Study Models Pressure At Connections

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HOUSTON—Standard rotary-shouleder drill stem connections (as defined by American Petroleum Institute standards) have a seal surface to carry internal pressure. It is assumed that the seal will hold the pressure from routine drilling operations, but drill stem tests, stuck pipe and other special situations can apply much higher differential pressures to the drill string.

Moreover, there have been no analytical solutions to predict the pressure that can be safely contained by an industry-standard connection. This can lead to extra cost and logistical difficulties in instances where significant pressures are expected. Some proprietary connections are designed for high-pressure service, but they may be unnecessary if the API connection can safely tolerate the expected pressure loads. However, without an accurate predictive equation, relying on the pressure capacity of an API connection injects uncertainty and risk into the drilling process.

FEA Modeling Process

To accurately describe the behavior of rotary-shouledered connections under internal pressure, finite element analysis (FEA) was used to find the seal stress of typical connections under variations of tensile loading and applied internal pressure. The full modeling process included applying makeup torque to a two-dimen-

![FIGURE 1](image1.png)

**FIGURE 2A**

**NC56 Seal Stress Versus Internal Pressure**

![FIGURE 2B](image2.png)

**Residual Study Of Data in Figure 2A**

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The modeling process was used on six types of API connections (NC38, NC50, NC56, 5½-inch FH, 6½-inch FH, and 6½-inch Reg), each in four sizes (ranging in outside diameters from 4½ to 8½ inches and inside diameters from 2 to 4½ inches). For each connection type, two reasonable box outside diameter sizes and pin inside diameter sizes were chosen, with all four combinations modeled. Each of these combinations was studied under three tensile loading scenarios (no tension applied, 30 percent of the connection tensile capacity, and 50 percent of the connection tensile capacity).

For each model, the average seal stress was read from the results. This required the stress at each node to be read individually, and those numbers then were averaged, accounting for the different radii at which they were taken (because the outer nodes trace a larger circle than the inner nodes, the outer nodes receive more weight in the final average).

The average seal stress values were used largely because an analytical predictive equation was desired. Since analytical equations generally deal in average cross-sectional stresses, the exact radial profile of the stress was not the subject of this study.

Modeling Complexity

Figure 2A shows the average seal stress versus the internal pressure applied for one of the 72 total models (NC56, 8-inch OD x 2½-inch ID with no tension applied). Considering the complexity of the model (nonlinear contact, nonlinear material properties, and complex geometry), the results are surprisingly linear. However, Figure 2B is a residual study of the data in Figure 2A (FEA data minus the linear regression prediction). It shows that some amount of nonlinearity is present. Still, visual examination suggests that an analytical model should be able to admirably predict the seal stress in these connections.

The trend seen in Figure 2A—the compressive seal stress goes down as internal pressure is applied—is consistent for all connection types, sizes and tension loads modeled. The most obvious physical reason for this is a “pump-open” force: The diameter change between the counterbore and the ID of the connection provides an area for pressure to push on (Figure 3). This force can be calculated as the projected area multiplied by the pressure applied.

However, modeling shows that the pump-open effect does not explain fully the reduction in seal stress with increasing internal pressure. The other physical cause for the reduction is a “ballooning” effect. As the internal pressure causes the box to expand radially, the Poisson effect will cause the box to get shorter and effectively reduce the stress between the seal shoulders.

Calculating the Poisson effect is straightforward for a cylinder; the Lamé equation can be used to find the axial strain caused by internal pressure. In a complex geometry such as a rotary-shouldered connection box, however, integrating that equation over the constantly changing threaded region is not a practical way to proceed. Instead, a representative cylinder can be found that accurately accounts for the physical effect in a way that is mathematically simpler.

The outside diameter of the representative cylinder is logically the OD of the connection. There are many possible op-

\[ \sigma_{seal} = \sigma_0 - P_i \frac{A_{ProjArea}}{A_{seal}} - E \cdot \frac{2v}{r_i^2} \left( P_i - P_o \right) \]

\[ \sigma_{seal} = \sigma_0 - P_i \left[ \frac{Q_c^2 - ID^2}{BD^2 - Q_c^2} + \frac{R_t^2}{(OD/2)^2 - R_t^2} \right] \]

\[ A_{ProjArea} = \text{Projected pressure area (in}^2) \]
\[ A_{seal} = \text{Seal (in}^2) \]
\[ BD = \text{Bevel diameter (in)} \]
\[ E = \text{Young’s modulus (psi)} \]
\[ ID = \text{Inside diameter (in)} \]
\[ OD = \text{Outside diameter (in)} \]
\[ p = \text{Pitch of the threads (in)} \]
\[ P_i = \text{Internal pressure applied (psi)} \]
\[ P_o = \text{External pressure applied (psi)} \]
\[ Q_c = \text{Counterbore diameter (in)} \]
\[ r_i = \text{Inside radius (in)} \]
\[ r_o = \text{Outside radius (in)} \]
\[ R_t = \text{Average thread pitch radius (in)} \]
\[ v = \text{Poisson ratio} \]
\[ \sigma_0 = \text{Seal stress prior to pressure application (psi)} \]
\[ \sigma_{seal} = \text{Seal stress (psi)} \]
tions for the representative inside diameter, from the counterbore diameter (the smallest) up to the ID of the connection (the largest). The counterbore is too flexible, and the calculated pressure effect is significantly greater than that actually found in the FEA modeling. Conversely, the full ID is too stiff, and does not allow enough of a seal-stress change with applied internal pressure. A good representative cylinder will have an ID between these two extremes.

There are several possibilities for this “in-between” size: counterbore/ID average, OD/ID average, pin-tip diameter, and various other thread dimensions. Many of these possibilities were investigated, and $R_t$ was chosen as the best number. Several other possibilities also gave reasonable results, but the average pitch radius of the threaded region is a standard parameter in the industry-accepted A.P. Farr equation used for API rotary-shouldered connections (API RP7G). Therefore, $R_t$ allows accurate representative calculations with a minimum of new information required.

**Predictive Equation**

The full equation starts with the average seal stress created by makeup torque and tension applied. This calculation is found in API RP7G, and can be simplified here to be only the starting value ($\sigma_0$). As internal pressure is added, the pump-open force and the ballooning effect reduce the seal stress. These two effects can be treated as separate terms in a predictive equation (Equation 1). The pump-open force and Poisson effect terms then can be expanded and simplified using the Lamé equation to find the Poisson effect (Equation 2).

Equation 2 can be used to find the seal stress as a function of the internal pressure applied, and the adjusted seal stress can be used to calculate the amount of internal pressure that the connection can safely seal. The bracketed term in the equation represents the slope of the line in a graph of seal stress versus internal pressure. The slope of the actual FEA data can be calculated for each model using linear regression.

To check the agreement between the analytical equation and the FEA models,

**FIGURE 4A**
Comparison of Predicted and Modeled Seal Pressures

**FIGURE 4B**
Improved Model/Analytical Equation Agreement

**TABLE 1**

<table>
<thead>
<tr>
<th>Example pipe</th>
<th>Connection type</th>
<th>OD (in)</th>
<th>ID (in)</th>
<th>Makeup torque (ft-lb)</th>
<th>Bevel diameter (in)</th>
<th>Tension applied (lb)</th>
<th>Pressure capacity (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3½&quot;, 13.30#</td>
<td>NC38</td>
<td>4½₄₈</td>
<td>2₇₈</td>
<td>12,353</td>
<td>4₉₄</td>
<td>381,870</td>
<td>14,085</td>
</tr>
<tr>
<td>4½&quot;, 16.60#</td>
<td>NC46</td>
<td>5₉₄₈</td>
<td>3₉₈</td>
<td>21,235</td>
<td>5₉₄</td>
<td>468,297</td>
<td>14,222</td>
</tr>
<tr>
<td>5&quot;, 19.50#</td>
<td>NC50</td>
<td>6₉₈</td>
<td>3₉₈</td>
<td>28,225</td>
<td>6₉₄</td>
<td>560,764</td>
<td>16,326</td>
</tr>
<tr>
<td>5½&quot;, 21.90#</td>
<td>5½ FH</td>
<td>6₉₈</td>
<td>3₉₈</td>
<td>34,713</td>
<td>6₉₄</td>
<td>620,604</td>
<td>15,419</td>
</tr>
<tr>
<td>6&quot;, 27.70#</td>
<td>6 FH</td>
<td>8</td>
<td>4₉₈</td>
<td>52,714</td>
<td>7₉₄</td>
<td>760,352</td>
<td>16,615</td>
</tr>
</tbody>
</table>

ODs, IDs, and bevel diameters are the worst allowed for premium class, S-grade drill pipe in each example (taken from Standard DS-1®, Volume 3). The makeup torques are those recommended by API as calculated according to API RP7G. The tensions applied are the tensile capacities of premium class tubes in each example (taken from Standard DS-1®, Volume 2).
a ratio between the predicted slope and the FEA slope can be calculated for each model (predicted slope divided by FEA slope). If the ratio is 1.0, the analytical equation is exactly predicting the model results. If the ratio is greater than 1.0, the equation is overpredicting the effect of pressure on the reduction of the seal stress, which would be conservative. If the ratio is less than 1.0, the equation is underpredicting the effect of pressure (nonconservative).

Considering the complex, nonlinear models that are being studied, the agreement between the simple linear equation and the modeling results shown in Figure 4A is fairly good and generally conservative. Still, there are several connections with inaccuracies of 20-30 percent, especially on the conservative side. The agreement can be greatly improved by considering the residuals of the data fit, as shown in Figure 2B.

The worst (most nonlinear) data points are those at the end of the data, but in many cases, those points are nonphysical. The internal pressure applied there is greater than the seal pressure, so the connection is very likely leaking. If all nonphysical data points (where the internal pressure applied is greater than the average seal pressure) are removed, the recalculated FEA slope is much better predicted by the analytical equation. This improved agreement is shown for the full, final dataset in Figure 4B.

It is important to note that the assumption is that if internal pressure exceeds the average seal stress, the seal will begin to leak. This is very likely true, but may be nonconservative; leaking may begin even with average seal pressures greater than the internal pressure. In fact, the actual criteria for leaking may be related more to the peak seal stress rather than the average cross-sectional stress. In short, the failure criteria for leaking may be suggested connections.

Implications

The equation developed from this work can accurately predict the average seal stress of a rotary-shouldered connection under typical loading scenarios. The consistent quality of the predictions verified by FEA modeling gives assurance that any pressure capacity calculations can be used with confidence. Table 1 lists the pressure capacities calculated for popular pipe sizes, and the capacity numbers are larger than one might anticipate. This suggests that many API connections could be used in higher differential pressures than those in which they typically are applied, possibly allowing operators to save money by not renting purpose-designed connections.

Again, this assumes that leaking begins when the internal pressure applied is equal to the average seal stress. Ideally, more research will be conducted into the precise relationship between leaking and seal stress. It may be that a simple expression such as “the seal fails when the internal pressure is greater than 90 percent of the average seal stress” will be adequate for predicting failure. On the other hand, further study may find that the exact radial profile of the seal stress determines when a connection leaks. The condition of the seal surface certainly will play a role, as well.

The predictive equation has been developed using physically representative parameters, but it has been verified for accuracy with detailed FEA modeling. The pressure capacity equation allows operators to gain insights into the connections they use every day and extend their capacities with confidence.