INSTALLATION LOADING AND STRESS ANALYSIS INVOLVED WITH PIPELINES INSTALLED BY HORIZONTAL DIRECTIONAL DRILLING

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Abstract

Pipelines installed by horizontal directional drilling (HDD) are subject to a combination of tension, bending, and external pressure. These installation loads, either individually or in combination, can be more severe than operational loads and may govern drilled path design or pipe specification. This is particularly true as the state of the art in horizontal directional drilling is advanced to larger pipe diameters and longer drilled lengths. This paper presents methods for calculating installation loads, including pulling forces, and analyzing combined stresses in steel pipe during installation and operation. Pipe to soil frictional and fluidic drag forces are discussed. A method for analyzing the effect of bends on pulling force is presented. Methods of analysis are illustrated with example calculations. The paper results from work done under the sponsorship of the Pipeline Research Committee at the American Gas Association and is taken from an engineering design guide produced specifically for HDD pipeline installation.

Installation Loads and Stresses

Pipeline properties (wall thickness, material grade) and pilot hole profiles must be selected such that the pipeline can be installed and operated without risk of damage. A pipeline installed by HDD can be examined for load and stress states by first breaking the problem into two distinct events: *installation* and *operation*. During installation the pipeline is subjected to:

> **tension** required to pull the pipe into the pilot hole and around curved sections in the hole, made up of,
>
> > **frictional drag** due to wetted friction between pipe and wall of hole,
> >
> > **fluidic drag** of pipe pulled through the viscous drilling mud trapped in the hole annulus,
> >
> > unbalanced gravity (**weight**) effects of pulling the pipe into and out of a hole at different elevations,
>
> **bending** as the pipe is forced to negotiate the curves in the hole,
>
> **external hoop** from the pressure exerted by the presence of the drilling mud in the annulus around the pipe (unless the pipe is flooded with a fluid at a similar pressure).

The stresses and failure potential of the pipe are a result of the interaction of these loads; therefore, calculation of the individual effects does not give an accurate picture of the stress limitations (Fowler and Langer) (Loh).

The purpose of this paper is to describe a reasonably simple means to estimate installation loads, calculate the resulting stresses, and determine if the overall HDD design is adequate. Loads and stresses experienced during installation of the pipeline are distinct from loads and stresses experienced during the service life of the pipeline and call for specific calculations and design checks. It is assumed in the method that follows that the pilot hole has been reamed approximately 12 inches larger than the pipeline diameter and that the annulus between the diameter and the reamed hole is filled with drilling mud of a known (or assumable) density. Reconsolidation of the formation surrounding the pilot hole will undoubtedly occur over time but if any significant formation pressure loads were exerted on the pipe during the pull back process, it is not expected that the pipe could be pulled in at all. Therefore, formation pressures are not considered in the analysis of the installation loads.

Pulling Load Calculation Method

Drilled Path Analysis

The first step in calculating pulling load is to analyze the drilled path. This analysis can be based on the designed drilled path or the "asbuilt" pilot hole. The drilled path centerline should be plotted in the two-dimensional vertical and horizontal planes with pipe length along the x-axis and distance from the reference line the y-axis. The entire drilled path should be broken into discrete sections: straight or curved. As many sections as necessary can be defined and there is no upper or lower bound on the length or arc length of any chosen section. Straight sections are those in which hole curvature is ideally zero but may actually have very slight curvature. Any pipe section with a net curvature less than that necessary to make the pipe deviate beyond the walls of the hole, which is roughly 12 inches larger in diameter than the pipe itself, can be considered a straight section. Curved sections should be short enough to assume one constant radius for the entire sweep of that section. If the radius of curvature in a particular curved section is variable, break that curved length into small enough sections to justify the assumption of constant radius of curvature in each smaller section. A plot of a simple designed drilled profile is shown in Figure 1.

As few sections as possible should be designated for the entire drilled path but as many as necessary to completely define its shape. The junctions between sections are assumed to be continuous (no sudden non-linearities in the shape of the pipe) and free of externally applied moment. The junction between a straight and curved section will constitute the beginning of the curvature for the curved section. It will later be modeled in this analysis as the end point of a simply supported beam. Curved sections may join straight sections or other curved sections but straight sections will always join curved sections on both ends since there is no reason to sub-divide long straight sections.

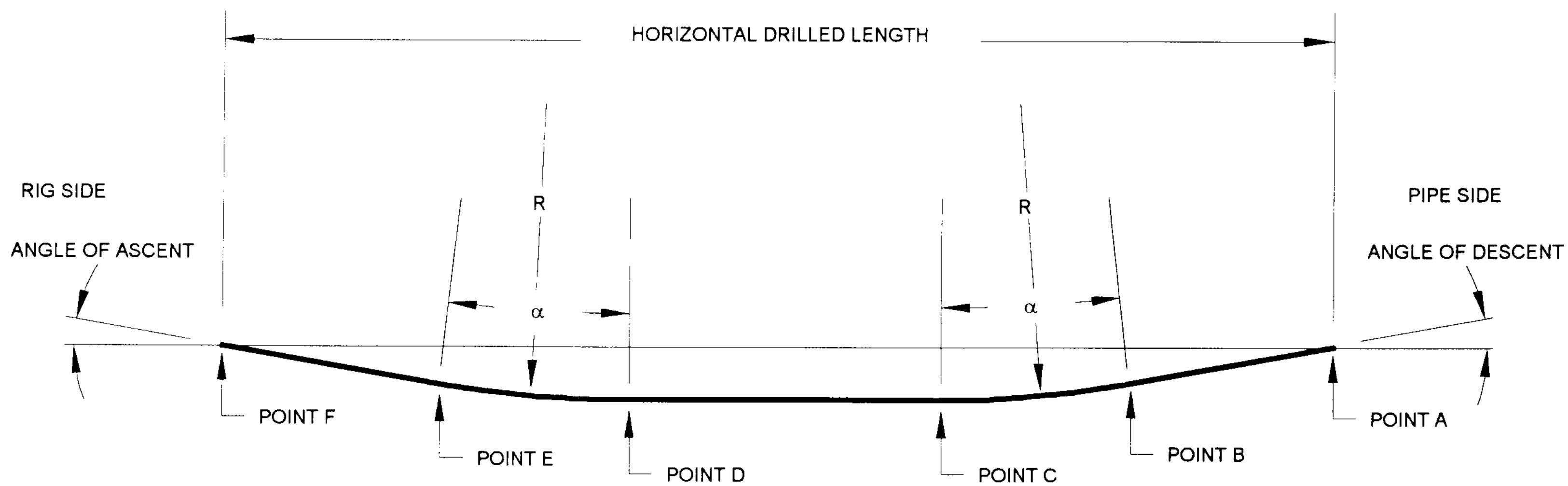


Figure 1
HDD Designed Drilled Profile

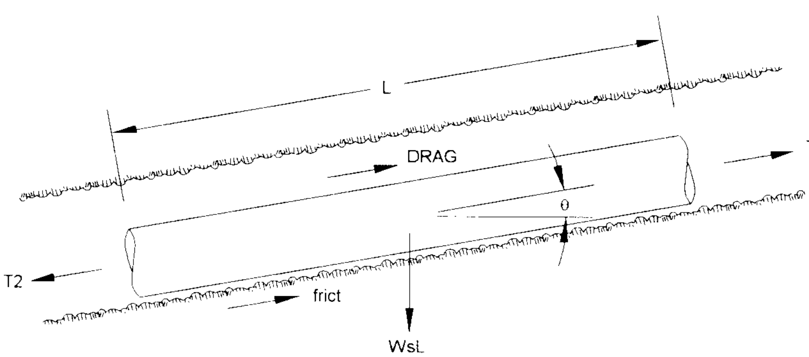
Pulling Loads

The load and stress calculation method described here begins with an elementary finite difference calculation of the pull force required to completely install the pipeline section from exit to entry point of the reamed pilot hole. The calculation is done in a way to define the maximum pulling loads which are assumed to occur at the moment the pipeline emerges from the entry point. Axial load in the pipe during the last instant of the pull back process will be distributed along its length from entry to exit. Cumulative axial load is composed of discrete axial loads occurring in each section of the hole due to friction between the pipe and the hole wall plus dynamic fluid friction required to make the pipe move through the viscous drilling mud. Because the hole wall/pipe frictional components are caused by the shape of the hole, the axial tension in the pipe at any given point can conveniently be considered to be confined to that particular location in the hole, regardless of which portion of the pipe is passing that point in the hole during any instant in the pull back process. This fact allows for worst case loads anywhere in the hole to be calculated only for the case where the pipe has just emerged from the hole.

Calculation - Straight Sections. The pipe is assumed to be pulled from the right to the left (as viewed in Figure 1) in all of the following models and calculations. The total pull force required to install the pipe is determined by summing the individual forces required to pull the pipe through each of the straight and curved sections defined in the hole profile. The modeling and calculation process must be done sequentially from right to left (i.e. from pipe side to rig side). Each straight section is modeled with variables as shown in Figure 2.

Figure 2

Straight Section Model



For any straight section, the left end tension, T2, is found from the static force balance,

$$T2 = T1 + |\,\text{frict}\,| + \text{DRAG} \pm Ws \times L \times \sin\theta \qquad (1)$$

The ± term is resolved as follows:

(-) if T2 tends downhole,
(+) if T2 tends upslope,
(0) if the hole section is horizontal, $\theta = 0$.

where,

T2 = tension at the **left** end of the section, in lbs, required to over come drag and friction.

T1 = tension at the **right** end of the section in lbs. This may be zero for the first section of the hole (Point A in Figure 1), or it may be determined by the drag of the pipe remaining on the rollers.

frict = friction between pipe and soil in lbs.

DRAG = fluidic drag between pipe and viscous drilling fluid in lbs.

Ws = effective (submerged) weight per foot of the pipeline plus internal contents (if filled with water) in lbs/ft.

L = length of section in feet.

θ = angle of the axis of the straight hole section relative to horizontal (zero equals horizontal, 90° is vertical)

Relationships defining the friction and fluidic drag terms are presented below. The absolute value for frict is used to insure that the it always acts in the proper direction relative to T2.

$$\text{frict} = Ws \times L \times \cos\theta \times \mu_{soil} \quad (2)$$

$$DRAG = 12 \times \pi \times D \times L \times \mu_{mud} \quad (3)$$

where,

μ_{soil} = average coefficient of friction between pipe and soil; recommended values between 0.21-0.30 (Maidla)

D = outside diameter of pipe in inches.

μ_{mud} = fluid drag coefficient for steel tube pulled through bentonite mud; recommended value 0.05 psi (NEN 3650).

Calculation - Curved Sections. Each curved hole section selected in the hole profile is modeled as shown in Figure 3 with the variables the same as the straight section, plus additional variables as defined below,

R = (constant) radius of curvature of the section in feet.

α = included angle of the curved section in degrees.

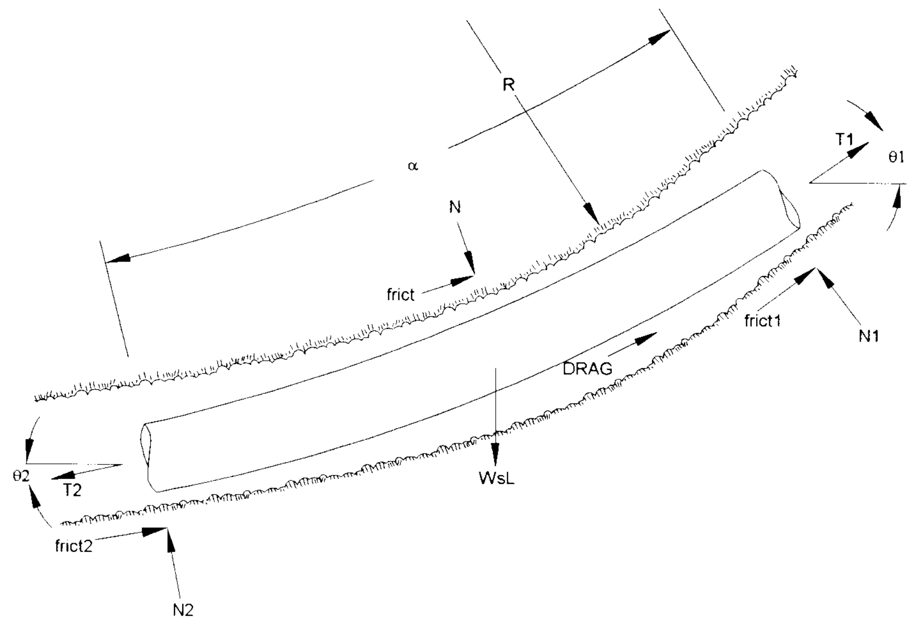
$\theta 1$ = angle in degrees from horizontal of T1, at right end of section.

θ2 = angle in degrees from horizontal of T2, at left end of section.

$$\theta = (\theta 1 + \theta 2)/2 \quad \text{(in degrees)}. \tag{4}$$

L is replaced by $L_{arc} = R \times \theta \times (\pi/180)$.

Figure 3
Curved Section Model



N1, N, & N2 = normal contact forces at right, center, & left points, respectively.

frict1, frict, & frict2 = frictional forces associated with normal forces at right, center, & left points, respectively.

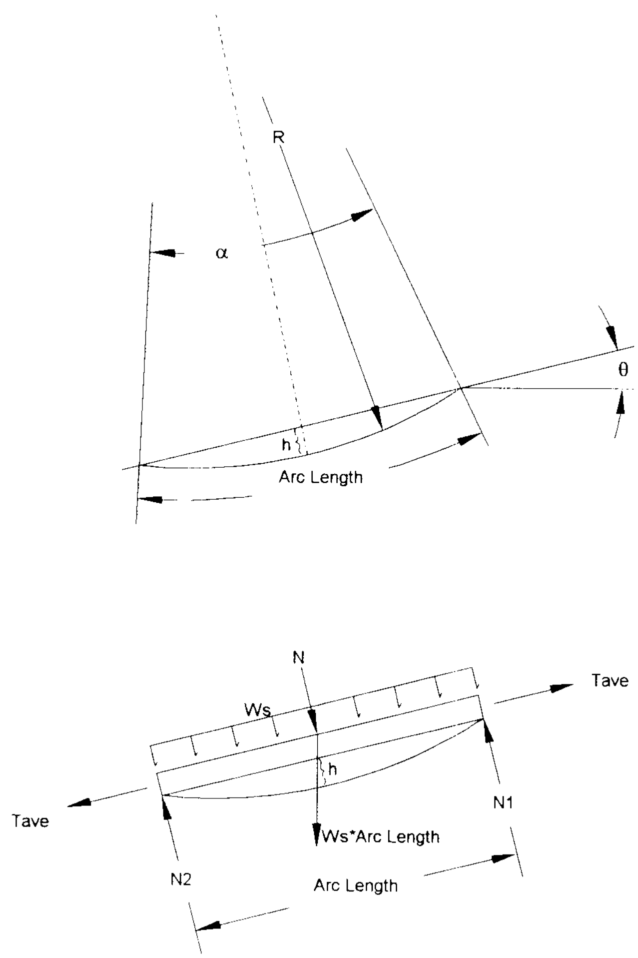
The distributed submerged weight of the pipe and contents are approximated to operate vertically at the center of the section despite the curvature of the pipe section.

To determine the normal forces of contact at the center and ends, each curved section is modeled as a beam in 3-point bending as shown in Figure 4. Loads on the beam are axial tension, T, plus distributed, submerged weight, Ws. A beam in 3-point, simply supported bending will not assume a constant radial, circular shape. For the bent pipe to fit in the circular hole section it must deflect enough to place its center at a point matching the displacement, h, of a circular arc with radius, R, where,

$$h = R \times [1 - \cos(\alpha/2)] \tag{5}$$

Note that this approximation of beam behavior in each curved section cannot be expected to be strictly accurate since more than three points of contact are likely for virtually any pipe/hole shape combination. However, since the objective is to determine normal contact forces and then calculate frictional forces exerted on the pipe as it is pulled through a curved section, this method is an adequate approximation and accounts for pipe stiffness relative to curve radius. The frictional forces become components in a later calculation of total pulling tension, T_{tot}. Sensitivity checks on the effect of T_{tot} in the overall stress analysis of the pipeline show that small variations in the value for T_{tot} do not grossly affect calculated stresses. Also, frictional loads are significantly affected by the assumed coefficients of friction, μ_{soil}, and drag, μ_{mud}.

Figure 4
3-Point Bending



The solution to the beam model to find N uses the vertical component of distributed weight, Ws × cos θ, as the main load and the arc length of the pipe section, L_{arc}, in place of L. From Roark's solution for elastic beam deflection,

$$N = [12 \times T \times h - (Ws/12) \times \cos\theta \times Y]/X \qquad (6)$$

where,

$$X = 3 \times L_{arc} - j/2 \times \tanh(U/2) \quad (7)$$

$$Y = 18 \times (L_{arc})^2 - j^2 \times [1 - 1/\cosh(U/2)] \quad (8)$$

$$j = (E \times I/T)^{1/2} \quad (9)$$

where,

E	=	Young's Modulus (2.9 x 10^7 psi for steel)
I	=	Bending Moment of Inertia in $inch^4$
U	=	$12 \times L_{arc} / j$ (10)
tanh	=	hyperbolic tangent
cosh	=	hyperbolic cosine

A value for T must be used to calculate both N and j. The proper value for T is the average of T1 and T2; therefore, an iterative solution is required to solve for T2 with accuracy. For hand calculations a few sensitivity checks on the value of T2 as the average, or T_{ave}, varies are quick to show that the assumed T_{ave} need not be exactly the average of T1 and T2.

$$frict = N \times \mu_{soil} \quad (11)$$

Therefore, end reactions are assumed to be N/2 and end friction forces are assumed to be f/2. Where N is a positive value (defined as downward acting as in Figure 4) it shows that the bending resistance and/or buoyancy of the pipe is sufficient to require a normal force acting against the top of the hole in order to get the pipe to displace downward by an amount equal to h. Where N is negative, the submerged pipe weight is sufficient to carry the pipe to the bottom of the curved section where an upward acting normal force is felt at the point of contact. Whether N is positive or negative in value, all friction values are taken as positive, acting in opposition to T2.

Assume forces acting along the curved path of the pipeline can be added as if acting in a straight line (as along a highly flexible, rope-like member).

Then,

$$T2 = T1 + 2 \times | frict | + DRAG \pm Ws \times L_{arc} \times \sin\theta \quad (12)$$

The $\pm$ term is resolved as follows:

(-) if T2 tends downhole,
(+) if T2 tends upslope,
(0) if the hole section is horizontal, $\theta = 0$.

The absolute value of frict is used because friction always retards pipe movement caused by T2.

Calculation - Total Pulling Load. The total force, T_{tot}, required to pull the entire pipeline into the reamed pilot hole is the sum of all straight and curved section values for ΔT, (T2 - T1).

$$T_{tot} = \Sigma_i \text{ (T2 - T1), for i sections} \tag{13}$$

Installation Stress Analysis

The worst case stress condition for the pipe will be located where the most serious combination of tensile, bending and/or hoop stresses occur simultaneously. This is not always obvious in looking at the hole profile because the interactions of the three loading conditions is not necessarily intuitive. To be sure that the point with the worst case condition is isolated it may be necessary to do a critical stress analysis for several suspect locations. In general, highest stresses will be felt at locations of tight radius bending, high tension (closer to the rig side), and high hydrostatic head (deepest point). The approach that follows is taken from the API Recommended Practice 2A-WSD. Lower case f's represent actual stresses, upper case F's represent allowable stresses.

Individual Loads

For any selected location in the drilled path profile that is suspected of being a critical stress location, first calculate the individual stresses for the specific loading conditions (tensile, bending, hoop stresses) and compare against allowable levels for these stress states. If no individual stress condition appears to cause overstress failure, the combined stress state is compared in two interaction equations presented as unity checks. That is, the combined stresses in the interaction equation must be less than 1.0 for the pipe to be safe from collapse by bending or hoop collapse in all regimes (plastic, elastic and transition).

Tensile stress.

$$f_t = T/A \tag{14}$$

where,

T	=	tension at the point of interest in lbs.
A	=	cross-sectional area of pipe wall in inches.

Bending stress.

$$f_b = (E \times D)/(24 \times R) \tag{15}$$

Hoop Stress.

$$f_h = (\Delta p \times D)/(2 \times t) \tag{16}$$

where,

t = pipe wall thickness in inches

and where Δp (psi) is equal to the difference between hydrostatic pressure exerted by the drilling mud in the hole acting on the outside of the pipe and the pressure from water, mud or air acting on the inside of the pipe, at the depth of the point of interest (Δp producing a compressive external hoop stress is taken as positive),

$$\text{external mud pressure} = \text{mud wt (ppg)} \times \text{depth (ft)}/19.25 \tag{17}$$

Compare each actual stress, f, to its allowable stress, F, as follows:

Tension.

$$F_t = 0.9 \times SMYS \tag{18}$$

where,

SMYS = Specified Minimum Yield Strength in psi

Bending.

$$F_b = 0.75 \times SMYS \quad \text{for } D/t \leq 1{,}500{,}000/SMYS \tag{19}$$

$$F_b = [0.84 - \{1.74 \times SMYS \times D/(E \times t)\}] \times SMYS \quad \text{for } 1{,}500{,}000/SMYS < D/t \leq 3{,}000{,}000/SMYS \tag{20}$$

$$F_b = [0.72 - \{0.58 \times SMYS \times D/(E \times t)\}] \times SMYS \quad \text{for } 3{,}000{,}000/SMYS < D/t \leq 300{,}000 \tag{21}$$

Hoop Buckling Stress.

$$f_h < F_{hc}/1.5 \tag{22}$$

where F_{hc}, critical hoop buckling stress, is a function of F_{he}, elastic hoop buckling stress as follows:

$$F_{he} = 0.88 \times E \times (t/D)^2 \quad \text{(for long, unstiffened cylinders)} \tag{23}$$

and,

$$F_{hc} = F_{he} \quad \text{for } F_{he} \leq 0.55 \times SMYS \tag{24}$$

For inelastic hoop buckling,

$$F_{hc} = 0.45 \times SMYS + 0.18 \times F_{he} \quad \text{for } 0.55 \times SMYS < F_{he} \leq 1.6 \times SMYS \tag{25}$$

$$F_{hc} = 1.31 \times SMYS/[1.15 + (SMYS/F_{he})] \quad \text{for } 1.6 \times SMYS < F_{he} \leq 6.2 \times SMYS \tag{26}$$

$$F_{hc} = SMYS \quad \text{for } F_{he} > 6.2 \times SMYS \tag{27}$$

Combined Loads

If all preliminary checks indicate that the loading on the pipe will not cause failure (overstress or buckling) due to a single load condition, the suspect stress locations must be checked for safety under interactive combined loading by conducting two unity checks; first a dual load condition (tension plus bending) and finally a full interactive load unity check which must be satisfied for combined tensile, bending and hoop stresses.

The unity check for combined stresses, **tensile and bending,** is:

$$f_t/(0.9 \times SMYS) + f_b/F_b \leq 1.0 \tag{28}$$

The unity check for full interaction of **tensile, bending and external hoop stresses** is:

$$A^2 + B^2 + 2\nu \times |A| \times B \leq 1 \tag{29}$$

where,

$$A = (f_t + f_b - 0.5 \times f_h) \times 1.25/SMYS \tag{30}$$

$$B = 1.5 \times f_h/F_{hc} \tag{31}$$

ν = Poisson's ratio (0.3 for steel)

Satisfying the unity check equation for combined loading at all particular locations of suspect serious stresses, after first satisfying all single-load condition stress cases at those locations, is sufficient to qualify the design for an HDD pipeline installation. If the unity check results in a value greater than one it does not mean that the pipeline will necessarily fail (by overstress or buckling) but it does indicate that the combined stress state places the design in a range where some test specimens under similar stress states have been found to be subject to failure.

The combined stress interaction analysis described above is useful for finding solutions to field problems where conditions differ from original design expectations. One typical case might be if a pilot hole has a few spots where the radius is tighter than designed and, thus, the unity check exceeds one. In this case it is possible to find a modified solution to the installation problem by varying one load parameter and checking its effect on the stress interaction. For example, if a certain HDD design profile fails the unity check with the pipe installed empty it may pass the unity check when filled with water, thus reducing hoop stress and decreasing buoyancy.

Example Pulling Load Calculation

Use the example HDD pilot hole profile as shown in Figure 5. For this example the hole is assumed to occupy only a single plane, i.e. there is no significant curvature into or out of the plane of the paper. The pipe side (right) and rig side (left) are at the same elevation and 1,500 feet apart. The total depth for the horizontal straight section is 100 feet below the entry/exit datum elevation. The total arc length of the centerline of the hole profile is 1,525 feet. The right curved section is a 20° arc at a 1,000 foot radius. The left curved section is a 14° arc at a 1,200 foot radius. The following particulars are given for this example installation:

D = 12.75 in	t = 0.25 in	Ws = -46.21 lb/ft
Pipe: Grade B Steel	SMYS = 35,000 psi	E = 2.9×10^7 psi
$\mu_{soil} = 0.3$	$\mu_{mud} = 0.05$ psi	

Mud wt. = 12 ppg This is a conservative (high drilled solids content assumed) mud weight value. 12 ppg = 89.76 lb/ft^3
Formation is predominately soft clay
Right side tension (pull-back) on the pipe as it enters the hole = 0
Pipe is installed empty

Examination of the geometry of the pilot hole allows for it to be broken down into five convenient sections for calculating pull forces. From right to left they are:

Section	Type	Angle	Length	T2
A to B	straight	$\theta = 20°$	L = 116.1 ft	↙
B to C	curved, R=1000 ft	$\theta = 10°$ $\alpha = 20°$	L_{arc} = 349.1 ft	↙
C to D	straight	$\theta = 0$	L = 500.3 ft	←
D to E	curved, R=1200 ft	$\theta = 7°$ $\alpha = 14°$	L_{arc} = 293.2 ft.	↖
E to F	straight	$\theta = 14°$	L = 266.2 ft	↖

Pulling Loads

Straight Section at Point B.

$$\Delta T_{BA} = T_B - T_A$$
$$= |\text{frict}| + \text{DRAG} - Ws \times L \times \sin\theta$$

$$|\text{frict}| = Ws \times L \times \cos\theta \times \mu_{soil}$$
$$= (-46.21\ \text{lb/ft})(116.1\ \text{ft})(\cos 20°)(0.3)$$
$$= 1{,}512\ \text{lb}$$

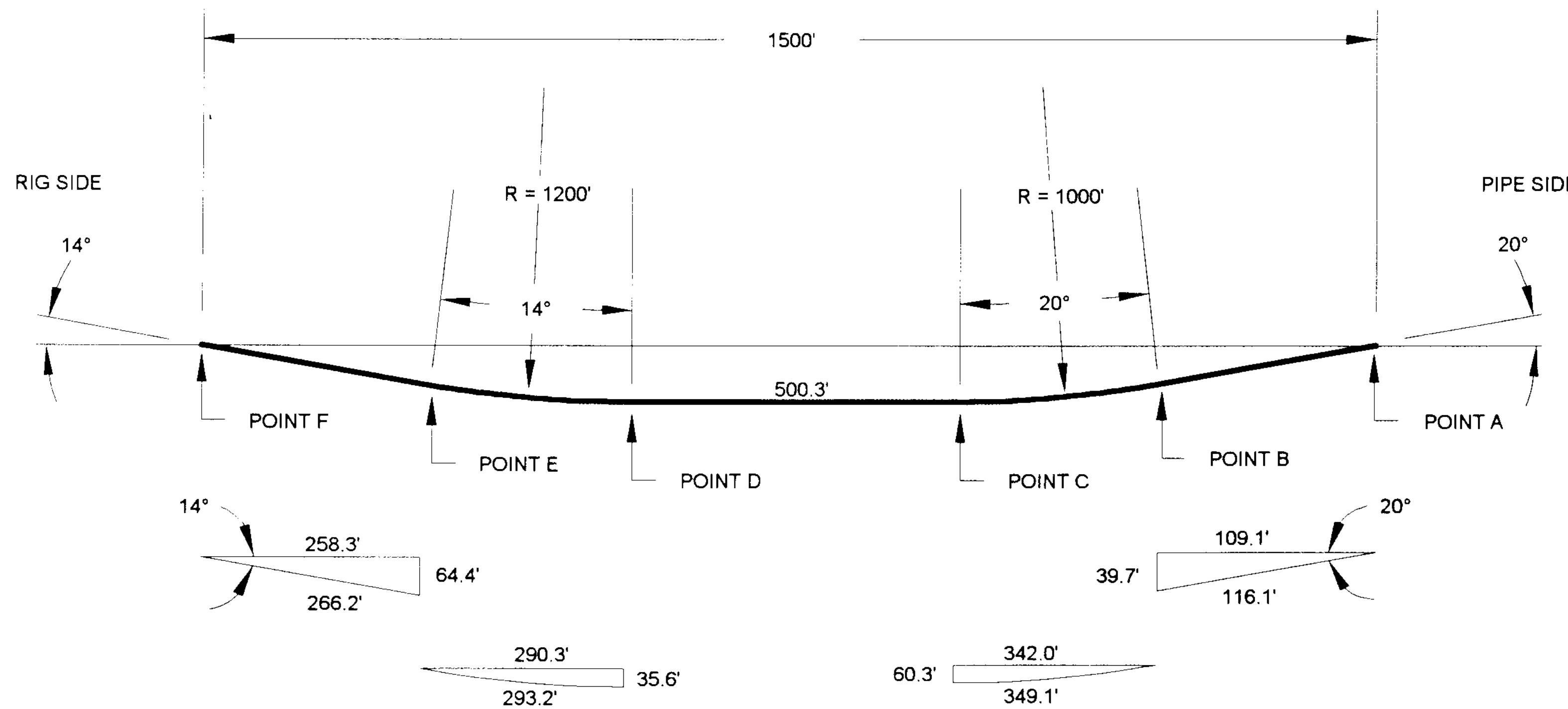


Figure 5
Example, HDD Pilot Hole Profile

$$\text{DRAG} = 12 \times \pi \times D \times L \times \mu_{mud}$$
$$= (12)(\pi)(12.75\ \text{in})(116.1\ \text{ft})(0.05\ \text{lb/in}^2)$$
$$= 2{,}790\ \text{lb}$$

$$Ws \times L \times \sin\theta = (-46.21\ \text{lb/ft})(116.1\ \text{ft})(\sin 20°)$$
$$= -1{,}835\ \text{lb}$$

$$\Delta T_{BA} = 1{,}512\ \text{lb} + 2{,}790\ \text{lb} - (-1{,}835\ \text{lb})$$
$$= \mathbf{6{,}137\ lb}$$

$$T_B = \Delta T_{BA} + T_A$$
$$= \mathbf{6{,}137\ lb} \quad \Leftarrow \text{Pull Load at Point B}$$

Curved Section at Point C.

$$h = R \times [1 - \cos(\alpha/2)]$$
$$= (1000\ \text{ft})(1 - \cos 10°)$$
$$= 15.19\ \text{ft}$$

$$I = \pi \times (D - t)^3 \times t/8$$
$$= \pi(12.75\ \text{in} - 0.25\ \text{in})^3(0.25\ \text{in})/8$$
$$= 191.75\ \text{in}^4$$

Assume T_{ave} for section = 10,000 lb to start iterative solution.

$$j = (E \times I/T_{ave})^{1/2}$$
$$= [2.9 \times 10^7\ \text{psi}\ (191.75\ \text{in}^4)/10{,}000\ \text{lb}]^{1/2}$$
$$= 745.7\ \text{in}$$

$$U = 12 \times L_{arc}/j$$
$$= (12)(349.1\ \text{ft})/(745.7\ \text{in})$$
$$= 5.62$$

$$X = 3 \times L_{arc} - (j/2) \times \tanh(U/2)$$
$$= (3)(349.1\ \text{ft}) - (0.5)(745.7\ \text{in})\tanh(5.62/2)$$
$$= 677.15\ \text{in}$$

$$Y = 18 \times (L_{arc})^2 - j^2 \times [1 - 1/\cosh(U/2)]$$
$$= (18)(349.1\ \text{ft})^2 - (745.7\ \text{in})^2 \times [1 - 1/\cosh(5.62/2)]$$
$$= 1{,}704{,}320\ \text{in}^2$$

$$N = [12 \times T_{ave} \times h - (Ws/12) \times \cos\theta \times Y]/X$$
$$= [(12)(10{,}000\ \text{lb})(15.19\ \text{ft}) - (1/12)(-46.21\ \text{lb/ft})(\cos 10°)(1{,}704{,}320\ \text{in}^2)]/677.15\ \text{in}$$
$$= 12{,}237\ \text{lb}$$

The positive value indicates that N acting down is the reaction normal force required at the top of the hole to bend the buoyant pipe (in this example) into the curve required to match the pilot hole.

$$\Delta T_{CB} = T_C - T_B$$
$$= 2 \times |\text{ frict }| + \text{DRAG} - Ws \times L_{arc} \times \sin\theta$$

$$|\text{ frict }| = N \times \mu_{soil}$$
$$= (12{,}237 \text{ lb})(0.3)$$
$$= 3{,}671 \text{ lb}$$

$$\text{DRAG} = 12 \times \pi \times D \times L_{arc} \times \mu_{mud}$$
$$= (12)(\pi)(12.75 \text{ in})(349.1 \text{ ft})(0.05 \text{ lb/in}^2)$$
$$= 8{,}389 \text{ lb}$$

$$Ws \times L_{arc} \times \sin\theta = (-46.21 \text{ lb/ft})(349.1 \text{ ft})(\sin 10^{\circ})$$
$$= -2{,}801 \text{ lb}$$

$$\Delta T_{CB} = 2(3{,}671 \text{ lb}) + 8{,}389 \text{ lb} - (-2{,}801 \text{ lb})$$
$$= \mathbf{18{,}533 \text{ lb}}$$

$$T_C = \Delta T_{CB} + T_B$$
$$= \mathbf{24{,}670 \text{ lb}} \quad \Leftarrow \text{ Pull Load at Point C before } T_{ave} \text{ assumption check}$$

Check accuracy of assuming T_{ave} = 10,000 at the beginning of this iterative solution.

$$T_{ave} = (T_C + T_B)/2$$
$$= (24{,}670 \text{ lb} + 6{,}137 \text{ lb})/2$$
$$= 15{,}404 \text{ lb}$$

Percent difference is (15,404 - 10,000)/10,000 × 100%, which is equal to 54%. This does not fall within an acceptable level of 10%, so the iteration process begins. Select 15,404 as the value for the assumed T_{ave}. Run through the same calculations and once again compare the calculated T_{ave} with the assumed T_{ave}. Continue the iteration until the percent difference is equal to or below 10%. In this example, a little iteration reveals that a more exact value for T_{ave} is 15,698 lb, and thus,

$$\Delta T_{CB} = 2(3{,}966 \text{ lb}) + 8{,}389 \text{ lb} - (-2{,}801 \text{ lb})$$
$$= \mathbf{19{,}122 \text{ lb}}$$

$$T_C = \mathbf{25{,}259 \text{ lb}} \quad \Leftarrow \text{ Pull Load at Point C}$$

Straight Section at Point D.

$$\Delta T_{DC} = T_D - T_C$$

$= |\text{frict}| + \text{DRAG} - Ws \times L \times \sin\theta$

$$|\text{frict}| = Ws \times L \times \cos\theta \times \mu_{soil}$$
$$= (-46.21 \text{ lb/ft})(500.3 \text{ ft})(\cos 0^{o})(0.3)$$
$$= 6{,}936 \text{ lb}$$

$$\text{DRAG} = 12 \times \pi \times D \times L \times \mu_{mud}$$
$$= (12)(\pi)(12.75 \text{ in})(500.3 \text{ ft})(0.05 \text{ lb/in}^2)$$
$$= 12{,}024 \text{ lb}$$

$$Ws \times L \times \sin\theta = (-46.21 \text{ lb/ft})(500.3 \text{ ft})(\sin 0^{o})$$
$$= 0 \text{ lb}$$

$$\Delta T_{DC} = 6{,}936 \text{ lb} + 12{,}024 \text{ lb} - 0 \text{ lb}$$
$$= \mathbf{18{,}960 \text{ lb}}$$

$$T_D = \Delta T_{DC} + T_C$$
$$= \mathbf{44{,}219 \text{ lb}}$$ ⇐ Pull Load at Point D

Curved section at Point E.

The iterative solution will yield $T_{ave} = 51{,}545$ lb. The calculation is as follows:

$$h = R \times [1 - \cos(\alpha/2)]$$
$$= (1200 \text{ ft})(1 - \cos 7^{o})$$
$$= 8.94 \text{ ft}$$

$$I = 191.75 \text{ in}^4$$

$$j = (E \times I/T_{ave})^{1/2}$$
$$= [(2.9 \times 10^7 \text{ psi})(191.75 \text{ in}^4)/(51{,}545 \text{ lb})]^{1/2}$$
$$= 328.45 \text{ in}$$

$$U = 12 \times L_{arc}/j$$
$$= (12)(293.2 \text{ ft})/(328.45 \text{ in})$$
$$= 10.71$$

$$X = 3 \times L_{arc} - (j/2) \times \tanh(U/2)$$
$$= (3)(293.2 \text{ ft}) - (0.5)(328.45 \text{ in})\tanh(10.71/2)$$
$$= 715.38 \text{ in}$$

$$Y = 18 \times (L_{arc})^2 - j^2 \times [1 - 1/\cosh(U/2)]$$
$$= [(18)(293.2 \text{ ft})^2 - (328.45 \text{ in})^2 \times [1 - 1/\cosh(10.71/2)]$$
$$= 1{,}440{,}532 \text{ in}^2$$

$$N = [12 \times T_{ave} \times h - (Ws/12) \times \cos\theta \times Y]/X$$
$$= [(12)(51{,}545 \text{ lb})(8.94 \text{ ft}) - (1/12)(-46.21 \text{ lb/ft})(\cos 7^o)(1{,}440{,}532 \text{ in}^2)]/715.38 \text{ in}$$
$$= 15{,}427 \text{ lb}$$

The positive value indicates that N acting down is the reaction normal force required at the top of the hole to bend the buoyant pipe (in this example) into the curve required to match the pilot hole.

$$\Delta T_{ED} = T_E - T_D$$
$$= 2 \times |\text{ frict }| + \text{DRAG} + Ws \times L \times \sin\theta$$

$$|\text{ frict }| = N \times \mu_{soil}$$
$$= (15{,}427 \text{ lb})(0.3)$$
$$= 4{,}628 \text{ lb}$$

$$\text{DRAG} = 12 \times \pi \times D \times L \times \mu_{mud}$$
$$= (12)(\pi)(12.75 \text{ in})(293.2 \text{ ft})(0.05 \text{ lb/in}^2)$$
$$= 7{,}047 \text{ lb}$$

$$Ws \times L_{arc} \times \sin\theta = (-46.21 \text{ lb/ft})(293.2 \text{ ft})(\sin 7^o)$$
$$= -1{,}651 \text{ lbf}$$

$$\Delta T_{ED} = 2(4{,}628 \text{ lb}) + 7{,}047 \text{ lb} + (-1{,}651 \text{ lb})$$
$$= \mathbf{14{,}652 \text{ lb}}$$

$$T_E = \Delta T_{ED} + T_D$$
$$= \mathbf{58{,}871 \text{ lb}} \quad \Leftarrow \text{ Pull Load at Point E}$$

Straight Section at Point F.

$$\Delta T_{FE} = T_F - T_E$$
$$= |\text{ frict }| + \text{DRAG} + Ws \times L \times \sin\theta$$

$$|\text{ frict }| = Ws \times L \times \cos\theta \times \mu_{soil}$$
$$= (-46.21 \text{ lb/ft})(266.2 \text{ ft})(\cos 14^o)(0.3)$$
$$= 3{,}581 \text{ lb}$$

$$\text{DRAG} = 12 \times \pi \times D \times L \times \mu_{mud}$$
$$= (12)(\pi)(12.75 \text{ in})(266.2 \text{ ft})(0.05 \text{ lb/in}^2)$$
$$= 6{,}398 \text{ lb}$$

$$Ws \times L \times \sin\theta = (-46.21 \text{ lb/ft})(266.2 \text{ ft})(\sin 14^o)$$
$$= -2{,}976 \text{ lb}$$

$$\Delta T_{FE} = 3{,}581 \text{ lb} + 6{,}398 \text{ lb} + (-2{,}976 \text{ lb})$$
$$= \mathbf{7{,}003 \text{ lb}}$$

$$T_F = \Delta T_{FE} + T_E$$
$$= \mathbf{65{,}874\ lb} \quad \Leftarrow \text{Pull Load at Point F}$$

Total Pull Load, T_{tot}. The total pulling load is simply the sum of all the individual loads which is equal to the pulling load at point F.

$$T_{tot} = \Delta T_{BA} + \Delta T_{CB} + \Delta T_{DC} + \Delta T_{ED} + \Delta T_{FE}$$
$$= T_F$$
$$= \mathbf{65{,}874\ lb}$$

Example Installation Stress Analysis

A complete analysis of the installation stresses experienced by the pipe requires stress calculations for any point where the combined stresses may be near a maximum. For this example case, examination of the pilot hole plot shows that the most likely location for high stress due to combined loading is at point E. This point is closer to the rig side and, therefore, will have relatively high local tension. It also is part of a tight radius curve (R = 1200 ft) and is near the deepest point with the highest hydrostatic mud pressure. For completeness and confidence, stresses at points C and D should also be checked.

Individual Stresses at Point E

From Figure 5, the depth at Point E is 64.4 ft below datum elevation.

Tensile stress.

$$f_t = T_E/A$$
$$= (58{,}871\ \text{lb})/[(\pi/4)(12.75^2 - 12.25^2)]$$
$$= 5{,}997\ \text{psi}$$

Bending stress.

$$f_b = (E \times D)/(24)(R)$$
$$= (2.9 \times 10^7)(12.75)/[(24)(1200\ \text{ft})]$$
$$= 12{,}839\ \text{psi}$$

External Hoop Stress.

$$f_h = (\Delta p \times D)/(2 \times t)$$
$$= (40.15\ \text{psi})(12.75\ \text{in})/[(2)(0.25\ \text{in})]$$
$$= 1{,}024\ \text{psi}$$

$$\Delta p = (12\ \text{ppg})(64.4\ \text{ft})/(19.25)$$
$$= 40.15\ \text{psi}$$

Allowable Tension.

$$F_t = 0.9 \times SMYS$$
$$= (0.9)(35{,}000\ psi)$$
$$= 31{,}500\ psi$$

Note that f_t is less than 31,500 psi, so tension is within allowable limits.

Allowable Bending.

$$F_b = [0.84 - \{1.74 \times SMYS \times D/(E \times t)\}] \times SMYS$$
$$\text{for } 1{,}500{,}000/SMYS < D/t \leq 3{,}000{,}000/SMYS$$

$$F_b = [0.84 - \{(1.74)(35{,}000\ psi)(12.75\ in)/(2.9 \times 10^7\ psi \times 0.25\ in)\}](35{,}000\ psi)$$
$$= 25{,}652\ psi$$

Note that f_b is less than 25,652 psi, so bending is within allowable limits.

Allowable Elastic Hoop Buckling.

$$F_{he} = 0.88 \times E \times (t/D)^2$$
$$= 0.88(2.9 \times 10^7)(0.25\ in/12.75\ in)^2$$
$$= 9{,}812\ psi$$

and,

$$F_{hc} = F_{he}$$
$$\text{for } F_{he} \leq 0.55 \times 35{,}000\ psi$$
$$F_{hc} = 9{,}812\ psi$$

Note that f_h is less than $F_{hc}/1.5$ = 6,541 psi, so external hoop stress is within allowable limits for buckling.

Combined Load Interactions at Point E

Since all individual stress checks are acceptable, the combined load interaction checks will now be examined.

Tensile and Bending.

$f_t/(0.9 \times SMYS) + (f_b/F_b) \leq 1.0$ [Unity check]

5,997 psi/(0.9 × 35,000 psi) + (12,839 psi/25,652 psi) = 0.69

0.69 < 1.0 so combined tensile and bending at Point E is acceptable.

Tensile, Bending, and External Hoop.

$$A^2 + B^2 + 2\nu \times |A| \times B \leq 1$$

$$\begin{aligned} A &= [f_t + f_b - 0.5 \times f_h] \times 1.25/SMYS \\ &= [5{,}997 \text{ psi} + 12{,}839 \text{ psi} - (0.5)(1{,}024 \text{ psi})](1.25)/(35{,}000 \text{ psi}) \\ &= 0.654 \end{aligned}$$

$$\begin{aligned} B &= 1.5\, f_h/F_{hc} \\ &= (1.5)(1{,}024 \text{ psi})/(9{,}812 \text{ psi}) \\ &= 0.157 \end{aligned}$$

$$(0.654)^2 + (0.157)^2 + (2)(0.3)(0.654)(0.157) = 0.51$$

0.51 < 1.0 so combined stresses at Point E are acceptable.

Operating Loads and Stresses

With one exception, the operating loads and stresses in a pipeline installed by HDD are not materially different from those experienced by pipelines installed by cut and cover techniques; therefore, past procedures for calculating and limiting stresses can be applied. The exception involves elastic bending. A pipeline installed by HDD will contain elastic bends. The pipe will not be bent to conform to the drilled hole as a pipeline installed by cut and cover is bent to conform to the ditch. Bending stresses imposed by the HDD installation method will generally not be severe. However, they should be checked in combination with other longitudinal and hoop stresses to insure that acceptable limits are not exceeded. The operating loads imposed on a pipeline installed by HDD are listed below:

internal pressure from the fluid flowing in it,

elastic bending as the pipe conforms to the shape of the drilled hole,

thermal resulting from the difference between the constructed (locked in) temperature and the operating temperature.

Formulas for calculating stresses produced by internal pressure and elastic bending have been previously listed (5.15 and 5.16) and are repeated below.

Bending stress.

$$f_b = (E \times D)/(24 \times R) \qquad (15)$$

Hoop Stress.

$$f_h = (\Delta p \times D)/(2 \times t) \qquad (16)$$

In this case Δp is equal to the difference between hydrostatic pressure exerted by groundwater acting on the outside of the pipe and the pressure from the fluid (gas) flowing inside of the pipe. Note that for this analysis Δp producing a tensile external hoop stress is taken as positive.

The formula for calculating thermal stresses is taken from ASME/ANSI B31.4

Thermal Stress.

$$f_t = (E \times k) \times (T_1 - T_2) \qquad (31)$$

where,

k = the coefficient of thermal expansion for steel (0.0000065 inches per inch per °F)(ANSI B31.4)

T_1 = Constructed temperature in °F

T_2 = Operating temperature in °F

Combined Stresses and Limitations

Combined stresses can be analyzed by calculating the maximum shear stress on a small element in the pipeline. This maximum shear stress should be limited to 45% of the SMYS of the pipe (ASME/ANSI B31.4). The maximum shear stress at any element is calculated using the following formula (Timoshenko and Gere).

$$f_v = (f_{hoop} - f_{long})/2 \qquad (32)$$

where,

f_{hoop} = the total hoop stress acting on the element

f_{long} = the total longitudinal stress acting on the element

Note that in this analysis all tensile stresses are positive and all compressive stresses are negative.

The total hoop stress is determined using equation 5.16. The total longitudinal stress is determined by taking the sum of the longitudinal stresses resulting from bending (equation 5.15), thermal (equation 5.31), and internal pressure (equation 5.33). Longitudinal stress from internal pressure is calculated as follows:

$$f_p = f_h \times \nu \qquad (33)$$

where,

ν = Poisson's ratio (0.3 for steel)

The maximum shear stress will occur in an element on the compressive side of an elastic bend and at the maximum distance from the neutral axis of the bend.

Example Operating Stress Analysis

Using the same example previously analyzed for installation stresses, check the combined operating stresses. Relevant data are listed below:

D = 12.75 in	t = 0.25 in	
Pipe: Grade B Steel	SMYS = 35,000 psi	$E = 2.9 \times 10^7$ psi

The total depth for the horizontal straight section is 100 feet below the entry/exit datum elevation and the groundwater table is assumed to be 10 feet below the entry/exit datum. The shortest radius of curvature is 1,000 feet on the right curve. The maximum allowable operating pressure is 720 psi. The construction temperature is 60 °F and the operating temperature is 80 °F.

Bending Stress.

$$f_b = (E \times D)/(24 \times R)$$
$$= (2.9 \times 10^7 \text{ psi})(12.75 \text{ in})/(24)(1000 \text{ ft})$$
$$= \pm 15{,}406 \text{ psi}$$

Hoop Stress.

$$\Delta p = 720 \text{ psi} - (90)(0.4333)$$
$$681 \text{ psi}$$

$$f_h = (\Delta p \times D)/(2 \times t)$$
$$= (681 \text{ psi})(12.75 \text{ in})/(2)(0.25 \text{ in})$$
$$= +17{,}366 \text{ psi}$$

Thermal Stress.

$$f_t = (E \times k) \times (T_1 - T_2)$$
$$= (2.9 \times 10^7 \text{ psi})(0.0000065)(60\ °F - 80\ °F)$$
$$= -3{,}770 \text{ psi}$$

Total Longitudinal Compressive Stress.

$$f_{long} = -15{,}406 \text{ psi} + -3{,}770 \text{ psi} + (17{,}366 \text{ psi})(.3)$$
$$= -13{,}966 \text{ psi}$$

Maximum Shear Stress.

$$f_v = (f_{hoop} - f_{long})/2$$
$$= [17{,}366 \text{ psi} - (-13{,}966 \text{ psi})]/2$$
$$= 15{,}666 \text{ psi}$$

The allowable shear stress is 15,750 psi (45% of 35,000 psi) which is greater than 15,666 psi. Therefore, the pipe specification is acceptable. It should be noted that violation of the 45% limit on shear stress does not necessarily mean that the pipeline will fail. Bending loads resulting from HDD installation are not sustained. That is, they may be relieved by plastic deformation. However, operating stresses should be maintained in the elastic range to provide a conservative design.

Conclusion

Unusual loads and load combinations can result from pipeline installation by HDD. These loads should be analyzed and allowed for in the design of pipe properties and drilled paths. Methods for accomplishing this for steel pipelines loaded during installation and operation have been presented in this paper. Governing criteria for installation stresses are based on limits developed for the design of tubular steel members in offshore platforms. Operating stress limits are founded in classical combined stress limiting theory for steel. These criteria will produce a conservative design and serve as an adequate check on pipe designed according to industry codes limiting the maximum allowable operating pressure.

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