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Coiled Tubing Life Prediction

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ABSTRACT

The paper addresses a recently developed mathematical model for coiled tubing fatigue life prediction. It is shown that the coiled tubing stress-strain condition is unique and is primarily characterized by plane elastic stress state induced by internal pressure and superimposed over extremely high plastic alternating bending strains.

The model was developed using full-scale tubing fatigue tests. In these tests three strength levels of coiled tubing material were tested at discrete pressure levels in ranges from 0 to 7500 psi using two types of gripper blocks: standard semicircle and universal V-shaped. It is revealed that the steady tangential stress component, induced by pressure, affects fatigue life in a nonlinear manner. Conventional failure theories do not work to describe and predict coiled tubing life. Instead, an algorithm based on equivalent strain as a function of principal strains is proposed. Constants of the function are defined in a way to achieve maximum correlation between model predicted life and actual life. Correlation coefficient became as high as 0.973.

Fatigue strength of coiled tubing material is expressed in terms of low-cycle S-N (strain versus life) fatigue line. This line is defined by reference point and slope. Fatigue life scatter is defined by lognormal distribution and its variation coefficient (standard deviation in terms of mean) is 0.11. That is, the test results are in close agreement with the model prediction.

Cumulative damage is expressed using Miner's rule and equivalent strain. Nonlinear equivalent strain respectively leads towards nonlinear cumulative damage expression.

Tables at end of paper

PROBLEM STATEMENT

Design in fatigue is a relatively simple problem when a detail is subjected to a uniaxial stress state with steady stress cycle components. For high-cycle fatigue of ductile metals under multiaxial stress state, the Tresca criteria (maximum shear stress failure theory) and von Mises criteria (distortion energy failure theory) are the most popular. But in low-cycle fatigue, these simple theories are unable to correlate the experimental results.

For instance, G.Z.Libertiny [1] notes that the von Mises criterion cannot allow for the effect of hydrostatic pressure. According to M.W.Brown and K.J.Miller [2], this situation has led to many criteria being suggested for correlating low-cycle, multiaxial fatigue, but no single criterion has been shown to have universal applicability. For example, M.Liddle and K.J.Miller [3] have tested tubes of 1% Cr-Mo-V steel in combined tension and torsion, for lives between 200 and 5000 cycles. Tests were controlled between constant strain limits. The authors were unable to correlate the results satisfactorily with a single criterion. As an alternative, they presented a series of constant life contours on a graph of maximum plastic shear strain range against the total tensile strain range, which is normal to the maximum shear plane.

D.L.McDiarmid [4] ran a comprehensive series of tests on thin-walled tubes of an aluminum alloy subjected to repeated internal pressurization and axial load. Using the results of other researchers' work on several materials **he derived an empirical correlation** relating the maximum alternating shear stress, τ_a, at the fatigue limit in terms of

NOMENCLATURE

δ_N - lognormal distribution shape parameter, it is a function of the life variation;
A - empirical constant (dimensionless);
Const - *S-N* line constant (defined by (6)
D - logarithmic ductility (19);
E - modulus of elasticity;
F - fatigue ductility coefficient (18);
m - empirical constant (dimensionless);
M - number of strokes to failure, test data;
M_r - predicted number of strokes to failure, it is defined by (10) as function of equivalent stresses, including constants m and A;
N - number of stress/strain cycles;
N_1 - number of cycles during one tubing stroke, as converted to the stress level S_M (7);
N_M - tubing median life in cycles defined at median strength, S_M;
R - correlation coefficient between $\ln M$ and $\ln M_r$ as random variables;
RA - percent reduction of area in tensile test;
S - equivalent cycle stress range (2); if plastic strain occurs, S should be found as the strain cycle range, ε, times modulus of elasticity, E;
S_a - cycle axial stress range (strain in terms of stress) imposed on outer fibers bent over the reel or bent over the gooseneck;
S_{conv} - normalizing stress unit constant, it is assumed that for English units $S_{conv} = 1$ kpsi; for other pressure units: $S_{conv} = 1$ kpsi $= 10^{3}$ Mpsi $= 1000$ psi $= 6.895 \cdot 10^{6}$ Pa $= 6895$ kPa $= 6.895$ MPa $= \dots$; S_{conv} coincides with respective conversion factor;
S_f - true fracture strength;
S_g - equivalent (combined) stress defined by (9) for tubing section subjected internal pressure and bent over the gooseneck;
S_M - tubing median fatigue strength (in terms of stress or strain) defined at median life, N_M;
S_r - equivalent (combined) stress defined by (8) for tubing section subjected internal pressure and bent over the reel;
S_t - tangential (hoop) stress on outer surface induced by internal pressure;
S_{ut} - ultimate tensile strength;
S_y - yield strength;

the alternating tensile stress, σ_τ, and mean tensile stress, σ_m, normal to the maximum shear stress plane

$$\tau_a = \sigma_A\left(0.58-0.225\sigma_\tau^{3/2}/\sigma_A^{3/2}\right)\left(1-2\sigma_{m\tau}/\sigma_{ut}\right)^{1/2} \tag{1}$$

Low-cycle fatigue of coiled tubing is a subject of serious investigations and publications. The results of full-scale and laboratory tests were published recently by S.M.Tipton, D.A.Newburn and K.R.Newman [5,6]. They analyzed a majority of known failure theories related to the low-cycle fatigue under multiaxial stress-strain conditions. They developed a life prediction model that is based on Miner's linear summation of fatigue damage [7]. In conclusion, the authors state that several life prediction parameters demonstrated success when combined with Miner's rule of damage summation for tests involving negligible internal pressure. However, hoop stresses resulting from increased pressure invoke damage mechanisms not accounted for by either the damage parameter or the linear damage summation. This indicates a need for the development of a valid nonlinear multiaxial damage summation algorithm [5,6].

To estimate low-cycle fatigue life under a multiaxial stress state, J.A.Collins [8] proposed techniques that involve the definition of an *equivalent stress* and *equivalent total strain range*. Equivalent total strain range is an appropriate function of the multiaxial plastic-elastic stress state. Then, life may be estimated using equivalent strain range and low-cycle fatigue lines (S-N lines) defined under uniaxial stress state. The equivalent uniaxial mean stress cycles may be converted to equivalent completely-reversed cycles by utilizing either the modified Goodman equations or some empirical expression based on specific material data. As J.A.Collins states, many questions still remain to be answered regarding the validity of this technique. However, it seems to be the best approach available at the present time.

STRESS/STRAIN CONDITION

Actual stresses, acting at various coiled tubing points and at various tubing locations between the reel and the well, are listed in Table 1. These stresses are typical and are defined for 1-¼ in. tubing OD, 0.087-in. wall thickness, and 5000 psi internal pressure. Other hardware parameters are listed in notes to Table 1. Analysis of the table data and field experience indicates that stresses induced by hoisting load and gripper pressure are relatively small, and they may be neglected in this consideration.

The coiled tubing stress-strain condition is unique and is primarily characterized by the following:

1. Alternating uniaxial plastic strains acting in the tubing's longitudinal direction. This direction is one principal direction. Strain cycles are induced by (*a*) tubing bending over the reel, and (*b*) tubing bending over the gooseneck. Bending strains are always far beyond the elastic limit, and they are the most damaging.

2. Steady (or static) biaxial stress state induced by internal pressure. Tangential and radial stresses due to pressure are the other two principal stresses, and they do not exceed the elastic limit. The static stress state may be converted into the equivalent uniaxial stress acting, for instance, in the longitudinal direction of the coiled tubing. Proportionality in this conversion between equivalent stress and actual biaxial stress state is in question and it is a subject of this presentation.

Stresses due to pressure are not damaging unless alternating axial stresses are applied. Damage due to alternating plastic strain cycles is intensified when pressure is increasing. For instance, during full-scale tests at negligible pressure, tubing life is close to 300 strokes where each stroke is counted as tubing motion from reel to well and back. At an elevated pressure of 5000 psi, the same tubing exhibits life from 50 to 55 cycles. Similar pressure influence on tubing life has been detected by S.M.Tipton, D.A.Newburn, K.R.Newman, L.W.Smith, E.J.Walker [5,6,9,10 et al.] and is experienced by coiled tubing in field applications.

Observing test results and published data, it has been tentatively defined that the coiled tubing full equivalent uniaxial alternating stress range could be expressed as

$$S = S_a + A S_t \left(\frac{S_t}{S_{conv}}\right)^{m-1} \quad if \quad S_a > S_y \tag{2}$$

$$S = 0 \quad if \quad S_a \le S_y$$

where stresses are in consistent units.

FATIGUE CURVE

Low-cycle *S-N* (Stress/Strain - Number of cycles to failure) curve is defined by so-called Coffin-Manson equation (see Appendix A). The curve is the sum of two lines, which are straight when plotted in double-logarithmic coordinates (Fig.3):

$$S = GN^b + FEN^c \,, \tag{3}$$

where the first term on the right side of equation reflects the elastic portion of the total strain range, and the second term reflects the plastic portion of the total strain range. It is evident that at short lives the plastic strain range dominates, that is, the second term of the total stress range is the largest portion of the Coffin-Manson expression. At longer lives, the elastic stress range (the first term of the equation) dominates.

Coiled tubing life is always less than 300 strokes. That is, plastic strains along with the respective second term of equation are decisive. For application purposes, the Coffin-Manson equation could be simplified and defined as

$$S = MEN^c \tag{4}$$

At $c=-\frac{1}{2}$, [11,12,13], and assuming that life has lognormal distribution [14] for reliability assessment:

$$\frac{N}{N_M}\left(\frac{S}{S_M}\right)^2 = \exp(\delta_N Z) \tag{5}$$

where the N_M, S_M and δ_N are empirical constants and they can be found by test data. The median fatigue line becomes (Z=0):

$$NS^2 = N_M S_M^2 = Const \tag{6}$$

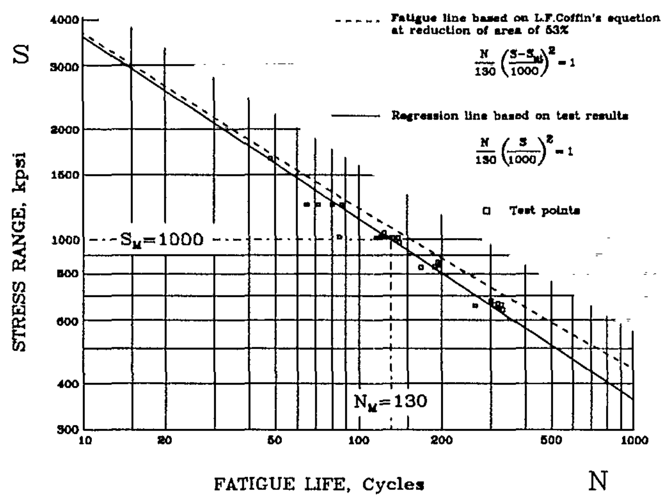

Figure 1

FATIGUE TESTS

A series of full-scale fatigue tests is planned to be conducted at company facilities. The program includes coiled tubing OD ranging from 1 to 1.75 in. at various wall thicknesses and materials. The first full set of tests was accomplished in spring 1992 using 1.25-in. OD by 0.087-in. wall coiled tubing. With standard semi-circular gripper blocks tests were run with three different materials:

1. QT-70, S_y=79.5 kpsi, and S_{ut}=85.7 kpsi,
2. SYMAX-80, S_y=91.9 kpsi, and S_{ut}=101 kpsi,
3. SYMAX-100, S_y=95.6 kpsi, and S_{ut}=113 kpsi.

With universal gripper blocks [15], tests were run using QT-70 coiled tubing.

A conventional coiled tubing unit was set up at a shallow test well 65-ft deep. The injector unit was equipped with 72 in. radius gooseneck. The coil diameter on the reel was in the range from 82 to 95 in. Tubing was run off the reel, over the gooseneck, and through the injector until the tubing's free end was below the top of the well casing. Each test *stroke* consisted of running the free end of the tubing 65 ft to the bottom of the well and back to the top of the casing. Tubing was run back and forth at constant internal pressure. Number of strokes to failure, *M*, was recorded. During a stroke, the most loaded and damaged sections had one cycle of bending stresses, S_{ar}, over the reel, and two cycles of bending stresses, S_{ag}, over the gooseneck.

For these sections, it may be counted as three strain cycles only when imposed stresses are identical, that is, radii of the reel and the gooseneck are identical. Actually, such condition is unlikely to happen. In general, summation should be performed using Miner's cumulative damage rule [7]:

$$N_1 = 1\left(\frac{S_r}{S_M}\right)^2 + 2\left(\frac{S_g}{S_M}\right)^2 \tag{7}$$

where

$$S_r = S_{ar} + A S_t\left(\frac{S_t}{S_{conv}}\right)^{m-1} \tag{8}$$

$$S_g = S_{ag} + A S_t\left(\frac{S_t}{S_{conv}}\right)^{m-1} \tag{9}$$

To begin the next test, a new tubing "sample" was reeled off until a section that had not been cycled came off the reel and reached the top of the casing. This resulted in a tubing sample length of approximately 125 ft long for each test and, therefore, this tubing length was discarded after each test. The tubing was filled with water and pressurized to the desired level. Failure was defined as when water squirted out of a crack or pin hole.

TEST RESULTS

It is assumed that low-cycle fatigue strength is proportional to the material ultimate strength. Under this assumption all test results have been converted to one standard strength level of 80 kpsi. For each sample, fatigue life in stress/strain cycles, N_1, was defined by equation (7). Then, expected life in strokes-to-failure becomes

$$M_r = \frac{N_M}{N_1} = \frac{N_M S_M^2}{S_r^2 + 2S_g^2} = \frac{Const}{S_r^2 + 2S_g^2} \tag{10}$$

In this evaluation, the constants, *m* and *A*, are links between actual life, *M*, and predicted life, M_r. We need to define such values of *m* and *A* at which correlation coefficient, *R*, between *M* and M_r becomes maximum and close to the unity:

$$(m,A) = (m,A \mid R^2 = \max)$$

In other words, if there exists a function

$$R^2 = f(m,A) \tag{12}$$

then the *m* and *A* will be found by putting

$$\frac{\partial f(m,A)}{\partial m} = 0\ , \qquad \frac{\partial f(m,A)}{\partial A} = 0$$

$$and \tag{13}$$

$$\frac{\partial^2 f(m,A)}{\partial m^2} < 0\ , \qquad \frac{\partial^2 f(m,A)}{\partial A^2} < 0$$

On Fig.2 the correlation coefficient is traced against *m* at *A=1*. By this graph, it has been chosen that *m=1.985* and *A=1* as solution for the test data.

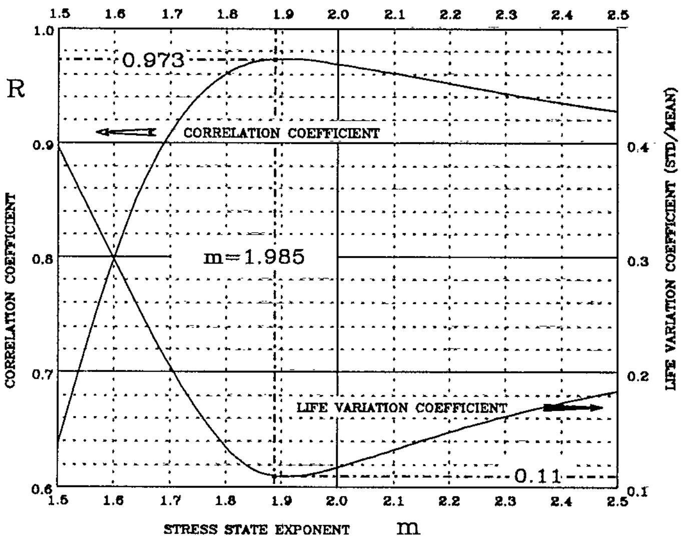

Figure 2

CONCLUSIONS

1. Low-cycle fatigue life for coiled tubing of 1.25 in outside diameter can be predicted using empirical equivalent stress expressed by equation (2) in terms of alternating stress range and hoop stress.
2. Fatigue *S-N* lines are defined by hyperbolic equations (5) and (6). They become straight lines when plotted in a double-logarithmic coordinate system.
3. The practical advantage of the equations (5) and (6) is that they simplify Miner's fatigue damage summation procedure.
4. A correlation coefficient of 0.973 was achieved between actual life and predicted life.
5. The paper presents a new approach in low-cycle life modeling. Therefore, the approach needs further verification by fatigue data from the field and from the lab.

REFERENCES

1. Libertiny, G.Z.: SHORT LIFE FATIGUE UNDER COMBINED STRESSES. *J. Strain Analysis*, Vol.2, 1967; 91

2. Brown, M.W. and Miller, K.J.: A THEORY FOR FATIGUE FAILURE UNDER MULTI-AXIAL STRESS-STRAIN CONDITIONS. *Proc. Inst. Mech. Engrs.*, London, 1973, Vol.187, 65/73; 745-755

3. Liddle, M. and Miller, K.J.: MULTIAXIAL HIGH STRAIN FATIGUE. *Proc. 3rd Int. Conf. Fracture*, Munich, 1973;

4. McDiarmid, D.L.: FAILURE CRITERIA AND CUMULATIVE DAMAGE IN FATIGUE UNDER MULTIAXIAL STRESS CONDITIONS. City University, London, UK; Ph.D. Thesis, 1972;

5. Tipton, S. M. and Newburn, D.A.: PLASTICITY AND FATIGUE DAMAGE MODELING OF SEVERELY LOADED TUBING. *Advances in Fatigue Lifetime Predictive Techniques*, Proc. 1st Symposium, San Francisco, CA, April 1990; Philadelphia, ASTM, 1992; 369-382

6. Newman, K.R. and Newburn, D.A.: COILED TUBING LIFE MODELING. SPE 22820, *SPE Annual Techical Conference & Exhibition*, Proceedings; Dallas, TX, October 6-9, 1991;

7. Miner, Milton A.: CUMULATIVE DAMAGE IN FATIGUE. *J.Appl.Mechanics*, Vol.12, *Trans. ASME*, Vol.67, 1945; A159-A164

8. Collins, J.A.: FAILURE OF MATERIALS IN MECHANICAL DESIGN. ANALYSIS, PREDICTION, PREVENTION. New York, J.Wiley, 1981; 629

9. Smith, L. W.: METHODS OF DETERMINING THE OPERATIONAL LIFE OF INDIVIDUAL STRINGS OF COILED TUBING. *SPE Workover and Well Intervention Seminar*, November 16, 1989, Aberdeen, Scotland

10. Walker, E. J.: HOW LOADS AFFECT COILED TUBING LIFE. *World Oil*, Vol.213, No.1; January, 1992; 47-49

11. Coffin, L.F.: DESIGN ASPECTS OF HIGH TEMPERATURE FATIGUE WITH PARTICULAR REFERENCE TO THERMAL STRESSES. *Trans. ASME*, Vol.78; April, 1955; 527-532

12. Langer, B. F.: DESIGN OF PRESSURE VESSELS FOR LOW-CYCLE FATIGUE, ASME Trans., *J. of Basic Engineering*, Vol.84, Series D, No.3, September, 1962; 389-402

13. Tavernelli, J. F. and Coffin, L. F. Jr.: EXPERIMENTAL SUPPORT FOR GENERALIZED EQUATION PREDICTING LOW CYCLE FATIGUE. Trans. ASME, *J. Basic Engineering*, Vol.84, Series D, No.4; December 1962; 533-541

14. Avakov, V. A.: FATIGUE STRENGTH DISTRIBUTIONS. *International Journal of Fatigue*, (UK) Vol.15, No.2, 1993; 85-91

15. Avakov, V.A. and Council, M.N.: UNIVERSAL GRIPPER BLOCKS FOR REELED TUBING INJECTORS. Paper 92-Pet-7, Presented at the *ASME Energy Sources Technology Conference and Exhibition*, January 25-30, 1992; Houston, TX; 4

Appendix A

COFFIN-MANSON EQUATION

The low-cycle *S-N* line has several modifications in literature. Here we will follow the original presentation made by L.F.Coffin and S.S.Manson [13]. Available approximations of the equation parameters are published by J.A.Collins [8]. The equation is

$$S = GN^{b}+FEN^{c} , \tag{14}$$

where the first term on the right side of equation reflects elastic portion of the total strain range, and the second term reflects plastic portion of the total strain range.

J.A.Collins states that b ranges from about -0.05 to -0.15 and c ranges from about -0.5 to -0.8. S.S.Manson defines them through material properties:

$$b = -0.08-0.18\log(S_f/S_{ut}) , \tag{15}$$

$$c = -0.52-0.25\log D-\log T , \tag{16}$$

$$T = \left[1-82\left(\frac{S_{ut}}{E}\right)\left(\frac{S_f}{S_{ut}}\right)^{0.18}\right]^{-1/3} , \tag{17}$$

$$F = 0.83\,D\,T , \tag{18}$$

$$D = -\frac{1}{2}\ln\frac{100-RA}{100} . \tag{19}$$

The material constant, G, is defined by S.S.Manson as

$$G = \frac{9}{4}S_{ut}\left(\frac{S_f}{S_{ut}}\right)^{0.9} . \tag{20}$$

It may be noted that at short lives the plastic strain range dominates, that is, the second term of the total stress range is the largest portion of the Coffin-Manson expression. At longer lives, the elastic stress range (the first term of the equation) dominates. The point at which the elastic and plastic lines intersect is called the *transition point* and the respective abscissa is called the *transition life* (Fig.3).

For design purposes, B.F.Langer [12] proposed:

$$N(S-S_{ut})^2 = ED . \tag{21}$$

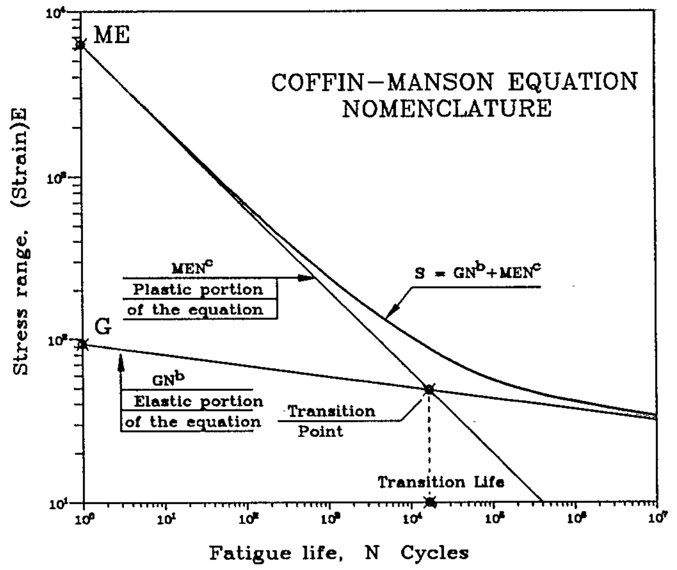

Figure 3

Table 1

COIL TUBING
TYPICAL STRESS/STRAIN CYCLE PATTERN AT VARIOUS LOCATIONS

Section location: *source of loading*		INNER TUBING SURFACE & OUTER RADIUS	INNER TUBING SURFACE & INNER RADIUS	OUTER TUBING SURFACE & OUTER RADIUS	OUTER TUBING SURFACE & INNER RADIUS
1. On the reel: *inner pressure &* *bending over reel*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_2=$ 33607 $\sigma_1=$ 382012	$\sigma_2=$ -5000 $\sigma_1=$ 33607 $\sigma_3=$ -382012	$\sigma_3=$ 0 $\sigma_2=$ 28607 $\sigma_1=$ 443787	$\sigma_2=$ 0 $\sigma_1=$ 28607 $\sigma_3=$ -443787
2. Space reel- -gooseneck & 4. gooseneck-p.beams: *inner pressure*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_1=$ 33607 $\sigma_2=$ 0	$\sigma_3=$ -5000 $\sigma_1=$ 33607 $\sigma_2=$ 0	$\sigma_3=$ 0 $\sigma_1=$ 28607 $\sigma_2=$ 0	$\sigma_3=$ 0 $\sigma_1=$ 28607 $\sigma_2=$ 0
3. Gooseneck: *inner pressure &* *bending over* *gooseneck*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_2=$ 33607 $\sigma_1=$ 224167	$\sigma_2=$ -5000 $\sigma_1=$ 33607 $\sigma_3=$ -224167	$\sigma_3=$ 0 $\sigma_2=$ 28607 $\sigma_1=$ 260416	$\sigma_2=$ 0 $\sigma_1=$ 28607 $\sigma_3=$ -260416
5. Between p.beams: *inner pressure &* *V-block load* *(upper sections)*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_1=$ 1822 $\sigma_2=$ 0	$\sigma_3=$ -5000 $\sigma_1=$ 1822 $\sigma_2=$ 0	$\sigma_2=$ 0 $\sigma_1=$ 60392 $\sigma_3=$ 0	$\sigma_2=$ 0 $\sigma_1=$ 60392 $\sigma_3=$ 0
6. Between p.beams: *inner pressure,* *hoisting load &* *V-block load*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_2=$ 1822 $\sigma_1=$ 56000	$\sigma_3=$ -5000 $\sigma_2=$ 1822 $\sigma_1=$ 56000	$\sigma_3=$ 0 $\sigma_1=$ 60392 $\sigma_2=$ 56000	$\sigma_3=$ 0 $\sigma_1=$ 60392 $\sigma_2=$ 56000
7. Below pressure beams: *inner* *pressure and* *hoisting load*	σ_r σ_t σ_a	$\sigma_3=$ -5000 $\sigma_2=$ 33607 $\sigma_1=$ 56000	$\sigma_3=$ -5000 $\sigma_2=$ 33607 $\sigma_1=$ 56000	$\sigma_3=$ 0 $\sigma_2=$ 28607 $\sigma_1=$ 56000	$\sigma_3=$ 0 $\sigma_2=$ 28607 $\sigma_1=$ 56000

σ_a - axial stress due to hoisting load or bending over reel and gooseneck; bending stress is defined as $\sigma_a=\sigma_b=\pm\varepsilon E\approx\pm dE/(2R)$, where R is bending radius over reel or gooseneck; σ_r - radial stress; σ_t - tangental stress; $\sigma_1,\sigma_2,\sigma_3$ - principal stresses.

Stresses are defined at following conditions: (1) Tubing size 1.25-in. OD x 0.087-in. Wall Thickness; (2) Yield strength 70000 psi (22251 lb axial yield load); (3) Applied allowable tensile stress 70000/1.25=56000 psi or 17800 lb hoisting load; (5) Inner pressure 5000 psi; (6) Bending radius over gooseneck 72 in.; (7) Bending radius over reel 42.25 in.; and (8) Pressure beam unit load 1287 lb/in. of beam length inducing ±31785 psi tangental bending stress, compression is neglected.

Table 2

COILED TUBING TEST DATA
1.25 *in.* OD, 0.087 *in.* wall thickness
m=1.895

SAMPLE	Ultimate Strength S_u *kpsi*	Bending Stress $S_{ag}=E\varepsilon_{ag}$ *kpsi*	Pressure P *psi*	Hoop Stress S_t *kpsi*	Predicted Life M_r *Strokes*	Recorded Life M *Strokes*	S-N Line Constant $N_M S_M^2$
1	85.7	436	200	1.1	321	240*	105645726
2	85.7	436	3000	17.2	145	143	139366392
3	85.7	436	5000	28.6	61	37	85365141
4	85.7	436	3000	17.2	145	135	131569671
5	85.7	436	4000	22.9	95	86	128692386
6	85.7	436	5000	28.6	61	56	129201294
7	85.7	446	4000	22.9	95	86	128692386
8	85.7	446	3000	17.2	145	141	137417211
9	85.7	446	5000	28.6	61	51	117665464
10	113	431	5000	28.6	107	115	152609453
11	113	431	6000	34.3	70	71	143230382
12	113	431	7500	42.9	39	40	144978886
13	113	431	3000	17.2	252	216	121082262
14	113	431	5000	28.6	107	102	135357949
15	113	431	7500	42.9	39	42	152227830
16	113	431	6000	34.3	70	66	133143735
17	101	395	6000	34.3	56	43	108582390
18	101	395	6000	34.3	56	49	123733421
19	101	395	5000	28.6	85	90	149499673
20	101	395	5000	28.6	85	82	136210813
21	101	395	4000	22.9	131	126	135751252
22	101	395	4000	22.9	131	112	120667780
23	85.7	457	3000	17.2	145	140	136442621
24	85.7	446	5000	28.6	61	60	138429958
25	85.7	446	3000	17.2	145	132	128645900
26	85.7	457	5000	28.6	61	50	115358298
27	85.7	457	5000	28.6	61	54	124586962
28	85.7	446	3000	17.2	145	140	136442621
						Mean	**130021352**
						STD	**14449594**
						STD/Mean	**0.111**

* Test halted after 240 strokes