



# UNIVERSITY OF MANITOBA

## Cascade Pup Trailer Chassis Redesign

MECH4860 Final Report

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Project Advisor: Sean O'Brien

Submitted by: Team #2 CCPD

Cody Peters

Christian Gruenke

Patrick Domingo

Dennis Yon Wu

**University of Manitoba  
Department of Mechanical Engineering**

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Dr. Paul Labossiere, P Eng.  
Associate Head, Undergraduate Program  
Department of Mechanical Engineering  
University of Manitoba, E2-327F  
66 Chancellors Cir, Winnipeg, MB. R3T 2N2  
(204) 474-8304

Dear Dr. Paul Labossiere,

Please find the enclosed Final Design Report for the Cancade Pup Trailer Redesign project.

The report begins by outlining the project definition which includes the problem statement, objectives, needs, and project scope. A concept generation process was then performed to develop concepts and choose a final design to move forward with. The report then details every aspect of the final design including an overview of the design, methodology, and specific details regarding the different methods used for evaluation of the neck chassis. The report concludes with the final design and a summary of how the design achieves all the needs described.

Thank you for your time, and please contact us with any concerns or feedback you may have.

Sincerely,

Cody Peters

Christian Gruenke

Dennis Yon Wu

Patrick Domingo

MECH 4860 – Engineering Design, Team 2  
Department of Mechanical Engineering  
University of Manitoba

## Executive Summary

Cascade CBI Limited is a truck trailer manufacturer that is currently experiencing issues in their manufacturing process for pup trailers. The current procedure for manufacturing the pup trailer chassis involves a process that requires multiple people and cranes to lift and flip the 30 foot long assembly in order to weld all the appropriate areas. This process is both dangerous and time consuming. Cascade has requested that a new neck portion of the trailer be designed so that the manufacturing and assembly process becomes quicker and safer.

A project definition was created and included to provide background information on the project as well as to identify the client's needs and methods for evaluating those needs. The team then collaborated on concept generation to come up with ideas for possible concepts. The concept creation process was followed by an iteration of screening and scoring processes that resulted in the final concept, the Fork C-channel design made of low carbon steel.

The team then started on the details of the design. The first step, and a requirement for the deliverables, was to analyze the current design using Finite Element Analysis. This was done to determine the maximum load that the current design would fail at. This maximum load would be the benchmark to test the new design against, assuring that the new design would be as strong as the current design. An analytical comparison was performed to compare welding strength to HuckBolt strength, resulting in 9.45 inches of weld equal to the clamping force of one HuckBolt. During this time, joints for the tube to C-channel connection and neck to frame connection were designed with the dimensions of the C-channel and tube approximated based on research and what Cascade currently uses for manufacturing. The benchmark loads were then applied to the new design and weak points were identified at the box to C-channel joint. Several iterations of joints and joint reinforcements were designed and analyzed until the joint was strong enough to resist yielding. The design was then subjected to known loads, leading to results that showed the C-channel and box met the desired factor of safety.

The final design is a Fork C-channel design constructed with HuckBolts that weighs 1007 lbs with a total cost of \$1344.30. The design was considered a success since the new design is able to be assembled without any linear or rotational movement which allows for cheaper, faster and safer manufacturing.

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## 1. Introduction

Cancade CDI Limited has tasked our team with designing a new neck chassis that can be manufactured and assembled in a safe and efficient manner when compared to their current design. In order to fully understand the problem and Cancade's needs, a definition of the project was created outlining background information, needs, metrics, and project scope. This report details the optimization process used to achieve the final designed neck chassis assembly. An analysis procedure was then used to evaluate the design's ability to satisfy the client's needs that were defined previously.

### 1.1. Background

Cancade is a manufacturing company located in Brandon, Manitoba. Cancade manufactures a wide range of gravel, agricultural, and commercial equipment, focusing largely on trailers. This project specifically focuses on the Cancade pup trailer, sometimes referred to as a pony trailer. The legal definition of a pony trailer is a non-powered vehicle that is towed by a rigid drawbar, with most of the load being carried by the axle [1]. The trailer consists of a box mounted to a twin rail frame via a hinge on the rear and a telescopic cylinder hoist on the front, as shown in Figure 1. The trailer is towed by a hitch which is connected to the frame by the neck. The chassis of the trailer is the entire rigidly connected body, including the frame, neck, and hitch.

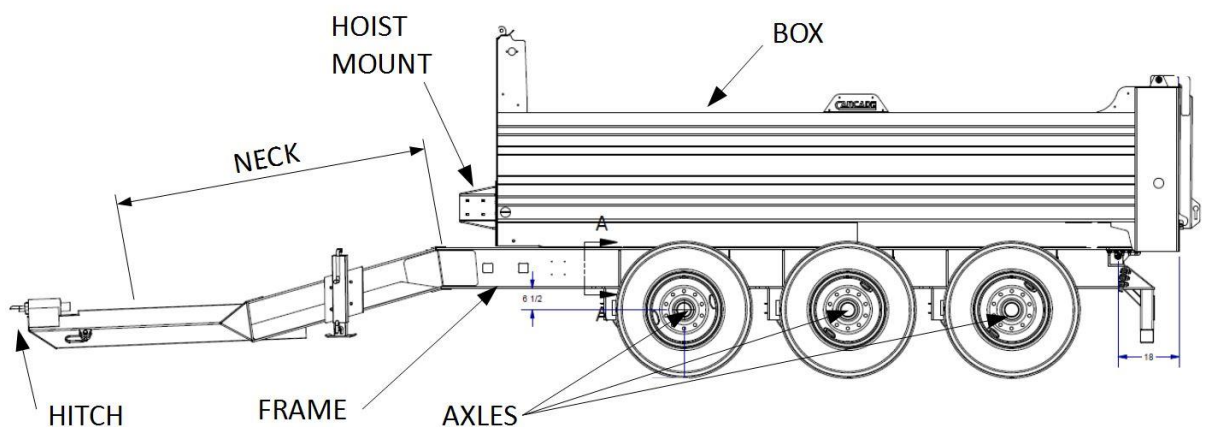


Figure 1. Diagram of a tridem pup trailer viewed from the side [2].

The Cancade pup trailer is manufactured with a grain or gravel box and is primarily used by commercial trucking, construction, and farming operations. Typical operating conditions for the trailer include rough terrain and uneven roads. The trailers often operate continuously for 24 hours a day [3] and are subject to loads beyond their rated capacity. For these reasons, a robust and strong design is required to operate in these conditions.

Cancade's manufacturing floor follows a process layout where the welding area is shown in Figure 2. The current trailer has the structural components of the chassis welded together, which is conventional with this type of trailer [3]. The welding process for the frame involves laying out the components on a fixture so that they can be properly aligned before welding. The welds must be made in the appropriate orientation relative to the chassis, as vertical and overhead welds are not permitted at Cancade [3]. Once the initial welds are made, the chassis must be repositioned to continue welding the rest of the required locations. Previously, a rotary jig was used to reposition the frame, but the use of this jig was discontinued due to infrastructure limitations. Currently, the frame is repositioned by manually lifting and rotating the chassis, which is time consuming and hazardous for the workers. This re-positioning of the assembly is the main issue, not specifically the welding process itself.

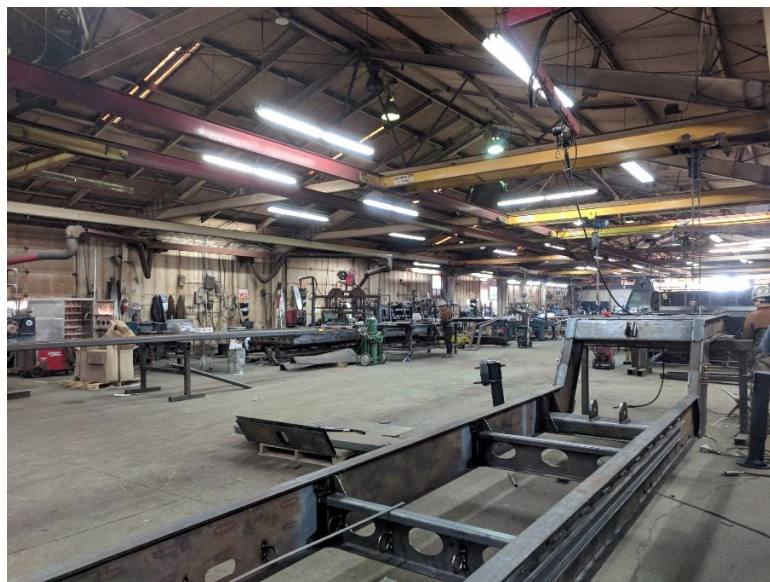


Figure 2. Cancade's welding area [3].

Because of the previously mentioned limitations of welding the trailer chassis, Cancade has started using HuckBolts to join their frames together. The specific HuckBolt used is the 16 mm

BobTail lockbolt [4], which requires a hydraulic gun shown in Figure 4 to crimp the fastener. This process has reduced restrictions in orientation, mitigating the need to reposition the frame during assembly. HuckBolts are the only alternative to welding that will be considered by our team due to the investment that Cancade has already made in the required equipment.



Figure 3. HuckBolt used by Cancade [3].



Figure 4. HuckBolt hydraulic gun [3].

Cancade's pup trailer is widely regarded as having one of the strongest necks in the market, with a robust design that has seen no warranty claims in the recent past [3]. Cancade requires a neck design that is safe and efficient to manufacture but is also analytically and numerically confirmed to be as robust as the previous design.

To ensure that the chassis neck design can be implemented and operated without risks, the neck design must comply with the imposed standards in Manitoba and Canada. The main standards that directly apply to the neck are the following.

- Highway Traffic Act
  - The portion of interest from the Highway Traffic Act is ruling 61(2) which states that the transported load must be secured and not fall off from the trailer [5]. This applies to debris built up along the trailer chassis.
- Vehicle Weights and Dimensions Limits in Manitoba
  - These standards limit the length of the trailer chassis and the neck portion. The maximum length allowed for a pup trailer is of 12.5 meters and the maximum allowed length of the neck is of 4 meters [1].

- Canadian Welding Bureau CSA W47.1 – Certification of companies for fusion welding of steel
  - The most common material used in pup trailers is steel. Therefore, in case the redesigned chassis neck requires to be welded it must follow the standard set by the Canadian Welding Bureau for fusion welding of steel [6].

In order to avoid infringing intellectual property rights, patents regarding welded and bolted pup trailer chassis were investigated. No patents were found specifically for the design of the chassis neck.

## 1.2. Problem Statement

The current manufacturing process for the Cancade pup trailer chassis is unsafe, inefficient and costly. The most critical section of the chassis regarding strength and manufacturing inefficiency is the neck. For these reasons a new design is required to allow for a safer and more efficient means of assembly.

The three main objectives of the project are as follows:

1. Redesign the structure of the pup trailer chassis neck for reduced manufacturing time and cost.
  - Current design of the chassis neck requires time consuming and costly manufacturing procedures such as the use of welds.
2. Redesign the chassis neck to allow for safer manufacturing and assembly processes.
  - Current design of the chassis neck is cumbersome and requires the chassis neck to be repositioned and moved during manufacturing, which poses a safety risk to the workers and potential for damage to the chassis neck. This is due to the re-positioning and not specifically the welding process.
3. Ensure that the new chassis neck design is as strong as the previous design.
  - The new chassis neck design must be able to withstand the same loading conditions as the current design without experiencing material yielding. Since the current chassis neck design has never been observed to fail during operation, the new design is expected to be successful as long as the new design is as strong as the current design.

### 1.3. Project Deliverables

In order to meet client requirements, this project contains the following deliverables:

1. A new design for the pup trailer chassis neck optimized for manufacturing time, cost, and safety.
2. Preliminary drawings of the new chassis neck design.
3. Finite element analyses of both new and current chassis necks which will be used as a basis for optimization procedures.
4. Analytical comparison of the HuckBolt and weld connection between chassis neck components.
5. Presentation regarding the project and results on December 6, 2018.

### 1.4. Project Needs and Target Specifications

The needs of the new chassis neck design are listed in Table I. The list also includes a value of relative importance for each need using numbers between 1 and 5, with 1 being the least important and 5 being the most important. These importance ratings were determined through consultation with the client.

TABLE I. LIST OF NEEDS.

Need ID	Need	Importance
1	<b>New design must be strong.</b> The new design must be as strong as the current design. Since the current design has not been observed to fail during operation, the new design is also expected to have the same performance. Meeting the current design's strength means being able to withstand the same maximum loading conditions without yield or excess deflection. Additionally, the new design must be able to withstand forces due to jackknifing.	5
2	<b>New design's overall geometry must not impede functionality of the trailer chassis.</b> In order to function properly, the trailer must adhere to the <i>Vehicle Weights and Dimensions Limits in Manitoba</i> regulations [1]. This will ensure that the trailer along with the towing vehicle operate in a safe manner while maintaining the required range of motion.	5

Need ID	Need	Importance
3	<p><b>New design's overall manufacturing process must be efficient.</b></p> <p>The new design must be more efficient to build. Manufacture efficiency includes the time to machine all required components, the time needed to connect the components together, and the ease of assembly.</p>	4
4	<p><b>New design must not require moving or repositioning of the structure during manufacturing and assembly.</b></p> <p>The current chassis neck designs require the lifting and re-positioning of the structure in order to weld all the seams and connections. The new chassis neck must be designed so that no re-positioning is required.</p>	4
5	<p><b>New design must withstand dynamic loading.</b></p> <p>In addition to meeting the same static load conditions as the current chassis design, the new design must also be able to withstand the same shock, vibration and other dynamic loading as the previous chassis design without yield or excessive deflection.</p>	4
6	<p><b>New design's overall cost is low.</b></p> <p>The total cost of the new design must be lower than that of the current design, including material and manufacturing costs.</p>	3
7	<p><b>New design must be able to operate for 8 years.</b></p> <p>New chassis neck design must be able to operate under normal conditions without requiring any part replacement or additional maintenance during its expected lifetime. This includes accounting for fatigue and other time dependent effects.</p>	3
8	<p><b>New design's required manufacturing and assembly processes must be safe.</b></p> <p>The neck design must minimize the potential for hazards to all persons involved in the manufacturing process.</p>	3
9	<p><b>New design must be light.</b></p> <p>The new chassis neck design must be close to the same weight as the current design. Additionally, it is preferable that the new design is lighter than the current design.</p>	2
10	<p><b>New design must be visually pleasing and appear strong.</b></p> <p>The neck portion of the new chassis design must appear to be sturdy and strong to the Cancade Engineering Department.</p>	2
11	<p><b>New design must have minimal debris buildup.</b></p> <p>New chassis neck must be designed in a manner such that no gravel or road debris will collect and fall off. [5].</p>	1
12	<p><b>New design must allow for easy attachment of peripheral components.</b></p> <p>For convenience in product assembly or maintenance, peripheral components such as trailer jacks must be able to attach onto the new chassis neck design.</p>	1

In order to measure if the needs of the client have been met, metrics were created so that each need could be quantified with a technical specification that is relevant to Cancade. Marginal and ideal values for each metric were determined in consultation with the client. The level of importance for each metric was derived from the customer needs. One of the metrics is quantified using a scoring matrix based on a percentage, labelled as SM. The metrics are listed in Table II.

TABLE II. METRIC IDEAL AND MARGINAL VALUES.

Metric ID	Need ID	Metric	Level of Importance	Marginal Value	Ideal Value	Units
1	1	Maximum tongue weight	5	7306	>7306	lbs
2	1	The factor of safety related to the maximum stress values observed	5	2	3	unitless
3	2	The amount of time it takes to manufacture the assembly	4	9	6	hour
4	2	Number of parts	4	14	<14	quantity
5	2	Equipment types used to manufacture and assemble	4	7	<7	quantity
6	3, 4	The total assembly weight	3	1050	750	lbs
7	2, 4	The total cost of the assembly (material and labor)	4	1722	<1722	\$
8	1, 5,11	The fatigue life of the assembly	5	8	10	year
9	6	Prevents debris buildup [5]	1	-	pass	binary
10	7	The existing peripheral components (jack, hitch adaptor, etc.) are compatible	1	-	pass	binary
11	8	The ability for the assembly to be manufactured and assembled in a safe manner	3	60%	100%	SM
12	9	Linear movement required during assembly	4	4	0	ft

<b>Metric ID</b>	<b>Need ID</b>	<b>Metric</b>	<b>Level of Importance</b>	<b>Marginal Value</b>	<b>Ideal Value</b>	<b>Units</b>
13	9	Rotational movement required during assembly	4	45	0	degree
14	10	Retains critical geometry	5	-	pass	binary
15	12	The engineering department approves that the design appears sturdy	2	-	pass	binary

Appendix A provides additional information on each metric. This includes reasons for the metrics and any relevant background information for the chosen marginal and ideal values.

### 1.5. Limitations and Constraints

The limitations and constraints were set by the client, while some were separately identified by the design team and then confirmed by the client. The limitations and constraints are listed and explained in Table III.

TABLE III. LIST OF LIMITATIONS AND CONSTRAINTS.

#	Limitation/Constraint	Additional Information
1	Design space	The design space is limited by a set distance between the hitch and the front of the frame. The height and width of the design space is also limited by the height and width of the frame.
2	Joint type	The client has requested the use of a HuckBolt, which is a permanent, rivet-like, fastener. This is because the client has previously invested in the proper equipment to install the fasteners and has had positive experiences with the HuckBolt. Welding that does not require any re-positioning will also be investigated and considered.
3	Hitch design	The current production hitch adaptor at the front of the neck chassis is used throughout Cancade’s production line and is a proven feature. Therefore, the hitch adaptor must remain unchanged.
4	Raw materials	The client has certain common materials that are available to them through their suppliers and distributors. These will be the material options available to the design team. The materials will also come in standard sizes and shapes which will also limit the design.
5	Manufacturing capabilities	The client has a limited amount of resources when it comes to manufacturing capabilities, such as the machines available and the machines’ capabilities. The design team will only use manufacturing practices that are readily available to the client.

#	Limitation/Constraint	Additional Information
6	Loading scenarios	<p>The client has requested two loading scenarios be used for the Finite Element Analysis and the design team has determined another. The loading scenarios are the following.</p> <ul style="list-style-type: none"> <li>• Beginning of the unloading sequence of the trailer while the trailer is not moving. The beginning sequence of unloading is the worst-case scenario as all the weight is going through the unloading cylinder <ul style="list-style-type: none"> <li>○ A fatigue analysis will be performed for repeated loading and unloading cycles of 30 minutes over a period of 10 years.</li> </ul> </li> <li>• A Force to simulate a bump on the road.</li> <li>• The design team will also evaluate the design during a jackknifing scenario which could potentially happen during every unloading.</li> </ul> <p>Because the new design will be as strong as the current design which has never failed, the new design will be considered strong enough in all other scenarios.</p>
7	Software limitations	<p>The students will only be using software for the numerical analysis that is available to them as students. This includes the FEA and CAD software.</p>
8	Cost analysis	<p>The design team will focus on the cost of material, labour, and processes. The design team is not able to provide an extensive cost analysis that includes concepts such as depreciation of machinery, amortization, taxes, or liability.</p>
9	Order of assembly	<p>The design team will not provide any information on the best order of assembly for the new neck design.</p>
10	Work hours available	<p>The design team consists of four undergraduate team members. The students have limited hours available due to other school obligations but will work as many hours needed to fulfill the deliverables.</p>
11	Technical capabilities	<p>All four team members are students who will strive to meet the deliverables to the best of their abilities. Yet the rigor of the FEA, fatigue analysis, or engineering drawings and models will be limited to the knowledge level of the undergraduate students.</p>

#	Limitation/Constraint	Additional Information
12	Student cost	The project has been given a budget of \$400 which the team may spend on travel, printing, and other costs associated with the project.
13	Project schedule	The project report has a final deadline of December 5, 2018, with other milestone deadlines throughout the project duration. This congested schedule will limit the options that can be explored as well as how large the project scope can become.
14	Client location	Candace is located in Brandon, MB and is an approximate 2.5-hour drive from the University of Manitoba. This long drive will limit the team in how much time they can spend at the client's facility.

## 1.6. Project Assumptions

In order to reduce the complexity and scope of the project, certain assumptions accepted by the client have been made. The following numbered list displays all the assumptions made during the project definition stage. It is assumed that:

1. The new chassis neck design will not fail under intended operation so long as it withstands the same loading conditions as the current chassis design.
2. The current chassis neck design has never structurally failed during operation.
  - The second assumption is important for the first assumption because the performance of the current chassis design is used as the base performance of the new design. There have been no reported instances of the current chassis structurally failing during operation. However, it is unknown whether all the actual instances of failure simply went unreported or if the owners continuously maintained the current chassis trailers.
3. There is no torque or moment applied to the trailer at the hitch.
  - The hitch is able to rotate in a way to prevent moments from being applied to the chassis neck. However, no actual tests have been performed to ensure that the assumption is true.
4. Maximum static loading during normal operation is during the beginning of the unloading process.

- This is the preliminary assumption into the worst-case (or highest-load) scenario that is shared by the client [3]. This assumption will be used as the basis for determining the trailer chassis external loads when going into concept development.
5. The trailer is operated every day for 24 hours a day, with the trailer fully loaded for half the time.
- The information regarding the product usage has been provided by the client [3]. This assumption will be used for the analysis of time-dependent effects such as fatigue.
6. Corrosion is prevented with Cancade's anti-corrosion method.
- Corrosion is an issue in the design of outdoor equipment for use over a long period. However, the current chassis manufacturing process involves a stage where corrosion-resistant coating is applied, which will also be applied to the new chassis design. To reduce the scope and complexity of the project, it is assumed that the coating by itself will be sufficient to prevent significant corrosion of the chassis neck over the lifetime of the new design.

## 2. Methodology of Concept Generation

With the project needs clearly defined, a systematic approach to establish the best design concept for the hitch was employed. This process involved generating concept ideas from external research and internal brainstorming. These concepts went through several stages of screening and scoring to determine the best concept to further optimize.

An external search was performed, and different competitor's pup trailer designs were examined with the objective to explore as many alternatives as possible. It was observed that most products used a fork, triangle, beam, or truss for the general shape of their hitch. A search of patents for similar products was also performed, however this did not result in any relevant design considerations.

At the early stage of the concept development process, the initial concepts generated and considered were no more than abstract ideas with significant room for change or improvement. The shape, cross-section and material were found to be fundamental aspects to each of the designs considered.

A set of criteria were determined for each category to evaluate how well each concept meets the product needs. The criteria are specific to each category and were used for initial concept screening. Each of the concepts in the categories were scored relative to the current hitch, acting as a baseline. The decisions made for this selection were based on intuition, so to offset any bias the selection process was performed individually by the project team members and the aggregate data was used to evaluate the concepts. The material, top three shapes, and cross-sections were chosen. It was clear that mild steel was the best option, so the final material was selected at this point.

The combinations of the three cross-sections and three shapes were used to create nine intermediate designs. A new set of criteria was established to evaluate the designs and were weighted by comparing the importance relative to one another. Another screening process was used to determine the four best concepts.

A more rigorous evaluation of the designs was performed to determine the final best concept to optimize for the final design. Using a scoring matrix, the fork shape with a C-channel cross-section in the rear, and a square tube at the front was determined to be the best concept.

### 3. Design Overview

With the general shape of the neck determined from the results of the concept development process, the specific details of the angles and dimensions of the neck was determined. A CAD model of the neck was created, as shown in Figure 5.

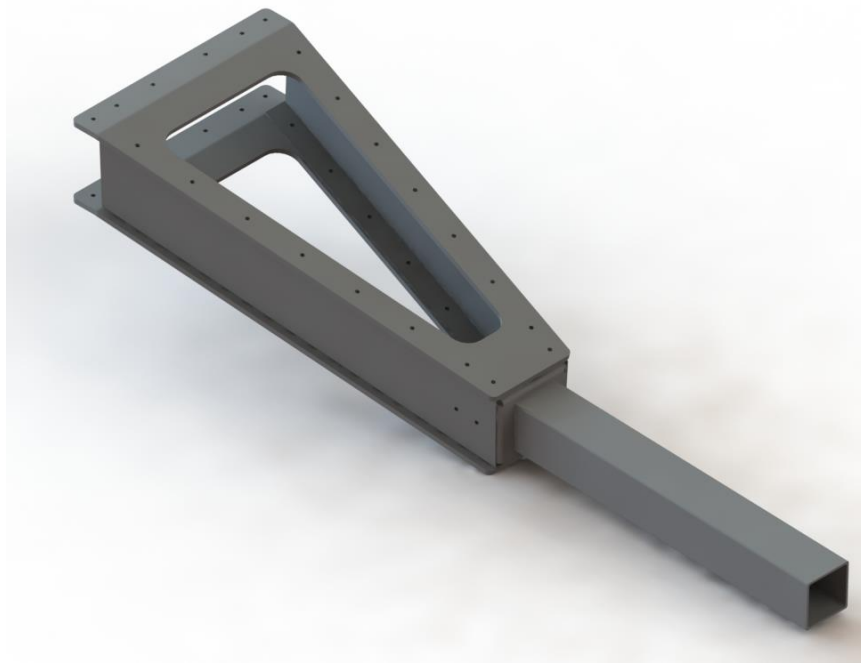


Figure 5. Initial new neck chassis design (HuckBolts hidden).

### 3.1. Square Tube

The location of the transition from the square tube and the C-Channel is determined from the point at which the tires of a towing vehicle will contact the square tube. This is shown in Figure 6 as 46 inches from the front of the hitch.

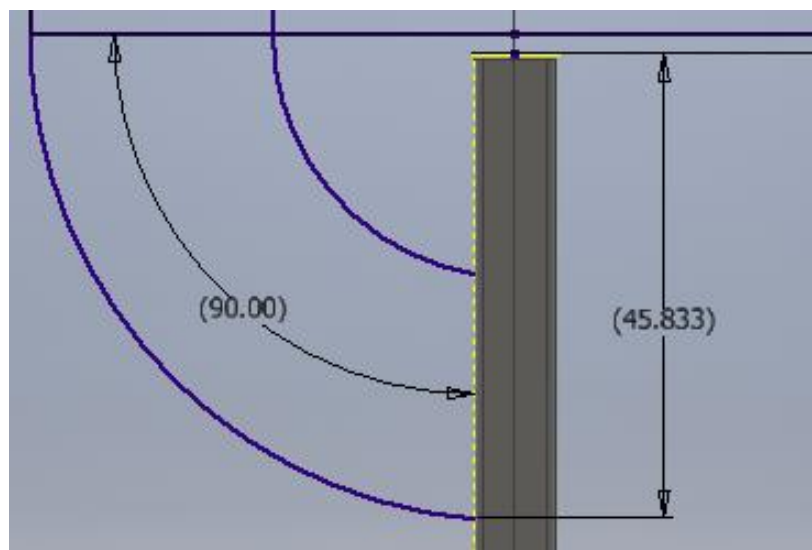


Figure 6. Geometric analysis for jackknifing location.

Intuitively, the hitch will be stronger with the shortest possible length of square tubing as the inertia to mass ratio of the C-Channel is greater than the square tubing. Eleven inches was considered adequate as a safety buffer, resulting in an approximate length of the square tubing of 57 inches, as shown in Figure 7.

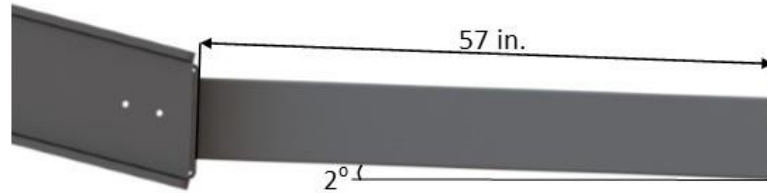


Figure 7. Angle of tilt of the square tube with respect to the ground.

The square tube is welded to a supporting structure with a 2 degree tilt angle relative to the frame, as shown in Figure 7. Two degrees in particular was chosen because it was considered acceptable by Cancade in their previous design [3]. This angle prevents any water accumulation on top of the tube and decreases the chances of corrosion.

The orientation of the hollow structural section is a square configuration, as shown in Figure 8. The square orientation was chosen over the diamond orientation because it reduces manufacturing time by reducing the amount of processing required to attach the hitch adapter. Additionally, it increases the range for jackknifing by having a smaller effective width while possessing the same moment of inertia.

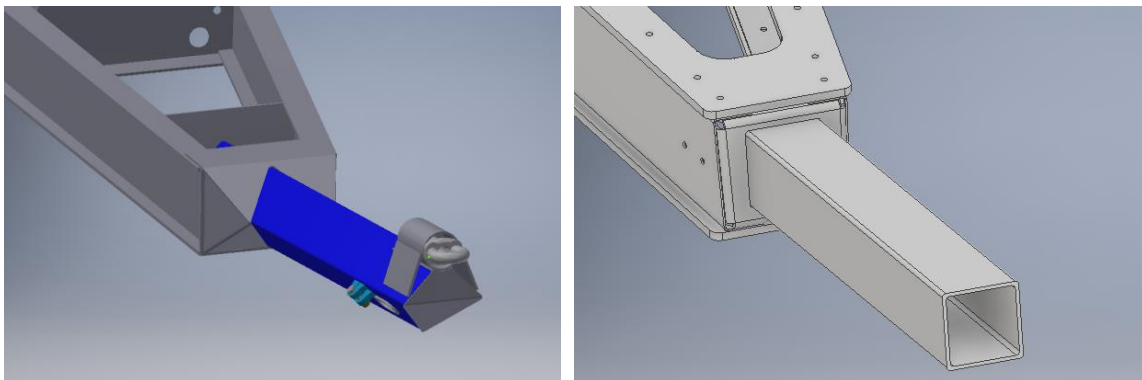


Figure 8. Square tube orientation comparison

### 3.2.C-Channel

The C-channels are made from a single bent metal sheet of uniform thickness. The angle of the C-channels with respect to the ground was restricted to 14.90 degrees, as shown in Figure 9. The angle is restricted because of the already established hitch location, square tube length, and overall neck length.

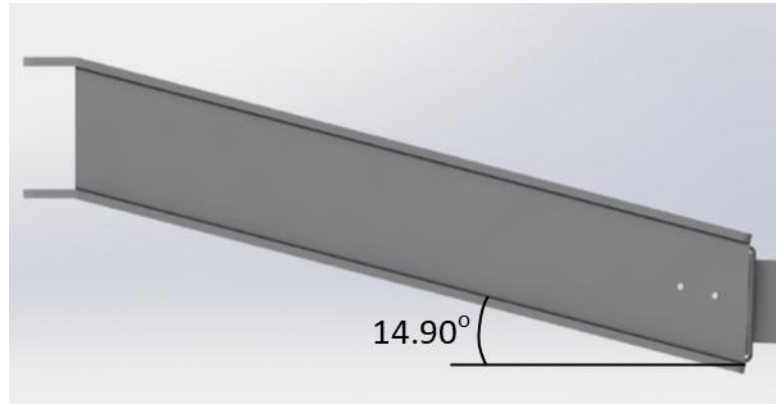


Figure 9. Angle of tilt of the C-channels with respect to the ground.

The angle of the C-channels with respect to the square tube is 80.5 degrees, as shown in Figure 10. This angle is restricted by the length of the square tube and the location of the C-channel with respect to the square tube.

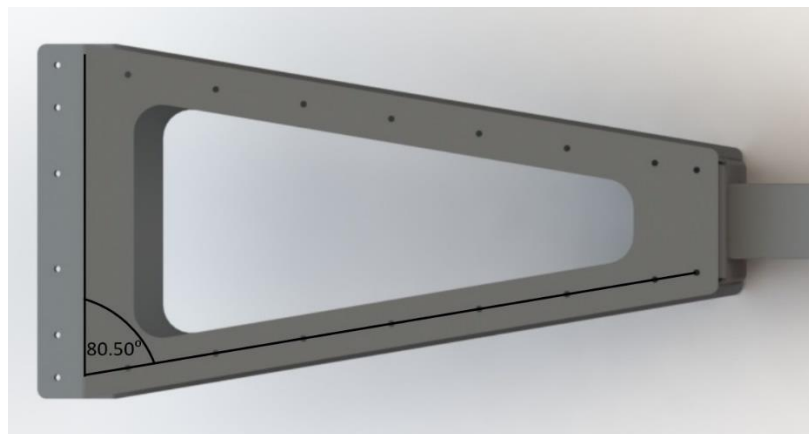


Figure 10. Angle of triangle section of the neck.

The location of the C-channel relative to the square tube was set so the C-channel was as close as possible to minimize material used, while still being acceptable for manufacturing. If the C-

channel is too close to the square tube, the member joining them will have a flange that is difficult to manufacture [7]. The minimum recommended flange length,  $D$ , can be determined using Equation 1, where  $t$  is the thickness, and  $R$  is the bend radius. The largest reasonable thickness for this member is 3/4 inches, with a bend radius of 5/8 in, results in a distance of 2.5 inches. To ensure the design is conservative, a distance of 3 inches was used, as shown in Figure 11.

$$D = 2.5t + R$$

Equation 1

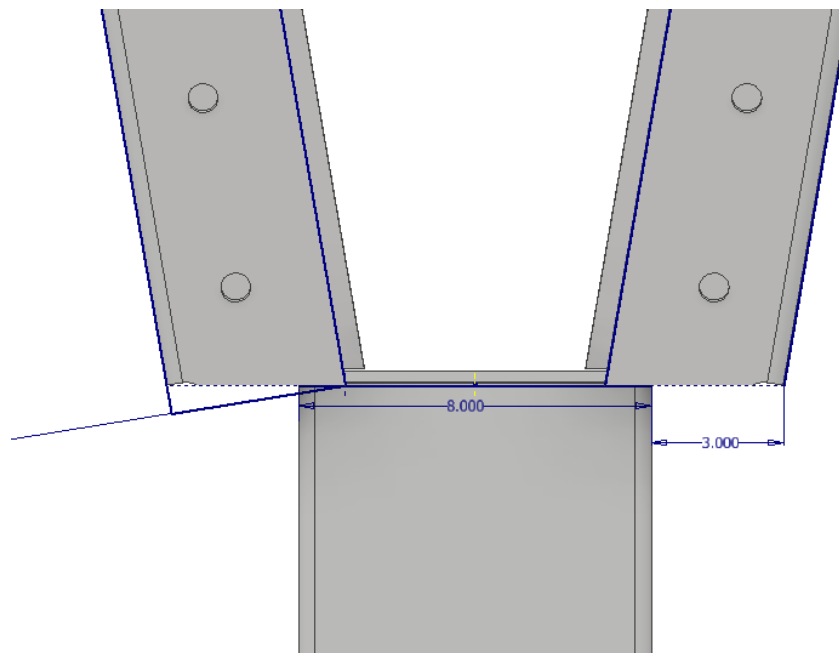


Figure 11. Location of C-channels with respect to the square tube.

### 3.3. Top and Bottom Plates

Two plates are bolted to the top and bottom of the C-channel, referred to as top and bottom plate, shown in Figure 12. These plates also bolt to the trailer frame, rigidly attaching the hitch to the frame without requiring any structural welding. Therefore, any repositioning of the frame is eliminated.

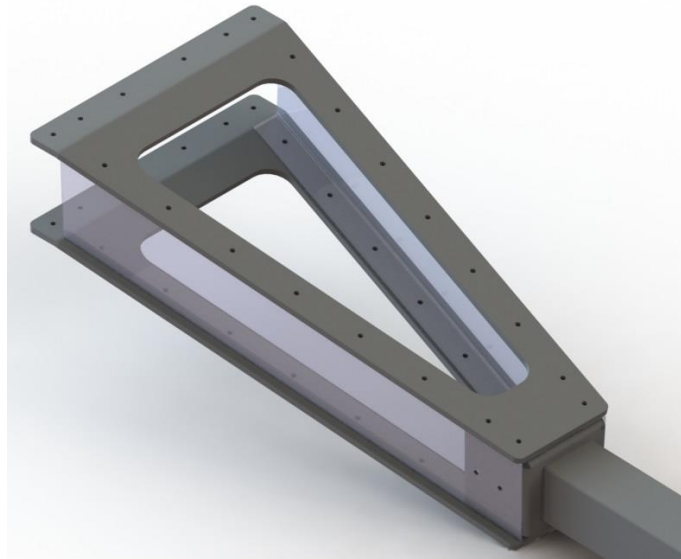


Figure 12. Top and bottom plate location and joining method.

These plates result in the neck having a higher bending stiffness as they increase the effective flange thickness of the C-channels. The use of these plates gives more opportunity for optimization compared to a single member. This is due to the C-channel geometry not relying on uniform thicknesses, as shown in Figure 13.

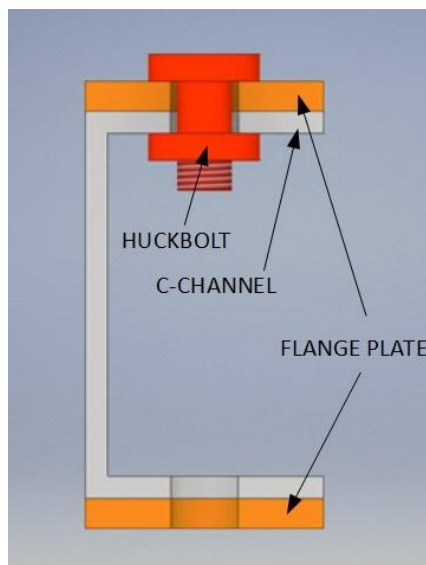


Figure 13. Effective flange thickness for c-channels with plates added.

### 3.4. Square Tube Support Structure

The supporting structure joint or “box” of the square tube is made from bending a steel sheet cut in the shape of a cross, shown in Figure 14.

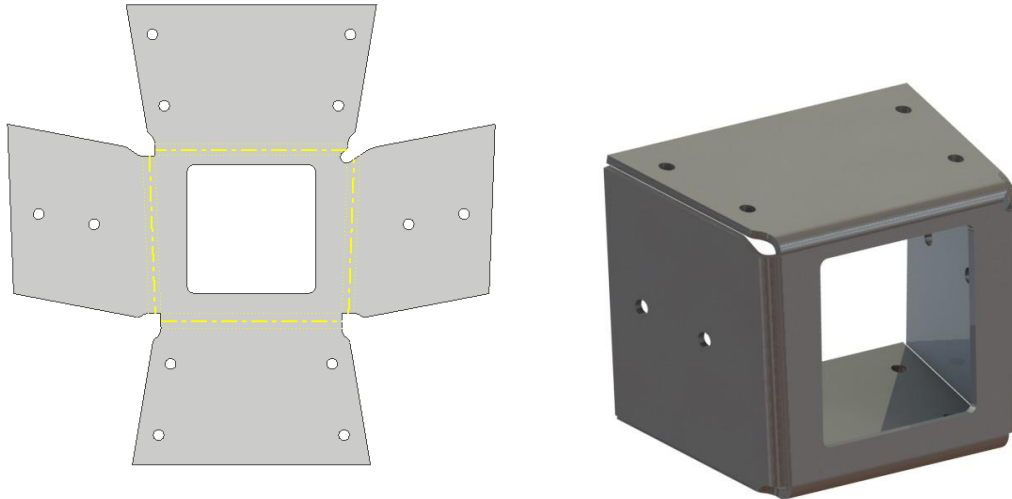


Figure 14. Box structure that supports the square tube.

The square tube is welded to the box, and the box is bolted to the C-channels and the plates as shown in Figure 15. This design allows for a modular assembly process. Even though the box-square tube assembly requires rotating during welding, the effort is minimal and does not require forklifts or cranes compared to rotating the whole neck structure.

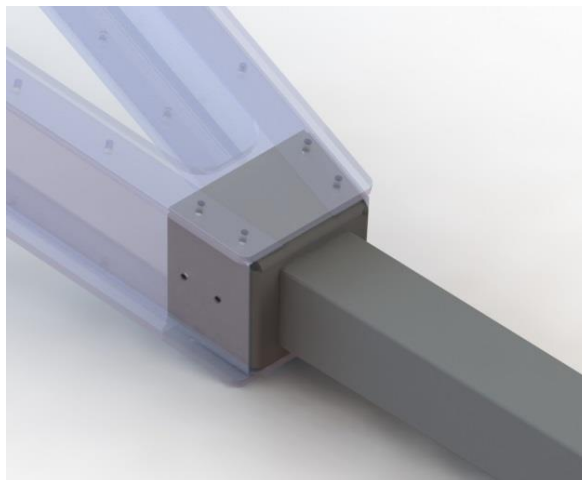


Figure 15. Location and joining method of the box.

### 3.5.HuckBolt Gun Clearance

The location of the HuckBolt holes allows for the HuckBolts to be positioned such that the head of the fastener is on the outside surface, as shown in Figure 16. Installing the Huckbolts in this position eliminates the risk of having the tail of the bolt getting caught on equipment or the operator. The appearance of the neck is visually more pleasant when the head of the HuckBolts are pointing outwards rather than inwards.

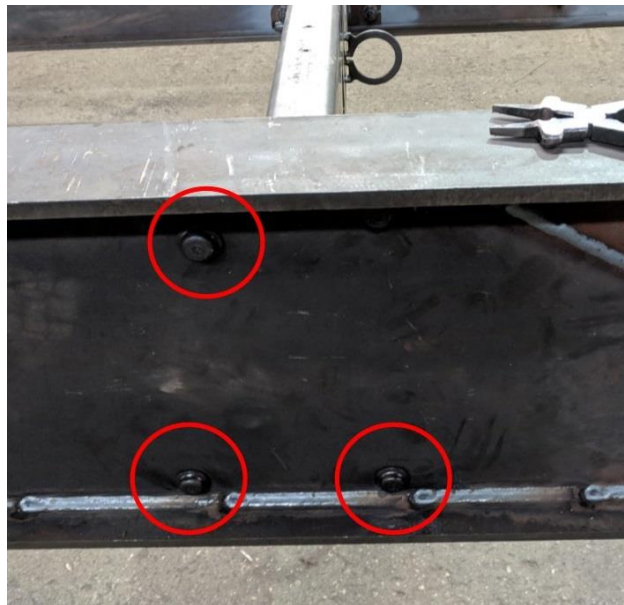


Figure 16. HuckBolt orientation, the head of the HuckBolt is facing outward.

The hydraulic gun requires a radial clearance zone around the hole of 1.5 inches from the center. Also, the height of the C-channel must be at least 7 inches for the gun to fit in between them. These clearances are determined by the dimensions of the hydraulic gun used, shown in Figure 17. The current clearance for holes is a radius of 2 inches and the height of the C-channel is of 11 inches, giving extra room for clearance.



Figure 17. Dimensions of the hydraulic gun used by Cancade.

The final thickness and dimensions of each of the components are optimized through finite element analysis and are described in section 4.6.3.

### 3.6. Analytical Comparison of HuckBolts to Welded Connections

One of the deliverables required by Cancade is an analytical comparison of welded connections between members and the use of HuckBolts. The analytical comparison was used to determine the bolt spacing equivalent to a welded connection. This potentially allows for the interchanging of the two joining methods in any design without significant changes in strength.

This report specifically considers a comparison of the I-beam section of the current neck design, and the C-channel section of the new neck design, shown in Figure 18 and Figure 19 respectively.

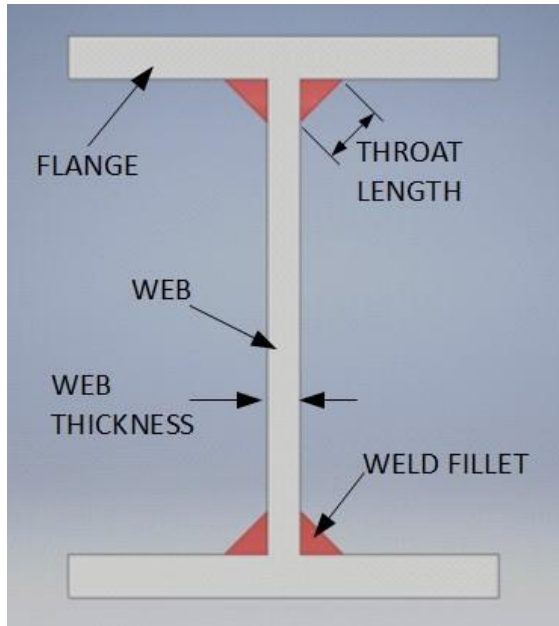


Figure 18. The cross-section of the current I-Beam design, where the weld fillet, shown in red, joins the web to the flange.

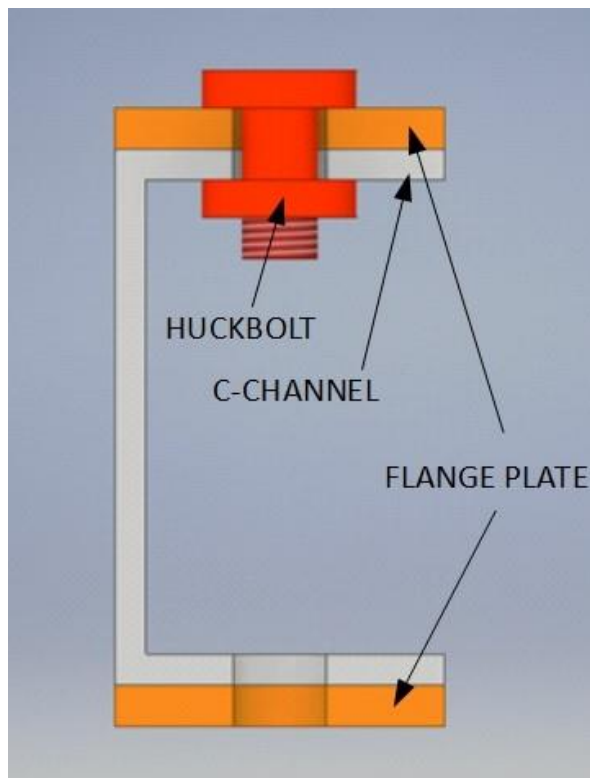


Figure 19. The C-channel from the new design, a HuckBolt, shown in red, connects the C-channel to the top plate and bottom plate.

If a welded structure is designed properly, the throat of the weld will be sized such that it will not fail during loading before the web of the structure fails [8]. The structure will fail when the load causes the stress in the web to reach the yield strength, shown by Equation 2, where F is the load, and A is the web area.

$$\sigma_{yld} = \frac{F}{A} \quad \text{Equation 2}$$

The web area shown in Figure 20 is equal to the web thickness, t, multiplied by the length along the edge, L.

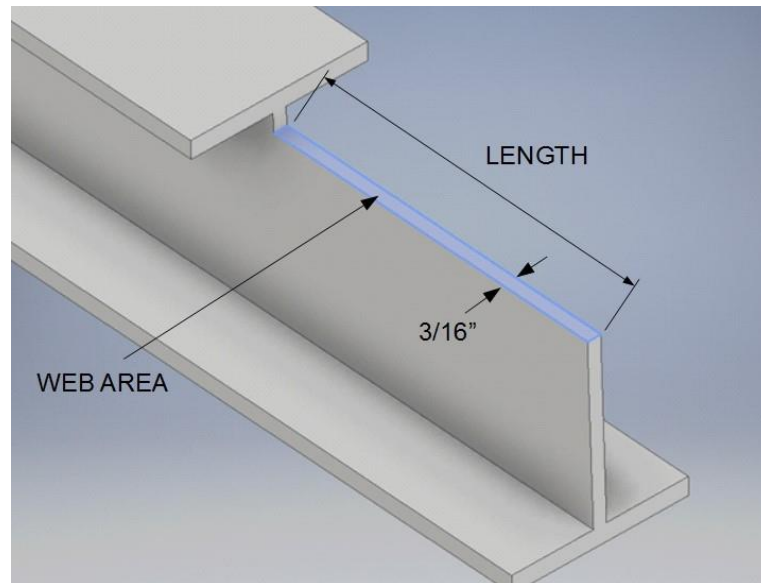


Figure 20. Area of the web that will fail first under a vertical load.

This results in Equation 3, which describes the minimum length of a particular cross-section required to withstand a force.

$$L = \frac{F}{t\sigma_{yld}} \quad \text{Equation 3}$$

The current design of the trailer hitch has a web thickness of 0.1875 inches and is made from 44W mild steel, which has a yield strength of 36000 psi [9]. This results in the relationship between the force applied to yield a particular length of cross-section, described by Equation 4.

$$L = \frac{F \text{ in}}{6.75 \text{ kip}}$$

Equation 4

The manufacturer specification for the proof load of the HuckBolt is 26000 lbf [4]. Applying this force to Equation 1 results in an equivalent bolt spacing of 9.45 inches, illustrated in Figure 21. This shows that spacing HuckBolts 9.45 inches apart results in a joint of the same strength as the current welded design in bending.

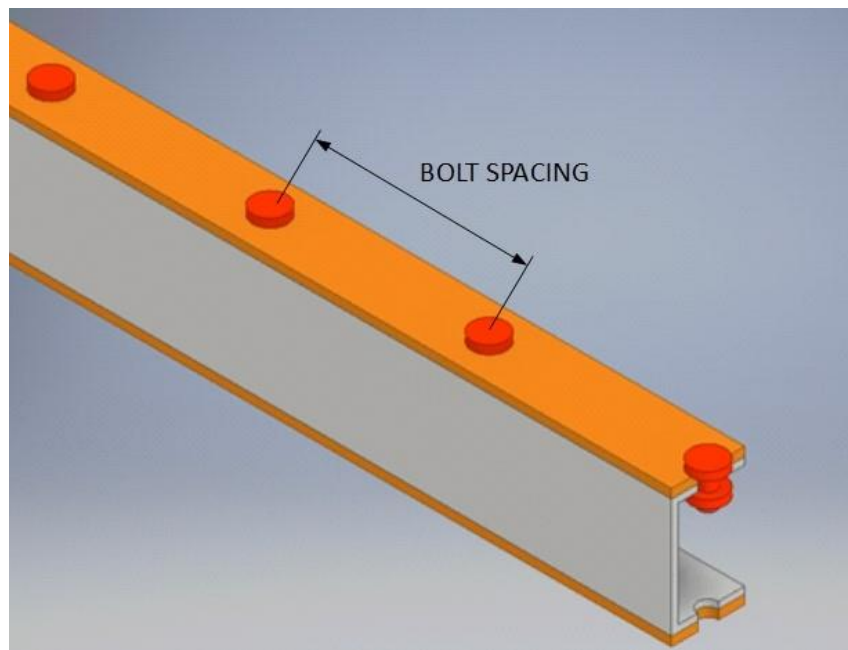


Figure 21. Spacing of the HuckBolts on a C-channel joined with a top and bottom plate.

#### 4. Numerical Analysis

After the general concept and shape of the final design was established, it was necessary to determine the actual dimensions of each of the different components. In order to find the optimal component sizes, finite element analysis (FEA) was performed on models of the current design and the new neck design created by the design team. This section describes how the current and new neck designs were modelled and summarizes the results of the FEA.

According to client requirements previously described in Section 1.4, the new neck structure must be just as strong as the current neck structure. The approach to fulfill this requirement was to ensure the new neck design was capable of withstanding the same max load as the current design. Therefore, FEA was performed on the current design to determine these maximum

loads. The maximum load obtained from the current design was applied to the new design model and the dimensions of the different components were adjusted to ensure the structure met strength requirements and minimal weight.

The pre-processing of structure geometry was performed from CAD software Autodesk Inventor. The geometry was imported to ANSYS Workbench 17 for further preprocessing. ANSYS Mechanical was used for all finite element calculations and post-processing.

#### 4.1. Loading Cases

The strength of the current neck design needed to be replicated in the new design. This strength was quantified using the maximum allowable load before structural yield. Three different loading scenarios were taken into account during FEA to approximate real loading situations during normal trailer operation.

One of the loading cases, referred to as Load Case 1, is a vertical load applied to the hitch as shown in Figure 22. This loading scenario approximates the bending that the neck experiences during the normal loading and unloading of the trailer. This loading case also represents a scenario wherein a bump in the road causes an upward force to be transmitted onto the trailer.

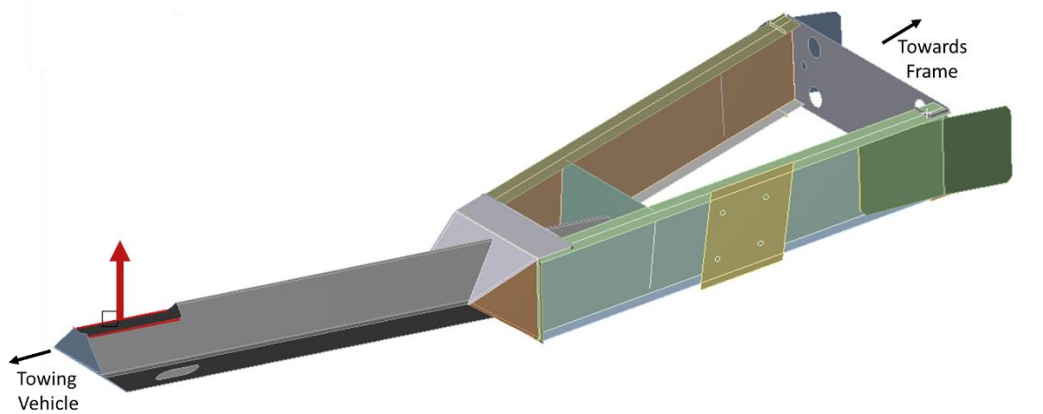


Figure 22. Location of vertical load for Load Case 1 representing bending scenario of normal operation.

Load Case 2 is similar to the first loading case in that they originate from the same location. However, Load Case 2 is oriented perpendicularly in a horizontal direction as shown in Figure 23. This loading scenario represents any oblique force-components of the same forces represented

by Load Case 1. However, Load Case 2 also represents any loads experienced by the neck as a result of sharp turns during road travel.

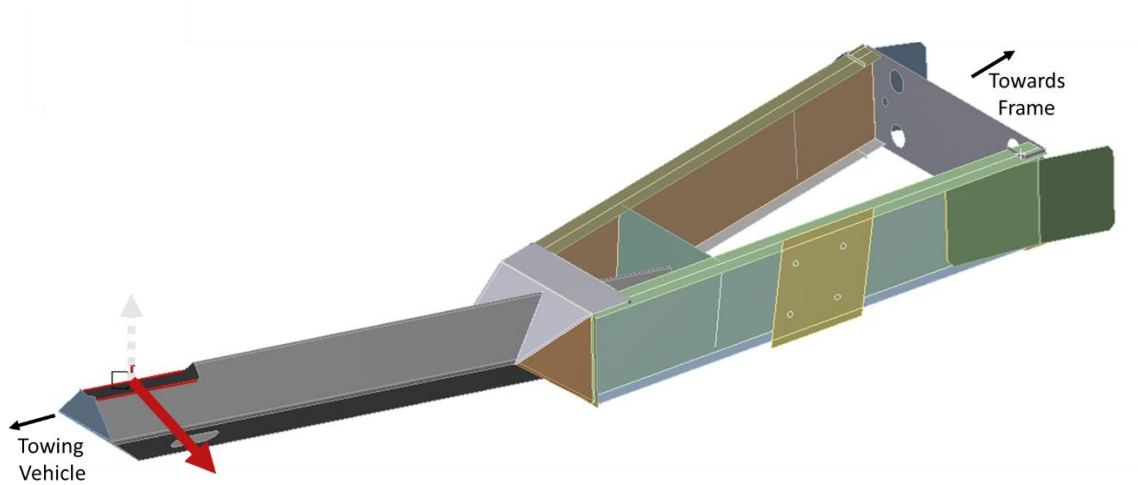


Figure 23. Location and orientation of force from Load Case 2.

The last loading scenario considered, Load Case 3, is applied at the location shown in Figure 24. This load is oriented horizontally with respect to the ground, however the load is located at approximately where jackknifing is expected to occur. The exact location of the jackknifing load with respect to the hitch is described in section 3.1.

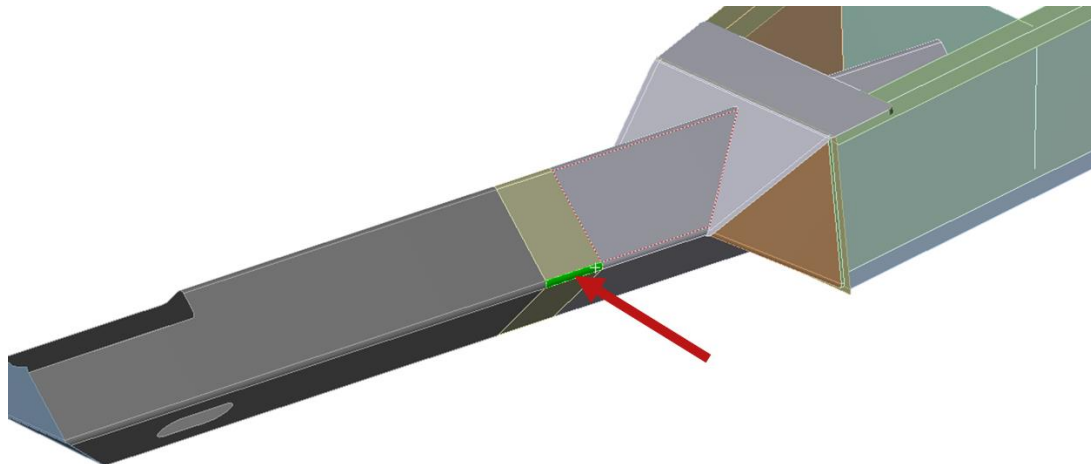


Figure 24. Location and orientation of force from Load Case 3; bright green surface represents the area of application of load.

## 4.2. Current Neck Design Finite Element Model

The strength of the current neck design was measured using the maximum allowable force for each of the three different loading cases. A finite element model of the current neck design was created to determine these maximum allowable forces. This subsection describes the details of the current neck design model.

### 4.2.1. Geometry

The CAD model of the entire trailer for the current design was provided by Cancade. The components considered to constitute the neck portion of the trailer are shown in Figure 25. Details regarding the classification of the different regions of Cancade's pup trailer can be found in section 1.1.

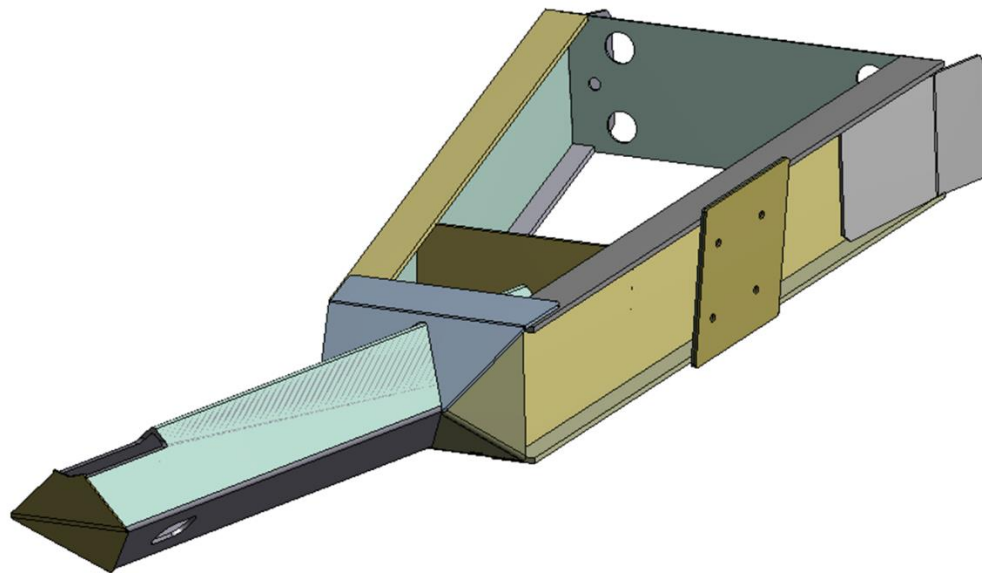


Figure 25. Overview of all components that constitute the “neck” during FEA.

In order to minimize computational time during analyses, all components in the neck assembly were modelled as shell elements. The inaccuracy that resulted from modelling the components as thin shells was considered negligible as all the components in the neck assembly are thin-shaped plates. Each of the components have thicknesses less than 1/20th of magnitude of its other dimensions, which is generally considered the minimum required for using shell elements [10].

### 4.2.2. Material Definition

For all finite element models created, all the bodies were defined as isotropic linear-elastic steel. The specific material used for each body and their corresponding material properties are listed in Appendix B. A material model that incorporated plasticity was unnecessary because the design team considered material yielding to be the point of structural failure. Furthermore, due to the uncertainty in the loading conditions for each trailer, an extra degree of safety is advantageous and reduces the risk of sudden failure during extreme conditions (such as overloading the trailer). Therefore, if the new neck structure was designed for yield, the maximum allowable load is also determined from the yielding of the current neck design.

### 4.2.3. Body Connections

The different components (or bodies) of the neck assembly are welded together in the real trailer. In the numerical model, all the different bodies are connected to each other by shared nodes as shown in Figure 26. The shell bodies share nodes along an edge, which represents an unrealistic connection between different bodies in terms of geometry and in the strength of the connection. In reality, the bodies should be attached along the geometry of the weld and the weld itself would behave differently from its parent material. However, due to the complexities associated with weld modelling as well as the time limitations of the project, high-fidelity modelling of connections between bodies was considered impractical. The single-edge shared node connections used in the neck assembly is considered to be a conservative method as it is a stronger type of connection compared to reality.

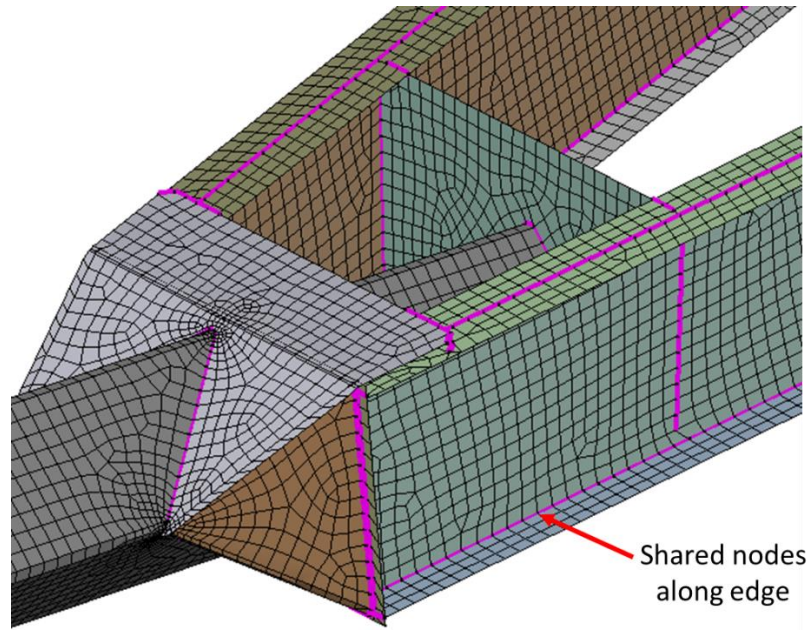


Figure 26. Shared-node connections between different components, indicated by pink lines.

#### 4.2.4. Boundary Conditions and Applied Loads

The current neck design model is loaded according to the three load cases described in section 4.1, in addition to a global gravity acceleration. For Load Case 1 and 2, the hitch load is applied to the edges shown in Figure 27, which is where the tube is welded to the hitch subassembly in reality.

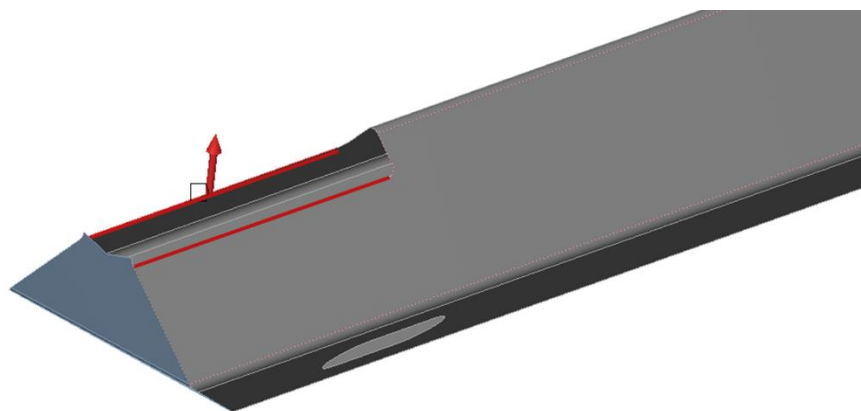


Figure 27. Location of edges (red line) for applied load in Load Case 1 and 2.

For Load Case 3, the load is applied over a length of 5 inches along the edge shown in Figure 28. The length is an approximation of the area of the impacting tire onto the tube. The jackknifing load is applied approximately 50 inches from the location of the hitch, obtained from section 3.1.

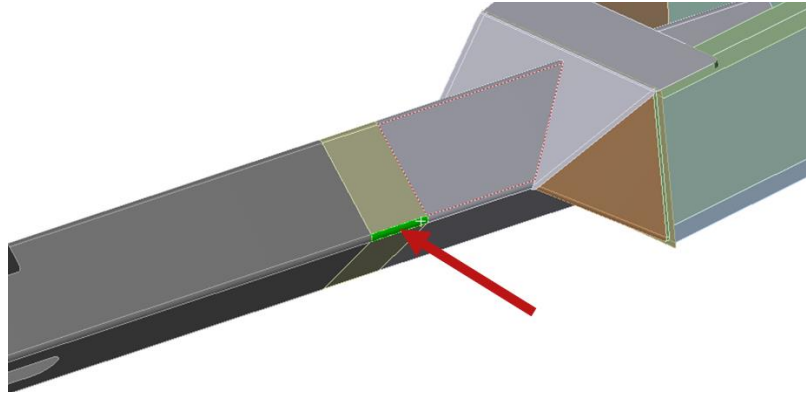


Figure 28. Green rectangular area for the applied jackknifing load.

For all load cases, the neck structure is fixed in space (displacement of zero along all axes) at the edges shown in Figure 29. These fixed edges are where the neck structure would normally be attached onto the rest of the trailer by welding.

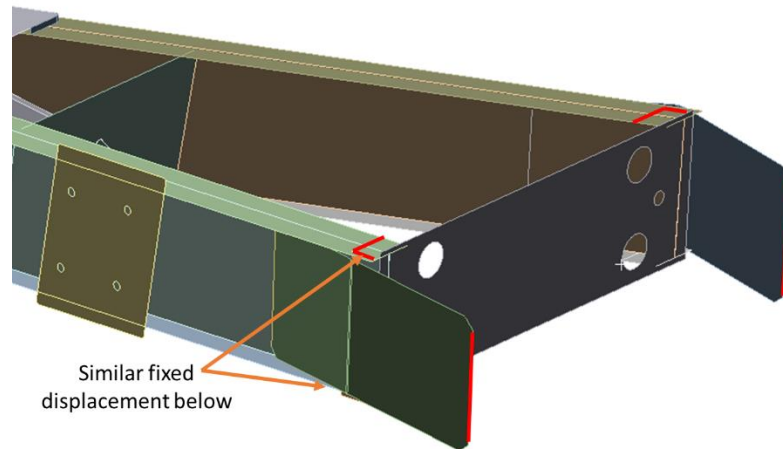


Figure 29. Zero displacement boundary condition along the red lines on the rear of current neck design.

#### 4.2.5. Mesh

For the entire model consisting of only shell elements, the mesh contains only quadrilateral or triangular elements. Due to the complexity of meshing procedures, ANSYS Mechanical was used

to automatically generate the mesh for the model. The default automatic method was found to be sufficient for generating a mesh with minimal problems (skewed elements, poor aspect ratio, etc.), because all the bodies consisted of mostly flat-surface shells.

Quadrilateral elements were given priority during the automatic meshing due to triangle elements generally having lower accuracy [10]. The majority of the elements in the different bodies are quadrilateral as shown in Figure 30, with some triangle elements placed in the complex geometry regions such as near the tube connection.

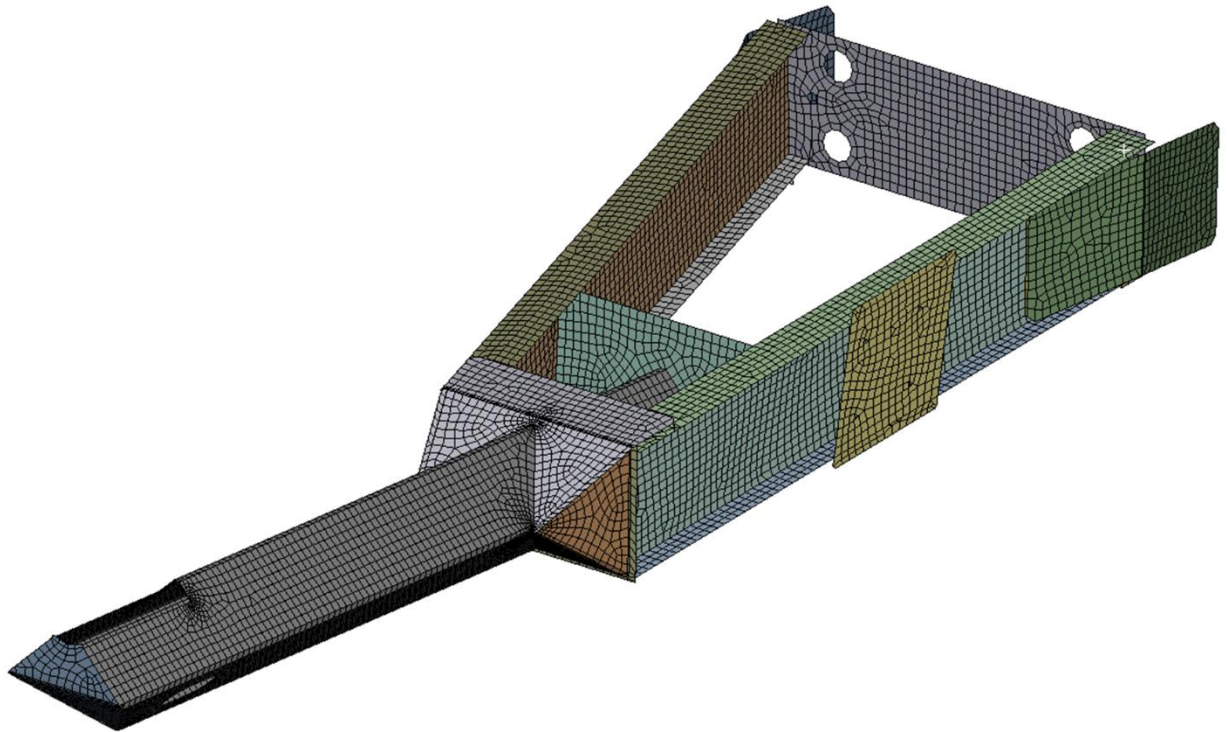


Figure 30. Sample mesh of neck structure; element size increased for visualization.

The quality of the shape of the elements in the model were checked by visual inspection to ensure that no elements had a skewed shape. However, no strict study of element quality was performed. The maximum element size for the whole structure was determined by a stress convergence study of five points (P1-P5) shown in Figure 31.

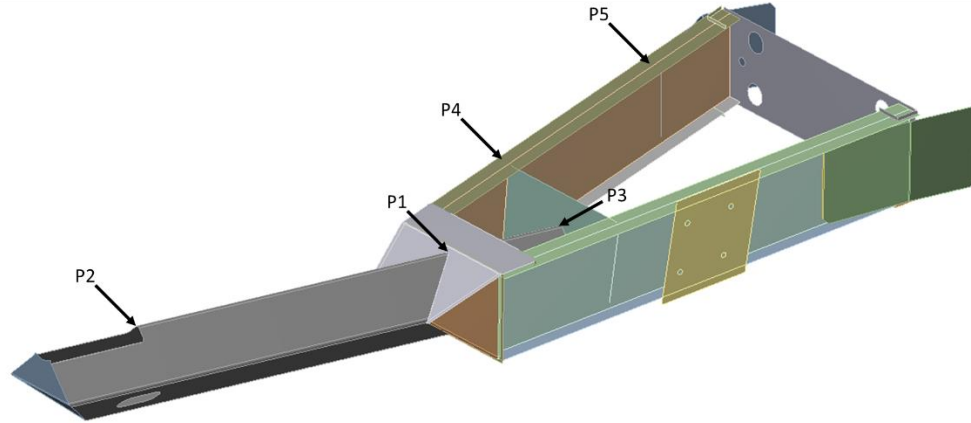


Figure 31. Location of stress points (P1-P5) used for measuring stress convergence.

Figure 32 shows the stress of point P1 for Load Case 1. The current neck model was found to have converged for stress after approximately 40000 elements were used. This process was performed at an intermediate stage, therefore the stress values shown in the figure do not represent final values. However, only the convergence of these values are important.

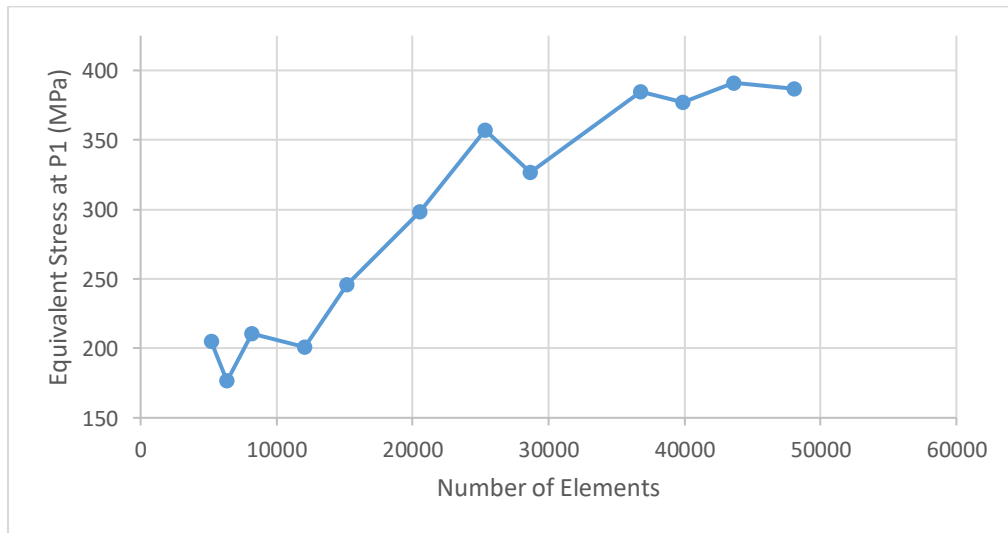


Figure 32. Convergence of stress at P1 as number of elements were increased.

The stress was considered to have converged if there was a difference of equal to or less than 5% between consecutive results at each point. The final mesh size was chosen from among the meshes where all five points have converged (after 40000 elements). The entire mesh convergence procedure was repeated for all load cases. The stress convergence for all points and for each load case can be found in Appendix C.

### 4.3. Results for Maximum Allowable Load

The magnitude of the load for each of the three load cases was varied through trial and error until the maximum equivalent stress was just below the material yield.

An overview of the distribution of safety factor (SF) of the current neck model is shown in Figure 33 to Figure 35 for Load Case 1. SF is defined as the ratio of the yield limit of the material to the equivalent stress result (von Mises). Therefore, SF is equal to one indicates a material that has just begun to yield. Since the current neck is made of several materials, safety factor conveniently shows which regions are closest to yielding.

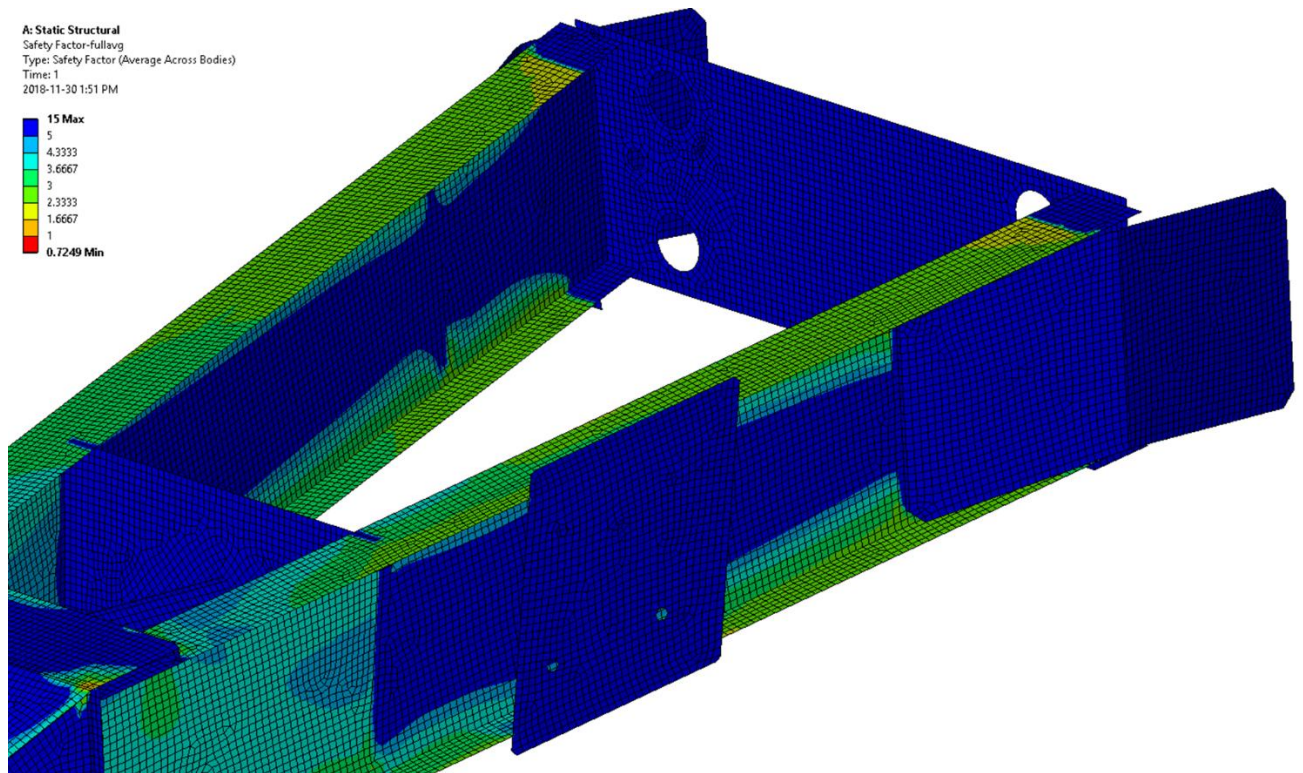


Figure 33. Contour of safety factor in rear of current neck design for maximum allowable load (Load Case 1).

The stress results for nodes shared between two bodies is averaged because all bodies are attached to each other at shared nodes. Overview results for the other load cases are shown in Appendix D.

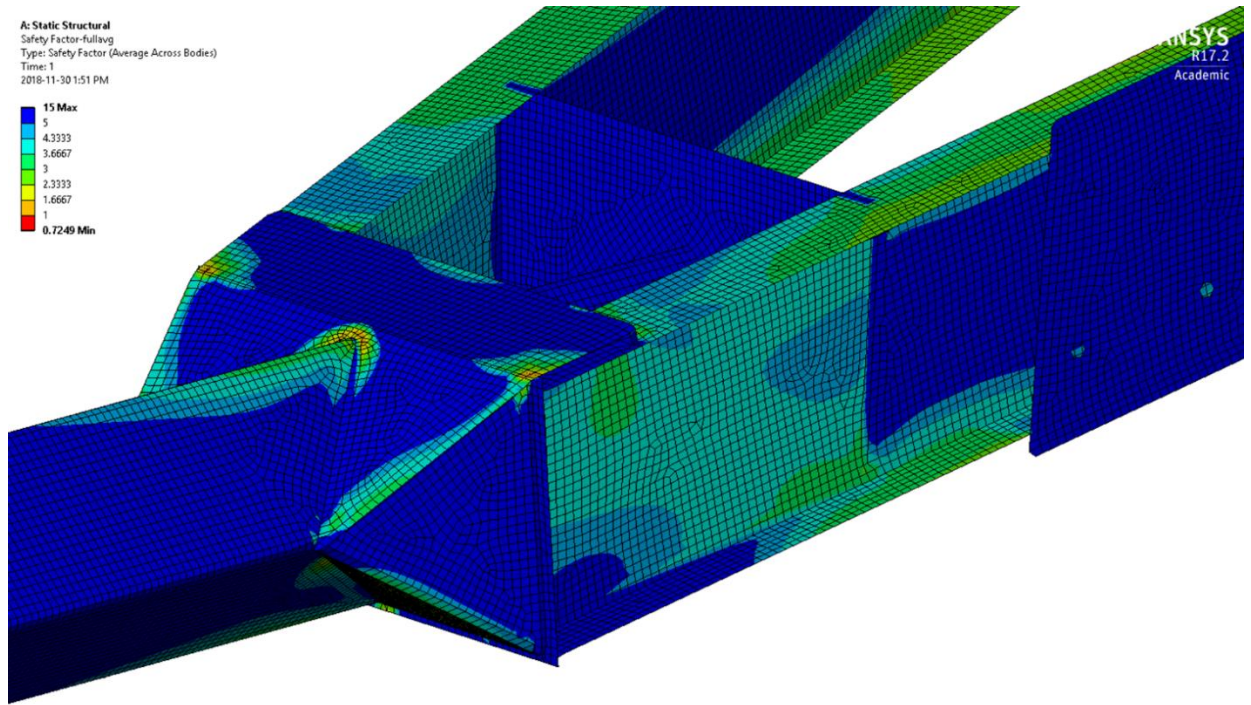


Figure 34. Contour of safety factor for middle portion of current neck design (Load Case 1).

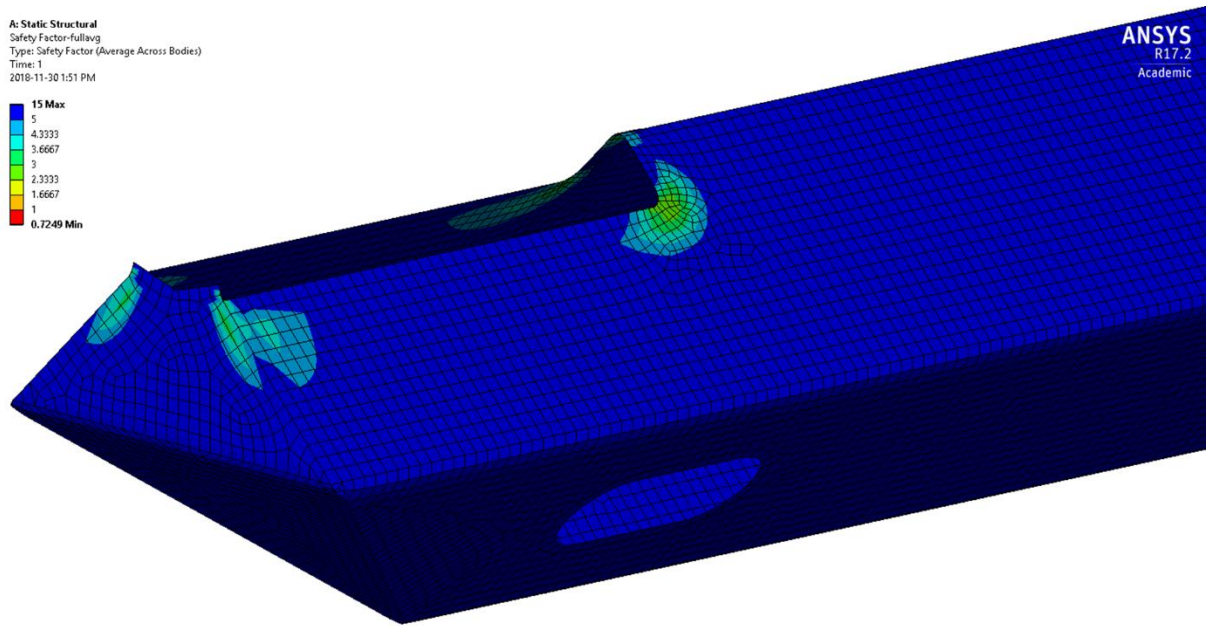


Figure 35. Contour of safety factor for end of tube region (Load Case 1).

The FEA yielded results with minimum SF of less than one, which indicated regions of material with equivalent stress above the yield limit. The issue with these regions, shown more closely in

Figure 36, was that they were stress singularities. These singularities are highly localized regions where stress increases without bound depending on the local element size. Therefore, they are not present in reality. Normally, the geometry of the region could be modified to remove these singularities. However, the complicated geometry of the offending regions made it impractical to fix due to the time constraints of the project.

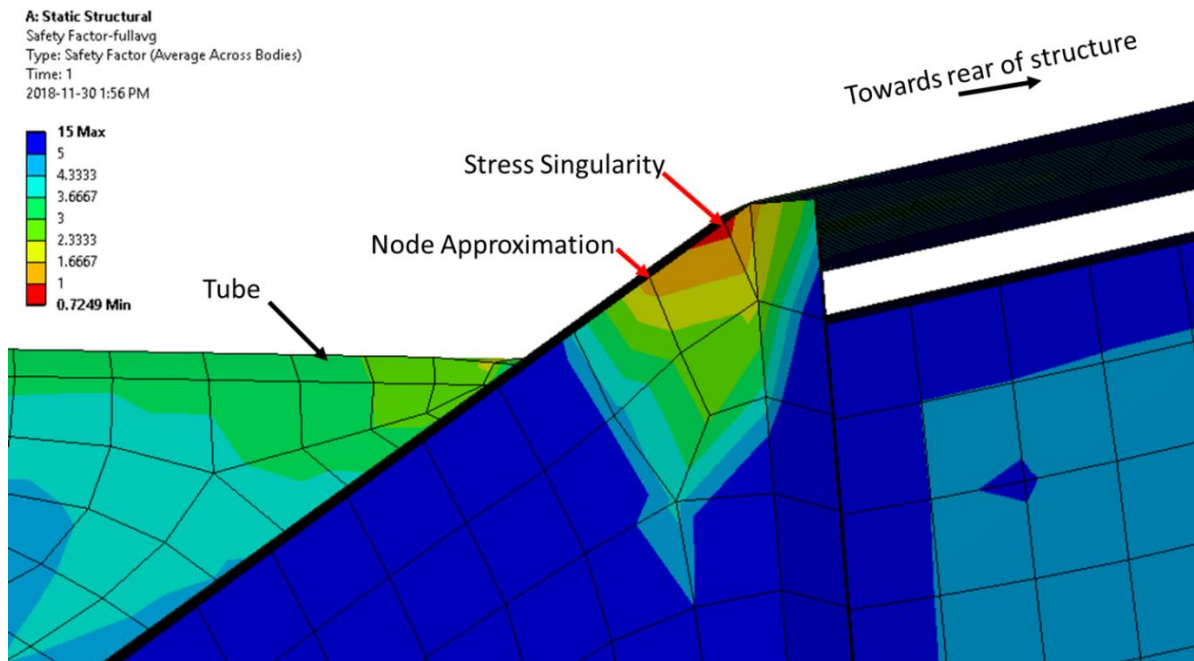


Figure 36. Location of stress singularity for Load Case 1.

The non-physical results due to the singularities were left in the model. Instead, the equivalent stress at a location of one element away from the stress singularity was used to approximate the real stress at that region.

Using the approximated maximum stresses, the maximum allowable loads were determined and summarized in Table IV for all load cases.

TABLE IV. MAXIMUM ALLOWABLE FORCE FOR EACH LOAD CASE.

	Load case		
	1	2	3
<b>Maximum allowable load, N</b>	32500	30000	38000
<b>Maximum allowable load, lbf</b>	7306.3	6744	8543

#### 4.4. New Design Finite Element Model

The new design, shown in Figure 37, was constructed differently from the current design with regards to its use of HuckBolts to replace most of the welds. Due to this, the numerical model of the new design was different from the current design with regards to how the connections between different bodies were handled. The details of the finite element model of the new design is described in this section.

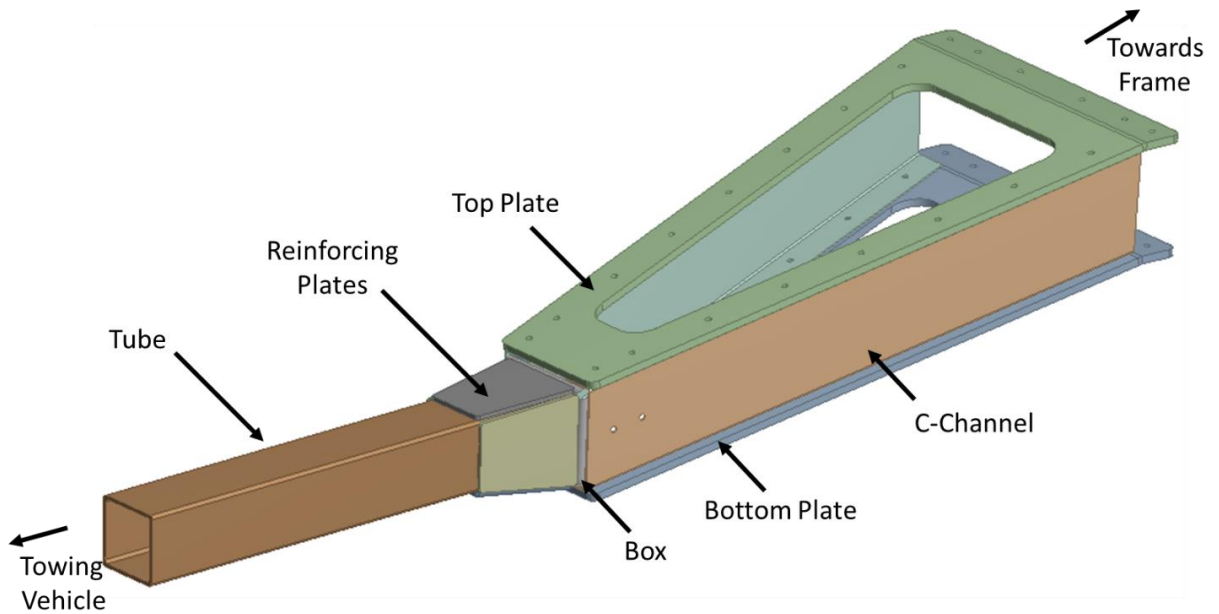


Figure 37. Overview of new design model.

#### 4.4.1. New Design Geometry

The geometry of the new design for use in FEA was taken directly from the CAD model described in section 3.

Similar to how the current neck design was modeled, all bodies in the new design were created using shell elements. The primary reason for the use of shell elements was to reduce the computational time needed to perform the analyses. All components in the new design were assumed as thin bodies with thicknesses less than 1/20th of its other dimensions. The assumption of thin-plate and the use of shell elements are later justified in the results of section 4.6.

The geometry of the tube-box reinforcing plates, as shown in Figure 38, was modified in order to achieve stress convergence in the model. The details of the reinforcing plates and the justification of their addition are later described in section 4.6.1.

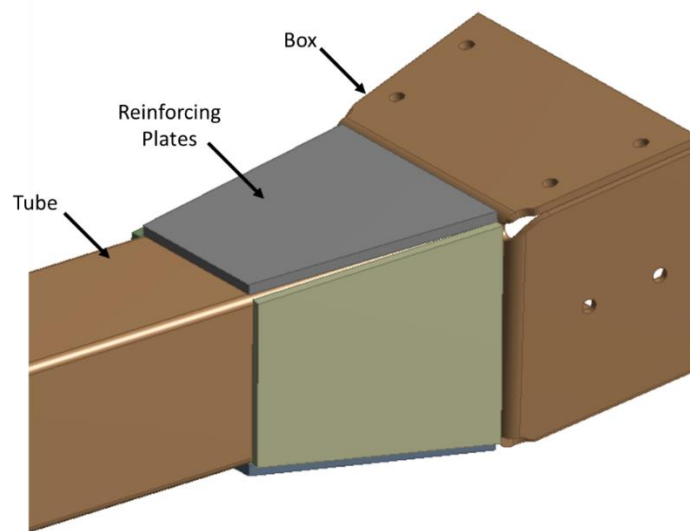


Figure 38. Reinforcing plates placed at tube-box connection.

If the geometry that was used in FEA was unaltered, stress singularities would form in the corners as shown in Figure 39.

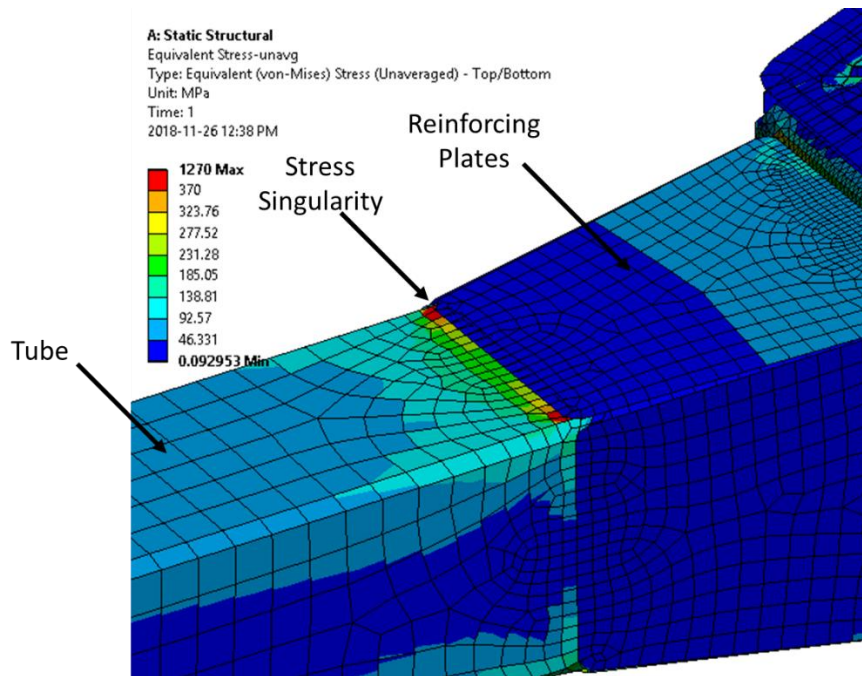


Figure 39. Location of stress singularity where corner of plate intersects with tube.

To rectify the problem, the plates were merged into a single body to form a sleeve-like structure shown in Figure 40. Since extra material was added to the reinforcing plates, the sleeve structure was expected to be slightly stronger. However, the design team considers the inaccuracy in stress results due to singularities a much larger problem than the modified geometry. Therefore, the modification to the plates was considered to be an acceptable change.

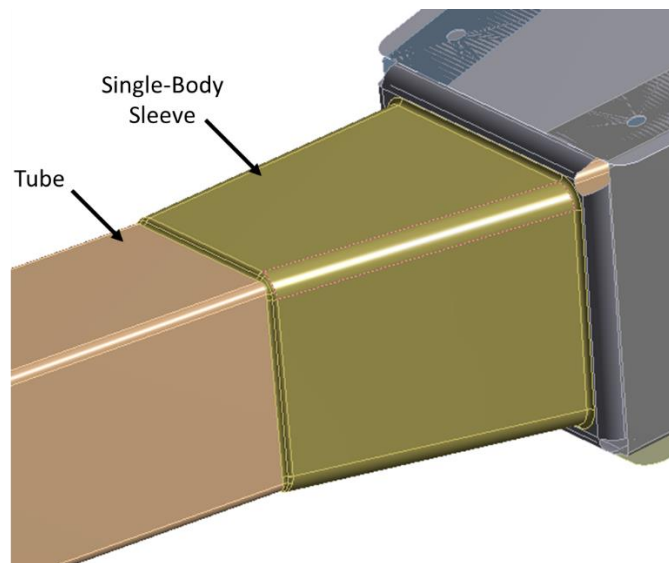


Figure 40. Sleeve component used as an alternative model to the discrete reinforcing plates.

### 4.4.2. New Design Material Definition

With regards to material assignments, all bodies in the model use a isotropic linear-elastic material definition for mild steel. The material property details are listed in Appendix B. Linear-elastic material model was used because the design team considered material yield as structural failure. Furthermore, the maximum allowable load applied to the new design model was also obtained using material yield. More details regarding the justification of linear-elastic material model is described in section 4.2.2.

### 4.4.3. New Design Body Connections

The connections between the different bodies of the new design consist of both welds and HuckBolts. The welded connections were modelled in a similar way to the current design: with single-edge shared nodes. These shared node connections were made in two main regions: the tube-box weld shown in Figure 41 and the tube-sleeve welds shown in Figure 42.

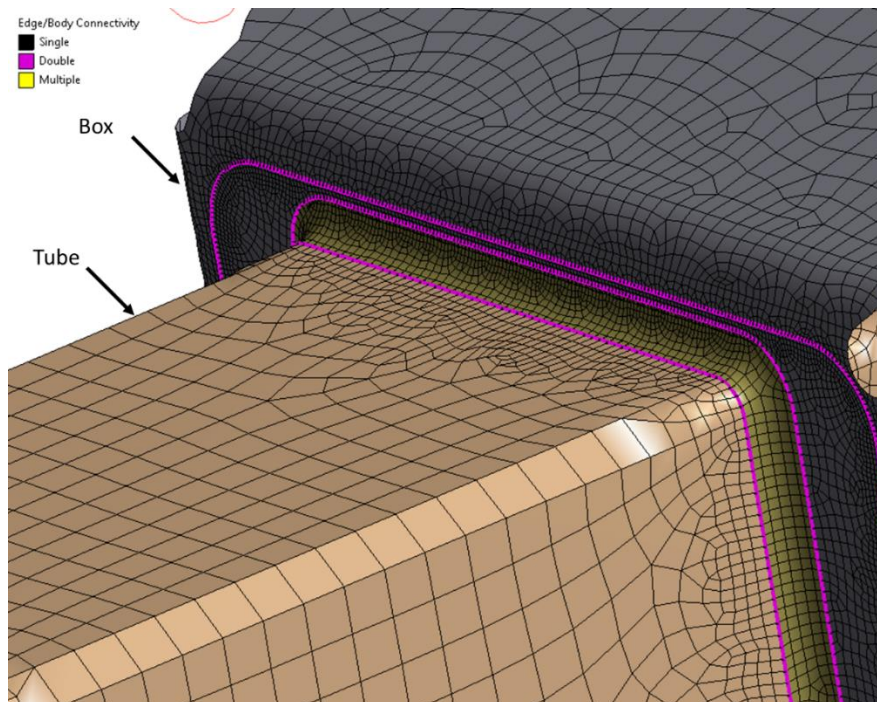


Figure 41. Shared nodes between tube and box components along the pink colored edges.

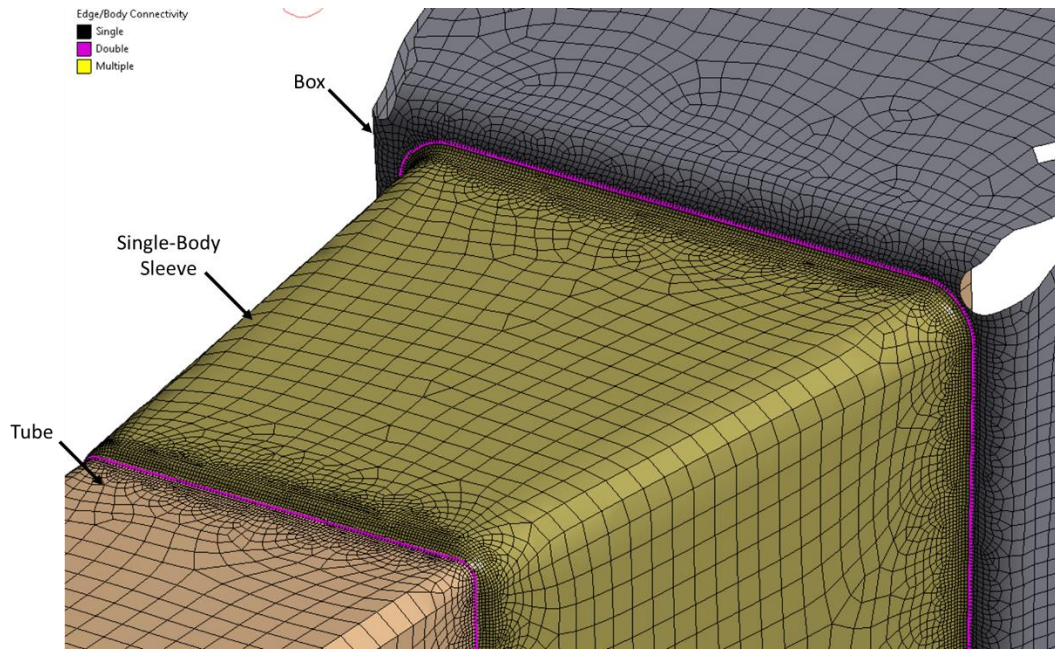


Figure 42. Shared nodes between tube and sleeve components along the pink colored edges.

The implementation of these shared nodes feature fillets that approximate the geometry of the weld. However, the fillets were just an extension of the parent body (a shell) and had the same material properties. The main reason for the fillets was to create a smooth intersection between the two bodies to avoid possible sharp corners.

The bolted connections throughout the rest of the structure were modelled using multi-point constraints (MPC) between the edges of the holes of each bolt-hole pair. A bolt-hole pair consists of two bolt openings from two bodies that are adjacent, as shown in Figure 43. The MPC connection constrains together the nodes that constitute the edges in each bolt-hole pair such that they have the same displacement (via constraint equation). The benefit of this connection method is that it is computationally efficient, which is advantageous as there are over 50 different bolt-holes in the new design.

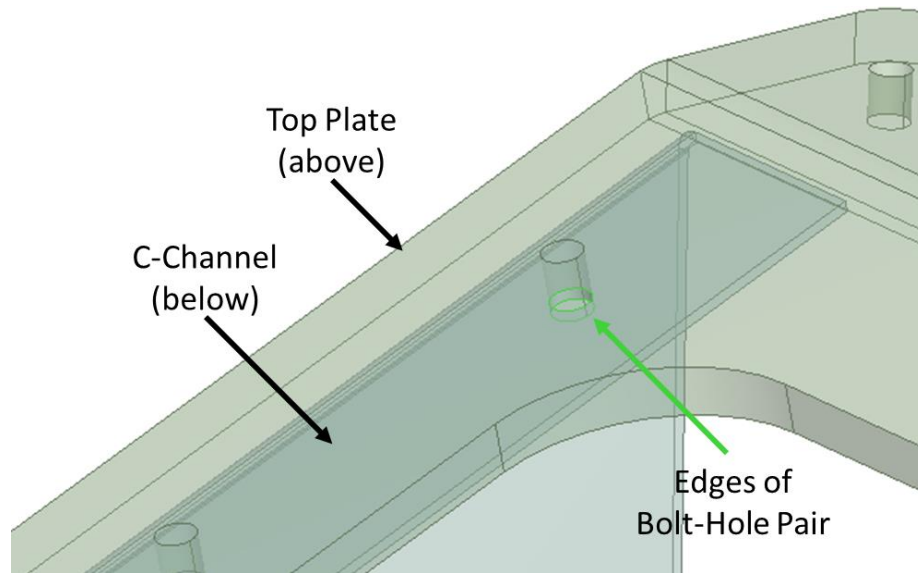


Figure 43. Example of bolt-hole pair and the edges used to define MPC connection.

A limitation of the MPC method to model bolted connections is that the clamping force and the resulting friction between the bodies are not represented. The design team considers the friction effects to be minimal, however, the exact impact or significance of this effect is unknown and future work is required.

#### 4.4.4. New Design Boundary Conditions and Applied Load

The boundary conditions and the loading of the new design model was made similar to the current design model in order to allow comparisons of strength to be made.

The maximum allowable load obtained from section 4.3 was applied to the new design model according to the three load cases described in 4.1. For the hitch loading (Load Case 1 and 2), the load was applied and distributed to the surface shown in Figure 44.

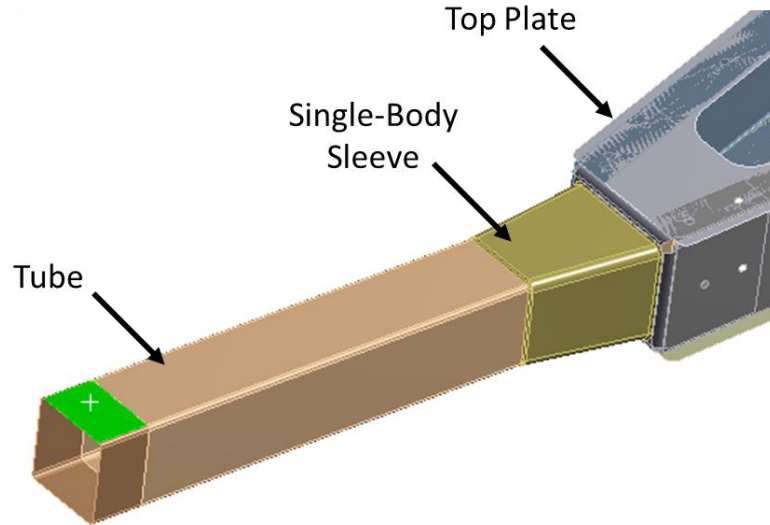


Figure 44. Area of application of load (green surface) for new design model.

The new neck design described in section 3 does not incorporate the geometry of the hitch to connect with the towing vehicle. Therefore, the application of the load onto the new design model is not realistic, as the specifics of the interface between the tube and hitch are currently unknown. However, the results of the FEA for the majority of the tube and the rest of the structure was expected to be valid due to Saint-Venant's principle. The principle states that the stresses sufficiently far from application of load are independent of the type of application of load. Since the application of the load is far away from the majority of the structure, the uncertainty of the tube-hitch interface is acceptable.

The neck was fixed in space (zero displacement in all directions) at the locations where the neck was connected to the rest of the trailer frame. The neck was connected to the frame and fixed on the edges of the bolt holes shown in Figure 45.

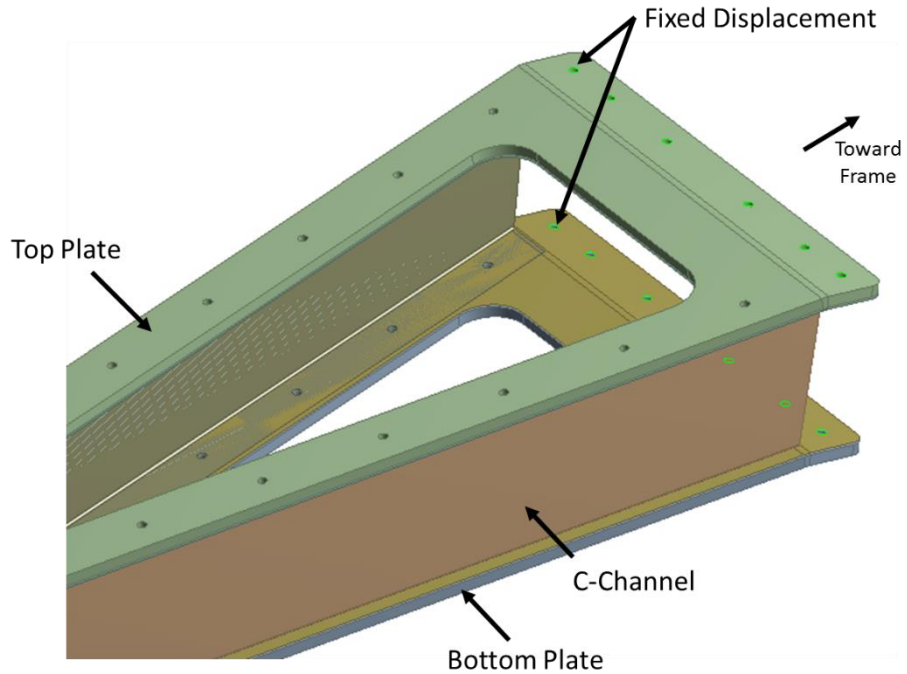


Figure 45. Fixed displacement for all bolt-hole edges (highlighted in green) at the rear of neck.

#### 4.4.5. New Design Mesh

The process to create the mesh for the new neck model was similar to the process used for the current neck model, described in section 4.2.5.

The mesh of the new neck model consisted of primarily quadrilateral elements automatically generated by ANSYS Mechanical defaults. The general mesh size of the structure was controlled by a global maximum element size. The global maximum element size was reduced until the points P1 to P5 shown in Figure 46 yielded convergent stress results.

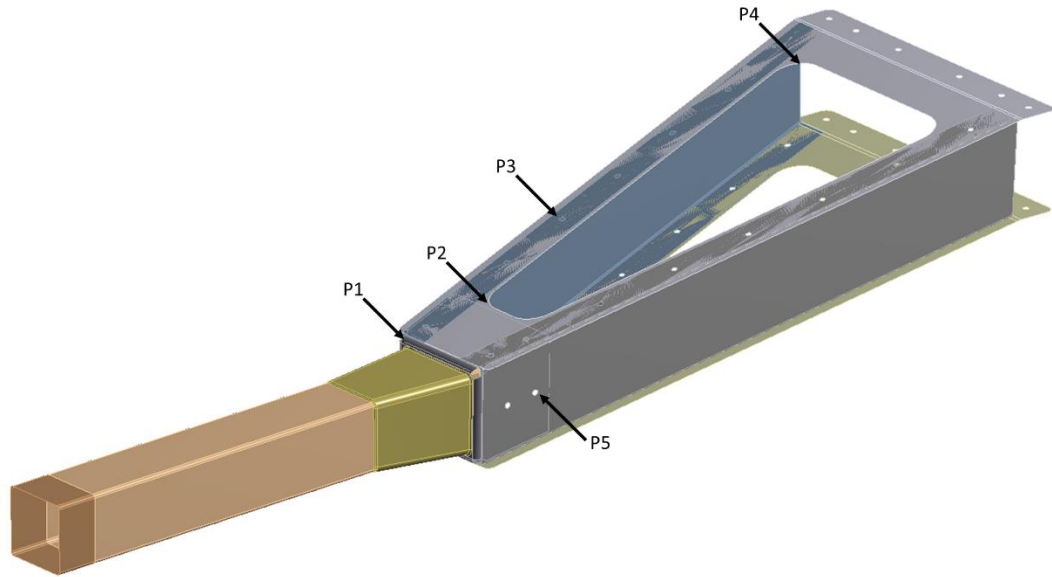


Figure 46. Points P1 to P5 in new neck model used to determine general mesh size.

Figure 47 shows the stress of point P1 as the maximum element size (and thus mesh size) was changed. The new neck model was found to have converged for stress after approximately 55000 elements were used.

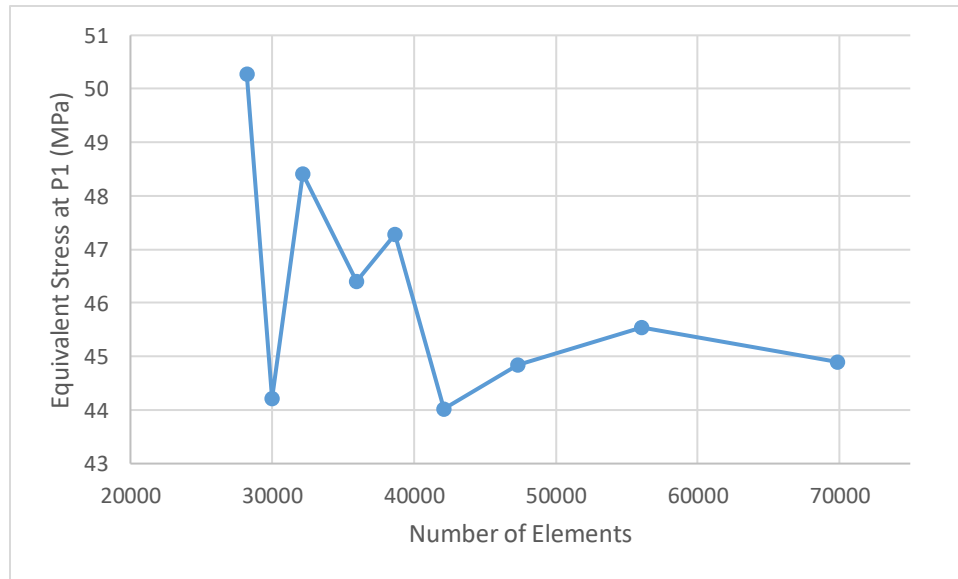


Figure 47. Convergence of equivalent stress at P1 as mesh size was changed for Load Case 1.

The stress results were considered to have converged (become mesh-independent) once the percent difference in consecutive stress results was equal to or less than 5%. The percent

differences between consecutive stress results for each point P1 to P5 are summarized in Figure 48.

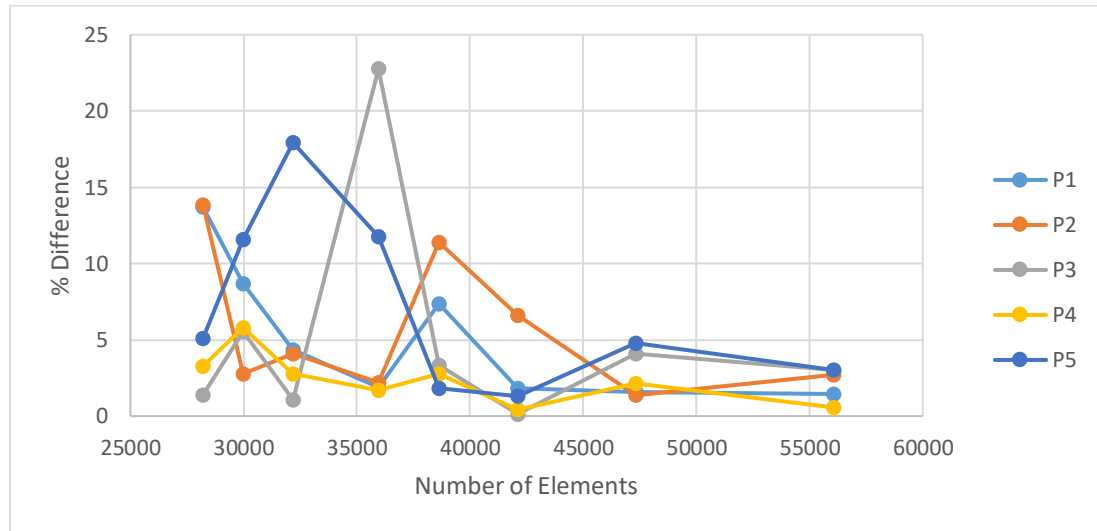


Figure 48. Percent difference between consecutive stress results for points P1 to P5 (Load Case 1).

The mesh convergence procedure was repeated for all load cases. Details of mesh convergence for Load Case 2 and 3 are shown in Appendix C.

The element size at the fillets between the sleeve and tube and between the tube and box were adjusted independently from the rest of the structure. The process for converging the small-feature fillets were the same as for the overall element size. The element sizes at these fillet locations was required to be smaller than the global element size as the fillets were a small feature and needed to be smooth. Figure 49 shows an example of the difference between the fillet mesh sizing and global sizing.

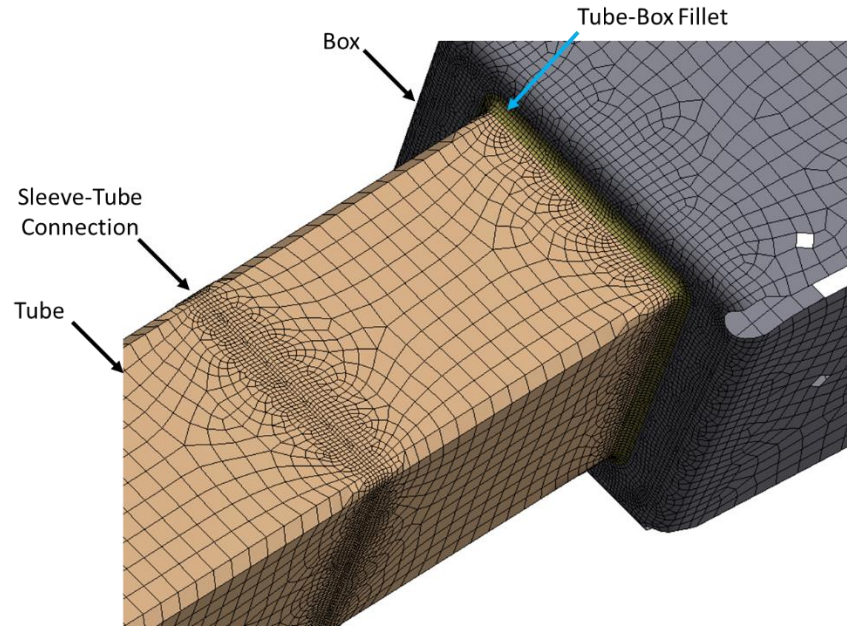


Figure 49. Smaller element sizing at fillet for tube-box and sleeve-tube (sleeve hidden).

#### 4.5. Results Overview of New Neck Model

This section provides an overview of the displacement and stress results for the optimized final design structure. Only results for Load Case 1 are shown because it has the highest stresses. Overview of results for other load cases can be found in Appendix E.

Figure 50 displays the total displacement for the whole structure. Displacement is mostly in the y-axis (vertical) direction, the same direction as the load applied for Load Case 1.

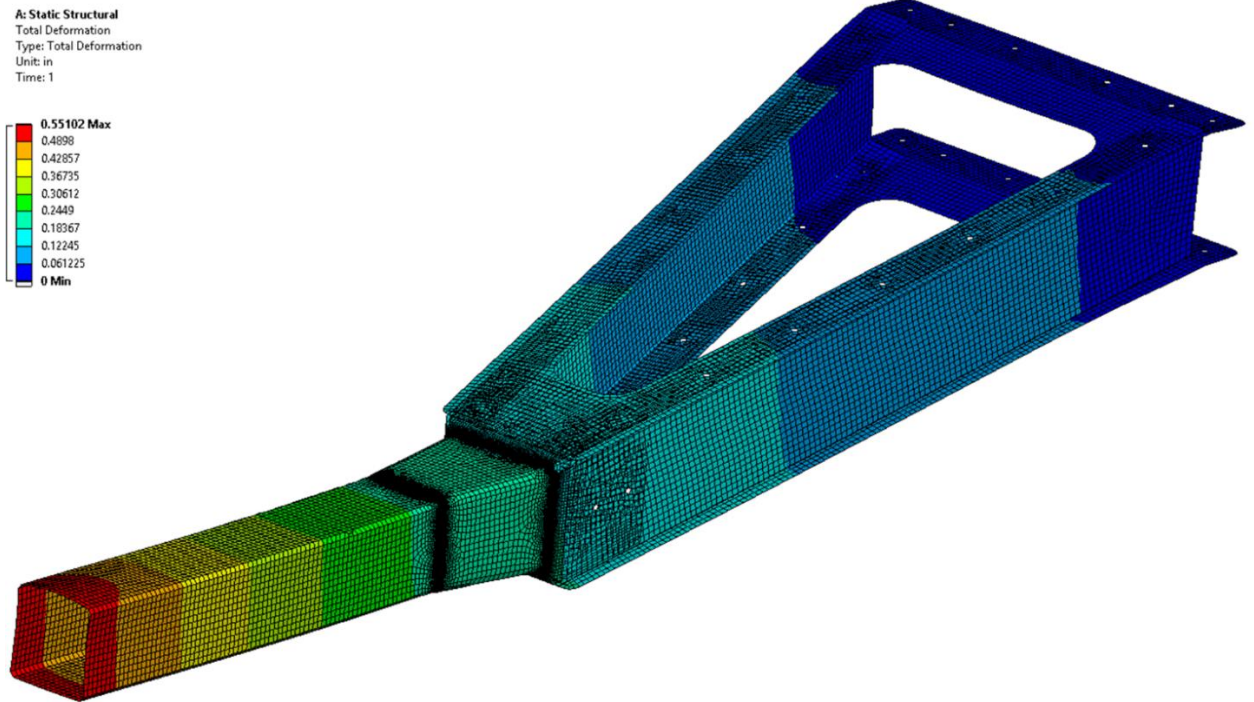


Figure 50. Total displacement in the structure (displacement scale of 12).

Figure 51 to Figure 54 shows the equivalent stresses (von Mises) of each major component in the neck structure. The stress results are averaged between shared nodes of different elements. All materials are made of 44W hot rolled steel, which has a yield point of 36000 psi.

A: Static Structural  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1

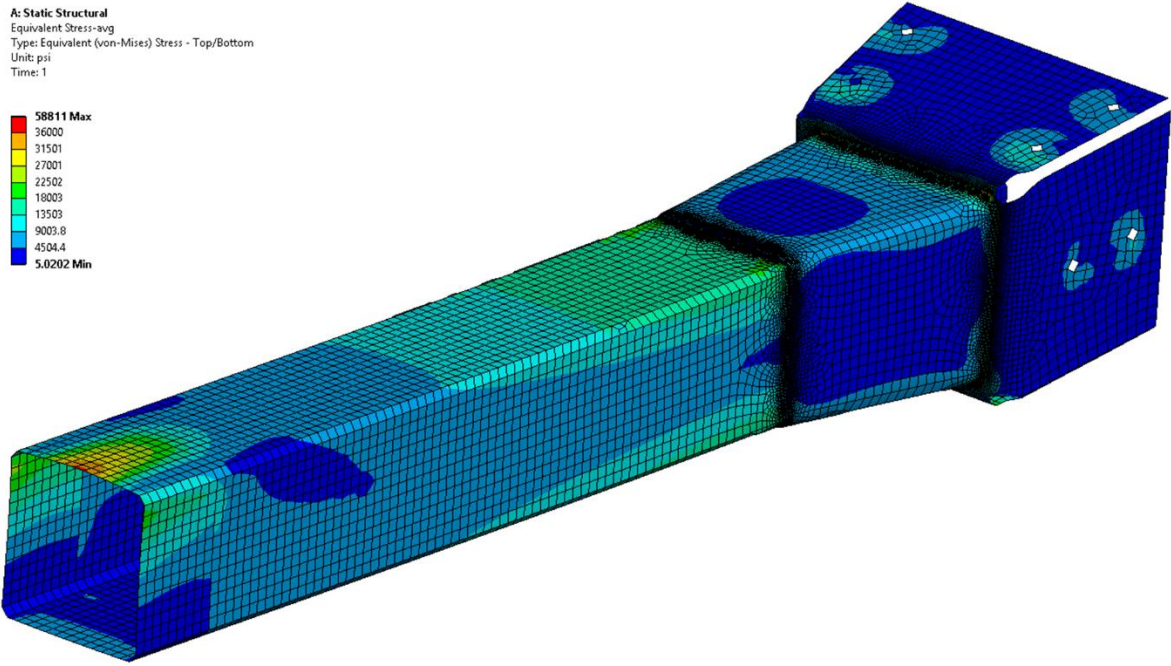
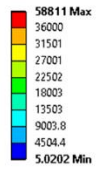


Figure 51. Equivalent stress contour of tube, sleeve and box (displacement scale of 12).

A: Static Structural  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1

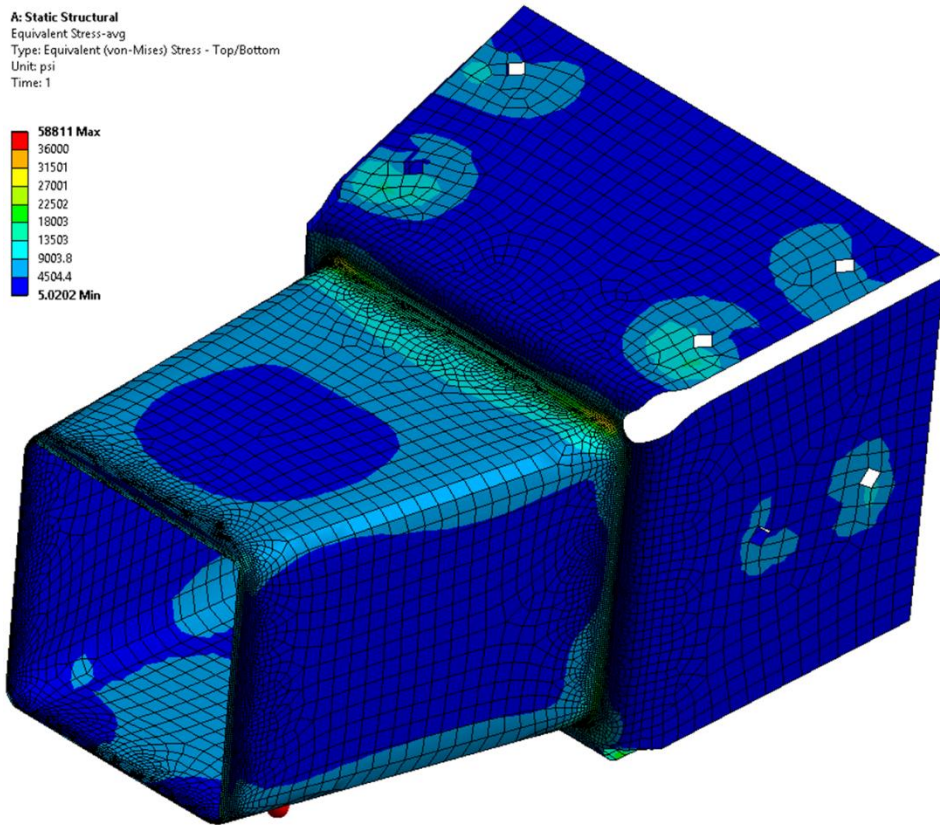
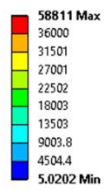


Figure 52. Equivalent stress contour of sleeve and box (displacement scale of 12).

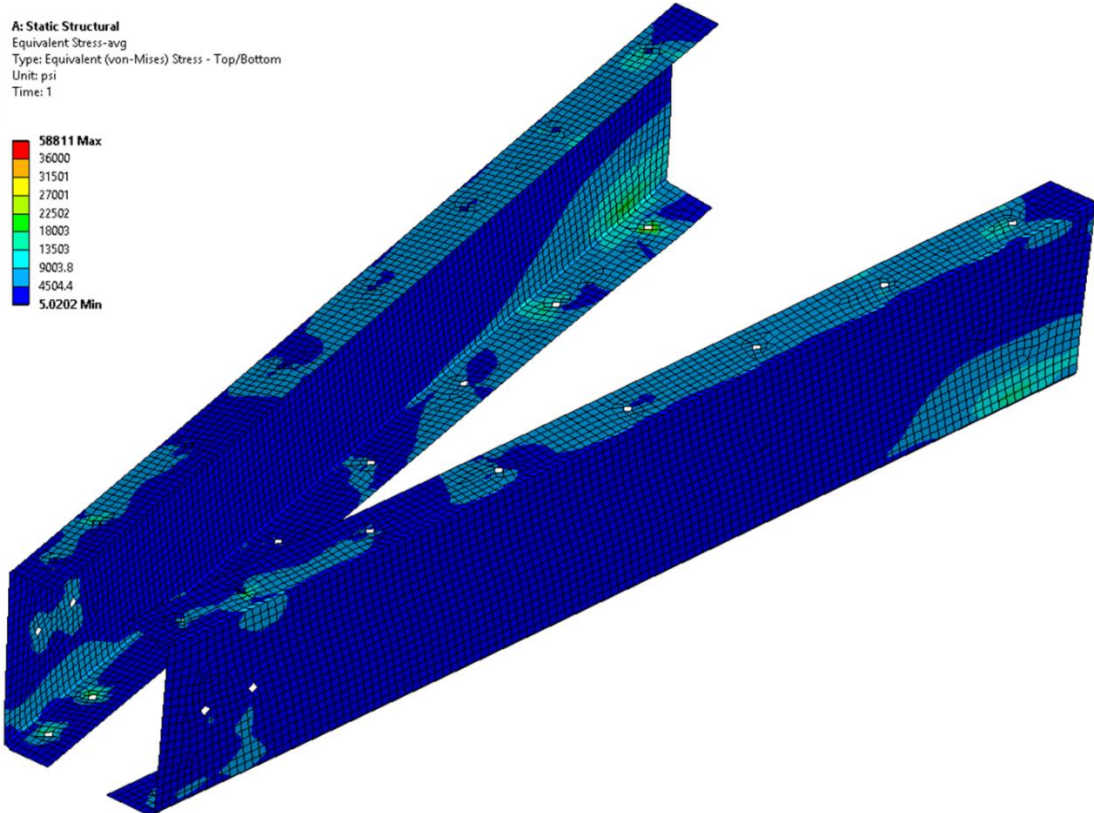


Figure 53. Equivalent stress contour of the two C-channels (displacement scale of 12).

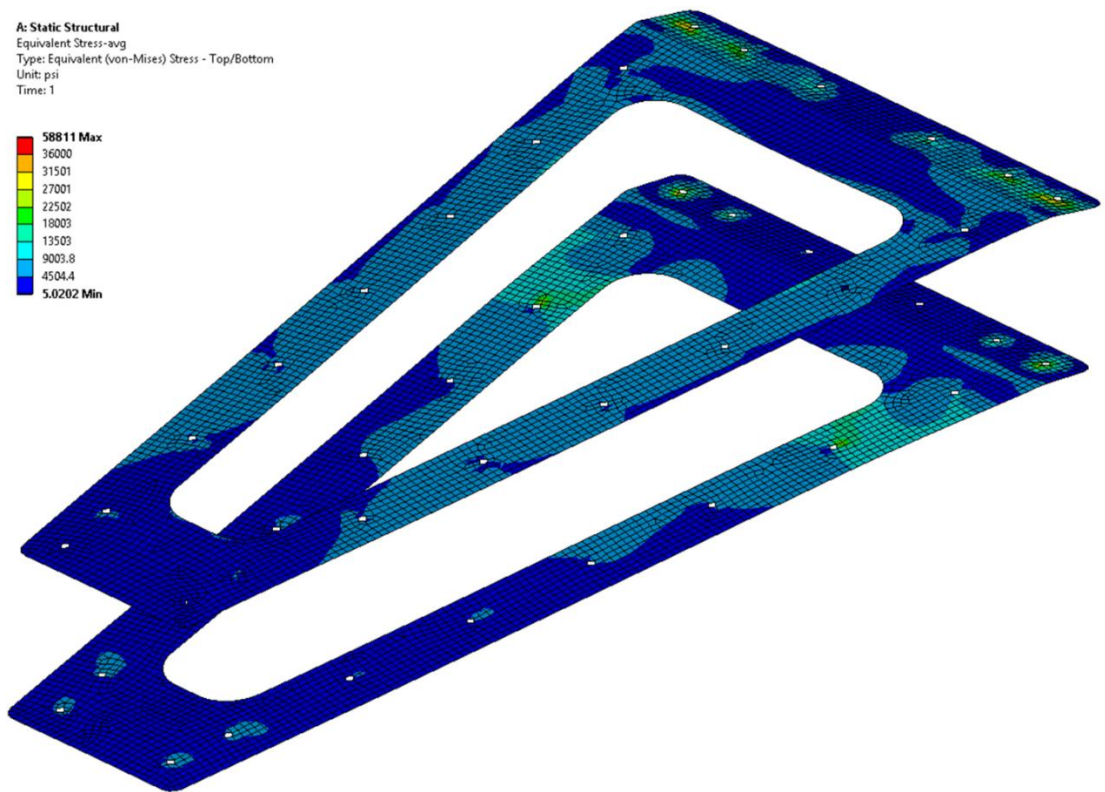


Figure 54. Equivalent stress contour of the top and bottom plates (displacement scale of 12).

The uniform thickness of each component was chosen to reduce local maximum stress to just below yield. The stress contour plots show large regions in the structure that are low stress, which indicates regions that have excessive material. If mass was a larger priority, further optimization can be performed in future work to reduce thickness in low stress regions. The structure would have lower weight but would be higher cost. For this project, cost is of higher importance than weight.

## 4.6. Optimization Procedure

FEA was used to optimize an initial neck model, which resulted in the final neck model described in section 4.4. This section describes the process that resulted in the addition of reinforcing members and the general procedure used to determine dimensions of each component.

### 4.6.1. Reinforcement of Tube-Box Connection

The design team expected the region at the connection between the tube and the box to be the weakest point in the structure for all three load cases. However, the extent and significance of the weak point became apparent only after the initial stress results (using maximum allowable load). The results prompted the addition of reinforcing members, which underwent two stages of modification.

Initial results for the tube-box connection is shown in Figure 55 for a linear-elastic material model. Structural failure at the tube-box connection was a certainty due to the high stresses in the region.

Gussets between the tube and the box were placed at each face of the tube as shown in Figure 56. However, the stresses were still severe and any changes to gusset size was considered insufficient for dealing with the stress concentration.

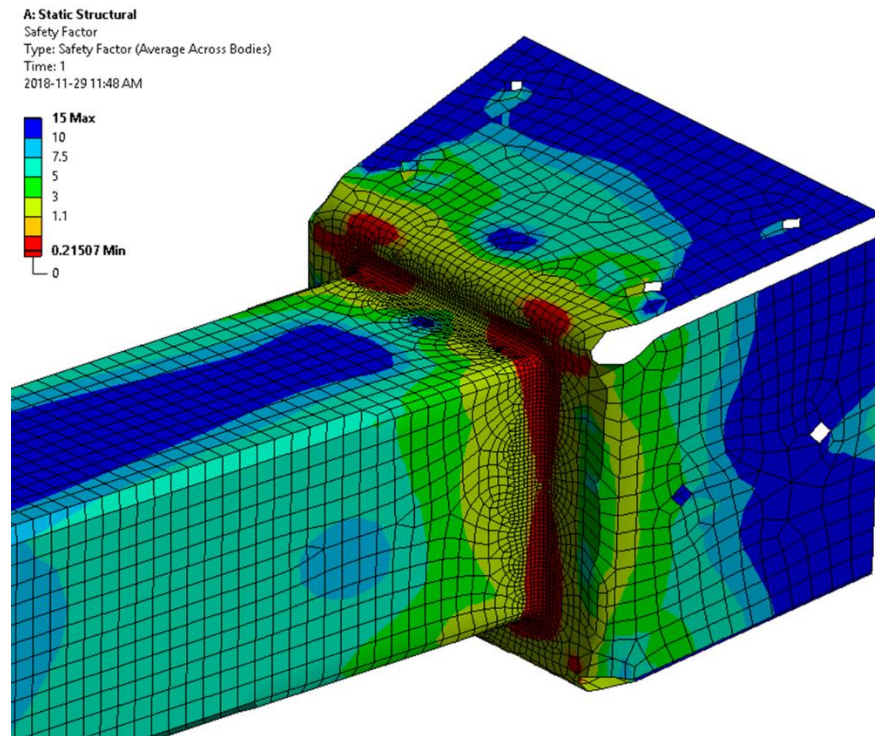


Figure 55. Contour of safety factor (ratio of material yield stress over equivalent stress) for non-reinforced tube-box.

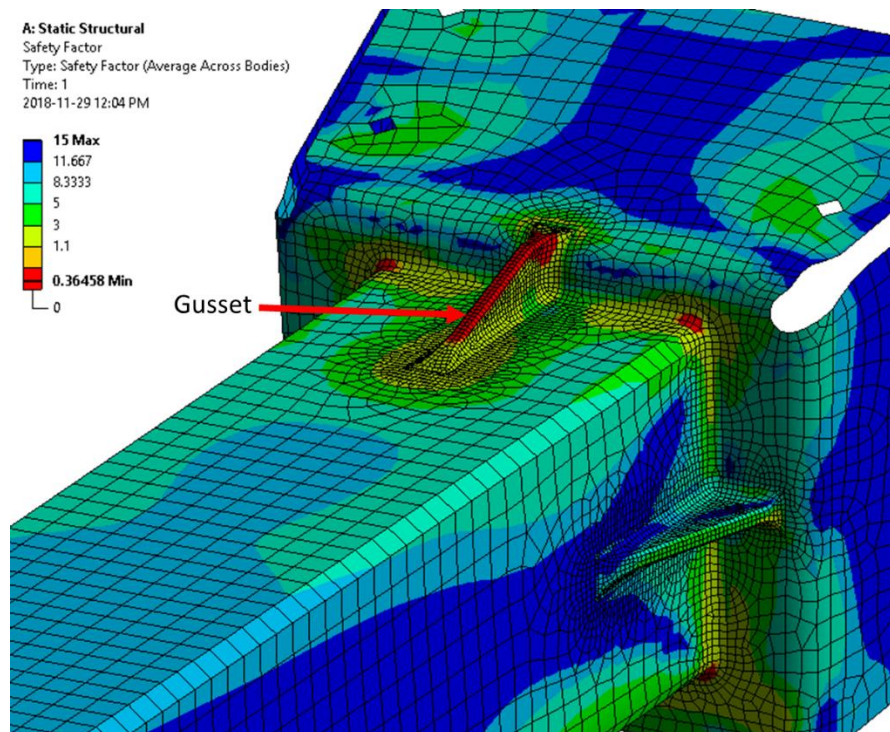


Figure 56. Safety factor results for gusset-reinforced connection.

A more drastic change was implemented in the final version of the new neck design described in section 4.4. The reinforcing gussets were replaced with four plates as shown in Figure 57. As later shown in the final results of section 4.5, these plates are much more effective at evenly distributing the load and reducing the stress results at the tube-box connection.

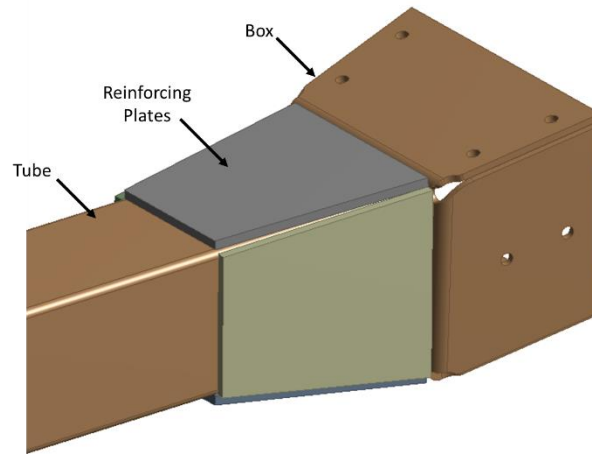


Figure 57. Reinforcing plates between tube and box connection.

The disadvantage of the addition of reinforcing members is the increase in manufacturing difficulty. There are more components to assemble and the four additional plates have to be welded onto both the tube and the box.

#### 4.6.2. Rear Bolt-Hole Plate

There was a stress concentration found during the stress analysis at the lower rear bolt-holes shown in Figure 58. The local stresses at these two locations were found to be much higher than all other regions along the C-channels.

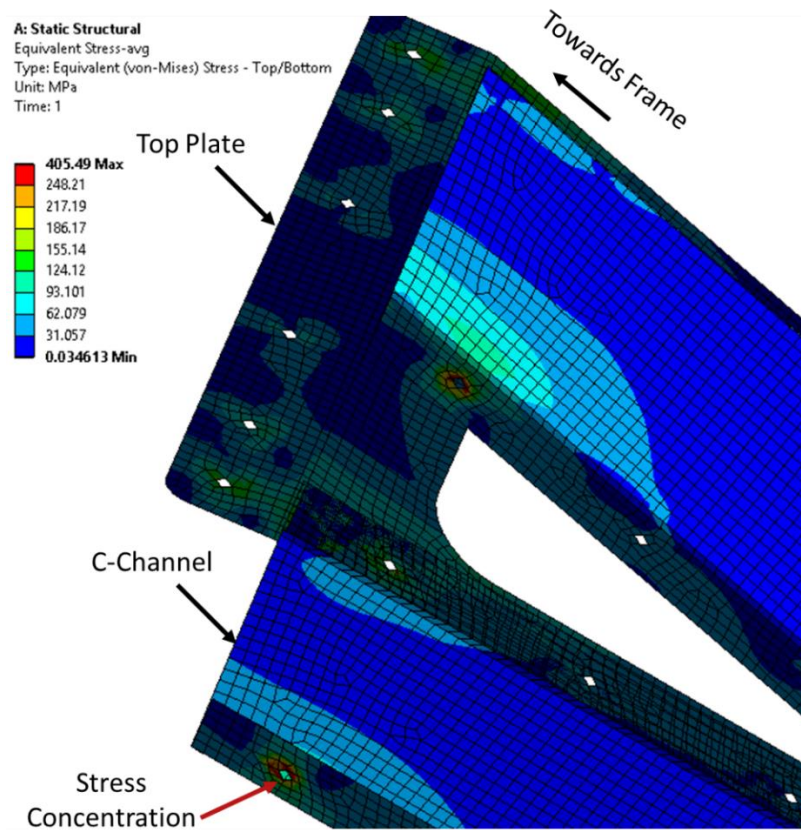


Figure 58. Stress concentration around lower rear bolt holes (bottom plate hidden).

A small thin plate was placed on the inside of the flange of the C-channel at each of the two HuckBolt holes. One of these thin plates is shown in Figure 59. The addition of these two plates reduced the maximum local stress by approximately 25%.

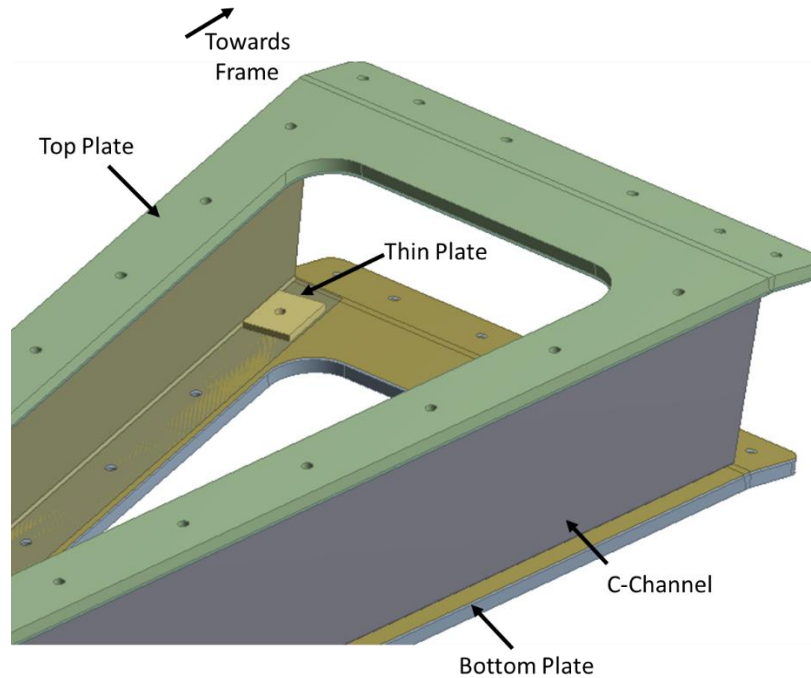


Figure 59. One of two thin plates added to reinforce lower rear bolt-hole.

The length of the plate was chosen arbitrarily but test simulations showed that additional length provided minimal benefit. The thickness of the plate (3/16 inches) was found through trial and error. The plate was made just thick enough such that the local stresses were below yield; additional thickness yielded no extra reduction in stress of the C-channel.

#### 4.6.3. Optimization of Component Dimensions

The procedure to determine the dimensions of the new neck design involved the size adjustment of each component until the material yield point was reached.

There were many possible solutions to how each component could be optimized as there were many combinations of dimensions that yielded the same stress. To simplify the optimization procedure, two of three dimensions of each component was fixed, leaving only the third dimension to be optimized (the thickness). For example, for the tube, the outside cross-section size and the longitudinal length are fixed while the thickness can be adjusted and optimized.

Some dimensions of the structure were fixed because of the design space, such as the length between hitch and frame. However, the other dimensions were fixed according to the default sizes that the client has provided. The default sizes of the new neck design was itself based on decisions by the client and the dimensions of the current neck design.

The thickness of each component was optimized independently until each component reached the material yield point. The determination of the final thicknesses involved a trial and error approach of using possible standard sheet-metal sizes. The final thickness results for each component was less than 1/20th of the other dimensions, therefore the use of shell elements described in section 4.4.1 was justified.

#### 4.7. Summary of Numerical Analysis

The objective of the numerical analysis performed was to determine the stress distribution in the structure in order to ensure the new neck design was as strong as the current neck.

Therefore, the new neck design was optimized for weight while ensuring that the structure can withstand the same maximum allowable load as the current design without material yield.

Two main numerical models were created for analysis. The first model involved the current neck design and was used to determine the maximum allowable load. The second model incorporated the previously obtained maximum allowable load. The new neck design model was used to analyze the strength of the structure, resulting in the addition of new components to reinforce stress concentration regions.

The numerical models had to be simplified due to project constraints. One of the major simplifications for both of the main models was the connection between different components or bodies. Welded components were modelled as bodies connected by shared nodes and did not include weld geometry or properties. Furthermore, bolted connections were modelled through multi-point constraints, which did not include bolt clamp force or friction interaction. These simplifications were necessary to complete the models within project constraints.

The various simplifications made in numerical modelling imposed a limit to the accuracy of the results. However, the numerical models still provide a reasonable approximation of the real structure. Additionally, the new neck design has a high safety factor with respect to expected

load conditions (described in more detail in section 6). Therefore, the new neck design optimized through the numerical process meets the needs and requirements of the project.

## 5. Fatigue Analysis

A fatigue analysis was performed to ensure that the optimized design lasts ten years under high usage under the rated load. The methods used are the stress-life approach taken from Shigley's Mechanical Engineering Design and the Canadian Standard CAN/CSA-S16.1, which utilize Stress vs. Number of cycle curves (S-N curves) to determine the life span of a structure [11] [12]. The Canadian standard for steel structures was used only for the welded locations. The purpose of this analysis is to give an approximation of the design life span.

### 5.1. Critical Locations

The fatigue analysis is a localized approach. The critical locations are analyzed and then optimized for the required life span. These locations are shown in Figure 60.

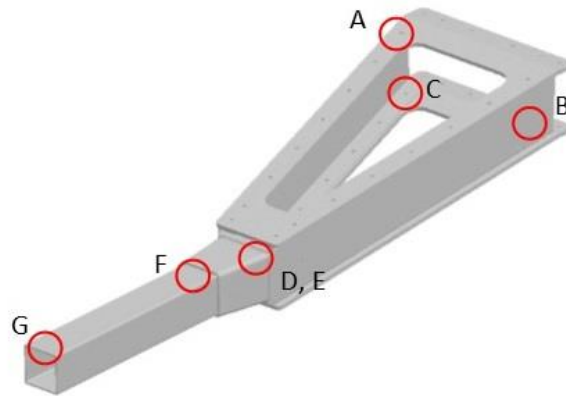


Figure 60. Critical locations for fatigue loading.

A finite element analysis of the new design under the rated load of 1500 lbs was performed. The results from this FEA are shown in Table V.

TABLE V. STRESS VALUES AT CRITICAL LOCATIONS.

Component	Location	Stress [Ksi]
C-channel	A - Top Flange	4.9
	B - Column	2
	C - Bottom Flange	6.2
HSS	G - Hitch	7
	F - Before Joint with Sleeve	3.8
	D - Joint with Box	2.2
Sleeve	E - Joint with Box	6.6

## 5.2. Simplifications and Assumptions

Stress concentrations due to the different structural geometries reduces the fatigue life of a structure. The approach taken to analyze fatigue is an analytical one, thus relying on experimental results to quantify the stress concentration factor of a geometry. The following simplifications and assumptions were made to approximate the new design geometry to the experimental data available.

- For the C-channel:
  - The thicknesses of the top and bottom plates were considered as part of the C-channel flange.
  - The C-channel was analyzed as three different parts. A top and bottom plate with holes and the flange. This is approximation was made due to the difficulty to find stress concentration factors for a C-channel with holes under bending.
- For the sleeves:
  - The sleeves were assumed to be a hollow structure with a uniform cross section. This helps approximate the fatigue strength by utilizing fatigue curves and stress concentration factors for hollow structural sections.

### 5.3. Number of Cycles

The life span of 10 years consists of 175200 cycles of loading and unloading. The duration of a cycle is of 30 minutes where the neck is subjected to the rated load of 1500 lbs for 15 minutes and unloaded for 15 minutes.

The duration of a loading and unloading cycle was determined as the most reasonable worst-case scenario for the usage of a pup trailer. A 15 minute period for loading and travel is a reasonable minimum time between trailer source and destination.

### 5.4. Material Properties

The material used for the C-channels, tube and the sleeve support is steel 44W. Based on the Canadian Standard CSA G40.20/G40.21 for structural quality steel, steel 44W has a tensile strength of 58 ksi and a yield strength of 36 ksi [13].

The fatigue strength of steel 44W was approximated using a general stress life curve for mild steels [14]. The fatigue strength of steel 44W for 175000 cycles is approximately 46 ksi. The S-N curve for mild steels is shown in Figure 61.

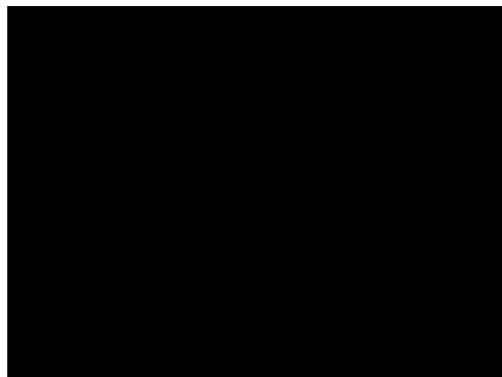


Figure 61. Stress vs. number of cycles curve for mild steels [14].

However, the introduced S-N curve for mild steels is not accurate to approximate the fatigue strength of the material when welded. Welding causes changes in the material properties and creates stress concentrations. Therefore, an S-N curve for a HSS welded to a plate is used to approximate the fatigue strength of the welded critical locations. The S-N curve shown in Figure

62 was obtained from the Canadian standard for steel structures CAN/CSA S16.1 and it applies to materials that meet the CAN/CSA G40.20/G40.21 standard [12].



Figure 62. S-N curve for a welded hollow structural section to a plate [12].

Category “E1” shown in Figure 63 is the category that refers to a tube welded to a plate, the values obtained from this curve are conservative values and already account for the stress concentrations originated from this type of joint. The fatigue strength for a tub welded to a plate under 175200 cycles is approximately 90 MPa (13.1 Ksi).

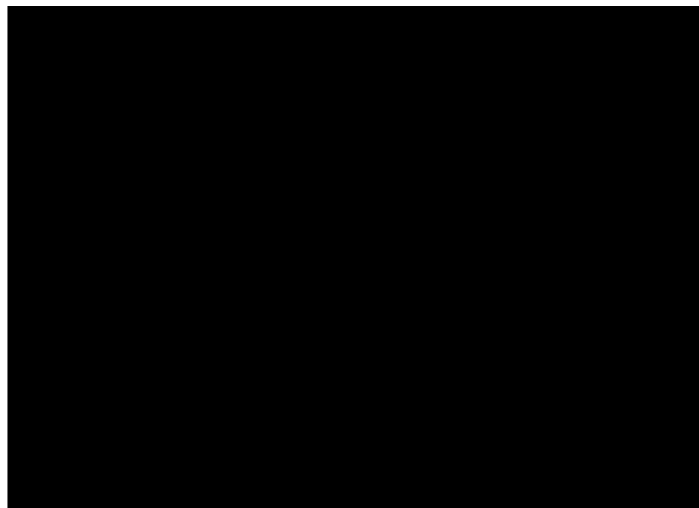


Figure 63. Detail "E1", welded hollow structural section to a plate [12].

Table VI summarizes the material properties of steel 44W and the fatigue strength at the critical locations.

TABLE VI. MATERIAL PROPERTIES OF STEEL 44W AT THE DIFFERENT LOCATIONS.

	C-Channel	HSS			Sleeve
Location	All three	Hitch	Hitch + Sleeve	Hitch + Box	Sleeve + Box
Material	Steel 44W				
Yield strength, ksi	44				
Ultimate tensile strength, ksi	60				
Fatigue strength	46	46	46	13.1	13.1

### 5.5. Modifying Factors

The values obtained from the S-N curves of materials are based on specimens tested under controlled environments [11]. Modifying factors are applied to account for differences in size, surface finish, load condition, temperature of operation and reliability. These factors affect the fatigue strength of the material and were taken from Shigley’s Mechanical Engineering Design [11]. Equation 5 shows the Marin equation.

$$S_f = k_a k_b k_c k_d k_e S'_f \tag{Equation 5}$$

Where:

- Ka = surface factor
- Kb = size factor
- Kc = load factor
- Kd = temperature factor
- Ke = reliability factor
- Sf = fatigue strength
- S'f = pristine fatigue strength

- **Surface Factor**

This factor depends on the finish quality of the structure. The following empirical relationship can be used to quantify the effect of the surface finish.

$$k_a = a S_{ult}^b \quad \text{Equation 6}$$

Where a and b are constants that depend on the characteristic surface finish of certain manufacturing methods. The values for a and b was determined from the Figure 64.

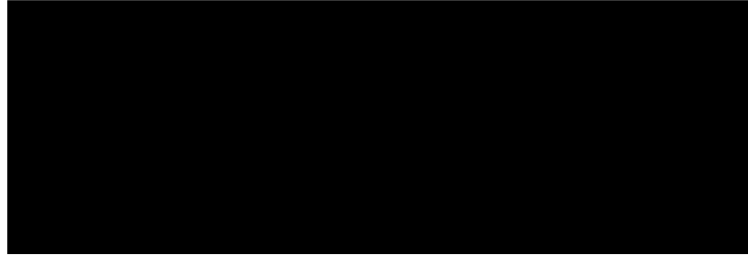


Figure 64. Surface factor constants [11].

- **Size Factor**

Equation 7 was used to find the effect of size on the fatigue strength of the components. Equation 8 and Equation 9 was used to determine the effective diameter for a non-rotating rectangular structural shape.

$$k_b = \begin{cases} 0.879d^{-0.107} & 0.11 \leq d \leq 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \leq 10 \text{ in} \end{cases} \quad \text{Equation 7}$$

$$A_{0.95\sigma} = 0.05hb \quad \text{Equation 8}$$

$$d_e = 0.808\sqrt{hb} \quad \text{Equation 9}$$

- **Load Factor**

The fatigue strength of a structure will vary depending on the type of loading scenario. The load factors used to account for the type of load are shown in Equation 10.

$$k_c = \begin{cases} 1 & \text{Bending} \\ 0.85 & \text{Axial} \\ 0.59 & \text{Torsion} \end{cases} \quad \text{Equation 10}$$

- **Temperature Factor**

This factor relates to the ductile-to-brittle transition that metals exhibits with respect to temperature. At temperatures lower than 25° Celsius (room temperature), metals are more likely to fail due to brittle fracture. And for temperatures higher than room temperatures, metals are more likely to fail due to yielding. Equation 11 is used to determine the temperature factor.

$$k_d = 0.975 + 0.432(10^{-3})T_F - 0.115(10^{-5})T_F^2 + 0.104(10^{-8})T_F^3 - 0.595(10^{-12})T_F^4 \quad \text{Equation 11}$$

For temperatures in the rage of 70° F to 1000°F

- **Reliability Factor**

This factor accounts for the variations in the material properties. Equation 12 was used to factor the desired reliability into the fatigue strength of the material.

$$k_e = 1 - 0.08z_a \quad \text{Equation 12}$$

$z_a$  is the transformation variate, and its values are obtained from Figure 65.

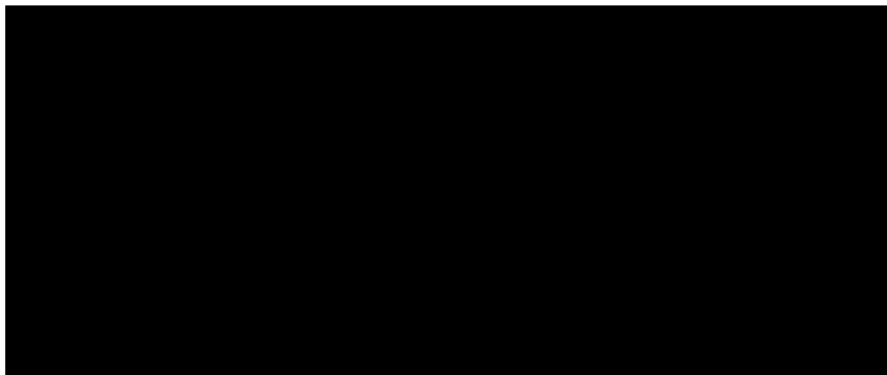


Figure 65. Reliability percentage and its respective reliability factor [11].

Table VII shows the value for each of the Marin’s modifying factors for the non-welded locations of the C-channel and HSS.

TABLE VII. VALUES OF THE MARIN FACTORS FOR THE NON-WELDED CRITICAL LOCATIONS.

Location	C-channel			HSS
	Top Plate w/ hole	Web	Bot Plate w/ Hole	At Hitch and Hitch + Sleeve
<b>Surface</b>	0.780	0.780	0.780	0.906
<b>a</b>	14.4	14.4	14.4	2.7
<b>b</b>	-0.718	-0.718	-0.718	-0.265
<b>Size</b>	0.832	0.833	0.830	0.679
<b>de</b>	1.67	1.65	1.71	6.46
<b>b</b>	4	0.375	4	8
<b>h</b>	1.0625	11.11	1.125	8
<b>Loading</b>	1	1	1	1
<b>Temperature</b>	1	1	1	1
<b>Reliability</b>	0.702	0.702	0.702	0.702

- **Stress Concentration Factors**

Stress concentration factors weakens the fatigue strength of a component. Since the S-N curve for a welded HSS to a plate already accounts for the stress concentrations, the only stress concentration factor required to be determined is for the flanges of the C-channels with holes under bending.

The stress concentration factor to the flanges with holes under bending was determined using Figure 66.

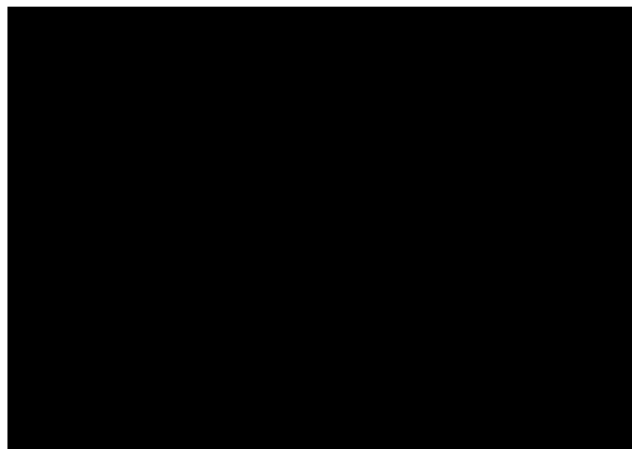


Figure 66. Stress concentration factor to a plate with a hole under bending [11].

The values used from the flange of the C-channel and the stress concentration factor determine are shown in Table VIII.

TABLE VIII. STRESS CONCENTRATION FACTOR PARAMETERS

	<b>Top Flange</b>	<b>Bottom Flange</b>
<b>Width of the plate (W)</b>	4 inches	4 inches
<b>Diameter of the hole (d)</b>	0.69 inches	0.69 inches
<b>Thickness of the plate (h) (Flange + Plate)</b>	1.0625 inches	1.125 inches
<b>d/w</b>	0.1725	0.1725
<b>d/h</b>	0.65	0.61
<b>Kt</b>	2.1	2.1

## 5.6. Fatigue Failure Criterion

The new neck design is subjected to a fluctuating simple loading. This effect from this type of loading is characterized by the mean stress and the amplitude of the stress. The value of these stresses is affected by the stress concentration factors and can be found using Equation 13 and Equation 14 [11].

$$\sigma_m = K_t \frac{\sigma_{MAX} + \sigma_{MIN}}{2} \quad \text{Equation 13}$$

$$\sigma_a = K_t \frac{|\sigma_{MAX} - \sigma_{MIN}|}{2} \quad \text{Equation 14}$$

The fatigue failure criterion used to evaluate the fatigue life of the new design is modified-Goodman. Equation 15 represents the modified Goodman criterion. The factor of safety  $n$  is the variable to be determined. If  $n < 1$ , the new design will fail due to fatigue before reaching the 10 years life span. If  $n > 1$ , the new design will have a life span of 10 years or longer.

$$\frac{\sigma_a}{S_f} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \quad \text{Equation 15}$$

Even though the minimum requirement for the new design to be successful is a factor of safety of 1, the objective is to achieve a factor of safety of 2 to account for all the approximations and assumptions made during the analysis.

For the welded locations, the permissible stress range was obtained using the number of cycles on the S-N curve for welded tube on a plate. The permissible stress range was compared to the actual stress range at the location of interest. If the actual stress range is lower than the permissible stress range, the new design is capable of withstanding such a stress cycle for the given life span [12].

## 5.7. Results

Table IX shows the results obtained for the fatigue strength, mean and amplitude stress and the factor of safeties.

TABLE IX. STRESS VALUES FOR THE CRITICAL LOCATIONS AND FATIGUE FACTOR OF SAFETY.

Stresses							
Component	C-Channel			HSS		Sleeve	
Location	Top Flange	Web	Bottom Flange	Hitch	HSS + Sleeve	HSS + Box	Sleeve + Box
<b>Actual Sf</b>	20.97	20.99	20.9	19.9	19.9	13.1	13.1
<b>Max</b>	4.0	2	6.2	7	3.8	2.2	6.6
<b>Min</b>	0	0	0	0	0	0	0
<b>Mean</b>	4.9	1	6.2	3.5	1.9		
<b>Amplitude</b>	4.9	1	6.2	3.5	1.9		
<b><math>n_f</math></b>	2.72	14	2.15	4.07	7.47	6.53	1.98

The results from the fatigue analysis show that the new design will achieve a life span of 10 years with a minimum factor of safety of 2 under continuous cycles of 30 minutes of loading and unloading.

## 6. Evaluation of Target Specifications

The design team compared the design specifications to the project target specifications. Table X summarizes the evaluation of the different target specifications.

TABLE X. EVALUATION OF TARGET SPECIFICATIONS

ID	Metric	Level of Importance	Marginal Value	Ideal Value	Achieved Value	Pass /Fail	Units
1	Maximum tongue weight	5	7306.3	>7306.3	7306.3	PASS	lbf
2	The factor of safety related to the maximum stress values observed	5	2	3	2	PASS	unitless
3	The amount of time it takes to manufacture the assembly	4	9	6	5	PASS	hour
4	Number of parts	4	14	<14	12	PASS	quantity
5	Equipment types	4	7	<7	5	PASS	quantity
6	The total assembly weight	3	1050	750	1006	PASS	lbs
7	The total cost of the assembly (material and labor)	4	1722	<1722	1344.30	PASS	\$
8	The fatigue life of the assembly	5	8	10	10	PASS	year
9	Prevents debris buildup [5]	1	-	PASS	PASS	PASS	binary
10	The existing peripheral components (jack, hitch adaptor, etc.) are compatible	1	-	PASS	PASS	PASS	binary
11	The ability for the assembly to be manufactured and assembled in a safe manner	3	-	PASS	PASS	PASS	binary

ID	Metric	Level of Importance	Marginal Value	Ideal Value	Achieved Value	Pass /Fail	Units
12	Linear movement required during assembly	4	4	0	0	PASS	ft
13	Rotational movement required during assembly	4	45	0	0	PASS	degree
14	Retains critical geometry	5	-	PASS	PASS	PASS	binary
15	The engineering department approves that the design appears sturdy	2	-	PASS	PASS	PASS	subj

### 1. Maximum tongue weight

The maximum tongue weight that was loaded onto the tongue of the current neck design before material yielding was 7306.3 lbf. This load was then successfully applied to the new design, also without material yielding. This result means that the new neck design is as strong as the current design. This is one of the most important objectives that the new design must fulfill in order to ensure that Cancade's zero warranty claim history can continue.

### 2. Factor of Safety

Cancade's minimum required factor of safety is two. This safety factor was successfully achieved by the design team. There was four loading scenarios that were used to analyze the factor of safety. These were: normal tongue weight in the vertical direction, tongue load in the horizontal direction, jackknife loading, and fatigue loading. The fatigue loading produced the lowest safety factor of two and this was the value that was used. The other loading scenarios produced much higher factors of safety.

### **3. Manufacturing time**

The time for manufacturing the current design is nine hours. This is due to unnecessary time needed for flipping the entire assembly as well as welding time. It takes 15 minutes and two workers to flip the assembly and needs to be done four times for each assembly. Out of the nine hours, five hours is needed for set-up and tacking, and the other four hours is strictly for welding.

Through evaluation of the new design and consultation with Cancade, it was determined that the new design would take five hours to set-up and fasten together. This result is due to the fact that new design consists of a similar number, size, and complexity of parts. The tacking process also is very similar in difficulty, process, and time to install the HuckBolts [3]. The four hours due to welding was eliminated. The total time for manufacturing is five hours.

### **4. Number of parts**

One of the goals of the project was make the neck easier to manufacture. This goal was achieved by decreasing the number of parts in the assembly from 14 to 12. The number of parts of the current design is a total of 14 parts consisting of sheet metal parts and a square tube. The new design consists of 12 parts, again consisting of sheet metal parts and a square tube. The decrease in the number of parts is due to the new design utilizing a C-channel which can be bent from a single piece of sheet metal. In comparison, the I-beams of the current design are welded together from three pieces of sheet metal.

### **5. Equipment types**

The design meets the ideal target specification for number of equipment types.

The number of types of equipment used in the assembly process is related to the complexity of the process as well as the time needed. More equipment types involved in the process potentially means more workers, higher skilled workers, and more time taken.

The current design requires various hand tools, two MIG welder, two forklifts, and a crane. Since the new design no longer needs to be flipped, there is no need for the two forklifts. Since there is only minimal welding needed, one MIG welder is sufficient.

Various hand tools are still needed and now with the use of Huck Bolts, the installation gun will also be needed. However, the installation gun costs less than a welder and does not use consumables nor does it need a certified worker. The single crane will still be needed as some of the parts are heavy and will need to be maneuvered into position as was done before. The number of equipment types was reduced by eliminating the use of forklifts.

## **6. Assembly weight**

The total weight of the current design is 800 lbs and the total weight of the new design is 1000 lbs. The extra weight is due to the fact that C-channels are not as strong in bending as I-beams and the top and bottom plates connecting the two are quite thick and heavy. The weight is within our marginal target, although it does not meet the ideal target.

Lower assembly weight allows for easier assembly and typically results in lower cost due to most of the weight made up by the weight of sheet metal. Cancade has specified that the new design can be as much as 300 lbs over the current weight.

## **7. Total assembly cost**

The total cost of the new design assembly including material, labour, and HuckBolts is \$1344.30, a decrease of \$377.70 from the current design. The marginal value is the same cost of the current design, \$1722.

The new design has a total material cost of \$820.02. This cost comes from \$0.80 per lb of sheet metal (914.44 lbs) and \$1.00 per lb of square tube (88.47 lbs). The total cost of labour is \$375 which comes from a labour time of five hours at a cost of \$75 per hour. The total cost of the HuckBolts is \$149.28 which comes from using 48 HuckBolts which carry a cost of \$2.00 per HuckBolt and \$1.11 per collar.

## **8. Fatigue life**

The new design achieves a life span of 10 years under loading and unloading cycles at the rated load every 30 minutes, with a factor of safety of 2.

## **9. Debris buildup prevention**

The new neck design protects from debris buildup. The current design easily allows for utilizing angled pieces of sheet metal welded onto corners and flanges where buildup is probable. The new design uses C-channels that have one less side that would have buildup when compared to I-beams. The rest of the problem areas will also be remedied by adding sheet metal.

## **10. Peripheral component compatibility**

Cascade uses certain sub-assemblies and hardware such as the hitch components and the jack assembly that must be used and installed onto the new design. These components are still able to be installed on the new design in a convenient manner and meets target specification.

## **11. Safety**

The new neck design has accomplished the elimination of any rotational movement of the whole frame during the attachment of the neck, which was the biggest safety risk. The safety risk was due to having to lift up the entire assembly with two people, one crane, and two forklifts. The new processes in the manufacturing of the new neck design are HuckBolt fastening and bending of sheet metal, both of which are already practiced at Cascade. The elimination of the biggest safety risk in the current chassis manufacturing process while not creating other safety risks results in the manufacturing process of new neck design being safer than the current design. Therefore, the new neck design achieves the target specification in terms of safety.

## **12. Linear movement**

The assembly area for the neck chassis allows for up to four feet of linear movement. This movement is allowable since the crane is already there and can be used. The new design does not need any linear movement for the whole assembly and achieves the ideal target specification.

## **13. Rotational movement**

The current design requires that it be rotated by 90 degrees four times to allow for the proper welding practices to take place and eliminate vertical welding requirements. This process is time consuming, dangerous, and involves multiple workers and forklifts. Since the new design does not need any vertical welding on the whole assembly and the HuckBolts can be installed easily in any orientation, this rotational movement is eliminated. This meets the ideal specifications.

## **14. Critical geometry**

The geometry of the new design allows for the proper function of the trailer without impeding the vehicle towing it, as discussed in Section 3.1. The hitch does not exceed the width of the frame, has adequate ground clearance, and does not exceed the height of the frame. The interface between the frame and the hitch is integrated such that both structures can be assembled without interference.

## **15. Sturdiness**

Because of the industry that the pup trailer is used in and with the type of client who purchases these trailers, the appearance of sturdiness is important. The current design exuded confidence in the perceived strength of the assembly and the desire was that the new design would as well. Both Dylan Mullin and Tanner Pidborchynski have given their approval that the design looks sturdy.

## 7. Conclusions

Cancade is a truck trailer manufacturer that is currently experiencing issues in their manufacturing process for pup trailers. The current procedure for manufacturing the pup trailer chassis involves a process that requires multiple people and cranes to lift and flip the 30 foot long assembly in order to weld all the appropriate areas. This process is both dangerous and time consuming. Cancade has requested that a new neck portion of the trailer be designed so that the manufacturing and assembly process of the pup trailer becomes quicker and safer.

The final design is a Fork C-channel design constructed with HuckBolts that weighs 1007 lbs with a total cost of \$1344.30. The design is considered a success since the new design can be manufactured using a safer, cheaper, and faster method while still maintaining the same degree of strength. Cancade required a full report detailing the new design, a finite element analysis of both the new and current design, and an analytical comparison between HuckBolt's and welding connections. All of these deliverables have been achieved.

An analytical comparison between the strength of welded connections and HuckBolt connections was performed. The team performed research to examine welding standards and determine the proper way to evaluate welds. The team calculated the strength needed for a simple welded I-beam and compared it to the specifications of the HuckBolt. The analysis resulted in a spacing requirement for HuckBolts to equal the weld.

Finite Element Analysis was used to guide the design process. This was done to determine the maximum load required to cause material yield in the current design. This maximum load was the benchmark used to test the new design. The new design was ensured to be as strong as the current design. The new design will sustain Cancade's history of zero warranty claims on the pup trail neck chassis in the future.

A complete initial assembly was created on Inventor CAD and the correct HuckBolt spacing was implemented. Joints for the tube to C-channel connection and neck to frame connection were designed with CAD using the dimensions of the C-channel and tube. The dimensions were approximated based on research and current Cancade designs. The joints were made in such a

way that Cancade would have the ability to manufacture them. The joint design involved flanges extending from the box to the tube at an angle, which reinforced the structure's weak points.

The specifications of final design meets most of the ideal target specifications. All other target specifications meet the marginal value at the very least. A thorough evaluation of all the target specifications was performed to reliably determine the success of the design. This evaluation ensured that the design was complete and achieved the goals and needs that were put forth in the project description.

The design team recommends that the next step is to build a test prototype for the purposes of destructive testing to confirm the strength. The next step would then be to build another prototype for endurance testing. This would mean cycling the neck chassis through various loads to simulate different road environments and loading and unloading conditions. With these tests performed, the product would then be reviewed to determine if production is the next step to be taken.

There are many benefits that come with the new neck design. The use of HuckBolts allows for Cancade to assemble more neck chasses and get them out quicker to their customers and distributors. Cancade can now build the neck assembly on more of a make-to-order system since they can make them quicker. This allows for finished goods to sit at the warehouse. The new design also is much safer which allows for less incidents and time loss due to injury. HuckBolts also are simpler to install than welding, allowing Cancade to use less expensive labour than what welding requires. Finally, the new neck assembly is \$378 lower than the current design, which will increase the profit for Cancade and promote growth in the company.

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## Appendix A

This appendix provides additional information about the project background. The metrics for the project are described in a greater detail in Table A1, including reasons for the metrics and any relevant background information for the chosen marginal and ideal values.

TABLE A1. ADDITIONAL INFORMATION FOR EACH METRIC

Metric ID	Additional Information
1	<p><b>Supported load</b></p> <p>The load that the chassis needs to support is based on the rules and regulations of highway vehicles. The marginal value of 37,000 lbs is the maximum legal weight that a pup trailer with Cancade’s specifications can haul. The ideal value of 40,000 lbs is the weight that they would prefer for the chassis to account for excessive loading [3].</p>
2	<p><b>The factor of safety related to the max stress values observed</b></p> <p>Cancade uses a minimum safety factor of 2 in their designs but would like to use a safety factor of 3 if possible.</p>
3	<p><b>The amount of time it takes to manufacture the assembly</b></p> <p>One of the main objectives in this design is to reduce manufacturing time. The current manufacturing time of their entire chassis assembly is 70 hours. Cancade would like to decrease the manufacturing time to 60 hours.</p>
4	<p><b>Number of parts</b></p> <p>The current production neck chassis uses 16 different components during assembly. Reducing the number of components will allow for more efficient manufacturing.</p>
5	<p><b>Equipment types</b></p> <p>The current neck chassis requires 7 different types of equipment used for assembly such as hand tools, lifting tool, forklifts, or welding equipment. By reducing the number of types of equipment used, the neck chassis will be more efficient to build.</p>
6	<p><b>The total assembly weight</b></p> <p>Cancade has stated that the total current chassis weight of 2,385 lbs plus an additional 300 lbs is an acceptable weight. However, it is desirable that the new chassis weight be below the current production weight of 2,385 lbs.</p>
7	<p><b>The total cost of the assembly</b></p> <p>Cancade would like to lower the current cost of manufacturing and assembling the chassis, which is currently \$5250.</p>

8	<p><b>The fatigue life of the assembly</b></p> <p>The pup trailer will experience deteriorating road conditions that will cause the chassis to twist and bend through multiple cycles. After a sufficiently high number of loading cycles the chassis could fail. Cancade has a 5-year warranty and would like to see their trailers last without repairs for at least 8 years, but preferably 10 years.</p>
9	<p><b>The ability of the assembly to resist debris buildup</b></p> <p>The rules and regulations for highway trailers requires that there not be any debris buildup on ledges of the trailer that could potentially fall off and harm other drivers on the road [5].</p>
10	<p><b>The existing peripheral components (jack, hitch adaptor, etc.) are compatible</b></p> <p>The jack and hitch adaptor are components that are installed onto the current production chassis. These components are not to be re-designed and must be installed in the same manner.</p>
11	<p><b>The ability for the assembly to be manufactured and assembled in a safe manner</b></p> <p>Safety is a high priority for Cancade, therefore the new design must be constructed in the safest manner following CANCADE's safety protocols.</p>
12	<p><b>Linear movement required during assembly</b></p> <p>The current neck chassis must be moved in a linear manner, particularly in the vertical direction. Ideally, the assembly process for the new design will not require linear movement due to the fact that they work as a project layout where the assembly is built in one area.</p>
13	<p><b>Rotational movement required during assembly</b></p> <p>The current neck chassis must be moved in a rotational manner. Ideally, the assembly process for the new design will not require rotational movement.</p>
14	<p><b>Retains critical geometry</b></p> <p>The pup trailer has a certain range of motions that affect how it reacts to the truck pulling it, such as in situations like tight corners and backing up in reverse. The trailer also frequently experiences jackknifing. It is imperative that the nature of these motions not be altered by changing critical dimensions.</p>
15	<p><b>The engineering department approves that the design appears sturdy</b></p> <p>The type of customers who buy these pup trailers pay attention to how strong a chassis may look. Cancade would like a design that aesthetically looks strong and durable. Dylan Mullin and Tanner Pidborchynski of the engineering department will approve the appearance of the design.</p>

## Appendix B

Table B2 lists the material properties of the two types of steel used for the current and new neck designs. Almost all components for both design use 44w hot rolled steel (also referred to as A36). The only component made of a different material is the tube of the current design, which consists of A500C steel.

TABLE B2. MATERIAL PROPERTIES OF STEELS USED [15] [16].

	44W	A500C
Density, $lb/in^3$	0.284	0.284
Yield Strength, psi	36000	50000
Ultimate Strength, psi	58000	616000
Young's modulus, ksi	29000	29000
Poisson's ratio	0.260	0.3
Shear modulus, ksi	11500	11600

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Any stress results obtained from any numerical model used for this project was ensured to have converged for all selected points (P1-P5) in the structure. This section provides the mesh convergence information for all points, all load cases, and for each of the two neck models. Convergence plots are shown as percent difference between the stress results of two consecutive mesh iterations. The stress near a selected point (P1-P5) was considered to have converged once the percent difference was consistently below 5%.

## Current Neck Models

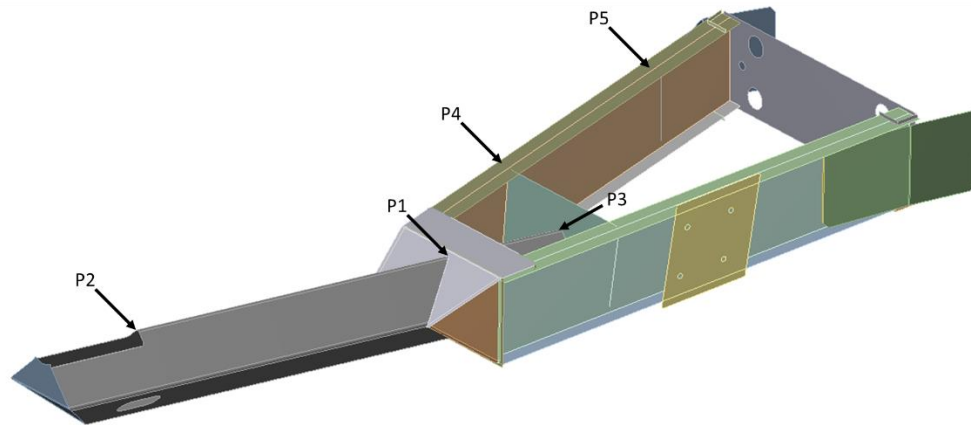


Figure C1. Location of stress points (P1-P5) used for measuring stress convergence in current neck model.

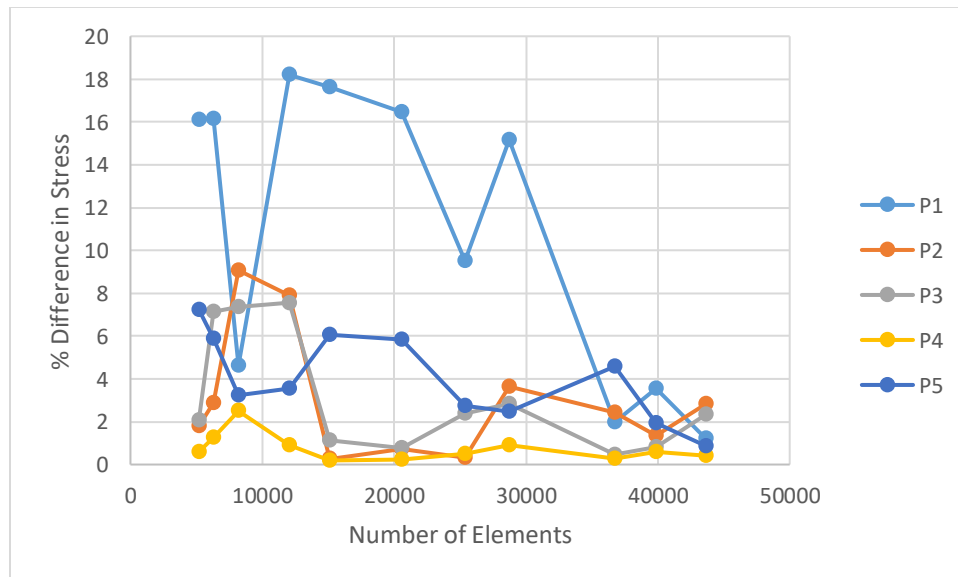


Figure C2. Stress convergence of points P1-P5 as mesh density is increased (Load Case 1, current neck).

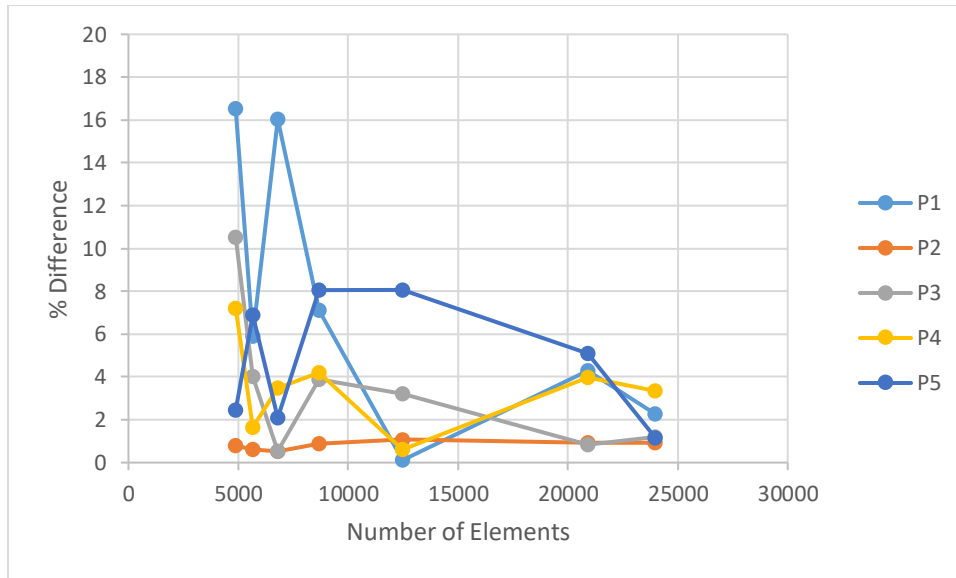


Figure C3. Stress convergence of points P1-P5 as mesh density is increased (Load Case 2, current neck).

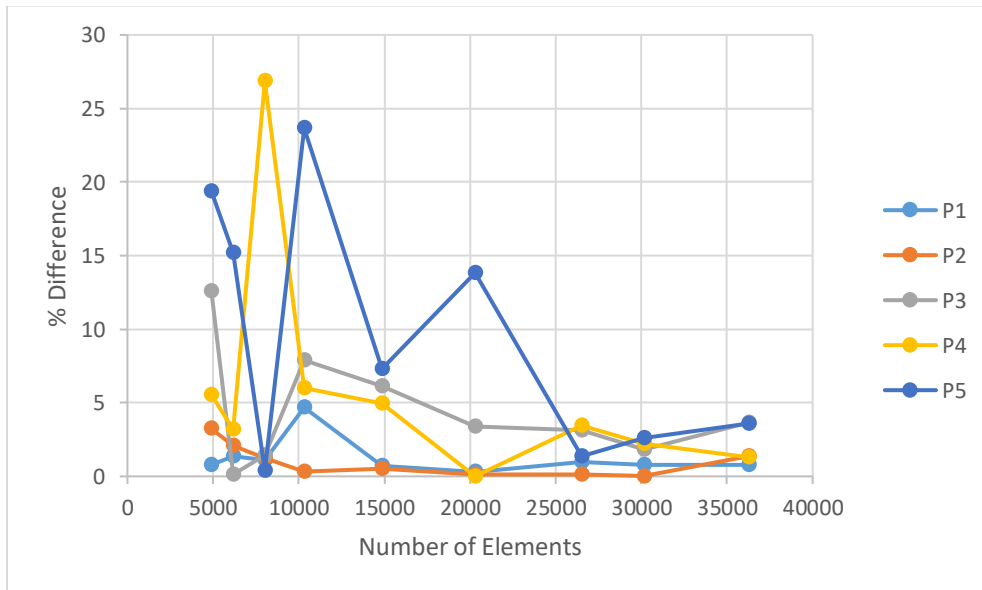


Figure C4. Stress convergence of points P1-P5 as mesh density is increased (Load Case 3, current neck).

## New Neck Models

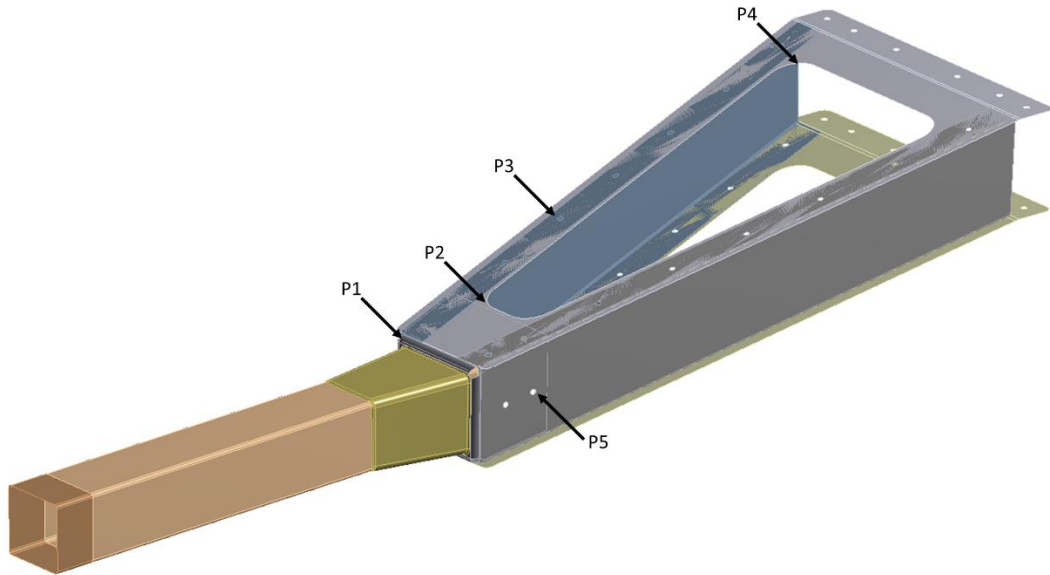


Figure C5. Location of stress points (P1-P5) used for measuring stress convergence in new neck model.

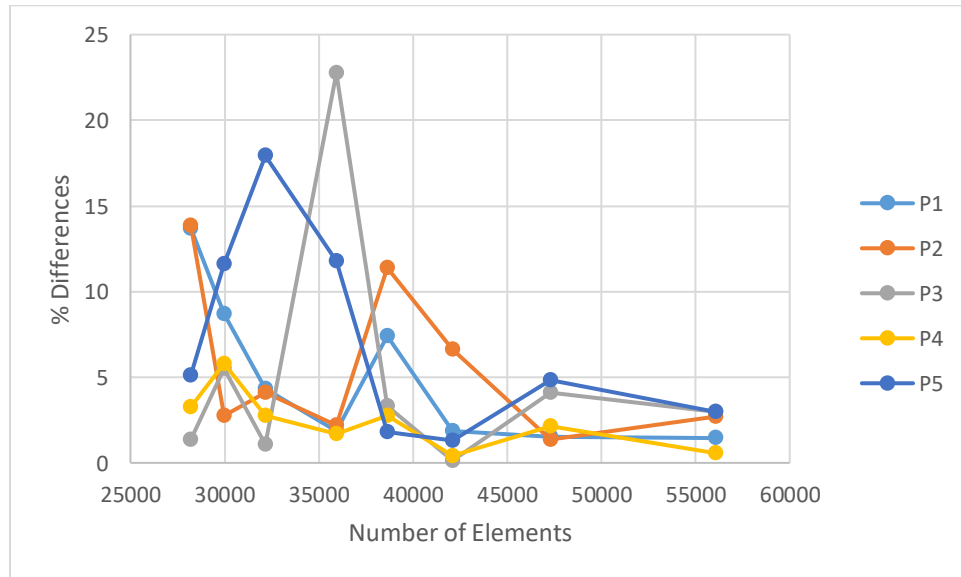


Figure C6. Stress convergence of points P1-P5 as mesh density is increased (Load Case 1, new neck).

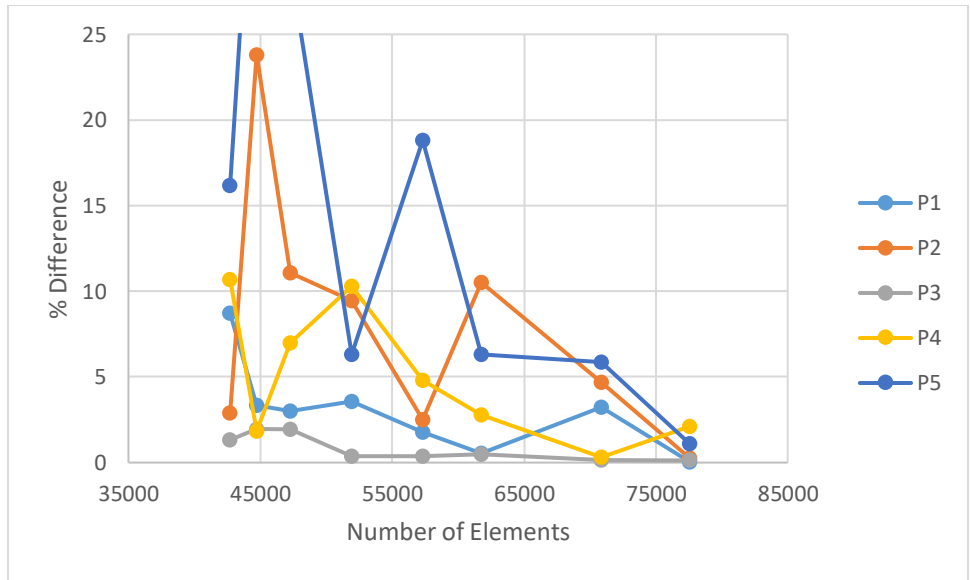


Figure C7. Stress convergence of points P1-P5 as mesh density is increased (Load Case 2, new neck).

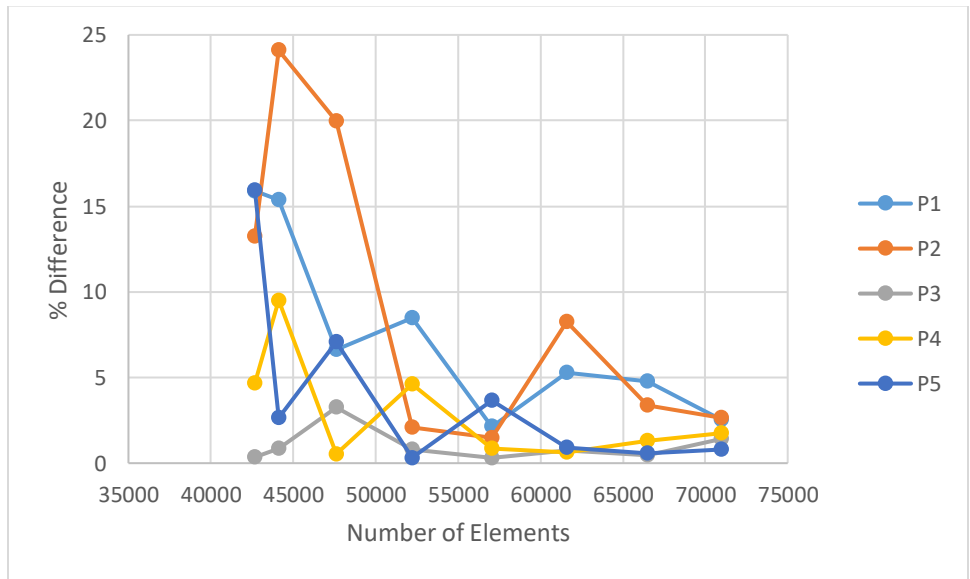


Figure C8. Stress convergence of points P1-P5 as mesh density is increased (Load Case 3, new neck).

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This section provides an overview of equivalent stress results in the current neck model for Load Case 2 and 3. These two load cases represent a horizontal load at the hitch and at the jackknife impact location. Results for Load Case 1 are shown in section 4.3.

The contour plots display the safety factor (SF), which is a ratio of the yield point of the material to the equivalent stress result. SF is defined as the ratio of the yield limit of the material to the equivalent stress result (von Mises). Therefore, SF is equal to one indicates a material that has just begun to yield. The stress results are averaged between shared nodes of different elements. Furthermore, stress results are also averaged between shared nodes of different bodies.

## Load Case 2

**A: Static Structural**  
Total Deformation  
Type: Total Deformation  
Unit: in  
Time: 1  
2018-12-04 8:51 PM

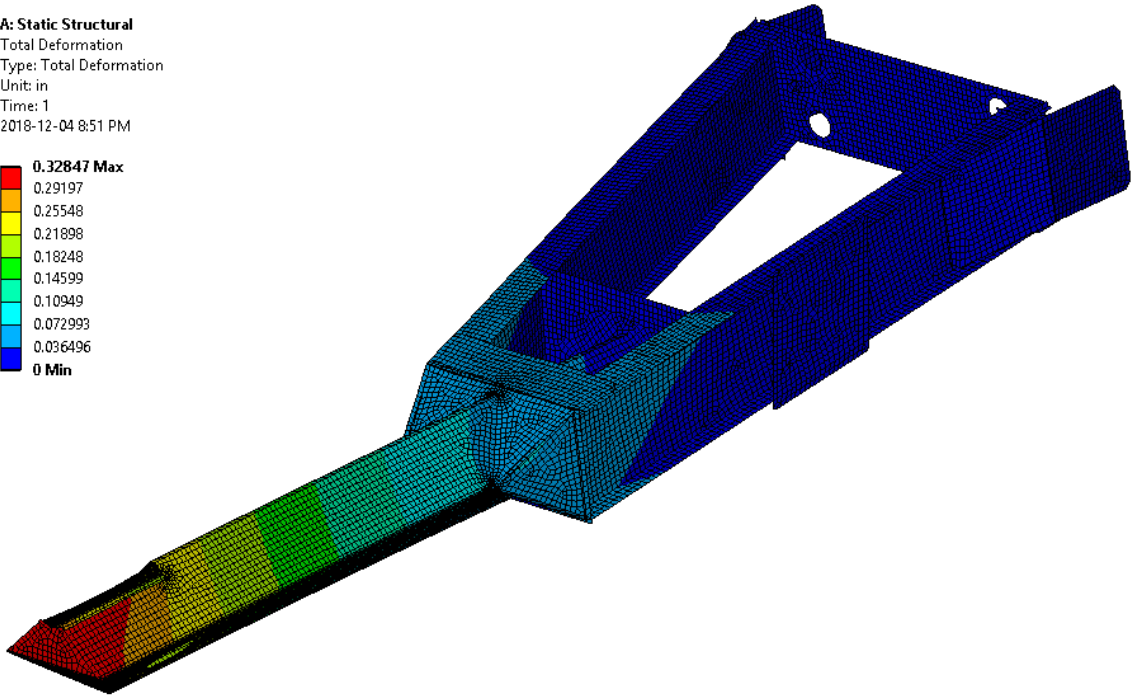
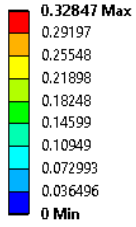


Figure D1. Total deformation for current neck model (displacement scale 12; Load Case 2).

**A: Static Structural**  
Equivalent Stress-fullavg  
Type: Equivalent (von-Mises) Stress (Average Across Bodies) - Top/Bottom  
Unit: psi  
Time: 1  
Custom Obsolete  
2018-12-04 8:53 PM

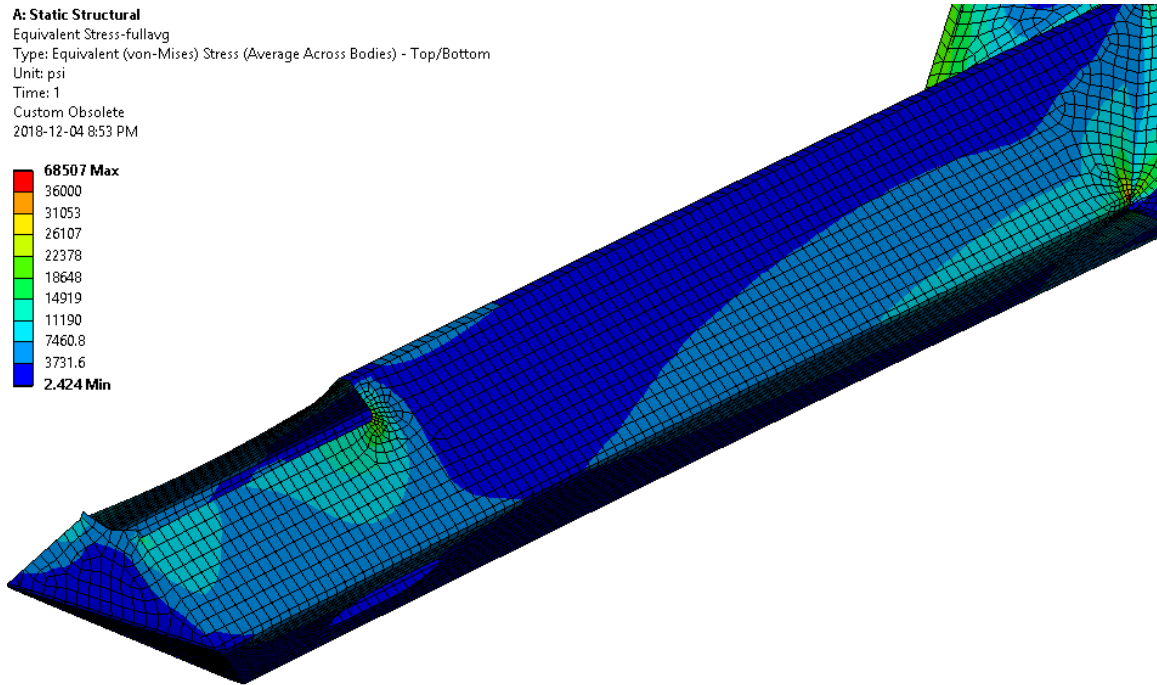
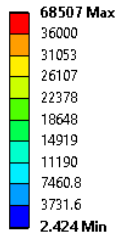


Figure D2. Equivalent stress results for tube (Displacement scale 12; Load Case 2).

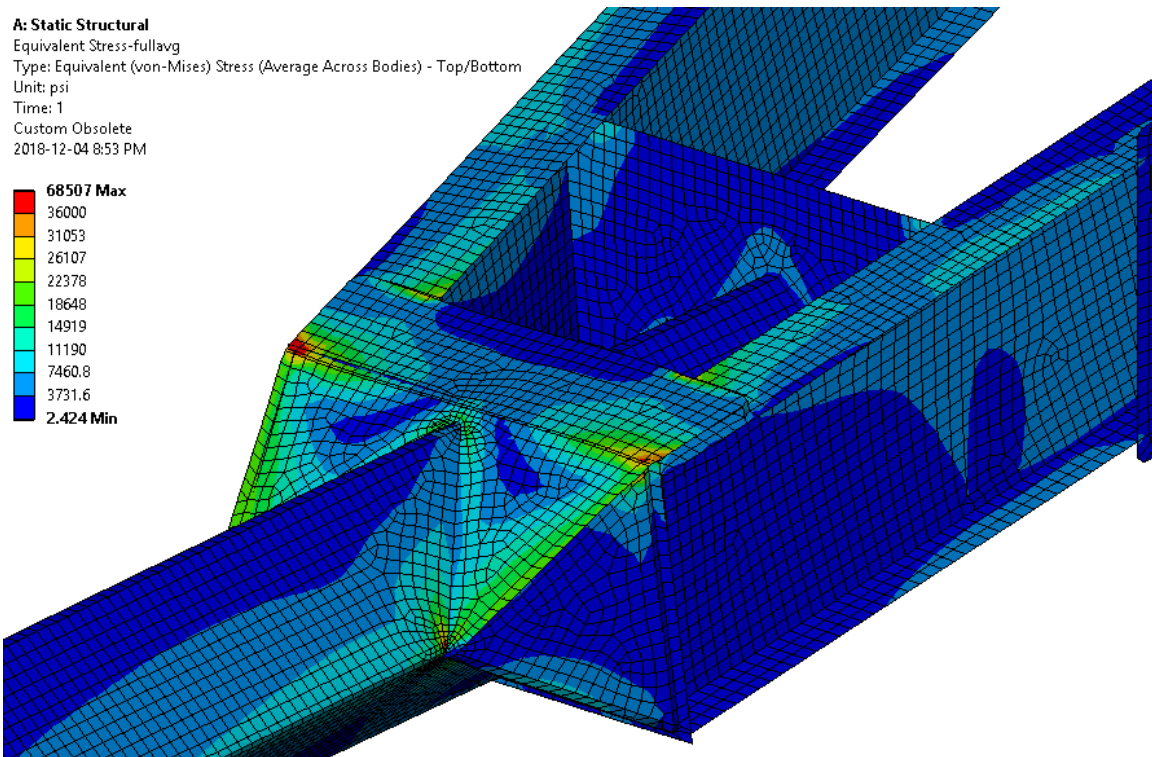


Figure D3. Equivalent stress results for middle region of current neck (Displacement scale 12; Load Case 2).

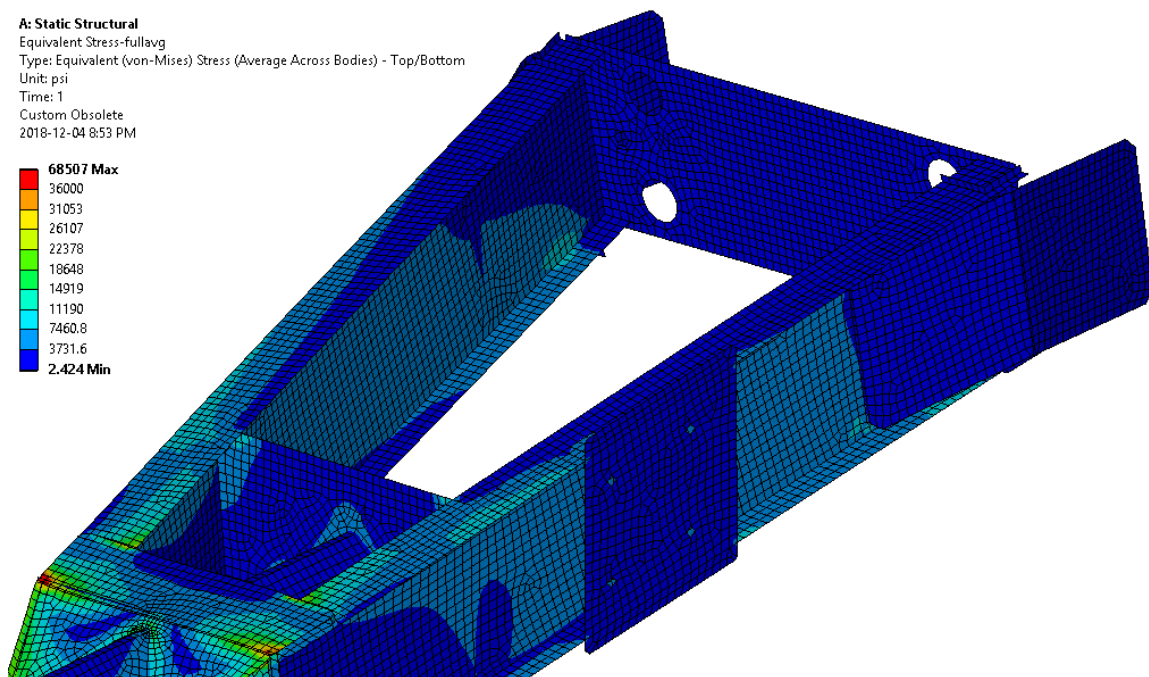


Figure D4. Equivalent stress results for rear of current neck (Displacement scale 12; Load Case 2).

### Load Case 3

**A: Static Structural**  
Total Deformation  
Type: Total Deformation  
Unit: in  
Time: 1  
2018-12-04 8:59 PM

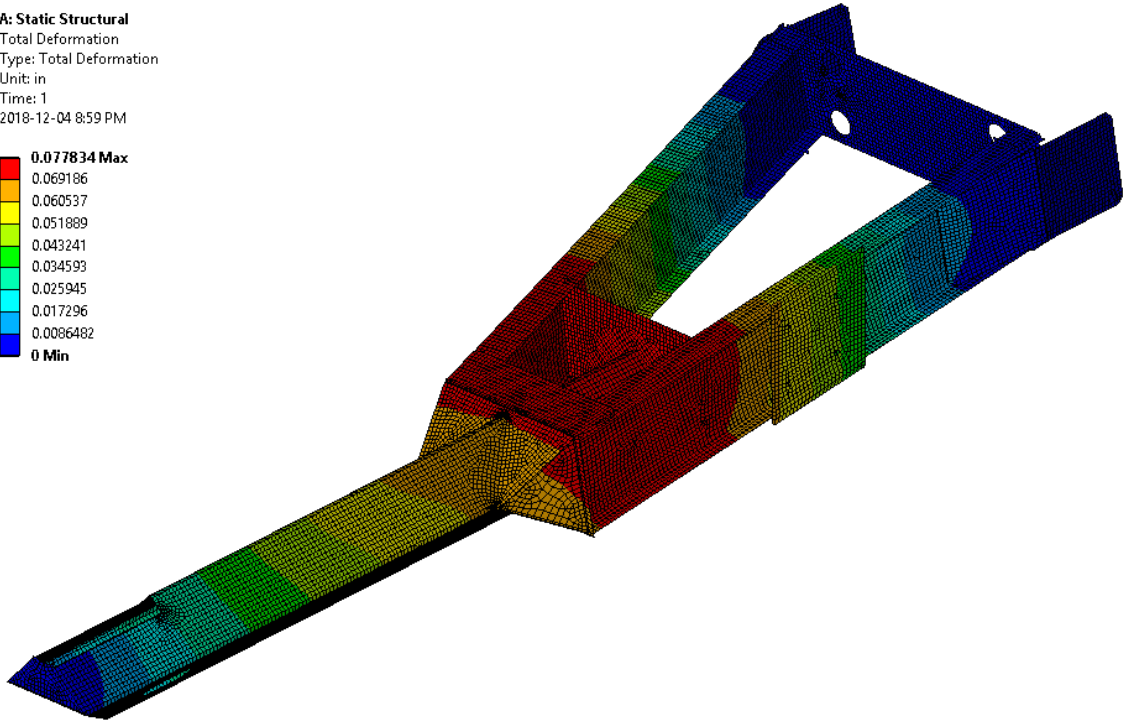
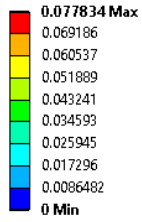


Figure D5. Total deformation for current neck model (displacement scale 12; Load Case 2).

**A: Static Structural**  
Safety Factor-fullavg  
Type: Safety Factor (Average Across Bodies)  
Time: 1  
2018-12-04 9:00 PM

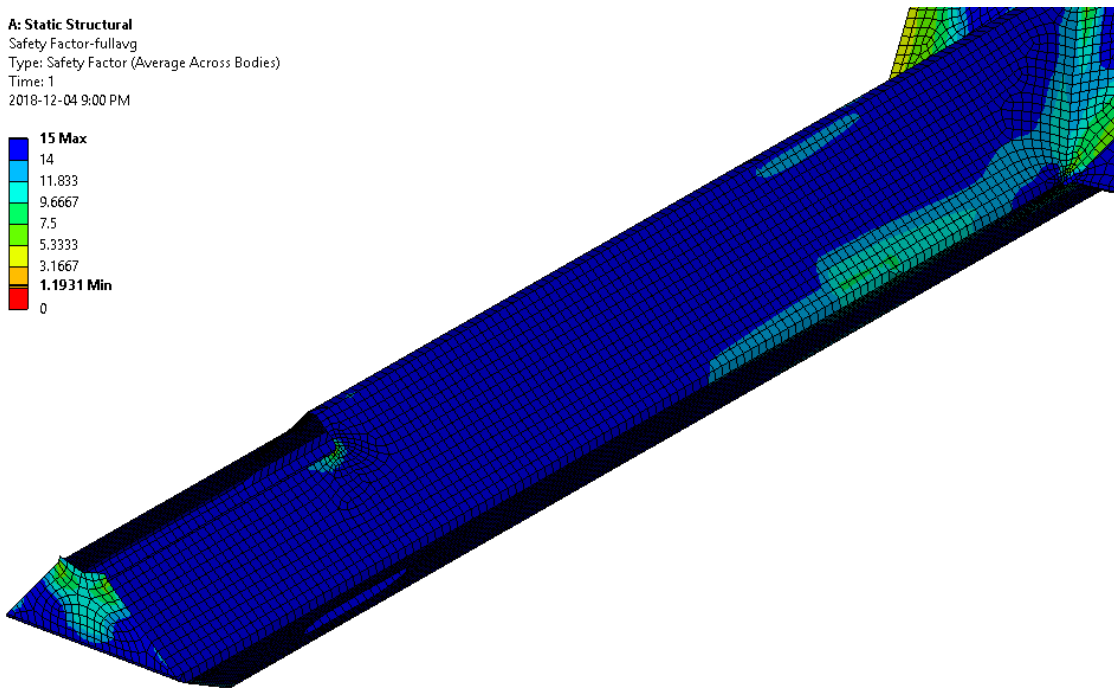
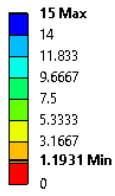


Figure D6. Equivalent stress results for tube (Displacement scale 12; Load Case 3).

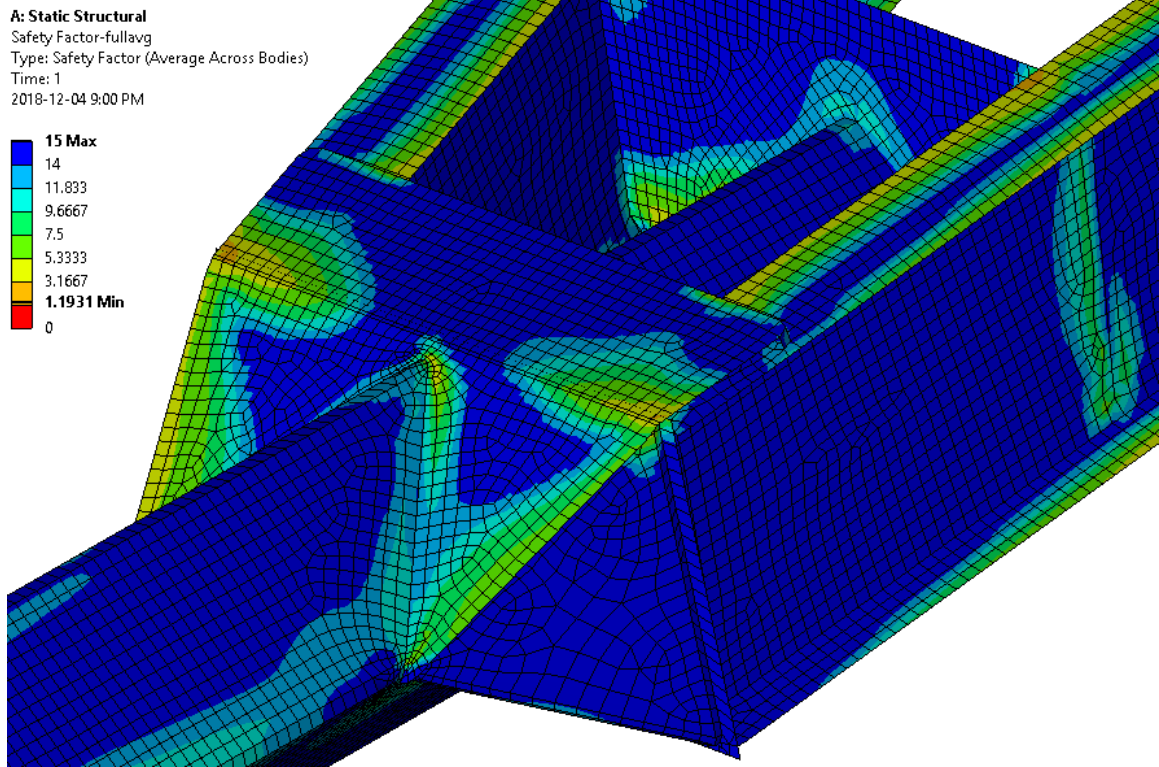


Figure D7. Equivalent stress results for middle region of current neck (Displacement scale 12; Load Case 3).

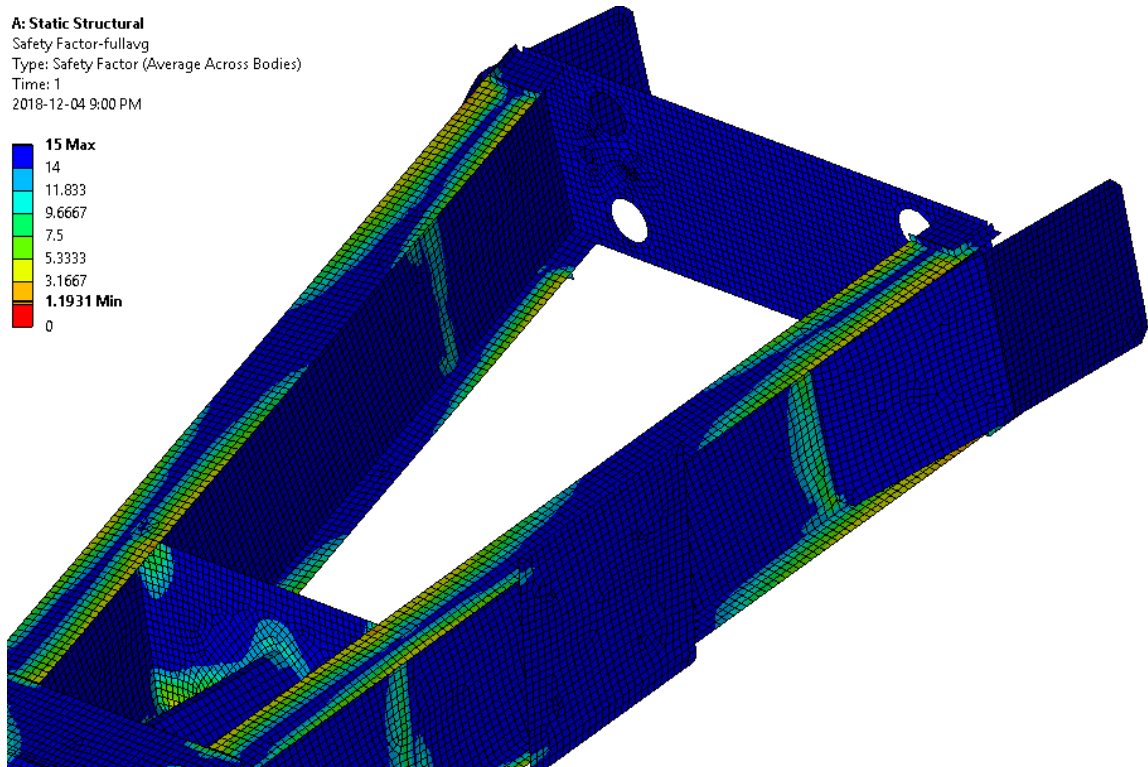


Figure D8. Equivalent stress results for rear of current neck (Displacement scale 12; Load Case 3).

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This section provides an overview of equivalent stress results in the new neck model for Load Case 2 and 3. These two load cases represent a horizontal load at the hitch and at the jackknife impact location. Results for Load Case 1 are shown in section 4.5.

The stress results are averaged between shared nodes of different elements. All materials are made of 44W hot rolled steel, which has a yield point of 36000 psi.

## Load Case 2

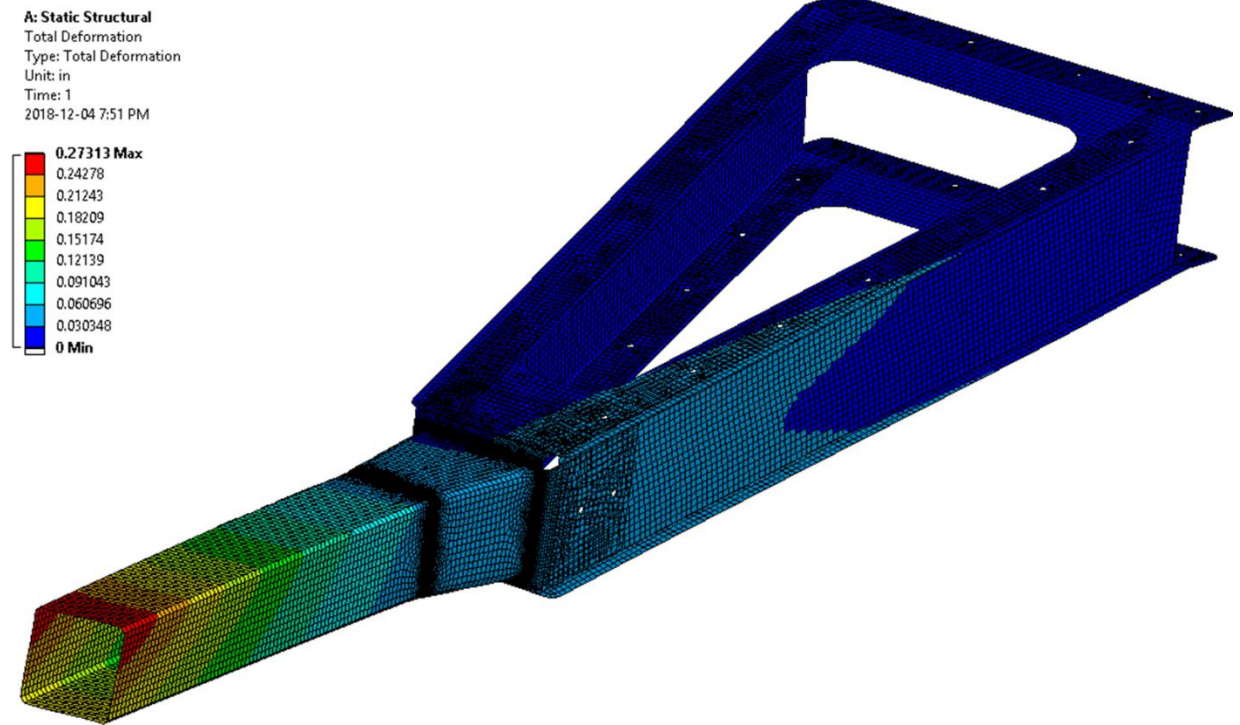


Figure E1. Total deformation for new neck model (displacement scale 12; Load Case 2).

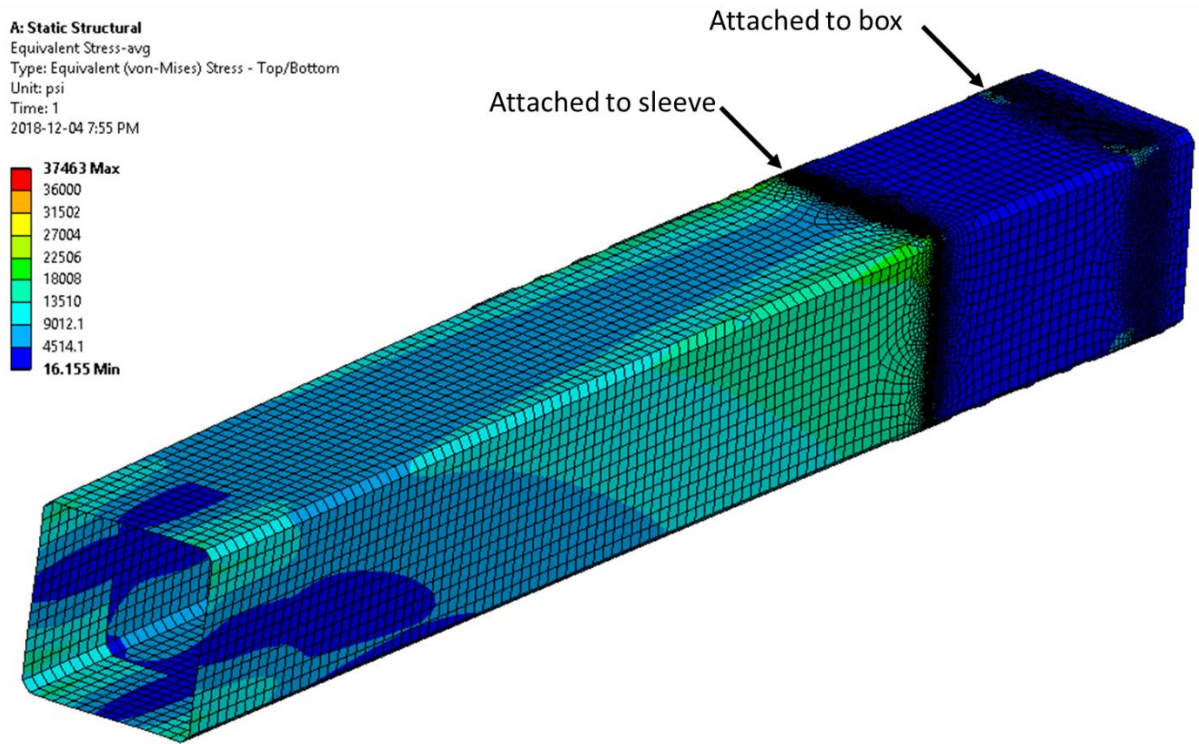


Figure E2. Equivalent stress results for tube (Displacement scale 12; Load Case 2).

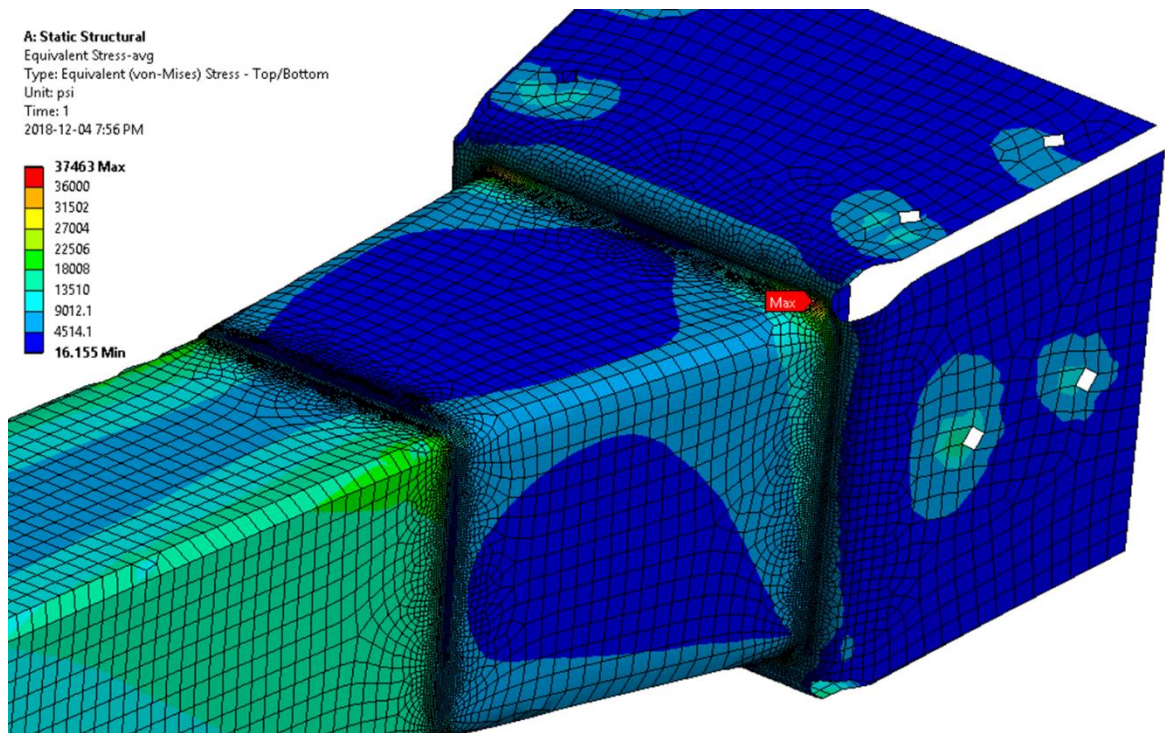


Figure E3. Equivalent stress results for sleeve and box (Displacement scale 12; Load Case 2).

**A: Static Structural**

Equivalent Stress-avg

Type: Equivalent (von-Mises) Stress - Top/Bottom

Unit: psi

Time: 1

2018-12-04 7:58 PM

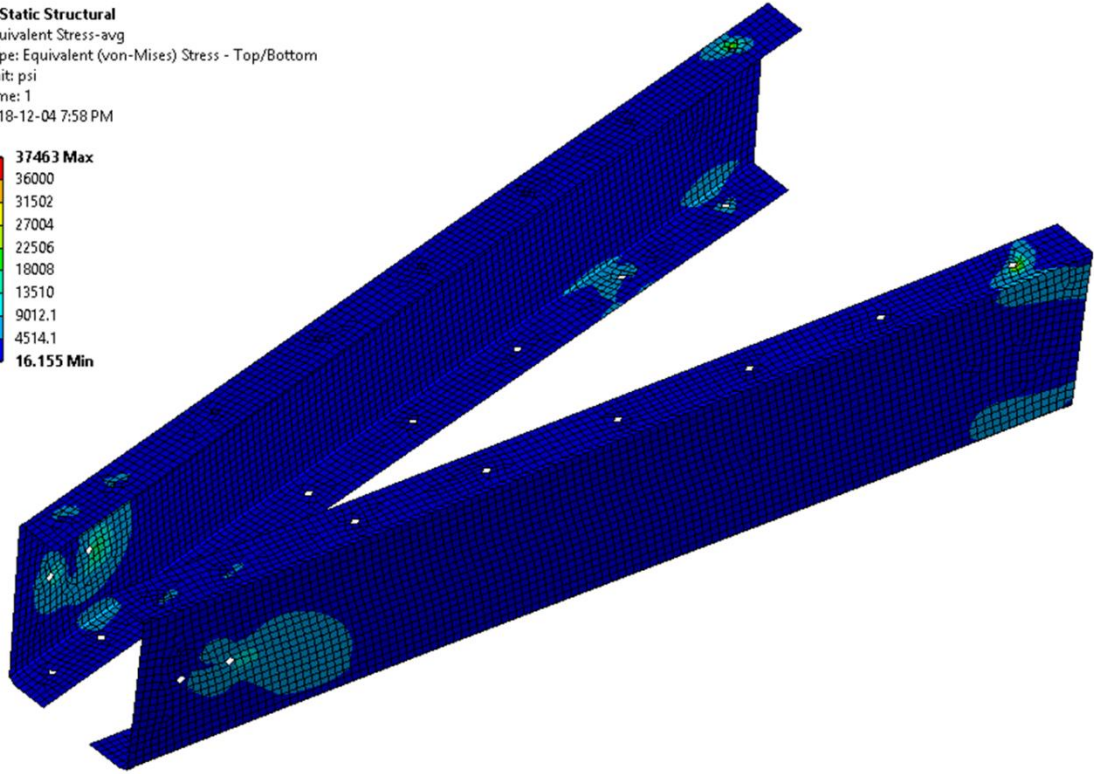
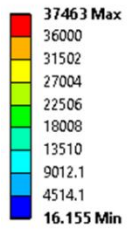


Figure E4. Equivalent stress results for C-channels (Displacement scale 12; Load Case 2).

**A: Static Structural**  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1  
2018-12-04 7:59 PM

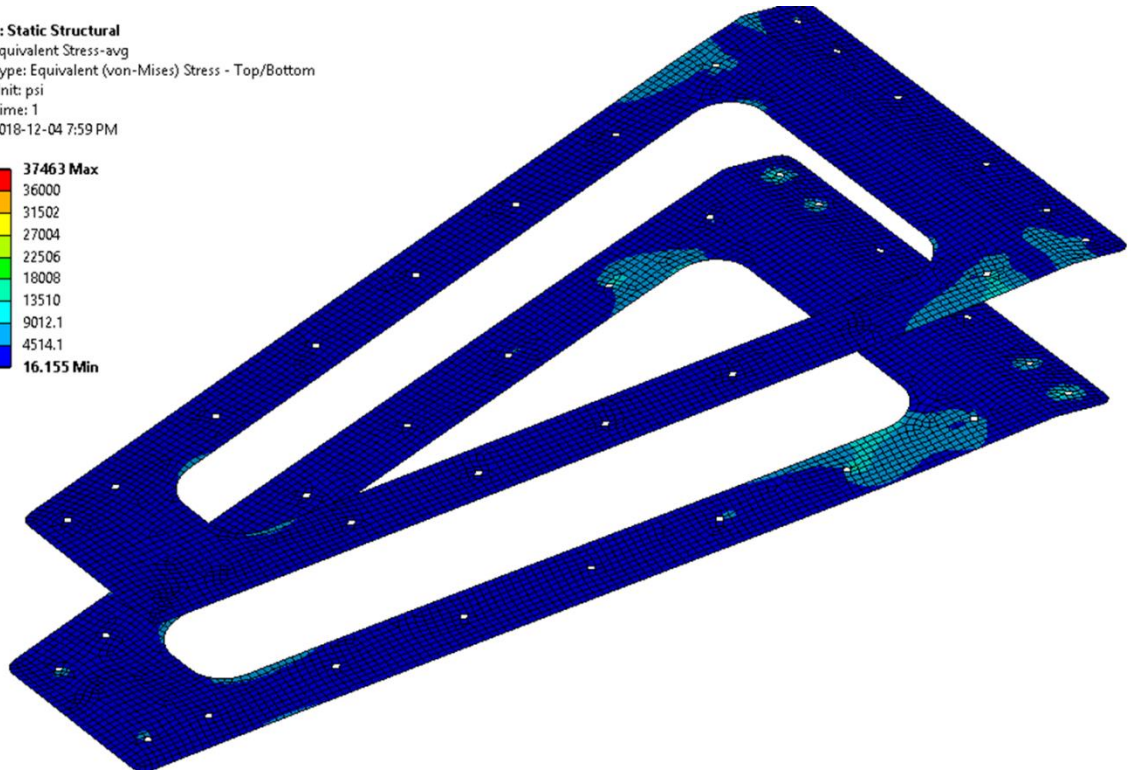
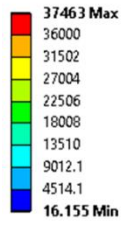


Figure E5. Equivalent stress results for Top and Bottom plates (Displacement scale 12; Load Case 2).

### Load Case 3

**A: Static Structural**  
Total Deformation  
Type: Total Deformation  
Unit: in  
Time: 1  
2018-12-04 8:40 PM

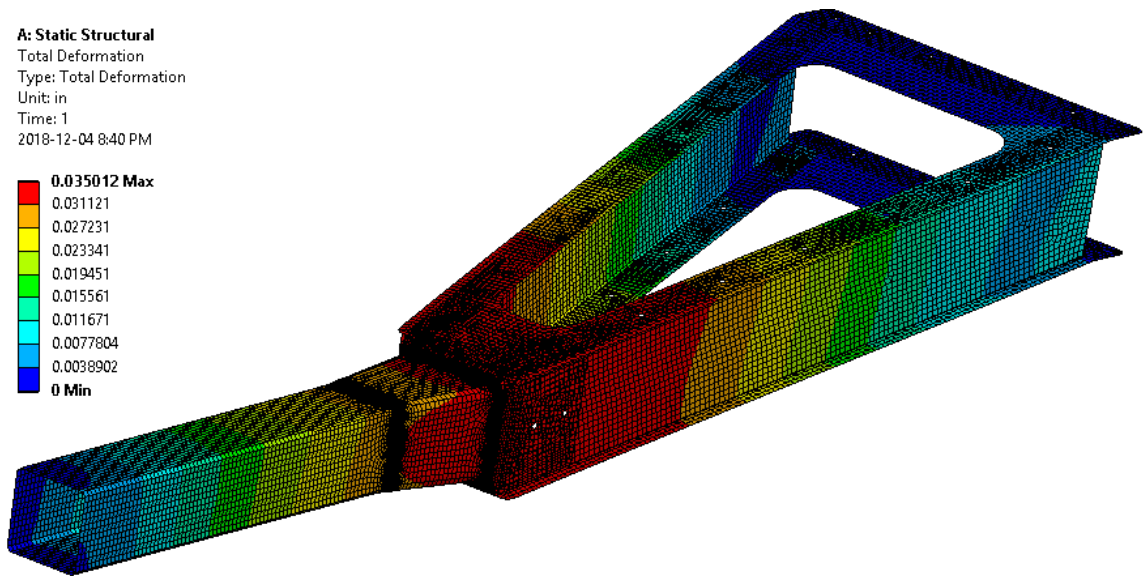
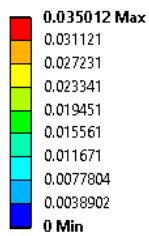


Figure E6. Total deformation for new neck model (displacement scale 12; Load Case 3).

**A: Static Structural**  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1  
2018-12-04 8:42 PM

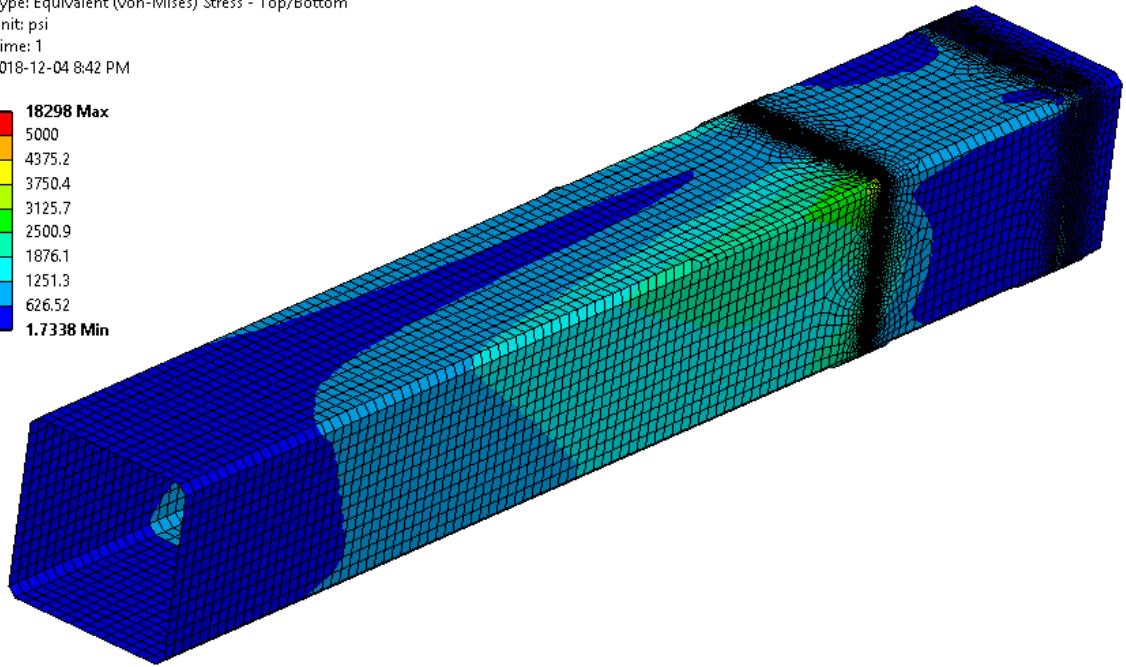
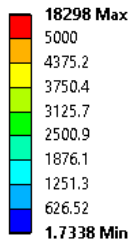


Figure E7. Equivalent stress results for tube (Displacement scale 12; Load Case 3).

**A: Static Structural**  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1  
2018-12-04 8:44 PM

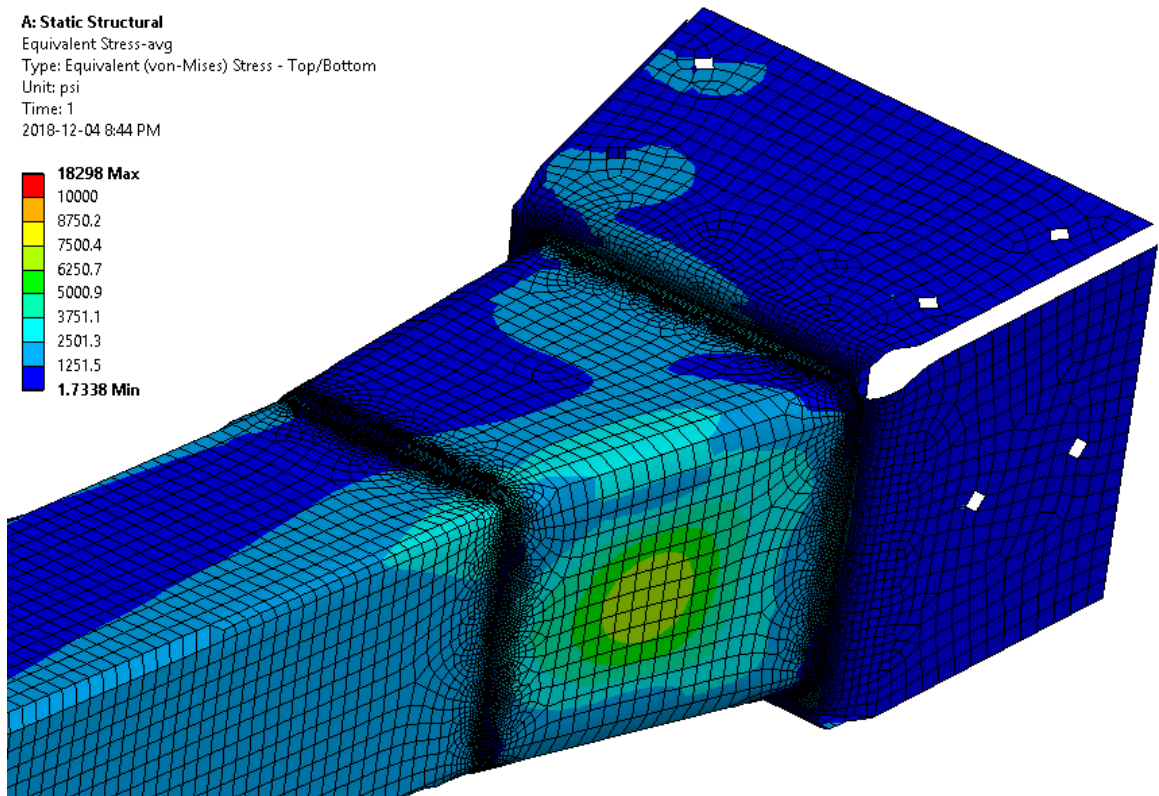
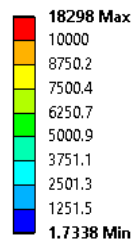


Figure E8. Equivalent stress results for sleeve and box (Displacement scale 12; Load Case3).

**A: Static Structural**  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1  
2018-12-04 8:45 PM

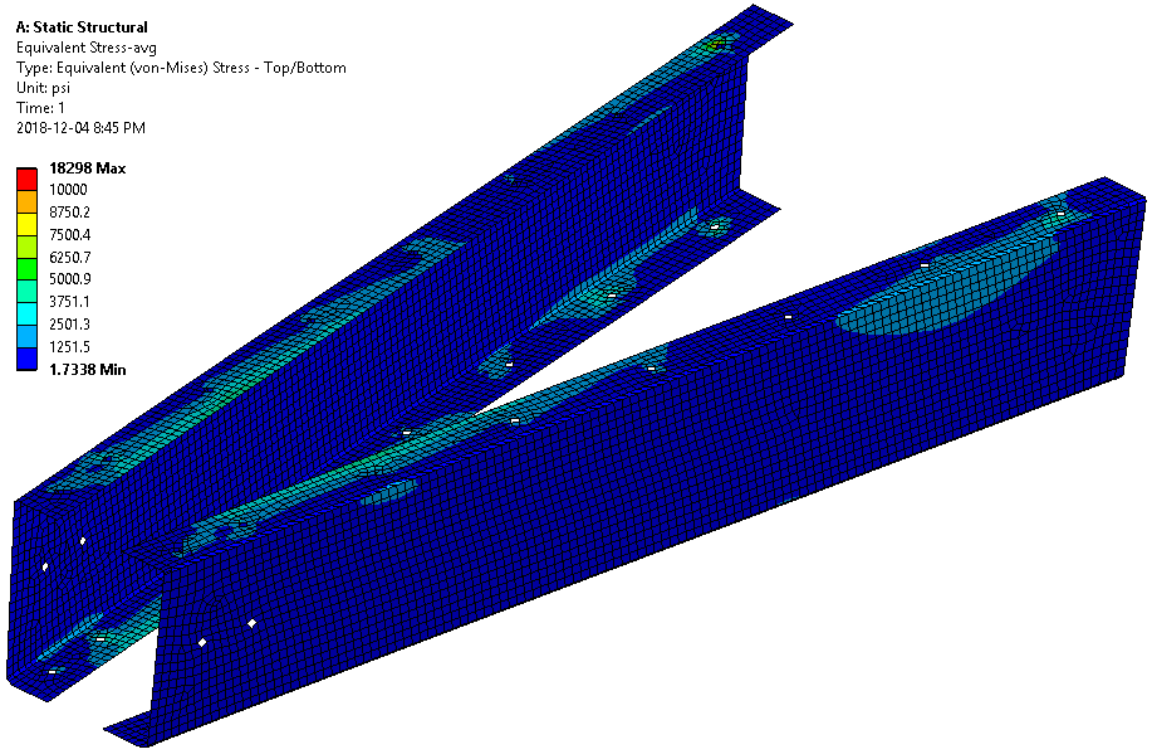
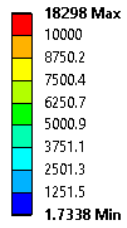


Figure E9. Equivalent stress results for the C-channels (Displacement scale 12; Load Case3).

**A: Static Structural**  
Equivalent Stress-avg  
Type: Equivalent (von-Mises) Stress - Top/Bottom  
Unit: psi  
Time: 1  
2018-12-04 8:47 PM

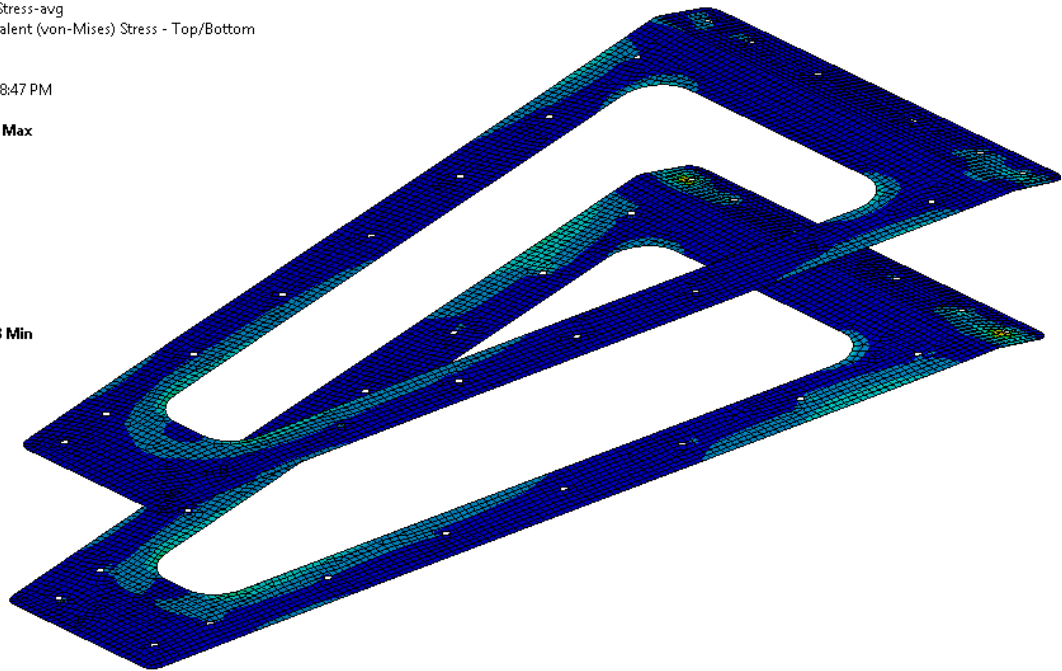
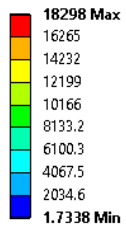


Figure E10. Equivalent stress results for Top and Bottom plates (Displacement scale 12; Load Case3).

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This section provides the preliminary drawings of all the main components of the final neck design.

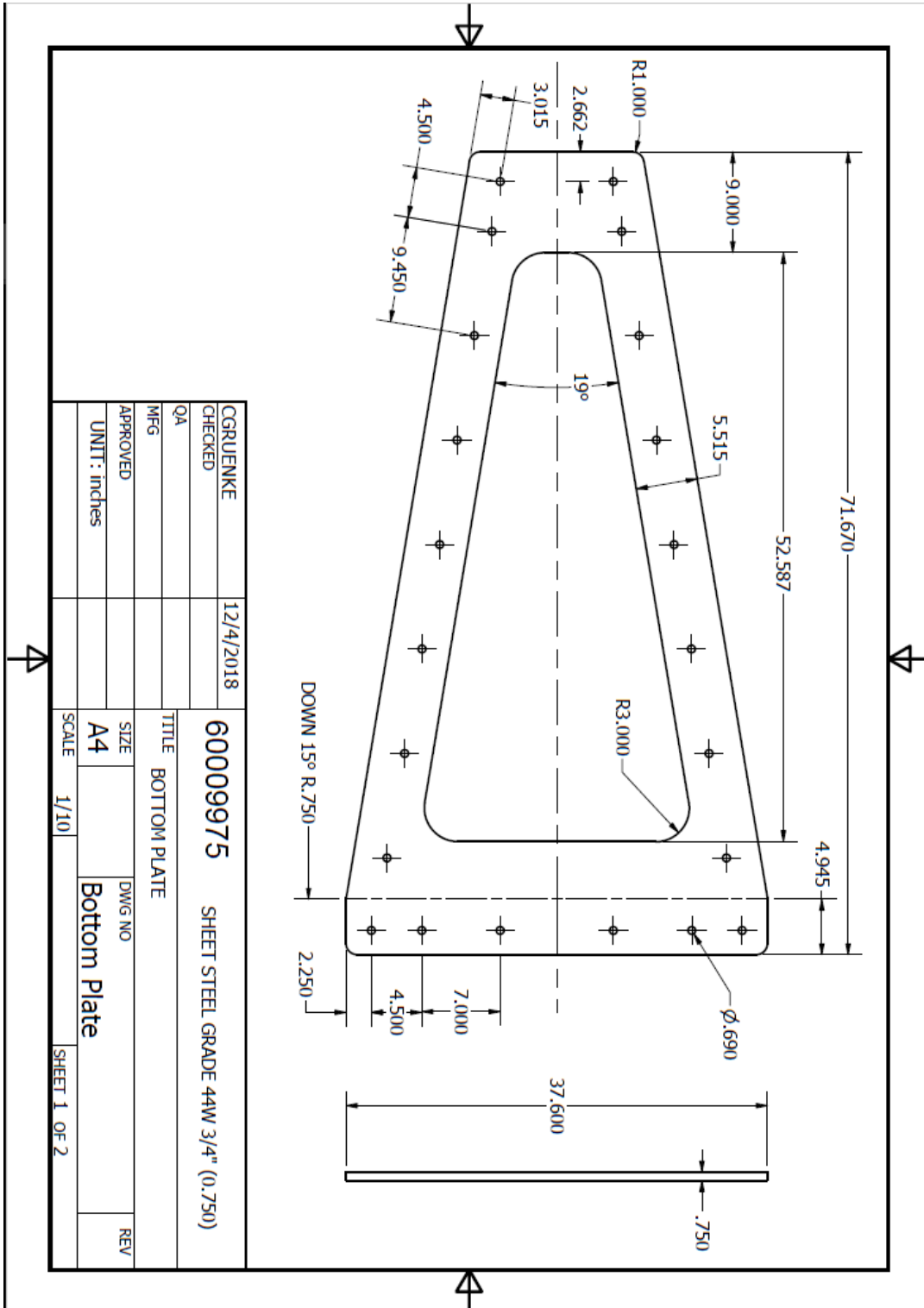


Figure F1. Bottom plate drawing.

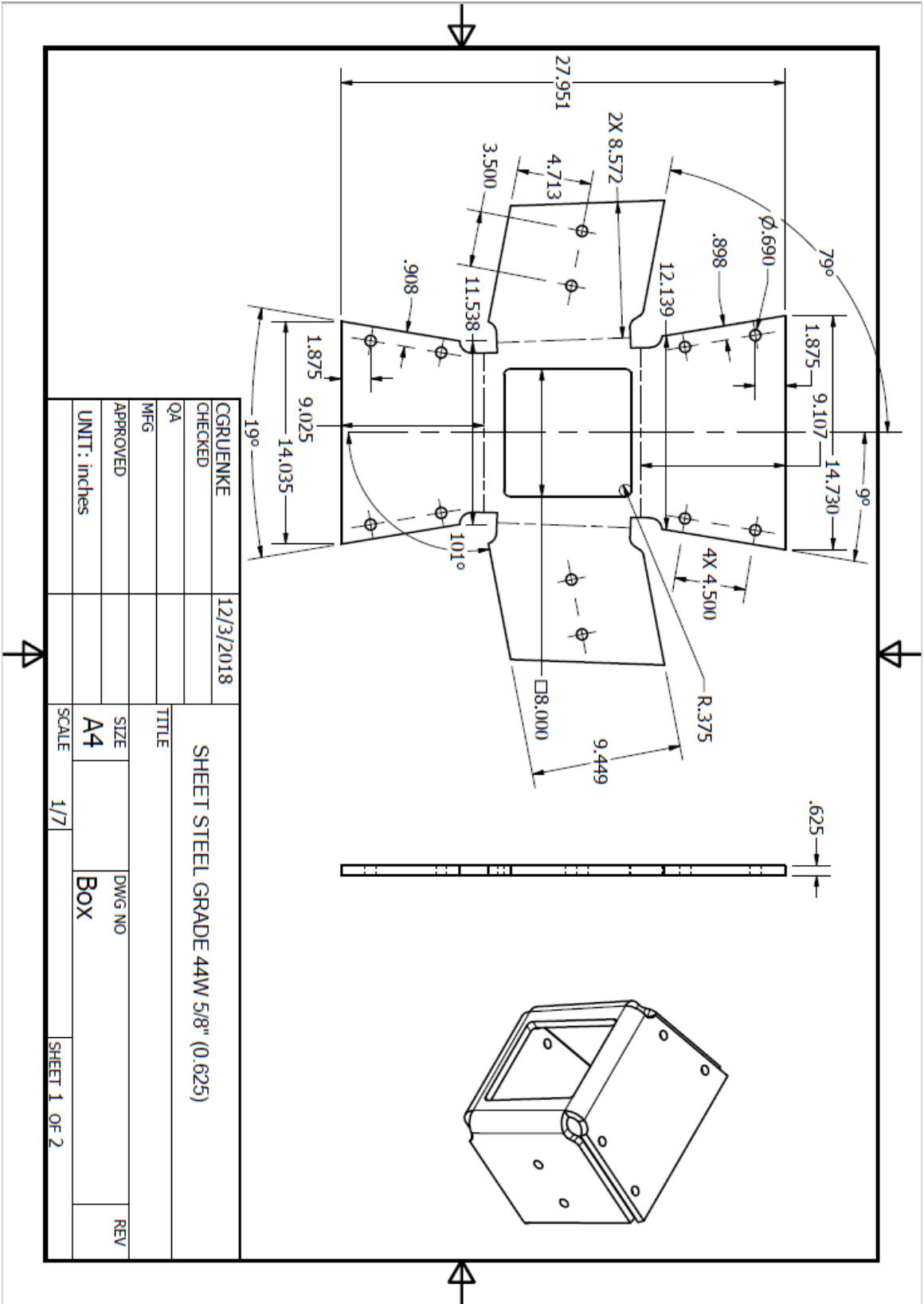


Figure F2. Box drawing.

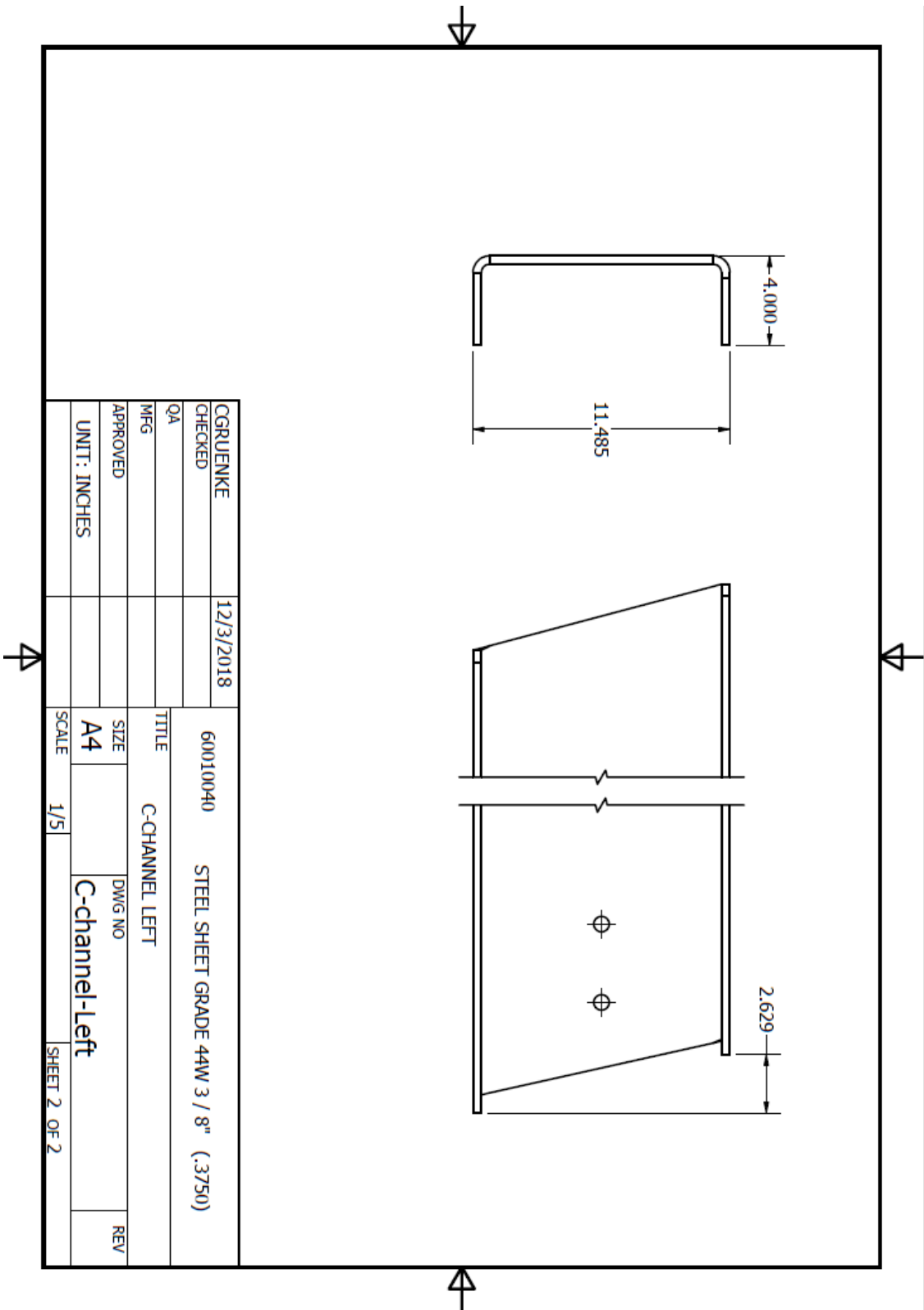


Figure F3. C-channel drawing.

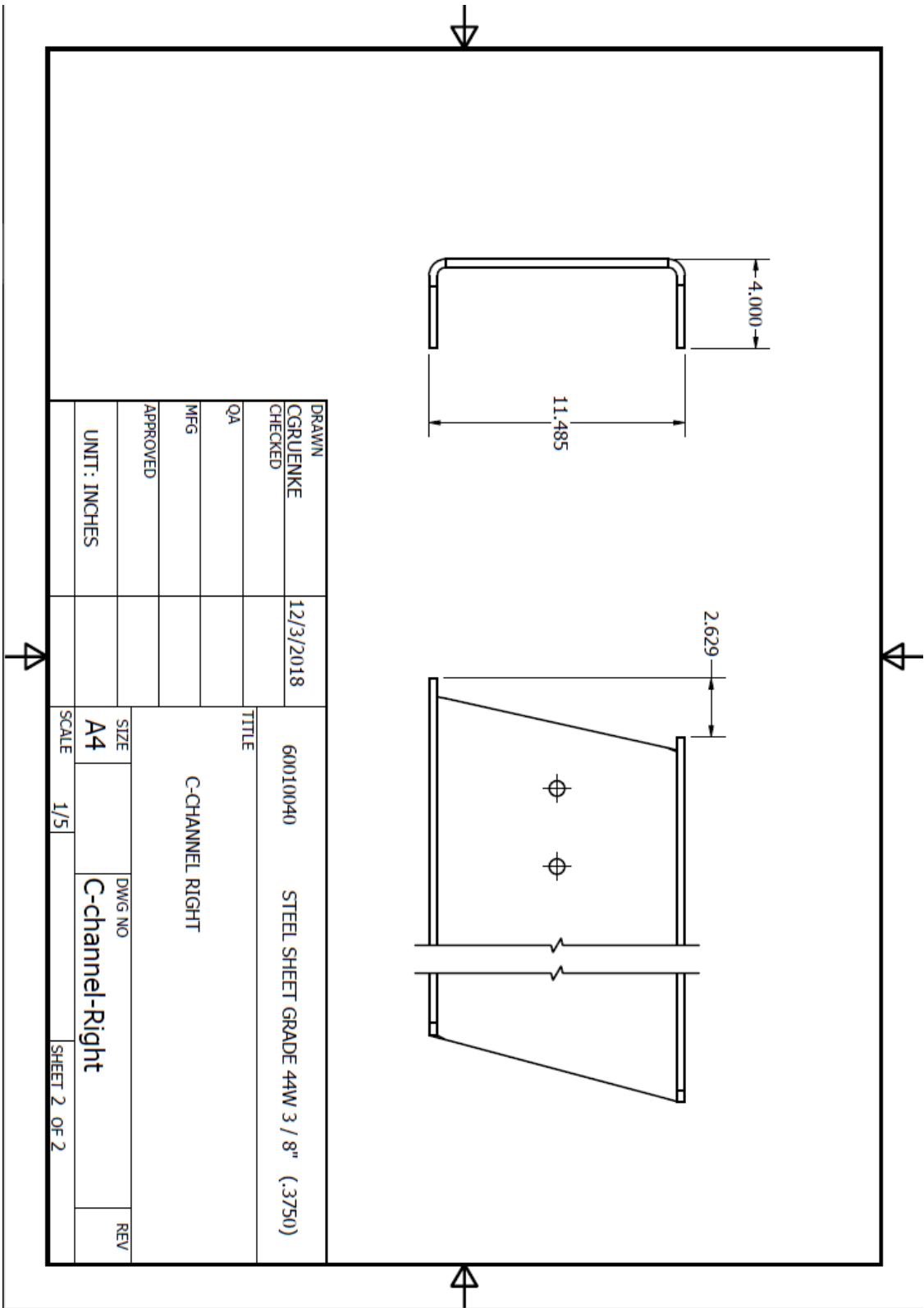


Figure F4. C-channel drawing.

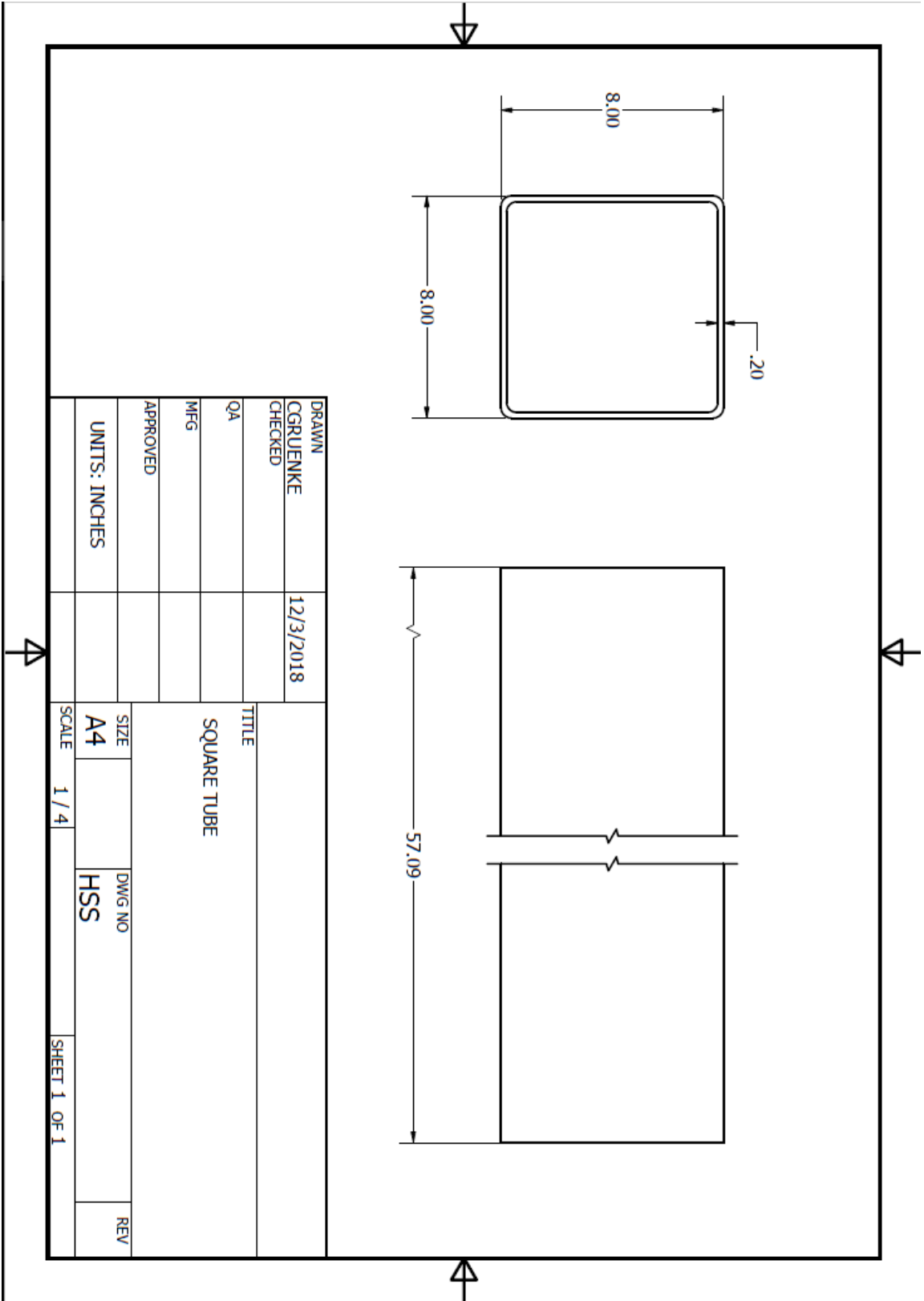


Figure F5. Tube drawing.

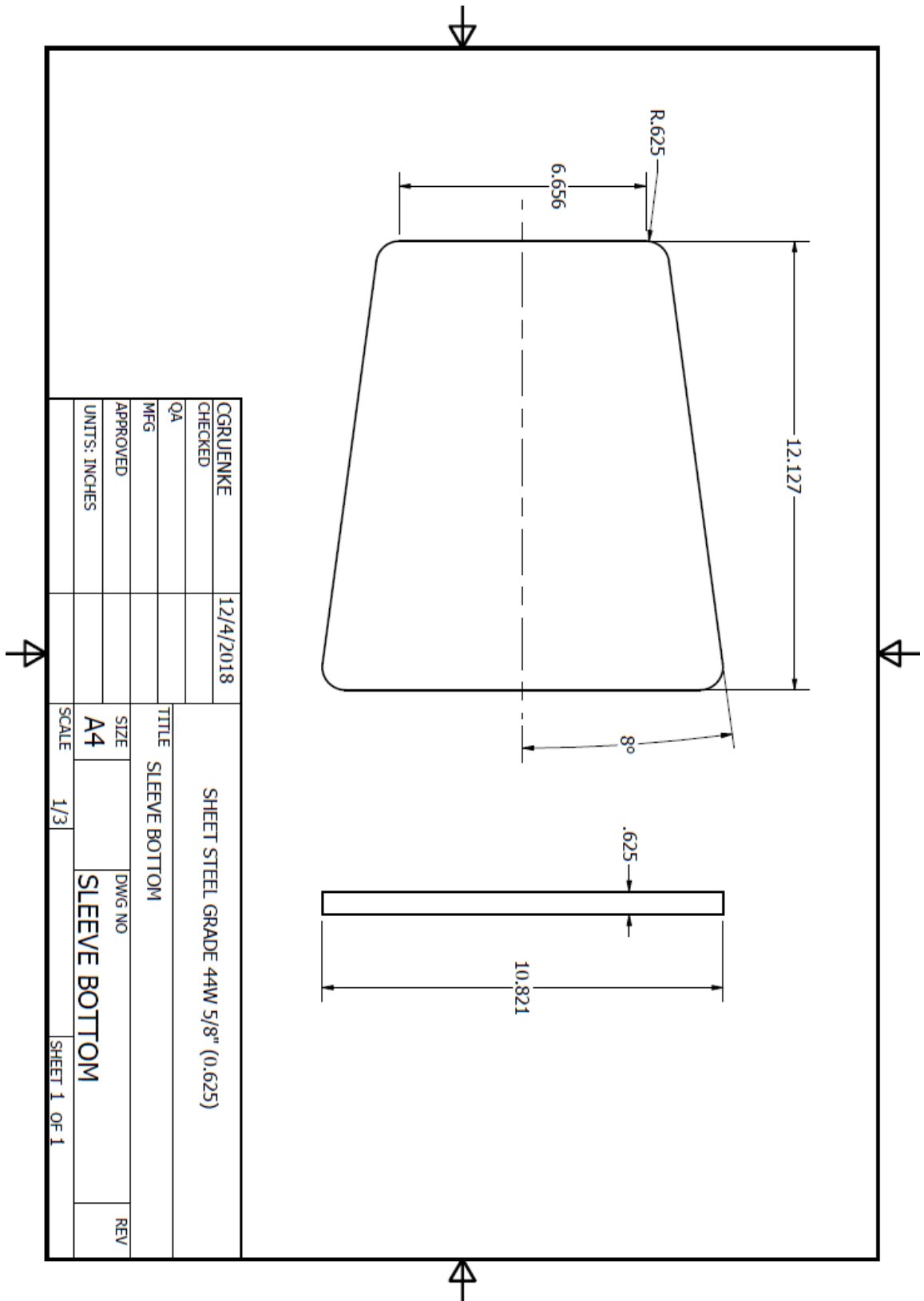


Figure F6. Sleeve (plate) drawing.

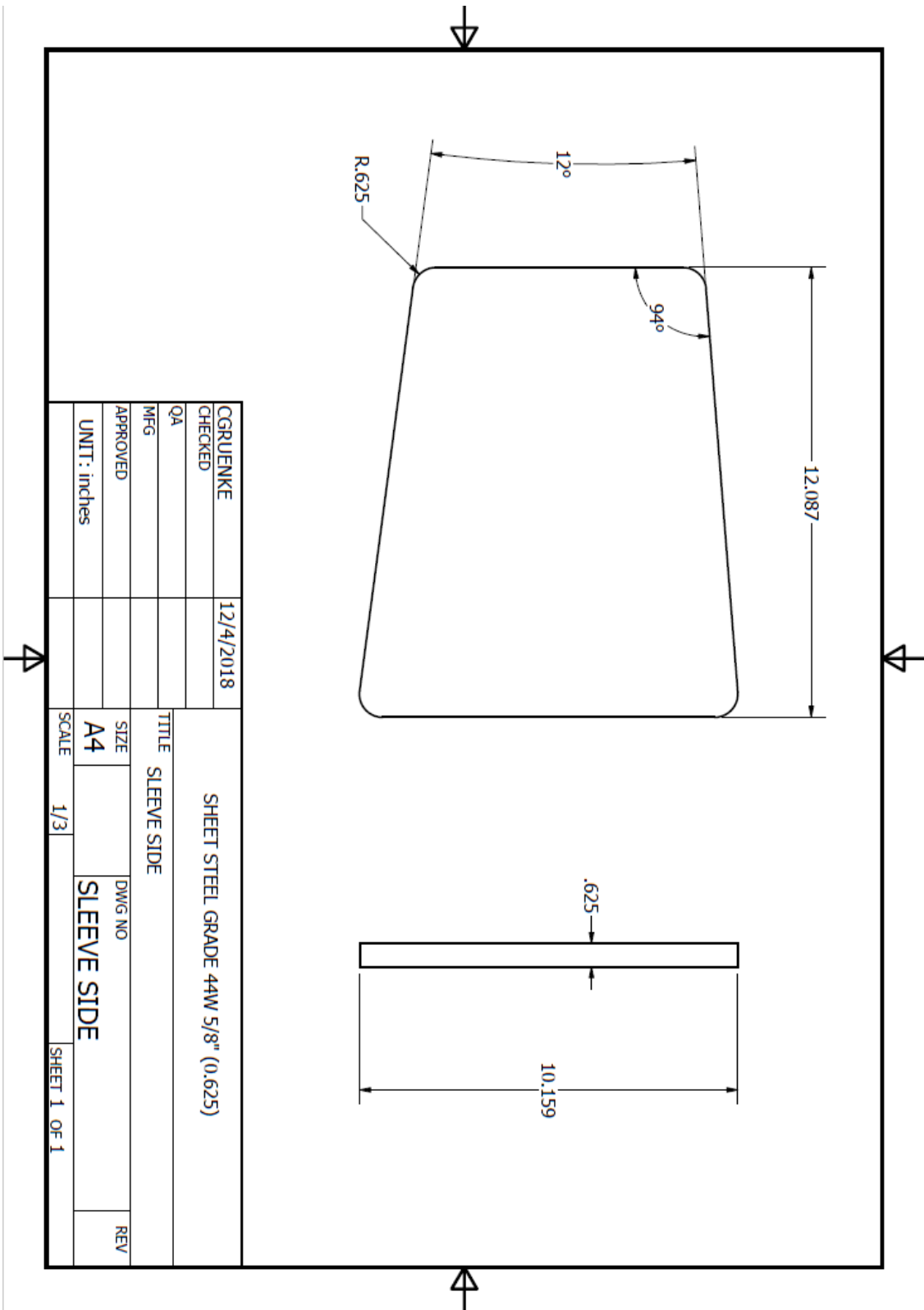


Figure F7. Sleeve (plate) drawing.

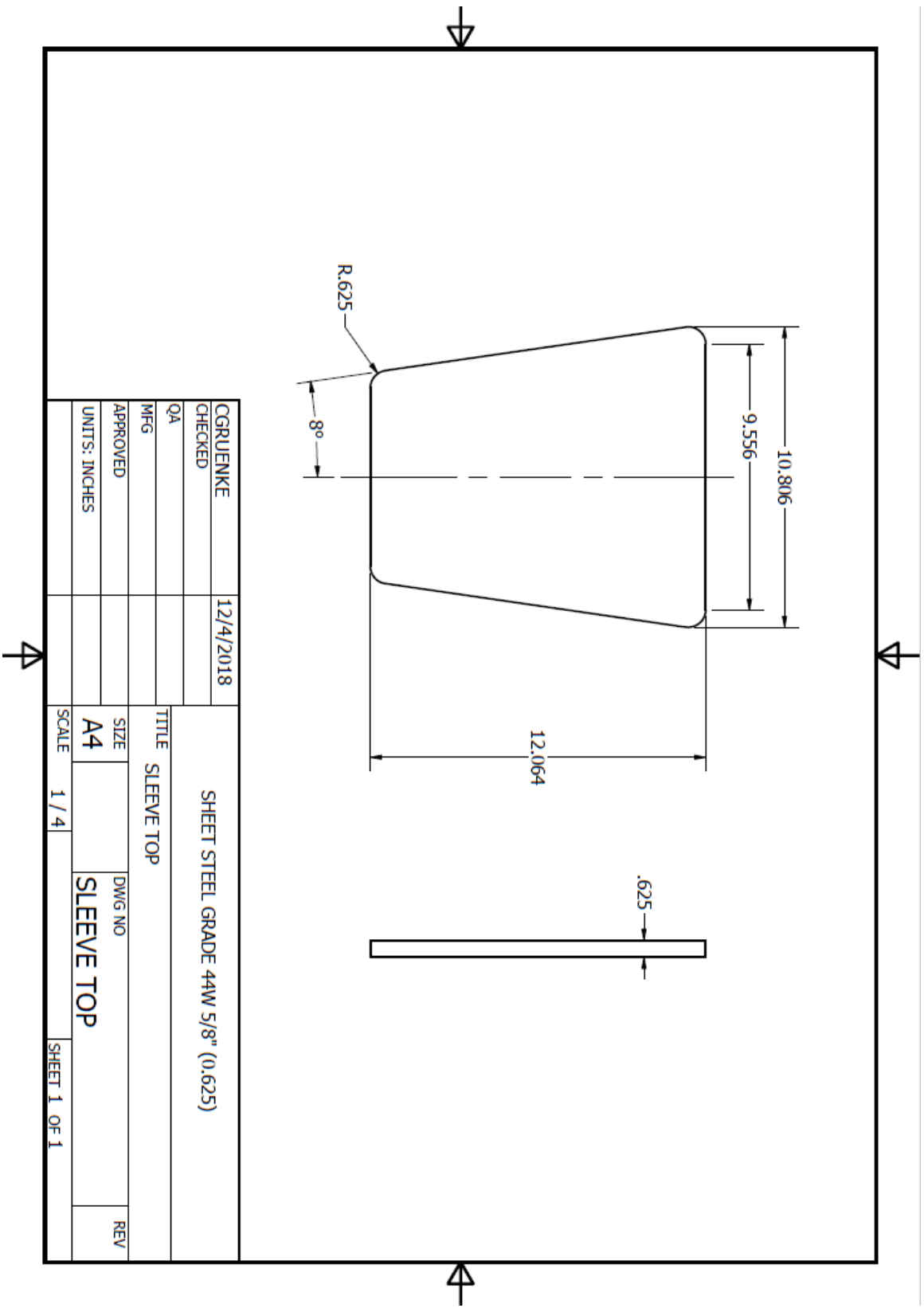


Figure F8. Sleeve (plate) drawing.

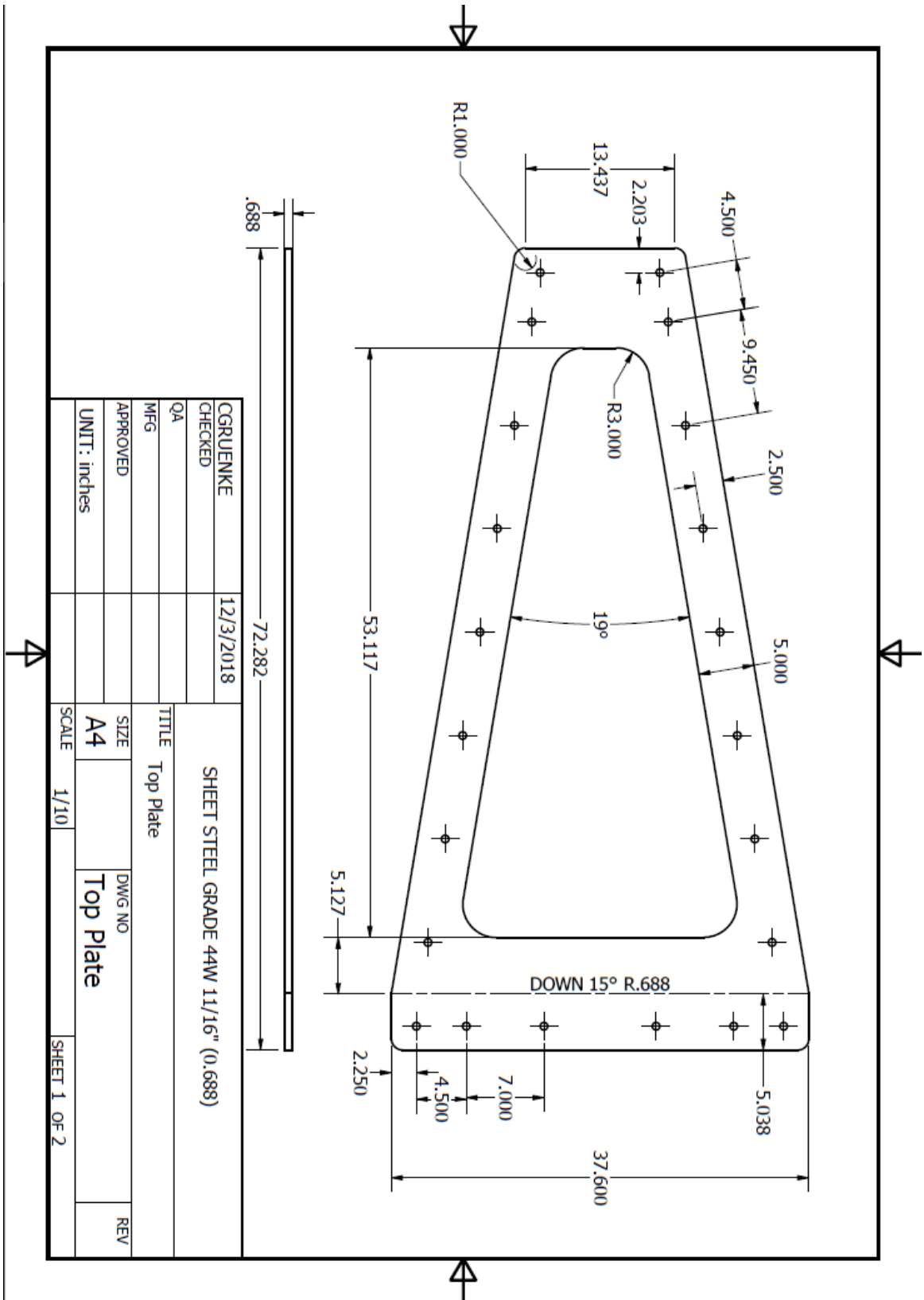


Figure F9. Top plate drawing.