlated. This will be a challenge to overcome, but due to the ability to raise more money it shouldn't hinder our final product too harshly.

4.0 Design Concepts

4.1 Concept 1: Manual

The manual transmission is the team's first concept generated for the drive train portion of the baja. As seen below in Figure 4.1, the manual transmission works by connecting multiple sets of gears, each of which has a different gear ratio.



(Figure 4.1, Manual Gearbox)

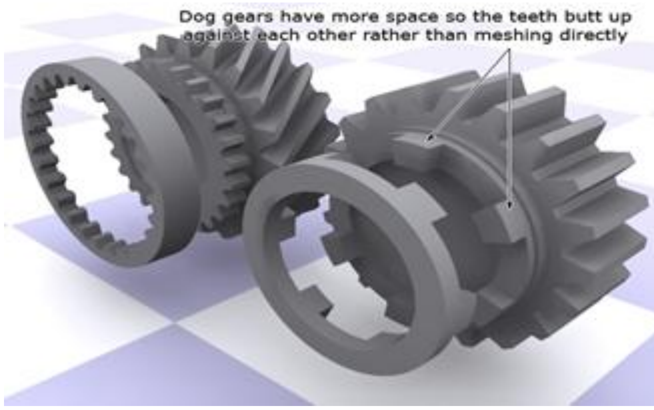
The gear sets are driven by an input shaft that is connected to a clutch. The clutch acts as a means of transmitting power by engaging a spring or hydraulic lever that will connect or disengage the input shaft to the set of gears. In order to shift from one ratio to the next, a shift fork is used to choose each gear. The shift fork is connected to a lever that the operator may use to select any gear he/she wishes. The gears are engaged by the use of a dog ring that is connected to one of the shift forks. In a manual transmission, the dog ring and gear must match up in a 1:1 ratio in order to shift into each gear. In order to do this the clutch must be engaged each time the operator wants to shift into another gear.

With the manual transmission comes a mix of both pros and cons. An advantage of the manual transmission is that it is reverse capable. Our customer, Dr. John Tester, required that we have a reverse gear integrated into our drive train. Because almost every manual gearbox is designed with a reverse gear, it will be very easy for the team to take an existing manual gearbox with reverse-capabilities, or to design a manual gearbox that can have a reverse gear integrated into it. Also, manual gearboxes have been tested for many years now and are a reliable means of transmitting power with multiple gear ratios. It is also a fairly cost effective design because most of the parts, such as gears and bearings, for this transmission are mass produced.

There are however a few disadvantages of running a manual transmission. First, they are in nature a bit heavier because of the need for a clutch and reverse gear. This also makes the gearbox larger because of the needed clutch and reverse shaft. A manual transmission also causes a loss of power that will occur in between each shift. When engaging and disengaging each gear ratio, the clutch must be used. Between each shift there is a brief period where there is no power being transmitted to the wheels. This can negatively impact the teams overall performance in both the Hill Climb and Acceleration tests, since there will be wasted time engaging and disengaging the clutch between each shift.

4.2 Concept 2: Sequential

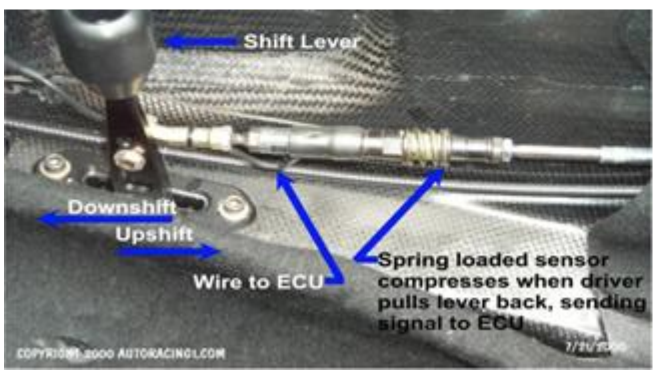
The next concept the team considered for a possible design solution is a sequential transmission. The sequential transmission is a derivative of the manual transmission with slight differences in the shifting mechanism. As seen in Figure 4.2, there is a difference in the geometry of the dog rings and the way which the gears and dog rings engage.



(Figure 4.2, Sequential (front) Vs. Manual (back) dog ring)

Because the sequential dog ring has a square geometry and more room to engage to the gear, it allows the gear and ring to engage at different speeds, as opposed to a manual gearbox which requires both the gear and ring to be spinning at the same 1:1 ratio in order to engage the two. This means the clutch does not have to be engaged each time the operator wants to shift from one gear ratio to the next.

There is also a different manner in which the sequential transmission selects each gear ratio. The shifting mechanism and selector work by only allowing the operator to shift either up or down a single step in gear ratio. If the operator desires to select 4th gear from 2nd gear, he/she must engage 3rd gear from 2nd gear, and then 4th gear from 3rd in sequence. The gear selection can be achieved by use of a shift lever as seen in Figure 4.3.



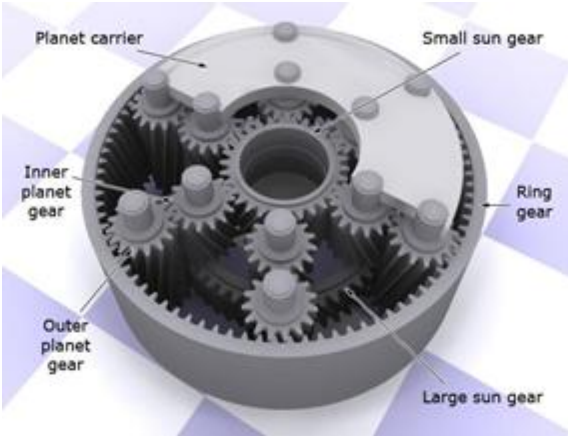
(Figure 4.3, Sequential Shift Lever)

There are many advantages when using a sequential transmission. First there is little loss of power because the clutch does not need to be engaged in between each shift. This means there will be a minimal amount of time that there is no power being transmitted to the wheels, which, in effect, means an increase in performance when accelerating. Also, sequential transmissions are generally smaller and more compact, which also means that the gearbox will be lighter as well. The sequential transmission is also easy to operate because the clutch needs to be used only when starting from a stop. Each gear shift after is completed by pulling or pushing a lever which will engage the gear ratio below or above the current gear. Finally, the sequential transmission is more reliable and stronger than a standard manual transmission. A countershaft is generally used to transmit power to the gear ratios, which means that the gears in a sequential transmission will experience about half the force of a normal manual transmission.

There are a few disadvantages of using a sequential transmission as well. Most sequential transmissions do not have a reverse integrated into the design of the gearbox, as they are generally used on motorcycles and off-road applications where reverse is not needed. It will be a difficult task to integrate a reverse into a pre-existing design. Also, if the team decides to integrate a reverse into an existing transmission, it will be an added expense to the production.

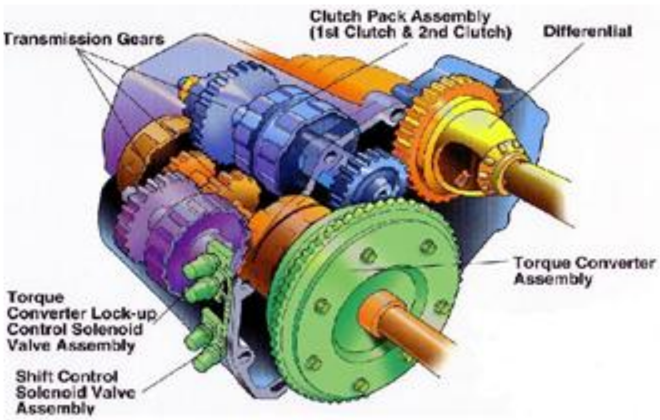
4.3 Concept 3: Automatic

The third concept the Drive Train Team came up with was to design an automatic transmission. The automatic transmission has great attributes - it will shift gears on its own via the compound planetary gear system shown in Figure 4.4, which is connected to a torque converter by the small sun gear. The planet carrier then turns the ring gear by trying to spin the opposite direction of the torque converter, but due to a one-directional clutch system, the ring gear instead turns and becomes the output power. The compound planetary gear set allows for four forward gear ratios, and one reverse.



(Figure 4.4, Compound Planetary Gear System)

The automatic transmission has many advantages: 1) there is no power loss in shifting gears, 2) it has a high gear ratio range to allow for various speeds and terrains, 3) it is very reliable, and 4) the transmission has a reverse gear capability. However, there are many disadvantages for this type of transmission: the cost tends to be higher than most other kinds of transmissions, since it can shift gears by itself, and thus has more components, also resulting in a larger size. An example of an automatic transmission is shown in Figure 4.5.



(Figure 4.5, Automatic Transmission)

4.4 Concept 4: Direct Drive

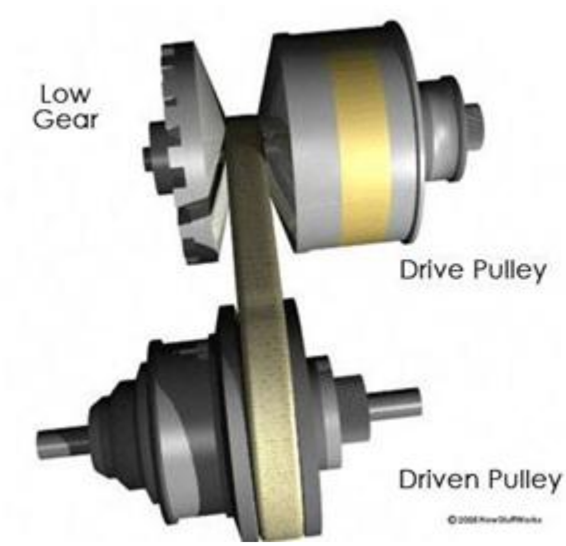
The fourth concept the Drive Train Team came up with was a direct drive transmission, which is essentially a manual transmission with just one gear on the input shaft, and another on the output shaft, i.e. a single gear ratio, which is shown in Fig. 4.6.

(Figure 4.6, Direct Drive)

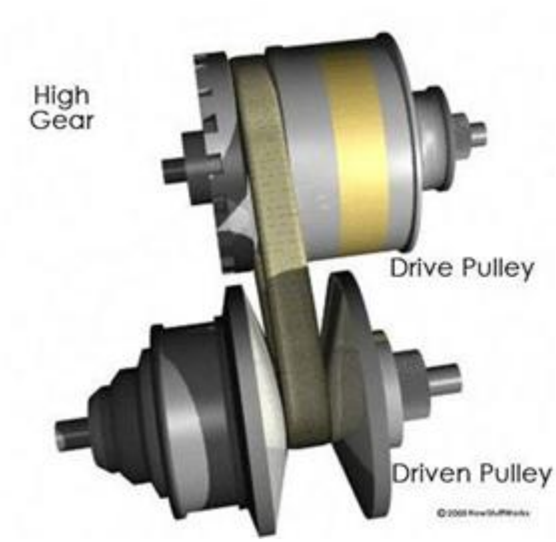
This system has many advantages: the cost is extremely low, the setup is very simple, small, and light, as the transmission only requires two gears; the one-to-one ratio also calls for a highly efficient system, as there aren't many losses due to friction if the gearbox is designed correctly. However, for the design of the Baja, a Direct Drive Transmission will not be effective in events (such as the Hill Climb due to a static gear ratio), and will lose points in the design category, as the system does not allow for a reverse gear.

4.5 Concept 5: CVT Belt

The CVT (Continuously Variable Transmission) is the next concept being considered. The first type of CVT being considered is belt driven. A CVT has a desirable attribute of being able to change its overall gear ration without the need of shifting gears, thus saving time in acceleration; CVTs are also relatively light for the gear ratio ranges they offer. This is due to the capability of a CVT to change its pulleys' diameters via rotational force. A simple way of looking at a CVT is that it is a pulley system where the pulleys can change size to optimize the output power as shown in Figures 4.7 and 4.8 below. The CVT changes the pulleys' diameters by linearly shifting allowing the belt to slip down into the larger gap causing the relative pulley size to shrink, as in Figure 4.7 with the drive pulley, or grow if the gap has shrunk, as in Figure 4.8.



(Figure 4.7, CVT Belt)



(Figure 4.8, CVT Belt)

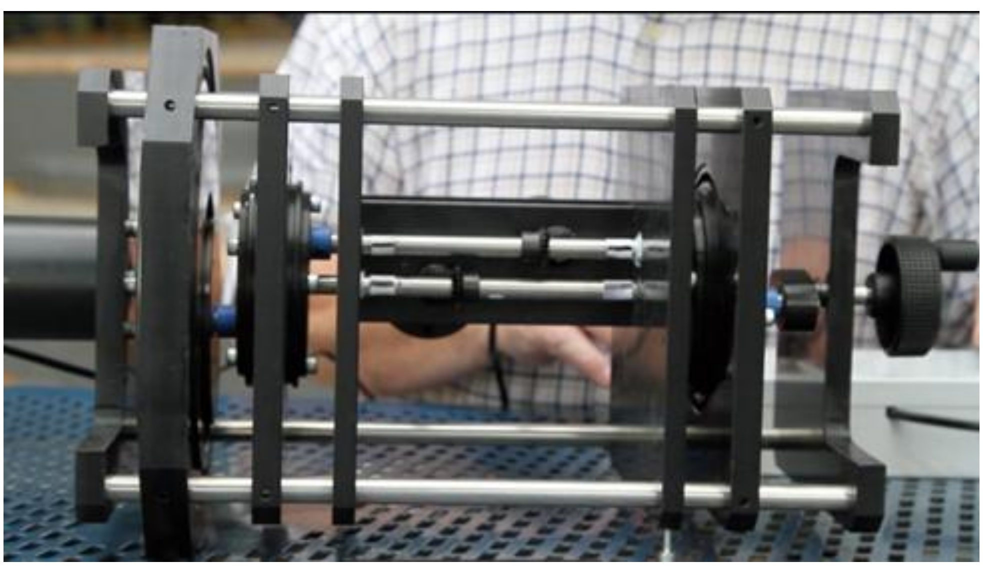
This is all great reason to use a CVT for fast acceleration and easy optimization of engine power. The disadvantages of a CVT though are also high in importance when considering overall performance in an off road vehicle. These include being limited in max power a CVT can handle due to the fact that the system is dependent on the belt's friction on the pulley to transmit power and this fact also results in relatively low efficiency compared to other designs.

4.6 Concept 6: CVT Gear

For these reasons the final consideration is a gear type CVT. The benefit of this design is that is keeps the changing gear ration of a normal CVT without the need of relying on the friction of a belt to transmit the power. This in conjunction with all the other benefits of a belt type CVT is very desirable for overall performance.

A CVT Gear setup works by transmitting power trough a double planetary gear system. The sun gear in both gear sets are connected to each other by a single shaft, ratio shaft, that is rotated by an external electric motor. This external motor actually sets the gear ratio by adjusting how fast the planet gear has to spin in order to keep up with how fast the ring gear is spinning. In the first gear set the planet gear is the input from the drive engine, and the ring gear is attached to a shaft that also attaches to the second ring gear. In the second gear set the planet gear is the output that goes to the drive shafts to the wheels. The technology for this variable capability is accomplished by rotating the ratio shaft at different speeds from the drive shaft to allow the gear sets to force each other to change how fast and which direction the output shaft is turning. As implied the CVT Gear setup does have the capability of reverse without additional weight.

The disadvantage of a gear type CVT is that is a very new technology. This proposes the issue of figuring out if we have the time and facilities to develop this type of CVT. Another disadvantage is as you can see in Figure 4.9 that it is relatively large and complicated with the number of moving parts and the number of gears.



(Figure 4.9, CVT Gear)

With these considerations, if we are capable of getting assistance from the inventor, Stephen Durnin, we may be able to design and manufacture this type of transmission for our specific application.

5.0 Concept Selection

Using Dr. John Tester's requirements, as well as other necessary criteria, the Drive Train Team came up with eight different criteria in which to rank each of the six ideas, as well as weights for each criterion, shown the decision matrix in Table 5.1. The criteria generated (cost, gear ratio range, efficiency, weight, simplicity of design, reliability, size, and capable of a reverse gear) were then ranked for type of transmission on a scale of one to five, five being the best for the vehicle, and one being the worst. Totals were then calculated by summing the products of the rank and weight of each criterion for each transmission. This led to two top choices: the Sequential and CVT Gear transmissions, with a manual transmission in third place. The team will now be analyzing all three gearboxes, as instructed by Dr. Srinivas Kosaraju, because of the complexity of the CVT Gear box. From our calculations we will then decide which concept to implement into the Baja vehicle.

(Decision Matrix, Table 5.1)

Scale 1-5 5 = Best, 1 = Worst	Cost	Gear Ratio Range	Efficiency (Loss of Power)	Weight	Simplicity of Design	Reliability	Size/Volume	Reverse Gear Capable	Total
Sequential	3	5	5	4	3	4	4	3	3.95
Manual	3	5	4	3	4	4	3	4	3.85
CVT Belt	2	3	2	3	5	2	5	1	2.35
CVT Gear	2	5	4	3	3	4	3	5	3.85
Automatic	2	4	3	3	2	4	2	4	3.2
Straight (One Gear Ratio)	5	2	5	5	5	5	5	1	3.75
Customer Weighting	15%	15%	20%	10%	5%	10%	5%	20%	

6.0 Final Design Selection and Analysis

This section will introduce the final decision matrix of the drive train, the analysis of the Hill Climb and Acceleration Tests, and into detail of the of the team's final assembly.

6.1 Selection

This is the Decision Matrix the Drive Train Team came up with in order to determine which of the transmissions, manual or sequential, to begin analyzing and testing:

(Decision Matrix, Table 6.1)

Scale 1-5 5 = Best 1 = Worst	Cost	Gear Ratio Range	Efficiency (Loss of Power)	Weight	Simplicity of Design	Reliability	Size/Volume	Reverse Gear Capable	Total
Sequential	3	5	5	4	3	4	4	3	3.95
Manual	3	5	4	3	4	4	3	4	3.85
Customer Weighting	15%	15%	20%	10%	5%	10%	5%	20%	

From the data that the Decision Matrix provides, it becomes clear that the team should pursue evaluation of the sequential transmission, as it will be more efficient, lighter, and overall smaller than a manual gearbox.

6.2 Analysis of Sequential Transmission

6.2.1 Hill Climb

To begin the engineering analysis for the Hill Climb Challenge, the team used Google, the transmissions text, and equations stated in texts from previous classes in order to find the static, friction, and drag forces on the Baja. Any equations not shown in this section have been stated in the References/Equations section at the end of this report. Based on last year's Hill Climb in El Paso, Texas, last year's average wind speed in May in Portland, Oregon, as well as the altitude of Portland, the team assumed a hill angle of 60 degrees, a coefficient of drag of 0.62, a wind speed of five miles per hour, and an air density of 0.00228 slugs per cubic foot. Then, the team assumed a coefficient of friction of 0.16, based off a table in the transmissions textbook for driving over dirt, the maximum surface area of the Baja to be 9.98 square feet, the power of the supplied Briggs & Stratton engine to be 8.5 horsepower, or 4675 pound-feet per second, and the weight of the Baja to be six hundred pounds.

With these assumptions in mind, the team generated that the static force pulling down on the Baja to be 519.615 pounds, the friction force to be 48 pounds, and the drag force to be 0.379 pounds, leading to a total resisting force of 567.994 pounds. To calculate the velocity of the Baja travelling up the hill, we divided the power of the engine from the total resisting force, giving us a velocity of 5.616 miles per hour. From this, Team Drive Train decided to be conservative, and assume an overall resisting force of 600 pounds and a vehicle velocity of 6 miles per hour. Assuming this 600 pound force resisting the vehicle's upward momentum, an upward angle of 60 degrees, and the engine's production of 8.5 horsepower, the team can achieve a velocity of 6 miles per hour up the hill, as stated previously. The maximum torque the engine can output is 14.5 foot-pounds in the range of 1800-2800 rpms.

With this in mind, the team used a gear ratio formula in order to find the optimum ratio that would keep the engine between 1800-2800 rpm at a 60 degree incline. First the angular velocity of the wheels were found to know exactly how fast the engine speed is, in relation to how fast the wheels would need to be turning using this formula:

$$\omega = \frac{v_{Baja}}{R_{Wheel}} = 91.67\ rpm$$

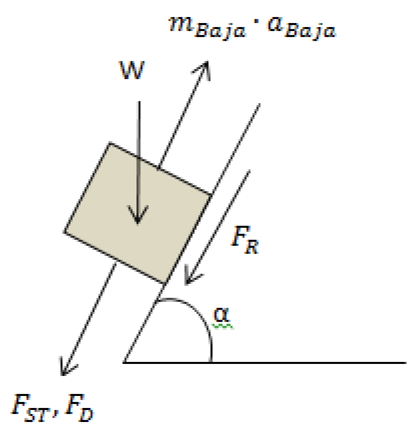
Relating the angular momentum to the engine speed to find the gear ratio needed using these formulas:

$$Gear\ Ratio_{min} = \frac{N_{min}}{\omega} = 19.63{:}\,1$$

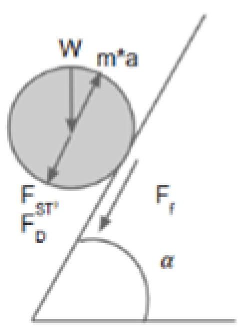
$$Gear\ Ratio_{max} = \frac{N_{max}}{\omega} = 30.54{:}\,1$$

$$Gear\ Ratio_{avg} = \frac{N_{max} - N_{min}}{2\omega} = 25.089{:}\,1$$

We found that the minimum gear ratio that is needed at 1800 rpm is 19.63:1, and the ratio needed at 2800 rpm is around 30.54:1. The team decided to take the average of these two ratios at 25.089:1 so that the engine would stay in the range of 1800-2800rpm.



(Figure 6.1, Hill Climb Free Body Diagram)



(Figure 6.2, Hill Climb Free Body Diagram of Wheel)

6.2.2 Acceleration

The Drive Train Team has set the goal of achieving a 100 foot distance in 4 seconds from a dead stop on pavement. To start our calculations, the team used the basic dynamics formula to find the required constant acceleration to achieve the desired goal:

$$x = 100\ feet$$

$$t = 4\ seconds$$

$$Initial\ Velocity = \ v_0 = 0\ \frac{ft}{s}$$

$$x = v_0 \cdot x \cdot t$$

$$a = \frac{2x}{t^2} = 12.5\ \frac{ft}{s^2}$$

$$v_{final} = a \cdot t^2 = 33.733\ \frac{ft}{s}$$

Then, using the forces of resistance on the vehicle (drag force, rolling resistance, etc.), the team calculated what the total resistance force on the vehicle would be for the Low Gear when $v = 0\ \frac{ft}{s}$, and the High Gear when $v = 33.733\ \frac{ft}{s}$:

$$m = 18.65\ lbm$$

$$c_A = 9.92\ ft^2$$

$$c_D = 0.62$$

$$F_{accel} = m \cdot a$$

$$F_{roll} = f_R \cdot m \cdot g$$

$$F_{drag} = \frac{1}{2} \cdot \rho_{air} \cdot c_D \cdot A \cdot v^2$$

$$F_{total,Low\ Gear} = 241\ lbf$$

$$F_{total,High\ Gear} = 250\ lbf$$

Using this information and the engine torques at idle and top angular velocities (1800 rpm and 2800 rpm, respectively), the team then calculated what the needed overall gear ratios were for the Low and High Gears for the acceleration challenge:

$$R_{tire} = 11\ in = 0.9166\ ft$$

$$\tau_{Low} = 10\ ft \cdot lbf$$

$$\tau_{High} = 13\ ft \cdot lbf$$

$$Ratio = \frac{\frac{F_{total}}{2} \cdot R_{tire}}{\tau}$$

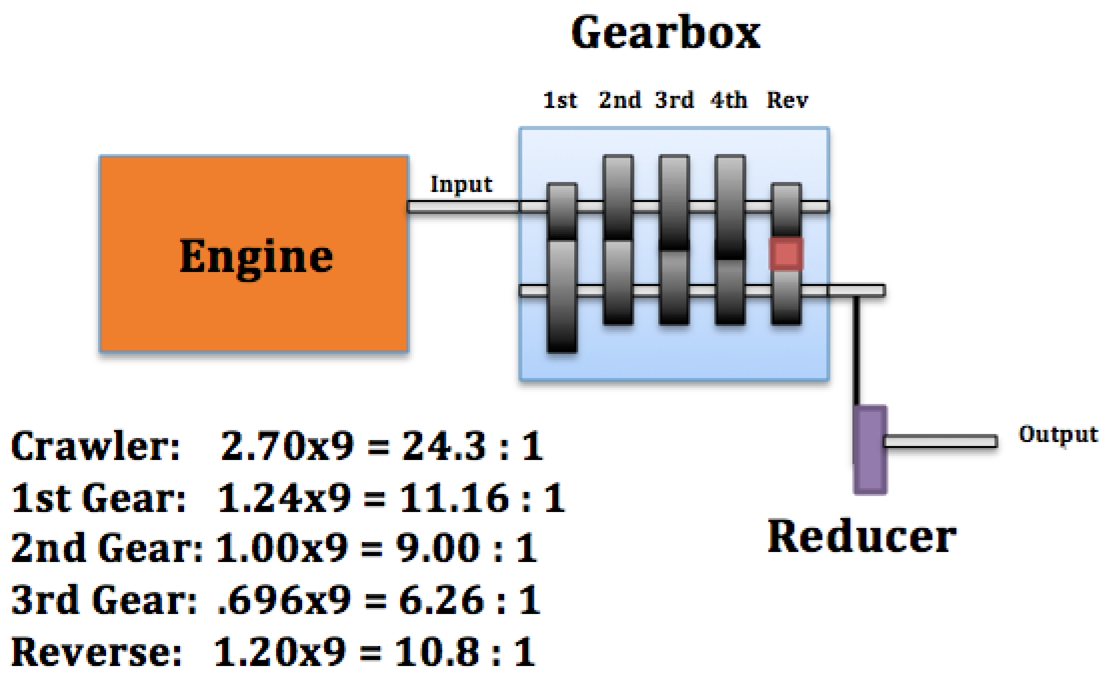
From this, we found the Low Ratio needed to be 11.05:1 and the High Ratio to be 8.8:1. From the power curve calculations we calculated that the middle gear should be around 9:1.

6.2.3 Gear Ratio Selection

After all the calculations, all the ratios will be as follows:

Engine to gearbox ratio:	1:1
Crawler:	2.70:1
First gear:	1.24:1
Second gear:	1:1
Third gear:	0.696:1
Reverse gear:	1.20:1
Reducer ratio:	9:1

Figure 6.3 shows a basic concept representation of our drive train layout:



(Figure 6.3, Gear Selection)

6.2.4 Gear Specifications

After more calculations, Team Drive Train has decided to go with 7075-T6 aluminum for the gear material, and found the following tooth sizes for each gear:

Crawler:	Pinion: 15	Gear: 41	
Fist Gear:	Pinion: 25	Gear: 31	
Second Gear:	Pinion: 28	Gear: 28	
Third Gear:	Pinion: 33	Gear: 23	
Reverse:	Pinion: 15	Gear: 15	Gear 2: 18
Reducer:	Pinion: 10	Gear: 90	

The minimum factor of safety out of all the gears came out to be 6.2.

6.2.5 Shafts

Using the DE-Goodman equation and the calculated torque and moment values (shown below), the Drive Train Team decided upon 4340 Normalized Steel as the shaft material with a diameter of 0.5 inches:

$$K_f = K_{fs} = 1$$

$$\frac{1}{n} = \frac{16}{\pi d^3}\{\frac{1}{S_e}\left[4\left(K_f M_a\right)^2 + 3\left(K_{fs} T_a\right)^2\right]^{\frac{1}{2}} + \frac{1}{S_{ut}}\left[4\left(K_f M_m\right)^2 + 3\left(K_{fs} T_m\right)^2\right]^{\frac{1}{2}}\}$$

As there is no midrange moment or alternating torque in either the input or output shafts, the alternating moment and midrange torque values become:

Input Shaft:

$$M_a = 578\ in \cdot lbf$$

$$M_m = 0\ in \cdot lbf$$

$$T_a = 0\ in \cdot lbf$$

$$T_m = 192\ in \cdot lbf$$

Output Shaft:

$$M_a = 578\ in \cdot lbf$$

$$M_m = 0\ in \cdot lbf$$

$$T_a = 0\ in \cdot lbf$$

$$T_m = 528\ in \cdot lbf$$

From all this, the factors of safety come out to be 2.94 and 2.00 for the input and output shafts, respectively.

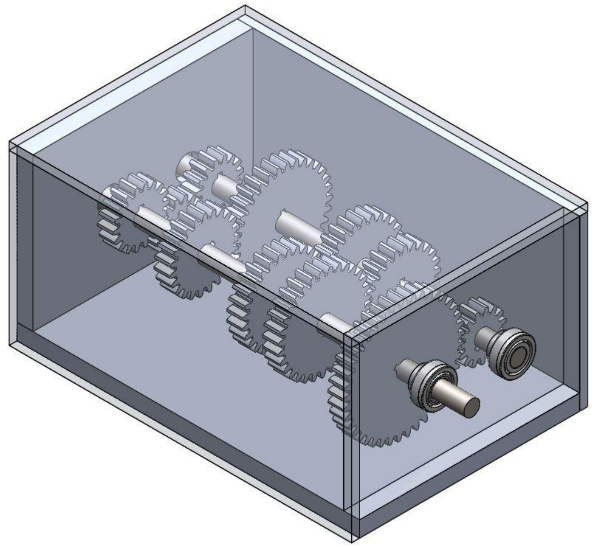
6.2.6 Bearings

After doing more calculations, Team Drive Train has decided on purchasing Open Steel Ball Bearing from the McMaster-Carr website. The specifications work for a shaft diameter of 0.5 inches, and the bearings have an outside diameter of 1.125 inches, a width of 0.375 inches, and a dynamic load capacity of 600 pounds. This leads to a factor of safety of 2.3 for our purposes. Figure 6.4 shows an example of a bearing being ordered:

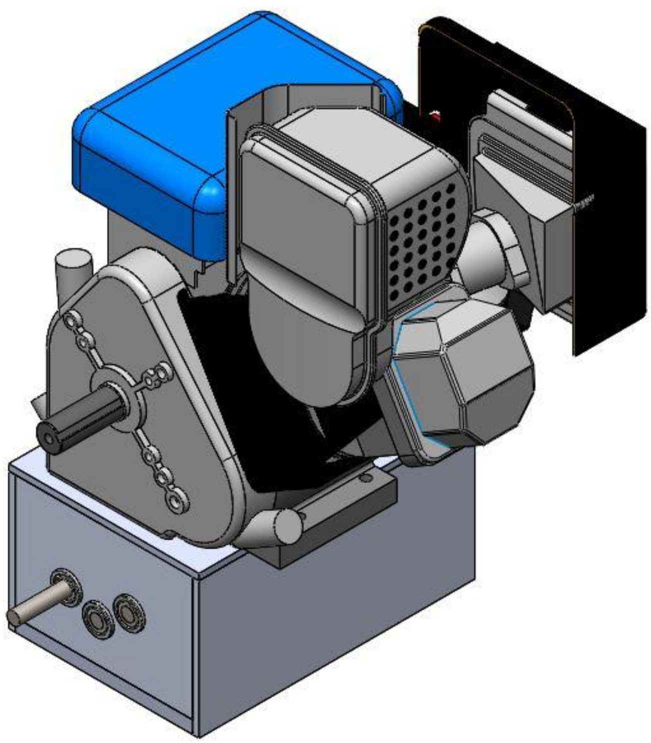
(Figure 6.4, Open Steel Ball Bearing)

6.2.7 Final Assembly

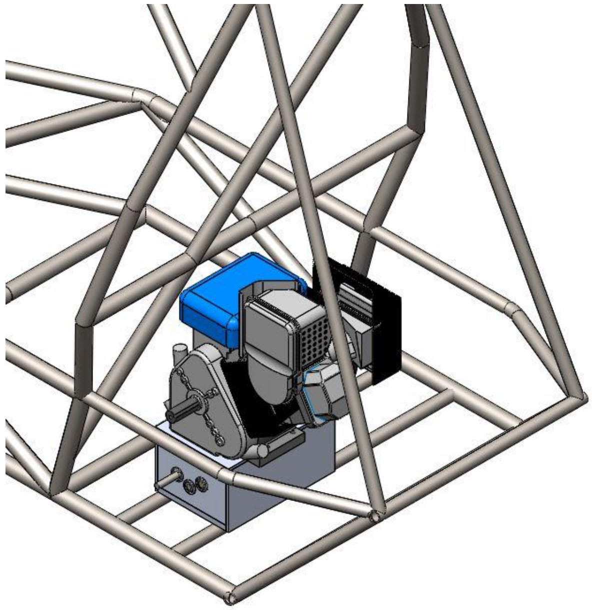
In Figures 6.4 and 6.5 are preliminary CAD Assemblies of the drive train transmission and engine combination. It is currently sitting at 11 in x 7.5 in x 10.5 in and weighs 15.7 pounds. This design is very optimal, as it is compact, lightweight, and efficient. 6.6 and 6.7 portray where the gearbox will sit on the most current frame design.



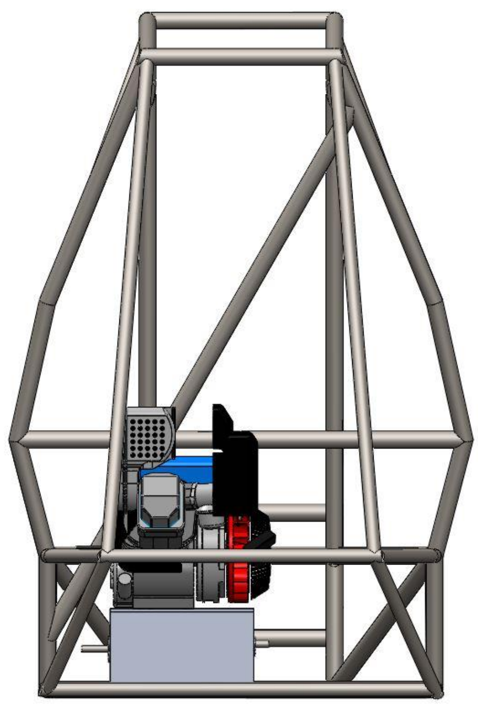
(Figure 6.5, Isometric View of Gearbox)



(Figure 6.6, Gearbox Assembly)



(Figure 6.7, Isometric View of Gearbox on Frame)



(Figure 6.8, Back View of Gearbox on Frame)

7.0 Bill of Materials

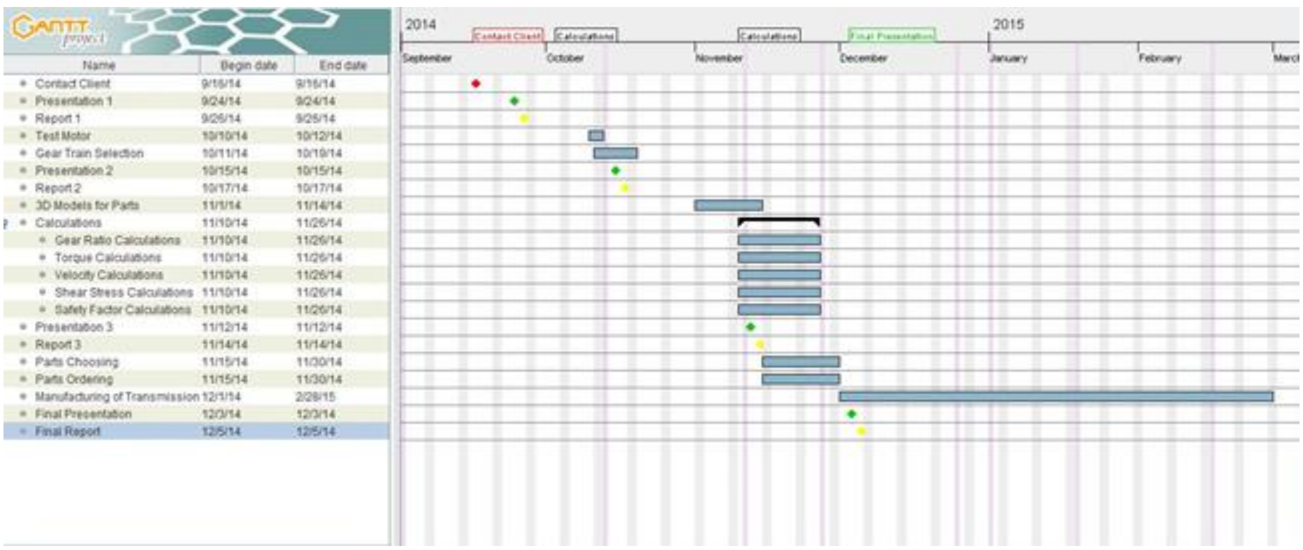
Shown below is a Bill of Materials created for the current design of the overall gearbox. If our team were to have nothing donated/on site, the total cost would be $1905.60 – however, due to donations from companies, such as Industrial Metal Co. (IMS), as well as the differential from last year's Baja, the total cost so far for the entire gearbox is $379.16.

(Table 7.1, Bill of Materials)

Materials	Quantity	Cost for One Unit of Material	Overall Cost of Each Material	Free/Donated
7075 T6 Aluminum (4" diameter, 2' bars)	1	$307.44	$307.44	x
7075 T6 Aluminum (3" diameter, 5' bars)	1	$298.87	$298.87	x
7075 T6 Aluminum (2" diameter, 4' bars)	1	$87.24	$87.24	x
6061 T6 Aluminum (0.5" thick, 1'x3' plates)	1	$164.92	$164.92	x
6061 T6 Aluminum (0.25" thick, 1'x3' plates)	1	$76.69	$76.69	x
4340 Normalized Steel (5/8" inch diameter, 5' bar)	2	$95.64	$191.28	x
Bearings	6	$7.36	$44.16	
Clutch	1	$300.00	$300.00	
Differential	1	$400.00	$400.00	x
80 tooth sprocket	1	$25.00	$25.00	
10 tooth sprocket	1	$10.00	$10.00	
		Total	$1,905.60	
		Total, subtracting free/donated	$379.16	

8.0 Project Plan

The Drive Train Team has been progressing through the project's plan, as displayed in the team's Gantt Chart, shown in Figure 8.1. However, the "Calculations" and "Parts Choosing" sections were pushed back a couple of weeks, due to illnesses throughout the team and because two of the Drive Train Team members were away between November 4-6, 2014 at SEMA, an automotive convention, attempting to find sponsors for the entire Baja team. Luckily, a couple companies seemed interested in sponsoring the group, which will immensely help the team.



(Gantt Chart, Figure 8.1)

9.0 Conclusion

In conclusion, the Drive Train Team chose to use and analyze the design of the sequential transmission for the Baja vehicle, due to its superiority over the manual transmission, for the Baja Team's purposes. After selecting which transmission to implement into the vehicle, the team calculated the forces against the vehicle in the Hill Climb Challenge. Using this data, the Drive Train Team then calculated the gear ratios needed to climb up the hill in a reasonable amount of time. From there, the team assumed moving 100 feet in around 4 seconds for the Acceleration Test, letting the vehicle's top speed hit around 35 miles per hour; gear ratios again were calculated to take these values into account. Afterwards, Team Drive Train compared the gear ratios needed for the Hill Climb and Acceleration Tests, respectfully, which led to the final gearbox assembly.

10.0 References/Equations

10.1 References

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5. Mating Pinion and Gear Picture http://www.daerospace.com/MechanicalSystems/GearDescFig2.png
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10.2 Equations

1. $W = 600lbf$
2. $f_R = 0.16$
3. $c_D = 0.62$
4. $P = 8.5\ hp = 4675\ \frac{ft \cdot lbf}{s}$
5. $\alpha = 60°$
6. $A = 9.98\ ft^2$
7. $\rho_{air} = 0.002278233594351\ \frac{slug}{ft^3}$
8. V$v_{wind} = 5pmh = \frac{22}{3}\frac{ft}{s}$
9. $F_{St} = W \cdot \sin(\alpha) = 519.615\ lbf$
10. $F_R = f_R \cdot W \cdot \cos(\alpha) = 48\ lbf$
11. $F_D = \frac{1}{2} \cdot \rho_{air} \cdot c_D \cdot A \cdot {v_{wind}}^2 = 0.379\ lbf$
12. $F_{total} = F_{St} + F_R + F_D = 567.994 \sim 600\ lbf$
13. $v_{Baja} = \frac{P}{F_{total}} = 8.236\ \frac{ft}{s} = 5.616\ mph$
14. $R_{Wheel} = 11\ in = 0.916\ ft$
15. $N_{min} = 1800\ rpm$
16. $N_{max} = 2800\ rpm$
17. $T_{Wheel} = R_{Wheel} \cdot F_{total} = 550\ ft \cdot lbf$