APPLE HARVESTING PLATFORM

(Preorder NOW with a $100 Deposit)

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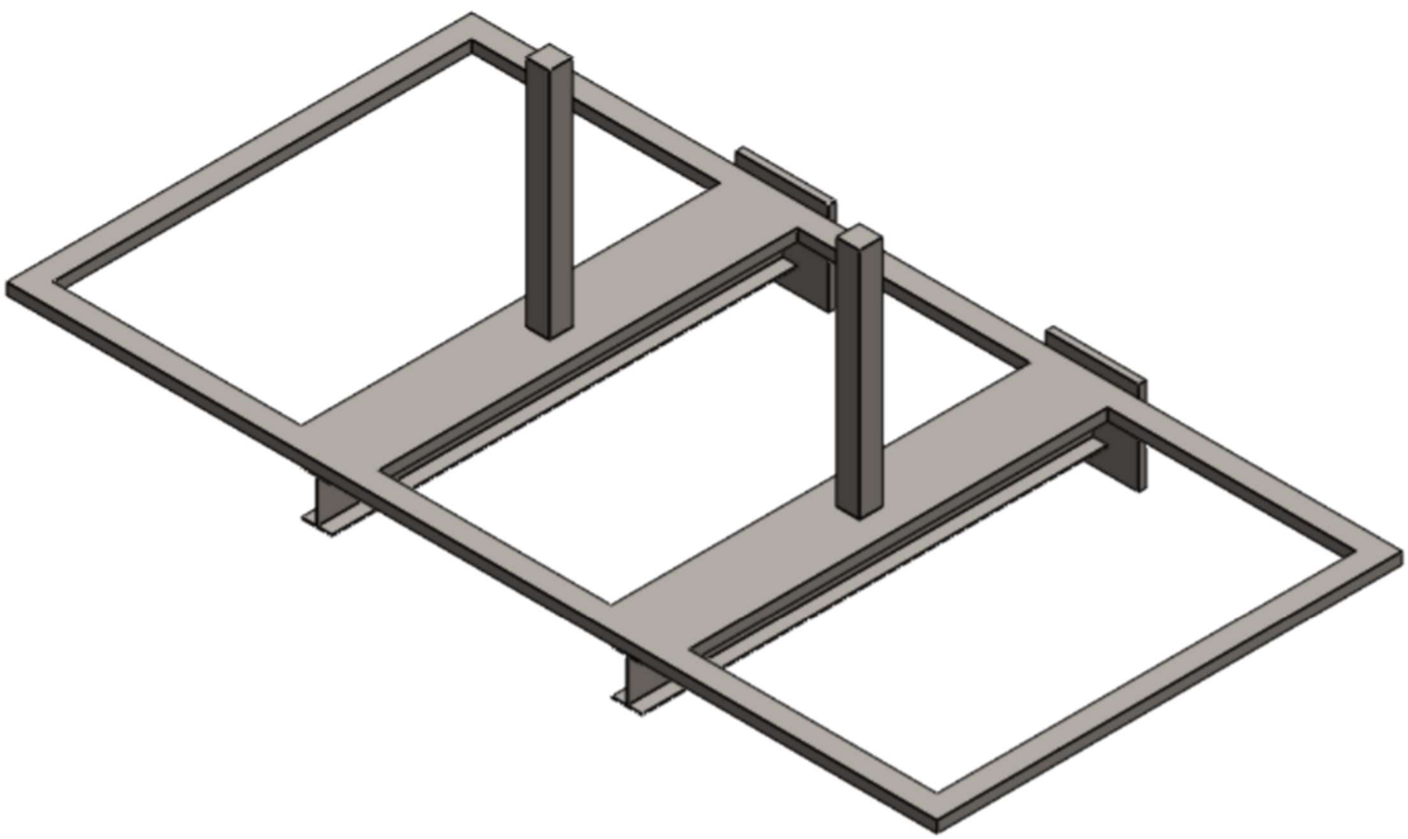
COURSE: EME150A

December 6th, 2019



Abstract

This report presents our team's final design of the apple harvesting vehicle platform. The platform will be mounted onto a vehicle and will support workers as they pick apples in orchards across the country. The design consists of three main components: the frame, grids, and support beams, which are joined together and bolted to the vehicle mounting plates. For the frame, rectangular slots are cut in a steel plate and the grids are put in to provide a light, strong surface to support the workers. The frame also features two rods that stick up perpendicular to the frame surface, which are used to secure the workers via rope and carabiner. The frame and grids are welded to two supporting I-beams, which connect to the two mounting plates on the vehicle.

The most important accomplishment of this design is its ability to withstand the maximum stress conditions provided by the workers and their apples. Several materials were considered, with static stress calculations, buckling and deflection analysis, fatigue life assessment, and cost analysis all being performed before finalization of the design. With some key assumptions, our design satisfied all of the requirements for market.

Fatigue life calculations ultimately resulted in a safety factor of 120 and a lifetime of $7 * 10^6$ cycles. These values outperformed the service requirement of 5 years. Throughout the report, two scenarios were considered. Scenario 1 corresponds to when the platform is static or not moving, and Scenario 2 details when the platform accelerates vertically at $0.25g$. For Scenario 1, the maximum shear stress of the platform frame and I-beam was 301.15 psi and 4735 psi, respectively. In Scenario 2, they came out to be 326.5 psi and 5134.1 psi. Our chosen material of A36 steel is more than capable of handling these stresses with its yield strength of 36.3 ksi. Maximum deflection under maximum load conditions also fell under the maximum requirements, and our weight came out to 1300 lbs. Exchanging the middle sections of the plate for the lightweight grids reduced the platform weight by a factor of ten.

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Project Objectives and Design Summary

The overall objective of the project set forth by AG Systems Mechanical Design Group was to design a safe and efficient apple harvesting vehicle. More specifically, we were tasked with designing the vehicle platform, which would carry workers and their apples and be raised via hydraulic lift. The need for this was in response to the National Institute for Occupational Safety and Health, which estimates there to be thousands of fall-related injuries each year in the Agricultural Industry.

For our platform to be competitive on the market and up to industry standard, there was a list of requirements it had to meet. One was that it had to be joined (either via welding or bolts) to two flat, square mounting plates (one-inch thick, 12-inch sides) which connected to the rest of the vehicle. The platform itself had the following requirements:

Table 1: Platform Requirements

	Requirement(s)		
Category	**1**	**2**	**3**
Static Loading (safety factor = 2.5)	Withstand concentrated load of 2 workers (225 lb each) and two apple bins (50 lb each)	Withstand ground impulse force (vertical acceleration of 0.25g)	Withstand horizontal impulse force of 1000 lb
Dimensions	10 ft width x 5 ft length	Maximum height of 6 ft fully extended	
Deflection (safety factor = 1.5)	Limit to 2 inches anywhere on platform		
Buckling	Resistance to buckling (safety factor = 3)		
Fatigue	Service life of 5 years, 50 days per year (safety factor = 100)	10 times each day the vehicle hits a vertical ground impulse of 0.25g	1000 times each day two workers mount the platform
Other	Corrosion Resistance	Low Cost	

Preliminary research brought us to similar products already on the market. One such product was the deicing truck, which features a small platform supported by a hydraulic lift, with the purpose of supporting the ground staff to safely stand and move around on while performing the de-icing process of airplanes. Another was the bucket truck used by PG&E and other utility companies for access to power lines. Like the de-icing truck, the bucket trucks have a hydraulic lift arm attached to a “bucket” for the person to stand in when servicing power lines. Both of these examples allowed our team to begin analyzing how to best address the axial, bending, and torsional loads that would be present in our design.

Figure 1: Bucket Truck

We ultimately wanted our design to be similar to the current vehicle platform that our potential customer was already using. This entailed a simple, rectangular grid platform with vertical poles sticking out that workers would hook themselves to via safety chords. The major changes we made had to do with the material and supports beams. We selected A36 steel I-beams to bolt onto the mounting plates and run along the five-foot length, essentially acting as cantilever beams in bending and torsion. Mounted atop the beams is an A36 steel plate with rectangular slots cut in three sections. We chose to fill these slots with pre-purchased steel grids, with the intention of reducing the weight and cost of the platform. We finally decided to paint every surface with anti-corrosive paint to avoid wear and tear.

A more detailed look into the design of the platform will follow, along with information on the material, CAD models, load and fatigue analysis, and further recommendations.

Design Presentation

Our platform design can be broken up into three major components—the frame, the grids, and the support beams. These pieces will be joined together in the end to form the full product.

Frame

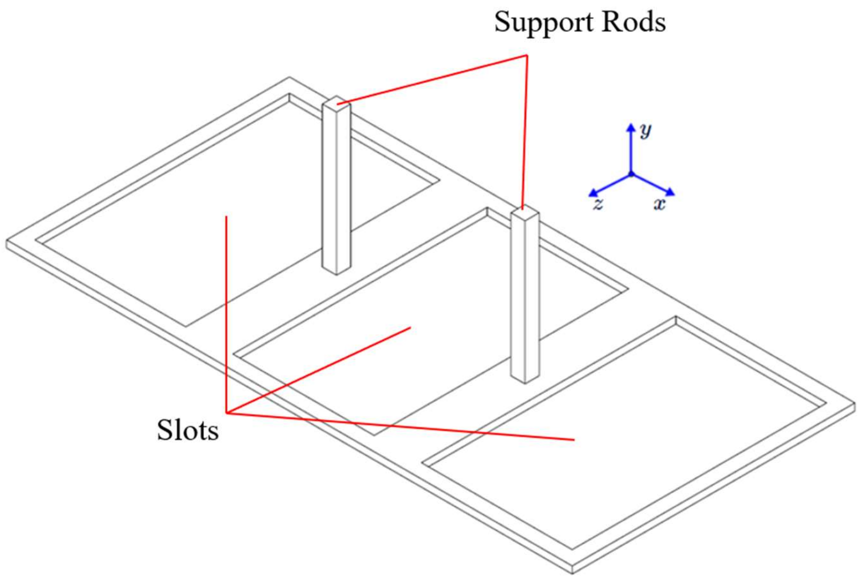

Figure 2: Platform Frame

The frame is the largest component and serves to contain the grids and support the workers. Two support rods stick out at around 1/3rd and 2/3rd of the 10-foot width that the workers can hook onto via rope and carabiner. These rods serve to secure them when they reach out beyond the length of the platform.

Grids

Steel grids will be joined to the frame and fill the empty slots. Like the frame, the grids will serve to support the weight of the workers and their apples. They come in standard widths and can be bolted or welded to each other. Steel and aluminum grids are commonplace in many industries and are typically used for walkways, stairs, and factory platforms. The holes on the surface and hollow interior significantly reduce the weight in comparison to solid block steel, while the high strength allows it to withstand the loads present.

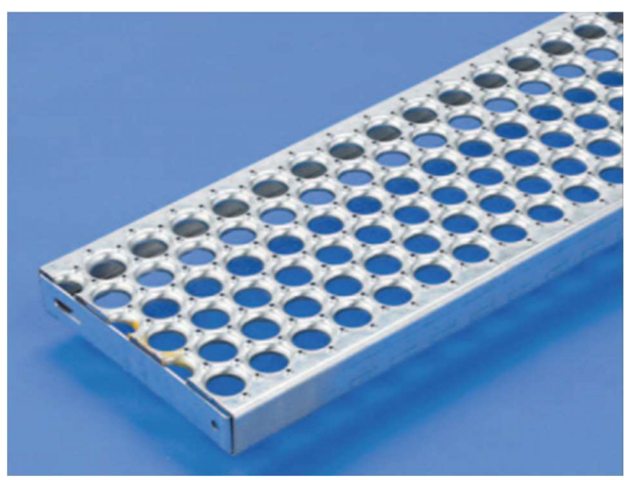

Figure 3: Grids

<u>Support Beams</u>

Two steel I-beams will serve as the main supports of our platform. One 6x9” beam will be bolted onto each mounting plate and extend across the entire five-foot length. The connection between the I-beam face and mounting plate was the area of interest in our stress analysis. The tops will be welded onto the platform frame in the two rectangular sections between the slots. These beams can be purchased in specified height and width combinations for any length.

Figure 4: I-Beam Supports

The material used for the entire platform will be A36 Steel. Material properties for this alloy can be found in Appendix C. Initially, our team had narrowed down possible materials to steel, aluminum, and titanium. The biggest factor in material selection was how well it would perform under various loading scenarios. If the material was strong enough to hold up to our calculated stresses for the specified lifetime, cost and practicality were then considered. Titanium, while the strongest by far, was also the most expensive. Aluminum was cheaper, but also offered the least in favorable material properties. Steel offered properties well-suited to fit the needs of the platform at a modest cost. In addition to this, the suppliers for the grids and I-beams offer their products in A36 steel, making the shaping process more convenient.

Design Documentation

This section details specific dimensions for each platform component.

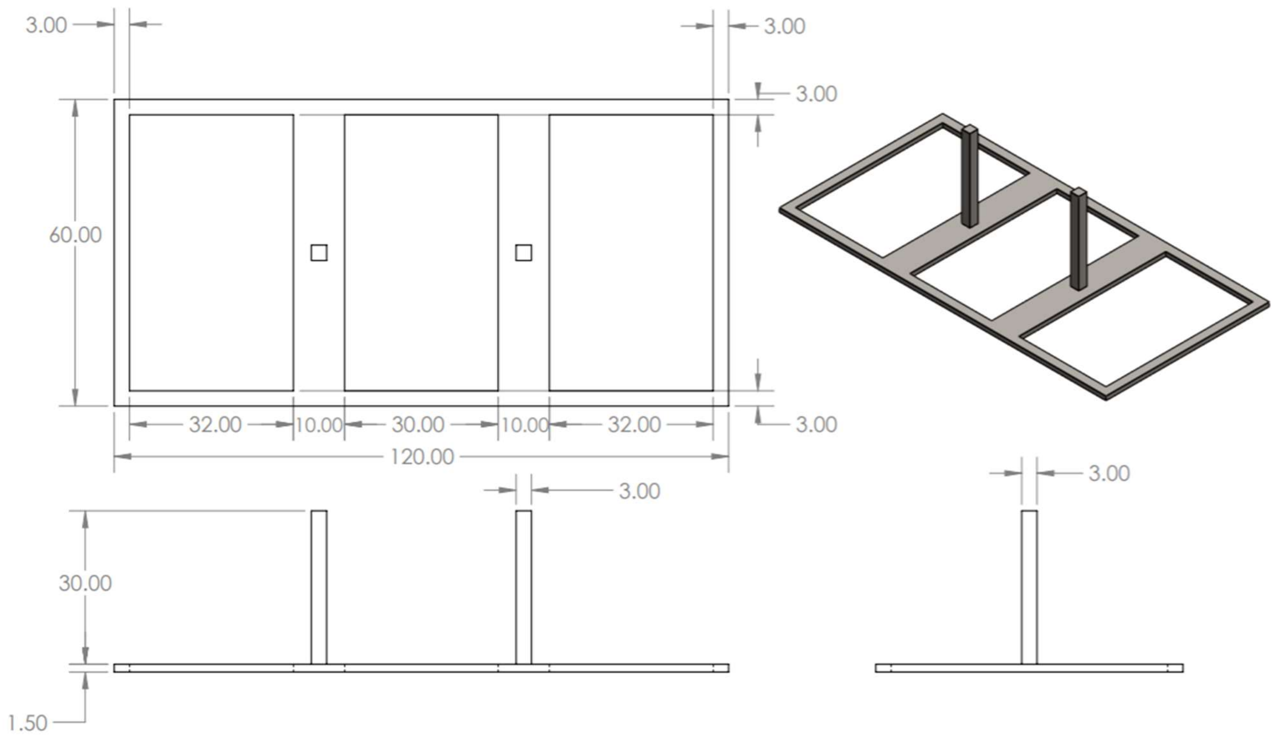

Figure 5: Platform Frame Drawing

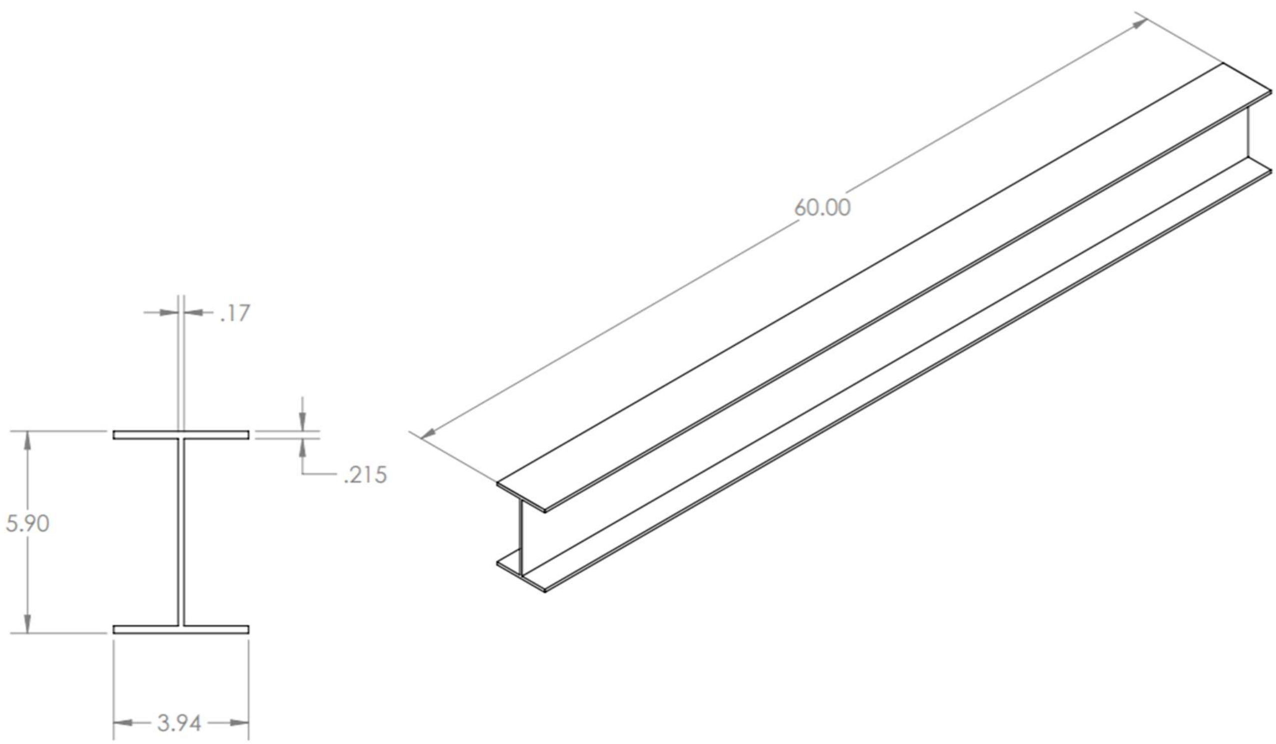

Figure 6: I-Beam Drawing

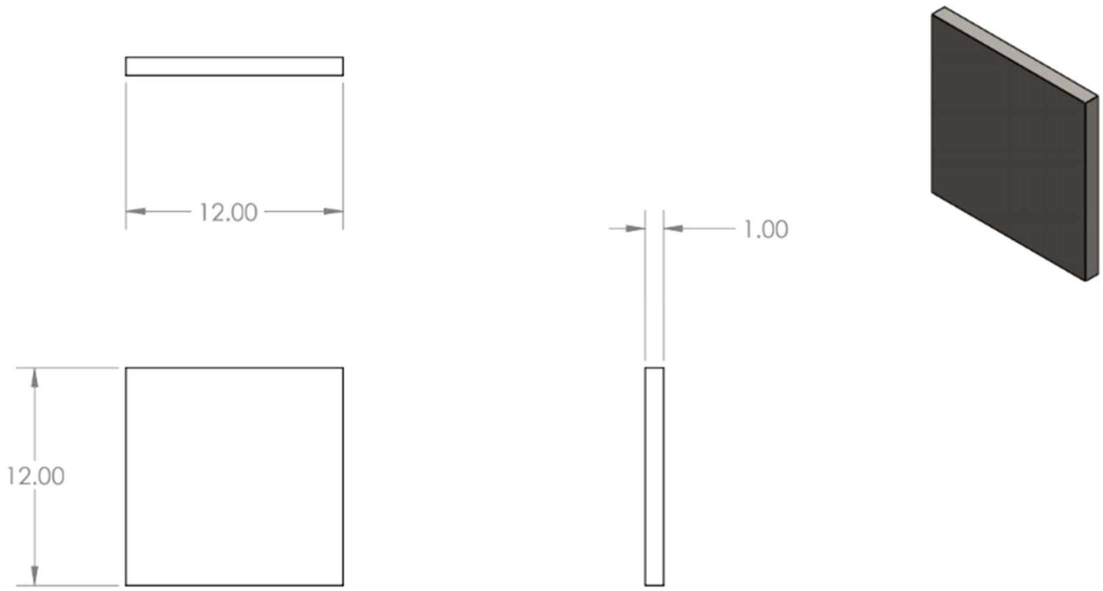

Figure 7: Mounting Plate Drawing

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Platform Frame		1
2	W6X9		2
3	Mounting Plate		2

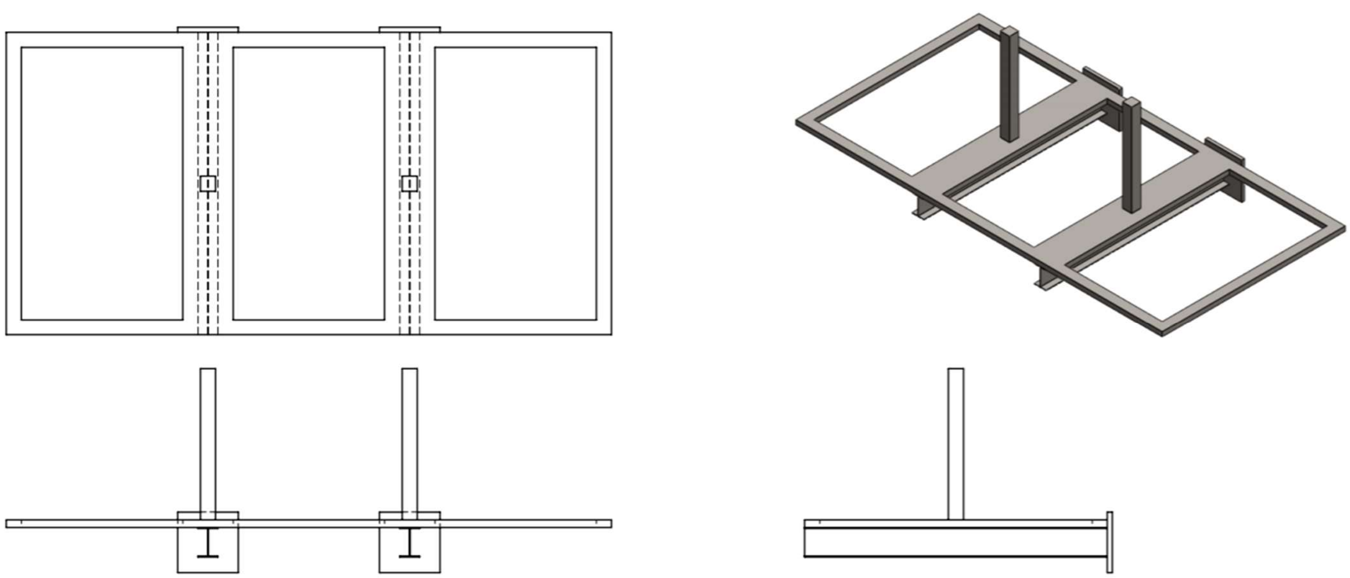

Figure 8: Final Assembly Drawing (All dimensions are included in the components drawing)

Design Analysis

In order to simplify stress, deflection, and fatigue analysis, several assumptions are made regarding the loads and platform itself. These assumptions are listed below:

1. Both I-beams used to support the platform frame are fixed at one end, at each corresponding mounting plate. The I-beams act as cantilever beams, producing reaction forces and moments at the contact point of mounting plate and beam.
2. The supporting forces provided by the I-beams on the platform frame are uniformly distributed along the contact surface between the frame and beam.
3. The vertical deflection of the platform frame is equivalent to the deflection of the I-beams.
4. The design is symmetric with respect to the z-axis. Both reaction forces (at the I-beam and mounting plate interfaces) are equal when no external loading is present (only weight).
5. The platform frame is solely supported by two I-beams.
6. All components are made of the same material and thus have uniform density.
7. The weight of the bolts used to connect the I-beams and platform frame is negligible.
8. The changes in cross section due to screw holes are negligible.
9. The grids are rigid and will not deflect during loading.
10. Each I-beam only receives torsion from the unsupported side sections of the frame.

The following formulas and models are used in the analyses. A full page of nomenclature is included in Appendix A.

(A) Geometric properties:

Moment of inertia

$$(1)\ I = \frac{1}{12}bh^3;\ (2)\ I_{total} = \sum \overline{I_k} + A_k(dy)^2$$

(B) Deflections:

Angle of twist for I-beam

$$\delta_{y_2} = -\phi L = \frac{-TL^2}{GI_T}$$

Castigliano's Theorem

$$U = \int_0^L \frac{\big(M(x)\big)^2}{2EI(x)}dx$$

(C) Stresses:

Axial stress

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

Bending stress

$$\sigma_b = \frac{Mc}{I}$$

Transverse shear stress

$$\tau_{\text{transverse,rectangular}} = \frac{3V_{max}}{2A}$$

$$\tau_{\text{transverse,I-bea}} = \frac{R_{beam}}{A_{web}}$$

Torsional stress [1]

$$\tau_{\text{torsional,I-bea}} = Gt_w\phi' = \frac{Tt_w}{I_T}$$

Principle stresses

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_{max} = \max\left\{\left|\frac{\sigma_1 - \sigma_2}{2}\right|, \left|\frac{\sigma_1 - \sigma_3}{2}\right|, \left|\frac{\sigma_2 - \sigma_3}{2}\right|\right\}$$

(D) Buckling:

Long Columns with Central Loading

$$P_{\text{cr}} = \frac{C\pi^2 EI_{\text{min}}}{L^2}$$

(E) Fatigue design:

Multiaxial Fatigue

$$S_{a\_multiaxial} = \sigma_{von\ Mises} = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}$$

Goodman Correction for Effective Stress Amplitude

$$S_a^{\mathrm{eff}} = S_{notch,a}\left(\frac{S_u}{S_u - S_{not\quad,m}}\right)$$

Marin factors

$$K_i = a_i S_{ut}^{b_I}$$

Note: a and b constants are taken from Parameter Table for Marin Surface Modification. [2]

Fatigue factor of safety

$$N_{design} = \left(\frac{S_a^{eff}}{a}\right)^{1/b}$$

$$a = \frac{(fS_{ut})^2}{S_e}; b = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right)$$

Note: f is taken from fatigue strength fraction figure. [2]

Fatigue factor of safety

$$n_f = \frac{N_{design}}{N_{expected}}$$

(F) Further analysis:

Temperature Effects

$$\delta_T = \alpha \Delta T L_0$$

$$\sigma_T = \alpha \Delta T E$$

Mean Stress

$$S_m = S_{ut}\sqrt{1 - \frac{S_u}{S_e}}$$

$$S_m = \frac{S_y}{1+R}$$

Stress ratio

$$R = \frac{S_{min}}{S_{max}}$$

<u>Models:</u>

A) Static design: MSS theory
B) Fatigue analysis: Method by Shigley, Mischke, and Brown [4]

In picking a material for the platform, the first priority was being able to withstand the loads that would be applied to it with limited deformation. After stress calculations and considerations of price and longevity, we concluded that A36 Steel was the top choice out of the three due to its favorable strength properties. The high density was initially an issue, but changes to the thickness of the plate and the addition of grids made the weight more manageable. Titanium was thought to be the first choice, but the cost made it very unappealing to consumers. Aluminum was in contention, but the low ultimate tensile strength caused concern over fracture after repeated use. Ultimately steel offered the most "bang for the buck" in terms of material properties and cost.

Table 2: Properties of Candidate Materials [3][4][5]

Material	**Density (lb/in^3)**	**Elastic Modulus (ksi)**	**Poisson's Ratio**	**Yield Strength (ksi)**	**Ultimate Strength (ksi)**
A36 Steel Alloy	0.284	29000	0.26	36.3	79.8
6061-T6 Aluminum	0.0975	10000	0.33	40	45
Titanium Ti-6Al-4V (Grade 5)	0.160	16500	0.3	120	131

In order to determine whether our design would satisfy customer requirements, analysis at points of maximum stress in three dimensions were made for two situations. Situation 1 is when the platform is static, and Situation 2 is when the platform accelerates vertically at $0.25g$. To address customer concerns over the strength, deflection, stability, and fatigue, the analytical results related to these concerns are shown in the table below: (MSS theory is used to determine if our design satisfies the strength requirement)

Table 3: Strength of Platform Analytical Results

	Platform Frame	I-beam
Situation 1	$\tau_{max} = 301.15$ psi	$\tau_{max} = 4735$ psi
Situation 2	$\tau_{max} = 326.5$ psi	$\tau_{max} = 5134.1$ psi
Design criteria	$7260 \text{ psi} = \frac{S_y}{2n} \geq \tau_{max}$	

Table 4: Deflection of Platform Analytical Results

	Maximum Deflection
Situation 1	$\lvert\delta_{\max}\rvert = 0.667$ in
Situation 2	$\lvert\delta_{\max}\rvert = 0.676$ in
Design criteria	$1.6 \text{ in} = \frac{2}{n} \geq \lvert\delta_{\max}\rvert$

Table 5: Stability of Platform Analytical Results

	Critical Load
Situation 1/ Situation 2	$P_{\mathrm{cr}} = 75.47$ kips
Design criteria	$P_{\mathrm{cr}} \geq n \times P = 3$ kips

Table 6: Fatigue Life Platform Analytical Results

	Life cycle	Safety factor on life
Situation 1	$N_f = 10^7$ cycles	$n_f = 200$
Situation 2	$N_f = 7 \times 10^6$ cycles	$n_f = 120$
Design criteria	$N_f > 2.5 \times 10^5$ cycles	$(n_f)_{\min} > 100$

As shown in the tables above, we can conclude that our design satisfies every customer requirement.

Recommendations

Temperature Effects:

(a) Temperature effects on dimensions:

A change in operating temperature can cause a body to change its dimensions, which will also lead to variation in stresses within the cross-section. When the temperature increases, the dimensions of the components will expand; whereas if the temperature decreases, parts of the entire platform will contract. Thermal stress will also arise due to the temperature change.

For example, if our customers use this product in Sacramento, we know that the temperature can vary throughout the year from 28.5°F to 105°F. [6] If the dimensions of our product are measured when the surrounding temperature is around 77°F, the variation in temperature will lead to the change indicated below: ($\alpha = 6.5 \times 10^{-6}\ °\mathrm{F}^{-1}$)[7]

$$(\%\delta_T)_{\max} = \frac{\delta_{T,i}}{L_{0,i}} = \alpha \Delta T_{\max} = 0.0182\%$$

$$(\%\delta_T)_{\min} = \frac{\delta_{T,i}}{L_{0,i}} = \alpha \Delta T_{\min} = -0.031525\%$$

From the calculation above, we can see that the temperature variation will lead to the change in geometric properties. This means the value of the stresses at the critical sections will change. Further analysis is needed to determine whether the temperature change during operation will cause the value of stress to fail the static and deflection requirements.

(b) Temperature effects on stresses:

Besides the temperature effects on stresses caused by dimensional variation as mentioned above, the manufacturing process will also lead to the occurrence of stresses that are not taken into consideration during static analysis. For instance, large stresses will occur during welding, since parts experience high heat to join together. For welding steel, parts are typically heated to around 2720 °F. [8] The surrounding area near the place where the melting occurs will experience large stress, as indicated below:

$$(\sigma_T)_{\max} = \alpha (\Delta T)_{\max} E = 498 \text{ ksi}$$

From this, we observe that improper welding processes can lead to distortion and fracture in the critical region. Therefore, further analysis is needed to determine the best way to implement welding.

(c) Temperature effects on the modulus of elasticity:

Because the atomic bonds in materials are much easier to stretch when the temperature increases, the modulus of elasticity E decreases with increased temperature.[9] This means the change in temperature from the reference temperature we take during our design will lead to a change in the deflection and stresses. Therefore, further analysis is needed to determine how the modulus of elasticity is affected by the change in temperature.

Stress Ratio:

Stress ratio R is defined as the ratio of minimum stress to maximum stress:

$$R = \frac{S_{min}}{S_{max}}$$

From different values of R, we use the Goodman correction to predict the life N_f from data in the fully reversed condition. However, this method just approximates the value of N_f. Therefore, further tests are needed to determine the best-fit S-N curves under different R conditions to better understand the related fatigue properties of our design.

Standard Selections:

For fluctuating stress, there exists several fatigue failure criteria, including Modified Goodman and Langer Failure Criteria, Gerber and Langer Failure Criteria and ASME-Elliptic and Langer Failure Criteria. The selection of standards is important because each one will provide different results. For example, when calculating the midrange steady component of stress S_m:

Modified Goodman and Langer Failure Criteria: $S_m = S_{ut}\sqrt{1 - \frac{S_u}{S_e}}$

Gerber and Langer Failure Criteria and ASME-Elliptic and Langer Failure Criteria: $S_m = \frac{S_y}{1+R}$

From the formulas shown above, we will achieve different value of stresses. Therefore, further analysis is needed to determine which criteria can provide the most conservative solution for our design.

Appendices

Appendix A

Nomenclature:

In this section, the following terms are used:

a: Acceleration.

A: Weight of apples carried by each worker.

g: Gravitational acceleration magnitude, which is equal to 32.17405 ft/s^2.

H: Weight of the worker.

I: Moment of inertia.

I_T: St Venant torsion constant

m: Mass.

$\mathcal{M}$: Reference total mass for calculating distributed load.

S_e: Endurance limit.

T: Torsion.

V: Volume.

α: Angular acceleration.

σ: Normal stress.

δ: Deflection.

λ_L: Mass per unit length.

ω: Weight per unit length or distributed load.

ρ: Density.

τ: Shear stress.

Physical and Geometric Properties:

(1) I-beam:

The geometric properties of I-beam can be acquired from the manual related to the geometric properties of structural shapes. [10] The following values are acquired from this kind of manual.

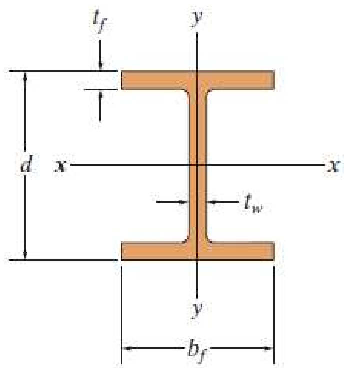

Figure A: I-Beam – Cross Section

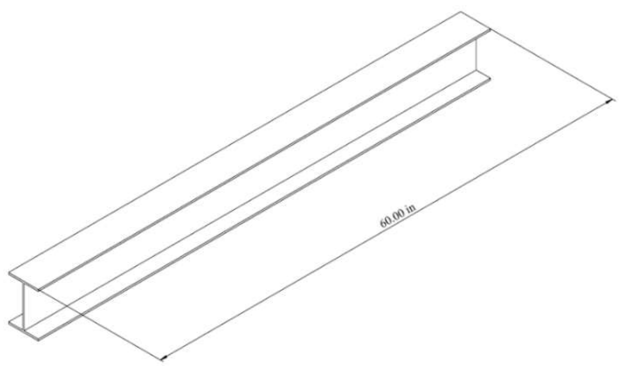

Figure B: I-Beam – Isometric View

Table A: Physical and Geometric Properties of I-beams

Designation $\lceil d \rceil \times \lambda_L$	Area A	Depth d	Web thickness t_w	Flange		I_{xx}	I_{yy}
				Width b_f	Thickness t_f		
in×lbm/ft	in^2	in	in	in	in	in^4	in^4
W6×9	2.68	5.9	0.170	3.940	0.215	16.4	2.19

Therefore, we can calculate the mass and weight of one I-beam:

$$m_{\mathrm{I-Beam}} = \lambda_L \times L_{\mathrm{I-Bea}} = 45\ \mathrm{lb_m}$$

Because 1 lbm has a weight of 1 lbf on planet Earth.

$$W_{\text{I-Bea}} \quad = m_{\text{I-Beam}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_{\text{m}}} = 45\ \text{lbf}$$

(2) Platform frame:

To determine the physical and geometrical properties of platform frame, we separate this component into seven section as shown below:

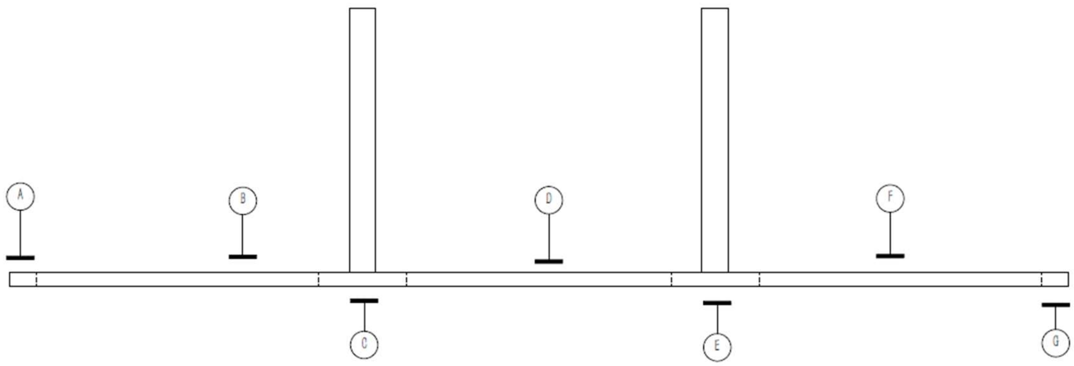

Figure C: Front View of Platform Frame.

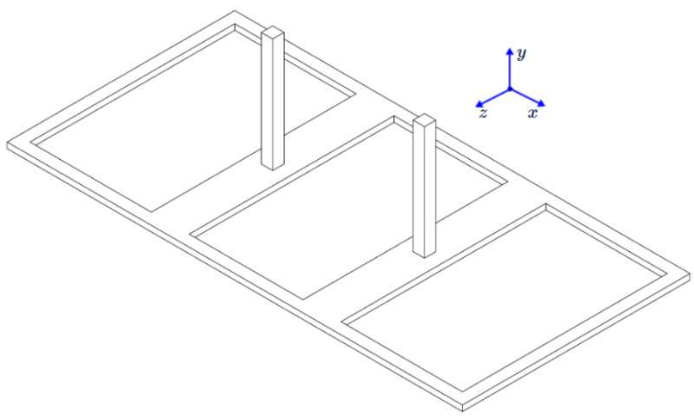

Figure D: Isometric View of Platform Frame.

The geometric properties of platform frame can be determined from the formulas in the Design Analysis section. Therefore, the moment of inertia should be:

$$I_{zz} = \begin{cases} 16.875\ \text{in}^4;\ \text{for section A, C, E, G} \\ \ 1.6875\ \text{in}^4;\ \text{for section B, D, F} \end{cases}$$

(From the assumptions made in the previous section, we only need to determine the moment of inertia at the section C and E that are not supported by I-beams.)

Also, from the material properties in Appendix C, we can calculate the mass and weight of the sections mentioned above:

$$m_{\text{section A}} = \rho_{\text{A36}} \times V_{\text{section A}} = 76.95\ \text{lb}_\text{m};\ W_{\text{section A}} = m_{\text{section A}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 76.95\ \text{lbf}.$$

$$m_{\text{section B}} = \rho_{\text{A36}} \times V_{\text{section B}} = 82.08\ \text{lb}_\text{m};\ W_{\text{section B}} = m_{\text{section B}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 82.08\ \text{lbf}.$$

$$m_{\text{section C}} = \rho_{\text{A36}} \times (V_{\text{section C}} + V_{\text{rod}}) = 333.45\ \text{lb}_\text{m};\ W_{\text{section C}} = m_{\text{section C}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 333.45\ \text{lbf}.$$

$$m_{\text{section D}} = \rho_{\text{A36}} \times V_{\text{section D}} = 76.95\ \text{lb}_\text{m};\ W_{\text{section D}} = m_{\text{section D}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 76.95\ \text{lbf}.$$

$$m_{\text{section E}} = \rho_{\text{A36}} \times (V_{\text{section E}} + V_{\text{rod}}) = 333.45\ \text{lb}_\text{m};\ W_{\text{section E}} = m_{\text{section E}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 333.45\ \text{lbf}.$$

$$m_{\text{section F}} = \rho_{\text{A36}} \times V_{\text{section F}} = 82.08\ \text{lb}_\text{m};\ W_{\text{section F}} = m_{\text{section F}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 82.08\ \text{lbf}.$$

$$m_{\text{section G}} = \rho_{\text{A36}} \times V_{\text{section G}} = 76.95\ \text{lb}_\text{m};\ W_{\text{section G}} = m_{\text{section G}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 76.95\ \text{lbf}.$$

(3) Safety stair treads:

To reduce the cost of manufacturing new parts, we use the product, Perf-O Grip® Metal Safety Stair Treads, as grids to be installed in the slots available on the platform frame. From the product data sheet, we can acquire the following information: [11]

Table B: Properties of Steel Grids

Material Gauge	Width	$\lambda_{L,S}$ (lbm/ft)	Depth
14-gauge steel	10 in	3.5	1.5 in
	12 in	4.3	1.5 in

Then, we can figure out the mass and weight of the stairs installed in the certain sections mentioned above:

$$m_{\text{Stair,section B}} = \sum_{i=1}^{n} \left(\lambda_{L,S}\right)_i \times L_{\text{required,B}} = 50.85\ \text{lb}_\text{m};\ W_{\text{section B}} = m_{\text{section B}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 50.85\ \text{lbf}.$$

$$m_{\text{Stair,section D}} = \sum_{i=1}^{n} \left(\lambda_{L,S}\right)_i \times L_{\text{required,D}} = 47.25\ \text{lb}_\text{m};\ W_{\text{section D}} = m_{\text{section B}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 47.25\ \text{lbf}.$$

$$m_{\text{Stair,section F}} = \sum_{i=1}^{n} \left(\lambda_{L,S}\right)_i \times L_{\text{required,F}} = 50.85\ \text{lb}_\text{m};\ W_{\text{section F}} = m_{\text{section B}} \times \frac{1\ \text{lbf}}{1\ \text{lb}_\text{m}} = 50.85\ \text{lbf}.$$

Appendix B

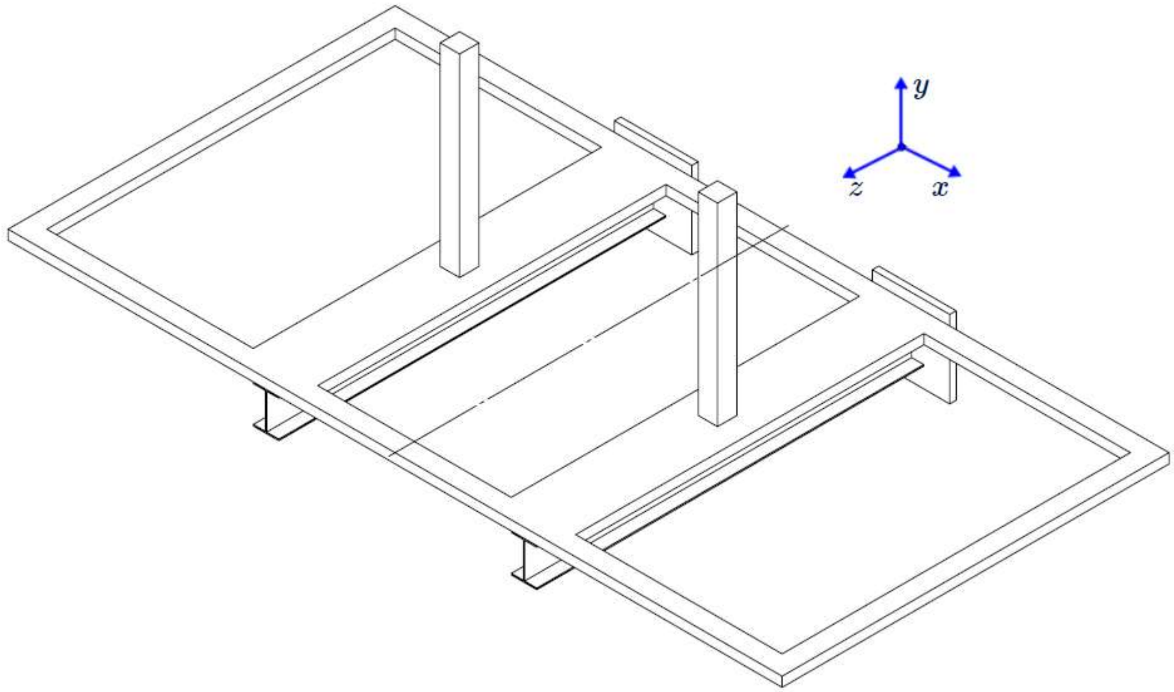

Figure E: Full Assembly – Isometric View

In order to determine if our design satisfies the static criteria, we need to consider the critical section on each component. For the platform frame, the critical section is where the bending moment is largest. For I-beams, the critical section is where the I-beam is fixed to the mounting plate. First, the forces, moments, and torque on the components are determined:

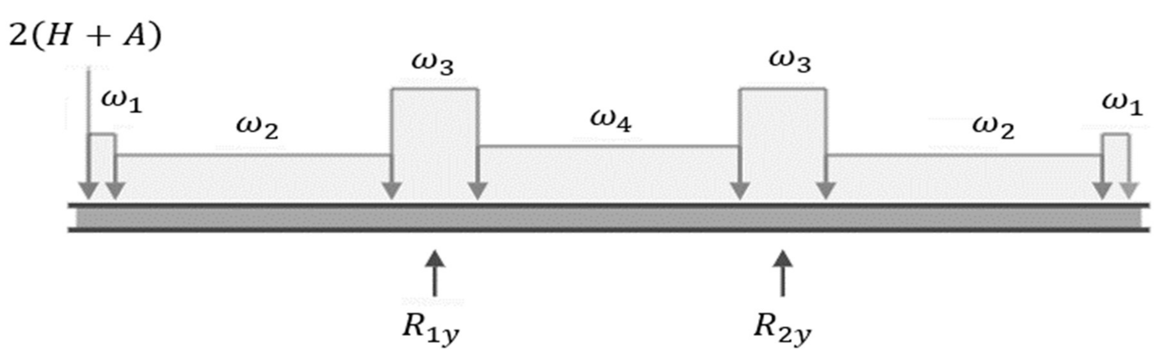

Figure F: Free-body Diagram of Platform Frame for Deflection Analysis in xy-plane.

($\omega_i = W_i/w_i$, where M_i is the total weight of the section, w_i is the width of this section.)

By free-body diagram shown above, the following relationships are set up:

$$\Sigma F_{\text{vertical}} = R_{1y} + R_{2y} - 2(H + A) - \omega_1 \times 6 - \omega_2 \times 64 - \omega_3 \times 20 - \omega_4 \times 30 = \mathcal{M}a$$

$$\begin{aligned}\Sigma M = {} & 2(H + A) \times 40 + R_{2y} \times 40 + \omega_1 \times 3 \times (30 - 1.5) + \omega_2 \times 32 \times (40 - 3 - 16) \\ & - \omega_4 \times 30 \times (15 + 5) - \omega_3 \times 10 \times (5 + 30 + 5) \\ & - \omega_2 \times 32 \times (5 + 30 + 10 + 16) - \omega_1 \times 3 \times (5 + 30 + 10 + 32 + 1.5) = 0\end{aligned}$$

If the platform is static (No acceleration, $a = 0$), the reaction forces are:

$$R_{1y} = 1705.5 \text{ lbf}; R_{2y} = 55.45 \text{ lbf}.$$

Then, the stresses on each component are evaluated:

(I) For platform frame

By singularity function, we can get:

$$q(x) = -2(H + A)\langle x\rangle^{-1} - [\omega_1\langle x\rangle^0 - \omega_1\langle x - 3\rangle^0] - [\omega_2\langle x - 3\rangle^0 - \omega_2\langle x - 35\rangle^0] - [\omega_3\langle x - 35\rangle^0 - \omega_3\langle x - 45\rangle^0] \ldots$$

$$\ldots - [\omega_4\langle x - 45\rangle^0 - \omega_4\langle x - 75\rangle^0] - [\omega_3\langle x - 75\rangle^0 - \omega_3\langle x - 85\rangle^0] - [\omega_2\langle x - 85\rangle^0 - \omega_2\langle x - 117\rangle^0] - [\omega_1\langle x - 117\rangle^0] \ldots$$

$$\ldots + R_{1y}\langle x - 40\rangle^{-1} + R_{2y}\langle x - 80\rangle^{-1}$$

By taking the integral on $q(x)$, we can get $V(x)$ and $M(x)$: (As long as the $q(x)$ is complete, integration constants are unnecessary for $V(x)$ and $M(x)$.)

$$\begin{aligned}V(x) = {} & -2(H + A)\langle x\rangle^0 - [\omega_1\langle x\rangle^1 - \omega_1\langle x - 3\rangle^1] - [\omega_2\langle x - 3\rangle^1 - \omega_2\langle x - 35\rangle^1] - [\omega_3\langle x - 35\rangle^1 - \omega_3\langle x - 45\rangle^1] \\ & - [\omega_4\langle x - 45\rangle^1 - \omega_4\langle x - 75\rangle^1] - [\omega_3\langle x - 75\rangle^1 - \omega_3\langle x - 85\rangle^1] - [\omega_2\langle x - 85\rangle^1 - \omega_2\langle x - 117\rangle^1] \\ & - [\omega_1\langle x - 117\rangle^1] + R_{1y}\langle x - 40\rangle^0 + R_{2y}\langle x - 80\rangle^0\end{aligned}$$

$$\begin{aligned}M(x) = {} & -2(H + A)\langle x\rangle^1 - 0.5[\omega_1\langle x\rangle^2 - \omega_1\langle x - 3\rangle^2] - 0.5[\omega_2\langle x - 3\rangle^2 - \omega_2\langle x - 35\rangle^2] - 0.5[\omega_3\langle x - 35\rangle^2 - \omega_3\langle x - 45\rangle^2] \\ & - 0.5[\omega_4\langle x - 45\rangle^2 - \omega_4\langle x - 75\rangle^2] - 0.5[\omega_3\langle x - 75\rangle^2 - \omega_3\langle x - 85\rangle^2] \\ & - 0.5[\omega_2\langle x - 85\rangle^2 - \omega_2\langle x - 117\rangle^2] - 0.5[\omega_1\langle x - 117\rangle^2] + R_{1y}\langle x - 40\rangle^1 + R_{2y}\langle x - 80\rangle^1\end{aligned}$$

From the expression of singularity function, we can determine the maximum value of shear force and bending moment, as shown below:

$$|M_{max}| = 27103 \text{ lbf} \cdot \text{in}$$

$$|V_{max}| = 875.78 \text{ lbf}$$

$$(I_{zz})_{\text{corresponding}} = 16.875 \text{ in}^4$$

Therefore, we can determine the following stresses with the conditions presented above:

(1) Bending stress:

$$\sigma_b = \frac{M_{max}(\frac{t_p}{2})}{(I_{zz})_{\text{corresponding}}} = 602.3 \text{ psi}$$

(2) Transverse shear stress:

$$\tau_{\text{transverse}} = \frac{3V_{max}}{2A} = 292 \text{ psi}$$

(II) I-beam

In yz-plane,

Figure G: Free-body Diagram of Beam with Larger Load in yz-plane.

Considering all the loads on the beam from the free-body diagram, we can get:

$$\omega_{\text{total}} = \omega_{\text{Beam}} + \omega_{\text{Supporting}} = \frac{W_{\text{I-Beam}}}{L} + \frac{R_{1y}}{L} = 29.175 \text{ lbf/in}$$

Then,

$$R_{beam} = \omega_{\text{total}}L = 1750.5 \text{ lbf}$$

$$M_{beam} = \frac{\omega_{\text{total}}L^2}{2} = 52515 \text{ lbf} \cdot \text{in}$$

Also, based on the assumption, we can determine the torsion by following steps:

$$T = 2(H + A) \times 40 + \omega_1 \times 3 \times (30 - 1.5) + \omega_2 \times 32 \times (40 - 3 - 16) + \omega_3 \times 4.03 \times \left(\frac{4.03}{2}\right) = 26187.57 \text{ lbf}$$

Then, we can determine the following stresses:

(1) Bending stress:

$$\sigma_b = \frac{M_{beam}(\frac{d}{2})}{I_{xx}} = 9446 \text{ psi}$$

(2) Transverse shear stress:

$$\tau_{\text{transverse}} = \frac{R_{beam}}{A_{web}} = 653.2 \text{ psi}$$

(3) Torsional shear stress:

$$\tau_{\text{torsional}} = G t_w \phi' = \frac{T t_w}{I_T} = 239.5 \text{ psi}$$

Also, for I-beam, I_T is equal to:

$$I_T = I_{xx} + I_{yy} = 18.59 \text{ in}^4$$

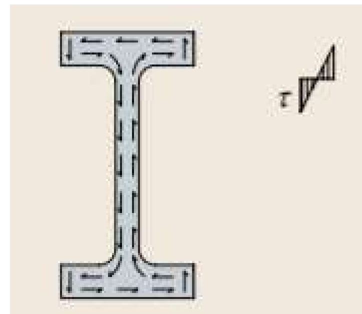

Figure H: Torsion in I-Beam

Therefore, the value of $\tau_{\text{torsional}}$ should be:

$$\tau_{\text{torsional}} = 239.5 \text{ psi}$$

In xy-plane,

Based on the assumptions, there exist some stresses on flange of I-beam,

$$\omega_7 = \frac{R_{1y}}{b_f} = 432.9 \text{ lbf/in}$$

Therefore, the maximum stress at the flange is:

$$\sigma_{\text{b,flange}} = \frac{\frac{1}{8}\omega_7 b_f^2}{L t_f\left(\frac{d}{2} - \frac{t_f}{2}\right) + \frac{1}{12} t_f^3} = 22.9 \text{ psi}$$

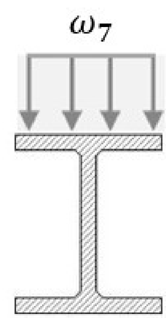

Figure I: I-Beam Loading

After finishing the calculations mentioned above, we need to finally determine the value of $\tau_{\max}$ under MSS theory.

(1) Platform frame:

The point of greatest stress is located at top surface of the cross section. Then, we can find the $\tau_{\max}$:

$$\sigma_x = 602.3 \text{ psi} \,;\, \sigma_y = 0 \,;\, \tau_{xy} = 0 \;;$$

$$\Rightarrow \sigma_1 = 602.3 \text{ psi} \,;\, \sigma_2 = 0 \,;\, \sigma_3 = 0 \;;$$

$$\Longrightarrow \tau_{ma} == \max\left\{\left|\frac{\sigma_1 - \sigma_2}{2}\right|, \left|\frac{\sigma_1 - \sigma_3}{2}\right|, \left|\frac{\sigma_2 - \sigma_3}{2}\right|\right\} = 301.15 \text{ psi}$$

Also, by design criteria, the system should satisfy: ($n = 2.5$)

$$\frac{S_y}{2n} \geq \tau_{max}$$

Because $S_y/2n = 7260$ psi for A36 steel, τ_{ma} satisfies the inequality. Therefore, we can determine that the platform meets the static requirement.

(2) I-beam:

From the previous calculation, we can determine that the worst point is located at the top corner. Then, we can find the $\tau_{\max}$ at this point:

$$\sigma_x = 9446 \text{ psi}; \sigma_y = 22.9 \text{ psi}; \tau_{xy} = 239.5 \text{ psi};$$

$$\because \sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\Rightarrow \sigma_1 = 9470 \text{ psi}; \sigma_2 = 17 \text{ psi}; \sigma_3 = 0 \text{ psi};$$

$$\Longrightarrow \tau_{max} == \max\left\{\left|\frac{\sigma_1 - \sigma_2}{2}\right|, \left|\frac{\sigma_1 - \sigma_3}{2}\right|, \left|\frac{\sigma_2 - \sigma_3}{2}\right|\right\} = 4735 \text{ psi}$$

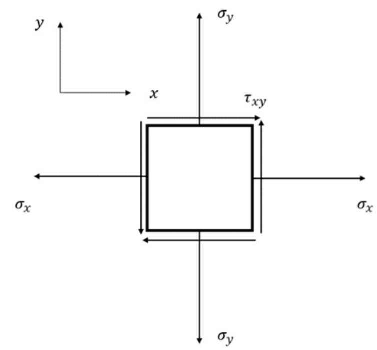

Figure J: Principal Stress Diagram

Also, by design criteria, the system should satisfy: ($n = 2.5$)

$$\frac{S_y}{2n} \geq \tau_{max}$$

Because $S_y/2n = 7260$ psi for A36 steel, τ_{ma} satisfies the inequality. Therefore, we can determine that the platform meets the static requirement when the platform stays still.

While the platform keeps still or undergoes constant speed lifting, a ground impulse force exists which cam causes the platform to accelerate vertically at $0.25g$.

In this situation, the force and moment for the platform can be determined by:

$$\Sigma F_{\text{vertical}} = \mathcal{M}a$$

$$\Sigma M_{\text{centeroid}} = I_{\text{centeroid}}\alpha$$

Because the I-beam avoid the whole platform rotating, $\alpha = 0$. Therefore, we can get:

$$R_{1y} = 1856.8125 \text{ lbf}; R_{2y} = 206.8125 \text{ lbf}.$$

Using the same steps mentioned above, we can acquire the new set of values $\tau_{\max}$:

Platform frame	I-beam
326.5 psi	5134.1 psi

Therefore, we can see that design criteria, $\frac{S_y}{2n} \geq \tau_{max}$, can still be satisfied.

Appendix C

Table C: Properties of Candidate Materials

Material	**Density (lb/in^3)**	**Elastic Modulus (ksi)**	**Poisson's Ratio**	**Yield Strength (ksi)**	**Ultimate Strength (ksi)**
A36 Steel	0.284	29000	0.26	36.3	79.8
6061-T6 Aluminum	0.0975	10000	0.33	40	45
Titanium Ti-6Al-4V (Grade 5)	0.160	16500	0.3	120	131

Appendix D

Because of the existence of horizontal impulse forces from accidental collisions, which can be as great as 1000lbf, the platform frame may buckle.

Figure K: Horizontal Impulse Forces from Accidental Collisions.

In this case, because I-beam can produce forces in the horizontal directions so that only unsupported side sections of the frame experience compressive forces. Due to the existence of I-beam, we can treat the sections as long columns with central loading under one end free and one end fixed. Therefore, we can determine: ($C = 0.25$)

$$P_{\mathrm{cr}} = \frac{C\pi^2 E I_{\mathrm{min}}}{L^2}$$

For this situation,

$$L = 40\text{ in}, E = 29 \times 10^6\text{ psi}, I_{\mathrm{min}} = \frac{1}{12} \times 2 \times 3 \times 1.5^3 = 1.6875\text{ in}^4.$$

Therefore,

$$P_{\mathrm{cr}} = 75.47\text{ kips}$$

Also, by design criteria, the system should satisfy: ($n = 3$)

$$P_{\mathrm{cr}} \geq n \times P$$

Therefore, our design will not buckle.

Appendix E

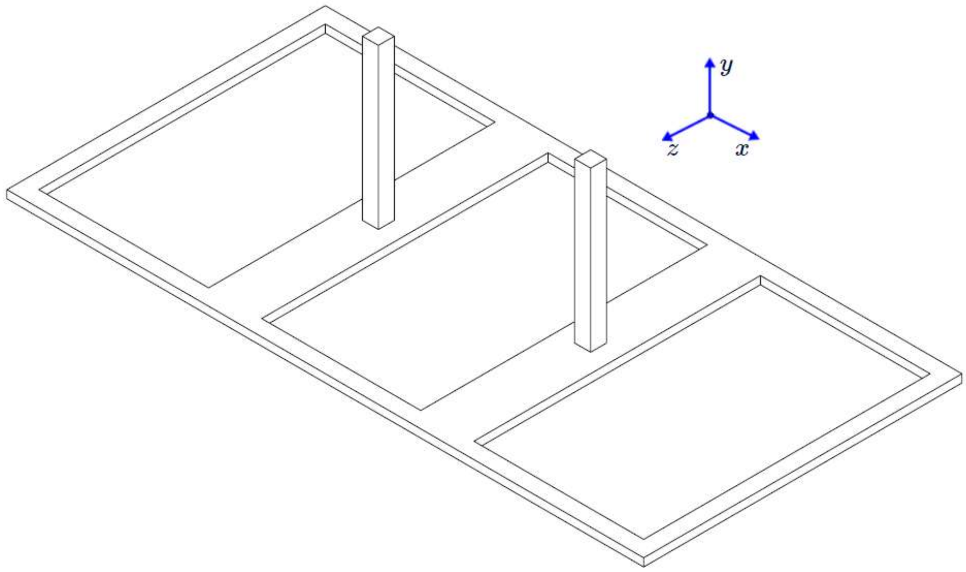

Figure L: Platform Frame Isometric View

In order to satisfy the design criteria, we need to consider the point of greatest deflection under the greatest-deflection condition. This is located at the outside corner when two workers stand on it. To solve the value of $|\delta_{\max}|$ in y-direction, four steps are taken:

Step 1: Determine the forces on the components.

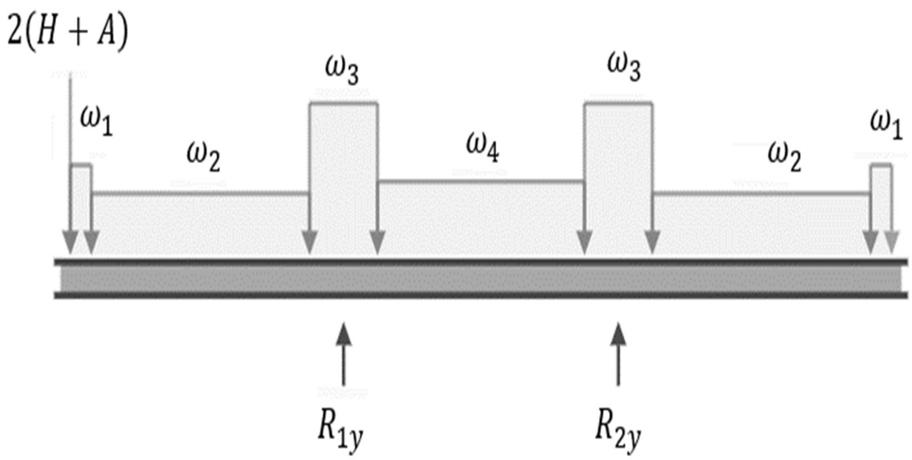

Figure M: Free-body Diagram of Platform Frame for Deflection Analysis in xy-plane.

$$\Sigma F_{\text{vertical}} = R_{1y} + R_{2y} - 2(H + A) - \omega_1 \times 6 - \omega_2 \times 64 - \omega_3 \times 20 - \omega_4 \times 30 = \mathcal{M}a$$

$$\Sigma M = 2(H + A) \times 40 + R_{2y} \times 40 + \omega_1 \times 3 \times (30 - 1.5) + \omega_2 \times 32 \times (40 - 3 - 16) - \omega_4 \times 30 \times (15 + 5) - \omega_3 \times 10 \times (5 + 30 + 5) - \omega_2 \times 32 \times (5 + 30 + 10 + 16) - \omega_1 \times 3 \times (5 + 30 + 10 + 32 + 1.5) = 0$$

If the platform is static (no acceleration, $a = 0$), the reaction forces are:

$$R_{1y} = 1705.5 \text{ lbf}; R_{2y} = 55.45 \text{ lbf}.$$

Step 2: Determine the value of δ_{y_1} due to the deflection of I-beam:

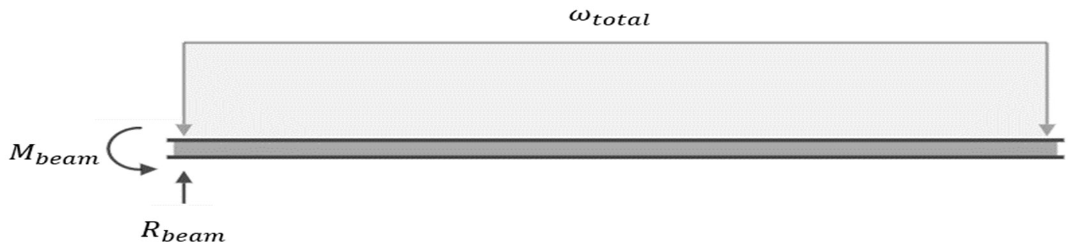

Figure N: Free-body Diagram of Beam with Larger Load in yz-plane.

Considering all the loads on the beam from the free-body diagram, we get:

$$\omega_{\text{total}} = \omega_{\text{Beam}} + \omega_{\text{Supporting}} = \frac{W_{\text{I-Beam}}}{L} + \frac{R_{1y}}{L} = 29.175 \text{ lbf/in}$$

$$\delta_{y1} = -\frac{\omega_{\text{total}} L^4}{8E(I_{xx})_{\text{beam}}} = -0.099 \text{ in}$$

Step 3: Determine the value of δ_{y_2} due to the angle of twist for the I-beam:

From our assumptions, we can determine the torsion T on the I-beam closest to the load by:

$$T = 2(H + A) \times 40 + \omega_1 \times 3 \times (30 - 1.5) + \omega_2 \times 32 \times (40 - 3 - 16) + \omega_3 \times 4.03 \times (\frac{4.03}{2})$$

Then, we can determine the value of δ_{y_2} due to the angle of twist of the I-beam

$$\delta_{y_2} = -\phi L = \frac{-TL^2}{GI_T} = \frac{-TL^2}{\frac{E}{2(1 + \nu)} I_T}$$

For I-beams, I_T is equal to:

$$I_T = I_{xx} + I_{yy}$$

Therefore, the value of δ_2 should be:

$$\delta_{y_2} = -0.441 \text{ in}$$

Step 4: Determine the value of δ_{y_3} due to the deflection of the platform frame:

From singularity functions between $x \in [0,38.03]$, we get:

$$q(x) = 2(H + A)\langle x\rangle^{-1} - [\omega_1\langle x\rangle^0 - \omega_1\langle x - 3\rangle^0] - [\omega_2\langle x - 3\rangle^0 - \omega_2\langle x - 35\rangle^0] - \omega_3\langle x - 35\rangle^0$$

By taking the integral, we can get $V(x)$ and $M(x)$: (As long as the $q(x)$ is complete, integration constants are unnecessary for $V(x)$ and $M(x)$)

$$V(x) = 2(H + A)\langle x\rangle^0 - [\omega_1\langle x\rangle^1 - \omega_1\langle x - 3\rangle^1] - [\omega_2\langle x - 3\rangle^1 - \omega_2\langle x - 35\rangle^1] - \omega_3\langle x - 35\rangle^1$$

$$M(x) = 2(H + A)\langle x\rangle^1 - \frac{1}{2}[\omega_1\langle x\rangle^2 - \omega_1\langle x - 3\rangle^2] - \frac{1}{2}[\omega_2\langle x - 3\rangle^2 - \omega_2\langle x - 35\rangle^2] - \frac{1}{2}\omega_3\langle x - 35\rangle^2$$

Then, by Castigliano's Theorem, we can get: ($L = 38.03$ in)

$$U = \int_0^L \frac{\left(M(x)\right)^2}{2EI(x)} dx$$

$$\because \delta_{y3} = \frac{\partial U}{\partial P} = \int_0^L \frac{M(x)}{EI(x)} \frac{\partial M(x)}{\partial P} dx$$

$$\text{Also}, P = 2(H + A); \ \frac{\partial M(x)}{\partial P} = x$$

$$\Longrightarrow \delta_{y3} = -0.127 \text{ in}$$

Finally, we can acquire the value of $|\delta_{\max}|$ by:

$$|\delta_{\max}| = |\delta_{y_1}| + |\delta_{y_2}| + |\delta_{y_3}| = 0.667 \text{ in}$$

Also, by design criteria, the system should satisfy: ($n = 1.25$)

$$1.6 \text{ in} = \frac{2}{n} \geq |\delta_{\max}|$$

Therefore, our design satisfies the static deflection criteria.

A ground impulse force exists which can cause the platform to accelerate vertically at $0.25g$.

In this scenario, the force and moment for the platform can be determined by:

$$\Sigma F_{\text{vertical}} = \mathcal{M}a$$

$$\Sigma M_{\text{centeroid}} = I_{\text{centeroid}}\alpha$$

Because the I-beam prevents the whole platform from rotating, $\alpha = 0$. Therefore, we can get:

$$R_{1y} = 1856.8125 \text{ lbf}; R_{2y} = 206.8125 \text{ lbf}.$$

Using the same steps mentioned above, we can acquire the new set of values for deflection:

$$\delta'_{y1} = -\frac{\omega'_{\text{total}} L^4}{8E(I_{xx})_{\text{beam}}} = -0.1076\text{in}$$

$$\delta'_{y2} = \delta_{y2}$$

$$\delta'_{y3} = \int_0^L \frac{M'(x)}{EI(x)} \frac{\partial M'(x)}{\partial P} dx = -0.127 \text{ in}$$

$$|\delta_{\max}| = |\delta'_{y_1}| + |\delta_{y_2}| + |\delta'_{y_3}| = 0.676 \text{ in}$$

Therefore, we can see that design criteria, $\frac{2}{n} \geq |\delta_{\max}|$, can still be satisfied even with added ground impulses.

Appendix F

(a) Life cycle:

In order to analyze the fatigue life of our design, we assume that the cross section of the I beam behaves like a rectangular filleted bar in bending. From this, we can perform a fatigue calculation in a conservative way based on the chart of theoretical stress concentration factor K_t. Because the filleted radius is r = 0.3 in, [12] the dimensions indicated in the figure below can be used to find the ratios needed for K_t.

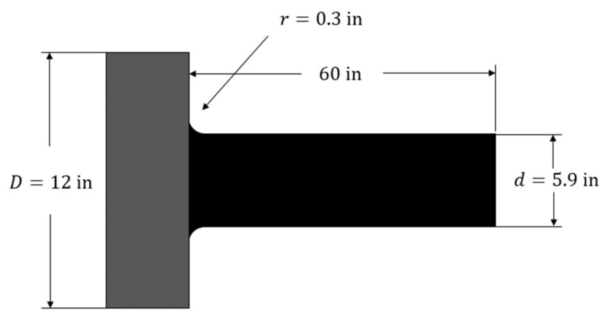

Figure O: Cross Section for Fatigue Life Analysis

$$\frac{D}{d} = 2.034 \qquad \frac{r}{d} = 0.051$$

From the graphs [2], these dimensions yield:

$$K_{\text{t\_bending}} = 2.3 \text{ and } K_{\text{t\_torsion}} = 2.8$$

Notch Sensitive for bending $q_b = 0.9$

Notch Sensitive for torsion $q_{tor} = 0.9$

$$\sigma_{notch,x} = \sigma_x K_{\text{t\_bending}} = 21726 \text{ psi}$$

$$\sigma_{notch,y} = \sigma_y K_{\text{t\_bending}} = 59.94 \text{ psi}$$

$$\tau_{not} = \tau_{xy} K_{\text{t\_torsion}} = 670.6 \text{ psi}$$

Because $S_{ut} < 70$ ksi for A36 steel, we can determine: [1] [13]

Endurance limit: $S'_e = 0.5S_{ut} = 39900$ psi

Surface factor: $K_a = 2.7(S_{ut})^{-0.265} = 0.135$

Side factor: $K_b = 0.91(d)^{-0.157} = 0.689$

Modified endurance strength: $S_e = K_a K_b S'_e = 3711$ psi

Stress concentration factor for bending: $K_f = 1 + q(K_t - 1) = 2.17$

Stress concentration factor for torsion: $K_{fs} = 1 + q(K_{t_{torsion}} - 1) = 2.62$

$\sigma_{x,a} = \frac{\sigma_{notch,x}}{2} = 10863$ psi	$\sigma_{y,a} = \frac{\sigma_{notch,y}}{2} = 29.97$ psi	$\tau_{xy,a} = \frac{\tau_{notch}}{2} = 335.3$ psi
$\sigma_{x,m} = \frac{\sigma_{notch,x}}{2} = 10863$ psi	$\sigma_{y,m} = \frac{\sigma_{notch,y}}{2} = 29.97$ psi	$\tau_{xy,m} = \frac{\tau_{notch}}{2} = 335.3$ psi

$$\sigma_{1,a} = \frac{\sigma_x+\sigma_y}{2} + \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + (\tau_{xy,a}^2)} = 10879 \text{ psi}$$

$$\sigma_{2,a} = \frac{\sigma_x+\sigma_y}{2} - \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + (\tau_{xy,a}^2)} = \text{-16 psi}$$

$$\sigma_{1,m} = \frac{\sigma_x+\sigma_y}{2} + \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + (\tau_{xy,m}^2)} = 10879 \text{ psi}$$

$$\sigma_{2,m} = \frac{\sigma_x+\sigma_y}{2} - \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + (\tau_{xy,m}^2)} = \text{-16 psi}$$

$$\sigma_{eq,a} = \frac{\sqrt{2}}{2}\sqrt{\left(\tau_{1,a} - \tau_{2,a}\right)^2 + \tau_{1,a}^2 + \tau_{2,a}^2} = 10887 \text{ psi}$$

$$\sigma_{eq,m} = \sigma_{1,m} + \sigma_{2,m} + \sigma_{3,m} = 10863 \text{ psi}$$

By Goodman correction,

$$S_a^{\text{eff}} = \sigma_{eq,a}\left(\frac{S_u}{S_u - \sigma_{eq,m}}\right) = 25150 \text{ psi} = 173.3 \text{ MPa}$$

From S-N graph of A36 Steel [13], we can get:

$$\left(N_f\right)_{\text{static}} = 10^7 \text{ cycles.}$$

Because our design only needs 1000 cycles/ day for 5 years with 50 days of picking annually, the design criteria is satisfied.

The ground impulse force mentioned previously can change the stress. Following the same steps mentioned above, we can get:

$$\left(N_f\right)_{\text{with acceleration}} = 7 \times 10^6 \text{ cycles.}$$

Therefore, our design still satisfies the design criteria.

(b) Safety factor on life:

In order to calculate the safety factor on life, we use a different method to predict the safety factor n_f.[14] Recall from the analysis on the stress element for the I-beam at the worst point:

$$\sigma_x = 9446 \text{ psi}; \sigma_y = 22.9 \text{ psi}; \tau_{xy} = 239.5 \text{ psi};$$

For $S_{a\_multiaxial}$,

$$S_{a,multiaxial} = \sigma_{von\ Mises} = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2} = 9461.51 \text{ psi}$$

For $S_{m\_multiaxial}$,

$$S_{m,multiaxial} = \sigma_{x,m} + \sigma_{y,m} = \frac{\sigma_x + \sigma_y}{2} = 4734.45 \text{ psi}$$

Because $S_{ut} = 79.8$ ksi for A36 steel, we can determine: [1] [13]

$$S_e' = 0.5 S_{ut}$$

By Goodman correction:

$$S_{a,multiaxial}^{\text{eff}} = S_{a,multiaxial}\left(\frac{S_{ut}}{S_{ut} - S_{m,multiaxial}}\right) = 10057 \text{ psi}$$

Because safety factor on life n_f, yields a safety factor with respect to S_e' and S_{ut} with the assumption that there is no damage summation of stress history cycles, we need to figure out how $S_{a,multiaxial}^{\text{eff}}$ is related to $N_{design,multiaxial}$. From the figure below we can get:

$$f \approx 0.88$$

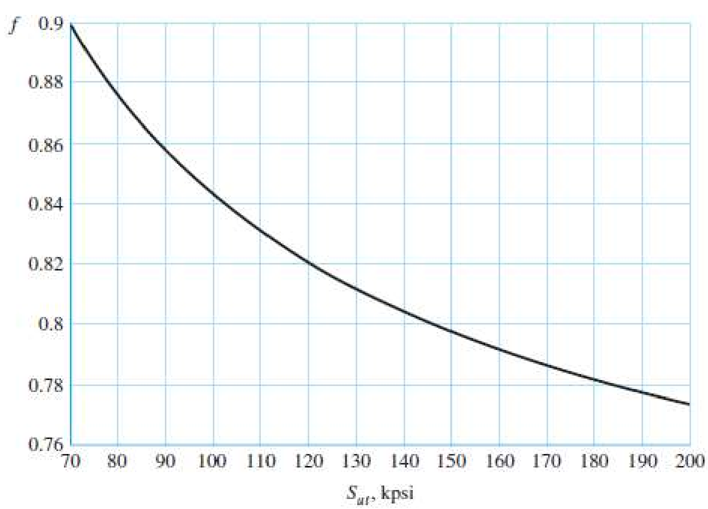

Figure P: Fatigue Strength Fraction Diagram. [2]

Then, using the equation from the formulas section we can get:

$$N_{design,multiaxial} = 5 \times 10^7$$

$$\therefore n_f = \frac{N_{design,multiaxial}}{N_{expected,multiaxial}} = 200$$

Because the design standard is $n_f > 100$, our design satisfies the design criteria.

The ground impulse force again changes this stress. Following the same steps mentioned above, we can get:

$$N'_{design,multiaxial} = 3 \times 10^7$$

$$n_f = \frac{N'_{design,multiaxial}}{N'_{expected,multiaxial}} = 120$$

Therefore, our design still satisfies the design criteria.

Appendix G

Total cost comes out to $6672.77 for our final product. This consists of material cost for each component with an estimate cost of the machining, welding, and other processes needed to form the platform. We believe this to be a major improvement over our preliminary designs, which were up to ten times heavier and used significantly more material.

Two major components of our design, the supporting I-beams [15] and platform grids[11], will be purchased from outside suppliers. No material removal will be needed for these parts, as they are already fit to the dimensions of our platform. The only process needed would be to join them together and to the frame. The purchased grids will be bolted together and onto the platform frame with 1.25-inch diameter bolts[15]. These same bolts will also be used to attach the reinforcement plates [17] of the I-beams to the mounting plates. The I-beams themselves will be welded to the platform along their 5-foot topside.

The other major component, the steel plate[18], is also available to be purchased in the specified size. This will require machining to cut out the rectangular slots that the grids will be placed into. Some of the material removed from the solid plate will be repurposed via welding to form the 3x3 inch rods perpendicular to the platform, which will be welded to the platform at their base.

Table D lists each component of the platform and adds an estimated processing cost for the assembly process. Again, this includes machining, welding, bolting, and painting [19] (to avoid corrosion).

Table D: Estimated Cost

Component	Quantity	Total Price
A36 Steel Plate (10' x 5' x 1.5")	1	$ 3,628.50
I-Beams (6" x 9") 5' length	2	$ 114.58
Steel Grids (10" width)	7	$ 653.87
Steel Grids (12" width)	2	$ 213.84
Bolts	164	$ 983.92
Reinforcement Plates	4	$ 11.76
Anti-Corrosive Paint	1	$ 66.30
Estimated Machining Processes		$ 1,000.00
Total		**$ 6,672.77**

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[16] 1-*1/2"-6x3" A325 Type 1 Heavy Hex Structural Bolt,* AFT Fasteners and Industrial Supply

[17] S*uper-Strut Steel Channel Angle Plate, Electro Galvanized Finish - V325EG*, Sustainable Supply

[18] *A36 Steel Plate 1.5-inch thick,* Midwest Steel and Aluminum

[19] *Rust-Oleum 1 Gallon Red Primer*, MSC Industrial Supply, Inc