2nd European CT Roundtable

Coiled Tubing Forces and Stresses Modeling Improvements

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Tubing Forces - BHA Bending Friction

As CT sizes and services increase, the sizes are also increasing. When a long, stiff BHA is bent around a curve in the well, additional friction is created with the wellbore. This additional friction can significantly change the forces required in the CT to convey the BHA. Initial tubing forces models (TFMs) did not consider this additional friction.

Figure 1 shows three cases for BHA bending. The first case shows a rigid, unbent tool in the curvature of the well. This case shows the longest rigid length the tool can have and still fit in the curve. There is no additional friction if the tool is no longer than this rigid length. The maximum rigid length of the tool is:

$$L_r = 2\sqrt{(4R + 2d)\Delta}$$

where:

d = tool diameter
R = radius of curvature of the wellbore
Δ = the radial clearance between the wellbore and the tool

Since 4R is obviously much larger than 2d, this equation can be simplified to:

$$L_r = 4\sqrt{R\Delta}$$

The second case shows the tool somewhat bent, but only contacting the wellbore at one point on the top. This case applies for tool lengths between L_r and L_2, where:

$$L_2 = 4\sqrt{3R\Delta}$$

In the third case the tool is bent further, and wraps around the wellbore for a distance. In this case the tool is in contact with the upper surface of the wellbore for an extended distance. This case applies for tool lengths greater than L_2.

The additional axial force required to overcome the friction caused by the BHA bending when the tool length is greater than L_r is given by:

$$F_{BHA} = 4P\mu$$

where μ is the friction coefficient and P is given by:

$$P = \frac{2EI}{RL}$$

where:

I is the moment of inertia of the tool
E is the modulus of elasticity
L is the tool length or L_2, whichever is less

When the additional friction caused by the BHA is included in a TFM, the result can be very significant. Figure 2 shows an example model output without including the BHA friction, while Figure 3 shows the same output with BHA friction included. Sometimes the additional friction from a BHA passing around a curve or dog leg can be significant enough to cause lock up to occur, and prevent further penetration into the well.

Stress Modeling Update

This author wrote an SPE paper 23131, "Coiled-Tubing Pressure and Tension Limits", several years ago which develops the equations to calculate the CT limits. (A copy of this paper is attached for convenience to the reader). This is an update to that paper with a correction to one of the equations, and the development of an improved method of calculating the limits. To clearly understand this update the reader must first be familiar with the equations and development presented in that paper. This update uses the same nomenclature as is defined in that paper.

Correction:

When the axial force is less than 0, if it is assumed that the CT is buckled, and the axial force, F_a, in equation 4 of the SPE paper should be the equivalent force, F_e, which is given by the following equation:

$$F_e = F_a + A_o P_o - A_i P_i$$

Equation 11 of the paper now becomes:

$$\sigma_a = \frac{F_a}{A_o - A_i} + \frac{F_e R r_o}{2I}$$

When the axial force is equal to or greater than zero the axial stress remains as given in equation 1 of the SPE paper.

Improved Solution:

The SPE paper substitutes the equations for the 3 stresses directly into the von Mises equation and solves for the unknown pressure. The algebraic manipulation required to develop this solution is very tedious. This algebraic manipulation must be repeated for to solve for either of the pressures. This becomes even more difficult with the correction given above.

In this development, the solution to the von Mises equation has been generalized by writing each of the three principal stress equations (radial, hoop and axial stresses) in the following form:

$$\sigma_i = \alpha_i + \beta_i P$$

In this equation i is 1 through 3 for the 3 principal stresses. P is the unknown pressure being solved for (either P_i or P_o). For the burst limits P is the internal pressure, P_i, and for the collapse limits P is the external pressure, P_o.

Substituting these generalized equations for the principal stresses into the von Mises equation (equation 18 in the SPE paper) yields the following:

$$AP^2 + BP + C = 0$$

or:

$$P = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A}$$

where:

$$A = \sum_{i=1}^{3} \beta_i^2 - (\beta_1\beta_2 + \beta_2\beta_3 + \beta_1\beta_3)$$

$$B = \sum_{i=1}^{3} \beta_i(2\alpha_i - \alpha_j - \alpha_k)$$

$$C = \sum_{i=1}^{3} \alpha_i^2 - (\alpha_1\alpha_2 + \alpha_2\alpha_3 + \alpha_1\alpha_3 + \sigma_y^2)$$

where j and k are the two values from 1 to 3 not equal to i and not equal to each other.

The Lame equation for the hoop stress at the inside surface (equation 15 of the SPE paper) is re-written as follows:

$$\sigma_h = P_i(\varphi - 1) - P_o\varphi$$

where:

$$\varphi = \frac{2r_0^2}{r_o^2 - r_i^2}$$

The SPE paper defines the inner most curve of 4 cases as the "limit curve". This limit curve is then multiplied by a safety factor to establish the "working limit curve". The unknown in this burst calculation is the internal pressure, P_i. For the burst limits the 4 cases are:

Case	Diameter	P_o
1	Nominal	0
2	1.06* Nominal	0
3	Nominal	P_{o_max}
4	1.06* Nominal	P_{o_max}

Table 1 - Burst Cases

For the collapse limits, the external pressure P_o is the unknown, and the following 4 cases are used:

Case	Diameter	P_i
1	Nominal	0
2	1.06* Nominal	0
3	Nominal	P_{i_max}
4	1.06* Nominal	P_{i_max}

Table 2 - Collapse Cases

The following table summarizes the α and β terms used:

	$P = P_i$ = UNKNOWN **Burst**	$P = P_o$ = UNKNOWN **Collapse**
Radial Stress $\sigma_r = -P_i$	$\alpha_1 = 0$ $\beta_1 = -1$	$\alpha_1 = -P_i$ $\beta_1 = 0$
Hoop Stress **Eq. H 9**	$\alpha_2 = -\varphi P_o$ $\beta_2 = \varphi - 1$	$\alpha_2 = P_i(\varphi - 1)$ $\beta_2 = -\varphi$
Axial Stress $F_a \geq 0$ **SPE paper Eq. 1** **(If not buckled)**	$\alpha_3 = F_a/(A_o - A_i)$ $\beta_3 = 0$	$\alpha_3 = F_a/(A_o - A_i)$ $\beta_3 = 0$
Axial Stress F_a<0 **Eq. H 2** **(If buckled)**	$\alpha_3 = \frac{F_a}{A_o - A_i} + \frac{Rr_o}{2I}(F_a + P_o A_o)$ $\beta_3 = -\frac{A_i R r_o}{2I}$	$\alpha_3 = \frac{F_a}{A_o - A_i} + \frac{Rr_o}{2I}(F_a - P_i A_i)$ $\beta_3 = \frac{A_o R r_o}{2I}$

Table 3 - α and β Terms

This solution procedure is easy to implement in software and is more flexible than previous solutions.

The results of these equations with and without helical buckling are interesting. Figure 4 shows an example of the results of these equations without helical buckling and without "imaginary points". Imaginary points are points that are mathematically possible but would require negative pressure either inside or outside the CT. These curves are similar to a single von Mises ellipse. There are a few points of inflection due to the 4 cases that are included and the elimination of the imaginary points.

Figure 5 shows the same results as Figure 4, but with helical buckling included. The left side of this curve is truncated significantly due to the inclusion of the helical buckling stresses. Whether or not to include these helical buckling stresses in CT limit considerations is currently a topic of discussion. The localized yielding that would occur if these limits were exceeded allows the CT to take on a slightly helical shape. However, the author is not aware of any failures that have occurred due to this yielding. Thus, the author recommends that these additional buckling stresses not be included in CT limit considerations.

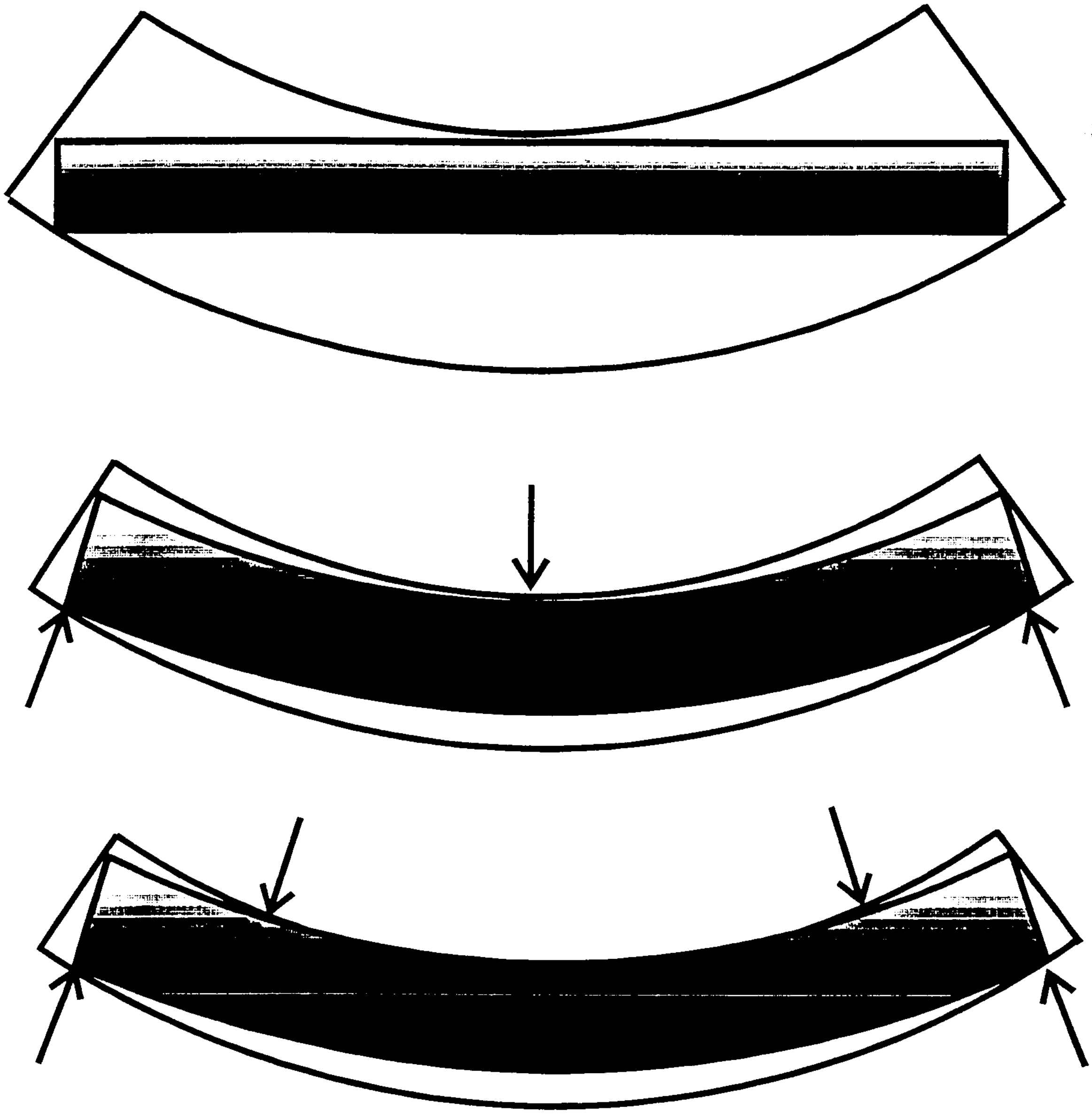

Figure 1

BHA Bending Cases

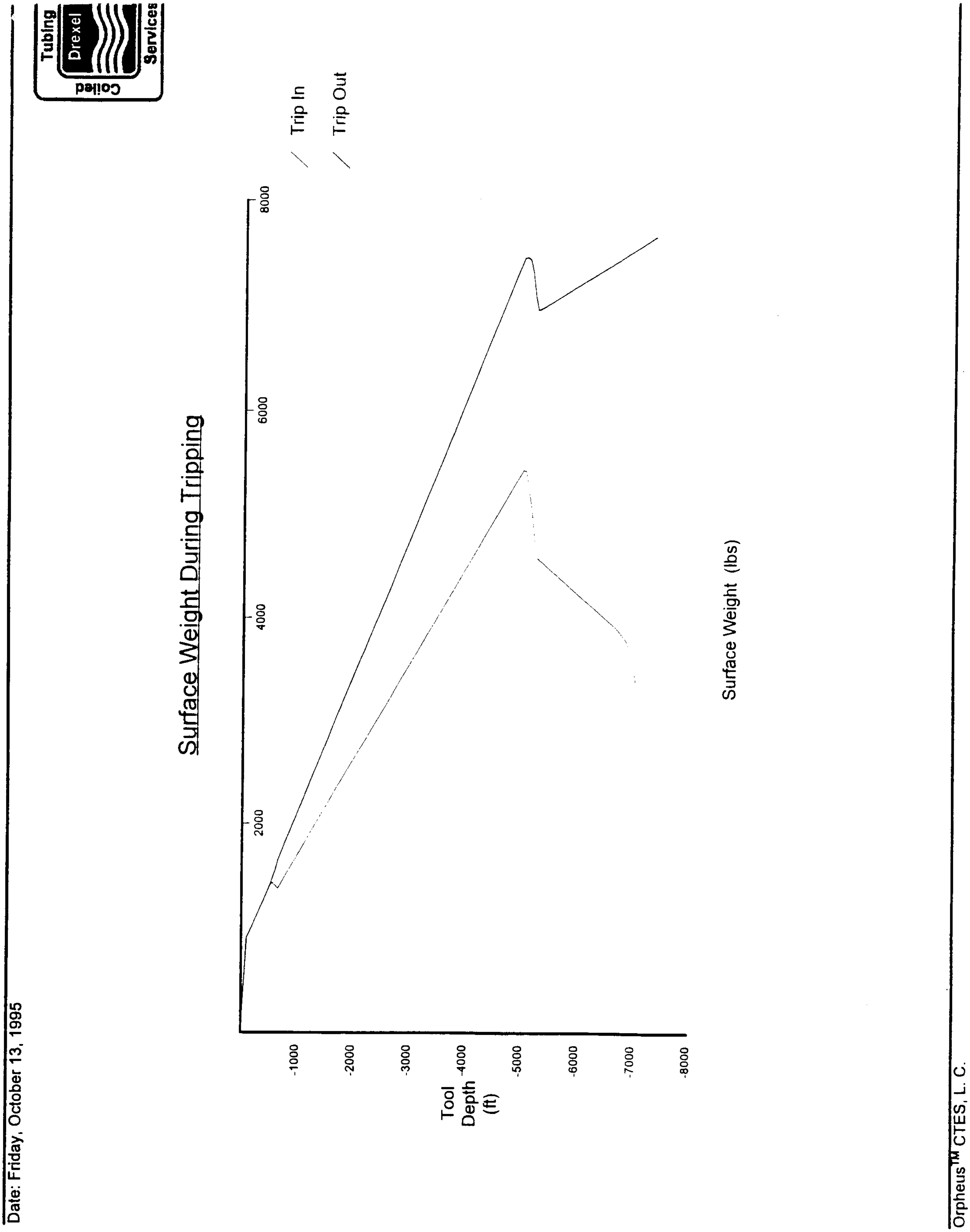

Figure 2
Weight vs Depth - Without BHA Bending Friction

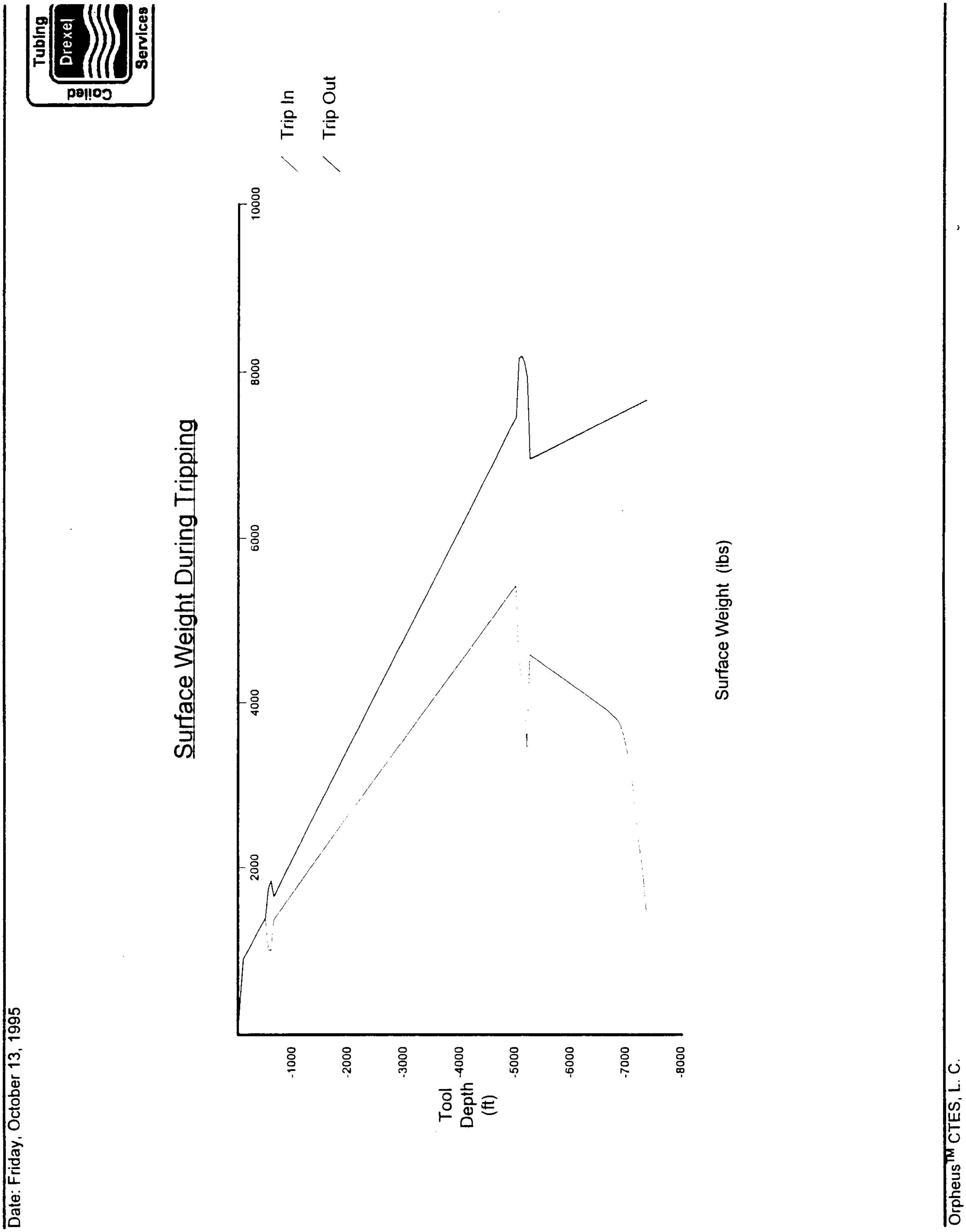

Figure 3
Weight vs Depth - With BHA Bending Friction

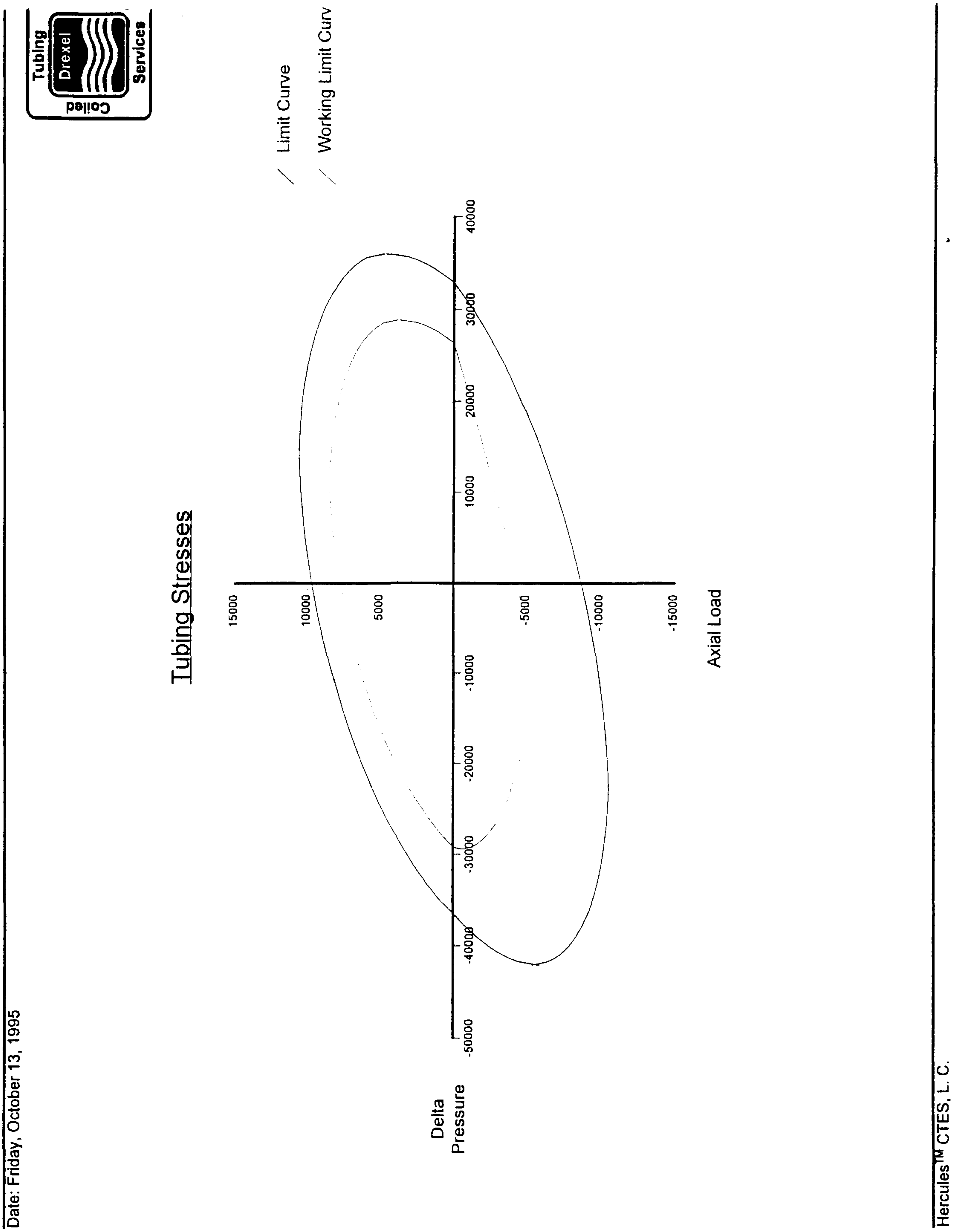

Figure 4
CT Limits - 1.5" X .109", 80ksi, Without Helical Buckling

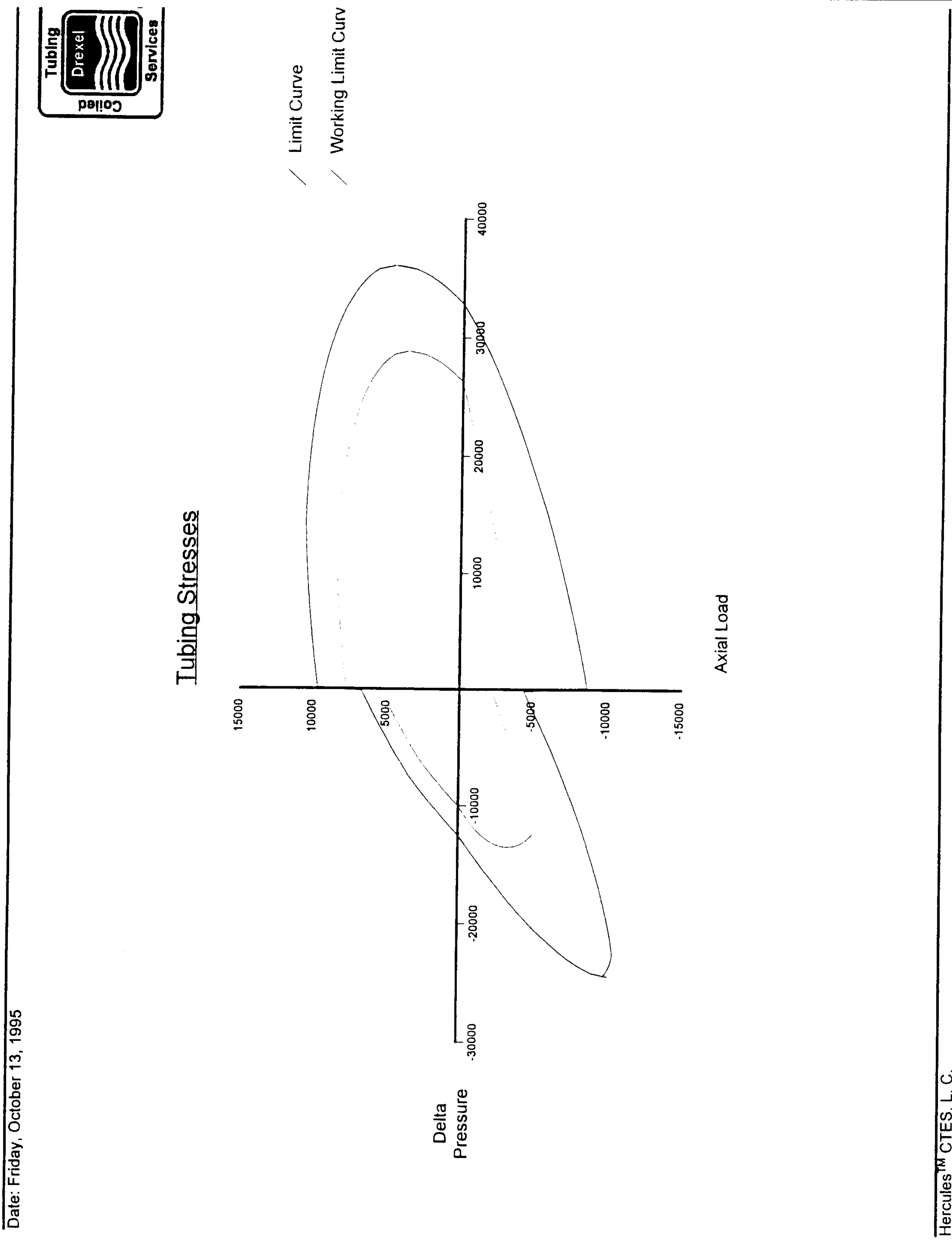

Figure 5
CT Limits - 1.5" X .109", 80ksi, With Helical Buckling