Estimation of Thread Stresses in 3/8" and 1/2" Ferrule Type Compression Tube Fittings for Knechtion Repair

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PROBLEM STATEMENT

- Knechtion Repair asked Winston Machinery Services (WMS) to provide a stress estimation as to the expected shear stresses resulting in the threads of a 3/8” and 1/2” ferrule compression type tube fitting and nut.

- WMS performed this calculation using classical methods available from literature within the public domain.

- The analysis conducted assumes that tightening the connection per the various manufacturers’ recommendations does not yield the threads making up the connection.

- The analysis conducted assumes the components making up the connection to be concentric with no eccentricity.

- The analysis conducted pertains only to 304 and 316 stainless steels.

METHODOLGY

- Using ASME B31.3 allowable stresses for stainless steel tubing, a maximum pressure load was determined for various wall thicknesses for the tubing diameters of interest.

- Pressure loads were then considered to be acting on the silhouette of the internal cross-section from which a nominal axial load acting parallel to a straight tube's axis was determined.

- To determine a worst-case loading condition, nominal tubing dimensions as well as minimum and maximum tubing tolerances were considered.

- Using equations presented in Bickford and Nassar's "Handbook of Bolts and Bolted Joints" the thread shear area was determined for a given number of threads engaged and resulting shear stresses determined.

To calculate a design wall thickness, ASME B31.3 provides the following equations

$$t = \frac{PD}{2(SEW + PY)} \qquad \text{(ASME B31.3 Eq. 3a)}$$

$$t = \frac{P(d + 2c)}{2[SEW - P(1 - Y)]} \qquad \text{(ASME B31.3 Eq. 3b)}$$

Where

$$P = applied\ internal\ pressure\ (psi)$$

$$D = nominal\ outside\ diameter\ (in)$$

$$S = stress\ value\ for\ material\ as\ taken\ from\ B31.3\ Table\ A-1$$

$$E = quality\ factor\ from\ Table\ A-1B$$

$$W = weld\ joint\ strength\ reduction\ factor$$

$$Y = Coefficient\ from\ B31.3\ Table\ 304.1.1$$

$$d = inside\ diameter\ of\ tube\ (in)$$

$$c = sum\ of\ mechanical\ allowances$$

Figure 1: ASME B31.3 Wall Thickness Calculations

Rearranging B31.3 Equation 3a to solve for pressure yields

$$P = \frac{SEW}{\frac{D}{2t} - Y}$$

and for B31.3 Equation 3b

$$P = \frac{SEW}{(1 - Y) + \frac{d + 2c}{2t}}$$

Where in all calculations the following values were assumed

$$S = 20{,}000\ psi$$

$$E = 1.0\ for\ seamlesss\ tubing$$

$$W = 1$$

$$c = 0$$

Figure 2: Rearrangement of B31.3 Wall Thickness Calculations

c =	0	sum of mechanical allowances
OD =	0.375	outside diameter (in)
E_s =	1.00	ASME B31.3 Quality Factor ASTM A-269 seamless (See Table A-1B)
E_{fw1} =	0.80	ASME B31.3 Quality Factor ASTM A-269 fusion welded single butt seam (See Table A-1B)
E_{fw2} =	0.85	ASME B31.3 Quality Factor ASTM A-269 fusion welded double butt seam (See Table A-1B)
S =	20,000	ASME B31.3 allowable stress (psi)
W =	1.00	

	Tubing Wall Thickness (in)					
Parameter	0.035	0.049	0.065	0.083	0.095	
d =	0.305	0.277	0.245	0.209	0.185	inside diameter (in)
A_i =	0.073	0.060	0.047	0.034	0.027	internal cross-sectional area (in^2)
Y =	0.400	0.400	0.395	0.358	0.330	coefficient (See ASME 31.3 Section 304.1

Allowable Pressure per ASME B31.3 Equation 3.a using nominal diameter and wall thickness

P_s =	4,035	5,837	8,034	10,520	12,170	*Seamless*
P_{fw1} =	3,228	4,669	6,427	8,416	9,736	*Single fusion welded tubing wall*
P_{fw2} =	3,429	4,961	6,829	8,942	10,345	*Double fusion welded tubing wall*

Allowable Pressure per ASME B31.3 Equation 3.b using nominal diameter and wall thickness

P_s =	4,035	5,837	8,034	10,520	12,170
P_{fw1} =	3,228	4,669	6,427	8,416	9,736
P_{fw2} =	3,429	4,961	6,829	8,942	10,345

From ASME 31.3, Table 304.1, if

$$Y < \frac{D}{6}$$

then

$$Y = 0.4$$

else

$$Y = \frac{d + 2c}{D + d + 2c}$$

where 'c' is the sum of mechanical allowances

Figure 3: Allowable Pressure Calculations for Nominal 3/8" Tubing

c =	0	sum of mechanical allowances
OD =	0.5	outside diameter (in)
E_s =	1.00	ASME B31.3 Quality Factor ASTM A-269 seamless (See Table A-1B)
E_{fw1} =	0.80	ASME B31.3 Quality Factor ASTM A-269 fusion welded single butt seam (See Table A-1B)
E_{fw2} =	0.85	ASME B31.3 Quality Factor ASTM A-269 fusion welded double butt seam (See Table A-1B)
S =	20,000	ASME B31.3 allowable stress (psi)
W =	1.00	

	Tubing Wall Thickness (in)					
Parameter	0.035	0.049	0.065	0.083	0.095	
d =	0.430	0.402	0.370	0.334	0.310	inside diameter (in)
A_i =	0.145	0.127	0.108	0.088	0.075	internal cross-sectional area (in^2)
Y =	0.400	0.400	0.400	0.400	0.383	coefficient (See ASME 31.3 Section 304.1

Allowable Pressure per ASME B31.3 Equation 3.a using nominal diameter and wall thickness

P_s =	2,966	4,253	5,804	7,657	8,893
P_{fw1} =	2,373	3,403	4,643	6,125	7,115
P_{fw2} =	2,521	3,615	4,933	6,508	7,559

Allowable Pressure per ASME B31.3 Equation 3.b using nominal diameter and wall thickness

P_s =	2,966	4,253	5,804	7,657	8,893
P_{fw1} =	2,373	3,403	4,643	6,125	7,115
P_{fw2} =	2,521	3,615	4,933	6,508	7,559

From ASME 31.3, Table 304.1, if

$$Y < \frac{D}{6}$$

then

$$Y = 0.4$$

else

$$Y = \frac{d + 2c}{D + d + 2c}$$

where 'c' is the sum of mechanical allowances

Figure 4: Allowable Pressure Calculations for Nominal 1/2" Tubing

Wall thickness *0.035”* *0.049”* *0.065”* *0.083”* *0.095”*

Nominal tube OD with specified wall thickness

	0.035”	0.049”	0.065”	0.083”	0.095”
Allowable Pressure Force per ASME B31.3 Equation 3.a using nominal diameter and wall thickness					
F_s =	248	359	495	648	749
F_{fw1} =	199	288	396	518	600
F_{fw2} =	211	305	420	551	637
Allowable Pressure Force per ASME B31.3 Equation 3.b using nominal diameter and wall thickness					
F_s =	248	359	495	648	749
F_{fw1} =	199	288	396	518	600
F_{fw2} =	211	305	420	551	637

Nominal tube OD +0.005” with wall thickness -15%

	0.035”	0.049”	0.065”	0.083”	0.095”
Allowable Pressure Force per ASME B31.3 Equation 3.a					
F_s =	206	296	405	534	620
F_{fw1} =	165	237	324	427	496
F_{fw2} =	175	252	344	454	527
Allowable Pressure Force per ASME B31.3 Equation 3.b					
F_s =	206	296	405	534	620
F_{fw1} =	165	237	324	427	496
F_{fw2} =	175	252	344	454	527

Nominal tube OD - 0.005” with wall thickness +15%

	0.035”	0.049”	0.065”	0.083”	0.095”
Allowable Pressure Force per ASME B31.3 Equation 3.a					
F_s =	293	427	586	764	878
F_{fw1} =	235	342	469	611	702
F_{fw2} =	249	363	498	649	746
Allowable Pressure Force per ASME B31.3 Equation 3.b					
F_s =	293	427	586	764	878
F_{fw1} =	235	342	469	611	702
F_{fw2} =	249	363	498	649	746

Figure 5: Axial Force Results 3/8” Tubing As Determined from Allowable Pressure Loads

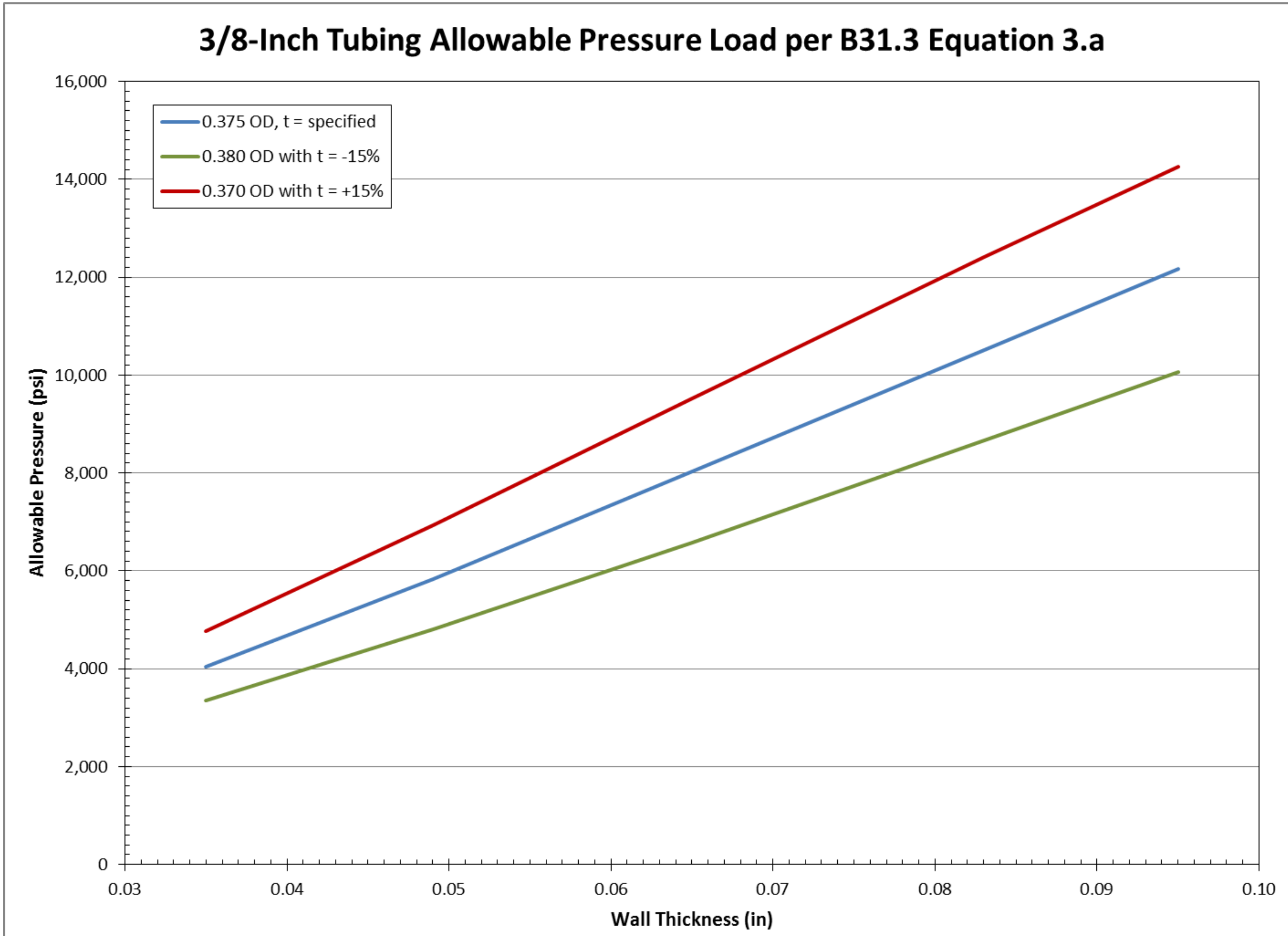

Figure 6: 3/8” Tube Allowable Pressure Loading Per B31.3 per Wall Thickness at a Resulting Operating Stress of 20,000 PSI

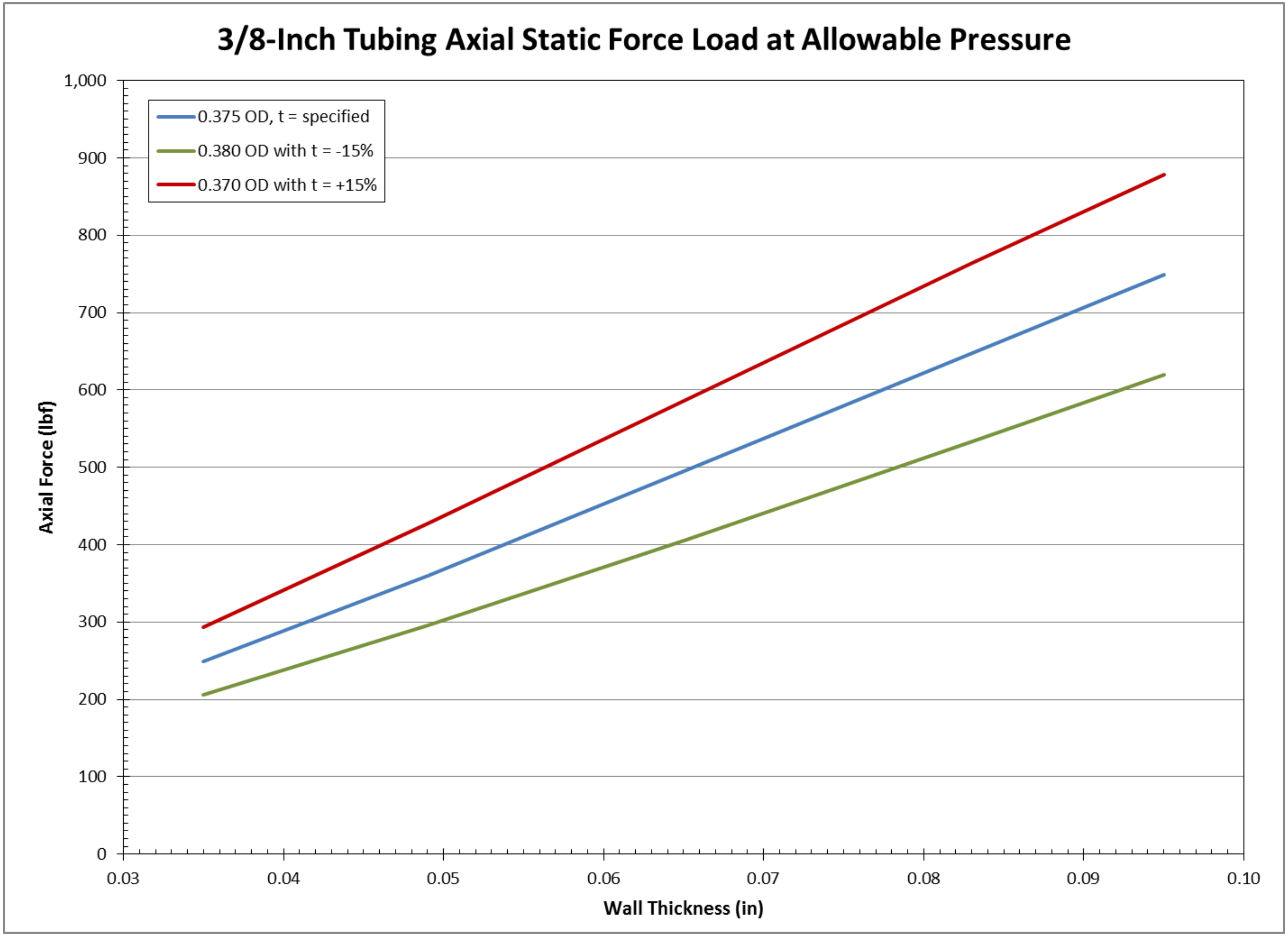

Figure 7: 3/8” Tube Axial Force per Wall Thickness for Allowable Pressure Loading Resulting in an Operating Stress of 20,000 PSI

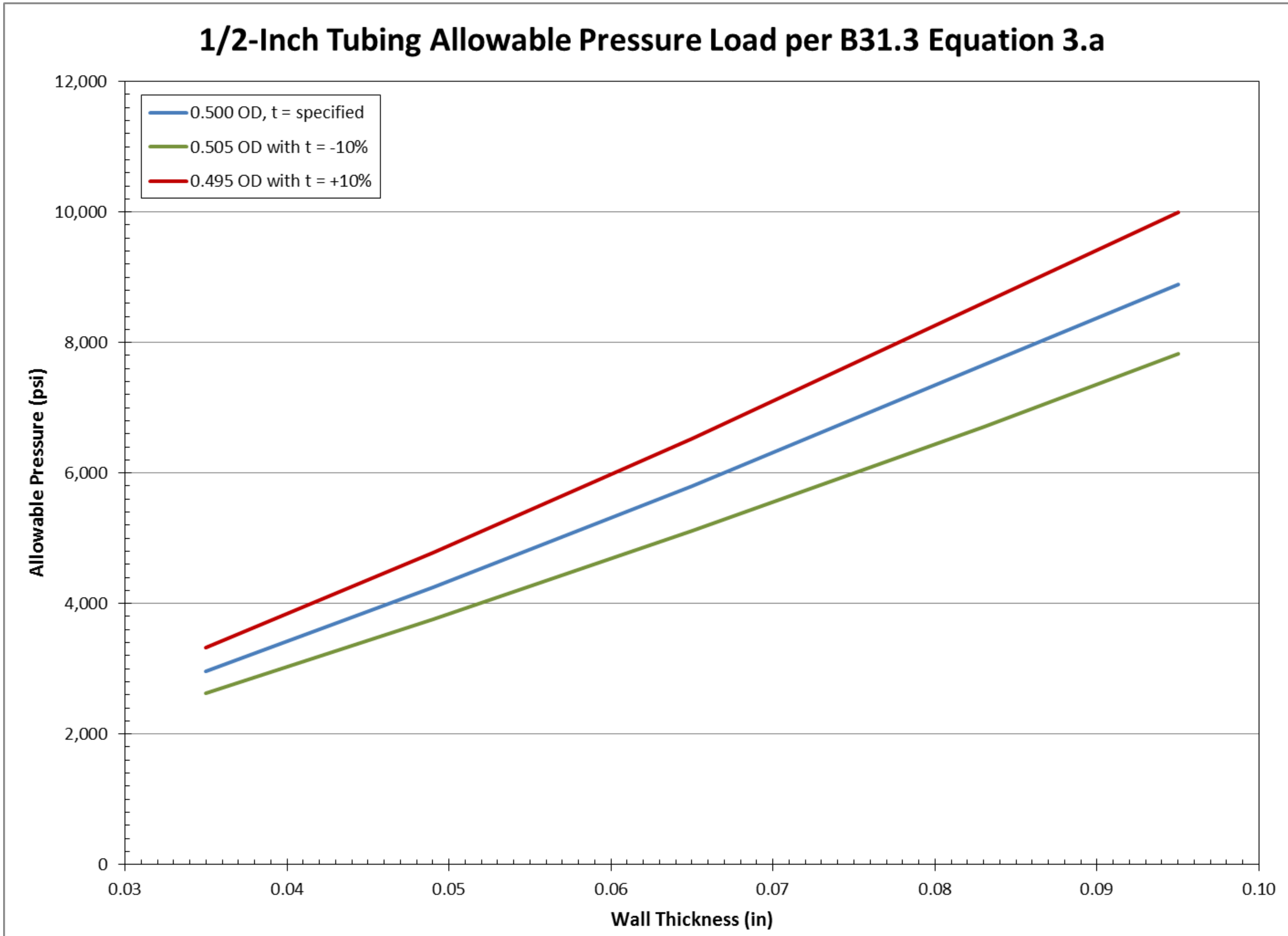

Figure 8: 1/2” Tube Allowable Pressure Loading Per B31.3 per Wall Thickness at a Resulting Operating Stress of 20,000 PSI

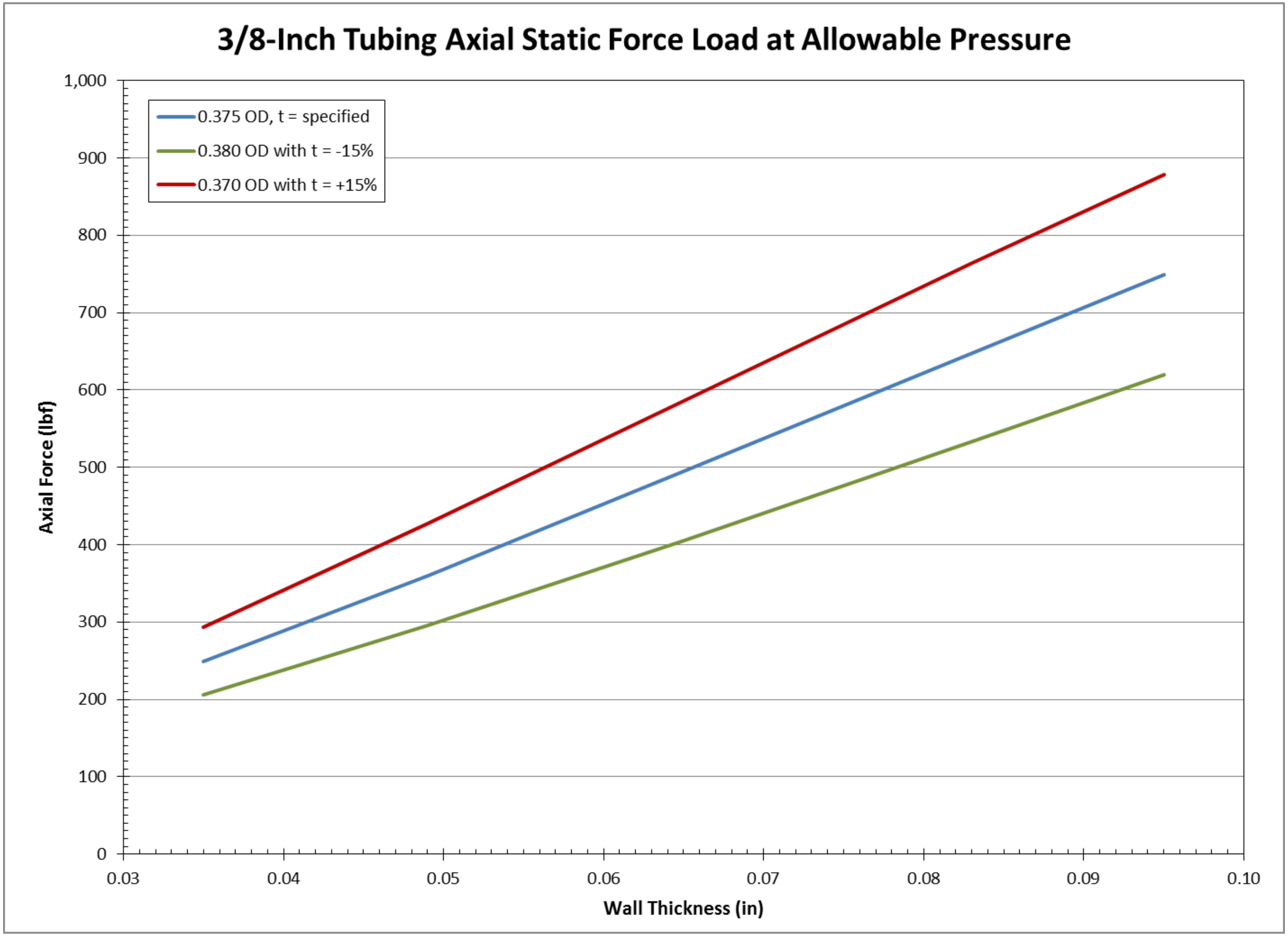

Figure 9: 1/2" Tube Axial Force per Wall Thickness for Allowable Pressure Loading Resulting in an Operating Stress of 20,000 PSI

	9/16-20UN	
$D =$	0.375	nominal thread diameter (in)
$D_{1max} =$	0.5162	maximum minor diameter of internal thread (in)
$D_{2max} =$	0.5341	maximum pitch diameter of internal thread (in)
$d_{1min} =$	0.5544	minimum major diameter of external thread (in)
$d_{2min} =$	0.5268	minimum pitch diameter of external thread (in)
$TPI =$	20	threads per inch
$p =$	0.05	thread pitch (in)

External Thread Shear Area

$$AS_{ext} = \frac{\pi \cdot LE \cdot D_{1max}}{p}\left[\frac{p}{2} + 0.57735(d_{2min} - D_{1max})\right]$$

Source: Bickford and Nassar

	Number of Threads Engaged						
Parameter	2	3	4	5	6	7	
$LE =$	0.100	0.150	0.200	0.250	0.300	0.350	Length of thread engagement (in)
$A_{shr.ext} =$	0.101	0.151	0.202	0.252	0.303	0.353	External thread shear area (in)
$\tau_{s\_0.035} =$	2,461	1,641	1,231	985	820	703	Resulting shear stress on fitting for 0.035 inch tube wall thickness (psi)
$\tau_{s\_0.049} =$	3,561	2,374	1,780	1,424	1,187	1,017	Resulting shear stress on fitting for 0.049 inch tube wall thickness (psi)
$\tau_{s\_0.065} =$	4,901	3,267	2,451	1,960	1,634	1,400	Resulting shear stress on fitting for 0.065 inch tube wall thickness (psi)
$\tau_{s\_0.083} =$	6,418	4,278	3,209	2,567	2,139	1,834	Resulting shear stress on fitting for 0.083 inch tube wall thickness (psi)
$\tau_{s\_0.095} =$	**7,425**	4,950	3,712	2,970	2,475	2,121	Resulting shear stress on fitting for 0.095 inch tube wall thickness (psi)

Worst case thread shear stress results on the external thread

Internal Thread Shear Area

$$AS_{int} = \frac{\pi \cdot LE \cdot d_{1min}}{p}\left[\frac{p}{2} + 0.57735(d_{1min} - D_{2max})\right]$$

Source: Bickford and Nassar

	Number of Threads Engaged						
Parameter	2	3	4	5	6	7	
$LE =$	0.100	0.150	0.200	0.250	0.300	0.350	Length of thread engagement (in)
$A_{shr.ext} =$	0.128	0.192	0.256	0.320	0.384	0.448	External thread shear area (in)
$\tau_{s\_0.035} =$	1,942	1,295	971	777	647	555	Resulting shear stress on fitting for 0.035 inch tube wall thickness (psi)
$\tau_{s\_0.049} =$	2,810	1,873	1,405	1,124	937	803	Resulting shear stress on fitting for 0.049 inch tube wall thickness (psi)
$\tau_{s\_0.065} =$	3,867	2,578	1,934	1,547	1,289	1,105	Resulting shear stress on fitting for 0.065 inch tube wall thickness (psi)
$\tau_{s\_0.083} =$	5,064	3,376	2,532	2,026	1,688	1,447	Resulting shear stress on fitting for 0.083 inch tube wall thickness (psi)
$\tau_{s\_0.095} =$	5,859	3,906	2,929	2,343	1,953	1,674	Resulting shear stress on fitting for 0.095 inch tube wall thickness (psi)

Figure 10: Thread Shear Stress Calculation for 3/8" Tube Operating at Allowable Wall Stress of 20,000 PSI

	3/4-20UNEF	
D =	0.5	nominal thread diameter (in)
D_{1max} =	0.7037	maximum minor diameter of internal thread (in)
D_{2max} =	0.7218	maximum pitch diameter of internal thread (in)
d_{1min} =	0.7419	minimum major diameter of external thread (in)
d_{2min} =	0.7142	minimum pitch diameter of external thread (in)
TPI =	20	threads per inch
p =	0.05	thread pitch (in)

External Thread Shear Area

$$AS_{ext} = \frac{\pi \cdot LE \cdot D_{1max}}{p}\left[\frac{p}{2} + 0.57735(d_{2min} - D_{1max})\right]$$

Source: Bickford and Nassar

	Number of Threads Engaged						
Parameter	2	3	4	5	6	7	
LE =	0.100	0.150	0.200	0.250	0.300	0.350	Length of thread engagement (in)
$A_{shr.ext}$ =	0.137	0.206	0.275	0.343	0.412	0.481	External thread shear area (in)
$\tau_{s\_0.035}$ =	1,330	887	665	532	443	380	Resulting shear stress on fitting for 0.035 inch tube wall thickness (psi)
$\tau_{s\_0.049}$ =	1,907	1,271	953	763	636	545	Resulting shear stress on fitting for 0.049 inch tube wall thickness (psi)
$\tau_{s\_0.065}$ =	2,602	1,735	1,301	1,041	867	743	Resulting shear stress on fitting for 0.065 inch tube wall thickness (psi)
$\tau_{s\_0.083}$ =	3,433	2,289	1,716	1,373	1,144	981	Resulting shear stress on fitting for 0.083 inch tube wall thickness (psi)
$\tau_{s\_0.095}$ =	**3,987**	2,658	1,994	1,595	1,329	1,139	Resulting shear stress on fitting for 0.095 inch tube wall thickness (psi)

Worst case thread shear stress results on the external thread

Internal Thread Shear Area

$$AS_{int} = \frac{\pi \cdot LE \cdot d_{1min}}{p}\left[\frac{p}{2} + 0.57735(d_{1min} - D_{2max})\right]$$

Source: Bickford and Nassar

	Number of Threads Engaged						
Parameter	2	3	4	5	6	7	
LE =	0.100	0.150	0.200	0.250	0.300	0.350	Length of thread engagement (in)
$A_{shr.ext}$ =	0.171	0.256	0.341	0.427	0.512	0.597	External thread shear area (in)
$\tau_{s\_0.035}$ =	1,070	714	535	428	357	306	Resulting shear stress on fitting for 0.035 inch tube wall thickness (psi)
$\tau_{s\_0.049}$ =	1,535	1,023	767	614	512	439	Resulting shear stress on fitting for 0.049 inch tube wall thickness (psi)
$\tau_{s\_0.065}$ =	2,094	1,396	1,047	838	698	598	Resulting shear stress on fitting for 0.065 inch tube wall thickness (psi)
$\tau_{s\_0.083}$ =	2,763	1,842	1,382	1,105	921	789	Resulting shear stress on fitting for 0.083 inch tube wall thickness (psi)
$\tau_{s\_0.095}$ =	3,209	2,140	1,605	1,284	1,070	917	Resulting shear stress on fitting for 0.095 inch tube wall thickness (psi)

Figure 11: Thread Shear Stress Calculation for 1/2" Tube Operating at Allowable Wall Stress of 20,000 PSI

CONCLUSIONS

- Results show that for the conditions considered and the assumptions made, the estimated thread shear stresses for the thread types considered are significantly less than the yield strength of either 304 or 316 stainless steel which is generally taken at room temperature to be 35 ksi and 37 ksi respectively.

- The analysis made considers the forces developed with the tubing wall operating at an allowable operating stress of 20 ksi per ASME B31.3

- Considering that many plant systems, such as instrument air, operate at pressures significantly less than those necessary to create the maximum allowable stress in the tubing wall, a corresponding decrease in thread stress and a higher factor of safety obtained in compression fitting type connection.

- The analysis effort undertaken is only a stress estimation based on the methodology employed. Where pressures approaching those necessary to produce the maximum allowable tubing wall stress of 20 ksi are present, it is prudent that due diligence be applied - especially where critical systems and personnel safety concerns are present.

DISCLAIMER

The thread stress estimation conducted is relevant only to the conditions considered and the assumptions made. It does not consider the many combinations of temperature, operating pressure, mechanical loading, support, restraint, environmental, maintenance, and/or assembly influences to which a compression fitting type connection considered is subject to. Thus, Winston Machinery Service makes no warranty declaration, either explicit or implied, related to the work presented and its use, nor does Winston Machinery Service accept responsibility for damages or injury resulting from the use of the work presented.