MAE 4344 – Senior Design

Parallel Squeeze Hay Bed: Design Analysis and Optimization

Design Project Final Report

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1.0 Introduction and Background

The Parallel Squeeze Arm Hay Bed teams consists of Jacob Brown, Bailey Bruns and Amber Kirkland. All three students are senior mechanical engineering students at Oklahoma State University and are completing their senior design project under the supervision of Dr. Robert Taylor.

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2.0 Problem Statement

The core of this project is a full engineering analysis and optimization of the HosWel parallel squeeze hay bed manufactured by Better Built Enterprises. Firstly, the design team will optimize each element of the hay bed and moving arm assembly by considering various loading cases in order to minimize weight and material use. Secondly, the team will assess the manufacturing process of each element in order to further reduce the cost of production. Finally, the team will develop a full manufacturing drawing package – accompanied with all necessary engineering calculations – for the optimized hay bed design.

3.0 Deliverables

- Full manufacturing drawing package for Squeeze Arm Haybed
- All calculations supporting engineering analysis and optimization
- Recommendations for any design and manufacturing process changes
- Justification for all design and manufacturing process optimization

4.0 Problem Approach and Work Completed

4.1 CAD Modeling

Initially, the client provided the design team with a set of 2D CAD drawings as well as a flash drive of 3D Solidworks models. The 2D CAD drawings provided by the client have been scanned and organized according to part number. Each file name begins with the part number followed by a brief description of the part's function. Additionally, these part numbers have been referenced in our new, updated drawings for easy referencing between the old and new drawing sets.

The Solidworks models provide to the design team were of the client's "spike bed" and were created by a team from SWOSU. After working through a number of the provided drawings, it was determined that many of the files either do not function properly or are in need of adjustment in order to be used as manufacturing drawings. Nonetheless, some of these models – such as the frame model – were useful to the design team due to similarities between the spike bed and the hay bed. Furthermore, because of the similarities between this project and the spike bed project completed by our peers, it was agreed that the two teams would collaborate on modeling parts that are the same for both the hay bed and the spike bed such as the headache rack, bumper, and side skirts. Thus, our team produced 3D models and drawings for the hay bed frame, headache rack, and squeeze arms while the responsibility for our bumper and side skirts was assigned to the spike bed team.

Frame Modeling

The frame was modeled using the given Solidworks model from the spike hay bed. This model did not have consistent C-channels for the cross member construction and individual pieces were not fixed to a reference piece. In addition, the C-channels used for the cross members had the same cross-section as the C-channels for the main frame. A lighter C-channel is used for the cross member pieces and therefore the cross-section sketch was altered. The new C-channels were then attached to the main frame runners and fixed in place.

Headache Rack

The 3D model for the headache rack was developed using the 2D paper drawings provided by the client. Using the sheet metal tool in Solidworks, each of the components that had specification in the paper drawings were modeled as bent sheets of 12-gauge steel. However, some of the parts, such as the inner supports for the louvers and the compartment doors, did not have drawings. Therefore, these parts were neglected in the final drawing package, but were modeled in the 3D headache rack assembly for aesthetic purposes. Additionally, the provided 2D drawings for the bent sheet metal parts lacked information regarding the specific bending parameters used. Thus, some of the dimensions between our new model and the old models differ slightly – usually less than one-tenth of an inch, though.

Squeeze Arms

The squeeze arm portion of the haybed has never been modeled previously and was modeled strictly from the 2D CAD drawings originally provided by our client. The dimensions of the parts given fit together in the assemblies very well, however, many of the drawings do not reflect the design changes that our client has made to newer beds over the years. The team has attempted to correct the designs as changes that were noticed, but many of the new dimensions could not be verified due to time constraints. Estimates were made in the drawings and will require verification before submission as a final manufacturing package. There are likely many more design changes still to be found.

The team decided to separate the 5000-1 assembly into two separate assemblies (5000-1 & 5000B-1) due to the fact that the Cross Arm Pivot Mount and all related parts do not change according to bed size, however, the hitch plate and related parts do change according to bed size. The Cross Arm Pivot Mount Assembly and the Hitch Plate Assembly have been modeled and drawn separately in their own sub-assemblies. Assemblies 6500 and 7000 were modeled as a single large assembly instead of multiple smaller assemblies due to the fact that those assemblies were modeled as a single large assembly and time constraints prevented the team from remodeling into smaller assemblies.

Some of the parts that have undergone more substantial design changes are is 6507, 6505, 7005 and 6503. 6507 is a cover on the outer beam (6501) from the final arm assembly (6500) that moves with the arms as they squeeze in and out. The new design covers has been extended on one size and are now bolted to the arms directly. 6505 are the shims that are welded to the outside of the outer beam (6501). The shims bridged the gap between the cross tube (6x6) and the outer beam (5x5) and allowed the Client to utilize standardized rectangular tubes. Those shims have been changed from bent strips that cover the corners to flat strips of 10 and 7 Ga steel. They are easier to manufacture and require less work than the previous designs. The shims have been dimensioned in the drawings according to the client's supplied specifications. The spinner mounting tube (7005) and the arm (6503) have both been redesigned to accommodate a square attachment instead of the previous circular tube design. Our client stated that they made the change to decrease the amount of stress on the pin that holds the spinners to the spinner mounting tube together. The dimensions that were used have not been verified according to the newer existing models, but have been estimated according to the geometry and space available. As a result a square cut was made in the end of the arm with the size of 2.375 x 2.375 inches and the spinner mounting square tube was designed at 2.5 x 2.5 inches with a 0.5 inch thickness. The remaining parts of the spinner assembly required little adjustment to fit the new model. Any other adjustments made to parts and assemblies provided are noted within the supplied drawings and in the team logbooks.

4.2 Stress Calculations

As discussed in this project's proposal, the team analyzed the hay bed for the loads experienced when lifting or hauling large hay bales and towing rear-hitch or gooseneck trailers. The following assumptions were made during this analysis:

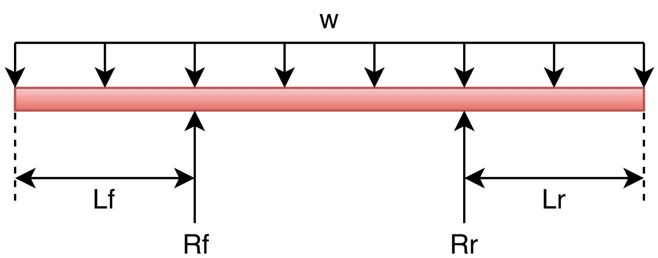
- *Material type: ASTM A36 Steel*
    - *Tensile Yield Strength: 36 ksi*
    - *Compressive Yield Strength: 22 ksi*

4.2.1 Hauling Stress Calculations

In order to analyze the hay bed while hauling hay bales, the design team has completed a full stress analysis of the frame of the hay bed. In order to begin, the frame was divided into two main categories for the analysis: frame runners and cross members. The maximum hauling weight was chosen to be 4400lbs, and the self-weight of the bed was assumed to be 2000lbs.

Frame Runners – 2 Supports

A two-dimensional loading model was developed to analyze the two main frame runner C3x6.0 channels irons from which the entire hay bed receives support. In this first case, it was assumed there were two attachment points between the hay bed and the frame. In order to load the beams, it was assumed that the 4400lbs load and 2000lbs self-weight of the bed were evenly distributed along the length of the two 100 inch beams – resulting in a uniformly distributed load of 32lb/in per beam. The free body diagram for this beam loading model is shown below:



FBD 1: 2 Support Case

Using the method of singularity functions [1], the shear force and bending moment across the length of the beam (graphical examples shown in section 8.1) were evaluated in an excel spreadsheet based upon the shear and moment singularity functions listed below:

$$V(x) = R_f < x - L_f >^0 - w < x >^1 + R_r < x - (L_t - L_r) >^0 \tag{1}$$

$$M(x) = R_f < x - L_f >^1 - \frac{w}{2} < x >^2 + R_r < x - (L_t - L_r) >^1 \tag{2}$$

Where:

R_f = support reaction at the front of the beam
R_r = support reaction at the rear of the beam
L_t = total length of beam
L_f = location of front support reaction (measured from front)
L_r = location of rear support reaction (measured from rear)
x = position on beam (measured from front)
w = distributed load

Initially, the support locations L_f and L_r were assumed to be 5 inches from the front and 12 inches from the rear, respectively. These locations are somewhat variable, however, as the hay beds are mounted to truck frames by HosWel dealers, and not BBE.

After calculating the shear force and bending moment along the span of the hay bed using equations (1) and (2), the obtained values of shear and moment were used to calculate the shear stress and bending stress experienced in the beam along its span. The equations for transverse shear stress and normal bending stress are as follows:

$$\tau = \frac{VA'\bar{y}}{Ib} \tag{3}$$

$$\sigma = \frac{My}{I} \tag{4}$$

Using the cross sectional properties listed in section 10.3, the shear and bending stresses were evaluated at three points within the C-channel cross section. (For specifics on point locations, see section 10.3) From this, the transverse shear stress and the normal bending stress were calculated using equations (3) and (4), respectively. Additionally, in order to display the relatively negligible effects of transverse shear stress, the maximum combined shear stress was also calculated at each of the three cross sectional points using the following equation:

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} \tag{5}$$

The values obtained from the stress calculations are summarized in the following table:

Maximum Stresses		
Point A (Center of Cross Section)		
Transverse	1597	psi
Normal	0	psi
Max Shear	1597	psi
Point B (Just Before Flange Thickens)		
Transverse	453	psi
Normal	11332	psi
Max Shear	566	psi
Point C (Edge of Cross Section)		
Transverse	0	psi
Normal	18994	psi
Max Shear	9497	psi

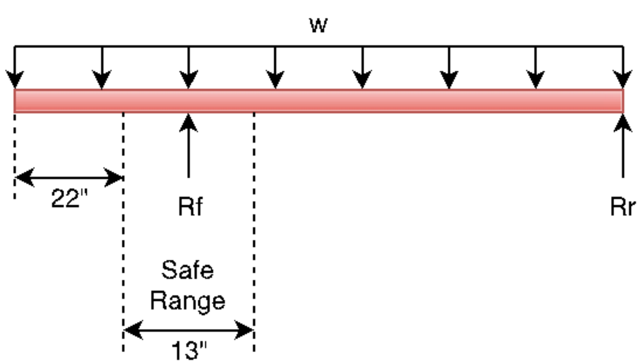
Table 1: Max Stresses - 2 Support Case

Using the maximum stress in the table above, the factor of safety was calculated by comparing the maximum stress against the compressive yield strength of the material.

$$n = \frac{yield\ stress}{max\ stress} \tag{6}$$

From this, the factor of safety was found to be 1.16 in the main C-channels – although this number is low, this is a conservative estimate and can be improved by moving the locations of the supports ($L_f = 15in$ and $L_r = 12in$, for instance, would give a safety factor of 1.65). Furthermore, as will be examined in the next section, adding a third support to these main C-channels will also increase the safety factor.

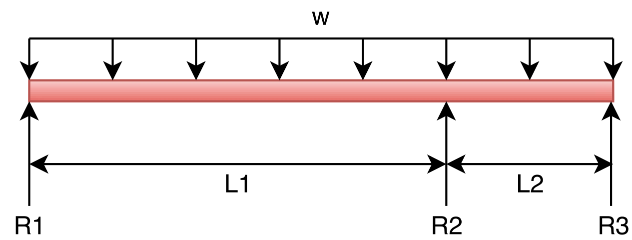
Additionally, it seemed reasonable to represent the support locations in terms of a range where the supports may safely be placed. Assuming the rear support is located at the end of the beam, the diagram below displays the "safe range" in which the front support may be placed while maintaining a factor of safety of 1.5.



FBD 2: Safe Range of Front Support

Frame Runners – 3 Supports

Since it is also common to attach these hay beds to the truck frame with three supports instead of two, analysis was also performed for this case. The free body diagram below is representative of this three-support case.



FBD 3: 3 Support Case

As can be seen from the free body diagram, this case is statically indeterminate. Thus, deflection tables were used in order to solve for the reactions [2]. The equations necessary for solving the reactions are shown below:

$$R_1 = \frac{M_1}{L_1} + \frac{wL_1}{2} \tag{7}$$

$$R_2 = wL_1 + wL_2 - R_1 - R_3 \tag{8}$$

$$R_3 = \frac{M_1}{L_2} + \frac{wL_2}{2} \tag{9}$$

$$M_1 = -\frac{wL_2^3 + wL_1^3}{8(L_1 + L_2)} \tag{10}$$

As with the two-support case, singularity functions were developed in order to obtain the shear force and bending moment along the span of the beam.

$$V(x) = -w < x >^1 + R_1 < x >^0 + R_2 < x - L_1 >^0 + R_3 < x - (L_1 + L_2) >^0 \tag{11}$$

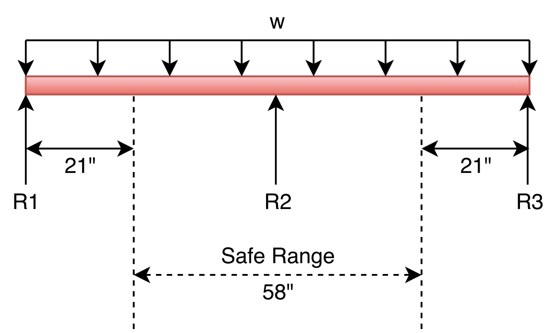
$$M(x) = -\frac{w}{2} < x >^2 + R_1 < x >^1 + R_2 < x - L_1 >^1 + R_3 < x - (L_1 + L_2) >^1 \tag{12}$$

Using equations (3), (4), and (5), the stresses were solved. R_2 was assumed to be located at the center of the span of the beam, and the maximum stresses found in the cross section are summarized in the following table:

Maximum Stresses		
Point A (Center of Cross Section)		
Transverse	1182	psi
Normal	0	psi
Max Shear	1182	psi
Point B (Just Before Flange Thickens)		
Transverse	335	psi
Normal	4323	psi
Max Shear	2187	psi
Point C (Edge of Cross Section)		
Transverse	0	psi
Normal	7246	psi
Max Shear	3623	psi

Table 2: Max Stresses - 2 Support Case

Placing the maximum stress into equation (6), a safety factor or 3.04 was found using this configuration with R_2 placed at the center of the beam. Since this safety factor is above the target safety factor of 1.5, the team investigated the effect of moving the middle support away from the center and the range of placement that would maintain the desired safety factor. After recalculating with various positioning, it was concluded that the middle support had a 58-inch "safe range" where the target safety factor was kept. Of course, the closer the support is to the center, the higher the safety factor will be, but this information allows for some flexibility during installation. The following figure is representation of the "safe range" of the center support:



FBD 4: Safe Range of Middle Support

Cross Members

In order to calculate the stresses experienced in the cross members of the frame from hauling heavy hay bales, the loadable area of the hay bed was divided into ten areas based upon the geometry of the cross members with respect to the loadable area. (The division of areas can be seen graphically in section 8.2.) The load from the 4400lbs hay bales and 2000lbs self-weight was divided by the loadable area in order to determine the average pressure that would be felt upon the loadable area. From geometry, the total loadable area was determined to be:

$$A_{total} = 6628\ in^2$$

Therefore,

$$P = \frac{Load}{A_{total}} = \frac{(4400 + 2000)\ lbs}{6628\ in^2} = 0.96\ psi$$

Next, each of the ten areas assigned to the ten cross members were multiplied by the pressure in order to determine the load imparted upon each of the ten cross members. Then, in a process similar to that with the frame runners, the load was divided by the length of the beam in order to determine the distributed load imparted upon each of the ten cross members. Finally, using the same type of singularity functions and the stress equations (with updated parameters) as with the frame runners, the factor of safety was calculated for each of the C3x4.1 channel irons used as cross members (cross sectional properties listed in section 10.3). The following tables summarize the calculations of each step and the final resulting safety factors:

Areas [in^2]	
1	155.0
2	311.9
3	968.0
4	1172.3
5	1357.8
6	860.9
7	806.0
8	528.9
9	155.0
10	311.9

Load Fraction [lbs]	
1	149.7
2	301.2
3	934.7
4	1132.0
5	1311.1
6	831.3
7	778.3
8	510.7
9	149.7
10	301.2

Load Distribution [lbs/in]	
1	6.4
2	12.9
3	11.5
4	14.0
5	16.2
6	15.1
7	14.2
8	6.3
9	6.4
10	12.9

Safety Factors	
1	13.95
2	6.93
3	8.21
4	6.78
5	5.85
6	14.07
7	15.03
8	15.03
9	13.95
10	6.93

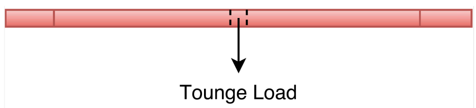
4.2.2 Towing Stress Calculations

This hay bed has attachment points for towing both rear hitch and gooseneck trailers. The assumptions made about towing capacities are as follows:

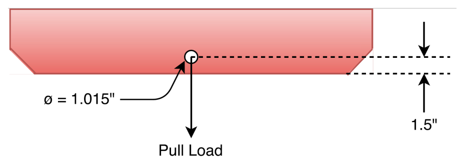
- *Rear hitch towing capacity: 15,000lbs*
- *Gooseneck towing capacity: 30,000lbs*

Rear Hitch

To determine if the hitch is able to support a suggested trailer pull load of 15,000 pounds, the bending and shear stress on the hitch attachment section were analyzed.



FBD 6: Rear Hitch – Side View



FBD 5: Rear Hitch - Top View

According to both Ford and Chevrolet hitch design specification sheets, between 10-15% of the pull load can be estimated as the downward load on the hitch. The cross-sectional area of the hitch plate attachment is therefore in tension from this downward load. Applying the normal stress equation:

$$\sigma = \frac{F}{A} \quad (13)$$

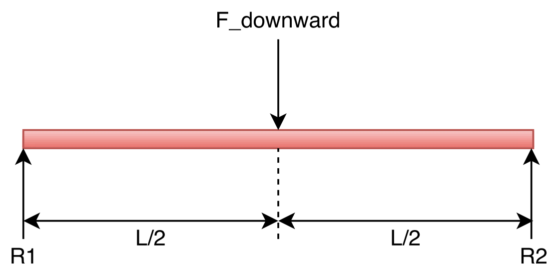
Where F is 15% of the pull load and A is the cross-sectional area of the attachment member, the safety factor can be determined. This analysis resulted in a safety factor of n = 36. In order to determine the safety factor for the horizontal pull load, the tear out stress in the ball receiver was calculated using the same method as will be discussed in detail in section 4.2.3 regarding the ears on the lifting mechanism. The diameter of the receiver hole and the thickness of the plate were given in the client's binder of drawings. This analysis resulted in a safety factor of n = 1.827. For both downward load and pull load, according to the calculated safety factors the bed is more than capable of towing a 15,000-pound trailer. The results are summarized in the table below.

Tongue Weight		Pull Weight	
Tongue Load	2250lbs	Pull Load	15000lbs
Width	3in	Hole Diameter	1.015in
Thickness	0.75in	Thickness	0.75in
Area	2.25in^2	Area	0.761in^2
Stress	1ksi	Stress	19.7ksi
Factor of Safety	36	Factor of Safety	1.8

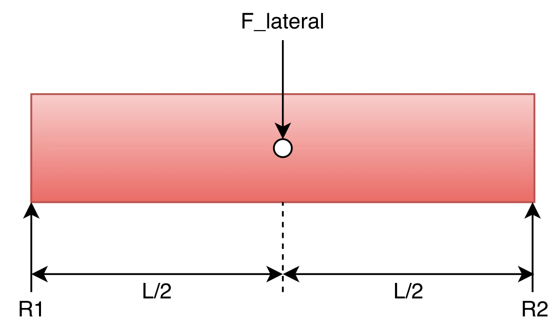
Table 4: Tounge Weight Safety Factor *Table 4: Pull Weight Safety Factor*

Gooseneck

The gooseneck trailer mount on the frame is one C6x13 channel iron positioned in a horizontal direction (such that it forms an "n" shape). Since the trailer will produce both downward and lateral forces on the channel iron, it must be analyzed for both cases. The following free body diagrams illustrate the forces imparted upon the gooseneck mount:



FBD 8: Gooseneck Trailer Mount - Side View



FBD 8: Gooseneck Trailer Mount - Top View

It was assumed that during dynamic movements (such as hitting a pothole/hard braking) the trailer might experience one full G of acceleration in both the downward and lateral directions. (For clarity, this would be equivalent to 2 times the normal gravity in the downward direction). Furthermore, it was assumed that the downward force – the king pin weight – from the trailer was equal to one-fifth of weight of the trailer. That is to say, for a 30,000lbs trailer:

$$F_{downward} = (0.2)(30{,}000lbs)(2\ G) = \ 12{,}000lbs$$

$$F_{lateral} = (30{,}000lbs)(1\ G) = \ 30{,}000lbs$$

Using the following singularity functions, the shear force and bending moments were solved for (Note: the singularity functions remain the same for both the downward and lateral directions)

$$V(x) = \ R_1 < x >^0 - F < x - \frac{L}{2} >^0 + R_2 < x - L >^0 \tag{14}$$

$$M(x) = \ R_1 < x >^1 - F < x - \frac{L}{2} >^1 + R_2 < x - L >^1 \tag{15}$$

Again using the stress equations (3-5), along with the cross sectional properties found in section 10.3, the safety factors were calculated. The safety factors were found to be 0.13 in the downward direction and 0.46 in the lateral direction. Thus, this gooseneck mount is not suited for the target towing capacity of 30,000lbs. Rather, the safety factor approaches 1.5 only when the trailer weight is reduced significantly. Although the calculated safety factors are very low compared to the original target number, the C-channel being used is inadequate for such large loading conditions. Thus, the maximum pulling and king pin weights that will maintain a safety factor of 1.5 are summarized below:

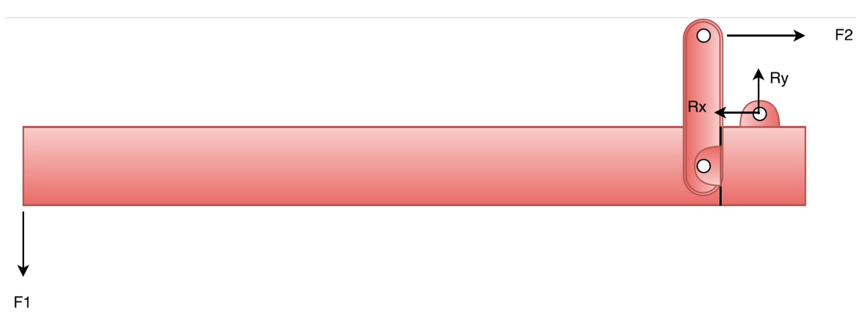
Pulling Weight	King Pin Weight
Safety Factor = 1.5	Safety Factor = 1.5
Dynamic Acceleration = 1G	Dynamic Acceleration = 1G
Max Bending Stress = 14.6ksi	Max Bending Stress = 14.6ksi
Yield Stress = 22ksi	Yield Stress = 22ksi
Pulling Capacity = 9300lbs	King Pin Capacity = 525lbs

Table 5: Gooseneck Trailer Towing Capacities

4.2.3 Lifting Arm Stress Calculations

Ears and Pins

For the parallel squeeze arm analysis, we established a free body diagram and loading model in order to solve for the reactions on the hinge that attaches to the frame.



FBD 9: Lifting Arm and Linkage Model

The first assumption made was that the weight of the bale is equally split between the two arms. Therefore, half the weight of the bale is the total load in the first loading model that assumes normal gravity conditions and the arm is horizontal holding the bale. An ear-link assembly is used to attach the arm to the cross-tube member which is attached to the frame with an ear hinge joint. This hinge joint is what holds the load of the bale and therefore loading analysis was centralized at this point. A pin is slid through two ears on the frame and a single ear on the cross-tube member and then welded in place. For the sake of these calculations, this pin is considered a bolt or a "non-permanent fastener" according to *Shigley's Mechanical Design [1]*.

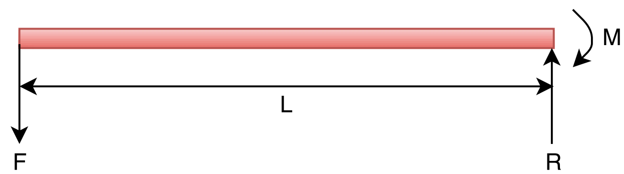
To confirm the integrity of the design, bending, shear and edge stresses were all calculated according to stress equations given in chapter 8 of *Shigley's*. The safety factors for each of the scenarios were determined by dividing the yield strength of the material, A-36 steel, by the calculated stress.

Failure by bending of the pin is determined with the bending stress equation: $= \frac{M}{I/c}$, where the bending moment is M=Ft/2, where F is the shearing force and t is the grip of the pin, and I/c is the section modulus for the pin. Failure of the pin by pure shear is determined by finding the shear stress in the pin with $= \frac{F}{A}$, where A is the cross-sectional area of the pin. Rupture of the plate by pure tension is found by determining the normal stress in the plate with $= \frac{F}{A}$, where A is the net cross-sectional area of the plate. Edge stress or tear-out is estimated by ensuring the pin is at least 1.5 diameters away from the edge of the plate and is also calculated by $\sigma = \frac{F}{(W-d)t}$, where W is the smallest width from the center of the pin to the edge of the plate, d is the diameter of the pin, and t is the plate thickness.

First the reaction forces at the ear were determined by summing the moments around the center of the pin. All lengths were measured on the model bed available in Elgin and are subject to slightly change depending on the type of bed desired by the customer (i.e. standard or extended). The grip of the pin is the total thickness of all ears under loading. The calculations are given in section 10.4. Under the given loading conditions, the pin and ear assembly is more than capable of handling the load of a one bale at both 0G and 1G conditions. As shown in the calculations, the critical element is the potential for horizontal tear out of the pin due to the hydraulic reaction force.

Main Arms

Continuing with the lifting analysis, the bending in the arms was calculated as well. A free body diagram, along with its singularity functions, is shown below for one of the arms. This is a worst-case scenario, as the bending moment in the arm will be the largest for this horizontal orientation.



FBD 10: Main Arm Loading Model

$$V(x) = -F < x >^0 - R < x - L >^0 + M < x - L >^{-1} \tag{16}$$

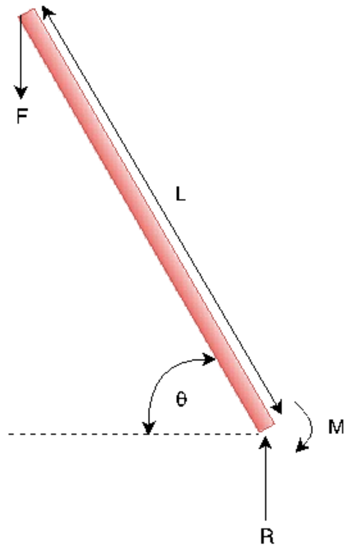
$$M(x) = -F < x >^1 - R < x - L >^1 + M < x - L >^0 \tag{17}$$

Using the cross-sectional properties listed in section 10.3, the stresses were calculated using equations (3-5) and various loading conditions in order to find the maximum lifting weight the arms can achieve while maintaining a factor of safety of 1.5. From the calculations, it was concluded the arms have a maximum lift weight of 1800lbs.

Additionally, as it seems like a common practice, the team investigated the effect of driving with a hay bale hanging from the arms. Using 1800lbs (the max rated static lift weight), and a dynamic acceleration of 1G, the team calculated the angle at which the arm needs to be raised in order to safely transport the bale under the given conditions. The singularity functions and free body diagram and for these calculations are shown below.

$$V(x) = -Fcos(\theta) < x >^0 - Rcos(\theta) < x - L >^0 + M < x - L >^{-1} \tag{18}$$

$$M(x) = -Fcos(\theta) < x >^1 - Rcos(\theta) < x - L >^1 + M < x - L >^0 \tag{19}$$



FBD 11: Main Arm at an Angle

Again, using the same process as before, the maximum stresses where found at various angles of inclination until a safety factor of 1.5 was achieved. From the calculations, the angle of inclination must be greater than or equal to 60^o in order to meet the desired level of safety. The overall results of the bending arm calculations are summarized in the table below:

Static	Dynamic
Safety Factor = 1.5	Safety Factor = 1.5
Dynamic Acceleration = 0	Dynamic Acceleration = 1G
Angle of Inclination = 0°	Lifting Load = 1800lbs
Yield Stress = 22ksi	Yield Stress = 22ksi
Lifting Capacity = 1800lbs	Angle of Safe Transport = 60°

Table 6: Main Arm Static and Dynamic Results

5.0 Evaluation of Design

As mentioned and tabulated above, one significant conclusion reached is regarding the safety factors of the components of the bed. From the safety factors the team calculated, some items, such as the cross members of frame, are vastly over-built, while other components are undersized for the client's desired loading capabilities. A safe, but economical, design according to the client should have a safety factor of near 1.5. Therefore, component sizes or loading conditions must be reevaluated in order to have more realistic safety factors across all of our analyzed components.

In the hauling considerations, the critical components are the frame runners. Under the loading conditions give, the stress calculated on the frame runners produced extremely large safety factors. Therefore, a lighter C-channel could be used which would reduce both weight and cost of the manufacturing. It is not recommended that too many of the frame runners are removed in order to maintain the rigidity of the bed liner.

Between the two towing assemblies, the gooseneck mount proves to be the critical component. With the tools available to calculate the stress induced on this component, as explained above in section 4.2.2, a trailer of only 9300lbs is able to be towed. This is considerably lower than the requested towing capacity of 30,000lbs. However, this mount connects to the frame and therefore the load from the trailer might be distributed across the frame. With a more complex analysis method such as Finite Element Analysis in Solidworks, the load distribution through the frame could be analyzed and a safety factor could be calculated. A recommendation is made to further analyze this component to determine the maximum load the trailer is capable of hauling.

The critical element of the lifting mechanism is the rectangular tube lift arm. This element, as explained in section 4.2.3, fails the safety factor requirement under the common specifications of holding a bale at past a certain angle while potentially driving over a pothole in a pasture. A recommendation is made to either use a slightly thicker tube or to strengthen the supports at the connection point.

6.0 Future Work

With the deliverables for this project completed, suggestions regarding work for a future team are outlined here:

The Client proposed that the frame be redesigned based on the example of a competitor's bed. Either a completely new frame would be designed or the existing frame would be changed based on the suggestions made by the team. Both designs should be able to handle a larger towing load, weigh less, cost less or a combination of these three.

The Client also proposed redesigning the link assembly. Using a three-point system would reduce the number of moving parts, creating a more efficient system. While the current link assembly is more than capable of supporting the given load, a three-point system would smooth the movement.

7.0 Remaining Design Decisions

The recommendations for optimization will be the most important remaining decisions for the team. These recommendations will be based on the final analysis of the hay bed including the frame and squeeze arms. The design team will make recommendations on how to improve the loading capabilities of the hay bed. Regarding the lifting capabilities of the arms, we will also make sure that the hydraulic cylinders are properly sized to generate the amount of force necessary to lift the bale.

8.0 Revised Budget Requirements with Summary Table

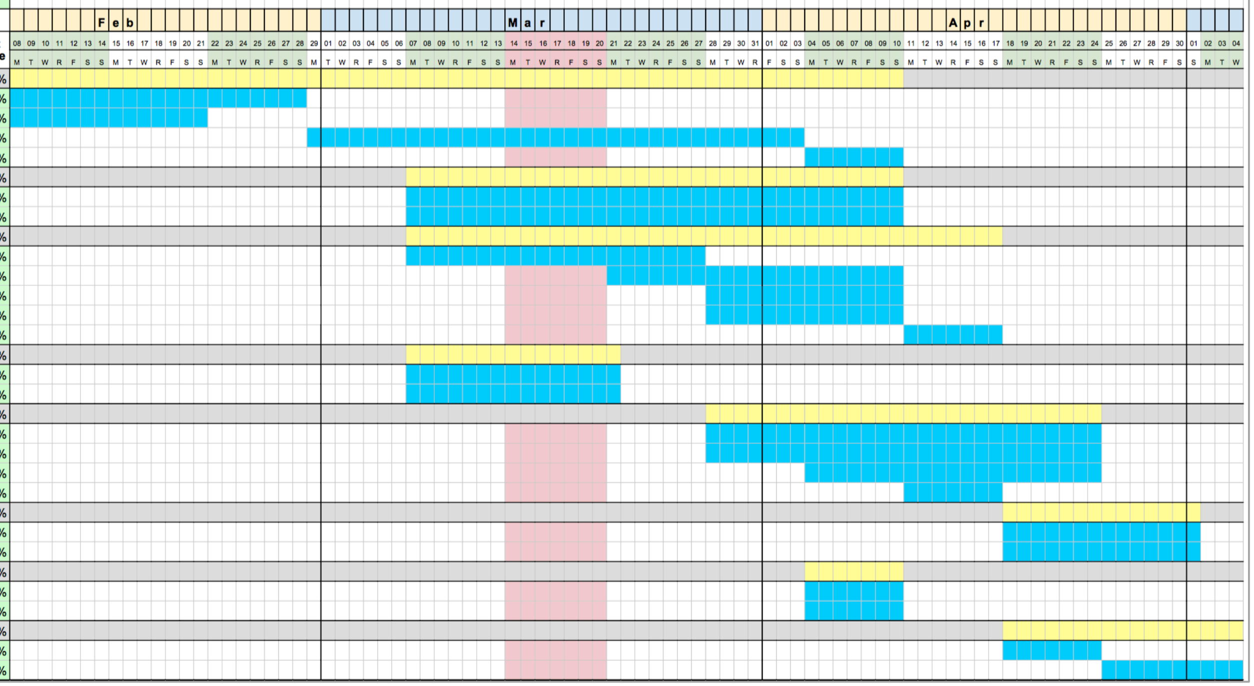
The team made one trip to Elgin, OK and had no additional expenses. As such, the updated itemized budget requirements are as listed below:

One Trip to Elgin, OK	
Miles Per Trip	280
Cost Per Mile	$0.54 (IRS Rate, 2016)
Cost Per Trip	$151.20
Total Budget	$151.20

Table 7: Final Project Budget

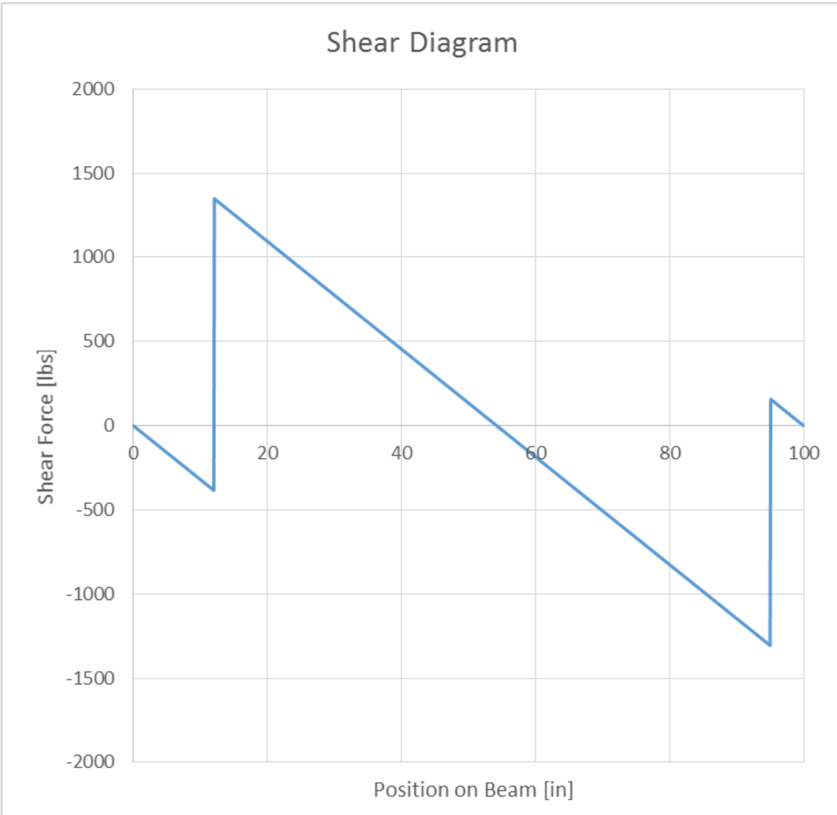
9.0 Gantt Chart

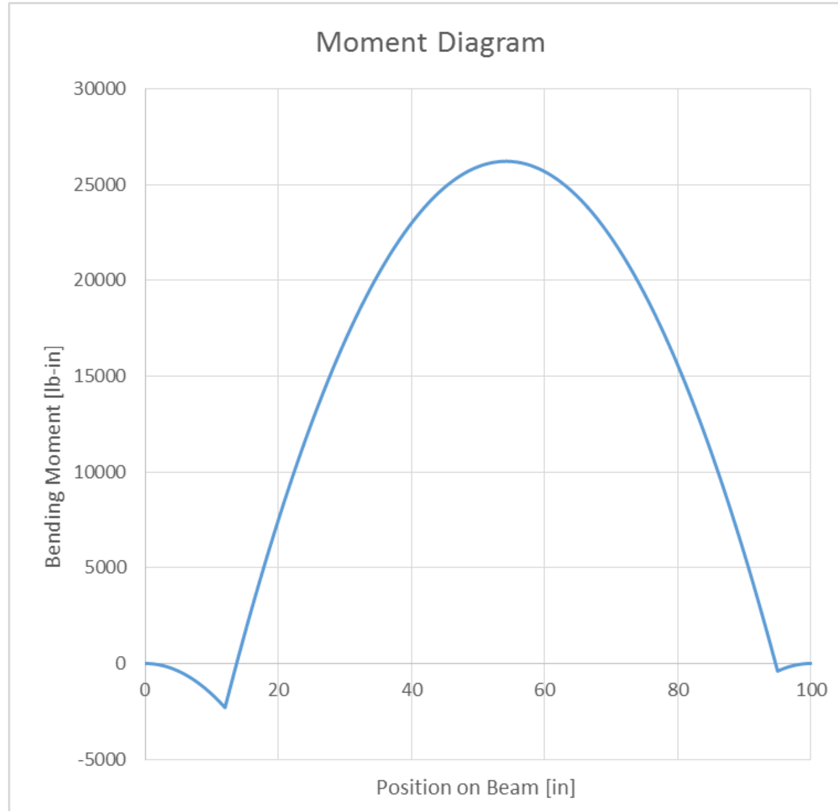
Senior Design	Project Start Date:		February 8, 2016	
BBE Parallel Squeeze Hay Bed				
Task	Start Date	End Date	Duration (days)	Percent Complete
1.0 Initial CAD Modeling	02-08-2016	04-10-2016	63	100.00%
1.1 Review Provided CAD Drawings	02-08-2016	02-28-2016	21	100.00%
1.2 Scan Provided Drawing Binder	02-08-2016	02-21-2016	14	100.00%
1.3 Construct 3D Model of Hay Bed	02-29-2016	04-03-2016	35	100.00%
1.4 Revise 3D Model (after input from client)	04-04-2016	04-10-2016	7	100.00%
2.0 Loading Consideration	03-07-2016	04-10-2016	35	100.00%
2.1 Research Dynamic Loading Models	03-07-2016	04-10-2016	35	100.00%
2.2 Develop Loading Models (3 Cases)	03-07-2016	04-10-2016	35	100.00%
3.0 Optimization of Structure	03-07-2016	04-17-2016	42	100.00%
3.1 Analyze/Optimize Frame	03-07-2016	03-27-2016	21	100.00%
3.2 Analyze/Optimize Squeeze Arms	03-21-2016	04-10-2016	21	100.00%
3.3 Analyze/Optimize Headache Rack	03-28-2016	04-10-2016	14	100.00%
3.4 Analyze/Optimize Accessories	03-28-2016	04-10-2016	14	100.00%
3.5 Revise Design (after input from client)	04-11-2016	04-17-2016	7	100.00%
6.0 Progress Report #1	03-07-2016	03-21-2016	15	100.00%
6.1 Create Oral Presentation	03-07-2016	03-21-2016	15	100.00%
6.2 Create Written Report	03-07-2016	03-21-2016	15	100.00%
4.0 Optimization of Manufacturing Process	03-28-2016	04-24-2016	28	100.00%
4.1 Optimize Frame Manufacturing	03-28-2016	04-24-2016	28	100.00%
4.2 Optimize Squeeze Arms Manufacturing	03-28-2016	04-24-2016	28	100.00%
4.3 Optimize Headache Rack Manufacturing	04-04-2016	04-24-2016	21	100.00%
4.4 Revise Design (after input from client)	04-11-2016	04-17-2016	7	100.00%
5.0 Final CAD Modeling	04-18-2016	05-01-2016	14	100.00%
5.1 Create Manufacturing Drawing Set	04-18-2016	05-01-2016	14	100.00%
5.2 Package Up Parts and Drawing Set	04-18-2016	05-01-2016	14	100.00%
6.0 Progress Report #2	04-04-2016	04-10-2016	7	100.00%
6.1 Create Oral Presentation	04-04-2016	04-10-2016	7	100.00%
6.2 Create Written Report	04-04-2016	04-10-2016	7	100.00%
6.0 Presentation of Results	04-18-2016	05-04-2016	17	100.00%
6.1 Create Final Oral Presentation	04-18-2016	04-24-2016	7	100.00%
6.2 Create Final Written Report	04-25-2016	05-04-2016	10	100.00%



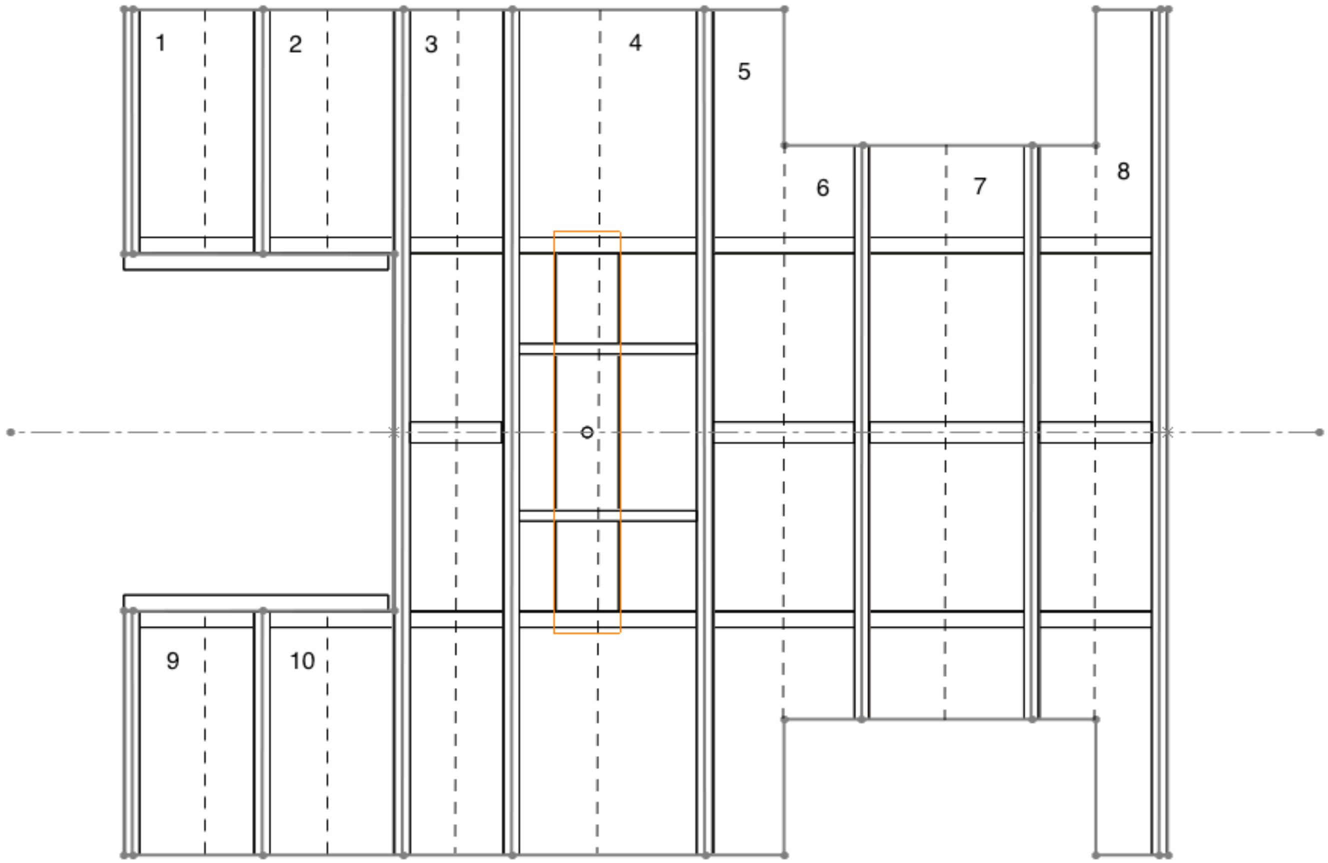
10.0 Appendix

10.1 Example Shear and Moment Diagrams



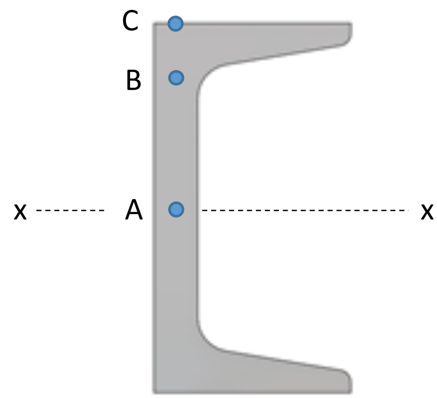


10.2 Cross Member Area Distribution

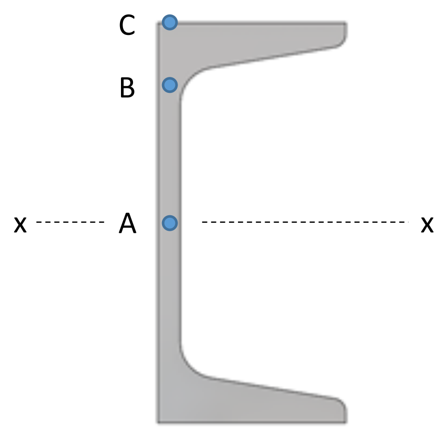


10.3 Cross Sectional Properties

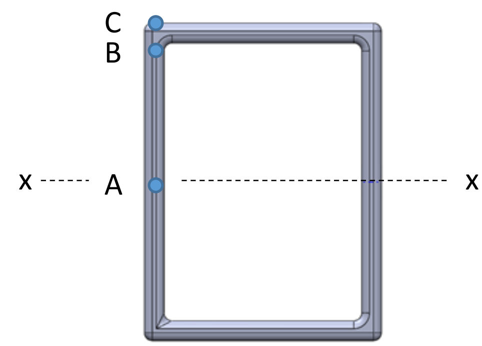
C3x6.0 Channel Iron		
I_x	2.07	in^4
Point A (Center of Cross Section)		
b	0.356	in
A'	0.88	in^2
ȳ	0.99	in
c	0	in
Point B (Just Before Flange Thickens)		
b	0.356	in
A'	0.56	in^2
ȳ	0.441	in
c	0.8949	in
Point C (Edge of Cross Section)		
b	1.596	in
A'	0	in^2
ȳ	0	in
c	1.5	in



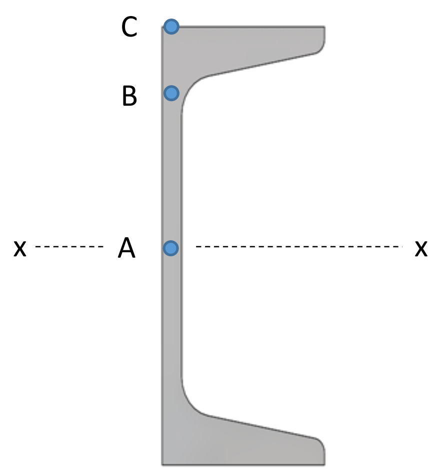
C3x4.1 Channel Iron		
I_x	1.65	in^4
Point A (Center of Cross Section)		
b	0.17	in
A'	0.603	in^2
ȳ	1.095	in
c	0	in
Point B (Just Before Flange Thickens)		
b	0.17	in
A'	0.4505	in^2
ȳ	0.491	in
c	0.8949	in
Point C (Edge of Cross Section)		
b	1.41	in
A'	0	in^2
ȳ	0	in
c	1.5	in



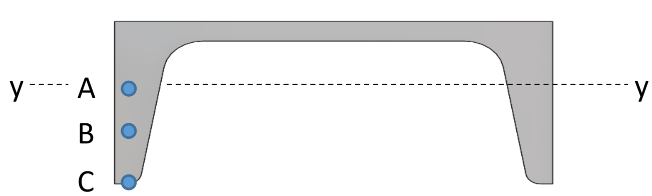
4x3x5/16" Rectangular Tube		
I_x	7.07	in^4
Point A (Center of Cross Section)		
b	0.625	in
A'	1.99	in^2
ȳ	1.31	in
c	0	in
Point B (Just Before Flange Thickens)		
b	0.625	in
A'	0.9375	in^2
ȳ	1.8438	in
c	1.6875	in
Point C (Edge of Cross Section)		
b	3	in
A'	0	in^2
ȳ	0	in
c	2	in



C6x13 Channel Iron - Vertical		
I_x	17.4	in^4
Point A (Center of Cross Section)		
b	0.44	in
A'	1.91	in^2
ȳ	1.91	in
c	0	in
Point B (Just Before Flange Thickens)		
b	0.44	in
A'	0.92	in^2
ȳ	2.75	in
c	2.26	in
Point C (Edge of Cross Section)		
b	2.16	in
A'	0	in^2
ȳ	0	in
c	3	in



C6x13 Channel Iron - Horizontal		
I_y	1.1	in^4
Point A (Neutral Axis)		
b	1.08	in
A'	1.10	in^2
ȳ	0.70	in
c	0	in
Point B (Midway Between A and C)		
b	0.68	in
A'	0.44	in^2
ȳ	1.19	in
c	0.83	in
Point C (Edge of Cross Section)		
b	0.2	in
A'	0	in^2
ȳ	0	in
c	1.64	in



10.4 Lifting Arm Assembly Cross-sectional Properties and Calculations

Shear of Pin Against Rx – 1G	
Force	13200lbs
Area	0.811in^2
Shear Stress	16200psi
Factor of Safety	2.21

Shear of Pin Against Ry – 1G	
Force	1100lbs
Area	0.811in^2
Shear Stress	1360psi
Factor of Safety	26.5

Tensile Stress – 1G	
Force	26400lbs
Width	3.77in
Diameter	1.02in
Thickness	1in
Area	2.75in^2
Normal Stress	9590psi
Factor of Safety	3.76

Tear-Out Vertical – 1G	
Force	1100lbs
Width	2.82in
Diameter	1.02in
Thickness	0.5in
Shear Stress	1220psi
Factor of Safety	18.1

Tear-Out Horizontal – 1G	
Force	26400lbs
Width	2.82in
Diameter	1.02in
Thickness	1in
Shear Stress	14600psi
Factor of Safety	1.50

11.0 References

[1] Budynas, Richard G., J. Keith. Nisbett, and Joseph Edward. Shigley. *Shigley's Mechanical Engineering Design*. New York: McGraw-Hill, 2011. Print.

[2] *Western Woods Use Book; Structural Data and Design Tables*. Portland, Or.: n.p., 1973. Print.