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Instability of mechanically lined pipelines under large deformation

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ABSTRACT

Mechanically lined pipe consists of a load bearing carbon steel outer pipe with a thin corrosion resistant alloy (CRA) liner and is a proven and economical solution for oil and gas pipelines. The bond between the two pipes is purely mechanical and under large bending loads plastic deformation of the liner results in non-reversible liner wrinkling. This wrinkling impedes flow and significantly reduces the performance of the pipeline. This study uses non-linear finite element techniques to describe the wrinkling of lined pipe under pure bending and examines the effect of initial material stresses and contact surface friction. This model is then used to test a design failure criterion equation of DNV GL for design of mechanically lined pipelines. In single directional bending to failure the DNV GL failure criterion proves to be conservative in predicting the onset of liner wrinkling for all initial conditions and pipe geometries tested. Under low cycle repeated loading the safety factor in the predicted DNV GL failure curvature is sensitive to the magnitude of initial pre-stress. The results presented herein provide evidence that the magnitude of initial material pre-stress in mechanically lined pipe needs to be considered in designing the CRA lined pipeline even at relatively low curvatures.

1. Introduction

Due to the often corrosive nature of the fluid transported by oil and gas pipelines it is essential to protect the internal surface of the pipelines against corrosion to ensure their structural integrity remains for the design life of the project. Constructing pipelines of sufficient mechanical strength from corrosion resistant materials however is prohibitively expensive. A more economical solution is to use a mechanically lined pipe, consisting of a load bearing carbon steel outer pipe that provides the required structural properties with a thin corrosion resistant alloy (CRA) liner to protect the pipelines internal surface [1]. Mechanically lined pipes are manufactured through a hydraulic expansion process centred around expanding and relaxing the CRA liner pipe inside the outer and relying on the differential elastic contraction of the two pipe materials to provide an interference fit. The magnitude of the mechanical bond, or ”gripping strength”, is a function of the residual hoop and radial stresses induced through the manufacturing process and the surface interactions between the liner and outer pipe [2].

The bond between the two pipes is purely mechanical with no metallurgical bonding between the two materials. As a result of the nature of this bond, during installation and operation it is possible for the liner and backing steel to move independently or become detached [3]. As the pipeline undergoes bending, plastic deformation of the liner can result in non-reversible wrinkling in the liner that can impede fluid flow and significantly reduce the pigging and fatigue performance of the pipeline [4]. Furthermore, this liner wrinkling can occur without any visible damage to the pipe outer making in-field detection difficult [5].

Given the economic benefits of mechanically lined pipe over clad pipe, there has been a concerted effort focussed on describing the bifurcation of the CRA liner experimentally [6–8] and in more recent years modelling the pipeline using numerical methods [1,3,5,9,10]. Together, this body of work has identified the mode of failure for mechanically lined pipe. As bending is induced in the pipeline the section of the liner under axial compression begins to plastically deform before detaching from the outer pipe. Further application of bending loads induces ovalisation of the cross section of the entire structure followed by bifurcation and complete collapse [11]. This failure mode has been repeated experimentally [6,7] and successfully replicated using finite element models [3,5,7,10].

Recent numerical studies have enabled parametric analyses of the pipeline geometry, manufacturing steps and residual material stress conditions to examine their effect on pipeline bifurcation and liner

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wrinkling. Attempts have been made to identify the parameters responsible for delaying the onset of liner wrinkling and maximising the moment carrying capacity of the pipeline [3,5]. However at present there is still no formal consensus for the "as delivered" condition for mechanically lined pipe. Indeed the international standards for subsea pipelines, most notably API 5LD and DNV-OS-F101, state that the degree of mechanical bonding strength or gripping force required in mechanically bonded line pipe is a matter of agreement between supplier and customer [12,13]. There are no formal requirements for the magnitude of this gripping force. The test regimes required by API 5LD and DNV-OS-F101 are intended only to act as a quality control measure and are not intended to accurately quantify the gripping force [14]. As a result the values for the experimentally determined initial pre-stress published in the literature span a considerable range.

The main objective of the previous work has been to describe the bifurcation of the pipe through single directional bending to failure and explore methods for delaying this failure. To date, there has been less of a focus on the effect of loading and unloading on the response of the pipe liner at lower curvatures. The pipelines are most susceptible to over bending and permanent deformation during installation on the sea bed. The most economical method for laying sub-sea pipeline of small diameter is the reel-lay method. Reeling involves welding pipe sections together and then spooling the pipe onto a large reel on the installation vessel. To lay the pipeline, the pipe is taken off the reel, straightened and then allowed to rest on the sea bed as the installation vessel moves forward [15]. This reeling and subsequent straightening places repeated bending strains of substantial magnitude on the pipeline that induces plastic deformation in the pipe liner and outer making the pipeline vulnerable to liner wrinkling and wrinkle fatigue.

The purpose of this present study is to examine the appropriateness of an empirically derived prediction criterion for the onset of non-reversible liner wrinkling recommended by DNV GL design guidelines [4]. Therefore, the objectives of this study were to (i) use the finite element method to describe the bifurcation of mechanically lined pipe under pure bending (ii) examine the effect of initial conditions on the moment-curvature response of the pipeline and the onset of liner wrinkling (iii) predict the curvature for the onset of liner wrinkling for different pipeline geometries and compare with an empirically determined failure prediction criterion, and finally (iv) predict the behaviour of the liner in response to repeated sub maximal bending loads to ensure the failure criterion is sufficiently conservative for repeated loading.

2. Constitutive modelling of cyclic behaviour

The constitutive model used in this paper uses the linear additive decomposition of strain into reversible, irreversible isotropic and irreversible kinetic hardening components. Although inspired from Karrech et al. [16], the current model relaxes the viscous and frictional effects by considering an associated elasto-plastic behaviour with von Mises plasticity potential. Therefore, the increment of strain is expressed as:

$$d\epsilon_{ij} = d\epsilon^{e}_{ij} + d\epsilon^{i}_{ij} + d\epsilon^{\kappa}_{ij} = d\epsilon^{e}_{ij} + d\epsilon^{p}_{ij} \tag{1}$$

The superscripts e, i, and κ denote the elastic isotropic, and kinetic strains respectively. We use a parameter M that describes the linear combination of strain such that $d\epsilon^{i}_{ij} = (1-M)d\epsilon^{p}_{ij}$ and $d\epsilon^{\kappa}_{ij} = Md\epsilon^{p}_{ij}$. Isotropic hardening involves the uniform expansion of the yield surface and kinetic hardening takes into account the shifting (translation) of the yield surface within the space of principal stresses. Combining these hardening mechanisms is preferred when cyclic loading phenomena such as the Bauschinger effect are relevant. The elastic behaviour is described by Hooke's law:

$$d\sigma_{ij} = C_{ijkl}\left(d\epsilon_{kl} - d\epsilon^{p}_{kl}\right) \tag{2}$$

where C_{ijkl} is the fourth order elastic tensor, ϵ_{ij} is the second order deformation tensor and σ_{ij} is the second order Cauchy stress tensors denoting respectively the strain and Cauchy stress. We use a von Mises limit of elasticity of the form $f(\hat{\sigma}_{ij}) = f(\sigma_{ij} - \alpha_{ij}) = \sqrt{3\hat{J}_2} = q$, where $\hat{J}_2 = \frac{1}{2}\hat{s}_{ij}\hat{s}_{ij}$ is the second invariant, $s_{ij} = \sigma_{ij} - p\delta_{ij}$ is the deviatoric stress, $p = \sigma_{ii}$ is the hydrostatic pressure, δ_{ij} is the Kronecker symbol, and $\hat{s}_{ij} = s_{ij} - \alpha_{ij} + \frac{1}{3}\alpha_{ii}\delta_{ij}$. Following Karrech et al. [16], we consider a back-stress of the form:

$$d\alpha_{ij} = c(N)d\epsilon^{\kappa}_{ij} \tag{3}$$

where $c(N)$ is a function of the number of loading cycles. The function $c(N)$ can be identified experimentally; in the current paper we consider it of the $c(N) = \frac{c_1}{1+c_2\ln\left(\frac{N}{\tau}+1\right)}$, given that we focus on a small number of cycles. The constants c_1, c_2, and τ can be seen as material properties. Since the material is associated, the flow rule can be defined by:

$$\dot{\epsilon}^{p}_{ij} = \dot{\lambda}\frac{\partial f}{\partial \sigma_{ij}} \tag{4}$$

where $\dot{\lambda}$ is a positive parameter. Using Hooke's law (2) and the flow law (4), it can be seen that

$$\Delta(\sigma_{ij})_{n+1} = C_{ijkl}\left(\Delta\epsilon_{kl} - \Delta\lambda\frac{\partial f}{\partial\sigma_{ij}} - \Delta\lambda\frac{\partial^2 f}{\partial\sigma_{ij}\partial\sigma_{kl}}\Delta(\sigma_{ij})_{n+1}\right) \tag{5}$$

By rearranging the terms, we deduce the Cauchy stress increment

$$\Delta(\sigma_{ij})_{n+1} = D_{ijkl}\left(\Delta\epsilon_{kl} - \Delta\lambda\frac{\partial f}{\partial\sigma_{ij}}\right) \tag{6}$$

where $D_{ijkl} = \left(C^{-1}_{ijkl} + \Delta\lambda\frac{\partial^2 f}{\partial\sigma_{ij}\partial\sigma_{kl}}\right)^{-1}$. This model was implemented using the Fortran user materials subroutine of Abaqus, UMAT.

3. Finite element model

The response of the mechanically lined pipe under pure bending loads was examined numerically using non-linear finite element modelling. The model was created and developed using ABAQUS finite element software [17]. The analyses considered non-linear geometry that involves the finite strain theory to describe the outer and liner pipes. These pipes were assigned elastic-plastic material properties with isometric hardening and plastic behaviour described through stress-strain curves as reported in the literature for commercially available lined pipe materials [7]. To enable the use of user defined pre-stress in the hoop, axial and radial directions the material orientations for the liner and outer pipe were assigned in a local cylindrical coordinate system.

The finite element model was constructed as a three dimensional model from solid, deformable, eight-noded elements (C3D8R), both for the liner an outer pipe with three elements in thickness and 120 elements around the half circumference. The meshing in the axial direction is described in Table 1 and Fig. 1. Following a short parametric study on the liner and outer pipe separately the model was set such that increasing the mesh density had little effect on the numerical results. This indicates the performance of the model in terms of convergence. The interaction between the liner and the outer pipe was modelled as finite sliding hard contact with the outer pipe as the master and the liner as the slave. Geometric symmetry of the pipe has been utilised

Table 1
Axial mesh density for pipe model regions showing number and length of elements in axial direction.

Region	L_{region} [mm]	$N_{element}$ [−]	$L_{element}$ [mm]
1	120	24	5
2	680	68	10