

# **The Big Cherry**

## **Design and Fabrication of a Heavy Duty Engine Hoist**

**By:**

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**Report Submitted as Part of the Course:**

ABEN 487: Senior Design Project II

**In Cooperation With:**

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**Fargo, North Dakota**

**May 2009**

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May 8, 2009

Mr. Elton G. Solseng and Mr. James A. Moos  
Department of Agricultural and Biosystems Engineering  
North Dakota State University  
NDSU Dept. 7620  
PO Box 6050  
Fargo ND 58108-6050

Dear Mr. Solseng and Mr. Moos:

The project to design a heavy duty engine hoist has come to completion. This is the final report for the project. Within this report, you will find descriptions of the final design and the methods used to come to completion. There are also a complete set of drawings and calculations used to assist in the design.

The designed engine hoist, nicknamed 'The Big Cherry', is designed to meet all the required objectives brought forward in the fall semester of 2008. It is recommended that the prototype be used within the designed (4000 lb) limit and on even ground. This engine hoist is a very heavy piece of equipment and should be operated with care.

This submitted report is a sign of our completion of the project. This letter formally gives you the sole right to the design, all drawings, and calculations. If a prototype is built, we recommend testing its structural integrity before operation. We thank you for giving us the opportunity to design this piece of equipment. It was a great honor working with you.

Sincerely,

Ryan Nagel  
Team Leader

## **EXECUTIVE SUMMARY**

There was a need for a heavy lifting device in the Agricultural and Biosystems Engineering Department at North Dakota State University. The project undertaking was to design a heavy duty version of an engine hoist, resembling a cherry picker. This project was chosen because of its interesting intentions.

The objectives of this project were as follows:

- 2 ton (4000 pound) capacity at full extension length of 8 feet
- Maximum lift height of 9 feet at full extension
- Wide casters for travel over uneven terrain
- Hydraulic cylinder system of sufficient capacity
- Collapsible for doorway mobility and storage

The new design was broken into four main sections: the upperstructure, lowerstructure, hydraulic cylinder, and caster wheels. The upperstructure is the attachment point for the object being lifted. It is the main lifting beam. The lowerstructure serves as the base for the piece of equipment. A hydraulic cylinder is used to lift the load. Attached to the lowerstructure are the caster wheels which allow transporting.

Extensive finite element analysis (FEA) was conducted on the upperstructure and lowerstructure. This helped to ensure a design that is structurally sound, but not bulky. Several different load cases were applied to the upperstructure and lowerstructure including a standard 4000 pound vertical load, a simulated forward load, and a side load. The upper and lowerstructures' structural integrity is up to par. It meets the goal of 1.5 to 2.0 safety factor.

The final prototype is to be constructed of tubing and sheet metal. The upperstructure is composed of a 5 inch by 3 inch by 0.5 inch tube with a 4 inch by 2 inch by 0.5 inch extendable arm. There are several 0.25 inch thick gussets used for support and for the attachment point for the cylinder. The lowerstructure is composed of 2 inch by 3 inch by 0.3125 inch thick tubing, 2 inch by 3 inch by 0.25 inch thick tubing, 1.5 inch by 2.5 inch by 0.25 inch thick tubing, 0.25 inch thick sheet metal and 0.375 inch thick sheet metal.

The hydraulic power unit has two options because prototyping did not occur. First, a hand pump powered hydraulic long ram jack of 8 ton (16000 pound) capacity is one option. The other option is to do a complete system using an electrically powered hydraulic pump, directional control valve, flow control, pressure relief valve, and a cylinder.

Since the prototype has not been constructed, testing before operation is recommended. Several regular cyclic loads and a few irregular loadings should be done to insure structural integrity before full operation. The cost is estimated at \$2,475. A prototype should be built, tested, and used as desired by the operator as long as it is in the recommended load range of 4000 pounds at full extension.

## **I. GENERAL INTRODUCTION**

### **1.1 Overview**

There are many lifting devices used in industry. They can range from cherry pickers, overhead hoists, to skid loaders and forklifts. Skid loaders and forklifts utilize engine power to lift the objects while cherry pickers and overhead

hoists utilize simple mechanics to lift heavy objects. A cherry picker design consists of a base with a hydraulic cylinder attached to the lifting arm. The cylinder extends and retracts, lifting and lowering the heavy objects.

### **1.2 Rationale**

The need for these heavy lifting devices is to lift heavy objects which may not be able to be lifted with man power alone. They also allow one man to simply lift and maneuver heavy objects instead of multiple people being needed. Simple machines over powered machinery can be beneficial in the cost of application and the usability indoors. They can also be manufactured so they are physically smaller, require less maintenance and are able to be stored taking up minimal space.

### **1.3 Objectives**

The project is to design and build a prototype lifting device to lift heavy objects. The prototype will be in the form of a heavy duty cherry picker. It will have a base which a hydraulic cylinder and a lifting beam will attach to. It lifts and maneuvers heavy objects with one man. It has a storage position to allow it to take up a minimal amount of space while not in use. The prototype was designed using Pro Engineer software. Finite Element Analysis (FEA) was performed using Algor software. The project kept money in perspective.

## **2. PROJECT DESCRIPTION**

### **2.1 Rationale**

With the prototype build of the lifting machine, nicknamed 'The Big Cherry', a single operator will be able to do what was once done by a team of two or more men. The Big Cherry will allow the operator to raise heavy objects off the floor, roll them to the desired location, and lower them in the desired position. Items will be moved from laboratory to laboratory, laboratory to transport device, and/or transport device to laboratory with ease.

The device helps eliminate the operator's work load while transporting. With its simple kinematics, a hydraulic system of sufficient lifting capacity is used to raise the desired load off the floor using this hoist. Then, the operator pushes The Big Cherry to its desired location.

Currently, at the Agricultural and Biosystems Engineering Department, a skidsteer, manufactured by Bobcat, is used to move class aids such as engines, transmissions, hydraulic components, and any other class aids in the Agricultural and Biosystems Engineering (ABEN) laboratories. The mobility and usefulness of this oversized machine is limited while inside the building. Tight spaces, engine exhaust, operating expenses, and fatigue during operation are just a few of the disadvantages to using the Bobcat skidsteer.

With the aggressive North Dakota winters that often appear, it is a major drawback to have to open the big shop doors to use the skid steer indoors. Not only does the room cool quickly, but the operation requires a two member team, one to open the shop door and another to operate the skidsteer, to do it effectively.

### **2.2 Physical Description**

A physical description of The Big Cherry can be imagined as a cherry picker on steroids. The Big Cherry is able to lift 4000 pounds at a full extension length of 8 feet. This allows The Big Cherry to place the heavy objects in the back of the department pickup or in the middle of the flat bed trailers.

The Big Cherry has a maximum lift height of approximately nine (9) feet which allows it to pick up the tallest class aid, an engine on a stand, and place it in the department pickup.

The Big Cherry has large, wide, heavy duty casters. The reason for the wideness is to travel over uneven surfaces, such as the floor drains in the laboratories. Also, a diameter of five to six inches was desired. This will help with the mobility and smooth transitions of the device over the floor drains and over uneven surfaces.

The hydraulic cylinder must be sufficient to control the 2 ton (4000 pound) load. The cylinder must have the ability to lower the load with precision. The load cannot suddenly drop causing a large impact on the floor and the lifted object. The load is to be raised and lowered either by hand pump or mechanically, by means of a hydraulic pump, directional control valve, pressure relief valve, and hydraulic cylinder. The decision will be based on funding capabilities.

The storability and maneuverability of our design was also another goal. The Big Cherry is designed to be collapsible. The legs and lifting arm are foldable for compact storage, but designed strong enough to handle the intended loads. In order to optimize the use of our device, the collapsible version fits through the doorways connecting the laboratories.

### **2.3 Constraints**

The first constraint requires the project to be completed by the beginning of May 2009. This will be the deadline for the design, cost estimate, and final report.

Another constraint is the cost. The funding for this project has not been decided, but must be considered in the design. A cost report will be required before construction is started. Funding to build the device will be provided, but is limited and will require an estimate before approval. The estimation must include material cost, cylinder(s) cost, caster wheel cost, and preventive maintenance, including priming and painting. The final project is to be painted to prevent corrosion and make it appealing to the eye.

As part of the design, FEA will be incorporated for stress estimation. This will assist in building the hoist to an adequate safety factor. The overall design should not be bulky and over built, but structurally sound.

### 3. Design Considerations

In this chapter, the process taken to design the heavy duty engine hoist is summarized. It is divided into four main sections: the **upperstructure**, **lowerstructure**, **hydraulic cylinder** and **caster wheels**. In order to understand this report, it is vital that the terminology used is recognized and understood.

#### 3.1 Upperstructure

The upperstructure is the support beam that is used to raise and lower the load to a desired height. It is very similar to a lever arm with the fulcrum placed at one end. Within the upperstructure is the **pinned pivot point**, the **pinned cylinder point**, the **pinned lift point**, and an **extendable arm**. The pinned pivot point is the attachment point between the upperstructure and the lowerstructure. It acts as the 'fulcrum' on a lever arm. The pinned cylinder point is the point of attachment for the hydraulic cylinder. It acts as the 'force' on a lever arm. The pinned lift point is the point that the load is attached to. It is the point where the 'load' is applied on a lever arm. Figure 3.1 and 3.2 show the comparison between the lever arm and upperstructure.

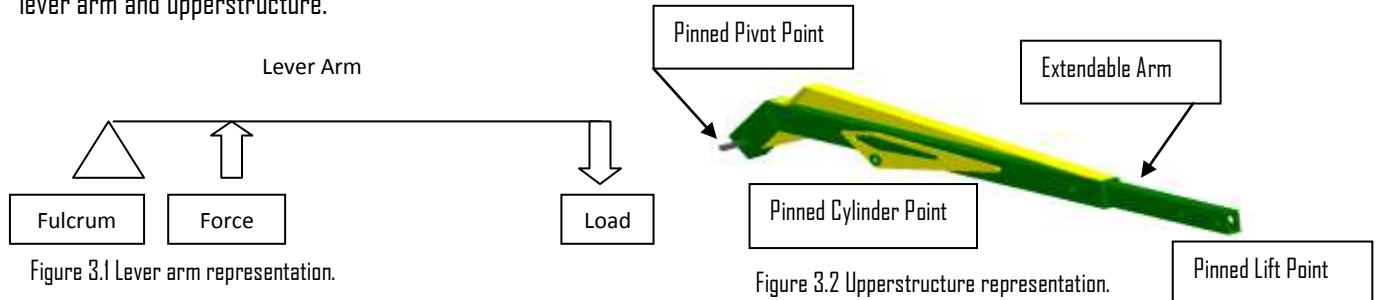


Figure 3.1 Lever arm representation.

Figure 3.2 Upperstructure representation.

Within the main beam of the upperstructure is an extendable arm. This extendable arm allows the engine hoist to lift a load at various lengths from the pinned pivot point. It acts as an extension to a lever.

#### 3.2 Lowerstructure

Attached to the upperstructure is the lowerstructure. The lowerstructure serves as the base for the upperstructure. It is the portion of the engine hoist that allows traveling to occur. It also, serves as the base support for the hydraulic cylinder. Figure 3.3 is a representation of a lowerstructure.



Figure 3.3 Lowerstructure representation.

### **3.3 Caster Wheels and Hydraulic Cylinder**

Attached to the lowerstructure are the **caster wheels**. These wheels are used to transport the load to its desired location. In order to lift the load to be transported, a **hydraulic cylinder** attached to both the upperstructure and lowerstructure is used. A hydraulic cylinder with a certain tonnage capacity is used.

This terminology will be used throughout this report. For clarification, please refer to this section for future reference.

## 4. DESIGN of UPPERSTRUCTURE

### 4.1 Considerations

The upperstructure design of the assembly is what holds the weight being lifted. It must be structurally sound to lift, maneuver, and withhold any extra forces that may be factors while the machine is in use. From here, the weight will be pushed, lifted, lowered, and swung around. Since the beam will be used for hoisting objects, it needs a hook or mechanism to fix itself to the lifted objects. Table 4.1 illustrates some of the design criteria for the upperstructure.

Table 4.1 Upperstructure specifications.

Max lifting capacity	4,000 pounds
Max lifting height	9 feet
Max extension length	8 feet
Max lowered height	3 feet
Minimum safety factor	1.5-2

### 4.2 Design Steps

The first step was to come up with a basic design layout. After a rough estimate of the dimensioning, Pro-Engineer was used to model the designs. After the design was modeled, Algor was used to complete the FEA.

The extended length needs to be eight feet but it needs to be foldable for storage. The dimensions for the main beam will be approximately 6 feet with an extension of about 2 feet. The extension piece will be a total of 3 feet to allow one foot of interaction within the first beam. This extension piece will also be able to retract into the main beam, allowing a shorter length for storage.

After brainstorming, several ideas were considered. These ranged from round tube stock design, a square tube stock design, to a double tube stock design. Due to the extreme weights that will be lifted, it was assumed that a structural support would be needed to minimize the stress. Two ideas were a triangular support and a rectangular support. These two ideas are analyzed. Another area where weakness could occur is where the lifting mechanism is attached. This attachment has to be added as a way of attaching a cylinder or other lifting mechanism and also to increase the structural integrity of the beam. A final option comes from the pinned end where the lifting beam will connect to the base support. This could either be a straight connection or an angular connection. Both ways will be explored.

The first design involved a tube design. Two tubes, one that slides into another would be used as the main lifting beam. The dimensions were 7 inch diameter and 0.5 inch sidewall and a 6 inch diameter with a 0.5 inch sidewall. One issue with this construction is the amount of weight per foot in the materials. The 7 inch diameter tube is approximately 42 pounds per foot. Figure 4.1 shows a simple representation of what a round tube design might look like.

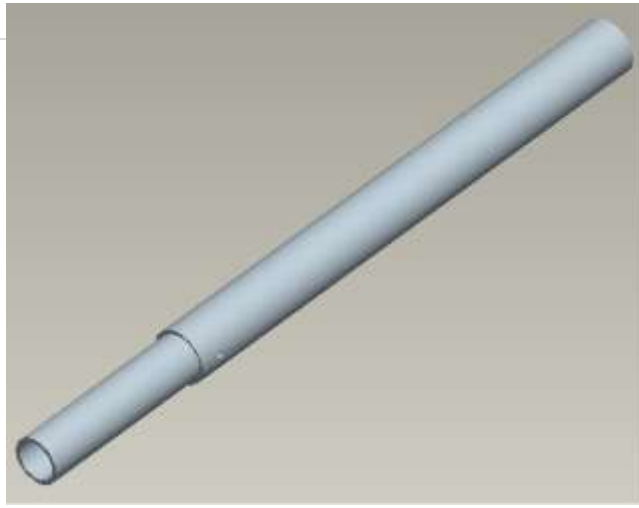


Figure 4.1 Design 1.

The second design was influenced from other engineers. It consisted of a two beam lifting arm. Two beams of smaller caliber would be placed side by side to lift the objects. The main tube dimensions were 3.5 inch by 2.5 inch by 0.25 inch. This design would allow for smaller steel tubing to be used. This design can be seen in Figure 4.2.

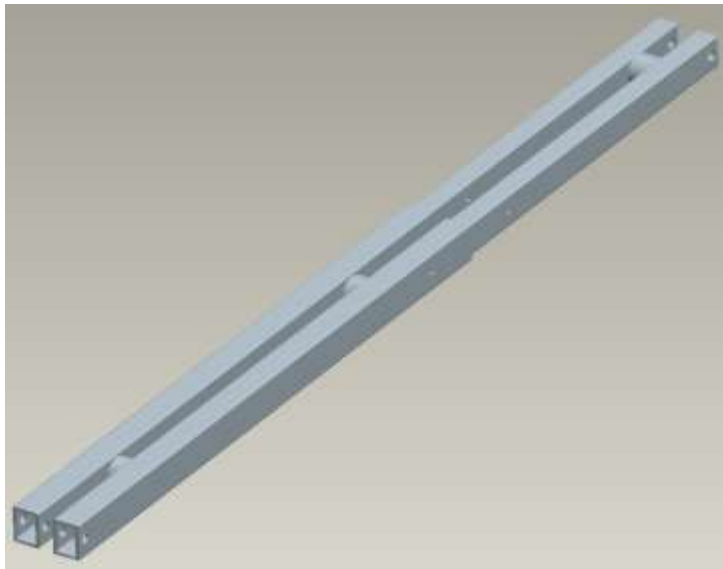


Figure 4.2 Design 2.

The third design also came from other engineers. Design 3 consisted of a 3.5 inch by 3.5 inch by 0.25 inch thick square tube. This beam was designed to be similar to a truss. Although this design would allow for smaller tubes, weight and cost to manufacture becomes a big issue. Due to these issues, this consideration was discarded. Figure 4.3 is a preview of what design 3 may have looked like.

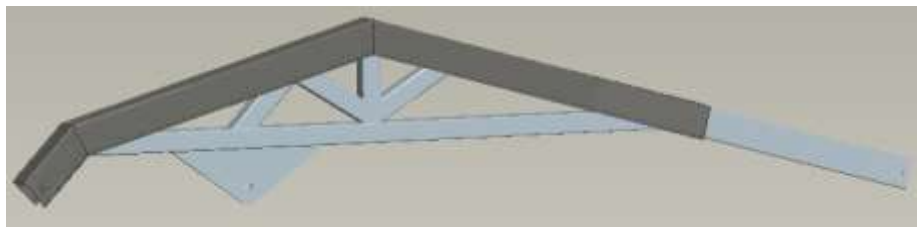


Figure 4.3 Design 3.

Design 4, shown in Figure 4.4, had a 4 inch by 4 inch by 0.25 inch thick square tube. The pinned end was set at a 45 degree angle. Gussets were put in the angle to support the bending moment. The gussets are 0.25 inch steel. The structural truss support on top is used to help reduce bending in the beam when the beam is loaded and is also constructed of 0.25 inch steel. The cylinder mount, shown in Figure 4.5, was designed to increase the structural integrity of the beam. It consists of a plate 0.3125 inch thick. The front and rear points are shaped to help disburse loadings. The inside is open to allow for more welding surfaces, which in part will increase the structural integrity. The tubes and sheet metal are ASTM A36.

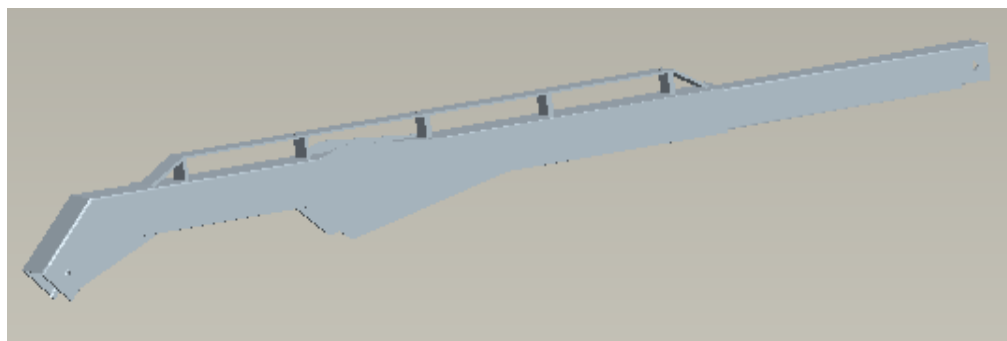


Figure 4.4 Design 4.

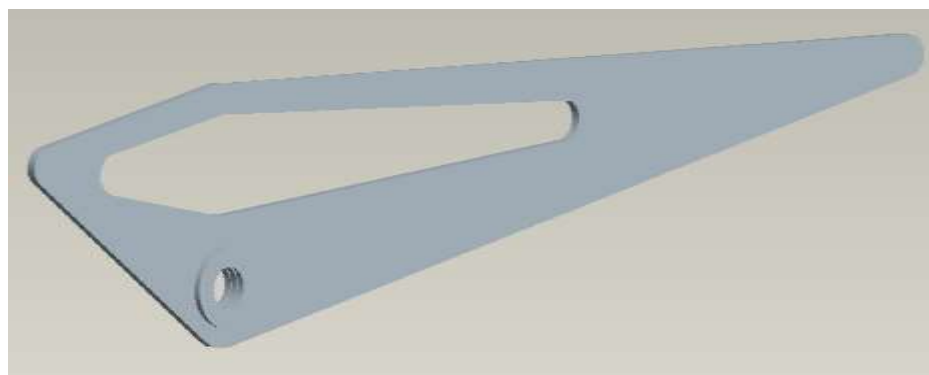


Figure 4.5 Cylinder Gusset.

After initial analysis of this design, it proved to be inadequate, specifically, the bottom of the main beam, the points where the sliding beam was positioned, and the holes. Figure 4.6 shows how the assembly was loaded for the analysis. A load of 4000 lbs was placed on the right end. The points where the hydraulic cylinder would be placed was fixed in the y-direction, and the pinned end that attaches to the lowerstructure was constrained in all directions. This load casing was used for all upperstructure beam loadings and should be referred back to.

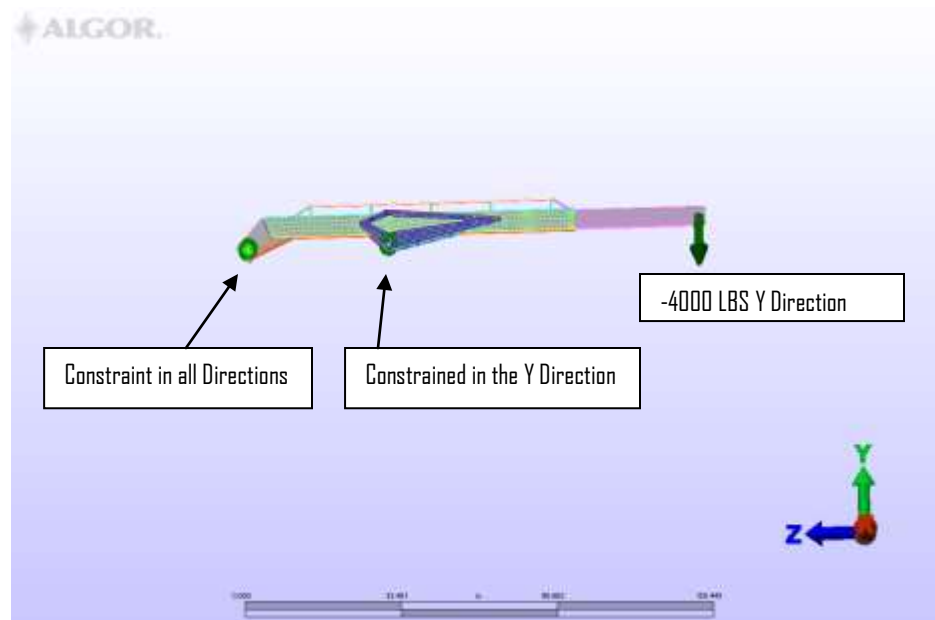


Figure 4.6 Loading on Design 4.

Figure 4.7 shows the results of this load case. The beam had a max stress of 78,555 pounds per square inch which is unacceptable.

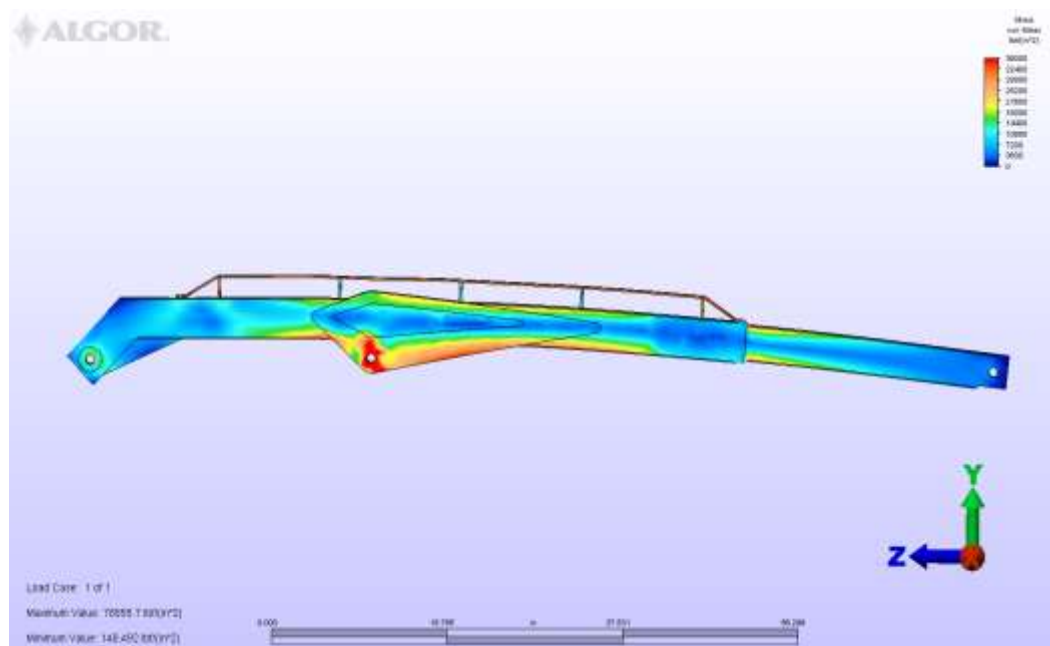


Figure 4.7 FEA results of Design 4.

After these initial designs were made, rough analysis was done using Algor. The analysis showed that the small tubes being used were not adequate for the design loading necessary. The next round of designs incorporated structural steel, ASTM A500 for the tube design and ASTM A36 steel for the other parts. The ASTM A500 has a yield strength of 46000 pounds per square inch. Steel of ASTM A36 has a yield strength of 36000 psi.

The following design 5 consisted of a 4 inch by 4 inch by 0.5 inch thick square tube design. The square tube designs are a less expensive option than rectangular tubes. This main beam in Figure 4.8 was tested with a straight end and a 45 degree angle pinned end which helped spread the forces over the beam. Both triangular and rectangular supports were tried.

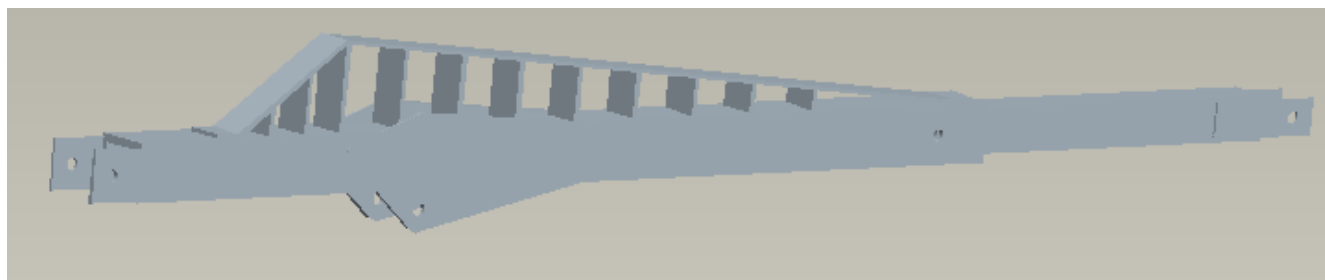


Figure 4.8 Design 5.

Figure 4.9 shows the FEA of the beam. The maximum stress was 145,889 pounds per square inch. The stress was localized in the pinned end holes, the cylinder mount holes, and the sleeve area where the inside beam ended.

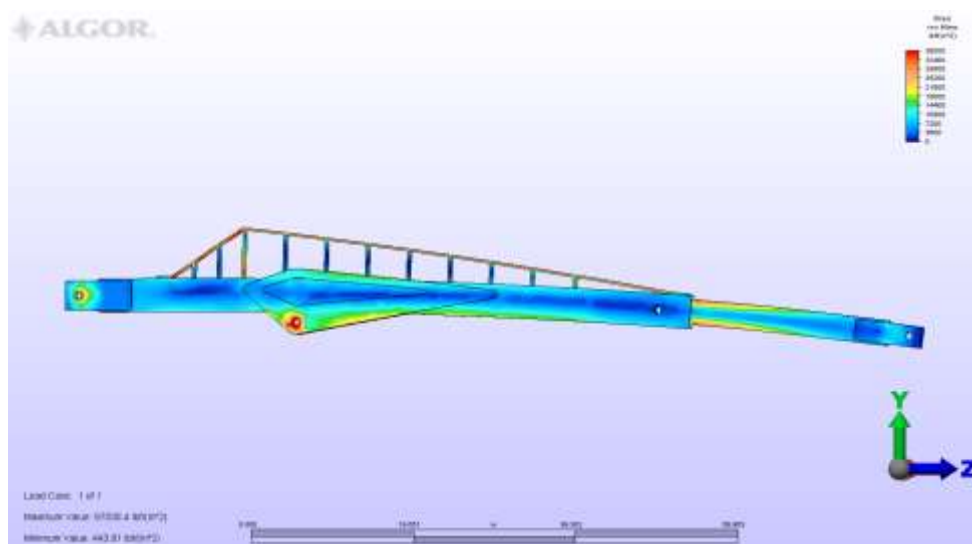


Figure 4.9 FEA results of Design 5.

After analyzing the last loadings, a few design areas were changed. One thing that was added were bushings or thicker material where the pins would be. Bushings were added in the next model. In order to help reduce the stress on the top of the beam, the angled beam seen in Figure 4.10 was used. Another design change then came in the truss design itself with a solid member instead of multiple gussets. This helped increase the strength while also increasing the manufacturability. This heavier built design can be seen in Figure 4.11. The results for design 7 are shown in Figure 4.12



Figure 4.10 Design 6.

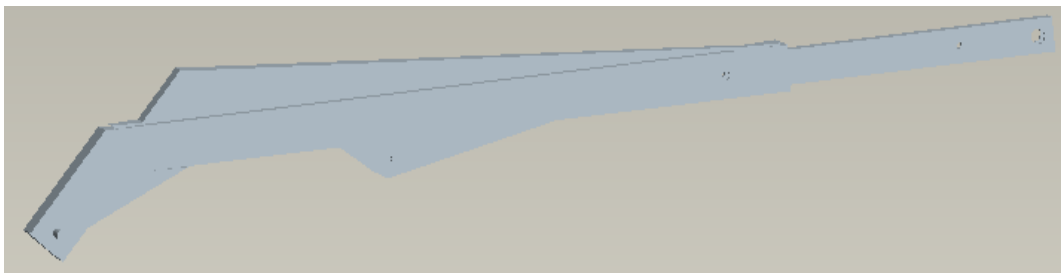
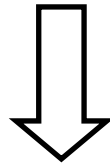


Figure 4.11 Design 7 .

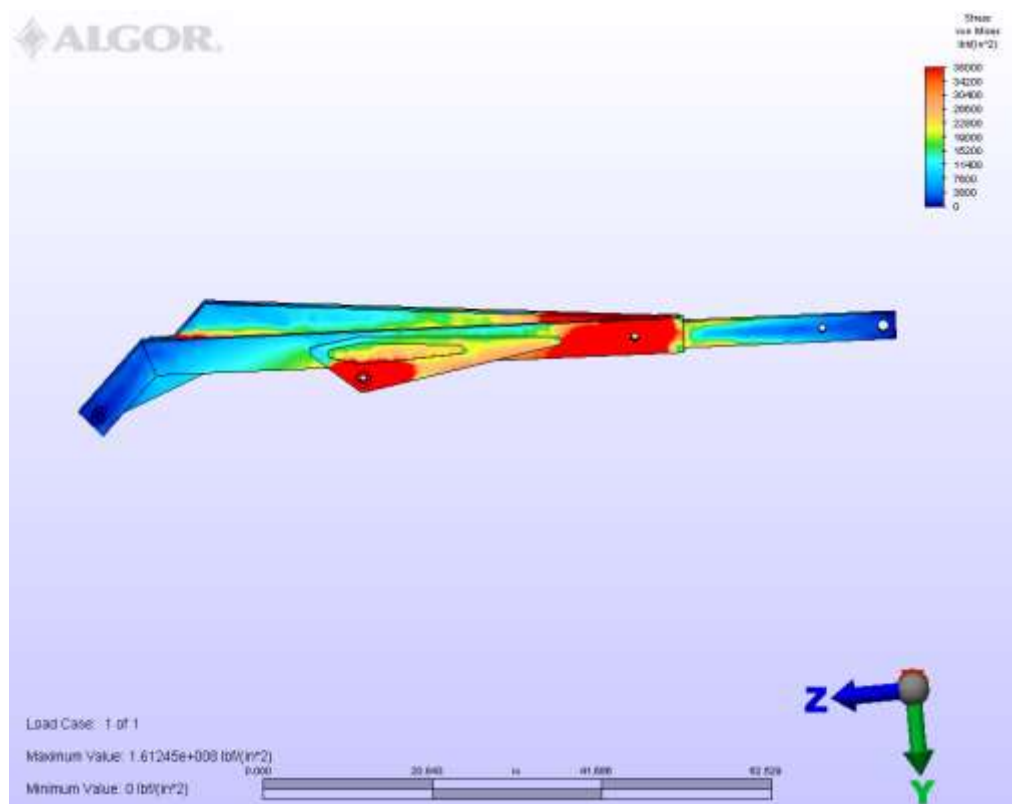


Figure 4.12 FEA Results of Design 7.

The stress in this design was located in the cylinder mounts and the end where the sliding beam sits inside the main beam. The maximum stress was about 161,000 pounds per square inch.

The selection of square tubes was changed to rectangular tubes due to inadequate results and a better area moment of inertia in this circumstance. Design 8 was a 6 inch by 4 inch by 0.5 inch thick main beam shown in Figure 4.13. The angled end and the solid truss support were retained from Design 7.



Figure 4.13 Design 8.

This beam passed the analysis with adequate stress and safety factor. The results can be shown in Figure 4.14. Since this was a heavy design and was adequately built, a 5 inch by 3 inch by 0.5 inch tube of similar design was also tried.

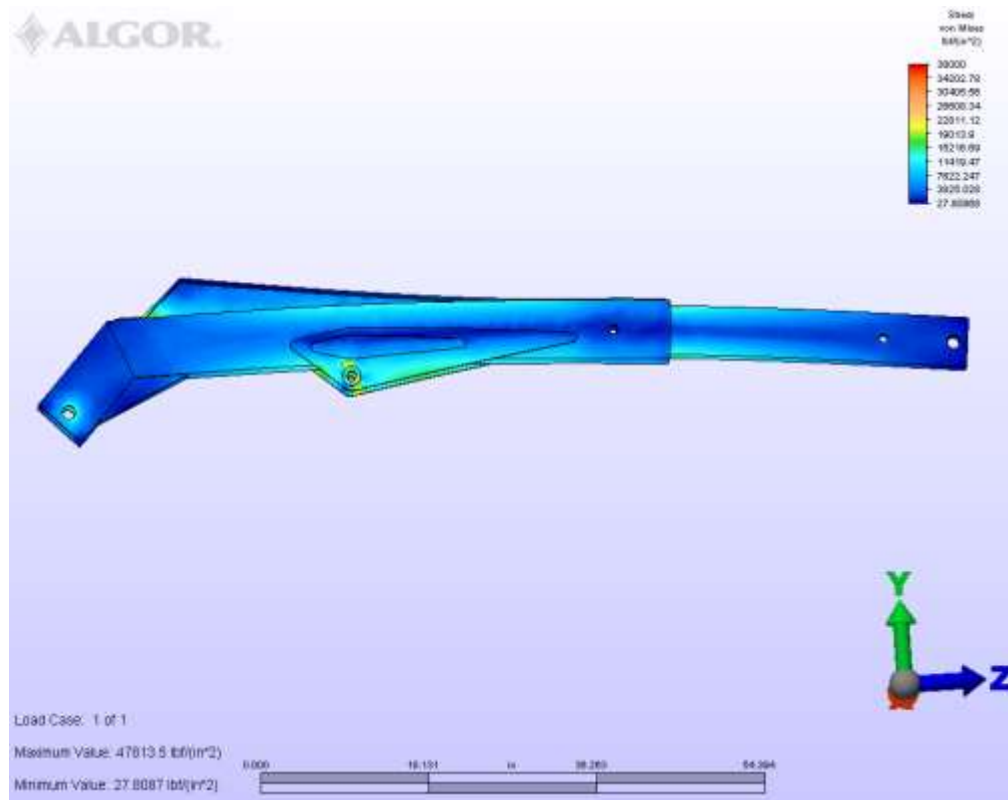


Figure 4.14 FEA results of Design 8.

### 4.3 Final Design

The final design, Design 9, utilizes a 5 inch by 3 inch by 0.5 inch tube (Figure 4.15). It was able to handle the loading of 4,000 pounds and incorporate a reasonable safety factor, of 1.5. Figure 4.16 and 4.17 show the stress and factor of safety in the beam.

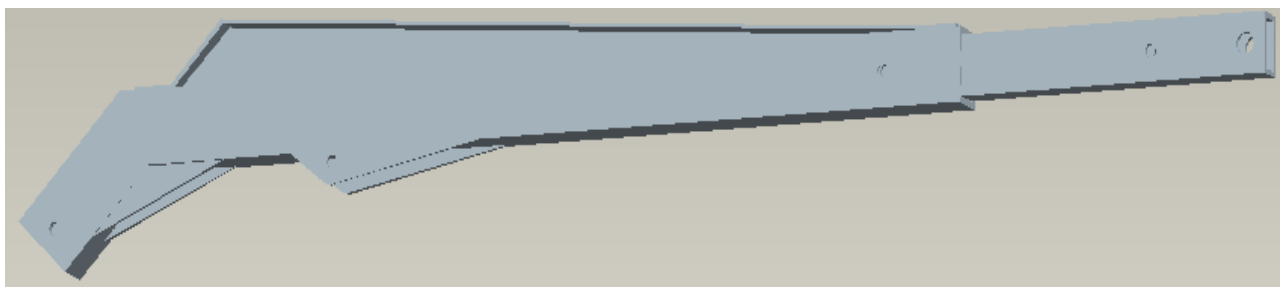


Figure 4.15 Design 9.

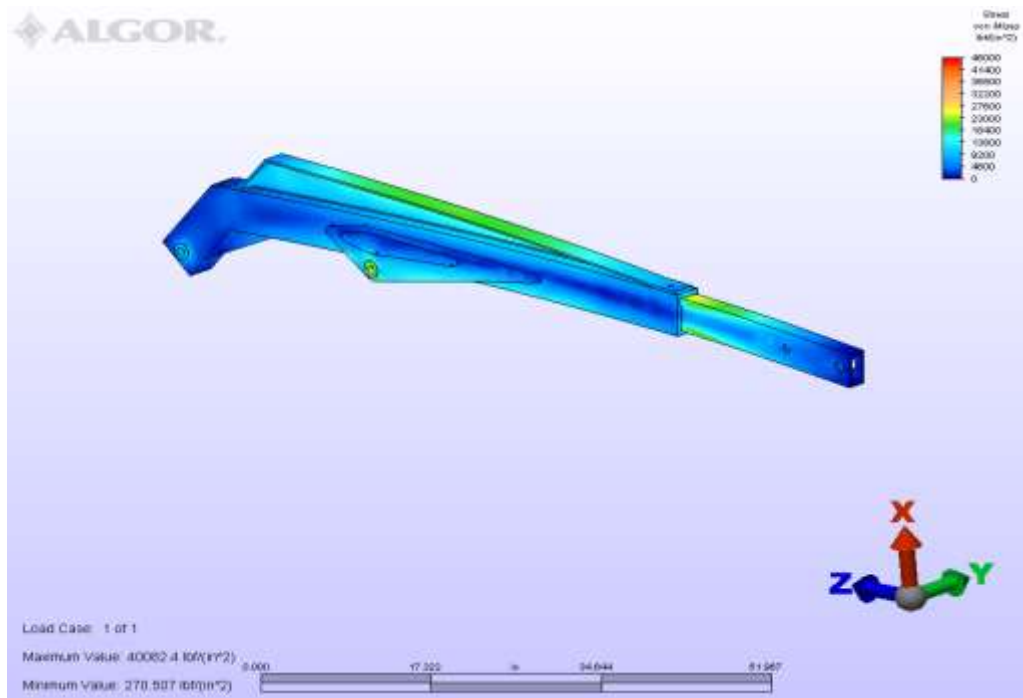


Figure 4.16 FEA results of Design 9.

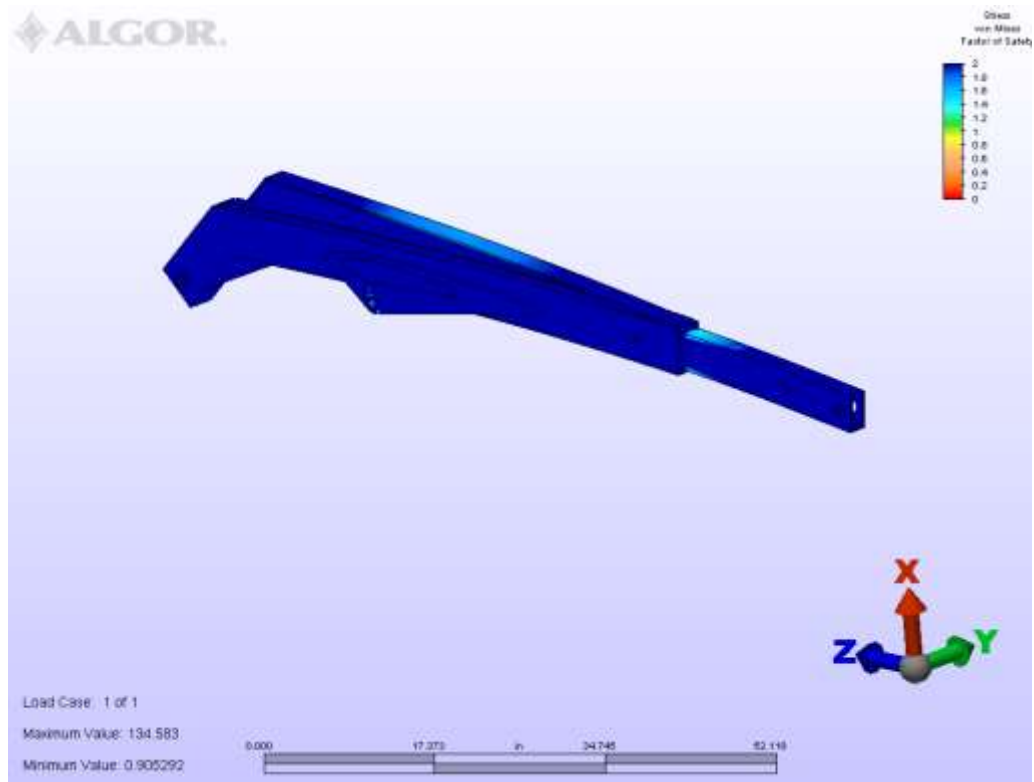


Figure 4.17 FEA factor of safety of Design 9.

## 5. DESIGN OF LOWERSTRUCTURE

### 5.1 Considerations

As mentioned earlier in this report, the lowerstructure serves as the base for the upperstructure. The pinned pivot point functions as the main connection point between the lowerstructure and upperstructure. The base of the hydraulic cylinder also attaches to the lowerstructure. All these factors needed to be considered to determine the optimal design. The first step in deciding the design of the lowerstructure was to explore all the available alternatives.

### 5.2 Design Steps

There are many different alternatives that were considered. The first one involved a complete design using formed sheet metal rather than steel tubing. This design would allow elimination of stress concentration areas by increasing the cross-sectional area in the high stress regions, but would dramatically increase our input cost. Also, manufacturability of the parts would not be possible by the available equipment. The cost of the steel would include the material and labor required to cut and form the steel. After much consideration, this alternative was disregarded. A steel tubing with minimal sheet metal structure was chosen.

The initial steel tubing design started off with the upright beam at a  $90^\circ$  angle. Figure 5.1 is a representation of this design. After much consideration, a decision was made that this design was very under designed for its intended purpose. A large amount of extra bracketing would be needed to make the large load stable.

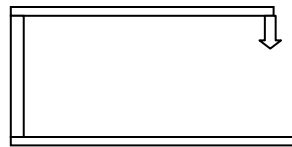


Figure 5.1.  $90^\circ$  angle design.

Following this initial thought, an angled design was then started. Figure 5.2 was the first initial design. It consists of the angled supports at  $75^\circ$  from the horizontal, foldable and extendable legs, the cylinder base support, caster wheel supports, and the pinned pivot point which is the point of attachment for the upperstructure. This initial model became the layout for the final result.

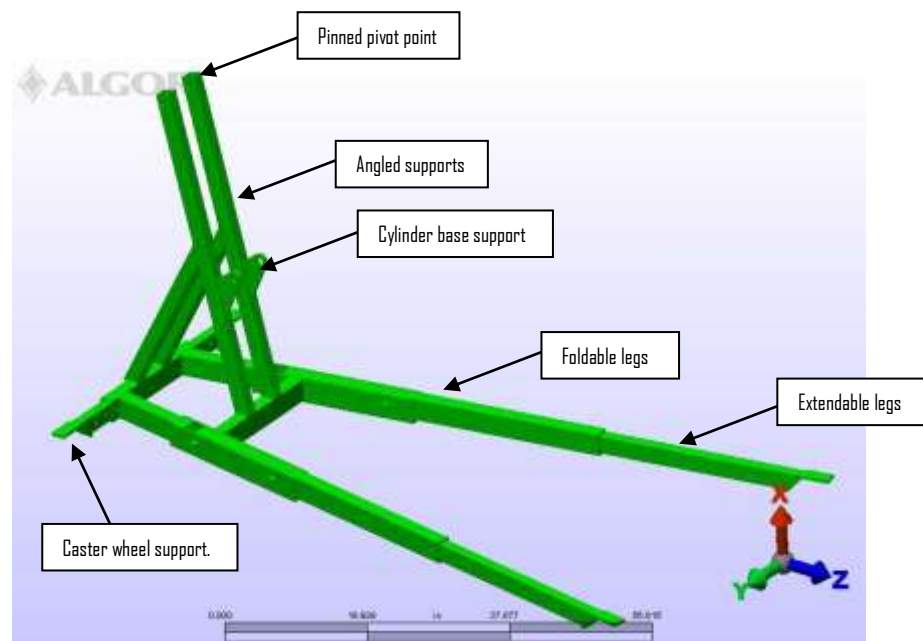


Figure 5.2. Initial design representation.

The initial angled design, pictured earlier, was designed in ProEngineer, and then, analyzed using FEA. By using Algor, loads and boundary conditions were applied. Figure 5.3 shows the loading and constraints used to analyze the lowerstructure. These loads did not change for normal loading throughout the design process. Additional special load cases were also performed.

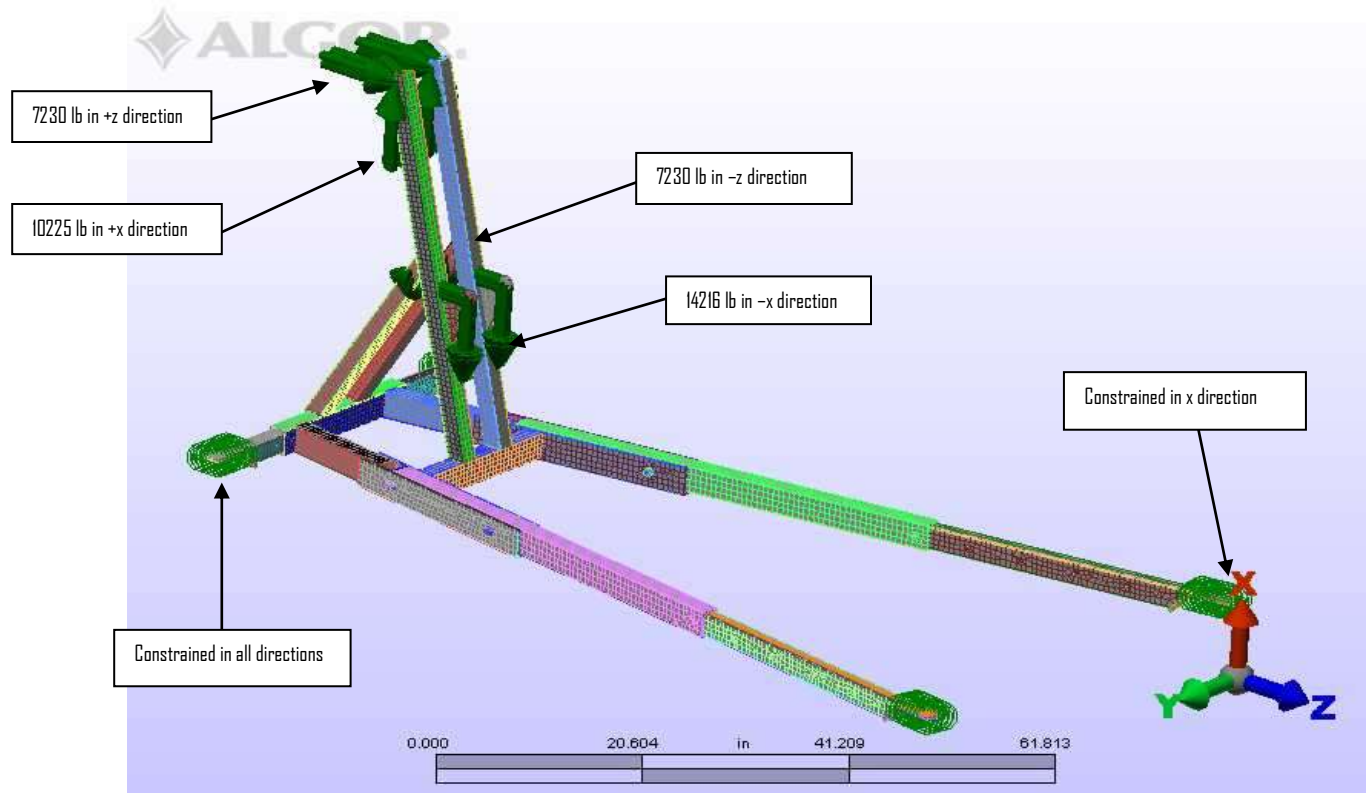


Figure 5.3. FEA forces and boundary conditions on initial run.

As you will notice, the upperstructure loading is broken down into loading at the pinned pivot point and the cylinder support. At the pivot pin, a force of 7230 pounds force is applied towards the front of the model and 10,225 pounds force is applied in the upward direction to simulate the loading occurring from the connection with the upperstructure. In order to simulate the loading produced from the hydraulic cylinder, 7230 pounds force is applied towards the rear of the model and 14216 pounds force is applied in the downward direction.

In this design, 2 inch by 2 inch by 0.25 inch thick square tubes are used for the supports connected to the upperstructure. For the rest of the structure, 2 inch by 3 inch by 0.25 inch thick tubes were used accompanied by 1.5 inch by 2.5 inch by 0.25 inch tubes that slide inside the 2 inch by 3 inch by 0.25 inch thick tubes. The structure used to allow foldability is a "U" shaped 0.25 inch thick formed sheet metal. The tubes are comprised of ASTM A500 steel and the formed sheet metal is ASTM A36.

In this scenario, the simulation of a 4000 pound load is applied to the lowerstructure. The outside caster supports are constrained from any movement. Figure 5.4 represents the results acquired from Algor.

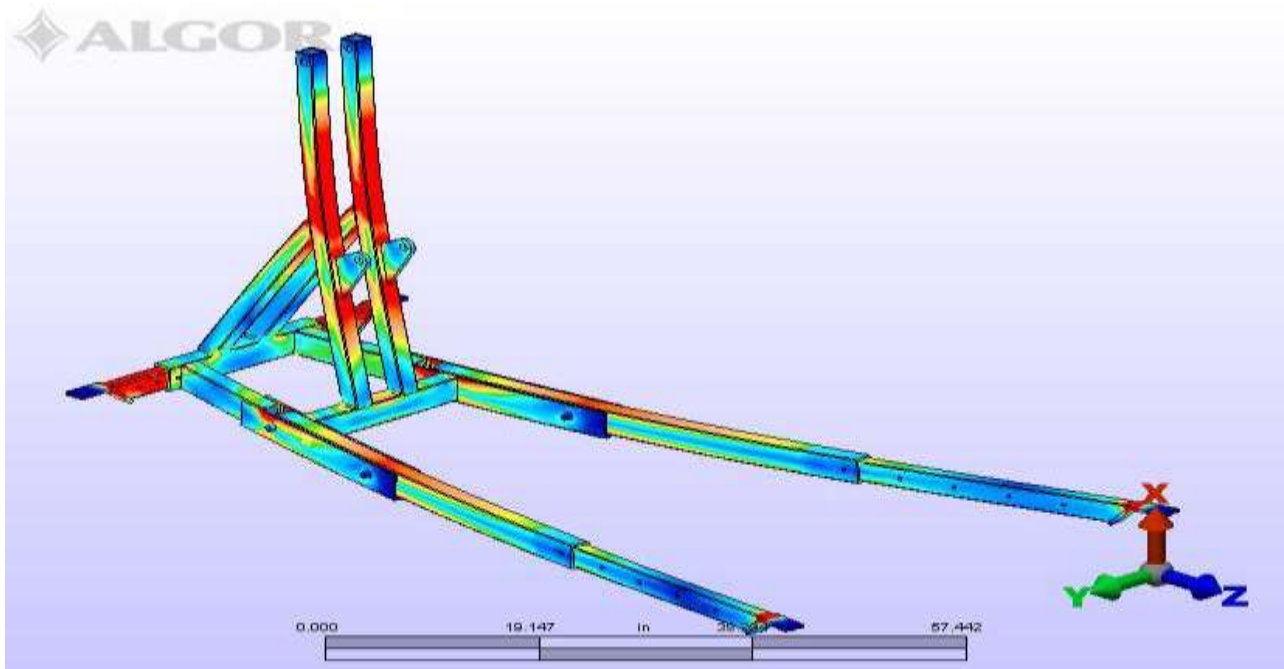


Figure 5.4. FEA results for initial design.

This design is not sufficient to hold the load. The red areas are where the stress level reached the yield strength (46000 pounds per square inch) of the ASTM A500 steel. The 2 inch by 2 inch by 0.25 inch thick square tubes used for the upright supports are insufficient for the load case. However, after further discussion with Dr. Thomas Bon, it was realized that the initial constraints were wrong. This loading will not be the normal cyclic loading. This loading case is a highly unlikely scenario on the recommended even ground operation.

In the second FEA run of the initial design, the center wheel caster was also constrained along with the end casters. Figure 5.5 shows the setup.

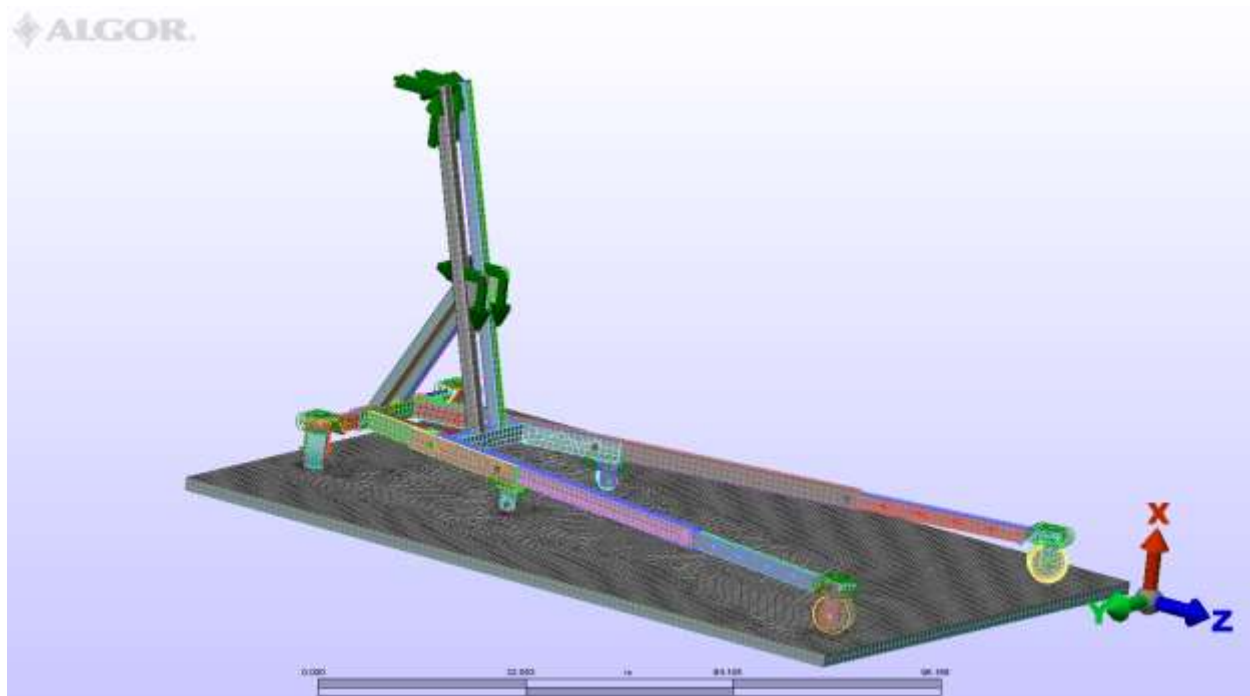


Figure 5.5 FEA constraints for second design.

This analysis showed an improvement from the first initial run, but showed a lot of potential concerns. But once again, after further consultation with Dr. Thomas Bon, it was realized that this analysis is not right either. The rear casters should not be constrained as they will have the ability to move in the upward direction.

So, once again, another analysis with different constraints was conducted. However, this time, the constraints were correct. Only the center and front casters were constrained. A slight design change also was done. The rear upright supports were adjusted upward to help support the bending moment. The results are shown in Figure 5.6. It is apparent that the stresses are quite high, especially in the upright supports. The moment created at the pivot point and hydraulic cylinder support is too high for the upright supports. Also, the folding support shows some potential concerns of buckling from heavy compression and tension.

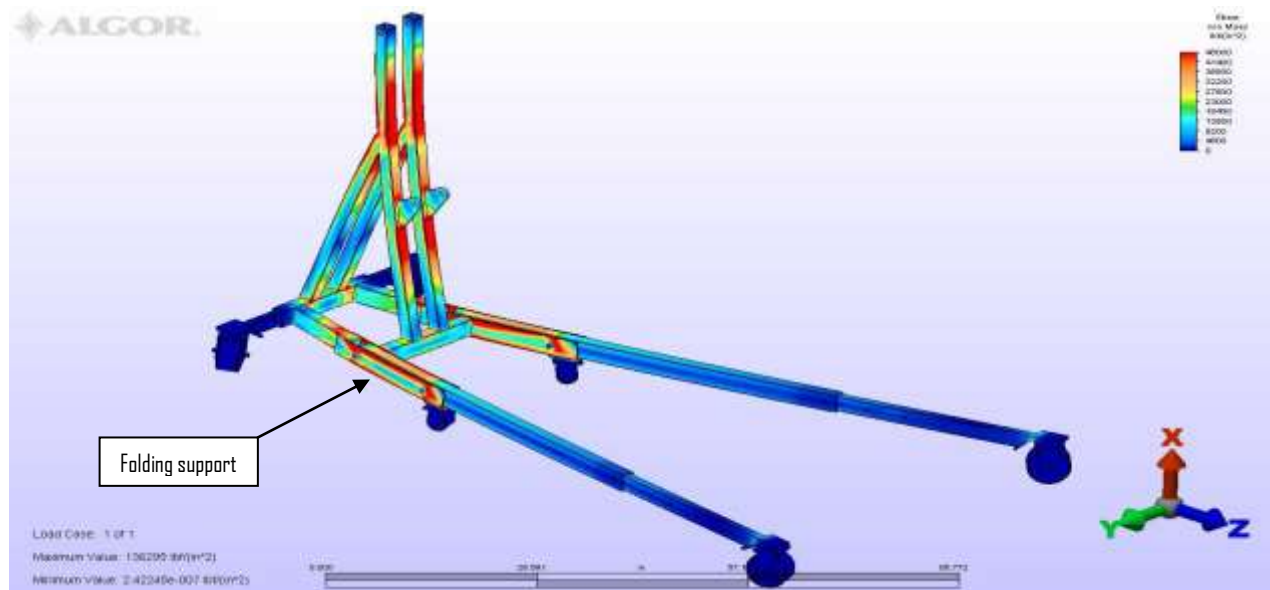


Figure 5.6 FEA results.

From these results, it is apparent that the upright supports needed to be increased in size in order to meet the requirements. However, meeting the requirements is not sufficient. A safety factor of 1.5 to 2 needs to be demonstrated for this machine to hold its structural integrity. So from the results, the tubes were increased in size to 2 inch by 3 inch by 0.25 inch thick and analyzed again.

Figure 5.7 shows the results. It is still apparent that the design is not sufficient. With the max load stress set to the yield strength of the ASTM A500 tubes (46,000 psi), there are still several hot spots. Several concerns exist in this model with the main concern being the upright supports having very high concentrated stress around the connection point. In order for a successful design, this stress needs to be distributed.

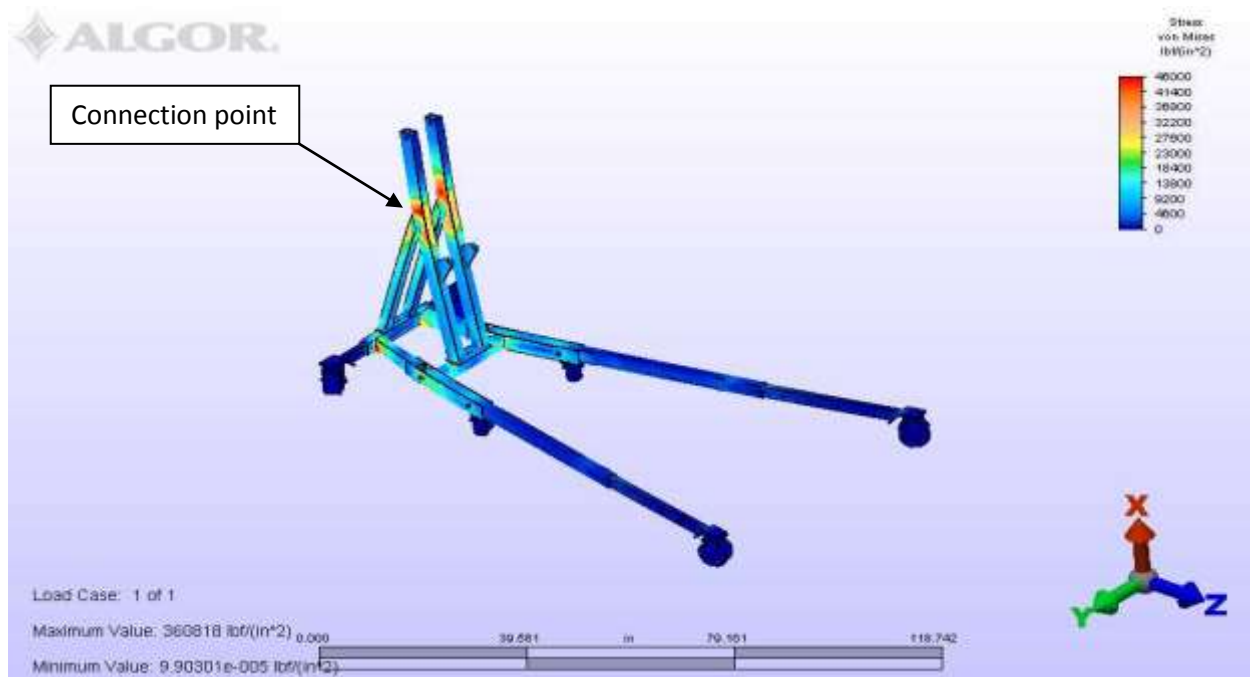


Figure 5.7. Results for FEA.

For the next design, gussets were added to the upright supports, as seen in Figure 5.8, in order to spread the stress concentration out. This helped tremendously in the upright supports. However, it transmitted more of the load into the bottom frame, in particular the folding support. So from this model, it was found that additional material is required in the folding supports.

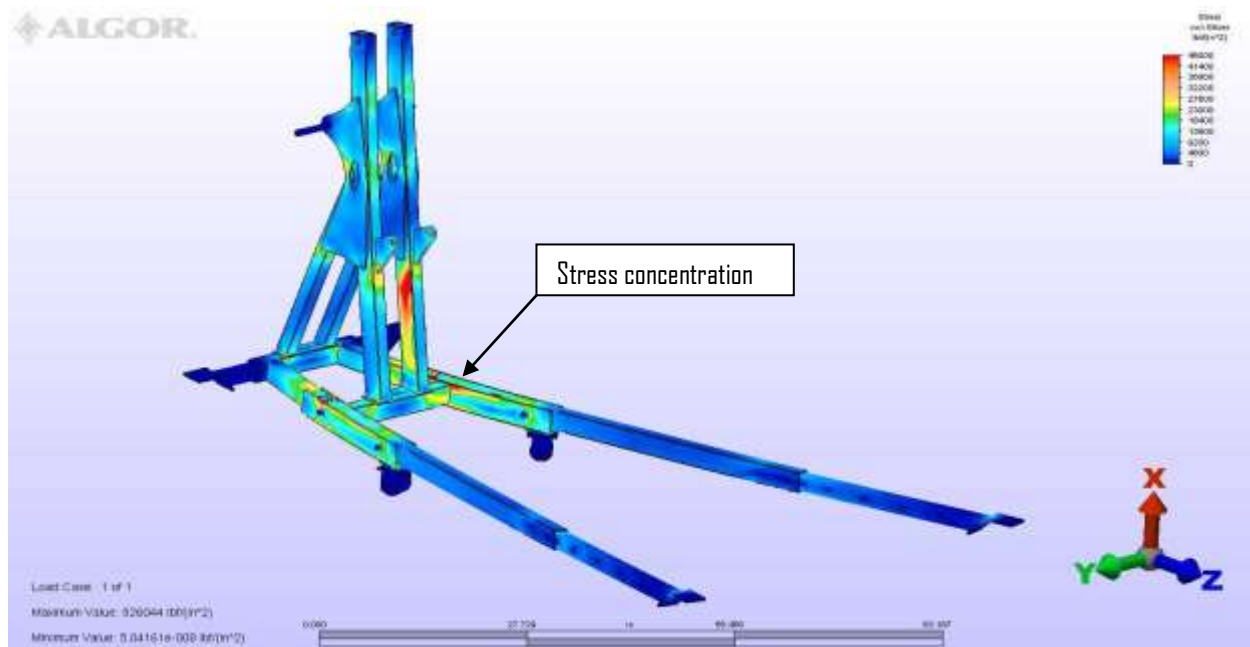


Figure 5.8. FEA results.

In the next design, top covers were added to help support the base. In this configuration, the top covers serve as top support to the folding support. However, as seen in Figure 5.9 and 5.10, this idea did what it was designed to do, but

contributed to stress in another area, the area with the smallest cross-section. Also in this design, the folding support was increased in length so the welded area would increase in perimeter, limiting the stress in the weld.

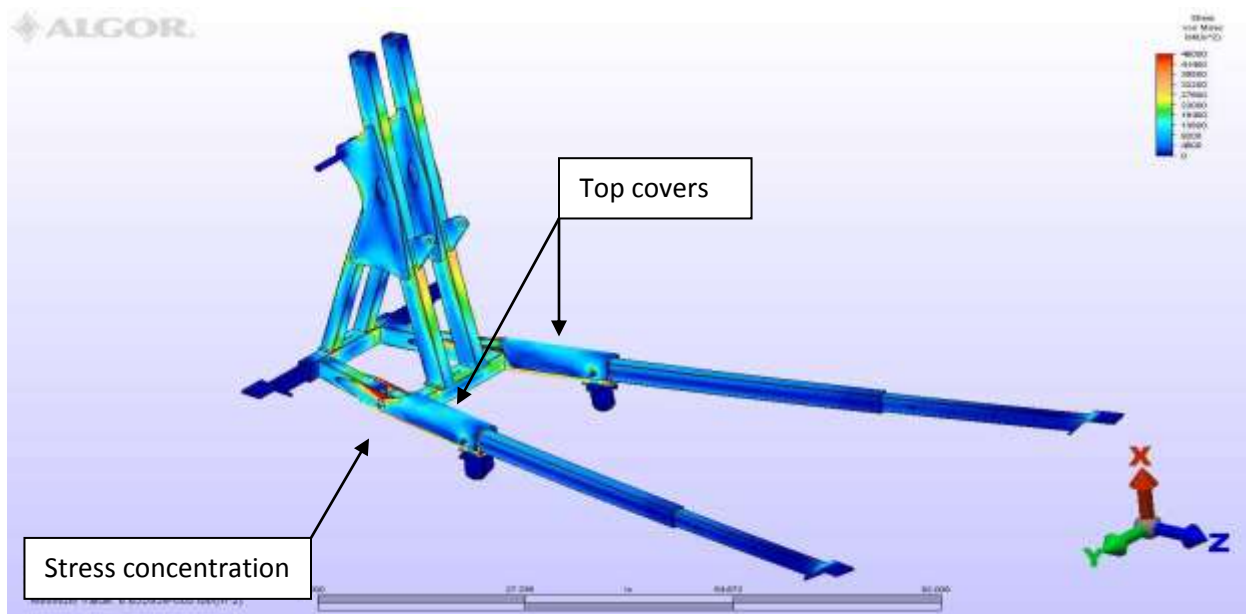


Figure 5.9. Outcome from FEA.

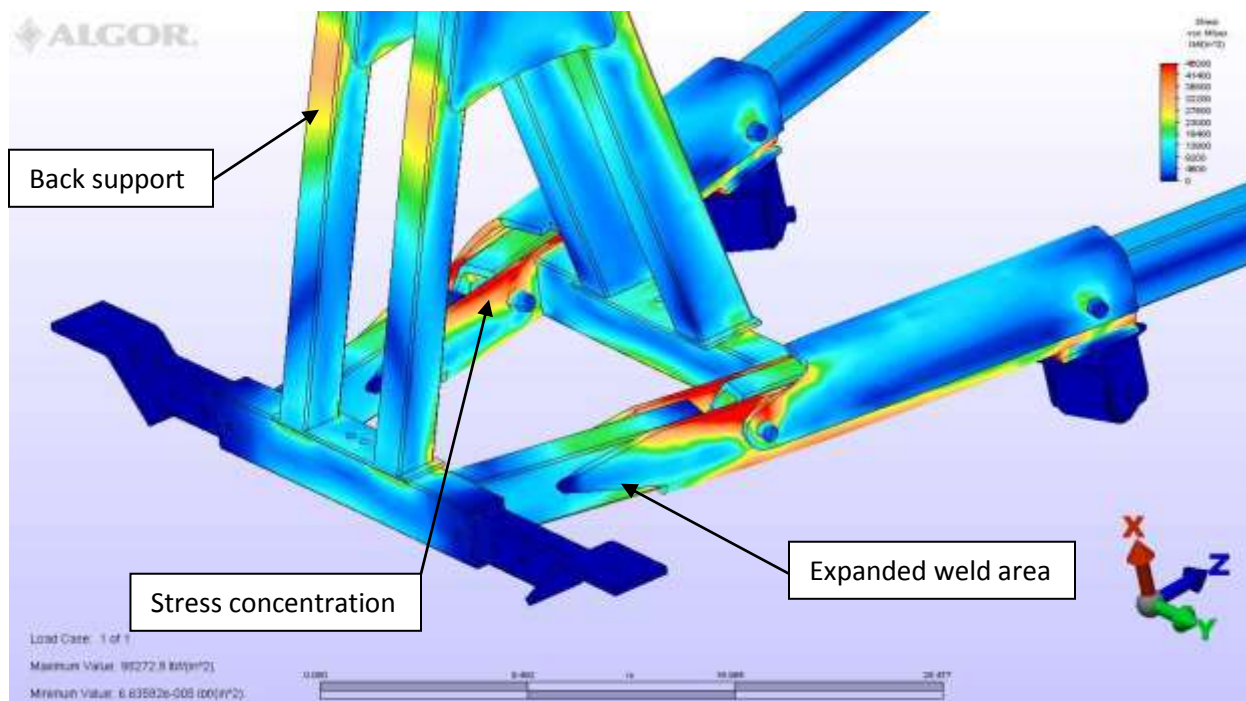


Figure 5.10. FEA stress concentrations and weld area.

Since the folding support is composed of only 0.25 inch thick formed sheet metal, a decision was made to increase this to 0.375 inch thick. Also, slight stress concentrations are still being seen on the upright supports. For the next design, the angled and back supports were increased in size to 3 inch by 3 inch by 0.25 inch thick tubing. Also, an

additional pin was inserted into the top cover to add more strength to the folding support. Figures 5.11 and 5.12 represent the results found.

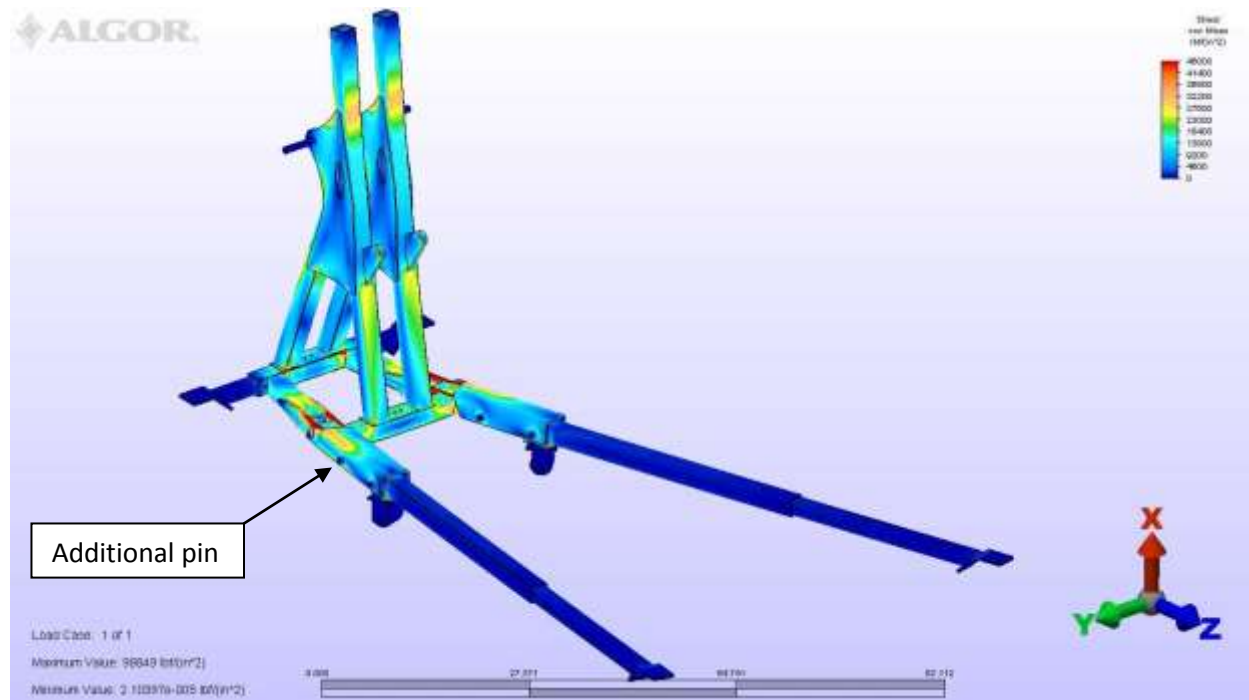


Figure 5.11. FEA results using 3 inch by 3 inch tubing.

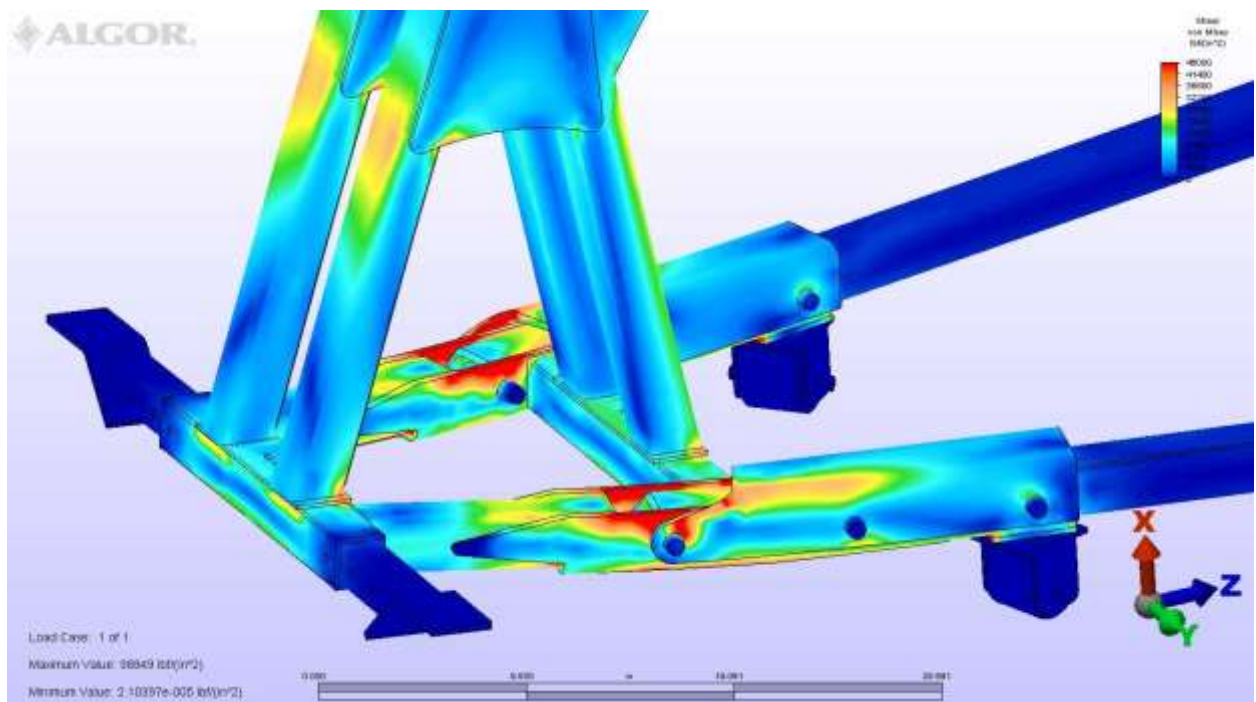


Figure 5.12. FEA results from 3 inch by 3 inch tubing.

From these results, it seems the new tube size will not help in dispersing the stress. So, the angled supports were changed back to reduce weight and to make manufacturing easier. However, because the 0.25 inch thick wall was insufficient, the next size (0.3125 inch) was selected.

Since the folding supports are an obvious weakness to the lower structure, additional steel was added to the sides and bottom for increased structural integrity. These supports are directly welded to the inside, outside, and bottom of the folding supports. The outside support lies against the folding support and inside the top cover.

This additional steel in both the folding support and upright supports adds additional weight and cost, but is required for the structural soundness. Figures 5.13, 5.14, and 5.15 are the results found.

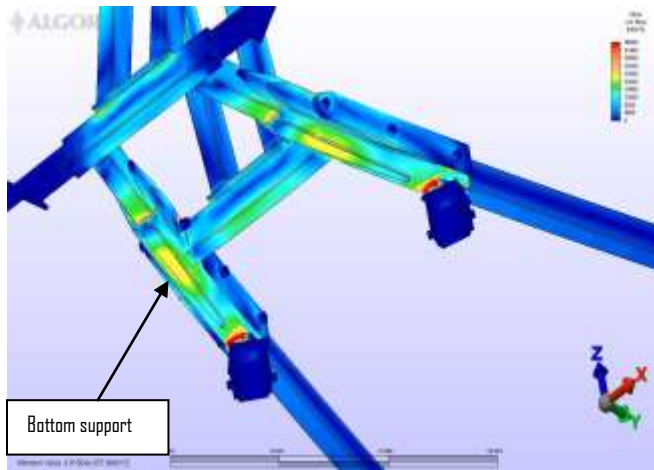


Figure 5.13. FEA 1 results using additional supports.

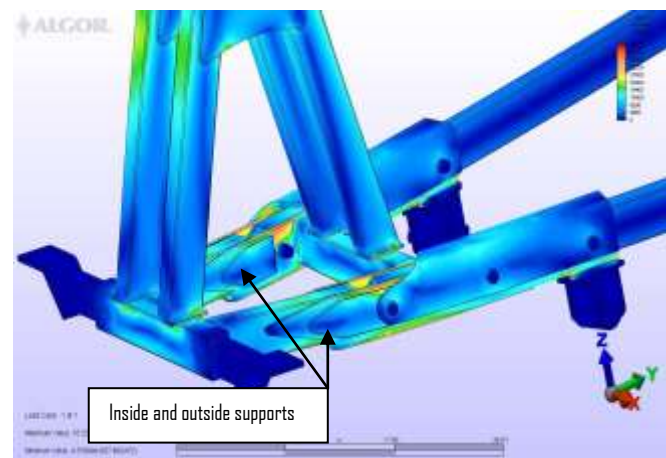


Figure 5.14. FEA 2 results using additional supports

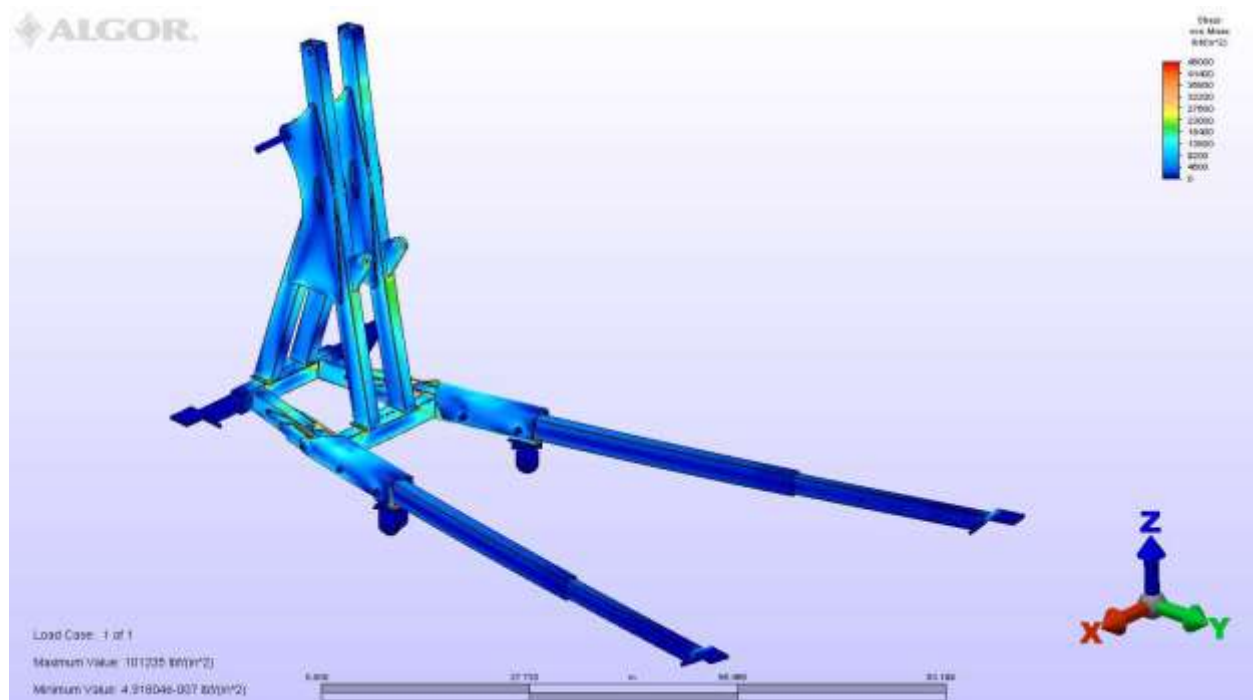


Figure 5.15. FEA 3 results using additional supports.

### 5.3 Final Design

The design process has evolved to a very structurally sound lowerstructure design. However, because special load cases have not been applied, an additional design iteration was completed. An additional truss support was added to the folding support, and the extra piece added to the inside of the folding support was removed. Figures 5.16. and 5.17 illustrate the added bottom truss support and the final design.

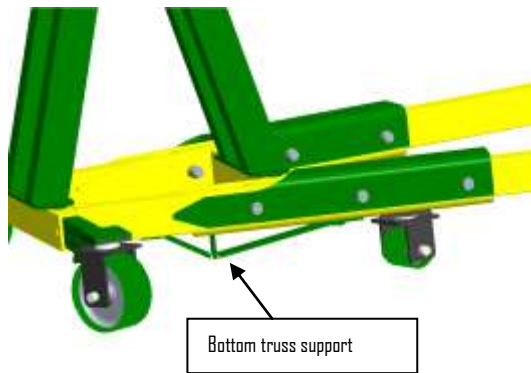


Figure 5.16. Final design bottom truss illustration.



Figure 5.17. Final design model.

One final Algor analysis was conducted to make sure the design was structurally sufficient. The same loading was applied to this model as before. The results were rather convincing that the design is sufficient. Figures 5.18 and 5.19 show the FEA on the model. It still has noticeable stress concentration areas, however, they are not high enough to cause any issues. The reddish colored areas have a stress value of 36,000 pounds per square inch; this is the yield point of the ASTM A36 steel.

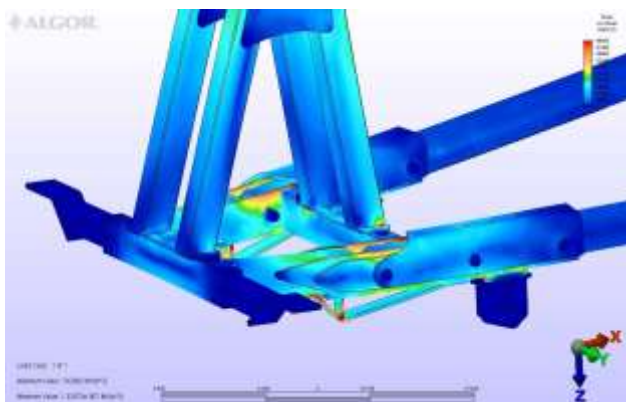


Figure 5.18. Final design FEA result 1 with normal load without beam.

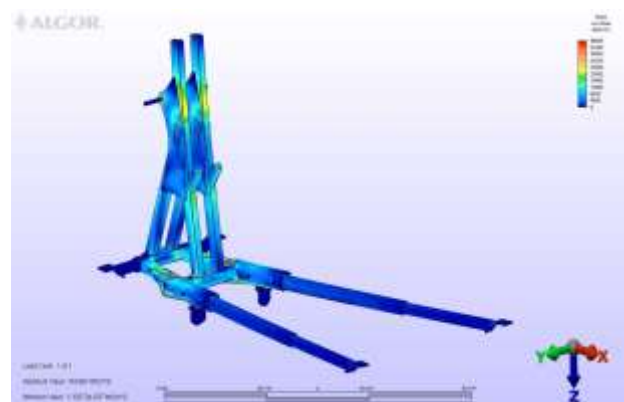


Figure 5.19. Final design FEA 2 result with normal load without beam.

After this rigorous load was conducted, the beam was attached and further analysis was conducted. In the first case, the 4000 pound load was placed on the beam, and the center and front casters were constrained as before. The cylinder support attached to the upperstructure was constrained in the z direction. The normal forces experienced by the cylinder supports on the lowerstructure were also applied. Figure 5.20 shows the setup.

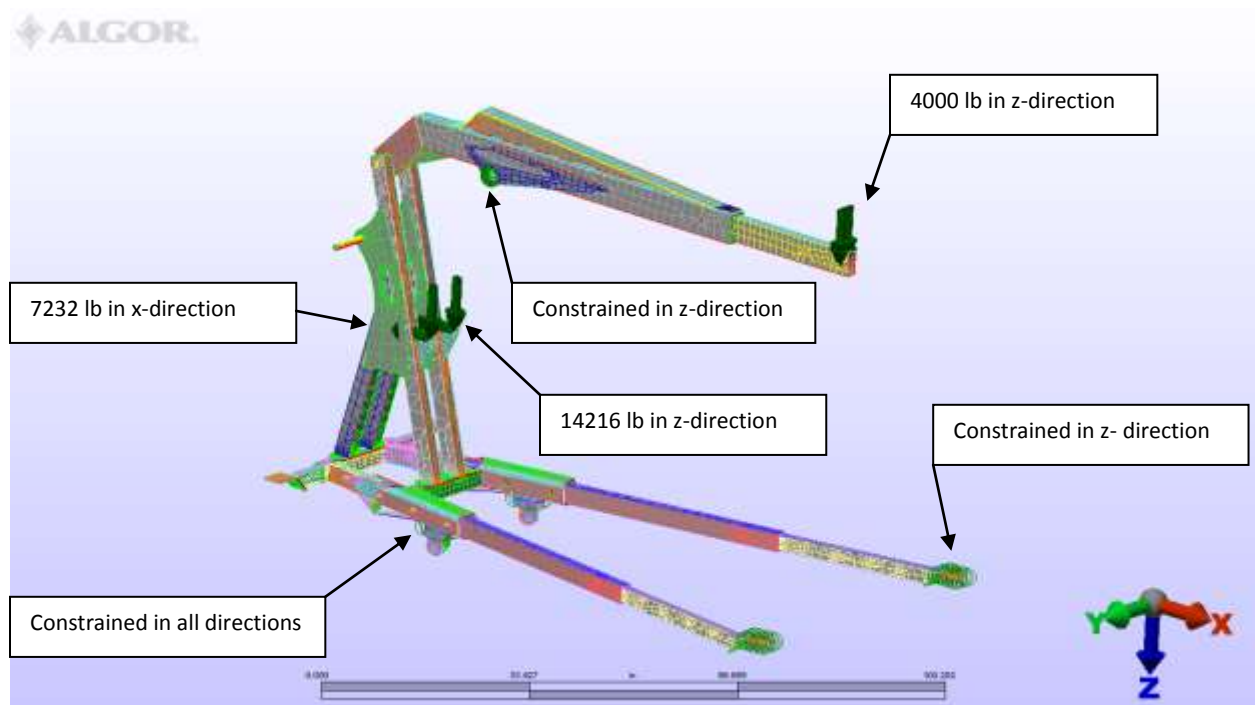


Figure 5.20. Final design FEA setup with normal load on beam.

From this loading case, it was found that the lowerstructure design is sufficient for this load case. By applying the simulated loads to the upper and lowerstructure, and allowing the upperstructure to pivot on the cylinder support, it is concluded that the design is sufficient to hold the load. Figure 5.21 shows this point. Figure 5.22 demonstrates how the design, also, meets our required factor of safety of 2.

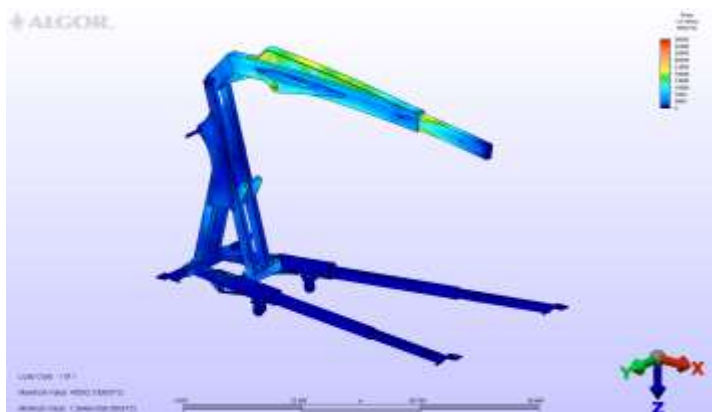


Figure 5.21. Final design FEA stress results with beam.

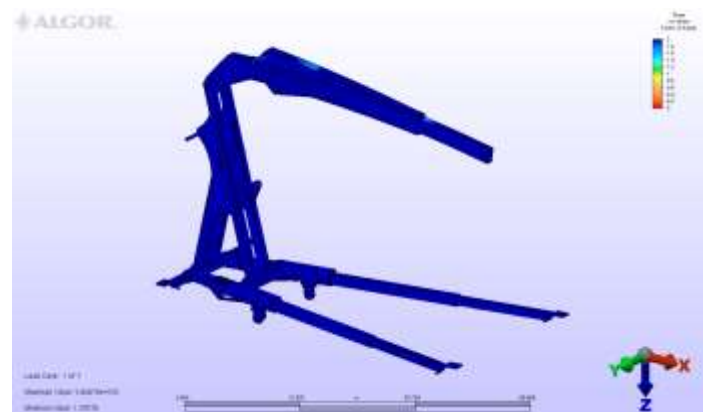


Figure 5.22. Final design FEA factor of safety results with beam.

In the next load case, a load was applied in both the z and x direction. This load simulates a 4000 pound load being applied at a 30 degree angle forward from the z axis. The analysis is done to simulate the hoist picking up a load near a wall that the hoist cannot reach. In this case, the legs of the hoist are resting against an obstacle that does not allow the recommended vertical lifting. Figure 5.23 is the representation of this.

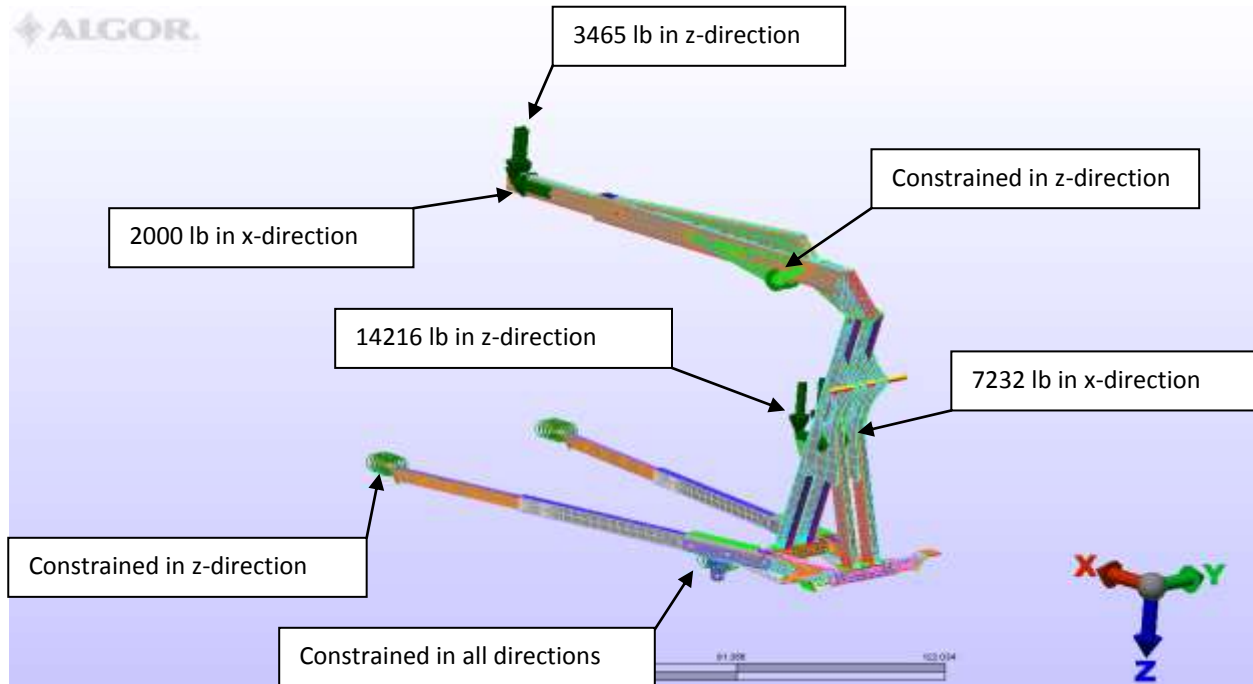


Figure 5.23 Final design FEA setup with forward load on beam.

The results from this analysis are shown in Figures 5.24 and 5.25. The stress produced in the rear supports is slightly higher than expected. From this forward load, it is expected that more load will be transmitted into the rear of the lowerstructure. Once again, the lowerstructure passes this test.

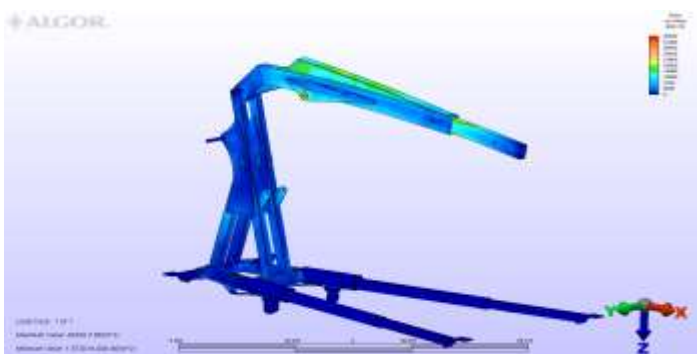


Figure 5.24. Final design FEA stress results with forward load on beam.



Figure 5.25. Final design FEA factor of safety result with forward load on beam.

The final analysis is done to simulate a side load on the engine hoist. In this situation, a 4000 pound load is being hoisted at a 30 degree angle from the z axis towards the y axis. This is a situation where the hoist is not directly underneath the object when lifting, forming a side load on the hoist. In reality, the hoist would probably self align. However, this design is going to account for this problem. Figure 5.26 illustrates the setup.

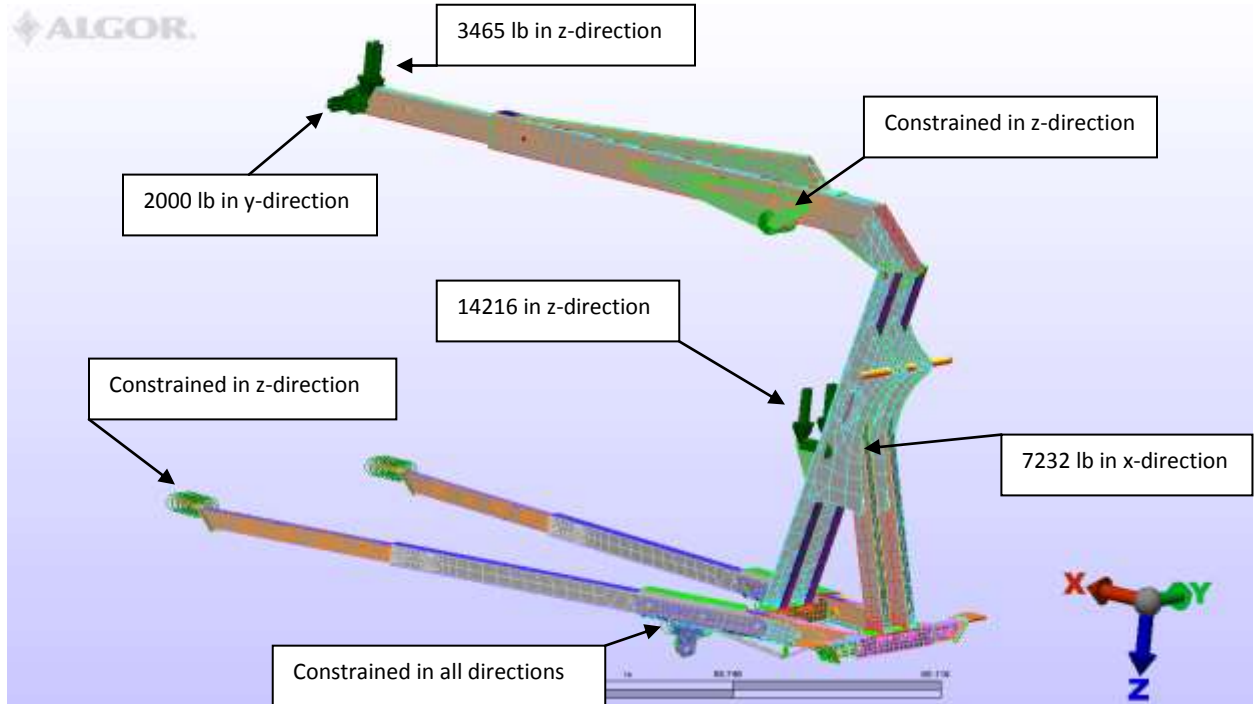


Figure 5.26. Final design FEA setup with side load on beam.

The results are also conclusive that the hoist is structurally sound. The analysis' result is shown in Figure 5.27. The stress produced on the lower structure is higher than any of the other tests. However, it still held its structural integrity. From this point forward, the lower structure design is complete.

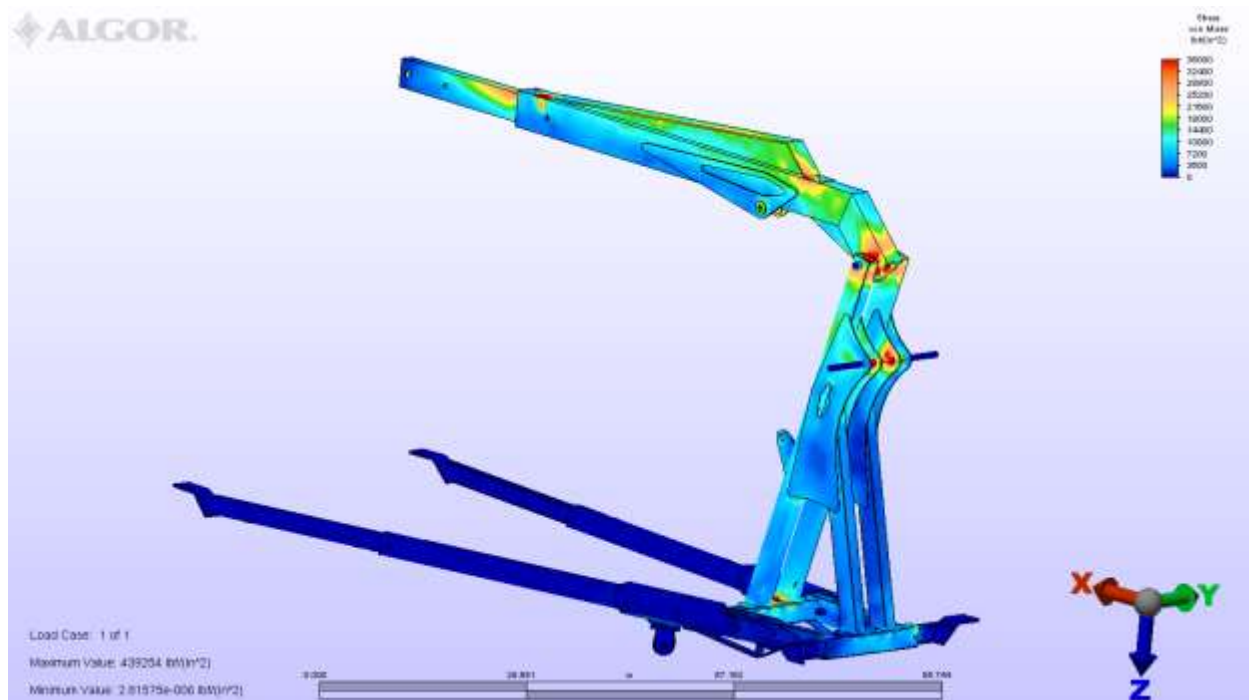


Figure 5.27 Final design FEA stress results with side load on beam.

Although the final design took several iterations, it has come to completion. Within this design, many considerations were taken into account including manufacturability, cost, maneuverability, and ergonomics. The lowerstructure design is composed of only three different sizes of steel tubing and 2 different sizes of sheet metal. They are 2 inch by 3 inch by 0.25 inch tubing, 2 inch by 3 inch by 0.3125 inch tubing, 1.5 inch by 2.5 inch by 0.25 inch tubing, 0.375 inch and 0.25 inch sheet metal. Figure 5.28 shows the location of the sizes. By using these sizes in our application, cost will be lowered.

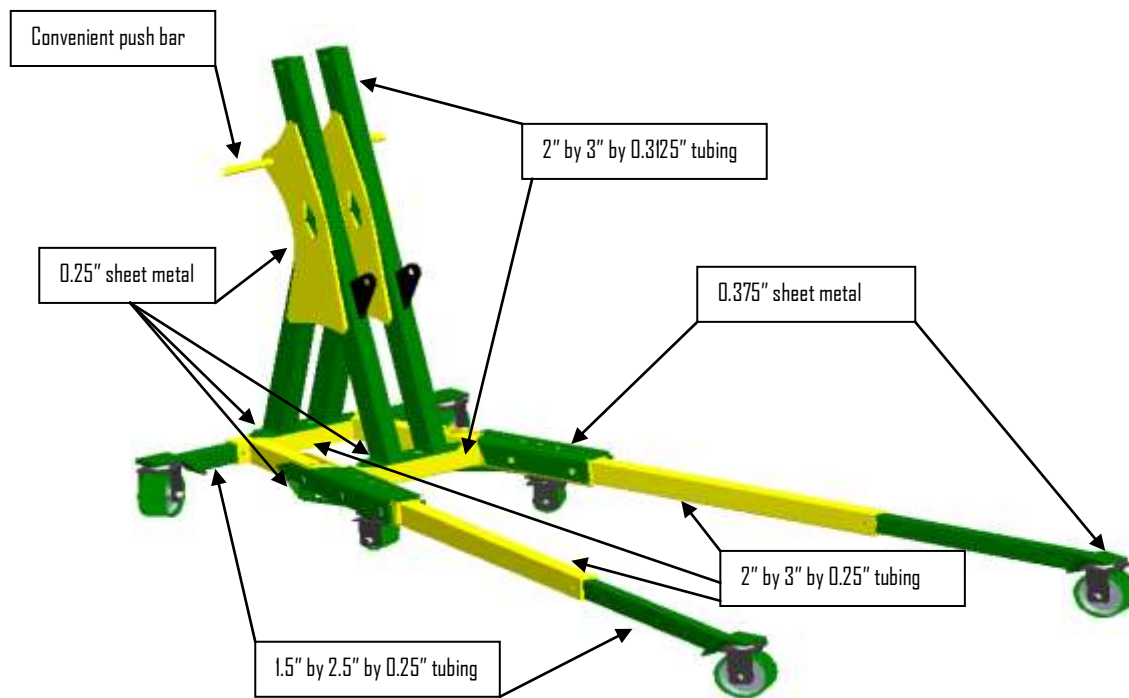


Figure 5.28. Lowerstructure part information.

## 6. DESIGN OF HYDRAULIC POWER

### 6.1 Considerations

The hydraulic power is the heart of the engine hoist. It is the apparatus used to lift the load off the ground and lower it back into the desired location. Hydraulic long ram cylinders are the most commonly used cylinders on an engine hoist. They can range in load capacity from as little as 3 ton to as much as 20 ton. However, there are several alternatives that could be used.

Before exploring alternatives, however, a desired load capacity needs to be chosen. From the calculations, which are found in the Appendix B, it was found that in order to raise a load of 4000 lb at the maximum extension length, at least an 8 ton jack is needed. There are several cylinders that meet this requirement. A pneumatic cylinder, a pneumatic over hydraulic cylinder, a bottle cylinder, a hand pump powered hydraulic cylinder, dual cylinders, and an electrically powered system are all possibilities.

First, a pneumatic cylinder is an air powered cylinder. Figure 6.1 is a pictorial representation of a pneumatic cylinder. In order to lift the load, a very high pressure needs to be applied to the base end of the cylinder. This high pressure is not available in our working environment. After finding this piece of information, the pneumatic cylinder alternative was discarded.



Figure 6.1. Pneumatic cylinder representation.  
(Source: www.airhydraulics.com)



Figure 6.2. Pneumatic over hydraulic cylinder representation.  
(Source: www.northerntool.com)

The pneumatic over hydraulic cylinder brought an interesting idea to the table. With this cylinder, a low pressure air powered cylinder is used to pump the hydraulic cylinder which raises the load. Figure 6.2 is a representation of a pneumatic over hydraulic cylinder. This option would eliminate the need for manual operation and would possibly speed the lifting rate. However, an air hose would need to be connected at all times. With the distances that we are looking at, this alternative does not suit our needs. The air hose would not reach required distances and would always need to be dragged along, adding a safety hazard.

A high capacity bottle cylinder was also considered.

With these cylinders, as seen in Figure 6.3, the extension distance is not suitable for our application. It would need to be positioned very near the pivot pin. The stress would, then, be concentrated directly on the pivot pin and surrounding metal. With the high loads that we are looking at, this one is not suitable for our application.

However, a hand pump powered hydraulic cylinder is. As you can see in Figure 6.4, this pump uses a manually powered pump to extend. With its long stroke length, this cylinder would give the engine hoist a mechanical advantage and would not concentrate the stress in one location.



Figure 6.3. High capacity bottle jack representation.  
(Source: www.northerntool.com)

Dual powered hydraulic cylinders were also considered. In this configuration, two low capacity hand pump powered hydraulic cylinders are placed side by side. This allows for two lower capacity cylinders to be used. However, in order to synch the cylinders, additional equipment will be needed.

One final option was also evaluated: an electrically powered hydraulic system. In this system, an electric motor would be connected to a hydraulic pump. The hydraulic pump would be plumbed to a single-acting hydraulic cylinder. Within the system would be the pump, a directional control valve, a pressure relief valve, a flow control valve, and a cylinder. With this option, the speed of operation would be greatly increased when compared to the previous options. However, two drawbacks exist. The first is the fact that an electrical cord or battery will always need to be present, and the other drawback is cost. With cost being a driving factor, this alternative has been designed, as seen in Figure 6.5. However, this design has not been incorporated into the system.



\*Figure 6.4. Hand pump powered hydraulic jack representation.  
(Source: [www.northerntool.com](http://www.northerntool.com))

A 1.5 gallon per minute fixed displacement pump powered by a 12 Volt DC motor is used. If the pressure relief valve is set to 2000 pounds per square inch, a 3.25 inch diameter cylinder will provide sufficient extension power. The flow control is a very essential piece of equipment during retraction of the cylinder. This will influence the lowering rate. It will not allow the load to crash suddenly to the ground when actuated.

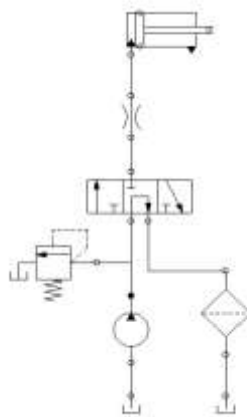


Figure 6.5. Hydraulic system design.

## **6.2 Discussion**

Of the possible alternatives, it is recommended that a hydraulic long ram cylinder be used if the resources are not available to buy a total system. This piece of equipment is virtually the heart the machine. If this piece were to not work, the machine would be incapable of doing any operation. With a total system, operation of the engine hoist would be quick and efficient. However, a tradeoff exists between cost and efficiency. It is recommended that a hand pump powered hydraulic long ram cylinder be used for the initial use. In the future, a total system could be incorporated into this design. Both systems are user friendly and can be interchanged very easily.

## 7. CASTER WHEEL SELECTION

### 7.1 Considerations

The caster wheels are what make The Big Cherry mobile. The casters need to support the weight and be durable enough to be rolled around in a shop. The casters come in two styles, rigid and swivel. For the design six casters are needed; two for the front, two for the middle, and two for the rear. The front casters will be rigid while the rest are swivel.

### 7.2 Exploring Alternatives

The first task in determining caster size was finding out the loadings on each caster. For determining this it was assumed when loaded the rear casters would not be carrying a load. This would put the entire load on the front and middle casters. Setting this up as a statics problem the front casters would have about 1750 pounds each while the middle casters would have about 225 pounds of force each. After looking at our base design we felt the middle casters should have a safety factor of 3. This would make the middle casters at about 700 pounds. The rear casters will be 500 pound casters.

The overall dimensions for the casters need to be in compliance with the lower structure assembly. This meant that the front and rear casters had to be 3 inches higher than the middle casters. The first sizing called for a 7.5 inch front and rear caster and a 4.5 inch middle caster. A 4.5 inch caster that could support the 700 pounds did not exist. We then tried to size the middle caster first. The smallest size caster that could support the weight was a 5 inch caster. This meant the front and rear casters had to be 8 inches tall.

Two companies that produced heavy duty casters were Hamilton Casters and Shepherd Caster Corporation. After researching, Hamilton Casters was more inexpensive than Shepherd Caster Corporation and Mr. Ryan Nagel has previous experience with the company.

### 7.3 Discussion

The following table 7.1 represents the casters specified for The Big Cherry.

Table 7.1 Caster Specifications

<b>Placement</b>	<b>Rigid/Swivel</b>	<b>Part #</b>	<b>Price</b>
Front	Rigid	R-A-63PH	\$75.30
Middle	Swivel	S-A-63RH	\$144.66
Rear	Swivel	W-320-P-3/4	\$32.04

## 8. COST ESTIMATE

For the cost estimate, the materials were priced out through local businesses, factories, or outlets close to Fargo. The least cost scenario was used for the cost estimate. Since the hydraulics have not been selected, a \$300 allowance was used for estimation. An estimation of \$50 was used for paint also. The contingency allowance of \$200 is used for any unexpected expenses or increased prices that may occur. Table 8.1 represents the cost estimation.

Table 8.1 Cost estimate for the prototype.

<b>Big Cherry</b>					
<b>Cost Estimate</b>					
	Steel Size	Quantity	Price	Supplier	Location
<b>Lowerstructure</b>					
--Tube	3" x 2" x 0.3125"	24'	\$ 345.00	McNeilus	Fargo
--Tube	3" x 2" x 0.25"	139"	\$ 92.67	Mid America	Fargo
--Tube	2.5" x 1.5" x 0.25"	92"	\$ 57.00	Discount Steel	Twin Cities
--Pipe	1"	21"	\$ 15.00	Mid America	Fargo
<b>Upperstructure</b>					
	5" x 3" x 0.5"	78"	\$ 167.00	Discount Steel	Twin Cities
	4" x 2" x 0.3125"	36"	\$ 44.50	Discount Steel	Twin Cities
<b>Sheet Steel</b>					
	1/4"	1780 sq. in	\$ 135.00		Fargo
	3/8"	1570 sq. in	\$ 171.00		Fargo
<b>Additional Cost</b>					
	--cutting and bending		\$ 300.00		Fargo
<b>Pins</b>					
	1" @ 6"	9	\$ 90.00	Northern Tool	Fargo
<b>Casters</b>					
	8" height Rigid	\$75.30 x 2	\$ 150.60		
	5" height	\$32.04 x 2	\$ 64.08		
	8" height Swivel	\$144.66 x 2	\$ 289.32		
<b>Hydraulics</b>					
			\$ 300.00		
<b>Paint</b>					
			\$ 50.00		
<b>Contingency</b>					
			\$ 200.00		
		<b>Total</b>	<b>\$ 2,471.17</b>		

## **9. RECOMMENDATIONS AND OPERATING PROCEDURE**

In order to insure safe operation once fabrication is completed, it is necessary to test this engine hoist before full operation. It should be tested to insure structural soundness. The hoist should be put through normal cyclic loadings with a few irregular load cases.

A regular load case scenario would be lifting the lab instructional equipment, such as the Cummins diesel engine or tractor transmission, onto the back of the Department's Ford F-250 pickup. To simulate an irregular load, a heavy class aid should be positioned so the hoist cannot lift vertically, such as close to the wall. This will test the effects of a forward and twisting load.

During transfer, it is important that the load is lifted to the lowest level possible, moved to its desired location, and raised up at that point. Moving a load at a high height is very unsafe and NOT recommended. The load should not be lowered at a high rate. A smooth controlled drop is recommended. This may be practiced with a lighter load before using The Big Cherry on heavy equipment. All pins need to be installed before operation.

If the hoist passes these tests, normal operation may resume. However, if a failure were to occur, further investigation must be done before further operation or repairs are made. Investigation may include material analysis, failure analysis, further FEA, and/or weld failure analysis. It is up to the operator to keep this hoist in top shape.

## **10. CONCLUSION**

The Big Cherry has been designed to meet the required objectives. A lifting device capable of hoisting 4,000 pounds at 8 feet length has been designed. After the cost analysis was conducted, it was estimated to be \$2,475. If resources are available, a prototype should be built. Engineering analysis has been performed on The Big Cherry. The design is both structurally sound and manufacturable.

## 11. ACKNOWLEDGEMENTS

### **Mr. Elton Salseng**

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--Cooperator

### **Mr. James Moos**

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--Cooperator

### **Mr. Tom Bon**

--Instructor, Agricultural and Biosystems Engineering Department

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### **13. APPENDIX INDEX**

Appendix A

-Design Drawings

Appendix B

-Sample Calculations

# Appendix A

# Appendix B