Mechanical and Manufacturing Engineering

MECH 4860 Engineering Design

Crane Brake Mounting Block Design

Final Design Report

Prepared for

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Manitoba Hydro

Prepared by

Team 16: Craynium Designers

December 1, 2014

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November 24, 2015

Prof. Paul E. Labossiere
75 Chancellors’ Circle

E1-546, EITC
Faculty of Engineering
University of Manitoba

- CRAYNIUM DESIGNERS PROJECT REPORT SUBMISSION -

Dear Prof. Paul E. Labossiere

The team Craynium Designers is pleased to present to you our initial report entitled*Crane Brake Mounting Block Design* on December 1, 2014.

This report discusses the methods and analysis used to design a crane brake mounting blockthat would be capable of handling the loads and torque reactions caused by the crane brake. The team initially brainstormed 26 design concepts. Through careful screening and scoring of each design the team selected three initial design concepts that would be further modified. The three designs consisted of the 2 block, prism sheet metal, and the hourglass design. The three concepts were later modeled using Solidworks, and FEA was performed to determine how the mount will react to the loading forces.

If there are any inquiries regarding this report do not hesitate to contact any members of the team via their University of Manitoba email.

Sincerely,

Kelsie Sweryda
Team Lead

Document Enclosed: *Crane Brake Mounting Block Design*

Document enclosed*: Crane Brake Mounting Block Design – Final Design Report*

Executive Summary

This report explains the design methods, and the necessary analysis needed to develop a crane brake mounting block made from G40.21 44W steel. This report also covers our sponsoring company's background, the problem statement that was presented by the customer, and needs we identified in order to develop the optimum design for the customer.

Initially the team developed 26 rough brake mount ideas. Through careful screening and scoring of each mount idea, three final designs were selected for the initial designs. These designs were carefully analysed based on static stresses obtained through finite element analysis (FEA) in order to determine the minimum thickness of the steel. Our FEA showed that the minimum thickness of the steel required was very small. As a result, we selected 1/8" steel which is the thinnest steel commercially available.

The 3 selected concept designs we selected were; the hourglass, prism sheet metal, and the 2 block design. These designs were modelled using Autodesk Inventor and Solidworks in order to determine the stress distributions obtained from the applied loading forces. The designs were required to support the weight of the brake, which was 145 pounds as well as handle 300% of the motors' full load torque. This value was calculated to be 546 foot pounds. FEA was performed on each initial design, and modifications were performed in order to determine the optimum dimensions for each design. Finally the three designs were evaluated in terms of manufacturing and material cost and engineering drawings were produced for each design.

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1 Introduction

Manitoba Hydro is our sponsoring company for this design. The company would prefer a brake mount block capable of supporting the reaction forces the brake exerts when it is engaged. This report will outline the company background, the problem statement, methods of analysis, and the selected design for this project.

1.1 Problem Background

Manitoba Hydro is a crown corporation that was founded in 1961. It is a major electric powered and natural gas utility located in Manitoba. [1]. The company currently operates 15 generating stations and supplies powered to over 520,000 electric customers and 260,000 natural gas customers [1]. The generating station that is considered for this project is the Pine Falls Generating Station. The station is located on the lower stretch of the Winnipeg River and was built in the late 1940's [1]. The station utilizes six turbine generators. Each turbine rotates at a speed of 94.7 rpm in order to generate annually an average of 620 million kWh [1].

Two powerhouse cranes are located at the Pine Falls generating station on the north and south sides of the plant. Manitoba Hydro intends to refurbish the powerhouse cranes in order to bring them up to current maintenance and safety standards. This ensures safe and reliable operation during future unit operation. Manitoba Hydro currently uses the Crane Manufacturers Association of America, CMAA, safety standards that will be discussed later.

The refurbishment of the powerhouse cranes includes the replacement of the main hoist motor. As a result of replacing the motors, a new braking system will need to be implemented on the main hoist. Manitoba Hydro requires a new mounting block design for the new brake system, which was previously designed by a contractor. Our team is required to design three brake mount block designs which will be compared to the design supplied by the contractor. Our client informed us that the contractor's design is already being implemented in the braking system and is currently being operated. Even though the contractors' design is already in use at the Pine Falls Generating Station, the company is interested in implementing one of our designs into identical braking systems in other generating stations that are also being refurbished to safety standards.

1.2 Problem Statement

The new brake mount block designed by our team, is to be based on the centerline of the main hoist shaft and the top of the trolley platform as shown in Figure 1. This image was supplied by the client in order to supply the team with the mandatory dimensions.

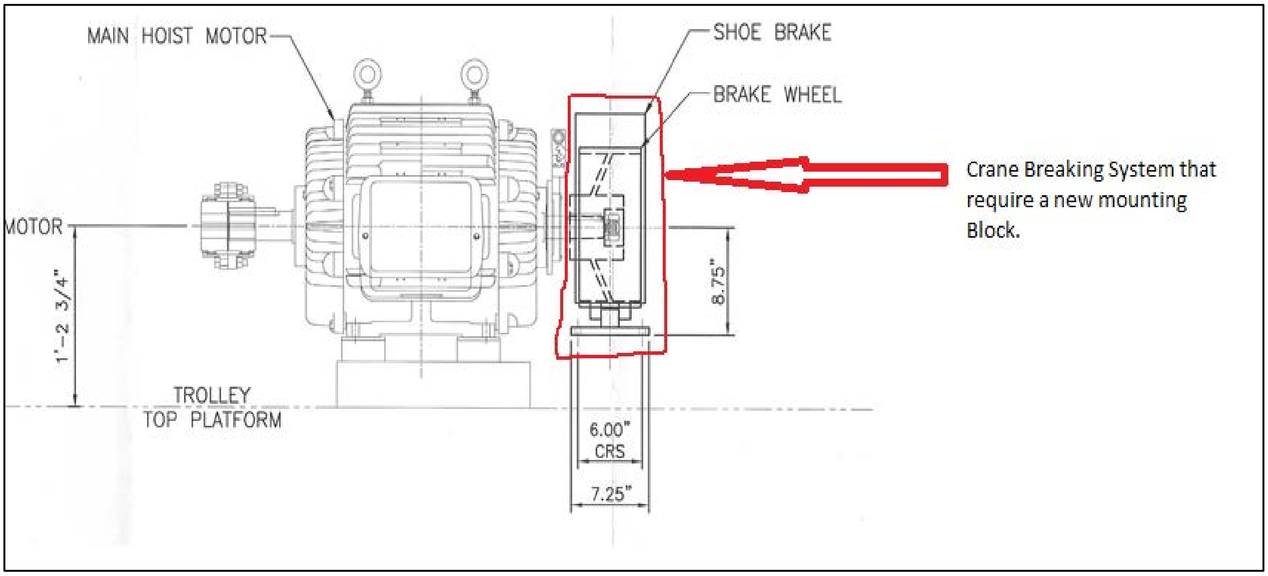


Figure 1: Crane Brake System (Supplied by the client).

Ultimately, our client requires that our team create three different concepts for a new brake mounting block. The block must correctly align the brake with the centerline of the motor shaft, and effectively bear the weight of the brake and the applied torque from the main hoist motor. The new block designs will also allow for the brake to be easily mounted and fastened, as well as being easily removable for maintenance purposes. Lastly, our client requires us to show thorough analysis on each of the three block designs and provide a recommendation on which of the three designs should be implemented in the new brake system.

1.3 Project Objectives

The purpose of this project is to apply practical and theoretical engineering knowledge to formulate a solution for the problem presented to us by our client. Our main objective is to provide our client with three different design concepts that satisfy the criteria stated in our problem statement, as well as fit within the constraints and limitations of our project.

The main objective in this project can be broken into three subcategories. First collect all the information required to fully define the problem and begin developing solutions for a new brake mounting block. This includes collecting information from our client, research from the internet and literature resources, and gathering information from project stakeholders. Second, use all information gathered in step one to begin formulating as many conceptual designs as possible. This step also includes the performance of preliminary analysis in the form of concept screening and scoring to narrow down the number of concepts before the final step. Finally, we will perform detailed analysis and refinement of the concepts using FEA and CAD software to select our final three designs to be presented to our client.

1.4 Needs and Target Specifications

For this project our group is expected to deliver three conceptual designs with detailed analysis verifying that each design satisfies the fit, form, and function required by our client. To ensure that we meet the expectations of our customer, we developed a set of project needs based on the information that Manitoba Hydro provided. TABLE I represents the developed list of needs. Each need was rated on a scale of one to five based on its’ importance in meeting the clients’ expectation.

Table II shows a list of metrics and target specifications, each of which corresponds to one or more needs. The metrics also have a rating of importance that is directly dependant on the rating of the need that it corresponds to.

TABLE I: PROJECT NEEDS

Need #	Need	Importance rating
1	The block is easily welded to the trolley and the brake is easily mounted onto block	2
2	The block satisfies current safety standards.	5
3	The block satisfies current maintenance standards.	5
4	The block is effective in bearing the applied loads and torque reactions.	5
5	The block correctly aligns the brake with the centerline of the shaft.	5
6	The block Remains Rigid throughout its lifetime.	5
8	The block fits within the space provided.	5
9	The block Design is cost efficient.	2
10	The block lasts a minimum of 50 years.	4
11	The block design maintains the functionality of the brake or trolley.	5
12	The block maintains the accessibility of the hoist and brake.	3

TABLE II: METRICS AND TARGET SPECIFICATIONS

Metric#	Need	Metric	Imp.	Unit	Marginal	Ideal
1	1	Total time it takes to install	2	Hr.	<2	<1
2	2, 3	Satisfies CMAA	5	N/A	Yes	Yes
3	4, 6	Allowable stresses of the block	5	MPa	Tensile/Comp/ Combined<182.05 Shear<109.25	Tensile/Comp/ Combined<150 Shear<85
4	5, 8,11, 12	Overall dimensions of the block	5	inches	h = 6, l < 18.75, w< 10	h = 6, l = 17, w= 7.25
6	9	Cost of block design	2	CAD$/ unit	<2000	<500
7	10	Block service life	4	Year	> 20	> 40
8	9, 8	Mass of the block	5	lb	< 40	< 20
9	5, 9	Dimension tolerances of the block	5	inches	$\pm$ 0.12	$\pm$ 0.04

1.5 Constraints and Limitations

During our meeting with our client's representative engineer, we were provided with the list of constraints and limitations. This includes the following; the mount must be capable of bearing a torque of 546.6 lb·ft in both clockwise and counter clockwise direction. This torque is equivalent to 300% of the full load torque applied by the motor. The motor used on the crane is the NEMA B which has a full load rpm of 865 and 30 horsepower, resulting in a full load torque of 182.2. The mount will also be required to support the weight of the brake, which is 145 lbs. The material to be used in the block design is restricted to G40.21 44W only; a mild grade steel. The mount must support the brake system 6" from the trolley to allow for proper alignment of the brake with the centerline of the motor shaft. The braking system will be mounted to the block using grade 5 fasteners to suit the holes through brake. The design concepts are required to satisfy CMAA, Crane Manufacture Association of America, safety and maintenance standards. Finally the project must be completed by December 1, 2014.

2 Details of the final Designs

In this section, it is explained the methods of refining each design concept that was selected for this project. There were three initial designs and each one was carefully modified and tested using FEA on Solidworks and Inventor. Two different software were used to ensure validity of the FEA tests.

2.1 Final Design 1

This section explains the first final design. With the use of Solidworks, a model of the block design was made. Figure 5 shows the model.

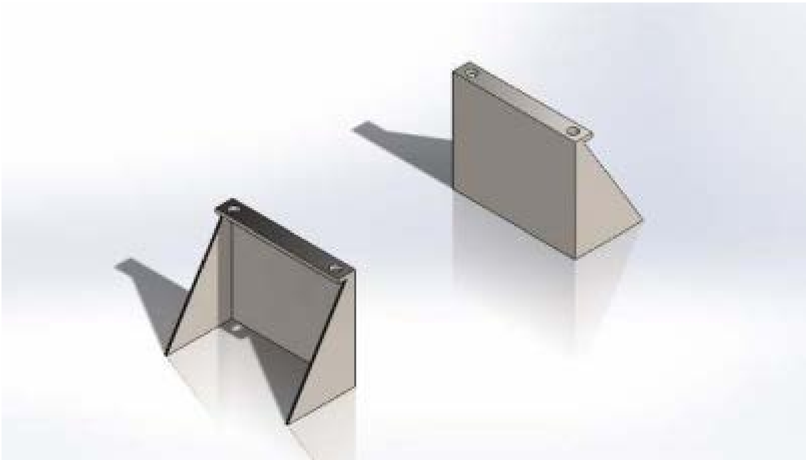


Figure 2: Solidworks model of the design

2.1.1 Design Details

This design is composed of two separate and identical structures. The design is a triangular shape that is made from four pieces; the top plate, two side plates, and a back plate. The triangular sides are designed to support the tortional forces that the mount will be supporting.

The design was tested under the known loads that it would experience. From the FEA analysis the maximum stress of the design was found to be 2.75MPa. Figure 6 illustrates the von mises stress of the design.

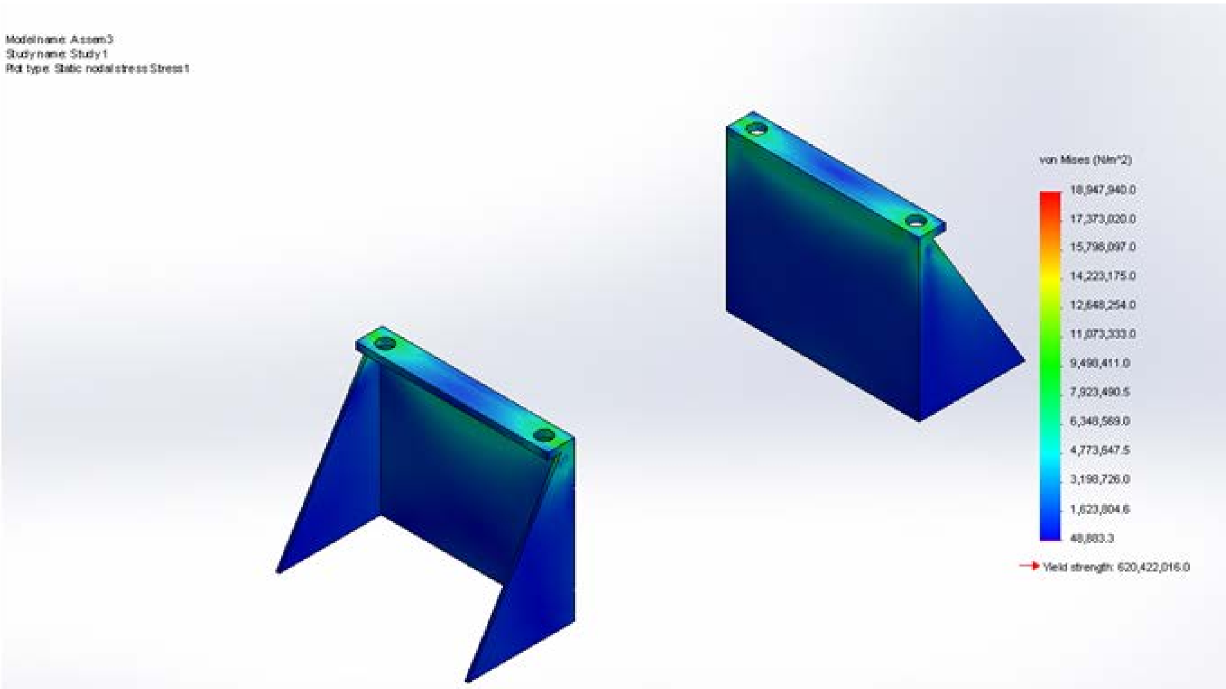


Figure 3: Stress results using FEA on Solidworks

This design had a maximum displacement of 0.000671 inches. As a result the design was adjusted in order to lower the displacement value. The sides of the mount were expanded to have the same thickness of the top of the mount which is 1”.Figure 7 illustrates the second design.

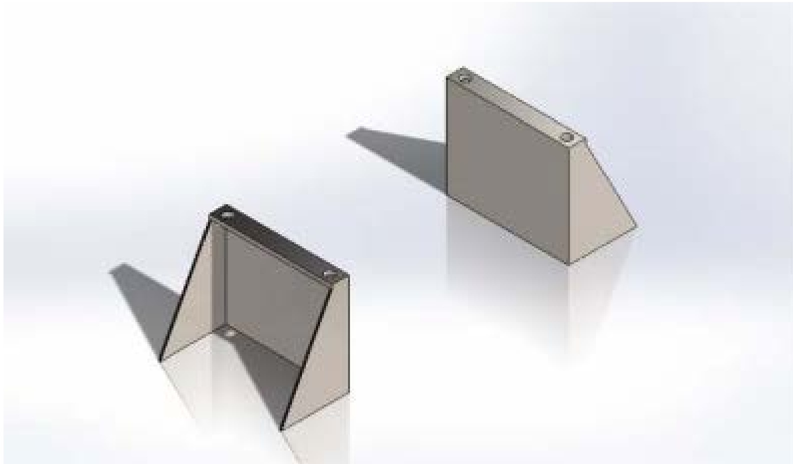
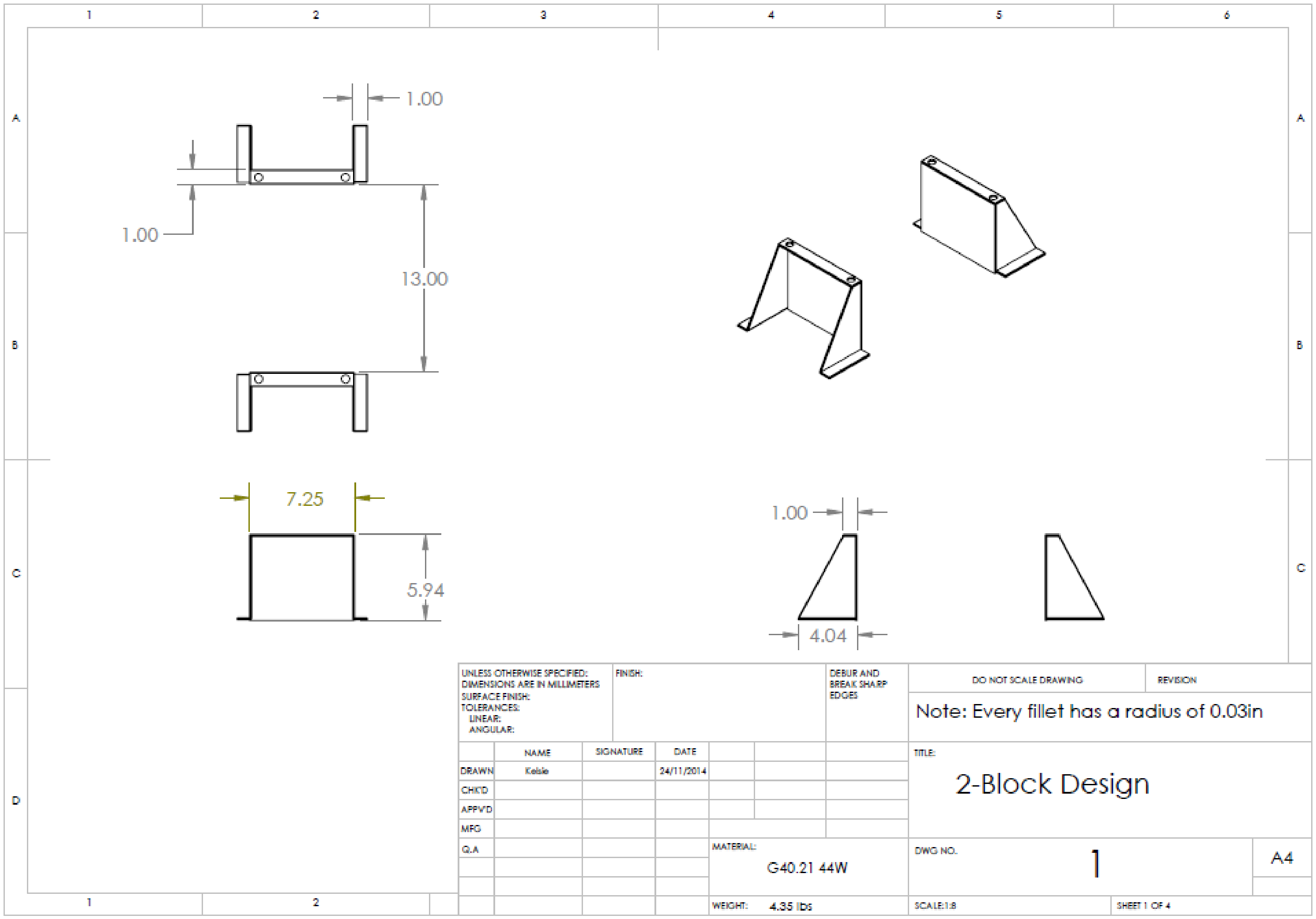


Figure 4: Modified concept design


1.00
1.00
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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBUR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
Note: Every fillet has a radius of 0.03in
NAME
SIGNATURE
DATE
DRAWN
Kelsie
24/11/2014
CHK'D
APPV'D
MFG
Q.A
TITLE:
2-Block Design
MATERIAL:
G40.21 44W
DWG NO.
1
A4
WEIGHT:
4.35 lbs
SCALE:1:8
SHEET 1 OF 4

Figure 5: Design 1 drawing

2.1.2 Design Verification

FEA was tested on the design using both Inventor and SolidWorks. The model was tested with and with out the brake brake shoes. This was to observe how the part reacts with the support of the brake shoe since this structure help prevents the top plate from bending. The stresses in the model must meet our client's requirements: allowable compressive stress of $\sigma_{C_{all}} = 0.6\sigma_{yp} = 26.4\ ksi$, allowable tensile stress of $\sigma_{T_{all}} = 0.6\sigma_{yp} = 26.4\ ksi$, allowable shear stress of $\tau_{all} = 0.36\sigma_{yp} = 15.8\ ksi$, and combined stress $\sigma_{comb} = \sqrt{(\sigma_z{}^2 + \sigma_y{}^2 - \sigma_z\sigma_y + 3(\tau_{yz})^2} \leq \sigma_{T_{all}}$.

2.1.3 Finite Element Model 1 Results

An FEA was performed on the second design in order to determine how the mount would react to the forces. Figure 5 illustrates the reaction of the brake mount.

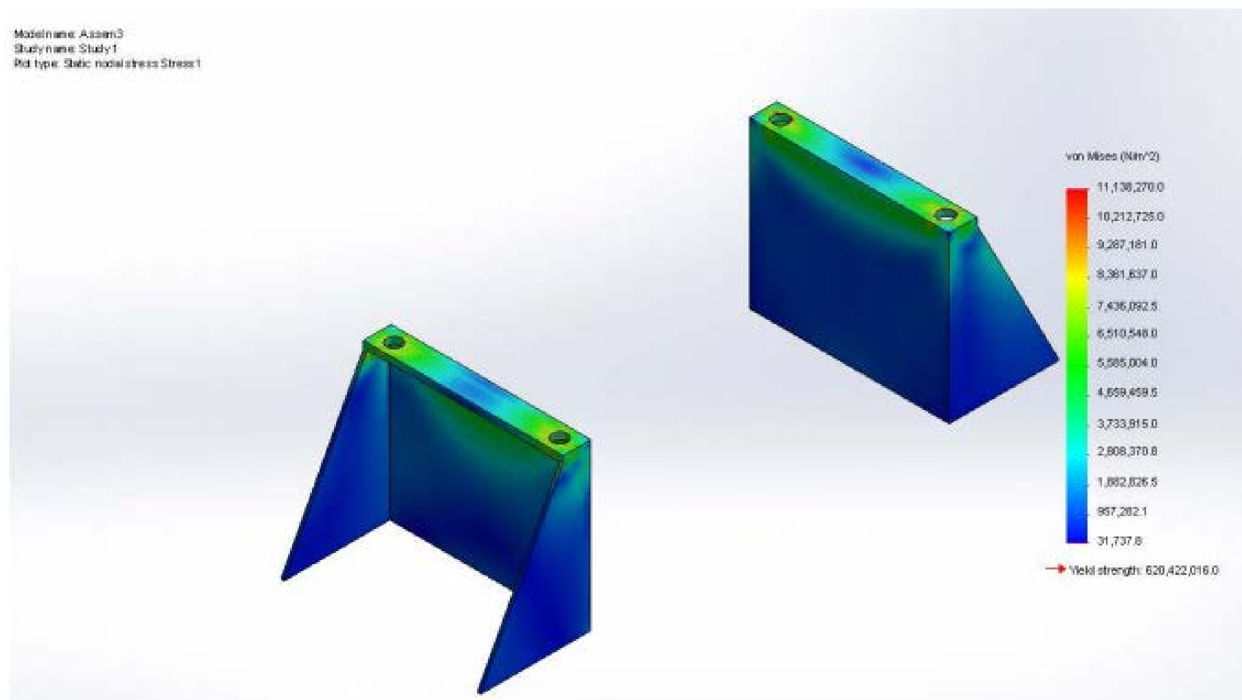

Figure 6: Stress analysis of modified design

The maximum stress of the design was found to be 1.616ksi and the maximum displacement was 0.000534 in. This design showed to be better than the previous design but it was determined that sheet metal would be a better solution for the design. Since the minimum thickness available for the specified steel was 1/8", the design was modeled as if pieces were welded together. Figure 6 illustrates the modified design.

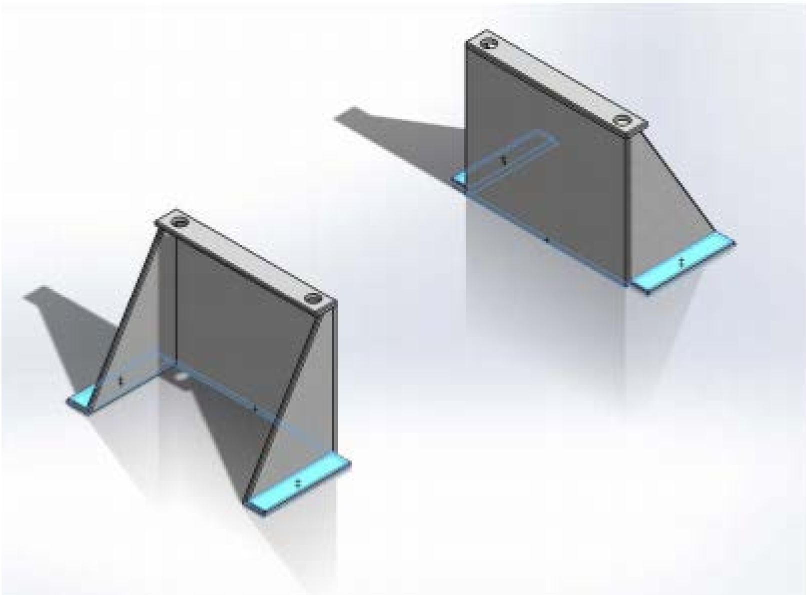

Figure 7: Final design

The bottom of the plate has a base for simplicity of welding and there are four separate pieces that are required to be welded together. The pieces are the top plate, back plate, and two side plates that are 1/8" thick. This design was tested using FEA on Solidworks. Figure 7 illustrates the von mises stress results of the design.

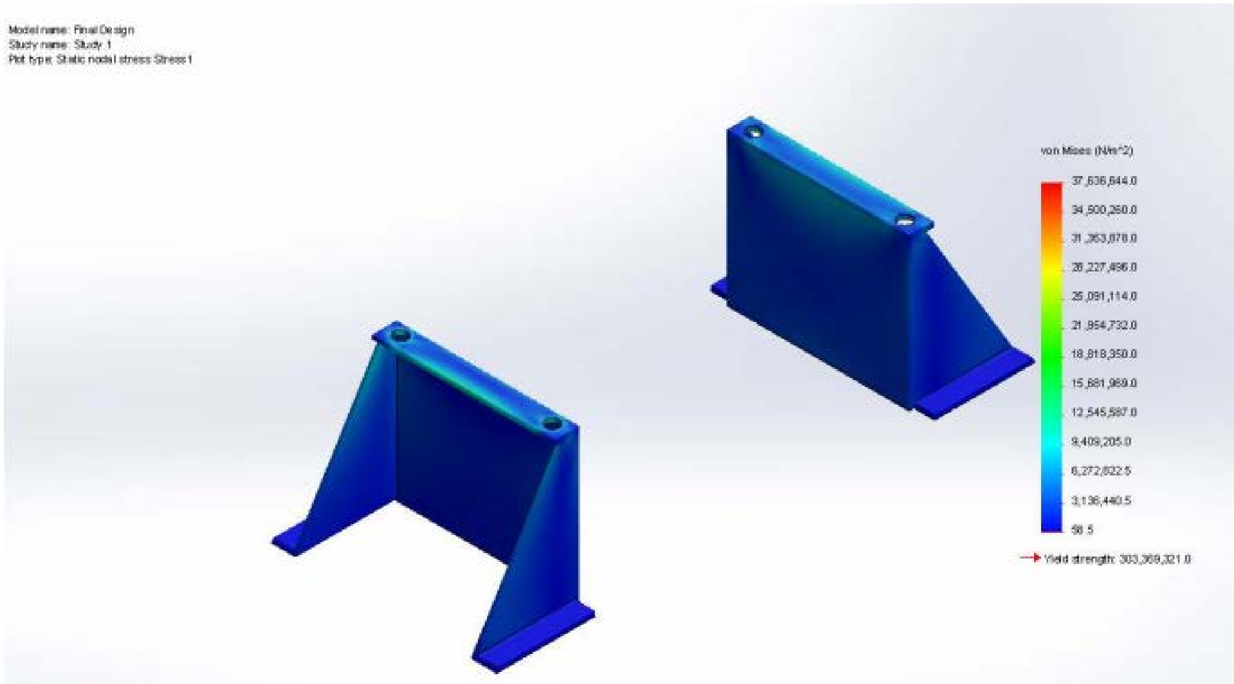

Figure 8: Stress analysis of final design

This design had a maximum stress of 5.459ksi and a maximum displacement of 0.000894 in. The test also stated that the design had a minimum factor of safety of 3.5. Figure 8 illustrates the factor of safety results.

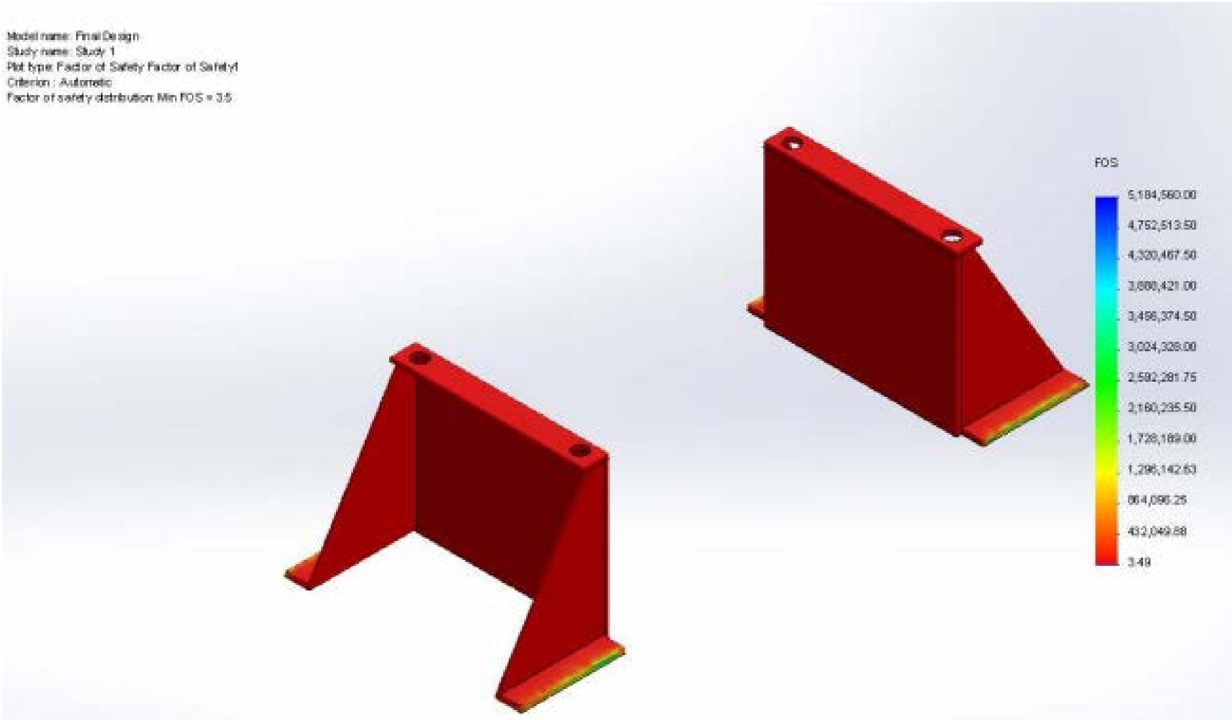

Figure 9: Factor of safety of final design

The design appears to absorb the stress in all locations of the mount but most of the stress is concentrated at the bolt hole locations. This design was determined to be the final design of the two block concept. Figure 9 illustrates the brake mount and the dimensions of the design.

The block was also tested using Inventor, with this software we were able to obtain the stress information listed in below.

TABLE 3: DATA FOR DESIGN 1

Name	Minimum	Maximum
Volume	11.6603 in^3	
Mass	3.30686 lbmass	
Von Mises Stress	0.000367758 ksi	14.9182 ksi
1st Principal Stress	-4.85078 ksi	8.99142 ksi
3rd Principal Stress	-16.8387 ksi	0.693853 ksi
Stress XX	-10.2973 ksi	7.16129 ksi
Stress XY	-1.73189 ksi	3.59085 ksi
Stress XZ	-6.05473 ksi	6.09392 ksi
Stress YY	-7.98698 ksi	6.82263 ksi
Stress YZ	-4.23337 ksi	4.8378 ksi
Stress ZZ	-13.4146 ksi	7.4669 ksi

Based on the data provided by Inventor, the combined stress was calculated based on the stress in the x and z direction. By using the combined stress equation it was found that the combined stress for this design was 12.84 ksi.

[redacted]

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[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]

2.1 Final Design 2

This section includes features of the second design, with drawings, engineering verifications and cost of material and manufacturing. Final design 2 has three components, two back-to-back C-shape beams made of bent metal sheets and two metal strips as bridges connecting the beams. See Figure 12.

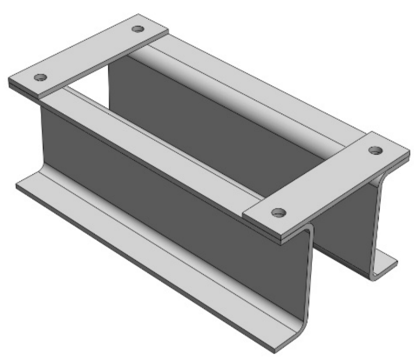

Figure 10: Final Design 2

2.1.1 Design Features

The model is 16 inches long, 8 inches wide, 6 inches tall, and its weight is 22.14 lbs. Each beam is 5.75 inches tall, its top and bottom flange is inch wide. The beams are 2 inches apart. The thickness of the sheets is ¼ inch. On the top of each flange, two holes of 0.56 in diameter are drilled to attach the bolts. The centre of each hole is 1 inch from the end edge and 1 inch from the side edge. The two bridging strips are 8 inches long and 2 inches wide. Each strip has a hole at each end, and the centre of the hole is 1 inch away from the edge. To be consistent with the beams, the thickness of the strips is also ¼ inch. Figure 11, presents the detailed dimension of the design.

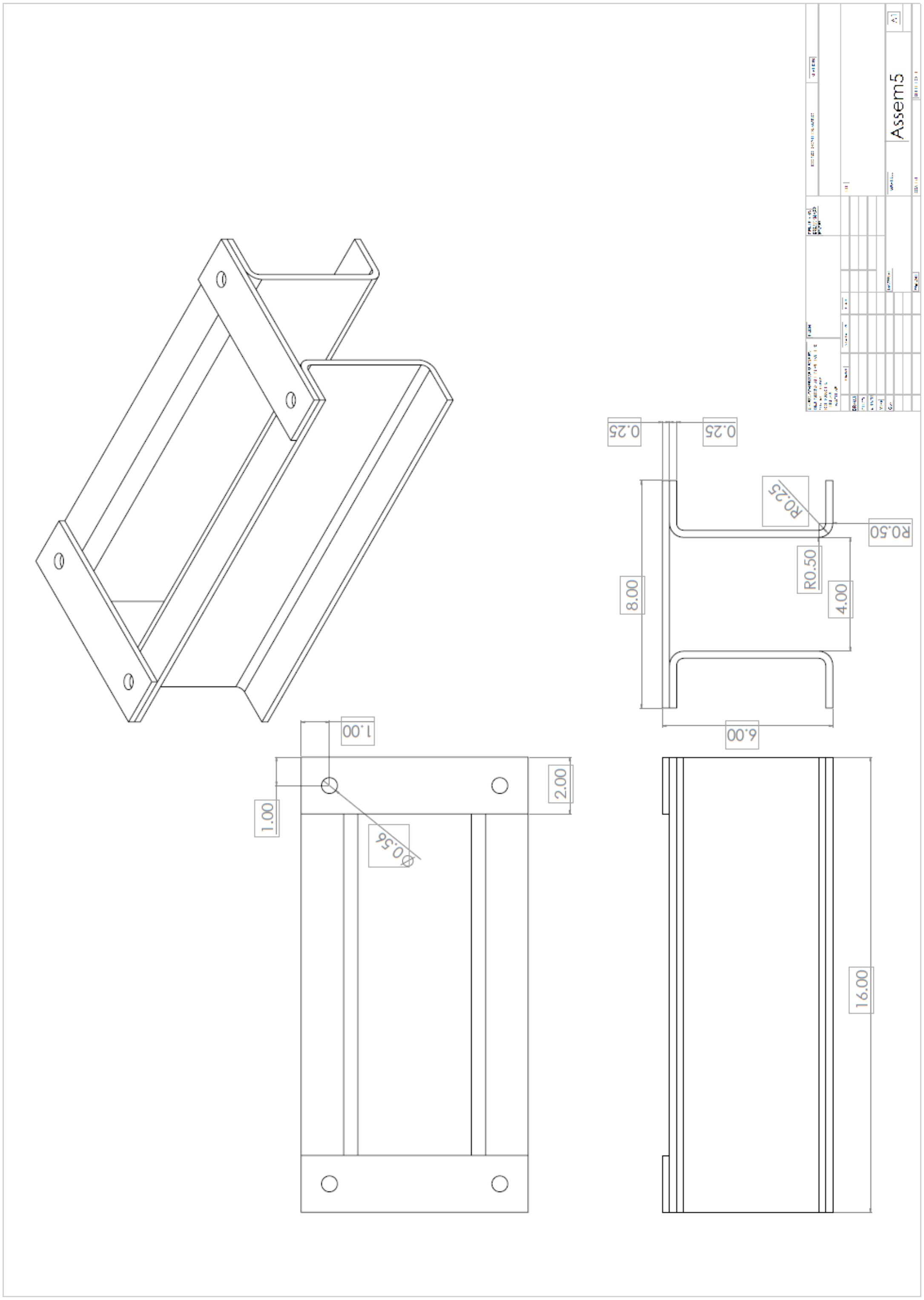

Figure 11: Final Design 2 Drawing

The vertical components of the design are to handle the weight of the brake wheel and the vertical components of the torque reaction from the shaft. The length of the beams give a larger moment of inertia, which makes the mount harder to deform from the torque reaction, since the torque reaction is the major load in this project.

The bridges stay on the top surface to connect two beams, and use their tension to prevent the beams from bending apart. The bottom flanges are designed for welding the mount onto the trolley; the top flanges are used to attach the brake wheel on the mount with bolts.

2.1.2 Design Verification

Finite element analysis is applied to the model, with both Inventor and SolidWorks. The analysis is using two approaches, the one using brake shoes attached to simulate the loads and the one without shoes. The stresses in the model must meet our client's requirements: allowable compressive stress of $\sigma_{C_{all}} = 0.6\sigma_{yp} = 26.4\ ksi$, allowable tensile stress of $\sigma_{T_{all}} = 0.6\sigma_{yp} = 26.4\ ksi$, allowable shear stress of $\tau_{all} = 0.36\sigma_{yp} = 15.8\ ksi$, and combined stress $\sigma_{comb} = \sqrt{(\sigma_z{}^2 + \sigma_y{}^2 - \sigma_z\sigma_y + 3(\tau_{yz})^2} \leq \sigma_{T_{all}}$.

2.1.3 Finite Element Model 1 Results

With Inventor, FEA are applied to the model without the brake shoes mounted on, as explained in previous section. Figure 12 shows that the maximum von Mises stress is 12.94 ksi, located at the contact surface of the beam and the bridge.

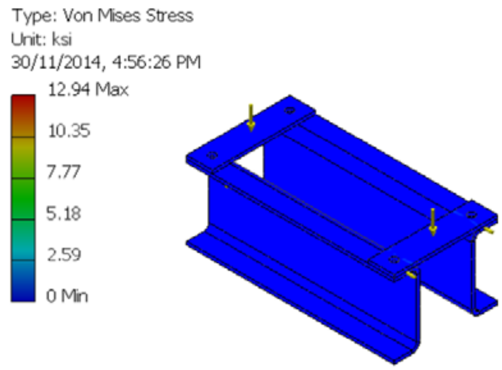

Figure 12: von Mises stress

Figure 13 shows the maximum tensile stress, 3.6 ksi approximately. The maximum tensile stress is under the allowable tensile stress 26.4 ksi.

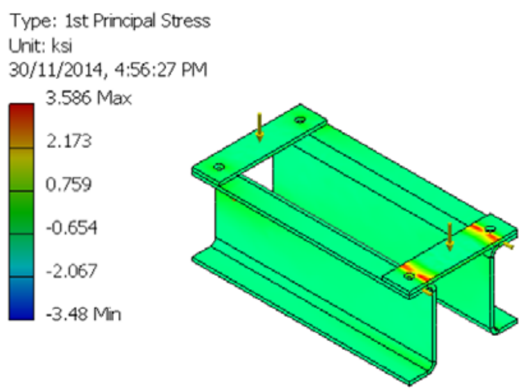

Figure 13: 1st principal stress

Figure 14 indicates that the maximum compressive stress is 16.44 ksi which is below the allowable compressive stress 26.4 ksi.

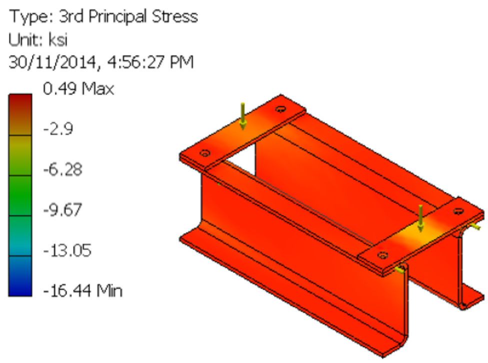

Figure 14: 3rd principal stress

The maximum shear stress is obtained, as shown in Figure 15. It is 5.1 ksi and it is under the allowable shear stress, 15.8 ksi.

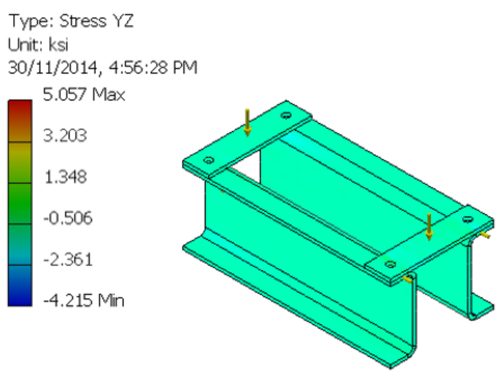

Figure 15: shear stress YZ

Table 4 includes all of the detailed stresses for further analysis. The combined Stress is calculated to be $\sigma_{comb} = 14.38\ ksi$. Use combined stress to compare with allowable tensile stress 26.4 ksi. The model passes the criterion.

TABLE 4: FEA RESULTS FROM INVENTOR

Name	Minimum	Maximum
Volume	78.0738 in^3	
Mass	22.1417 lbmass	
Von Mises Stress	0.00248132 ksi	12.9323 ksi
1st Principal Stress	-3.47762 ksi	3.58319 ksi
3rd Principal Stress	-16.4239 ksi	0.484744 ksi
Stress XX	-5.92582 ksi	1.23314 ksi
Stress XY	-0.887464 ksi	2.22621 ksi
Stress XZ	-2.62886 ksi	3.09763 ksi
Stress YY	-6.27832 ksi	2.13081 ksi
Stress YZ	-4.21235 ksi	5.05383 ksi
Stress ZZ	-14.2816 ksi	3.58049 ksi
Strain XZ	-0.000117815 ul	0.000138823 ul
Strain YY	-0.0000750412 ul	0.000142724 ul
Strain YZ	-0.00018878 ul	0.000226492 ul
Strain ZZ	-0.000395148 ul	0.000116357 ul

2.1.4 Finite Element Model 2

With SolidWorks, we use second approach for FEA, attaching shoes on the model to simulate the loads. Four bolts and nuts are used to lock two shoes, and two bridges onto the two beams perpendicularly. Loads are applied on the shoes as explained in previous section. Figure 16 shows the configuration of this model.

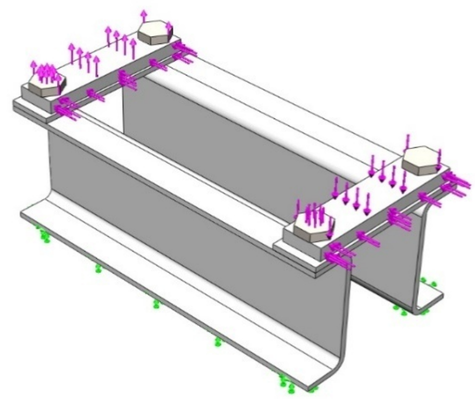

Figure 16: Load applications

Results are obtained from SolidWorks and are shown in this section. Figure 17 shows that the maximum von Mises stress of the model is 2.5 ksi.

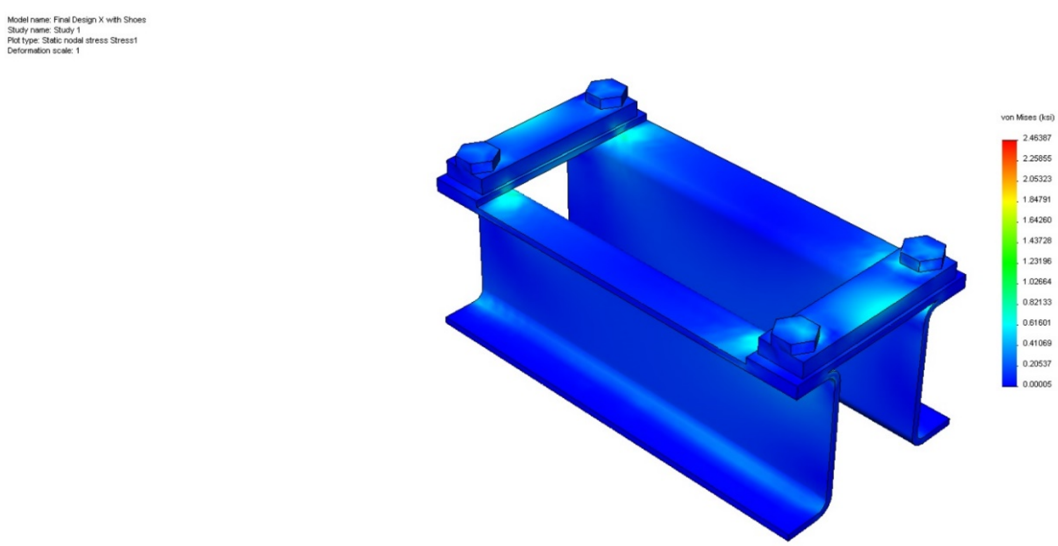

Figure 17: von Mises stress

Figure 18 shows that the maximum tensile stress is 5.0 ksi which is less than the allowable tensile stress 26.4 ksi.

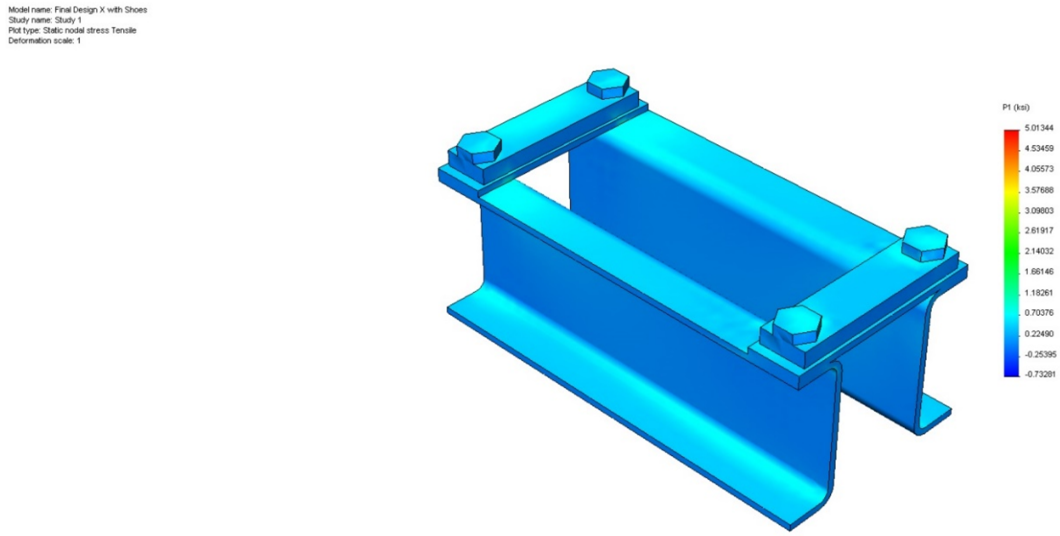

Figure 18: 1st Principal Stress

Figure 19 shows that the maximum compressive stress, 1.7 ksi, is under the allowable compressive stress, 26.4 ksi.

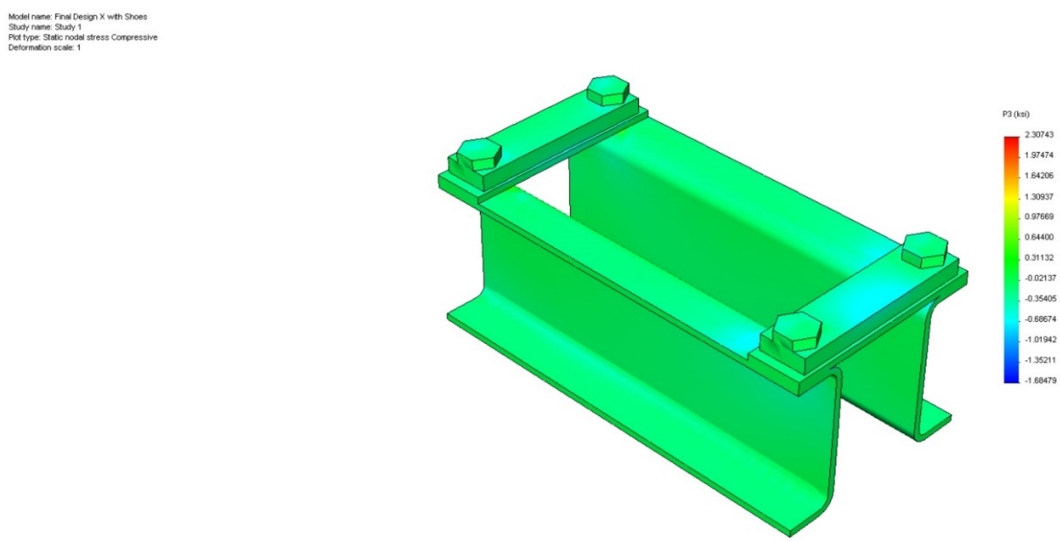

Figure 19: 3rd Principal stress

The maximum shear stress is found to be 0.65 ksi in ZY-direction, shown in Figure 20. It is below the allowable shear stress 15.8 ksi.

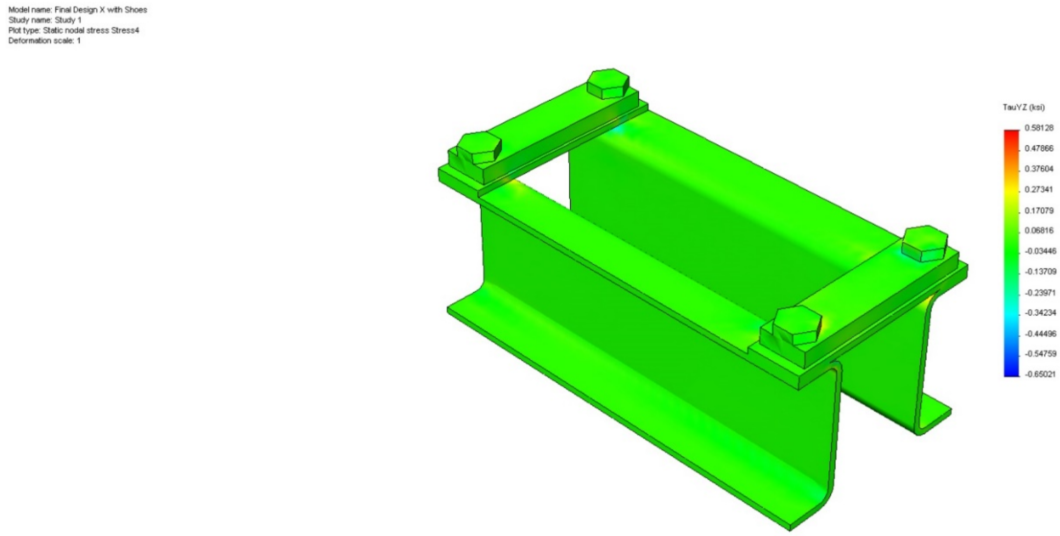

Figure 20: Maximum Shear

Table 5 includes all of the detailed stresses for further analysis.

TABLE 5: RESULTS FROM SOLIDWORKS

Name	Minimum	Maximum
Volume	78.0738 in^3	
Mass	22.1417 lbmass	
Von Mises Stress	0.00005ksi	2.46387ksi
1st Principal Stress	-0.73281ksi	5.01344ksi
3rd Principal Stress	-1.68479ksi	2.30743ksi
Stress XX	-1.24076ksi	2.88792 ksi
Stress XY	-0.49679ksi	0.26287ksi
Stress XZ	-0.30224ksi	0.30626ksi
Stress YY	-1.26713 ksi	2.44497 ksi
Stress YZ	-0.65021ksi	0.58128ksi
Stress ZZ	-1.46324ksi	4.88766 ksi

The combined Stress is calculated to be $\sigma_{comb} = 4.38\ ksi$. Use combined stress to compare with allowable tensile stress 26.4 ksi. The model passes the criterion.

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[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
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2.2 Design 3

This section describes the specific features of our third design including engineering drawings, part analysis, and approximate design manufacturing and maintenance costs. An isometric view of the design is shown in the figure below.

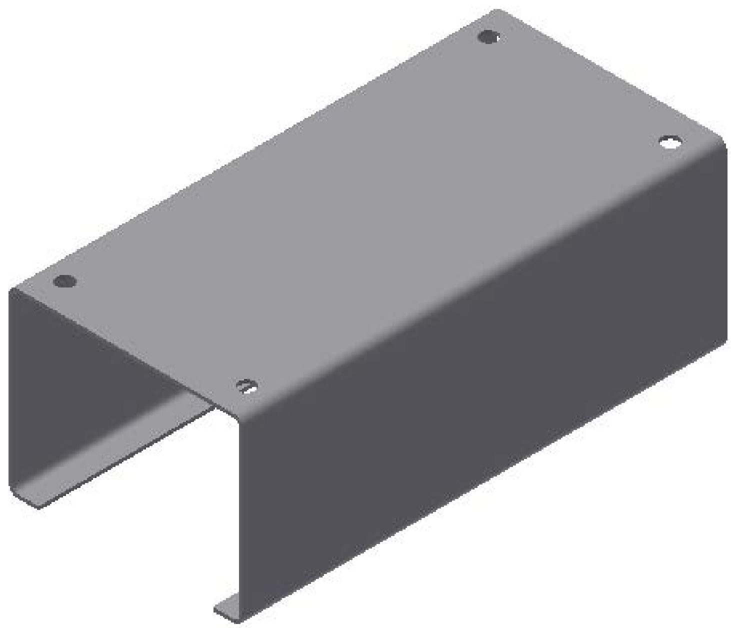
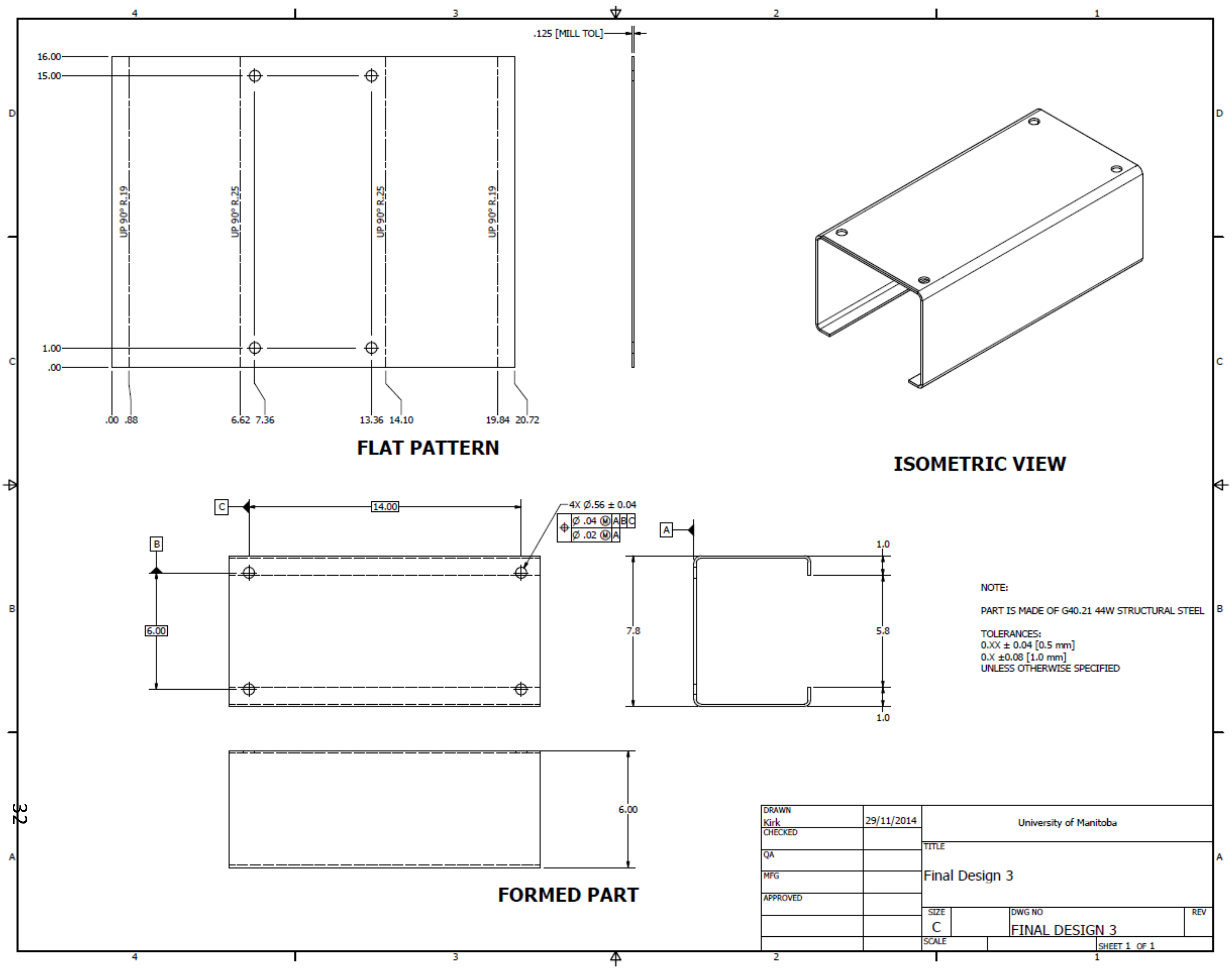


Figure 21: Final Design 3

2.2.1 Design Features

The sheet used in this design is 1/8 of an inch in thickness, which allows for the appropriate strength for bearing the applied loads of the brake and motor. The block has an overall heigh of 6 inches to allow for proper alignment of the brake with the centerline of the motorshaft. The mount also has an overall length of 16 inches and an overall width of 7.75 inches. All forming bend radii are 0.25 of an inch or 200% of the thickness;although 150% of the thickness is standard for the radius [4] we made it larger to reduce stresses in the bends of the mount. Other features of the design include 1 inch flanges at the base to provide a more stable base and more weldable area, and four clearance holes on top of the mount block for a bolted connection with brake shoes. Other features of the block include a high moment of inertia to resist possible bending due to the moment caused by the torque reaction of the motor. Further details about the design, including flat patterns and manufacturing tolerances, can be found in the following preliminary engineering drawing. Again it should be noted that the drawings in this report are preliminary drawings done by students who do not have experience creating drawing for parts used in industry.


FLAT PATTERN
UP 90° R.19
UP 90° R.25
UP 90° R.25
UP 90° R.19
16.00
15.00
1.00
.00
.00 .88
6.62 7.36
13.36 14.10
19.84 20.72
.125 [MILL TOL]
ISOMETRIC VIEW
4X Ø.56 ± 0.04
Ø .04 Ⓜ A B C
Ø .02 Ⓜ A
14.00
6.00
7.8
5.8
1.0
1.0
6.00
FORMED PART
NOTE:
PART IS MADE OF G40.21 44W STRUCTURAL STEEL
TOLERANCES:
0.XX ± 0.04 [0.5 mm]
0.X ±0.08 [1.0 mm]
UNLESS OTHERWISE SPECIFIED
DRAWN
Kirk
29/11/2014
CHECKED
QA
MFG
APPROVED
University of Manitoba
TITLE
Final Design 3
SIZE
C
DWG NO
FINAL DESIGN 3
REV
SCALE
SHEET 1 OF 1

2.2.2 Design Verification

The following section will describe in detail the two FEA simulations used to verify the mount will satisfy our client's criteria. The results of the two FEA simulations are presented using various figures and a table to summarize all the maximum and minimum values obtained for the different stresses.

2.2.3 Finite Element Model 1 Results

During the process of verifying the design we found that applying forces without the brake shoes on the design yielded more critical stresses. Figure 21 plots the results of our FEA using Von Mises stress criterion.

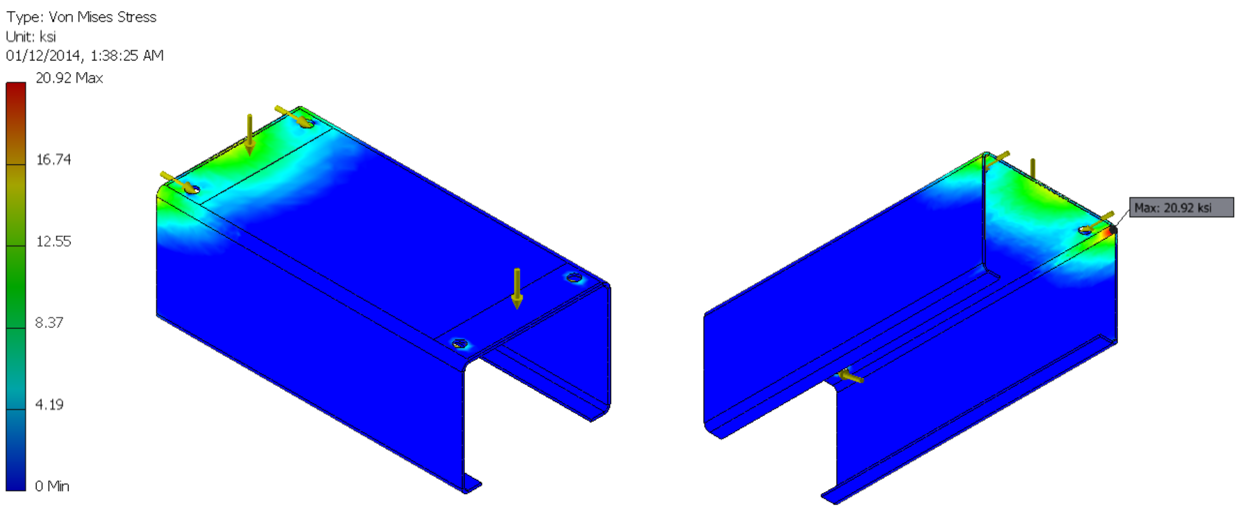


Figure 22: Von Mises Stress Criterion (second view shown for max stress location)

As seen in the figure above the maximum stress produced was located at the bend of the plate. Using this analysis we verified that the part satisfies the criteria provided by our client;

which is that the maximum stress must be less than or equal to 26.4 ksi (60% of the yield point of the material).

Also by using the Inventor software we were also able to obtain the a diagram of the 1st principle stresses in the part. This stress describes the maximum stresses (positive) in the part with no shear stresses, therefore we used this the obtain an accurate approximation of the maximum tensile forces present in the part. A schematic of the 1st principle stress distribution is shown in figure 22.

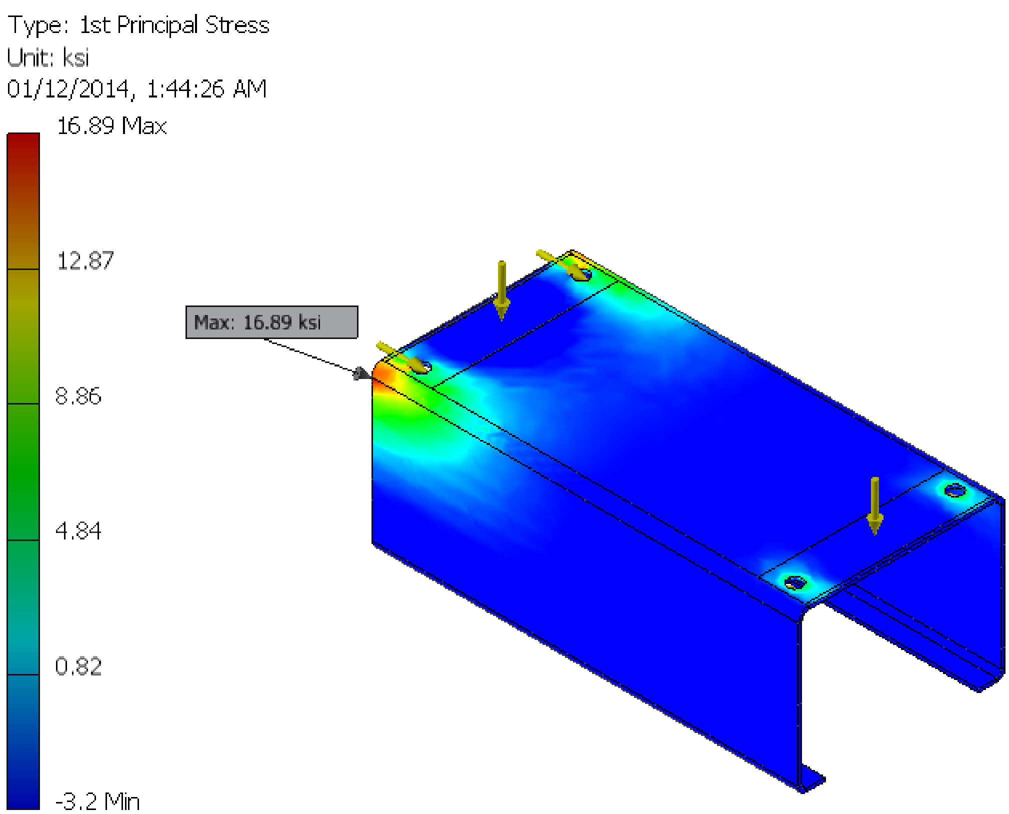


Figure 23: 1st Principle Stress

As seen above the maximum tensile stress can be approximated to be 16.98 ksi which is also below our customer's criteria of 26.4 ksi.

In the same way, we obtained the 3rd principal stresses which we used to approximate the maximum compressive stresses in the mount. Figure 23 shows a plot of the 3rd principle stress distribution in the part, where the compressive stresses are represented as negative values.

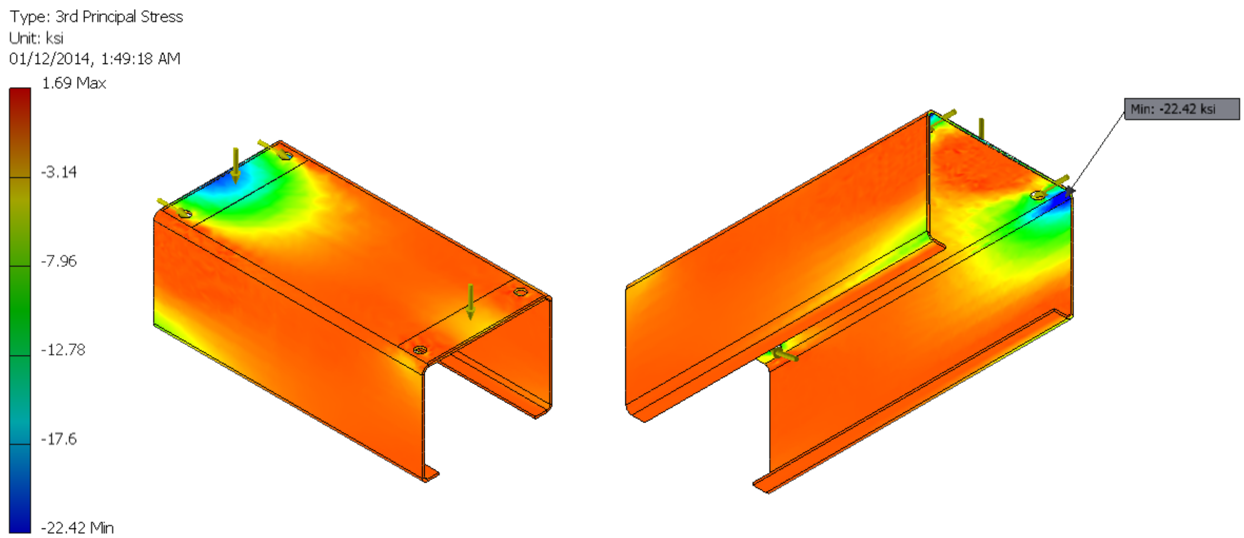


Figure 24: 3rd Principle Stress

We found the maximum compressive stresses to be much greater than the tensile and combined stresses as expected due to the combination of the weight of the brake and torque reaction of the motor. We obtained a value of -22.42 ksi, which is still below the stress criteria of 26.4 ksi set by our client.

Lastly we also verified that the shear stresses in our material also satisfied our clients criteria of 15.8 (36% of the yield point). Using inventor software we were also able to calculate the maximum values for the three shear stresses, τ_{xy}, τ, and τ_{yz}. We found that τ_{yz}had the highest value for shear stress, located at the bend of the side plate. The figure 24 is a schematic of the τ_{yz}shear stress distribution throughout the part.

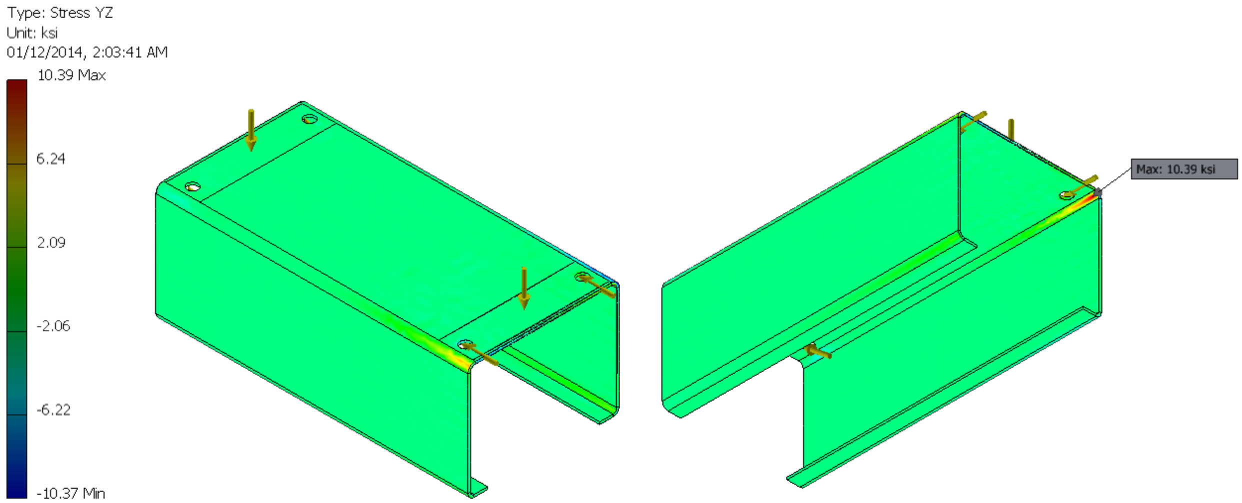


Figure 25: Maximum Shear Stress

Our maximum shear stress value was obtained to be -10.9 ksi, which is lower than our clients criteria of 15.8 ksi. Note that in the figure each view has its own coordinate system, and that the negative value is an indication of direction not magnitude.

In our second FEA we perfomed the same analysis but with the brake shoes. We found that in this design the stresses were very similar or less than in the previous model.

Provided in the following table are the maximum and minimum values for the various stresses in our FEA without the brake shoes.

TABLE 6: FEA MODEL 1 SUMMARY

Name	Minimum	Maximum
Volume	41.4114 in^3	
Mass	11.7442 lb mass	
Von Mises Stress	0.00310786 ksi	20.9057 ksi
1st Principal Stress	-3.20032 ksi	16.8825 ksi
3rd Principal Stress	-22.4087 ksi	1.68578 ksi
Stress XX	-9.69468 ksi	9.7011 ksi
Stress XY	-3.53316 ksi	4.56359 ksi
Stress XZ	-8.60966 ksi	8.37383 ksi
Stress YY	-20.214 ksi	16.867 ksi
Stress YZ	-10.3593 ksi	10.3803 ksi
Stress ZZ	-17.2746 ksi	15.8395 ksi

2.2.4 Finite Element Model 2

With the brake shoes incorporated in the analysis we found that the stresses were generally reduced or were approximately the same in our third design. Figure 25 shows FEA using Von Mises stress criterion.

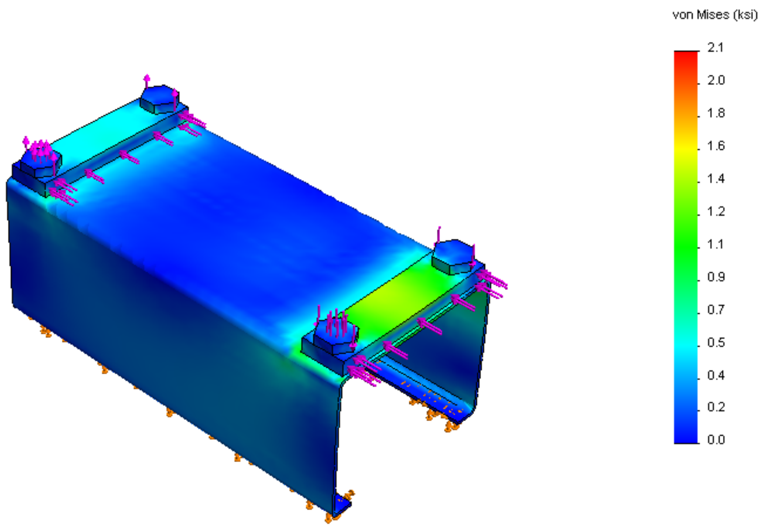


Figure 26: Von Mises 2

In the figure seen we can observe that the maximum stress was reduced from 21.8 ksi to 2.1 ksi. This is because when the bearing force was directly applied to the clearance holes it created stress concentrations at the bends of the mount. With the brake shoes attached those stress concentration were significantly decreased.

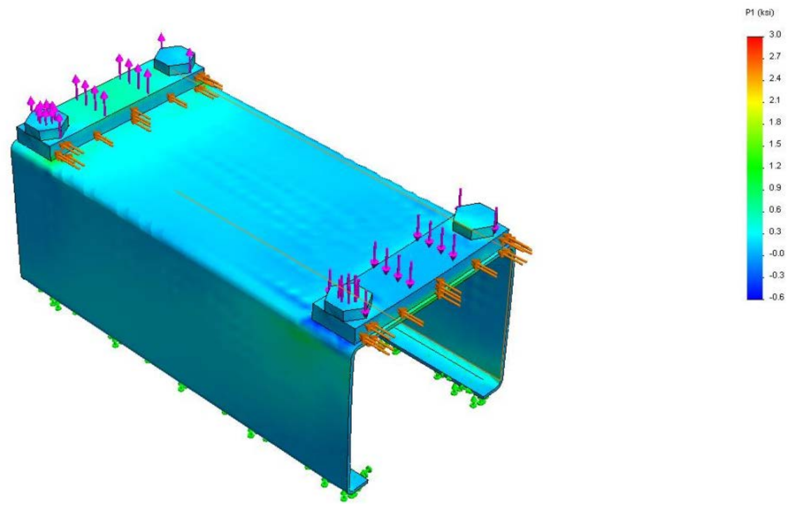


Figure 27: First principle Stress

As was the case for the Von Mises stress the first principle stress, in other words the maximum tensile stress was reduced from 16.96 ksi to 3.0 ksi with the addition of the brake shoes. The third principle stress obtained in the second FEA model had the most significant drop, as it went from 23.4 ksi to 1.6 ksi of compressive force.

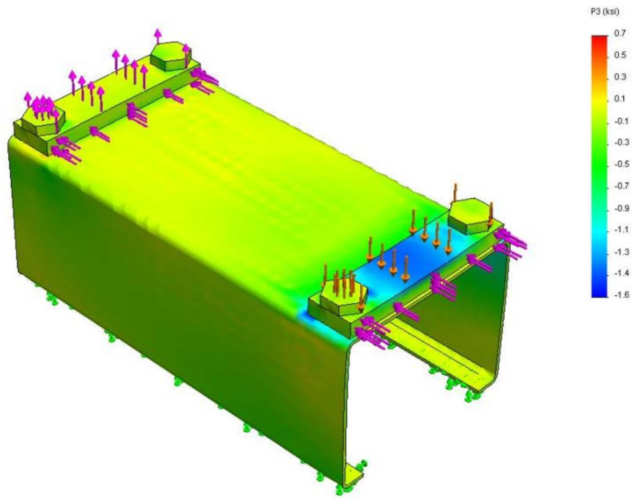


Figure 28: Third Principle Stress

Lastly the maximum shear stress the part was obtained to be -0.7 ksi with the brake shoes, as opposed to 10.9 ksi which was obtained without the brake shoes.

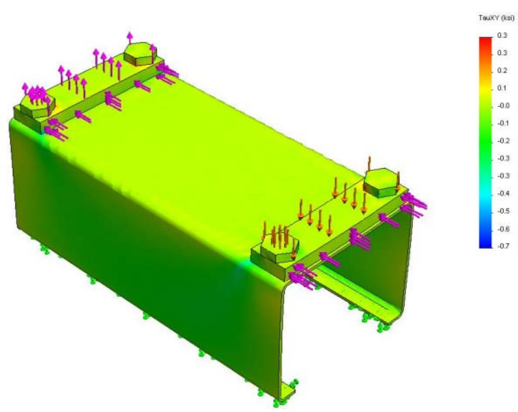


Figure 29: Shear in XY

To summarize all the values obtained in the second FEA are seen in Table 6 for comparative reasons. As mentioned previously the addition of the brake shoes significantly changes the results of our FEA because it provides the mount with more rigidity and also prevents stress concentration that was generated in the previous FEA model.

TABLE 7: FEA MODEL 2 SUMMARY

Name	Minimum	Maximum
Volume	41.5187 in^3	
Mass	11.7747 lb mass	
Von Mises Stress	0.0005 ksi	2.1399 ksi
1st Principal Stress	-0.6419 ksi	3.0255 ksi
3rd Principal Stress	-1.6444 ksi	0.7112ksi
Stress X	-0.9528 ksi	1.4662 ksi
Stress XY	-0.6844 ksi	0.3418 ksi
Stress XZ	-0.5473 ksi	0.4405 ksi
Stress Y	-1.5067 ksi	1.0668 ksi
Stress YZ	-0.5541 ksi	0.4989 ksi
Stress Z	-1.4429 ksi	2.9863 ksi

2.2.5 Combined Stress

The Combined stress is calculated as follows:

$$\sigma_{comb} = \sqrt{(\sigma_z{}^2 + \sigma_y{}^2 - \sigma_z\sigma_y + 3\left(\tau_{yz}\right)^2}$$

Using the values obtained in our FEA (listed in the tables above) the highest value of combined stress is calculated using the stresses in the y and z directions, along with the shear in yz.

$$\sigma_{comb} = 26.11\ ksi$$

Therefore our clients last criteria of having the combined stress less than or equal to 60% of the yield point of the material is also satisfied.

[redacted]

[redacted]

[redacted]

[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]

3 Recommendations

The three designs were compared to each other based on size, complexity, labour time, and reliability. In terms of size final design is the smallest, followed by design 3, leaving design 2 as the largest design. However, although final design 1 has the least amount of material than the other two designs, the material comes in 2X2 foot pieces of sheet metal; the same amount of material as final design 3 would have to be purchased. Final design 2 is made from 1/4 inch sheets of steel and has the highest material costs. For manufacturing costs final design 3 was the cheapest followed by final design 1, and lastly final design 2 being the most expensive. As for complexity final design 1 is composed of 2 separate pieces and may be difficult to install on the trolley. Final design 1 also has the most welds, along with forming of the sheets and will take the longest to make. Final design 2 is also fairly complex as it requires forming of the sheets and the welding of 2 pieces on those sheets with additional bending of the sheets. Final design 3 is the simplest to manufacture out of all 3 as it was composed of a single sheet of steel that is formed and requires no extra welds besides welding to the trolley platform. Therefore final design 3 has also the least amount of approximated labour time. As for the results of the FEA, all parts passed the clients criteria of stresses, and therefore satisfied the safety maintenance standards provided by the client. Therefore this is not considered in our recommendation. Considering all these factors our recommendation is final design 3. It is the simplest design to manufacture and install, and is lowest in cost. It fully satisfies the provided criteria, and will perform the functions required.

4 Conclusion

In this report, it explains the design methods, and the necessary analysis needed to develop a crane brake mounting block made from G40.21 44W steel. The report also covers our sponsoring company's background, the problem statement that was presented by the customer, and needs we identified in order to develop the optimum design for the customer.

There were 26 initial brake mount ideas but with careful screening and scoring, found in Appendix B, the mount ideas were summed down to three final designs. Based off specific calculations that were explained in each design section of this report, the thickness of each design would be made from the thinnest material available on the market. The minimal thickness of the steel was found to be 1/8".

The three designs that were selected were later compared with each other in order to recommend to the customer the best of the three designs. Since the first two concepts required a lot of welding it was concluded that these would be the weaker designs. The installation of the first two concepts would take longer than the third one as well. As a result these designs would cost more for installation due to labour time. On a final note it was observed that the total cost of the third part was less than the first two concepts. Based on these observations, it was concluded that the third design suggestion would be the optimum design for this project.

Bibliography

[1] Manitoba Hydro, "Pine Falls Generating Station," 2014. [Online]. Available: https://www.hydro.mb.ca/corporate/facilities/gs_pine_falls.shtml.

[redacted]

APPENDICES

1 Search Result and Concept Generation

This section describes the methods of research and idea generation in detail used by our team to develop design concepts for our project. Our team utilized the five step method to effectively develop concepts. This method includes clearly defining our project, which allowed our team to efficiently generate concepts that provide a viable solution to our problem. We also utilized effective external and internal search methods to develop design concepts based on the information collected from past experience, our client, textbooks, brainstorming, and many other forms of information. As we developed more concepts we classified them into categories according to their manufacturing process and type of structure. Classifying the concepts allowed our team to combine different ideas and improve concepts. Also, since our project requires our team to design a general and basic (although crucial) block to mount a brake on to, we did not encounter any patents that directly relate to our project or concepts we developed. Lastly, since our client did not want our team to see the design developed by the contractor, and since there is no competitive market for this type of product, we were not able implement benchmarking of competitor designs into our concept generation process.

2.2 External Search Results

Our primary source of external search results was our client. In order for our team to clearly define the project and generate concepts that provide an effective solution for the problem, we needed to clearly communicate with our client and obtain all the information about the project. During our meeting, our client provided us with various drawings, one of which was the layout of the trolley top platform and main hoist configuration, allowing for clear understanding of the space constraints and size limitations of our concepts. The layout drawing can be seen in Figure 2.

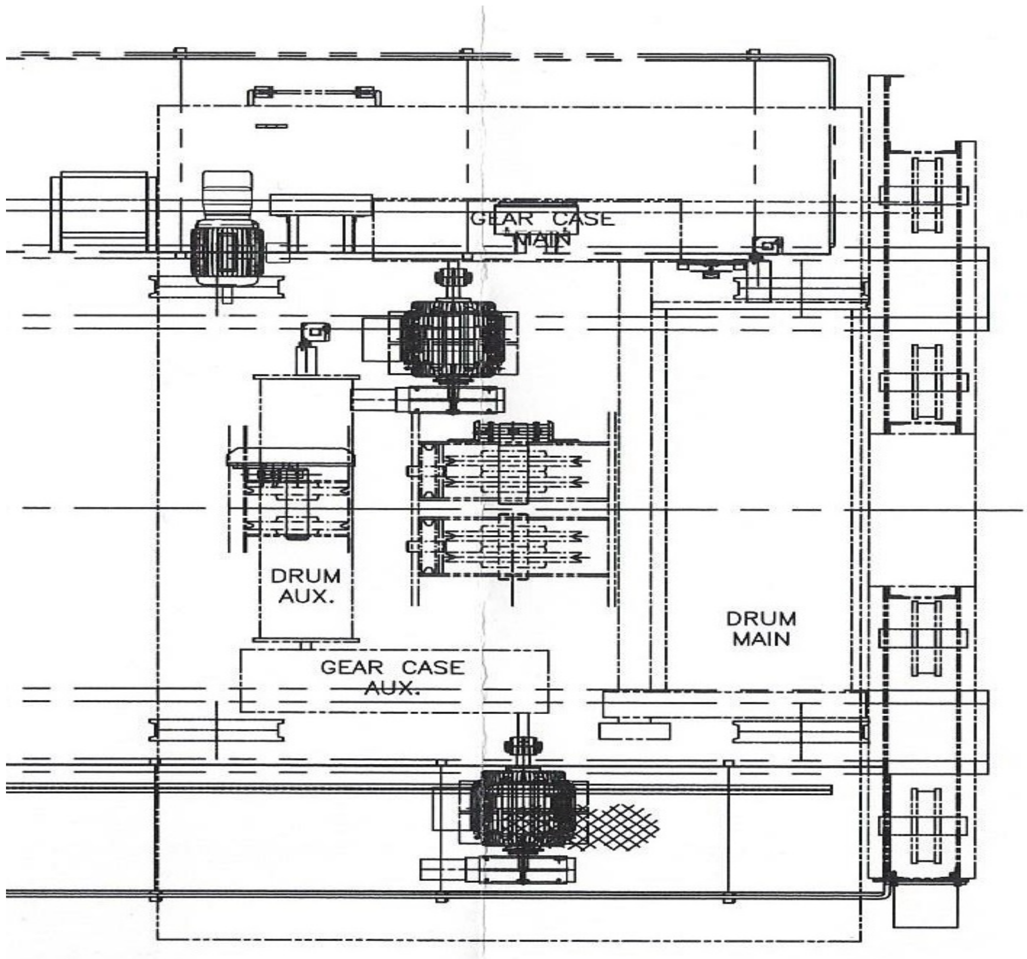


Figure 30: Power House Crane Layout. **[1]**

Another drawing provided to us by our client, is a detailed drawing of the main hoist motor and brake system. From this drawing we obtained the dimensional requirements of the block and the clearance holes for bolting the brake onto the block. The drawing also includes information obtained from the CMAA Safety and Maintenance standards; that directly applied to the mounting block and its requirements. We considered these standards when developing different types of structures for various conceptual designs. The detailed drawing of the main hoist motor and brake system is shown in Figure 3.

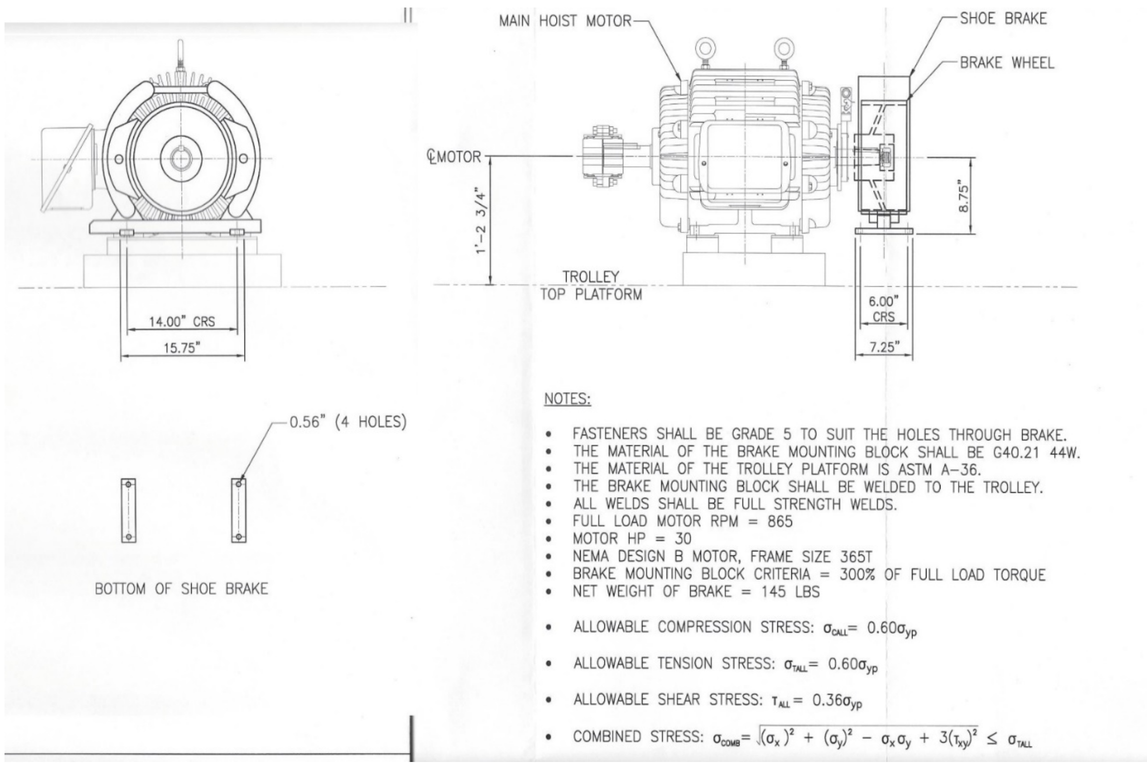


Figure 31: Detailed Drawing of Main Hoist Motor and Brake System.

After generating the list of needs from the information received from our client, we assigned metrics to each need and developed target specifications. These target specs were developed using various web resources, client’s performance expectations, and engineering knowledge we have acquired during our studies. The target specifications served as a catalyst in our concept generation process; it provided us a reference for the basic design requirements and resulted in efficient concept generation.

Our group also obtained useful information from various textbooks, which was considered in the development of our design concepts. [redacted].Knowledge of these processes aided in the development of concepts that were practical in terms of cost and time required to develop.

We also considered equations for average shear and compressive stresses when developing our conceptual designs. When developing our conceptual designs we kept in mind the different variables in the stress equations. To effectively bear average compressive stresses and general shear stresses we considered the size of cross-sectional where forces are applied; refer to

$$\sigma = \frac{P}{A} \text{ and} \tau_{ave} = \frac{P}{A}$$

Where,

σ = axial stress

τ_{ave}= average shear stress

P = applied load

A = cross-sectional area

Therefore the majority of concepts we developed have large areas where the forces are applied, to reduce the stresses in the structures.

We also considered shear stresses in beams, as many of the concepts we developed where beam type structures. In our beam designs we ensured that we had small values for moment of areas (Q), and large values for the moment of Inertia (I) and the length of the bases (t) to reduce the amount of shear stresses in the beams. The equation for shear stress in beams is:

$$\tau_{ave} = \frac{VQ}{It}$$

Where,

V = shearing force

Q = Moment of Area

I = Moment of Inertia

t = thickness in material perpendicular to force

The shear in our structures occurs in the horizontal plain as a result of the torque reactions transferred from the rotation of the shaft in the main hoist motor, to the brake and therefore the brake mount block itself. So the values of Q, I, and t were considered accordingly in the horizontal plain.

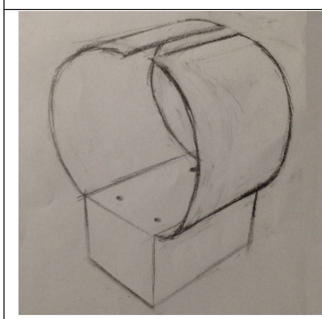
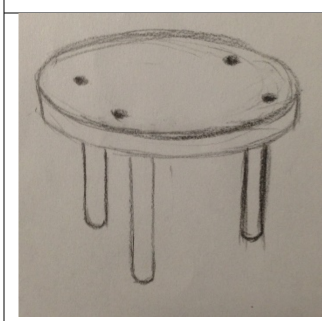
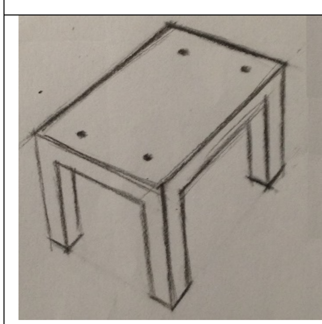
2.3 Internal Search Results

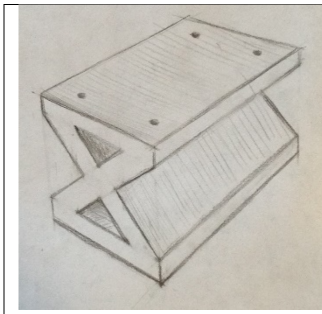
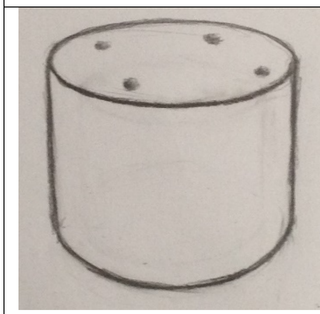
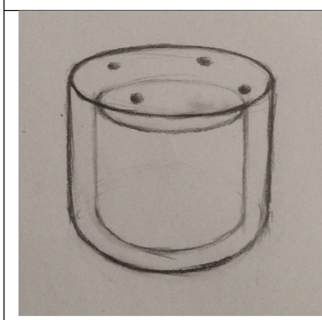
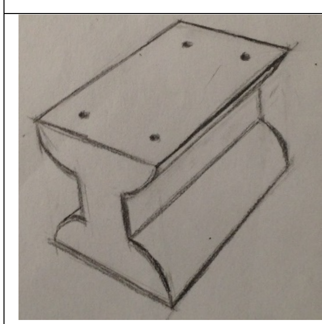
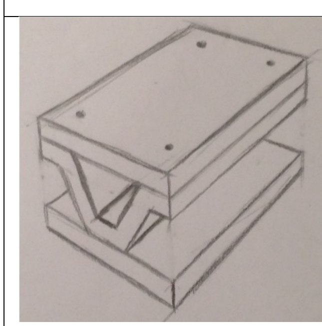
The information collected in our external searches provided a solid foundation for our team to start generating concepts and perform methods of internal searches. Our initial step in generating concepts as a team; was to first brainstorm and generate concepts individually. This allowed each individual member to be creative in their designs, and allowed our team to come up with as many concepts as possible. Generating concepts individually made it easier for other team members to welcome all ideas and suspend judgements.

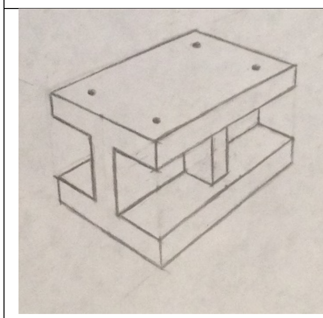
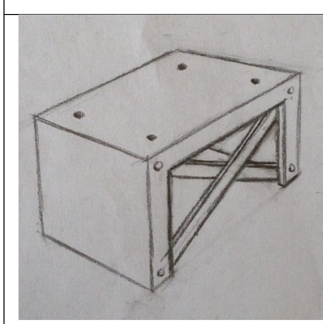
Using the Gallery Method, each team member presented their ideas to the group. The Gallery Method is done by creating quick sketches of each concept to effectively communicate ideas. The sketches provide visual aids that served as stimuli for combining ideas, developing new ideas, and ultimately developing new concepts. Along with the visual aids each member also provided a brief explanation for what their idea was and how they developed it. We found that clear communication of the ideas also promoted further development of the ideas by other members of the group. We also categorized the concepts, in terms of structural types and manufacturing processes. We had truss structures, beams, and sheet metal structures. For manufacturing processes we looked at expendable-mold castings, and laser cutting sheet metal followed by the welding of the sheets. By categorizing the ideas, we were able to combine ideas in the same categories, as well as mix ideas from different categories to develop new concepts. Overall we found that a mixture of individual methods for internal searches combined with group methods for internal searches, allowed our group to effectively generate conceptual ideas. The following section in our report provides a table containing the conceptual ideas that we screened and scored, as well as a brief description of the design and the intent of the designer.

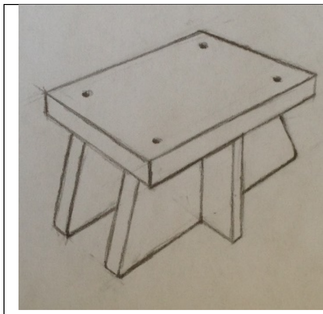
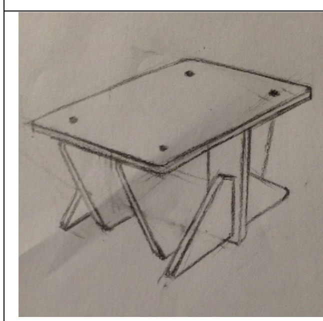
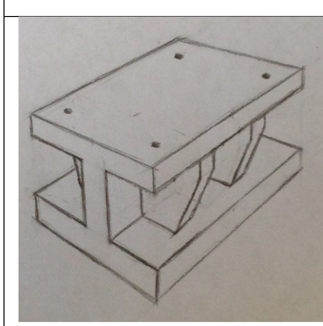
3 Design Concepts and Analysis

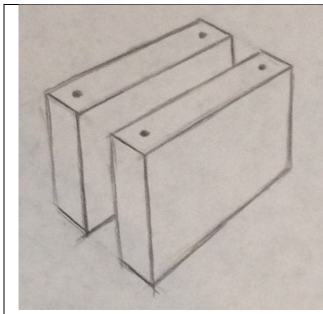
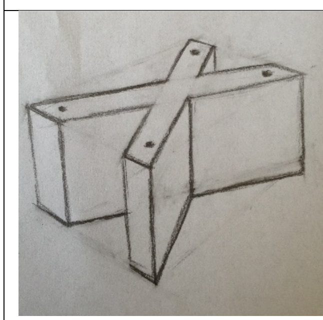
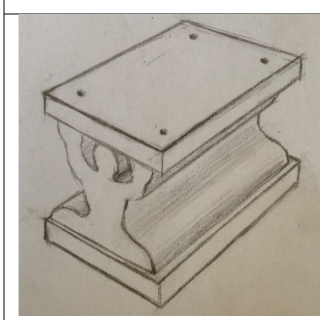
Each member in our group was tasked with generating a minimum of 4 concepts individually. As a group, we then generated new concepts by combining and modifying concepts developed individually and by generating totally new designs all together. Our team generated a total of 26 conceptual designs; Figure 4a to Figure 4z are the sketches of the 26 concepts generated during process.

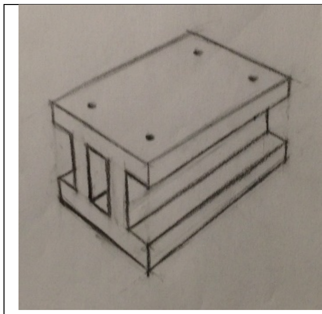
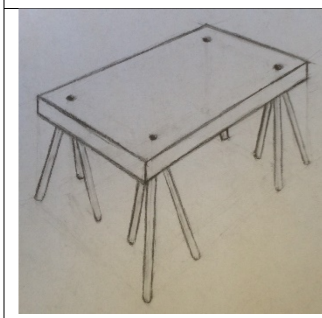
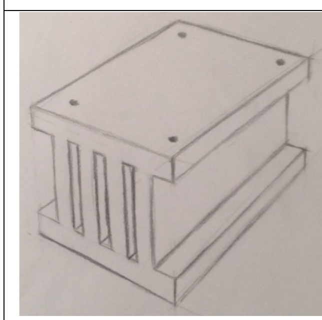
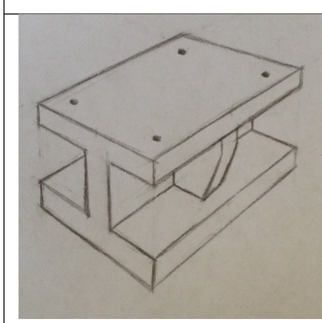
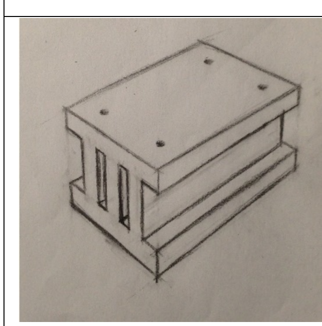
	(a) Cube-Dust Shield	Design A is called *Cube-Dust Shield*. It is a solid rectangular block with two pieces of metal wrapping around the brake wheel to protect it from dust.
	(b) Three-Leg Circular Top	Design B is called *Three-Leg Circular Top*. It has a circular piece of flat top, bolted underneath the brake wheel, with three supporting legs. The legs forms an equilateral triangle to support the weight and torque reaction, and still keep balance of the brake.
	(c) Four Leg Cubicle	Design C is called *Four Leg Cubicle*. It has a rectangular piece of metal with a flat top. The brake wheel is bolted on the top. The cubicle has four supporting legs at each corner to take the weight

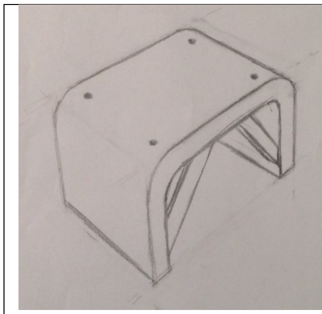
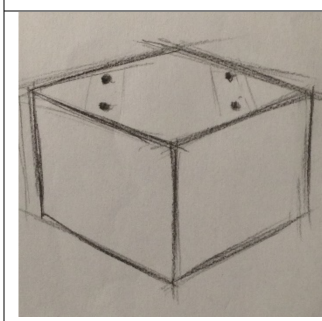
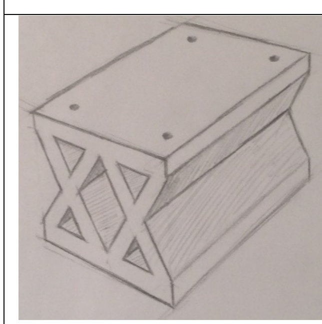
	(d) X-Shape Flat Top	Design D is a beam with a cross-section of "X" shape. The beam has two pieces metal with flat top and bottom. The design of a beam is based on the consideration of torque reaction which is the major load.
	(e) Solid Cylinder	Design E is a solid cylinder with a solid body (no holes). This design is simple, stable, hard to fail, and easy to manufacture.
	(f) Hollow Cylinder	Design F is a hollow cylinder. It is similar to Design E, but has a hollow circular cross-section, in order to remove unnecessary material from a solid cylinder
	(g) Hourglass Flat Top	Design G is a beam with a cross-section shaping like an hourglass, which has two semi-circles on the top and bottom with thin neck in the middle. It is revolved from an I-beam but reduced some of the unnecessary material
	(h) V-Shape Flat Top	This design was created from the idea of an I Beam. However since the conventional I beam has only one vertical component it is not designed for handling compressive forces. So we substituted a v shaped cross-section. This design has a relatively low manufacturing cost as it could be done by laser cutting sheets of metal, forming the metal (bending), and finished by welding the pieces together. The mounting holes would

		have to be machined as they are required to be threaded. The design if refined to the correct specifications, it will also effectively bear the torque reactions applied by the hoist's motor. A limitation in the design would be that there is limited welding area for large sections of sheet metal. This could lead to cracking at the weld joints.
	(i) I-Beam Support	This design is also built on the same principles of the previous concept. However instead of making a completely new cross section, we added supports to prevent failures such as buckling from compressive stresses. This design would also be made welding together laser cut sheets of metal; there is no bending or forming in the manufacturing process. The mounting holes on top would be clearance holes, and therefore would be made during the laser cutting of the top sheet. This has a relatively low cost of manufacturing and is easy to manufacture. Some limitations to this design may be due to the many edges and corners, many fillets at the edges and corners to prevent safety hazards. In this design there is also excessive material that may not contribute to the overall ability of the design to bear the forces applied.
	(j) Truss Flat Top	This design was created as a light and sturdy mounting block. This design includes laser cut sheet metal that is welded together, along with some bolted supports at the sides of the block. This concept benefits from being one of the lighter designs as it decreases the cost of material. Manufacturing wise the material is also very easy to make. A limitation of this design would be that additional hardware (bolts and nuts) would have to be included, so the cost may increase. Also the amount of area that could be welded to the trolley platform

	(k) Inverted Conical	This concept is designed to have a wider base than its top, forming a triangular shape which is better for bearing the compressive stresses and torque reactions. This concept is manufactured by welding together laser cut sheets; which is low in cost. The top portion consists of a single rectangular plate. The base has four triangular sheets welded to another rectangular sheet. The top piece is then welded to the base of the structure. The sheets in this design are meant to be thin so that it is as light as possible. Also the sheets are arranged in a way that will allow the block design to effectively bear the compressive force and the applied torque reactions. A limitation of this design would again be the amount of fillets and chamfers required to prevent any safety hazards. There is a lot of welding that must be done on this piece which would make it more difficult to analyze in FEA
	(l) Prism Sheet Metal	This design is similar to the previous but there are more pieces welded on the base. The triangular shapes welded onto the base will provide support to the rest of the block design, and provides a more stable foundation. It creates a wider base as well as provides support to the rectangular piece welded in the center of the design. Since this design has more pieces there is more welding to be done, as well as more laser cutting, thus increasing the cost of manufacturing. Each piece must also be machined so that all sharp edges and corners are filleted
	(m) I-Beam with Two Supports	This design is similar to “concept I” except with two supports. The added support will allow for the beam to more effectively bear the applied loads. The cost of the design will be slightly be more expensive as a result of the extra support, however will be much sturdier

	(n) Two Straight Block	The two straight blocks is a very simple design that is easy to manufacture and will be relatively low in cost of material. The cost of manufacturing will increase slightly since the holes must be threaded as well as have Having threaded holes on the design will make it easier for the whoever is change the brake in the system, to easily mount and dismount the brake. This design will also be very easy to analyze using FEA, as well as easier to optimize the dimensional values. Since this design is also simple it will be easier to modify and improve in the final design stage
	(o) Extruded X-Shape	Design (O): Extruded X-shape- This design is basically a solid block with a X marked on it and the rest of the part get extruded. It is a very strong design in term on the load it can handle just as much the solid block. Moreover, it is less material compared to solid block. It may be difficult to machine and more challenging to weld to the trolley during the initial installation. This design has more aesthetic compare to solid block design.
	(p) Muscle Man	Design (P): Muscle man design reflect the idea that the team is designing a strong block that will be subjected to some load and should be able to carry that load. It is an I-beam structure that is redesigned to have human features. The design has very strong base and will likely support the required load of the brake system. It is definitely has an appealing look and customers will just know the functionality of the design from the look of it. It maybe a challenge to machine the human looking shape for this design.

	(q) Dual I-Beam	Design (Q): Dual I-Beam-As stated in the name, this is an I-beam design with two connected I-beams. The reason for doubling the I-beam is to increase the load handling capacity of the design. The design should be able to carry more loads then a normal I-beam design. This design is more appealing compare to just an I-beam or solid block.
	(r) Tripod Flat Top	Design (R): Tripod Flat Top is a design with triple stand on each side of flattop. The idea was to minimize the material. It can handle a fairly good amount of load. However, the design may fail due to buckling due to the size of the stands.
	(s) Quad I-Beam	Design (S): Quad I-Beam- Is a I-beam with four I-shape, and it is intended to be much stronger and carry more load than the regular I-beam. The design is very stable but the down side is that it is complex to manufacture.
	(t) V Support I-Beam	Design (T): V Support I-Beam –Is basically and I-Beam with side extrusion. With the assumption that I-Beam can fail in the middle this part was added to avoid that failure in the middle. It is a simple design. It may be a little complex design but it depends on the choice of material and method of manufacturing.
	(u) Triple I-Beam	Design (U): Triple I-Beam – like design S & Q, it is just another I-beam with three (3) I-beam with same intention of increasing the capability of carrying more load. This design will carry a load as much as the regular block design. It maybe complex to manufacture and maybe more material compare to other designs.

	(v) Sheet Metal Table W/ Gussets	The sheet-Metal table with gussets is a layer of sheet metal bent into the shape of a table. The direction of each bend run in the direction of rotation of the motor. Two flat pieces of sheet metal is then connected between the each leg and the top part of the table shape. This is to support the legs and prevent them from buckling under the forces they are required to support. Welding the flat pieces to the table would be very difficult in the inside section near the corners of the bend. This would be because it is a tight area and it would be difficult for the welders to weld in there. This could create a point of weakness that would have the possibility of failing. Also this structure would be one of the lighter concepts because it is made from sheet metal.
	(w) Diamond Flat Top	The diamond flat top is a block shape design where the tips of the block run parallel with the direction of rotation of the motor. This design is very simplistic and manufacturing would be very fast. Unfortunately this design would be very heavy in comparison to the other designs. As a result to the extensive amount of material that would be required, the cost of this design would be much higher than most other concepts.
	(x) Vertical Dual X-Shape	The Vertical dual X-Shape consists of 4 criss crossing beams that intersect each other to create 2 X-shaped beams. The beams are supported by a flat bar on top and on bottom the X-shaped bars. The bars run in the direction of rotation of the motor. This design is very complex and would require a lot of welding to create it. The weld locations are very tight and it would be difficult to create good welds. As a result the weld spots could be weak and there is a greater chance of imperfections occurring. The complexity

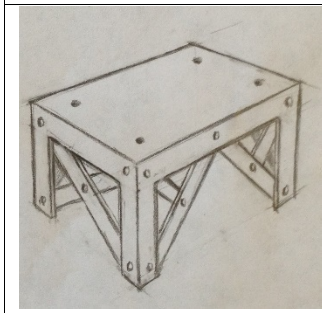
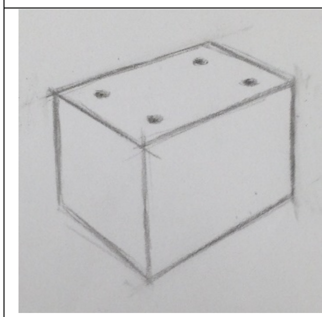
		of the design could make the design costly in the manufacturing aspect.
	(y) Saw Horse	The saw horse is a table shape design that is supported by links. All the components of the design is bolted together making this design more reliable since welding imperfections would not be a concern. Since the design consists of multiple pieces and would be very time consuming to build. The truss design is a very strong orientation and this design would be one of the stronger designs. As a result the structure will not require much material and this would save on material expenses.
	(z) Simple Solid Block	The simple solid block is a block made entirely out of steel. The flat sides run parallel to the direction of rotation of the brake motor. It was assumed that the stress would be concentrated at two of the corners rather than one of the corners like the diamond flat top. This design would be very heavy since it is composed of solid steel. It is very simplistic so manufacturing would be fast and cheap. Also the design would be very costly in material expense.

Figure 32: 26 Sketches of Concepts with Description.

3.1 Concept Analysis and Selection

This section describes our methods for screening and scoring the concepts we generated. In detail it will describe our selection and weighting of the scoring criteria, as well as the process of analyzing the sensitivity of the scoring criteria. Our screening and scoring was done in three phases. The first is the screening process, secondly is the initial scoring, and lastly is final scoring after performing sensitivity analysis and reweighting the scoring criteria.

3.1.1 Screening

The screening process was used by our team as a quick and efficient method for narrowing the amount of concepts to a reasonable size for scoring. It served as a preparation for a more detailed scoring process. Our team generated the following criteria to screen the concepts:

Cost - The cost of material and manufacturing for the concepts;

Safety - We considered safety of technician performing maintenance on the equipment;

Weight - One of the customer's requirements is that the design has to be light;

Manufacturability - The ease of manufacturing the design;

Durability - The design has to last for as long as possible;

Load Handling- The ability of the design to bear applied loads;

Aesthetics-This criterion is pretty much base on the look of the design;

Size- The design has to fit in the space provided, as well as maintain the accessibility of the brake system.

The screening of the concepts was done by choosing one concept as a reference, to which all other concepts are compared to using the criteria listed above. The chosen reference concept is concept Z (Simple Solid Block). Table III shows the results of the screening process and Table IV lists the all the concepts with their alphabetical representation.

TABLE VIII: SCREENING TABLE.

	Concept Variable																									
Selection Criteria	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z
Cost	-	-	+	-	0	+	-	-	-	-	+	+	+	+	-	-	+	-	-	-	-	+	-	-	-	0
Safety	+	-	-	-	0	-	0	-	-	-	-	+	0	+	-	-	+	-	0	-	-	-	-	0	-	0
Weight	-	+	+	+	-	+	+	+	+	+	+	+	+	+	+	-	+	+	0	+	+	+	-	+	+	0
Manufacturability	-	-	0	-	0	-	-	-	-	-	-	-	-	0	-	-	-	-	-	-	-	+	0	-	-	0
Durability	0	-	-	0	0	-	0	-	-	-	-	+	0	-	-	-	0	-	0	-	0	-	0	0	-	0
Load Handling	0	-	-	0	+	-	-	-	0	0	0	0	0	0	0	-	0	-	0	-	0	-	0	0	-	0
Aesthetics	+	+	+	+	0	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	0	+	+	0
Size	-	+	+	+	-	+	+	+	+	+	+	+	+	+	+	+	+	+	0	+	+	+	-	+	+	0
Maitenance	-	-	-	0	0	0	0	-	0	-	0	0	-	0	-	-	-	-	0	-	0	0	0	-	-	0
Pluses	2	3	4	3	1	4	3	3	3	3	4	6	4	5	3	2	5	3	1	3	3	5	0	3	3	0
Zeros	2	0	1	3	6	1	3	0	2	1	2	2	3	3	1	0	2	0	6	0	3	1	5	3	0	0
Minuses	5	6	4	3	2	4	3	6	4	5	3	1	2	1	5	7	2	6	2	6	3	3	4	3	6	0
Net	-3	-3	0	0	-1	0	0	-3	-1	-2	1	5	2	4	-2	-5	3	-3	-1	-3	0	2	-4	0	-3	0
Rank	19	19	7	7	14	7	7	19	14	17	6	1	4	2	17	26	3	19	14	19	7	4	25	7	19	7
Continue?	no	no	yes	yes	no	yes	yes	no	no	no	yes	yes	yes	yes	no	no	yes	no	no	no	yes	yes	no	yes	no	yes

TABLE IX: CONCEPTS WITH ALPHABETICAL REPRESENTATION.

Concepts	
A	Cube-Dust Shield
B	Three-Leg Circular Top
C	Four Leg Cubicle
D	X-Shape Flat Top
E	Solid Cylinder
F	Hollow Cylinder
G	Hourglass Flat Top
H	V-Shape Flat Top
I	I-Beam Support
J	Truss Flat Top
K	Inverted Conical
L	Prism Sheet Metal
M	I-Beam with Two Supports
N	Two Straight Block
O	Extruded X-Shape
P	Muscle Man
Q	Dual I-Beam
R	Tripod Flat Top
S	Quad I-Beam
T	V Support I-Beam
U	Triple I-Beam
V	Sheet Metal Table W/ Gussets
W	Diamond Flat Top
X	Vertical Dual X-Shape
Y	Saw Horse
Z	Simple Solid Block

Through the screening process, the amount of concepts was reduced from twenty six to thirteen concept designs that were left to be scored. The scoring matrix will be explained in the next section to further narrow down the designs to the more satisfactory design concepts.

3.1.2 Scoring

In the scoring section the team compared each criteria against each other to determine the value of the criteria. A matrix was created in order to do this comparison; Table V below shows the weight percentage matrix of the criteria. According to the table it shows that the selected factor aesthetics had no importance; as a result this criterion was removed from the list.

TABLE X: WEIGHTING OF CRITERIA.

Criteria	A. Cost	B. Safety	C. Weight	D. Manufacturability	E. Durability	F. Load Handling	G. Aesthetics	H. Size	I. Maintenance
A. Cost		B	A	D	E	F	A	A	I
B. Safety			B	B	E	F	B	B	B
C. Weight				D	E	F	C	H	I
D. Manufacturability					E	F	D	D	D
E. Durability						F	E	E	E
F. Load Handling							F	F	F
G. Asethetics								H	I
H. Size									H
I. Maintenance									
	A	B	C	D	E	F	G	H	I
Score	3	6	1	5	7	8	0	3	3
Weight (%)	8.33	16.7	2.778	13.9	19.4	22.2	0	8.33	8.33

After screening out the less preferred concept designs, the remaining 13 designs were scored in each criterion. Figure 5 illustrates the ranking of all 13 design concepts and the 6 designs that were selected as a result are highlighted in blue.

		Concept											
Selection Criteria	Weight %	Four-Legged Cubicle		X-Shape Flat Top		Hollow Cylinder		Hourglass		Inverted Conical		Prism Sheet Metal	
		rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight
Cost	8.33	3	0.2499	2	0.1666	4	0.3332	2	0.1666	5	0.4165	4	0.3332
Safety	16.67	1	0.1667	3	0.5001	2	0.3334	4	0.6668	2	0.3334	3	0.5001
Weight	2.78	3	0.0834	4	0.1112	4	0.1112	2	0.0556	4	0.1112	3	0.0834
Manufacturability	13.89	3	0.4167	4	0.5556	3	0.4167	2	0.2778	3	0.4167	3	0.4167
Durability	19.44	2	0.3888	2	0.3888	1	0.1944	4	0.7776	3	0.5832	4	0.7776
Load Handling	22.22	1	0.2222	3	0.6666	2	0.4444	4	0.8888	3	0.6666	4	0.8888
Size	8.33	4	0.3332	4	0.3332	1	0.0833	3	0.2499	4	0.3332	3	0.2499
Maintenance	8.33	3	0.2499	2	0.1666	2	0.1666	5	0.4165	2	0.1666	2	0.1666
Score		2.1108		2.8887		2.0832		3.4996		3.0274		3.4163	
Rank		11		9		12		2		8		3	
Continue?		No		No		No		Develop		No		Develop	

		Concept													
Selection Criteria	Weight %	I-Beam 2 Supports		2 Blocks		Dual I-Beam		Triple I-Beam		Sheet Metal w/ Gussets		Vertical Dual X-Shape		Simple Block	
		Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight	Rating	Weight
Cost	8.33	3	0.2499	4	0.3332	4	0.3332	2	0.1666	4	0.3332	2	0.1666	1	0.0833
Safety	16.67	4	0.6668	4	0.6668	3	0.5001	3	0.5001	3	0.5001	1	0.1667	5	0.8335
Weight	2.78	3	0.0834	2	0.0556	4	0.1112	3	0.0834	5	0.139	2	0.0556	1	0.0278
Manufacturability	13.89	2	0.2778	5	0.6945	4	0.5556	2	0.2778	3	0.4167	2	0.2778	5	0.6945
Durability	19.44	4	0.7776	4	0.7776	3	0.5832	4	0.7776	3	0.5832	3	0.5832	5	0.972
Load Handling	22.22	4	0.8888	4	0.8888	3	0.6666	4	0.8888	3	0.6666	4	0.8888	5	1.111
Size	8.33	2	0.1666	3	0.2499	3	0.2499	3	0.2499	4	0.3332	2	0.1666	1	0.0833
Maintenance	8.33	2	0.1666	4	0.3332	3	0.2499	3	0.2499	2	0.1666	3	0.2499	5	0.4165
Score		3.2775		3.9996		3.2497		3.1941		3.1386		2.5552		4.2219	
Rank		4		1		5		6		7		10		NA	
Continue?		Develop		Develop		Develop		Develop				No		No	

TABLE XI: NEW WEIGHTING OF CRITERIA.

Criteria	A. Cost	B. Safety	C. Manufacturability	D. Durability	E. Load Handling	F. Aesthetics	G. Size	H. Maintenance
A. Cost		A	A	D	E	A	A	A
B. Safety			C	D	E	B	B	H
C. Manufacturability				D	E	C	C	C
D. Durability					E	D	D	D
E. Load Handling						E	E	E
F. Aesthetics							G	H
G. Size								H
H. Maintenance								
	A	B	C	D	E	F	G	H
Score	5	2	4	6	7	0	1	3
Weight	17.86%	7.14%	14.29%	21.43%	25%	0	3.57%	10.71%

TABLE XII: CONCEPT RANKING 2.

Selection Criteria	Weight%	Hourglass		Prism Sheet Metal		I-Beam 2 Support	
		Rating	Weight	Rating	Weight	Rating	Weight
Cost	17.86	2	0.3572	4	0.7144	3	0.5358
Safety	7.14	4	0.2856	3	0.2142	4	0.2856
Manufacturability	14.29	2	0.2858	3	0.4287	2	0.2858
Durability	21.43	4	0.8572	4	0.8572	4	0.8572
Load Handling	25	4	1	4	1	4	1
Size	3.57	3	0.1071	3	0.1071	2	0.0714
Maintenance	10.71	5	0.5355	2	0.2142	2	0.2142
Score			3.4284		3.5358		3.25
Rank			3		2		4
Continue?		Develop		Develop		No	
Selection Criteria	Weight%	2 Blocks		Dual I-Beam		Tripple I-Beam	
		Rating	Weight	Rating	Weight	Rating	Weight
Cost	17.86	4	0.7144	4	0.7144	2	0.3572
Safety	7.14	5	0.357	3	0.2142	3	0.2142
Manufacturability	14.29	5	0.7145	4	0.5716	2	0.2858
Durability	21.43	4	0.8572	3	0.6429	4	0.8572
Load Handling	25	4	1	3	0.75	4	1
Size	3.57	3	0.1071	3	0.1071	3	0.1071
Maintenance	10.71	4	0.4284	3	0.3213	3	0.3213
Score			4.1786		3.3215		3.1428
Rank			1		5		6
Continue?		Develop		No		No	

Following the initial scoring, our team performed sensitivity analysis on the scoring criteria. In our sensitivity analysis the most drastic change in the weighting was the safety criteria. Initially being 16.67%, the weight of the safety was changed to be 7.14%. Reason being is since the six designs that were selected in the initial scoring process had relatively high ratings for safety; therefore all the designs being scored in the second are assumed to be safe. So we decided to reweight the criteria with that in mind. It is important to note that we also wanted to maintain the weight of the criteria that were crucial to the performance of the block. Therefore the two criteria that were increased the most in our second time weighting was the manufacturability and the cost. Table VI shows the new weight percentage of the criteria and Table VII shows the second round of scoring.

3.1.3 Recommendation of Concepts

The three final design concepts that were chosen during the second round of scoring are the hour glass beam, the prism sheet metal structure, and the two blocks design. We ensured that all of these are designed to excel in the durability, load handling, safety, and maintenance criterion. These designs were also estimated to be the cheapest and easiest to manufacture. These three designs will be the concepts we will be working with and refining during our final stage of our design process. We will also be performing detailed analysis using FEA and CAD software to optimize the performance of these concepts.

4 Design Refinement and Analysis

The selected three designs were carefully analysed using Inventor and SolidWorks. The design was then refined to design the optimum design that met the design limitations.

4.1 Design 1

Design 1 is two separate blocks that support the braking component. Figure 2 displays the initial design where two solid blocks are positioned directly below the brake mount.

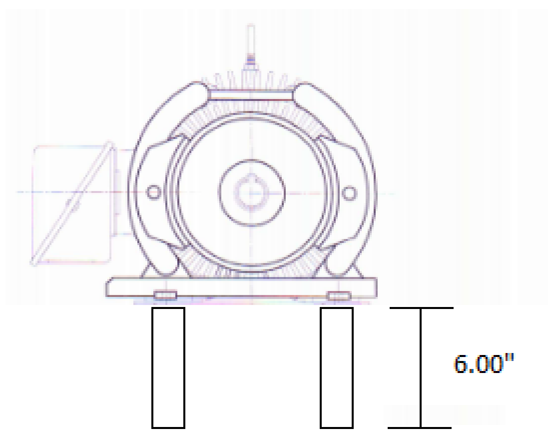


Figure 33: Initial design of the two block concept

Calculations were performed in order to select the best thickness of the design. The weight of the brake mount was given by the client which was 145 lbs. With static forced each block would hold support halve the weight of the brake mount which would be 72.5 lbs. The following stress equation would calculate the minimum area of the brake mount design.

$$\sigma = \frac{F}{A}$$

Since the allowable compression stress is equal to 60% the allowable yield stress that was explained by the client, which is 44 ksi, the minimum area of the brake mount was found to be 0.0016 in^2. By assuming the brake mount would be the same thickness of the bottom of the brake which is 7.25" the minimal thickness of the brake mount was found to be $5.756X10^{-6}$ in. This value is very small so it was determined that the minimum thickness of the specified steel would be able to support the design. The next concept that was considered was buckling. The minimum moment of inertia solved by using the following equation that would determine the minimum thickness of the brake design.

$$F = n\pi^2 \frac{EI}{L^2}$$

In the previous equation, n is equal to 1, the modulus of elasticity of the steel is 200,000 MPA, and the length, L, is equal to 6 ". By calculation the previous equation with the known vertical force, the moment of inertia was found to be $9.1202X10^{-6}$ in^4. This value was used to calculate the minimal thickness of the brake mount which was found to be $2.64664x10^{-7}$ m. This thickness is very small as well as in the previous calculation. The final calculation performed was regarding the torque that would be applied to the brake mount. This would produce a shearing stress on the mount which was determined to be 36% of the maximum yield stress. This information was also provided by the customer. As a result the maximum shear stress was found to be 15.84 ksi. The maximum torque that would be applied to the mount was found to be 546.24 ft lb. By using the previous stress equation the minimum area of the mount was found to be $2.368x10^{-2}in^2$. Since the width of the brake mount was assumed to be 7.25" the minimum thickness of the mount would need to be $3.262x10^{-3}$in.

As a conclusion from the previous calculations the minimum thickness of the mount would be 3.262×10^{-3}in. Since this value is very small the brake mount was designed as a solid molded piece of steel consisting of a thickness of 1/8". The design was changed from a solid block design to a triangular design in order to compensate for the torque reactions caused by the brake. This design was designed and tested using FEA on SolidWorks. Figure 3 illustrates the initial design.

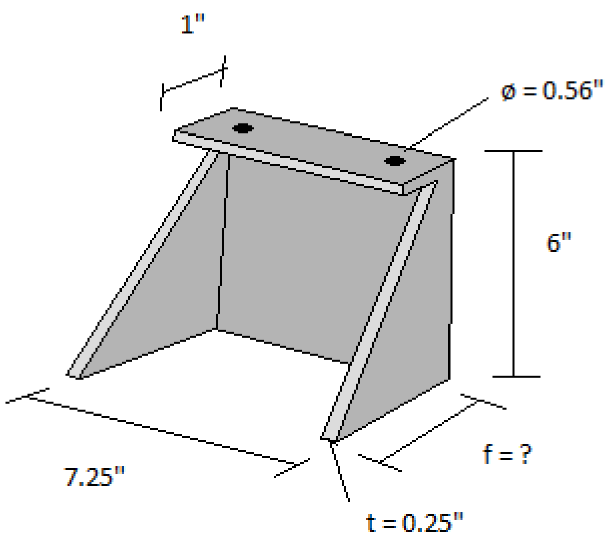


Figure 34: Second design concept

Some dimensions of the block were known but the dimension of *f* was unknown. It was decided that this dimension would be determined through FEA on SolidWorks. The dimension will start at 4" and through FEA this value could either increase or decrease. The orientation of the two blocks is shown in Figure 4.

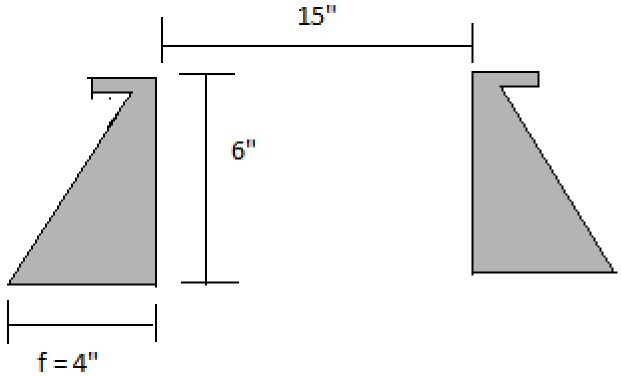


Figure 35: Brake mount block orientation

Design 2 Refinement and Analysis

One of the final concepts selected is Hourglass beam, shown in Figure 35. It is 16 inches long and 8.3 inches wide, and 6 inches tall. The center support is 1 inch in thickness. Its weight is 75.36 lbs. Finite element analysis is applied to the model, and the maximum von Mises Stress is 0.9 ksi, and the maximum displacement is 0.0185 inch.

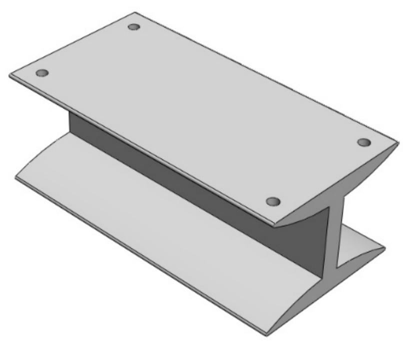


Figure 36: design model

The brake wheel is attached to the mounting block by bolts and nuts. Nuts require a flat surface at contact. Therefore, the hourglass concept is replaced by an I-beam. The model of the I-beam is shown in figure 36

. The beam is 16 inches long, 8 inches wide and 6 inches tall. The top and bottom plats are 0.25 inch thick. The center support is 0.5 inch in thickness. The weight of the entire beam is 30.6 lbs.

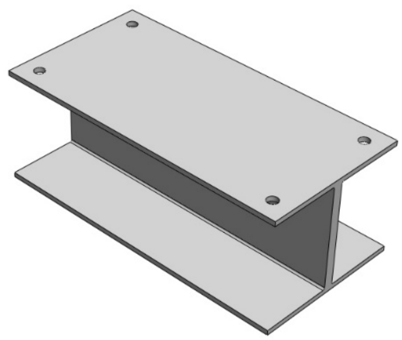


Figure 37: design model

Finite element analysis is applied on this beam. Four bolts are attached on the top plate of the beam. In this analysis, loads are applied on the bolts for simulation. There are four horizontal forces, 114.25 lbs each, applied on the bolts; two 127.67 lbs forces pressing down vertically on two bolts; and two 127.67 lbs forces pulling up vertically on the other two bolts, shown in Figure 37.

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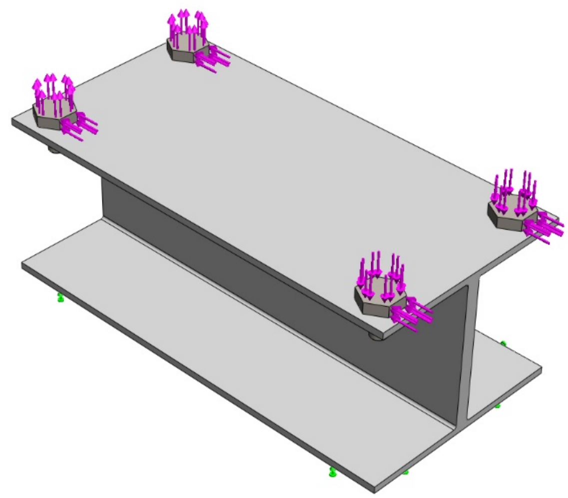

Figure 38: I beam

Figures below present the results from the simulation. The maximum von Mises stress is 7.89 ksi and the maximum displacement yields at 0.77 in. The corners of the top flange experience the most deflection, caused by their large distance from the vertical support. Optimization is needed for this design.

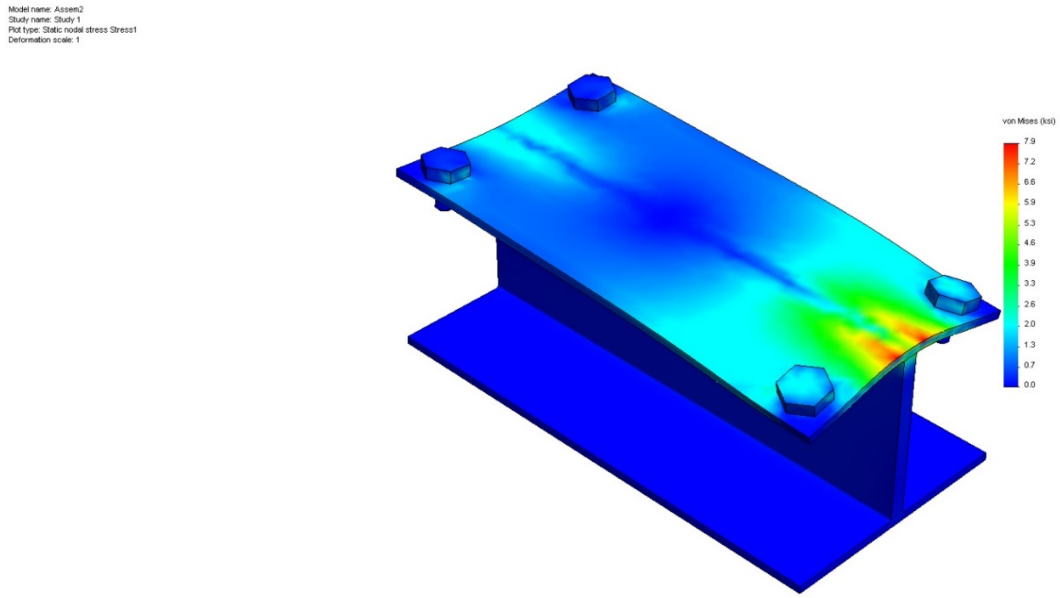

Figure 39: von Mises

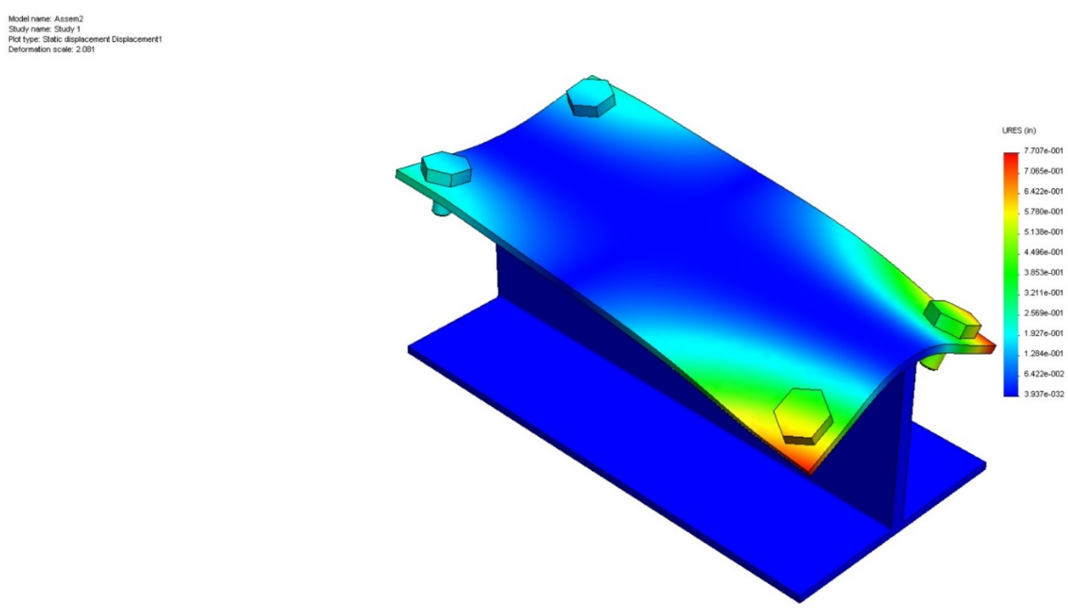

Figure 40: displacement

The center vertical support is then split into two components, cutting the I-beam into two C-shape beams. Each beam is 16 inches long and 6 inches tall. The beam is bent from a sheet metal with 1/8 in thickness. The top and bottom flanges are 2 inches wide, shown in Figure 39

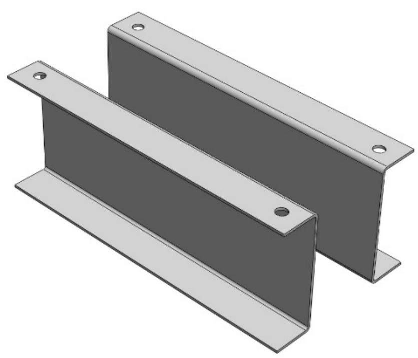

Figure 41: design model

Finite element analysis is applied on this model. Same bolts attachment and loads are applied. Shown in Figure 40. Due to excessive displacements calculated in this model, SolidWorks fails to simulate. Further optimization is needed for this design.

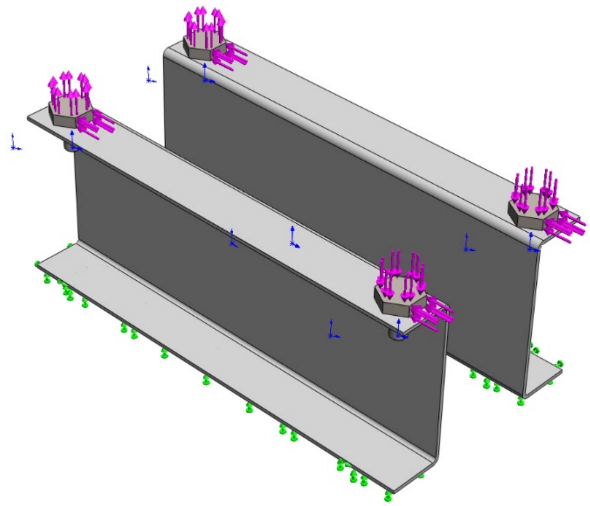

Figure 42: load application

To prevent excessive displacement, a sheet metal with ¼ inch thickness will be used and on the top of the beams, two bridges made of metal strips will connect the two beams. Each strip is 2 inches wide, 8 inches long, and ¼ inch thick, see Figure 41. A final design is reached.

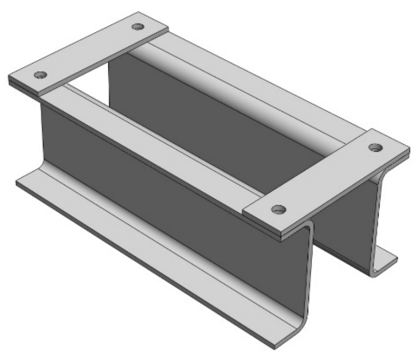

Figure 43: final model

5 Final Design 3

This design was derived from concept k; We found that in our initial FEA that there were many components in the design that did not effect the mounts ability to bear load. A figure of the FEA analysis on the original design is shown below.

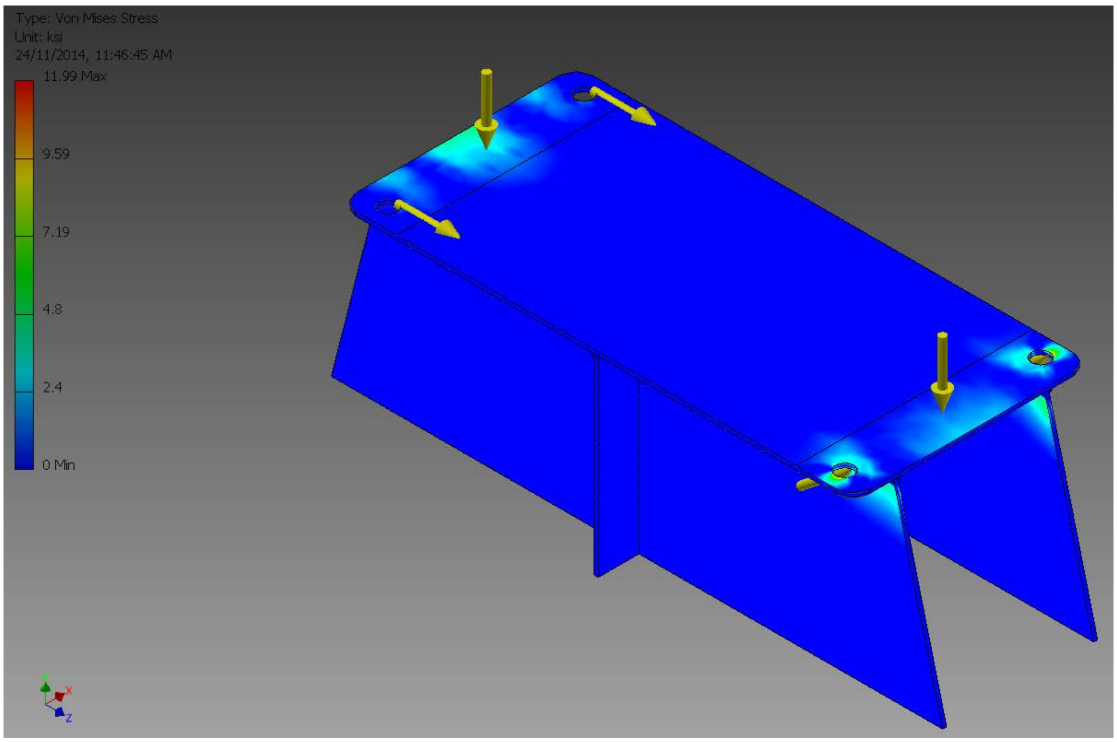

Figure 44: initial design FEA

As seen in the figure the majority of the stresses is concentrated around the clearance holes for the bolts. You can also see that there is vertually no stress in the center of the mount. Following the initial FEA performed on the part, the block was redesigned as shown in figure 43. As for modifications we altered the design by removing the center plate which bore no stress. We also altered the side plates by using two longer sheets as opposed to 4 sheets. We also made flanges so that there would be another layer of sheet metal at the clearance holes and that there would be more weldable area on the bottom of the mount.

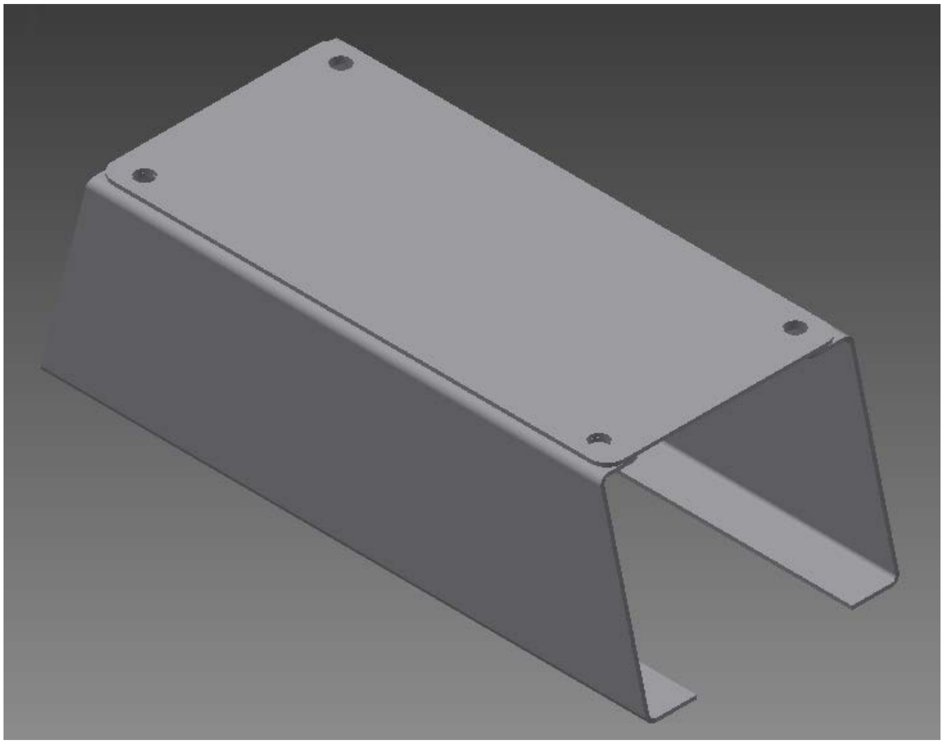

Figure 45: Modified Part 1

We verified that the design would perform to the specifications required by our customer by using detailed fea analysis. In our FEA we used a sheet metal thickness of 0.125 inches as we found during our research that it was the thinnest available for G40.21 44W strutural steel. Since we found that it was more than sufficient (as you will see later in this section) we made no further alterations to the design.

The figure below illustrates how the forces were applied and the Von Mises Stress distributions of the part in the second iteration of FEA.

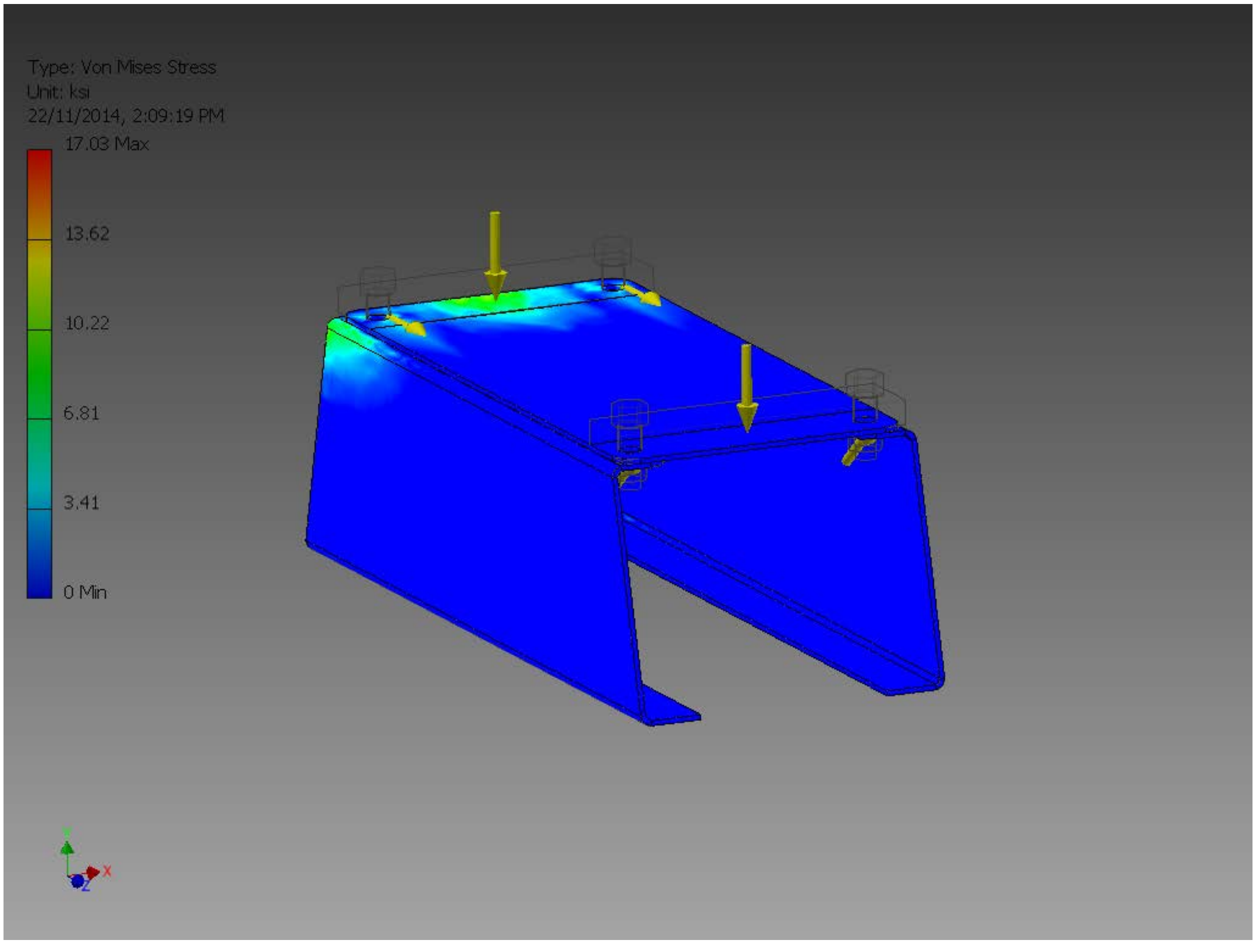

Figure 46: 2nd FEA

As seen in the figure above, using Von Mises Stress criterion, we verified that the part satisfies the criteria provided by our customer which is that the maximum combined stress must be less than or equal to 26.4 ksi (60% of the yield point of the material). Although it satisfied the criteria we found that we could improve the design, by reducing the cost of manufacturing. So, to impove the design we made it into one piece of sheet metal with 4 bends. By doing this we reduce the total amount of length in inches to laser cut, along with eliminating the welding required in the part. We found that this significantly reduced the cost. A figure 45 shows an isometric view of our third final concept.

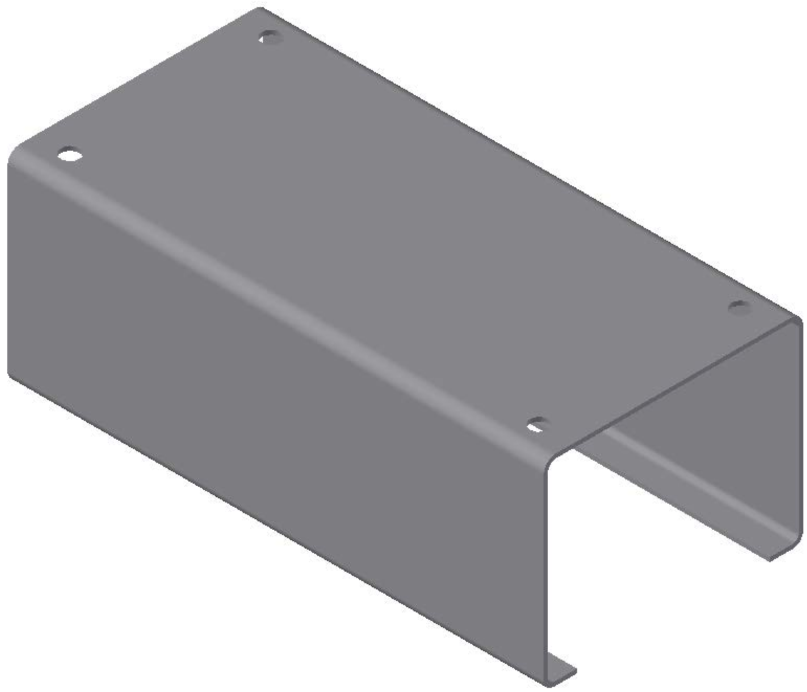

Figure 47: Final Design 3