Carabiner Design Analysis

ME 442

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Abstract

The goal of this project is to analyze carabiner cross-sectional areas. This data will show which design is preferred for climbing applications. The project contents include design selections, design drawings, mathematical design analysis, design comparisons, and conclusions of the data from mathematical analysis. Methods used include Castigliano's theorem, curved beam theory, and fatigue analysis through the Soderberg and Smith-Watson-Topper criterion, as well as utilizing Basquin's equation to find predicted failure cycles. Results concluded that the circular cross section is superior to the triangle cross section in terms of deflection and fatigue, and that a design that will concentrate the force closer to one of the bars is best in order to lower the moment arm and therefore decrease the normal bending stress acting on the most sensitive area of the part, significantly increasing the factor of safety and reducing the total deflection.

Product Definition

Functionality:

Carabiners are an incredibly useful tool that can be used in numerous applications, from high-stress applications like climbing or rescue operations to lower-stress environments like holding keys to lanyards. For this specific application, the design is catered towards climbing applications. This means the forces are predominantly experienced in the longitudinal direction in a tensile manner via ropes. The calculations will also assume a total force of 10 kN. While the design is relatively mature and not many new designs can be developed for these, it was decided to utilize the lessons learned in the ME442 Machine Design course to analyze and develop a better form factor for these. These carabiners must support high tensile loads with minimal deflection or failure and be easily and readily attached/detached to the loads being linked. Using these requirements, the following constraints were developed.

Constraints:

- **Dimensions:** The carabiner must be compact and lightweight for portability, with an overall size suitable for attaching to gear loops, straps, or small ropes. It should accommodate items 12 mm in diameter, making it versatile for hammocks, tool lanyards, or key gear attachments.
- **Materials:** A material that balances strength, cost, and corrosion resistance is essential. Aluminum 7075-T6 is selected for its adequate strength, good machinability, and lower cost than high-strength alloys. Stainless steel may also be considered for marine or high-wear environments where corrosion is a greater concern. The properties of these materials were found in the ASM material data sheets [1].
- **Costs:** The carabiner must be affordable for medium-scale production, ideally using die casting, CNC machining, or forging. To remain competitive, cost targets should be kept below $2.00 per unit in bulk production.
- **Environment:** The carabiner should function reliably in various outdoor settings, including exposure to rain, UV radiation, dirt, and temperature variations. It must maintain mechanical integrity during repeated use and offer moderate resistance to abrasion and corrosion.

Conceptual Design

Design 1: Circular Carabiner (Original Design)

- Simple curvature profile
- Simpler load distribution
- Traditional manufacturing and well-established safety limits
- **Pros:** Symmetrical, easy to manufacture, predictable behavior under load
- **Cons:** Stress concentration at curvature; heavier than needed

Design 2: Triangular Carabiner (Modified Design)

- Angled sides to reduce overall weight and potentially shift stress paths
- Intended to minimize bending stresses and shift more load into axial components
- Optimized for manufacturing by extrusion or CNC
- **Pros:** Potential weight savings; geometric stiffness
- **Cons:** Corners introduce higher stress concentrations (needs fillet radii), less intuitive stress distribution

Using the design specifications seen above, the drawing for Conceptual Design 1 can be found in Figure 1, and the drawing for Concept 2 can be found in Figure 2. The designs themselves for this stage were all based off of what was commercially available already in order to get a baseline for the design and provide a springboard with which to jumpstart the calculations.

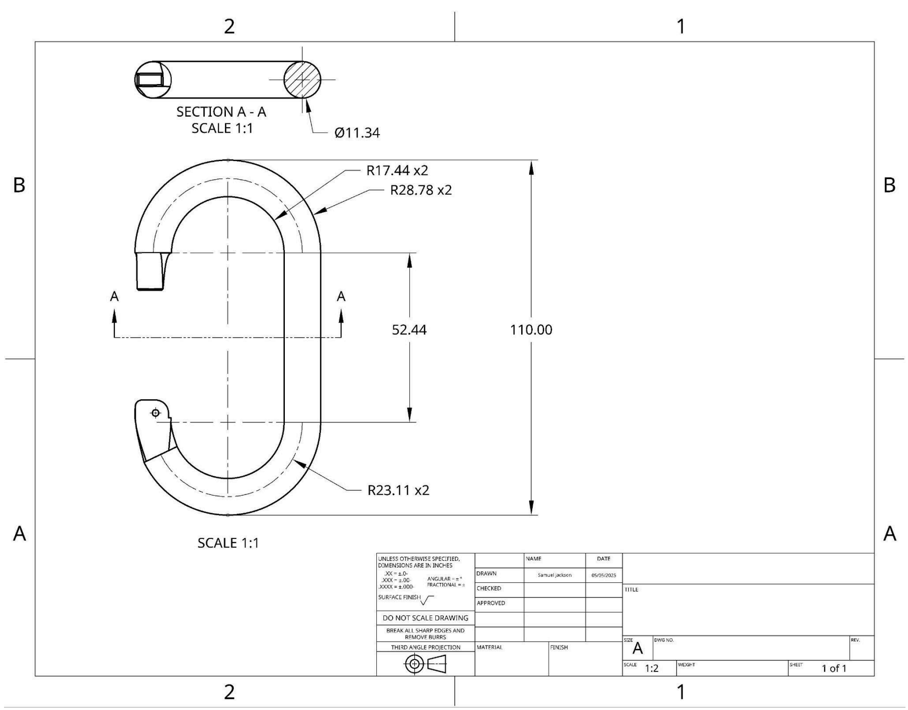

Figure 1. Initial drawings for the carabiner with a circular cross-section

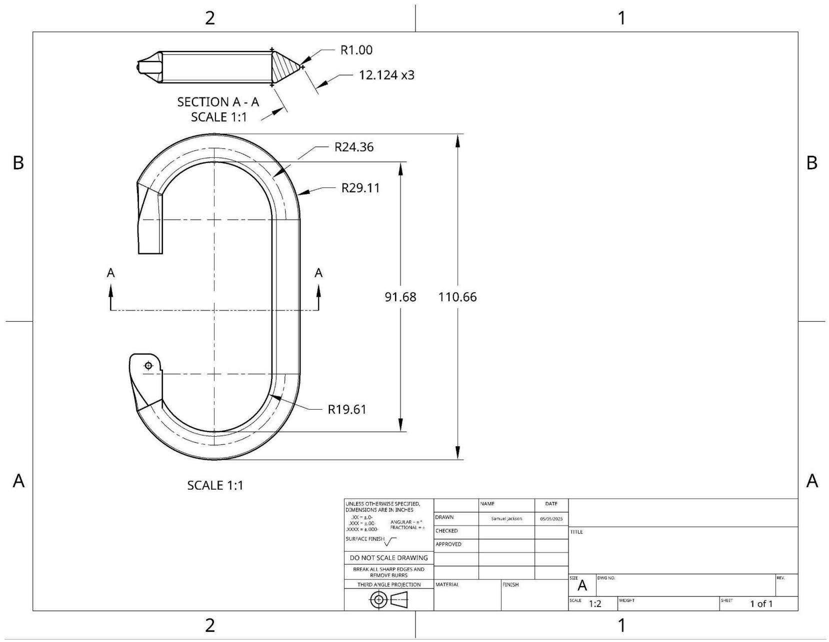

Figure 2. Initial drawing for the carabiner with a triangular cross-section

Decision Basis:

There are four criteria that will be used as a basis for the final decision, and in order to achieve these criteria, the design process will be done to fulfill them specifically. Those four are:

1. Load capacity and stress safety margin (via von Mises + Soderberg / Smith-Watson-Topper criteria)
2. Weight and deflection performance
3. Manufacturability and material usage
4. Safety and fatigue resistance under cyclic loading

In order to determine the validity of the designs and dial in the dimensions, force analysis calculations were performed. According to research, carabiners can lose more than half their strength when their gate is open [4]. As such, the force analysis was done specifically for the open gate condition, as this provides the worst case scenario for the carabiner and its loading conditions. As mentioned, for the following calculations, the force is assumed to be 10kN total maximum for the carabiner.

Mathematical Calculations:

Calculations were performed for both the circular and triangular cases. The deflection δ of the section of interest is calculated as given by solving through a simplified curved beam case [2],

$$\delta = \frac{FR^2}{AEe}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta + \frac{FR^2}{AEe}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta - \frac{2FR}{AE}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta + \frac{CFR}{AG}\int_0^{\frac{\pi}{2}} cos^2\theta d\theta \qquad (1)$$

integrating from 0 to $\frac{\pi}{2}$,

$$\delta = \frac{FR^2}{AEe}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta + \frac{FR^2}{AEe}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta - \frac{2FR}{AE}\int_0^{\frac{\pi}{2}} sin^2\theta d\theta + \frac{CFR}{AG}\int_0^{\frac{\pi}{2}} cos^2\theta d\theta \quad (2)$$

which gives,

$$\delta = \frac{\pi}{4}\frac{FR^2}{AEe} - \frac{\pi}{4}\frac{FR}{AE} + \frac{\pi}{4}\frac{CFR}{AG} \quad (3)$$

which simplifies to,

$$\delta = \frac{\pi}{4}\frac{FR}{A}(\frac{R}{Ee} - \frac{1}{E} + \frac{C}{G}) \quad (4)$$

This deflection δ represents the total deflection of the curved member, where the force *F*, the radius *R*, the material's Young's modulus *E*, the shear modulus *G*, the transverse shear correction factor *C*, and the eccentricity *e* are situational constants. Here, the eccentricity is given by,

$$e = \bar{r} - r_c \quad (5)$$

where r_c is the distance from the central axis of the carabiner to the centroid of the cross-section, and $\bar{r}$ for the circle is given by,

$$\bar{r} = \frac{R^2}{2(r_c - \sqrt{r_c^2 - R^2})} \quad (6)$$

and $\bar{r}$ for the triangle is given by,

$$\bar{r} = \frac{A}{-b + [\frac{br_o}{h}]ln(\frac{r_o}{r_i})} \quad (7)$$

where r_o and r_i represent the outer and inner radii of the carabiner, respectively, and h represents the triangle's height as measured from the base. After the total deflection was found, the ratio of the total deflection divided by the length of the curved beam section to the yield strength divided by the elastic modulus was compared. This is essentially a strain comparison, comparing the experienced normalized deflection to the deflection at yield. For the curved beam section, the length is taken by finding the arc length. The material is expected to remain in the elastic region if the following condition is satisfied,

$$\frac{2\delta}{\pi R} < \frac{S_y}{E} \quad (8)$$

After these calculations were made, fatigue analysis was performed to investigate the performance of the carabiners under cyclic loading conditions. The modified endurance limit S'_e is given by,

$$S'_e = 0.4\, x\, S_u \quad (9)$$

where S_u is the ultimate strength of the material. For this application, 7075 T-6 aluminum was used, so the endurance limit of 40% of the ultimate strength [3]. The corrected endurance limit is found with the Marin factors through,

$$S_e = k_a k_b k_c k_d k_e k_f S_e^{'} \quad (10)$$

The values for these variables are found in the appendix. The point of interest on the curved beam carabiner member is at point C in Figure 3. The total normal stress for this point of interest is found with,

$$\sigma_n = \frac{F}{A} + \frac{Mc}{I} \quad (11)$$

For the fatigue analysis, the stress amplitude σ_a was found with,

$$\sigma_a = \frac{\sigma_{vm}}{2} = \frac{\sqrt{\sigma_n^{\ 2} + 3\tau_{xy}^{\ 2}}}{2} \quad (12)$$

where σ_{vm} represents the Von Mises stresses. For this specific application, the carabiner is mainly going to be experiencing tensile loads, and it is highly unlikely that the total normal force is going to be compressive. For each case, the minimum stress is 0, and the maximum stress is simply the stress amplitude. This also implies that the stress amplitude will be equal to the mean stress, or $\sigma_a = \sigma_m$. Because there are no shear stresses in the cross section at the point of interest, the stress amplitude is simply equal to half of the total normal force at the cross section. Distortion Energy (DE) theory was used for this case because the application uses aluminum, a ductile material. After this, the factor of safety was found with,

$$n = \left(\frac{\sigma_a}{S_e} + \frac{\sigma_a}{S_y}\right)^{-1} \quad (13)$$

Here, the Soderberg criterion is used instead of the Goodman criterion to be more conservative [2]. For all calculations, a Python program was used to calculate the values to hasten the process.

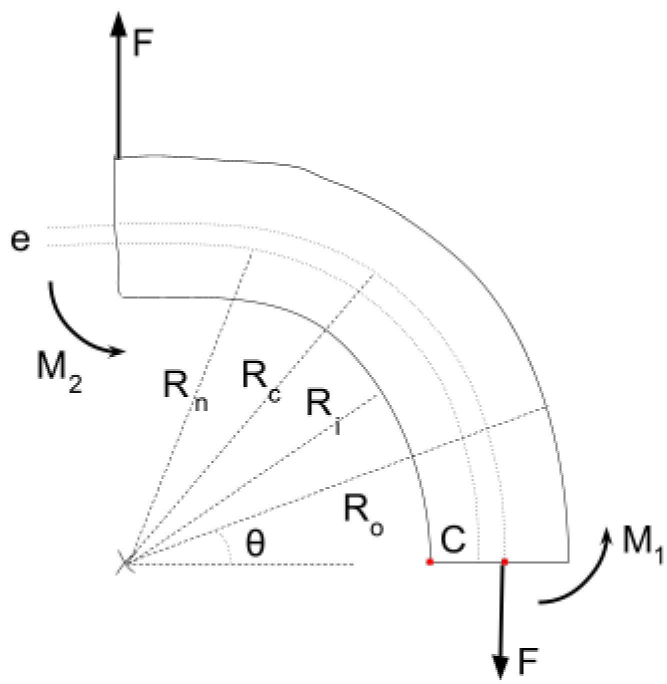

Figure 3. FBD for the curved beam carabiner member

Circular Cross Section:

The circular cross section, with the dimensions provided in Figure 1, exhibited a deflection of approximately 0.573 mm, a deflection to length ratio of 0.0158, and a yield strength to modulus ratio of 0.007. With the deflection to length ratio greater than the yield strength to modulus ratio, this implies that the material will likely yield upon loading. The Soderberg factor of safety was found to be 0.309, which is far below the desired factor of safety of one. While the material may endure many cycles before failure, it is not ideal that the carabiner yields upon loading.

Triangular Cross Section:

The process for the triangle carabiner was the same with the dimensions provided in Figure 2, also yielding similar results, with the main difference being that the eccentricity was slightly higher as the neutral axis lay on the boundary of the lower third of the cross section. The deflection was slightly better at 0.728 mm, and the deflection to length ratio was 0.019, still surpassing the yield strength to modulus ratio of 0.00705. The fatigue factor of safety per the Soderberg criterion was 0.26, about 0.03 higher than the circular cross section. Both of these values were still unacceptable, as the carabiner would fail far earlier than desired.

Preliminary Design

After calculating the deflections for the two designs, it was decided that dimension changes were necessary. Using these new dimensions, open failure modes were analyzed again to determine the new strengths, expected factors of safety, and open gate deflections. The drawings for the updated designs for the circular and triangular cross sections can be found in Figure 3 and Figure 4, respectively.

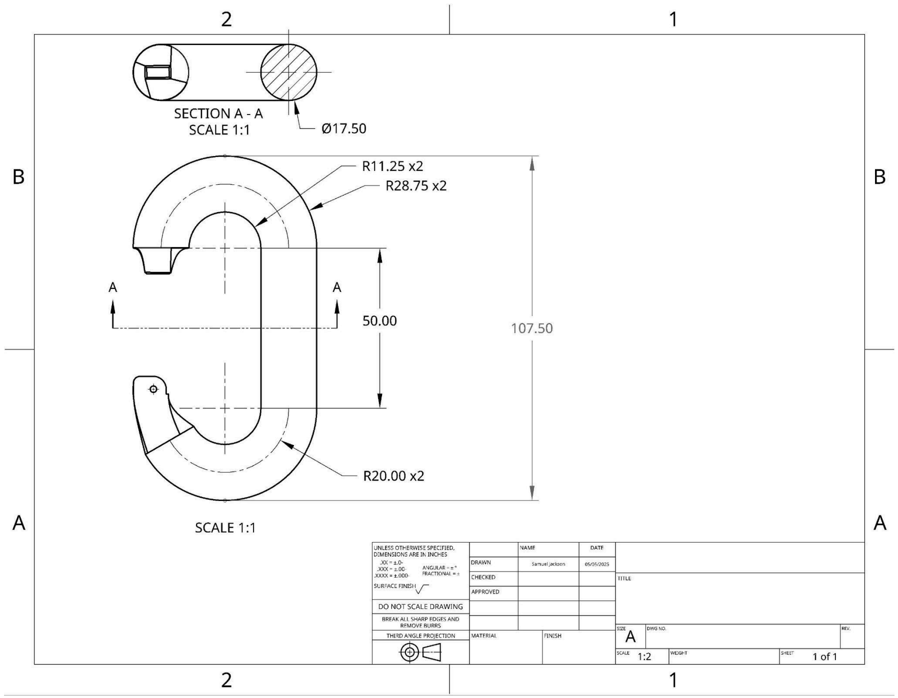

Figure 4. Updated drawing of the carabiner with circular cross-section

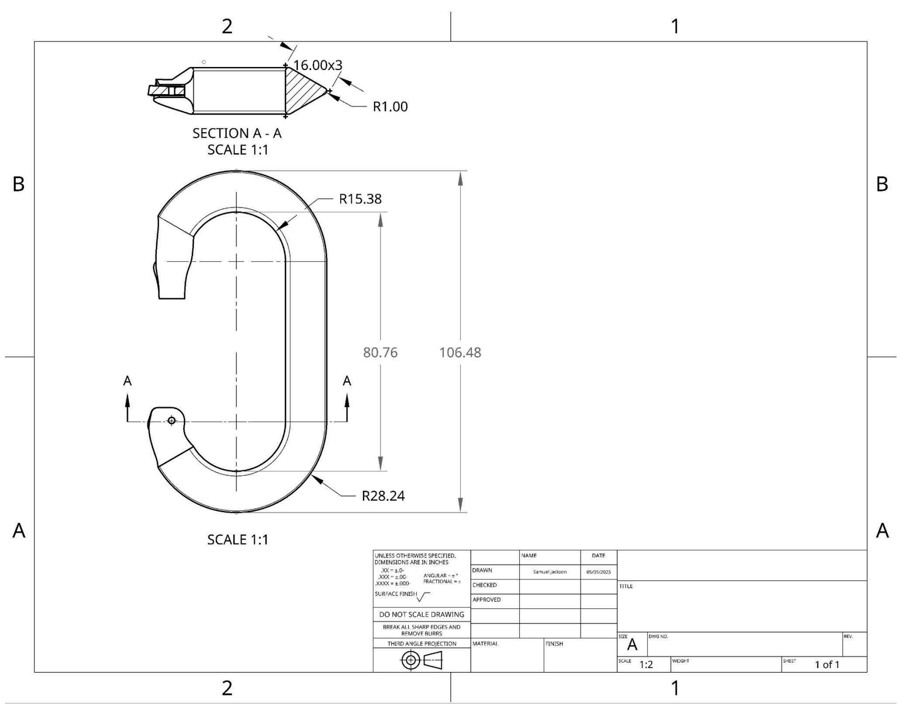

Figure 5. Updated drawing of the carabiner with a triangular cross-section

Circular Cross Section:

Using the same process as defined above in the conceptual design, as well as the values outlined in Figure 4, the deflection for the updated design is 0.37mm. Still, the deflection ratios imply that the material will yield, with the deflection to length ratio being 0.011 and the yield strength to modulus ratio being 0.007. This is a better value than that found in the previous iteration, proving that the design is indeed an improvement. After finding this value, it was decided to move forward with the calculated factor of safety calculations. The factor of safety improved by a small margin, increasing from 0.309 to 0.353, but this is still much lower than the desired factor of safety of close to or over 1.

Triangular Cross Section:

With the same calculation pathway as was used with the dimensions outlined in Figure 5, the deflection was found to be 0.41 mm, worse than the circular cross section. The deflection to length ratio was 0.013, and the yield strength to modulus ratio was 0.007, meaning the cross section would yield at the first load. The fatigue factor of safety worsened with the triangular cross section at 0.31, about .04 lower than the circular cross section. Important to note that the circular cross section for the Figure 4 design was about 122.71 mm^2, and about 110.85 mm^2 for the triangle cross section. This was explored to explain the differences in factors of safety. However, in the process of altering dimensions, it was found that the R_c had a larger effect on the deflection and the fatigue factor of safety.

Other considerations

When calculating stress values for this project, an open-latch situation was the ideal option. In general, circular cross-sections withstand more weight than the triangular cross-sections. While calculating closed-latch applications would account for normal operation, safety is an important factor when it comes to climbing carabiners. Calculating deflection for open-latched carabiners gives the best numbers to determine what will happen in a worst-case scenario. Aspects such as user misoperation, latch abuse/wear, and freak climbing accidents are better accounted for when checking deflection conditions of an open-latched member. This is because the latch offers additional strength when closed, evenly dividing the force concentration between the latch and wall members on a carabiner. This only works because the provided assumptions account for potential disparities between real-life carabiner load distribution and the examples provided in this project.

While the numbers themselves make a good case for the carabiner of circular cross-section, there are other important factors that need to be taken into account. Rope fraying in climbing is a massive issue, as with every fiber that breaks, the strength of the rope can drop considerably. Having a carabiner with a triangular cross-section can make this issue more of a problem. The angles that the triangular cross section can introduce would end up forcing the rope to take equally sharp angles while in the carabiner, strongly decreasing the rope's life expectancy. This decrease in life expectancy means that ropes in use with a triangular cross-sectioned carabiner will have far fewer operation cycles before becoming unsafe, potentially causing injury or death to the climbers using them and the climbers/spectators below them. Adding on to this point, these sharp edges on carabiners of triangular cross-section are generally more dangerous. The sharper edges and points on carabiners of triangular cross section are more likely to cut into clothing and skin on normal use; furthermore, they can also cause worse injuries on impact or user misoperation. Since companies want to minimize lawsuits, there would have to be an amazing counterpoint to the heightened safety concerns.

Another reason circular cross sections are superior in carabiner design is the cheaper production cost. In general, creating a carabiner with a circular cross section is less complicated and costs less than creating the same carabiner with a triangular or other cross section. Carabiners are created from long rods of metal that, one cut to proper size, undergo die-wrapping, stamping, and heat-treating processes to ensure a fair balance between strength and production quantity. These processes are far easier to apply on circular rods because of their relatively even stress and shape distributions within their cross-section. With triangular cross sections, extra machine costs and precautions are needed to maintain shape and minimize defects. On top of this, creating a perfect triangular cross section generates more waste metal than a circular cross section, further increasing production costs.

Final decision

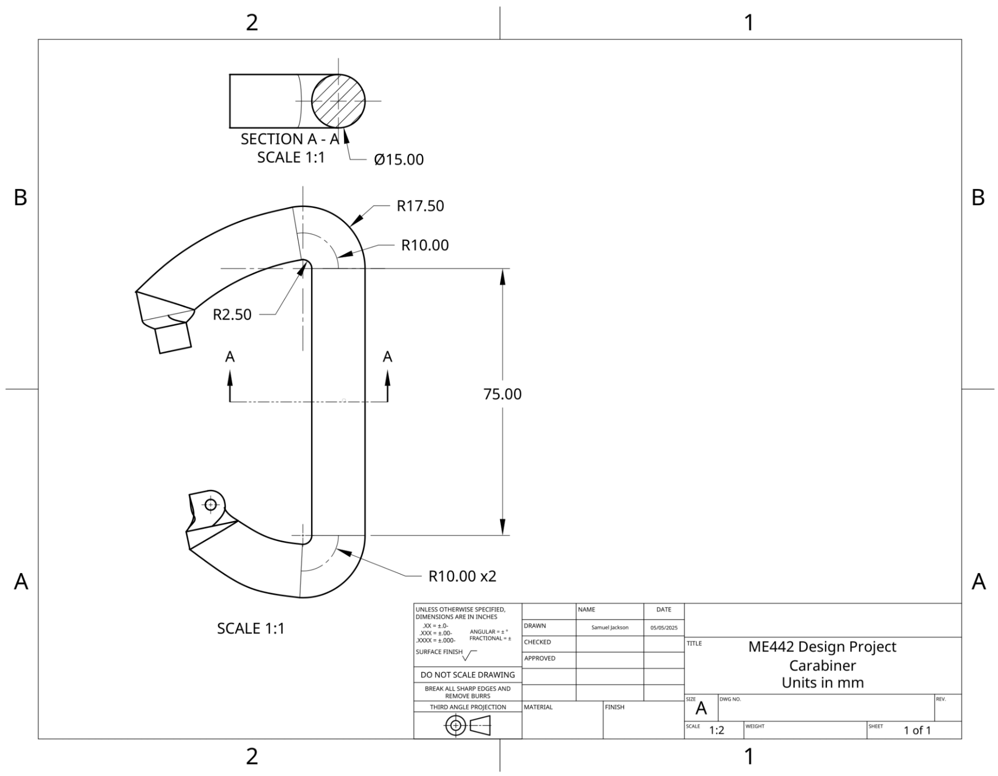

Figure 6. Drawing for the final design

After doing the proper calculations to determine important numbers like the factor of safety, stresses, and deflection, it has become clear that the carabiners with circular cross-sections are the strongest and, therefore, safer option. Not only this, but they are cheaper and easier to manufacture costing manufacturers and investors less in the long run. The manufacturing processes required to make carabiners of circular cross-section would more than likely have a higher success rate, resulting in less material, and therefore money, waste. Still, the factor of safety for both designs is much lower than desired.

While the circular cross section is superior to the triangular one, it was determined that in order to increase the strength of the carabiners, the point of force concentration needed to be shifted to allow most of the force to be transmitted through the beam further from the gate. During the fatigue analysis with the Soderberg criterion, it was found that changing the distance from the center axis of the carabiner to the centroid of the cross section had a dramatic impact on the factor of safety. Further analysis showed that this is due to the axial stress due to bending as seen in Equation 14. The moment,

$$M = r_c F \tag{14}$$

dramatically decreases when r_c is lowered. By shifting the application of the force through r_c, it allows the gate to act more as a support member, as opposed to one of the main strength elements. The reason for this is due to the large mitigation of bending stress. Under the defined assumptions for this project, the only aspect of Von Mises stress that isn't equal to zero is σ_x. Due to the horizontal shift in force concentration, there is a large bending stress that accounts for most of the σ_x value. This reveals a crucial revelation: the location of F, or applied force, must be brought closer to the main supporting "wall" of the carabiner. By doing this, the length used in the stress calculation's bending moment significantly decreases, offering greatly improved calculations for Von Mises stress and deflection. In typical climbing applications, the "D-shape" design is superior to the oval shape design for these reasons. Figure 6 illustrates the final design changes made opposed to the preliminary design changes. The design forces the ropes to move closer towards the main bar when tension is applied.

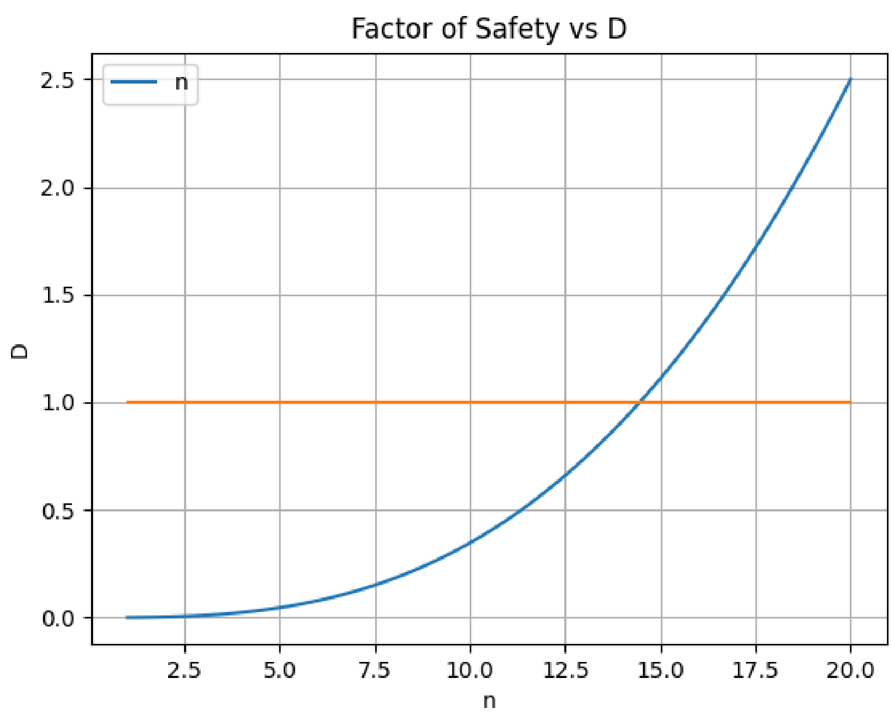

Figure 7. Graph of Factor of Safety as D changes

In order to analyze the appropriate dimensions for the final design, the factor of safety was analyzed in response to changes in a certain variable. For this analysis, the centroidal radius was fixed at 10 mm to allow for standard climbing ropes to fit in the confines of the corner of the carabiner, allowing the force to be distributed closer to the intended spot when the rope is forced towards that area when placed in tension. This leaves the diameter of the carabiner *D* as the remaining variable that can be shifted geometrically. Figure 7 shows the factor of safety *n* as *D* increases graphed with Python. It was found that the fatigue factor of safety is 1 when *D* is approximately 14.6 mm, and anything above would give a fatigue factor of safety of over 1. For this reason, *D* was chosen to be 15 mm. These changes yielded a deflection to length ratio of 0.00155 and a yield strength to modulus ratio of 0.0070, as well as a fatigue factor of safety of 1.1097. This factor of safety value is based off of Soderberg criterion, meaning it is likely more conservative and therefore safer. This is much better than the previous iterations, as it guarantees that the carabiner will not yield upon the first load. Still, as aluminum is a non-ferrous material, it cannot be assumed that the carabiner is expected to have infinite life, even if the factor of safety is over 1 [2] [4].

To find the cycle life for this carabiner design, the cycle life *N* is given by,

$$N = \left(\frac{\sigma_{ar}}{a}\right)^{\frac{1}{b}} \tag{15}$$

where *a* and *b* are the Basquin equation constants, given by Equations 16 and 17,

$$a = \frac{(fS_{ut})^2}{S_e} \tag{16}$$

$$b = -\frac{1}{3}log\left(\frac{fS_{ut}}{S_e}\right) \tag{17}$$

where f is the fatigue strength fraction. fS_{ut} altogether represents the fatigue strength, which is 159 MPa at 500 million cycles for 7075 T-6 aluminum, according to Matweb [6]. For the reversed stress amplitude σ_{ar} is given by,

$$\sigma_{ar} = \sqrt{\sigma_{max}\sigma_a} \tag{18}$$

This reversed stress amplitude is in accordance with the Smith-Watson-Topper criterion, which is very good for aluminum [2]. With the above calculation pathway, the cycle life was predicted to be *N* = 540574.05, or 540,000 cycles, meaning this carabiner is highly unlikely to break within its service life, as these calculations assume maximum load.

This carabiner is predicted to not yield at its maximum load of 10kN and is not likely to fail when subjected to cyclic loading. The circular profile will also make this product easy to manufacture. This carabiner's mechanical and simplistic properties makes it a highly effective design.

Works Cited

[1] *ASM material data sheet*. Available at: https://asm.matweb.com/search/specificmaterial.asp?bassnum=ma7075t6 (Accessed: 09 May 2025).

[2] Budynas, R. G., & Nisbett, J. K. (2024). *Shigley's Mechanical Engineering Design* (11th ed.). McGraw-Hill Education.

[3] PTC Help Center. (n.d.). https://support.ptc.com/help/wrr/r12.0.1.0/en/index.html#page/wrr/ReferenceGuide/prediction/endurance_limits.html

[4] Blair, K., Custer, D., Graham, J., Okal, M., & Massachusetts Institute of Technology Center for Sports Innovation. (2002). Analysis of fatigue failure in D-shaped carabiners. MIT Center for Sports Innovation. https://web.mit.edu/sp255/www/reference_vault/Fatigue_Presentation.pdf#:~:text=Carabiner%20Load%20Analysis.%20%E2%80%A2%20Worst%20case%20scenario%20is%20factor%202&text=Used%20to%20evaluate%20stress%20concentrations%20in%20carabiner%20to%20predict%20failure%20area.

[5] ScienceDirect. (n.d.). Fatigue of materials. In ScienceDirect Topics. Elsevier. Retrieved May 11, 2025, from https://www.sciencedirect.com/topics/materials-science/fatigue-of-materials#:~:text=Fatigue%20strength%20and%20fatigue%20limit

[6] MatWeb. (n.d.). *Aluminum Alloy 7075-T6; 7075-T651 (UNS A97075) – Material property data*. Retrieved May 11, 2025, from https://www.matweb.com/search/datasheet.aspx?MatGUID=4f19a42be94546b686bbf43f79c51b7d

Appendix

Variables used in calculations	Values
F	10000 N
R_i	17.44 mm
R_o	28.78 mm
R_c	23.11 mm
A_{circle}	100.99 mm^2
E	71.7 GPa
G	26.9 GPa
δ	0.844 mm
σ_a	277.46 MPa
σ_m	277.46 MPa
S_e	123.94 MPa
S_u	572 MPa
n	0.23

Table 1. The values used and found in calculations for the first circular cross-section

Variables used in calculations	Values
F	10000 N
R_i	13.75 mm
R_o	26.25 mm
R_c	20 mm
A_{circle}	122.71 mm^2
E	71.7 GPa

Variables used in calculations	Values
G	26.9 GPa
δ	0.373 mm
σ_a	277.46 Mpa
σ_m	277.46 Mpa
S_e	123.94 Mpa
S_u	572 Mpa
n	0.35

Table 2. The values used and found in calculations for the second circular cross-section

Variables used in calculations	Values
F	10000 N
R_i	19.74 mm
R_o	33.49 mm
r_c	24.36 mm
A	110.85 mm^2
E	71 GPa
G	26.9 GPa
δ	0.728 mm
σ_a	379.38 Mpa
σ_m	379.38 Mpa
S_e	123.94 Mpa
S_u	572 Mpa
n	0.26

Table 3. The values used and found in calculations for the first triangular cross-section

Variables used in calculations	Values
F	10000 N
R_i	19.74 mm
R_o	33.59 mm
r_c	20 mm
A	110.85 mm^2
E	71 GPa
G	26.9 GPa
δ	0.411 mm
σ_a	379.38 Mpa
σ_m	379.38 Mpa
S_e	123.94 Mpa
S_u	572 Mpa
n	0.31

Table 4. The values used and found in calculations for the second triangular cross-section

Variables used in calculations	Values
k_a	.92
k_b	.85
k_c	.85
k_e	.814
S'_e	228.8 Mpa
S_e	123.93
σ_a	281.13

Variables used in calculations	Values
σ_m	281.13
S_{ut}	572 Mpa

Table 5. Fatigue values for calculations of circular cross-sectional area

Variables used in calculations	Values
k_a	.92
k_b	.85
k_c	.85
k_e	.814
S'_e	228.8 Mpa
S_e	123.93
σ_a	379.38
σ_m	379.38
S_{ut}	572 Mpa

Table 6. Fatigue values for calculations of triangular cross-sectional area

$\delta = \frac{\pi F R^2}{2AeE} - \frac{\pi F R}{2AE} + \frac{\pi c F R}{2AG}$	The equation for deflection in a curved beam
$r_n = \frac{A}{\int \frac{dA}{r}}$	Equation for the location of the centroidal axis in a curved beam.
$A = \pi r^2$	Equation for the circular cross sectional area for the carabiner.
$A = \frac{1}{2} bh$	Equation of the triangular cross sectional area for the carabiner
$I = \frac{\pi d^4}{64}$	Equation for moment of inertia for a circle

$I = \frac{sh^3}{36}$	Equation for moment of inertia for a triangle
$\sigma_{vm} = \sqrt{(\sigma_x)^2}$	Equation for 2D Von Mises stress assuming $\sigma_y = \tau_{xy} = 0$
$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = \frac{1}{n}$	Equation for Soderberg factor of safety

Table 7. The equation list

ME 442—Mechanical Engineering Design I

Team Self-Evaluation of Project Contribution

Project Title: Carabiner Design Analysis

Group Member	**I**	**II**	**III**	**IV**	
Student Name (Print)	Terrence Andre San Gabriel	Mody Bawi	James Cline	Sam Jackson	
Signature	Terrence Andre San Gabriel	Mody Bawi	James Cline	Sam Jackson	
Individual Contribution (%)	25%	25%	25%	25%	Total **100%**