E91 Machine Design: Lab 2

Analysis of Stresses in Machines with FEA Techniques

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ABSTRACT

In this lab, a 3-ton arbor press was modeled and assembled in Solidworks. The stresses produced in the press during use were then simulated through finite-element analysis with Solidworks Simulation. A variety of simulation setups were used to examine their comparative validity, although no comparison to measured stress values has been made. Among all simulations, the more complex simulations (with all elements nonrigid and involved in simulation) produced similar stress values. At a pressing load of 2000 lbf (a press torque of 1000 lbf-in), maximum stresses of around 11 MPa/maximum stresses of around 300 MPa/43 ksi were produced in the gear teeth of the pinion and piston.

Introduction

Computerized finite-element analysis (FEA) is a valuable tool in the modern engineer's toolkit, allowing the simulation of extremely complicated stress situations that would be impossible to analyze using analytical methods. It also can allow simultaneous simulations of multi-part assemblies, as well as simulation of dynamic and nonlinear behaviors. However, without proper application, finite-element analysis tools yield poor results: as the common saying goes, "garbage in, garbage out."

Because of their utility in machine design applications, being able to correctly apply finite-element analysis tools to machine design problems – especially those involving complex assemblies – is of critical importance to machine designers. The goal of this lab was to examine the application of finite-element analysis tools in these situations, specifically through analysis of the stresses in the hand arbor press in the Engineering Department Machine Shop. The arbor press was modeled in SolidWorks and then simulated using Solidworks Simulation Premium, using a variety of constraint and load methods.

Background

The arbor press examined is a 3-ton press manufactured by Greenerd. The press may be seen in Although the materials used in its construction were not determined empirically, they are assumed to be cast iron in the frame, and cold-rolled steel in the piston and pinion.

The press has a 12" vertical capacity and a 6" throat. Its piston (the member that provides pressure) measures 1.750", and is 16.75" long. As can be seen from the picture, one side of the piston is flattened,

and a 12-pitch American National Standard coarse rack gear[1] is cut into the flattened surface. This rack is driven by a 12-tooth, 12-pitch ANS coarse pinion, which is cut into the end of a 1.186"-diameter steel bar (Figure 2).
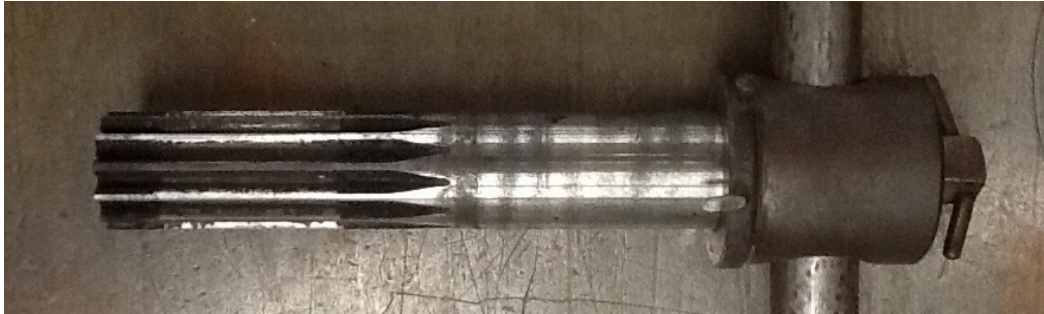

Figure 2 - Pinion Gear

All three components were measured using standard measuring equipment (dial calipers, rulers): non-critical dimensions (throat depth, large fillet radii) were approximated. Additionally, because of their complex nature, some features were interpolated between using a series of measurements. These measurements were then taken and used to develop 3D CAD models of all three components in Solidworks. The individual components were then combined in a Solidworks Assembly for simulation. These components and the completed assembly may be seen in Appendix 1: Solidworks Parts Images.

During use, the part to be pressed (commonly a bearing in a housing or some other press-fit part) is placed under the arbor press piston. The pinion is then turned using the crank visible in Figure 1. This gives significant mechanical advantage to the user, resulting in a 45:1 increase in force (Greenerd 2011). Within the arbor press, forces are developed as shown in Figure 3. Torque applied at the pinion produces a reaction force F_Z. This is resisted by the frame through F_R. The piston, meanwhile, produces a reaction force F_P through the press frame. In most cases, the press is used on a yielding body, which means that $F_Z > F_P$; however, for the purposes of this lab, they are assumed to be equal (the press is pressing on a non-yielding body).

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[1] The determination of the gear tooth form is approximate at best. Measurements of the gear system were taken using simple measurement tools (calipers, ruler, etc) without any specific gear-measurement equipment or techniques. These measurements were then compared to gear tables in Machinery's Handbook (pgs. 2030-2068), and the 12-pitch coarse specification was determined to best match measured values.
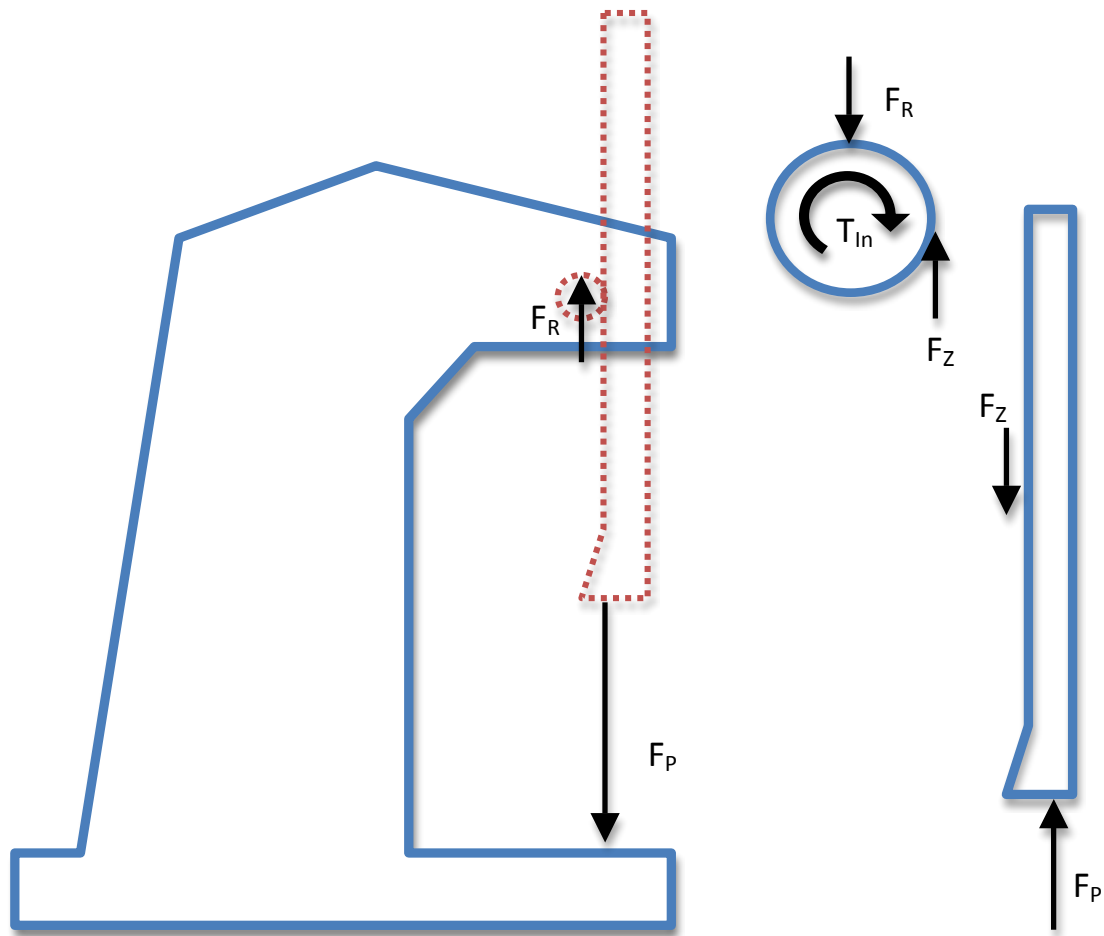
**Figure 3 - Arbor Press Free-Body Diagram**

Simulation

To simulate the stresses induced in the press by this combination of forces, a series of different finite-element analysis setups were used, in order to determine the relative performance of each method and examine whether certain methods were prone to inaccuracies in simulation. For all simulations, the arbor press frame was defined to be malleable cast iron, with a yield strength of 275 MPa (40 ksi) and a modulus of elasticity of 190 GPa (27 Mpsi) ; the pinion and piston components were defined to be AISI 1020 CR steel, with a yield strength of 350 MPa (51 ksi) and a modulus of elasticity of 205 GPa (30 Mpsi). Additionally, the base of the frame was defined to be fixed for all simulations. With the exception of the separated simulations, no-penetration contact was defined between the touching surfaces of the pinion and frame, the piston and frame, and the teeth in contact between the piston and the pinion. Some cosmetic features (fillets in noncritical areas) were suppressed during simulation to cut down simulation time.

Images of each simulation are included in

Part 2: Pinion

Part 3: Piston

Image 4: Completed Assembly
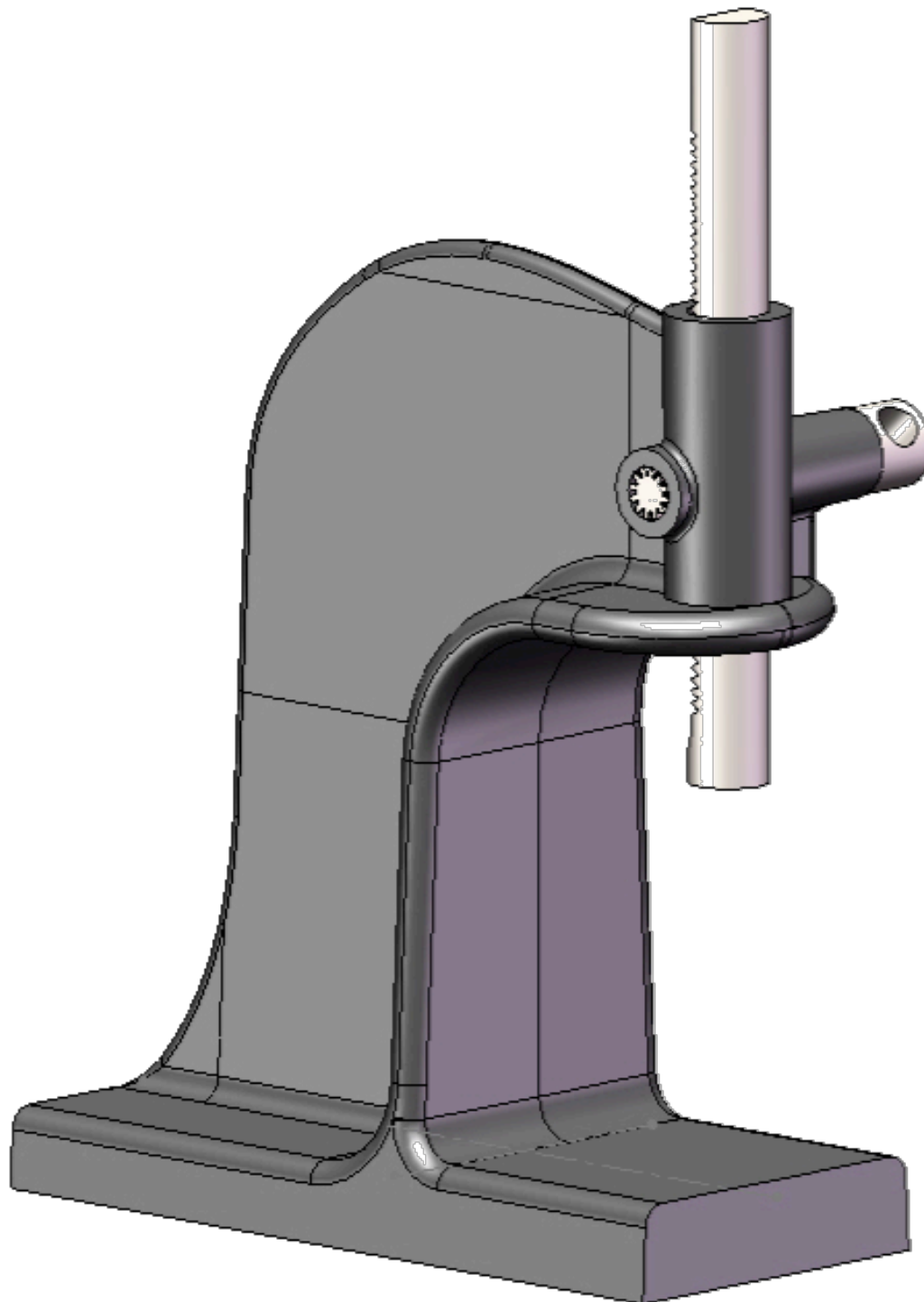
Appendix 2: Simulation Images. **It should be noted that deformation magnification scales differ between images**; although deflections may appear significantly different, they are not.

Unfortunately, because the stresses in the arbor press could not be experimentally examined (by, for example, applying strain gages to the press and subjecting it to load), the accuracy of the test methods used cannot be determined. However, methods that produce obviously incorrect results can be identified from the tests conducted here, allowing the selection of a method for "first-approximation" stress determination.

Force On Piston

In these simulations, the end face of the pinion (where the handle attaches) was defined as fixed against rotation about its longitudinal axis (to simulate its being rotationally fixed), and a 2000 lbf load was applied to the bottom of the piston in an upwards direction.

- Rigid Piston and Pinion: In this simulation, the pinion and piston were defined as rigid, but were still included in the simulation. Because these components were defined as rigid, stresses within them were not calculated. Within the frame, areas of maximum stress were the inside of the large radius at the top of the work volume (11 MPa/1.5 ksi), and at the bearing surface where the pinion meets the frame (20 MPa/2.9 ksi). The first photo shows these stress distributions within the frame: the second photo shows a section of the frame taken along its longer base axis.
- Nonrigid Piston and Pinion: In this simulation, the pinion and piston were defined as nonrigid, thus allowing the stresses developed within them to be determined. Stresses within the frame are similar to those experienced with a rigid piston and pinion, except that the maximum stress experienced is higher (50 MPa/7.2 ksi), and occurs where the pinion enters its housing. The first image shows the entire press frame, while the second shows a section taken along the longer base axis of the frame, just on the far side of the frame's "spine".
  In the pinion and piston, stresses are also highest near to where the pinion enters its housing, as shown by the third, fourth and fifth images (the fourth and fifth are sections also along the longer base axis, and the color scales in these images are NOT the same as those in the images of the frame). Stresses are significantly higher here (300 MPa/43 ksi). Significant stresses are also experienced on the "back" side of the pinion, possibly due to bearing against the frame.

Torque on Pinion

In these simulations, a torque of 1000 lbf-in was applied to the face of the pinion where the handle enters, while the end of the piston was defined as fixed.

- Rigid Piston and Pinion: This simulation setup yielded significantly lower stresses. The maximum stress developed in the frame was only 450 KPa (65 PSI), and occurred at the front inner face of the piston shaft housing (visible in the second image, a section taken along the longer base axis).
- Nonrigid Piston and Pinion: The stresses produced by this simulation setup were comparable to those produced by the previously attempted methods. In the frame (the first image), maximum stresses were again at the inside of the large radius at the top of the work volume (10 MPa/1.4

ksi), and at the bearing surface where the pinion meets the frame (20 MPa/2.9 ksi). In the pinion and piston (the second image) stresses were again quite high, approaching 300 MPa (43 ksi) at the base of the gear tooth fillets. Again, elevated stress distributions were seen throughout the pinion gear, especially at the back and top of the gear. Further investigation into the source of these stresses is needed, but it is suspected that they are a combination of bearing stresses and torsional stresses[2]

Separated Simulation

- Frame: The frame was tested apart from the other components by applying a 2000 lbf force at the curve marking the intersection between the piston and pinion shaft housings. The other components were excluded from this simulation. The highest stresses experienced in the body of the frame occurred again at the inside of the large radius at the top of the work volume (13 MPa/1.9 ksi), and at the bearing surface where the pinion meets the frame (30 MPa/4.3 ksi). Because of the force application method, the stress values at the bearing surface are not considered representative: however, the frame stresses are.
- Piston and Pinion: The piston and pinion were simulated independently of the frame, with the frame excluded from the simulation. The pinion was constrained with a hinge constraint on its main surface (where it contacts its shaft housing) and a torque of 1000 lbf-in was applied to the face where the handle would enter. Meanwhile, the piston was constrained with a sliding constraint along its main surface (where it contacts its shaft housing), and its end was held fixed. The maximum stresses produced in this simulation were significantly higher than in other simulations (up to 5,500 MPa/800 ksi): however, their limited area (visible in the third and fourth images) makes it clear that these were bearing stresses caused by the corner of one gear impacting the other gear's tooth, and not an accurate depiction of the actual stress state present in the component. While the resolution of the two images is insufficient to determine with any confidence what the stresses at the expected areas of maximum stress (the fillets of the gear teeth) are, they are within the range of hundreds of MPa – within an order of magnitude of the values predicted by the other methods attempted.

Conclusion

As mentioned above, because the stresses in the arbor press could not be experimentally examined, the accuracy of the test methods used cannot be determined. However, based on the above results, some conclusions can be drawn about the relative accuracy of the various methods.

First, it appears that simulation quality (or at least simulation results consistency) may actually improve with higher levels of modeling accuracy: for example, the furthest outlying results (high and low stress) both occurred in simulations where either some components were treated as rigid for the purposes of simulation (Torque on Pinion – Rigid Piston and Pinion), or were constrained by virtual constraints (Separated Simulation – Piston and Pinion). In the more complex simulations, while the resolution of the

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[2] Since stress plots are reported in von Mises stress, the directionality of the stress is unclear.

images taken is insufficiently high to precisely determine stress values in the assembly, it is clear that the stress values are all within a similar range, and that areas of high stress are consistent across simulations.

Second, the importance of fully understanding the function of the different simulation options is clearly demonstrated in this lab. During an early simulation run (Force on Piston – Nonrigid Pinion and Piston), a fixed constraint was mistakenly applied to the end of the pinion instead of a nonrotational constraint. This slight difference produced a wildly different result – the arbor press frame bent simultaneously backwards as expected, but also twisted around its longitudinal axis about the end of the pinion handle – which was not representative of the real-life behavior of the press.

Finally, there is significant disagreement between the specified load capacity of the press and simulation results. At a 1-ton load, the pinion and piston are already approaching yield, according to simulation results. It is likely that the steel used in the pinion and piston is not in fact AISI 1020, and instead has a far higher yield strength. However, this disagreement may also be due to inaccuracies in simulation, something which cannot be determined without empirical measurement of the stresses produced during use.

Avenues for future work include:

- Measurement of stress in simulation at specific locations through use of Solidworks' "probe" capability. This would allow more precise comparison of each testing methodology.
- Investigation of directional stresses in frame. As mentioned above, since the stress plots in this lab report von Mises stress, the directionality of the stress is unclear.
- Comparison of simulation output from Solidworks to other FEA programs (for example, ANSYS). Some preliminary comparisons were done between Solidworks and ANSYS for the purposes of this lab: although the process of transferring Solidworks models into ANSYS was determined to be too unwieldy for the purposes of this lab, the results of basic simulations carried out in the two programs were extremely similar.
- Comparison to real-life stress generation. Possibly as an E59 project/a final project for this course, strain gages could be attached to the arbor press and the stresses produced by pressing operations could be determined. These could then be compared to the stresses predicted by Solidworks and/or ANSYS.
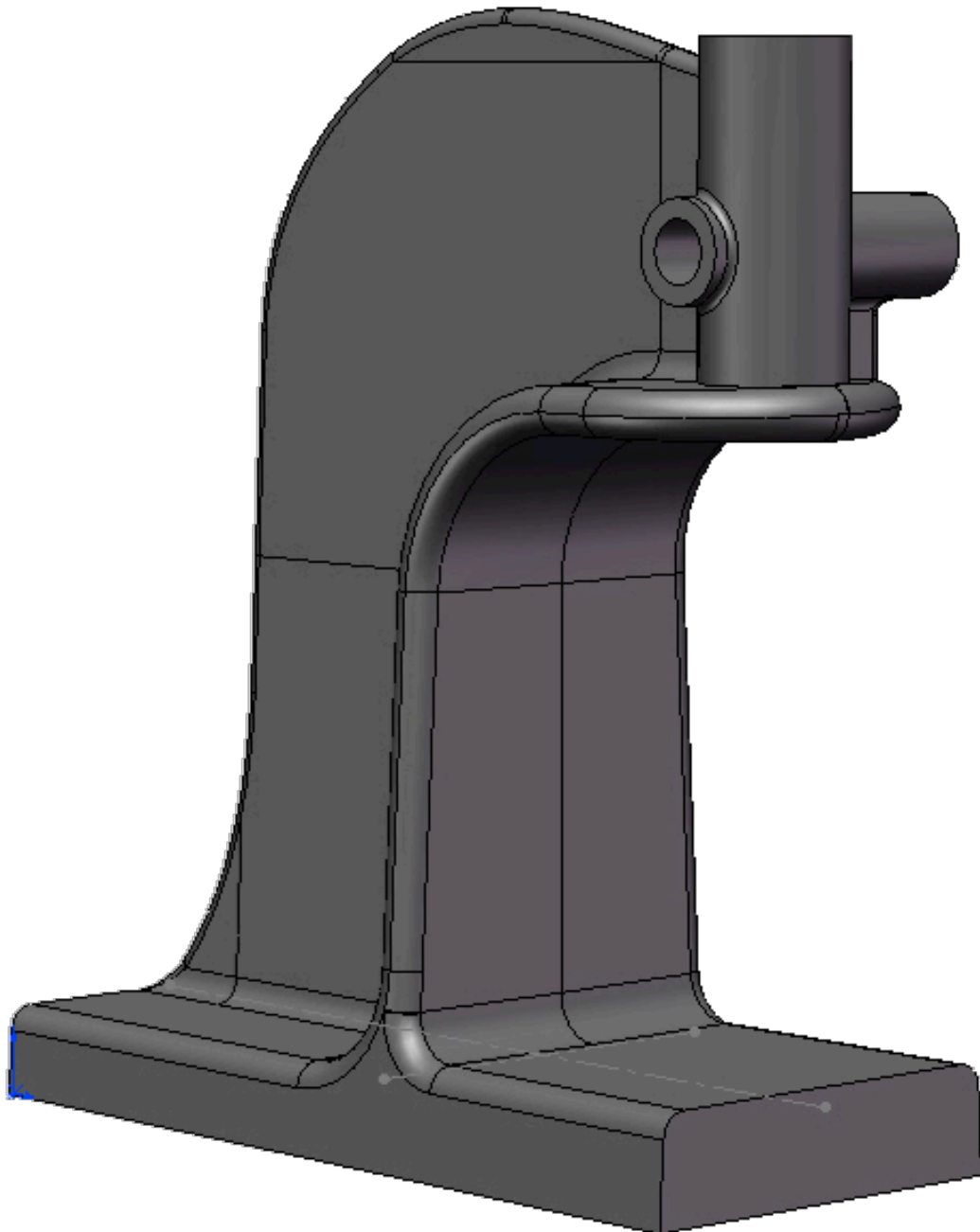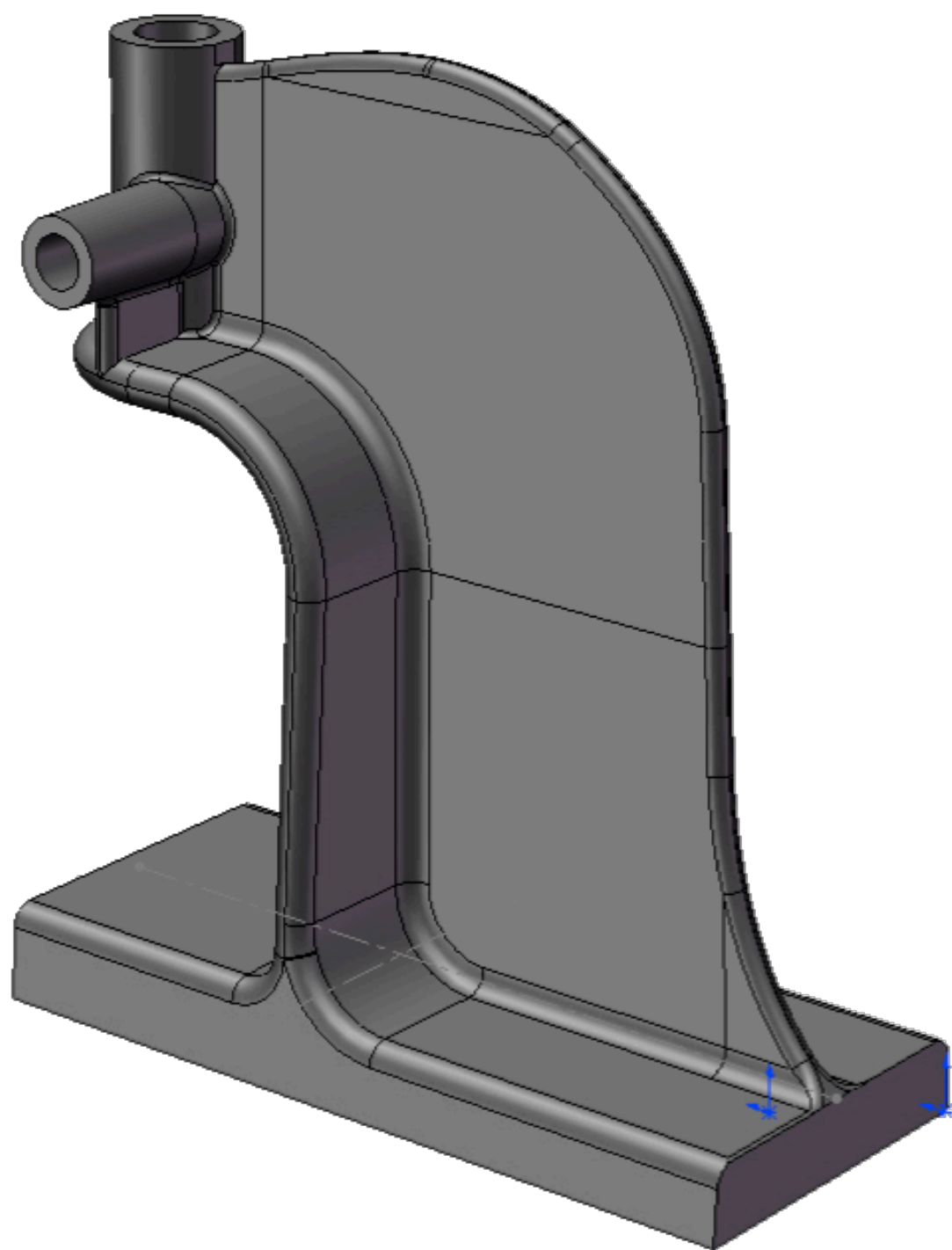
Works Cited

Greenerd. *No. 3 Bench Press.* 2011. http://www.greenerd.com/arborpress-detail.aspx?pressid=6 (accessed Dec. 2, 2011).

Industrial Press. *Machinery's Handbook.* 28th. New York: Industrial Press, 2008.
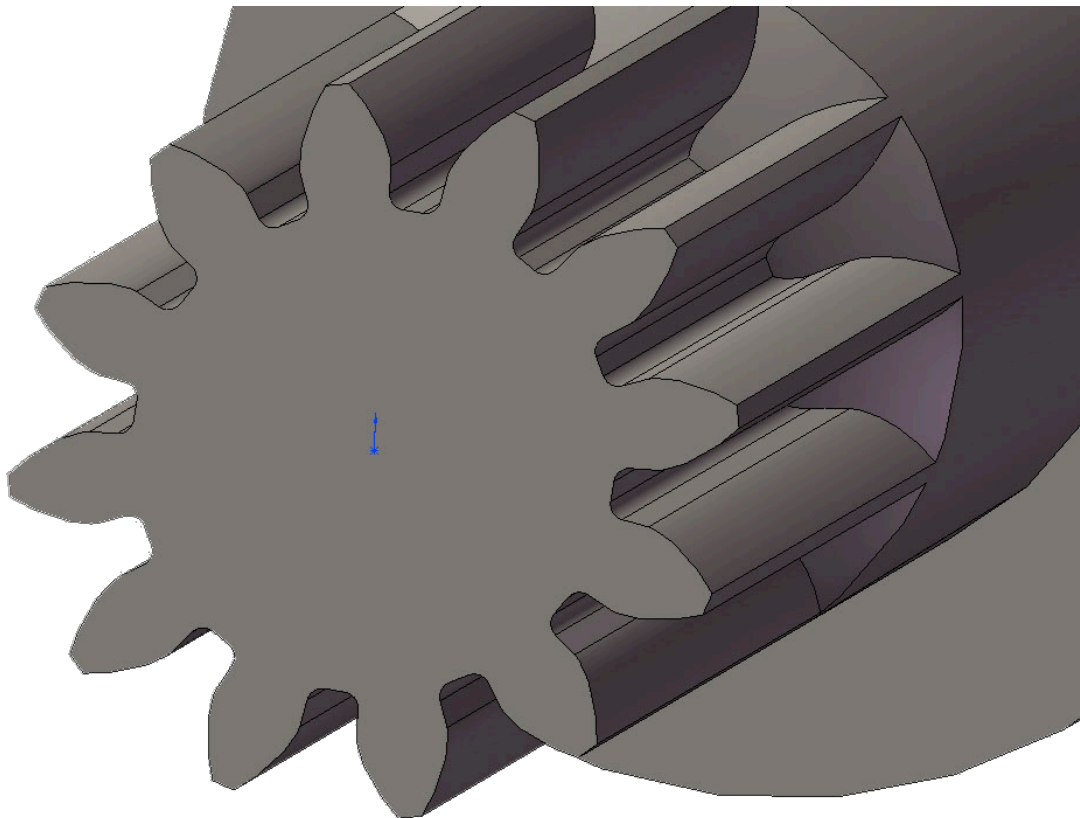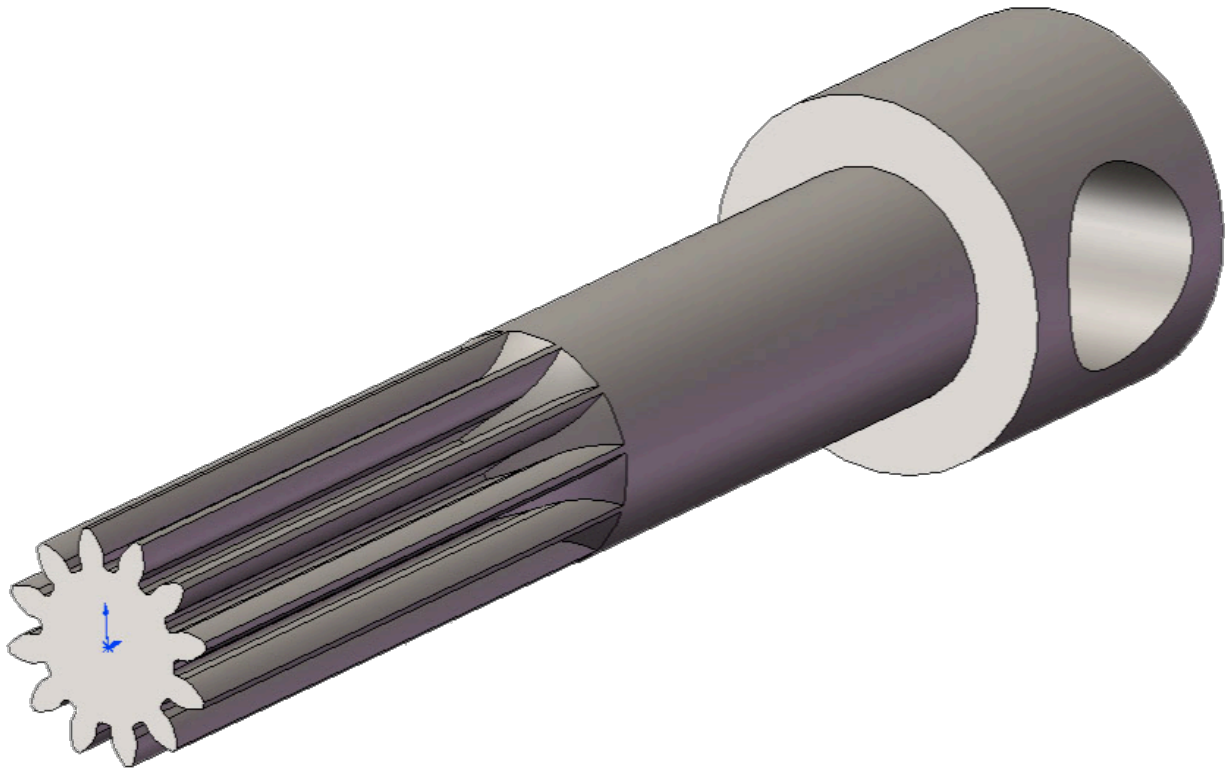
Appendix 1: Solidworks Parts Images

Part 1: Frame
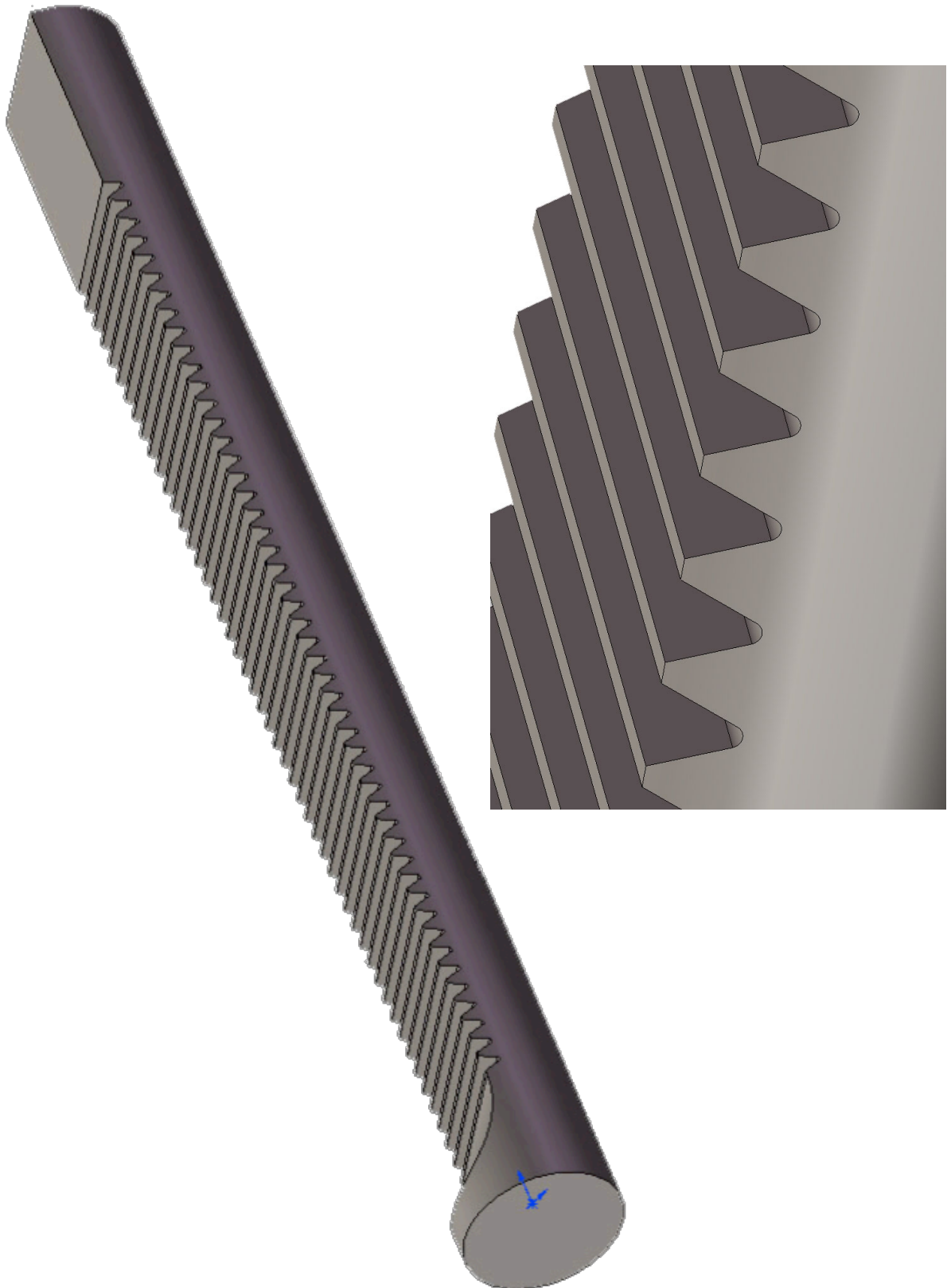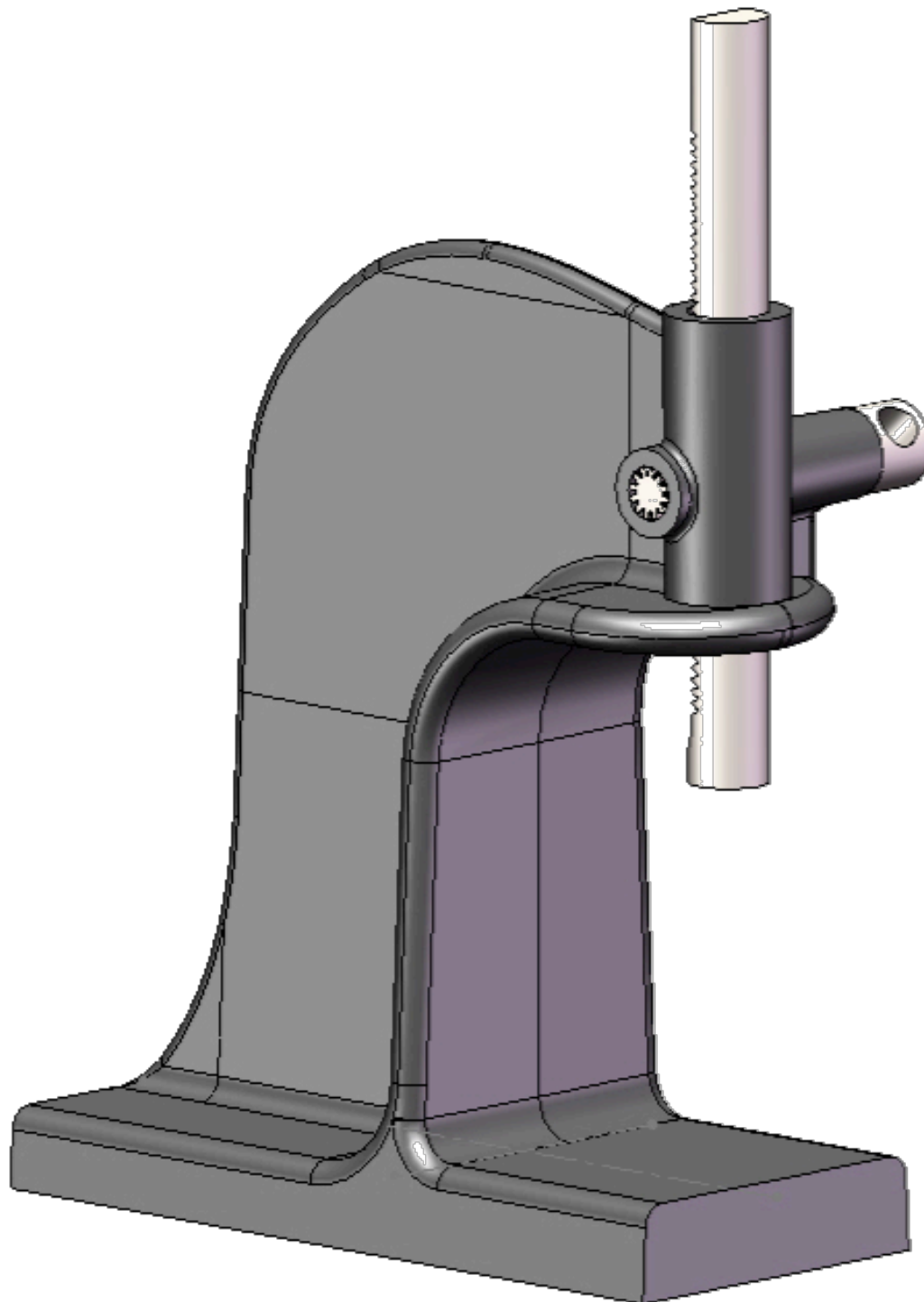
Part 2: Pinion

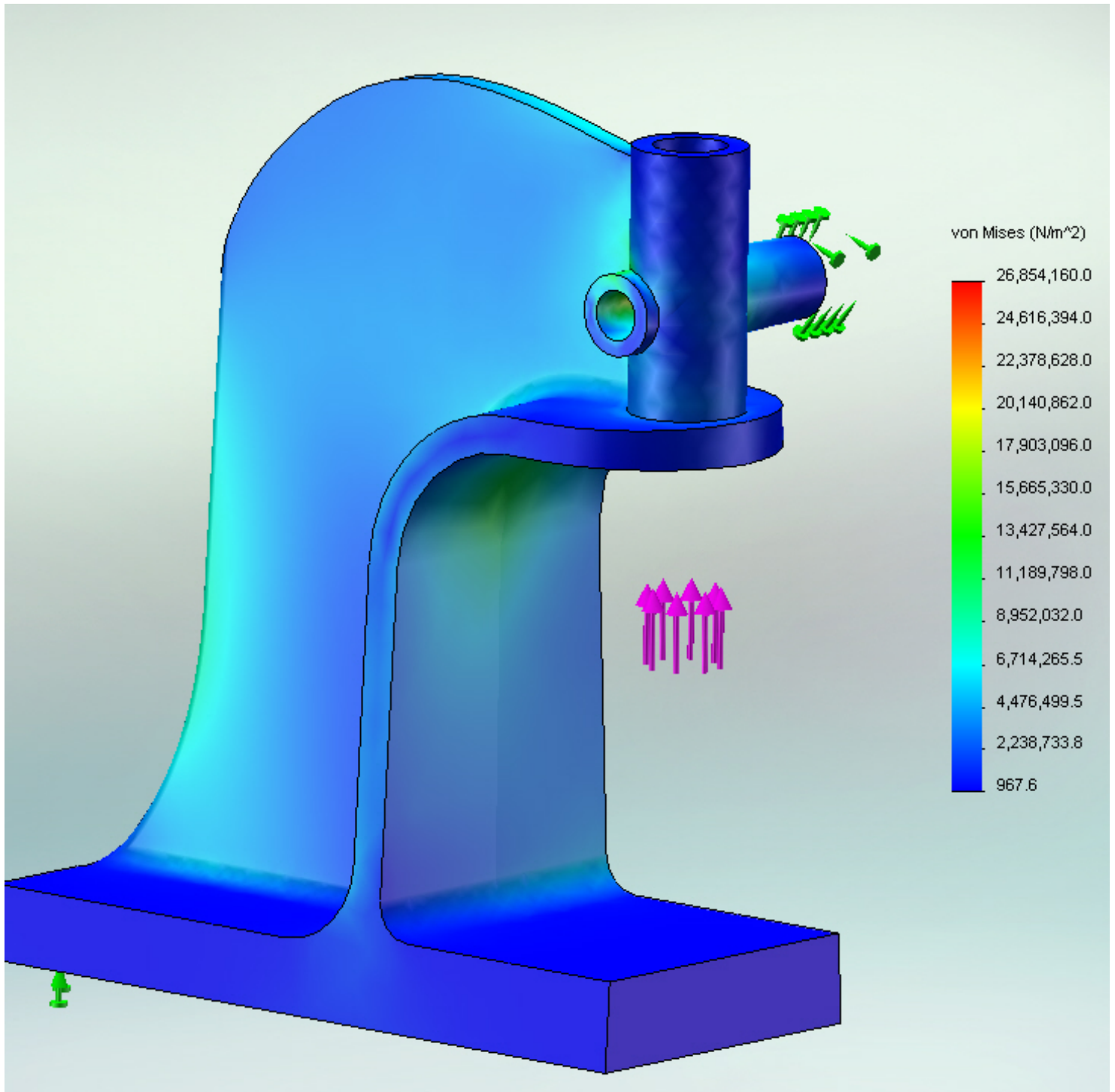Part 3: Piston

Image 4: Completed Assembly

Appendix 2: Simulation Images

Test 1: Force On Piston – Rigid Pinion and Piston

von Mises (N/m^2)

26,854,160.0
24,616,394.0
22,378,628.0
20,140,862.0
17,903,096.0
15,665,330.0
13,427,564.0
11,189,798.0
8,952,032.0
6,714,265.5
4,476,499.5
2,238,733.8
967.6

me: ArborPress
me: Fixed Pinion
: Static nodal stress/Stress2
alue: 0 to 2.68542e+007 N/m^2
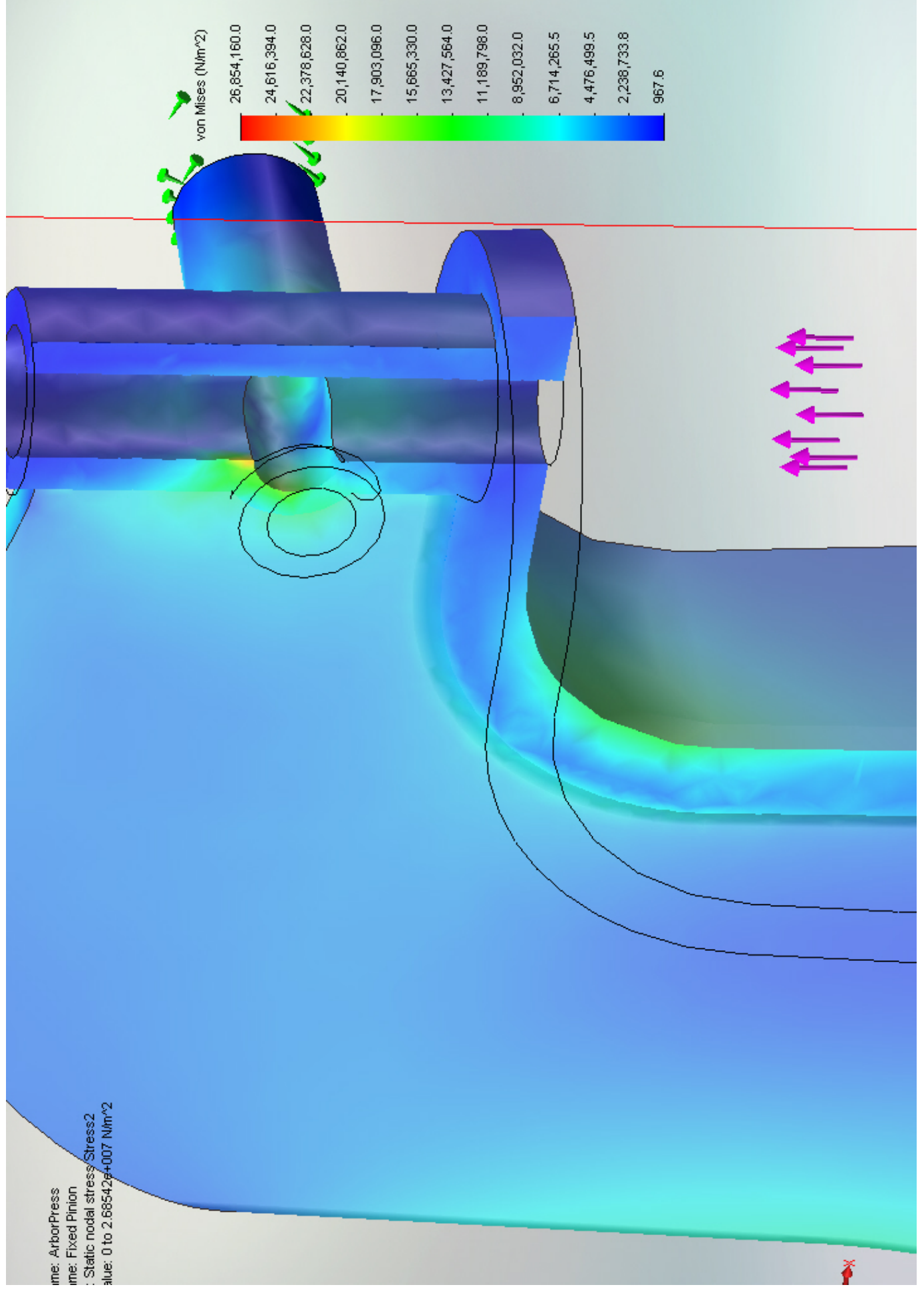
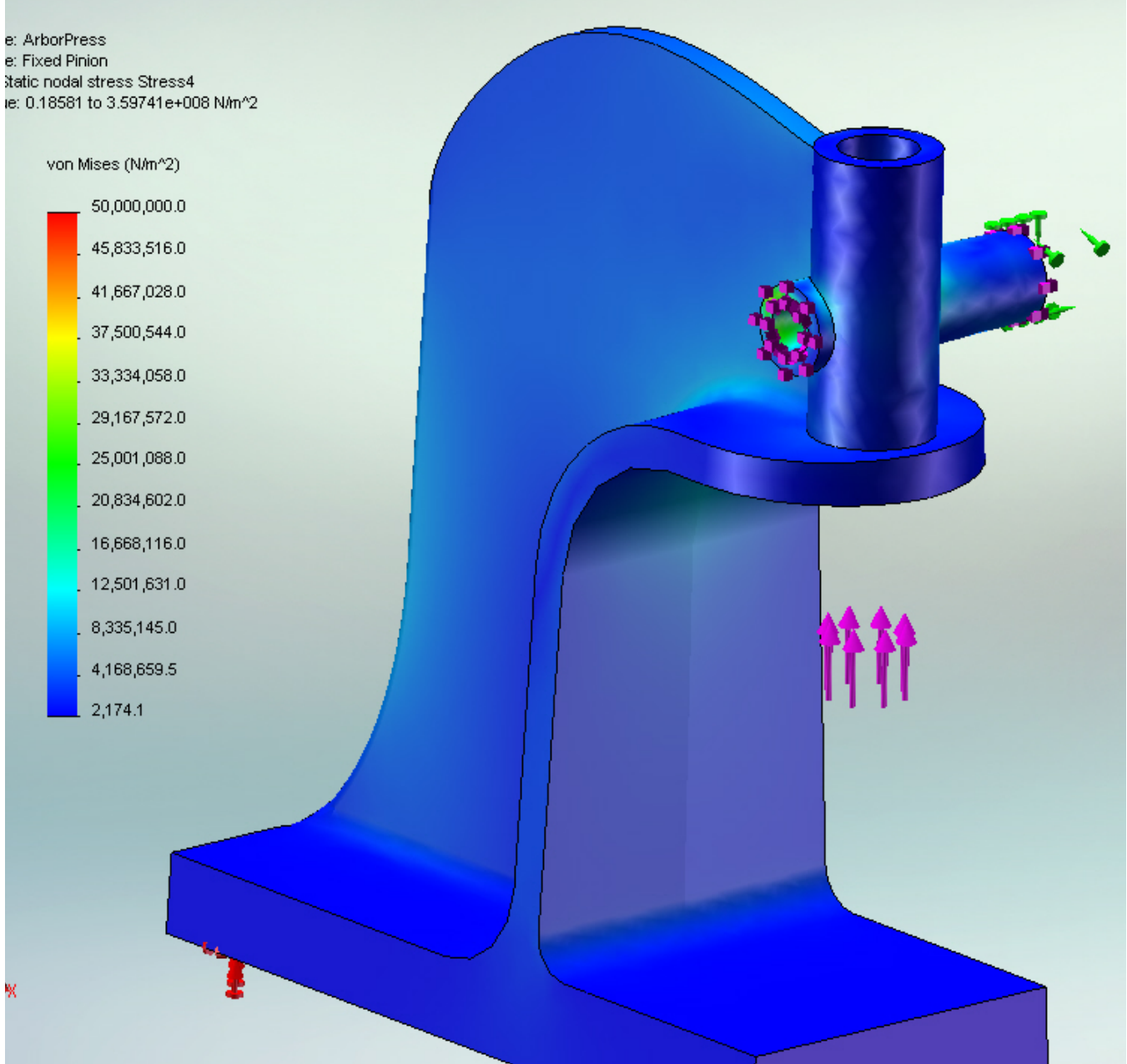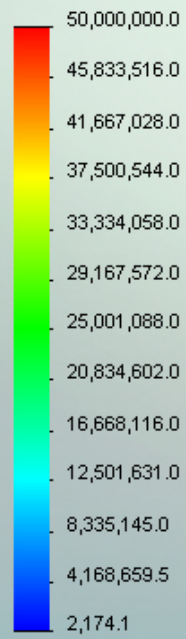**Test 2:** Force on Piston – Nonrigid Piston and Pinion

Study name: ArborPress
Fixed Pinion
Static nodal stress Stress4
Range: 0.18581 to 3.59741e+008 N/m^2

von Mises (N/m^2)

50,000,000.0
45,833,516.0
41,667,028.0
37,500,544.0
33,334,058.0
29,167,572.0
25,001,088.0
20,834,602.0
16,668,116.0
12,501,631.0
8,335,145.0
4,168,659.5
2,174.1

von Mises (N/m^2)

359,741,216.0
329,762,976.0
299,784,704.0
269,806,464.0
239,828,208.0
209,849,952.0
179,871,696.0
149,893,440.0
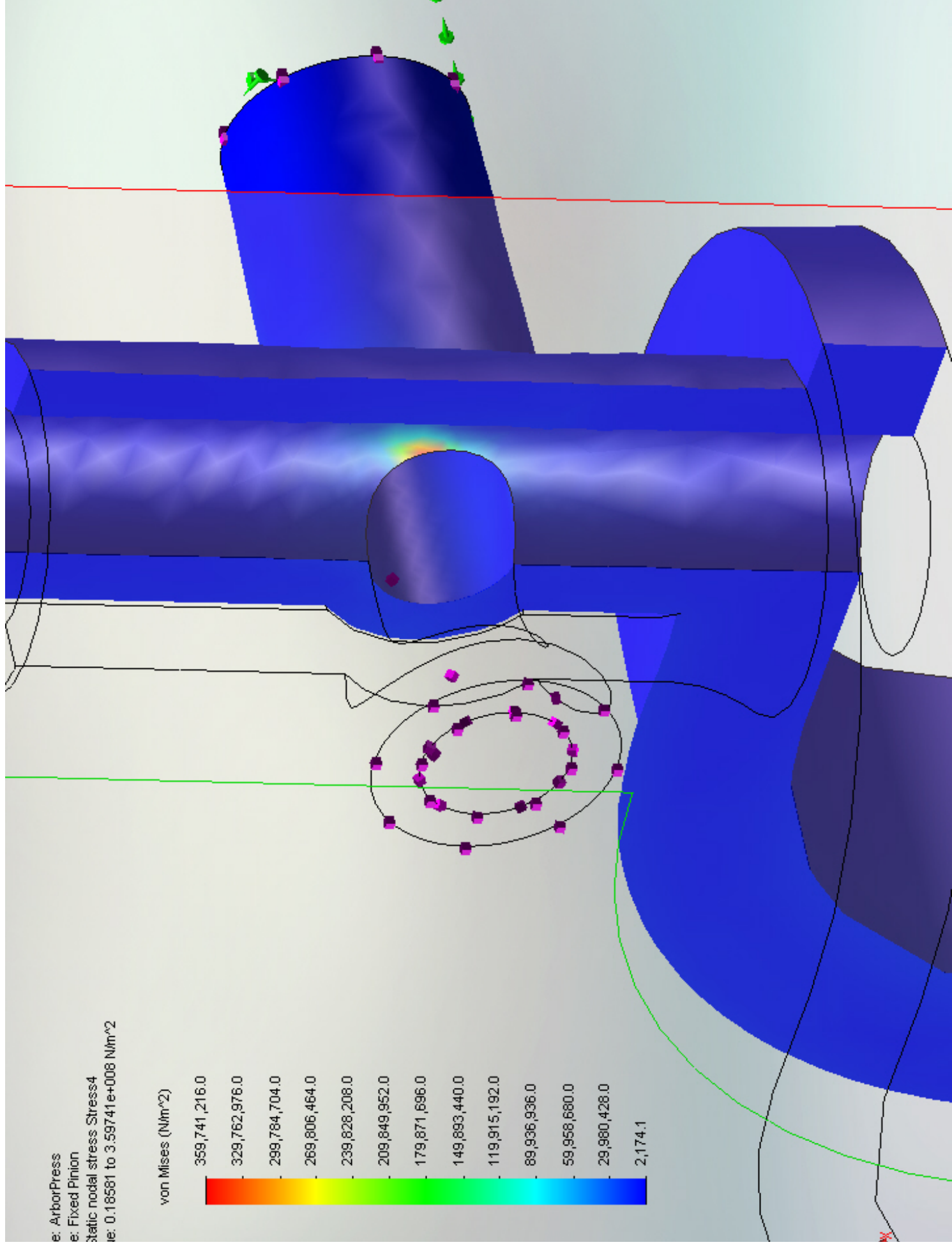119,915,192.0
89,936,936.0
59,958,680.0
29,980,428.0
2,174.1
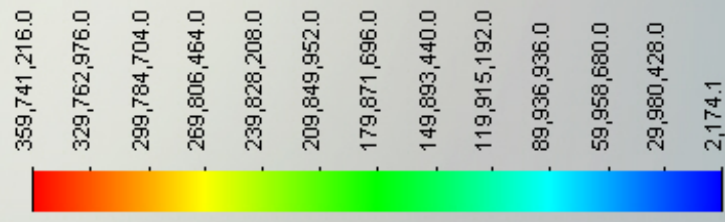
e: ArborPress
e: Fixed Pinion
static nodal stress Stress4
ue: 0.18581 to 3.59741e+008 N/m^2

von Mises (N/m^2)

359,741,216.0
329,762,784.0
299,784,352.0
269,805,920.0
239,827,472.0
209,849,040.0
179,870,608.0
149,892,176.0
119,913,736.0
89,935,304.0
59,956,868.0
29,978,434.0
0.2

von Mises (N/m^2)

359,741,216.0
329,762,784.0
299,784,352.0
269,805,920.0
239,827,472.0
209,849,040.0
179,870,608.0
149,892,176.0
119,913,736.0
89,935,304.0
59,956,868.0
29,978,434.0
0.2

Deformation scale: 1

von Mises (N/m^2)

359,741,216.0
329,762,784.0
299,784,352.0
269,805,920.0
239,827,472.0
209,849,040.0
179,870,608.0
149,892,176.0
119,913,736.0
89,935,304.0
59,956,868.0
29,978,434.0
0.2

Test 3: Torque on Pinion – Rigid Piston and Pinion

von Mises (N/m^2)

885,223.4
811,455.8
737,688.1
663,920.4
590,152.8
516,385.1
442,617.4
368,849.8
295,082.2
221,314.5
147,546.8
73,779.2
11.5
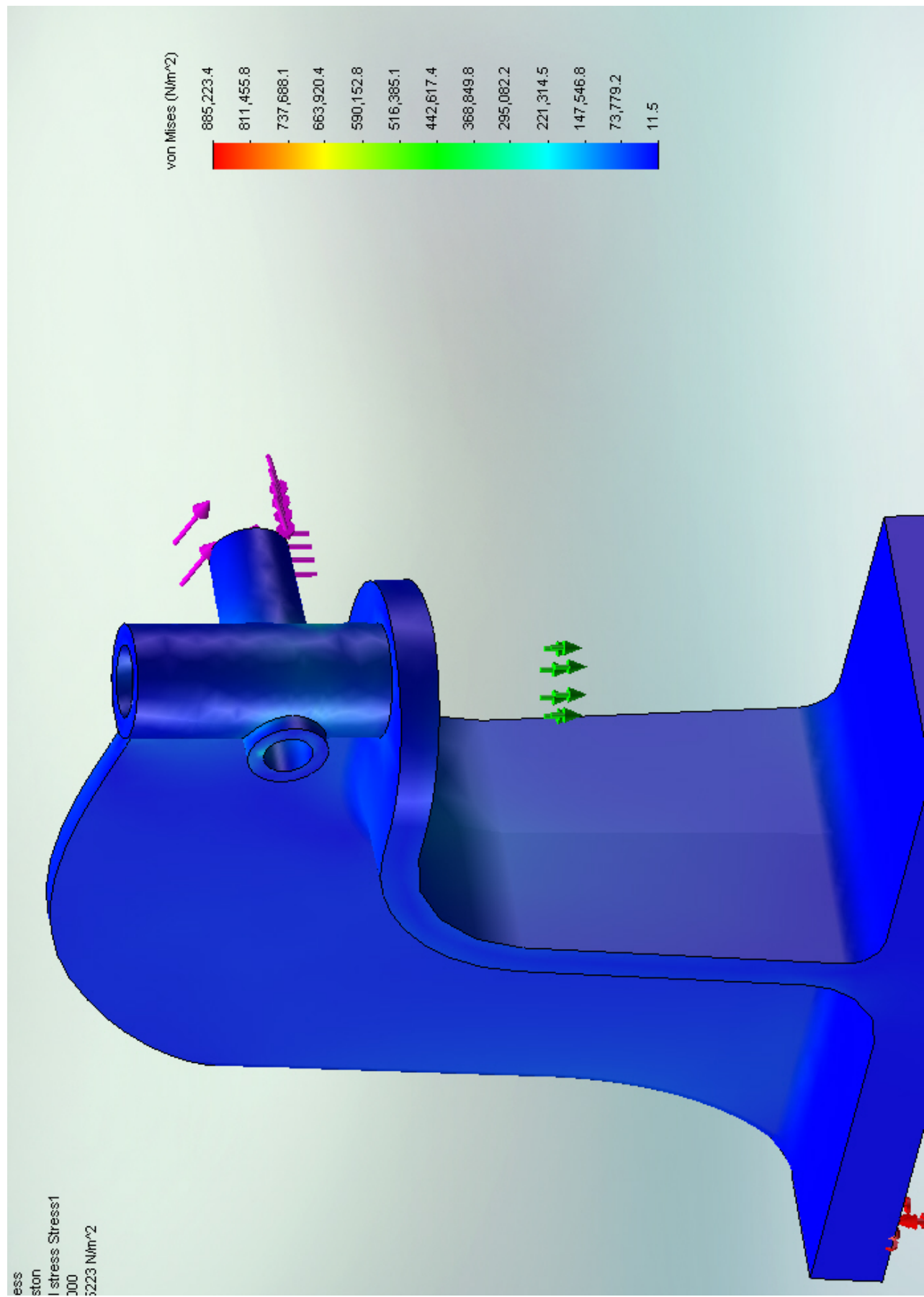
Test 4: Torque on Pinion – Nonrigid Pinion and Piston

von Mises (N/m^2)

55,817,976.0
51,166,604.0
46,515,232.0
41,863,860.0
37,212,488.0
32,561,116.0
27,909,744.0
23,258,372.0
18,607,000.0
13,955,627.0
9,304,255.0
4,652,882.5
1,510.6
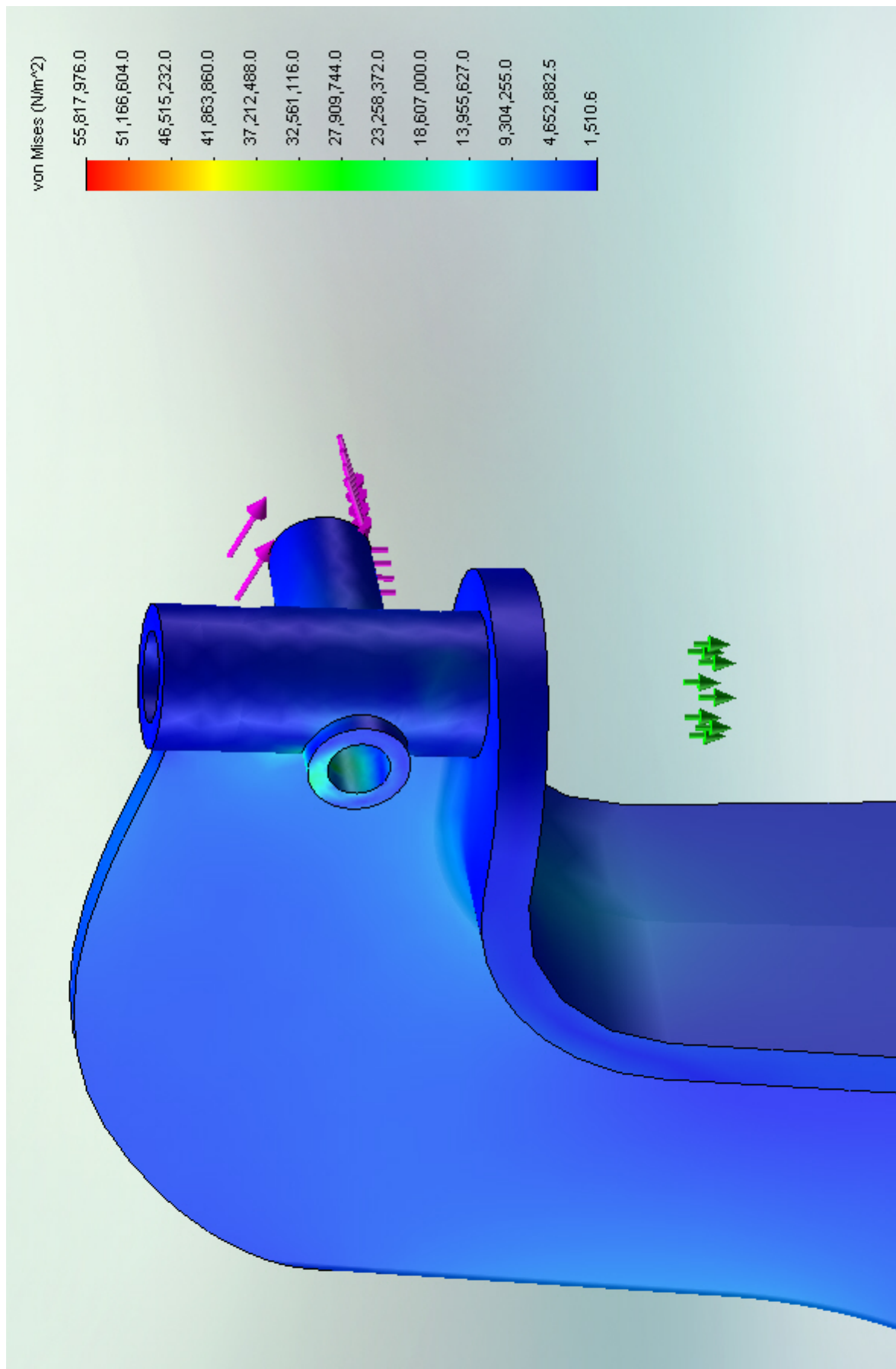
Model name: ArborPress
Study name: Fixed Piston
Plot type: Static nodal stress Stress2
Deformation scale: 1
Global value: 0.245023 to 3.04085e+008 N/m^2

von Mises (N/m^2)

304,085,280.0
278,744,832.0
253,404,400.0
228,063,968.0
202,723,520.0
177,383,088.0
152,042,640.0
126,702,200.0
101,361,760.0
76,021,320.0
50,680,880.0
25,340,440.0
0.2
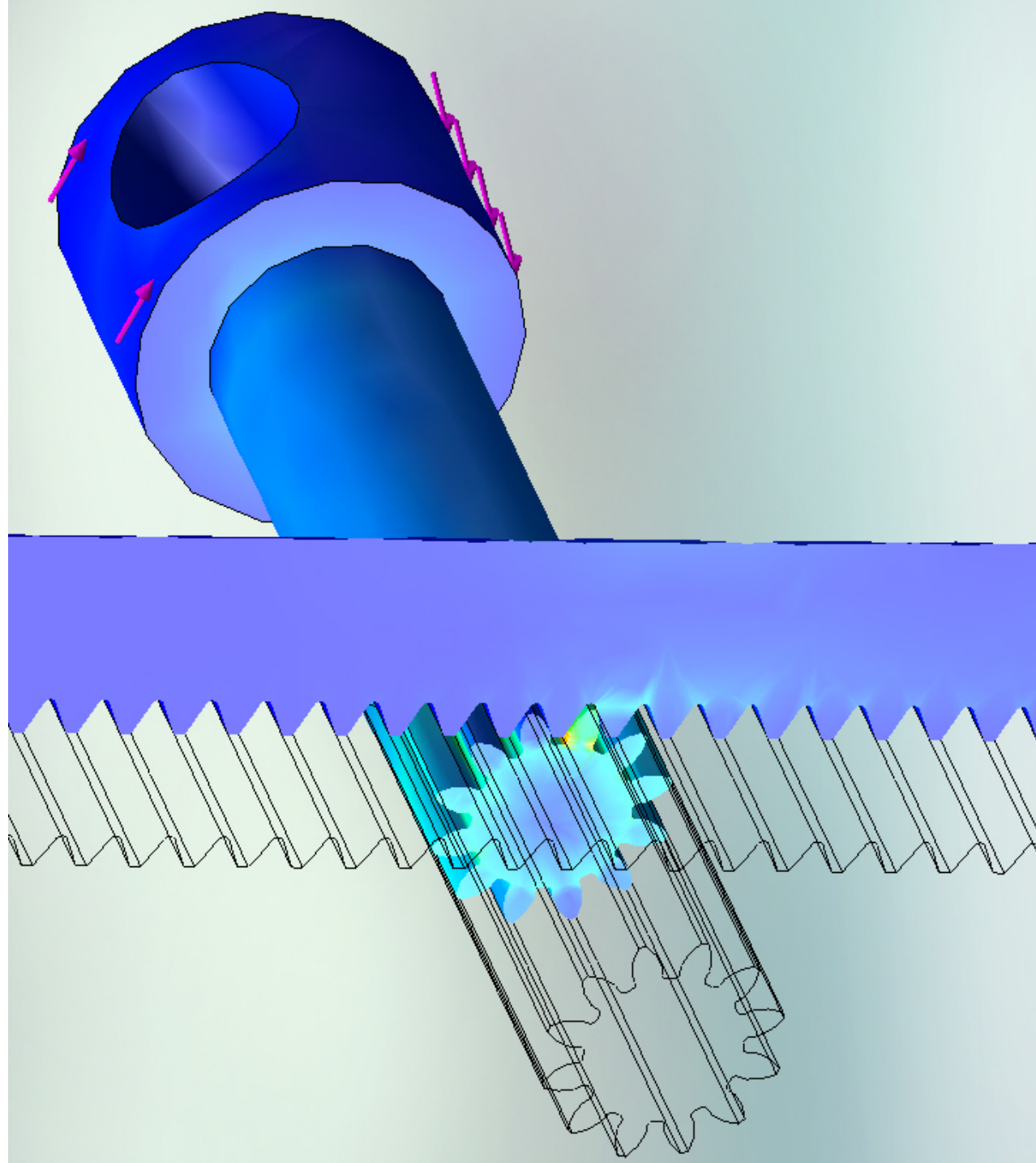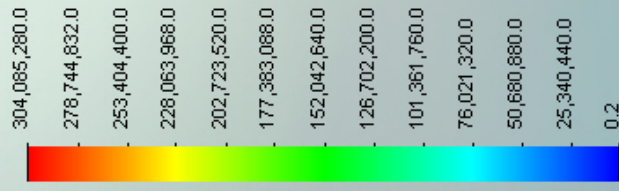
Test 5: Frame Solo Test

Model name: ArborPress
Study name: Frame
Plot type: Static nodal stress Stress1
Deformation scale: 1292.18

von Mises (N/m^2)

40,641,548.0
37,254,844.0
33,868,140.0
30,481,434.0
27,094,730.0
23,708,026.0
20,321,320.0
16,934,616.0
13,547,912.0
10,161,207.0
6,774,502.5
3,387,798.0
1,093.4

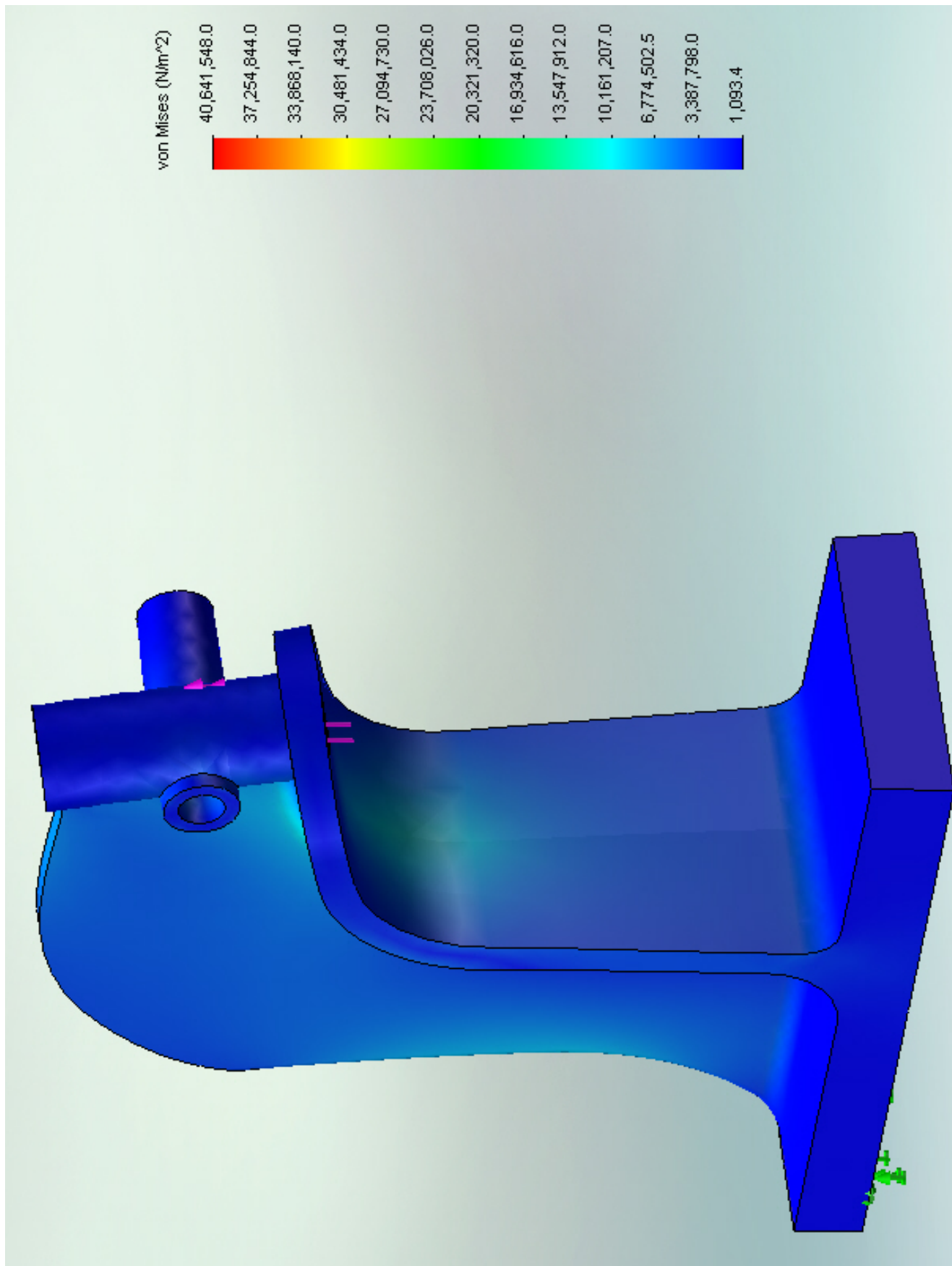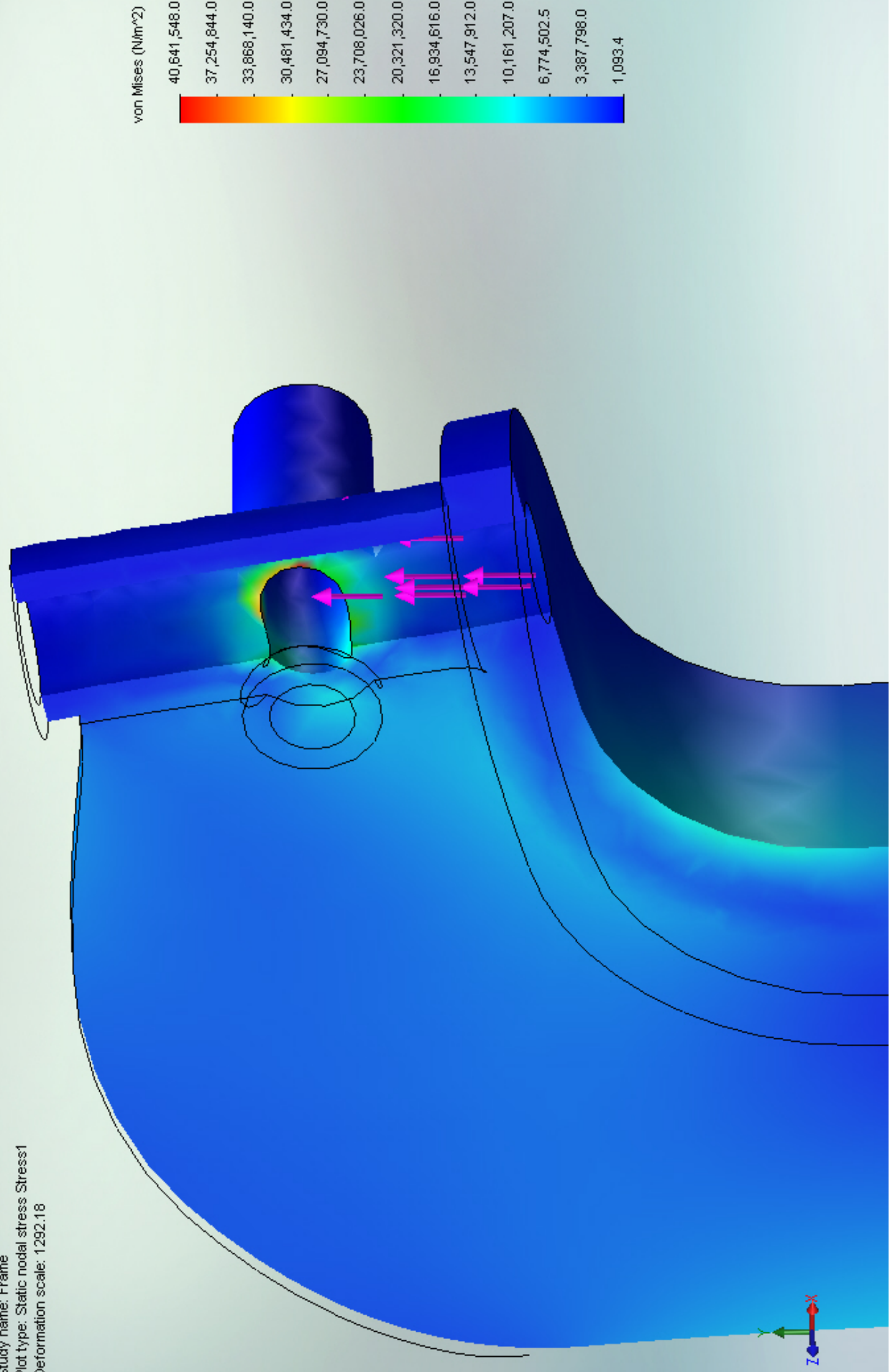**Test 6:** Pinion and Piston Solo Test

Model name: ArborPress
Study name: Pinion and Piston
Plot type: Static nodal stress Stress2
Deformation scale: 1
Global value: 8.31743 to 5.4194e+009 N/m^2

von Mises (N/m^2)

5,419,404,800.0
4,967,787,520.0
4,516,170,752.0
4,064,553,728.0
3,612,936,448.0
3,161,319,424.0
2,709,702,400.0
2,258,085,376.0
1,806,468,224.0
1,354,851,200.0
903,234,112.0
451,617,088.0
8.3

von Mises (N/m^2)

2,995,050,496.0
2,745,463,040.0
2,495,875,328.0
2,246,287,872.0
1,996,700,288.0
1,747,112,832.0
1,497,525,248.0
1,247,937,664.0
998,350,144.0
748,762,624.0
499,175,104.0
249,587,552.0
8.3

Model name: ArborPress
Study name: Pinion and Piston
Plot type: Static nodal stress Stress3
Deformation scale: 1
Global value: 8.31743 to 5.4194e+009 N/m^2