Effects of Tension and Pressure on Torque

The amount of torque required to either further tighten or loosen the connection under tension and pressure was determined from the threaded model and the same method of summing thread forces \( \times \) radii. Using 0.021 friction coefficient, the torques were:

Nominal tapers at three turns, torque = 5538 ft•lb
Addition of 500 kips tension = 5628 ft•lb
Addition of 11,220 psi pressure = 10,480 ft•lb

Therefore, tension has negligible effect and pressure has a significant effect. The lack of effect by tension is due to loss of preload force on the stab flanks being equal to the increase of force on the load flanks, thereby producing no change on total torque. This behavior was also recognized by Clinedinst.

Effect of Yielding on Torque

Part II of this paper discusses elastic and plastic solutions with the threaded finite element model that used to determine maximum strain for 5.4 turns, 9350 psi and 660 kips tension. The assembly interference was for a fast pin and slow box, with interferences of 0.0079 in. at the pin end and 0.0231 in. at the box end. Stresses were compared in Fig. 12 of Part II and showed significant reductions due to yielding of the P-110 material with the stress/strain relationship shown in Fig. 11 of Part II.

The same solutions were used to compare the effects of load and yielding on torque with results given in Table 7.

Torque reduction due to yielding when assembled was 11 percent and the reduction at full load was 16 percent. The reductions are not particularly high but do show a source of variation in the torque/turn relationship that can occur for makeup.

The elastic assembly solution can also be used to check above equation (3) for torque. The radius at the middle of the 3.25 in, length of threads was 3.418 in. The inside radius was 3.116 in. and the outside radius was 3.833 in., which gives the following calculated values:

\[
\delta = \frac{0.0079 + 0.0231}{2} = 0.0155 \text{ in. (interference at middle)}
\]

\[
P_c = \frac{30 \times 10^4(0.1255)(3.418^2 - 3.116^2)(3.833^2 - 3.418^2)}{2(3.418)^2(3.833^2 - 3.116^2)} = 6939 \text{ psi}
\]

\[
T = 0.0278(3.418)^2(3.25)(6939) = 7324 \text{ ft•lb}
\]

This value is within 3 percent of the finite element torque of 7545 ft•lb. and supports the above comparisons and conclusions that equation (3) is independent of the thread taper fit.

Conclusions

1 Calculation of contact pressure in the threads by the shrink-fit idealization gives reasonable results. The main limitations of this method is inability to account for tension effects.
2 The foregoing conclusion also applies to connection modeling without the threads, although greater accuracy than the shrink-fit method can be obtained for geometric end effects.
3 Tension reduces connection sealability by decreasing contact pressure on the stab flanks. Thread sealability is controlled by the stab flanks as opposed to the load flanks.
4 The change in root to crest clearances due to assembly, pressure, and tension is very small and may be negligible to leak tightness.

5 Based on the assembly test results and thread compound used, the coefficient of friction for makeup is 0.021. While this may seem low, similar tests and analysis methods on tool joints have given agreement with API for tool joint coefficient of friction of 0.08.
6 Equation (3) can be used to calculate assembly torque for 8 Round connections. However, variables such as surface finish, coatings, and thread lubricant can affect torque. Another possible variable is speed of makeup. Excessive rotation speed results in thermal energy by heating the connection, thereby absorbing input torsional energy and appearing as higher required torque. The rotation speed for the test connection was very small with negligible heat loss.
7 Tension has no effect on torque, while pressure increases the amount of torque necessary to either further tighten or loosen the connection.

SI Conversion Factors

<table>
<thead>
<tr>
<th>Unit</th>
<th>Conversion Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 in.</td>
<td>2.54 cm</td>
</tr>
<tr>
<td>1 ft•lb</td>
<td>1.356 N•m (or J)</td>
</tr>
<tr>
<td>1 lb</td>
<td>4.448 N</td>
</tr>
<tr>
<td>1 psi</td>
<td>6895 Pa</td>
</tr>
<tr>
<td>1 ksi</td>
<td>6.895 MPa</td>
</tr>
</tbody>
</table>

References


DISCUSSION

M. B. Allen

Asbill, Pattillo, and Rogers are to be commended for presenting an informative paper on the behavior of an API 8 Round threaded connector. They have demonstrated some of the benefits of finite element analysis of threaded connectors. However, there are a number of items that should be considered in evaluating their results and conclusions.

Plasticity

Plasticity is important in analyzing threaded connectors. Although plasticity was included in some of the computer runs, it significantly affects results that the authors have obtained from completely elastic runs.

The authors mention that failure to include plasticity in the assembly model in Part I resulted in high bearing stresses (247,463 psi) on the thread flanks and also accounts for the failure to match test data in Figs. 8 and 9. Although differences between predictions and test data are smaller under loaded conditions than for makeup, omission of plasticity in high stress regions near threads make the model too stiff, allowing the highly stressed regions to carry more load than they should, and causes the magnitudes of predicted stresses on the coupling outside diameter and pin inside diameter to be lower than the measured stresses in Figs. 10 through 13 of Part I.

By using an elastic model, calculated stresses go above yield during makeup and at low loads. Using ultimate stress in failure criteria would have resulted in unrealistically low...
failure loads, so the authors adopted failure criteria that allowed calculated stresses to exceed the ultimate stress of the material. Instead of basing thread shear and fracture failure criteria on ultimate stress or ultimate strain, an average was used. For fracture, only stresses at mid wall were considered. It was also assumed ‘that the limiting condition in the box was that twice the maximum shear stress could be above yield over a distance of 1/2 the engaged thread length measured from either the pin end or the box end.’ The authors imply that if half of the threads have stresses above yield and half have stresses below yield, then the median thread has a stress equal to yield and failure occurs. Any failure criterion based on thread stresses calculated with an elastic model will be incorrect since an elastic model does not allow realistic redistribution of thread loads and does not correctly calculate deformation and bearing stresses in the threads.

No failure criterion for jumpout (thread separation) was mentioned. Because a completely elastic model is too stiff, predicted displacements are low. An elastic model is unlikely to predict a jumpout failure since a significant amount of plastic deformation is involved. An elastic model is also less likely to give a true representation of thread separation and leak resistance than one with correct mechanical properties.

The authors use of a strain safety factor is misleading. Although the thread strain of 3.6 percent calculated at 660 kips and 10,000 psi is small compared to the 13.6 percent API minimum and the measured 21 percent elongation, no test is made to determine how fast it is changing with increasing load, which would indicate how close to failure the thread may be. Because of the complex relationship between load and stresses in the connector, to properly evaluate the factor of safety, it is necessary to calculate the load at failure and divide by the design load.

A final point on the importance of plasticity is illustrated in Part III in discussing the effects of yielding on torque. Assembly torque to 5.4 turns with a fast pin and slow box predicted by the elastic model is 13 percent higher than the torque predicted by the more accurate elastic-plastic model.

Plasticity has an important effect on stresses, strains, and displacements in threaded connectors. To omit this from the model decreases cost, but also decreases accuracy and should only be done if the documented discrepancy between predictions and test data, as well as the 13 percent difference in calculated torque, are acceptable. In my experience, the cost of a plastic analysis is less than twice the cost of an elastic analysis with gap nonlinearities and proper load incrementation. In my opinion, the best available answer is worth the additional cost and should be pursued, especially if it significantly changes the solution.

**Torque Correction Factor**

It seems reasonable to assume that an increase in thread length should not proportionally increase end effects. \( F_E \) is the multiplicative factor used in Part III to correct the hand calculated torque for end effects. As presented in the paper, this correction factor does not depend on the number of threads actually influenced by end effects. It implies that the magnitude of end effects is proportional to the overall torque and thread length or that \( F_E \) must be re-evaluated for different thread lengths. The following alternative for including end effects is therefore submitted:

\[
T' = T \left( \frac{NT - NE}{NT} + F_E \frac{NE}{NT} \right)
\]

where

- \( T' \) = corrected torque
- \( T \) = uncorrected torque
- \( NT \) = total number of engaged threads
- \( NE \) = number of threads influenced by end effects

This way, the formula may be applicable to other connectors with different thread lengths since \( NT \) can increase and not require reevaluation of \( F_E \).

**Unthreaded Model**

The authors have performed a valuable service in demonstrating the use of their unthreaded model. This model can significantly reduce the time and cost involved in finite element analysis of threaded connector. One suggestion to improve its accuracy would be to couple coincident nodes of the pin and box in such a way that forces are transmitted at an angle 30 deg from the axis of symmetry, the direction normal to the thread flanks, instead of coupling them axially. It should also be noted that the unthreaded model makes several simplifications and should be properly verified with tests or with a detailed model before accepting results at face value, as was done in the paper. Differences could be important, as in Fig. 7 of Part III.

**Authors’ Closure**

The discussion seems to imply that the analysis of threaded connections can only be done with plastic solutions; i.e., elastic solutions are incorrect. If so, this implication made in a broad sense is wrong. Up to maximum operating conditions, that is casing design loads using typical safety factors of 1.6 for tension and 1.2 for burst and normal assembly interference, elastic solutions give very good results. For analyses in which tension failure and/or burst loads are desired, plasticity becomes increasingly significant as larger volumes of material are stressed beyond the elastic limit. The first author has performed other threaded connection analyses using plasticity and found the above to be true. To automatically assume all threaded connection solutions must include plasticity is unwise engineering judgment. Another consideration of plasticity is that a stress-strain behavior must be used. This means that one particular material (and material behavior can significantly vary among manufacturers even for the same grade) must be used and, therefore, the analysis is somewhat limited. For these reasons, plastic finite element analyses of casing connections are usually performed only for the most critical situations, such as fatigue analysis or known severe service conditions (i.e., deep, high-pressure, sour gas wells).

The predicted stresses in Figs. 10 through 13 of Part I are not particularly lower than measured as stated in the Discussion. In Fig. 10, the theoretical results of the threaded model are in the most part about 14 percent higher (not lower) than measured; and similarly, in Fig. 11 the theoretical hoop stresses were mostly higher than measured. In Figs. 12 and 13, the theoretical stresses are both high and low. However, for those who are experienced in experimental stress analysis, it will be realized that the comparison of stresses in Part I is very good, particularly in view of the severity of some of the gradients.

Regarding the discussor’s comments on the use of midwall stresses and thread shear stress, our paper did not intend to evaluate integrity of the threads themselves—either for shearing or jumpout. The purpose of this investigation was not to attempt to redefine a criteria for thread jumpout. This has been very adequately analyzed by W.O. Clinedinst and adopted by API.

The reason for using midwall stresses in the evaluation in Part II is that the general behavior of the connection is governed by membrane (average) stresses. For example, the variation in pressure hoop stress in a cylinder from the inside surface to the outside surface is a secondary stress, not a primary stress, and failure cannot occur by a one-time ap-
application of a secondary stress. Only the membrane stress is primary and, therefore, of main concern (see ASME Boiler & Pressure Vessel Code, Section VIII, Division 2, Appendix 4).

In regard to the strain safety factor in the coupling of 3.7 on pp. 142 and 143 of Part II and strain rate, the increase in strain occurred at a decreasing rate with applied load and, therefore, the conclusion was conservative and not misleading. This behavior is as expected since the additional load carried by the coupling partial thread decreases with applied load and progressive yielding. If thread jumpout was of interest, then strain and strain rate in the pin at the last engaged thread (where jumpout originates) would be of major concern. However, as stated in the foregoing, the evaluation of failure by thread jumpout was not one of the purposes of this study.

The discussion regarding the torque correction factor, $F_3$, being independent of thread length is correct for API LT & C connections, and probably ST & C as well. However, there is some uncertainty as to the suggested equation for determining the end effects factor. From p. 151 of Part III, the value of $F_3$ is shown to be 1.10; therefore, it is assumed that the discussor's term in parenthesis should also equal 1.10 for the 7-in.-29-lb/ft connection evaluated. From Fig. 8 of Part III, the number of threads affected by the ends is 16 (8 at each end) and should be reasonably constant. Using NT = 26, NE = 16, and $F_3 = 1.10$, the value of the parenthesis is 1.06 and not 1.10. The only value of NE that gives 1.10 is 26, which is NT and cannot be correct. An alternative equation is given in the forthcoming, in which the 1.16 term accounts for the extra 494 ft-lb created by the end effects.

\[
T' = 7 NT - 16 + 1.16(16)(7)
= 7 NT - 16 + 18.56
\]

where

\[
T = \text{torque per thread, and including the two factors, } F_1 \text{ and } F_3, \text{ gives}
\]

\[
T = 2(1.15)(2 \mu R^2 LP_e),
\]

where

\[
L = 0.125 \text{ in., which gives}
\]

\[
T = 0.0479 \mu R^2 P_e, \text{ ft lb/thread}
\]

For NT = 26, $T' = 5540$ ft. lb, which agrees with the test torque.

The suggestion regarding the unthreaded model of coupling coincident nodes in order to transmit forces normal to the flanks, instead of axially, is a good idea; however, this would not behave correctly with the interface elements. Radial interference for makeup was input at the interface elements and, if the coincident nodes were coupled 30 deg to the centerline, unrealistic axial forces would occur from the axial component of the coupled forces.

However, the desired behavior could be modeled with two interface elements per thread (52 elements for 26 engaged threads) with each element being oriented normal to the thread flank. This would give improved modeling of the thread loading and allow the stab flank assembly preload to be relieved with tension. However, the objective of the investigation included simulating the hand equation assumption of smooth cylinders.

\[
T' = 7 NT - 16 + 1.16(16)(7)
= 7 NT - 16 + 18.56
\]