MAE 321: ENGINEERING DESIGN
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HEAVILIFT CRANE

FINAL DESIGN REPORT

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Executive Summary

The HeaviLift crane is a structure specified to lift a 1000 pound load to a height of 4 feet in approximately 3.5 seconds; however, the structure must be constructed from only 30 cubic inches of aluminum and 20 cubic inches of plastic. Crane performance is evaluated using an equation based on the lift height, weight lifted, lift time, volume of aluminum used, and assembly time. Since there are several performance factors, various approaches to performance optimization are possible. The ultimate goal is to design a crane with the highest performance score within the allotted time and resources.

Since height and lifting time are inversely proportional and coupled, the load could be lifted to a great height at the expense of a long lifting time and vice-versa; therefore, neither factor should be the focus of optimization. Instead, design philosophy should emphasize structural integrity and simplicity. A maximum performance score can then be achieved by designing a crane that is both easily assembled and lifts a large amount of weight.

The crane's design is essentially a supported cantilever beam. A pair of parallel aluminum cantilevers project horizontally from the vertical I-beam support. This simple design is efficient since less material is used than in some other designs in which the main beams project upwards at an angle. The reduced length also decreases the possibility of buckling due to compressive loads as well as increasing the stiffness.

Deflective and torsional rigidity are provided by 2 diagonal members, which are secured to the I-beam above the main cantilever beams. Pulleys along the central axis guide the steel cable from the reel, over the I-beam, and down the front of the cantilever beam. Plastic brackets and spacers, prefabricated to the exact required lengths, increase the structural rigidity and reduce the likelihood of buckling. The crane also uses a unique composite member for lateral support. This member is composed of two of the supplied steel pins, as well as 11 inches of steel cable, and is connected using cotter pins. It runs diagonally from the top pin on the I-beam to the opposite cantilever beam, where it is secured at a distance of 21.75 inches from the I-beam. In addition to significantly increasing the torsional rigidity of the structure, this member reduces the load on the diagonal aluminum members.

The aluminum in the final design weighs approximately 2.8125 pounds. The final crane withstood 351.25 pounds with minimal deflection and no sign of failure. In an attempt to lift 481.25 pounds, failure occurred because a plastic spacer unexpectedly detached, causing an asymmetric load which resulted in out of plane buckling. Analyses had predicted that the structure would buckle out of plane at 570 pounds, but this failure weight was based on completely symmetric loading.

Introduction

Objectives, Specifications & Constraints

The ultimate HeaviLift Crane is a lightweight structure capable of lifting approximately 1000 pounds of weight. Specifications require the crane lift the load vertically 4 feet while maintaining a horizontal distance of 3 feet from the nearest edge of the I-beam support. Additionally, this vertical displacement must be completed within 3.5 seconds. Unfortunately, rapid acceleration means there are additional (D'Alambert) forces. Therefore, the crane must support more weight than a statically calculated number to withstand initial jerks.

The entire structure must be fabricated from 30 cubic inches of 2024-T3 aluminum. Quarter inch thick stock is used for all of the members. 20 cubic inches of TIVAR UHMW plastic pieces (from half inch stock with a maximum dimension of 48 inches) are used for the plastic brackets and spacers. Half inch diameter steel pins (maximum of 7 with 14 washers) are the only means of securing each member. This means none of these pinned joints provide any restoring moment in theory. The entire structure is secured to a rigid steel I-beam with an 8 inch flange doubler and extender. This I-beam acts as a vertical support for the crane.

Lifting force is provided by a motor with a maximum power of 3 hp at 1760 rpm. The motor is geared down 30 to 1, providing enough speed reduction and torque multiplication so that the final output shaft is driven at approximately 1 revolution per second. 16 feet of steel cable capable of withstanding 2000 pound of static tensile load is run through a 4 inch diameter TIVAR UHMW take up reel, and 9 steel pulleys (3 each of 2 inch, 3 inch and 4 inch diameters) are provided for guiding this cable about the crane structure.

Basic Design Concept

HeaviLift crane performance is evaluated by the formula shown in Equation 1. Therefore, the ideal crane is one that is easily assembled, quickly lifts the most weight to the greatest height, and uses the least amount of aluminum. Based on this equation, assembly time appears to be the most significant factor, so building an easily assembled crane was an important goal. Also, the structure was designed for maximum strength in order to achieve a high weight credit factor.

$$Performance = \frac{Height[in]}{48in} \times \frac{3.5s}{Time_{lift}[s]} \times \frac{30in^3}{Aluminum_{use}[in^3]} \times 25 \times WCF \times \frac{20min}{Time_{assembly}[\mathrm{min}]}$$

Equation 1. Performance evaluation formula.

Preliminary crane designs involved multi-beam truss structures. Unfortunately, the limited amount of materials made these cranes infeasible. Subsequent configurations focused on simpler cantilevers with some sort of support to prevent the cantilever from rotating. A purely horizontal beam was chosen as the cantilever because the increased rigidity of an angled beam did not justify the extra required material. Furthermore, compression related buckling was deemed more severe with an angled beam. This led to the goal of having the diagonal members carry tensile loads rather than compressive loads. In order to minimize compressive loads further, Team A decided on using two thinner cantilevers with the pulley placed between them. Plastic brackets were added to the two lower cantilevers to reduce their effective length and thereby reduce their chance of buckling. The distribution of loads over two independent cantilever members is expected to reduce the chance of buckling and offer some counterbalance when subjected to torsional loads.

Since the amount of aluminum posed a significant constraint, the cross-sectional areas of the diagonal supports were reduced in the final design. From the initial test, it was evident that these members did not require the entire ¼ inch by 1 inch cross-section used in the cantilevers. In the final design, they were constructed using ¼ inch by ¾ aluminum.

Assembled Crane

The following is a picture of the proposed assembled crane. Notice the plastic H-bracket and the composite diagonal tensile member designed to increase lateral and torsional rigidity.

Figure 1a. Assembled crane.

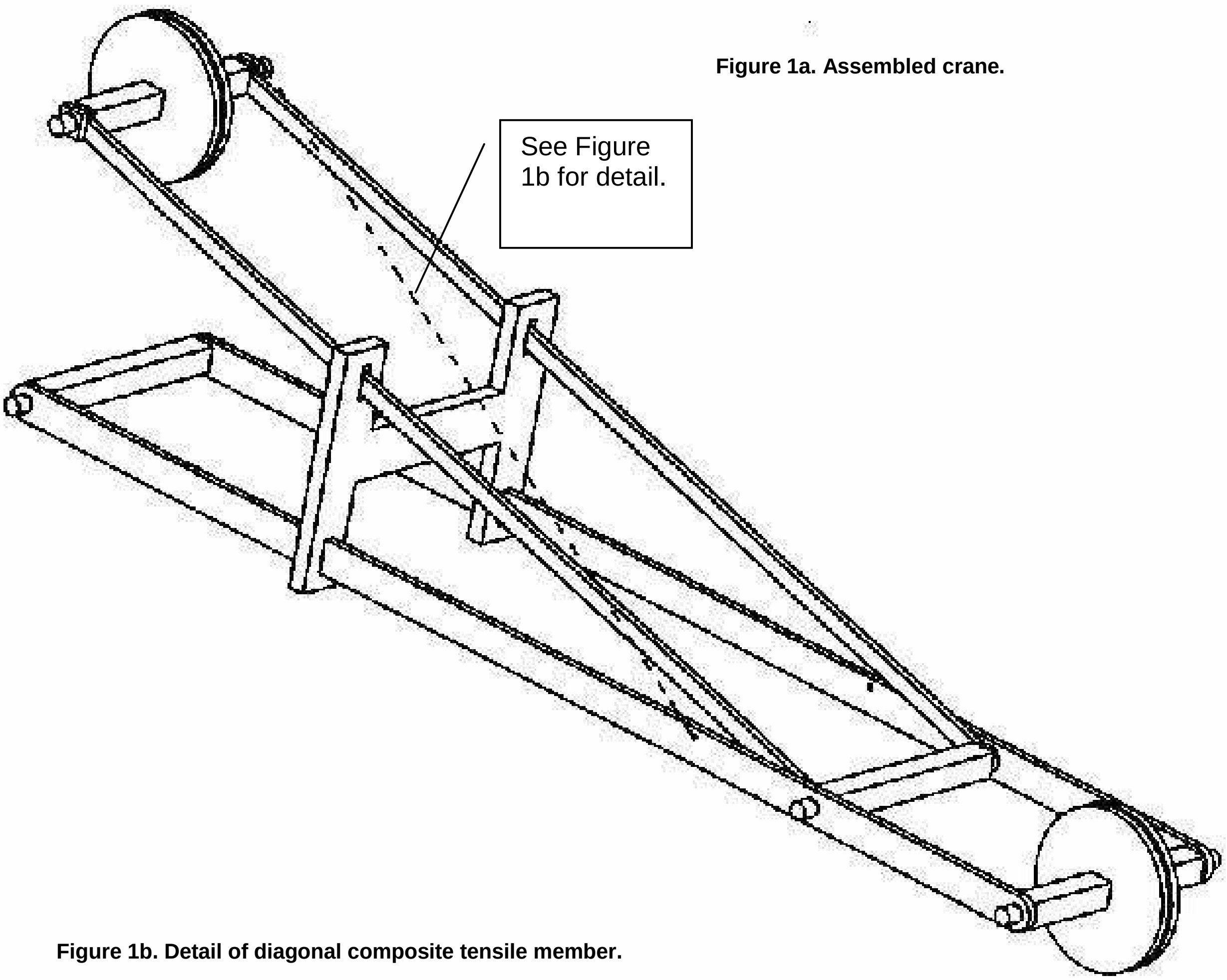
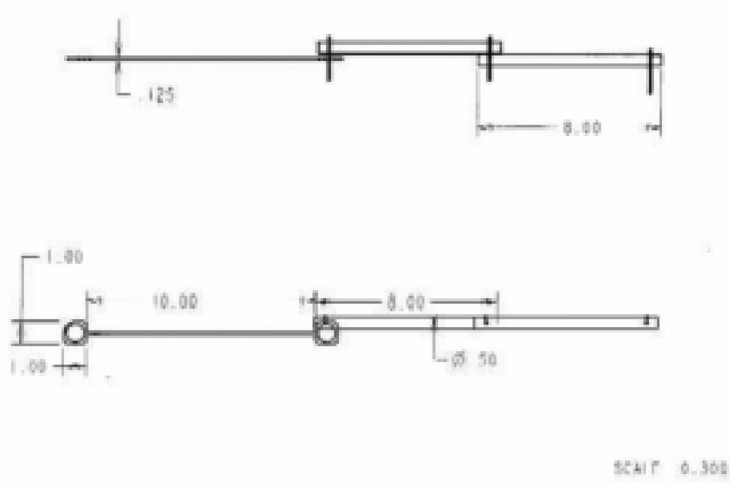


Figure 1b. Detail of diagonal composite tensile member.



Major Design Processes: Flowchart and Precedence Matrix

Figure 2. Flowchart of major processes.

Figure 3. Precedence matrix based on flowchart.

Detailed Design and Analysis

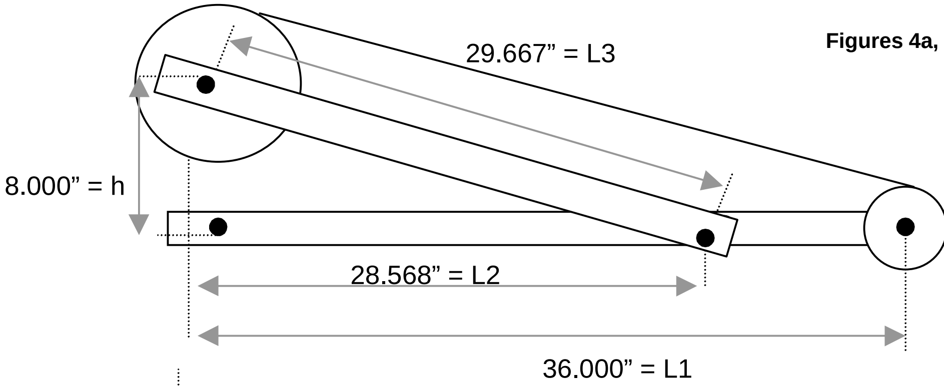
Design Philosophy

The design philosophy is to maximize the structural integrity of the crane at the expense of using more aluminum. Assembly time is also an important factor, requiring a simple and easily assembled crane. The use of horizontal cantilevers is justified since it offers the most efficient use of aluminum. Much less aluminum is necessary for a horizontal beam than for an angled member.

Mathematica graphs (please see Appendix C) were used as aids in the optimization process. Normal stresses, critical buckling critical loads, and deflection were plotted against L2, (the distance from the I-beam to the attachment of the diagonal support). These optimization plots showed a L2 length of about 28.6 inches would offer good deflective resistance while retaining sufficient lateral stability. Initially, there were some doubts regarding the imminent buckling of L2. However, when the prototype test ran with an L2 of 20, it was clear that the 16 inches of cantilever did not offer sufficient torsional or deflective resistance, and buckling was not a problem. Therefore, L2 was increased to 28.6 inches for the final design. The 28.6 inches for L2 left 7.4 inches of an effective cantilever at the end of the crane. Extending the support members to the end of the crane and thereby creating a truss structure would have been the optimal design. Unfortunately, there was not enough aluminum to build this truss design.

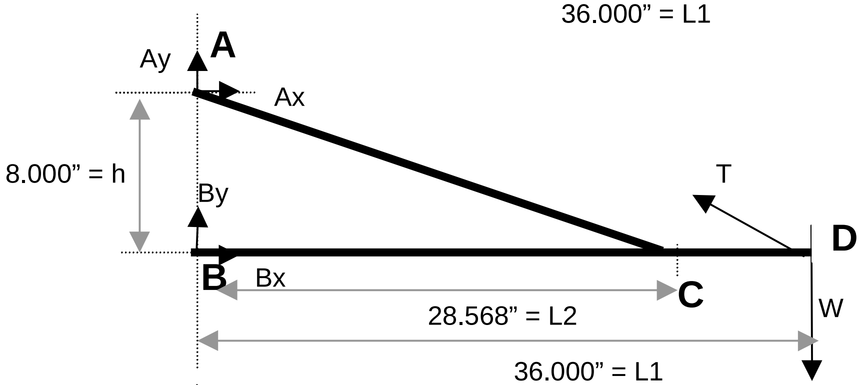
In order to maximize material usage efficiency, two unused steel pins and the 13 inches of excess cable were utilized as supports. The prototype testing showed the crane had a tendency to tilt to the left when viewed from the front despite its symmetrical construction. A bubble level showed that the I-beam was indeed slightly tilted. Furthermore, the final crane continued to tilt left even when no load was applied. Therefore, the last addition to the final design was to link the steel pins and cable with cotter pins diagonally across the I-beam to create a lateral restoring force when needed.

Design Loads & Structural Analysis

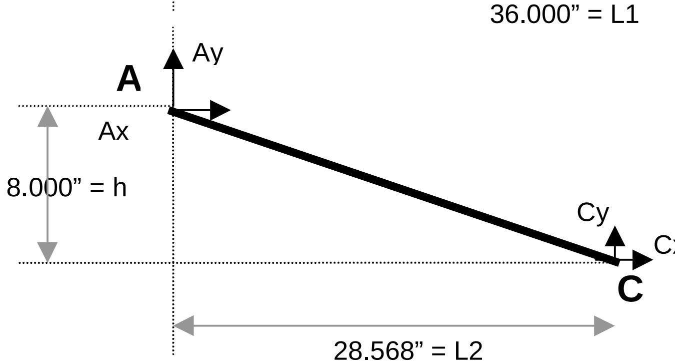


Figures 4a, b, c, d. Free body diagrams.

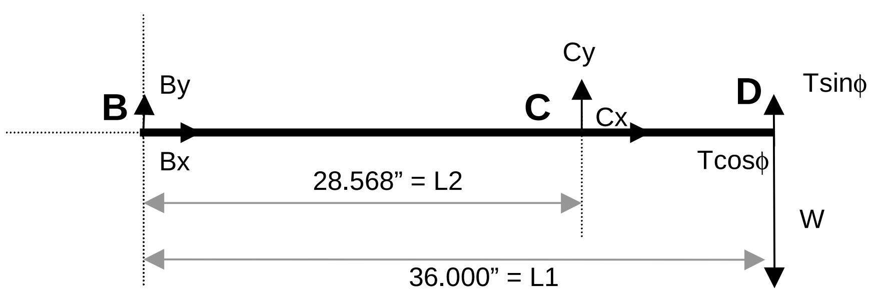
Equations 2. Reaction forces.



$$\sum Fx = Ax + Bx - Tx = 0$$
$$\sum Fy = Ay + T\sin\phi + By - W = 0$$
$$\sum Mb = -Axh + L1T\sin\phi - L1W = 0$$



$$\sum Fx = Ax + Cx = 0$$
$$\sum Fy = Ay + Cy = 0$$
$$\sum Ma = L2Cy + hCx$$



$$\sum Fx = Bx - C - T\cos\phi = 0$$
$$\sum Fy = By - Cy + T\sin\phi - W = 0$$
$$\sum Mb = -L2Cy + L1T\sin\phi = 0$$

$$Ax = \frac{L1(T\sin\phi - W)}{h}$$
$$Ay = -\frac{L1}{L2}(T\sin\phi - W)$$
$$Bx = T\cos\phi - \frac{L1(T\sin\phi - W)}{h}$$
$$By = \frac{L1}{L2}(T\sin\phi - W) - T\sin\phi + W$$
$$Cx = \frac{-L2Cy}{h} = -Ax$$
$$Cy = \frac{L1}{L2}(T\sin\phi - W)$$

Axial Stresses

Internal forces in the members (as shown in Figures 4c, 4d) are used to determine the tensile and compressive stresses. For the diagonal support beams, the k stress concentration factor is approximately 2.2. The main cantilevers are under compression. Because of its larger cross-sectional area, the k stress concentration factor is 2.19.

It should be noted that point C, the point common to both the diagonal members and the cantilevers, undergoes two stresses: 42500 psi from the diagonal members and -14900 psi from the cantilevers. Point B (where the cantilevers join the I-beam) has a compressive stress of -19200 psi, while point D (the loaded end) has a stress of 4200 psi. The Mathematica graphs show how each of these varies with L2. However, only the stress in the diagonal members (AC) is dependent on L2.

Diagonal Support (AC)

$$h/w = d/w = 0.5/0.75 = 0.667$$

$$K = 2.2$$

$$K + 0.35K = 3.0$$

$$L1 = 36.0in$$

$$L2 = 28.6in$$

$$\phi = 14.0°$$

$$W = T = 500lb$$

$$h = 8in$$

$$XArea_{diagonal} = 0.25in \cdot (0.75 - 0.5)in = 0.0625in^2$$

$$\sigma_{ac\,\max} = k \cdot \frac{\sqrt{\left(L1 \cdot \frac{T\sin\phi - W}{h}\right)^2 + \left(\frac{-L1}{L2} \cdot T\sin\phi - W\right)^2}}{2 \cdot XArea_{diagonal}}$$

$$\sigma_{ac\,\max} = 42.5E3psi$$

Equations 3. Axial stress calculations.

Main cantilever (BCD)

$$d/w = 0.5/1.0 = 0.5$$

$$K = 2.19$$

$$L1 = 36.0in$$

$$L2 = 28.6in$$

$$\phi = 14.0°$$

$$W = T = 500lb$$

$$h = 8in$$

$$XArea_{cantilever} = 0.25in \cdot (1.0 - 0.5)in = 0.125in^2$$

$$\sigma_{b\,\max} = -k \cdot \frac{\left(T\cos\phi - L1 \cdot \frac{(T\sin\phi - W)}{h}\right)}{2XArea_{cantilever}}$$

$$\sigma_{b\,\max} = -19.2E3psi$$

$$\sigma_{c\,\max} = k \cdot \frac{\frac{-L1}{h} \cdot (T\sin\phi - W)}{2Xarea_{cantilever}}$$

$$\sigma_{c\,\max} = -14.9E3psi$$

$$\sigma_{d\,\max} = -k \cdot \frac{T\cos\phi}{2Xarea_{cantilever}}$$

$$\sigma_{d\,\max} = -4.2E3psi$$

Deflection

In this analysis, section C-D of the structure is modeled as a cantilever section with a rigid point C. The diagonal cable added some deflection resistance as well. For a cantilever beam, Equations 4 govern the maximum vertical deflection. The inertia term is taken as twice that of each cantilever beam, assuming the two beams can be approximated by doubling I. This means deflection should be halved. Note that the Pro/Mechanica analysis uses only one beam subject to the 500 pound load, so Equations 4 will have only half the deflection. The Mathematica plots also show how the deflection varies inversely with L2.

$$\delta = (FL^3)/(3EI)$$
$$I = (1/12)bh^3$$
$$F = (T\sin\phi - W)$$
$$L = (L1 - L2)$$
$$\delta = [4(T\sin\phi - W)(L1 - L2)^3]/(Ebh^3)$$
$$\delta = -0.12in$$

Equations 4. Deflection calculations

$$T = 500lb$$
$$W = 500lb$$
$$\phi = 14.0°$$
$$L1 = 36.0in$$
$$L2 = 28.6in$$
$$E = 1\times10^7\ psi$$
$$b = 0.5in$$
$$h = 1.0in$$

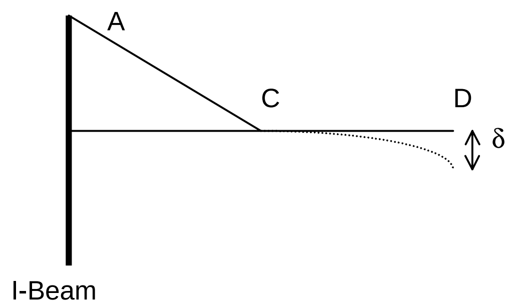


Figure 5. Deflection of member C-D.

Elongation

$$\Delta = FL/AE$$
$$T = W = 500lb$$
$$E = 1\times10^7\ psi$$
$$\phi = 14.0°$$
$$L1 = 36.0in$$
$$L2 = 28.6in$$
$$\sqrt{Ax^2 + Ay^2} = 1.77E3lb$$
$$Xarea_{cantilever} = 0.125in^2$$
$$Xarea_{diagonal} = 0.0625in^2$$
$$\Delta_{CD} = [-T\cos\phi(L1 - L2)]/(2XArea_{cantilever}E) = -0.0014in$$
$$\Delta_{BC} = [-Bx(L2)]/(2XArea_{cantilever}E) = -0.025in$$
$$\Delta_{AC} = [\sqrt{Ax^2 + Ay^2}(L3)]/(2XArea_{diagonal}E) = 0.042in$$

Equations 5. Elongation calculations.

Deflection of members cause elongation, which is roughly approximated by Equations 5. The cross-sectional areas of two beams is taken into consideration. It appears that the diagonal support member will undergo the maximum amount of elongation of 0.042 inches with a 500 pound load. The other members are under compression and thus undergo a decrease in length.

Buckling

Buckling is another possible mode of failure, especially in the cantilever from the I-beam to the attachment point of the diagonal supports. This length, L2, is 28.6 inches. The plastic H-bracket was placed in the middle of L2, thereby reducing the effective length by half. Buckling in the plane in the clamped-clamped mode is another possibility since the diagonal supports and H-bracket limits the angular motion of the cantilever beams. Similarly, the out of plane mode is also clamped-clamped. Thus, the effective length for both modes is L2/2.

Under these conditions, the critical force in the cantilever for buckling out of plane is 2500 pounds. The reaction force at the I-beam end of the cantilever (Bx) is about 2100 pounds, so buckling is possible. However, the buckling resistance is actually stronger because there are two cantilevers as well as the diagonal supports to increase rigidity.

Mathematica plots of buckling critical loads as a function of L2 can be found in Appendix C.

$$I_{inplane} = \frac{1}{12} bh^3$$

$$b = 0.25in$$

$$h = 1in$$

$$E = 10^7\ psi$$

$$L2 = 28.6in$$

$$F_{crinplane} = \frac{\pi^2 EI}{(0.5 \cdot L2/2)^2} = 40.3E3lb$$

$$I_{outofplane} = \frac{1}{12} bh^3$$

$$b = 1in$$

$$h = 0.25in$$

$$E = 10^7\ psi$$

$$L2 = 28.6in$$

$$F_{croutofplane} = \frac{\pi^2 EI}{(0.5 \cdot L2/2)^2} 2.52E3lb$$

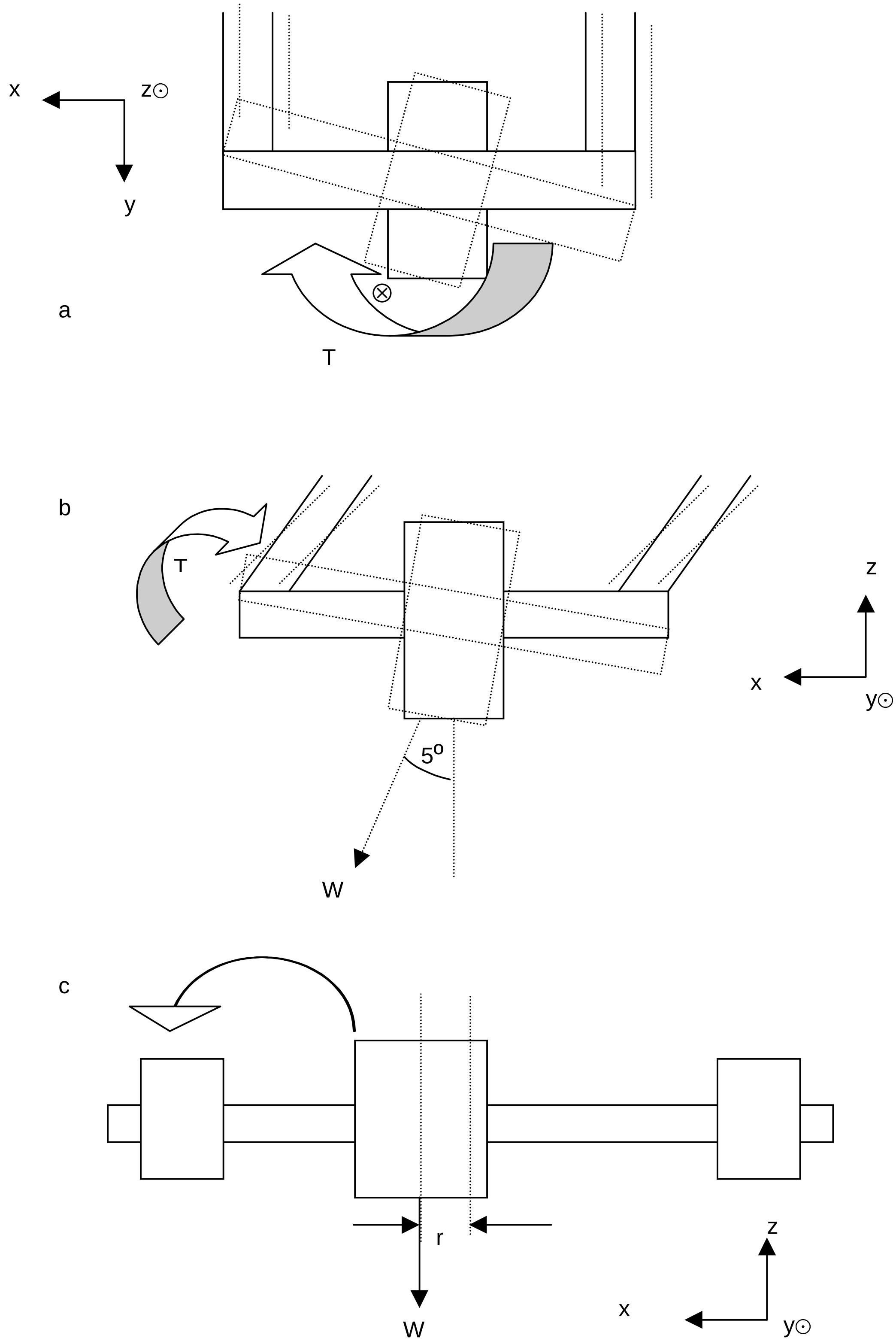
Equations 6. Buckling calculations.

Torsion

There are three manners in which torsion can be applied to the crane. The first figure (top view) shows a rotation of the pin about the z axis. This compresses one cantilever beam while stretching the other. The second figure (front view) shows a rotation about the y axis which subjects the entire structure to torsion.

Calculations are based on a 5 degree maximum rotation of the system with a 500 pound weight. The last figure shows a frontal view of the pulley shifting sideways in the x direction. This will create a moment about the y axis and tilt the crane.

Equation 7a assumes that the displacement due to the bending causes purely axial loads on the horizontal members. The stresses generated in this manner are negligible (63.4 psi). Equation 7b assumes that the unstable weights cause a 5 degree rotation. The



Figures 6. Three types of possible torsion. (a) Top view of rotation about the z axis. (b) Front view of rotation about the y axis. (c) Front view of displacement in the y direction.

polar moment of inertia for a rectangular cross section is used, but the torque is divided by 2 since there are 2 members. Because the beams have a small cross-sectional area, torsion can cause shearing stresses on the magnitude of 5400 psi. Equation 6c shows the result if the pulley displaces horizontally. This type of shear can also be quite significant (3450 psi).

These equations are based on having one cantilever instead of two. The beam is also independent from the additional rigidity provided by the diagonal supports. Therefore, the actual stresses are expected to be lower.

Equations 7. (a) Rotation about the z axis. (b) Rotation about the y axis. (c) Translation in the y direction.

a

$$F_{pulley} = 500lb \times \sin(5°) = 43.6lb$$

$$M_{pulley} = F_{pulley} \times r = 87.2lb \times 1in = 43.6lb \cdot in$$

$$F_{beam} = \frac{M_{pulley}}{W/2} = \frac{43.6lb \cdot in}{5.5/2} = 5.8lb$$

$$\sigma_{beam} = \frac{F_{beam}}{A} = \frac{15.8lb}{0.25in^2} = 63.4psi$$

b

$$\phi \leq 5°$$

$$J = \frac{1}{12} wh\left(w^2 + h^2\right)$$

$$T = \frac{\phi GJ}{2L} = \frac{5° \times \frac{2\pi}{360°} \times 3.7E6psi \times \left(\frac{1}{12} \times \frac{1}{4} \times 1\left(\left(\frac{1}{4}\right)^2 + 1^2\right)\right)}{2 \times 36.0in} = 99.3lb \cdot in$$

$$\tau_{\max} = \frac{T}{hw^2}\left(3 + 1.8\frac{w}{h}\right) = \frac{99.3lb \cdot in}{1in \times 0.25^2 in^2}\left(3 + 1.8\frac{0.25in}{1in}\right) = 5.4E3psi$$

c

$$T = Wr = 500lb \times 0.125 = 62.5lb \cdot in$$

$$\tau_{\max} = \frac{T}{hw^2}\left(3 + 1.8\frac{w}{h}\right) = \frac{62.5lb \cdot in}{1in \times 0.25^2 in^2}\left(3 + 1.8\frac{0.25in}{1in}\right) = 3.45E3psi$$

Vibration Analysis/Resonance

The crane is modeled as a cantilever with an external point load at its end. The external force is lower than the gross weight since there exists an upward tension in the cable. The net downward load is 378.7 pounds with a 500 pound load (Equation 8). An effective spring constant, k, is used for a cantilever, and an equivalent mass is determined to evaluate the natural frequency of the crane. The base of the cantilever is doubled to account for the two parallel beams. The calculated natural frequency of the crane under these conditions is 0.72Hz.

This should be compared to the frequency of the motor (0.83 Hz). Therefore, it is possible that the frequency of the motor will pass through the 0.72 Hz natural frequency of the crane during the initial few inches.

$$F = T - T\sin\phi$$

$$k = \frac{3EI}{l^3}$$

$$I = \frac{1}{12}bh^3$$

$$m = m_c + 0.23m_d$$

$$m_d = \rho V$$

$$\omega_n = \sqrt{k/m}$$

$$\omega_n = \sqrt{\frac{\dfrac{3\cdot 10^7\, psi\cdot\left(\frac{1}{12}\cdot 0.5in\cdot(1in)^3\right)}{(36in)^3}}{\left(500lb - 500lb\sin(14.0°) + 0.23\cdot\left(0.1\frac{lb}{in^3}18in^3\right)\right)\cdot\dfrac{lb_m}{386.4lb}}}$$

$$\omega_n = 5.22\, rad/s$$

$$f_n = 0.83Hz$$

$$f_{motor} = \frac{1760rev}{min}\cdot\frac{1}{30}\cdot\frac{min}{60s} = 0.98Hz$$

Equations 8. Vibration analysis

Out of Plane Analysis

Figures 7 shows a top of the crane under a purely lateral (out of plane) load of 500 lb. Here, the composite diagonal cable is assumed to provide all of the restoring force. A force balance in the directions parallel to the weight shows that the cable must support a tension of 1746 lb. Since the steel cable can withstand 2000 pounds of tension and the steel pins are even stronger, the most likely form of failure would be the cotter pins bending or slipping.

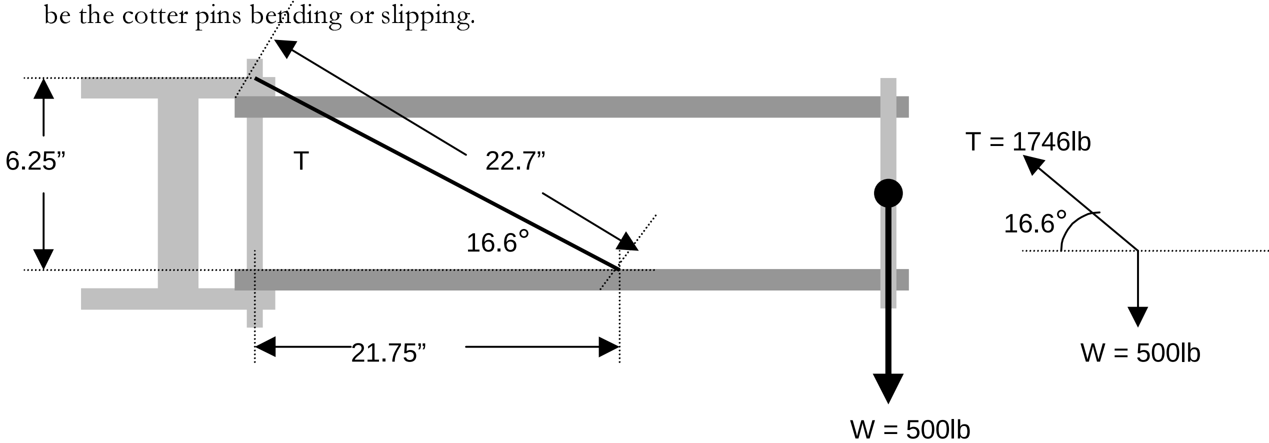


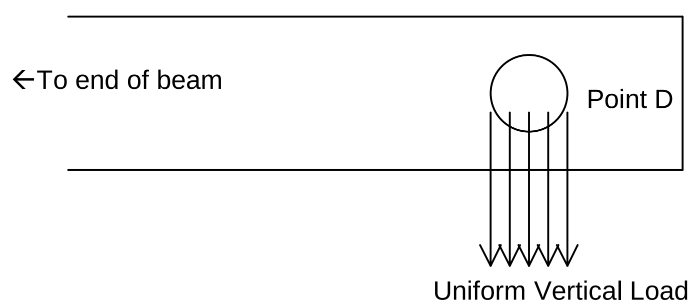
Figure 7. Out of plane schematic.

Pro/Mechanica Simulations

Pro/Mechanica was utilized to simulate the behavior of one horizontal member under a vertical load of 500 lbs. Specifically, member B-D (Figure 4b) was modeled in the software. Every effort was made to approximate the actual load and structural constraints in the model. Essentially, the analysis of the member can be simplified as a cantilever. Points B and C are rigidly constrained in the x, y, and z directions. This was done with the diagonal supports, a series of plastic spacers and pulleys, and a composite, diagonal tensile member. Therefore, the simulation reduces to cantilever C-D with end C being fixed and a load of 500lbs at D. The simulation model deviates from actuality in that real-life physical constraints give way under deformations resulting from the subjected loading conditions, thus allowing some motion along all three directions.

<u>Deflection</u>

The computer model assumed a vertical load equal to 378 lbs., the net vertical load applied to point D via the pulley (Equation 8). The net load is uniformly applied across the bottom curvature of the hole, as shown below:

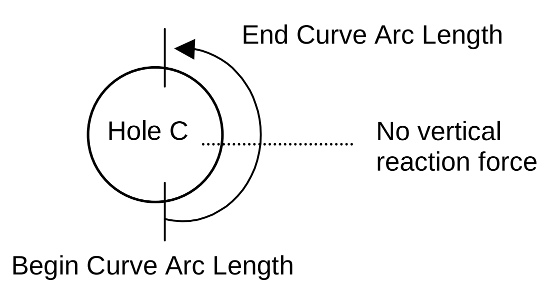


Appendix B Figure 1 shows the results of the iterative convergence analysis performed by Pro/Mechanica. Under the specified conditions and constraints, the simulation shows a deflection of section C-D occurring about point C. The vertical deflection is reported as -0.035 inches. The simulation also shows a dilation of the hole at point D, although no numbers were reported by the software to quantify this observation. The theoretical deflection analysis performed with Equations 4 yields a theoretically computed deflection value of -0.06 inches (for one horizontal beam, alone). The discrepancy between the two numbers results from the approximation for the applied load at the end of the beam. For the theoretical analysis, the net applied load was assumed to be at the end of the beam, acting effectively as a point load at this geometric point. In the computer simulation, however, the load was uniformly distributed about the bottom surface of the hole. Even with the simplifying assumptions in the theoretical analysis, the calculated value is within 15% of the PRO/MECH simulation. Neither value is truly representative of the actual deflection, however. As mentioned before, the beam does not behave like an actual cantilever, since physical constraints are not rigid.

<u>Edge Reaction Force (Y-direction)</u>

The edge reaction force along the curve of the hole at Point C was simulated. Refer to Appendix B Figure 3 for a representation of the vertical reaction force as a function of the curve arc length defined by Figure 8.

Figure 8. Stress concentration at hole C.



As expected, the plot reveals that the maximum vertical reaction force is found at the bottom-most point of the hole. Progression through the curvature reveals that there is a point at which there is no vertical reaction force, which is labeled above. The theoretically determined vertical reaction force at point C (refer Equations 2) is an average point load value which does not take into account local stress concentrations around the hole. The computed value for the reaction force at point C is about 477 lbs., which is less than most reaction values found about the curvature of the hole. The discrepancy between the theoretical analysis and the computer simulation is a result of the method of analysis. The theoretical analysis was a "macro-analysis," which treated the beam surface as essentially a flat contact surface with uniform reaction forces. Appendix B Figure 3 shows local reaction forces around the hole, and these data are not informative on the overall structural rigidity. Rather, these local forces help to point out sections where the applied forces may yield stresses close to yield stress values for the chosen material. Again, the simulated values are not truly indicative of the actual situation because point C is actually reinforced by the addition of two tensile members alongside of the main member under analysis, effectively increasing the cross-sectional area of the hole. Also, these reported reaction forces are for one member only, but in the actual crane configuration (2 identical members), the reaction forces are about half of the simulated values.

Maximum Principal Stresses

Appendix B Figures 1 and 2 depict a fringe plot of the stress distribution along the structural member. From the color-code, the order of magnitude of the stresses can quickly be grasped. The fringe plot is not a true representation of the stress distributions in this member, as this member is actually not subjected to the applied vertical load without any other members aiding in tension and compression. A true fringe plot should include the entire crane structure.

Failure Load & Mode Prediction

Table 1. Summary of major parameters and variables for a load of 500 pounds.

Parameter	*Value*	*Units*
W	500	lb
L1	36.000	in
L2	28.568	in
L3	29.667	in
ϕ	14	
k diagonal	3	
k cantilever	2.19	
x-area diagonal with hole	0.0625	in^2
x-area cantilever with hole	0.125	in^2
Ax	-1704.29	lb
Ay	477.26	lb
Bx	2189.37	lb
By	-98.5276	lb
Cx	1704.29	lb
Cy	-477.26	lb
Fac	1769.86	lb
σacmax	42476.6	psi
σbmax	-18167.8	psi
σcmax	-14929.6	psi
sdmax	-4249.22	psi
δ	-0.124	in
Fcr in plane	20566.6	lb
Fcr out of plane	2519.42	lb
Natural frequency	0.98	Hz

Analyses shows that, for a 570 pound load placed symmetrically between the two horizontal members, member BC (see Figure 4b) will buckle out of plane at its critical buckling force of 2500 pounds. Also at this load, the member AC (see Figure 4b) is within 2000 psi of the aluminum's yield stress, so tensile failure is also a concern. Other members should not be near failure with this load. Any load placed asymmetrically would result in buckling out of plane at less than 570 pounds.

Structural Weight Analysis

The first crane, with a constant ¼ inch by 1 inch cross-section throughout, weighed in at 2.9375 pounds, which is equivalent to 29.375 in^3 of aluminum. This crane lifted 350 pounds, giving the prototype a strength to weight ratio of 119.1 pounds per pound of aluminum.

The final crane lifted 351.25 pounds with a structural weight of 2.8125 pounds (28.125 in^3 of aluminum). Therefore, the strength to weight ratio was 124.9 pounds per pound of aluminum. It is expected that if the unexpected mode of failure had not taken place (see Final Test Results), the crane would probably have held at least 450 pounds, giving a much greater strength to weight ratio of 160 pounds per pound of aluminum.

Final Design Drawings

Assembled Crane & Individual Members

Please see Appendix A.

Test Results

Iteration	Max Load	Height	Time	Failure Mode	Weight Crane	Strength / Weight	Assembly Time
1	350 lbs	44"	4.4 s	Out of plane buckling	2.9375 lbs	119.1 lb/lb Al	1min 44 s
2	351.25 lbs	36"	3.44 s	Out of plane buckling	2.8125 lbs	124.9 lb/lb Al	1 min 7 s

Table 2. Summary of results.

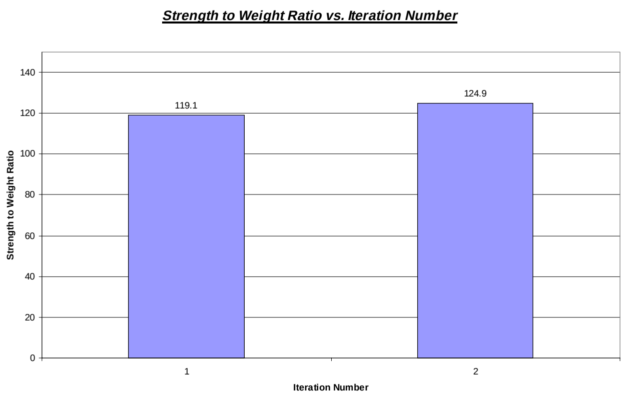


Figure 9. Strength to weight ratio vs. iteration number.

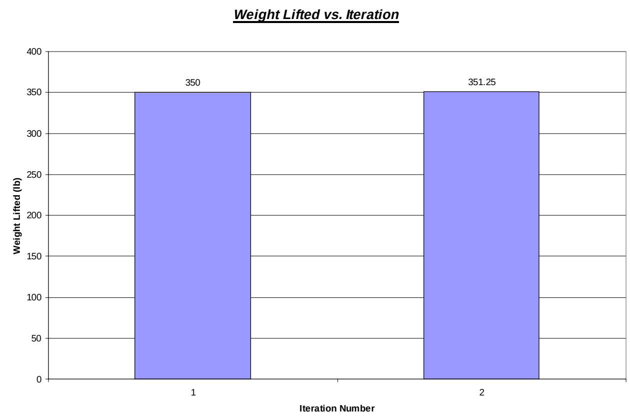


Figure 10. Weight lifted vs. iteration number.

Initial Design Results

The initial design had diagonal support beams attached to the main cantilevers at a distance of 20 inches from the I-beam pinholes. The I-beam's schematics showed a distance of 58.5 inches between the floor and the top of the I-beam. Taking into account the minimum 48 inch vertical displacement in the required crane's specifications, it leaves a maximum of 8 inches between the top of the crane and the 48 inch mark. This resulted in a length of 21 inches for the diagonal support. Cross-sectional dimensions were constant in all the members at 1 inch by ¼ inch. The total weight of the aluminum members was 2.9375 pounds.

During our optimization process, it was clear that the cross-sectional areas of the diagonal supports could be reduced. This was verified during the prototype testing, when these supports showed almost no sign of deformation, while the cantilever beam deflected significantly, which eventually caused the final failure. Under a 400 pound load, permanent deformation from previous lifts caused an asymmetric load that resulted in out of plane buckling.

The highest successful lift for this prototype was 350 pounds at a height of 44 inches in 4.4 seconds. The crane failed to reach the designed height of 48 inches due to deflection of the cantilever section. When loaded, the cantilever deflected more than expected. Furthermore, the I-beam schematics did not include the heights of the raised wooden platform and the newly included polyurethane padding. These two modifications in the I-beam decreased the distance between the floor and the top of the I-beam by 6 inches. With this reduced height, the bottom of the unloaded and undeflected crane was already at 48 inches.

Final Design Results

The final crane design was significantly stronger in both deflection and torsional resistance than the prototype. An initial load of 65 pounds caused virtually no deflection in the crane. As a result, the weight was immediately increased to 351.25 pounds. At this weight, the crane lifted a vertical distance of 36 inches in 3.44 seconds. It is estimated that the crane would have continued lifting to at least 46 inches had the motor not been stopped prematurely. Furthermore, deflection was minimal and no sign of torsion or buckling was evident. It was clear that the diagonal composite member was extremely beneficial since it immediately went into tension when the crane was loaded, thereby removing much of the load from the cantilever beams.

The final test for the crane was at 481.25 pounds. Under this load, it appeared that the plastic spacer on the left side of the front pulley popped out under lateral load, causing the pulley to slide to one side of the pin, and thus causing uneven loading, which twisted the crane and caused the cantilever to buckle out of plane. Further study of this failure from video is necessary since it is also possible that the cantilever section buckled first, which then caused the spacer to fall out.

The improvement of the final crane is even more substantial considering that the final crane was lighter (2.8125 pounds) by 4.3%.

Conclusion

The main objective in our final crane design was to maximize the performance score by designing a crane which is easily assembled while lifting a large amount of weight. Additionally, from observations of the testings of the first prototype and the cranes from the other groups, lateral bending of the entire crane structures to the left (when facing the I-beam head on) was observed. Thus, another objective in redesigning our crane was to counter-balance this lateral bending. This was achieved by adding the diagonal composite member. Finally, a significant amount of deflection was observed in the cantilever section of the first prototype. This was a limiting factor in the maximum height the weight could be lifted. In the final design, this deflection was minimized by shortening the length of the cantilever section through the lengthening of the diagonal support.

The final crane lifted 351.25 pounds in 3.44 seconds, to a height of 36 inches, with an assembly time of 1 minute and 7 seconds.

Suggestions for Further Improvement

It is clear from the final testing that the structure needs to be strengthened even further to resist out of plane buckling. The buckling mode can be increased by inserting more pins into the main cantilever. However, this option may not be feasible since all the extra pins are used in the composite torsional member. Additionally, the plastic spacers need to be more resistant to dislodging under lateral load. This can be achieved by drilling the pinhole further upward in the cross-section or by reducing the amount of sanding around the spacers' edges. However, these modifications are expected to increase assembly time marginally.

Further work on the optimization of material usage may also prove beneficial. Since the diagonal supports still appear to be stronger than the cantilevers, more material should be apportioned to the cantilevers.