# Product Analysis

Trailer Winch System

**Group 5**
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# Summary

The product is a trailer winch system used to pull heavy objects onto a trailer. It is attached to the trailer, either temporarily via bolts/pins or permanently via welding, and pulls the object attached to it towards it when the crank handle is rotated.

Analysis of the part was based on the maximum 2,000 lbf weight rating for the winch. Each critical component was analyzed based on the force/torque transmitted to it, and the force/torque it transmits to a subsequent part. Both traditional analytical techniques and SolidWorks FEA were used. All components of the system were considered AISI 1020 CD Steel with fatigue SN curves derived from ASME Carbon Steel. According to analytical techniques, none of the components fail due to yielding, but the rod fails due to fatigue based on the maximum rating. According to SolidWorks FEA, none of the components fail due to yielding, but the rod and the handle all fail due to fatigue based on the maximum rating.

# Introduction

The product is operated by attached a strap between the shaft/drum of the gear and the desired object. The entire housing must be attached to a stable material either by welding or by bolts/pins. The handle transmits a torque to the worm via a transmitting rod, and the worm rotates the gear and gear shaft/drum. This compound system allows for a single person to exert enough force to pull objects weighing up to 2,000 lbf.

The target market is for pulling vehicles, typically either ones that are stuck and require a large force to move, or being able to transport vehicles in general. The benefits of this winch of an electric winch are a lower price and the only power source necessary is the operator. Analyses performed on the components of the system are traditional FBD/stress and fatigue equations, and more advanced SolidWorks FEA (Finite Element Analysis). The maximum load on the system was taken as 2,000 lbf and all components were tested for fatigue, with infinite life based on 10^6 cycles.

# Method and Findings from Analytical Analysis

**Universal assumptions:**

- All components were analyzed based on AISI 1020 CD Steel with fatigue SN curves derived from ASME Carbon Steel.
- Based on Table A-20, the yield strength is 57 kpsi and the tensile strength is 68 kpsi.
- Fine machined fillets have a radius of 0.04 in.
- Yielding factor of safety does not include stress concentration factors due to being a ductile material.

**Gear and Gear Shaft:**

- The gear was analyzed based on its tooth. The assumptions made were:
  - The entire reaction load acts on one tooth.
  - The reaction force acts on the tip of the gear tooth.
- The geometrical dimensions of the tooth were analyzed via a caliper.*
- FBDs indicate that when the gear and gear shaft receive a force of 2000 lbf from the load, it transmits a force of 577 lbf to the worm. **
- The critical location of the gear tooth was found to be at the base of the tooth where there is a fine machined fillet.
  - A force acting a gear tooth will generate a stress that is propagated down throughout the tooth. This stress is greatest where the tooth meets the “wall” of the gear (the base of the tooth), which in this case has a very fine machined fillet.
- The Lewis Factor equation was used to determine the maximum stress acting on the gear tooth. This value was determined to be 29.3 kpsi. The yielding factor of safety was determined to be 1.943.
- With the maximum stress, fatigue was calculated based on zero based cycles (the minimum stress was taken to be 0) and the factor of safety is determined via ASME Elliptic. *
- The stress concentration factor of 1.470, Marin factors $k_a = 0.883$ (CD), $k_b = 1.022$ (non-rotating, rectangular cross section), and $k_c = 1$ (pure bending), the tensile strength of 68 kpsi, and the maximum stress of 29.3 kpsi were used to find the fatigue factor of safety of 1.252. This indicates that the part has infinite life. ***

* These steps are repeated for all subsequent components and will not be listed again for said components.
** FBDs for all components are included in the appendix.
*** Mathematical steps for all components are included in the appendix.

## Worm:

- The assumptions made were:
  - The force transmitted to the worm from the gear is perfectly normal with the worm tooth face.
  - Only one tooth from the gear acts on the worm at any given time.
  - The sketch profile of the worm tooth is assumed to be rectangular instead of trapezoidal for equation calculations so that forces on it are 2D instead of 3D and can be broken down into “x” and “y” components.
- FBDs indicate that when the worm receives a force of 577 lbf from the gear tooth, it transmits a torque of 392 lbf·in to the transmitting rod.
- The critical location of the worm tooth was found to be at the base of the tooth where there is a fine machined fillet.
  - A force acting a worm tooth will generate a stress that is propagated down throughout the tooth. This stress is greatest where the tooth meets the “wall” of the worm (the base of the tooth), which in this case has a very fine machined fillet.

- The maximum stress from the axial stress equation was found to be 6.21 ksi. The yielding factor of safety was determined to be 9.18.
- The stress concentration factor of 1.288, Marin factors $k_a = 0.883$ (CD), $k_b = 1.021$ (non-rotating, "rectangular" cross section), and $k_c = 0.85$ (pure axial), the tensile strength of 68 kpsi, and the maximum stress of 6.21 kpsi were used to find the fatigue factor of safety of 6.02. This indicates that the part has infinite life.

Transmitting Rod:
- The assumptions made were:
  - The pin that connects the rod and the worm is neglected as a component to keep the project component count at five.
  - The worm and the rod are rigidly connected, such that the torque transmitted from the worm is perfectly transmitted to the rod.
- FBDs indicate that when the transmitting rod receives a torque of 392 lbf·in from the worm, it transmits a torque of 392 lbf·in to the handle.
- The critical location of the transmitting rod was found to be at the fine machined fillet between the larger diameter and the smaller diameter near the top of the object. A diagram is provided in the appendix.
  - While there is a fillet at the top of the rod, the hole in the rod has a much greater effect on its structural integrity. The stress is on the top of this hole due to torsion.
- The maximum shear stress from the shear stress equation was found to be 15.04 ksi. The yielding factor of safety was determined to be 3.79.
- The stress concentration factor of 2.31, Marin factors $k_a = 0.883$ (CD), $k_b = 0.965$ (rotating system), and $k_c = 0.59$ (pure torsion), the tensile strength of 68 kpsi, and the maximum shear stress of 15.04 kpsi were used to find the fatigue factor of safety of 0.543. This indicates that the part has finite life.
- Equivalent fully reversed stress is calculated as 54.1 kpsi, and after finding f = 0.9, a = 219, b = -0.185, the number of cycles until failure is calculated as 1950 cycles.

Handle:
- The assumptions made were:
  - The handle was treated as a straight rectangular bar.
- FBDs indicate that when the transmitting rod receives a torque of 392 lbf·in from the transmitting rod, it transmits a force of 49.0 lbf to the operator.
- The critical location of the handle was found to be at hole where the torque is applied. If the assumption of a straight bar were not made, the critical location would be at the bend in the bar.
  - For a straight bar, the area of greatest stress is at the hole in the bar. This hole compromises the structural integrity of the bar. If the bar is bent however, the bend in the bar compromises the structural integrity even more so.
- The maximum shear stress from the shear stress equation was found to be 2.36 kpsi.
- The yielding factor of safety was determined to be 24.2.
- The stress concentration factor of 1 (none applicable), Marin factors $k_a = 0.883$ (CD), $k_b = 0.808$ (rotating system), and $k_c = 0.59$ (pure torsion), the tensile strength of 68

kpsi, and the maximum shear stress of 2.36 kpsi were used to find the fatigue factor of safety of 4.07. This indicates that the part has infinite life.

Housing:
- The assumptions made were:
  - The housing is rigidly attached by its base using the whole surface area (e.g. welding) as to not add in pins as components.
  - The surface holding the housing is assumed to be unmovable and indestructible.
- FBDs indicate that when the housing receives two forces of 1000 lbf each from the gear shaft, it transmits a force of 2000 lbf to the surface that is holding it down.
- The critical location of the housing is found to be at the hole were the gear shaft passes through.
  - In a solid wall, a hole to fit a shaft through will compromise the structural integrity the greatest and suffers from the greatest stress at that hole.
- The maximum stress from the axial stress equation was found to be 7.22 kpsi.
- The yielding factor of safety was determined to be 7.89.
- The stress concentration factor of 1 (none applicable), Marin factors $k_a = 0.883$ (CD), $k_b = 0.910$ (non-rotating, rectangular cross section), and $k_c = 0.85$ (pure axial), the tensile strength of 68 kpsi, and the maximum shear stress of 7.22 kpsi were used to find the fatigue factor of safety of 1.233 This indicates that the part has infinite life.

| Component | Receives | Transmits | $\sigma_{max}$ | $\boldsymbol{n_y}$ | $k_a$ | $k_b$ | $k_c$ | $K_f, K_{fs}$ | $\boldsymbol{n_f}$ |
|---|---|---|---|---|---|---|---|---|---|
| **Gear** | 2000 lbf | 577 lbf | 29.3 kpsi | **1.943** | 0.883 | 1.022 | 1 | 1.470 | **1.252** |
| **Worm** | 577 lbf | 392 lbf·in | 6.21 kpsi | **9.18** | 0.883 | 1.021 | 0.85 | 1.288 | **6.02** |
| **Transmitting Rod** | 392 lbf·in | 392 lbf·in | 15.04 kpsi | **3.79** | 0.883 | 0.965 | 0.59 | 1.235 | **0.543*** |
| **Handle** | 392 lbf·in | 49.0 lbf | 2.36 kpsi | **24.2** | 0.883 | 0.808 | 0.59 | 1 | **4.07** |
| **Housing** | 2000 lbf | 2000 lbf | 7.22 kpsi | **7.89** | 0.883 | 0.910 | 0.85 | 2.41 | **1.233** |

Table 1. *Values from Analytical Analysis*

*The transmitting rod lasts for 1950 cycles before failure.

## Methods and Findings from Computational Analysis

All computational analyses were performed using SolidWorks FEA as the tool and technique of choice. The weak points of the component were identified visually in the FEAs.

All FEAs were performed as follows:

- A static study was performed where a portion of the component is fixed and a force/torque is applied based on the previous section.
  - The fixture is a fixed geometry.
  - The external load is either a force or a torque acting on a face, with magnitudes based on the values obtained from the FBDs.
  - The mesh was set to a fine mesh density.
  - The material is set to AISI 1020.
- A fatigue study was performed using the static study.
  - The event is 10^6 zero based cycles of the static study (cycles from 0 to the maximum load of the static study)
  - The fatigue SN curve was derived from the material elastic modulus based on ASME Carbon Steel curves.

SolidWorks FEA provides data on Von Mises stress (includes maximum stress the part undergoes), FOS for the Von Mises stress, fatigue life cycle (includes minimum cycles before failure), and load factor (FOS) for fatigue.

| Component | Fixture | Loading |
|---|---|---|
| Gear | Shaft of the gear | Force acting normal on the –z face of the top-most tooth |
| Worm | Wall of the hole in the middle of the worm | Force acting normal on the top face of the tooth |
| Transmitting Rod | Hole where the pin that rigidly connects the rod and worm is currently missing from | Torque acting on the entire rod (larger diameter) |
| Handle | The upper portion of the handle | Torque acting on where the hole is that would normally connected to the rod |
| Housing | Entire bottom of housing | Two forces acting in the x direction on the two holes where the gear shaft would go |

Table 2. *Loading and Fixtures*

| Component | $\sigma_{max}$ | $\boldsymbol{n_{y,min}}$ | $N_{min}$ | $\boldsymbol{n_{f,min}}$ |
|---|---|---|---|---|
| Gear | 0.006 kpsi | 8062 | 1e6 cycles* | 3700 |
| Worm | 1.746 kpsi | 29.2 | 1e6 cycles* | 31.26 |
| Transmitting Rod | 46.5 kpsi | 1.097 | 28200 cycles | 0.449 |
| Handle | 43.8 kpsi | 1.164 | 33830 cycles | 0.477 |
| Housing | 20.8 kpsi | 2.46 | 1e6 cycles* | 1.093 |

Table 3. *Values from Computational Analysis*

* Indicative of infinite life

Fatigue Von Mises FEAs:

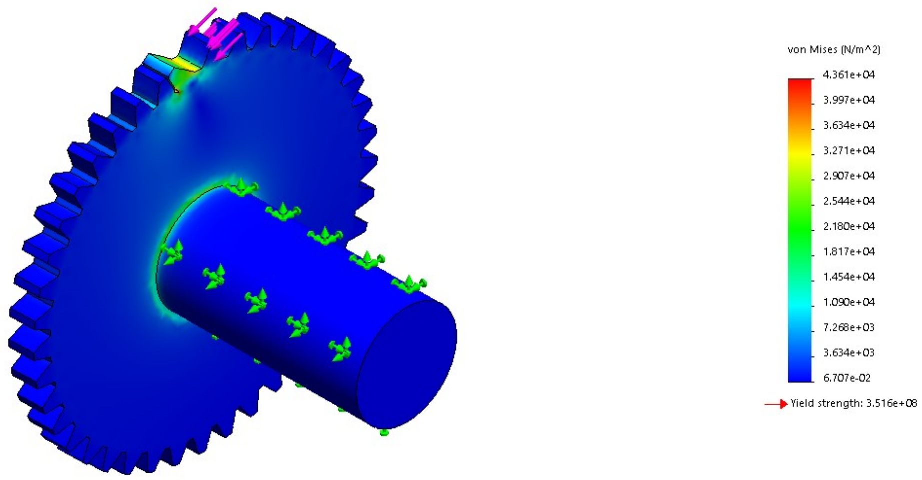


Figure 1. *Gear and Gear Shaft Von Mises FEA*

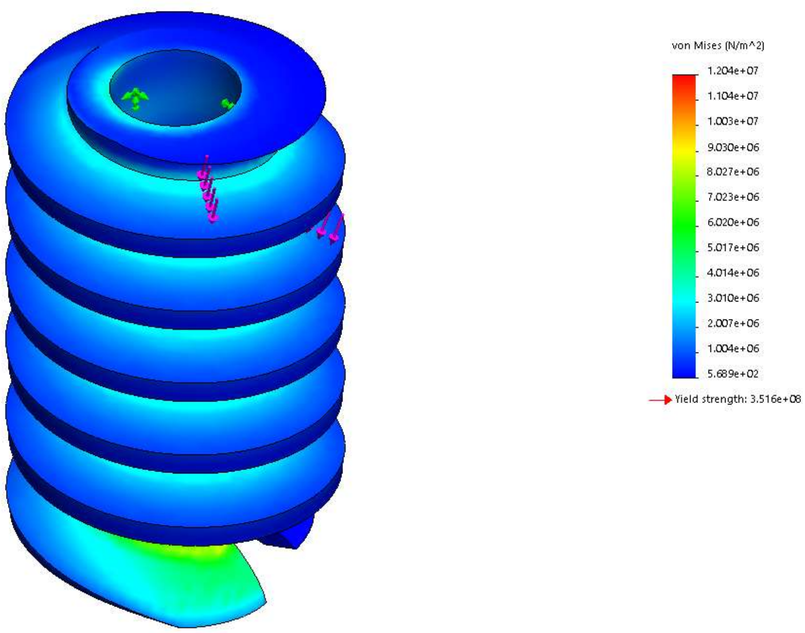


Figure 2. *Worm Von Mises FEA*

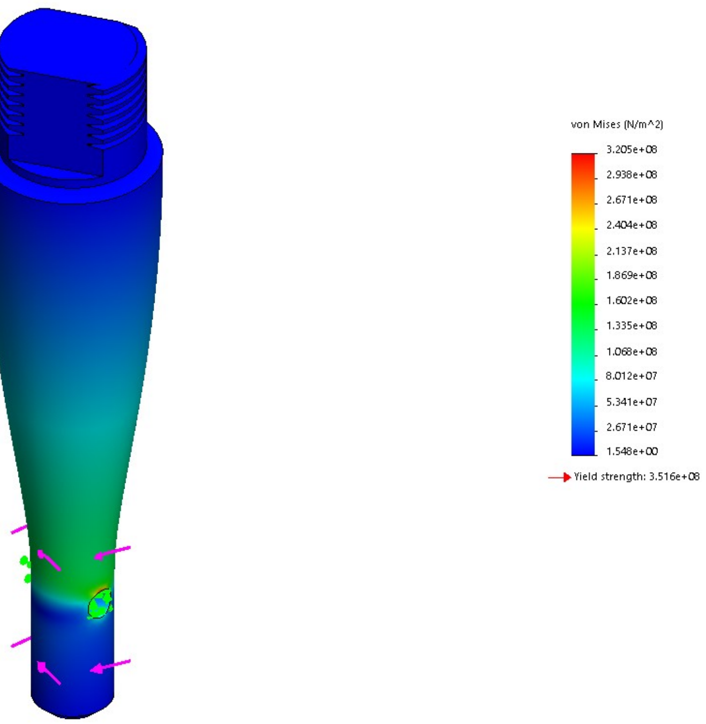


Figure 3. *Transmitting Rod Von Mises FEA*

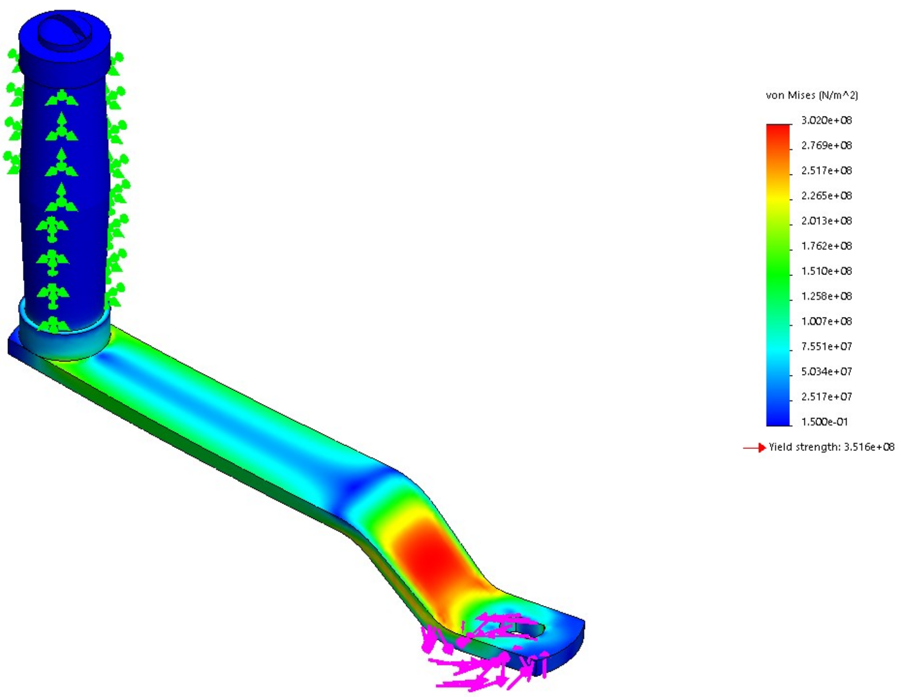


Figure 4. *Handle Von Mises FEA*

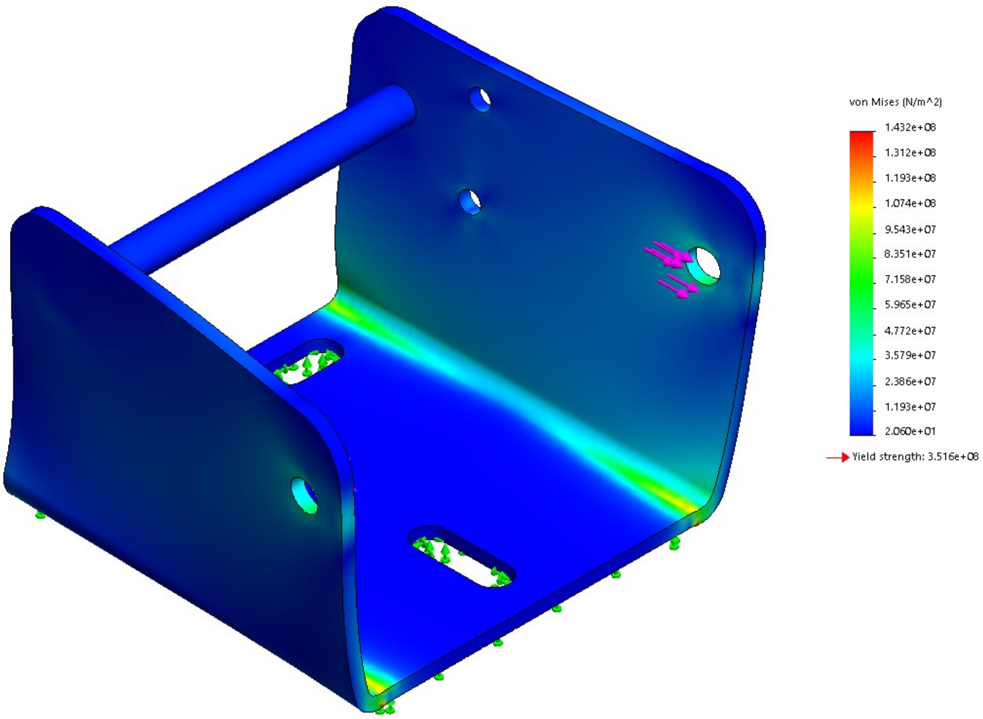


Figure 5. *Housing Von Mises FEA*

| Component | Analytical Weak Point | Computational Weak Point |
|---|---|---|
| Gear | Base of tooth | Base of tooth, fillet between gear wall and shaft |
| Worm | Base of tooth | Base of tooth, especially at the bottom |
| Transmitting Rod | Hole in the rod | Hole in the rod |
| Handle | The hole in the bar (straight bar) | The bend in the bar |
| Housing | The holes where the gear shaft resides | The holes where the gear shaft resides, the fillets were the walls bend |

Table 4. *Weak Points Comparison between Analytical and Computational*

The weak points between the two methods however were similar, with the only differences being that SolidWorks does not identify only the weakest point, but multiple weak points.

The other FEA diagrams are includes in the appendix.

# Conclusions

The computational analysis with SolidWorks FEA correlates fairly well with the more traditional analytical analysis. Generally, the SolidWorks FEA values' magnitude were greater than the magnitude of the values from the equations. This suggests that the analytical analysis is in line with the computational analysis, aside from lacking the advanced mathematics that SolidWorks FEA is capable of performing by deforming fine meshes. The only exception to this was the handle. The values from the equations suggests that the handle should not yield and have an infinite fatigue life. SolidWorks FEA agrees with the non-yielding nature of the component, but disagrees with the fatigue analysis by suggesting it has a much lower total life of 33830 cycles.

The differences between the analytical analysis and the computational analysis can also be attributed to the assumptions made in the former. Here are the assumptions again, the reasoning behind them, and what they may have inadvertently caused.

Gear:

- The entire reaction load acts on one tooth.
    - In actuality, multiple teeth from the gear will intermesh with the worm, and will experience different loads as the gear rotates. This resulted in the analytical analysis in being a little less conservative by assuming it experiences a little less stress than it actually does.
- The reaction force acts on the tip of the gear tooth.
    - While there is a force on the tip of the gear tooth, the gear tooth experiences a force gradient down towards its base. A more uniform deformation of the tooth could result in a much smaller bending moment at the base, which results in the part lasting much longer than if the force were concentrated on the tip.

Worm:

- The force transmitted to the worm from the gear is perfectly normal with the worm tooth face.
    - The teeth on the gear are not angled. Due to being a non-helical gear, only a portion of the teeth actually intermesh with the worm as the two rotate. This results in greater concentration, and could result in the part having shorter life.
- Only one tooth from the gear acts on the worm at any given time.
    - As mentioned in the gear assumptions, multiple teeth from the gear would be intermeshed with the worm. This means the analytical analysis performed for the worm is less conservative by assuming it experiences less stress than it should.
- The sketch profile of the worm tooth is assumed to be rectangular instead of trapezoidal for equation calculations so that forces on it are 2D instead of 3D and can be broken down into "x" and 'y" components.
    - This is the greatest assumption made for the worm. The actual forces applied to the worm should be 3D, so a "z" component would have to be accounted for.

This suggests that less force is actually transmitted from the gear to the worm, which may make calculations for subsequent components inaccurate.

Transmitting Rod:

- The pin that connects the rod and the worm is neglected as a component to keep the project component count at five.
    - This was a decision made to fit the criteria of the project. Its effect however should actually be reasonably negligible if the pin is made of a sufficiently strong material. If the pin is indestructible, the worm and the rod should always be perfectly rigidly connected and transmit torque perfectly. In the much more likely scenario where the pin is capable of failure and deformation, some of the torque is lost to the pin as it transfers it to the worm.
- The worm and the rod are rigidly connected, such that the torque transmitted from the worm is perfectly transmitted to the rod.
    - As mentioned above, this scenario is true for an indestructible pin, but a realistic situation would have some torque lost. This could lead to less torque being transferred perfectly from the worm to the handle (and vice versa).

Handle:

- The handle was treated as a straight rectangular bar.
    - This is the greatest assumption and led to the greatest difference between analytical and computational analyses. The bend in the bar sacrifices a lot of structural integrity and the FEA capitalizes on this by showing the significant decrease in total life the handle has. Had the analytical analysis included the much more organically shaped bar, it may have not predicted infinite life.

Housing:

- The housing is rigidly attached by its base using the whole surface area (e.g. welding) as to not add in pins as components.
    - This is not a completely idealistic assumption. In practical scenarios, the housing would be secured to its respective platform with little tolerance with welding being a very practical option particularly for long term use.
- The surface holding the housing is assumed to be unmovable and indestructible.
    - Because this report is only on the winch itself, this was a safe assumption and doesn't change how the values were calculated very much.

While some of these assumptions compromised how much stress the device actually deals with, they made the analytical analyses feasible, and were therefore a necessary decision.

Potential Improvements to Product:

Based on SolidWorks FEA, the gear and the worm components of the winch are well beyond safe, with the gear suffering minimal damage from tooth interface, and the worm similarly suffering minimal damage from having the axial force applied on its teeth.

The housing, while safe, is very close to finite life. Its fatigue factor of safety sits at 1.093. Changes to the housing is unnecessary, but given this situation it may be advisable to make some small improvements to the housing such as having thicker walls, which would decrease the amount of stress the component receives by increasing the area over which the force is applied.

**Transmitting Rod:**

The transmitting rod is the weakest of all the components. Although it does not yield, its total life is only 28200 cycles. Aside from using a material with greater yielding and tensile strength, the analytical analyses grant insight into how to improve its factor of safety.

The transmitting rod can be improved by having a greater diameter for its main body. The shear stress the rod experiences is inversely related to its diameter, where cubic diameter remains in the denominator of the equation. E.g. by increasing the diameter by just 0.02 inches, the shear stress the rod experiences decreases by 850 psi (based on analytical equations).

The other improvement applicable to the transmitting rod is decreasing the pin hole that goes straight through the rod. A smaller radius of that pin hole results in a smaller q value, which can lead to a smaller stress concentration factor. However, unintended effects such as a smaller pin to hold the rod and the worm together could greatly affect product use, so it is important to reevaluate the pin is a critical component if such a decision is considered.

**Handle:**

As mentioned in the report, the bend in the bar results in a significant loss of total life. A straight bar would move the weak point from the bar to the hole where the bar and the transmitting rod is connected. This change would move the weak area from being at the bend in the bar to the whole where the handle and the transmitting rod are connected.

Aside from the bend in the bar, a greater thickness in the bar would also improve its life by decreasing how much shear stress it experiences. In general, this is true for greater cross-section area. E.g. by increasing the thickness to 0.3 in. the shear stress is decreased by 111 psi (based on analytical equations). This would lead to both greater yielding factor of safety and fatigue factor of safety.

## References:

Budynas, R. G., & Nisbett, J. K. (2015). Shigley's Mechanical Engineering Design (10th ed.). New York, NT: McGraw-Hill.

Lewis Factor Equation For Gear Tooth Calculations
LLC. Edge - https://www.engineersedge.com/gears/lewis-factor.htm

Standard Drill Bit Sizes For Cnc Machining (conversion Tables) https://www.3dhubs.com/knowledge-base/standard-drill-bit-sizes-cnc-machining-conversion-tables

1 Ton Capacity Hand Winch, Harbor Freight Tools
https://www.harborfreight.com/media/catalog/product/i/m/image_20523.jpg

# Appendix A: Force Analysis

### Gear and Gear Shaft:

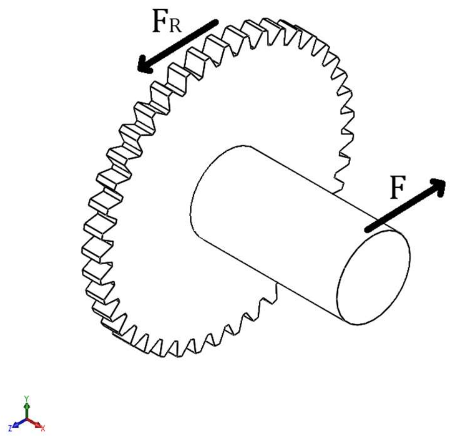


Figure 6. *FBD of Gear and Gear Shaft*

$F = 2000\ lbf$

$+\circlearrowleft \sum M_x = 0: -(2000\ lbf)\left(\frac{1.5\ in.}{2}\right) + F_R\left(\frac{5.2\ in.}{2}\right) = 0$

$F_R = 576.9230769\ lbf$

### Worm:

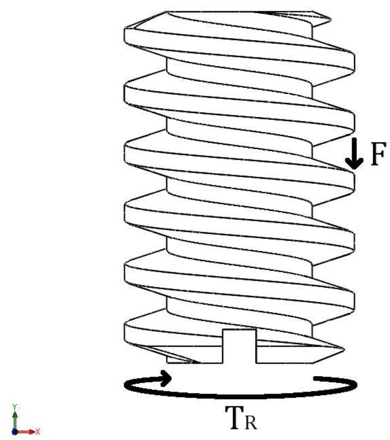

Figure 7. *FBD of Worm*

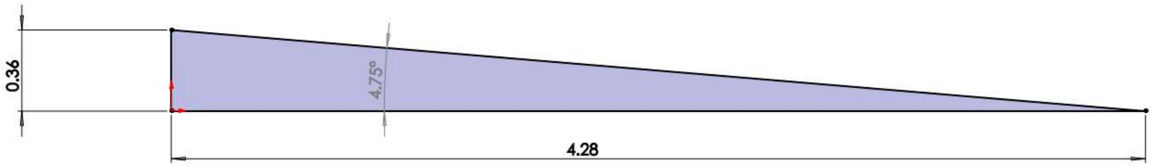


Figure 8. *Profile Angle of Worm*

$$F = 576.9230769\ lbf$$
$$\theta = \tan^{-1}\left(\frac{0.356\ in.}{2\pi\left(\frac{1.363\ in.}{2}\right)}\right) = \tan^{-1}\left(\frac{0.356\ in.}{4.281990787}\right) = 4.75°$$
$$F_x = F\cos\theta = (576.9230769\ lbf)\cos(4.75°) = 574.9416361\ lbf$$
$$+\circlearrowleft \sum M_y = 0: -T_R + (574.9416361\ lbf)\left(\frac{1.363\ in.}{2}\right) = 0$$
$$T_R = 391.822725\ lbf \cdot in. = 32.65189375\ lbf \cdot ft$$

Transmitting Rod:

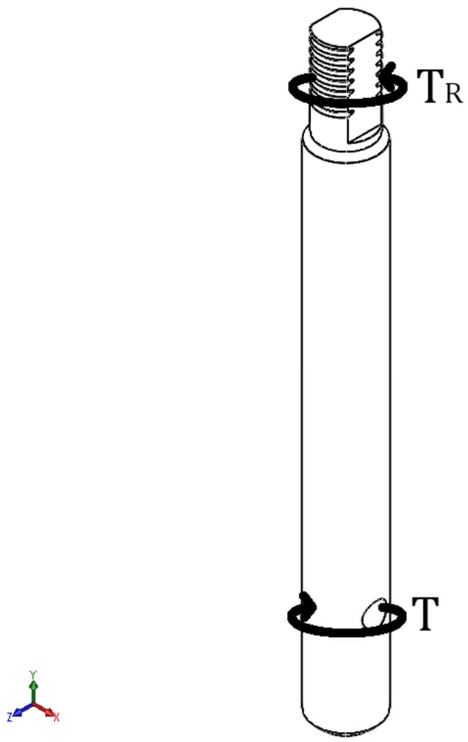


Figure 9. *FBD of Transmitting Rod*

$$T = 391.822725\ lbf \cdot in.$$

$$+\circlearrowleft \sum M_y = 0: -T + T_R = 0$$
$$T_R = 391.822725\ lbf \cdot in.$$

Handle:

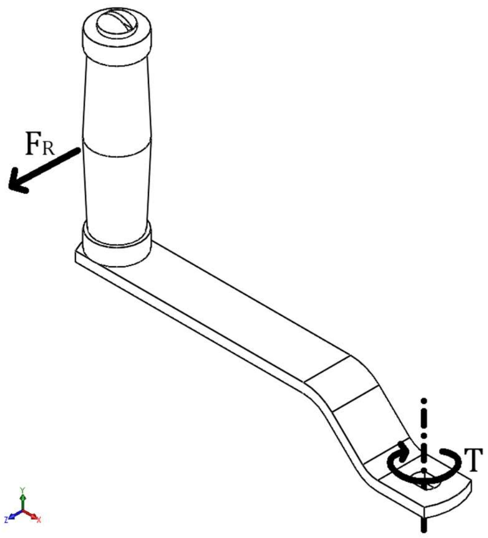


Figure 10. *FBD of Handle*

$$T = 391.822725\ lbf \cdot in.$$
$$+\circlearrowleft \sum M_y = 0: -(391.822725\ lbf \cdot in.) + F_R(8\ in.) = 0$$
$$F_R = 48.97784063\ lbf$$

Housing:

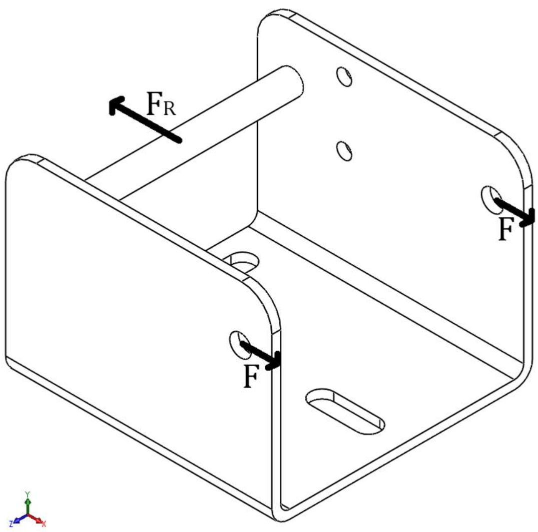


Figure 11. *FBD of Housing*

$$F = \frac{2000\ lbf}{2} = 1000\ lbf$$
$$+\rightarrow \sum F_x = 0: 2F - F_R = 0$$
$$F_R = 2000\ lbf$$

The reaction force on the housing is divided between the number of pins holding the housing down. The scenarios are:

1. 2 pins holding the housing on the far wall
2. 3 pins holding the housing on the base
3. 1 "pin" holding the housing via the rod attached to it
4. 1 "pin" holding the housing via welding the base of the housing to a platform

# Appendix B: Stress Analysis

AISI 1020 Steel
$S_{ut} = 68\ kpsi; S_y = 57\ kpsi\ [Table\ A - 20]$

**Gear and Gear Shaft:**

Lewis Equation:
$$\sigma = \frac{W_t P}{FY}$$

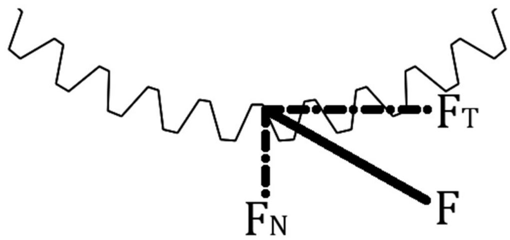


**Figure 12.** *Force Components on Gear Tooth*

$$Wound\ up\ cable\ diameter = 1.3855\ in.$$

$$+\circlearrowleft \sum M_O = 0: (2000\ lbf)\left(\frac{1.3855\ in.}{2}\right) - F_T\left(\frac{5.2\ in.}{2}\right) = 0$$
$$F_T = 532.8846154\ lbf$$

$$W_t = F_T = 532.8846154\ lbf\ [tangential\ tooth\ load]$$
$$P = \frac{\pi}{p} = \frac{\pi}{0.39} = 8.055365778\ [diametral\ pitch]$$

$F = 0.3685\ in. [face\ width]$
$Y = 0.397\ [Lewis\ Form\ Factor]$

$\sigma = 29.34204973\ ksi$

**Worm:**

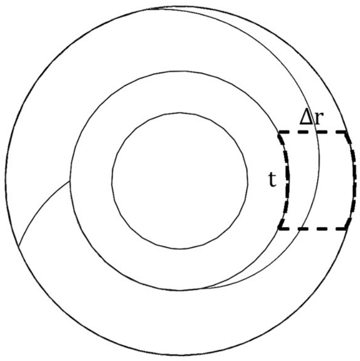


Figure 13. *Area of Worm Axial Force Acts On*

$$\sigma = \frac{P}{A} = \frac{576.9230769\ lbf}{t\Delta r} = \frac{576.9230769\ lbf}{(0.3685\ in.)\left(\frac{1.363\ in. - 0.8585\ in.}{2}\right)} = 6.206535502\ kpsi$$

$\sigma = 6.206535502\ kpsi$

**Transmitting Rod:**

$$\tau = \frac{Tr}{J} = \frac{(391.822725\ lbf * in)\left(\frac{0.51\ in.}{2}\right)}{\frac{\pi(0.51\ in.)^4}{32}} = 15.04351081\ kpsi$$

$\tau = 15.04351081\ ksi$

**Handle:**

Treating it as if it were a flat bar.

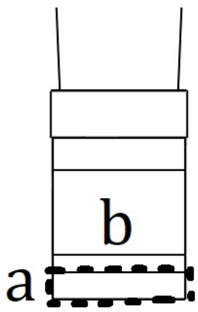


Figure 14. *Cross-Sectional Area of "Flat" Bar*

$$\tau = \frac{Tr}{J} = \frac{Tr}{\frac{ab(a^2+b^2)}{12}} = \frac{(391.822725\ lbf * in)\left(\frac{0.2\ in.}{2}\right)}{\frac{(0.2\ in.)(0.9860\ in.)((0.2\ in.)^2 + (0.9860\ in.)^2)}{12}}$$
$$= 2.355588033\ kpsi$$
$$\tau = 2.355588033\ kpsi$$

Housing:

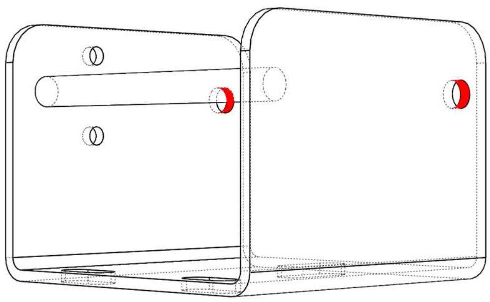

Figure 15. *Area of Housing Force Acts On*

$$\sigma = \frac{P}{A} = \frac{2(1000\ lbf)}{2\frac{(2\pi)\left(\frac{0.469\ in.}{2}\right)(0.1880\ in.)}{2}} = 7.220203379\ kpsi$$
$$\sigma = 7.220203379\ ksi$$

## Appendix C: Analytical Failure Analysis

Gear and Gear Shaft:

Yielding:

$$n_y = \frac{S_y}{\sigma} = \frac{57\ kpsi}{29.34204973\ kpsi} = 1.942604573$$

Fatigue:

$$S_e = 0.5 S_{ut} k_a k_b k_c$$
$$k_a = a{S_{ut}}^b = (2.70)(68\ kpsi)^{-0.265} = 0.8825694743\ [machined]$$
$$d_e = 0.808\sqrt{hb} = 0.808\sqrt{0.249\ in. * 0.3685\ in.}$$
$$= 0.24475399\ in.\ [non-rotating, rectangular\ cross\ section]$$
$$k_b = \left(\frac{d_e}{0.3}\right)^{-0.107} = \left(\frac{0.24475399\ in.}{0.3}\right)^{-0.107} = 1.022016454$$
$$k_c = 1\ [pure\ bending]$$

$S_e = 0.5(68\ kpsi)(0.8825694743\ )(1.022016454)(1) = 30.66801783\ ksi$

$K_f = 1 + q(K_t - 1)$

$\sqrt{a} = 0.246 - (3.08)(10^{-3})(68\ kpsi) + (1.51)(10^{-5})(68\ kpsi)^2 - (2.67)(10^{-8})(68\ kpsi)^3$
$= 0.0979870656$

$r = 0.04\ in. [assuming\ very\ fine\ machining\ tools]$

$$q = \frac{1}{1 + \frac{0.0979870656}{\sqrt{0.04\ in.}}} = 0.6711700711$$

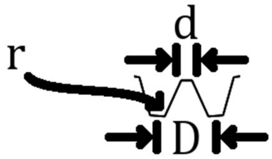


Figure 16. *Applicable D, d, and r Values for Gear Stress Concentration Factor*

$$\frac{r}{d} = \frac{0.04\ in.}{0.3185\ in.} = 0.125588697;\ \frac{D}{d} = \frac{0.3715\ in.}{0.3185\ in.} = 1.166405024$$

$K_t = 1.7\ [Figure\ A - 15 - 6]$

$K_f = 1 + (0.6711700711)(1.7 - 1) = 1.46981905$

$\sigma_{max} = 29.34204973\ kpsi$
$\sigma_{min} = 0$
$$\sigma'_m = K_f \frac{\sigma_{max} + \sigma_{min}}{2} = 21.56375183\ kpsi$$
$$\sigma'_a = K_f \left|\frac{\sigma_{max} - \sigma_{min}}{2}\right| = 21.56375183\ kpsi$$

ASME Elliptic:

$$\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2 = \frac{1}{{n_f}^2}$$

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2}} = \sqrt{\frac{1}{\left(\frac{21.56375183\ kpsi}{30.6467485\ kpsi}\right)^2 + \left(\frac{21.56375183\ kpsi}{57\ kpsi}\right)^2}} = 1.252430757$$

The part has infinite life in a fluctuating load according to the ASME Elliptic.

Worm:

Yielding:

$$n_y = \frac{S_y}{\sigma} = \frac{57\ kpsi}{6.206535502\ kpsi} = 9.18386755$$

Fatigue:

$$S_e = 0.5 S_{ut} k_a k_b k_c$$
$$k_a = a {S_{ut}}^{b} = (2.70)(68\ kpsi)^{-0.265} = 0.8825694743\ [machined]$$
$$d_e = 0.808\sqrt{hb} = 0.808\sqrt{t\Delta r} = 0.808\sqrt{(0.3685\ in.)\left(\frac{1.363\ in. - 0.8585\ in.}{2}\right)}$$
$$= 0.2463461018\ in.\ [non-rotating, "rectangular"\ cross\ section]$$
$$k_b = \left(\frac{d_e}{0.3}\right)^{-0.107} = \left(\frac{0.2463461018\ in.}{0.3}\right)^{-0.107} = 1.02130765$$
$$k_c = 0.85\ [pure\ axial]$$

$$S_e = 0.5(68\ kpsi)(0.8825694743)(1.02130765)(0.85) = 26.04973622\ kpsi$$

$$K_f = 1 + q(K_t - 1)$$

$$\sqrt{a} = 0.246 - (3.08)(10^{-3})(68\ kpsi) + (1.51)(10^{-5})(68\ kpsi)^2 - (2.67)(10^{-8})(68\ kpsi)^3$$
$$= 0.0979870656$$

$$r = 0.04\ in.\ [assuming\ very\ fine\ machining\ tools]$$

$$q = \frac{1}{1 + \frac{0.0979870656}{\sqrt{0.04\ in.}}} = 0.6711700711$$

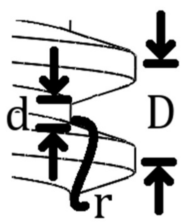


Figure 17. *Applicable D, d, and r Values for Worm Stress Concentration Factor*

$$\frac{r}{d} = \frac{0.04\ in.}{0.10\ in.} = 0.4;\ \frac{D}{d} = \frac{0.356\ in.}{0.10\ in.} = 3.56$$

$$K_t = 1.4\ [Figure\ A - 15 - 6]$$

$K_f = 1 + (0.6711700711)(1.4 - 1) = 1.288468028$

$\sigma_{max} = 6.206535502\ kpsi$
$\sigma_{min} = 0$
$\sigma_m = K_f \dfrac{\sigma_{ma} + \sigma_{min}}{2} = 3.936395924\ kpsi$
$\sigma_a = K_f \left|\dfrac{\sigma_{max} - \sigma_{min}}{2}\right| = 3.936395924\ kpsi$

ASME Elliptic:

$$\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2 = \frac{1}{{n_f}^2}$$

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2}} = \sqrt{\frac{1}{\left(\frac{3.936395924\ kpsi}{26.04973622\ kpsi}\right)^2 + \left(\frac{3.936395924\ kpsi}{57\ kpsi}\right)^2}} = 6.018890025$$

The part has infinite life in a fluctuating load according to the ASME Elliptic.

Transmitting Rod:

Yielding:

$$n_y = \frac{S_y}{\tau} = \frac{57\ kpsi}{15.04351081\ kpsi} = 3.789009143$$

Fatigue:

$S_e = 0.5 S_{ut} k_a k_b k_c$
$k_a = a{S_{ut}}^b = (2.70)(68\ kpsi)^{-0.265} = 0.8825694743\ [machined]$
$k_b = \left(\dfrac{d}{0.3}\right)^{-0.107} = \left(\dfrac{0.42\ in.}{0.3}\right)^{-0.107} = 0.9646378536\ [rotating\ system]$
$k_c = 0.59\ [pure\ torsion]$
$S_e = 0.5(68\ kpsi)(0.8825694743\ )(0.9646378536)(0.59) = 17.07828006\ ksi$

$K_f = 1 + q(K_t - 1)$

$$\sqrt{a} = 0.246 - (3.08)(10^{-3})(68\ kpsi) + (1.51)(10^{-5})(68\ kpsi)^2 - (2.67)(10^{-8})(68\ kpsi)^3 = 0.0979870656$$

$r = 0.2\ in.$

$$q = \frac{1}{1 + \frac{0.0979870656}{\sqrt{0.2\ in.}}} = 0.8202733918$$

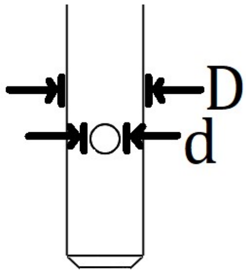


Figure 18. *Applicable D and d Values for Transmitting Rod Stress Concentration Factor*

$$\frac{d}{D} = \frac{0.2\ in.}{0.51\ in.} = 0.392156827$$

$$K_{ts,B} = 2.6\ [Figure\ A - 15 - 10]$$

$$K_{fs} = 1 + (0.8202733918)(2.6 - 1) = 2.312437427$$

$$\tau_{max} = 15.04351081\ kpsi$$
$$\tau_{min} = 0$$
$$\sigma'_m = \sqrt{\left(K_f \sigma_{m,bending} + K_f \sigma_{m,axial}\right)^2 + 3\left(K_{fs}\tau_m\right)^2} = \sqrt{3}K_f \frac{\tau_{max} + \tau_{min}}{2}$$
$$= 30.12657938\ kpsi$$
$$\sigma'_a = \sqrt{\left(K_f \sigma_{a,bending} + K_f \frac{\sigma_{a,axial}}{0.85}\right)^2 + 3\left(K_{fs}\tau_a\right)^2} = \sqrt{3}K_f \left|\frac{\tau_{max} - \tau_{min}}{2}\right|$$
$$= 30.12657938\ kpsi$$

ASME Elliptic:

$$\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2 = \frac{1}{{n_f}^2}$$

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2}} = \sqrt{\frac{1}{\left(\frac{30.12657938\ kpsi}{17.07828006\ kpsi}\right)^2 + \left(\frac{30.12657938\ kpsi}{57\ kpsi}\right)^2}} = 0.5430334488$$

The part has finite life in a fluctuating load according to the ASME Elliptic.

$$(\sigma'_{rev})^{eqv} = \frac{\sigma'_a}{1 - \frac{\sigma'_m}{S_{ut}}} = \frac{30.12657938\ kpsi}{1 - \frac{30.12657938\ kpsi}{68\ kpsi}} = 54.09089975\ kpsi$$

$$f = 0.9\ [Figure\ 6-18]$$

$$a = \frac{(fS_{ut})^2}{S_e} = \frac{(0.9 * 68\ kpsi)^2}{17.07828006\ ksi} = 219.3101405$$

$$b = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{0.9 * 68\ kpsi}{17.07828006\ kpsi}\right) = -0.184769097$$

$$N = \left(\frac{(\sigma'_{rev})^{eqv}}{a}\right)^{1/b} = \left(\frac{54.09089975\ kpsi}{219.3101405}\right)^{-1/0.184769097} = 1950.918382 = 1950\ cycles$$

Handle:

Yielding:

$$n_y = \frac{S_y}{\sigma} = \frac{57\ kpsi}{2.355588033\ kpsi} = 24.19777957$$

Fatigue:

$$S_e = 0.5S_{ut}k_ak_bk_c$$
$$k_a = aS_{ut}{}^b = (2.70)(68\ kpsi)^{-0.265} = 0.8825694743\ [machined]$$

$$d_e = 0.808\sqrt{ab} = 0.808\sqrt{0.2\ in. * 0.9860\ in.}$$
$$= 0.3588102295\ in.\ [non-rotating, "rectangular"\ cross\ section]$$
$$k_b = \left(\frac{d}{0.3}\right)^{-0.107} = \left(\frac{0.3588102295\ in.}{0.3}\right)^{-0.107} = 0.9810280812$$
$$k_c = 0.59\ [pure\ torsion]$$
$$S_e = 0.5(68\ ksi)(0.8825694743\ )(0.9810280812)(0.59) = 17.36845828\ ksi$$
No stress concentrations appear applicable.

$$K_{fs} = 1$$

$$\tau_{max} = 2.355588033\ kpsi$$
$$\tau_{min} = 0$$
$$\sigma'_m = \sqrt{\left(K_f\sigma_{m,bending} + K_f\sigma_{m,axial}\right)^2 + 3\left(K_{fs}\tau_m\right)^2} = \sqrt{3}\,\frac{\tau_{max} + \tau_{min}}{2} = 4.079998155\ kpsi$$
$$\sigma'_a = \sqrt{\left(K_f\sigma_{a,bending} + K_f\frac{\sigma_{a,axial}}{0.85}\right)^2 + 3\left(K_{fs}\tau_a\right)^2} = \sqrt{3}\left|\frac{\tau_{max} - \tau_{min}}{2}\right| = 4.079998155\ kpsi$$

ASME Elliptic:

$$\left(\frac{\sigma_a'}{S_e}\right)^2 + \left(\frac{\sigma_m'}{S_y}\right)^2 = \frac{1}{{n_f}^2}$$

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma_a'}{S_e}\right)^2 + \left(\frac{\sigma_m'}{S_y}\right)^2}} = \sqrt{\frac{1}{\left(\frac{4.079998155\ kpsi}{17.36845828\ kpsi}\right)^2 + \left(\frac{4.079998155\ kpsi}{57\ kpsi}\right)^2}} = 4.072127903$$

The part has infinite life in a fluctuating load according to the ASME Elliptic.

Housing:

Yielding:

$$n_y = \frac{S_y}{\sigma} = \frac{57\ kpsi}{7.220203379\ kpsi} = 7.894514463$$

Fatigue:

$$S_e = 0.5 S_{ut} k_a k_b k_c$$
$$k_a = a{S_{ut}}^b = (2.70)(68\ kpsi)^{-0.265} = 0.8825694743\ [machined]$$

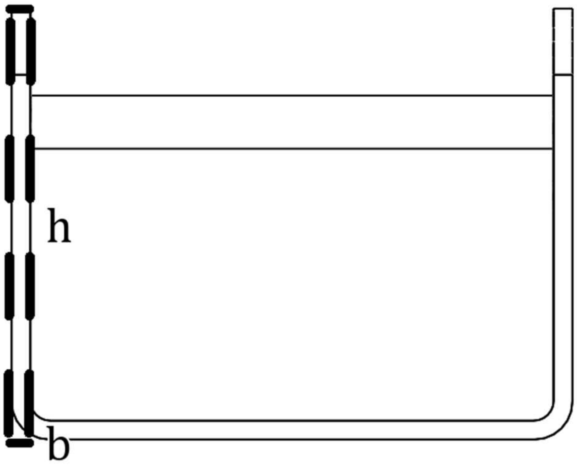


Figure 19. *Cross-Sectional Area of Housing*

$$d_e = 0.808\sqrt{hb} = 0.808\sqrt{(4.3125\ in.)(0.1880\ in.)}$$
$$= 0.7275365888\ in.\ [non-rotating, "rectangular"\ cross\ section]$$
$$k_b = \left(\frac{d_e}{0.3}\right)^{-0.107} = \left(\frac{0.7275365888\ in.}{0.3}\right)^{-0.107} = 0.90956451$$
$$k_c = 0.85\ [pure\ axial]$$

$$S_e = 0.5(68\ kpsi)(0.8825694743\ )(0.90956451)(0.85) = 23.19958688\ kpsi$$

$$K_f = 1 + q(K_t - 1)$$

$$\sqrt{a} = 0.246 - (3.08)(10^{-3})(68\ kpsi) + (1.51)(10^{-5})(68\ kpsi)^2 - (2.67)(10^{-8})(68\ kpsi)^3 = 0.0979870656$$

$$r = 0.2345\ in.$$

$$q = \frac{1}{1 + \frac{0.0979870656}{\sqrt{0.2345\ in.}}} = 0.8317064737$$

$$\frac{d}{w} = \frac{0.469\ in.}{4.3125\ in.} = 0.1087536232$$
$$K_t = 2.7\ [Figure\ A - 15 - 1]$$

$$K_f = 1 + (0.8317064737)(2.7 - 1) = 2.413901005$$

$$\sigma_{max} = 7.220203379\ kpsi$$
$$\sigma_{min} = 0$$
$$\sigma_m = K_f \frac{\sigma_{max} + \sigma_{min}}{2} = 17.42885619\ kpsi$$
$$\sigma_a = K_f \left|\frac{\sigma_{max} - \sigma_{min}}{2}\right| = 17.42885619\ kpsi$$

ASME Elliptic:

$$\left(\frac{\sigma_a'}{S_e}\right)^2 + \left(\frac{\sigma_m'}{S_y}\right)^2 = \frac{1}{{n_f}^2}$$

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma_a'}{S_e}\right)^2 + \left(\frac{\sigma_m'}{S_y}\right)^2}} = \sqrt{\frac{1}{\left(\frac{17.42885619\ kpsi}{23.19958688\ kpsi}\right)^2 + \left(\frac{17.42885619\ kpsi}{57\ kpsi}\right)^2}} = 1.232894474$$

The part has infinite life in a fluctuating load according to the ASME Elliptic.

## Appendix D: Computational Report

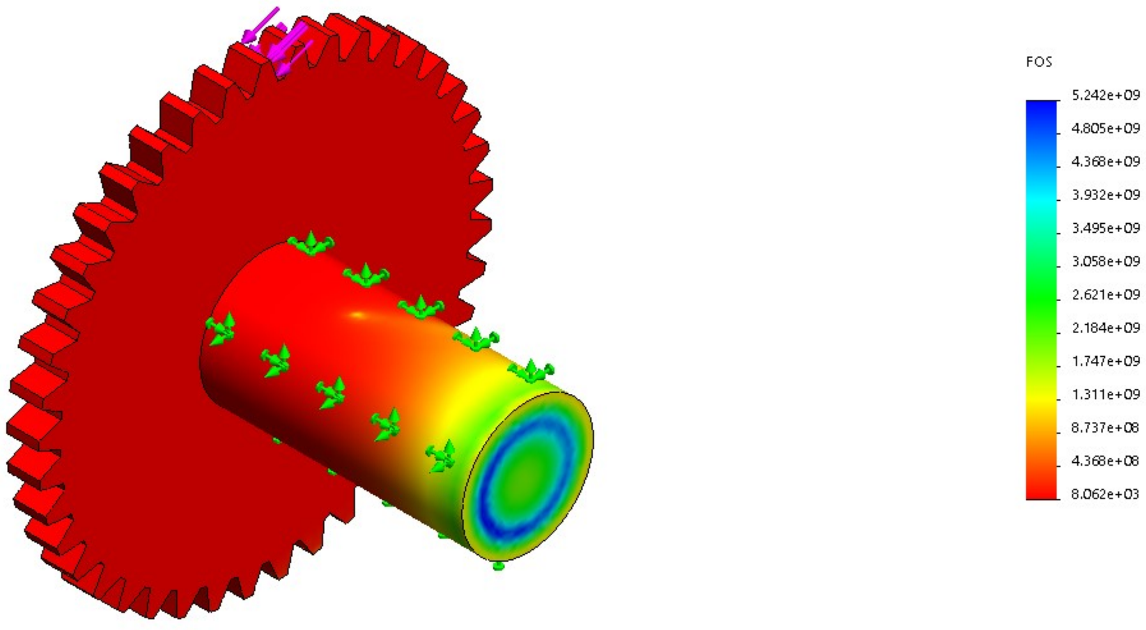


Figure 20. *Gear and Gear Shaft Von Mises Factor of Safety FEA*

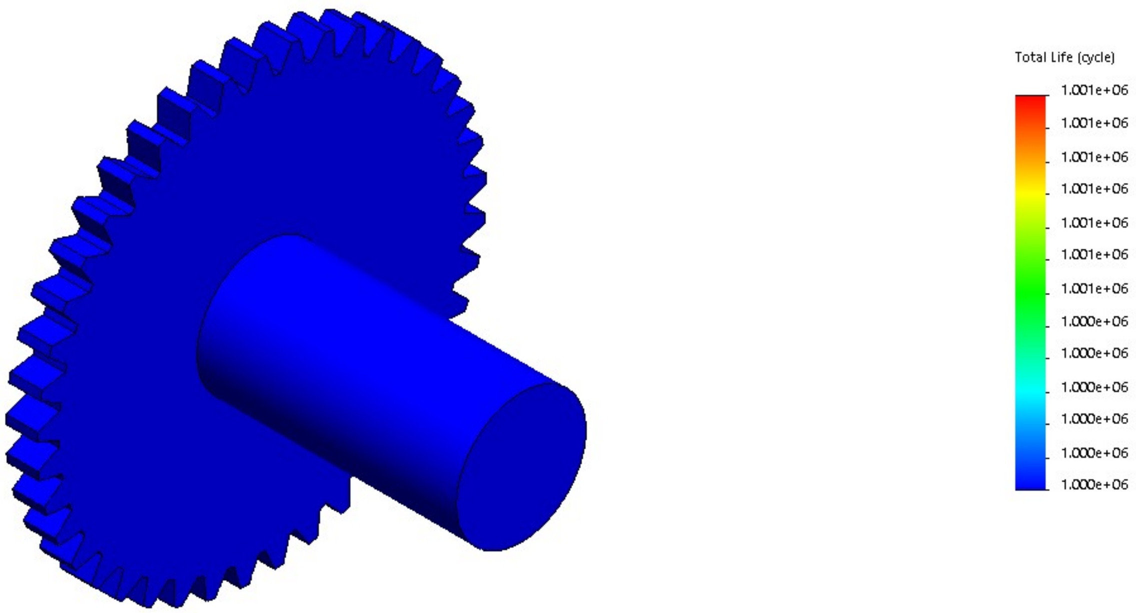


Figure 21. *Gear and Gear Shaft Fatigue Total Life FEA*

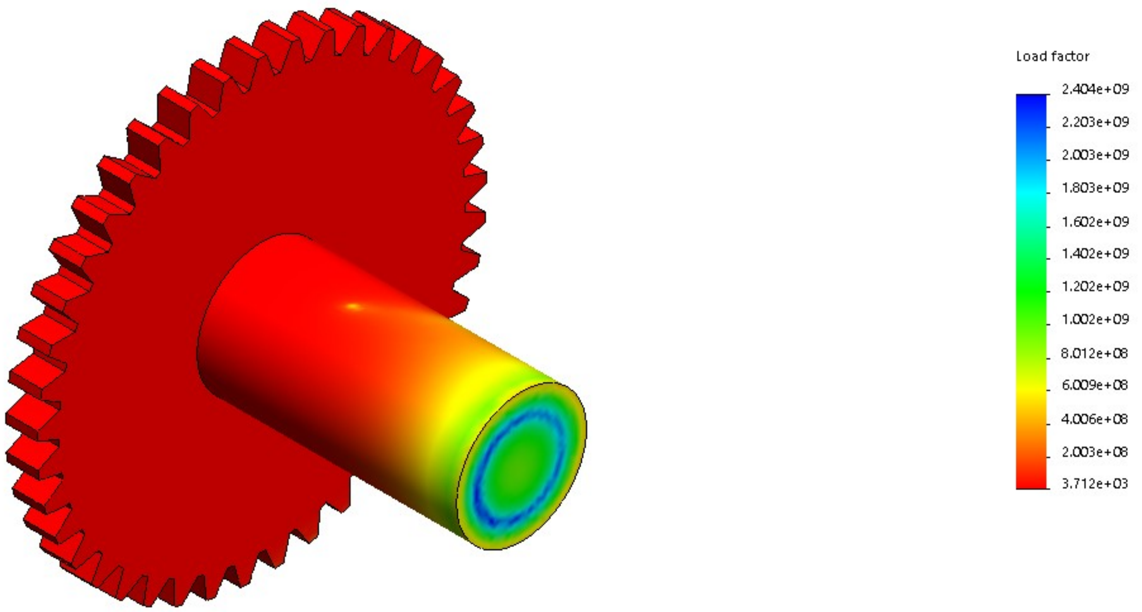


Figure 22. *Gear and Gear Shaft Fatigue Load Factor (Factor of Safety) FEA*

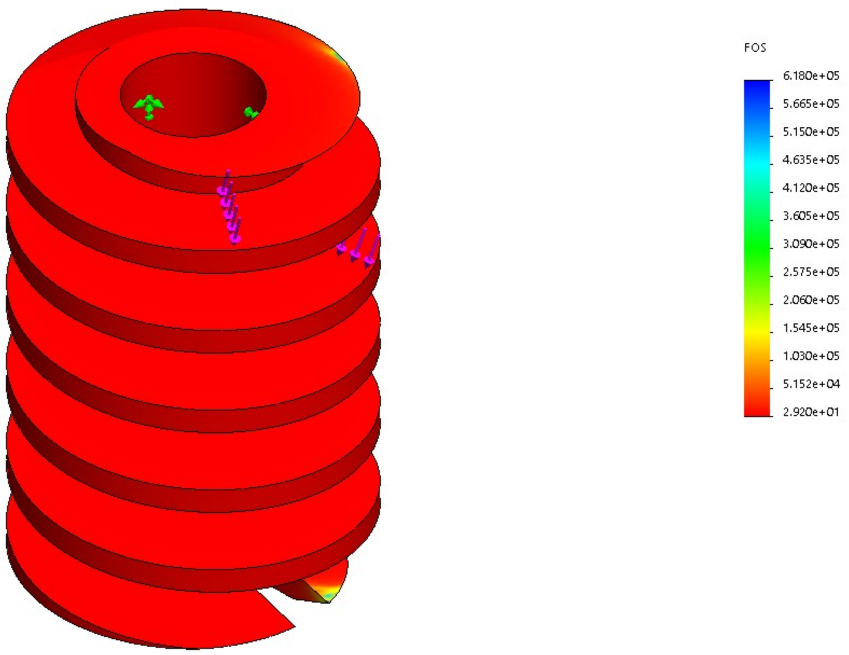


Figure 23. *Worm Von Mises Factor of Safety FEA*

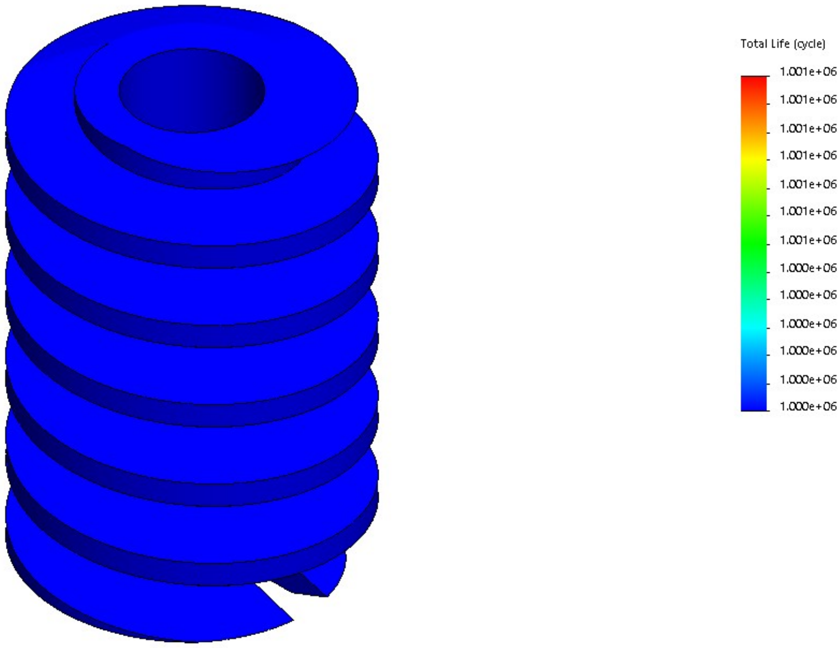


Figure 24. *Worm Fatigue Total Life FEA*

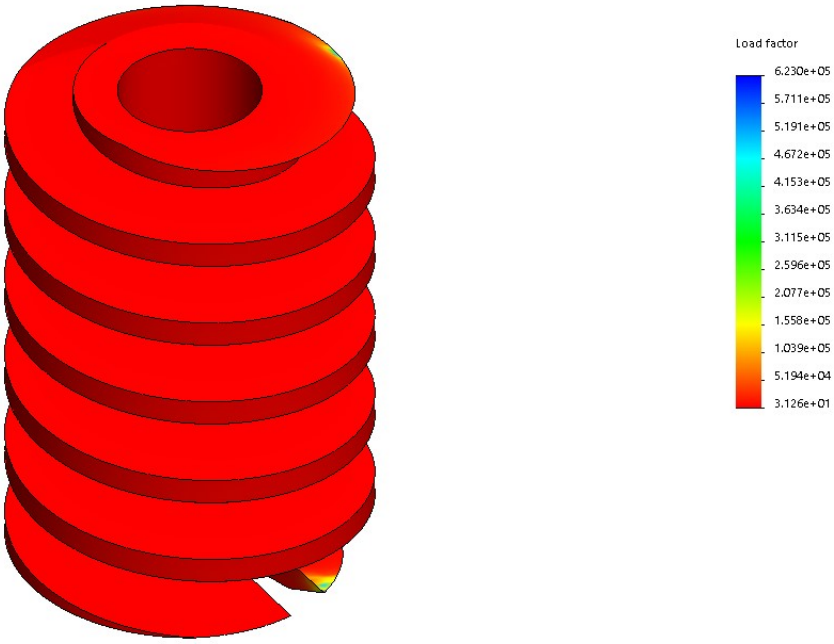


Figure 25. *Worm Fatigue Load Factor (Factor of Safety) FEA*

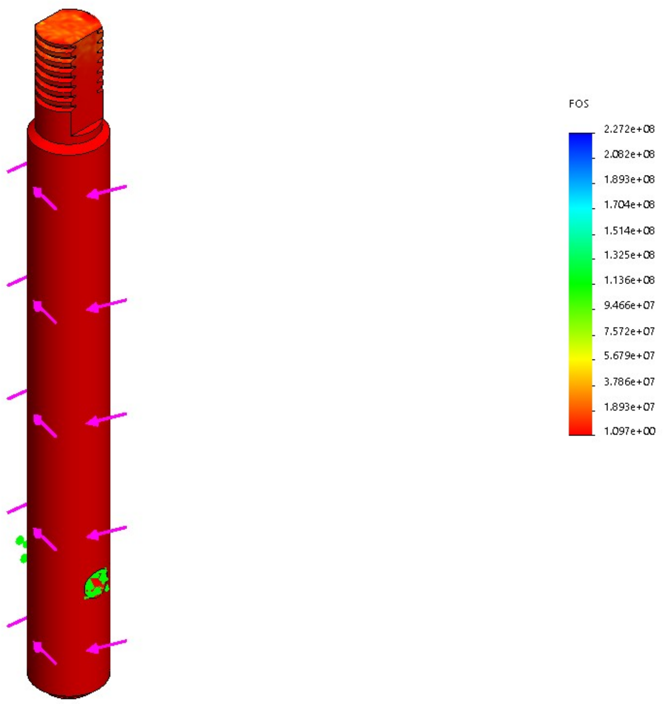


Figure 26. *Transmitting Rod Von Mises Factor of Safety FEA*

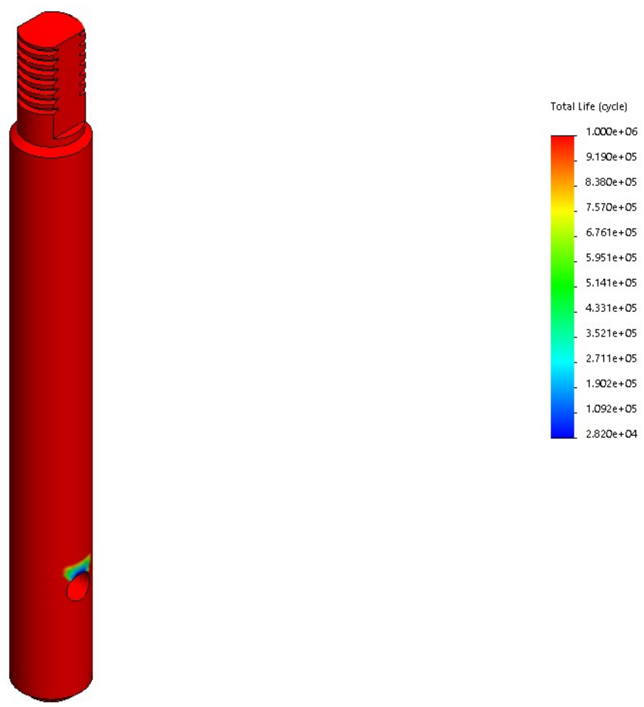


Figure 27. *Transmitting Rod Fatigue Total Life FEA*

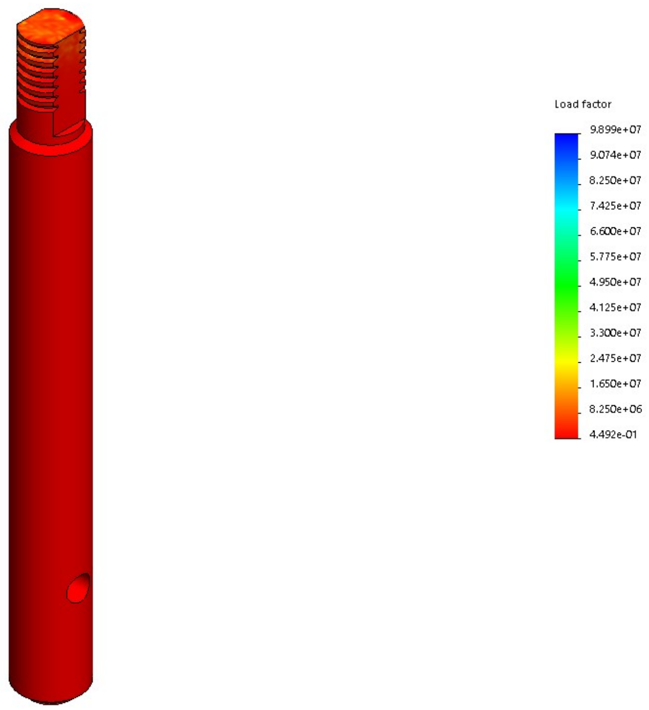


Figure 28. *Transmitting Rod Fatigue Load Factor (Factor of Safety) FEA*

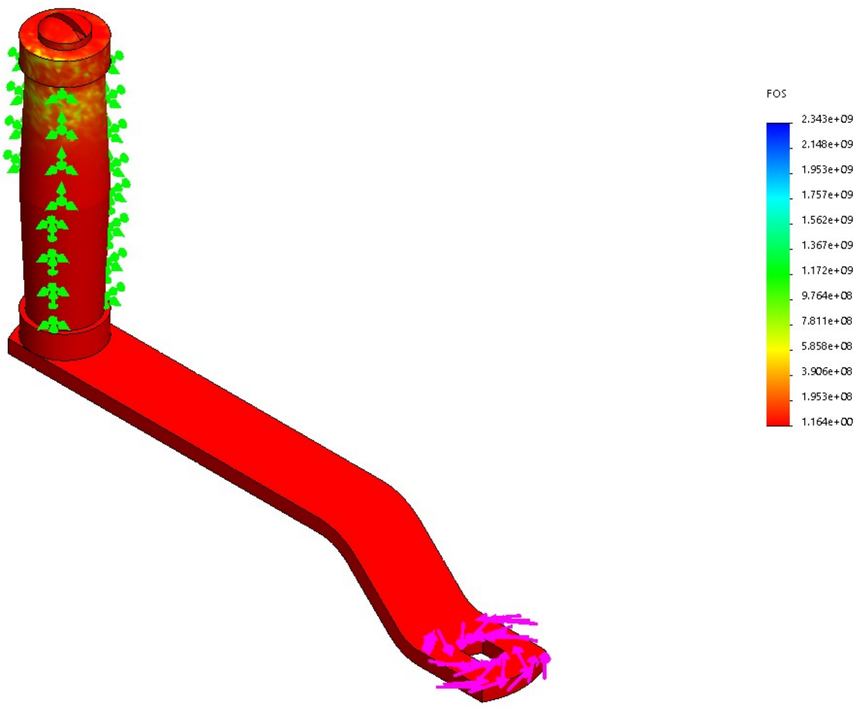


Figure 29: *Handle Von Mises Factor of Safety FEA*

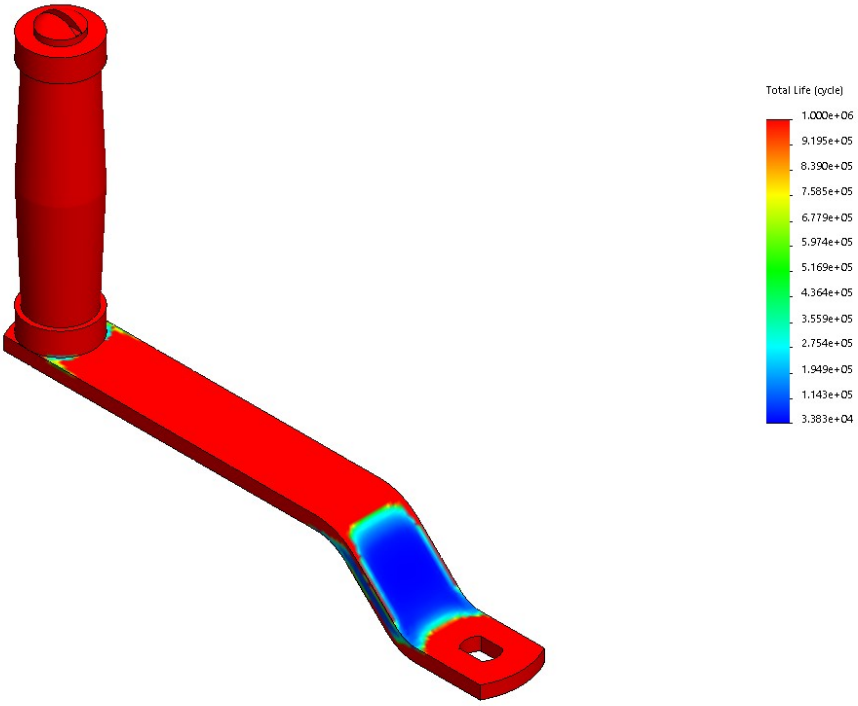


Figure 30. *Handle Fatigue Total Life FEA*

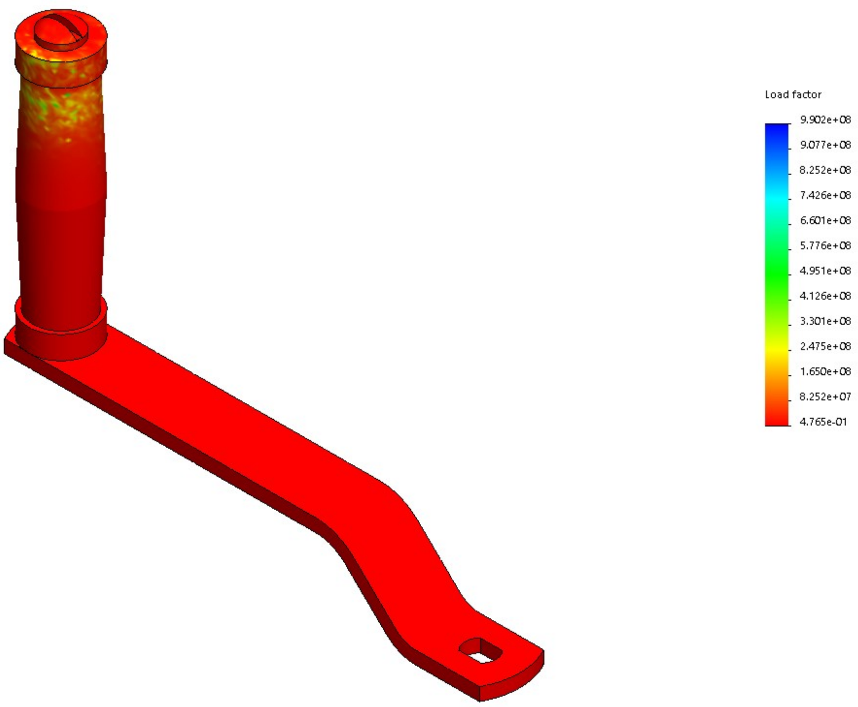


Figure 31: *Handle Fatigue Load Factor (Factor of Safety) FEA*

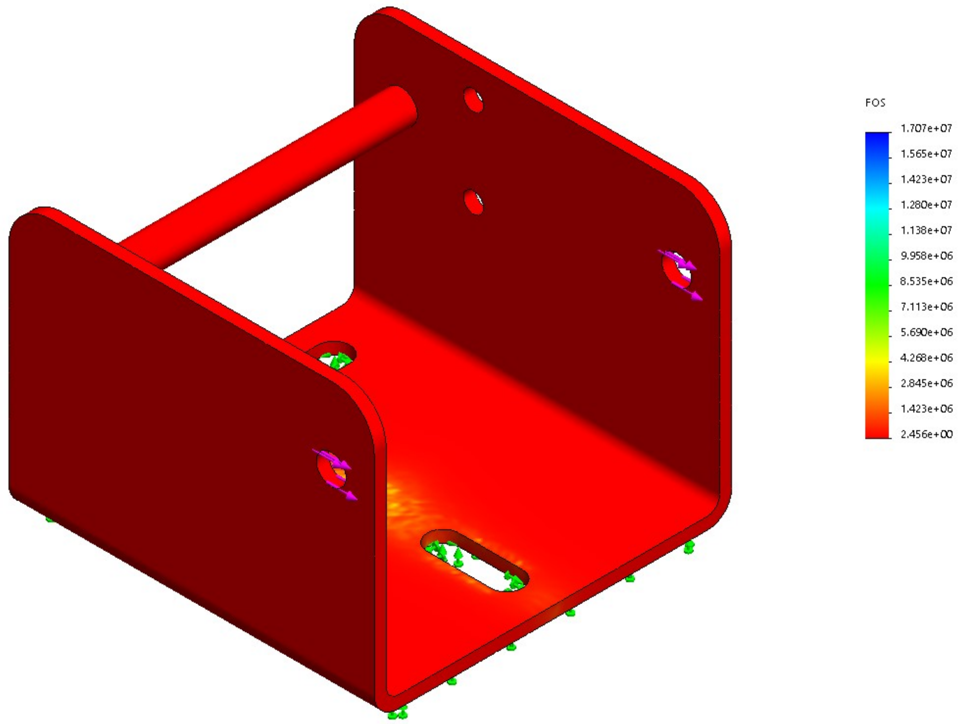


Figure 32: *Housing Von Mises Factor of Safety FEA*

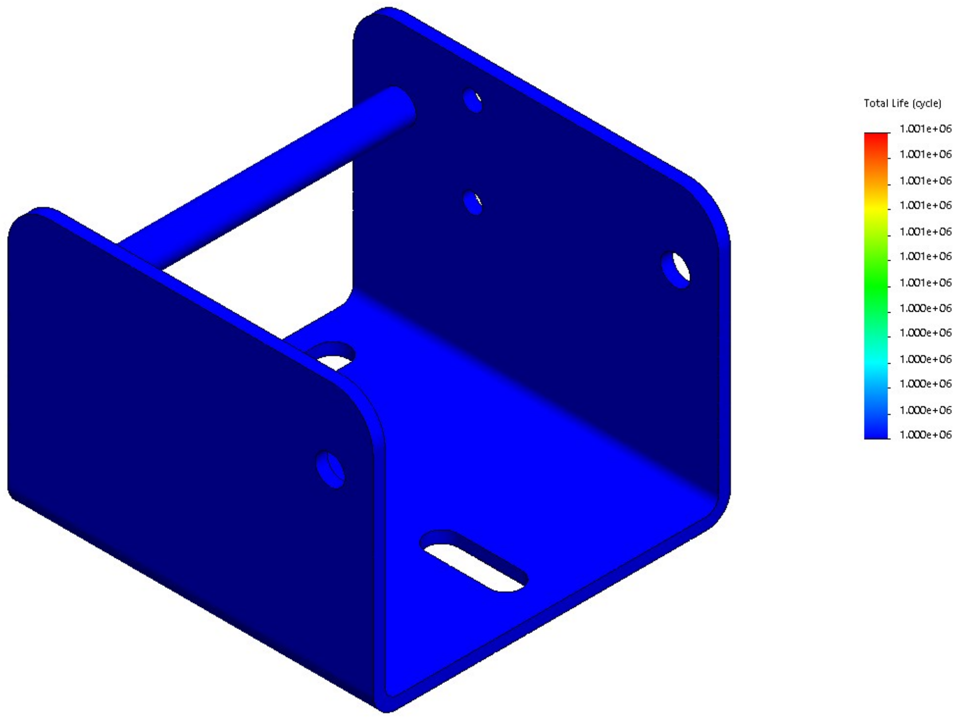


Figure 33: *Housing Fatigue Total Life FEA*

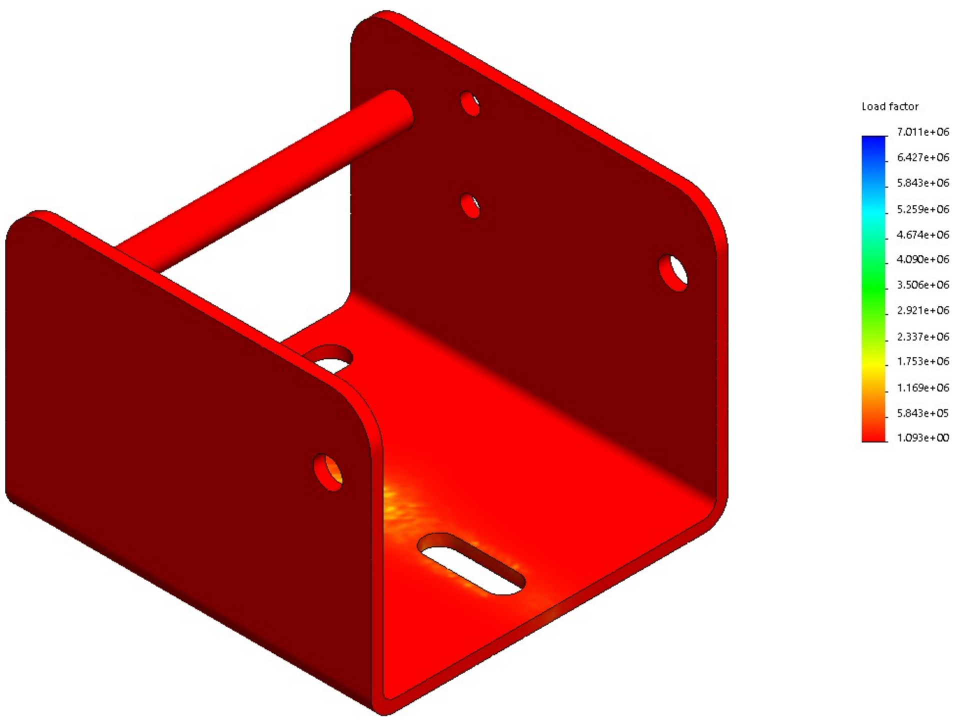


Figure 34: *Housing Fatigue Load Factor (Factor of Safety) FEA*

# Appendix E: Drawings

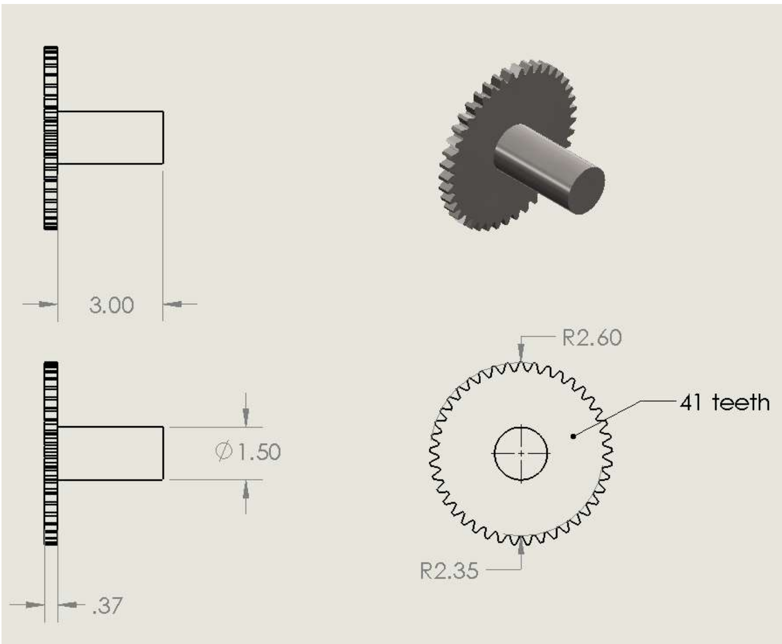


Figure 35: *Drawing of Gear and Gear Shaft (Dimensioned in Inches)*

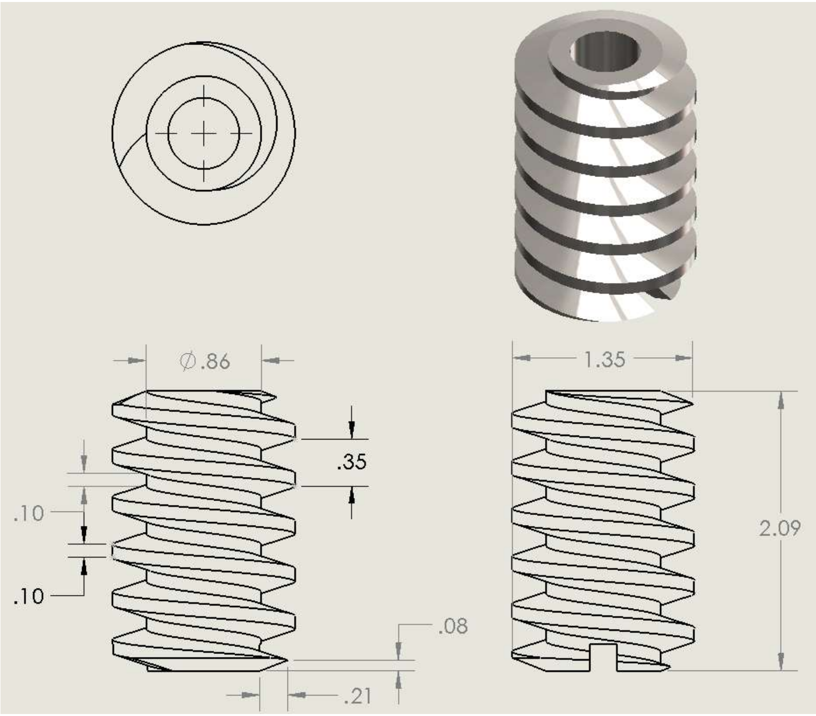

Figure 36: *Drawing of Worm (Dimensioned in Inches)*

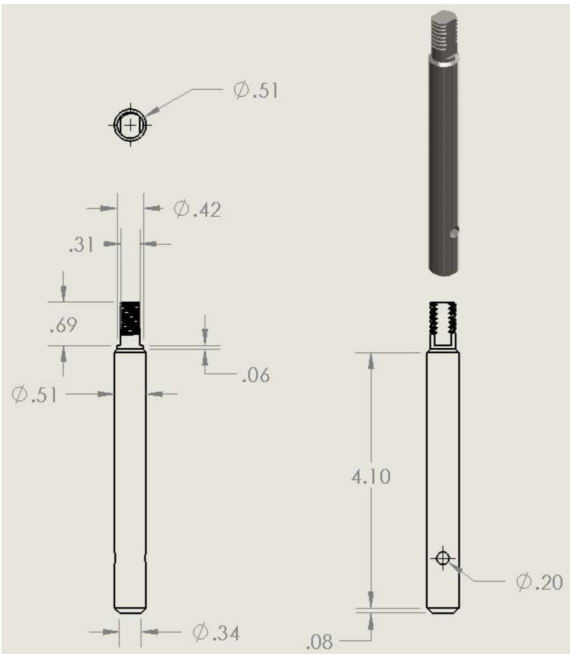

Figure 37: *Drawing of Transmitting Rod (Dimensioned in Inches)*

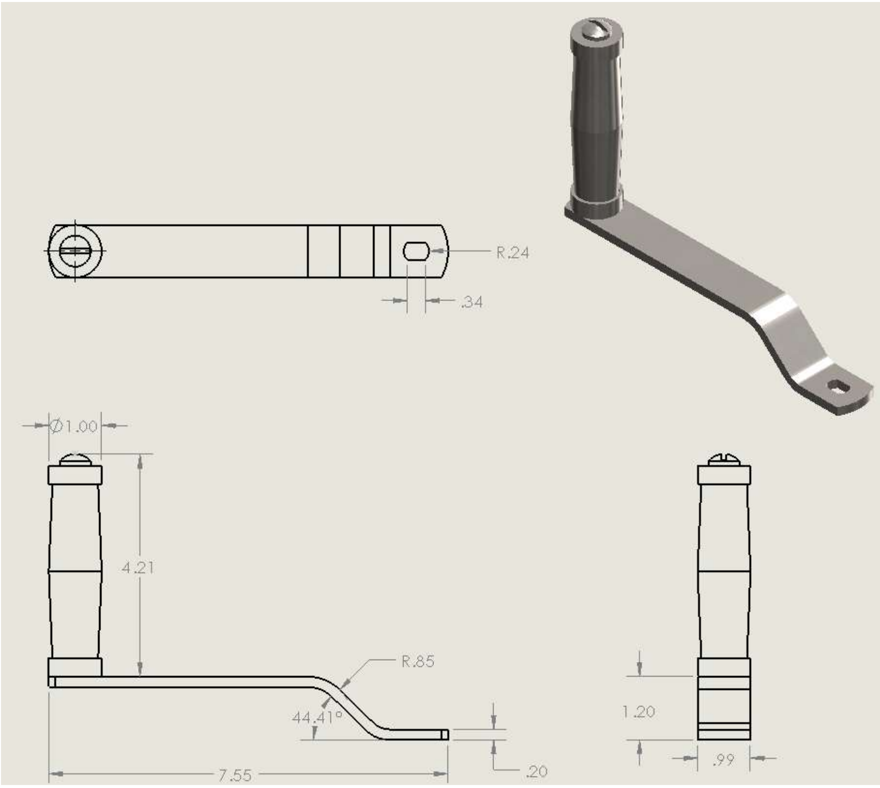

Figure 38: *Drawing of Handle (Dimensioned in Inches)*

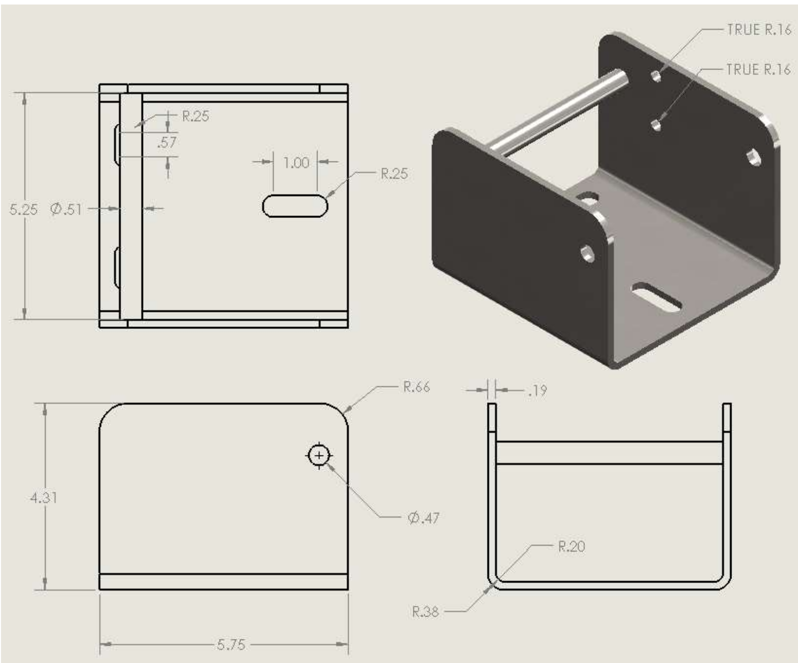

Figure 39: *Drawing of Housing (Dimensioned in Inches)*

# Appendix F: Reference Tables

**Figure 6–18**

Fatigue strength fraction, $f$, of $S_{ut}$ at $10^3$ cycles for $S_e = S'_e = 0.5S_{ut}$ at $10^6$ cycles.

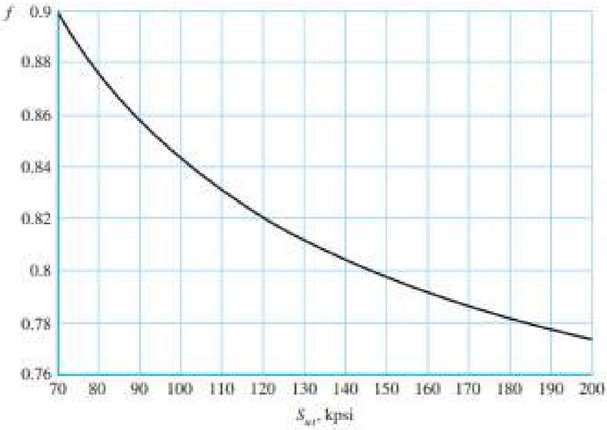


Figure 40: *Figure 6-18 from Shigley's Mechanical Engineering Design (10th ed.)*

**Figure A–15–1**

Bar in tension or simple compression with a transverse hole. $\sigma_0 = F/A$, where $A = (w - d)t$ and $t$ is the thickness.

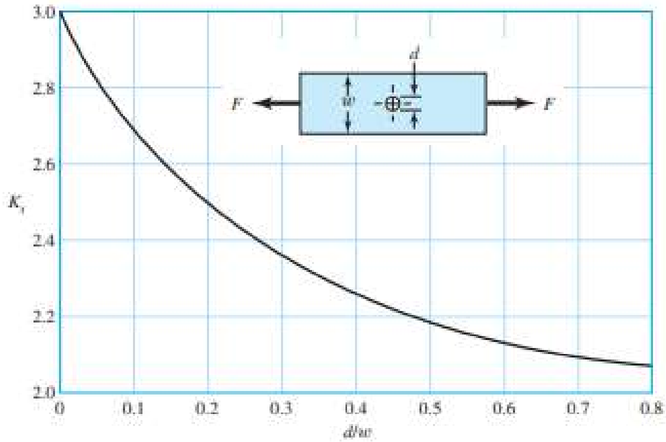


Figure 41: *Figure A-15-1 from Shigley's Mechanical Engineering Design (10th ed.)*

**Figure A–15–6**

Rectangular filleted bar in bending. $\sigma_0 = Mc/I$, where $c = d/2$, $I = td^3/12$, $t$ is the thickness.

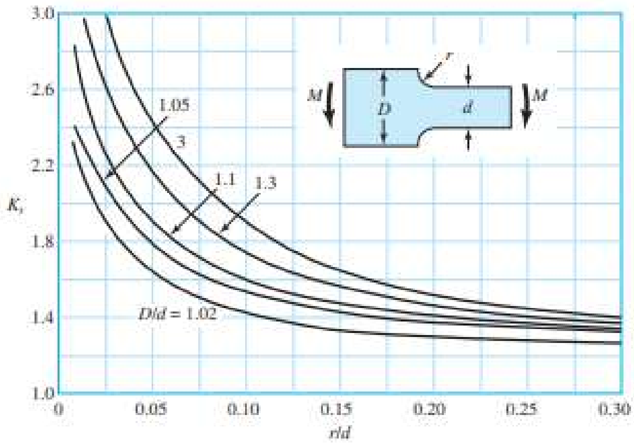


Figure 42: *Figure A-15-6 from Shigley's Mechanical Engineering Design (10th ed.)*

**Figure A–15–10**

Round shaft in torsion with transverse hole.

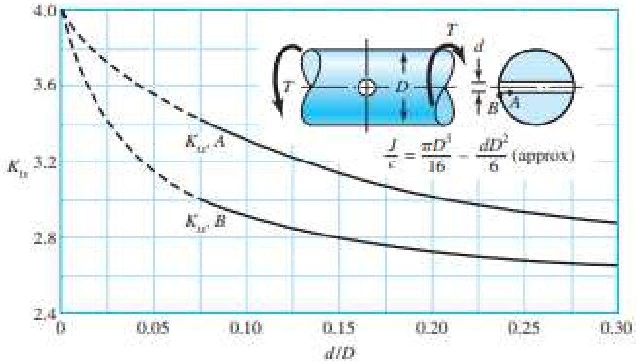


Figure 43: *Figure A-15-10 from Shigley's Mechanical Engineering Design (10th ed.)*

**Table A–20**

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolled (HR) and Cold-Drawn (CD) Steels [The strengths listed are estimated ASTM minimum values in the size range 18 to 32 mm ($\frac{3}{4}$ to $1\frac{1}{4}$ in). These strengths are suitable for use with the design factor defined in Sec. 1–10, provided the materials conform to ASTM A6 or A568 requirements or are required in the purchase specifications. Remember that a numbering system is not a specification.] *Source:* 1986 SAE Handbook, p. 2.15.

| 1<br>UNS No. | 2<br>SAE and/or AISI No. | 3<br>Process-ing | 4<br>Tensile Strength, MPa (kpsi) | 5<br>Yield Strength, MPa (kpsi) | 6<br>Elongation in 2 in, % | 7<br>Reduction in Area, % | 8<br>Brinell Hardness |
|---|---|---|---|---|---|---|---|
| G10060 | 1006 | HR | 300 (43) | 170 (24) | 30 | 55 | 86 |
| | | CD | 330 (48) | 280 (41) | 20 | 45 | 95 |
| G10100 | 1010 | HR | 320 (47) | 180 (26) | 28 | 50 | 95 |
| | | CD | 370 (53) | 300 (44) | 20 | 40 | 105 |
| G10150 | 1015 | HR | 340 (50) | 190 (27.5) | 28 | 50 | 101 |
| | | CD | 390 (56) | 320 (47) | 18 | 40 | 111 |
| G10180 | 1018 | HR | 400 (58) | 220 (32) | 25 | 50 | 116 |
| | | CD | 440 (64) | 370 (54) | 15 | 40 | 126 |
| G10200 | 1020 | HR | 380 (55) | 210 (30) | 25 | 50 | 111 |
| | | CD | 470 (68) | 390 (57) | 15 | 40 | 131 |
| G10300 | 1030 | HR | 470 (68) | 260 (37.5) | 20 | 42 | 137 |
| | | CD | 520 (76) | 440 (64) | 12 | 35 | 149 |
| G10350 | 1035 | HR | 500 (72) | 270 (39.5) | 18 | 40 | 143 |
| | | CD | 550 (80) | 460 (67) | 12 | 35 | 163 |
| G10400 | 1040 | HR | 520 (76) | 290 (42) | 18 | 40 | 149 |
| | | CD | 590 (85) | 490 (71) | 12 | 35 | 170 |
| G10450 | 1045 | HR | 570 (82) | 310 (45) | 16 | 40 | 163 |
| | | CD | 630 (91) | 530 (77) | 12 | 35 | 179 |
| G10500 | 1050 | HR | 620 (90) | 340 (49.5) | 15 | 35 | 179 |
| | | CD | 690 (100) | 580 (84) | 10 | 30 | 197 |
| G10600 | 1060 | HR | 680 (98) | 370 (54) | 12 | 30 | 201 |
| G10800 | 1080 | HR | 770 (112) | 420 (61.5) | 10 | 25 | 229 |
| G10950 | 1095 | HR | 830 (120) | 460 (66) | 10 | 25 | 248 |

Figure 44: *Figure A-20 from Shigley's Mechanical Engineering Design (10th ed.)*

# Appendix G: FMEA Table

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | FAILURE MODE AND EFFECTS ANALYSIS | | | | | |
| Item: | Haul Master Worm Hear Hand Winch | | | | Responsibility: | | Pierce Grimm, Matthew Ramsey | | | | | |
| Model: | V1.1 | | | | Prepared by: | | Matthew Ramsey | | | | | |
| Core Team: | Pierce Grimm, Havinda Herath, Daniel Laughlin, Chris Phu, Matthew Ramsey | | | | | | | | | | | |
| Component & Function | Potential Failure Mode | Potential Effect(s) of | S e v | C l a s s | Potential Cause(s)/ | O c c u r | Current Process | D e t e c | R P N | Recommended Action(s) | Responsibility and Target Completion Date | |
| Handle, provide torque force to move components, using a max of 50lbf | Unintended function | Operation impared, poor appearance | 7 | YS | Surface damage. Inproper use | 7 | Detection | 2 | 98 | Possible use of stronger material, repositioning/ redesign of connection to next parts | Daniel Laughlin | |
| Housing, hold all pieces together and attachment to allow to hold a max of 2000lbf pulling | Unintended function | Unstable, Poor appearance, Noise | 8 | YS | Inproper mounting, damage sustained | 2 | Detection | 2 | 32 | Repositioning of components of part | Matthew Ramsey | |
| Handle Bolt, ensure connection between wormgear and handle at a max pull of 2000 lbf | Partial/ Overfunction Over time | Inoperative, erratic operation, noise | 8 | YS | Inproper tightening and possible loosening | 7 | Detection | 3 | 168 | Use of stronger materials, possible use of other confibolt configurations | Kavinda Herath | |
| Worm Gear, provide connection to drum gear while providing a gear reduction of 1:40 of lbf with the drum gear. | Partial/ Overfunction Over time | Rough, inoperative, noise | 6 | YS | Friction from connection between drum gear | 1 | Prevention | 2 | 12 | Metal finish and lubrication | Chris Phu | |
| Drum Gear, provide force to stap maintining a force while providing a gear reduction with the worm gear of 1:40of 2000 lbf | Partial/ Overfunction Over time | Customer Dissatisfaction. Rough, inoperative, noise | 7 | YS | Friction from connection between worm gear. Exceed Recommended Loading. Exceed Life Cycle. | 1 | Prevention | 2 | 14 | Possible use of stronger material. Increase Width of Gear Tooth Face. | Pierce Grimm | |

Figure 45: *FMEA Table*

# Appendix H: Bonus

Provided Monte Carlo code was modified to check different possible diameters for the transmitting rod in the trailer winch device. Horsepower and RPM were made to remain constant. Original diameter of transmitting rod was 0.51 in.

Considering the FEA analyses for the transmitting rod, decreasing the diameter to the 0.45 in. as suggested by the program (the diameter with the greatest number of iterations) would not be advantageous. A smaller diameter would directly result in receiving greater shear stress, and would cause the component to have an even lower factor of safety in both yielding and fatigue.

```
%%Below is the code provided with a change in the T to 16044 kpsi
% The same materail was used and the T is take from the stress that the
% transmitting rod received.
% it was determined that the diameter could be smaller.  It is currently
% .51 in but the code determines that it could be between .4517 to .4509

%% Demo of Monte Carlo Simulation for Uncertainty and Sensitivy Analysis in MED
%For this demo see the problem in  5-38 in Shigley 10 ed.

% A 1020 CD Steel shaft is to transmit 20 HP at 1750 RPM. Determine the
% minimum diameter for the shaft to provide a minimum factor of safety of 3
% based on the maximum sheer stress theory. (Static failure)


%% Notes before we begin:
 % Use the command 'rand' to generate a random number from a uniform
 % distribution. Use the command 'randn' to draw from a normal
 % distribution. Other distributions are possible but we will stick with
 % these for now. To draw a single number from a uniform distribution
 %between two parameters [a,b], use 'r=a+(b-a)*rand()' or 'randi' if you
 %want an integer. To draw a single number from a normal distribution with
 %a mean and standard deviation [mean,std] use 'r=std*randn()+mean'.
 clc


%% Define the variables you will hold as constant and uncertain
FS=3; %factor of safety is a requirement.
iter= 100000; %10,000 iterations for a simple analysis should be sufficient.

HPMean=20; HPstd=2.5; %I choose Normal distribution for power
SpeedMin=1700; SpeedMax=1800; %I choose Uniform distibution for speed
SyMean=57*10^3; Systd=3; % I choose normal dist for yield strength


%% Deterministic Solution
% It is helpful to be sure you have the deterministic part correct before
% trying to explore uncertainty.

% Variables from the problem as given
HP=20; %horse power
N=1750; %rpm
Sy= 57*10^3; %psi

% Calculate Torque
T=15044*HP/N; %Equation for relation speed, torque and HP
TauMax=Sy/(2*FS); %Equation for Max Shear stress = (16*T/pi)*d^-3

d=((32*15044*HP*FS)/(N*pi*Sy))^(1/3);
disp('Deterministic min. diameter is =')
disp(d)

%% Area for Monte Carlo Simulation
%Going to find the resulting distribution of minimum diameters based on the
%uncertain variabels and use those dist to determine the liklihood of
%meeting the Factor of saftery requirment with the diameter from the
%deterministic part.

tic; %keep track of your comptation speed

d_Prob=zeros(iter,1);% Pre-allocate to RAM for speed
% for i=1:iter
%     % in each loop, we are going to build a diameter distribution
%     %Begin by refreshing the random number generator
%     rng('shuffle');
%     %Now draw from the distribution to find each value for this iteration
%     HP=HPstd*randn()+HPMean;
%     Speed=SpeedMin+(SpeedMax-SpeedMin)*rand();
%     Sy=Systd*randn()+SyMean;
%
%     %Perform Calculations as in deterministic part.
%     T=63025*HP/Speed;
%     TauMax=Sy/(2*FS);
%     d_Prob(i,1)=((32*63025*HP*FS)/(Speed*pi*Sy))^(1/3);
% end

%% Quicker way (that is a little less random)
% generate the entire vector of possible first
    HP=HPstd.*randn(iter,1)+HPMean;
    Speed=SpeedMin+(SpeedMax-SpeedMin).*rand(iter,1);
    Sy=Systd.*randn(iter,1)+SyMean;

%Perform Calculations as in deterministic part.
     T=63025*HP./Speed;
     TauMax=Sy./(2*FS);
     d_Prob=((32*15044.*HP.*FS)./(Speed.*pi.*Sy)).^(1/3);


%% Display Results

D_sort=sort(d_Prob); %Sort for calculations later

disp('Mean of Min. Diameter is =')
disp(mean(D_sort));
disp('Std of Min. Diameter is =')
disp(std(D_sort))

histogram(D_sort)

toc
```

Figure 46. *Modified Monte Carlo Code for Design for Uncertainty*

```
Deterministic min. diameter is =
    0.4517

Mean of Min. Diameter is =
    0.4509

Std of Min. Diameter is =
    0.0191

Elapsed time is 9.168550 seconds.
```

Figure 47: *Results from Running Modified Monte Carlo Code*

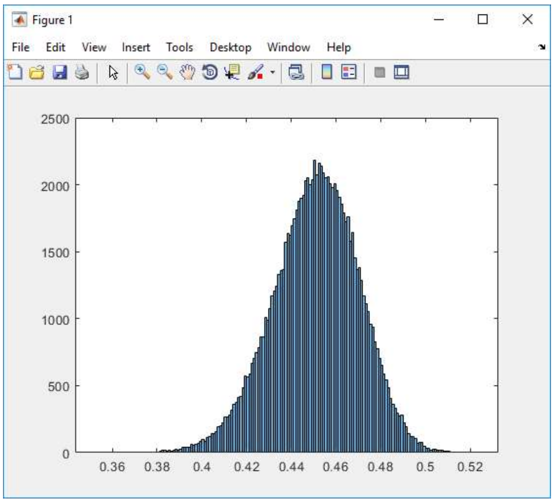


Figure 48: *Normal Distribution of Possible Transmitting Rod Diameters, Iterations v. Diameter*