Component Design

Scissor Lift

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August 2nd, 2022

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I. Introduction

ATV Solutions is developing a mobile, portable scissor lift for at home use. Most scissor lifts are primarily sold for commercial use and are much too large and expensive for the common consumer. However, repairs and renovations are a critical part of home ownership and are made easier through the use of a scissor lift. These tasks often involve hard-to-reach places, whether it's a routine job such as cleaning the gutters or a more ambitious project like repainting a home's exterior. Any job high above the ground is inherently dangerous, especially for older adults and those with physical impairments. This risk is amplified when using a ladder, which can be extremely unstable if improperly secured. This product provides the ordinary homeowner with a safer and easier way to carry out these tasks for a reasonable cost.

Since this product will be constantly raised and lowered, presumably with the consumer on it, the ATV Solutions engineering team has conducted various analyses on the critical components identified in the design. Specifically, overall stress analysis, buckling analysis, and fatigue analysis for cyclic loading have been calculated, and the failure modes have been identified. This report presents the modelled system as well as the individual components in Onshape along with the material selection. In addition, the loads on each component are shown through free body diagrams, and the stresses due to those loads are calculated. Based on a maximum of 900lbf distributed across the top plate, bucking calculations are also shown to assess the safety of critical components and identify possible points of redesign. Lastly, fatigue analysis was conducted to determine if any components will fail after repeated raising and lowering of the lift within the range of the predicted life of 10,000 hours of usage.

II. Overall System

The full model of the overall system can be seen below in both the open and closed position. This system consists of the critical components: the top plate, bottom plate, linkages, and connecting rod, which are analyzed in this report. It also contains safety railings, wheels with casters for portability, and lowerable legs for use during operation. During analysis of the system, it became clear that the maximum stresses will occur in the collapsed configuration, and thus analysis was conducted primarily on this state.

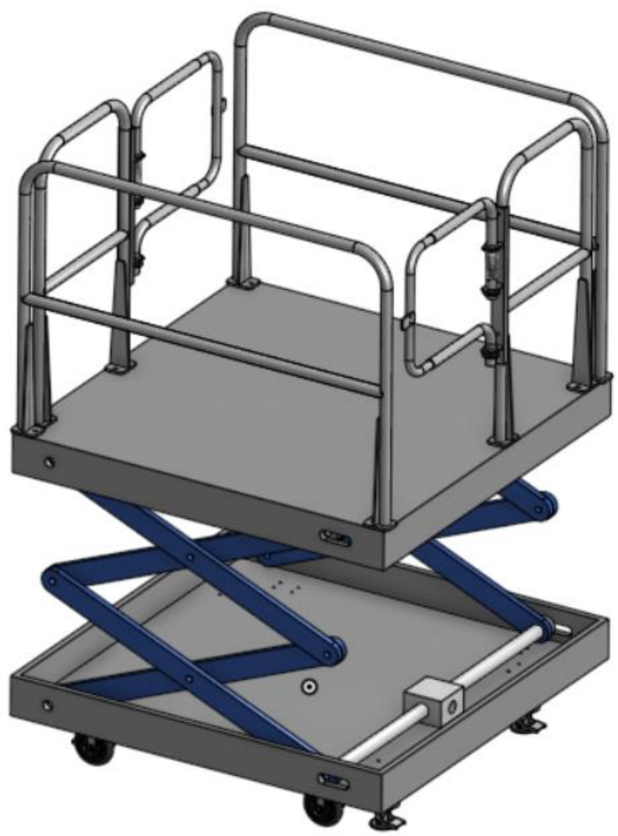

Figure II.1: Scissor Lift Model in Raised Position

Figure II.2: Scissor Lift Model in Lowered Position

A free body diagram of the entire system can be seen below in Figure II.3, showing an offset applied load:

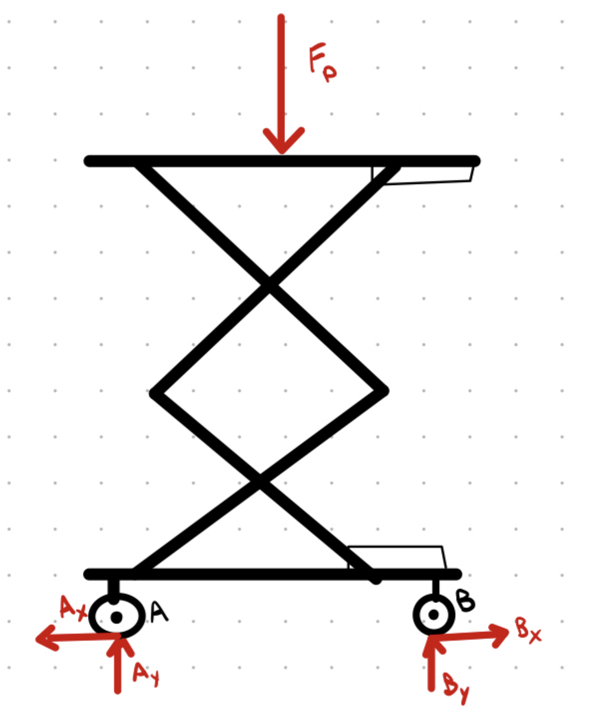

Figure II.3 FBD of Full System

This free body diagram was then used to create force and moment balances to solve for the support reactions. This was calculated in MATLAB, (see Appendix B for code) and the resulting solutions are shown in Figure II.4 below. In addition to the reaction forces in each member, it was concluded that the drive force on the system must be 9.26 kip, which will be taken into account in our Power Transmission analysis, as it is important that the chosen motor achieves this spec.

```
F_D = 9.2617e+03
C_x = -2.2546e+03
C_y = -326.3577
D_y = -307.0943
E_x = -1.1516e+04
E_y = -1.1405e+03
G_y = 307.0943
H_x = -1.5831e+04
H_y = -1.0905e+03
I_x = -6.5698e+03
I_y = -733.4520
J_x = 6.5698e+03
J_y = 376.3577
K_x = 4.3151e+03
K_y = 0
```

Figure II.4: Reaction forces on linkage members (lbf)

The overall system was analyzed using SolidWorks and the stress distribution can be seen below in Figure II.5. As previously mentioned, this analysis was done in the closed position as that is the position where the highest stress is present.

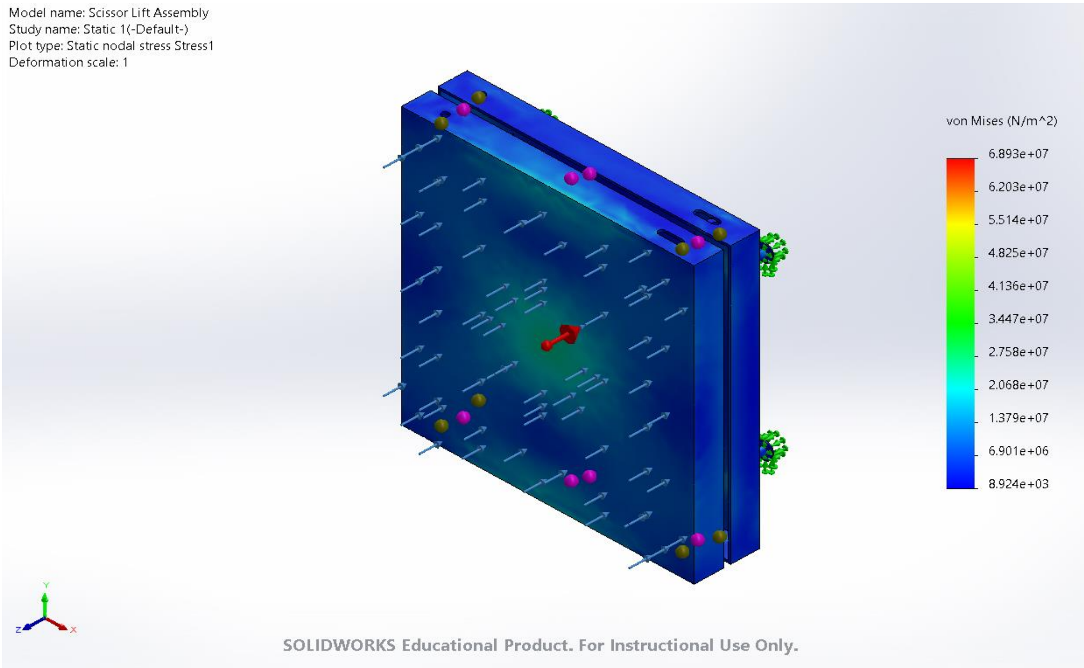

Figure II.5: Stress Map for Overall System

III. Top Plate Analysis

a. Model and Drawing of Component

The top plate was modeled in Onshape and can be seen in the assembly below in the isolated state, with both the isometric and side view shown. This component is the plate that is raised and lowered during usage, and the part where the user will stand on, so it will carry the direct load. The material selected for this component is structural A36 steel with powder coating to for a higher strength and corrosion resistant design.

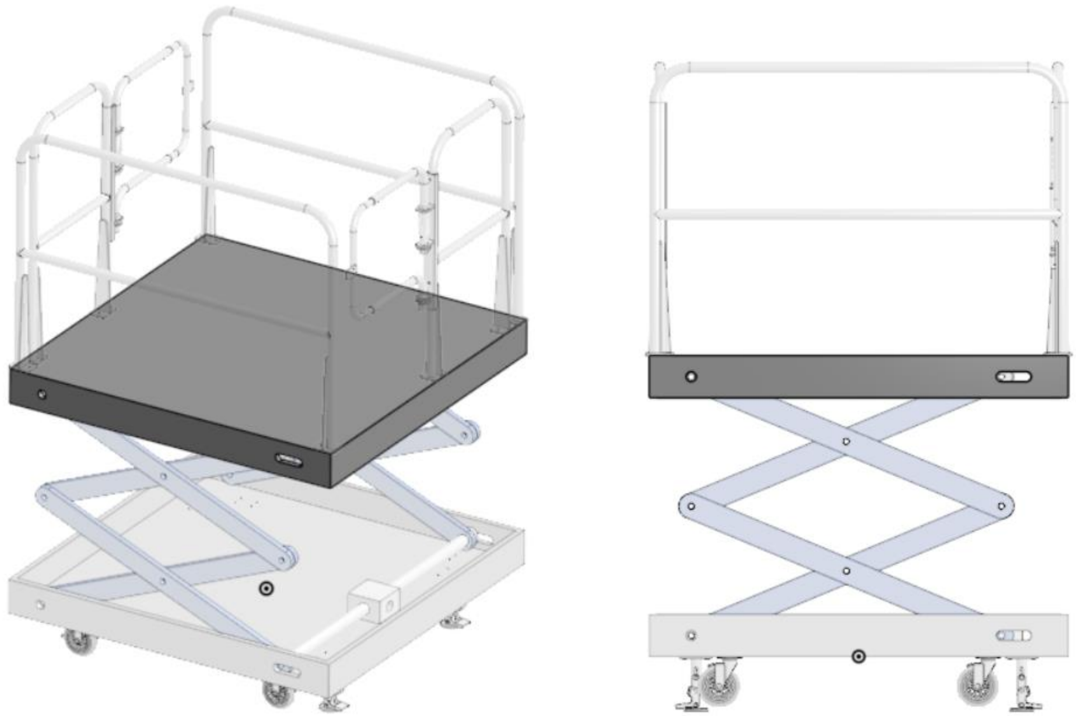

Figure III.1: Isolated Top Plate in the Raised Position, for Isometric and Side Views

b. Free Body Diagram

A free body diagram for the top plate including the reaction forces of the linkages and the applied load can be seen in Figure III.2 below.

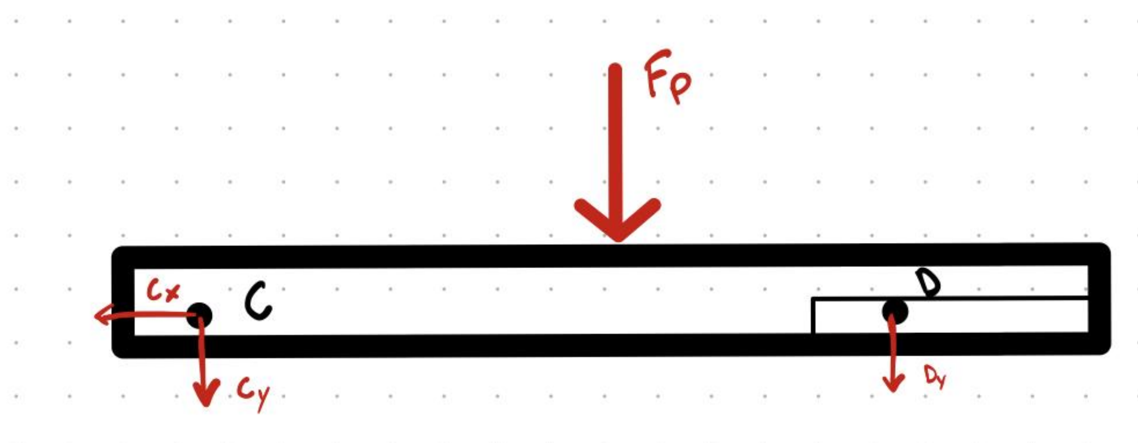

Figure III.2 FBD of Top Plate

c. Stress Calculations and Finite Element Analysis

The stress analysis for the top plate was done in SolidWorks, where the maximum stress was found to be in the center of the plate, as seen below in Figure III.3. The maximum stress in the top plate was found to be 0.2392 MPa

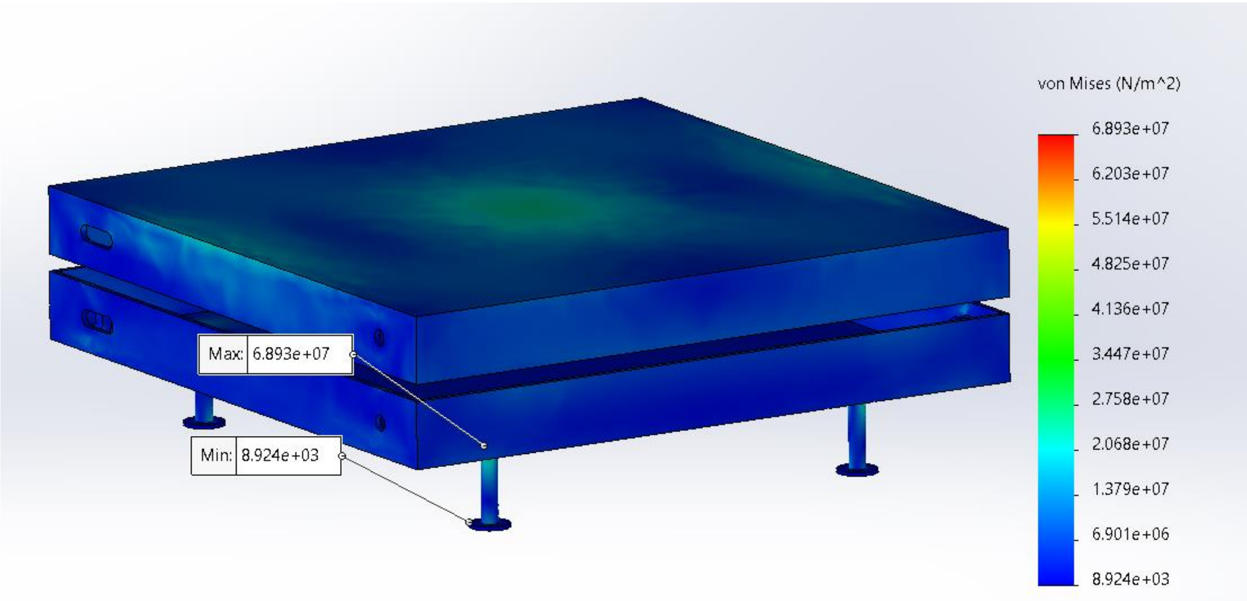

Figure III.3 FBD of Top Plate

d. Buckling Analysis

There is no buckling potential on the top plate since it is not in compression along either of its vulnerable, horizontal axes.

e. Fatigue Analysis

As the scissor lift undergoes cyclic loading on the top plate over the course of its life, the top plate will undergo fatigue. Using the ultimate tensile strength of A36 Steel and assuming the plate will behave as a non-rotating rectangular beam, the following calculations were carried out in Excel and can be seen below in Figure III.4. The maximum and minimum stresses on the top plate were taken from the previous SolidWorks simulation. From these calculations, it was determined that infinite life was achievable with a factor of safety of 572.82.

FATIGUE CALC FOR TOP PLATE - STRUCTURAL STEEL

TENSILE STRENGTH	Sut	400	MPa	
	MIN TENSILE STRENGTH FOR A36 STEEL			
ENDURANCE LIMIT	Se'	200	Mpa	
SURFACE COND FACTOR	ka	0.828374202		
	ka=a*Sut^b			
	SURFACE FINISH: MACHINED			
	a	3.04		
	b	-0.217		
SIZE MOD FACTOR	kb	0.735881936		
	NON ROTATING MEMBER - RECTANGLE			
	de=0.808*sqrt(h*b)			
	h	9.525	3/8 in	
	b	1524	5 ft	
	de	97.35008531		
	51<de<254 mm			
	kb=1.51*de^-0.157			
LOAD MOD FACTOR	kc	1		
	BENDING kc=1			
TEMP MOD FACTOR	kd	1		
	ASSUME ROOM TEMP			
RELIABILITY FACTOR	ke	0.702		
	RELIABILITY OF 99.99%			
MISC FACTOR	kf	1		
	ASSUME NO MISC			
CRIT ENDURANCE LIMIT	Se	85.58581995	Mpa	
	Se=Se'*ka*kb*kc*kd*ke			
AMPLITUDE COMPONENT	siga	0.122654	MPa	
	sigmamax	0.2477	MPa	*FROM FEA*
	sigmamin	0.002392	MPa	
	siga=(sigmamax-sigmamin)/2			
MIDRANGE COMPONENT	sigm	0.125046	MPa	
	sigm=(sigmamax+sigmamin)/2			
MOD-GOODMAN				
SAFETY FACTOR	n	572.8275034		
	1/n=(siga/Se)+(sigm/Sut)			
	1/n	0.001745726		

Figure III.4 Fatigue Calculations for Top Plate in Excel

IV. Bottom Plate Analysis

a. Model and Drawing of Component

The bottom plate was modelled in Onshape and can be seen in the assembly below in the isolated state, with both the isometric and side view shown. This is the component that attaches to the wheels and legs, which will contact the ground and stays stationary during use. The material selected for this component is Aluminum 7075 to for a lightweight and corrosion resistant design.

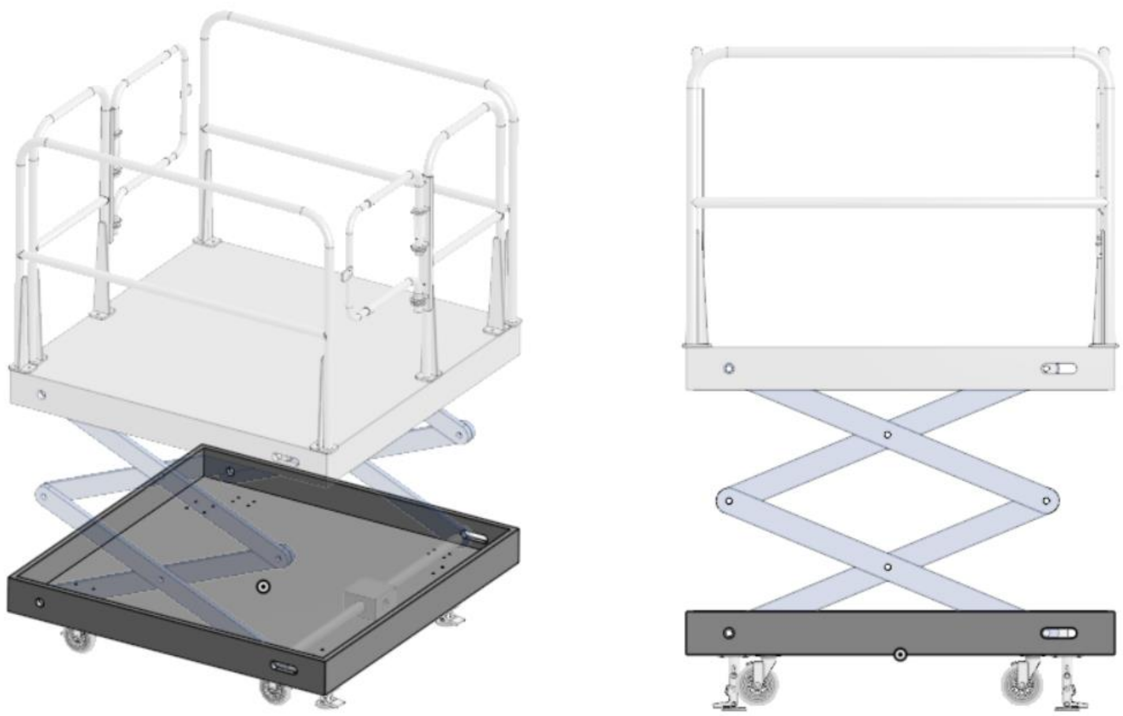

Figure IV.1: Isolated Bottom Plate in the Raised Position, for Isometric and Side Views

b. Free Body Diagram

A free body diagram for the bottom plate including the reaction forces of the linkages can be seen in Figure IV.2 below

Figure IV.2 FBD of Bottom Plate

c. Stress Calculations and Finite Element Analysis

The stress analysis for the bottom plate was done in SolidWorks, where the maximum stress was found to be where the legs are mounted, as seen below in Figure IV.3. The maximum stress in the bottom plate was found to be 68.93 MPa.

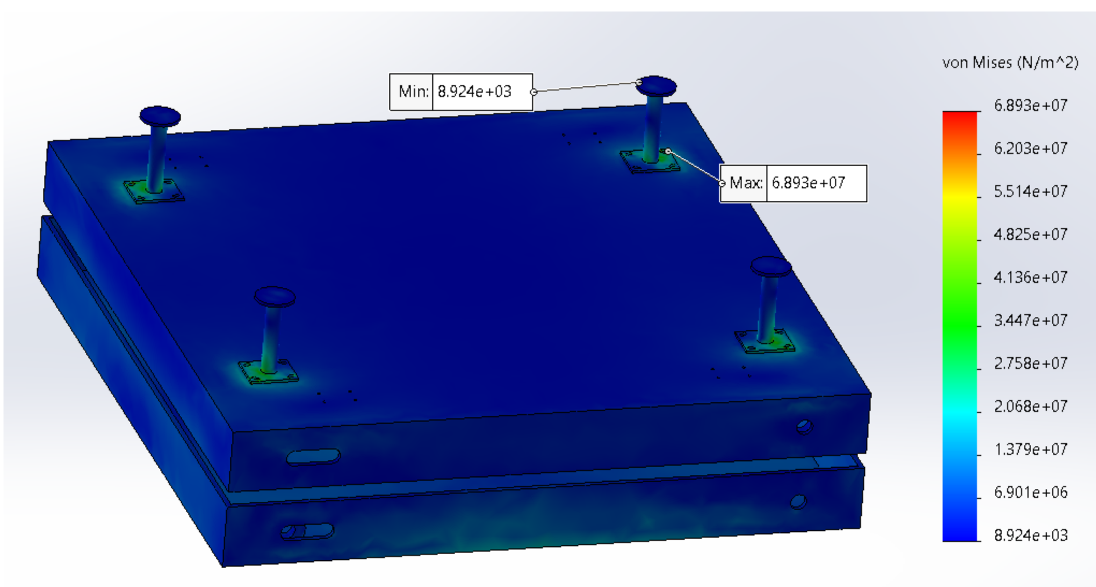

Figure IV.3 FBD of Bottom Plate

d. Buckling Analysis

There will be no risk of buckling in the bottom plate. While there is certainly compression, none of these loads are oriented with the vulnerable, horizontal axes of the thin plate.

e. Fatigue Analysis

As the scissor lift undergoes cyclic loading the bottom plate will undergo fatigue as the support legs are constantly being loaded. Using the ultimate tensile strength of Aluminum 7075 and assuming the plate will behave as a non-rotating rectangular beam, the following calculations were carried out in Excel and can be seen below in Figure IV.4. The maximum and minimum stresses on the bottom plate were taken from the previous SolidWorks simulation. From these calculations, it was determined that infinite life was achievable with a factor of safety of 2.76.

FATIGUE CALC FOR BOTTOM PLATE - ALUMINUM 7075

TENSILE STRENGTH	Sut	570	MPa	
	MIN TENSILE STRENGTH FOR ALUMINUM 7075			
ENDURANCE LIMIT	Se'	285	Mpa	
SURFACE COND FACTOR	ka	0.767094272		
	ka=a*Sut^b			
	SURFACE FINISH: MACHINED			
	a	3.04		
	b	-0.217		
SIZE MOD FACTOR	kb	0.735881936		
	NON ROTATING MEMBER - RECTANGLE			
	de=0.808*sqrt(h*b)			
	h	9.525	3/8 in	
	b	1524	5 ft	
	de	97.35008531		
	51<de<254 mm			
	kb=1.51*de^-0.157			
LOAD MOD FACTOR	kc	1		
	BENDING kc=1			
TEMP MOD FACTOR	kd	1		
	ASSUME ROOM TEMP			
RELIABILITY FACTOR	ke	0.702		
	RELIABILITY OF 99.99%			
MISC FACTOR	kf	1		
	ASSUME NO MISC			
CRIT ENDURANCE LIMIT	Se	112.937678	Mpa	
	Se=Se'*ka*kb*kc*kd*ke			
AMPLITUDE COMPONENT	siga	34.460538	MPa	
	sigmamax	68.93	MPa	*FROM FEA*
	sigmamin	0.008924	MPa	
	siga=(sigmamax-sigmamin)/2			
MIDRANGE COMPONENT	sigm	34.469462	MPa	
	sigm=(sigmamax+sigmamin)/2			
MOD-GOODMAN				
SAFETY FACTOR	n	2.735218247		
	1/n=(siga/Se)+(sigm/Sut)			
	1/n	0.365601539		

Figure IV.4 Fatigue Calculations for Bottom Plate in Excel

V. Linkage Analysis

a. Model and Drawing of Component

The linkages were modelled in Onshape and can be seen in the assembly below in the isolated state, with both the isometric and side view shown. The linkage is one part which is used eight times in the assembly. These components are connected to each other and pushed along the slots in the two plates, which will either raise or lower the top plate. The material selected for this component is structural A36 steel with powder coating to for a higher strength and corrosion resistant design.

Figure V.1: Isolated Linkages in the Raised Position, for Isometric and Side Views

b. Free Body Diagram

Free body diagrams were drawn for the individual linkages to show the forces acting on and between them, which can be seen below in Figure V.2.

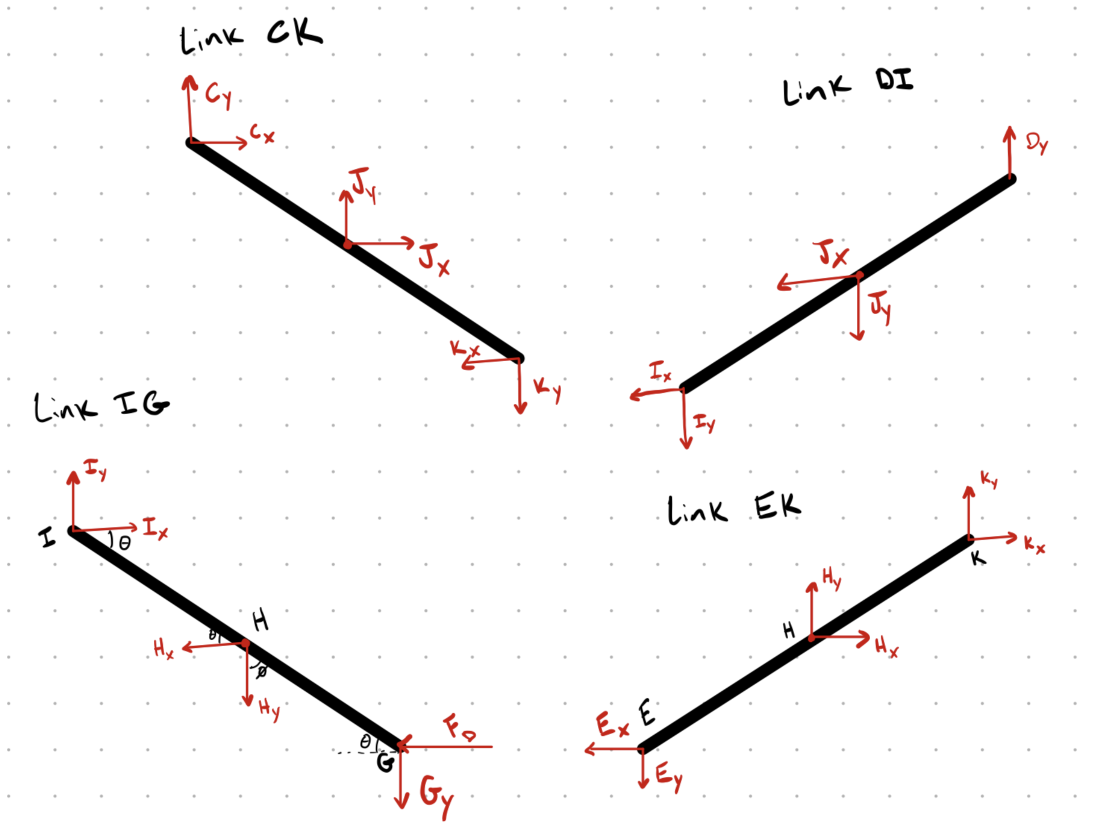

Figure V.2 FBD of Linkages

c. Stress Calculations and Finite Element Analysis

A static analysis was conducted on the overall assembly in SolidWorks to solve for the resultant forces of the linkages. These components were assumed to behave as simple beams and were treated as such during static analysis. To note, this analysis was done with a weight of 900 lbs evenly distributed evenly throughout the top plate. This information can be found below in Table V.1.

Table V.1: Static Forces and Moments on Linkage Members

Beam Name	Joints	Axial(N)	Shear1(N)	Shear2(N)	Moment1(N*m)	Moment2(N*m)	Torque(N*m)
Linkage IG	1	-156.501	-776.648	-57.1211	-5.21189	16.2436	-1.9818
	2	1,958.63	212.304	131.644	4.11339	-25.5089	8.58802
	3	468.073	-1,119.14	-215.542	-17.7503	70.5138	9.44987
Linkage EK	1	-132.827	-367.559	-139.334	-1.76519	-28.1131	0.0274159
	2	-833.266	-5.39394	8.04607	5.26174	-11.1735	-6.65995
	3	824.985	-100.609	-8.04607	-0.341914	-17.9366	6.65995
Linkage CK	1	908.927	-148.238	-236.586	7.55637	3.24595	21.6429
	2	-844.266	-32.0894	-29.1133	-8.02675	-12.273	10.2674
	3	-1,934.45	-138.316	43.3868	17.5524	39.5376	-25.2011
Linkage ID	1	424.605	-258.871	61.3697	-1.04743	35.0721	0.635493
	2	-359.679	-102.823	0.697556	-4.17028	-25.5092	-6.21794
	3	-706.211	-76.9849	5.19373	-1.5778	4.07216	-7.17929
Linkage IG – OPP SIDE	1	535.257	-1,256.37	-86.6462	-17.0355	-13.6932	-9.41358
	2	1,137.14	-39.9148	50.9422	-5.22841	-16.5387	8.79163
	3	-658.463	-435.841	167.92	-12.4453	-49.4772	-16.6528
Linkage EK – OPP SIDE	1	-1,203.45	215.272	-62.9129	-3.27873	-18.9884	-17.139
	2	982.468	-52.7612	29.4881	-8.11463	2.11225	-11.401
	3	-974.187	-53.242	-29.4881	-9.91611	-1.96522	11.401
Linkage CK – OPP SIDE	1	186.53	274.948	97.2116	0.16252	-3.78916	-18.0718
	2	497.601	-447.067	204.746	20.8373	35.9152	-16.2412
	3	-1,543.87	-238.018	31.6851	-10.9002	12.1444	17.5826
Linkage ID – OPP SIDE	1	66.3495	184.017	-25.5549	12.3943	22.1859	9.27391
	2	-594.419	120.049	-291.427	26.3971	-30.8754	17.5336
	3	-727.491	264.783	-188.646	-6.97347	-31.8906	9.5096

SolidWorks analysis was done on the scissor lift in the lowered position, as the lowered position was determined to have higher stresses due to the smaller angle between the linkages. The maximum stress in the linkages was 6.827 MPa and the minimum stress was 0.5995 MPa. The resulting stress map of the linkages with minimum and maximum stresses can be seen below in Figure V.3.

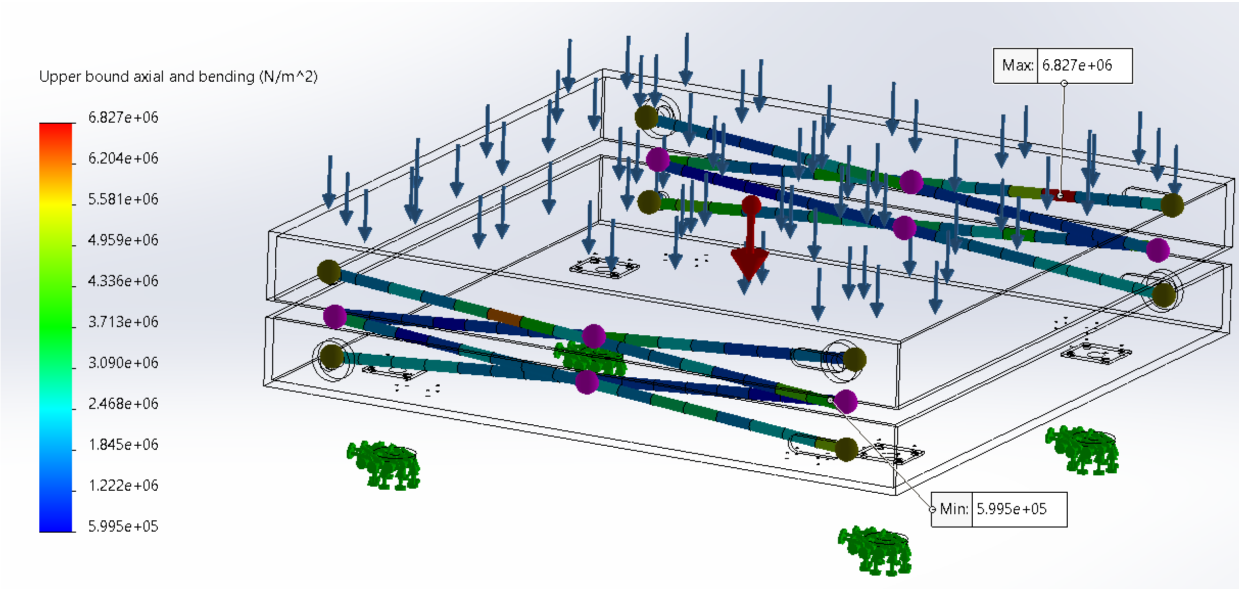

Figure V.3 Stress Map for Linkages

The SolidWorks Analysis also yielded the resulting stresses on each beam, which can be seen below in Table V.2.

Table V.2: Stresses on Linkage Members

Beam Name	Joints	Axial(N/m^2)	Bending Dir1(N/m^2)	Bending Dir2(N/m^2)	Torsional (N/m^2)	Upper bound axial and bending(N/m^2)
Linkage IG	1	69,307.6	545,227	485,507	-229,834	1.10004e+06
	2	867,393	430,311	762,442	995,976	2.06015e+06
	3	207,290	-1.8569e+06	-2.1076e+06	1.09593e+06	4.17179e+06
Linkage EK	1	-58,823.3	-184,660	840,277	3,179.5	1.08376e+06
	2	-369,019	550,442	333,966	-772,372	1.25343e+06
	3	-365,351	35,768.3	-536,111	772,372	937,230
Linkage CK	1	-402,526	-790,488	97,018.8	2.50998e+06	1.29003e+06
	2	-373,890	-839,695	366,829	1.19074e+06	1.58041e+06
	3	-856,689	1.83619e+06	-1.18175e+06	-2.92264e+06	3.87462e+06
Linkage ID	1	-188,040	109,574	1.04828e+06	73,699.9	1.34589e+06
	2	-159,287	-436,262	762,448	-721,112	1.358e+06
	3	-312,751	-165,057	-121,713	-832,602	599,521
Linkage IG – OPP SIDE	1	237,043	-1.78212e+06	409,279	-1.09172e+06	2.42844e+06
	2	503,592	-546,955	494,329	1.01959e+06	1.54488e+06
	3	291,606	1.30193e+06	-1.47883e+06	-1.93127e+06	3.07236e+06
Linkage EK – OPP SIDE	1	-532,959	-342,995	567,548	-1.98766e+06	1.4435e+06
	2	-435,094	848,889	63,133.4	-1.32221e+06	1.34712e+06
	3	-431,426	-1.03735e+06	58,738.9	1.32221e+06	1.52751e+06
Linkage CK – OPP SIDE	1	-82,606.2	-17,001.5	-113,255	-2.09584e+06	212,863
	2	220,367	2.17984e+06	-1.07348e+06	-1.88354e+06	3.47368e+06
	3	-683,715	-1.1403e+06	-362,987	2.0391e+06	2.187e+06

Beam Name	Joints	Axial(N/m^2)	Bending Dir1(N/m^2)	Bending Dir2(N/m^2)	Torsional (N/m^2)	Upper bound axial and bending(N/m^2)
Linkage ID – OPP SIDE	1	29,383.4	1.29659e+06	-663,119	1.07552e+06	1.98909e+06
	2	263,243	-2.76146e+06	-922,840	2.03342e+06	3.94754e+06
	3	-322,175	-729,510	953,185	1.10285e+06	2.00487e+06

d. Buckling Analysis

Because of the high compressive loads on the linkages, bucking analyses must be completed. These analyses will be done for both in-plane and out-of-plane buckling to identify the first buckling mode. Then, the safety factor will be calculated and compared with the safety factor for yielding to determine the limiting factor.

Each linkage is geometrically identical. Therefore, the limiting linkage will be that with the highest applied force. This component can be identified by probing the stresses on each linkage. Through the SolidWorks FEA above, it was determined that Linkage IG experiences the highest compressive axial load, which is 1.959 kN this will be used for the following analysis.

In general, buckling can be defined by the following set of equations. Equation V.1 is the J.B. Johnson Equation for Buckling and Equation V.2 is the Euler Equation for Buckling.

$$P_{cr,J} = \left[S_y - \left(\frac{1}{CE}\right)\left(\frac{S_y S_R}{2\pi}\right)^2\right](A) : S_R \leq C_C \quad \text{(Eq. V.1)}$$

$$P_{cr,E} = \frac{C\pi^2 EI}{L^2} : S_R \geq C_C \quad \text{(Eq. V.2)}$$

The correct equation is determined by C_c and S_R , where C_c is the Critical Slenderness Ratio

$$C_C = sqrt\left(\frac{2C\pi^2 E}{S_y}\right) \quad \text{(Eq. V.3)}$$

And S_R is the Slenderness Ratio, defined as

$$S_R = \frac{L}{sqrt\left(\frac{I}{A}\right)} \quad \text{(Eq. V.4)}$$

Starting with **In-Plane Buckling**, C_c is calculated using the following values.

C = 1 (pin-pin loading).

Young's Modulus (E) = 200 GPa

Yield Strength (S_y) = 250 MPa

$$C_C = sqrt\left(\frac{2C\pi^2 E}{S_y}\right) = sqrt((2 * 1 * \pi^2 * 200 * 10^9)/(250 * 10^6)) = \mathbf{125.664}$$

S_R is calculated using these additional values

b = 1” = 0.0254 m

h = 3.5” = 0.0889 m

L = 48.186” = 1.2239 m

$$I = \frac{bh^3}{12} = 1.4872 * 10^{-6}\, m^4$$

$$A = bh = 0.002258\, m^2$$

$$S_R = \frac{L}{sqrt\left(\frac{I}{A}\right)} = \frac{1.2239}{sqrt\left(1.4872 * \frac{10^{-6}}{0.002258}\right)} = \mathbf{47.692}$$

Since $S_R < C_C$, we use the J.B. Johnson Equation Plugging in:

$$P_{cr} = \left[S_y - \left(\frac{1}{CE}\right)\left(\frac{S_y S_R}{2\pi}\right)^2\right](A) = \mathbf{523,800\ N}$$

This gives us a critical load of 523.8 kN. Before determining the actual factor of safety for buckling, it is also important to consider the **Out-of-Plane** buckling which may occur at a lower load. The calculation is the same except for changes to the following values:

C = fixed-fixed = 1.2

b = 3.5” = 0.0889 m

h = 1” = 0.0254 m

$$I = \frac{bh^3}{12} = 1.2140 * 10^{-7}\, m^4$$

Solving **Out-of-Plane** for C_C and S_R

$$C_C = sqrt\left(\frac{2C\pi^2 E}{S_y}\right) = sqrt((2 * 1.2 * \pi^2 * 200 * 10^9)/(250 * 10^6)) = \mathbf{137.658}$$

$$S_R = \frac{L}{sqrt\left(\frac{I}{A}\right)} = \frac{1.2239}{sqrt\left(1.4872 * \frac{10^{-6}}{0.002258}\right)} = \mathbf{166.916}$$

In this case, S_R is less than C_C, so the Euler Equation is used.

$$P_{cr} = \frac{C\pi^2 EI}{L^2} = \frac{1.2\pi^2(200 * 10^9 * 1.214 * 10^{-7})}{1.2239^2} = \mathbf{191{,}973\ N}$$

This gives us a critical load of 191.973 kN, which is less than the in plane critical load of 523.8 kN. Since the actual load on this limiting linkage is 1.959 kN, the final factor of safety for overall buckling would be **97.995**. This value is quite high. However, given the overall scale of the system, reducing the thickness of the linkages would not provide much of reduction of weight or cost to the final product. Additionally, making the beams thinner than an inch could reduce a potential purchaser's confidence in our system, since our primary claim is the safety and stability of the Scissor Lift.

e. Fatigue Analysis

As the scissor lift undergoes cyclic loading and is raised and lowered over the course of its life, the linkages will undergo fatiguing. Using the ultimate tensile strength of A36 Steel and treating each linkage as a non-rotating rectangular bar, the following calculations were carried out in Excel and can be seen below in Figure V.4. The maximum and minimum stresses on the linkages were taken from the previous SolidWorks simulation. From these calculations, it was determined that infinite life was achievable with a factor of safety of 24.28.

FATIGUE CALC FOR LINKAGE - STRUCTURAL STEEL

TENSILE STRENGTH	Sut	400	MPa	
	MIN TENSILE STRENGTH FOR A36 STEEL			
ENDURANCE LIMIT	Se'	200	Mpa	
SURFACE COND FACTOR	ka	0.828374202		
	ka=a*Sut^b			
	SURFACE FINISH: MACHINED			
	a	3.04		
	b	-0.217		
SIZE MOD FACTOR	kb	0.839274101		
	NON ROTATING MEMBER - RECTANGLE			
	de=0.808*sqrt(h*b)			
	h	88.9	3.5 in	
	b	25.4	1 in	
	de	38.39539144		
	7.62mm<D<51mm			
	kb=1.24*de^-0.107			
LOAD MOD FACTOR	kc	1		
	BENDING kc=1			
TEMP MOD FACTOR	kd	1		
	ASSUME ROOM TEMP			
RELIABILITY FACTOR	ke	0.702		
	RELIABILITY OF 99.99%			
MISC FACTOR	kf	1		
	ASSUME NO MISC			
CRIT ENDURANCE LIMIT	Se	97.61071518	Mpa	
	Se=Se'*ka*kb*kc*kd*ke			
AMPLITUDE COMPONENT	siga	3.11375	MPa	
	sigmamax	6.827	MPa	*FROM FEA*
	sigmamin	0.5995	MPa	
	siga=(sigmamax-sigmamin)/2			
MIDRANGE COMPONENT	sigm	3.71325	MPa	
	sigm=(sigmamax+sigmamin)/2			
MOD-GOODMAN				
SAFETY FACTOR	n	24.28198235		
	1/n=(siga/Se)+(sigm/Sut)			
	1/n	0.041182799		

Figure V.4: Fatigue Calculations for Linkages in Excel

VI. Connecting Rod Analysis

a. Model and Drawing of Component

The connecting rod was modelled in Onshape and can be seen in the assembly below in the isolated state, with both the isometric and side view shown. This is the component which connects both sets of linkages to the power screw, allowing them to be driven in unison. The material selected for this component is structural A36 steel with zinc plating to for a higher strength and corrosion resistant design. This connecting rod is made up of two rods welded to a block which will have a threaded hole for the lead screw. Since the weld connection will be further analyzed in a later report, the rod portion of the beam was primarily analyzed in this section.

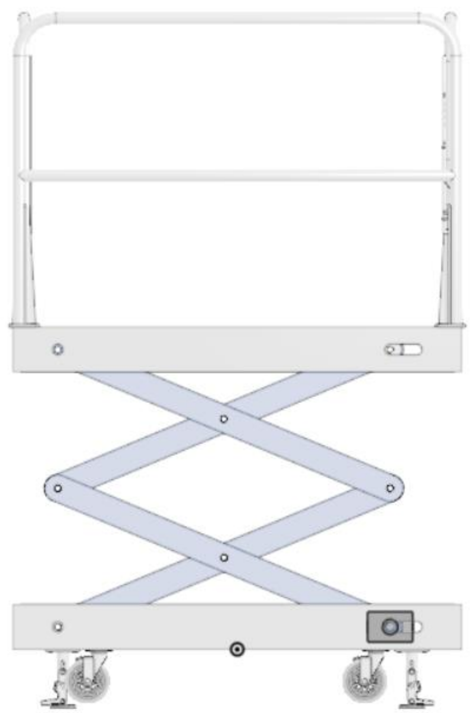

Figure VI.1: Isolated Connecting Rod in the Raised Position, for Isometric and Side Views

b. Free Body Diagram

The free body diagram weas drawn for the connecting rod to show the drive force and reaction forces of the linkages acting on it in the XZ plane, which can be seen below in Figure V.2.

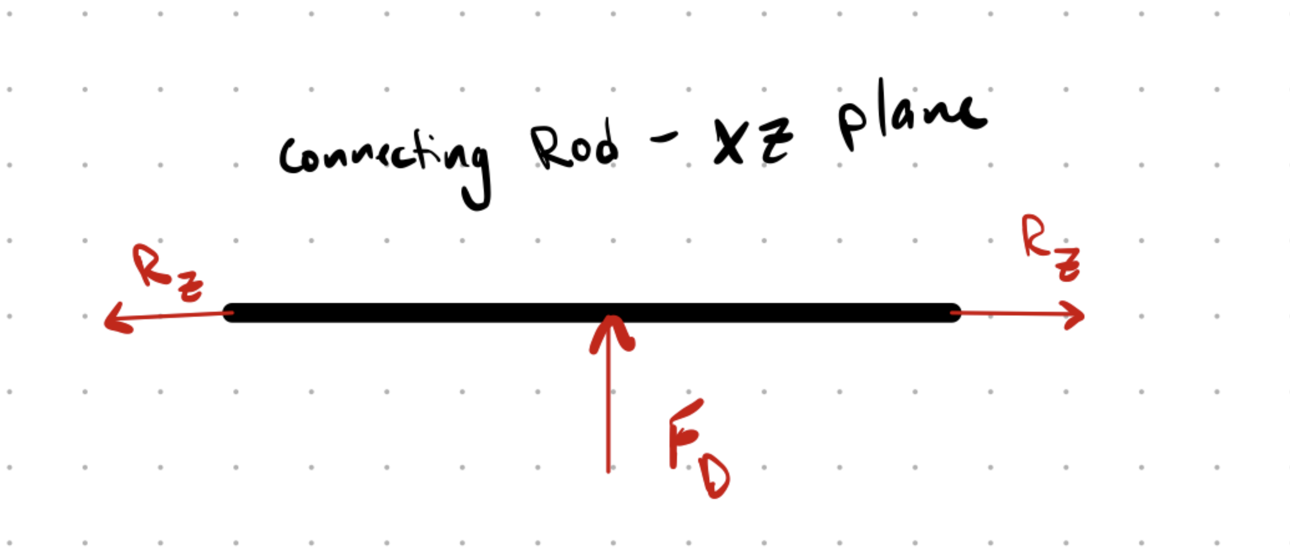

Figure VI.2 FBD of Connecting Rod

c. Stress Calculations and Finite Element Analysis

The connecting rod was analyzed using SolidWorks FEA to ensure that it will not fail. However, due to its nature as a part of the drive system, it is currently unknown what the drive force will be. This will be further explored in the Power Transmission report. Therefore, the weld connection was ignored, and the rod sections were analyzed to check for bending. Due to this, the connecting rod was treated as two beams, which would be symmetrical on either side of the rod. The stresses shown below in Figure VI.3 illustrate the bending stresses in the rod due to its connection with the linkages and its interface with the bottom plate. The maximum stress in this component was found to be 8.141 MPa.

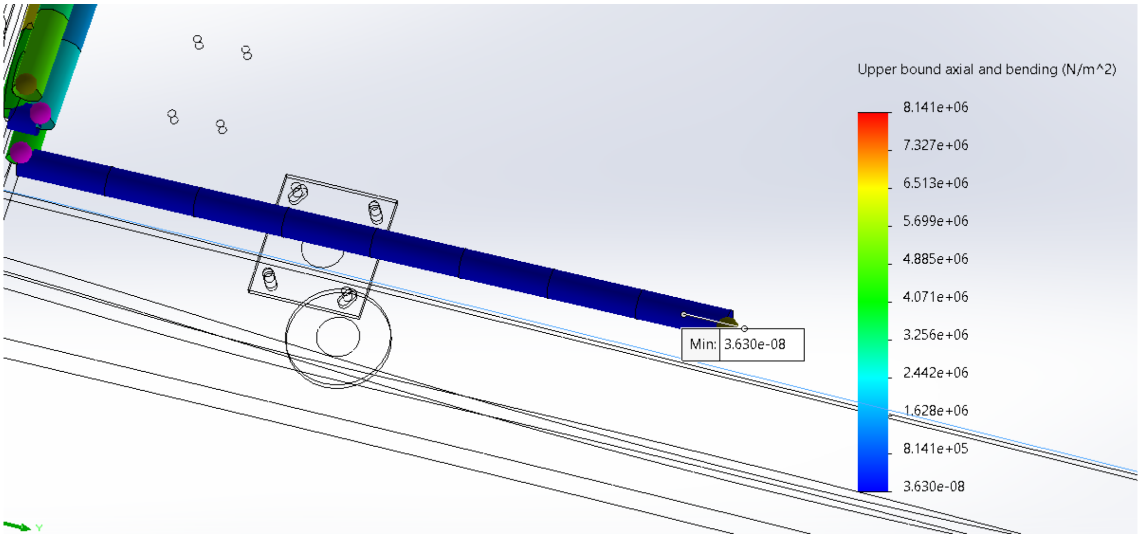

Figure VI.3 Stress Map of Connecting Rod

d. Buckling Analysis

There is no buckling potential on the connecting rod since there is no significant compressive loading along the axis.

e. Fatigue Analysis

As the scissor lift undergoes cyclic loading and is raised and lowered over the course of its life, the connecting rod will undergo fatigue due to being driven back and forth. Using the ultimate tensile strength of A36 Steel and treating the connecting rod as a circular beam, the following calculations were carried out in Excel and can be seen below in Figure VI.4. The maximum and minimum stresses on the connecting rod were taken from the previous SolidWorks simulation. From these calculations, it was determined that infinite life was achievable with a factor of safety of 20.48.

FATIGUE CALC FOR CONNECTING ROD - STRUCTURAL STEEL

TENSILE STRENGTH	Sut	400	MPa	
	MIN TENSILE STRENGTH FOR A36 STEEL			
ENDURANCE LIMIT	Se'	200	Mpa	
SURFACE COND FACTOR	ka	0.828374202		
	ka=a*Sut^b			
	SURFACE FINISH: COLD DRAWN			
	a	3.04		
	b	-0.217		
SIZE MOD FACTOR	kb	0.905934259		
	NON ROTATING MEMBER - CIRCLE			
	de=0.370*d			
	D	50.8	2 in	
	de	18.796		
	7.62mm<D<51mm			
	kb=1.24*de^-0.107			
LOAD MOD FACTOR	kc	1		
	BENDING kc=1			
TEMP MOD FACTOR	kd	1		
	ASSUME ROOM TEMP			
RELIABILITY FACTOR	ke	0.702		
	RELIABILITY OF 99.99%			
MISC FACTOR	kf	1		
	ASSUME NO MISC			
CRIT ENDURANCE LIMIT	Se	105.3635408	Mpa	
	Se=Se'*ka*kb*kc*kd*ke			
AMPLITUDE COMPONENT	siga	4.070499982	MPa	
	sigmamax	8.141	MPa	*FROM FEA*
	sigmamin	3.63E-08	MPa	
	siga=(sigmamax-sigmamin)/2			
MIDRANGE COMPONENT	sigm	4.070500018	MPa	
	sigm=(sigmamax+sigmamin)/2			
MOD-GOODMAN				
SAFETY FACTOR	n	20.48795839		
	1/n=(siga/Se)+(sigm/Sut)			
	1/n	0.048809158		

Figure VI.4: Fatigue Calculations for Connecting Rod in Excel

VII. Summary

Since the Scissor Lift is intended to provide a safe alternative to ladders, engineering safe and reliable components is crucial to the success of the product. In this report, the team analyzed the 4 main components and determined the safety factor for each under expected loading conditions. Overall, our minimum safety factor for fatigue was 2.74, which is more than enough to be confident in our design. The maximum safety factor was 572.83, which is a bit higher than needed, but will not result in a significant amount of excessive cost or reduced mobility.

VIII. Appendix

Appendix A – Group Member Contributions

This contribution distribution was agreed on by all group members.

PJ Fries: 25% Contribution: Component Modelling, SolidWorks Assembly Work, SolidWorks Analysis, Report Writing

Emilie Hardel: 25% Contribution: Component Modelling, Free Body Diagrams and MATLAB Code, SolidWorks Analysis, Report Writing

Lauren Kreder: 25% Contribution: Component Modelling, Fatigue Calcs and Excel Calc, Introduction, Report Writing

Justin Miller: 25% Contribution: Component Modelling, Buckling Assessment, Summary, Report Writing

Appendix B – MATLAB Code for Reaction Force Calculations

```
clear

clc


%system force calculator


syms F_D F_P A_x A_y B_x B_y C_x C_y D_y E_x E_y G_y H_x H_y I_x I_y J_x J_y K_x K_y L L_AD L_AB theta W_link W_topPlate rho_link rho_topPlate V_link V_topPlate W_bottomPlate V_bottomPlate rho_bottomPlate L_DCentroid


g = 386.089; %in/s^2

rho_topPlate = 0.1; %lbm/in^3, density of Al

rho_bottomPlate = 0.1;

rho_link = 0.291; %structural steel

V_link = 175.777; %in^3

V_bottomPlate = 4759.928/2;

V_topPlate = 4762.243/2;


W_link = 50; %lbf

W_topPlate = 483.452;

W_bottomPlate = 483.217;
```

```
F_P = 100;

theta = 9;

L = 24;

L_AB = 37.063;

L_AD = 40;

%constants: W_link, W_topPlate, W_bottomPlate, F_P, all lengths

%variables: 15 unknowns

%Overall system- external loads and ground reactions


OS_sumFx = B_x - A_x == 0;

OS_sumFy = A_y +B_y -F_P - 4*W_link - W_topPlate/2 - W_bottomPlate/2 == 0;

OS_sumMA = B_y*L_AB - F_P*L_AD/2 -(W_link+W_topPlate+W_bottomPlate)*L_AD/2 == 0;

sol = vpasolve([OS_sumFy, OS_sumMA], [A_y, B_y]);

A_y = sol.A_y

B_y = sol.B_y


%linkage force/moment balances



%top plate


L_CCentroid = 22.76; %from onshape

L_CD = 2*L*cosd(theta);


TP_sumFy = -F_P - D_y - C_y - W_topPlate == 0;

TP_sumMC = -W_topPlate*L_CCentroid - F_P*L_CD/2 - D_y*L_CD;

topPlateSol = vpasolve([TP_sumMC,TP_sumFy],[C_y,D_y])

topPlateSol.C_y

topPlateSol.D_y


%bottom plate

L_ECentroid = 23.21;

BP_sumFy = E_y +G_y == 0;

BP_sumME = G_y*L_CD - W_bottomPlate*L_ECentroid;

BPsol = vpasolve([BP_sumME,BP_sumFy],[E_y, G_y]);
```

```
BPsol.E_y

BPsol.G_y


%link IG


IG_sumFx = I_x -H_x - F_D == 0;

IG_sumFy = I_y - H_y - G_y - W_link == 0;

IG_sumMI = -H_x*sind(theta)*L - H_y*L*sind(90-theta) - W_link*L*sind(90-theta) - F_D*2*L*sind(theta) - G_y*2*L*sind(90-theta) == 0;


%link EK


%compatibility

%comp1 = E_x == F_D;


EK_sumFx = -E_x + H_x +K_x == 0;

EK_sumFy = K_y + H_y - E_y - W_link == 0;

EK_sumMK = H_x*L*sind(theta) - H_y*L*sind(90-theta) + W_link*L*sind(90-theta) -E_x*2*L*sind(theta) + E_y*2*L*sind(90-theta);


%link CK


CK_sumFx = C_x + J_x - K_x == 0;

CK_sumFy = C_y +J_y - K_y - W_link == 0;

CK_sumMC = J_x*L*sind(theta) + J_y*L*sind(90-theta) - W_link*L*sind(90-theta) - K_x*2*L*sind(theta) - K_y*2*L*sind(90-theta);


%link DI


DI_sumFx = -J_x - I_x == 0;

DI_sumFy = D_y - J_y - I_y - W_link == 0;

DI_sumMD = -J_x*L*sind(theta) + J_y*L*sind(90-theta) + W_link*L*sind(90-theta) - I_x*2*L*sind(theta) + I_y*2*L*sind(90-theta);


solution =
vpasolve([IG_sumFy,IG_sumMI,IG_sumFx,EK_sumMK,EK_sumFy,EK_sumFx,CK_sumMC,CK_sumFy,CK_sumFx,DI_sumMD,DI_sumFy,DI_su
mFx,TP_sumFy, TP_sumMC],[F_D C_x C_y D_y E_x E_y G_y H_x H_y I_x I_y J_x J_y K_x K_y]);
```

```
F_D = double(solution.F_D)

C_x = double(solution.C_x)

C_y = double(solution.C_y)

D_y = double(solution.D_y)

E_x = double(solution.E_x)

E_y = double(solution.E_y)

G_y = double(solution.G_y)

H_x = double(solution.H_x)

H_y = double(solution.H_y)

I_x = double(solution.I_x)

I_y = double(solution.I_y)

J_x = double(solution.J_x)

J_y = double(solution.J_y)

K_x = double(solution.K_x)

K_y = double(solution.K_y)
```