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TORSION IN FLEXIBLE PIPES, UMBILICALS AND CABLES UNDER LOADOUT TO INSTALLATION VESSELS

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ABSTRACT

Operations where a flexible pipe, umbilical or cable is loaded out from an on-shore spool to the hold of an installation vessel can cause a build-up of torsion. In unfavourable cases, the torsion has been known to cause spiralling or various forms of damage to the tensile armour.

This paper presents a vocabulary for the description of torsion. It then gives a short review of design codes, enumerates known failure modes (some of which have not always been identified as torsion related), and discusses the mechanisms of torsion generation, with an emphasis on the effect of internal friction. It concludes with some ideas on torsion prevention.

INTRODUCTION

During production, transport and installation, non-bonded flexible pipes, umbilicals and cables (in the following, collectively referred to as “flexibles”) are handled by "paying out": the flexible is pulled along a route, which may involve any combination of reel, chute, stinger, caterpillar, turntable, tensioners and so forth.

Under some conditions during load out, the flexible may start to roll along its axis, and significant torques may develop. This can result in "pig tailing" (the flexible takes a spiral shape), or various other local damage, which can be severe enough to make the flexible unserviceable. The later damage mechanisms may be mistaken for the effect of excessive curvature, or for the effect of excessive compression (birdcageing). These torsion issues are particularly prevalent under loadout operations – when flexibles are paid out from onshore storage into the hold of an installation vessel.

The financial impact of damage to a flexible can be significant. Lack of insight into the mechanism of the failure, combined with the fear of litigation between partners in a project have made this a taboo topic. Since failures cause material damage, but no risk to safety or the environment, authorities do not get involved either. The author has had the opportunity to work with some failures and study their mechanism, and has arrived to the conviction that the issue is widespread enough to be of general interest.

The present publication is peculiar in that industrial actors known to this author, even though they may have worked diligently to address the issue, have insisted on anonymity. This has caused a delay of several years in this publication, and unfortunately, the present text does not report or refer to some excellent data that has been collected in situ. It is hoped that

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this presentation is still of interest to the industry, and will contribute to the understanding and control of this issue.

Ribbon theory is a branch of mathematics that studies the relation between the 3D geometry of a ribbon (a curve with a side mark), its twist, and roll. It is shown [1][2] that in a 3D geometry, the marking on a flexible can change orientation in the absence to twist and hence torque. The author does not know of presentations of the issue accessible to engineers.

Fylling et al. [3] studied the resistance to the change of curvature plane in a flexible. While they did not consider loadout torsion, their findings are highly relevant here.

People handling ropes, including, rock climbers, have valuable empirical knowledge on the issue at hand. Geometric effects have implications about how ropes are stored (lapped in a figure of 8 or in a heap, rather than coiled). Modern belay and abseil devices keep the rope in a plane within the device. Devices that do not (figure-of-8, Italian hitch) tend to cause torsion: the rope rolls through device during abseil.

Longva et al. [9][10] have created a FEM software for the study of loadout torsion, based on some of the ideas belatedly presented here.

Design codes offer very limited guidance on torsion, and this will be reviewed in the following.

The present paper starts with introducing the notion of *writhe*, and generally provide a rigorous vocabulary for the study of torsion. Flexibles are sometimes given a longitudinal marking. This marking may be on the side of the pipe, and after a few bends, may be on the top of the pipe, *in the absence of any torque in the pipe*.

The paper then describes some torsion-related failure modes, to prevent their confusion with failures related to excessive curvature or compression.

It then provides a brief review of how torsion under handling is covered in design codes.

The paper then studies the causes of torque in the pipe during load out, based on forensic work carried out by the industry and the author.

Finally, based on the understanding of the mechanisms generating torque, possible strategies for the mitigation of load out torsion are discussed.

NOMENCLATURE

c_i	Coulomb friction coefficient
$\bar{e}_i$	Base vector
G	Writhe
K_T	Torsional stiffness
M_f	Friction moment
M_x	Torque
p	Pitch length
P_j	Contact pressure
R	Radius of a layer
R	Roll
$\dot{R}$	Roll rate
S	Link
s	Curvilinear coordinate along a tendon
T	Torque
W	Friction work
α	Angle around the cross section
δ	Slip
Δ	Slipped distance
ΔR_G	Writhe-induced roll increment
ΔR_S	Total roll increment
ΔR_τ	Twist-induced roll increment
ε	Elongation of a tendon trajectory
τ	Twist
κ	Curvature
ξ	Curvilinear coordinate along the pipe
$\bar{\omega}$	Rotation vector
$\bar{\bar{\Omega}}$	Rotation matrix

GEOMETRY AND VOCABULARY

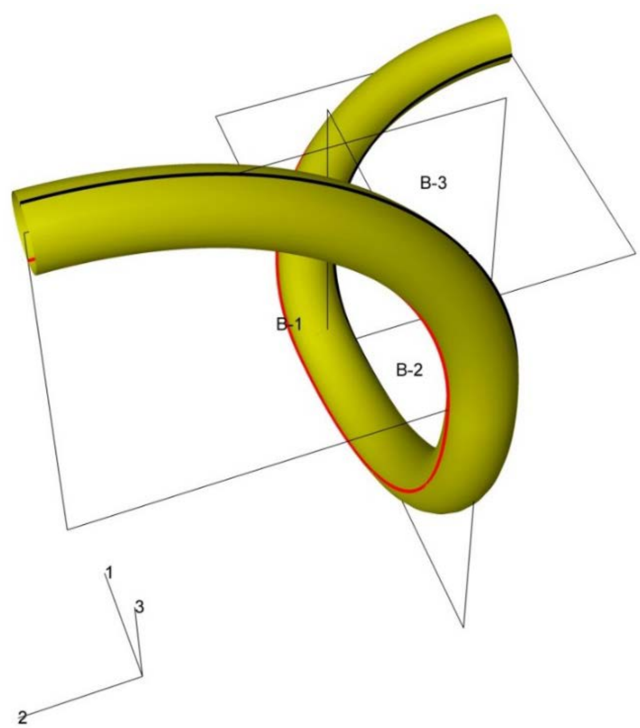

Figure 1: The roll angles at both end of this pipe are different because of writhe. The pipe shown here has no

twist. "B-i" indicates an area in which the pipe is bent around axis i.

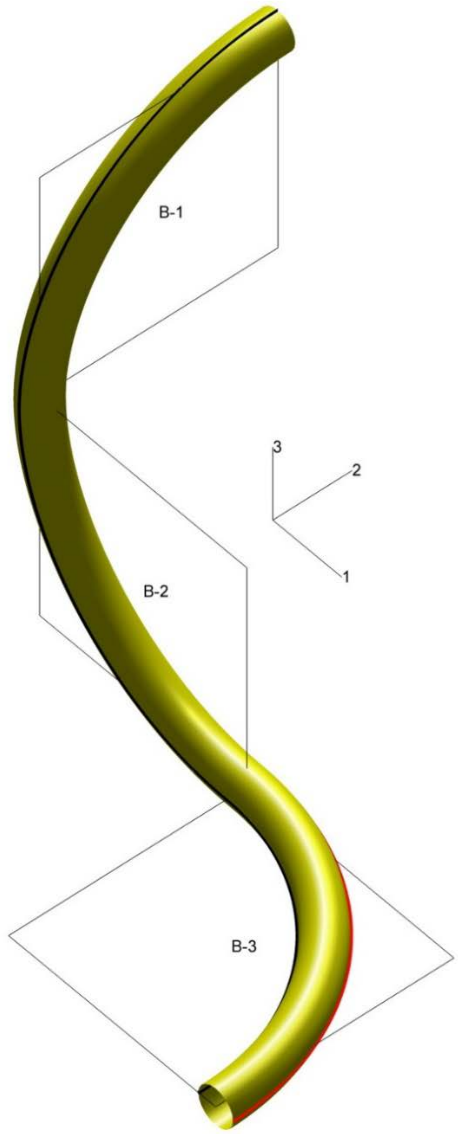

Figure 2 Same as Figure 1, different perspective

Figure 1 and Figure 2 show a pipe segment in which each of its 3 thirds are *bent*. Examination of the markings on each of the thirds shows that there is no "twist" or "torque" (to be defined later). Still the sequence of bends is such that the black marking, which is at 9 o'clock at the lower end of the pipe, is at 12 o'clock at the upper end.

Figure 3 shows a more complicated example of a pigtail geometry. It has zero twist, and yet the roll angle changes continuously along the pipe. (In a real pig tailed pipe, there will be twist superimposed to the purely geometric effects.)

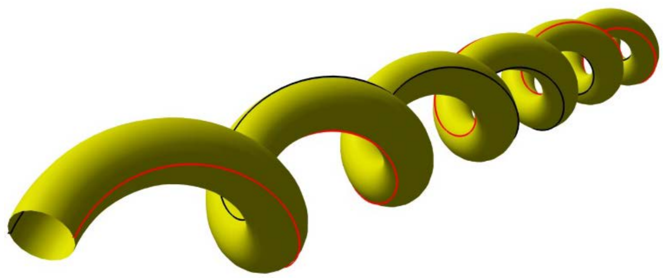

Figure 3: A "pig tail" (positive sign) is a more complicated example of *writhe* (positive sign). The pipe shown here has no *twist*: at all points along the pipe, the black marking is exactly perpendicular to the pipe circumference.

The geometry of the pigtail can be seen as a "smoothed" version of the one shown in Figure 1: in the pigtail, the plane of curvature varies continuously while it varies in steps in Figure 1.

In both examples above, dividing the difference in roll angle by the length of the pipe segment would lead to the wrong conclusion that the pipe is undergoing twist, while it is only bent.

It can be shown that this effect, known as *writhe, can only occur on flexibles that are not restrained to a single plane*. This is simply a consequence of the commutativity of rotations around a given axis (the axis normal to the single plane). Two practical examples of this situation spring to mind: the free span in a vessel's turntable, and a pig tail as tends to appear in the presence of torque.

Definitions

Curvilinear coordinate (s) [m] The length measured along the pipe. The origin is a fixed point on the loadout route, not a material point on the pipe. In the context of loadout torsion, curvilinear coordinates increase in the direction of planned transport of the pipe. For example, when transferring the pipe from a spool on land to the ship, *s* increases towards the ship. The term "downstream" (respectively "upstream") is sometimes used for the direction of increasing (respectively, decreasing) curvilinear coordinates.

Roll (*R*) [deg] or [rad]: Angle of rotation around the pipe's axis, measured as the angle by which the pipe must be rolled to get a vector that is normal to the pipe and in a vertical plane, to point from pipe axis to the marking on the pipe. As a consequence of the definition, roll is undefined if the pipe is vertical. In practice, roll is measured on a horizontal, or near horizontal section of the pipe.

Roll is defined to be positive for rotations that are seen as clockwise when looking downstream.

Figure 4 illustrates one procedure to measure roll angle. Note that using the instrument as held here, it is necessary to subtract 90 degrees to the measured roll to obtain a value consistent with the above definition.

Figure 4 Roll angle measurement

Total roll increment (ΔR_S) [deg] or [rad]: The difference between values of roll measured at the same instant at two positions along the pipe:

$$\Delta R_S = R(s_2) - R(s_1)$$

with $s_2 > s_1$.

Link (S) [deg/m] or [rad/m], or link: The derivative of roll along the length of the pipe. Over a length L of pipe the (average) link is equal to *ΔR/L*. More precisely

$$S(s) = \frac{\partial R(s)}{\partial s}$$

Twist (τ) [deg/m] or [rad/m]: Intuitively, twist is the part of the link that is proportional to torque by a factor that is the torsional stiffness. This cannot be taken as a definition because we wish to define the torsional stiffness as a ratio of torque to twist, leading to a circular system of definitions.

For a *straight pipe* (or for a pipe within a horizontal plane) $\tau(s) = \partial R(s)/\partial s$ but this is *not* the general definition we need either.

Figure 5: A straight segment of pipe under twist (positive sign).

General definition: Let s be the curvilinear coordinate. At any point along the pipe one defines the local orthonormal reference system $\bar{\bar{e}}$ where $\bar{e}_1$ is tangential to the pipe, $\bar{e}_2$ is radial pointing *towards the marking on the pipe* and $\bar{e}_3 = \bar{e}_1 \times \bar{e}_2$, where $\times$ is the cross product. This local reference system (not be confused with the Frenet-Serret reference system), undergoes rotation between two neighbouring points along the pipe. This rotation can be characterised by a rotation gradient vector $\bar{\omega}$. The direction of $\bar{\omega}$ is the axis of the rotation and its length is equal to the rotation gradient [rad/m]. Twist is defined as the component of $\bar{\omega}$ in the direction of $\bar{e}_1$.

Figure 6 shows a twist-free pipe. Seen in the tangential direction, $\bar{e}_2$ and $\bar{e}_3$ of neighbouring reference systems (in black) are aligned. Seen from a normal direction, the rotation gradient vector $\bar{\omega}$ (in red) is in the plane of the cross section. Figure 7 shows the same pipe with twist superimposed. Seen in the tangential direction, $\bar{e}_2$ and $\bar{e}_3$ of neighbouring reference systems are no longer aligned and the rotation gradient vector has an axial component.

The same expressed in mathematical notations: The matrix of rotation gradient (with respect to s) of $\bar{\bar{e}}$ can be computed as

$$\bar{\bar{\Omega}} = \lim_{x \to 0} \frac{\bar{\bar{e}}(s + ds) \cdot \bar{\bar{e}}^{-1}(s)}{ds}$$

and from it the vector of rotation gradient $\bar{\omega}$ of the local reference system is found as

$$\bar{\omega} = [\Omega_{23} \quad \Omega_{31} \quad \Omega_{21}]^T$$

and twist is the axial component of the rotation rate

$$\tau = \bar{\omega} \cdot \bar{e}_1$$

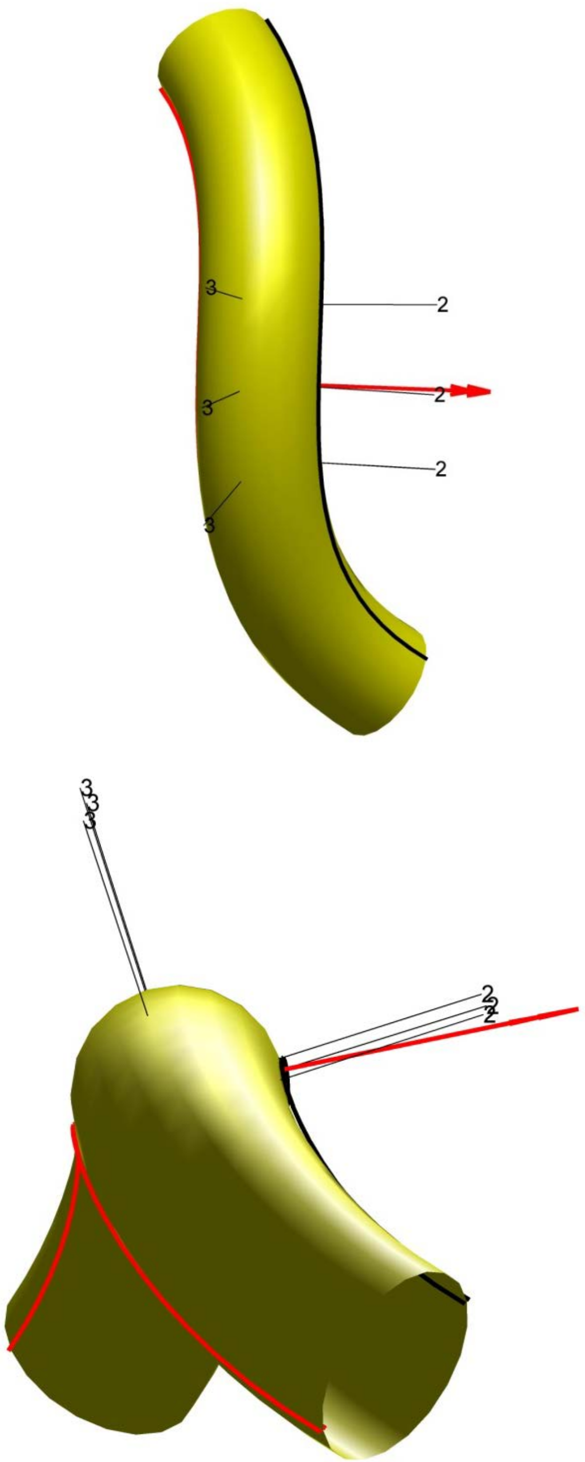

Figure 6 Geometry of a twist free pipe. The same object is seen from two perspectives.

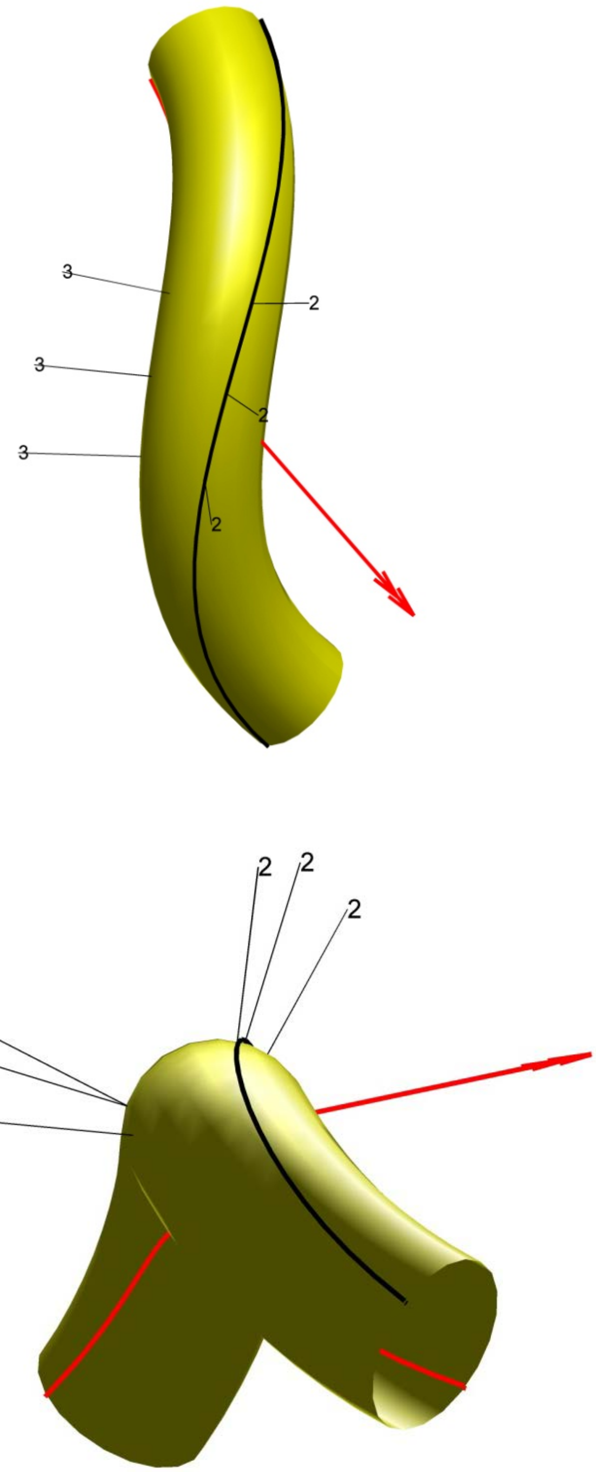

Figure 7 Geometry of a pipe with twist (positive sign). The same object is seen from two perspectives.

Writhe (G) [deg/m] or [rad/m]: Writhe is simply the link minus the twist.

$$S = \tau + G$$

This is the Călugăreanu–White–Fuller theorem in ribbon theory.

Torque (M_x) [Nm]: Also known as torsional moment, a moment around the pipe's axis. A pipe with a positive internal torque has a positive *twist*.

Torsional stiffness (K_T) [Nm^2/deg] or [Nm^2/rad] The ratio of the torque to the twist.

$$M_x = K_T \cdot \tau$$

Twist-induced roll increment (ΔR_τ)[deg] or [rad]: The part of the difference in roll angle between two points, that is due to twist in the pipe.

$$\Delta R_\tau = \int \tau \, ds$$

Writhe-induced roll increment (ΔR_G) [deg] or [rad]: The part of the difference in roll angle at two point, that is due to writhe in the pipe.

$$\Delta R_G = \int G \, ds$$

Just like writhe added to the twist gives the link, one can show that

$$\Delta R_S = \Delta R_\tau + \Delta R_G$$

Roll rate [deg/m] or [rad/m] is the change in roll at a given point along the loadout route (as opposed to a given material point on the pipe) for each meter *t* of pipe paid out.

$$\dot{R} = \frac{\partial R}{\partial t}$$

Roll rate and twist must not be confused. The roll rate is a time derivative while twist is a gradient (a space derivative), but the unit of "time" is "the time it takes to pay out 1m".

Measuring twist

Torque is related to twist, as stress is related to strain. In operation, in order to assess the torque, one simple approach is to assess the twist, and multiply it by torsional stiffness obtained, for example, from simulation.

Because twist is small, one needs to measure the roll at points that are sufficiently far apart. It is then important to subtract the writhe-induced roll to the total roll difference to get the twist-induced roll. The only practical way to do that (baring precise measurement of the geometry by photogrammetry) is to ensure that the writhe is zero, by choosing a segment along which the pipe bends only in a single plane.

For example loading a flexible pipe out to a vessel equipped with a turntable, the free span between the deck of the vessel and the bottom of the turntable can have a significant writhe, which can also vary significantly with changes in the geometry of the free span. Misinterpreting this writhe as twist can give the impression that the pipe is under a heavy torque.

Frenet-serret torsion

The concept of Frenet-Serret torsion is briefly addressed here, only to be dismissed as irrelevant to the present issue. This is done to eliminate a known source of confusion.

By definition [4][5], the Frenet-Serret reference system associated to a curve has: a vector tangent to the curve, one vector pointing inside the curvature, and one vector orthogonal to the two previous vectors. This differs from the definition of the reference system $\bar{\bar{e}}$ above: $\bar{e}_2$ is pointing towards the marking, and not inside the curvature. Indeed $\bar{e}_2$ is well defined for a marked straight pipe, while the Frenet-Serret reference system is not.

The Frenet-Serret *torsion* is defined as the dot product of the rotation gradient of that reference system with the tangent vector. The Frenet-Serret torsion is thus neither a link, a writhe, a twist nor a torque as defined in this text. So far, Frenet-Serret torsion has not been found to be relevant in the present context.

TORSION RELATED FAILURE MODES

Pigtailing

Figure 3 shows a flexible with an idealised pigtail geometry: it is twist free, but writhed, and indeed there is a total roll increment between both ends. A straight flexible under twist (Figure 5) also has total roll increment between both ends, but due to twist, and not writhe. Pigtailing occurs when twist is exchanged for writhe.

When is pigtailing triggered? Looking at it from the energy point of view: the energy of the flexible is equal to the sum of the energies related to torsion, bending, elongation, and the work of external forces on the flexible segment. For a flexible under tension, with no bending moments applies at the end, and a low enough torque, the straight twisted flexible is the state of lowest energy. As torque increases, the amount of torsion energy increases, until a critical point is reached, where the energy of the system can be reduced by reducing twist by taking a 3D trajectory (spiral), even though the 3D trajectory implies some bending energy and some work to pull together both ends to compensate for the longer trajectory.

Because absorbing changes in roll angle without twist requires significant deviations from a straight line, some pulling-in of the ends is necessary. This of course requires work against tension. This work is minimized, at the cost of larger curvatures, by keeping the pigtailed area short. It can indeed be observed in small scale experiments that as tension increases, pigtailing occurs at higher twist levels, with a more localized pigtail.

Because a pigtail localizes deformation in a system which is hysteretic (slip of the tensile armour), it cannot be reversed under tension, just like a bar kinked during plastic compressive buckling cannot be straightened by tension. Hence in operation, applying tension to prevent pigtailing can be allowable, but putting a pig tail under the same tension can be damaging.

During S-lay, steel pipelines may be deformed plastically in the hog bend (over the stinger). There are then several mechanisms of instability. One of them is that the pipe on the seafloor is liable to minimize its internal energy by twisting, thus making it possible to keep a curvature close to the residual curvature without changes in pipeline heading on a larger scale.

The results of spiralling and pigtailing look the same, but there is a significant difference. In flexible spiralling, roll and twist is introduced in order to reduce curvature energy. In pig tailing, curvature is introduced in order to reduce twist energy.

Strangling

When a flexible with two tensile layers is twisted in the slack direction, the torsional stiffness typically comes from the reaction of inner components (like pressure armour and carcass in a flexible pipeline) against the inward motion of the inner tensile layer. If the torque is excessive, the inner components may potentially collapse, allowing the inner tensile layer to move inwards, causing a local concentration of twist, thus forcing the outer tensile layer to birdcage.

This is distinct from birdcageing due to wall compression, in which all tensile layers typically move outwards.

Herniation

If a flexible is twisted in the tight direction (assuming again that there are only two tensile layers), then the inner layer is in compression. Simultaneously, the outer layer's lay angle is slightly reduced, which reduces its packing.

An unstable situation, referred to as herniation, has been experienced where strands of the inner layer deform plastically and bulge through a gap between two strands of the outer layer.

A potential variation of this failure mode might occur in flexibles in which a high tensile polymer layer is wrapped around the outer tensile layer. Under a twist in the slack direction, the outer tensile layer could herniate its way through a weakness in the polymer layer.

In-layer buckling

Potentially, if a flexible is twisted in the tight direction and no herniation occurs, or, as has been experienced, if a flexible is twisted in the slack direction, and the outer armour layer is wrapped in plys of high tensile membrane, then in-layer buckling may occur.

In in-layer buckling, the axially compressed tendons jointly undergo plastic buckling around their strong axis, taking a curved path within their own layer.

Skew-kinking

When a steel tube or a flexible pipe is subjected to an excessive bending moment, it first ovalises, then kinks, forming a hinge with an axis orthogonal with the axis of the pipe, and parallel to the internal moment carried by it.

If a torque is added to the bending moment, then a kink can still form, but experience shows that it axis is no longer orthogonal with the axis of the pipe. The hinge is thought to be in a direction “between” the orthogonal to the pipe axis and the direction of the moment vector (sum of the bending moment and torque vectors). The hinge follows another direction than the direction of the moment vector in order to limit the length over which the hinge intersects the pipe and hence limit the amount of plastic work needed to create the hinge.

Experience indicates that skew kinking can occur at levels of curvature that could be safe in the absence of torque.

Unlocking of the pressure spiral

This failure mode is mentioned in DnV-RP F206 [6], Table C1. Torsion with the same sign as the pressure spiral can cause it to unlock, which in turn can cause later failure of the pressure containment.

TORSION IN DESIGN CODES

API 17B [7], section 5.4.1.7.2 prescribes the use of a yield criterion: the axial stress in the compressive layer must not exceed 0.72 of SMYS.

5.4.1.7.2 The maximum acceptable torsion derives from the following two scenarios, depending on the direction of the applied torsion:

a. The outer tensile armor layer is turned inwards and pressed against the internal layer (in which case the allowable tension causes overstressing of the tensile armor) by inducing a stress corresponding to its structural capacity (defined by Section 5.3.1.4—of API Specification 17J multiplied by the utilization factor, as specified in Table 6 of API Specification 17J).

b. The outer tensile armor layer is turned outwards and pressed against the outer layers, leading to a gap between the two tensile armor layers in which case, the damaging torsion, induces a gap between tensile armor layers, (in which case, the damaging torsion, induces a gap between tensile armor *layers equal to half the thickness of the tensile armor wire). The allowable torsion for this case should be calculated from the damaging torsion using a safety factor not less than 1.0.*

API 17J [8], section 6.3.1.4 states:

The structural capacity shall be either the yield strength or 0.9 times the ultimate tensile strength of the material where tensile testing can accurately identify only this later property.

Experience suggests that some of the failure modes described above can occur at torques that would be deemed safe by API 17B.

MECHANISMS OF TORSION GENERATION

Observations

Observations made during loadout operations, from observations by alert operators to well-planned and extensive measurement campaigns carried out by the industry, have provided important clues. These clues provided guidance to select, from a range of hypotheses, the mechanisms presented here. For reasons of confidentiality, the sources of these observations may not be revealed here. The author pledges to acknowledge these sources, should they choose to come forward.

The scenario relevant in the following is a flexible, paid out from an onshore storage spool, traveling a route over roller and tensioner, to the side of an instalation vessel. The flexible is led to the deck of the vessel, on which its route turns aft, and then down through a hatch. From the hatch, the flexible has a free span (along which a variety of tools can be used to control its geometry) as it gos down into a turntable, and turns left intot he turntable. The free span is illustrated in Figure 8.

Considering a fixed point along the route of the flexible, roll is found to build up progressively (in the negative direction), and eventually reach a constant level. During the roll build-up, the flexible, while mostly fixed once laid down in the turntable, has a positive twist in the turn table (as the flexible is coiled along a vertical axis, layer by layer, the writhe in the turntable is negligible). Once the roll build-up ceases, the twist in the turntable essentially disappears.

During the roll build-up, roll appears first downstream (on deck just above the hatch) as is observed to propagate upstream along the loadout route. This propagation does not seem to be hindered by bends or friction against rollers and tensioners along the route. When the roll build-up has ceased, the twist along the loadout route, from the onshore spool or turntable to the hatch, is roughly constant.

Superimposed to the "long term" build-up and stabilisation of roll mentioned above, "short term" transients were observed, with the negative roll rate being strengthened or weakened. The propagation of roll upstream along the route was very clear in these short term transients. In one case, the roll rate reversed (became positive), leading to a negative twist in the turntable. At the same moment, it was found that the free span from the hatch to the turntable, which usually resembled a negative spiral had been manipulated into a positive spiral.

Cranking

Applying a force in a direction orthogonal to the plane of curvature of a pipe amounts to loading a crank and will obviously produce a torque.

If one considers the above-mentioned free span in a turntable, and imagines that the lower and upper ends of the free span are restrained from rotating, then the total roll increment is fixed. This implies that changing the writhe along the free span will cause twist, and hence torque.

Hence a more subtle variant of cranking can occur when forcing the pipe to move within its plane of curvature: the in-plane forced displacement may require a force with a component orthogonal to the plane of curvature: one can thus unwittingly be cranking the pipe.

Simulations have shown that cranking can cause damaging torque in flexibles [9][10].

Inductive reasonning

Cranking, does not explain the progressive build and stabilisation of roll that is observed in several operations. Roll rates were observed also in periods when the shape of the free span was being kept practically constant. So in addition to cranking, another one must be at play too.

In a thought experiment, consider that the flexible is a conservative mechanical system (no friction, no heat production): it is a hyper-elastic system. Its internal energy only depends on its curvature along its length. Further, the flexible being rotation-symmetric, its internal energy density at one cross section does not depend on the direction of the curvature, only on its intensity. Similarly, the flexible being uniform along its length, the total internal energy along the route does not change if the flexible is paid out along the route.

Let the flexible be force to follow a given route. The flexible is let sort out its roll (minimisation of the internal energy with respect to roll degrees of freedom, leading to uniform twist along the route, or zero twist, depending on boundary conditions). As the flexible is then paid out along the route, the internal energy (integrated along the route) is unchanged, and uniform twist is still the lowest state of energy.

Hence, in the absence of dissipative phenomena (friction) there can be no other source of torque than torques applied at the end of the flexible. This is not compatible with the observations: the "motor" of loadout torsion must necessarily involve friction. In a different setting involving flexibles [3], it was remarked that a curved flexible resists any change to its curvature plane: this requires to mobilise the slip of tensile armours along the relevant segment, generating friction, and thus requiring a torque.

In the case of a load out operation, observations point to the free span between hatch and turntable as a source of torsion. Indeed this free span is characterised by a progressive change of the plane of curvature along its length (Figure 8).

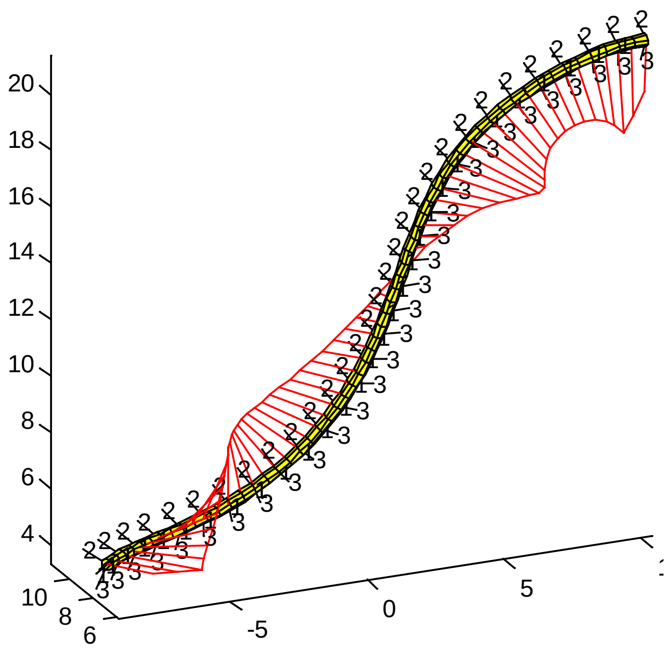

Figure 8 Free span geometry. "1-2-3" marks local reference system, with e.g. "2" pointing in the direction of a longitudinal marking. The curvature vectors are in red.

Analytical model of roll

Assuming that the geometry followed by the flexible is known, and using a local approximation of locally constant curvature [11], one can compute the amplitude of the displacement of a tendon analytically. The total distance followed by any material point on the tendon, as the flexible is rolled a full circle, is 4 times the amplitude. If the contact pressures and friction coefficients are known, the amount of friction work (per unit length of flexible, for a given curvature) can be assessed. The torque per unit length is then simply the work, divided by the total roll angle 2π.

The elongation of the trajectory of a tendon is

$$\varepsilon = r\kappa\cos^2\alpha \cdot \sin\left(\frac{2\pi}{p}\xi\right)$$

where r is the mean radius of the layer, κ is the curvature, α is the lay angle, p the pitch length and ξ the curvilinear coordinate along the flexible. Let s be the curvilinear coordinate along the tendon, so that $\xi = s\cos\alpha$, which is replaced in the above equation. By integration with respect to s, the displacement under curvature of a point of the tendon, with respect to the rest of the flexible

$$\delta = -\frac{p}{2\pi} r\kappa\cos\alpha \cdot \cos\left(\frac{2\pi}{p} s\cos\alpha\right)$$

The total displacement of the tendon over a full round of roll is hence

$$\Delta = \frac{p}{2\pi} r\kappa\cos\alpha \cdot 4$$

If the friction shear per unit surface for both faces of the tendon is τ then the work per unit length of flexible to roll it one round is

$$W = \Delta\tau 2\pi r$$

which divided by 2π gives the torque per unit length

$$\frac{dT}{ds} = \frac{2p}{\pi}\tau r^2\kappa\cos\alpha$$

for each layer. Noting that

$$\frac{p}{2\pi r} = \frac{\cos\alpha}{\sin\alpha}$$

one can rewrite

$$\frac{dT}{ds} = 4\tau r^3\kappa\frac{\cos^2\alpha}{\sin\alpha}$$

From similar considerations on a straight flexible being bent, the friction bending moment of the flexible (accounting for Coulomb friction only, not for material stiffness) is known to be

$$M_f = 4\tau r^3\frac{\cos^2\alpha}{\sin\alpha}$$

for each layer. So finally: to roll the flexible requires a torque

$$\frac{dT}{ds} = M_f\kappa$$

In [3] it is shown that the above equation is valid independently of the structure of the product. For example it is also valid in a flexible where friction is *not* dominated by the tensile armour.

Asymptotic friction moment

The asymptotic friction moment M_f is the limit at high curvature of the moment generated by friction between layers. If the contact pressure at all layer interfaces and the friction coefficients are known, this can be computed as

$$M_f = \sum_{i=1}^{nlay} 4\,\tau_i\, r_i^3\frac{\cos^2\alpha_i}{\sin\alpha_i}$$

where r_i is the mean radius of layer i, α_i is the lay angle, and τ_i the sum of the contact-shear stresses *on both sides of the layer*. The shear stress is found as the product of the contact pressure by the Coulomb coefficient.

Load-out operations take place at low effective tensions, where contact pressure is *probably* dominated by the shrinkage of the plastic layers. This hypothesis needs to be verified. With the above assumption, the pressure P_j inside a plastic layer j of radius r_j, thickness t_j, and yield stress $\sigma_{y\,j}$, is

$$P_j = \frac{t_j\,\sigma_{y\,j}}{r_j}$$

which gives rise to shears stresses summed over both sides of tensile layer i

$$\tau_i = 2c_i \sum_j P_j$$

where the sum is over all plastic layers which contain tensile layer i, and c_i is the Coulomb friction coefficient.

An alternative is to obtain M_f experimentally. A moment-curvature graph must be obtained under the relevant tension and temperature conditions. Then, the graph must be corrected (Figure 9) by subtracting the sum of the elastic moments due to the curvature of each individual component of the cross section. The asymptotic value of the correct graph is M_f. In practice:

$$M_f(\kappa) = M(\kappa) - \kappa\frac{\partial M}{\partial\kappa}(\kappa)$$

where κ a "high enough" level of curvature, such that the contribution from the friction moment is close to its asymptotic value).

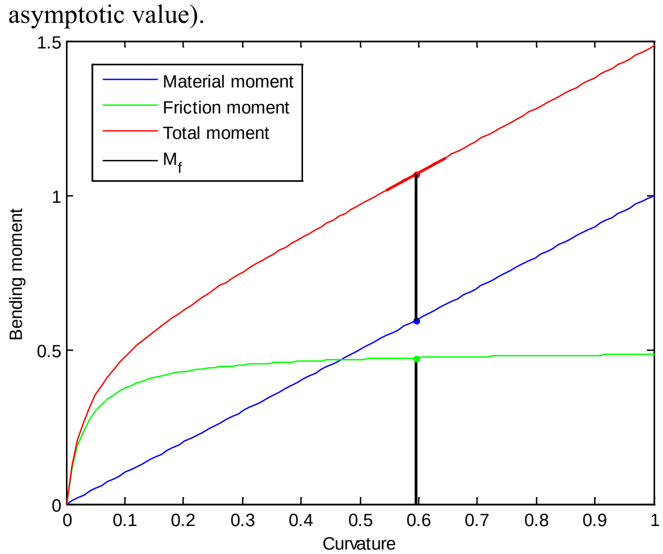

Figure 9 Computing M_f from bending test data. The vertical black segments are both of length M_f

Combined changes of curvature plane and intensity

More generally, as a material point travels with the flexible along the loadout route, it will experience simultaneous changes of curvature plane and intensity.

In the absence of non-linear effects, the bending moment needed to apply an incremental curvature to an unloaded, but not necessarily straight flexible is independent of the unloaded geometry of the flexible. The friction part of this moment is (for a large enough increment) of given intensity M_f and of direction opposed to the direction of the curvature increment.

To express the same in other words, let us consider the trajectory, in the curvature diagram, of a material point of the flexible as it is subjected to changes of curvature (Figure 10). The friction resistance that appears when displacing in the curvature diagram can be represented by an arrow of length M_f, tangent to the trajectory, and of direction opposite to that of movement along the trajectory.

Using the vocabulary of polar coordinates, this arrow can be decomposed in a radial part (part of the arrow parallel to the direction between the origin and the point of the trajectory) and an angular part (the part of the arrow orthogonal to the above direction). The radial part is readily interpreted as the value of a bending moment which vector is normal (in physical space) to the plane of curvature.

The angular part is interpreted using the energy considerations introduced above: the length of the angular part of the resistance, multiplied by the curvature, is a torque per unit length of flexible.

Figure 10 shows the curvature along the flexible (black curve). The dots correspond to equally spaced points along the flexible. The red vector is tangent to the curve, and has been normalized (to an arbitrary length, to be kept constant as one changes point along the curve). The contribution to torque dT/ds from the point under consideration is proportional to the (signed) area of the blue triangle: *the torque per unit length induced by the change of curvature plane is the cross product of the curvature vector with a vector of length* M *and of direction tangent to* dM/dt.

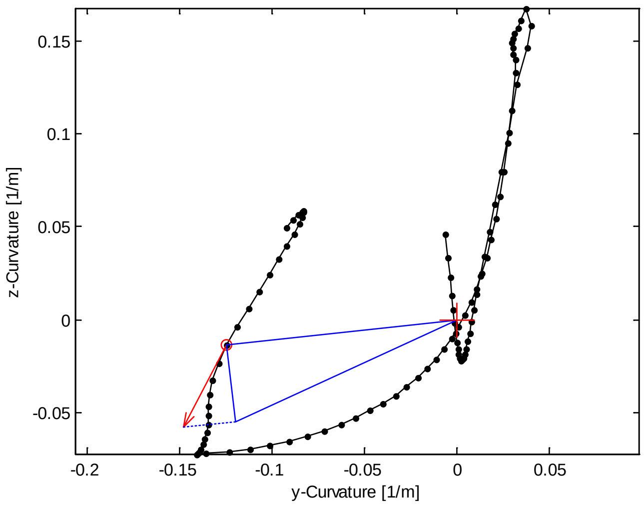

Figure 10 Illustration of torque evaluation

Torque integration

At this stage, a simplification can be introduced. It is assumed that integrating the above torque-per-unit-length along the free span gives the difference between the internal torques at the top, respectively bottom of the span. This assumption is strictly true if the flexible is restricted to follow a given route (or free span geometry) exactly – for example by being confined within a hypothetic frictionless steel tube shaped like the free span. In reality, the high levels of curvatures, combined with multiple supports of the flexible result in a more complicated picture.

Effect of roll rate

Since torque is generated by changes in curvature plane *as experienced by a cross section of the flexible while moving along its route*, the flexible may roll during pay-out to accommodate the change of curvature plane with a minimum of friction work involved – this is precisely what happens early in the roll build-up phase.

This effect is simply to account for in the torque integration, and thus it is possible to produce a curve, which, for a given fee span geometry and flexible characteristics, show the relation between torque and roll rate (Figure 11). In the example shown, the curve is nearly linear. Combined with a constant torsional stiffness from the upstream route, this leads to expect exponential transients in roll – which is in accordance with observations.

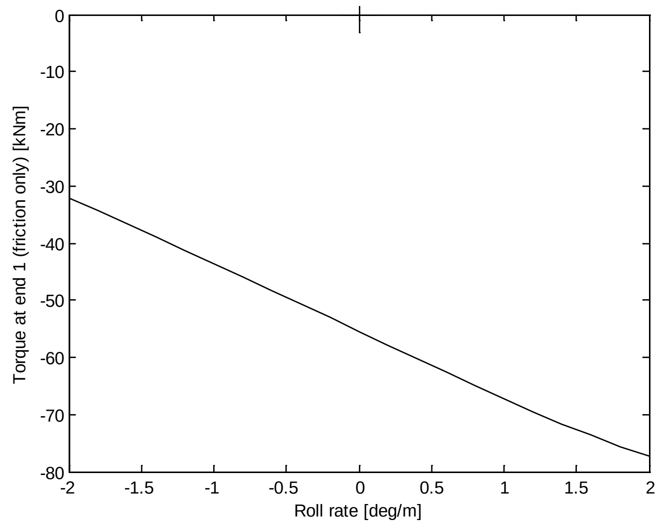

Figure 11 Torque as a function of roll rate

Discussion

There is good qualitative agreement between the properties of the mechanism identified, and the observation: the roll "motor" is in the free span between hatch and turntable, or more generally speaking, in segments along the route with high curvatures and slowly changing curvature plane. There is a an equilibrium between the roll-rate dependant torque generated in the free span, the torque in the twisted flexible in the turntable, and the increasingly twisted flexible in the loadout root from on-shore spool to the hatch at the top of the free span.

For a given friction level, and for a given geometry, it is fairly straightforward to compute the torque. Computations, which may unfortunately not be reported here, showed that the torque that can be generated in this way is high enough to cause a variety of failures.

The numerical model presented here (in-house software Jordan) can provide some important insights.

1. The torque depends on the route geometry
2. Plane routes do not induce torque
3. Two bends in different planes, separated by a straight segment, generate no torque.
4. For a given curvature, a change of curvature plane produces a torque which is proportional to the length over which the curvature plane changes: a progressive change of plane is worse than an abrupt one. (Bear in mind that the model loses validity when the curvature plane changes over a length approaching the pitch length.)

5. The torque depends on the roll rate.

Uncertainties on geometry

A major weakness of Jordan is that it requires the input of a free span geometry. The idea is that this geometry would be obtained from a beam element model (ABAQUS, RIFLEX, ORCAFLEX and so forth). The above-mentioned FEM software is not well suited to include the torsion in the analysis. Hence Jordan estimates the torsion from a geometry found ignoring torsion, which is a source of error.

The present publication was delayed by several years due to confidentiality commitments. In the mean time, an in-house beam element software, LORO, using the present model to estimate the torsion, and accounting for the torsion in the force equilibrium that leads to the geometry, was developed by Longva and Sævik [9][10]. This required the introduction of a specialised solution with a meshed that does not move with the pay-out of the flexible.

Uncertainties on friction

One of the major unknowns in this analysis is the contact pressures between the layers that need to be used. In fatigue analysis of flexibles under installation offshore, or under operation, this contact pressure is dominated by tension in the flexible (for flexibles with an internal pressure: by the wall tension).

Three mechanisms to that contribute to friction in flexibles under low tension have been considered:

1. Tendons in the tensile armour are plastically deformed into spirals under production. If the curvature is not perfectly suited to the geometry, the spirals may be constricting the "payload" of the flexible. Simple computations show that the contact pressures this could induce are very low.
2. As written above, polymer layers, for example a medium-density polyethylene (MDPE) outer sheath are subjected to thermal expansion. They are extruded around inner layers, and shrink while cooling. They may relax their stresses in storage, and again shrink and expand under diurnal temperature variations. The contact pressures this can develop are more significant, of the order of magnitude of one MPa.
3. Speculatively, it might be important to account for an amplification effect: because for friction from sheath shrinkage, curvature cause a little tension in tendons – which increases contact pressure and friction.

Finally, in the cases that were investigated, the changes in curvature plane occurred over several pitch length of the tensile armour. For some products with particularly long pitch lengths (AKS umbilicals, for example) this assumption, used in the above calculations would not hold.

Steady state analysis of torsion

Observations and the proposed mechanism indicate that we are dealing with a "torque saturation" phenomenon: the roll increases until the torque developed in the free span is in equilibrium with the torque of the pipe under torsion on the ship's deck. Hence, a conservative approach to evaluate an operation is to compare the torque developed in the free span *at steady state* (roll rate equal to zero), with the torque that the pipeline can tolerate.

The boundary condition in such an analysis would be fixed roll degree of freedom at top of the free span and zero torque at the bottom (since, at steady state, in the absence of roll, the flexible laid in the turn table is not twisted).

The analysis can be carried out assuming a span shape (as in the Jordan model), or better, making no such assumption and determining it by requiring equilibrium – including the effect of friction-induced torque (as in the LORO model).

MITIGATION STRATEGIES

If, as it seems, roll build-up is related to the shape of the free span, in a way that can be quantified, then this should allow to control the shape of the free span to limit or cancel torque.

A preliminary remark is that it is not necessary to eliminate torque completely. Torque must just be kept within what the pipe can handle without damage and without spiralling.

Based on the mechanism described above, three possible strategies present themselves. Each strategy by itself should allow to control roll, but nothing prevents to combine them. The third strategy may be the most practical.

1. Keep curvatures small, because the torque is linearly proportional to the curvature. While this is successfully applied on land, this may not be practical when handling large diameter flexibles in the confine of a ship hold.
2. Change the curvature's intensity, not its plane, because this keeps the triangle in Figure 10 flat. Concretely, this means having bends that are in a single plane each, separated by straight segments of pipe.
3. If the plane of curvature must be changed without straightening the pipe "then 'twere well it were done quickly". Changing the curvature plane over a short segment of pipe limits the "number" of triangles with large areas in Figure 10. This may be unintuitive, but *a "natural" looking, slow change of curvature plane in the*

sagbend-to-coil area is the worst possible scenario. A flexible traveling through a spiral is an optimal torque engine, and indeed, pigtails once formed, rather travel along with the pipe.

Roller pair

Point 3 above suggests a way of steering the pipe in "distinct curvatures". Figure 12 two bends, one in a vertical plane, one in a horizontal one. In the vertical bend, the internal moment is a vector of direction 2. In the horizontal bend, the internal moment is a vector of direction 3. Hence, in order to have equilibrium in the geometry described above, it is necessary to apply an external couple at the transition point between both bends. This can be achieved by means of two rollers as shown in the figure. Alternatively the rollers can be replaced by caterpillar belts to keep the contact pressures on the pipe within the acceptable.

It must be noted that the present design will cause abrupt changes of curvature plane, while it is inspired by a theory that has been developed assuming slow changes in curvature planes. One would also need to consider the effect of tension, and stability issues. Generally speaking this idea would need to be developed with due caution.

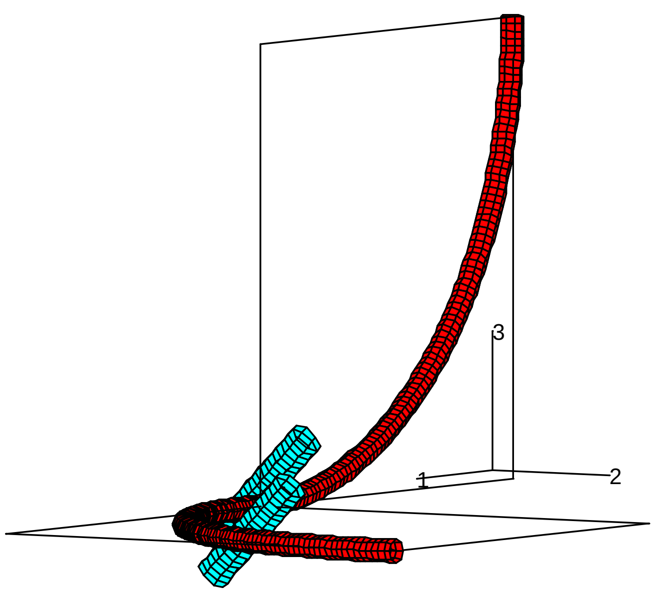

Figure 12 Two rollers, a short distance along the flexible to each other, to transform a vertical into a horizontal curvature over a short distance

CONCLUSIONS

When dealing with loadout torsion, it is important to distinguish between roll, roll rate, link, twist, writhe and torque, just as one must distinguish between displacements, velocities, deformation tensors, strains and stresses in continuum mechanics.

The most unfamiliar concept is arguably the *writhe*. A spacecraft could be given the following sequence of quarter turns: yaw left, pitch down, and yaw right. At the end of the sequence, the axis of the craft is pointing towards the same star as before. The sky appears to have turned clockwise around this axis *yet* the craft never rolled: it writhed.

Torsion induced failures are not always recognized at such, and this could be contributing to an underreporting of such issues.

Relevant design codes do not cover these failure modes, and there are reasons to believe that failure may occur at level of torque below what may be calculated as safe according to API 17B.

Internal friction, in flexibles that are paid out along a route with change of curvature plan in a curved section, causes torque. A model for the evaluation of this torque is presented. The model agrees with observations made during load out operations, and it provides several important insights in how the shape of, for example, a free span, affects the level of torque.

The existence of such models opens for the creation of codified guidance on the handling of flexibles, of software allowing to simulate an operation before the operation is undertaken, or before installations are built. It also opens for the creation of real-time simulator that would allow to train operators and provide them with insight to effectively operate existing installations.

The paper discussed some strategies to control load out torsion. While development work will still be needed, there is good reason to hope that in the near future, models of loadout torsion will be used, but in design rather than forensic analysis.

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given by the crew of an installation vessel: I was welcomed on board to "Oh, so *you* are Dr. Twist".

Congratulations to Prof. Torgeir Moan, who can look back on an amazing career that shaped a whole research environment in Trondheim (and beyond). He lectured me in several courses, way back in the early nineties, and he was a rigorous teacher. Among other things, his teachings in the use of probabilistic methods, including sampling methods and distribution updating made a lasting impression on me.

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