Bumper Lift Assist Design

S&V Manufacturing

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Executive Summary

The following document outlines Team 6's final design proposal for MECH 4860, Engineering Design, Fall 2017. The objective of this report is to provide a detailed design for a lift assist mechanism to aid in raising and lowering S&V Manufacturing's line of Bison Grille Guard products. S&V Manufacturing is a Winnipeg based company that designs, manufactures, and distributes high strength grille guards for semi-trucks, pickup-trucks, and SUVs. The grille guard's purpose is to protect the vehicle and its passengers in the event of a collision with wildlife. The grille guards accomplish this by having high strength aluminum designs, which in turn, makes them heavy and cumbersome for the operators to raise and lower. The design of a grille guard lift assist mechanism is critical to ensure the safety of the vehicle operates and allow for S&V Manufacturing to remain competitive in the marketplace.

The teams proposed lift assist mechanism is an assembly containing 16 components, including two custom manufactured 1.35 [Nm/º] torsional springs. The addition of the torsional springs promise to reduce the grille guards weight by up to 75.21 [%], while only requiring a maximum force of 14.33 [lbf] to raise the grille guard from the lowered position. This proposed lift assist design has an estimated unit cost of $632.92 CAD which can be reduced to $349.02 CAD per unit when the number of units produced is increased to 100. There is also tremendous potential to further reduce the unit cost of the lift assist design by increasing the production quantity.

The latch design proposed by the team is a modification of the tow pin design currently being used by S&V Manufacturing. This advanced tow pin design consists of five separate components which include multiple keys and key seats, allowing for easy locking and unlocking of the grille guard. The proposed latch design has an estimated unit cost of $0.00 CAD since all required materials can be sourced from S&V Manufacturing's scrap material. All manufacturing procedures required to assemble the latch mechanism are already in place at S&V Manufacturing. This design will reduce the time it takes to lock and unlock the grille guard as well as ensure the grille guard remains in its upright position during operation of the vehicle.

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Glossary of Terms

Eye Bolt – Bolt type where the fastener material is curved into a circular handle to tighten or loosen the bolt.

Grille Guard – A device installed on the front of a vehicle, usually a semi-trailer, to provide protection to the vehicle and its occupants in the case of a collision.

Hinge Point – Attachment location between the grille guard and the mounting bracket which allows the grille to rotate 90^o between the raised and lowered positions.

Mounting Bracket – A bracket secured to the frame of a vehicle which is used to mount the grille guard.

MTS – Abbreviation for “made to specifications”. Term used in procurement to symbolize that the part is custom made to the design specifications.

OTS – Abbreviation for “off the shelf”. Term used in procurement to symbolize that the part is a stock component purchased from a supplier implemented in the assembly without any additional modifications.

Tow Pin – Solid steel peg which is inserted into a series of parallel holes in the grille guard and mounting bracket used to secure the grille guard in the raised position.

1 Introduction

This report outlines the design of a lift assist and latch mechanism for S&V Manufacturing's line of Bison Grille Guard products. The report starts by introducing the customer background, which includes product information, current concerns, and notable competitors. This is followed by the lift assist and latch mechanism design requirements and specifications provided by the customer. Finally, the detailed design description of the proposed lift assist, and latch mechanisms are presented.

1.1 Problem Statement

S&V Manufacturing requires a lift assist system to aid vehicle operators in lowering and raising their Bison Grille Guard products. The lift assist system needs to reduce the overall force required by the operators to raise and lower the grilles, while also making the process fast, and safe. S&V Manufacturing has also requested a latch system to replace the two eye bolts that are currently used to secure the grille guard. This will reduce the time needed to lock the grille guard in the raised position as well as improve the process for the operators. The latch and lift assist mechanisms need to be adjustable, and compatible with all S&V Manufacturing Bison Grille Guard designs.

1.2 Project Objectives

The objective of this project is to design a lift assist and latch system to aid in the lowering and raising of S&V Manufacturing's line of Bison Grille Guards. The lift system needs to be adjustable, and compatible with all the current grille guard products. The deliverables for S&V Manufacturing are the full designs for the proposed lift assist and latch mechanisms. This includes preliminary engineering CAD Models, preliminary engineering drawings, and a preliminary bill of materials (BOM).

1.3 Background

This section provides background information on S&V Manufacturing. Company background, current product information, and issues related to the existing products are detailed in the following sections. Notable competitors are also mentioned to outline the status of the marketplace.

1.3.1 Company Background

S&V Manufacturing is a Winnipeg based manufacturing company that specializes in the production of high strength Bison Grille Guards. Their signature grille guard designs are made from high strength aluminum to provide superior protection to drivers in the unfortunate case of a collision with wildlife. S&V Manufacturing offers a wide variety of products ranging from grille guards suited for smaller pickup trucks and SUV's, to larger semi-trailers as well as custom work to suit the customer's individual needs [1]. All aspects of production are handled by S&V Manufacturing including the design and fabrication of the grille guards as well as their distribution. S&V Manufacturing has partnered with Volvo and MACK dealers to create a wide distribution network that spans across North America [2].

1.3.2 Product Background

The Bison Grille Guard systems produced and sold by S&V Manufacturing consist of two major components: the grille guard, and the mounting bracket. Each grille type has a unique mounting bracket which depends on the style of the grille, as well as the type of vehicle it will be mounted to.

The grilles are made from 6063 T4 and T5 aluminum with varying thicknesses, depending on the model, and 44W steel for the mounting brackets. The grille guards range in weight from 175 [lbs] to 225 [lbs] depending on the model, while each bracket weighs approximately 140 [lbs] [3].

To install the grille guard, the mounting bracket is bolted to the frame of the vehicle. In some cases, the vehicle's stock bumper must be removed to accomplish this. The grille guard is then fastened to the mounting brackets hinge point using two 20 [mm] bolts. The hinge points allow the grille guard to move between the raised and lowered positions. Figure 1 shows the grille guard in the lowered position, also revealing the mounting bracket.

Figure 1: Freightliner Cascadia Evolution 2018 (FCASE-4B) [4]

While in the raised position, the grille guard is secured to the mounting bracket by a 1 ¾ [in.] steel tow pin as well as two 20 [mm] eye bolts, as shown in Figure 2.

Figure 2: S&V Manufacturing Bison Grille Guard on a Freightliner Semi-Trailer [5]

1.3.3 Current Product Concerns

The current grille guard system set-up, outlined in section 1.3.2, forces the operator to handle the full weight of the grille guard when it is being lowered or raised. Both S&V Manufacturing and the vehicle operators find this to be problematic as the grille guard must be lowered in order to gain access to the engine, which can be required multiple times a day. Larger semi-trailers require oil changes and engine checks every one to two days, meaning that raising and lowering the grille can be a daily occurrence for the operators. There are also unforeseen engine issues like breakdowns on the side of the road where operators are forced to handle the grille guards in remote areas. This could result in potentially dangerous conditions such as icy or uneven roads. Due to the weight of the grille guards as well as the risk to operator safety, S&V Manufacturing feels that this is a problem that needs to be addressed.

The eye bolt method of locking the grille guard in the raised position, described in section 1.3.2, is also an area of concern to S&V Manufacturing. Since the tow pin, and both eye bolts are completely removable, they are frequently misplaced by the vehicle operator.

1.3.4 Marketplace

While there are many companies that design and manufacture grille guards, only two major companies offer some form of lift assist for their products. Herd North America and Magnum Trailer & Equipment Inc. both offer a version of a lift assist and latch mechanism for their respective grille guard products. There is limited public information for either of these products. The team reached out to both companies for further information on the products they currently offer however we did not receive a response.

The information that was available on the two competing products outlined below helped with the concept generation and selection for both the lift assist and the latch mechanism designs. This information helped by demonstrating what technology is currently being used, the features of the current designs that appeal to customers, and, more importantly, how these products can be improved. This information was also used in benchmarking our final design.

1.3.4.1 Herd Lift Assist

The Herd Lift Assist, designed and manufactured by Herd, claims to reduce the grille's weight by 50-75 [%] resulting in a grille weight as light as 13 [lbs] in the lowered position for certain models [6]. The Herd Lift Assist is available for all Herd grille models and is currently patent-pending. Herd does have a Herd Lift Assist Retrofit Kit Installation Guide for its Defender series of grille guards on its website [7]. This installation guide provides some insight into Herd's designs. The retrofit lift assist consists of two modules attached to the bottom side of the mounting bracket. Each module, as shown in Figure 3, consists of two torsion springs.

Figure 3: Herd's Retrofit Lift Assist for Defender Series Grille Guards [7]

One end of each spring is seated in a retaining hole on the side of the module, while the other free end rests under the bottom end of the grille guard. The springs deform as the grille guard is lowered, which results in the reduction of the grille's weight. The system is bolted to the mounting bracket using three ⅜ [in.] fully threaded hex bolts which allow for easy installation [7]. The springs are secured to the spring hanger subassembly using ⅛ [in.] hair pins which allows for spring replacement if repairs are required. It is important to note that this is only the retrofit design and that the standard Herd Lift Assist could operate differently.

Herd also offers an optional Slam Latch to provide easy locking and unlocking of their grille guards [8]. The Slam Latch consists of a spring-loaded mechanism bolted to the top inside surface of the grille guard and two hook ramps welded to the top face of the mounting bracket. The mechanism has two spring-loaded hooks which are controlled by a handle. When the grille guard is in the raised position, the two spring-loaded hooks are seated behind the hook ramps, securing the grille guard to the mounting bracket. When the Slam Latch handle is pulled, the two spring-loaded hooks disengage the hook ramps and the grille guard can be lowered. Figure 4 shows Herd's Slam Latch on a Herd Defender series grille guard. For reference, the grille guard in Figure 4 is in the lowered position.

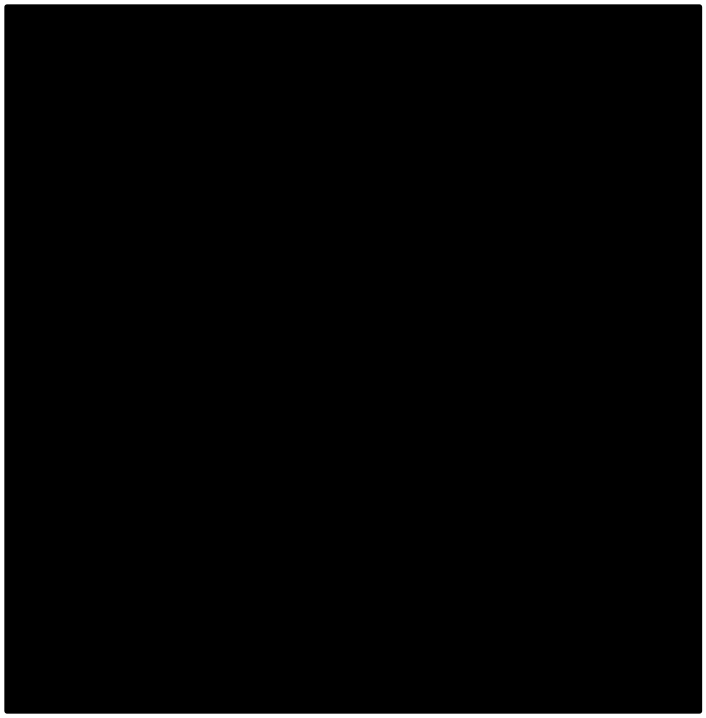


Figure 4: Herd Slam Latch Mounted to a Defender Series Grille Guard [9]

1.3.4.2 Titan 2000 Easy Lift

Magnum Trailer, the company who offers the Titan 2000 Easy Lift, is S&V Manufacturing's second major competitor. Magnum claims the Titan 2000 Easy Lift will reduce the grilles weight by up to 95 [%] when in the lowered position [10]. Magnum also offers a latch system similar to the Herd Slam Latch called the Magnum Quick Latch [10]. Magnum Trailer provides limited information for their Easy Lift and Quick Latch systems. The team was unable to find any relevant images, designs, or specifications for either product.

2 Constraints and Specifications

This section provides a brief overview of the design constraints and target specifications for the lift assist and latch mechanism designs. All constraints and target specifications mentioned in this section were emphasized by S&V Manufacturing throughout the design process. The target specifications for the lift assist and latch mechanism design were based on the customer needs outlined in Appendix A.

2.1 Lift Assist Mechanism

The dimensions of the lift assist mechanism are the major constraints of the design. The lift assist mechanism must be compatible with bracket widths ranging from 808 [mm] to 1150 [mm], and be no less than 305 [mm] to the ground [3]. The lift assist mechanism cannot impede any of the current functionality of the grille guard. Besides the physical dimensions of the lift assist mechanism there are no other set constraints. However, all other lift assist parameters were given marginal values, and ideal values. The marginal values represent the minimum acceptable performance that the client will still consider successful, while the ideal value represents performance that exceeds the customers expectations.

Each specification has been noted with the preferred or given unit of measurement, a marginal value, and an ideal value. The specifications for the lift assist mechanism are shown in TABLE I.

TABLE I: LIFT ASSIST SPECIFICATIONS OUTLINED BY THE CLIENT [3]

Metric	Units	Marginal Value	Ideal Value
Weight of the grille in the lowered position	lbf	20	~13
Weight of the lift assist mechanism	lbs	40	20 to 30
Operating temperature	°C	-30 to +40	-40 to +50
Width of lift assist mechanism	mm	808 to 1150	750 to 1200
Diameter of the hinge point hole	mm	20	20
Time to lower/raise the grille	secs	30	15
Clearance from ground to the mechanism	mm	>305	>305
Cycles until failure	cycles	>365	>1000
Grille guard weight	lbs	225	250
Cost to the customer	$ CAD	~300	Included in current price
Maintenance cost	$ CAD	Minimized	0
Manufacturing cost	$ CAD	~300	Minimized
Time to perform maintenance	mins	15	0
Time in between scheduled maintenance	months	4	Infinite

2.2 Latch Mechanism

There are no physical constraints for the latch mechanism however, all latch parameters were given marginal values, and ideal values. The marginal values represent the minimum acceptable performance that the client will still consider successful, while the ideal value represents performance that exceeds the customers expectations.

Each specification has been noted with the preferred or given unit of measurement, a marginal value, and an ideal value. The specifications for the lift assist mechanism are shown in TABLE II.

TABLE II: LATCH SPECIFICATIONS OUTLINED BY THE CLIENT [3]

Metric	Units	Marginal Value	Ideal Value
Weight of the latch mechanism	lbs	10	5
Operating temperature	°C	-30 to +40	-40 to +50
Time to secure/release the grille	secs	5	1
Strength of latch mechanism	lbs	29,000	>29,000
Cycles until failure	cycles	>365	>1000
Grille guard weight	lbs	225	250
Maintenance cost	$ CAD	Minimized	0
Time to perform maintenance	mins	15	0
Time in between scheduled maintenance	months	4	Infinite

It should be noted that there is some crossover between the lift assist and latch mechanism specifications. The cost of the latch mechanism is not shown in TABLE II. The cost to the customer for the lift assist mechanism is shown in TABLE I to be approximately $300 [CAD]. This is the cost given by the client for both the lift assist and latch mechanisms combined. The specification given by the client included the cost for both mechanisms despite only being displayed in TABLE I.

3 Detailed Design

This section outlines the detailed design for the proposed lift assist and latch mechanisms. Both designs are independent of one another and will therefore be discussed separately. Each design description will include an overview of the proposed design, an explanation of important design features, assembly procedure, operation, and cost breakdown. A full explanation of the conceptual design process can be found in Appendix B showing the evaluation method used to select the final lift assist and latch mechanism designs.

3.1 Lift Assist

The single-spring lift assist design was selected as the optimal configuration to meet all the customer's needs. The single-spring design was given the name because it consists of a single torsional spring at each hinge point. The full detailed description of the design for the single-spring lift assist mechanism is outlined in the following sections.

3.1.1 Design Overview

The single-spring lift assist mechanism consists of 16 separate components that aid in the raising and lowering of the Bison Grille Guard models. The design includes of two separate, but identical modules each consisting of eight components. A single module is attached at each mounting bracket hinge point as shown in Figure 5. To simplify the explanation of the proposed design, the overview will discuss a single module of the lift assist design.

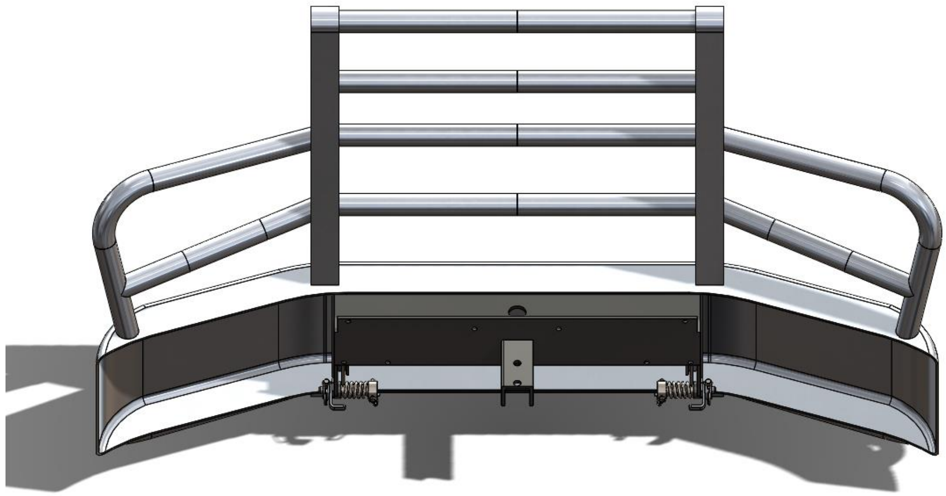


Figure 5: The Bison Grille Guard and Lift Assist Mechanism [11]

The lift assist mechanism utilizes a high strength custom torsional spring whose body is fixed to a steel spring housing by one of its moment arms. This in turn fixes the spring to the grille guard. Additionally, the spring's second radial moment arm is fixed to the body of the mounting bracket.

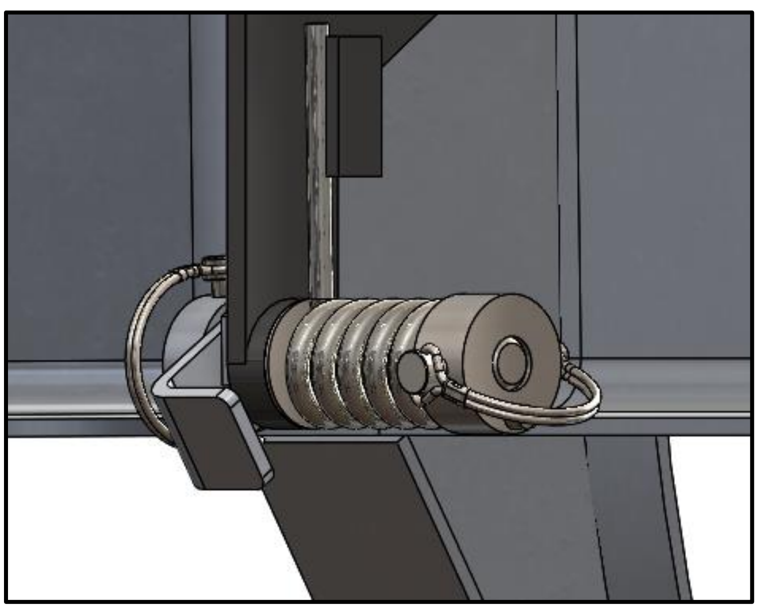


Figure 6: Single Module of the Lift Assist Mechanism [11]

When the grille is lowered, it rotates 90° away from the mounting bracket, which leads to an angular deformation of 90º in the torsional spring. In the lowered position when the angular deformation of the spring 90º, there is 95.5 [J] of energy stored in each spring. This accounts for 45.2 [%] to 58.1

[%] of the energy required to raise the grille guard from the lowered position. A view of the grille guard in both the raised and lowered positions can be seen in Figure 7 and Figure 8.

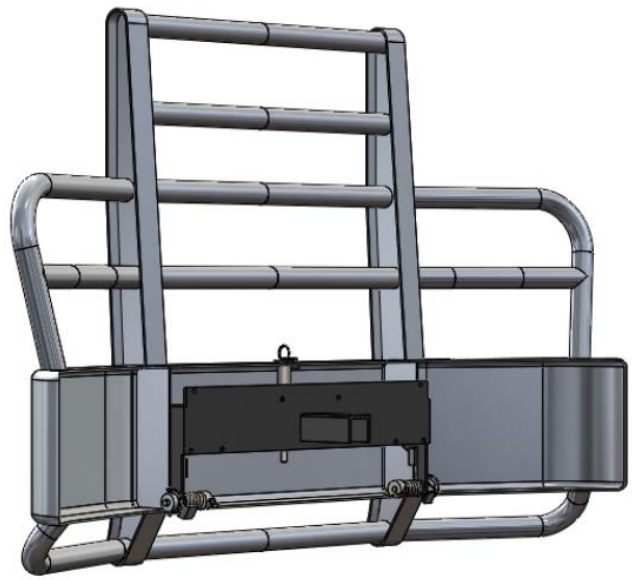


Figure 7: Bison Grille Guard in the Raised Position [11]

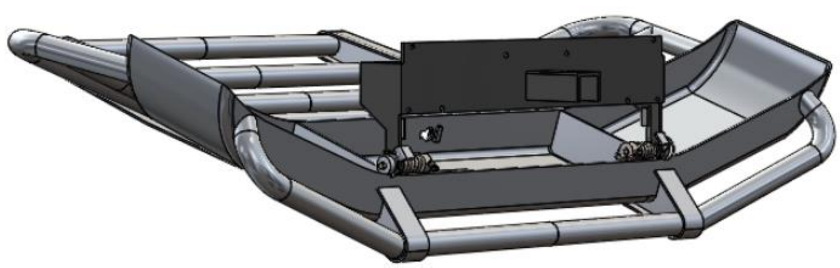


Figure 8: Bison Grille Guard in the Lowered Position [11]

In addition to reducing the force required to raise or lower the grille guard, the lift assist mechanism also acts as a fastener at the hinge point between the mounting bracket and the guard. The proposed design will replace the 20 [mm] bolt previously used to secure the grille guard to the mounting bracket. Figure 9 illustrates this fastening function, with the mounting bracket arm shown in black on the right, and the grille guard arm shown in silver on the left.

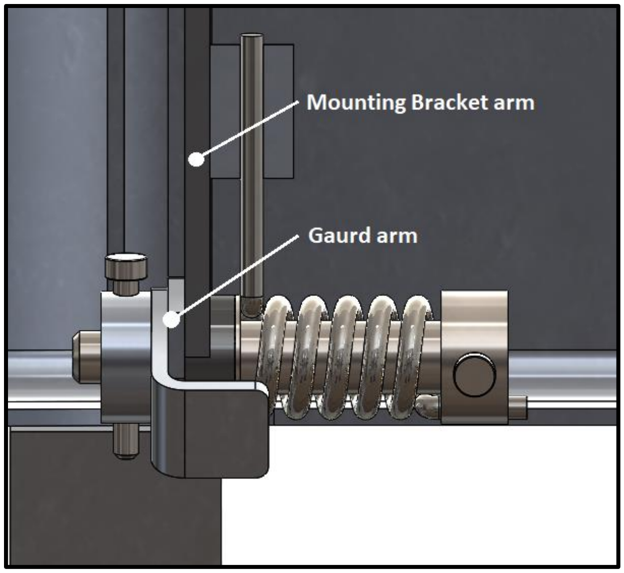


Figure 9: Secondary Fastening Function of the Lift Assist Mechanism [11]

All Bison Grille Guard designs have a common 20 [mm] hole at the hinge point of the grille guard and mounting bracket. The current grille guard set-up used by S&V Manufacturing uses 20 [mm] bolts to secure the assembly. The proposed lift assist mechanism design utilizes 19 [mm] diameter mounting shafts that both the spring housing, and grille guard are fastened to via 18-8 stainless steel locking pins. These mounting shafts secure the entire assembly to the mounting bracket. This shaft is also subjected to the majority of the bending force introduced by the spring, while also transmitting the springs deformation energy between the grille and mounting bracket.

Each hinge point of the grille guard assembly will be outfitted with a lift assist mechanism module. The left and right hinge points will have identical lift assist mechanism modules mirroring one another as seen in Figure 10.

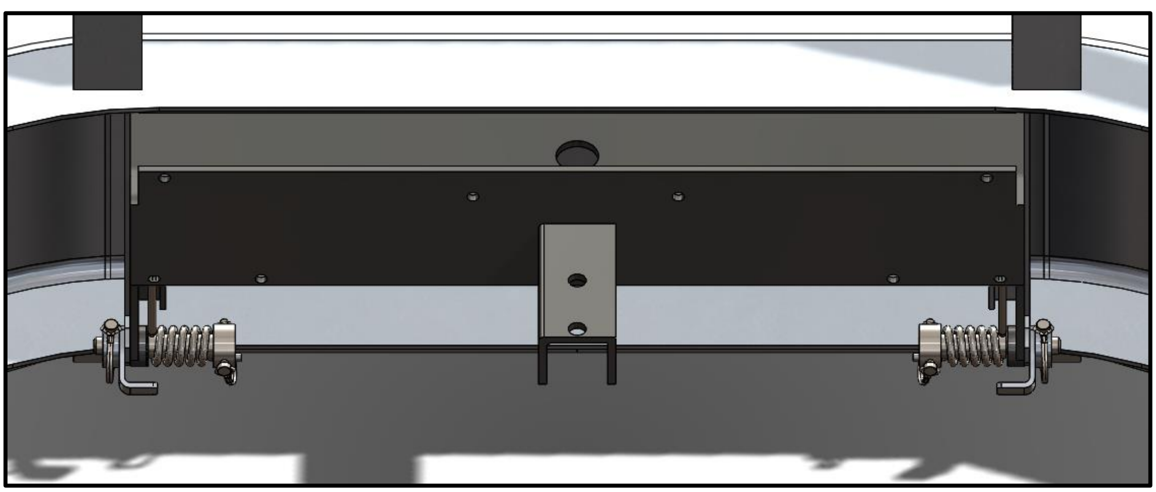


Figure 10: Identical Lift Assist Components at the Left and Right Hinge Points [11]

3.1.2 Detailed Design Features

A breakdown of each component of the lift assist mechanism design is shown in this section. This will include part dimensions, figures, and special considerations for each component.

3.1.2.1 Torsional Spring

The high strength torsional spring utilized for this design, shown in Figure 11, is a custom spring manufactured by Western Spring and Wire Ltd [12].

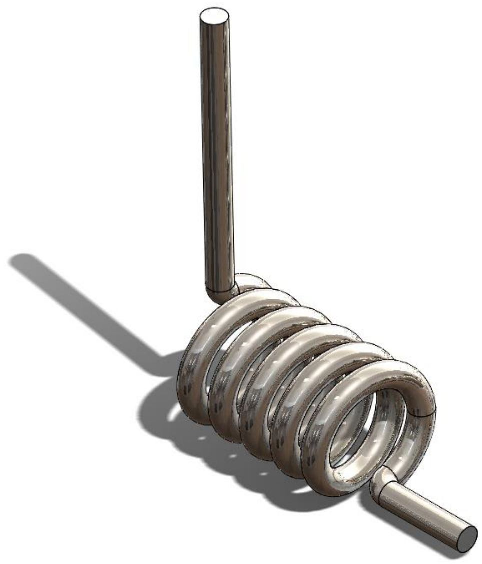


Figure 11: High Strength Torsional Spring [11]

The spring has a wire diameter of 8.7122 [mm] and an inner coil diameter of 30 [mm], which results in an outer coil diameter of 47.4244 [mm]. One end of the spring has a 30 [mm] axial moment arm which mates with the lift assist mechanisms spring housing. The other end of the spring has a 100 [mm] moment arm which mates to the mounting bracket body via an additional mounting bracket body piece called the spring seat. The two moment arms are located 180° away from one another. Figure 12 depicts these moment arms as well as the 30 [mm] inner coil diameter.

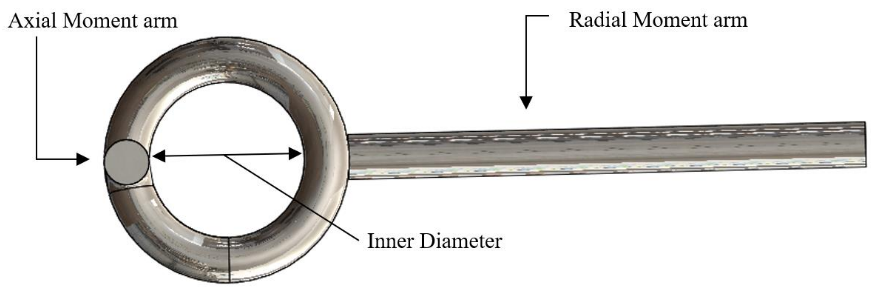


Figure 12: Side View of the Torsional Spring Showing the Moment Arms and Inner Diameter [11]

The rotational pitch of the wire is 12 [mm] with 5.5 body coils. The resulting body length of the spring is 74.7122 [mm]. When subjected to 90° of deformation during the lowering of the grille guard, the spring, which has a torsional spring rate of 1.3519 [Nm/°], yields a torque value of 121.671 [Nm]. The spring is manufactured from commercial grade chrome silicon, whose material properties can be referenced in Appendix G. The weight of the spring is 0.373 [kg]. Preliminary engineering drawings of the spring can be referenced in Appendix F for exact dimensions and feature locations. Each lift assist assembly will utilize two torsional springs, each with a different coil direction. The opposing coil directions will ensure the moment arms of both torsional springs will fit in the desired fastening locations at each hinge point. Full design calculations for the torsional spring can be found in Appendix C.

3.1.2.2 Spring Housing

The spring housing, which holds the spring and fixes it via its axial moment arm, is made from 6061-T6 aluminum as seen in Figure 13.

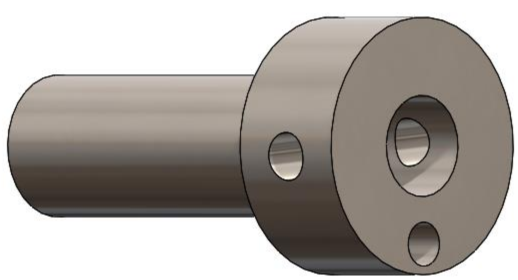


Figure 13: Spring Housing [11]

The spring housing has an inner diameter of 20 [mm], an outer diameter of 28 [mm], and a body length of 76 [mm]. The shoulder of the spring housing, which mates with both the axial arm of the spring and connects the spring housing to the 20 [mm] diameter mounting shaft, has an outer diameter of 50 [mm] and a length of 20 [mm]. The coincident holes in which the anti-rotational locking pin is placed is 0.375 [in.] in diameter. The hole for the axial moment arm of the spring is 8.2 [mm] in diameter. They are located 90° away from one another when referenced from the shoulder end of the spring housing.

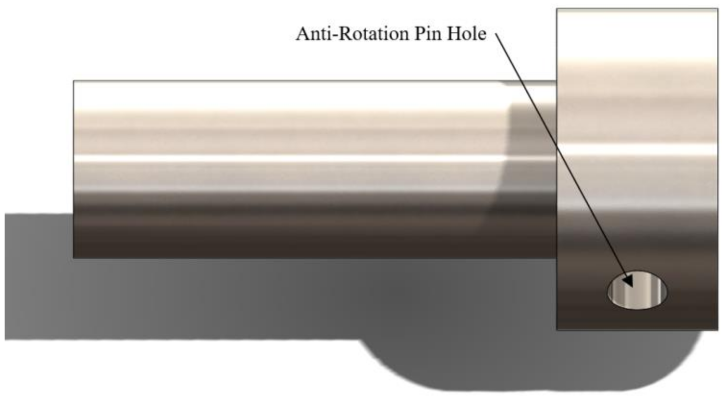


Figure 14: Side View of the Spring Housing, Showing the Locking Pin Hole [11]

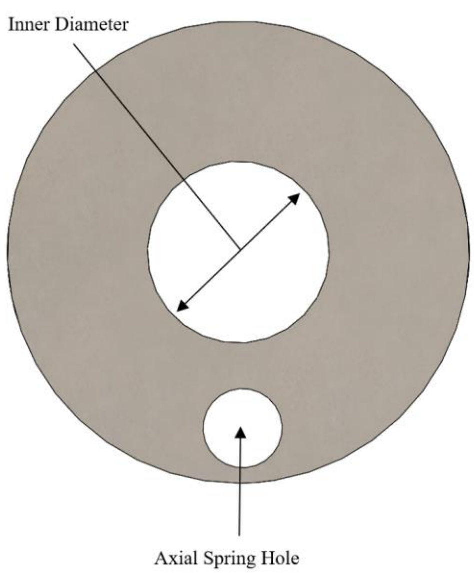


Figure 15: End View of the Spring Housing Showing the Axial Spring Arm Hole [11]

The material properties for this component can be referenced in Appendix G, while preliminary drawings can be referenced in Appendix F, for all dimensions and their locations. It is also worth noting that a steel washer positioned between the mounting bracket arm and spring housing. This washer is made from 4130 steel with an inner diameter of 21 [mm] and an outer diameter of 45 [mm]. Material properties for the washer can also be referenced in Appendix G, while engineering drawings can be referenced in Appendix F.

3.1.2.3 Mounting Shaft

The ¾ [in.] mounting shaft is utilized to transmit the deformation energy from the spring to the grille guard. It is the structure which the spring housing and spring are mounted too. It also acts as the fastener for securing the grille guard to the mounting bracket. A model of the mounting shaft can be seen in Figure 16.

Figure 16: 3/4 [in.] Mounting Shaft [11]

The ¾ [in.] mounting shaft is 6.50 [in.] in length. Two ⅜ [in.] holes are drilled into the shaft after purchase. One hole is centered 12.70 [mm] from the end of the shaft while the other is centered 20.25 [mm] away from the opposite end. The two holes are drilled 90° away from one another. The shaft material is 4140 steel, whose material properties can be referenced in Appendix G. Preliminary engineering drawings for the ¾ [in.] mounting shaft can also be referenced for exact dimensions and locations in Appendix F.

3.1.2.4 Locking Pin

The 18-8 stainless steel locking pins with rounded wire retainers used in the design can be seen in Figure 17. Two of these high strength pins are utilized to lock the ¾ [in.] diameter shaft into orientation with the grille guard, while also locking the spring housing to the mounting shaft.

Figure 17: 18-8 Stainless Steel Locking Pin [11]

The technical drawings, provided by McMaster-Carr can be referenced in Appendix F. The material properties for 18-8 stainless steel can be found for reference in Appendix G.

3.1.2.5 Additional Components

In addition to the features mentioned above, there are two components that would need to be added onto S&V Manufacturing's current Bison Grille Guard designs. Both components would be welded on following the current S&V Manufacturing assembly procedures for the grille guard and mounting bracket.

The first additional component would be the spring seat which would be added onto the inside of the mounting bracket hinge arm. Its purpose is to resist the rotation of the spring, allowing the torsional spring to deform, and aid in reducing the force required to raise and lower the grille guard. The lift assist mechanism and mounting bracket can be seen in Figure 18 and Figure 19 with and without the spring seat.

Figure 18: Lift Assist Mechanism and Mounting Bracket without the Spring Seat [11]

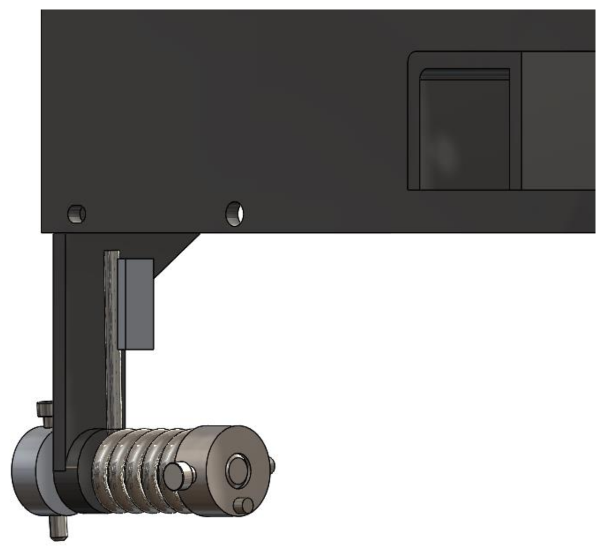


Figure 19: Lift Assist Mechanism and Mounting Bracket with the Spring Seat [11]

The spring seat, which can also be seen by itself in Figure 20 is critical to the design, as its positioning allows for the torsional spring to be subjected to a closing force, rather than an opening force when deformed.

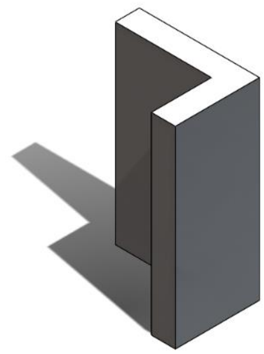


Figure 20: Spring Seat Design [11]

This is important as torsional springs only work properly when closing. It is made from ¼ [in.] 44W plate steel which is consist with the material currently used by S&V Manufacturing for their mounting brackets. Its material properties can be referenced in Appendix G. The preliminary engineering drawings for the spring seat are available for reference in Appendix F.

The second additional component, which will be added onto the grille portion of the Bison Grille Guard assembly, is the grille pin housing, which can be seen in Figure 21.

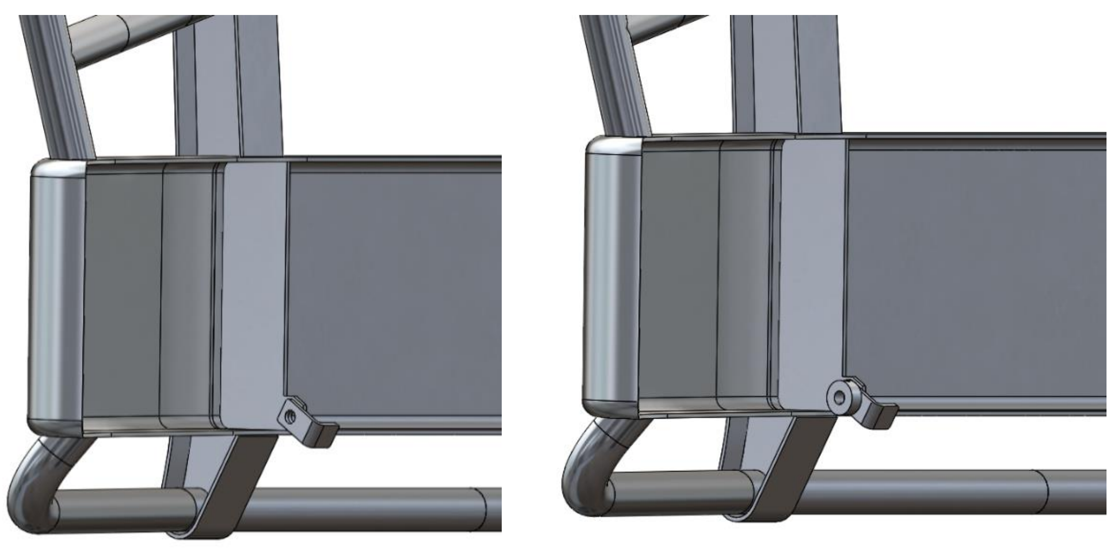


Figure 21: Grille Guard with and without the Mounting Shaft Housing [11]

The grille guard mounting shaft housing is made from the same material as the grille guard, 6160 T6 aluminum. This component allows for the shaft and spring housing to be locked into the same rotation as the grille guard. The mounting shaft housing is 20 [mm] thick with a 50 [mm] outer diameter and a 20 [mm] inner hole diameter. It has a ⅜ [in.] hole centered on its side going all the way through its body, which would house one of the locking pins. The material properties for this component are available for reference in Appendix G. Preliminary engineering drawings for the mounting shaft housing are also available in Appendix F.

3.1.3 Numerical Analysis Results

Two separate finite element analysis simulations were utilized to verify the lift assist mechanism design. The analysis simulated the stresses on the lift assist mechanism while the grille guard was in the lowered position. This situation produces the highest stress state for the design. The first simulation provided proof that the locking pins do not fail during operation as well as showed the general stress distribution in the assembly. Figure 22 shows the general stress distribution results from the first finite element analysis simulation.

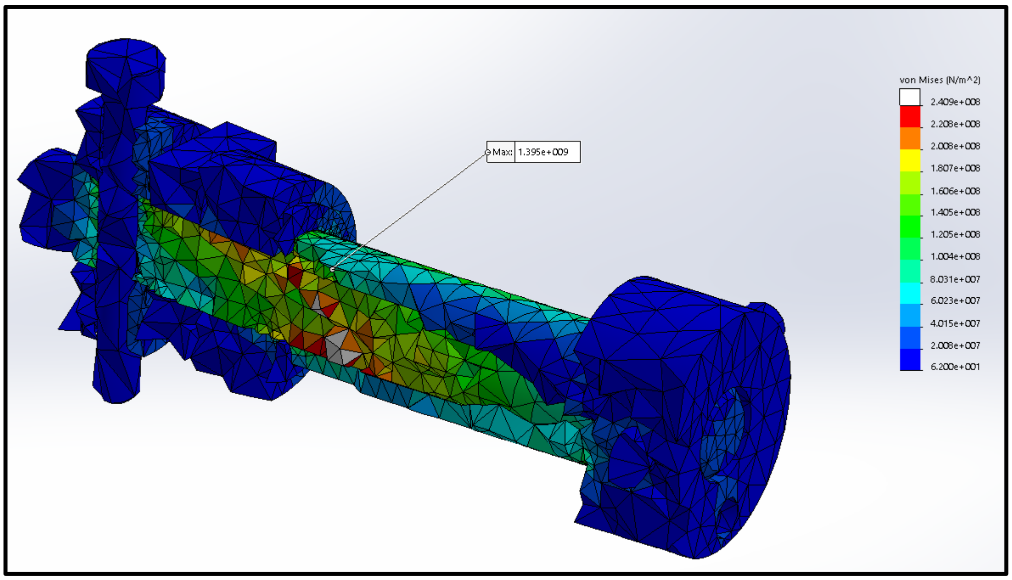


Figure 22: General Stress Distribution from the First Simulation [11]

The first simulation showed a high stress area in the assembly, located at the contact point between the mounting bracket, washer, and spring housing. This area was further investigated using a second simulation which utilized manual mesh control to increase the precision of the mesh in the area of interest. The general stress distribution from the second simulation can be seen in Figure 23.

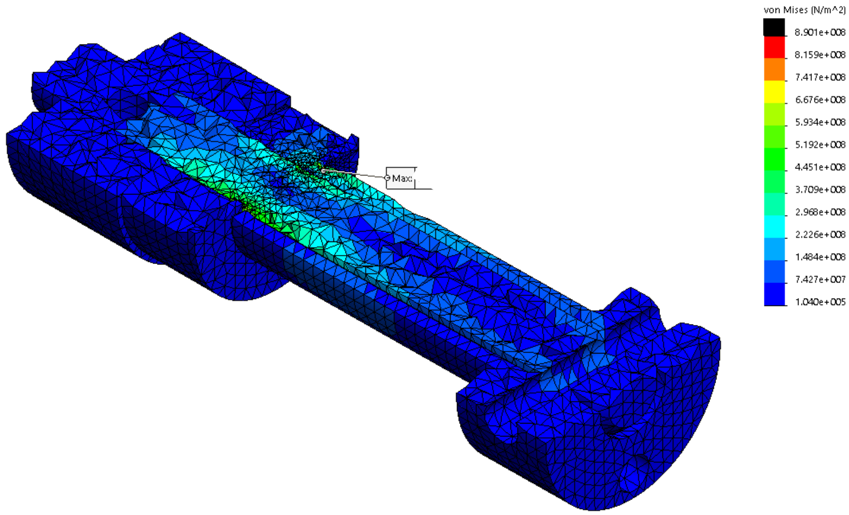


Figure 23: General Stress Distribution from the Second Simulation [11]

The second simulation found high stresses in the same location as the first simulation. The maximum stress in this region was found on the mounting bracket, with a value of 558 [MPa]. This maximum stress exceeds the mounting bracket yield strength of 303 [MPa] [13]. The material used for the mounting bracket, 44W steel, is a very soft steel. The team recommends that the material around the hinge point of the mounting bracket be changed to a material with a minimum yield strength of 600 [MPa]. There is evidence that the stresses calculated by the simulations are higher than what would be observed in reality. Therefore, a sample lift assist mechanism could be manufactured, and practical stress testing could be performed as a proof of concept to avoid having to change the material of the mounting bracket.

The finite element analysis proves that stress throughout the rest of the lift assist design is in an acceptable range. An in-depth discussion and explanation of the finite element simulations and results are available for reference in Appendix D.

3.1.4 Assembly

It is critical to the client that this design be easy to assemble, as it will need to be assembled by S&V Manufacturing in their warehouse, as well as in the field by some vehicle operators.

The following assembly instructions assume the mounting bracket has already been attached to the vehicle. The instructions on how to fasten the mounting bracket to the vehicle are excluded from this report as it is information that is already provided by S&V Manufacturing.

Step 1: Position the grille and mounting bracket

The grille guard needs to be positioned in such a way so that the 20 [mm] hinge point holes on the grille guard are in line with the 20 [mm] holes on the mounting bracket. The grille guard must be in its raised position, or how it would sit on the mounting bracket when the vehicle is in operation, as shown in Figure 24.

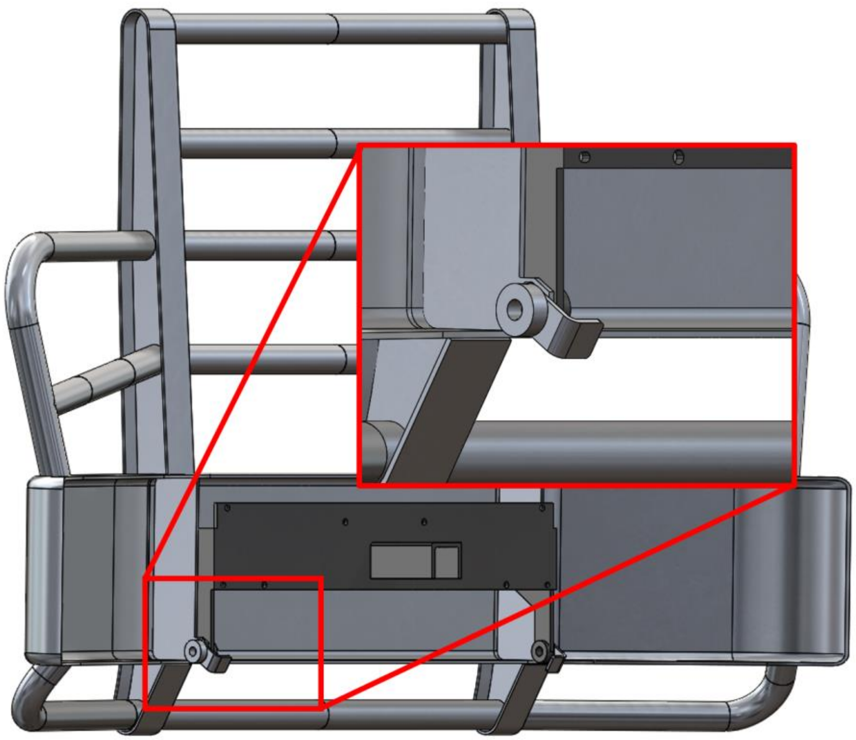


Figure 24: Step 1 of the Lift Assist Assembly Procedure [11]

Step 2: Insert the ¾ [in.] mounting shafts

The ¾ [in.] mounting shafts must be inserted through both sets of holes in the grille guard and mounting bracket, at both the left and right hinge points. It is important that the hole, which is positioned 20.25 [mm] from the end of the mounting shaft, is facing the outside of the grille guard assembly. This step is shown from two different vantage points in Figure 25 and Figure 26.

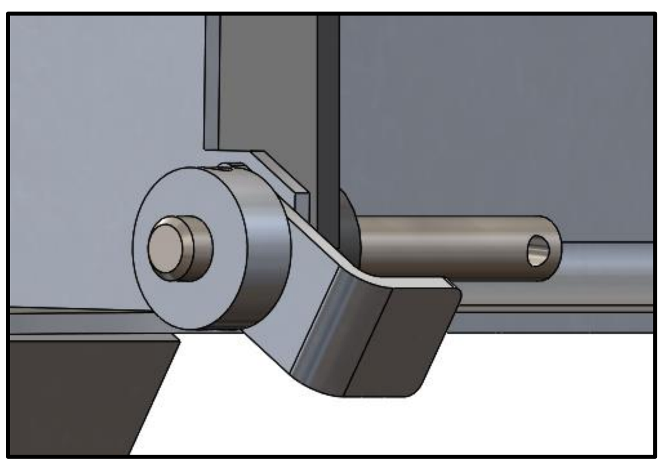


Figure 25: Step 2 of the Lift Assist Assembly Procedure [11]

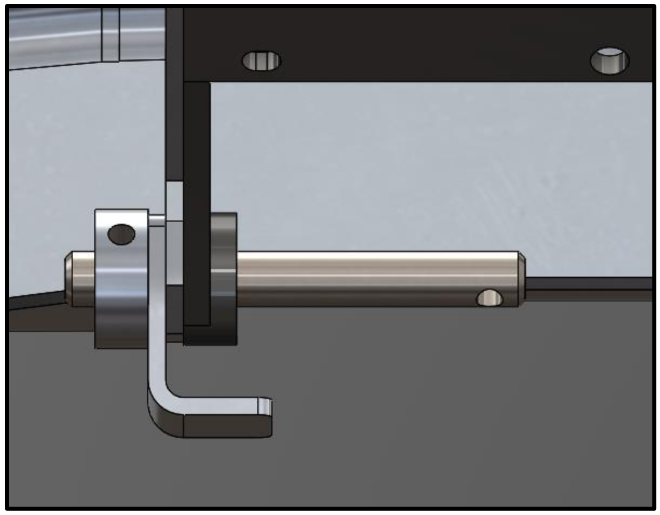


Figure 26: Step 2 of the Lift Assist Assembly Procedure [11]

Note, that in Figure 26 the hole at the top of the mounting shaft housing should be in line with the hole in the mounting shaft. Visual inspection for this alignment is sufficient. This step must be completed at both hinge points before moving onto Step 3.

Step 3: Insert the first assembly pins

The first assembly pin needs to be inserted through both the hole in the mounting shaft housing, and the hole in the mounting shaft itself. Assure that the pin is locked on both hinges before moving onto Step 4. Step 3 can be seen in Figure 27 and Figure 28.

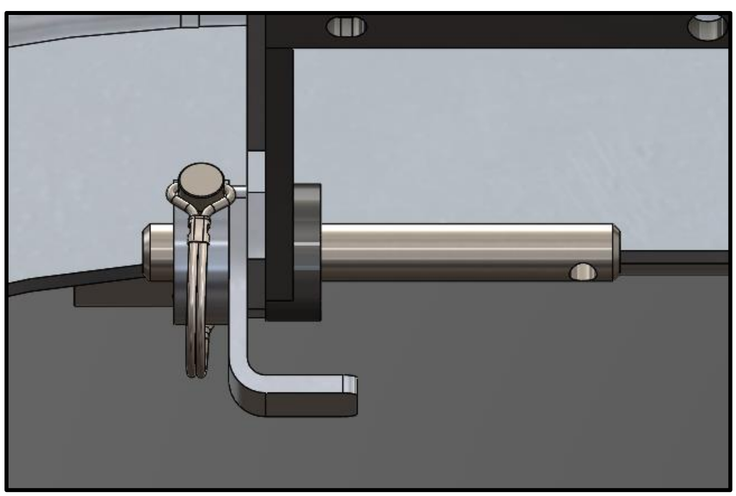


Figure 27: Step 3 of the Lift Assist Assembly Procedure [11]

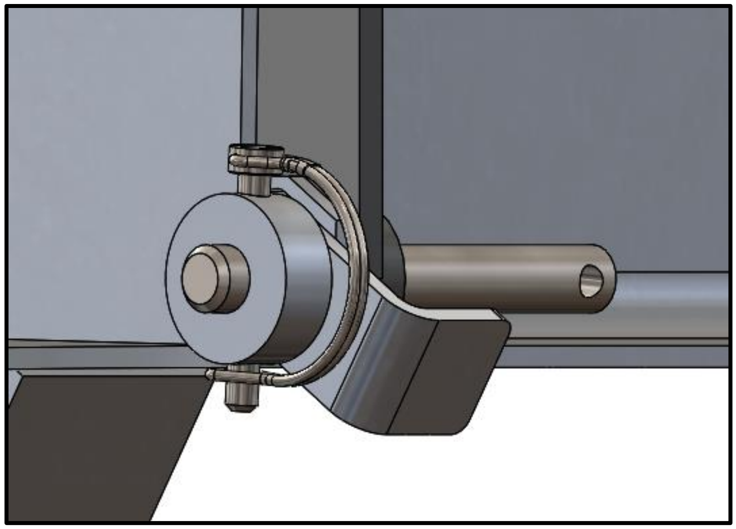


Figure 28: Step 3 of the Lift Assist Assembly Procedure Showing the Locked Pin [11]

Step 4: Inset the washers on to the mounting shafts

Inset the washer over the ¾ [in.] mounting shaft, all the way to the base of the shaft at the mounting bracket hinge point as shown in Figure 29. This step should be performed for both lift assist modules.

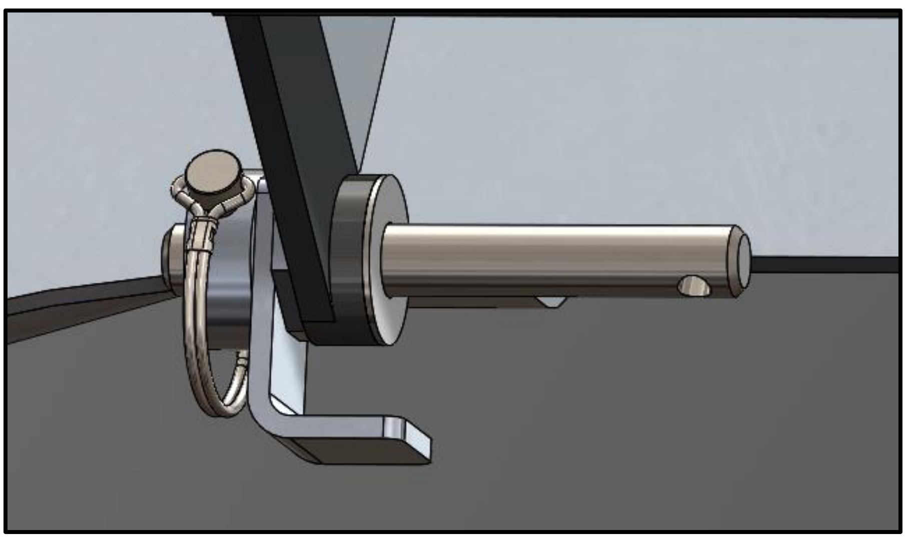


Figure 29: Step 4 of the Lift Assist Assembly Procedure [11]

Step 5: Insert the springs and spring housings

Next, the spring housing with the spring already mounted, needs to be inserted over the ¾ [in.] mounting shaft. Figure 30 shows the mounting of the spring housing without the spring as a visual aid.

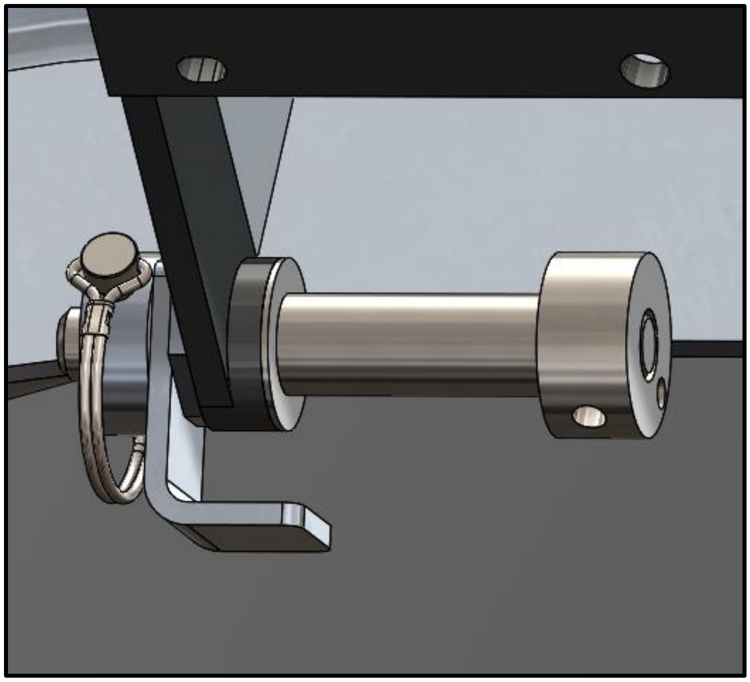


Figure 30: Step 5 of the Lift Assist Assembly Procedure Showing only the Spring Housing [11]

Assure the proper rotation of spring before mating the spring and the spring housing, as the left and right hinge points are mirrored. In Figure 31, the spring properly coils “into the page” while being properly seated in the mounting brackets spring seat.

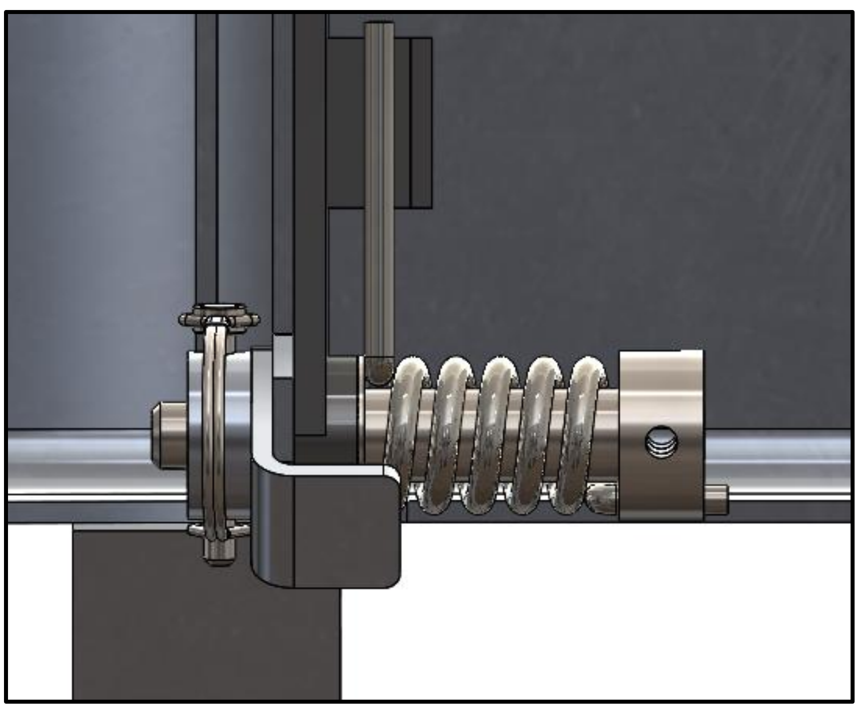


Figure 31: Step 5 of the Lift Assist Assembly Procedure [11]

Note, the second hole on the ¾ [in.] mounting shaft will line up with the pin holes on the spring housing. This is the only possible configuration of the assembly if the axial moment arm of the spring is properly mated to the spring housing.

Step 6: Insert the remaining pins

At this point the remaining locking pins can be added to complete the assembly. Assure the pins are locked at both hinge points. The lift assist mechanism should now be fully assembled and ready for operation as seen in Figure 32, Figure 33 and Figure 34.

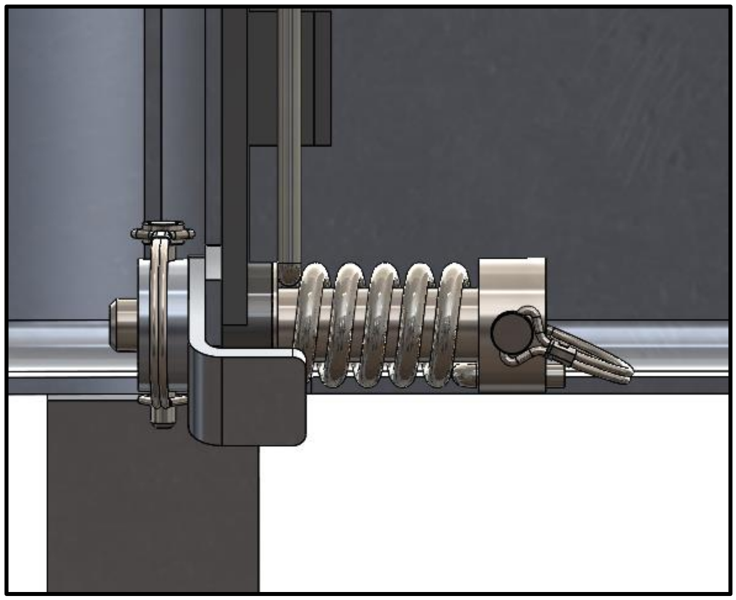


Figure 32: Step 6 of the Lift Assist Assembly Procedure [11]

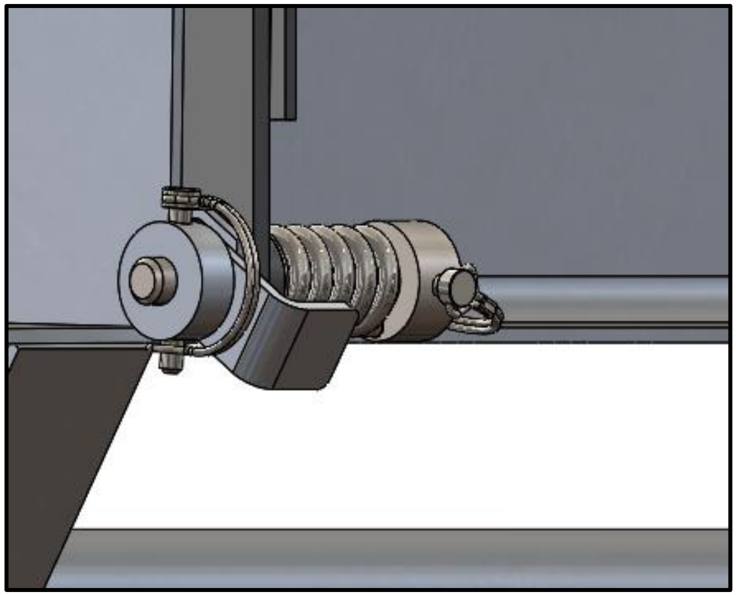


Figure 33: Step 6 of the Lift Assist Assembly Procedure [11]

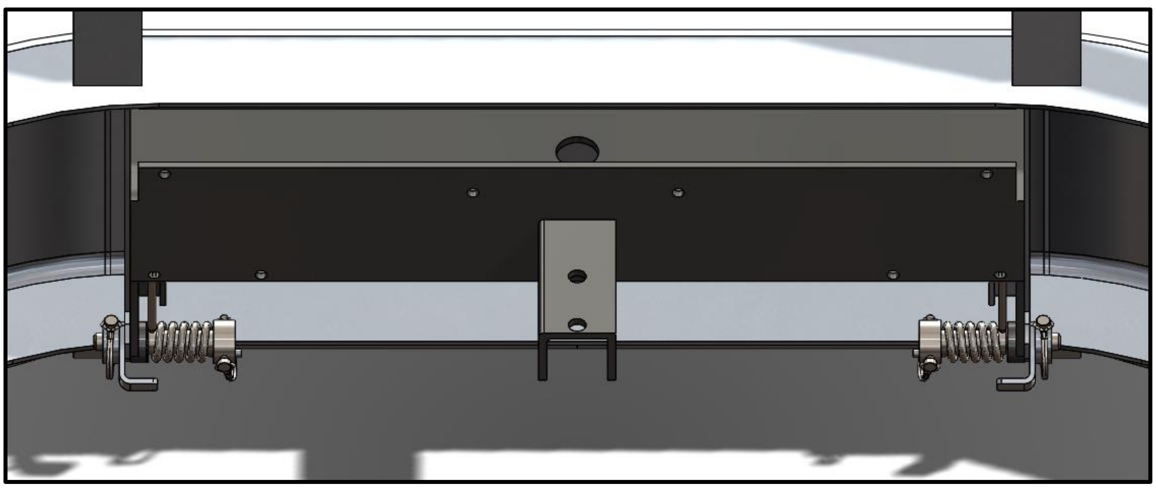


Figure 34: Fully Assembled Lift Assist Mechanism [11]

3.1.5 Operation

In order to operate the lift assist mechanism, assure that the vehicle in which the Bison Grille Guard is mounted to is parked safely away from traffic, and that the engine is off. Assure that the road conditions in front of the vehicle are safe, free from ice, liquid, or debris.

To utilize the lift assist mechanism to lower the Bison Grille Guard, simply grasp the front of the grille, as close to chest height as possible and pull it away from the front of the vehicle, stepping backwards. The grille can be seen in the raised position in Figure 35.

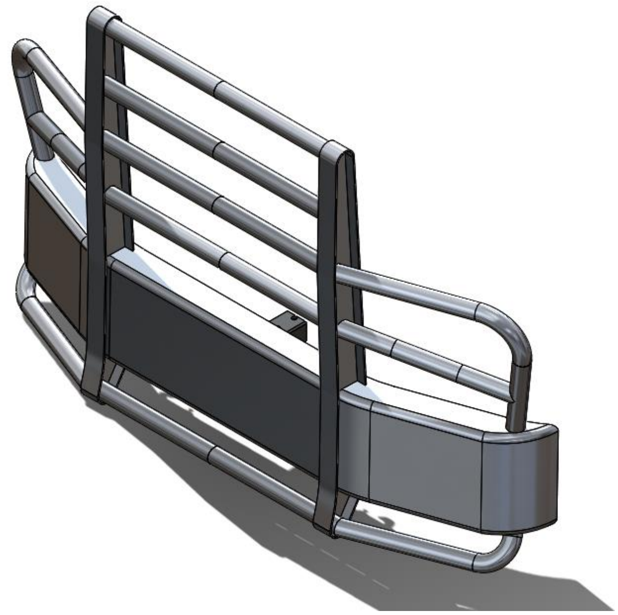


Figure 35: The Bison Grille Guard in the Raised Position [11]

As the grille guard begins to move away from the front of the vehicle, it should also begin to swing down towards the ground. It should be lowered using the legs, rather than the back as much as possible to avoid injury. The grille guard should continue to travel towards the ground until it stops in the bottom position, illustrated in Figure 36.

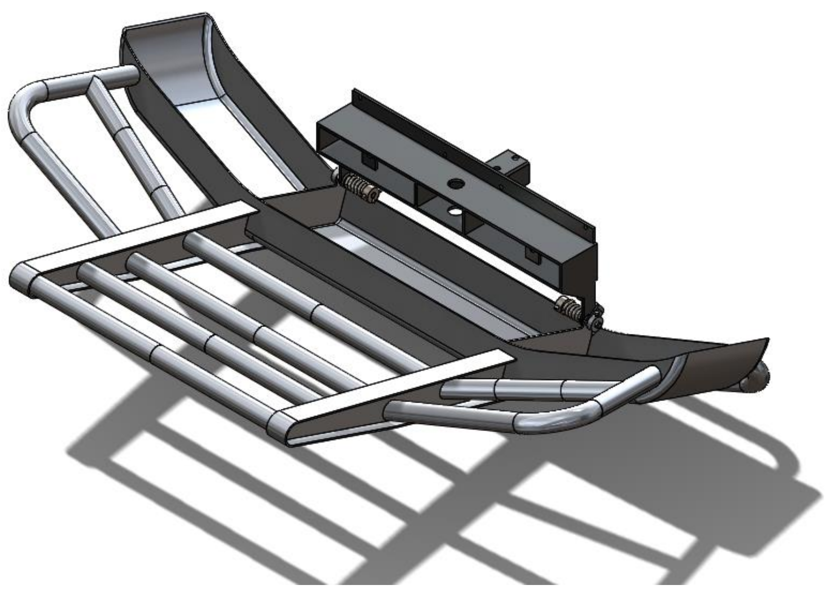


Figure 36: The Bison Grille Guard in the Lowered Position [11]

In order to raise the grille, grasp the top bar with both hands, bent down with a flat back. Lifting with the legs, raise the grille guard up and towards the front of the vehicle. Continue to raise the grille in this manner, keeping both hands on the grille at all times. The grille should never require more than 30 [lbf] of human force to raise or lower for any model of S&V Manufacturing's Bison Grille Guards.

3.1.6 Cost

This section outlines the cost associated with the design and manufacturing of the single-spring lift assist mechanism. This cost includes the raw materials, procurement, manufacturing, assembly, and maintenance of the lift assist system. TABLE III shows the breakdown of the manufacturing costs associated with the design.

TABLE III: SINGLE-SPRING LIFT ASSIST COST BREAKDOWN

Part #	Part Name	Description	Quantity	Unit	Unit Cost	Cost
LA002	RS Spring	1.35 Nm/º Torsional Spring CCW Coiling Direction	1	EACH	$94.26 [12]	$94.26
LA002	LS Spring	1.35 Nm/º Torsional Spring CW Coiling Direction	1	EACH	$94.26 [12]	$94.26
LA003	Spring Housing	20 mm Inner Diameter 28 mm Outer Diameter 50 mm Diameter Shoulder	2	EACH	$200.95 [14]	$401.9
LA004	Mounting Shaft	6.5" Long ¾" High-Strength 4140 Steel Alloy Rod	2	EACH	$7.10 [15]	$14.20
LA005	Locking Pin	⅜" 18-8 Stainless Steel Lock Pin with Wire Retainer	4	EACH	$6.59 [16]	$26.36
LA006	Spring Seat	¼" 44W Plate Steel L-Shaped Bracket	2	EACH	$0.00	$0.00
LA007	Grille Pin Housing	20 mm Thickness 20 mm Inner Diameter 20 mm outer Diameter	2	EACH	$0.00	$0.00
LA008	Washer	38 mm Outer Diameter 21 mm Inner Diameter	2	EACH	$0.97 [17]	$1.94

Although some parts, such as the pins and washers, are purchased in bulk, the costs shown in TABLE III are for a single lift assist assembly. Costs are expected to decrease as the purchase quantity is increased. TABLE IV shows the expected cost per assembly for an assembly quantity of 100 lift assist units.

TABLE IV: SINGLE-SPRING LIFT ASSIST COST BREAKDOWN FOR 100 ASSEMBLY UNITS

Part #	Part Name	Description	Quantity	Unit	Unit Cost	Cost
LA002	RS Spring	1.35 Nm/º Torsional Spring CCW Coiling Direction	1	EACH	$7.19 [12]	$7.19
LA002	LS Spring	1.35 Nm/º Torsional Spring CW Coiling Direction	1	EACH	$7.19 [12]	$7.19
LA003	Spring Housing	20 mm Inner Diameter 28 mm Outer Diameter 50 mm Diameter Shoulder	2	EACH	$148.98 [14]	$297.96
LA004	Mounting Shaft	6.5 Long ¾" High-Strength 4140 Steel Alloy Rod	2	EACH	$4.19 [15]	$8.38
LA005	Locking Pin	⅜" 18-8 Stainless Steel Lock Pin with Wire Retainer	4	EACH	$6.59 [16]	$26.36
LA006	Spring Seat	¼" 44W Plate Steel L-Shaped Bracket	2	EACH	$0.00	$0.00
LA007	Grille Pin Housing	20 mm Thickness 20 mm Inner Diameter 50 mm outer Diameter	2	EACH	$0.00	$0.00
LA008	Washer	38 mm Outer Diameter 21 mm Inner Diameter	2	EACH	$0.97 [17]	$1.94

The total cost for the single-spring lift assist mechanism is $632.92 CAD, which can be reduced to $349.02 CAD per unit if 100 assembly units are produced. These total costs do not include the cost of assembly of the lift assist mechanism. The single-spring lift assist mechanism was designed to be assembled at S&V Manufacturing with limited specialized tooling as outlined in section 3.1.4. Therefore, the assembly costs will be limited to the hourly wage of the employees tasked with assembly. There will also be no maintenance costs associated with the single-spring lift assist mechanism over the one-year warranty of the product.

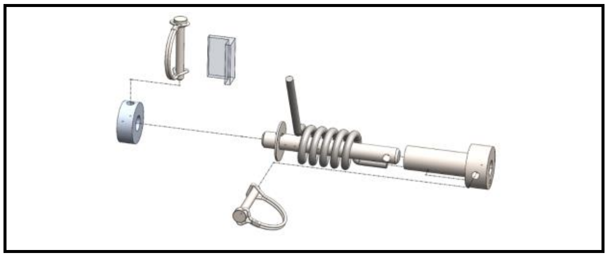
3.1.7 Preliminary Bill of Materials

This section discusses the preliminary bill of materials (BOM) for the lift assist assembly. The BOM includes the part number, description, quantity, procurement type, and the vendor or manufacture the part was sourced from for each component of the design. The vendors and manufactures listed in the BOM are the preferred suppliers based on price, convenience, or recommendation from the client. The vendors in the BOM represent who supplied the stock material or part, while the manufacturer represents who supplied custom manufactured components. If both a vendor and a manufacturer are listed for a certain component, a stock part was purchased and then sent to a manufacturer for additional modifications.

TABLE V: PRELIMINARY BOM FOR THE LIFT ASSIST MECHANISM

Lift Assist Mechanism BOM

Assembly Name: Single-Spring Lift Assist
Assembly #: LA-ASSEM01
Approval Date: 11/26/2017
Part Count: 16
Total Cost: $632.92 CAD



Part #	Part Name	Description	Quantity	Procurement Type	Vendor	Manufacturer
LA002	RS Spring	1.35 Nm/º Torsional Spring CCW Coiling Direction	1	MTS	-	Western Spring and Wire Ltd. [12]
LA002	LS Spring	1.35 Nm/º Torsional Spring CW Coiling Direction	1	MTS	-	Western Spring and Wire Ltd. [12]
LA003	Spring Housing	20 mm Inner Diameter 28 mm Outer Diameter 50 mm Diameter Shoulder	2	MTS	-	Kinetic Machine Works Ltd. [14]
LA004	Mounting Shaft	6.5” Long ¾” High-Strength 4140 Steel Alloy Rod	2	OTS	McMaster-Carr Supply Company [15]	S&V Manufacturing Ltd.
LA005	Locking Pin	⅜” 18-8 Stainless Steel Lock Pin with Wire Retainer	4	OTS	McMaster-Carr Supply Company [16]	-
LA006	Spring Seat	¼” 44W Plate Steel L-Shaped Bracket	2	MTS	-	S&V Manufacturing Ltd.
LA007	Grille Pin Housing	20 mm Thickness 20 mm Inner Diameter 50 mm outer Diameter	2	MTS	-	S&V Manufacturing Ltd.
LA008	Washer	38 mm Outer Diameter 21 mm Inner Diameter	2	OTS	McMaster-Carr Supply Company [17]	-

3.2 Latch

The advanced tow pin latch was selected as the optimal latch design to meet all the customer’s needs. The advanced tow pin design was given the name because it works off the existing tow pin design currently used by S&V Manufacturing. The full detailed description of the design for the advanced tow pin latch mechanism is outlined in the following sections.

3.2.1 Design Overview

The advanced two pin latch mechanism is a design that consists of five separate components that, when assembled, will enable the grille guard to be lock and unlocked in a matter of seconds. The integration of the advanced tow pin and the grille guard assembly can be seen below in Figure 37.

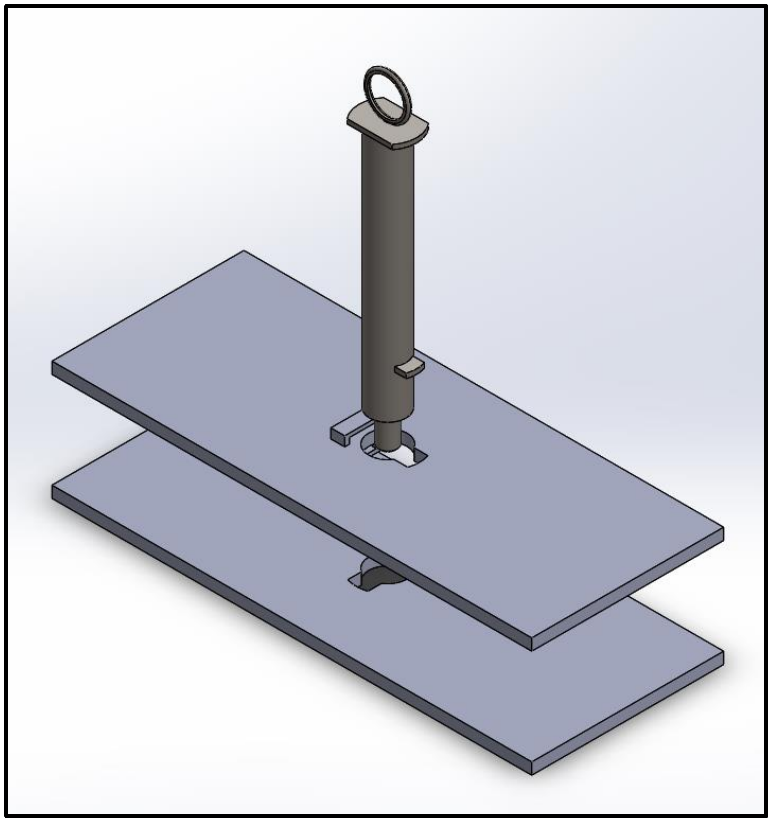


Figure 37: Advanced Tow Pin Assembly [18]

The existing tow pin is intended to be inserted through the holes on the grille guard and mounting bracket to secure the grille guard in the raised position. When the grille guard needs to be lowered, the tow pin is removed, and the eye bolts are loosened. With the suggested modifications, the full removal of the tow pin is no longer required for lowering the grille guard. This will eliminate the potential for the vehicle operator to misplace the tow pin which was a concern outlined by the client. A key is welded on to the existing tow pin design. The two holes on the grille guard and mounting bracket are modified to provide key ways for the tow pin key. Two key seats are designed to hold the pin in place, which are welded on to the top and bottom faces of the grille guard next to the tow pin hole. The details of the key holes and key seats are shown in Figure 38.

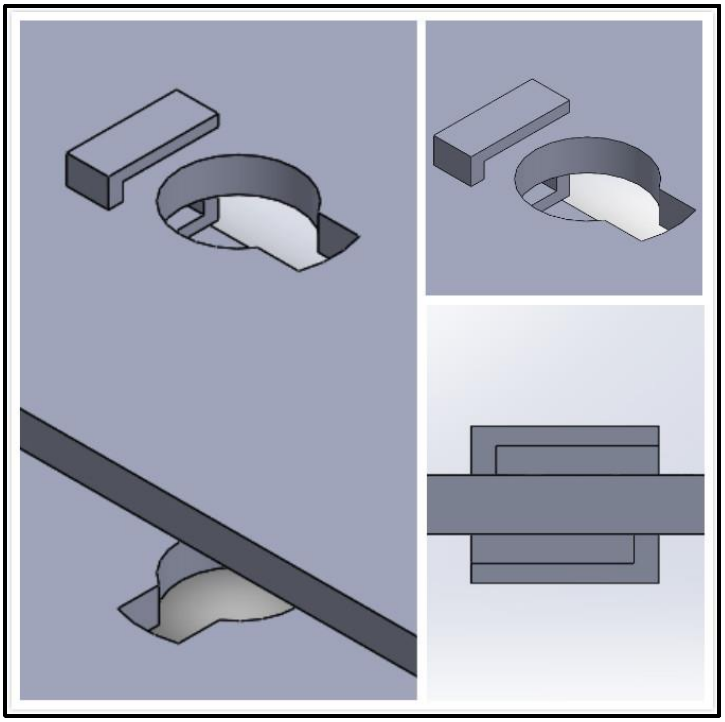


Figure 38: Key Holes and Key Seats [18]

The advanced tow pin will be first inserted through the hole on the grille guard, and rotated 90° to allow the key to be inserted through the hole on the bracket. Once the tow pin is fully inserted, another 90° rotation is needed to lock the key in the key seat. The key seat on the bottom of the grille is used for locking the tow pin when it is in the raised position.

This design also allows for the option of a spring to be added, where one end will be attached to the top of the tow pin and another end is attached to the bottom grille guard bar. By adding this spring, the advanced tow pin can be raised with little to no force supplied by the operator. Furthermore, the spring can also eliminate the potential vibration of the tow pin when it is inserted in the locked position.

3.2.2 Detailed Design Features

A breakdown of each component of the advanced tow pin design is presented in this section. The main three parts of the design include the tow pin keys, the mounting bracket key ways, and the mounting bracket key seats. The breakdown of the components will include part dimensions, figures, and special considerations for each component.

3.2.2.1 Tow Pin Keys

The tow pin features two separate tow pin keys. The top face of the tow pin has a varied diameter to allow it to act as one of the keys. The top face has a maximum diameter of 60.33 [mm] with a minimum diameter of 49.21 [mm]. The second key is located 90º from the maximum diameter of the top face of the tow pin and 86.35 [mm] from the bottom of the pin. The key has a width of 20.00 [mm], a length of 30.16 [mm], and a thickness of 6.35 [mm].

3.2.2.2 Key Ways

The two key ways are cut into the top face of the grille guard and the top face of the mounting bracket. The key ways allow the second tow pin key to pass through when the tow pin is inserted to lock the grille guard. The two key ways are located 90º apart from one another. Both key ways have lengths of 11.11 [mm] and widths of 20.00 [mm].

3.2.2.3 Grille Guard Key Seats

The advanced tow pin design features two key seats attached to the grille guard. One key seat is on the top face of the top plate of the grille guard, while the other key seat is located on the bottom face of the top plate. Both key seats have openings in opposite directions to allow locking in one direction and unlocking in the other. Both grille guard key seats have heights of 6.35 [mm], depths of 11.00 [mm], and widths of 40.00 [mm].

3.2.3 Numerical Analysis Results

Numerical analysis was performed on the advanced tow pin to validate the design and prove that is will be able to handle normal working conditions. The existing tow pin used for the proposed latch design is already a proven component of the current grille guard system and therefore was excluded from the analysis. It was assumed that the tow pin would be strong enough to withstand normal operating forces, and it was neglected from the finite element analysis. The analysis in this section will be focused on the added keys and key seats.

The tow pin key was analyzed for the situation for which it must support itself against gravity. A simulated force of 24.24 [N] was applied to the top face of the tow pin, while the key was fixed. The analysis determined that the maximum stress on the geometry was 1.283 [MPa] which was below the yield strength of the material. A stress concentration was located on the key as predicted. The high stress concentration could be eliminated by adding material to the connection between the tow pin and the key, or removing the sharp corners on the geometry by adding fillets. The finite element analysis results validated our proposed latch design.

The key seats were also analyzed using finite element analysis to ensure they would function as intended. The application of the fixture and load can be seen in Figure 39. An external load of 30.00 [N] was applied to the inner surface of the key seat. The 30.00 [N] simulated the operator pulling on the tow pin while it is in the locked position. The results showed a maximum stress on the key seat of 1.374 [MPa]. The material used has a yield strength of 227.5 [MPa] [19] which shows that our key seat will be able to withstand normal operating conditions.

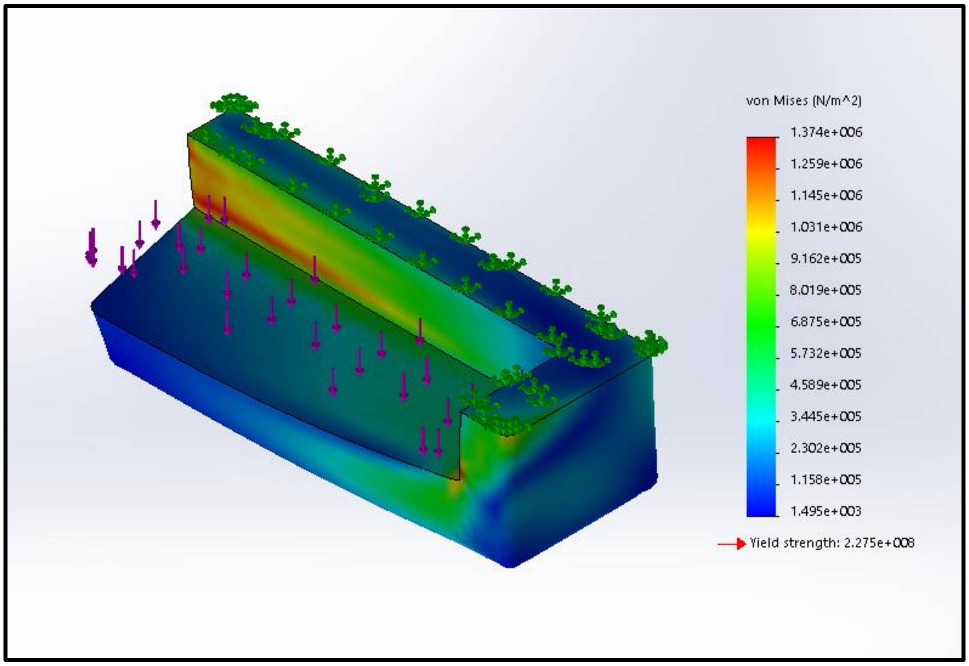


Figure 39: Stress Distribution Plot for the Advanced Tow Pin Key Seat [18]

The full numerical analysis of the advanced tow pin latch mechanism can be found in Appendix E.

3.2.4 Assembly

The assembly for the advanced tow pin latch mechanism is straightforward and requires no specialized tooling. The assembly of the latch mechanism will be done in the S&V Manufacturing workshop with their existing equipment, before being shipped to customers. The assembly of the advanced tow pin can be completed in the four steps listed below.

Step 1: Cut the key ways

Cut a key way with a length of 14.00 [mm], and a width of 24.00 [mm] in the top plate of the grille guard at the location of the tow pin hole. The top edge of the key way should be convex, with a radius of 19.05 [mm]. This key way can is shown in Figure 40.

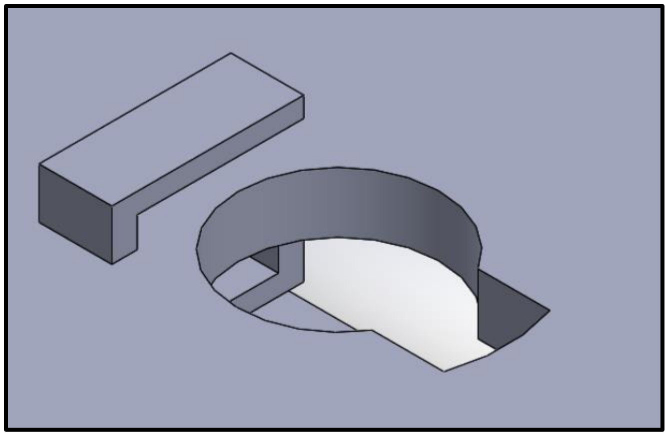


Figure 40: Tow Pin Key Way Located on the Top Plate of the Grille Guard [18]

Cut a second key way, with dimensions identical to the first, on the top plate of the mounting bracket. The second key way should be cut so that it is 90^o away from the key way on the top plate.

This step should be done during the fabrication of the grille guards and mounting brackets at the same time as the holes for the tow pin are being cut.

Step 2: Attach the keys to the tow pin

Using scrap ¼ [in.] 44W steel, cut a key with a length of 11.11 [mm] and a width of 20.00 [mm]. The key should be cut in such a way that the bottom edge of the key sits flush with the tow pin. Meaning, the bottom edge of the key should be concave with a radius of 19.05 [mm].

Once the key is cut to the specified dimensions, it will be welded to the tow pin at the location specified by the preliminary drawings found in Appendix F.

Step 3: Mount the key seats on the grille guard

Using 6061 T6 aluminum, assemble two key seats with the dimensions specified by the preliminary engineering drawings found in Appendix F. Weld one key seat to the top face of the top plate of the grille guard, and one to the bottom face. The openings of the key seats should face in opposite directions as shown in Figure 41.

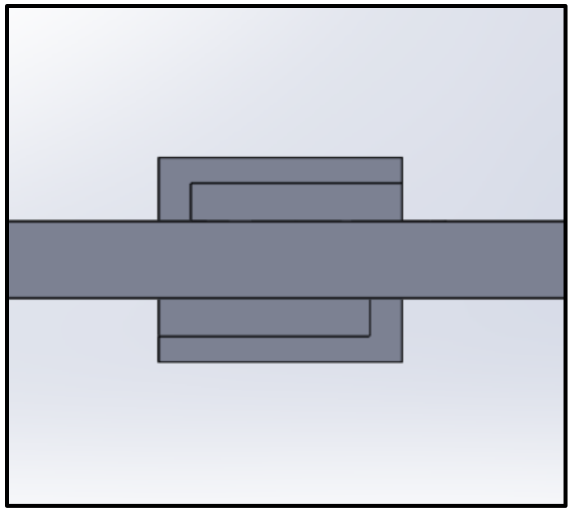


Figure 41: Top Plate of the Grille Guard Showing the Two Key Seats [18]

Step 4: Insert the tow pin

When the grille guard is in the closed position, insert the tow pin so that the key passes through the key way on the top plate of the grille guard. Then, rotate the tow pin 90º so that the key can pass through the second key way on the top plate of the mounting bracket. Once the top face of the tow pin is flush with the top surface of the grille guard, rotate the tow pin so that the top face of the tow pin is in the grille guard key seat as shown in Figure 42.

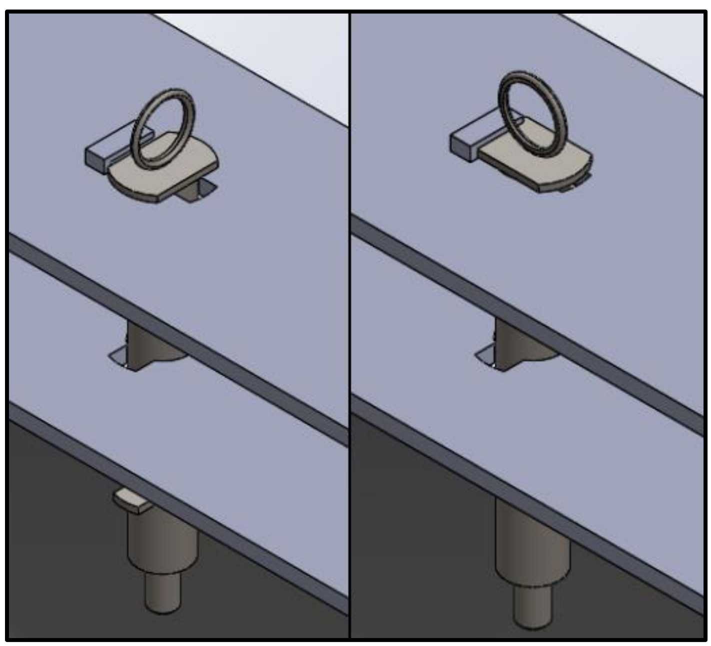


Figure 42: Locking the Tow Pin [18]

3.2.5 Operation

The advanced tow pin design is a slight modification on the existing tow pin currently used by S&V Manufacturing. This will ensure operation of the proposed latch system will be straightforward for the vehicle operators.

To unlock the grille guard from the mounting bracket, turn the tow pin 90^o clockwise to unseat the top tow pin key from the grille guard key seat. Once the key is unseated, pull up on the tow pin until the second tow pin key hits the bottom face of the top plate of the grille guard. Then turn the tow pin an additional 90^o clockwise to lock the second tow pin key into the grille guard key seat located on the bottom face of the top plate of the grille guard. The tow pin is now unlocked from the mounting bracket and is free to be lowered.

To lock the grille guard to the mounting bracket, turn the tow pin 90^o counter clockwise to unseat the second tow pin key from the grille guard key seat. Once the key is unseated, let the tow pin fall until the top face of the tow pin is resting on the top face of the grille guard. Turn the tow pin an additional 90^o counter clockwise to seat the top face of the tow pin in the grille guard key seat. The grille guard is now locked to the mounting bracket.

3.2.6 Cost

This section outlines the cost associated with the design and manufacturing of the advanced tow pin latch mechanism. This cost includes the raw materials, procurement, manufacturing, assembly, and maintenance of the latch system. TABLE VI shows the breakdown of the manufacturing costs associated with the design.

TABLE VI: ADVANCED TOW PIN LATCH MECHANISM COST BREAKDOWN

Part #	Part Name	Description	Quantity	Unit	Unit Cost	Cost
LS01	Key Seat	6.35 mm Height 11.00 mm Depth 40 mm Width	2	EACH	$0.00	$0.00
LS02	Key	30.16 mm Length 20.00 mm Width 6.35 mm Thickness	1	EACH	$0.00	$0.00

It is important to note that the material used to manufacture the keys and key seats used in the advanced two pin design is the same material used for the construction of the S&V Manufacturing mounting brackets. Because of the size of the keys and key seats, they will most likely be able to be manufactured using scrap material from the mounting brackets with negligible added cost to the client. The cost of the tow pin itself is not included in the cost breakdown since it is a part which is already included in the current grille guard assembly.

The total cost of the advanced tow pin latch mechanism is $0.00 CAD.

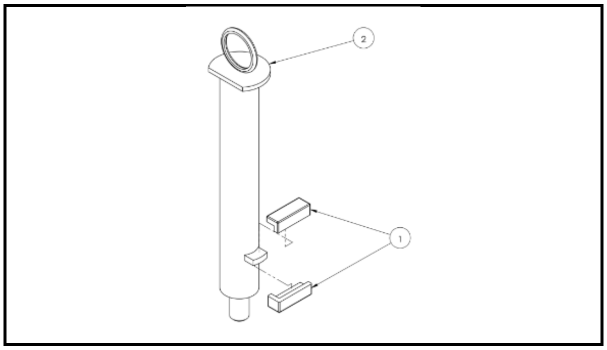
3.2.7 Preliminary Bill of Materials

This section discusses the preliminary bill of materials (BOM) for the latch mechanism. The BOM includes the part number, description, quantity, procurement type, and the vendor or manufacturer the part was sourced from for each component of the design. The vendors and manufactures listed in the BOM are the preferred suppliers based on price, convenience, or recommendation from the client. The vendors in the BOM represent who supplied the stock material or part, while the manufacturer represents who supplied custom manufactured components. If both a vendor and a manufacturer are listed for a certain component, a stock part was purchased and then sent to a manufacturer for additional modifications.

TABLE VII: PRELIMINARY BOM FOR THE LATCH MECHANISM

Latch Mechanism BOM

Assembly Name:	Advanced Tow Pin
Assembly #:	LS-ASSEM01
Approval Date:	11/26/2017
Part Count:	4
Total Cost:	$0.00 CAD



Part #	Part Name	Description	Quantity	Procurement Type	Vendor	Manufacturer
-	Tow Pin	Tow pin currently used by S&V Manufacturing	1	MTS	-	S&V Manufacturing Ltd.
LS01	Key Seat	6.35 mm Height 11.00 mm Depth 40 mm Width	2	MTS	-	S&V Manufacturing Ltd.
LS02	Key	30.16 mm Length 20.00 mm Width 6.35 mm Thickness	1	MTS	-	S&V Manufacturing Ltd.

4 Recommendations

This section discusses the team's recommendations moving forward with the proposed lift assist and latch mechanism designs. The team feels there are opportunities to decrease the cost of the designs as well as increase the reliability.

To reduce the cost of the lift assist design we recommend exploring different companies to manufacture the spring housing component of the design which is the majority of the overall cost. The company listed in this report, Kinetic Machine Works Ltd., was the only company who responded to our request for a quote. It is our belief that the other companies we reached out to did not respond as they were aware we were students. An established and respected company such as S&V Manufacturing will have no problem finding a manufacturer and negotiating the price. The same is true for the other components of the design.

Increasing the quantity will also decrease the price for almost all lift assist components. It was shown the cost decreased by approximately 45% by increasing the quantity from one to 100 units. Therefore, manufacturing the lift assist mechanism on a larger scale will reduce the cost of each unit.

During our numerical analysis of the proposed lift assist design, high stress areas were discovered near the hinge points of the mounting bracket. The stresses found were near the yield stress of the material for that specific mounting bracket model. However, since the lift assist design was only analyzed on certain mounting bracket models, the maximum stress cannot be determined for all grille guard configurations. Since 44W steel is a relatively weak material, we recommend that the material of the mounting bracket near the hinge point be changed to a material with a minimum yield strength of 600 [MPa]. We are also led to believe that the way in which the assembly was constrained during the analysis could have led to the increase in stresses observed. Therefore, a sample lift assist mechanism could be manufactured, and practical stress testing could be performed as a proof of concept. Our team is confident that a stronger grille guard material will be able to handle the increased torque caused by the lift assist mechanism.

5 Conclusion

In conclusion, a lift assist mechanism was designed to aid in the raising and lowering of S&V Manufacturing's line of Bison Grille Guard products. A latch mechanism was also designed to lock the grille guard in the raised position while also allowing for easy unlocking and lowering. The final lift assist design assembly consists of two modules with a total 16 components. The design's two 1.35 [Nm/º] torsional springs promise to reduce the grille guards weight by up to 75.2 [%] and require a maximum force of 14.33 [lbf] to raise the grille guard from the lowered position. This proposed lift assist design has an estimated unit cost of $632.92 CAD which can be reduced to $349.02 CAD by increasing the assembly quantity to 100 units. The proposed latch design is a modification of the current tow pin design utilized by S&V Manufacturing. This advanced tow pin design consists of an assembly of keys and key seats which allows for easy locking and unlocking of the grille guard. The latch design has an estimated unit cost of $0.00 CAD since all manufacturing materials and procedures are currently available to S&V Manufacturing.

It is our recommendation that S&V Manufacturing implement both the lift assist and latch mechanism designs for a total estimated cost of $349.02 CAD per unit, with an assembly quantity of 100 units. The addition of these components will increase the value of an S&V Manufacturing Bison Grille Guard as well as protect the vehicle operators from potential injuries.

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Appendix A

Customer Needs and Design Criteria

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1 Introduction

A solid understanding of the customer needs was important to the design process as they were used to determine the design criteria for the lift assist and latch mechanisms. This section discusses the customer needs that were outlined by the client for the lift assist and latch mechanism designs, and how those customer needs were converted to design criteria. The method used to rank the importance of each design criterion is also discussed for both the lift assist and latch mechanisms. The process of determining the customer needs and design criteria was an iterative process, therefore, the information shown in this section is the final iteration of that process.

2 Customer Needs

To understand the customer needs for the product, the team scheduled and conducted an onsite interview with the client. By observing both how the Bison Grille Guards were manufactured, as well as how they were currently being lowered and raised, the team collected some critical information. This included information on the grille guards as well as specifications governing the design of the lift assist mechanism. These specifications included how the lift assist should function, general dimensions of both the future product and current grille guard system, and estimated manufacturing costs. Based on the information gathered from the meeting, a preliminary list of customer needs was created. The list was further organized into a hierarchy based on similarities, as shown in TABLE I.

TABLE I: PRELIMINARY CUSTOMER NEEDS FOR LIFT ASSIST MECHANISM [1]

<table>
<tr><td>Simplicity

The product needs to:
Be manufacturable
Be lightweight
Have easily sourced materials
Be assembled at S&V Manufacturing
Function without power
Have minimal complex components
Be affordable</td><td>Reliability

The product needs to:
Maintain its desired usage over time
Function in all North American climates
Maintain its integrity over the lift of the warranty
Be robust</td></tr>
<tr><td>Functionality

The product needs to:
Be operated with one hand
Open when expected
Close when intended</td><td>Compatibility

The product needs to:
Be adjustable/versatile
Be non-intrusive to the current bracket and grille guard designs
Reduce the weight of the grille guard</td></tr>
<tr><td colspan="2">Maintainability

The product needs to:
Operate with minimal maintenance
Have minimal maintenance costs</td></tr>
</table>

A list of preliminary customer needs was also assembled for the latch system. There are many similarities between the needs for the lift assist and latch mechanisms. The lift assist, and the latch mechanism are two separate designs that will undergo their own design processes, therefore, both the lift assist, and latch mechanisms require their own specific set of customer needs. The preliminary customer needs list for the latch mechanism is shown in TABLE II.

TABLE II: PRELIMINARY CUSTOMER NEEDS FOR LATCH MECHANISM [1]

Simplicity	**Reliability**
The product needs to: • Be manufacturable • Have easily sourced materials • Be assembled at S&V Manufacturing • Function without power • Have minimal complex components • Be affordable	The product needs to: • Maintain its desired usage over time • Function in all North American climates • Maintain its integrity over the lift of the warranty
Functionality	**Compatibility**
The product needs to: • Be operated with one hand • Open when expected • Close when intended • Be easily and safely accessed by the operator	The product needs to: • Be adjustable/versatile • Be non-intrusive to the current bracket and grille guard designs • Safely secure all models of grille guard
Maintainability	
The product needs to: • Operate with minimal maintenance • Have minimal maintenance costs	

Throughout the discussion and investigation, the team eliminated the redundant or undefined needs from the lists shown in TABLE I and TABLE II. With additional investigation and discussion, the lists were reduced and prioritized by assigning a level of importance to each customer need. Some needs were excluded from the prioritized list since they were closer to customer preferences rather than customer needs. For example, the need for the lift assist mechanism to operate without power was stated by the client as a preference since that is the style of system being used by their competitors.

It should be noted that the importance value assigned to each need was determined by the team and not the client. However, the client reviewed and signed off on the list of needs and their assigned importance values. TABLE III and TABLE IV show the prioritized list of customer needs for the lift assist and latch systems, respectively.

TABLE III: PRIORITIZED LIST OF CUSTOMER NEEDS FOR LIFT ASSIST MECHANISM

Customer Needs		
Number	Need	Importance
1	The lift assist mechanism will reduce the weight of the grille	5
2	The lift assist mechanism will be adjustable and versatile	5
3	The lift assist mechanism will open and close when expected	5
4	The lift assist mechanism will reduce the current operating time	5
5	The lift assist mechanism is non-intrusive to the current grille guard and bracket set-up	5
6	The lift assist mechanism will be safe	5
7	The lift assist mechanism will maintain its integrity over the life of the warranty	4
8	The lift assist mechanism will function as intended in all North American climates	4
9	The lift assist mechanism will function as intended with minimal maintenance	3
10	The lift assist mechanism will have a low manufacturing cost.	2
11	The lift assist mechanism will have a low maintenance cost.	2
12	The lift assist mechanism will have straightforward maintenance.	2
13	The lift assist mechanism is lightweight.	1

TABLE IV: PRIORITIZED LIST OF CUSTOMER NEEDS FOR LATCH MECHANISM

Customer Needs		
Number	Need	Importance
1	The latch mechanism will be strong enough to hold the grille guard in the raised position.	5
2	The latch mechanism will be adjustable for all S&V designs.	5
3	The latch mechanism will reduce the current operating time for locking and unlocking the grille guard.	5
4	The latch mechanism will open and close when expected.	5
5	The latch mechanism will be compatible with the current grille guard and mounting bracket set up.	5
6	The latch mechanism will be safe.	4
7	The latch mechanism will function as intended in all North American climates.	4
8	The latch mechanism will function as intended with minimal maintenance.	3
9	The latch mechanism will have a low manufacturing cost.	2
10	The latch mechanism will have low maintenance cost.	2
11	The latch mechanism will have straightforward maintenance.	2
12	The latch mechanism will be lightweight.	1

These prioritized needs are used to determine the design criteria for the lift assist and latch mechanisms. Determining the customer needs and their respective levels of importance, shown in TABLE III and TABLE IV, was an iterative process that was adjusted throughout the design process in collaboration with the client to better understand their needs.

3 Design Criteria

The design criteria defined below were determined based on the customer needs defined in section 2. The criteria for both the lift assist and latch mechanisms are discussed, followed by the determination of the weight of each individual criterion. The design criteria should be viewed as a as the range from marginal value to optimal value. It is important to note that some criteria outlined below cannot be represented by a value but are still included since they were emphasized by the client. For example, product safety cannot be measured in units instead the perceived safety of a design is used to evaluate different concepts.

3.1 Lift Assist Mechanism

Based on the customer needs and target specifications, 12 criteria were established to evaluate the conceptual lift assist designs. The explanations below describe what specifications a conceptual design had to meet be successful for each criterion.

Grille Weight Reduction: The vertical force applied at the top edge of the grille guard when in the lowered position is the force needed to raise the grille guard. It is this force that is being minimized by the lift assist mechanism. The maximum allowable force for a design to successfully meet this criterion is 20 [lbs]. However, the smaller the force applied by the operator, the more successful the design. The ideal design has weight reduction capabilities that could easily be adjusted to any value required by the customer, including the optimal value of less than 13 [lbs].

Width Compatibility: The widths of S&V Manufacturing's line of Bison Grille Guard mounting brackets range from 750 [mm] to 1200 [mm] at the hinge point. A successful lift assist design must be compatible with all designs and fit within the range of 808 [mm] to 1150 [mm] which covers the majority of S&V Manufacturing's sales. It is important to note that the width of the lift assist mechanism itself does not have to fall within the specified range, it only needs to fit brackets within said range. The optimal design must meet all mounting bracket widths, while having the capability for expansion to larger designs in the future.

Weight Compatibility: The weight of S&V Manufacturing's Bison Grille Guards ranges from 175 [lbs] to 225 [lbs]. A successful design must satisfy the grille weight reduction criterion, outlined above, for **all** S&V Manufacturing grille models. An optimal design would be compatible with all current grille models while also having the capability to increase the weight range up to 250 [lbs]. A high score will correlate to a high perceived weight bearing ability, and large weight bearing range. A score of one will indicate a perceived weight bearing ability equivalent to the marginal value range of 175 [lbs] to 225 [lbs].

Height Compatibility: Depending on the vehicle type, the distance from the bottom of the mounting bracket to the ground must be within the range of 305 [mm] to 406.4 [mm]. A successful lift assist design will be no more 305 [mm] to the ground, or obstruct any of the grille guard components. However, the optimal design should be contained within the space between the hinge points. If an axis is drawn from one hinge point to another, and a cylinder is constructed with a radius of 80 [mm], a successful design would be fully contained within that space. An illustration showing this design space can be seen below in Figure 1.

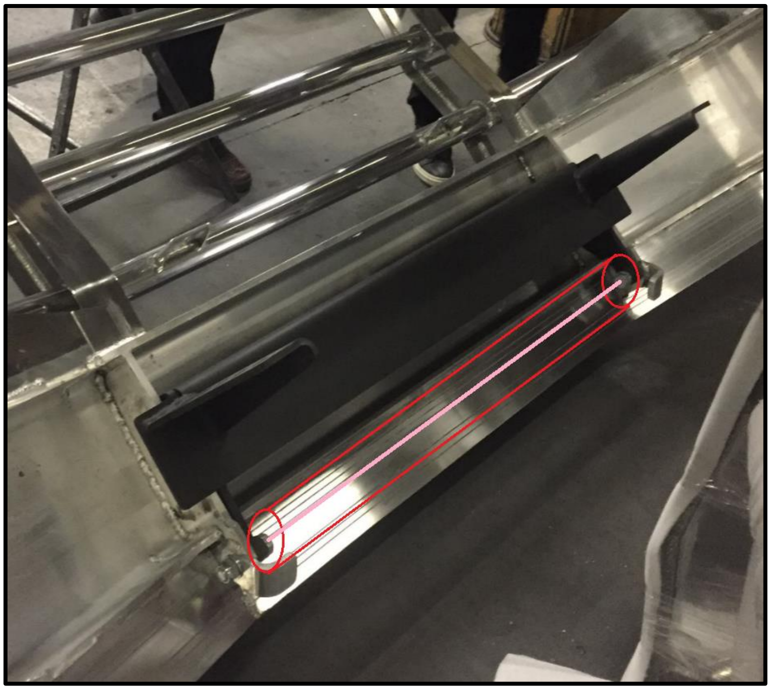

Figure 1: The Optimal Design Space to meet the Height Compatibility Criteria [2]

A design must be 305 [mm] away from the ground to be considered. The closer it is to the optimal design space, the higher the score will be.

Product Safety: A design is considered safe for the operator if there are no exposed pinch points, sharp edges, hot surfaces, high velocity components, or high voltage components. A design that scores well for this criterion is a design which does not add any additional potentially hazardous components to the current grille guard system.

Product Reliability: A successful design will have the ability to retain its functionality and performance over its guaranteed lifetime. A design that scores well for this criterion would effectively outlast any reasonable customer requirements.

Operable in all Working Conditions: The design's ability to retain its performance in all roadside weather conditions, including dust, rain, hail, ice, snow, as well as temperatures ranging from -30 [^{o}C] to 40 [^{o}C]. The optimal design will be able to function in temperature ranges of -40 [^{o}C] to 50 [^{o}C].

Maintenance Cost: The required maintenance for the lift assist mechanism will have a minimal cost over the life of the product. The optimal lift assist design will have no required maintenance.

Maintenance Frequency: The required scheduled maintenance should be performed as infrequently as possible without affecting the safety or performance of the lift assist system. The minimal time allowable between scheduled maintenance is four months. The optimal lift assist design will have no required maintenance.

Maintenance Simplicity: The required maintenance for the lift assist mechanism will be able to be performed by one person, ideally the operator of the vehicle, in under 15 minutes, with little to no specialized tooling. As well, this criterion includes the ease of repair of the lift assist mechanism in the case of unpredictable part wear or failure. A design will achieve a higher score if it has fewer custom parts. It would be considered ideal if the customer could repair the mechanism without having to send it back to S&V Manufacturing.

Design Cost: The cost of the lift assist design, including materials and manufacturing, will be minimized. A successful design will have a design cost of $300 [CAD] or less for both the lift assist **and** latch mechanisms. The lower the lift assist design cost, the more successful the design.

Lightweight: The weight of the lift assist mechanism will be minimized. A successful design will have a weight of less than 40 [lbs]. The optimal lift assist design will have a weight between 20 and 30 [lbs].

3.2 Latch Mechanism

Based on the customer needs and specifications, eight criteria were established to evaluate the conceptual latch designs. The explanations below outline what specifications a conceptual design must meet be successful for each criterion.

Latch Strength: The latch strength is the force required for the latch to fail when in the closed position. A successful latch design will have a latch strength equal or greater than the current tow pin system.

Operating Time: The operating time is the time it takes to lock or unlock the grille guard in the raised position. A successful latch design will reduce the operating time in comparison to the current eye bolt design. The shorter the operating time, the more successful the design.

Product Reliability: A successful design will have the ability to retain its functionality and performance over its guaranteed lifetime. A successful design will also open when expected, and close and remain close when intended.

Product Safety: A design is considered safe for the operator if there are no exposed pinch points, sharp edges, hot surfaces, high velocity components, or high voltage components. A design that scores well for this criterion is a design which does not add any additional potentially hazardous components to the current latch design.

Maintenance Cost: The required maintenance for the latch mechanism will have a minimal cost over the life of the product. The optimal latch design will have no required maintenance.

Maintenance Frequency: The required scheduled maintenance should be performed as infrequently as possible without affecting the safety or performance of the latch mechanism. The minimal time allowable between scheduled maintenance is four months. The optimal latch design will have no required maintenance.

Maintenance Simplicity: The required maintenance for the latch mechanism will be able to be performed by one person, ideally the operator of the vehicle, in under 15 minutes, with little to no specialized tooling. The optimal lift assist design will have no required maintenance.

Design Cost: The cost of the latch design, including materials and manufacturing, will be minimized. A successful design will have a design cost of $300 [CAD] or less for both the lift assist **and** latch mechanisms as previously mentioned. The lower the latch design cost, the more successful the design.

4 Criteria Evaluation

The criteria outlined in section 3.1 and 3.2 do not hold the same importance in determining the success of the final design. The team evaluated the different criteria to distinguish which were more important contributing factors to overall design success. A hit matrix was utilized to determine the weights of each criterion. A hit matrix evaluates the criteria by comparing each criterion in a head-to-head fashion. Whichever criterion was deemed more important to the design success based on the customer needs was awarded a hit. Each criterion was then assigned a fractional weight based on the number of hits it accrued, divided by the total number of hits in the matrix. This weight was utilized to distinguish the importance of each criterion.

Two different matrices were utilized; one for the criteria governing the lift assist, and one for the criteria governing the latch mechanism. The criteria weighting matrices can be seen below in TABLE V and TABLE VI. It should be noted that the winner of each criterion comparison was decided by the team based on the customer needs.

TABLE V: THE HIT MATRIX FOR THE LIFT ASSIST MECHANISM CRITERION

CRITERIA			Grille Weight Reduction	Deisgn Compatibility: Width	Deisgn Compatibility: Weight	Deisgn Compatibility: Height	Product Safety	Operable in all Working Conditions	Reliability over Product Lifetime	Maintenance: Cost	Maintenance: Frequency	Maintenance: Simplicity	Design Cost	Lightweight
			A	B	C	D	E	F	G	H	I	J	K	L
	Grille Weight Reduction	A		A	A	A	A	A	A	A	A	A	A	A
Deisgn Compatibility	Width	B			B	B	B	B	B	B	B	B	B	B
Deisgn Compatibility	Weight	C				C	C	F	C	C	C	C	C	C
Deisgn Compatibility	Height	D					E	F	G	D	D	D	K	D
	Product Safety	E						F	E	E	E	E	E	E
	Operable in all Working Conditions	F							F	F	F	F	F	F
	Reliability over Product Lifetime	G								G	G	G	G	G
Maintenance	Cost	H									H	J	K	H
Maintenance	Frequency	I										I	K	L
Maintenance	Simplicity	J											K	J
	Design Cost	K												K
	Lightweight	L												
HITS			11	10	8	4	7	9	6	2	1	2	5	1
WEIGHT			0.167	0.152	0.121	0.061	0.106	0.136	0.091	0.03	0.015	0.03	0.076	0.015

TABLE VI: THE HIT MATRIX FOR THE LATCH MECHANISM CRITERION

CRITERIA			Latch Strength	Design Compatibility	Product Safety	Reduces Current Operating Time	Reliability over Product Lifetime	Maintenance			Design Cost
								Cost	Frequency	Simplicity	
			A	B	D	E	F	G	H	I	J
	Latch Strength	A		A	A	E	F	A	A	A	A
	Design Compatibility	B			B	E	B	B	B	B	B
	Product Safety	D				E	D	D	D	D	D
	Reduces Current Operating Time	E					E	E	E	E	E
	Reliability over Product Lifetime	F						F	F	F	F
Maintenance	Cost	G							H	G	J
Maintenance	Frequency	H								I	J
Maintenance	Simplicity	I									J
	Design Cost	J									
HITS			6	6	5	8	5	1	1	1	3
WEIGHT			0.167	0.167	0.139	0.222	0.139	0.028	0.028	0.028	0.083

The most important design criteria for the lift assist mechanism were determined to be the grille weight reduction, design weight compatibility, and product safety. The most important criteria for the latch system were the operating time, the latch strength, and design compatibility.

5 Conclusion

Through multiple meetings and on-going email communication with S&V Manufacturing, the preliminary customer needs for the lift assist and latch mechanism needs were determined. After organizing the needs based the importance emphasized by the client, the design criteria were determined for the lift assist and latch mechanism designs. The design criteria were then ranked using the hit matrix method. This lead to the determination that the ability to reduce the weight of the grille guard, the ability to reduce the weight of **all** grille guard models, and the product safety were the most important design criteria for the lift assist mechanism. The evaluation also determined that the time is takes to lock or unlock the grille guard, the latch strength, and the compatibility with all grille guard models were the most important design criteria for the latch mechanism. The design criteria are used to evaluate the conceptual designs shown in Appendix B.

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Appendix B

Concept Generation and Selection

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1 Introduction

This appendix discusses the concept generation, evaluation, and selection of the lift assist and latch mechanism designs. The process used to determine the final lift assist and latch mechanism designs is outlined in the following sections.

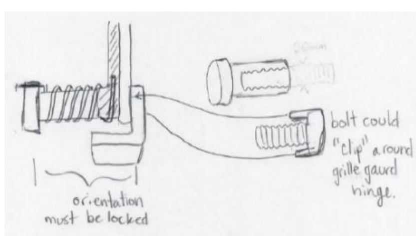
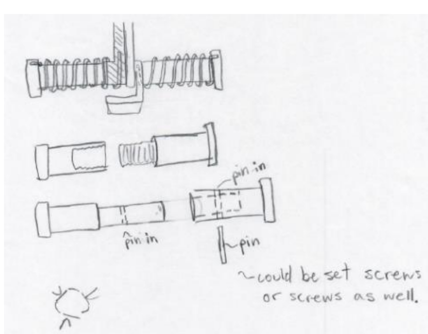
2 Lift Assist Mechanism

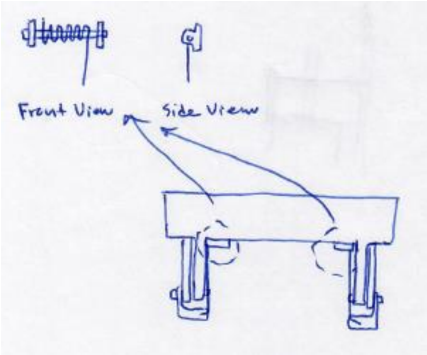
This section will discuss the design of the lift assist mechanism. The design process began with concept generation where a total of 20 conceptual designs were presented. The designs were then grouped by similarities to reduce the number of redundant concepts which yielded 12 unique conceptual lift assist designs. The designs were then evaluated based on the design criteria outlined in Appendix A. A second form of analysis was also performed to further distinguish between potential final designs. The full process that was utilized is described in the sections below to provide further detail.

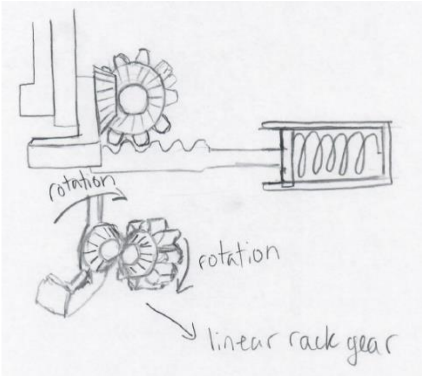
2.1 Conceptual Design Generation

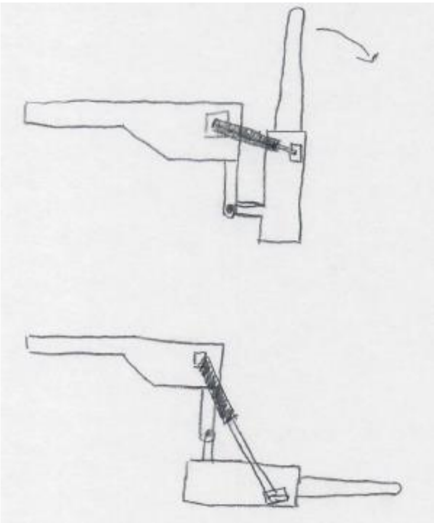
This section presents each of the conceptual designs generated by the team. The 12 conceptual lift assist designs are introduced in TABLE I. Each design is accompanied by a sketch and a brief description.

TABLE I: CONCEPTUAL LIFT ASSIST MECHANISM DESIGNS

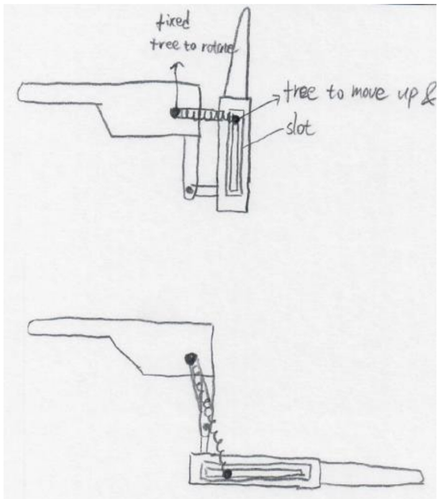
#	Design Name/Sketch	Description
1	**Hinge-Mounted Single Spring** 	• This design uses two large coiled springs which are attached to the two hinge points. One spring located at each hinge point. • One end of the spring is fixed vertically to the mounting bracket arm, while the other end is fixed to a section of the grille guard. • As the grille guard is raised and lowered, the bracket and grille create an angular displacement in the springs which creates resistance.
2	**Equilibrium Double-Spring** 	• The design uses two large coiled springs on each hinge point. A total of four springs for the entire lift assist set-up. • The spring system would be secured to both the mounting bracket as well as the grille guard. • Like the single-ended spring system, the angular displacements between the bracket and grille guard would result in resistance created by the springs. • The two springs on either side hinge point would oppose each other and be in equilibrium.

<table>
<tr><td>3</td><td>Bracket Mounted Spring

</td><td>

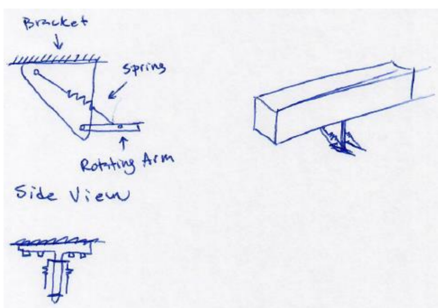
Two spring systems are attached under the bracket (above the hinge points).
One end of the spring will be fixed on the system housing, and another end of the spring rest on the grille guard back stops.
As the grille hinge section rotates, the spring end would also rotate, creating resistance.
This design is similar to both competitor products, the Herd Lift Assist (patent pending) and the Titan 2000 Easy Lift. It is both a feasible design, and can be used as a benchmark through the evaluation process.
</td></tr>
<tr><td>4</td><td>Hypoid-Pinion-Rack Gear System

</td><td>

This design uses hypoid-pinion-rack gear system.
The system converts rotary motion to linear motion.
The horizontal rack system consists of a spring, a cylindrical housing, and a horizontal link bar with rack at the end.
As the link bar moves back and forth, the spring would be in compression or tension, which creates mechanical force to aid the operator in lowering or raising the grille.
</td></tr>
<tr><td>5</td><td>Hydraulic System
</td><td>

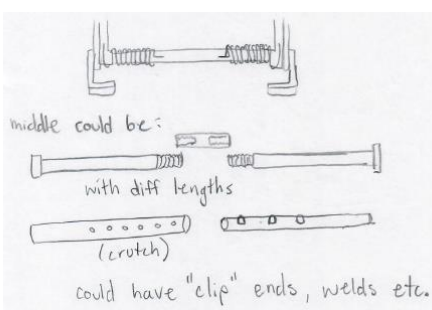
The design uses two hydraulic pistons as the lift assist mechanism.
One end of the hydraulic piston is mounted on the outside out the bracket, while another end is fixed on the grille.
The system is mounted by using mounting bolts, which the mounting location could be easily adjusted for different applications.
</td></tr>
</table>

<table>
<tr>
<td>6</td>
<td>Spring-Slot System
</td>
<td>

The design uses a spring with one end is fixed on the bracket, and another end is free to move within a slot that is fixed on the grille.
The spring is bolted on the bracket that is rotation-free.
A metal slot panel is either welded or bolted on the grille.
The slot-side of the spring moves up and as the raising or lowering of the grille.
The metal slot panel can also be mounted on the bracket, while the spring is mounted on the grille.

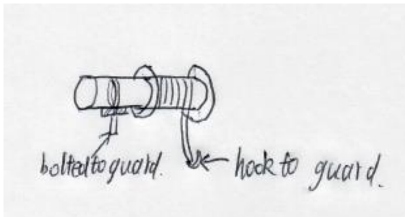
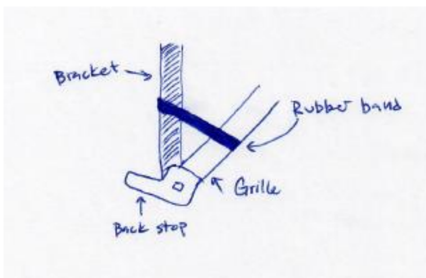
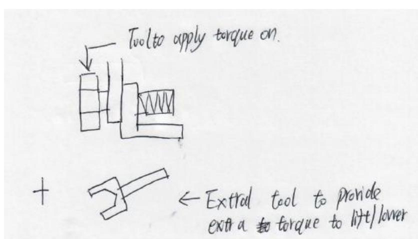
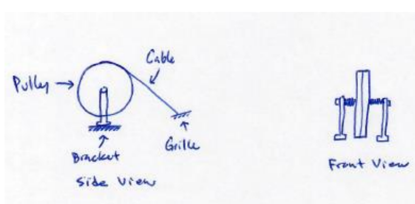
</td>
</tr>
<tr>
<td>7</td>
<td>Kick-Stand Spring System
</td>
<td>

A metal piece is mounted and fixed under the bracket as the stand system.
One end of the spring is attached to the stand, and another end is attached to a rotating arm.
The rotating arm is attached to the grille, which creates resistance when the grille guard is being lowered.
This design is inspired by the bike kick stands.

</td>
</tr>
<tr>
<td>8</td>
<td>Adjustable Rod with Spring
</td>
<td>

This design uses a single long rod is pinned between both hinge points.
Two coiled springs are attached at both hinge points.
The long rod also eliminates the bending moments.
Adjustable by using different lengths of the rod.

</td>
</tr>
</table>

9	**Independent Torque Spring Housing**  	• The housing has two housing components. One side of housing is fixed to the bracket and another side of housing is attached to a wire and is free to rotate. • The free end of the wire is attached to the grille guard. • This design acts like an individual module that can be placed on different locations on the grille for different locations.
10	**Elastic Band Design**  	• The elastic band design includes a strong elastic rubber band on each side of the grille. • The rubber band will provide a resistance that will help to lower the grille guard.
11	**External Torque Wrench**  	• This external torque wrench design will require the operator to use a special tool to produce torque to lift the guard.
12	**Spring-Loaded Pulley**  	• The spring-loaded pulley includes a pulley that is preloaded by a spring. • Once the latch is released, the pulley attached to cable will provide a torque to reduce the weight of the grille guard.

2.2 Conceptual Design Evaluation

The conceptual designs presented in section 2.1 were evaluated based on their ability to meet the design criteria. Each designs ability to meet a given criterion was given a score from one to five. A design was given a score of one for a given criteria if it met, but did not exceed the marginal design value. A score of five was awarded to a design if it reached or exceeded the optimal value for a specific criterion. A design would we awarded a score between two and four if it was able to achieve between the marginal and optimal values for that criterion. For design criteria where there was no specified marginal or optimal value, the design was scored based on its perceived ability to achieve the level of performance outlined in the criterion definition. In situations where a specific design achieved optimal levels of perform based on certain aspects of a criterion but fell short on others, a judgement call was made based on engineering intuition.

Below is an example of how the conceptual designs were scored. For this example, the width compatibility design criterion was used. The score and a justification are shown below for scores of 1, 3, and 5.

1 – Meets the marginal target specification for the design width. The lift assist mechanism is compatible for mounting brackets ranging in widths from 808 [mm] to 1150 [mm].

3 – Exceeds the marginal target specification for the design width, but does not meet the optimal value OR meets the optimal target specification for the design width but requires modification to do so. The lift assist mechanism is compatible for mounting brackets ranging in width from 750 [mm] to 1200 [mm].

5 – Meets or exceeds the optimal target specification for the design width for all grille guard models. The lift assist mechanism is compatible for mounting brackets ranging in width from 750 [mm] to 1200 [mm].

All conceptual designs were ranked based on the design criteria individually by each team member. Each hit was multiplied by the criterion weights established in Appendix A. The design scores were multiplied by a factor of 20 to allow the total score for a design to be evaluated out of 100 points. The score given by each team member was averaged to allow for independent scoring and to eliminate designer bias.

The hope was that this evaluation method would yield a single conceptual design for the lift assist mechanism that stood apart from the rest. However, TABLE II shows that the design scores are separated into two distinct groupings. The designs highlighted in green performed well with scores grouped closely together, while the designs highlighted in red scored worse.

TABLE II: LIFT ASSIST MECHANISM CONCEPT EVALUATION MATRIX

LIFT ASSIST MECHANISM														
			Deisgn Compatibility						Maintenance					
Criteria		Grille Weight Reduction	Width	Weight	Height	Product Safety	Operable in all Working Conditions	Reliability over Product Lifetime	Cost	Frequency	Simplicity	Design Cost	Lightweight	Total out of 100
Weight (X/1)		0.167	0.152	0.121	0.061	0.106	0.136	0.091	0.03	0.015	0.03	0.076	0.015	
Designs \ Weight		3.333	3.03	2.424	1.212	2.121	2.727	1.818	0.606	0.303	0.606	1.515	0.303	
Single Rod with Springs	Scores 1-5	3.875	2.125	2.5	4.5	4.125	4.375	4.375	4.5	4.375	4.125	3.75	3.5	72.73
Kick Stand Lift		4.125	4.25	4.125	3	4.125	4.125	4.375	4.625	4.25	3.875	4	4.5	82.08
Hydraulic Lift		3.25	3.25	3.375	1.75	4.125	2.875	3.875	2	2.5	2.25	2.125	2.625	61.97
Hinge-Fixed Single-Spring		4	4.75	4.25	4.75	3.75	4.625	4.125	4.375	4	5	4.875	4.75	87.58
Grille Mounted Torque Spring		4.625	3.75	4.25	2.75	3.5	4.5	4.125	4	4.25	3.25	2.75	3.75	78.6
Geared Lift		3	3.75	2.5	3.5	3.25	3.625	2.875	3.375	3.25	2.5	2	2.75	62.08
Equilibrium Double-Spring		4.5	4.125	4.5	4.75	3.75	4.625	4.25	4.25	4.125	4.25	4.125	4.375	86.44
Bracket Mounted Spring Lift		4	4.5	4.5	4.125	4.125	4.25	4.5	4.375	4.125	4.25	4.375	4	85.72

The highest scoring design for the lift assist mechanism were the Kick Stand Lift (82.08), the Bracket Mounted Spring Lift (85.72) the Equilibrium Double-Spring (86.44) and the Hinge-Fixed Single Spring (87.58). All other conceptual designs were eliminated from further analysis and evaluation. The reason for the elimination for each lift assist conceptual design is outlined below to provide more insight into how the scoring was performed.

The Hydraulic Lift system achieved the lowest score of 61.97. It scored poorly for many reasons, including but not limited to:

- Its large design size which required one end of the hydraulic mechanism to be attached to the grille guard. This would lead to the height of the system to interfere with the grille guard itself and would require modification and optimization for each specific grille configuration.
- Its elevated cost due to the fact that pneumatic or hydraulic pistons would need to be purchased rather than developed or manufactured by S&V Manufacturing.
- Its specialized maintenance, which would require a trained professional to perform any maintenance or repairs.

The Gear Lift system achieved the second lowest score of 62.08. It scored poorly for many reasons, including but not limited to:

- Its complex design which required the development of gearing ratios to convert the rotational motion of the grille guard into linear motion. This would lead to higher costs, a heavier system, and the potential for high maintenance or repair costs.
- Its inability to satisfy the weight compatibility criterion. Finding a gearing ratio that would effectively move both the lightest grille guard, and the heaviest grille guard would be an engineering challenge.

The Single Rod with Springs system achieved a score of 72.73. It scored poorly for reasons including but not limited to:

- Its design and manufacturing costs which would be elevated by the fact that the system would most likely have to be custom made to be compatible with the range of mounting bracket widths.
- Its increased design weight due to excess material.

The Grille Mounted Torque Spring system achieved a score of 78.6. This design scored poorly in a few areas which led to its elimination. This design scored poorly for reasons including but not limited to:

- Its inability to satisfy the height compatibility criterion. The design required a cable to attach to the grille guard which would vary between different grille guard models.
- Its complex design which consisted of multiple moving and custom-made components would increase the design, manufacturing, and potential maintenance costs.

2.3 Sensitivity Analysis

Due to the number of successful designs for the lift assist and the fact that their scores were so closely grouped, further analysis of these designs was required in order to determine how to proceed. A sensitivity analysis was performed to re-evaluate the most successful designs. This second evaluation utilized a different method of scoring the concept designs for each criterion.

The sensitivity analysis utilized the hit matrix evaluation method to determine which design performed best for each criterion. A hit matrix was constructed for each criterion to rank the designs head-to-head in that category. Each hit was multiplied by the criterion weights established in Appendix A, and then summed to provide an overall score to rank the designs from strongest to weakest. This process not only established which design was strongest for each criterion, but also which design was the strongest overall. Below is an example of the hit matrix used to compare the designs performance for grille weight reduction criterion.

TABLE III: THE HIT MATRIX COMPARING LIFT ASSIST MECHANISM DESIGNS FOR GRILLE WEIGHT REDUCTION

GRILLE WEIGHT REDUCTION		Kick Stand Lift A	Hinge-Fixed Single-Spring B	Equilibrium Double-Spring C	Bracket Mounted Spring Lift D
Kick Stand Lift	A		A	C	D
Hinge-Fixed Single-Spring	B			C	D
Equilibrium Double-Spring	C				D
Bracket Mounted Spring Lift	D				
HITS		1	0	2	3
SCORE		0.17	0	0.33	0.5

TABLE III shows that the Bracket Mounted Spring Lift performed best for that specific criterion. That is due to the design's ability to have up to six springs available for potential energy storage, while other designs are limited to four springs such as the Equilibrium Double-Spring, or two springs like the Kick Stand Lift and Hinge-Fixed Single-Spring. A second important criterion in determining design success was the width compatibility criterion, whose hit matrix is displayed below.

TABLE IV: THE HIT MATRIX COMPARING LIFT ASSIST MECHANISM DESIGNS FOR DESIGN WIDTH COMPATIBILITY

DESIGN COMPATIBILITY - WIDTH		Kick Stand Lift	Hinge-Fixed Single-Spring	Equilibrium Double-Spring	Bracket Mounted Spring Lift
		A	B	C	D
Kick Stand Lift	A		B	C	D
Hinge-Fixed Single-Spring	B			B	B
Equilibrium Double-Spring	C				C
Bracket Mounted Spring Lift	D				
HITS		0	3	2	1
SCORE		0	0.45	0.3	0.15

The strongest design in this category, the Hinge-Fixed Single-Spring, was determined to be the best performer due to its small size. This design will theoretically fit for any mounting bracket width since the mechanism is contained to the hinge point. This design sits in the exact ideal position outlined by the client in our customer needs, and target specifications meetings. The ideal position is located on the inside of the mounting bracket at the hinge point which can be seen below in Figure 1. This was deemed to be the ideal position since it is the only point that each grille guard model has in common with the exact same dimensions and clearances. A lift assist mechanism in this location will also not impede the grille guard's functionality.

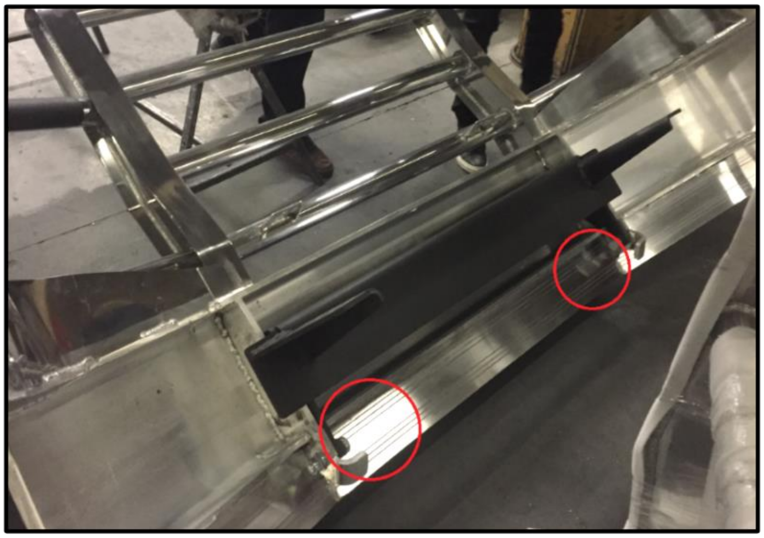


Figure 1: The Ideal Lift Assist Mechanism Location [1]

The Equilibrium Double-Spring design also performed well in this category, as it is based on the same hinge point except with spring features on either side. The Kick Stand Lift and Bracket Mounted Spring Lift did not perform as well, due to the fact they would be mounted directly onto the mounting bracket, which is the black steel frame visible in Figure 1. Mounting to the bracket itself detracts from the design's ability to meet this criterion since there are numerous mounting bracket models, meaning variability on the mounting placement. This leads to poor design compatibility between different bracket models and widths.

Based on the results of the sensitivity analysis, the final rankings for the lift assist mechanism designs were:

1. Equilibrium Double-Spring – 2.06
2. Bracket Mounted Spring Lift – 1.94
3. Hinge-Fixed Single Spring – 1.52
4. Kick Stand Lift – 0.39

These results were presented to the client to receive input before selecting a final design. The client agreed with the criteria, concept evaluation process, and sensitivity analysis process. Additional insight was also provided that had not been considered in previous discussions with the client.

The client noted that when the mounting bracket and grille guard assembly was shipped to the customers, it was sometimes disassembled for easier installation onto the vehicles. Therefore, it is critical to have the lift assist mechanism design be easy to reassemble by the customer. However, one of the strengths of the top rated design, the Equilibrium Double-Spring was simplicity.

Throughout optimization the difficulty of assembly would reduced for the customer – S&V Manufacturing agreed with this evaluation.

After speaking with S&V Manufacturing it was determined that the two designs that would be considered moving forward were the Hinge-Mounted Single-Spring and Equilibrium Double-Spring designs.

After the meeting it was determined that the Hinge-Mounted Single-Spring design would be used as the final design as its cost was significantly lower than that of the Equilibrium double spring. The fewer springs contributed to substantially cost savings.

2.4 Final Design Selection

The final lift assist design selection was the Hinge-Mounted Single-Spring. The Hinge-Mounted Single-Spring is pictured below in Figure 2 below.

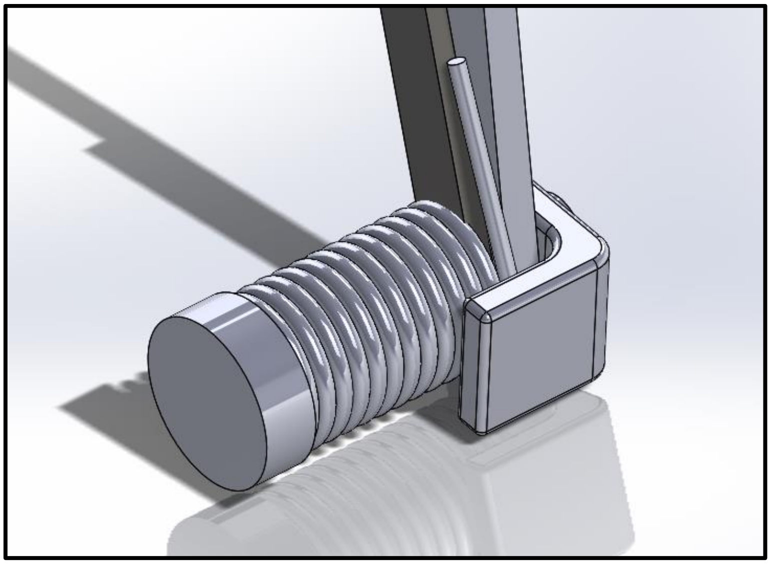


Figure 2: The Hinge-Mounted Single-Hinge Design Concept with the Mounting Bracket and Grille Guard [2]

The working principal of the design is that a spring in torsion will produce a torque (T) measured in [Nm] related to the angle of twist the spring is subjected to from it's equilibrium position (θ) measured in [rad], and it's spring constant (K) measured in [Nm] [3]:

$$T = K\theta$$

The angle that the spring will be subjected to is equal to the angle the grille guard moves through from the opening to closing positions. This angle is approximately 90° as seen below in Figure 3 and Figure 4.

Figure 3: Grille Guard in the Raised Position (0°) [4]

Figure 4: Grille Guard in the Lowered Position (90°) [4]

The lift assist mechanism will be mounted onto the hinge where the grille attaches to the mounting bracket. It will also be used as the fastener at this point, by threading together as seen in Figure 5 below.

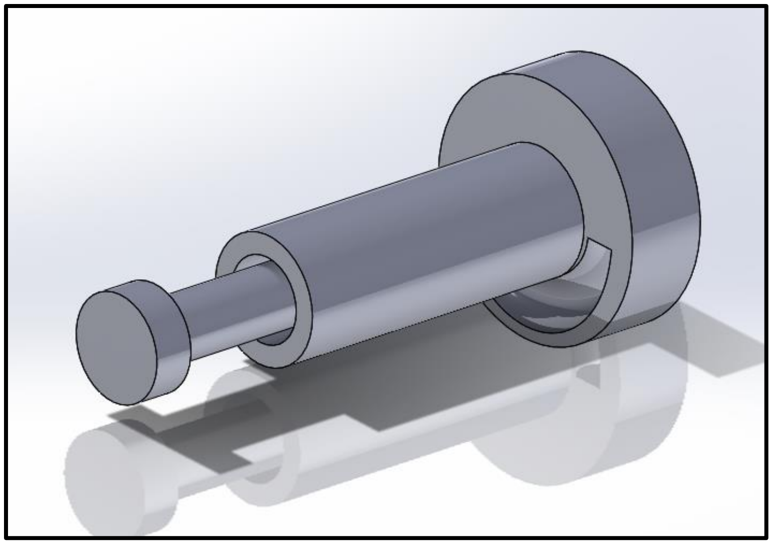


Figure 5: The Male (Left) and Female (Right) Lift Assist Components [2]

The reason for this mounting position is because it is subjected to the angular displacement which is critical to the design. This hinge is also the only completely constant feature throughout all S&V Manufacturing Bison Grille Guards. This hinge can be seen below in Figure 6.

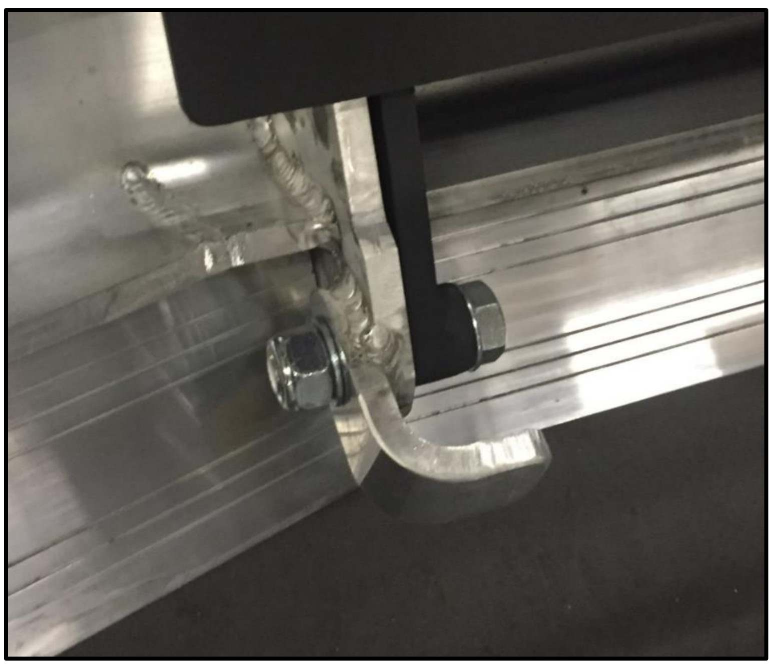


Figure 6: The Hinge Connecting the Grille to the Mounting Bracket [1]

The black structure in the image is the mounting bracket, which is attached to the frame of the vehicle and is locked in orientation. The silver structure is the grille guard. The L-shaped member protruding out the bottom of the page is what stops the grille in the lowered position, and holds it at 90°. The grille guard would rotate to the lowered position into the page from the view shown above.

The proposed design would secure the grille and bracket together utilizing the same 20 [mm] bolt that is currently used by S&V Manufacturing. However, the bolt would instead thread into a female component which would include a structure which the spring would seat into and lock. This structure would be locked in the same orientation as the grille, so as the grille guard rotates about the hinge point, the spring would want to twist with it. The L-Shaped structure seen in Figure 2, would seat onto the grille portion of the hinge, allowing for this locked orientation. However, this twist would be opposed by the springs mating surface with the mounting bracket, leading to a twist out of it's equilibrium position and an increase in torque. This torque value would linearly increase, reaching a maximum value as the grille reaches the lowered position. A better view of how the spring would seat onto the mounting bracket can be seen below in Figure 7.

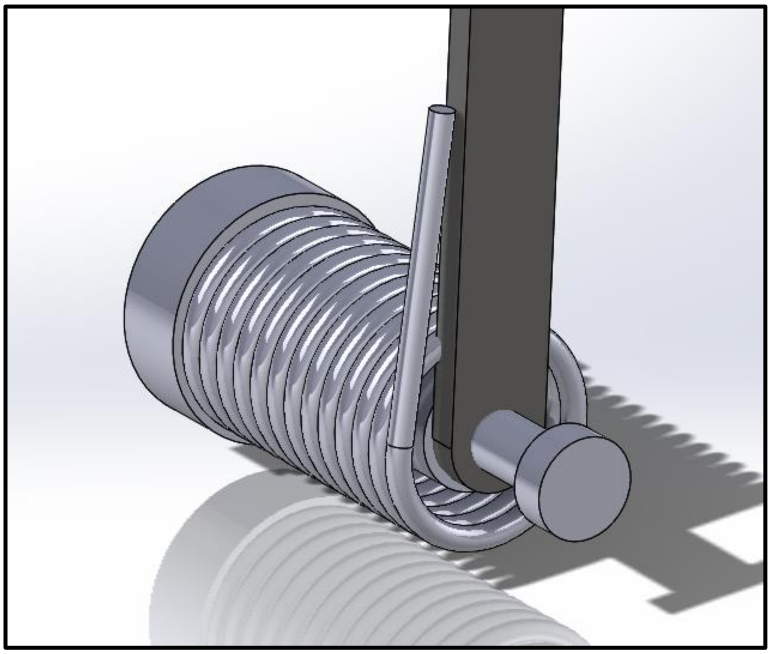


Figure 7: Potential Mating Configuration for the Spring and Mounting Bracket Arm [2]

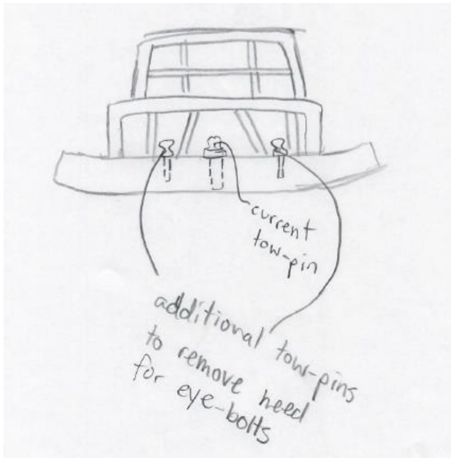
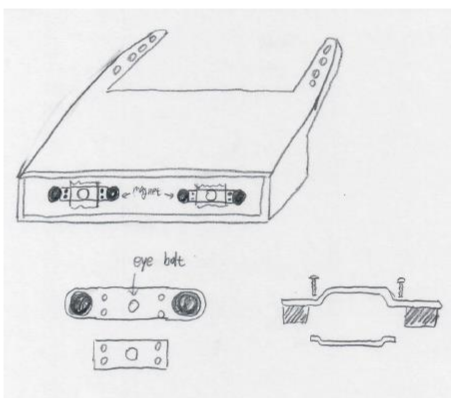
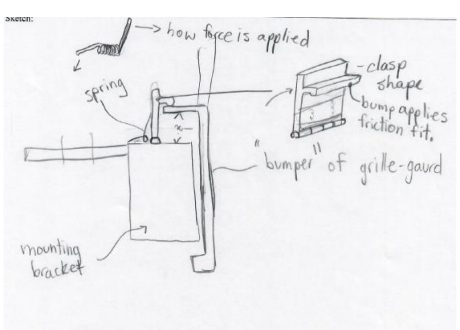
3 Latch Mechanism

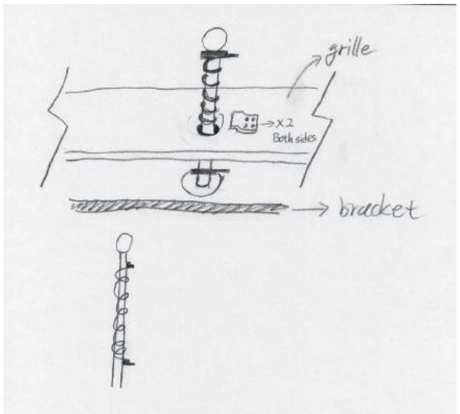
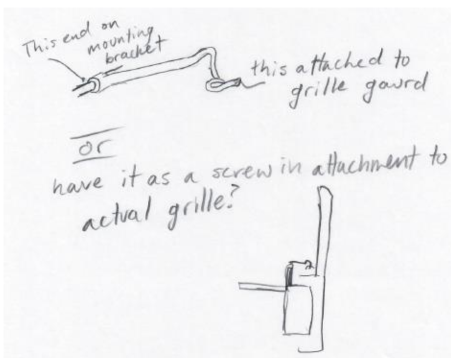
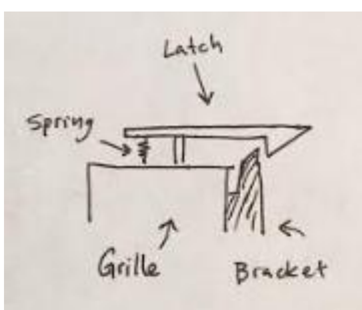
This section will discuss the design of the latch mechanism. The design process began with concept generation where a total of six conceptual designs were presented. The designs were then evaluated based on the design criteria outlined in Appendix A. A second form of analysis was also performed to further distinguish between potential final designs. The full process that was utilized is described in the sections below to provide further detail.

3.1 Conceptual Design Generation

This section presents each of the conceptual designs generated by the team. The six conceptual latch designs are introduced in TABLE V. Each design is accompanied by a sketch and a brief description.

TABLE V: CONCEPTUAL LATCH MECHANISM DESIGNS

<table>
<tr><th>#</th><th>Design Name/Sketch</th><th>Description</th></tr>
<tr><td>1</td><td>Simple Drop Pin

</td><td>• The simple drop pin design is an improved version of the current tow pin design. The simple drop pin concept will provide higher tolerance on important geometry.
• The simple drop pin design can also be implemented as multiple drop pins in order to eliminate the need for eye bolts.</td></tr>
<tr><td>2</td><td>Magnet Latch

</td><td>• This magnet latch design is designed to prevent the grille assembly from unexpectedly falling when the tow pin and eye bolts are being removed.
• The magnet latch design can either be bolted or welded between top and bottom surface of the bracket in current grille assembly.</td></tr>
<tr><td>3</td><td>Spring Loaded Latch Vertical

</td><td>• The spring-loaded latch design will be mounted on to the mounting bracket with a torsional spring behind a clasp-shape latch.
• The latch will provide a friction force to secure the guard in the expected location.</td></tr>
</table>

4	**Advanced Tow Pin** 	• The advanced tow pin is a modified version of the current tow pin design. The modified tow pin will secure the grille in the raised position when engaged. • Pressing the tow pin head underneath the grille can disengage the tow pin. The grille guard can then be lowered.
5	**Hook Latch** 	• The hook latch design will be fixed to the mounting bracket. The design will hook onto the grille guard to secure it in place.
6	**Spring-Loaded Latch Horizontal** 	• The spring loaded horizontal latch would be mounted onto the grille guard bumper. • It would have an additional piece built onto the mounting bracket which would extend up to the latch. It would hook over to secure the guard in place. • The latch would be spring-loaded to keep it shut, and allow it to maintain its locked position when not in use.

3.2 Conceptual Design Evaluation

The conceptual designs presented in section 3.1 were evaluated based on their ability to meet the design criteria. Each designs ability to meet a given criterion was given a score from one to five. A design was given a score of one for a given criteria if it met, but did not exceed the marginal design value. A score of five was awarded to a design if it reached or exceeded the optimal value for a specific criterion. A design would we awarded a score between two and four if it was able to achieve between the marginal and optimal values for that criterion. For design criteria where there was no specified marginal or optimal value, the design was scored based on its perceived ability to achieve the level of performance outlined in the criterion definition. In situations where a specific design achieved optimal levels of perform based on certain aspects of a criterion but fell short on others, a judgement call was made based on engineering intuition.

All conceptual designs were ranked based on the design criteria individually by each team member. Each hit was multiplied by the criterion weights established in Appendix A. The design scores were multiplied by a factor of 20 to allow the total score for a design to be evaluated out of 100 points. The score given by each team member was averaged to allow for independent scoring and to eliminate designer bias.

The hope was that this evaluation method would yield a single conceptual design for the latch mechanism that stood apart from the rest. However, TABLE VI shows that the design scores are separated into two distinct groupings. The designs highlighted in green performed well with scores grouped closely together, while the designs highlighted in red scored worse.

TABLE VI: LATCH MECHANISM CONCEPT EVALUATION MATRIX

LATCH											
Criteria		Latch Strength	Design Compatibility	Product Safety	Reduces Current Operating Time	Reliability over Product Lifetime	Maintenance: Cost	Maintenance: Frequency	Maintenance: Simplicity	Design Cost	Total out of 100
Weight (X/1)		0.167	0.167	0.139	0.222	0.139	0.028	0.028	0.028	0.083	
Designs \ Weight		3.333	3.333	2.778	4.444	2.778	0.556	0.556	0.556	1.667	
Spring Loaded Latch Vertical	Scores 1-5	3.75	3.375	3.75	4.25	4	4.125	4.5	4.375	3.5	77.22
Simple Drop Pin		4.5	4	4.25	4	4.625	4.75	5	5	4.75	86.88
Magnet Latch		3.125	3.25	3.5	4	4	4	4.75	4.25	2.75	71.67
Advanced Tow Pin		4.25	4	3.625	4.75	3.75	4.375	4.75	4	4	83.06
Spring Loaded Latch Horizontal		4	3.375	4.25	4.5	4.375	3.875	4.75	4.625	3.125	81.11

The highest scoring latch mechanism designs were the Spring-Loaded Latch – Horizontal (81.11), the Advanced Tow Pin (83.06), and the Simple Drop Pin (86.88).

The Magnet Latch achieved the lowest score of 71.67. It scored poorly for many reasons, including but not limited to:

- Its inability to satisfy the latch strength criterion. Finding a magnet that would be strong enough to hold the grille, yet also be weak enough to be easily opened by a human would be an engineering challenge. Alternatively, an electromagnet design could be used but that would result in a complex and expensive design.
- Its inability to satisfy the design cost criterion. The cost of the materials required for this design would render the design infeasible.

- Its inability to satisfy the design compatibility criterion since the mounting location for the latch mechanism would have to be different for each grille guard model to allow for a proper connection between both faces of the magnet latch.

The Spring-Loaded Latch – Vertical achieved a score of 77.22. This design scored well for most criteria however there were a few flaws in the design which lead the design to be eliminated. This design was eliminated for the following reason:

- Its inability to satisfy the design compatibility criterion. The variance in height between the mounting bracket and the top of the bumper portion of the grille guard would force each grille guard to have a modified version of the latch system.

3.3 Sensitivity Analysis

Due to the number of successful designs for the latch mechanism and the fact that their scores were so closely grouped, further analysis of these designs was required in order to determine how to proceed. A sensitivity analysis was performed to re-evaluate the most successful designs. This second evaluation utilized a different method of scoring the concept designs for each criterion.

The sensitivity analysis utilized the hit matrix evaluation method to determine which design performed best for each criterion. A hit matrix was constructed for each criterion to rank the designs head-to-head in that category. Each hit was multiplied by the criterion weights established in Appendix A, and then summed to provide an overall score to rank the designs from strongest to weakest. This process not only established which design was strongest for each criterion, but also which design was the strongest overall. TABLE VII below shows the comparison of the top three latch designs for the latch strength criterion.

TABLE VII: THE HIT MATRIX COMPARING LATCH DESIGNS FOR LATCH STRENGTH

LATCH STRENGTH		Simple Drop Pin	Advanced Tow Pin	Spring Loaded Latch - Horizontal
		A	B	C
Simple Drop Pin	A		B	A
Advanced Tow Pin	B			B
Spring Loaded Latch - Horizontal	C			
HITS		1	2	0
SCORE		0.167	0.333	0

The Advanced Tow Pin design came first with two hits while the simple drop pin placed second. These two designs are very similar, and based off the same governing principal. The Advanced Tow Pin, simply expands on the Simple Drop Pin design, adding in a spring-loaded friction latch for additional safety and strength. Both designs rely on shear stress through the bar which results in high strength latch concepts.

For certain criteria, each design's performance was so close that there was no clear winner, therefore no hits were rewarded. This can be seen below in TABLE VIII which compares the latch designs for their ability to reduce the operating time.

TABLE VIII: THE HIT MATRIX COMPARING LATCH DESIGNS FOR OPERATING TIME

REDUCE OPERATING TIME		Simple Drop Pin	Advanced Tow Pin	Spring Loaded Latch - Horizontal
		A	B	C
Simple Drop Pin	A		-	-
Advanced Tow Pin	B			-
Spring Loaded Latch - Horizontal	C			
HITS				
SCORE		0	0	0

This result occurred since all the designs would greatly reduce the operating time to lock and unlock the grille guard, compared to the current eye bolt system. The difference between the operating time for each design would be small, and for our purposes at this point, considered negligible.

The design compatibility was another critical criterion used to determine the latch design success. The hit matrix evaluating the conceptual latch designs based on their compatibility is show in TABLE IX.

TABLE IX: HIT MATRIX COMPARING THE LATCH SYSTEM FOR DESIGN COMPATIBILITY

DESIGN COMPATIBILITY		Simple Drop Pin	Advanced Tow Pin	Spring Loaded Latch - Horizontal
		A	B	C
Simple Drop Pin	A		A	A
Advanced Tow Pin	B			B
Spring Loaded Latch - Horizontal	C			
HITS		2	1	0
SCORE		0.333	0.167	0

The Simple Drop Pin and Advanced Tow Pin designs were both award hits for this matrix due to their proven compatibility with S&V Manufacturing's current line of Bison Grille Guards. These two designs work on the principal of the standard tow pin which is already incorporated in all grille guard designs. The tow pin hole, which is already in place on current S&V Manufacturing grille guard models can be seen below in Figure 8.

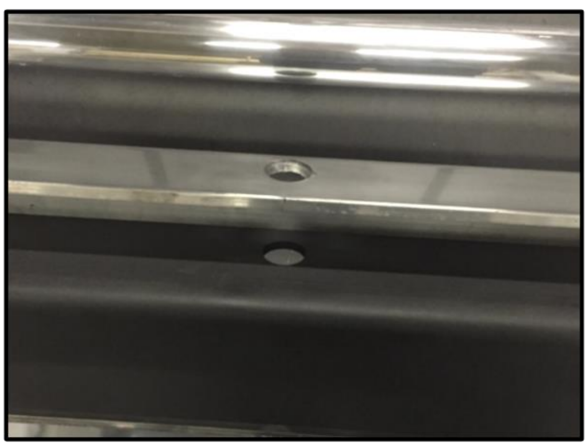


Figure 8: Tow Pin Hole on the S&V Manufacturing Grille Guards and Mounting Brackets [1]

Both designs utilize a similar hole placed near the center of the top face on the grille guard. The Simple Drop Pin received a higher score than the Advanced Tow Pin since it requires no additional fixturing. The Spring-Loaded Latch – Horizontal design did not receive any hits, due to the variability of the height between the grille guard and mounting bracket for different grille guard models. This variability in terms of height would require customization of the latch mechanism for each grille guard model.

The final rankings for the latch system designs were:

1. Advanced Tow Pin – 0.917
2. Simple Drop Pin – 0.778
3. Spring Loaded Latch – Horizontal – 0.139

These results were presented to the client, so their input could be considered before a final design was selected. The client agreed with the evaluation method of the latch designs. With client's approval, the Advanced Tow Pin design has been selected as the final design to be further optimized. However, the client again introduced additional information to consider. There were complaints that some competitor latch products have been failing due to loosening over time from vibration from the vehicles. Based on this knowledge, the client requested that the latch should minimize vibrations in the grille. Through further discussion, the team and client agreed that the Advanced Tow-Pin design could accomplish this with tight tolerancing, and with the aid of the spring.

3.4 Final Design Selection

The final latch design being utilized for optimization is the Advanced Tow Pin concept. A model of this design can be seen below in Figure 9. In this figure the grey rectangular surface at the top of the model is representative of the grille guard bumper, while the black rectangular surface is representative of the mounting bracket.

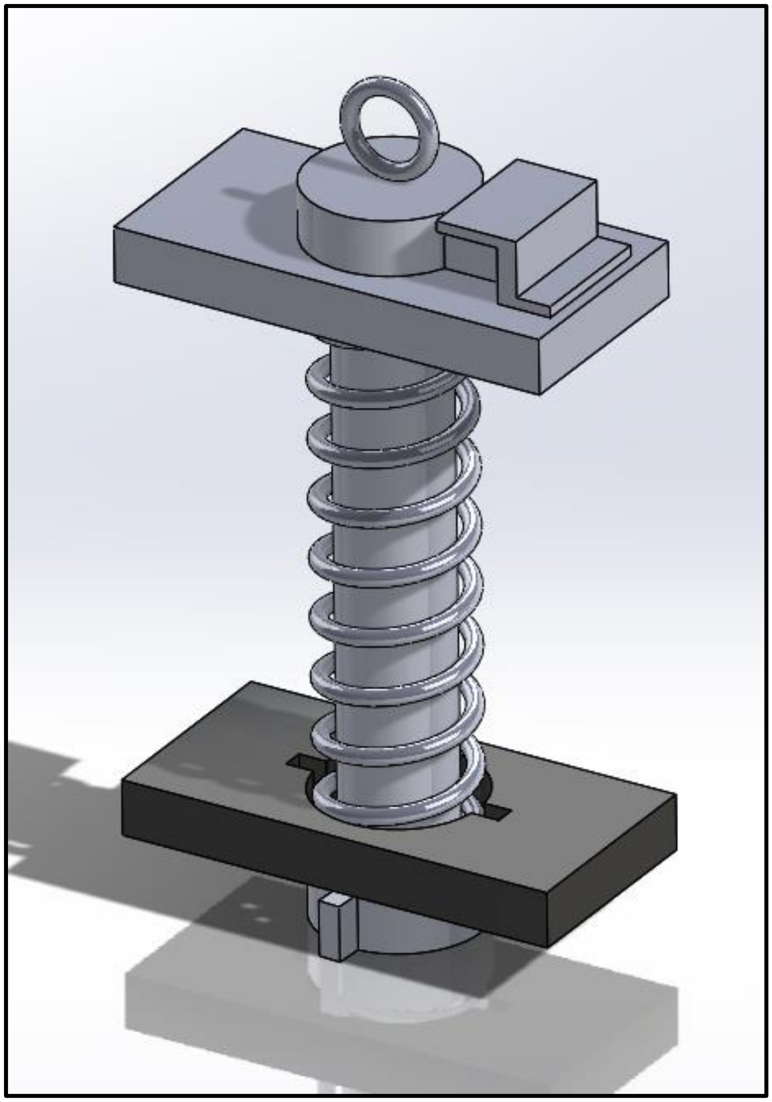


Figure 9: The Advanced Tow Pin Design Concept [2]

This design works based on the already proven tow pin which is currently being utilized by the S&V Manufacturing. The tow pin, which is dropped vertically through a hole in the grille, and then two holes in the mounting bracket, is used as the main method of bearing the load of the grille guard. It also functions as a tow hook if the need arises. However, due to its current simplicity and the tendency for it to be misplaced by the operators, it is supplemented with two 20 [mm] eye bolts that screw in through the front of the bumper.

[redacted]

Figure 10: Current Tow Pin and Eyebolt Configuration [5]

The Advanced Tow Pin would eliminate the need for the 20 [mm] eye bolts by improving upon the current tow pin design, while retaining the intended towing functionality.

The advanced tow pin would keep the same tow pin dimensions and load bearing surfaces, while adding a keyway to stop the tow pin from being able to be fully separate from the grille guard as seen below in Figure 11.

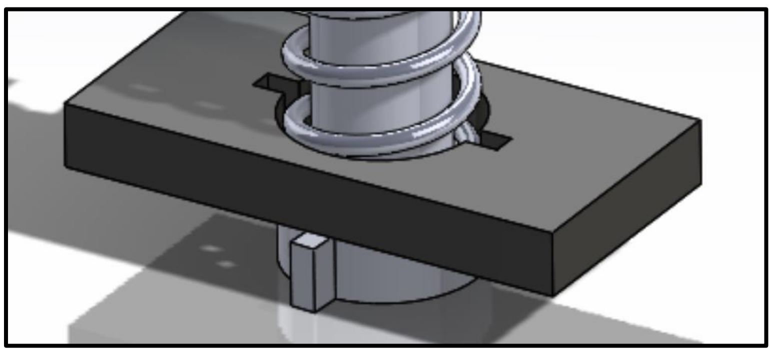


Figure 11: The Locking Keyway on the Advanced Tow Pin Design [2]

This keyway would be paired with a turn-to-lock system on the top of the grille bumper, adding additional security, and strength to the tow pin. When the operator of the vehicle intends to lower the grille, they would simply turn the tow pin to release it. It would then raise to a neutral position, where it would be held by a spring (secured to the pin keyway) to keep it clear from interfering with the mounting bracket as the grille is raised and lowered. Additionally, this spring would provide tension to the system when in the locked position to help eliminate vibrations occurring from road conditions. The Advanced Tow Pin in the unlocked position can be seen below in Figure 12.

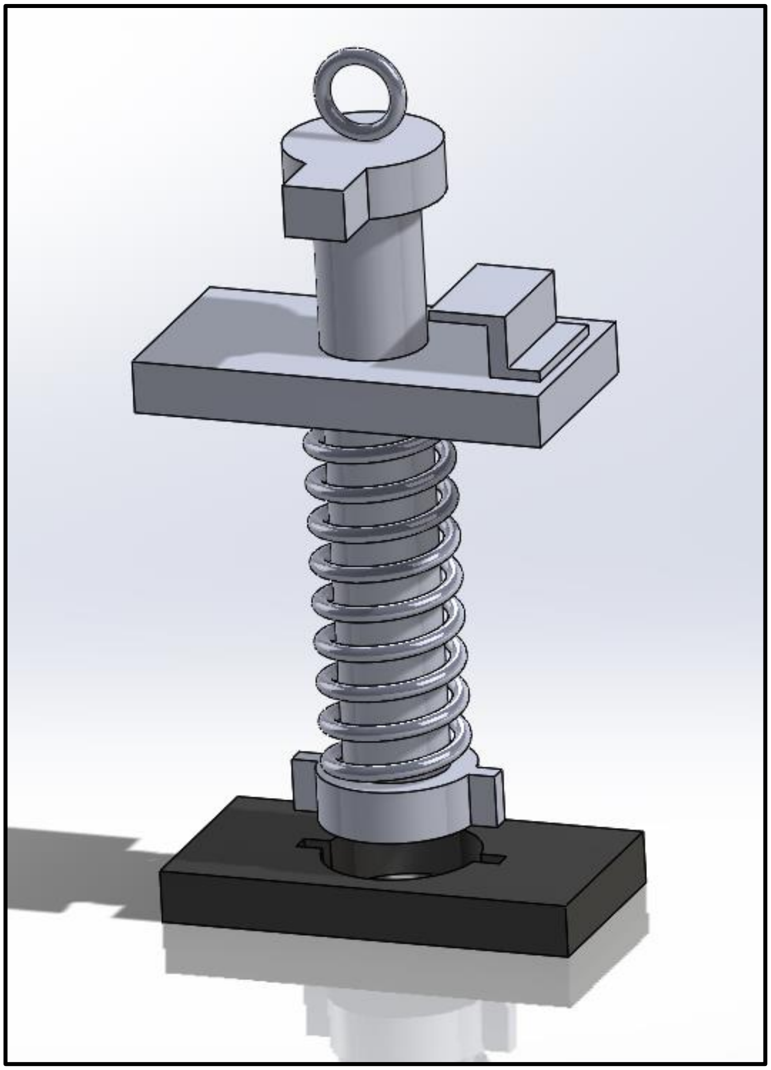


Figure 12: The Advanced Tow Pin in the Unlocked Position [2]

4 Conclusion

In conclusion, the Hinge Fixed Single-Spring was selected as the final lift assist design while the Advanced Tow Pin was selected as the final latch mechanism design. The Hinge Fixed Single-Spring lift assist design utilized two torsional springs attached at the mounting bracket's hinge point. One spring arm is attached to the grille guard, while the other is secured to the mounting bracket. It is design so that the spring will deflect as the grille guard is lowered which will reduce the applied force that is required. The Advanced Tow Pin design is a redesign of the current S&V Manufacturing tow pin. The Advance Tow Pin uses a series of keys and keyways on the mounting bracket and grille guard to easily lock and unlock grille guard. This removes the need for the eye bolts which are used in the current system to secure the grille in the raised position. The final lift assist, and latch mechanism design were determined by evaluating the conceptual designs using the design criteria from Appendix A. Input from S&V Manufacturing also led to the determination and selection of the Hinge Fixed Single-Spring and Advanced Tow Pin designs.

Works Cited

[1] M. Mahon. *S&V Manufacturing Grille Guard Image*. Winnipeg: Design Eng., Univ Manitoba, Winnipeg, MB, September 2017.

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[3] Lee Spring Limited. *Designing & Specifying Compression, Extension and Torsion Springs* [Online]. Available: http://www.leespring.com/downloads/uk/leespring_engguide.pdf. [19 November 2017].

[4] Truck Tire Carrier. *Semi-Truck Deer Guard* [Online]. Available: http://semitruckdeerguard.blogspot.ca/2013/09/semi-truck-deer-guard.html. [25 October 2017].

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Appendix C

Analytical Analysis

Table of Contents

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1 Introduction

This appendix discusses the preliminary analysis performed to determine the spring parameters required for the proposed lift assist and latch mechanism designs. The initial static and dynamic analysis methods utilized to determine the appropriate spring rates for the lift assist and latch mechanisms are outlined in the following sections. The calculations used to determine all additional spring parameters such as the wire diameter, inner and outer coil diameters, spring deflection, number of coils, and the spring pitch are provided.

2 Static and Dynamic Analysis

Two different types of preliminary analysis were performed on the conceptual lift assist design in order to determine the amount of torque required by the springs. A static analysis was used to determine the spring torque required for a constant operator applied force. This method is also used in reverse to determine the applied force required for a constant spring rate. An energy analysis approach was also taken to determine how much the energy was required to raise the grille guard from the lowered position, and how much of that energy was handled by the spring. A combination of these two analytical methods were what was used to determine the springs needed for the lift assist mechanism.

A static analysis method was also used to determine the spring rate needed for the latch mechanism extension spring. That analysis is also outlined in this section.

2.1 Static Analysis

A static analysis was conducted to determine the amount of torque required by the spring to raise the grille under the given conditions. The free body diagram (FDB) of the grille guard can be seen in Figure 1. This diagram shows the three major forces applied to the grille guard which include; the force applied by the operator (F_A), the weight of the grille guard (W_G), and the torque applied by the spring (T_S).

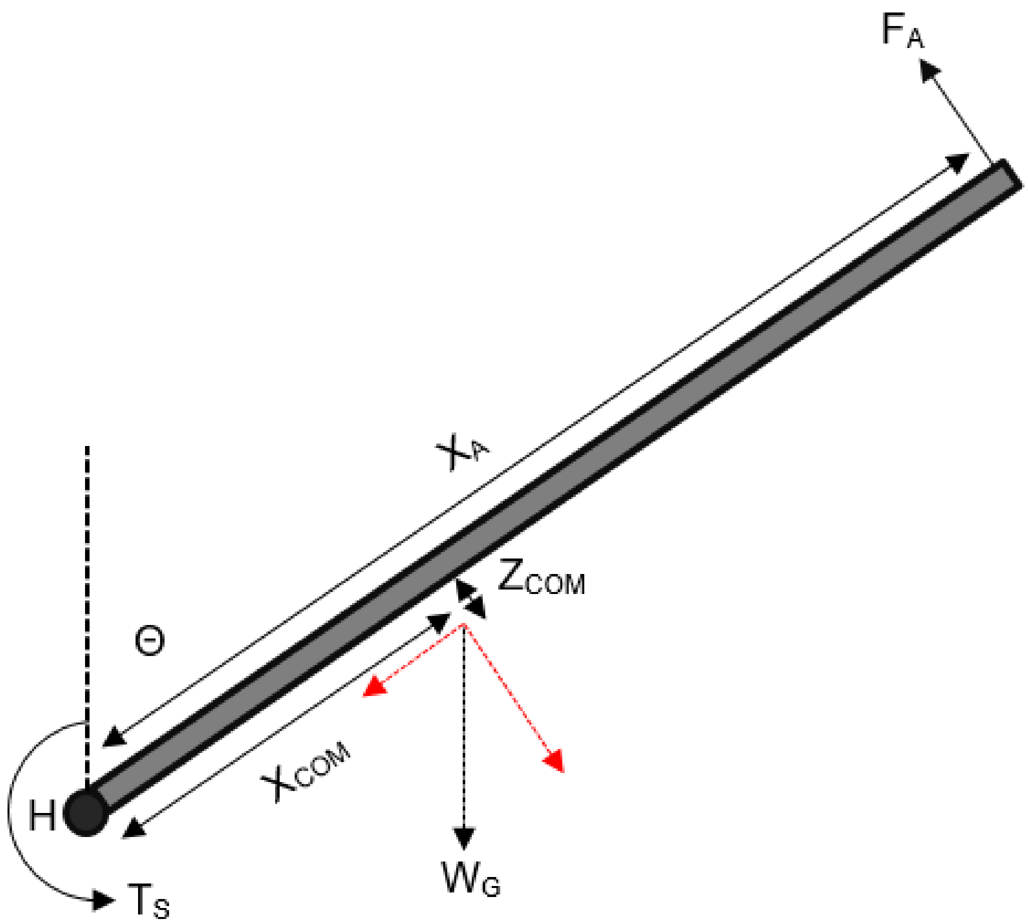

Figure 1: Free Body Diagram of the Bison Grille Guard [1]

This analysis assumes that the force applied by the operator, F_A, is applied at the top edge of the grille guard, perpendicular to the front face. The location of the applied force remains constant throughout the motion of the grille. This assumption was made to simplify the analysis as well as to remove the uncertainty of how the operators choose to raise the grille.

A moment balance about the hinge point, H, was performed based on the FDB shown in Figure 1. The result of the moment balance about point H is shown in the equation below.

$$\sum M_H = 0 \rightarrow T_S = W_G(X_{COM}\sin\theta + Z_{COM}\cos\theta) - F_A X_A$$

This equation can be used to determine the required torque at any angle for any grille model as long as the dimensions and grille guard mass are known. For our purposes, this equation was used to determine the torque required by the spring at various angles for grille models ranging from 175 [lbs] to 225 [lbs] when a constant force of 20 [lbf] was applied by the operator. Equation 1 was then combined with Equation 2 [2] in order to determine the spring rate, K, required for the design. It is important to note that this refers to the equivalent spring rate K_{eq} as the number of springs which will be used in the design is not considered in this section of the analysis.

$$T_S = K_{eq}\theta$$

A 3D model of a S&V Manufacturing Bison Grille Guard was created based on drawings given by the client. Unfortunately, the drawings were not completely accurate as they are missing some critical dimensions. It should be noted that there were also no engineering drawings for the majority of the grille guards manufactured at S&V Manufacturing. After discussing with the client, we learned that the design of each grille is not documented, and each grille is more or less unique to the vehicle that it is being mounted too. Therefore, we were only given rough dimensions to use moving forward. A model was created using the dimensions that were available from the client.

This grille guard model had a total weight of 190.21 [lbs], and a center of mass (COM) with and X-component and Z-component of 0.323 [m] and 0.09884 [m], respectively. The grille guard also had a height of 1.26 [m]. Applying that information to Equation 1 determined how the required torque from the spring was affected by the angle of the grille guard.

Since S&V Manufacturing was unable to provide us with the dimensions of the smallest or largest grille models, it was assumed for our purposes that the dimensions of the grille guards do not change when the weight is increased. It is understood that this is not the case however it was the best available option with the resources provided. Therefore, the largest and smallest grille guard were analyzed using weights of 225 [lbs] and 175 [lbs], respectively. The results of this analysis are shown in TABLE I.

TABLE I: SPRING RATE CALCULATED USING STATIC ANALYSIS

Grille Model	Θ [rad]	$W_G\cos\Theta$ [N]	$W_G\sin\Theta$ [N]	T_S [Nm]	K_{eq} [Nm/rad]
Grille 1 [190 lbs]	1.57	0.00	846.41	161.25	102.65
	1.37	165.13	830.14	172.32	125.37
	1.18	323.91	781.98	172.45	146.38
	0.98	470.24	703.76	161.65	164.66
	0.79	598.50	598.50	140.33	178.68
	0.59	703.76	470.24	109.31	185.56
	0.39	781.98	323.91	69.77	177.67
	0.20	830.14	165.13	23.25	118.40
	0.00	846.41	0.00	-28.48	
Grille 2 [175 lbs]	1.57	0.00	778.72	139.39	88.74
	1.37	151.92	763.75	149.57	108.82
	1.18	298.00	719.44	149.69	127.06
	0.98	432.63	647.48	139.76	142.36
	0.79	550.64	550.64	120.14	152.97
	0.59	647.48	432.63	91.60	155.50
	0.39	719.44	298.00	55.22	140.63
	0.20	763.75	151.92	12.42	63.25
	0.00	778.72	0.00	-35.17	
Grille 3 [225 lbs]	1.57	0.00	1001.21	211.25	134.49
	1.37	195.33	981.97	224.34	163.22
	1.18	383.15	925.00	224.50	190.56
	0.98	556.24	832.47	211.73	215.66
	0.79	707.96	707.96	186.51	237.47
	0.59	832.47	556.24	149.81	254.32
	0.39	925.00	383.15	103.04	262.40
	0.20	981.97	195.33	48.01	244.50
	0.00	1001.21	0.00	-13.18	

The results shown in TABLE I reveal a large variation in equivalent spring rate ranging from 63.25 [Nm/rad] to 254.32 [Nm/rad].

Equation 1 was rearranged into Equation 3 which was used to determine the required force for a given equivalent spring rate.

$$F_A = \frac{W_G(X_{COM}\sin\theta + Z_{COM}\cos\theta) - K_{eq}\theta}{X_A}$$

The required applied force for six different grille models ranging from 175 [lbs] to 225 [lbs] were plotted to show how the applied force varies throughout the motion of the grille guard. Like in the previous calculations, the dimensions and the locations of the center of masses do not change between grille models, only the mass changes based on our assumption. Plotting Equation 3 for various grille guard weights was an iterative process to observe how the equivalent spring rate effected the required force and to select a spring rate which satisfies the design criteria. The iterative process determined that the maximum allowable equivalent spring rate was 160 [Nm/rad]. With this spring rate, the require force is less than 20 [lbf] for all grille guard models in the lowered position. It is important to note that the required force is greater than 20 [lbf] for certain models during the process of raising the grille guard. However, if the spring rate was increased to ensure that the required force never exceeded 20 [lbf], the lighter grilles would not be able to be lowered to the full 90º as the spring would be too stiff. The results, as presented in Figure 2, show that there is no single spring that would be able to ensure that the required applied force would be 20 [lbf] or less. Using an equivalent spring rate of 160 [Nm/rad] results in a maximum applied force of 30.86 [lbf] for the 225 [lbs] grille guard model at a grille angle of 45º, and a minimum applied force of 0.04 [lbf] for the 175 [lbs] grille guard model at a grille angle of 90º. If the equivalent spring rate was less than 160 [Nm/rad], both the maximum and minimum required forces would be increased. If the equivalent spring rate was greater than 160 [Nm/rad], the maximum required force would decrease, and the minimum required force would become negative, meaning the force would be in the opposite direction holding the grille in the lowered position.

Figure 2: Force Vs. Grille Angle for Varying Grille Guard Weights

2.2 Energy Analysis

An energy analysis was performed to determine how much energy is currently being used to raise and lower the grille guards, and how much the energy will be reduced by the addition of a lift assist system. The energy that is currently required to raise the grille is the work to move the grille from 90^o to 0^o. The formula used to calculate work is shown in Equation 4 where F is the force, and d is the distance [3].

$$W = Fd$$

However, since the grille guard is rotated about a hinge point the force will become torque, T, and the distance will become the angle, Θ, as shown in Equation 5.

$$W = T\theta$$

As shown previously in Equation 1, the torque required to raise the grille guard is not constant throughout the motion of the grille. Therefore, the torque, T, is given by Equation 6 where W_G is the force caused by the mass of the grille guard, X_{COM} and Z_{COM} signify the location of the center of mass, and Θ is the grille guard angle.

$$T = W_G(X_{COM}\sin\theta + Z_{COM}\cos\theta)$$

Combining Equation 5 and Equation 6 and integrating the resulting formula, yields the equation for the work required to raise or lower the grille guard. This is shown in

$$W = \int_{0^o}^{90^o} W_G(X_{COM}\sin\theta + Z_{COM}\cos\theta)d\theta = W_G(X_{COM} + Z_{COM})$$

Based on Equation 7, the energy required to raise or lower the lightest and the heaviest grille guard models without a lift assist mechanism is 328.5 [J] and 422.3 [J], respectively.

The energy provided by the spring is given by the following equation [2].

$$E_S = \frac{1}{2}K\theta^2$$

Since we were looking for a spring rate, K, which would provide a required force of less than 20 [lbf] throughout the motion of the grille guard, we calculated how much energy that would take. Using Equation 5 it was determined that lowering or raising the grille guard by only applying a force of 20 [lbf] uses 176.1 [J]. Therefore, to raise the heaviest grille guard using a maximum applied force of 20 [lbf], the lift assist system would need to supply 246.2 [J] of energy. Based on Equation 8, the spring would have to have an equivalent spring rate of 218.8 [Nm/rad] which is far greater than what is suitable for the design as shown in section 1.

Using an equivalent spring rate of 160 [Nm/rad] would require an operator force of 14.9 [lbf] and 25.7 [lbf] to raise or lower the lightest and heaviest grille guard models, respectively.

3 Spring Design

This section outlines the design of the springs for both the lift assist and latch mechanisms. These springs were designed based off Lee Spring Limited's Engineering Guide for Designing & Specifying Compression, Extension and Torsion Springs [4].

3.1 Torsional Spring

The torsional spring was designed based on two major constraints; the inner coil diameter, D_i, which needed to be 30 [mm] to fit around the spring housing, and the torsional spring rate, S_θ, which needed to be approximately 1.396 [Nm/deg] per spring based on the following calculation, manipulated from the total 160 [Nm/rad] ideal total spring rate to lift the necessary grille models:

$$S_\theta = \frac{160\ Nm}{radian} \times \frac{\pi\ radians}{180°} \div \frac{2\ hinges}{assembly} \times \frac{1\ spring}{hinge} = 1.396\ \frac{Nm}{degree}\ per\ spring$$

With these parameters, an excel solver was utilized to determine the necessary dimensions to create a spring with the required spring rate. The governing equation for this solver was:

$$S_\theta = \frac{Ed^4}{3667nD}$$

Where E is the elastic modulus of the spring material [N/mm^2], d is the wire diameter of the spring [mm], n is the number of active spring coils, and D is the mean coil diameter of the spring [mm]. The mean coil diameter, D, was calculated using the following equation, based on the 30 [mm] inner diameter (D_i) constraint:

$$D = \frac{D_i + D_o}{2} = \frac{D_i + (D_i + d)}{2} = D_i + \frac{d}{2}$$

This result is a final spring rate equation of:

$$S_\theta = \frac{Ed^4}{3667nD} = \frac{Ed^4}{3667n(D_i + \frac{d}{2})}$$

This equation was solved by iteratively increasing the spring wire diameter, d, by 0.25 [mm] from zero to 2.5 [mm], while also increasing the active number of coils by the equation:

$$n = 0.5 + \mathbb{Z}, (where\ \mathbb{Z} = 0,1,2 \ldots)$$

The wire diameter was adjusted in this way in order to obtain a final diameter which would match an industry stock wire diameter. The equation for the active number of coils was used in order to result in a spring whose two moment arms were located 180° away from one another, as this was a requirement of the lift assist mechanism design. The spring material utilized was commercial chrome silicon, which has a modulus of elasticity, E, of 207,000 [MPa]. The excel solver utilized, yielded the spring specifications seen in TABLE II based off the above equations.

TABLE II: DESIGNED SPRING PARAMETERS FOR THE TORSIONAL SPRING.

D_i	Inner Diameter	30.0	mm
d	Spring Wire Diameter	8.5	mm
D_o	Outer Diameter	47.0	mm
D	Mean Diameter	38.50	mm
E	Modulus of Elasticity	207,000	N/mm
n	Active Coils	5.5	#
S_θ	Torsional Spring Rate	1.392	Nm/°

Additionally, it was decided the spring would have a helical pitch, p, of 12 [mm]. This decision was made to reduce the friction between the active coils of the spring. The minimum pitch for a spring is its spring wire diameter, resulting in the coils helically stacking upon one another. Increasing the pitch increases the active spring length, resulting in a better distribution of the springs applied force, while also reducing friction, and increasing the life of the spring [4]. Based on the pitch, the body length of the spring, L_0, was calculated.

$$L_0 = (n + 1)p = (5.5 + 1)12 = 78\,mm$$

It was also decided that the spring would have one axial moment arm (Figure 3), parallel to the helical axis of the spring, while the other moment arm would extend radially from the spring helix (Figure 4). The two moment arms would be located 180° away from one another.

Figure 3: Axial Spring Moment Arm [4]

Figure 4: Radial Spring Moment Arm [4]

3.2 Extension Spring

The extension spring for the latch mechanism needs to be able to accomplish a number of different functions. When the latch mechanism is unlocked, it needs to be able to hold the tow pin extended less than 10 [mm] below the surface of the grille guard. The spring also needs to be able to deform so it could be pushed all the way to the lock position. It was determined that the human force required to lock the spring should be approximately 31 pounds, to mirror the maximum force utilized to lift the grille guard.

Knowing extension springs are governed by Hooke's Law, the distances between major features needed to be established in order to properly design this extension spring. Critical distances and lengths for this design are included in TABLE III.

TABLE III: CRITICAL DISTANCES FOR THE EXTENSION SPRING

L_p	Pin Length	258.83	mm
h_t	From Grille Tube to Pin Hole	220	mm
h_h	From Grille Tube to Hanging Pin Position	73.87	mm
h_l	From Grille Tube to Locked Pin Position	215.24	mm
L_b	Length from Grille to Mounting Bracket	10	cm

These critical distances can be visualized in Figure 5.

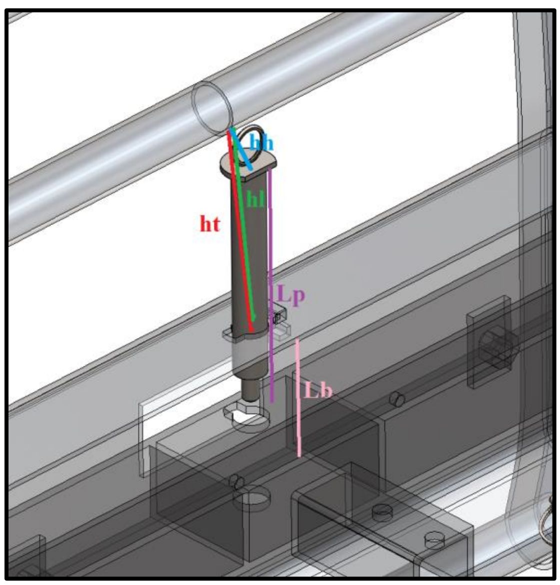

Figure 5: Critical Extension Spring Distances [5]

The spring rate, K, needed could be calculated based on the free body diagram of the pin in the hanging position. This free body diagram can be seen in Figure 6, where the weight of the pin is 2.5 [kg].

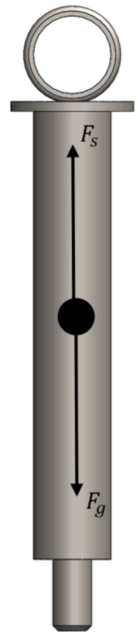

Figure 6: Free Body Diagram for the Pin in the Hanging Position [5]

F_s is the force applied by the extension spring, and F_g is the force due to gravity acting on the pin. Since the pin is hanging in this position, the equation of force in the y-direction is as follows:

$$F_y = 0 = F_s - F_g = K \cdot x - m \cdot g = K \cdot x - 2.5\ kg \cdot 9.81\ m/s^2$$

or,

$$K = \frac{24.525\ N}{x}$$

In order to solve this equation for the spring rate K, the deformation of the spring, x, must be known. The deformation of the spring is expressed by the equation:

$$x = h_h - L_o = 78.87 - L_0$$

Where L_0 is the free body length of the spring, or the spring in a relaxed state. The free body length can be calculated by the equation:

$$L_0 = (n + 1)d + 2(D - d)$$

Where d is the spring wire diameter, D is the mean spring coil diameter, and n is the active number of coils. Even with these relationships, the spring rate equation from the free body diagram has many unknowns (S, n, D, and d) as seen below.

$$S = \frac{24.525}{h_h - (n + 1)d + 2(D - d)}$$

Therefore, it was decided the spring rate would be solved for iteratively to assure that the critical distances above were satisfied. The length from the hanging pin to the grille, h_h was constrained so it would be as close to, without exceeding, 73.87 [mm]. This would assure that there would always be proper clearance between the pin and the mounting bracket when raised and lowered. The wire diameter, *d*, was iteratively increased by a factor of 0.125 [mm] from zero to a maximum of 1 [mm]. The number of coils *n*, was iteratively increased by integer values. As well, the amount of human force needed to push the pin all the way to the lock position was set to be minimized with an upper value of 31 pounds of force applied. The equation used to set this constraint was:

$$F_l = S \cdot (h_l - L_0) = \frac{24.525}{h_h - L_0}(h_l - L_0)$$

With all these constraints, the iterative solver created in excel yielded a final spring rate of 0.731623 [Nm], a required locking force of 28.88 [lbs], a spring wire diameter, *d*, of 0.75 [mm], an inner coil diameter, D_i, of 4 [mm], a mean coil diameter, *D*, of 4.75 [mm], 40 active spring coils, *n*, and a length from the grille tube to the pin, h_h of 72.27 [mm]. The spring length results in a distance between the pin and the grille guard of 98.40 [mm]. This provides sufficient clearance between the pin, and the mounting bracket and grille guard. The free body length of this spring was calculated to be 38.75 [mm].

This spring rate calculated using a second method outlined by Lee Spring [4] to verify the results of the first calculation.

$$S = \frac{Gd^4}{8nD^3}$$

Where *G* is the modulus of rigidity. The modulus of rigidity for music wire, the material used for the proposed spring, is 79,300 [MPa]. Using the spring parameters calculated above, this equation yields an identical extension spring rate of 0.731623 [Nm]. This confirms the design methodology used for the extension spring.

TABLE IV: DESIGNED SPRING PARAMETERS FOR THE EXTENSION SPRING

D_i	Inner Diameter	4.00	mm
d	Spring Wire Diameter	0.75	mm
D_o	Outer Diameter	5.50	mm
D	Mean Diameter	4.75	mm
E	Modulus of Rigidity	79,300	N/mm
n	Active Coils	40.00	#
L_0	Free Body Length	38.75	mm
S	Spring Rate	0.7316	N/mm

4 Spring Selection

This section outlines the spring parameters that will be implemented in the final lift assist and mechanism designs. All spring parameters shown in this section were finalized and quoted by Western Spring and Wire Ltd. All revised spring parameters and manufacturing costs of the springs is shown along will the finalized energy analysis of the lift assist mechanism.

4.1 Torsional Spring

The final torsional spring parameters based on the quote provided by Western Spring and Wire can be seen in TABLE V. Based on these spring parameters, the total energy stored by these springs will be 190.93 [J]. This will result in an operator energy reduction ranging from 45.2 [%] to 58.1 [%], depending on the grille model. These springs will also result in a weight reduction in the lowered position of 75.2 [%] to 96.7 [%], depending on the grille model.

TABLE V: FINAL TORSIONAL SPRING PARAMETERS [6]

Parameter	Value	Unit
Spring Material:	Chrome Silicon	-
Wire Diameter:	8.71	mm
Wire Length:	803.06	mm
Spring Weight:	0.38	kg
Coil Inner Diameter:	30.0	mm
Coil Outer Diameter:	47.42	mm
Pitch:	12.0	mm
Number of Active Coils:	5.86	-

The full quote provided by Western Spring and Wire Ltd. can be seen in Appendix H.

4.2 Extension Spring

The final extension spring parameters based on the quote provided by Western Spring and Wire can be seen in TABLE VI. Western Spring and Wire Ltd. advised that this spring design is overstressed due to the required deflection of the spring and that it will likely fail. This will be presented to S&V Manufacturing so that they are aware of the risks associated with the spring component of the latch mechanism.

TABLE VI: FINAL EXTENSION SPRING PARAMETERS [6]

Parameter	Value	Unit
Spring Material:	Music Wire	-
Wire Diameter:	0.75	mm
Coil Inner Diameter:	4.0	mm
Coil Outer Diameter:	5.50	mm
Pitch:	0.75	mm
Number of Active Coils:	39.97	-

The full quote provided by Western Spring and Wire Ltd. can be seen in Appendix H.

5 Conclusion

In conclusion, the analysis shown in this section led to the determination that the lift assist mechanism would utilize two chrome silicon torsional springs with individual spring rates of 1.352 [Nmm/°]. These torsional springs have a wire diameter of 8.71 [mm], a pitch of 12.00 [mm], and an inner and outer diameter of 30.00 [mm] and 47.42 [mm], respectively. With these two torsional springs, the lift assist will reduce the energy to raise the grille by 45 [%] to 58 [%], depending on the grille model. The analysis also determined that the latch mechanism music wire extension spring has a spring rate of 0.732 [Nmm/°], and a pitch of 0.75 [mm]. This extension spring has an inner and outer diameter of 4.00 [mm] and 5.50 [mm], respectively. Due to Western Spring and Wire Ltd. advising this spring would be overstressed, it is not recommended to be utilized on the final design.

Work Cited

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Appendix D

Lift Assist Numerical Analysis

Table of Contents

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1 Introduction

This appendix discusses the finite element analysis methods utilized to verify the lift assist mechanism design. The following sections will outline the simulation setup utilized as well as all relevant considerations, the results from the simulation, an explanation and discussion on the results, and recommendations for future designs. All analysis below was run on the SolidWorks application, using an assembly simulation, which allows multiple parts to be analyzed together, with numerical consideration applied to the way parts mate and interact with one another. All the materials utilized for this analysis, with the relevant yield strength can be seen in TABLE I.

TABLE I: MATERIALS USED FOR NUMERICAL ANALYSIS

Item #	Description	Material	Yield Point
1	Locking Pin	18-8 Steel (AISI 304 Steel Equivalent)	206 MPa [1]
2	Grille Pin Housing	6061 T6 Aluminum	275 MPa [2]
3	Spring Housing	4140 Steel	890 MPa [3]
4	Grille Guard	6061 T6 Aluminum	275 MPa [2]
5	Mounting Bracket	44W Steel (ASTM A36 Steel Equivalent)	250 MPa [4]
6	Washer	4130 Steel	670 MPa [5]
7	Mounting Shaft	4140 Steel	890 MPa [3]

The lift assist mechanism needed to have some components simplified in order to conduct a proper analysis. As seen in Figure 1, the grille guard, mounting bracket, and locking pins needed to be simplified. Note, in Figure 1 the numbers on the corresponding features match the item numbers in TABLE I.

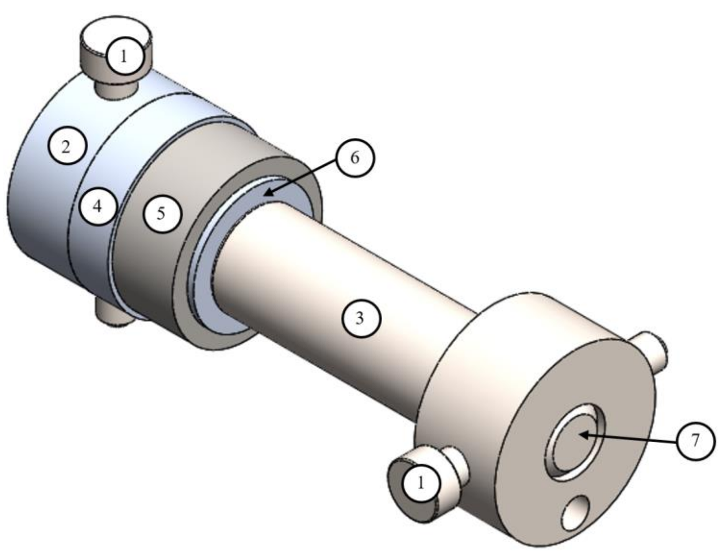

Figure 1: Simplified Lift Assist Mechanism Model [6]

The grille guard and bracket were simplified because in reality they are quite large. Since the majority of the structures are far away from the hinge point, they would not be experiencing any stress. As well, the simplified representation allowed for a much smaller number of mesh elements during the finite element analysis, leading to a more refined mesh overall and a shorter simulation time. The locking pin was simplified because the wire "locking" portion of its geometry has no impact on the simulation. The geometry is also complex, and fine geometry features could decrease the quality of the mesh overall if included. The spring which applies the stress to the system was removed from the simulation model. Instead, the resulting stresses from the deformation of the spring was subjected to the numerical simulation. This was done to avoid the large displacement and deformation that the spring will experience during the motion of the grille guard. Instead, a "non-linear" simulation was utilized. SolidWorks Nonlinear Stress Analysis allows for stresses and deformations to be calculated under dynamic loads, large component deformations, and non-linear materials, including springs [7]. This provided more accurate results for the team's design scenario, although it does increase the simulation time.

2 Simulation 1 - Setup

The following section will outline the setup for the finite element model utilized in the first numerical simulation.

2.1 Interference Inspection

Interference Detection is recommended when running a simulation on SolidWorks to identify interferences between components, especially in complex assemblies where it is difficult to determine interferences visually [8]. This is a necessary step because any unnecessary clearances or poor shrink fits can unexpectedly decrease the accuracy of the simulation. The result of the SolidWorks Interference Detection can be seen in Figure 2.

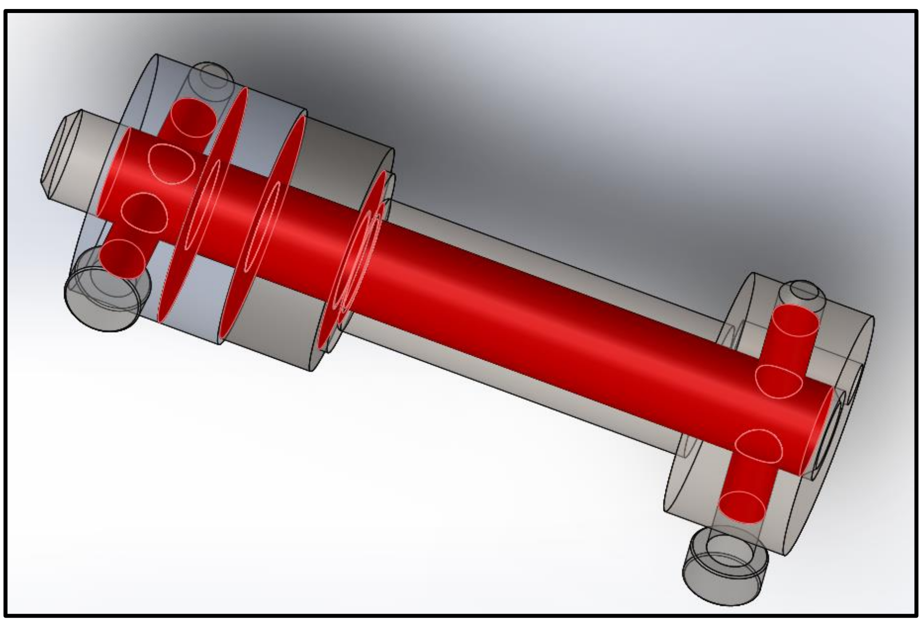

Figure 2: Results of the SolidWorks Interference Detection [6]

Figure 2 shows that there is no interference on the lift assist mechanism assembly, as all the coincident regions in the assembly are indicated in red. This means they match their required operational condition. The results of this inspection assure that the next step in the simulation process can begin, as the inspection results of the Interference Detection were satisfied.

2.2 Simulation Properties

The SolidWorks solver used was the FFEPlus solver, which uses an iterative approach to obtain the simulation solution. The iterations are run until SolidWorks finds that the error in the solution reduces to an acceptable value [9]. As well, the simulation was set to have surface to surface bonding contact in order to achieve a higher accuracy. The simulation was set to "improve accuracy for no penetration contacting surfaces (slower)". This setting means that the simulation is tailored for when contact between mating surfaces is not a solid bond. Rather, as soon as surfaces contact one another, the resulting reaction forces were calculated and considered [10]. Furthermore, the simulation implemented h-adaptive meshing which increased how fine the mesh was in the model around complex geometries or areas of high stress [11]. The "accuracy bias" was set to 80%, with the number of iterations set to two. The low number of iterations was set so that an acceptable mesh density could be manually improved based on analysis of the results from previous iterations of simulation. Each simulation iteration took 35 minutes to complete, therefore the "mesh coarsening" option was selected to increase the speed of the simulations, which allows the software to coarsen the mesh in areas where a highly defined mesh is not needed [11].

2.3 Component Connections

The global component contact for the simulation was set to "no penetration". Two additional "rigid connections" were also applied between each of the locking pins and their respective holes. A "rigid connection" connects faces of one solid body to another, only allowing them to deform rigidly as a group, with the distance between the two surfaces remaining constant [12]. This allowed the locking pin and its respective hole to act like a real-life pin, being properly retained by the structure around it. This relationship is visualized in Figure 3, where the blue geometries are the simplified locking pins.

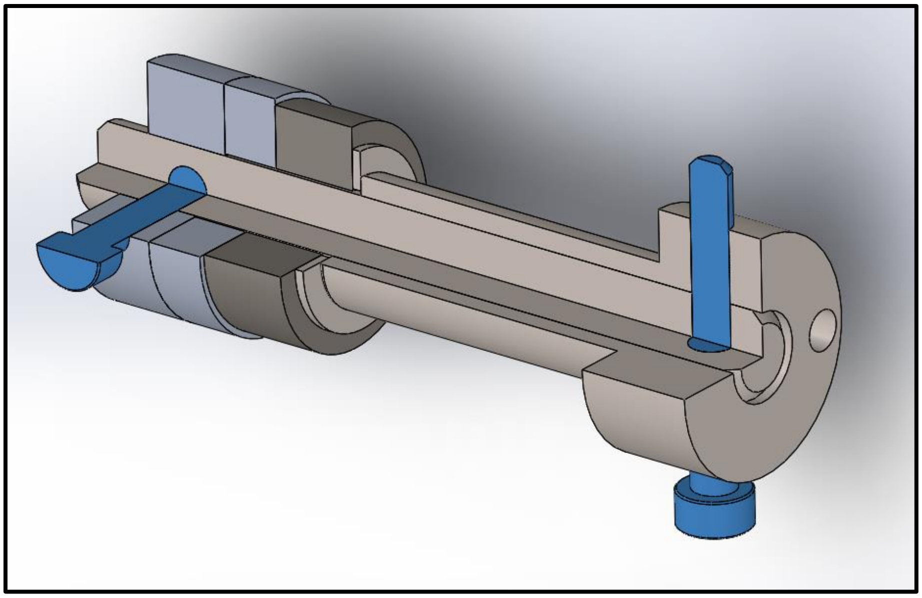

Figure 3: Simplified Locking Pin Simulation Assembly [6]

2.4 Mesh Set-up

The initial mesh set prior to the first h-adaptive loops was a standard global mesh (uniform mesh) of 3.8 [mm] with a tolerance of 0.2 [mm]. Because of the coarseness of this mesh it required further refinement for following simulation studies. This was true when the stress concentrations were from bending the mounting shaft and spring housing were considered. The mesh was further refined to have a local mesh control with a mesh size of 1.8 [mm] and an aspect ratio of 1.5 in the highlighted area seen in Figure 4. The coarse global mesh was used everywhere else on the simulation model.

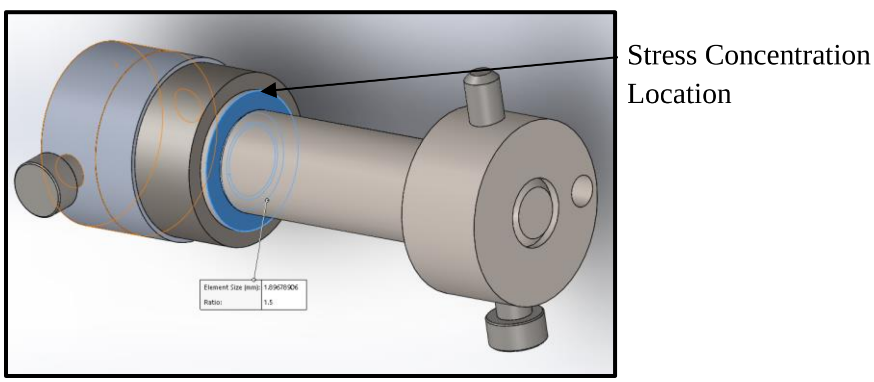

Figure 4: Location of the Local Mesh Control [6]

2.5 External Loads and Fixtures

There were two external loads that would be applied to the grille guard. The first was the load of the weight of the grille guard itself, while the second was the torque provided by the lift assist mechanisms torsional spring. Because the grille guard is such a complex geometry, and due to the necessity of customization of grille guard design by S&V Manufacturing it was not possible to obtain the exact force that was applied by the grille to the lift assist system. However, since a similar system is already used by S&V Manufacturing to hold and fix the grille guard to the mounting bracket, it was assumed it would be adequate to only test the torque applied to the lift assist mechanism.

The simulation model has two fixtures utilized. The first fixture was the surface of the simplified grille guard coincident to the mounting bracket, while the other fixture was the outer surface of the simplified mounting bracket as seen in Figure 5.

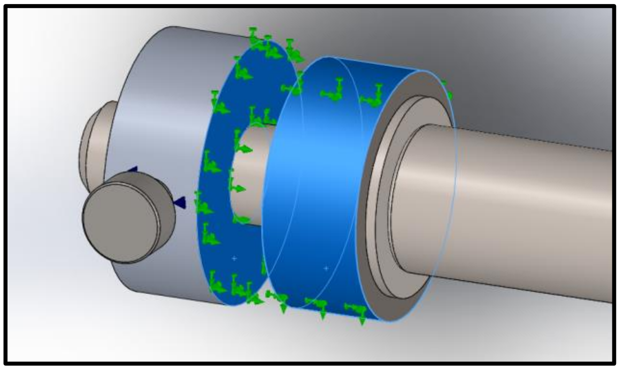

Figure 5: Simulation Fixture Set-Up [6]

This fixture scenario allowed these pieces to move and operate like a hinge fixture. This fixture scenario was selected instead of the built in SolidWorks hinge fixture since this allowed the assembly to mimic the actual intended movement of the lift assist mechanism. The applied torque on the model can be seen in Figure 6. The applied torque was created by applying an equivalent 6,000 [N] force in the negative Z-axis direction.

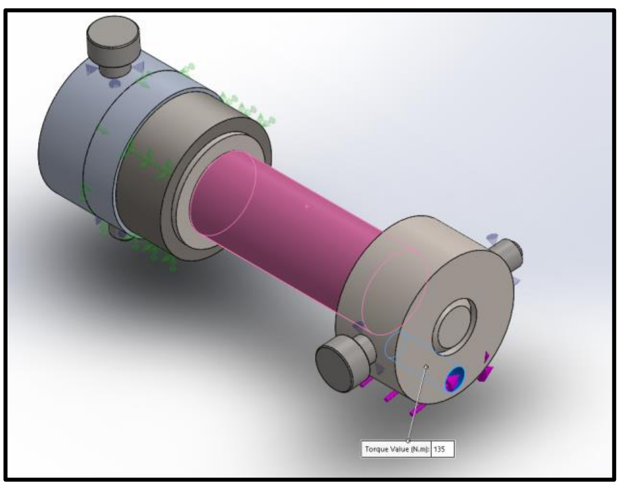

Figure 6: External Applied Torque Simulating the Torque from the Torsional Spring [6]

3 Simulation 1 – Results and Analysis

With the conclusion of the h-adaptive simulation, the strain energy error reported by SolidWorks was less than 5.3 [%], suggesting a successful simulation. However, despite this, the maximum resulting von Mises stress did not converge. This was due to a stress concentration issue which will be discussed. This would be avoidable if the SolidWorks h-adaptive simulation were able to record the stress values throughout the duration of the simulation, because better convergence was calculated under a more limited number of iterations. However, this is not the case.

3.1 General Result

Figure 7 shows the stress results on the exterior of the simulation model. The maximum stress of 139.5 [MPa] was located on the external portion of the assembly.

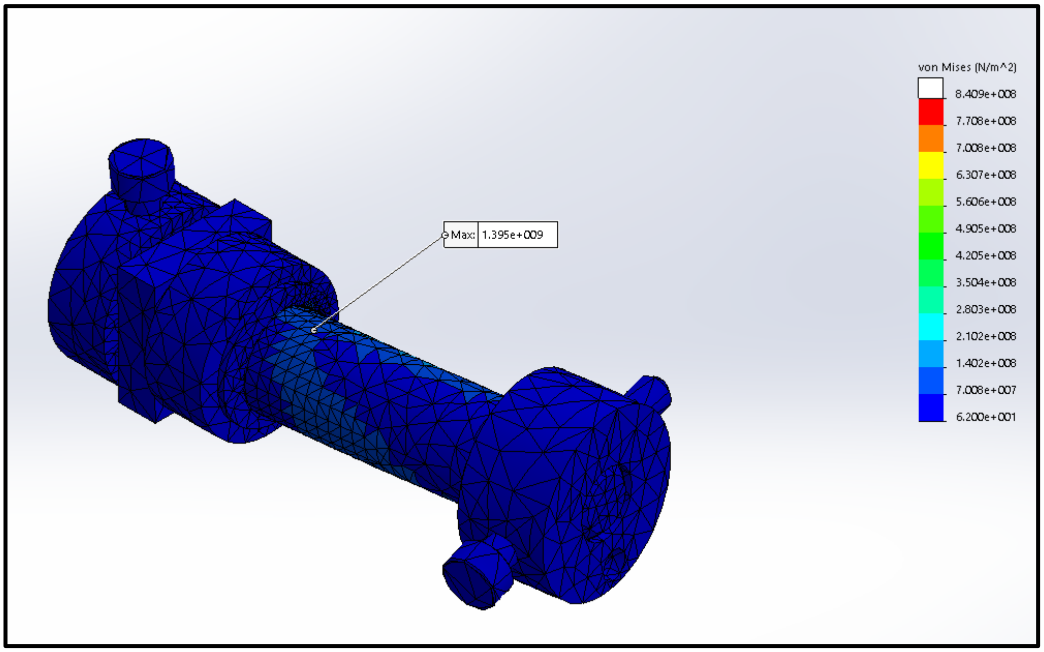

Figure 7: General Simulation Stress Results [6]

A mesh inclusive section view better shows the increasing stresses through the assembly as seen in Figure 8. The stress trend becomes more obvious in this view. However, it still was not entirely clear, due to the stress concentration which was higher than the rest of the stresses within the assembly.

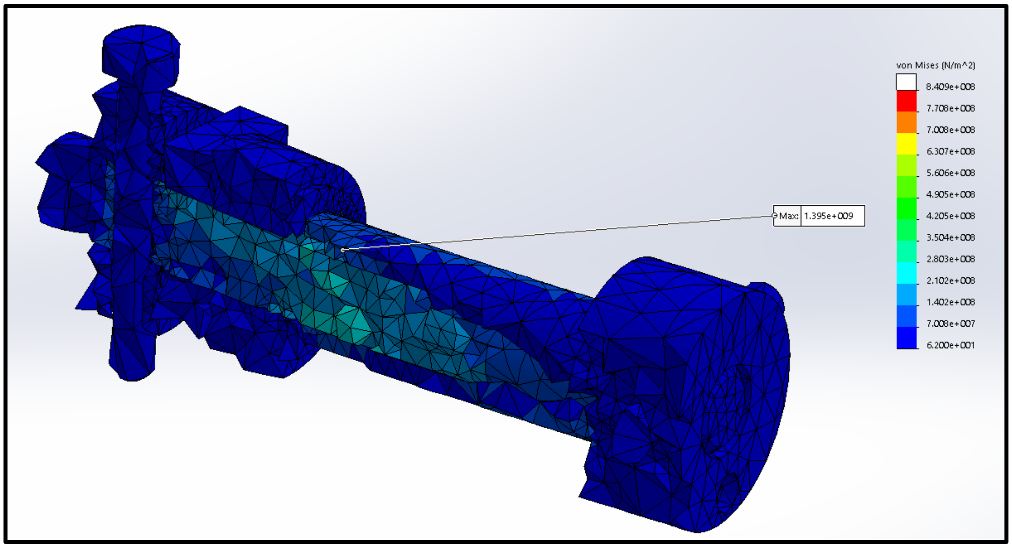

Figure 8: Mesh Inclusive Section View of the Stress Distribution [6]

This uncertainty of the stress distribution can be combated by adjusting the stress scale in order to better visualize how the stress is acting inside the part.

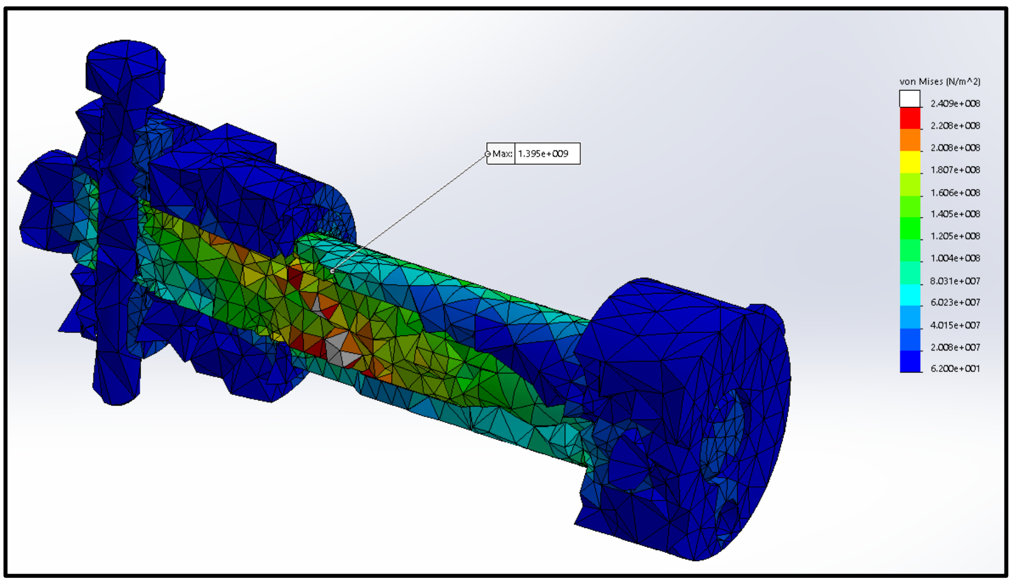

Figure 9: Mesh Inclusive Section View of the Stress Distribution with Adjusted Scale [6]

The view shown in Figure 9 gives a far better view of the stress trend through the assembly. Still, the stress concentration was not visible. The view above does show how the 6,000 [N] applied force, which represented the equivalent torque applied by the torsional spring seats into the spring housing, results in a large bending moment to the mounting shaft and spring housing. As a result of this bending moment, the spring housing and mounting shaft were subjected to a large tensile stress. In order to properly analyze the stress concentration, another view of the stress distribution needed to be investigated.

3.2 Stress Concentration

Figure 10, shows a section view of the stress distribution on a plane perpendicular to the plane in Figure 9.

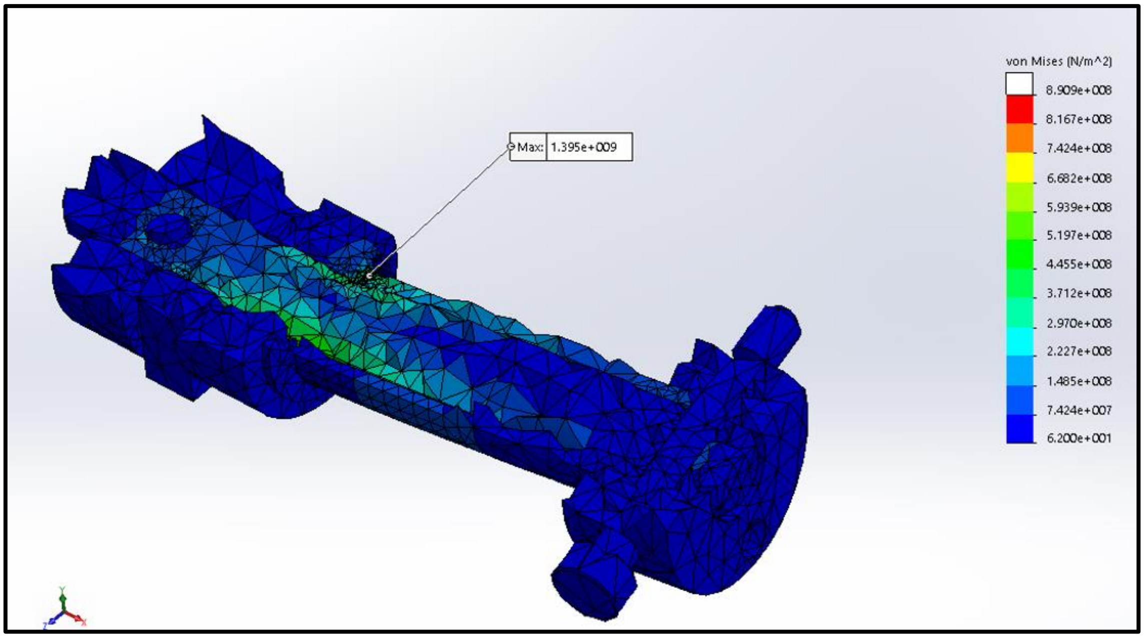

Figure 10: Rescaled Stress Distribution Showing a Center Plane and the Relevant Stress Concentration [6]

This mesh inclusive cross section view shows the stress trend near the stress concentration. The enlarged view in Figure 11 shows the stress trend near the stress concentration.

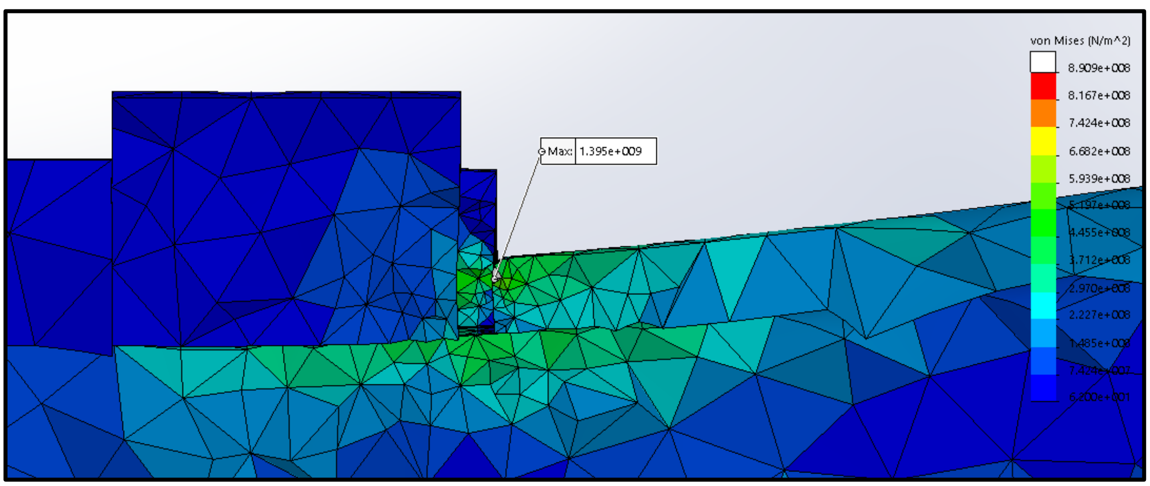

Figure 11: Stress Concentration Location [6]

As shown in Figure 11, there was a single element experiencing a heightened level of stress. This stress value was not reliable, since in a surface to surface connection, if the mesh is not fine enough the resulting forces between two surfaces could be applied to very few contact nodes. That results in a reduced surface area, and heightened stresses in that location. In order to verify that that stress concentration was accurate a more refined mesh needed to be considered.

3.3 Factor of Safety Plot

The factor of safety is visualized below in Figure 12.

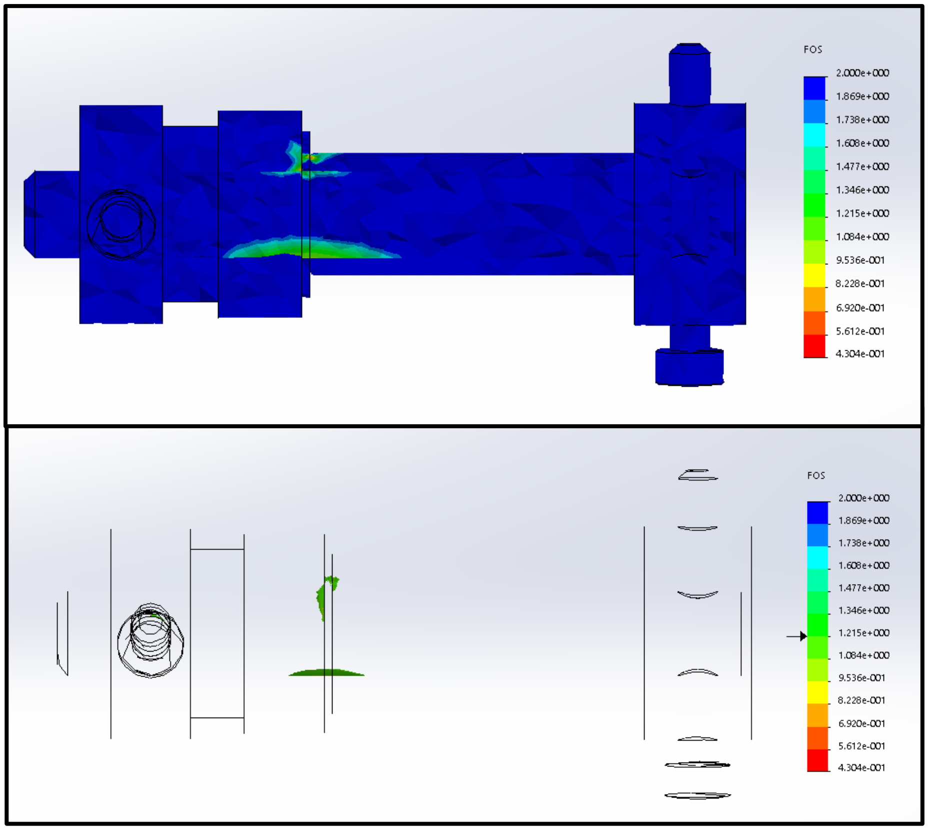

Figure 12: Factor of Safety Plot and ISO Clipping for Factor of Safety Values Below 1.215 [6]

As shown above in Figure 12, the overall factor of safety for the design was satisfied. Unfortunately, there was some heightened stress, resulting in a lowered factor of safety around the washer and spring housing. The ISO Clipping image is a SolidWorks feature which allows surfaces of a specified value from a plot to be viewed individually [13]. In the case shown in Figure 12, factor of safety values of less than 1.215 are all that are displayed.

Overall, it was shown through the above analysis that overall the lift assist design is acceptable. However, the stress at the contact point between the washer and pin housing is a potential risk to the designs success and safety. Even with this uncertainty, there is no risk of catastrophic failure due to fracture.

In order to unequivocally prove the validity of the lift assist mechanism design using SolidWorks the simulation result needed to be refined with an adjusted simulation set up.

4 Simulation 2 – Set-Up

The following section will outline the setup of the second SolidWorks setup which was used to provide a final conclusion on the identified potential risks which were raised from the initial simulation.

4.1 Simplifications

The results of the first simulation indicated that the locking pin was strong enough and that there was no possible risk of material failure under the loading condition created by lowering the grille guard. Due to this, the pins were replaced in the second simulation with pin connectors in SolidWorks as seen in Figure 13.

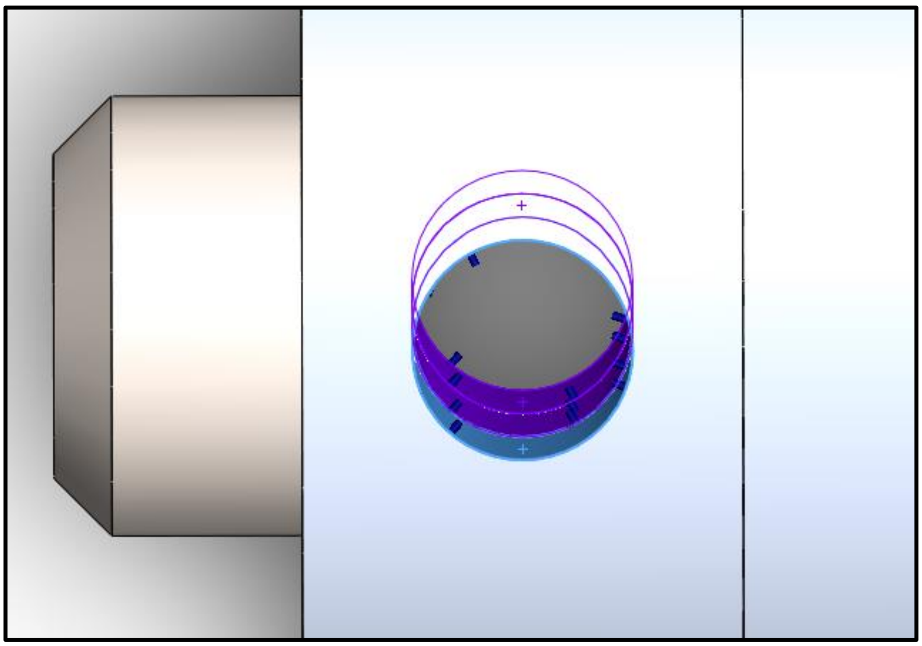

Figure 13: SolidWorks Pin Connectors Replace the Locking Pin from the First Simulation [6]

The connection type for both pins was set to no translation and no rotation as long as the material strength properties of AISI 304 Steel were not exceeded. Pin Connectors are best utilized in SolidWorks when wanting to know the effects of pins on their adjacent parts or the full assembly, rather than the actual stresses and strains in the pins themselves [14].

4.2 Interference Inspection

The interference inspection for this simulation yielded the exact same result as the first simulation. This is due to the fact that there were no changes in geometry outside of the removal of the locking pins.

4.3 Simulation Properties

The simulation was changed to a non-adaptive study, rather than an h-adaptive study. A non-adaptive study requires the user to set the mesh manually rather than automatically by the SolidWorks engine. This decision was made since properly analyzing the stress concentration region within this simulation required certain regions to have much finer meshes than others. This is not an option available when utilizing h-adaptive meshing, as it is not compatible with manual mesh control. In order to continue utilizing h-adaptive meshing the whole region would need to be changed to a mesh fine enough to fully understand behaviour of the stress concentration. If the mesh was set to this level of precision, the simulation would consume more than 12 hours per iteration.

4.4 Component Connections

The component connections remained consistent from those outlined in section 2.3. However, due to the removal of the simplified locking pins, their relevant component connections were replaced with pin connectors as explained in section 4.1 and Figure 13.

4.5 Local Mesh Set-up

Continuing from the results obtained in the previous study, the stress was expected to concentrate on the contact area between the spring housing and washer. The blue region indicated in Figure 14 is the area in which this stress concentration was expected. A “split line” feature was created in this region in order to reduce the mesh element size from the global value to 0.8 [mm] with an preceding layer ratio of 1.2. The “split line” feature allows for this, as it enables the ability to separately define and constrain different portions of one feature surface within SolidWorks.

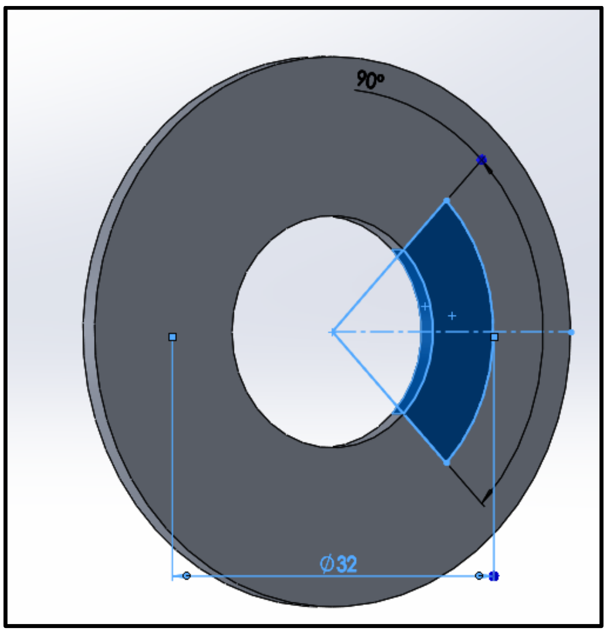

Figure 14: Split Surface on the Washer, Utilized to Increase a Finer Mesh in the Areas of High Stress Concentration [6]

The resulting mesh in the higher stress region can be seen in Figure 15. The results in the following sections confirmed that this higher mesh area was of adequate size.

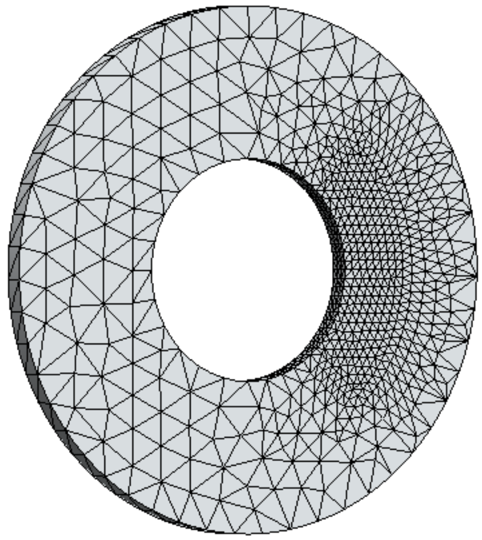

Figure 15: Resulting Mesh on the Washer [6]

Like the manually adjusted mesh control for the washer, the spring housing area from Figure 16 indicated in blue was where stress concentrations were expected.

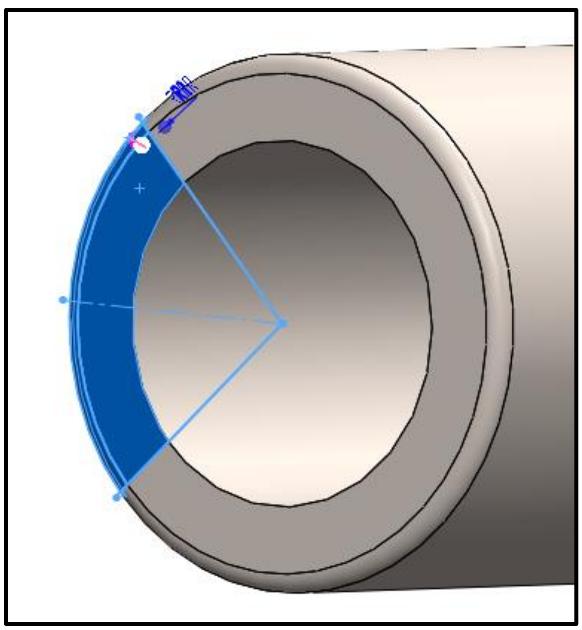

Figure 16: Split Surface on the Spring Housing, Utilized to Increase a Finer Mesh in the Areas of High Stress Concentration [6]

The resulting mesh in the higher stress region can be seen in Figure 17. The mesh control for this region matched that of the washer.

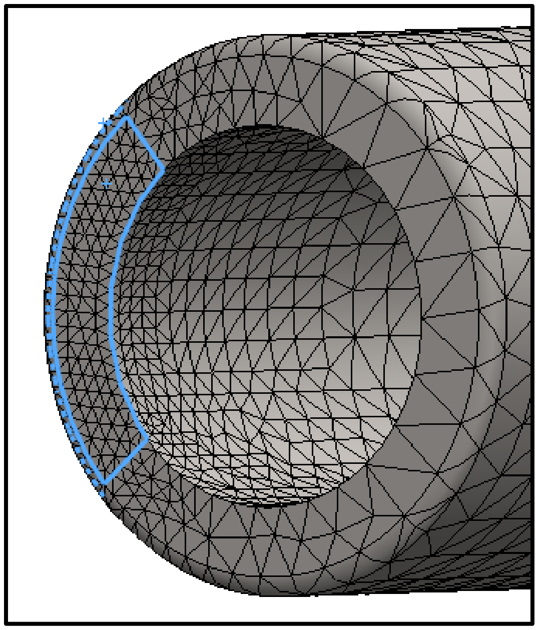

Figure 17: Resulting Mesh on the Spring Housing [6]

The mounting shaft also needed improved mesh control in its higher stress regions as seen in Figure 18. The mesh in the relevant region seen in Figure 18.

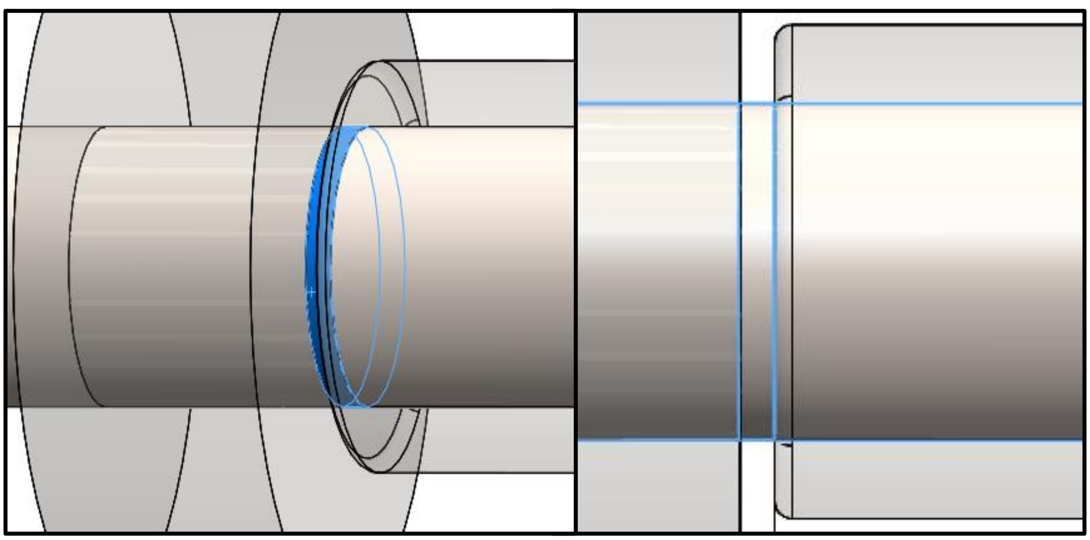

Figure 18: Split Surface on the Mounting Shaft, Utilized to Refine the Mesh in the Areas of High Stress Concentration [6]

The mesh in the relevant region seen in Figure 19 utilized the same 0.8 [mm] element size with a preceding layer ratio of 1.2.

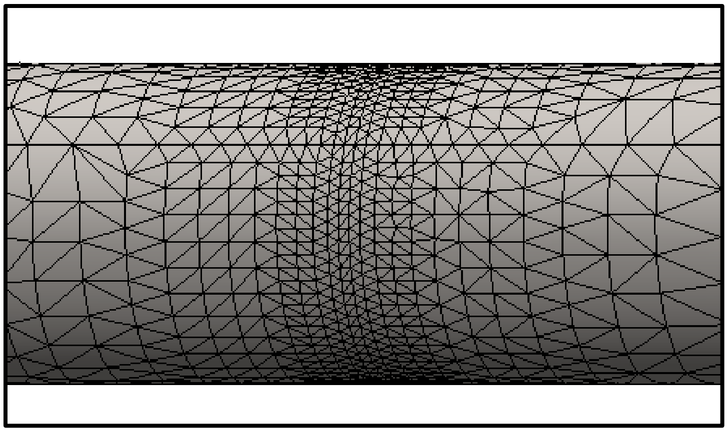

Figure 19: Refined Mesh on the Mounting Shaft [6]

4.6 Global Mesh Set-up

With the local mesh control established, the rest of the model was set to have a global standard mesh with an initial element size of 3 [mm] and a 0.15 [mm] tolerance. The resulting mesh for the model can be seen in Figure 20.

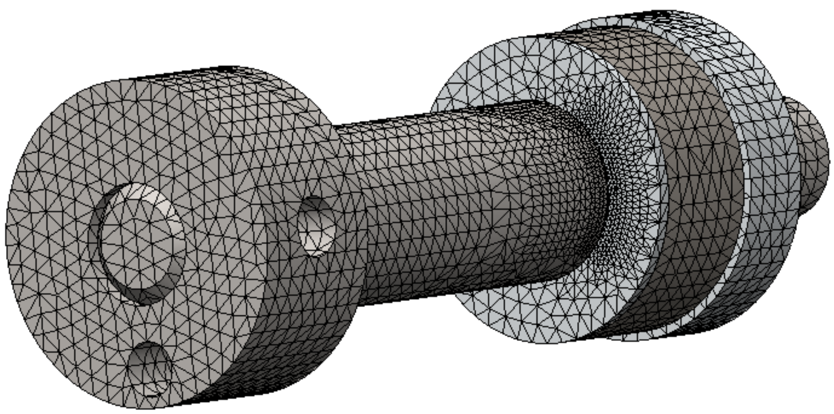

Figure 20: Global Mesh for the Second Simulation [6]

As the enlarged view in Figure 21 shows, the mesh details blended into the global mesh in the region of higher stress.

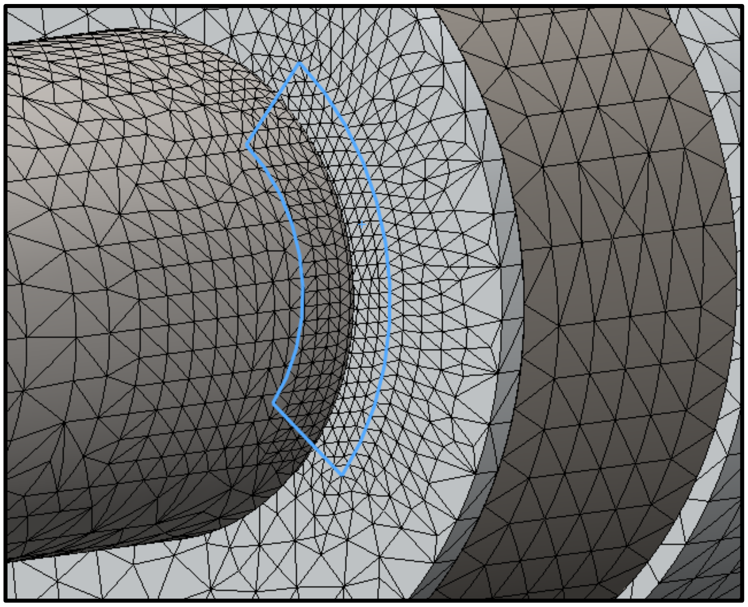

Figure 21: Manually Controlled and Refined Mesh in the Higher Stress Region of the Second Simulation [6]

5 Simulation 2 – Results and Analysis

The following section will outline the results and analysis of the second simulation.

5.1 General Results

The results from this simulation are similar to the results obtained from the first simulation. The bending moment created by the torque supplied by the lift assist mechanisms torsional spring created a stress concentration in a similar region. The stress concentration region was close to mating surface between the washer and spring housing. The stress throughout the rest of the assembly, including the grille pin housing was far below the yield point as seen in Figure 22.

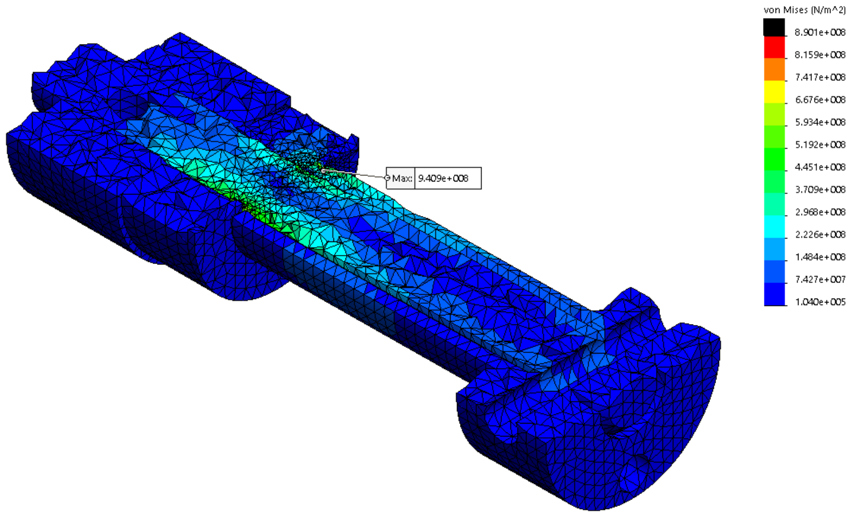

Figure 22: Overall Stress Distribution for the Second Simulation [6]

An enlarged view of the stress concentrations showed the stress trends through the different features, shown in Figure 23.

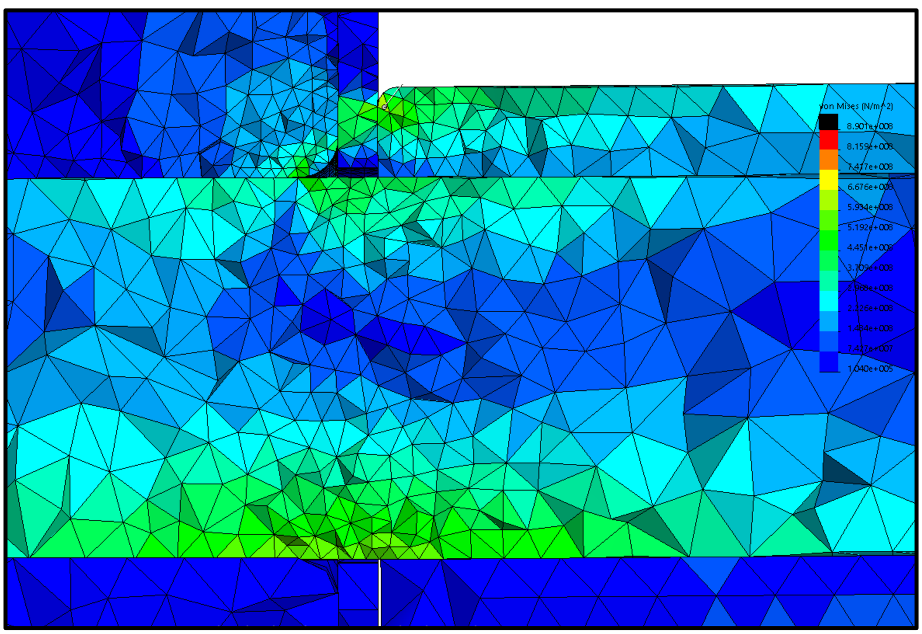

Figure 23: Closer View of the Stress Trends in the Second Simulation [6]

Figure 23 shows that the mounting shaft stresses were far below the yield strength of the assembly which is 890 [MPa]. The highest stress in the region was 558 [MPa] resulting in a factor of safety of 1.6. The spring housing which is made with the same material also had a stress node with a Von Mises stress of 558 [MPa] resulting in a factor of safety of 1.6. As shown in Figure 23, the stresses in the simplified grille region and grille pin housing are extremely low and no where near their yield strength of 275 [MPa]. This second simulation also gave a high accuracy visualization of the stress concentrations in the lift assist assembly.

5.2 Stress Concentrations

The first stress concentration shown in the model was on the 4130 steel washer which has a yield strength of 670 [MPa]. The stress distribution is visualized below in Figure 24.

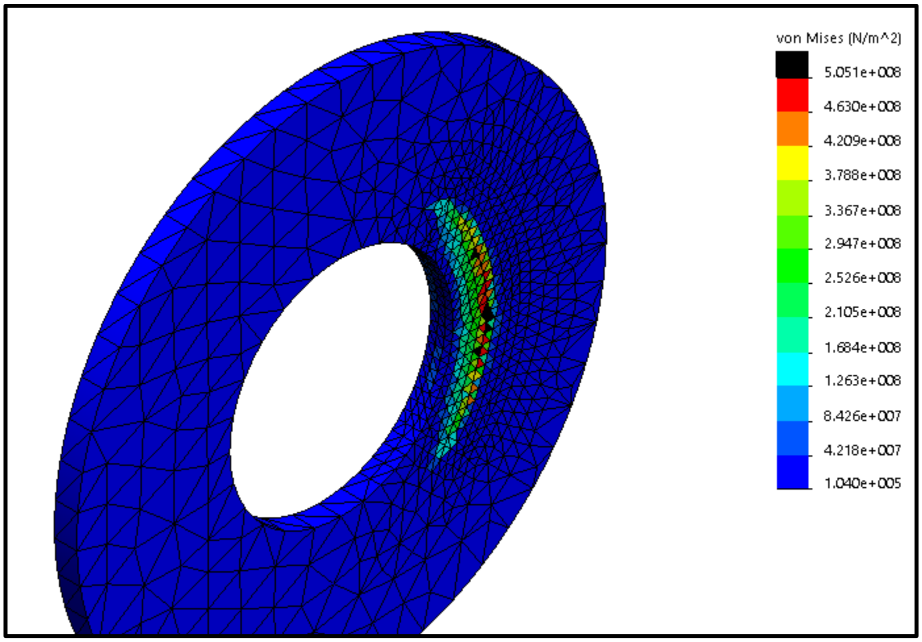

Figure 24: Stress Concentrations on the Washer [6]

As shown in Figure 24, the maximum stress in the washer was approximately 505.1 [MPa] which is below the yield strength of the material, resulting in a factor of safety of 1.33. The stress concentrations in the washer are due to the resulting contact force from the spring housing during bending.

Even though this feature is not at risk of failing, there was some speculation as to why the stress in this region was so high. It was postulated by the team that the stress resulting from the contact between the washer and spring housing had some inaccuracies. In order to check this, a simple stress hand calculation over this region was performed using the equation:

$$\sigma = F/A$$

Utilizing tools in SolidWorks, it was found that the compressive force acting on the high stress region was 20,000 [N] while the high concentration area in Figure 24 was 78 [mm^2]. These values yielded an overall stress in that region of:

$$\sigma = \frac{20{,}000N}{78mm^2} = 256\ MPa$$

This stress value is much lower than the simulation value, leading the team to believe there could be some stress singularities within this region.

The highest stresses in this simulation occurred on the mounting bracket, which can be seen in relation to the overall stress distribution in Figure 25 outlined by the red circle.

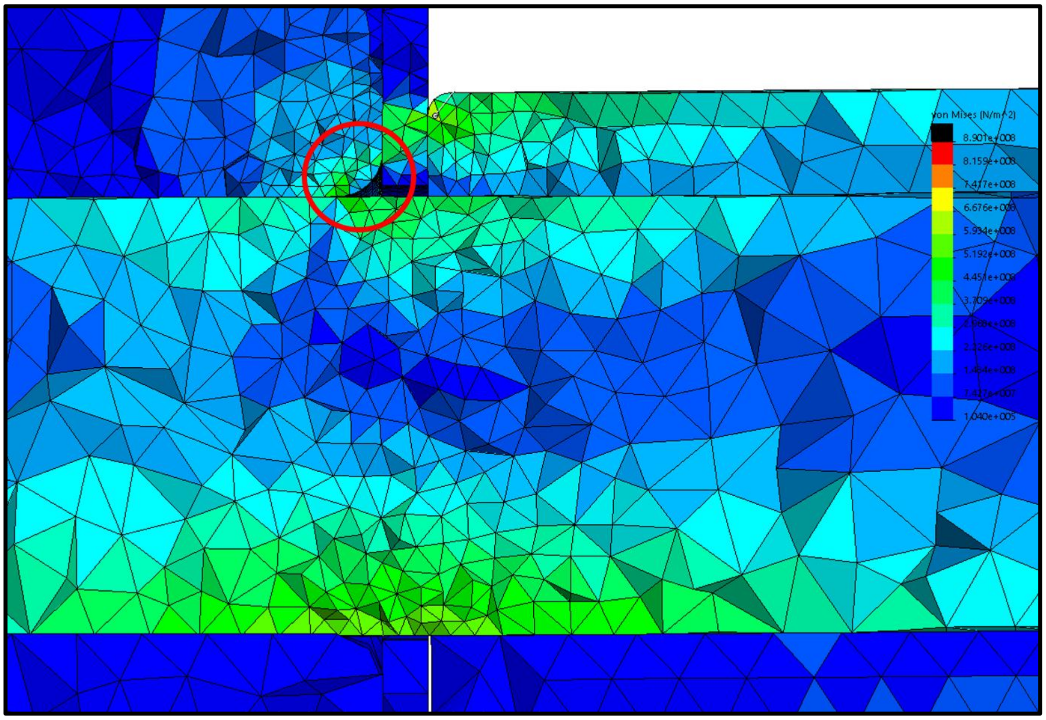

Figure 25: Area of Maximum Stress in the Second Simulation [6]

An enlarged view of this high stress region on the mounting bracket can be seen with stress probing in Figure 26.

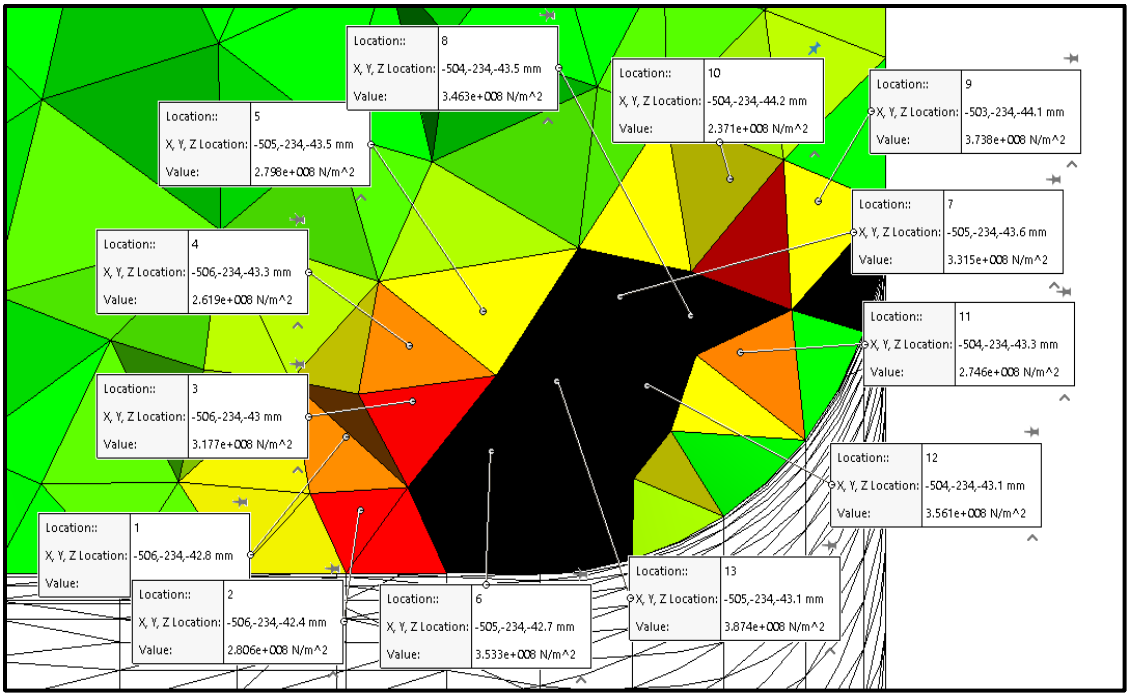

Figure 26: High Stress Region of the Simulation Located on the Mounting Bracket [6]

As shown in this figure, there were multiple nodes with stress values above the yield stress of the mounting bracket, with the maximum stress reaching 558 [MPa]. However, it is worth noting that the mounting bracket is made from 44W Steel, which is an extremely low strength steel, yielding at only 303 [MPa]. The risk of failure in this part would be best mitigated by changing the material at the hinge of the mounting bracket. An additional reason for this high stress is the method in which the simulation needed to be constrained, which can be seen in Figure 27.

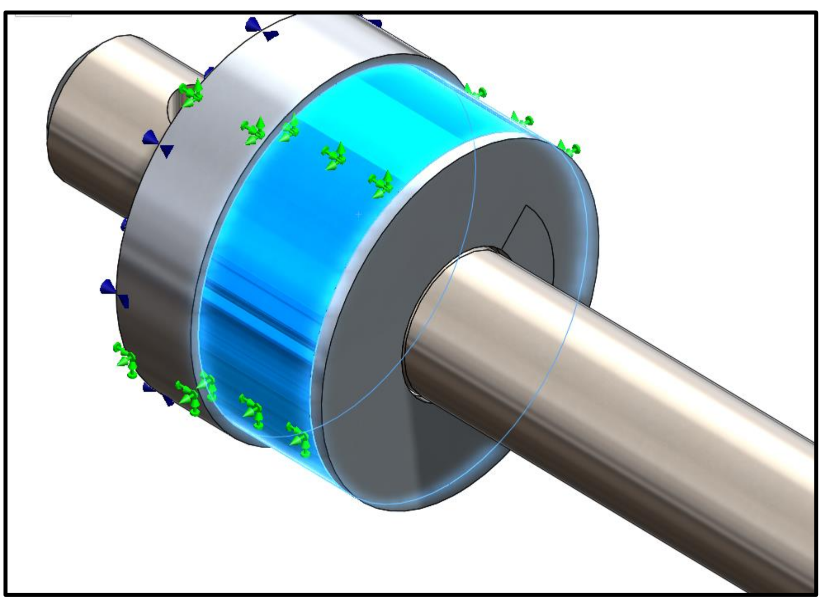

Figure 27: Mounting Bracket Fixture [6]

This method was necessary in order to simulate the hinge action of the lift assist mechanism. However, it results in the mounting bracket being constrained to a higher degree than it would be in reality which can also result in higher stresses, and a lack of ability for a feature to reduce its strain energy through small scale elastic deformations. Due to this constraint it is likely the stresses in this region could be higher than a real-life model would be. Despite this, it is not possible to recommend the utilization of this part without changing the mounting bracket material form 44W steel to a higher strength material at the hinge point.

6 Recommendations

In order to successfully achieve the main design goal of the design a bumper lift assist mechanism the steel utilized at the hinge point of the mounting bracket needs to be changed. This recommendation is based on a number of factors which have determined that this would be necessary in order to successfully incorporate this lift assist mechanism to the Bison Grille Guard system.

To properly aid in the raising and lowering of the grille guard from the hinge points, or from another location near the hinge points, a torque of at least 122 [Nm] is required in order to aid the process to the specifications outlined by the customer. From the spring design section found in Appendix C, the spring capable of providing this torque utilized in this design is optimized based on the geometric constraints set by the mounting bracket and grille geometry. It cannot be shortened, constricted, or expanded or it would begin to interfere with additional geometries in the Bison Grille Guard system while also moving further away from its optimized state. Because of this, the contact stress on the mounting bracket cannot be easily be reduced. Based on this, it is the team's recommendation that if this lift assist design was utilized, the bracket material at the hinge point should be changed to have a material with at least a yield strength of 600 [MPa]. Alternatively, real world testing could be utilized on a sample part as a proof of concept to prove that the stress is not high enough to fail, as the finite element analysis may have yielded a maximum stress higher than would be observed in reality.

7 Conclusion

To conclude, the lift assist mechanism has a reasonable safety factor and is very unlikely to fail. However, the mounting bracket at the hinge point does have potential to fail mainly due to the weak material utilized. It is recommended that to utilize this lift assist mechanism, a stronger material should replace the current mounting bracket material at the hinge point. This material should have a yield strength of at least 600 [MPa].

Works Cited

[1] McMaster-Carr. *18-8 Stainless Steel Locking Pin with Wire Retainer* [Online]. Available: https://www.mcmaster.com/#98480a140/=1afiifp. [28 November 2017].

[2] ASM Aerospace Specification Metals Inc.. *Aluminum 6061-T6; 6061-T651* [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=ma6061t6. [1 December 2017].

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Appendix E

Tow Pin Numerical Analysis

Table of Contents

List of Figures

1 Introduction

This section will discuss the numerical analysis performed on the advanced tow pin latch design. The construction of the 3D model and the numerical analysis performed are presented in detail to reinforce the preliminary calculations and ensure the final design is reliable.

2 3D Modeling

A 3D model of S&V Manufacturing's original tow pin was created using SolidWorks. There were no existing engineering drawings of the tow pin, therefore all measurements were taken from a sample tow pin provided by S&V Manufacturing. The material used for the tow pin is ASTM A36 steel with a yield strength of 250 [MPa] [1]. Detailed preliminary engineering drawings were prepared for the standard tow pin. These drawings can be found in the Appendix F.

The advanced tow pin design allows for the operator to lock the tow pin when it is in the raised or lowered positions without the need for it to be completely removed. This is achieved by the addition of two keys and two keys. The first key in the advanced tow pin design is a modification of the original top plate of the tow pin. The top plate is cut into the shape shown in Figure 1 to act as a key.

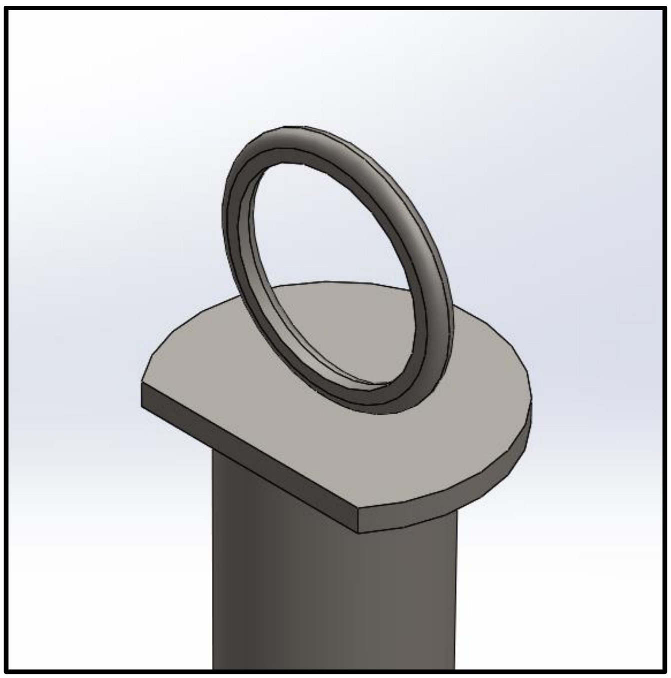


Figure 1: Modification on the Top Plate [2]

The second key is made from the same material as the tow pin, ASTM A36 steel. The key has a thickness of 6.35 [mm], a width of 20.00 [mm], and a length of 11.11 [mm]. The second key will be welded on to the body of the tow pin 202.58 [mm] from the top plate, 90^{o} from its straight edge. The position of the second key on the body is critical to allow for clearance between the grille guard and the bracket. However, this also shows that the position of the second key varies for different grille system as the clearance changes. Figure 2 shows the general placement of the second key for the advanced tow pin design.

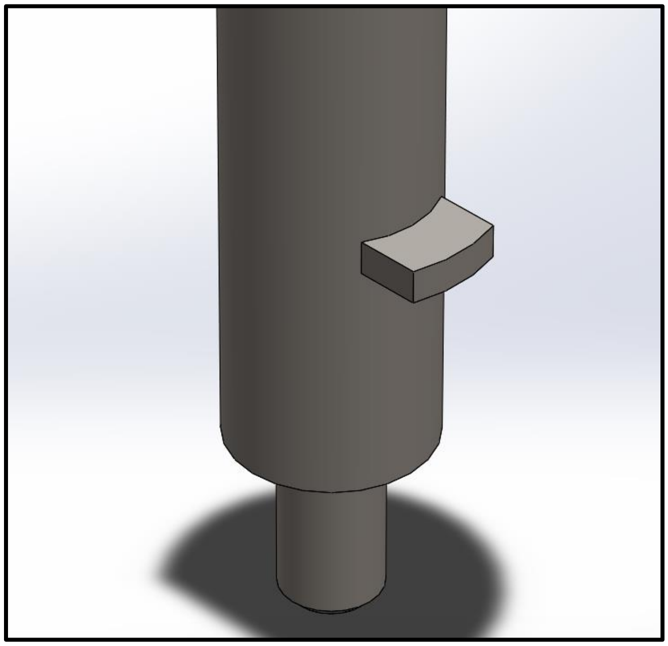


Figure 2: Second Key Placement [2]

The key seats for the advanced tow pin design are made from stock ¼ inches aluminum plates, to match the appearance of the grille guard. Figure 3 shows the key seats welded on both sides of the grille guard.

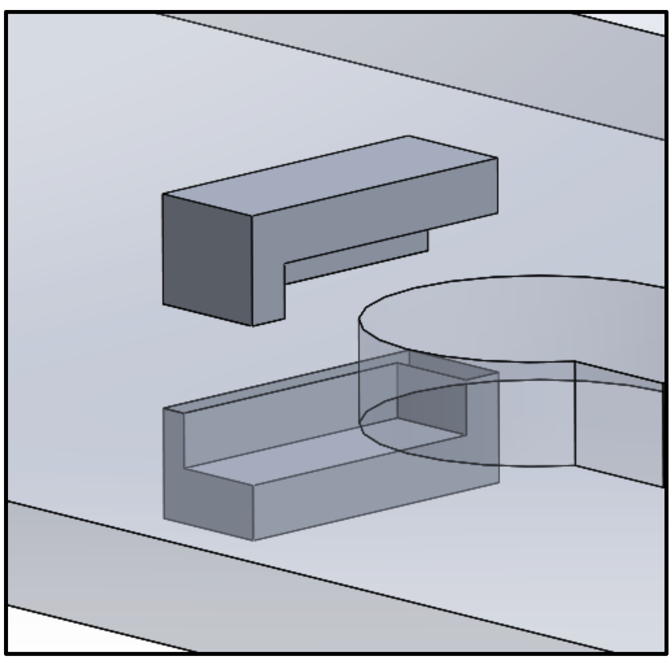


Figure 3: Advanced Tow Pin Key Seats [2]

As shown in the figure above, the openings of the key seats are opposite to each other for locking the pin in the raised and lowered positions. The material used for the key seat is 6061 T6 aluminum with a yield point of 275 [MPa] [3].

Modifications are needed on the existing tow pin holes on both grille guard and mounting bracket in order to fit the shape of the advanced to pin key. Figure 4 shows the modification has been made to the key holes.

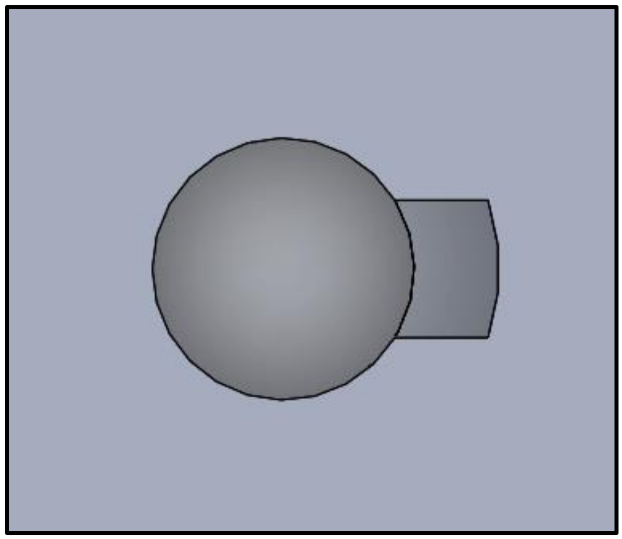


Figure 4: Grille Guard Key Hole [2]

With all the individual parts of the advanced tow pin modeled in SolidWorks, the assembly of the latch system was then constructed. Figure 5 shows the advanced tow pin in raised and lowered position respectively.

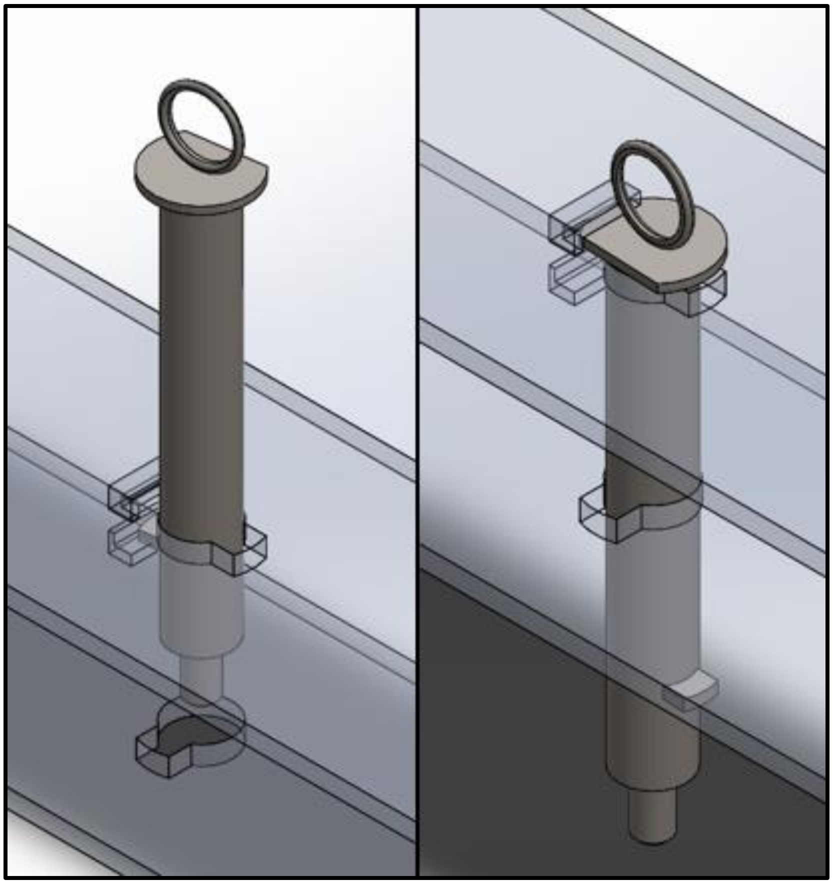


Figure 5: Latch System in Raised and Lowered Positions [2]

3 Finite Element Analysis

As mentioned in the previous section, the existing tow pin used for the latch design is already a proven component of the current grille guard system. Therefore, it was assumed that the tow pin will be strong enough to withstand normal operating forces, and it was neglected from our finite element analysis. The analysis in this section was focused on the added key and key seats.

3.1 Tow Pin Key

The fixture on the modified tow pin was set to be on the bottom surface of the key. The fixture geometry was intended to simulate the situation where the tow pin is raised and locked in the key seat. Since there are no other external forces acting on the tow pin, the key will only be bearing the weight of the tow pin. A gravity force of 24.24 [N] was applied. A medium density mesh was created for the model. Figure 6 shows the locations of fixture and force applied.

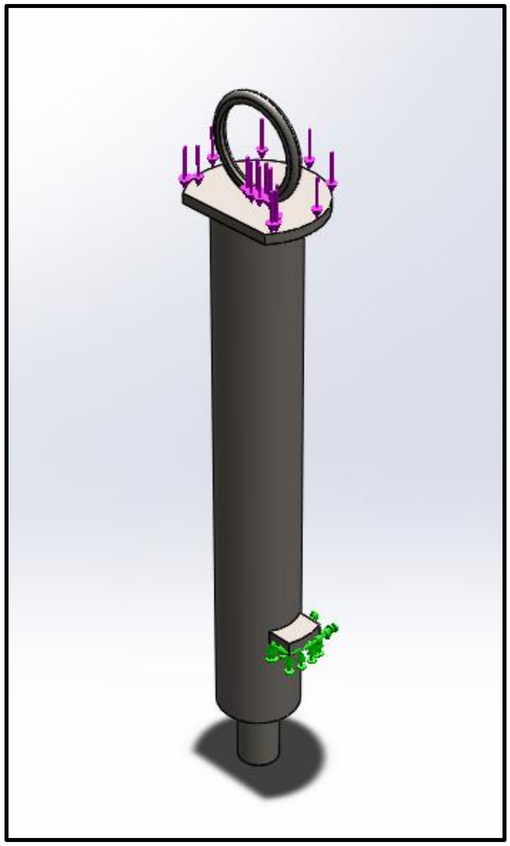


Figure 6: Fixture Geometry and Applied Force [2]

As shown in Figure 7, the maximum stress on the geometry is 1.283 [MPa] which is below the yield strength of the material. A stress concentration was located on the key as predicted. This high stress concentration could be eliminated by either adding material to the connection between the pin and key, or removing the sharp corners on the geometry by the addition of fillets. The finite element analysis results validated our proposed latch design.

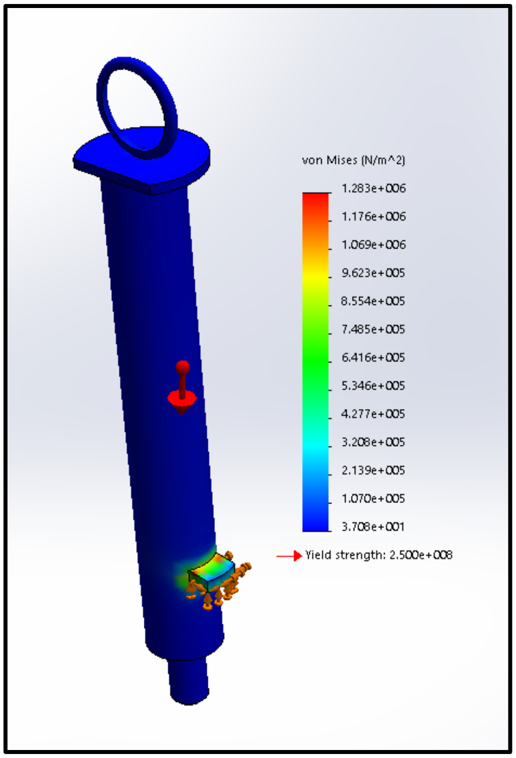


Figure 7: Stress Distribution Plot for the Tow Pin Key [2]

3.2 Key Seats

To ensure the design can function as intended, the key seats were also analyzed using finite element analysis. The application of the fixture and load is summarized in Figure 8. An external load of 30 [N] was applied to the inner surface of the key seat. The 30 [N] simulates the operator pulling on the tow pin while it is in the locked position.

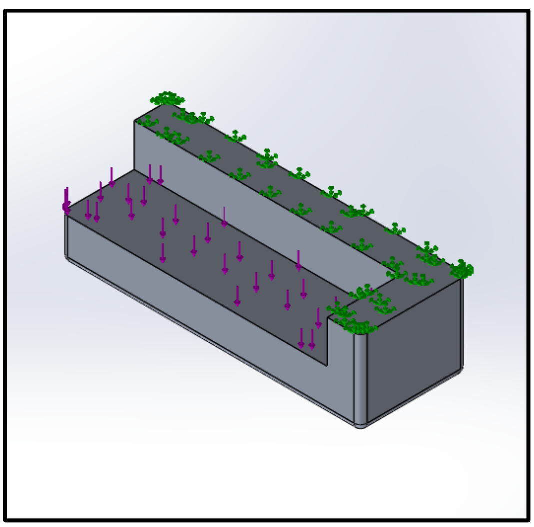


Figure 8: Fixture and Forced Applied on Key Seat [2]

The analysis was run using a medium density mesh. The results showed a maximum stress on the key seat of 1.374 [MPa]. The material used has a yield strength of 227.5 [MPa] which shows that our key seat will be able to withstand normal operating conditions. A stress distribution plot of the key seat structure is shown in Figure 9.

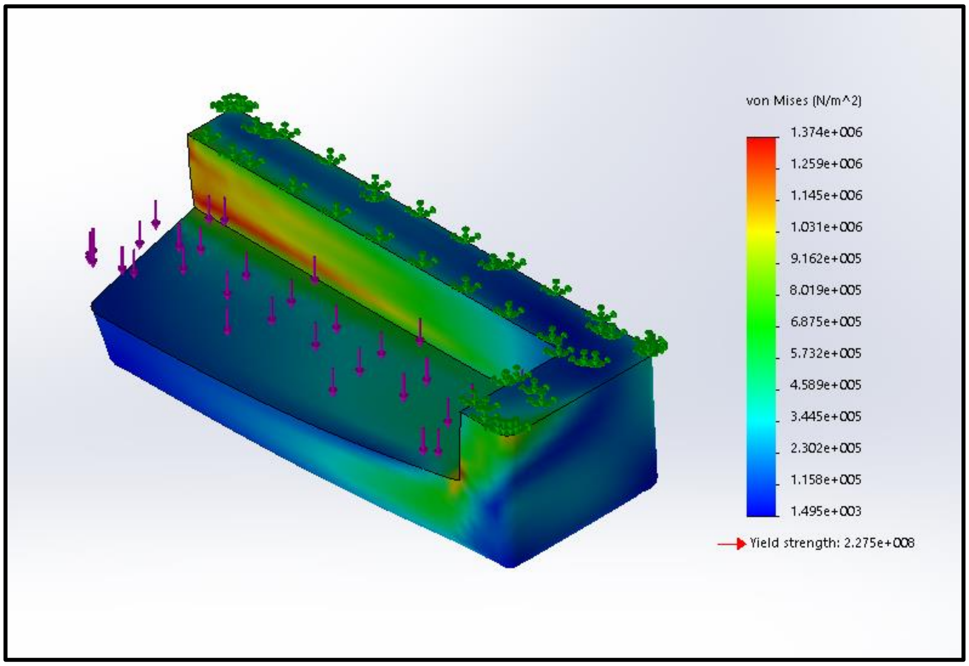


Figure 9: Stress Distribution Plot for the Key Seat [2]

4 Conclusion

In conclusion, the finite element analysis preformed on the advanced tow pin design reinforced the latch system selection by proving its ability to withstand normal operating conditions. The analysis revealed that the maximum stress in the tow pin key is 1.283 [MPa], while the force in the key seat is 1.374 [MPa]. Due to the nature of the operating conditions for the advanced tow pin design, the maximum stress for the key and key seat are negligible compared to the yield strength of the materials used.

Works Cited

[1] MatWeb. *ASTM A36 Material Properties* [Online]. Available: http://www.matweb.com/search/datasheet.aspx?matguid=afc003f4fb40465fa3df05129f0e88e6. [2 December 2017].

[2] S. Wang. *3D Tow Pin Model.* Winnipeg: Design Eng.,Univ Manitoba, Winnipeg, MB, November 2017.

[3] ASM Aerospace Specification Metals Ltd. *Aluminum 6061 T6; 6061-T651* [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=ma6061t6. [2 December 2017].

Appendix F

Preliminary Engineering Drawings

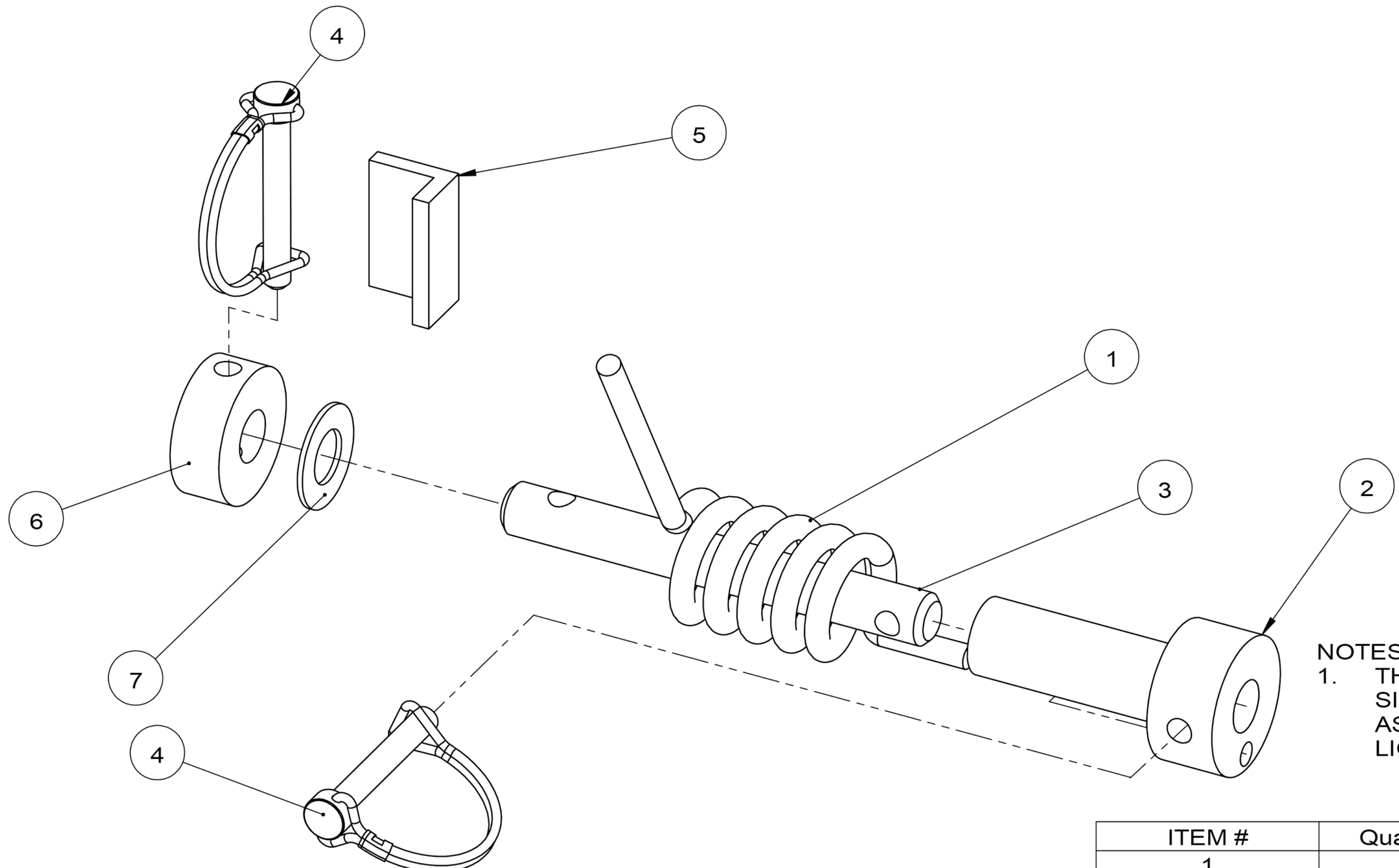
Table of Contents

1 Introduction

This appendix presents the preliminary engineering drawings for the lift assist and latch mechanism designs. Drawing for the entire assembly as well as each individual component of the design are shown for both the lift assist and latch mechanisms. It is important to note that all drawings are preliminary. Additional information and investigation is needed in order to finalize the drawings provided.

2 Lift Assist Mechanism

This section presents the preliminary engineering drawings for the lift assist mechanism. All drawings were created using SolidWorks based on the 3D models created in Appendix D.



NOTES:
1. THIS ASSEMBLY DRAWING IS FOR LEFT HAND SIDE ASSEMBLY ONLY. RIGHT HAND ASSEMBLY ASSEMBLY IS SYMMETRIC WITH LIGHT HAND SIDE ASSEMBLY.

ITEM #	Quantity	DESCRIPTIONS	PART NUMBER
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2	1	SPRING HOUSING	LA003
3	1	MOUNTING SHAFT	LA004
4	2	LOCK PIN	LA005
5	1	SPRING SEAT	LA006
6	1	GRILLE PIN HOUSING	LA007
7	1	WASHER	LA008

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
1	17/12/03		DWN / CKD	Z. XIN

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE: LIFT ASSIST MECHANSIM ASSEMBLY

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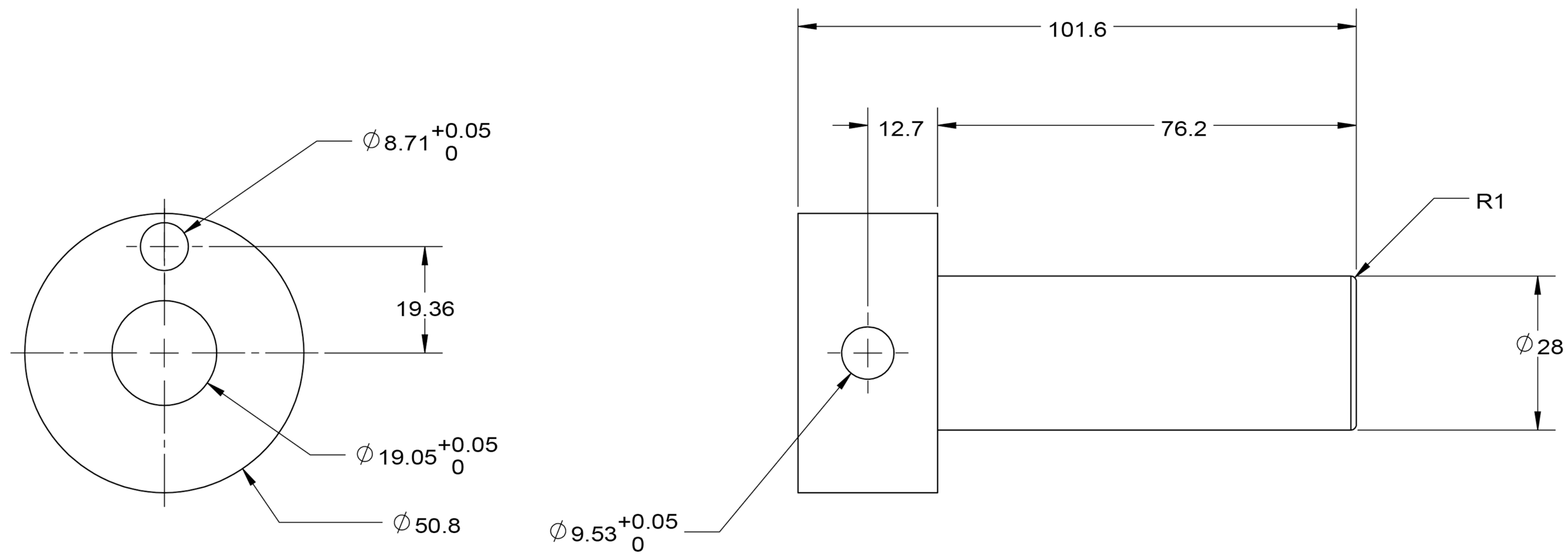
F4

F5

REV 1

LA003

DWG. NO.



NOTES:
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3. BREAK SHARP OF R1±0.1.

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			CKD NAME #	
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			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
1	17/12/03		DWN NAME #	ZXIN
			CKD NAME #	

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MECH 4860 Engineering Design

TITLE:
SPRING HOUSING

MATERIAL
HIGH STRENGTH AISI 4140 STEEL (ETD150)

FINISH MACHINED SURFACE
N/A

SIZE B

DWG. NO. LA003

REV 1

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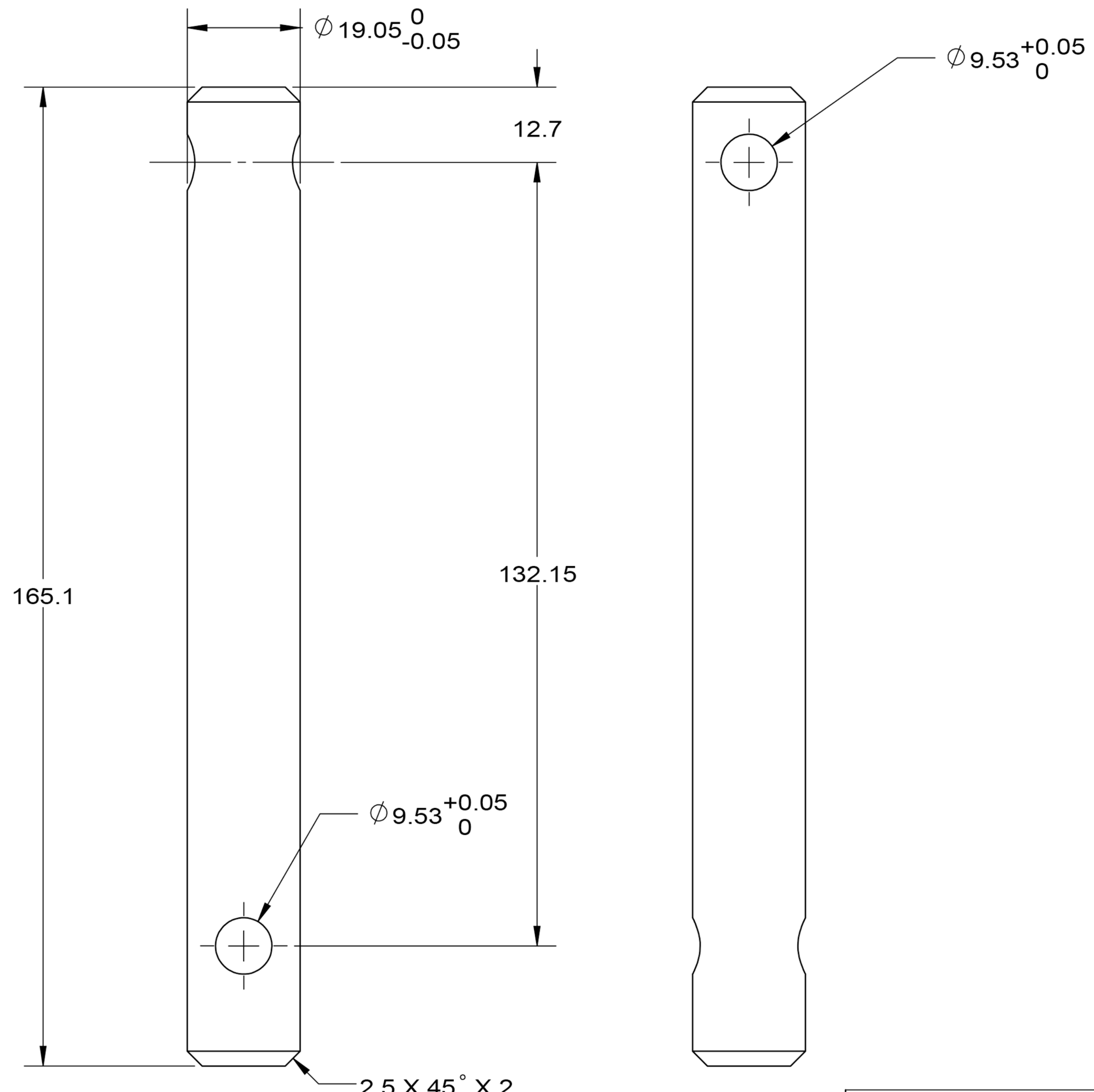
WEIGHT: N/A

SHEET 1 OF 1



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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/12/03		DWN	Z. XIN
			CKD	

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE: MOUNTING SHAFT

MATERIAL: HIGH STRENGTH AISI 4140 STEEL (ETD150)

FINISH: MACHINE FINISH N/A

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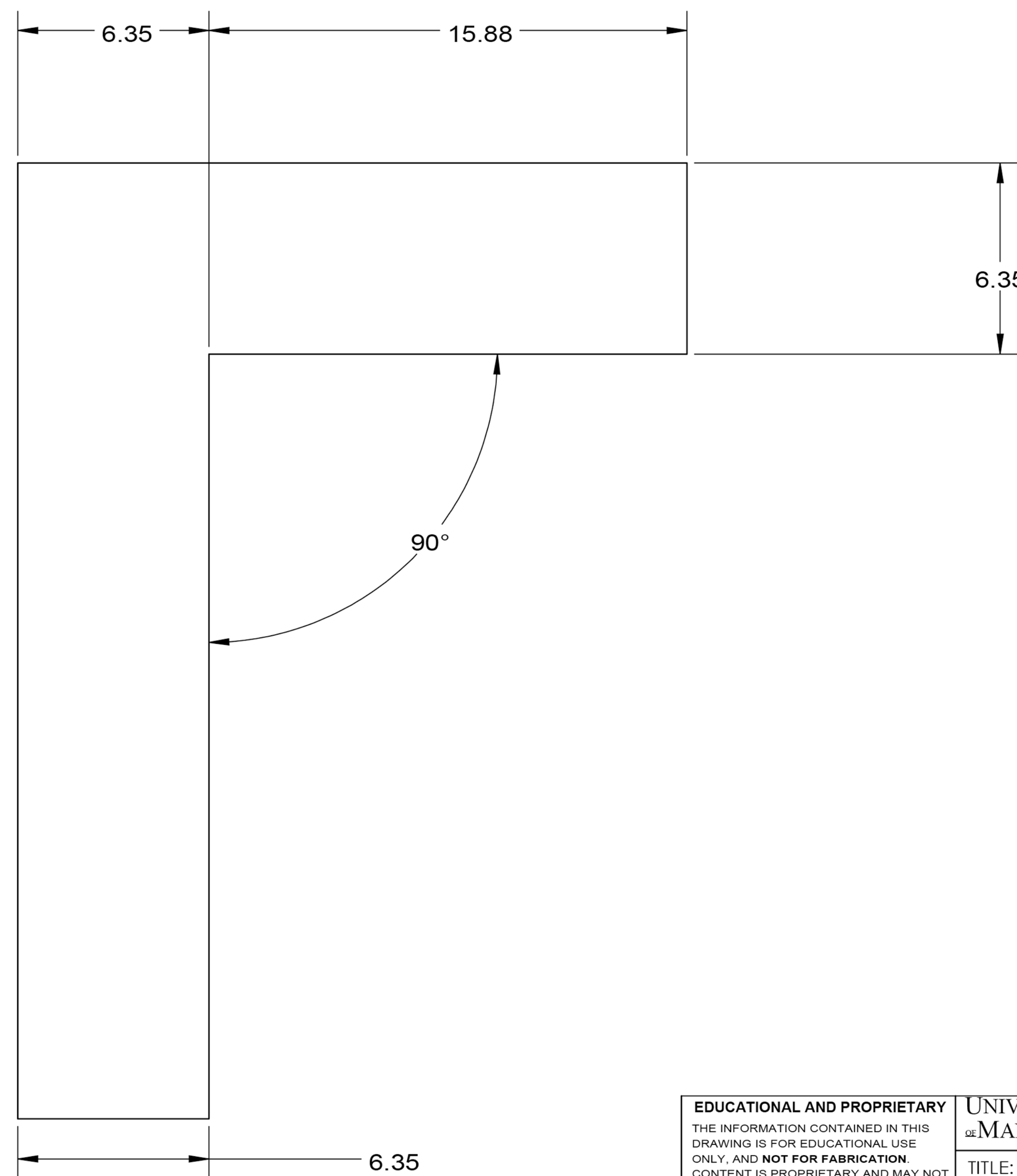
DWG. NO. LA004 REV 1

F6

F7



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TITLE: SPRING SEAT

MATERIAL: STEEL 44W

FINISH: MACHINE FINISH N/A

SIZE: B | DWG. NO. LA006 | REV 1

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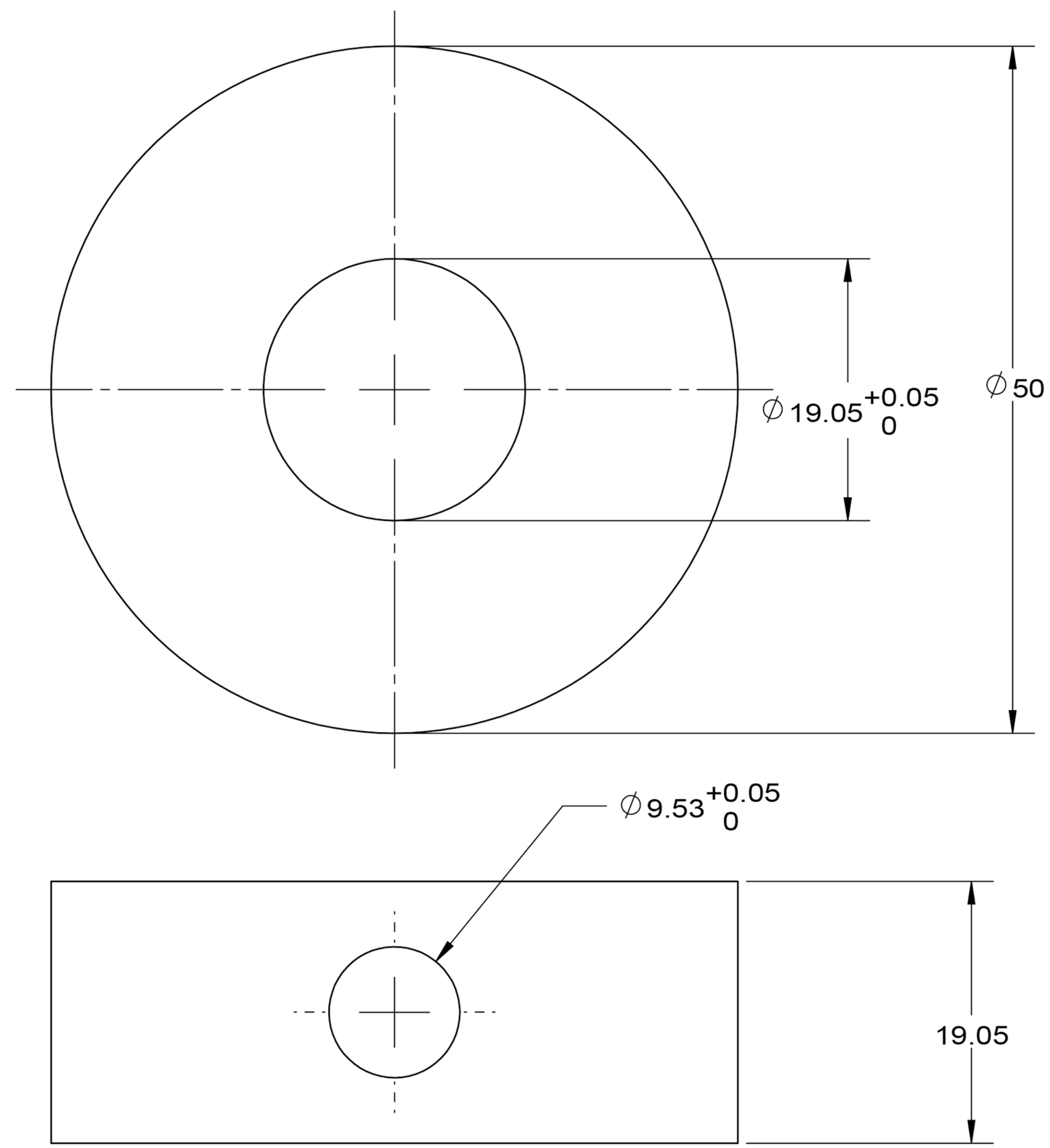

NOTES:
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3. BREAK SHARP OF R1±0.1.

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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/12/03		DWN	Z. XIN
			CKD	

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE: GRILLE PIN HOUSING

MATERIAL: ALUMINUM 6061 T6

FINISH: MACHINED FINISH N/A

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DWG. NO. LA007 REV 1

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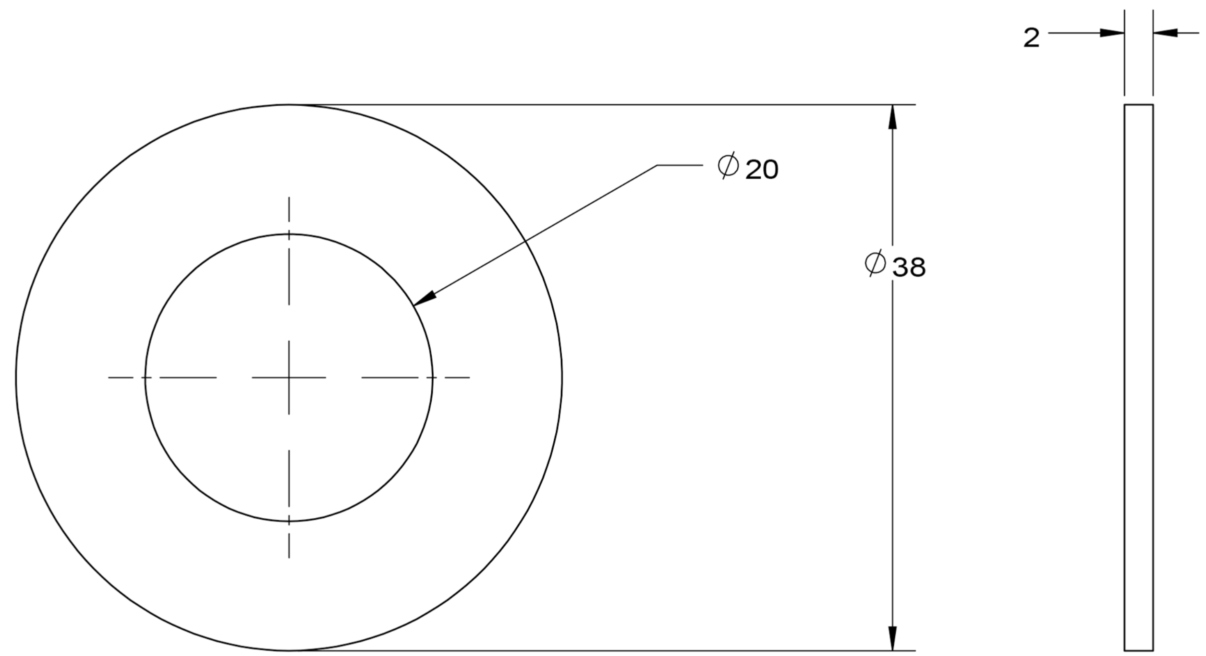

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3. BREAK SHARP OF R1±0.1.

REV	DATE Y/M/D	DESCRIPTION		
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			CKD	

EDUCATIONAL AND PROPRIETARY

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE: WASHER

MATERIAL: 7075 T6 ALUMINUM

FINISH: MACHINE FINISH
N/A

SIZE	DWG. NO.	REV
B	LA008	1

DO NOT SCALE DRAWING | SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

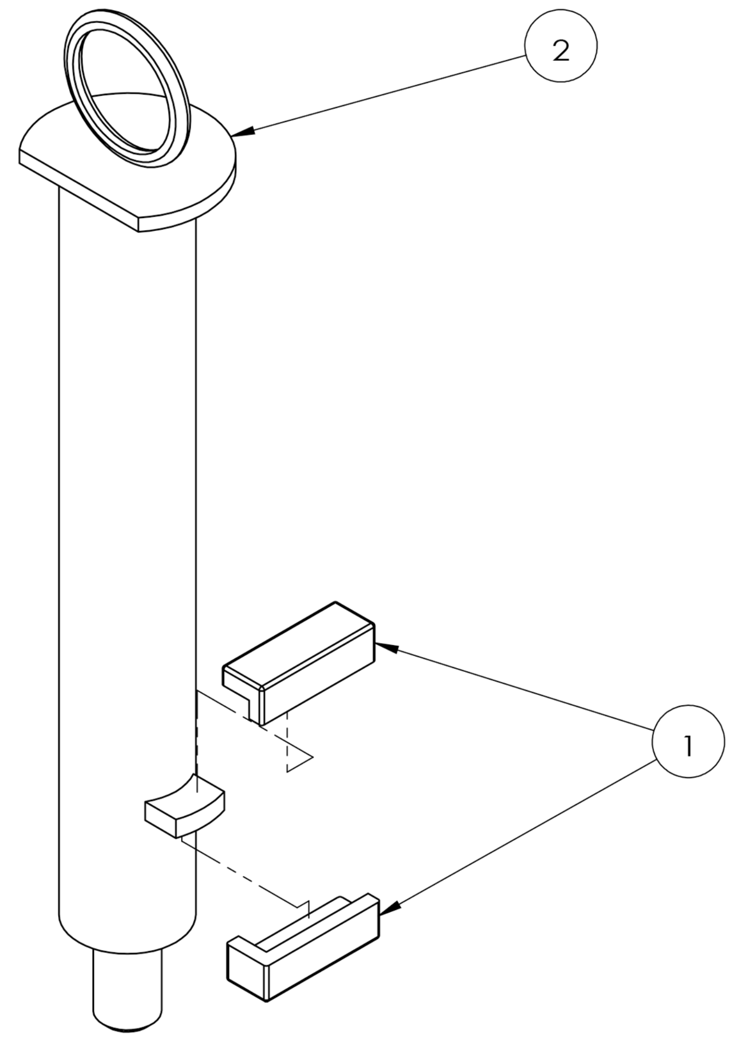
3 Latch Mechanism

This section presents the preliminary engineering drawings for the lift assist mechanism. All drawings were created using SolidWorks based on the 3D models created in Appendix E.

F10

REV 1

DWG. NO. LS-ASSEM01



ITEM #	Quantity	DESCRIPTIONS	PART NUMBER
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2	1	ADVANCED TOW PIN	LS002

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			CKD	

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TITLE:

LATCH SYSTEM ASSEMBLY

SIZE: B | DWG. NO.: LS-ASSEM01 | REV: 1

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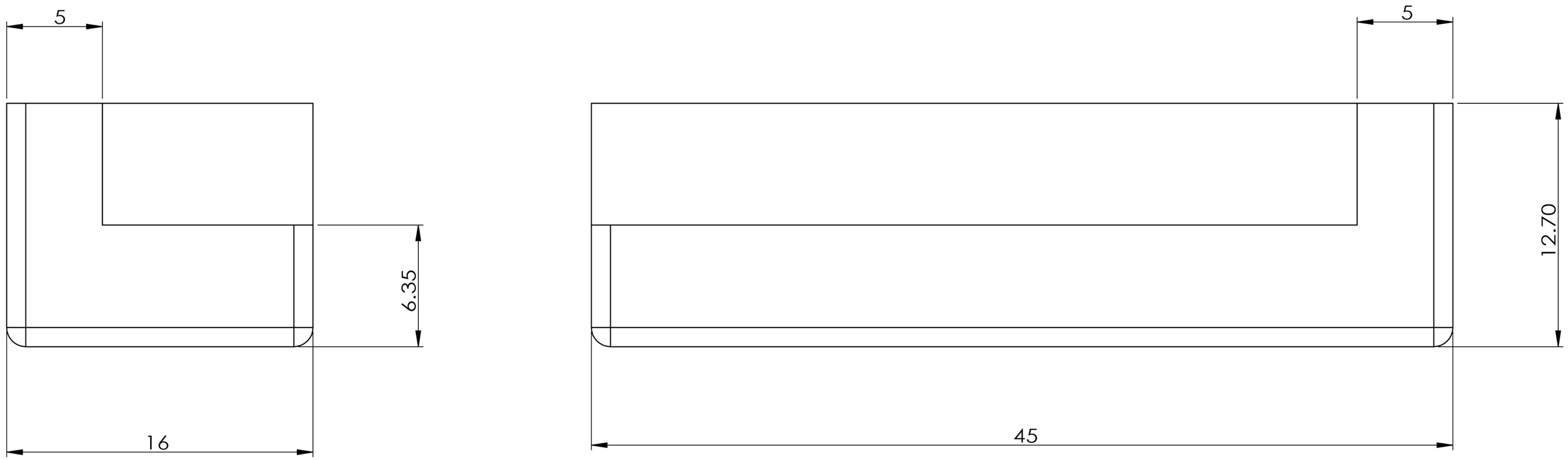

NOTES:
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3. BREAK SHARP OF R1±0.1.

REV	DATE Y/M/D	DESCRIPTION		
1	17/11/27		DWN	SWANG

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE: KEY SEAT

MATERIAL: 6061-T4 (SS)

SIZE: B | DWG. NO.: LS001 | REV: 1

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F12

REV 1

LS002

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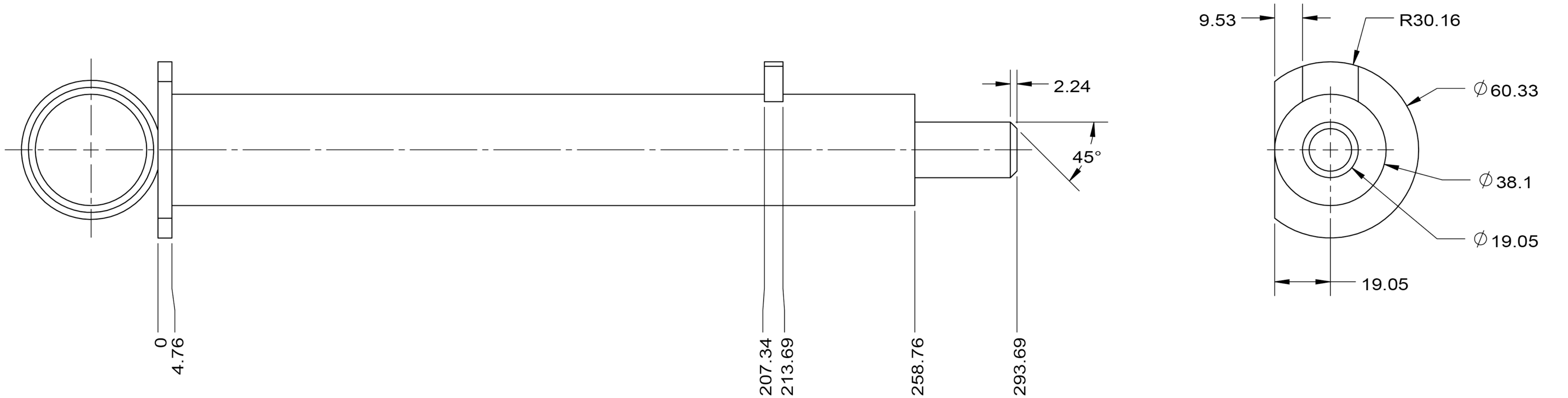

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			CKD	
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			CKD	
			DWN	
			CKD	
1	17/11/27		DWN	SWANG
			CKD	

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MATERIAL
ASTM A36 Steel

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 Engineering Design

TITLE:
ADVANCED TOW PIN

SIZE	DWG. NO.	REV
B	LS002	1

SCALE: 2:3 | SHEET 1 OF 1

Appendix G

Material Properties

Table of Contents

List of Tables

1 Introduction

This appendix discusses the materials used for the lift assist and latch mechanism designs. All relevant materials properties for all design components are presented in the following sections.

2 6061 T6 Aluminum

The material properties for 6061 T6 aluminum can be found in TABLE I. This type of aluminum is used for the grille guard, the grille pin housing, and the grille guard key seats.

TABLE I: 6061 T6 ALUMINUM MATERIAL PROPERTIES [1]

Material	Aluminum 6061 T6	
Brinell Hardness	95	
Specifications Met	N/A	
Unit System	*English*	*Metric*
Density	0.0975 lb/in^2	2,700 kg/m^3
Tensile Strength	45,000 psi	310 MPa
Yield Strength	40,000 psi	276 MPa
Poisson Ratio	0.33	0.33
Modulus of Elasticity	10,000 ksi	68.9 GPa
Shear Modulus	3,770 ksi	26 GPa
Shear Strength	30,000 psi	207 MPa

3 7075 T6 Aluminum

The material properties for 7075 T6 aluminum can be found in TABLE II. This type of aluminum is used for the washer in the lift assist mechanism design.

TABLE II: 7075 T6 ALUMINUM MATERIAL PROPERTIES [2]

Material	Aluminum 7075 T6	
Brinell Hardness	95	
Specifications Met	N/A	
Unit System	*English*	*Metric*
Density	0.0975 lb/in^2	2,700 kg/m^3
Tensile Strength	45,000 psi	310 MPa
Yield Strength	40,000 psi	276 MPa
Poisson Ratio	0.33	0.33
Modulus of Elasticity	10,000 ksi	68.9 GPa
Shear Modulus	3,770 ksi	26 GPa
Shear Strength	30,000 psi	207 MPa

4 Alloy Steel Wire ASTM A401 (Commercial Chrome Silicon)

The material properties for ASTM A401 alloy steel wire can be found in TABLE III. This type of steel is used for the torsional springs in the lift assist mechanism design.

TABLE III: ASTM A401 ALLOY STEEL WIRE MATERIAL PROPERTIES [3]

Material	ASTM A 401	
Rockwell Hardness (min.)	C48	
Specifications Met	N/A	
Unit System	*English*	*Metric*
Density (Referenced as AISI 4140)	0.284 lb/in^2	7,850 kg/m^3
Tensile Strength (min.)	241,000 psi	1,660 MPa
Yield Strength	Not Rated	Not Rated
Bulk Modulus	23,200 ksi	160 GPa
Poisson Ratio	0.29	0.29
Modulus of Elasticity	29,000 ksi	200 GPa
Shear Modulus	11,600 ksi	80 GPa
Shear Strength	30,000 psi	207 MPa

5 AISI 4140 High Strength Steel (ETD 150)

The material properties for AISI 4140 high strength steel can be found in TABLE IV. This type of steel is used for the lift assist mechanism mounting shaft and the spring housing.

TABLE IV: AISI 4140 HIGH STRENGTH STEEL MATERIAL PROPERTIES [4]

Material	High Strength Steel AISI 4140 (ETD 150)	
Brinell Hardness	302	
Specifications Met	ASTM A108	
Unit System	*English*	*Metric*
Density (Referenced as AISI 4140)	0.284 lb/in^2 [5]	7,850 kg/m^3
Tensile Strength	150,000 psi	1,034.2 MPa
Yield Strength	130,000 psi	896.3 MPa

6 18-8 Steel (AISI 304)

There were limited material properties available for 18-8 steel, therefore AISI 304 steel was used as an equivalent material since they have similar properties. The material properties for AISI 304 steel can be found in TABLE V. This type of steel is used for the locking pin in the lift assist mechanism design.

TABLE V: 18-8 STEEL MATERIAL PROPERTIES

Material	18-8 Steel [6]		AISI 304 Steel Equivalent [7]	
Brinell Hardness	Not Rated		123	
Specifications Met	N/A		N/A	
Unit System	*English*	*Metric*	*English*	*Metric*
Density	Not Rated	Not Rated	0.289 lb/in^2	8,000 kg/m^3
Tensile Strength	Not Rated	Not Rated	73,200 psi	505 MPa
Yield Strength	Not Rated	Not Rated	31,200 psi	215 MPa
Poisson Ratio	Not Rated	Not Rated	0.29	0.29
Modulus of Elasticity	Not Rated	Not Rated	29,000 ksi	200 GPa
Shear Modulus (min.)	Not Rated	Not Rated	28,000 ksi	193 GPa
Shear Strength	Not Rated	Not Rated	Not Rated	Not Rated

7 44W Steel (ASTM A36)

There were limited material properties available for 44W steel, therefore ASTM A36 steel was used as an equivalent material since they have similar properties. The material properties for ASTM A36 can be found in TABLE VI. This type of steel is used for the grille guard mounting bracket and the spring seat.

TABLE VI: 44W Steel Material Properties

Material*	Steel 44W [8]		ASTM A36 Equivalent [9]	
Brinell Hardness	Not Rated		Not Rated	
Specifications Met	N/A		N/A	
Unit System	*English*	*Metric*	*English*	*Metric*
Density	0.0975 lb/in^2	2,700 kg/m^3	0.282 lb/in^2	7,800 kg/m^3
Tensile Strength (min.)	65,000 psi	448 MPa	58,000 psi	400 MPa
Yield Strength	44,000 psi	303 MPa	36,300 psi	250 MPa
Poisson Ratio	Not Rated	Not Rated	0.26	0.26
Modulus of Elasticity	Not Rated	Not Rated	29,000 ksi	200 GPa
Shear Modulus	Not Rated	Not Rated	11,500 ksi	79.3 GPa
Shear Strength	Not Rated	Not Rated	Not Rated	Not Rated

Works Cited

[1] ASM Aerospace Specification Metals Inc. *Aluminum 6061-T6; 6061-T651* [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=ma6061t6. [1 December 2017].

[2] ASM Aerospace Specification Metals Inc. *Aluminum 7075-T6; 7075-T651* [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA7075T6. [1 December 2017].

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[4] O'Neal. *EDT 150 Bar* [Online]. Available: http://www.onealsteel.com/alloy-steel-bar-cd-etd-150.html. [1 December 2017].

[5] McMaster-Carr. *Standard Steel Rods* [Online]. Available: https://www.mcmaster.com/#standard-steel-rods/=1ailham. [1 December 2017].

[6] British Stainless Steel Association. *Cutlery Stainless Steel Grades '18/8', '18/10' and '18/0'* [Online]. Available: http://www.bssa.org.uk/topics.php?article=91. [1 December 2017].

[7] ASM Aerospace Specification Metals Inc. *AISI Type 304 Stainless Steel* [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=mq304a. [1 December 2017].

[8] Wilkinson Steel and Metals. *Hot Rolled Steel Bars* [Online]. Available: http://wilkinsonsteel.ca/products/plate-bar-structural-steel/hot-rolled-steel-bars/. [1 December 2017].

[9] MatWeb. *ASTM A36 Material Properties* [Online]. [Accessed 1 December 2017]. Available: http://www.matweb.com/search/datasheet.aspx?matguid=afc003f4fb40465fa3df05129f0e88e6. [1 December 2017].

Appendix H

Manufacturing Quotes

Table of Contents

1 Introduction

This appendix presents the manufacturing quotes provided by companies for certain lift assist and latch mechanism components. The components that require custom manufacturing are the torsional springs and the spring housings for the lift assist design, and the extension spring for the latch mechanism design. It should be noted that the extension spring is not included in the final latch mechanism design, however a quote is provided.

2 Custom Manufacturing Quotes

- The quote for the manufacturing of the torsional springs was provided by Western Spring and Wire Ltd [1]. The full quote can be found on pages H4 to H7.
- The quote for the manufacturing of the extension springs was provided by Western Spring and Wire Ltd [1]. The full quote can be found on pages H8 and H9.
- The quote for the manufacturing of the spring housings was provided by Kinetic Machine Works Ltd [2]. The full quote can be found on page H10.

Report

Advanced Spring Design

Developed for:

My Customer
Customer Street
Customer City, Illinois 0, United States

Developed by:

WESTERN SPRING AND WIRE
55 KEITH ROAD
WINNIPEG, Manitoba R3H 0H7, Canada
Phone: 204 694-7128
Email: gcunha@wsw.ca

Torsion Spring, Standard - Round Wire

Part #

Material Chrome Silicon, Commercial

Description

Configuration Pitched **Coiling Direction** Optional **Grade** Commercial

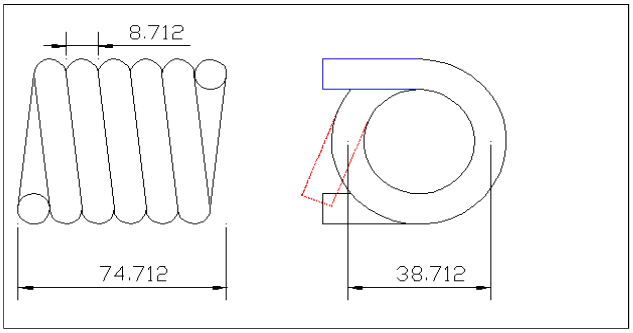
☐ **Shot Peened** ☐ **Use Wahl Factor** ☐ **Equal Arm Lengths**

Wire Diameter [mm]	**8.7122**	Arbor Dia. [mm]	28.3198	Pitch [mm]	**12.0000**
Wire Length [mm]	803.0583	Coil ID [mm]	**30.0000**	Additional Feed [mm]	0.0000
Wire Weight [kg]	0.376336	Coil Mean Diameter [mm]	38.7122	Allowable Body Length [mm]	
Minimum Tensile Strength (MTS) [MPa]	1668	Coil OD [mm]	47.4244	Support Shaft Diameter [mm]	
		Coil Dia. Tol. (+/-) [mm]	0.9906	Load Tol. (+/-) [N-mm]	23404.1451
Spring Rate [N-mm/deg]	1351.9279	Minimum Coil ID [mm]	29.0094	Free Angle Tol. (+/-) [deg]	17.3117
Spring Index	4.4434	Maximum Coil OD [mm]	48.4150		
Natural Frequency, One End Fixed [Hz]	203	Moment Arm Length 1	**30.0000**		
		Moment Arm Length 2	**100.0000**		
Natural Frequency, Both Ends Fixed [Hz]	405				

	Free	Cycle Torque 1	Cycle Torque 2	Other Torque	Set
Torsional Moment [N-mm]	**0**				92027.
Contact Force, Arm 1 [N]	**0**				3067.5981
Contact Force, Arm 2 [N]	**0**				920.2794
Moving Arm's Angle [deg]	180.0000				248.0716
Angle Between Arms [deg]	0.0000				291.9284
Deflection [deg]	**0**				68.0716
ID Stress [MPa]	**0**				
OD Stress [MPa]	**0**				
% of Min. Tensile Strength	**0**				**85.0**
Body Coils	**5.5000**				5.6891
Active Coils	5.8563				
Body Length [mm]	74.7122				
Min. Coil ID [mm]					27.7227

Wire Available	No
Next Smaller Wire Size	8.0000
Next Larger Wire Size	9.0000
Max. Std. Wire Size	**15.8750**

Design Status:



western spring & wire

manufacturing & distribution of precision springs, wire forms & display racking

QUOTATION NUMBER: CS 37085

To: TAG:

Date: Nov 23, 2017

Phone: (000) 000-0000

Attention: MICHAEL

Your reference:
Drawing: WSW37085
Revision:
Specs: CS .343x1.867

Desc: TORSION SPRING
Mat.: CHROME SILICON
Fin.: PLAIN FINISH
Ends: MACHINE CUT ENDS

---

*IMPORTANT: ALL Quotations are based upon the buyer accepting OVERRUNS or UNDERRUNS not exceeding 5% on quantity of product ordered.

Quantity	Unit price
2	94.2600 EACH

NOTE(S): QUOTE BASED ON 1 LH AND 1 RH WOUND

DELIVERY: 1 TO 2 WEEKS

F.O.B: OUR PLANT,WINNIPEG,MANITOBA

NOTE: Delivery Time is an estimate only, based on Production Capacity and orders received as of the quotation date.

GST & PST EXTRA IF APPLICABLE.

TERMS: TBA .

ALL POs MUST INCLUDE TAX EXEMPTION NOs AND/OR CERTIFICATES, IF APPLICABLE.

COMMERCIAL TOLERANCES, AS DICTATED BY THE AMERICAN SOCIETY FOR TESTING AND MATERIALS (ASTM), WILL APPLY TO ALL SPRINGS MANUFACTURED UNLESS OTHERWISE SPECIFIED.

SUBJECT TO WESTERN SPRING AND WIRE LTD.'S STANDARD TERMS AND CONDITIONS OF SALE.

Western Spring and Wire Ltd.
Per: GEORGE CUNHA
(204) 694-7128
FAX: (204) 694-7129
gcunha@wsw.ca

***** PLEASE NOTE THAT ALL PRICES ARE SUBJECT TO RAW MATERIAL SURCHARGES *****
***** IN EFFECT AT THE TIME OF ORDERING. THESE CHARGES, IF APPLICABLE, *****
***** WILL BE SHOWN AS A SEPARATE ITEM ON YOUR INVOICE. *****

western spring & wire

manufacturing & distribution of precision springs, wire forms & display racking

QUOTATION NUMBER: CS 37085

To: TAG: Date: Nov 30, 2017

Phone: (000) 000-0000

Attention: MICHAEL

Your reference:
Drawing: WSW37085
Revision:
Specs: CS .343x1.867

Desc: TORSION SPRING
Mat.: CHROME SILICON
Fin.: PLAIN FINISH
Ends: MACHINE CUT ENDS

---

*IMPORTANT: ALL Quotations are based upon the buyer accepting OVERRUNS or UNDERRUNS not exceeding 5% on quantity of product ordered.

Quantity	Unit price
200	7.1900 EACH

NOTE(S): QUOTE BASED ON 1 LH AND 1 RH WOUND

DELIVERY: 3 TO 4 WEEKS F.O.B: OUR PLANT,WINNIPEG,MANITOBA

NOTE: Delivery Time is an estimate only, based on Production Capacity and orders received as of the quotation date.

GST & PST EXTRA IF APPLICABLE. TERMS: TBA .
ALL POs MUST INCLUDE TAX EXEMPTION NOs AND/OR CERTIFICATES, IF APPLICABLE.

COMMERCIAL TOLERANCES, AS DICTATED BY THE AMERICAN SOCIETY FOR TESTING AND MATERIALS (ASTM), WILL APPLY TO ALL SPRINGS MANUFACTURED UNLESS OTHERWISE SPECIFIED.

SUBJECT TO WESTERN SPRING AND WIRE LTD.'S STANDARD TERMS AND CONDITIONS OF SALE.

Western Spring and Wire Ltd.
Per: GEORGE CUNHA
(204) 694-7128
FAX: (204) 694-7129
gcunha@wsw.ca

***** PLEASE NOTE THAT ALL PRICES ARE SUBJECT TO RAW MATERIAL SURCHARGES *****
***** IN EFFECT AT THE TIME OF ORDERING. THESE CHARGES, IF APPLICABLE, *****
***** WILL BE SHOWN AS A SEPARATE ITEM ON YOUR INVOICE. *****

Report — Advanced Spring Design

Developed for:

My Customer
Customer Street
Customer City, Illinois 0, United States

Developed by:

WESTERN SPRING AND WIRE
55 KEITH ROAD
WINNIPEG, Manitoba R3H 0H7, Canada
Phone: 204 694-7128
Email: gcunha@wsw.ca

Cylindrical Extension Spring, Round Wire

Part # **Material** Music Wire

Description

Hand	Optional	**Configuration**	Close Wound	**End Type**	Machine
Grade	Commercial	**Condition**	Not Preset/Not Peened		

Wire Diameter [mm]	**0.7500**
Wire Dia. Tol. (+/-) [mm]	0.0102
Pitch [mm]	**0.7500**
Coil Spacing [mm]	0.0000
Arbor Diameter [mm]	3.5388
Minimun Tensile Strength (MTS) [MPa]	2290
Free Length Tol. (+/-)	1.3000
Spring Rate [N/mm]	**0.732**
Spring Index	6.3333
Natural Frequency [Hz]	147
Maximum OD [mm]	5.9064
Minimum ID [mm]	3.5936
Wire Availability	No
Next Smaller Wire Size	0.7000
Next Larger Wire Size	0.8000

Coil ID [mm]	**4.0000**
Coil Mean Diameter [mm]	4.7500
Coil OD [mm]	5.5000
Coil Diameter Tol. (+/-) [mm]	0.4064

	End 1	End 2
Tapered Coils per End	0.0000	0.0000
Last Coil Dia. Reduction [mm]	0.0000	0.0000
Hook Radius at Load [mm]	2.3750	2.3750
Hook Radius off Body [mm]	1.5000	1.5000
Hook Extension [mm]	0.0000	0.0000
Hook Length [mm]	3.1250	3.1250
Hook Angle from Tip [deg]	90.0000	90.0000
Hook Gap [mm]	1.4246	1.4246
Hook Wire Dia. [mm]	N/A	
Hook Matl.		

Active Coils	39.9742
Total Coils	39.9742
Dead Coils	0.0000
Body Length [mm]	30.7307
Total Hook Length [mm]	6.2500
Body Coils Life	
Hook Life (Torsion)	
Hook Life (Bending)	
Initial Tension (IT) [N]	4.1430
Max. Obtainable IT [N]	4.9716
Mean Obtainable IT [N]	4.1430
Min. Obtainable IT [N]	3.3144
Initial Tension OK?	OK
Coil Stress at IT [MPa]	147
% Coil Stress at IT	6.4

	Free	Cycle Load 1	Cycle Load 2	OtherLoad	Set
Load [N]	**0**		141.027		29.033
Load Tolerance (+/-) [N]	**0**		5.9061		
Length [mm]	36.9807		223.9807		
Deflection [mm]	**0**		**187.0000**		34.0029
Coil Torsional Stress [MPa]	**0**		5005		
% Coil Torsional Stress	**0**		218.6		
Hook Bending Stress [MPa]	**0**		9483		
% Hook Bending Stress	**0**		414.2		
Hook Torsional Stress [MPa]	**0**		5054		
% Hook Torsional Stress	**0**		220.8		

Body Wire Length [mm]
597.2709

Total Wire Length [mm]
627.1537

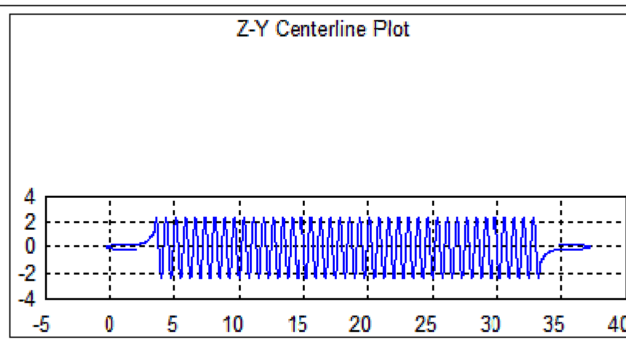
Wire Weight [kg]
0.002178

Additional Feed [mm]
0.0000

Design Status:

Warning(s):

1. Excessive Coil Torsion Stress
2. Excessive deflection.
3. Excessive Bending Stress
4. Excessive Hook Torsion Stress
5. Excessive Hook Bending Stress



western spring & wire

manufacturing & distribution of precision springs, wire forms & display racking

QUOTATION NUMBER: MU 37108

To: TAG: Date: Nov 30, 2017

Phone: (000) 000-0000

Attention: MICHAEL

Your reference:
Drawing: WSW37108
Revision:
Specs: MU .030x0.216x1.455

Desc: EXTENSION SPRING
Mat.: MUSIC WIRE
Fin.: PLAIN FINISH
Ends: REG HOOKS OVER CENTER

---

*IMPORTANT: ALL Quotations are based upon the buyer accepting OVERRUNS or UNDERRUNS not exceeding 5% on quantity of product ordered.

Quantity	Unit price
100	5.3200 EACH

DELIVERY: 3 TO 4 WEEKS F.O.B: OUR PLANT,WINNIPEG,MANITOBA

NOTE: Delivery Time is an estimate only, based on Production Capacity and orders received as of the quotation date.

GST & PST EXTRA IF APPLICABLE. TERMS: TBA .
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SUBJECT TO WESTERN SPRING AND WIRE LTD.'S STANDARD TERMS AND CONDITIONS OF SALE.

Western Spring and Wire Ltd.
Per: GEORGE CUNHA
(204) 694-7128
FAX: (204) 694-7129
gcunha@wsw.ca

***** PLEASE NOTE THAT ALL PRICES ARE SUBJECT TO RAW MATERIAL SURCHARGES *****
***** IN EFFECT AT THE TIME OF ORDERING. THESE CHARGES, IF APPLICABLE, *****
***** WILL BE SHOWN AS A SEPARATE ITEM ON YOUR INVOICE. *****

Kinetic machine works ltd.
ASME B51
509 MERCY STREET P.O. BOX 275
SELKIRK, MANITOBA, CANADA
R1A2B2
PHONE 1-204-785-1536
OR 1-204-785-1955
FAX 1-204-785-1731

MACHINING - MANUAL and CNC – WATERJET and LASER CUTTING
MILLWRIGHTING – MANUFACTURING CONSULTING – SHIP BUILDING – WELDING

Quotation

Quote To: University of Manitoba

Winnioeg, MB

Quote Number:	10678			**Contact:**	Michael Reimer
Quote Date:	11/30/17	**Expires:**	12/30/17	**Inquiry:**	
Customer:	CASH			**Terms:**	Net 30 Days
Salesman:	House			**Phone:**	
Ship Via:	PICK UP			**FAX:**	

Note:

Please allow 2-3 weeks for Pick up or delivery.

If needed tpbe shipped by Kinetic a $50.00 delivery charge for the whole lot would be added.

Thanks for the opportunity to quote.

Item	Part Number Description	Revision	Quantity	Price
1	INNER PIN HOUSING Pin Housing	REV 1	1	$253.45 /EA
			2	$200.95 /EA
			200	$148.98 /EA

By Kevin Naruse

Kinetic Machine Works Ltd.

Works Cited

[1] G. Cunha. [Private Communication]. 23 November 2017.

[2] K. Naruse. [Private Communication]. 29 November 2017.