MECH 3409

Final Report

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Introduction

Discussion on adapting different modes of transportation has been a primary topic for the government, including different sectors concerned with the public. E-bikes allow for a variety of different users to contribute in adopting cleaner uses of transportation. The e-bike design presented in this report has been carefully considered, analyzed under different criteria, and different calculations have been studied to ensure the optimal design. The environmental benefits of e-bikes are unmatched as no harmful emissions are released that could contribute to fossil fuel and air pollution issues.

Idea generation in the design of this ebike was primarily focused on making a user-friendly bike that will allow different users to be comfortable with the new idea of a motorized bicycle. In order for this design to have an impact and create a significant difference in the environment, many people must adapt the new concept. Considering this, the bike frame design went against common e-bike design, which tend to resemble more motorcycle-like frames and focused on a frame that is familiar to users.

In terms of material selection, the main materials considered were nylon, aluminum and steel. The main material under analysis for this report would be steel. Through the use of steel the report analyzes the static analysis of the material under ductile material failure theories, such as the distortion energy theory or (Von Mises Stress) which focuses on how ductile materials yielding is mainly affected by angular distortion energy [11]. Fatigue failure analysis is also considered primarily for the material steel, for the cyclic loading applied to the bicycle frame. The gear analysis done throughout this report is then done corresponding to the AGMA method.

Overall, this report focuses on creating a design for an e-bike that can meet the different safety criteria while also adapting a familiar design to most bicycle and non-bicycle users that can apply a simple gearbox. Focusing on the simplicity of the design is the primary focus to attract as many users as possible and ultimately create an impact in benefiting the environment and reducing harmful emissions.

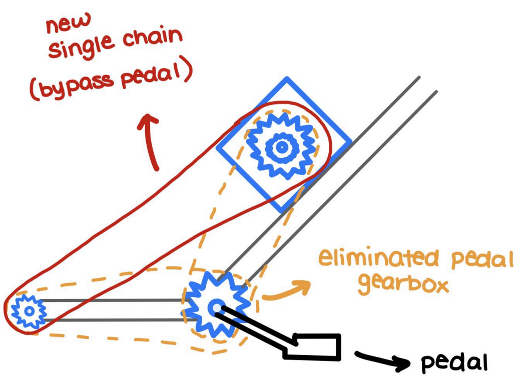
Idea Generation & Concept Selection

<table>
<tr><th>Frame Designs</th><th>Gearbox Designs</th></tr>
<tr><td>Option #1

This frame below was initially selected for the simplicity of the design, however when incorporating the gearbox, it posed restrictions with location and logical spacing and function. Initially the thought was that the frame would include pedals so that the bike would be a hybrid where it would have electrical power and manual power. Incorporating the gearbox and manual pedaling became a challenge, therefore we opted to keep it without pedals and run strictly by electric power.


[2]</td><td>Option #1

The gearbox design had to be connected with the pedal gearbox and with the rear axle gear drive. By eliminating the pedal for manual power, we bypassed the pedal and went directly from the body mounted gearbox to the rear axle drive. This prevented us from having to design a second gearbox for the pedal.


</td></tr>
<tr><td>Option #2

This frame below was selected because it resembled more of a typical bike frame design and allowed us to place the gearbox on the frame and create a pedal foot rest platform where the typical pedal</td><td>Option #2

The gearbox location had to be reviewed since it seemed to be too high on the frame and interfering with the rider's leg. We were considering lowering the gearbox but would have to change the</td></tr>
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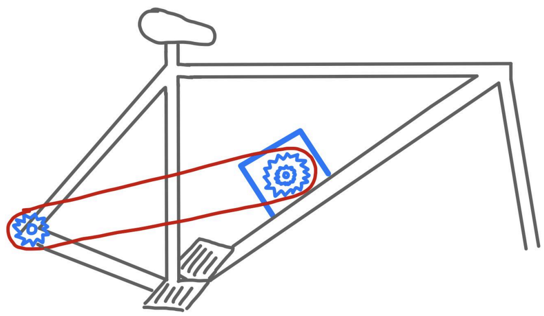
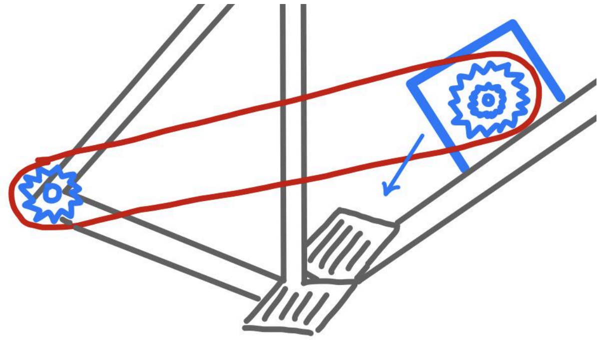
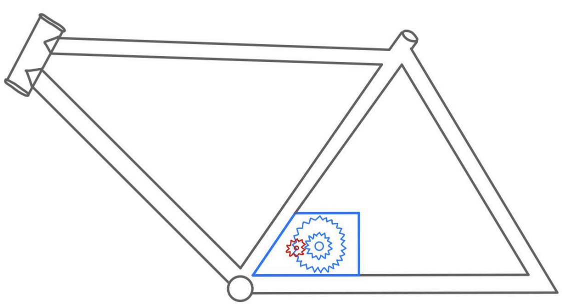
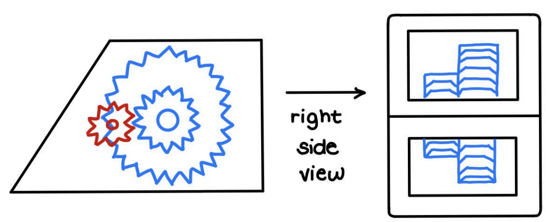
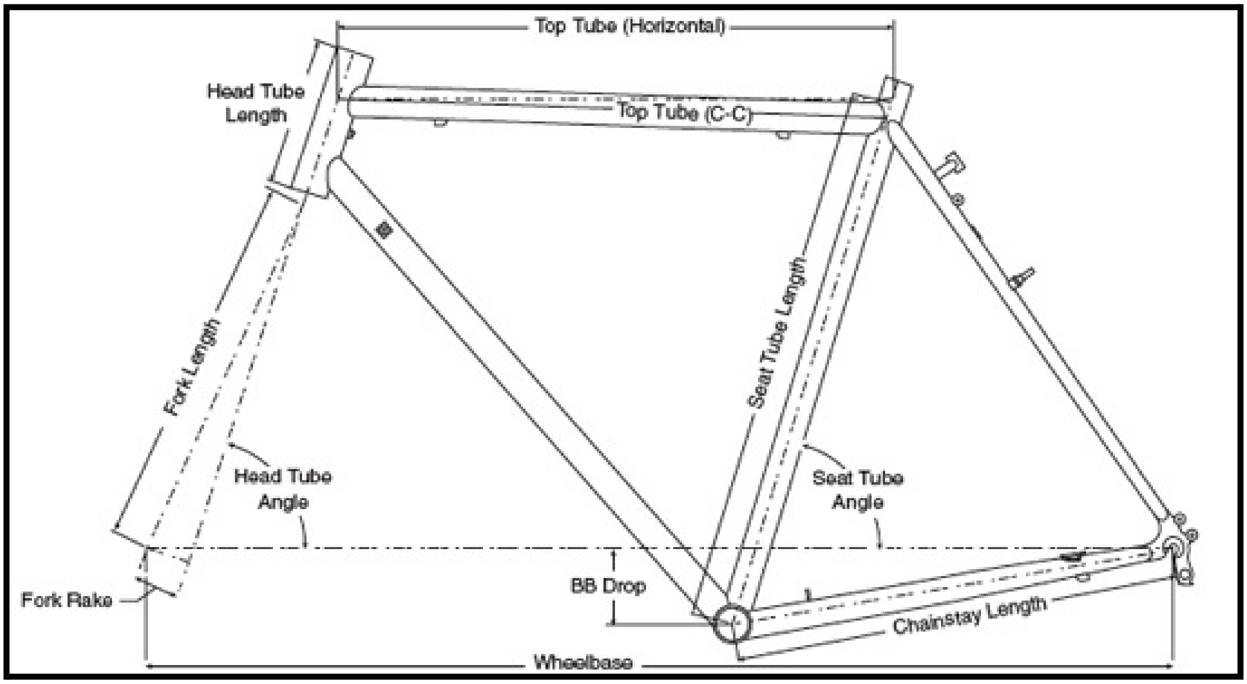
arm would be placed. However we did realize that the location of the gearbox on the frame would be too high for it to drive the rear axle because of its location that could interfere with one's leg. 	shape of the enclosed design to fit appropriately. 
Option #3 The overall design of the frame was kept for simplicity however, some of the dimensions were slightly altered, mainely the rear triangle. It was extended so that the gearbox could fit behind the seat tube but also allow for a wheel to fit without any interference. 	Option #3 The gearbox location was moved behind the seat tube to be closer to the rear axle. As for the gears themselves, we decided to add another smaller spur gear (pinion) to the side of the compound gear and have the motor attached to the pinion instead of directly to the compound gear. We also changed the style of the gearbox from just simply being square to an angled square shape so that the gearbox could fit inside of the frame and also have the chain run in and out of it. 

Selection

Option #3 was the confirmed selection amongst the group for both the gearbox and the frame because it provided the most efficient location in relation to the bike frame style so that it would not hinder the feet and leg positions of the rider. In order for the e-bike to look traditional, the frame design remained typical to a street bike and not distinguish it from a motorcycle design. It is also more familiar for people so that they recognize and distinguish the difference when they are sharing a pathway with others to provide safe spacing. There is also less material being used but the design incorporates several triangular shapes (rigid structure) that provide the strongest form of support for loads. The gearbox housing was originally a standard square shape but was modified with a 45 degree cut to accommodate to fit in the frame so that it can be placed as low as possible. The gears themselves are simple enough to run the 700 rpm motor but strong enough to withstand the 10 Nm torque and the 65kg rider without failure.

Preliminary Design

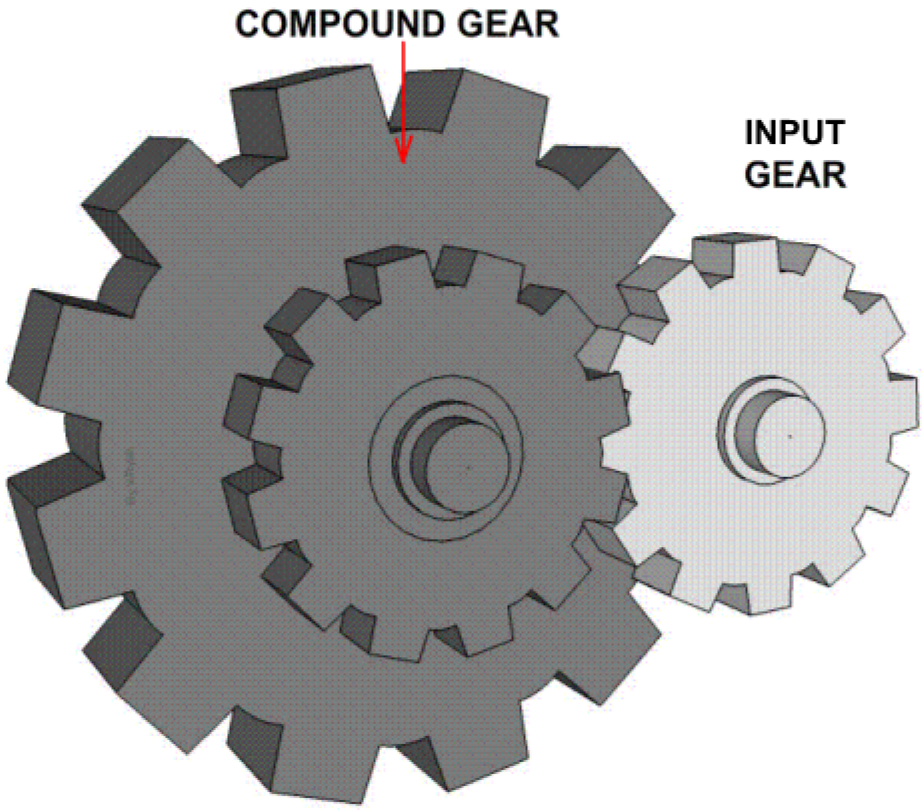
Bike Frame Preliminary Design



[6]

The Frame design was built considering the dimensions/angles of the chainstay, seat stay, seat tube, top tube, down tube, head tube, fork and wheelbase. Considering the aspects of the rider, we chose the build to be that of an average heighted person while taking into consideration that the bike had to be made to withstand the weight of 65kg. Another constraint that we had to account for was that the bike had to undergo daily wear and tear since it is to be used for everyday rides that ranged 5km/day for 10 years. With these constraints in mind we built our frame based on the standard dimensions used to build a medium size bike and made slight adjustments to dimensions to suit our purpose (dimensions provided in Excel spreadsheet). Once the dimensions were complete, we had to choose the most optimal material for the job since the bike had to withstand loading and boundary conditions imposed on it such as static start-up, horizontal, and vertical loadings. We chose to assume total Force acting mainly on the seat tube as the rider's weight times the gravitational acceleration will be applied at that point, creating a bending moment. After researching, we picked two materials that were best suited for frame work, AISI 4130 Alloy Steel and Aluminum 6061. After working out the design calculations and safety factors 4130 steel met the requirements and was the better choice against stress and fatigue while also meeting the cycle requirement of 5km/day for 10 years by having a N cycle of 299669 compared to Aluminum not even meeting the first cycle yield check. The safety factor for 4130 for Goodman was 0.85 (conservative) and 1.01 for Gerber. All in all, although 6061 weighs less, costs less, and does not rust, it is more rigid causing the ride to be unpleasant, prone to fatigue failure meaning that it does not last as long needing more maintenance and is harder to repair since it deforms easily and is subject to cracking. In the end, we decided to go with 4130 steel even with cons such as being more expensive, the bike being more weighted and prone to rusting, steel offers a better comfort in rides by being shock absorbent, able to handle beatings from road wear and tear. The material also offers better protection from fatigue and shows early signs when reaching failure in terms of the frame bending rather than immediately cracking.

Gearbox Preliminary Design



[1]

The gearbox selection included a compound gear in order for the larger gear to connect to the rear wheel drive with a chain. At the same time the smaller compound gear connects to the input gear (pinion) which is ultimately driven by the motor. Once the motor is powered, it will spin the pinion (input gear) which will turn the compound gear to power the rear wheel. The size of the gears were suited to allow for the spur gear design to work more efficiently together, therefore we selected trial dimensions 1 in, 2 in and 4 in, diametral pitch of 10, and face width of 1.25 in (pinion) and 1.5 in for the compound gear. From here we ran tests in the CAD software (Solidworks) and calculations in an

Excel spreadsheet to determine the best dimensions of the gear to allow for optimal meshing and to withstand the torque and speed applied. Our final dimensions were 1.5 in (15 teeth), 4 in (40 teeth) and 6 in (60 teeth) and a diametral pitch of 10 (detailed calculations and dimensions in Excel spreadsheet). The material selected for the gears was to incorporate strength yet lightweight for added efficiency and power to meet the criteria. We researched three basic types of materials; nylon, aluminum and steel. Based on our research we determined that steel was the optimal choice and is the most common material used in gears, which has been proven overtime to be the most durable, therefore we went with this selection and sacrificed a bit of extra weight for durability and longevity. If reduced noise and vibrations levels were a primary factor over durability, nylon would have been chosen and if corrosion resistance was a primary factor, aluminum would have been considered. Since neither of those were a major concern, we opted for steel.

Analysis & Iterative Design

Gearbox Analysis

Gear Fundamentals	Gear 1 (Pinion)	Gear 2	Gear 3
Face Width (F, in)	1.25	1.5	1.5
Pressure Angle (Φ)	20	20	20
Diametral Pitch (Pd, teeth/in)	10	10	10
Pitch Diameter (d, in)	1.5	4	6
Torque (T, lb.in)	88.51	88.51	88.51

Tangential Transmitted Load (Wt, lbf)	
Wt_1	118.013333
Wt_2	44.255
Wt_3	29.5033333

Interference of Spur Gears	Np_est (estimate)	Ng_est (estimate)	Gear Ratio (m)	N_P	N_G
Test 1	10	20	2	14.1607591	28.32151613
Test 2	12	30	2.5	14.6370799	36.59270146
Test 3	15	40	2.666666667	14.763407	39.36908533

For the biggest gear, after the final test (test 3) we decided that 60 teeth would be best to complete the compound gear.

AGMA Stress Equations				
Description	All	Gear 1	Gear 2	Gear 2
Bending Geometry Factor (J)	-	0.25	0.62	1
Surface Geometry Factor (I)	0.116874	-	-	-
Elastic Coefficient (Cp)	2300	-	-	-
Dynamic Factor (Kv)	-	1.26	1.43	1.57
Overload Factor (Ko)	1	-	-	-
Surface Condition Factor (Cf)	1	-	-	-
Size Factor (Ks)	-	1.031763328	1.050108711	1.05238402
Load Distribution Factor (Km)	1.01	-	-	-
Rim-Thickness Factor (Kb)	1	-	-	-
Transverse Diametral Pitch (Pd)	10	-	-	-
Face Width of the Narrower Member (F, i	-	1.25	-	-
Pitch Diameter of the Pinion (dw1, in)	-	1.25	-	-

(US Units)	Gear 1 (MPa	Gear 2 (MPa	Gear 3 (MPa)
AGMA Bending	4958.53134	866.068643	393.8731215
AGMA Contact Stress	66997.7798	44094.6478	37393.40118

AGMA Strength Equations			
Description	All	Gear 1	Gear 2
Bending Strength (St, psi)	-	66910	63045
Contact Strength (Sc, psi)	-	254500	238400
Hardness Ratio Factor (C_H)	1	-	-
Stress Cycle Factor (Y_N)	0.928346106	-	-
Stress Cycle Factor (Z_N)	0.948436889	-	-
Reliability Factor (K_R)	1	-	-
Temperature Factor (K_T)	1		-
Brinell Hardness	-	700	650
Number of Load Cycles (N)	100000000	-	-
Reliability (Rel)	0.99	-	-

Safety Factors	Gear 1	Gear 2	Gear 3
S_F	12.5270234	67.5784543	148.5950095
S_H	3.60276399	5.12777323	6.046718062

The gearbox analysis follows the AGMA recommended method, which analyzes the bending stress and contact stress of the three gears in the design. Through this method the safety factors were also found and compared to reveal the gears that are most critical in the design and also what the gears were more critical to (bending or contact stress)

The first step of this analysis was to define and measure fundamental aspects of the gears, recorded in the “Gear Fundamentals” section of the excel spreadsheet. A pressure angle of 20 degrees was chosen, a larger pressure angle will allow for less interference of the gears but at the same time 20 degrees ensure that it is not high enough where the radial force will increase. A reasonable face width of 1.25, 1.5 and 1.5 for gear 1, gear 2 and gear 3 respectively was chosen, ensuring that the gearbox was as compact as possible. The diametral pitch was then chosen to be 10 based on it being the optimal choice for a medium sized gear system. The pitch diameter was then found by using solidworks to determine the sizing. The pitch diameters were 1.5, 4 and 6 for gear 1, gear 2 and gear 3 respectively. The final component of this table was the torque which as part of the information given a torque of 10 N.m is what the motor produces, which when converted to lbf.in is 88.51.

Following this, in order to compute the bending and contact stress the transmitted load is required, this was found using formula (13-33) in the Shigley's Mechanical Engineering Design textbook, which involves using the pinion diameter and torque.

Ensuring that there is minimal interference of gears formula (13-11) was used to find the minimum required teeth, which will have no interference, using K=1, for full depth teeth. This formula was used primarily between the two gears in the design which will be in contact which is gear 1 and 2. An estimate was made initially before using the formula, but the calculated values were referred to. Three estimates were made and compared using the actual values from the formula (13-11) for the pinion and (13-12) for the gear. The best test that was done was found to be test 3 as it allowed for an optimal pitch diameter. To find the number of teeth for the largest gear it was decided 60 teeth would be the best to complete the compound gear.

Beginning to calculate the AGMA bending stress for each gear, the overload factor (Ko) was found to be 1 for a uniform power source. The dynamic factor remaining the same for all gears was found using equations (14-27) and (14-28) with a Qv of 5. The size factor (Ks) was found using the formula in section (14-10), using the face width, diametral pitch and lewis form factor from (table 14-2), interpolation was done for gear 2 which had 40 teeth. Following this, the load distribution factor (Km/Kh) was found using formula (14-30) for uncrowned teeth, commercial, enclosed units, and centered gears. Cpf, was found for gears with a face width between 1 and 17 in and Ce was used for all other conditions. The rim-thickness factor was 1 for a uniform gear. Finally, the spur-gear bending geometry factor (J/YJ) was found using (Fig. 14-6) for the mating of each gear.

Calculating the AGMA contact stress the first factor calculated was the elastic coefficient (Cp/ZE), using (Table 14-8), choosing steel as the pinion material. Following this, the spur gear surface geometry factor (I/ZI) is calculated for external gear meshes which would be between gear 1 and gear 2, mG is calculated from the ratio of these two gears, mN=1 for spur gears and the pressure angle would be 20 degrees as mentioned. Then the surface condition factor (Cf/ZR) would be 1 as no values are currently given by AGMA.

Finally the factors required to find the safety factors SF and SH. Gear bending strengths (St) and gear contact strengths (Sc) are found using (Fig. 14-2) and (Fig. 14-5) for through hardened, grade 1 steel. Stress-Cycle Factor (YN) and (ZN) is found for 10^8 cycles and the hardness ratio factor CH will be 1 since the pinion and gear hardness ratio is less than 1.2.

After all these factors were found the bending and contact stress was found for each gear and the safety factors as well. The calculated contact stress proved to be significantly greater than the bending, which is comprehensive as the gear teeth will be in constant contact with each other. This would also explain why the safety factors for contact are much lower than for bending. The calculated safety factors show that contact is more critical and important to consider for all gears, and pitting in the gear teeth would be more of a concern than fatigue fracture.

Bike Frame Analysis

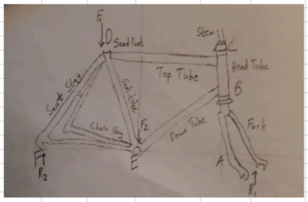
Materials	Ultimate tensile strength (Mpa)	Yield Tensile Strength (MPa)	Density (g/cm^3)	Poisson's ratio	Module Elasticity (GPa)
AISI 4130 Alloy Steel	670	435	7.85	0.27-0.30	205
Aluminum 6061	310	276	2.7		68.9

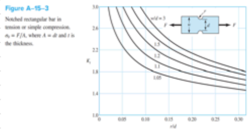
Applied Loads/Loading Conditions	weight (kg)	Load(N)
Total Weight	65	637.7
F1(load on seat)	55	539.6
F2(load on peddle)	10	98.1
R2		434.1
R1		203.6

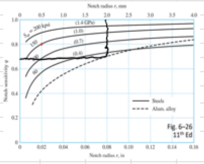
Location	Angle(Degrees)	Right Angle sect.
E	73.1	
F	53.8	45(above horizontal), 8.8(below)
D	53.1	22.1(above vertical), 30(below)

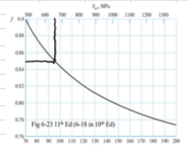
Frame Dimensions	Length (m)	Outer Diameter (m)	Inner Diameter (m)	Area (m^2)
AE(cos45)	0.687			
EF	0.423	0.016	0.012	0.0126
ED	0.430	0.032	0.028	0.0126
CD	0.420	0.032	0.028	0.0126
BC	0.101	0.032	0.028	0.0126
AC (sin45)	0.612			
AB		0.024	0.02	0.0126
FD	0.051	0.016	0.012	0.0126
AD(cos45)	0.768			
AF	1.110			

ED is the critical beam being looked at. So assume total force acting on it. Frame treated as beam/shaft









*Treat as 2D Problem

Stress Calculations	Loads (N)	Tension/Compression	Moment (N*m)(M_a, M_m	$Stress_a$ (MPa)
EF	380.2	Tension		
DF	531.6	Compression		
ED	Highest/Critical = 637.7		257.2	193

No rotation, so assume no Torque

Machined	
a	3.04
b	-0.217

Notch	Steel	Aluminum
radius	2mm	2mm
Neuber √a	0.3271	
K_t	2.3	2.3
q	0.81	0.7

Strength Calculations	S_{ut} (Mpa)	S_e' (Mpa)	K_a (machined)	K_b (bending)	K_c (bending)	K_d (assume room temp.)	K_e (99.9)	K_f	S_e
4130	670	335	0.74	0.95	1	1	0.753	2.06	177.5
6061	310	155	0.88	0.95	1	1	0.753	1.91	97.1

Stress Calculation	$Stress_{max}$' (Mpa)	n_y	Yield check		f	a	b	N	n_f Goodman	n_f Gerber
4130	328.7	1.32	Pass	Better Material	0.855	1848.9	-0.170	299669	0.85	1.01
6061	305.4	0.90	Fail							

Assuming stress is unidirectional so $Sigma_a$ and $Sigma_m$ are equal meaning Sigma max is both combined. These are also repeated loads.

Finite Life predicted

Steel was the more reliable material.

AISI 4130 Steel	(Mpa)
Sigma_a	164.3
Sigma_m	164.3

6061 Aluminum	(Mpa)
Sigma_a	152.7
Sigma_m	152.7

AISI 4130	(Mpa)
Sigma_ar	217.7

A	1057345.2
B	1057345.2

The frame calculations started by taking the previous loading conditions and forces from the previous report and determining which part of the frame was the most critical. To start off, we first make a free body diagram of the frame and state where the reaction and load forces are. Since the wheels will be touching the surface, our two reactions, R1 and R2 are respectively on the front and back wheels and our F1 will be focused on the seat tube where the rider will be seated and F2 on the bike pedal. After designing the bike frame with our own calculated angles/dimensions according to the standard M size bike, we then proceed to solve for our reactionary forces by finding the moment at point A (where R1 is situated) and multiplying our forces with their respective distance from point A horizontally. Then we find the summation of forces in the y-direction and now that we have two equations with two unknowns we can isolate one of them and substitute it to the other equation to solve for the one of the reaction forces and then plug the one we solve for to the first one to solve for the second reaction force. Now that we have the reaction forces, moments, lengths, diameters, area and angles we can proceed to solve for the factor of safety using the knowledge from chapters 6 and 7. We made the assumption and based on research, we found that the seat tube was the most critical point that had to support the load after calculating the loads on the seat stay and chain stay which had 531.6N and 380.2 N respectively compared to when we assumed full body load on seat tube, 637.7N, rather than splitting it to 2 forces to do shaft stress calculation. We then calculated the Moment of the seat tube and assumed the moment is unidirectional and a repeated load meaning that M_a and M_m are the same and since the shaft does not rotate we assumed it to have no Torque but rather bending moment mainly.

Next, we proceeded to find the strength calculation as we learned in chapter 6 to find our Endurance limit and marin factors especially, our K_f to find out $Sigma_a$ and $Sigma_m$. The first step is to find our materials property information and with the help of sites [3] and [4]. Once we have our ultimate tensile strength and yield tensile strength in Mpa for both AISI 4130 and 6061 Aluminum we can divide the ultimate strengths by two to find S_e'. We then find K_a, K_b, K_c, K_d, and K_e to solve for S_e. Our first factor is machined and we get the a and b from table 6-2 and use equation 6-18 to solve for K_a. For K_b we use equation 6-24 to solve for the diameter for a non rotating round and plug that into equation 6-20 respective to being between 2.79 and 51mm. K_c is 1 for bending from 6-26 and K_d we assume 1 for the room temperature. For K_e we wanted a 99.9 reliability and got the value from table 6-4. By multiplying the S_e' with the marin factors, we got S_e for both materials. To calculate K_f we used the table and charts provided in the excel, using the notch radius, neuber for steel, K_t and q to solve for it using equation 6-32.

To end with, we solved for the stress calculations, $Sigma_a$ and $Sigma_m$ using equation 7-2 and then used 7-15 to solve for maximum von mises stress which turned out to be 328.7 Mpa for steel and 305.4 Mpa for aluminum. Equation 7-16 was used to find first cycle yielding and only steel passed the check making steel the suitable choice. We had to solve for the number of cycles using 6-59 to find $Sigma_{ar}$ and using equation 6-13, 6-14 and figure 6-23 for f we solved for N using 6-15. All that was left to do was calculate the DE-Goodman, equation 7-7, and DE-Gerber, equation 7-11, by solving for A and B using equation 7-6. With this we got a conservative safety factor of 0.85 Goodman and 1.01 for Gerber.

Final Design

The final design meets all of the design considerations where the frame structure is a single piece, no weld part that is designed to support a rider weight of 65 kg. It will extend beyond 10 years average riding distance as the stress life cycle for steel is 299,669 cycles, which although has a finite life is still strong enough to withstand use for 10 years. The gearbox uses steel gears to help the motor drive the ebike. The steel has a good combination of lightweight yet durable and cost effective. We made sure that the material and the gear design would be able to withstand the necessary loads as well as any failure that could occur within the gears themselves. This design was the best for looks and functionality as we decided to keep a classic bike look and a simple gearbox as to welcome all riders to try this ebike.

This is a picture of our final design put together

Safety Factor for Static & Fatigue Lifetime (Steel)

n_static = $\frac{S_y}{\sigma'} = \frac{435}{329.4}$ = 1.32

n_fatigue (Goodman) = 0.85

n_fatigue (Gerber) = 1.01

(Both Goodman and Gerber on Excel Spreadsheet)

We performed multiple tests in the Excel spreadsheet to decide on the best possible designs for both the frame and the gearbox. After calculating the safety factors and choosing the best scenario, we determined that the final design should be able to withstand any static failure and although having a finite life, will last at least 10 years, withstanding the fatigue lifetime.

Bill of Materials

Parts	Quantity
Bike Frame	1
Pinion	1
Gear	1
Big Gear	1
Small Pin	1
Large Pin	1
Gearbox Top	1
Gearbox Bottom	1

References

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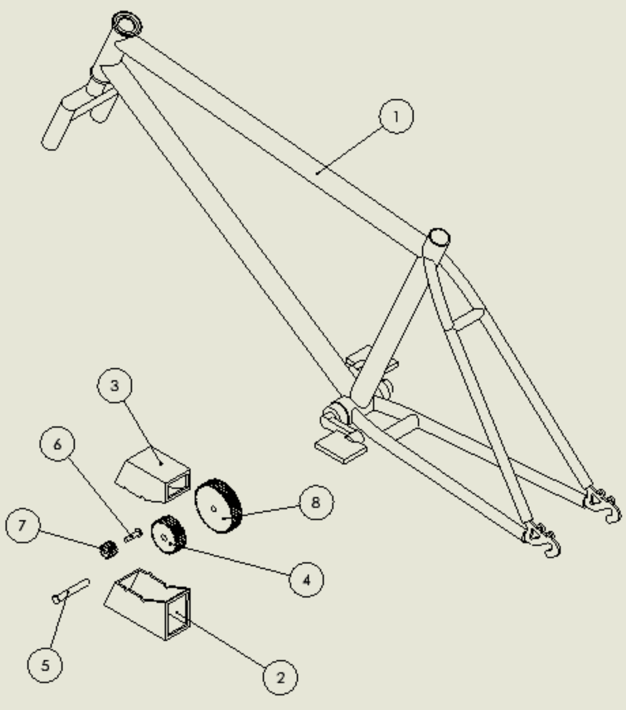
Appendix 1 – Assembly drawing



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	MECH 3409 Complete Bike Frame	Bike frame that holds the gearbox and is the main part of the ebike	1
2	Gear Holder Test	Bottom of the gearbox holder that rests on the bike frame	1
3	Gear Holder Test 2	Top of the gearbox holder to enclose the gears inside	1
4	Inch - Spur gear 10DP 40T 20PA 1.5FW --- S40N3.0H2.0L0.75N	Smaller gear in the compund gear and is driven by the pinion	1
5	Gearbox Pin#1.smg	This pin holds the compund gear in place	1
6	Gearbox Pin#2.smg	This pin holds the pinion in place	1
7	Inch - Spur gear 10DP 15T 20PA 1.25FW --- S15N3.0H2.0L0.5N	Smallest gear and is the driver (pinion)	1
8	Inch - Spur gear 10DP 60T 20PA 1.5FW --- S60N3.0H2.0L0.75N	Larger gear in the compund gear and is also driven by the pinion (also where the chain gets attached)	1

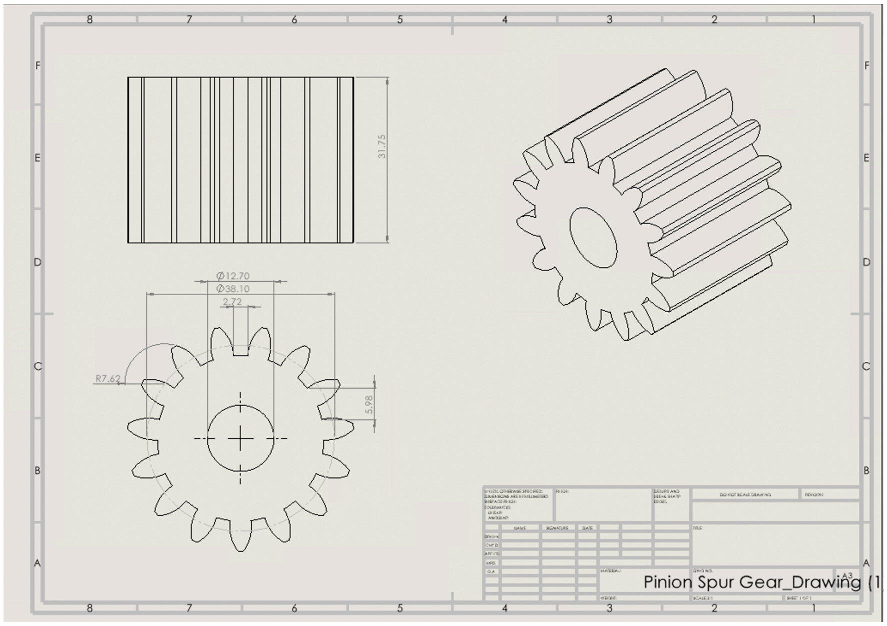
Final eBike Assembly Drawing

A3

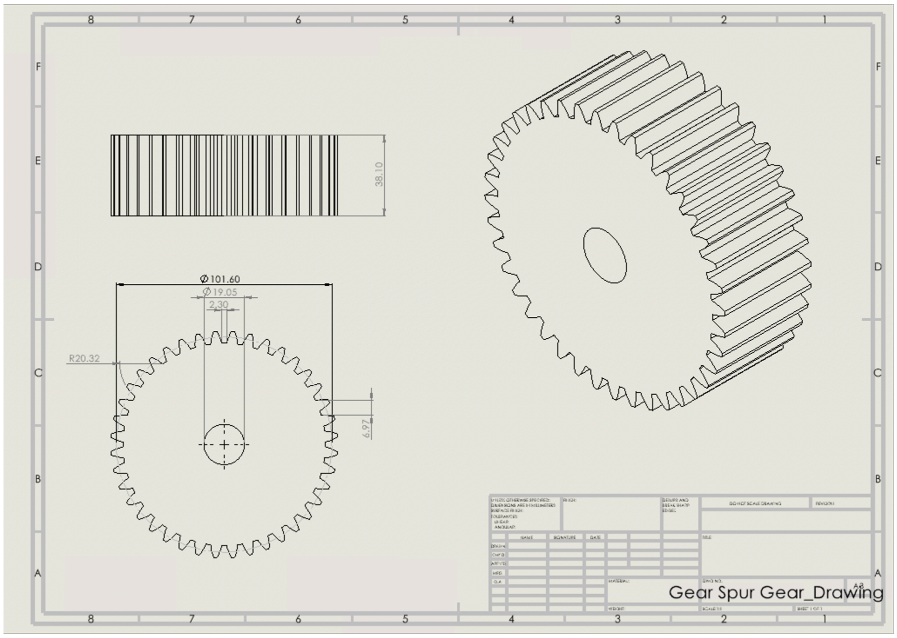
Appendix 2 – Engineering drawings of individual parts, including dimensions and tolerances

Gearbox Drawings

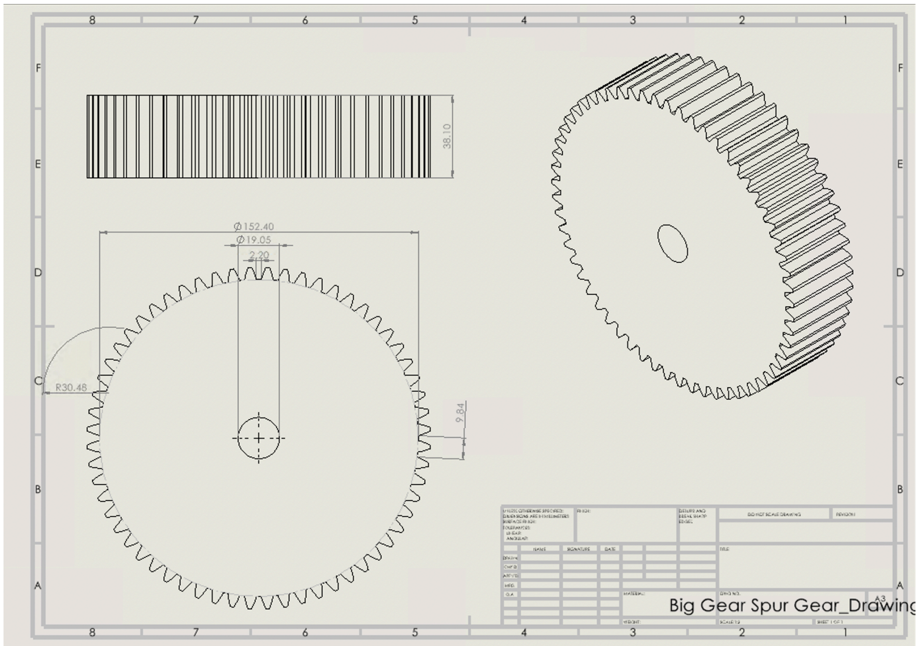
Pinion



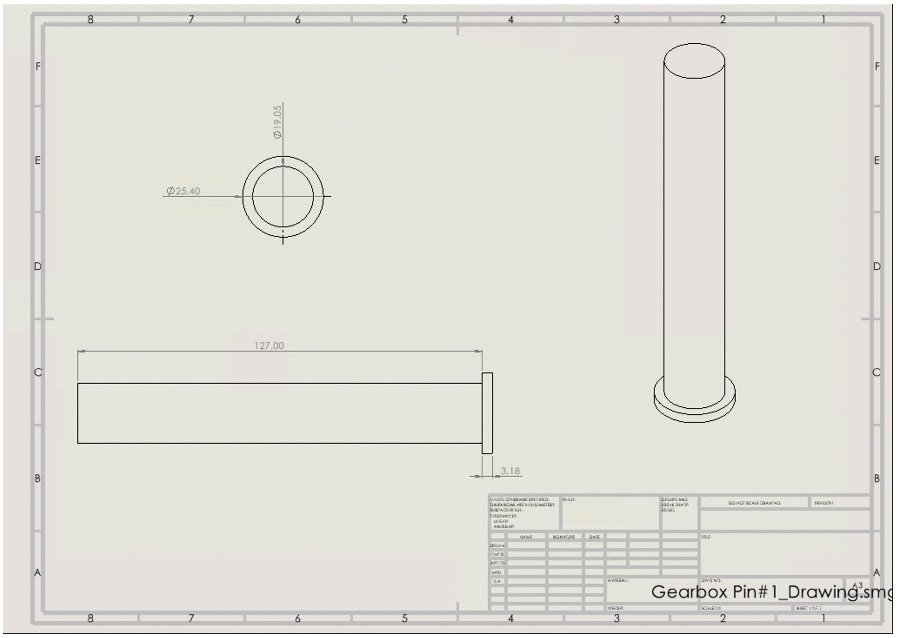
Medium Gear



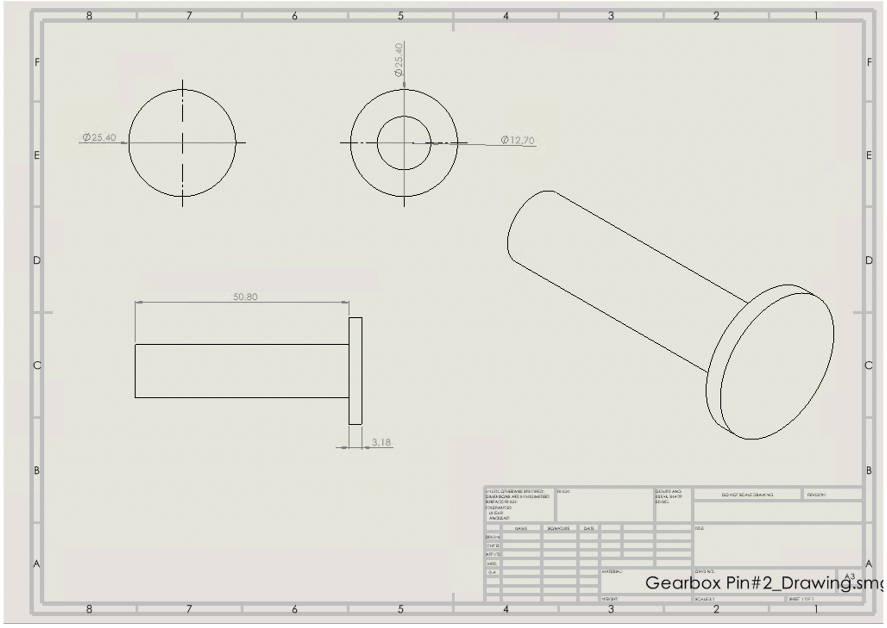
Big gear



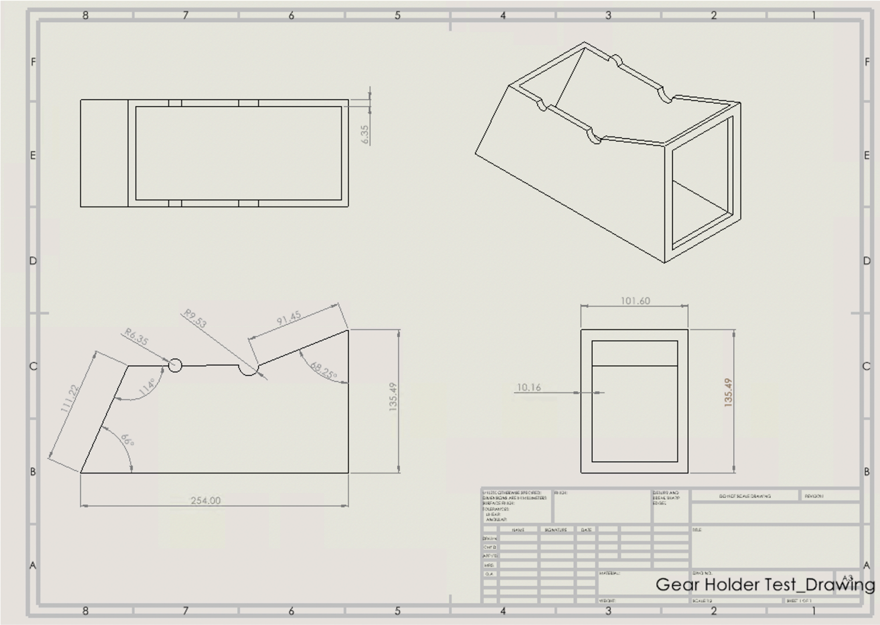
Pin #1


Ø19.05
Ø25.40
127.00
3.18
Gearbox Pin#1_Drawing.smg

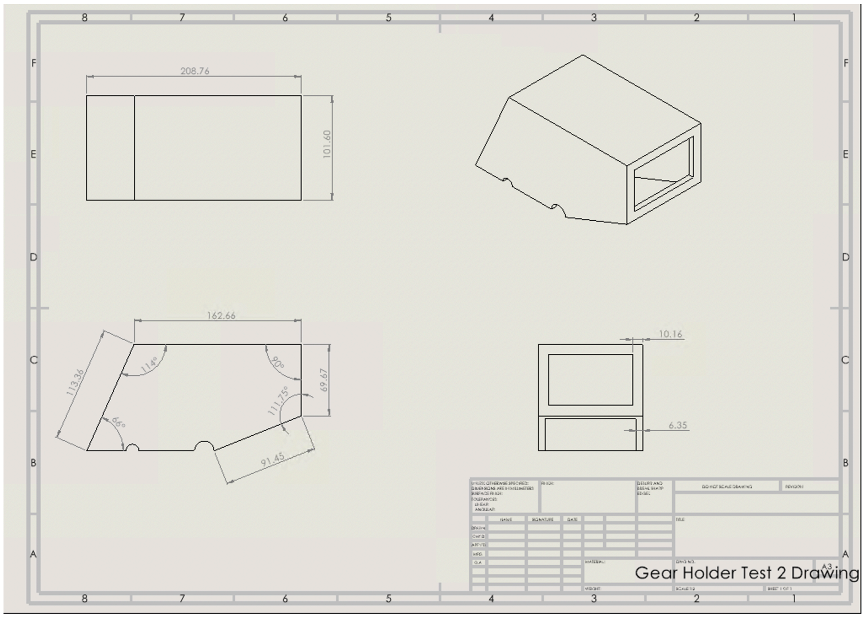
Pin #2


Ø25.40
Ø25.40
Ø12.70
50.80
3.18
Gearbox Pin#2_Drawing.smg

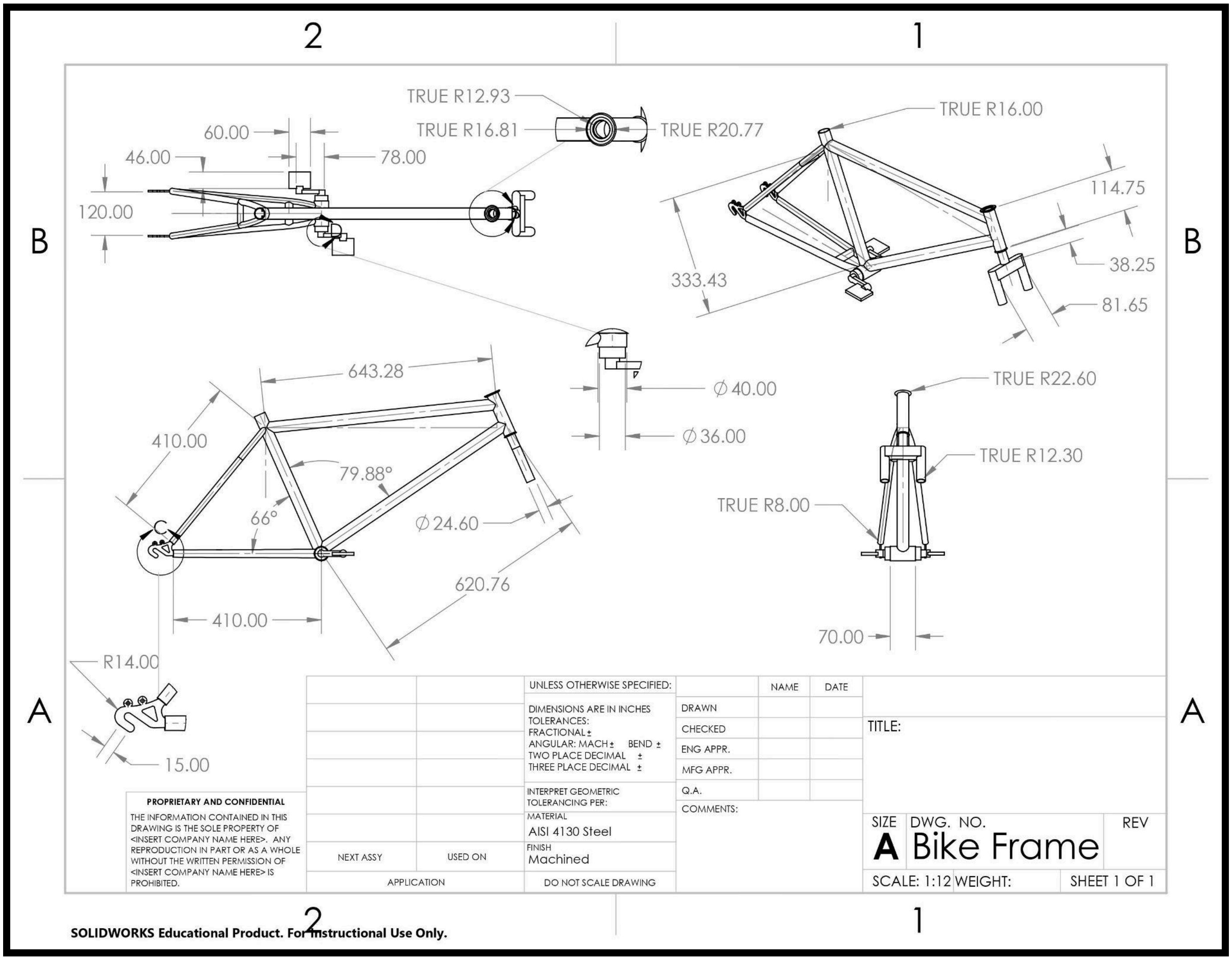
Bottom Half of Gear Holder



Top Half of Gear Holder

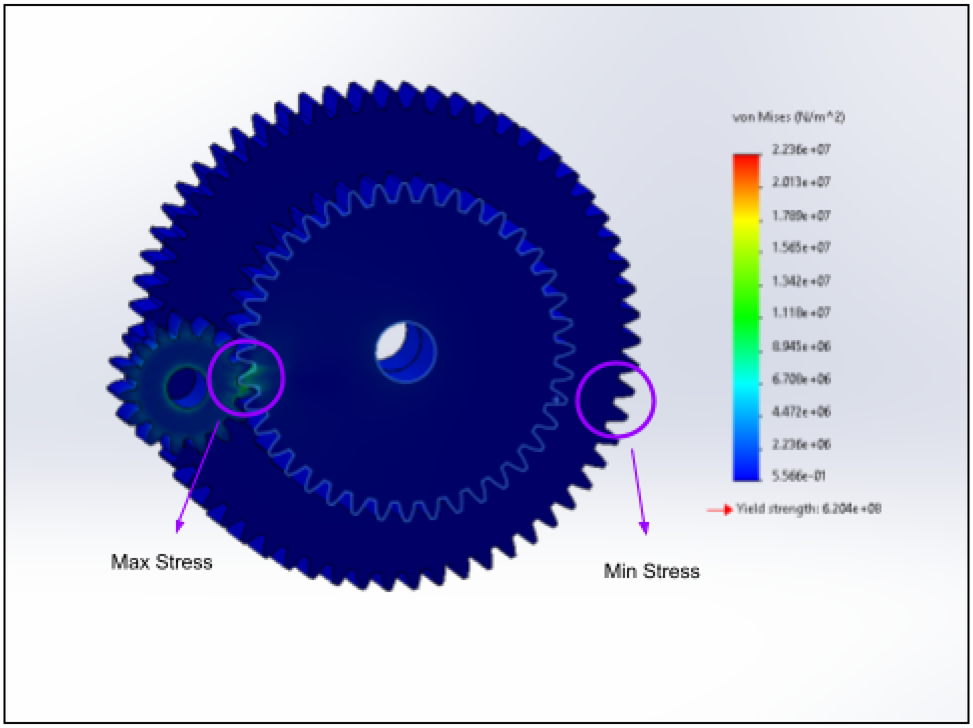


Bicycle Frame Drawing

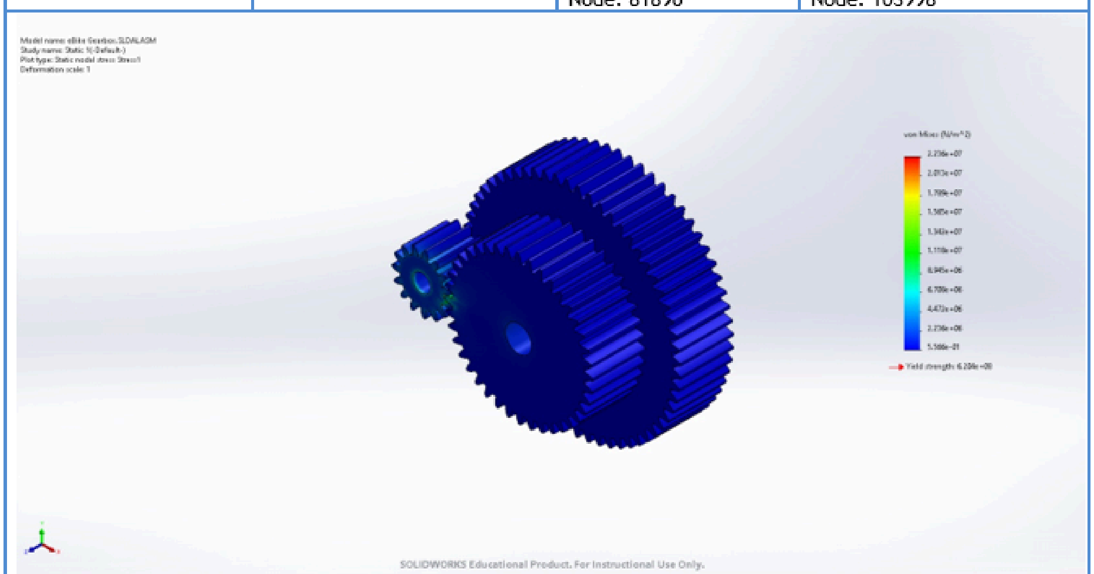


Appendix 3 – Detailed data from FEM

Gearbox FEA:

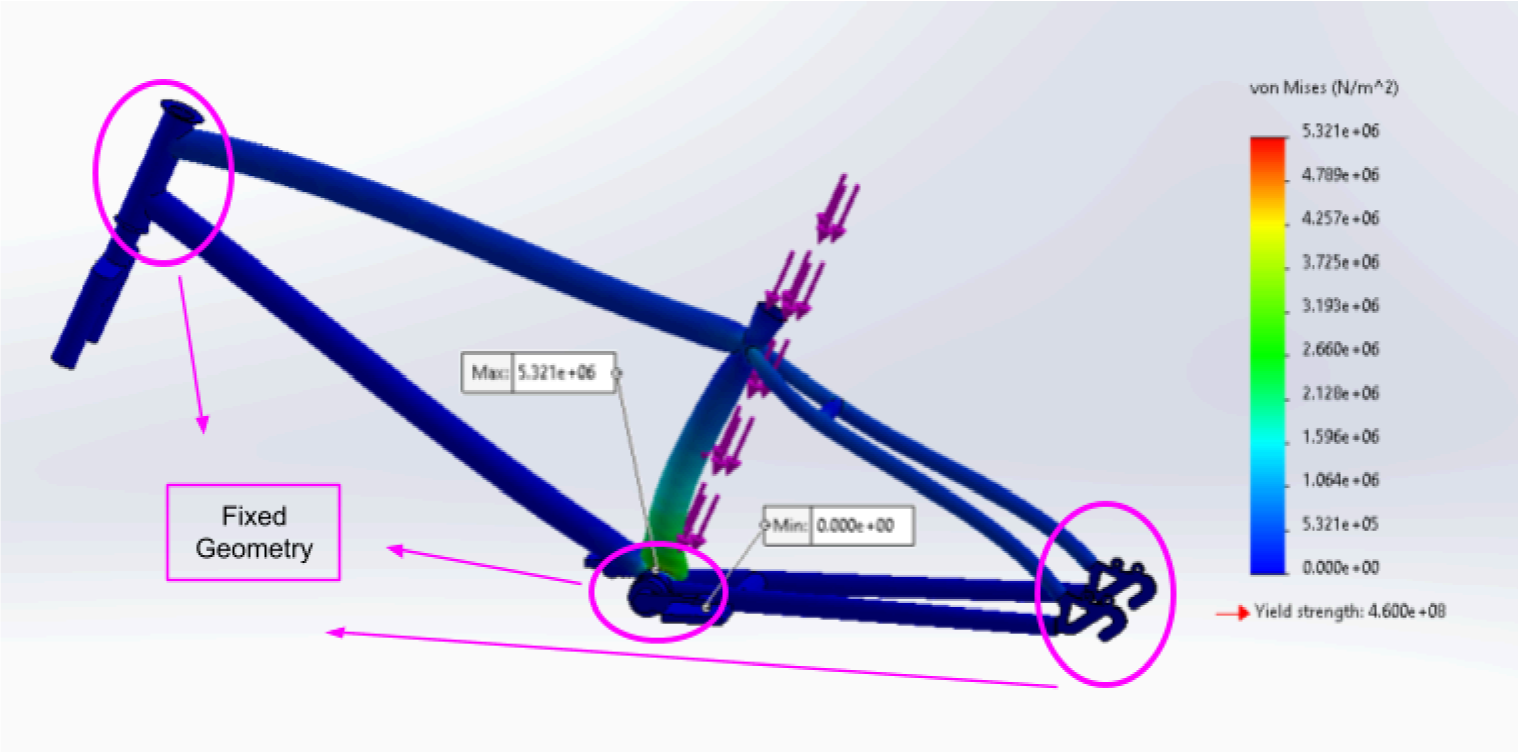


Name	Type	Min	Max
Stress1	VON: von Mises Stress	5.566e-01N/m^2 Node: 81896	2.236e+07N/m^2 Node: 103998



eBike Gearbox.SLDALASM-Static 1-Stress-Stress1

Frame FEA:


von Mises (N/m^2)
5.321e+06
4.789e+06
4.257e+06
3.725e+06
3.193e+06
2.660e+06
2.128e+06
1.596e+06
1.064e+06
5.321e+05
0.000e+00
Yield strength: 4.600e+08
Max: 5.321e+06
Min: 0.000e+00
Fixed Geometry

Appendix 4 – Analysis based on Excel spreadsheet or program

Gearbox Excel Spreadsheet:

Gear Fundamentals	Gear 1 (Pinion)	Gear 2	Gear 3
Face Width (F, in)	1.25	1.5	1.5
Pressure Angle (Φ)	20	20	20
Diametral Pitch (Pd, teeth/in)	10	10	10
Pitch Diameter (d, in)	1.5	4	6
Torque (T, lb.in)	88.51	88.51	88.51

Tangential Transmitted Load (Wt, lbf)	
Wt_1	118.013333
Wt_2	44.255
Wt_3	29.5033333

Interference of Spur Gears	Np_est (estimate)	Ng_est (estimate)	Gear Ratio (m)	N_P	N_G
Test 1	10	20	2	14.1607591	28.32151613
Test 2	12	30	2.5	14.6370799	36.59270146
Test 3	15	40	2.666666667	14.763407	39.36908533

For the biggest gear, after the final test (test 3) we decided that 60 teeth would be best to complete the compound gear.

AGMA Stress Equations				
Description	All	Gear 1	Gear 2	Gear 2
Bending Geometry Factor (J)	-	0.25	0.62	1
Surface Geometry Factor (I)	0.116874	-	-	-
Elastic Coefficient (Cp)	2300	-	-	-
Dynamic Factor (Kv)	-	1.26	1.43	1.57
Overload Factor (Ko)	1	-	-	-
Surface Condition Factor (Cf)	1	-	-	-
Size Factor (Ks)	-	1.031763328	1.050108711	1.05238402
Load Distribution Factor (Km)	1.01	-	-	-
Rim-Thickness Factor (Kb)	1	-	-	-
Transverse Diametral Pitch (Pd)	10	-	-	-
Face Width of the Narrower Member (F, i	-	1.25	-	-
Pitch Diameter of the Pinion (dw1, in)	-	1.25	-	-

(US Units)	Gear 1 (MPa	Gear 2 (MPa	Gear 3 (MPa)
AGMA Bending	4958.53134	866.068643	393.8731215
AGMA Contact Stress	66997.7798	44094.6478	37393.40118

AGMA Strength Equations			
Description	All	Gear 1	Gear 2
Bending Strength (St, psi)	-	66910	63045
Contact Strength (Sc, psi)	-	254500	238400
Hardness Ratio Factor (C_H)	1	-	-
Stress Cycle Factor (Y_N)	0.928346106	-	-
Stress Cycle Factor (Z_N)	0.948436889	-	-
Reliability Factor (K_R)	1	-	-
Temperature Factor (K_T)	1		-
Brinell Hardness	-	700	650
Number of Load Cycles (N)	100000000	-	-
Reliability (Rel)	0.99	-	-

Safety Factors	Gear 1	Gear 2	Gear 3
S_F	12.5270234	67.5784543	148.5950095
S_H	3.60276399	5.12777323	6.046718062

Bike Frame Excel Spreadsheet:

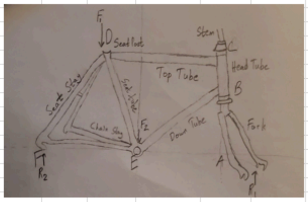
Materials	Ultimate tensile strength (Mpa)	Yield Tensile Strength (MPa)	Density (g/cm^3)	Poisson's ratio	Module Elasticity (GPa)
AISI 4130 Alloy Steel	670	435	7.85	0.27-0.30	205
Aluminum 6061	310	276	2.7		68.9

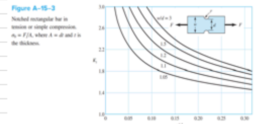
Applied Loads/Loading Conditions	weight (kg)	Load(N)
Total Weight	65	637.7
F1(load on seat)	55	539.6
F2(load on peddle)	10	98.1
R2		434.1
R1		203.6

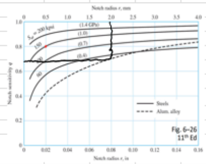
Location	Angle(Degrees)	Right Angle sect.
E	73.1	
F	53.8	45(above horizontal), 8.8(below)
D	53.1	22.1(above vertical), 30(below)

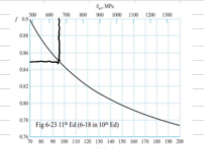
Frame Dimensions	Length (m)	Outer Diameter (m)	Inner Diameter (m)	Area (m^2)
AE(cos45)	0.687			
EF	0.423	0.016	0.012	0.0126
ED	0.430	0.032	0.028	0.0126
CD	0.420	0.032	0.028	0.0126
BC	0.101	0.032	0.028	0.0126
AC (sin45)	0.612			
AB		0.024	0.02	0.0126
FD	0.051	0.016	0.012	0.0126
AD(cos45)	0.768			
AF	1.110			

ED is the critical beam being looked at. So assume total force acting on it. Frame treated as beam/shaft









*Treat as 2D Problem

Stress Calculations	Loads (N)	Tension/Compression	Moment (N*m)(M_a, M_m)	$Stress_a$ (MPa)
EF	380.2	Tension		
DF	531.6	Compression		
ED	Highest/Critical = 637.7		257.2	193

No rotation, so assume no Torque

Machined	
a	3.04
b	-0.217

Notch	Steel	Aluminum
radius	2mm	2mm
Neuber √a	0.3271	
K_t	2.3	2.3
q	0.81	0.7

Strength Calculations	S_{ut} (Mpa)	S_e' (Mpa)	K_a (machined)	K_b (bending)	K_c (bending)	K_d (assume room temp.)	K_e (99.9)	K_f	S_e
4130	670	335	0.74	0.95	1	1	0.753	2.06	177.5
6061	310	155	0.88	0.95	1	1	0.753	1.91	97.1

Stress Calculation	$Stress_{max}$' (Mpa)	n_y	Yield check		f	a	b	N	n_f Goodman	n_f Gerber
4130	328.7	1.32	Pass	Better Material	0.855	1848.9	-0.170	299669	0.85	1.01
6061	305.4	0.90	Fail							

Assuming stress is unidirectional so $Sigma_a$ and $Sigma_m$ are equal meaning Sigma max is both combined. These are also repeated loads.

Finite Life predicted

Steel was the more reliable material.

AISI 4130	(Mpa)
Sigma_ar	217.7

AISI 4130 Steel	(Mpa)
Sigma_a	164.3
Sigma_m	164.3

6061 Aluminum	(Mpa)
Sigma_a	152.7
Sigma_m	152.7

A	1057345.2
B	1057345.2

Attribution Table

Part of Report	Group Member
Introduction	Maria
Idea Generation & Concept Selection	Brigette
Preliminary Design	Gearbox - Brigette Frame - Abiman

Analysis & Iterative Design	• Gear Analysis - Brigette & Maria
	    ○ Written description by Maria
	• Frame Analysis - Abiman & Khaalid
	    ○ Written description by Abiman

Final Design	Brigette & Maria
Appendix 1	Brigette
Appendix 2	Maria & Abiman
Appendix 3	Gearbox FEA - Brigette Frame FEA - Khaalid
Appendix 4	Gear Analysis - Brigette & Maria Frame Analysis - Abiman & Khaalid
CAD Designs	Gearbox - Brigette & Maria Frame - Khaalid & Abiman