DEPARTMENT OF MECHANICAL ENGINEERING

MECH 4860

Final Project Report

HERD North America Inc:
Dynamic Wildlife Collision Test Apparatus & Method

PROJECT ADVISOR

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Dr. Nan Wu, Dr. Paul Labossiere
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Dear Dr.Wu and Dr. Labossiere,

Our team, Engineering Design Group 6, is pleased to provide you with our Final Project Report entitled "HERD North America Inc: Dynamic Wildlife Collision Test Apparatus & Method".

The report showcases the engineering design methodology our team took to develop the drop test design. The report outlines how the drop test design meets our client's needs and specifications. Ultimately an analysis of the tower support structure, wildlife model cart assembly, bumper support structure, power system and operational components is provided. We are confident that our design will yield a more than satisfactory product. Thank you for your time and input that provided for us throughout this report.

Please contact Cole Ridd via e-mail if you have any questions or concerns regarding this report. He will be more than happy to resolve and issues.

Sincerely,

Joshua Beauchemin

Brad Poirier

Ian Silk

Cole Ridd

Enclosed Document: Final Project Report

Executive Summary

A dynamic wildlife collision test apparatus and method was successfully designed for HERD North America Industries Inc. The task of this project was to develop a wildlife collision test apparatus and method to simulate a dynamic collision between a moose model and a line of HERD AeroPLUS bumpers. HERD currently does not have any dynamic testing methods and static testing methods with bumpers in the past have yielded results that do not emulate a real life crash scenario.

The determination of client needs and target specifications initiated the project, followed by the development of constraints and limitations. Some of the most important needs include, safety in regards to assembly and operation, repeatability of results, and the simulated collision resembles a real wildlife collision. Some constraints and limitations include, the ability of HERD or one of HERD's suppliers to be able to manufacture the test apparatus and the test location is limited to HERD's property.

Following a detailed research phase, the project was divided into various design components, including the moose model, test apparatus and power method. Concepts for each component were considered and analyzed though concept generation and concept selection phases before a final design was determined.

The final design of this project features a drop test style design were a steel tower support provides the structure for which a wildlife model cart assembly with moose model insert is collided into a HERD bumper mounted on a bumper support structure. The gravity powered system is powered further with the addition of spring and rope assemblies. Operation of the apparatus is conducted through the use of a winch, safety stop, quick release mechanism and sheaves. This system simulates an impact between a 200 kg animal and a semi-trailer moving at 80 km/s at a total cost of $41086.09.

Lastly, the final deliverables required by the client were provided. These deliverables include CAD models of the test apparatus and bumper support structure, a detailed moose model design, provided through both written description and CAD model, and a methodology describing the collision test, provided in the form of a procedural list for the testing method.

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1. Introduction

This report will provide details on our final design of the dynamic wildlife collision test method our team designed for HERD North America. It offers insight into the calculations performed to come to conclusions on more in depth design aspects of certain components of the design. In addition it provides rationale and reasoning for the selection of various components. A cost analysis is provided for the major components of the design. Failure mode and effect analysis is provided as well as finite element analysis of the tower support structure.

1.1. Background Information and Problem Statement

HERD North America offers collision protection for trucks and semitrailers with full bumper packages, grill guards and bumper guards. The trucking industry demands the best protection for its front ends as well as light and aerodynamic options. HERD has proven the effectiveness of their designs through static finite element analysis, customer testimonials and testing, including a vibration test and static load test. The current vibration test consists of forcing compressed air at a bumper while it is attached to a steel mount and support structure. The vibration test apparatus is pictured in Figure 1.

Figure 1: Current vibration test apparatus consisting of a compressed air delivery system and steel mount and support structure [1]

The current static load test consists of applying a single point load to a fixed bumper with increasing force over time until failure of the bumper occurs. These static tests do not show consistent data in comparison to dynamic highway collisions. The result of the failed bumper from the static load test is pictured in Figure 2. To meet new industry demands of lighter and easier designs HERD is upgrading their bumpers to a new design. One challenge for the new design is to make the new design lighter without compromising strength or ease of use. HERD currently has no procedure to test dynamic wildlife collisions with their current or new bumper designs. As a result, HERD requires a test method to compare the strength of the current, proven designs with their new designs. Our team will provide HERD with a dynamic collision test method that supports a collision between their bumpers and a wildlife stand in.

Figure 2: Failed deformed bumper following the current static load test utilized by HERD [2]

Figure 3 shows the AeroPLUS Bumper that our team will be working with to develop a CAD model test apparatus and dynamic collision procedure to illustrate the impact between the Bumper and a moose replica dummy.

Figure 3: HERD's current AeroPLUS Bumper [3]

1.2. Project Objectives

In order for HERD to compare proven designs with current designs, multiple project objectives were developed. HERD has requested the design of a test procedure, with CAD models and drawings, of a collision between their AeroPLUS Bumper and a moose model. The design will allow HERD to draw conclusions regarding the strength of their new design when the bumper collides with wildlife under highway conditions. In order to draw these conclusions, the expectation is that once the project is complete, HERD will retrofit the test apparatus to test current bumper designs. All the conclusions HERD requires will be analyzed through the use of a video camera during the collision.

The design team must provide the following three deliverables to successfully meet HERD's requirements:

1. CAD models of a test apparatus to support the AeroPLUS Bumper that can withstand repetitive collisions and can be retro fitted to support other HERD bumper models.
2. A moose model that replicates the size and density of an average moose and that can either be used for multiple tests or that is economical to reproduce for single use testing.
3. Test procedure a method to cause a dynamic collision, at a variety of locations, on the bumper and a moose model.

In summary, HERD requires a test procedure that will provide comparative data between their current bumpers and their research and design bumper designs.

1.3. Customer Needs and Target Specifications

In order to determine the customer needs and subsequently develop metrics and specifications for those needs, our group went through a thorough consultation with HERD and our project advisors. We had an initial meeting with HERD on September 15, 2016, which provided us with the necessary information to develop a preliminary list of customer needs. After consulting with Ed Hohenberg, and clarifying some of the requirements of the project with HERD, we came up with a final list of needs, and their respective priority levels. These

needs and priority levels have been confirmed by HERD. Our list of customer needs is provided below in TABLE I.

TABLE I: LIST OF CUSTOMER NEEDS AND PRIORITY LEVELS

Need No.	Customer Need	Priority Level
1	The test apparatus is compatible with the bumper provided by HERD.	5
2	The apparatus survives the collision.	5
3	When repeated, the test provides consistent results.	5
4	The test is safe to assemble and operate.	5
5	The collision occurs at a vertical location on the bumper that simulates the height of the centre of gravity of a moose.	5
6	The dummy transmits a similar amount of energy to the bumper in the collision.	5
7	The apparatus can be broken down into parts movable by a forklift.	4
8	The test can simulate a variety of collision speeds.	4
9	The test can simulate a variety of collision locations on the bumper.	4
10	The dummy can simulate a variety of different weights.	4
11	The collision mechanism allows for observation by camera.	4
Need No.	**Customer Need**	**Priority Level**
12	The design of the dummy allows for reuse.	4
13	The apparatus and the dummy are designed using only commonly available materials.	3
14	The apparatus and the dummy can be manufactured by HERD, or its suppliers.	3
15	The apparatus and the dummy do not exceed budget provided by client.	2

In the table of needs above, and in the table of metrics and specifications below, the apparatus and the dummy refer to separate elements of the final product. The dummy refers to

the physical model of a moose that is provided for the test, while the apparatus refers to every other physical element of the test.

Need 12 refers to the reuse of the dummy design. This need encompasses two broad conceptual designs that will be explored in the conceptual design phase. It could refer to a physical model that survives multiple impacts, and the same model is used over and over with no waste. Or, reusing the design could also refer to a design that involves a single use physical model, after which a new model is required. The design of the dummy is the key reusable item, not the physical dummy itself.

The priority levels in right column of the TABLE I represent the importance of satisfying the corresponding need, with a 1 being the least important and a 5 being the most important. The priority levels were determined through internal consultations.

After determining the customer needs, a list of 18 quantifiable metrics were developed to allow the customer needs to be accurately measured. These metrics represent tests that can be applied to the final product to determine if the design meets the needs of the customer. The specifications on the right of the table represent the target, or ideal, specifications of the design. Some of the specifications, such as the maximum speed of the test, were acquired during our meeting with our client during our site visit of HERD. Other specifications, such as the weight and height of a moose, were researched, and others, such as the energy transferred between a moose and a bumper in a collision were derived. These metrics have all been confirmed by HERD. The needs and specifications are provided in TABLE II.

TABLE II: LIST OF METRICS ADRESSING NEEDS, WITH CORRESPONDING SPECIFICATIONS

Metric No.	Corresponding Need No.	Metric	Units	Target Specification
1	1	Apparatus attaches to the bumper.	Pass/Fail	Pass
2	2	Cycles to failure of the apparatus.	No. of Cycles	500
3	3	Percent difference in collision force in successive trials.	Percentage	≤ 3%
4	4	Maximum spread of debris from collision location.	m	10m

5	5, 9	Vertical movement range of the dummy with respect to the bumper.	m	2.5m
6	5	Height of the collision location between the bumper and the centre of gravity of the dummy.	m	1.5m
7	6	Energy transferred from moose to bumper in collision	J	49kJ
8	6, 10	Weight of the dummy.	kg	800kg [4]
9	7	Maximum size of one piece of the apparatus.	m^3	1000m^3, with a maximum of 10m in any one direction
10	8	Maximum simulated collision speed.	km/h	100km/h
11	8	Range of simulation speeds available.	km/h	40km/h range (not absolute speed)
12	9	Horizontal movement range of the dummy with respect to the bumper.	m	2.5m
13	10	Dummy design allows for different dummy weights	kg	Range of 800kg
14	11	Percent of bumper visible to the camera at moment of impact.	Percentage	100%
15	12, 15	Cost of the dummy.	Canadian Dollars	$10,000, when combined with Metric 18
Metric No.	**Corresponding Need No.**	**Metric**	**Units**	**Target Specification**
16	13	Commonly available materials such as steel, aluminum, rubber, etc.	Pass/Fail	Pass
17	14	Simple and common machining techniques such as welding, metal cutting, etc.	Pass/Fail	Pass
18	15	Cost of the apparatus.	Canadian Dollars	$10,000, when combined with Metric 15

The calculation of the amount of energy in the collision is a two-step process. It is based on a moose weighing 800kg being at rest at the start of the collision, and a fully loaded 8-axle

truck (weighing 62,500 kg [5]) travelling at 80 km/h hitting the moose. First, conservation of linear momentum is applied, and assuming the final velocities of the moose and the truck to be identical, a final velocity of the moose is determined. Second, the final velocity of the moose is used to calculate the kinetic energy gained by the moose in the collision, representing the total energy transfer in the collision. This model also assumes that the only energy changes in the moose-truck system are in the kinetic energy of the two objects, in a conservative manner.

The goal of determining the moose collision height with the bumper was to make the centre of gravity of the moose hit the bumper. Four assumptions were made to determine the impact height. The first was that the collision was with a large moose, with a shoulder height of 2m [4]. The second assumption is that the height of this moose's legs is 1m [6]. Third, this model assumes that the moose has no weight in its legs, and fourth, it assumes that its body is perfectly oval in shape. These assumptions make it easy to calculate that the centre of gravity of a moose is at a point 1.5m off the ground.

1.4. Constraints and Limitations

To fully define the scope of a project, the design limitations and constraints must be identified. By identifying the limitations and constraints our team can address their implications and will increase the efficiency in our team's concept creation and decisions making processes. In addition, by having a well-defined scope, scope creep will be eliminated allowing for a final product that will be finished on time and completely satisfies our client.

To identify the limitations and constraints we met and consulted with our client. This meeting provided us with information that allowed us to identify the customer needs, which in turn allowed for the determination of the project and design limitations and constraints. The limitations and constraints are documented below. They have been broken down into two sections, the first being for the project and the second being for the product design.

The project had a very limited time window. The deadline was firm and the project absolutely could not go past December 7th, 2016. This led to having a small window to complete our project, including the concept generation, concept selection and final concept phases.

1.4.1. Product Design

The product has been broken down into three main design components: the test apparatus, the moose replica dummy and the dynamic collision method. Each of the components has their own limitations and constraints. These include design space, materials, manufacturability and other specific limitations depending on the design component. These are provided in the sub-sections below.

Before going into the individual component methods, there are two limitations and constraints that encompass all three main design components. The first is the material selection. The material that will be used for the components must be commonly available to HERD North America. This means that expensive, high tech materials are not to be used in the component design. With regards to the dynamic collision method, any vehicle or equipment that would be used in the method has to be sourced within Canada or the United States. Our team will had to consult with our client to ultimately determine if a material could be used in a design since they will be fabricating the components and conducting the tests.

Secondly, our client provided our team with a soft budget for the entire design. We had to attempt to keep the project in the $10,000 range. This budget is very soft, and acted more as a guideline than anything. HERD told us not to worry if it was exceeded as they believe they can source raw materials for much less than any prices we can find, and manufacture the test structure and components in house.

1.4.1.1. Test Apparatus

The test apparatus was treated as an open concept that will vary in function depending on the dynamic collision method. Regardless of the method there will need to be apparatuses to hold both the AeroPLUS bumper and the moose dummy before the test is commenced. Together they will be referred to as the test apparatus.

The test apparatus must be designed in a manner that HERD or ones of HERD's suppliers may manufacture the test apparatus. The test apparatus had to be able to be lifted by a forklift at HERD's facilities so that it may be loaded for transportation to a test site. This limited the test apparatuses size and weight.

1.4.1.2. Wildlife Replica Dummy

The design constraint for the moose replica dummy was that it must survive repeated use or be simple and cheap to produce such that HERD can manufacture it on site. When designing the moose replica dummy, we had to compare the cost to life span of a potential design. For the scope of our project we were originally asked to simulate a moose, however this was scaled down to simulating a deer. The design we choose for a wildlife replica dummy had to be able to be re-configured/altered to different weight and sizes to as small as 20 kg.

1.4.1.3. Dynamic Collision Method

The greatest limitation of the dynamic collision method is safety. Since the will be no formal crash test experts conducting these tests, safety is a great concern. The test must be safe enough to be performed by the staff of HERD North America. We had to rely on our intuition and consultation of the client to determine whether a concept for the dynamic collision method was feasible to safely perform.

1.4.1.4. Design Space

To be able to make concept selections the design space of our project must be fully defined. To define the design space, we met with our client. The design space was then clearly outlined. HERD wants to be able to perform the dynamic collision test on their company's property. It could be performed inside their manufacturing facility, product storage tent or outside on their property grounds. Figure 4 shows HERD's property from Google Earth satellite imaging. Its approximate property lines are highlighted in yellow.

Figure 4: HERD North America approximate property lines [7]

From the scale provided in Figure 4 the approximate dimensions of the property are 250 m by 175 m. Labelled, is one of the manufacturing facilities and the product storage tent, denoted by the red A and B respectively. The inside of the manufacturing facility (A) has an approximate 20 m by 20 m area that could be used for testing. The ceiling of said space has an approximate height of 6.5 m. In addition the floor is paved with smooth concrete and a garage door that is approximately 5 m tall and 5 m wide. The inside of the product storage tent (B) is entirely open

and can have remaining products removed if necessary. The area inside is approximately 25 m by 60 m and the maximum height of the ceiling is roughly 9 m. The ground is gravel as opposed to the paved floor in facility (A). Neither, ceiling of A or B is to be used to support the test apparatus. Finally, the rest of the outdoor property is covered in gravel or tall grass. If a concrete anchoring platform or pile is required in the final design of a concept, our client has indicated that they would allow it to be created on their property. Finally, if a test is to be performed in the outdoors, it should be able to perform the safely and accurately in various weather conditions. For example, the test should perform the same on a clear sunny day versus a cloudier, colder day with some rain. Any concept we create must be able to be performed within these constraints.

1.5. Research

At the beginning of the project, a comprehensive external research phase was conducted in order to ensure a thorough exploration of potential designs and alternatives. The main methods utilized to conduct this phase include the determination of relevant industry leaders, literature review, patent search, and viewing visual media, such as videos, schematics and pictures. Figure 5 shows a flow diagram of the external research phase.

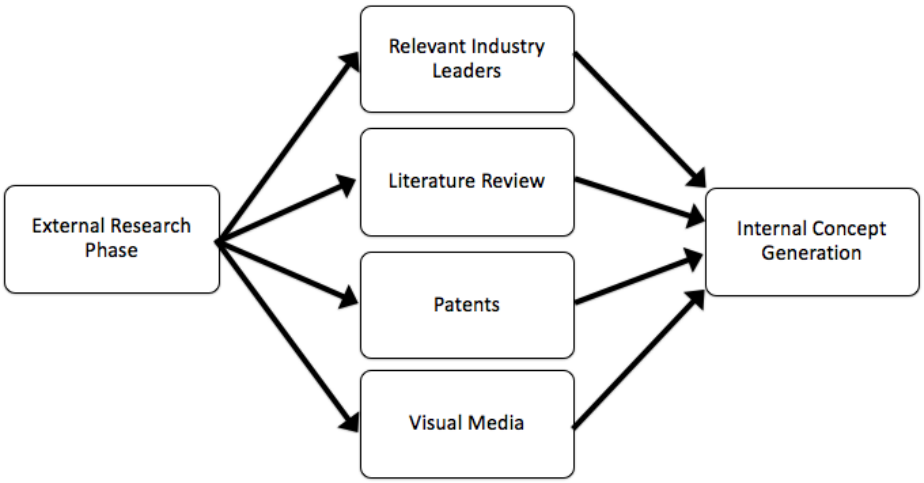


Figure 5: Flow diagram indicating external research phase components.

1.5.1. Relevant Industry Leaders

The Swedish National Road and Transport Research Institute, or *Statens väg- och transportforskningsinstitut* (VTI) was the most relevant industry leader found in regards to wildlife collision simulation. VTI has done extensive vehicle crash testing involving a full size moose model and has detailed descriptions of crash test equipment and function on their webpage [8]. Another relevant industry leader in wildlife crash testing is Germany's ADAC automobile club that has conducted crash tests on model wild boars [9]. These organizations give general insight and provide benchmarks to the industry of wildlife collision simulation.

1.5.2. Literature Review

A search for relevant literature was also conducted. Two papers published by VTI were found to be the most relevant: Magnus Gens' master's thesis titled "Moose Crash Test Dummy" and "Evaluation of the moose dummy Mooses II with a view to consumer guidance," authored by Ylva Matstoms. Gens' 2001 master's thesis describes the construction of a moose model and details the dynamic collision test apparatus and method used to validate the model. For the purpose of this project, this paper provides excellent insight into a possible moose model design and test apparatus and method. A possible limitation to a reusable moose model is presented and should be considered. After 70 km/h the moose model began to warp and deform and would require additional reinforcement to endure repetitive collisions [10]. Figure 6 shows the moose model and set up of the simulated vehicle collision.

Figure 6: Image from Magus Gens' master's thesis showing the moose model and vehicle crash simulation set up [10].

In a follow up to Gens' thesis, Matstoms' 2003 report evaluates the moose model for durability and realistic behaviour during a dynamic collision. Matstom also offers suggestions towards moose accidents in order to provide consumer guidance [11].

1.5.3. Patents

Several patents were found detailing vehicle crash test apparatus. The patents are all quite similar in that only slight variations of the test apparatus differentiate one from another. Also, few details are given to the exact construction and methodology of the test apparatus; rather, the general idea of the design is claimed. A patent held by Chirstopher Dragan of Toyota

details a vehicle simulated crash test apparatus consisting of a pair of rails, a vehicle buck and a suspension system to support the buck above the rails. An actuator accelerates the rails with the attached vehicle buck to simulate the crash. This patent does not include information about the power system but is especially useful when considering a fixed track rail system design for this project [12]. Figure 7 shows a drawing from the submitted patent.

Figure 7: Drawing showing rails, a vehicle buck and a suspension system from Christopher Dragan's patent [12].

A patent from Breed Automotive Technology Inc. details a sled test apparatus and method for simulating a pre-impact crash event before a collision. A pre-impact crash event refers to an event such as applying the brakes before a collision or striking a smaller obstacle before colliding into a larger object. The apparatus consists of the pre-impact force-generating device using a bungee sled with spring loaded moving seat and an impact generating piston. This patent is of interest when considering a collision simulated by a piston as opposed to an accelerated vehicle [13]. Figure 8 shows a sketch of the apparatus.

Figure 8: Sketch of the apparatus consisting of the vehicle sled, pre-impact simulating moving seat and impact piston [13].

A patent held by Walter Cerny of Ford Motor Co details an apparatus for vehicle crash test simulation consisting of a crash vehicle towed by a cable behind which is a data recording unit mounted to a follower vehicle. A braking system exists to ensure the follower vehicle is not involved in the collision. The design detail of interest in this patent is the towing cable functionality, as data collection is not a requirement of this project [14]. Figure 9 shows a sketch of the apparatus.

Figure 9: Sketch of the apparatus consisting of the vehicle, follower vehicle with data recording equipment and towing cable mount [14].

1.5.4. Visual Media

Visual media was analyzed in order to gather observations regarding test configurations and methods. The Mythbuster's episode, "Moose Mayhem," involved a simulated vehicle crash

between a car and a moose. In the episode, the moose model design used was based on Gens' master's thesis design. The moose model was constructed from rubber plates cut from a CNC machine supported with wire cable. The Mythbusters contracted a company specializing in crash simulations to perform the test. The company's apparatus consisted of a truck attached to a 2:1 pulley system powering the test vehicle, which included a steering hub assembly attached to a cable for guidance. A small hill stopped the vehicle after the crash until a remotely controlled actuating braking system was installed [15]. Figure 10 shows a screenshot of the Mythbuster moose model and test apparatus.

Figure 10: Screenshot from Mythbusters's, "Moose Mayhem" episode showing the moose model, test vehicle and apparatus, including cable pulley system and cable guiding system [15].

A simulated vehicle moose collision test performed by VTI was also viewed. The video gave good insight into the test apparatus and how the dynamic collision occurred. Details of the apparatus useful to this project include the rail guiding system used to direct the vehicle to the collision, as well as the cable pulley system used to accelerate the vehicle [16]. Figure 11 shows a screenshot of the VTI moose model and test apparatus.

Figure 11: Screenshot from VTI's moose crash simulation. The moose model shown is Magus Gens' master's thesis design [8].

Finally, ADAC automobile club's video of their wild boar crash simulation provided additional insight into wildlife crash simulation methodology and apparatus's. In the video a human operator can be clearly seen driving the vehicle into the collision. Using a human driver for this project addresses issues of guidance, accuracy, braking and powering system but introduces serious safety concerns [17]. Figure 12 shows a screenshot of the ADAC boar model and vehicle collision.

Figure 12: Screenshot from ADAC automobile's club wild boar simulation vehicle collision [17].

In summary, the provided research allows for a solid foundation and fundamental understanding of wildlife vehicle collisions prior to the concept generation phase. This will allow for more effective generation of practical solutions and increase the likelihood of success for the final chosen design.

2. Concept Development Phase

Once we had fully defined our project, we needed to develop concepts to analyse and use an engineering methodology to systematically remove all concepts but the best concepts. This was done in four phases and in four parts: We first brainstormed and developed concepts to screen in phase three. The second phase took the project needs from TABLE I and developed them into specific needs for each component of the design. Phase three was a subjective screening of all the concepts developed, to narrow the field of designs that would be compared in detail in the final selection phase.

These four phases were conducted on each of the four main components of our design: the power system, the dummy design, the aiming method, and the apparatus of the test. After the completion of the concept development and selection for each component, the best pieces were mixed and matched to determine the best two overall designs, which were further analysed to determine the best design. The details of the concept development phase are provided in Appendix A. A summary of the process is provided in this section to explain the rationale behind choosing the freefall design.

Phase 1: Concept Generation

We developed a total of 36 different component concepts and eight different combined components to screen. Some of the component concepts are not universal, and rely on the selection of certain concepts in different component categories (for example, using an old semi truck as a bumper mount would rule out using gravity as the power source completely, and implies that the best power method is using the truck's engine). Some of the concepts were as simple as using a truck to tow either the dummy or the bumper into a fixed, stationary bumper

or dummy, while some concepts were as complex as designing an air cannon or solenoid to launch the dummy into the bumper.

Phase 2: Defining and Prioritizing Component Specific Needs

The second phase required to determine a concept to pursue is to take the needs developed in TABLE I and specialize them for each component. Not all project needs are applicable for each concept. For example, the dummy model is independent from the Need 1 (compatibility with the AeroPLUS bumper) and therefore that need doesn't need to be considered when filtering the bumper concepts. Likewise, some needs are relevant on the component scale but aren't relevant when considering the project as a whole. For example, one of the needs developed for the power system dealt with the compatibility of the power system, specifically stating that "The power system is compatible with different types of moose models and bumpers."

The apparatus had a broad range of needs that needed to be addressed, such as:

- The compatibility with the AeroPLUS bumper
- The durability under impact loading
- The accuracy of the test method when the specific apparatus was used
- The portability of the apparatus
- The range of speeds allowed by the apparatus
- The weight of the apparatus; this was closely related to the portability of the apparatus
- The ease of observation of the test allowed by the apparatus
- The cost of the apparatus

The dummy had much fewer needs that needed to be addressed as compared to the apparatus, including:

- The compatibility with the rest of the apparatus
- The durability under impact loading
- The adjustability of the weight of the dummy

- The cost of the dummy
- How realistic the dummy is when compared to a moose or deer

The power system had a relatively unique set of needs developed to evaluate the concepts. These needs included:

- The consistency of the test with the power system
- Remote use of the power system
- The accuracy of the power system; this is related to the consistency
- The portability of the power system
- The cost of the power system
- The size and complexity of the power system

Finally, needs were developed for the aiming system, including:

- The durability under impact loading
- The consistency of testing with the aiming concept used
- The adjustability of the impact location
- The speed allowed with the aiming concept
- The ease of testing observation allowed by the aiming system

As a team, we took all the needs for each component separately and compared them against one another to determine the priority levels and accurate priority weights of each need. Each need was given a score between 0 and 2 when compared with the other needs for the component. The total scores for each need were summed and compared against the total points distributed to determine the need score as a percentage of the total points awarded. This process showed that the durability was the most important consideration when designing the apparatus; that the compatibility with the rest of the test apparatus and the realism of the dummy were the most important dummy design considerations; the testing consistency was the most important power system need; and that the consistency and safety provided by the aiming system were key needs.

Phase 3: Concept Screening

The first stage of the concept screening process involved determining whether moving the dummy into a fixed bumper or moving the bumper into a fixed dummy was more beneficial. This entire calculation is provided in Appendix A. The evaluating factor was the amount of speed required to move the moving component and maintain an equivalent collision. Since the bumper is lighter than the dummy, it requires a higher final velocity for an equivalent collision and therefore we will fix the bumper in place and move the dummy towards it.

The second part of concept screening involved comparing all the concepts generated against the determined needs in each component category. If we subjectively determined that a concept is strong at addressing a need, it was assigned a plus. If a component was weak at addressing a need it was given a minus, and if a concept is neutral with respect to a concept it is given a same. The total plusses for each concept were summed and the number of minuses were subtracted to determine the top four concepts in each category. From this, the best four concepts for each component were carried through to the concept selection phase. The top four apparatuses are:

- The vertical tower
- The crane
- The pendulum
- The swinging arm

The top three bumper supports are:

- The large mass mount
- The small mass mount
- The steel truss “vibration mount”

The top four dummy concepts are:

- Sandbags in tires
- A water container
- Pure plastic

- Rubber over a mass

The top four power system concepts are:

- Gravity
- Springs
- A cable with a winch
- A truck pulling the dummy

Finally, the top four aiming concepts are:

- Cables in tension
- Mechanical aiming
- Straight rails
- Curved rails

These concepts are carried forward into the concept scoring and selection phase.

Phase 4: Concept Scoring and Selection

The final stage in the concept selection process is scoring the best concepts as determined by the concept screening, then taking these scores and applying them to the components in the overall concepts we generated. The two concepts with the highest scores must be further analysed to determine the best one design to pursue in our detailed design.

The concept scoring process was done by giving each component a score between one and four for each component need, and multiplying this score by conceptual need weight determined in Phase 2. The best concept at addressing each need got a score of four, while the worst component at addressing a given need was assigned a one.

After this process was complete, two overall group concepts remained. The first was the "Merry-go-Round" concept, consisting of spring power, mechanical aiming, any dummy design (the top scoring dummy design was the sandbags in tires design so we selected that one) and the swinging arm apparatus. The second was the "Drop Test" concept, consisting of gravity and springs for power, cables in tension as an aiming system, any dummy design, and the vertical tower apparatus. Upon further analysis we determined that the compression on the cables

during the drop test would have a severe negative effect on their accuracy in the drop test, and changed the aiming system to a mechanical system.

3. Final Concept Selection

To determine a final concept we needed to perform a preliminary analysis to determine the system dynamics of each concept. By determining the dynamics, it is then possible to approximate important variables such as general forces that would be acting on the test apparatus' and the work or power required to accelerate the moose dummy to the desired collision velocity. This will enable us to further compare the concepts in terms of the product requirements. The following sections will describe the processes and results taken for the concepts preliminary analyses.

3.1. Drop Test Preliminary Analysis

The drop test system is one of the two concepts that require evaluation in the form a preliminary analysis. To do this, the concept must be converted into a dynamic model that is representative of how it will perform in the real world under ideal conditions. This concept utilizes the force of gravity in free fall acting on the mass as well as some undetermined spring/elastic system to aid in accelerating the moose model to collision velocities. The simple nature of the system allows for a simplified analysis. First, the concept is converted into a 1 dimensional model. This simplifies and idealizes the type of motion we expect to observe from this concept. Next, the system is converted into a simple vertical spring mass system. The wildlife model is converted into a point mass, m (kg), and has two forces acting on the mass; the force of gravity, g (m/s^2) and the spring force. The spring/elastic is assumed to have an un-stretched length of x_L (m), a stretched length of x_o (m) and a spring constant K (N/m). The height at which the mass is held above the ground is denoted by h (m) and the final velocity of the wildlife model the instant before the collision is V_f (m/s).

TABLE III: SPRING MASS SYSTEM VARIABLES

Variable	Notation	Unit
Mass	m	kg

Variable	Notation	Unit
Gravity	g	m/s^2
Un-stretched Spring Length	x_L	m
Stretched Spring Length	x_o	m
Spring Constant	K	N/m
Height of Model	h	m
Instantaneous Velocity at Collision	V_f	m/s

In this model, we will assume that the moose model is separated from the spring when the spring reaches its un-stretched length. This allows for the moose to achieve its maximum possible velocity from the spring before relying on gravity to accelerate it to the desired final test velocity. The system is shown in Figure 13 below.

This analysis will be used to the determine spring property requirements for a corresponding drop height. We will analyze the system with the use of an energy balance equation. The potential energy stored in the system can be calculated and then equated to the equivalent kinetic energy that mass would obtain. The equation relating potential and kinetic energy is shown below in equation (1) below. The potential energy (P.E.) and kinetic energy (K.E.) are both expressed in the units of Joules (J).

$$K.E. = P.E. \tag{1}$$

The potential energy of the system can be determined with the variables previously mentioned in TABLE III and Figure 13 above. The kinetic energy of the system has been determined to be 49 kJ as previously stated at the beginning of this section. The potential energy of the system can be determined using the following equation.

$$P.E. = mgh + \frac{1}{2}K{x_o}^2 \tag{2}$$

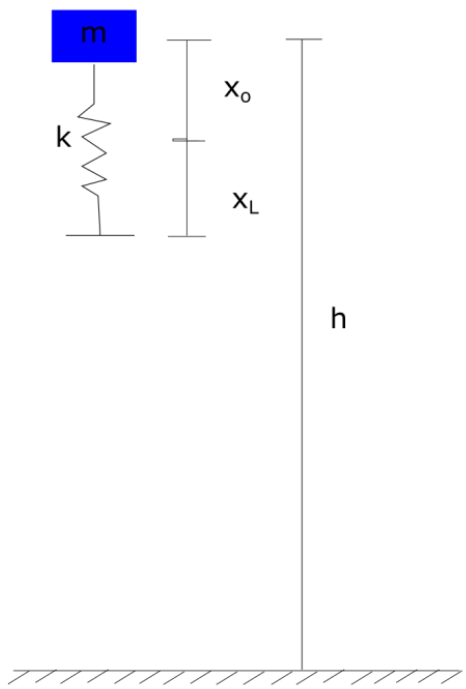


Figure 13: Spring mass simplification of Drop Test [18]

It is important to note that the spring must be a linear spring for equation (2) to be valid. This will be discussed in further detail in section 5.5.1. For this analysis it is assumed that the spring is linear. In addition, we will assume all but one of the variables stated in TABLE III and then calculate unknown variable. TABLE IV states the assumed variables, leaving the spring constant (*K*) to be determined.

TABLE IV: ASSUMED VARIABLE VALUES FOR DROP TEST PRELIMINARY ANALYSIS

Variable	Value	Unit
m	200	kg
g	9.81	m/s^2
x_L	0.5	m
x_o	0.5	m
h	6.096	m
V_f	22.22	m/s
K.E.	49	kJ

Rearranging equation (2), we can isolate the spring constant value. Then inserting the known variables we can solve for the spring constant.

$$K = \frac{2(K.E. - mgh)}{{x_o}^2} = \frac{2(49\times10^3 \ - 200\times9.81\times6.096)}{(0.5)^2} = 296317 \text{ N/m} \qquad (3)$$

This value is large for one single spring acting on the wildlife model. However, if the design assumes that there are multiple springs in parallel, the spring constant is then divided by that number since the potential energy is distributed among those multiple springs. Springs that provide enough force would need to be sourced or created. The modularity of this concept is desirable in the sense that many variables can be altered to achieve the desired collision energy. By altering the mass of the dummy, the spring constant and the drop height many crash velocities and scenarios can be achieved. The test bumper can be moved into different positions to provide the desired collision location with the wildlife model. The wildlife model would need to be rigidly fixed to the guide rail system or would need to have an apparatus that accelerates to model on target and then allows the model to collide freely with the test bumper. A drawback to a rigidly fixed model is that there is a loss in accuracy of a real world collision when the model collides with the bumper since the model can only travel in the vertical direction.

3.2. Merry-Go-Round Preliminary Analysis

The Merry-Go-Round is the second concept that requires a preliminary analysis. In this analysis the momentum of the wildlife model before impacting the bumper will be equated to that of a 200kg deer just after the moment of impact from a fully loaded semi-truck travelling at 80km/h. To calculate momentum, the following equation is used.

$$p = mv \qquad (4)$$

Where *p* (kg-m/s) is momentum, *m* is mass (kg) and *v* (m/s) is velocity. The momentum of a 200kg deer the instant after impacting the semi-truck travelling at 80 km/h (or 22.22 m/s) is calculated. We assume that the collision is completely inelastic the instant after the collision and that the deer is accelerated to the semi-trucks velocity instantaneously.

$$p = 200 \times 22.22 = 4444 \ \frac{kg - m}{s} \tag{5}$$

This amount of momentum required by the moose dummy the instant before the impact with the bumper during the Merry-Go-Round Test. A diagram of the Merry-Go-Round test is shown in Figure 14. The variables for this test include the mass of the wildlife model *m* (kg), the tangential velocity of the wildlife model V_T (m/s), the radius of the model mounting arm *r* (m) and the angular position of the model ϑ (radians). These are presented in TABLE V.

TABLE V: MERRY-GO-ROUND TEST VARIABLES

Variable	Notation	Unit
Mass	m	kg
Model Tangential Velocity	V_T	m/s
Mounting Arm Radius	r	m
Angular Position	θ	radian

Initially the wildlife model is at rest at position ϑ_o = 0 radians and V_o = 0 m/s. The mass of the model is assumed to be 200kg. The radius of the mounting arm is assumed to be 1m. Therefore, the moose model must be have a tangential velocity at the time of impact with the bumper of 80km/h or 22.22 m/s to have the amount of momentum calculated in equation (5).

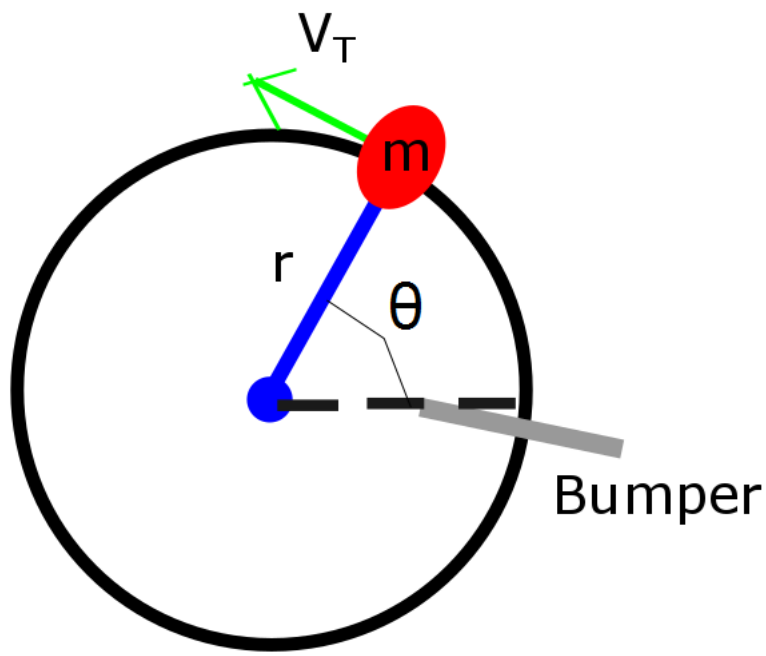


Figure 14: Merry-Go-Round test simplified diagram [19]

The final angular position of the wildlife model, or in other words the collision point of this test will occur at $\vartheta_f = 2\pi$ radians. The assumed variables for this test are presented in TABLE VI.

TABLE VI: ASSUMED VARIABLE VALUES FOR MERRY-GO-ROUND TEST

Variable	Value	Unit
m	200	kg
V_T	22.22	m/s
r	1	m
θ	2π	radian

To obtain the collision velocity of 80km/h in one revolution, the amount of power transmitted to the system must be calculated. To do so, the acceleration required to bring the model up to the collision speed must be calculated. Since the wildlife model is travelling about an ark, the angular velocities and accelerations are to be calculated. Equations (6) and (7) are used.

$$\omega = \frac{V_T}{r} \text{ rad/s} \tag{6}$$

$$\alpha = \frac{a_T}{r} \text{ rad/s}^2 \tag{7}$$

This introduces three new variables which will be used in determining the power requirement. The three variables are angular velocity ω (rad/s), the angular acceleration α (rad/s^2) and the tangential acceleration a_T (m/s^2). Before the power can be determined, the acceleration of the model must be determined. Ignoring air resistance and frictional losses, the dynamic system relating the angular velocity and angular acceleration is given by a simple equation.

$$\Delta\omega^2 = 2\alpha\Delta\theta \tag{8}$$

Rearranging equation (8), we can solve for the angular acceleration. Then rearranging equation (7), the tangential acceleration can be determined. The angular acceleration is calculated in equation (9).

$$\alpha = \frac{\Delta\omega^2}{2\Delta\theta} = \frac{(V_T/r)^2}{2\times 2\pi} = \frac{(22.22/1)^2}{2\times 2\pi} = 39.29 \text{ rad/s}^2 \tag{9}$$

From the calculated angular acceleration, the tangential acceleration is determined to be 39.29m/s^2. Next, the torque T (N-m) required to accelerate the wildlife model to the tangential acceleration is calculated.

$$T = Fr = ma_T r \tag{10}$$

Inserting our assumed and calculated variables into equation (10), we calculate the torque.

$$T = 200\times 39.29 \cdot 1 = 7858 \text{ N-m} \tag{11}$$

Knowing the torque, the power P (kW) required to accelerate the wildlife model to the collision velocity is calculated. The equation for power is described below. In addition the calculated values that we previously used are inserted into the equation.

$$P = T\omega = T\frac{V_T}{r} \tag{12}$$

$$P = 7858 \times \frac{22.22}{1} = 174.61 \text{ kW} \tag{13}$$

For sourcing a power source, in the case of a motor, it is important to determine the rotation per minute *RPM* (rev/min) at which the wildlife model would be travelling. This is calculated using a simple conversion.

$$RPM = \omega\frac{60}{2\pi} = 22.22 \times \frac{60}{2\pi} = 212.19 \tag{14}$$

Thus the required RPM of the rotating arm and wildlife model would need to be 212.19. The calculation results for the Merry-Go-Round analysis are tabulated in TABLE VII.

TABLE VII: MERRY-GO-ROUND PRELIMINARY ANALYSIS CALCULATION RESULTS

Variable	Value	Unit
a_T	39.29	m/s^2
T	7858	N-m
P	174.61	kW
RPM	212.19	rev/min

For sourcing purposes many motors use Imperial units. Therefore, the required torque and power have been converted to ft-lbf and HP (horsepower) respectively. The results of the conversion are presented in TABLE VIII.

TABLE VIII: MERRY-GO-ROUND IMPERIAL TORQUE AND POWER VALUES

Variable	Value	Unit
T	5795.84	ft-lbf
P	234.16	HP

The method of powering the Merry-Go-Round is undetermined. The concept scoring provided the best power source to be spring force however upon review a more suitable source would be a motor or engine. Using a motor or engine would allow for a gear reduction to be

used injunction so that the desired torque and RPM could be achieved for the rotating arm while maintaining lower engine or motor operating RPM. The approximate 235 HP required to power the arm can be reduced by increasing the radians that the arm travels before impacting the model. Unfortunately, if the rotating arm and wildlife model have to rotate more than 1 full revolution, a system for timing the placement of the bumper in front of the wildlife model would need to be added. A benefit of this test is that is closely simulates a collision accurately in the sense that the collision direction occurs parallel with the ground. This concept would require some type of breakaway system for the wildlife model upon colliding with the bumper. We believe it to be necessary due to the large forces involved in the collision that would act upon the rotating arm apparatus if the wildlife model was fixed to the arm. In addition an accurate guide track would be required to keep the wildlife model on an accurate trajectory. The test bumper can be moved into different positions to provide the desired collision location with the wildlife model.

3.3. Final Selection

During the final concept scoring, the Drop Test concept scored higher than the Merry-Go-Round test with scores of 12.55 to 11.47 respectively. After completing a preliminary analysis our team was able to use the results as well as input from our team advisor and client to determine the final test concept to proceed with into the design stage. The chosen concept was the Drop Test.

There were many reasons for choosing the Drop Test over the Merry-Go-Round. The first being safety. Having a large rotating mass being accelerating to high velocities in a very short distance comes with risks. For example, if the model becomes decoupled with the rotating arm during the acceleration phase it could be launched in any direction. This applies to any of the other rotating parts as well. The safety of the test operators is a top priority. Having the collision occur in a linear direction during the Drop Test reduces the chances of injury or property damage if a failure in the test system occurs. Secondly, the modularity of the Drop Test was greater than that of the Merry-Go-Round. Being able to alter the drop height and mass of the wildlife in conjunction with springs that can be attached or detached is appealing to our

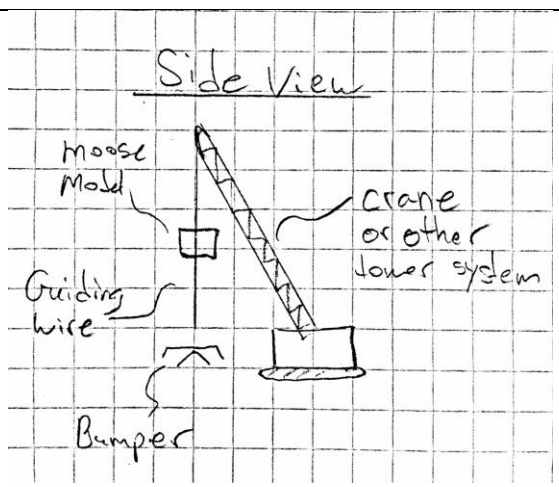
client. Lastly, the simplicity of the drop test is appealing. Having less moving parts is a benefit to the Drop Test. A complex system will have more possibilities for failure than a simpler system. The simpler system is likely to be less costly and providing a failure in a component, be easier to replace or repair.

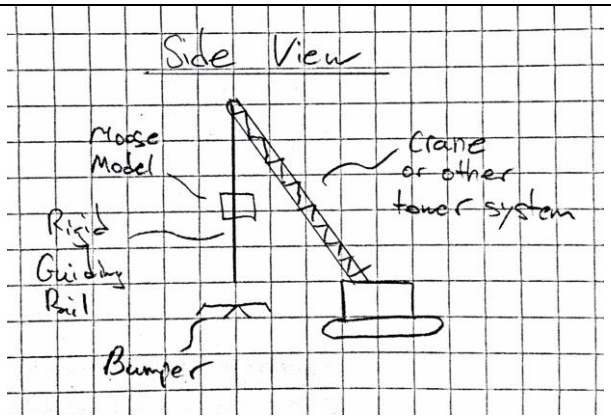
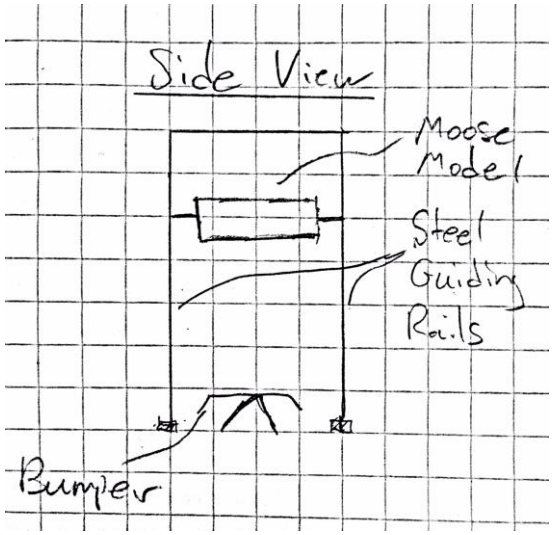
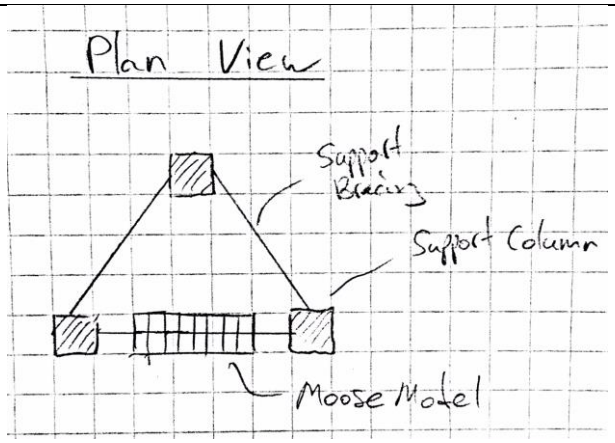
Going forward with the Drop Test will require designing a system that can safely and accurately impact the wildlife model with the test bumper. This will require a support structure with guide rails, a bumper support structure, springs or elastics to accelerate to model and a method for loading the model and springs with potential energy before unloading it instantaneously.

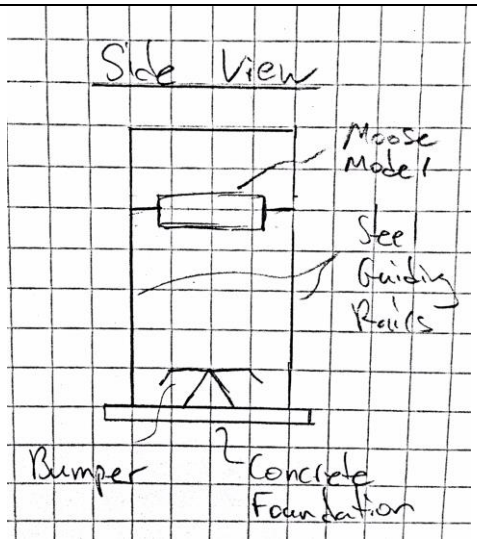
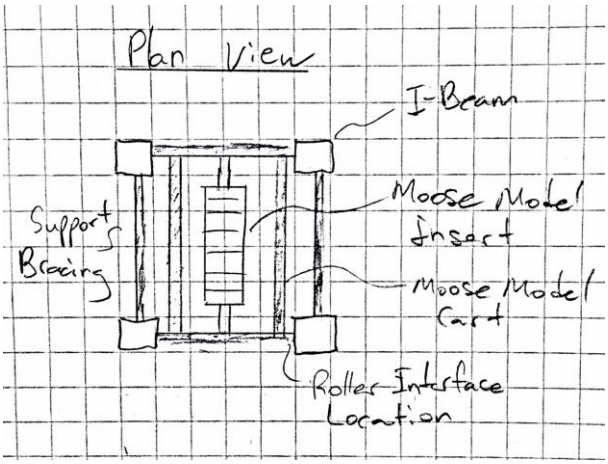
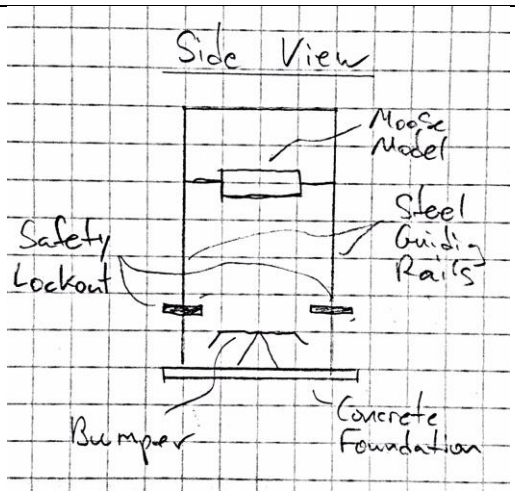
4. Final Design Optimization

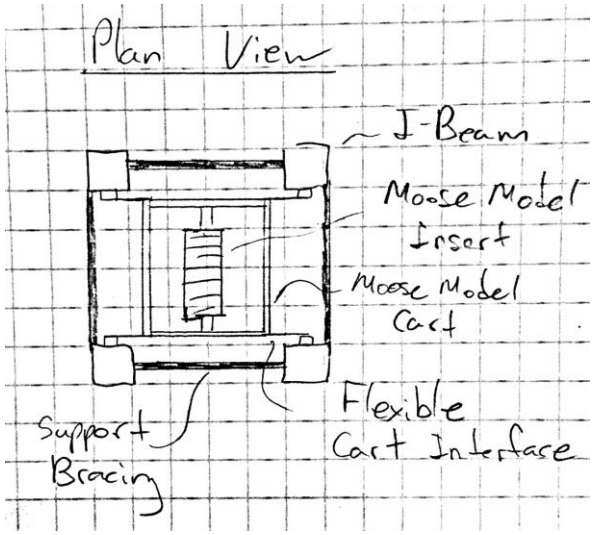
The purpose of this section is to outline the development and progression of the drop test final design strategy. Different methods of performing the drop test were considered while refining the strategy down to the final overall design. This iterative approach of the final design process is outlined in TABLE IX.

TABLE IX: FINAL DESIGN OPTIMIZATION [20]

Iteration	Drawing	Description
1		Guiding wire directs the dropped rubber plate wildlife model design to the bumper. A crane or other tower system supports the wire and wildlife model.

Iteration	Drawing	Description
2		To meet accuracy requirements specified by the client the loose guide wire is replaced with a rigid steel rail.
3		An extra support rail is added to provide more stability to the system. The wildlife model is incorporated into a cart assembly. This design also incorporates a spring power system to reduce the height of the apparatus.
4		An extra support rail is added to provide more stability to the system. This provides three support columns.

Iteration	Drawing	Description
5	 	An extra support rail is added to provide more stability to the system. This provides four support columns. The wildlife model cart assembly is positioned in the center of the support columns for added stability. Additionally, a concrete foundation is added to support the entire tower structure.
6		For added safety a lockout system is added to the design. To satisfy cost requirements the wildlife model is now constructed from steel wheels.

Iteration	Drawing	Description
7		A flexible interface between the support columns and the wildlife model cart assembly is added to prevent jamming during the drop test. Rubber tires are also added to the face of the concrete foundation to dampen the system following a collision.

5. Final Design Details

After determining the needs and requirements of our design and conducting a detailed concept selection and conceptual design, we are fully prepared to begin a detailed design. The detailed design will include a detailed description of the design space we have at our disposal. It then progresses into the design of the tower support structure, including the concrete foundation and the steel tower. Next, we detail the design of the wildlife dummy carriage, including the design of our wildlife model, the rollers used to ensure that the design has a predictable collision location, and the frame of the carriage.

The fourth component of our detailed design is the bumper support structure. This looks at the theory behind using strain energy to determine stress in objects experiencing impact loading, and uses this to design the ideal bumper support structure. Following that is the spring powering system design, which considers a vast array of springs that could be used to power the collision simulation, including detailed designs of both helical expansion and flat springs.

Finally, we design the operational components of the collision modelling apparatus, including the winch used to prime the collision, and the required cable and sheaves to haul the wildlife carriage to the top of the tower. This section also details the design of safety featured added to the design, and the quick release mechanism added to the structure.

5.1. Drop Test Design Components and Design Space

To help confine the final detailed design, we need to identify the design space we have, and we need to identify the components we must design and the space we must design the components in. This is broken down into two sections, the first being the required components and the second being the design space.

5.1.1. Design Components

There are four main components that we need to design to complete the test apparatus. They are:

- The test tower
- The dummy cart
- The power system
- The bumper support structure

In addition to these main components, we need to design a method to raise the dummy cart to the top of the test tower, and a method to ensure that the dummy cart does not begin freefall until the test is initiated by the test operators. This small component is called the test mechanism.

The test tower is the first main component of the test. This component includes the tower that houses the dummy cart as it is dropped on top of the bumper, as well as the concrete pad that the whole test is housed on. This component must operate in unison with the dummy cart, allowing for a nearly frictionless drop of the dummy onto the bumper. It also must stay stable when the dummy is raised to the top to begin the test. It must allow for the test mechanisms to operate smoothly. Finally, the concrete pad must be large enough to contain all

components of the test. An ideal scenario for the apparatus as a whole is that the whole test is self-contained within the footprint of the test structure.

The dummy cart is the next main test component. By definition, it must weigh 800kg. This weight generates enough gravitational potential energy when the dummy is raised to the top to make springs a feasible powering system. An ideal dummy cart will have a modular weight addition/removal system, so tests of different weights may be conducted. The actual dummy must be made of rubber, as rubber is a good, durable substitute for actual moose flesh. The dummy cart must also be accurately, reliably, and with a relative lack of friction, be guided to the desired bumper collision location.

The power system is the third major test component. It must provide the remainder of the energy that the freefall of the dummy does not to accurately simulate the desired collision, in the form of spring potential energy. The springs chosen must not provide a torque to the dummy cart. The springs chosen must fit within the test tower, or just slightly outside of it. The power system also may include rope or cable to attach the top of springs with small deformations to the dummy cart. Springs may extend above the top of the bumper support structure, but may only extend a long distance beyond this if the spring chosen is designed to have large deformations. The springs must not obstruct the travel of the dummy cart in any way.

The final major component is the bumper support structure. The bumper support structure must be compatible with the AeroPlus bumper provided by Herd. The structure must be durable enough to survive a collision with the dummy cart. It must fit inside the support structure, but it also must be able to be moved in one direction to simulate collisions with a moose at different horizontal locations on the bumper. It must me light enough that this motion can be completed with either a forklift, or a front-end loader that Herd owns. It finally must be able to be fixed to the concrete pad to prevent bumper motion during impact.

The minor component is the testing mechanism. It consists of a winch that must be able to pull the fully loaded dummy up the tower, while loading the springs that provide the extra spring force to the dummy. It attaches to a cable that also must be able to support this load.

This cable must go around pulleys that guide the cable to the dummy cart. Finally, the test mechanisms must include a release mechanism that can be operated remotely and reliably when the dummy is brought to the top of the tower.

5.1.2. Drop Test Design Space

Herd has provided little in the way of restrictions to our test space. We may design the tower as wide, long, or tall as we desire, if it is necessary. They have given us permission to recommend the installation of a concrete pad to house this test apparatus. Therefore, the design space limitations are all guidelines for us to design our contraption.

The first and second guideline place height restrictions on the bumper stand, the power system, and the tower. These height restrictions are relatively arbitrary. The design would work just as well with any combination of freefall heights selected, and would only affect the design of the aforementioned components. However, determining the most effective balance between freefall height, and the height of the bumper stand only affects the scale of the power system and the tower. This determination is outside the scope of the project, as Herd is more concerned with the methods used to provide the collision, i.e. using a helical spring or using a flat spring, than optimizing the spring energy used as compared to the gravitational energy used.

The first guideline that we have put in place is that the total freefall height is 10 feet. This allows for a good balance of freefall, with heavy duty springs still being required to provide the remainder of the force. The second requirement we have put in place is that the bumper will be held 3 feet above the ground, and therefore the dummy must be dropped from 13 feet. Changing the height that the bumper is held at affects the allowable deformation of the springs, as we wish that the springs do not pass the top of the bumper by a large amount at full extension. This also affects the design of the bumper support structure, as a higher height allows for more impact to be absorbed by beams of a smaller cross-section. However, as we already discussed, this is outside the scope of this project, as Herd is more concerned with the test method than anything else.

5.2. Tower Support Structure

The tower support structure provides the structure for which the wildlife model cart assembly will be dropped, as well as the attachment points for the spring power system and pulleys. The tower support structure consists of the tower itself, constructed from structural steel and the concrete foundation.

5.2.1. Structural Steel

The structural steel comprising the tower support structure consists of four 18 feet high main I-beam columns. The I-beam support column bases form a rectangle that are 105 inches apart in one direction and 121 inches apart in the other. This placement forms a rectangular support structure. The columns are supported by steel cross bracing at the top of the structure in order to withstand the forces of the loaded wildlife model cart assembly and the dynamic collision.

5.2.2. Concrete Foundation

The concrete foundation provides the base support and anchoring point for the structural steel. The concrete foundation consists of a mesh of reinforcing steel filled with concrete to form a visible concrete pad with bolted anchoring point at surface level. The dimensions for the concrete base are 4 m x 4 m x 0.25 m and includes 36 J bolts as anchoring points.

5.2.3. Tower Analysis

In order to determine the stresses in the tower, we must first conduct a static truss analysis on the tower to determine the forces at all points in the members of the structure and consequently the stresses in the truss members. A skeleton structure was used as a basis for this analysis and is shown in Figure 15. This figure labels dimensions, member notation and forces that will be used in the analysis.

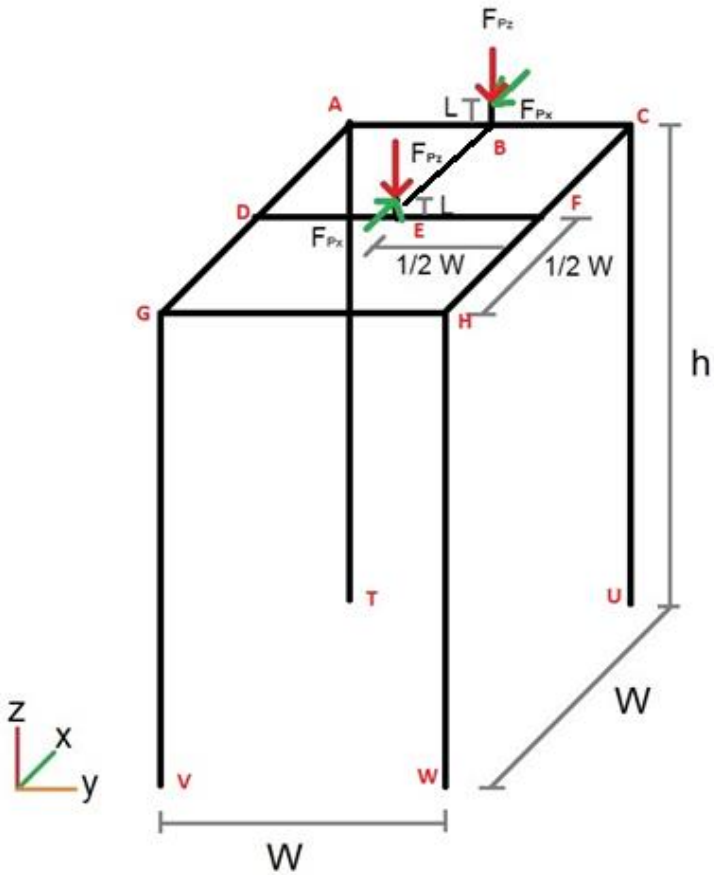


Figure 15: Tower structure truss frame [21]

For our structure we chose to use ASTM A992 Wx12 steel beams. The dimensions for this beam type are presented in TABLE X, and the properties of ASTM A992 are presented in TABLE XI. It should be noted that the columns have the web in parallel with the forces in the x direction. The cross beams will have the web perpendicular to the ground.

TABLE X: TRUSS W - BEAM DIMENSION

Dimension	**Value (in)**
Flange Width	3.97
Web Thickness	0.2
Height	11.97

Dimension	Value (in)
Flange Thickness*	0.5

TABLE XI: Wx12 ASTM A992 MATERIAL AND SECTIONAL PROPERTIES

Property	Value	Unit
Yield Stress	60000	psi
Young's Modulus	29000	ksi
Cross-sectional Area	6.152	in^2
Moment of Inertia (strong)	156.16	in^4
Moment of Inertia (weak)	150.94	in^4

5.2.3.1. Force Analysis

As this is a three-dimensional structure, this means that the forces and applied moments in all directions for all members and joints must be equal to zero, or

$$\sum F_x = \sum F_y = \sum F_z = 0 \tag{15}$$

$$\sum M_x = \sum M_y = \sum M_z = 0 \tag{16}$$

The first member we wish to analyze is member BE. An FBD of member BE is shown in Figure 16.

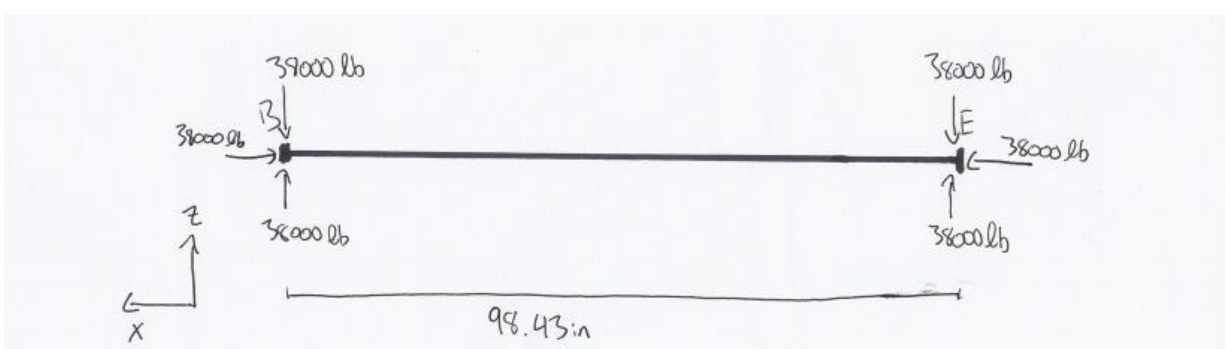


Figure 16: FBD of member BE [22]

Member BE has inputs of 38000î lb - 38000k̂ lb at E, and -38000î lb – 38000k̂ lb at B. In addition, there is a moment of 342 klb-in applied at B and an equal and opposite moment applied at E by the displacement of the sheaves from the joints. The forces in the x direction and the applied moments are equal and opposite, and therefore the sum of the forces in the x direction and all the applied moments to the member are balanced out. There are still residual forces that must be balanced in the member equal to -76000k̂ lb. Due to symmetry we can determine that there must be a reaction force of 38000k̂ lb at both points B and E from member's ABC and DEF, respectively, as well as a force of 38000î lb and -38000î lb, respectively.

Next we can analyze member's ABC and DEF. An FBD of member ABC is shown in Figure 17.

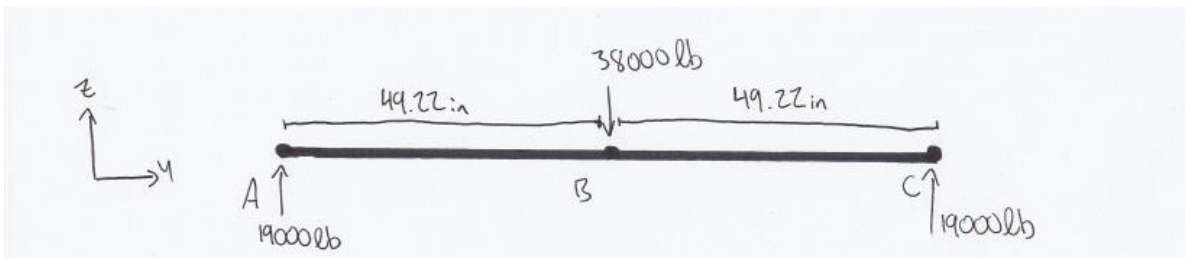


Figure 17: FBD of member ABC [23]

These members have input forces of -38000k̂ lb, as well as -38000î lb and 38000î lb, at points B and E, respectively. Through symmetry, we can observe that there are required to have equal support forces at points A, C, D, and F, all equal to 19000k̂ lb. In addition, there is This satisfies both the requirements for the members' sum of forces in the z direction to be equal to zero and the sum of the moments in the z direction to also equate to zero.

The third group of similar members that need to be analyzed are member's ADG and CFH. An FBD of member ADG is shown in Figure 18.

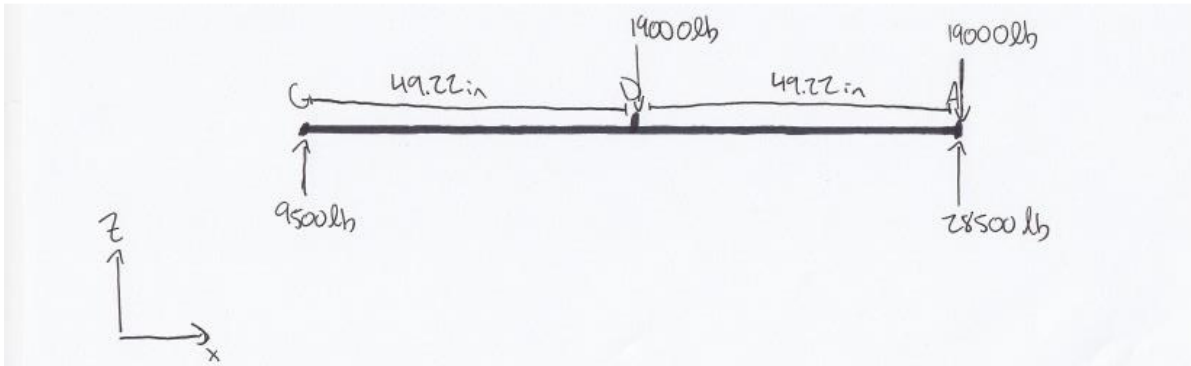


Figure 18: FBD of member ADG [24].

The reaction forces at the ends of these beams must be equal to 38000$\hat{k}$ lb. Because this is not a case of symmetry, the moment and force balance equations must be used to determine the proportions of the reaction forces at A and C, as well as at G and H. Taking the sum of the moments at A gives:

$$\sum M_A = 19000lb * 49.22in - R_G * 98.43in = 0$$

$$R_G = 9500lb$$

Summing the forces present in the beam gives:

$$\sum F_z = -19000lb - 19000lb + 9500lb + R_A = 0$$

$$R_A = 28500lb$$

The x direction inputs at points A and D balance themselves, as do the x direction inputs at C and F.

The only members left to analyze are the supports AT, CU, GV, and HW, and the final cross bracing GH. Since the only forces at points G and H are in the z direction, and GH runs perpendicular to the Z direction, it is a zero force member and no further analysis is required on it. The supports AT, CU, GV, and HW are parallel to the applied force from the node at their top, and therefore the force in them is equal to the force at nodes A, C, G, and H, respectively.

5.2.3.2. Stress Analysis

There are three concerns that must be considered in the stress analysis of the structure are bucking, bending stress and axial stress. Not all members will experience buckling. Only the members under axial compression need to be checked when looking at buckling. Likewise, only members that have applied moments need to have a bending stress analysis conducted. Finally, members experiencing axial forces must have axial stress checked.

If the stress in any member is greater than the yield stress of the steel used, it is considered to fail and the member needs to be changed to comply with the yielding stress in the cases of bending and axial stress. If the critical buckling force is less than the force in a member in compression, then that member will buckle, and changes to the geometry of the structure must be made to ensure that the members will not buckle.

3.3.2.2.1. Members in Bending

There are five members in the structure that experience bending stress: ABC, ADG, BE, CFH, and DEF. Members ABC and DEF are identical and can be analysed together, as can members ADG and CFH. The general equation for the maximum bending stress is:

$$\sigma_{Bending} = \frac{M_{Max}c}{I} \tag{17}$$

Where *c* is the distance from the bending axis to the point in question and *I* is the inertia of the beam in the direction of the bending.

In member's ABC and DEF, the max moment is in the middle of the beam, at nodes B and E, respectively. The moment at that point is equal to the force at A times half the length of the beam, or 935085$\hat{k}$ lb-in. The distance between the bending axis and the farthest point from it is half the height of the member, or 5.955 in, and the inertia in the direction of bending is 156.16 in^4. This information gives a maximum bending stress of 35.66$\hat{k}$ ksi. Additionally, the inputs in the x direction are identical as the inputs in the z direction. The difference between the bending in the z and x directions is the orientation of the I beam. Since the I beam is not symmetrical on all four sides, the moment of inertia of the beam changed. This beam is oriented so that the weak bending direction is in the I direction. This orientation has a moment

of inertia of 150.94 in^4. The distance to the maximum bending point is also smaller, and is only 1.985 in. This info gives bending stress of 12.30î ksi.

These stresses need to be combined to determine the overall stress in the member and compare it to the yield strength of the beam. Since there is no shear stress in the beam, we can assume that the stresses in the x and z direction are equal to two of the three principal stresses in the member, with the third being equal to zero. These can be used to calculate the von Mises stress in the beam using the formula:

$$\sigma_{vM} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2}{2} + \frac{(\sigma_2 - \sigma_3)^2}{2} + \frac{(\sigma_3 - \sigma_1)^2}{2}} \tag{18}$$

Where $\sigma_{1,2,3}$ are the principal stresses of the beam, and $\sigma_{x,z}$ are equal to any of them. Since σ_x and σ_z are bending stresses with equal compressive and tensile components, when calculating the term with the difference of the two stresses, one must be taken as a negative and one must be taken as a positive. As this term is squared, it does not matter which stress is assumed to be compressive and which one is tensile. The von Mises stress in ABC and DEF is 43.15 ksi, and this is the stress compared to the yield strength.

Members ADG and CFH are identical in bending to member's ABC and DEF in the z direction, as they have the same cross section and length, however they have half the force and therefore moment at the maximum moment point. This means that members ABC and DEF have a maximum bending stress of 17.83 ksi. There is no bending in the x direction in these beams.

Member BE has moments applied by the sheaves at each end, and therefore the bending stress is constant along the length of the member. The inertia and thickness are the same as the rest of the members in bending. Using the moment already determined in section 5.2.3.1 we get a maximum bending stress of 13.04 ksi.

3.3.2.2.2. Members Experiencing Axial Stress

There are seven members in the structure that experience axial stress: the supports AT, CU, GV, and HW, as well as the members BE, ADG and CFH. Under axial forces, the stress is defined as:

$$\sigma_{Axial} = \frac{F}{A} \tag{19}$$

F is the axial force in the member and *A* is the cross-sectional area in a member. Since all members have identical cross-sections, they all have cross-sectional areas of 6.152 in^4. Using the forces determined in each member in section 5.2.3.1, we find that the stress in members AT and CU are 4653 psi, the stresses in members GV and HW are 1551 psi, and the stress in member BE is 6204 psi. The stress in ADG and CFH is 3088.43 psi

We have two different normal stress components in members BE, ADG, and CFH: an axial stress and a bending stress. The bending stresses have maximum tensile components on the bottom face and maximum compressive components on the top face. Adding the maximum compressive component with the maximum compressive axial stress gives a maximum stress in the member of 19.24 ksi in memberBE. Members ADG and CFH have total stresses equal to 20.92 ksi.

3.3.2.2.3. Members Experiencing Compression

Any members experiencing compressive forces must be checked to confirm that there is no chance of buckling in the member. This is done using Euler's buckling criterion:

$$F_{Max} < F_{crit} = \frac{\pi^2 EI}{L_e^2} \tag{20}$$

Where F_{Max} is the maximum force in a member, F_{crit} is the critical buckling stress in a member, *E* is Young's modulus for the beam's material, *I* is the moment of inertia in the beam in the weak direction, and L_e is the equivalent length of the beam, and is a function of the connections on the beam. We assume that all beams ends are fixed to resist both a moment

and applied force in all directions. This is a conservative assumption as this makes L_e = 2L, while assuming that all connections are pinned gives an L_e = L and a higher critical force.

The members that must be checked for bending are the supports AT, CU, GV, and HW, as well as the members BE, ADG and CFH. All members are made of the same material and have the same cross-section, at 29000 ksi and 150.94in^4, respectively. The supports have a length of 216 in, and using a L_e of 2L, this gives the supports a critical force of F_{crit} = 231.49 klb. Since the critical force is much larger than any of the forces transmitted through the supports, there is no risk of buckling in the supports. Likewise, member BE has a length of 47.42 in and an F_{crit} = 4.803 Mlb, which is again much higher than the force carried by the member. Members ADG and CFH have the same length of member under compression as member BE, and so the critical force for BE can be applied for members ADG and CFH. Therefore, there is no chance of buckling in any of the members in the structure.

3.3.2.2.4. Summary of Forces and Stresses on Structural Members

Tables XII to XV provide a summary of the forces and stresses in the members of the structure.

TABLE XII: FORCES AT ALL NODES IN ALL MEMBERS IN THE STRUCTURE

Member	Node	z Force (lb)	x Force (lb)
BE	B	-38000	-38000
	E	-38000	38000
ABC	A	-19000	-19000
	B	38000	38000
	C	-19000	-19000
DEF	D	-19000	19000
	E	38000	-38000
	F	-19000	19000
ADG	A	-9500	19000
	D	19000	-19000
	G	-9500	0
CFH	C	-9500	19000
	F	19000	-19000
	H	-9500	0
AT	A	28500	0
	T	-25800	0
CU	C	28500	0
	U	-28500	0
GV	G	9500	0
	V	-9500	0
HW	H	9500	0
	W	-9500	0

TABLE XIII: BENDING STRESSES IN APPLICABLE MEMBERS

Members in Bending	Maximum Bending Stress (ksi)
ABC	43.15
ADG	17.83
BE	13.04
CFH	17.83
DEF	43.15

TABLE XIV: AXIAL STRESSES IN APPLICABLE MEMBERS

Members Under Axial Stress	Axial Stress (psi)
AT	4653
CU	4653
GV	1551
HW	1551
BE	6204

TABLE XV: COMBINED AXIAL AND BENDING STRESSES IN APPLICABLE MEMBERS

Members with Multiple Stress Components	Combined Bending and Axial Stress (ksi)
BE	19.24
ADG	20.92
CFH	20.92

5.3. Wildlife Model Cart Assembly

The wildlife model cart assembly consists of three distinct components, including the steel support structure, wildlife model insert and roller interface. All components contribute to the weight of the wildlife model totalling 800 kg.

5.3.1. Steel Support Frame

The steel support frame provides structural stability and connects the roller interface to the wildlife model insert. The steel support frame is made from thick durable steel to ensure survival through repeated collision simulations. The overall dimensions of the rectangular steel support frame are 100” x 112”.

The steel support frame was designed to withstand not only the vertical forces experienced during loading scenarios but also for deflection. Two types of deflection can affect the columns that the carriage rollers follow. These two deflections are loading deflection and temperature deflection. To ensure that neither deflection will cause the carriage to bind in the frame, a cantilever beam analysis was completed.

The 2-ton trolley connection to the carriage was designed to deflect in the case of deflection. The square tubing used will have the top and bottom surfaces removed from the tube as seen in Figure 19.

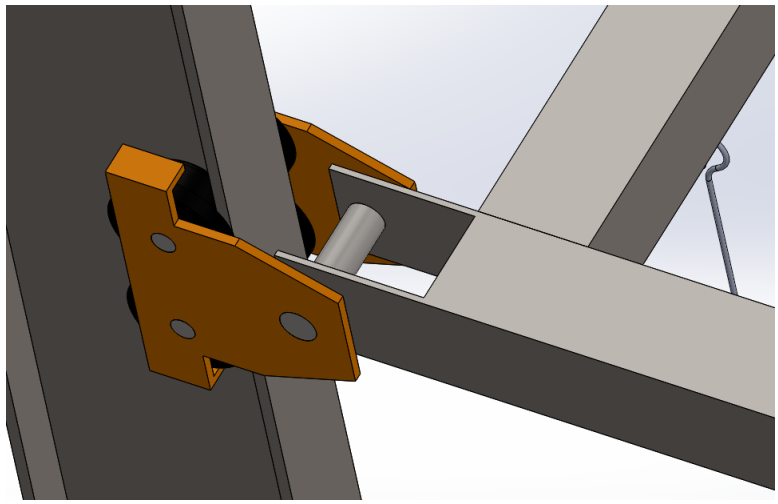


Figure 19: Carriage square tubing with the top and bottom surfaces removed.

By removing these two surfaces the trolleys have the ability to move without binding too the columns. The two sides will act as cantilever beams. This cantilever beam action will allow them to deflect elastically and therefore allow deflection in the column support. Analysis of the column will be taken for a 5 mm deflection at the midpoint height of the beam and a maximum load of 38.7klbs or 17.6 kg.

A 5-mm deflection at the midpoint of the column will result in a horizontal force of 313N using similar triangles. With the horizontal force and the maximum deflection we can analyze the beam deflection formula to find the needed length of the extrusion.

$$L = [\frac{3\delta EI}{P}]^{\frac{1}{3}} \tag{21}$$

With δ=0.005m, E=29 GPA, I = 7.9×10^{-7} and P=17554. Solving for length we find 0.1m. Since there are two walls to the tubing we must divide this length by two and find an extrusion depth of 0.05m or 2-inches to be necessary. The necessary extrusion can be found in Figure 20.

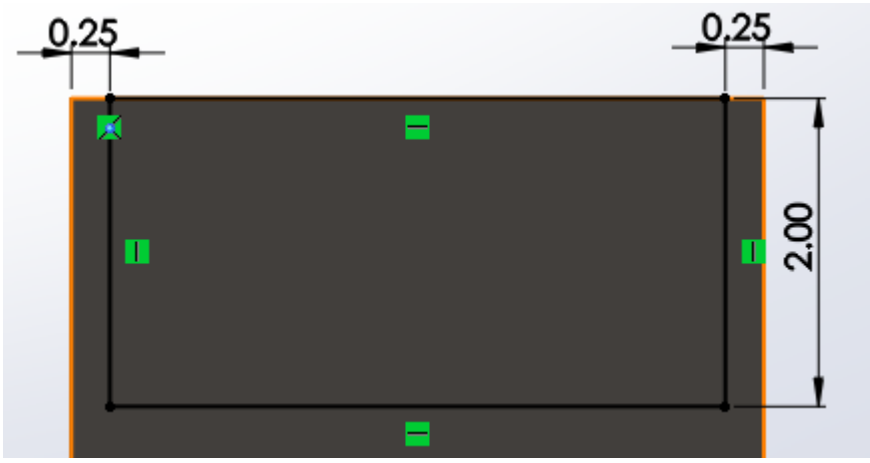


Figure 20: Support frame extrusion dimensions.

5.3.2. Wildlife Model Insert

The wildlife model insert consists of steel rims with deflated sand filled tires and a steel support shaft. Originally, the design consisted of water cut rubber plates, similar to the moose model design by VTI previously discussed. A local Winnipeg based provider, Argus Industries, was contacted to determine the cost of the rubber plates. A quote was provided at $9,920.12 far exceeding the project budget. Rubber tires supported with rugged steel rims offer a cost effective solution. The tires are deflated and filled with sand to reduce recoil after the collision. The tires are also modular, allowing the wildlife model to be scaled to different sizes and weights replicating different animals. The steel support shaft passes through the center of the steel rims and connects to the steel support frame via detachable bolted connection. Multiple connection points run along the frame of the cart to allow for various impact locations on the bumper. Four 40.7” x 11.3” wheels mounted on a 2.7 m long, 3” diameter solid shaft comprises the wildlife model insert at the 800 kg full weight capacity of the wildlife model cart assembly.

5.3.3. Roller Interface

The roller interface consists of a roller connection to the tower support structure and a flexible connection between the wildlife model cart assembly steel support frame. The roller connections used in this application are four, two-ton beam trolleys by Princess Auto shown Figure 21.

Figure 21: 2-ton beam trolley by Princess Auto [25].

A flexible connection is needed in order to prevent jamming between the wildlife model cart assembly and tower support structure. In an ideal circumstance, the wildlife model cart assembly and tower support structure would be perfectly aligned and allow for a smooth drop during a collision test. In reality, imperfections in alignment will contribute to a jam. To prevent this, a flexible steel plate connection is placed between the roller connection and steel support frame, allowing for a smooth drop during a collision test.

5.4. Bumper Support Structure

The bumper support structure is the structure that mounts and supports the bumper. It must comply with a number of requirements, including:

- It must hold the top of the bumper 3 feet above the ground.
- It must survive impact, including potential torque around the connection with the bumper.
- It must be compatible with the AeroPlus bumper.
- It must be mobile.
- It must me movable with a forklift.
- It must be able to be fixed to the ground for testing.

Most of these requirements are easily achieved through designing a model with SolidWorks, with no analysis done. A more thorough analysis must be conducted on the beam(s) that support the bumper, as that is the component that is most likely to fail due to impact.

5.4.1. Support Structure Concepts

In Section 2 and Appendix A we concluded that the best concept for a bumper support structure is a truss-like structure, based off of our vibration rig design. From this design, we came up with two separate concepts for a support structure: a stump design, and a quad-pod design. The stump design consists of one vertical metallic beam located right under the

connection between the bumper and the rest of the structure. The quad-pod consists of four identical metallic beams, positioned in a circle centred on the bumper connection, with each beam at a 90° or 180° from all other beams. These beams are then placed at a 45° vertical angle.

The advantage of using a stump design is that it would be much cheaper than the quad-pod design, for two reasons. The first reason is that there is only one beam used, instead of four beams. The second reason is that the single beam could be build weaker, as bending stresses in impact are much greater than axial stresses from impact, as Section 6.4.2 details. A disadvantage of using the stump design is that it provides no resistance to any non-vertical loading, and would easily fail if the dummy hit a bar on the bumper in an unanticipated manner.

The quad-pod is a much more versatile design. It resists both vertical and horizontal loading, for the collisions where the dummy does not hit the bumper head on. A disadvantage of this design is that it must be built to resist both bending and axial stress, and therefore will cost more.

5.4.2. General Impact Loading Theory

Both the stumps and the quad-pod must be able to support the bumper under an impact loading. Stress in a member due to impact loading can be determined by using the strain energy transferred to the member under impact loading. For members of a uniform cross-section experiencing axial loading, the stress due to impact loading is:

$$\sigma_{axial} = \sqrt{\frac{2U_m E}{V}} \tag{22}$$

Where σ_{axial} is the axial stress in the member, U_m is the potential energy due to impact, E is Young's modulus for the material, and V is the volume of the beam. The bending stress due to impact is given by the equation:

$$\sigma_{bending} = \sqrt{\frac{6U_m E}{L\left(\frac{I}{c^2}\right)}} \quad (23)$$

Where $\sigma_{bending}$ is the bending stress in the member, L is the length of the member, I is the moment of inertia of the member, and c is the distance between the neutral axis of the member and the surface with maximum bending stresses.

These equations make a few assumptions, however. The first assumption is that no energy is dissipated (i.e. through sound) during impact, and that the striking object (the dummy) will not bounce off of the structure and retain energy. In real tests, both of these assumptions are not satisfied; there will not be perfect energy transfer between the objects. However, these assumptions lead to a theoretical solution with more energy transferred, and therefore more stresses in the members than in reality. This makes the assumptions good, conservative assumptions that will add a safety factor to the structure [26].

From the equations of stress due to impact loading, we can determine design goals for the design of members to withstand impact loading: we want members with a large volume, and we want materials with a low Young's modulus, but a high yield strength.

5.4.3. Member Cross-Sectional Analysis

The cross-section of the stump and the quad-pod do not have to be the same. The quad-pod must have a cross-section that optimises both overall volume, and the ratio of I/c^2, while the stump just needs to have a high volume design. Therefore, the cross-section of the quad-pod must be carefully analyzed.

The two cross-sections we will consider are a solid square, and a solid circle. These cross sections were chosen as they are the densest rectangular and circular cross-sections, respectively, and therefore make efficient use of the space available for the stand. To compare the two, they will be given identical cross-sectional areas, and the shape that has the highest ratio of I/c^2 will be selected as the best design. For simplicities sake, we will assign each cross-

section an area of 1. This gives the square dimensions of height (h) = 1 and base (b) = 1, and gives the circle a radius of √(1/π). The moment of inertia of a square is given with the formula:

$$I = \frac{1}{12}bh^3 = \frac{1}{12}h^4 \tag{24}$$

Since c = h/2, we can determine that, for a square:

$$\frac{I}{c^2} = \frac{1}{12}\frac{h^4}{\frac{h^2}{4}} = \frac{1}{3}h^2 = \frac{1}{3} \tag{25}$$

For a circle,

$$I = \frac{1}{4}\pi r^4 \tag{26}$$

In the case of a circle, c = r = √(1/π). Therefore,

$$\frac{I}{c^2} = \frac{1}{4}\frac{\pi r^4}{r^2} = \frac{1}{4}\pi r^2 = \frac{1}{4}\frac{\pi}{\pi} = \frac{1}{4} \tag{27}$$

Since the ratio I/c^2 is higher for a solid square cross-section than it is for a solid circular cross-section, the ideal cross-section for the quad-pod is a solid square. For simplicities sake, the cross-section of the stump will also be a solid square, as its cross-section isn't a major factor in its ability to absorb energy.

5.4.4. Material Selection

There are two material properties that are important to a material's ability to absorb energy: its yield strength, and its modulus. Both formulas for stress in an impact loaded member have the relation:

$$\sigma \propto \sqrt{E}$$

Therefore, the criterion we will use for material selection is the maximization of the ratio of σ/√E. Appendices 3-12 were browsed in *Mechanical Elements in Machine Design* [27] to determine a short list of materials with high yield strength, and a low modulus of elasticity. These properties will be used to determine their ratio of σ/√E, and the material with the highest ratio is the ideal material to build the support structure out of. If steel is not the material chosen, other factors such as weldability will have to be considered. The materials considered, their relevant properties, and their overall ratios are listed in TABLE XVI.

TABLE XVI: MATERIALS CONSIDERED FOR THE CONSTRUCTION OF THE SUPPORT STRUCTURE [27].

Metal	Alloy	Yield Stress		Youngs Modulus		σ/√E
Steel	AISI 9255 Q&T400	1980	MPa	207	GPa	4351.911
Iron	ASTM A897M 1600/1300/01	1300	MPa	152	GPa	3334.43
Aluminum	7075-T6	503	MPa	71.1	GPa	1886.397
Magnesium	AZ-91 MgA19Zn1(A)	155	MPa	45	GPa	730.677
Titanium	Ti-3Al-13V-11Cr Aged	1210	MPa	110	GPa	3648.287
Brass	Free-cutting Muntz metal	414	MPa	15	GPa	3380.296

From TABLE XVI, we can observe that heat treated AISI 9255 steel is the superior choice by a wide margin, with aged titanium in second and Muntz metal in a distant third, at only about 75% the efficiency of steel at dispersing strain energy.

5.4.5. Quad-pod Design and Analysis

The quad-pod consists of four legs, equally spaced radially, and at the same angle with respect to the bumper and ground. While the analysis will be done assuming that the load applied from the dummy is purely a vertical load, and therefore horizontal support is unnecessary, the legs of the quad-pod must have some horizontal component because it is unlikely that this assumption will hold up in actual tests, and the horizontal component in the legs is necessary to resist any non-vertical forces. Therefore, we chose to have the legs take an angle of 45° with the ground. As discussed above, this analysis assumes that the strain energy is

split evenly between the bending of the beam and the axial stress on the beam. This assumption gives the formula for the max stress in one leg of the quad-pod:

$$\frac{\sigma}{\sqrt{E}} = \sqrt{\frac{2(U_m cos\theta)}{V}} + \sqrt{\frac{6(U_m sin\theta)\left(\frac{h}{2}\right)^2}{LI}} \tag{28}$$

The angle between the member and the ground is denoted by θ. Additionally, we added in a factor of safety on the energy transferred into the members.

The final variables that need to be determined are the energy transferred to the members during the collision, and the height of the members of the quad-pod. The energy transfer was determined in section 3 and is equal to 49kJ. The length of each member can be determined using Pythagorean Theorem for an isosceles triangle with non-hypotenuse side length of 2.5 feet, or .762m. This height required of the stand is assumed, and will be confirmed with a SolidWorks model. The length of a member at a 45° angle with the ground and this height is 1.08m. If the length assumed is too long, the stress absorbed by the member will decrease, however the safety factor applied will be satisfactory to eliminate a length change of a few centimetres.

With each member taking equal amounts of energy, and adding in a factor of safety of 2, each member must absorb 24.5kJ of energy. The only variable that can be adjusted to accommodate this is the height of the cross-section (and the base of the cross-section, as it is a square cross-section). Solving the stress for one member in bending and axial stress equation, above, for height gives:

$$h = \frac{\sqrt{E}}{\sigma}\left(\sqrt{\frac{2U_m cos\theta}{L}} + \sqrt{\frac{18U_m sin\theta}{L}}\right) \tag{29}$$

Applying this for a steel member gives the quad-pod a required cross-section of 16cm x 16cm, and a length of 1.08m per member.

5.4.6. Stump Design and Analysis

The stump design and analysis is much simpler than the quad-pod analysis. As it is perfectly vertical, it will not experience bending stress, and therefore only axial stress needs to be considered. The stress due to a pure axial stress under impact loading is given as:

$$\frac{\sigma}{\sqrt{E}} = \sqrt{\frac{2U_m}{V}} \tag{30}$$

Like the quad-pod, we will assume that the stump is 2.5 feet, or 0.762 metres, and resists an impact of 49kJ. Likewise, a safety factor of 2 will be assumed in the analysis of the stump. Since all the variables are known, we can rearrange the stress equation for the stump to solve for the required cross-sectional height (and base). The height of the cross-section is equal to:

$$h = \frac{\sqrt{E}}{\sigma}\left(\sqrt{\frac{2U_m}{L}}\right) \tag{31}$$

With a height of 0.762m, an applied energy of 49kJ, a safety factor of two, and using steel, we can determine that we require a stump with dimensions: 76cm x 12cm x 12cm.

5.4.7. Summary of Bumper Support Analysis

To determine the ideal design for a bumper support, we designed a bumper that met a few requirements, such as portability and durability. While most of the dimensioning and design of the bumper stand was done in SolidWorks, the cross-sections of the supports and the material that they are made out of had to be more carefully analyzed. We analyzed the bumper support using stain energy methods for impact loading, and from this were able to determine that the ideal cross-section for the bumper supports is a solid square cross-section. Using the same methods, we determined that the ideal material to construct the supports out of is heat treated AISI 9255 steel.

There were two support designs considered. The first is a central quad-pod, containing four members at a 45° angle to the ground and at 90° angles to each other. These quad-pod

members need to be 1.08m long, and have square cross-section with sides 16cm long. The second support designed is a vertical stump. It is subjected to purely axial loading, and must be 76cm long, with a square cross section with sides 12cm long.

Choosing the right bumper support is important, as they are costly to produce and will fail easily if under-designed. As the stump design is much more likely to fail when experiencing non-centred loads, the quad-pod design is the superior design, even if it is much more expensive than the stump design.

5.5. Power System

While freefall will provide some acceleration to the dummy, a large vertical freefall distance is required to accumulate enough kinetic energy in the dummy through freefall alone. To lower the freefall distance required, we looked at additional methods of adding potential energy in the form of spring energy to the dummy at the start of freefall. The power system consists of the components required to mount the spring to the rest of the test apparatus, the components required to transmit the energy generated by the power system to the dummy, such as ropes or cables, and the spring itself.

5.5.1. Spring Analysis

The spring design chosen to power the freefall of the dummy dictates the requirements of the method used to transmit power to the dummy and the method used to attach the spring(s) to the rest of the test apparatus. Therefore, the spring must be designed before the rest of the power system.

There are four main categories of springs to choose a spring from, based on the direction the spring transmits force in. These categories include:

- Push springs
- Pull springs
- Radial springs
- Torsional springs [27]

Push springs and pull springs have energy stored in them through axially applied force, and release the energy axially. The differences between push and pull springs is the direction that the force exerted by the spring is relative to the spring. Push springs push objects away from them; while pull springs pull objects towards them. Radial springs are like push and pull springs; however they exert the energy stored in them radially in all directions. The final spring type is a torsional spring. This spring has energy stored in it and releases energy through the application of torque. As these springs supply a torque to an object, the force supplied generally changes directions throughout the deflection of the spring. As we are designing an aid to freefall, we want a spring that supplies an axial force in one direction only; torsional springs are not useful to us. While radial springs could theoretically work for our application, they are designed to release energy stored in all directions, while we are only concerned with force in one direction. Therefore, we will only look further at push and pull springs.

Push and pull springs operate in similar, though opposite, manners. They will either push or pull two objects apart or together. To accelerate the dummy towards the ground, we can either push it away from the top of the tower, or pull it towards the ground. Both methods have advantages and disadvantages. Push springs would be anchored to the top of the tower, and push the dummy towards the ground. An advantage of this is that the spring can push directly against the dummy; no power transmission is required. However, as springs push or pull in both axial directions equally, anchoring the springs to the top of the tower would put unnecessary force on the top of the tower. This would mean that the tower would have to be overbuilt to accommodate for the spring force at the top.

Pull springs would be attached to the ground and pull the dummy towards them. An advantage of this is that the opposing force from the spring would be exerted on the ground, meaning that the support structure does not need to be overbuilt. A disadvantage of using pull springs is the relatively short deformation that they allow. While springs can theoretically be built to support any deformation, building a spring to allow for a deflection of 10 feet would be unreasonably large for this application. Therefore, a method needs to be determined to transmit the power from the end of the spring to the dummy. A notable exception to this rule is

the constant force spring, which can be built within reasonable limits to support a 10-foot deformation.

As designing a power transmission system for pull springs is simpler than designing the support structure to allow push springs at the top, we decided to analyse pull springs for this design. Conceptually, this decision makes more sense too. As we are looking to bring the dummy towards the bumper, it makes sense to pull the dummy and ground together, rather than pushing the dummy away from the sky.

There are four primary designs of pull springs. They are:

- Helical springs
- Flat springs
- Drawbar springs
- Constant force springs

Pulling (expansion) helical springs consist of several coils of wire curled into a cylindrical shape. The coils are very close to each other, or touch, under no load. When the spring is loaded, the coils are pulled apart, storing energy in the motion of pulling the coils apart [27]. Helical springs are the most common spring design, and are used in all applications. Figure 22 shows an example of an expansion helical spring.

Figure 22: An expansion helical spring under no load conditions [28].

A flat spring, also known as a cantilever spring or a leaf spring, consist of at least one flat strip of spring material loaded as a cantilever or as a simply supported beam. They are loaded by applying a load either on the end of the beam for a cantilever spring or in the middle of a simply supported beam, storing strain energy in them. They release energy attempting to get back to a straight beam. An advantage of these springs is that they allow a large force to be applied in a small space. They are simple to analyse, as the analysis is like that of a cantilever or a simply supported beam [27]. Typical applications of leaf springs are in automotive suspensions. Figure 23 shows an example of a leaf spring.

Figure 23: A leaf spring in an automotive application. The leaf spring is bent around the bottom of the axle [29].

A drawbar spring is a modification on a helical spring. It takes a pushing (compression) helical spring, and inserts two looped wires inside the device. The ends of the looped wires are then pulled and the helical spring is forced to compress. This design allows for a compression helical spring to be used instead of its cousin, the extension (pulling) helical spring. Figure 24 shows a drawbar spring. This design will not be considered in the spring analysis as the analysis is relatively similar to that of a helical spring. However the two wires wrapped inside provide extra complexity and cost, which can be avoided by using a helical expansion spring [27].

Figure 24: Drawbar spring under no load conditions [30].

The final primary design of a pulling spring is the constant force spring. A constant force spring is formed out of a coiled strip of spring metal. The metal is manufactured to naturally curl. As the tongue of the spring is pulled, it forces the spring to straighten, storing energy in the deformation of the beam from curled to straight. Once an initial extension of the spring is surpassed, the force applied by the constant force spring is almost constant over the length of the pull. The force is dependent on many factors, such as Young's modulus of the spring stock, as well as width and thickness of the stock and the radius of curvature of the coil. A large advantage of these springs is that their displacement from no load has next to no size limits like

the helical spring or flat spring. However there are other issues such as mounting the coil and twisting or buckling when it is extended over a long distance [27]. Figure 25 shows a constant force spring.

Figure 25: A constant force spring [31].

The remaining three springs must be analyzed thoroughly, including considerations of the power transmission method and the mounting method, to choose the ideal spring for aiding the freefall of the dummy. As the helical spring is the most common spring and the most versatile spring, it will be analyzed first. The remaining designs will be compared to the helical spring.

5.5.1.1. Linear Spring Analysis

The first step to analyzing linear springs is to determine their spring constant *k* and to determine the maximum force *F* that they supply. This step is the same for all linear springs and can be calculated once, then applied to both the helical spring and the flat spring. As the constant force spring is not a linear spring, this initial step is not applicable to its analysis.

From our initial drop test calculations and design space determinations, we can determine initial parameters to base the design of our springs off. The first parameter we have is that the total energy imparted by this collision must be 49kJ. This includes the energy provided by freefall and the energy provided by the springs. We also defined the mass of the dummy as 800kg and the total length of freefall as 10 feet. These two parameters will give the total gravitational potential energy of the collision. Finally, we defined the height of the bumper as 3 feet. To avoid any risk of impact between the springs and the dummy and/or drop cart, the maximum height of the flat or helical spring will be designed to be less than 3 feet. However, some leeway from this is allowable.

The total energy required from the springs can be calculated from the parameters above. Assuming that there are no losses in energy in the dummy over the 10-foot fall, the total energy of the system can be written as:

$$\Delta KE = \Delta GPE + \Delta SPE \tag{32}$$

Where *KE* is kinetic energy, *GPE* is gravitational potential energy, and *SPE* is spring potential energy, all in similar units of energy. Rearranging, and expanding out *ΔGPE* per its definition, we get the required spring energy as:

$$\Delta KE - mgh = \Delta SPE \tag{33}$$

Where *m* is the mass of the dummy in kg, *g* is the acceleration due to gravity on earth (*g* =9.81m/s^2), and *h* is the height of the drop in m. The energy is given with in units of Joules (J). With the parameters defined earlier in this section, we get a required spring potential energy of 25.1kJ which is equivalent to 18,500ft-lb.

The definition of spring potential energy is:

$$\Delta SPE = \frac{1}{2}k\delta^2 \tag{34}$$

Where *k* is the spring constant in lb/ft, and *δ* is the deflection from no load of the spring in feet, giving energy in ft-lb. We know that the maximum height of our spring at any point is 3ft or 36in. A horizontal flat spring, such as the one that will be considered, requires as much room below it as above it for its rebound stroke, and therefore the maximum deflection it can offer is half of the total height available, or 18in. Using 3in on either side as leeway, we will design a flat spring with a deflection of 15in.

Helical springs are much tougher to determine an initial deflection of, as the deflection must be added to the free length of the spring, which is a function of the number of coils in the spring and the thickness of the spring wire. A general rule of thumb to start the analysis is to assume that the deflection of the spring is roughly the same as the free length of the spring. Allowing for 6in of deviation between the free length and the deflection of the spring, we assume that a deflection of 15in will work for a helical spring. This assumption must be confirmed at the end of the helical spring analysis.

As the deflection is assumed to be the same for the helical and flat springs, their spring constant and maximum forces will be the same, too. Rearranging the spring potential energy equation, we determine the spring constant using:

$$k = \frac{2 * \Delta SPE}{\delta^2} = 23680 lb/ft$$

Hooke's law for linear springs can be applied to determine the maximum force in the spring. Hooke's law says:

$$F = k\delta$$

Where *F* is the maximum force applied by the spring, in lb. As force and displacement are directly proportional, the maximum force will occur at the point of maximum displacement. With the determined maximum displacement of 15in, we expect a maximum change in force applied by the spring of 29600lb. This force is carried forward into the analysis on the individual spring designs. This is much too large a force for standard helical springs, and needs to be divided amongst a set of springs to support this load. In addition, as the springs are going to be attached to supply force away from the centre of mass of the dummy, we want an even number of springs to remove any torque that may be applied by an odd number of springs spread across the sides of the wildlife dummy/drop cart.

To split the forces among multiple springs, we must use springs in parallel. This allows the spring constant to stay constant for every spring, and divides the force evenly among all springs with equal deflection. Using the analysis in section 5.5.1.1.1, maximum realistic wire diameter, and light service, we can calculate a maximum load realistic on a spring of 7500lb, which requires 4 springs to apply 29600lbs of force. However, this is the case under ideal conditions, and it doesn't consider initial spring tension. Therefore, we will consider a set of 8 helical springs in parallel, each with a required operational load of 3700lbs.

For flat springs, the only concern in load applied is applying a torque to the dummy. This is mitigated by having an even number of springs. In addition, since flat springs are quite large, it is tough to fit more than two springs into the footprint of the apparatus. Therefore, we will consider a system of two flat springs each applying 14800lbs.

5.5.1.1.1. Helical Spring Design and Analysis

Helical expansion springs exert a pulling force when the coils of the spring are pulled apart. The force applied by the spring is an attempt to restore the coils of the spring to their initial position. Helical springs are generally designed and manufactured so that the coils are touching in a manner that requires an initial force to be applied to the spring to separate the coils and get to the lower operational limit of the spring. Once the lower operational limit of the spring is reached, the spring operates like a normal spring, as shown in Figure 26.

Figure 26: Load vs deflection chart for a helical expansion spring. Pi is the initial tension in the spring [32] .

The initial tension in a spring must be determined experimentally. It is generally between 10% and 25% of the maximum design force of the spring, and this force cannot be used as part of the operation of the spring [27]. The initial force tends to decrease as the spring index decreases [32]. The initial force must be added as a constant to the maximum operating force when determining the stresses in the spring.

The first main consideration in the design of a helical spring is the stresses present in the spring when it is fully loaded. As loading the spring twists the spring wire, the dominating stress is shear stress. The shear stress in a helical spring is determined using the following:

$$\tau = \frac{8KF_{max}C}{\pi D_w^2} \tag{35}$$

Where τ is the maximum stress present in the spring, F_{max} is the maximum force present in the spring and is a sum of the required operating force and the initial force. C is the spring index which is a function of the wire diameter and the mean spring diameter, K is the Wahl factor and a function of the spring index and D_w is the spring wire diameter. The shear stresses in the spring must be below the allowable shear of the spring material, which depends on the material chosen and the type of loading that the spring experiences.

The type of loading that the spring experiences can be broken into three categories: heavy, average, and light service. Light service loading is defined as “Static loads or up to 10 000 cycles of loading with a low rate if loading (nonimpact)” [27]. This loading description perfectly

describes our application of the springs. While there will be impact during the test, the springs will be unloaded at that point, and our test apparatus doesn't have to survive 10 000 cycles. However, we will analyze the springs using an average service loading to build in a safety factor to the spring design.

There are five low cost materials recommended by *Machine Elements in Mechanical Design* [27] to use as spring wire. They include:

- Hand-drawn high carbon steel A227
- Music wire A228
- Oil-tempered high carbon steel A229
- Chromium-vanadium steel A231
- Chromium-silicon steel A401

The two strongest materials are A401 and A228, with A231 having an average strength and A227 and A229 being the weakest of the five. As we are looking for high strength materials for the springs, we will consider A228 and A401 first. The main difference between A228 and A401 for our application is the possible wire sizes: A228 typically has a max wire diameter of 1/8in [27], while A401 has a much larger maximum stand wire size of up to 1in [33]. All other materials considered have a maximum standard wire size of 5/8in. Therefore, A401 will be the material considered in the first iteration of the helical spring design. As it is the most expensive of the five materials [27], if any materials look feasible after a design with A401, then they will be used to lower the cost of the spring.

The allowable shear stress in a material decreases as the wire diameter increases, but the wire diameter is also a function of shear stresses amongst other factors, thus the allowable shear stress and wire diameter must be iteratively calculated using the following steps:

1. Assume a τ_{max}
2. Calculate the required D_w
3. Find the corresponding τ_{max} in Figure 27 below
4. Repeat steps 2 and 3 until $\tau_{max,new}$ equals $\tau_{max,old}$

Figure 27 gives the allowable shear stress as a function of wire diameter for all three service levels described.

From Figure 27, we can see that for wires larger than 1/2in, the allowable shear stress does not change dramatically. We will therefore assume that the allowable shear stress for any wire diameter larger than 1/2in is constant. For an average service wire, we can observe that τ_{max}=125ksi at D_w > 1/2in. Therefore, this will be the first shear in our iteration of τ_{max}.

The final consideration we must make before determining the stresses in the spring is the spring index and Wahl factor of the spring. The spring index of the spring is given in equation (36) below.

Figure 27: Design shear stresses for A401 wire as a function of wire diameter [27].

$$C = \frac{D_m}{D_w} \tag{36}$$

Where D_m is the mean diameter of the spring. The Wahl factor of a spring is defined as:

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} \tag{37}$$

As C increases, K decreases. We are looking to minimize the product of these two variables to lower the overall shear in the spring. As such we can select C to best suit our needs and then we can determine $C*K$ as a function of C, as shown in Figure 28.

Figure 28 shows the ideal value of C would be just below 2. However, any value of C below 5 makes manufacturing the spring difficult and increases the risk of cracks forming in the spring during manufacturing [27]. In addition, having the spring index below 6 increases the manufacturing cost of the spring. Therefore, a marginal spring index of 5 is required, but the spring should be optimized to have C > 6 for cheap manufacturing.

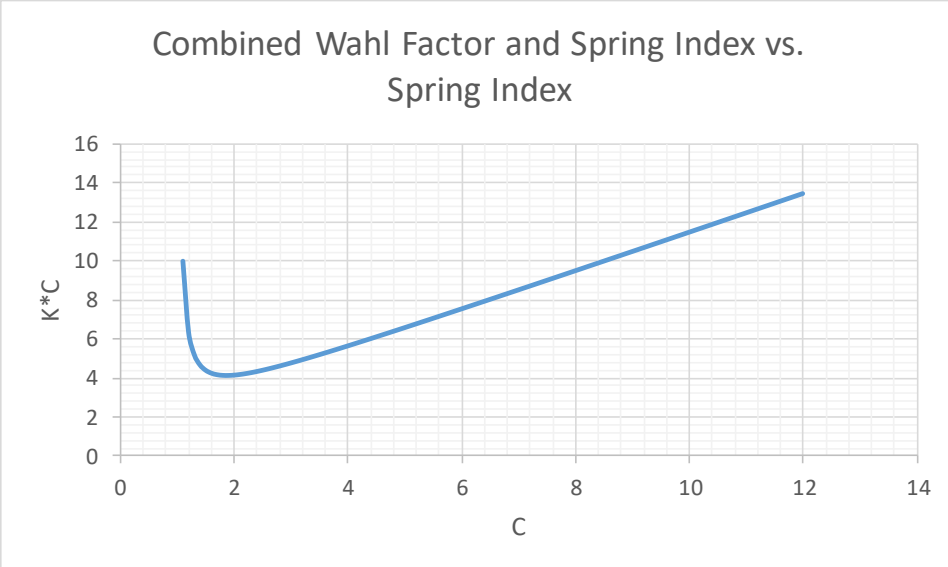


Figure 28: Comparison of C and K*C in a spring.

The maximum force is the sum of the initial tension in the spring and the force required to generate the required energy. As we are using spring indices between 5 and 6, we will assume that the initial tension required is 25% of the operational force of 3700lb. Therefore, we must design for a maximum force of 4625lb.

We know have enough information to calculate an initial wire diameter from the allowable shear stress. The diameter of the wire required can be found using the equation:

$$D_w = \sqrt{\frac{8KF_{max}C}{\pi\tau_{max}}} = 0.786in \tag{38}$$

The next largest standard spring wire size is 13/16in [33]. The stresses caused by using this wire diameter will be less than the design stress of 125ksi. However, the assumed design stress is equal to the design stress for a wire of 13/16in diameter, satisfying step 4 in our iterative wire diameter calculation. Therefore a wire diameter of 13/16in is the final wire diameter of our design.

From this determination, we can see that A401 is the only common spring material usable for our design. To use a weaker material, we would need to increase the wire diameter beyond the 13/16in required for A401. This isn't practical, as any other material has a maximum

wire diameter of only 5/8in and would therefore fail due to high shear stresses of our application. While it is theoretically possible to use a cheaper material with a larger wire diameter, the cost of manufacturing a spring with a non-standard wire diameter would likely exceed the cost saved by using a cheaper spring material.

Using the same calculation, we can optimise *C* to reduce manufacturing costs. Using a design stress of 125ksi and a wire diameter of 13/16in, we can calculate the required *C*K* of the design.

$$C * K = \frac{\pi \tau_{max} D_w^2}{8F_{max}} = 7.007 \tag{39}$$

The corresponding *C* that gives *C*K* < 7.007 is *C* = 5.47, giving a *C*K* = 7.003. This is the maximum spring index that we can use, however it may be decreased when determining the number of coils in the spring.

The final consideration in the spring is number of coils in the spring. This is determined using the following equation:

$$N_a = \frac{\delta G D_w}{8FC^3} \tag{40}$$

Where *G* is the shear modulus of A401, equal to 11.2×10^6psi [27], N_a is the number of coils present in the spring and δ is the deformation of the spring in inches. In addition, we use *F* instead of F_{max} in this calculation since this calculation only looks at the operational displacement of the spring. All factors aside from *C* and N_a are fixed, thus N_a can only be adjusted by changing the *C* of the spring. As we wish for the loops on the end of the spring to be parallel, the number of coils must be either an integer number or half an integer number. With the current values, we get N_a = 28.17. Since *C* = 5.47 is the maximum allowable spring index, we must increase N_a to the closest half integer to avoid increasing *C*. This gives us 28.5 coils in the spring, and we can determine the final spring index of the spring from the following:

$$C = \sqrt[3]{\frac{\delta G D_w}{8FN_a}} = 5.45 \tag{41}$$

From the above information, we can determine the precise geometry of the body of our spring, which is displayed in TABLE XVII.

TABLE XVII: GEOMETRY OF THE CHOSEN HELICAL EXPANSION SPRING

Wire Diameter	0.8125	in
Mean Diameter	4.428	in
Outer Diameter	5.24	in
Inner Diameter	3.616	in
Free Length	23.15	in
Operating Length	38.12	in

The mean diameter is calculated from the definition of the spring index and the wire diameter. The outer diameter (*OD*) and inner diameter (*ID*) are given using the following equations:

$$OD = D_m + D_w; \tag{42}$$

$$ID = D_m - D_w \tag{43}$$

The coils are all touching at no load so the free length is given as:

$$L_{free} = W_d * N_a \tag{44}$$

The operating length is the sum of the free length and the deflection of the spring:

$$L_{operating} = L_{free} + \delta \tag{45}$$

The final consideration that must be made is the geometry of the ends of the loop. Various end loop geometries can be used for this application. We will assume that the chosen end loop geometry is a full loop end, as shown in Figure 29. This loop is likely to fail in two places. Referring to Figure 29, the points of failure are at point *A* due to bending stress and point *B* due to torsion.

Figure 29: Geometry and nomenclature for the stresses in the loops at the end of an expansion spring [27].

Machine Elements in Mechanical Design [27] recommends that R_1 and R_2 are at least double the diameter of the wire to avoid high stresses. If the radii are kept constant with the mean radius of the spring, this makes R_1 and R_2 equal to 2.21in.

Adding on $4*R_1$ to the free length of the base helical spring gives an overall free length of the spring of 32in. This length, along with the outer diameter of the spring and the wire diameter, can be used to get an approximate price for the springs, using the instant quote feature at *www.thespringstore.com* [33]. The overall cost of 8 springs with the dimensions in TABLE XVII is $5073 US.

5.5.1.1.2. Flat Spring Design and Analysis

The second possibility for springs to use in our design is a set of two flat (also known as cantilever) springs that each support 14800lb. There are two principal equations governing the design of flat springs in a cantilever support structure (as opposed to a simply supported flat spring that is much longer overall):

$$\sigma = \frac{6FL}{bt^2} \tag{46}$$

$$F = \frac{\delta E b t^3}{4L^3} \tag{47}$$

Where σ is the normal allowable bending stress in the spring, F is the force exerted by the spring, L is the length of the spring (perpendicular to the deflection), b is the width of the cantilever (perpendicular to both the length and the deflection), t is the cantilever thickness (parallel to the deflection), δ is the deflection of the spring, and E is Young's modulus of the material chosen. These equations assume that the spring has a negligible weight.

From these equations, it can be seen that materials that have low modulus of elasticity and high yield strengths are required for constructing the optimal spring. More specifically, the optimal material will have the highest ratio σ/E. Materials considered are provided from Appendix 3-12 of *Machine Elements in Mechanical Design* [27]. From screening that specified section, we can determine the best material for each type of metal provided. The types of metal are steel, iron, aluminum, magnesium, titanium, and brass. TABLE XVIII provides the yield strength, modulus, and ratio of σ/E for the superior alloy of each metal.

TABLE XVIII: METAL'S CONSIDERED FOR THE LEAF SPRING AND RELATIVE PROPERTIES [27].

Metal	Alloy	Yield Stress		Youngs Modulus		σ/E
Steel	AISI 9255 Q&T400	310	ksi	30000	ksi	0.010333
Iron	ASTM A897 230/185/01	185	ksi	23000	ksi	0.008043
Aluminum	7075-T6	73	ksi	10400	ksi	0.007019
Magnesium	AZ-91 MgA19Zn1(A)	22.5	ksi	6500	ksi	0.003462
Titanium	Ti-3Al-13V-11Cr Aged	175	ksi	16000	ksi	0.010938
Brass	Free-cutting Muntz metal	60	ksi	103000	ksi	0.000583

From this analysis, we can observe that titanium is the best metal to make the flat spring out of, with a ratio of σ/E = 0.0109. However, steel is a very close second, with a σ/E = 0.0103. Since steel is much cheaper than titanium, and cost is a concern in this project, the benefits of using a titanium spring are not worth the extra cost incurred by the titanium material. Therefore, we will design the flat spring out of SAE AISI 9255 steel, quenched and tempered at 400°F.

The design of the spring can be done by solving the stress governing equation for force, and the force governing equation for thickness:

$$\frac{\sigma b t^2}{6L} = F \tag{48}$$

$$t^3 = \frac{4L^3 F}{\delta E b} \tag{49}$$

Combining these equations, we can solve for the thickness of the flat spring as a function of its length L:

$$t = \frac{4}{6} * \frac{L^2 \sigma_{design}}{\delta E} \tag{50}$$

σ_{design} is the design allowable stress. This is calculated by multiplying the allowable stress σ by a safety factor of 4. This accounts for both a small general safety factor and the stress

concentration around the joint between the spring and the power transmission system. The stress concentration around the joint could be as high as 3, depending on the joint chosen. This gives σ_{design} = 77.5ksi. The above equation can now be used to calculate *t* as a function of the spring length. The stress governing equation can be rearranged again to determine the required width of the spring as a function of the length and thickness:

$$b = \frac{6FL}{\sigma_{design} t^2} \tag{51}$$

For any given spring length, we can determine a valid spring by determining a theoretically valid spring width and thickness. To calculate a list of various valid widths and thicknesses we iterated these equations in Microsoft Excel, increasing the length by 0.25in after each iteration.

The assumptions made in these calculations only hold up under certain geometry conditions. For these equations, we require the following three assumptions:

$$\frac{\delta}{L} < 0.3; \qquad \frac{b}{t} < 10; \qquad \frac{L}{b} > 5$$

Of these three assumptions, the limiting assumption in this case is $b/t < 10$. The shortest flat spring that meets this requirement is 150in (12.5 feet) long, with a width of 25.75in and a thickness of 2.58in.

This is a massive beam, and this size presents some issues that must be addressed. The first is the effects that the weight of the beam has on the stresses present in the beam. The weight of the beam will provide a distributed load across the beam with a force per unit length of:

$$\omega = \rho b t \tag{52}$$

Where ω is the weight per unit length in lb/in and ρ is the density of the beam in lb/in^3 (which is a constant 0.283lb/in^3). This gives a weight per unit length of 18.8lb/in. This can be applied to the beam, with a cantilever support as shown in Figure 30.

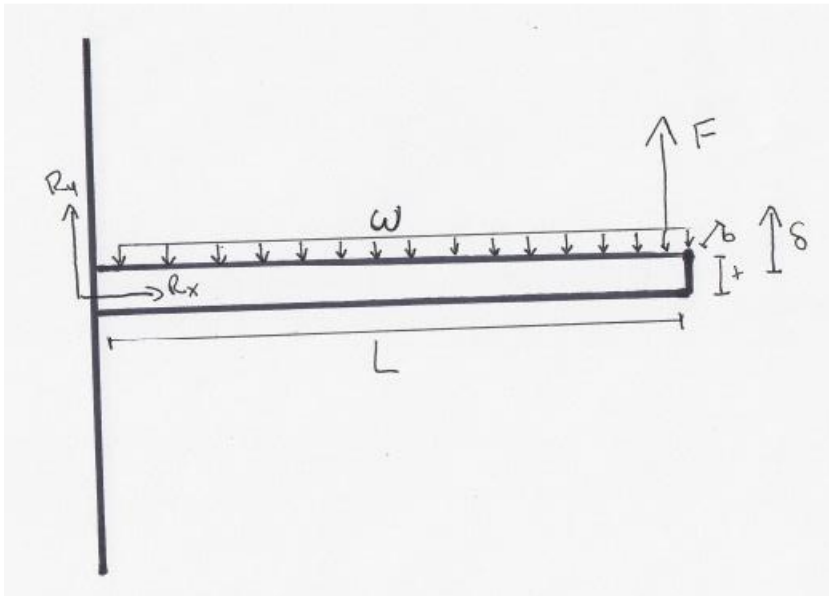


Figure 30: Free body diagram of our flat spring in bending [34].

Assuming that *x* is 0 at the furthest point from the cantilever support, the bending moment can be written as:

$$M = Fx - \omega x^2 \tag{53}$$

Where ωx^2 is the effect of the bending moment. Since the stresses in the beam are proportional to the bending moment, this seems like a benefit of using a flat spring. It can be said that the stresses are lowered because of the beam's weight. However, this is not the case. Cantilever beams operate by storing strain energy when they are given a deformation. The strain energy for a cantilever in bending is determined using the equation:

$$U = \int_0^L \frac{M^2}{2EI} dx \tag{54}$$

Where *U* is the strain energy in the beam, which is equivalent to the energy added to the system using a flat spring. The strain energy is proportional to moment squared. As the moment and bending stress in a beam decreases, the strain energy in the beam also decreases. To counteract this, either the force applied to the beam must increase, putting excess strain on the winch used to lift the dummy that the helical springs don't supply, or to increase the length, and therefore weight and cost of the beam.

The second concern is the largest concern with using a solid steel cantilever beam, and that is the cost of using such a massive beam. One quote, from *MetalsDepot* [35], put the cost of the equivalent amount of steel as $9092 US. This cost assumes that 4 steel bars 150in x 12in x 1in are equivalent to the 150in x 26in x 2.5in springs we require. This cost also does not account for the cost of the heat treatment required on the steel beam, or the cost occurred by lengthening the beam to increase its moment.

As the steel flat springs would cost at least 80% more than equivalent helical springs to implement, plus flat springs would be much larger than equivalent helical springs, and therefore much more difficult to fit inside the test apparatus, we will rule out flat springs as a possibility for our spring selection in this design.

5.5.1.1.3. Constant Force Spring Design and Analysis

The final spring that we will look at in detail is a constant force spring (CF spring). CF springs, unlike flat and helical springs, are non-linear springs and therefore do not have a spring constant *k*. A load vs. deflection graph for a constant force spring is shown in Figure 31:

Figure 31: Load vs. Deflection curves for a CF spring and for a linear helical expansion spring [36].

As Figure 31 shows, after an initial extension of the spring is surpassed (generally defined as 1.5*barrel diameter [37]), the force of the spring stays constant. Therefore, we cannot use spring potential energy to determine the required force, but we must calculate the work done by the springs to determine the force required from the springs. The work done by a CF spring is:

$$W = F * \delta * cos\theta \tag{55}$$

Where *W* is the work done by the spring, *F* is the force required to do that work over the known deflection δ, and ϑ is the angle between the line of action of the force and the direction that the spring moves, which is 0 in this case. The work done by the spring is equal to the potential energy that the CF spring must add to the dummy. This means that the work done by the CF spring needs to be equal to 25.1kJ.

Unlike linear springs, CF springs are not limited to relatively short deflections. They may theoretically be used with an infinite displacement, meaning that the spring can directly attach to the dummy, removing a need for a power transfer system if oriented vertically. However, when CF springs are given large deflections, issues begin to arise. According to Spring-I-Pedia, "Unless properly designed, a long, free extended portion of a spring may have a tendency to buckle or curl, particularly at the start of a return stroke. While there is no theoretical limit to such a spring's extension, it is advisable to confine any long extended portion in a way that will prevent curling [37]." To prevent curling, we would have to provide an external guide to the spring. If the spring is oriented vertically, such as in Figure 32, providing an external guide for the spring is very difficult. This would require the dummy to move perfectly smoothly along tracks and not get caught on any guides so that the CF spring works properly.

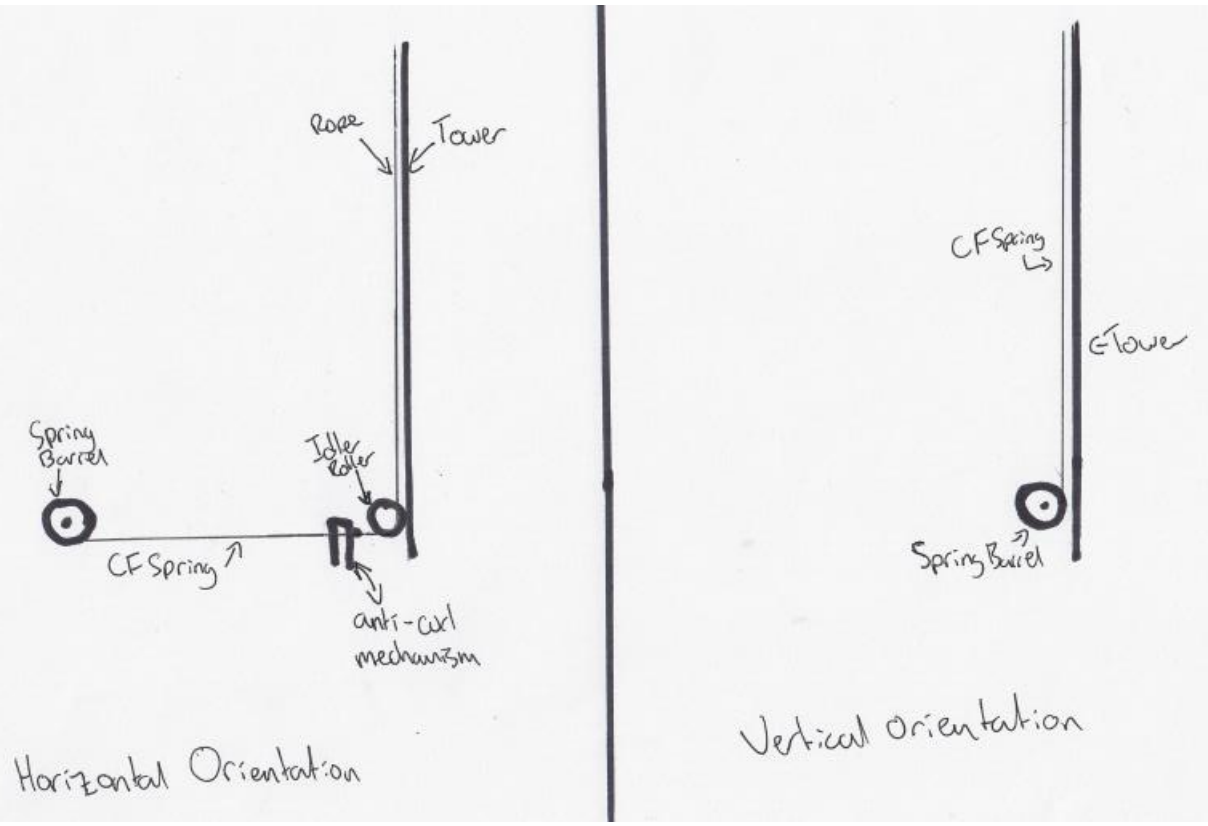


Figure 32: Orientation options for a CF spring [38].

The other orientation option in Figure 32 is a horizontal orientation. This involves the CF spring being extended horizontally along the ground, and being attached to a power transmission system to deliver the power to the dummy. An advantage of this orientation is

that the spring's potential deflection isn't limited by the height of freefall. This orientation also allows for designing the spring to mitigate the potential for curl. However, it removes the primary advantage of using CF springs in that a power transmission system is required to transmit power from the spring to the dummy.

The next consideration in designing a CF spring is the mounting of the spring to the ground. There are two common mounting methods: spool mounting, as shown in Figure 33, and cavity mounting, such as in Figure 34.

Figure 33: Spool mounted constant force spring [39].

Figure 34: Cavity mounted constant force spring [39].

Both of these methods have major drawbacks, however. The first option with spool mounting requires shafts and bearings to be designed to accommodate the spools for smooth operation. This vastly increases the complexity of a CF spring as compared to mounting a helical or flat spring. The use of a cavity spring presents two large challenges. The first is that there is spring energy lost to friction between the cavity and the spring. This can be compensated for by adding more springs. The second issue with cavity mounted springs is the hysteresis experienced by the spring from the friction, causing the spring force to be absent for the first part of freefall. This is a major issue as it equates to work lost by the spring during freefall, resulting in a lower energy collision.

Due to the issues discussed with constant force springs, including the issues about their orientation and the issues involving mounting the springs, we decided not to do a full analysis on the springs as it was clear that they were not the simplest solution to powering the drop test.

5.5.1.2. Spring Selection

In conclusion, we analyzed three types of pulling springs: helical springs, flat springs, and constant force springs. After a conceptual overview of constant force springs, we determined that they would not be the simplest or cheapest solution, as every spring used would require

mounting on a shaft with bearings, whereas flat springs or helical springs have much simpler mounting options.

We fully analyzed using either two flat springs, or eight helical springs, to add the required energy to the dummy. If we used flat springs to power the simulation, we would require two flat springs made of heat treated AISI 9255 steel, 150in long by 26in wide by 2.5in thick to support the required load. This option costs roughly $9100 US. If we used a helical spring to power the simulation, we would require eight springs made of ASTM A401 steel, with a spring wire diameter of 13/16in, outer diameter of 5-1/4in and an overall free length of 32in. This option costs roughly $5100. Therefore, since the helical springs are more compact and cheaper, we will use the helical spring described earlier in this section.

5.5.2. Rope Selection

The next challenge is transmitting the power generated by the spring to the drop cart. The power supplied from the spring is output at the upper loop, and the power must be supplied to the drop cart. The power transmission system for the springs must be small enough to fit through the loop at the end of the spring. This gives a maximum diameter of 3.5 inches of the power transmission system, allowing for clearance between the wire and the transmission system. If this maximum width is impossible to satisfy, it is possible to alter the spring end shape. However that would be extra customization on the spring, which would increase the spring cost.

The most limiting power transmission requirement is that it must not restrict the motion of the dummy during freefall. This removes the possibility of using any rigid connection between the spring and the dummy. To satisfy this requirement, as well as the size requirement posed by the spring end size, we chose to use a rope to transmit power from the spring to the dummy. The largest advantage that a rope has is that it can be designed to have enough tensile strength to support the loads presented by the spring, but when the rope loses tension, it will spool and the drop cart will fall unhindered.

The rope orientation is depicted in Figure 35 below. It features a sling of rope to fit around the spring hook and at the drop cart attachment.

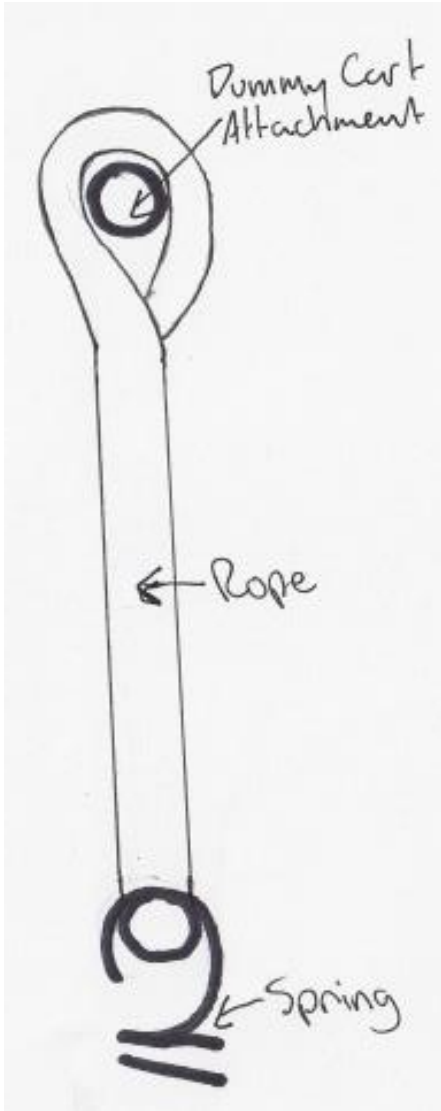


Figure 35: Power transmission setup using a rope to transmit power from the springs to the rod attached to the dummy [40].

There are two main classes of ropes to choose from: metal (or wire) ropes, and non-metallic ropes. Wire ropes consist of many small metallic wires bunched into rope strands, with the rope strands twisted around each other to form a cable of wires. Advantages of using wire ropes are that they are much tougher; you can get more force out of a wire rope than a non-metallic rope with a similar diameter. Another advantage of using a wire rope is that there is little to no elongation in the rope, and any elongation can be calculated using a variation on Hooke's law:

$$\varepsilon = \frac{WL}{EA} \tag{56}$$

Where *W* is the applied load, *L* is the length of the rope, *E* is the apparent elastic modulus of the rope, and *A* is the metallic cross-section of the rope [41]. The lack of elongation means that there is little strain energy stored in the rope, and more energy is transferred from the springs to the dummy. A disadvantage of using a wire rope is the cost. Wire ropes generally cost more than non-metallic ropes of a similar diameter. Another is the spool radius of the fallen wire rope will be larger than a nylon rope since it is not as flexible. This may cause the wire rope to land in between the test bumper and the wildlife model adding a source of error during the collision.

The second class of rope is a non-metallic rope. This class consists of types of rope such as nylon rope, manila rope, and polyester rope, among other options. These ropes all have varying responses to inputs like abrasion, corrosion, heat, or UV. In general, these ropes are weaker than similar diameter wire ropes, with more elongation (up to a strain of 0.25 at maximum rated load [42]). The main advantage of non-metallic ropes is that they are much cheaper than equivalent. The two strongest non-metallic ropes are nylon and polyester, with nylon being slightly stronger but will stretch more than polyester under load [[42], [43]].

The three ropes that will be compared are polyester, nylon, and wire. For polyester and nylon, a factor of safety of 5 will be worked into their maximum loads, with the factor of safety functioning as general safety, plus a mechanism to account for the extra strain expected in the rope. The wire rope will have a factor of safety of 2, to act as a general safety factor. The rope must support the force from one of the eight helical springs, equal to 4625lb. Therefore, the nylon and polyester ropes must support a load of 23125lb, while the wire rope must support a load of 9250lb. The total length of rope required is approximately 20 feet for every spring (with some excess for tying off the rope, etc. This means that in total, the required rope length is 160 feet for the springs. The required rope diameter and its respective cost are provided in TABLE XIX.

TABLE XIX: COMMON ROPE SPECIFICATION AND PRICES [[44] [45] [46] [47] [48]]

Rope Type	Rope Style	Rope Diameter	Max Load	Cost
Nylon		1-1/16 in	25200 lb	$1.80/ft**
Polyester		1-1/8 in	24800 lb	$1.52/ft***
Wire	6x7 Class IPS Fiber Core, Bright	3/8 in	11720 lb	$2.33/ft
	6x19 IPS Bright-XIP	5/16 in	9380 lb	$3.19/ft

** Cost is for a 600ft spool of 1 in rope [47]. *** Cost is for a 600ft spool of 1-1/4 in rope [48].

As all the ropes listed above satisfy the load requirements and the size requirements of the rope, the ultimate selection criterion is the cost of the rope. As this is heavily dependent on the supplier of the rope, no recommendation on the rope selection will be made. All the ropes listed in TABLE XIX satisfy the requirements of the design.

5.5.3. Spring Mounting

We considered two methods to anchor the springs to the ground. The first option is to use a simply supported beam, such as in Figure 34, to attach the springs to. As there are 8 springs, with 4 on each side of the dummy, we would require two simply supported beams, one for each spring.

Figure 36: Simply supported beam example [49].

Each beam would support 4 springs, roughly evenly spaced out, and held in place with clips. The advantages of this method is that the springs are free to be placed in different locations along one axis, which will help align them so their applied force acts as close to vertically as possible. Working against this concept is the height that this design will add to the springs, and the cost and complexity required to use these as an anchor.

The other option is to design a screw hook, like in Figure 37: A screw hook , to attach each spring individual spring to the ground.

Figure 37: A screw hook [50]

An advantage of this is that it is cheaper than a simply supported beam, and it takes up much less room and adds much less height to the max height of the spring. A disadvantage is that it locks in the location of the springs, allowing for little to no adjustments in their location.

The biggest advantage of using a screw hook as an anchor is that the design of the hook is identical to the required design of the spring loop, save for a gap provided on the spring hook to allow the spring loop and the screw hook to attach. This means that an A401 spring hook would have a required wire diameter of 13/16in, and have a mean loop diameter of 4.425in. For these reasons, we recommend that a screw hook is used to anchor the spring to the ground.

5.6. Operational Components

The operational components of the Drop Test are additional the components that are required for the test to be performed. For the springs and model to be loaded and lifted to the drop height, they need a mechanism to provide that force and then to release the model. In addition, safety measures must be built into the system to ensure that the setup and operation of this testing apparatus is safe at all times. These components are highlighted in this section.

5.6.1. Winch

To lift the wildlife model up to the desired drop height as well as loading the springs in tension, the desired method will be by winch and cable. This method was chosen because it is safe, reliable and will guarantee that the test can be loaded and held in position before it is released. From the helical spring analysis, we determined that the equivalent mass of the cart system and tensioned springs is 38.7klb.

There are two winches that were investigated. The Ramsey HD-P 177.9 [51] and the DP model 45 [52]. Both were rated above the desired 38.7klb load range. TABLE XX provides relevant for the information for our test regarding both models.

TABLE XX: IMPORTANT WINCH MODEL SPECIFICATIONS

Product	Rated Pull (lb)	Line Speeds (ft/min)	Recommended Cable Size (in)	1st Layer Capacity (ft)	Total Cable Capacity (ft)
Ramsey HD-P 177.9	40 000	13.8	0.787 (20 mm)	29	215
DP Model 45	45 000	23	0.75	29	250

Both models exceed the 38.7klb load limit. In addition both have a 1st layer capacity of 29ft which is desirable our test apparatus. The length of cable able to be spooled as the first layer of the winch when it is retracting the cable is important since after the first layer is wrapped, the second layer begins to wrap over top of the first. This causes the outer diameter at which the cable is being spooled about to increase. Increasing the outer diameter at which the cable spools will reduce the load that the winch can lift. With regards to our test apparatus, this means that the 1st layer capacity must be greater or equal to the maximum effective drop height (distance between the test bumper and cart height before release) of the cart. For our design, we have specified the maximum effective drop height to be 10ft. Both winches exceed that value and will provide their rated pull if less than 19ft of cable is spooled on the winch drum before retraction. Both winches provide ample total cable capacity at 215ft and 250ft, which allows for more than enough cable length.

Finally the recommended cable size varies slightly but not by much (only 0.037in) and is not of concern. The Ramsey winch has a slower pull speed which is desirable since the cart stopping height is visually determined when pulling the cart up to the desired drop height. Based on the information provided to us by the manufactures, the Ramsey HD-P 177.9 winch would be a suitable selection for its application. The cost of this winch was obtained from a supplier in Winnipeg. The supplier is Westrock Battery and a conversation with a sales representative provided a unit price of $15,199 for a new model.

5.6.1.1. Winch Cable

The winch has been determined to be the Ramsey HD-P 177.9. The winch has a recommended cable size of 20mm steel wire. Suitable wire with a minimum breaking strength of 68 114 lbs [53] was sourced from an Idahoan company Bluejay Industrial Inc. It has a price of $3.83 US per foot. For our application we will need approximately 45 feet of cable. The breakdown of the total cable length is given in the table below. This is assuming we are using a sheave with a 16in outer diameter sheave.

TABLE XXI: WINCH CABLE LENGTH BREAKDOWN

Section	Approximate Length (ft)
Cart to middle of sheave	15.75
Front Sheave to back sheave	8.20
Curves along front and back sheave	2.10
Back sheave to winch	17.00
Connections	1.00
Total	44.05

Rounding up to the nearest foot brings the total to 45ft and the total cost of the cable is $172.35 US which roughly translates to $229.80 CAD at the Canadian dollar being 0.75 of a US dollar.

5.6.2. Sheaves

The use of a winch and cable system will require sheave to guide the cable as it is being retracted to and from the winch over the top frame of the structure. These sheaves must be able to withstand the 38.7klb tension running across their perimeter during the loading scenario. In addition, the minimum bore of the sheave must be able to fit a shaft that can withstand the maximum stresses from the force of the tensioned cable running over the sheave.

Based on the geometry of the structure, the approximate angle of wrap angle of the cable over the sheave is 90˚. This will result in a reaction force on the shaft of the sheave that is equal to the magnitude of the tension vectors across the sheave. These vectors and the resulting reaction force are depicted in Figure 38 below.

Figure 38: Reaction forces on a pulley due to 90° wrap angle [54].

From the figure above, the tension in the F_{T1} and F_{T2} directions with respect to our analysis are 38.7klb. The resultant force acting on the support is the root sum square of those magnitudes. The final value is shown in the following equation.

$$\bar{F}_{support} = \sqrt{\bar{F}_{T1}{}^{2} + \bar{F}_{T2}{}^{2}} = \sqrt{38.7^2 + 38.7^2} = 54.73 \text{ klb} \tag{57}$$

The sheave support shaft will be required to withstand said amount of force across a shear plane. To determine the shaft diameter we will need to select a material first. A suitable material for this application would be AISI 4130 steel [55]. The yield strength of this material is σ_Y = 66.7 ksi [56]. The shear strength is then approximated to be:

$$\tau_y \approx 0.58 * \sigma_y \tag{58}$$

This results in a shear strength value of 38.686 ksi. Knowing the shear strength and the reaction force we can determine a minimum area for the shaft design. This is determined by the following equation.

$$A = \frac{F}{\tau_y} = \frac{54.73\ klb}{38.686\ ksi} = 1.415\ \text{in}^2 \tag{59}$$

The minimum area is determined to be 1.415 in^2. Adding a safety factor of 2 brings the minimum area to 2.830 in^2. For a solid steel shaft the required diameter would then be:

$$OD_{solid} = \sqrt{\frac{4 * A}{\pi}} = \sqrt{\frac{4 * 2.830}{\pi}} = 1.035\ \text{in} \tag{60}$$

If we assume an outer shaft diameter of 5in, then we a hollow shaft can be used. The minimum thickness of the shaft would be:

$$ID_{hollow} = \sqrt{{OD_{hollow}}^2 - \frac{4 * A}{\pi}} = \sqrt{5^2 - \frac{4 * 2.830}{\pi}} = 4.626\ \text{in} \tag{61}$$

Therefore, the minimum thickness of a hollow shaft would be 0.364in (approximately 3/8in). The 5 in outer diameter was chosen to accommodate bearings that would be placed at the ends of the shafts. Loading cycle analysis is recommended before finalizing the shaft thickness or diameter but for this section we will assume it to be safe for many loading cycles. The next step is to determine a sheave.

Since the working load limit of the required sheave is 54.73klb, most suppliers do not carry stock options. This may require custom ordering of sheaves for the pulley use. We requested a quote from Gunnebo Johnson for a price on their 16in finish bored sheave model #4320. Information regarding the wrap angle (90°) the line speed (13.8ft/min) and load (38.7 klb) were given. Their representative said that a custom sheave would possibly be needed to be manufactured if the sheave model did not meet our specifications. An approximate price of a similar type of sheave was $925 US [57] which is roughly $1233 CAD at the Canadian dollar being 0.75 a US dollar.

5.6.3. Safety Stop

Safe use of the drop test design is paramount in avoiding injury to the end users. With safety in mind we designed a built-in safety stop to ensure that the 800 Kg carriage could be securely held in place at multiple heights. Throughout the models, safety components of the design will be modeled in yellow to ensure easy visibility and understanding of the safety components. Mechanical stops are the most inherently safe stops due to their lack of moving components and the reliability of solid objects. Mechanical stops can come in many forms including; pins, blocks, latches, ratchet gears, etc. For the design of our gravity drop system the safety bins and blocks are the most suitable to ensure the carriage could not fall during the operational setup process

Safety blocks can be used at the base of the structure. The blocks would be placed in the channel of the W beams. Thus, if the carriage was to be released unexpectedly, the carriage arms and the rollers would come in contact with the block and be stopped.

Figure 39 and Figure 40 show the dimensions and the operational position of the safety stop block.

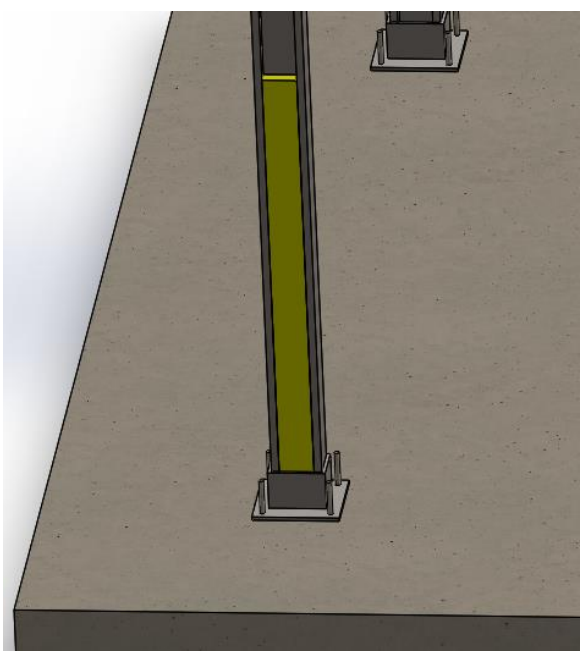


Figure 39: Block safety stop, placed in operational position [58].

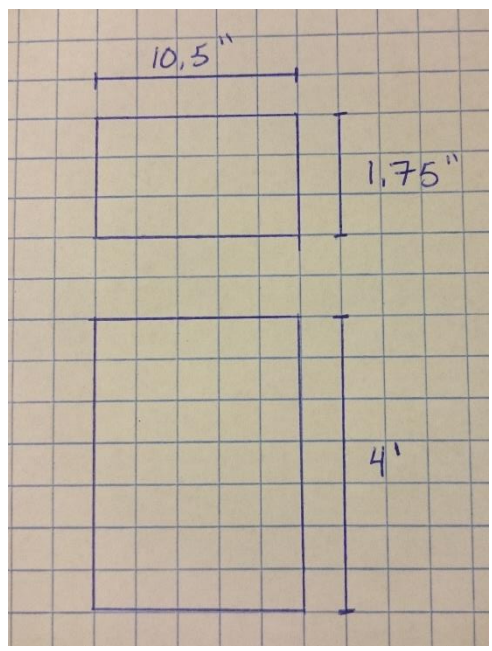


Figure 40: Block safety stop dimensions [58].

The next safety stops to be considered were the use of safety pins. The safety pins could be designed into the W beams. These safety pins can be inserted, through the W beams, at three foot intervals up to nine feet. With multiple height options available, the end user has the ability to move the carriage to differing heights, depending on the needs of the tests they are conducting, safely and easily.

Shear stress was used to determine the necessary thickness of the safety pins. The shear stress can be calculated using:

$$\tau = \frac{F}{A} \tag{62}$$

With $\tau_{\max}$ = 78 GPA for ASTM A108 [59] and F=7848 N. Accounting for 4 pins around the structure and a safety factor of 4, the required area of a pin is 0.0001 m^2. Converting to inches a 0.5-inch diameter safety stop pin would be sufficient to secure the carriage at any height without the springs attached. The safety pin operational positioning and dimensions are found in Figure 41 and Figure 42.

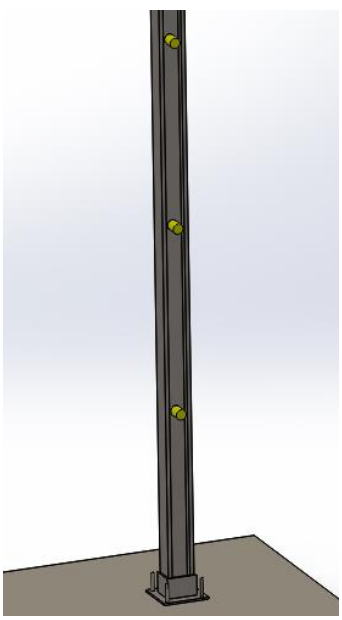


Figure 41: Pin safety stop, placed in operational position [58].

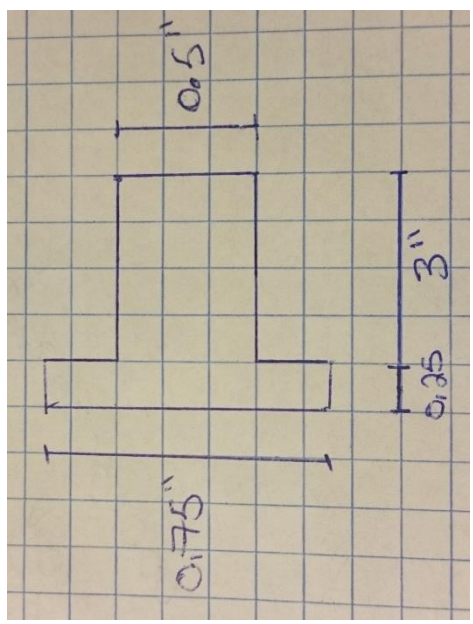


Figure 42: Pin safety stop dimensions [58].

The ability of the safety stop pin to be moved to different vertical locations allows more freedom for the user. This maneuverability and the ability to safely and securely hold the carriage the safety pins will be used in the final design.

5.6.4. Quick Release Mechanism

The quick release mechanism will allow for the loaded system to be released instantaneously from a safe distance. This is a very important component of the test since it is required to hold up the entire loaded drop cart and release it instantaneously. Two options were looked at for the quick release. The first being a small section of nylon rope attached between the drop cart and winch cable. The small section would be cut when the loaded drop cart is to be released. A suitable diameter of rope wold be 1-1/2 in diameter rope, with a breaking strength of 48600 lbf minimum breaking strength [46]. The main benefit of this option is that the nylon rope is cheap and long lengths could be acquired for multiple test runs. Drawbacks are that that this option would require a cutting mechanism that can be operated from a safe distance. In addition, the rope lengths may lose strength when they are cut to the appropriate size and cause them to be unreliable under heavy loads. This option was discarded.

The second option was a mechanical quick release mechanism. This mechanism is known as the Sea Catch. This product was designed for various industries where loads are required to be securely held and released from safe distances [60]. It has been used for scientific purposes, specifically drop testing. The sea catch has safety features, such as a hitch pin to ensure that the load being lifted cannot be released until the operator removes the pin. The pin can be removed by having a line running to the pin and pulling the line. Once the hitch pin is removed, by pulling the release lever will release the load. The release lever can be operated from any angle between 90 degrees from perpendicular to the line of the load. It is important to note that the hitch pin is not needed to lock the Sea Catch. It is an added safety measure. A depiction of the operation of a Sea Catch unit is shown in Figure 43.

The benefits of this option are much greater than those of the nylon rope option. For this design we will go ahead with the TR-15 model. The biggest drawback for this option is its

cost, which is $10608 per unit [61]. This cost was obtained from a company called Trotac Marine which is located in Victoria, BC. More information about this product will be presented following Figure 43.

Figure 43: Sea Catch Operation [60]

Figure 44: Sea Catch important geometries [60].

The specifications of the model (TR-15) that would be used for our test are presented in TABLE XXII: SEA CATCH MODEL TR-15 IMPORTANT SPECIFICATIONS Figure 44 above provides a diagram of the unit as well as the associated geometries.

TABLE XXII: SEA CATCH MODEL TR-15 IMPORTANT SPECIFICATIONS [60]

Specification (Model TR-15)	**Value**	**Unit**
Safe Working Load	50200	lb
Breaking Weight	251000	lb
Unit Weight	69	lb
Drop Shackle Diameter (B)	2	in
Support Shackle Diameter (F)	2.16	in
Drop Shackle Width (A)	2.38	in
Shackle Size	1-3/4	in
Effort to Release	460	lb
Cost	10608	$CAD

6. Test Procedure

A detailed test procedure was requested by the client as a final deliverable. In order to satisfy the client's request and to ensure clear communication of test functionality, a procedural list was created. It should be noted that the provided test procedure applies only to the repeatable simulation of a wildlife collision and not to the fabrication or construction of the test apparatus. The test procedure list is shown in TABLE XXIII. [25]

TABLE XXIII: TEST PROCEDURE LIST

No.	Procedure Description
1	Engage winch to raise wildlife model cart assembly to set up position.
2	Secure safety pins under wildlife model cart assembly.
3	Place bumper affixed to bumper support structure in position.
4	Secure spring power system from concrete pad to wildlife model cart assembly.
5	Remove safety pins under wildlife model cart assembly.
6	Re-engage winch to final desired drop position.
7	Engage quick release mechanism to trigger drop.
8	Re-attached quick release mechanism to wildlife model cart assembly.
9	Repeat steps 1 and 2.
10	Bumper affixed to bumper support structure may be viewed or removed for inspection.

7. Failure Analysis

Proper failure analysis allows engineers to identify a multitude of failure mechanisms that may be present in their designs. Using failure analysis, we can thoroughly and systematically analyze all the working components, as a whole and as individual parts. With this systematic approach, we can find and address many issues that would have otherwise been left unaddressed when the product was prototyped or placed in production.

In the following section the root cause of failures will be identified. By identifying these failures in the design process we can implement controls that will reduce the frequency and severity and increase our detection ability.

7.1. FMEA

Failure modes and effects analysis allows our team to identify manners in which the test apparatus may fail and eliminate or reduce this risk of failure. By analysing the potential failures of our system at this point we can modify and remove hazards from the design before they became apparent during the manufacturing process and operation of the system.

The first step of our analysis was identifying the test procedure that the system would go through during normal operation. The normal operation steps can be found TABLE XXIII. With the required test procedure developed we began developing failure modes that could occur for each of these steps. Each of these failure modes were identified for their effects, their causes and their current controls. Next the system was evaluated for the severity of each failure, the frequency of each failure and the detection effectiveness of each system. The severity, frequency and detection were all rated on a scale from one to ten.

The severity of the failure mode was rated as a 1 if the effect, of the failure, was of a minor nature that would not cause any substantial effects on system performance. Alternatively, it was rated as 10 if the effect could cause serious personal injury or noncompliance with government regulations. The frequency was rated as a 1 if the failure rate was 1<20,000 and a 10 if the failure rate was 1<2. Finally, the detection effectiveness was rated a 1 if the likelihood of detection was very high and a 10 if there was absolute certainty of non-detection.

Once we had ratings for the severity, frequency and detection of the failure mode we multiplied their values together to achieve a risk priority number. Due to the multiplication of the values, the higher the RPN means a higher overall danger the failure mode would be to the system.

Our FMEA table can be found in TABLE XXIV. The table is organized by the function, which is the order of operations in the test procedure.

TABLE XXIV: FAILURE MODES AND EFFECTS ANALYSIS SUMMARY

Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (1-10)	Potential Cause(s) of Failure	Frequency (1-10)	Current Controls	Detection (1-10)	RPN
Engage winch to raise wildlife model cart assembly to set up position.	Moose carriage binds on W beams	Carriage remains stuck	5	Structural support bending	4	Calculations/Simulations of loaded structure	4	80
Engage winch to raise wildlife model cart assembly to set up position.	Winch stalls	Carriage cannot be raised or lowered	7	Improper rating of winch	3	Supplier specifications of winch	3	63
Secure safety pins under wildlife model cart assembly.	Pins are secured incorrectly	Carriage will hit damper	10	Improper test procedure instructions	2	In-depth test procedure	2	40
Secure safety pins under wildlife model cart assembly.	Pins do not handle the moose carriage load	Carriage will hit damper	10	Insufficient strength in safety stops	1	Calculations/Simulations of loaded safety stop	2	20
Place bumper affixed to bumper support structure in position.	Bumper support bolt head is sheared off	Delay/Repair	3	Forklift operator without enough experience	1	Only forklift certified operators can move bumper support	9	27

Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (1-10)	Potential Cause(s) of Failure	Frequency (1-10)	Current Controls	Detection (1-10)	RPN
Secure spring power system from concrete pad to wildlife model cart assembly.	Spring wires are tangled	Improper disassembly	1	Improper test procedure	5	In depth test procedure for assembly and disassembly	3	15
Remove safety pins under wildlife model cart assembly.	Lift cable breaks	Carriage is dropped	10	Spring force too large	1	Safety stops can be removed from outside the drop zone. Maximum spring load specifications	2	20
Re-engage winch to final desired drop position.	Winch stalls	Carriage cannot be raised or lowered	7	Improper rating of winch	3	Supplier specifications of winch	3	63
Re-engage winch to final desired drop position.	Spring breaks	Spring assembly is shot away	10	Spring strength to weak	2	Maximum spring load specifications, Safe zone during operation	3	60
Re-engage winch to final desired drop position.	Lift cable breaks	Carriage is dropped	10	Spring force too large	1	Maximum spring length	3	30
Re-engage winch to final desired drop position.	Moose carriage binds on W beams	Carriage remains stuck	5	Structural support bending	4	Calculations/Simulations of loaded structure	4	80

Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (1-10)	Potential Cause(s) of Failure	Frequency (1-10)	Current Controls	Detection (1-10)	RPN
Engage quick release mechanism to trigger drop.	Quick release does not unhook	Lower carriage and unhook manually	8	Improper rating of quick release	2	Supplier specifications of quick release	4	64
Engage quick release mechanism to trigger drop.	Moose carriage rebounds on impact	Data will be inconsistent	7	Improper elastic rebound calculations of moose model	3	In depth analysis of the energy transfer during impact	3	63
Engage quick release mechanism to trigger drop.	Moose carriage binds on columns	Test cannot be completed	8	Warping due to load	2	Stress and deflection analysis of the test structure	3	40
Re-attach quick release mechanism to wildlife model cart assembly.	Bumper fails under impact	Moose cart model drops	4	Bumper joint has a bad weld	2	Tire absorption at the base of columns	8	64
Re-attach quick release mechanism to wildlife model cart assembly.	Moose support bar fails	Test data is inaccurate	8	Improper design analysis	3	In depth analysis of forces on the support bar	2	48

Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (1-10)	Potential Cause(s) of Failure	Frequency (1-10)	Current Controls	Detection (1-10)	RPN
Bumper affixed to bumper support structure may be view or removed for inspection.	Inconsistent impact energy	Data will be less valuable	7	Improper design analysis	3	In-depth system analysis	2	42
General usage	Concrete cannot handle test collision impacts	Concrete will degrade and become uneven	2	High Force being transmitted through beams to the pad	1	Reinforce concrete with repair to combat deterioration	1	2

7.2. Tower Structure FEA

Finite element analysis was performed on the truss structure. After the tower analysis in section 5.2.3 we determined it would be wise to verify our calculations with an FEA analysis. In addition we are able to determine the maximum deflection of the structure to verify that a cart would be able to run along the columns without locking. By using FEA this can verify that a tower structure made from commonly available steel beams can be used in the creation of a support structure.

7.3.1. Methodology

The Solidworks model was created from the truss skeleton in Figure 15 shown in section 5.2.3. The 3 dimensional sketch of the structure in Solidworks is shown in Figure 45 below.

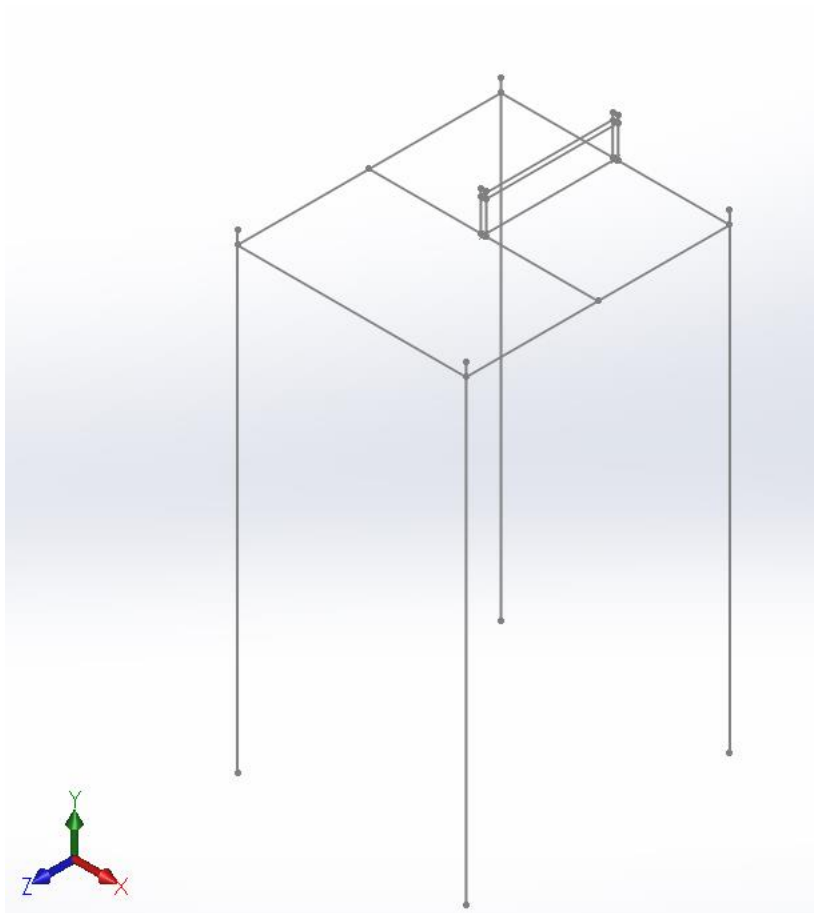


Figure 45: Solidworks three dimensional sketch of the tower structure truss model [62].

As you may observe there is a difference in the cross beams where the sheaves are placed. An additional set of beams are added in line with the Z- axis force component to reduce stress concentrations at the sheave support connection points with their respective beam. This was added using intuition that there would need to be additional support for the sheave supports. Notice that the sheave supports are separated into two sections to accommodate the sheave placement. A zoomed in look at the separated sheave supports and additional cross bracing between supports is shown in Figure 46 below.

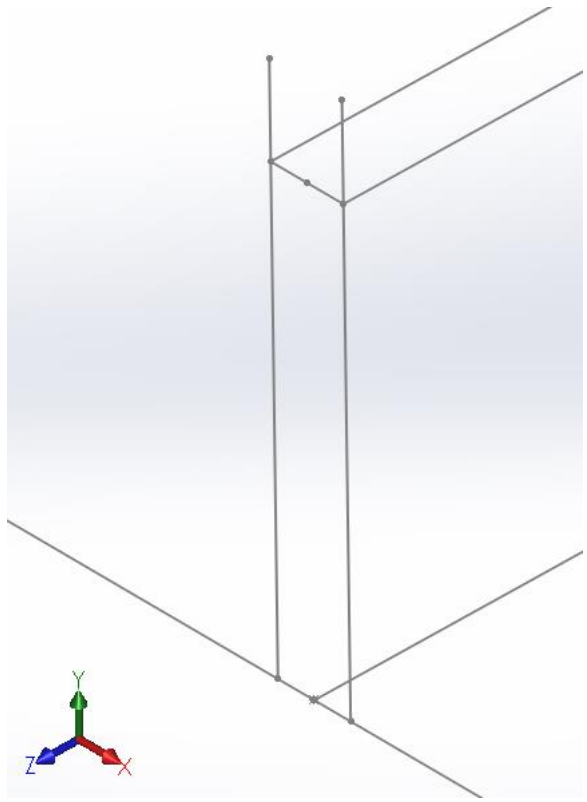


Figure 46: Additional cross bracing between pulley supports in truss tower structure sketch [63]

In the previous two figures, there are lengths that pass above of lengths which does not resemble the sketch in the hand analysis section. This is due to the compensation for the beam cross sections in Solidworks. This alters the design slightly from the design of the hand calculations, thus is a source of error in the analysis. The cross sections for the columns, outer perimeter beams and lower z-axis cross beam is the Wx12 steel beam. Its Solidworks cross section is shown in the figure below.

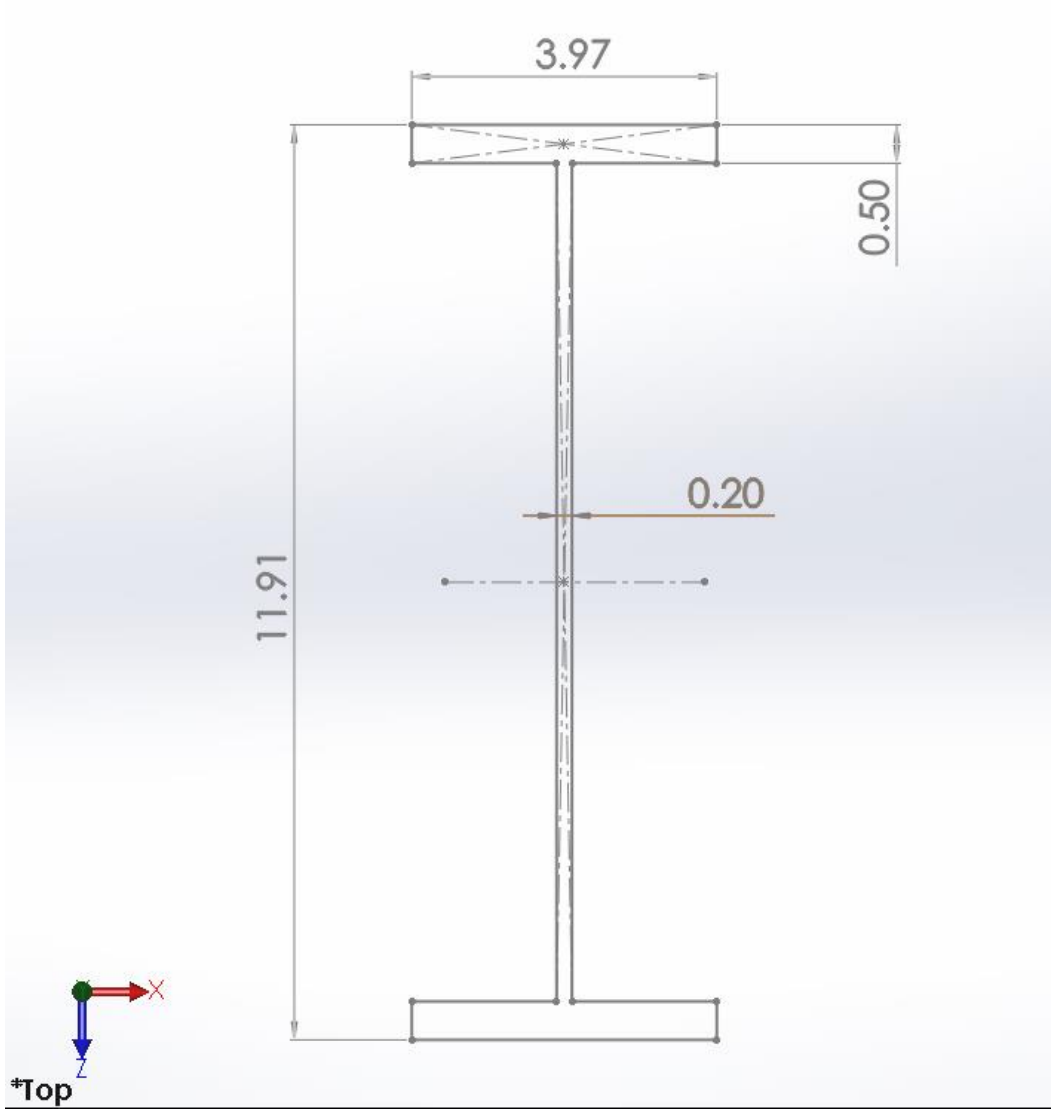


Figure 47: Wx12 steel beam cross section sketch and dimensions [64]

The cross section for the sheave supports and sheave support bracing is a 3x5 ansi inch C-channel beam taken from the Solidworks library. The dimensions for the c-channel cross section are shown in Figure 48 below.

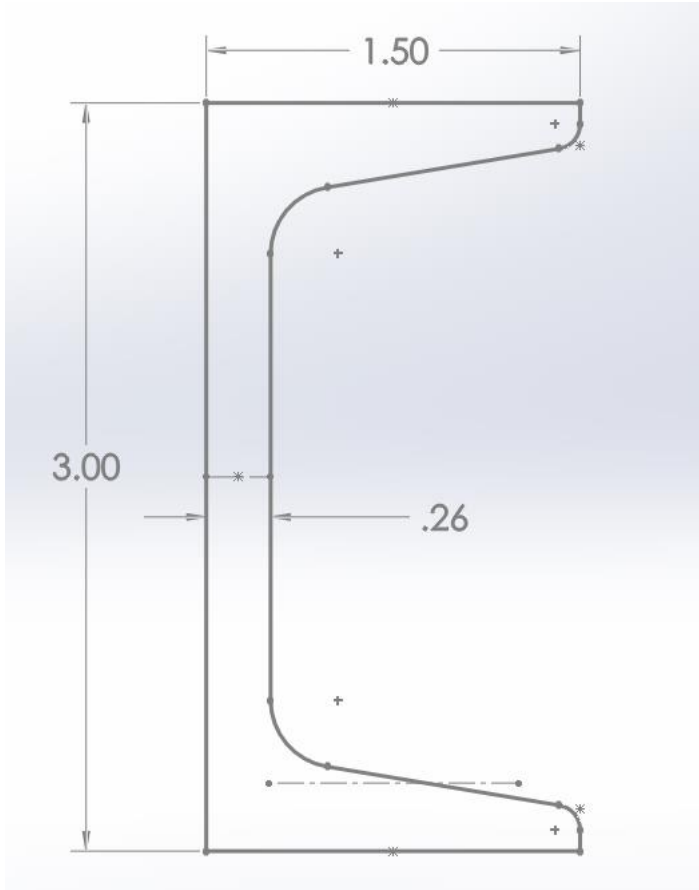


Figure 48: 3x5 c-channel cross section [65]

The section properties for the two types of beams are provided in TABLE XXV below.

TABLE XXV: SOLIDWORKS MODEL BEAM CROSS SECTIONS

Property	Beam Type	
	Wx12 beam	**3x5 c-channel**
Area [A] (in_2)	6.15	1.47
Moment of inertia [I_{xx}](in^4)	156.14	1.85
Moment of Inertia [I_{yy}](in_4)	150.94	0.25

The material that is being used for the analysis is ASTM A992 steel. This material is used for both types of cross sections, making the entire truss structure a homogeneous material. The material properties for this steel are shown below in TABLE XXVI

TABLE XXVI: MATERIAL PROPERTIES FOR ASTM A992 STEEL [66]

Property	Value	Unit
Yield Stress	60000	psi
Young's Modulus	29000	ksi
Shear Modulus	11,200	ksi
Poisson Ratio	0.3	N/a

These are used to create a custom material in Solidworks since ASTM A992 is not part of their stock library of materials. Now that the material and cross sections are specified they are added to the sketch skeleton as structural members using the weldments function. This creates a structure that is simplified into a truss with nodes at all connection points between members and free ends. The weldment truss model is shown in Figure 49 with the nodes, fixtures and loads present.

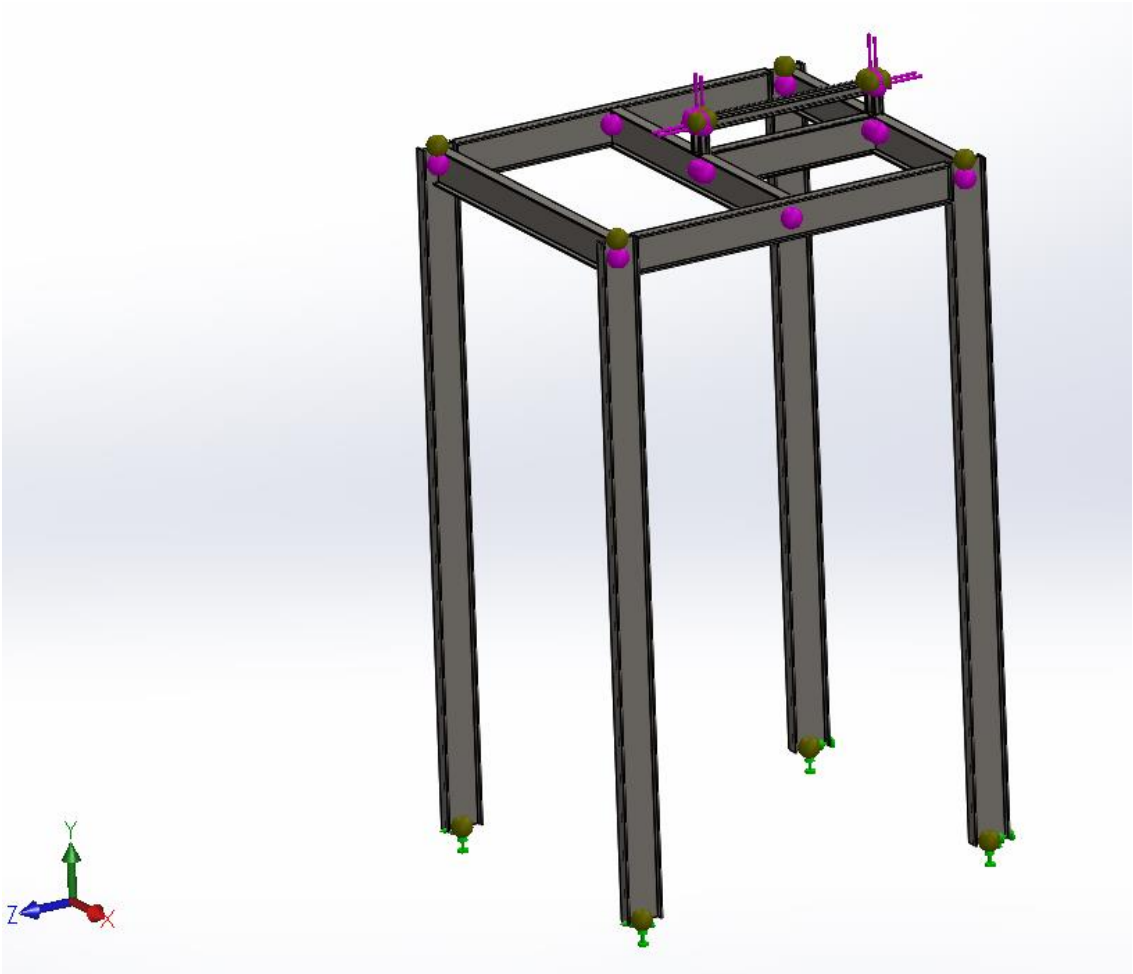


Figure 49: Tower structure truss model with nodes, fixtures and loads [67]

The green arrows represent complete fixtures at the base of the tower columns and the purple arrows near the sheave supports represent loads that would be acting on the sheave supports. The loads are transferred from the cables that run over the sheaves with 38.7klb of tension running through them. A close up of the loads is shown in Figure 50.

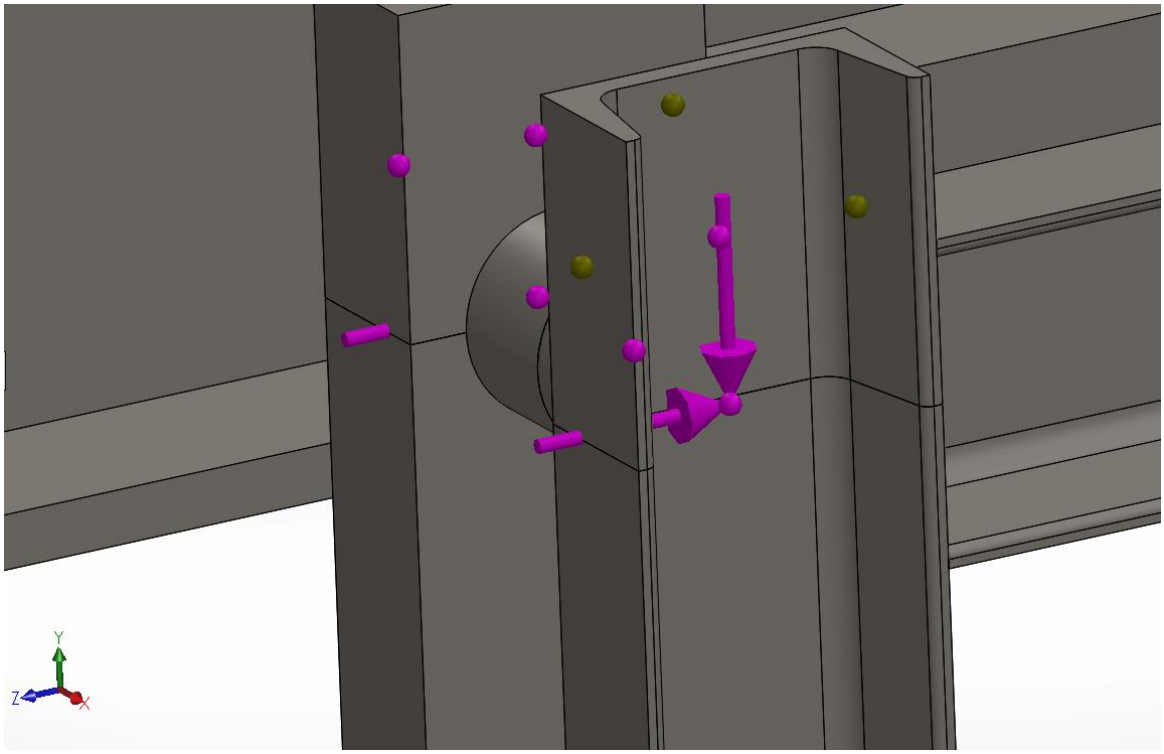


Figure 50: Close up view of the force vectors acting on the sheave support nodes [68]

The force vectors in the z-axis at the outer sheave are in the opposite direction of the vectors in the previously depicted figure. The magnitudes of the loads are shown in the TABLE XXVII below.

TABLE XXVII: LOAD MAGNITUDE AND DIRECTIONS FOR TRUSS STRUCTURE STATIC ANALYSIS

Sheave Support	**Load Vector Direction Axis**	**Magnitude (lb)**
Central	Y-axis	-19,350
	Z-axis	-19,350
Outer	Y-axis	-19,350
	Z-axis	19,350

The loads are half the calculated load (38.7klb) because the force will be split between the two supports after it is transferred through the sheave shafts. Mesh control was applied to the structure. The element sizes that were used in three simulations are shown in TABLE XXVIII below. The number of elements was held constant at 15 and then the element size was incrementally reduced.

TABLE XXVIII: STATIC ANALYIS RESULT CONVERGENCE

Iteration	Element size (in)	Highest upper bound axial and bending stress (psi)
1	5	38840.2
2	2.5	38840.2
3	0.5	38840.3

This shows that the results of the model were consistent while increasing mesh density. A reason why there is only a tenth of a difference between the plots is because the truss model mesh will creates an approximation of the structural members. These members act as theoretical truss member and will not experience stress concentrations at edges or high stress points.

7.3.2. Results

Figure 51 to Figure 53 below shows the three different stress plots for the structure that are compared with the analytical results.

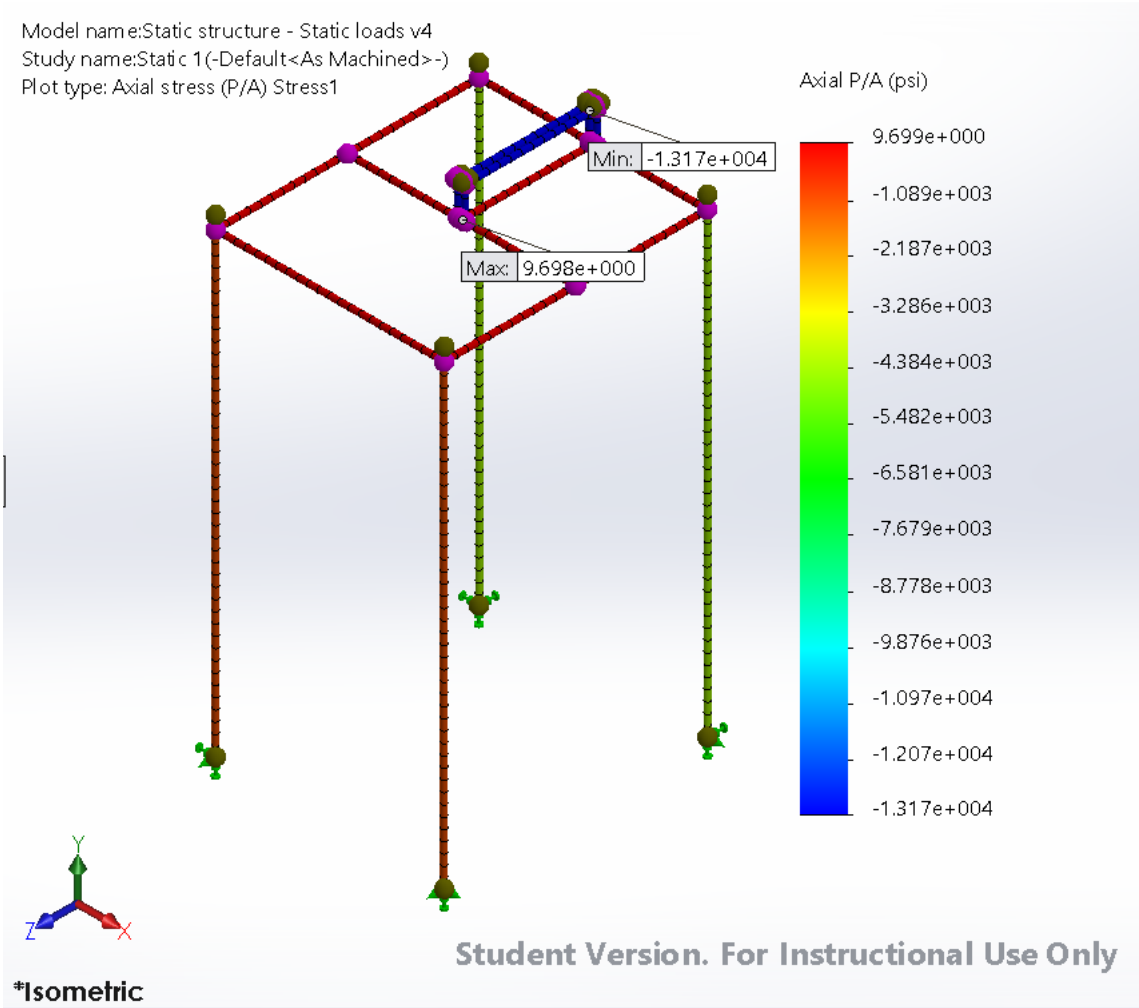


Figure 51: Axial stress plot from truss structure analysis [69]

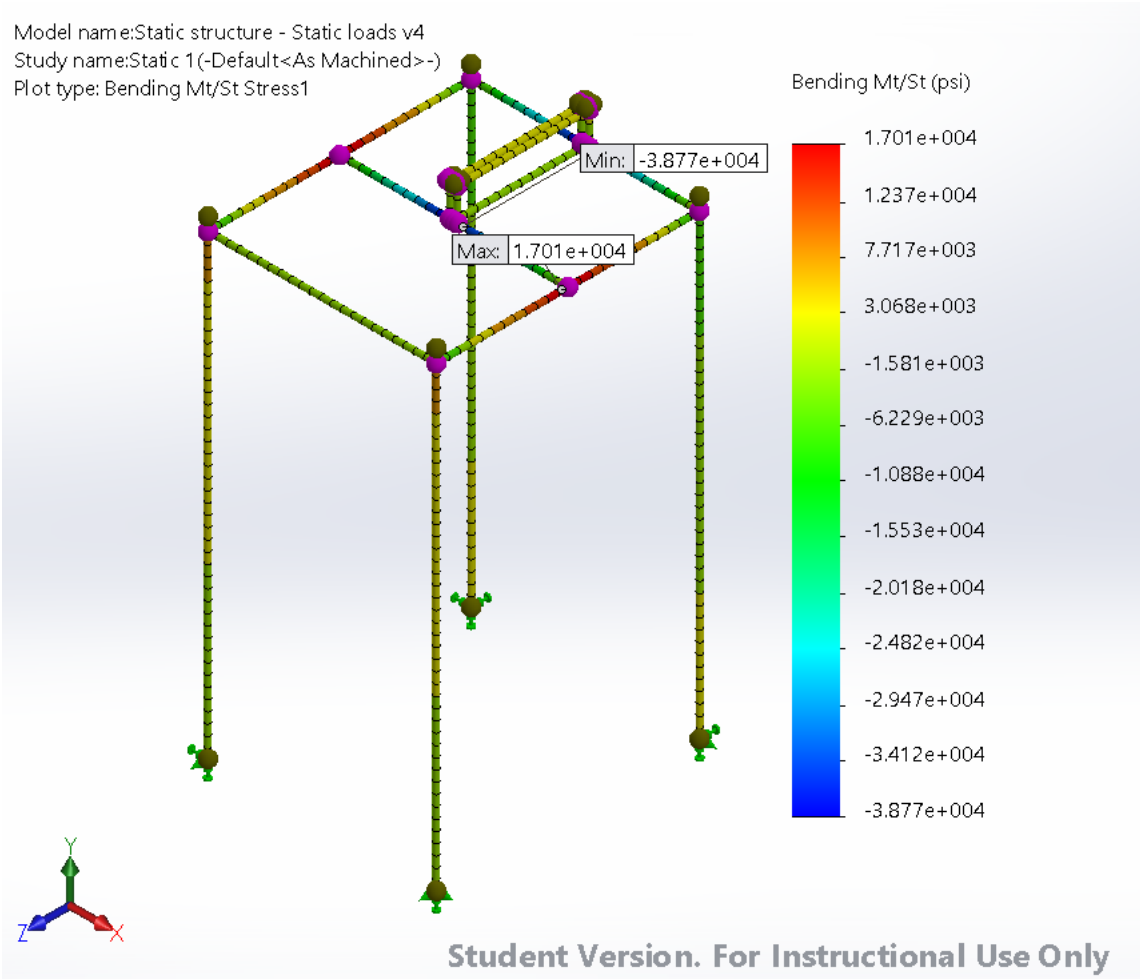


Figure 52: Upper bound bending in y-axis plot from truss structure analysis [70]

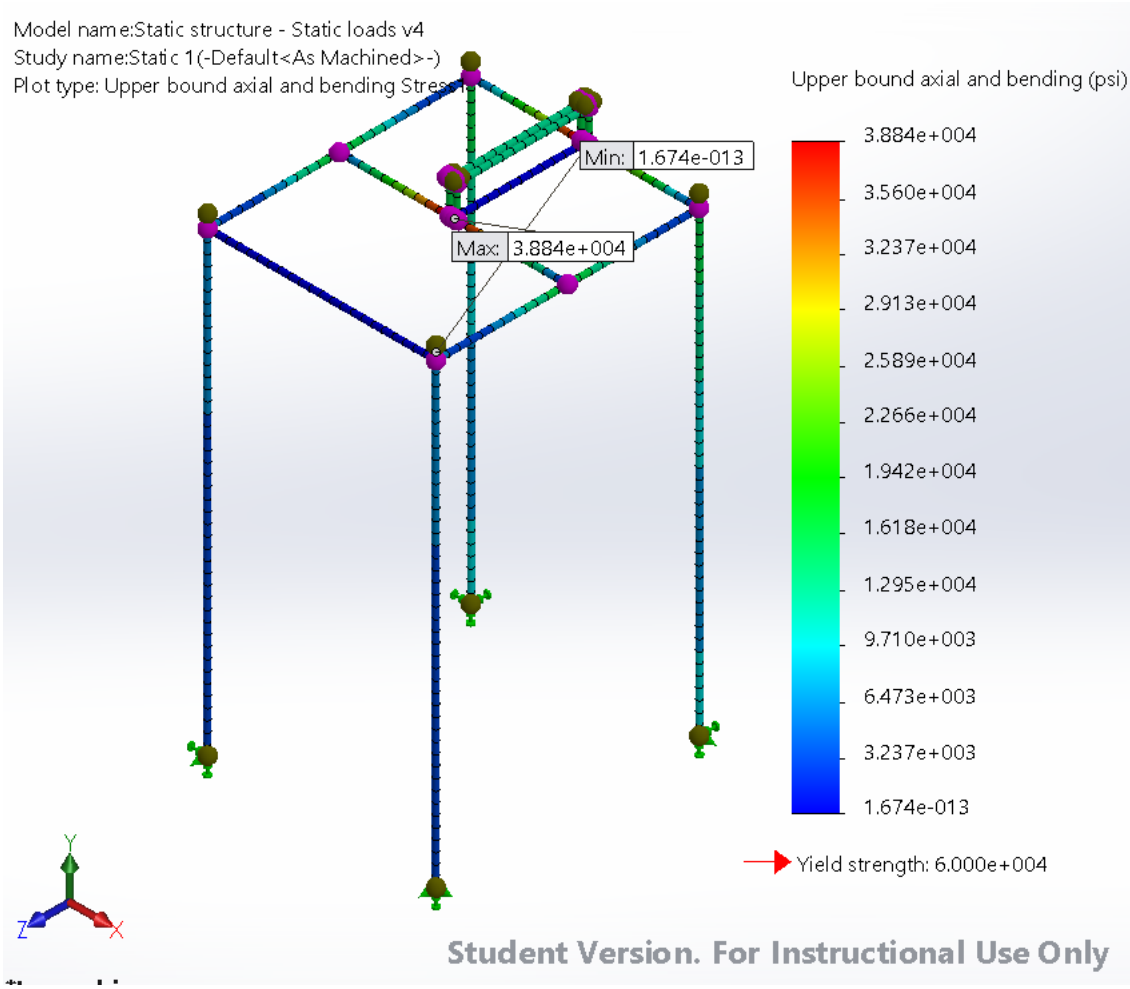


Figure 53: Combined upper bound bending and axial stress plot from truss structure analysis [71]

The results of the experimental FEA stress analysis are compared with those calculated analytically in section 5.2.3. A comparison of results is presented in the TABLE XXIX below.

TABLE XXIX: COMPARISON OF ANALYTICAL AND EXPERIMENTAL RESULTS FOR TOWER STRUCTURE

Member	Axial stress (psi)			Bending stress (ksi)			Combined Bending and Axial Stress (ksi)		
	Analyt.	Experi.	% Error	Analyt.	Experi.	% Error	Analyt.	Experi.	% Error
AT	4653	4709	1.19	N/A	----	----	N/A	----	----
CU	4653	4709	1.19	N/A	----	----	N/A	----	----
GV	1551	1582	1.96	N/A	----	----	N/A	----	----
HW	1551	1582	1.96	N/A	----	----	N/A	----	----
BE	6204	68.84	98.90	13.04	0.135	99.97	19.24	0.204	99.94
BE’	----	1317	----	----	1.177	----	----	14.23	----
ABC	N/A	-----	----	43.15	38.77	10.15	N/A	----	----
ADG	3088.43	176.7	94.27	17.83	17.01	4.60	20.92	38.84	46.43
CFH	3088.43	176.7	94.27	17.83	17.01	4.60	20.92	38.84	46.43
DEF	N/A	-----	-----	43.15	38.77	10.15	N/A	-----	-----

The results show low percentage error for the columns of the support structure but in the beam we see much higher errors. Part of this error is due to the addition of member *B'* during the modelling process of the structure. In addition, this structure is not a true truss as was assumed during the analytical analysis. Its members were still treated as beams during the Solidworks analysis. Based on these results, further analysis is required before this design should be implemented. The maximum stress values for both the analytical and experimental results show overall stresses below the yielding strength of the material. We believe a tower structure of this design with similar materials would be feasible to construct and implement with the test method.

Figure 54 below shows a side by side comparison of the models to compare members. The notation in the skeleton truss drawing is the same notation used in Solidworks when measuring a member's stress.

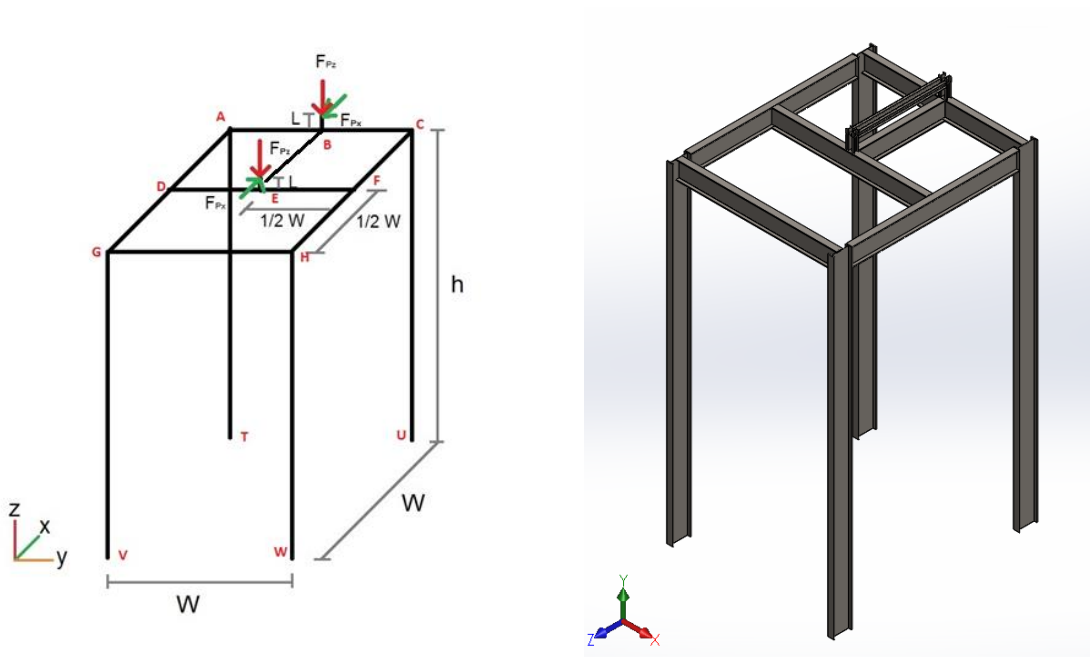


Figure 54: Notation used in measuring and comparing truss member stresses for Solidworks FEA [72]

Figure 55 to Figure 58 below show the displacement results from the analysis in all three axis as well as residually.

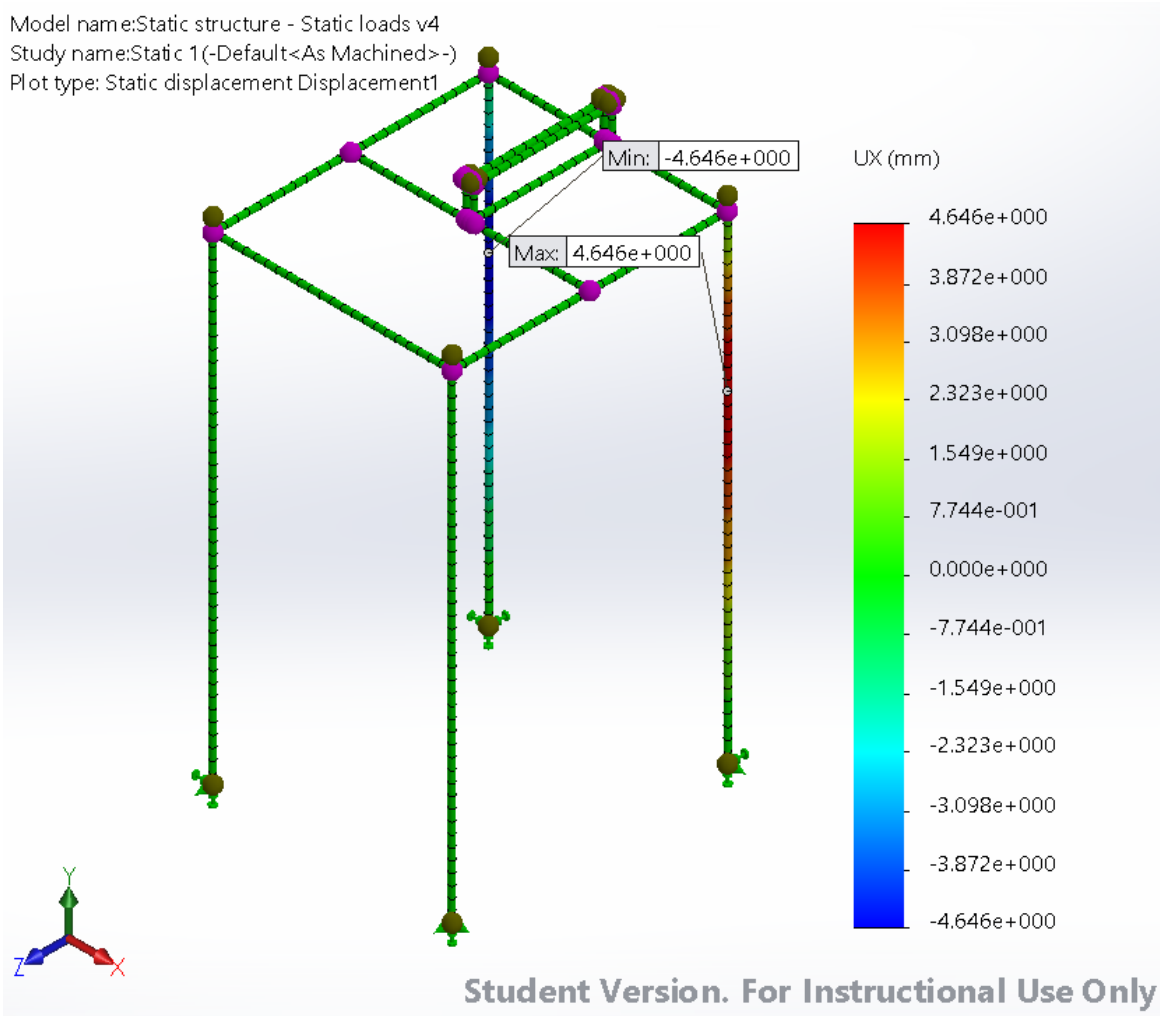


Figure 55: Displacement in x-axis plot from truss structure analysis [73]

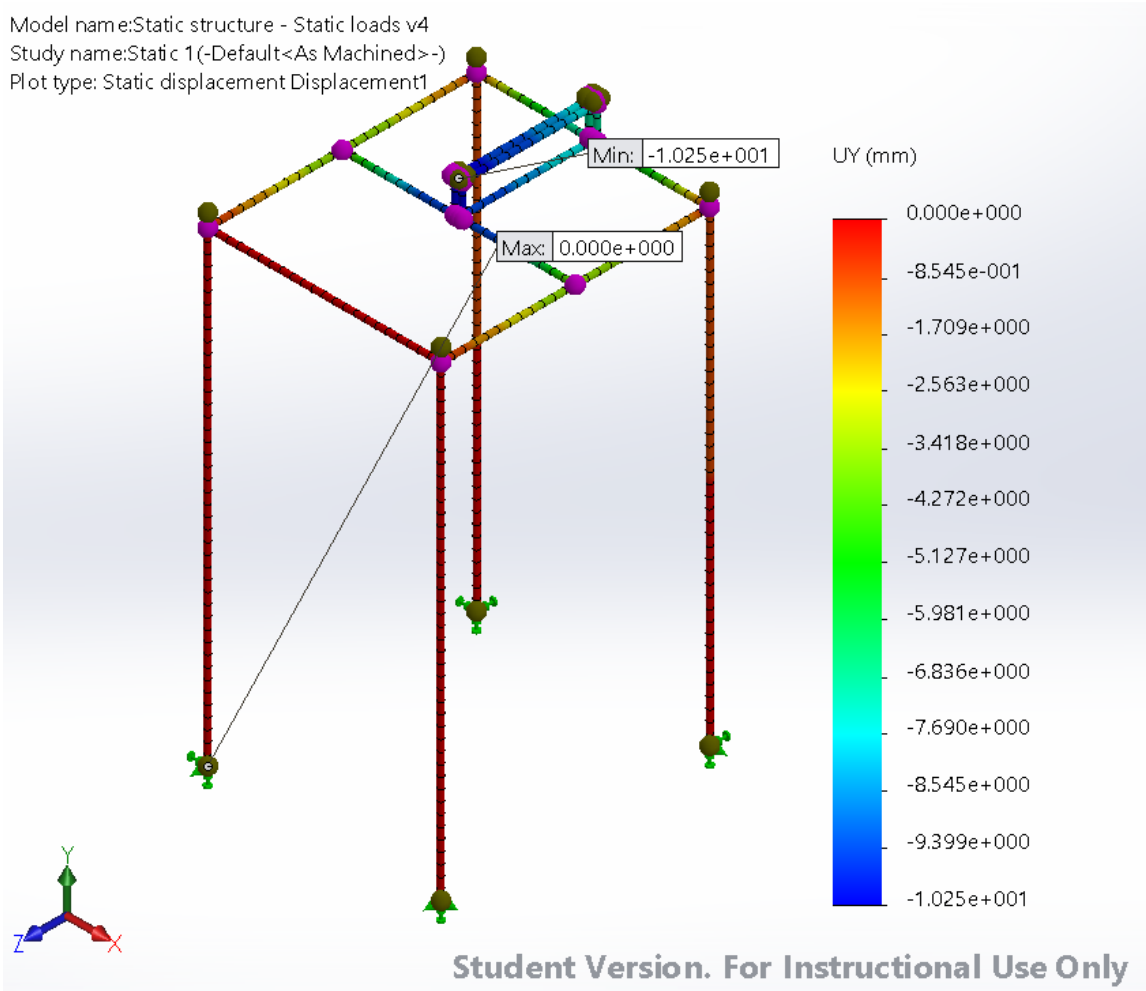


Figure 56: Displacement in y-axis plot from truss structure analysis [74]

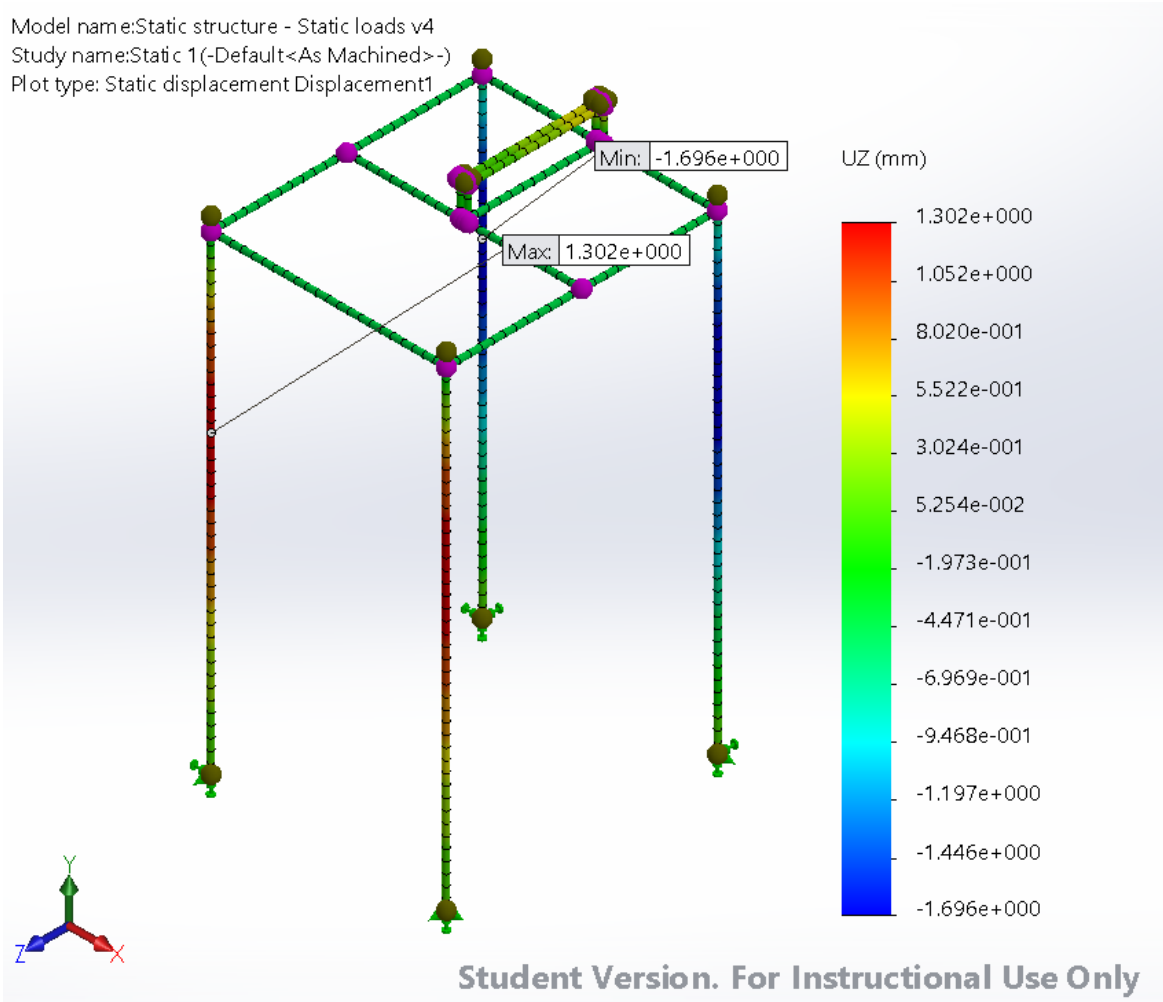


Figure 57: Displacement in z-axis plot from truss structure analysis

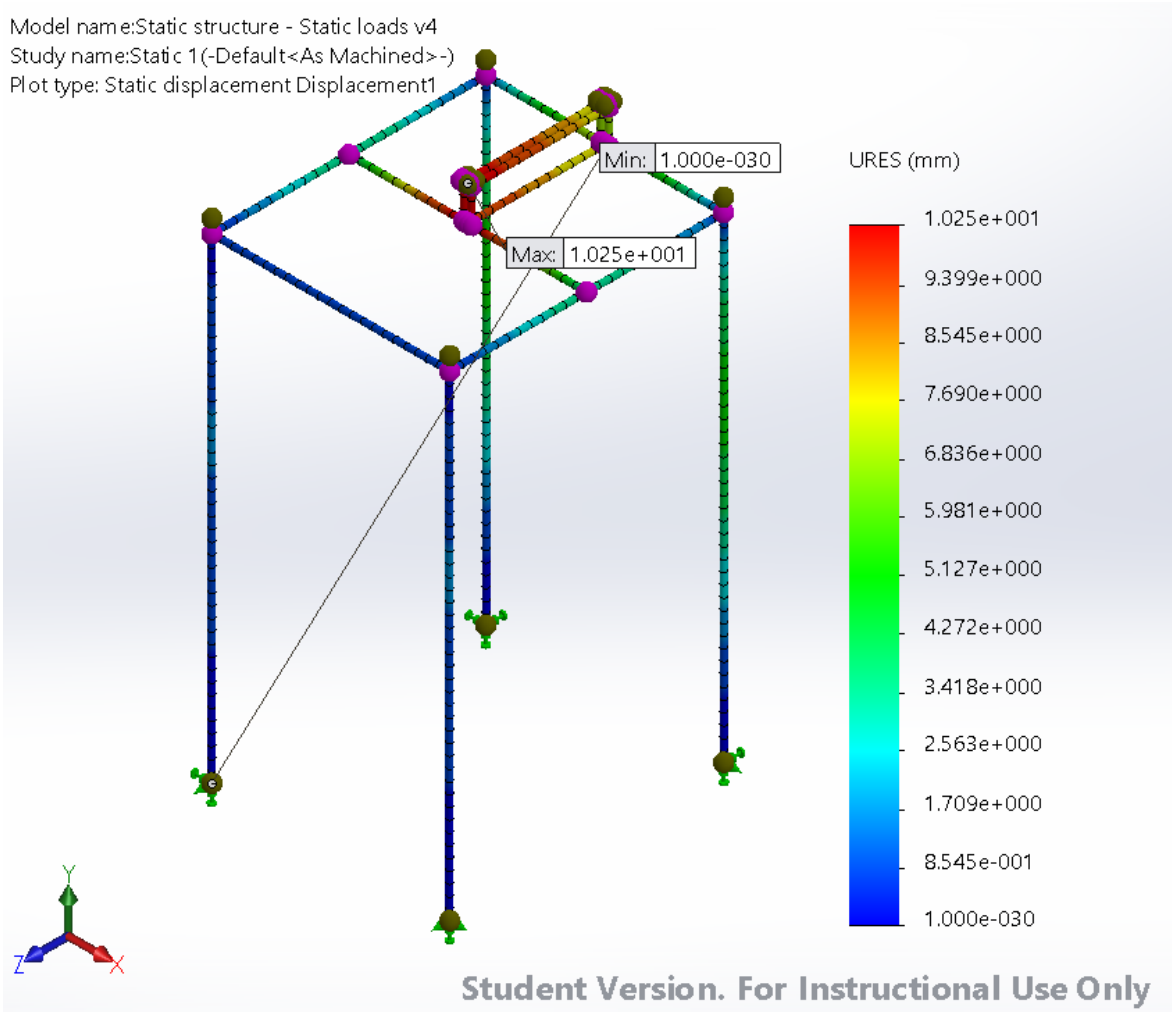


Figure 58: Residual displacement plot from truss structure analysis [75]

Based on the displacement plots we see that the point that incurs the greatest displacement is at the sheave mount. This deflection is 10.25 mm in the negative y axis direction. Point of interest are the deflections of the columns in the x and z axis. If the deflection is over 5 mm in the x –axis the drop cart may lock up. Excessive deflection in the z-axis may cause the cart to lock up as well. The maximum magnitude of deflection in the x-axis on a single column is 4.646 mm. The maximum magnitude of deflection of a single column in the z-axis is 1.302 mm. The deflection in the x-axis will allow for the cart to operate without locking up due to the flexible plates that are mounted to the rollers. If enough play is provided to the guide rollers (total of >2*1.302 mm = 2.604 mm) then there should be no lock up as well. The residual maximum deflection of a single column

is shown in Figure 59 below. Figure 60 depicts an exaggerated deflection of the entire structure for aiding in the visualization of the deformed structure.

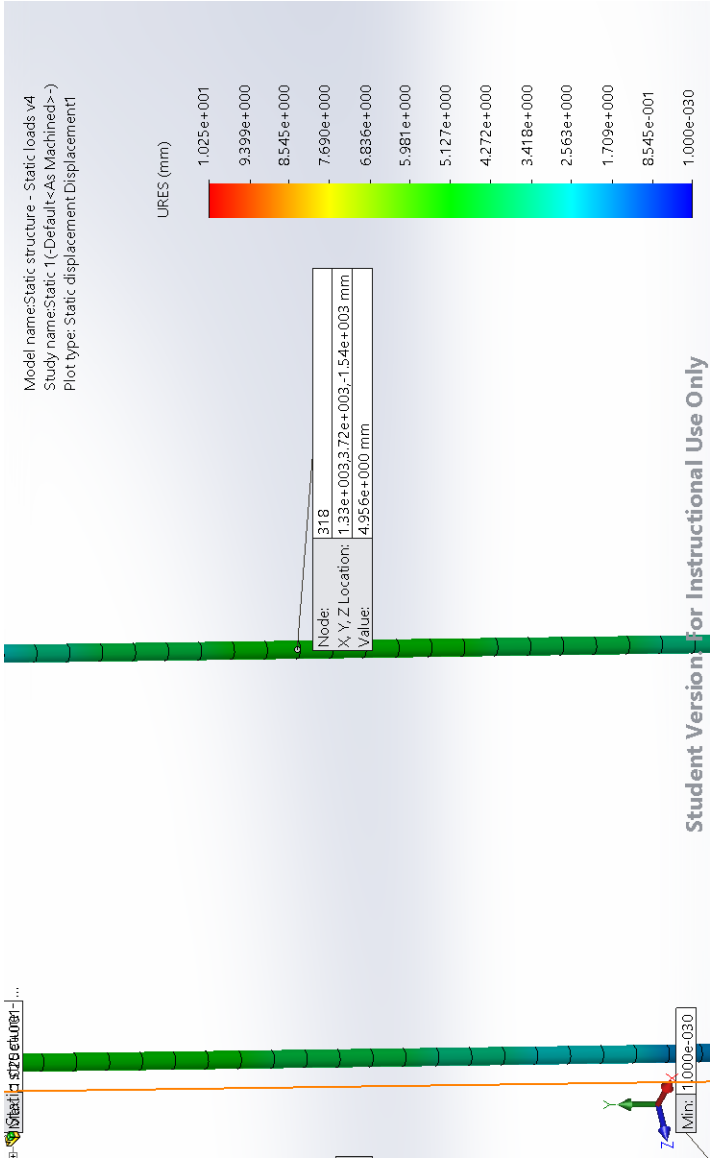


Figure 59: Maximum residual displacement of a single column from truss structure analysis [76]

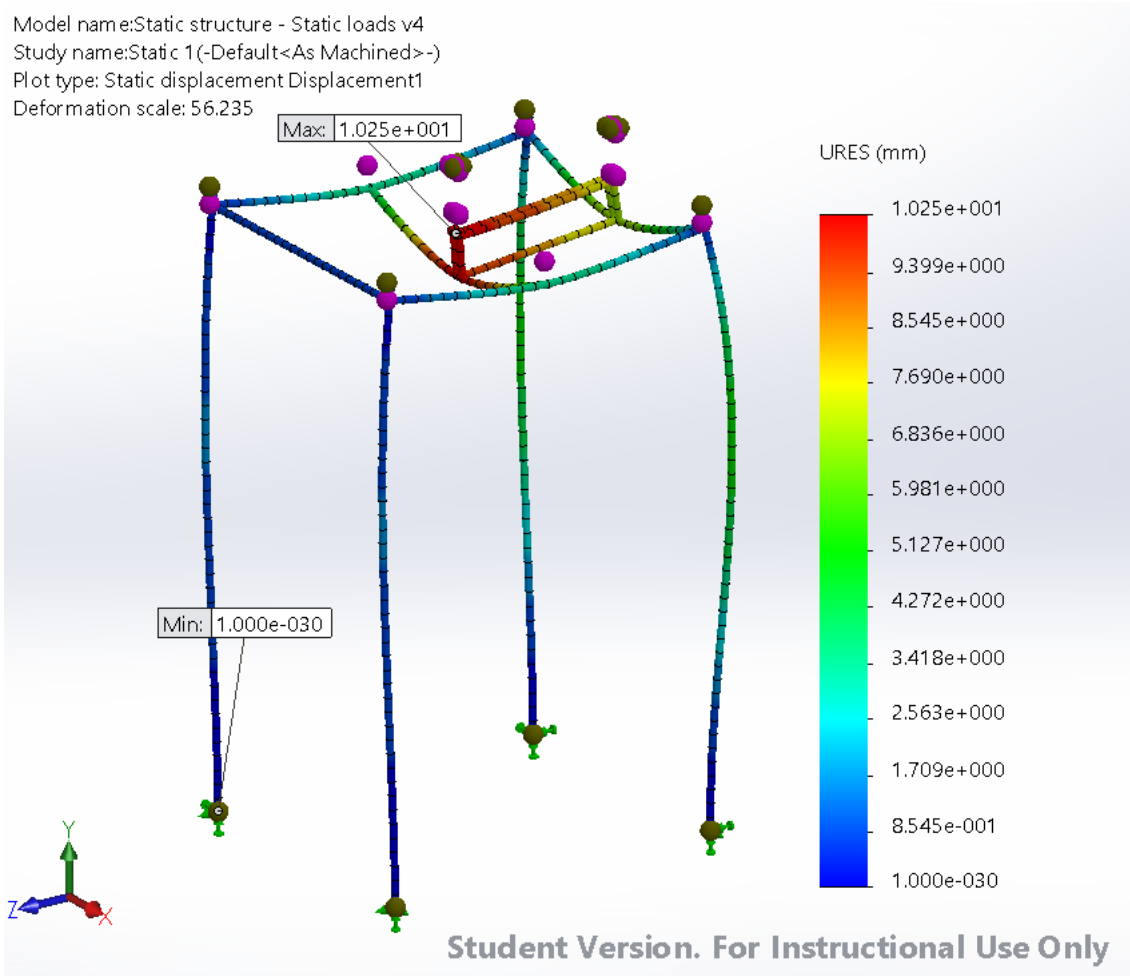


Figure 60: Exaggerated combined deflections of truss structure [77]

There are no analytical calculations to compare the results of the experimental results against. However, we know that the axial stresses in the columns were accurately recreated in the simulation we compared with the analytical results because of the small error (>1.2%). With this information we can conclude that the forces causing the buckling deformation are accurate and thus the deflections in the columns are accurate. This reinforces the design of the drop cart in the sense that if given enough play in the rollers (> than 1.302 mm in z-axis for a single column) and small amounts of deflection with a flexible material, the cart will not lock up during the loading scenario. Unless the tower vibrates at its resonant frequency when the cart is released the deflections should not exceed the maximum static deflections.

8. Overall Design Summary

The final design of our moose drop test assembly combines the four main components that we analyzed through the final design details section. The combination of the four components delivers a functional moose drop test concept that HERD can use to recreate dynamic collisions. Through analysis, we have found the necessary energy requirements to recreate an impact between a 200 kg animal and a semi-truck moving at 80 km/h. An impact with the 800-kg carriage dropping at 40 km/hr will produce the equivalent force of 38 700 lbs or 49 KJ.

The carriage supporting the moose was analyzed giving a structure able to withstand a collision with energy equal to 37 800 lbs. Square tubing, 4.5” x 4.5”, with ½ walls, 2 tonnes I beam rollers and a 3” solid rod was used to reinforce and add weight to the moose model. The rendered components are found in Figure 61.

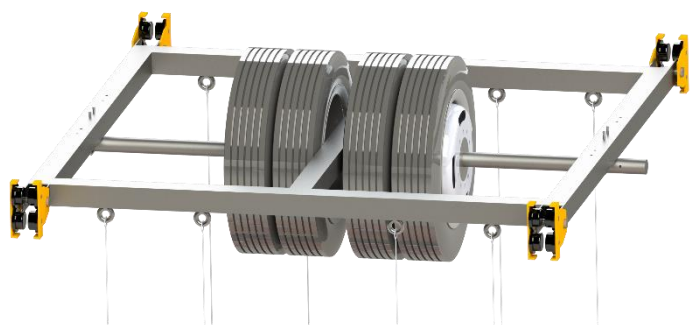


Figure 61: Carriage with moose model [58].

Helical springs are used to add power, along with gravity, to the system to replicate the collision forces on a highway. The spring forces allow the height of the tower structure to be decreased so that it is more compact and allows for easier installation due to smaller beam members. The required forces were found to be equivalent to the spring dimensions outlined in TABLE XXV.

TABLE XXX: SPRING DIMENSIONS.

Dimension	Value (in)
Wire Diameter	0.825
Mean Diameter	4.428
Free Length	23.15
Operating Length	38.12

The spring dimensions add the necessary energy to the moose model to reach 49 KJ. With 49 KJ of energy the system reaches the desired bumper with the desired energy. The spring design is found in Figure 62.

Figure 62: Spring model [58].

The final component we added to the assembly was the bumper support. Analyzing the forces of the collision we found the dimensions 2.5 in x 2.5 in and 60 in tall were necessary for the stumps to withstand the impact. Cross bracing was also implemented to allow the support to be moved horizontally and to allow for non-axial loading. This functionality allows the user to apply the impact across the bumper at multiple points. The bumper support model can be found in **Error! Reference source not found.**.

Figure 63: Bumper support model [58].

Assembling these components results in the overall design of the moose drop test .The final render of the moose drop test assembly with the tower support structure, can be found in Figure 64.

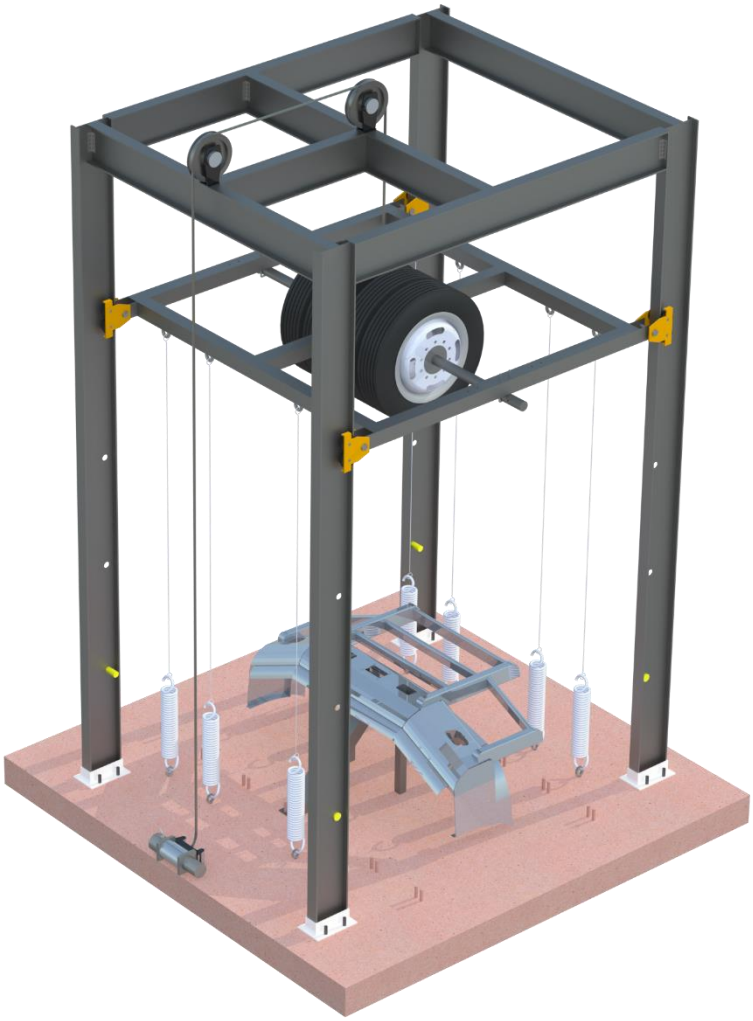


Figure 64: Moose drop test assembly [58].

9. Cost Analysis

An initial total project budget requirement given by the client was $10,000. As the project progressed and the magnitude of the project became more apparent, the client reduced the prioritization to stay within a $10,000 budget. To ensure the project remained as close to the original budget of $10,000 as possible, a detailed cost analysis was conducted. The scope of the analysis consists solely of material costs and does not consider labour, installation, maintenance or other possible associated costs and provides a conservative estimate. The cost analysis summary is shown in TABLE XXXI.

TABLE XXXI: COST ANALYSIS SUMMARY

Design Component	Description	Reference	Cost
Concrete Foundation	4m x 4m x 0.25m with Rebar and 36 J Bolts	[78]	$2,500.00
Tires and Wheels	-4 295-75R22.5 Goodyear Tires -4 22.5" x 8.25" Steel Wheels	[79], [80]	$800.00
Rollers	4 2-Ton Beam Trollies - Princess Auto	[25]	$680.00
Springs	8 Helical Springs, ASTM A401 Steel, 13/16" Spring Wire Diamter, 5-1/4" Outer Diameter, 32" Length	[33]	$5,100.00
Ropes	160 ft Nylon Rope	[47]	$288.00
Winch	- Ramsey HD-P 177.9 Hydraulic Winch - 45 ft of 20 mm Steel Wire Rope	[81]	$15,428.80
Sheave	2x 16 in Outer Diameter wire Rope Sheave	[57]	$2466.67
Quick Release Mechanism	Sea Catch TR-15	[61]	$10,608.00
Steel	-A572/A992 Steel H Beam, 20' Length x 4 -3" Dia. 1045 TGP-Precision Shafting, 8' Length -4-1/2"x4-1/2"x1/4" A500 Square Steel Tube, 24' Length -2" Dia. 4140 Steel Round, 2'Length	[35]	$3,214.62
Final Cost			$41,086.09

The concrete foundation quote was provided by Haywood Concrete Ltd, located in Haywood, Manitoba. All prices in the summary are in Canadian Dollars. Costs for this project could be further reduced by buying used parts, fabricating parts such as the quick release mechanism, or scaling down the performance requirements of the apparatus, allowing the performance specifications of the design components to be reduced, thereby requiring less components (ie. Less structural steel) or smaller components (ie. Smaller winch and pulleys).

10. Conclusion

A dynamic wildlife collision test apparatus and method was successfully designed for Herd North America Industries Inc. The final overall design satisfies the needs provided by the client. The client's needs were organized into the three major components of the design, the test apparatus, moose wildlife model and power method.

The most important test apparatus needs include compatibility with the HERD AeroPLUS bumper, durability and consistency. These needs are satisfied as the test apparatus component of the final design allows for consistent and repeatable collisions by utilizing a solid rail drop system. The bumper support structure was designed specifically to be compatible with the HERD AeroPLUS bumper through the integration of the universal mounting plate by Herd. In addition, the concrete base and steel components allow the test apparatus component to be durable enough to withstand numerous cycles of collision testing. Although the design is not portable, portability was defined as one of the least important needs and therefore does not significantly impact client satisfaction.

The most important moose wildlife model needs include compatibility with the rest of the design and elastic composition resembling a real moose. These needs are satisfied as the wildlife model cart assembly is directly integrated into the power method and test apparatus making it fully compatible with the final design. Additionally, the sand filled rubber tires of the moose wildlife insert provides a realistic simulation of the composition of a real moose. A bonus is the moose wildlife insert can be scaled to simulate weights of various animals by changing the number and size of tires.

The most important power method needs include consistency and adjustability. These needs are satisfied as the helical spring and rope assembly, along with gravity, deliver consistent and reliable power to the wildlife model cart assembly. Power can also be easily adjusted by removing springs and by reducing the drop distance from the wildlife cart assembly to the bumper. The final impact speed of 80 km/h is lower than the original impact speed specified by the client at 100 km/h. As max speed was defined

as one of the least important needs and the importance of this need was further reduced through client feedback, the final impact speed is not considered to significantly impact client satisfaction.

The final cost of the design was determined to be $41086.09. This cost exceeds the original design budget of $10,000. Cost was defined as the least important need for all three design components, including test apparatus, wildlife model and power method. In order to satisfy needs of higher importance, cost was sacrificed, resulting in a final design cost higher than the original budget. Cost and design parameters were carefully optimized by the team to ensure the greatest possible client satisfaction.

Recommendations to the client are provided as follows in order to ensure successful implementation of the final design. The team recommends the client initiate collision testing scaled back from the provided design conditions. This includes testing at a lower speed and lower force by reducing the drop distance, removing the spring components and reducing the weight of the moose model by removing tires and/or using smaller and lighter tires. Thereafter, the client may scale up through several progressing collision tests until the design conditions provided by this report are met. This process is necessary to verify the performance of the design and to reduce the risk of design failure during implementation.

Lastly, the final deliverables required by the client are provided. These deliverables include CAD models of the test apparatus and bumper support structure, a detailed moose model design, provided through both written description and CAD model, and a methodology describing the collision test, provided in the form of a procedural list for the testing method.

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Appendix A – Concept Development and Selection

Table of Contents

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Concept Generation

Concept generation was broken down into two distinct phases. The two phases described in this section are “Internal Concept Generation” and “Group Concept Generation”. This was done in order to ensure that our team generated a wide variety of concepts. Having large amounts of concepts is recommended because it increases problem solving productivity. The following sections of the report will describe the processes that our team took in order to generate a wide variety of concepts.

Internal Concept Generation

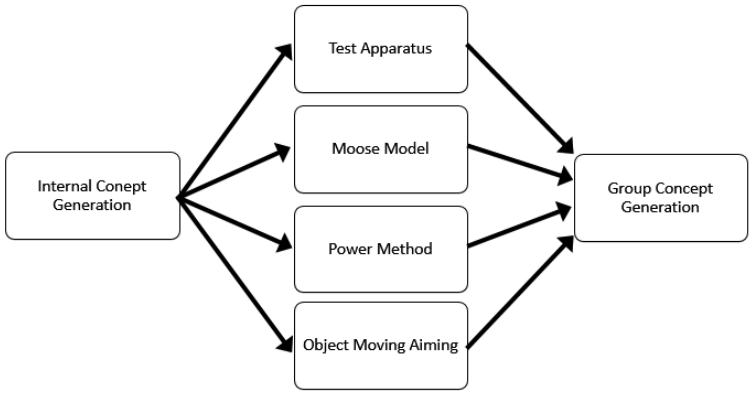


Figure 1: Concept generation phase breakdown.

Figure 1 breaks down the process our team followed to complete our internal concept generation. Once the background and research phases were developed and reviewed the team could continue the process with internal concept generation. The first step taken was individual brainstorming of the four main components of our project. The test apparatus, moose model, power method and object moving and aiming. With everyone having brainstormed their own concepts and ideas for the four main components we scheduled a team brainstorming meeting.

The team brainstorming meeting allowed all members to present their individual concepts to the group. Once all the individual concepts were put forward the team made comments and built new ideas off the ideas brought forward. This led to the development of the concepts found in Figure 2.

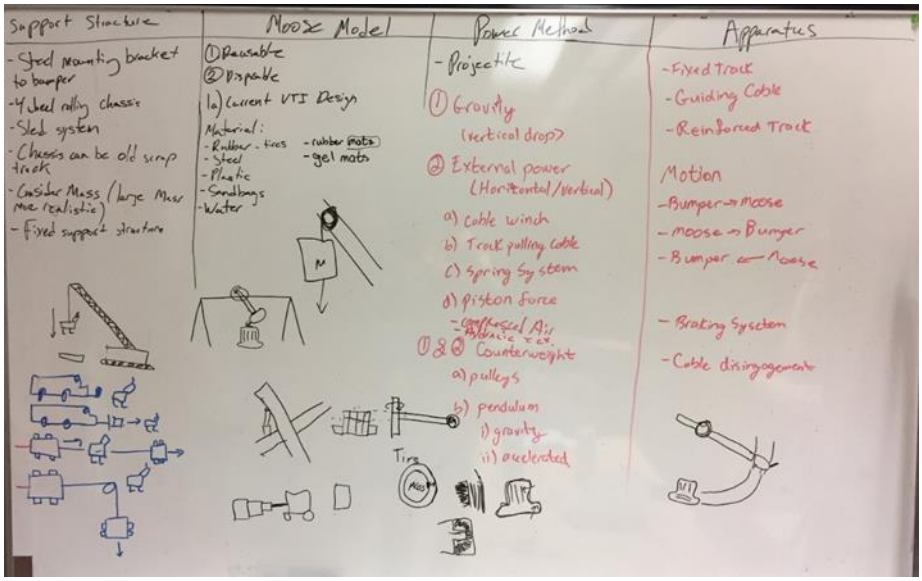
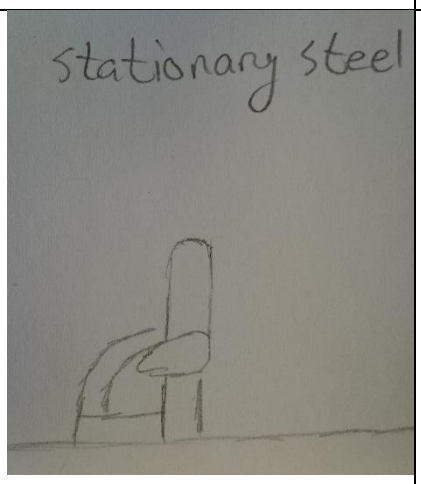
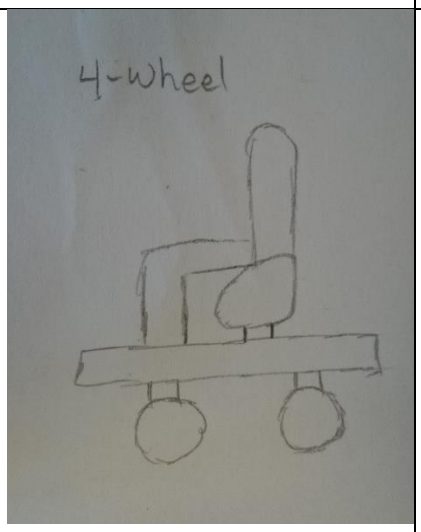
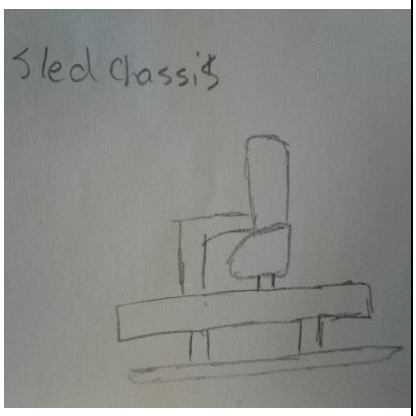
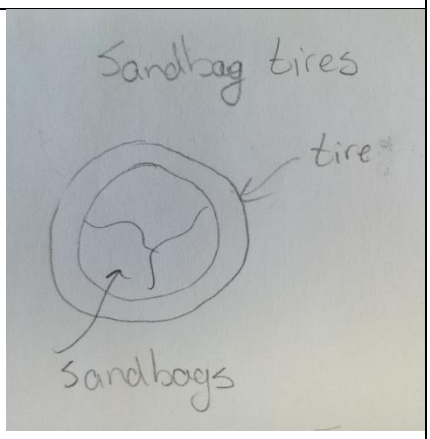
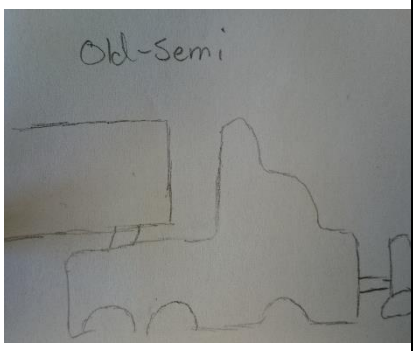
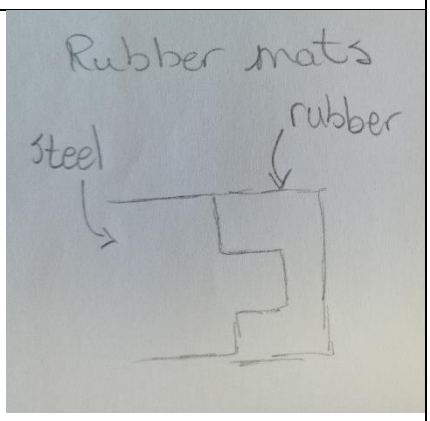


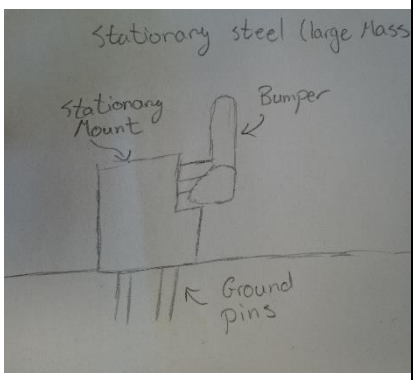
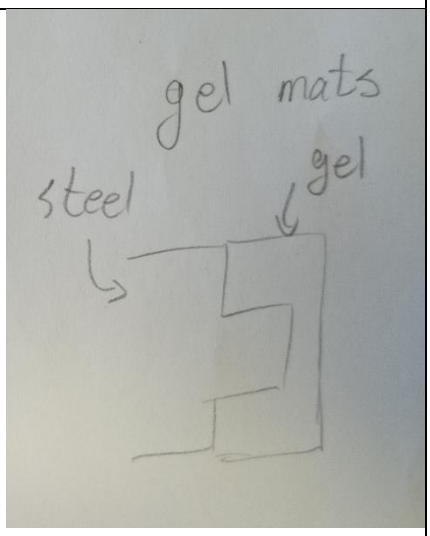
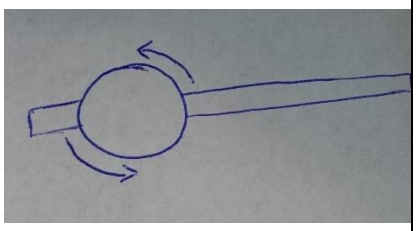
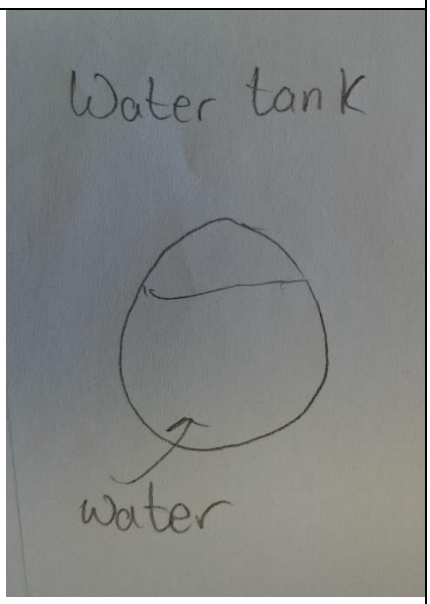
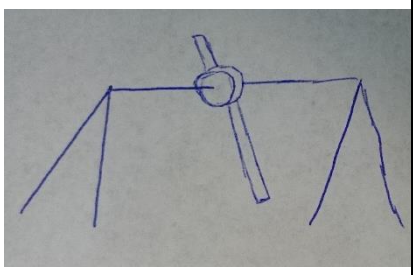
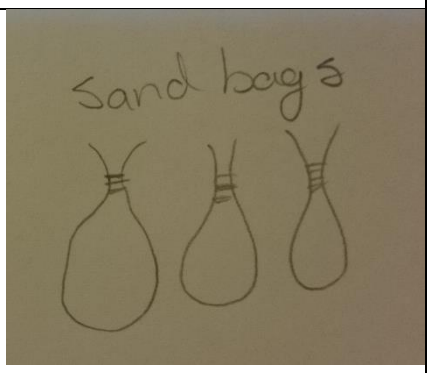
Figure 2: Individual concept generation amalgamation [1]

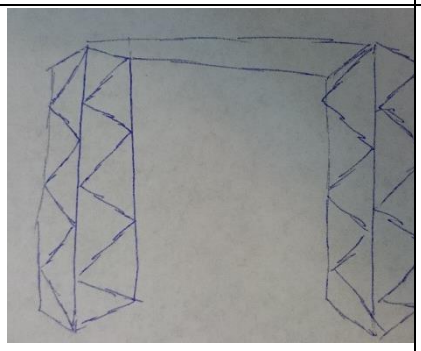
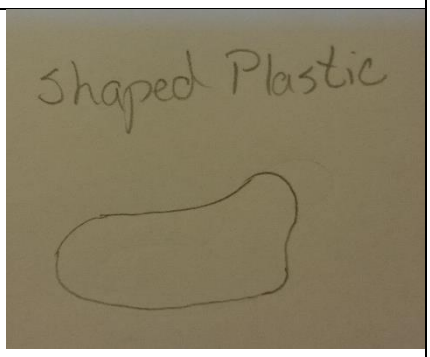
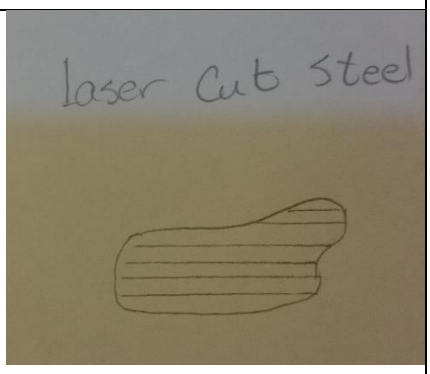
With the individual concepts combined we could see the feasible solutions and the concepts that would be best suited for our test. In order to keep our ideas organized we developed TABLE I and TABLE II.

TABLE I: SUPPORT STRUCTURE AND MOOSE MODEL CONCEPTS [2]

Test Apparatus		Moose Model	
Stationary steel mounting bracket to bumper		VTI design [3]	

Test Apparatus		Moose Model	
4-wheel rolling chassis System		Myth busters design [4]	
Sled chassis system		Sand-bags loaded into tires	
Attach bumper to old semi		Rubber mats overtop of mass	

Test Apparatus		Moose Model	
Large mass support to replicate truck mass		Gel mats overtop of mas	
Pivot point (Horizontal)		Water tank	
Pivot point (Vertical)		Sand-bags	

Test Apparatus		Moose Model	
Vertical Tower		Shaped Plastic	
Vibrations test		Laser Cut Steel	

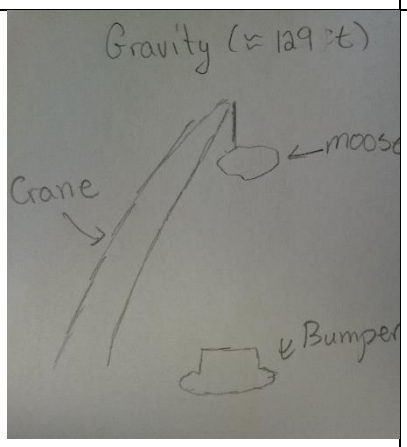
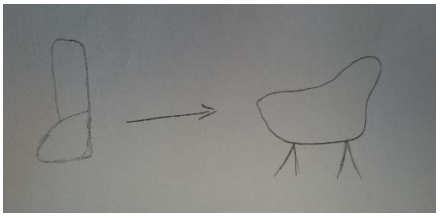
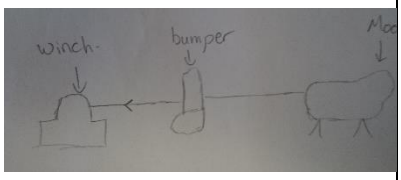
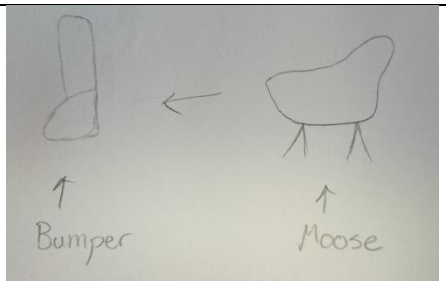
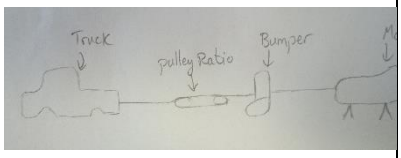
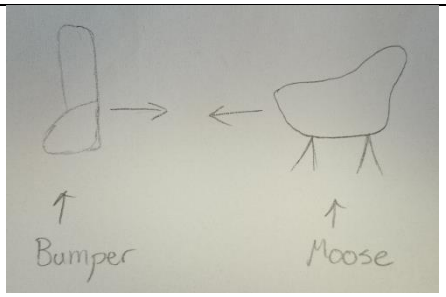
The support structure concept generation was used to find the best method to support the bumper. At this point we did not consider whether the bumper, the moose or both would be moving. Once we had developed a sufficient number of concepts we took the momentum equation into consideration.

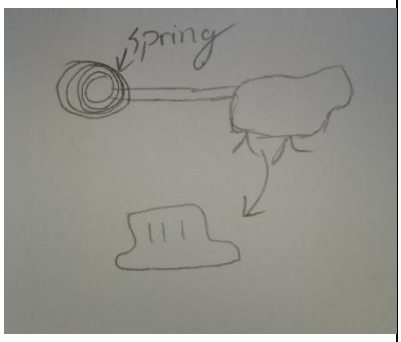
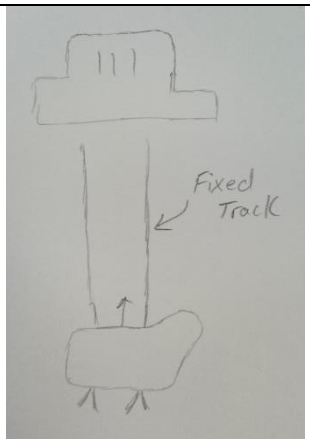
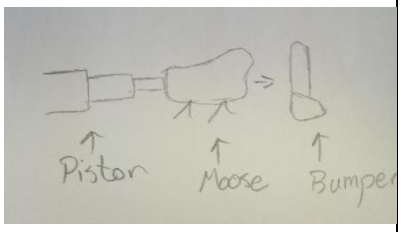
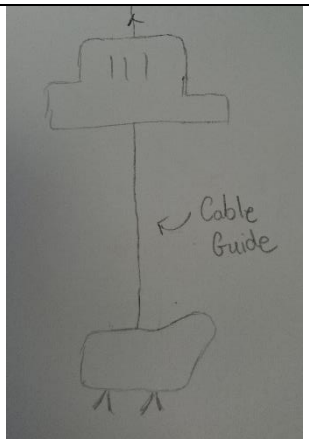
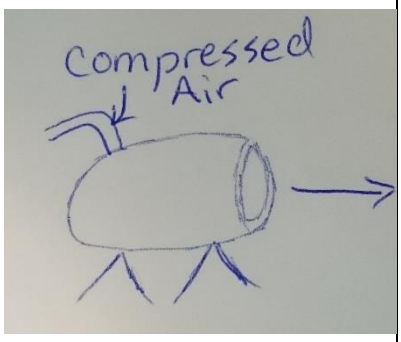
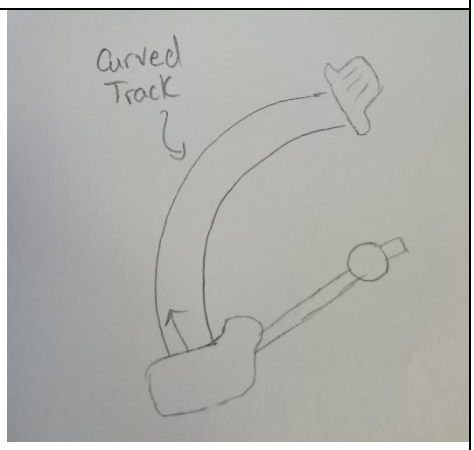
$$p = mv \tag{1}$$

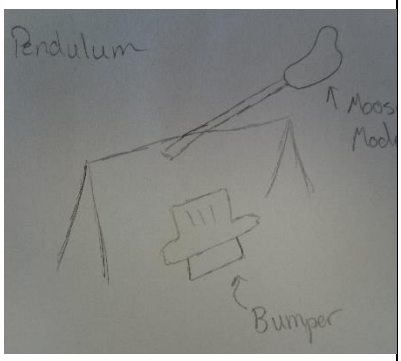
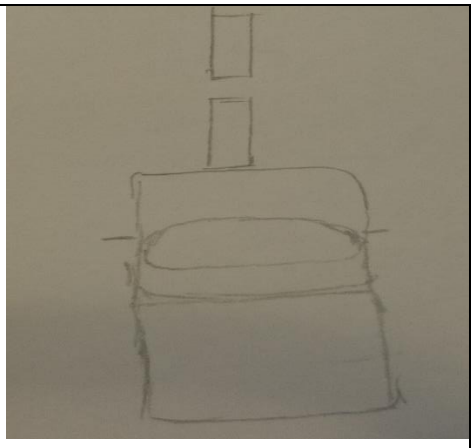
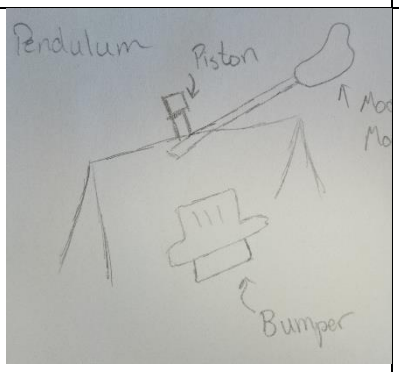
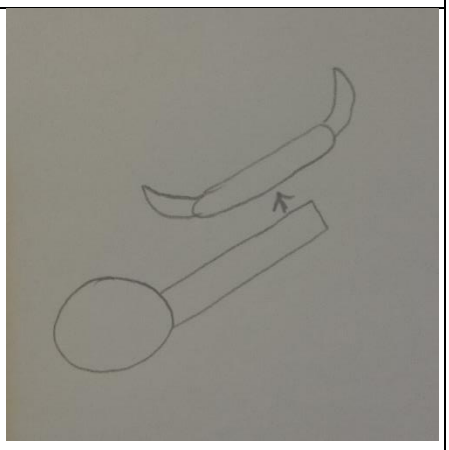
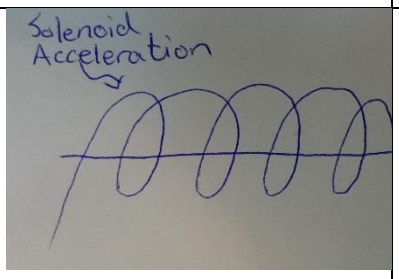
In the equation of momentum, the p is momentum, m is mass and v is speed. We will keep the momentum of our test procedure similar to that of a collision that a HERD bumper would experience on the road. The momentum of a fully loaded 8-axle truck (weighing 62,500kg [5]) would be difficult to recreate for this test. There would be advantages conserve momentum by moving the moose model and approximating the bumper as stationary. Selection of the best process to move the moose model will be completed in the concept selection section.

Our concept generation of moose models included many different ideas. There are advantages to simpler methods such as the old tire, sandbag and water methods due to their low manufacturing costs and simplicity. The VTI test moose and Mythbusters test moose are advantageous in regards to being the most life like during a collision scenario. However they would require high upfront costs and may not endure multiple test collisions.

TABLE II: POWER METHOD AND OBJECT MOVEMENT AND AIMING CONCEPT GENERATION [2]

Power Method		Object Movement and Aiming	
Gravity (vertical drop)		Bumper to Moose	
Cable winch		Moose to Bumper	
Truck pulling through pulleys		Moose and Bumper	

Spring system (Catapult)		Straight fixed track	
Power Method		**Object Movement and Aiming**	
Piston impulse		Guiding cable	
Compressed air (Air cannon)		Curved fixed track	

Pendulum (gravity only)		Painted guide line	
Power Method		**Object Movement and Aiming**	
Pendulum (gravity + another source)		Close range: pendulum, linear impact	
Solenoid acceleration		Laser	

Power method concept generation led to the creation of many diverse solutions. Advantages to the pendulum and gravity type solutions include allowing for high accuracy of collision speeds and location of impact between multiple tests. However, a disadvantage could be increased manufacturing required for the support structures. Comparatively, advantages of a truck powered systems would be its low cost. HERD

vehicles could be used for the tests and the manufacturing cost for the support structures would be cheaper. Drawbacks of the truck powered systems could come from the inability to recreate accurate test speeds as well as the roughness of the gravel test space could introduce unwanted test variables.

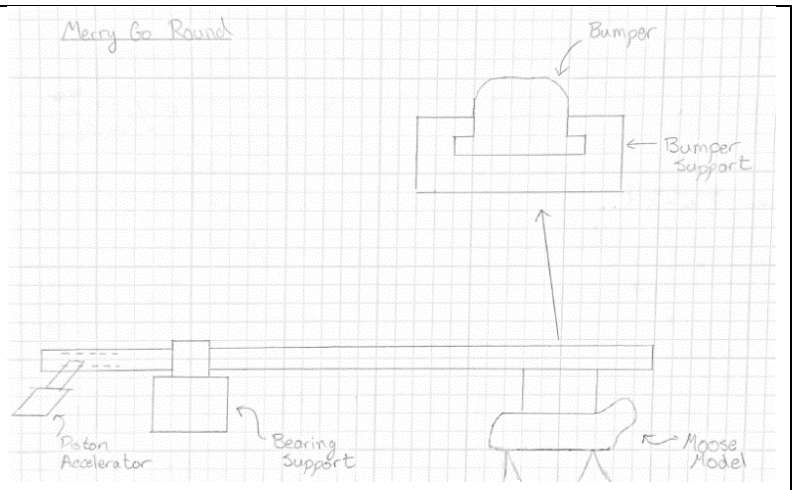
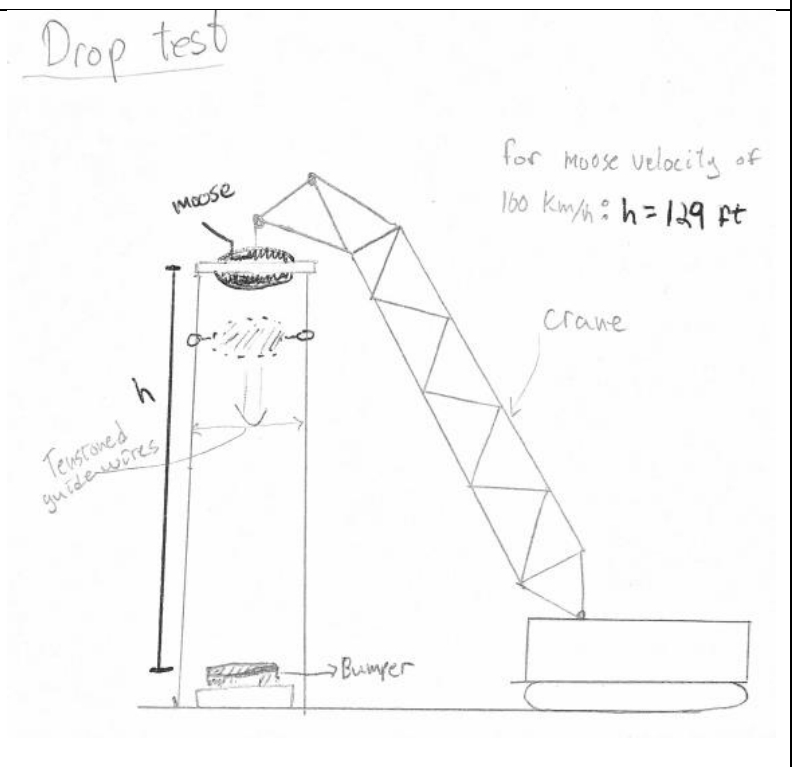
The “Object Movement and Aiming” concept generation was used to determine the ideal system for collision impact accuracy and test repetition accuracy. Fixed track type methods would allow for the most repetitive accuracy due to their robust designs however, they would require higher manufacturing costs and time. Laser guidance would allow a visual accuracy check for human operated machines but would be complex and costly to integrate into an automated system. Finally, moving both the moose model and the bumper would increase the number of variables in the system.

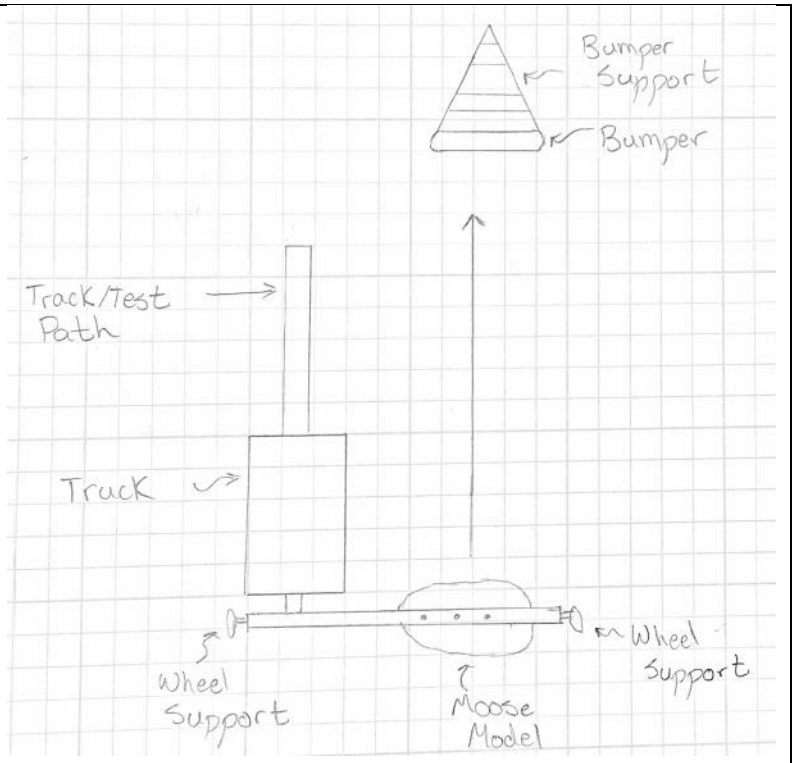
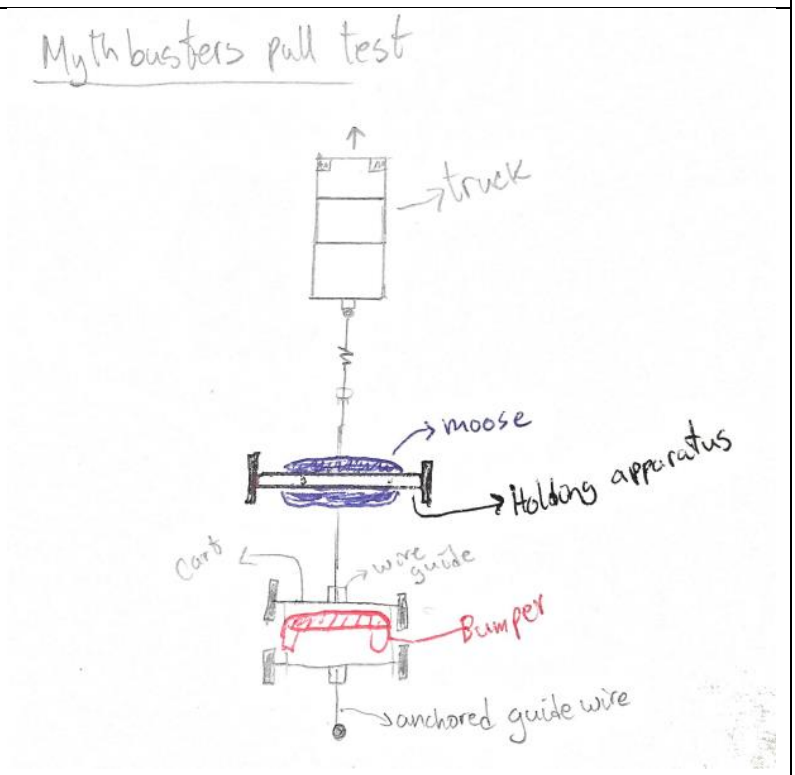
The concept generation section of our project led to a wide variety of solutions to complete the project objective outlined by the client. Many of these aspects still need to be combined and refined to be viable solutions of our project. However, from this concept generation we will be able to find the best solution.

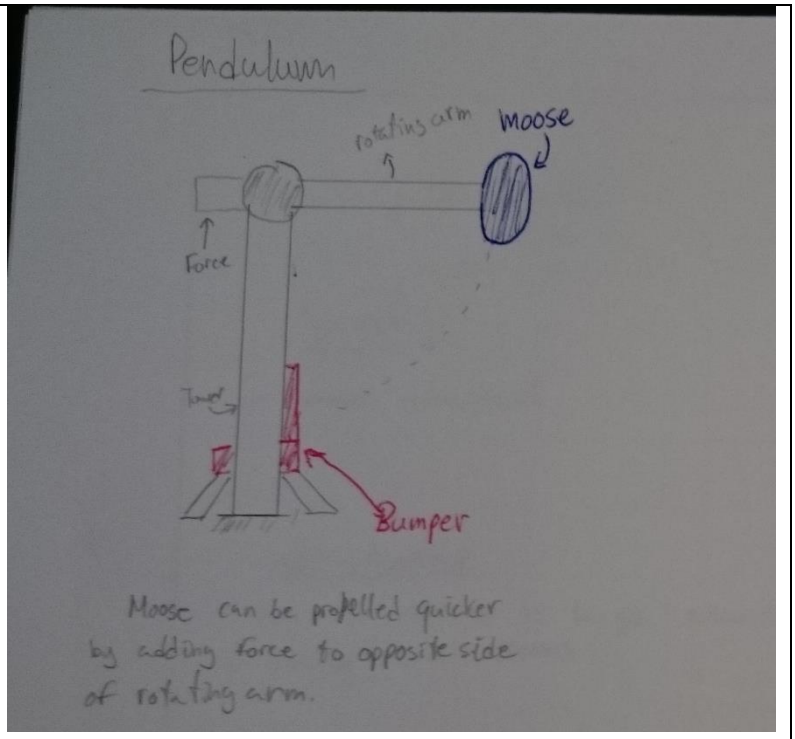
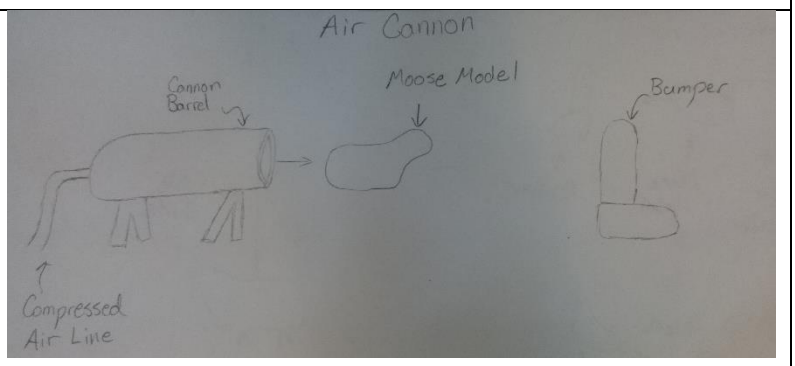
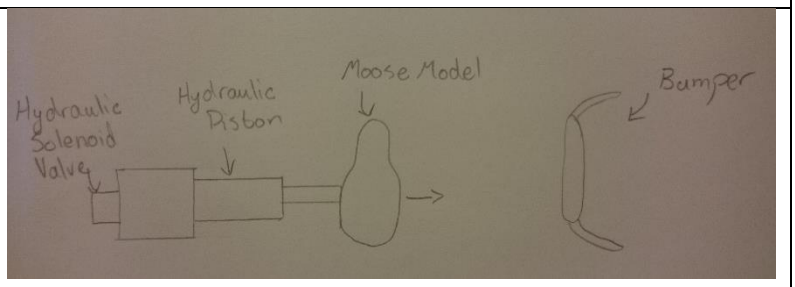
Group Concept Generation

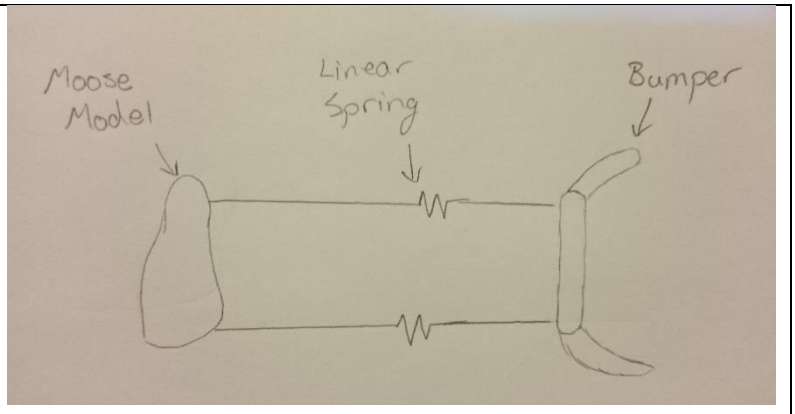
Once our team had sufficient concept generation for the various components of our design we began to combine ideas. From TABLE I and TABLE II there were many great concepts. With these concepts, we needed to determine the best method to combine them to meet the customer needs. TABLE III summarizes the combination of our concept generation. We utilized basic screening techniques to remove the generated concepts that would not fully solve the problem or would be unfeasible to create. With the remaining concepts, we strove to combine them in new ways that would best solve our objective.

TABLE III: CONCEPT GENERATION COMBINATION [2]

Merry Go Round: Horizontal beam with pivot point. Accelerated by piston? At one end. Moose dangling? From the other end.	
Drop Test: Use gravity to accelerate moose model from approximately 129 ft.	

Truck Hitch Apparatus: Tow structure behind a truck with support wheels? To speed. Moose model hanging? From structure.	
Myth busters Pull Test: Use a truck and pulleys to accelerate the bumper into the moose model. Truck pulling will replicate mass of semi-truck.	

Pendulum: Use gravity and external force to rotate the moose model into the bumper.	
Air Cannon	
Hydraulic Solenoid Valve	

Linear Spring Pull	

With the team concept generation combination complete we approached our client to continue building on and reviewing our ideas. Our team met with the client to review our initial concepts and receive feedback on our designs. From our concepts and the feedback from the client we were prepared to complete a more rigorous screening process.

Concept Selection

Our team was required to select one or two design concepts to pursue into the detailed design from the large selection of concepts that were developed during concept generation. This process needs to be systematic. Instead of choosing the design that is subjectively our favourite, we had to implement a process to objectively choose the superior design concepts. This process takes the work done in developing the project needs and develops criteria for judging generated concepts. However, it does not make sense to use all the project needs to judge all components in the design. Therefore, one of the first steps in the concept selection is to develop a set of needs specific to each component of the design. The next step is to adjudicate the strengths and weaknesses of all the designs with respect to the needs developed for each component with the intent of screening out the majority of the concepts at an early stage in the selection process. At the same time, all the needs for each concept are ranked against each other to provide objective priority levels for each need, and to provide weights for each need, to assist in a final concept scoring. The concept scoring is the penultimate step in the conceptual design, and it will give a score between one and four to each component in the design.

After the component conceptual scoring is complete, the final step is to take overall concepts developed in concept generation and apply the scores of each component present in that concept to determine the best overall concept. The concept with the highest score advances onto the detailed design. We will select a maximum of two overall concepts to continue after this phase. Figure 3 visualises the steps that the concept selection phase will cover.

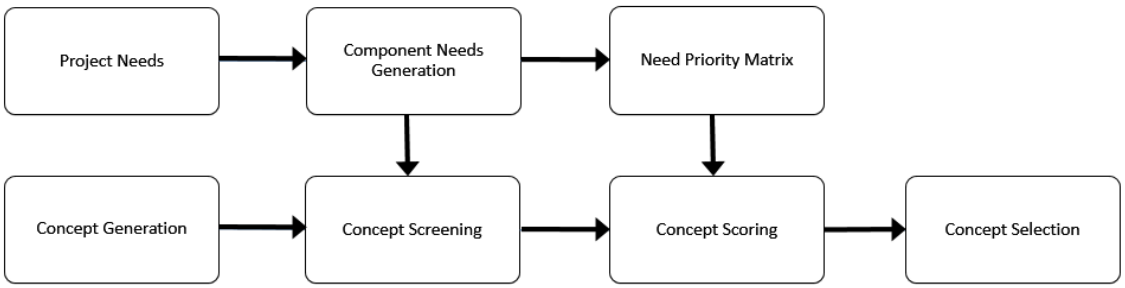


Figure 3: Process of concept selection

Determination of Dynamic and Static Collision Components

The first step in the whole concept selection is determining the moving component in the simulated crash. This needs to be done before the rest of the concept selection because the outcome of the moving component selection could render some concepts impractical or impossible. The two dynamic scenarios we are examining are:

1. Dummy in motion, with a stationary bumper.
2. Bumper in motion, with a stationary dummy.

The goals of this analysis are to determine the energy transfer required in an idealized collision between a truck and a moose, and apply this result to determine the velocity required for the dummy hitting a stationary bumper and the bumper hitting a stationary dummy.

Idealized Truck and Moose Collision

To simplify this collision scenario, a few assumptions were made. The first assumption is that the truck provides only enough force to overcome the energy losses occurred by driving at its instantaneous velocity. This means that the truck will provide enough force to perfectly counteract forces such as air resistance and friction, but will not apply a braking force or extra acceleration at any point during the collision. The second simplifying assumption is that the collision is perfectly inelastic, where the final velocity of the moose is equal to the final velocity of the truck. The third assumption is that the collision is between two infinitely small particles in a 1D plane. This assumption removes any energy or momentum transfer in dimensions that we are not concerned

about on this level, such as vertical or rotational. This also ensures that the two centres of gravity are colliding with each other.

The mass of the moose in this collision is assumed as slightly above the average weight of a deer, or 200 kg [6]. The mass of the truck is the maximum allowable truck weight in Manitoba, or 62,500 kg for an "RTCA B-Train" with 8 or more axles [5]. The initial speed of the truck is assumed to be 80 km/h (22.22 m/s), with the moose stationary. The momentum conservation equation is:

$$m_{1i}v_{1i} + m_{2i}v_{2i} = m_{1f}v_{1f} + m_{2f}v_{2f} \tag{2}$$

Where m is the mass in kg, v is the speed in m/s, terms denoted by the subscripts 1 represent the truck, 2 represent the moose, i represent initial conditions, and f represents the final conditions. To solve for the final velocity, the equation is rearranged as

$$\frac{m_{truck}v_{truck,i}}{m_{truck} + m_{moose}} = v_f \tag{3}$$

$$v_f = \frac{(62500kg) * (27.78m/s)}{(62500kg) + (800kg)} = 27.15m/s = 79.74km/h$$

With the final and initial velocities of the moose known, an energy calculation can be performed to determine the overall energy transfer between the truck and the moose in the collision. For this, we assume that the only energy transfer in the collision system is kinetic energy from the truck to the moose (i.e. no energy loss to friction, sound, heat generation, etc.). The equation of the energy transfer to the moose is as follows, where E is the energy in the moose in J, v_2 is the final velocity of the moose (in m/s) and v_1 is the initial moose velocity.

$$\Delta E = \frac{1}{2} * m * (v_2^2 - v_1^2) \tag{4}$$

$$\Delta E = 0.5 * 200kg * (22.22^2 m/s) = 49.07kJ$$

Therefore, a simulated collision must transfer 49.07 kJ between the dummy and the bumper to accurately simulate a collision with a truck.

Moose Dummy in Motion

The velocity calculations for a dummy in motion hitting a stationary bumper are the reverse (where v_2 is zero and ΔE = -49.07kJ) of the previous equation for a dummy with a mass of 800 kg if the bumper is fixed in place, meaning a dummy velocity of 79.74 km/h is required.

Test Bumper in Motion

The bumper provided to us by HERD has a mass of 146 kg. Reversing the energy equation (3), we can calculate for velocity (assuming a final velocity of zero).

$$\sqrt{\frac{\Delta E}{0.5 * m}} = v_1 = \sqrt{\frac{2 * 49.07kJ}{146kg}} = 25.93m/s \ = 93.33km/h$$

Therefore, a bumper velocity of 93.33 km/h is required to accurately simulate a collision. Since the required test velocity of moving the dummy to the fixed bumper is lower than the test velocity of moving the bumper to a fixed dummy, we will move the dummy towards a fixed dummy. In addition, as we are designing the dummy but not the bumper, we can increase the dummy weight and reduce the required collision speed for an equivalent collision while we cannot modify the bumper in this manner.

Needs of Various Test Components

While detailed needs and metrics were developed for the overall design in Section **Error! Reference source not found.** of the main body of this report, those needs aren't necessarily applicable to all individual components of the design. For example, the speed of the collision test has no effect on the design of the moose model, so speed does not need to be considered when choosing a moose model. There may be needs provided in Table I in the report proper that need to be explored in more detail for a component. An example of this is the durability of the moose model. If the moose model is destroyed after every collision, it needs to be serviceable so that the same design can

be repeated for multiple tests. Therefore, needs dealing with design durability are broken down into multiple needs. Finally, there may be needs that were not mentioned in Table I in the report proper that are relevant to making accurate and effective components, such as the elasticity of the dummy. Thus, needs were developed on a component level of the design with the intent of aiding the precision of concept selection.

TABLE IV to TABLE VII list the different detailed needs developed for the specified component; with a unique corresponding need number. They also list the corresponding need (if applicable) in Table I in the report proper.

TABLE IV: TESTING APPARATUS SPECIFIC NEEDS

Need No.	Corresponding Need No.	Corresponding Need	Detailed Need
T1	1	Compatability	Test apparatus is compatible with bumper provided by HERD
T2	2	Durability	Test apparatus survives the collision
T3	5, 9	Accuracy	Test apparatus allows for consistent collision location
T4			Test apparatus allows for vertical collision location movement
T5			Test apparatus allows for horizontal collision location movement
T6	7	Portability	Test apparatus can be broken down into parts movable by a forklift
T7	8	Speed	Test apparatus works for a variety of simulated collision speeds
T8	10	Weight	Test apparatus supports variety of weights of moose
T9	1		Test apparatus supports provided HERD bumper
T10	11	Observation	Test apparatus allows for close observation of test by video cameras
T11	15	Cost	Test apparatus can be produced in an inexpensive fashion

TABLE V: MOOSE DUMMY SPECIFIC NEEDS

Need No.	Corresponding Need No.	Corresponding Need	Detailed Need
D1	1	Compatibility	Dummy can be easily attached to apparatus
D2	2, 12	Durability	Dummy survives collision
D3			Long term, dummy design is sustainable
D4	7, 10	Weight	Dummy weight is adjustable
D5	15	Cost	Dummy can be produced and maintained at a low cost
D6	None	Dummy authenticity	Dummy density is similar to moose
D7			Dummy elasticity is similar to moose
D8			Dummy shape, when experiencing high force, is similar to moose at no acceleration

TABLE VI: POWER SYSTEM SPECIFIC NEEDS

Need No.	Corresponding Need No.	Corresponding Need	Detailed Need
P1	3	Consistency	Power system provides acceleration that is repeatable and predictable
P2	4	Safety	Power system can be operated remotely
P3	6, 8	Simulation Accuracy	Simulated speed provided by power system can easily reach 100km/h
P4			Simulated speed is adjustable
P5	7	Portability	Power system allows for easy mobility
P6	15	Cost	Power system would be cheap to produce or source
P7	None	Design compatability	Power system is compatable with different types of moose models and bumpers
P8		Space	Powering system does not require a large footprint
P9		Complexity	Power system is easy to design and manufacture

TABLE VII: AIMING MECHANISM SPECIFIC NEEDS

Need No.	Corresponding Need No.	Corresponding Need	Detailed Need
A1	2	Durability	Aiming mechanism is not damaged in collision
A2	3	Consistency	Aiming mechanism provides consistant collision location
A3	4	Safety	Aiming mechanism controls motion of moving component through acceleration and impact
A4	5, 9	Adjustability	Aiming mechanism allows for vertical impact location adjustments
A5			Aiming mechanism allows for horizontal impact location adjsutments
A6	7	Portability	Aiming mechanism can be easily transported by forklift
A7	8	Speed	Aiming mechanism works well for a variety of collision speeds
A8	10	Weight	Aiming mechanism works consistantly for variety of moose weights
A9	11	Observation	Aiming mechanism does not impede observation by video camera

Concept Screening

After the needs were developed for each component, we proceeded into the concept screening phase. In this stage, the concepts for each component were brought together and evaluated against each criterion as defined by the needs that were developed for each specific component. Concepts will be scored with a plus if the need under consideration can be considered an advantage of choosing that concept. If the criteria are neither an advantage nor disadvantage of the concept, then the concept is issued a 0. If the criterion is a disadvantage of the concept then a minus is issued. At the end, the pluses and minuses are totaled and the difference between the pluses and the minuses gives a net score for the design. The four concepts with the best net score advance into the concept selection phase. TABLE VIII to TABLE XII lists the results of the concept screening for each component.

At this point in the concept analysis, it was invaluable to split up the test apparatus into two components. One component is required to launch the dummy at the bumper – this is named the dynamic test apparatus. Another component is required

to hold the bumper and absorb the force from the collision – this is named the static bumper mount. Both these parts of the test apparatus can be judged on similar needs, and some concepts will overlap. The concepts evaluated for the dynamic test apparatus are:

- Vertical Tower
- Pendulum
- Swinging Arm
- Cannon Tube
- Truck Hitch Rig
- Cart with Wheels
- Cart with a Sled
- Crane

The concepts being considered for the static bumper mount are:

- Cart with Wheels
- Cart with a Sled
- Stationary Mount with a Large Mass
- Stationary Mount with a Small Mass
- Vibration Rig

TABLE VIII shows the concept screening for the dynamic test apparatus concepts. Need's T1 and T9 are irrelevant to the dynamic bumper mount and therefore every concept is given a same for these criteria. Using a crane to drop the dummy onto the bumper was ranked second after the concept screening. However, the cost of this solution is prohibitive to HERD, and after consulting with them we have decided to not pursue this concept any further. Therefore, the four concepts that will be evaluated in more detail in the concept selection are:

- Vertical Tower
- Pendulum

- Swinging Arm
- Cannon Tube

TABLE IX shows the concept screening for the static bumper mount. Need T8 is irrelevant to the static bumper mount and therefore all concepts were given the same score for this criterion. As there were fewer concepts for the static bumper mount and since the static bumper mount is independent of the overall concept selection (any bumper mount will work with any other combination of components) only three bumper mounts will be carried forward to concept selection. If the bumper mount is not independent of the final concept (i.e. some concepts didn't work with some overall conceptual designs) then four bumper mount concepts would need to be carried through to selection. Since this is not the case, we can only choose three to go through. This is convenient because three of the bumper mount concepts tied for the best net score in screening, and the other two concepts tied for the worst rank in the screening. The three concepts moving forward into concept selection are:

- Stationary Mount with Large Mass
- Stationary Mount with Small Mass
- Vibration Rig

TABLE VIII: DYNAMIC TEST APPARATUS CONECPT SCREENING

	Concept							
Need	**Vertical Tower**	**Pendulum**	**Swinging Arm**	**Cannon Tube**	**Truck Hitch Rig**	**Cart with Wheels**	**Cart with a Sled**	**Crane**
T1	0	0	0	0	0	0	0	0
T2	+	-	-	+	0	-	-	+
T3	+	+	+	0	-	-	-	+
T4	+	-	-	+	-	-	-	+
T5	+	+	+	+	0	0	0	+
T6	-	0	0	+	+	+	+	-
T7	+	+	+	0	+	+	+	+
T8	+	+	0	-	0	+	+	+
T9	0	0	0	0	0	0	0	0
T10	+	+	+	-	0	-	-	+
T11	0	+	+	+	+	+	+	-
PLUSES	7	6	5	5	3	4	4	7
SAMES	3	3	4	4	6	3	3	2
MINUSES	1	2	2	2	2	4	4	2
TOTAL	6	4	3	3	1	0	0	5
RANK	**1**	**3**	**4**	**4**	**6**	**7**	**7**	**2**

TABLE IX: STATIC BUMPER MOUNT CONCEPT SCREENING

	Concept				
Need	**Cart with Wheels**	**Cart with a Sled**	**Stationary Mount with Large Mass**	**Stationary Mount with Small Mass**	**Vibration Rig**
T1	+	+	+	+	+
T2	-	-	+	+	0
T3	-	-	+	+	+
T4	0	0	0	0	0
T5	+	+	-	-	-
T6	+	+	0	+	0
T7	-	-	+	0	+
T8	0	0	0	0	0
T9	+	+	+	+	+
T10	-	-	0	0	+
T11	0	0	+	+	+
PLUSES	4	4	6	6	6
SAMES	3	3	4	4	4
MINUSES	4	4	1	1	1
TOTAL	0	0	5	5	5
RANK	**4**	**4**	**1**	**1**	**1**

TABLE X shows the concept screening for the model used as a moose dummy. The needs developed for a moose dummy fall into two major categories: accuracy of the model and manufacturing ease of the model (cost, complexity). While there are two designs that are excellent replicas of moose (the VTI and Mythbusters moose), these dummies are prohibitively costly and complex for our purposes. On the other hand, a sandbag is very cheap and easy to produce and source, however it is not a realistic replica of a moose. These two extremes of concepts both landed low on our screening rankings because they are not a good balance of the two main requirements of the moose. The top four designs moving onto the concept selection phase for the moose dummy are:

- Sandbags inside Tires
- Water Container
- Rubber over a Mass
- Pure Plastic

TABLE XI shows the concept screening for the powering systems of the test. We produced five concepts ranked four or higher for this component because there was a tie for the fourth best concept. Since the power system chosen is partially dependant on other parts of the design, such as the dynamic bumper mount (it makes no sense to use a truck to power a dummy though a cannon tube), we must narrow this down to four and only four designs to give the final rankings equal weight across all dependant concepts. In this case, we can remove one of the concepts tied for fourth overall, the "Truck Pulling through Pulleys" concept, after consulting with HERD. They were concerned about the space required and the accuracy produced by using a truck as a power source, and seemed more enthusiastic about other power sources such as gravity. Therefore, the four concepts carried through to the concept selection are:

- Gravity
- Spring System
- Cable Winch

- Piston Impact

TABLE XII shows the concept screening for the aiming system. The top ranked concepts are:

- Cables in Tension
- Mechanical Aiming
- Straight Rails
- Curved Rails

However since the straight and curved rails are almost identical concepts, but the straight rails are superior, we will eliminate the curved rails at this point and use the Painted Line concept as the fourth concept in the aiming system concept screening.

TABLE X: MOOSE DUMMY CONCEPT SCREENING

	Concept								
Need	**VTI Design**	**Mythbusters Design**	**Sandbags in Tires**	**Rubber over mass**	**Gel over mass**	**Water container**	**Sandbag**	**Plastic**	**Steel**
D1	0	0	+	+	+	+	0	+	+
D2	+	+	+	+	+	0	0	+	+
D3	+	+	+	+	+	+	+	+	+
D4	-	-	+	0	0	+	+	-	-
D5	-	-	+	0	0	+	+	+	+
D6	0	0	0	-	-	0	-	-	-
D7	0	0	0	0	0	-	-	0	-
D8	+	+	+	+	0	+	-	+	+
PLUSES	3	3	6	4	3	5	3	5	5
SAMES	3	3	2	3	4	2	2	1	0
MINUSES	2	2	0	1	1	1	3	2	3
TOTAL	1	1	6	3	2	4	0	3	2
RANK	**7**	**7**	**1**	**3**	**5**	**2**	**9**	**3**	**5**

TABLE XI: POWER SYSTEM CONCEPT SCREENING

	Concept								
Need	**Gravity**	**Cable Winch**	**Truck Pulling Through Pullies**	**Truck Pulling without Pullies**	**Spring System**	**Piston Impulse**	**Compressed Air**	**Pendulum**	**Solunoid**
P1	+	0	0	0	+	+	-	+	0
P2	+	+	-	-	+	+	+	+	+
P3	0	+	+	+	0	0	-	-	+
P4	+	+	+	+	+	+	+	0	0
P5	+	+	+	+	+	+	0	0	0
P6	+	+	+	+	+	0	-	+	-
P7	+	+	+	+	+	+	-	0	-
P8	+	-	0	-	+	+	+	+	+
P9	0	+	+	+	0	-	-	0	-
PLUSES	7	7	6	6	7	6	3	4	3
SAMES	2	1	2	1	2	2	1	4	3
MINUSES	0	1	1	2	0	1	5	1	3
TOTAL	7	6	5	4	7	5	-2	3	0
RANK	**1**	**3**	**4**	**6**	**1**	**4**	**9**	**7**	**8**

TABLE XII: AIMING SYSTEM CONCEPT SCREENING

	Concept					
Need	**No Aiming System**	**Cables in Tension**	**Straight Rails**	**Painted Line**	**Mechanical**	**Curved Rails**
A1	+	+	0	+	-	0
A2	-	+	+	0	+	+
A3	-	+	+	-	+	+
A4	-	0	-	-	+	-
A5	-	+	-	-	+	-
A6	+	+	-	-	-	-
A7	-	+	+	+	+	0
A8	-	-	+	+	+	+
A9	+	+	+	+	-	+
PLUSES	3	7	5	4	6	4
SAMES	0	1	1	1	0	2
MINUSES	6	1	3	4	3	3
TOTAL	-3	6	2	0	3	1
RANK	**6**	**1**	**3**	**5**	**2**	**4**

Need Priority Level Determination

The following step in selecting a concept to pursue is to determine the priority levels of the needs of each component relative to each other. This is done by rating each need against the rest of the needs for each component and giving it a rating of 0, 1, or 2. If a 2 is issued to a need, it means that the given need is a higher design priority than the need it is compared against, and oppositely the compared against need is given a 0. If a need is assigned a 0 against another need, it means that the need given the 0 is less important to the design than the need that it is compared against, and oppositely the compared against need is given a 2. If a need is given a 1, it means that that need is about equal with the need it is being compared against, and the other need also receives a 1. A need cannot be compared against itself.

TABLE XIII to TABLE XVI list the need comparisons for each component and gives each need a final weight based on the priority levels of all the needs. The needs being evaluated are in the columns, while the needs used as comparisons are in the rows. At the bottom of each column, the total points for that need are summed and then divided by the total number of points issued to get a final weight. The final weight will then be used in the final component concept selection to determine the overall score of each concept.

TABLE XIII shows the need priority matrix for the test apparatus. It shows that the most important needs of the test apparatus are needs T1, T2, and T3, which are compatibility, durability, and consistency respectively. The least important needs for the test apparatus are T6 and T11, which are cost and portability of the test apparatus. This analysis shows that the building blocks of a successful test apparatus is the consistency and durability (compatibility will be dealt with as a design requirement in the detailed design) while portability and cost can be improved through optimizing the design at a later date.

TABLE XIV shows the need priority matrix for the needs of the moose dummy. It shows that needs D1 and D7 are the most important needs. These needs are the

compatibility of the dummy with the rest of the design and the elasticity of the dummy which affects the collision mechanics of the test. The lowest priority needs are D2 and D5, dealing with the one off durability of the dummy and the cost of the dummy. Durability of the dummy received a weight of 0, however it is kept in the list of needs because it had an effect in the concept screening stage. This analysis shows that we should focus our design efforts on having a dummy that is compatible with the apparatus and has a realistic elasticity of a moose, while optimising the cost.

TABLE XV shows the need priority matrix for the power method of the test. It shows that the highest priority needs are P1 and P4, which are the consistency and adjustability of the power provided to the system. The lowest priority needs include P3 and P6, which are the max speed of the test and the cost of the power system. The lack of a necessity to hit the target top speed comes from consultations with HERD. The relative damage to the bumpers is the most important outcome of the test and the damage can be easily extrapolated with respect to test speed. A test at 60 km/h would indicate the same points of failure for a given test bumper that would be apparent at 100 km/h.

TABLE XIII: TEST APPARATUS NEED PRORITIZATION AND WEIGHTING

	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11
T1	XXXXX	2	2	0	0	0	0	0	1	0	0
T2	0	XXXXX	0	0	0	0	0	0	0	0	0
T3	0	2	XXXXX	0	0	0	1	1	0	1	0
T4	2	2	2	XXXXX	2	0	2	2	1	2	0
T5	2	2	2	0	XXXXX	0	1	1	0	1	0
T6	2	2	2	2	2	XXXXX	2	2	1	2	1
T7	2	2	1	0	1	0	XXXXX	1	0	1	0
T8	2	2	1	0	1	0	1	XXXXX	1	1	0
T9	1	2	2	1	2	1	2	1	XXXXX	2	0
T10	2	2	1	0	1	0	1	1	0	XXXXX	0
T11	2	2	2	2	2	1	2	2	2	2	XXXXX
TOTAL	15	20	15	5	11	2	12	11	6	12	1
WEIGHT	**0.136364**	**0.181818**	**0.136364**	**0.045455**	**0.1**	**0.018182**	**0.109091**	**0.1**	**0.054545**	**0.109091**	**0.009091**

TABLE XIV: MOOSE DUMMY NEED PRIORITIZATION AND WEIGHTING

	D1	D2	D3	D4	D5	D6	D7	D8
D1	XXXXX	0	0	1	0	0	1	0
D2	2	XXXXX	2	2	2	2	2	2
D3	2	0	XXXXX	0	0	0	1	0
D4	1	0	2	XXXXX	0	0	2	1
D5	2	0	2	2	XXXXX	2	2	1
D6	2	0	2	2	0	XXXXX	2	1
D7	1	0	1	0	0	0	XXXXX	0
D8	2	0	2	1	1	1	2	XXXXX
TOTAL	12	0	11	8	3	5	12	5
WEIGHT	**0.214286**	**0**	**0.196429**	**0.142857**	**0.053571**	**0.089286**	**0.214286**	**0.089286**

TABLE XV: POWERING SYSTEM NEED PRORITIZATION AND WEIGHTING

	P1	P2	P3	P4	P5	P6	P7	P8	P9
P1	XXXXX	0	0	0	0	0	0	0	0
P2	2	XXXXX	0	1	0	0	1	1	0
P3	2	2	XXXXX	2	1	1	1	1	2
P4	2	1	0	XXXXX	0	0	1	0	0
P5	2	0	1	2	XXXXX	0	0	2	1
P6	2	2	1	2	2	XXXXX	1	2	1
P7	2	1	1	1	2	1	XXXXX	1	1
P8	2	1	1	2	0	0	1	XXXXX	1
P9	2	2	0	2	1	1	1	1	XXXXX
TOTAL	16	9	4	12	6	3	6	8	6
WEIGHT	**0.25**	**0.140625**	**0.0625**	**0.1875**	**0.09375**	**0.046875**	**0.09375**	**0.125**	**0.09375**

TABLE XVI: AIMING SYSTEM NEED PRIORITIZATION AND WEIGHTING

	A1	A2	A3	A4	A5	A6	A7	A8	A9
A1	XXXXX	2	2	0	0	1	0	0	1
A2	0	XXXXX	1	0	0	0	0	0	1
A3	0	1	XXXXX	0	0	0	0	0	1
A4	2	2	2	XXXXX	2	2	1	1	2
A5	2	2	2	0	XXXXX	2	1	1	1
A6	1	2	2	0	0	XXXXX	0	0	1
A7	2	2	2	1	1	2	XXXXX	0	1
A8	2	2	2	1	1	2	2	XXXXX	1
A9	1	1	1	0	1	1	1	1	XXXXX
TOTAL	10	14	14	2	5	10	5	3	9
WEIGHT	**0.138889**	**0.194444**	**0.194444**	**0.027778**	**0.069444**	**0.138889**	**0.069444**	**0.041667**	**0.125**

TABLE XVI shows the need priority matrix for the test's aiming system. From this matrix, we determined that the top needs of the aiming system are A2 and A3, which

are the consistency and safety given by the aiming system. The lowest priorities are A4, A7 and A8. These needs are the vertical height adjustments of the test, and the effects of the test variables (weight, speed) on the effectiveness of the aiming system. The vertical height adjustments are less important than the horizontal collision location adjustments because the test is designed to simulate an average moose. There is little range in the height of an average moose, while there is a range of locations that the bumper can contact a moose laterally.

Component Concept Scoring

This is the penultimate stage of the concept selection phase. It combines the need priority levels developed in section 0 and the four surviving concepts for each component from section 0. Each of the remaining concepts is ranked against the other three concepts, and a score between one and four is issued to the component concept, with a four being given to the best concept. There can be no ties between concepts; one must be the best with respect to each need. The scores of each concept for each criterion are multiplied by the weight determined for each need, and that becomes the final score of each concept for the specified need. The final scores are summed, and the highest score is determined to be the superior concept for each component. TABLE XVII to TABLE XXI show the detailed results of concept scoring for each component.

For this section, as for the concept screening, the test apparatus concepts will be split up into two sections: the dynamic test apparatus and the static bumper mount. At this stage, there are no overlapping concepts between the two components of the design. For the scoring, as for the screening, if a need is irrelevant to the component, all components will be assigned a score of zero for that criterion. This is the only time a zero will appear in the concept scoring. All other times a design will be assigned a score between one and four.

TABLE XVII shows the concept scoring for the dynamic test apparatus. The concepts were scored in T2 on the likelihood of the apparatus surviving the collision, with the apparatus' that are removed from the collision location receiving higher scores,

and the apparatus' that are part of the collision receiving lower scores. The Vertical Tower concept received a low score in T3 due to concerns that the freefall distance required to make a vertical tower test apparatus work could introduce turbulence that shifts the dummy slightly. This is not a concern at all in the mechanically aimed apparatus' such as the Pendulum and the Swinging Arm. T7 ranks the concepts on the perceived difficulty of adjusting the speed of the test with the given apparatus.

The top-ranking concept from the concept scoring is the Vertical tower, with a score of 2.5. Second place is the Swinging Arm, the Cannon Tube came in third, and the Pendulum was the worst of the four scored concepts.

TABLE XVII: DYNAMIC TEST APPARATUS CONCEPT SCORING

		Concept			
Need	**Weight**	**Vertical Tower**	**Pendulum**	**Swinging Arm**	**Cannon Tube**
T1	0.13636364	0	0	0	0
T2	0.18181818	4	2	1	3
T3	0.13636364	2	3	4	1
T4	0.04545455	4	1	2	3
T5	0.1	2	1	4	3
T6	0.01818182	1	2	3	4
T7	0.10909091	3	1	2	4
T8	0.1	3	4	2	1
T9	0.05454545	0	0	0	0
T10	0.10909091	4	2	3	1
T11	0.00909091	1	3	4	2
	TOTAL	2.47272727	1.709090909	2.054545455	1.85454545
	RANK	**1**	**4**	**2**	**3**

TABLE XVIII shows the concept scoring for the fixed bumper mount. Need T1 was assigned the maximum score for all concepts because it is a design requirement. In addition to this, all the needs were interpreted in the same way as for the dynamic test apparatus

TABLE XVIII: FIXED BUMPER MOUNT CONCEPT SCORING

		Concept		
Need	**Weight**	**Stationary Mount with Large Mass**	**Stationary Mount with Small Mass**	**Vibration Rig**
T1	0.13636364	3	3	3
T2	0.18181818	3	1	2
T3	0.13636364	2	1	3
T4	0.04545455	2	3	1
T5	0.1	2	3	1
T6	0.01818182	1	3	2
T7	0.10909091	3	1	2
T8	0.1	0	0	0
T9	0.05454545	2	1	3
T10	0.10909091	1	2	3
T11	0.00909091	1	2	3
	TOTAL	2.090909091	1.618181818	2.1
	RANK	**2**	**3**	**1**

The top ranking fixed bumper mount concept is the Vibration Rig concept with a score of 2.1, followed very close by the mount with a large mass, also scoring 2.1. These two designs therefore need further analysis to determine the superior design, and will both be taken into the detailed design.

TABLE XIX: MOOSE DUMMY CONCEPT SCORING

		Concept			
Need	**Weight**	**Sandbags in Tires**	**Water Container**	**Rubber over Mass**	**Pure Plastic**
D1	0.214286	4	2	3	1
D2	0	2	1	3	4
D3	0.196429	3	4	2	1
D4	0.142857	4	3	2	1
D5	0.053571	2	4	1	3
D6	0.089286	3	4	2	1
D7	0.214286	3	2	4	1
D8	0.089286	3	2	1	4
	TOTAL	3.303571429	2.821428571	2.5	1.375
	RANK	**1**	**2**	**3**	**4**

TABLE XIX shows the concept scoring process for the moose dummy design. For need D1, the pure plastic design scored low because it would need to be solid. This would make attaching it to anything with any sort of strength quite difficult. D2 was kept in for informational purposes only as it has no impact on the final design choice. D3 took two factors into account: the cost of the dummy design and the likelihood that the dummy is destroyed in a collision, and ranked the concepts against those criteria. Plastic scored low because it has relatively low fracture strength. Meanwhile, rubber designs received lower scores because of the high cost of rubber (or rubber tires) in the quantity needed when compared to the cost of plastic. Likewise, for D5, the more expensive rubber designs scored lower. D6 uses the fact that about three quarters of a moose's body weight is water [7] to rank the water container as the closest to the density of the moose. The plastic scores the worst because most plastics are much less dense than the water benchmark. D8 deals with the shape of the moose dummy when experiencing high acceleration forces matching that of a moose standing still on a road. Since plastic is quite rigid, it scores quite high, while the rubber over mass design is not meant to be shaped like a moose and therefore gets a lower score.

Overall, it was determined that the "Sandbags in Tires" concept was superior, with a score of 3.3. A relatively second was the water container, with the rubber over mass in third and the pure plastic model being the last place concept.

TABLE XX shows the concept scoring process for the powering method concepts. P1 proved difficult to rank the designs. We determined that there may be some losses due to air resistance with a gravity powered design, and some frictional losses using springs, but these losses are small compared to the losses using a winch or a piston. The higher the losses results in higher unpredictability, thus the lowering the score. P3 ranked the powering systems on how easy it would be to use the concepts to hit the target speed. P4 ranked the concepts on how easy it would be to adjust the target speed. Gravity received a low mark in P5 because the structure required to induce freefall would not be very portable. P6 notes that gravity is free to use, while springs strong enough to induce the required acceleration are not common and therefore likely

more expensive. P7 acknowledges that anything will have gravity acting on it, while other designs will require modifications to the moose model.

After scoring the power systems, it was determined that the Spring System would be superior, with a score of 3.4. Gravity was a really close second, also scoring 3.4, while the piston and cable winch rounded out the designs. Our team noted that it would be possible to combine gravity and spring power, and that this may create the superior design.

TABLE XX: POWER SYSTEM CONCEPT SCORING

		Concept			
Need	**Weight**	**Gravity**	**Springs**	**Cable Winch**	**Piston Impulse**
P1	0.25	4	3	2	1
P2	0.140625	4	2	1	3
P3	0.0625	2	4	1	3
P4	0.1875	2	4	1	3
P5	0.09375	1	4	2	3
P6	0.046875	4	2	3	1
P7	0.09375	4	2	3	1
P8	0.125	4	3	1	2
P9	0.09375	2	4	3	1
	TOTAL	3.40625	3.4375	1.90625	2.1875
	RANK	**2**	**1**	**4**	**3**

TABLE XXI shows the scoring matrix for the aiming systems considered. Need A1 was interpreted as judging the risk of damage occurring during the collision. A3 gave the cables a low score because any slack in the cables could allow swaying to occur, reducing the control of the dummy. However, A4 and A5 note that cables provide the easiest mechanism to adjust the aim of the dummy with. A6 notes that the painted line is not mobile at all, however mobility would not be an issue for the painted line as it does not need to be stored. Therefore the painted line receives the lowest grade on this, but with an asterisk, noting that it may not be relevant to this concept.

TABLE XXI: AIMING SYSTEM CONCEPT SCORING

		Concept			
Need	**Weight**	**Cables in Tension**	**Mechanical**	**Straight Rails**	**Painted Line**
A1	0.138889	3	1	2	4
A2	0.194444	2	4	3	1
A3	0.194444	2	4	3	1
A4	0.027778	4	2	3	1
A5	0.069444	4	3	2	1
A6	0.138889	4	3	2	1
A7	0.069444	1	2	3	4
A8	0.041667	2	1	3	4
A9	0.125	3	1	2	4
	TOTAL	2.666666667	2.680555556	2.527777778	2.125
	RANK	**2**	**1**	**3**	**4**

After scoring the aiming systems, we concluded that the Mechanical aiming is the best concept, with a score of 2.7, however the Cables in Tension is a very close second place. Straight Rails is in third, with a painted line coming in last, however all these concepts had similar scores.

Final Concept Scoring

The final step in determining the best overall component is to combine all the component concepts into overall designs, mixing and matching components until the best design is discovered. Eight concept designs were developed in the concept generation section. These designs can be broken down into components, and the scores of the components in these designs summed and compared against the other concepts to determine the best design. If a component concept was eliminated in the concept screening, it is given a score of zero in this analysis. In addition to the eight concepts developed, the best concepts from each component can be combined to create new overall concepts to be evaluated. Note that the static bumper mount is always independent of the remainder of the design and will not be considered in this analysis.

The first concept is named the “Merry-go-Round”. Its components are as follows, with the component score in brackets after the component.

- Power System
    - Spring (3.44)
    - Winch (1.91)
    - Piston (2.19)
- Aiming System: Mechanical (2.68)
- Dummy Design: Any. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Swinging Arm (2.05)

The power system components are relatively independent; there is no simple way to combine them into a superior design. The overall concept score is 11.47.

The second concept is named the “Drop Test”. Its components are as follows, with the component score in brackets after the component.

- Power System
    - Gravity (3.41)
    - Gravity plus Piston (around 3.41)
    - Gravity plus Springs (above 3.41)
- Aiming System: Cables in Tension (2.67)
- Dummy Design: Any. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Vertical Tower (2.47)

The power system can be gravity or gravity plus a combination of springs or a piston. As it is tough to quantify what the combination of concepts will do to the design, a score is assigned as the sum of 0.6*(score of each component). For the gravity plus piston, this gives a score of 3.36. A combination of gravity and springs gives a score of 4.11. This power plus the rest of the components gives a score of 12.55.

The third concept is named the “Truck Hitch Test”. Its components are as follows, with the component score in brackets after the component.

- Power System: Truck without Pullies (0)
- Aiming System: Painted Line Guide (2.12)

- Dummy Design: Any. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Truck Hitch Apparatus (0)

This concept gives an overall score of 5.42.

The fourth concept is named the “Mythbusters Test”. Its components are as follows, with the component score in brackets after the component.

- Power System: Truck with Pully Power (0)
- Aiming System: Painted Line (2.12)
- Dummy Design: Any, but prefer Mythbusters Dummy (0)
- Apparatus: Cart with Wheels (0)

This concept gives an overall score of 2.12.

The fifth concept is named the “Pendulum”. Its components are as follows, with the component score in brackets after the component.

- Power System: Gravity and Piston (3.37)
- Aiming System: Mechanical (2.68)
- Dummy Design: Any, except for water container. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Pendulum Apparatus (1.71)

The power score for gravity plus a pendulum for this concept were determined looking at the power for the drop test. This concept has a combined score of 11.06.

The sixth concept is named the “Air Cannon”. Its components are as follows, with the component score in brackets after the component.

- Power System: Compressed Air (0)
- Aiming System: Mechanical (2.68)
- Dummy Design: Pure plastic (1.38)
- Apparatus: Cannon Tube (1.85)

This concept gives an overall score of 5.91.

The seventh concept is named the “Hydraulic Solenoid Valve”. Its components are as follows, with the component score in brackets after the component.

- Power System: Solenoid and Piston (1.31)
- Aiming System: Mechanical (2.68)
- Dummy Design: Any. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Cart with Wheels (0)

The power for this concept was scored using the combination method explained looking at the “Drop Test” concept, with a solenoid score of 0 and a piston score of 2.19. This concept gives an overall score of 7.29.

The final concept is named the “Linear Spring Pull”. Its components are as follows, with the component score in brackets after the component.

- Power System: Spring Power (3.44)
- Aiming System: Mechanical (2.68)
- Dummy Design: Any. Highest scoring is Sandbag in Tires (3.30)
- Apparatus: Cart with Wheels (0)

This concept gives an overall score of 9.42.

The final concept to consider is a combination of the best concepts on a component level. These concepts are:

- Power System: Springs (3.44)
- Aiming System: Mechanical (2.68)
- Dummy Design: Sandbag with Tires (3.30)
- Apparatus: Vertical Tower (2.47)

However, the mechanical aiming system is incompatible with the vertical tower, as the collision is not a close range collision. Therefore, the next best aiming system must be used, which is the “Cables in Tension” with an overall score of 2.67. At this point, it can be seen that this design is the same as the “Drop Test”, with the exception

of the power system being the combination of gravity and a spring with a score of 4.11. Therefore, the best possible design is the Drop Test. TABLE XXII summarizes the concepts and their scores.

TABLE XXII: SUMMARY OF CONCEPT SCORING

Concept Name	Concept Score
Merry-go-Round	11.47
Drop Test	12.55
Truck Hitch test	5.42
Mythbusters Test	2.12
Pendulum	11.06
Air Cannon	5.91
Hydraulic Solenoid	7.29
Linear Spring Pull	9.42

From this analysis we conclude that the best design is the “Drop Test”. This test gives the best possible overall concept selection score. Therefore, we will continue onto a detailed design with the “Drop Test”. However, the second highest score belongs to the “Merry-go-Round” design. This concept will also be carried into the detailed design, with a final decision on overall design coming after preliminary calculations are done on both designs.

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