Wheelchair E-Bike Attachment

Technical Report

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Summary

This report outlines the processes for designing and manufacturing an e-bike attachment for a wheelchair. The design is based on the Batec Mobility Scrambler 2 because of its versatile nature and ability for the user to remain in their wheelchair. Design was broken down into smaller components such as attachment mechanism, fork and wheel plates and mid-frame. FEA was used to conduct stress and fatigue analysis based on TBIS 4210 for each stage in the design. A frame prototype was produced using manufacturing equipment available such as 2D laser, tube bender and tube laser.

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Glossary

CAD	Computer Aided Design
E-Bike	Electric Bike
FEA	Finite Element Analysis
ISO	International Organization for Standardization
TBIS	Taiwan Bicycle Industry Standard

Introduction

Purpose

The purpose of this technical report is to develop an e-bike designed for wheelchair users. Specialized adaptive equipment can be a substantial cost to most people living with a disability. This project was a way to improve my design skills while also using available resources to reduce the cost of manufacturing my own equipment.

Background

I was born with a physical disability and use a wheelchair for mobility. Growing up I have enjoyed spending time outdoors but to be able to keep up with others on gravel roads or grass fields it takes a lot of effort to push a regular wheelchair. At times I was able to use golf-carts or side by side to make it easier, however those solutions still had their problems. They are not easily transportable, hard to transfer from chair to vehicle and do not have the ability to bring the wheelchair to the end destination.

In the current adaptive equipment market, there are a few options that try to solve this problem. The first is an attachment that connects to the wheelchair turning it into a trike and is powered by a hub motor. This attachment is made by Batec Mobility and costs upwards of $10 000 before options.

Figure 1 Batec Scambler Attached to Rider

Figure 2 Batec Scrambler 2

The second option available on the market is a four-wheeled electric Rig that utilizes many off-the-shelf bicycle parts to reduce end user cost. The Rig requires the user to transfer out of their chair but has room to carry it on the back. The Rig produced by Not A Wheelchair costs $9 500 for the highest options chosen.

Figure 3 The Rig

The final option is a reverse trike designed for the adaptive mountain biking community. The Bowhead Reach is also powered by an electric motor and has full articulating front suspension. The Reach also requires users to transfer and cannot transport a wheelchair. Its price starts at $14 999.

Figure 4 Bowhead Reach

After researching what was available on the market, I believe the Batec Scrambler to be the best option because I can remain with my wheelchair and transport it to different locations.

Scope

This report documents the wheelchair e-bike attachment design and manufacturing project. The scope of this project was to design the e-bike frame for a wheelchair like the Batec Scrambler 2 and manufacture a prototype.

The design and manufacturing can be broken down into the following parts and tasks:

- Frame Design
    - Determine motor and wheel criteria
    - Design and model attachment mechanism
    - Design and model wheel plates
    - Design and model mid frame including adjustment plates and stand
- Stress Analysis
    - Locking Plate
    - Front fork and wheel plates
- Fatigue Analysis
    - Front Fork and wheel plates
    - Mid frame
- Cost Analysis
    - Raw Material
    - Purchased parts with suppliers
    - Labour

- Manufacturing
    - Assembly Drawing
    - Part Drawings
    - Welding Fixtures
    - Assembled Prototype

Motor and Wheel Selection

Using Grin Technologies motor information, the eZee front hub was chosen for this application. It is a geared 36V-250 rpm motor that has good torque suitable for cargo hauling and steep hills. Having good torque is important since the frame will be attached to a wheelchair and there will be more rolling resistance and weight as compared to a typical bike. A 20” rim was chosen because it would be better suited for off-road capabilities and easier to acquire tires for.

Based on this selection a theoretical unloaded speed that could be reached is 31 km/h from calculations in (Appendix B). A more accurate theoretical top speed of 25 km/h was determined using Grin Technologies motor simulator that considers the motor, controller, battery and rolling resistances.

Attachment Mechanism

Loads

The attachment point of the frame is located 140 mm behind the footplate of the wheelchair. A scale was used to measure the max weight as the user was leaning as far forward as possible to mimic a worst-case weight distribution. The result was a weight of 43 kg on the footplate. In the actual design the chair will be tilted back 15 degrees distributing more weight onto the wheelchair tires and off the e-bike.

Figure 5 Wheelchair on Scale

Hardware

This design requires the use of pins to lock the chair to the e-bike attachment. To avoid machining customs pins, standard 5/16" clevis pins were chosen because of their length and availability. Based on hand calculations the pins would not permanently deform under shear stress (Appendix C). Based on the same calculations 5/16" bolts were chosen for the locking plate pivot point. The bolts are partially threaded to allow for the non-threaded portion to take the shear force rather than the threads that would have a higher stress concentration. 5/16" bolts were standardized for all remaining bolted components on the prototype. Shear calculations for the wheel plates determined the 5/16" were acceptable (Appendix C).

Design

The design concept for the attachment mechanism is based on the Batec Mobility system. It allows the e-bike to quickly attach or detach from the wheelchair with a single lever that releases the lock.

Figure 6 Batec Attachment Mechanism

The attachment mechanism is not only there to lock the e-bike to the wheelchair but is also crucial in lifting the front castor wheels off the ground. A phone level app was used to measure the angle required to lift the front castors of the wheelchair off the ground. A 15° angle lifted the castors approximately 80 mm off the ground. Once the angle and the 85 mm spacing between pins was decided the remaining dimensions and geometries of the locking plate and angle plate were determined through 3D printing trial and error until the lock worked smoothly.

Figure 7 3D Printed Test Parts

After finalizing the dimensions, tab and slots were used to connect the individual plates to make alignment for welding easier. The center 1.25” square tube has 4 holes 1” apart to allow the frame length to be adjusted incrementally.

Plastic washers are used between the locking plates and angle plates to avoid metal on metal binding.

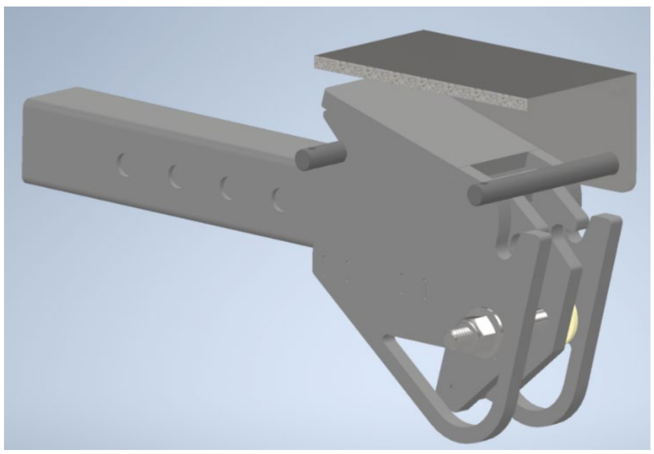

Figure 8 Attachment Mechanism CAD Model

Stress Analysis

Loads and Constraints

A fixed constraint was used on the bolt hole of the locking plate. Initially a pin constraint was tired but was underdefined for this analysis. It was assumed that the entire 1390 N calculated from the wheelchair weight could rest on a single plate. The load was angled slightly off vertical to replicate the position the pin would rest in the locked position.

FEA Results

The plate thickness was determined by simulating various ones. Using 0.25” thick plate resulted in a max stress of 157 MPa and a 0.02 mm displacement. The max stress falls below the yield stress of A36 steel (Appendix D).

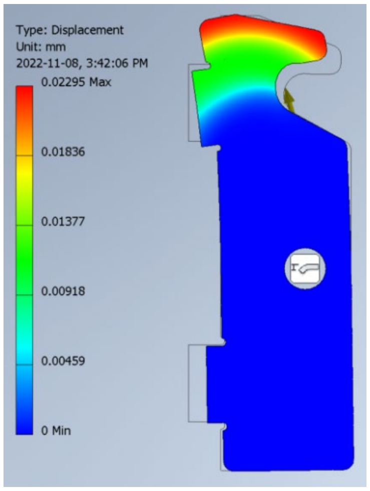

Figure 9 Locking Plate FEA Displacement

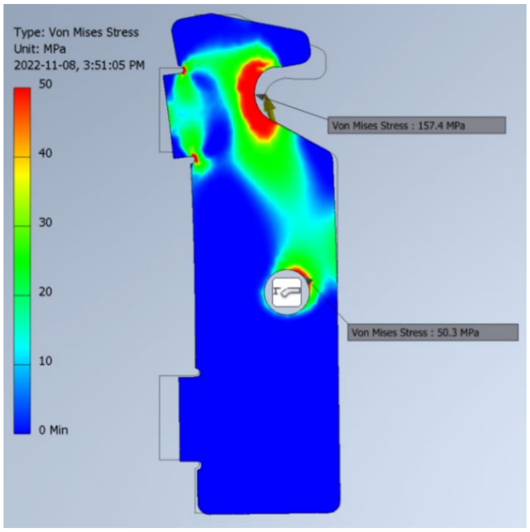

Figure 10 Locking Plate FEA Stress

Fork and Wheel Mounts

Constraints

To make manufacturing easier going further all laser cut plates will be 0.25” thick to match the locking plate allowing all parts to be cut together when manufactured. The motor required the fork dropouts to be 100 mm apart and have 10 mm slots to accept the axle posts (Appendix E). Dropouts needed to be positioned to avoid the 20” wheel interfering with the fork bend.

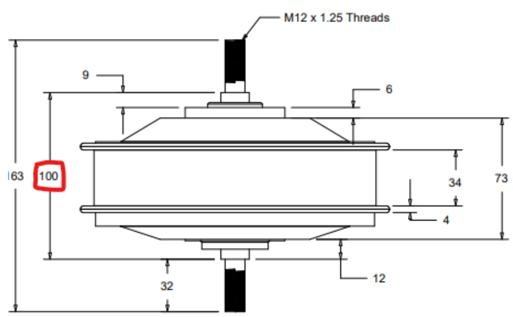

Figure 11 eZee Motor Dimensions

Repurposing an existing bike front fork and steering tube would simplify the design. An old kick scooter was sourced from Facebook Marketplace because it has a longer steering tube for a battery mount that most bikes do not have.


Figure 12 Kick Scooter

Fork Mounts

An important aspect of this design was to have most of the critical components bolt together so if something did not work on the physical prototype it could be easily replaced. The fork mounts weld onto the scooter forks but allow for the wheel plates to bolt on. The outside face dimensions between fork mounts when welded meets the 100 mm requirement to ensure that if the wheel plates ever get cut in a different thickness the motor will always fit. Lower tabs were included to add ballast for traction and reduce the front wheel wobbling at higher speeds.

Figure 13 Fork Mount CAD Model

Wheel Plates

The front wheel plates were designed with a 100 mm offset and 40° rake angle to allow the 20” wheel to fit the repurposed fork while maintaining a reasonable handlebar position for the user.

A disk and caliper system were chosen for brakes instead of rim friction brakes. IS brake mount dimensions (Appendix F) were used on the left wheel plate to ensure that standard bike calipers could fit this fork system.

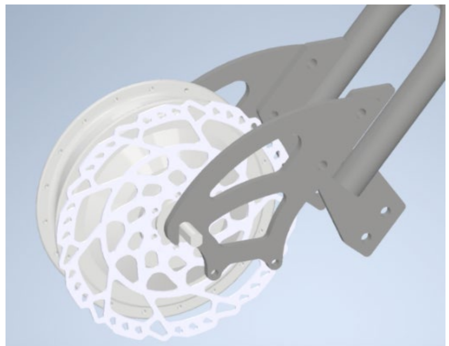

Figure 14 Wheel Plate CAD Model

Stress Analysis

The following stress analysis tests are based on the TBIS which uses the ISO standard 4210 as their basis. In all FEA simulations there are high stress concentrations located around the fork mounts where welds could not be accurately represented. These stresses will not be considered in the analysis.

Where loads are placed and how they are split up can influence FEA results. There were two options for loading the front forks of the model. First, were two individual loads acting on the dropouts (Figure 15) and the second uses a test bar with a single load at the center (Figure 16). Both options cause different amounts of torque on the fork tubes that results in a higher stress for the test bar load. Going further, the test bar loading scenario will be used for all FEA analysis because it is a closer representation of having a load acting on a wheel axle.

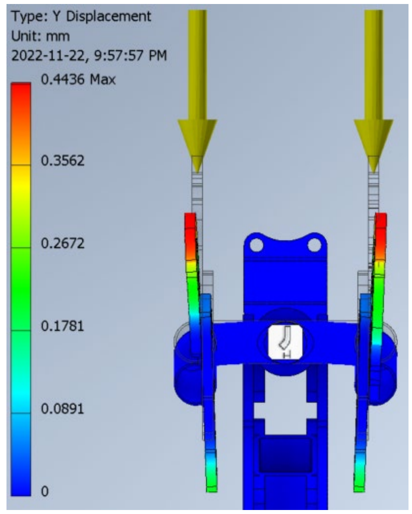

Figure 15 Two Individual Loads Analysis

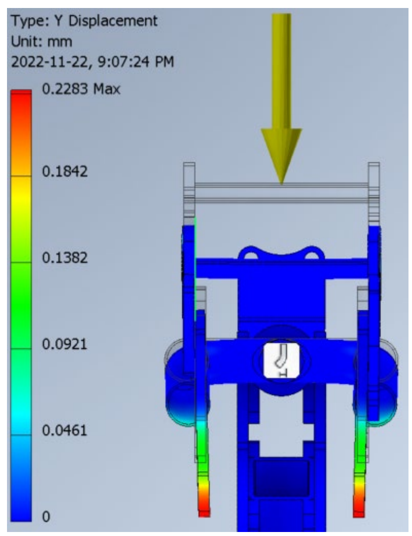

Figure 16 Single Load With Test Bar Analysis

Motor Torque

Loads and Constraints

An electric motor can cause a lot of torque to bicycle dropouts and often requires a torque arm to avoid axle spin. The eZee motor puts out a max torque of 63.7 Nm during acceleration according to Grin Technologies motor simulator. The torque acts in the opposite direction of the wheel direction in the FEA model. Pin constraints were used on the bolt locations.

FEA Results

The dropouts had a max stress of 93 MPa (Figure 17) which falls below the yield strength of A36 steel (Appendix D), and the displacement is negligible at a max of 0.03 mm (Figure 18). There is a higher stress concentration in the test bar because Inventor does not allow material properties to be increased to infinite strength. If the test bar was sufficiently strong, then a higher stress would be expected in the wheel plates.

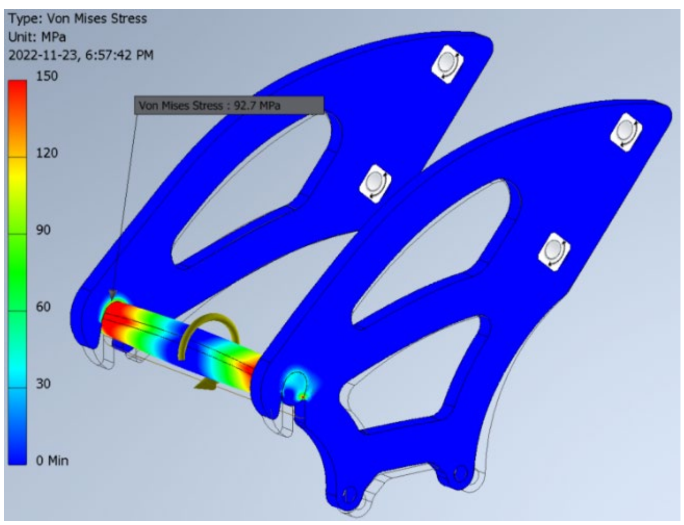

Figure 17 Wheel Plate FEA Von Mises Stress

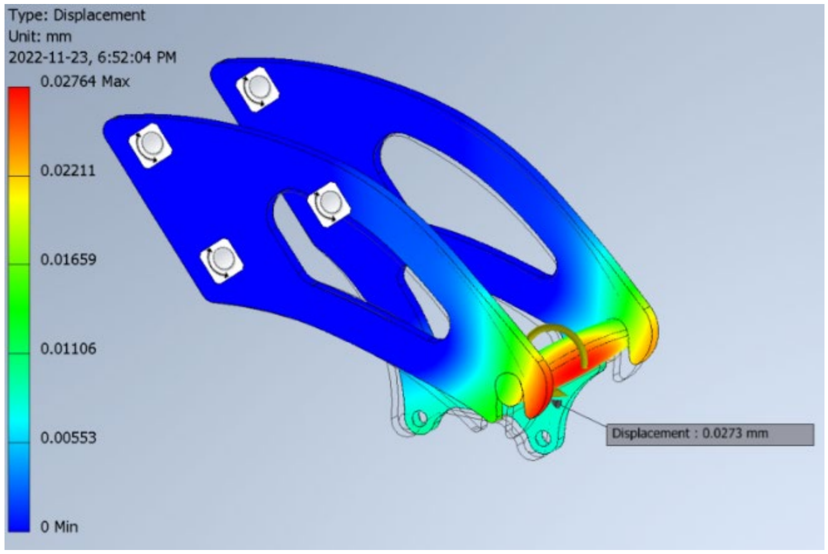

Figure 18 Wheel Plate FEA Displacement

Static Bending Test

Loads and Constraints

According to the TBIS front fork static bend test the steering tube is fixed horizontally. A 1000 N load is applied vertically in the downwards direction on a test bar that puts a distributed 500 N to each side.

Hand Calculation Results

For manual calculations the front fork was simplified to a single tube acting as a cantilever beam. The 1000 N load was halved to 500 N since the analysis is on only one of two tubes. This simplification resulted in a Von Mises stress of 274 MPa (Appendix G). Conducting FEA on a simplified model resulted in similar results to the hand calculations (Figure 19).

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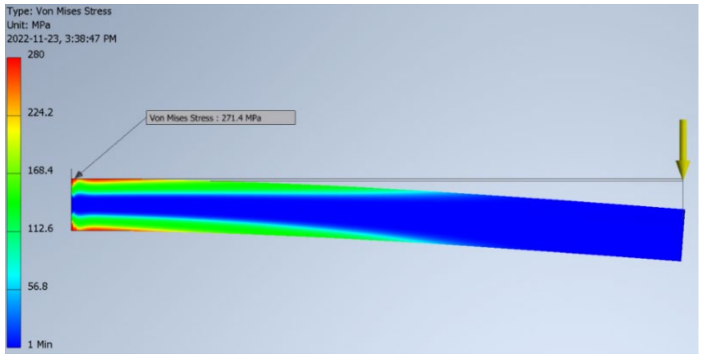

Figure 19 Static Bending Test Simplified FEA

FEA Results

A max stress of 391 MPa (Figure 20) was seen in the scooter fork that exceeds the yield strength of 1020 steel but remains below the ultimate strength. The fork would plastically deform due to this stress but not fracture. A max stress of 73 MPa in the wheel plates falls below the yield stress of A36 steel. These FEA results do not match the hand calculations because of the added geometry and load complexity. As discussed earlier the test bar adds a torque on the tube that is neglected in the hand calculations

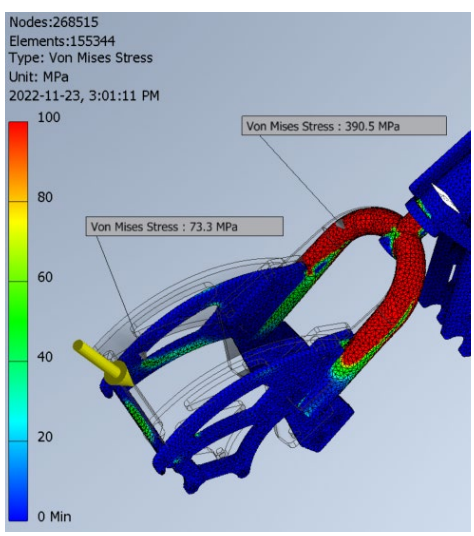

Figure 20 FEA Static Bending Test Results

Fatigue Analysis

Loads and Constraints

According to the TBIS front fork – bending fatigue test the steering tube is fixed horizontally and a ±450 N load is applied vertically on the test bar. The TBIS requires the fork to last 120 000 cycles with no fractures for this test.

Hand Calculations

For manual calculations the front fork was simplified to a single tube acting as a cantilever beam. The 450 N load was halved to 225 N since the analysis is on only one of two tubes. This simplification resulted in a Von Mises stress of 123 MPa (Appendix G). Conducting FEA on a simplified model resulted in similar results to the hand calculations (Figure 21).

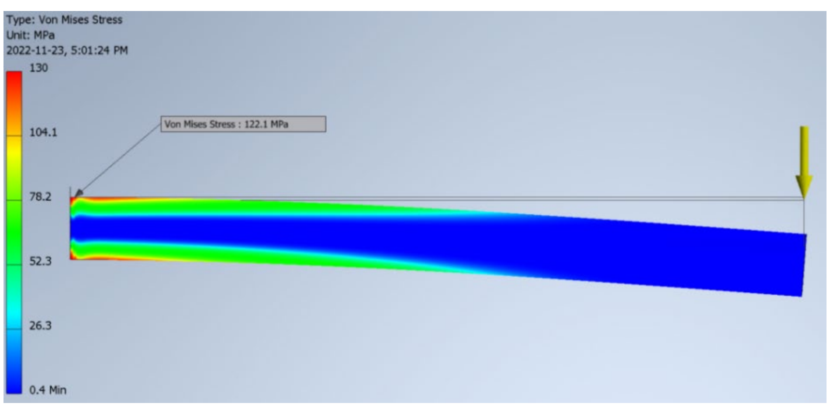

Figure 21 Fatigue Test Simplified FEA

FEA Results

The wheel plates had a max stress of 33 MPa (4.8 ksi) and 178 MPa (26 ksi) in the fork tube (Figure 22). Plotting both stresses on an s-n curve for 1020 steel

(Figure 23) results in an infinite life cycle which exceeds the TBIS requirement. The stress in the wheel plates was plotted as 1020 steel instead of A36 because s-n curves are rare to find for specific materials. Since the stress is so low it can still be assumed to have an infinite life. Like the static bending test, the hand calculations do not match the assembly FEA because of the complex geometry and torque from the test bar.

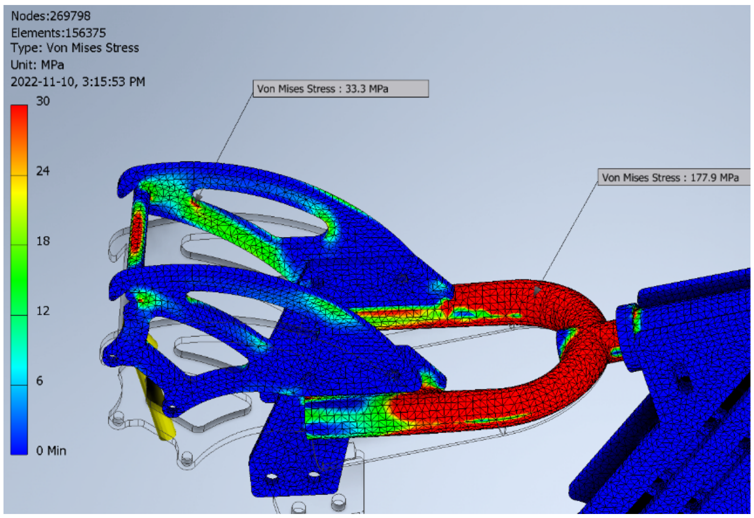

Figure 22 FEA Bending Fatigue Results

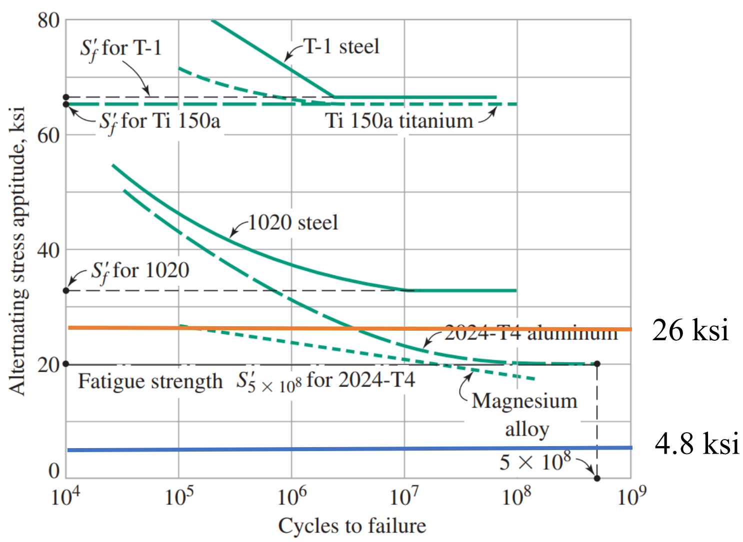

Figure 23 S-N Curve for various metals

Mid Frame

Design

The final part of the frame was designing the connection point between the steering fork and the attachment mechanism. A simple stand was designed to keep the frame upright when not connected to a wheelchair and act as a mudguard when a plastic cover is added in the future.

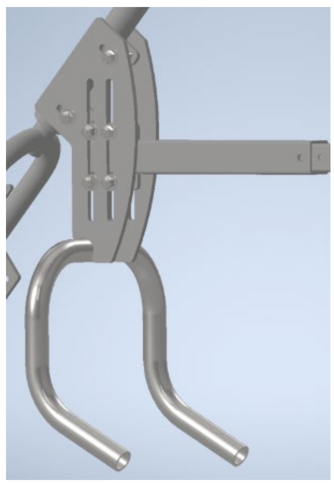

Figure 24 CAD Model Mid Frame

To remain consistent with the fork design, the adjustment plates (Figure 25) are meant to be fully bolted together so that the fork or plates themselves could be easily changed for future iterations. Slots allow for 3.75” of vertical travel to ensure compatibility with different wheelchair heights. Another two smaller slots serve two functional purposes. First, they allow for discrepancies between perfect CAD locating and real world locating of the steering tube tabs after welding. Second, it allows for the rake angle of the steering tube to have a slight adjustment allowance.

Figure 25 Assembled Prototype Adjustment Plate

Full Frame Analysis

Stress Analysis

Loads and Constraints

The frame stress analysis was used to determine the max G's that the different frame components could handle before yielding. Hand calculations (Appendix G) were done to calculate the reaction force of 264 N that the weight of the frame, battery and rider would put on the front dropouts. This force is at 1 G and used in FEA to find the initial stress in each component. From the initial stress in the fork tube and adjustment tube it was estimated based on the materials yield strength that the frame could handle 5 G's. A load of 1321 N was then used in FEA to verify the stress at 5 G's.

The attachment mechanism was constrained at the pin locations where it mates to the wheelchair.

FEA Results

The 1 G analysis (Figure 26) showed a Von Mises Stress of 65 MPa in the front forks, 52 MPa in the adjustment tube and 21 MPa in the angle plates. These values were used to determine the estimated max G's the frame could handle.

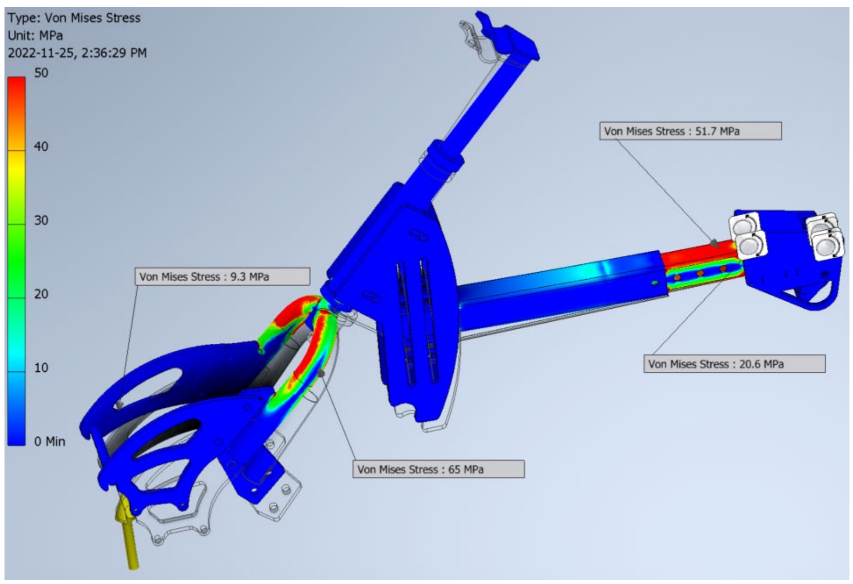

Figure 26 1 G FEA Von Mises Stress

A second analysis (Figure 27) was done to verify the estimated 5 G's from the hand calculations (Appendix G). Results showed a Von Mises Stress of 325 MPa in the front forks, 259 MPa in the adjustment tube and 103 MPa in the angle plates. These results remained below the yield strength of 1020, A500 and A36 steel respectively.

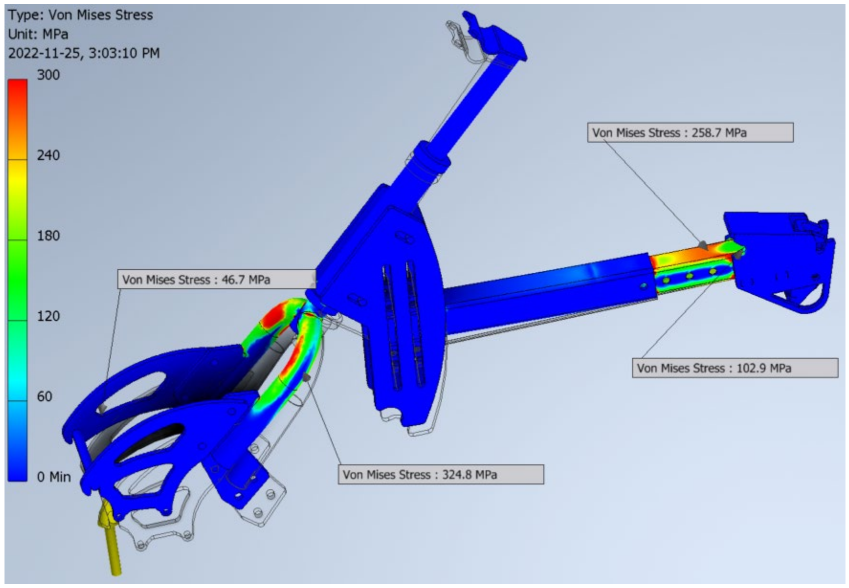

Figure 27 5 G FEA Von Mises Stress

Fatigue Analysis

Loads and Constraints

According to TBIS frame – fatigue test with horizontal forces the rear dropouts are pinned, and the front dropouts are a guided roller. To adapt the constraints for FEA, the attachment mechanism has pinned constraints and the front dropouts are left unconstrained since a roller does not prevent rotation in FEA.

Hand Calculations

To do manual calculations on the mid frame the 1.25" square tube was simplified to a single tube acting as a cantilever beam. This simplification resulted in a Von Mises stress of 23 MPa (Appendix G). Conducting FEA on a simplified model resulted in similar results to the hand calculations (Figure 28).

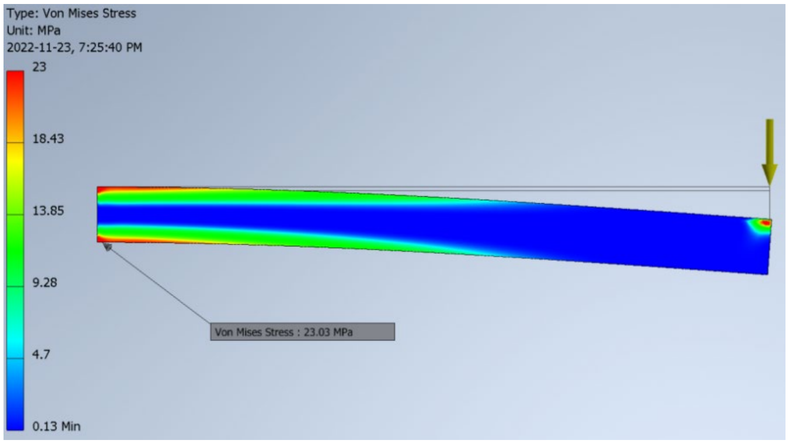

Figure 28 Adjustment Tube Simplified FEA

FEA Results

The mid frame had a max stress of 26 MPa (3.77 ksi) (Figure 29) that would result in an infinite life based on the earlier S-N curve. When neglecting the higher stress concentrations where the 1.5” tube transitions to the 1.25” tube the stress matches the hand calculations.

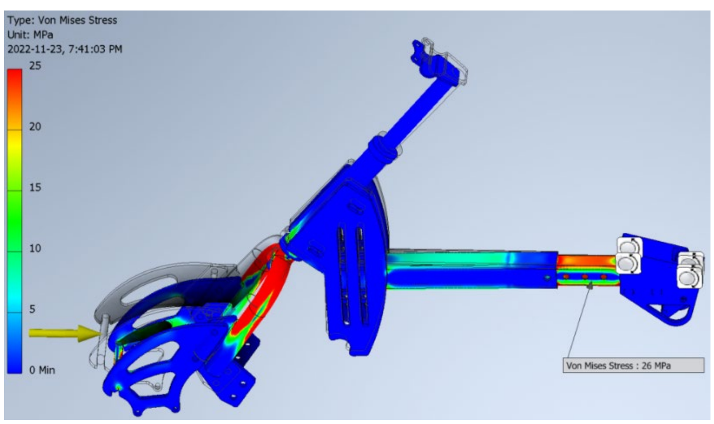

Figure 29 FEA Frame Fatigue Results

Manufacturing

Laser Cutting

A 2D fiber laser was used to cut the 0.25" sheet metal parts. All the parts were cut to the same thickness so that they could be nested together rather than waiting for different sheet thicknesses to run through production. Minor post processing such as removing mill scale was done before it moves to welding or assembly.

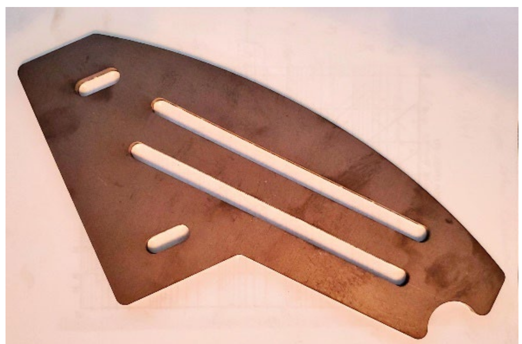

Figure 30 Laser Cut Adjustment Plate

Tube Cutting and Bending

To manufacture the stand the point coordinates from the drawing in (Appendix H) were first programed on the BLM E-Turn52. This allowed the operator to set bending parameters and visualize if the bend was possible to do. Five millimeters had to be added to the side straight sections to allow the machine's clamp enough room to grab the tube. From the bending program an estimated straight cut length was given to the tube laser. The straight tube was then cut using a BLM LT8 Fiber laser (Figure 31) and sent back to be bent.

Figure 31 Tube being Laser Cut

Welding

Two fixtures were designed to ensure proper location of the fork mounts and steering tube tabs (Figures 32 and 33). Both fixtures were 3D printed and intended to be used only for tacking purposes and then removed to complete the welds. Tig welding was the chosen method of welding to avoid burning through when joining the 0.25" plates to 0.049" wall tubing.

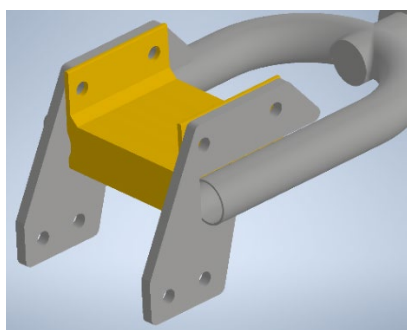

Figure 32 Fork Mount Weld Fixture

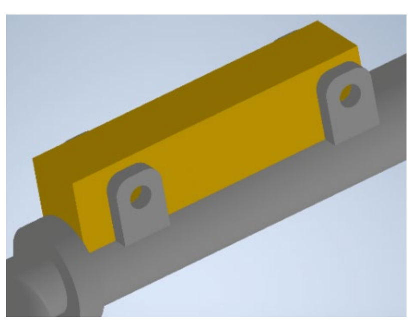

Figure 33 Steering Tube Tab Weld Fixture

In real practice the steering tube tab fixture was clamped down to the fabrication table and initially tacked (Figure 34). The heat from just four tacks was enough to melt the PLA print and cause the tabs to warp inwards. The printed fixture had to be broken free and replaced with a 1” wide aluminum block to maintain the proper spacing and alignment between the tabs shown in (Figure 35).

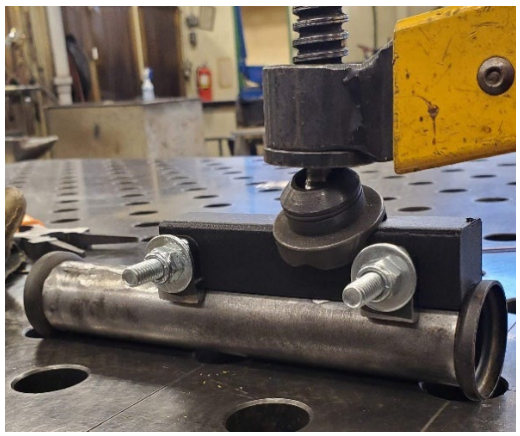

Figure 34 Steering Tube Tabs Welding Setup

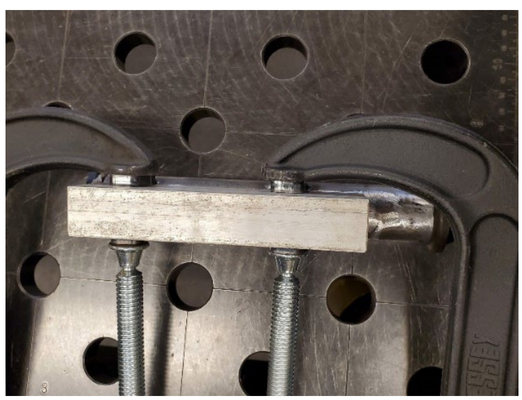

Figure 35 Clamped 1" Spacer

When welding the fork plate, the printed fixture also melted and warped but the 100mm dropout width was maintained. However, after assembling the wheel plates the dropout width decreased to 94 mm over the added distance.

Cost Analysis

Raw materials for this project would cost $157.8 when sourcing from Metal Supermarkets. The cost per foot of material could be decreased if bought in bulk from a steel distributor. Metal Supermarkets have added fees because they cut material to size.

The purchased parts cost $2 191.95 which was the only real upfront cost of this project. All other costs in this analysis are theoretical for the purposes of calculating a prototype cost.

The final cost category is the services that cost $10 995. These subtotals are all estimates based on the work that was completed for this project. Shop rates for tube bending and laser cutting were not included because they were provided by Mercury Specialty Products.

A completed wheelchair e-bike prototype would cost $13 345 without machine time cost. Excluding the electronics, the prototype frame would cost $11 210. Although the cost is high and in the range of other adaptive bikes on the market the initial design cost would be spread out in production, bringing down the price per unit.

Wheelchair E-bike Cost Breakdown						
Raw Materials	**Cost**	**Unit**	**QTY**	**Subtotal**	**Source**	**Comments**
1/4" Steel Sheet	$ 119.69	ea.	1	$ 119.69	Metal Super Markets	19.5" x 19.5" stock estimate
1.5"x0.1 SQ Tube	$ 1.17	in	12	$ 13.98	Metal Super Markets	
1.25"x0.083" SQ Tube	$ 2.63	in	8	$ 21.05	Metal Super Markets	
3D Printer Filament	$ 0.03	gram	110	$ 3.08	Amazon	
			Total:	$ 157.80		
Purchased Parts	**Cost**	**Unit**	**QTY**	**Subtotal**	**Source**	**Comments**
Scooter Frame	$ -	ea.	1	$ -	FB Marketplace	Received for free
Clevis Pins	$ 14.99	ea.	1	$ 14.99	Princess Auto	Assorted kit - Needed 5/16"
Sping	$ 14.99	ea.	1	$ 14.99	Princess Auto	Assorted kit
Horizontal Bar Clamps	$ 21.27	ea.	1	$ 21.27	Amazon	
Electric Motor	$ 491.00	ea.	1	$ 491.00	Grin Technologies	MeZee250F_L10
Disk Rotor	$ 12.00	ea.	1	$ 12.00	Grin Technologies	Magura 6-bolt, 203mm
Wheelbuild	$ 80.00	ea.	1	$ 80.00	Grin Technologies	
Battery	$ 932.00	ea.	1	$ 932.00	Grin Technologies	B3620Li-DT, 36V 19.3 Ah
Battery Charger	$ 50.00	ea.	1	$ 50.00	Grin Technologies	36V 2.0A ST3
Controller	$ 320.00	ea.	1	$ 320.00	Grin Technologies	V5 Baserunner_L10
Throttle	$ 19.00	ea.	1	$ 19.00	Grin Technologies	T-Htwist_Slim_50cm
Torque Arm	$ 25.00	ea.	1	$ 25.00	Grin Technologies	
Assorted Cable/Display	$ 206.00	ea.	1	$ 206.00	Grin Technologies	
5/16" Nyloc Nuts	$ 0.07	ea.	15	$ 1.00	Mid-Canada Fasteners & Tools	
5/16" Hex Bolt Grade 5	$ 0.31	ea.	7	$ 2.19	Mid-Canada Fasteners & Tools	Partially Threaded
5/16" Hex Bolt Grade 5	$ 0.15	ea.	8	$ 1.19	Mid-Canada Fasteners & Tools	Full Thread
Washers	$ 0.06	ea.	23	$ 1.32	Mid-Canada Fasteners & Tools	
			Total:	$ 2,191.95		
Services	**Cost**	**Unit**	**QTY**	**Subtotal**	**Source**	**Comments**
Design	$ 100.00	hour	100	$ 10,000.00	Student	Estimate (8wks averaging 10hrs)
FEA Analysis	$ 80.00	hour	10	$ 800.00	Student	Estimate
Welding	$ 25.00	hour	6	$ 150.00	Student	Estimate
Labour	$ 15.00	hour	3	$ 45.00	Student	Estimate
Machine Time	$ -	hour		$ -	Mercury Specialty Products	Shop Rate N/A as employee
		Total:		$ 10,995.00		
		Grand Total:		$ 13,344.75		

Figure 36 Cost Analysis

Conclusions

The original scope of designing an e-bike frame was exceeded by including manufacturing and building a physical prototype. A design was created around the use of available manufacturing resources and using standardized components.

The FEA stress analysis concluded that under a 1000 N static bending load the front forks would plastically deform. It was also found that forks and adjustment tube could endure up to 5 G's. Fatigue analysis concluded that all components could have infinite life cycles, exceeding the TBIS minimum life cycles.

Manufacturing showed that welding heat can play a big factor for fixturing and final dimensions. Working collaboratively with operators and welders helps improve future design considerations.

This project showed me how difficult it is to simulate the real world without gathering real world data. I learned that aspects of design and manufacturing are out of your control and it's a process. In the end this project has created a prototype that is a solid base for future iterations and further development.

Recommendations

Design

To improve on this project the fork should be designed and fabricated for this use case rather than salvaging an existing scooter fork. Key components such as bearings and bearing guides can be reused or purchased but the fork itself should be bent from a thicker walled tubing more than 0.049" and use a steering tube with a thicker wall as well. The fork should also be longer so that the wheel plates can be shorter. A thicker walled tube will not only increase the strength but also reduce the burn through risk when welding.

The four individual tabs on the steering tube could be replaced by tubes cut on a tube laser (Figure 37). This change will make welding easier since a fixture will not be needed to ensure the tabs don't warp out of parallel from heat and will provide a more distributed area to weld to the steer tube. For the same reason the four tabs on the center tube should be replaced by two larger plates or drill holes through to remove the need for welding altogether.

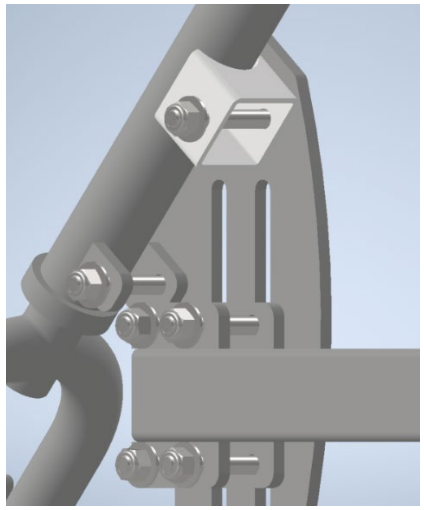

Figure 37 Steering Tube Tab Recommendation

Manufacturing

During the welding process the 3D printed fixtures melted and warped while tacking plates in place. 3D printed fixtures should be replaced by metal fixtures or redesign parts such as the tabs to only require clamps while welding. When welding the fork mounts the wheel plates should be assembled and a fixture used to maintain dropout width at the actual dropouts rather than the mounting point.

Welding processes should be explored to reduce excessive amounts of heat concentrations and control warping.

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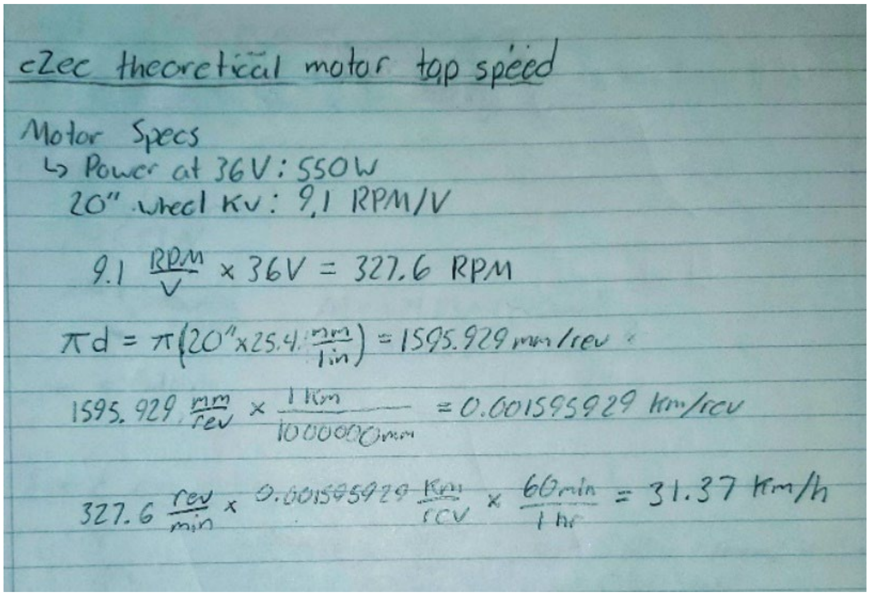
Appendix A – Batec Scrambler Reference Photos

Figure 38 Batec Under Chair Pins

Figure 39 Batec Adjustment Plates

Figure 40 Batec Front Forks

Appendix B – Motor Calculations



eZee theoretical motor top speed

Motor Specs
↳ Power at 36V: 550W
20" wheel Kv: 9.1 RPM/V

$$9.1\ \frac{RPM}{V} \times 36V = 327.6\ RPM$$

$$\pi d = \pi\left(20'' \times 25.4\ \frac{mm}{1in}\right) = 1595.929\ mm/rev$$

$$1595.929\ \frac{mm}{rev} \times \frac{1\ Km}{1000000\ mm} = 0.001595929\ km/rev$$

$$327.6\ \frac{rev}{min} \times 0.001595929\ \frac{Km}{rev} \times \frac{60\ min}{1\ hr} = 31.37\ Km/h$$

Figure 41 Theoretical Motor Calculations

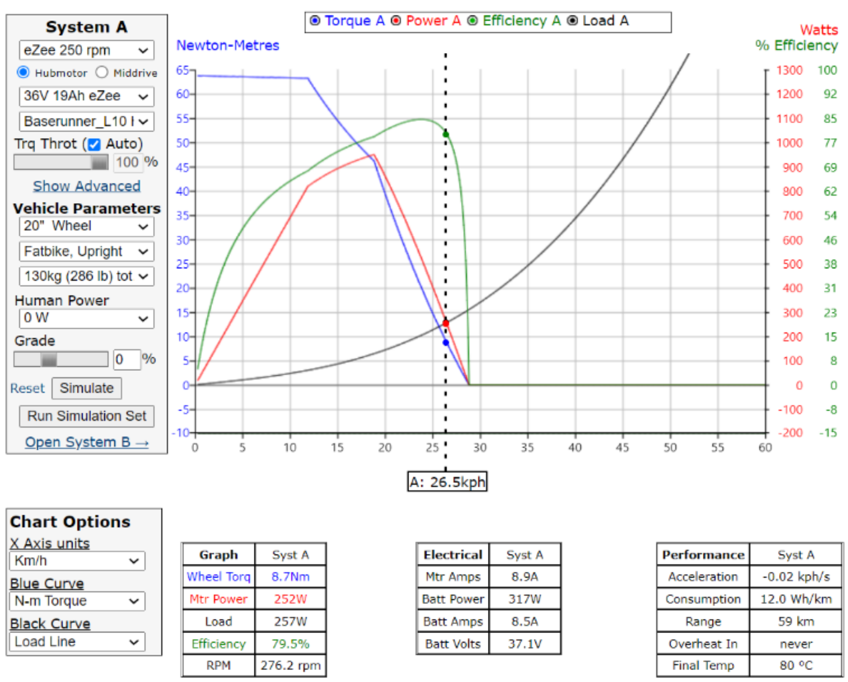
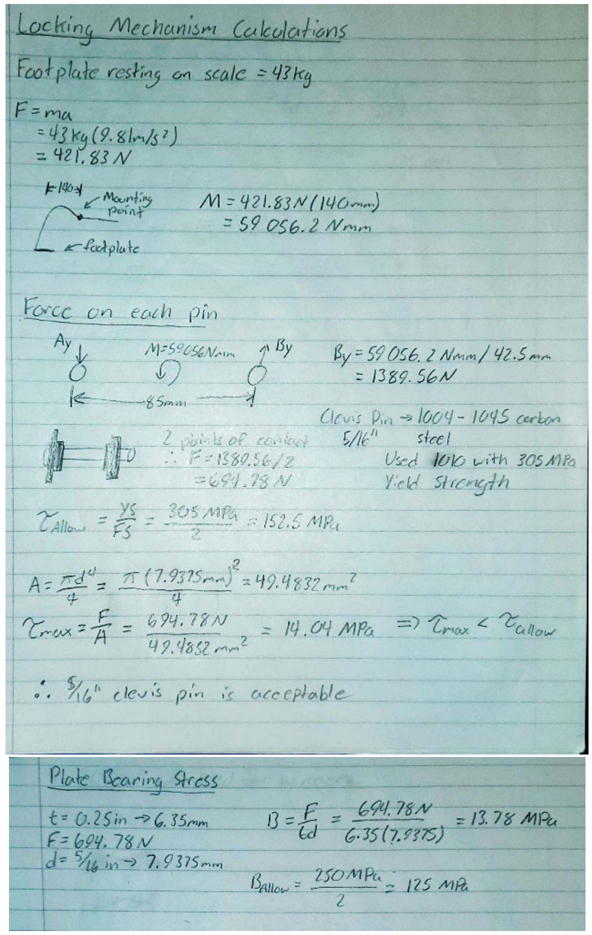


Figure 42 Grin Tech Motor Simulator

Appendix C – Hardware Calculations



Locking Mechanism Calculations

Footplate resting on scale = 43 kg

$F = ma$
$= 43\,kg(9.81\,m/s^2)$
$= 421.83\,N$

$M = 421.83\,N(140\,mm)$
$= 59\,056.2\,Nmm$

Force on each pin

$B_y = 59\,056.2\,Nmm / 42.5\,mm$
$= 1389.56\,N$

Clevis Pin → 1004 - 1045 carbon steel 5/16"

Used 1010 with 305 MPa Yield Strength

2 points of contact
$\therefore F = 1389.56/2$
$= 694.78\,N$

$\tau_{Allow} = \frac{YS}{FS} = \frac{305\,MPa}{2} = 152.5\,MPa$

$A = \frac{\pi d^2}{4} = \frac{\pi(7.9375\,mm)^2}{4} = 49.4832\,mm^2$

$\tau_{max} = \frac{F}{A} = \frac{694.78\,N}{49.4832\,mm^2} = 14.04\,MPa \Rightarrow \tau_{max} < \tau_{allow}$

$\therefore$ 5/16" clevis pin is acceptable

Plate Bearing Stress

$t = 0.25\,in \rightarrow 6.35\,mm$
$F = 694.78\,N$
$d = 5/16\,in \rightarrow 7.9375\,mm$

$B = \frac{F}{td} = \frac{694.78\,N}{6.35(7.9375)} = 13.78\,MPa$

$B_{Allow} = \frac{250\,MPa}{2} = 125\,MPa$

Figure 43 Locking Pin Hand Calculations

Wheel plate bolt selection										
	metric		imperial					Bolt 1		
Abolt =		mm^2	0.0454	in^2	Jj =	0.080413	in^4	Ttix =	7994.04	psi
Horizontal Load =	-289	N	-64.97	lb	Tv =	1490.473	psi	Ttiy =	6710.4	psi
Vertical Load =	602	N	135.335	lb	Th =	-715.526	psi	T1 =	10154	psi
Horizontal dist to CG =	132.16	mm	5.20295	in	Mx =	-135.055	in lb			
Vertical dist to CG =	52.8	mm	2.07874	in	My =	704.1415	in lb	Bolt 2		
r =	23.903	mm	0.94107	in	Mnet =	839.1968	in lb	Ttix =	7994.04	psi
Yield Strength =		Mpa	57	ksi				Ttiy =	6710.4	psi
Safety Factor =	1.5							T2 =	10965	psi
					Tmax =	10965	psi			
					Tallow =	21.926	ksi			
					Acceptable	Yes				

Figure 44 Wheel Plate Bolt Calculations

Appendix D – Material Mechanical Properties

Table 1 1020 Steel Mechanical Properties

Mechanical Properties	Metric	Imperial
Tensile Strength, Ultimate	420 MPa	60900 psi
Tensile Strength, Yield	350 MPa	50800 psi

Table 2 A36 Steel Mechanical Properties

Mechanical Properties	Metric	Imperial
Tensile Strength, Ultimate	400 - 550 MPa	58000 - 79800 psi
Tensile Strength, Yield	250 MPa	36300 psi

Table 3 A500 Mechanical Properties

Mechanical Properties	Metric	Imperial
Tensile Strength, Ultimate	310 MPa	45000 psi
Tensile Strength, Yield	270 MPa	39200 psi

Appendix E – Motor Dimensions

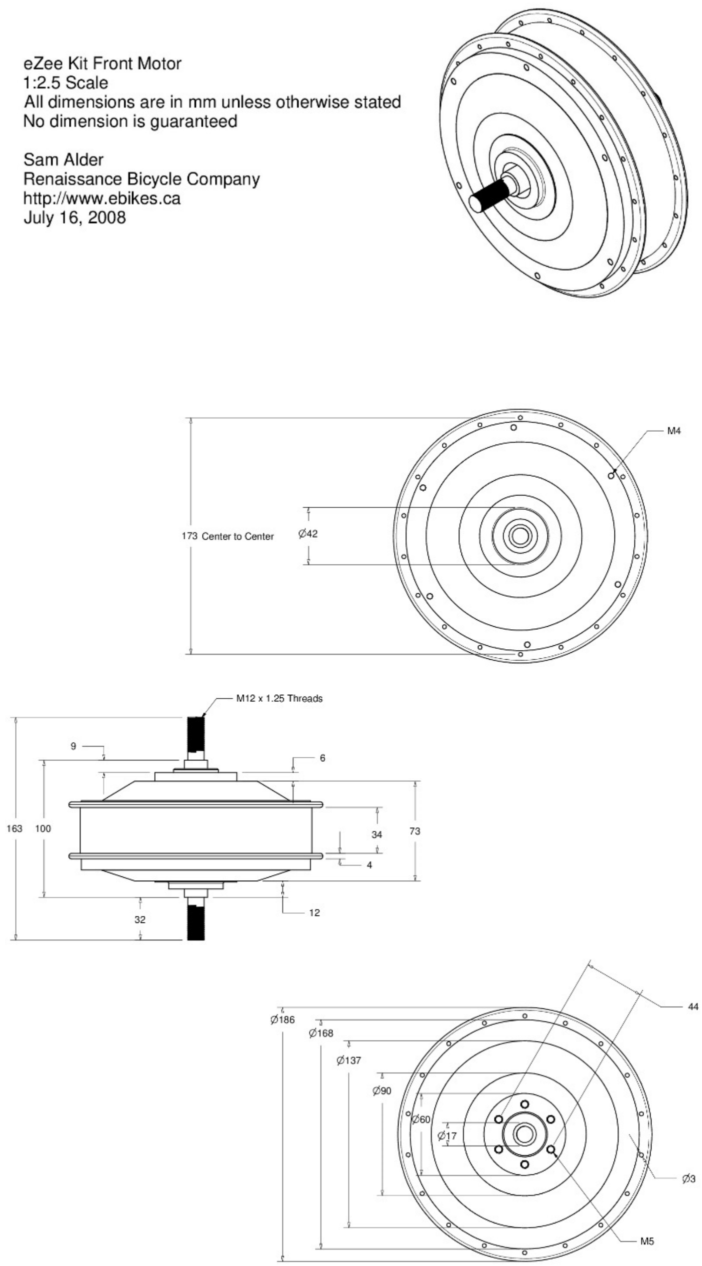


Figure 45 eZee Kit Front Motor Dimensions

Appendix F – IS Mount Dimensions

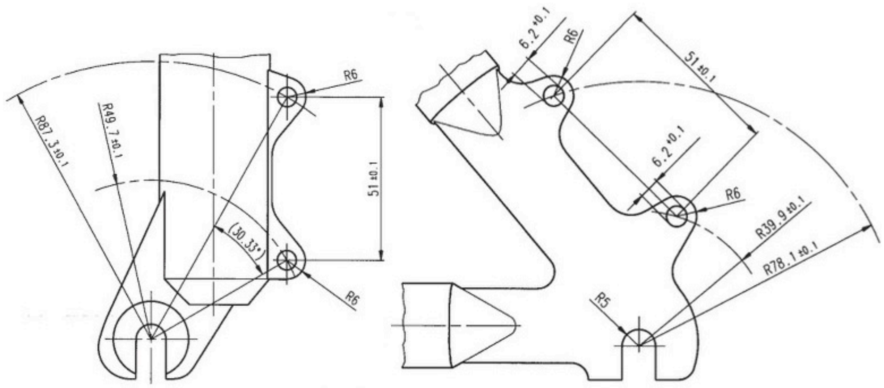
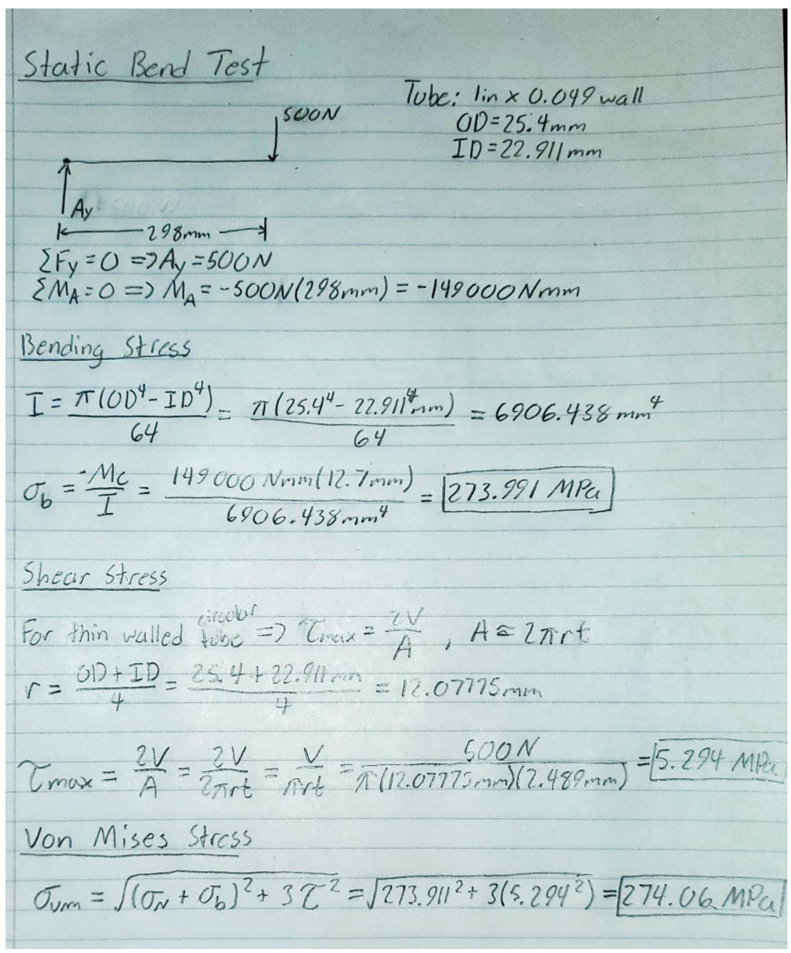


Figure 46 IS Mount Dimensions

Appendix G – Stress Analysis Hand Calculations



Static Bend Test

Tube: 1in x 0.049 wall

OD = 25.4mm

ID = 22.911mm

500N

A_y

298mm

$\sum F_y = 0 \Rightarrow A_y = 500N$

$\sum M_A = 0 \Rightarrow M_A = -500N(298mm) = -149000Nmm$

Bending Stress

$$I = \frac{\pi(OD^4 - ID^4)}{64} = \frac{\pi(25.4^4 - 22.911^4 mm)}{64} = 6906.438\, mm^4$$

$$\sigma_b = \frac{-Mc}{I} = \frac{149000\, Nmm(12.7mm)}{6906.438mm^4} = \boxed{273.991\ MPa}$$

Shear Stress

For thin walled circular tube $\Rightarrow \tau_{max} = \frac{2V}{A}$, $A \approx 2\pi r t$

$$r = \frac{OD + ID}{4} = \frac{25.4 + 22.911mm}{4} = 12.07775mm$$

$$\tau_{max} = \frac{2V}{A} = \frac{2V}{2\pi r t} = \frac{V}{\pi r t} = \frac{500N}{\pi(12.07775mm)(2.489mm)} = \boxed{5.294\ MPa}$$

Von Mises Stress

$$\sigma_{vm} = \sqrt{(\sigma_N + \sigma_b)^2 + 3\tau^2} = \sqrt{273.911^2 + 3(5.294^2)} = \boxed{274.06\ MPa}$$

Figure 47 Static Bend Test Hand Calculations

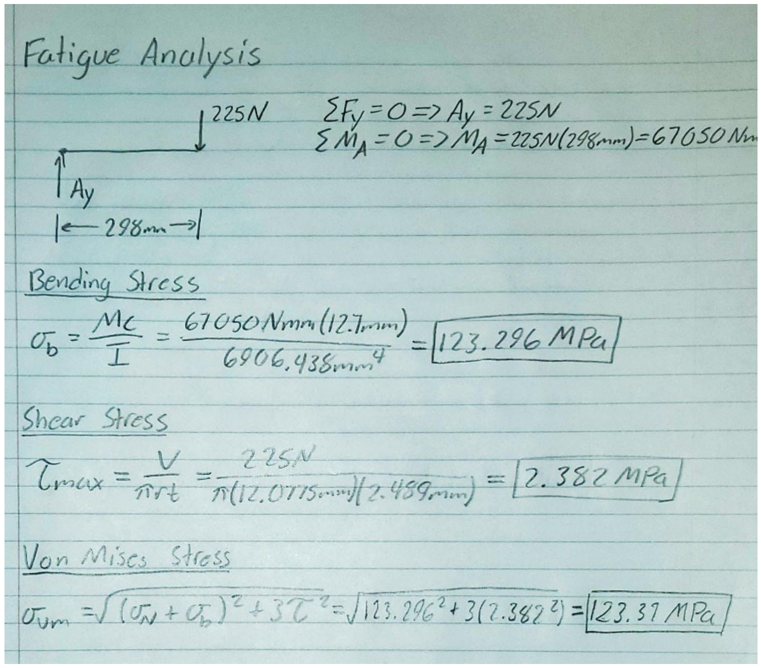


Figure 48 Fork Fatigue Hand Calculations

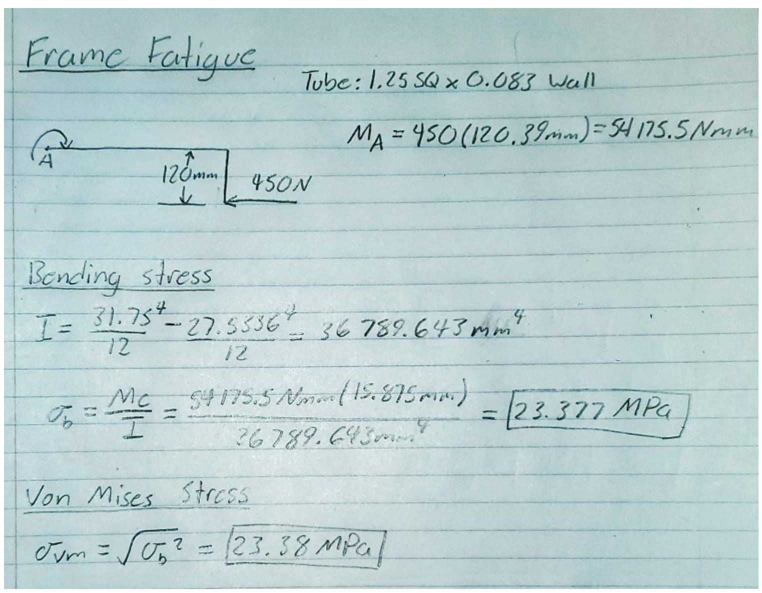
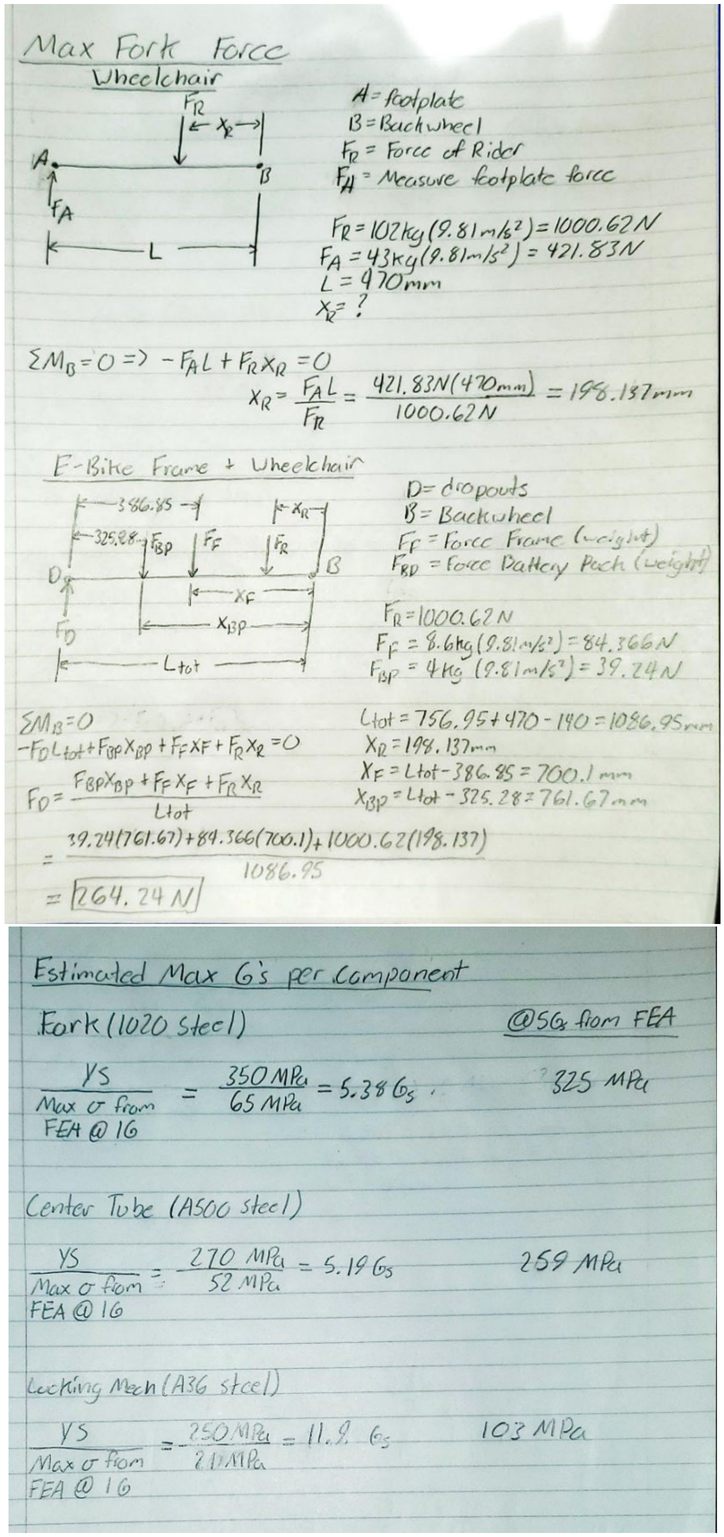


Figure 49 Frame Fatigue Hand Calculations



Max Fork Force

Wheelchair

A = footplate
B = Backwheel
F_R = Force of Rider
F_A = Measure footplate force

$F_R = 102kg(9.81m/s^2) = 1000.62N$
$F_A = 43kg(9.81m/s^2) = 421.83N$
$L = 470mm$
$x_R = ?$

$\Sigma M_B = 0 \Rightarrow -F_A L + F_R x_R = 0$

$x_R = \frac{F_A L}{F_R} = \frac{421.83N(470mm)}{1000.62N} = 198.137mm$

E-Bike Frame + Wheelchair

D = dropouts
B = Backwheel
F_F = Force Frame (weight)
F_{BP} = Force Battery Pack (weight)

$F_R = 1000.62N$
$F_F = 8.6kg(9.81m/s^2) = 84.366N$
$F_{BP} = 4kg(9.81m/s^2) = 39.24N$

$\Sigma M_B = 0$
$-F_D L_{tot} + F_{BP}x_{BP} + F_F x_F + F_R x_R = 0$

$F_D = \frac{F_{BP}x_{BP} + F_F x_F + F_R x_R}{L_{tot}}$

$L_{tot} = 756.95 + 470 - 140 = 1086.95mm$
$x_R = 198.137mm$
$x_F = L_{tot} - 386.85 = 700.1mm$
$x_{BP} = L_{tot} - 325.28 = 761.67mm$

$= \frac{39.24(761.67) + 84.366(700.1) + 1000.62(198.137)}{1086.95}$

$= \boxed{264.24N}$

Estimated Max G's per Component

Fork (1020 steel) — @5G from FEA

$\frac{YS}{\text{Max } \sigma \text{ from FEA @ 1G}} = \frac{350MPa}{65MPa} = 5.38Gs$ — 325 MPa

Center Tube (A500 steel)

$\frac{YS}{\text{Max } \sigma \text{ from FEA @ 1G}} = \frac{270MPa}{52MPa} = 5.19Gs$ — 259 MPa

Locking Mech (A36 steel)

$\frac{YS}{\text{Max } \sigma \text{ from FEA @ 1G}} = \frac{250MPa}{21MPa} = 11.9Gs$ — 103 MPa

Figure 50 Frame Max G Hand Calculations

Appendix H – Manufacturing Drawings