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DISCUSSION OF ASTM F 1962
or
"HOW ARE THE PULLING LOAD FORMULAS DERIVED AND HOW ARE THEY USED?"

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ABSTRACT: ASTM F 1962, *Standard Guide for Use of Maxi-Horizontal Directional Drilling for Placement of Polyethylene Pipe or Conduit Under Obstacles, Including River Crossings*, has become an increasingly used procedure for initially verifying the feasibility of the use of HDD for placing HDPE pipe in major installations. This document provides estimates of required pulling tension on the pipe due to the various drag forces imposed during the maxi-HDD operation, and also considers the potential vulnerability to collapse under the imposed external pressures. Whereas the latter loads are evaluated using generally understood principles in pipeline design, the estimated pulling loads are partially based upon cable installation principles from the communications and electric power industries. The present paper provides the basis for the development of the corresponding formulae related to the pulling loads within ASTM F 1962, and also discusses various issues that have arisen in their application.

1. INTRODUCTION

Horizontal directional drilling (HDD) encompasses a wide range of equipment and technology, ranging from small rigs used for distribution and service applications to large machines capable of installing pipelines up to 54-inches, for distances as much as 8,200 ft or greater (HDD Consortium 2008). The latter category is commonly referred to as maxi-horizontal directional drilling (maxi-HDD). ASTM F 1962, *Standard Guide for Use of Maxi-Horizontal Directional Drilling for Placement of Polyethylene Pipe or Conduit Under Obstacles, Including River Crossings*, was originally approved in 1999, following development within the F17.67 Trenchless Technology Subcommittee of the ASTM[1], and reissued in 2005. A detailed description of ASTM F 1962, and its application, was provided during NO-DIG 2006 (Petroff, 2006). The ASTM document provides overall guidelines for a maxi-HDD operation, addressing preliminary site investigation, safety and environmental considerations, regulations and damage prevention, bore path layout and design, implementation, and inspection and site cleanup. One of the significant contributions of ASTM F 1962 is the provision of a rational, analytical method for selecting the polyethylene pipe strength based upon the estimated installation loads on the polyethylene (PE) pipe. Thus, ASTM F 1962 provides a means of determining project feasibility, as well as initial design information. Such results could be further refined by competent engineering expertise, including an analysis of pipe and soil characteristics and interaction, often including the use of relatively sophisticated software tools.

ASTM F 1962 attempts to make reasonable or conservative assumptions in the process of estimating the tensile loads and lateral loads directly associated with the HDD pipe placement process as applied to the

[1] Originally known as the American Society for Testing and Materials.

PE pipe. The loads are compared to the corresponding strength to evaluate the ability of the pipe to withstand the loads, with a reasonable margin of confidence. There are essentially two categories of loads to be considered. For example, the pipe must withstand the laterally or radially imposed forces due to the drilling fluid pressure without collapse. The general magnitude of these forces and their effects are evaluated using generally understood principles in pipeline design, accounting for the time-dependent properties and other behavioral characteristics of PE pipe. In comparison, the prediction of the required pulling load on the pipe is somewhat more problematic, and requires the use or development of an appropriate theoretical model, for which there is not necessarily universal agreement within the industry. Thus, the present paper provides the basis for the development of the corresponding formulae related to the pulling loads and their usage within ASTM F 1962, and also discusses various related issues that have arisen in their application. In particular, the model used for developing the estimated pulling loads is in part based upon cable installation principles from the communications and electric power industries.

2. DESCRIPTION OF ASTM F 1962

Figure 1 illustrates a typical geometry for a maxi-HDD operation, corresponding to a river crossing. The indicated path corresponds to that shown in ASTM F 1962 and comprises four segments, including those spanning the pipe entry to exit point (L_2, L_3, L_4) and the additional length L_1 which allows for handling at both ends and possible other effects (path curvature, thermal contraction, stretching, etc.). The intermediate horizontal segment, L_3 , may be of zero length.

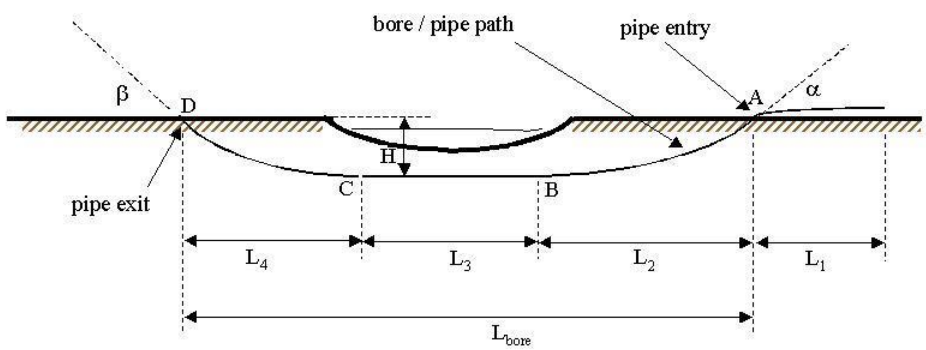

Figure 1 Typical maxi-HDD route (river crossing)
(Source: Outside Plant Consulting Services, Inc.)

Using the above terminology, ASTM F 1962 provides a set of recursive relations to predict the required pull force -- T_A , T_B , T_C , and T_D -- corresponding to the leading end of the pipe reaching point A, B, C and D (Figure 1). Thus,

$$T_A = e^{\nu_a \alpha} \cdot \nu_a \cdot w_a \cdot (L_1 + L_2 + L_3 + L_4) \qquad [1a]$$

$$T_B = e^{\nu_b \alpha} \cdot (T_A + \nu_b \cdot |w_b| \cdot L_2 + w_b \cdot H - \nu_a \cdot w_a \cdot L_2 \cdot e^{\nu_a \alpha}) \qquad [1b]$$

$$T_C = T_B + \nu_b \cdot |w_b| \cdot L_3 - e^{\nu_b \alpha} \cdot (\nu_a \cdot w_a \cdot L_3 \cdot e^{\nu_a \alpha}) \qquad [1c]$$

$$T_D = e^{\nu_b \beta} \cdot (T_C + \nu_b \cdot |w_b| \cdot L_4 - w_b \cdot H - e^{\nu_b \alpha} \cdot [\nu_a \cdot w_a \cdot L_4 \cdot e^{\nu_a \alpha}]) \qquad [1d]$$

where w_a and w_b are the empty (above ground) and buoyant weights of the pipe, respectively, and ν_a and ν_b. are the corresponding "coefficients of friction", as defined below. The pipe entry angle α and exit angle β are expressed in radians, where one radian equals $180^o / \pi$. Equations 1 are sufficiently general to consider the possible implementation of anti-buoyant measures to reduce the otherwise high values of w_b for plastic (i.e., PE) pipe. Typically, the maximum pull force will tend to occur towards the end of the installation; e.g., T_C or T_D . Unfortunately, due to the relative complexity of Equations 1, it is not difficult for mathematical errors to be introduced during their application, or even for typographical errors to arise when transcribing the formulae.

In addition to the tensile loads at the individual stages of progress (points A, B, C, and D), an incremental tension, ΔT, must be added to account for the drag effect of the mud ("fluidic drag"), which is determined from the magnitude of the "hydrokinetic pressure", ΔP:

$$\Delta T = \Delta P \cdot (\pi/8) \cdot (D^2_{hole} - D^2) \qquad [2]$$

where D_{hole} is the diameter of the borehole and D is the diameter of the pipe; ΔP is the incremental drilling fluid pressure in the borehole at the leading end of the pipe during the pullback operation, which is in addition to the hydrostatic pressure corresponding to the head (depth) of relatively dense drilling fluid. The incremental tension, ΔT, is properly added to the local tension T_A , T_B , T_C , or T_D as specified in Equations 1, for each of the four points, but is *not cumulative*; e.g., the value of T_A inserted into Equation 1b is that given by Equation 1a, as written, and *not* $T_A + \Delta T$. This is correctly indicated in ASTM F 1962, but is not always properly implemented by users. In any case, this estimate of ΔT is usually low compared to the tension estimates given by Equations 1, for practical installations, for typically assumed values of ΔP, e.g., 4 - 8 psi (Svetlik, 1995); a value of 10 psi is conservatively recommended in ASTM F 1962.

Finally the effect of the bending stress variation across the pipe cross-section is added to the average stress corresponding to the results of Equations 1 and 2. This effect is typically not major for the flexible PE pipe, for the relatively large radii of curvature generated by the steel drill rods, unless the pipe is of very large diameter. The net resulting peak tensile stress is required to be less than the safe pull tensile stress (SPS) of the PE pipe (Petroff, 2006).

3. THEORETICAL BASIS FOR MODEL

The theoretical model used to develop Equations 1 is based upon conventional Coulomb friction, which assumes that drag forces on the pipe are proportional to the local normal bearing forces applied at the pipe surface, with the proportionality constant designated as the "coefficient of friction". Such bearing forces may be due to the dead (empty) weight of the pipe where above ground, the buoyant weight of the submerged pipe (possibly mitigated by anti-buoyancy measures), bearing/bending forces associated with pulling a stiff pipe around a curve, or bearing forces resulting from (previously induced) axial tension tending to pull the pipe snugly against any locally curved surfaces. In addition, there is a contribution due to the drilling fluid/slurry flowing along the length of the pipe.

Frictional Drag Due to Weight and Buoyancy

Based upon the Coulomb model, the primary source of resistance for installing the pipe is due to the friction between the pipe and the borehole surface and/or the exterior surface upon which the pipe is supported as it enters the borehole. In the absence of anti-buoyancy techniques, the frictional drag developed within the borehole is generally much greater than that developed outside, even assuming a possibly lower coefficient of friction within the wet (lubricated) borehole, due to the very large buoyant weight for the empty PE pipe. The basic methodology for calculating the combined effects of weight and buoyancy, including gravity and the developed drag resistance, has been provided by Svetlik (1995), and, with the exception of the various exponential terms, is the basis of Equations 1. Suggested values for the coefficients of friction are provided in ASTM F 1962, consistent with industry recommended values (Svetlik, 1995). In particular, the suggested 0.3 value for υ_b, applicable to PE pipe within the borehole, is also consistent with more recently obtained values based upon controlled laboratory studies (El-Chazli et al, 2005).

The buoyant weight of the submerged pipe, in combination with the corresponding coefficient of friction, is typically the major factor in determining the required pull force. ASTM F 1962 provides formulae for determining the buoyant weight under various conditions, and is a direct function of the density of the drilling fluid/slurry. For the purpose of design calculations, a conservatively high specific gravity of 1.5 is suggested, although such a high density is not generally recommended during actual operation due to increased required pumping (hydrokinetic) pressure and corresponding elevated risk of hydrofracture (drilling fluid leakage).

Pipe Stiffness at Bends

Bearing/bending forces are significant for pipes with high material stiffness or large diameter when installed in boreholes with limited clearance or relatively sharp bends. However, for flexible PE pipes, installed in boreholes with recommended minimal clearances (e.g., 50% of pipe diameter), along gradually curved paths created by the steel drill rods, such effects may be ignored.

Capstan Effect at Bends

Although pipe stiffness effects may be ignored for PE pipe under the conditions described above, there is nonetheless a potentially important effect due to discrete route bends or gradual path curvature that should not be ignored. Tensions induced in the PE pipe as it negotiates the bore path, become amplified at such curves due to the tensile forces tending to pull the pipe against the curved surface. Such effects are independent of the pipe stiffness, pipe diameter, borehole clearance, radius of curvature or direction of curvature. Thus, bends in opposite directions do not cancel, but are cumulative, with the total cumulative angle traversed being the significant parameter. This phenomenon is referred to as the "capstan effect" since it is the operating principle of the capstan winch, as illustrated in Figure 2.

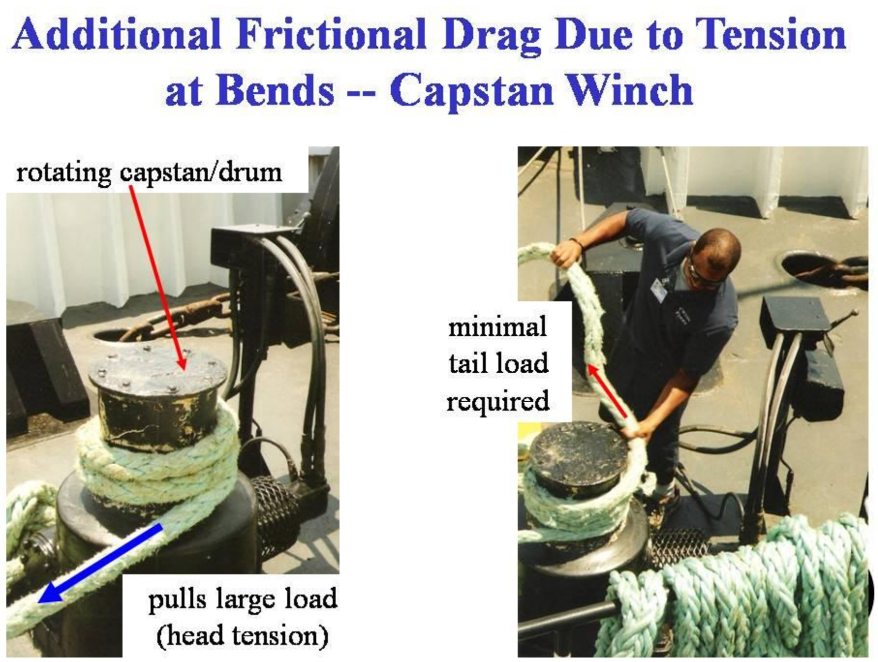

Figure 2 Example of "capstan effect"
(Courtesy of Outside Plant Consulting Services, Inc.)

The following relationship illustrates the basic phenomenon for the idealized case of a weightless, flexible rope, cable or pipe:

$$F_2 = F_1 \cdot e^{\nu \theta} \qquad [3]$$

where F_1 represents axial tension ("tail load") at the entry point of a bend of magnitude θ (radians), ν is the local coefficient of friction between the installed item and wall surface of the cavity, and F_2 is the required axial tension at the exit point of the bend. The exponential term $e^{\nu \theta}$ represents an amplification factor due to the discrete or cumulative bend θ. In practice, the impact of the actual weight of the pipe, or possible stiffness, may be reflected in the preceding tension, F_1. Detailed discussions of the mechanism and associated formulae for various path geometries are provided in utility cable industry literature (Buller, 1949), (Rifenburg, 1953), (Griffioen, 1993), which consider the more complex case of a cable with finite weight as pulled around a horizontal or vertical bend of finite extent.

Due to the compounding nature of such effects, in some situations they can become a major source of drag, possibly controlling practical placement distances. For example, typical fiber-optic telecommunication cables are light (weight) and flexible, but are limited in placement distances by such effects when installed by conventional pulling techniques. (Recently introduced "blown-cable" techniques, discussed below, essentially eliminate this degrading effect and therefore allow extremely large placement distances.)

The capstan effect, as characterized by Equation 3, is the source of the various exponential terms in Equations 1. For convenience, the exponential amplification factor has been assumed to be applied to the pipe at the ends of the curved segments (points B and D). This tends to be a conservative procedure, in order to simplify the analysis, but is reasonable as illustrated in the more precise computer calculations presented below.

For the geometry shown in Figure 1, with relatively shallow entry and exit angles, and no additional deliberate path bends, the effect of the exponential terms is not major. However, for more complex paths, the effect would be more significant. For example, mini-HDD applications tend to include paths with additional deliberate curvature due to the need to avoid known obstacles or follow a curved right-of-way, as well as more subtle curvature due to path corrections characteristic of these typically less precisely controlled installations. The latter non-deliberate curvature is a function of the soil conditions, operator skill, and equipment and is therefore difficult to accurately quantify. Nonetheless, such effects can have a major impact and are reflected in some mini-HDD design methodologies (Slavin, 2007), as discussed below.

Hydrokinetic Surface Drag (Fluidic Drag)
The effect of the shear forces directly imparted on the pipe by the drilling fluid ("fluidic drag") has been handled in a widely disparate manner within the industry. Some design procedures assign an effective shear stress to be applied to the outer surface of the pipe, on the order of 0.025 lbs/in^2 or greater (Puckett, 2003), (ASCE 108). Depending upon the pipe diameter, loads of such magnitude often tend to be sufficiently large as to preclude the need to consider the frictional drag associated with the other effects generally accepted to be of importance (buoyancy, route bends, etc.). Other sources (Svetlik, 1995) tend to de-emphasize these shear forces, indicating such effects are usually considered to be negligible, consistent with the above discussion.

Similar to consideration of the capstan effect, the development of Equation 2, for use within ASTM F 1962, is also based upon a technology used to place cables into ducts. The blown-cable method (Griffioen, 1993) refers to a technique for using high-speed air, to effectively install light-weight (fiber-optic) cables within ducts. In this procedure, high pressure air (e.g., 100 psig) is applied at one end of the duct (cable feed end) with the far end (cable exit) of the duct open to the atmosphere (0 psig). This results in a rapid flow of air in the annular space between the cable and duct surfaces to the open end, which applies useful shear (drag) force to the outer surface of the cable, acting in the direction of installation. This force, in combination with a cable pusher, is able to successfully place the cable into the duct. In the analysis of this technique, the magnitude of the total shear force applied to the cable surface within the duct is estimated as:

$$\Delta T = \Delta P \cdot (\pi/4) \cdot (D^2_{duct} - D^2_{cable}) \cdot D_{cable} / (D_{duct} + D_{cable}) \qquad [4]$$

where D_{duct} and D_{cable} represent the inner diameter of the duct and outer diameter of the cable, respectively. In this case, ΔP is equal to the pressure difference at the opposite ends of the cable, as it traverses the duct path. Since this pressure differential will vary from 0 psi as it enters the duct to the full applied pressure (e.g., 100 psi) when exiting the duct, the corresponding shear force ΔT will vary accordingly, with the maximum occurring at the exit.

Equation 4 is based upon the assumption that the rapidly moving annular air mass is approximately in equilibrium, with the difference in air pressures at the ends balanced by the retarding shear forces acting upon the air mass at the duct and cable surfaces. These shear forces are also assumed to be uniformly distributed across the duct and cable surfaces, such that the fraction applied to the cable is equal to its fraction of the combined surface areas; i.e., $D_{cable} / (D_{duct} + D_{cable})$. This model ignores the detailed microscopic viscosity and shear stress behavior at the cable surface due to the moving fluid (air), and focuses on the gross macroscopic behavior corresponding to the pressure differential across the length of cable in the duct. This approach is an obvious oversimplification of a complex process but has the advantage of allowing a useful estimate of the effective shear force acting on the cable based on a known or assumed pressure differential.

In the case of the cable in the duct, the direction of air (fluid) flow is towards the leading end of the cable, such that the shear forces tend to help drag the cable into the duct. For the case of the pipe in the borehole, the direction of drilling fluid flow is assumed to be in the opposite direction, away from the leading end of the pipe, such that the shear forces tend to inhibit the placement of the pipe into the borehole, resulting in an incremental tension load ΔT. Using the above model, this additional force may be estimated as:

$$\Delta T = \Delta P \cdot (\pi/4) \cdot (D^2_{hole} - D^2) \cdot D / (D_{hole} + D) \qquad [5]$$

where ΔP now corresponds to the hydrokinetic pressure at the leading end of the pipe in the borehole. The fraction $D / (D_{hole} + D)$ in Equation 5 is less than ½; e.g., for a borehole 50% greater than the pipe diameter, this quantity would be equal to 0.4. For simplicity, the value of ½ is conservatively assumed which directly results in Equation 2. The effect of the drilling fluid properties would be manifested in the magnitude of the required ΔP term to maintain fluid flow, which may vary (increase) as the installation progresses, corresponding to a varying value for the fluidic drag term, ΔT. For practical design purposes, a single reasonable or conservative value for the hydrokinetic pressure term is assumed, which, as discussed above, is considered to be on the order of 10 psi or less, assuming proper drilling fluid control and usage in a well-controlled maxi-HDD operation, and corresponds to only minimal drag on the pipe.

Industry Investigation of Fluidic Drag

Several investigations in the industry have attempted to evaluate the magnitude of the fluidic drag. In some cases the procedure is based upon monitoring the pulling force applied to the pipe, possibly deduced from the loads applied at the drill rig, and essentially subtracting the contributions due to all other effects. In particular, Adedamola et al (2002) evaluated the effect based upon field experiments using PE pipe, and Puckett (2003) evaluated the effect based upon actual installations with steel pipe. This procedure suffers from the risk of inadvertently omitting some phenomena, or over- or underestimating these other effects, possibly due to assumed values of the coefficients of friction, ignoring the insidious capstan effect, pipe stiffness, etc. or other assumptions inherent in the method applied. As a result, the determined values tend to be unreliable, and may be inordinately high if they reflect the impact of other effects that may not have been explicitly considered in the employed model.

A very interesting discussion of the nature of the fluidic drag, and its potential magnitude, is provided in a recent paper by Duyvestyn (2009). This paper also presents results of an investigation that attempts to more accurately characterize the fluidic drag, and compares the resulting pull loads to “observed” results in three actual installations, as well as that predicted by ASTM F 1962 and the method adopted for the installation of steel pipe by the American Gas Association (Hair et al, 1995). Due to the technique used to select the parameters (e.g., coefficient of friction) to best match the observed loads as deduced by measurements at the drill rig, all three procedures (ASTM F 1962, Hair et al, and that recommended by Duyvestyn) provide a reasonable match with the peak observed load in two of the installations, with the AGA method significantly over-predicting the peak pull load in the third installation. The method recommended by Duyvestyn provides the best match in all cases.

The derivation of Equation 5 (and the resulting Equation 2) is an admitted oversimplification of a complicated fluid flow process, including the assumption that the drilling fluid always flows towards the rear of the pipe -- which assumption is shown by Duyvestyn to not be correct beyond the “crossover” point, beyond which the drilling fluid flows in the opposite direction, towards the drill rig and pipe exit point. Nonetheless, the parametric dependence of the incremental tension on the borehole and pipe diameters is remarkably similar to that provided by Duyvestyn’s Equation 5, as obtained from his referenced Wellplan drilling analysis tool. These relationships only differ by a sign (plus vs. minus) in the denominator of the respective equations. It is not presently clear why there should be such a fundamental difference in the dependency upon these basic parameters. In any case, the peak borehole pressure, as well as the peak fluidic drag component, is apparently at a peak at the crossover point (Duyvestyn, 2009), suggesting that the estimate given by Equation 5 (or Equation 2) is conservative, as intended for design purposes.

Hydrokinetic Pressure Resistance

In addition to the fluidic drag imposed along the longitudinal surface of the pipe, the presence of the hydrokinetic pressure ΔP acting at the leading end of the pipe, across its exposed cross-section, tends to resist the movement of the pipe into the borehole, similar to that experienced in the blown-cable technique (Griffioen, 1993). This effect is usually not considered in HDD analyses, including ASTM F 1962, but may be considered to be analogous to the resistance experienced when pushing a rod or cable into a high pressure chamber. The magnitude may be estimated by the pressure increment ΔP applied to the exposed cross-sectional area of the pipe. However, the combined magnitude of both of these effects is generally only a small fraction of the safe pull stress of HDPE pipe, for any size or available wall thickness, and well within the degree of uncertainty in present methods for estimating the pull load.

The magnitude of the hydrokinetic pressure has been assumed to be that necessary to maintain proper drilling fluid flow. However, a non-uniform pulling rate for the pipe may cause additional incremental pressure. Such surge effects may be minimized by pulling at essentially constant speed, and any possible incremental loads on the pipe, either radial or longitudinal, would be withstood by the greater short term strength of the HDPE product (Svetlik, 1995).

4. PREDICTION vs. DESIGN

In spite of attempts by the HDD industry to develop more accurate predictions of pulling loads in practical applications, it is generally acknowledged that present methods often produce installation forces that differ significantly from field observations. Attempts to align the predictions with field measurements, by selecting appropriate quantitative values of relevant parameters, are ultimately only useful if able to more accurately predict -- apriori – the required pulling loads in subsequent applications, using the developed model. This condition has not yet been demonstrated, essentially because of the complexity and wide variability associated with operations in non-engineered materials and environments -- i.e., soil and rock. For example, controlled HDD field experiments have resulted in significantly different pull loads in what would appear to be almost identical conditions (Knight et al, 2002). In a sequence of two installations through the same borehole, pulling the same pipe, the measured pull force, using a gauge mounted at the leading end of the pipe, varied by a factor of almost two to one. For the second installation, the pipe experienced a pull load of approximately ½ that of the first pipe installation, with the peak occurring at a different point along the path. Obviously, the conditions were therefore not identical, possibly due to the second pipe pullback operation enjoying a slightly straighter borehole path as a result of the second pass of the reamer, a more stabilized borehole, and/or non-identical drilling fluid/slurry conditions with lower friction and drag effects. This experience raises the question as to what does it mean to make an accurate prediction, when there is such a wide variation in results in what would appear to be essentially the same installation ?

Thus, although there may -- or may not -- be readily available explanations for significantly different pull loads in supposedly identical situations, it appears that the present state of understanding of the overall process is insufficient to allow complete characterization and accurate prediction of the pulling load experienced in any given HDD installation. Nonetheless, rather than be overly concerned with the ability

– or inability -- to make accurate predictions of the pulling force, it should be recognized that engineering design procedures are available that provide confidence in the ability to successfully install PE pipe using HDD. For example, since all the factors affecting the installation are not readily quantified in advance of an installation, it is possible that probabilistic analyses would be the most appropriate approach, based upon what would appear to be "random" variations of significant parameters (friction, fluidic drag, unplanned variations in path curvature, etc.). In this case, the predicted pulling load would be expressed in statistical parameters, such as expected (average) pulling load and standard deviation, for which it would still be necessary to statistically characterize the "random" variability of the assumed individual significant parameters. The resulting probabilistic distribution of the pulling load would then be used to select compatible equipment and pipe strength, based upon the high end of the distribution (e.g., 95th percentile load). Thus, the focus would be on a means of arriving at a proper (conservative) design strength, rather than attempting to accurate predict a single value of an inherently widely varying, essentially random quantity.

However, at present, an alternate practical approach is typically employed in the industry. An admittedly imperfect model, with predictions based upon imperfectly understood variables, is used with conservative values of these variables to determine a conservative estimate of the pull load, which is hopefully not overly conservative. This is essentially the procedure adopted in ASTM F 1962.

ASTM F 1962

In application of Equations 1 and 2, ASTM F 1962 suggests reasonable or conservative values for the various coefficients of friction, drilling fluid density (affecting buoyant weight) and hydrokinetic pressure. Furthermore, as indicated above, the exponential amplification factor for the capstan effect has been assumed to be applied to the pipe at the ends of the curved segments in Figure 1, generally overestimating the impact of this effect, assuming no additional route bends or significant curvature are present. ASTM F 1962 then compares the corresponding peak axial stress, including the contribution of the local peak bending stresses, to the safe pull stress (SPS) for the PE pipe. This strength level is conservatively selected to avoid permanent deformation under assumed temperature and load duration. Due to the combination of these effects, no additional safety factor has been introduced for the typically well-controlled maxi-HDD operation. (See Equation 21 of ASTM F 1962.)

Thus, ASTM F 1962 is intended to provide a design procedure rather than a prediction method. In any given case, it is likely that the pulling loads based upon Equations 1 and 2, with the suggested quantitative values for the various parameters, will somewhat overestimate the maximum observed load. For example Petroff (2006) discusses the calculated pullback force for two separate applications, including the installation of a 28-inch pipe and a 30-inch pipe. The total pullback force experienced at the drill rig – including the significant additional force necessary to pull the drill rods and reamer -- was only slightly higher than the calculated force per ASTM F 1962, using the suggested conservative assumptions. (Similar to other methods, ASTM F 1962 only addresses the required pull force applied to the pipe itself, and is not intended to reflect the load contributions attributed to the drill rods and reaming operation.) Results are also provided to illustrate the sensitivity of the estimated pull loads on the assumed values of the parameters, including coefficient of friction and drilling fluid density.

Although ASTM F 1962 suggests quantitative values to be used for the various parameters in the equations, it would generally be preferable that the selected values for the important parameters (e.g., drilling fluid density and coefficient of friction) reflect the specific conditions of each installation, possibly based upon related preliminary tests and careful control and monitoring of conditions during the actual installation. Depending upon the degree of confidence inherent in this procedure, the engineer may decide to apply a load factor (> 1.0) to the estimated pull force to provide greater assurance of success.

5. ASTM F 1962 RESULTS and MACHINE-GENERATED CALCULATIONS

Equations 1 represent a simplification with respect to the application of the capstan effect, for which the exponential amplification of Equation 3 is conveniently applied at the ends of the curved segments, rather than distributed continuously along the segment. It is therefore of interest to perform more precise sample calculations, using a computer generated incremental model that correctly distributes the frictional drag, buoyancy, and capstan effects along the length, as applied to a practical example.

Figure 3 illustrates the results for the calculated pulling tension, using ASTM F 1962, for a 2,500 foot installation of 24-inch HDPE pipe, of DR 11 wall thickness, at a depth of 35 feet. The excess pipe length L_1 is 100 feet and the pipe entry and exit angles are 10° and 15°, respectively. In this example, the coefficient of friction on the surface, ν_a , has been assumed to be 0.1, assuming roller supports for the pipe outside the borehole, and that in the borehole, ν_b , to be 0.3 -- as suggested in ASTM F 1962. The results include the nominal case, without the employment of anti-buoyancy measures, as well as that corresponding to the commonly applied procedure in which the pipe is filled with water to reduce buoyancy. The safe pulling tension (based upon the safe pull stress applied to the cross-section, ignoring the relatively low bending stresses) is also shown. In the nominal case, based upon the assumed parameters, the peak pulling load is predicted to occur at the end of the bore path, at point D, at which it apparently exceeds the safe pulling tension. However, if anti-buoyancy techniques are used, such as filling the pipe with water, the frictional drag forces are dramatically reduced and the peak pulling load, which in this case also occurs at the end of the installation, is determined to now be well below the safe pulling tension. In general, depending upon the relative values of the various parameters in Equations 1, the peak tension may occur prior to the end of the installation (point D).

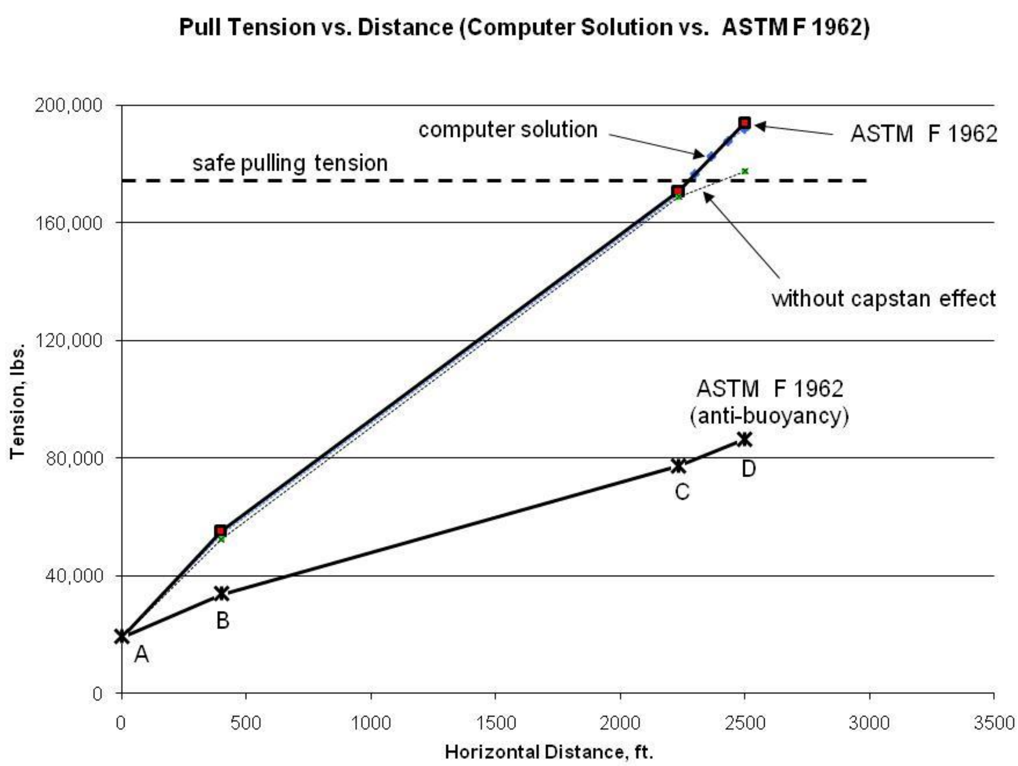

Figure 3 Results of ASTM F 1962 and computer generated incremental model

Figure 3 also includes the results of the computer generated solution for the nominal case, in the absence of anti-buoyancy techniques. It is apparent that the results using ASTM F 1962 are extremely close to the computer solution at all four points A, B C and D.

As an additional matter of interest, the results for the pulling load determined without consideration of the capstan effect are also included. In this case, for which the only bends considered are essentially those due to the entry and exit angles, the error introduced by ignoring the capstan effect is relatively small; i.e., less than 10%. For other cases, such as mini-HDD installations, for which there may be additional deliberate planned route bends, as well as cumulative, unplanned path curvature resulting from path deviations and corrections, ignoring the capstan effect may lead to major discrepancies.

6. EXTRAPOLATION of ASTM F 1962 to MINI-HDD APPLICATIONS

In comparison to the original intended application of ASTM F 1962 to maxi-HDD operations, the basic model has been extended to application to mini-HDD installations (Slavin, 2007). This methodology is based upon various assumptions and a gross simplification of Equations 1, including ignoring the term L_1, and ignoring the contribution ΔT of Equation 2. However, for the mini-HDD case, it is necessary to consider and apply the capstan effect to the unplanned, but typical, path curvatures associated with path deviations and corrections. These effects are of wide variability and of much greater significance than that in a well-controlled maxi-HDD operation, employing larger diameter, stiffer drill rods. Existing data verify the validity of this overall procedure, also indirectly verifying the validity of the basic ASTM F 1962 model and procedure. Furthermore, for the mini-HDD applications, a load (safety) factor >1.0 is recommended to be applied to the calculated pull load (at point D, assumed to be equal, or close, to the maximum pull force load experienced by the pipe). The load factor is intended to account for the various simplifications, as well as the less carefully controlled nature of a mini-HDD installation relative to a sophisticated maxi-HDD operation, resulting in potentially greater path deviations and corrections than estimated in the mini-HDD model (Slavin, 2007).

7. SUMMARY

ASTM F 1962, *Standard Guide for Use of Maxi-Horizontal Directional Drilling for Placement of Polyethylene Pipe or Conduit Under Obstacles, Including River Crossings*, provides a rational procedure for initially verifying the feasibility of the use of HDD for placing HDPE pipe in major installations. The various formulae for calculating required pull loads are derived from previous formulations in the directional drilling industry, supplemented with principles familiar to the electric power and telephone industry for placing cables within ducts. The resulting methodology accounts for the weight of the PE pipe, including buoyancy, and the capstan effect due to the degrading effect of tension at route bends, as well as surface drag effects associated with the flow of drilling fluid. The effect of pipe stiffness has been appropriately ignored for PE pipe. For convenience, the corresponding equations for calculating the estimated pull load on the pipe have been somewhat simplified, but have been shown to closely match more precise computer solutions based upon the same physical principles.

6. REFERENCES (Alphabetical Order)

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