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Novel stiffeners exploiting internal pressurisation to enhance buckling behaviour under bending loads

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ABSTRACT

The paper proposes a novel type of stiffener designed to bear bending loads by exploiting internal pressure effects. The stiffener is made of two adjacent thin-walled pipes ($r/t \geq 50$) jointed with a connecting strip. Such a structure is shown to have higher performance against buckling failure compared to a single pipe and its geometry allows for good exploitation of internal pressurisation.

The study is conducted by using the FEA software ANSYS and the analysis technique is the linear perturbation buckling analysis. Internal pressure ranges from 0 to 1.4 MPa. The buckling mechanisms are observed for a set of models with different values of length, wall thickness and geometric variation of the cross-section. It is shown that two different buckling modes can take place. However, for a given geometry, the level of pressure can alter the behaviour and lead to one mode rather than the other one.

Potential of the presented structure is maximised by the use of high performance materials and a possible aerospace engineering application is presented.

1. Introduction

Many engineering applications see the use of slender parts and thin-walled components. These allow limiting the amount of material and consequently the weight whilst providing structural strength and rigidity. On the other hand, design of slender parts and thin-walled components has to keep in consideration that they tend to exhibit structural stability problems, i.e. buckling failure. Therefore extensive studies have been made on this matter.

The first studies on buckling were carried out in XVIII century by Euler who solved the problem of lateral buckling of compressed members and further research was made starting from the XIX century [1,2]. Most of the relevant literature dates back to the XX century, when many authors investigated on buckling of thin cylindrical shells [3–5]. It was found that the buckling behaviour of cylindrical shells subjected to bending depends on the material parameters as well as geometric parameters, especially the diameter–thickness ratio D/t [6]. Thick wall pipes ($D/t < 40$ for steel) exhibit a plastic buckling [7]. On the other hand, thin wall pipes ($D/t > 100$ for steel) buckle at first in the elastic region. Thus, the buckling load depends by some material parameters (Young's Modulus E and Poisson ratio ν) but not by the material yield stress σ_y. Some authors [2,3,8] conducted studies on the elastic buckling of circular pipes under bending and their results of the buckling load differ significantly within a wide range of values. The reason of this variability is that the behaviour (and the strength) of circular pipes is strongly depending on other factors than material properties, such as diameter–thickness ratio D/t, constraints and cross-section flattening.

Authors proved that internal pressure increases the limit buckling load of thin cylindrical shells for different load conditions such as axial load [9], torsion [10] and bending [11,12]. In the case of bending, benefits of pressure are not only related to the stabilisation effect but also to an anti-flattening effect which decreases the cross-section inertial moment reduction and, therefore, the stress in the bent pipe [13]. It was also proved that pipe curvature has effects on buckling phenomenon and curved pipes buckle earlier than the corresponding straight pipes [14,13] and [12].

In our previous work [15] we showed that, for a given pipe, there exists an optimal pressure to which buckling failure and yield failure have the same limit. This allows us to fully exploit the bearing capacity of the pipe and maximise its limit bending load. Therefore, we proposed an optimisation of the pipe based on internal pressure and pipe curvature. Now we carry on with maximising limit load for thin-walled cylinders by studying a slightly more complex geometry made of two adjacent pipes firmly connected each other by a strip. Such stiffener, that we will be referring to it as "pinched pipe", is subjected to internal pressure and bending moment and its buckling behaviour is studied by FEA. We show that for a certain pipe radius r and length L, the pinched pipe

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exhibits higher buckling load over weight ratio than the single pipe. Moreover, the geometry of the pinched pipe still allows us to internally pressurise the structure and exploit the consequent benefits. This structure reveals high potential in application wherein load withstanding capacity is needed.

The paper is organised as follow: Section 2 describes the FE model; Section 3 explicates the analysis procedure; Section 4 is divided into four subsections presenting the numerical results; Section 5 discusses the obtained results; Section 6 provides some remarks on the applications and feasibility of the presented structure; Section 7 concludes the paper.

2. FE model

The FEA software used to perform buckling analysis is ANSYS 15.0. The model geometry consists in a right-prismatic shape. Three surfaces can be identified: a middle plate, which will be referred to as “strip”, and two adjacent and identical cylindrical shells symmetrical with respect to the strip and connected to the latter along the edges. The entire structure is referred to as pinched pipe. Fig. 1 shows the cross-section geometry with the relevant geometric parameters; Fig. 2 shows the meshed model and reference system.

The problem is modeled with SHELL181 which is a four-nodes shell element. The mesh size, based upon a sensitivity study, is set to 3 mm along the axial direction and approximately 4 mm along the hoop direction (corresponding to 0.06 and 0.08 times the radius r, respectively). Bending moments around the X-axis are applied to the nodes of the two stiffener ends, rotating them downwards. The ends are constraints to behave as rigid planes. The other acting load is pressure applied to the inner surfaces. The material model is linear and isotropic with typical properties of a steel. In order to reduce the computational power required by the simulations only half of the pinched pipe is modeled taking advantage of the symmetry of the problem. Therefore, additional constraints are added to the XY plan (middle plane along the span).

Table 1 summarises the model parameters.

3. The analysis

The FE simulations hereby conducted aim to quantify the limit buckling moment for the different geometric parameters and values of internal pressure.

As shown in our previous work [15], linear and non-linear analysis can lead to somewhat different results. Linear buckling analysis is essentially an eigenvalue problem. Its formulation is the following [16]:

$$(\mathbf{K} + \lambda_j \mathbf{S})\boldsymbol{\psi}_j = \mathbf{0} \tag{1}$$

where $\mathbf{K}$ is the stiffness matrix, $\mathbf{S}$ is the stress stiffness matrix, λ_j is the jth eigenvalue (used to multiply the loads which generated $\mathbf{S}$) and $\boldsymbol{\psi}_j$ is the jth eigenvector of displacements. The simulation time is generally short and more eigenvalues and the corresponding eigenvectors can be obtained in order to predict what the buckling modes will be like; on the other hand, it might over-estimate the actual buckling load.

Non-linear buckling analysis is a static analysis wherein large deflections are accounted for. Loads are applied in gradual steps to seek the load level at which the structure becomes unstable. The problem is solved with the Newton–Raphson method, an iterative process used to solve non-linear equations. In a structural analysis, the problem formulation is the following:

$$\mathbf{K}_i^{\mathbf{T}} \Delta \boldsymbol{u}_i = \boldsymbol{F}^{a} - \boldsymbol{F}_i^{nr} \tag{2}$$

$$\boldsymbol{u}_{i+1} = \boldsymbol{u}_i + \Delta \boldsymbol{u}_i \tag{3}$$

where the subscript i represents the current iteration, $\mathbf{K}_i^{\mathbf{T}}$ is the tangent stiffness matrix, $\boldsymbol{u}_i$, $\boldsymbol{u}_{i+1}$ and $\Delta\boldsymbol{u}_i$ represent vectors of degree of freedom values, $\boldsymbol{F}^{a}$ is the vector of applied loads and $\boldsymbol{F}_i^{nr}$ is the vector of restoring loads corresponding to the element internal loads. Non-linear analysis is more accurate but presents some disadvantages. The main ones are the longer computational time and, in case of buckling, the need to trigger the instability by means of geometric imperfections and/or perturbation loads.

Hereinafter we adopt a different analysis technique to overcome the limitations of the two techniques above. The ANSYS theory refers to such analysis procedure as *linear perturbation buckling analysis*. It consists in two phases: (i) a non-linear static analysis up to a point prior to buckling; (ii) a linear buckling analysis (eigenvalue problem) starting from the preloaded

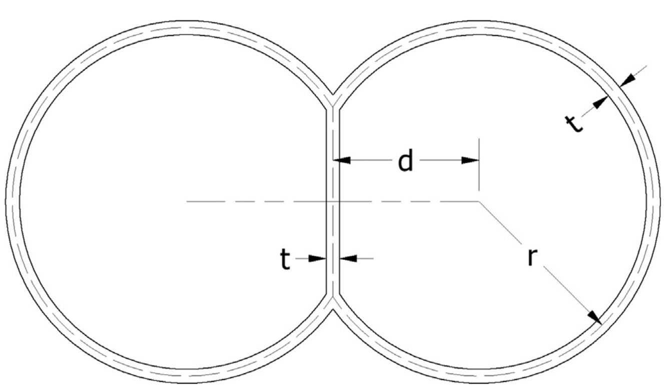

Fig. 1. Cross-section geometry and main geometric parameters: radius r, semi-distance d and wall thickness t.

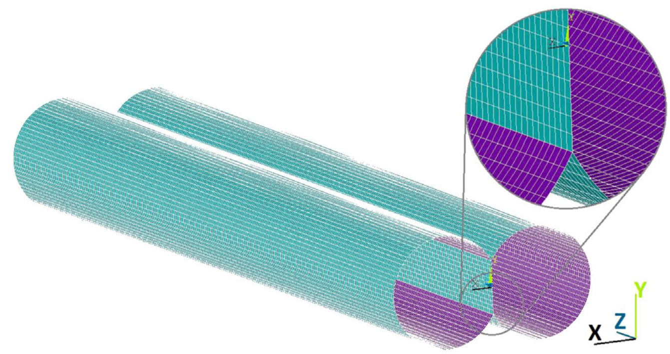

Fig. 2. FE model. Only half span of the pinched pipe is modeled by taking advantage of the symmetry.

Table 1
FE model parameters.

Parameter	Type	Value
Pipe radius, r	Fixed	50 mm
Pipes semi-distance, d	Variable	[40 mm, 48 mm]
Length, L	Variable[a]	[600 mm, 2200 mm]
Wall thickness, t	Variable	0.25 mm, 0.50 mm, 1.00 mm
Internal pressure, p	Variable	[0 MPa, 1.4 MPa]
Young Modulus, E	Fixed	200 GPa
Poisson's ratio, ν	Fixed	0.3

[a] The problem is mainly investigated with a model 1600 mm long although different values are analysed to study the influence of length.