MSE 320: Project Report Part 2

Mechanical Design and Analysis of Proposed Electric Three- Wheeler

For Dr. [redacted]

Group Members:

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Letter of Transmittal

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Dr. [redacted]
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Dear Dr. [redacted]

It is indeed a great pleasure for us to be able to design a single-passenger trike vehicle. This report is being submitted on behalf of Team 18, to fulfill the project 2 requirements as part of MSE 320. This report is due on December 4, 2019, and is being submitted by [redacted], Sepehr Rezvani and [redacted]

The purpose of this report is to present detailed design & analysis of the vehicle's chassis and further development of its power transmission performed during part 2 of this assigned project, in which the team will design a three wheeled, single passenger vehicle.

Thank you for supporting this project and reviewing this submission. The team looks forward to working closely with you to address any comments you may have.

Sincerely,

[redacted]

Member of Team 18, MSE 320

Executive Summary

This project is the further development of the three wheeled, single passenger vehicle powered by an Emrax 207 AC Synchronous Electric Motor. The scope of the second part of the project includes development of a compatible power transmission with detailed analysis and calculation of its components. In addition, a detailed design and analysis of the vehicle's chassis has been included. The report describes the measures that were taken to ensure complete safety of the vehicle and a break down of the cost of each component of the vehicle has also been reported.

Introduction

In this project we are tasked with designing the chassis and drivetrain for our three-wheeler trike. We decided to design a trike with an electric motor for the eco-friendly young male who enjoys a powerful vehicle for use within the city. Having this in mind we opted for aggressive safety factors and a space frame chassis to protect our passenger and a tadpole shape for better steering and stability. These factors make for a safe and sporty vehicle that is environmentally friendly by leveraging the BC electric car tax rebate as a plus of this vehicle.

Description of Design - Drivetrain

The drivetrain design required to integrate our motor required the use of various transmission components that we were made familiar with during the semester. A belt and gear system were used together as a demonstration of knowing both systems. Whilst our reduction ratio requires 2nd belt drive in addition to our 1st belt drive, we opted to implement a gear drive instead. This was due to the high torque as we got closer to the driving wheel and gears provide more safety than belts for this use. Gears can also be lubricated using carbon shavings, increasing their life. Additionally the gears allow us to create mounting plates for the drivetrain for better overall transmission rigidity. This is important when considering a powerful drive train as ours, if mounting is not adequate then we can experience a lot of vibration on the body which is not desired.

Exploded View

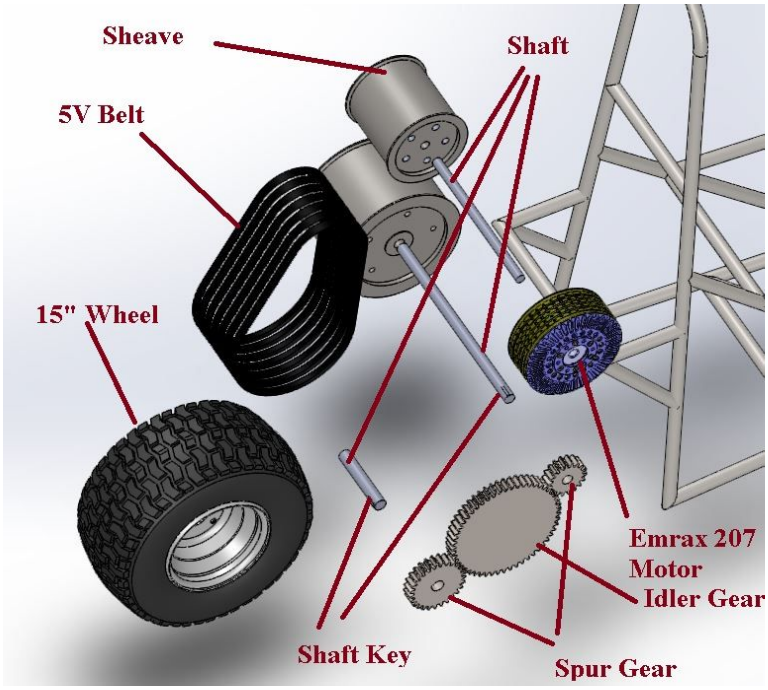

Fig 1. Exploded view of Drive Train

Description of Main Components

Emrax 207 AC Synchronous Electric Motor

The emrax 207 is an electric motor running off of a 3 phase voltage source with 95% efficiency. A rotating magnetic field is generated due to the source and another magnetic field is generated in the rotor as it tries to rotate and catch up to the stator magnetic field. This motor selection was done to align ourselves with the automotive industry who are pivoting to electric cars.

Transmission Design

The three-wheeled vehicle designed for this project consists of three major transmission components, the first component is a belt drive, it's driving sheave is connected to the motor through a shaft that provides the motor's rotational speed to the input sheave, the belt drive transmits power from motor to belt's output shaft that is connected to a set of spur gears. The simple gear train transmits power from the belt's output shaft to the gear's output shaft that drives the rear wheel.

Prior to designing this transmission,a set of initial constraints were defined and are discussed below.

Initial Constraints	Values of constraints	How they were defined
Maximum Speed (km/h)	60	Car is designed to be driven in the city,according to the speed limit.
Total speed reduction	3.03	Calculated from motor output RPM and required RPM for car at the wheels
Speed reduction from belt drive	1.55	Defined to reduce the number of belts and optimize the size of the belt drive.
Gear train speed reduction	1.48	Calculated from the difference of Total reduction and belt drive speed reduction

Table 1. Constraints, Ratios, and Reductions

Belt Design

The primary objective of using belts are that, they have quite a few advantages over chain drive. First of all, belts are lighter than chains so their contribution to the overall weight of the vehicle is low. Although chains can be made stronger than belts, their greater mass increases drivetrain inertia.Conventional roller chain drives suffer the potential for vibration, as the effective radius of action in a chain and sprocket combination constantly changes during revolution.Belt drives are designed to avoid this issue by operating at a constant pitch radius.

In addition, belts are quieter than chains, do not rust and require no lubrication.Chains are used to transmit high torque and belts are used for high speed applications. In our case the torque requirement is fairly low because it is a single occupancy vehicle. The belt drive is a significant part of this vehicle's transmission drive. Certain parameter had to be theoretically calculated and matched to standard values provided in the tables of textbook and industrial belt selection guide. The table presented below, mentions the key values required to design the belt drive.

Parameters	Values	Constraints & Method
Service Factor (SF)	1.2	Light shock loading, motor running daily for 6-8 hours. (Table 7-1, textbook)
Design Power	144.8 HP	$Design\ power\ =\ SF * \ Motor\ pow$

Belt Selected	5V Belt	From Figure 7-13, Textbook. Using design power & input rotational speed.
Input Sheave Diameter (D1)	7.10 inch	Calculated and then corrected according to standard values
Output Sheave Diameter (D2)	10.90 inch	Calculated and then corrected according to standard values
Power rating of belt	22.80 HP/ Belt	Baldor V-belt drive selection handbook, according parameter calculated above.
Number of Belts required	7 Belts	$No.\ of\ Belts = \frac{Design\ power}{power\ rating\ of belt}$
Belt Length (L)	56 inch	Calculated and corrected according to standard values. Table 7-2, textbook.
Center Distance between Sheaves (CD)	13.7 in	Calculated and corrected.
Angle of Wrap, Input Sheave(⌀1)	164 Degrees	Calculated
Angle of wrap, output sheave (⌀2)	195.4 Degrees	Calculated

Table 2.Belt Drive Parameters

Detailed calculation of the values presented here are shown in the Appendix.

Gear Design

The primary objective of using gears in this transmission design was to reduce the number of belts. Although, the total speed reduction required for the output shaft could have been achieved utilizing only a belt drive, the number of belts required for this application would have been higher by a significant number, a single reduction gear was implemented to keep the design compact and less bulky according to client specification. The transmission consists of a set of three gears, first gear is a pinion, the gear in the middle is an idler gear and the larger output gear. The application of idler gear was to increase the distance between output belt sheave and output gear shaft. Calculation for important parameter were done employing the procedures learned in class. These parameters are presented in the table below.

Parameters	Values	Constraints & Method
Pressure Angle	20 degrees	most widely used pressure angle in the industry.
Diametral Pitch (Pd)	6/inch	Figure 9-11, textbook
Diameter of Pinion (Dp)	3 inch	Calculated
Diameter of Gear (Dg)	4.5 inch	Calculated
Number of pinion teeth (Np)	18 teeth	Teeth chosen > 17 teeth
Diameter of Idler Gear	48 teeth	Calculated
Number of gear teeth (Ng)	27 teeth	Calculated
Center Distance (C)	12.75 inch	Calculated
Face Width	2 inch	Calculated
Sacp	1009.5 MPa	Calculated and corrected.
Sacg	977 MPa	Calculated
Stp	129.2 MPa	Calculated

Stg	108.27 MPa	Calculated

Minimum Values were calculated for Sacp, Sacg, Stp, Stg. Values found for the material selected are higher than these calculated here, to withstand forces acting on the gears. SAE 4340 alloy steel were chosen for both pinion and gear according to the calculation.

Detailed calculation of the values presented here are shown in the Appendix.

Shaft design

Shaft design calculations and design selection came after gear and belt selection. Shaft design is critical for achieving an optimal design and shaft failure analysis is very important because shafts connect different subsystems in a machine design.

There are a few points worth mentioning before doing shaft calculations. Generally, shafts encounter three main types of stress:

1) Steady Torsional Shear Stress

2) Vertical Shear Stress

3) Direct Loading

Main focus of this report will be on the first type. Additionally, the following figure shows the distribution of vertical shearing stresses on such a circular cross section. Note that the maximum shearing stress is at the neutral axis of the shaft, that is, at the diameter. Therefore, that would be our point of interest for stress analysis.

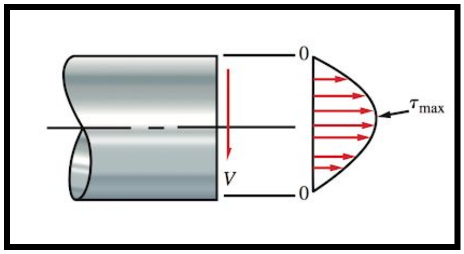

Fig. 2 Stress Distribution across Shaft

In the Belt Selection section of this report, reasoning behind specific type of selection (5-V) and the quantity of belts are mentioned. It is important to note that for this case where V-Belts are chosen, power being transmitted causes torsion, and transverse forces on the elements causes bending moments. Next, the following figure is the basis for shaft design calculations, where x-axis and y-axis are ratio of Torsional Shear Stress over Yield Strength in Shear, and ratio of Reverse Bending Stress over Endurance Strength, respectively. Therefore, rule of thumb for shaft design calculations is to not enter the Failure Zone according to the following figure.

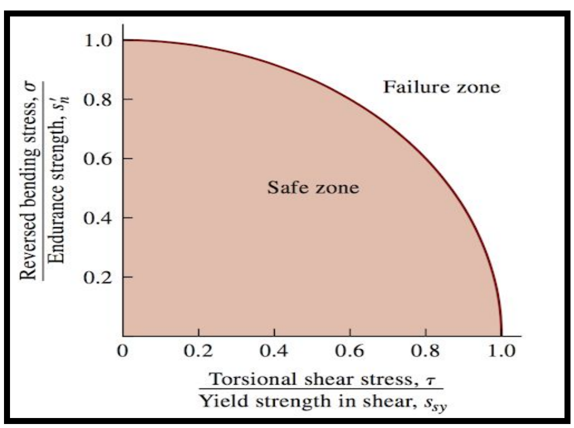

Fig 3. Failure Zone and Safe Zone

The following figure is a rough sketch of the design, and it was the first step of the process. Starting off on the top left of the figure, motor powers the system is transferred through 3 shafts (1-3), 2 sheaves (B and C), and 2 gears (D and E). There are other components, but to reach the goal of this section, these components had enough information to calculate proper diameter of 3 shafts.

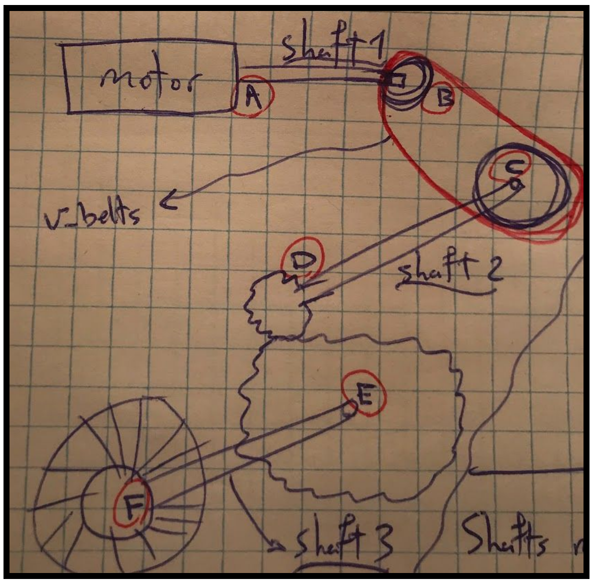

Fig 4. Drivetrain Sketch

The assumption was that shafts 1-3 must have diameter values in ascending order, because shaft 1 is connected to the motor and requires high speed and low torque, opposingly, shaft 3 is connected to the wheel and requires high torque and low speed. Calculation of shaft diameters goes as follows:

Diameter equation for shaft design: $D = [(\frac{32N}{\Pi}) \times \sqrt{(\frac{K_t \times M}{S_n'})^2 + (\frac{3}{4})(\frac{T}{S_y})^2}\ ]^{(\frac{1}{3})}$

For simplicity, it was assumed that there are no gears or bearing in the middle of all shafts. Therefore, the first part of the equation involving ‘M’ can be ignored. Therefore, the following equation will be used for shaft diameter calculation in this report.

$$D = [(\frac{32N}{\Pi}) \times \sqrt{(\frac{3}{4})(\frac{T}{S_y})^2}\ ]^{(\frac{1}{3})}$$

Variables are calculated as follows:

N = 2 → Design Factor. This value was based on the assumption of dynamic loading.

$T = \frac{63'000 \times (P)}{n}$ → where P and n are power input on the shaft {hp}, and rotations per minute {rpm}, respectively. Power is assumed to stay unchanged.

S_y = (235’000 psi) Material’s yield → according to the following figure.

Note: All three shafts were assumed to be made of ‘SAE 1340 Steel’.

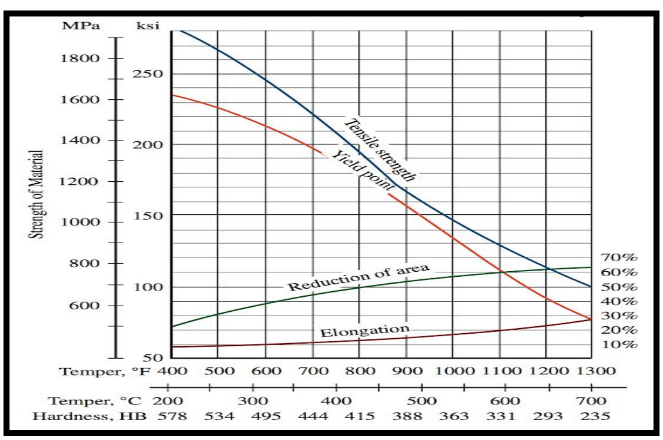

Fig 5. Yield Stress Graph

$$T_1 = \frac{63'000 \times (120.69)}{1386.6} = (5'483.5)\ Ib.in \Rightarrow D_1 = [(\frac{32(2)}{\Pi}) \times \sqrt{(\frac{3}{4})(\frac{5'483.5}{235'000})^2}\ ]^{(\frac{1}{3})} = (0.744)\ in$$

[Diameter of shaft 1]

Gear diameters: $D_B = 7.10\ in$ // $D_C = 10.9\ in$ // $D_D = 3\ in$ // $D_E = 4.5\ in$

[B and C sheaves, D and E gears]

Rotations Per Minute calculations for sheave C and gear E:

$n_B = 1386.6\ (rpm)$ ⇒ (angular velocity at B) $1386.6\ \frac{rotations}{min} \times \frac{(2\Pi)\ radians}{1\ rotation} \times \frac{1\ min}{60\ sec}$

$\Rightarrow \omega_B = (145.2)\ rad/s \Rightarrow$

(tangential velocity on the radius of sheave B) $V_B = r.\omega_B = (\frac{7.10}{2})(145.2) = 515.5\ in/s \rightarrow$

know: $V_B = V_{Belt} = V_C$

$\Rightarrow \omega_C = \frac{V_C}{r_C} = \frac{515.5}{5.45} = (94.58)\ rad/s$

$\Rightarrow n_C = (94.58)\ \frac{rad}{s} \times \frac{60\ sec}{1\ min} \times \frac{1\ rotation}{2\Pi\ rad} = (903.2)\ rpm$ → at sheave C

Finally → $n_C = n_D$ and $n_E = n_D \times \frac{n_D}{n_E}$ = (602.13) rpm

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Diameter calculations for shafts 2 and 3:

$$T_2 = \frac{63'000 \times (120.69)}{903.2} = (8418.37)\ Ib.in \Rightarrow D_2 = [(\frac{32(2)}{\Pi}) \times \sqrt{(\frac{3}{4})(\frac{8418.37}{235'000})^2}\ ]^{(\frac{1}{3})} = (0.8582)\ in$$

[Diameter of shaft 2]

$$T_3 = \frac{63'000 \times (120.69)}{602.13} = (12'627.62)\ Ib.in \Rightarrow$$

$$D_3 = [(\frac{32(2)}{\Pi}) \times \sqrt{(\frac{3}{4})(\frac{12'627.62}{235'000})^2}\ ]^{(\frac{1}{3})} = (0.9824)\ in$$ **[Diameter of shaft 3]**

Main Technical Specifications

Distance	86 Miles/charge
Charging Time(With BOSCH EV200 -3.3kW)	4.5 hours
Maximum Speed	60 Mph

Features of Machine

Safety

Features	Description
Seat Belts Load limiters	Implementation of this mechanism allows, the force applied by the seat belt to the chest rise only to a point where serious injury is unlikely in case of an accident[6]
Dynamic Stability(Understeer/Oversteer)	For safety purposes, one of the requirements of the design was to make sure that the vehicle should be not oversteer. Having tadpole as the configuration, the vehicle will more likely understeer rather than oversteer[4]
Space-Frame Chassis	It uses small tubes placed in trusses like to structure to avoid any bending stresses. That implies that the members of frame will only experience compressive and tensile forces.And the circular tubes has bending moment and torsional advantages over their

	counterpart[3]
Braking Assistance System	As the vehicle is designed to be driven on the road with varying traffic, we will be implementing brake booster system that typically uses electric motor to reduce the force that the driver needs to apply on the brake pedal to generate the desired level of vehicle braking. Recent german accident data show significant safety benefits of this technology.[5]
Inertial Measurement Unit	It is a device that measures the three linear acceleration components and the three rotational rate components (6 dof) of a vehicle. This device would be responsible for tracking the orientations of the vehicle, in case of an accident it will shut down the motor.
Aggressive Safety factors	When designing the transmission belts and gears, the safety factor considered was higher than the actual requirement. For example, the belt was designed for moderate shocking rather than light shocks.

Cost

Component	Price(USD)

Motor(EMRAX 207 -Air Cooled)	$3236
Battery(15kWh- Panasonic NCR 18650)	15kWh*($200/kWh) = $3000
Front Wheels(Magliner Pneumatic Wheel -10 inch)	2*$46 = $92
Rear Wheel(Mickey Thompson Sportsman -15 inch)	$290
Material Cost for Frame(SAE 1060)	38kg*($2000/1000kg) = $76
Seat(MasterCraft Safety Nomad Suspension Seat)	$245
Brake Pads(Advics) + ABC Controller(Dorman OE Solution)	$127+$1467 = $1594
Belts(Dayton 13V737)	7*$31 = $217

The components mentioned here are not the only components that are required to manufacture an Electric Vehicle. The cost for power electronics, manufacturing and labour are not included here.

Detailed Design and Analysis - Chassis

The chassis was designed using the space frame model which uses truss members to create a network of fully constrained joints around the driver. This makes is a very safe design compared to the monocquaq design that uses a sheet metal surrounding which does not resist bending stress which are prevalent when turning or the reaction of a suspension.

The ISO standard pipe structural member - (26.9 x 3.2)mm was used in designing the chassis in order to simply procuring the round stock needed to build the frame. This way we can easily acquire the round stock and then have the pipe cut into appropriate member sizes. The members are then mitered and welded to form the truss joint. Additionally, pipes are excellent in resisting stresses in any direction normal to the surface due to their symmetry, hence they are the best hollow structural member to use for a chassis.

The members are to be constructed out of AISI 1018 mild/ low carbon steel. This is the most commonly used material for chassis design due to its excellent weldability and homogenous grain structure making it a good choice as a structural member. In addition to being very cheap, and easy to machine it has a good balance of toughness, strength and ductility [1].

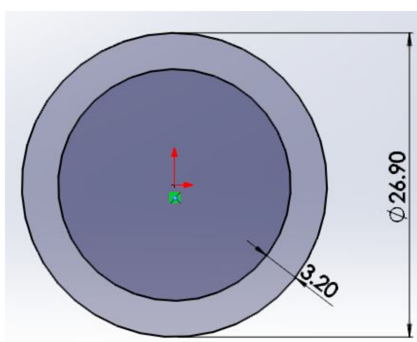

Fig 6. ISO Structural Member Cross Section in mm

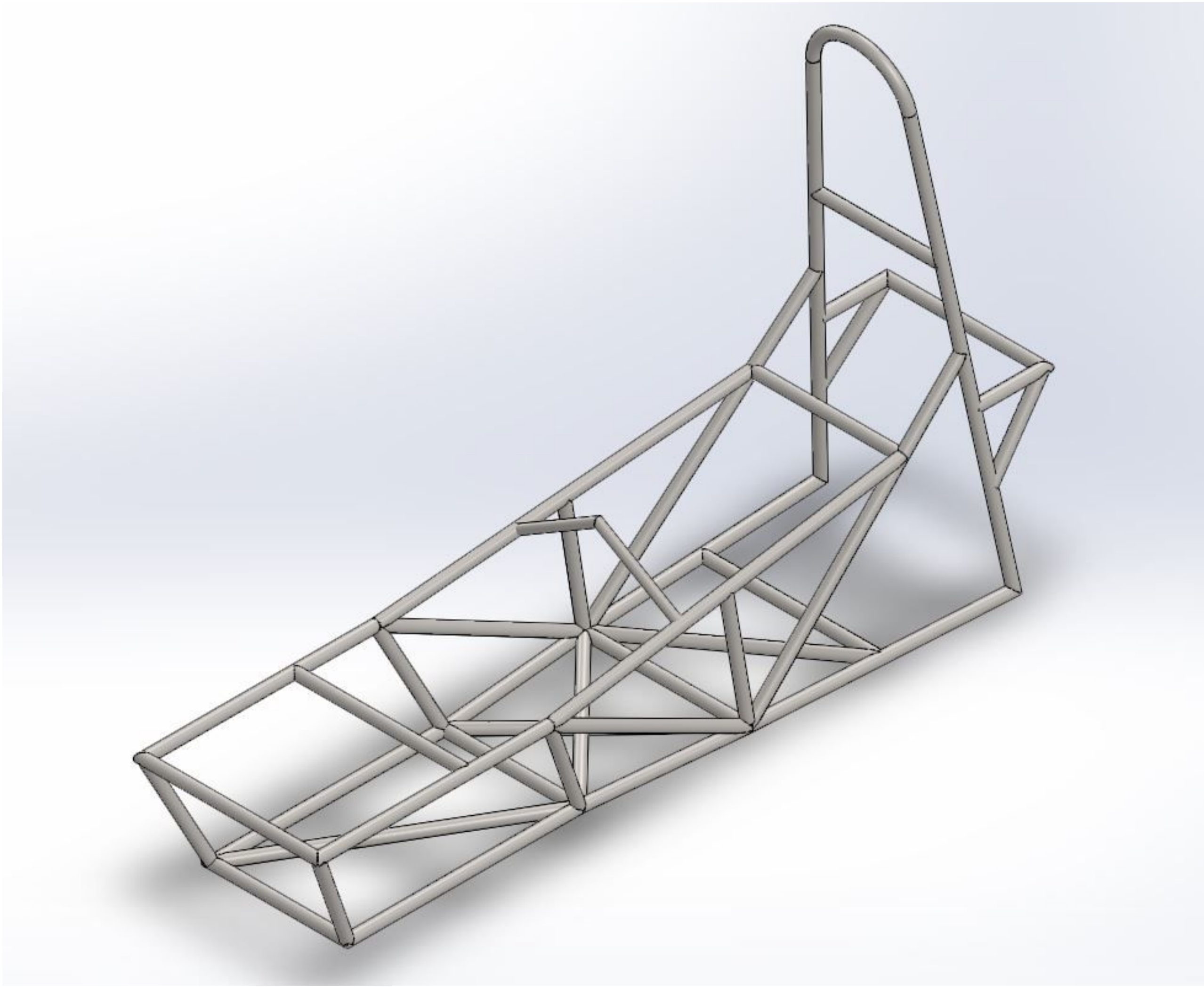

Fig 7. Chassis Rendering in Isometric View

Structural Design and Analysis

The structural design was tested against the payload of the transmission components. These components will be driving the rear wheel and as a result will be placed in the rear. To support these components, various mounts such as the motor mount, belt mount and gear mount will be required to be designed and then held up by the two trusses and beams behind the roll cage.

The mass of the payload using the materials option and mass properties option in Solidworks was given out to be approximately 36Kg after inputting the correct dimensions and material properties for the various components.

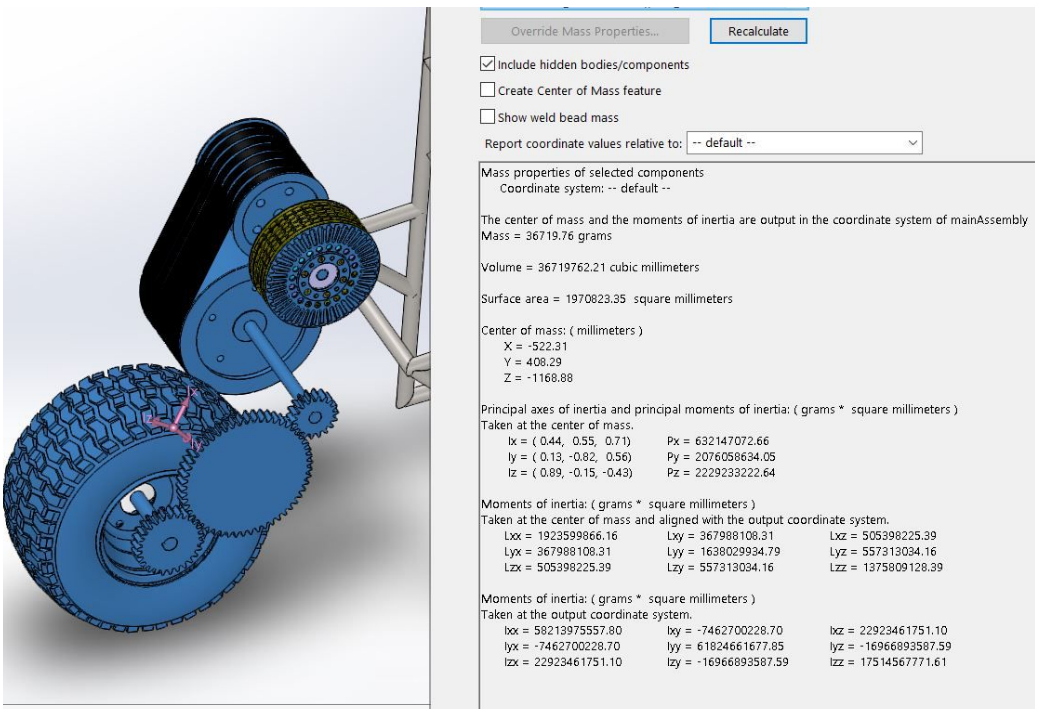

Fig 8. Mass Properties for Drivetrain Payload

This mass is then converted into a force and moment acting on our beam and truss support. To simplify our hand calculations, the centroid of the payload was assumed along the line of action of all the components. Also, the moment was found by taking the distance from the approximate centroid of the drivetrain and our beam support as shown in the image below.

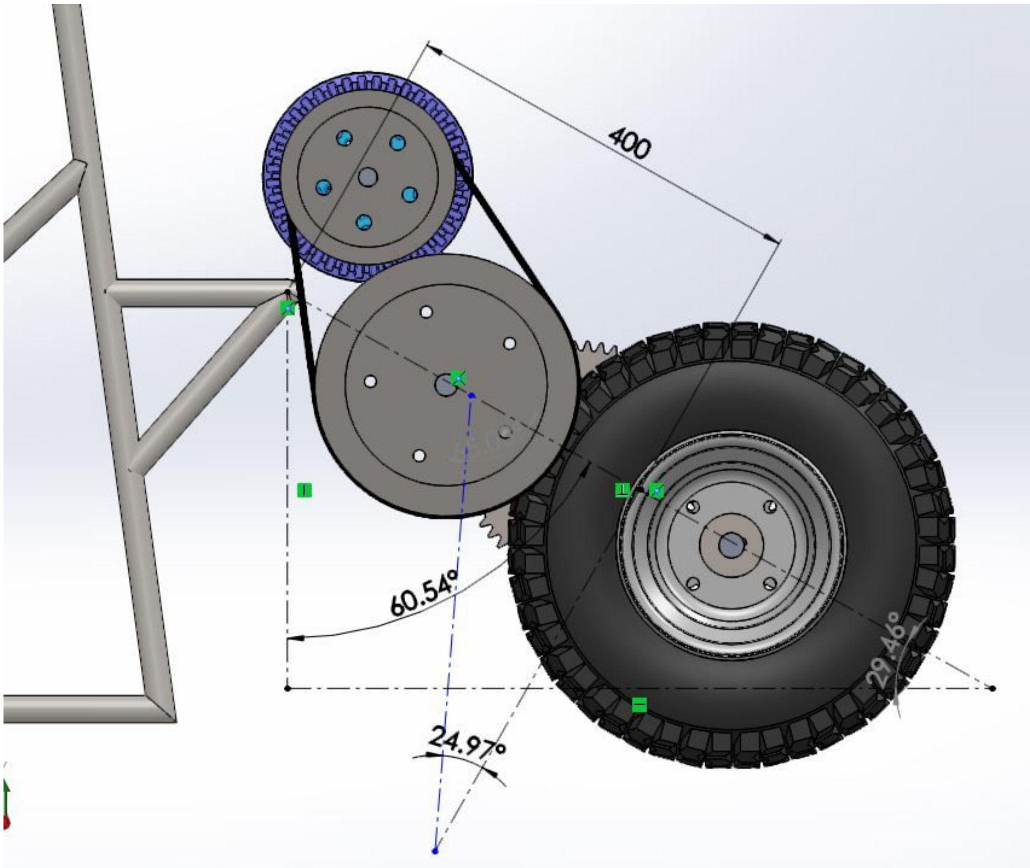

Fig 9. Assumed Centroid and Line of Action for Drive Train

Point Force on Beam Centroid (N)	$353N = 36Kg \times 9.81NKg^{-1}$
Torque on Member (Nm)	$69.5Nm = 353N \times 0.4Cos(60.5)m$

Table 2. FEA Load Conditions

FEA was adopted to test the chassis under the loading conditions obtained above. This simulation was limited due to the lack of computing power used normally in FEA calculations hence a simpler mesh was created out of coarser elements. Ideally elements should be close to the tolerance size of the members obtained. The mesh control options selected are shown below.

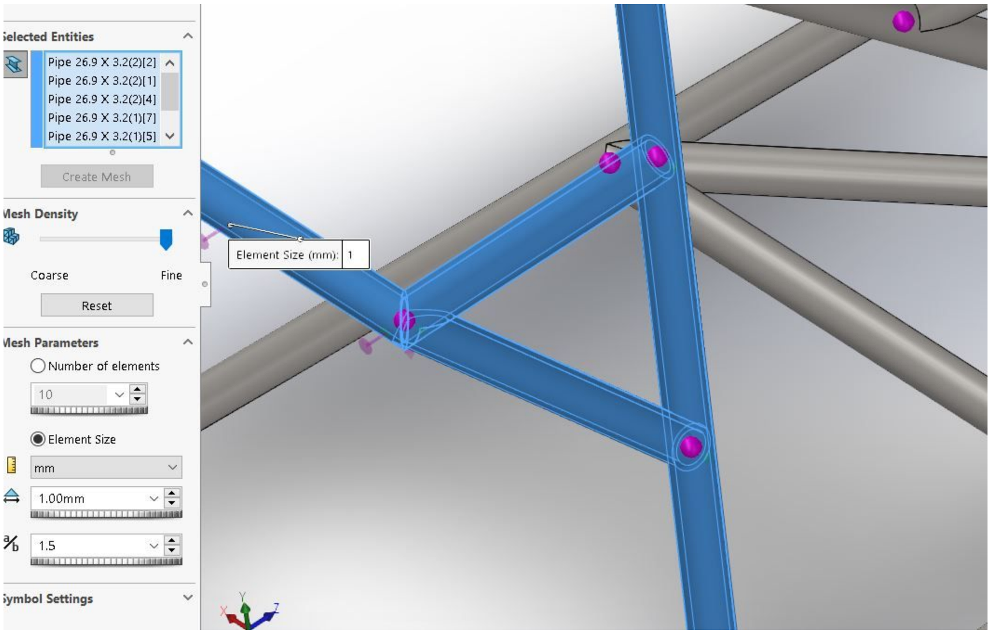

Fig 10. Mesh Element Size and Mesh Density

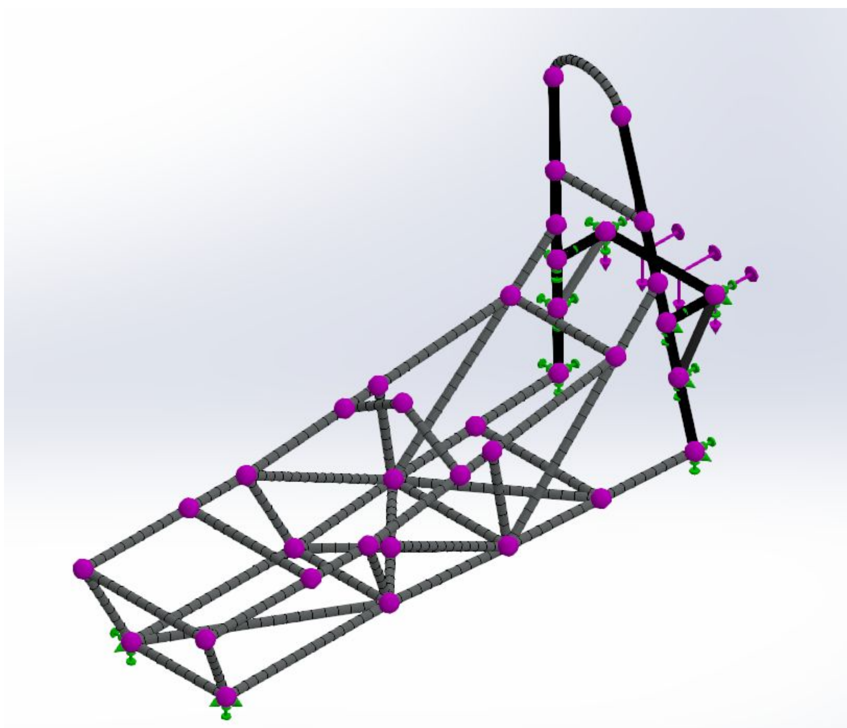

Fig 11. Solidworks Mesh with Fixtures, Joints, and Constraints

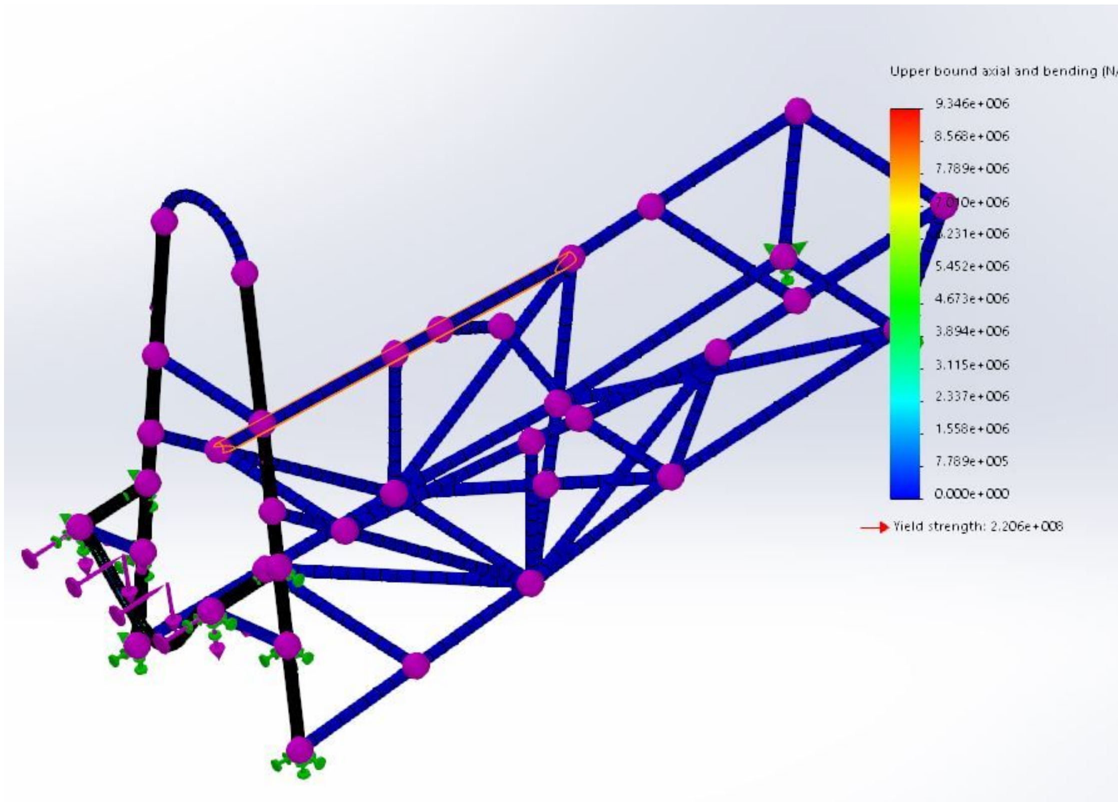

Fig 12. Axial Stress and Bending Color Gradient

The maximum shearing stress for the chassis is $9.346 \times 10^{6} Nm^{-2}$ as per the FEA analysis. The comparable hand calculation for shearing stress is shown below:

$$Shearing\ Stress = Shearing\ Stress \div Shearing\ Area$$

$$Shearing\ Area = \frac{\pi}{4}(D^2 - d^2) = 2.383 \times 10^{-4} m^2$$

$$Shearing\ Force = \frac{353N}{2.383 \times 10^{-4} m^2} = 1.48 \times 10^{6} Nm^{-2}$$

$$Shearing\ Force\ (Torsion) = (Radius \times Torque) \div Polar\ Moment\ of\ Inertia$$

$$Torque = 69.5Nm \div 2\ (2\ Supports) = 34.75Nm$$

$$Polar\ Moment\ of\ Inertia\ =\ \tfrac{\pi}{32}(D^4 - d^4)$$

$$Shearing\ Force\ (Calculated : Torsion)\ =\ 1.37 \times 10^7 Nm^{-2}$$

$$Total\ Shearing\ (Calculated)\ =\ 1.518 \times 10^7 Nm^{-2}$$

$$Percentage\ Deviation\ =\ \tfrac{(Actual - FEA)}{Actual} = 38\%$$

$$Safety\ Factor\ =\ \tfrac{Yield\ Stress}{Actual\ Stress} = 14.5\ Times$$

The stresses whilst having a 38% deviation are still in the ballpark region hence we can take the worse of the two as our assumed stress. Additionally, our yield strength is very high so despite the uncertainty, our chassis will still be more than capable of supporting the transmission components.

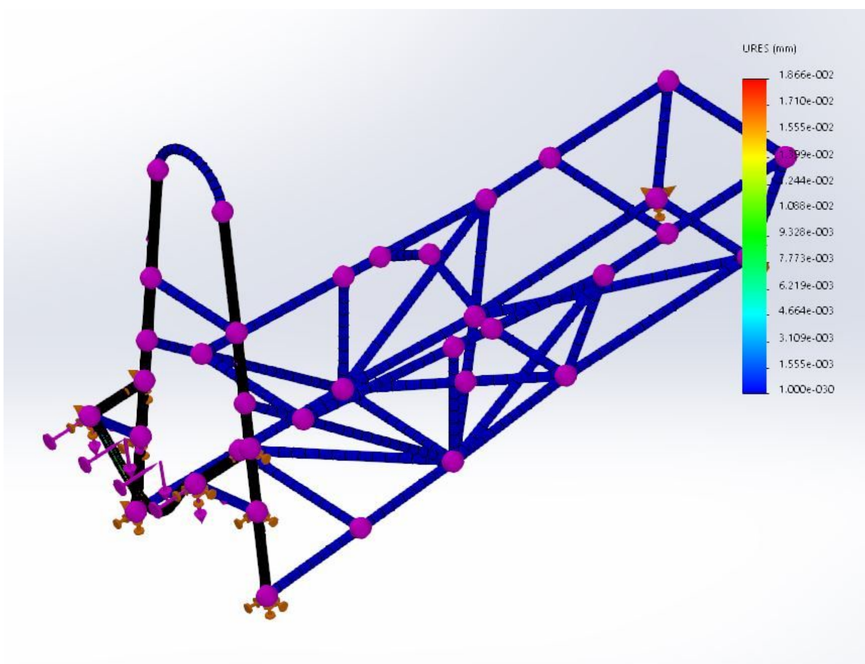

Fig 13. Displacement Color Gradient

Engineering Drawings

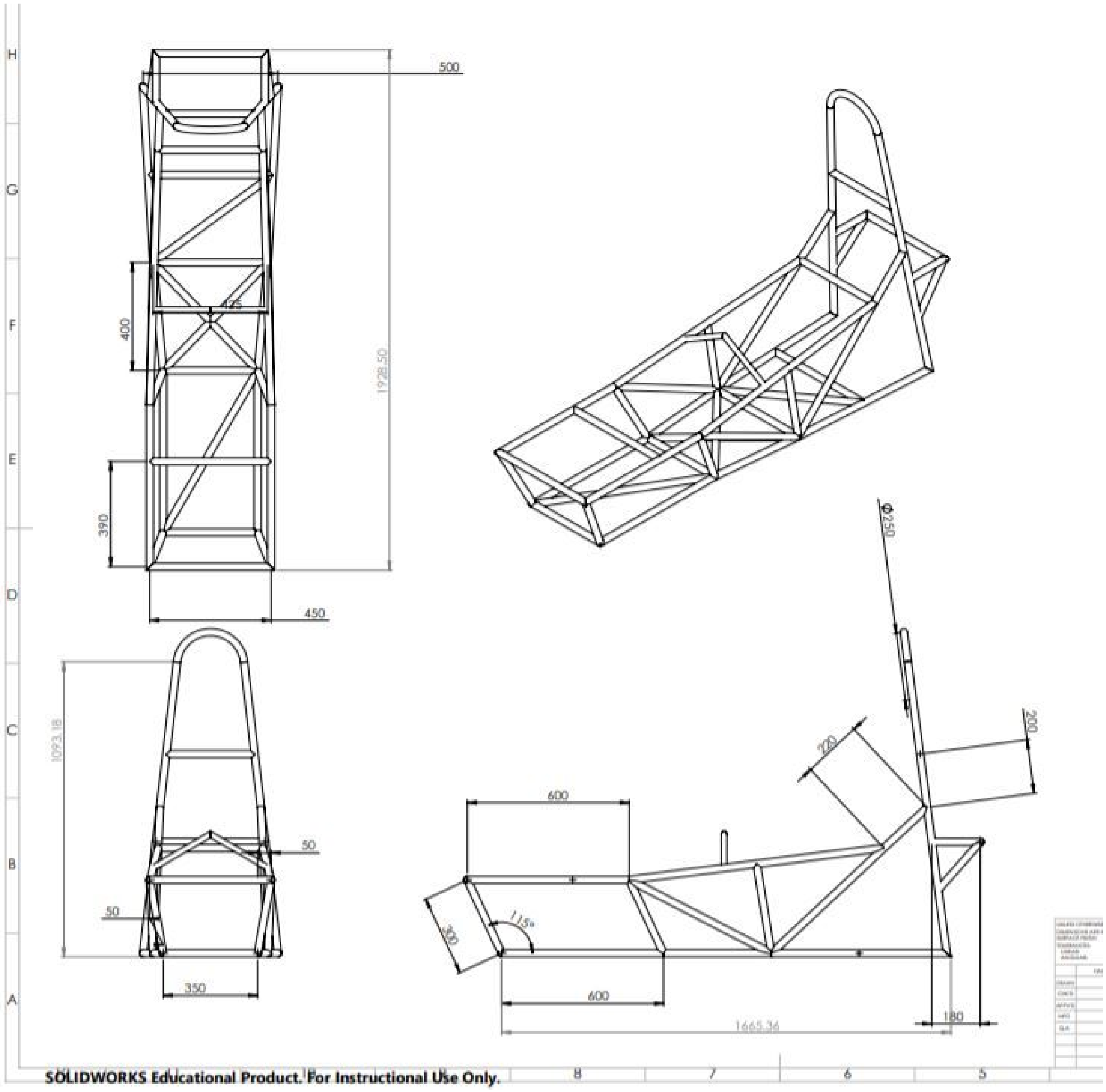

Fig 14. Chassis Dimensions in mm

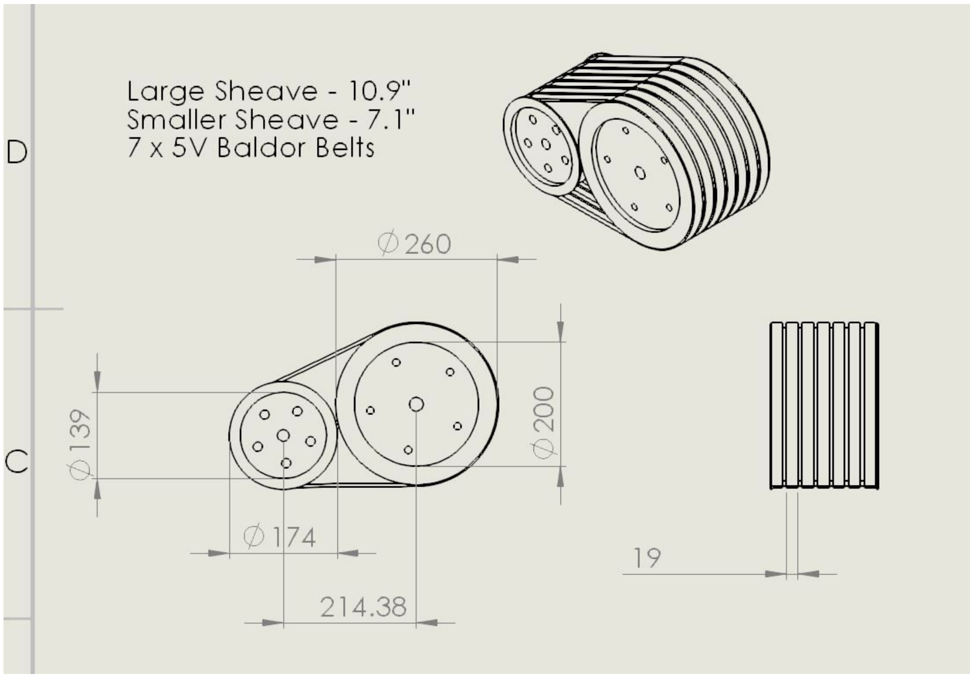

Fig 15. Belt Drive in mm

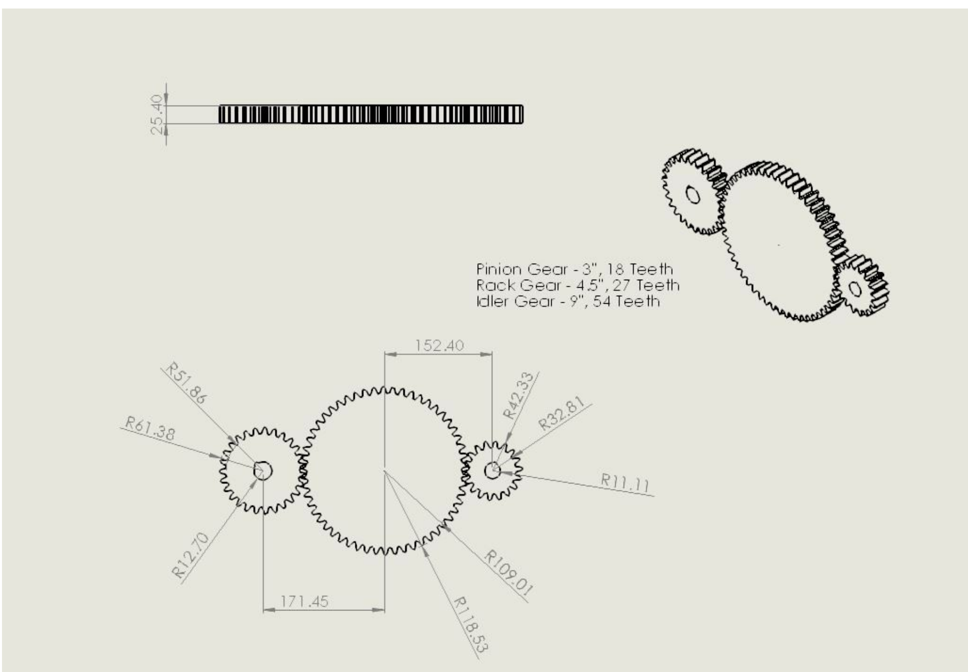

Fig 16. Gear Drive System in mm

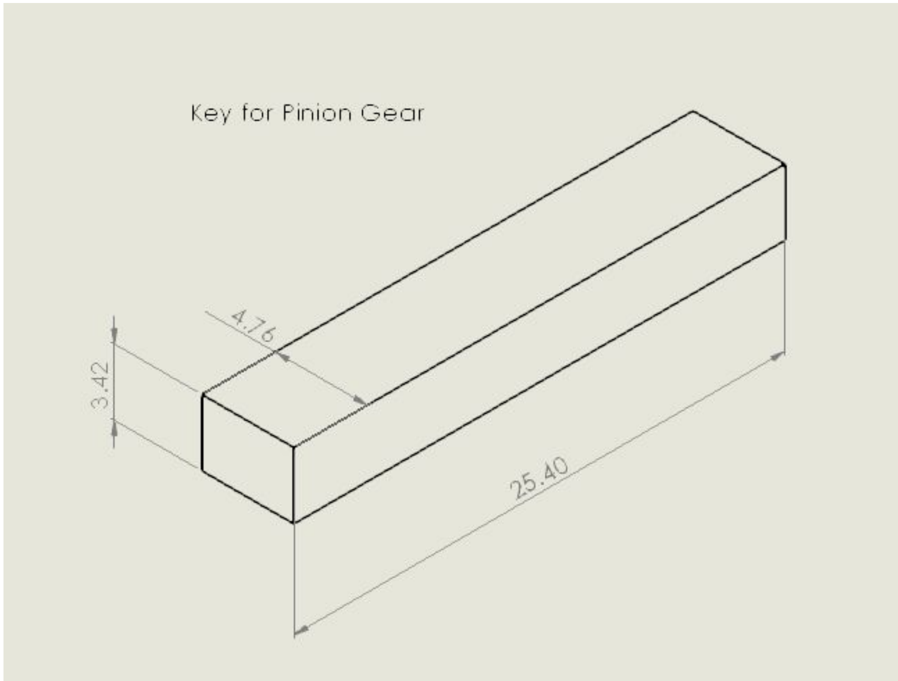

Fig 17. Key for Pinion Gear

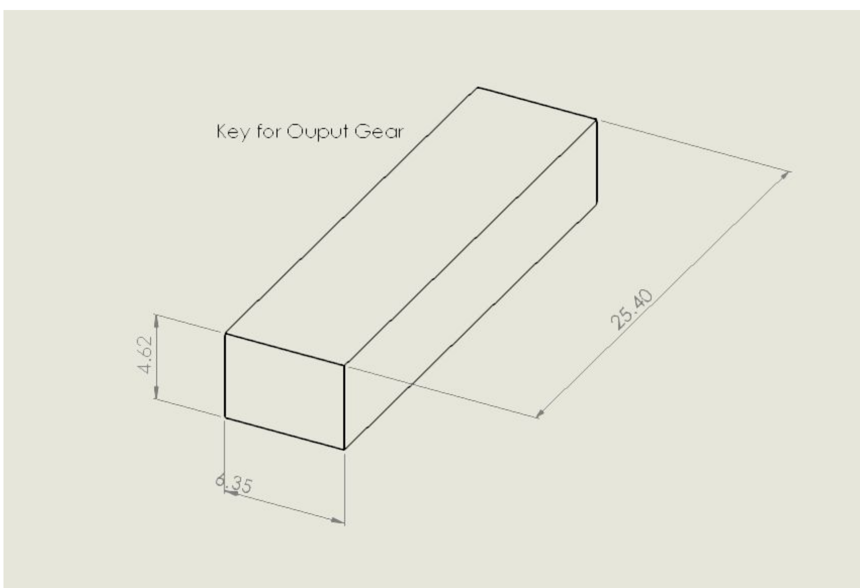

Fig 18. Key for Output Gear to Wheel

Recommendations

Belt Drive

Replacing V-belts with synchronous belts has been recognized as a best practice to reduce energy requirements in industrial and commercial applications, according to a recent report from the US Department of Energy.[6]

Synchronous belts are considered to be environmentally friendly compared to V belts, at the time of installation, V-belt drives run at 95%-98% efficiency. Those percentages will decline to approximately 93% over the life of the belt if it is properly maintained. If V-belt drives are not re-tensioned as part of a preventive maintenance program, they can drop to as low as 80% efficiency during the life of the belt.Once properly tensioned, synchronous belts will maintain a 98% efficiency rating throughout the life of the belt, without the need for costly maintenance. Electrical savings due to the constant 98% efficiency standard of a synchronous belt are significant in many applications.[6]

Gear Drive

When gears work at high loads and speeds, the noise and vibration caused by the rotation of the gears is considered a big problem. However, since noise problems tend to happen due to several causes in combination, it is very difficult to identify the cause.

The following are ways to reduce noise and these points should be considered in the design stage of gear systems.[7]

- ❏ **Use High-Precision Gears**
  Reduce the pitch error, tooth profile error, runout error and lead error. Grind teeth to improve the accuracy as well as the surface finish. [7]
- ❏ **Use a Better Surface Finish on Gears**
  Grinding, lapping and honing the tooth surface, or running in gears in oil for a period of time can also improve the smoothness of tooth surface and reduce the noise.[7]
- ❏ **Ensure a Correct Tooth Contact**
  Crowning and end relief can prevent edge contact. Proper tooth profile modification is also effective. Eliminate impact on tooth surface[7]
- ❏ Use High Vibration Damping Material
  - ❏ Cast iron gears have lower noise than steel gears. Use of gears with the hub made of cast iron is also effective.[7]

Shaft Design

The main reference for shaft design calculations of this project was "Machine Elements in Mechanical Design" textbook (included in References section). Even though calculations are relatively accurate, number of assumptions were significant. First assumption was made for type of shaft material and that is not feasible in a real-life project. Materials are most often alloys in real engineering designs and their material ratio differs depending on their application. However, in this project, all shafts were assumed to be made of the same material (SAE 1340 Steel). Next, in almost every real design project, every shaft has varying diameters (at least 2 different), but here was assumed to be constant for all three diameters in this project. Another assumption that is never accurate in real life, is power efficiency. Although this trike would has a much higher power efficiency than its competitors, it is nothing close to 100 percent. In order to find the true power efficiency of this system it requires a lot more information that was not available to our team and zero waste was chosen for simplify our calculations. Lastly, to have more realistic shaft design, other dimensions must be accurate because shafts connect different subsystems, every error in connecting subsystems decreases shaft calculations' accuracy. For example, it can be seen in Figure 4 that shafts connects multiple gears, sheaves or belts either directly or indirectly and any wrong assumption in those sections will add inaccuracy to shaft design regardless of how well shaft design calculations are done.

Chassis

The chassis can be further optimized to reduce stresses and conserve more material. To implement this time is required to create detailed models so that the accurate moment of inertia of the components can be used to make for a more accurate loading condition. Additionally a powerful computer like a mainframe can be used to carry out the FEA analysis to remove any errors we suspect in computation such as truncation error. In industry, FEA is usually done with computers with access to 1Tb of RAM in contrast to 8Gb of RAM used for our analysis.

References

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Appendices

Emrax 208 Motor Datasheet

EMRAX 207 Technical Data Table

Type / Technical data	EMRAX 207 High Voltage			EMRAX 207 Medium Voltage			EMRAX 207 Low Voltage		
Air cooling = AC Liquid cooling = LC Combined cooling = Air + Liquid cooling = CC	AC	LC	CC	AC	LC	CC	AC	LC	CC
Ingress protection	IP21	IP65	IP21	IP21	IP65	IP21	IP21	IP65	IP21
Cooling medium specification (Air Flow = AF; Water Flow = WF – if inlet water temperature and/or ambient temperature are lower, then continuous power is higher)	AF speed 25 m/s; 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C	AF speed 25 m/s; 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C	AF speed 25 m/s; 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C	inlet WF 8 l/min - 40°C; ambient air 25°C
Weight [kg]	9,1	9,4	9,3	9,1	9,4	9,3	9,1	9,4	9,3
Diameter ø / width [mm]	207 / 85								
Battery voltage range [Vdc]	500 (*580 – to get 7000 RPMp)			300 (*350 – to get 7000 RPMp)			115 (*135 – to get 7000 RPMp)		
Peak motor power (few min at cold start / few seconds at hot start) [kW]	**80**								
Continuous motor power (depends on the motor RPM 3000 - 5000) [kW]	20 - 32	20 - 32	25 - 40	20 - 32	20 - 32	25 - 40	20 - 32	20 - 32	25 - 40
Maximal rotation speed [RPM]	6000 (*7000 peak)								
Maximal motor current (for 2 min if cooled as described in Manual for EMRAX motors) [Arms]	200			320			800		
Continuous motor current [Arms]	100			160			400		
Maximal peak motor torque [Nm]	160								
Continuous motor torque [Nm]	80								
Torque / motor current [Nm/1Aph rms]	**0,83**			**0,54**			**0,20**		
Maximal temperature of the copper windings in the stator and also max. temp. of the magnets [°C]	120								
Motor efficiency [%]	93-98%								
Internal phase resistance at 25 °C [mΩ]	12,0			5,7			0,8		
Input phase wire cross-section [mm²]	10,2			15,2			38		
Induction Ld/Lq [µH]	125/130			52/56			7,2/7,5		
Controller / motor signal	sine wave								
Specific idle speed (no load RPM) [RPM/1Vdc]	15			22			58		
Specific load speed (depends on the controller settings) [RPM/1Vdc]	11 – 15			18 – 22			50 – 58		
Magnetic field weakening (for higher RPM at low torque) [%]	up to 100								
Magnetic flux – axial [Vs]	0,0393			0,0257			0,095		
Temperature sensor in the motor	kty 81/210								
Number of pole pairs	10								
Rotor Inertia (mass dia=160mm, m=4,0kg) [kg*cm²]	256								
Bearings SKF _ FAG	**R/R 6206/6206** or R/AR 6206/7206 or AR/AR 7206/7206 (»O« orientation)								

*For a few seconds.
Maximal battery voltage is 600 Vdc (EMRAX 207 High Voltage). Maximal RPM must not be exceeded.
It is possible to weaken the magnetic field (up to 100%) to get higher RPM at existing battery voltage. Maximal RPM must not be exceeded.
These data are valid for the motors, which were sold after January 2014.
EMRAX motors that had been made before May 2012 have 30% lower power/torque and RPM than new generation of EMRAX motors.

Belt Drive Calculations

- Peak motor Power = 80kW = 120.7 HP
- max RPM being used from the motor = 3300 RPM
- motor runs for 6-8 hours daily (exxagarated hours have been taken to ensure safety, and to use a higher service factor (SF))
- Using Table 7-1 [Constraints: Light shock loading and running for 6-15 hours/day]
  → SF = 1.2
- Design Power = service factor × Peak motor power
  → DP = $(120.7)(1.2) = 144.8$ HP
- Belt selection
  → From Figure 7-13 of Book, a $\boxed{\text{5V belt}}$ is recommended for 144.8 HP at 3300 RPM – input rotational speed.
- Nominal speed ratio or speed reduction
  → $\text{Ratio} = \dfrac{3300 \text{ RPM}}{2100 \text{ RPM}} = 1.535$
- Input Sheave Size:
  → As a guide to selecting a standard sheave size, we comput imput sheave size that would produce a belt speed of $V_b = 4900$ ft/min
  → Belt speed, $V_b = \dfrac{D_1 n_1}{2}$
  ∴ Required diameter to produce V_b of 4900 ft/min → $D_1 = \dfrac{2V_b}{n_1} = \dfrac{2(4900 \text{ ft/min})}{3300 \text{ rev/min}} \times \dfrac{12 \text{ in}}{\text{ft}} \times \dfrac{1 \text{ rev}}{2\pi}$
  Input Sheave size → $D_1 = 6.45$ in or 163.8 mm
- Using theoretical input sheave size to calculate standard input and output sheave diameters at required ouput RPM.

from Baldor V belt selection handbook, Standard V belt found: Corrected value.
Nominal Ratio = 1.55; Pitch Driver = 7.10 inch; Pitch Driven = 10.90 inch
RPM, output = 2180; Power Rating = 22.30 HP/Belt.

- center Distance:
  → Trial CD, using nominal acceptable range for CD → $D_2 < CD < 3(D_2 + D_1)$
  $10.90 < CD < 54$
  Trial CD = 13.5 inch.
- Belt length using trial CD: → $L = 2CD + 1.57(D_2 + D_1) + \dfrac{(D_2 - D_1)^2}{4CD}$ ⇒ L = 55.52 inch
  Using Table 7-2, nearest standard bet length = 56 inch.
  ∴ Corrected CD:
  → $B = 4L - 6.28(D_2 + D_1) = 110.96$ inch | $CD = \dfrac{110.96 + \sqrt{(110.96)^2 - 32(10.90 - 7.10)^2}}{16}$
  $CD_{corrected} = 13.7$ in
- Angle of wrap on belt of the sheave:
  → θ_1 (smaller sheave) $= 180 - 2\sin^{-1}\left(\dfrac{D_2 - D_1}{2CD}\right)$
  $\theta_1 = 164^\circ$
  Larger sheave: $\theta_2 = 180 + 2\sin^{-1}\left[\dfrac{D_2 - D_1}{2CD}\right] = 195.94^\circ$.

Gear Drive Calculations

- Output torque - 128 ft·lbs — to shaft.
  Output horsepower - 34.1 hP
  Output RPM to shaft - 1400 RPM

- Velocity Ratio = $VR = \frac{2100}{1400} = 1.48$
- Design Power $\Rightarrow P_{des} = K_o P = (2)(34.1) = 68.2$ hP

K_o - over-load factor found from Table 9-1
Power Source - moderate shock
Driven-machine - moderate shock

- From figure 9-11, at 2100 RPM w/ Pdesign = 68.2hP
  $\rightarrow P_d = \frac{6}{in}$ or $m = 4$

→ Gear Parameters
- * N_P (Teeth Pinion Gear) = 18 teeth (Assumption > 17 teeth)
- * D_P (Diameter of Pinion) = $N_P(m)$ or $\frac{N_P}{P_d}$ = 3inch or 76.2 mm
- * N_G (Teeth of Gear) = $N_P(VR)$ = 27 teeth
- * D_G (Diameter of Gear) = $\frac{N_G}{P_d}$ = 4.5 inch

→ Final output speed $\{ n_G = n_P(N_P/N_G) \Rightarrow n_G = \frac{2100}{1}\left(\frac{18}{27}\right) \Rightarrow 1386.6$ RPM

→ Center Distance = $C = (N_P + N_G)\, m/2$
$C = \frac{(N_P + N_G)}{2 P_d} = 3.75$ inch

→ Pitch Line Speed
* $V_t = \frac{\pi D_P n_P}{60{,}000} = 8.29$ m/s

→ Transmitted load : Tangential force
$W_t = \frac{P}{V_t} = \frac{25353.8\,W}{8.29\ m/s} = 3058N$

→ Face width, F (using nominal value) $\Rightarrow F = 12m = 12(4) = 48$ mm

→ Pressure Angle, $\phi = 20°$. (most widely used value)

Factors in Stress Analysis
- K_o, over load factor = 2
- Size factor, K_s = 1
- Reliability factor, K_R = 0.99
- load distribution factor, K_m = 1.212

$J_P = 0.31$, $J_G = 0.37$
$C_{pf} = 0.054$
$K_B = 1$
$SF = 1$
$K_V = 1.1$

Adjustment for number of cycles: fig 9-21, 9-22
$Y_{NP} = 0.94$ $Z_{NP} = 0.91$ $Y_{NG} = 0.96$ $Z_{NG} = 0.92$

S_c = 928 mPa

S_{acP} = 1009.5 MPa

Using S_{acP} = 100.9 mPa, figure 9-19 shows required hardness HB = 365 for through grade 1 steel. Figure A4-5 SAE 4340 oil quenched and tempered.

S_{acG} = 977 mPa

Belt Selection Table

BALDOR • MASKA

5V & 5VX BELTS

	Ratio	Pitch Driver in	Pitch Driven in	Center Distance in	RPM Driver = 1750 Driven RPM	HP / Belt 5V	HP / Belt 5VX	RPM Driver = 3500 Driven RPM	HP / Belt 5V	HP / Belt 5VX	*Belt Number
151	1.46	10.30	15.00	30.05	1202	28.02	33.28	-	-	-	1000
152	1.47	6.30	9.25	17.73	1192	12.34	16.02	2384	17.95	27.04	600
153	1.47	8.50	12.50	23.43	1190	20.99	25.39	-	-	-	800
154	1.47	9.00	13.20	24.98	1193	22.94	27.55	-	-	-	850
155	1.47	10.90	16.00	31.78	1192	30.22	35.80	-	-	-	1060
156	1.48	8.00	11.80	21.87	1186	19.02	23.22	-	-	-	750
157	1.50	12.50	18.70	34.37	1170	35.49	42.04	-	-	-	1180
158	1.51	7.50	11.30	20.65	1162	17.08	21.11	-	-	-	710
159	1.51	9.25	14.00	26.64	1156	24.04	28.76	-	-	-	900
160	1.52	4.40	6.70	16.25	1149	5.07	8.38	2299	6.99	14.24	500
161	1.53	4.65	7.10	15.73	1146	6.01	9.34	2292	8.52	15.92	500
162	1.53	4.90	7.50	15.21	1143	6.93	10.28	2287	10.01	17.55	500
163	1.53	5.90	9.00	16.23	1147	10.75	14.30	2294	15.76	24.27	560
164	1.54	5.20	8.00	14.57	1138	8.02	11.40	2275	11.72	19.45	500
165	1.54	6.70	10.30	20.07	1138	14.08	17.89	2277	20.30	30.01	670
166	1.54	7.10	10.90	21.29	1140	15.68	19.62	2280	22.30	32.65	710
167	1.54	9.75	15.00	30.46	1138	26.22	31.23	-	-	-	1000
168	1.55	5.50	8.50	15.44	1132	9.21	12.66	2265	13.51	21.57	530
169	1.55	6.30	9.75	18.82	1131	12.48	16.17	2262	18.19	27.30	630
170	1.55	8.50	13.20	25.36	1127	21.24	25.66	-	-	-	850
171	1.55	10.30	16.00	29.22	1127	27.95	33.15	-	-	-	1000
172	1.56	8.00	12.50	23.80	1120	19.26	23.49	-	-	-	800
173	1.56	9.00	14.00	26.83	1125	23.19	27.82	-	-	-	900
174	1.57	5.90	9.25	18.03	1116	10.92	14.50	2232	16.02	24.63	600
175	1.57	7.50	11.80	22.25	1112	17.27	21.32	-	-	-	750
176	1.58	11.80	18.70	34.89	1104	33.46	39.56	-	-	-	1180
177	1.59	7.10	11.30	20.95	1100	15.67	19.58	2199	22.30	32.60	710
178	1.61	4.40	7.10	15.92	1085	5.11	8.39	2169	7.07	14.27	500
179	1.61	4.65	7.50	15.40	1085	6.04	9.34	2170	8.59	15.93	500
180	1.61	13.20	21.20	38.79	1090	38.02	45.10	-	-	-	1320
181	1.62	9.25	15.00	25.80	1079	23.95	28.63	-	-	-	900
182	1.63	4.90	8.00	14.79	1072	6.95	10.27	2144	10.07	17.54	500
183	1.63	5.20	8.50	15.66	1071	8.14	11.52	2141	11.91	19.68	530
184	1.63	6.30	10.30	18.36	1070	12.47	16.12	2141	18.20	27.24	630
185	1.63	6.70	10.90	19.57	1076	14.06	17.84	2151	20.31	29.94	670
186	1.64	5.50	9.00	16.52	1069	9.32	12.78	2139	13.71	21.79	560
187	1.64	9.75	16.00	29.62	1066	26.14	31.10	-	-	-	1000
188	1.65	5.90	9.75	17.61	1059	10.91	14.47	2118	16.04	24.58	600
189	1.65	8.00	13.20	23.21	1061	19.21	23.41	-	-	-	800
190	1.65	8.50	14.00	24.68	1063	21.18	25.56	-	-	-	850

5V & 5VX BELTS