Coiled tubing is a continuous steel pipe reeled on the reel, and is used in well completion, workovers, drilling and production. Currently about 75% of the coiled tubing jobs are: nitrogen kick-offs, acidizing and cleanouts. Other are: cementing, fishing and logging. Recently the use of coiled tubing becomes more important in drilling operations, especially for horizontal wells drilling and completion.

Because of limitations in the use of coiled tubing: life limits due to low cycle fatigue (on the real and in injector), allowed axial forces and pressures (combined stresses), change of dimensions (diameter and ovality) it is necessary to understand and examine interrelation of these limitations to determine overall coiled tubing working parameters and durability.

Coiled tubing stresses

The most important performance properties of coiled tubing are its rated values for axial tension, burst pressure and collapse pressure. The stresses are caused by internal and external pressures and the axial force that is either tension or compression (API Bulletin 5C3, 1985).

Axial tension loading results from the weight of the coiled tubing suspended below the point of interest. The stress when coiled tubing is in tension is defined as:

$$\sigma_t = \frac{F}{A_s} \quad \text{(tension)} \quad (1)$$

When the coiled tubing is in compression, and the hole is vertical, the helical buckling load is zero. This means that coiled tubing will buckle when we apply some compressive load. The total maximum axial stress is then:

$$\sigma_s = \frac{F}{A_s} \left[ \frac{R \cdot r_0}{2 \cdot \ell} \right] \quad \text{(compression)} \quad (2)$$

Body yield strength is the tensinal force (Eq. 1) required to cause the pipe body to exceed its elastic limit. Because the API allows the reduction of nominal wall thickness (minimum is 87.5% of nominal) it would be necessary to control wall thickness all the time.

Burst pressure rating is the calculated minimum internal pressure that will cause the coiled tubing to rupture in the absence of external pressure and axial load. For proper determination of this pressure it is recommended to use Barlow's equation for thick wall pipes. The API burst-pressure rating is based on this equation:

$$P_{(b)} = 0.875 \cdot \frac{2 \cdot \sigma_t \cdot t}{D_s} \quad (3)$$

 Collapse pressure rating is the minimum external pressure that will cause the coiled tubing walls to collapse in the absence of internal pressure and axial loading.

The radial and hoop stresses can be calculated using Lame's equations:

$$\sigma_r = \frac{p \cdot r_0 \cdot (r_0^2 - t^2) - p_s \cdot r_0^2 \cdot (r_0^2 - r^2)}{r^2 \cdot (r_0^2 - r_1^2)} \quad (4)$$

$$\sigma_\theta = \frac{p \cdot r_0^2 \cdot (r_0^2 + r^2) - p_s \cdot r_0^2 \cdot (r_0^2 + r_1^2)}{r^2 \cdot (r_0^2 - r_1^2)} \quad (5)$$

As a result of API collapse test data four collapse modes have been adopted (Pattillo, 1985):

**Elastic collapse:** if the axial or hoop stress which causes the tube to buckle is below the yield stress of the material, then the collapse mode characterizing the instability is termed elastic collapse.

**Plastic collapse:** as the moment of inertia of the tube increases (or equivalently, the length decreases) or as the diameter of cross section decreases (or thickness increases), the point will be reached where buckling does
not occur until the axial or hoop stress exceeds the material's yield strength.

**Transition collapse.** This collapse mode results from an anomaly in the statistical determination of minimum curves for elastic and plastic collapse and does not actually exist.

**Yield collapse.** This mode is based on the initial yield at the inner radius of the tube.

Necessary relationships for calculating API minimum collapse resistance with the appropriate \( (D_{nt}) \) relationship are given in Table 1.

Furthermore in the presence of a significant axial stress, API recommends the following equation:

\[
\sigma_i = \frac{p_i}{t} \]

The computation of allowed stresses is then done with the use of effective yield strength of material. It is essential to state that increasing of axial load lowers the yield strength of material and therefore the collapse resistance.

Because the combination of stresses is usually present in coiled tubing applications the Huber-von Mises-Hencky Yield Condition or Distortion Energy Theory of Failure (HMH) is used for determination of yield conditions:

\[
(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 = 2 \cdot \sigma_d^2 \]

The result of that analysis is a triaxial load capacity diagram, a graphical representation of the effect of anticipated loads (Johnson et al., 1985).

<table>
<thead>
<tr>
<th>MODE</th>
<th>EQUATION</th>
<th>APPLICABLE RANGE ( (D_{nt}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>ELASTIC</td>
<td>[ p_o = \frac{2 \cdot (\sigma_1 \cdot \frac{D_b}{t} - 1)}{2} \left( \frac{D_b}{t} \right)^{1/2} ]</td>
<td>( \left( \frac{D_b}{t} \right)^{1/2} \geq \frac{2 \cdot F_b}{F_c} )</td>
</tr>
<tr>
<td>TRANSITION</td>
<td>[ p_o = \left( \sigma_1 \right) \cdot \frac{F_b}{F_c} - F_c ]</td>
<td>( \left( \frac{D_b}{t} \right)^{1/2} \geq \frac{3 \cdot F_b}{F_c} \left( \frac{\sigma_1}{F_c} \right) \cdot \left( F_c - F_b \right) )</td>
</tr>
<tr>
<td>PLASTIC</td>
<td>[ p_o = \left( \sigma_1 \right) \cdot \frac{F_b}{F_c} - F_c ]</td>
<td>( \left( \frac{D_b}{t} \right)^{1/2} \geq \frac{3 \cdot F_b}{F_c} \left( \frac{\sigma_1}{F_c} \right) \cdot \left( F_c - F_b \right) )</td>
</tr>
<tr>
<td>YIELD</td>
<td>[ p_o = 2 \cdot (\sigma_1) \cdot \frac{D_b}{t} - 1 \left( \frac{D_b}{t} \right)^{1/2} ]</td>
<td>( \left( \frac{D_b}{t} \right)^{1/2} \geq \frac{2 \cdot F_b}{F_c} )</td>
</tr>
</tbody>
</table>

At the time the ellipse of plasticity was calculated in the manner of axial loads without buckling stresses in compression region.

Recent works (Newman, 1991) include the influence of all three principal stresses, and distinguish the compression and tension region of axial loads. Because the criterion predicts that yielding would first occur at the inner surface, only the inner conditions are considered.

For \( r = r_i \):

\[
\sigma = -p_i \]

and tangential stress due to internal pressure on inner surface is:

\[
\sigma_h = \frac{(r_i^2 + r_o^2) \cdot p_i - 2 \cdot r_i \cdot p_o}{r_o^2 - r_i^2} \]

To allow calculations we can write:

\[
\beta = \frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \]

Then:

\[
\sigma_h = \beta \cdot p_i - \beta^2 \cdot p_o \]

Substituting (8) and (11) into (7) we obtain:

\[
\alpha \cdot p_i^2 - \gamma \cdot p_i + \delta = 0 \]

Where:

\[
\alpha = \beta^2 + \beta + 1 \]
When we solve (12) for $p_i$, substituting $\sigma_i = -p_i$ we obtain:

$$o, = -\frac{2}{2} - \frac{\sqrt{\alpha - \beta}}{2}$$

(22)

The difference between the tensile yield hoop stress and the compressive yield hoop stress is:

$$\Delta \sigma_h = \sigma_{ht} - \sigma_{hc} = 2 \cdot \sqrt{\sigma_h^2 - 0.75 \cdot (\sigma_h - p_i)^2}$$

(23)

For purpose of the analysis, ovality was defined as:

$$Ovality \% = \left(\frac{r_{th}-1}{r_{th}}\right) \times 100$$

(25)

Because of tolerances in pipe diameter ($D_{max} = D \pm 0.254$ mm), thickness ($t_{max} = t + 0.2032$ mm; $t_{min} = t - 0.127$ mm) and real decreasing of pipe wall thickness because of diameter increasing or loss of material due to corrosion, it is important to check and control the effective wall thickness.

**Materials for coiled tubing production**

First coiled tubing (about 25 years ago) was produced from as-rolled steel, cold worked and tempered. The
yield strength of these materials ranged from 345 MPa to 552 MPa. As the needs and stresses during the use have progressed it was necessary to use heat treating operations to raise the properties. According to newest laboratory and field testings (Coburn, 1993) (cycling tests over the wheel, O-rings Sulphide Stress Cracking tests, metallographic examination) it was stated that:

- higher strength tubing (621 MPa or 689 MPa) performs better in coiled tubing pressure cycle tests
- the relation between the growth of the pipe diameter and the number of cycles is linear
- the number of cycles until failure is longer for higher strength materials
- the slope of the outer diameter growth line is less and the number of cycles until failure is longer for quenched and tempered martensite structure than for as-rolled and tempered steels
- the slope of the outer diameter growth line is less and the number of cycles until failure is longer for tubing operating at lower internal pressure
- increasing wall thickness reduces hoop stress for the system operating pressure
- in sulphide stress cracking tests, the quenched and tempered martensite microstructure is superior to the microstructure of perlite and ferrite, at the same strength level

Coiled tubing life considerations

Coiled tubing when reeled on the standard reel (depending to the traffic/road restrictions) is bent. He is also bent through uncoiling and running over the gooseneck. The tube can be bent to the yield radius of curvature, before the material begins to yield. Beyond that radius of curvature the material is plastically deformed. The bending radii that are encountered in service are beyond the radii of the elastic limit. This means that the coiled tubing is plastically deformed several times during one trip in and out of the hole (Fig. 2).

Yield radius of curvature is given by the following equation:

\[ \frac{E}{D} \frac{d}{r} \]

To fulfil this condition, the reel radius for coiled tubing OT-700 with diameter 50.8 mm, must be 11 meters.

Low cycle fatigue life of coiled tubing is associated with data scatter regardless of test accuracy, stability of material properties, or loading conditions. To predict safe coiled tubing operation with desired risk, a coiled tubing life-strain-reliability function was developed.

Data for the life-strain-reliability were obtained from full-scale tests performed in a company facility. Using equivalent strain function and low cycle fatigue line equation, test data were converted into one line loading level. This transformation function allows treatment of
data from various materials and loads as one homogeneous statistical group. Straight line regression line was fitted with a correlation coefficient of 0.975, and the distribution exhibits a variation coefficient of 11%.

Joint reliability function (N-S-Q Equation) \( Q(N,S) = \exp \left\{ \frac{1}{1.7675} \ln \left[ \frac{N}{23(6895000)^{0.363}} \right] \right\} \) (Avakov & Foster, 1994):

According to Avakov et al. (1993) conversion factor from coiled tubing strokes during a job to equivalent bend cycles is 2.3.

Conclusion

Triaxial load capacity diagrams (ellipse of plasticity) provide a useful tool in analysis of possible use of coiled tubing for a given application. The effect of service load conditions can be evaluated using both API and HMH calculations. The effect of pipe body dimensional tolerances or ovality can also be displayed.

We can also state that the life of coiled tubing increases with increased wall thickness. Also the life increases with increased gooseneck and reel radii, and increased strength of coiled tubing material. Stresses of coiled tubing are in plastic range, and its average actual life is in range from 30 to 200 strokes depending on loads. Fatigue model alone is not sufficient in determination of failure occurring. Monitoring the real diameters and possible mechanical damages is necessary. When ovality is present, the calculations are made with minimum yield stress of material, and minimum wall thickness. Because of the assumptions built into the calculation it is also good to account safety factor to allow a safety margin.

Nomenclature

- \( A_h \) — cross sectional area of coiled tubing, \( m^2 \)
- \( D_n \) — nominal pipe diameter, \( m \)
- \( E \) — Young’s Modulus, \( Pa \)
- \( F \) — force, \( N \)
- \( I \) — moment of inertia of coiled tubing section, \( m^4 \)
- \( N \) — number of cycles
- \( p_i \) — pressure inside the coiled tubing, \( Pa \)
- \( p_r \) — burst pressure, \( Pa \)
- \( p_o \) — pressure outside the coiled tubing, \( Pa \)
- \( Q \) — reliability
- \( r \) — radial axis from the center of the coiled tubing, \( m \)
- \( R \) — the radial clearence between the CT and the hole, \( m \)
- \( r_h \) — yield radius of curvature, \( m \)
- \( r_i \) — inside radius of the coiled tubing, \( m \)
- \( r_{IA} \) — inside radius of the tube at the major axis, \( m \)
- \( r_{IB} \) — inside radius of the tube at the minor axis, \( m \)
- \( r_o \) — outside radius of the coiled tubing, \( m \)
- \( r_{OA} \) — outside radius of the tube at the major axis, \( m \)
- \( r_{OB} \) — outside radius of the tube at the minor axis, \( m \)
- \( t \) — wall thickness, \( m \)
- \( \sigma \) — stress, \( Pa \)
- \( \sigma_a \) — axial stress, \( Pa \)
- \( \sigma_d \) — material yield strength, \( Pa \)
- \( (\sigma_d)_{ge} \) — effective yield strength, \( Pa \)
- \( \sigma_{he} \) — compressive yield stress in hoop direction, \( Pa \)
- \( \sigma_{ht} \) — tensile yield stress in hoop direction, \( Pa \)
- \( \Delta \sigma_{OH} \) — difference between hoop yield stresses, \( Pa \)
- \( \sigma_f \) — radial stress, \( Pa \)
- \( \sigma_t \) — tangential or hoop stress, \( Pa \)

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REFERENCES