Versateam Engineering



FINAL DESIGN REPORT

Integrated Cab Suspension System for Versatile L4WD Tractor

MECH 4860 - TEAM 1

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December 1st 2014

Dr. Paul Labossiere,
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Dear Dr. Paul Labossiere,

Our team is pleased to submit the attached report in regards to the proposal sent by Buhler Versatile to design a cab suspension system for the Versatile L4WD tractor. The report is written and formatted in accordance with IEEE standards and the criteria set by MECH 4860. Moreover, the cab suspension system design is compliant with the industry safety standards.

Our client, Buhler Versatile, requested our team to design a cab suspension system to reduce cab vibrations as customers have made complaints in regards to the vibrations causing operator discomfort. This report illustrates the detailed stages of the design process.

The design process involved generating concepts, evaluating them based on the criteria provided by the client and selecting the final concept. The suspension system is composed of a combination of a shock absorber and a spring mounted in parallel, reducing vibrations by 79%, satisfying the client's performance goal. As requested by the client, the cab suspension system is designed to be implemented on the rear cab mounts.

The team would like to give special thanks to our advisor, Ed Hohenberg, for the assistance he provided to the team in facilitating the analysis and enabling this project to be completed. The team also gives special thanks to Michael Langan and Buhler Versatile for allowing us an opportunity to experience an industry project upon finalizing the undergraduate degree.

Sincerely,

Versateam Engineering

Executive Summary

This report outlines the design and analysis procedure followed in designing a cab suspension system for Buhler Versatile's L4WD tractor. Buhler Versatile received complaints stating that vibrations were being transmitted to the operator when the tractor was operated on hard surfaces. As a result, the client has commissioned Versateam Engineering to design a marketable and functional cab suspension system.

Buhler Versatile determined that the offending vibrations originated from the interaction of the tracks with hard road surfaces while travelling at speeds above 20km/h. Since the cab is not isolated from the frame, certain offending frequencies were being transmitted to the cab, causing operator discomfort.

The client specified that the goal for the new system was to reduce vibration by 25-50% when compared to the current mounting system. The main design constraint is that the system must satisfy all applicable safety standards and regulations. Low cost, durability and low maintenance were also desired features, as well as the ability to manufacture the system in house or purchase the necessary components from a supplier. The key requirement of the new design is that it must be marketable in order for the L4WD tractor to remain competitive in the marketplace.

The final design consists of a combination of a coil spring and shock absorber mounted at both rear corners of the cab. This design provides the best ability to isolate the cab from the offending vibrations and damp any cab oscillations. It is estimated that the proposed cab suspension system will reduce vibrations by 79%, thereby improving operator comfort. Additionally, this design is comprised of components known to provide successful suspension characteristics, legitimizing the design in the eyes of consumers and aiding its marketability.

CAD model drawings have been created for each individual component as well as for the complete system assembly. Additionally, the overall tractor model provided by the customer has been updated with the suspension system retrofitted into the existing model. Engineering drawings have been produced for the components to be fabricated in house and product requirements have been provided for the components to be purchased from outside suppliers.

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1.0 Introduction

Established in 1932, Buhler Versatile is the largest Canadian agricultural equipment manufacturer. The company headquarters are in Winnipeg, Manitoba, Canada. One of the many different agricultural vehicles that Buhler Versatile currently manufactures is the Versatile L4WD Tractor, shown in Figure 1.



Figure 1: Versatile L4WD Tractor [1]

The L4WD tractor is currently equipped with a cab mounting system that offers very little vibration damping, but satisfies the Rollover Protection System (ROPS) test for 64,000 lbs [2]. Since the cab is fixed to the frame with only small rubber pads for damping, the vibration created when the tractor is driven on hard surfaces is transmitted throughout the tractor and ultimately to the operator. The small rubber dampers have some damping attributes; however, they do not fully eliminate the vibrations that resonate with the human body, causing operator discomfort. The current mounting system is shown in Figure 2, with the rubber mounts indicated by the red arrows. This illustrates how the cab is bolted to the tractor frame with very little means of preventing the transmission of vibration to the cab.



Figure 2: Current rear mount

The L4WD's track drive system, which can be seen in Figure 1, dramatically increases traction and field performance in comparison to standard tractors with tires. However, the tracks cause a unique vibration problem. The vibrations are caused by the tracks when the lugs interact with a hard road surface. Since the tracks are fixed rigidly to the frame, these vibrations are transmitted throughout the tractor. On a typical tractor, the air in the tires dampens the majority of the vibrations. Since the tracks do not provide any damping and there is no isolation system between the tracks and the frame, these vibrations have a greater impact on the comfort of the operator.

By referencing *ISO Standard 2631: Mechanical Vibration and Shock – Evaluation of Human Exposure to Whole-Body Vibration* and cab vibration data provided by Buhler Versatile, the team has observed that due to the relatively high amplitude of vertical vibrations transmitted to the cab at frequencies of 5, 8 and 27 Hz, the operator would potentially feel nauseous and feel discomfort in the upper back, neck and shoulder areas [3]. Since each person has a unique sensitivity to vibration, some operators may be impacted more severely than others.

Currently, one of Buhler Versatile's direct competitors has a suspended cab system that reduces the vibration transmitted to the operator. The competitor's system is complex, comprised of sway bars in addition to active suspension components at all four corners of the cab. Buhler Versatile is looking to implement a suspended cab system to compete in the market while still meeting the mandatory safety standards. The cab suspension must also follow the company's model of designing simple and easy to maintain products.

Lastly, when the team refers to 'offending frequencies' in the report, these are the frequencies which the design aims to reduce, or if possible eliminate. Though objects cannot actually offend, this terminology will be used for the purposes of this report.

1.1 Problem Statement

Buhler Versatile, manufacturer of the Versatile L4WD tractor, has acknowledged that their track drive system is transmitting vibration to the operator when the tractor is operated on hard surfaces above speeds of 20 km/h. Some of these vibrations resonate with natural frequencies of the human body, leading to operator discomfort. A successful cab suspension design will be marketable, compliant with all applicable safety standards and reduce cab vibration by 25-50%. Detailed drawings and analysis of a proposed design of a cab suspension system are presented in this report.

1.2 Objectives

The purpose of this design project is to design a cab suspension system to reduce vibrations transmitted to the operator of the Versatile L4WD tractor. The suspension system will improve operator comfort when travelling on hard surfaces, allowing for prolonged operation without causing discomfort to the operator. This system will also enable the tractor to compare more favorably to the other competitors in the marketplace.

A successful design will be a suspended cab system consisting of the current front mounts with suspension components integrated into the rear mounts that reduces vibration by 25-50% when compared to the current mounting system. In designing the suspension system, the team will focus on isolating vibrations at 5Hz. If the 5Hz frequency can be successfully isolated, the higher order vibrations at 8HZ and 25Hz will also be isolated. This will be explained in

further detail in Section 3.0 of this report. The suspension system will be integrated into the existing tractor without interfering with the articulated steering system or changing the design of the cab. However, Buhler Versatile has acknowledged that they are prepared to make alterations to the design of the mounting brackets to accommodate a new suspension system.

1.3 Needs and Technical Specifications

The needs of the design were compiled with the assistance of the client and are ranked by importance in TABLE I. The primary customer need for the suspension system is marketability. A major driver of this project is that a competitor has an appealing product that our client is not yet able to match. Additionally, it is mandatory for any design to meet all safety requirements before it is released to the market.

TABLE I: DESIGN CRITERIA

Need	Description	Metric	Importance
Design Marketability	Able to compete with competitors products	Subjective Pass/Fail	5
ROPS Compliant	Must pass testing	ROPS Standard	5
OECD Compliant (see Appendix B)	Must pass testing	OECD Standard	5
Vibration reduction of 25-50%	Impacts operator comfort	Vibration Amplitude	5
Flexible assembly options	Option for existing mount or new design in manufacturing	Pass/Fail	4
Durability	Product quality, customer satisfaction	Pass field testing	4
Reliability	Product quality, customer satisfaction	Pass field testing	4
Low Cost	Cost control	CDN Dollars	3
Manufacturability	Either use in house manufacturing capabilities or off the shelf products	CDN Dollars	3
Ease of Installation	Easy to swap with old system and repair in the field	Pass/Fail	3

Other design criteria are durability, reliability, low cost, manufacturability and ease of installation. Ideally, all parts for the design will be possible to manufacture in-house or be available off the shelf from an outside supplier, which would improve the manufacturability and reduce cost.

Finally, the vibration reduction target provided by the client will be used as a goal for our design. The team will aim for as much reduction of the offending frequencies as possible, with 25-50% representing an initial target. The team is tasked with designing a suspension system to solve a vibration problem stemming from the tractor's use of tracks, with the goal of maximizing operator comfort and product marketability.

1.4 Constraints and Limitations

The main constraints in designing a cab suspension system for the Versatile L4WD are industry regulations and space constraints on the tractor.

1.4.1 Space Constraint

As shown in Figure 3, there is a maximum of 217 mm between the bottom of the tractor frame and the bottom of where the mounting bracket on the cab could be moved to.



Figure 3: Mounting Bracket Space Constraint

This is a rigid constraint as the front cab mounts will not be changed. Additionally, the overall orientation of the cab relative to the frame may not change as that would interfere with the steering and braking controls.

The cab mounting bracket, located on the frame, may be redesigned as long as it does not change the height of the cab and does not interfere with the articulated steering system. Figure 4 highlights the articulated steering system as the tractor is turning.



Figure 4: Articulated Steering System

The client has specified that a 25 mm clearance between the steering system and mounting bracket is required. Additionally, the cab is a purchased component and may not be altered in any way.

1.4.2 ROPS

The Rollover Protection Structure (ROPS) standard is designed to ensure operator safety in the event of a vehicle rollover. Primary restrictions imposed by the ROPS criteria are that the cab must not be crushed beneath the weight of the tractor, and that the cab must not become detached from the tractor in a rollover scenario.

The cab is a purchased component and is used on the current production model of this tractor. The cab itself is ROPS compliant, in that it will not be crushed in a rollover scenario. The

design constraint presented by the ROPS standard is that all mounting hardware must be strong enough to ensure that the cab remains fixed to the frame in a rollover scenario. This will be accounted for in the design process and verified through testing by Buhler Versatile as it is mandatory for the system to be ROPS compliant before being released to the market.

1.5 FMEA

Failure Modes and Effects Analysis (FMEA) is a project management tool used to identify, quantify, prioritize and reduce risk associated with different aspects of a project [4]. Though it is more commonly associated with process improvement, FMEA can still be effectively applied to any project as a risk management tool. The goal is to develop action plans for the highest impact risks, contingency plans for the medium risks and monitor the low risks throughout the project.

The team must determine the different ways that each parameter can fail so that each failure mode can be analyzed to determine its impact, or Risk Priority Number (RPN). The potential failure modes are scored on a scale from 1 (good) to 10 (bad) based on their Severity, Occurrence and Detection. The severity ranges from 10 where a human can get injured, 6 where a malfunction arises, to 1 where it's unnoticed and does not impact performance [4]. Examples on the occurrence scale are daily (10), monthly (7), annually (4), and once every 6-100 years (1) [4]. Detection is scored between undetectable (10), to manual monitoring and inspection (5), to all defects are obvious and never reach the customer (1), where a no-inspection system can be implemented with routine monitoring [4].

The RPN is calculated by multiplying the severity, occurrence and detection values for each individual risk, and can range from 1 to 1000. The higher RPN's are the ones to focus on first by determining an appropriate mitigating action and determining the new RPN based on the outcome. The mitigation of risk is an iterative process that is never finished, so it is important to continue to monitor and take action to reduce risk as necessary. In TABLE II, three critical parameters for our project were examined in detail to ensure that the design process was successful. These parameters represent the critical requirements of the client.

TABLE II: FMEA

Critical Parameter	Potential Failure Mode	Potential Failure Effect	Severity	Potential Causes	Occurrence	Current Controls	Detection	RPN	Recommended Action	Responsibility and Target Date	Action Taken	Severity	Occurrence	Detection	New RPN
Market-able	Design not marketable	Customers don't purchase tractor	9	Design doesn't inspire confidence	7	None	6	378	Improve design appeal	Team members 12/02/14	11/13/14	4	4	5	80
Reduces Vibration	Over/under damping	Isolation lost	6	Incorrect C value for shock	10	None	5	300	Determine and apply correct damping constant C	Team, handoff to client	11/27/14	6	2	3	36
	Cab bottoms out	Suspension life reduced	6	Springs have less vertical travel than shock	7	Damper placed between cab and frame	5	210	Ensure shock bottoms out before spring	Pengkai, Brent 11/29/14	11/27/14	6	3	3	54
Satisfies ROPS	Bolts not sufficient	Design fails ROPS test	10	Improper tolerances, calculation error	3	Safety testing	2	60	Continue current ROPS testing	Client	Ongoing	10	3	2	60

Upon consulting with the client and taking into consideration the results of the FMEA analysis, the team decided that marketability is of utmost concern. As such, the suspension system will be designed to be immediately recognizable as an effective suspension system by a potential customer.

With the problem well defined by the client, the design concepts will now be presented and scored according to the criteria laid out in this section, and the team will select one to proceed with to the final design stage.

2.0 Concept Generation and Selection

Through the use of effective research methods, such as brainstorming and TRIZ, the team generated various original concepts to satisfy the design criteria. The concept generation phase was split into two main categories: designing space for the suspension components and selecting the appropriate combination of suspension components.

2.1 Mounting Brackets

The most restricting limitation that was identified by the team was the lack of space available to implement a suspension system between the cab and the frame. To resolve this limitation, the team began by generating concepts in which the mounting brackets would be redesigned to accommodate the suspension components. The concepts are shown in TABLE III.

TABLE III: CONCEPTS TO CREATE SPACE FOR SUSPENSION SYSTEM

Concept	Description
Raise entire cab	Raising the entire cab would provide the space necessary to install a suspension system. The suspension would be mounted on the current rear mounting brackets and spacers would be added to the front cab mounts to level the cab.
Create exoskeleton around cab	With an exoskeleton designed around the cab, the suspension system would be mounted between the exoskeleton and the tractor frame.
Lower current rear mounting brackets	This would create room in between the bottom of the cab and the mounts such that a wide variety of suspension components could be installed.
Design L-shaped brackets to come out the sides of the cab	Brackets would be bolted to the bottom of the cab, then come out to the sides and up like an L in order to generate space for a suspension system.
Design L-shaped brackets to come out the rear of the cab	Similar to the sides, except the brackets would come out the rear of the cab.
Remove current brackets all together	This would create room such that a variety of suspension systems could be installed and mounted directly to the tractor frame via a plate from the cab mount location.

An altered cab mounting bracket will be employed to mount any combination of suspension components, outlined in Appendix D, between the frame and the cab.

2.1.1 Lowered Rear Mounting Bracket

Since the size of the suspension system is limited by the small space between the cab and the original mounting bracket without modifying the bracket, the only option is to change the damping material. As this would not provide a suitable solution, redesigning the brackets is necessary to provide a space in which to install any suspension components. The team generated four ways to create more space: lower the rear bracket, side-mounted suspension system, rear-mounted suspension system and raised the cab. The lower rear bracket design is shown below in Figure 5.

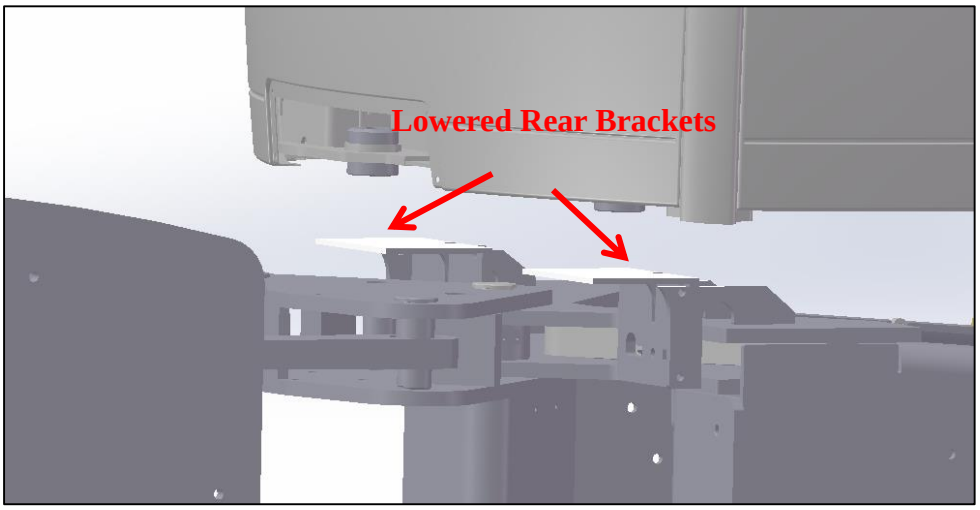


Figure 5: Lowered Rear Brackets

Lowering the rear brackets would allow for more space to install the suspension system directly between the cab and frame. The height between the cab and the tractor frame is 217 mm, while the original height of the bracket connecting the cab with the frame is 196 mm. This new design would lower the bracket height to between 80 mm and 120 mm, allowing a suspension system of 100-150 mm to be installed. The main concern for this type of bracket is that it must not interfere with the articulated steering system.

2.1.2 Side-Mount Bracket

The side-mounted suspension system would eliminate the need to create more room beneath the cab by moving the suspension components to the side of the cab. The bracket is shown in Figure 6.

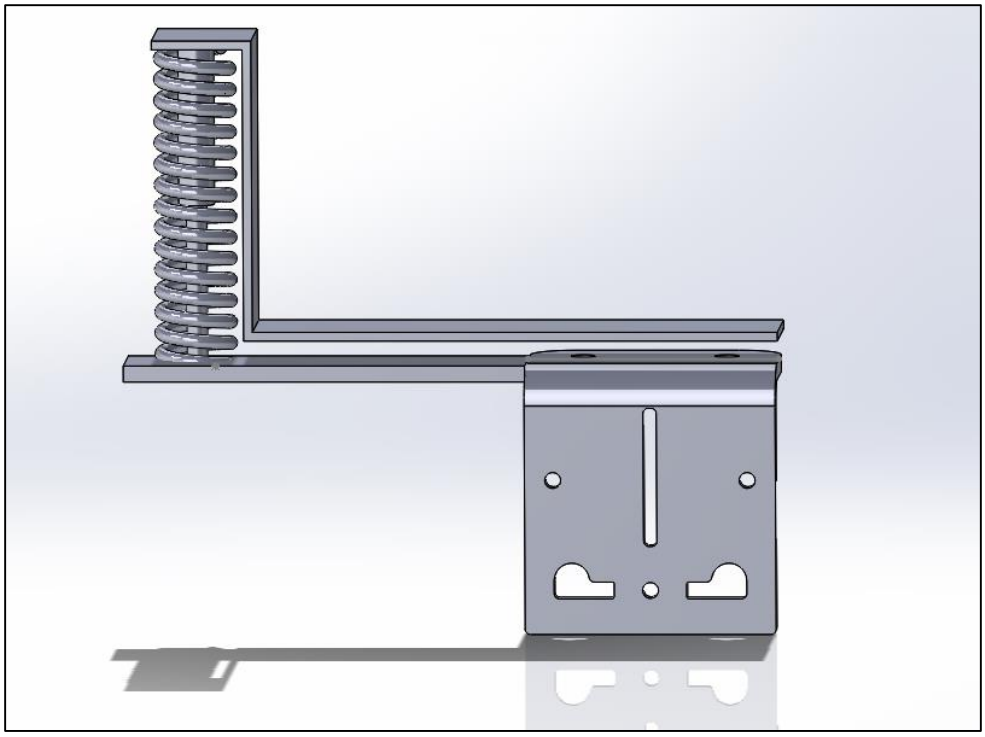


Figure 6: L-Bracket with external shock

In order to mount the suspension on the side of the cab, the top of the bracket needs to be extended so that it protrudes out from the side of the cab. The main constraint with this design is the space limitation on the side of the cab. The space directly adjacent to the cab is occupied by ergonomically designed fuel tanks that cannot be altered. Additionally, the bracket would be subjected to significant force, which would incur large stress on the bracket, necessitating the use of gussets, which would further interfere with the other tractor components on the side of the cab.

2.1.3 Rear-Mount Bracket

The rear-mount bracket is similar to the side-mount bracket in that it moves the suspension components out from under the cab. The top of the bracket is extended to the rear of the cab as shown in Figure 7.

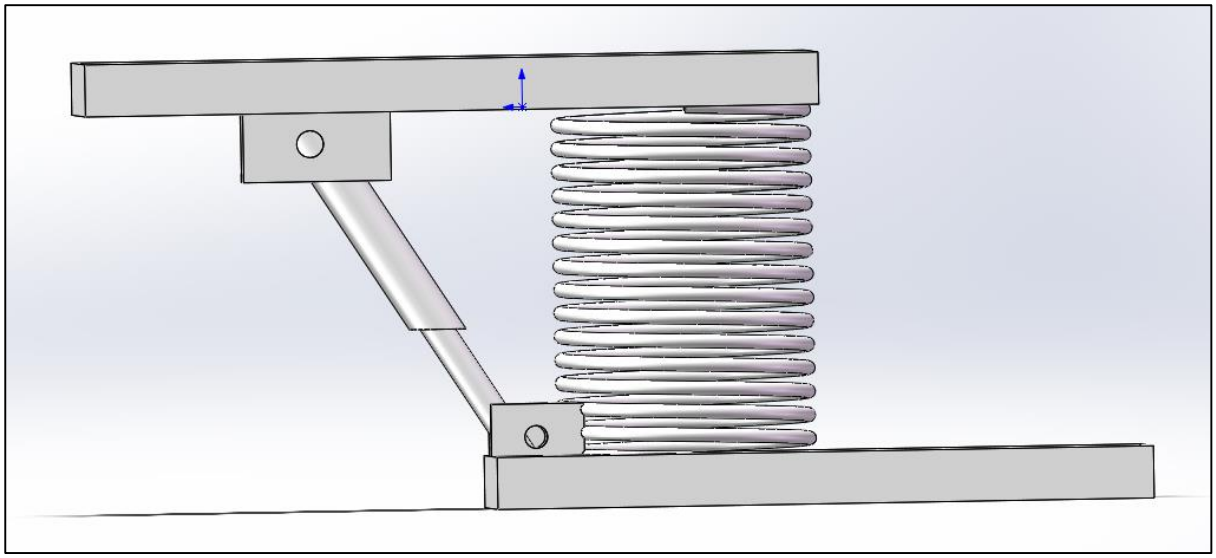


Figure 7: Rear-Mounted Suspension System

Figure 7 shows a shock absorber and coil spring mounted to the rear of the cab. Extending the bracket to protrude behind the cab allows for a variety of suspension components to be installed without being constrained by the small space beneath the cab. One possible configuration is shown here but the same principle may be applied to install other types of suspension.

2.1.4 Cab Elevation

The cab may be elevated by installing spacers on the front cab mounts to level the cab, compensating for the suspension components installed on the rear mounts. The main limitation with this concept is that it would involve altering the configuration of all seating, steering and foot pedals within the cab to compensate for the higher cab mounting position.

2.2 Concept Screening

With an abundance of concepts generated, the team began a process of screening all of the concepts against a number of important criteria in order to eliminate those that were not feasible within the scope of this project. The team then moved to an evaluation process using each of the suspension concepts paired with each of the space generating concepts and compared each variation against the tractor's current design for reference. The reference

design had a score of zero, and each concept with a score of 2 or higher was highlighted and carried to the next round of screening.

The suspension components in consideration are as follows:

A – Modified Damping Material

B – Air Bag

C – Rubber Spring

D – Shock Absorber

E – Shock Absorber and Damping Material Combination

F – Shock Absorber and Spring Combination

G – Isolation Mounts

Ref. – (Current Design)

The following tables show the evaluations and results of each concept compared against the reference.

TABLE IV: SCREENING OF CONCEPT VARIATIONS USING LOWERING BRACKET

	1. Concept Variations with Lowered Brackets							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	-	+	+	+	+	+	0	0
Ease of Manufacturing	0	-	0	0	0	0	0	0
Aesthetically Pleasing	0	+	+	+	+	+	+	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	0	-	0	0	0	0	-	0
Ability to still use current design	+	-	+	0	0	+	+	0
Ease of integration to current design	0	-	0	0	0	0	0	0
Cost	0	-	0	-	-	-	-	0
Maintenance interval	0	-	0	0	-	0	0	0
Pluses	2	3	4	3	3	4	3	0
Sames	6	0	5	5	4	4	4	9
Minuses	1	6	0	1	2	1	2	0
Net	1	-3	4	2	1	3	1	0
Continue	NO	NO	YES	YES	NO	YES	NO	NO

TABLE V: SCREENING OF CONCEPT VARIATIONS USING SIDE-MOUNT BRACKET

	2. Concept Variations with side mounted L-Brackets							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	-	0	0	+	+	+	+	0
Ease of Manufacturing	-	-	-	-	-	-	-	0
Aesthetically Pleasing	0	-	-	+	+	0	0	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	-	-	-	-	-	-	-	0
Ability to still use current design	-	-	-	-	-	-	-	0
Ease of integration to current design	0	-	0	0	0	0	0	0
Cost	0	-	0	-	-	-	-	0
Maintenance interval	0	0	0	0	0	0	0	0
Pluses	1	1	1	3	3	2	2	0
Sames	4	2	4	2	2	3	3	9
Minuses	4	6	4	4	4	4	4	0
Net	-3	-5	-3	-1	-1	-2	-2	0
Continue	NO	NO	NO	NO	NO	NO	NO	NO

TABLE VI: SCREENING OF CONCEPT VARIATIONS WITH REAR-MOUNT BRACKET

	3. Concept Variations with rear mounted L-Brackets							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	-	+	+	+	+	+	0	0
Ease of Manufacturing	-	-	-	-	-	-	-	0
Aesthetically Pleasing	0	-	0	0	+	+	-	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	-	-	-	-	-	-	-	0
Ability to still use current design	-	-	-	-	-	-	-	0
Ease of integration to current design	-	-	-	-	-	-	-	0
Cost	0	-	0	0	-	-	-	0
Maintenance interval	0	-	0	-	-	-	0	0
Pluses	1	2	2	2	3	3	1	0
Sames	3	0	3	2	0	0	2	9
Minuses	5	7	4	5	6	6	6	0
Net	-4	-5	-2	-3	-3	-3	-5	0
Continue	NO	NO	NO	NO	NO	NO	NO	NO

TABLE VII: SCREENING OF CONCEPT VARIATIONS WITH RAISED CAB

	4. Concept Variations with Raised Cab							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	-	+	+	+	+	+	0	0
Ease of Manufacturing	-	-	-	-	-	-	-	0
Aesthetically Pleasing	-	-	-	-	-	-	0	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	-	-	0	0	0	0	0	0
Ability to still use current design	-	-	-	-	-	-	-	0
Ease of integration to current design	-	-	-	-	-	-	-	0
Cost	-	-	-	-	-	-	-	0
Maintenance interval	0	-	0	0	0	0	0	0
Pluses	1	2	2	2	2	2	1	0
Sames	1	0	2	2	2	2	4	9
Minuses	7	7	5	5	5	5	4	0
Net	-6	-5	-3	-3	-3	-3	-3	0
Continue	NO	NO	NO	NO	NO	NO	NO	NO

TABLE VIII: SCREENING OF CONCEPT VARIATIONS WITH EXOSKELETON CONCEPT

	5. Concept Variations with Exoskeleton around Tractor							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	0	0	0	0	0	0	0	0
Ease of Manufacturing	-	-	-	-	-	-	-	0
Aesthetically Pleasing	-	-	-	-	-	-	-	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	0	0	0	0	0	0	0	0
Ability to still use current design	-	-	-	-	-	-	-	0
Ease of integration to current design	-	-	-	-	-	-	-	0
Cost	-	-	-	-	-	-	-	0
Maintenance interval	-	-	-	-	-	-	-	0
Pluses	1	1	1	1	1	1	1	0
Sames	2	2	2	2	2	2	2	9
Minuses	6	6	6	6	6	6	6	0
Net	-5	-5	-5	-5	-5	-5	-5	0
Continue	NO	NO	NO	NO	NO	NO	NO	NO

TABLE IX: SCREENING OF CONCEPT VARIATIONS WITH NO REAR MOUNT BRACKET

	6. Concept Variations with removed rear brackets and mounted to frame							
Selection Criteria	A	B	C	D	E	F	G	Ref.
Marketability against competitor	0	+	+	+	+	+	0	0
Ease of Manufacturing	0	-	0	0	0	0	0	0
Aesthetically Pleasing	0	+	0	+	+	+	0	0
Vibration Reduction	+	+	+	+	+	+	+	0
Durability	-	-	0	0	0	+	0	0
Ability to still use current design	+	+	+	+	+	+	+	0
Ease of integration to current design	0	-	0	0	0	0	0	0
Cost	0	-	0	-	-	-	-	0
Maintenance interval	-	-	0	0	0	0	-	0
Pluses	2	4	3	4	4	5	2	0
Sames	5	0	6	4	4	3	5	9
Minuses	2	5	0	1	1	1	2	0
Net	0	-1	3	3	3	4	0	0
Continue	NO	NO	YES	YES	YES	YES	NO	NO

From TABLES IV-IX, seven concept variations, highlighted in yellow, were identified as being sufficiently better than the current system. These seven concepts were then entered into an additional screening in order to further narrow the concepts to three feasible designs. The difference for this matrix, as seen in TABLE X, is the concepts were additionally screened with the ease of complying with the ROPS standard, which is a strict design constraint.

TABLE X: SCREENING OF TOP 7 CONCEPTS WITH EASE OF ROPS COMPLIANCE INCLUDED

	Top Scoring Concepts Screened Against Ease of ROPS Compliance						
Selection Criteria	1C	1D	1F	6C	6D	6E	6F
Marketability against competitor	+	+	+	+	+	+	+
Ease of Manufacturing	0	0	0	0	0	0	0
Aesthetically Pleasing	+	+	+	0	+	+	+
Vibration Reduction	+	+	+	+	+	+	+
Durability	0	0	0	0	0	0	+
Ability to still use current design	+	0	+	+	+	+	+
Ease of integration to current design	0	0	0	0	0	0	0
Cost	0	-	-	0	-	-	-
Maintenance interval	0	0	0	0	0	0	0
Ease of complying with ROPS	+	+	+	0	0	0	0
Pluses	5	4	5	3	4	4	5
Sames	5	5	4	7	5	5	4
Minuses	0	1	1	0	1	1	1
Net	5	3	4	3	3	3	4
Rank	1	7	2	6	4	5	3
Continue?	YES	NO	YES	NO	NO	NO	YES

After completing this screening, concepts 1C – lowered bracket with rubber spring, 1F – lowered bracket with shock absorber and spring, and 6F – shock absorber and spring mounted directly to frame, were ranked as the top three concepts.

2.3 Criteria Importance Weighting

After our initial screening processes, Versateam engineering emerged with three concepts that were brought forward to the next stage of scoring. In this stage of our concept selection process, the team evaluated the importance of the different criteria of the design in order to fairly determine the appropriate weight of each criterion. The team generated the decision matrix, shown in Figure 8, and sent it to the client to have them evaluate it to ensure the weighting was done according to their specifications.

		Marketability	Manufacturability	Athestics	Vibration Reduction	Durability	Mass	Ease of Integration
	Criteria	A	B	C	D	E	F	G
A	Marketabiltiy		A	A	A	A	A	A
B	Manufacturability			C	D	E	B	B
C	Athestics				D	E	C	C
D	Vibration Reduction					D	D	D
E	Durability						E	E
F	Mass							G
G	Ease of Integration							
	Total Hits	6	2	3	5	4	0	1
	Weightings	0.285714	0.095238	0.142857	0.238095	0.190476	0	0.047619

Figure 8: Design Criteria Weighting by Client

The Weightings row at the bottom of the matrix in Figure 8 shows the weights of the criteria as determined by our client. For sensitivity purposes, we also included the weighting determined by the team, in Figure 9, to see if it would dramatically change the results.

		Marketability	Manufacturability	Athestics	Vibration Reduction	Durability	Mass	Ease of Integration
	Criteria	A	B	C	D	E	F	G
A	Marketabiltiy		A	A	A	E	A	A
B	Manufacturability			B	D	E	B	G
C	Athestics				D	E	C	G
D	Vibration Reduction					E	D	D
E	Durability						E	E
F	Mass							G
G	Ease of Integration							
	Total Hits	3	2	1	4	6	0	2
	Weightings	0.142857	0.095238	0.047619	0.190476	0.285714	0	0.095238

Figure 9: Design Criteria Weights by our team

The biggest discrepancy was with the weighting for marketability, which our client deemed much more important than the team. Additionally, our client placed more emphasis on vibration reduction and less on manufacturability compared to our internal weightings.

2.4 Concept Weighing and Scoring

Using the weights determined by our matrices as shown in Figure 8 and Figure 9, we scored the top three design concepts from our screening process. We performed the scoring using both our design criteria weights as well as those provided by our client for sensitivity purposes. The results of this scoring are shown in TABLE XI and TABLE XII.

TABLE XI: SCORED CONCEPTS WITH CLIENT'S CRITERIA WEIGHTING

Weighted Design Selection							
		Lowered bracket with spring and shock combo		Modified mounting with spring and shock combo		Lowered bracket with rubber spring	
		1F		6F		1C	
Criteria	**Weight**	**Score**	**Weighted**	**Score**	**Weighted**	**Score**	**Weighted**
Marketability	0.285714	5	1.42857	5	1.42857	4	1.142856
Manufacturability	0.095238	4	0.3809524	4	0.3809524	4	0.3809524
Aesthetics	0.142857	4	0.571428	3	0.428571	3	0.428571
Vibration Reduction	0.238095	4	0.95238	4	0.95238	4	0.95238
Durability	0.190476	4	0.761904	3	0.571428	4	0.761904
Mass	0	0	0	0	0	0	0
Ease of Integration	0.047619	4	0.190476	3	0.142857	4	0.190476
			0		0		0
Total			4.285710381		3.904758381		3.857139381
Rank			1		2		3
Develop			YES		NO		NO

TABLE XII: SCORED CONCEPTS WITH INTERNAL CRITERIA WEIGHTING

Weighted Design Selection							
		Lowered bracket with spring and shock combo		Modified mounting with spring and shock combo		Lowered bracket with rubber spring	
		1F		6F		1C	
Criteria	**Weight**	**Score**	**Weighted**	**Score**	**Weighted**	**Score**	**Weighted**
Marketability	0.142857	5	0.714285	5	0.714285	4	0.571428
Manufacturability	0.095238	4	0.3809524	4	0.3809524	4	0.3809524
Aesthetics	0.047619	4	0.190476	3	0.142857	3	0.142857
Vibration Reduction	0.190476	4	0.761904	4	0.761904	4	0.761904
Durability	0.285714	4	1.142856	3	0.857142	4	1.142856
Mass	0	3	0	3	0	3	0
Ease of Integration	0.095238	4	0.380952	3	0.285714	4	0.380952
			0		0		0
Total			3.571425381		3.142854381		3.380949381
Rank			1		3		2
Develop			YES		NO		NO

With the results of our scoring matrices, and after consultation with our client, our team selected and will develop concept 1F, which is the concept that uses a spring and shock in combination, mounted between the tractor cab and a lowered mounting bracket.

This design has excellent marketability due to its recognition as a suspension system, will be easily integrated and manufactured, and will allow for successful cab vibration reduction. Appropriate springs and shocks are widely available from suppliers and can be sourced cost effectively while modified mounting brackets and hardware are available on site at Buhler Versatile.

3.0 Technical Considerations

In order to design a functional suspension system, isolation, damping and transmissibility must be analyzed, as each of these characteristics heavily influences the behavior of the system.

3.1 Vibration Isolation

Vibration isolation is a technique in which vibration is prevented from entering the isolated components. The isolation of a system is a function of its natural frequency, which is dictated by its mass and spring characteristics. In our case, we are attempting to isolate the cab from the vibrations transmitted to it by the frame. As such, the mass of the cab and the spring characteristics of the suspension system dictate its natural frequency and ultimately, its isolation [5].

In order to isolate the offending vibrations, the natural frequency of the cab must be lower than the lowest offending frequency divided by $\sqrt{2}$ as shown in equation 1.

$$f_n = \frac{f_o}{\sqrt{2}} \quad \text{(eq 1)}$$

Where f_n is the natural frequency of the cab and f_o is the lowest offending frequency. The natural frequency of the cab is dependent on the spring constant, k, of the suspension system as shown in equation 2.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m_{eq}}} \quad \text{(eq 2)}$$

Where m_{eq} is the equivalent mass of the cab. In this scenario, the mass of the cab is fixed and cannot be altered as it is a purchased component. As such, the only way to modify its natural frequency is to modify the spring characteristics of the cab suspension system.

3.2 Damping

Determining the appropriate spring constant of the suspension system ensures that the cab is isolated from the offending frequencies. However, the suspension system must consist of

damping components to stop the cab from oscillating in the event of an excitation. Damping is a property that ensures that the vibration energy of the cab is dissipated and the cab returns to its equilibrium position.

The damping of a system is typically described by its damping ratio, shown in equation 3.

$$\zeta = \frac{c}{c_c} \quad \text{(eq 3)}$$

Where ζ is the damping ratio, c is the damping constant of the system and c_c is the system's critical damping constant, shown in equation 4.

$$c_c = 2\sqrt{km_{eq}} \quad \text{(eq 4)}$$

It can be seen that the desired damping characteristics of the system are dictated by the spring characteristics. Figure 10 shows the vibration response of a system for various damping ratios.

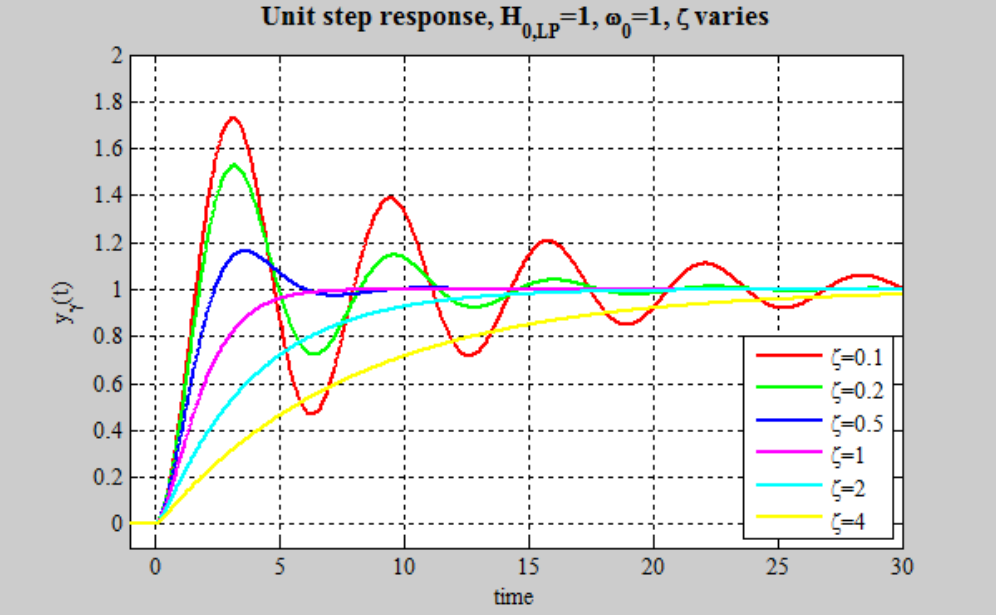


Figure 10: Vibration Response for Various Damping Constants [6]

A system with very little damping is called an underdamped system. An underdamped system has a high amplitude of oscillation and oscillates for longer before returning to equilibrium, as shown by the red line in Figure 10. A damping ratio of 1, shown by the pink line in Figure 10 signifies a critically damped system and returns to equilibrium quickly without oscillating.

Conversely, an overdamped system with a damping ratio above 1 does not oscillate, but takes longer to return to equilibrium, as shown by the yellow line. Typically, a critically damped system is desirable as it returns to equilibrium as fast as possible and does not oscillate.

3.3 Transmissibility

The damping ratio of the system also relates to the system's transmissibility, described by equation 5.

$$T = \sqrt{\frac{1+(2\zeta\frac{f_o}{f_n})^2}{[1-(\frac{f_o}{f_n})^2]^2+[2\zeta\frac{f_o}{f_n}]^2}} \qquad \text{(eq 5)}$$

Transmissibility, *T*, is the ratio of the system's response to an input. A transmissibility of 1 signifies that the system's response has the same amplitude as the excitation. The forcing frequency is represented by f_o and natural frequency by f_n.

As it relates to suspension systems, a lower transmissibility is desired for a smoother ride. A damping ratio close to 1.0 is desirable, as it ensures the system returns to equilibrium as fast as possible. However, from equation 5, it can be seen that transmissibility and damping ratio are negatively proportional. That is, altering the system to make the damping ratio more desirable makes the transmissibility less desirable, as shown in Figure 11.

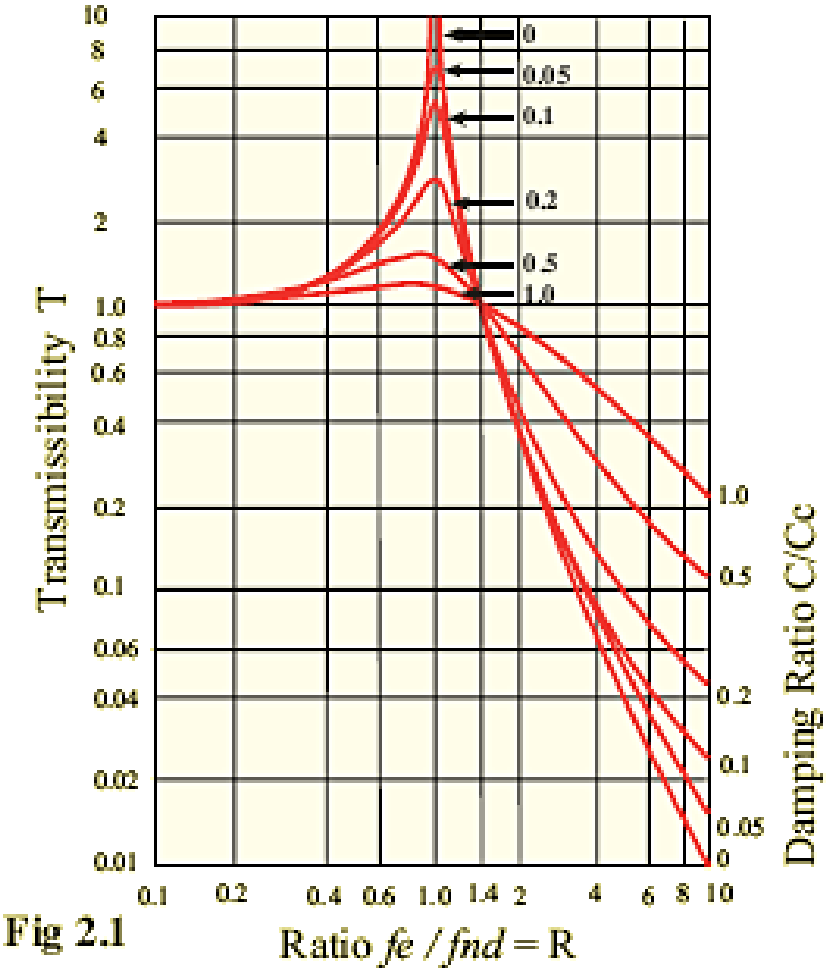


Figure 11: Transmissibility as it Relates to Damping Ratio [7]

The negative relationship is evident in the isolation region in Figure 11, characterized by a frequency ratio above 1.4. Raising the damping ratio from underdamped to approach critical damping raises the transmissibility, which is detrimental to isolation.

A vibrating system resonates at a frequency ratio of 1.0, shown by the peak in Figure 11. It can be seen that the damping ratio also affects the resonant peak. A damping ratio closer to 1.0 damps the resonant peak but raises the transmissibility in the isolation region. Conversely, reducing the damping ratio to 0 lowers the transmissibility in the isolation region but allows the resonant peak to reach a large amplitude, corresponding to a system with uncontrollable vibrations at the resonant frequency.

The team concluded that a damping ratio of $\zeta = 0.5$ would be a reasonable compromise to control the transmissibility peak at the cab's natural frequency and reduce the transmissibility in the isolation region.

3.4 Methods and Assumptions

The methods and assumptions used to analyze the behavior of the current system and determine the required suspension characteristics are outlined below.

3.4.1 Single Degree of Freedom Assumption

To simplify the analysis, the cab is modeled as a single degree of freedom system vibrating in the vertical direction. As shown in Figure 12, the cab is assumed to act as a single mass attached to the tractor frame by a spring and damper suspension system.

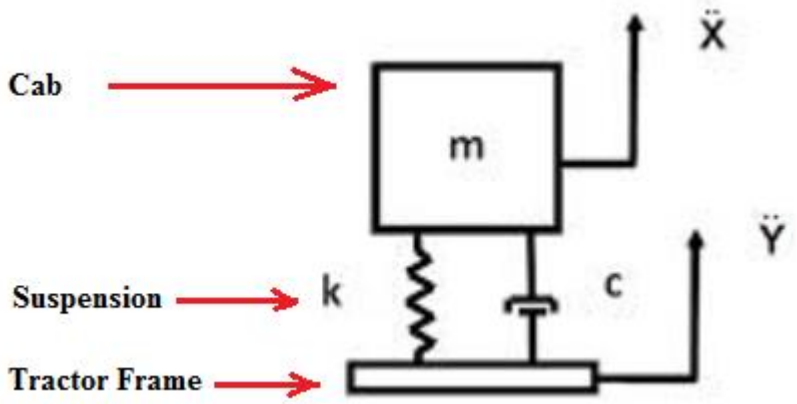


Figure 12: Single Degree of Freedom Vibration Model

Where k is the spring constant, c is the damping constant, m is the mass of the cab, $\ddot{X}$ is the vertical acceleration of the cab and $\ddot{Y}$ is the vertical acceleration of the frame. The track drive system is mounted rigid to the cab and transmits the vibrations caused by the tracks interacting with the road surface directly to the frame. As such, the tractor frame is assumed to act as the base that transmits the vibrations to the cab.

3.4.2 Kinetic Energy Equivalence Method

In order to perform the kinetic energy equivalence analysis, the two front mounts of the cab were assumed to act as hinges. Figure 13 depicts the center of mass of the cab as a point, CM

that hinges about the front of the cab, point O. The rear mounts are replaced with a spring of spring constant k, located at the rear cab mounts at a distance L from point O. The center of mass is located at a distance L_1 from point O, where L_1 is assumed to be 55% of the way from the front of the cab to the rear, shown by the point on the beam. It should be noted that the front mounts are assumed to be rigid and do not experience any vibration.

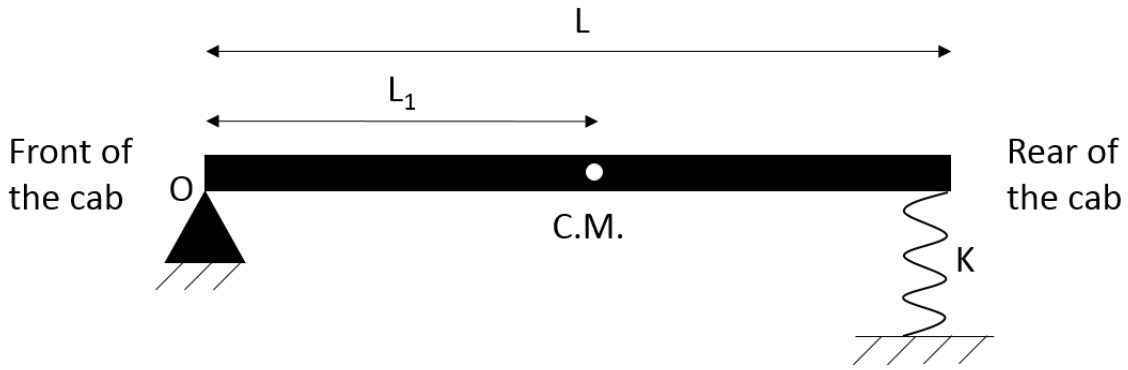


Figure 13: Centre of Mass Assumption

As there was no information available from the client regarding the center of mass of the cab, the team assumed the center of mass to be located at 55% the distance from the front of the cab to the rear because the operator's seat is located at the rear of the cab. This is a conservative assumption because assuming the center of mass to be closer to the middle reduces the weight acting on the rear of the cab, which makes finding suitable isolation characteristics more difficult.

If the center of mass is actually further back, the determined characteristics will still be valid. Additionally, the height of the center of mass is assumed to be on the cab floor. Since the cab hinges according to the small angle approximation, the height of the center of mass may be neglected as it has little effect on the behavior.

Figure 14 illustrates the displacement relationships of the cab, where $\dot{x}_{CM}$ and $\dot{x}_{eq}$ represent the rate of displacement of the center of mass of the cab and the equivalent mass of the cab situated at the rear mounts, respectively.

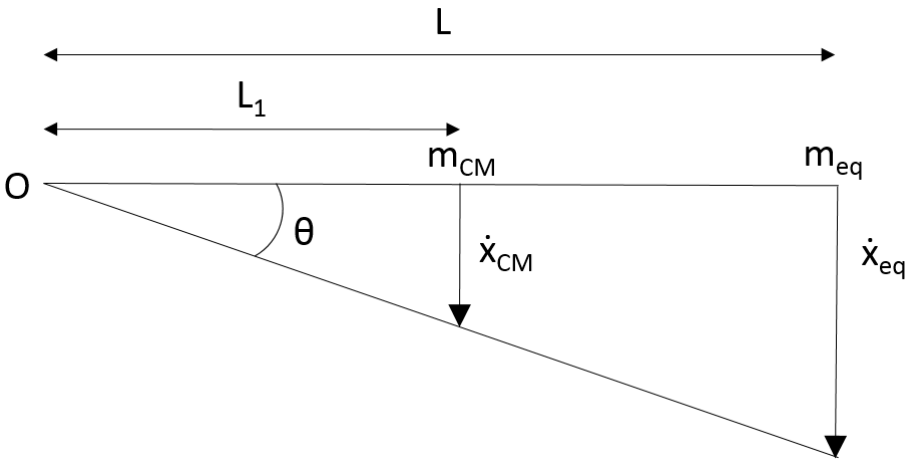


Figure 14: Displacement Relationships

From the similar triangles property, the following relation determined.

$$\frac{L_1}{L} = \frac{\dot{x_{CM}}}{\dot{x}_{eq}} \quad \text{(eq 6)}$$

Equation 7 is used to describe the kinetic energy equivalence.

$$\frac{1}{2} m_{eq} \dot{x}_{eq}{}^2 = \frac{1}{2} m_{CM} \dot{x}_{CM}{}^2 \quad \text{(eq 7)}$$

Where m_{CM} is the mass of the cab acting at the center of mass and m_{eq} is the equivalent mass acting on the springs at the rear mounts. Equation 7 can be simplified as shown in equation 8.

$$m_{eq} \dot{x}_{eq}{}^2 = m_{CM} \dot{x}_{CM}{}^2 \quad \text{(eq 8)}$$

Using the similar triangles property yields equation 9.

$$\frac{m_{eq}}{m_{CM}} = \left(\frac{L_1}{L}\right)^2 \quad \text{(eq 9)}$$

The equivalent mass can be calculated using Equation 10.

$$m_{eq} = (\frac{L_1}{L})^2 . m_{CM} \quad \text{(eq 10)}$$

Now, considering the vibration data provided by the client, the desired isolation region can be identified and the corresponding natural frequency can be calculated using equation 1.

$$f_n = \frac{f_o}{\sqrt{2}}$$

From equation 2, the desired spring constant can be calculated.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m_{eq}}}$$

That is, the required spring constant to get the natural frequency to a level to isolate the desired offending frequencies can be calculated using equation 2, above.

3.4.3 Pitch Frequency Method

In order to perform the pitch frequency analysis, the assumptions regarding the front mounts and center of mass outlined in section 3.4.2 were employed. The pitch frequency analysis considers the cab's angular motion about the front mounts. By referencing Figure 14, the summation of moments about the front mounts, point O, is zero as shown in equation 11.

$$\Sigma M = 0 \quad \text{(eq 11)}$$

Inserting the cab's angular forces into equation 11 yields equation 12.

$$I\theta'' + kx_{eq}L = 0 \quad \text{(eq 12)}$$

Where I is the moment of inertia of the cab and θ'' is the angular acceleration about point O.

The moment of inertia about point O is calculated using equation 13.

$$I = mL_1{}^2 \quad \text{(eq 13)}$$

Equation 13 is the moment of inertia equation for a solid rod about point O. This approximation was deemed appropriate for the cab because the center of mass is assumed to be located on the floor due to the small angle approximation as described above.

From Figure 14, equation 14 was generated.

$$tan\theta = \frac{x_{eq}}{L} \quad \text{(eq 14)}$$

Applying the small angle approximation, equation 14 reduces to equation 15.

$$\theta = \frac{x_{eq}}{L} \quad \text{(eq 15)}$$

And

$$\theta'' = \frac{x_{eq}''}{L} \quad \text{(eq 16)}$$

Using equations 13, 15 and 16, equation 12 can be written as equation 17.

$$mL_1{}^2 \frac{x_{eq}''}{L} + kx_{eq}L = 0 \quad \text{(eq 17)}$$

The natural frequency of the cab can then be calculated using equation 1 as shown in equation 18.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{\frac{mL_1{}^2}{L^2}}} \quad \text{(eq 18)}$$

Now, it can be seen that equation 18 can be obtained by combining equation 2 and equation 10 from the kinetic energy method analysis. The kinetic energy method and the pitch frequency method yield the same result.

4.0 Technical Analysis

Using the methods and assumptions outlined in section 3.0, the analysis was conducted to analyze the current system and determine the required characteristics of the proposed suspension system. The Vibration Data section outlines the information provided by the client, and the following sections outline the analysis the team performed on the design.

4.1 Vibration Data

The client has supplied the team with vibration data in the form of Fast Fourier Transform (FFT) plots, showing the vibration amplitude and corresponding frequencies at several locations on the tractor. The relevant FFT plot describes the vibration in the vertical direction on the floor of the cab, as shown in Figure 15.

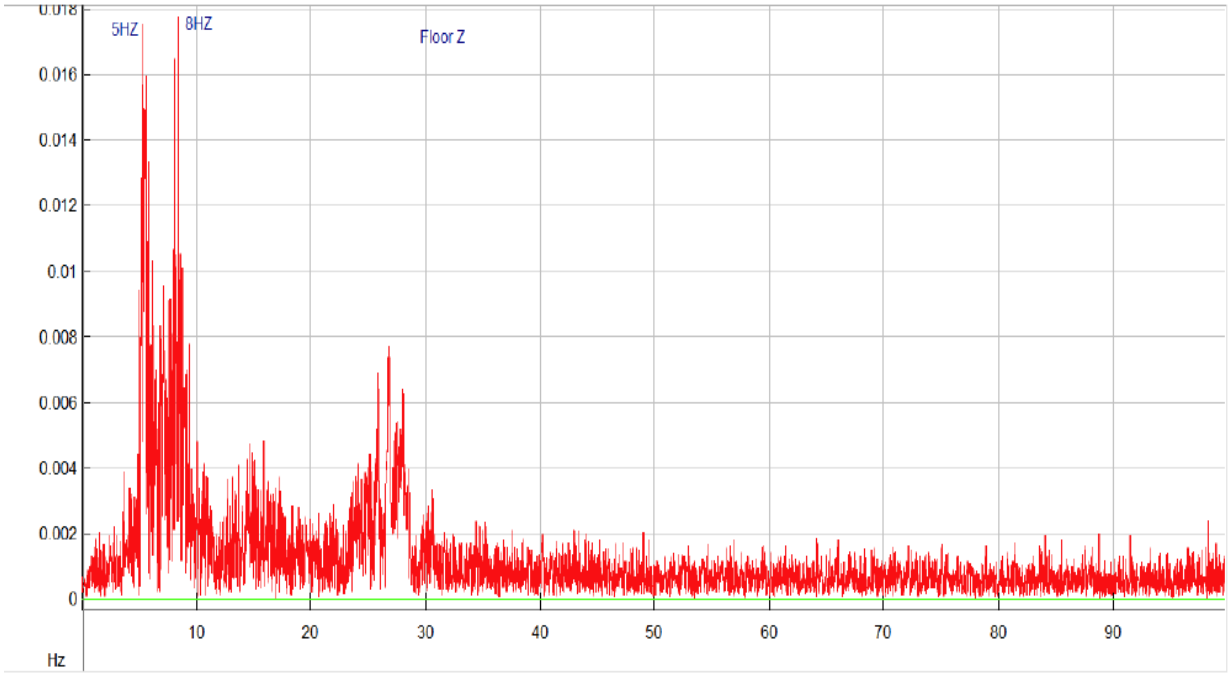


Figure 15: FFT Plot for Cab Floor - Vertical Direction

Based on this FFT plot, it can be seen that the cab experiences spikes in the amplitude of vibrations at frequencies of 5, 8 and 27 Hz.

The general approach to isolate the cab from the offending vibrations is to isolate the cab from the lowest frequency, as this will also isolate the cab from the higher order offending

frequencies. Considering that the lowest offending frequency is 5 Hz, the target natural frequency for the cab is calculated using equation 1.

$$f_n = \frac{f_o}{\sqrt{2}}$$

$$f_n = \frac{5}{\sqrt{2}} = 3.5\ Hz$$

That is, the goal of the suspension system is to bring the natural frequency of the cab to 3.5 Hz.

4.2 Analysis of Current System

In order to gain an in depth understanding of how vibration is transmitted throughout the tractor, the current cab mounting system was analyzed. The current mounting system includes two rubber pads that are mounted at each corner beneath the cab as illustrated in Figure 16.



Figure 16: Current Cab Mounts

The lower rubber pad is mounted between the black cab bracket and the red frame bracket, and the second pad is mounted above the cab bracket and below the washer and the end of the bolt. The rubber damping material is provided by LORD, which is an external supplier. Relevant material characteristics of the rubber pads are shown in TABLE XIII.

TABLE XIII: CHARACTERISTICS OF RUBBER DAMPERS

Part number	Commercial Name	Chemical Type	Maximum Axial Load at Specified Deflection	
			lbs @ in	N @ mm
CB-2204-5	Natural rubber	Natural polyisoprene	1000 at 0.09	4450 at 2.3
CB-2205-9	Natural rubber	Natural polyisoprene	2825 at 0.09	12566 at 2.3

The stiffer rubber pad, part number CB-2205-9, is the lower pad mounted between the cab and frame brackets and is responsible for providing the spring behavior to the cab. Its spring constant is calculated in equation 19.

$$k = \frac{12566\ N}{0.0023\ m} = 5463478\ \frac{N}{m} \quad \text{(eq 19)}$$

The natural frequency of the cab can then be calculated using equation 2. The mass of the cab is 1160kg. The client advised the team that the operator mass is generally assumed to be 75kg, resulting in a total weight of 1235 kg for the cab and operator.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m_{eq}}} = \frac{1}{2\pi}\sqrt{\frac{5643478}{(1235/4)}} = 21\ Hz$$

The equivalent mass is equal to the total mass of the cab is divided by 4 because the rubber pads at all 4 corners share the weight of the cab and operator. It follows that, from equation 1, the isolation region can be determined

$$f_o = \sqrt{2} f_n = (\sqrt{2})(21) = 30Hz \quad \text{(eq 21)}$$

That is, all vibrations above 30 Hz should be isolated. By referencing the FFT plot in Figure 15, it can be seen that this precisely describes the behavior of the current system.

4.3 Analysis of Desired Suspension Characteristics

Using the techniques outlined above, the desired characteristics of the suspension system may be determined. The client requested that the suspension system be integrated into the rear cab mounts while keeping the current front mounts unchanged. To simplify the behavior of the cab in this configuration, it is assumed that the cab hinges about the front mounts.

4.3.1 Spring Characteristics

The desired spring characteristics were calculated using the kinetic energy method and the pitch frequency method.

4.3.1.1 Kinetic Energy Method

As outlined above, the kinetic energy method essentially calculates the equivalent mass of the cab acting on the suspension components situated at the rear cab mounts. The equivalent mass is dependent on the location of the center of mass and is calculated using equation 11.

$$m_{eq} = (\frac{L_1}{L})^2 . m_{CM}$$

The mass of the cab and operator is 1235 kg and the center of mass is located 55% of the way to the rear of the cab from the front.

$$m_{eq} = (0.55)^2 . (1160 + 75) = 373\ kg$$

However, 373kg is the equivalent mass of the cab acting on the two rear mounts. For the equivalent mass acting on each rear mount, the value is divided by two,

$$m_{eq} = \frac{373}{2} = 186.5\ kg$$

186.5 kg is the equivalent mass acting on each rear mount. Since the lowest offending frequency is 5 Hz, the cab must have a natural frequency of $\frac{5}{\sqrt{2}} = 3.5$ Hz in order to isolate the cab from all offending frequencies.

Now, from equation 2, the desired spring constant can be calculated.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m_{eq}}}$$

$$\frac{5}{\sqrt{2}} = \frac{1}{2\pi}\sqrt{\frac{k}{186.5}} = 92034\ \frac{N}{m}$$

That is, the required spring constant to get the natural frequency to 3.5 Hz, to isolate the 5 Hz offending frequency, is 92034 N/m.

4.3.1.2 Pitch Frequency Method

An alternate analysis was conducted by considering the cab's angular motion about the front mounts. The cab's moment about the front mounts is shown in equation 17.

$$m{L_1}^2\frac{x_{eq}''}{L} + kx_{eq}L = 0$$

The natural frequency of the cab can then be calculated as shown in equation 18.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{\frac{m{L_1}^2}{L^2}}}$$

$$\frac{5}{\sqrt{2}} = \frac{1}{2\pi}\sqrt{\frac{k}{(\frac{1235}{2})(0.55)^2}}$$

$$k = 92034\ \frac{N}{m}$$

The kinetic energy method and the pitch frequency method yield the same result. The required spring constant to isolate the cab from the 5 Hz offending frequency is 92034 N/m. By isolating the 5 Hz frequency, the higher frequency offending vibrations are also isolated.

4.3.2 Damping Characteristics

As described in section 3.2, the suspension system must include damping to return the system to equilibrium after excitation. Using equation 4, the critical damping constant can be calculated.

$$c_c = 2\sqrt{km_{eq}} = 2\sqrt{(92034)(186.5)} = 8286\ \frac{N.s}{m}$$

It was decided that the desired damping ratio to have the best compromise of damping and transmissibility is $\zeta = 0.5$. Using equation 3, the desired damping characteristics of the suspension system can be calculated.

$$\zeta = \frac{c}{c_c}$$

$$0.5 = \frac{c}{8286}$$

$$c = 4143\ \frac{N.s}{m}$$

The damping components of the suspension system must have a damping constant of 4143 Ns/m to keep the transmissibility at the resonant peak to roughly 1.5 and to exhibit good isolation characteristics in the isolation region.

5.0 Design Details

In order to satisfy the client's criteria, this cab suspension system is designed to reduce vibration transmitted to the operator and pass the ROPS standard of 64000 lbs. The system consists of shock absorbers and metal coil springs mounted in parallel. This is an effective design because it is marketable and it satisfies the technical analysis. The tractor with the suspension system installed is shown in Figure 17.

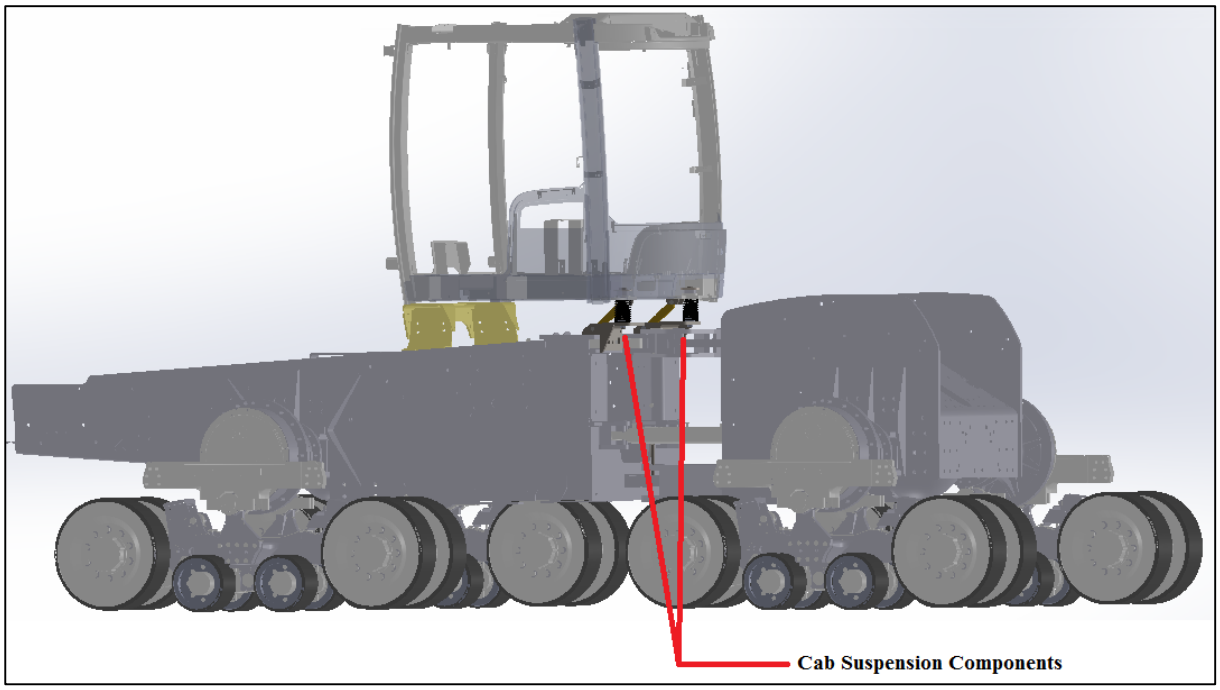


Figure 17: Tractor with Suspension System Installed.

As shown, the suspension system is active only on the rear cab mounts. This was requested by the client to simplify the system. Additionally, with only two mounts with active suspension components, the need for sway bars is eliminated.

Figure 18 shows the suspension system in more detail.

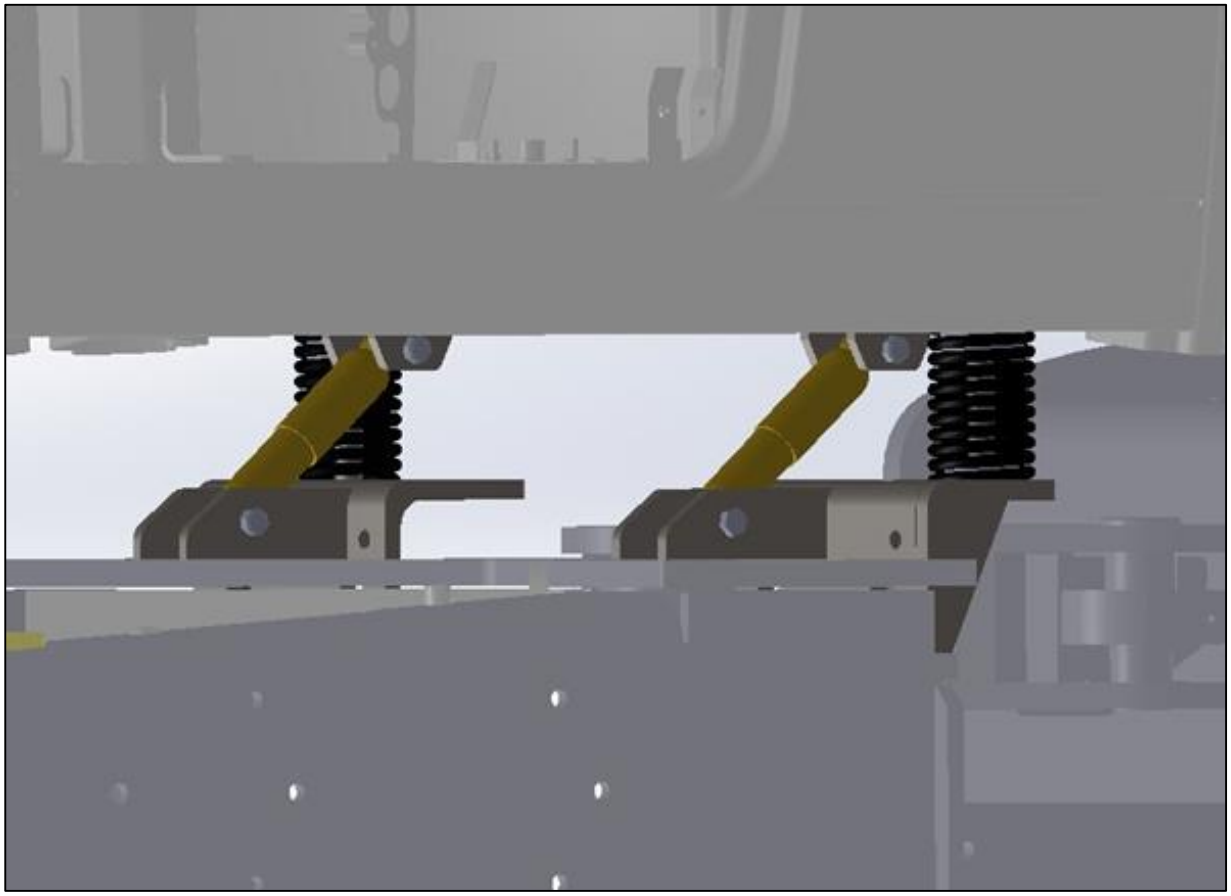


Figure 18: Overall Cab Suspension System

The system consists of redesigned mounting brackets, shock absorbers and metal coil springs. The function of each of these components and their contribution to the system as a whole are explained below.

5.1 Brackets

The current cab mounting brackets are designed to mount the cab rigid to the frame with a small rubber damper between the cab and frame brackets as shown in Figure 16, above. This mounting system does not accommodate suspension components due to space constraints. As such, the mounting brackets were completely redesigned to allow for the suspension components to be mounted between the cab and the frame, as shown in Figure 19.

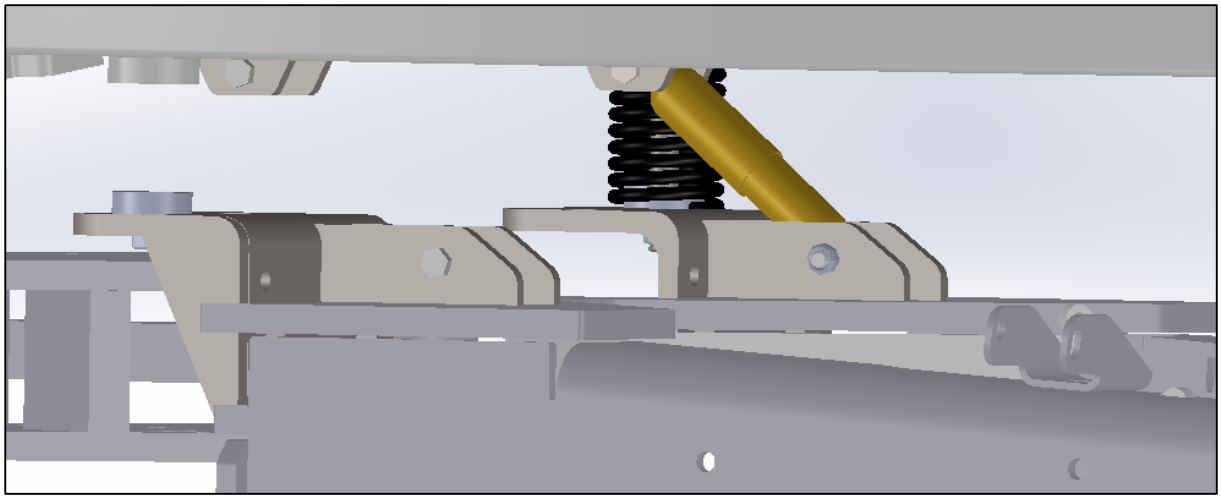


Figure 19: Redesigned Mounting Brackets

The brackets were redesigned to sit lower to the frame, creating space beneath the cab to mount the suspension system. The bracket on the left in Figure 19, displays the space created by the new bracket. The bracket on the right demonstrates how the components are mounted to the bracket. Gussets are welded in the bend of the bracket to provide strength and rigidity. FEA was performed on the brackets and the results are shown in Appendix F. The maximum deflection of the plate under the weight of the cab is 0.09 mm.

Shock mounts were incorporated into the brackets in order to provide solid mounting points for the shock absorbers. Spring spacers were also included to keep the spring in position during operation as shown in Figure 20.

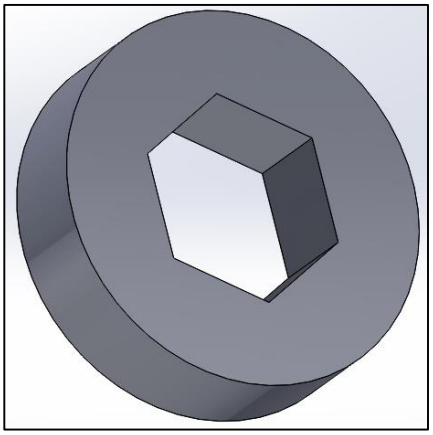


Figure 20: Spring Spacer with Hexagonal Hole to Hold Nut.

The spacer is designed to fit inside the spring, keeping it from moving laterally during operation. This feature is shown in Figure 21.

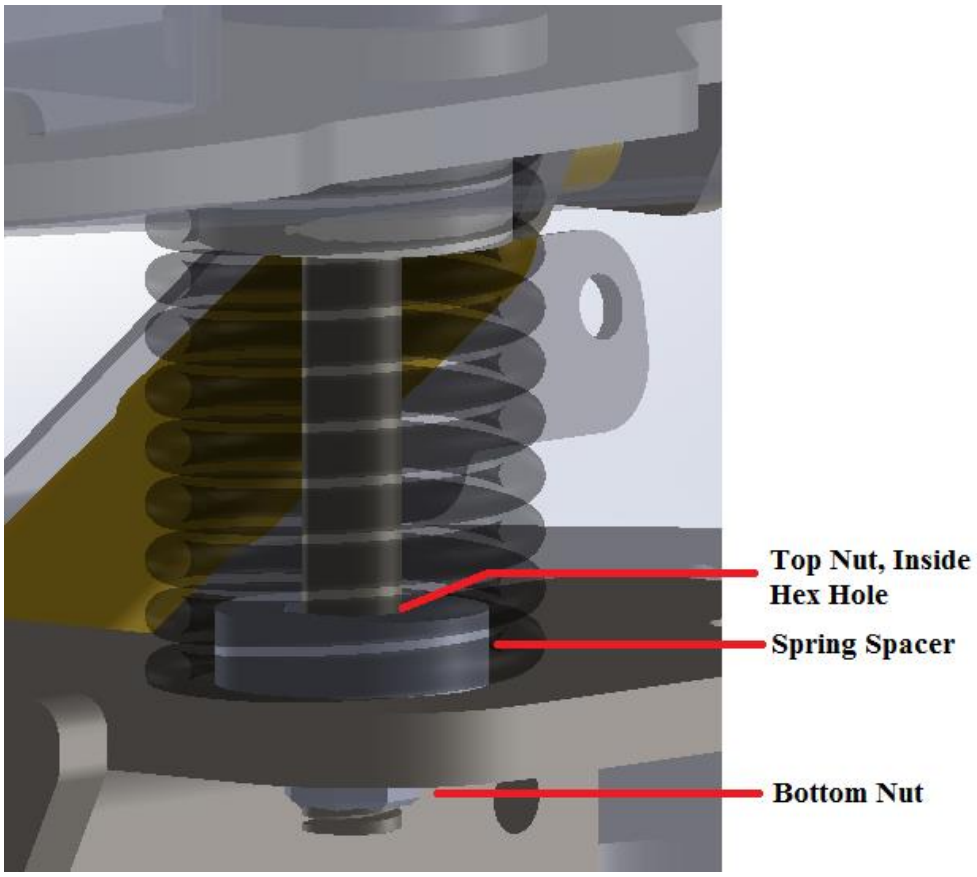


Figure 21: System to Hold Bolt Stationary at Bottom Mounting Bracket.

The hexagonal hole cut into the spring spacer is another design feature to keep the mounting bolt stationary relative to the bottom bracket. The hex hole is used to seat a nut to fix the bolt to the bracket from the top. When combined with the nut beneath the bracket, the bolt is kept stationary relative to the bracket. This keeps the bolt from interfering with the articulated steering system when the suspension is compressed.

5.2 Springs

The most effective method of vibration reduction is isolation. The current mounting system consists of extremely stiff rubber pads and results in an elevated natural frequency of 21 Hz as shown in the Technical Analysis section. In order to reduce the natural frequency, a softer suspension is needed. It was determined that a spring constant of 92034 N/m is needed to

reduce the natural frequency to 3.5 Hz, thereby isolating the offending vibration at 5 Hz and any higher frequency vibrations.

The natural frequency of the system is controlled by the spring constant of the suspension. The suspension is designed with a 180 mm spring of spring constant 89343 N/m as shown in Figure 22.

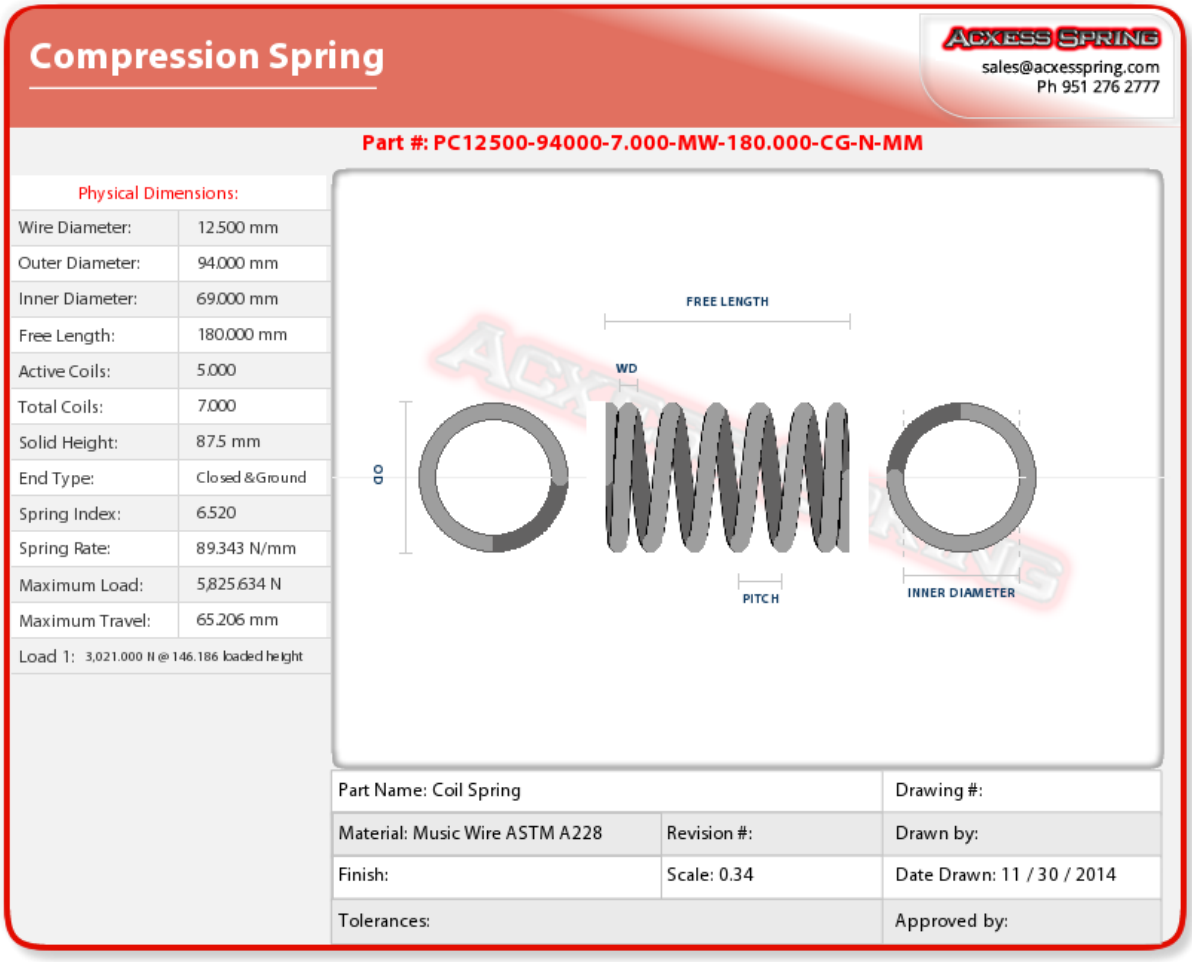


Figure 22: Custom Spring Design [8]

This spring is sufficient in bringing the natural frequency below 3.5 Hz, and will isolate all of the offending frequencies above 5 Hz. The spring will sag 34 mm from the weight of the cab itself but due to the length of the spring, there is enough spring travel remaining to sufficiently support the cab.

5.3 Shock Absorbers

The second method of vibration reduction is damping. Every piece of mechanical equipment needs some form of damping to dissipate vibration energy. For this application, shock absorbers were selected to provide damping to the system.

From the technical analysis, it was determined that a damping constant of 4143 Ns/m was ideal for this system. In consulting with shock absorber manufacturers, it was determined that this corresponds to a medium duty cab suspension shock absorber, shown in Figure 23.



Figure 23: Monroe 66101 Cab Suspension Shock

Monroe, the shock absorber manufacturer, has the best selection of cab suspension shock absorbers. However, they refuse to release the damping characteristics of their products. They have advised that the above shock absorber, part number 66101, is suited to our application.

5.4 ROPS Compliance

The proposed shocks have a 15000 lb (6818 kg) tensile strength. Combined with the strength of the front mounts, they would pass the 64000 lb ROPS criteria in tension. However, this does not account for the shear strength of the rear suspension. This is resolved by mounting the springs with 1" diameter grade 8 bolts, having 141370 lbs of double shear strength, detailed in Figure 24.

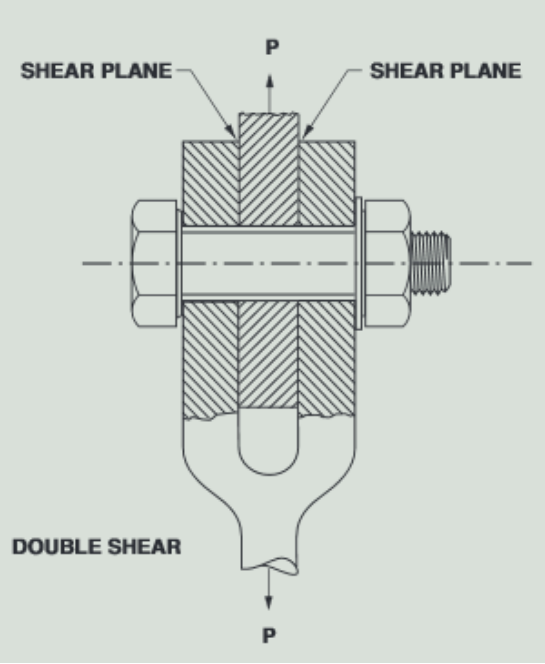


Figure 24: Diagram of bolt in double shear

The bolts are positioned concentrically within the spring as shown in Figure 25 on the following page.

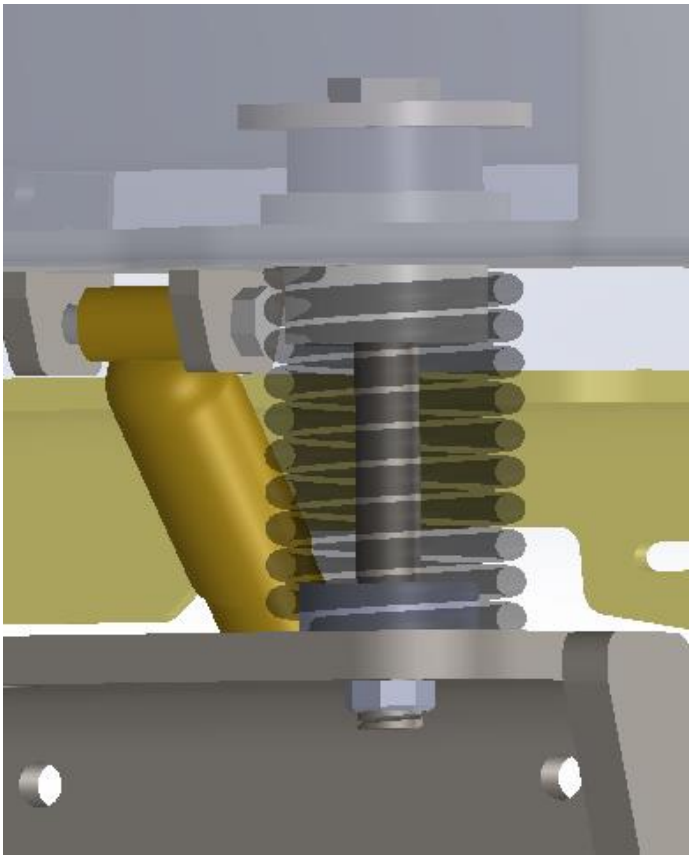


Figure 25: Mounting Bolts

Additionally, 5/8'' Grade 8 bolts are used to mount the shock absorbers to the brackets, which have 55122 lbs. double shear strength. Combined with the original front mounts, the cab mounting system satisfies the ROPS criteria of 64000 lbs.

5.5 Overall Design and Functionality

The suspension system as whole is designed to meet the three main criteria set by the client: marketable, ROPS compliant and functional. Each component is designed to specifically address these three needs.

5.5.1 Operation and Functionality

The suspension system is designed to oscillate to absorb the vibrations that are transmitted to the cab so that they are not transmitted to the operator. The spring is used to suspend the cab and control its natural frequency. The spring characteristics were specifically determined to

isolate the offending vibration at the lowest frequency peak of 5 Hz and all the vibrations of higher frequencies.

The shock absorbers are used to damp vibrations and prevent the cab from undesired oscillation in the event of an impulse. An impulse could be due to hitting a large bump or hole. Shock absorbers also provide overall damping to the cab when travelling on uneven terrain as would be encountered in a field.

The long bolts mounted through the spring, shown in Figure 25, are strictly for safety purposes to ensure that the tractor passes the ROPS criteria. The bolts do not provide any suspension characteristics. As such, the bolt is mounted in a way that does not interfere or influence the suspension behavior when the system is in normal operation. However, in the event of a vehicle rollover, the bolt will become loaded in shear or tension, depending on the scenario, and provide sufficient mounting strength to ensure that the cab remains fixed to the tractor.

To keep the bolt from interfering with the suspension during normal operation, the bolt is fixed rigid to the bottom cab mounting bracket. As the suspension compresses, the bolt remains stationary relative to the bottom bracket so it does not move downward and interfere with the articulated steering system. The bolt is free-floating in the vertical direction relative to the top mounting bracket as there is room above the bracket. As the suspension compresses, the top of the bolt moves vertically toward the cab. This motion has no influence on the system's behavior as the bolt does not interfere with anything beneath the cab.

5.5.2 Overall Characteristics

The overall suspension performance characteristics are outlined in this section. System transmissibility and a theoretical vibration reduction will be presented.

5.5.2.1 System Transmissibility

By altering the spring and damping characteristics of the cab mounting system, the transmissibility was also changed. The actual transmissibility of the proposed suspension system is shown in Figure 26.

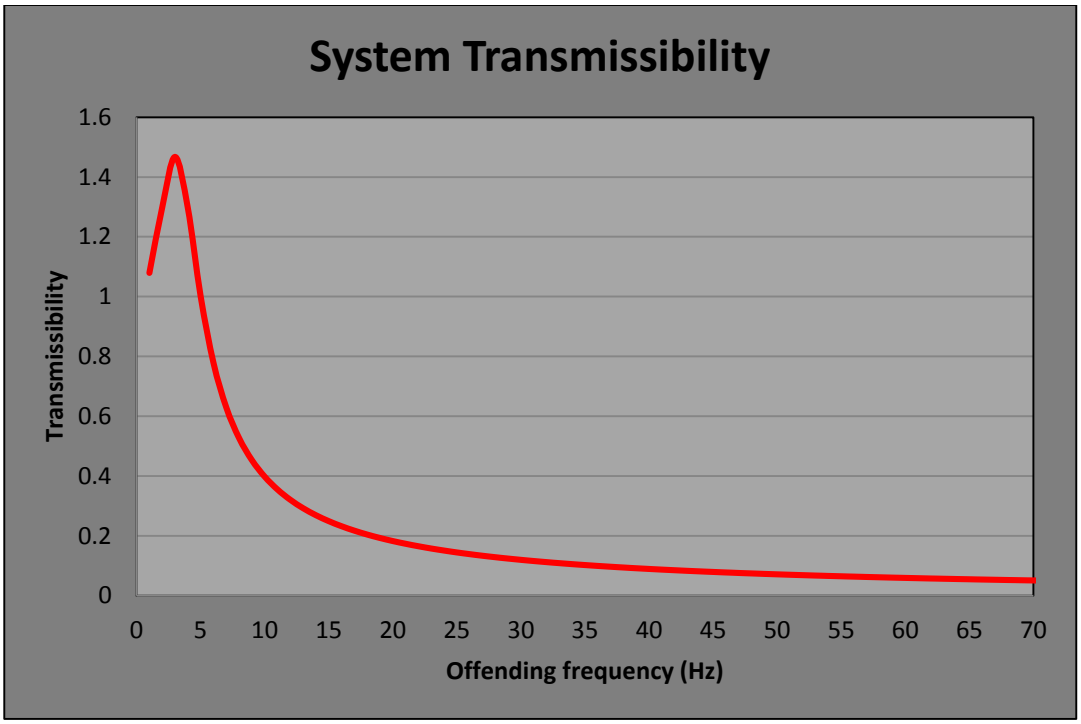


Figure 26: System Transmissibility

The suspension is designed to isolate vibrations above 5 Hz. As shown above, the transmissibility at 5 Hz is 1.0 and decreases as the frequency increases. This signifies that the amplitude of the cab's response to offending vibrations at frequencies above 5 Hz is smaller than the amplitude of the offending vibration. As such, the offending frequencies above 5 Hz are theoretically isolated.

5.5.2.2 Theoretical Vibration Reduction

In order to estimate the overall vibration reduction, the transmissibility of the suspension system was compared to that of the current system. The maximum transmissibility occurs at a frequency ratio of 1.0, where the cab resonates as the offending frequency and the cab natural frequency are equal.

By referencing Figure 26, it can be seen that the maximum transmissibility for the new suspension system is 1.5. Similarly, by referencing Figure 11, the maximum transmissibility for the current system is 7, since the damping ratio of the rubber mounting pads is roughly 0.05 [9].

Now, the overall reduction can be calculated using equation 20.

$$\%\ Reduction = \frac{T_C - T_S}{T_C} \quad \text{(eq 20)}$$

Where T_C is the transmissibility of the current system and T_S is the transmissibility of the suspension system. The actual percent reduction in transmissibility is calculated below.

$$\%\ Reduction = \frac{7 - 1.5}{7} = 79\%$$

That is, the maximum response to an input excitation is reduced by 79% for the proposed suspension system. Additionally, the suspension system is designed to isolate the peak offending frequencies at 5, 8 and 27 Hz. The amplitudes of those frequencies are theoretically reduced to the point that they are negligible. However, the overall vibration reduction of the system is unknown and the system will have to be prototyped to determine the exact overall vibration reduction.

5.5.3 Manufacturing

This section outlines the approach to be taken to manufacture the parts designed by the team. All necessary engineering drawings are provided in Appendix E. Most parts are made from A36 steel, and should be laser cut from a plate of appropriate thicknesses.

5.5.3.1 Tolerances

The suspension system was engineered to the client's internal specifications for tolerances and clearances. In consulting with the client, it was decided that the tolerance for hole diameters in the brackets is 0.5 mm.

5.5.3.2 Clearances

The brackets were redesigned to be lower to the cab and closer to the articulated steering system. The client requested a minimum clearance of 25 mm between the brackets and articulated steering system.

5.5.3.3 Welds

The brackets are fabricated from a series of smaller components welded together. The client informed the team that they typically use fillet welds no less than 4 mm for most applications and would prefer that those welds be used. Upon further analysis, 6 mm fillet welds are recommended for 12 mm plate and 8 mm fillet welds are recommended for 19 mm plate. The engineering drawings include all relevant welding information.

5.5.3.4 Bill of Materials

The bill of materials for the cab suspension system is shown in TABLE XIV.

TABLE XIV: BILL OF MATERIALS

Item No.	Supplier	Part number	Description	Quantity	Source
1.	Monroe	66101	Shock absorber	2	[10]
2.	Axcess Spring	PC 12500 94000-7.000-MW-180.000-CG-N-MM	Compression spring	2	[11]
3.	Designed by team	Part 01	Lower shock mounts	4	Designed by team
4.	Designed by team	Part 02	Mount	2	Designed by team
5.	Designed by team	Part 03	Upper shock mount	4	Designed by team
6.	Designed by team	Part 04	Spring spacer	2	Designed by team
7.	BoltDepot	2591	Hex nuts, grade 8 steel, 1"-8	2	[12]

TABLE XIV: BILL OF MATERIALS (continued)

Item No.	Supplier	Part number	Description	Quantity	Source
8.	Designed by team	Part 05	Upper spring retainer	2	Designed by team
9.	Designed by team	Part 06	Mount support	2	Designed by team
10.	BoltDepot	2588	Hex nuts, grade 8 steel, 5/8”-11	4	[13]
11.	BoltDepot	7209	Hex bolts, grade 8 steel, 1"-8 x 10"	2	[14]
12.	BoltDepot	675	Hex bolts, grade 8 steel, 5/8"-11 x 4-1/2"	4	[15]
13.	Fastenal	1103762	5/8" 18-8 stainless steel thick flat washer	16	[16]
14.	Versatile	86043576	Washer 1.06” IDX5.25”ODX.375”	2	N/A
15.	Versatile	86043028	Isolator – Cab 1” Bolt	2	N/A
16.	Versatile	86044340	Nut Hex Lock 1-8UNC GR8 Pl	2	N/A

Quantities are defined for one tractor suspension system, and references are provided for external suppliers. Parts that were designed by the team are to be manufactured in house.

5.5.3.5 Cost

The cost analysis includes the cost of purchasing all the components needed for the suspension system. The labour cost associated with manufacturing and assembly is not considered in this analysis. Moreover, the cost of the components to be manufactured in house is not considered. Since the manufactured parts are directly replacing current production components, the net cost for the parts is assumed to be zero. TABLE XV below shows the cost break-down of the purchased suspension system components.

TABLE XV: ITEM COST

Item	Supplier	Part number	Description	Qty.	Cost($/unit)	Total($)
1.	Monroe	66101	Shock absorbers	2	44.43 [17]	88.86
2.	Axcess Spring	PC 12500 94000-7.000-MW-180.000-CG-N-MM	Compression springs	2	99.6 [8]	199.2
3.	Designed by team	Part 01	Lower shock mounts	4	*	*
4.	Designed by team	Part 02	Mount	2	*	*
5.	Designed by team	Part 03	Upper shock mount	4	*	*
6.	Designed by team	Part 04	Spring spacer	2	*	*
7.	BoltDepot	2591	Hex nuts, grade 8 steel, 1"-8	2	1.83	3.66
8.	Designed by team	Part 05	Upper spring retainer	2	*	*
9.	Designed by team	Part 06	Mount support	2	*	*
10.	BoltDepot	2588	Hex nuts, grade 8 steel, 5/8”	4	0.27	1.08
11.	BoltDepot	7209	Hex bolts, grade 8 steel, 1"-8 x 10"	2	19.4	38.8
12.	BoltDepot	675	Hex bolts, grade 8 steel, 5/8"-11 x 4.5"	4	1.8	7.2
13.	Fastenal	1103762	Washer .688”ID X 1.5”OD X.188”	16	4.94	79.04
14.	Versatile	86043576	Washer 1.06” IDX5.25”ODX.375”	2	*	*
15.	Versatile	86043028	Isolator – Cab 1” Bolt	2	*	*
16.	Versatile	86044340	Nut Hex Lock 1-8UNC GR8 Pl	2	*	*
					Total Cost($)	**417.84**

5.5.3.6 Assembly

The assembly of the suspension system will take place in a similar manner to how the current cab is installed on the tractor frame. However, there will need to be some slight modifications to the assembly process.

During cab manufacture, shock mounts and upper spring retainers need to be welded onto the existing assembly G18204 as displayed in Appendix E. The 10" bolt, bolt isolator and large washer (86043028, 86043576) are also recommended to be installed through their necessary holes in assembly G18204 in the cab at this time for ease of assembly onto the tractor frame. Alternatively, the shock mounts and spring retainers may be welded in house once the cab is received. The 10" spring mounting bolt, rubber pad and large washer may also be installed in house.

Once the cab is delivered, it can be hoisted onto the new frame mounts with the coil springs carefully placed between the mounts and the cab. The long bolts can then be threaded down through the nuts placed within the bottom spring retainers until the bolt heads are sitting flush with the upper washers and isolators. The bottom lock nuts can then be installed and torqued to act as a jam with the nuts in the spring retainers and keep the bolts in place. The shock absorbers can now be installed, again by referencing the assembly drawing.

6.0 Summary and Recommendations

Buhler Versatile has commissioned the design and analysis of a cab suspension system for their Versatile L4WD tractor. The manufacturer determined that a cab suspension system was necessary after they received complaints stating that vibrations were being transmitted to the operator when the tractor was operated on hard surfaces. Additionally, Buhler Versatile requested a cab suspension system to enhance the marketability of their product and compare more favorably to their competitors.

The client determined that the offending vibrations originated from the interaction of the tracks with the hard road surfaces while travelling at speeds above 20km/h. Because the cab is not isolated from the frame, certain offending frequencies were being transmitted to the cab, causing operator discomfort.

The client specified that the goal for this design is to have a marketable cab suspension system to reduce vibration by 25-50% over the current mounting system. Additionally, the design must satisfy the ROPS standard, ensuring operator safety in the event of a vehicle rollover.

The final design consists of a redesigned mounting bracket which accommodates a combination of a coil spring and shock absorber mounted at each rear corner of the cab. This design isolates the cab from the offending vibrations and provides damping to the cab in the event of a large excitation. It is estimated that the proposed cab suspension system will reduce vibrations by 79%, thereby improving operator comfort. Additionally, this design is comprised of components known to provide successful suspension characteristics, legitimizing the system in the eyes of customers and aiding its marketability.

CAD models have been created for each individual component and a complete system assembly. Additionally, the overall tractor model provided by the customer has been updated with the suspension system retrofitted into the existing model. Engineering drawings have been produced for the components to be fabricated in house and product requirements have been provided for the components to be purchased from outside suppliers.

The team recommends that this design be prototyped and tested to ensure adequate performance before being put into production. The analysis is heavily reliant on assumptions regarding information that was not available within the time available for the project, and is based on simplified methods to model the system's behavior.

When testing the prototype, the team recommends that special consideration be given to the behavior of the system under large excitations. According to these calculations, the spring will stack just before the shock bottoms out. If this is the case during operation, the system will still perform as designed, but spring life will be compromised.

Additionally, the system is designed with ease of manufacturability in mind, but it is not optimized. The components to be fabricated in house may need to be altered to aid in their manufacturability depending on the available equipment and plant layout.

7.0 References

[1] [Online]. Available: http://i.ytimg.com/vi/Ggu7XxTgdF4/maxresdefault.jpg. [Accessed 22 November 2014].

[2] Caterpillar, *Rollover Protection Structure*, Peoria: Caterpillar, 2010.

[3] U. Katu, R. Desavale and R. Kanai, "Effect Of Vehilce Vibration On Human Body - RIT Experience," Department of Mechanical Engineering, Rajarambapu Institute of Technology, Sakharale, 2010.

[4] H. S. Gitlow and D. M. Levine, Six Sigma for Green Belts, Upper Saddle River, NJ: Pearson Education Inc., 2005.

[5] Rogers Corporation, 2012. [Online]. Available: http://www.rogerscorp.com/documents/2303/hpf/shared/Vibration-Isolation-and-Damping-the-Basics-Color.pdf. [Accessed 5 November 2014].

[6] E. Cheever, 2014. [Online]. Available: http://lpsa.swarthmore.edu/Transient/TransInputs/TransStep/imgBC1.gif. [Accessed 20 November 2014].

[7] Farrat Engineering Excellence, 2014. [Online]. Available: http://www.farrat.com/images/theory05.gif. [Accessed 17 November 2014].

[8] Acxess Spring, "Custom Springs," The Spring Store, 2014. [Online]. Available: http://www.thespringstore.com/pc531-3656-9-40-ot-8-000-cg-n-in.html. [Accessed 28 November 2014].

[9] Farrat Engineering Excellence, 2012. [Online]. Available: http://www.farrat.com/detailed-vibration-isolation-theory-68.html. [Accessed 17 November 2014].

[10] Tenneco Inc., "MONROE Shocks and Struts," MONROE, 2013. [Online]. Available: http://www.monroe.com/en-US/catalog/e-Catalog/66101. [Accessed 24 November 2014].

[11] Acxess Spring, "The Spring Store," Acxess Spring, 2014. [Online]. Available: http://www.thespringstore.com/pc531-3656-9-40-ot-8-000-cg-n-in.html. [Accessed November 2014].

[12] Bolt Depot Inc., "Product Catalogue," Bolt Depot Inc., 2014. [Online]. Available: https://www.boltdepot.com/Product-Details.aspx?product=2591. [Accessed 26 November 2014].

[13] Bolt Depot Inc., "Product Catalogue," Bolt Depot Inc., 2014. [Online]. Available: https://www.boltdepot.com/Product-Details.aspx?product=2588. [Accessed 26 November 2014].

[14] Bolt Depot Inc., "Product Catalogue," Bolt Depot Inc., 2014. [Online]. Available: https://www.boltdepot.com/Product-Details.aspx?product=7209. [Accessed 26 November 2014].

[15] Bolt Depot Inc., "Product Catalogue," Bolt Depot Inc., 2014. [Online]. Available: https://www.boltdepot.com/Product-Details.aspx?product=675. [Accessed 26 November 2014].

[16] Fastenal, "Products - Washers," Fastenal, 2014. [Online]. Available: https://www.fastenal.com/web/products/details/11103762. [Accessed 28 November 2014].

[17] Amazon, "Vehicle Parts," Monroe, 2014. [Online]. Available: http://www.amazon.com/Monroe-66101-Gas-Magnum-Suspension-Absorber/dp/B000JWK6EO. [Accessed 28 November 2014].

Appendix A – Project Scheduling

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Introduction

The following sections outline our Work Breakdown Structure as well as our Project Schedule in the form of a Gantt chart. These tools were used to develop a plan of how the team was going to delegate and determine the deadlines for each task.

Our team generated a Work Breakdown Structure divided into four separate phases. Each phase displays the necessary tasks that must be completed for the project to be a success. The Work Breakdown Structure is shown in Figure 1, on the following page, followed by our project schedule which is in the form of a Gantt Chart displayed in Figure 2.

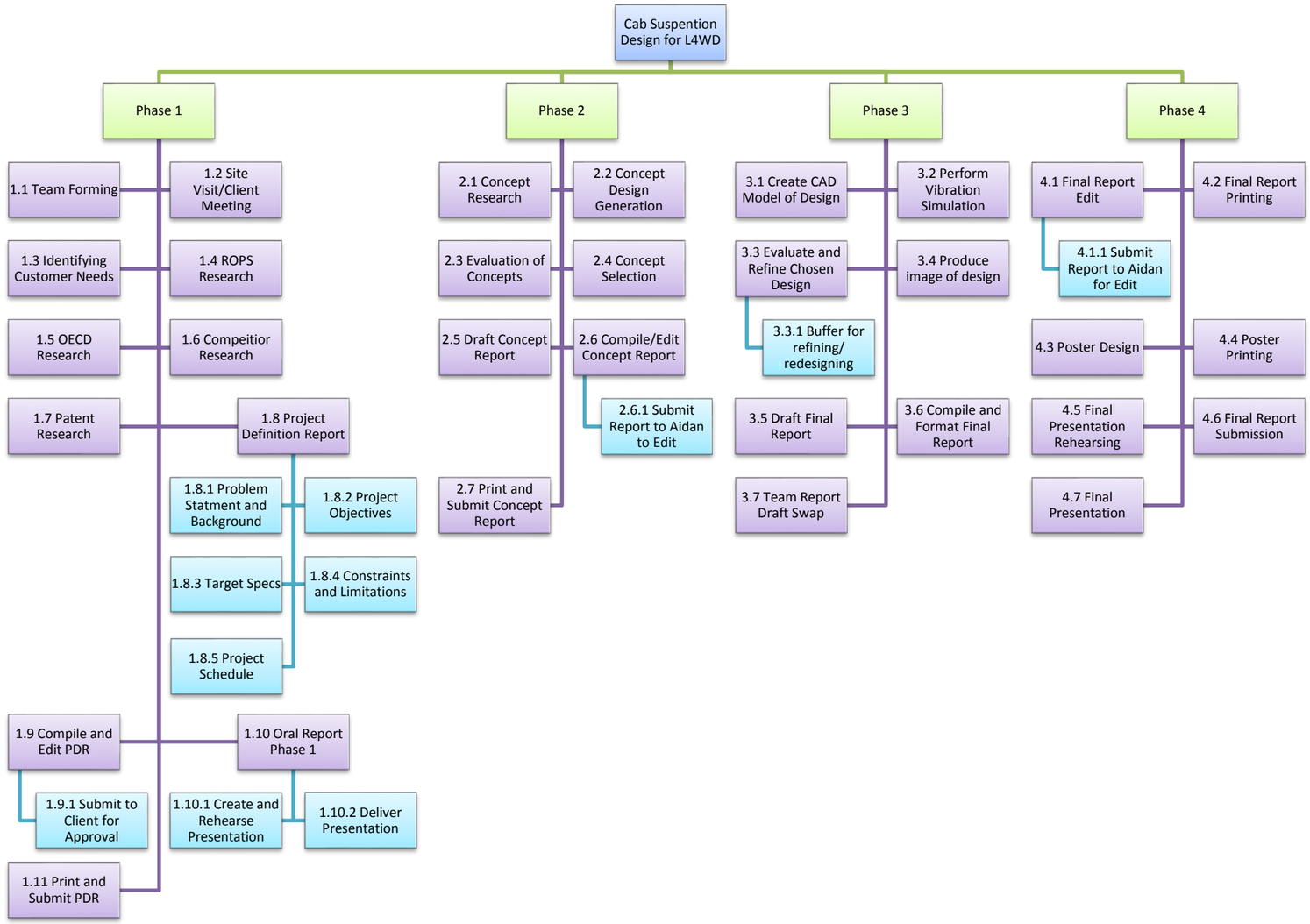


Figure 1: Work Breakdown Structure

Versatile L4WD Cab Suspension Design Project Schedule

Versatteam Engineering

Project Lead: Andrew Grossart
Project Start Date: 9/10/2014 (Wednesday)
Today's Date: 11/30/2014 (Sunday)

Daily ☐ Show Weekends

WBS	Task	Members Resonsible	Start	End	Work Days	% Done
1	**Phase 1**	[Initials]	Wed 9/10/14	Thu 10/02/14	17	100%
1.1	Team Forming	AG BM AH PS DP	Wed 9/10/14	Wed 9/10/14	1	100%
1.2	Site Visit/Client Meeting	AG BM AH PS DP	Thu 9/11/14	Thu 9/11/14	1	100%
1.3	Idenitfying Customer Needs	AG BM AH PS DP	Thu 9/11/14	Wed 9/17/14	5	100%
1.4	ROPS Research	AH	Tue 9/16/14	Wed 9/24/14	7	100%
1.5	OECD-Code 2 Research	PS	Tue 9/16/14	Wed 9/24/14	7	100%
1.6	Competitor Design Research	AG BM DP	Tue 9/16/14	Wed 9/24/14	7	100%
1.7	Competitor Patent Research	AG BM DP	Tue 9/16/14	Wed 9/24/14	7	100%
1.8	Project Definition Report	AG BM AH PS DP	Tue 9/16/14	Fri 9/26/14	9	100%
1.8.1	Problem Background/Statement	DP	Wed 9/17/14	Wed 9/24/14	6	100%
1.8.2	Project Objectives	BM	Wed 9/17/14	Wed 9/24/14	6	100%
1.8.3	Target Specs	PS	Wed 9/17/14	Wed 9/24/14	6	100%
1.8.4	Constraints and Limitations	AH	Wed 9/17/14	Wed 9/24/14	6	100%
1.8.5	Project Schedule (WBS/Gannt)	AG	Wed 9/17/14	Wed 9/24/14	6	100%
1.9	Compile and Edit PDR	AG BM AH PS DP	Wed 9/24/14	Thu 9/25/14	2	100%
1.9.1	Submit to Client for approval	AG	Fri 9/26/14	Fri 9/26/14	1	100%
1.10	Oral Report Phase 1		Fri 9/26/14	Mon 9/29/14	2	100%
1.10.1	Create and Rehearse Presentaion	AG BM AH PS DP	Fri 9/26/14	Mon 9/29/14	2	100%
1.10.2	Deliver Presentation	AG BM AH PS DP	Thu 10/02/14	Thu 10/02/14	1	100%
1.11	Print and Submit PDR	AG	Mon 9/29/14	Mon 9/29/14	1	100%
2	**Phase 2**	[Initials]	Mon 9/29/14	Fri 10/24/14	19	100%
2.1	Concept Research		Mon 9/29/14	Fri 10/03/14	5	100%
2.2	Concept Design Generation		Sat 10/04/14	Fri 10/10/14	5	100%
2.3	Evaluation of Concepts		Fri 10/10/14	Fri 10/17/14	5	100%
2.4	Concept Selection		Fri 10/17/14	Fri 10/17/14	1	100%
2.5	Draft Concept Report		Fri 10/17/14	Mon 10/20/14	2	100%
2.6	Compile and Edit Concept Report		Mon 10/20/14	Thu 10/23/14	4	100%
2.6.1	Submit Draft to Aidan for Editing		Wed 10/22/14	Thu 10/23/14	2	100%
2.7	Print and Submit Concept Report		Fri 10/24/14	Fri 10/24/14	1	100%
3	**Phase 3**	[Initials]	Mon 10/20/14	Mon 11/24/14	25	100%
3.1	Create CAD Model of Design		Mon 10/20/14	Fri 10/24/14	5	100%
3.2	Perform Vibration Simulation		Thu 10/23/14	Wed 10/29/14	5	100%
3.3	Evaluate and Refine Chosen Design		Wed 10/29/14	Thu 11/06/14	7	100%
3.3.1	Buffer Time for refining/redesigning		Wed 10/29/14	Thu 11/06/14	7	100%
3.4	Produce Image of Final Design		Thu 11/06/14	Thu 11/06/14	1	100%
3.5	Draft Final Report		Thu 11/06/14	Thu 11/13/14	5	100%
3.6	Compile and Format Final Report		Fri 11/14/14	Mon 11/24/14	7	100%
3.7	Swap Report draft with other team		Mon 11/24/14	Mon 11/24/14	1	100%
4	**Phase 4**	[Initials]	Tue 11/25/14	Tue 12/02/14	5	87%
4.1	Final Report Editing		Tue 11/25/14	Fri 11/28/14	3	100%
4.2	Final Report Printing		Thu 11/27/14	Fri 11/28/14	2	100%
4.3	Poster Design		Tue 11/25/14	Mon 12/01/14	4	100%
4.4	Poster Printing		Mon 12/01/14	Mon 12/01/14	1	100%
4.5	Final Presentation Rehearsing		Thu 11/27/14	Mon 12/01/14	3	66%
4.6	Final Report Submission		Mon 12/01/14	Mon 12/01/14	1	100%
4.7	Final Presentation		Tue 12/02/14	Tue 12/02/14	1	0%

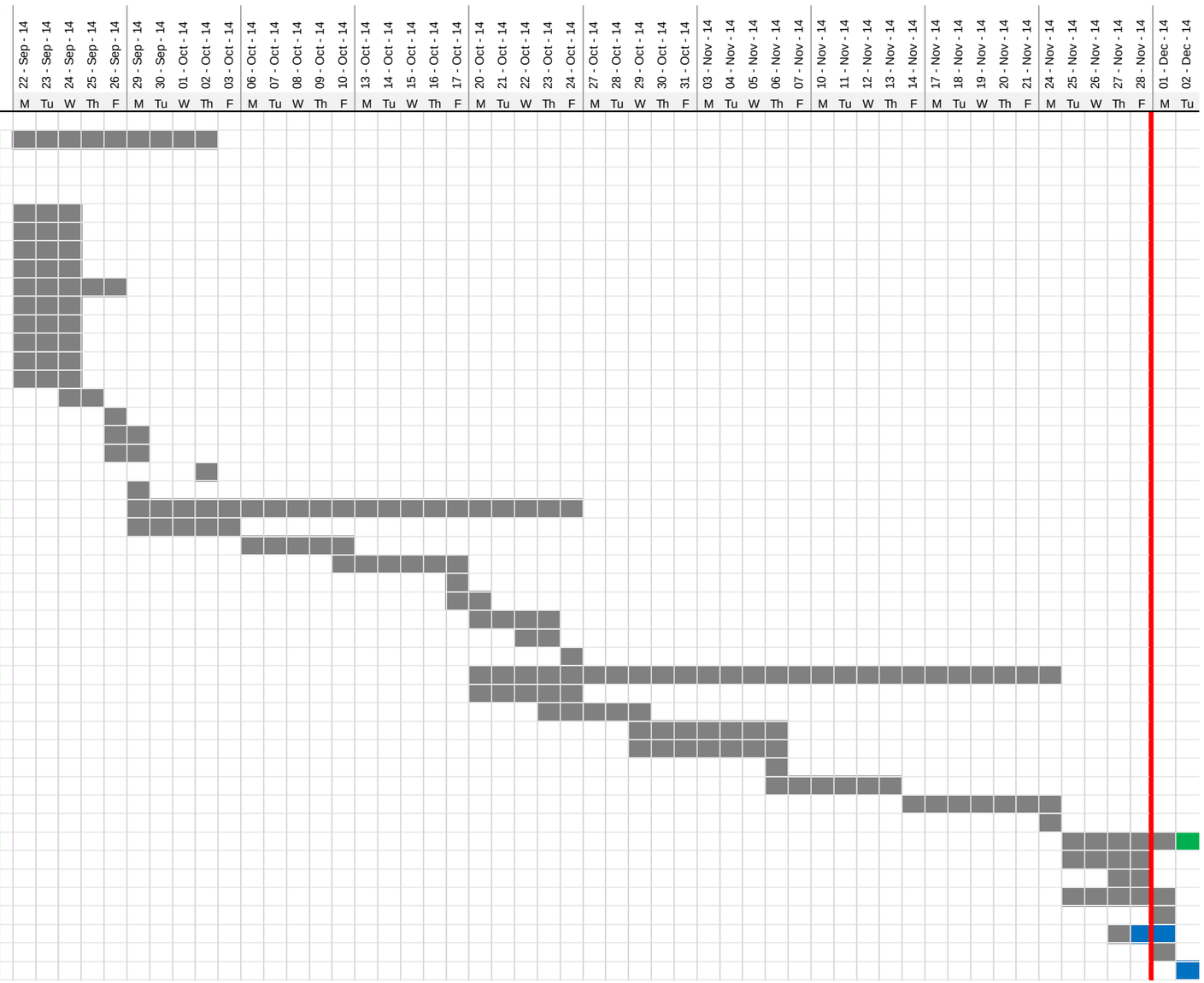


Figure 2: Gantt Chart

Appendix B – Codes and Standards

Prior to introducing new concepts to the suspension system of the tractor, the team conducted extensive research to ensure that the concepts meet the relevant codes and standards. This section includes information regarding ROPS and OECD regulations.

The main applicable standard for this cab suspension system is the Rollover Protection Structure (ROPS) regulation. ROPS is an industry standard designed to protect the operator in the event of a vehicle rollover. The standard consists of two main regulations:

- The cab will not be crushed beneath the weight of the tractor.
- The cab will not become detached from the tractor.

The cab is a purchased component, will not be altered for this project and is compliant with ROPS in regards to not being crushed, so it is not a limitation. However, the cab staying fixed to the tractor in a rollover scenario is directly related to its mounting configuration. Thus, this is an important consideration when altering the mounting system. After our suspension system is designed and constructed, it will need to pass ROPS testing before it is allowed into the marketplace.

Another applicable standard is the Organization for Economic Co-operation and Development (OECD) standard, a set of 9 codes for agricultural and forestry equipment. The relevant code for this project is *OECD Code 4: OECD Standard Code for the Official Testing of Protective Structures on Agricultural and Forestry Tractors*. This code is essentially a rephrasing of the ROPS standard and is also a constraint for our design [1].

The ROPS and OECD standards are applicable to the construction and design of the suspension system. Additionally, *ISO 2631: Mechanical Vibration and Shock – Evaluation of Human Exposure to Whole-Body Vibration* is applicable in evaluating human response and tolerance to vibration and will be referenced when designing the suspension system to ensure vibration frequencies known to cause human discomfort are isolated [2].

References – Appendix B

[1] "OECD Code 4," July 2012. [Online]. Available: http://www.oecd.org/tad/code/Code%204%20-%20Final.pdf. [Accessed October 2014].

[2] U. Katu, R. Desavale and R. Kanai, "Effect Of Vehilce Vibration On Human Body - RIT Experience," Department of Mechanical Engineering, Rajarambapu Institute of Technology, Sakharale, 2010.

Appendix C – Relevant Patents

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Introduction

In order to develop a genuine design for the cab suspension system, it is important to conduct adequate research on the patents that are owned by other individuals or competitors in the industry. The main patent the team considered prior to generating the conceptual designs for the cab suspension system is patent number 2619043.

The tractor cab suspension system in this patent is composed of multiple suspension components that support the cab and reduce the transmission of vibration from the frame of the tractor. The suspension system is arranged such that one end of the cab is composed of resilient suspension members, where the members at the other end of the cab are composed of air cushions. Each air cushion is coupled with a rollover protection member for the purpose of preventing cab-frame detachment during rollover scenarios. TABLE I below represents some basic information regarding this patent.

TABLE I: TRACTOR CAB SUSPENSION SYSTEM PATENT DETAILS

Patent title	Tractor cab suspension system
Patent number	2619043
Inventors	Walter, Benjamin G (Canada)
	Antal, Dean (Canada)

Owners	Walter, Benjamin G (Canada)
	Antal, Dean (Canada)
Applicants	Ben and Donna Walter Farms LTD. (Canada)
Agent	ADE & Company Inc.

As shown in Figure 1, the cab is supported by the rollover protection member (60), which prevents the detachment of the cab from the frame during rollover scenarios. The upper mount (62) is a component of the cab body, while the lower mount (64) is part of the tractor frame. The rollover protection member enables linear sliding movement of the cab relative to the frame, which allows the air cushion (76) to act as a suspension member in the tractor.

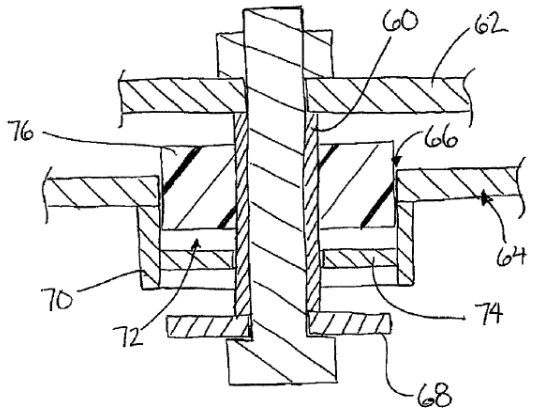


Figure 1: Tractor cab suspension system patent drawing [1]

TABLE II contains the names and numbers of the components in Figure 1 for reference.

TABLE II: COMPONENT LABELS

Component number	Component name
60	Rollover protection member
62	Upper mount
64	Lower mount
66	Aperture
68	End plate
70	Cylindrical collar extension
72	Space gap
74	Retainer ring
76	Air cushion

Reference – Appendix C

[1] B. G. Walter and D. Antal, "Tractor Cab Suspension System". Canada Patent 2619043, 2 July 2009.

Appendix D – Suspension Components

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Introduction

This appendix discusses the suspension components that were considered for implementation into the suspension system. Our suspension system will consist of generic suspension components adapted to our application using original brackets and mounting configurations. The suspension components that were considered are: shock absorbers, struts, rubber springs and anti-vibration mounts. Each of these components is introduced below.

Shock Absorber

The most basic suspension component is a shock absorber, which is a device used to dampen impulses by restricting the fluid flow contained within the shock itself. Generally, there are two types of shock absorbers: pneumatic and hydraulic. Pneumatic shocks use air as the working fluid while hydraulic shocks use oil. As shown in Figure below, the shock uses a simple piston-cylinder construction. The fluid flow in the shock is restricted via valves that slow the travel of the piston within the cylinder, giving it its damping characteristics. Shock absorbers are generally coupled with coil springs where the shock is used to dampen spring oscillations.



Figure 1: Shock [1]

A shock absorber setup is appealing because it is a relatively simple setup, commonly used for similar applications in the industry. Shock absorbers are available from a variety of suppliers in many different sizes and configurations, leading to a variety of damping characteristics.

Shock absorbers have great marketability as they are intuitively recognized as a suspension component by the customer and are widely accepted as a viable component of a suspension system.

Strut

An alternative to the shock absorber is the strut. A strut is essentially a unit consisting of a shock absorber and coil spring. The coil spring aids in suspending, while the shock absorber is used to dampen the spring oscillations. The coil spring is mounted around the shock absorber, as shown in Figure.



Figure 2: Strut [2]

As with shock absorbers, struts are appealing because they are commonly used for similar applications and offer a wide variety of configurations and damping characteristics. They are considered to have good marketability due to their widespread use in the industry.

Rubber Spring

Due to its elastic properties, rubber is generally effective in vibration damping applications. The physical properties of rubber allow it to absorb energy without damaging the structure of the material. When subjected to dynamic loading, the molecules of rubber convert loading energy into heat via internal molecular friction. As can be seen in Figure below, rubber springs have a higher rate of amplitude reduction than steel springs. This is because rubber converts mechanical stress energy into thermal energy, which is easier to dissipate, allowing a higher rate of impact absorption.

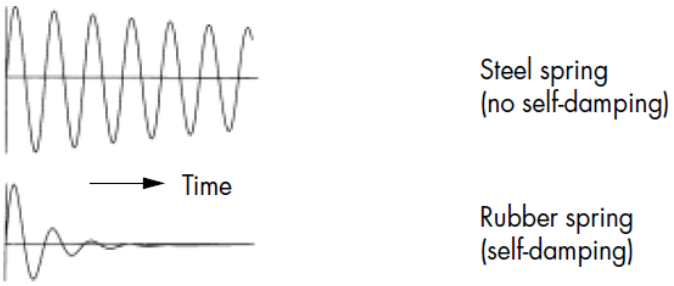


Figure 3: Spring Damping Comparison [3]

Figure 4 below is an example of a double convolution rubber spring manufactured by Timbren.



Figure 4: Rubber Spring [4]

Rubber springs can be used without a shock because of the damping characteristics of the rubber.

Anti-Vibration Mount

An anti-vibration mount is a mounting device that is designed to absorb compressive, tensile and shear loading in all three directions, achieving multi-directional vibration damping. A schematic can be seen below in Figure.

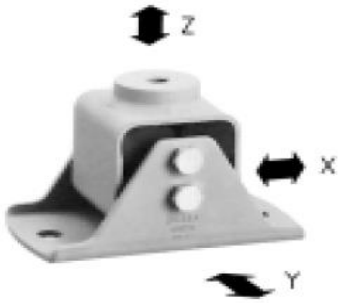


Figure 5: Multi-directional mount for compression, tension and shear loading [3]

The geometrical design of this component allows it to be easily integrated into the tractor, as the support base can be bolted directly to the frame and the top of the mount can be mounted to the underside of the cab.\

References – Appendix D

[1] Works Performance Shocks, "Direct Cycle Parts," 2014. [Online]. Available: http://www.directcycleparts.com/images/works-performance-shocks_ds310094.jpg. [Accessed October 2014].

[2] Auto Sphere, "Monroe Quik Strut," 2014. [Online]. Available: http://www.autosphere.ca/carcarebusiness/files/2013/10/Monroe-Quik-Strut.jpg. [Accessed October 2014].

[3] Rosta, "Rosta Anti-Vibration Mountings," [Online]. Available: http://www.rostainc.com/pdfs/applications/Rosta_Catalogues/Antivibration%20Mountings.pdf. [Accessed October 2014].

[4] Timbren, "Double Convolution Aeon Spring," 2014. [Online]. Available: http://timbren.com/double-convolution/. [Accessed October 2014].

Appendix E – Engineering Drawings

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Introduction

The following appendix section contains all relevant engineering part and assembly drawings required for the implementation of the proposed design. These drawings are all preliminary, and must be confirmed by a professional engineer before sending them to manufacturing.

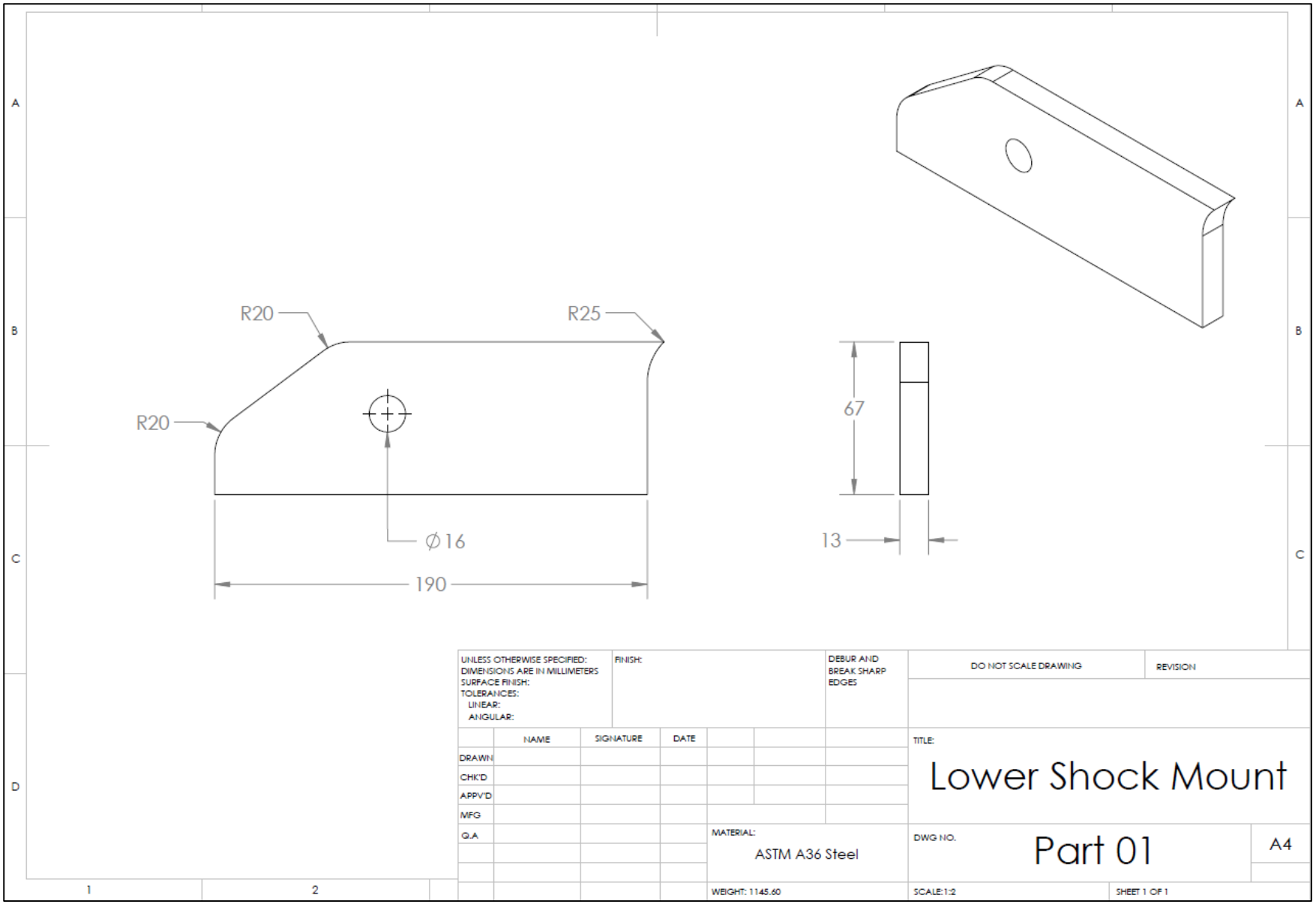


Figure 1: Lower Shock Mount Drawing

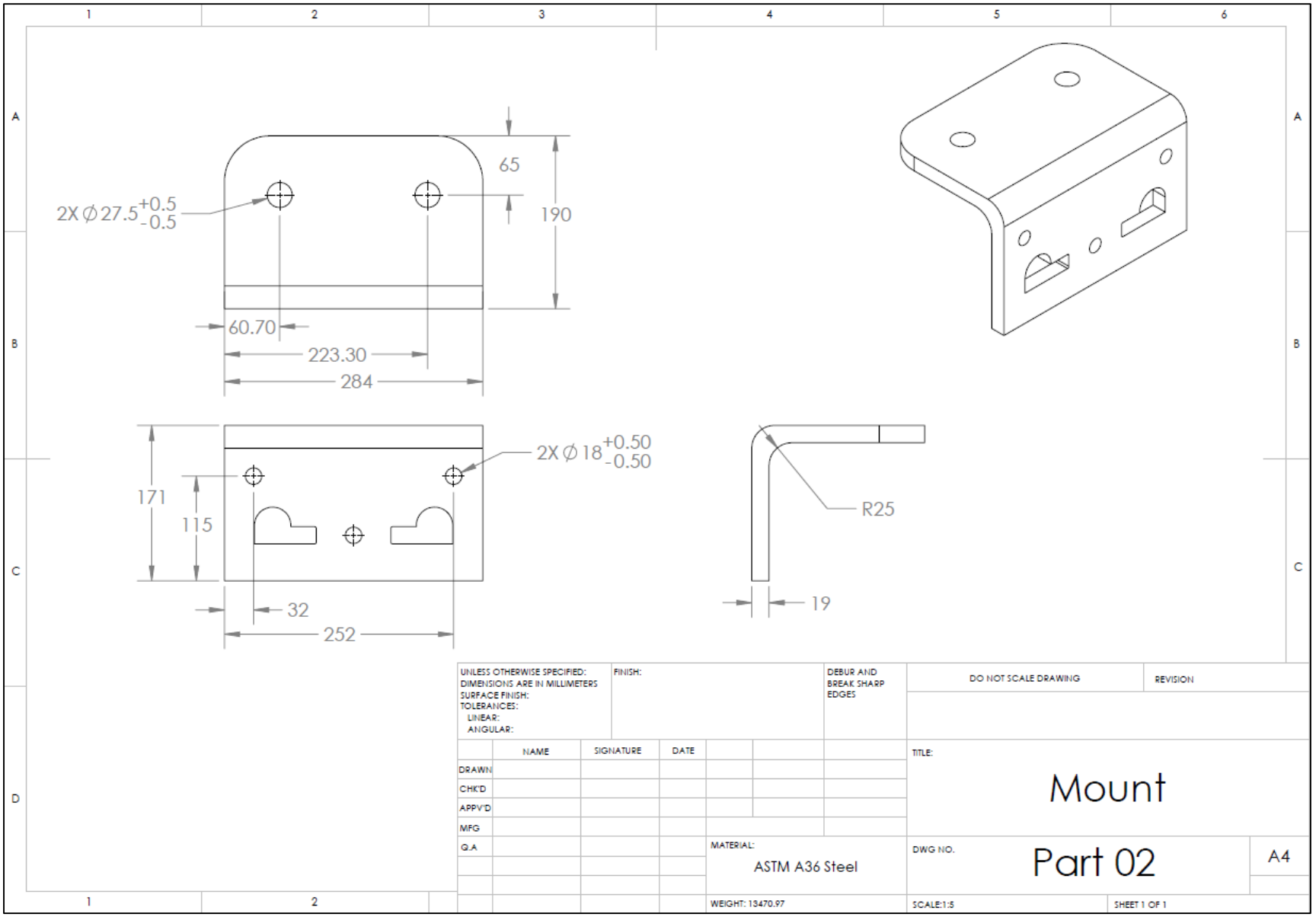


Figure 2: Mounting Bracket Drawing

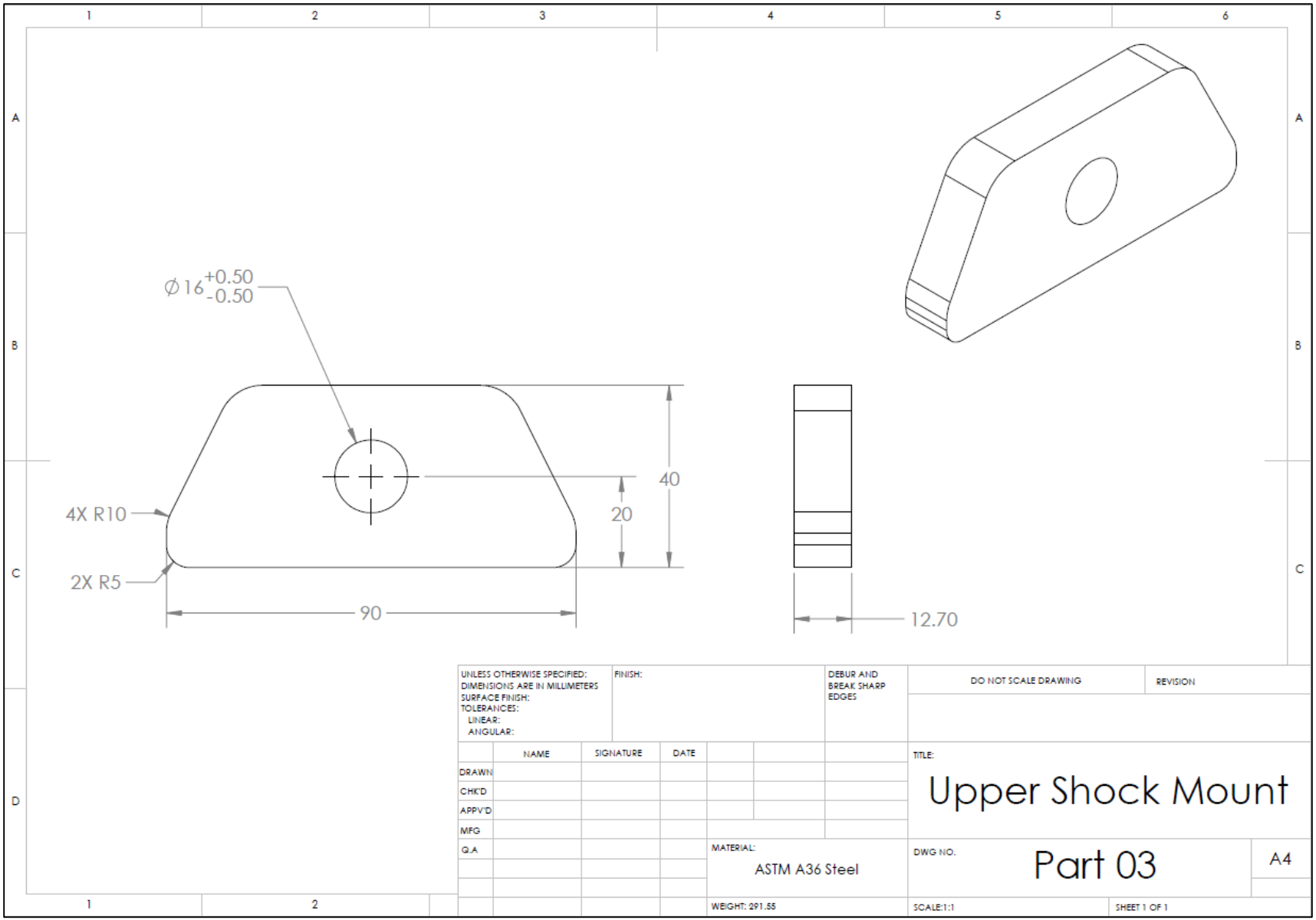


Figure 3: Upper Shock Mount Drawing

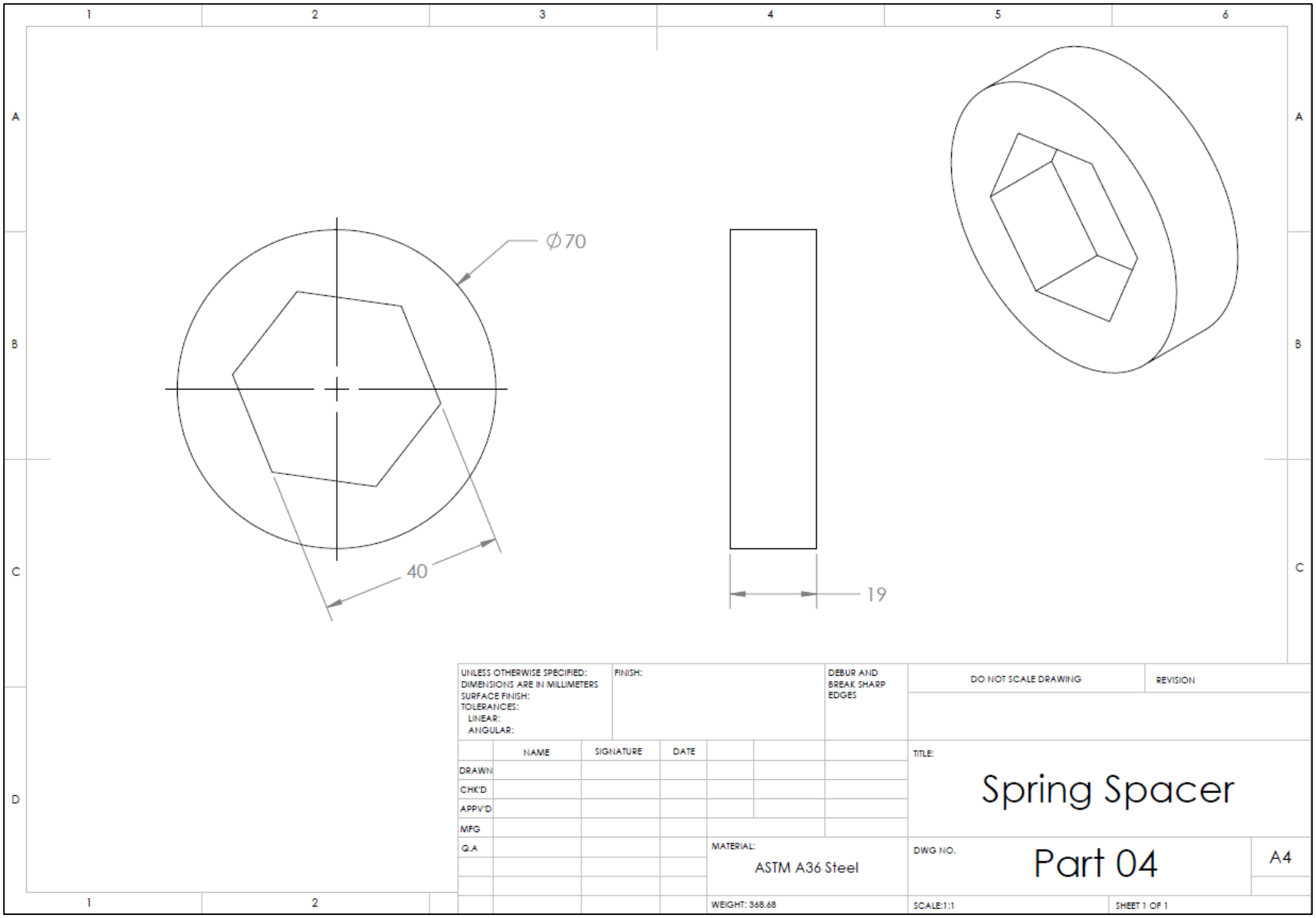


Figure 4: Spring Spacer Drawing

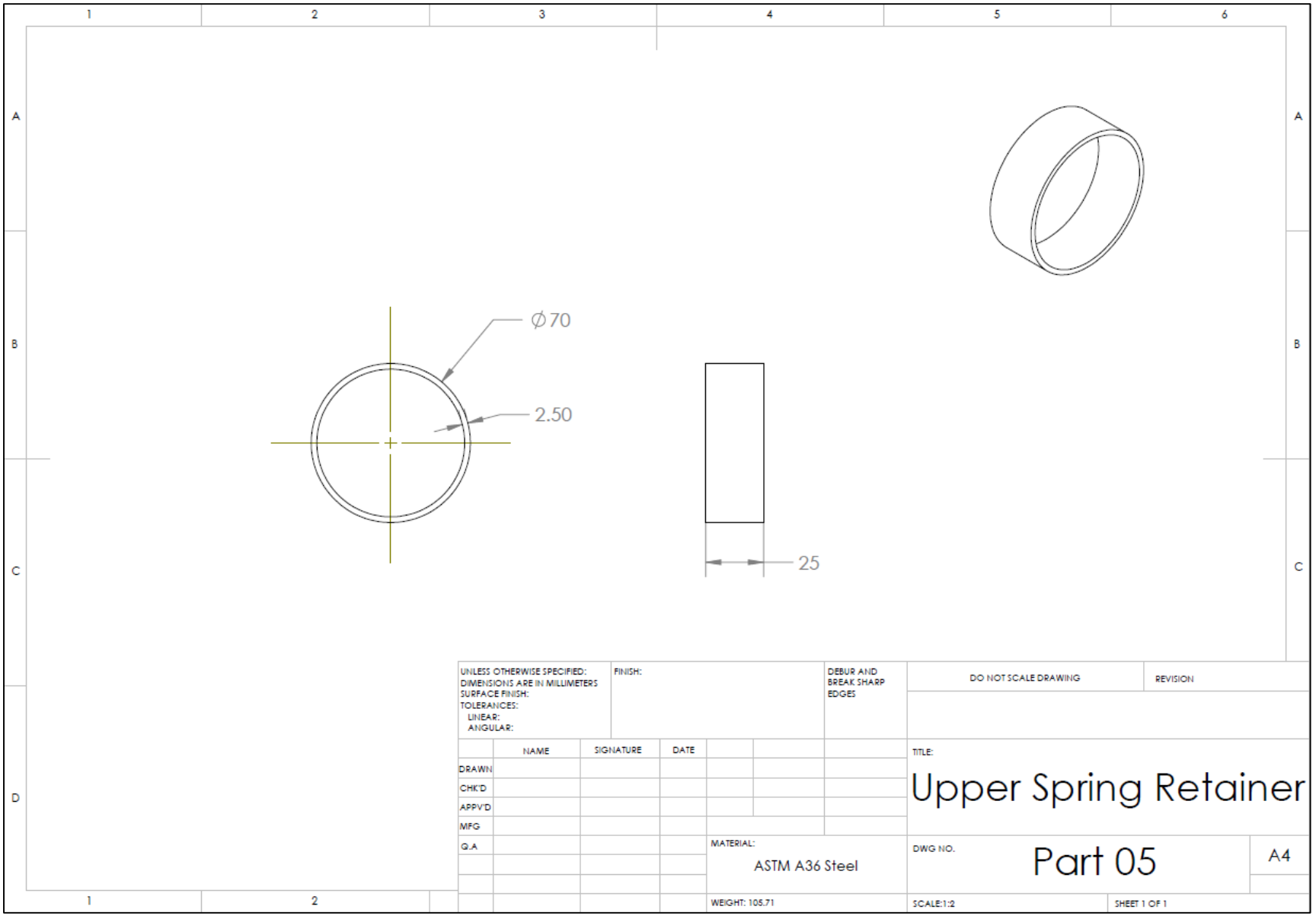


Figure 5: Upper Spring Retainer Drawing

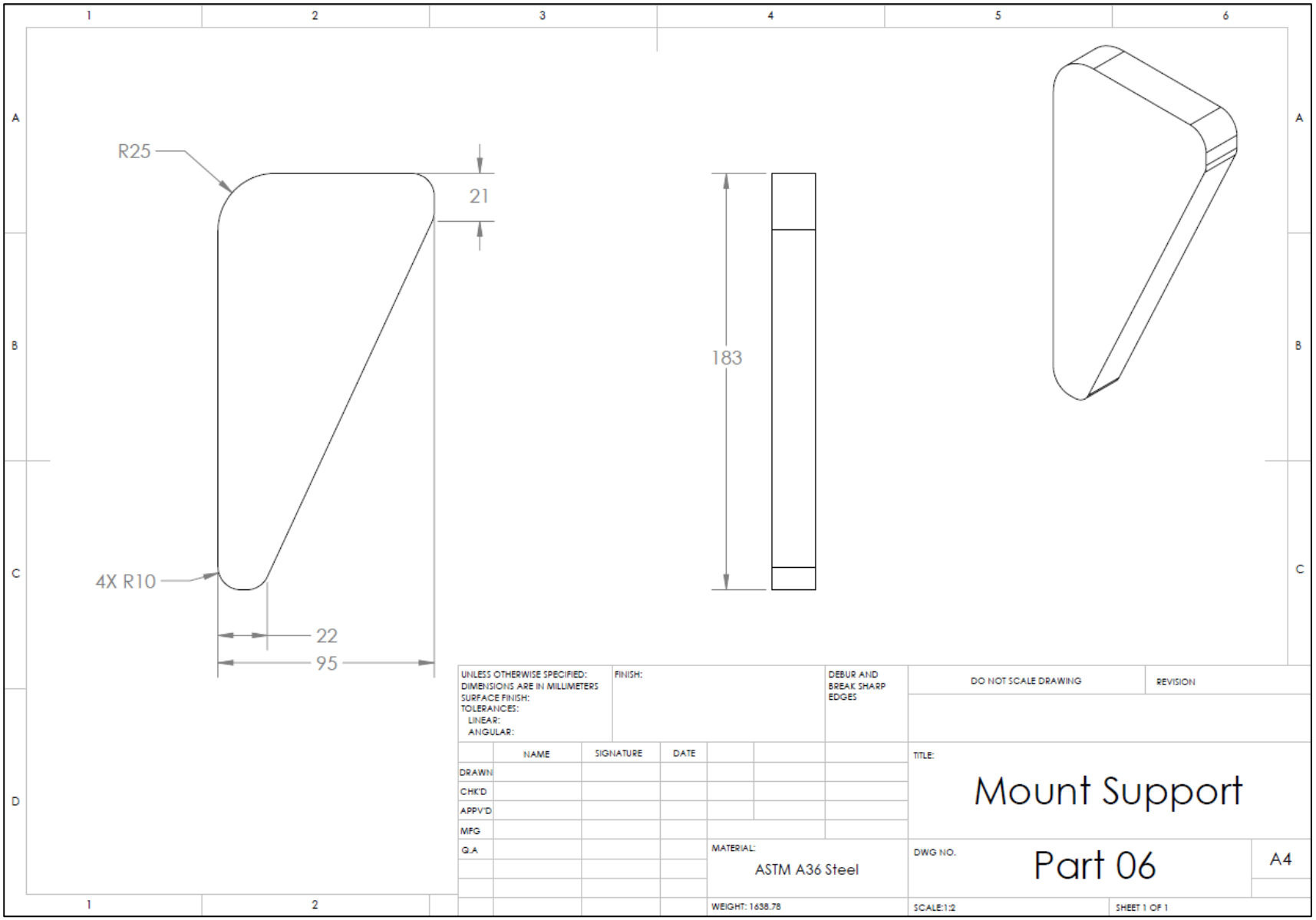


Figure 6: Mount Support Drawing

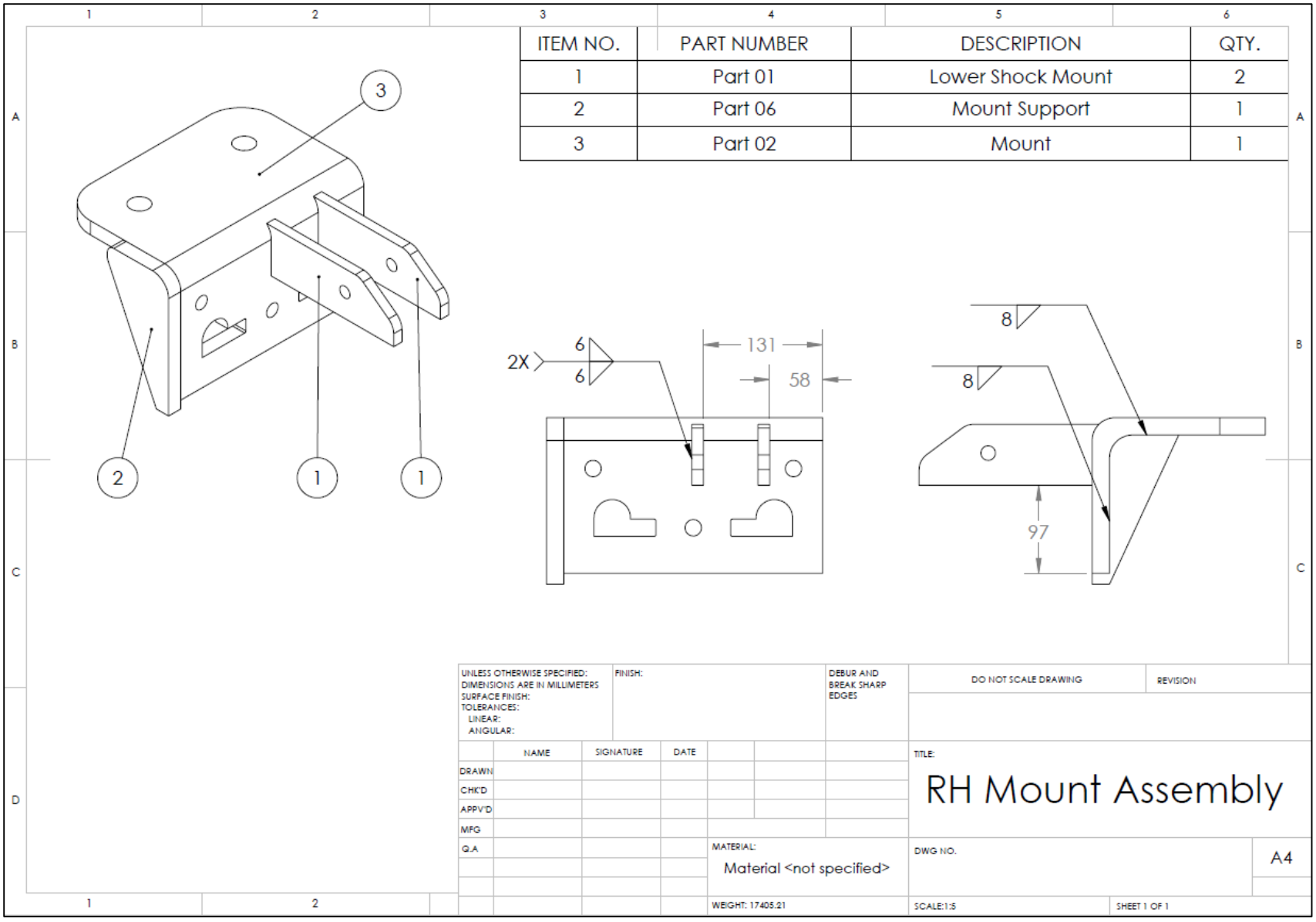


Figure 7: RH Mount Assembly Drawing

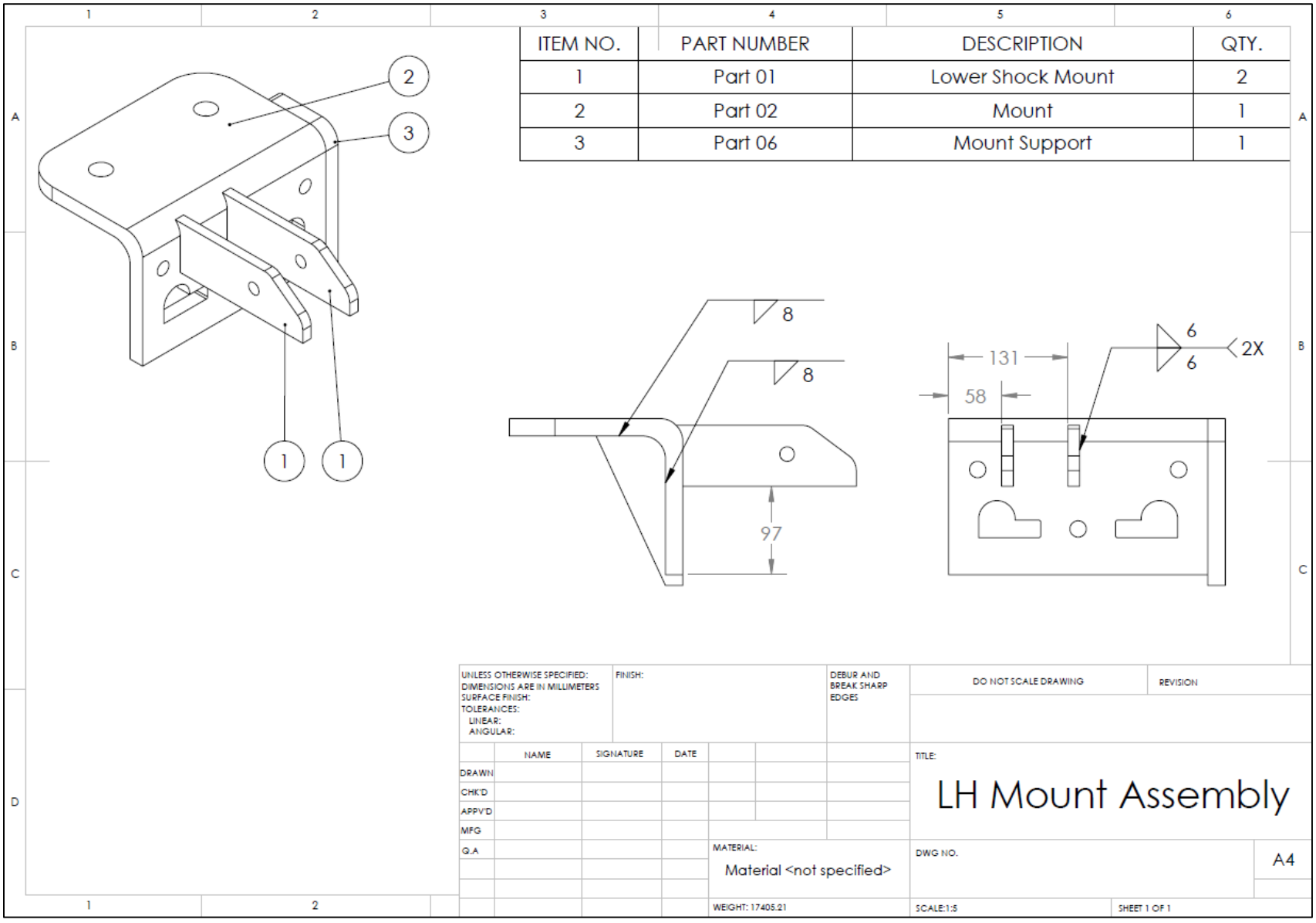


Figure 8: LH Mount Assembly Drawing

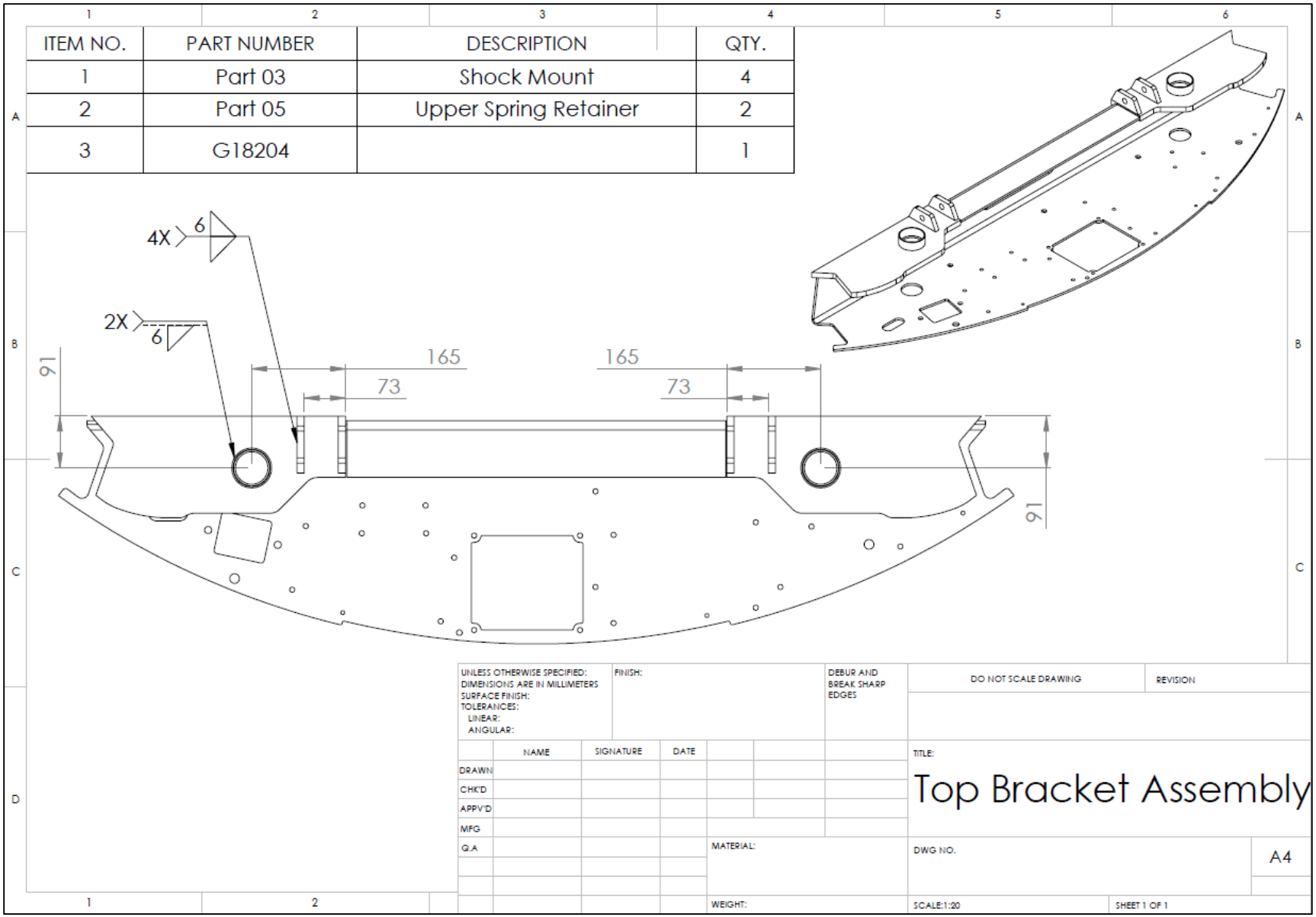


Figure 9: Top Bracket Assembly Drawing

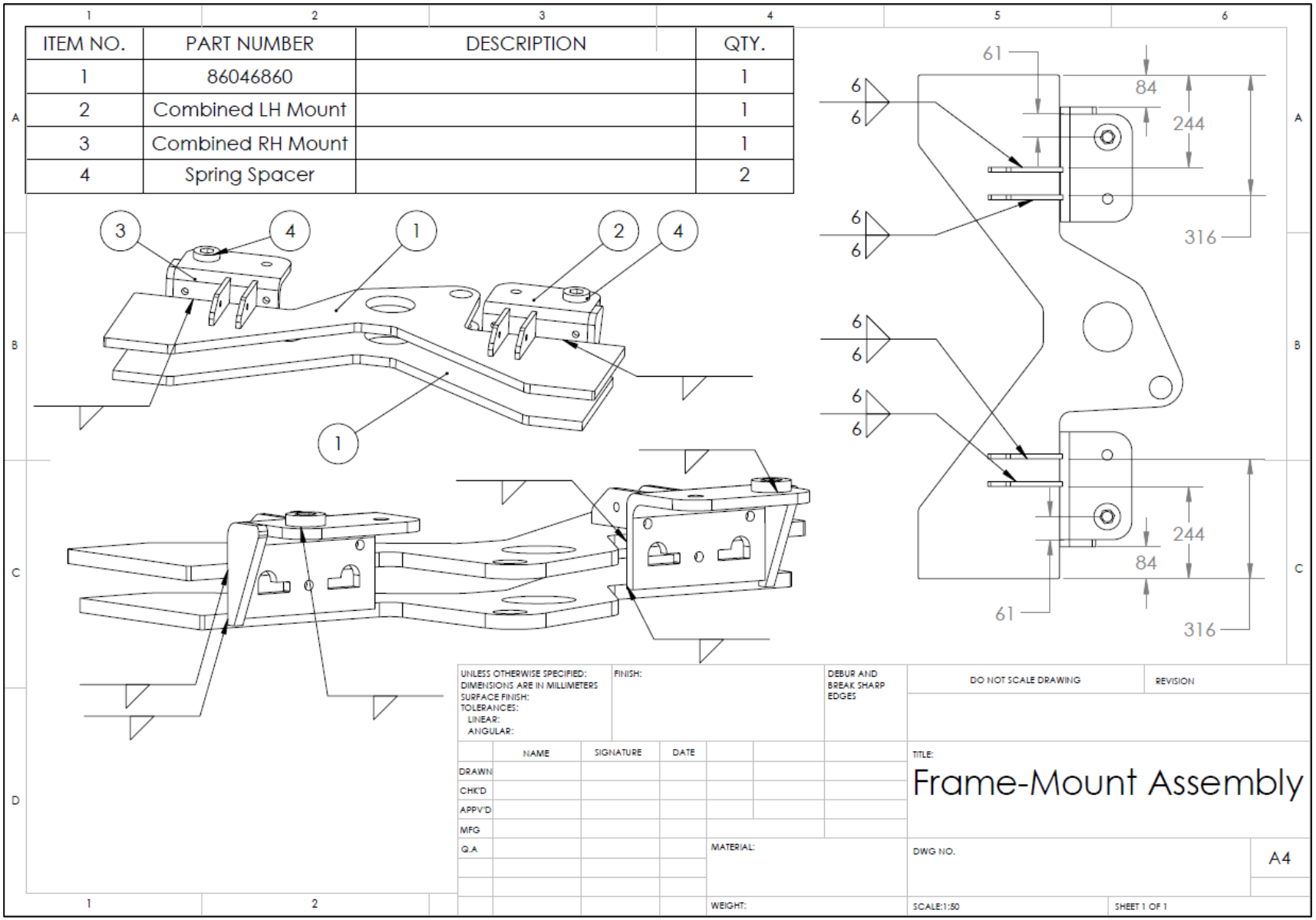
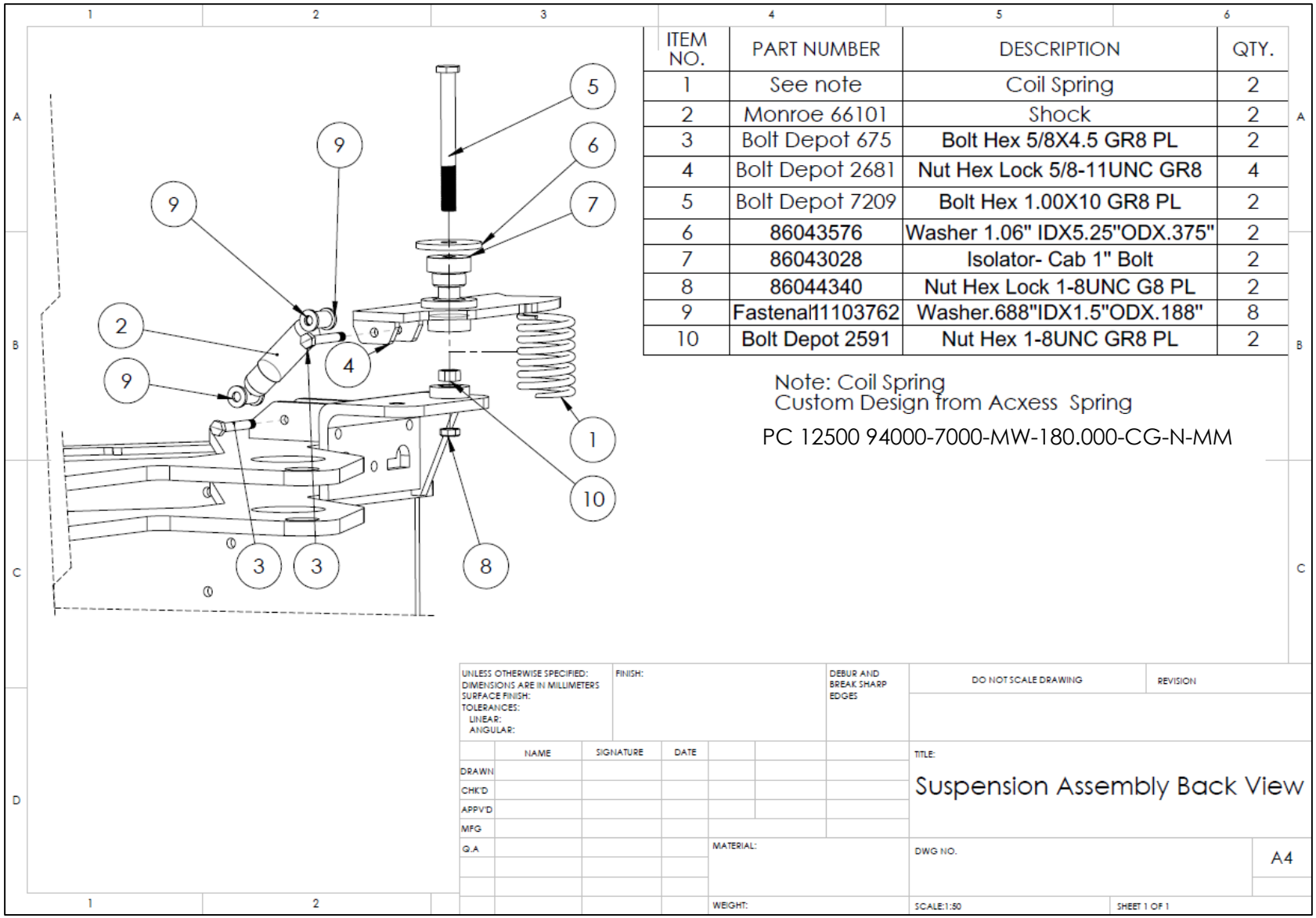


Figure 10: Frame-Mount Assembly Drawing



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	See note	Coil Spring	2
2	Monroe 66101	Shock	2
3	Bolt Depot 675	Bolt Hex 5/8X4.5 GR8 PL	2
4	Bolt Depot 2681	Nut Hex Lock 5/8-11UNC GR8	4
5	Bolt Depot 7209	Bolt Hex 1.00X10 GR8 PL	2
6	86043576	Washer 1.06" IDX5.25"ODX.375"	2
7	86043028	Isolator- Cab 1" Bolt	2
8	86044340	Nut Hex Lock 1-8UNC G8 PL	2
9	Fastenal11103762	Washer.688"IDX1.5"ODX.188"	8
10	Bolt Depot 2591	Nut Hex 1-8UNC GR8 PL	2

Note: Coil Spring
Custom Design from Acxess Spring

PC 12500 94000-7000-MW-180.000-CG-N-MM

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

Suspension Assembly Back View

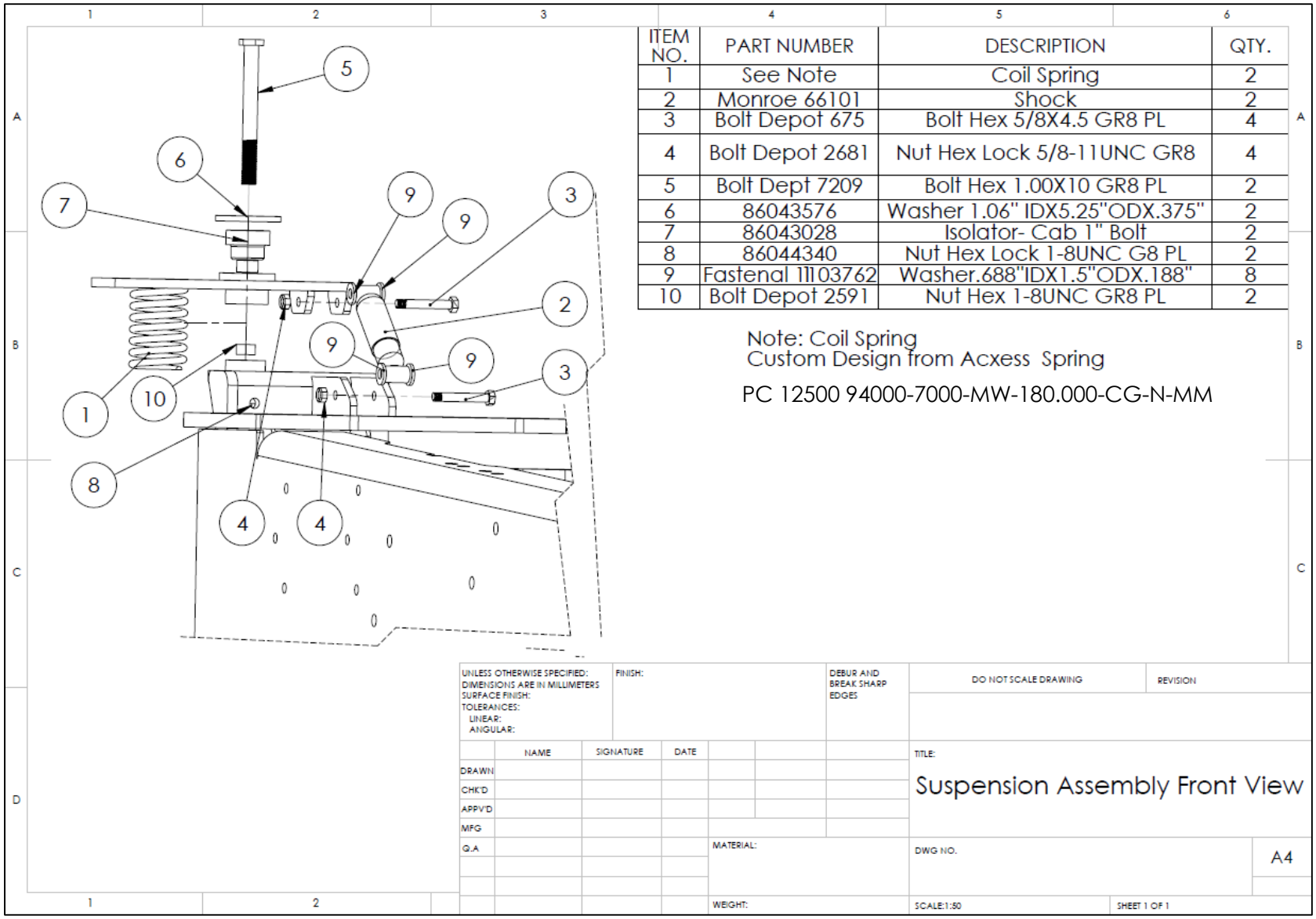
DWG NO.

A4

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SHEET 1 OF 1

Figure 11: Suspension Assembly Back View Drawing



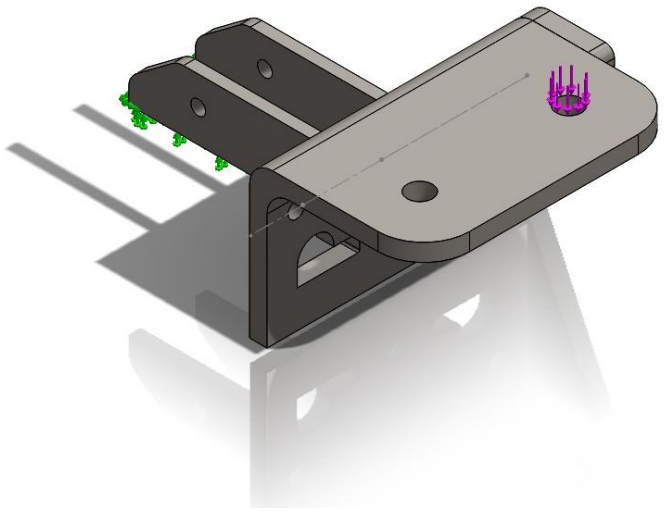
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1	See Note	Coil Spring	2
2	Monroe 66101	Shock	2
3	Bolt Depot 675	Bolt Hex 5/8X4.5 GR8 PL	4
4	Bolt Depot 2681	Nut Hex Lock 5/8-11UNC GR8	4
5	Bolt Dept 7209	Bolt Hex 1.00X10 GR8 PL	2
6	86043576	Washer 1.06" IDX5.25"ODX.375"	2
7	86043028	Isolator- Cab 1" Bolt	2
8	86044340	Nut Hex Lock 1-8UNC G8 PL	2
9	Fastenal 11103762	Washer.688"IDX1.5"ODX.188"	8
10	Bolt Depot 2591	Nut Hex 1-8UNC GR8 PL	2

Figure 12: Suspension Assembly Front View Drawing

Appendix F – FEA Simulation on Mounting Bracket Assembly

The following Appendix section contains a preliminary FEA simulation on the designed Mounting Bracket Assemblies in order to validate the designs ability to support the static weight of the cab.

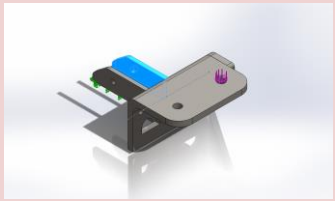
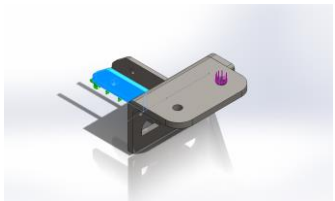
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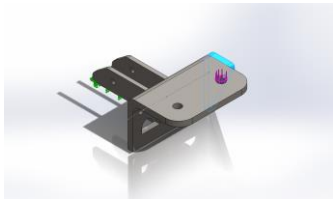


Model name: Combined RH Mount

Current Configuration: Default

Solid Bodies

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		Weight:132.015 N	
Fillet4 	**Solid Body**	Mass:1.64304 kg Volume:0.000209304 m^3 Density:7850 kg/m^3 Weight:16.1018 N	F:\Part Drawings and Parts\Part 06.SLDPRT **Nov 27 19:30:40 2014**

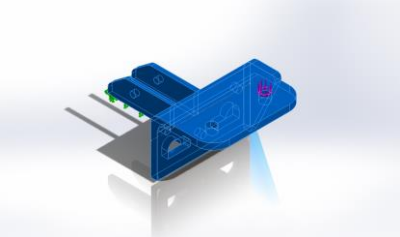
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Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off
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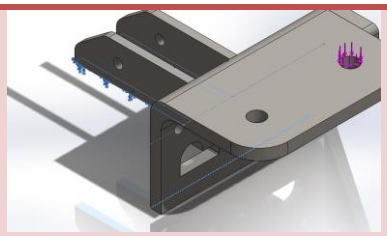
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Pressure/Stress	N/m^2

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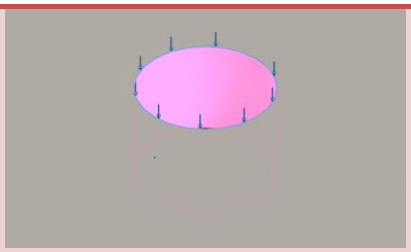
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Loads and Fixtures

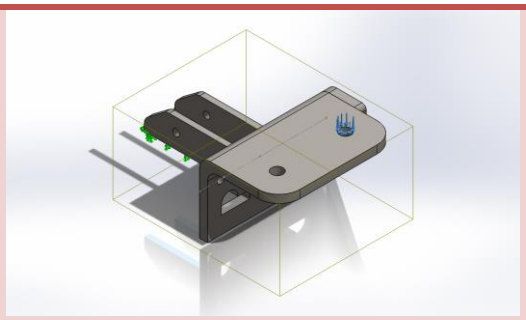
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Resultant Forces

Components	X	Y	Z	Resultant
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Reaction Moment(N·m)	0	0	0	0

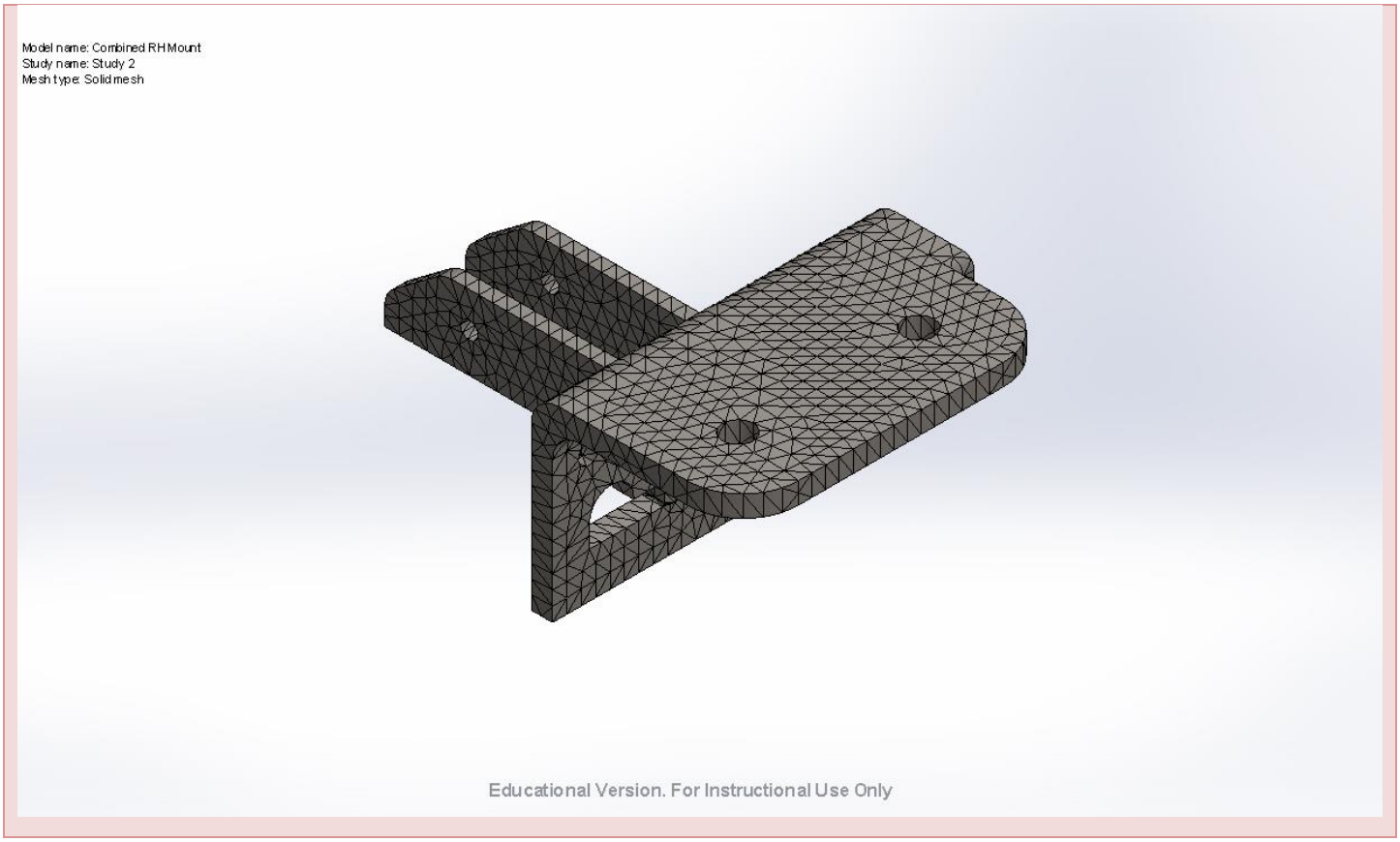
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Contact Information

Contact	Contact Image	Contact Properties
Global Contact		**Type:** Bonded **Components:** 1 component(s) **Options:** Compatible mesh

Mesh Information

Mesh type	Solid Mesh
Mesher Used:	Curvature based mesh
Jacobian points	4 Points
Maximum element size	0 mm
Minimum element size	0 mm
Mesh Quality	High
Remesh failed parts with incompatible mesh	Off
Total Nodes	15671
Total Elements	8832
Maximum Aspect Ratio	17.954
% of elements with Aspect Ratio < 3	98.4
% of elements with Aspect Ratio > 10	0.0793
% of distorted elements(Jacobian)	0
Time to complete mesh(hh;mm;ss):	00:00:05
Computer name:	CEMELAB-10



Resultant Forces

Reaction Forces

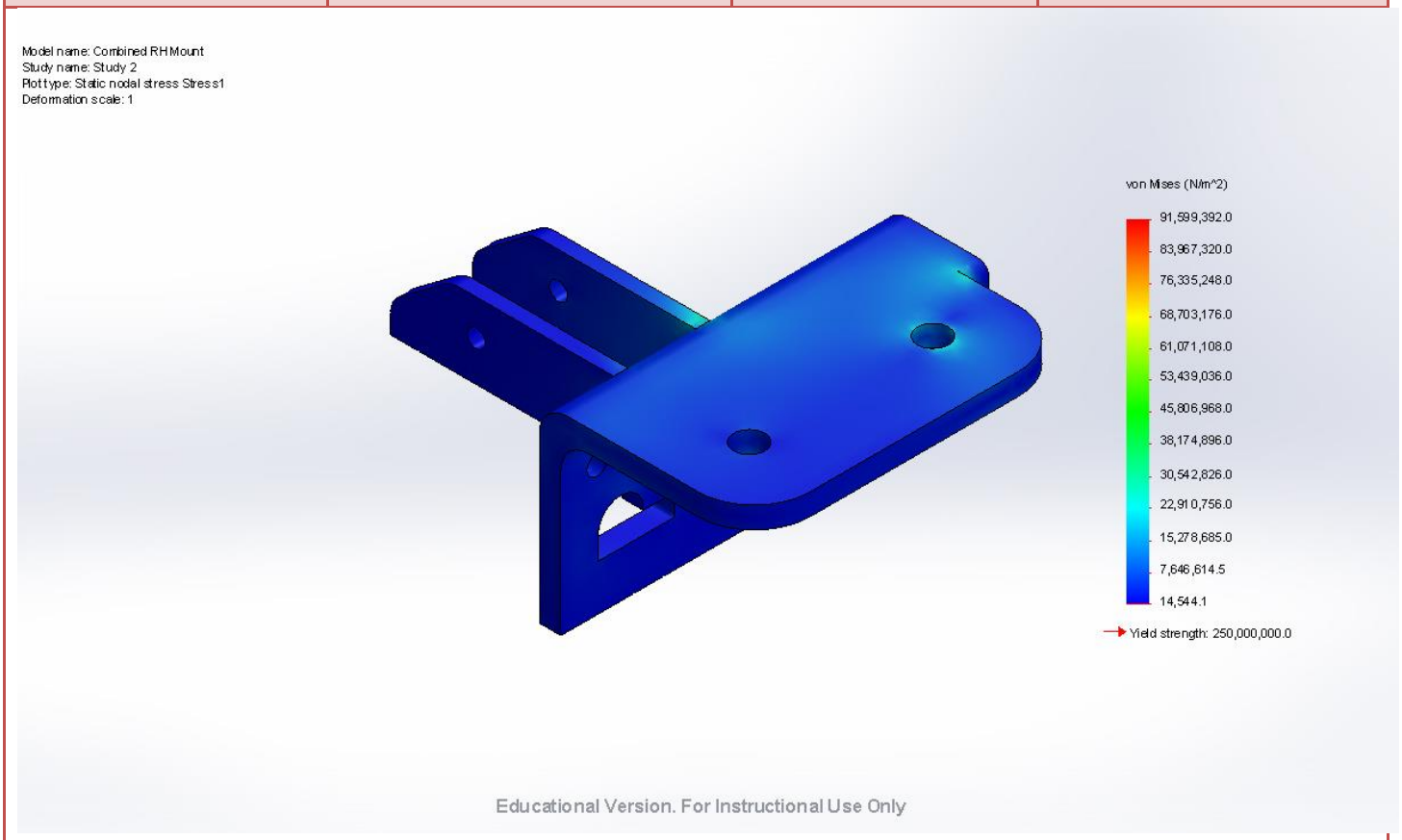
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Entire Model	N	0.134003	3020.72	-0.214073	3020.72

Reaction Moments

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N·m	0	0	0	0

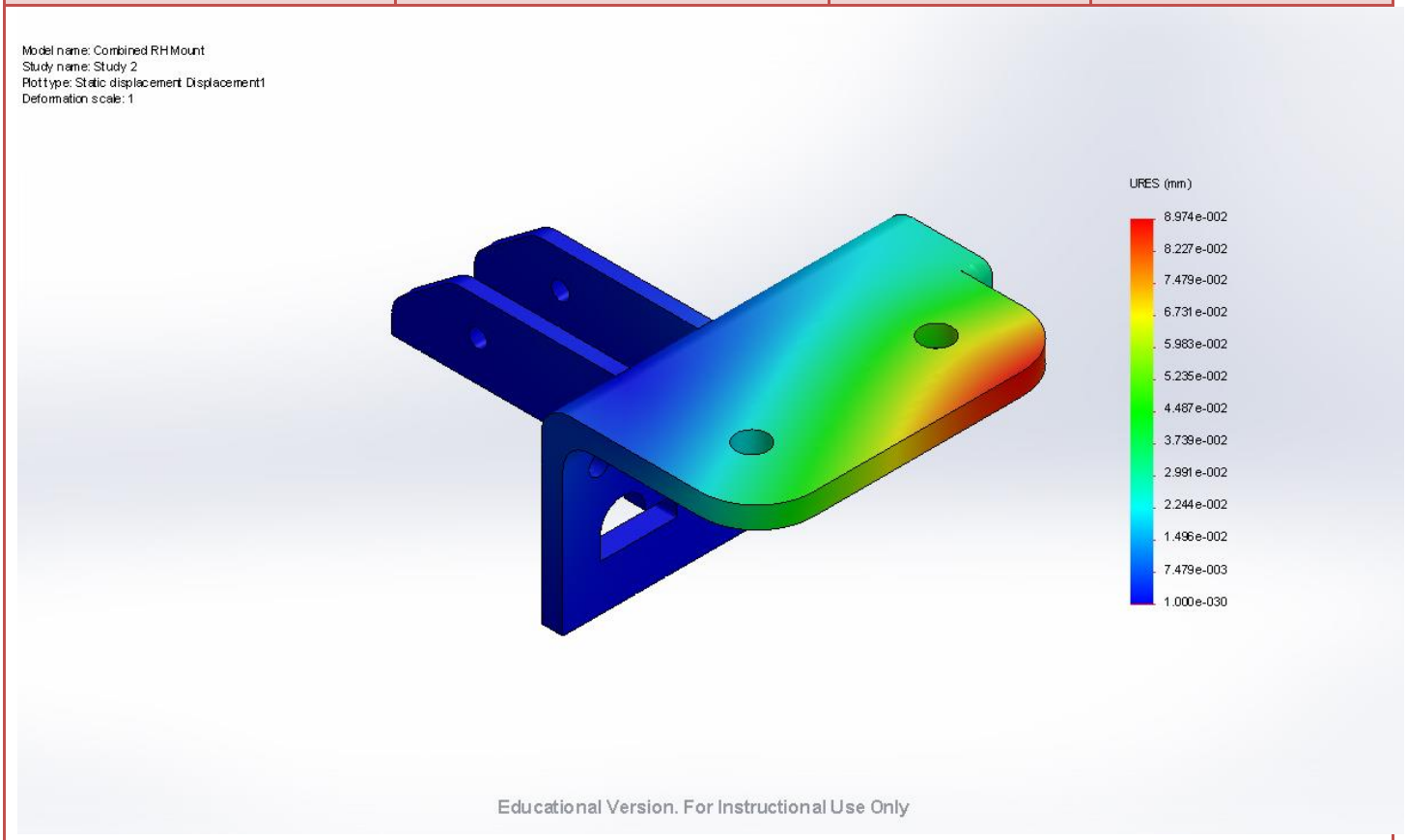
Study Results

Name	Type	Min	Max
Stress1	VON: von Mises Stress	14544.1 N/m^2 Node: 2793	9.15994e+007 N/m^2 Node: 15660



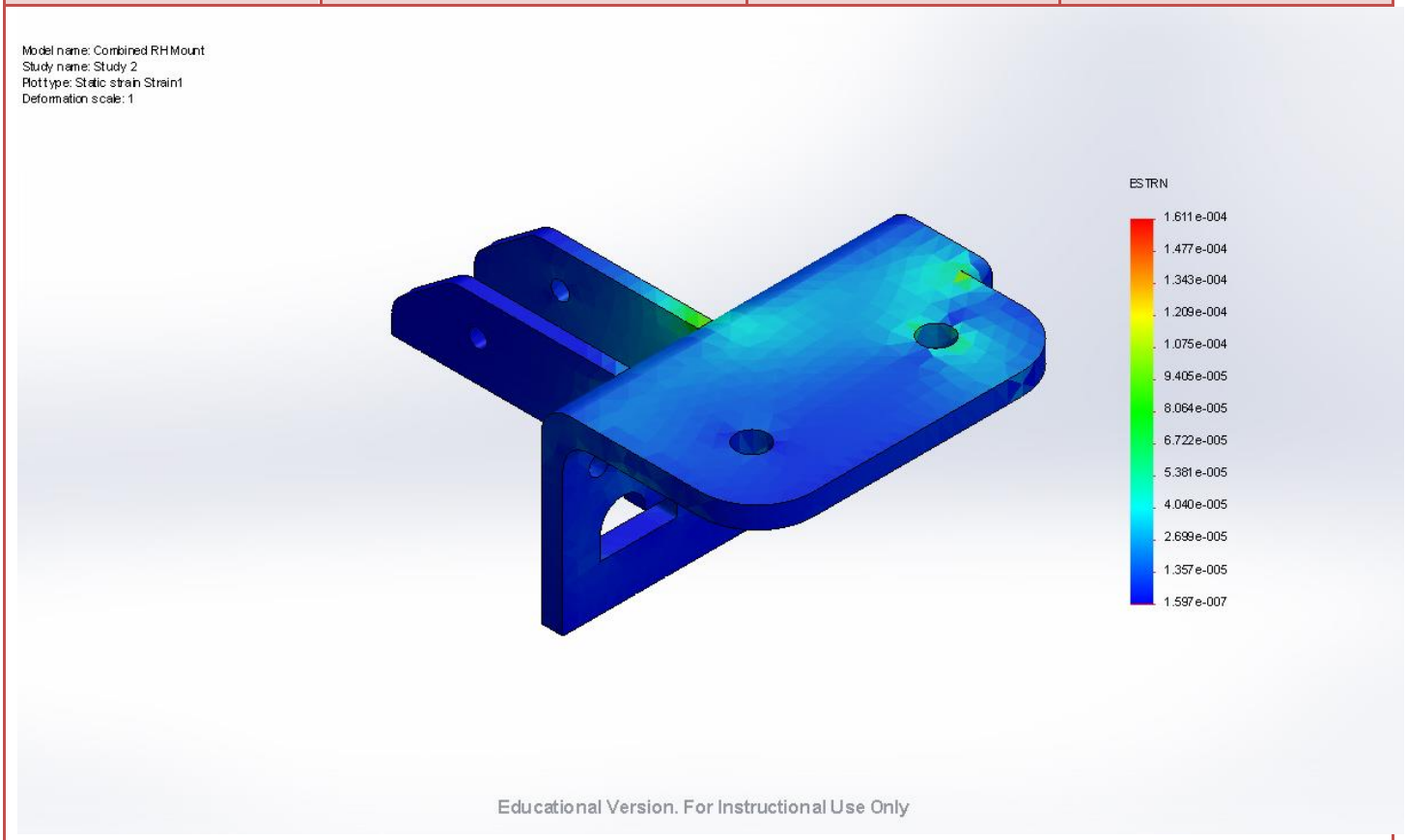
Combined RH Mount-Study 2-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 23	0.089744 mm Node: 3264



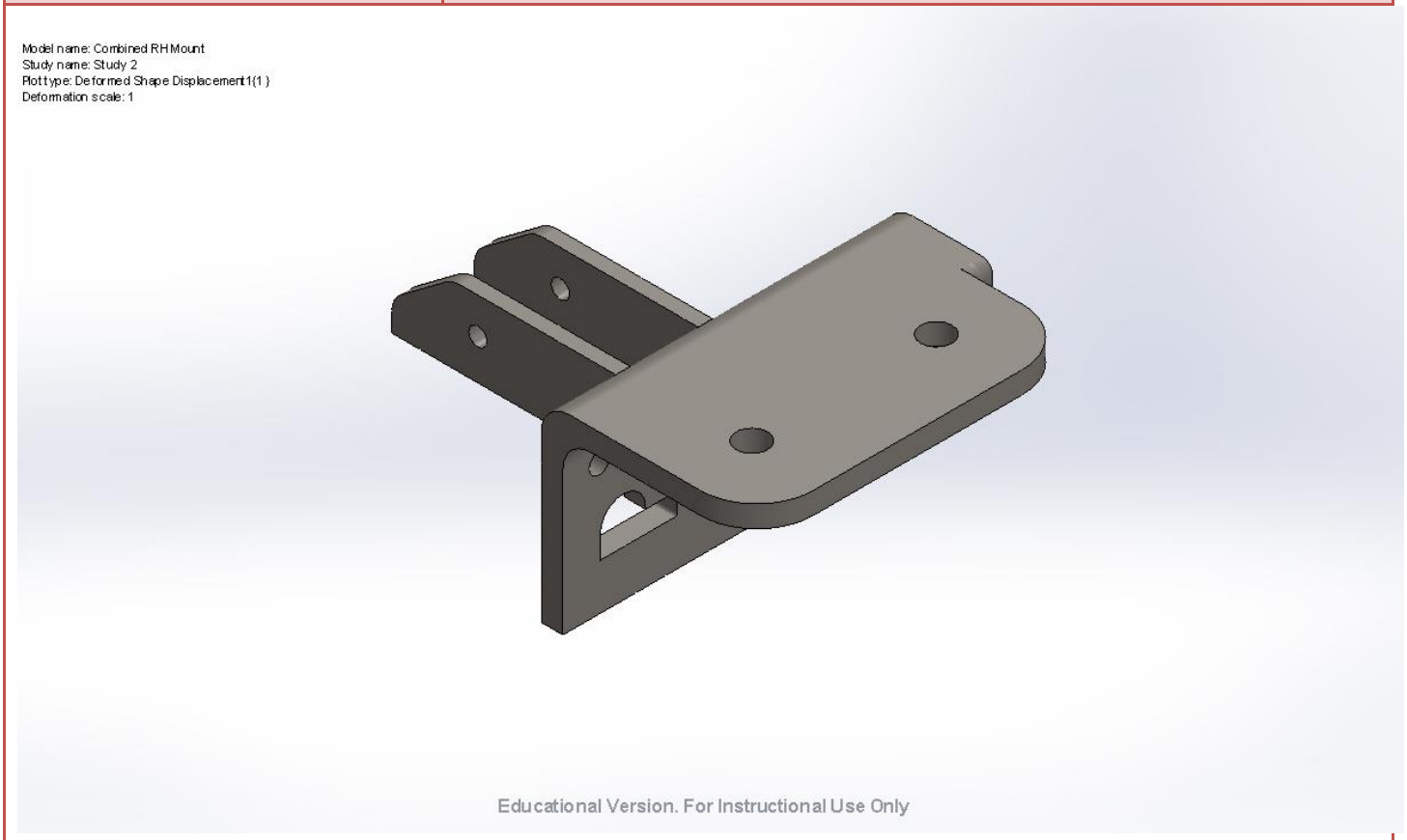
Combined RH Mount-Study 2-Displacement-Displacement1

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	1.59717e-007 Element: 1361	0.000161116 Element: 8828

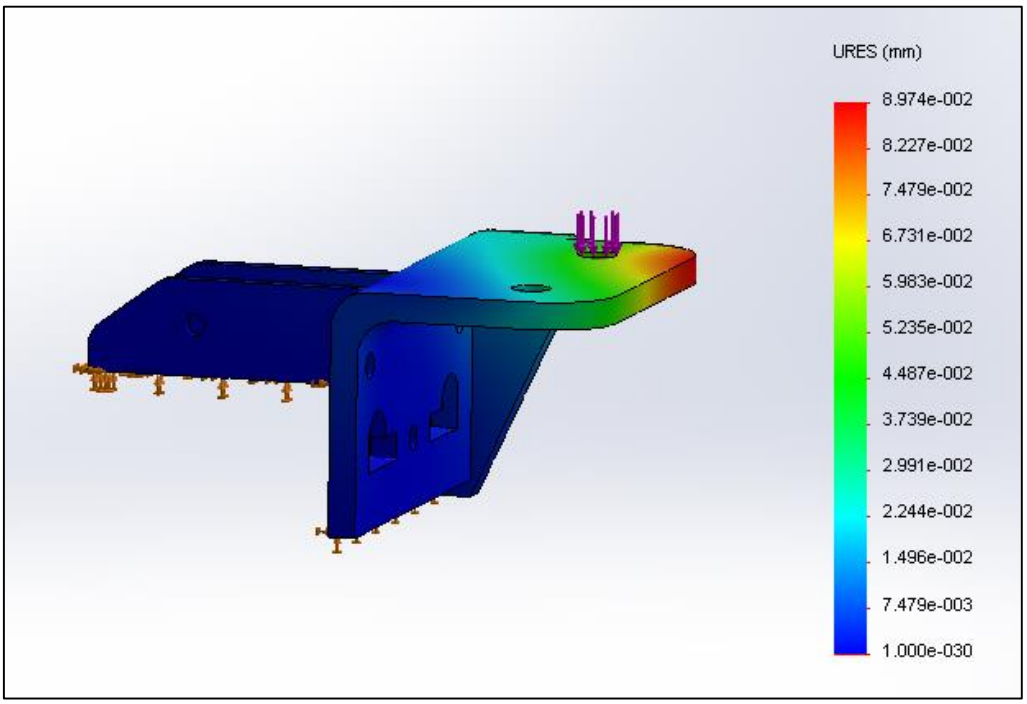


Combined RH Mount-Study 2-Strain-Strain1

Name	Type
Displacement1{1}	Deformed Shape



Combined RH Mount-Study 2-Displacement-Displacement1{1}



Displacement of support bracket under static loading – displacement to scale. Maximum displacement is 0.09 mm under static loading

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