Influence of Stiffeners and Buckling Arrestors on the Behaviour of Offshore Pipelines under Bending

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Abstract: On the basis of past experimental results and of a simple analytical formulation it is shown that the presence of stiffeners and buckling arrestors can substantially alter the strain and stress distribution in offshore pipelines under bending. A simple formula is provided to estimate such effects.

Keywords: Stiffeners, buckle arrestors, offshore pipelines, bending.

1. INTRODUCTION

Pipelines are vital components in the energy systems of all economically developed countries and are designed to sustain the effects of a wide range of loading conditions resulting from internal and external pressure and bending during installation and operations.

It is known that a locally damaged offshore pipeline may fail progressively over long distances by a propagating collapse failure driven by the hydrostatic pressure of the seawater [1]. In fact, the external pressure required to thrust a propagating collapse is much smaller than the pressure necessary to initiate it when the pipe is undamaged. Since for deepwater pipelines it would be excessively costing to design the pipeline in order to prevent a propagating collapse failure, it is convenient to install buckle arrestors, such as thick-wall rings, at intervals along the pipeline, see Figure **1**. A series of such stiffeners will limit the extent of damaged pipe in event of an accident. Also, for pipelines installed by J-Lay, such stiffeners are used as pipe support collars.

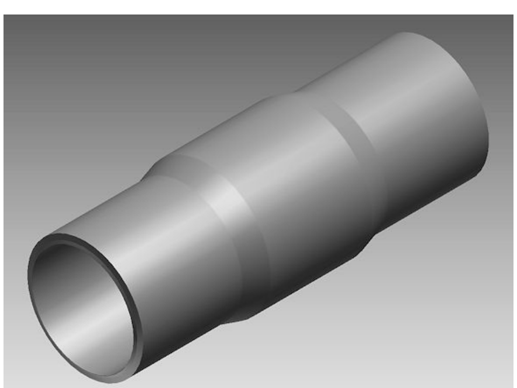

Figure 1: A buckle arrestor.

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However, it has been noticed in the past by the present author that the presence of stiffeners and buckling arrestors results in preventing the natural ovalisation of the pipe under bending [2-5]. The ovalisation is due to the well-known von Karman-Brazier effect and its inhibition can substantially alter the expected strain and stress distribution in the pipe walls.

As such, the purpose of the present paper is to present a review of the previous findings and remark a simple analytical expression which can provide a straightforward evaluation of the effects of stiffeners and buckle arrestors on the bending of pipelines which can occur especially at the installation stage [6] without making resort to complex finite element analyses.

2. TEST RESULTS

By and large, testing a section of a circular cylindrical shell in purely bending loading is carried out on the basis that the test specimen deforms according to simple bending beam theory. Primarily this implies that while the material remains elastic the application of purely bending moment will induce maximum tensile and compressive strains that are identical in magnitude. A typical test rig for a medium diameter pipe, of about 700mm diameter, is shown in Figure **2**. The test rig applies a four-point bending condition with the central section of the test pipe assumed to be subjected to bending action only, with no, or at most very little, shear or axial forces.

The form of load-deformation plot from such a rig is shown in Figure **3** for a pipe with a D/t ratio of about 40. From the limit state point of view the two relevant conditions are the maximum moment and corresponding strain, for load-controlled conditions of design, and the strain at which the reduction of load-bearing capacity first occurs, which relates to displacement-controlled design conditions. Following

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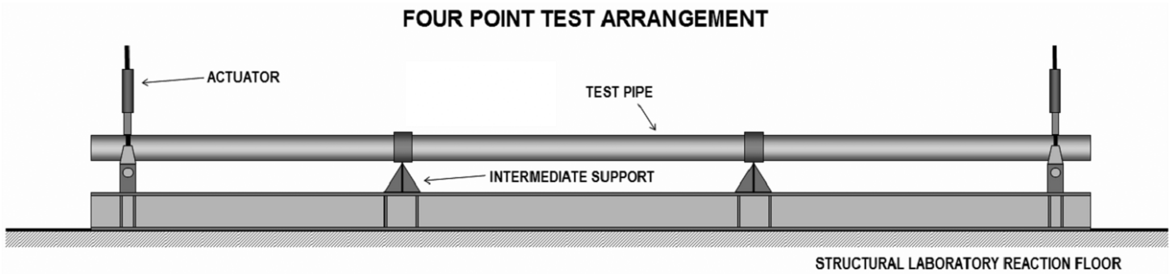

Figure 2: Typical four-point bending test arrangement.

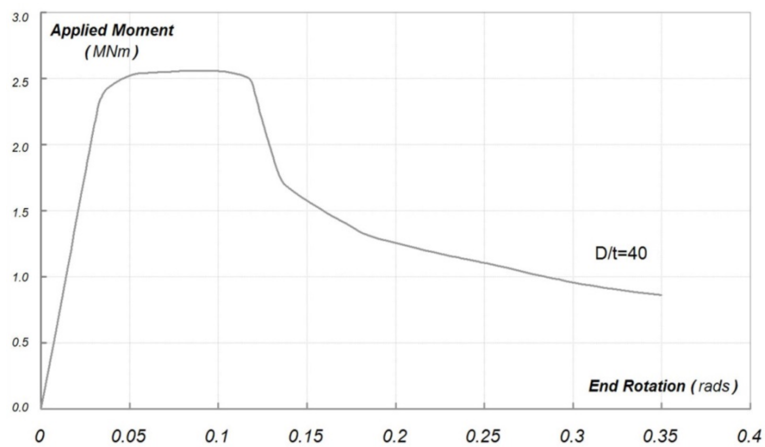

Figure 3: Ends rotation vs. applied moment.

the attainment of that strain, as the loading is further applied; the pipe develops a very local form of buckling.

Since the pipe is assumed to be an extremely simple structural element, and the simple beam theory holds true, it has been common practice to assume that the axial strains have identical values in tension and compression and that the strains can be calculated directly from the curvature or the vertical displacements of the central section of the pipe. The ultimate strain values from tests in which the pipe has been loaded to the point of local buckling have usually been inferred from measurements of the deformations. Only recently have strain gauges been attached to the test pipe to measure axial strains directly.

Some time ago tests [7] were carried out on 152mm diameter pipe to determine the minimum curvature to which the pipe could be deformed prior to local buckling occurring. An arrangement similar to that in Figure **2** was used, and strain gauges to measure axial and circumferential strains were attached at intervals of 100mm apart along the central test section. Very thick steel collars were used to protect the pipe at the supports and at the loading points. The collars were machined to fit very closely around the pipe to ensure no localised loading was applied to the pipe wall. As a result, the pipe was fully prevented from ovalising at all these points. In the design of the test rig it was assumed that a central test section of about 5D would suffice to ensure that end effects due to the loading conditions would diminish to a negligible level along the major part of that section. Figure **4** shows results of the axial strain values along the top and bottom of the pipe section for two levels of the applied loading. It is evident the axial strains are fairly uniform along the length of the test section but there are significant

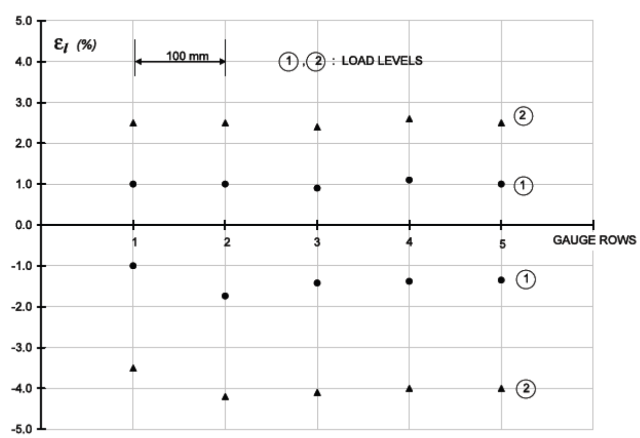

Figure 4: Results from a 152mm diameter pipe bend test [7].

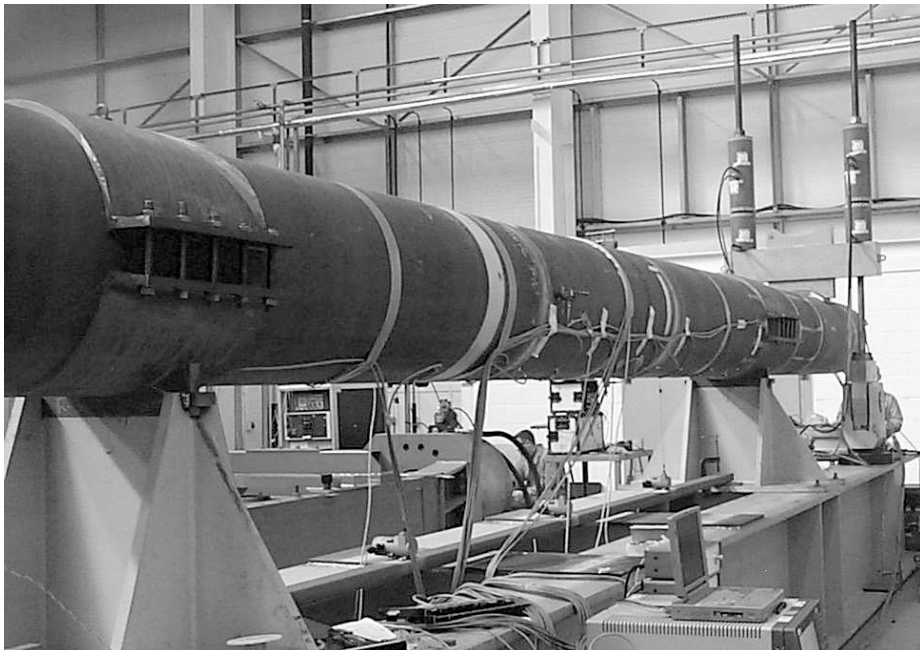

Figure 5: Saddles at the supports and load application points.

differences in the averaged values of the compressive strains compared to the tensile strains.

At that time the evident anomaly between the measured strains with the expected values vis-à-vis the simple bending theory was not followed up, and even after checking that the strain gauges were correctly positioned and the instrumentation was functioning properly the cause of the anomaly was not further investigated.

Some time later, proving tests were carried out on a section of 609mm diameter pipes containing a thin liner made from a corrosion resistant material [8]. The purpose of the tests was to determine accurately the level of strain to which the pipe could be bent before the liner buckled locally. The test arrangement of Figure **2** had a loading arm 2m long to create the moment in the central section of the test pipe. The test section was arranged to be 3.5D. The load was applied to the test pipe using straight bars and loose yokes around part of the pipe circumference. Saddles were used at the loading points and at the supports, see Figure **5**. In this case not only the pipe section was not prevented from ovalising but, on the contrary, the localised actions at the loading points and at the supports were such to cause a local degree of ovalisation. A number of axial strain gauges were attached along the top and bottom centre lines of the pipe at intervals from the support points. The values of strain were monitored as the load values were progressively increased. Figure **6** shows plots of the values for the top and bottom gauges averaged along the test sections and plotted against the corresponding value of applied load.

It is evident from Figure **6** that there is a again systematic difference between the averaged strains along the top and the bottom of the pipe, with a reversed result with respect to Figure **4**. At the maximum load level, the averaged axial tensile strains were 1.28 times the corresponding averaged compressive strains.

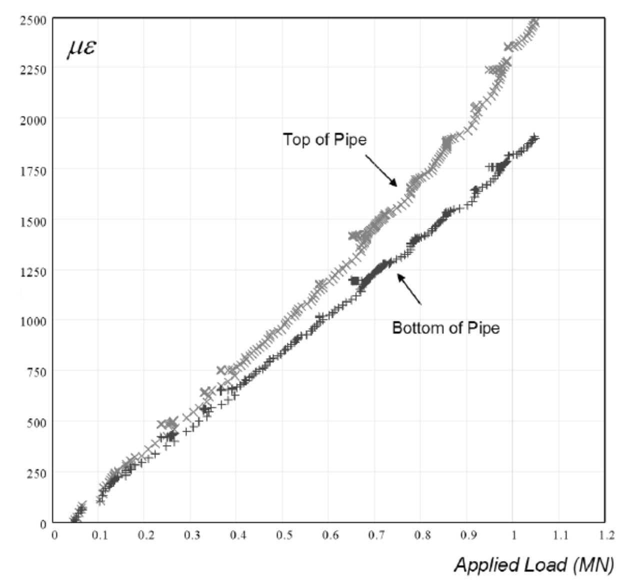

Figure 6: Averaged strain values plotted against corresponding values of applied loading [8]: Maximum Ratio of Tensile to Compressive averaged strains = 1.28 (*D=609.6*mm, *t=18.9*mm, (*D/t=32*), X65 material).

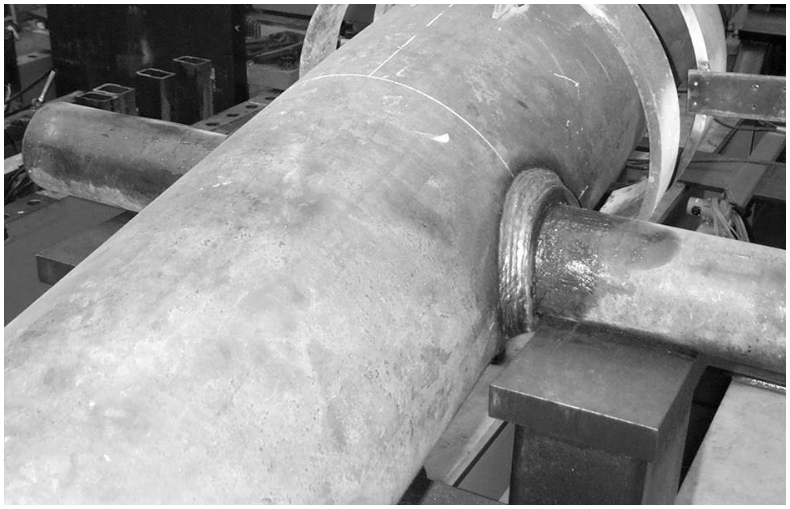

Figure 7: Test arrangement with modified support and load application points.

In view of the importance of the results of the tests in providing the allowable levels of strain for the lined pipe an investigation was made with regard to the underlying cause of the anomaly. This is described extensively in [4], with the aid of several finite element models intended to replicate the conditions in bending tests, or in pipelines that have changes in cross-section and are subjected to bending, with special attention paid to the constraint arrangements.

The investigation determined that the cause lay in the effect of the imposed ovalisation applied by the saddles at the load points. This result pointed to a proposal for the modification of the loading application in which the loads were applied, not through local stiffening of the pipe wall or saddles, but by means of a shaft through the neutral axis of the pipe, as shown in Figure **7**.

The test pipe was fitted with strain gauges, as before, and also gauges to measure the ovality of the pipe. The values of the axial strains measured by the gauges along the test section of the pipe were very uniform. As expected, with the modified loading and support arrangement, the averaged measured values of compressive strains agreed very closely with the corresponding values of the tensile strains, see Figure **8**.

This said it can be expected that if the ovalisation is prevented, as it is in the case of applied stiffeners of buckling arrestors, the compressive stress will exceed the value of tensile ones under bending, whereas if some sort of localised ovalisation is imposed, the tensile stress will exceed the value of compressive ones.

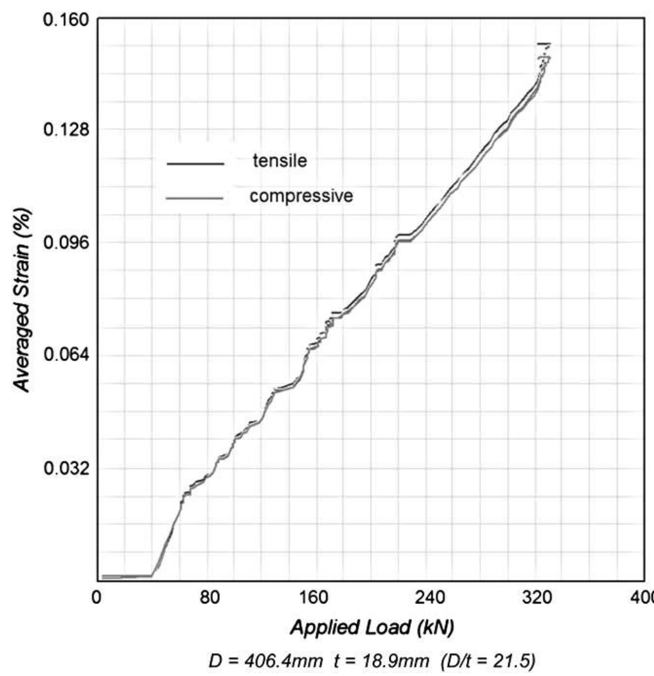

Figure 8: Results from modified bend test.

In the next Section a simple approximate analytical approach will be discussed in order to allow a numerical evaluations of such effects.

3. DERIVATION OF A SIMPLE FORMULA

In order to provide a simple tool to evaluate the effect of imposed or prevented ovalisation, reference is made to the classical Ritz's approach [9] and to a modified set of Donnell's strain and curvature changes [10]. As a matter of fact, the Ritz's method has been extensively used by structural engineers well through the middle of the twentieth century until it has progressively lost ground to its more versatile localised form, i.e. the finite element method. Nevertheless, many formulae of primary practical importance have been found by this mean, which still form the basis of our understanding of a large number of mechanical problems [11]. A difficulty of the Ritz's method certainly consists in the extensive calculations required, but the appearance of computer algebra systems (CAS), that are software programs which allow manipulation of mathematical expressions in symbolic form, has now made possible the treatment of many problems abandoned in the past.

The advantage of the proposed procedure lies in the extreme simplicity of its final expression, which can give a meaningful physical insight into the parameters which govern the problem at hand and can also offer a first validation to subsequent three-dimensional and computationally expensive analyses.

Donnell's equations [10] have been used with a considerable degree of success for the analysis of elastic and plastic buckling of thin-walled circular cylinders. The basic assumptions at the basis of Donnell's theory have proved to be able to deal with several deformation modes with a satisfactory degree of accuracy and for this reason they can be considered to be able to represent also the cases in which loading is not symmetrical with respect to the axis of the cylinder. However, Donnell's equations are not well adapted to solution by Fourier series since some of the high-order derivatives found in the formulation sometimes lead to divergent trigonometric series. Even if in the present Ritz's approach reference is naturally made to an energy expression and no differential equations are involved, nevertheless a modified set of strain and curvature changes are employed.

A circular cylindrical shell of infinite length is taken into consideration. With reference to an element in the middle surface of the shell, the coordinate axes are directed with the *x*-axis in the axial direction of the cylinder, the *y*-axis in the circumferential direction and the *z*-axis in the radial direction. *u, v*, and *w* are the components in the *x, y*, and *z* directions of the displacement of a generic point. Said φ the central angle, the strains are assumed to be;

$$\varepsilon_x = \frac{\partial u}{\partial x}, \quad \varepsilon_\varphi = \frac{1}{r}\frac{\partial v}{\partial \varphi} - \frac{w}{r}, \quad \gamma_{x\varphi} = \frac{\partial u}{r\,\partial \varphi} + \frac{\partial v}{\partial x} \tag{1}$$

where *r* is the radius of the middle surface of the shell.

In calculating the expression of the strain energy, the changes of curvature of the middle surface of the shell are also required. To this scope the following approximate expressions are assumed;

$$\chi_x = \frac{\partial^2 u}{\partial x^2}, \quad \chi_\varphi = \frac{1}{r^2}\left(w + \frac{\partial^2 w}{\partial \varphi^2}\right), \quad \chi_{x\varphi} = \frac{1}{r}\left(\frac{\partial^2 w}{\partial \varphi\,\partial x} + \frac{\partial v}{\partial x}\right) \tag{2}$$

and in order to evaluate the deformation induced by two opposite forces, *F*, acting along a vertical diameter at a certain section $x = 0$, the components of displacement varying along the length of the cylinder are taken in the form;

$$\begin{aligned} u &= C_1\, R\alpha\, e^{-\alpha_1 x} \sum_{n=1}^{\infty} \frac{1}{n}\left[A_n \sin(n\varphi) + B_n \cos(n\varphi)\right]\left[M_1 \sin(\alpha_2 x) + N_1 \cos(\alpha_2 x)\right] \\ v &= C_2\, e^{-\alpha_1 x} \sum_{n=1}^{\infty} \left[A_n \cos(n\varphi) - B_n \sin(n\varphi)\right]\left[M_2 \sin(\alpha_2 x) + N_2 \cos(\alpha_2 x)\right] \\ w &= C_3\, e^{-\alpha_1 x} \sum_{n=1}^{\infty} n\left[A_n \sin(n\varphi) + B_n \cos(n\varphi)\right]\left[M_3 \sin(\alpha_2 x) + N_3 \cos(\alpha_2 x)\right] \end{aligned} \tag{3}$$

where $\alpha_1,...,N_3$ are constants that must be calculated for the case of loading at hand. It is worth noticing that no particular physical meaning can be attributed a priori to the constants $\alpha_1,...,N_3$ and that the statement of the problem does not involve any form of local or global buckling.

Essentially, this is the key differentiation of the present approach with respect to classical formulations which assume the change of curvature in the direction of the generatrix to be equal to zero [11]. It must also be pointed out that the derivatives of the displacement field (3) result discontinuous with respect to the section of symmetry and, therefore, the displacement field cannot be considered kinematically admissible over the whole length of the tube, that is for $x \in]-\infty, +\infty[$. However, the extended Ritz's method provides a solution that fulfils the displacement conditions approximately, as well as the conditions of equilibrium and the static boundary conditions.

As anticipated, the present assumption implies a considerable computational effort at a symbolic level to define the total strain energy by integration over the surface of the shell of the strain energy per unit area in terms of Eq.(3). Therefore, the whole operation has been performed by means of ad hoc routines written with the aid of the symbolic system Mathematica® [12].

The equations for calculating the constants $\alpha_1,...,N_3$ have been first obtained by imposing the total potential energy to be a stationary value and then solved. The results have been expanded in series, trigonometrically fit and simplified in order to obtain a practical expression.

The end result can be summarised in the following formula, which provides the top and bottom mid-surface strain on account of the deformation induced by two opposite forces, *F*, acting along the vertical diameter at the mid-span;

$$\varepsilon = -\frac{2\left(1-\nu^2\right)F}{Ert} e^{-\frac{\beta x}{2}} \cos \beta x \tag{4}$$

where *t* is the thickness of the shell and *E* and λ stand for the Young's modulus and the Poisson's ratio, respectively.

β is given by;

$$\beta = \frac{\pi^2 \sqrt[4]{\left(1-\nu^2\right)}}{4\sqrt{r^3/t}} \tag{5}$$

Eq. (4) has been extensively validated against FE results in [5]. Here a different expression is also proposed, following the same line of reasoning but a different approximation,

$$\varepsilon = -\frac{64\left(1-\nu^2\right)F}{Ert} e^{-2\beta x}\left(1-\frac{\beta^2 x^2}{2}+\frac{\beta^4 x^4}{24}\right) \tag{6}$$

which depicts a more rapidly decaying value of the longitudinal strain even though its value in correspondence of the concentrated load is much higher than that yielded by Eq.(4). However, since a concentrated force constitutes a mathematical idealisation, formulae (4) and (6) can be effectively compared at a distance *L>D*.

It is worth noticing that the expression of what can be considered the natural half-wavelength of the problem results proportional to the term $\sqrt{r^3/t}$, whereas in the case of circular shells subject to axial symmetric loading , it is proportional to $\sqrt{rt}$.

According to Eqs. (4) and (6), Figure **8** shows the value of the top and bottom strains along the axis *x* of the pipe of Figure **7** induced by two opposite forces of magnitude 1.118MN.

The proposed formulae can be straightforwardly employed to evaluate the order of magnitude of the difference in top and bottom strains with regard to a tested sample of the previous Section. In fact, for the pipe characterised by *D=609.6*mm, *t=18.9*mm and X65 material [8], for an applied load of 1.118MN the absolute value of the top and bottom strains calculated according to the simple bending theory is 2.3504×10^{-3}. Eq. (6) yields the additional strain at the mid-span on account of the local deformation induced by the concentrated loads, that is 3.5062×10^{-4}. By adding this latter quantity to the tensile strain and subtracting it from the compressive one, it follows that the ratio of tensile to compressive strains is equal to 1.35, with a difference from the experimentally measured ratio of about 5.5% (see Figure **6**). This can be considered a quite satisfactory result, bearing however in mind that in the actual testing arrangement the supports and the applied loads are not opposite, as it is the hypothesis leading to Eqs. (4) and (6), but distant 2m apart.

In order to evaluate the result of a prevented ovalisation as a consequence of the presence of stiffeners or buckle arrestors, the previously considered forces, *F*, can be reversed in order to neutralize the von Karman-Brazier effect [2] and the subsequent distribution of the strain and stress distribution in the pipe walls can be once again obtained by Eqs.(4) and (6).

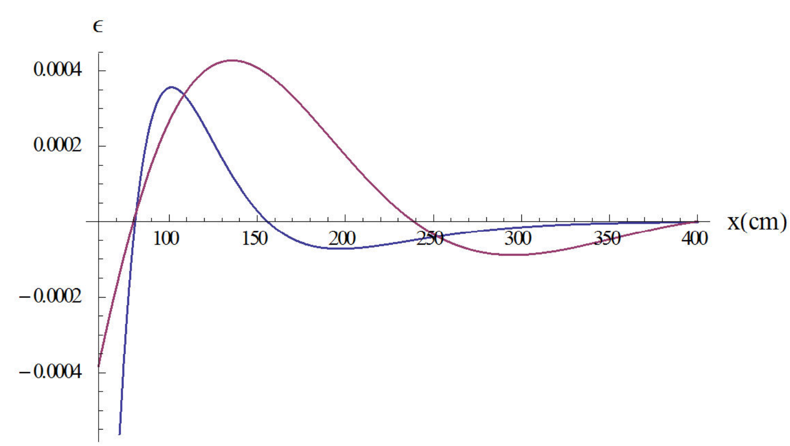

Figure 8: Top and bottom strains induced by two opposite forces of magnitude 1.118MN in the pipe of Figure **7** (D=609.6mm, t=18.9mm) according to Eqs. (4), blue, and (6), magenta.

CONCLUSIONS

The present work has presented a review of previous findings and a simple analytical expression, which can provide a straightforward evaluation of the effects of stiffeners and buckle arrestors on the bending of pipelines, which can occur especially at the installation stage.

On these bases, the seemingly anomalous values of measured axial strain in aforementioned tests can be explained very straightforwardly. The proposed formulation offers a physical insight into the mechanics of the problem in the fashion of many classical results still widely used in the engineering practice and can turn useful both at the design and at the assessment stage of offshore pipelines.

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REFERENCES

[1] Palmer AC and Martin JH. Buckle propagation in submarine pipelines, Nature 1975; 254: 46-48. https://doi.org/10.1038/254046a0

[2] Guarracino F. On the analysis of cylindrical tubes under flexure: theoretical formulations, experimental data and finite elements analysis. Thin-Walled Structures 2003; 41(2-3): 127-147. https://doi.org/10.1016/S0263-8231(02)00083-6

[3] Guarracino F, Fraldi M and Giordano A. Analysis of testing methods of pipelines for limit state design. Applied Ocean Research 2008; 30(4): 297-304. https://doi.org/10.1016/j.apor.2008.12.003

[4] Guarracino F and Walker AC and Giordano A. Effects of Boundary Conditions on Testing of Pipes and Finite Element Modelling. Int J Press Ves Piping 2009; 86(2-3): 196-206. https://doi.org/10.1016/j.ijpvp.2008.09.009

[5] Guarracino F. A Simple Formula for Complementing FE Analyses in the Estimation of the Effects of Local Conditions in Circular Cylindrical Shells. CMES – Comp Mod Eng Sci 2011; 72 (3): 167-184. https://doi.org/10.3970/cmes.2011.072.167

[6] Guarracino F and Mallardo V. A refined analytical analysis of submerged pipelines in seabed laying, Applied Ocean Research 1999; 21(6): 281-293. https://doi.org/10.1016/S0141-1187(99)00020-6

[7] Ellinas CP, Walker AC, Langfield GN and Vines MJ. A Development in the Reeling Method for Laying Subsea Pipeline, in: Proc. 1st Petroleum Tech. Australian Conf Perth Australia 1985.

[8] Walker AC, Holt A, Guarracino F and Wilmot D. Test Procedure for Pipe and Pipeline Material, Offshore Pipeline Technology Conference, Amsterdam, Holland 2003.

[9] Guarracino F and Walker AC. Energy Methods in Structural Mechanics, Telford, London, UK 1999. https://doi.org/10.1680/emism.27572

[10] Donnell LH. Stability of Thin-Walled Tubes under Torsion, NACA Rep 1935; 479.

[11] Timoshenko SP and Woinowsky-Kreiger S. Theory of Plates and Shells, McGraw-Hill, New York, USA, 1959.

[12] Wolfram S. The Mathematica Book, Wolfram Media, Inc Champaign, IL, USA, 2003.

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