ME 3330-02

DESIGN OF MACHINE ELEMENTS

DESIGN PROJECT: TRUCK BED RAMPS

BY:

IN PARTIAL FULFILLMENT OF REQUIREMENTS

FOR THE COURSE

UNDER THE GUIDANCE OF DR. JESSICA P.M. FICK

UNIVERSITY OF WISCONSIN
PLATTEVILLE

INTRODUCTION:

The overall goal of this project was to work as a team to design and create a truck bed ramp system used in the loading of heavy equipment into and out of the truck bed. The initial step was to formulate requirements that the target customer would value the most. A key feature was a ramp system which could be deployed from the truck bed and then be stored in the bed or truck after use. The product should also be able to be used with relative ease and convenience. To assist with installation and removal of the ramp system, the system needed to be lightweight and require minimal installation. Another key criterion was ensuring the ramps were wide enough for various applications and provided maximum safety for the consumer. The overall maintenance and cleaning of the system should be simple and require minimal time. It also, under no circumstances, require removal of the tailgate. Lastly, the system should be universal for various bed sizes. The possibility for a variety of models, to fit all bed sizes was considered. The overall use of the ramp system would accommodate motorcycles, four wheelers, riding lawn mowers, and other vehicles of these nature. With these customer requirements in mind, the next stage was to research various aspects that are important in the design process.

BACKGROUND:

The entire class was split into four “expert groups” comprised of a single member from each of the different project groups. One expert group was tasked with researching information pertaining to safety regulations that may be imposed, finding standard dimensions of truck beds, and equipment which may be loaded in the beds. Three different bed sizes were found with the lengths being 5 feet 8 inches, 6 feet 4 inches, and 8 feet. The ramp was designed to fit into an 8-foot bed; however, its length could be scaled to fit all 3 bed sizes. For accuracy, the bed of a 2017 Chevy Silverado Crew Cab was measured. Measured dimensions included bed height from ground level, distance between wheel-wells, and tailgate height/length. These dimensions were then used in the final design process. The payload capacity of various trucks was found. The maximum payload ranged anywhere from 2,000 pounds to 3,000 pounds. It was noted this capacity would include both the ramp system and the equipment. These ratings were for trucks which could handle the least amount of weight out of all the different truck classes. The weight of various ATVs was found, with the average being 300 to 700 pounds. The standards for the lowest rated capacities were considered under the assumption that if these trucks could handle the load then the same could be said about any truck model. Various safety laws and requirements were found but were focused more on commercial applications. Of these laws and regulations, one looked at how a load must be secured. Another looked at OSHA standards on ramp requirements and grade. The design chosen was not suited for commercial applications, so these laws were not considered.

Another expert group was assigned ergonomics. The main objective within this group was to find information that would aid in the design considerations pertaining to customer comfort and ease-of-use. After conceptualizing the final design, the main concerns regarding ergonomics was the overall weight of the assembly, the individual weight of the ramps, and the height of the platform. Out of all the expert group's findings, one research paper provided ample information regarding the ergonomics of the design. How Much Weight Is Too Much for Manual Lifting: Determining a Weight Limit Guideline for Team-Effort Lifting Tasks; by Piyush G. Chapla, aims to find a base lifting capacity for workers on an individual level and team level [3]. In the paper, Chapla details many existing guidelines and research regarding the same topic. Customer requirements deemed it necessary to be able to remove the frame and ramp assembly at the customers convenience. In the group's findings, it was revealed that OSHA standards states 55 pounds lifted more than once should be the limit for any individual. However, the larger the object lifted, the weight to cause a critical injury decreases as size increases [3]. For the ramps, the design would require the customer be able to pull out, lower, raise, and push in the ramps from the truck bed with relative ease. Chapla found a lift height of 53 inches is recommended to be the maximum any person should attempt while remaining relatively safe. OSHA states that a lift of more than 25 pounds below the knee is not desirable [3]. From this information, it can be concluded that no matter the assembly weight, due to the size of the product, it is recommended the customer to install and uninstall the assembly using team lifts only. If the entire assembly should weigh over 110 pounds, a three-person team lift is recommended to avoid serious injury. It is recommended that if possible, the ramp weight should be within ±10 lbs. of 25lbs for safe, injury free operation. If these recommendations are followed the final product should be relatively easy and convenient for the customer to use.

An expert group was formed to research and find existing truck ramps in the current market. Most of the existing ramps on the market are very similar, if not close to the final design of the project. Many of the truck ramps have a load capacity of around 1500 lbs. This is a very conservative load capacity considering the vehicles that are loaded into the bed of trucks. The ramps on the market are usually made of steel and/or aluminum. These materials have a good balance between weight, strength, and cost. In the design, a decision was made to pick aluminum for its low weight and high strength. The price for most ramps on the market range from $50 to almost $2000. The simply designed ramps are normally priced lower than the $2000 high priced intricately designed ramps. From initial impressions the ramp design would push the price point up to the $2000 and up range. A lot of the ramps that currently exist are foldable, which is a good design for compact situations. Length of the ramps ranged from 60 inches to 80 inches. Overall, when looking at the currently existing ramps on the market, the average price point is around $200. Most ramps are made of a combination of aluminum and steel and are on average 6 feet long.

Patents was the final expert group's research topic. The goal of this project is to make a marketable product that would be competitive in the current market. This would not be achievable if the product violated a patent. Many patents for truck ramps exist and as many as possible were reviewed. While some patents found were for products that had a similar design, none of the patents were protecting any specific design that was used in this group. While no patents were found that the design violates, it is hard to be certain that there are no patents in existence for this design. Before this product could be marketed, the design would have to be taken to a professional to ensure that no patents are being violated. After that, this product could get its own patent to protect it from being copied.

PROPOSED CONCEPT:

After consideration of the customer requirements along with information gathered through expert groups, three designs were proposed. The first design consisted of two single ramps which would be stored on either side of the truck bed in a permanent enclosure (Figure 1). This design however failed to meet several of the customer requirements due to safety and the inability to be removed. Major factors in not choosing this design was due to already prevalent similar products which are patented, and the overall design complexity necessary to achieve a working system. The second design was a more complex design that allowed the entire truck bed to tilt up and down, allowing for an easier and safer loading of the equipment (Figure 2). This concept met several requirements; however, the cost and installation of this product would be high. This also would have required an alteration of the trucks initial design. The final concept was ultimately chosen to be a platform, in which two ramps slide out and are stored underneath the raised platform (Figure 3). This design was simple, easy to use, removable, and met all the customer requirements. This design allowed for easy installation and removal, making it either a permanent fixing or temporary use. The imprint of the system did take away from the usable bed space, however it is minimal and would not affect most every day occurrences if not removed. The ramp system is a wide and stable platform which minimizes safety concerns while loading equipment. The ramps simply pull out from underneath the platform and remain securely attached.

CALCULATIONS:

An analysis was conducted for the ramp system based on bending experienced while loading and unloading the equipment, as well as the fatigue it would experience over time (Figure 7). Aluminum T6061-T6 was the selected material to be used. The loading used was 1000 pounds which is above the standard weight of an ATV. With this is mind, each ramp would be supporting 500 pounds. These rough estimates were deemed conservative enough to be used in the calculations. The cross-section of the ramp used incorporated the two edge beams only as this would be the most critical section. The weight would be distributed between these two beams via the horizontal rungs between them. The bending of the ramp with respect to the yield strength resulted in a safety factor of 4. This safety factor is generally high but was reassuring given a conservative load estimate. Because of this, cases of loading that exceed the 500-pound load would still not cause failure. Next, the fatigue of the ramp was looked at. The grade of aluminum selected is lightweight yet still offers a desirable yield strength and ultimate tensile strength. After calculating the fatigue of the ramp, the safety factor was found to be 2.1. This safety factor, being greater than 1, shows the ramp is designed for infinite life due to fatigue. However, since the endurance limit does not apply to aluminum, it is concluded that the ramp cycle life is adequately high.

In order to see if the design followed the recommendations of the ergonomic expert group, and to aid in the hinge failure calculations, finding an estimate of the weight of the assembly and its individual parts was deemed necessary. The calculations for the weight can be seen in Figure 4(A). Based off the final design the total volume of Aluminum 6061-T6 was found for the frame and the ramps individually. When using the English Engineering Unit system, because of the conversion factor, when looking at weight it can be simplified that 1 lbm. is equal to 1 lbf. Thus $W = \rho V$. The density of aluminum was found and multiplied by volume to obtain the weight. [2] The frame is approximated to be 62.94 pounds and the individual ramps 33.23 pounds; thus, the approximate weight of the complete assembly is 107 pounds.

To allow for one complete ramp stretching from the deck to the ground, there is a hinge and bolt connecting each ramp to a section accounting for the truck's tailgate. Assuming the max load upon each ramp is 500lbs added to the approximate weight of the ramp, a bolt shear calculation was done (Figure 5). When designing the hinge, a ½ inch grade 5 steel bolt was used. The hinge had 4 shear areas that act on each bolt. First the initial tensile force was calculated using 0.9 as the coefficient for the static loading. Then it was found the final force that the bolt could withstand is 41,000 pounds. It is concluded that the bolt diameter is overdesigned for the type of loadings that will be applied to the ramps. However, the calculations only looked at static loading, so that ½ inch bolt could be an added safety factor for improper loading of the ramps, such as sudden and abrupt loading, or other unsafe loading methods.

Because the ramps have a height of 2 inches, and the opening for them between the frame support beams is 2.5 inches, when the frame is bearing the load from cargo, the deflection must be less than 0.5 inches. This would ensure the ramps to be slid back to their original position. The deflection calculations (Figure 4B) were done on the support beams spanning across the front, as the other support beams following would experience similar deflection. The load was assumed to be a point load at the center with magnitude 500 pounds. From some quick approximation, this is half of the assumed max load of 1000 pounds. The load is halved due to the left and right side of the frame sharing the load equally. Since there are multiple supports spanning across the entire length of the frame, and the load would most likely be experienced as a distributed load across the area contacted by each of the object's tires, 500 pounds is likely more than the support beam would experience. Using Castigliano's energy method, the maximum deflection due to bending and shear on the beam would not be expected to exceed 0.004 inches. From the findings, deflection can be ruled out as a concern, as the ramps will have ample space for normal operation.

Once loaded, the frame will have to support the weight of the machine while the truck undergoes normal operation. As seen in Figure 6, fatigue calculations were done to the support bars at the top of the frame and buckling calculations were done to the bar that holds up the top of the frame. When looking at the support bars, 500 pounds was used in the center of the bar. This is the worst case-scenario since moving the weight in any direction would either distribute the load to other bars or decrease the moment that the load causes. With fatigue calculations, the safety factor comes out to be 12.74. This number is quite high, and the bar could likely be made smaller. However, it is beneficial to have an increased safety factor since every bump that the truck goes over is essentially a loading cycle. When looking at the support beam that holds up the top of the frame, buckling calculations were performed (Figure 6). With a safety factor of 165,400, this piece will likely never fail. This piece could be made smaller, but from a manufacturing standpoint it is convenient to keep stock uniform to what is used for the top of the frame. This calculation was added initially to ensure the frame could support itself under load. After completing the calculation, the location was not of concern, but the calculation was kept incorporating course curriculum.

RESULTS AND DISCUSSION:

The final design was found to meet the customer requirements set forth. Its highly sturdy structure would support a variety of different equipment, while sustaining nearly infinite life at a high endurance strength (Figure 8). The consumer market for this product would be a nominal part of an already niche market. However, for those consumers that wish to acquire a safer and easier to use product and are willing to spend additional money, the design is sure to be appealing. The average consumer in this market often are looking for a cheap and quick solutions and find satisfaction with the basic two ramp systems. For the price the design is forecasted to cost, many customers would simply purchase a trailer for loading applications. The proposed product however is a space saving alternative and could even be used in conjunction with a trailer. Marketing for this product could be achieved through smaller scale situations where the product would be most useful. These advertisements could be demonstrated through media such as magazines targeted towards the utility vehicle market, YouTube and other video streaming media, and a simple product website. This would help to boost sales and allow for targeted consumers to know about the product. The product could be profitable, but a high profit margin would be difficult to achieve. Sales are predicted to be small compared to cheaper alternatives like the standard two ramp system. Since this product is targeted for a small market, it is recommended the design be revamped. In redesign, improvements could be made in overall weight, size, and material reduction. From calculations it is obvious some elements are overdesigned. Reducing these elements could make the product more competitive. This product should have the ability to be patented. The design is unique, and when compared to similar products, it does not overtly copy them. However, the design would need to be professionally analyzed. Overall, consumers using the standard two ramp system have experienced situations where the ramps have become unsecured, unstable, and unsafe. This product would eliminate risk, and uncertainty. The proposed design is sturdy, heavy-duty, and safely allows consumers to load and unload recreational vehicles and more. Therefore, the design fulfills the necessary requirements for this project.

CONCLUSION:

This design project allowed for insight on what is required of professionals to create a functional design while meeting specific criteria. Overall it has shown the importance of teamwork and communication when connecting each step of the design process into a final product. It also emphasized the importance of the course work, and how it is effective in a professional position. Analysis of critical points on the assembly provided valuable insight on the design. This insight is critical in formulating design changes. If used correctly, the computational methods demonstrated in this course streamlines design for maximum safety, convenience, and economy. Each aspect of the design was looked at to see exactly which calculations needed to be done to ensure there would not be failure. After completing these calculations, the product was ensured to be able to perform its designated task. The product’s customer requirements, which were set forth in the beginning of the process, were included into the design. Overall, the product could use improvements. However, in its current state it would adequately perform to customer standards. Based on current products in the market for this same application, this product would not be a replace all system but rather an additional option for consumers to consider.

APPENDIX:

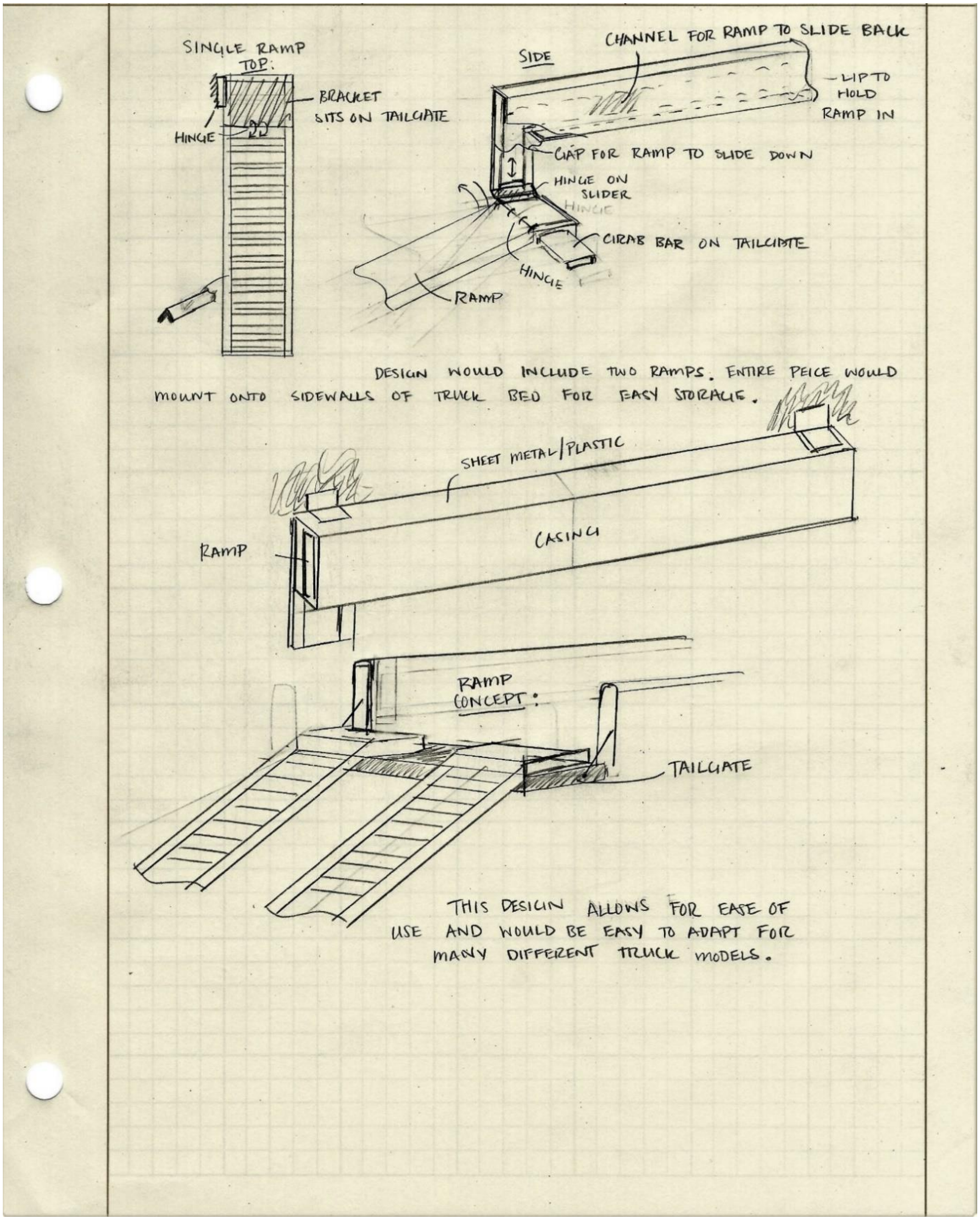


FIGURE 1. SINGLE RAMP CONCEPT

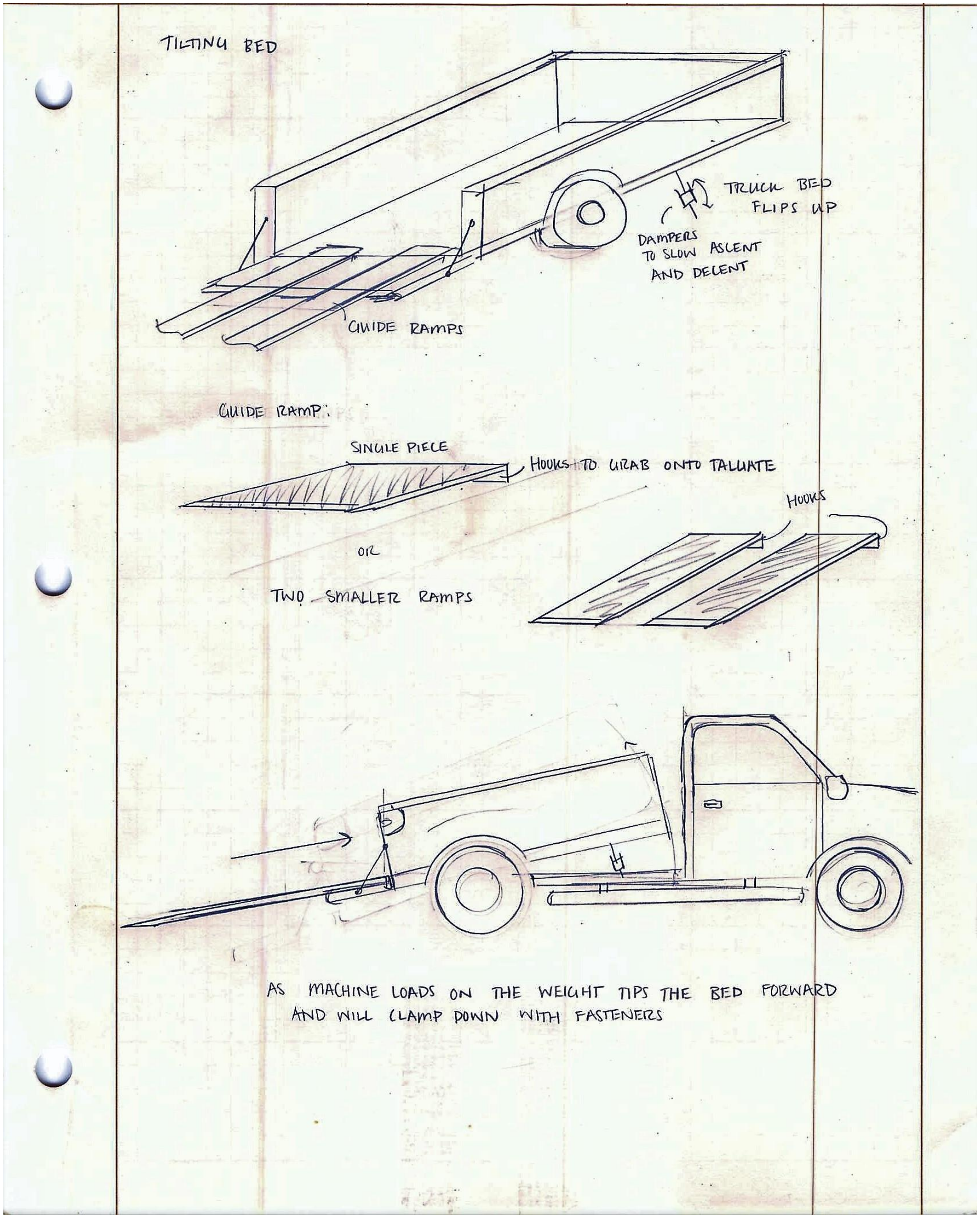


FIGURE 2. TILTING BED CONCEPT

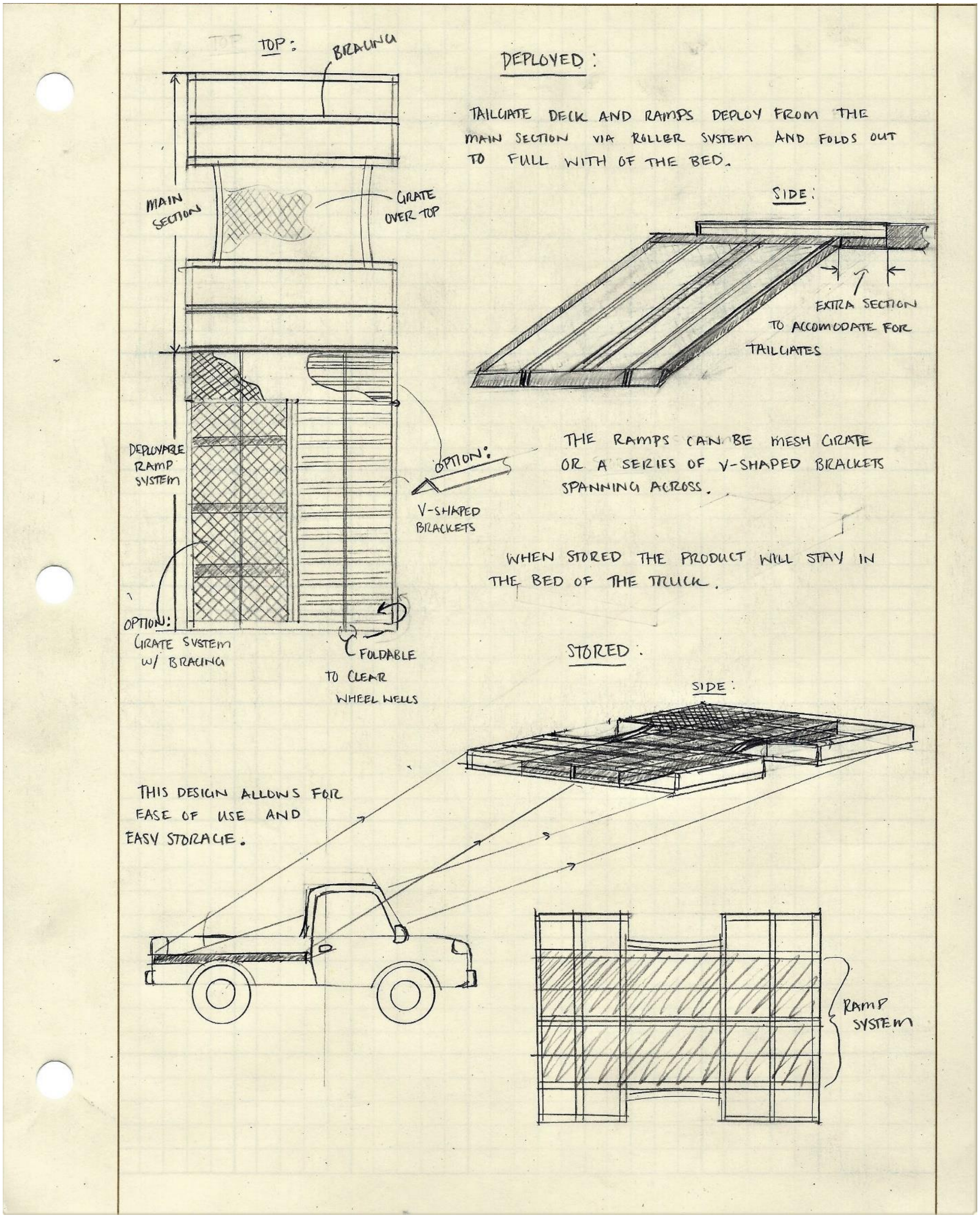
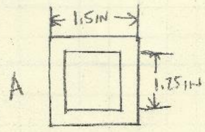


FIGURE 3. CHOSEN CONCEPT

FOR FRAME, EVERY BAR HAS A CROSS SECTIONAL AREA



$$A = 1.5IN \times 1.5IN - 1.25IN \times 1.25IN = 0.6875\ IN^2$$

FRAME CONSISTS OF 6, 96 IN BARS
14, 23.25 IN BEAMS
15, 2.5 IN COLUMNS

$$V = 6(96IN(0.6875IN^2)) + 14(23.25IN(0.6875IN^2)) + 15(2.5IN(0.6875IN^2))$$

$$V = 645.5625\ IN^3$$

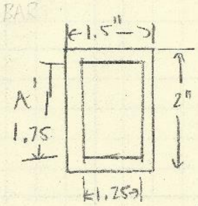
DENSITY OF ALUMINUM 6061-T6 IS KNOWN TO BE $0.0975\ LB/IN^3$

$$W = \rho V \qquad W = 0.0975LB/IN^3\ (645.5625\ IN^3)$$

$$\hookrightarrow \underline{W = 62.94\ LBS}$$

THE ESTIMATED FINAL WEIGHT OF THE FRAME IS 62.94LBS

EACH RAMP HAS TWO SEPARATE SECTIONS AND COMPRISES OF BARS WITH CROSS-SECTION A AND A'



$$A = 0.6875IN^2 \qquad A' = 0.8125IN^2$$

THE RAMP SECTION COMPRISES OF
2, 69.5 IN BARS W/ CROSS-SECTION A'
17, 19.5 IN BARS W/ CROSS-SECTION A

$$V_{RAMP} = 2(69.5IN(0.8125IN^2)) + 17(19.5IN(0.6875IN^2))$$

$$\hookrightarrow V_{RAMP} = 340.84375\ IN^3$$

$$W_{RAMP} = \rho V_{RAMP} = 340.84375LBS\ (0.0975LB/IN^3)$$

$$\hookrightarrow \underline{W_{RAMP} = 33.232\ LBS}$$

THE TAILGATE SECTION COMPRISES OF
2, 24 IN BARS W/ CROSS-SECTION A'
5, 19.5 IN BARS W/ CROSS-SECTION A

$$V_{TAIL} = 2(24IN(0.8125IN^2)) + (5(19.5IN(0.6875IN^2))$$

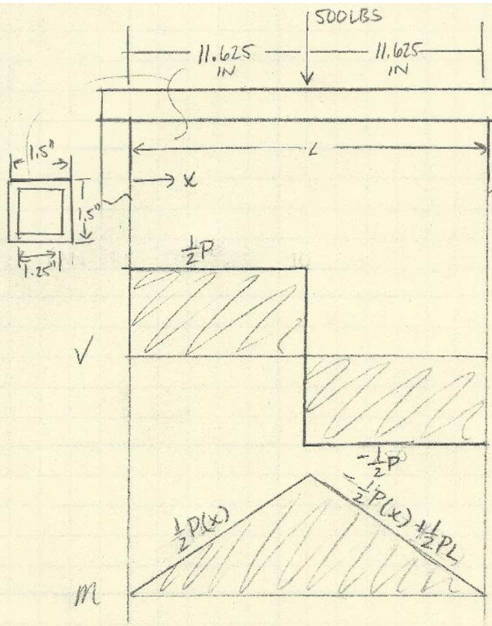
$$\hookrightarrow V_{TAIL} = 107.656\ IN^3$$

$$W_{TAIL} = \rho V_{TAIL} = 0.0975\ LB/IN^3\ (107.656IN^3)$$

$$\hookrightarrow \underline{W_{TAIL} = 10.5\ LBS} \qquad W_{TOTAL} = W_{FRAME} + 2(W_{TAIL} + W_{RAMP}) = \underline{106.672\ LBS}$$

THE ESTIMATED TOTAL WEIGHT IS AROUND 107 LBS

FIGURE 4 (A). WEIGHT CALCULATION



REACTIONS ARE 250LBS EACH OR IF THE LOAD 500LBS IS REFERED TO AS P; $\frac{1}{2}P$.

USING CASTIGLIANOS METHOD

$$U = U_{SHEAR} + U_{BEND}$$

$$U = \int_0^{L/2} \frac{3V^2}{5GA}dx + \int_{L/2}^{L} \frac{3V^2}{5GA}dx + \int_0^{L/2} \frac{m^2}{2EI}dx + \int_{L/2}^{L} \frac{m^2}{2EI}dx$$

$$\int_0^{L/2} \frac{3V^2}{5GA}dx = \frac{3}{5GA}\int_0^{L/2} \frac{P^2}{4}dx = \frac{3P^2(x)}{20GA}\Big|_0^{L/2} \rightarrow = \frac{3P^2L}{40GA}$$

$$\int_{L/2}^{L} \frac{3V^2}{5GA}dx = \frac{3}{5GA}\int_{L/2}^{L} \frac{P^2}{4}dx = \frac{3P^2(x)}{20GA}\Big|_{L/2}^{L}$$

$$\rightarrow = \frac{3P^2(L)}{20GA} - \frac{3P^2(L)}{40GA}$$

$$U_{SHEAR} = \frac{3P^2L}{40GA} + \frac{3P^2L}{20GA} - \frac{3P^2L}{40GA} = \frac{3P^2L}{20GA}$$

$$\int_0^{L/2} \frac{m^2}{2EI}dx = \frac{1}{2EI}\int_0^{L/2} \frac{1}{4}P^2x^2dx = \frac{1}{2EI}\left(\frac{1}{12}P^2x^3\right)\Big|_0^{L/2} = \frac{P^2L^3}{192EI}$$

$$\int_{L/2}^{L} \frac{m^2}{2EI}dx = \frac{1}{2EI}\int_{L/2}^{L} \frac{1}{4}P^2x^2 - \frac{P^2xL}{2} + \frac{P^2L^2}{4}dx = \frac{1}{2EI}\left(\frac{1}{12}P^2x^3 - \frac{1}{4}P^2x^2L + \frac{P^2xL^2}{4}\right)\Big|_{L/2}^{L}$$

$$= \frac{P^2L^3}{24EI} - \frac{P^2L^3}{8EI} + \frac{P^2L^3}{8EI} - \frac{P^2L^3}{192EI} + \frac{P^2L^3}{32EI} - \frac{P^2L^3}{16EI} = \frac{-11P^2L^3}{192EI} + \frac{P^2L^3}{16EI} = \frac{P^2L^3}{192EI}$$

$$U_{BEND} = \frac{P^2L^3}{192EI} + \frac{P^2L^3}{192EI} = \frac{P^2L^2}{96EI}$$

MAX DEFLECTION IS THE DERIVATIVE OF ENERGY WRT LOAD

$$U = U_{SHEAR} + U_{BEND} = \frac{3P^2L}{20GA} + \frac{P^2L^2}{96EI}$$

FIGURE 4 (B1). PLATFORM DEFLECTION CALCULATIONS

MOMENT OF INERTIA:

$$I = \frac{bh^3}{12} - \frac{b'h'^3}{12} = \frac{1.5(1.5)^3}{12} - \frac{(1.25)(1.25)^3}{12}$$

$$= \frac{1.5^4}{12} - \frac{1.25^4}{12} = 0.2184\ IN^4$$

$$\delta = \frac{\delta U}{\delta P} = \frac{3PL}{10GA} + \frac{PL^2}{48EI}$$

E IS KNOWN TO BE 10,000 KSI
$E = 10{,}000[10]^3$ PSI
G IS KNOWN TO BE 3770 KSI
$G = 3770[10]^3$ PSI

CROSS SECTIONAL AREA, A IS $A = 0.6875\ IN^2$

$$\delta = \frac{3(500LBS)(23.25IN)}{10(3770[10^3]PSI)(0.6875IN^2)} + \frac{(500LBS)(23.25IN)^2}{48(10{,}000[10^3]PSI)(0.2184IN^4)}$$

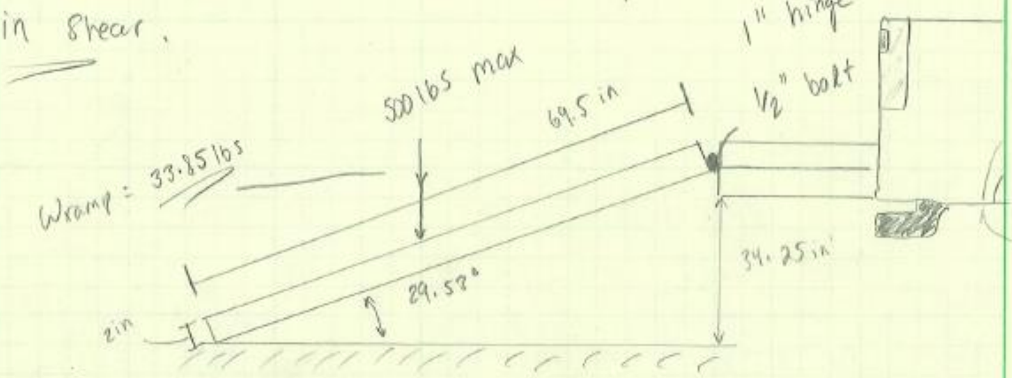
$$\delta = (0.00135 + 0.002578)IN$$

$$\boxed{\delta_{max} = 0.003924\ INCHES}$$

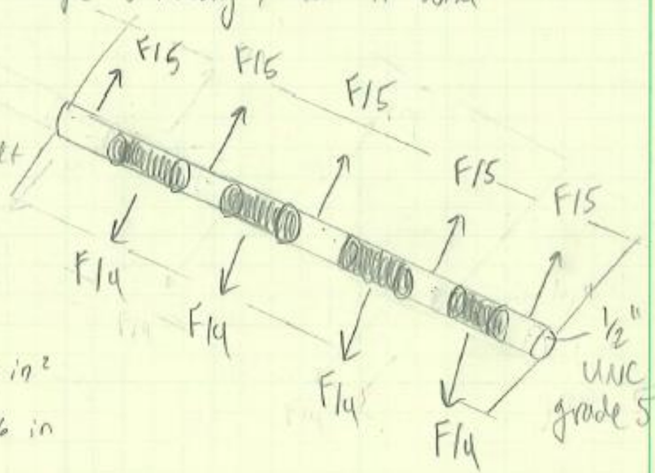
THE MAX DEFLECTION EACH SUPPORT BEAM OF THE FRAME IS EXPECTED TO NOT EXCEED 0.004 INCHES.

FIGURE 4 (B2). PLATFORM DEFLECTION CALCULATIONS

- Assume that the max load per ramp applied to the bolt is 500lbs + Wramp in Shear.



- Use 1/2" bolt diameter (grade 5 steel)
- Assume No friction in hinge assembly, and it will apply to the bolt



☆ Find the max force bolt will take

TABLE 10-1

(At) Tensile Area stress: .1419 in^2

(d1) Minor diameter: .4056 in

TABLE 10-4

(Sp) Proof load: 85 ksi

(Sy) Yield Strength: 92 ksi

FIGURE 5 (A1). BOLT FORCES CALCULATION

- F_i is initial tensile force
- Usually $F_i \simeq (.9)(A_t)(S_p)$, involving static loading. We will use this for ease of calculations

☆ $F_i = (.9)(A_t)(S_p) \Rightarrow F_i = (.9)(.1419)(85.0)$

$$= 10.855 \text{ kip}$$

- We will be concerned about the bolt in shear if it exceeds our load combined with ramp weight

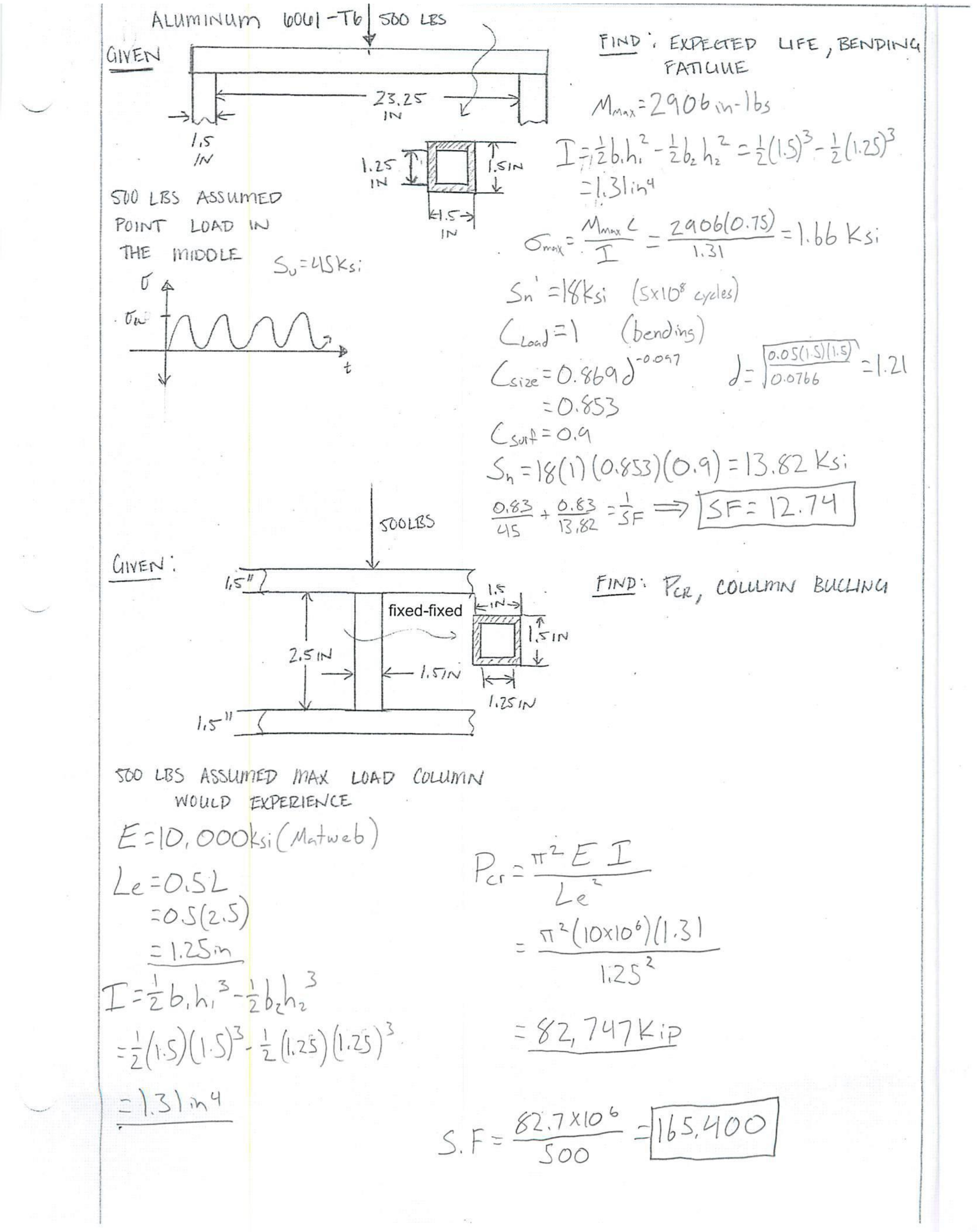
$$\tau_{bolt} = \frac{F}{4A} \geq S_{ys}$$

$$F = 4\left(\frac{\pi}{4}\left(1/2\right)^2\right)\left(.58\left(92 \times 10^3\right)\right)$$

$$= 41908.846 \text{ lbs}$$

- When looking at this maximum load (force) that we can apply, we should consider using a smaller diameter or less expensive material to reduce cost.

FIGURE 5 (A2). BOLT FORCES CALCULATION



ALUMINUM 6061-T6 | 500 LBS

GIVEN

23.25 IN

1.5 IN

1.25 IN

1.5 IN

1.5 IN

500 LBS ASSUMED POINT LOAD IN THE MIDDLE

$S_u = 45\,ksi$

σ, σ_m, t

FIND: EXPECTED LIFE, BENDING FATIGUE

$M_{max} = 2906$ in-lbs

$I = \frac{1}{2}b_1h_1^2 - \frac{1}{2}b_2h_2^2 = \frac{1}{2}(1.5)^3 - \frac{1}{2}(1.25)^3$

$= 1.31\,in^4$

$\sigma_{max} = \frac{M_{max}c}{I} = \frac{2906(0.75)}{1.31} = 1.66\,Ksi$

$S_n' = 18\,ksi$ (5×10^8 cycles)

$C_{Load} = 1$ (bending)

$C_{size} = 0.869d^{-0.097}$ $d = \sqrt{\frac{0.05(1.5)(1.5)}{0.0766}} = 1.21$

$= 0.853$

$C_{surf} = 0.9$

$S_n = 18(1)(0.853)(0.9) = 13.82\,Ksi$

$\frac{0.83}{45} + \frac{0.83}{13.82} = \frac{1}{SF} \Longrightarrow$ **SF = 12.74**

500LBS

GIVEN:

1.5"

fixed-fixed

1.5 IN

1.5 IN

2.5 IN

1.5 IN

1.25 IN

1.5"

FIND: P_{CR}, COLUMN BUCLING

500 LBS ASSUMED MAX LOAD COLUMN WOULD EXPERIENCE

$E = 10{,}000\,ksi$ (Matweb)

$L_e = 0.5L$

$= 0.5(2.5)$

$= 1.25\,in$

$I = \frac{1}{2}b_1h_1^3 - \frac{1}{2}b_2h_2^3$

$= \frac{1}{2}(1.5)(1.5)^3 - \frac{1}{2}(1.25)(1.25)^3$

$= 1.31\,in^4$

$P_{cr} = \frac{\pi^2 E I}{L_e^2}$

$= \frac{\pi^2(10\times10^6)(1.31)}{1.25^2}$

$= 82{,}747\,Kip$

$S.F = \frac{82.7\times10^6}{500} =$ **165,400**

FIGURE 6. BENDING AND BUCKLING OF PLATFORM

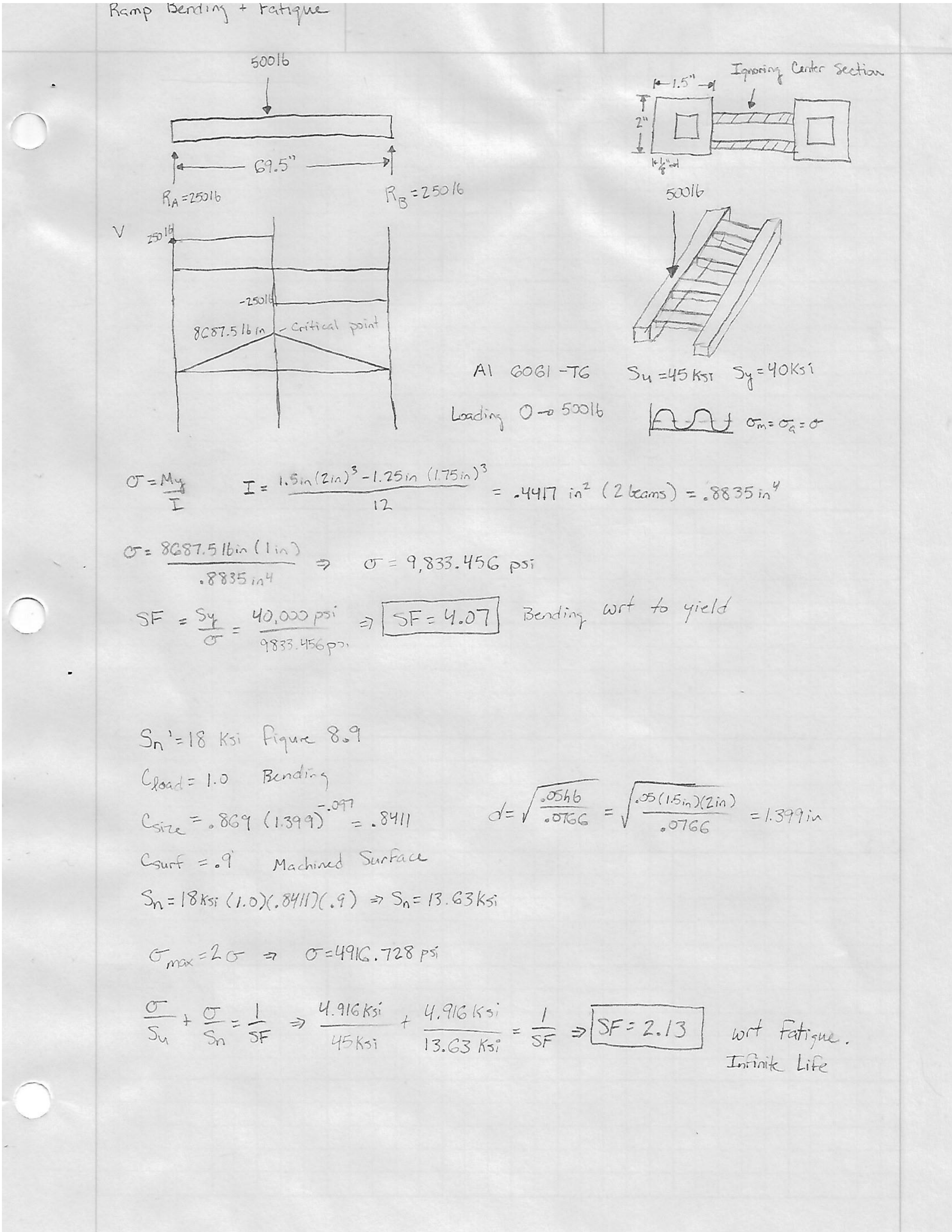


FIGURE 7. BENDING AND FATIGUE OF RAMP

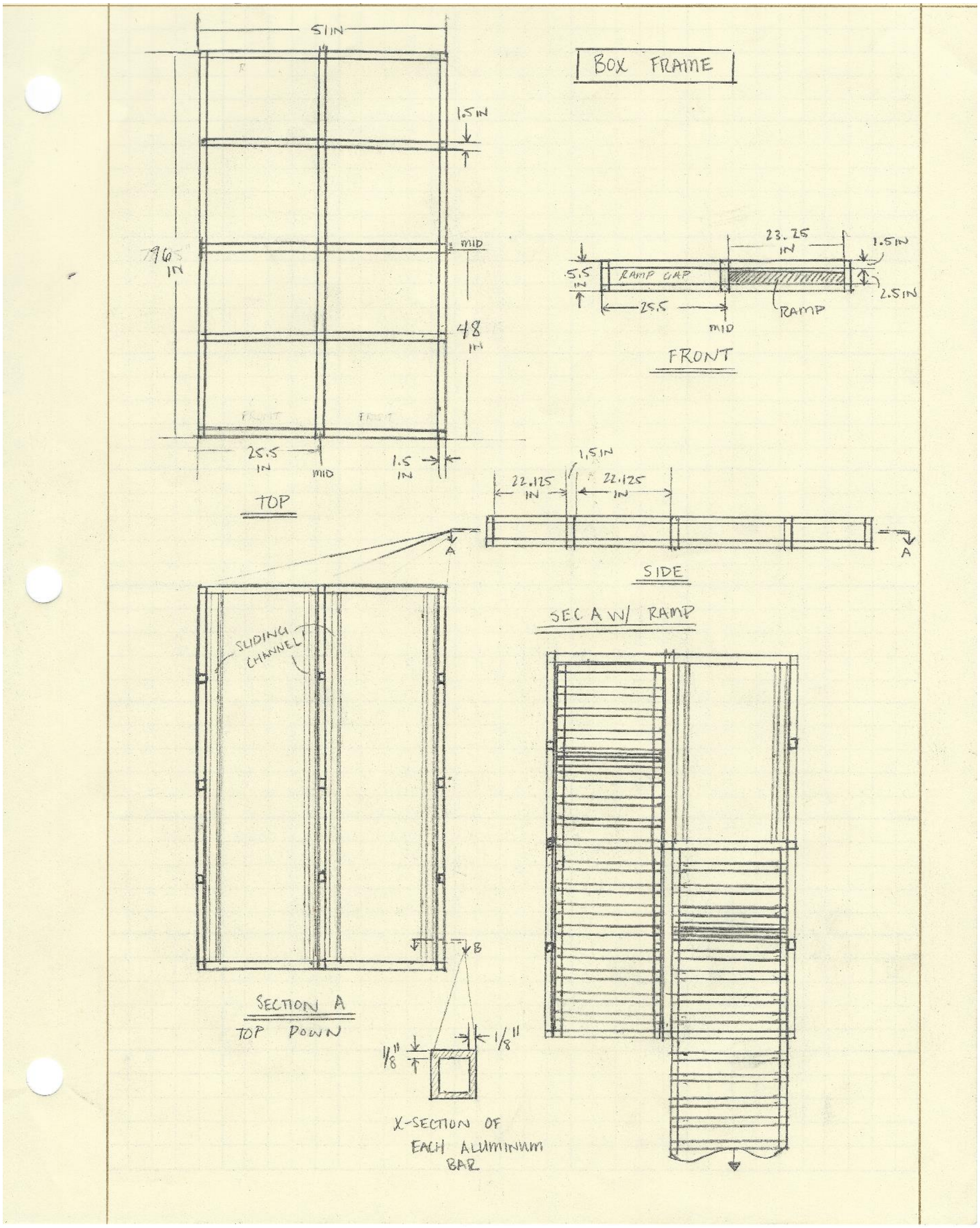


FIGURE 8 (1). FINAL PRODUCT

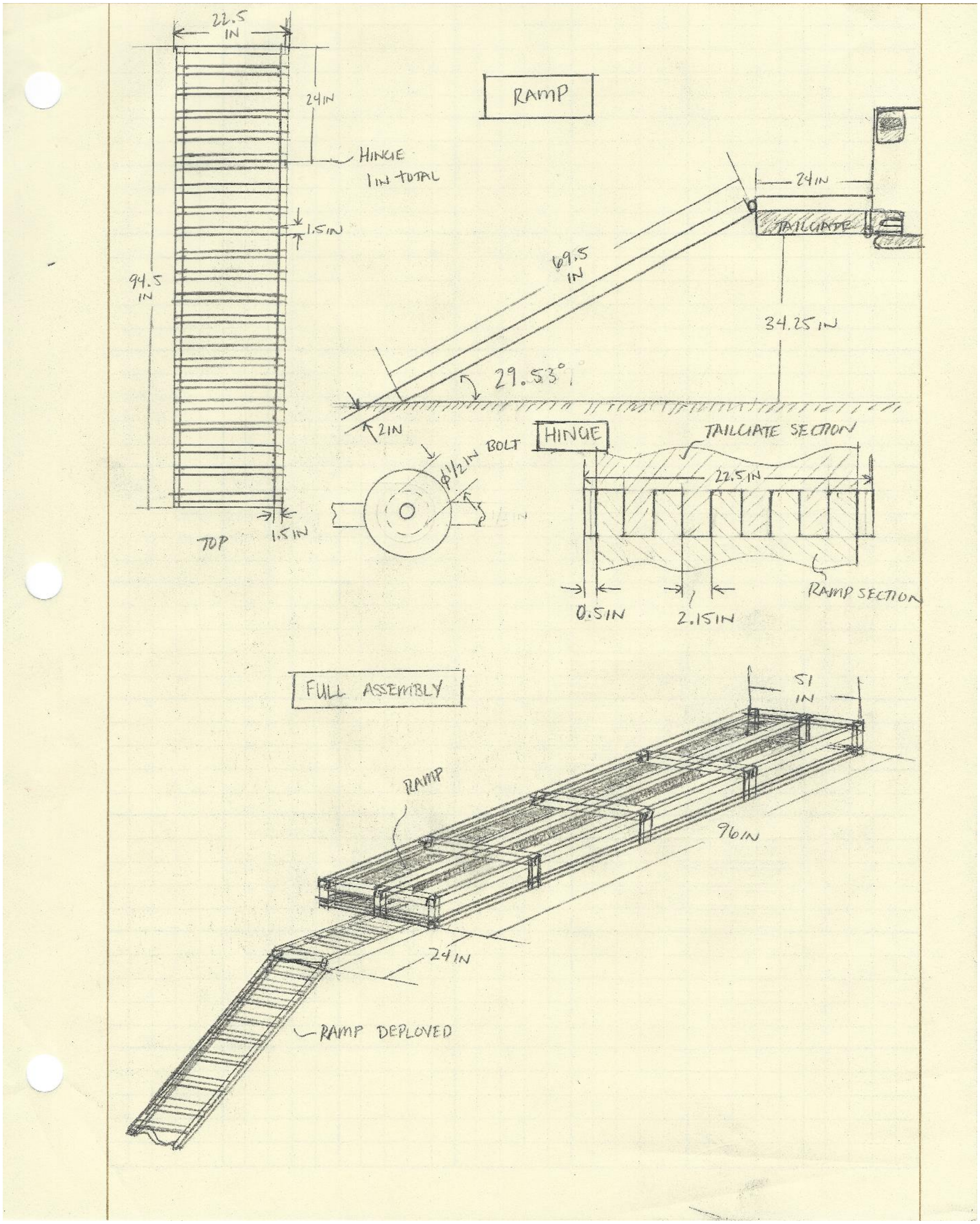


FIGURE 8 (2). FINAL PRODUCT

REFERENCES

[1] Juvinall, C. Robert, and Marshek, M. Kurt, [Fundamentals of Machine Component Design, 4th Ed.], Wiley and Sons, Inc., Hoboken, New Jersey (2000).

[2] "Online Materials Information Source," 25 April 2019, <http://www.matweb.com>

[3] Chapla, Piyush G. 2004. "How Much Weight Is Too Much for Manual Lifting: Determining a Weight Limit Guideline for Team-Effort Lifting Tasks.". Thesis. <http://archives.njit.edu/vol01/etd/2000s/2004/njit-etd2004-111/njit-etd2004-111.pdf.>