University of Victoria
Department of Mechanical Engineering
MECH 360

Final Report: Custom Electric Drive Unit

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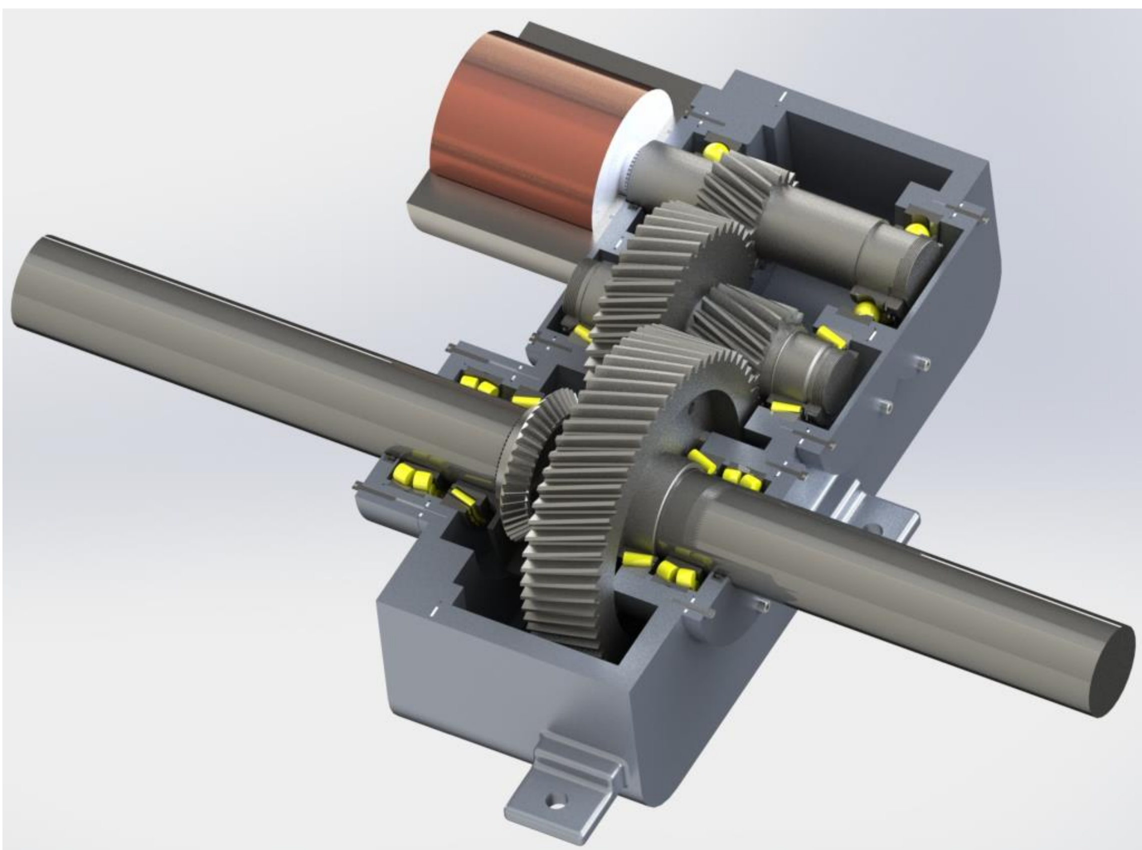

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Contents

Figures

Tables

1.0 Introduction

Electric vehicles have recently been gaining popularity as an efficient and environmentally friendly alternative to internal combustion vehicles. This report outlines the design of a single axle electric drive unit for use in midsize sedans. For this report, the Nissan Altima midsize sedan has been chosen as the specific design recipient.

1.1 Objectives

In designing this system, there are a number of parameters which have been defined. The unit should provide comparable performance to the equivalent internal combustion vehicle. In addition, the axle drive unit should have a minimum lifetime of 12 years of average driving. Assembly of the gearbox should be reliable and accurate.

1.2 Constraints

The drive unit should not be larger than 1320x1397mm, which is the average size of the engine and transmission in most midsize sedans. The drive unit should be a fully contained unit complete with electronics and mounting features for the axles and the frame.

2.0 Defining the Design

Before a detailed design could begin, several parameters had to be selected. After investigating a number of commercially available electric motor options, the first-generation Nissan Leaf electric motor was chosen. This motor has numerous valuable features. It provides a proven electric motor which is widely available for production units. The main parameters of this motor are outlined below in Table 1.

Table 1: Nissan Leaf motor specifications [1]

Nissan Leaf Motor Key Specifications	
Maximum Torque	280 Nm
Top speed	10390 RPM
Power	80 kW

2.1 Choosing the Optimal Gear Ratio

The gear ratio is the parameter that has the greatest impact on the efficiency of the drive unit. In order to determine the optimal gear ratio, a MATLAB model of the system was developed. The key physical parameters of the vehicle dynamics were modelled. In order to develop an accurate representation of average driving, the United States' Environmental Protection Agency (EPA) standard driving cycle was used [2]. Using the velocity and time information, the required forces and torques from the motor could be determined with a given gear ratio. There is a large range of acceptable gear ratios, so the most efficient configuration was sought. Using the published efficiency data of the Nissan Leaf motor and inverter system [3] coupled with the required torques and speeds from the EPA cycle, an optimization was run on the gear ratio to determine the most efficient ratio for operation. From this optimization, an overall gear ratio of 7.872:1 was determined to be ideal. Figure 1 shows a heat map of the drive unit operational torque and speed over the EPA drive cycle.

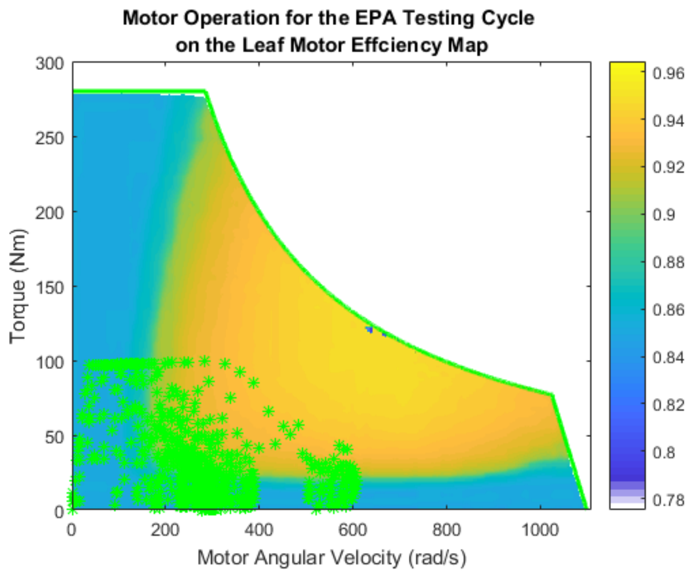

Figure 1: Axle Drive Unit EPA drive cycle heat map superimposed on the system efficiency map

The implementation of this optimization can be found in Appendix F.

2.2 General Layout

With the overall ratio of 7.872:1 determined, research was performed to determine the appropriate number of shafts and intermediate gear ratios. It is recommended for ratios between 5:1 and 25:1 that a two-stage reduction is used [4]. Based on the loads experienced by the gears, a parallel axle gearbox would be preferred to a planetary arrangement. The most common pressure angle is 20°. In order to prevent undercutting, a minimum of 17 teeth are required on a gear with pressure angle 20°. Using this constraint, the other two gears were determined to require 35 and 65 teeth respectively for an overall ratio of 7.872:1. While this constraint gave the maximum possible efficiency, it was determined that the center distance of the bearings required a larger diameter between the first two shafts. For this reason, the ratios were reconfigured to be 17:40 and 18:60, which retained a very similar efficiency and enables a simpler gearbox geometry. The geometric data for the chosen gears is presented in Table 2.

Table 2: Selected Gear Properties. See Appendix B for complete data

	Gear 1	**Gear 2**	**Gear 3**	**Gear 4**
Number of Teeth	17	40	18	60
Module (mm)	4	4	4	4
Diameter (mm)	72.36	170.27	76.62	255.4
Pressure Angle (°)	20	20	20	20
Helix Angle (°)	20	20	20	20
Face Width (mm)	52	52	64	64

Figure 2 shows the complete assembly section view of the gearbox. The red outlined part is the main housing and the pink is the main housing cover. These sections are both from a single cast part.

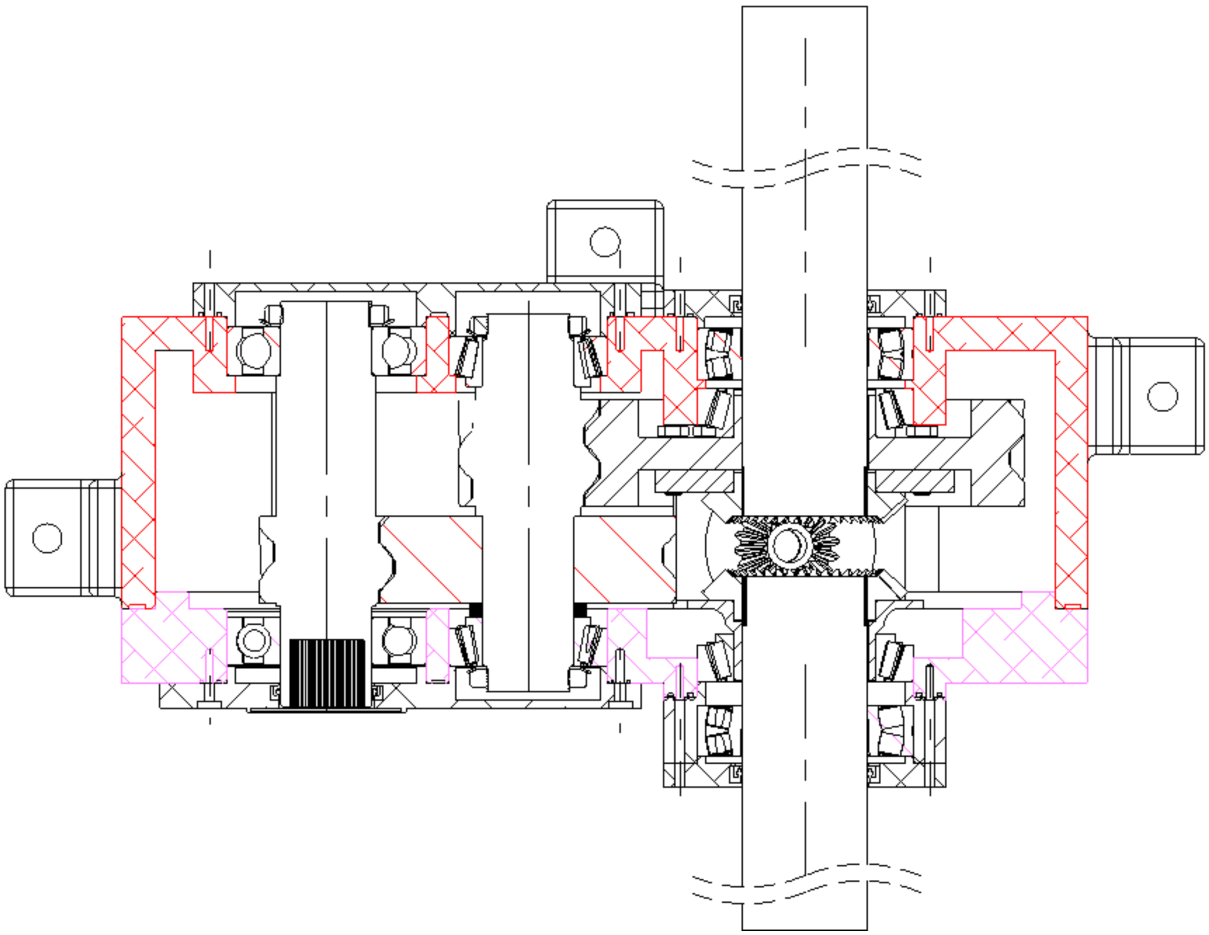

Figure 2: Section View of Gearbox Assembly

The complete assembly has four gears, four shafts and eight bearings. The details and specifications will be found in the later sections of this report.

2.3 Design Process

The gearbox design provided an ideal challenge where constant iteration was required to achieve a viable product. One of the primary lessons learned involved defining repeatable test procedures. When automating a calculation, such as a gear stress or shaft bending moment diagram, multiple hand calculations were done to verify that the implementation was correct. Generating MATLAB models of all of the primary components of the design: gears, shafts, bearings, and fasteners allowed iteration to occur rapidly. Various parameters such as tooth pressure angle could be varied, and all of the forces and loads would be re-calculated for each part. Wherever possible, all of the relevant calculations were performed before the Solidworks CAD design began. This ensured that the work was relevant and would be accurate. The

MATLAB model is more flexible than a CAD model, so it was important to refrain from CAD until the design was nearly finalized. This helped minimize any redundancies in the work.

3.0 Shaft Design

For a two-stage gearbox, three main shaft designs were needed: an input shaft to connect to the Nissan Leaf electric motor to the gearbox, an intermediate shaft to reduce speed and increase torque, and two drive shafts to output the final torque and speed to the front wheels. The material used for all the gears is AISI 4023 normalized alloy steel because of its high hardness and better machinability compared to the other choice from an ASM report on gear material choice [5]. Because two pinions will be machined onto the shafts, the shaft material must also be AISI 4023. The material properties, as well as the endurance strength for each shaft is shown in Appendix A. The regions where gears are machined into the shafts will be carburized for increased hardness. The regions where the bearings sit must not be carburized to maintain machining precision. A nominal, minimum safety factor of 2.0 was used in the design of the shafts. In addition, the ability to assemble the gearbox was considered in all of the designs. Most safety factors are greater than two because the shafts were increased to the nearest available bearing size. The design of all the gears and bearings for each shaft will be mentioned, but the design of each is explained in detail in their respective section (‘Gear Design’ and ‘Bearings’ sections).

3.1 Input Shaft

The input shaft is 243mm long with a diameter of 55mm. A 17-tooth pinion gear is machined onto the shaft and mates with the gear of intermediate shaft. The machined pinion has a fillet radius of 1mm (callout A of Figure 3) between the pinion body and shaft. This results in a small stress concentration due to the fillet radius. The safety factor at this point was found to be 6.53. This calculation can be found in Appendix D. The Nissan Leaf motor has a 41mm diameter, 40 tooth male spline of 40mm length machined onto the end of the rotor shaft. To transmit the torque from the motor, the input shaft has a 40-tooth female spline with an inner diameter of 41mm and a length of 40 mm machined into one end of the shaft (Callout B of Figure 3). The thickness of the input shaft at the female spline must be optimized for the shear stress resulting from the torsion applied by the motor. To be safe, the outer diameter of the first shaft is 55mm, resulting in a wall thickness of 5mm at the female spline. The shear stress in the female spline is calculated to be 19.28MPa, resulting in a safety factor of 21.52, establishing that it is in fact not a determining factor of the design.

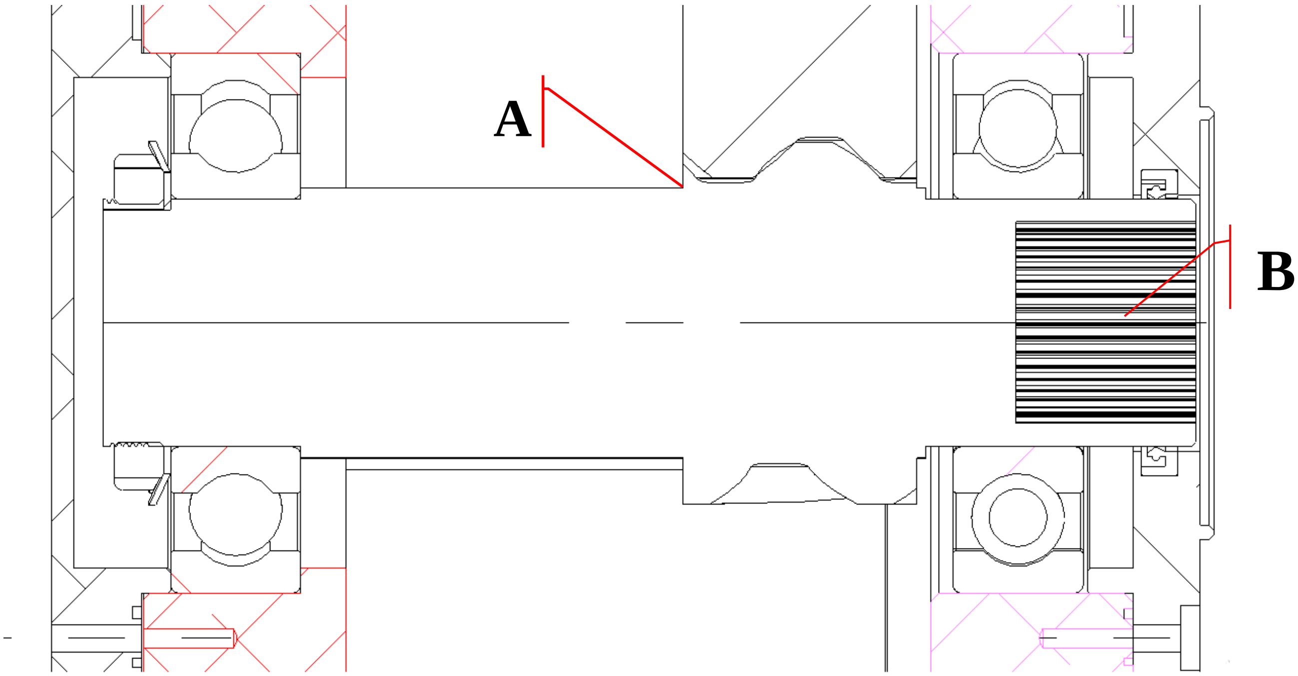

Figure 3: Input Shaft Arrangement

The shaft is supported by two SKF 6311M deep groove ball bearings. The ball bearing shown on the left side of Figure 3 is used to axially locate the position of the shaft with locating features and supports the full axial load. The left bearing is axially unconstrained to account for any changes in temperature. The shaft is constrained to the bearings using a SKF KM 11 lock nut and washer [6].

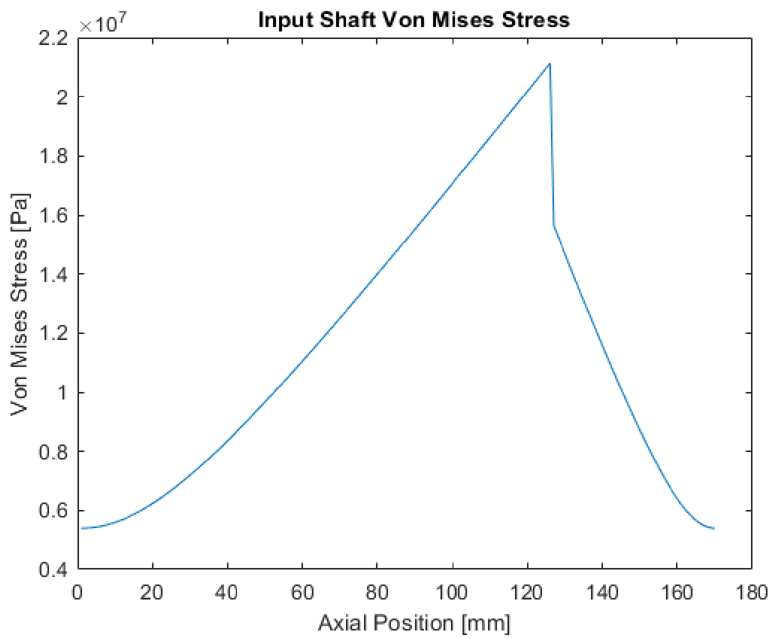

Figure 4: Input Shaft Nominal von Mises Stress

From Figure 4, the maximum nominal von Mises stress in the shaft is 21.14MPa, resulting in a safety factor of 6.199. The maximum stresses at the points A and B respectively were 25.3MPa and 12.55MPa. This gave safety factors of 6.53 and 9.07. These calculations are available in Appendix D.

3.2 Intermediate Shaft

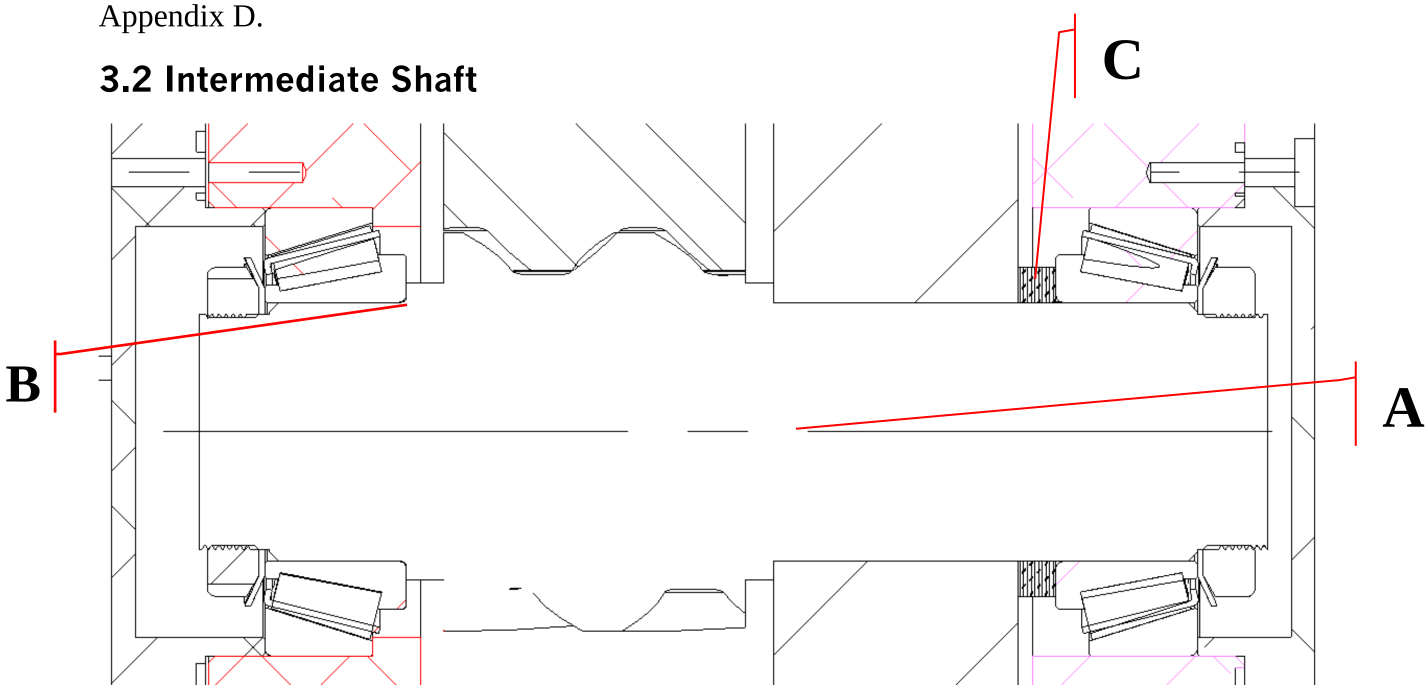

Figure 5: Intermediate Shaft Arrangement

The 55mm diameter intermediate shaft is 212mm long and supports a pinion that is machined into the shaft and a keyed gear blank. The keyed gear meshes with the pinion of the input shaft and is constrained to the shaft with a 16x10mm parallel key. The location of the intermediate shaft is constrained by a pair of tapered roller bearings secured axially by locating features in the housing. A shim (Figure 5 Callout C) presses the gear blank against the shaft shoulder. This shim is compressed by the SKF KM 10 lock nut [7] on the shaft. Hand fatigue calculations can be found Appendix D. The stresses at point A where the keyway interfaces with the shaft step, were $\sigma'_a = 77.24$MPa and $\sigma'_m = 4.08$MPa. This leads to a safety factor of 1.45. It would be ideal to improve this safety factor. Some possible solutions would be to move the keyway away from the shaft step by using a smaller key. Callout B is the step in the shaft at the bearing. At this point, $\sigma'_a = 32.7$MPa and $\sigma'_m = 0.5$MPa. This results in an acceptable safety factor of 3.45.

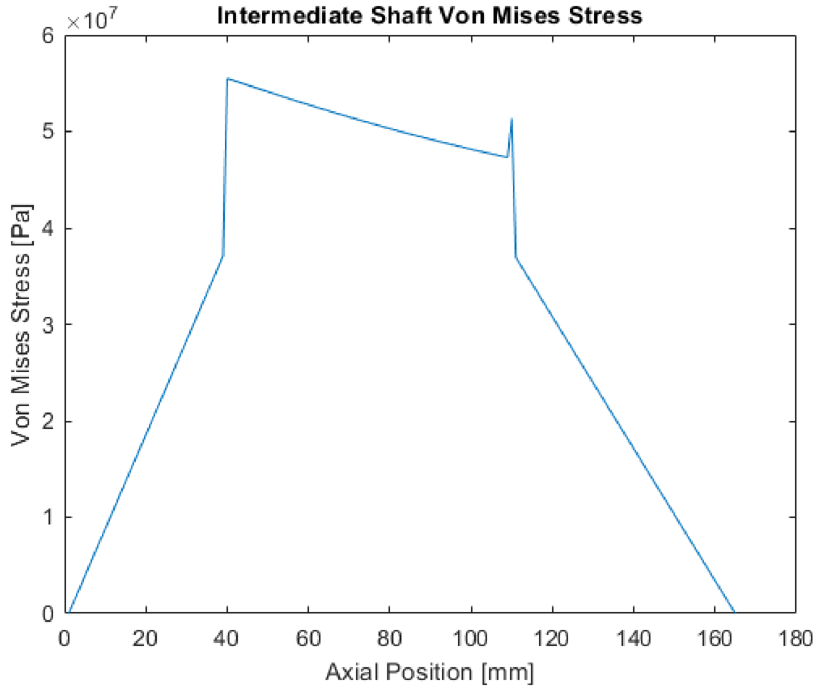

Figure 6: Intermediate Shaft von Mises Stress

From the design calculations in Appendix D and the graph in Figure 6, the maximum nominal von Mises stress of the intermediate shaft is 55.51 MPa, resulting in a safety factor of 2.36.

3.3 Output/Drive Shafts

The output shaft has a gear that is bolted directly onto the differential. The differential is supported by the housing through two tapered roller bearings. The housing of the differential supports the bending and axial loads caused by the gear. The differential bearings and housing are designed to provide clearance for the two drive shafts which connect the gearbox to the wheels. The gear is hollowed to save weight and partially enclose the differential. The gear bore was specified to ensure that the rim factor remains 1. The housing is made of cast iron. The spider gears will be investigated in the future. Since the housing supports the axial and bending loads, the drive shafts simply experience the torque from the gear. The axles are 75mm thick to withstand the maximum torque available from the motor. The end of the drive shafts is splined to connect to the bevel gears in the differential. This method was chosen for ease of assembly and to meet the industry standard.

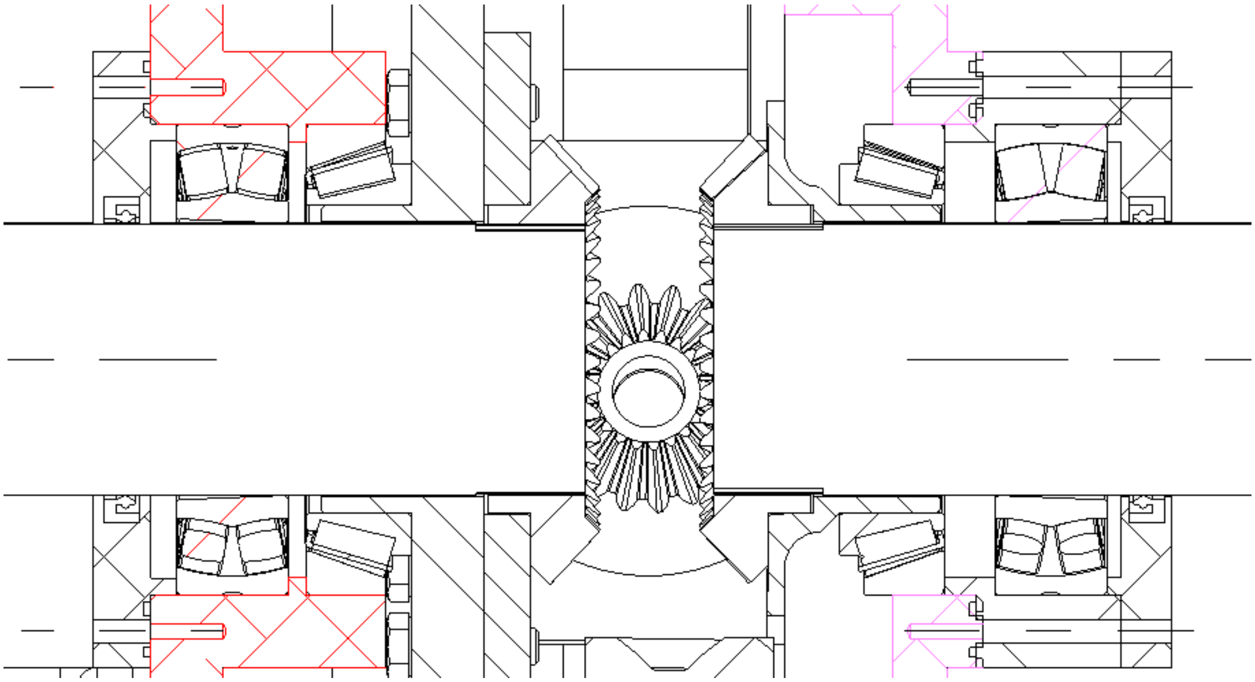

Figure 7: Output Shaft Arrangement

The maximum nominal bending moment is determined to be 552.8 Nm. Figure 8 shows a plot of the equivalent moment over the length of the differential.

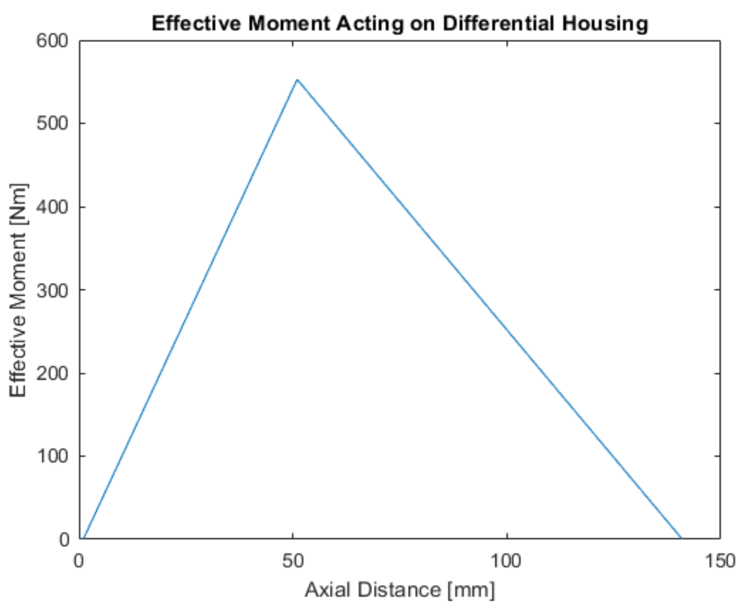

Figure 8: Effective Moment Acting on Differential Housing

Given the complex geometry of the part, hand stress calculations were infeasible. Accordingly, preliminary Solidworks FEA was performed on the housing to evaluate its strength.

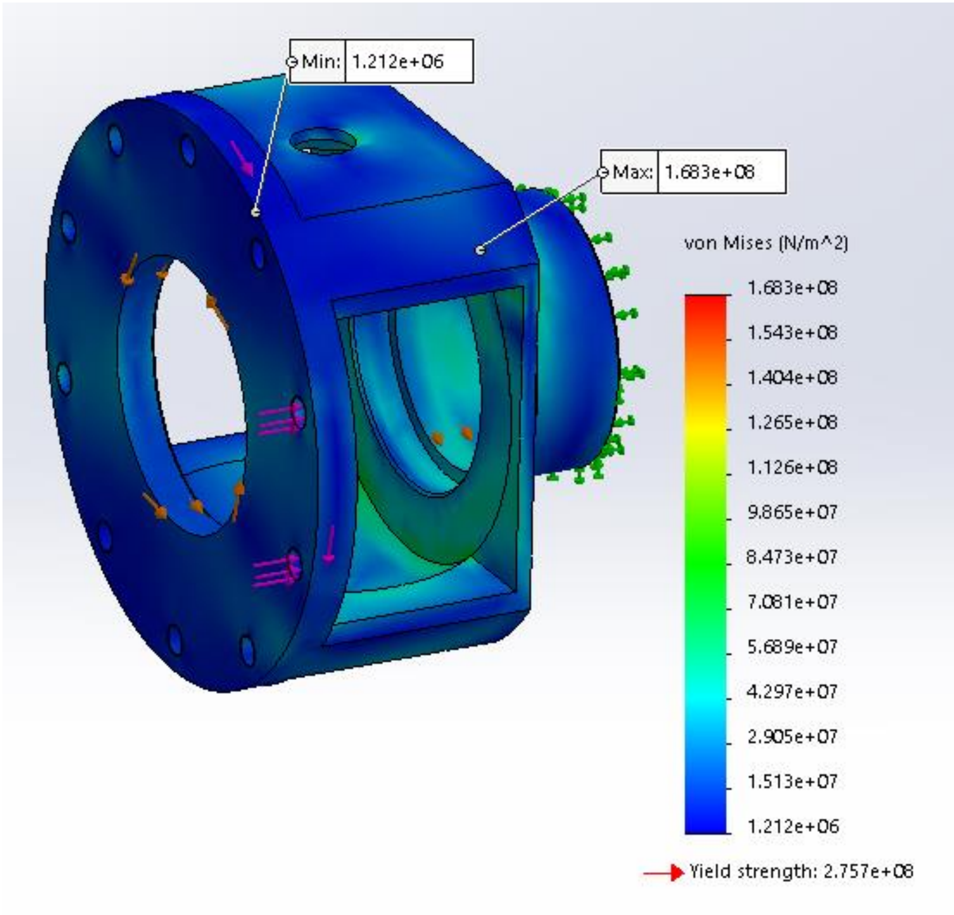

Figure 9: Differential Housing Finite Element Analysis

The housing requires some further optimization to reduce stress concentrations in the corners, but currently can withstand the applied loads and torques from the drive system. Solidworks estimates a safety factor of 1.64. In addition, given the complex nature of the loading of this part, we recommend actual lifecycle testing on a prototype to obtain a more realistic lifetime prediction. The 10mm wall thickness appears to be a safe design choice.

4.0 Gear Design

Research showed, that for the gear design, with an overall gear ratio of 7.843:1, a two-stage gearbox was the best choice [4]. Helical gears are advantageous in automotive applications due to their increased durability and quieter operation. For this reason, all of the gears in the drive unit are helical.

Since the diameters of the two pinions are very similar to the shaft diameters, the pinions will be machined directly onto the shafts. Therefore, it is crucial to choose a high strength material which can be easily machined before it is hardened. The material of the gear blanks and the gears machined onto the shafts must remain the same, as mismatched gear hardness will dramatically increase the likelihood of surface failure of the gear teeth. As detailed in Appendix A, the material is AISI 4023 steel. Appendix B contains all the specific data for each gear.

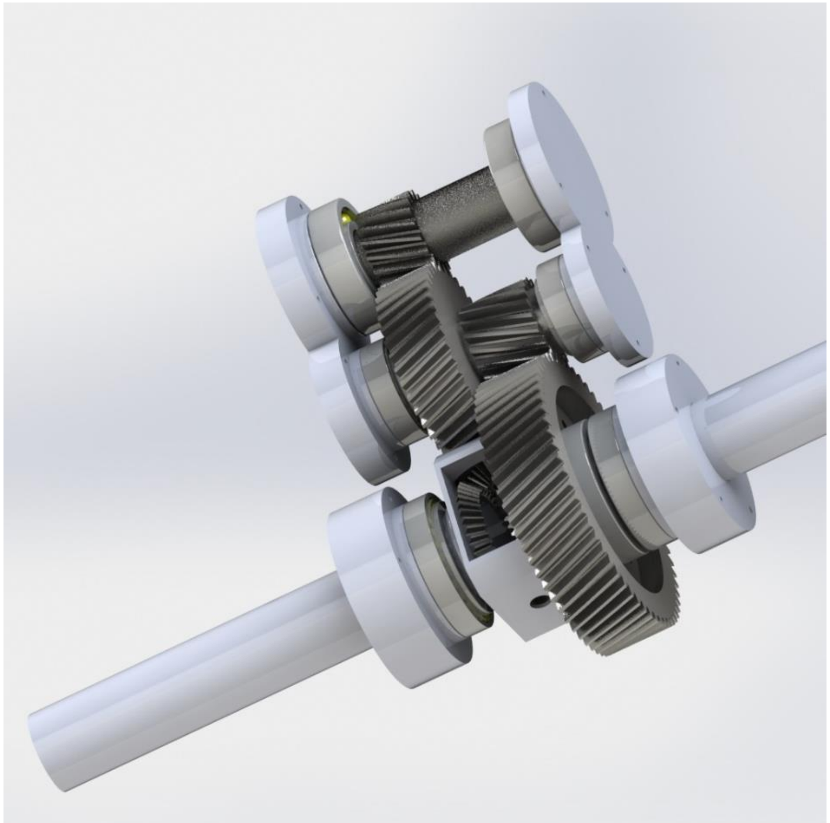

Figure 10: Shaft and Gear Render

4.1 Primary Stage

The first set of gears connecting the input shaft and the intermediate shaft have 17 and 40 teeth, respectively. The pinion is machined into the input shaft, and the gear on the intermediate shaft is a gear blank with a 16x10mm parallel key. [8] The key material is AISI 1020 cold rolled steel. The key has a safety factor of 8.97. These calculations are available in Appendix D.

From MATLAB, the bending stress on the input shaft pinion is 198.9MPa and the stress on the intermediate shaft gear is 168.3MPa, with safety factors of 1.74 and 2.06 respectively. The pitting surface stress was also considered. The surface stress between the two gears is 941.8MPa with a safety factor of 1.76. In addition, to ensure the quiet and smooth operation a contact ratio greater than or equal to 1.5 is desirable. The axial contact ratio of the primary stage is 1.506, which is ideal for the system. A sample calculation of these stresses is available in Appendix D.

4.2 Secondary Stage

The secondary stage of the gearbox is similar to the primary stage; with the pinion being machined into the shaft, and the gear being a gear blank bolted to the differential on the drive shaft. The number of teeth and gear ratio for the secondary stage is 18 for the pinion and 60 for the gear. Finally, it was determined that the bending stresses on the primary and secondary gears are 262.6MPa and 215.6MPa with safety factors of 1.32 and 1.6 respectively. To reduce the surface stresses, the face width of the gears was set to the maximum bound of 16 times the module [9], resulting in a face width of 64mm. The surface stress between the two gears is 1108.2MPa with a safety factor of 1.27. While these safety factors are lower than the value of 2 used for the shaft design, these are valid based on our operational duty cycle. This safety factor is based on the car operating at the absolute maximum torque for the entire operational lifecycle of the vehicle which is not the intended operating condition. The axial contact ratio of the second pair is 1.85 which is well above the recommended value of 1.5 but still less than 2 where indeterminate loads become a problem.

5.0 Bearings

The specifications of the bearings chosen for the design of the gearbox are shown in Table 3.

Table 3: List of Bearings

	Bearing 1	**Bearing 2**	**Bearing 3**	**Bearing 4**	**Bearing 5**	**Bearing 6**
Shaft #	1	1	2	2	3	3
SKF Model #	6311 M	6311 M	33311	33311	32017 X	32017 X
C (kN)	74.1	74.1	136	136	171	171
P (kN)	2.29	6.41	7.70	8.63	6.38	11.06
N (RPM)	10500	10500	4462.5	4462.5	1339	1339

C is the rated load for 1 million cycles, P is the equivalent load, N is the shaft rotational shaft

5.1 Equivalent Bearing Loads

Bearings are an essential part of machine design; as they are used to maintain relative motion between the shafts and gears of the gearbox. There are two critical design constraints that must be met to ensure safe operation and an adequate bearing lifetime. The rotational speed and reaction forces within the bearing limit the available bearing options. Due to the use of helical gears, the axial load on the bearings is an important consideration, eliminating the use of cylindrical roller bearings, which cannot support an axial load.

The input shaft is supported by two 55mm SKF 6311M deep-groove ball bearings [10]. The choice of bearings is limited by the maximum shaft speed of 10,500 RPM and the requirement to handle axial loads. The 10,500 RPM top speed is not suitable for most bearings other than deep groove ball bearings. The force exerted by the first gear is expressed in Cartesian coordinates as:

$$\mathbf{F_{G1}} = [2.815, -7.734, 2.815] \text{ kN}$$

Bearing 1 supports a radial load of 4.15kN and an axial load of 2.82kN, while bearing 2 supports a radial load of 4.1kN and an axial load of 0kN. The reaction forces of the two bearings are expressed as:

$$\mathbf{F_B} = [F_r, F_a] \text{ kN}$$

$$\mathbf{F_{B1}} = [4.15, 2.82] \text{ kN}$$

$\mathbf{F_{B2}}$ = [4.1, 0] kN

The equivalent dynamic load (P) of the bearing 1 is found to be 6.41 kN using equation 1, an axial load bearing factor (Y) of 1.45, and a radial load bearing factor (X) of 0.56 [10]. The right bearing has an equivalent load of 2.29kN.

$$P = XF_r + YF_a \qquad \text{Eq. (1)}$$

The intermediate shaft rotates slower than the input shaft at a maximum speed of 4,462 rpm. The reduced speed allows for tapered roller bearings to be used. Two 55mm SKF 33111 single-row tapered roller bearings [11] will be installed to support the intermediate shaft. The intermediate shaft consists of two gears whose forces are represented as:

$\mathbf{F_{G2}}$ = [6.26, -17.197, -6.26] kN

$\mathbf{F_{G3}}$ = [-2.817, 7.739, 2.817] kN

Bearing 3 supports a radial load of 12.37kN and an axial load of 1.72kN, while bearing 4 supports a radial load of 14.70kN and an axial load of 1.72kN. Referencing the free-body diagrams and force analysis of Appendix D, the reaction forces in the bearings are determined to be:

$\mathbf{F_{B3}}$ = [15.70, 4.432] kN

$\mathbf{F_{B4}}$ = [-9.233, 6.285] kN

Using an axial and radial bearing load factor, of 1.6 and 0.4, respectively. Equation 1 determines that the equivalent dynamic load of bearing 3 is 7.70 kN and bearing 4 is 8.63kN.

$\mathbf{F_{G4}}$ **=** [6.035, -16.58, 6.035] kN

There are four bearings inside the gearbox that are located on the output shaft, two are used to support the differential assembly, and the other two support the load of the drive shafts. The 2 SKF 32017 X tapered roller bearings [12] support the differential housing and the radial and axial loads caused by gear 4. The two SKF 22215 E double spherical roller bearings support the drive shafts [13]. Due to the unknown loads due to connecting the drive shafts to the gearbox spherical roller bearings were chosen based on their ability to handle the external moments caused by loads outside the gearbox. Bearing 5 supports a radial load of 5.38kN and an axial load of 3.13kN, while bearing 6 supports a radial load of 17.09kN and an axial load of 3.13kN.

$\mathbf{F_{B5}}$ = [11.92, 6.035] kN

$\mathbf{F_{B6}}$ = [4.66, 6.025] kN

Using the equivalent load equation, the equivalent load for bearing 5 is determined to be 6.38kN and 11.06 kN for bearing 6 with X = 0.56 and Y = 1.35

5.2 Bearing Lifetime

To determine the lifetime of the bearings subjected to equivalent loads, the L_{10} and L_{10h} lifetimes are calculated using equations 2 and 3, respectively. The L_{10} lifetime expresses the amount of cycles required for 90% of a sample of identical bearings to fail from fatigue.

$$L_{10} = \left(\frac{C}{P}\right)^p \qquad \text{Eq. (2)}$$

$$L_{10h} = \frac{10^6}{60N}\left(\frac{C}{P}\right)^p \qquad \text{Eq. (3)}$$

Where p is 3 for ball bearings and 10/3 for roller bearings. The L_{10} and L_{10h} values for each bearing are shown in Table 4.

Table 4. Bearing lifetimes

	Bearing 1	**Bearing 2**	**Bearing 3**	**Bearing 4**	**Bearing 5**	**Bearing 6**
L_{10} (millions of cycles)	1547	33731	14332	9798	57747	9203
L_{10h} (hours)	2455	53542	53528	36595	718927	114580

Based on the duty cycle of the gearbox, the minimum revolutions (in millions) for the shafts over 12 years are: 666, 283, and 85 for shafts 1-3, respectively. The lifetime ratings exceed the minimum lifetime requirement of 12 years. The maximum Weibull distribution factor (K_R) is 0.44 for bearing 3, resulting in a maximum probability of failure of 3% over the 12-year design life. Appendix C contains detailed summary information on all eight bearings.

6. Lubrication, Seals, and Gaskets

High speed gearboxes such as this require lubrication to minimize friction, increase lifetime and help distribute heat. The various aspects of delivering this lubrication as well as containing the system will be discussed in this section.

6.1 Lubrication

The lifetime of bearings and gears strongly depends on the method of lubrication and the type of lubricant used. The ideal lubricant will eliminate all friction between the bearings and their inner and outer races. Contrary to the splash oil lubrication method used in internal combustion engines, it is recommended to use a spray oil lubrication system for gearboxes that are designed for electric motors. The oil splash method is advantageous for internal combustion engines, as there is minimal torque transmitted through the shafts when they are not rotating or rotating slowly. However, for an electric motor, the highest torque in the shafts occur when the shafts rotate slowly, generating the largest loads in the bearings and gears. To negate the high loading for small shaft rotational speeds, a precisely aimed spray of oil is pumped directly into the gearbox, covering the bearings and gears. The oil splash method also causes churning within the gearbox, reducing the efficiency of the torque transmitted through the gears, an oil spray system does not have this caveat. Another important consideration for electric vehicles is that regenerative braking increases the fluctuating load, creating high load reversals which must be accounted for.

Three bearings are of main concern, the high-speed bearing sustaining the axial load on the input shaft and the bearings on the intermediate and drive shaft sustaining the axial load, resulting in high magnitude forces. The high speed of the input shaft generates significant heat due to friction and must be continuously cooled. In order to achieve a constant flow rate of oil and to combat the high heat generation and friction, a forced lubrication system is to be installed. Using equation 4, the minimum flow rate for each bearing is calculated.

$$Q = \frac{0.19E-5}{T_{in}-T_{out}}\, d\, \mu\, N\, F \qquad \text{Eq. (4)}$$

The minimum flow rate required for the oil spray system is 0.792 LPM, constrained by the flow rate requirement of the intermediate bearing subjected to the axial load.

A 1/8” NPT threaded hole has been machined into the gearbox casing, allowing the gearbox to be outfitted with a 120° stainless steel full-cone spray nozzle [14], providing 0.792 LPM of oil at 40 psi.

6.2 Seals and Gaskets

Seals are necessary for the lifetime of the gearbox, as seals are used to contain all the lubricant within the gearbox, and to keep any dirt that could damage the gears and bearings out of the gearbox. Seals were only needed at the input and outputs of the gearbox, which are the only sections where oil or dirt could enter or exit the housing. The seals chosen for the gearbox are SKF standard dynamic seals; the seals used for the input shaft and each output shaft are HMS5 V seals, size 55”x68”x8” and 75”x90”x10” respectively. These dynamic seals are optimally fitted to the shafts and are capable of expansion and contraction due to thermal changes in the gearbox.

The gaskets between the main body and lid of the housing cases and the end caps for the shafts prevent leakage between two joined parts; gaskets allow for machined parts to have larger tolerances when machining the mating features, as the gaskets prevent oil leakage even with imperfect mating surfaces. All the gaskets for the housing body and lid, the input/intermediate shaft end caps, and the drive shaft end caps are specified in engineering drawings in Appendix E. The material used for each gasket is multi-layered steel due to its low clamp load to seal, high bolt load retention because of the rigidity of the steel, and the high-pressure sealing capability (up to 3625.94 psi).

7. Housing and Fasteners

7.1 Housing Design

The housing of the gearbox is designed with few key features to ensure a consistent and repeatable assembly procedure. The housing consists of two main structural pieces and five outer caps. The main housing parts are designed with three shaft openings with the shoulders for the bearings to be located. The bearing placed on the shoulder on the shaft opening constrains the concentricity and the axial position. To clamp the bearings, the shoulders are only on one side of the main housing parts. All five enclosure parts have extruded lips to press on to the bearings. These have been specifically designed to ensure that they only contact the outer race, leaving the inner race clear. All mating locating features are chamfered on the lips to ensure smooth mating. The housing parts will be made from cast aluminum; this decision is primarily to save weight. Some surfaces, such as the bearing seats and flat mating faces will require machining after being cast. In addition, to prevent oil from leaking out of the gearbox, recessed gaskets have been designed to seal all mating flat faces. Three bolt flanges are used to mount the system to the car chassis. The location of these flanges is arbitrary and would be based on the vehicle that the system is mounted in.

To assemble the gearbox, each shaft has its bearings, gear blank, key and shim pre-assembled, as well as the differential already bolted to the final gear blank, and the seals being installed in their respective locations. The input shaft and intermediate shaft are then placed into the main body of the housing, with the lock nut and lock washer holding each shaft in place, and one output shaft is assembled through the main body and splined into the differential. The lid of the body with the gasket assembled is then partially installed, leaving room for any final adjustments or alignment, and the second drive shaft is splined into the differential. The housing lid is then bolted to the main body, and finally the end caps with gaskets are bolted onto the housing, and the motor is installed into the input.

7.2 Fasteners

The minimum engagement length for the M6 bolts is calculated to be 4.76mm using equation 5. Where the tensile area (A_t) is 20mm^2, the diameter (D) is 6mm and the pitch (P) is 1.

$$L_e = \frac{4A_t}{\pi(D-0.64952p)} = \frac{4(20mm^2)}{16.81mm} = 4.76mm \qquad \text{eq. (5)}$$

For threaded holes the minimum engagement length is larger, calculated to be 10.79mm using equation 6. Where the thread engagement ratio (J) is 2.2667, calculated as the ratio between the tensile strength of the bolt (170MPa) and the tensile strength of the hole material (75MPa).

$$L_{e,threaded} = J * L_e = (2.2667)(4.76mm) = 10.79mm \qquad \text{eq. (6)}$$

Accordingly, all of the tapped holes in the case are 20mm deep. This extra clearance means that the holes don't require bottom tapping which reduces costs.

The most critical fatigue loaded bolts are the 10 M10 bolts that hold gear 4 to differential housing. In order to prevent separation when the gear is loaded in reverse a preload of 34kN or 90% of the proof strength. This provides a safety factor of separation of 42. These calculations are in Appendix D.

Conclusions

The Nissan Altima electric axle drive unit features include an optimized gear ratio, strong and reliable gearing, and bearings which provide a long lifetime for the unit. The unit provides a 0-100km/h acceleration of 9.1s which is similar to the stock Nissan Altima acceleration time of 8.9s. The axle drive unit also meets future emissions targets and will help reduce greenhouse gas emissions from vehicles within the popular midsize sedan segment. The entire assembly weighs 120kg. This should provide weight savings from the average internal combustion engine and transmission which weighs about 160kg [15]. This weight can be re-allocated for the batteries of the vehicle. The gear box has been designed with a 12-year lifetime. The there is only a 3% chance that any bearing will fail before this point. In order to ensure that those lifetime predications are accurate, the gearbox has been lubricated with oil. This oil system will need to be chosen to ensure that a minimum flow rate of 0.754LPM is attained through the nozzle. In addition, the output and input shafts have dynamic seals to keep the oil inside. Finally, custom gaskets have been designed for all mating cast surfaces to ensure that oil is retained within the case.

There are a few areas where future work could improve the design. The case should have a FEA topology simulation performed to better understand the loads it bears and reduce the thickness of non-critical locations. This would reduce a significant amount of weight. In addition, the first shaft could likely be hollowed out to save some weight and rotational inertia. Finally, more iteration of the keyway on the intermediate shaft could help to reduce the stress concentration at that location. Finally, the invertor assembly was not included as its parameters are unknown.

The Nissan Altima electric drive unit provides a viable conversion for this internal combustion vehicle. For more details on the process, please refer to Appendix D for hand calculations, Appendix E for all part technical drawings, and Appendix F for relevant MATLAB code.

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Appendix A: AISI 4023 Steel Material Properties

Table 5: 4023 Steel Data

Ultimate Tensile Strength (S_{ut})[MPa]	**724**
Yield Strength (S_y)[MPa]	**415**
Modulus of Elasticity (E)[GPa]	**205**
Poisson's Ratio (υ)	**0.29**
Shear Modulus (G)[GPa]	**80**

Table 6: Shaft corrected endurance limit – non carburized

Material Property	Input Shaft	Intermediate Shaft	Output Shaft
S_e' [MPa]	362	362	362
C_{size}	0.806	0.806	0.7822
C_{load}	0.7	0.7	0.7
C_{surf}	0.788	0.788	0.788
C_{temp}	1	1	1
$C_{reliability}$	0.814	0.8147	0.814
S_e [MPa]	131	131	127.1

Table 7: Gear Bending Failure Coefficients - carburized

Material Property	All Gears
S_{fb}' [MPa]	520
K_L	1
K_T	1
$K_{reliability}$	1.5
S_{fb} [MPa]	346

Table 8: Surface Failure Coefficients - carburized

Material Property	All Gears
S_{fc}' [MPa]	1250
C_L	1
C_T	1
C_H	1
$C_{reliability}$	1
S_{fc} [MPa]	1250

Table 9: AISI 1020 Material Data

Ultimate Tensile Strength (S_{ut})[MPa]	**469**
Yield Strength (S_y)[MPa]	**393**
Modulus of Elasticity (E)[GPa]	**200**
Poisson's Ratio (υ)	**0.29**
Shear Modulus (G)[GPa]	**80**

Table 10: Fatigue Strength of AISI 1020 in Key

Material Property	Key
S_e' [MPa]	234.5
C_{size}	1
C_{load}	1
C_{surf}	0.883
C_{temp}	1
$C_{reliability}$	0.702
S_e [MPa]	145.36

Appendix B: Gear Tables

Table 11: Gear Properties

Gear	Gear 1	Gear 2	Gear 3	Gear 4
Number of Teeth	17	40	18	60
Module (mm)	4	4	4	4
Pressure Angle (°)	20	20	20	20
Helix Angle (°)	20	20	20	20
Face width (mm)	52	52	64	64
Pitch Diameter (mm)	72.4	170.3	76.6	255.4
Quality Index	11	11	11	11
Ka	1.25	1.25	1.25	1.25
Ks	1	1	1	1
Kb	1	1	1	1
Ki	1	1	1	1
Km	1.7	1.7	1.7	1.7
Kv	0.9034	0.9034	0.9301	09301
J	0.44	0.52	0.55	0.67
I	0.1795	0.1795	0.2148	0.2148
Cp ($Pa^{1/2}$)	18703	18703	18703	18703
Cf	1	1	1	1
Axial Contact Ratio	1.506	1.506	1.854	1.854

Appendix C: Bearing Tables

Table 12: Bearing Properties

	Position			
Property	**Input Shaft**	**Intermediate Shaft**	**Differential**	**Driveshafts**
SKF Product Number	6311 M	33111	32017 X	22215 E
Class	Deep groove ball	Tapered Roller	Tapered Roller	Spherical Roller
Inner Diameter (mm)	55	55	85	75
Outer Diameter (mm)	120	95	130	130
C (kN)	74.1	136	171	217
X	0.56	0.4	0.4	0
Y	1.45	1.6	1.35	1
Equivalent Load P 1 (kN)	2.29	7.7	6.38	8.65
Equivalent Load P 2 (kN)	6.41	8.63	11.06	8.02
Smaller L10 (x10^6 cycles)	1547	9798	9203	46218