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COLLAPSE PRESSURE OF DAMAGED SANDWICH PIPES

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Abstract. *Sandwich pipes represent an alternative solution for the offshore industry due to their good performance under external pressure and improved thermal insulation, which avoids paraffin formation. During operation, sandwich pipes can be damaged by the impact of anchors, rocks or any kind of heavy objects that are common in marine petroleum fields. The presence of mechanical damage can reduce the collapse pressure of a sandwich pipe. The aim of this work is to investigate the reduction of the collapse pressure of sandwich pipes under external pressure caused by mechanical damage. A three-dimensional nonlinear numerical model is developed according to the finite element method. A sandwich pipe is modeled, consisting of two concentric API X-60 grade steel pipes and an annular layer filled with polypropylene. The numerical model simulates the introduction of mechanical damage (denting process) followed by the application of external pressure on the sandwich pipe. Using the numerical model, a parametric study is carried out to determine the collapse pressure for different sandwich pipe geometries and dent depths. Additionally, two different adhesion conditions between the interfaces of the different layers are considered: no adhesion and perfect adhesion. From the obtained results, it is observed that the initiation pressure of damaged sandwich pipes is directly related to the magnitude of the damage and the condition of adhesion between the steel pipes and the annular layer.*

Keywords*: Sandwich Pipe, Mechanical Damage, Collapse Pressure, Finite Element Method.*

1. INTRODUCTION

The oil and gas offshore industry faces many challenges while operating in deep waters as the water depth is constantly increasing. One of these challenges is to design pipelines and risers capable to withstand high internal and external pressures and to produce a good thermal insulation, avoiding the formation of paraffin. In this case, recent studies (Netto *et al.*, 2002) have shown that sandwich pipes represent an alternative solution for this challenge due to their good performance under high external pressure and improved thermal insulation. Sandwich pipes are composed by two steel pipes concentrically assembled with an annular space, which can be filled with a proper non-structural insulation material. This optimized system is designed to produce combined structural strength and thermal insulation. The inner and outer steel pipes are designed against failure under internal and external pressure, respectively. The three layers are able to work together to resist combined high external pressure (Netto *et al.*, 2002) and bending loads (Pasqualino *et al.*, 2002), typically due to installation processes. The factors governing the collapse and the propagation of buckles in pipe-in-pipe systems under external pressure have been extensively studied by Kyriakides (2002), Kyriakides and Vogler (2002) and Pasqualino *et al.* (2002).

During operation, sandwich pipes can be subjected to mechanical damages due to the impact of anchors, rocks or any kind of heavy objects that are common in marine petroleum fields, such as equipment dropping from the platforms (Kyriades *et al.*, 1984). These mechanical damages induce geometric distortions, or defects, like dents, ovalizations, smooth localized buckles and wrinkles (Pinheiro *et al.*, 2006). The presence of these defects can reduce the collapse pressure of a sandwich pipe and may eventually lead to a local collapse of the line. When a local failure occurs, depending on the external pressure, the buckle may propagate to large distances at a high speed, collapsing the whole pipeline (Kyriakides and Netto, 2000, Netto and Kyriakides, 2000). The minimum pressure that can initiate buckle propagation is known as the initiation pressure (P_i), which depends on the damage magnitude, pipe geometry and material properties (Park and Kyriakides, 1996, Estefen *et al.*, 1992). Once initiated, the propagation phenomenon can only be arrested if the external pressure is lower than a value at which the buckle propagates (propagation pressure, P_p), or if buckle arrestors are used (Netto and Estefen, 1996, Kyriakides *et al.*, 1998). The propagation pressure depends on the geometric and material properties of the pipe (Dyau and Kyriakides, 1993, Pasqualino and Estefen, 2001). Then, once a mechanical damage has occurred, it is important to evaluate if the induced defect represents a threat to the global structure, i.e., if it can cause a local collapse of the line which, depending on the external pressure, may propagate and collapse the entire pipeline. Thus, to ensure a safe operation of sandwich pipes, it is necessary to make a consistent assessment of defects induced by mechanical damages, avoiding the collapse of the line.

The aim of this work is to investigate the reduction of the collapse pressure of sandwich pipes under external pressure caused by mechanical damage. The mechanical behavior of a sandwich pipe is reproduced with the development of a three-dimensional nonlinear numerical model according to the finite element (FE) method. A sandwich pipe is modeled, consisting of two concentric API X-60 grade steel pipes and an annular layer filled with polypropylene. The numerical model simulates the introduction of mechanical damage (denting process) followed by the application of external pressure on the sandwich pipe. Using the numerical model, a parametric study is carried out to determine the collapse pressure for different sandwich pipe geometries and dent depths. Additionally, the influence of the adhesion between the interfaces of the different layers is taken into account, assuming two different adhesion conditions: no adhesion and perfect adhesion. From the obtained results, it is observed that the initiation pressure of damaged sandwich pipes is directly related to the magnitude of the damage and the condition of adhesion between the steel pipes and the annular layer.

2. NUMERICAL MODEL

A three-dimensional nonlinear numerical model was developed according to the finite element (FE) method, with the aid of the commercial program ABAQUS release 6.5 (ABAQUS, 2006). A sandwich pipe was modeled, consisting of two concentric steel pipes and an annular layer. The inner and outer pipes material was assumed as API X-60 grade steel and the annular layer was considered to be filled with polypropylene, due to its wide availability and relative low cost. The basic structural and thermal design requirements are considered in an overall perspective, i.e., the annular layer material is intended to not only produce thermal insulation but also provide, together with inner and outer pipes, sufficient resistance against burst or collapse of the system. The model incorporates plasticity and admits large strains and large rotations (geometric nonlinearity).

To investigate the reduction of the collapse pressure of sandwich pipes under external pressure caused by mechanical damage, a denting tool with spherical shape was modeled as an analytical rigid surface to damage the sandwich pipe. The contact between the outer pipe external surface and the denting tool was simulated with the aid of contact surfaces, assuming small sliding between the two surfaces.

Two conditions of adhesion between the interfaces of the steel pipes and the annular layer were considered: perfect adhesion and no adhesion. For the perfect adhesion condition, a tie constraint is applied at the nodes along the interfaces between each pair of contact surfaces. For the no adhesion condition, the contact between each pair of surfaces along the interfaces was defined assuming small sliding and a frictionless tangential behavior.

2.1. Model Geometry

Three different sandwich pipe geometries were considered for the numerical model, referred to as Case 1, Case 2 and Case 3. The geometries considered are described in Tab. 1, were D_i is the internal diameter of the inner pipe, t_i is the inner pipe thickness, t_a is the annulus thickness and t_e is the outer pipe thickness. These geometries were defined keeping constant the internal diameter of the inner pipe and varying the thicknesses t_i, t_e and t_a. Figure 1 presents a sketch of the sandwich pipe geometry, showing its transversal section with the two pipes concentrically assembled with an annular layer between them. The length of the model (L) was assumed to be equal to $5D$, where D is the external diameter of the outer pipe. Consequently, different sandwich pipe lengths were assumed, according to the geometry considered. This length was selected in order to reduce the analysis computational time without affecting the accuracy of the results of collapse pressure. In addition, to minimize computational time in the numerical analysis, a quarter-symmetry model was used, considering planes of symmetry in the longitudinal and transversal directions, as shown in Fig. 2. The denting tool was modeled as a spherical analytical rigid surface with a diameter equal to $0.5D_i$.

Table1. Sandwich pipe geometries for the cases studied

Case	D_i (mm)	t_i (mm)	t_e (mm)	t_a (mm)	D (mm)
1	152.40	3.175	3.175	6.35	177.80
2	152.40	4.7625	4.7625	19.05	209.55
3	152.40	6.35	6.35	31.75	241.30

The sandwich pipe model was developed with an initial imperfection around the pipe perimeter (w_0), which is defined as the difference between the nominal and the actual external radius and distributed around the transversal section according to the following equation:

$$w_o(\theta) = -\varDelta_o \frac{D}{2}\cos 2\theta \tag{1}$$

where θ is the polar coordinate and Δ_0 is the initial out-of-roundness. The initial out-of-roundness (Δ_0) is given by:

$$\Delta_o = \frac{D_{max} - D_{min}}{D_{max} + D_{min}} \tag{2}$$

where D_{max} and D_{min} are the maximum and minimum external diameters, respectively. Two different values of Δ_0 were adopted, a very small value and 0.5%.

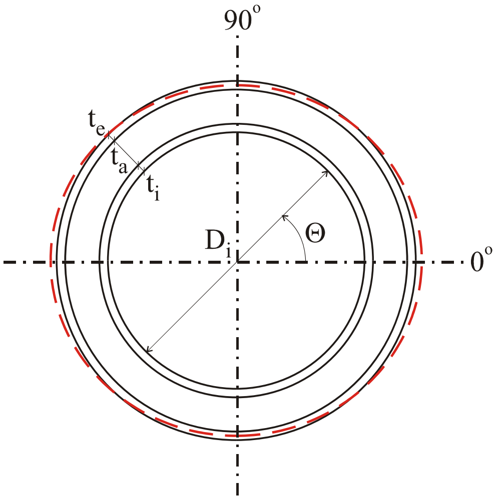


Figure1. Sketch of the sandwich pipe section, indicating the initial imperfection by the dashed line

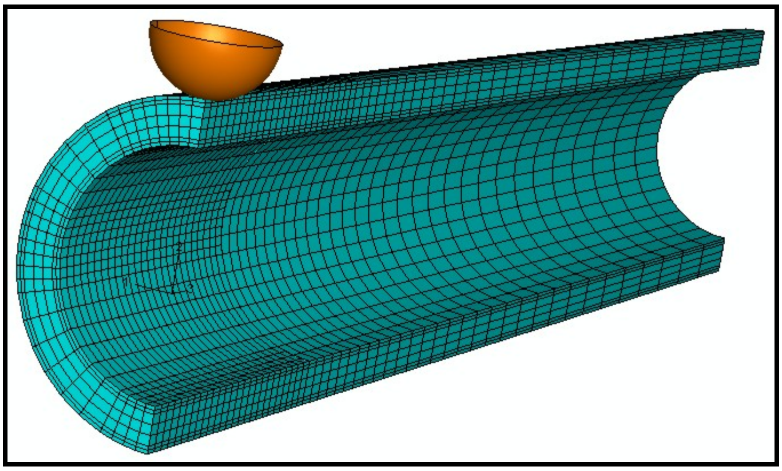


Figure 2. Sandwich pipe finite element mesh and the analytical rigid surface simulating the denting tool

2.2 Material Properties

To reproduce the mechanical behavior of a sandwich pipe, distinct material constitutive models were adopted in the numerical model. The inner and outer pipes were made of API X-60 grade steel and the annulus was considered to be filled with polypropylene. For the steel pipes, a plastic constitutive behavior was used within the potential flow rule, assuming the von Mises yield function under isotropic hardening. A modulus of elasticity and a Poisson ratio of, respectively, 206 GPa and 0.3 were adopted as elastic properties. Figure 3 shows the nominal true stress-strain curve for the API X-60 grade steel adopted in the numerical model. The polypropylene was modeled as a hyperelastic, incompressible material. A uniaxial tensile curve obtained for Moplen polypropylene (Castello and Estefen, 2006), employed in submarine pipe coatings, was used to calibrate a potential deformation energy model (Marlow model) available in ABAQUS (ABAQUS, 2006). Figure 4 shows the stress-strain curve obtained at ambient temperature for the Moplen polypropylene, whose density and maximum tensile strength are 0.9 and 22.8 MPa, respectively. Thermal effects on the material properties of polypropylene were not considered in this work; these effects will be taken into account in future works.

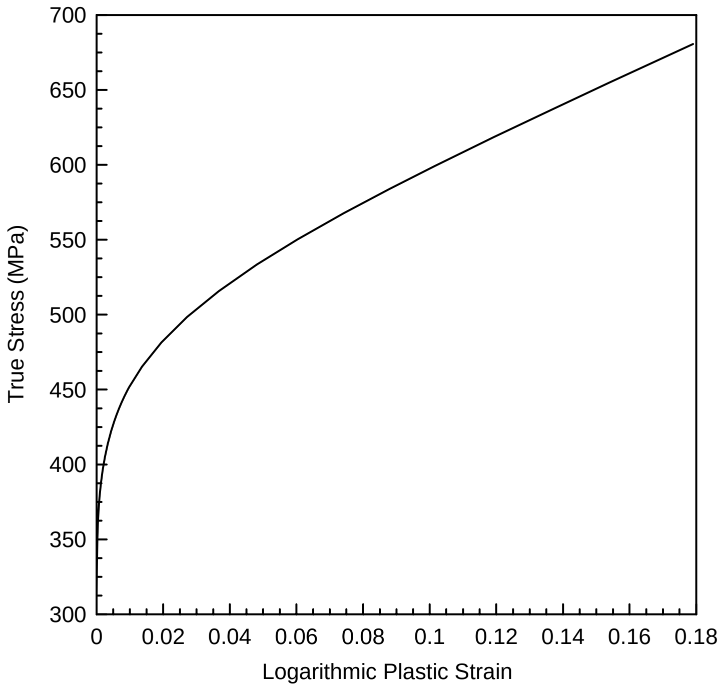


Figure 3. True stress versus logarithmic plastic strain of the API X-60 grade steel

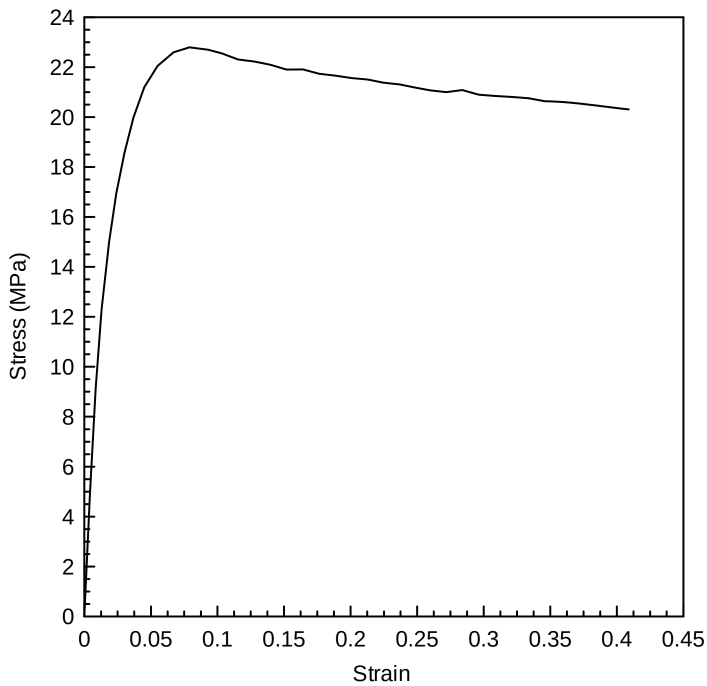
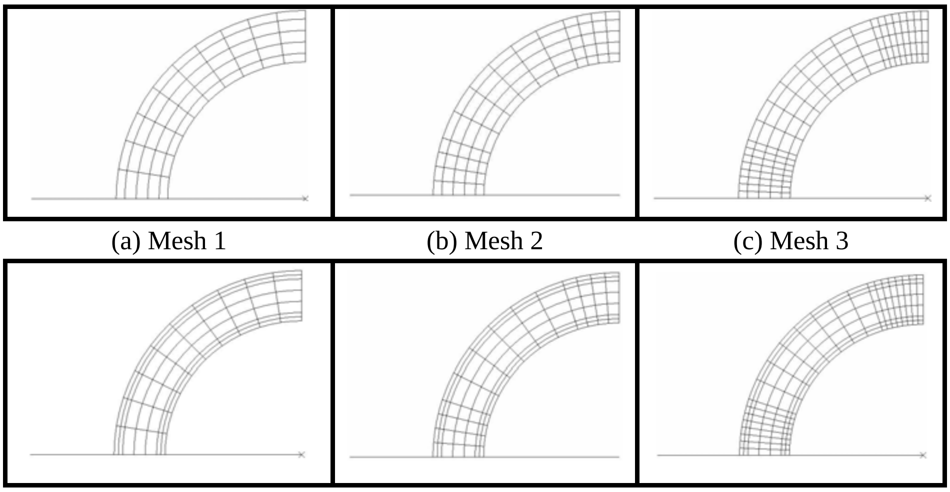


Figure 4. Stress-strain curve of Moplen polypropylene (ambient temperature)

2.3. Mesh Sensitivity Study

A mesh sensitivity study was carried out on a two-dimensional sandwich pipe model under external pressure to determine the mesh refinement in the circumferential and radial directions (1-2 plane). The objective of this study was to select a mesh refinement capable to accurately generate results of collapse pressure without the need of an excessive computational time in the numerical analyses. The three different sandwich pipe geometries, described in Tab. 1 (Cases 1, 2 and 3), were considered and six different mesh refinements were analyzed for each geometry (Meshes 1 to 6). In order to minimize the computational time in the numerical analyses, a quarter-symmetry model was used, adopting two planes of symmetry. The two-dimensional sandwich pipe model was developed with an initial imperfection around the pipe perimeter (w_0), prescribed according to Eq. (1), which induced an initial out-of-roundness (Δ_0) of 0.5%.

The two-dimensional model FE mesh for the inner and outer pipes was generated using ABAQUS CPE8 biquadratic plane strain element, with eight nodes and two translation degrees of freedom per node (ABAQUS, 2006). For the annular layer, this element type was used with a hybrid formulation (CPE8H) to account for the incompressible hyperelastic material behavior (ABAQUS, 2006). For the inner and outer pipes, Meshes 1, 2 and 3 comprise one element in the radial direction, while Meshes 4, 5 and 6 present two elements in this direction. This refinement scheme in the radial direction is the same for the three considered sandwich pipe geometries. For the annular layer, two, three and four elements were used in the radial direction for the Cases 1, 2 and 3, respectively. Figure 5 shows the six different FE meshes analyzed in the mesh sensitivity study for the Case 2. Since large amount of plastic deformation is expected near the positions of 0º and 90º, the mesh is more refined around this area in the circumferential direction. The number of elements used in each analyzed mesh for the Case 2 is listed in Tab. 2. Both the no adhesion and the perfect adhesion conditions between the interfaces of the steel pipes and the annular layer were considered. The perfect adhesion condition is obtained by sharing the nodes along the interfaces.



(a) Mesh 1 (b) Mesh 2 (c) Mesh 3

(d) Mesh 4 (e) Mesh 5 (f) Mesh 6

Figure 5. Finite element meshes analyzed in the mesh sensitivity study (Case 2)

Table 2. Number of elements for each FE mesh (Case 2)

Mesh	Number of Elements		
	Circumferential Direction	Radial Direction	Total
1	10	5	50
2	14	5	70
3	24	5	120
4	10	7	70
5	14	7	98
6	24	7	168

Table 3. Results of the mesh sensitivity study for the no adhesion condition

Case	Mesh 1		Mesh 2		Mesh 3		Mesh 4		Mesh 5		Mesh 6	
	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)
1	5.70	88.50	5.67	110.00	5.58	269.00	5.70	93.70	5.58	165.10	5.55	290.40
2	14.40	116.80	14.38	128.50	14.24	273.00	14.40	94.00	14.32	232.20	14.24	292.80
3	25.62	90.50	25.62	134.10	25.48	284.30	25.50	94.20	25.48	261.80	25.44	293.50

Table 4. Results of the mesh sensitivity study for the perfect adhesion condition

Case	Mesh 1		Mesh 2		Mesh 3		Mesh 4		Mesh 5		Mesh 6	
	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)	P_{co} (MPa)	CPU Time (s)
1	20.00	68.00	20.00	150.30	19.92	201.40	19.98	89.90	19.92	183.20	19.20	221.40
2	39.56	83.60	39.56	151.00	39.40	223.00	39.44	100.00	39.40	208.00	39.36	247.80
3	56.52	126.30	56.52	154.80	56.28	236.00	56.46	156.60	56.36	221.70	56.28	252.00

The results of the collapse pressure (P_{co}) obtained by the mesh sensitivity study are shown in Tabs. 3 and 4 for the no adhesion and the perfect adhesion conditions, respectively. As can be observed for each adhesion condition, the values of collapse pressure obtained for the six meshes are very similar, showing no significant difference between them. The Mesh 5 was selected considering that its refinement can accurately generate results of collapse pressure without an excessive computational time in the numerical analyses.

2.4. Finite Element Mesh

The three-dimensional model FE mesh for the inner and outer pipes was generated using ABAQUS C3D8 eight-node hexaedrical linear element, with three translation degrees of freedom per node (ABAQUS, 2006). For the annular layer, this element type was used with a hybrid formulation (C3D8H) to account for the incompressible hyperelastic material behavior (ABAQUS, 2006). The mesh refinement in the circumferential and radial directions (1-2 plane) was determined according to the mesh sensitivity study on a two-dimensional sandwich pipe model under external pressure.

The mesh is divided into two parts with different levels of refinement. The mesh is more refined in the region in which the contact with the denting tool is expected to occur, defined within a length equal to D from the pipe middle section. This finer mesh comprises twenty five elements along the axial direction and thirty elements around the circumference, considering that a change in the two-dimensional Mesh 5 was done to keep an aspect ratio of 1 to 1 in the region were higher stress gradients are induced during the denting process. The coarser part of the mesh, defined within a length equal to $4D$, comprises twenty elements around the circumference and thirty elements in the axial direction, smoothly decreasing the refinement level in the axial direction until the pipe edge. The refinement of the coarser mesh in the circumferential and radial directions is not equivalent to Mesh 5. The coarser and finer meshes were joined through a tie constraint. Figure 2 shows the model FE mesh used for the sandwich pipe geometry referred as Case 2.

2.5. Boundary Conditions and Loading

To simulate the symmetry conditions, the normal displacements of the nodes on plane 1-2 and on plane 2-3 at $x_1 = 0$ were constrained. An axially constrained condition was applied at the pipe edge, simulating a large pipe length. The nonlinear analysis was comprised of three load steps, simulating the denting process, the elastic return (spring back) and the application of external (hydrostatic) pressure. To simulate the mechanical damage, the displacement of the denting tool (analytical rigid surface) was prescribed in order to generate different magnitudes of damages, equivalent to $0.1D_i$, $0.25D_i$ and $0.5D_i$. The hydrostatic pressure simulation was done using the Riks method (arc-length control).

3. RESULTS

Figure 6 shows finite element results of the von Mises stress at the end of the denting step for the sandwich pipe geometry referred to Case 2, with an initial out-of-roundness (Δ_0) of 0.5%, and a denting tool displacement of $0.5D_i$. The stress level verified at the metal layers around the damaged region is considerable larger than the material yield stress (413 MPa). The same is not verified for the polypropylene layer, where the stress level is inferior to the necking stress observed in Fig. 4. It shows that the polypropylene still behaves elastically even after the denting step.

Tables 5, 6 and 7 present the obtained FE results for Cases 1, 2 and 3, respectively, where d_i is the denting tool displacement, d is the residual dent depth (after spring back) and P_i is the initiation pressure of the damaged sandwich pipe. Here, two different bonding conditions between metal and polypropylene, perfect adhesion and no adhesion, were evaluated. Also, the effect of different initial out-of-roundness (0.5% and infinitely small) were investigated.

Figures 7 and 8 present the finite element results of the initiation pressure versus the dent depth related to the internal diameter of the inner pipe (d/D_i) for the no adhesion and the perfect adhesion conditions, respectively. From the obtained results, it is observed that the initiation pressure of damaged sandwich pipes is directly related to the magnitude of the damage and the condition of adhesion between the steel pipes and the annular layer. Considering the same geometry and adhesion condition, as the dent depth increases, the initiation pressure is decreased. Additionally, higher initiation pressures were obtained for the perfect adhesion condition. It can be noted that, for the same adhesion condition, the initiation pressure of the sandwich pipe is increased when the pipe stiffness is reduced. The results of initiation pressure obtained for the geometry referred to Case1 (thinner pipe) are higher than those obtained for Cases 2 and 3 for both adhesion conditions. The effect of initial out-of-roundness on the initiation pressure is small but very interesting. For both adhesion conditions, the initial out-of-roundness of 0.5 decreases the initiation pressure, as verified for Cases 1 and 2. Inversely, it increases P_i for the thicker annulus (Case 3).

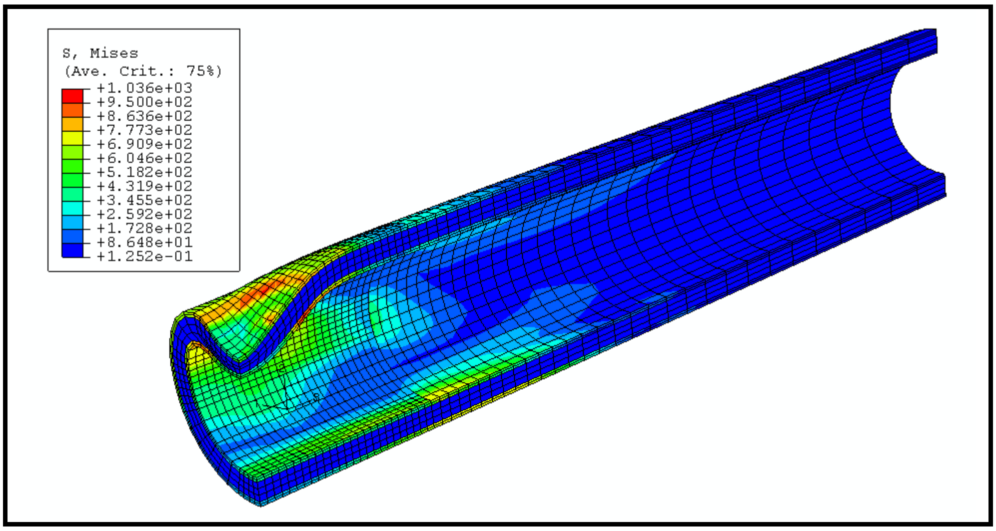


Figure 6. Finite element results of von Mises stress at the end of the denting step (Case 2 and Δ_0 = 0.5%)

Table 5. Finite element results for Case 1

Δ_o	d_i	d_i (mm)	No Adhesion			Perfect Adhesion		
			d (mm)	d/D_i (%)	P_i (MPa)	d (mm)	d/D_i (%)	P_i (MPa)
Infinitely Small	0.10D_i	15.24	13.77	9.03	19.14	12.71	8.34	27.57
	0.25D_i	38.10	33.73	22.13	13.74	38.94	25.55	19.77
	0.50D_i	76.20	67.26	45.44	8.43	67.03	43.99	14.01
0.5%	0.10D_i	15.24	13.72	9.00	18.00	13.27	8.71	26.70
	0.25D_i	38.10	33.74	22.14	13.29	33.14	21.74	19.45
	0.50D_i	76.20	67.29	44.15	8.31	67.06	44.00	14.05

Table 6. Finite element results for Case 2

Δ_o	d_i	d_i (mm)	No Adhesion			Perfect Adhesion		
			d (mm)	d/D_i (%)	P_i (MPa)	d (mm)	d/D_i (%)	P_i (MPa)
Infinitely Small	0.10D_i	15.24	12.44	8.16	34.16	12.98	8.52	47.94
	0.25D_i	38.10	33.18	21.77	25.00	33.08	21.71	36.60
	0.50D_i	67.20	68.19	44.75	16.80	66.09	43.37	27.56
0.5%	0.10D_i	15.24	12.41	8.14	29.16	12.96	8.50	46.76
	0.25D_i	38.10	33.15	21.75	23.28	33.00	21.65	36.19
	0.50D_i	76.20	68.16	44.72	16.32	66.03	43.32	27.37

The same results are presented in Figs. 9 and 10 with normalized values of initiation pressures, referred to collapse pressure of intact sandwich pipes. The collapse pressures used to normalize the initiation pressures, given in Tab. 8, were obtained from numerical analyses using the two-dimensional model (developed for the mesh sensitivity study). As expected, the initiation pressure is smaller than the collapse pressure of the perfect geometry for both bonding conditions, when the initial out-of-roundness is infinitely small. Conversely, the same is not verified for the no adhesion condition, when Δ_0 is equal to 0.5%. Also, for perfect adhesion and small damages. It can be explained by the collapse pressure of the sandwich pipe, which is strongly reduced by the increase of the initial out-of roundness, mainly when no adhesion is taken into account (see Tab. 8). As mentioned, the influence of the initial out-of-roundness on the initiation pressure is very small, since the induced damage is much larger that the initial out-of-roundness. The collapse mode can also explain this phenomenon. The damaged sandwich pipe collapses under the U-shape failure mode, while the ovalized pipe collapses under the dog-bone failure mode. The former requires more energy than the later, which explains higher collapse pressures.

Table 7. Finite element results for Case 3

Δ_o	d_i	d_i (mm)	No Adhesion			Perfect Adhesion		
			d (mm)	d/D_i (%)	P_i (MPa)	d (mm)	d/D_i (%)	P_i (MPa)
Infinitely Small	$0.10D_i$	15.24	11.39	7.48	41.52	16.76	11.00	58.56
	$0.25D_i$	38.10	32.52	21.34	31.68	36.10	23.69	45.30
	$0.50D_i$	76.20	74.34	48.78	22.98	67.62	44.37	37.02
0.5%	$0.10D_i$	15.24	11.33	7.44	42.30	12.46	8.18	66.69
	$0.25D_i$	38.10	32.50	21.32	35.46	33.08	21.70	52.47
	$0.50D_i$	76.20	67.53	44.31	26.64	67.97	44.60	38.79

Table 8. Collapse pressures (two-dimensional model) used to normalize the initiation pressures

Case	P_{co} (MPa)			
	Infinitely Small Δ_o		Δ_o = 0.5%	
	No Adhesion	Perfect Adhesion	No Adhesion	Perfect Adhesion
1	26.43	37.65	5.67	25.25
2	44.88	64.90	15.60	46.65
3	65.20	89.10	29.00	64.90

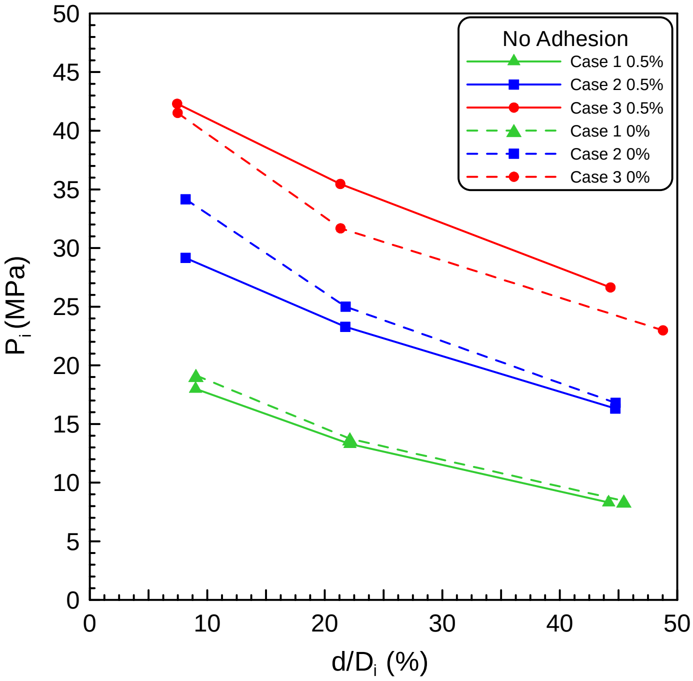


Figure 7. P_i versus dent depth (d/D_i) for the no adhesion condition

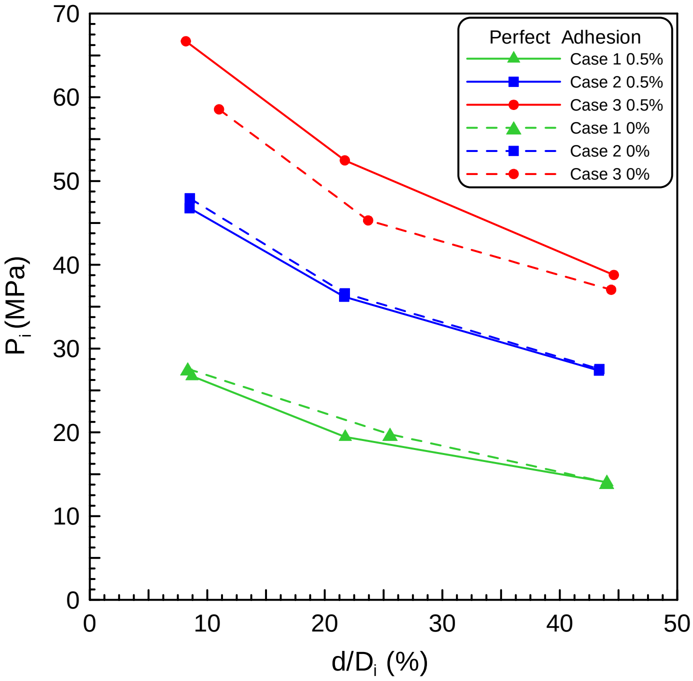


Figure 8. P_i versus dent depth (d/D_i) for the perfect adhesion condition

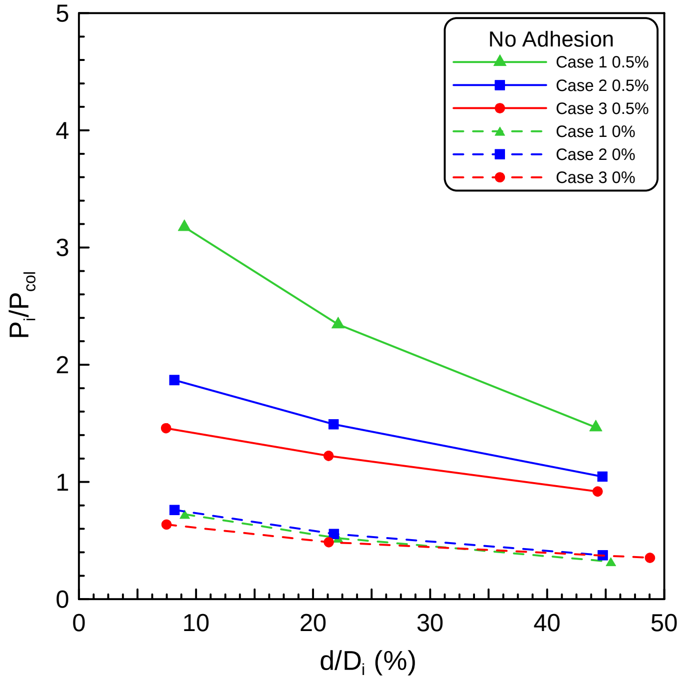


Figure 9. P_i/P_{co} versus dent depth (d/D_i) for the no adhesion condition

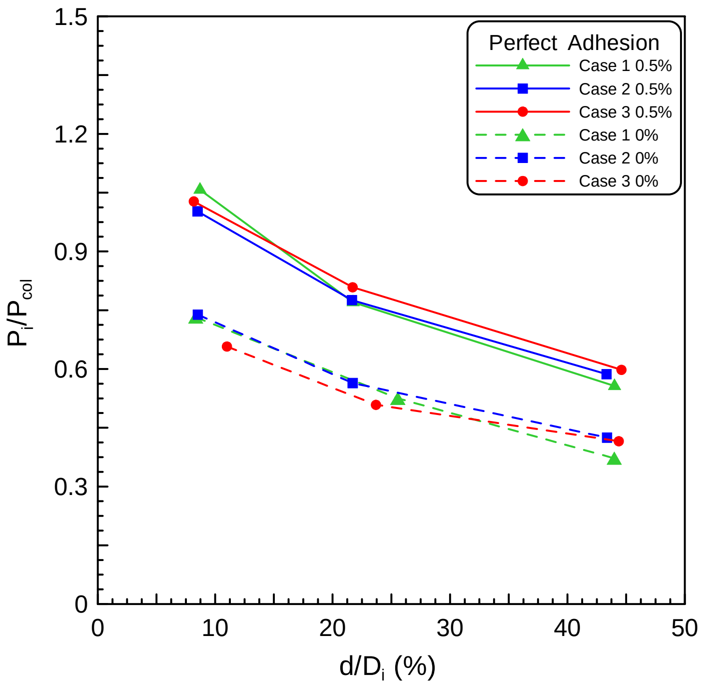


Figure 10. P_i/P_{co} versus dent depth (d/D_i) for the perfect adhesion condition

4. CONCLUSIONS

The reduction of the collapse pressure of sandwich pipes under external pressure caused by mechanical damage is numerically studied. A finite element model is developed to reproduce the mechanical behavior of a sandwich pipe subjected to the introduction of mechanical damage (denting process) and the application of external pressure. The sandwich pipe is modeled with two concentric API X-60 grade steel pipes and an annular layer filled with polypropylene. Using the numerical model, a parametric study is carried out to determine the collapse pressure for different sandwich pipe geometries and dent depths. Two different adhesion conditions between the interfaces of the different layers were assumed: no adhesion and perfect adhesion. The effect of the initial out-of-roundness is also investigated.

Based on the obtained numerical results, it is observed that the initiation pressure of damaged sandwich pipes is directly related to the magnitude of the damage and the condition of adhesion between the steel pipes and the annular layer. Considering the same geometry and adhesion condition, as the dent depth increases, the initiation pressure is decreased. Additionally, higher initiation pressures were obtained for the perfect adhesion condition. It can be noted that, for the same adhesion condition, the initiation pressure of the sandwich pipe is increased when the pipe stiffness is reduced.

The effect of initial out-of-roundness on the initiation pressure is small but very interesting. For both adhesion conditions, it decreases the initiation pressure. This tendency is inverted for thicker annulus, when the initial ovalization slightly increases the initiation pressure.

As expected, the initiation pressure is smaller than the collapse pressure of the perfect geometry for both bonding conditions, when the initial out-of-roundness is infinitely small. Conversely, the same is not verified for the no adhesion condition, when a ovalization of 0.5% is considered. It can be explained by the collapse mode. The damaged sandwich pipe collapses under the U-shape failure mode, while the ovalized pipe collapses under the dog-bone failure mode. The former requires more energy than the later, which explains higher collapse pressures.

Thermal effects on the material properties of polypropylene will be considered in a future work. Additionally, the annular layer will be assumed to be filled with a different material. In this case, PEEK *(polyetheretherketone)* will be adopted to carry out future numerical analyses.

5. ACKNOWLEDGEMENTS

The authors would like to acknowledge the financial support from the Human Resources Program of the National Petroleum Agency (ANP) and the Submarine Technology Laboratory of COPPE/UFRJ for the development of this research work.

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