Study of Mechanical Jack

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MAE 493: Honors Thesis

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Abstract:

In this study a scissor jack was structurally analyzed and compared to a FEA model to study the structure of the jack. The system was simplified to a 2D system, and one of the truss members was analyzed for yielding, fatigue, and buckling. The FEA and analytical static safety factors were 0.613 and 0.614 which were nearly identical leading to the conclusion that the analysis was correct and that the members did fail under the maximum loading conditions. The member would buckle as well under that loading as the critical load, which was $2939\ lbf$, was way below the compressive load acting on the member, which was $16111\ lbf$.

Introduction

To perform maintenance, build, or repair work on an object, it is necessary to fix it at a position where the location of work is easily accessible. It would be harder to move or lift heavier objects like cars than a smaller and lighter object like a book. For such cases, a mechanical jack is used to lift the load of the heavy object so appropriate work can be done on the it. Depending on the lifting load, an appropriate jack would be chosen. Most jacks can be operated by a single operator unless it requires a significant amount of force to lift an object up, like buildings. There are several types of jacks including hydraulic and mechanical jacks employed in industry and everyday life. Hydraulic jacks have a higher loading capacity compared to mechanical jacks as the fluid does some work along with the operator [4]. Some mechanical jacks can be powered electrically.

Mechanical jacks are structural devices which do not necessarily require any power. They operate on a principal of mechanical advantage. Mechanical advantage is the ratio of the force required to perform the necessary work to the applied work [1]. A common mechanical jack design is a scissor jack. They have four pinned member joints and one lead screw which provides the scissor jack its mechanical advantage. Scissor jacks are used for car repair and maintenance and they are an easy tool to lift up objects such as four wheeled vehicles without the use of heavy-duty jacks and can be carried easily to remote locations. Scissor jacks are easy to carry, provide the necessary mechanical advantage and get the job done in case there is a flat tire in the middle of the road. A graphic depicting the jack in action is shown below.

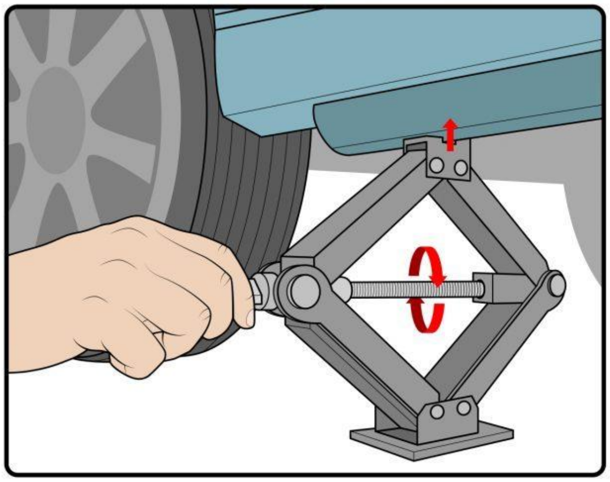

Figure 1. Scissor Jack Operation to Lift Vehicle [6]

This report will exclusively focus on an existing scissor jack and its structural analysis as scissor jacks are purely mechanical. The report includes static, dynamic, and FEM studies on a simplified version of one of its members.

During this study, many assumptions were made in order to correctly use the analytical models without a lot of complexity. All assumptions will be stated within each section of the report. The main one is simplifying the scissor jack to a 2D 4-link truss system with a lead screw in between as shown in the figure below:

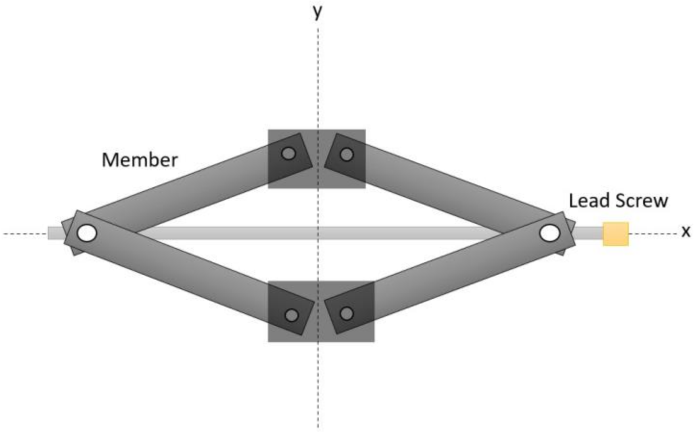

Figure 2. 2D Simplification of Scissor Jack System into a 4-Link System

Objectives

The main objective of this study is to understand and analyze the one of the primary components of a chosen scissor jack. These components include the members and the lead screw, but studies were only conducted on the main member. All sub-objectives are mentioned below:

1. Choose a scissor jack with two-location contact on the lead screw.
2. Perform static analysis using a conservative theory.
3. Perform fatigue analysis using a conservative theory.
4. Create a simplified CAD model of the chosen scissor jack member using SolidWorks.
5. Perform Finite Element Analysis (FEA) using ANSYS on the member model and compare the static safety factor results to the analysis done in part 2.
6. Check for buckling in the member.

Scissor Jack

Choosing a simple scissor jack that can be easily modeled was important. There were many kinds of scissor jacks varying in loading capacity, the number of contact points on the lead screw, and method of operation. The loading capacity depended on the material and design of the jack. Some needed a hooked arm to provide more torque for a faster operational time and some required a wrench or power drill with a hex head bit like the one in figure 3.

The design that was analyzed was a two-location contact scissor jack. This meant that the lead screw opened and closed the jack at both ends. This set-up would provide double the torque to open or close it which in turn reduces operational time. When it comes to manufacturing, it is better to manufacture a one-ended lead screw as it does not require introducing threads from both ends. A one-end lead screw would be easier to manufacture while keeping the structural integrity similar to the two-end version. Pictures of the base scissor jack utilized in this study are included below:

a b c

Figure 3. Chosen Scissor Jack. a) front closed, b) top, c) front opened

This design has a loading capacity of 6500 lbf and is made of mild carbon steel. The supplier was not clear about the material, so material properties were based on AISI 1020 steel as they stated the members were made of low carbon steel.

Static Analysis

To re-iterate, the chosen scissor jack was simplified to the 2D version of the jack member. Since the members are designed to primarily bear compressive loads and not any bending, any bending on the structure is neglected and that leaves the 2D body in figure 4 to be analyzed.

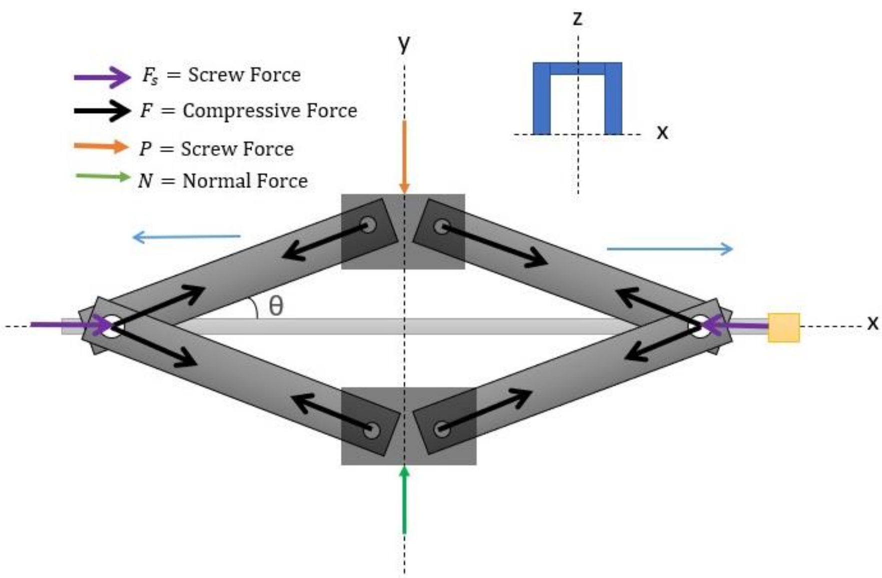

Figure 4. 2D Simplification of the Scissor Jack and Loading on the Members

The free body diagram for members is shown in figure 5. Since the weight of the jack was only 13.5 lbf, which is significantly smaller compared to its loading capacity, it was not taken into consideration. This just makes the normal force equal to the load on top. Due to symmetry about the y and x axis, the compressive force in all the members is of the same magnitude. There is a screw force exerted ono the members from the lead screw at both ends. The top and bottom load bearing, and mount plates are ignored, although they do add a significant bear up a significant amount of load.

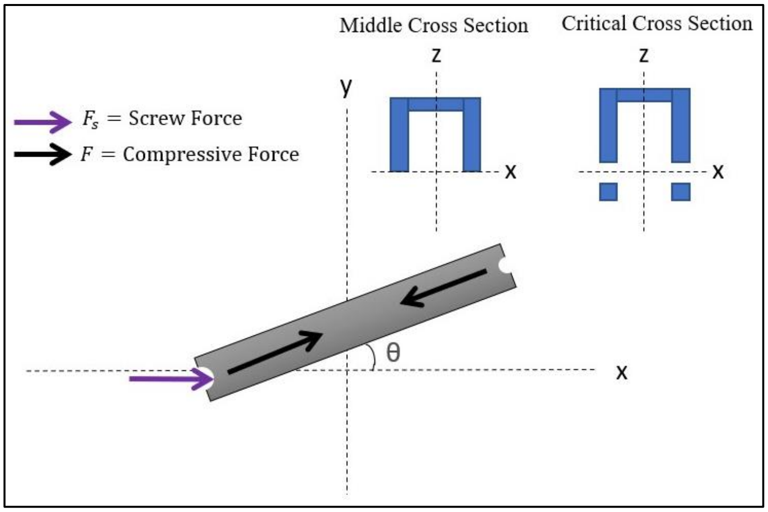

Figure 5. Free Body Diagram of a Single Member and its General Middle and Critical Cross-Sectional Areas

The angle θ was measured at a point where the jack touched the bottom of a Nissan Sentra. This angle was measured to be 11.6°. The way that it was measured is illustrate in the drawing below:

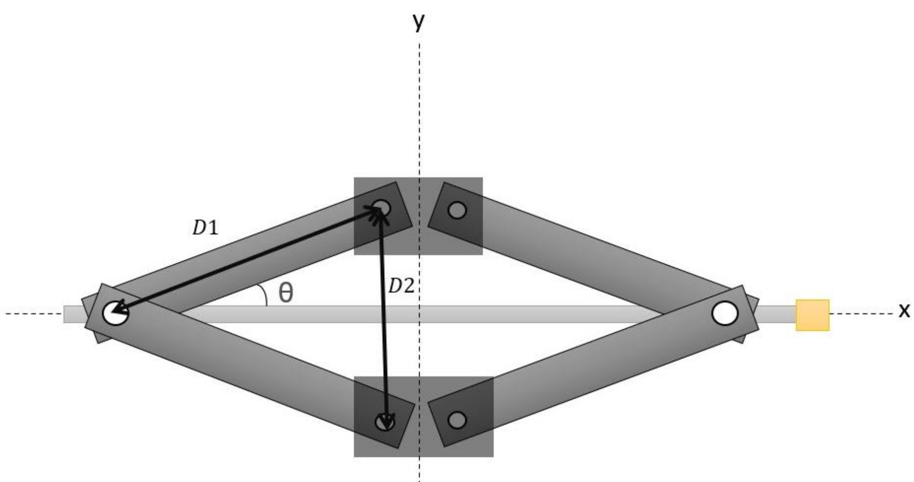

Figure 6

$$\theta = \tan^{-1}\frac{D_2/2}{D_1} \quad \textbf{(1)}$$

To re-iterate, this mode is simplified 2D version of the actual jack member. Since the members are designed to primarily bear compressive loads and not any bending, any bending on the structure is neglected and that leaves this 2D figure in figure 4 to be analyzed.

Relevant Dimensions:

- Sy = 57 kpsi (ASIS 1020 [5])
- Sut = 68 kpsi (ASIS 1020 [5])
- E = 297 x 10^3 kpsi
- Thickness = 13/128 in
- Height = 1 7/8 in
- Width = 2 in
- Pin hole diameter = 1 in
- Length from hole to hole = 10.5 in

Given the dimensions of the pin hole, the stress concentration factor was found to be $k_t = 2.08$ as referenced in Shigly [5].

Force Balance on Member:

$$F = N/\left(2sin(\theta)\right) = 16111\ lbf \ (\mathbf{2})$$

$$Fs = 2Fcos(\theta) = 31559\ lbf \ (\mathbf{3})$$

Stress at the Critical Location under Compression of the Member:

$$\sigma_a = F/A = 44.7230\ kpsi \ (\mathbf{4})$$

Static Safety Factor:

$$n_s = S_y/(k_t\ \sigma_a) = 0.61275 \ (\mathbf{5})$$

The Langer model was used to the calculate the static safety factor. This came out to be extremely low. This could be attributed to the manufacturer providing misinformation about the material used and a simplified model of the member as it has ridges, dents, and bends around itself. As of now, this model indicated that the structure would fail under static loading. In real life, some of the loads can be supported by the top and bottom plates and even if there is yielding, it is mostly due to over-simplification.

FEA of Member

CAD:

The member was modeled in SolidWorks to later conduct finite element analysis on. Initially, the member was more accurately modeled as shown in figures 7 and 8. The screw pin side of the member is more accurate compared to the bolt pin side. The actual member is made of bent sheet metal with blanks and ridges. But the CAD model was mainly made of extrusions and cuts. But it was soon discovered it is better to simplify it further.

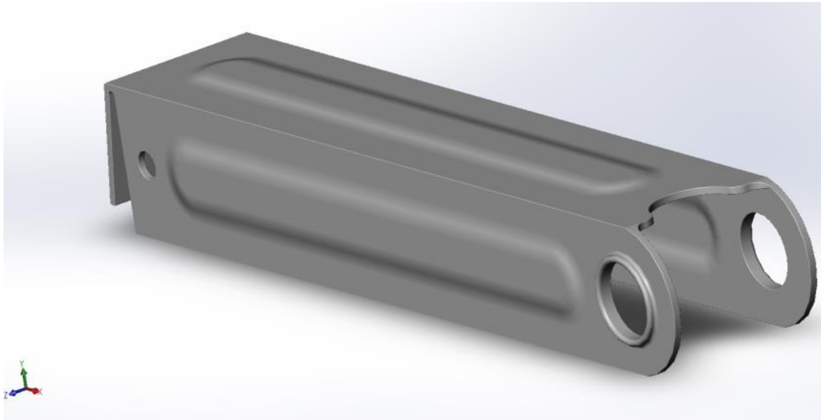

Figure 7. Member CAD Rev 1: Isometric View

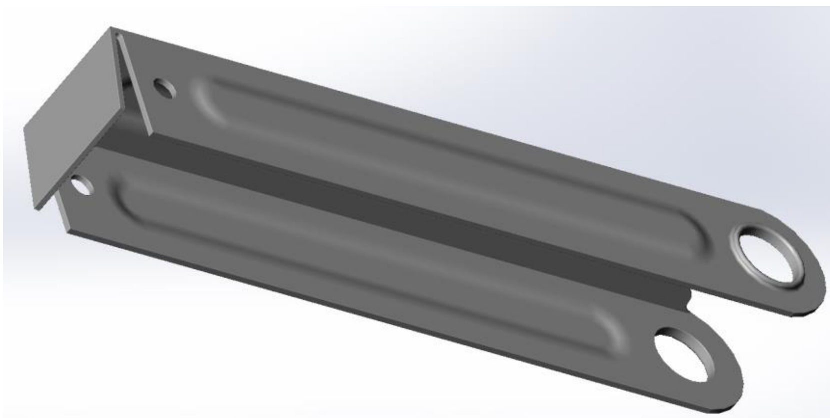

Figure 8. Member CAD Rev 1: Back View

The second revision was modeled as the one shown in figure 9, But it still had the top semi-circle hole. To assess, its significance, a simulation was conducted on the simplified member with and without the hole. The CAD model without the hole is shown in figure 10.

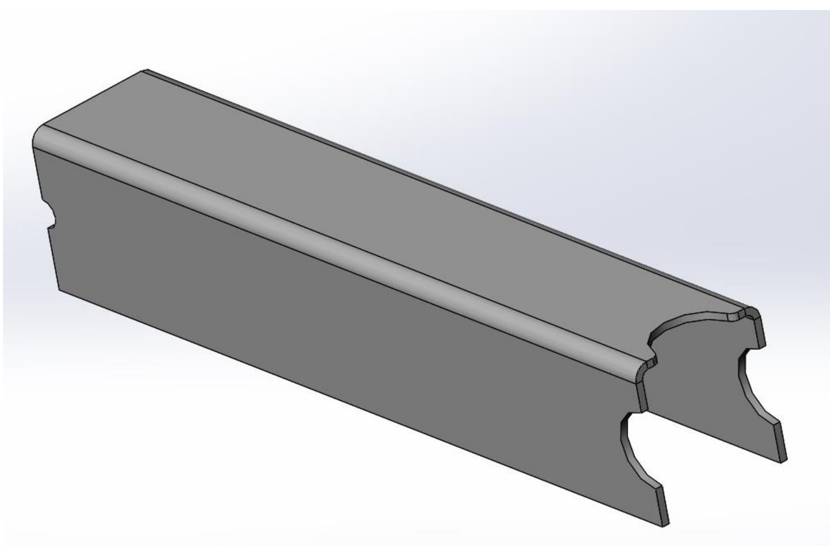

Figure 9. Member CAD Rev 2

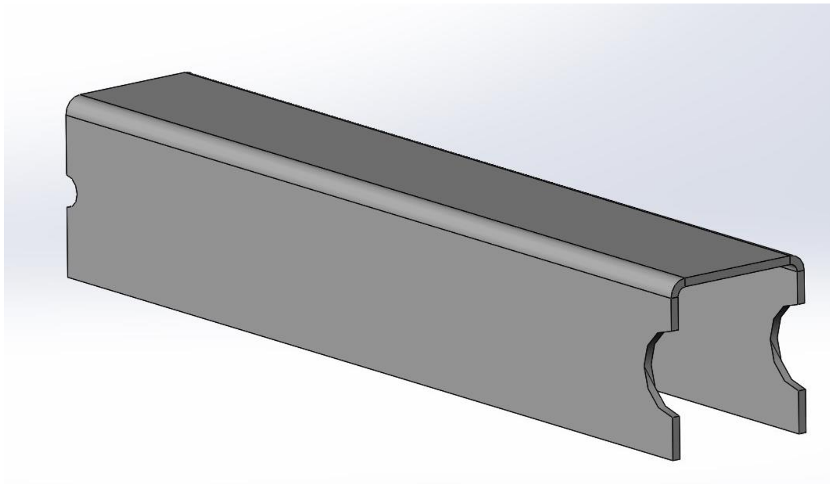

Figure 10. Member CAD Rev 3

Mesh:

A default mesh was used to analyze both members. A default mesh with 7287 nodes was used to compare the models. The percent difference between the static safety factors for the Rev 2 and Rev 3 was 0.0085%. Since it was significantly small, the FEA was conducted on Rev 3.

Table 1. Mesh Convergence Table

Nodes	ns
7287	0.70604
229397	0.68215
333843	0.64599
644255	0.60816
1452761	0.61465

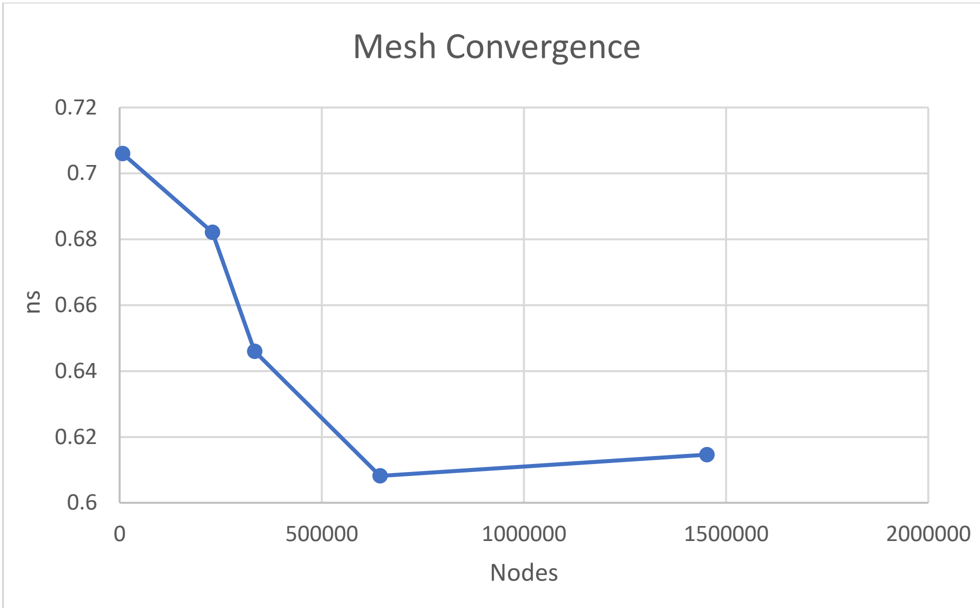

Figure 11. Mesh Convergence Plot

The safety factor was converging when the nodes of a quadratic triangular mesh were converging to 1452761 nodes. This mesh resulted in the same safety factor as the one found analytically.

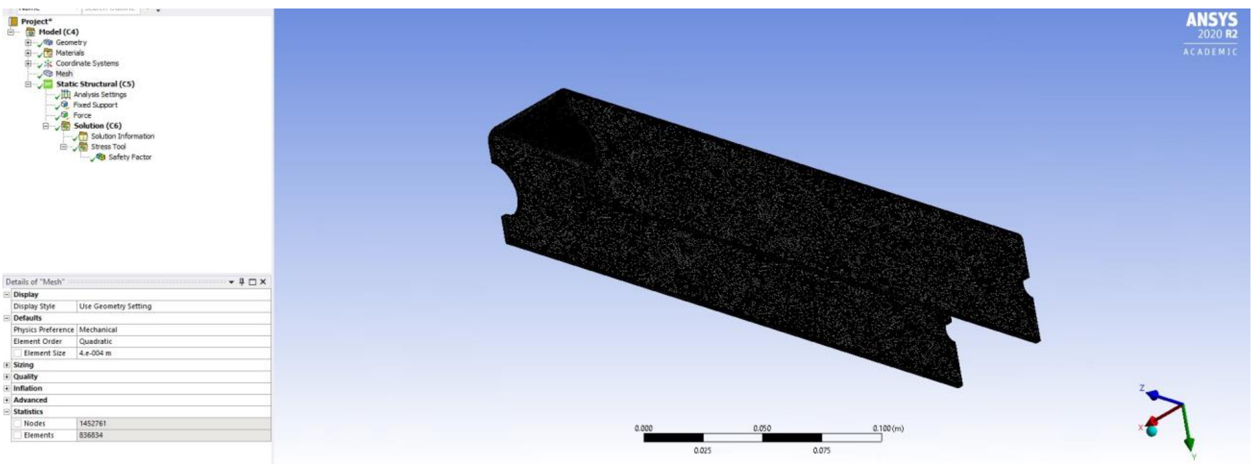

Figure 14. Mesh on the Member

Loading:

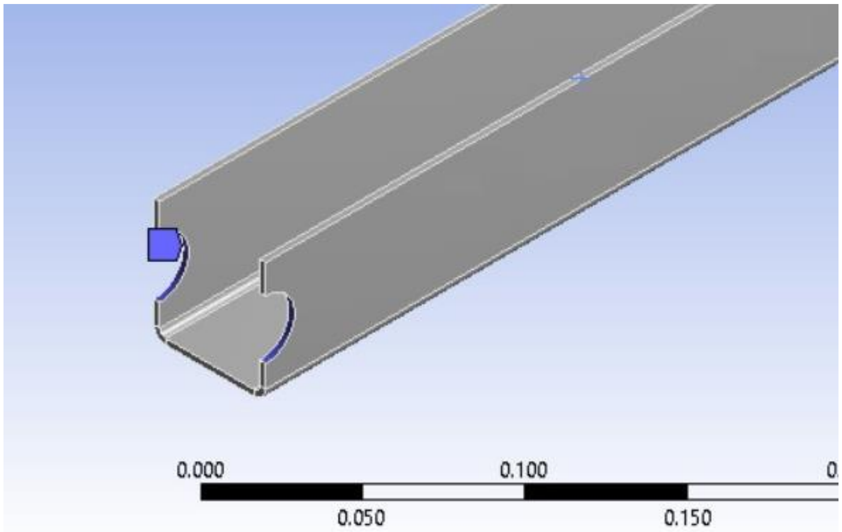

Figure 12. Fixed Support at the Pin Hole on the Member

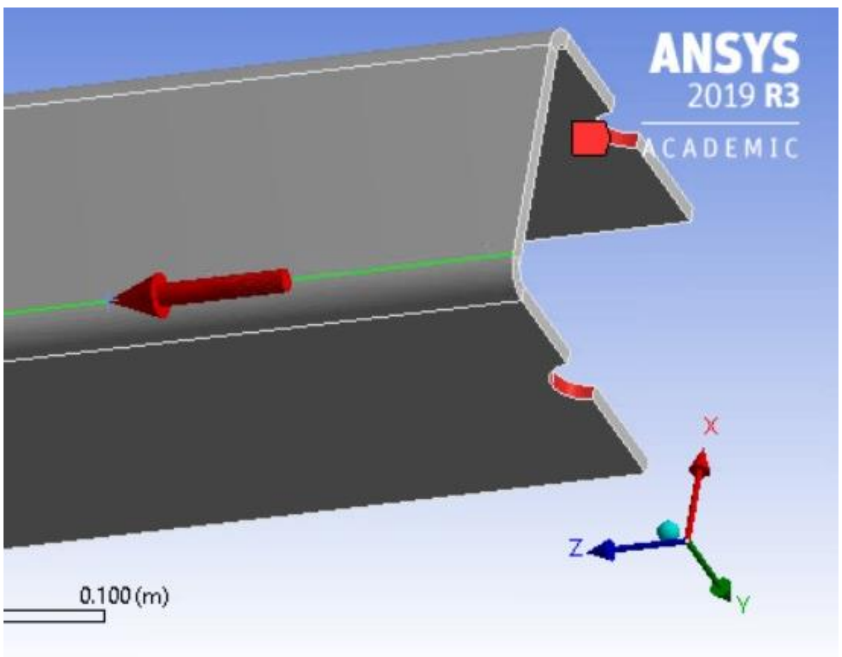

Figure 13. Compressive force F acting on the Bolt Holes on the Member

The pin holes were set as fixed supports as shown in the figure 12 and the compressive force F acted on the bolt holes as shown in figure 13. This set-up was ideal because on the jack, although the members can pivot about the pins, they are still fixed pins relative to the member. But they do move relative to the ground. The compressive force F comes from the loading which acts at the top of the structure, so it makes sense for F to act on the bolt holes at the top of the member and along the member.

Deformation:

Most of the deformation is in the region where the loads are acting. In the actual member, there are gear like teeth which are used to mesh and align the top and bottom members together. These mesh teeth were a little bent upon inspection so this might explain some of that observation. Or else, it was a manufacturing issue. The maximum deformation was 0.0179 in. which is small enough but also keeping in mind the thickness of the sheet metal is about 0.102 in.

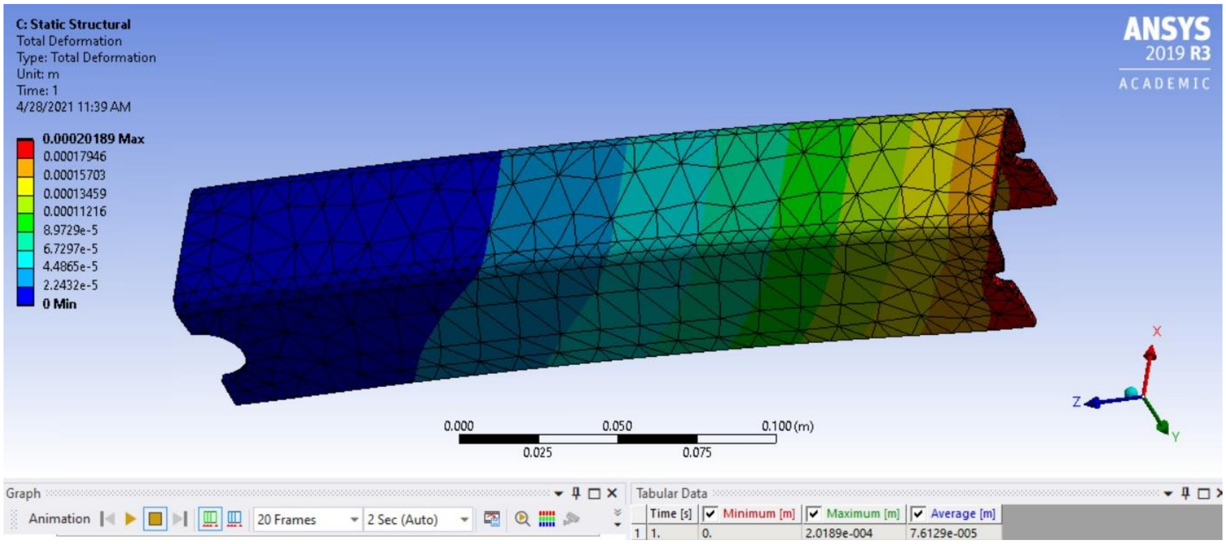

Figure 14. Deformation of the Member when the Mesh has 7287 nodes.

Safety Factor:

The unaveraged minimum safety factor for max shear stress was 0.614 for the optimal mesh. According to this, the member would fail in static loading as well, being consistent with the analytical calculations. One of the major inaccuracies could come from wrong material choice, inaccurate CAD model, and the angle θ being too small. The material was changed from Hot-Rolled 1018 steel to Cold-Drawn 1020 steel but that was not enough. The angle was measured based on height of the bottom of the Nissan Sentra from the ground. This particular jack could potentially not be designed for such low cars. An angle of 45° gave a static safety factor greater than 1 but that is not the initial angle used in this simulation and the previous calculations.

Buckling

Next, buckling of the member was checked. Elastic buckling of a pin-supported long slender column was assumed. The members are not that long but compared to its other dimensions, was a pretty good estimate to use this particular buckling model. For the column to not buckle and stay in stable equilibrium, the axial load acting upon it must be less than the critical load of the column. The least moment of inertia would be about the x axis in the cross-section diagram of figure 14. The middle cross section was taken into account as buckling would only care about the member's general profile. The inertia was calculated using the following formulas written in the tables:

Centroid Relative to y:

$$\bar{y} = \frac{\sum A_i y_i}{\sum A_i} \ \mathbf{(6)}$$

$$I_x = \sum \overline{I_x} + A_i dy_i^2 \ \mathbf{(7)}$$

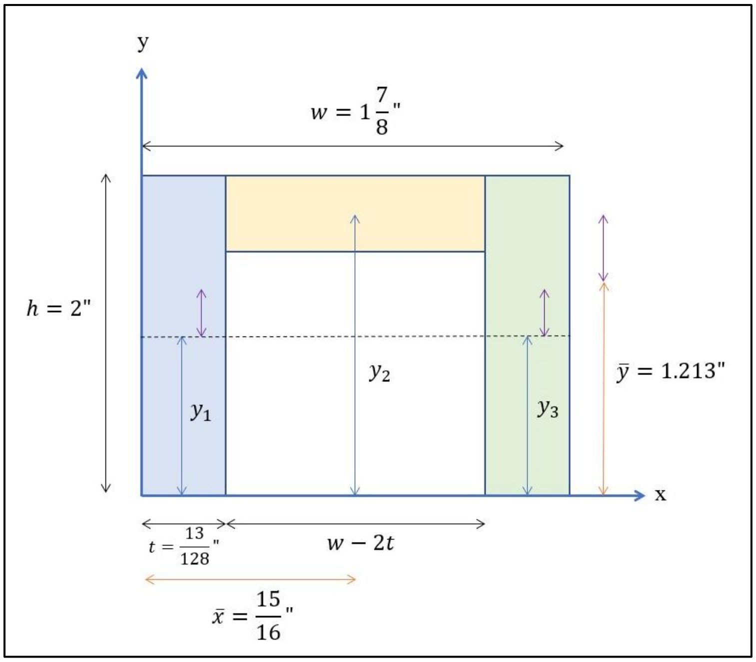

Figure 14. Centroid Diagram of the General Cross Section

Table 2. Centroid Calculation Table

	Ai(in^2)	yi(in)	Ai*yi(in^3)
1	0.203125	1	0.203125
2	0.1698	1.72265625	0.292506695
3	0.203125	1	0.203125
sum	**0.57605**		**0.698756695**
y			1.213014377

Table 3. Inertia Calculation Table

	I(in^4)	dyi = y-yi(in)	Ai*dyi^2(in^4)
1	0.016927	0.213014377	0.009216822
2	0.01415	0.509641873	0.044102925
3	0.016927	0.213014377	0.009216822
sum	**0.048004**		**0.062536569**
Ix			0.11054072

The critical load:

$$P_{cr} = \pi^2 EI/(KL)^2 \ (\mathbf{8})$$

$$P_{cr} = \ 2939 \ lbf$$

Since the compressive force F is more than the critical load ($16111 \ lbf \ < \ 2939 \ lbf$) by whole order of magnitude, the member does seem to buckle at peak loading. This shows that the members will fail under static loading and buckling as well.

Fatigue Analysis for 10^2 Cycles

The fatigue safety factor was calculated for an endurance strength of the member designed to operate at 10^2 but the member seems to fail statically first. The conservative theory, Soderburg, was applied to find the fatigue safety factor. The fatigue safety factor considering all the right extra factors was 0.56058. The structure fails for infinite cycles as well as expected. All constants are given shown in the MATLab code in Appendix A.

Lead Screw Analysis

The lead screw was analyzed following the example problem 8.1 on pg. 411 of *Shigley's* Mechanical Engineering Design (Tenth Edition).

Assumptions:

1. There is no collar.
2. Same frictional coefficients as in example 8.1.
3. Axial nominal normal stress σ is on the x-axis.
4. Thread-root bending stress σ_B is on the y-axis.
5. For σ_B, each thread carries 38% of F as specified in 8.1.
6. Yield strength of the lead screw is similar to the members as they are the same material. The screw probably has extra surface processing for surface durability. Assume $S_y = 120 \ kpsi$.

Given:

$d = 31/64 \ in$ (major diameter, measured)

$p \ = 1/32$ in (screw pitch, measured)

$d_m = d_c = d - p/2$ in (pitch diameter and collar pitch diameter)

$f = f_c = 0.08$ (Frictional coefficients)

$l = 2(0.785)p$ in (lead with respect to both contact point which were each 0.785 in long)

$d_r = d - p$ in (minor diameter)

$F_s = 1611 \ lb_f$ (screw force previously calculated based on maximum loading)

Illustration:

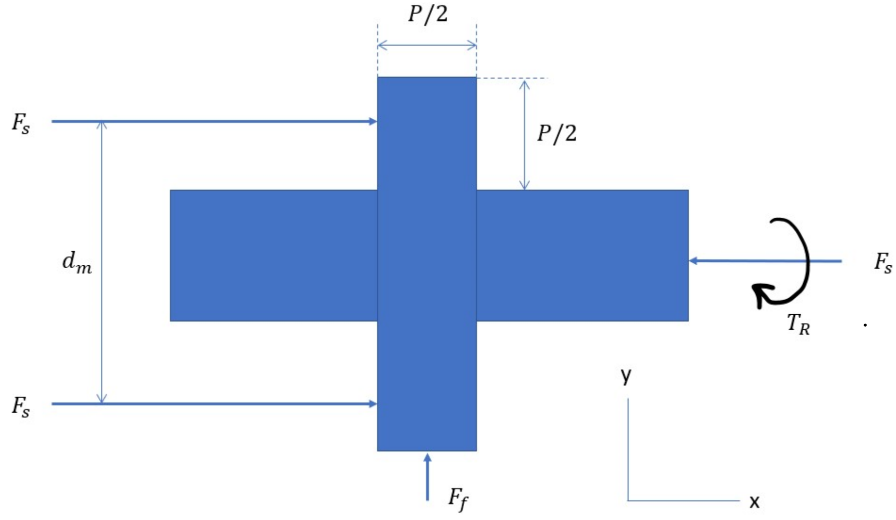

Image 1. Simplified Cross-Sectional Representation of Thread Useful in Finding Shear and Axial Stresses on the Lead Screw (figure inaccurate and not to scale, referencing Figure 8-8 in *Shigley's*) where z axis is into the page.

Torque Required to Turn the Screw Against the Load:

The screw force F_s is responsible for producing the torque required to turn the lead screw and open the jack.

$$T_R = \frac{F_s d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) + \frac{F_s f_c d_c}{2} \ (\mathbf{9})$$

$$T_R = 1432.1 \ lb_f$$

Body Shear Stress due to Torsional Moment:

The torque T_R induces a shear stress onto the lead screw acting on the teeth.

$$\tau_s = \frac{16 T_R}{\pi d_r^3} \ (\mathbf{10})$$

$$\tau_s = -7.0836 \ kpsi$$

Axial Nominal Normal Stress:

According to image 1, the axial nominal normal stress is on the negative x-axis as the screw force F_s is pointing the same direction.

$$\sigma = \frac{4F_s}{\pi d_r^2} \ (\mathbf{11})$$

$$\sigma_x = -195.7 \ kpsi$$

Thread-Root Bending Stress:

As each thread will be at angle by their nature, there is a thread-root bending stress along the y axis.

$$\sigma_B = \frac{6(0.38|F_s|)}{\pi d_r(1)p} \, (\mathbf{12})$$

$$\sigma_y = 1617.5 \; kpsi$$

Principal Stresses:

The first principal stress is just the thread-root bending stress, while the other two principal stresses are calculated from Eq (3—13) in *Shigley's*.

$$\sigma_1 = \sigma_y = 1617.5 \; kpsi$$

$$\sigma_2 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_s^2} = 0.25607 \; kpsi \; (\mathbf{14})$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_s^2} = -195.96 \; kpsi$$

3-D Stresses:

$$\sigma_x = -195.7 \; kpsi \qquad \tau_{xy} = 0$$

$$\sigma_y = 1617.5 \; kpsi \qquad \tau_{yz} = 0$$

$$\sigma_z = 0 \qquad \tau_{xz} = \tau_s == -7.0836 \; kpsi$$

Von Mises Stress:

Using Eq (5—12) in *Shigley's,* the Von Mises Stress is calculated.

$$\sigma' = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} = \mathbf{1723.7} \; kpsi \; (\mathbf{15})$$

As a check, the von Mises stress was calculated:

Using Eq (3—12) in *Shigley's,* the Von Mises Stress is calculated.

$$\sigma'_{check} = \frac{\sqrt{\left(\sigma_x - \sigma_y\right)^2 + \left(\sigma_y\right)^2 + (-\sigma_x)^2 + 6(\tau_s)^2}}{\sqrt{2}} = \mathbf{1723.7} \; kpsi \; (\mathbf{16})$$

Maximum Shear Stress:

Using Eq (3—16) in *Shigley's,* the maximum shear stress is calculated given $\tau_{max} = \tau_{1/3}$.

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \mathbf{906.71}\ kpsi\ (\mathbf{17})$$

Failure:

The safety factor of the lead screw is calculated given the yield strength of the lead screw and maximum shear stress acting on it when it is just about to be opened.

$$n_l = \frac{S_y}{\tau_{max}}\ (\mathbf{18})$$

$$n_l = 0.132$$

Discussion:

The calculated safety factor from the lead screw analysis indicates that the lead screw would statically fail under the maximum loading conditions as specified by the manufacturer of the jack used in this study. Assuming the product's design was tested and approved for mass production according to standard guidelines, the analysis done in this paper indicates that there may be one or more assumptions which must have resulted in the failing safety factor. For example, the lead screw material could have been a harder steel instead of low carbon steel and the surface of the screw could have been coated with some material to add strength.

The screw was also analyzed under lower loading conditions than the maximum, but the load had to be significantly reduced by more than one magnitude for the screw to yield designable results. This indicates that it was not just this particular piece that could not handle the maximum specified load but the analysis was flawed in general.

More analytical error could stem from inaccurate measurements of the lead screw parameters and a flawed simplified analysis of the jack from the previous sections which may have carried over.

In conclusion, further modifications to the analysis must be made as the over-simplified model of the jack may have contributed to the failing safety factors. Accurate material properties and design measurements must be acquired as well for better results.

Conclusions

The simulations results are consistent with the analytical model. Both results show the scissor jack fails statically when applied the jack's loading capacity. This could be due to multiple factors. This includes the oversimplification of the member, not accounting for the other components of the jack as a whole, and conservative estimates of the model. But it is possible the structure was weakly designed and would not be recommended to be used a working mechanical jack at full loading capacity. In conclusion, all analysis conducted on the member indicate it is not structurally sound.

Resources

[1] https://www.britannica.com/technology/mechanical-advantage

[2] https://www.google.com/url?sa=i&url=https%3A%2F%2Fwww.autoguide.com%2Fbest-floor-jacks&psig=AOvVaw0O_eXdbWnXCbAgJHProgJ2&ust=1619543788946000&source=images&cd=vfe&ved=0CA0QjhxqFwoTCOjM5PO0nPACFQAAAAAdAAAAABAF

[3] https://www.metrohydraulic.com/mechanical-jacks/

[4] Hibblers – Mechanics of Materials, 10th edition

[5] Shigley - Mechanical Engineering Design, 10th edition

[6] https://www.google.com/url?sa=i&url=https%3A%2F%2Fwww.autoguide.com%2Fbest-floor-jacks&psig=AOvVaw1dK0uTT8639Uk-HbR2yQaF&ust=1619931650420000&source=images&cd=vfe&ved=0CA0QjhxqFwoTCJD_4-bZp_ACFQAAAAAdAAAAABAE

MAE 493 - Study of Mechanical Jack

Aishwarya Ledalla, Dr. Srinivas Kosaraju

```
clear
clc
```

Assumptions

```
% Assume 2D simplified structure so only axial stresses
% Can ignore weight since small(W = 13.5 lbs)
% symmetrical  about both axes
% Two lead screw contact points
% AISI 1018 Mild Steel
% 95% reliable
```

Compressive Force and Screw Force

```
% Given:
P = 6500; % maximum load in lbf
W = 13.5; % weight of jack in lbf

bolt_dis1 = 4.53125/2; % in
bolt_dis2 = 11; % in
theta = atand(bolt_dis1/bolt_dis2); % minimum angle in degrees

N = P; % normal force in lbf
% Force balance on bottom plate:
F = N./(2*sind(theta)); % compressive force in lbf
% Force balance on the member where the screw acts:
Fs = 2.*F.*cosd(theta); % screw force in lbf
```

Axial Stress at the Critial Location

```
% Dimentions of the c-ross section of the member at the hole:
l = 11; % length (in)
```

```
h = 1 + 7/8; % height (in)
w = 2; % width (in)
t = 13/128; % thickness (in)
d = 1; % diameter of hole (in)

% area of cross section at the pin hole and regular area
A = [(h.*w)-(h-t).*(w-2.*t)-(2.*d.*t) (h.*w)-(h-t).*(w-2.*t)];

sigma_a = (F./A)./1000; % axial stress in kpsi
```

Static Yielding Safety Factor

```
% Given:
% Cold Drawn AISI 1050 Steel
Sy = 94; % yield strength in kpsi
Sut = 123; % ultimate tensile strength in kpsi
E = 297*10^3; % Young's Modulus in kpsi

% Langer Line:
kt = 2.08;
ns = Sy./(kt.*sigma_a)';
```

Fatigue Safety Factor

Surface Factor:

```
% Cold-Drawn:
a = 2.7; % kpsi
b = -0.265;
ka = a.*Sut.^b;
```

Size Factor:

```
% Effective Diameter of Members:
A_95 = 0.05.*A; % in^2
de = 2.*sqrt(A_95./pi); %in

for i = 1:length(de)
    if de(i) > 2 % in
        kb(i) = 0.91.*de(i).^-0.157;
    else
        kb(i) = 0.879.*de(i).^-0.107;
    end
end
```

Loading Factor:

```
kc = 0.85; % 0.85 for axial
```

Temperature Factor:

```
kd = 1; % T < 70 degrees
```

Reliability Factor:

```
ke = 0.868; % 95% reliability
```

Miscellaneous-Effects Factor:

```
kf = 1; % assuming no corrosion
```

Material Endurance Strength:

```
Se_p = 0.5.*Sut; % kpsi
```

Material Fatigue Strength:

```
N = [10^3]; % number of cycles
f = 0.9; % for Sut < 70 ksi
A = ((f.*Sut).^2)./Se_p;
B = (-1/3).*log10(f.*Sut./Se_p);
Sf = A.*N.^B;
```

Endurance Strength:

```
Se = (ka.*kb.*kc.*ke.*kf.*Se_p); % kpsi
```

Fatigue Strength:

```
Sf = (ka.*kb.*kc.*ke.*kf.*Sf); % kpsi
```

Soderberg - conservative estimate

```
ne = 1./(kt.*sigma_a./Se)';
nf = 1./(kt.*sigma_a./Sf)';
```

Buckling of Member

```
% Least Moment of Inertia
L = 10.5; % in
I = 0.11054072; % in^4
K = 1; % both sides pinned
Pcr = E.*I.*(pi^2)./(K*L^2); % critical load in lbf
```

```
table(theta,N,F,Fs,Pcr,'VariableNames',{'Theta(degrees)','N(lbf)','F(lbf)','Fs(lbf)','Pcr(lbf)'})
table(ka,kb,kc,kd,ke,kf)
sigma_a = sigma_a';
table(ns,nf,ne,sigma_a,'RowNames',{'Hole cross section','Middle Cross Section'})
```

```
ans =

  1×5 table

    Theta(degrees)    N(lbf)    F(lbf)    Fs(lbf)    Pcr(lbf)
    ______________    ______    ______    _______    ________

        11.638         1000     16111      31559       2939


ans =

  1×6 table

      ka              kb              kc      kd     ke      kf
    _______    ________________    ____    __    _____    __

    0.75429    1.0757    1.0503    0.85    1     0.868    1


ans =

  2×4 table

                              ns        nf         ne       sigma_a
                            ______    _______    _______    _______

    Hole cross section      1.0105    0.71242    0.39579    44.723
    Middle Cross Section    1.5803     1.0878    0.60433    28.597
```

Lead Screw Analysis

```
d = 31/64; % in
p = 1/32; % in
dm = d - p./2;
f = 0.08; % frictional coeff
l = 2.*-4.215.*p;
fc = 0.08; % frictional coeff
dc = dm; % in
dr = d - p; % in
```

Required Torque to Turn Screw

```
TR = (Fs.*dm./2).*((l+pi.*f.*dm)./(pi.*dm-f.*l))+(Fs.*fc.*dc./2); % lbf in
% Load-Lowering Torque
TL = (Fs.*dm./2).*((pi.*f.*dm-l)./(pi.*dm+f.*l))+(Fs.*fc.*dc./2); % lbf in
```

Stresses

```
tau_s = 10^(-3).*16.*TR./(pi.*dr.^3); % shear stress along xz in kpsi
sigma_x = -10^(-3).*4.*Fs/(pi.*dr.^2); % axial nominal normal stress in kpsi
sigma_y = 10^(-3).*6.*(0.38.*Fs)./(pi.*dr.*p); % thread-root bending stress in kpsi (assume 38%
of Fs per thread)
```

Principal Stress

```
sigma2 = (sigma_x./2)+sqrt((sigma_x./2).^2+tau_s.^2); % kpsi
sigma3 = (sigma_x./2)-sqrt((sigma_x./2).^2+tau_s.^2); % kpsi
```

Von Mises stress

```
sigmap = sqrt(((sigma_y-sigma2).^2+(sigma2-sigma3).^2+(sigma3-sigma_y).^2)./2); % kpsi
sigmap_check = (1/sqrt(2)).*sqrt((sigma_x-sigma_y).^2+(sigma_y).^2+(-sigma_x).^2+6.*(tau_s).^2);
% kpsi, check to see if I'm getting the same result
```

Maximum Shear Stress

```
tau_max = (sigma_y-sigma3)./2; % kpsi
```

Lead Screw Static Safety Factor

```
nl = 120./tau_max;

table(sigma2,sigma3,sigmap,sigmap_check)
table(TR,TL,tau_max,nl)
table(tau_s,sigma_x,sigma_y,sigmap)
```

```
ans =

  1x4 table

    sigma2     sigma3     sigmap    sigmap_check
    _______    _______    ______    ____________

    0.25607    -195.96    1723.7       1723.7


ans =
```

```
  1×4 table

      TR        TL       tau_max      nl
    ______    ______    _______    _______

    -129.4    2534.4    906.71     0.13235


ans =

  1×4 table

     tau_s      sigma_x    sigma_y    sigmap
    _______    _______    _______    ______

    -7.0836    -195.7     1617.5     1723.7
```

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Simulation Results:

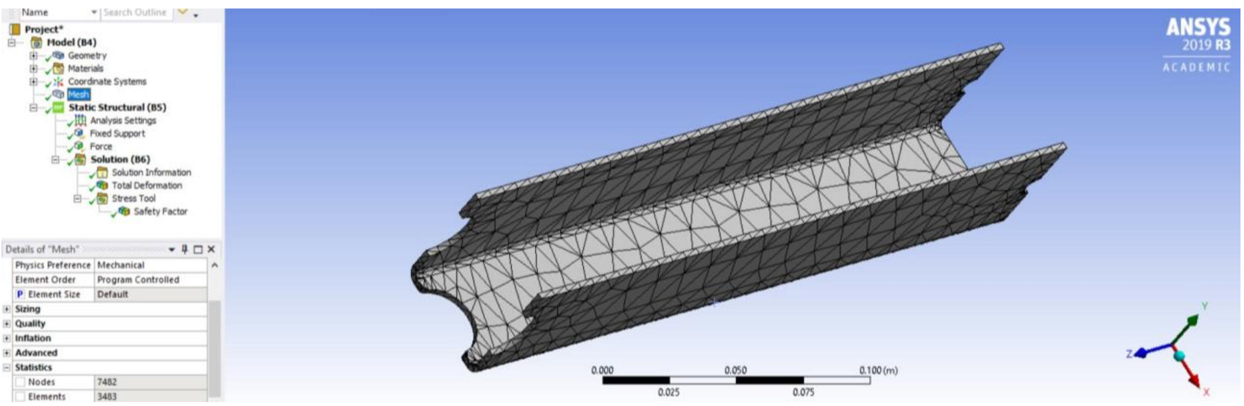


Figure 15. CAD Rev 2: Mesh with 7482 Nodes

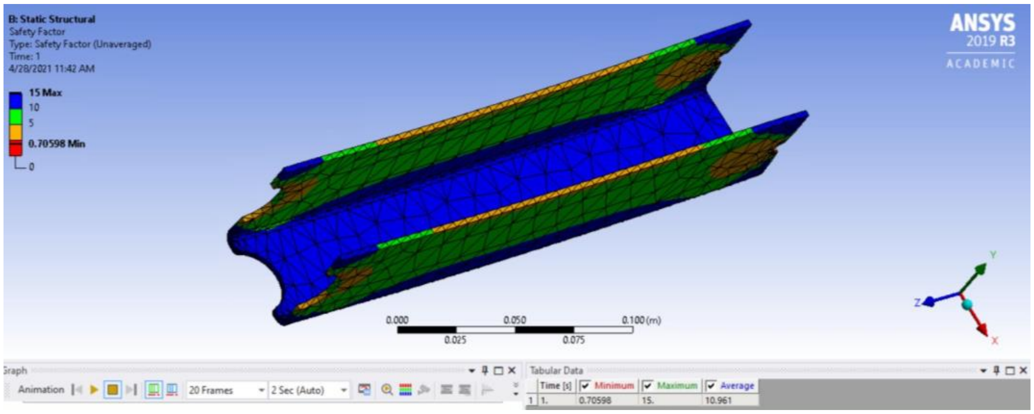


Figure 16. CAD Rev 2: Static Safety Factor when Mesh has 7482 Nodes

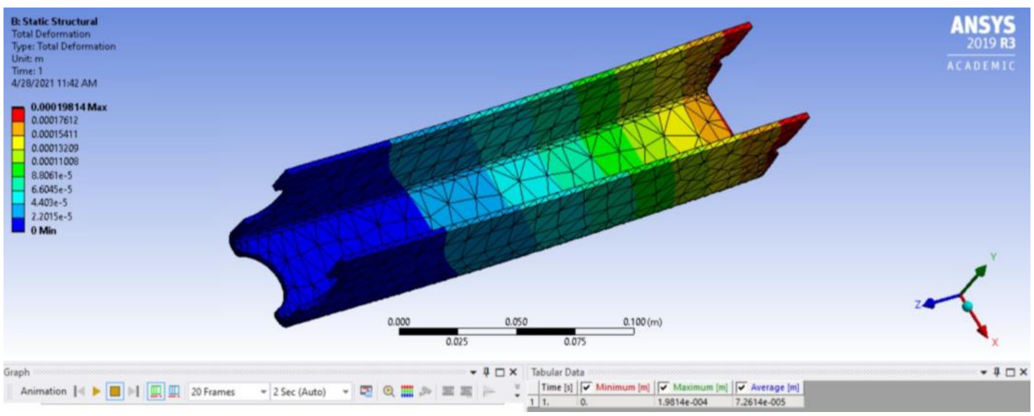


Figure 17. CAD Rev 2: Deformation when Mesh has 7482 Nodes

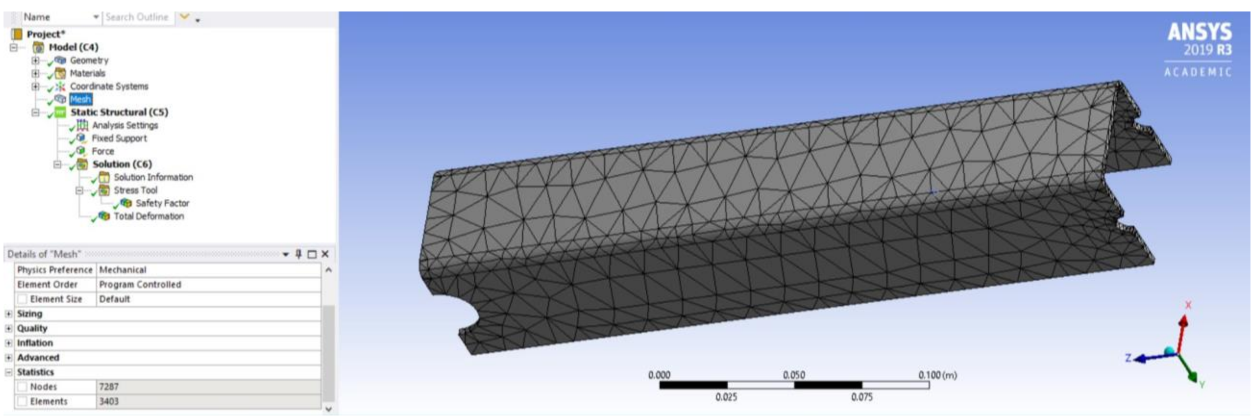


Figure 18. CAD Rev 3: Mesh with 7287 Nodes

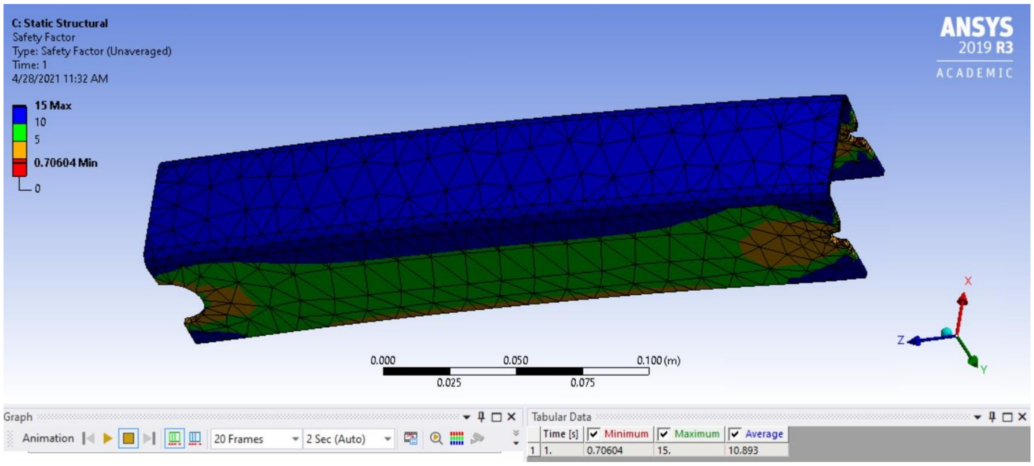


Figure 19. CAD Rev 3: Static Safety Factor when Mesh has 7287 Nodes

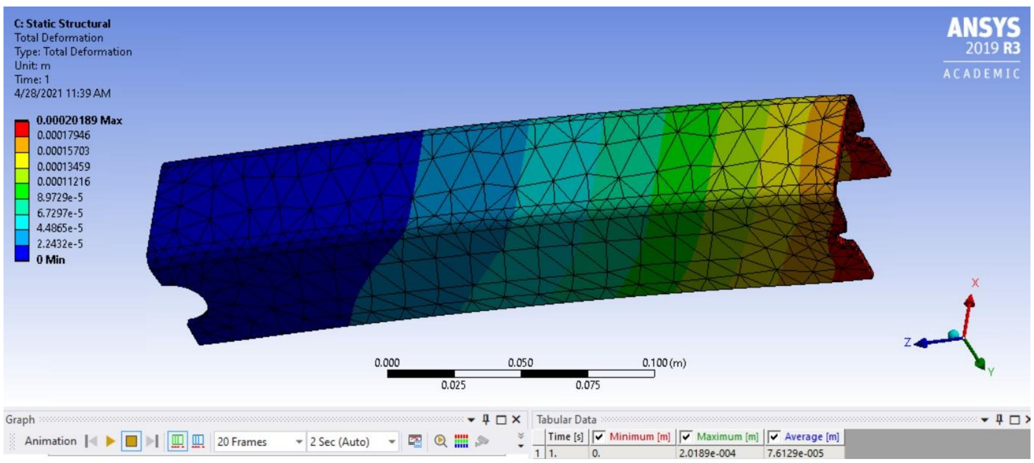


Figure 20. CAD Rev 3: Deformation when Mesh has 7287 Nodes

Mesh convergence:

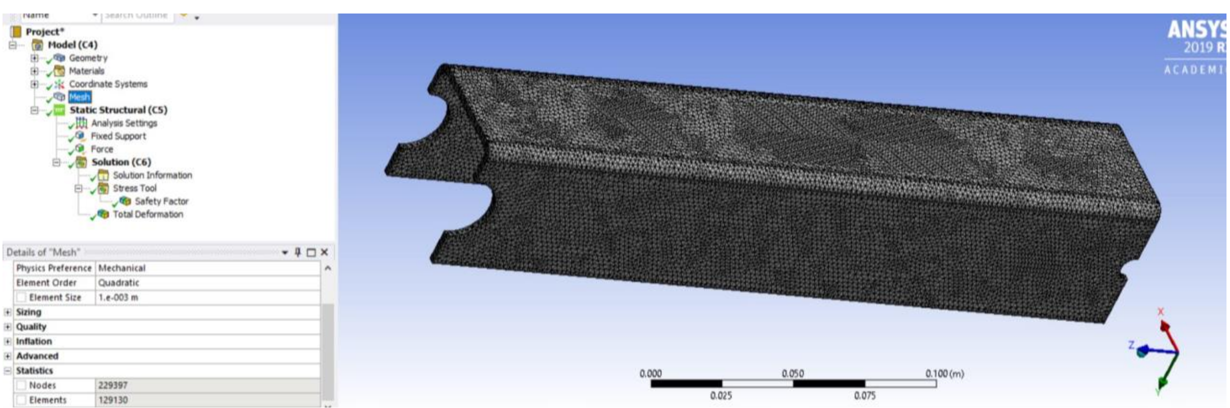


Figure 21. CAD Rev 3: Mesh with 229397 Nodes

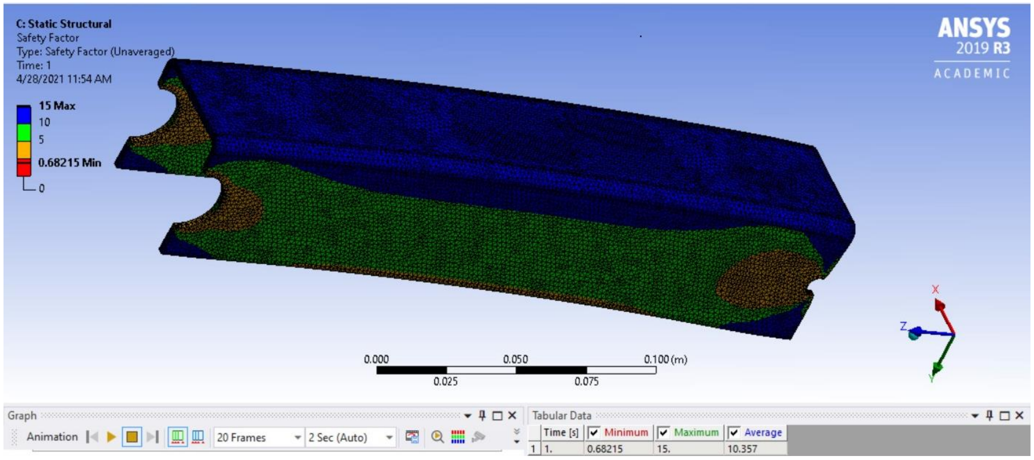


Figure 22. CAD Rev 3: Static Safety Factor when Mesh has 229397 Nodes

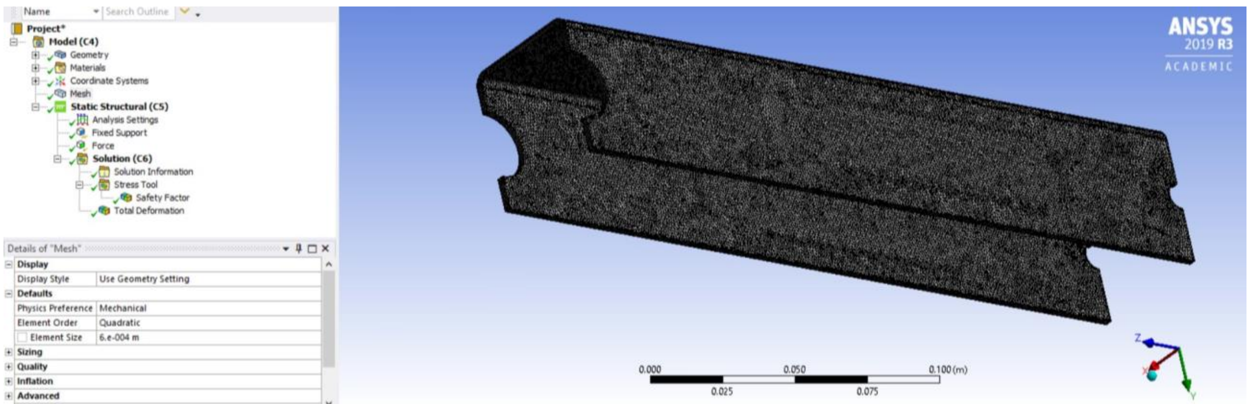


Figure 23. CAD Rev 3: Mesh with element size of $6 \times e^{(-0.004)}m$

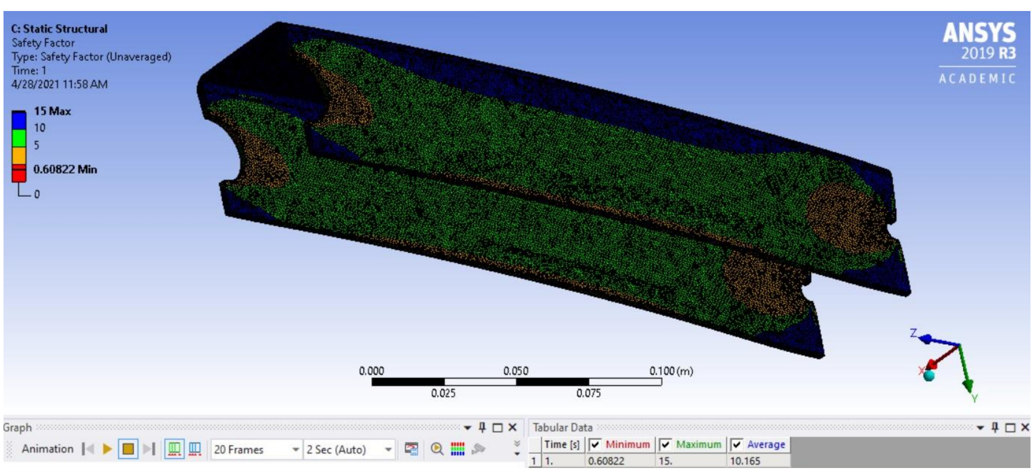


Figure 24. CAD Rev 3: Static Safety Factor when Mesh for the Previous Mesh

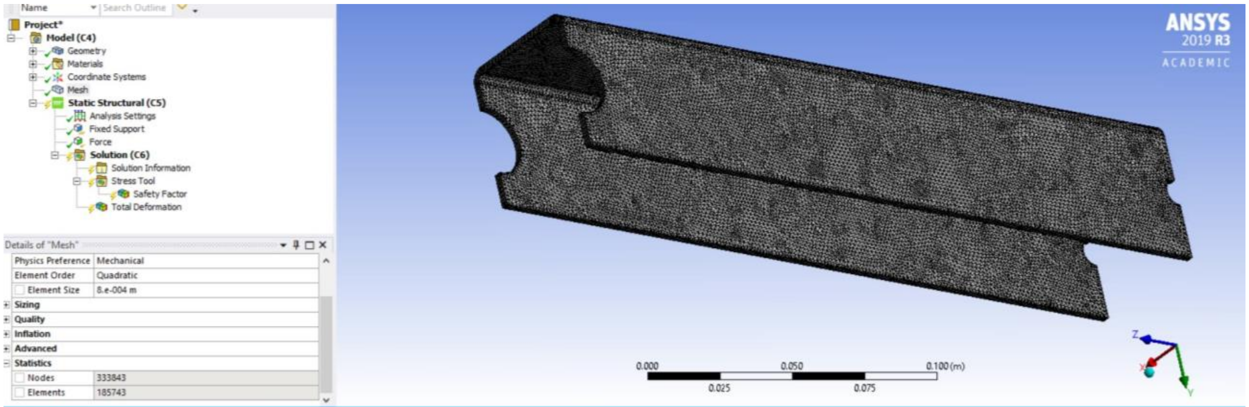


Figure 25. CAD Rev 3: Mesh with 333843 Nodes

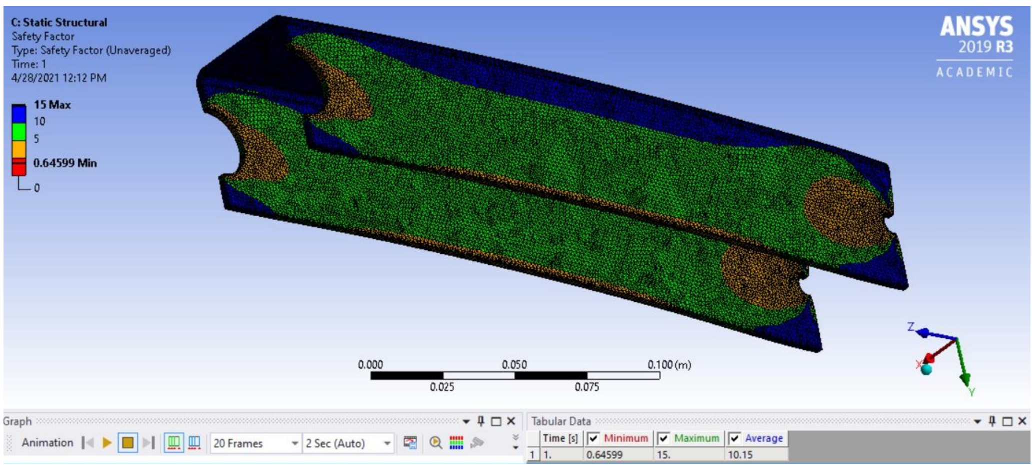


Figure 26. CAD Rev 3: Static Safety Factor when Mesh has 333843 Nodes

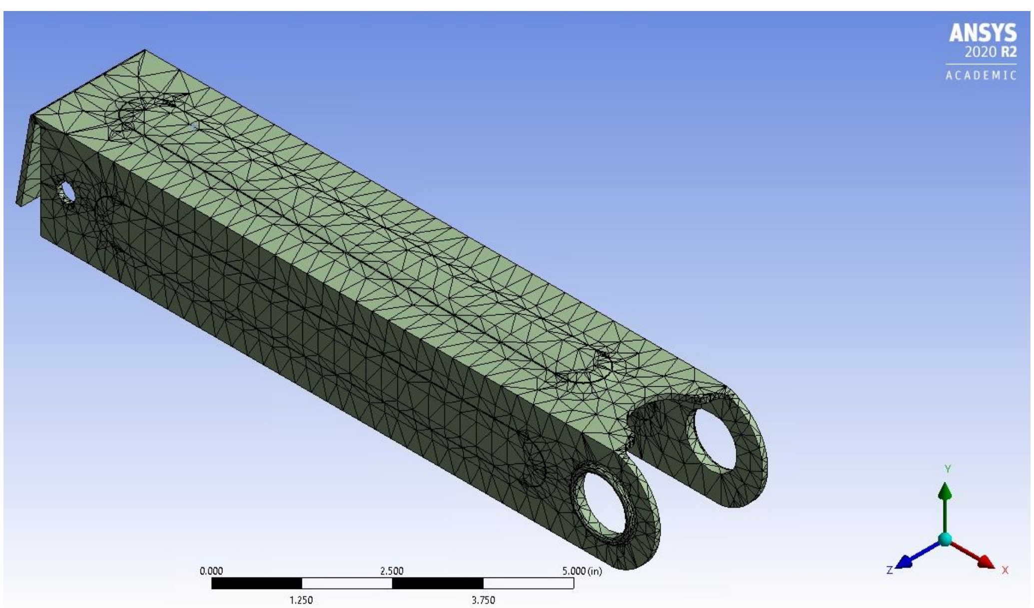
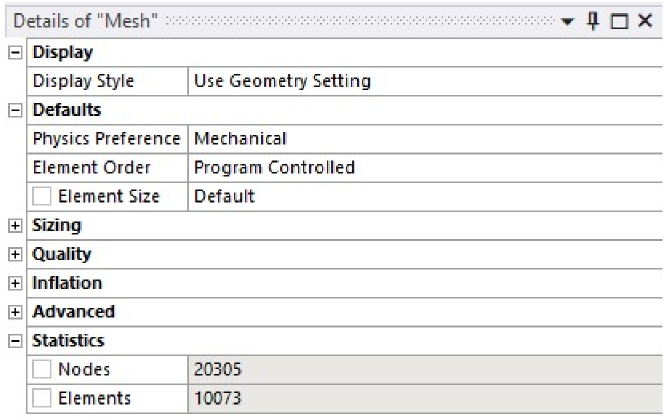


Figure 27. CAD Rev 1: Mesh



Details of "Mesh"

Display	
Display Style	Use Geometry Setting
Defaults	
Physics Preference	Mechanical
Element Order	Program Controlled
Element Size	Default
Sizing	
Quality	
Inflation	
Advanced	
Statistics	
Nodes	20305
Elements	10073

Figure 28. CAD Rev 1: Mesh of 20305 Nodes

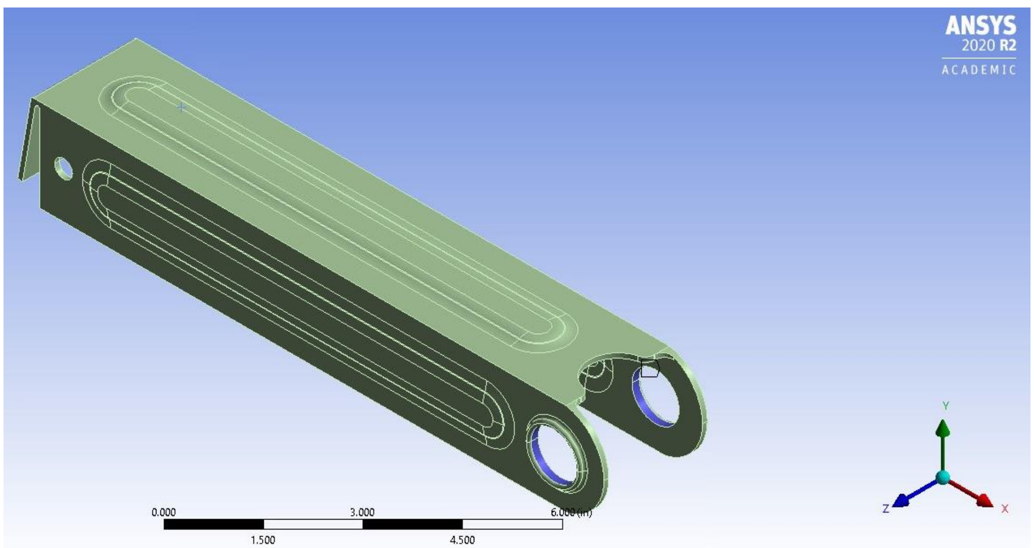


Figure 29. CAD Rev 1: Fixed Support

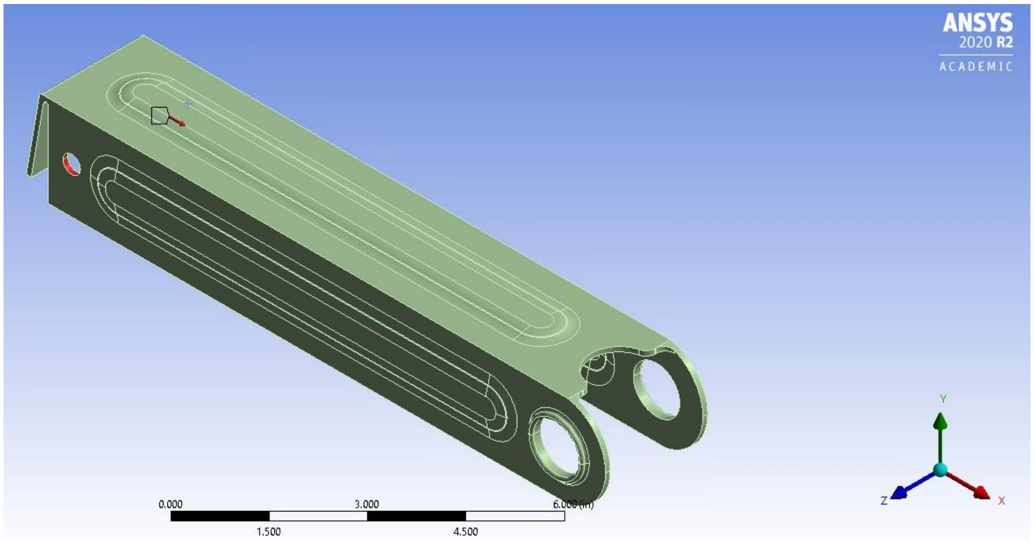


Figure 30. CAD Rev 1: Compressive Force at Bolt Holes

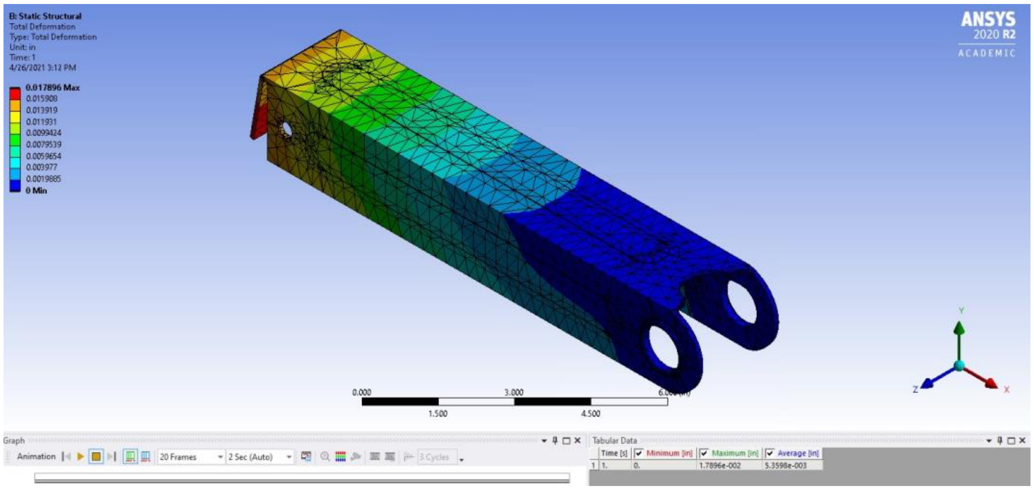


Figure 32. CAD Rev 1: Deformation when Mesh has 20305 Nodes

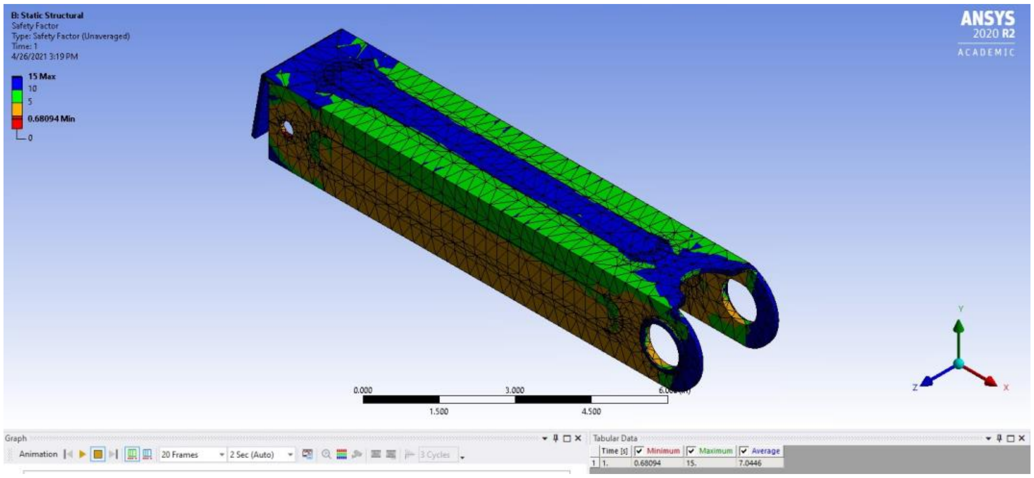


Figure 32. CAD Rev 1: Static Safety Factor when Mesh has 20305 Nodes