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PRACTICAL PULL-IN LENGTHS FOR OFFSHORE THERMOPLASTIC PIPE LINER SYSTEMS

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ABSTRACT

Pulling corrosion-resistant, thermoplastic pipe into a corroded steel pipeline is proving to be more technically viable and cost effective than conventional methods for refurbishing aging pipelines. This is especially the case in difficult to access areas, such as onshore wetlands and shallow offshore water.

This study made use of basic physics equations governing the frictional forces associated with pulling a thermoplastic pipe into a steel host pipe. The equations include the effects of friction coefficient, bend angles, bend radii, pipe weight and stiffness, and pull rope weight and strength. Pipe and rope density was a key parameter because including the effect of flooding the pipes with water was a key aspect of this work. Software based on the basic physics equations was used to simulate a variety of pull-in scenarios that are compared in this paper. The pipeline geometry for this work was from a recently completed case study which evaluated pulling HDPE pipe (both bare and high strength fiber reinforced) into an out-of-service pipeline for new use as a treated, produced-water, outfall line. The pipeline runs from the shoreline, over a rocky outcrop and then continues 7.7 kilometers to a final proposed offshore, disposal location.

This study demonstrates that pulling a flooded liner into a flooded host pipe can dramatically increase the length of pipe able to be pulled compared to a dry host pipe. The relative lack of bends in offshore pipelines is also a key factor in achieving long, continuous pulls.

INTRODUCTION

This study was initiated when a pipeline operator wanted to evaluate the feasibility of reusing a thirty-plus year-old, out-ofservice, 16-inch pipeline as a water outfall pipeline for a new onshore water treatment plant. The target disposal location of treated produced water was 7.7 km offshore. The new water outfall pipeline would transport a treated mixture of produced water and surface rain water, a mixture which is an especially corrosive liquid due to the dissolved oxygen in the stream. A thermoplastic liner, considered non-corrosive, has obvious appeal because it often offers a cost-effective solution, compared to a new steel pipeline. The basic approach was to reuse the existing 16-inch, out-of-service, de-ballast pipeline as a conduit for a High Density Polyethylene (HDPE) liner.

The maximum length of plastic lined pipe reported in the offshore industry literature appears to be just over 5 km, with the pull being divided into segments no longer than 1.5 km [1,2]. The objective of the feasibility study leading to this paper was to provide calculations supporting the feasibility of pulling an HDPE liner the entire 7.7 km pipeline length, in a single pull operation. The pipeline runs from the shoreline, over a rocky outcrop and continues 7.7 km to the proposed offshore disposal location. While the pipeline is essentially straight in the sense that there are no intentional bends, the pipeline does follow the sea bottom contour, and therefore has many large radius, vertical-plane bends. A key focus of this paper is the significance of these bends to the required pull-in forces.

HDPE is a relatively low tensile strength material. This low strength limits the length of pipe that can be pulled into the host pipe in a single pull operation. The study therefore also looked at high strength, fiber reinforced HDPE pipe to understand how additional, high strength, helical and axial fiber can improve both the cost efficiency of the pull-in operation, and the final pipeline’s pressure rating and flow rate.

In the subject study, both loose fit and tight fit designs were considered. For the unreinforced, bare HDPE cases, both “tight fit” and “loose fit” liners were considered. The bare HDPE, tight fit liner relies entirely on the existing host pipe to carry the pressure-induced hoop stress, and as a result, generally has a large diameter-to-thickness ratio (DR). The loose-fit liner (bare HDPE) must carry the pressure hoop stress, and therefore has a DR consistent with the applied pressure. The high strength fiber-reinforced HDPE pipe is designed to carry the pressure load and can be either loose fit or tight fit, depending on flow rate and other requirements.

This study made use of basic physics equations governing the frictional forces associated with pulling a rope and thermoplastic pipe into a steel host pipe. The software that implemented these equations included the effects of friction coefficient, bend angles, bend radii, pipe weight, pipe stiffness, and pull rope weight. The host pipe was treated as a rigid conduit. Studying the effect of flooding the pipes with water was a primary objective of this work. The equations used in the software are provided in this paper as well as a simple, easy to use, pull force formula that does not include the effect of pipe bending stiffness on friction at host pipe bends.

This work shows the feasibility of a 7.7 km pull-in of a high strength fiber reinforced pipe. Making this pull feasible was the combination of four key factors:

1. The use of low weight, high strength rope and liner kept the friction forces low, while offering the capability of high pull tensions.
2. The low weight benefit was amplified signifi cantly by flooding the host pipe and liner with sea water. The liner has nearly the same density as the water (near neutral buoyancy). The use of Ultra High Molecular Weight Polyethylene (UHMWPE) rope, rather than steel rope was also critical in this regard.
3. The relative lack of bends in this pipeline, and in offshore pipelines in general, was also a key factor in achieving the 7.7 km pull.
4. The use of a liner that did not require a diameter reduction box (as typically employed for HDPE “tight fit” liners), together with the associated additional tension at the insertion point, led to significantly reduced pull forces, and thus, potentially longer pull lengths.

FRICTION MECHANICS EQUATIONS

For a rope or pipe sliding through a perfectly straight host pipe, the required increase in tension to overcome friction is

(1) $$\Delta T = T_o - T_i = L[|\omega|\mu\cos\theta + \omega\sin\theta]$$

where T_o is the required pull force at the host outlet, T_i is the tension in the pipe or rope where it enters the host pipe, L is the length of the host pipe, μ is the coefficient of friction, ω is the weight per unit length of the rope or pipe, and θ is the slope of the host pipe due to elevation change.

Consider now, a perfectly flexible rope at a host pipe bend. If one ignores weight induced friction and elevation change within the bend, very simple formulas can give the "amplification" of the tension in the rope as it passes through the bend:

(2) $$T_o = T_i e^{\mu\beta};\ \Delta T = T_i (e^{\mu t} - 1)$$

where T_o and T_i now refer to the tension at the inlet and outlet of the bend, and β is the bend angle in radians. Note that bend radius does not enter this formula. Eq. (2) is applicable to a rope or pipe that contacts the inside of the host pipe bend. For a pipe with bending stiffness, sufficient tension must exist in the pipe to force the pipe to conform to the bend, without contacting the outer bend surface. If the tension is not sufficient, the resulting contact at the outer bend surface will lead to normal contact forces that generally increase ΔT at the bend compared to (2).

Equations (1) and (2) can be used sequentially along the length of the host pipe to analyze any number of straight segments connected by bends. The calculation process is started at the host pipe inlet where T_i is known.

Eqs. (1) and (2) can be combined to provide a formula that is again limited to perfectly flexible rope:

(3) $$\frac{\Delta T}{\Delta T\ ref} = \left[\frac{T_i}{\Delta T\ ref}\ \mu\beta + 1 \right] \left[\frac{e^{\mu\beta} - 1}{\mu\beta} \right]$$

where $\Delta T\ ref$ is defined using Eq. (1) without concern for any bends. In this formula, β is the bend angle corresponding to the length L used to calculate $\Delta T\ ref$. Formula (3) is not as useful as (1) and (2) for performing piecewise calculations of pull tension along the host pipe. However, it is useful, as an approximate formula, to quickly estimate the tension in a complete pipeline. For this use, the value of β is simply the sum of all bend angles in the pipeline where all angles are taken to be positive.

The longest completed pull-in known to the authors employed an electric motorized pig to pull a cable 16.2 km through an offshore 24-inch pipe containing crude oil [3]. Using the pull device's maximum designed pull force, weight, and friction information, and including a suggested friction coefficient between 0.20 and 0.25, equations (1) and (3), indicate that the

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total bend angle (β) for that pipeline would be in the range of 230° to 400°. This range is about 14° to 25° of bending per kilometer. Longer pulls are believed to be possible, depending on the pipeline details and the motorized pig design [3].

This work employed a semi-empirical approach to include the pulled pipe's bending stiffness effect when negotiating host pipe bends. Formulas were developed for bend contact normal forces which essentially treat the bending as three-point bending. To calculate the bend normal forces at both the inside bend surface and the outside bend surface, the diametral clearance between the pulled pipe and the host pipe is combined with the pipe bending stiffness, the host pipe bend radius, and axial tension. The friction force at the bend is then simply the sum of μ, times these three normal forces. The derivation results in an empirical coefficient that is set based on pull force data, or some other type of calibration such as finite element simulation. The derivation does not address ovality effects other than through the empirical coefficient. For each bend, the following equations are used to test for outer bend surface contact ($N_o > 0$) before using (2). If outer bend contact occurs, then the friction force at the bend is given by:

(4) if $N_o > 0$ then $\Delta T = \mu(N_i + 2N_o)$

where N_i is the bend inner surface normal force magnitude and N_o is the magnitude of the two bend outer surface normal forces. The value of N_i is given by:

(5) $$N_i = \Upsilon \frac{8s}{R^2\beta}$$

where s is the pulled pipe's bending stiffness (typically $s = EI$), Υ is the empirical calibration factor and R is the pulled pipe's bend radius calculated by:

(6) $$R = R_H + \frac{(D_H - D)}{2}\left[1 + \left(\frac{4}{\beta}\right)\left(\frac{\sin(\beta/2)}{1 - (\cos\beta)}\right)\right]$$

where R_H is the host bend radius and $D_H - D$ is the diametric clearance between the pulled pipe and host pipe. The value of N_o is then calculated by:

(7) $$N_o = \frac{N_i \cos(\beta/2) - T \sin\beta}{1 + \cos\beta}$$

where T is the tension at the start of the bend. If $N_o \le 0$, then no outer bend contact occurs, and the bend friction force is calculated using (2).

The empirical factor Υ was set at 1.5 in the current calculations based on finite element simulations of pulling through a single host bend.

COMPARISON OF KEY MATERIAL PROPERTIES

Pulling a liner into a pipeline typically involves two components in the pull string: the pull rope and the pulled pipe. Consideration must be given to the strength of both components. With rope or bare HDPE pipe, a theoretical upper limit on the pull length for a perfectly straight host pipe can be determined from the maximum allowable axial stress of the material, its weight density, and an assumed friction coefficient:

$$(8) \qquad L_{max} = \frac{\sigma_{max}}{\rho\mu}$$

Table 1 shows densities and strengths of the rope materials and plastic pipe in this study. Eq. (8) is not applicable to the fiber reinforced pipes; therefore, these pipes are not included in the table. Using $\mu = 0.3$ for steel rope, UHMWPE rope, and for HDPE pipe, gives L_{max} values of 53, 390, and 8 km, respectively. Assuming the materials are submerged in sea water, the UHMWPE and HDPE benefit significantly compared to steel, and the L_{max} values increase to 61, 6900, and 100 km. The UHMWPE and HDPE can have even larger values of L_{max} if their inherently lower friction coefficients are used in place of 0.3 (perhaps 0.2 for HDPE [4] and 0.1 for UHMWPE [5]). Please note that the point of this calculation does not suggest that such long pulls are practical. Rather, it is to show: (i) the potentially dramatic effect on practical pull distances offered by the much lower densities of the HDPE and UHMWPE compared to steel, and (ii) how this effect is further amplified when these materials are submerged in water. Also, note that the UHMWPE and HDPE are slightly less dense than seawater and that L_{max}, as defined here, would approach infinity as the density of the materials approaches the density of the water (neutrally buoyant).

	specific gravity	strength (MPa)	L_{max} (dry) (km)	L_{max} (submerged) (km)
steel wire rope	7.839	1220	53	61
UHMWPE rope	0.970	1120	390	6900
HDPE pipe	0.945	24	8.0	100

Table 1. Density and Strength of the Rope and Pipe Materials and Theoretical “L_{max}“ Values

PIPELINE GEOMETRY

The pipeline geometry used in the calculations for this study was digitized from a sea bottom contour plot. The total pipeline length is 7.7 km. The 1169 digitized points are joined by 1168 pipe segments. The average segment length is about 6.6 m, but the segment length was varied as needed to represent the bottom contour. The software automatically calculated the bend angle between each segment. The bend angle behavior is summarized in Table

2. It can be seen that only 10% of the bends involve greater than 1° change in direction. The largest bend angle is ~12°. Only 5 bends have angles of at least 5°. Summing all the bend angles gives a "total" bend angle of 578°. This is a fairly large bend angle, considering that the pipeline is essentially "straight". For the calculations in this paper, it was assumed that the pipeline is perfectly straight when viewed from above. No attempt was made to account for bridging that likely occurred due to the inherent stiffness of the steel pipe. As the digitizing was extracted from a fairly low resolution plot of the bottom contour, the digitization could have introduced more bends than in the actual pipeline. The total bend angle, from summing just the bends with angle of at least 1°, is 39% of the 578° (223°). Figure 1 shows the pipeline elevation, versus its distance from shore. The vastly different vertical and horizontal scales significantly exaggerate the bends in this plot. The large change in slope at about 1100 m is due to crossing the rocky outcrop.

The radii of the pipe bends were input to the calculation for each of the model's pipe joints. For this study, all joints were given the same bend radius of 50 times the host pipe's nominal diameter (50 D). Sensitivity to assumed bend radii was determined by additionally considering 25 D and 100 D bends for the fiber reinforced liner. This liner design was used for the sensitivity study because it has the largest bending stiffness and thus would show the greatest sensitivity to the assumed bend radii.

	Number	Fraction
$0° <= \beta < 0.1°$	178	15.3%
$0.1° <= \beta < 0.5°$	597	51.2%
$0.5° <= \beta < 1.0°$	271	23.2%
$1° <= \beta < 2°$	97	8.3%
$2° <= \beta < 5°$	19	1.6%
$5° <= \beta < 12°$	5	0.4%
totals	1167	100%
total for angles >1°	121	10%

Table 2. Summary of Pipeline Bend Behavior

SIMULATION CASES

Simulations were run for four liner designs. Table 3 summarizes the key features of the four designs.

The first case simulated a tight fit HDPE liner. A tight fit liner is given a temporary diameter reduction using a diameter reduction box as the liner enters the host pipe. The diameter reduction is retained during pull-in by maintaining a minimum specified pull tension. Upon completion of the pull-in, the liner expands due to viscoelastic recovery and to the addition of internal pressure. In this study, it was assumed that the liner diameter during pull-in was 95% of the host ID. The liner diameter-to-thickness ratio (DR) was 40. The minimum pull tension to maintain the diameter reduction was assumed to be 34.6 kN. This force limit was introduced in the simulations by specifying the tension in the liner at the host pipe inlet as 34.6 kN. This axial force was ~14% of the 242 kN assumed axial strength of the HDPE liner. Note that the lowest tension in the line is always at the host inlet in this study and this will generally be the case unless there are steep slopes at which the second term of (1) is negative and larger than the first term.

Case Description	HDPE Pipe DR	Weight (kg/m)	Specific Gravity (g/cc)	Pressure Rating (bar)	Axial Strength (kN)
Tight Fit Bare HDPE	40	9.5	0.945	NA	242
Loose Fit Bare HDPE	21	14.2	0.945	6.9	360
Round Reinforced HDPE	33.5	10.5	0.929	13.8	1273
Folded Reinforced HDPE	32	15.7	0.922	13.8	1433

Table 3. Pull-In Case Descriptions with Pipe Weight, Density, and Axial Strength

The second case simulated a loose fit HDPE liner. For this liner type, the liner had to carry the operating internal pressure load of 6.9 bar, which required a DR = 21. The OD of the loose liner was assumed to be 85% of the host pipe ID. Unlike the tight fit liner case, no minimum tension was required to maintain a reduced diameter during pull-in. The tension in the pipe at the inlet to the host pipe was therefore input as zero. This liner had an assumed axial strength of 360 kN, which is larger than for the tight liner case due to its heavier wall.

The third case considered a fiber-reinforced HDPE pipe used as a loose fit liner. The HDPE pipe was contra-helically wrapped with fabric made from high strength fiber. In this type of design, the high strength fiber carries the internal pressure load. The reinforced pipe for this study was designed to carry twice the internal operating pressure of the bare HDPE loose fit liner (even higher pressure ratings could have been designed). This increased operating pressure allowed the target flow rate to be maintained, relative to the original steel pipe. Because the HDPE pipe in this case does not carry the pressure load, its DR ratio was 33.5. To provide increased pull force capability, axial tapes made from high strength fiber were also part of the pipe construction. The strength of the combined axial tapes was 1270 kN (the core pipe strength was not included). The liner's OD was about 80 % of the steel host pipe's ID. Due to the added axial fiber, the bending stiffness was 1.7 times that of the bare DR 33.5 core pipe.

The fourth case also considered a high strength, fiber reinforced HDPE liner and again doubled the bare loose fit liner's operating pressure. In this case the pipe was assumed to be folded into a "C" shaped cross-section before pull-in so it was able to have a larger, tight fit, diameter, while still having a significantly reduced effective cross-section size during pull-in. The maximum folded dimension was assumed to be 78% of the steel pipe's ID. This larger diameter pipe had more axial fiber than the previous smaller reinforced design, and had an axial fiber strength of 1430 kN. The bending stiffness of the folded pipe was 1.5 times that of its round bare DR 32 HDPE core pipe. The folded pipe's bending stiffness was ~9% smaller than the bending stiffness of the smaller loose fit reinforced round pipe, even though the flow diameter was ~25% larger and the axial pull-in strength ~13% larger.

The simulations used three different steel rope sizes and one UHMWPE rope size. The rope strengths and weights are summarized in Table 4. The submerged weights of the rope in sea water are also included.

Rope Description	Strength (kN)	Weight (kg/m)	Specific Gravity (g/c)	Submerged Weight (kg/m)
Steel, 3/4 - inch	255	1.6	7.84	1.3
Steel, 1-inch	449	2.8	7.84	2.4
Steel, 2-inch	1718	11.0	7.84	9.6
UHMWPE, 2-inch	1580	1.4	0.98	-0.063

Table 4. Pull Rope Descriptions

PULL-IN SIMULATION RESULTS

Pull-in simulations were run for the four basic pipe designs of Table 3. The simulations were repeated for a range of friction factors to illustrate the degree of sensitivity to the assumed friction behavior. Simulations were done for "dry" hovst pipes and "submerged" host pipes. For "submerged" cases, the liner and the host pipe were assumed to be filled with sea water. Most cases were run with the assumed bend radius of 50 D, but a few cases were run with 25 D and 100 D bend radii. It is generally believed that the highest coefficient of friction (CF) will occur between the steel rope and the steel pipe, followed by HDPE against steel pipe, and then UHMWPE against steel pipe. In the simulation cases, the assumed friction is varied to show the effect of this key parameter, but all cases assumed the same relative CF behavior (as shown in the tables).

Tables 5 through 8 summarize the pull simulation results for the four pipe designs of Figure 3. The first 5 columns define the key inputs to the simulations and the last three columns summarize the key results. All simulations were run for the full 7.7 km pipeline. However, if either the strength of the pipe or rope was exceeded in the calculation, it was possible to determine the distance that could be achieved in a single pull. This "Max Pull Length" is tabulated for each case. If the tabulated value is less than 7.7 km, it was not possible to complete the 7.7 km pull. The tabulated "Pull Force" is the maximum pull force, and is either the force required for the 7.7 km pull, or the force that would have caused failure in the pipe or rope. The last column identifies the rope or the pipe as the limiting segment.

Pull Condition	Rope Type	Assumed Pipe CF	Assumed Rope CF	Assumed Bend Radii	Max Pull Length (km)	Pull Force (kN)	Limiting Segment
dry	steel 3/4"	0.2	0.3	50 D	3.2	242	pipe
dry	steel 3/4"	0.3	0.4	50 D	1.9	242	pipe
dry	steel 3/4"	0.4	0.5	50 D	1.4	242	pipe
submerged	steel 3/4"	0.2	0.3	50 D	4.0	255	rope
submerged	steel 3/4"	0.3	0.4	50 D	2.6	255	rope
submerged	steel 3/4"	0.4	0.5	50 D	1.8	255	rope
submerged	steel 1"	0.2	0.3	50 D	4.7	242	pipe
submerged	steel 1"	0.3	0.4	50 D	2.8	242	pipe
submerged	steel 1"	0.4	0.5	50 D	1.9	242	pipe

Table 5. Tight Fit HDPE Liner Pull-In Simulation Summary

Table 5 for the Tight Fit Liner shows that none of the simulated cases resulted in a successful 7.7 km pull-in. For dry pipes, the maximum length was 1.4 to 3.2 km, depending on the friction assumption. Pipe strength was always limiting for the dry pipe cases. This means that stronger steel rope, or the use of UHMWPE rope, would not improve these "dry" pipe results. Submerged pipe cases with ¾-inch steel rope had pull lengths between 1.8 and 4.0 km, and the rope strength was limiting in these cases. This limitation implied that a stronger rope could improve pull lengths.

Choosing a 1-inch steel rope resulted in the pipe strength being limiting in all cases. However, the possible pull lengths were still only 1.9 to 4.7 km. These pipestrength-limited pull lengths would not be increased by using stronger or lighter rope.

Pull Condition	Rope Type	Assumed Pipe CF	Assumed Rope CF	Assumed Bend Radii	Max Pull Length (km)	Pull Force (kN)	Limiting Segment
dry	steel 1"	0.2	0.3	50 D	5.7	360	pipe
dry	steel 1"	0.3	0.4	50 D	3.4	360	pipe
dry	steel 1"	0.4	0.5	50 D	2.2	360	pipe
submerged	steel 1"	0.2	0.3	50 D	≥ 7.7	359	N/A
submerged	steel 1"	0.3	0.4	50 D	6.1	449	rope
submerged	steel 1"	0.4	0.5	50 D	4.6	449	rope
submerged	steel 2"	0.3	0.4	50 D	6.1	1,718	rope
submerged	UHMWPE 2"	0.3	0.2	50 D	7.5	360	pipe
submerged	UHMWPE 2"	0.4	0.3	50 D	5.4	360	pipe

Table 6. Loose Fit HDPE Liner Pull-In Simulation Summary

Table 6 shows the results for the Loose Fit Liner simulations. In the dry pipe cases, none of the friction assumptions resulted in a successful 7.7 km pull, and all cases resulted in the pull length being limited by the pipe strength, rather than by the rope strength. The possible pull distances were increased by 60% to 80%., relative to the Tight Fit Liner. For the submerged cases with 1-inch steel rope, the lowest friction assumption predicted a successful 7.7 km pull. The higher friction cases had feasible pull lengths of 4.6 to 6.1

km. The 1-inch steel rope was limiting in the unsuccessful cases. Using a larger 2-inch steel rope provided no improvement for the submerged cases since the peak tension was occurring at the very beginning of the pull when the only resistance was from the weight and friction of the pull rope. Since weight, friction, and strength all increase in tandem with rope size, further increases in rope size could provide no benefit. A pull rope with a higher strength-toweight ratio than that of steel rope (UHMWPE 2”) was able to provide ~20% longer pull distances. However, the benefit of the lighter and stronger rope was still not sufficient to allow the full 7.7 km pull(the pipe strength became the limiting factor).

Pull Condition	Rope Type	Assumed Pipe CF	Assumed Rope CF	Assumed Bend Radii	Max Pull Length (km)	Pull Force (kN)	Limiting Segment
dry	UHMWPE 2"	0.2	0.1	50 D	≥ 7.7	458	NA
dry	UHMWPE 2"	0.3	0.2	50 D	7.5	1,273	pipe
dry	UHMWPE 2"	0.4	0.3	50 D	5.2	1,273	pipe
dry	UHMWPE 2"	0.2	0.1	25 D	≥ 7.7	472	NA
dry	UHMWPE 2"	0.2	0.1	100 D	≥ 7.7	454	NA
submerged	UHMWPE 2"	0.2	0.1	50 D	≥ 7.7	93	NA
submerged	UHMWPE 2"	0.3	0.2	50 D	≥ 7.7	256	NA
submerged	UHMWPE 2"	0.4	0.3	50 D	≥ 7.7	654	NA
submerged	UHMWPE 2"	0.5	0.4	50 D	7.3	1,273	pipe
submerged	UHMWPE 2"	0.2	0.1	25 D	≥ 7.7	113	NA
submerged	UHMWPE 2"	0.2	0.1	100 D	≥ 7.7	73	NA

Table 7. Round High Strength Fiber Reinforced HDPE Liner Pull-In Simulation Summary

Table 7 shows the results for pulling in the unfolded High Strength Fiber Reinforced Liner. Due to the higher strength-toweight ratio of this liner system, (compared to the bare HDPE liner), only the UHMWPE rope was simulated. For the three dry pipe cases with 50 D bends, only the lowest friction case was predicted to be successful at 7.7 km. For the two higherfriction cases, the pipe strength was limiting. Adding more axial high strength fiber could theoretically provide some increased pull distance. The four submerged 50 D cases showed that all friction assumptions led to a successful 7.7 km pull, except for the highest friction assumption. Again, the pipe strength was limiting. The cases that considered 25 D and 100 D bends instead of 50 D bends, led to relatively small changes in the predicted pull force.

Pull Condition	Rope Type	Assumed Pipe CF	Assumed Rope CF	Assumed Bend Radii	Max Pull Length (km)	Pull Force (kN)	Limiting Segment
dry	UHMWPE 2"	0.2	0.1	50 D	≥ 7.7	694	NA
dry	UHMWPE 2"	0.3	0.2	50 D	6.6	1,433	pipe
dry	UHMWPE 2"	0.4	0.3	50 D	4.7	1,433	pipe
dry	UHMWPE 2"	0.2	0.1	25 D	≥ 7.7	730	NA
dry	UHMWPE 2"	0.2	0.1	100 D	≥ 7.7	683	NA
submerged	UHMWPE 2"	0.2	0.1	50 D	≥ 7.7	166	NA
submerged	UHMWPE 2"	0.3	0.2	50 D	≥ 7.7	463	NA
submerged	UHMWPE 2"	0.4	0.3	50 D	≥ 7.7	1,179	NA
submerged	UHMWPE 2"	0.5	0.4	50 D	6.3	1,433	pipe
submerged	UHMWPE 2"	0.2	0.1	25 D	≥ 7.7	203	NA
submerged	UHMWPE 2"	0.2	0.1	100 D	≥ 7.7	130	NA

Table 8. Folded High Strength Fiber Reinforced HDPE Liner Pull-In Simulation Summary

Table 8 shows the simulation results for the Folded High Strength Fiber Reinforced HDPE Liner. The behavior was very similar to that of the unfolded (round) pipe of Table 7. For the friction cases that did not lead to successful 7.7 km pulls, the distances were 10 to 15% shorter than for the smaller diameter round pipe of Table 7. For the successful cases, the pull forces were 50 to 80% larger. The most important factor limiting pull distances for the folded pipe, relative to the smaller reinforced pipe, was its 22% larger diameter that led to 50% greater weight (60% greater when submerged). Another factor was that, although ~13% more axial fiber was used in this pipe's design, this additional fiber was not enough to offset the increased weight and friction. More axial fiber could have been used in the design, and it is expected that pull distances could be achieved which exceed those shown in Table 7.

DISCUSSION

It must be recognized that axial pipe strength in this theoretical, comparative study is straight pipe strength, and that a safety design factor against combination bending and axial loading failure, was not employed. Even for the nominal 50 D bends of this study, the bending effect on maximum allowable tension could be significant and should be considered in a more rigorous pull-in calculation. This study also did not consider how the pull rope would be inserted into the host pipe. This rope insertion process can be more limiting to the length of a practical pull than the strength of the pull rope and pipe. A typical method for rope insertion is to have a pipeline pig pull the rope. The maximum practical distance for this method will depend on the same parameters and properties for pipe pull-in, discussed above. However, rope pull-in also depends on how much force can be generated by the pig. A pig's pull force is determined by the maximum pressure differential that can be sustained across the pig. Using a self-propelled crawler device is another approach that has shown significant potential when near neutral buoyancy pull-in rope can be employed to reduce the required pull forces [3].

SUMMARY

It is clear from the theoretical friction formulas that the "compounding" of pull forces at bends has a very significant effect on pull forces. This compounding occurs even at small angle bends and therefore can have a significant effect even in pipelines that would generally be considered straight. This force-compounding also means that tension introduced at the insertion point of the pull (such as for a tight liner's diameter reduction) will significantly increase the required pull force.

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SMARTPIPE TECHNOLOGIES

While the bend sequence and distribution along the pipeline affect the required pull force, the total bend angle appears to be a useful characterization of a pipeline's lack of straightness.

Although bend radius theoretically does not affect pull force for perfectly flexible rope, bend radius does affect pull force when the pulled pipe has significant bending stiffness. The effect of the bending stiffness is that it causes contact at the outside of host pipe bends, which can lead to significant additional friction. Once there is sufficient tension to avoid contact at the outer bend surface, friction force at that bend is again independent of bend radius.

For the liner types considered in this study, the bare HDPE tight fit liner shows the most limited pull length. This is due to the need for tension at the inlet of the pull to maintain the diameter reduction during the pull, and to the inherent low strength of the HDPE, compared to the high-strength fibers used in the reinforced pipes. None of the modeled tight fit liner cases were able to achieve the 7.7 km pull of this study. UHMWPE rope provided no benefit to maximum pull distance for this liner because the pipe strength limit was reached with steel rope.

The loose fit liner was able to achieve the 7.7 km pull only when the pipes were submerged and the lowest friction coefficients were assumed. The use of UHMWPE rope had some benefit to the maximum possible pull distance for this liner design when the pipes were submerged, since the steel rope weight to strength ratio was more limiting than that of the liner.

The two high-strength fiber reinforced liners were both theoretically able to be pulled the 7.7 km distance in the dry condition for the lowest friction assumptions. When these liners were submerged, only the worst case friction assumptions prevented achieving the full pull length. The UHMWPE rope was generally a benefit to these cases, in that the strength-to-weight ratio of the steel rope was insufficient to pull itself the full 7.7 km for all but the lowest assumed friction factors. The ability to design the high-strength axial tapes to achieve a required strength is clearly a benefit of these liner designs. Another key advantage is the ability to design for significantly higher internal pressures than is possible with loose fit bare HDPE.

CONCLUSIONS

Relatively straight offshore pipelines combined with flooding the pipes and selecting liner and pull rope materials with nearneutral buoyancy, provide the opportunity for much longer pull-ins than typically considered feasible. A high-strength, fiber reinforced liner (folded or round) also provides significant additional potential gains in pull distances, operating pressures, and flow rates.

RECOMMENDATIONS

While the basic friction formulas of this study are simple and easy to implement, the practical complexities of liner ovalization and bending stiffness effects at bends could lead to inaccurate pull force predictions. The exponential term with $\mu\beta$ (friction coefficient times bend angle) in the pull force formulas means that uncertainty in either μ or β will lead to disproportionately larger uncertainty in the pull force predictions. To improve the model and predictions, more opportunities must be found to compare the model to measured pull behavior. Pull-in force data for cases in which pipeline bend geometry information and/or for which friction coefficients are well known would be most useful. Finite element modeling should be used to gain a better understanding of ovality and pipe bending stiffness effects.

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