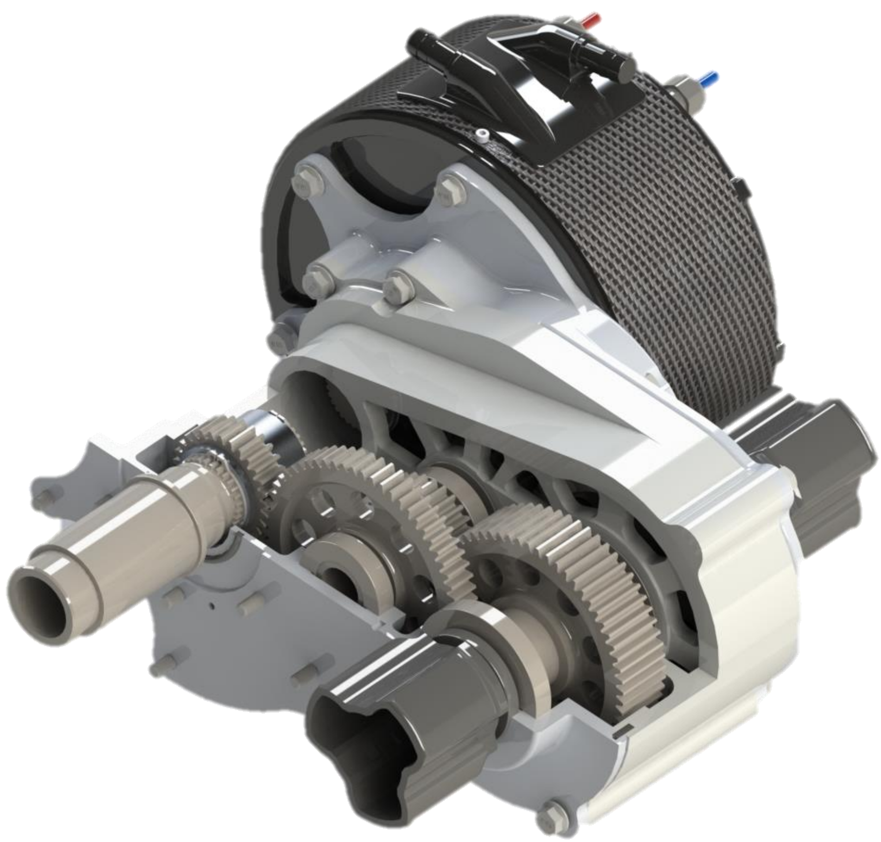
Rear Drivetrain Design Overview



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Introduction

The role of the rear drivetrain is to enable the vehicle to achieve its acceleration and top speed specifications. Within this capacity, the components of the rear drivetrain must be robust should they need to power the vehicle without the assistance of the front motors. Robust, in our application, implies a greater safety factor and decreased weight optimization.

Motor Selection

The Nova 30 motors, manufactured by Plettenberg, were chosen for their extremely high power to weight ratio. Their small form factor contributed to our ability to package them on the vehicle and the purpose-built motor controllers allowed for increased confidence in the team's ability to control the motors once they arrived in house. The motor specifications, found in table 1, create the base for the calculations to identify a gearing ratio for the rear drivetrain.

Table 1: Nova 30 Motor Specifications

Nova 30	
Power Max	up to 30 kW
Approximate Weight	5 - 6,5 kg
RPM Max	2.500 - 5.000 1/min
Torque Max	80 Nm
Voltage Nominal	80 - 140 V
eta max	90 % incl. controller

As discussed in the lap simulation section a physics based model was developed for the vehicle for several of the dynamic events. By adjusting parameters within the simulation, a correlation was developed between the gear ratio and the time in the acceleration event. A gear ratio of 4.5 to 1 provides the fastest acceleration event time, and was chosen for the rear transmission. The decision was made to fix the gear ratio of the rear transmission to reduce complexity and increase reliability.

Packaging Architecture

Implementation of the ratio on the vehicle is a balance between packaging and power transmission efficiency. The motors for the rear drivetrain were mounted within the rear chassis and use half shafts to drive the rear wheels. Each motor drives a single wheel removing the need for a physical differential moving the solution of differing wheel speeds around corners to the controls team. The motor shafts face each other to allow the car to taper at the rear, as seen in figure 1. A chain driven rear drivetrain would require a large sprocket, which would be difficult to package inside of the rear frame, and would not transmit torque as efficiently as a gear box. A single pinion and gear pair would be a similar size as the sprocket, as they are implementing the same gear ratio, and with the motor shafts facing each other the gears would be even larger to ensure the stub shafts clear the motor housing.

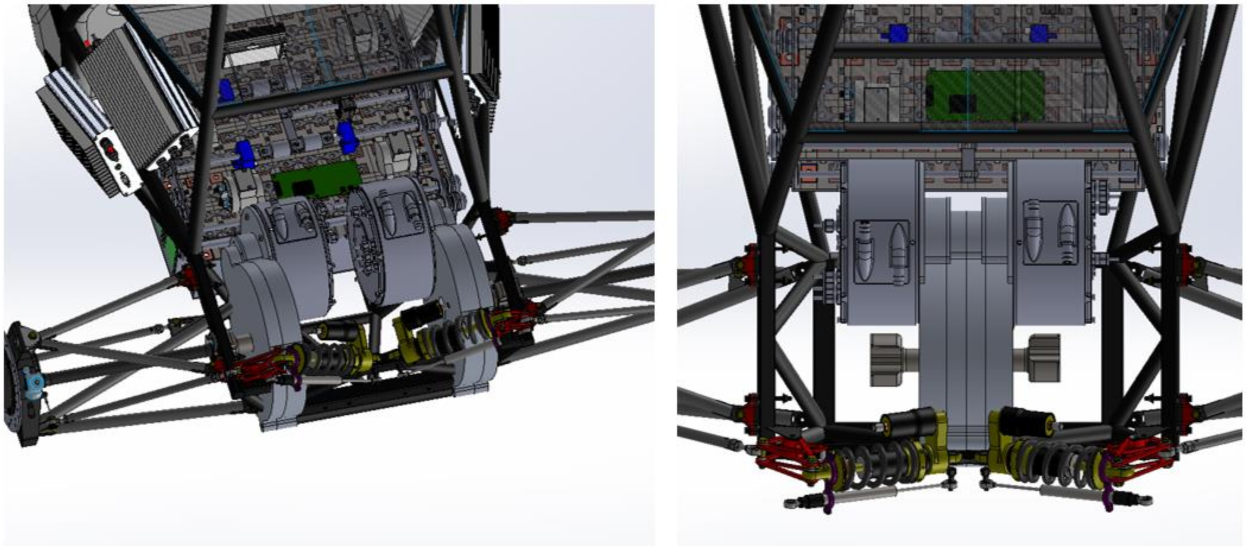

Figure 1: Rear Drivetrain Packaging Taper

To ensure a tight packaging with the proper ratio a two-stage rear transmission was chosen. Each stage has a gear ratio of 2.13 to 1 so that the total gear ratio remains 4.5. The 2-stage gear reduction also provides a greater center distance without the volume of a single stage. An early 2-stage transmission design can be seen in figure 2. The smaller overall volume of the transmission makes packaging within the rear frame tighter. With an architecture determine that will package and transmit torque effectively the next consideration is how strong the gears much be so that they will not break during operation.

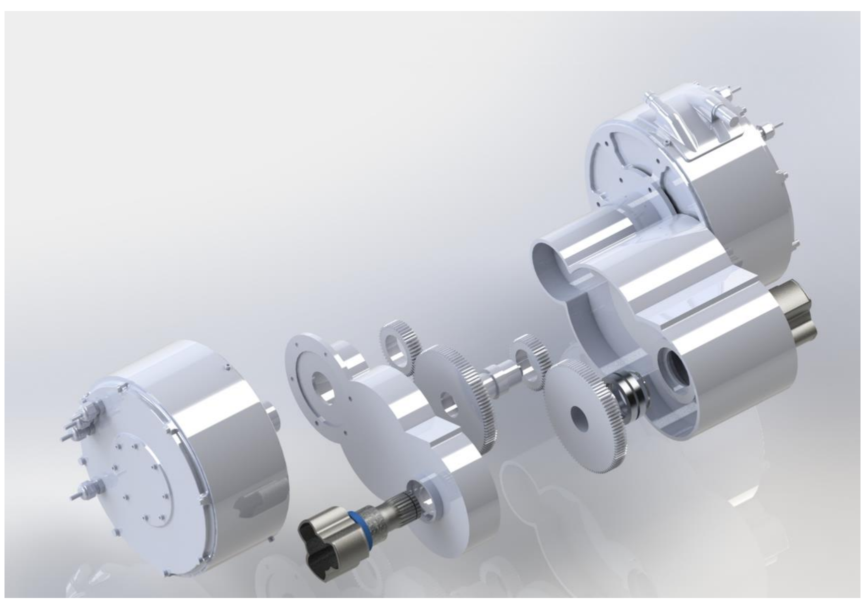

Figure 2: 2-Stage Transmission Design

Gear Calculations

There are two separate consideration for the design of gears: the bending strength of the tooth and the fatigue strength of the contact surface. An inadequate bending strength will result in the tooth breaking under load while an inadequate fatigue strength will cause the surface of the gear tooth to pit reducing the integrity of the tooth and adding contaminants to the gear box. For the initial hand calculations, the following equations are used for bending stress and strength:

$$S_n = S_n{}' C_L C_G C_S k_r k_t k_{ms}$$

$$\sigma = \frac{F_t P}{bJ} K_v K_o K_m$$

Where the correction factors are:

Strength	Stress
K_r = 0.814	K_v = sqrt((78 + sqrt(V))/(78))
K_t = 1.0	K_o = 1
K_ms = 1.4	K_m = 1.3

The equations for fatigue stress and strength are as follows:

$$S_H = S_{fe} C_{Li} C_R$$

$$S_{fe} = 0.4(Bhn) - 10[ksi]$$

$$\sigma_H = C_P \sqrt{\frac{F_t}{b d_p I} K_v K_o K_m}$$

Where the correction factors are:

Strength	Stress
C_r = 1.0 (for 99% reliability)	K_v = sqrt((78 + sqrt(V))/(78))
C_li = determined from cycle life	K_o = 1
	K_m = 1.3

Discussions where held with Edgerton Gear to help inform material selection and gear design. The material for the gears was chosen to be 4140 steel which has a hardness of 302 Bhn. With a gas nitriding post process, the surface hardness was brought up to 555 Bhn. Pairing all this information with the Nova 30 torque and speed map a Matlab code was developed that out puts a 3-axis graph that can communicate the safety factory of the gears at any operating condition of the motor. The bending safety factor map is figure 3 and the surface fatigue safety factor map is figure 4.

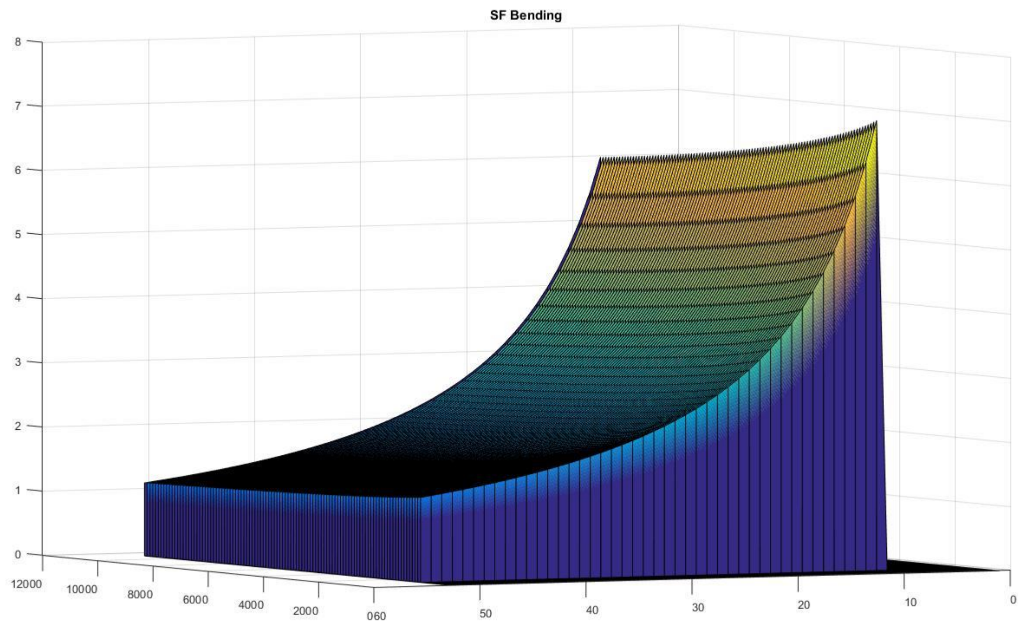

Figure 3: Bending Safety Factor Map

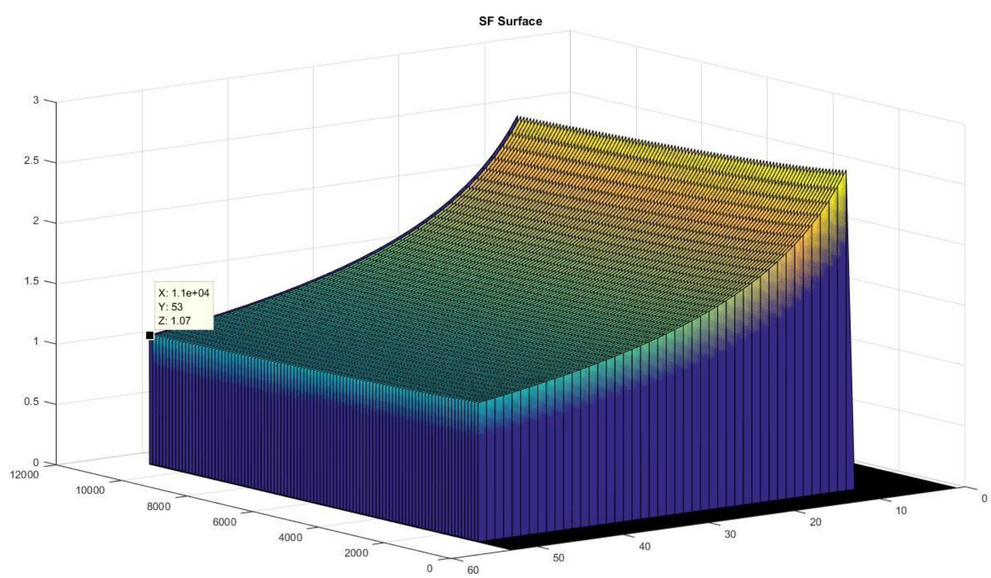

Figure 4: Surface Fatigue Safety Factor Map

Support Software

The maps confirm that the most critical point for the gears is at maximum speed and maximum torque. To create more informative models of the gear operations two software packages were chosen to analyze the gears at the maximum operating conditions. The packages are Windows LDP and RMC developed by the Ohio State Gear Lab and KISSsoft, and industry standard software package for determining gear and shaft loading. Windows LDP provided insights into the surface contact stress as seen in figure 6 and how micro geometries might be used to round the crown of the tooth to reduce stress concentrations at the edges. I addition to this it provided a useful way of visualizing the bend stress of a gear and where failure in bending would occur, seen in figure 5. However limited access of the software resulted in an inability to utilize all the features that may have been available.

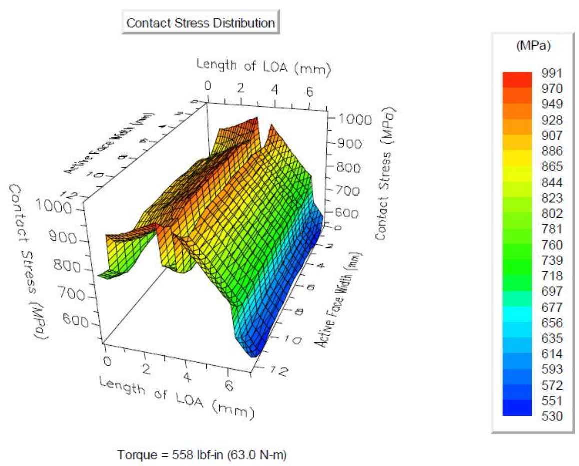

Figure 6: Contact Stress Concentrations

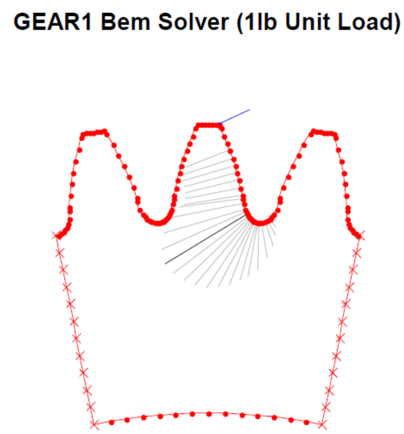

Figure 5: Bending Stress Visualization

KISSsoft allowed for a continuation of this analysis and incorporated the shafts with the gears. The output from KISSsoft provided bending and surface fatigue safety factors for the first stage as 1.14 and 0.73 respectively. The surface fatigue safety factor falls below 1 because they added strength of the gas nitriding process was not expressed in the software iteration. For the second stage, similar safety factors were achieved of 1.14 for bending and 0.76 for surface fatigue. The shaft analysis in KISSsoft mirrored the hand calculations and provided bearing loads which were used to pick the NSK bearings. These bearing loads were also used to model the stresses on the transmission housing for FEA. Future analysis of the Torque, Stress, Force, and Displacement graphs could lead to optimized shaft designs where thicknesses are varied at certain locations to create uniform stress distributions and displacements.

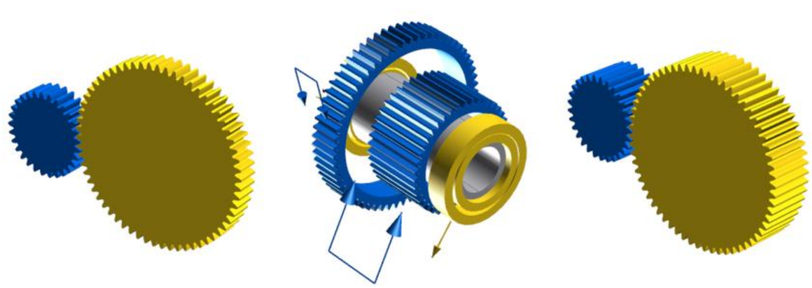

Figure 7: KISSsoft Gear Analysis

Solidworks Analysis

The transmission housing was analyzed using Solidworks FEA tools. The load cases for the housing were taken from the shaft analysis provided by hand calculations and KISSsoft. These loads were joined by the bearing press fits modeled as a fix displacement which was obtained from the NSK shaft tolerance and fit documentation and the weight of the motors on the housing modeled as a point load at the center of mass of the two motors. In addition to looking at the materials strength and stress at each location special consideration was also given to the deflection of the housing based on the forces. To ensure proper meshing of the gears the housing must deflect as little as possible under high loads. If too much deflection occurs, it may lead to poor meshing and uneven ware. The center of the gear box is cast using Aluminum A356. The yield strength of A356 is 22 ksi and as a fatigue estimate the scale in the FEA is set to 11 ksi. The bearing races in figure 8 are completely red because of this small scale, the races are steel and not cast aluminum, and because the forced displacement requires a much higher force to move the steel the same distance as the aluminum housing. In real life the bearings will be in compression so they high stress in this model can be ignored.

The results of the FEA on the housing were that the center housing, which is the limiting factor when it comes to strength, experiences stresses which fall below half the yield stress. The outer housing, with is manufactured from 6061-T6 experiences much of the same loading scenario but has a strength of almost twice that of the cast component and thus is not a concern. The decision was made to cast the center housing to reduce manufacturing costs, increase part complexity, and as an exploration of production processes for a larger scale.

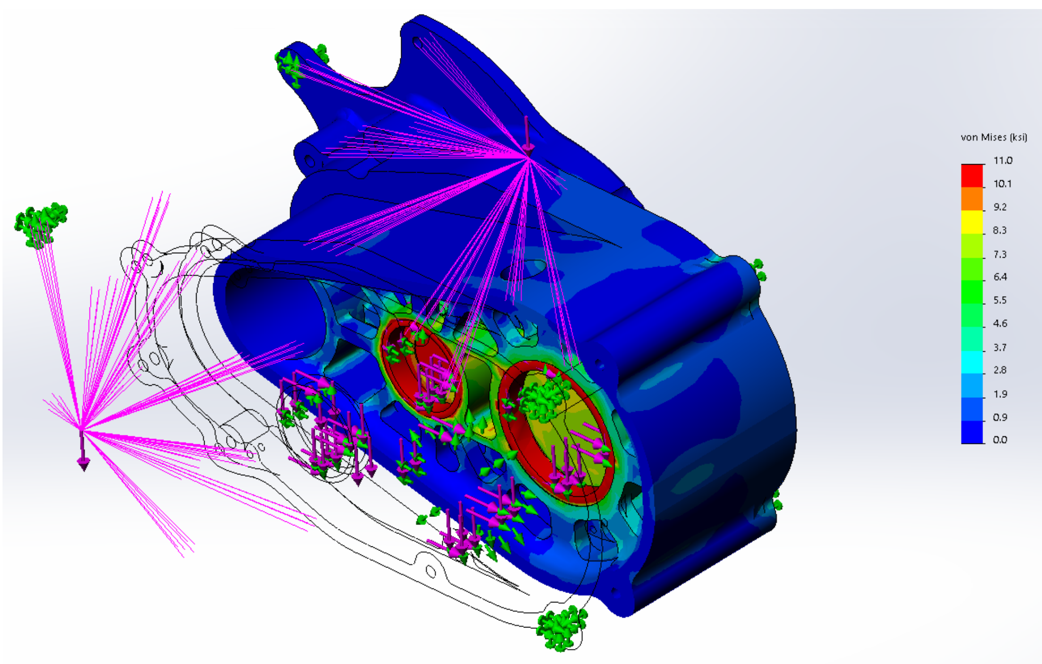

Figure 8: Housing FEA