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MECH 4860 – Engineering Design
Final Design Report

WESTCARD - ENGINE TEST CALIBRATION FIXTURE AND PROCEDURE DESIGN

Team 15: UMCal Consulting Group

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Date Submitted: December 2, 2013

[redacted]

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December 2, 2013

WestCaRD

Winnipeg, Manitoba,
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Re: Engine Test Calibration Fixture and Procedure Design

To Mr. Baillie,

Our design team has completed phase three of the *Engine Test Calibration Fixture and Procedure Design* project. The final design report is attached for you to review.

The report includes our documented final design for the redesign of an Engine Test Calibration Fixture and Procedure. Also included are concept designs that were considered, a detailed analysis of the final design, a cost summary and payback period, and a procedure for the operational use of the final design.

If you have any issues or concerns with the final design report or any of the information presented within it, please feel free to contact our project manager, Dylan Guenette at or by e-mail at

Thank you for your time and consideration,

Dylan Guenette
Eric Hornby
Lynne Kincaid
Heather McCrea

Signature of Approval: _________________________ Date Signed: ______________________

Attached: Final Design Report

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Executive Summary

The thrust calibration system used at the General Electric Test, Research and Development Centre in Winnipeg, MB has been identified as costly and inconvenient. UMCal has been tasked with the creation of a new calibration design and procedure that will eliminate the current need to import specialized contracted equipment and personnel from the United States. The new system must maintain the current standards utilized by GE, which include, but are not limited to: accuracy, repeatability, and precision. In addition to these standards, the client has also requested that the design provide a payback period of no greater than five years.

A total of three concepts for the engine thrust simulation and total of two concepts for the engine weight simulation were considered. Of the considered concepts, a single concept for each of the thrust simulation and engine weight simulation were selected. The two concepts brought forth to the Final Design Generation phase are the simple lever concept for the thrust simulation and the turnbuckle concept for the engine weight simulation.

The final design consists of three components; engine thrust simulation, engine weight simulation, and calibration pylon. A thorough stress and cost analysis was performed on each component, and used to determine the feasibility of the conceptual designs. To accompany the conceptual design, a calibration procedure document outlining the required tasks and set up steps needed to perform the calibration testing using the new equipment is provided. In addition, CAD models and the accompanying system drawings are also provided.

Upon completion of the analyses, UMCal successfully designed a calibration system capable of meeting the design criteria set by the client. However, UMCal was unable to meet to the client's desired payback period of five years within the given budget. UMCal recommends that the current calibration equipment continue to be contracted until further details regarding cost of manufacturing can be accurately predicted, or the frequency of calibration at the GE facility increases.

1. Introduction

An aircraft's engines are critical to its operation, as they provide power for electricity and lift generation. Performance testing of gas turbine engines is therefore essential to ensuring the safe passage of air travelers worldwide. This report presents the final design for an engine thrust calibration system and procedure, to be used at the General Electric Testing, Research and Development Centre (hereafter referred to as "GE TRDC") to ensure validated engine performance testing at the facility. This project is sponsored by WestCaRD, a not-for-profit organization using the facility for learning and development purposes. UMCal is a student-based consulting group which has been challenged with completing this project.

1.1 Project Background

General Electric Aviation (hereafter referred to as "GE") owns a cold-weather gas turbine engine testing facility on the grounds of the Winnipeg International Airport. The daily operation of this facility is overseen by Standard Aero employees who conduct a variety of tests on new GE engines as part of the performance verification process. The major component of this facility is a thrust stand and frame, to which an engine is attached and run to gather data on its thrust performance. A labeled picture of this facility is shown in Figure 1 [1].

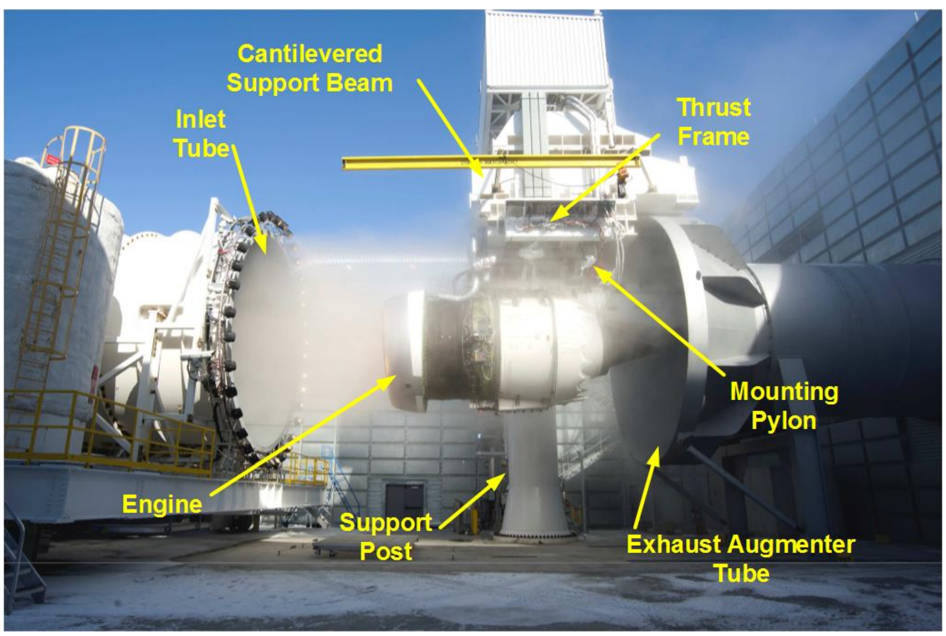

Figure 1: Schematic drawing of thrust stand at GE engine test facility [1].

In order to ensure the accuracy of the data gathered at this facility, the thrust stand must first be calibrated using known input values. It is imperative that the measurement equipment being used is calibrated to specific industry and GE standards. Figure 2 shows the underside of the thrust frame. Labeled in this figure are the load cells that require calibration and the mounting locations that the proposed design must interface with.

[redacted]

Figure 2: Thrust frame.

The current method of thrust stand calibration used at the GE TRDC requires specialized equipment and personnel to be contracted from the United States, and is therefore costly and time-consuming. For this reason, WestCaRD has sponsored this project which challenges UMCal to develop an in-house alternative to the current calibration method. The client has requested the design of calibration equipment, and a procedure for its use, that will allow Standard Aero to perform the required calibration using only internal resources. [redacted]

[redacted] To reduce costs, the client has also requested that an existing A-frame be incorporated into the design if at all possible. This A-frame is part of the current calibration equipment, and is shown in Figure 3.

Figure 3: A-frame used in current calibration system.

1.2 Problem Objectives and Deliverables

The main purpose of this project is to design an in-house thrust calibration system for the GE TRDC facility. This design must accurately simulate the effects of a running engine on the facility's thrust frame. To accomplish this, the system must simulate engine thrust up to a maximum specified value, as well as simulate the weight of the engine on the frame.

The specific objectives and deliverables of this project were developed through open communication between the client and UMCal. A list of defined objectives and deliverables are shown in TABLE I.

TABLE I: PROJECT OBJECTIVES AND DELIVERABLES

	Project Objectives
1	The design will allow the client to perform in-house calibrations.
2	The design will produce calibration curves similar to those created by the current equipment.
3	The design will meet the same tolerances for accuracy, reliability and precision as outlined in the GE testing procedure.
4	The design will be less expensive than the currently utilized equipment.

	Project Deliverables
1	Detailed CAD model of calibration fixture
2	System drawings
3	Bill of materials
4	Cost estimation
5	Calibration procedure document
6	Project poster
7	Final report
8	Final presentation

Overall, the objectives of this project are to create a design that will integrate seamlessly into the current system, meet required standards for accuracy, repeatability and precision, and will do so at a low cost to the client.

Deliverables one through five in TABLE I are specific project deliverables requested by the client. Deliverables six through eight are requirements set by the MECH 4860 course. UMCal has worked tirelessly in order to develop a design which meets the stated objectives and provide all identified deliverables.

1.3 Project Needs and Specifications

The project needs and specifications were determined by UMCal after an initial meeting with the GE and Standard Aero representatives at the GE TRDC facility [3]. The identification of these needs will ensure that the final proposed design fulfills all client expectations. The importance of each need, ranked on a scale from 1 to 5, has been established by the team and verified by the client. On this scale, 1 represents a need of little importance and 5 indicates an essential need. These rankings proved useful to the team when weighting the importance of the evaluation criteria during the design selection phase. A summary of the needs identified by UMCal are shown in TABLE II.

TABLE II: PROJECT NEEDS AS IDENTIFIED BY UMCAL

#	Needs		Importance
1	The thrust calibration equipment	accurately simulates engine thrust	5
2	The thrust calibration equipment	is capable of testing entire range of engine thrust	5
3	The thrust calibration equipment	allows for small thrust increments	5
4	The thrust calibration equipment	accounts for engine mass	5
5	The thrust calibration equipment	provides suitable outputs to create correction curve	5
6	The thrust calibration equipment	accurately reads thrust simulation load	5
7	The thrust calibration equipment	produces repeatable results	5
8	The thrust calibration equipment	will have minimal hysteresis error	5
9	The thrust calibration equipment	is cost competitive with current calibration method	4
10	The thrust calibration equipment	allows for quick calibration procedure	4
11	The thrust calibration equipment	is easy to set up and tear-down	4
12	The thrust calibration equipment	is compatible with current testing system	5
13	The thrust calibration equipment	is easily upgradeable	4
14	The thrust calibration equipment	lasts a long time	4
15	The thrust calibration equipment	is easily stored	2

Due to the high tolerances typically required within the aerospace industry, all items related to accuracy and repeatability were given the highest importance ranking. GE could not be confident in the performance of its engines without accurate and repeatable test results, and consequently, tested engines would not receive the required certification. Needs related to system integration were also assigned a ranking of five, as the client indicated it is essential for the new calibration to integrate into their testing system seamlessly.

From the needs identified in TABLE II, metrics were developed to establish quantifiable measurements for target specification. Numerical target values have been generated for each specification [redacted] and are shown in TABLE III.

TABLE III: METRICS AND TARGET SPECIFICATIONS

Metric #	Need #	Metric	Unit	Marginal value	Ideal value
1	1	Distance from thrust simulation application point to engine centerline [3].	in	0	0
2	1	Consistency of the thrust simulation load.	% FS	Pk-Pk: 0.1	0
3	2	Maximum producible load of thrust simulation method [3].	klbf	85 -105	105
4	2	Operating range of thrust simulation measuring instrumentation [3].	klbf	4000 – 85	0 - 105
5	3	Step size capability of thrust simulation method [4].	lb	≤ 100	≤ 100
6	4	Engine weight simulation [3].	lb	9,900 - 10,100	10,000
7	5	Output range of thrust simulation measuring instrumentation [5].	V	0 - 10	0 - 10
8	6	Specified sampling rate of thrust simulation measuring instrumentation [5].	Hz	10	10 - 100
9	6	Specified accuracy of thrust simulation measuring instrumentation [2].	% FS	± 0.05	0
10	7	Specified repeatability of thrust simulation measuring instrumentation [2].	% FS	± 0.05	0
11	8	Specified hysteresis of thrust simulation measuring instrumentation [2].	% FS	± 0.1	0
12	9	Payback period of implementing new calibration method	Yrs	3 - 5	3
13	10	Time to perform calibration [6].	hrs	< 3	2
14	11	Time to set up calibration equipment [6].	hrs	< 8	5
15	11	Time to tear down calibration equipment [6].	hrs	< 8	5
16	12	Physical interfaces to current testing system	List	Lifting chain attachment points	
				Pin locations	
				Thrust frame clearance	
				Measurement-instrumentation connections	
17	13	Maximum allowable load on design structures [3].	klbf	85 - 105	105
18	14	Equipment life span	Yrs	20 - 30	30
19	15	Indoor storage space requirements	m^3	10	0

The current system used at the GE TRDC facility calibrates thrust up to a maximum value of 85,000 $[lbf]$. This range is sufficient for current engines tested at the facility and has been specified by the client as the minimum thrust range requirement for this design.

However, the client has also projected [redacted] may require calibration up to $105{,}000\ [lbf]$ of thrust. In order to reduce costs in future facility upgrades, UMCal will be designing a system capable of a $105{,}000\ [lbf]$ thrust load for all design components except the calibration load cell.

UMCal was unable to create a project budget from the specified payback period alone. [redacted]

2. Design Options Considered

The project began with the project definition phase where all deliverables, objectives, needs and specifications were identified. Following this was the concept selection phase, where the goal was to create, evaluate and refine as many concepts as possible until the final concept was chosen and brought forward into the design phase of the project.

During the concept selection phase, the team completed a weighted evaluation of promising design concepts for both the thrust and weight simulations. To do this, UMCal developed a list of design criteria and assigned each a weighted percentage based on its importance. Each concept was ranked on its ability to meet the criteria and assigned an overall score based on the sum of all weighted criteria rankings. The concepts that scored highest were expected to best meet the design needs and specifications while remaining cost competitive with the current calibration method. The list of criteria used to determine the overall concept score is shown in TABLE IV.

TABLE IV: CONCEPT SCORING CRITERIA

Criteria	Reason for Criteria
Ease of Assembly	Client would like to reduce overall time required for the setup of equipment before beginning the calibration.
Cost	Client has expressed that current equipment is expensive and would prefer that a new design be as cost effective as possible. Client has asked that new equipment provide a payback period of 3-5 years.
Ease of Use	Important that once equipment has been setup, the calibration is performed with no difficulties as client has minimal time to spend preparing for testing.
System Accuracy	System must be capable of incorporating measuring equipment. It is important that the system is accurate as the engines are generally being tested for the first time. Design must ensure that there are no losses throughout system (e.g. through linkages)
Safety	Safety of the system is paramount. The design must function so that there is no chance of any individual being at risk.
Lifespan	Client has expressed the desire for new design to have a significant lifespan. Team has set goal of at least 20 years.
System integration	The new design must be capable of integrating with the current interface, as well the current mounting locations.

Criteria	Reason for Criteria
Load Sensitivity	The design is expected to be capable of providing appropriate step size intervals for the loadings required.
Required load input	The design must be able to provide the require load input range as stated by the client.
Storage	The design should be easily stored away when not in use.
Low Maintenance	Client would prefer if minimal maintenance is required in the up keep of the equipment.
Ease of upgrades	Client has expressed interest in the idea of the design being easily upgradable for testing of more powerful engines in the future

Potential final design concepts brought forward from the concept generation phase were selected based on the weighted evaluation scores in conjunction with client feedback.

The client expressed interest in three concepts for the thrust simulation and requested further pursuance for each. The three concepts for thrust simulation brought forward are shown in Figures 4, 5 and 6, and described below.

Simple Lever Thrust Simulation Concept

The simple lever concept uses a ball screw that is mounted at ground level and actuated in the horizontal direction. A single beam utilizes a fulcrum mechanical advantage to transfer the ball screw input load to the engine thrust line. The simple lever concept is shown in Figure 4.

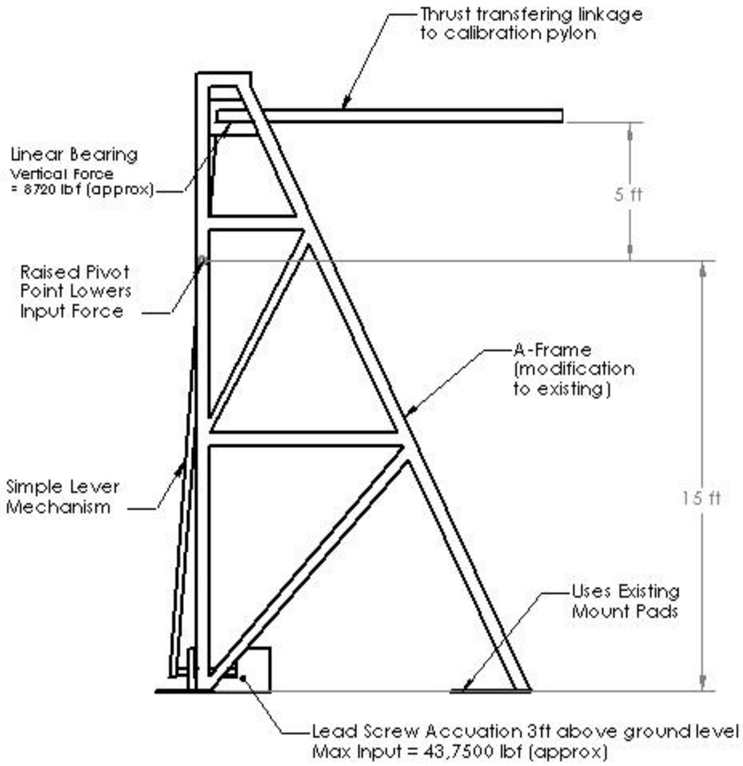

Figure 4: Simple lever thrust simulation.

Modified Simple Press Thrust Simulation Concept

The modified simple press concept utilizes a ball screw that is attached to a hollow threaded rod. As the ball screw is extended, the two attached linkages will move up and outwards. The motion of the ball screw is transferred to the upper linkage and through the rigid frame. As the upper linkage is pushed outwards, it creates a pulling motion through the connection between the thrust frame and linkages. The pulling motion will serve to simulate the thrust of an engine. This concept is a modification of a simple press mechanism. The modified simple press concept is shown in Figure 5.

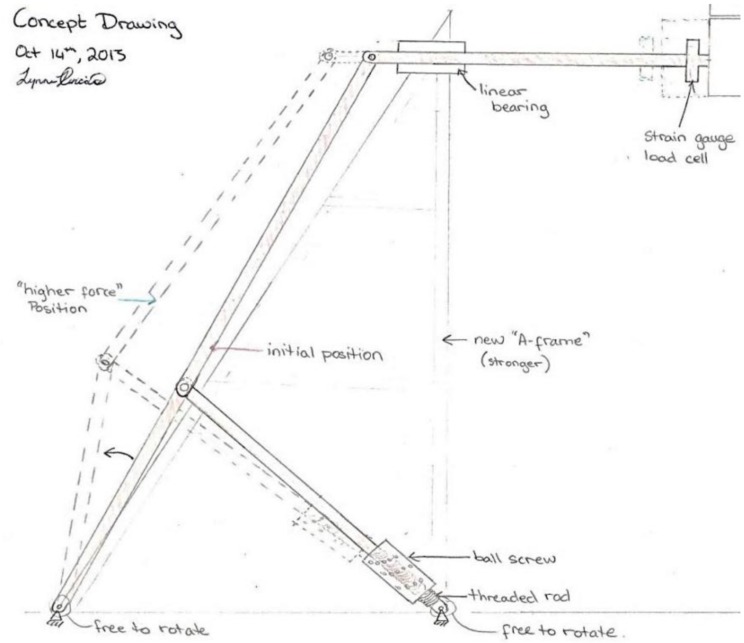

Figure 5: Modified simple press thrust simulation.

Reference Thrust Simulation Concept

The reference concept uses an A-Frame with a ball screw mounted horizontally in line with the engine thrust line. As the length of the ball screw is shortened, the mounting pylon is pulled towards the A-Frame, simulating engine thrust. The reference concept is shown in Figure 6.

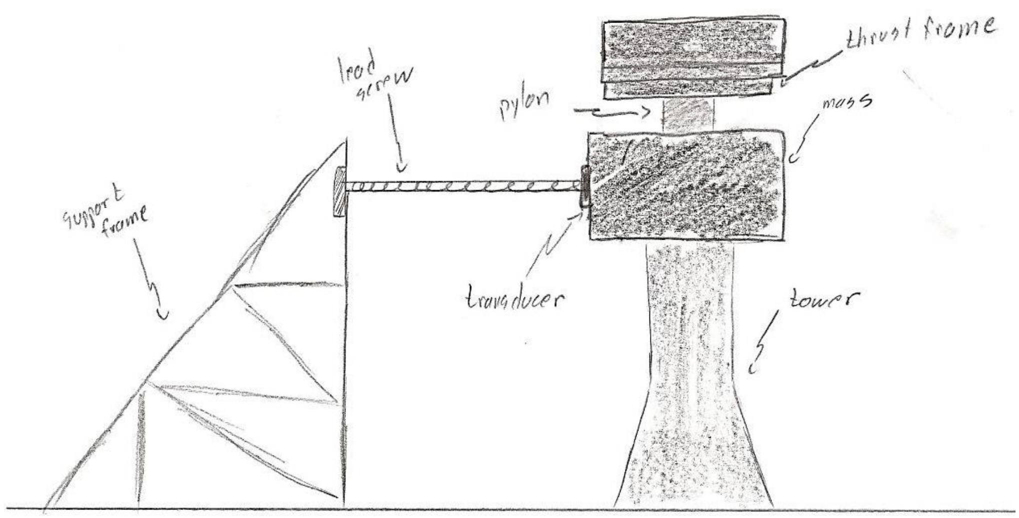

Figure 6: Reference design for thrust simulation.

For weight simulation, the client expressed interest in two of concepts for further analysis and consideration. These concepts are shown in Figures 7 and 8.

Turnbuckle Weight Simulation Concept

The turnbuckle concept contains two cables that are attached to both ends of a turnbuckle. One cable will be mounted to the ground and the other attached to the pylon. As the turnbuckle is rotated, tension will be applied to both cables causing downwards force on the calibration pylon, thereby simulating the weight of an engine. This concept is shown in Figure 7.

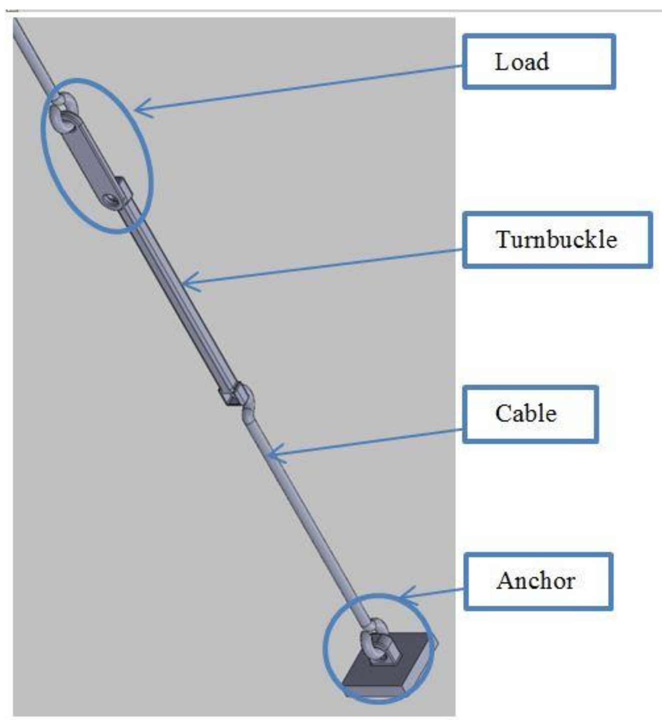

Figure 7: Turnbuckle engine weight simulation concept.

Reference Weight Simulation Concept

The reference weight concept is designed to use a predetermined set weight that will be suspended from the calibration pylon. The suspended weight will simulate the weight of a hanging engine. This concept is shown in Figure 8.

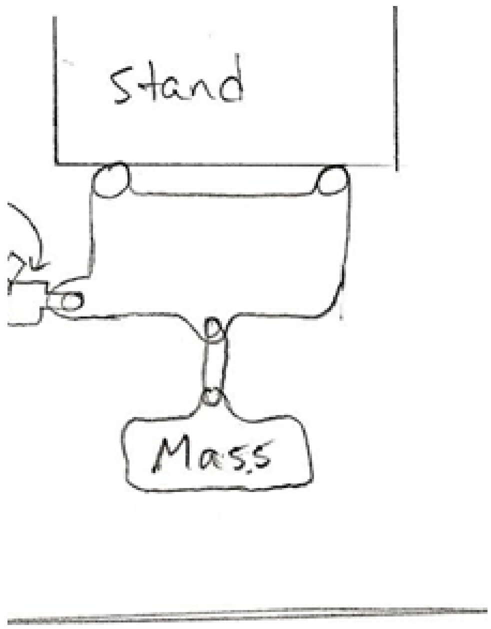

Figure 8: Reference design for weight simulation.

The team then set out to verify the feasibility of implementing the concepts selected by the client through a technical analysis and costs comparison. Through this analysis, a single concept for each of the thrust simulation and engine weight simulation were selected to be brought forth for the Final Design Generation phase.

2.1.1 Thrust Simulation

After a preliminary analysis of three thrust simulation concepts, the team possessed sufficient information to create a preliminary cost estimation for each of the three concepts. The results of the cost analysis revealed that implementing the simple press design would be significantly more expensive than the other concepts. Consequently, the simple press was not brought forward for further development.

The cost analysis also showed that implementation of the simple lever and the reference design would cost approximately the same. UMCal ultimately chose the simple lever design as the final thrust simulation concept as it was believed to benefit the calibration system in the following ways:

- The lower ball screw location allows for easier use and maintenance.
- The reduced required input load will allow for a smaller, less expensive, ball screw.

Further details of the cost analysis performed during the concept generation phase are found in Appendix D.

2.1.2 Engine Weight Simulation

Upon completing analyses of the top two weight simulation concepts, the team was able to confirm that the turnbuckle concept was superior to the set weight concept with respect to cost. A cost comparison between the concepts revealed that the set weight concept is almost eight times the cost of the turnbuckle concept. Further details of the cost analysis performed during the concept generation phase are found in Appendix D.

Furthermore, the set weight would be cumbersome to transport throughout the facility in comparison to the turnbuckle, which would easily be carried by one person. However, the client expressed concern that the turnbuckle design may produce a force that is not truly vertical. During the refinement of the design, UMCal addressed this issue in the manner discussed in Section 3.3.

Further details of the cost analysis performed during the concept generation phase are found in Appendix D.

3. Final Design Concept

The selected final design concepts for the engine thrust and weight simulation have been developed further. A calibration pylon which serves to transfer the simulation loads into the thrust frame has also been designed in detail. Each designed component of the calibration system required hand calculations, detailed CAD models, high level computational analysis and cost estimations. A detailed description of the final calibration system design is provided in the sections to follow.

3.1 Design Overview

The calibration design proposed by UMCal consists of three major components: Engine Thrust Simulation, Engine Weight Simulation and Calibration Pylon. An overview of each component and its relation to the overall calibration system is described in the sub-sections to follow. For reference, the main component locations with respect to the facility layout are shown in Figure 9.

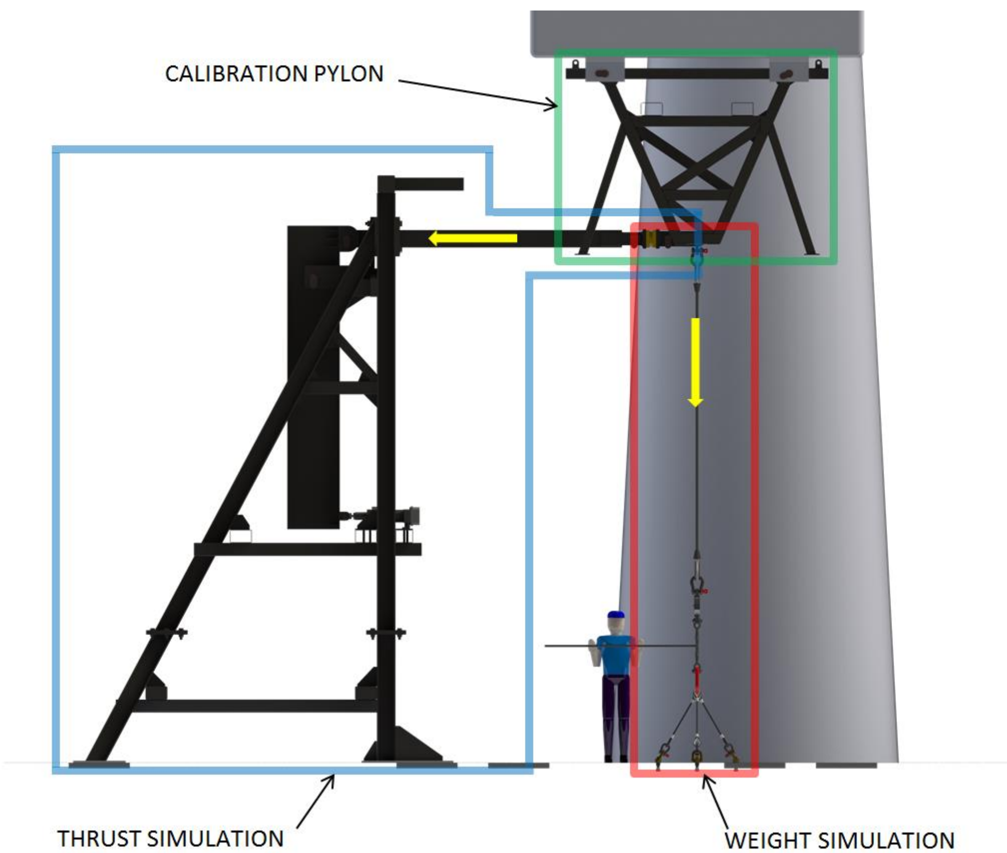

Figure 9: GE TRDC facility layout with major components labelled.

Engine Thrust Simulation

The thrust simulation method incorporates a simple lever design that is retrofitted into the existing A-frame. In this design, a force input at mid-frame level results in a "pulling" force along the engine centerline. A single beam lever rotates about an optimized pivot point to achieve this force transfer. The lever provides a mechanical advantage by reducing the input load required to obtain the desired thrust simulation force during calibration. To resist the bending forces associated with this design, an I-beam was chosen to act as the lever. The I-beam is responsible for transferring the force from the ball screw to a pylon connecting rod located at the engine center line. The connecting rod serves to transfer this force into a horizontal load acting on the calibration pylon, and in turn thrust frame. The complete engine thrust simulation assembly is shown in Figure 10.

Figure 10: Engine thrust simulation assembly.

Engine Weight Simulation

In order to the replicate the effect of an engine hanging during the calibration procedure, it is required that a force be applied to the thrust frame in a downwards direction. For the engine weight simulation, a wire rope tensioned by a turnbuckle will create the downwards force. The wire rope is connected to the suspended calibration pylon at one end, and the ground at the other. As the turnbuckle is rotated, the tension within the wire rope increases. Subsequently, a downwards vertical force of a desired magnitude is applied to the calibration pylon and in turn the force is transferred to the thrust stand, thus simulating the weight of an engine. The lower portion of the engine weight simulation assembly is shown in Figure 11.

Figure 11: Weight simulation assembly.

Calibration Pylon

The calibration pylon is a structural assembly suspended from the thrust stand. Its purpose is to transfer the applied loads from both the engine thrust simulation and the engine weight simulation to the thrust stand. The calibration pylon interfaces with the underside of the thrust stand via four pin connections. A render of the complete pylon is shown in Figure 12

Figure 12: Calibration pylon.

3.2 Thrust Simulation Design

UMCal has designed a fixed frame with pivoting lever system to simulate the engine thrust. This system incorporates an existing A-frame at the GE TRDC facility, with modifications to accommodate a structural pivoting lever. Other main elements of this design include an input load actuation system and a connecting rod to transfer the thrust load from lever to calibration pylon. The detailed design and analysis of the complete thrust simulation system is outlined in the sections to follow.

3.2.1 Structural Lever

Due to its large size and interfacing requirements within the system, the structural lever was the first thrust simulation system component to be designed. This component became the base around which all other system components were designed. An overview of the entire lever assembly is shown in Figure 13 with major components labelled.

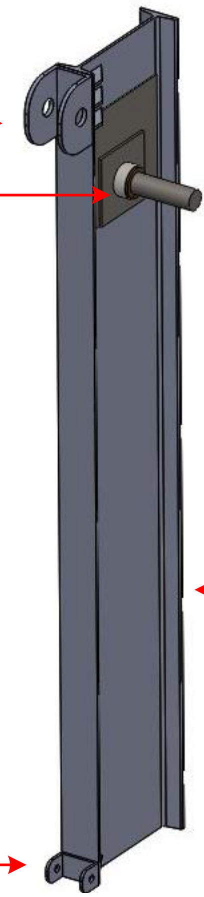

Figure 13: Structural lever main components.

Design Criteria

A set of design criteria was established prior to the detailed design and analysis of the structural lever. These criteria are shown in TABLE V.

TABLE V: STRUCTURAL LEVER DESIGN CRITERIA

Specification	Value
Maximum thrust load [lbf]	105,000
Minimum safety factor	2

In addition, the lever is required to interface with the existing A-frame using minimal modifications, efficiently transfer load from the base input to the thrust line output, and withstand a combined loading from the both the 105,000 $[lbf]$ as well as the corresponding input load.

The engine thrust calibration system was designed for a 105,000 [lbf] thrust load where possible, therefore the lever was designed with the goal of meeting this desired loading capacity. Should the need for calibration of higher thrust loads increase from the current $85{,}000$ [lbf], this design may be used without modification until a thrust load of 105,000 $[lbf]$ is reached. The overall lever design maintains a minimum safety factor of 2.

Design Process and Decisions

The lever design began with the selection of the optimal pivot point location. This location was dependent on several factors:

- Maximum bending moment
- Input-to-thrust load ratio
- Horizontal translation at base
- Vertical load component at the engine centerline

A main design restriction for a lever of this length is deflection at the beam ends. Significant deflection at either the load application point or the thrust simulation point would result in an increased required input load and higher levels of off-axis loading. As such, the reduction of bending moment was a primary concern. A secondary concern was the ratio of input load to obtain the required maximum thrust output load. The higher the pivot point placement above the input to output midpoint, the lower the input load. An additional consideration was the travel distance of the lever base over the thrust load range. [redacted] the thrust frame experiences a horizontal displacement of approximately $0.467[in]$ at $85{,}000\ [lbf]$ [7]. This plot was extrapolated to approximate a thrust frame displacement of $0.512\ [in]$ at $105{,}000\ [lbf]$. For a feasible design, it was necessary for the translated displacement at the load input level to be within a reasonable range for ball screw actuation. Finally, as off-axis loading of both the load cell (at engine centerline) and ball screw (at load input level) is undesirable, a reduced vertical component of the lever load was necessary.

The variation of each parameter was examined as the pivot point was moved from one end of the lever to the other. The pivot point location was which best met all the above considerations was found to be 1.5 $[ft]$ below the thrust simulation line. This placement provides the following benefits:

- Low bending moment of 1,890,000 $[lb \cdot in]$, a significant reduction when compared to a pivot point placement at the input to output load midpoint.
- Input load of $16{,}726$ $[lbf]$ is less than a fifth of the thrust load ($105{,}000$ $[lbf]$).
- Low vertical load component of $2{,}917$ $[lbf]$.
- Acceptable base actuation of 3.14 $[in]$ at maximum load.

Once a pivot point was selected, the main beam component was sized under a worst-case loading scenario. This involved fixing the pivot point with the maximum thrust load of $105{,}000$ $[lbf]$ at the top and a balanced input load at the bottom. The input load was calculated to balance the moment of the thrust load. A free body diagram of the loading scenario is shown in Figure 14.

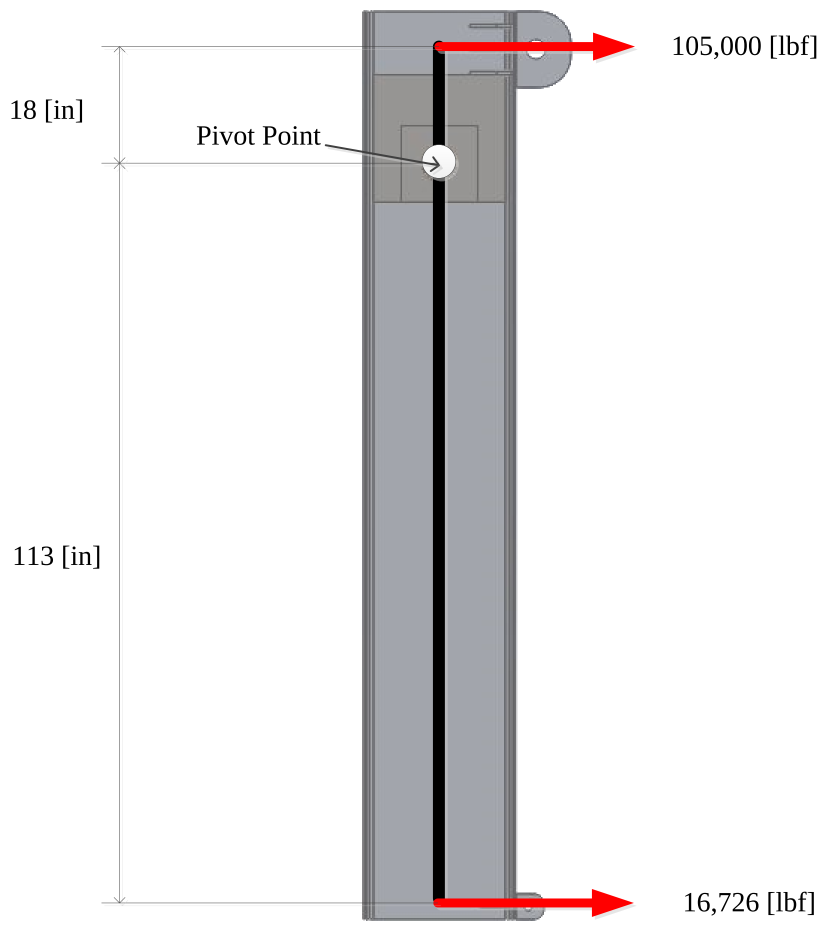

Figure 14: Input and output loads on structural lever.

The maximum bending moment of the beam occurs at the balanced pivot point, calculated as:

$$M_{max} = F_T \cdot h_t$$

$$M_{max} = 105{,}000\ [lbf] \cdot 18\ [in] = 1{,}890{,}000\ [lb \cdot in]$$

To withstand this bending moment, the required section modulus of the beam was calculated as follows:

$$S = \frac{M_{max}}{\sigma_{yield}} \cdot Design\ Factor$$

Assuming ASTM-A36 structural steel material properties and incorporating a safety factor of 2, the minimum required section modulus is:

$$S = \frac{1{,}890{,}000\ [lb \cdot in]}{22{,}000\ [psi]} \cdot 2 = 172.82\ [in^3]$$

A variety of beam types with the required section modulus were considered including solid bar, box beams (square and rectangular), and I-beams (wide flange and standard). A section modulus of this magnitude narrowed down the beam selection to standard and wide-flanged I-beams. Potential beams that underwent final evaluation are shown in TABLE VI [8].

TABLE VI: BEAMS CONSIDERED FOR LEVER DESIGN

Designation	Depth $[in]$	Width $[in]$	Section Modulus $[in^3]$	Weight $\left[\frac{lb}{ft}\right]$
W18 X 106	18.73	11.20	204.0	106
W21 x 101	21.36	12.29	227.0	101
W27 X 84	26.71	9.960	213.0	84
S24 X 80	24.00	7.000	175	80

The I-beams found in TABLE VI all met the minimum required strength of the lever. Parameters of weight and size were therefore considered for final selection. As the weight per foot of all beams translate into a total weight of slightly under $1{,}000\ [lb]$ over an $11.75\ [ft]$ length, any decrease in weight became crucial to maintain a reasonable design weight. The final lever beam selected was the S24 X 80 Standard Shape I-beam. This beam proved lightest by its advantageous use of area. Although deeper than the W18 X 106 and W21 X 101 beams, the $7\ [in]$ width of the S24 X 80 was deemed beneficial for mounting within the A-frame, as it will provide good clearance for a pivot support structure.

The second main component of the lever design is the pivot bar. The pivot bar was designed to withstand the loads imposed by the lever design including a safety factor of 2.

The bar was sized to resist both shear and bending loads. The shear load is a result of the reaction forces acting on the bar. These forces were combined into a single force vector that includes the sum of the thrust and input loads, as well as an estimated lever weight. Bar material of AISI-C1144 stress relieved steel was selected, due to its high strength properties [9]. The bar loading is shown in Figure 15.

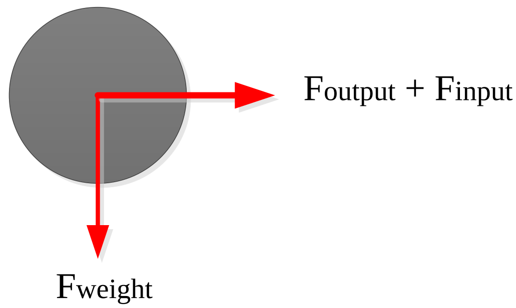

Figure 15: Forces acting on pivot bar.

The previously calculated input load of $16{,}726\ [lbf]$ was calculated as simply the force required to balance a $105{,}000\ [lbf]$ thrust load about the lever pivot point. This load however, did not take into account inefficiencies within the system such as deflection of the beam and friction at the pivot. To account for some of these losses, the pin was sized with an increased input load of $17{,}500\ [lbf]$. In addition, the beam weight was increased to $1{,}250\ [lbs]$ to account for additional lever weight such as beam reinforcements and connection hardware.

A total reaction load of the input load, thrust load, and beam weight was calculated as follows:

$$F_R = \sqrt{F_W^2 + (F_T + F_{in})^2}$$

$$F_R = \sqrt{1{,}250^2 + (105{,}000 + 17{,}500)^2} = 122{,}506\ [lbf]$$

The required bar diameter to resist shear loading failure may be calculated from the following derivation:

$$\tau = \frac{F_R}{A} \cdot Design\ Factor \rightarrow\ A = \frac{F_R}{\tau} \cdot Design\ Factor$$

For a round solid bar with loading divided over two cross sectional areas,

$$A = 2 \cdot \frac{\pi D^2}{4}$$

$$\therefore D = \sqrt{\frac{2}{\pi} \cdot \frac{F_R}{\tau} \cdot Design\ Factor} = \sqrt{\frac{2}{\pi} \cdot \frac{122{,}506\ [lbf]}{89{,}900\ [psi]} \cdot 2} = 2.335\ [in]$$

The second form of loading that the bar must resist is bending. With the bar designed to span 9 $[in]$ between the fixed supports, while maintaining 1 $[in]$ clearance on either side of the I-beam lever, a bending moment is introduced. The required diameter to resist bending moment failure may be calculated from the following derivation:

$$\sigma_{yield} = \frac{M_{max} \cdot y}{I} \cdot Design\ Factor \rightarrow D = \sqrt[3]{\frac{32 \cdot M_{max}}{\pi \sigma_{yield}} \cdot Design\ Factor}$$

The moment on the bar was modeled assuming fixed supports on either end of the bar and a condensed point load at its midpoint. This was taken for a conservative calculation as the point load will actually be distributed along the length of the bearing, effectively reducing the bar length subjected to bending. The loading scenario is shown in Figure 16.

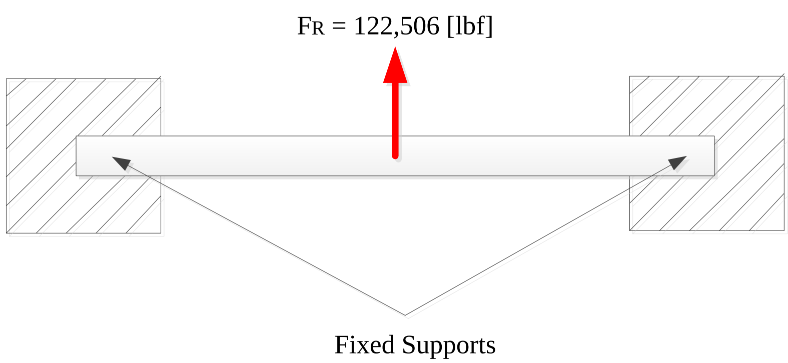

Figure 16: Loading scenario on pivot bar.

For this loading scenario the maximum bending moment, located at the bar midpoint, was calculated using the following equation:

$$M_{max} = \frac{P \cdot l}{8} = \frac{122{,}506\ [lbf] \cdot 9\ [in]}{8} = 137{,}819\ [lb \cdot in]$$

From the last two equations, the minimum required diameter to resist bending moment failure was calculated to be 3.15 $[in]$. The final pivot bar selection was rounded up to a nominal 3.5 $[in]$ diameter bar of AISI-C1144 stress relieved steel.

To provide smooth rotation of the pivot bar under such high loading, a self-lubricating sleeve bearing was incorporated around the pivot bar. The use of a self-lubricating bearing rather than incorporating the use of conventional oil or grease was deemed critical for this design as the latter two both require full shaft rotations to form a lubricant film. The benefits of this bearing selection include:

- Better lubrication for high load applications with minimal oscillating rotation [10].
- Better for use in cold temperatures, where oil or grease might freeze [10].
- Less maintenance and need for oil application [10].

Because the pivot bar undergoes minimal rotation, a more complex bearing such as roller or ball was deemed unnecessary. The bearing will be press fit within the I-beam at the pivot location, allowing the lever to rotate freely about the pivot bar. This configuration was deemed best for ease of installation within the existing A-frame. In addition, it allows for easy bar and bearing replacement, should they wear over time.

The bearing capacity and life is typically sized based on bearing pressure and velocity. Because this pivot bar is undergoing very small and slow oscillating motion, the bearing velocity became negligible in sizing. Bearing pressure was calculated as follows:

$$P = \frac{F}{d \cdot L} = \frac{122{,}506\ [lbf]}{3.5\ [in] \cdot 3\ [in]} = 11{,}667\ [psi]$$

The above calculations are for the Isostatic TU Steel Backed PTFE Lined sleeve bearing (P/N 501157). With a maximum bearing pressure of $36{,}250\ [psi]$, this bearing provides a safety factor of 3.1. In addition, this bearing is rated to $-200\ ^{\circ}C$, making it more than suitable for the icing facility application should winter calibration be desired. Further details of the pivot interface are shown in Figure 17.

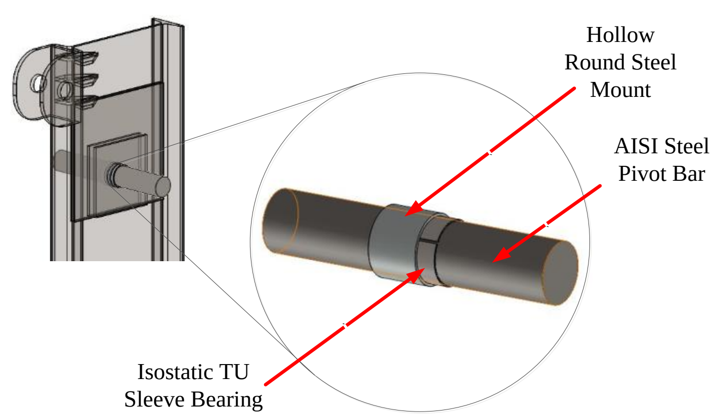

Figure 17: Structural lever pivot point interface.

The final components of the lever design are the load attachment points at each end. Simple receivers for a pin attachment are made of custom C–channel sections fabricated by welding three individual structural steel plates. These plates must be able to withstand the shear loading of the mounting pins at maximum load. To achieve the required shear area, both plate thickness and hole depth were varied as shown in Figure 18.

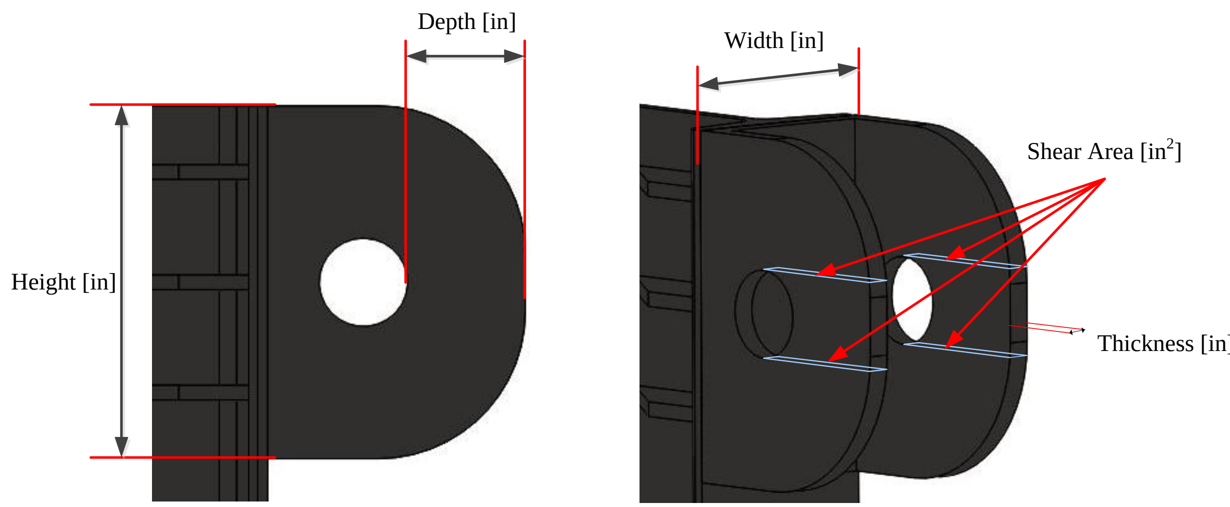

Figure 18: Load attachment plates.

With the load distributed over two plates, the required shear area of each plate was calculated using the following formula:

$$A = \frac{P}{\tau} \cdot Design\ Factor \ \rightarrow \ t \cdot d = \frac{1}{2} \cdot \frac{P}{\tau} \cdot Design\ Factor$$

The final dimensions for the top and bottom attachment points along with their loading are summarized in TABLE VII.

TABLE VII: STRUCTURAL LEVER DESIGN CRITERIA

Location	Plate Thickness [in]	Height [in]	Length [in]	Width [in]	Pin Diameter [in]	Max Rated Load [lbf]	Safety factor
Thrust Line	0.75	12	8.75	7.0	3.25	105,000	2
Input Load Line	0.50	4	4.5	7.0	1.25	17,500	2

For the above load ratings, ASTM A36 Structural Steel was again used for material properties, as this material is readily available and inexpensive when compared to higher strength steels. With set plate thicknesses, the required pin hole depth for the top and bottom connection points was calculated to be 6.36 $[in]$ and 1.59 $[in]$ respectively. These values were maintained in positioning of the pin holes within the plate.

Cost Analysis

The estimated cost of this lever design is $992.74. This estimate includes only material level costs and does not account for manufacturing as it will be largely dependent on in-house manufacturing capabilities and preferred contractors. Details of the lever design cost estimate are outlined in TABLE VIII.

TABLE VIII: STRUCTURAL LEVER COST SUMMARY

Item	Qty	Unit Cost	Total Cost
S24 X 80 I-beam	12 [ft]	$ 50.00/ft	$ 600.00
ASTM A36 Steel Plate, 1/4”	7 [ft^2]	$ 5.67/ft^2	$ 39.69
ASTM A36 Steel Plate, 1/2”	3 [ft^2]	$ 11.33/ft^2	$ 33.99
ASTM A36 Steel Plate, 3/4”	2 [ft^2]	$ 18.94/ft^2	$ 37.88
AISI-C1144 Stress Relieved Steel Round Bar, 3-1/2 OD x 18” L	1	$ 202.12	$ 202.12
Structural 1020 Steel Round Tube, 4” OD w/ 01/4” wall thickness	1 [ft]	$ 60.06/ft	$ 60.06
Isostatic TU Sleeve Bearing P/N 501157. 3-1/2” ID x 3-11/16” OD x 3” L	1	$ 19.00	$ 19.00
Total			$ 992.74

The cost estimate for the S24 X 80 I-beam was based on a quote from Brunswick Steel [11]. Although the supplier does offer this beam, it does not always carry it in inventory. The provided estimate was based off a similar sized beam (W24 X 62), priced at $39.00/ft. As the S24 X 80 is less common, its cost estimate was increased to $50.00/ft for a conservative cost estimate.

3.2.2 System Actuation Design

Once the structural lever was sized, a required load input was determined. Again, this load was increased slightly to account for losses in the system as well as bring the load input to a more nominal value of 17,500 $[lbf]$. The next step in design of the thrust simulation system was to ensure an adequate actuation system exists. In terms of the actuation system design, UMCal opted to size and source basic components to ensure design feasibility.

Design Criteria

The design criteria of the thrust simulation actuation system focused mainly on physical requirements to drive the system; requirements related to available instrumentation were given less emphasis. The system actuation design criteria are listed in TABLE IX.

TABLE IX: SYSTEM ACTUATION DESIGN CRITERIA

Specification	Value
Available thrust Load Output [lbf]	17,500
Minimum Total Lead [in]	3.14
Safety factor	2

In addition, the actuation system must be of reasonable size to fit within the A-frame assembly. During the concept generation phase it was decided that hydraulic systems should be avoided if possible, thereby increasing design simplicity and ease of set-up as well as decreasing cost.

Design Process and Decisions

To generate $105{,}000\ [lbf]$ at the engine thrust line, the lever must translate $0.5\ [in]$ at the top and a corresponding $3.14\ [in]$ at the load actuation line. UMCal considered both lead and ball screws to achieve this actuation, based on the team's findings during the concept generation phase. Of these options, ball screws were found to offer greater selection for the high load of this application. In sourcing out an appropriate ball screw, the team searched for screws with as small a lead as possible to minimize the size of load increments on the thrust frame. In terms of the ball screw precision and accuracy, UMCal opted for the lower values when possible as it was deemed unnecessary to the system. In conjunction with the client, UMCal concluded that the ability of the system to input exact load increments was not necessary, as long as the actual load imposed on the thrust frame is accurately measured and recorded by the calibration load cell. This parameter is therefore controlled by the calibration load cell, and not the actuation system. By accepting lower accuracy in the ball screw, cost could be reduced.

UMCal has selected an E-Drive Actuators ball screw system for the thrust simulation actuation design. The E-Drive Eliminator HD series offers complete ball screw assemblies for high load applications [12]. These assemblies range up to load outputs of $25{,}000\ [lbf]$

and require only a motor drive to complete the system. Of the numerous model variations, the HD618-06-MB-E-U4 model was selected for this application. Specifications of the actuator are summarized in TABLE X.

TABLE X: E-DRIVE HD618-06 BALL SCREW ACTUATOR SPECIFICATIONS

Model	Rated Thrust Load [lbf]	Max Thrust Load [lbf]	Max Travel Length [in]	Lead [mm]	Max Required Torque [lb in]	Dynamic Capacity [lbf]	Weight [lbs]
HD-618-06	18,000	30,000	6.0	12	1,500	27,840	240

This model was selected for its rated thrust load, just above the required $17{,}500\ [lbf]$ in addition to its shorter travel length of $6\ [in]$. This length will provide adequate room for the required $3.14\ [in]$ actuation of the lever, as well as provide additional length for deformation in the complete thrust simulation system. A benefit of having a small available travel length is a reduced size in the actuator system, thereby making the mounting of the actuator within the A-frame easier. Furthermore, the dynamic capacity of the chosen ball screw was not a concern because the desired travel life of the ball screw is much lower than $10^6\ [in]$ and the specified dynamic load capacity of the ball screw exceeds the rated thrust load. A variation of the HD618-06 is shown in Figure 19.

[redacted]

Figure 19: Eliminator HD ball screw actuator ([redacted][12].

Unlike the above figure, the actual ball screw assembly does not include a motor and thus, one must be sourced separately. Therefore, with the selection of a ball screw actuator complete, sizing a motor to drive the system was required. Based on the actuator parameters, the motor input must deliver a maximum torque of $1{,}500\ [lb \cdot in]$ and interface with the linear actuator. The motor must also be capable of delivering incremental rotation and maintaining position, rather than continuous revolution. Due to the nature of rotation, stepper motors and servo motors were considered. Although used in typical ball and lead screw applications, stepper motors (even with a reduction gearbox) are not able to produce the required torque of this high load system. In order to achieve the torque requirement of this system, a servo motor with gearbox combination was selected. The selected motor drive system includes a Baldor AC brushless servo motor with a 10:1 ratio gearhead. Specifications of both units are shown in TABLE XI.

TABLE XI: BALDOR MOTOR DRIVE SYSTEM SPECIFICATIONS

Brand	Model	Max Input Torque [lbin]	Max Output Torque [lbin]	Max Current [A]	Gear Ratio	Stages
Baldor	BSM90N-2150AF [13]	N/A	354	22.1	N/A	N/A
Baldor	GBSM90-MRP155-10 [14]	159	1505	N/A	10	1

The final output torque of the motor drive system meets the required $1{,}500\ [lb \cdot in]$ torque of the linear actuator with use of a 160 VDC electrical input, and a rotational speed of up to $500 [rpm]$. In addition, the –AF series indicates that this motor has an encoder feedback device for positional input.

The linear actuator, servo motor, and gearhead make up the complete mechanical actuation system of the thrust simulation design. The complete actuation assembly is shown in Figure 20.

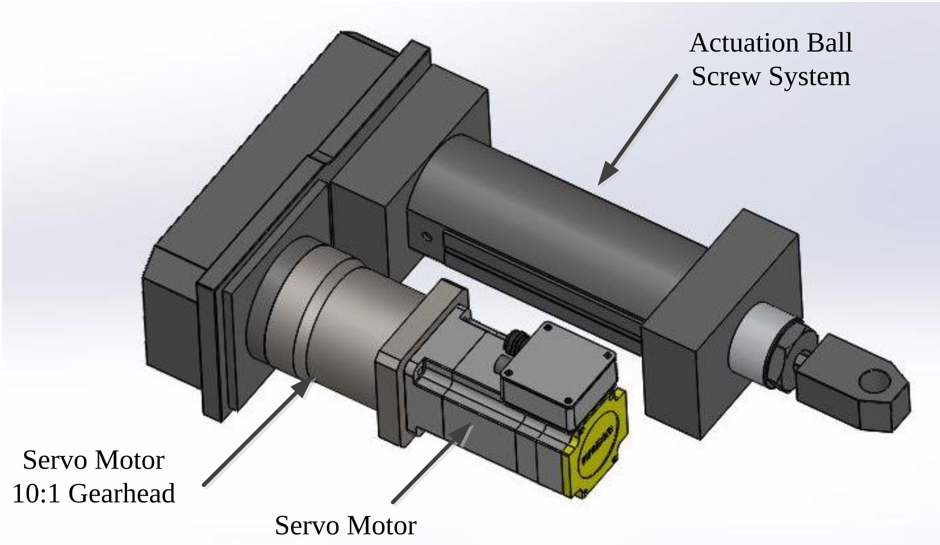

Figure 20: Actuation system assembly.

As previously mentioned, this is a baseline design to ensure system feasibility. In selecting the components of the actuation system, UMCal ensured that the selected component companies offer a variety of similar models with varying instrumentation requirements. As such, the client may more easily modify this design to suit existing instrumentation at the facility.

Cost Analysis

A rough cost summary for the main actuation system components was calculated, with pricing quotes obtained from Red Line [15]. The system cost estimation is $23,921 and the summary is shown in TABLE XII.

TABLE XII: SYSTEM ACTUATION COST SUMMARY

Item	Qty	Unit Cost	Total Cost
E-Drive Eliminator HD618-06 Ball Screw Linear Actuator [15]	1	$ 19,495.00	$ 19,495.00
Baldor BSM90N-2150AF Brushless Servo Motor [16]	1	$ 2,202.00	$ 2,202.00
Baldor GBSM90-MRP155-10 Servo Motor Gearhead [16]	1	$ 2,224.00	$ 2,224.00
Total:			$23,921.00

The servo motor cost found in the above table includes the cost of the encoder. Additional costs, such as fasteners, are not incorporated in this estimate.

3.2.3 Pylon Connecting Rod Design

The connecting rod is the component that connects the structural lever to the calibration pylon. The load cell responsible for measuring the thrust simulation load is housed within this rod to ensure that the entire magnitude of the force being applied to the pylon is recorded.

Design Criteria

The connecting rod serves the purpose of transferring a horizontal load from the lever through to the calibration pylon. As such, the connecting rod must be capable of withstanding the maximum applied load of $105{,}000\,[lbf]$ used to simulate the engine thrust. The connecting rod must mate with the calibration pylon at one end and the lever on the other, and must accommodate a load cell mount. TABLE XIII summarizes the loading criteria of the connecting rod.

TABLE XIII: PYLON CONNECTING ROD DESIGN CRITERIA

Specification	Value
Minimum horizontal load $[lbf]$	105,000
Desired safety factor	2

Furthermore, UMCal has chosen to use A36 structural steel as the primary connecting rod material wherever possible, as it is readily available and inexpensive when compared to higher strength metals.

Design Process and Decisions

The first step in designing the connecting rod was to determine the cross-sectional area required to withstand the applied thrust loading. As this member will only be subjected to tensile loading, bending was not considered. The cross-sectional area was simply determined by dividing the applied force by the material's yield strength, as shown in the equation below.

$$A_{min} = \frac{\sigma}{F}$$

Using the tensile yield strength of structural steel ($36[ksi]$) and an applied force of $210{,}000\ [lbf]$ to ensure a safety factor of 2, the minimum cross-sectional area was determined to be $5.83[in^2]$. To select the size of the structural steel tubing, the cross-sectional area was rounded up to match the area of nominal sized tubing. A tube selection of $8 \text{ x } 3 \text{ x } 5/16"$ with a cross-sectional area of $5.85[in^2]$, was chosen. This member size also allows room for mounting plates, of sufficient height to accommodate the $3[in]$ pin hole, to be welded to the tubing walls. The pin hole is needed for the connections between the calibration pylon and simple lever. Figure 21 illustrates the pin hole and its surrounding dimensions. All dimensions are in inches.

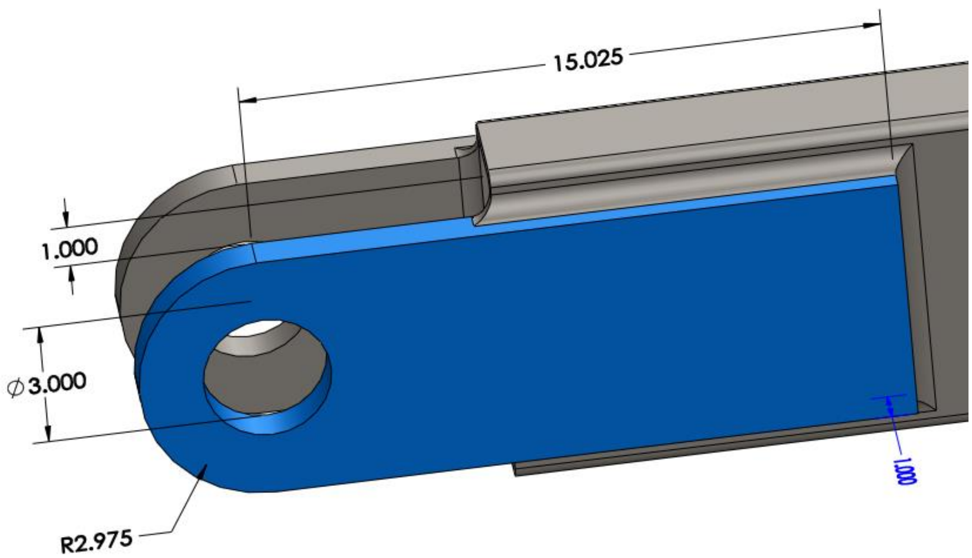

Figure 21: Connecting rod pin hole sizing.

Since the cross-sectional area of the tubing will decrease at the location of the pin hole, the height of the mount plates need to be sized to compensate for the area loss. With the use of the minimum required cross-sectional area previously stated and selecting a width of $1\ [in]$ for each of the two plates that are to be welded to tubing, the necessary height of the plate may be obtained using the equation below.

$$A = b \cdot \left(h - D_{pin}\right)$$

Solving for h,

$$h = \frac{A}{b} + D_{pin}$$

Where:

- b is the overall thickness of the 2 plates (2 $[in]$)
- D_{pin} is the diameter of the pin being used of 3 $[in]$
- A is the previously determined area of 5.85 $[in^2]$
- h is the required height of the plate

With the above stated values, the minimum value of h was calculated to be 5.915 $[in]$. A nominal size of 6 $[in]$ was chosen due to ease of availability. Since the connecting rod also requires mounting to the load cell, two additional flange connectors are needed. The flange connectors will be mounted to both ends of the load cell, and then be attached to the connecting rod. One male flange connector will be required at one end of the load cell so it may mount to the connecting rod, while the other flange is required to be a female connection so that the load cell may connect to the calibration pylon. Figure 22 displays the overall assembly of the connecting rod with the load cell attached.

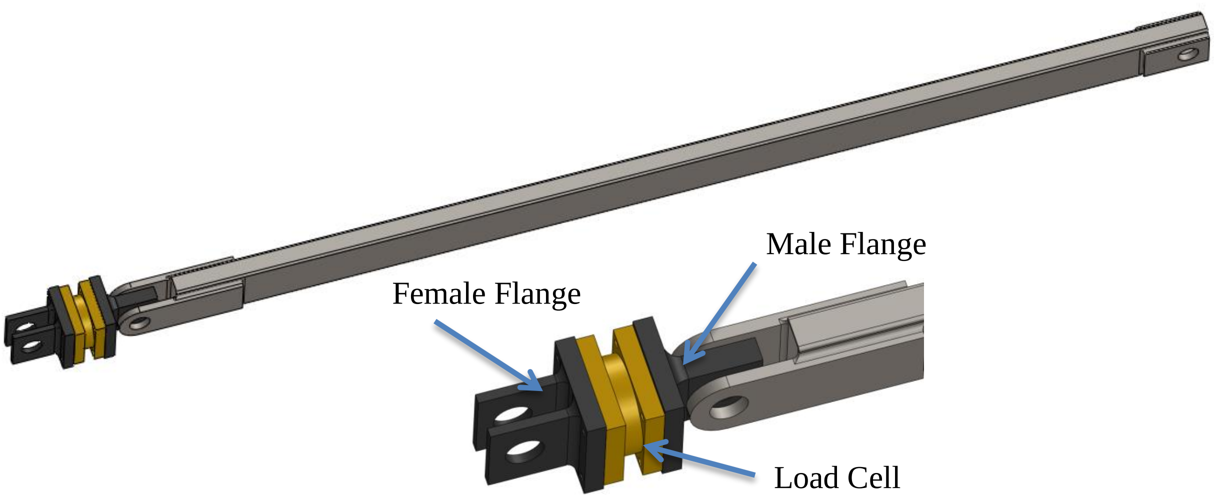

Figure 22: Connecting rod assembly.

The flange connectors needed to support the loading are required to either be cast in iron, or be custom fabricated by a third party using high strength steel. Should the flanges be cast, UMCal recommends that the material used be Malleable Cast Iron. UMCal recommends this material as it contains similar material properties to that of ASTM A36 structural steel in regards to density and tensile strength.

In selecting a load cell for use with this design, it was of critical importance that the client's standards for accuracy, calibration, and precision be met or exceeded. After examining options from several manufacturers, UMCal selected the Interface Model 1632 load cell as the best fit for this design. The relevant specifications for this component, as well as the requirements set forth by the client, are shown in TABLE XXVII.

TABLE XIV: LOAD CELL SPECIFICATIONS [17]

Parameter	Unit	Interface 1632	Client Specification
Accuracy	% FS	±0.05	±0.05
Hysteresis	% FS	±0.05	±0.1
Range	lbf	100,000	85,000
Output voltage	mV/V	4.0	-0.1 to 6.0
Repeatability	% FS	±0.005	±0.05

As shown in TABLE XIV, the chosen load cell meets or improves upon each of the required parameters laid out by the client. UMCal is therefore confident that this load cell will function to the client's expected standards.

Cost Analysis

A cost estimation for implementing the connecting rod is shown in TABLE XV. All components have been sourced as ASTM structural steel. This is a preliminary cost estimate, and does not include the costs for labour associated with the fabrication of the connecting rod. The total cost of materials needed to implement the connecting rod is estimated at $8,847.29.

TABLE XV: PYLON CONNECTING ROD COST SUMMARY

Item	Qty	Unit Cost	Total Cost
Steel Plate 12" X 12" X 2" [18]	1	$108.17	$108.17
Steel Plate 12" X 12" X 1.5" [18]	2	$83.33	$166.66
Steel Plate 12" X 18" X 1" [18]	2	$60.23	$120.46
Tube 150" X 8" X 3" X 5/16" [18]	1	$654.00	$654.00
Interface Model 1632AJH-100k load cell [17]	1	$7603.00	$7603.00
Interface load cell mating cable, 50 ft [17]	1	$195.00	$195.00
Total			$ 8847.29

3.2.4 A-Frame Design

The largest structural component of the calibration system design is the A-frame to which the thrust simulation mechanism is mounted. This component is currently owned by the client as part of the previous calibration equipment, and must be modified in order to accommodate the proposed new design.

Design Criteria

Before beginning the design process, the required design criteria were determined. As this component houses the pivot bar and linear actuator, it must be capable of resisting both a $105{,}000\ lbf$ load at the thrust line and a $17{,}500\ [lbf]$ load at the linear actuator mounting position. The required design criteria for the A-frame are summarized in TABLE XVI.

TABLE XVI: A-FRAME DESIGN CRITERIA

Specification	Value
Minimum thrust load $[lbf]$	105,000
Minimum ball screw load $[lbf]$	17,500
Desired safety factor	2

Design Process and Decisions

As the current A-frame was originally designed to support a thrust loading of only $85{,}000\ [lbf]$, the ability for the frame to support a higher thrust load of $105{,}000\ [lbf]$ was unknown. Therefore, a preliminary FEA study was performed on the frame in its existing configuration, assuming the current in-line thrust loading method and a force of $105{,}000\ [lbf]$. The results of this study suggest that the existing A-frame would have no issues supporting the $105{,}000\ [lbf]$ load. Detailed results of this analysis will be discussed in Section 4.2 As UMCal is proposing a new method of applying the thrust simulation load, further analysis was performed to ensure that the A-frame is capable of supporting the additional loads and moments induced by the lever design. To ensure the structural integrity of the A-frame under the new loading conditions, modifications and reinforcements were applied to the design.

As mentioned in Section 3.2.1, the lever design rotates about a pivot point located $18\ [in]$ below the engine thrust line and $117\ [in]$ above the actuation line of the input load. To support the lever at its pivot point, it was necessary to add new mounting points to the rear of the A-frame. These mounting points are shown in detail in Figure 23. The rear structural beams have been made translucent to improve visibility, and the mounting blocks have been coloured red.

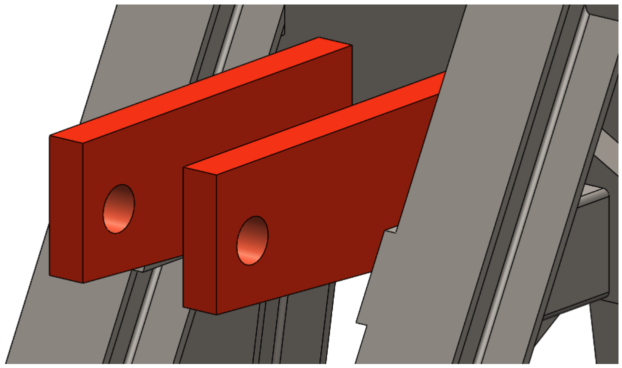

Figure 23: Pivot lever mount points.

Initial hand calculations were performed in order to size the thickness of the steel plate used for these mounting points. Using the compressive yield strength of A36 structural steel and a reaction force of 120,366 $[lbf]$ acting horizontally towards the front of the frame as determined previously in Section 3.2.1, the required contact area was determined to be 5.5 $[in^2]$ by the following equation:

$$A_c = \frac{P}{\sigma_y}$$

The pivot bar was specified to be 3.5 $[in]$ in diameter, and incorporating a safety factor of 2, the required cross-sectional area is thereby achieved using steel plate with a standard thickness of 3 $[in]$.

To support these mounting points for the lever, it was necessary to reinforce the existing frame structure. This was accomplished by means of two steel plates on each side of the vertical structural beams. These plates are to be welded to the existing structural wide flange beams. In addition, bolts will run through the existing holes in the A-frame in order to transfer to the load into the structure. The bolts are to be 1 ½"-6 UNC, Grade 8 steel hex cap screws, 16 $[in]$ in length, and washers are to be used on both contact faces. Bolt lengths of 16 $[in]$ are required in order to secure the plates on each side of the front structural members, and Grade 8 bolts are specified due to the magnitude of the applied load on the plates. The two plates are to be made of 2 $[in]$ A36 structural steel or equivalent. Square structural steel tube size of 8 ” x 8 ” x ¼” are to be placed in between the new mounting plate and the rear structural wide flange members for additional reinforcement.

Running beneath the lever supports is a 6"x 4" x ¼" rectangular structural member, connected to the 8"x 8 ″x ¼" braces on each side. All components are to be welded together. Further detail of these supports is shown in Figure 24. As shown previously, the rear structural members have been made translucent, and the components being discussed have been coloured red for visibility.

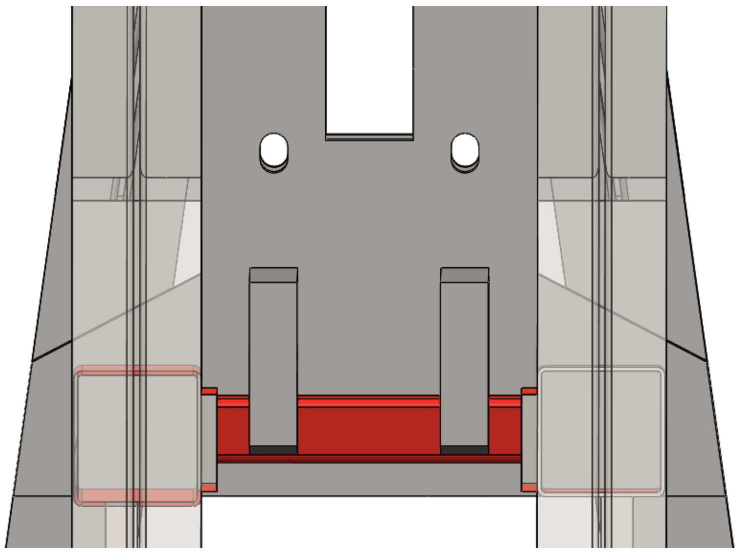

Figure 24: Structural reinforcement for pivot lever mount points.

Once the additions have been made to the A-frame, the lever is to be hoisted into place with a crane, the pivot bar inserted into the mounting blocks and welded in place. The connecting rod is then to be hoisted into place and bolted to the lever.

In order to mount the ball screw and motor system, a steel plate is to be bolted to the upper set of forklift pockets. The required thickness of this plate was determined using the ball screw load of 17,500 $[lbf]$ and the set mounting hole diameter of 1 $[in]$ that is provided with the chosen ball screw system. With four mounting holes on the ball screw housing, a load of 4,375 $[lbf]$ is to be applied to each mounting hole. The required thickness of the plate was determined by means of the following equation:

$$\sigma_y = \frac{P}{A}$$

$$22000 = \frac{4375}{A}$$

$$A = \frac{4375}{22000} = 0.2\ [in^2]$$

The required plate thickness was determined to be $0.5\ [in]$, keeping a safety factor of 2 and allowing for steel plate of standard size to be used,. The ball screw mounting holes are threaded for 1”-8 UNC screws and are $1.63[in]$ deep. Accounting for the additional thickness of the mounting plate requires a 1”-8 UNC cap screw of $2.25\ [in]$ in length. Grade 5 screws were chosen due to their availability and negligible cost increase over Grade 2 screws. As the load capacity of a single $1[in]$ Grade 5 cap screw in single shear is given as $56{,}548\ [lbf]$, the possibility of the ball screw mounts failing is not of concern [19].

Spacers are to be used to offset the actuator assembly, in order to vertically align the actuator and the lever connection point. These spacers are to be 2 ½ $[in]$ in diameter and 2 ¾ $[in]$ in height, and are to have their centres drilled to allow passage of mounting bolts.

In order to fasten the ball screw mounting plate to the A-frame, an $8''$ x $6''$ x $1/4''$ member is to first be welded to the front structural beams of the frame. This member and the front forklift pocket support are then to be drilled to accept four 1”-8 UNC bolts, $10\ [in]$ in length, which will secure the mounting plate to the A-frame. The ball screw mounting system is shown in Figure 25 with new components shown in red for clarity.

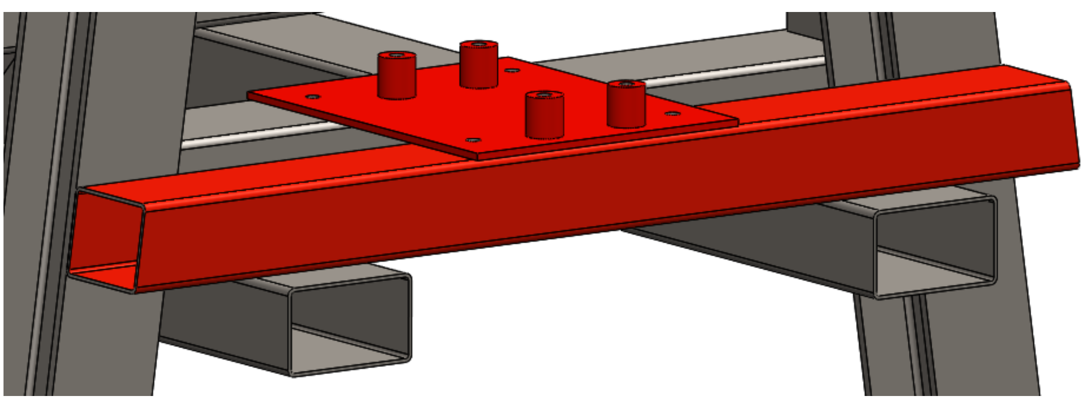

Figure 25: Reinforcement for ball screw mounting.

An image of the complete A-frame with all required modifications is shown in Figure 26.

Figure 26: A-frame design modifications.

Cost Analysis

The cost analysis of the required A-frame modifications revealed a total cost of \$7,415.85, which was much higher than anticipated. This was due largely to the quantity of steel plate in 2 [in] and 3 [in] thicknesses, which was required in order to achieve the desired safety factor. In addition, bolts of the required dimensions were not initially anticipated when estimating the cost of the modifications, and added a significant portion to the material cost. A complete breakdown of the material and equipment costs is shown in TABLE XVII.

TABLE XVII: A-FRAME MODIFCATIONS COST SUMMARY

Item	Qty	Unit Cost	Total Cost
3” A36 steel plate, 2x4 ft [20]	1	$1,592.72	$1,592.72
2” A36 steel plate, 4x4 ft [20]	2	$1,437.44	$2,875.54
1” A36 steel plate, 1x1 ft [20]	1	$93.51	$93.51
1” A36 steel plate,4x8 ft [20]	1	$1,254.40	$1,254.40
½” A36 steel plate, 4x2 ft [20]	1	$210.72	$210.72
1-1/2”-6 UNC x16” SAE J429 Grade 8 Hex Cap Screw [21]	4	$131.39	$525.56
1-1/2” USS Standard Washer, Zinc-Plated Steel, package of 5 [22]	2	$7.02	$14.04
Ultra-Coated Grade 8 Steel Hex Nut, 1-1/2”-6 UNC [23]	4	$8.20	$32.80
8x8x0.25” A36 structural steel, 2 ft [24]	1	$100.36	$100.36
6x4x0.25” A36 structural steel, 2 ft [25]	1	$50.76	$50.76
4x4x0.25” A36 structural steel, 6 ft [24]	1	$106.92	$106.92
6x6x0.25” A36 structural steel, 4 feet [24]	1	$115.84	$115.84
8x6x0.25” A36 structural steel, 8 feet [25]	1	$365.12	$365.12
1”-8 UNC x 10” Grade 5 Zinc-Plated Steel Hex Head Cap Screw [21]	4	$13.16	$52.64
Ultra-Coated Grade 8 Steel Hex Nut, 1”-8 UNC, package of 5 [23]	1	$9.12	$9.12
1”-8 UNC x 2-1/4” Grade 8 Hex Cap Screw [21]	4	$3.95	$15.80
Total			$7,415.85

3.3 Weight Simulation Design

Simulating engine weight is critical to the calibration process as the weight of a hanging engine affects the friction loading of the moving thrust frame during testing. In turn, this friction affects the load cell readings. UMCal has chosen to simulate this weight using a wire rope which is tensioned via a turnbuckle.

Design Criteria

Ensuring a safe engine weight simulation is a major criterion for this design due to the hazardous nature of wire ropes under tension. When a rope suddenly breaks, the stored energy within the rope causes it to swing, risking harm to anyone within its reach. To ensure that this situation never occurs, UMCal has designed this system to have a safety factor above and beyond all standards. Unless specified, all components of the design will be designed with an ultimate strength at least five times greater than the working load. Since damage and yielding is harder to observe in wire ropes, all components of the design that incorporate wire ropes will be sized such that the ultimate strength is at least ten times greater than the working load. To reduce the risk of damaging the rope, UMCal recommends that proper care be taken when handling and storing the rope. Before every use, the rope should be inspected for kinks, bird cage and core protrusions [26]. Furthermore, UMCal recommends that the rope is annually inspected by a professional to further ensure the safety of users. Unless specified otherwise, all other components of the design will be designed with an ultimate strength at least five times greater than the working load. For additional information regarding inspections, refer to Appendix A.

Another criterion for the weight simulation is its minimum load capability. Currently the calibration procedure hangs a fixed mass, weighing $10{,}000\ [lbf]$, from the calibration pylon. The proposed weight simulation method must meet or exceed this value. The client identified in the initial meeting that a weight simulation with variable outputs would be preferred as it would allow for more advanced calibration in the future [3].

The final requirement of this design is that it must create a completely vertical load. Once the thrust simulation load is applied, the calibration pylon and the upper rope attachment point will move horizontally. Consequently, an undesirable horizontal force component may be transferred into the pylon through the rope. TABLE XVIII summarizes the design criteria for the weight simulation component of the calibration system.

TABLE XVIII: WEIGHT SIMULATION DESIGN CRITERIA

Specification	Value
Minimum safety factor for wire ropes	10
Minimum safety factor for remaining components (unless justified otherwise)	5
Minimum load capacity $[lbf]$	10,000
Load direction	Vertical

Design Process and Decisions

The first step in the weight simulation design process was calculating the horizontal force component induced when the thrust frame deflects. UMCal was able to estimate this horizontal force component using thrust versus deflection data provided by the client. A graph showing the thrust frame deflection due to thrust is shown in Figure 27.

Figure 27: Thrust frame deflection vs. thrust [7].

By using ratios developed from similar triangles, the horizontal force component can be calculated using the following expression:

$$\frac{\Delta H}{V} = \frac{F_H}{F_V} \rightarrow F_H = F_V \cdot \frac{\Delta H}{V}$$

Since the pylon deflection (ΔH) at a $105{,}000$ $[lbf]$ thrust load is only 0.512 $[in]$, the horizontal force component (F_H) amounts to 21.3 $[lbf]$ when the tension in the wire rope (F_V) is $10{,}000$ $[lbf]$. Therefore, the horizontal for component is a negligible 0.2% of the vertical applied force. As a result, UMCal deemed it unnecessary to incorporate a design component that would remove the minimal horizontal load.

To size the sub-components of this design, UMCal began at ground level and worked up. The first components to consider are the concrete anchors used to fix the tensioned rope to the ground. During the earlier phases of the project, concrete wedge anchors were identified as capable of resisting the entire load. However, wedge anchors have a threaded stud that would permanently protrude from the ground once installed, creating an undesired safety hazard during normal operation of the facility. For this reason, a double expansion anchor was chosen, as no part of the anchor will extend above the concrete surface once installed. Figure 28 shows the geometry of a typical double expansion anchor before installation.

[redacted]

Figure 28: Double expansion concrete anchor ([redacted] [27].

The capabilities of the double expansion anchor are less than that of the wedge anchor. Consequently, more than one expansion anchor is required in order to resist the tensile load. Confast Fastening Systems offers a $3/4\ [in]$ double expansion anchor with an ultimate pull out strength of $16,962\ [lbf]$ in 3,000 $[psi]$ concrete [28]. The client has specified that the facility's concrete has a strength of 5,800 $[psi]$ or greater [29]. This will increase the performance of the anchor, however this increase is not quantifiable without performing a test as per ASTM standard E488 [30]. Confast recommends 4:1 as minimum safe working load [28]. A four-legged bridle will be used to split the load equally among four concrete anchors. Assuming an evenly split tensile load between the four anchor points, the ground attachment portion of the weight simulation design is rated up to $16,962\ [lbf]$.

Hoist rings were chosen to transfer the load through the bridle and into the concrete anchors. These linkages were chosen because their two rotational degrees of rotational freedom prevent side loading: 360 degrees of swivel around the vertical axis and 180 degrees of pivot around the pin. Avoiding a side load on a connection linkage is critical because it decreases the linkage`s lifting capabilities, which may result in design failure. Figure 29 illustrates how the hoist ring's rotational degrees of freedom ensure an in-line loading.

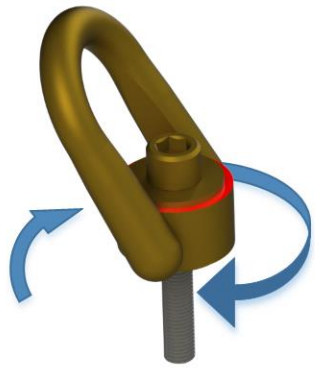

Figure 29: Degrees of freedom in a hoist ring.

In order for the hoist ring to interface with the concrete anchors, the bolt size and thread pitch must match that of the anchor. Furthermore, the hoist ring must also have a load capacity that is equal to or greater than that the concrete anchors so as to not limit the capabilities of the system. The 3/4"– 10 x 3.5 hoist ring from The Crosby Group's catalogue exceeds the requirements with a $7000\ [lbf]$ working load and an ultimate load that is 4.5 times greater [31].

A four-legged bridle is connected to the hoist rings, and will distribute the tensile load equally between the concrete anchors. There are several important characteristics of a bridle that UMCal considered when during the selection of this component. The first characteristic is the relationship between the bridle's loading capabilities and the angle that each "leg" makes with the horizontal. As this angle decreases (shown as ϕ in Figure 30), the loading capacity of the bridle also decreases. In order to ensure an efficient, yet low cost bridle, UMCal specified that each leg of the bridle must form an angle of greater than 60 degrees with the ground.

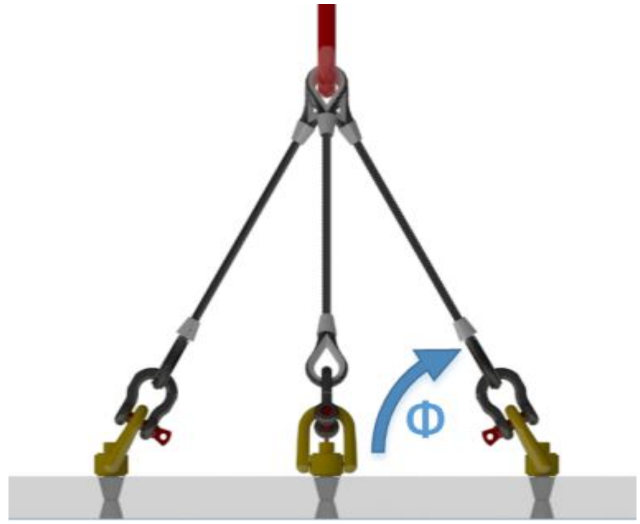

Figure 30: Four-legged bridle for weight simulation design.

The second characteristic is the method by which the wire rope loops rest in the sling link. Since the bottom of the sling link has constant radius geometry, the legs will terminate at different heights above the ground. As each of the legs must be the same length, basic trigonometry will show that the horizontal travel of the legs must be different. Consequently, the concrete anchors will be positioned in a rectangular pattern, rather than a square.

Figure 31: Staggered bridle connections

Peak Trading offers a four legged bridle with ¾” diameter wire ropes that have a working load of 19,000 $[lbs]$ and an ultimate load that is ten times greater. This more than satisfies the safety requirement set by UMCal which states that all tensioned ropes must maintain an ultimate strength ten times greater than the working load. Peak Trading also indicates that the minimum available leg length is 2.75 $[ft]$ [32]. When mounted at 60 degrees with the horizontal, this length conveniently locates the turnbuckle at chest level which is ergonomically desirable for the user.

The turnbuckle is responsible for tensioning the wire rope to the desired load. The turnbuckle consists of a frame and two studs which are attached to either end of the object being tensioned. The studs have opposite threads such that when the turnbuckle frame is turned the studs come together, thereby creating a tension force on the attached objects. UMCal has selected a turnbuckle with a maximum capacity less than that of the concrete anchors. This reduces the risk of surpassing the safe working load of each component's working load, thereby creating a safer design as the turnbuckle will not be able to create a large enough tensile force to damage the wire rope. Wecall offers a turnbuckle with a thread diameter of 1-1/8" with a rated working load of $15{,}200\ [lbf]$ and an ultimate load that is five times greater [33]. In addition, UMCal has recommended a using a hot dipped galvanized turnbuckle, for improved corrosion resistance. Jaw fittings were chosen for the ends of the studs as they are more secure and have higher capacities than that of standard hook fittings [26]. The complete turnbuckle assembly, complete with jaw fittings, is shown in Figure 32.

Figure 32: Turnbuckle with jaw fittings.

Attached to the upper jaw fitting of the turnbuckle, and in line with the vertical load, is a dynamometer. This device correlates the deflection of a flexing beam to the tensile load and displays the reading on a dial gauge. Check Line offers a dynamometer capable of reading loads up to $20{,}000\ [lbf]$, with an ultimate load five times greater. This gauge features an easy to read 10 inch diameter dial which will be located at approximately eye-level with the user [34]. This will allow the user to know the applied load while tensioning the wire rope. As a result, the risk of over loading the system is further reduced. The dynamometer selected for this design is shown in Figure 33

[redacted]

Figure 33: Dillon AP Mechanical Dynamometer ([redacted] [34].

The remaining portion of the weight simulation system incorporates a long wire rope with linkages on both ends that attach the rope to the dynamometer and calibration pylon. Crosby provides an equation that can be used to calculate the required minimum breaking strength of a wire rope [35].

$$Breaking\ Strength\ = \frac{Working\ Load \cdot Design\ factor}{Termination\ efficiency} = \frac{15{,}200 \cdot 10}{0.9} = 169{,}000\ lbf$$

In this equation, the working load was set to the maximum capability of the turnbuckle, the safety factor is set to ten as per UMCal's safety requirements, and the termination efficiency is equal to that of a Flemish eye. Lexco's specifications table for 6x19 galvanized wire rope specifies a 1-3/8" rope thickness with a breaking strength of 172,000 $[lbf]$ [36].

UMCal has chosen a Flemish eye as the style of the rope's terminating loop. The

Flemish eye was chosen as it does not distort the rope and no wire ends are left exposed, therefore the risk of rope un-laying is eliminated. Furthermore, Flemish eye termination has an efficiency of 90% or greater and is considered the most reliable form of loop connections. Flemish eyes are created by dividing the strands of the rope in two and re-laying the strands in opposite direction until a loop is formed. The ends of the rope are then encapsulated by a pressed metal sleeve [35].

To ensure the shape of the loop is preserved, UMCal suggests incorporating a thimble on the inner surface of the loop. The thimble will also spread the load over more wires and protect the wires from crushing and pinching, further increasing the safety and longevity of the system [37]. Screw pin shackles have been chosen as the end linkages over the round pin shackles for two reasons:

- They do not require a locking pin, which is difficult and time-consuming to install
- Only the correct size of pin will fit, thereby increasing safety [26].

Crosby offers a shackle with a $1\,[in]$ pin diameter, which meets the strength requirements. However, the inner opening of this shackle is too small to allow the thimble and wire rope to pass through. Therefore a larger, stronger shackle is required to have a working interface. Crosby's $1\,\tfrac{1}{4}\,[in]$ G209 screw pin shackle has a working load limit of $27{,}000\,[lbs]$, an ultimate load that is six times greater, and a throat opening of $2\,\tfrac{1}{4}\,[in]$, thereby meeting both size and strength requirements [38]. The complete wire rope end attachments are shown in Figure 34.

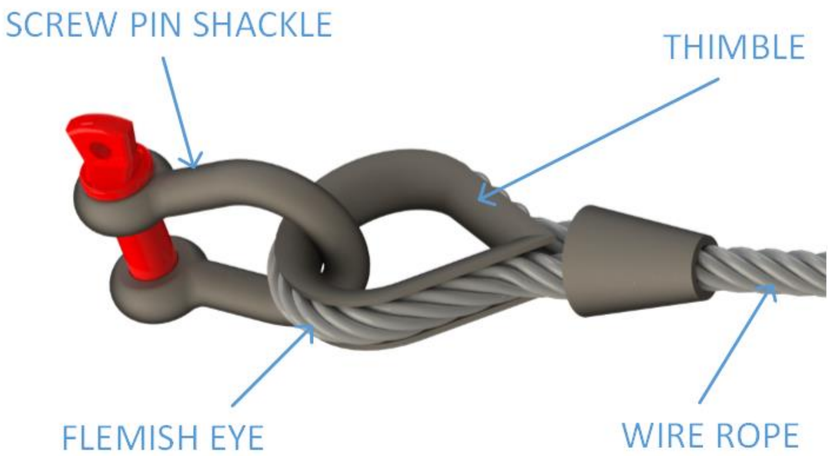

Figure 34: End attachments for wire rope.

TABLE XIX summarizes the safety factor for each of the components in this design. Components that incorporate a wire rope (numbers 3 and 7 in the table) have a safety factor greater than ten to ensure the rope will never break and create a life-threatening situation. All other components, except the concrete anchors, have a safety factor of five. The concrete anchors have a lower safety factor of 4 as it was recommended by the manufacturer, Confast. Coupled with the fact that the concrete pressure is greater than Confast`s anchor specifications, the safety factor is likely much greater than 4.46.

TABLE XIX: WEIGHT SIMULATION DESIGN SAFETY FACTORS

Item #	Item	Ultimate Capacity (lbf)	Actual Load (lbf)	Safety factor
1	Double Expansion Concrete Anchor [28]	16,962	3,800	4.46
2	3/4"-10 x 3.5" Hoist Ring [39]	31,500	3,800	8.29
3	4-Legged Bridle with shackles [32]	190,000	15,200	12.50
4	1-1/8" Galvanized Turnbuckle [33]	76,000	15,200	5.00
5	20,000 [lbf] Dynamometer [34]	100,000	15,200	6.58
6	1-3/8" Screw Pin Shackle [38]	162,000	15,200	10.66
7	6 x 19 IWRC Wire Rope [40]	172,000	15,200	11.32

Cost Analysis

A cost estimation for the weight simulation design was prepared. The dynamometer is the most expensive component of the design, accounting for almost half of the system’s cost. However, UMCal believes that this component, or one with the same capabilities, is essential to ensure the wire rope is not tensioned beyond its limits. Furthermore, the cost estimation does not include costs associated with tax, shipping, duty or man-hours required for assembly. TABLE XX summarizes the quantity and cost of each of the components required to the weight simulation design. The total estimated cost of materials required for the implementation of the weight simulation concept is $2,666.04. There will be no additional manufacturing costs as the components only need to be assembled before use.

TABLE XX: WEIGHT SIMULATION DESIGN COST SUMMARY

Item	Qty	Unit Cost	Total Cost
Double Expansion Concrete Anchor [28]	4	$4.14	$16.56
3/4"-10 x 3.5" Hoist Ring [39]	4	$146.76	$587.04
4-Legged Bridle with shackles [41]	1	$326.52	$326.52
1-1/8" Galvanized Turnbuckle [42]	1	$22.40	$22.40
20,000 Lb Dynamometer [34]	1	$1,156.00	$1,156.00
1-3/8" Screw Pin Shackle [38]	2	$129.96	$259.92
6 x 19 IWRC Wire Rope [40]	12	$21.41	$256.92
1-3/8" Galvanized Wire Rope Thimble [37]	2	$20.34	$40.68
Total			$ 2,666.04

3.4 Calibration Pylon Design

The final component of the system design is the calibration pylon. The calibration pylon is the centerpiece which connects all other components to the thrust frame. It contains mounting points to which the thrust loading from the A-Frame and the engine weight simulation from the turnbuckle assembly will be applied. All forces applied to the calibration pylon will be transferred to the thrust frame so the load cells may be properly calibrated for engine testing.

Design Criteria

There are several key criteria that must be met by the calibration pylon design. Since the pylon links both the thrust simulation and weight simulation loads to the thrust frame, it must be designed to resist both forces while maintaining a desired safety factor. Table XVIII summarizes the requirements of the pylon design, as identified by the team and client.

TABLE XXI: CALIBRATION PYLON DESIGN CRITERIA

Specification	Value
Minimum horizontal load $[lbf]$	105,000
Minimum vertical load $[lbf]$	15,200
Desired Safety factor	2

In addition to the above listed criteria, the pylon should be easily transportable by on site equipment. As such, the design will incorporate transportation via forklift. UMCal has also opted to use A36 structural steel as the main pylon material, given that it is readily available and inexpensive when compared to higher strength metals.

Design Process and Decisions

The first step in designing the pylon was identifying the locations of all the mounting points. These include the mounting points to the thrust frame, thrust simulation load, weight simulation load and lifting lugs. Following the identification of these points, UMCal determined the loading scenarios to which the calibration pylon would be subjected and the required internal reaction forces needed to achieve a static equilibrium. A triangle was selected as the basic shape of the calibration pylon. This shape was selected since the applied loadings were to be applied at the engine's center of gravity, located at the center of the calibration pylon mounting points, and at a height of $20\,[ft]$ above the ground. A preliminary CAD model of the calibration pylon is shown in Figure 35.

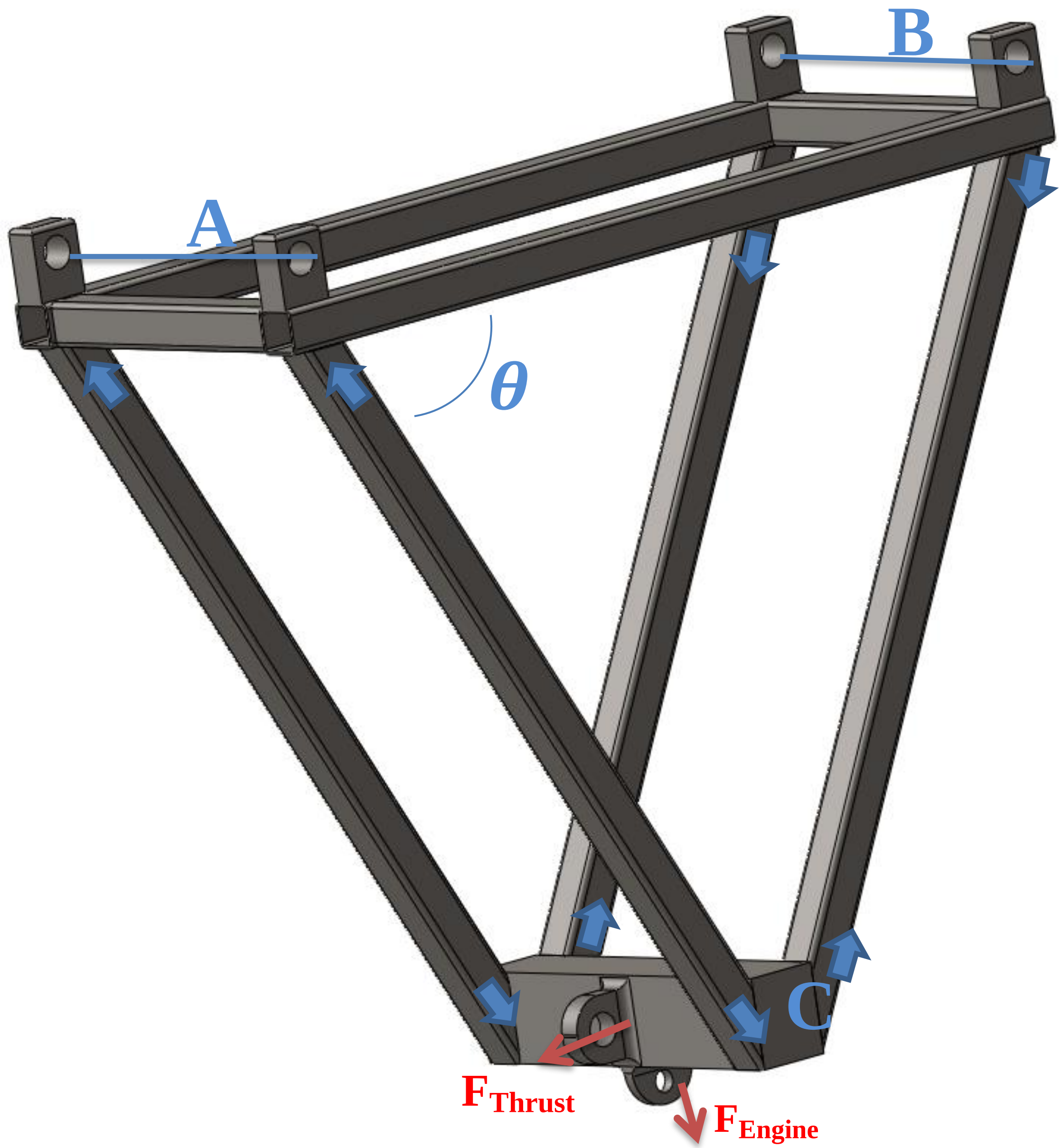

Figure 35: Calibration pylon loading points and reaction forces.

Figure 35 shows that the applied loadings will be transferred through a total of four members, two of which will be in compression and two of which will be in tension. As such, the applied loadings will be split among all four members. The following equations outline the process used to determine the reaction forces at the pin mounting locations. Each equation contains two unknown variables.

$$2 \cdot F_{cb} \cdot \cos(\theta) + 2 \cdot F_{ca} \cdot \cos(\theta) = F_{Thrust}$$

$$2 \cdot F_{cb} \cdot \sin(\theta) + 2 \cdot F_{ca} \cdot \sin(\theta) = F_{Engine}$$

Where:

- F_{cb} is the internal reaction force of members CB
- F_{ca} is the internal reaction force of members CA
- F_{Thrust} is the force of an engine's thrust
- F_{Engine} is the force of the weight of a hanging engine
- θ is the vector angle of F_{cb} and F_{ca}

Solving both equations simultaneously, the solutions to F_{cb} and F_{ca} are obtained. The solutions are shown below.

$$F_{ca} = \frac{F_{Thrust} \cdot \sin(\theta) - F_{Engine} \cdot \cos(\theta)}{4 \cdot \sin(\theta) \cdot \cos(\theta)}$$

$$F_{cb} = \frac{F_{Thrust} \cdot \sin(\theta) + F_{Engine} \cdot \cos(\theta)}{4 \cdot \sin(\theta) \cdot \cos(\theta)}$$

Inputting the values for $F_{Thrust} = 210{,}000\ [lbf]$ and $F_{Engine} = 30{,}400\ [lbf]$, the internal force values for F_{ca} and F_{cb} were determined. The values inputted are twice the size of the required loading, so that a desired safety factor of 2 may be reached. In addition, since the value of θ is known, the reaction forces may be broken into their respective X and Y components. The results of the internal forces of the members are summarized in TABLE XXII.

TABLE XXII: INTERNAL FORCES OF CALIBRATION PYLON

Force	Value
$\boldsymbol{F_{ca}}$	99, 236 $[lbf]$ (Compression)
$\boldsymbol{F_{cb}}$	116, 633 $[lbf]$ (Tension)
$\boldsymbol{F_{xa}}$	48, 269 $[lbf]$
$\boldsymbol{F_{ya}}$	86, 706 $[lbf]$
$\boldsymbol{F_{xb}}$	56, 730 $[lbf]$
$\boldsymbol{F_{yb}}$	101, 906 $[lbf]$

Upon completion of determining the internal forces, a minimum required cross-sectional area for the beams connecting between points AC and BC (as shown in Figure 35) was calculated. The minimum required cross-sectional area was determined by dividing the applied force by the material's yield strength. The equation below was used to determine the minimum cross-sectional area for an individual member.

$$A_{min} = \frac{\sigma}{F}$$

Since 2 of the 4 members will be in compression under the specified loading conditions, a compressive yield strength of $22\ [ksi]$ for A36 structural steel was used. This will ensure that the minimum cross-sectional area needed will allow for a desired safety factor of 2. The minimum cross-sectional area was determined to be $4.51\ [in^2]$. From the obtained cross-sectional area, a nominal size of structural steel tubing was selected with an area closest to that desired, to allow for both manufacturability and availability. Steel tubing was selected for the design due to its ease of assembly for connecting adjoining members together. Furthermore, steel tubing allows for a large peripheral weld at the joints, and a higher torsional stiffness value than that of an I-Beam. Based on the previous calculations, structural rectangular tubing with dimensions of $5''$ x $3''$ x $3/8''$, with a cross-sectional area of $4.78\ [in^2]$ was selected. This selection of structural tubing was chosen as it is it is readily available and will only require alterations at the end of each tubing section for joints, when necessary.

A 3D CAD model was created using the structural members feature in SolidWorks. This allowed UMCal to perform a detailed FEA to validate the aforementioned hand calculations. Several design iterations were necessary before reaching the final design. The iterations showed that gussets, shear plates and larger radius welds were required in certain areas to lower the stress in certain components and subsiquently obtain a safety factor of 2. In addition, tube bracings were added in order to minimize the effective stresses contained within the pylon. The additional bracing also served to increase the overall torsional stiffness of the calibration pylon. Further details regarding how the FEA was performed and the corresponding results are provided in Section 4.3.

Non-structural components were also added to the design in order to meet the design criteria for transporability and ease of set-up. Two rectangular tubing members are located just above the pylon's center of gravity, spread 32 $[in]$ apart from the inside faces. These tubes allow for transportation of the pylon via forklift. Four legs were also designed of similar tubing size so that the structure is capable of supporting itself in an upright position when the pylon is not in use. Lifting lugs are located above the calibration pylon mounting points, which allow the pylon to be raised into position via chains. Finally, since the calibration pylon is made entirely from A36 structural steel, the pylon will be manufactured using only weldments to connect any adjoining members together. The calibration pylon was designed in such a manner that disassembly is not required, and it will remain intact when not in use. The final calibration pylon design including the outline of main components is seen in Figure 36. The total weight of the calibration pylon was calculated to be 2331.50 $[lbs]$.

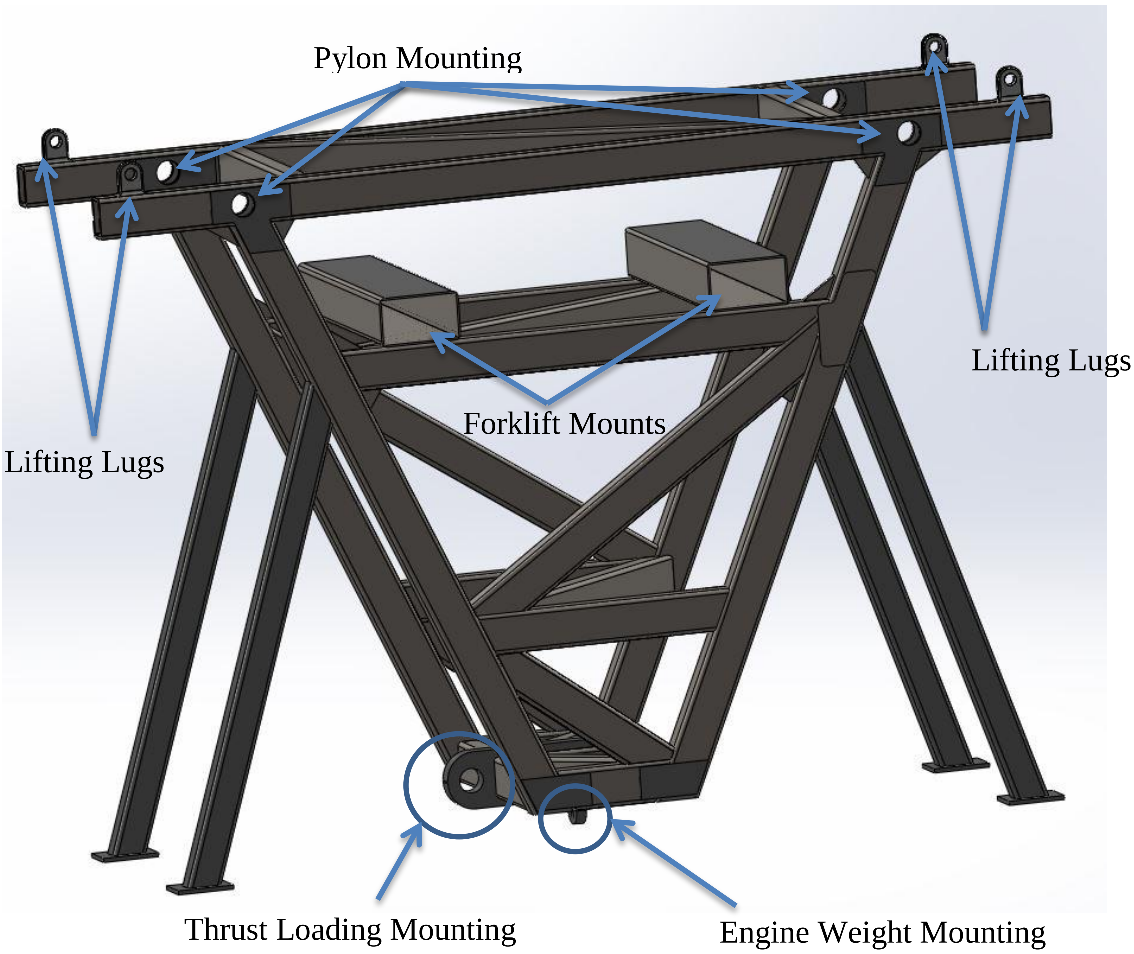

Figure 36: Final design of calibration pylon.

Cost Analysis

In addition to the ensuring that the calibration pylon design meets the requirements of the client, a cost analysis was performed to verify whether the pylon is a feasible design from a financial perspective. Research has been completed to provide a cost analysis of the overall material required for the design and is included in TABLE XXIII. The total cost of materials needed for the calbration pylon is estimated at $5,768.85.

TABLE XXIII: CALIBRATION PYLON COST SUMMARY

Item	Qty	Unit Cost	Total Cost
Tube 18" X 5" X 3" X 3/8" [18]	3	$ 54.36	$ 163.08
Tube 24" X 5" X 3" X 3/8" [18]	2	$ 72.48	$ 144.96
Tube 32" X 5" X 3" X 3/8" [18]	2	$ 96.64	$ 193.28
Tube 34" X 5" X 3" X 3/8" [18]	2	$ 102.68	$ 205.36
Tube 36" X 5" X 3" X 3/8" [18]	2	$ 108.72	$ 217.44
Tube 48" X 5" X 3" X 3/8" [18]	1	$ 144.96	$ 144.96
Tube 61" X 5" X 3" X 3/8" [18]	2	$ 184.22	$ 368.44
Tube 65" X 5" X 3" X 3/8" [18]	4	$ 196.30	$ 785.20
Tube 73" X 5" X 3" X 3/8" [18]	3	$ 220.46	$ 661.38
Tube 75" X 5" X 3" X 3/8" [18]	1	$ 226.50	$ 226.50
Tube 83" X 5" X 3" X 3/8" [18]	4	$ 250.66	$ 1,002.64
Tube 120" X 5" X 3" X 3/8" [18]	2	$ 362.40	$ 724.80
Tube 30" X 10" X 6" X 3/8" [18]	2	$ 200.10	$ 400.20
Steel Plate 12" X 12" X 1/4" [18]	14	$ 14.66	$ 205.24
Steel Plate 12" X 12" X 1/2" [18]	2	$ 29.53	$ 59.06
Steel Plate 12" X 12" X 1 1/4" [18]	1	$ 69.63	$ 69.63
Steel Plate 12" X 24" X 2" [18]	1	$ 196.68	$ 196.68
Total			$ 5,768.85

3.5 Design Summary

The proposed re-design of the calibration system is capable of simulating both engine thrust and weight. The engine simulation loads act upon a calibration pylon that is suspending from the trust frame. The pylon serves to transfer the loads into the thrust frame and in turn the load cells being calibrated. Figure 37 is a labeled render of the final calibration system design within the GE TRDC facility.

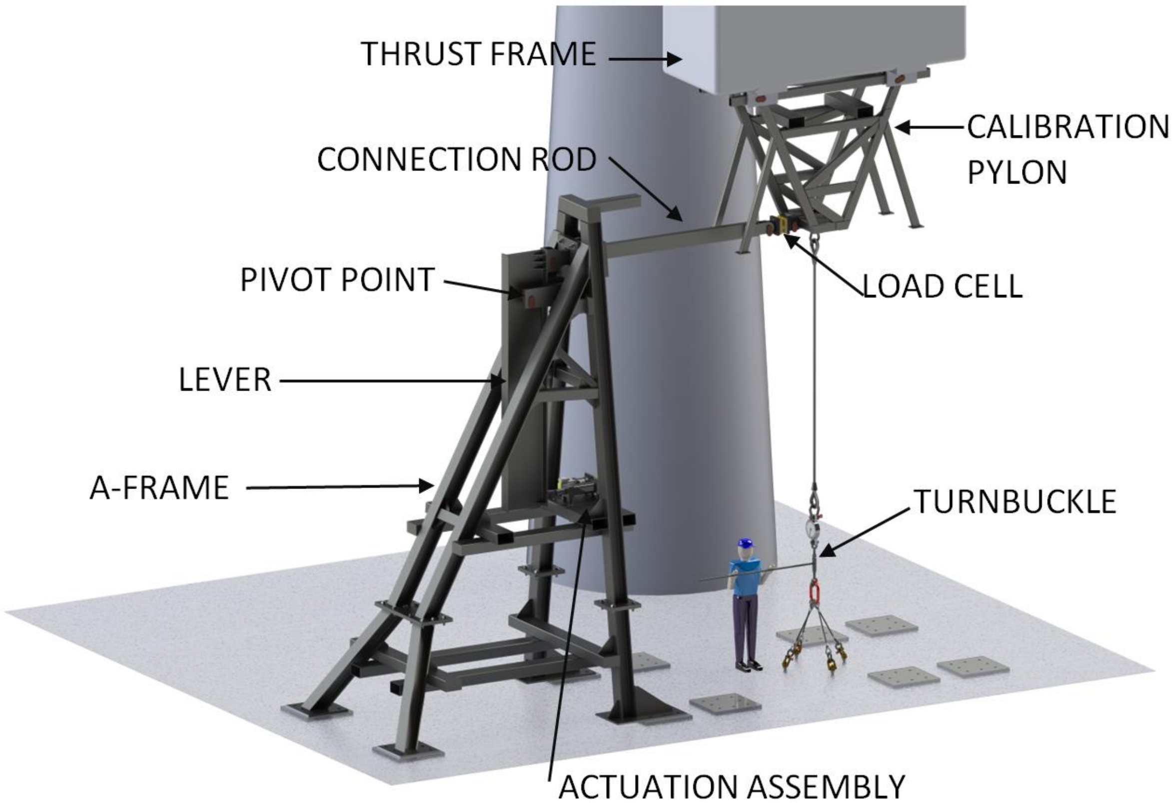

Figure 37: Final render of calibration system design.

Upon completing the final design, UMCal created an Engine Thrust Calibration Setup and Pre-Inspection Procedure was created to compliment the proposed design. This procedure documents all steps required to install each component of the calibration design. Not included in this document are the steps required to gather data from the measurement instrumentation since specifying measurement instrumentation was outside of the scope of this project. A copy of the Engine Thrust Calibration Setup and Pre-Inspection Procedure is provided in Appendix A.

Another project deliverable requires UMCal to provide design drawings, therefore assembly drawings of each system component were created and are included in Appendix B. These drawings include a bill of materials and critical dimensions for each assembly.

4. Finite Element Analysis

To validate the preliminary hand calculations provided in Section 3, FEA was performed using the Solidworks Simulation software. For all three components, several iterations were required to achieve designs that meet the safety factor criteria while remaining low cost, manufacturable and simple. To decrease computation time, non-structural features of the design were excluded from the analysis during the design iterations. In addition, a number of assumptions were made while performing the FEA.

- Materials selected are free of any voids or defects within the structure's geometry.
- Materials selected are expected to behave in a Linear Elastic fashion.
- All joints that are welded are considered to contain welds of equal or greater strength than that of the welded material.
- Stress concentration at inifinitely sharp edges contained within the model were neglected as they were exceedingly large and unrealistic.
- As the number loading cycles is minimal (1-2 times per year), fatigue was not considered in the analysis of the calibration pylon.

4.1 Structural Lever

After performing hand calculations to determine the general required geometry of the structural lever, FEA was to validate the calculations and identify areas of peak stress that require reinforcement. To simulate the forces exerted on the lever, a thrust load of $105{,}000\ [lbf]$ was applied at the top pin attachment points and an actuation load of $17{,}500\ [lbf]$ was applied at the bottom pin attachment points, both in the same direction. The pivot diameter of the lever was fixed. This simulates the effect on the lever in a steady position at maximum thrust load. A summary of loads and constraints used to simulate the real loading of the lever are shown in TABLE XXIV.

TABLE XXIV: LOADS AND CONSTRAINTS FOR LEVER FEA

Loads				Constraints	
Type	**Location**	**Orientation**	**Value $[lbf]$**	**Type**	**Location**
Force	Top Pin Attachment Points	0	105,000	Fixed	Pivot rod interfacing diameter
Force	Bottom Pin Attachment Points	0	7,500		

The lever loading scenario and applied mesh is shown in Figure 38.

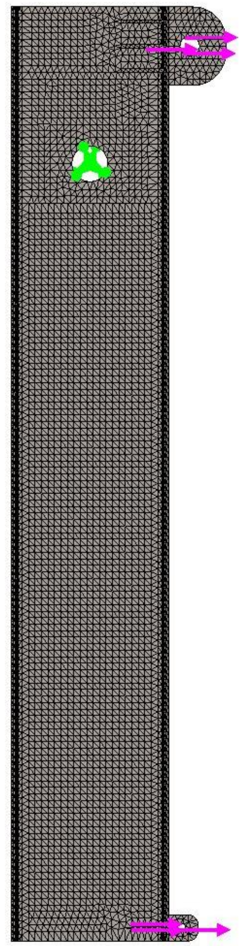

Figure 38: Lever mesh used for FEA.

The resulting stresses in the lever when subjected to the above loading scenario are shown in Figure 39. The view on the left shows the stress distribution including a safety factor of 2. Therefore, all red points are indications of reaching one half the yield strength of A36 structural steel. The view on the right shows the true stress distribution, where areas of red would indicate surpassing the actual material yield strength.

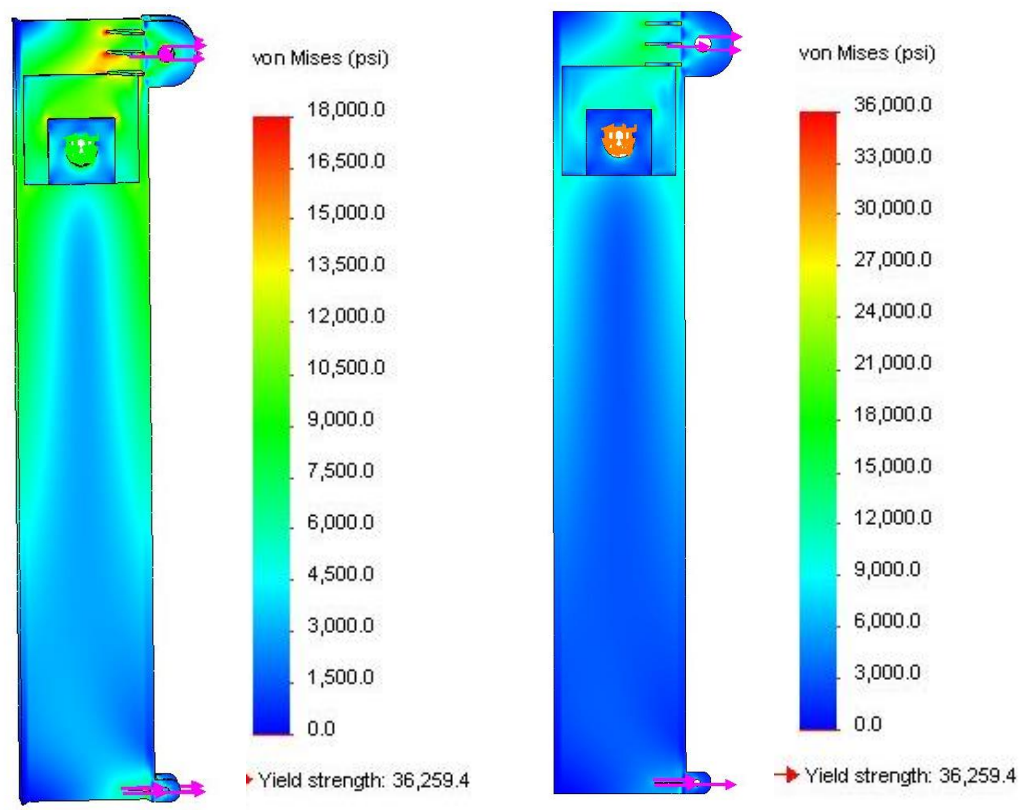

Figure 39: Stress distribution in lever.

The resulting lever design above was the result of many design iterations. Although sized for the appropriate bending moment, the I-beam experienced areas of high peak stresses around areas of load application. As such, the lever was reinforced with A36 structural steel plate around the pivot bar mounting area, in addition to gussets between the flange and web at thrust and actuation load points. These modifications effectively reduced the peak stresses in these areas.

The area of maximum stresses in the design is the upper flange area where the thrust load is applied. The use of gussets significantly reduces the stresses around the flange to web radius, however slightly higher than desired loads are still indicated at the gusset corners. As shown in Figure 40, peak loads of approximately $25{,}000\ [psi]$ are shown in these areas. UMCal has deemed these peak stresses acceptable as the load is still well below yield, the area of concern is minimal, and some stress is attributed to the sharp corner geometry of the model which would be lessened by a weld radius.

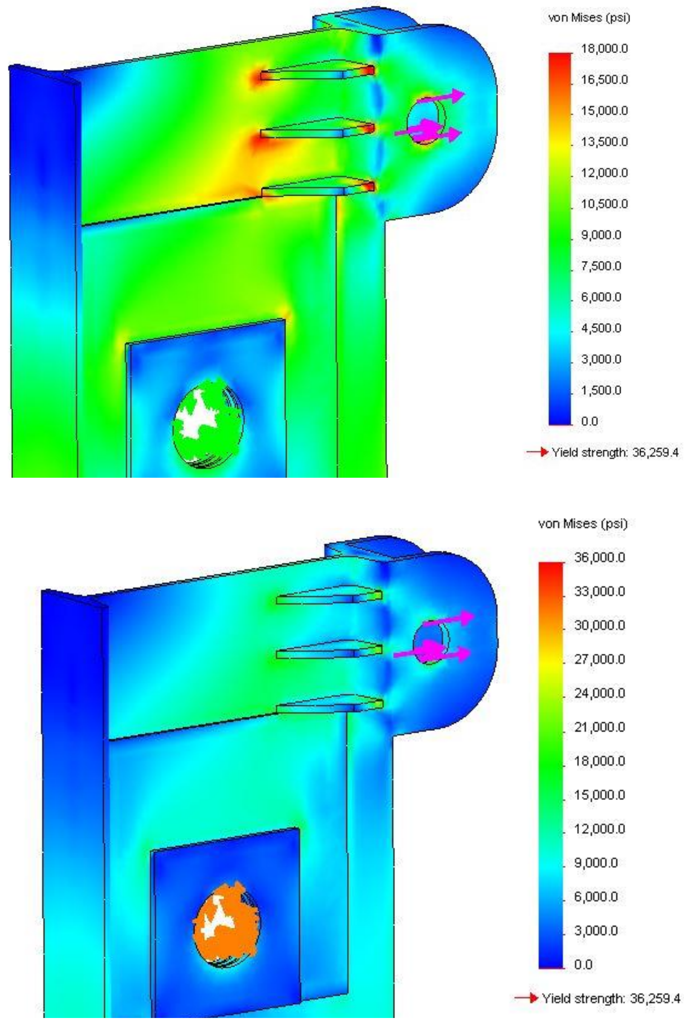

Figure 40: Peak stress region in lever.

FEA results were also used to approximate the maximum lever deflection. Results anticipate the maximum deflection of the lever to be $0.194\ [in]$ at the bottom lever actuation point. With a deflection of just over $3/16\ [in]$, over an $11\ [ft]$ beam span, this number is not unreasonable. Given that the lever does not reach yield, this deflection is acceptable. The lever deflection is visually represented in Figure 41.

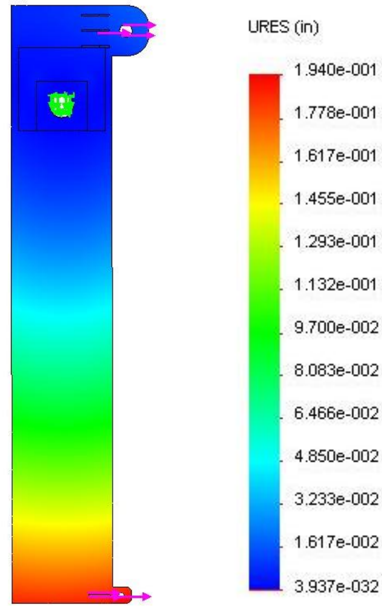

Figure 41: Deflection in lever.

A summary of FEA results is shown in TABLE XXV, including an estimated weight of the design.

TABLE XXV: RESULTS SUMMARY OF LEVER FEA

Parameter	Value
Max Deflection $[in]$	0.196
Max Stress $[ksi]$	25
Safety factor	1.44
Total Weight $[lbs]$	1,245

Due to the peak stress concentrations located at the reinforcing gussets, the actual safety factor of the design is 1.44. However, as the areas undergoing this stress are small and easier to repair or reinforce, this lower than desired factor is acceptable. The remainder of the lever meets or exceeds the desired safety factor of 2.

4.2 A-Frame

FEA was performed first on the currently existing A-frame design, to examine whether it would withstand an in-line thrust loading of 105,000 $[lbf]$. The loads and constraints used for this simulation are summarized in TABLE XXVI.

TABLE XXVI: LOADS AND CONSTRAINTS FOR A-FRAME FEA (IN-LINE)

Loads				Constraints	
Type	**Location**	**Orientation**	**Value [lbf]**	**Type**	**Location**
Force	Top plate	0	105,000	Fixed	Mounting points to concrete

A single 105,000 [lbf] force was applied, representing the applied thrust loading. The bottom of each mounting plate had a fixed constraint applied where it contacts the concrete pad. A graphical summary of the mesh, applied loads, and constraints for this simulation is shown in Figure 42.

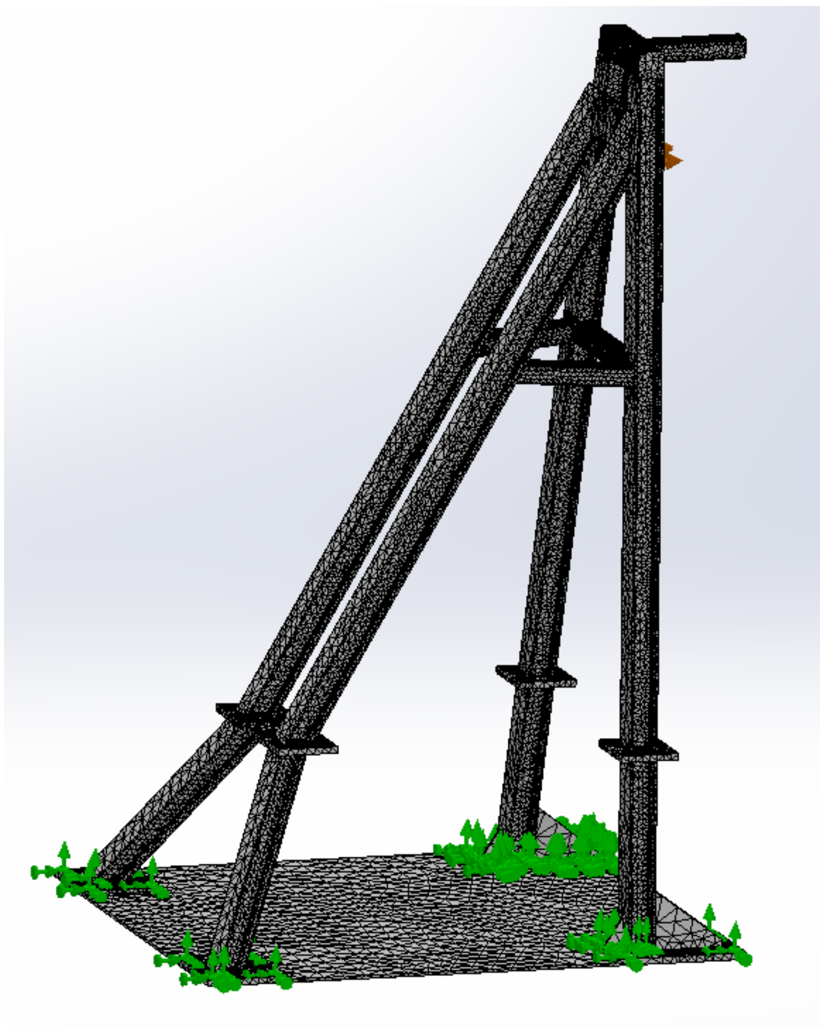

Figure 42: A-frame mesh used for FEA (in-line thrust simulation).

Figure 43 displays the results of this FEA simulation, with the high stress limit set to one half the yield strength, approximating a safety factor of 2. It is evident that while the bulk of the structure is capable of withstanding the higher load, reinforcement in key areas is necessary to ensure safety. While the FEA results show that certain areas of the frame are stressed beyond the safety factor, this is likely due to limitations of both the simulation software and the proficiency level with which UMCal is able to define the model constraints. Specifically, the bond between the upper plate and the four main structural members is not accurately represented by the simulation.

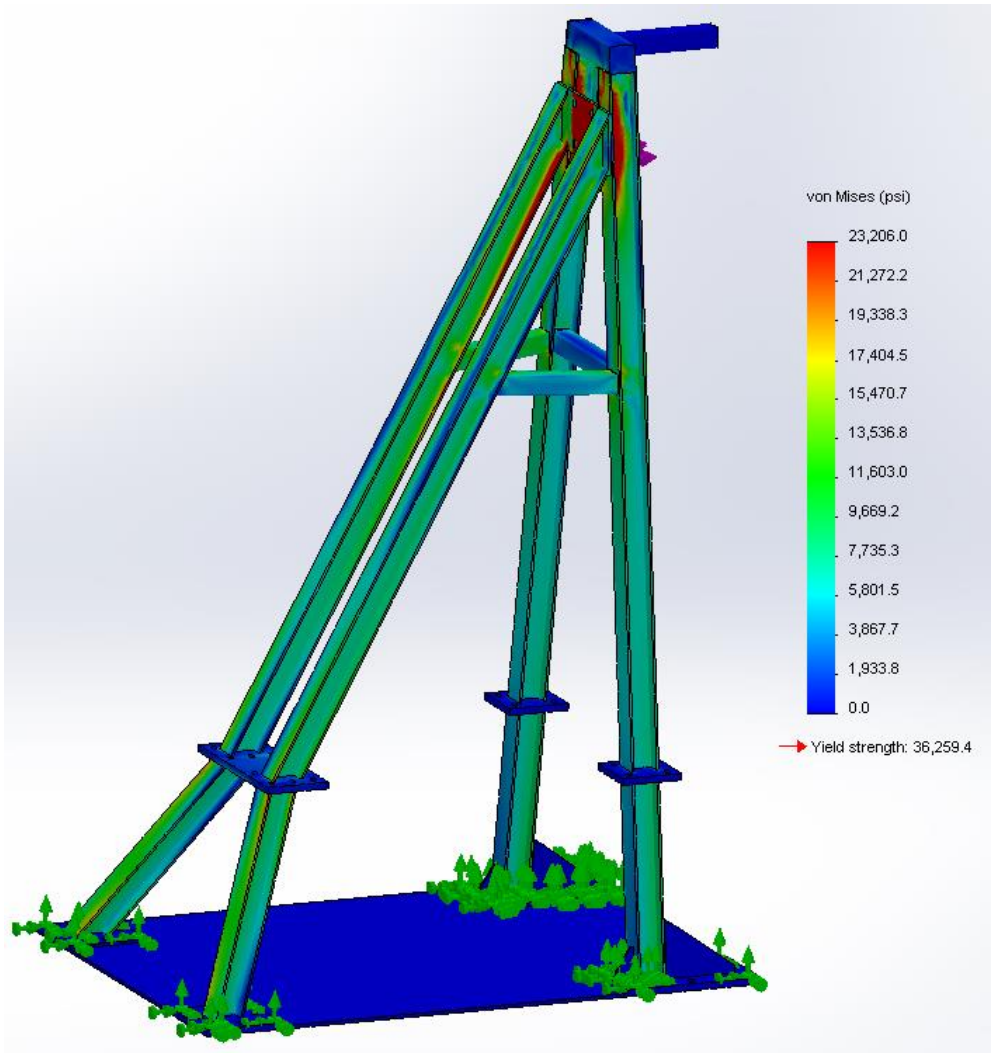

Figure 43: Stress distribution in A-frame (in-line thrust simulation).

With a final reinforced A-frame design complete, a second study was conducted to verify the load capacity of the modified A-frame. A summary of the applied loads and constraints for this study is shown in TABLE XXVII.

TABLE XXVII: LOADS AND CONSTRAINTS FOR A-FRAME FEA (LEVER)

Loads				Constraints	
Type	**Location**	**Orientation**	**Value** $[lbf]$	**Type**	**Location**
Force	Pivot bar	0	120366	Fixed	Mounting points to concrete

As before, a single force is applied horizontally towards the front of the A-frame. In this case, the load is applied to the pivot bar set into its two mounting blocks. Fixed constraints are again applied to each of the four mounting pads. The mesh conditions and suppressed features are unchanged from the previous simulation. A graphical summary of the mesh, applied loads, and constraints is shown in Figure 44.

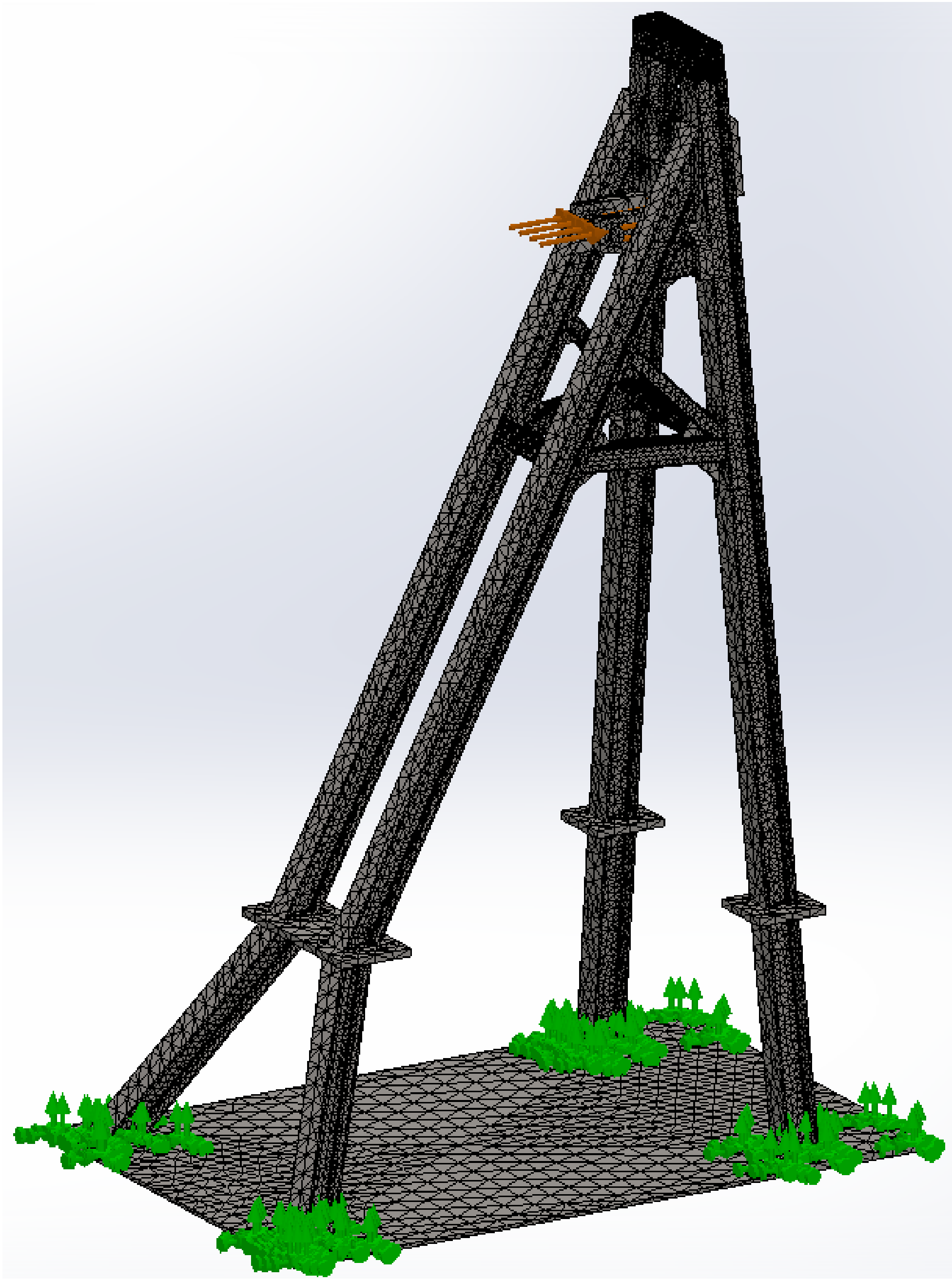

Figure 44: A-frame mesh used for FEA (lever thrust simulation).

Figure 45 displays the results of this FEA simulation, with the high stress limit again set to one half the yield strength of the material.

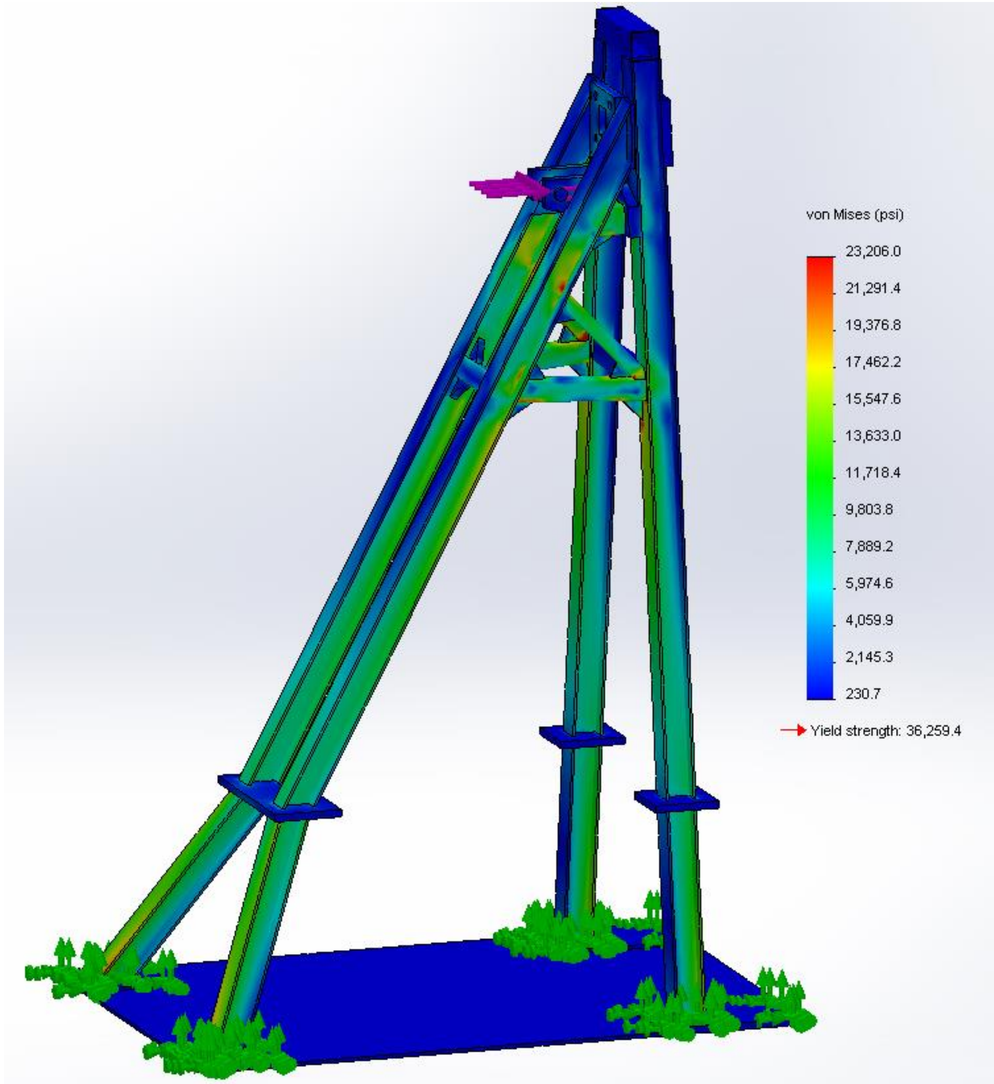

Figure 45: Stress distribution in A-frame (lever thrust simulation).

As the study was unable to take into account any welds in the finished structure, there are areas of stress concentrations at sharp corners where two surfaces meet. A closer view of these areas is shown in Figure 46. In addition to not fully accounting for welded surfaces, the simulation does not allow for the top bolted connection across the structural beams to be adequately modeled. As such, the load transfer across the front structural members is not entirely accurate, and in reality the load would be more evenly distributed across the entire frame which would further reduce the stresses.

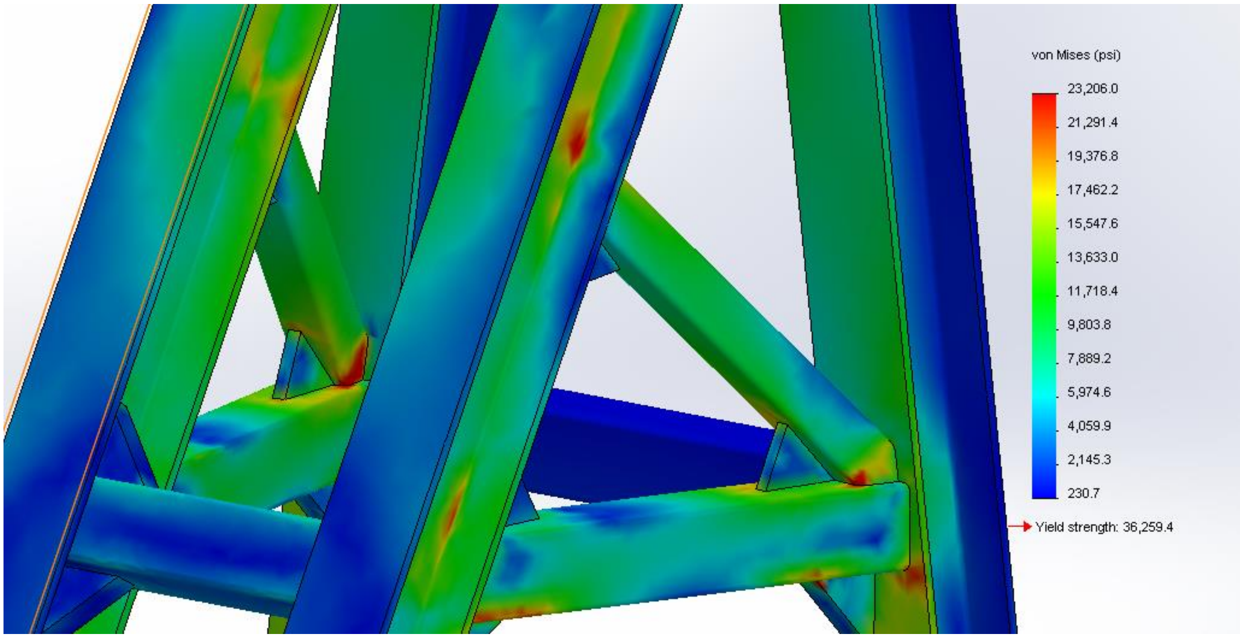

Figure 46: Stress concentrations in A-frame.

An analysis of the A-frame deflection was also performed in order to ensure that stresses were being properly distributed throughout the model and that the frame was not deflecting in a manner that would cause harm to the lever's actuation. The results of this analysis are shown in Figure 47.

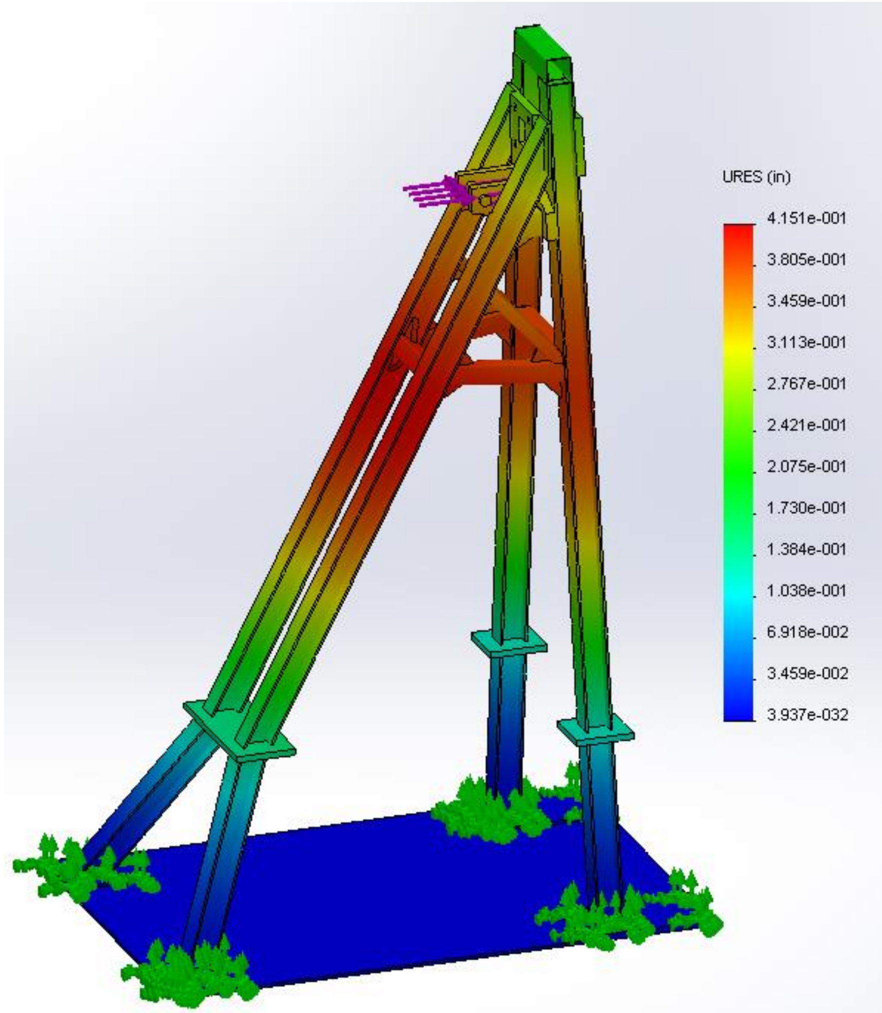

Figure 47: Deflection in A-frame due to the lever thrust simulation.

The complete simulation results are summarized in TABLE XXVIII:

TABLE XXVIII: RESULTS SUMMARY OF A-FRAME FEA

Parameter	Value
Max Deflection $[in]$	0.4151
Max Stress $[ksi]$	24.5
Safety factor	1.89

The safety factor obtained by the FEA simulation is slightly lower than 2. Although this is lower than the desired value, the result has been deemed acceptable for four reasons. First, inaccuracies in the model prevent loadings from being distributed in a completely accurate fashion. Second, the removal of weld beads and fillets creates stress concentrations where they otherwise would not occur. Third, the frame modifications were designed using the yield strength rather than the ultimate strength. Fourth, compressive yield strength was used in all calculations, rather than the higher tensile yield strength. These factors add an additional margin of safety that UMCal believes more than justifies the results of the FEA simulation.

4.3 Pylon

The first step in performing the FEA was specifying the loads and constraints to apply to the model. The FEA must verify that the calibration pylon is capable of resisting both the thrust simulation and weight simulation forces as it is responsible for transferring these forces into the thrust frame. The thrust simulation will apply a load of $105{,}000\ [lbf]$ in the horizontal direction, and the engine weight simulation will apply a load of $15{,}200\ [lbf]$ in the vertical direction. TABLE XXIX summarizes the loads and constraints used during the FEA.

TABLE XXIX: LOADS AND CONSTRAINTS FOR CALIBRATION PYLON FEA

Loads				Constraints	
Type	**Location**	**Orientation**	**Value** $[lbf]$	**Type**	**Location**
Force	Thrust Load Mount	0	105,000	Fixed Hinge	Pylon mounting points
Force	Engine Weight Mount	-90	15,200	Fixed Hinge	Pylon mounting points

Figure 48 illustrates the FEA stress distributions throughout the calibration pylon when subjected to a loads defined in TABLE XXIX. From the FEA analysis results, it is clear that the final design will able to comfortably withstand the necessary applied loadings, as the majority of the calibration pylon stress levels are far below the allowable stress of $36\ [ksi]$. The areas indicated in blue have stress levels of $12.5\ [ksi]$, less than half the allowable stress.

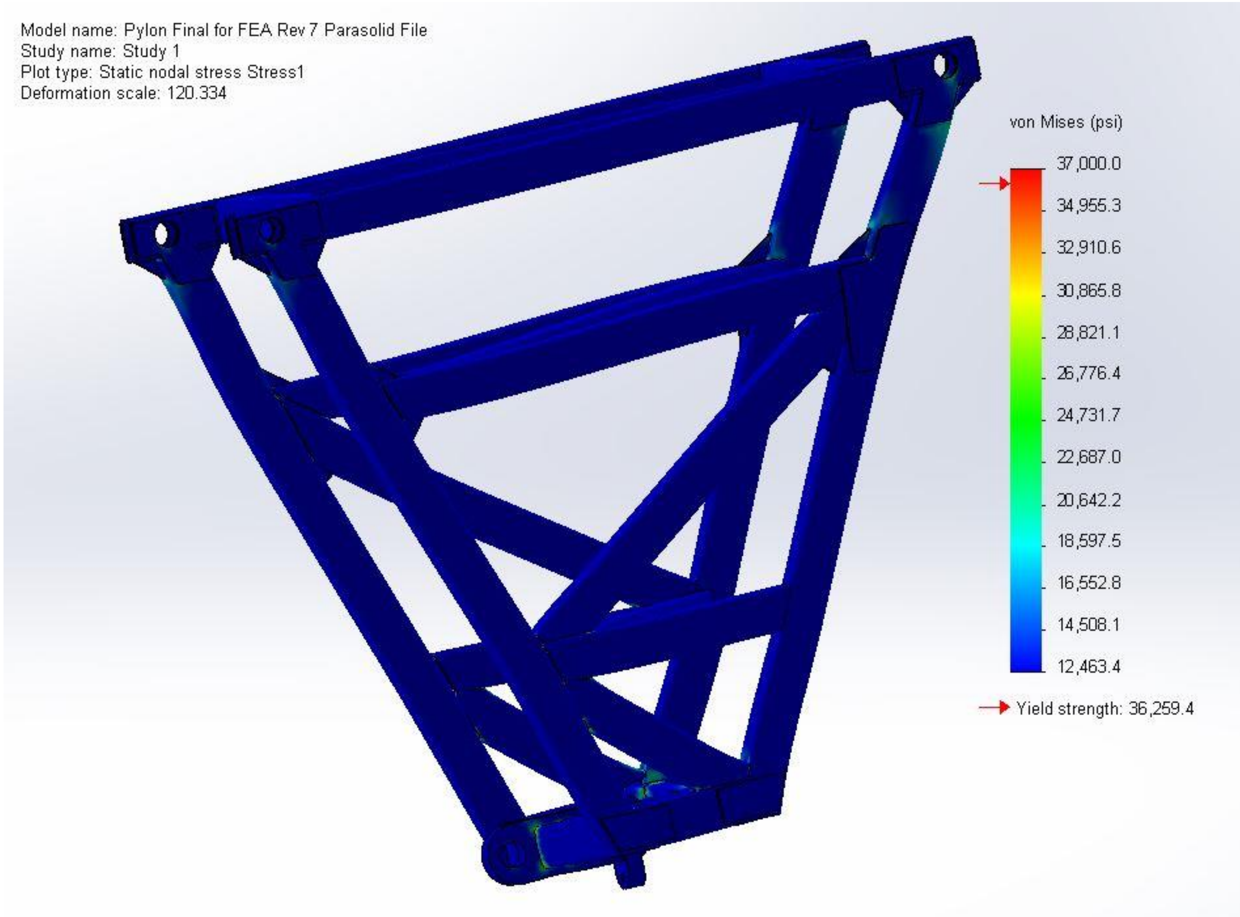

Figure 48: Stress distribution in calibration pylon.

However, there are certain areas of the calibration pylon design containing stress levels that are large in comparison to the rest of the calibration pylon, and are illustrated in Figure 49. The high stress levels in these areas are likely due to the sharp edges. Once the connecting joints are welded together creating a fillet radius, the high stresses would be reduced substantially. When disregarding the stress due to sharp edges, the maximum stress attained in the calibration pylon was computed to be 18.8 $[ksi]$. As such, the actual safety factor of the calibration pylon is 1.92.

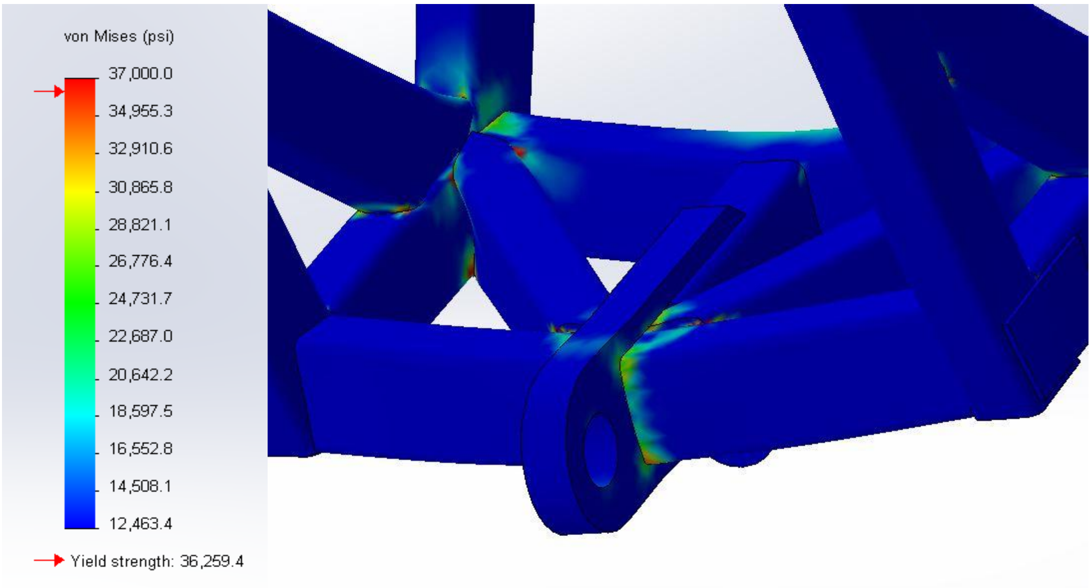

Figure 49: Stress concentrations in calibration pylon.

Figure 50 displays the overall displacement of the calibration pylon when subjected to the normal loading conditions. The total computed displacement of the calibration pylon under normal loading conditions was determined to be just above $0.07734\ [in]$.

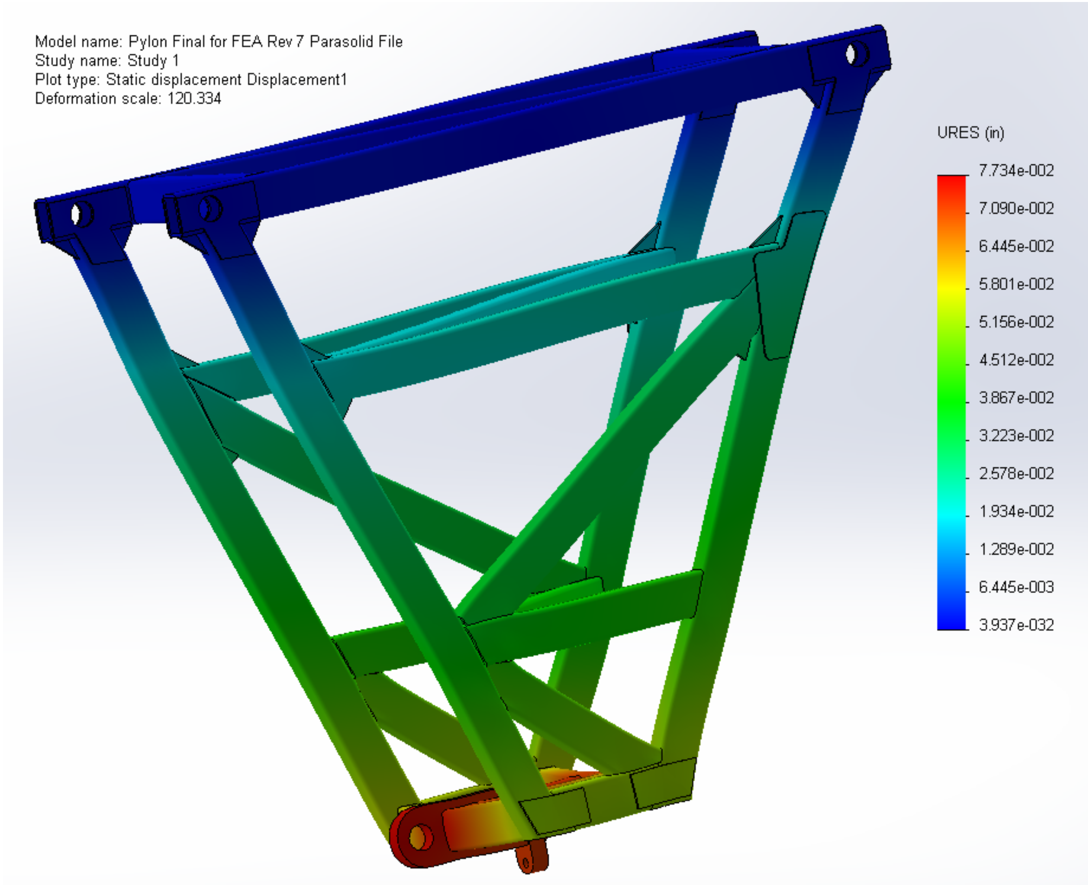

Figure 50: Deflection in calibration pylon.

TABLE XXX summarizes the results of the FEA conducted on the final revision of the calibration pylon.

TABLE XXX: RESULTS SUMMARY OF A-FRAME FEA

Parameter	Value
Max Deflection $[in]$	0.07734
Max Stress $[ksi]$	18.8
Safety factor	1.92
Total Weight $[lbs]$	2,331.50

Although the computed safety factor is slightly under the desired safety factor of 2, the calibration pylon design was designed using the selected material's yield strength and not the material's ultimate strength. As such, the team feels that the calibration pylon design will be sufficient in withstanding the applied loading scenarios.

5. Design Cost Summary

Having finalized the overall design of a new calibration system, UMCal has prepared a complete cost breakdown of the material and equipment required to put this design into use. This breakdown is shown in TABLE XXXI.

TABLE XXXI: DESIGN COST SUMMARY

Section	Component	Cost
Lever	S24 X 80 I-beam	$ 600.00
Lever	ASTM A36 Steel Plate, 1/4”	$ 39.69
Lever	ASTM A36 Steel Plate, 1/2”	$ 33.99
Lever	ASTM A36 Steel Plate, 3/4”	$ 37.88
Lever	AISI-C1144 Stress Relieved Steel Round Bar, 3-1/2 OD x 18” L	$ 202.12
Lever	Structural 1020 Steel Round Tube, 4” OD w/ 01/4” wall thickness	$ 60.06
Lever	Isostatic TU Sleeve Bearing P/N 501157. 3-1/2” ID x 3-11/16” OD x 3” L	$ 19.00
	LEVER TOTAL	**$992.74**
Actuation	E-Drive Eliminator HD618-06 Ball Screw Linear Actuator	$ 19,495.00
Actuation	Baldor BSM90N-2150AF Brushless Servo Motor	$ 2,202.00
Actuation	Baldor GBSM90-MRP155-10 Servo Motor Gearhead	$ 2,224.00
	ACTUATION TOTAL	**$23,921.00**
Connecting Rod	Steel Plate 12” X 12” X 2”	$108.17
Connecting Rod	Steel Plate 12” X 12” X 1.5”	$166.66
Connecting Rod	Steel Plate 12” X 18” X 1”	$120.46
Connecting Rod	Tube 150" X 8" X 3" X 5/16"	$654.00
Connecting Rod	Interface Model 1632AJH-100k load cell	$7603.00
Connecting Rod	Interface load cell mating cable, 50 ft	$195.00
	CONNECTING ROD TOTAL	**$8,847.29**
A-Frame	3” A36 steel plate, 2x4 ft	$1592.72
A-Frame	2” A36 steel plate, 4x4 ft	$2,875.54
A-Frame	1” A36 steel plate, 1x1 ft	$93.51
A-Frame	1” A36 steel plate,4x8 ft	$1,254.40
A-Frame	½” A36 steel plate, 4x2 ft	$210.72
A-Frame	1-1/2”-6 UNC x16” SAE J429 Grade 8 Hex Cap Screw	$525.56
A-Frame	1-1/2” USS Standard Washer, Zinc-Plated Steel, package of 5	$14.04
A-Frame	Ultra-Coated Grade 8 Steel Hex Nut, 1-1/2”-6 UNC	$32.80
A-Frame	8x8x0.25” A36 structural steel, 2 ft	$100.36
A-Frame	6x4x0.25” A36 structural steel, 2 ft	$50.76
A-Frame	4x4x0.25” A36 structural steel, 6 ft	$106.92

Section	Component	Cost
A-Frame	6x6x0.25” A36 structural steel, 4 feet	$115.84
A-Frame	8x6x0.25” A36 structural steel, 8 feet	$365.12
A-Frame	1”-8 UNC x 10” Grade 5 Zinc-Plated Steel Hex Head Cap Screw	$52.64
A-Frame	Ultra-Coated Grade 8 Steel Hex Nut, 1”-8 UNC, package of 5	$9.12
A-Frame	1”-8 UNC x 2-1/4” Grade 8 Hex Cap Screw	$15.80
	A-FRAME TOTAL	**$7,415.85**
Weight Simulation	Double Expansion Concrete Anchor	$16.56
Weight Simulation	3/4"-10 x 3.5" Hoist Ring	$587.04
Weight Simulation	4-Legged Bridle with shackles	$326.52
Weight Simulation	1-1/8" Galvanized Turnbuckle	$22.40
Weight Simulation	20,000 Lb Dynamometer	$1,156.00
Weight Simulation	1-3/8" Screw Pin Shackle	$259.92
Weight Simulation	6 x 19 IWRC Wire Rope	$256.92
Weight Simulation	1-3/8" Galvanized Wire Rope Thimble	$40.68
	WEIGHT SIMULATION TOTAL	**$2,666.04**
Pylon	Tube 18" X 5" X 3" X 3/8"	$ 163.08
Pylon	Tube 24" X 5" X 3" X 3/8"	$ 144.96
Pylon	Tube 32" X 5" X 3" X 3/8"	$ 193.28
Pylon	Tube 34" X 5" X 3" X 3/8"	$ 205.36
Pylon	Tube 36" X 5" X 3" X 3/8"	$ 217.44
Pylon	Tube 48" X 5" X 3" X 3/8"	$ 144.96
Pylon	Tube 61" X 5" X 3" X 3/8"	$ 368.44
Pylon	Tube 65" X 5" X 3" X 3/8"	$ 785.20
Pylon	Tube 73" X 5" X 3" X 3/8"	$ 661.38
Pylon	Tube 75" X 5" X 3" X 3/8"	$ 226.50
Pylon	Tube 83" X 5" X 3" X 3/8"	$ 1,002.64
Pylon	Tube 120" X 5" X 3" X 3/8"	$ 724.80
Pylon	Tube 30" X 10" X 6" X 3/8"	$ 400.20
Pylon	Steel Plate 12" X 12" X 1/4"	$ 205.24
Pylon	Steel Plate 12" X 12" X 1/2"	$ 59.06
Pylon	Steel Plate 12" X 12" X 1 1/4"	$ 69.63
Pylon	Steel Plate 12" X 24" X 2"	$ 196.68
	PYLON TOTAL	**$5,768.85**
TOTAL		**$49,611.47**

The total raw cost for all material and equipment is $49,611.77. Applying applicable Manitoba tax rates, the total then becomes $56,061.30. The client has specified a desired payback period of no more than five years in order for this project to be financially feasible, [redacted]

After discussion with the client regarding the cost of capital for this project, the client was unable to provide a figure for use and suggested that UMCal exercise good judgment as to a reasonable value. Cost of capital was therefore established as 6.35%, from data obtained by means of the Value Line database for the aerospace and defense industries [43]. This rate infers a 3.75% premium over the current Canadian risk-free rate of 2.6%, which is based on the ten-year Government of Canada bond yield. This rate is an appropriate estimate due to a lower than average proportion of debt of 21% within the sector, whereas the average proportion of debt for industry as a whole is approximately 30%.

The Net Present Value (NPV) of all costs associated with contracting the calibration technicians and equipment was calculated by means of the following equation, where CF is the cash flow to be adjusted, R is the cost of capital, and n is the year in which the cash flow occurs:

$$NPV = \sum_{n=1}^{5} \frac{CF}{(1+R)^n}$$

Using this equation to calculate the NPV for five years of calibration provides an NPV of [redacted] Detailed calculations may be found in Appendix C.

When the NPV of performing calibration for the next five years by means of contracting outside services is compared to the cost of UMCal's proposed design, the fact that labour costs have not been analyzed must be taken into account. As UMCal was not able to obtain a reasonable estimate for the cost of labour associated with building the proposed design, these costs would need to be incorporated into the cost estimate before a more accurate payback period could be established. As the costs for material alone are already higher than the NPV of contracting services for the next five years, the payback period of the proposed design would certainly be greater than five years and therefore not financially viable as per the client's specifications.

6. Recommendations

Based on the results of the NPV analysis, it is UMCal's recommendation that the client continue to bring in contractors to perform annual calibration. As the cost of the proposed design already yields a payback period above the five years which was specified by the client, the addition of labour costs would only further increase this period beyond the desired length. However, UMCal recommends that in the event the client were to perform calibration multiple times a year, the cost analysis be revisited as this would significantly change the NPV of contracting outside calibration services and therefore reduce the payback period. In addition, UMCal would like to stress that this design is largely conceptual in nature, and has primarily been undertaken as a feasibility study in order to assess whether such a design would be both financially and physically viable. UMCal therefore recommends that a further, in-depth design review be conducted by the client in order to verify all assumptions and calculations before proceeding with any implementation of this design.

7. Lessons Learned

Throughout the duration of this project the members of UMCal were continuingly learning. The lessons learned by the team members can be divided into two categories, technical information and good design process practices.

In terms of technical knowledge, UMCal became aware of everything from load cell calibration procedures to the common industry names for rigging fixtures. Before this project, UMCal's knowledge of calibration procedures, measuring instrumentation, force actuators and connection fixtures was very rudimentary. After three months of constant research the team members are more confident with these subjects. If ever presented with a similar challenge in the future, the members would be able to find component details easily and would be able to recommend suppliers and manufacturers with minimal searching.

In terms of best practices for the design process, UMCal members learned several valuable lessons. The first of these lessons occurred during the project definition phase of the project. Each team was required to define a list of project needs and determine quantifiable specifications for each. At first, this seemed like a simple task that would only be completed to meet the requirements laid out in the PDR rubric. However, once the team members began defining the specifications, it quickly became apparent that this was a difficult task, and was certainly necessary in order to create a successful design. Defining specifications forced the team members to think about the information that they would require from the client for the design phase, before the design phase even began. Without the incentive to gather information early, UMCal would have entered the design phase unprepared and the project schedule certainly would have suffered drastically. Subsequently, the members of UMCal learned from this experience and are appreciative of the course structure for aiding with the design process.

UMCal members also learned the importance of assigning a lengthy amount of time to the concept generation phase. When the project schedule was first completed, it seemed as though too much time was allocated towards generating concepts and not enough time set aside for developing the final design. Once the concept generation time came about, UMCal quickly became aware of just how much time is required to iterate through the different stages of screening, refinement, weighted evaluations, secondary refinement and preliminary analysis, each of which are required to ensure a suitable final design concept is chosen. The team members would have appreciated more time to evaluate the final design, but they all agree that putting an extended amount of time into the concept generation phase was well worth the effort. The members learned that it is important to exhaust all possible concepts before moving into the design phase as it would be devastating to think of a great idea when it is too late to implement.

Throughout the concept generation phase, the members of UMCal, each in their own way, learned the significance of documentation. For example, when conducting research, team members learned the importance of recording what valuable information was found and where it was found. There were countless occasions when team members were forced to find the same valuable information more than once. During this time, UMCal also learned that report drafts should be completed well before the submission date, approximately one week. The feedback provided from technical communication assistants and project advisors is priceless and sufficient time must be allocated to allow for their feedback and subsequent edits to the report.

The final and most significant lesson that the team members benefited from transpired during the final design stage of the project. Since the MECH 4680 course has a strict deadline, there is no flexibility in the project completion date. In order to meet the assigned deadline, it is imperative that the project scope was not allowed to increase throughout the project duration. It is essential to remember that the engineering design project resembles a feasibility analysis for the project rather than a design process that results in a ready-to-use design with all the added-value features. If given an infinite timeline, one could spend an infinite time refining the design and perfecting the analysis. However, time is not abundant and sometimes one must make a decision with the information that is available to them, and move on to the next task. The members of UMCal were forced to learn this lesson the hard way, as the time remaining was diminishing but the tasks to complete were multiplying.

The members of UMCal have identified countless more lessons than the ones mentioned here, each of wish they consider valuable and will strive to learn from as they are faced with more challenging projects in their future careers. UMCal is appreciative of the learning opportunity that they have been given and have worked endlessly to gain all benefits.

8. Summary

In conjunction with the client, UMCal has identified a detailed list of project objectives, as well as needs and specifications that have served to accurately define the client's problem. UMCal has used the identified specific needs to successfully create a conceptual design for a new calibration system for the GE TRDC facility. At the client's request, UMCal has created a conceptual design that is capable of testing the entire range of thrust desired, capable of simulating various engine weights and integrates with the currently utilized thrust frame.

The final design was divided into three separate components: engine thrust simulation, engine weight simulation, and calibration pylon. For each respective component, a thorough stress analysis using both hand calculations and computer software methods such Finite Element Analysis was performed to verify the validity of the designs.

After conducting the analyses of each component, UMCal was able to conclude that the respective designs were capable of withstanding the necessary loading required for the calibration testing. In addition, UMCal has designed all components specifically so they may comply with all specifications set forth in the needs and specifications section of the report.

Detailed CAD models of all the final design components have been created, along with a list of detailed drawings. Furthermore, a preliminary cost analysis of the materials required to fabricate each component has been created. With this report, UMCal has completed the design generation phase of this project and is submitting this report to the client as a feasibility analysis of the implementation of a new calibration system design for the GE TRDC facility.

9. Works Cited

[1] KGS Group. (2013, Feb.). *G.E. Research Development & Test Center Jet Engine Test Stand* [Online]. Available: http://www.kgsgroup.com/pdf/awards/ACEC_GE_AviationEngineTRDC.pdf. [October 19, 2013].

[2] *Instrument Calibration & Standards Laboratory Calibration Procedure*, General Electric Aviation Standard CP-F-3.8, 2007.

[3] R. Baillie, D. Whynot, J. Viramontes and M. Hibbs (private communication), September 17, 2013.

[4] R. Baillie, (2013, Sept. 23). "Project Deliverables and Requirements". Personal E-Mail.

[5] R. Baillie, (2013, Sept. 30) "Questions for Project Definition Report". Personal E-Mail.

[6] R. Baillie, D. Whynot, J. Viramontes and M. Hibbs, (private communication), October 15, 2013.

[7] R. Baillie, (2013, Oct. 30). "Force vs. Deflection". Personal E-Mail.

[8] F. P. Beer, E. R. Johnston, J. T. DeWolf, and D. F. Mazurek, *Mechanics of Materials, 6th Edition*, New York, NY: McGraw-Hill, 2012.

[9] Brunswick Steel. (2012, Sep. 23). *Reference Catalogue* [Online]. Available: http://www.brunswicksteel.com/reference-book.html. [November 26, 2013].

[10] S. Hudak. (1995). *Primer for Sleever Bearing Design* [Online]. Available: http://www.metalstop.com/engine/Bearing_Design_Guide.pdf. [November 28, 2013].

[11] Brunswick Steel, (private communication), November 14, 2013.

[12] E-Drive Actuators. (2013, Oct. 15) *Eliminator HD* [Online]. Available: http://www.edriveactuators.com/eliminator.html. [November 21,2013].

[13] Baldor. (2013, Jan. 9). *AC Brushless Servo Motors* [Online]. Available: http://www.baldor.com/products/servomotors/n_series/bsm_nseries.asp. [November 22, 2013].

[14] Baldor. (2013, Feb. 11). *Servo Rated Gearheads* [Online]. Available: http://www.baldor.com/products/servomotors/servo_gearheads/gearheads.asp. [November 27, 2013].

[15] W. Campbell, (2013, Nov. 29) "Budgetary E-drive quotation". Personal E-Mail.

[16] Motion Industries. (2013). *Electric Motors & Controls* [Online]. Available: http://www.motionindustries.com. [November 29,2013].

[17] Interface Advanced Force Measurement. (2013). *1600 Gold Standard Calibration LowProfile Load Cell* [Online]. Available: http://www.interfaceforce.com/index.php?1600-Gold-Standard%C2%AE-Calibration-LowProfile%E2%84%A2-Load-Cell-Calibration-Grade-Load-Cell&mod=product&show=15. [November 3, 2013].

[18] Speedy Metals. (2013). *Steel* [Online]. Available: http://www.speedymetals.com/. [November 19, 2013].

[19] Nucor Fastener. (2013) *Shear Strength* [Online]. Available: http://nucor-fastener.com/Files/PDFs/TechDataSheets/TDS_013_Shear_Strength.pdf. [November 24, 2013].

[20] MetalsDepot. (2013). *Steel Plate* [Online]. Available: http://www.metalsdepot.com/products/hrsteel2.phtml?page=plate&LimAcc=%20 &aident=. [November 19, 2013].

[21] McMaster-Carr. (2013). *Screws and Bolts* [Online]. Available: http://www.mcmaster.com/#screws/=pmedj4. [November 23, 2013].

[22] McMaster-Carr. (2013). *Washers* [Online]. Available: http://www.mcmaster.com/#standard-washers/=pmeelo. [November 23, 2013].

[23] McMaster-Carr. (2013). *Nuts* [Online]. Available: http://www.mcmaster.com/#nuts/=pmeed4. [November 23, 2013].

[24] MetalsDepot. (2013). *Steel Square Tube* [Online]. Available: http://www.metalsdepot.com/products/hrsteel2.phtml?page=sqtube&LimAcc=%20&aident=. [November 23, 2013].

[25] MetalsDepot. (2013). *Steel Rectangle Tube* [Online]. Available: http://www.metalsdepot.com/products/hrsteel2.phtml?page=rttube&LimAcc=%20&aident=. [November 24, 2013].

[26] Northwest Industrial Supply. (2011, June 11). *Sling and Rope Efficiencies* [Online]. Available: http://www.nwisupply.com. [November 16, 2013].

[27] MSC. (2013). *Concrete Anchors* [Online]. Available: http://www.mscdirect.com/product/68001320. [November 24, 2013].

[28] Confast. (2013, March 30). *Concrete Double Expansion Anchor* [Online]. Available: http://www.confast.com/products/technical-info/double-anchor.aspx. [November 22, 2013].

[29] R. Baillie, (2013, Nov. 3). "CG location of engine". Personal E-Mail.

[30] *Standard Test Methods for Strength of Anchors in Concrete* [Online]. ASTM Standard E448/E448-10. Available: http://www.astm.org/Standards/E488.htm. [November 26, 2013]

[31] The Crosby Group. (2013). *Hoist Rings* [Online]. Available: http://www.thecrosbygroup.com/Portals/0/docs/BrochuresandFlyers/HoistringEng.pdf. [November 21, 2013].

[32] Peak Trading Corporation. (2013). *4 Leg Wire Rope Bridles* [Online]. Available: ttp://www.peaktrading.com/productpages/default.aspx?ProductPageId=1403. [November 15, 2013].

[33] Wecall. (2013). *Turnbuckles* [Online]. Available: http://www.wecallinc.com/_datasheets/turnbuckles.pdf. [November 18, 2013].

[34] Check Line. (2013). *Mechanical Dynamometer* [Online]. Available: http://www.checkline.com/cable_tension_meters/AP%20Dynamometer. [November 21, 2013].

[35] The Crosby Group. (2013). *Wire Rope End Terminations* [Online]. Available: http://www.thecrosbygroup.com/Portals/0/docs/manuals/9992320.pdf. [November 22, 2013].

[36] Lexco. (2013). *6x19," Galvanized EIPS IWRC Wire Rope* [Online]. Available: http://www.lexcocable.com/6x19_galvanized.html. [November 19, 2013].

[37] US Cargo Control. (2013). *Wire Rope Thimbles* [Online]. Available: http://www.uscargocontrol.com/Rigging-Supplies-Hardware/Wire-Rope-Thimbles. [November 22, 2013].

[38] Westech Rigging Supplies. (2013). *Crosby, 1-3/8" G-209 Screw Pin Shackle, #1018598* [Online]. Available: http://www.westechrigging.com/shackle-screw-pin-crosby-g-209-138.html#.UpuRW8RDt-N. [November 23, 2013].

[39] McMaster-Carr. (2013). *Hoist Rings* [Online]. Available: http://www.mcmaster.com/#hoist-rings/=pecqie. [November 21, 2013].

[40] Unirope (private communication), November 7, 2013.

[41] Wecall (private communication), November 7, 2013.

[42] WireCo WorldGroup. (2013, March 15). General Purpose U.S. and Canada Price Schedule No. GP-313. [Online]. Available: http://www.wirecoworldgroup.com. [November 17, 2013].

[43] Value Line. (2013, Jan.). *Cost of Capital by Sector* [Online]. Available: http://w4.stern.nyu.edu/~adamodar/New_Home_Page/datafile/wacc.htm. [November 24, 2013].

Appendix A

Table of Contents

1. Introduction

In addition to the design of the engine thrust calibration system, UMCal created a supplementary work procedure document. This document covers the details of the mechanical system setup prior to calibration. In addition, the document outlines recommended inspection of the system equipment prior to use. This work procedure is to be used in addition to [redacted]

2. Calibration Work Procedure

A copy of the UMCal developed work procedure is attached in the pages to follow.

[redacted]

General Electric Test Research & Development Centre

[redacted]

StandardAero

[redacted]

for the University of Manitoba
Faculty of Mechanical Engineering
IDEAS Program

Sponsored by

WestCaRD

[redacted]

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Appendix B

Table of Contents

List of Drawings

1. Introduction

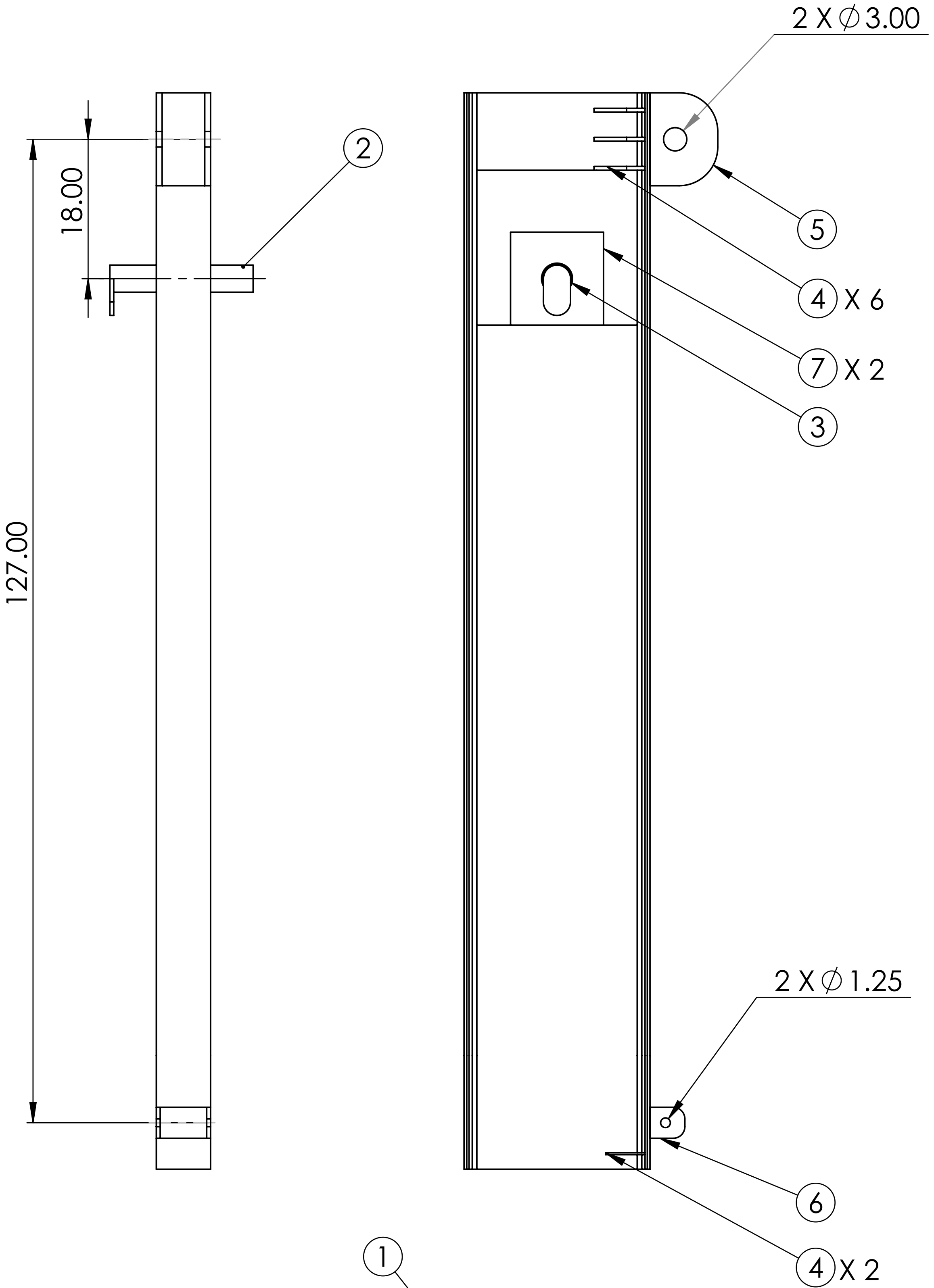
During initial conversation, the client requested that UMCal provides assembly drawings of the calibration re-design at the time of report submission. To ensure detailed drawings that accurately represent the system design, complete CAD models were required. CAD models for individual components of the design were created first and then integrated into a model of the facility to guarantee the design is fully compatible with the facility interface locations.

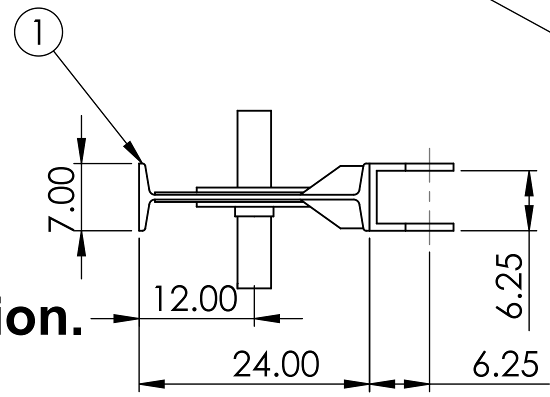
Once the CAD models were complete, the assembly drawings were created. The intent of these drawings is to provide the client with further details of the design and critical design or manufacturing features. Therefore each drawing incorporates a bill of materials and critical dimensions.

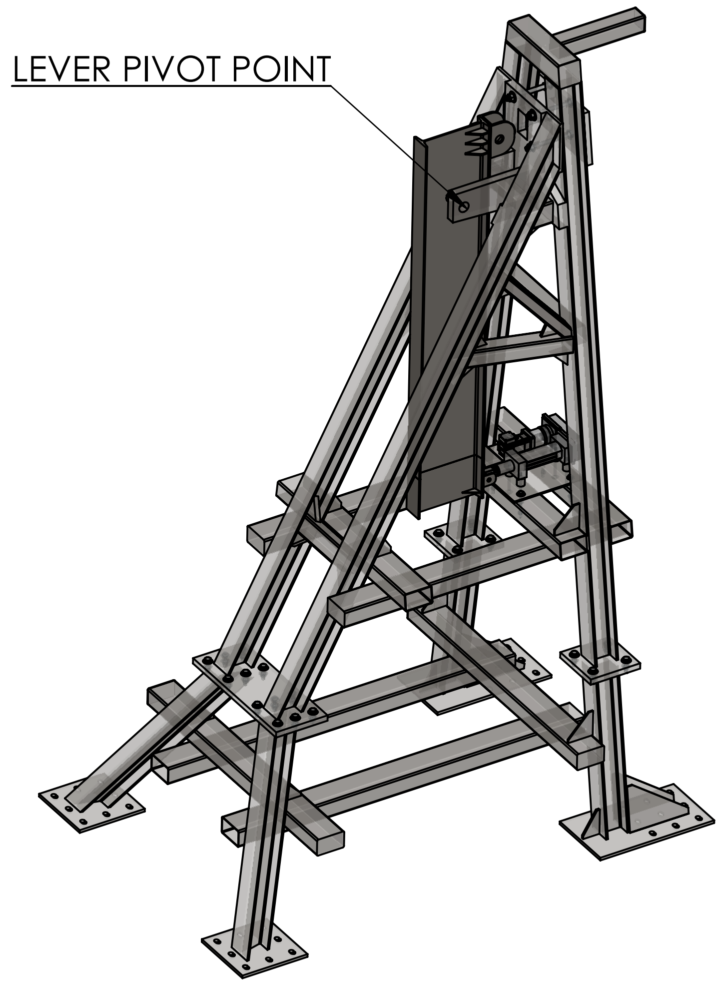
2. System Drawings

System drawings for the lever, connection rod, A-frame, weight simulation, and pylon are provided in drawings 1, 2, 3, 4 and 5 on the following pages.

#	PART DECRIPTION	QUANTITY
1	S24 X 80 I-BEAM, 12FT	1
2	ANSI-C1144 ROUND BAR, 3.5IN DIA	1
3	ISOSTATIC TU SELF LUBRICATING SLEEVE BEARING	1
4	GUSSET, 1/4IN ASTM A36 STEEL PLATE	8
5	UPPER MOUNT, 3/4IN ASTM A36 STEEL PLATE	1
6	LOWER MOUNT, 1/2IN ASTM A36 STEEL PLATE	1
7	REINFORMENT PLATE, 1/2IN ASTM A36 STEEL PLATE	2





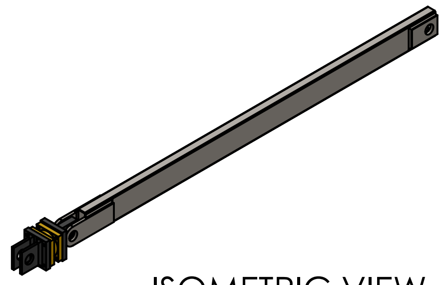


LEVER POSITION IN A-FRAME

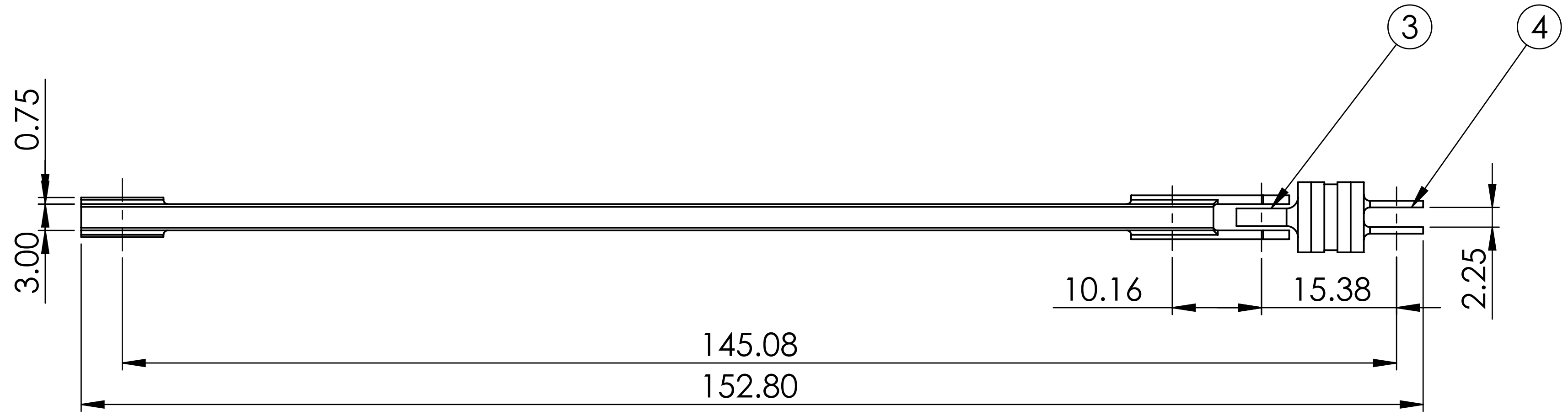
Consulting Group

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE: THRUST SIMULATION LEVER	
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: MACH ±0° 30' TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.002	MATERIAL: DRAWN	LK	30/11/2013	SIZE A3	DESCRIPTION: Assembly View
	FINISH: CHECKED	HM	01/12/2013		REV 0
	DRAWING NOT TO SCALE	NUMBER		SHEET 1 OF 5	B-4

#	PART DESCRIPTION	QUANTITY
1	1IN SHEAR/MOUNT PLATE	2
2	0.75IN SHEAR/MOUNT PLATE	2
3	MALE FLANGE	1
4	FEMALE FLANGE	1
5	INTERFACE MODEL 1632AJH-100K LOAD CELL	1



ISOMETRIC VIEW



SolidWorks Student Edition.
For Academic Use Only.

UJMal Consulting Group

TITLE: CONNECTION ROD

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: MACH ±0° 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.002

		NAME	DATE
MATERIAL:	DRAWN	LK	30/11/2013
FINISH:	CHECKED	DG	01/12/2013
DRAWING NOT TO SCALE	NUMBER		

SIZE A3

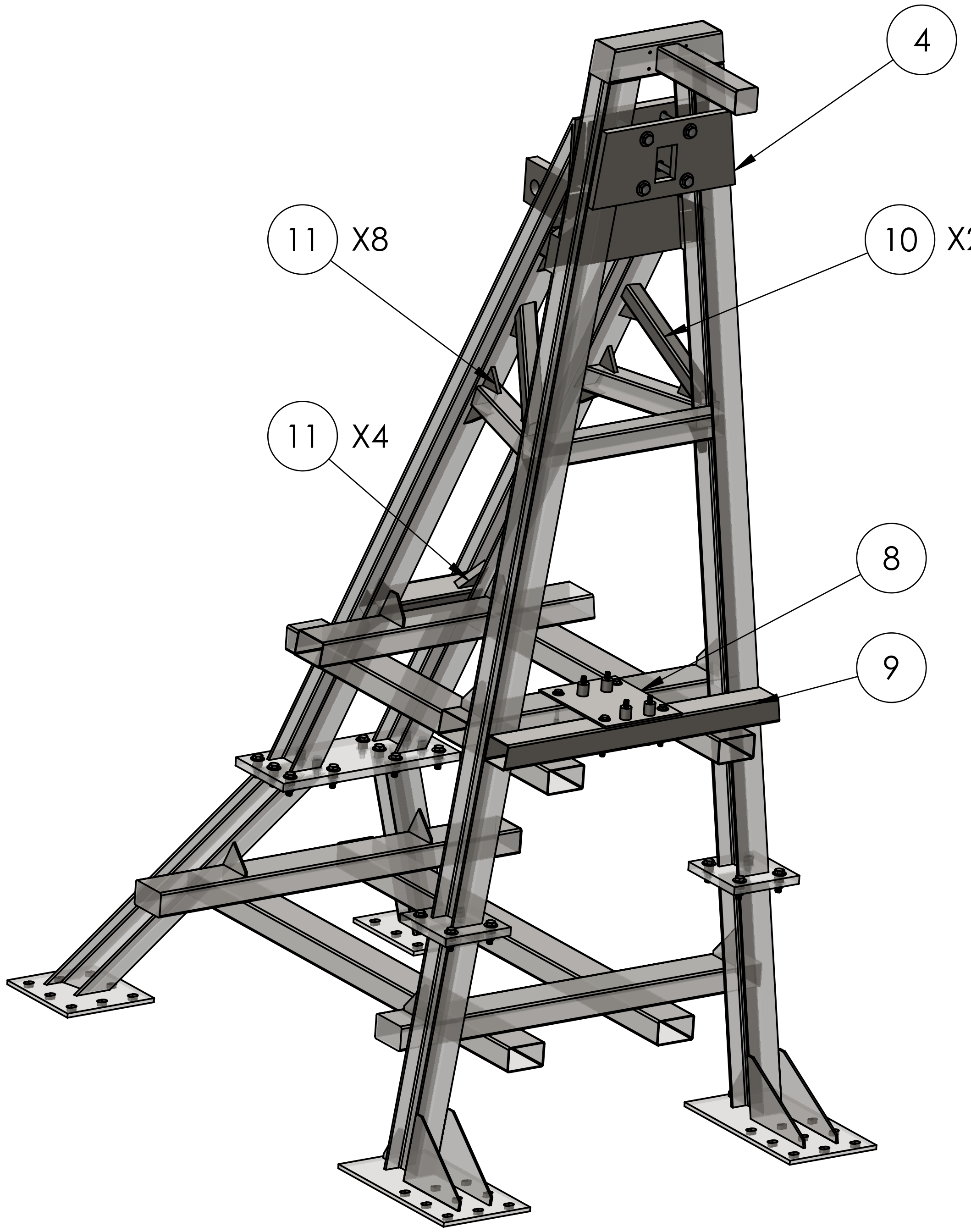
DESCRIPTION: Assembly View

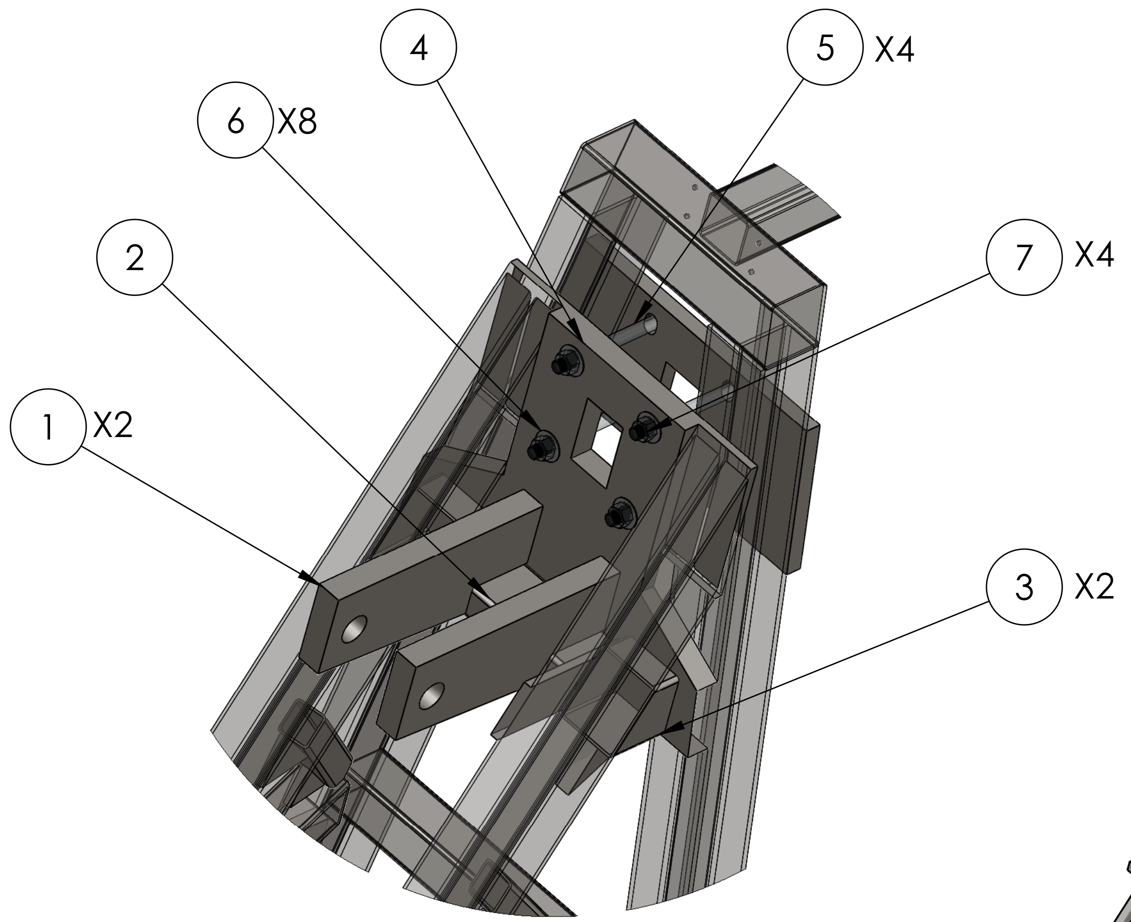
REV 0

SHEET 2 OF 5

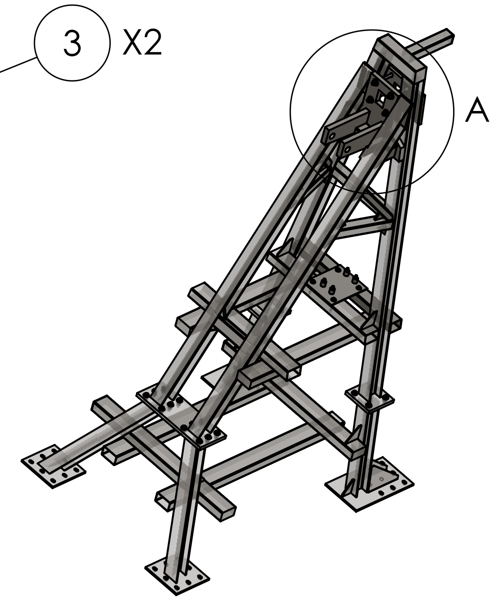
B-5

#	PART DESCRIPTION	QUANTITY
1	LEVER PIVOT SUPPORT	2
2	6IN X 4IN X 1/4INSTRUCTURAL MEMBER	1
3	8IN X 8IN X 1/4IN STRUCTURAL MEMBER	2
4	STRUCTURAL PLATE, 2IN	2
5	1 1/2-6, UNC, 16IN GRADE 8 STEEL HEX CAP	4
6	1 1/2 USS STANDARD WASHER, ZINC PLATED	8
7	1 1/2-6, UNC, ULTRA COATED GRADE 8 NUT	4
8	LEAD SCREW MOUNTING PLATE	1
9	8IN X 6IN X 1/4IN STRUCTURAL MEMBER	1
10	4IN X 4IN X 1/4IN STRCUTURAL MEMBER	2
11	GUSSET	12





DETAIL A
SCALE 1 : 20



SolidWorks Student Edition.
For Academic Use Only.

UMJal Consulting Group				TITLE: A-Frame Modifications	
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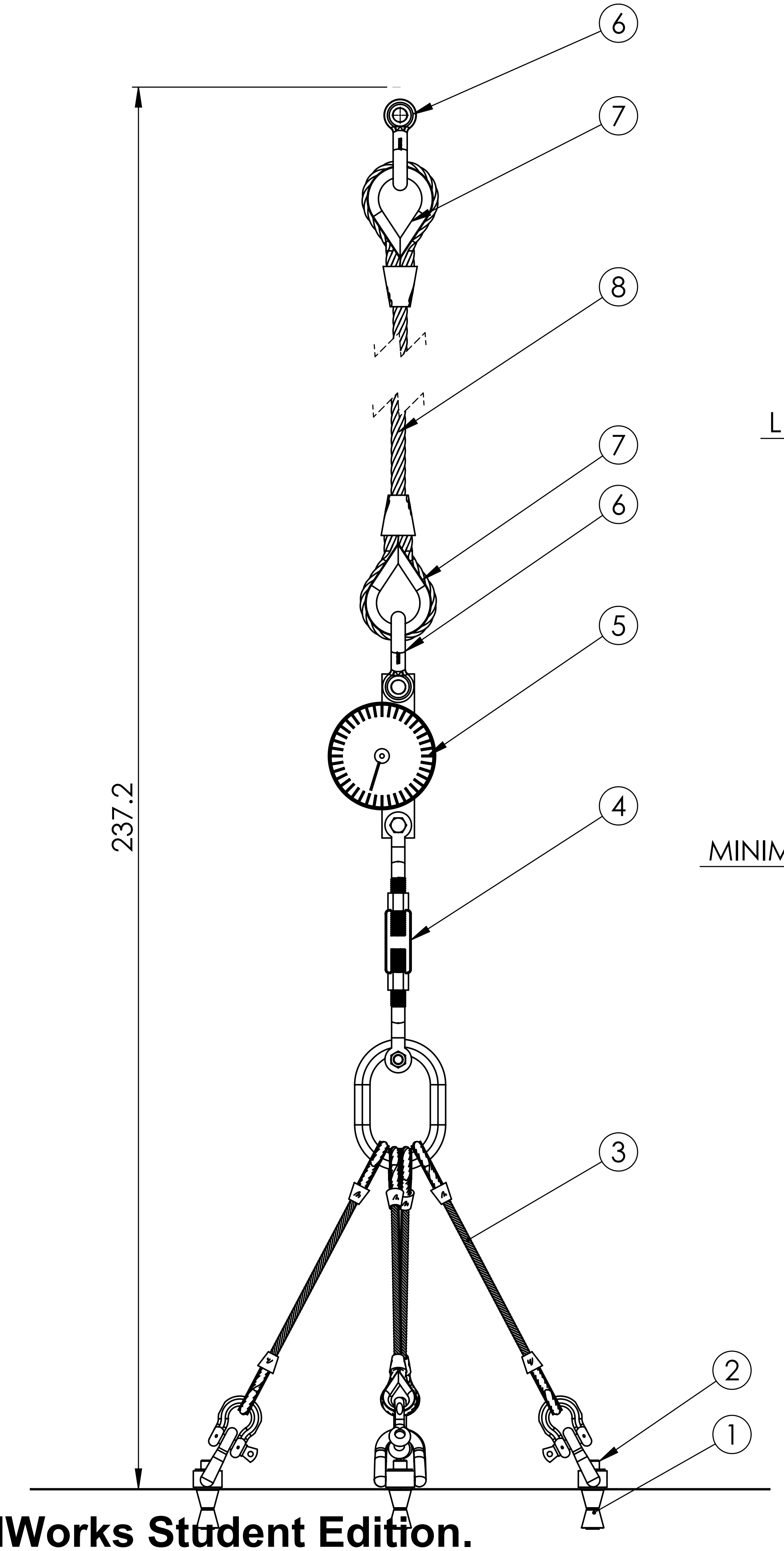
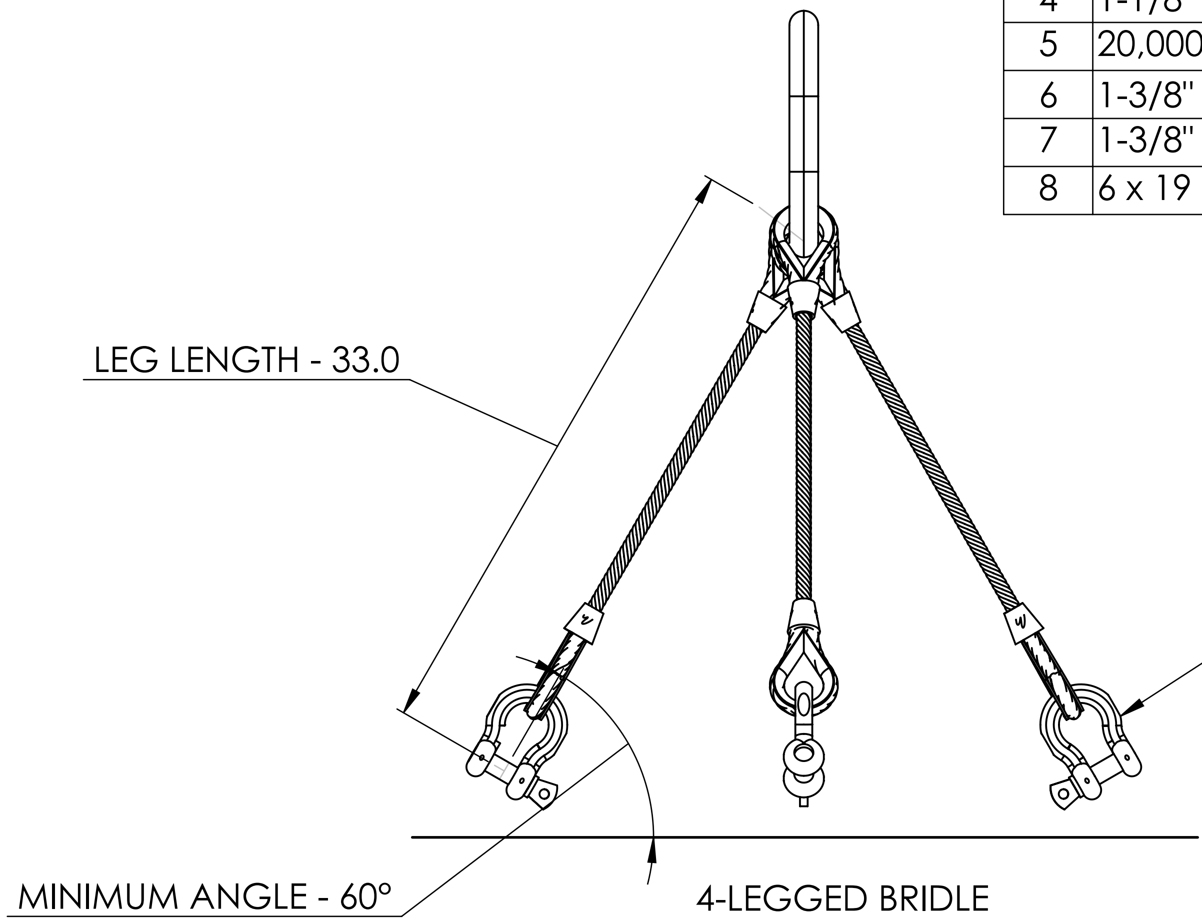
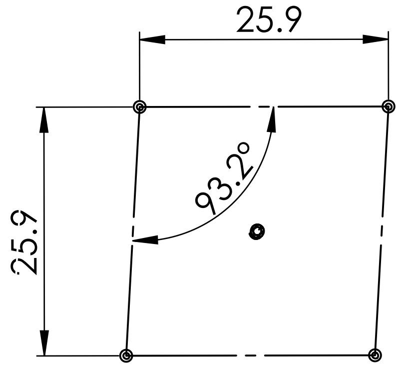
FINISH:	CHECKED	EH	01/12/2013		
DRAWING NOT TO SCALE	NUMBER			SHEET 3 OF 5	B-6

#	PART DESCRIPTION	QUANTITY
1	Double Expansion Concrete Anchor	4
2	3/4"-10 x 3.5" Hoist Ring	4
3	4-Legged Bridle with shackles	1
4	1-1/8" Galvanized Turnbuckle	1
5	20,000 Lb Dyanmometer	1
6	1-3/8" Screw Pin Shackle	2
7	1-3/8" Galvanized Wire Rope Thimble	2
8	6 x 19 IWRC Wire Rope	12

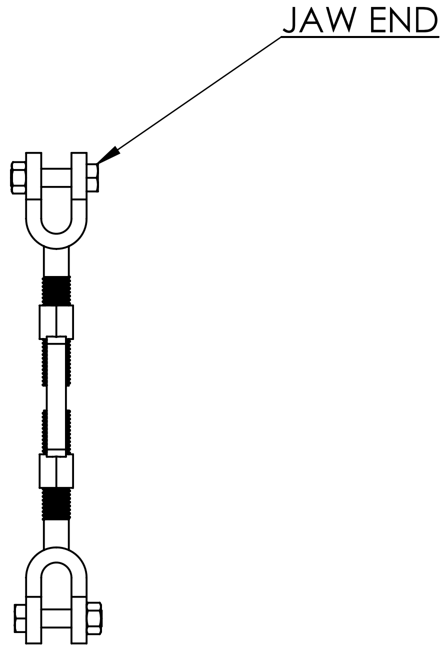




4-LEGGED BRIDLE



ANCHOR HOLE PATTERN



TURNBUCKLE

Consulting Group

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: MACH ±0° 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.002

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FINISH:	CHECKED	DG	01/12/2013
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TITLE: WEIGTH SIMULATION

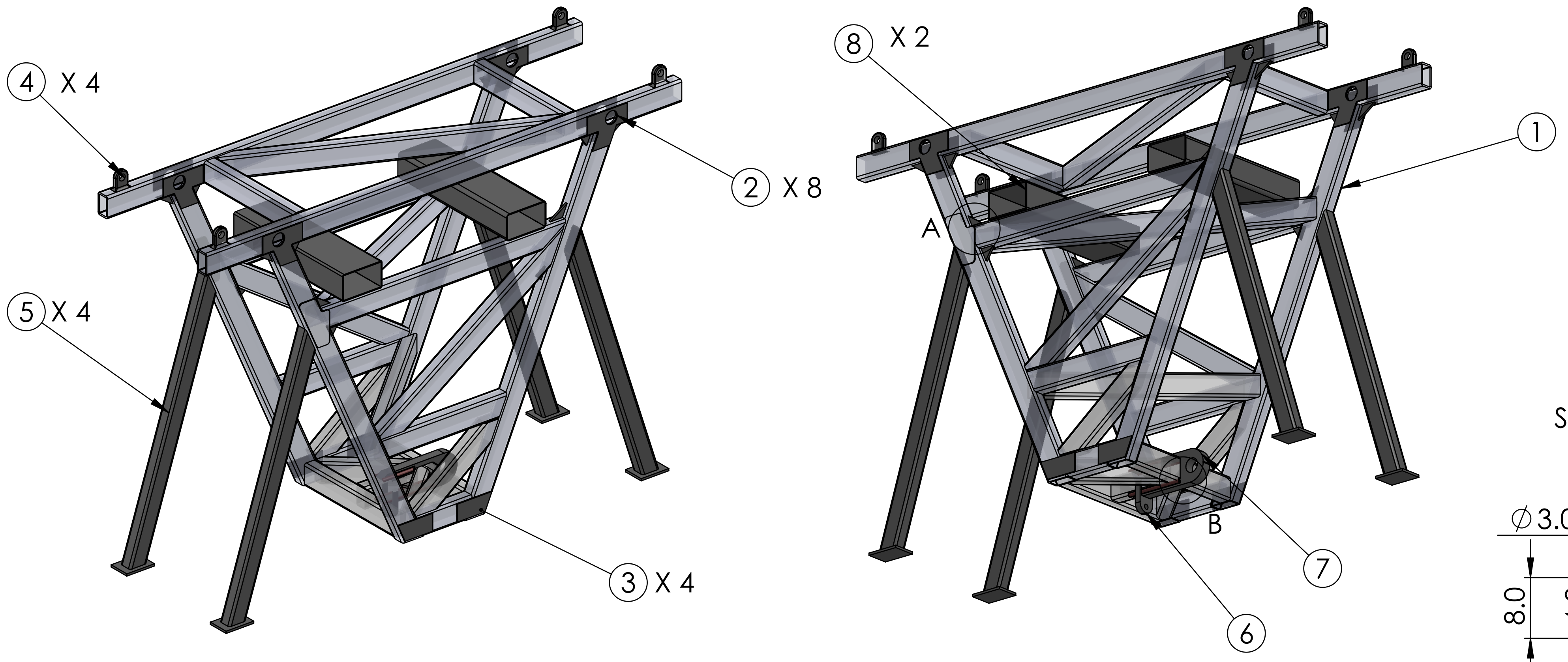
SIZE A3

DESCRIPTION: Assembly View

REV 0

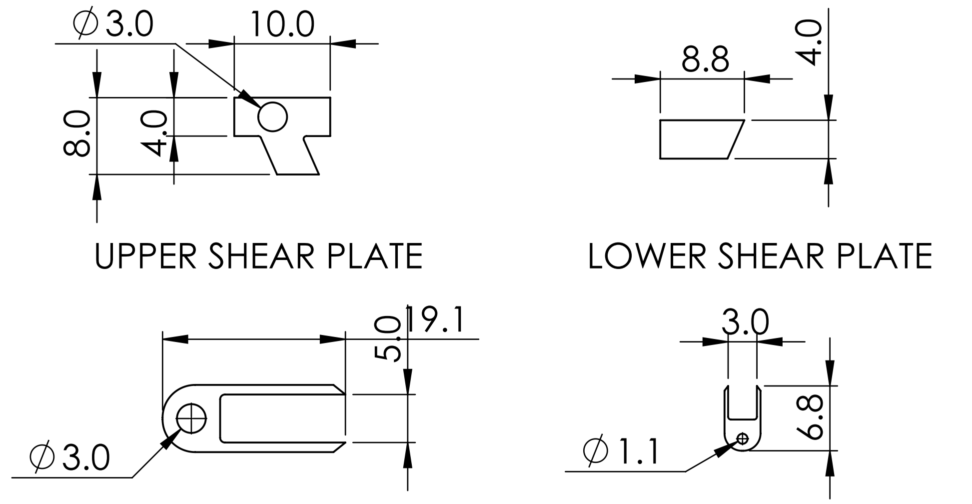
SHEET 4 OF 5

B-7



DETAIL A
SCALE 1 : 15

DETAIL B
SCALE 1 : 15

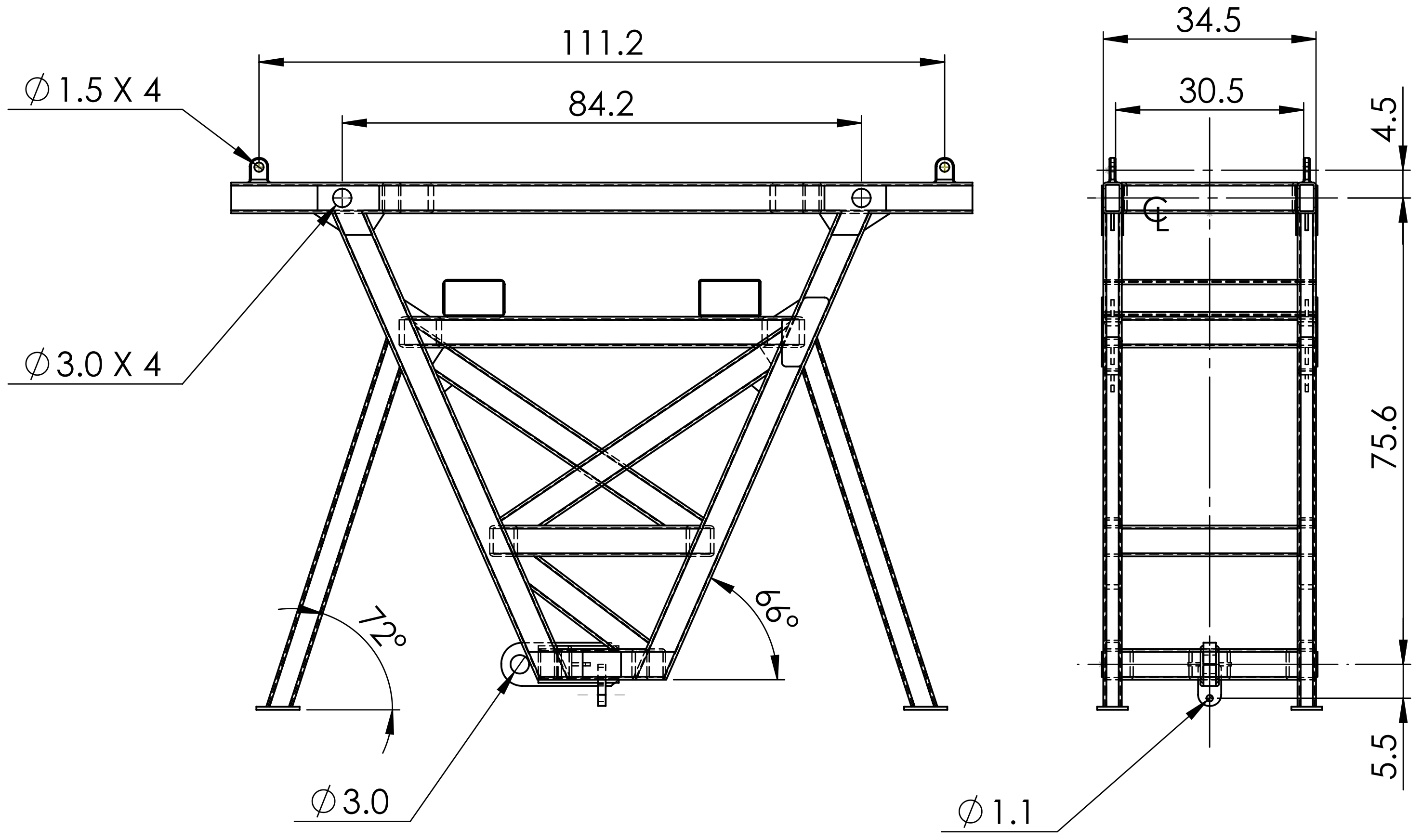


UPPER SHEAR PLATE

LOWER SHEAR PLATE

THRUST SIM MOUNT

WEIGHT SIM MOUNT



DIMENSION VIEWS

#	PART DECRIPTION	QUANTITY
1	TUBE FRAME	3
2	UPPER SHEAR PLATE	8
3	LOWER SHEAR PLATE	4
4	CHAIN MOUNT	4
5	STAND FOOT	4
6	WEIGHT SIM MOUNT	1
7	THRUST SIM MOUNT	1
8	FORKLIFT MOUNT	2
9	GUSSET	16

UJMal Consulting Group

TITLE: CALIBRATION PYLON

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: MACH ±0° 30'
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.002

		NAME	DATE
MATERIAL:	DRAWN	LK	30/11/2013
FINISH:	CHECKED	DG	01/12/2013
DRAWING NOT TO SCALE	NUMBER		

SIZE A3

DESCRIPTION: Assembly View

REV 0

SHEET 5 OF 5

B-8

Appendix C

Table of Contents

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2. Quotes and Cost Estimates

Cost estimates and quotes obtained by UMCal during the course of design are attached in the pages to follow.

Durham Instruments

Quote: 32169.CH

Products That Measure. Service That Measures Up.

Date: November 12, 2013
Company: University Of Manitoba - Mechanical
Attn: Eric Hornby

From: Chris White

Durham Instruments 900 McKay Road, Unit 3, Pickering Ont. L1W 3X8 Voice: 905-839-9960 Fax: 905-839-4325 www.disensors.com

Further to your request, I am forwarding the following quotation pertaining to the load cell. Please review the quotation and get back to us with any questions you may have.

Quote: 32169.CH

Item	Qty	Description	Unit Price	Amount
1	1	Model 1632AJH-100K Interface Gold Standard Load Cell 100,000 lbf. capacity, tension & compression, PT02E-12-8P connector, high precision base installed, factory installed calibration adaptor with 1-3/4"-12 UNF 3B (Male) thread, NIST traceable ASTM E74 calibration (3 runs with plots, coefficients)	7,603.000	7,603.00
2	1	Mating cable assembly, 50 ft. length with MC-002 mating connector, to open leads	195.000	195.00
			Sub Total	$7,798.00
			G.S.T.	$389.90
			Total	$8,187.90

Prices Quoted In Canadian Funds
Duty: Included Where Applicable
Taxes Extra Unless Noted
F.O.B. Pickering
Delivery: ~ 6 weeks
Terms: Net 30 days + 1.5% Per Month Past Due
Quotation Valid For: 30 Days

International Organization for Standardization
ISO CERTIFIED
ISO 9001:2008

SIGNED: Chris White

RED LINE

MOTION CONTROL INC
206 Brunswick
Pointe Claire, QC H9R-5P9

(T) 514 429 7733
(F) 514 429 7737

Quotation

Client:

University of Manitoba
E2-290 Engineering & Inforamtion
Winnepeg, MB R3T-5V6
Canada

Ship To

University of Manitoba
E2-290 Engineering & Inforamtion
Winnepeg, MB R3T-5V6
Canada

Date	Quote #
11/29/2013	SO-14-076
Att:	Heather McCrea
Rep	House

Item	Description	Qty	U/M	Cost/Unit	Total
HD618-06-MB-E-U4-...	HD Eliminator Series Heavy Duty Actuator -18,000lb Thrust -30,000lb max -Parallel Motor Pos #4 / Belt 1:1 Ration -Female Rod End -Bottom Mount -63mm Ballscrew x 12mm Pitch -2 x NC NPN Switches -All Steel Construction -Wgt: 240 lbs	1		19,495.00	19,495.00
E182-S3-B-1-N-0-2-1...	E180 Series Motor -TENV (IP55) -No Brake -Resolver feedback	1		4,900.00	4,900.00
SP3201	Unidrive SP without Keypad, 230Vac, Max Cont Output Current (hp): Normal Duty - 42A (15hp), Heavy Duty - 31A (10hp)	1		2,625.00	2,625.00
MISC	Cable set, Comprising Power and Resolver feedback cables, 3m	1		550.00	550.00
	Delivery Time: 6 weeks				
	Note: Budgetary - More application data is needed. Motor and drive could be reduced is speed is not a critical issue at full load. GST on sales			5.00%	1,378.50

Terms Credit Card/Cheque
FOB Pointe-Claire

Quotation is valid for 30 days

QST No. 1216972500
GST/HST No. 844640805

Subtotal CAD 27,570.00
Total CAD 28,948.50

3. Works Cited

[1] Value Line. (2013, Jan.). *Cost of Capital by Sector* [Online]. Available: http://w4.stern.nyu.edu/~adamodar/New_Home_Page/datafile/wacc.htm. [November 24, 2013].

Appendix D

Table of Contents

List of Figures

List of Tables

1. Technical Costs and Analysis

In order to ensure the feasibility of any designs moving forward, a brief cost estimate and structural analysis were prepared for each of the concepts that were brought past the screening phase. These were preliminary in nature, with a further in-depth analysis to follow later in the design process.

1.1. Thrust Simulation Concepts

The concepts pertaining to simulation of the engine thrust shall be addressed first, due to their simplicity when compared to the concepts for simulating thrust.

1.1.1. Reference Design

At the client's request, the team considered the current method of calibration as a reference design, and performed a preliminary stress and cost analysis as a baseline. Due to the complexity of analysis required, structural analysis will be limited to the connecting rod between the lead screw and thrust stand. A cost estimate will be restricted to the lead screw required to generate the thrust loading.

In order to determine the required sizing for a connecting rod, a rudimentary stress analysis was performed. Calculations were performed using a 105,000 $[lbf]$ load and a 150% safety factor The connecting rod was assumed to be a solid, round bar, subjected to purely axial loading. As the connecting rod will be in tension, buckling need not be considered. The results of the preliminary stress analysis for the connecting rod are summarized in TABLE I.

TABLE I: RESULTS OF PRELIMINARY STRESS ANALYSIS FOR REFERENCE DESIGN

Parameter	Value	Unit
Yield strength, A36 structural steel (σ_y)	36000	psi
Factor of safety (F.S.)	1.5	N/A
Applied load (P)	105000	lbf
Minimum required rod cross-sectional area	2.92	in^2
Required rod cross-sectional area, 150% F.S.	4.375	in^2
Required rod diameter, 150% F.S.	2.36	in

Depending on the method by which the connecting rod is fastened to both the lead screw and the thrust stand, additional area may be required to offset any stress concentrations. Any further calculations to determine this additional area will be performed at a later time, if necessary.

A rough cost estimate was prepared, taking into consideration the cost of the lead screw as well as the material for the connecting rod. The length of the connecting rod was estimated to be 15 feet, to take into account varying lengths of lead screw and the position of the connection point on the thrust frame. The results of the cost estimate are summarized in TABLE II.

TABLE II: PRELIMINARY COST ESTIMATE FOR REFERENCE DESIGN

Component	Quantity	Cost
Lead screw, 3-4 RH, with brass nut	1	\$633.37 [1]
Steel rod, 2 ½" diameter, 15 feet length	1	\$749.10 [2]
	TOTAL	\$1382.47

Upon completion of the preliminary cost estimate for the reference design, it appears to be within the budget constraints set forth by the client.

1.1.2. Simple Lever

Based on a high ranking from the weighted concept evaluation and positive customer feedback, the simple lever design was selected as one of the thrust simulation concepts for continued development. This concept consists of a support A-frame (preferably a modification to the existing A-frame), with a single lever mechanism to transfer applied load from the ground to the engine thrust line, located 20 feet above the ground. An optimized pivot point allows the input load, actuated by a lead screw system, to be significantly less than the required 105,000 $[lbf]$ at the engine thrust line. In order to ensure this design's feasibility prior to full development, preliminary stress and cost analyses were performed.

High level stress calculations were conducted on this concept to ensure its feasibility prior to moving forward in design phase. For the purpose of calculations on the load transferring lever at this stage, several design simplifications were assumed. For example, the horizontal force components of the applied load and thrust load were assumed to be constant with small lever deflection. Strength of the existing A-frame was not considered at this phase. Based on preliminary optimization, the load actuation point, lever pivot point, and thrust application point along the lever were placed at 3ft, 15ft and 20ft off the ground respectively. The selected beam for preliminary calculations is a 6x14in rectangular structural beam with a ¼ $[in]$ wall thickness. A basic drawing for the system can be seen in Figure 1.

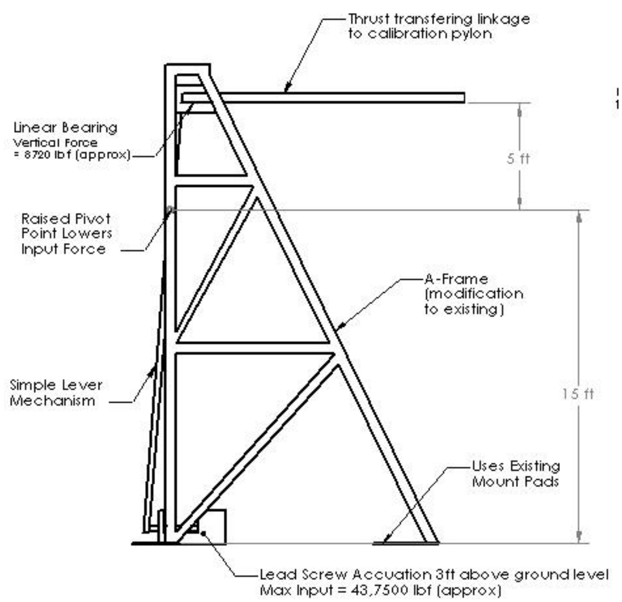

Figure 1: Simple lever thrust simulation concept.

The resulting applied load (at ground level) and the resultant thrust load were determined to be 43,750 $[lbf]$ and 105,000 $[lbf]$, respectively. Basic shear, bending moment and deflection calculations were performed based on these loads. Areas of concern for the lever were maximum stress due to bending moment, maximum deflection at the lever ends, vertical loading at the thrust application point, and overall lever weight. A summary of the results is shown in TABLE III.

TABLE III: PRELIMINARY CALCULATIONS FOR SIMPLE LEVER CONCEPT

Parameter/Calculation	Value
Beam Second Moment of Area, I_{xx}	2931.9 $[in^4]$
Yield Strength, σ_y	36,000 $[psi]$
Modulus of Elasticity, E	29 x 10^6 $[psi]$
Total Beam Weight, W	652.6 $[lb]$
Max Bending Moment, M_{max}	6,300,000 $[lb \cdot in]$
Max Stress, σ_{max}	15,041.4 $[psi]$
Factor of Safety	2.39
Max Deflection at Base, δ_{base}	2.00 $[in]$
Max Deflection at Top, δ_{top}	3.84 $[in]$
Max Vertical Load @ Thrust Line, $F_{T,y}$	8750 $[lbf]$

A rough cost estimate for this concept was achieved using estimated prices of the major system components. The current estimate accounts for material costs only. Although they are not included within this preliminary cost analysis, labor costs will be considered for final concept selection based on comparative design complexity. A basic bill of materials and associated design cost is shown in TABLE IV.

TABLE IV: PRELIMINARY COST ESTIMATE OF SIMPLE LEVER CONCEPT

Item	Qty	Unit Cost
Lead Screw (2 ½ in w/ bronze nut)	2	$250.00 [1]
24ft, Rectangular Structural Beam (6"x14")	1	$ 1,500.00 [3]
Linear Bearing	1	$250.00 [4]
A-Frame (use existing)	1	$0.00
	Total	$2,250.00

Moving forward with this design, the team will have to optimize the system to increase the lever strength in bending, while maintaining a reasonable component weight. In addition, the existing A-frame will require analysis to ensure it can withstand the new loading configuration. Modifications to the A-frame will have to be considered, with particular focus on the lever interface point. This area will likely require significant reinforcement due to a high reaction force at this point.

1.1.3. Modified Simple Press

The modified simple press concept consists of a three part linkage that transfers a linear force on the ground, to a horizontal tensile force along the engine center line. As shown in Figure 2, a ball screw lengthens the link B-C, causing the link A-C to rotate counter clockwise and consequently shifting link C-D to the left. This results in a tensile force along the engine centerline. The shaded linkages in Figure 2 illustrate the zero-load position, and the dotted linkages show how the linkages will articulate throughout the calibration process.

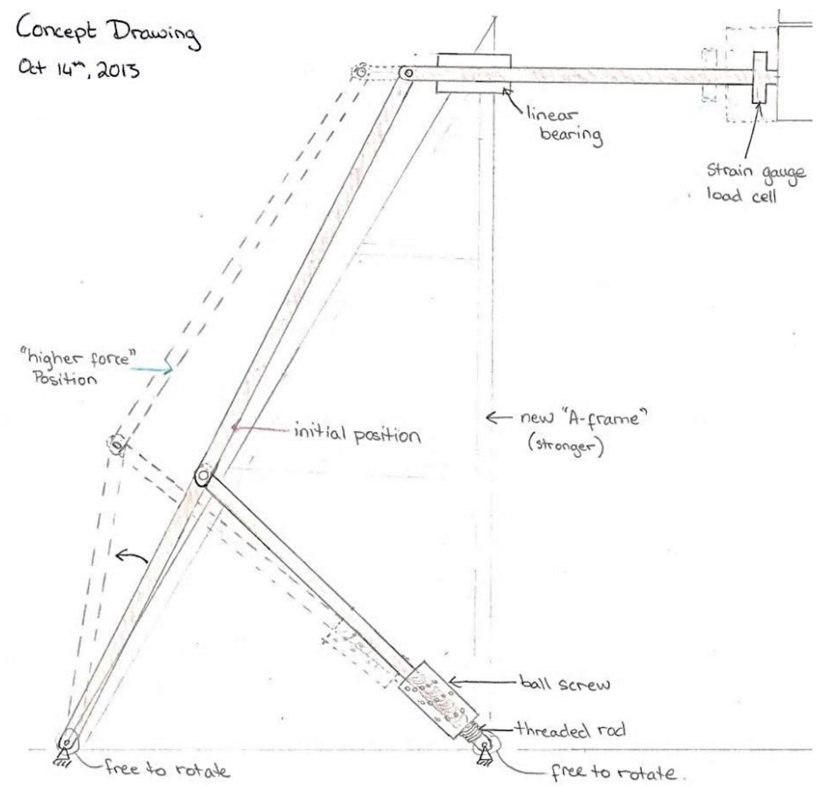

Figure 2: Modified simple press concept.

A preliminary set of calculations was completed to estimate the required length and radius of structural steel round bar required for the main linkages. These calculations can be found in the attached Appendix. From these estimates, we know that the required diameter of the round bar is approximately 4 $[in]$. However, due to the geometry of the linkages and the motion paths they follow, the ball screw will have to provide 340 $[klbf]$ in order to produce the desired 105 $[klbf]$ along the engine centerline. Ball screws of this capacity are uncommon; therefore, this design will have to incorporate two smaller ball screws working in parallel. Moreover, the system's geometry results in a 175 $[klbf]$ vertical force component acting along the engine centerline. A series of industrial linear bearings will have to attach to the upper most part of the A-frame in order to resist the vertical force and ensure only a horizontal, tensile force is acting along the engine center line. Furthermore, should this concept be selected, an in-depth analysis of the A-frame will need to be conducted in order to ensure it will withstand the required forces being applied. A summary of the calculations is provided in TABLE V.

TABLE V: SUMMARY OF PRELIMINARY MODIFIED SIMPLE PRESS ANALYSIS

Parameter	Value
Applied Force at ball screw	340 $[klbf]$
Vertical force acting at engine centerline	175 $[lkb]$
Required length of rod	33.61 $[ft]$
Required diameter of round bar	4 $[in]$

In order to provide a cost estimate for this concept, it has been assumed that all three linkages will be made from structural steel round bar of the same diameter. In addition, it is assumed that the current A-frame is strong enough to support the linkages without any modifications. TABLE VI summarizes the estimated cost of purchasing the materials and parts associated with the modified simple press concept.

TABLE VI: ESTIMATED COST OF MODIFIED SIMPLE PRESS CONCEPT

Part/ material	Cost/unit	Quantity	Total Cost
Steel rod	$120.32/ft	35 ft	4211.20 [2]
Ball screw	$50,000	2	$100,000 [5]
Linear Bearing (FL 64)	$300	3	$900 [6]
Total			$105111.20

1.2. Weight Simulation Concepts

With preliminary analysis complete for the thrust simulation methods, the team was able to perform similar analysis on the various proposed methods of simulating engine weight loads.

1.2.1. Set Weight

The first design to be analyzed is the set weight. This design is currently in use with the existing system (reference design) that the team is aiming to improve. As this design scored high during the concept weighting phase, it was decided that further analysis would be performed on the set weight design, to compare its feasibility against other potential concepts.

Using the set weight design is straight forward. The weight must be raised to the elevation where the pylon is resting and then be attached via pin connection. It is simple and effective, as the only requirement is a mass of predetermined weight which will interface with the trust frame pin connectors to keep the mass suspended during the calibration.

A disadvantage of this design is the effort required to move the mass from its storage location to the engine testing bay. If use of this design were to remain in use, a forklift would be required to move this mass. However, the client has expressed that they would prefer the mobility of the system to be simple.

In order to create a cost estimate for this design, it is assumed that a block of ASTM-A36 Structural Steel would be available and that it may be shaped to any desired size shape. TABLE VII summarizes the preliminary cost analysis of the set weight design.

TABLE VII: SUMMARY OF SET WEIGHT PRELIMINARY COST ANALYSIS

Dimensions		Total Volume (m^3)	Density (lb/in^3)	Total Weight (lb)	Cost ($/lb)	Cost ($)
Height (in)	12	35280	0.284	10019.52	$0.90	$9,004.98 [7]
Width (in)	49					
Length (in)	60					

From the results shown in TABLE VII, and based on the design cost, the set weight concept is a feasible design to simulate the weight of the engine. However, it should be noted that TABLE VII does not take into consideration the cost to manufacture the weight into its desired shape, nor does it include any required fasteners to mount the weight in place.

1.2.2. Turnbuckle

The next design concept is the turnbuckle, which will be used to simulate the engine weight during the calibration period. The turnbuckle was selected as it is a rather simple design to implement, and is inexpensive. The design will consist of two cables of different lengths, a turnbuckle, a measuring device (load cell), and some concrete anchoring points. The short length cable will be connected between the ground and the bottom end of the turnbuckle. The measuring device will then be connected between the top end of the turnbuckle and the bottom end of the long length cable. The longer cable will be connected between the measuring device and the pylon that is to be suspended from the thrust frame. The design can be seen in Figure 3 which outlines the main components of the design.

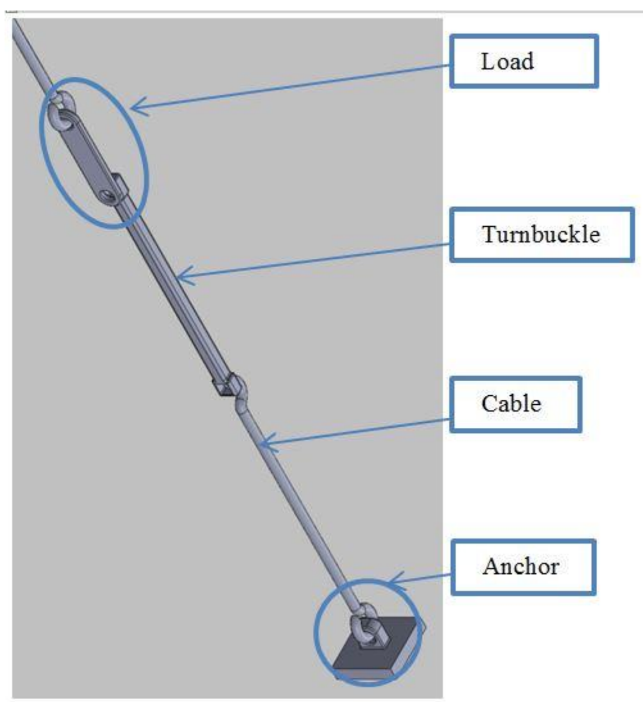

Figure 3: Highlighted features of turnbuckle design.

When the center of the turnbuckle is rotated, the inner threaded portions (left and right threading) pull the outer eyebolts closer together. As the eyebolts are brought closer together, the tension in the cable will increase. Using the measuring device to read the tension, the tension may be adjusted until it has reached the force that corresponds to the engine weight which the client wishes to simulate.

A disadvantage of the turnbuckle concept is that the tensioned cable anchored between the ground and the pylon will create a reaction force in the opposing direction of the applied thrust.

TABLE VIII summarizes the components required in order to implement the turnbuckle design. As well, some research has been completed to provide a preliminary cost analysis and is included in TABLE VIII. All components were selected to ensure a minimum safety factor of 2 based on the supplier's safe working conditions.

TABLE VIII: SUMMARY OF TURNBUCKLE PRELIMINARY COST ANALYSIS

Component	Cost	Quantity	Total ($)	Notes
Cable Short (1.375" Dia)	$6.99/ft	10	$69.90	Cable Size Based on Safe Working Load [8]
Cable Long (1.375" Dia)	$6.99/ft	30	$209.70	Cable Size Based on Safe Working Load [8]
Turnbuckle	$151.50/ft	1	$151.50	Selection Based on Force Rating [9]
Load Cell	$690.00/ft	1	$690.00	Selection Based on Force Rating [10]
Concrete Anchors	$44.79/ft	1	$44.79	Selection Based on Force Rating [11]
Total			**$ 1,165.89**	

From the proposed design cost stated by the client, it is evident that the turnbuckle concept will be a feasible alternative for the Engine Weight Simulation.

1.3. Summary of Results

Upon completion of preliminary cost and stress analyses, the modified simple press was excluded from any further consideration, owing to its high projected cost that was well above the desired budget. All other designs are possible from both a structural and a financial perspective.

2. Recommendations and Summary

Upon completion of the concept generation and preliminary analysis, the team was able to make recommendations with regard to the direction taken for the final design.

2.1. Thrust Simulation

After a preliminary analysis of three thrust simulation concepts, the team possessed sufficient information to select the best concept to develop further in the Final Design Generation phase of the project. One specific tool that is helpful for design selection is a cost comparison between the available options. TABLE IX summarizes the cost associated with implementing each of the thrust simulation methods that were analyzed. Note that the cost of fasteners, load cell and modifications to the current A-frame are not included as they are common among all designs.

TABLE IX: COST COMPARISON SUMMARY FOR THRUST SIMULATION CONCEPTS

Concept	Cost
Simple lever	$2,250.00
Modified Simple Press	$105,111.20
Reference	$1,382.47

Prior to this analysis, the simple lever concept was designated as the first choice for the thrust simulation due to positive client feedback. The cost analysis has shown that this concept is more expensive than the reference by approximately $1000. However, the team believes the additional cost will provide added value to the calibration system in the following ways:

- The lead screw will be on ground level, allowing for easier use and maintenance.
- The required input load is smaller, allowing for a smaller lead screw that will be less expensive to replace or upgrade.

Furthermore, the preliminary stress analysis confirmed that the required beam sizes are readily available from local suppliers. In addition, the calculated vertical force at the engine center is small enough to be supported by a single, fixed linear bearing. For these reasons, the team has selected the simple lever concept as the final thrust simulation concept that will be further refined in the Final Design Generation phase.

Also prior to the analysis, the client suggested analyzing the modified simple press concept as a backup in the event that the simple lever design is deemed insufficient. However, upon initial hand calculations, it was recognized that the input force required to simulate engine thrust was excessive for any lead screw available. The calculations revealed that the use of two ball screws would be required to reach the required input load. Consequently, the cost of implementation became significantly larger than the other two concepts and beyond the limit where meeting the desired payback period is attainable. As a result, the team recommends that the reference concept is used as a backup design, if the simple lever considered unfavourable in the future.

2.2. Engine Weight Simulation

Upon analyses of the top two weight simulation concepts, the team was able to confirm that the turnbuckle concept was the superior of the two with respect to cost. A cost comparison between the concepts reveals that the set weight concept is almost eight times the cost of the turnbuckle concept. TABLE X summarizes the cost of implementing the set weight and turnbuckle concepts.

TABLE X: COST COMPARISON SUMMARY FOR WEIGHT SIMULATION CONCEPTS

Concept	Cost
Set Weight	$9,004.98
Turnbuckle	$1,165.89

Furthermore, the set weight would be cumbersome to transport throughout the facility, in comparison to the turnbuckle which would easily be carried by one person. However, the client expressed concern that the turnbuckle design may produce a force that is not truly vertical. The team believes this concern can be addressed by attaching the lower cable to a linear guide rail on the ground level, which will ensure a fully vertical load throughout the entire calibration. For these reasons, the team has selected the simple lever concept as the final thrust simulation concept that will be further refined in the Final Design Generation phase.

3. Works Cited

[1] Roton. *2 1/2 - 4 Right Hand Acme Products* [Online]. Available: http://www.roton.com/Mating_Components.aspx?family=7059756. [Oct. 20, 2013].

[2] McMaster-Carr. *General Purpose Low-Carbon Steel* [Online]. Available: http://www.mcmaster.com/#standard-steel-rods/=p2z056. [Oct. 21,2013].

[3] Discount Steel. *Mechanical/Structural Steel Rectangular Tube* [Online]. Available: http://www.discountsteel.com/items/Mechanical_Structural_Steel_Rectangular_Tube.cfm?item_id=205&size_no=85. [Oct. 20, 2013].

[4] McMaster-Carr. *Linear Motion Carriages* [Online]. Available: http://www.mcmaster.com/#linear-motion-carriages/=p2z1el. [Oct. 21, 2013].

[5] R. Lipsett. *Why Lead Screws are the Best Fir for Many Linear Motion Applications - and How to Rightly Apply Them*. [Online] Available: http://www.thomsonlinear.com. [Oct. 18, 2013].

[6] D. Archambault (private communication), Oct. 22, 2013.

[7] McMaster-Carr. *Steel* [Online]. Available: http://www.mcmaster.com/#steel/=p2z5kq. [Oct. 23, 2013].

[8] US Cargo Control. *6x19 Bright Wire Rope EIPS IWRC (Steel Core)* [Online]. Available: http://www.uscargocontrol.com/Rigging-Supplies-Hardware/Bright-Wire-Rope-6x19-EIPS-IWRC-6x26. [Oct. 22, 2013].

[9] US Cargo Control, *Eye & Eye Galvanized Turnbuckles* [Online]. Available: http://www.uscargocontrol.com/Rigging-Supplies-Hardware/Eye-Eye-Galvanized-Turnbuckles. [Oct. 22, 2013].

[10] Vetek. *Loadcell tension "crane scale modell" 20 ton* [Online]. Available: http://www.vetek.com/Loadcell-tension-crane-scale-modell-20-ton/article#.UmnHX_mkpIl. [Oct 21, 2013].

[11] US Cargo Control. *Screw Pin Anchor Shackles* [Online]. Available: http://www.uscargocontrol.com/Rigging-Supplies-Hardware/Screw-Pin-Anchor-Shackles-Galvanized. [Oct. 22, 2013].