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DEVELOPMENT OF COILED TUBING STRESS ANALYSIS

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The use of coiled tubing is increasing rapidly with drilling of horizontal wells. To satisfy all requirements (larger mechanical stresses, larger fluid capacities) the production of larger sizes and better material qualities was developed. Stresses due to axial forces and pressures that coiled tubing is subjected are close to its performance limits. So it is really important to know and understand the behaviour of coiled tubing to avoid its break, burst or collapse in the well.

Ključne riječi: Savitljivi tubing, Naprezanja, Trajnost

Primjena savitljivog tubinga drastično je povećana izradom horizontalnih bušotina. Za udovoljavanje novim zahtjevima (povećana mehanička opterećenja, veće protočne količine) usvojena je proizvodnja savitljivih tubinga većih promjera i iz kvalitetnijih materijala. Naprezanja uslijed uzdužnih sila i tlakova kojima je savitljivi tubing izložen dosežu često njegova granična dozvoljena naprezanja, pa je neobično važno odrediti ponašanje i trajnost savitljivog tubinga kako ne bi došlo do lomova, rasprskavanja ili gnječenja unutar kanala bušotine.

Introduction

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 this 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:

$$\sigma_a = \frac{F}{A_s}(\text{tension}) \tag{1}$$

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:

$$\sigma_a = F \cdot \left[\frac{1}{A_s} + \frac{R \cdot r_o}{2 \cdot I}\right](\text{compresion}) \tag{2}$$

Body yield strength is the tensional 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 nominall) 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 rapture in the absence of external pressure and axial load. For proper determination of this pressure it is recomended to use Barlow's equation for thick wall pipes. The API burst-pressure rating is based on this equation:

$$p_{i(r)} = 0{,}875 \cdot \frac{2 \cdot \sigma_d \cdot t}{D_n} \tag{3}$$

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:

$$\sigma_r = \frac{-p_i \cdot r_i^2 \cdot (r_o^2 - r^2) - p_o \cdot r_o^2 \cdot (r^2 - r_i^2)}{r^2 \cdot (r_o^2 - r_i^2)} \tag{4}$$

$$\sigma_t = \frac{p_i \cdot r_i^2 \cdot (r_o^2 + r^2) - p_o \cdot r_o^2 \cdot (r_i^2 + r^2)}{r^2 \cdot (r_o^2 - r_i^2)} \tag{5}$$

As a result of API collapse test data four collapse modes have been adopted (Pattilo, 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

Table 1. API minimum collapse equations with applicable (D_n/t) range

MODE	EQUATION	APPLICABLE RANGE (D_n/t)
ELASTIC	$p_{cr}=\dfrac{3{,}22488\cdot 10^{11}}{\left(\dfrac{D_n}{t}\right)\cdot\left(\dfrac{D_n}{t}-1\right)^2}$	$\left(\dfrac{D_n}{t}\right)\geq\dfrac{2+\dfrac{F_B}{F_A}}{\dfrac{3\cdot F_B}{F_A}}$
TRANSITION	$p_{cr}=(\sigma_d)_e\cdot\left(\dfrac{F_D}{\dfrac{D_n}{t}}-F_E\right)$	$\dfrac{2+\dfrac{F_B}{F_A}}{\dfrac{3\cdot F_B}{F_A}}\geq\left(\dfrac{D_n}{t}\right)\geq\dfrac{(\sigma_d)_e\cdot(F_A-F_D)}{F_C+(\sigma_d)_e\cdot(F_B-F_E)}$
PLASTIC	$p_{cr}=(\sigma_d)_e\cdot\left(\dfrac{F_A}{\dfrac{D_n}{t}}-F_B\right)-F_C$	$\dfrac{(\sigma_d)_e\cdot(F_A-F_D)}{F_C+(\sigma_d)_e\cdot(F_B-F_E)}\geq\left(\dfrac{D_n}{t}\right)\geq\dfrac{\sqrt{(F_A-2)^2+8\cdot\left[F_B+\dfrac{F_C}{(\sigma_d)_e}\right]}+(F_A-2)}{2\cdot\left[F_B+\dfrac{F_C}{(\sigma_d)_e}\right]}$
YIELD	$p_{cr}=2\cdot(\sigma_d)_e\cdot\left[\dfrac{\dfrac{D_n}{t}-1}{\left(\dfrac{D_n}{t}\right)^2}\right]$	$\dfrac{\sqrt{(F_A-2)^2+8\cdot\left[F_B+\dfrac{F_C}{(\sigma_d)_e}\right]}+(F_A-2)}{2\cdot\left[F_B+\dfrac{F_C}{(\sigma_d)_e}\right]}\geq\left(\dfrac{D_n}{t}\right)$

$$F_A=2{,}8762+1{,}069\cdot 10^{-6}\cdot\frac{\sigma_d}{6895}+2{,}1301\cdot 10^{-11}\cdot\left(\frac{\sigma_d}{6895}\right)^2-5{,}3132\cdot 10^{-17}\cdot\left(\frac{\sigma_d}{6895}\right)^3 \qquad F_B=0{,}026233+5{,}0609\cdot 10^{-7}\cdot\frac{\sigma_d}{6895}$$

$$F_C=\left(-465{,}93+0{,}030867\cdot\frac{\sigma_d}{6895}-1{,}0483\cdot 10^{-8}\cdot\left(\frac{\sigma_d}{6895}\right)^2+3{,}6989\cdot 10^{-14}\cdot\left(\frac{\sigma_d}{6895}\right)^3\right)\cdot 6895$$

$$F_D=46{,}95\cdot 10^6\cdot F_F^3\cdot\left[\frac{\sigma_d}{6895}\left(F_F-F_B/F_A\right)\cdot(1-F_F)^2\right]^{-1} \qquad F_E=F_D\cdot F_B/F_A \qquad F_F=3\cdot(F_B/F_A)\cdot(2+F_B/F_A)^{-1}$$

not occur untitl 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 elastical and plastical 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_n/t) relationship are given in Table 1.

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

$$\frac{(\sigma_d)_e}{\sigma_d}=\sqrt{1-\frac{3}{4}\cdot\left(\frac{\sigma_a}{\sigma_d}\right)^2}-\frac{1}{2}\cdot\left(\frac{\sigma_a}{\sigma_d}\right) \qquad (6)$$

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 Misses-Hencky Yield Condition or Distortion Energy Theory of Failure (HMH) is used for determination of yield conditions:

$$(\sigma_t-\sigma_a)^2+(\sigma_r-\sigma_t)^2+(\sigma_a-\sigma_r)^2=2\cdot\sigma_d^2 \qquad (7)$$

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

At the time the elipse of plasticity was calculated in the manner of axial loads withouth buckling stresses in compresion region.

Recent works (Newman, 1991) include the influence of all three principal stresses, and distinguish the compresion 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 $r=r_i$:

$$\sigma_r=-p_i \qquad (8)$$

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

$$\sigma_h=\frac{(r_i^2+r_o^2)\cdot p_i-2\cdot r_o^2\cdot p_o}{r_o^2-r_i^2} \qquad (9)$$

To allow calculations we can write:

$$\beta=\frac{r_o^2+r_i^2}{r_o^2-r_i^2} \qquad (10)$$

Then:

$$\sigma_h=\beta\cdot p_i-\beta\cdot p_o-p_o \qquad (11)$$

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

$$\alpha\cdot p_i^2-\gamma\cdot p_i+\delta=0 \qquad (12)$$

Where:

$$\alpha=\beta^2+\beta+1 \qquad (13)$$

$$\gamma = p_o \cdot (2 \cdot \beta^2 + 3 \cdot \beta + 1) + \sigma_a \cdot (\beta - 1) \qquad (14)$$

$$\delta = p_o^2 \cdot (\beta+1)^2 + p_o \cdot \sigma_a \cdot (\beta+1) + \sigma_a^2 - \sigma_y^2 \qquad (15)$$

When we solve (12) for p_i;

$$p_i = \frac{\gamma \pm \sqrt{\gamma^2 - 4 \cdot \alpha \cdot \delta}}{2 \cdot \alpha} \qquad (16)$$

To simplify the presentation, the difference between the internal and external pressure is defined as:

$$\Delta p = p_i - p_o \qquad (17)$$

Computer program done on the Faculty of mining, geology and petroleum engineering in Zagreb, can compute and show the results of API and HMH calculations, and their graphical presentation and comparison.

For given values (r_o=15.9 mm, t=2.35 mm, σ_d=450 MPa) calculated data are given in Table 2 and the real ellipse of plasticity is given in Fig. 1.

Ovality

During its life coiled tubing becomes oval due to bending around the reel and over the gooseneck. This ovality and axial tensile force reduce the collapse pressure of coiled tubing. Using the equilibrium equations for square waveform plastic hinge distribution we can obtain the external pressure at which the oval tube will collapse (Newman, 1992).

$$p_o = \frac{-B + \sqrt{B^2 - 4 \cdot A \cdot C}}{2 \cdot A} \qquad (18)$$

where A, B and C are given in equations (19), (20) and (21).

$$A = \frac{r_{oA}^2 + r_{oB}^2}{\Delta\sigma_h} \qquad (19)$$

$$B = r_{oA}^2 + r_{oB}^2 - \frac{2 \cdot p_i}{\Delta\sigma_h} \cdot (r_{oA} \cdot r_{iA} + r_{oB} \cdot r_{iB}) - \\ -2 \cdot r_{oB} \cdot r_{iB} + \frac{2 \cdot t \cdot \sigma_{hc}}{\Delta\sigma_h} \cdot (r_{oA} + r_{oB}) \qquad (20)$$

$$C = \frac{p_i^2}{\Delta\sigma_h} \cdot (r_{iA}^2 + r_{iB}^2) - \frac{2 \cdot t \cdot p_i \cdot \sigma_{hc}}{\Delta\sigma_h} + \\ + p_i \cdot (r_{iB}^2 - r_{iA}^2 - 2 \cdot t \cdot r_{iA}) + 2 \cdot t^2 \cdot \sigma_{hc} \left[\frac{\sigma_{hc}}{\Delta\sigma_h} + 1 \right] \qquad (21)$$

When we solve equation (7) for σ_h substituting $\sigma_r = -p_i$ we obtain:

$$\sigma_{hc} = \frac{\sigma_a - p_i}{2} - \sqrt{\sigma_d^2 - 0.75 \cdot (\sigma_a - p_i)^2} \text{ (compressive)} \qquad (22)$$

$$\sigma_{ht} = \frac{\sigma_a - p_i}{2} + \sqrt{\sigma_d^2 - 0.75 \cdot (\sigma_a - p_i)^2} \text{ (tensile)} \qquad (23)$$

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

$$\Delta\sigma_h = \sigma_{ht} - \sigma_{hc} = 2 \cdot \sqrt{\sigma_d^2 - 0.75 \cdot (\sigma_a - p_i)^2} \qquad (24)$$

For purpose of the analysis, ovality was defined as:

$$\text{Ovality}\,\% = \left[\frac{r_{oA}}{r_{oB}} - 1 \right] \cdot 100 \qquad (25)$$

Because of tolerances in pipe diameter ($D_{max,\ min}$=D±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

Table 2. Calculated data (Matanović at all, 1997.)

Force, F (N)	Pressure, p_1 (10^5 Pa)	Pressure, p_2 (10^5 Pa)	$p_{ovál.}$ (10^5 Pa)
-30000	219,3	-636,2	
-20000	427,0	-705,0	
-10000	565,0	-704,2	
0	655,9	-655,9	-614,7
10000	678,9	-627,0	-581,0
20000	696,0	-592,3	-542,3
30000	706,9	-551,3	-498,6
40000	711,2	-503,7	-449,4
50000	708,1	-448,7	-394,4
60000	696,4	-385,2	-332,5
70000	674,2	-311,1	269,7
80000	638,0	-223,0	-182,7
90000	580,3	-113,4	-88,6
100000	292,6	41,2	

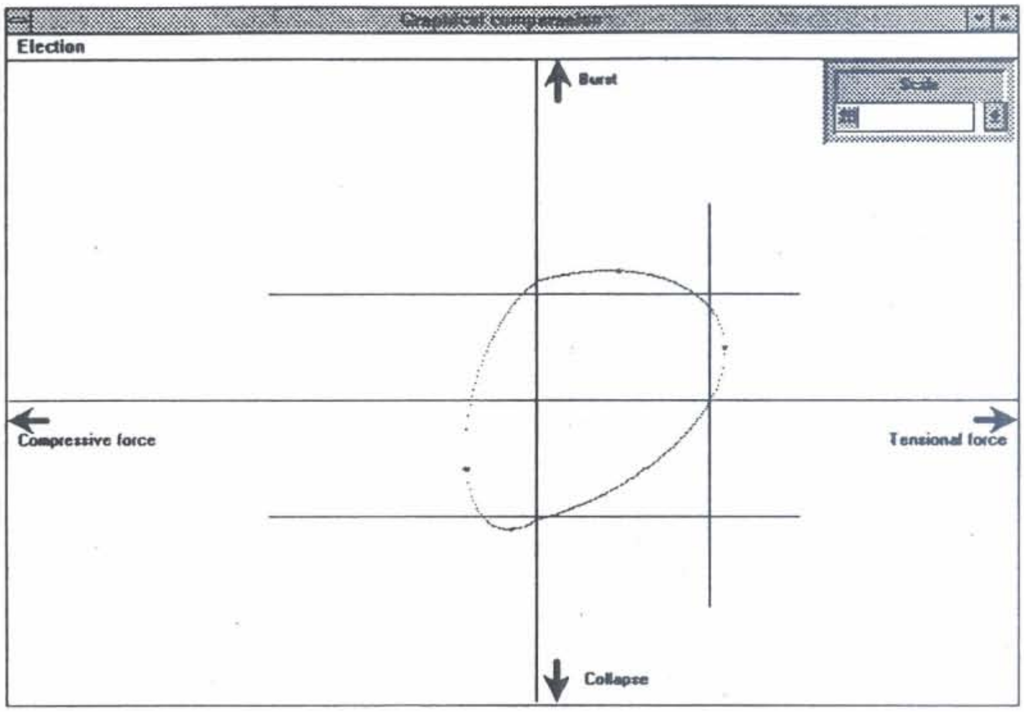

Fig. 1. Ellipse of plasticity with calculated data

yield strength of these materials ranged from 345 MPa to 552 PMa. 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, »C« ring 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 the quenched and tempered martensite structure than for as-rolled and tempered steels

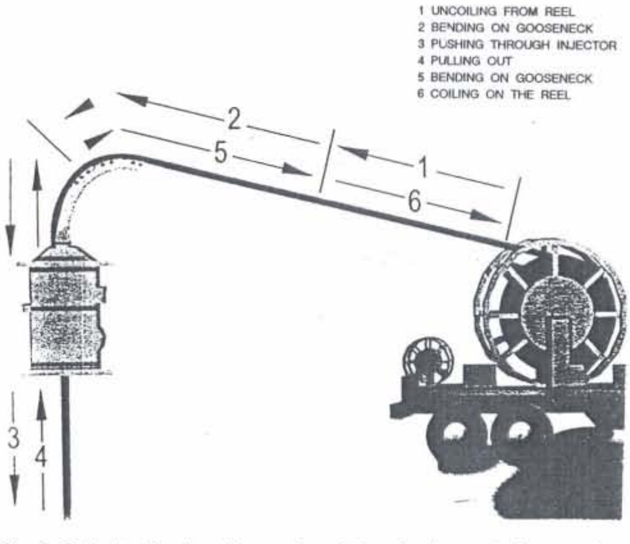

Fig. 2. Coiled tubing bending and straightening intervals from reel to the well

- 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 holef (Fig. 2).

Yield radius of curvature is given by the following equation:

$$r_b = \frac{E \cdot \frac{D}{2}}{\sigma_d} \tag{26}$$

To fulfil this condition, the reel radius for coiled tubing QT-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 homogenous 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) (Avakov & Foster, 1994) is:

$$Q(N,S) = \exp\left\{-\left(\frac{1}{1.7675}\ln\left[\frac{N}{23}\left(\frac{\sigma}{6895000}\right)^{2}\right]\right)^{16.919}\right\} \tag{27}$$

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_s – 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
$p_{i(r)}$ – 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
σ – stress, Pa
σ_a – axial stress, Pa
σ_d – material yield strength, Pa
$(\sigma_d)_e$ – effective yield strength, Pa
σ_{hc} – compressive yield stress in hoop direction, Pa
σ_{ht} – tensile yield stress in hoop direction, Pa
$\Delta\sigma_h$ – difference between hoop yield stresses, Pa
σ_r – radial stress, Pa
σ_t – tangential or hoop stress, Pa

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