Offshore Mechanics and Arctic Engineering, OMAE '98 Lisbon, July 5-9, 1998.

PIPELINE BUCKLING, CORROSION AND LOW CYCLE FATIGUE

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ABSTRACT

Subsea pipelines operating at high temperature can buckle vertically and laterally. The curvature and high strains in buckled pipelines can cause ovalisation, wrinkling and fracture. Additionally, low cycle fatigue and ratchetting may result from cyclic operation. Various factors affect the levels of strain generated by thermal buckling. Two factors: local corrosion and pipeline constraint conditions are of particular concern.

This paper presents a non-linear Finite Element study to investigate the effect of local corrosion on the strains generated at the apex of a lateral buckle caused by thermal expansion.

The study considered a thick wall 6" pipeline, trenched but not buried. The analysis used the ABAQUS Finite Element package, and included an investigation of the sensitivity of results to a number of parameters.

The start-up and shutdown of a thermally buckled pipeline can lead to large variations in bending stress. Local corrosion can increase the bending strains at the apex of a buckle. The large variations in bending stresses may result in low fatigue lives. This analysis has demonstrated these features and shows that this type of behaviour can be effectively modelled using finite element methods.

BACKGROUND

Oil and gas production technology is developing rapidly. New technologies and cost reductions have made the exploitation of high temperature and high pressure reservoirs economically viable. In the drive to reduce costs pipelines that would once have been trenched and buried to protect against impact and prevent buckling may now only be trenched (giving impact protection but allowing buckling). If there are unexpected changes due to internal corrosion or altered seabed restraint conditions then axial strains that are higher than anticipated may be generated. To be able to competently assess the implications of large buckles on pipeline integrity, a good understanding is required of post buckling behaviour, the effects of corrosion and restraint conditions.

INTRODUCTION

A detailed non-linear Finite Element study has been undertaken to improve the understanding of the post buckling behaviour of a pipeline. The analysis used the ABAQUS Finite Element package. The study considered a 6" flowline, trenched but not buried. Surveys of similar pipelines have shown buckles of greater than 5m peak displacement. The aim of the study was to investigate the effect on the pipe wall stress and strain of local corrosion at the apex of a lateral buckle.

The finite element model was validated in the first instance, by comparison with the predictions of widely accepted pipeline buckling models described by Hobbs[1].

An investigation of the sensitivity of results to the following influencing parameters was carried out.

- axial and lateral friction co-efficients,
- friction mobilisation distance,
- material properties and
- initial out-of-straightness.

The model was then used to estimate the change to the strain conditions resulting from a short length of corrosion.

CYCLIC BUCKLING BEHAVIOUR

The forces acting on the buckle while the pipeline is operating are shown in Figure 1. A state of equilibrium exists between, the longitudinal force (which is driving the deflection of the pipeline) and the pipeline bending stiffness, axial friction and lateral frictional restraint (which are preventing further deflection of the buckle). In this state the majority of the pipeline will be in compression; the buckle apex however, will be in tension because of the applied bending moment.

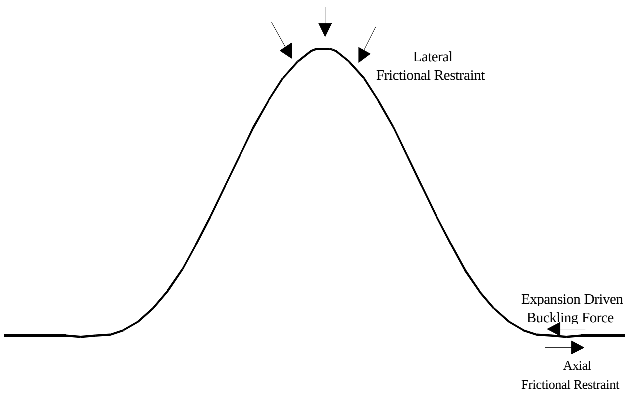


Figure 1 Forces acting on a buckle at high temperature

The axial force in the buckle decreases when the pipeline cools (unloads). Initially the buckle does not move, because the axial force and frictional resistance remain in balance, maintaining its position. At some point the axial force changes sign and the pipeline tries to contract, and the buckle begins to move inwards. At this point the lateral frictional forces change and act outwards, maintaining the deflection of the buckle. With a further reduction in the temperature, the contraction force increases and the lateral friction reaches its maximum sliding value. This outwards frictional force acts to maintain the deflected shape of the buckle. As shown in figure 2.

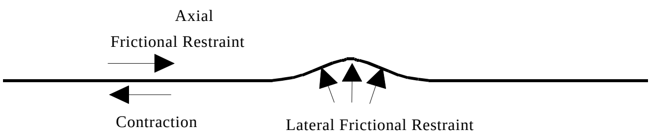


Figure 2 Forces acting on a buckle when cool

If the pipeline is depressurised and allowed to cool to ambient temperature then the buckle will still exist in the pipeline, although of much reduced amplitude. This residual deformation in the buckle is due to two distinct effects:

- Residual deformation of the pipe wall induced by plastic strain on loading;
- Friction forces acting to maintain the deflection of the buckle.

Both of these effects will lead to a tensile force in the pipeline, where the pipeline wants to contract and straighten but is prevented from doing so. This tensile force will act against the friction forces, and will provide a moment which will potentially cause a compressive stress at the buckle apex.

This residual deflection will act as an initial out-of-straightness feature for subsequent loading.

BUCKLING AND CORROSION

Internal corrosion will alter the buckling behaviour of a pipeline due to changes in the flexural rigidity, submerged weight and axial force. Each of these parameters is discussed briefly below, with respect to three different cases of internal corrosion damage: large scale uniform wall thinning (both axially and circumferentially), localised corrosion (e.g. axial grooves possibly created by mesa (CO_2) corrosion or circumferential preferential weld corrosion) and pitting. Only uniform wall thinning has been considered numerically.

Flexural rigidity

Large scale uniform corrosion around the circumference will reduce the flexural rigidity of the pipe. If the submerged weight and axial force remained unchanged (which is not the case), this would result in a concentration of the buckle feature. The buckle length would reduce as the curvature at the crown of the buckle increases, resulting in greater bending stresses/strains in the pipe wall.

Localised corrosion will reduce the flexural rigidity of the pipe to a lesser extent than large scale corrosion. If localised corrosion should occur at the apex of a buckle however, then only a small reduction in the flexural rigidity may lead to failure from local buckling.

Pitting corrosion will also reduce the flexural rigidity of the pipe, but to a lesser extent than either of the two previous cases.

Submerged weight

Large scale corrosion over the whole length of a pipeline will reduce the submerged weight of the pipeline. This will beneficially allow a buckle feature to increase in length due to the reduced lateral resistance. This effectively allows a buckle to ‘relax’, reducing the severity of the curvature at the crown of the buckle. This effect will be much less for localised or pitting corrosion.

Fully constrained axial force

Large scale uniform corrosion will reduce the cross sectional area of the pipe resulting in a drop in the fully constrained force (buckle driving force) of the flowline. This reduction will limit the lateral growth of a buckle feature. Localised or pitting corrosion however, will not significantly alter the axial force.

MODEL DEFINITION.

Large deformations may take place during pipeline buckling. Additionally high strains can be induced by the bending of a pipeline with a corroded section. To accurately model this behaviour, a non-linear analysis is required [3]. The ABAQUS [2] finite element program is one of the leading non-linear finite element codes and has therefore been used for this assessment.

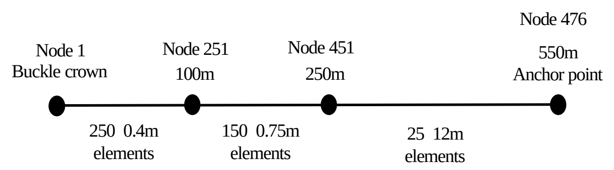
There are three components to the Finite Element model generated for this analysis. The first is the model of the pipe geometry. The second is a material model, and the third is the model of the interaction between the pipe and the seabed.

There are two main analysis tasks. The first is to validate the model and the second is to investigate the effects of localised corrosion.

To model the buckling behaviour of the flowline, a number of assumptions or simplifications have been made:

1. Stationary points, where the flowline does not move axially on heating exist between adjacent buckles.
2. The pipe lies in the bottom of a 'V' shaped trench with a 1:4 side slope.
3. The axial friction coefficient μ_a=0.5.
4. The 'effective' lateral friction coefficient up the trench side slope μ_u = 1.5.
5. The 'effective' lateral friction coefficient down the trench side slope μ_d = 0.5.
6. The buckle will be mode 1 and therefore symmetrical. Hence only one half (centred on the buckle apex) will need to be modelled.

The model consists of 476 nodes and 475 elements. The nodes define the element end points in space and the elements describe the pipe response to the applied loading. The arrangement of nodes and elements is shown schematically in the sketch below.



Model Length

At some location between adjacent buckles there will be a 'stationary point' (or anchor point) where the pipeline does not move axially on heating. Thus the length of pipe available to 'feed in' on expansion to any particular buckle will be the distance from the stationary point to the apex of the buckle. A model length of 500m was selected, equivalent to 1000m spacing between buckles. This is a relatively large spacing for buckles on an unburied pipeline, which may occur when there are local restraint conditions such as partial burial.

Element Selection

475 ABAQUS PIPE31 elements have been used. These are 2 node linear elements in space (3D rather than planar). Other elements are available, such as ELBOW31B, that can provide through thickness stress data, but these are computationally expensive and do not give direct curvature or bending moment information. Substituting ELBOW31B elements for PIPE31 elements in the model used generated the same displaced shapes and therefore, for the purposes of this study the PIPE31 elements were considered to be adequate. The element length is short (0.4m) in the region of the buckle to give improved definition of results in this area. Away from the buckle region the elements are longer (extending to 12m) to improve computer run time. A sensitivity analysis was performed to ensure that the model was stable for these element lengths.

Corrosion Modelling

The nominal wall thickness selected is 19.1mm. This is the value used over the whole pipe length for model validation. To investigate the effect of localised corrosion two depths of corrosion were modelled:

- 3mm deep, 16mm remaining wall thickness.
- 6mm deep, 13mm remaining wall thickness.

In both cases the corrosion is modelled as fully circumferential and the axial length is 4m. The corrosion was modelled by applying a reduced wall thickness definition to elements over a 2m length (4m by symmetry) at the apex of the buckle.

Fully circumferential corrosion is unusual in a pipeline, but it can be assumed to model trough-type corrosion in the base of the pipeline by the consideration of the second moment of inertia.

Pipe Material

Three material models were used; a fully elastic material, a model based on true stress - true strain data, and an idealised true stress-true strain model. The model based on true stress - true strain data has been taken from tests carried out on X65 linepipe. This is a lower bound curve since it is slightly below the standard specification. The idealised model is an approximation of X65 minimum specifications, with an SMYS of 448 MPa at 0.5% strain and a UTS (SMTS) of 530 MPa. The material models used have not taken account of the elevated temperatures. For the temperature range under consideration it is not anticipated that this would have a significant effect on the material properties. All of the material models include a thermal expansion co-efficient to allow the application of the thermal buckling load.

Seabed/Trench Interaction Model

The trench has been modelled as a 'V' shape. The trench applies a frictional restraint to the pipe, which varies depending on the direction of movement of the pipe. For this analysis the axial coefficient of friction is taken as 0.5. This is a standard figure for a sandy/silty seabed.

The lateral coefficient of friction on a flat seabed is higher than the axial coefficient, due to embeddment and the build-up of a bank of sand and silt ('berm') along the pipe as it moves. Lateral friction coefficients of 0.7 to 0.9 are used for sandy/silty seabeds, depending on the level of embedding of the pipe. In this case the pipe is moving up a slope. Therefore the restraint will be higher due to greater 'berm' build-up, and gravitational restraint. The effective coefficient of friction will therefore be greater.

For this analysis values of 1, 1.5 and 2 were considered. An up-slope friction coefficient of 1.5 has been found to give a good approximation to surveyed pipeline positions.

The down slope frictional restraint will be lower than that for a pipeline on a flat seabed. A figure of 0.5 has been chosen for this analysis. The sensitivity of the results to changes in the downslope friction co-efficient has been investigated.

Breakout behaviour (where initial restraint is higher but then drops off as the pipe starts to move) has not been modelled. It is more relevant to harder clay like soils, rather than soft sandy/silty conditions.

In the Finite Element model the frictional restraint of the trench on the pipe has been defined using several sets of non-linear springs. This is a common method of approximating the frictional restraint of the seabed on a pipe. The springs are defined such that they will allow a small amount of movement before applying a constant restraining force (F_f) to each element calculated as follows:

$$F_f = \mu \times W \times l_e$$

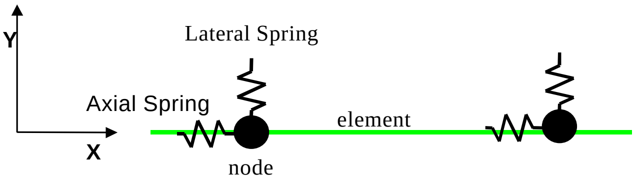
Where

μ = the effective coefficient of friction

W = the submerged weight per unit length of the flowline

l_e = the length of the element (distance between springs)

The arrangement of the springs relative to the nodes is shown in the sketch below:



Lateral or upheaval buckles usually occur where there is an initial defect or out-of-straightness. Within the analysis this has been created by the use of a gap contact element, which acts as a point support.

Loading

Two loads have been applied:

- Pressure (270 bar)
- Temperature (85°C)

These loads represent typical flowline operating conditions. The pressure load is applied first and represents the difference between the external pressure on the pipeline and the internal pressure of the fluid. The temperature load is applied after the pressure to mimic the steady heating of a pipe on startup. The temperature load causes the expansion of the model. This expansion provides the buckle driving load. For modelling cyclic operation, on reaching the peak temperature, the pressure is reduced to ambient and subsequently the temperature reduced. To reload, the pressure is re-applied, followed by the temperature.

MODEL VALIDATION

The model was validated by comparison with the predictions of elastic beam equations (after Hobbs[1]). The displaced shape plots shown in Figure 3 demonstrate that the analysis is broadly in agreement with the results of previous work. Similar models have also been used in the assessment of operational pipeline buckles and have been shown to give a good approximation to the surveyed displaced shapes.

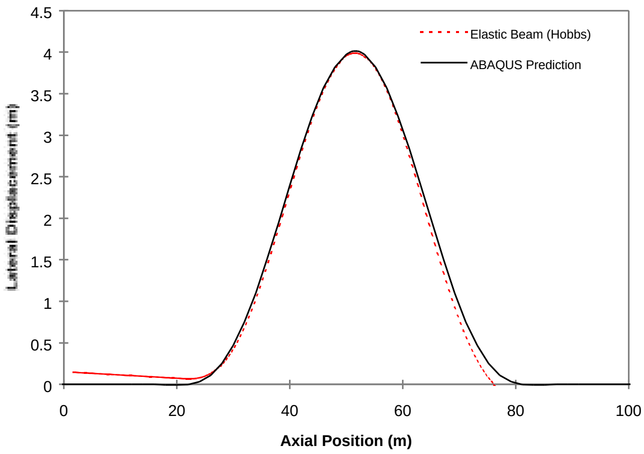


Figure 3 Displaced Shapes by FE and Elastic Beam Anlysis

MODEL SENSITIVITY

Initial Out-of-Straightness

Initial out-of-straightness' of 200mm, 300mm and 500mm were considered. For each of these cases the lateral displacement is plotted against temperature in Figure 4 and it can be seen that the results converge quickly. Initial out-of-straightness therefore, does not have a significant effect on the buckling behaviour.

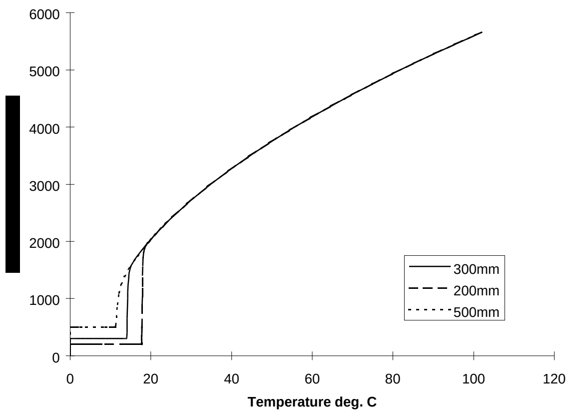


Figure 4 Effect of Initial Out-of-Straightness

Material Model

Three different material models have been considered: a fully elastic material, a material based on true stress-true strain data for an X65 sample and an idealised fit to minimum X65 specifications. The bending moment against curvature behaviour for each of these models has been plotted in Figure 5. The idealised model was used in all of the other analyses presented here.

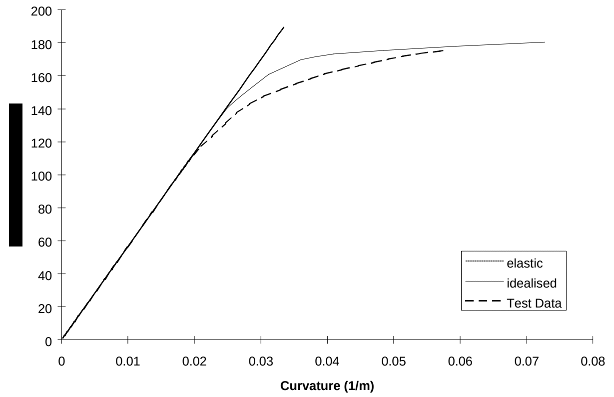


Figure 5 Material Model Differences

Element Density

To check that a sufficient number of elements are being used to accurately predict the behaviour a model has been generated with the number of elements halved. The bending moment against curvature behaviour for the full model and the reduced element model is plotted in Figure 6. Reducing the element density has no effect on the results and hence we can be confident that enough elements are being used to accurately predict the system response.

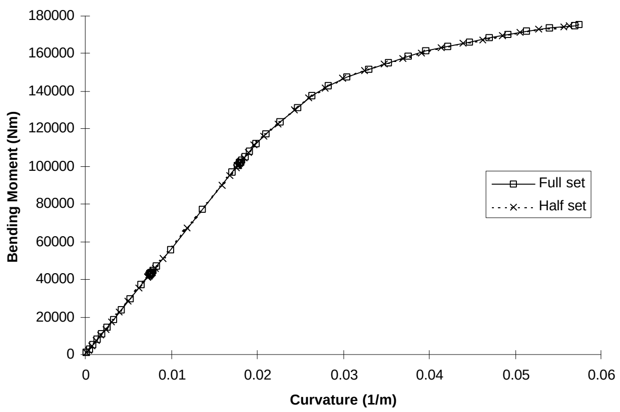


Figure 6 Effect of Element Density

MODEL FRICTION SENSITIVITY

Friction between the pipe and the sea bed affects the axial movement of the flow line, the up slope lateral movement and the down slope lateral movement. The up slope and down slope restraints are different due to the effect of sand building up against the pipe as it moves up-hill, and the combination of gravitational and frictional restraint.

The frictional constraint is applied using spring elements. The definition of these elements requires a certain amount of movement before the full force can be applied. This is known as the mobilisation distance. If the mobilisation distance is set too high then the restraint will not be realistic. If it is set too low then the model may be overconstrained and solving can be difficult. The mobilisation distance will not tend to have a significant effect on the stresses generated in the initial loading but may become significant if cyclic operation is being modelled, due to the small axial movements and changes in stress conditions on load reversal.

Sensitivity to friction coefficient

A 'base case' model has been selected with friction coefficients corresponding to the best estimates made for the FE analysis. The base case model from which the changes in behaviour are referenced has the following frictional coefficients.

Axial friction coefficient = 0.5

Lateral friction coefficient (down slope) = 0.5

Lateral friction coefficient (up slope) = 1.5

The results of the sensitivity study are shown in Table 1.

Factor	Frictional Coefficients			Effect On Model
	Axial	Lateral Down Slope	Lateral Up Slope	
Base case	**0.5**	**0.5**	**1.5**	
Axial	**0.3**	0.5	1.5	no influence
	0.7	0.5	1.5	no influence
Lateral Down	0.5	**0.3**	1.5	no influence
	0.5	**0.7**	1.5	no influence
Lateral Up	0.5	0.5	**1**	lower curvature
	0.5	0.5	**2**	higher curvature

Table 1 Results of Friction Sensitivity Study

The only factor to have any significant effect on the buckling behaviour as modelled is the coefficient of lateral friction up the slope. The value of 1.5 was selected on the basis that it was a reasonable, conservative, figure for a pipeline in a trench on a sandy/silty seabed with some natural burial.

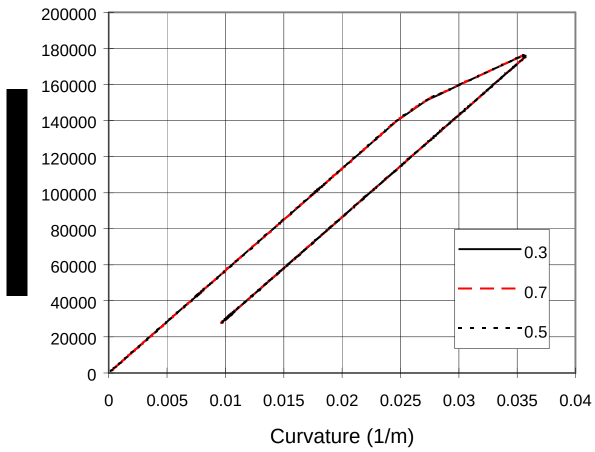


Figure 7 Bending moment against curvature at the buckle apex by downslope friction coefficient

As shown in Figure 7 the down slope friction coefficient has no effect on the bending moment curvature relationship. The range of effective friction coefficients considered, 0.3, 0.5 and 0.7, is conservative. Any level of embedding will quickly lead to a friction coefficient of 1 or more. In the absence of cold survey data with which to validate these figures, a conservative value must be used. In all cases the pipe returned to the original 'straight' position on cooling to ambient (300mm initial displacement).

The axial friction coefficient has a very minor influence on the thermal buckling behaviour. This may be explained by the relatively light weight of this pipeline (the submerged weight is around 45 kg/m) and the relatively short length under consideration (1000m between anchor points).

Sensitivity to Friction Mobilisation length

Friction mobilisation lengths of 10mm, 5mm, 1mm, 0.1mm and 0.01mm were considered. There was only a small effect on the displacement and stress at the buckle apex. Remote from the apex (where the movement is axial and very small) a significant difference was seen, with the lower mobilisation lengths giving a more realistic sectional force profile along the pipeline than the higher values. The value of 0.1 was selected as the optimum; below this level the model became over constrained and would not converge reliably.

EFFECT OF CORROSION

Three different levels of corrosion have been considered. All corrosion is considered to be fully circumferential along a 4m section of the pipe and is modelled as a locally reduced wall thickness. A temperature increase of 85°C and a pressure of 270 bar is used in all cases.

1. No Corrosion.
2. 16mm wall thickness along a 4m section at the apex of the buckle.
3. 13mm wall thickness along a 4m section at the apex of the buckle.

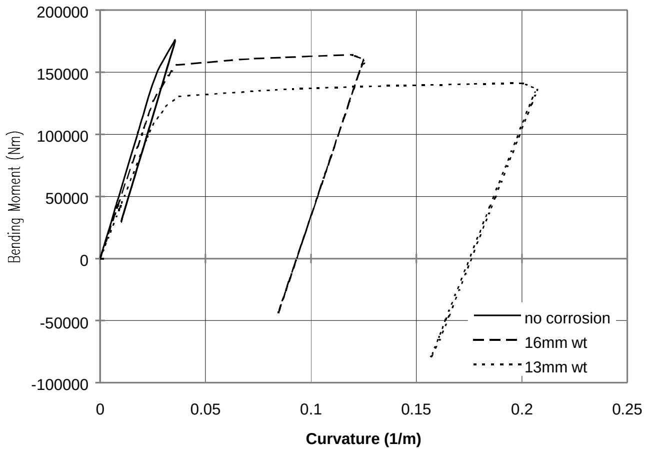


Figure 8 Bending moment curvature at the buckle apex for differing wall thicknesses

Figure 8 shows the bending moment curvature relationship for cases 1, 2 and 3.

Discussion

Corrosion causes the strain at the buckle apex to increase. The results of each case are considered below.

1. No Corrosion: The pipe just exceeds yield.
2. 16mm wall thickness by 4m length: The reduction in wall thickness over a relatively short length has resulted in yielding at a lower applied bending moment, due to the reduction in flexural rigidity and the reduced load capacity of the thinned region. A significant permanent curvature is also generated.
3. 13mm wall thickness by 4m length: A further reduction in wall thickness gives an even lower yield threshold and an even higher curvature. The onset of ratcheting is also evident.

The onset of ratcheting can be seen in Figure 8. For the uncorroded and the 16mm wall thickness case, following the initial loading the unloading and reloading paths are coincident. For the 13mm wall thickness case however, the unloading and subsequent reload path are slightly separated, indicating some plastic deformation at the end of the unloading cycle. This shows the onset of ratcheting. The maximum increase in strain found for this analysis was 1.17×10^{-5}. It is expected that this cyclic increase in strain would 'shake down' after a number of cycles [4]. For wall thicknesses lower than 13mm, ratcheting can be expected to become increasingly severe.

To investigate the effect of cyclic operation of a pipeline with a severe buckle, case 2 (3mm of corrosion, 16mm wall thickness) has been used. The resulting stress-temperature plot is shown in Figure 9.

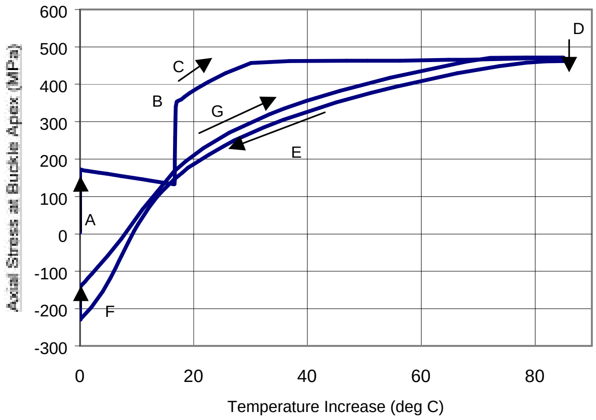


Figure 9 Stress - Temperature Relationship (stress at the buckle apex on the outside of the curve in the 16mm section)

Figure 9 shows the stress caused by the initial displacement (A in the figure) and that the pipeline buckles at a temperature increase of approximately 17°C (point B). At 30°C the pipeline yields and starts to permanently deform. After reaching the peak temperature the pressure is reduced to ambient (point D), leading to a reduction in stress. As the temperature reduces the stress relaxes. At approximately 10°C above ambient the tension at the buckle apex changes to compression. This occurs because the pipeline is trying to straighten out the curvature set into the pipe by the plastic deformation. If the pipeline follows the cycle F, C, D, E on startup and shutdown, the resulting stress range will be 700 MPa (–229 MPa to 471 MPa). The cyclic behaviour described here is fully elastic, following the plastic deformation resulting from the first cycle. The axial stress range at the buckle apex resulting from the buckling of the uncorroded pipeline was also calculated and is 483 MPa. The axial stress range that would be caused by variations in the internal pressure alone is omly 36 MPa.

The fatigue implications of cyclic operation at these stress ranges can be illustrated by plotting them on an appropriate S-N curve. The stress range has been used directly because the cycle is fully elastic. If plastic deformations were generated in every cycle then the strains would have to be used to derive an equivalent stress range. An S-N curve, two standard deviations below the mean, for parent plate in seawater with CP protection has been selected [5]. This curve is plotted in Figure 11; the stress ranges resulting from the Finite Element analysis are also marked.

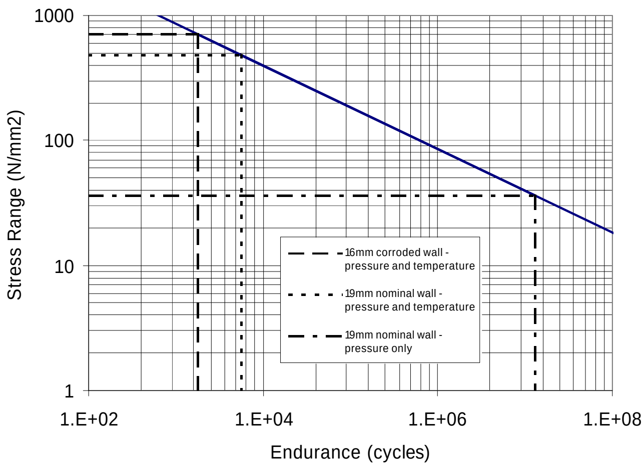


Figure 10 Fatigue life of a 6" pipeline, with buckling and buckling with corrosion

These results are summarised in table 2.

Wall thickness	Internal pressure and temperature	Stress range induced MPa	Cycles to endurance limit
19	270 Bar	36	> 1 million
19	270 Bar + 85°C	479	5500
16	270 Bar + 85°C	700	1773

Table 2 Cycles to fatigue endurance limit

For the corroded pipe (16mm wall thickness) the endurance limit will be reached after 1773 cycles. A fatigue life usage factor of 0.1 is generally considered acceptable in design [6]. For the pipeline considered in this study, with 3mm of corrosion, the design code could be exceeded in 177 cycles. For a typical flowline, one shut down every 2 weeks would be reasonable. This would give a design fatigue life of 7 years as opposed to a design fatigue life of 26 years with no corrosion. If there were a weld or other stress concentrating feature at the apex of the buckle, then the predicted fatigue life would be much shorter.

CONCLUSIONS

1. The detrimental effect of corrosion on a pipeline subjected to post-yield thermal buckling has been quantified using finite element methods.

2. The cyclic operation of pipelines that have been subjected to post –yield buckling can lead to low fatigue lives. For the case considered here, 3mm of corrosion (16mm wall thickness) reduces the fatigue life from 26 years to 7 years.

ACKNOWLEDGMENTS

The authors would like to acknowlege Andew Palmer and Associates for permission to publish this paper and acknowlege the contributions of Phil Hopkins, Malcom Carr, David Kaye and James Ingram

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