

Finite element modelling techniques and testing methods of submerged pipes

A. Giordano ¹, A.C. Walker ² and F. Guarracino ¹

¹ Dipartimento di Ingegneria Strutturale, Università di Napoli "Federico II"- ITALY

² Dept of Civil & Municipal Engineering, University College London - UK.

Abstract: *The purpose of the present work is to discuss some FEM procedures and experimental methods that are currently used in the pipeline industry and open the way to the possibility of developing new experimental apparatuses which can provide much more economical alternatives to traditional design codes and tests.*

Keywords: *Pipe, Pipeline, Collapse, FEM, Testing Method*

1. Introduction

Pipelines are used worldwide, onshore and offshore, and have now become vital components in the energy systems of all economically developed countries. Pipelines are designed to accommodate the effects of a wide range of loading conditions resulting from internal and external pressure, bending, etc. during installation and operations. The design calculations for pipelines are aimed at providing a safe, robust pipeline with an economical use of expensive material and installation equipment. Pipeline design calculations have traditionally been based on a limiting stress approach but since 1996 a limit state code has been developed. The use of the limit state approach provides a more comprehensive basis for the calculation of the ultimate conditions for pipes subjected simultaneously to pressure and bending loads. The ultimate state of the pipeline deformation or loading is calculated using a model that describes the characteristic ultimate moment or strain related to the geometry and material properties of the pipe. The design factors are calculated using statistical descriptions of the scatter of test results compared to the mean values together with the statistical descriptions of the variables composing the particular model, e.g. material strength, modulus etc. In the process described above, it is generally assumed that the scatter of tests results from minor and usually random variations in the variables included in the model. In the case of a pipe, these variations would generally relate to the differences in the geometries of the test pipes from their corresponding nominal values, say for pipe wall thickness, or out-of-roundness.

The pipelines are typically installed empty, i.e. filled with air at ambient pressure and only filled with oil or gas under pressure once installation is completed. A major risk experienced during the installation of these deep-water pipelines is from the pressure applied by the water causing the pipe to deform out of its initial round shape and deform into an almost flat configuration. This is called external pressure collapse and if not controlled can result in the total loss of the pipeline. The dimensions, i.e. diameter and wall thickness, and to a lesser degree the material properties of a very deep-water pipeline are therefore determined by the potential for external pressure collapse. It is evident that the wall thickness is the most relevant factor in the pipeline's capacity to sustain the loads imposed during installation and under operating conditions, as well as a considerable factor affecting pipeline costs. Minimum wall thickness requirements are strictly linked to the material specifications. Also, it is well known that two additional parameters are very important for the cross section capacity to withstand the external pressure: the cross section ovality (maximum value of fabrication ovality allowed by DNV OS-F101 is 1.0 and 1.5% at pipe ends and body, respectively) and the specified minimum yield stress through the pipe wall thickness. The increase in compressive yield strength in the circumferential direction and the reduction of the maximum cross section ovality will allow reducing the required steel wall thickness. The purpose of the present work is to revise some methods that are currently used in the industry and illustrate the drawbacks from some experimental procedures.

2. Theoretical background

At the beginning of the 1990s the offshore pipeline industry, in conjunction with regulatory authorities in UK and Norway, started to revise design guidelines for offshore pipelines. In fact, design guidelines in force at that time did not account for modern fabrication technology. A project called SUPERB was thus aimed to develop a SUBmarine Pipeline Reliability Based design guideline, together with a comprehensive set of recommendations and criteria for different load conditions. The guideline included the so-called limit state design approach, with safety factors defined using structural reliability methods.

The developed design guideline was incorporated in the DNV standards Submarine Pipeline Systems, DNV OS F101. The collapse resistance of the pipeline is calculated using the following equation

$$\left(\frac{P_{c,d}}{P_{el,d}} - 1 \right) \left[\left(\frac{P_{c,d}}{P_{y,d}} \right)^2 - 1 \right] = f_{0,d} \frac{P_{c,d}}{P_{y,d}} \frac{D_0}{t} \quad (1)$$

where D_0 is the nominal outer steel diameter, t is the nominal steel wall thickness and $f_{0,d}$ is the pipe initial ovality (not less than 0.5%),

$$f_{0,d} = \frac{(D_{\max} - D_{\min})}{D_0} \quad (2)$$

with $D_{\max}$ and $D_{\min}$ being respectively the maximum and minimum outer diameter. The design elastic collapse pressure is given by

$$p_{el,d} = \frac{2E}{1-\nu^2} \left(\frac{t}{D_0} \right)^3 \quad (3)$$

while the design yield pressure is

$$p_{y,d} = 2 SMYS \alpha_{fab} \alpha_U \frac{t}{D_0} \quad (4)$$

E is the Young modulus, ν is the Poisson's ratio and $SMYS$ is the specified minimum yield strength. The factor α_{fab} considers the effect of the fabrication process, which introduces different strength in tension and compression along the circumferential direction of the pipeline, due to cold deformations (Bauschinger effect). The factor α_U takes into account the different material qualification.

3. FEM analyses

In the case of bending, special attention must be paid in modeling the ends of the tubes being analyzed. As shown in previous works, particular boundary conditions can trigger localized effects which may influence the response of thin-walled pipes.

Mainly, the end of the pipe can be considered as restrained against ovalization or free of ovalising. These two conditions are relevant, for instance, to the presence or absence of special structural details such as stiffening rings that are commonly employed in submerged pipelines as buckling arrestors.

3.1 Restrained ovalization

This boundary condition can be simulated by constraining the displacements of the tube end nodes to those of a single node, placed at the centroid of the end section. Within the adopted computer program, ABAQUS® v.6.5, this is done using the *MPC BEAM type option. Actually, MPC type BEAM provides a rigid connection between two nodes, or one node set and a node, to constrain the displacements and rotations at the first node, or node set, to the displacements and rotations at the second one, which acts as reference node. This corresponds to the presence of a rigid beam between the joints involved. The MPC type BEAM constraint does not apply linearization of the displacements, so that it can be effectively used in cases where geometrically non-linear behaviour must be taken into account. Figure 1 graphically shows the above described constraint.

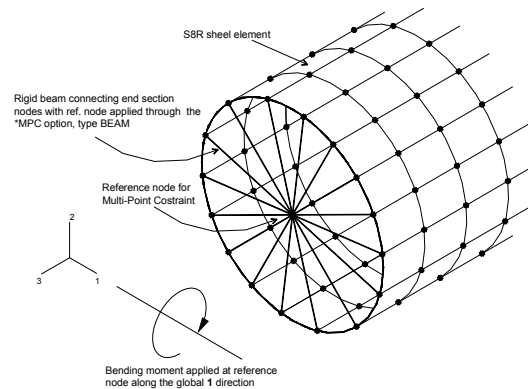


Figure 1 – kinematic constraint to enforce restrained ovalization

The bending action can be applied directly to the reference nodes as concentrated couples. In the case of the analyzed tubes, one end reference node was pinned and the other was left free of translating along the axial direction of the cylinder, in order to avoid interaction of axial strains. Similar results are obtained providing full restraint to one end node and applying the bending action to the other one. Figure 2 shows the deformed shape and longitudinal stresses distribution of a typical example, having a diameter/thickness ratio of 40, which is fairly common within the category of pipes usually employed in submerged pipelines. In such cases, a remarkable interaction between geometrical and material nonlinearities takes place. As a matter of fact, plastic deformations localize in the region where also the pipe buckles with subsequent severe distortion of the cross section shape. The generalized force-displacement curve appears as shown in the following figure 3, where the start point of the descending branch is associated with the onset of the local buckling phenomenon.

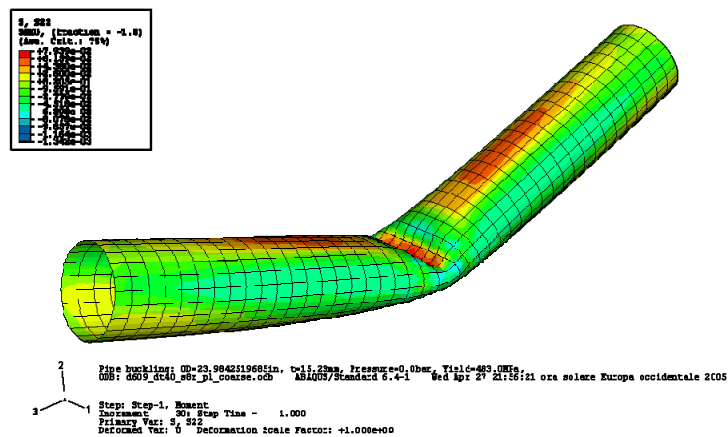


Figure 2 – Deformed shape of a bent tube modeled with restrained ovalisation

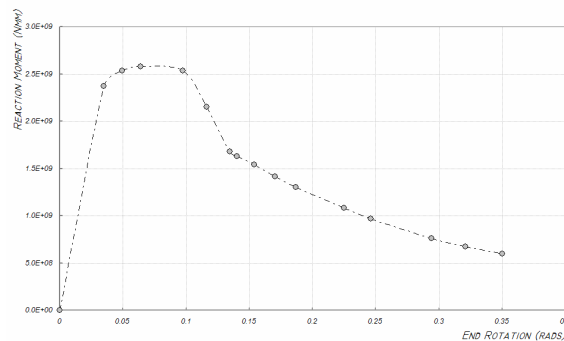


Figure 3 - End rotation vs. applied action curve for a D/t =40 pipe

As the bending process evolves, the prescribed boundary condition induces an unsymmetrical stress distribution. Actually, theoretical values of the longitudinal stress are equal at the compression and tensile sides, whether the finite element results show up to 30% differences. In figure 4, axial stresses acting on a mid-section ring are pictured, showing, in two subsequent analysis steps, how the afore mentioned asymmetry increases.

From the FE analyses on several specimens, it has been found that the onset of longitudinal wrinkles on the compression side, which can be considered as a physical evidence of stress asymmetry, and their effect on the pipe behavior strongly depend on the ratio of the diameter to the thickness. As a matter of fact, while for small thickness values longitudinal ripples localize and

buckling takes place, for relatively thick shells the growth of ripples on compression side generally results in a softening effect over the generalized moment-curvature response.

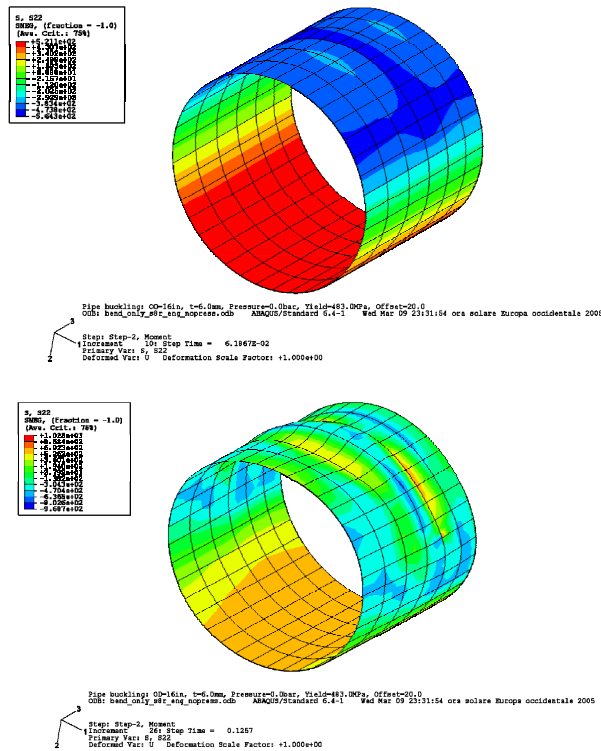


Figure 4 – Longitudinal stresses asymmetry due to restrained ovalization

Accordingly to these considerations, Figure 5 comparatively shows the deformed shapes of two different specimens, along with the corresponding moment-rotation curves.

In some cases, local buckling can manifest itself in more than one region. Such circumstance is likely to occur for very small thickness values, such as in figure 6, where a $D/t=60$ pipe is pictured.

This effect is mainly driven by the characteristic wavelength of the longitudinal wrinkles due to unsymmetrical stress distribution, which trigger the buckling phenomenon. Such wavelength increases with the pipe thickness (Guarracino, 2004). Quite obviously, the effect arising at one end diffuse inwards and couples with the one traveling from the other hand. This coupling may result

in the longitudinal compressive stress becoming larger than the critical value as expressed in (Guarracino, 2004) at mid length for thick specimens, where the described phenomenon does not develop fully, or in more than one section for thin tubes as shown in figure 6.

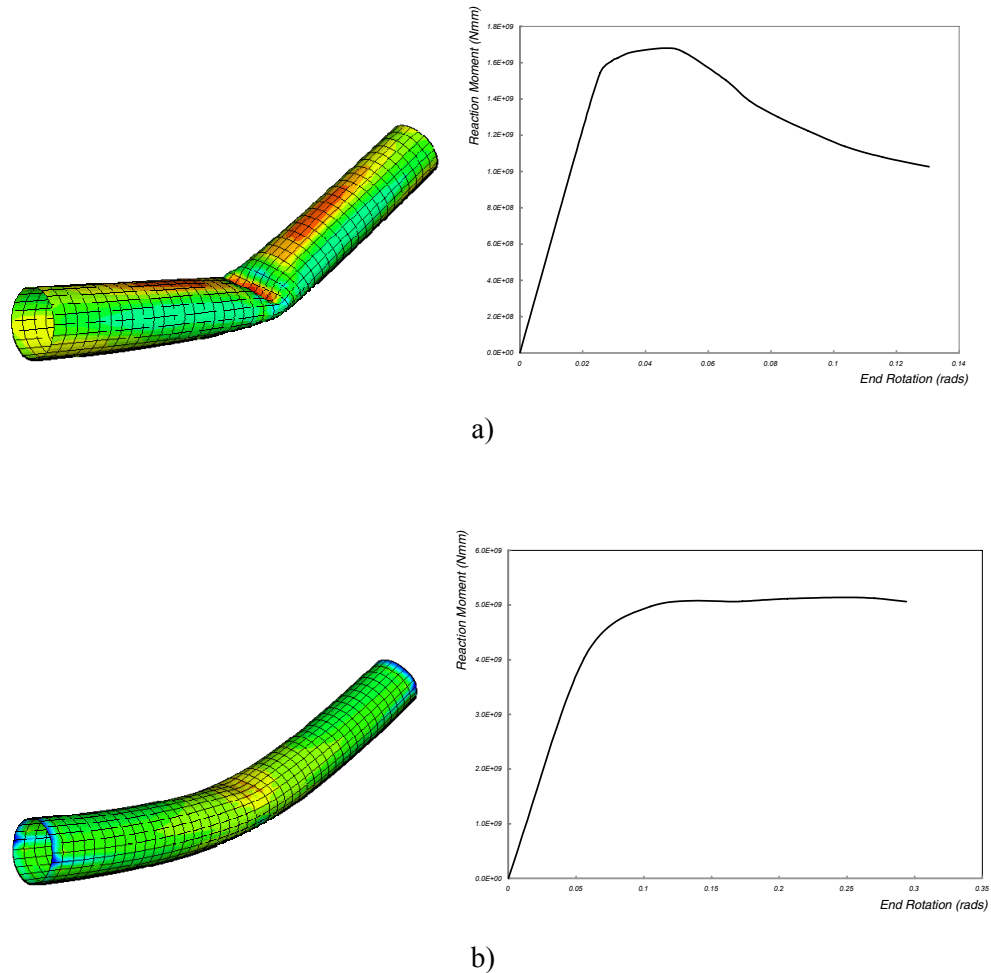


Figure 5 – Deformed configuration and moment-rotation curve for a thin pipe (a) and a thick pipe (b)

This considerations are to be revised when imperfections are present. In real cases, such imperfections are unavoidable for construction reasons. In particular, for example in submerged pipelines, the constructional and laying process features tube portions being subsequently welded.

The welding can introduce misalignment along the longitudinal axis. Such geometrical imperfection is very easily taken into account by modeling two half-tubes, which can be located in space so that a chosen misalignment is realized, and then defining the necessary transition elements. These elements, which actually model the weld, can also be provided with different material in order account for the mechanical properties of the weld material.

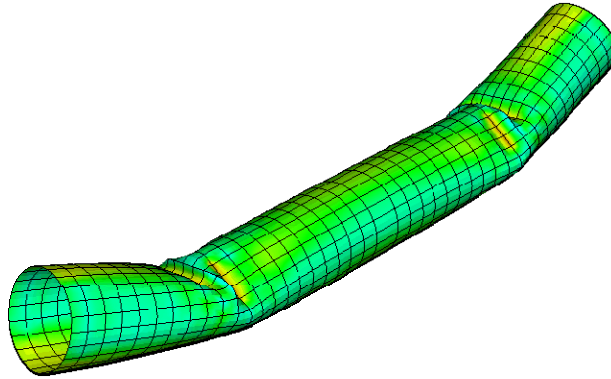


Figure 6 – Deformed configuration of a very thin pipe

This modeling issue is pictured in figure 7, where the misalignment is amplified in an unrealistic manner for the sake of clear representation.

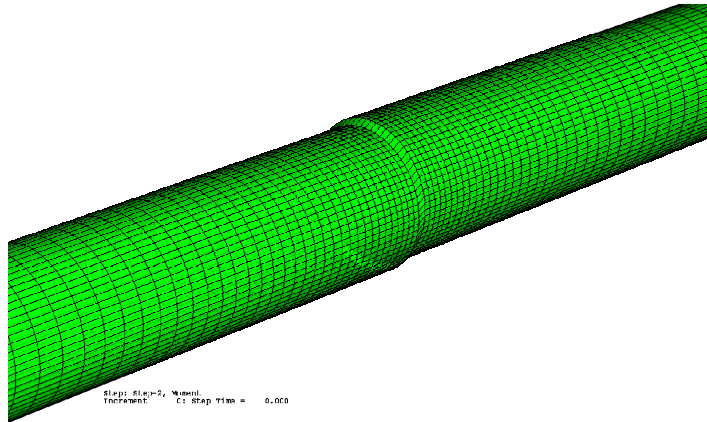


Figure 7 – Geometrical imperfection due to misalignment

In the same figure a necessary bias of the FE mesh towards the imperfection is also shown.

The imperfection may result either in *seeding* the buckling phenomenon, or in a softening effect over the response curve, depending on the D/t ratio. Of course, interaction between the two effects generally takes place.

3.2 Free ovalization

Different modeling schemes can be used to simulate such boundary condition. One very simple and straightforward approach is to adopt the same constraint as above, coupled with an equivalent concept of the De Saint-Venant principle in the beam theory. In this aim, the FE model must feature an appropriate length so that the localized effect due to the applied end constraint is no longer felt in the mid regions of the model. This approach, which may seem the easiest to apply, is not free from some severe shortcomings, mainly depending on the uncertainty of selecting the appropriate length, and on the fact that in some frequent cases the elements close to the ends easily experience severe, and unlikely, plastic deformations that affect the finite element results. Figure 8 shows the deformed shape of a 25 m long pipe with a D/t ratio of 30

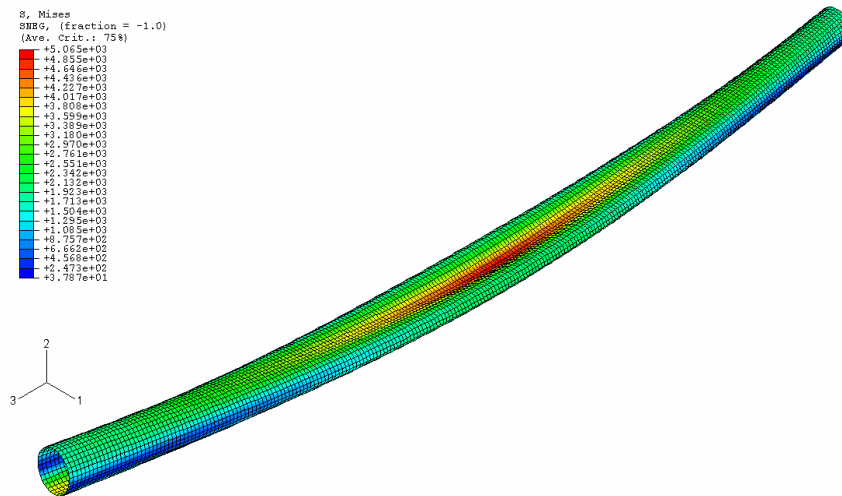


Figure 8 – Deformed configuration of 25m D/t=30 pipe

Another technique investigated by the authors is the sliding plane approach shown in figure 9. The modeling scheme features the contact between the set of nodes at one or both ends and an analytically rigid surface. The nodes are constrained to remain, in the deformed position, on the plane defined by the rigid surface, while being free of sliding on it. This is done, within the computer program, defining a master-slave contact with no friction properties. The surface is then rotated, so that the bending action is applied.

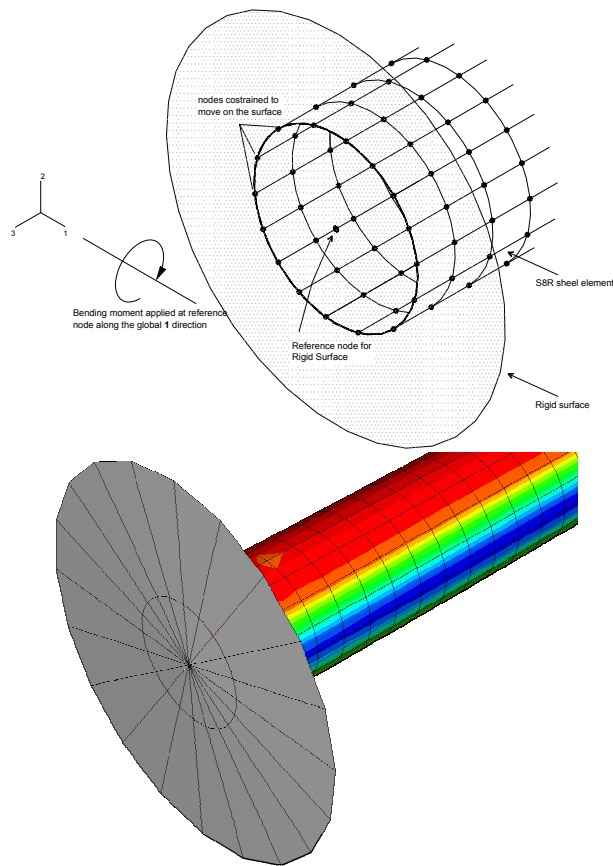


Figure 9 – sliding plane technique

In such modeling technique, the nodal rotations are not affected by the contact. For this reason, once the rigid surface is rotated, the cylinder generatrices may not remain orthogonal to the surfaces, i.e. to the cross section. Thus, this approach happens to be suitable mostly for thick tubes, which are less sensitive to local instability phenomena.

The following figure 10 shows a case in which the above circumstance takes place. Conversely, in case of thick pipes, this effect does not occur, and the described modeling technique proves to be effective, as in the following figure 11.

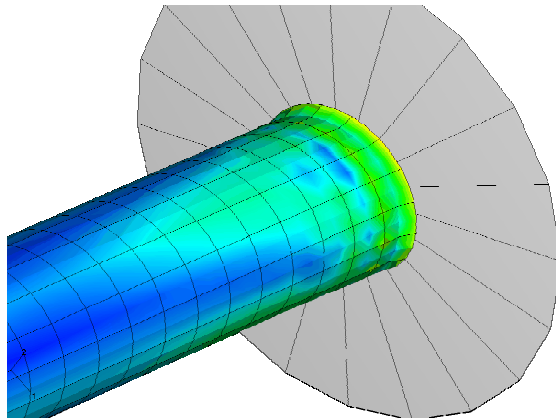


Figure 10 – End local buckling in a thin pipe w/sliding plane technique

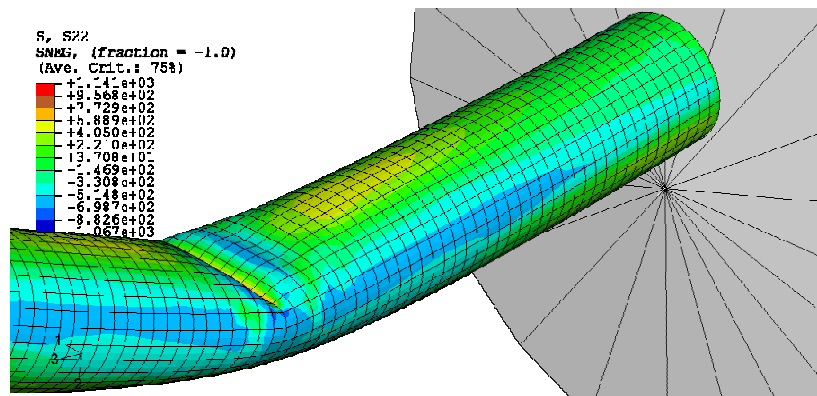


Figure 11 – Deformed configuration of a thick pipe w/sliding plane technique

Of course, as a consequence of the free sliding, translational lability is present. For this reason, some additional boundary conditions have to be provided. In the analyzed models, the two nodes on the horizontal diameter at mid section are restrained against vertical displacement. Such condition, while preventing external lability, do not affect the buckling modes and, inherently, the analysis results. The following figure 12 displays the above concept.

A further approach, which has proved to be the most effective, is the use of the kinematic coupling constraint, through which a chosen number of nodes (the “coupling” nodes) are constrained to the

rigid body motion of a single node (figure 13). The degrees of freedom that participate in the constraint can be selectively chosen. Such degrees of freedom at the coupling nodes can also be specified in a local coordinate system.

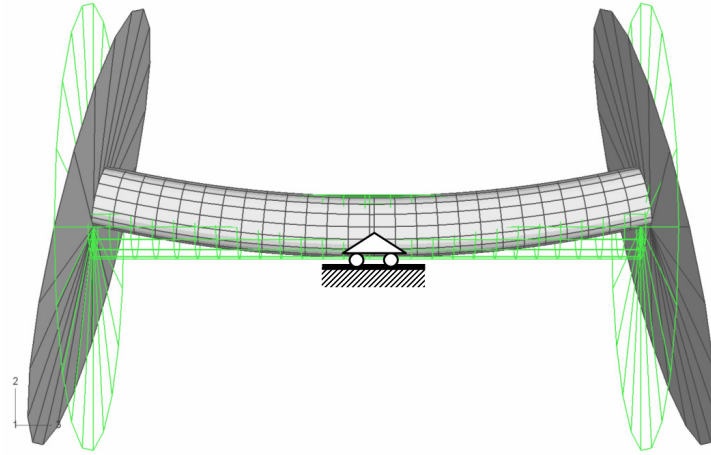


Figure 12 – Additional boundary condition for sliding plane technique

If this coordinate system is spherical or cylindrical, the kinematic coupling constraint can be used to prescribe a twisting or bending motion to the model without constraining radial motions. In particular, for spherical system, the application of a rotation to the reference node would result in a bending action onto the pipe which do not enforce restrained ovalisation of the cross section.

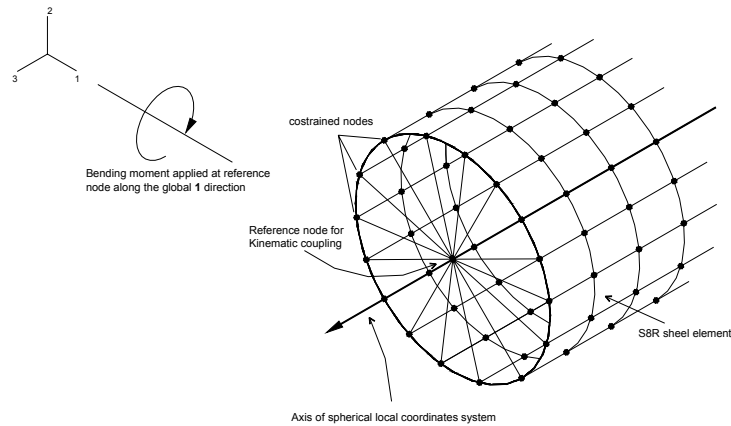


Figure 13 – Free ovalisation through kinematic coupling in spherical coordinates

The approach is also suitable for geometrically nonlinear analyses, as the coordinate system in which the constrained degrees of freedom are specified will rotate with the reference node.

Of course this approach is conceptually equivalent to the sliding plane technique, but is noticeably simpler since it does not require the definition of rigid surfaces. Besides, it has proven to provide smoother results, since the action is actually applied through kinematic constraints and not via direct forces transmitted by the rigid surface, which also requires the definition of interaction properties, generally difficult to manage.

The following figure 14 shows a deformed configuration of a bent pipe, analysed using the above modeling technique, in which the effect of free ovalization is fairly evident.

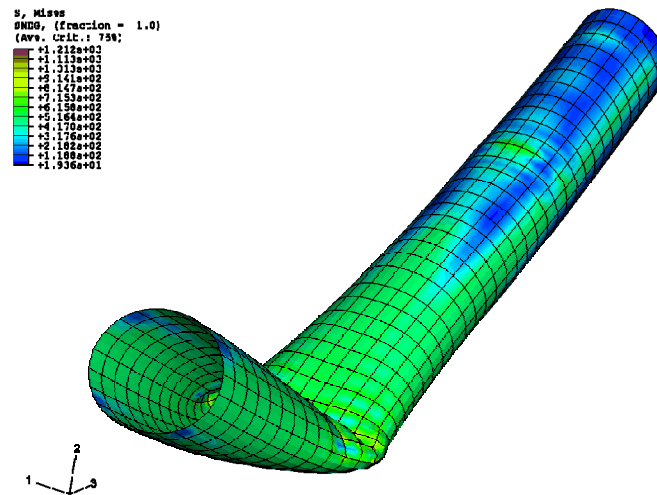


Figure 14 – Deformed shape of a pipe with free section ovalization through kinematic coupling

It is also noticeable that the analysis which the above figure refers to has been carried out using very few 8-node finite elements, which displays the exceptional performance and capability of such elements in geometrically and mechanically non linear problems.

3.3 Solution strategies for the nonlinear problem

Nonlinear static problems are often unstable. Such instabilities may be of a geometrical nature, such as buckling, or of mechanical nature, such as material softening. If the instability manifests itself in a global load-displacement response with a negative stiffness, the problem can be treated as a buckling or collapse problem. However, if the instability is localized, there will be a local

sudden transfer of strain energy from one part of the model to neighboring parts, and usual solution methods may turn out to be unreliable or even non-converging. This class of problems has to be solved either dynamically or with the aid of (artificial) damping; for example, by using dashpots. Besides, the used FEM code (ABAQUS/Standard v.6.6) provides an scheme for stabilizing unstable quasi-static problems through the automatic addition of volume-proportional damping to the model. This is recalled by including the STABILIZE parameter on any nonlinear quasi-static procedure. In order to assess the most suitable numerical procedure, parametric analyses have been carried out using each of the afore mentioned solution strategies.

In the experience gained in the analyses relevant to this paper, the use of the automatic stabilization procedure implemented in Abaqus appears to be the most effective.

4. Experimental testing

Very recently, several carefully conducted experiments on circular carbon steel tubes under bending (see Figures 15 and 16) have shown a noticeable difference between the strain gauge readings at the extrados and at the intrados of the originally circular section (Walker et al. (2003)). With respect to the resulting axial stresses, these differences can exceed the ratio of 1.25 to 1.

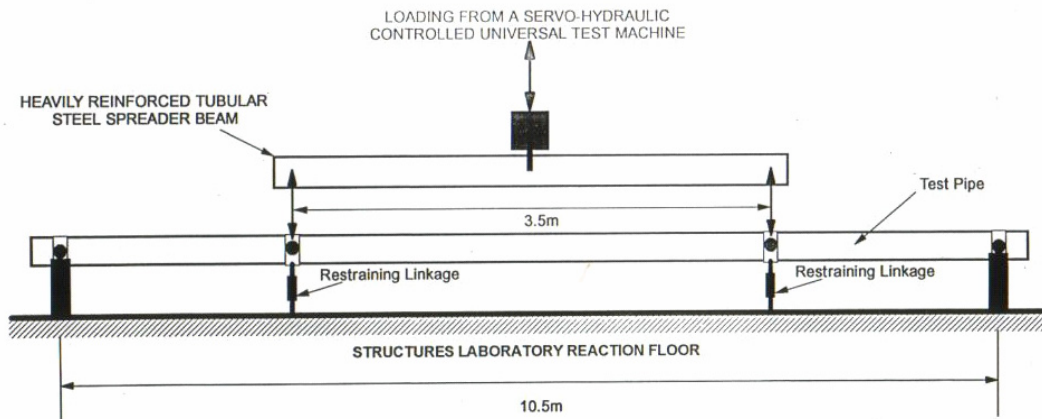


Figure 15: Four points bend test arrangement for steel pipes.

These findings could be ascribed to the onset of axial wrinkles at the compressed region of the bent tube, but also, more significantly in the present case, to the influence of the boundary conditions on the complex and highly sensitive response of the tube under bending.

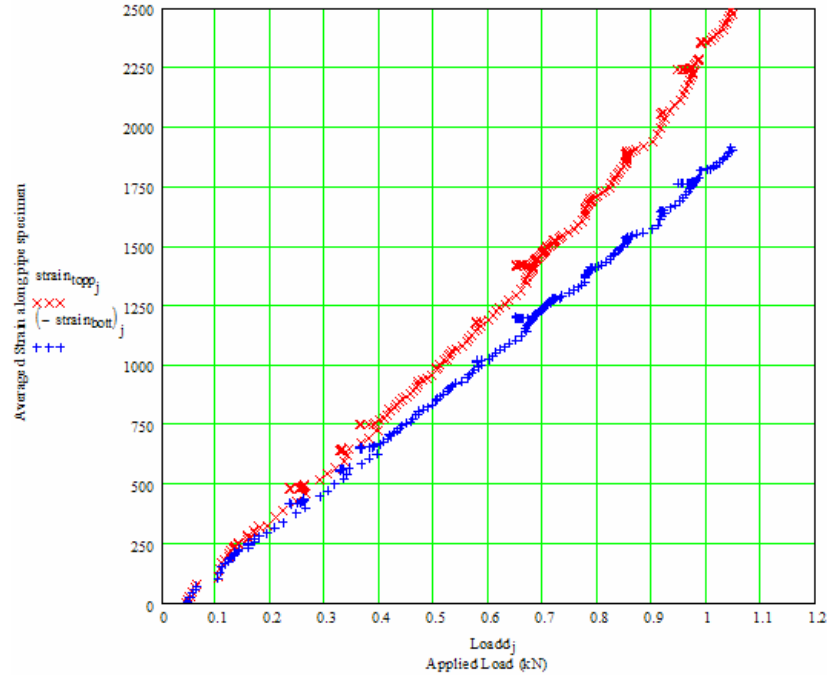


Figure 16: Plot of averaged strain along the top and bottom of pipe bend test specimen

An analytical treatment of the problem will be proposed in a forthcoming paper which leads to an increment (or decrement) in the bending stresses described by

$$\Delta\sigma_z = \frac{\beta}{2\alpha^2} \frac{\Delta r}{rt} \frac{\left(\frac{a_1}{B} + \frac{a_2}{Gt^3}\right)}{\left(\frac{a_3}{B} + \frac{a_4}{Gt^3}\right)} e^{-\alpha z} \sin\left(\alpha z + \frac{\pi}{2}\right) \quad (5)$$

where $B = EJ$, $\alpha = \sqrt[4]{\beta/(4B)}$, G is the shear modulus of elasticity and Δr is the degree of ovalisation that the pipe would experience without any constraint. $a_1, \dots, a_4$ are functions of the type of restraint that prevents the ovalisation.

5. conclusion

The choice of a design procedure is to be linked not only to its capacity to fit experimental data but also to the overall safety objective pursued. For deep-water pipelines safety is related to the specified mechanical characteristic of the material and to the geometrical characteristics of the line pipe. The FE analyses performed show that the actual capacity of the pipe to sustain bending is affected by several factors and therefore requires extensive and expensive testing. A test method that can replicate the effects of external pressure to cause the collapse of long pipelines and that is easy to set up and complete is currently under development and validation.

6. REFERENCES

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