REPORT
ME 329-04 Lab

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Date:	10/17/17	
RE:	**Critical Design Review**	

Introduction

The objective of this quarter long project is to convert a small combustion powered car into an electric vehicle with a cohesive system of a battery, motor, motor controller and transmission. While design consideration of each component is necessary for a functional vehicle, our team's primary focus is to design the two-speed transmission to optimize the torque provided from the motor into efficient motion. Using our knowledge of gears, shafts bearings and other mechanical design elements, we will choose the optimal design in electrifying this vehicle. This report outlines the full process of our design, from background research into iteration on a system level.

Background Research

Battery

The battery is a component that plays a major role determining both the power supplied to the motor for acceleration and the overall range available to the car. MIT's Electric Vehicle Team laid out "A Guide to Understanding Battery Specifications" that outlines key elements within a battery [1]. The nominal voltage of the battery is what is typically referred to when mentioning battery voltages. The voltage picked for a battery will rely heavily on the motor specifications for the car and the required acceleration. Capacity is another key factor to consider when selecting a battery. Measured in kilowatt-hours, the capacity is the amount of charge that a battery contains and is directly related to the car's range. A higher capacity will yield a longer discharge time while holding the current output constant. The current supplied to the system will again depend on compatibility with the selected motor. Voltage, capacity and amperage are the primary factors to consider for the vehicle to operate as planned.

There are other aspects of car batteries that need to be addressed. For example, a battery for an EV will obviously need to be rechargeable. Feasible electric car batteries typically fall into two categories: lead acid and lithium ion. Lithium ion batteries are generally accepted as the superior battery type, but this comes as a significant increase in price. Lithium batteries are not as heavy, and have a longer life cycle per cost. In addition, there is a phenomenon called the Peukert Effect which generally means that the faster a battery is discharged, the less efficient it is [2]. This effect is more significant for lead acid batteries than lithium ion batteries. Figures 1 & 2 from a senior application engineer at Mercedes-Benz Energy [3] show some comparisons between the two types:

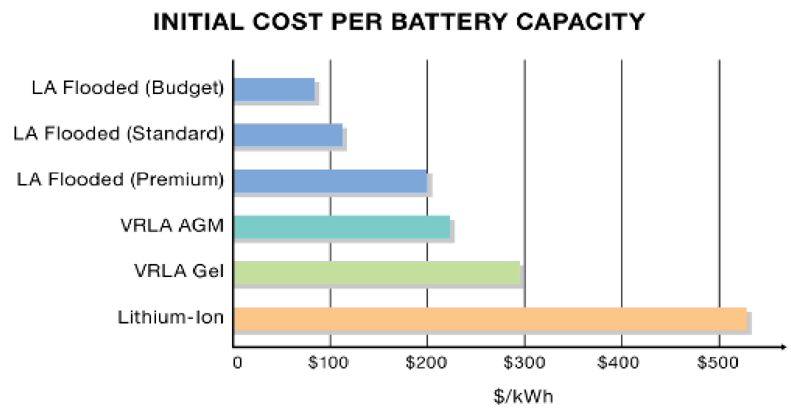

Figure 1. Initial cost per capacity for various EV batteries. Lead-acid and lithium-ion batteries fall on the extremes of cost per capacity.

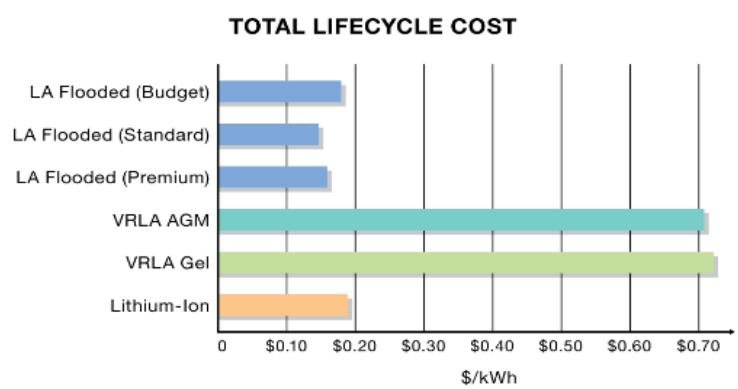

Figure 2. Total lifecycle cost for various EV batteries. Lithium-ion batteries are economical due to their long lifespan.

Based on the elements previously described for batteries, we have some idea of how to move forward when sizing a battery. The battery's capacity will be determined by the target range of 40 miles for the vehicle, as well as the overall weight of the vehicle. The battery voltage needs to be compatible with the AC-50 motor for proper motor control. In addition, the battery needs to be rechargeable with a sufficient life expectancy of 10 years. There are other aspects of the battery that can be examined, such as charging time, regenerative braking efficiency, but considering the focus of this project is on the transmission we will omit these other aspects.

Motor and Controller

The motor and motor controller were already specified to be the HPEVS AC-50 and a 650 Amp, 96V Curtis 1238, respectively. According to the HPEVS website [4], this motor can take a range of different currents and voltages to power it, with Figure 3 corresponding to performance at 96V and 650A.

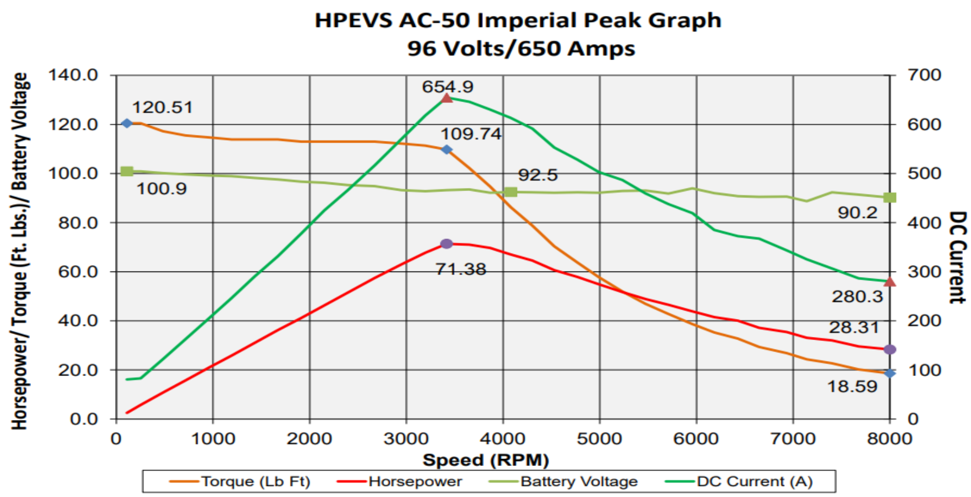

Figure 3. Dynamometer pull data at full throttle for HPEVS AC-50 motor. This dynamometer data shows that the motor has significant torque before a steep dropoff around 3500 RPM.

As shown in Figure 3, the torque begins to drop off after around 3500 RPM while the horsepower peaks at this speed. These values will be considered in designing when to shift the transmission from first to second gear. The controller is rated at 50KW with a max efficiency of 88%.

Transmission

Primary factors to consider when selecting the transmission are the weight, cost, reliability, and shift speed. This list is not exhaustive, and a more thorough decision matrix is in Attachment C. There are many different types of transmissions being used for the various IC and electric vehicles in common use. The factors that are the most important to the vehicle determine what transmission will be used. If shift time is of most concern and cost is not a factor, a dual-clutch transmission might be used. If reliability and cost are most important, an automatic planetary gear set transmission could be used, as most production vehicles use today. Overall most transmissions can be categorized into manual and automatic transmissions.

Automatic transmissions typically contain complicated torque converters, planetary gear sets, and valve bodies for controlling the transmission. Manual transmissions will be simpler and smaller in size and weight to help with packaging and weight distribution of the car. From each overall design more complicated systems can be added to tailor the transmission to fit its design requirements best.

Using an electric motor as our power source allows more mounting opportunities than a conventional internal combustion engine. With the electric motor not having a rotational idle speed, the motor can mount directly to the transmission input without a clutch. When the vehicle is not in motion the electric motor can come to a complete stop with the transmission. Therefore clutch is needed to separate the motor's rotation from the transmission when stopped, unlike with an internal combustion engine. While moving the clutch allows separation to change gears and match gear speeds. The electric motor controller has this capability, so that aspect of a clutch is unnecessary.

Shifting for a manual transmission would be a simple forward or back selector that would be manual or electrically controlled. Manually a lever would be used to select the high or low gear. If manually controlling rev matching the driver would have to rev match and change gears at the correct time. If rev matching is electrically controlled a switch would trigger the motor to speed up to match the speed of the gear. Electrically controlled shifting uses an actuator to move the shift fork/rail to change gears and would have use electronic rev matching. Using electrically controlled shifting is the best option due to the possibility of missing a gear when manually controlled. This would leave the car in neutral with rev matching or coming to a stop as the only way to select a gear.

Vehicle Loads

Determining the vehicle loads on the car is an important step in generating accurate simulations to model the performance of our car. When considering the vehicle loads for designing the conversion to an electric vehicle, there are a few different resistance factors to consider: rolling, drag, bearing and driveshaft. See Attachment D for the coded version that was used in simulation.

Rolling resistance will depend primarily on the aspects of the tires and its contact with the road [5]. A larger tire size and weight will cause the moment of inertia to increase, which will require higher torque transmission to generate the same amount of motion. In addition, the coefficient of friction between the tires and the road plays a part in the allowable acceleration when moving the car from rest to a given speed.

The drag force on the car is based on the cross-sectional area of the car. Fortunately we are not designing a new chassis, so we can use the existing data of the car to estimate drag forces for a given speed.

The drive shaft and bearing resistance will also play a part in the simulations for vehicle motion [6]. The drive shaft inertia can be lumped with the transmission inertia, and from a design perspective it will be better to be conservative with this in mind [7]. Bearings resistance will depend heavily on the type of bearing used. Two primary options to consider are fluid bearings and roller bearings. For example, if we specify fluid bearings, the main design parameters are bearing unit load, rotating speed and fluid viscosity. More specifics will be dealt with further along the design process, but at a minimum the bearings need to last for the target 10 year lifetime.

Detailed Requirements

Simulations

We created a vehicle simulation in MATLAB® to provide us with key functional requirements. The simulation itself used vehicle-specific information - drag coefficient, chassis weight, rolling resistance, etc. - along with two simulated runs to provide us with information regarding battery size, performance, and gear ratios. All runs were simulated with two occupants (320 lbs) and the simulation is printed in Attachment E.

The first simulated run was the commute of our targeted customer. This involved a freeway trip up and down a pass followed by travel on surface streets, and then the return trip. We assumed a time-to-speed of ten seconds. We also assumed that no regenerative braking could be harnessed down the hill, although the motor controller is capable of regenerative braking. These assumptions were conservative in terms of battery life, as the relatively fast acceleration uses more energy and the regenerative braking would mitigate against some energy use.

The simulated run was then transformed from vehicle speed and traction force to motor speed and motor torque using given transmission ratios and a set shift point. These were integrated over time then over motor position. The output of this integration was the total energy used by the motor, and by applying efficiency values for the differential and transmission (98% each), motor (80% at the lowest), and controller (88%) we were able to output a necessary battery capacity and factor of safety.

The second simulated run was at full-throttle, allowing the car to accelerate to its top speed over 90 seconds. This simulation used interpolated values for motor torque at each speed from a dynamometer pull that the motor manufacturer provided. This torque was transferred to the tires to provide a traction force, which was opposed by the same road forces as the first simulation. By iterating over time with a set shift point, our simulation provided an acceleration time to 60 mph as well as a top speed. Figure 4 shows the full acceleration, with 60 dealt within 8.2 s and a top speed of 93 mph.

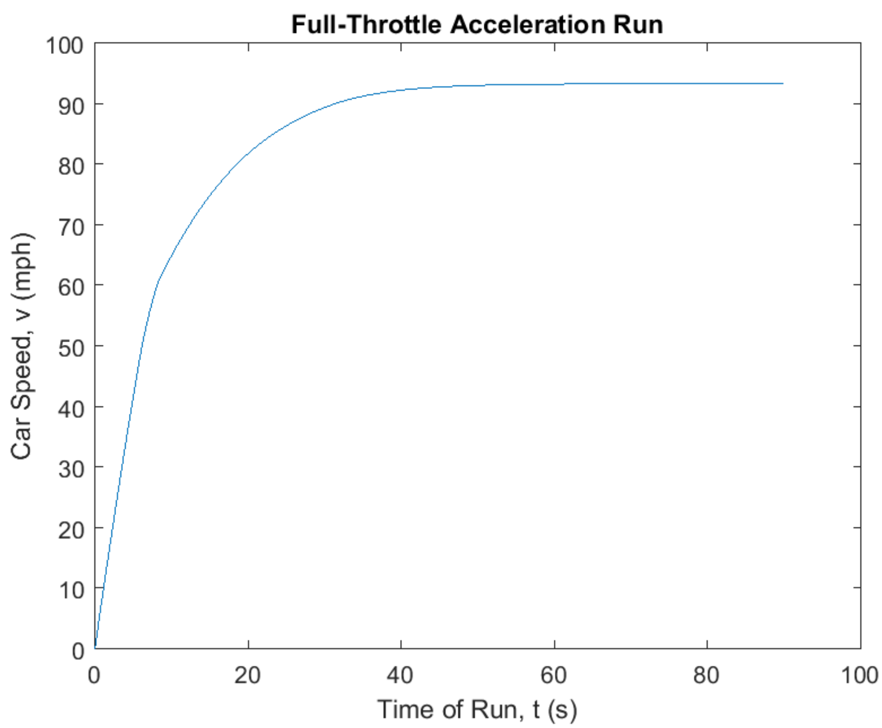

Figure 4. The car speed tracked over the simulated full-throttle run. The gear change occurs around 62 mph, and can be seen as a slight knee in the curve. The first gear handles the full acceleration to 60 mph, then the second gear is used more for cruising, and brings the car to a top speed of 93 mph in around 60 seconds.

The gear ratios were also decided as a result of these two simulations. The process was manual iteration, with initial values for gear 1, gear 2, and shift point first set at reasonable values, then adjusted up or down individually until a peak performance was found. It should be noted that the gear ratios had a negligible effect on battery usage, largely because the acceleration time during the first simulated run was minimal. Figure 5 shows the final result, with ratios of 1.4:1 and 0.714:1.

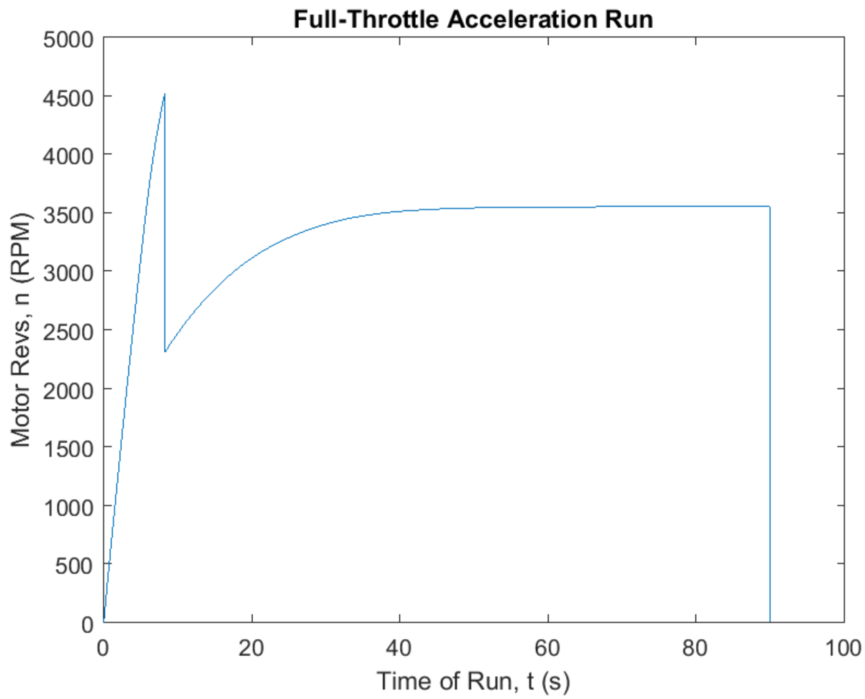

Figure 5. The motor speed during the simulated full-throttle run. At about 9 seconds in, the gear is changed. This can be seen in the large drop in motor speed, and also comes with a lower rate of change of motor speed.

<u>Spatial requirements</u>

Transmission functional requirements - When designing the transmission, the output needs to be compatible with the current car's implementation using a Borg and Warner Mustang T5 Transmission. To be specific, the tail shaft is a 28 spline, male shaft with a diameter of 1.1875 inches. This needs to fit within the drive shaft in Figure 6.

- Length to center of u-joint... Dimension #1 = 6 inches
- Barrel Diameter... Dimension #2 = 1 1/2 inches
- Barrel Length... Dimension #3 = 4 1/4 inches
- U-joint cap diameter... Dimension #4 = 1 1/16 inches
- U-joint width... Dimension #5 = 3 7/32 inches
- Series... 1310
- Spline count... 28

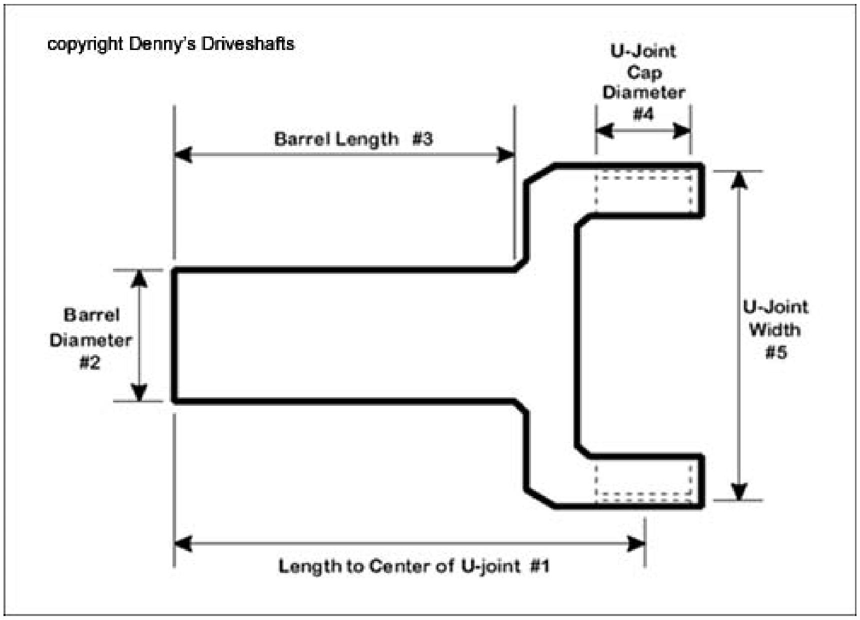

Figure 6. Driveshaft dimensions

From a spatial standpoint, the entire system needs to fit within the chassis in Figure 7. The front of the car currently contains the transmission and internal combustion engine in a trapezoid with a height of 29 inches and a base of 17.0 inches. The frame is 13 inches deep as viewed from the side of the car. Figure 7 is a rough model of the chassis dimensions using solid modeling.

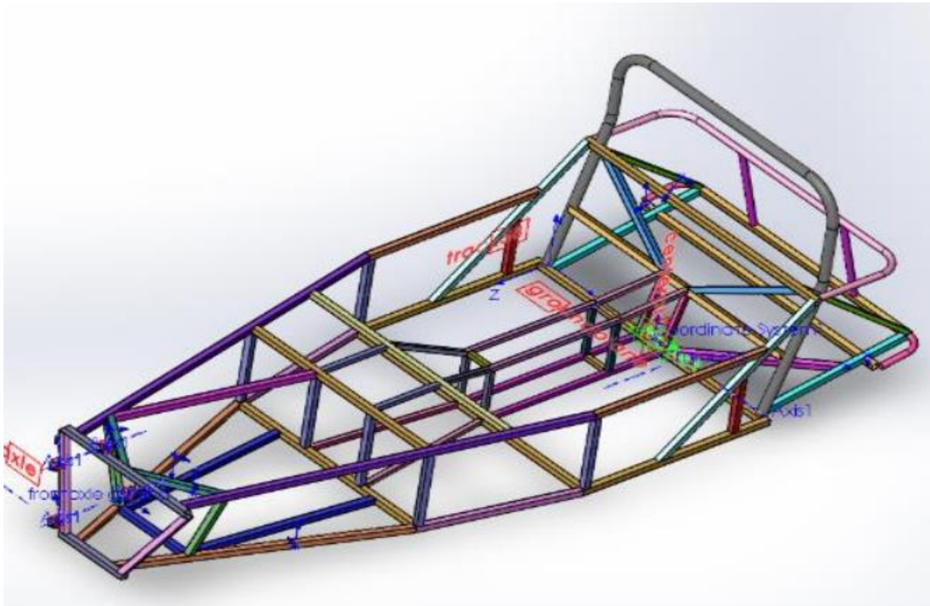

Figure 7. CAD model of chassis for car of concern. This car was designed in SolidWorks, and packaging requirements were based on the solid model.

All of the components that we specify, including the new transmission, batteries, controller, cooling system etc. will be constrained to this part of the car. It may be possible to fit some features outside of the front chamber, such as a portion of the battery, but we should not rely on this too heavily. The motor that is specified has a length just over 18 inches and a diameter of approximately 9 inches as shown in Figure 8 (units in inches).

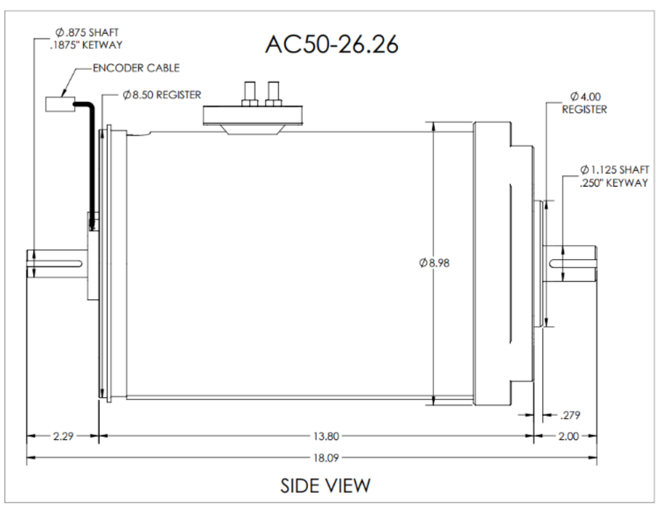

Figure 8. Schematic of HPEVS AC-50 Motor.

The motor will likely be located in the front of the designated chamber, and will leave approximately 11 inches of length for the transmission. The width and height of the transmission will depend primarily on the size of the gears that we choose. The maximum height of our transmission is 13 inches. With the input and output shafts vertically oriented, this leaves just over 6 inches for the average gear diameter. According to American Gear Inc. [8], a standard diametral pitch range for an AGMA quality of 10 at this diameter is between 20 and 200. Lastly, pending on final dimensions of the batteries, they will ideally be located on either side of the transmission where there is empty space in the chamber.

Table of Requirements

Table 1: Table of Requirements

Variable	Value	Units
Weight of Trans.	50	Lbf
Length of Trans.	11	inches
Height of Trans.	10	inches
Width of Trans.	5	inches
Input Shaft	1 ⅛ Inch Keyed	N/A
Output Shaft	1.1875 Inch 28-Spline	N/A
Input Torque	130	Lbf-Ft
Output Torque	180	Lbf-Ft
Input Speed	8000	RPM
Output Speed	11,200	RPM
Low Gear Ratio	1.40:1	N/A
High Gear Ratio	0.714:1	N/A

Transmission Concepts and Selection

Sequential Transmission:

Our first transmission concept is a sequential gearbox. Typically found in motorcycles and high-end race cars, this type of transmission in not common for production vehicles. One of its key features is the fact that you cannot skip gears due to the gear shifting mechanism. This plays in heavily with how the selector shaft operates. As shown in Figure 9 below, there are grooves in the selector shaft that are oriented vertically with a zig-zag. The selector fork contains a pin that rides in the groove. When the operator initiates a gear change the selector shaft rotates, and in order for the pin on the fork to stay within the slot the selector shaft, the entire fork will translate left or right. The collar that is connected to the selector fork will also move horizontally, and will lock in with one of the gears. Since the collar is connected to the output shaft via splines, when the collar meshes with one of the gears and rotates with that gear, torque and power will be transmitted to the output shaft. From a functionality standpoint, the sequential transmission is not used frequently because it is quite clunky at lower speeds. Shifting gears is loud and not smooth. Furthermore, implementing a sequential transmission is relatively expensive compared to other transmission types.

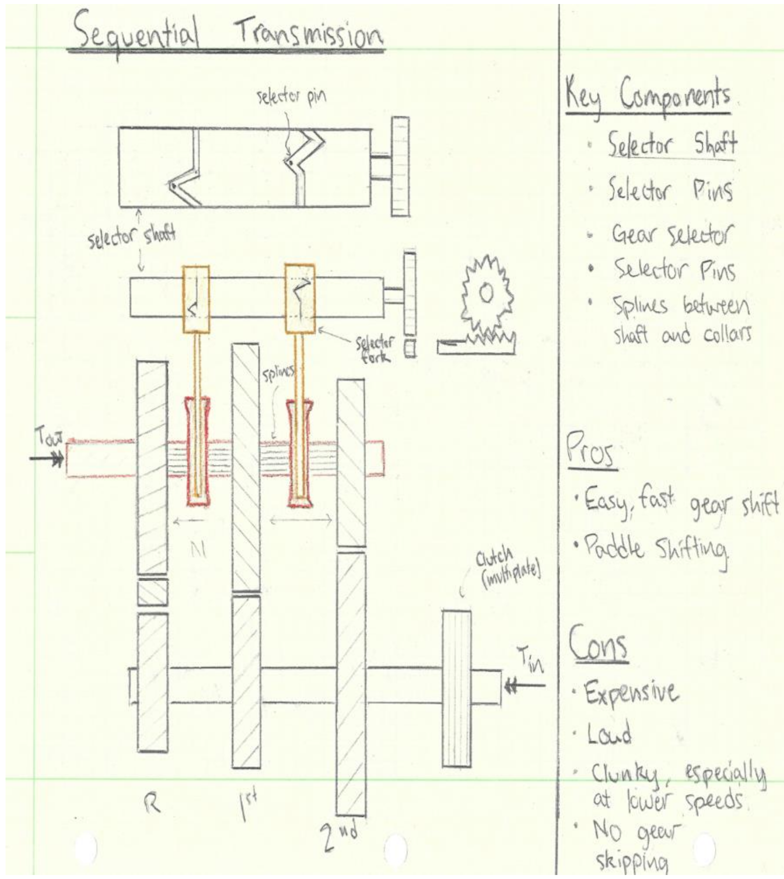

Figure 9. Basic layout for a two-speed sequential transmission.

Planetary Transmission:

The planetary transmission is an automatic transmission option that requires no shifting input because it is done by the car's controller. Since shifting done by the car's controller, this introduces more complexity. The main components of the transmission include a sun gear, planet gears, ring, arm, and clutch pack which can be seen in the figure 10 below. Since we are designing a two-speed transmission, a single gearset can provide both necessary ratios. The locked ring will be the lower gear ratio and the direct drive will be the higher gear ratio. Clutches lock up different components of the gearset, providing different ratios. The construction of this type of transmission is not very simple since it requires multiple gear surfaces and at least one clutch mechanism. This is also difficult to service by the user, and because this is a one off transmission serviceability is important. Relative to other transmissions, planetary transmission is also very expensive because of the complex components [5]. In Figure 10, the main components can be explicitly seen with a side and front view for better understanding. Torque comes in the left and is transmitted by the rotation of the planets around the ring gear being rotated by the sun gear or all the components locked up causing a direct drive. In order to have more ratios another planetary gearset must be attached. The gears are locked up by the use of a clutch pack also seen in the figure.

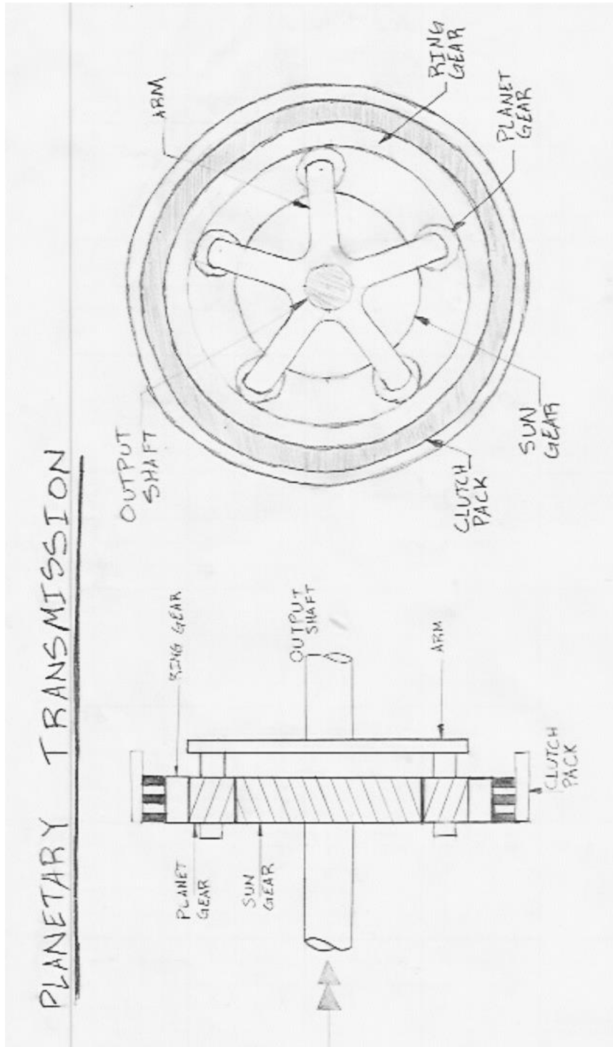

Figure 10. Basic layout for a 2-speed planetary transmission.

<u>Dual Clutch Transmission:</u>

The operation of a dual clutch transmission is similar to a regular manual transmission using clutches to shift between shafts. It utilizes two electronically actuated clutches instead of a pedal to shift.. This requires electronic components to be able to actuate the shift of the collar from one gear to another. With this actuated shifting the delay that is felt by a standard manual transmission is eliminated, creating faster shifting. The actual shaft of the dual clutch transmission is two separate shafts making it possible to smoothly shift between intermediate gears such as one to two or two to three with the gears running on separate shafts before shifting. Since we will only be using two gears for our transmission the dual clutch will only shift between first and second gear making it not an ideal candidate to design because of the complexity in only being able to shift one gear [6]. Looking at Figure 11, we can see that each set of gears is running on separate shafts and at the far left is the clutch mechanism that will lock one gearset or the other. Both input clutches are clearly identified with the second gear shaft running inside of the first gear shaft.

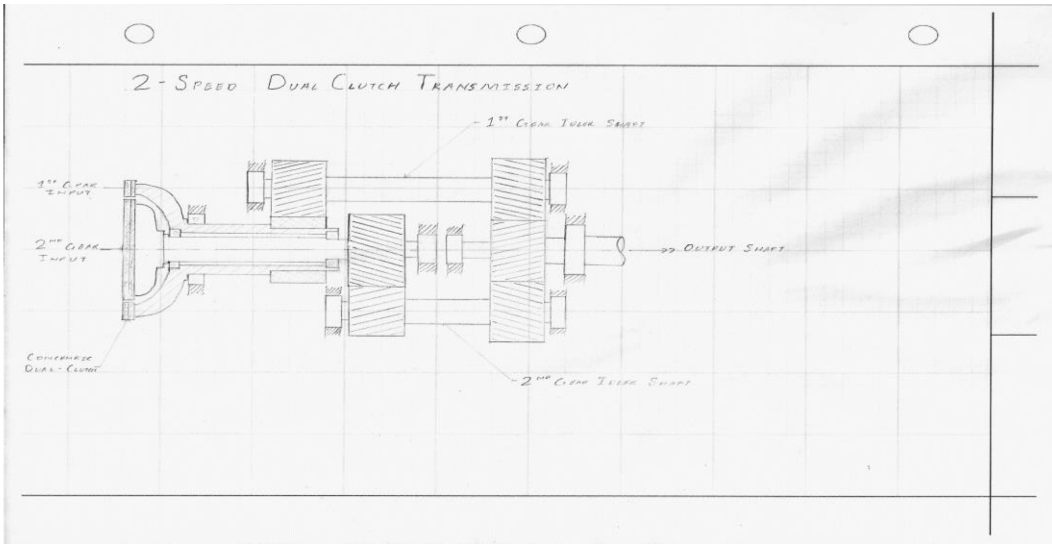

Figure 11. Dual-clutch layout for a two-speed transmission. This figure has exaggerated spacing between components for clarity. Key features of this system are the dual idler shafts and the concentric dual clutch.

<u>Dog Box Manual Transmission:</u>

A dog box transmission, as shown in Figure 12, gets its name from the large dog teeth used to engage the gears when shifting. When shifting, the dog teeth of one gear and the dog teeth of the sliding collar release and slide apart and are free to spin independently. The motor is used to match rotational speed of the next gear to the output shaft and collar. The collar then slides further, engaging the dog teeth of the next gear. Because the collar is rotationally fixed to the shaft, the next gear now transmits torque. The dog teeth have space between each other to account for minor rotation speed differences. This method of changing gears results in simple, yet very fast shifting. This type of manual dog box is most commonly found in racing where short shift times and high strength are required. A gearset with stronger gears and dog teeth engagement can be swapped into a standard manual transmission case. Dog teeth engagement is found in sequential transmissions as well. This type of transmission is not common on production vehicles due to the shifting firmness, loudness of the dog teeth engaging, and the necessity of a forceful gear change. These same reasons make dog boxes ideal for some racing transmissions.

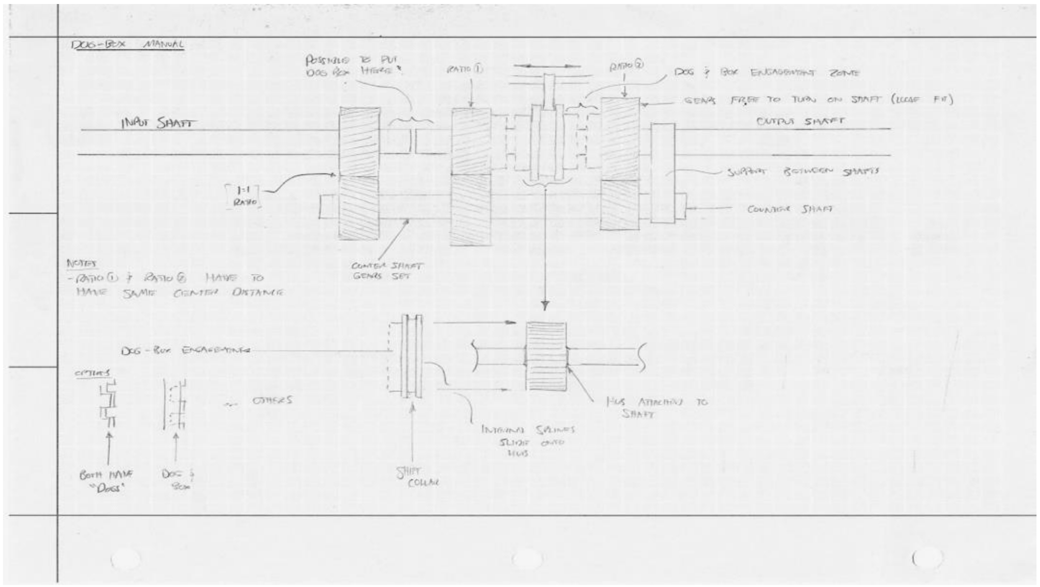

Figure 12. Basic layout of 2-speed dog box transmission.

Concept Selection

For the selection of the transmission we considered multiple engineering design factors and requirements of the user. These were defined in a house of quality matrix which consisted of quality characteristics and demanded quality (Attachment B). The quality characteristics were factors controlled by the engineers that are important in the design process to create parts of certain quality. The demanded quality requirements are requirements considered by the user that are not directly related to an engineering requirement. For example, the relationship between “easy to drive” and shift time is strong. This is considered when creating a weighted importance. With this weighted importance these values can then be used in the decision matrix created to determine which type of the four transmissions is the best option to engineer and for the user (Attachment C).

In our decision matrix we used two weighting factors in order determine what requirements were more important than others. One was completed in the actual decision matrix the other was values taken from the house of quality as mentioned. These two were then directly combined into one overall weight factor. We set the planetary gear option as our baseline so the others we rated relative to it. Each engineering requirement, quality characteristics, that was previously identified in the house of quality was then given a +1, 0, or -1 relative to the planetary transmission. After the completion of the decision matrix the dog box transmission option was what came out on top as what would be our best design considering the engineering and user requirements. After considering if this indeed made sense we were able to conclude that it is our best option moving forward since it is the most realistic option for design. Also, the gear ratio that was selected in the requirements section of the report does indeed work for this transmission type.

Detailed Design

Gears

The gears are the true workings of our transmission. Because we found that gear ratios of 1.4:1 and 1:1.4 were optimum for vehicle performance, we needed to specify gear face width, pitch, angles, and shaft centerline distance.

We selected face width, gear angles, and pitch before calculating CL distance. Face width was initially chosen to be 1 inch based upon pictures of similar transmissions. We chose 20 degree pressure and helix angle gears due to their strength and common availability. Diametral pitch was initially set a 8 tpi, as this often provides a compromise between wear life and bending fatigue life [12]. Gears were chosen to be Grade 2 Carburized, which is common for long-life gears [11].

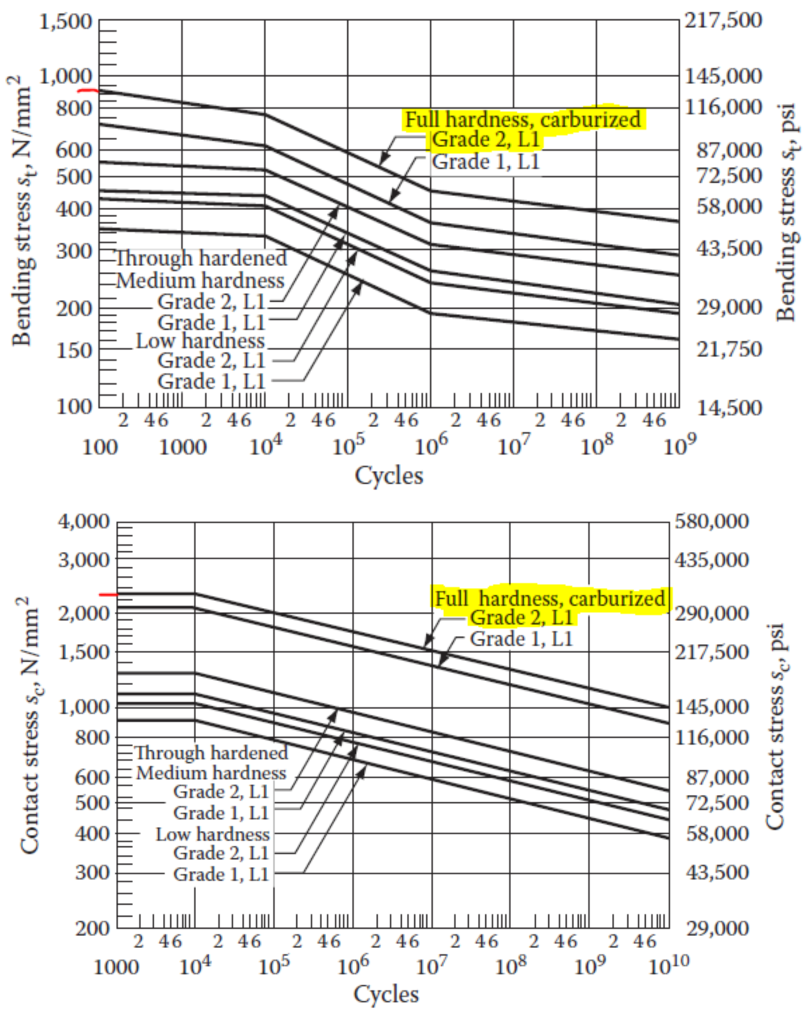

Figure 13. Figures used to determine statistical strengths of gears. These display statistical strengths for carburized Grade 2 gear steel at 99% certainty [11]. The red marks signify the nominal strength of the gears.

Load conditions for fatigue were required to determine the gear sizes. Loads were based upon MATLAB simulations that we used for determining vehicle range and performance, with the assumption that the end user commutes 20 times/mo for ten years and twice a month takes the car for a full 0-Vmax acceleration.

Shaft CL distance was related to the pinion and gear diameters of each gearset. By increasing CL distance, the gears became larger and could take larger loads. We used Miner's Law for cumulative damage to determine the necessary size of the gears. Essentially, AGMA gear life equations were used to determine stresses from contact and bending at every torque in each loading regime, then the CL distance was gradually decreased from a high value (5 inches) until the cumulative damage over 10 years for either bending or for contact was enough to break the gears. 99% reliability with derating factors from [11] were used.

We found that the gears would fail in contact with a 2.420 inch CL distance. We also reduced face width to 0.5 inches for packaging. This yielded gear pitch diameters in the table below, which set the standpoint for shaft calculations. We had a factor of safety on life of 10 with these diameters, and due to size constraints of our shaft we were unable to iterate further. The final gear sizes were based upon integer values for diametral pitch and tooth number. By rounding to 38 teeth on the pinion and 53 teeth on the gear at 20 TPI, we found final pitch diameters of 2.022 and 2.820 inches.

Table 2: Gear Values for Dimensions

Variable	**Pinion**	**Gear**	**Units**
Diameter	2.022	2.820	Inches
Pitch Diameters	2.009	2.182	Inches
Number of teeth	38	53	Teeth
Face width	0.5	0.5	Inches
Teeth per inch	20		Teeth/Inch
Centerline Distance	2.420		Inches

<u>Shaft</u>

Our transmission consists of an input and output shaft in parallel. Since we are not lining up the output and input shaft the transmission will have an offset, allowing us to better utilize the space we have allocated for the transmission.

The helical gears will be oriented to pull the shafts into the transmission, therefore the axial load will be carried by the bearings with the help of a thrust washer. This is to ensure the contact from the gear faces will be on only the thrust washer then only onto the race of the bearing. This eliminates the need for a step in the shaft.

The input shaft is solid forged 9310 AISI steel, including the gears. This simplifies the gear design, strengthens the shaft metallurgically, and reduces shaft features. There will not need to be a step where the bearing gets mounted due to the thrust washer, so the only stress concentrations will be at the gear steps.

The output shaft was designed as a hollow shaft with splines acting as the hub for the collar to slide over. Again, there are no other steps on the shaft decreasing the number of locations of stress concentrations.

To calculate shaft stresses, we began by drawing free-body diagrams of the shafts and gears. The worst case loading for either shaft occurred when the larger gear was engaged, so conservatively the larger gear was modeled as being engaged at all times.

First, a shaft diameter was estimated then all of the reaction calculations were completed followed by the deflection at the gears and the rotation at the bearings. Minimum values from [12] were selected with bearing rotation constrained to a maximum of 0.0005 radians and gear deflection less than 0.01 inches. We iterated the shaft's minimum diameter until both of these criteria were met with an ample factor of safety.

A similar process was utilized when sizing shafts for life. As usual, a critical point was found at the root of a step, and [13] was used to determine material fatigue life. An allowable stress of 70 ksi corresponded to our stress ratio of -0.57. Table 3 displays the relevant properties of each shaft for sizing.

Table 3: Dimensions and values for Shafts

Variable	**Input Shaft**	**Output Shaft**	**Units**
Elastic Modulus	30	29.9	Msi
Diameter (outer)	1.182	0.985	Inches
Diameter (Inner)	N/A	3/16	Inches
Stress Allowable	70	70	ksi
Stress Maximum	25.01	46.38	ksi
Safety Factor Life	2.80	2.11	N/A
Critical Speed	24238	20245	rpm

<u>Shifting</u>

The shifting mechanism of the car is comprised of a splined hub on the shaft, a collar with dog teeth, an arm to slide the collar, and an electromechanical system for actual shifting.

The gears on the output shaft spin freely on journal bearings when not engaged. These bearings are lubricated by oil ports on the hollow shaft, which will contain pressurized oil. When the arm presses the collar into a gear, the dog teeth lock the gear to the collar which transmits torque through the hub into the output shaft.

The shifting arm is similar to that of a manual transmission. A semicircular fork rides in a groove in the collar, and the shaft portion of the arm slides within two bushings in the housing. The fork axially locates the collar based upon three detents and a sprung pin. This is a system utilized in standard manual transmissions for ensuring that the transmission doesn't "slip" out of gear. These detents additionally provide a sensing point for Hall-effect sensors, which feed transmission position into the TCU.

Our transmission operates in the realm of the semi-automatic or automated manual transmission category. This means that gears are shifted by an electromechanical system at the transmission, but the driver inputs the times for the shifts. We found that a pair of solenoids provided ample force to shift the gears very quickly, and they would be actuated by a transmission control unit (TCU). The TCU would monitor motor speed, gearshift paddle inputs, and current gear position. On the input from a shift lever, the TCU would signal the motor controller to briefly 'blip' the power, allowing for smooth shifting. A reverse button would also be provided, and the TCU would ensure that it could only be activated when the motor was at a standstill.

Calculations for spline sizing in the hub had to begin with an educated guess of the size. Figure 14 displays the starting point for sizing of spline diameters, and we found that the diameter required (~0.7 in) was less than our shaft diameter, so our shaft diameter acted as the constraint for spline size. Machinery's Handbook also suggests a starting point of 1/3 the pitch diameter for spline length.

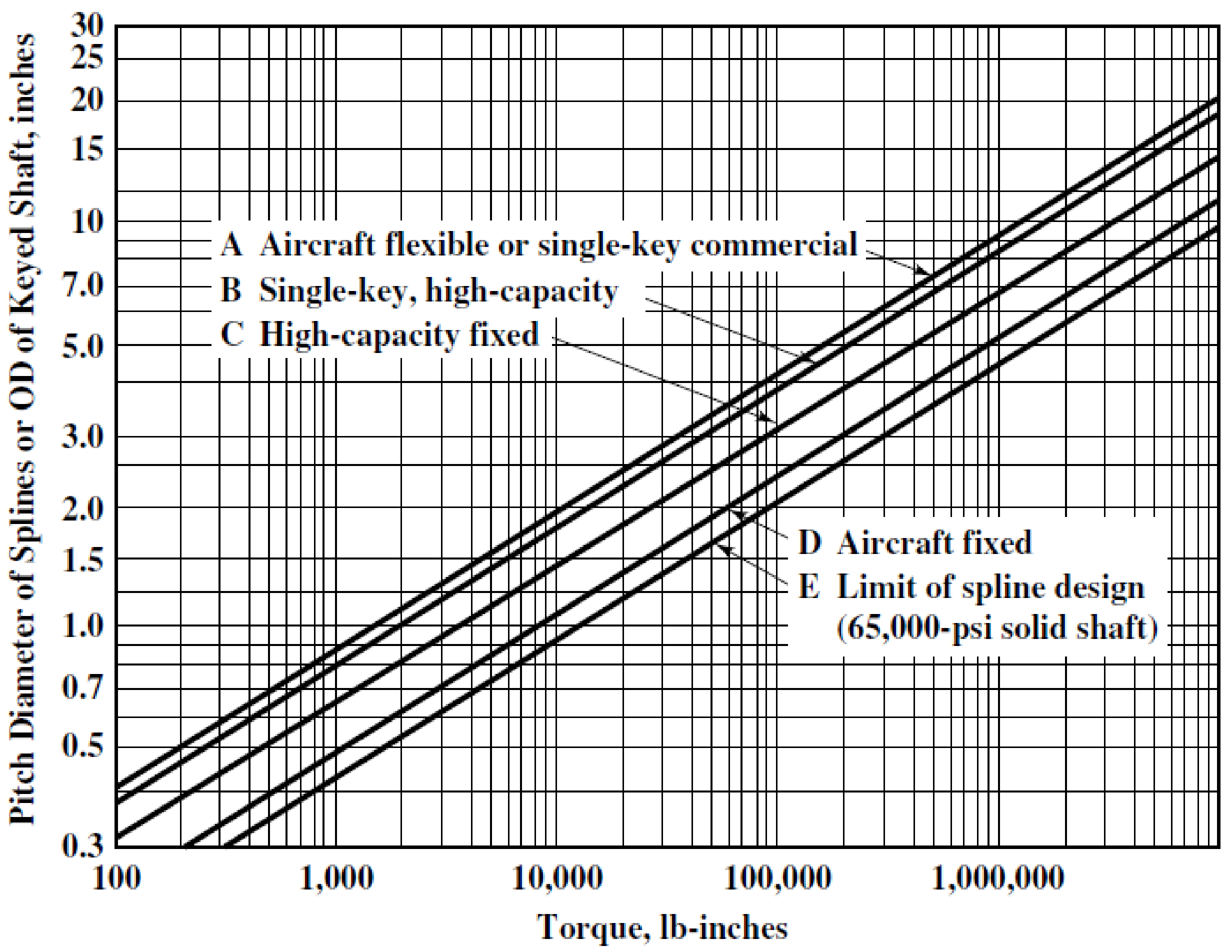

Figure 14. Basic spline sizing for power transmission applications from [11].

For the dog teeth, we began by assuming a number of teeth and general dimensions. Four dog teeth for each side of the collar allowed for ample clearance of teeth within the windows. Teeth were designed as squares with a uniform height of ¼ inch, which was borrowed from designs in other transmissions. The teeth were designed with a factor of safety of 5 to allow for the variable shock loading found in shifting. Table 4 displays the final size of the dog teeth. Calculations for these dog teeth were taken from [11].

Table 4: Table of Dog Teeth and Spline dimensions

Variable	**Value**	**Units**
Dog Teeth(DT) Dim.	L:0.25xW:0.25xH:0.18	Inches
SF on DT for Yield	5	N/A
Spline Root Diameter	0.995	Inches
Spline Height	0.125	Inches
Spline Pitch Diameter	1.245	Inches

Bearings

The primary metric for gauging the validity for the transmission bearings is their operation life. In order to develop these numbers, bearing loads need to be calculated based on parameters such as the input torque from the motor, gear type, pinion and gear dimensions and more. Since the main purpose of a bearing is to allow the transmission shafts to rotate along it major axis, there is assumed to solely be radial and or axial loads on any given bearing. Hand calculations that lay out the method behind estimating the loads for a combined radial and axial load for a single bearing are laid out in Attachment G. From here, estimated life for our transmission conditions based on Timken's C10 and L10 values are calculated. In Attachment F, we have supplied the MATLAB simulations that output the loads and life for each bearing based on max acceleration, uphill, and 65 mph flat speed conditions. Finally, with a six level variable loading model (the 3 dynamic conditions for each gearset engaged) an equivalent life was calculated.

The input shaft will have Timken 6206-RS Deep Groove Ball Bearings, and the output shaft will have Timken 6305-RS Deep Groove Ball Bearings. These are single contact seal bearings, and have excellent lubrication retention and greater torque capacity than their shielded or non-contact seal counterparts for the same bore size. The single shield aspect allows for relubrication from the open side. This specific bearing has a lower rated speed capability than the others, but our transmission will not exceed these specifications.

In terms of bearing loads, since we have specced the same deep groove type of bearing for each side of each shaft, each bearing has the capacity to hold both an axial and radial load. However, for a single shaft, it is typically not recommended for both bearings to take axial loads. This would render the shafts statically indeterminate, making load analysis more difficult. In the assembly of the transmission we opt for only one bearing per shaft to support the thrust load generated by helical gears. In contrast, the pure radial bearing for each shaft has no washer in contact, so no axial load is counteracted.

For the most efficient design, there will be an interference fit between the shaft and the bearing in order to avoid creep or turn that would increase wear, as well as supply advantageous preload to the bearing. Because bearings are not typically made for custom applications, and instead come in set intervals based on manufacturing standards, the exact shaft diameters have been designed based on our selected bearing bore sizes. We have selected a g6 tolerance range for the bearing fitting between the outer shaft and bearing bore diameters. A brief diagram on optimal shaft and housing fit selections is displayed in Figure 15. This comes in at a 25 micron interference which equates to .001 in. As such, based on our bearing bore sizes, the shaft diameters are specced at 1.181in and 0.984 in for the input and output shafts, respectively. More on the shaft specifications are covered in the shaft section of the detailed design.

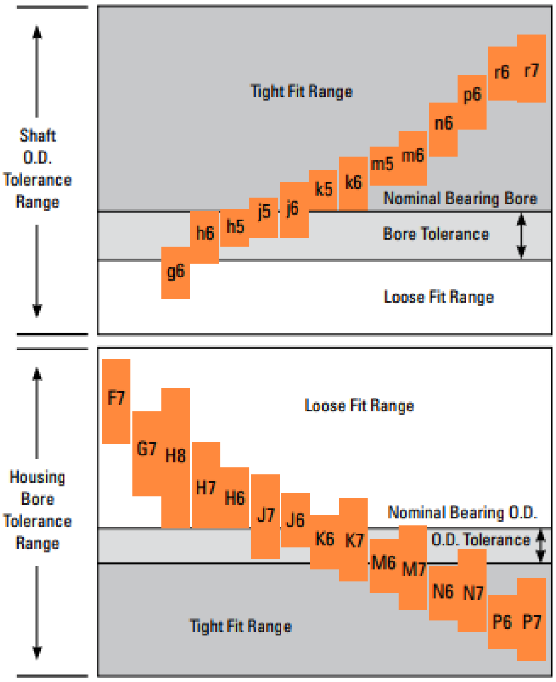

Fig. 2. Shaft and housing fit selection.

Figure 15: Image of different types of press fits for bearings on shafts.

The bearings fit into a figure eight housing, also with a 0.001 inch interference press fit as recommended per Timken. This figure eight housing will be bolted to the overall housing of the transmission. While there is a radial interference fit to prevent slipping between the inner race and the shaft surface, an axial clearance needs to exist between the inner race and the portion of the housing counteracting an axial load. Essentially, all of the axial load needs to be on the outer flat edge of the bearing so that friction between the housing and the moving part of the bearing is not present.

Table 5 contains all of the rated values from Timken's catalog. This includes dynamic and static loads, and dimensions of each bearing. Since Timken primarily deals with metric units in their catalogs, we had to convert these values to English units for our calculations. Metric numbers from the catalog are shown in parentheses and English units for our calculations are shown without parentheses. Loads based on statics and dynamics for the car are shown on the bottom half of the table. Each of the axial, radial and effective loads are based on a gear engagement of 30% in first gear and 70% in second gear throughout the car commute. The effective loads are what was input into Matlab simulations in order to estimate the life of the bearings.

Table 5: Rated values from Timken's Catalog for bearing loads and sizes of bearings

		Input Shaft		Output Shaft		Units
Bearing Number		6206-RS (Radial and Axial)	6206-RS (Radial Only)	6305-RS (Radial and Axial)	6305-RS (Radial Only)	
Static Load		2550 (11.30)	2550 (11.30)	2525 (11.20)	2525 (11.20)	lbf. (kN)
Dynamic Load		4400 (19.50)	4400 (19.50)	4650 (20.60)	4650 (20.60)	lbf. (kN)
Weight		.440 (.200)	.440 (.200)	.485 (.220)	.485 (.220)	lbf (kg)
Bore Diameter		1.181 (30)	1.181 (30)	0.984 (25)	0.984 (25)	in. (mm)
Outside Diameter		2.441 (62)	2.441 (62)	2.441 (62)	2.441 (62)	in. (mm)
Width		.630 (16)	.630 (16)	.669 (17)	.669 (17)	in. (mm)
Radial Load	Accel	278.9	206.3	206.3	278.9	lbf.
	Uphill	221.4	163.8	163.8	221.4	lbf.
	Flat	148.5	109.9	109.9	148.5	lbf.
Axial Load	Accel	44.4	--	59.1	--	lbf.
	Uphill	44.1	--	46.9	--	lbf.
	Flat	29.6	--	31.5	--	lbf.
Effective Load .	Accel	242.3	206.3	207.2	278.9	lbf
	Uphill	192.3	163.8	164.5	221.4	lbf
	Flat	123.0	109.9	110.3	148.5	lbf
Variable Load Life		3.69 (10^9) (2670)	4.27 (10^9) (3092)	6.61 (10^9) (4788)	2.53 (10^9) (1833)	revs (commutes)

Running through the MATLAB simulations shows that the first bearing fails at $2.53*10^9$ revolutions. This value equates to 1833 commutes for the combination of variable loads in the simulation. For a 10-year life of the transmission, if the car drives the commute once every other day, this gives us a factor of safety of 1.00. These simulations were run at a 90% reliability for each shaft.

Lubrication:

Gear and bearing lubrication require a source of pressurized oil to circulate through the transmission. Inadequate lubrication will result in accelerated and unmanageable wear, along with possible overheating of the gear sets. According to Dudley's Practical Gear Design Handbook, petroleum oils are the most common choice for gear lubrication. For a parallel shaft apparatus with a single gear reduction that we have chosen in our design, along with an ambient temperature between 1- C and 50 C, the recommended AGMA lubricant number for gears up to 8 inch diameters is 3-4. From an SAE rating perspective, this lubricant is between a No. 80 and No. 90 oil number, as shown in Figure 15 from [11].

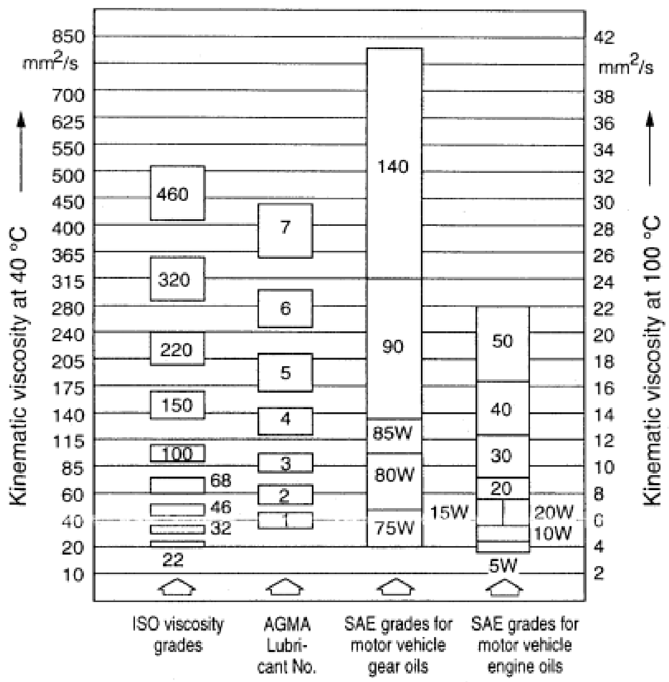

Figure 15. Image of viscosities for different types of lubricants from [11].

Since this is a convertible EV, most likely to be driven around in warm, sunny weather, we can aim for a generally higher viscosity fluid as the ambient temperature will be relatively warm. To start, we will work within the category of an AGMA No. 4 Gear Lubricant. At this rating, transmission fluids can be separated further into synthetic/non-synthetic lubes, extreme pressure (EP)/non-extreme pressure lubes.

To narrow the search between EP non-EP fluids, Dudley's Handbook brings up a concept of Gearset Lubricant Regimens. From a basic standpoint, different regimens denote different lubricant effectiveness's for a gear set. In general, the most optimal gearbox designs will fall within Regimen III, having an oil film thickness large enough to avoid direct contact between the gear surfaces. Two primary factors of pitchline velocity (fpm) and K-factor (psi) are used to determine which of the regimens a gearbox falls within. This is shown in the figure taken out of Dudley's:

Based on the calculations shown in the Attachment G regarding our lubricant regimen rating, we would expect our gearbox to be in the Regimen III range. Dudley's recommends that a Regimen II gearset can be upgraded to a Regimen III set with the implementation of an EP gear lubricant, but since we already fall within the most optimal range, we can opt for a lower performance transmission lubricant without the use of extreme pressure additives.

As per the above considerations, a possible gear lube to further analyze would be Mobil's DTE Extra Heavy product. This is a non-synthetic lube with slightly worse performance than synthetic oil, but due to our gearset falling within the optimal Lubricant Regimen III, we can opt for a more economical option. The data sheet for this specific gear oil denotes a viscosity of 135/150 cSt at 40C and 14.4 cSt at 100C.

We elected to have a dry sump lubrication scheme with an external electric pump for oil. This means that oil accumulates within an aptly named accumulator before being dispensed through various channels to the bearings and the hollow shaft. An electric pump would have to be specified based upon testing, as there is not a readily available analytical model for applying lubrication to gears from a hollow shaft. Around 1 liter (qt) of Mobil's DTE Extra Heavy gear oil would be necessary [11].

The journal bearings are going to be located under the gear on the output shaft since they are freely moving and not attached unless engaged with the collar. The calculations for this are completed in the MATLAB in Attachment F and all the requirements are outlined in Table 6. The oil will be coming in from the shaft and be continually discharging from between the gear and the shaft which will also go onto the gear faces to promote less wear. Between the shaft and bearing is also a 0.022in thick bushing material that is Cadmium (1.5% Nickel). The surface finish on the shaft where the bearing is in contact is 8 microinches which is a fine precision ground at those areas.

Table 6: Fluid Bearing Sizes for Gears on the Output Shaft

Variable	**Value**	**Units**
Bearing Diameter	0.985	Inches
Bushing Thickness	0.022	Inches
Clearance*	0.0125	Inches
Bearing Outer Diameter	1.0315	Inches

* Clearance based on 1.5 Inch Diameter from Shigley's [12]

<u>Additional Details:</u>

There are several auxiliary systems supporting the main power transmission pieces of a transmission. These include the housing, battery, seals, gear and bearing lubrication, and the parking brake.

For the housing of the transmission we will be having a three-panel removal system from the bottom for the sump, input shaft side, and top for the shifting mechanism assembly. At all of these panel attachments there will be a gasket to keep the fluid inside of the housing. The shafts will run output above the input shaft where the input will be attached with a coupling from the motor with set screws to hold it in place. The output shaft will be attached to the driveline using the slip yoke provided by the vehicle. Three hoses will run into the transmission for the oil to go into and out of the transmission. The upper hose will be for the high pressure coming from the pump to go into output shaft. the lower two will recycle the oil back into the pump. The transmission and motor will all be mounted together to the chassis of the vehicle ensuring that the output lines up to the already existent drive shaft.

After spending time researching the battery that will go into powering this vehicle, we have come across a major tradeoff. Either the battery will be very expensive, out-pricing the transmission significantly, or the battery will be extremely heavy, causing major inefficiencies for commutes. For example, a lead-acid battery system comprised of 8 12V batteries will need to have each battery at a 200Ah capacity. Some online browsing shows that these batteries can be obtained for under $2400 from certain suppliers, but unfortunately the weight totals up to over 900 lbs. This is not acceptable for our scenario. As such, we are sacrificing cost and going with the plausible 96V 22kWh Enderdel MP320-049 battery pack. It is expensive at over $10,000, but its lightweight nature is what is important when designing the transmission, which is what this project is primarily aimed at.

For the seals on the housing there are two locations that need to be considered where the shafts enter and exit. These locations will need a seal that does not allow the oil to escape therefore must be a tight enough fit but at the same time allow for smooth rotation of the shafts. Looking at the layout of the transmission the seal on the input shaft needs to fit snugger than the seal on the output since it is lower and will be closer to the oil in the system. From our shaft sizes of 1.182 and 1.379 we need to find out how tight the seals need to be on the shafts. Using an online retailer's website to find the seals required, the input shaft at approximately 30mm would need a 30mm Dia x 62mm Bore x 10mm Seal Nitrile Rubber Lip Double Lip with Spring Oil Seal [14]. For the output shaft, since it is smaller, approximately 25mm, a different seal will need to be used. Looking at the same website a similar seal is found that would work on the shaft. Both of these seals have a spring oil seal which means it grips onto the shaft with a spring around the seal that creates even pressure as to not let the oil out.

The parking brake will utilize an electromechanical locking pin, similar to that of a modern automatic transmission. The collar coupling the motor to the transmission has four detents. These will mate to a pull solenoid, which will have its piston removed from its constraining position when the brake button is toggled.

Conclusion

After completing the transmission detailed design it is important, as designers, to go back and verify that we have met our marks of the preliminary table of requirements. The weight of the transmission was constrained to be at a max 50 lbs. Based the overall volume of the assembly (115.43 in^3), as well as the specific weight of steel (.284 lbs/in^3), our estimated weight comes out to 32.78 lbs. This is accounting for the steel parts, and does not incorporate the oil weight, which should be around 4 lbs. The length, height and width of the entire transmission also satisfy our original requirements with values of 10 in, 10 in, and 3.7 in, respectively. In addition, based on numerous simulations analyzing the gears, shafts, and bearings, our transmission components reach or exceed the targeted life time of 10 years under the required commute conditions.

Although our design is complete for the scope of this project, there is obviously still more that could be done. Notable points in our design that could use work include the housing, pump/oil delivery system, mounting with vibrational analysis, and of course testing to verify. We can say with minimally 95% confidence that our transmission design would help make this car a proper electric sports car.

Attachments:

A) Works Cited
B) House of Quality
C) Decision Matrix
D) Bearing Sizing Image
E) Overall Table of Requirements
F) MATLAB Calculations
G) Hand Calculations

Attachment A: Works Cited

1. MIT Vehicle Team, “A Guide to Understanding Battery Specifications,” *web.mit.edu*, Dec-2008. [Online]. Available: http://web.mit.edu/evt/summary_battery_specifications.pdf. [Accessed: 17-Oct-2017].
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11. Radzevich, S. P., *Dudley's Handbook of Practical Gear Design and Manufacture*, 2nd ed. Boca Raton, FL: CRC Press, 2012.
12. Budynas, R. G., Nisbett, J. K., *Shigley’s Mechanical Engineering Design*, 9th ed. New York, NY: McGraw Hill, 2011.
13. *Military Handbook No. 5. Metallic Materials and Elements for Flight Vehicle Structures*. Dept. of Defense, 1962.
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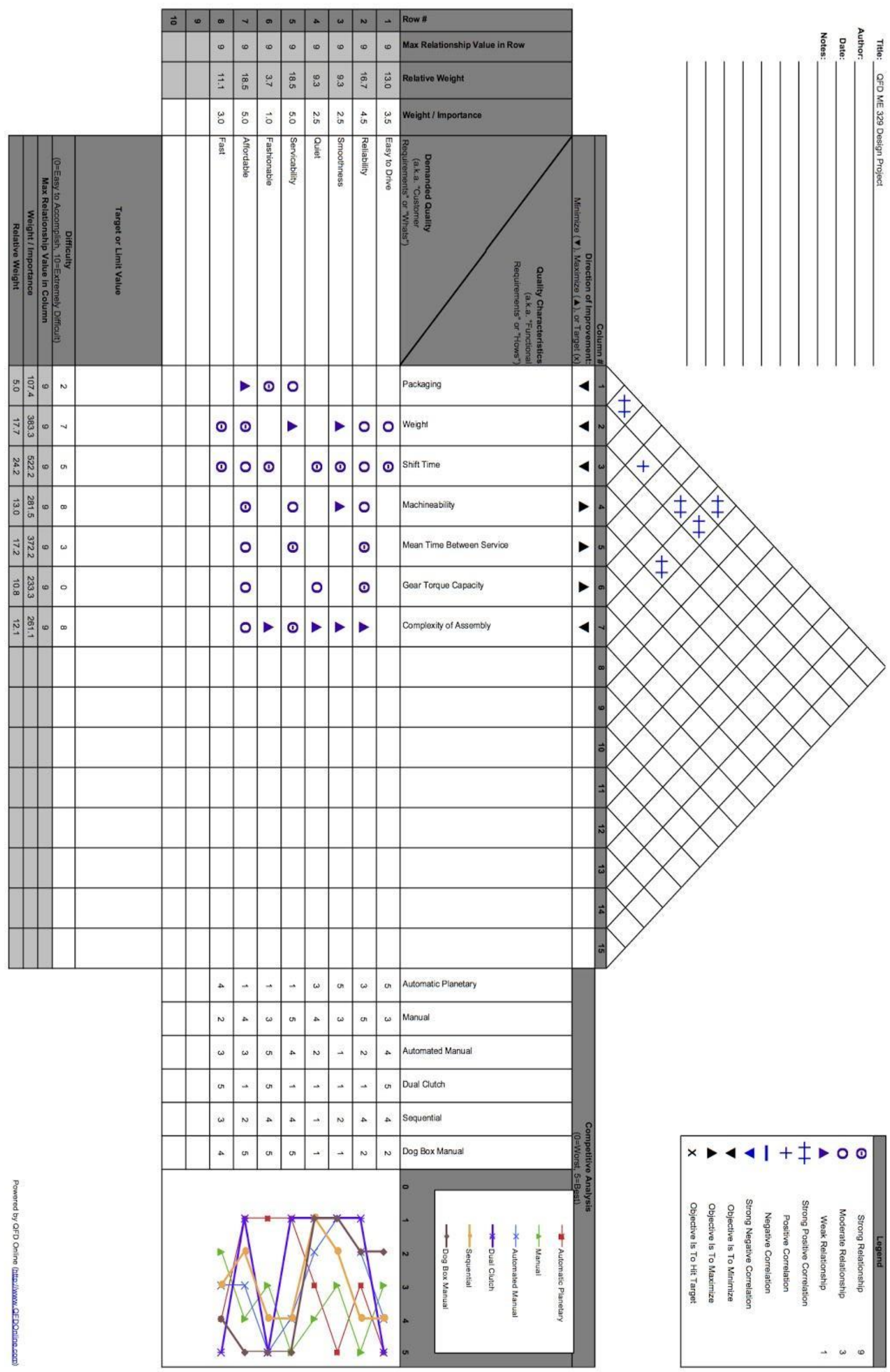
Attachment B: House of Quality



Attachment C: Decision Matrix

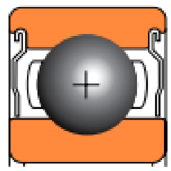
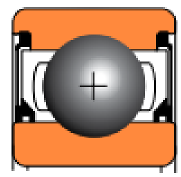
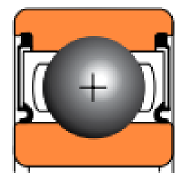
Decision Matrix ME 329

Qualities	Difficulty	Weight Factor	Weight Factor from House of Quality	Combined
Packaging	2	0.06	107.40	6.32
Weight	7	0.21	383.30	78.91
Shift Time	5	0.15	522.20	76.79
Machinability	8	0.24	281.50	66.24
Mean time Between service	3	0.09	372.20	32.84
Gear Toueque Capacity	1	0.03	233.30	6.86
Complexity of assembly	8	0.24	261.10	61.44
Totals	**34.00**	**1.00**	**2161.00**	**329.40**

Characteristics	Weight Factor	*Planetary*	Ratio	*Dog Box*	Ratio	*Sequential*	Ratio	*Dual Clutch*	Ratio
Packaging	6.32	**0**	0	**-1**	-6.3	**-1**	-6.3	**-1**	-6.3
Weight	78.91	**0**	0	**0**	0	**0**	0	**-1**	-79
Shift Time	76.79	**0**	0	**-1**	-77	**-1**	-77	**1**	76.8
Machinability	66.24	**0**	0	**1**	66.2	**0**	0	**1**	66.2
Mean Time Between Failure	32.84	**0**	0	**1**	32.8	**0**	0	**0**	0
Gear Tourque Capacity	6.86	**0**	0	**-1**	-6.9	**-1**	-6.9	**-1**	-6.9
Complexity of Assembly	61.44	**0**	0	**1**	61.4	**1**	61.4	**-1**	-61
	Totals	**0**		**70.53823529**		**-28.53823529**		**-10.5**	

Attachment D: Bearing sizing

TABLE 2. CHARACTERISTICS OF SHIELDS AND SEALS

Type	Shields One = Z Two = ZZ	Non-Contact Seals One = RZ Two = 2RZ	Contact Seals One = RS Two = 2RS
Construction			
Material	Low-carbon pressed steel	Nitrile Buna Rubber with steel case	Nitrile Buna Rubber with steel case
Speed Capability	High speed	High speed	Less than shield(s) and non-contact seal(s) due to seal lip contact
Operating Temperature	-50° C to +120° C	-40° C to +120° C	-40° C to +120° C
Grease Retention	Good	Better than shield(s)	Excellent
Dust Resistance	Good	Better than shield(s)	Excellent
Torque	Low	Low	Greater than shield(s) and non-contact seal(s) due to seal lip contact

NOTE: The above operating temperature ranges are for standard shielded and sealed bearings. If higher temperature capability is needed, alternative bearing, grease or seal materials may be considered. Please contact your Timken sales engineer for such requirements.

Attachment E: Overall Table of requirements

Variable	Value		Units
Weight of Trans.	50		Lbf
Length of Trans.	11		inches
Height of Trans.	10		inches
Width of Trans.	5		inches
Input Shaft	1 ⅛ Inch Keyed		N/A
Output Shaft	1.1875 Inch 28-Spline		N/A
Input Torque	130		Lbf-Ft
Output Torque	180		Lbf-Ft
Input Speed	8000		RPM
Output Speed	11,200		RPM
Low Gear Ratio	1.40:1		N/A
High Gear Ratio	0.714:1		N/A
Variable	**Pinion**	**Gear**	**Units**
Diameter	2.022	2.820	Inches
Pitch Diameters	2.009	2.182	Inches
Number of teeth	38	53	Teeth
Face width	0.5	0.5	Inches
Teeth per inch	20		Teeth/Inch
Centerline Distance	2.420		Inches
Variable	**Input Shaft**	**Output Shaft**	**Units**
Elastic Modulus	30	29.9	Msi
Diameter (outer)	1.182	0.985	Inches
Diameter (Inner)	N/A	3/16	Inches
Stress Allowable	70	70	ksi
Stress Maximum	25.01	46.38	ksi

Safety Factor Life		2.80		2.11	N/A
Critical Speed		24238		20245	rpm
Variable		**Value**			**Units**
Dog Teeth(DT) Dim.		L:0.25xW:0.25xH:0.18			Inches
SF on DT for Yield		2			N/A
Spline Root Diameter		0.995			Inches
Spline Height		0.125			Inches
Spline Pitch Diameter		1.245			Inches
Bearings					
Bearing Number		6206-RS (Radial and Axial)	6206-RS (Radial Only)	6305-RS (Radial and Axial)	6305-RS (Radial Only)
Static Load lbf. (kN)		2550 (11.30)	2550 (11.30)	2525 (11.20)	2525 (11.20)
Dynamic Load lbf. (kN)		4400 (19.50)	4400 (19.50)	4650 (20.60)	4650 (20.60)
Weight lbs (kg)		.440 (.200)	.440 (.200)	.485 (.220)	.485 (.220)
Bore Diameter in. (mm)		1.181 (30)	1.181 (30)	0.984 (25)	0.984 (25)
Outside Diameter in. (mm)		2.441 (62)	2.441 (62)	2.441 (62)	2.441 (62)
Width in. (mm)		.669 (17)	.669 (17)	.748 (19)	.748 (19)
Radial Load lbf.	Accel	278.9	206.3	206.3	278.9
	Uphill	221.4	163.8	163.8	221.4
	Flat	148.5	109.9	109.9	148.5
Axial Load lbf.	Accel	44.4	--	59.1	--
	Uphill	44.1	--	46.9	--
	Flat	29.6	--	31.5	--
	Accel	242.3	206.3	207.2	278.9
	Uphill	192.3	163.8	164.5	221.4

Effective Load lbf.	Flat	123.0	109.9	110.3	148.5
Variable Load Life Revolutions (commutes)		3.69 (10^9) (2670)	4.27 (10^9) (3092)	6.61 (10^9) (4788)	2.53 (10^9) (1833)

Variable	**Value**	**Units**
Bearing Diameter	0.985	Inches
Bushing Thickness	0.022	Inches
Clearance*	0.0125	Inches
Bearing Outer Diameter	1.0315	Inches

Attachment F: MATLAB Code

Table of Contents for MATLAB Attachment

Vehicle Resistance

ME 329 Lab project Ernesto Huerta

```
% This function's inputs are velocity and acceleration with respect to time
% inertias, weight front, and weight back that are constant.
```

Labeling

F_t: rear wheel traction force Wr, Wf: weight rear and front R_r, R_f: rolling resistance rear front Ra: Aero drag force La: aero lift Ma: aero moment r: radius of wheel h: com height l: wheelbase W: vehicle total weigh theta: angle of grade with resptect to time FT: rear wheel traction force

```
function [FT] = Force(v, a, theta, meq)

%conversions
v_c = v;   % in/s
a_t = a*1;      %

% Constants
m = meq;        % mass in Lbm
w = m;          % lbf
fo = 0.001;      % coeficient
fs = 0.005;      % coefficient
Wr = w/2;       % lbf
Wf = w/2;       % lbf
l  = 94;        % inches
h  = 20;        % inches
Cd = 0.66;      % coefficient
A  = 19.50;     % ft^2, front area
rho= 4.335e-5;  % lbs/in^3, density of Air
ha = 25;        % inches
d = 21.75;        % inches

% angular acceleration



% Rolling Resistance
fr = fo+3.24*fs*(v_c/100).^(2.5);
R_r = fr*Wr;
R_f = fr*Wf;

% Air Resistance
Ra = (Cd*rho*(v_c*17.6).^2*A*144/12/32.2)/2;

%



% Force
FT = Ra+R_r+R_f+w*sin(theta)+ m*a_t/12/32.174;



end
```

```
Not enough input arguments.
Error in Force (line 26)
v_c = v;   % in/s
```

Published with MATLAB® R2016b

Full Vehicle Simulation

This here is going to be a simulation to determine our required battery capacity.

```
clear all
close all
clc
format long g
format compact
```

Problem Variables

```
mch=510;       % Weight of the chassis in LBf
mmot=125;       % Weight of the motor and controller in LBf
mbatt=375;     % Weight of the battery in LBf
mtran=50;
mello=320;     % Weight of Dr. Mello without cycling, LBf

e1=1/1.4;      % Low gear
e2=1.4/1;      % High gear
rpmshift=4530;  % Motor RPM at which trans shifts
cd=.66;        % Drag coeficient of the car
Afront=19.5;    % Frontal area of the car, FT^2
dtire=21.75;    % Diameter of the tires
dif=3.45;      % Differential gear ratio
eta=.88*.98*.98*.8; % Constant efficiencies (controller, gears)
meq=mch+mello+mmot+mbatt+mtran+2/dtire*(15/7.5^2+20*(21.75/2)^2 ...
   +25*dif/e2+25*dif)
```

```
meq =
        1611.1205801861
```

Vehicle Kinetmatics

For this, I want to model a torque and angular velocity at the tire for the entire trip.

```
% Normal acceleration to 65mph will be set at 10s. Normal acceleration to
% 30 mph will be set at 5s.

% The conversion from vehicle speed (mph) to wheel angular velocity (RPM)
% is (mph)*C=(RPM)
C=17.6/(21.75*pi);
% The following matrix will track the trip time t (s) and vehicle speed v1
% (mph) for the whole trip.
mph=[[0:10]',linspace(0,65,11)']; % This line is an acceleration to 65
mph=[mph(:,1),mph(:,2)
   [11:(11+415)]',[linspace(65,65,416)']]; % This line is 7.5mi at 65
mph=[mph(:,1),mph(:,2)
   [427:(427+5)]',[linspace(0,30,6)]']; % This line is 0-30 mph
mph=[mph(:,1),mph(:,2)
```

```
    [433:433+300]',linspace(30,30,301)']; % This line is 2.5 mi at 30 mph
mph=[mph(:,1),mph(:,2)
    [734:734+554]',linspace(65,65,555)']; % This line is DOWN the grade

mph=[mph(:,1),mph(:,2)
    [1289:(1289+5)]',[linspace(0,30,6)]']; % This line is 0-30 mph
mph=[mph(:,1),mph(:,2)
    [1295:1295+300]',linspace(30,30,301)']; % This line is 2.5 mi at 30 mph
mph=[mph(:,1),mph(:,2)
    [1596:1596+554]',linspace(65,65,555)']; % This line is UP the grade

mph=[mph(:,1),mph(:,2)
    [2151:2151+10]',linspace(0,65,11)']; % This line is an
                                        % acceleration to 65
mph=[mph(:,1),mph(:,2)
    [2162:(2162+415)]',[linspace(65,65,416)']]; % This line is 7.5mi at 65

%figure
%plot(mph(:,1),mph(:,2))
```

Engine Torque Data

First, I need a matrix of motor speed and motor torque at full throttle. peak will have column 1 be motor speed in RPM and column 2 be motor torque in FT-LB.

```
peak=[0 128
    48        127.591
51        127.592
52        127.593
55        127.594
56        127.595
57        126.706
60        126.707
108       126.708
202       125.82
280       124.93
362       123.31
521       121.54
604       120.66
684       119.773
769       119.772
846       119.771
927       118.89111
1005      118.8911
1170      118.891
1251      118.003333
1330      118.00333
1413      118.003
```

1493	118.0022222
1575	118.002222
1651	118.00222
1812	118.0022
1895	118.002
1976	118.001111
2056	118.001111
2136	118.00111
2216	118.0011
2296	118.001
2469	117.121
2561	117.1211
2654	117.12111
2746	117.121111
2840	117.1211111
2928	117.12111111
3019	117.121111111
3197	116.2311
3280	116.231
3348	116.23111
3419	116.231111
3496	114.31
3573	111.95
3645	109.30
3794	103.99
3856	101.33
3937	98.09
4002	95.29
4078	91.75
4153	89.24
4224	86.58
4372	81.86
4449	79.65
4524	76.11
4602	73.46
4670	70.80
4757	68.15
4825	66.52
4984	61.21
5062	59.44
5149	57.67
5216	56.05
5311	54.28
5382	52.51
5456	50.74
5626	47.20

```
5699        46.32
5793        44.69
5868        42.92
5952        41.15
6012        40.42
6095        39.38
6187        37.61
6343        35.84
6411        34.96
6509        33.34
6568        32.45
6672        31.57
6740        30.68
6830        29.80
6981        28.03
7073        27.14
7132        26.26
7237        25.37
7405        24.19
7573        22.57
7621        21.98
7729        21.091
7788        21.09
7879        20.21
7935        19.32
8000        18.44];
% For continuous operation, the matrix cont will have column 1 be motor
% speed in RPM and column 2 be motor torque in FT-LB.
cont=[1000          26.26
2000        36.73
3000        41.15
4000        50.74
5000        42.04
6000        31.57
7000        22.86];
```

Motor Speed

```
rpm=[mph(:,1),mph(:,2)*5280*12/3600/dtire*2*dif/e1*30/pi];
i=1;
%figure
%plot(rpm(:,1),rpm(:,2))

% This loop is the gear shifts
while i<=length(mph')
    if rpm(i,2)>=4000
        rpm(i,2)=rpm(i,2)*e1/e2;
```

```
        rpm(i,3)=dif/e2;
    else
        rpm(i,2)=rpm(i,2);
        rpm(i,3)=dif/e1;
    end
    i=i+1;
end
%figure
%plot(rpm(:,1),rpm(:,2))
%figure
%plot(rpm(:,1),rpm(:,3))
```

Motor Loads

This section will be used to first determine the force that the car is up against at any speed. Second, it will be used to get a matrix F(t) based upon the trip data.

```
v=mph(:,2);              % Velocity in mph
a=[diff(v);0]*5280*12/60/60; % Acceleration in in/s/s
i=1;
while i<=length(a')
    if a(i,1)<0
        a(i,1)=0;
    end
    i=i+1;
end

theta=zeros(length(rpm'),1);
theta(733:1289,1)=-7;
theta(1596:2151,1)=7;
theta=atan(theta/100);
F=[mph(:,1),Force(v,a,theta,meq)];
T=[F(:,1),F(:,2)*dtire/24./rpm(:,3)]; % Actual motor torque in
                                       % FTLb at each second
i=1;
while i<=length(T')
    if T(i,1)<0
        T(i,1)=0;
    end
    i=i+1;
end
pos=cumtrapz(rpm(:,1),rpm(:,2)*pi/30);
Batt=trapz(pos,T(:,2))*3.76616e-7/eta % Battery cap. in kW-Hr
BattFS=28/Batt
```

```
Batt =
       16.0387862157689
```

```
BattFS =
       1.74576801656419
```

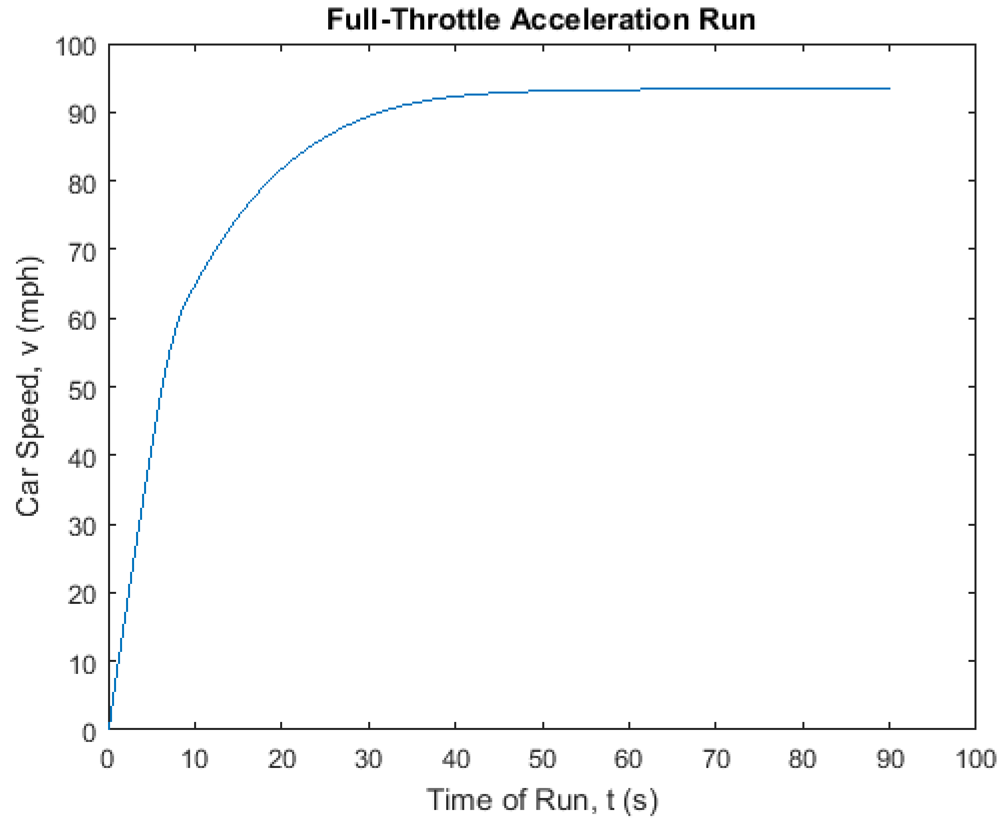
Performance

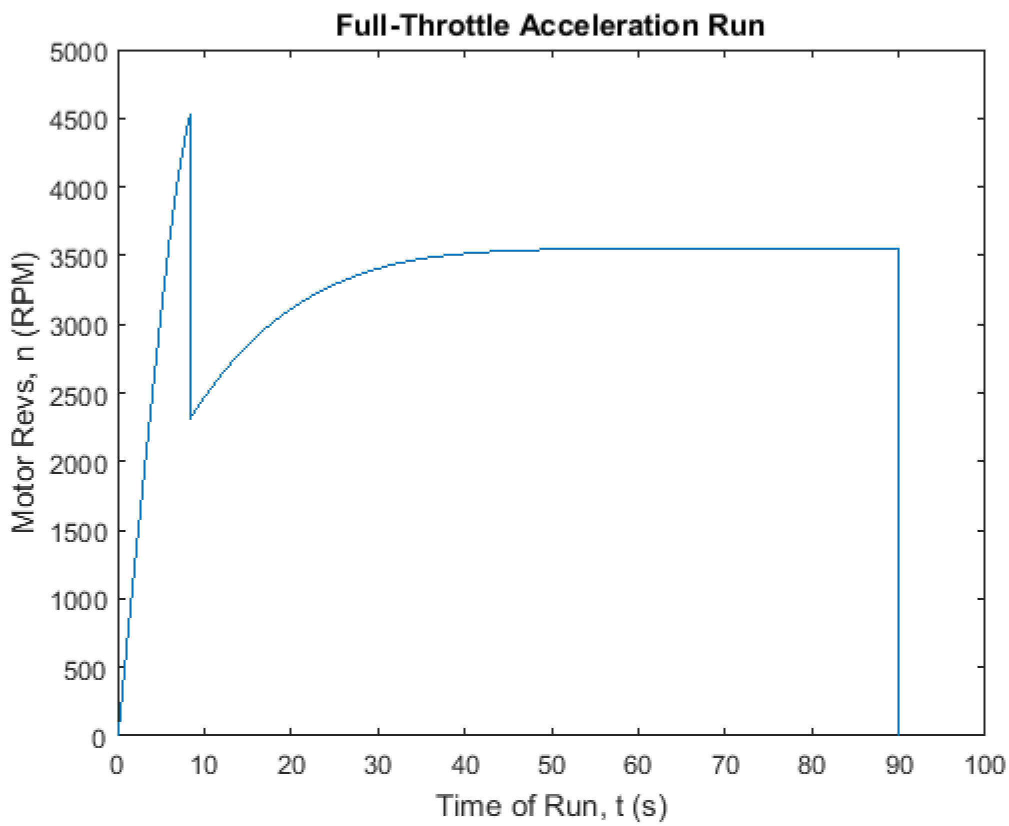
```
tstep=.05;   % Time step for full-throttle run
i=2;
acc=[0,0,0,0
  .1,.1,.1,.1]; % t,v,F,rev
while i<=90/tstep
  rev=acc(i,2)*5280*12/3600*2/dtire*dif/e1*30/pi;
  gear=e1;
  if rev>=rpmshift
    rev=rev*e1/e2;
    gear=e2;
  end
  acc(i,4)=rev;
  acc(i,3)=interp1(peak(:,1),peak(:,2),rev)/gear*dif*2/dtire*12;
  acc(i+1,1)=acc(i,1)+tstep;
  acc(i+1,2)=acc(i,2)+(acc(i,3)-Force(acc(i,2),...
    0,0.001,meq))/meq*32.174*tstep/5280*3600;
  i=i+1;
end
sixty=interp1(acc(:,2),acc(:,1),60)
TOP=acc(length(acc'),2)

figure
plot(acc(:,1),acc(:,2))
title('Full-Throttle Acceleration Run')
xlabel('Time of Run, t (s)')
ylabel('Car Speed, v (mph)')
%print('MPH','-dpng')

figure
plot(acc(:,1),acc(:,4))
title('Full-Throttle Acceleration Run')
xlabel('Time of Run, t (s)')
ylabel('Motor Revs, n (RPM)')
%print('RPM','-dpng')
```

```
sixty =
       8.18896528899922
TOP =
       93.3574113320112
```


Full-Throttle Acceleration Run
Car Speed, v (mph)
Time of Run, t (s)
0
10
20
30
40
50
60
70
80
90
100


Full-Throttle Acceleration Run
Motor Revs, n (RPM)
Time of Run, t (s)
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
0
10
20
30
40
50
60
70
80
90
100

Average Loading for Shaft Calculations

For 30 mph [433:433+300], [1295:1295+300] matrix times at this loading

```
n1=rpm(434,2)*600;     % Number of revolutions at this load in one commute
T1=T(434,2);           % Motor torque at this load, LB-FT

% 65 mph Flat
% [11:(11+415)], [2162:(2162+415)]
n2=rpm(12,2)*830;     % Number of revolutions at this load in one commute
T2=T(12,2);           % Motor torque at this load, LB-FT

% 65 mph Up the Grade
% [1596:1596+554]
n3=rpm(1597,2)*554;    % Number of revolutions at this load in one commute
T3=T(1597,2);          % Motor torque at this load, LB-FT

% 65 mph Down the Grade
% [734:734+554]
n4=rpm(735,2)*554;     % Number of revolutions at this load in one commute
T4=T(735,2);           % Motor torque at this load, LB-FT

% Full Acceleration
n5=sum(acc(:,4)*.05);  % Number of revolutions at this load in one run
T5=120;                % Motor torque at this load, LB-FT
```

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Gear Simulation

Written by William Sirski. This code is intended to size the gears for a 2-speed dogbox manual transmission. The intention is to use Miner's Law to calculate gear size based upon a ten-year lifespan with projected loading conditions.

```
format bank
format compact
clc
clear all
```

Loading Conditions

This portion of the code will outline the two loading regimes for the gears.

```
% Full acceleration (fun). This is a full 0-Vmax acceleration run, and I am
% assuming that this sort of fun thing happens twice a month.
run('ME32902VehicleSim.m')
close all
L1=2;                    % Runs/mo
L1=L1*12*10;             % Runs in 10-years
T1=acc(:,3)*dtire/24/dif;  % Torque transmitted by output in LBFT
R1=acc(:,2)*5280/3600;     % Speed  ft/s
R1=R1/dtire*24*2*pi*dif;   % Speed of output shaft in RPM

% Commute (not fun). This is the gear wear caused by the commute back and
% forth from Atascadero to Cal Poly. We will assume that this occurs 20
% times per month.
L2=20*12*10;             % Runs in 10-years
T2=T(:,2);               % Torque transmitted by motor in LBFT
R2=rpm(:,2).*rpm(:,3)/dif; % Speed of output shaft in RPM
i=1;
while i<=length(T')
   if 3<rpm(i,3)
      T2(i)=T2(i)*1.4;
   else
      T2(i)=T2(i)/1.4;
   end
   i=i+1;
end
T2; % Torque at output shaft in LBFT
```

```
meq =
        1611.1205801861
Batt =
       16.0387862157689
BattFS =
       1.74576801656419
sixty =
```

```
    8.18896528899922
TOP =
    93.3574113320112
```

Gears

First, some variables and constraints have to be defined.

```
Pn=20;          % Normal diametral pitch, TPI
Kd=1.5;         % Ka*Km*Ks/Kv, derating factor from Dudley 5.94
Ko=1;           % Overload factor, pends testing
Kb=1;           % Rim thickness factor
f=.5;           % Face width in IN
FS=10;          % Factor of safety
phin=20;        % Normal pressure angle, degrees
psi=20;         % Helix angle, degrees
L=10^7;         % Life for stresses
phit=atand(tand(phin)/cosd(psi)); % Tangential pressure angle, degrees
Pt=Pn*cosd(psi);% Tangential diametral pitch, TPI
J=0.5;          % Geometry factor from Shigley CONSERVATIVE from 14-7
Yn=1;           % Repeated stress factor, conservative for high-cycle,
                % from Shigley's 14-14
Kt=1;           % Temperature factor, assume T<250F
Kr=1;           % Reliability factor, assume 99% Shigley's 14-10
mg=1.4;         % Speed ratio
I=cosd(phit)*sind(phit)/2/(mg+1)*mg;% Pitting resistance factor,
                                    % Shigley's 14.23
Zn=1;           % Repeated stress factor from Shigley's CONSERVATIVE
                % for high-cycle, 14-15
St=130000;      % GRADE 2 CARBURIZED GEARS, psi
Sc=350000;      % GRADE 2 CARBURIZED GEARS, psi
Cp=2300;        % For steel gears from Shigley's 14-8
sigall=St*Yn/FS/Kt/Kr;  % Allowable bending stress, psi
sigallc=Sc*Zn/FS^.5/Kt/Kr;  % Allowable contact stress, psi

% Now for the fun part.
```

Gear Optimization

The goal is to minimize CL distance while maintaining unbroken gears. Miner's rule will be used to iterate for trips.

```
CL=5;     % Starting CL distance in inches
i=1;
minerb=0;   % Destruction from bending
minerc=0;   % Destruction from contact

while minerb<1&minerc<1
   minerb=0;   % Destruction from bending
   minerc=0;   % Destruction from contact
```

```
    dp=CL*2/1.4;
    dg=CL*2*1.4;
    % Destruction from commutes
    j=1;
    while j<=length(T2')
        Ft=T2(j)*2/dp;
        % Tangential force on teeth, LB
        sigapp=Ft*Ko*Kd*Pt/f/J*Kb;
        % Applied bending stress, psi
        minerb=minerb+1*R2(j)/L*sigapp/sigall;
        % Added destruction from 1s at this RPM
        sigappc=Cp*Ft*Ko*Kd/dp/f/I;
        % Applied contact stress, psi
        minerc=minerc+1*R2(j)/L*sigappc/sigallc;
        % Added destruction from contact
        j=j+1;
    end
    j=1;
    while j<=length(T1')
        Ft=T1(j)*2/dp;
        % Tangential force on teeth, LB
        sigapp=Ft*Ko*Kd*Pt/f/J*Kb;
        % Applied bending stress, psi
        minerb=minerb+tstep*R1(j)/L*sigapp/sigall;
        % Added destruction from tstep s at this RPM
        sigappc=Cp*Ft*Ko*Kd/dp/f/I;
        % Applied contact stress, psi
        minerc=minerc+tstep*R1(j)/L*sigappc/sigallc;
        % Added destruction from contact
        j=j+1;
    end
        CL=CL-0.001;
end

CL;
dp=CL*2/2.4/1.4;     % Pinion pitch diameter
dg=dp*1.4;           % Gear pitch diameter
dpb=dp*cosd(phit);   % Base circle diameter
dgb=dg*cosd(phit);   % Base circle diameter
ntp=round(dp*Pt)     % Number of teeth on pinion
ntg=round(dg*Pt)     % Number of teeth on gear
dp=ntp/Pt            % Actual available pinion pitch diameter
dg=ntg/Pt            % Actual available gear pitch diameter
CL=(dp+dg)/2         % Actual centerline distance, IN
efinal=dg/dp         % Actual gear ratio
```

```
ntp =
   38
ntg =
   53
dp =
        2.02193776770423
dg =
        2.82007109706117
CL =
         2.4210044323827
efinal =
        1.39473684210526
```

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```
clc; clear
```

Shaft Analysis

Author: Ernesto Huerta California Polytechnic State University, San Luis Obispo, CA
Date Created: 10/17/17 Description: This code is intended to solve for the diameters of the shafts of the two speed transmissions.

```
% Only one gear mesh is considered at a time but the radial forces are
% always considered since the grears are always in mesh. This is how we can
% consider the maximum moment.
```

Design Data

```
run('GearSim.m')
clc

% Material Properties of steel shaft
E2 = 30.0e6; % modulus of steel (psi)

% Gravity
g = 386.09;     % in/s^2
% Density
rho = .284;     % lb/in^3

% Design Stiffness Requirements
ThetaB = 0.0005; % allowable bearing rotation (rad)
DCD = 0.01; % change in center distance (in)
```

```
meq =
        1611.1205801861
Batt =
       16.0387862157689
BattFS =
       1.74576801656419
sixty =
       8.18896528899922
TOP =
       93.3574113320112
ntp =
   38
ntg =
   53
dp =
       2.02193776770423
dg =
       2.82007109706117
CL =
```

```
    2.4210044323827
efinal =
    1.39473684210526
```

Iteration Values, Shaft 2

Shaft Diameter Selection, Section Properties

```
D2 = 0.985;          % [in]
d2 = 3/16;           % [in]
I2 = pi/64*(D2^4 + d2^4);   % [in^4]

x1 = .8;       % Bearing Width approximation
x2 = 1/32;     % Space between bearing and gear
x3 = f;        % Gear Face width
x4 = 1.15;     % Space between gears for hub

% Lengths
Leg = x1/2 + x3/2 + x2;
Lgh = x3+x4;
Lhj = Leg;

L1 = Leg;
L2 = Leg + Lgh;
L3 = Leg + Lgh + Lhj;   % Length center of bearing to bearing
```

Shaft 1, Gear D (gear)

```
D1 = 1.182;

% Material Properties of cast shaft
E1 = 29.9e6; % modulus of steel (psi)

I1 = pi/64*(D1^4);
```

Shaft 2, Gear A (Gear) (output)

Gear Forces and dimensions

```
T_out = T2;
% Known reaction forces
% Forces on gear (Gear A)
WtA = T_out*2/dg;
W_totA = WtA/(cos(phin)*cos(psi));
WrA = W_totA*cos(phin);
WaA = W_totA*cosd(phin)*cosd(psi);

Hy = WrA';
Hx = WaA';
Hz = WtA';
```

```
Mhx = T_out';
Mhz = (dg/2)*Hx;
```

Shaft 2, Gear C (pinion)

```
% Forces on gear (Gear C)
WtC = T_out*2/dp;
W_totC = WtC/(cos(phin)*cos(psi));
WrC = W_totC*cos(phin);
WaC = W_totC*cosd(phin)*cosd(psi);

% Loads on the second set of gears locked in mesh.
WtB = WtC;
W_totB = W_totC;
WrB = WrC;
WaB = WaC;

By = WrB';
Bx = WaB';
Bz = WtB';
Mbx = 1.4*T_out';      % This is accounted fot in the gear ratio
Mbz = (dg/2)*Bx;

% Gy = By
% Hy = Cy
% Gz = Bz
% Hz = Bz
```

Reaction Forces on shaft 2

```
A2 = [ 1 0 0 0 0;
     0 -1 0 1 0;
     0 0 1 0 1;
     0 0 0 0 L3;
     0 0 0 L3 0];

b2 = [Hx; Hy-By; Hz+Bz; L2*Hz+Bz*L1; L2*Hy+Mhz-By*L1];

% x_2 = [Ex2, Ey2, Ez2, Jy2, Jz2]
x_2 = zeros(5,2578);
for n = 1:length(T_out)
   x_2(:,n) = A2\b2(:,n);
end

Ex2 = x_2(1,:);
Ey2 = x_2(2,:);
Ez2 = x_2(3,:);
```

```
Jy2 = x_2(4,:);
Jz2 = x_2(5,:);

% Deflection of shaft
% At first step point F
v2 = W_totA.*(L3)^3/(48*E2*I2);  % transverse deflection at gear engagement
v2_max = max(v2);

theta2 = W_totA.*(L3)^2/(16*E2*I2); % shaft rotation at the bearings
theta_max2 = max(theta2);
```

Time for Loading Consideration

Since the loading conditions are for one gear set operating for the whole ten years we can use a multiplication safety factor to increase the factor of safety to account for using both gears for the trip. We can assume that the will use the first gear set where the pinion is on the input about 60% of the time. Therefore, when this is being used the shaft will take approximately 60% of the load. Therefore, we can multiply the mean and alternating by the respective load consideration for the amount of time running.

```
% saftey factors on deformation allowable/applied
SFv = DCD/v2_max;
SFtheta = ThetaB/theta_max2; % brg rotation

% Check stresses for giggles is it stiffness driven?
M_max2 = W_totA(find(theta2==theta_max2))*(L3)/4;

kf = 2.5; % conservative fatigue stress concentration factor2
SIGEa2 = kf*M_max2*(D2/2)/I2; % bending stress
J2 = 2*I2; % polar MOI
Tau2 = kf*T_out(find(theta2==theta_max2))*(D2/2)/J2; % mean shear stress
SIGEm2 = (3^.5)*Tau2; % mises or effective mean normal stress

% max and min sigma
SIGEmin2 = .6*(SIGEm2 - SIGEa2); % max fluctuationg stress
SIGEmax2 = .6*(SIGEm2 + SIGEa2); % min fluc. stress

StressRatio = SIGEmin2/SIGEmax2; % stress ratio for Mil 5 fatiue curves

% Life for 10^6 cycles
SIGEmax_alw = 70000;    % [ksi]

% Safety factor for life
SF_life_shaft2 = 1.4*SIGEmax_alw/SIGEmax2
```

```
SF_life_shaft2 =
    2.11289352243122
```

Reaction Forces on shaft 1

```
A1 = [ 1 0 0 0 0;
     0 -1 0 1 0;
     0 0 1 0 1;
     0 0 L3 0 0;
     0 -L3 0 0 0];

b1 = [Bx; By-Hy; Bz+Hz; L1*Bz+Hz*L2; L1*By-Mbz+Hy*L2];

x_1 = zeros(5,2578);
for n = 1:length(Mbx)
    x_1(:,n) = A1\b1(:,n);
end

Dx1 = x_1(1,:);
Dy1 = x_1(2,:);
Dz1 = x_1(3,:);
Ay1 = x_1(4,:);
Az1 = x_1(5,:);

% Deflection of shaft
v1 = W_totB.*(L3)^3/(48*E1*I1);  % transverse deflection at gear engagement
v1_max = max(v1);

theta1 = W_totB.*(L3)^2/(16*E1*I1); % shaft rotation at the bearings
theta_max1 = max(theta1);

% saftey factors on deformation allowable/applied
SFv1 = DCD/v1_max;           % center distance
SFtheta1 = ThetaB/theta_max1; % brg rotation

% Check stresses
M_max1 = W_totB(find(theta1==theta_max1))*(L3)/4;

SIGEa1 = kf*M_max1*(D1/2)/I1; % bending stress
J1 = 2*I1; % polar MOI
Tau1 = kf*Mbx(find(theta1==theta_max1))*(D1/2)/J1; % mean shear stress
SIGEm1 = (3^.5)*Tau1; % mises or effective mean normal stress

% max and min sigma
SIGEmin1 = .4*(SIGEm1 - SIGEa1); % max fluctuationg stress
SIGEmax1 = .4*(SIGEm1 + SIGEa1); % min fluc. stress

StressRatio1 = SIGEmin1/SIGEmax1; % stress ratio for Mil 5 fatiue curves
```

```
SF_life_shaft1 = SIGEmax_alw/SIGEmax1
```

```
SF_life_shaft1 =

        2.79881478368156
```

Shaft Diameter Sizes

Shaft 2 (Output): 1.15 inches

```
D_shaft2 = D2
% conversion to milimeters
D2_mm = D_shaft2*25.4;
% Shaft 1 (Input ): 1.35 inches
D_shaft1 = D1
% conversion to milimeters
D1_mm = D_shaft1*25.4;
```

```
D_shaft2 =

                0.985

D_shaft1 =

                1.182
```

Critical Speed

```
W2 = L3*(pi/4)*D2^2;
W1 = L3*(pi/4)*D1^2;

Wcr2 = sqrt((48*E2*I2*g)/(W2*L3^3))
Wcr1 = sqrt((48*E1*I1*g)/(W1*L3^3))
```

```
Wcr2 =

        20245.6896413276

Wcr1 =

        24238.3952714583
```

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```
clc; clear
```

Splines

Author: Ernesto Huerta and Josh Plaskett California Polytechnic State University, San Luis Obispo, CA

Date Created: 10/17/17 Description: This code is intended to solve for the Dog Teeth on the two-speed transmission of the gears and hub for ten splines.

```
% Script to run for diameters
run('Shaft_analysis.m')
```

```
meq =
        1611.1205801861
Batt =
       16.0387862157689
BattFS =
       1.74576801656419
sixty =
       8.18896528899922
TOP =
       93.3574113320112
ntp =
   28
ntg =
   39
dp =
       1.41890369663455
dg =
       1.97633014888384
efinal =
       1.39285714285714
SF_life_shaft2 =
       1.89214835440314
SF_life_shaft1 =
       2.50984338331501
D_shaft2 =
                 0.985
D_shaft1 =
                 1.182
Wcr2 =
        18255.066721964
Wcr1 =
        21855.196377731
```

Design Data

Material type: AISI 9310 Carbonized

```
Su = 179000;       % [psi]

% Size of shaft from previous calculations.
N = 10;                % number of splines
D = D_shaft2;          % [in]
R = D/2;               % [in]

% diameter to the root
D_root = D+.01;

% spline height, look up for manufactuers handbook
h = .125;

D_pitch = D_root+2*h;       % [in]

% effective length
Le = (1/3)*D_pitch;        % [in], Machinery's Handbok

% clearance for spacing dog-teeth
c = .1;                  % [in]

% total length
Lt = Le+2*c             % [in]
```

```
Lt =

    0.615
```

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```
clc; clear
```

Dog teeth

Author: Ernesto Huerta and Josh Plaskett California Polytechnic State University, San Luis Obispo, CA

Date Created: 10/17/17 Description: This code is intended to solve for the Dog Teeth on the two speed transmission of the gears and hub

```
% files needed
run('Splines.m')
```

```
meq =
     1611.1205801861
Batt =
    16.0387862157689
BattFS =
    1.74576801656419
sixty =
    8.18896528899922
TOP =
    93.3574113320112
ntp =
   28
ntg =
   39
dp =
    1.41890369663455
dg =
    1.97633014888384
efinal =
    1.39285714285714
SF_life_shaft2 =
    1.89214835440314
SF_life_shaft1 =
    2.50984338331501
D_shaft2 =
           0.985
D_shaft1 =
           1.182
Wcr2 =
     18255.066721964
Wcr1 =
     21855.196377731
Lt =
           0.615
```

Design Data

```
% number of teeth
N = 4;
% Clearance
c_teeth = .1;

% Forceon teeth
T = 180;          % [ft*lb]
r = dp/2/12;      % [ft]
Ft = T/r ;        % [lbf]
h = 0.18;         % [in]

% Stresses for material AISI 9310 and safety factor
sf = 5;           % safety factor
Sy = 143000;      % [psi]
sigmaALW = Sy/sf;

% equation
%
% where t and b are the same for a square dog tooth

t = (((6/N)*Ft*h)/sigmaALW)^(1/3);

% Therefore, the tooth size is given by
Tooth_size = t
```

```
Tooth_size =
    0.306320587223769
```

Fluid Bearings

Table 12-5 P = 120-250 [psi]

```
D = D_shaft2;     % [in], journal bearing diameter
R = D/2;          % [in], journal bearing radius
Si = 20;
Pd = 250/20;

% clearance
% Based on 1.5 in journal rod
c = 0.00125;

% r/c ratio
ratio = R/c;

% bushing thickness
t = 0.022;        % [in]
```

```
% Table 12.4
T2 = 142;          % [deg F]
ho = 0.00065;      % [in]
delta = 0.0090;    %
Q = 153;           % [btu/s]
H = 0.008;         % [btu/s]
N = 2;

% Total flowrate
Q_tot = Q*N;

% Size outer Journal
D_outer = 2*(R+c+t);

% The Diameter of the Inner gear face
D_inner_gear_face = D_outer
```

```
D_inner_gear_face =

    1.0315
```

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ME 329 Fall 2017 - Bearing Analysis

Author: Nathaniel Furbeyre California Polytechnic State University, San Luis Obispo, CA **Date Created: 11/14/17** Description: This code is intended to solve for the estimated shaft bearing life in the two speed transmission.

```
clc; clear; clearvars
format
```

Design Data

```
Rpin = 1.015;      % pinion radius (in)
Rgear = 1.40;     % gear radius (in)

Lbb = 3.6125;      % length between centers of 2 bearings
Lb1g1 = 1.13125;   % length between bearing and closer of 2 gears for radial load sum moments
calc
Lb1g2 = Lbb - Lb1g1; %length between bearing and further of 2 gears for radial load sum moments
calc

% Factors and variables according to Shigley's Rolling Element Bearing Chapter
phi = 20;          % gear pressure angle (deg)
psi = 30;          % gear helix angle (deg)
V = 1;             % inner ring rotation factor
Xi = 0.56;         % for Fe calculation
Yi = 1.55;         % for Fe calculation

a = 3;             % exponent variable for ball bearings
```

```
X0 = .02;          % next three variables from Shigleys (Eqn 10-11)
theta = 4.441;
b = 1.439;
af = 1.5;          % application factor for a transmission
Rtot = .90;        % desired reliability for entire bearing system
Rbear = Rtot^(1/2);% single bearing reliability


% Torques / rpms retrieved from average torques of the Vehicle Simulation
Taccel = 120;       % torque from motor for full acceleration (ft-lb)
Tuphill = 95.27;    % torque from motor for uphill motion at 65 mph(ft-lb)
Tflat = 63.90;      % torque from motor for flat, 65 mph speed (ft-lb)
Tcomb = [Taccel, Tuphill, Tflat]; % Each torque in 3x1 matrix

naccel = 2.9805 * 10^5;   % revs for a commute at accel load
nuphill = 1.3714 * 10^6;  % revs for a commute at uphill load
nflat = 2.0456 * 10^6;    % revs for a commute at flat load
n_ave = .05 * naccel + .25 * nuphill + .5 * nflat; %average revs for commute

% Following numbers converted from kN to lbs from Timken Bearing Catalog
% for 6206-RS and 6305-RS deep groove bearings
C10_6206 = 19.5*224.809;   % radial/axial and radial only bearing for output shaft(lbs)
C10_6305 = 20.6*224.809;   % radial/axial and radial only bearing for input shaft(lbs)

L10 = 1000000;     % rated life for Timken bearings (revolutions)

% Estimated Weighting for Variable Loading
G1 = .3;            % estimated 30% of time in first gear
G2 = .7;            % estimated 70% of time in second gear
flat = .5;          % estimated 50% of time car going on flat roads
hill = .25;         % estimated 25% of time car going uphill
accel = .05;        % estimated 5% of time car at full acceleration
```

----- Dynamic Loading with all Modes of Motion (Full Accel, Uphill and Flat) -----

Shaft 1 Radial/Axial Loaded Bearing (6207-RS on Input Shaft)

first gear engaged - sum of moments to find bearing radial reaction

```
Fr1_1 = Lb1g1/Lbb * Tcomb/Rpin * sind(phi) * 12;
Fa1_1 = Tcomb/Rpin * sind(psi) * cosd(phi);
Fe1_1 = Xi*V*Fr1_1 + Yi*Fa1_1;
Xd1_1 = C10_6206^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fe1_1.^a);
Ld1_1 = Xd1_1 * L10;

% second gear engaged - sum of moments to find bearing radial reaction
Fr1_2 = Lb1g2/Lbb * Tcomb/Rpin * sind(phi) * 12;
Fa1_2 = Tcomb/Rpin * sind(psi) * cosd(phi);
Fe1_2 = Xi*V*Fr1_2 + Yi*Fa1_2;
Xd1_2 = C10_6206^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fe1_2.^a);
```

```
Ld1_2 = Xd1_2 * L10;

% weighted average loads, estimating 30% in first, 70% in second
B1r_ave_load = .3*Fr1_1 + .7*Fr1_2
B1a_ave_load = .3*Fa1_1 + .7*Fa1_2
B1e_ave_load = .3*Fe1_1 + .7*Fe1_2
```

```
B1r_ave_load =

  278.8816   221.4088   148.5045


B1a_ave_load =

   55.5483    44.1007    29.5795


B1e_ave_load =

  242.2736   192.3451   129.0107
```

Shaft 1 Radial Loaded Only Bearing (6207-RS on Input Shaft)

first gear engaged

```
Fr2_1 = Lb1g2/Lbb * Tcomb/Rpin * sind(phi) * 12 ;
Xd2_1 = C10_6206^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fr2_1.^a);
Ld2_1 = Xd2_1 * L10;

% second gear engaged
Fr2_2 = Lb1g1/Lbb * Tcomb/Rpin * sind(phi) * 12;
Xd2_2 = C10_6206^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fr2_2.^a);
Ld2_2 = Xd2_2 * L10;

% weighted average loads, estimating 30% in first, 70% in second
B2r_ave_load = .3*Fr2_1 + .7*Fr2_2
```

```
B2r_ave_load =

  206.3489   163.8238   109.8808
```

Shaft 2 Radial/Axial Loaded Bearing (6306-RS on Output Shaft)

first gear engaged

```
Fr3_1 = Lb1g2/Lbb * Tcomb/Rpin * sind(phi) * 12;
Fa3_1 = Tcomb/Rpin * sind(psi);
Fe3_1 = Xi*V*Fr3_1 + Yi*Fa3_1;
Xd3_1 = C10_6305^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fe3_1.^a);
Ld3_1 = Xd3_1 * L10;

% second gear engaged
Fr3_2 = Lb1g1/Lbb * Tcomb/Rpin * sind(phi) * 12;
Fa3_2 = Tcomb/Rpin * sind(psi);
Fe3_2 = Xi*V*Fr3_2 + Yi*Fa3_2;
Xd3_2 = C10_6305^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fe3_2.^a);
Ld3_2 = Xd3_2 * L10;

% weighted average loads, estimating 30% in first, 70% in second
B3r_ave_load = .3*Fr3_1 + .7*Fr3_2
B3a_ave_load = .3*Fa3_1 + .7*Fa3_2
B3e_ave_load = .3*Fe3_1 + .7*Fe3_2
```

```
B3r_ave_load =

  206.3489   163.8238   109.8808


B3a_ave_load =

   59.1133    46.9310    31.4778


B3e_ave_load =

  207.1810   164.4845   110.3239
```

Shaft 2 Radial Loaded Only Bearing (6306-RS on Output Shaft)

first gear engaged

```
Fr4_1 = Lb1g1/Lbb * Tcomb/Rpin * sind(phi) * 12;
Xd4_1 = C10_6305^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fr4_1.^a);
Ld4_1 = Xd4_1 * L10;

% second gear engaged
Fr4_2 = Lb1g2/Lbb * Tcomb/Rpin * sind(phi) * 12 ;
Xd4_2 = C10_6305^a * (X0 + (theta-X0) * (1-Rbear)^(1/b)) ./ ((af)^a * Fr4_2.^a);
Ld4_2 = Xd4_2 * L10;

% weighted average loads, estimating 30% in first, 70% in second
B4r_ave_load = .3*Fr4_1 + .7*Fr4_2
```

```
B4r_ave_load =

  278.8816   221.4088   148.5045
```

------------- Variable Loading (3 levels) ---------------

active gear engagement: 20/40 miles flat, 10/40 miles uphill, 10/40 miles   no power transmission (need to find which bearing fails first) see above design data section for estimates on percentages for elapsed \   time on variable loads

```
% Bearing constants for damage to failure as per Eqn 11-1 in Shigley's
K1 = Fe1_1(1)^a * Ld1_1(1);
K2 = Fr2_1(1)^a * Ld2_1(1);
K3 = Fe3_1(1)^a * Ld3_1(1);
K4 = Fr4_1(1)^a * Ld4_1(1);

% Equivalent loads on each bearing. The 6 terms come from acceleration,
% uphill and flat conditions for each gear. See design data for values.
B1_Feq = (G1 * (accel*Fe1_1(1)^a + hill*Fe1_1(2)^a + flat*Fe1_1(3)^a) + ...
    G2 * (accel*Fe1_2(1)^a + hill*Fe1_2(2)^a + flat*Fe1_2(3)^a)) ^ (1/3);
B2_Feq = (G1 * (accel*Fr2_1(1)^a + hill*Fr2_1(2)^a + flat*Fr2_1(3)^a) + ...
    G2 * (accel*Fr2_2(1)^a + hill*Fr2_2(2)^a + flat*Fr2_2(3)^a)) ^ (1/3);
B3_Feq = (G1 * (accel*Fe3_1(1)^a + hill*Fe3_1(2)^a + flat*Fe3_1(3)^a) + ...
    G2 * (accel*Fe3_2(1)^a + hill*Fe3_2(2)^a + flat*Fe3_2(3)^a)) ^ (1/3);
B4_Feq = (G1 * (accel*Fr4_1(1)^a + hill*Fr4_1(2)^a + flat*Fr4_1(3)^a) + ...
    G2 * (accel*Fr4_2(1)^a + hill*Fr4_2(2)^a + flat*Fr4_2(3)^a)) ^ (1/3);

% Bearing life for the variable loading experienced by the car in revs


B1_Leq_rev = K1 / B1_Feq^a;
B2_Leq_rev = K2 / B2_Feq^a;
B3_Leq_rev = K3 / B3_Feq^a;
B4_Leq_rev = K4 / B4_Feq^a;

% Bearing life for the variable loading experienced by the car in commutes
format long g
B_Leq_com = [B1_Leq_rev B2_Leq_rev B3_Leq_rev B4_Leq_rev] / n_ave

L_worst = min([B1_Leq_rev, B2_Leq_rev, B3_Leq_rev, B4_Leq_rev]) % revolutions
fail_com = L_worst / n_ave  % number of commutes until failure
fos = fail_com / 1825 % factor of safety for commute every day for 10 years
```

```
B_Leq_com =

  Columns 1 through 3

          2670.58505206862          3092.94751905362          4788.67179854759

  Column 4
```

```
          1833.85150192294


L_worst =

          2531728275.60847


fail_com =

          1833.85150192294


fos =

          1.00485013803997
```

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