Evaluating Stresses and Forces in Fasteners

Part 2

Review of Part 1 and Correction

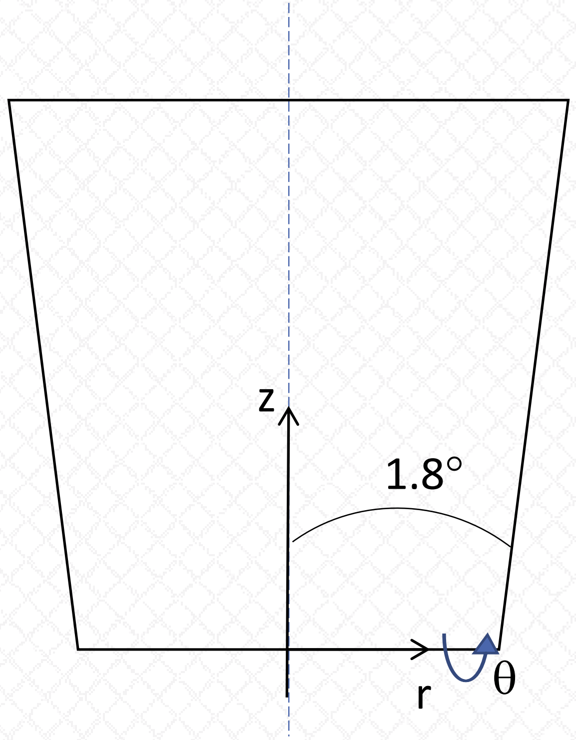
- In Part 1, we analyzed the stresses induced by screwing a 3/8-18 NPT brass male fitting into a mating cast Aluminum part
- We discovered that the mean thread stresses agreed quite closely with hand calculations of stresses across the thread effective shear area (the Ats parameter of slide 7, taken from https://mechanicalc.com/reference/bolted-joint-analysis#torque). This strongly suggests that a full finite element model of mechanical parts which include thread detail is unnecessary –even when thread failure is a primary concern
- It should be sufficient to simply estimate thread stress safety factors with standard hand estimates (like those in the link above), and instead create simplified finite element models which eliminate thread detail, but incorporate loads resulting from fastener tightening (preloading). An example of such an approach will be the focus of this presentation
- CORRECTION TO PART 1: slide 12 of part 1 described the applied screw displacement as shown below. This was a mistake. Each quarter turn was instead accompanied by a corresponding ¼ full thread advance (0.01389”)

used simply to establish the initial thread contact. Each subsequent load step turns the brass fitting by 90°, and translates it axially by one full thread (0.05556”).

- An axial load of 1 lbf is applied at load step 1 –just for added stability during

Simplified model

- We refer readers to part 1 for the geometry of the tapered fastener. For the present investigation, the important fact is that a 3/8-18 NPT fastener has a taper angle of 1.8° (more precisely, this is the half angle "between the taper and the center axis", as described here: https://en.wikipedia.org/wiki/National_pipe_thread)

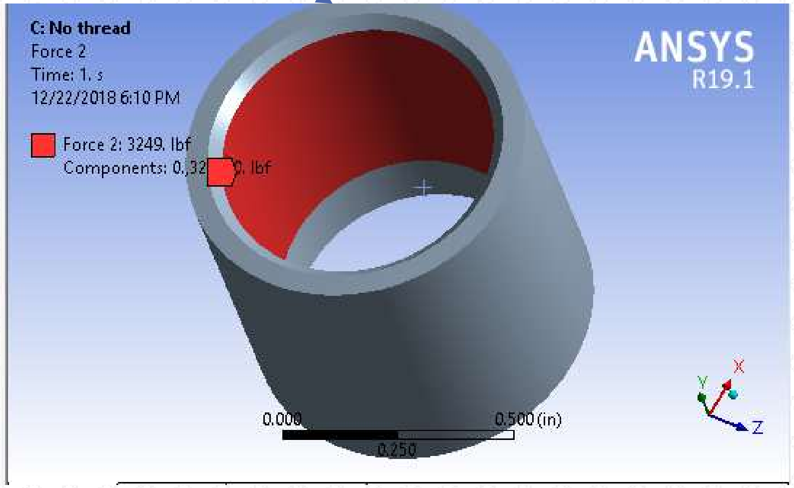


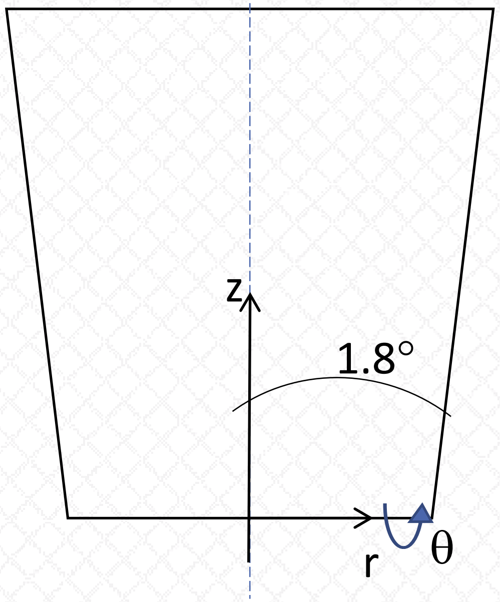
- We want to create a simplified model of the "at-risk" part (the female component) which eliminates the thread detail, but resolves the forces applied by the male fitting. Such a simplified version of the threaded region is shown at the left
- So, to begin, we'll take a look at the force balance of the original threaded part and that of the simplified part

Simplified model

- We already know how to calculate the axial thread force (this was done in part 1). The question before us is: Can we apply this axial load to the simplified thread surface and expect this applied load to reasonably approximate the stress-inducing loads applied to the female part by the actual male threaded part? So far, we have only a vague intuition that this should be the case. To test this, we'll have to take a more detailed look at the force balance between the threaded parts
- We can also apply the torque if necessary. But what about the radial force components (which surely exist)?
- Before we begin, It is important to remember that we are no longer interested in what happens at the threads. Rather, we are concerned about stresses in the rest of the part when tightening this fastener, so we're after a simplified representation that applies the same relevant forces.

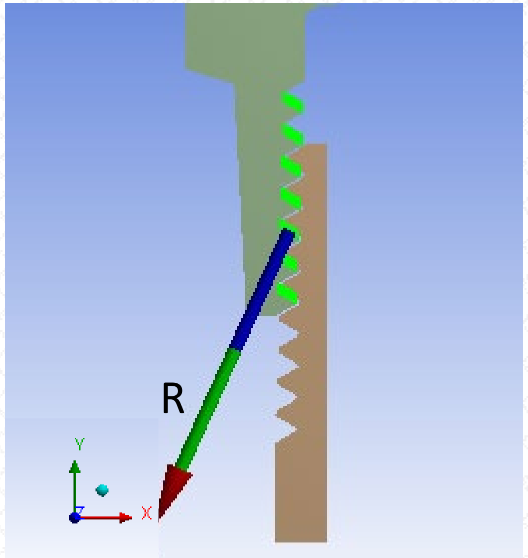
Threads removed –Tensile load applied…



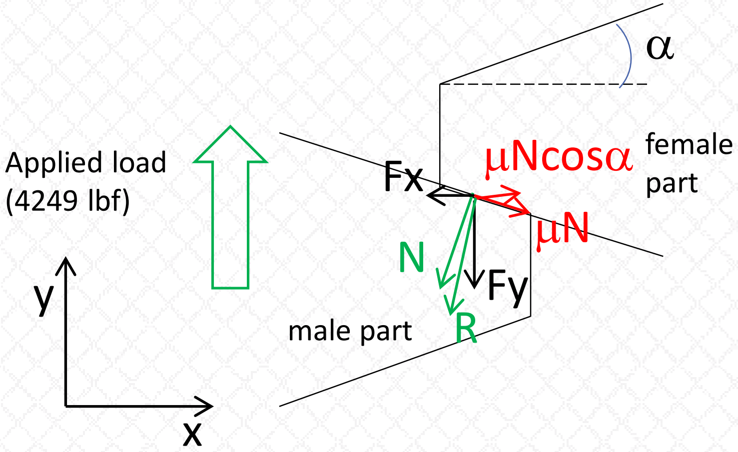


Force Balance: A similar 2D example

- To better understand the force balance in the NPT threaded assembly, it's easier to look at a 2D axisymmetric model. The axisymmetric model has the correct thread profile, but no helix angle. There is no way to "turn" the male part to induce the correct torque and associated forces, so we simply analyze the component in the correct position (after two full turns) with an external applied load of 3249 lbf (the axial load we ended up with in part 1).
- In what follows, we estimate an effective radial force component, assuming that the effective thread force reaction components may be simply resolved along the thread angle (in general, this will not be true because the reaction force is not evenly distributed, but it should get us close)



The resultant reaction force, R on the male part



$$F_x = \mu N\cos\alpha - N\sin\alpha \qquad (1)$$

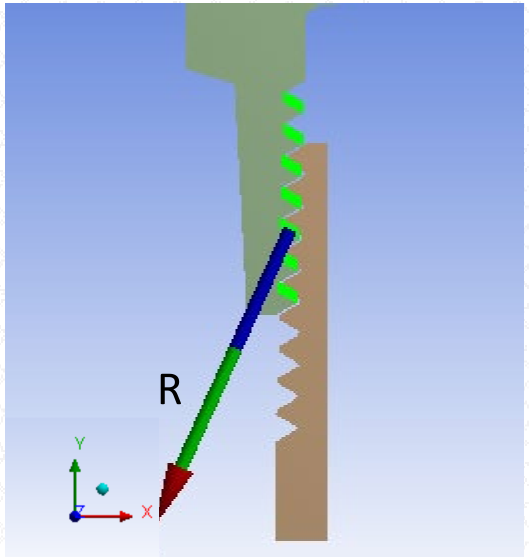
$$F_y = -\mu N\sin\alpha - N\cos\alpha \qquad (2)$$

Force Balance: A similar 2D example

- From part 1, we know that for a 3/8-18 NPT, the thread half-angle α = 30°. We also know that Fy = 3249 lbf. From this, we make the following estimate of the radial force, Fx and normal force, N by combining (1) and (2):

N = 3546.85 lbf

Fx = -1466.3 lbf (calculated by combining (1) and (2))



Radial force component

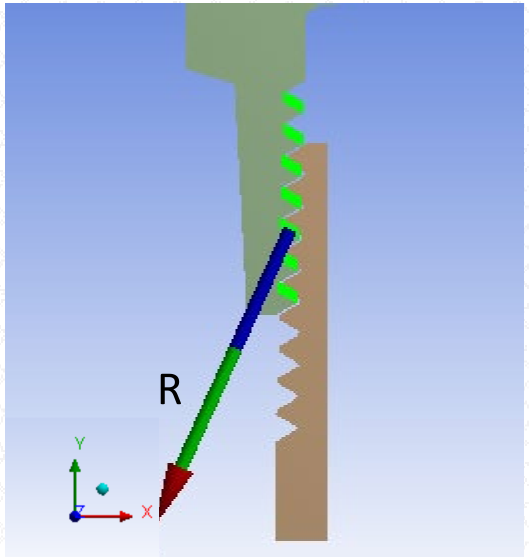
Time [s]	Force Reaction 3 (X) [lbf]	Force Reaction 3 (Y)
1.	-283.77	-389.23
2.	-1487.8	-3249.

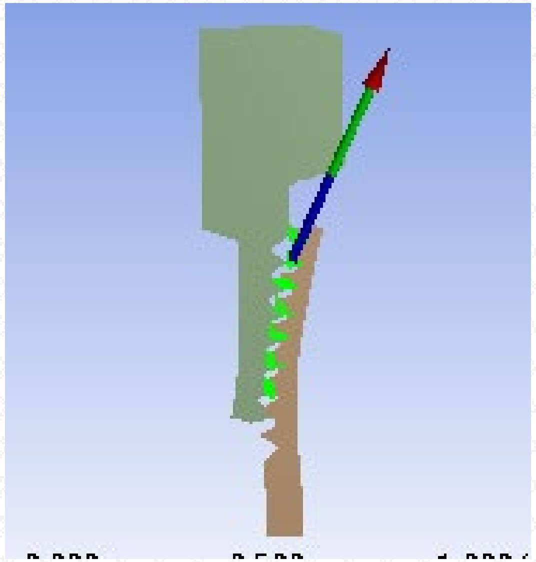
Compare the last time point

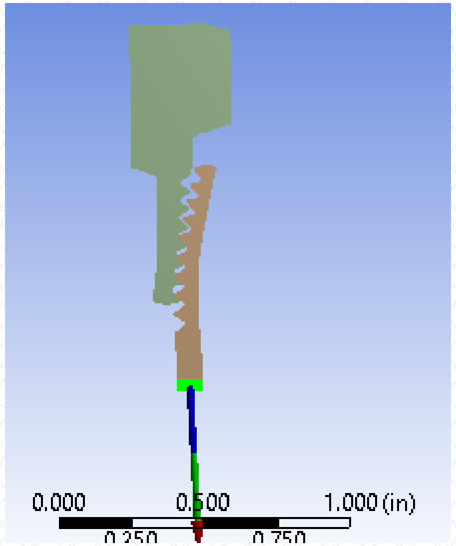
- We can compare this estimate to the actual reaction forces calculated by ANSYS (above, for the 2D example). The estimate is pretty good, considering all the simplifying assumptions that have been made
- Already, this force reaction on the male part tells us a lot about the underlying mechanics of this fastener system. To learn more, we turn to the female part

Force Balance: A similar 2D example

- The left figure below shows the reaction force on the male threads
- The middle figure shows the reaction force at the corresponding threads of the female part
- The last figure on the right shows the reaction force at the fixed boundary condition of the female part
- Note the relative magnitude of the radial component of force at the threads compared with that at the fixed boundary (the two figures on the left, vs. the one on the right). Some readers may wonder what happened to the radial force component in the female part between the middle and last figure (why isn't x-component in the middle figure balanced by the boundary condition in the figure on the right?)







Time [s]	Force Reaction 3 (X) [lbf]	Force Reaction 3 (Y)
1.	-283.77	-389.23
2.	-1487.8	-3249.

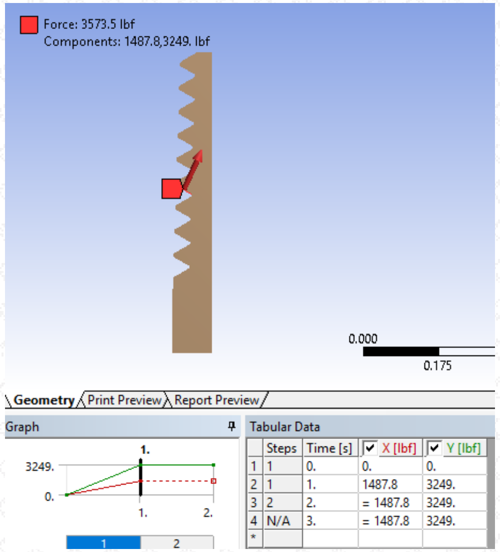
Time [s]	Force Reaction 3 (X) [lbf]	Force Reaction 3 (Y)
1.	283.81	389.32
2.	1487.8	3249.

Time [s]	Force Reaction 2 (X) [lbf]	Force Reaction 2 (Y)
1.	21.187	-389.32
2.	176.66	-3249.

Force Balance: A similar 2D example

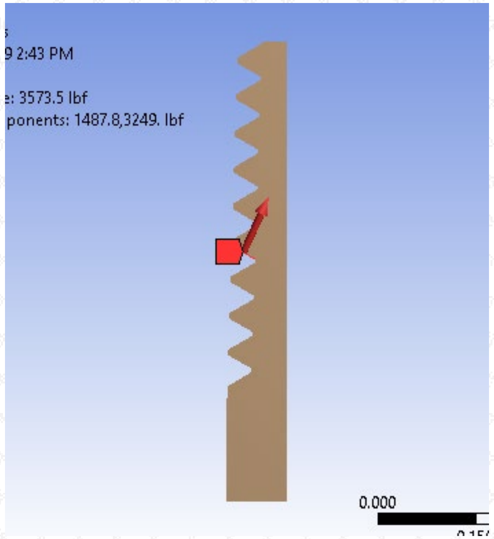
- To answer the last question, we must understand that a symmetric cylinder IS STATICALLY INDETERMINATE IN ANY PLANE WHICH CONTAINS THE AXIS (it is actually over-constrained)
- Another way of putting this is: A purely radial load will result in a deflection of an axisymmetric part WITHOUT ANY BOUNDARY CONDITIONS (thus making the addition of any boundary condition statically redundant)
- It remains only to determine what becomes of this radial load, and to convince ourselves that it's contribution to the female part's structural response may be neglected at sufficient distances from the thread resultant load (recalling our goal. See slide 4)
- A detailed mathematical treatment of this problem is beyond the current scope. Instead, we hope to convince readers with a more qualitative approach using ANSYS. We'll begin by applying the resultant thread load on the female part only

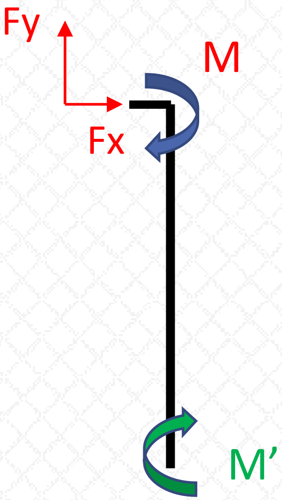
Female part only: resultant thread load applied at a single tooth

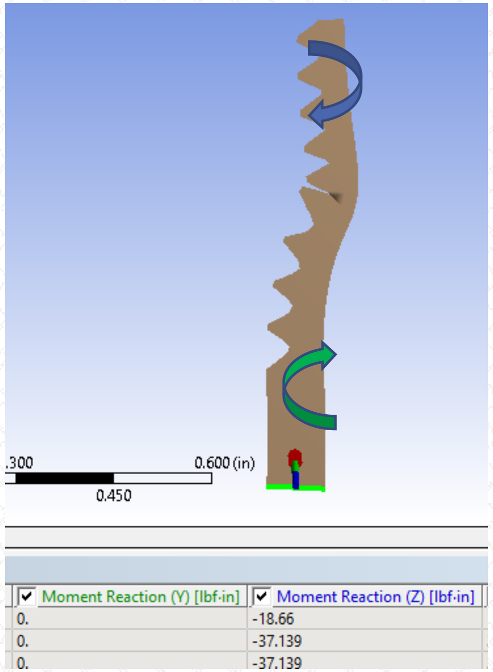


Force Balance: A similar 2D example

- A static force-equilibrium diagram for this system is seen in the middle diagram. In the axisymmetric system, the applied moment M and the resultant radial load are not balanced due to the static indeterminacy of this system.
- For our purposes, the practical consequence of this is that these loads behave like point loads in an elastic medium. That is, their affect on stress diminishes in inverse proportion to the square of the distance from the load (St. Venant's Principle). A quick estimate of the applied moment load M = 320 in-lbf*, but little over 10% of this value is actually reacted out at the fixed boundary (lower right)
- This is in stark contrast to the axial load (Fy) which IS statically determinate (must be reacted out in its entirety)



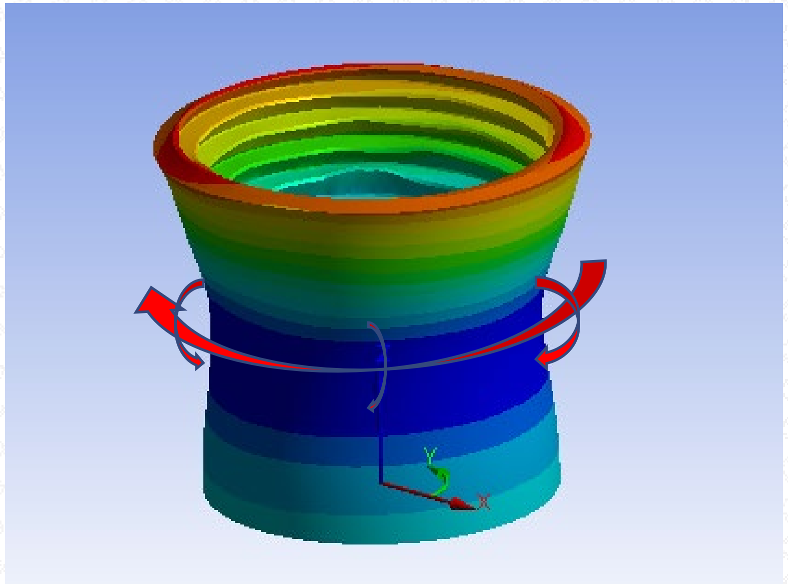




*On a "per $2\pi r$ basis"

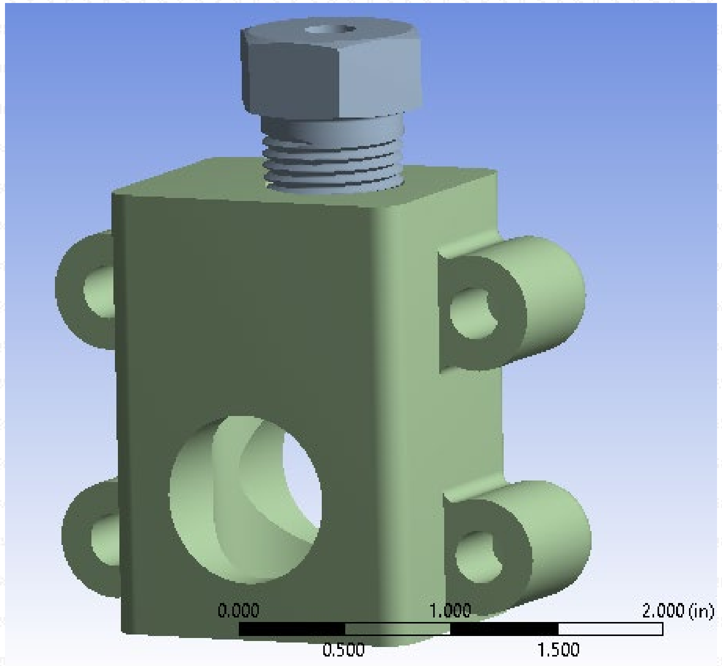
Conclusion (?)

- We're now in a position to understand the somewhat counter-intuitive structural response of the female part seen in part 1
- The resultant thread load on the female part causes a local circumferential twist
- The radial load component contributes overwhelmingly to this response and is largely unreacted at fixed boundaries
- Since this behavior is local (confined to the thread region only), we thus claim that this moment load, and the radial force contributing to it may be neglected in developing a statically equivalent model which ignores the thread region
- The observation is actually more general: Because of the statically indeterminate nature of radial loads in a cylinder, such loads may in general be ignored in elastic regions whose distance, R from the hole of a radius r is such that R>>r

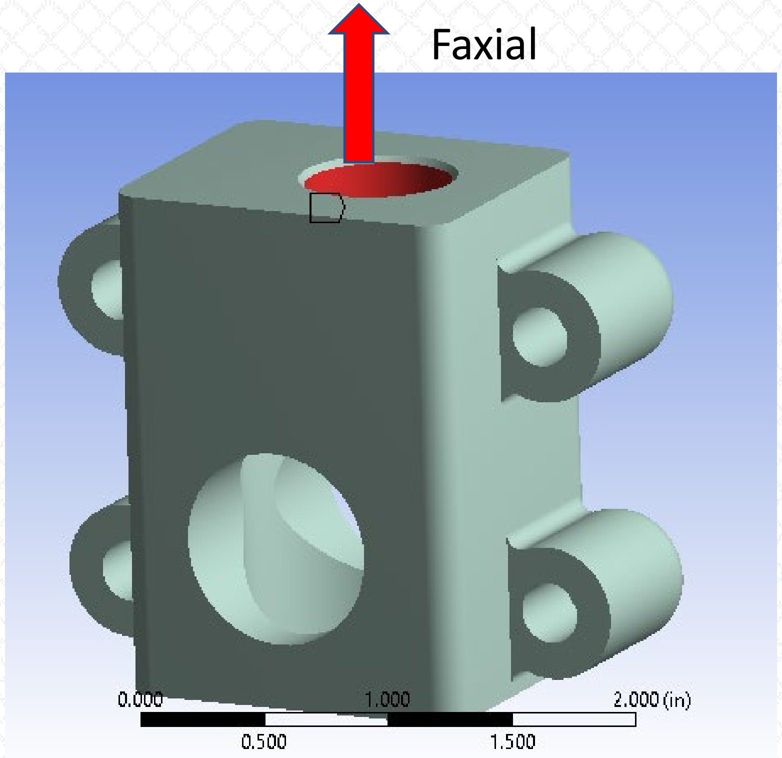


Test: An Example (two cases)

- We now want to test how well our conclusion holds on an arbitrary test example. We've imagined some hypothetical hydraulic component with an NPT fitting to see how well the simplifying assumptions we've developed hold up under scrutiny
- We'll analyze two cases, shown below. The first, analyzes the part with the full thread detail (and male component) retained. We then remove the thread detail and male part, while retaining the hole taper…



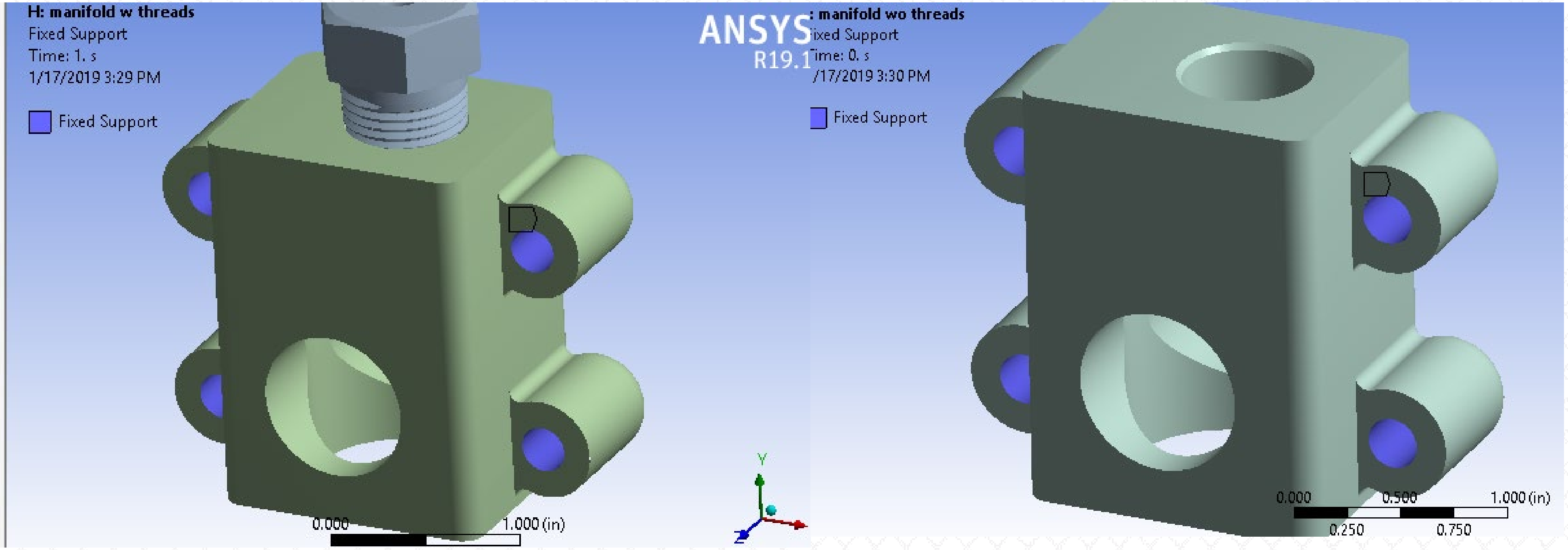
Case A: Full thread model



Case B: Simplified Model –Axial thread load

Test: An Example (Boundary Conditions)

- In each case, the boundary conditions are shown below (fixed at the mounting hole locations)

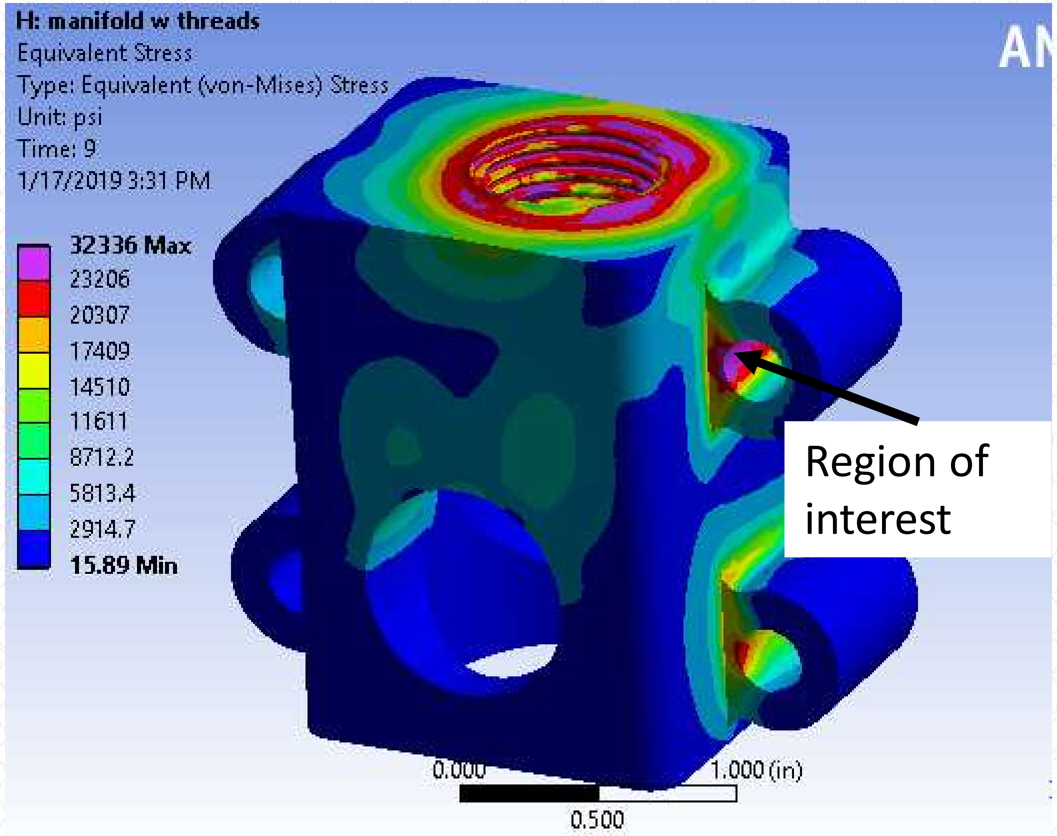


Case A: Full thread model

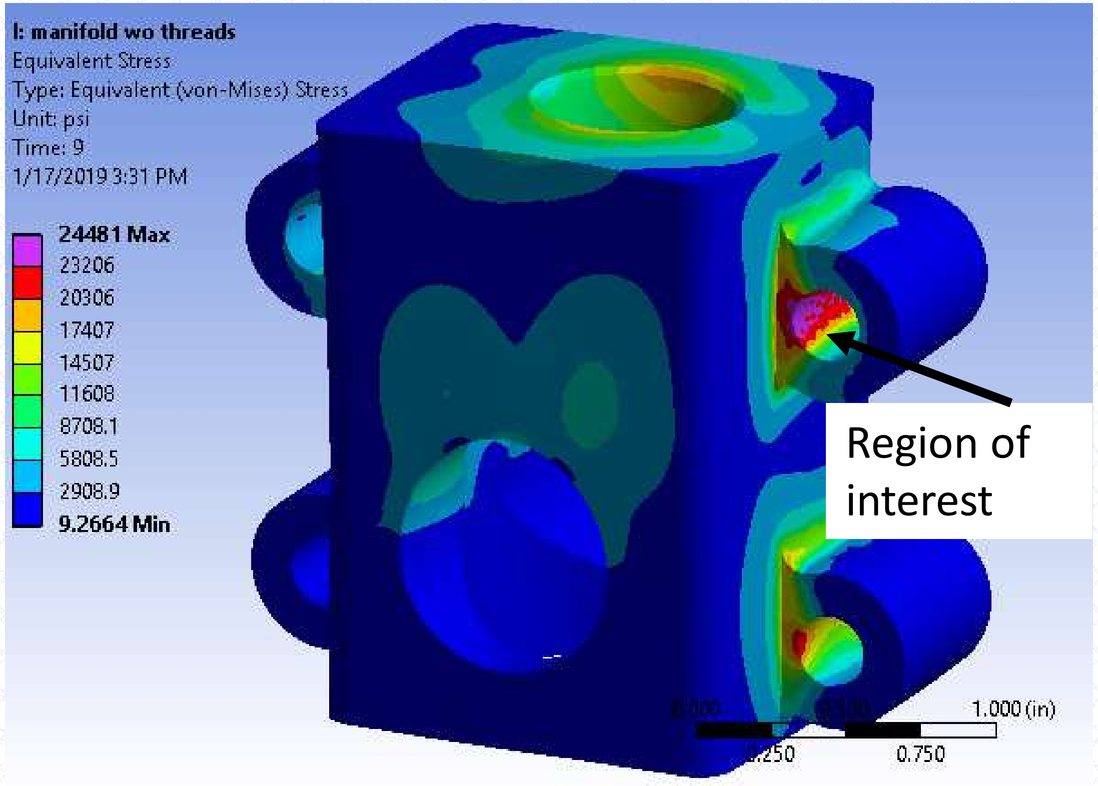
Case B: Simplified Model –Axial thread load

Test Case Results

- After turning the NPT fitting two full turns using the same technique outlined in Part 1, we see the following stress in the region of interest. The agreement is very good at R >> r as expected
- Once again, the stress contours are capped at the material yield stress (purple regions yield)



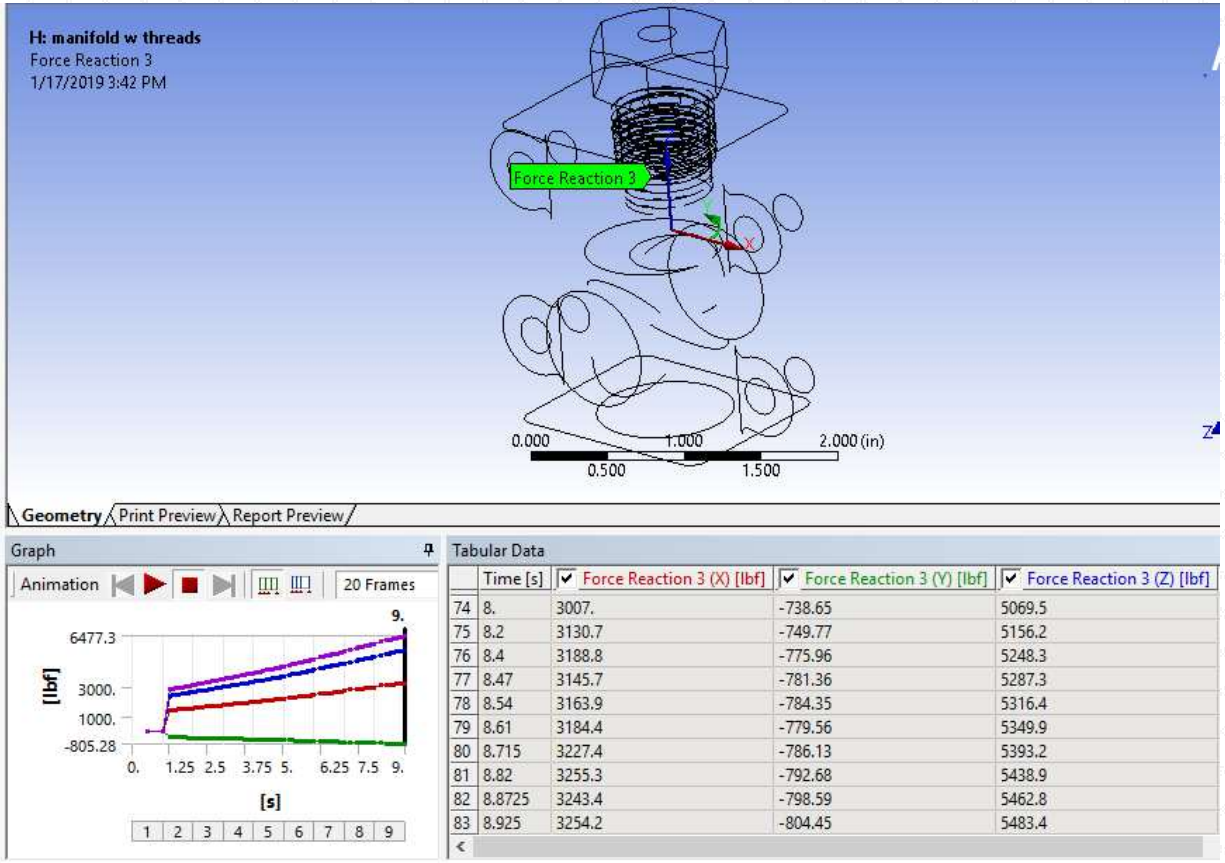
Case A: Full thread model



Case B: Simplified Model –Axial thread load

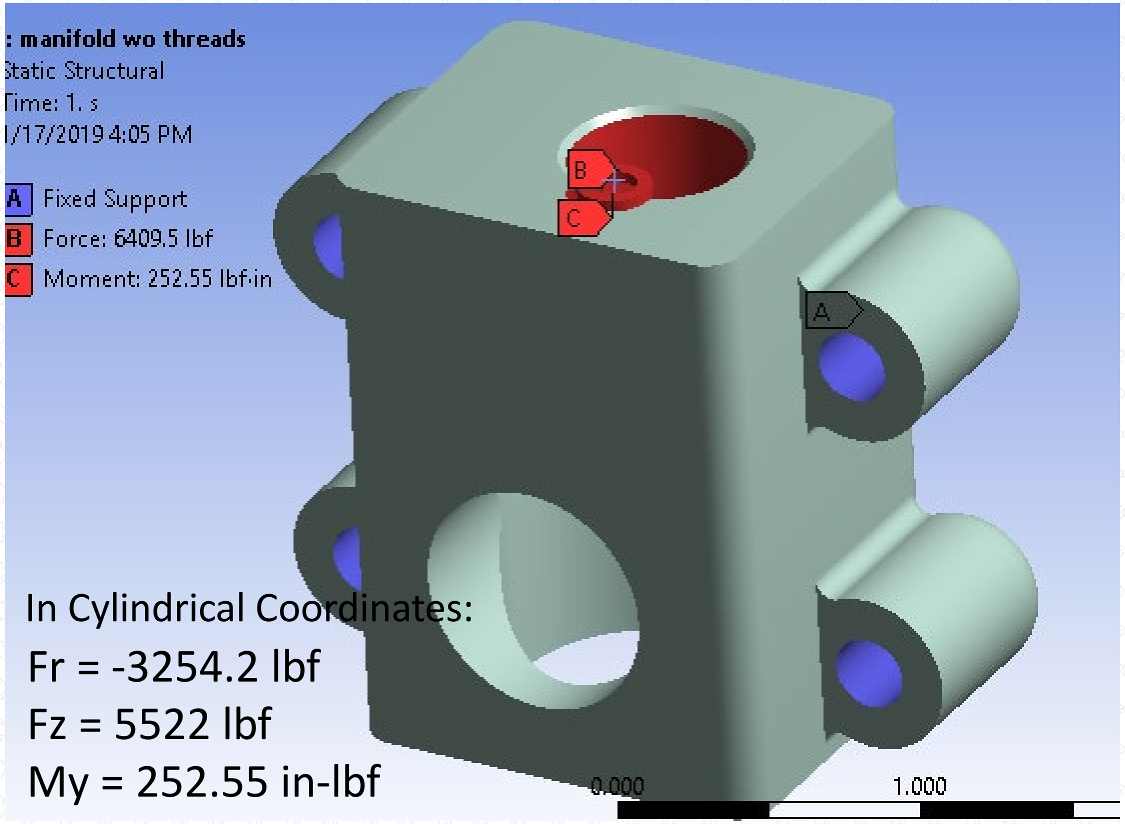
Confirmation and Lessons Learned

- To confirm further confirm our intuition, lets' just take the resultant thread reaction force and apply it in it's entirety to the simplified model –including all the details, such as radial force and torque
- We expect nothing much to change…



	Time [s]	Force Reaction 3 (X) [lbf]	Force Reaction 3 (Y) [lbf]	Force Reaction 3 (Z) [lbf]
74	8.	3007.	-738.65	5069.5
75	8.2	3130.7	-749.77	5156.2
76	8.4	3188.8	-775.96	5248.3
77	8.47	3145.7	-781.36	5287.3
78	8.54	3163.9	-784.35	5316.4
79	8.61	3184.4	-779.56	5349.9
80	8.715	3227.4	-786.13	5393.2
81	8.82	3255.3	-792.68	5438.9
82	8.8725	3243.4	-798.59	5462.8
83	8.925	3254.2	-804.45	5483.4

Case A: Full thread model



In Cylindrical Coordinates:

Fr = -3254.2 lbf

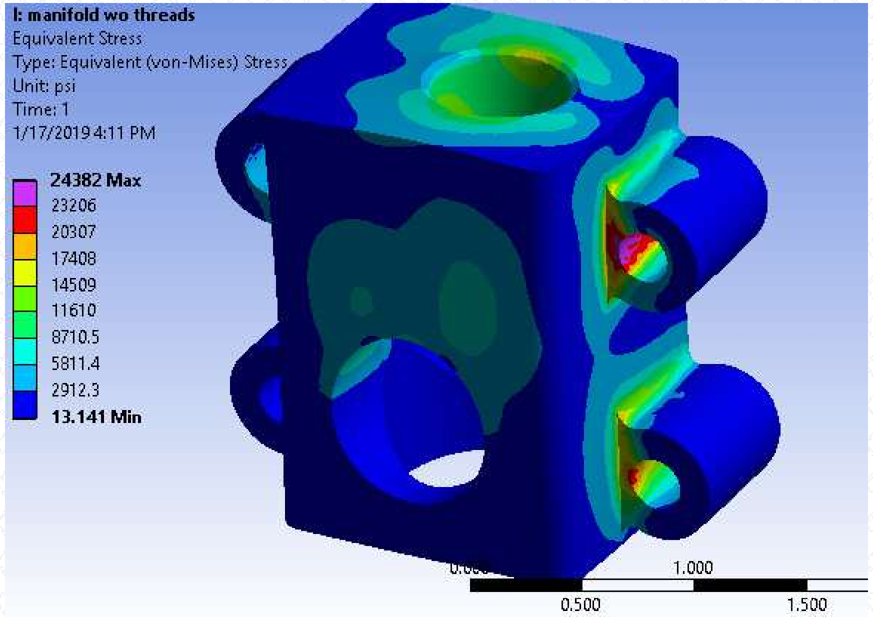
Fz = 5522 lbf

My = 252.55 in-lbf

Case B: Simplified Model –Axial thread load

Confirmation and Lessons Learned

- These results are so close that we conclude our intuition about the thread reaction forces was correct: For most cases, we can simply ignore the applied moment (which, although must be reacted, is typically too small to cause structural problems away from the thread region) and the thread radial reaction force
- The major discovery here is that the thread radial reaction, although substantial, is essentially confined to the thread region due to it's statically indeterminate nature (nothing at distances R >> 2 "feels" this load)

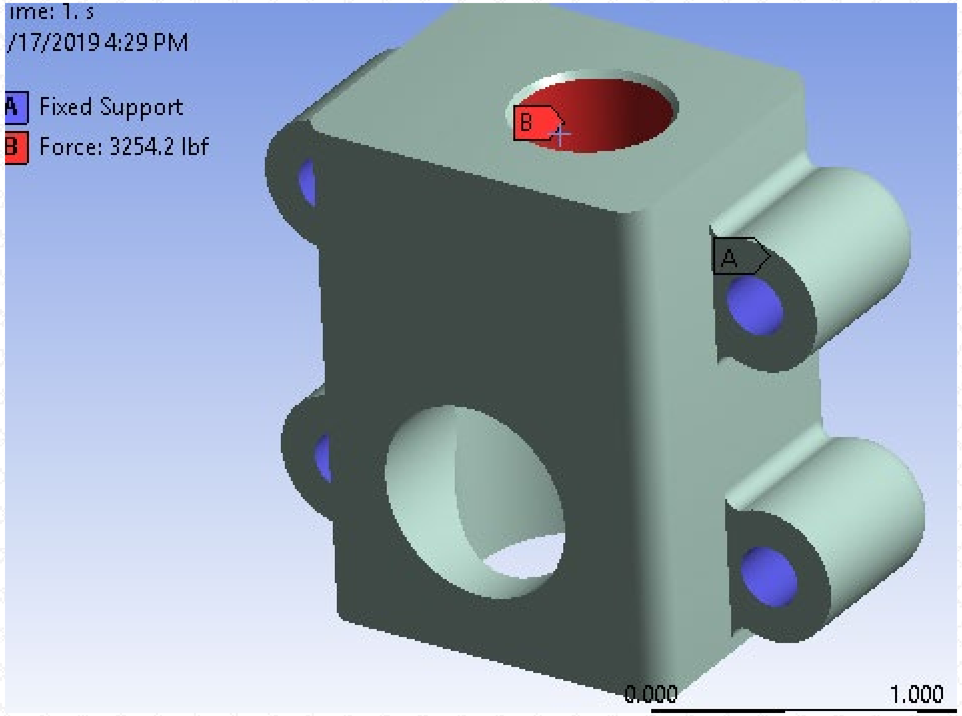


Simplified Model with all thread reactions applied to simplified thread surface

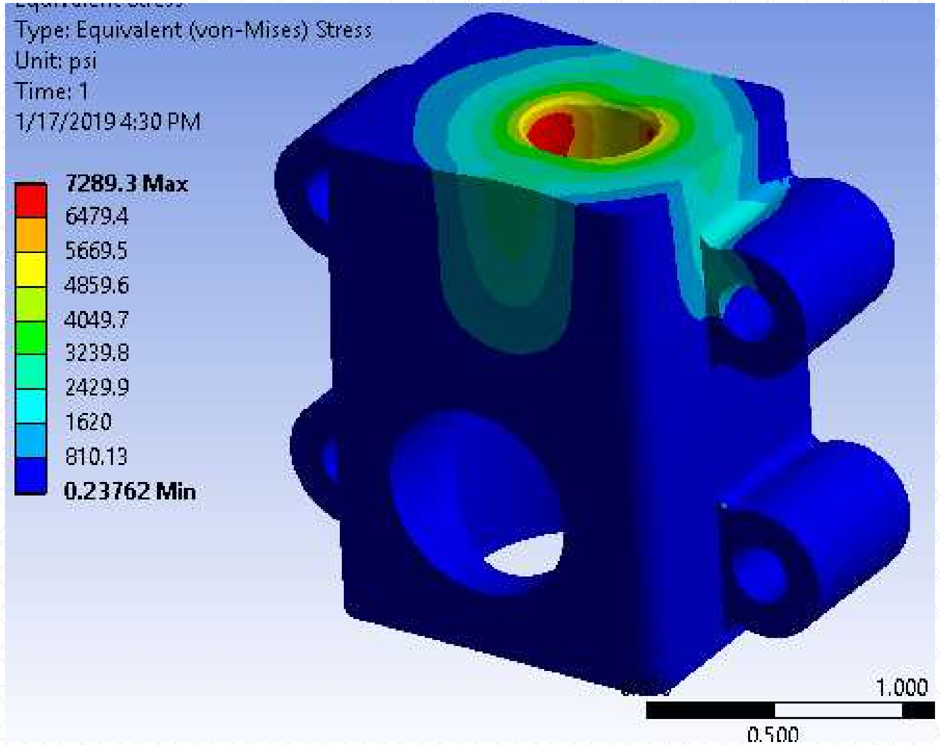
- This last discovery is good news, because it confirms what we find in the literature: Radial thread force reactions are never calculated. They simply aren't an engineering concern anywhere other than the threads themselves
- This means we don't have to calculate this force

Confirmation and Lessons Learned

- For final confirmation, let’s just look at the un-altered stress contours for the simplified system with only the radial load applied



Simplified Model with radial thread load only: Fr = -3254.2 lbf



- See how the max stress is not only low, but all stresses are die off as R >> r (our region of interest is roughly R = 3r)

Conclusions

- In this study (part 2 of 2), we sought only to find the simplest reasonable way to approximate the load imparted to an elastic medium by a tightened NPT fitting.
- Our conclusion was that all we really need to calculate is the thread axial (or pretension) load on the part, according to methods freely available in textbooks and online (see part 1), and apply this load to a simplified (surface) representation of the thread region
- The study showed that the rather substantial radial component of such loading may safely be ignored for all features at distances R>>r, both due to relatively low maximum stress and localization (a.k.a St. Venant's Principle)