A simple methodology to repair localized corrosion damage in metallic pipelines with epoxy resins

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Abstract

The present work is concerned with the analysis of epoxy repair systems for metallic pipelines undergoing elastic or inelastic deformations with localized corrosion damage that impair the serviceability. In the case of trough-thickness damage, the main focus is to assure an adequate application of the epoxy filler in such a way the pipe wont leak after repair. Such a procedure can be used or not associated with a composite sleeve that assures a satisfactory level of structural integrity. Examples concerning the use of repair systems in different damage situations are presented and analyzed showing the possibilities of practical use of the proposed methodology.

Keywords: corroded pipelines, epoxy repair systems, polymer matrix composites.

1 Introduction

Corroded pipelines with part-wall metal loss defects can be repaired or reinforced with a composite sleeve system. In these systems, a piping or vessel segment is reinforced by wrapping it with concentric coils of composite material after the application of epoxy filler in the corrosion defect. Nevertheless, so far, composite repair systems are not effective for through-thickness corrosion defects because generally they cannot avoid leaking. Information about requirements and recommendations for the qualification, design, installation, testing and inspection for the external application of composite repairs to corroded or damaged pipeline in petroleum, petrochemical and natural gas industries can be found in Jaske et al [1] or in the ISO Technical Specification 24817 [2]. Composite repair systems (patches) are also used in aircraft industry to repair cracks in order to extend the service life of metallic components [3, 4]. In this case, the size of the patch and bonding properties are very important. In the case of

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corroded pipelines conveying liquids, the geometry of the composite repair is simpler (a sleeve), but the main difficulties are the definition of the adequate composite thickness to assure a satisfactory level of structural integrity and to avoid leaking in the case of through-thickness defects.

The present paper is concerned with the analysis of epoxy repair systems for metallic pipelines undergoing elastic or inelastic deformations with localized corrosion damage that impair the serviceability. In the case of through-wall corrosion damage, the focus is to assure that the pipe wont leak after repair. The main motivation for the study presented on this paper are corrosion defects in produced water pipelines used in offshore oil exploitation. Since offshore platforms are hydrocarbon atmospheres, any repair method using equipment that may produce heat and/or sparkling is forbidden.

The damages derived from corrosion process in produced water pipelines in platforms cause very important economical losses because the operation must be stopped while the repair is being performed. The rehabilitation of this kind of corroded pipeline may eventually require an industrial climber and hence the application of the repair system must be as simple as possible. Although the operation pressure of these pipelines is not very high, the water temperature is between 60°C to 90°C, which can be a major shortcoming for the use of polymeric material as repair systems. Initially, it is presented in this paper a simple methodology to define the necessary thickness of the composite sleeve to assure the safe operation of corroded pipelines with part-wall metal loss defects. As a second step, it is presented a complementary procedure to repair through-thickness corrosion defects in pipelines using epoxy resins. The objective is to assure the pipeline wont leak under the operation pressure and temperature. Hydrostatic tests were carried out with water at room temperature and at 80°C to validate epoxy repair systems applied in offshore produced water pipelines. Examples concerning the use of repair systems in different damage situations are presented and analyzed showing the possibilities of practical use of the proposed methodology.

2 Mechanical analysis of composite sleeve reinforcement systems

The present section is concerned with the analysis of composite sleeve reinforcement systems for metallic pipelines undergoing elastic or inelastic deformations with localized part-wall metal loss that impair the serviceability. Different commercial repair systems based in fiber reinforced composite materials can be found: (a) dry fiberglass fabric to be wrapped with impregnation of liquid resin, (b) ready pre-cured layers ready to wrap around the pipe, (c) Flexible resin pre-impregnated bandage to be wrapped with water. No matter the application procedure, the basic idea of the reinforcement technique is to transfer the hoop stress in the pipe wall due to the internal pressure to the composite sleeve.

Most of the studies about these systems are concerned with the materials (matrix, fibers, adhesive) and application procedures. Only a few studies are concerned with the mechanical analysis of the repair system [5–9]. The main goal of this section is to summarize a new methodology, as simple as possible, to define the minimum thickness of composite material to assure: (a) the safety of repairs under operation conditions and/or (b) the lifetime extension under operation conditions. Such methodology, although simple, is able to account for different failure mechanisms (plasticity, corrosion, etc.).

2.1 Basic model – Pipe without localized damage

In a first step towards a simplified mechanical analysis of composite sleeve reinforcement systems, no localized imperfections or damage are considered. The pipe-composite sleeve system is modeled as two concentric cylinders, open at the ends, under internal pressure – an internal thin-walled cylinder with elastic-plastic behavior and a sleeve with orthotropic elastic behavior. The internal cylinder has an inner radius r_i and external radius r_o. The cylinder can be considered thin-walled if the wall thickness t is less than about $1/10$ of the internal radius ($t < r_i/10$). The sleeve has an internal radius r_o and external radius r_e. The system is subjected to an internal pressure P_i as shown in Fig.1.

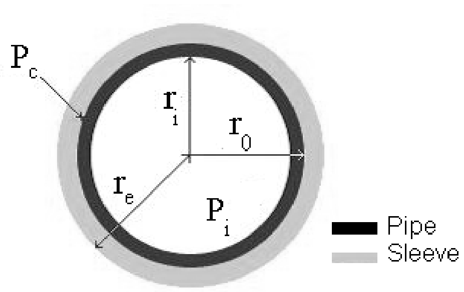

Figure 1: Pipe and sleeve with internal pressure.

The contact pressure between the pipe and the sleeve will be noted P_c. Assuming that the radial displacement in the contact surface is the same for both cylinders, it is possible to obtain analytical expressions for the stress, strain and displacement fields. With this expressions, it can be obtained the minimum composite sleeve thickness in order to verify a given safety criterion.

Generally unidirectional glass reinforced epoxy is used for the sleeve (epoxy resin is the matrix and the reinforcement is glass fiber). Neglecting a decrease in time of the polymer composite strenght due to the environment, a closed-form of the radial displacement u_r in the sleeve can be obtained (see appendix)

$$u_r(r) = -BKr^{-K}\left(\frac{1}{E_{\theta\theta}} + \frac{\nu_{r\theta}}{E_{rr}}\right) + CKr^{K}\left(\frac{1}{E_{\theta\theta}} - \frac{\nu_{r\theta}}{E_{rr}}\right) \quad (1)$$

With

$$K = \sqrt{\frac{E_\theta}{E_r}} \quad (2)$$

$$B = \left(\frac{-P_c r_e^{(K-1)}}{\left[r_o^{-(K+1)} r_e^{(K-1)} - r_o^{(K-1)} r_e^{-(K+1)}\right]}\right) \quad (3)$$

$$C = \left(\frac{-P_c r_e^{-(K+1)}}{\left[r_o^{-(K+1)} r_e^{(K-1)} - r_o^{(K-1)} r_e^{-(K+1)} \right]} \right) \tag{4}$$

P_c is the contact pressure between the sleeve and pipe, E_θ the extensional modulus in the tangential direction and E_r the extensional modulus in the radial direction and $\nu_{r\theta}$ the coefficient relating contraction in the circumferential direction to extension in the radial direction. The radial displacement in the sleeve is a function of the contact pressure P_c which is not known "a priori". If the pipe wall is thin, it can be shown that the stress component σ_θ and the radial displacement u_r are approximated by the following expressions (see appendix)

$$\sigma_\theta = \frac{P_i r_i - P_c r_o}{r_o - r_i} \tag{5}$$

$$u_r = r \left[\frac{\sigma_\theta}{E} + \left\langle \frac{\sigma_\theta - \sigma_y}{K} \right\rangle^{\frac{1}{N}} \right] \tag{6}$$

where E is the Young modulus of the pipe material, σ_y the yield stress, K and N are material parameters that characterize the plastic behavior of the material. K is the coefficient of plastic resistance and N is the hardening exponent. The angular brackets have the following meaning: $\langle x \rangle = \max\{0, x\}$. The term (σ_θ / E) corresponds to the elastic deformation and the term $\left\langle \frac{\sigma_\theta - \sigma_y}{K} \right\rangle^{1/N}$ to the plastic deformation in the pipe. From the practical point of view, it is important to define the sleeve thickness in order to assure a given maximum hoop stress criterion in the pipe

$$\sigma_\theta < \sigma_{\max} \text{ in the pipe} \tag{7}$$

In this case, since the minimum contact (external) pressure $(P_c)_{min}$ acting on the pipe necessary to assure (7) can be obtained analytically

$$\sigma_\theta = \frac{P_i r_i - (P_c)_{\min} r_o}{r_o - r_i} = \sigma_{\max} \Rightarrow (P_c)_{\min} = \frac{P_i r_i - \sigma_{\max}(r_o - r_i)}{r_o} \tag{8}$$

the minimum sleeve thickness to assure (7) may be obtained from the following compatibility condition

$$[u_r(r = r_o)]_{pipe} - \hat{u} = [u_r(r = r_o)]_{sleeve} \tag{9}$$

where $\hat{u}$ is the radial displacement of the external surface of the pipe due to the internal pressure P_{apl} the pipe was submitted to when the reinforcement was applied

$$\hat{u} = r_o \left[\frac{\hat{\sigma}_\theta}{E} + \left\langle \frac{\hat{\sigma}_\theta - \sigma_y}{K} \right\rangle^{\frac{1}{N}} \right] \text{ with } \hat{\sigma}_\theta = \frac{P_{apl}\ r_i}{r_o - r_i} \tag{10}$$

Compatibility condition (9) assures that the radial displacement of outer surface of the pipe and the inner surface of the sleeve must be the same (contact surface). Using this condition and considering eqs. (1) - (4) and (8) it comes that the minimum external radius $(r_e)_{\min}$ to assure the constraint (7) is the root of the function Φ

$$\Phi(r_e) = r_o \overbrace{\left[\frac{\sigma_{\max}}{E} + \left\langle\frac{\sigma_{\max}-\sigma_y}{K}\right\rangle^{\frac{1}{N}}\right]}^{[u_r(r=r_o)]_{pipe}} - \hat{u} - \tag{11}$$
$$- \overbrace{\left[B(r_e)\sqrt{\frac{E_\theta}{E_r}}\, r_o^{-\sqrt{\frac{E_\theta}{E_r}}}\left(\frac{1}{E_\theta}+\frac{\nu_{r\theta}}{E_r}\right) + C(r_e)\sqrt{\frac{E_\theta}{E_r}}\, r_o^{\sqrt{\frac{E_\theta}{E_r}}}\left(\frac{1}{E_\theta}-\frac{\nu_{r\theta}}{E_r}\right)\right]}^{[u_r(r=r_o)]_{sleeve}}$$

Where

$$B(r_e) = \left(\frac{-(P_c)_{\min} r_e^{(K-1)}}{\left[r_o^{-(K+1)} r_e^{(K-1)} - r_o^{(K-1)} r_e^{-(K+1)}\right]}\right) \tag{12}$$

$$C(r_e) = \left(\frac{-(P_c)_{\min} r_e^{-(K+1)}}{\left[r_0^{-(K+1)} r_e^{(K-1)} - r_0^{(K-1)} r_e^{-(K+1)}\right]}\right) \tag{13}$$

$$(P_c)_{\min} = \frac{P_i r_i - \sigma_{\max}(r_0 - r_i)}{r_0} \tag{14}$$

The internal pressure P_{apl} the pipe is submitted to when the sleeve is applied is one of the most important variables in the application of reinforcement systems. The wrong choice of this pressure may result in ineffective reinforcements as it is discussed in Costa-Mattos et al [8]. If $P_{apl\mathrm{B}B}$ is closer to P_i, the reinforcement must be very thick and will only share hoop stresses with the sleeve when a pressure surge above the value P_i occurs. Most of commercial repair systems recognize that reducing pressure during repair is a good practice but this pressure reduction is not quantified and is not a mandatory requirement. The choice of $\sigma_{\max}$ is also very important in order to define the role of the sleeve reinforcement [8]. The most obvious choice is the von Mises criterion. The pipe won't be submitted to permanent deformation provided the hoop stress is smaller than the yield stress

$$\sigma_\theta < \sigma_y \Rightarrow \sigma_{\max} = \sigma_y \tag{15}$$

2.2 Accounting for a localized corrosion damage

The expressions presented up to now are valid only if there are no localized imperfections or damage in the pipe section. In this section, a simple procedure to account for a localized damage is proposed. The basic idea is to suppose the maximum hoop stress close to a localized imperfection can be approximated considering the tangential stress for an undamaged cylinder corrected by a factor η which is a function of the geometry

$$\sigma_\theta = \eta\ (\sigma_\theta)_{undamaged} = \eta\ \frac{P_i r_i - P_c r_o}{r_o - r_i} \tag{16}$$

Most "engineering safety criteria" for thin-walled pipelines consider expressions similar to (16) and the only basic difference is the definition of how the "correction factor" η depends on the geometry. Generally these criteria have the following form

$$\sigma_\theta = \eta\ \frac{P_i r_i - P_c r_o}{r_o - r_i} < \hat{\sigma}_{\max} \tag{17}$$

where $\hat{\sigma}_{\max}$ is a material constant. If a criterion like (17) is considered, the same equations proposed in the last section can be used taking $\sigma_{\max} = \frac{\hat{\sigma}_{\max}}{\eta}$

$$\sigma_\theta = \frac{P_i r_i - P_c r_o}{r_o - r_i} < \frac{\hat{\sigma}_{\max}}{\eta} = \sigma_{\max} \tag{18}$$

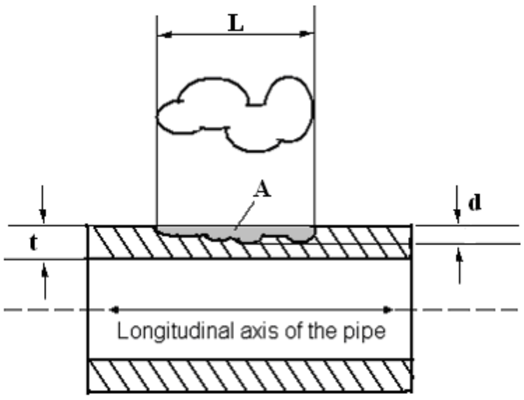

Figure 2: Metal loss in the pipe.

The adequate choice of the sleeve thickness assures that the maximum hoop stress verifies safety condition (17). The most widely used criteria for assessment of corrosion defects under internal pressure loading is a family of criteria described in Stephens and Francini [10] as the effective area methods. These include the ASME B31G criterion [11], the RSTENG 0.85 criterion (also Known as the modified B31G criterion). These criteria were developed in the beginning of the late 1960s and early 1970s to evaluate the serviceability of corroded gas transmission lines. The basic empirical assumption is that the strength loss due to corrosion is proportional to the amount of metal loss measured axially along the pipe. Other approaches can be considered (damage mechanics, assume that corrosion defects are blunt and hence they all fail by plastic collapse, etc.) but will not be discussed in the present paper. Other studies can be found in literature but in all of then the resulting metal loss is treated as a part-through defect in the pipe [12].

The effective area methods assume that the maximum depth profile lies in one plane along the axis of the pipe. To accommodate the irregular nature of most corrosion defects, a profile of the defect is measured and the deepest points are translated to a single axial plane for analysis, as illustrated in Fig. 2. These criteria may be expressed in the following form

$$\sigma_\theta < \bar{\sigma}\left[\frac{1-(A/A_o)}{1-(A/A_o)\left(M^T\right)^{-1}}\right] \Rightarrow \sigma_{\max} = \bar{\sigma}\left[\frac{1-(A/A_o)}{1-(A/A_o)\left(M^T\right)^{-1}}\right] \tag{19}$$

A is the area of defect in the longitudinal plane through the wall thickness, $A_o = Lt$ is the original cross-sectional area, M^T is the Folias factor for a through-wall defect, $\hat{\sigma}_{\max} = \bar{\sigma}$ in this case is the "flow stress", which is a computed parameter that is between the material's yield stress and ultimate strength. The B31G criterion assumes conservatively that $\bar{\sigma} = 1.1\ \sigma_y$ and also that the corrosion defect has a parabolic shape (this approximation results in the expression $A = (2/3)Ld$). Lastly, the B31G criterion uses a simplified two-term form of the Folias bulging factor that is applicable to $\left(L/\sqrt{2r_i t}\right)^2 \leqslant 20.0$ and $(d/t) < 0.175$. Hence, from (19), it comes that

$$\begin{gathered}\sigma_\theta < 1.1\sigma_y\left[\frac{1-(2/3)(d/t)}{1-(2/3)(d/t)\left(M_1^T\right)^{-1}}\right];\ M_1^T = \sqrt{1+0.8\left(\frac{L}{2r_i t}\right)^2} \\ \Rightarrow \sigma_{\max} = 1.1\sigma_y\left[\frac{1-(2/3)(d/t)}{1-(2/3)(d/t)\left(M_1^T\right)^{-1}}\right]\end{gathered} \tag{20}$$

The modified B31G criterion may be expressed in the following form

$$\sigma_\theta < \bar{\sigma}\left[\frac{1-0.85(d/t)}{1-0.85(d/t)\left(M_2^T\right)^{-1}}\right] \Rightarrow \sigma_{\max} = \bar{\sigma}\left[\frac{1-0.85(d/t)}{1-0.85(d/t)\left(M_2^T\right)^{-1}}\right] \tag{21}$$

with

$$\bar{\sigma} = \sigma_y + 68.94\ \text{MPa} \tag{22}$$

In this criterion the flow stress $\bar{\sigma}$ is given by the less conservative expression (22). In addition, rather than the parabolic shape resulting in the "2/3" area factor, this criterion utilizes the more accurate three-term expression (22) for the Folias bulging factor. These changes result in less conservative and more reliable estimates of failure pressure than the B31G criterion.

3 Epoxy repair systems for localized trough-thickness corrosion damage

In the present section it is proposed a complementary procedure to repair leaking defects in pipes using epoxy resins. The objective is to assure the pipe wont leak under the operation pressure and temperature. Composite sleeves can assure a satisfactory level of structural integrity for part-through corrosion defects but are not necessarily effective to avoid leakage for localized through-thickness corrosion defects. The repair methodology proposed in this section can be used or not associated with a composite sleeve in order to improve the effectiveness of the epoxy repair system.

The experimental set up at the laboratory was conceived to approximate a real repair operation, where the resin has to be applied in field conditions (which affect the quality of the resulting epoxy repair).

3.1 Defect sizing

The defect sizing is important to define limits to an effective use of the repair procedure. The dimension of the defect should be determined by the smaller ellipse with one axis parallel to the axis of the pipe that fully contains the area of the flaw (see Fig 3).

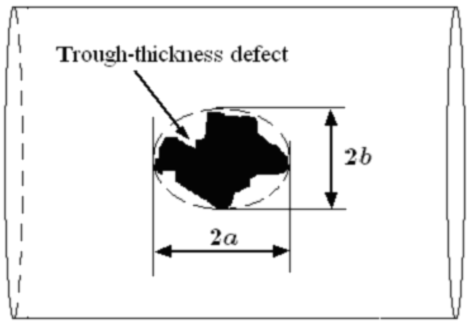

Figure 3: Defect sizing.

The maximum allowable defect size for the proposed repair procedure is defined by the semi major axis a of the ellipse which is given by:

$$a_{\max} \leqslant \max\left\{\frac{R}{10}, t\right\} \tag{23}$$

where R is the inner radius of the pipe and t the wall thickness. That means the maximum allowable dimension for the semi major axis a is the biggest value between the wall thickness t and $1/10$ of the inner radius R.

3.2 Proposed repair procedure

The repair methodology can be described as follows:

3.2.1 Surface preparation

Surface treatments often involve chemicals reactions, which produce surfaces modifications on adherends, or mechanical procedures, which improve adhesion by increasing mechanical interlocking of the adhesive to the adherend. By this way, the primary objective of a surface treatment is to increase the

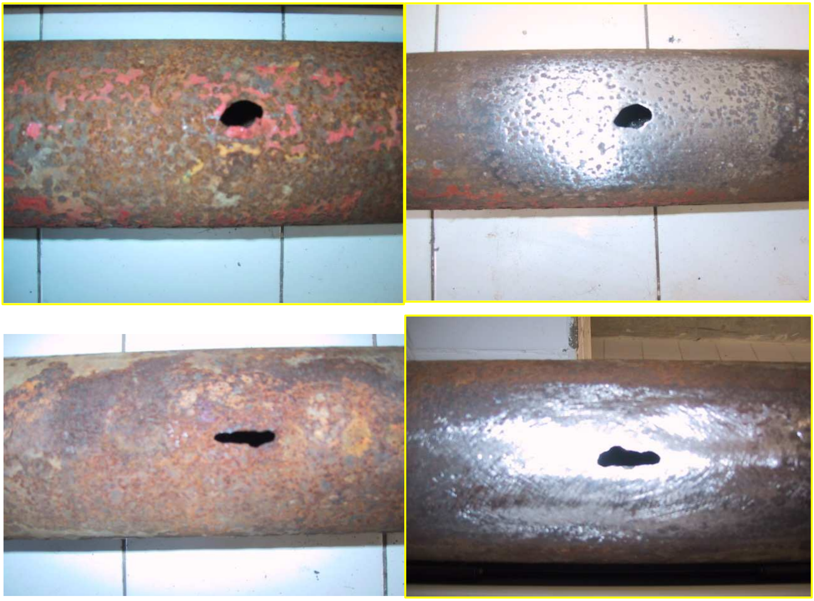

Figure 4: Surface preparation.

surface energy of the adherend as much as possible and/or improve the contact between the adhesive/adherend by increasing the contact area. Roughness or an increase in the surface area has been shown good results in improving adhesion. Subsequently, a relationship exists between good adhesion and bond durability.

In order to obtain the previous properties, sanding with sand paper 120 or 150 is used to achieve a white metal appearance and to remove some of the existing oxide layer. A final rinse with solvent was made to provide a free of oil, grease and dirt surface. After that the adhesive was mixed according to manufacture procedure and then the pipe was repaired. It is important to remark that, in a real situation, eventually the pipe is so corroded that sand paper should be used with extreme care (see Fig 4). Besides, since offshore platforms are hydrocarbon atmospheres, any method to mechanically rough up the surface (sandblasting, cutting, grinding) that may produce heat and sparkling is inadequate.

3.2.2 Introduction of a plastic cap inside the pipeline to avoid spilling of epoxy resin and application of the epoxy adhesive

A plastic cap with elliptical shape must be used to avoid resin to spill inside the pipe. Since the plastic material is very deformable, it is easy to introduce the cap inside the pipe. The cap is maintained in possition using a simple system of nylon strings (Fig. 5).

The cap should ensure internal layer of adhesive with approximately the same thickness of the

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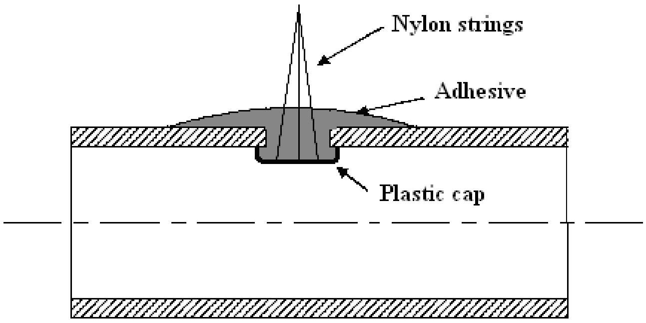

Figure 5: Plastic cap to avoid adhesive spilling.

pipe wall and with average dimension twice the size of the defect The Epoxy adhesive layer applied externally should have proximally 5 times the ellipse size. The thickness of this first layer must be at least equal to the thickness of the pipe. The epoxy layer should have a smooth boundary for improved performance and thickness higher than the pipe wall (see Fig 6). After application of the first layer, wait, according to manufacturer, the epoxy polymerization (maximum desirable 20 min), cut the the nylon strings at the surace and apply a second layer of adhesive without sanding the first one.

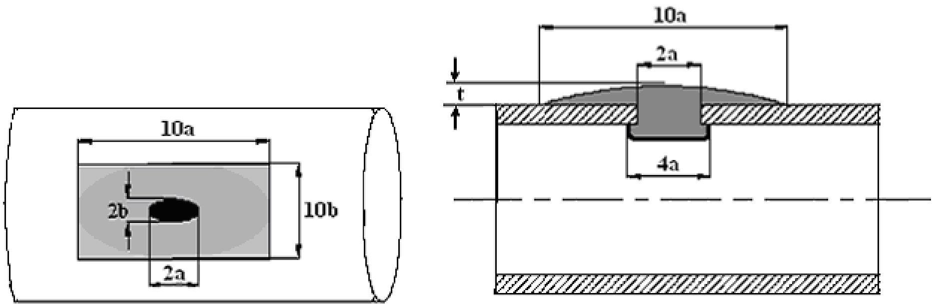

Figure 6: External epoxy adhesive layer.

For through-thickness defects with $a \leqslant 5mm$, it may be difficult to introduce the rubber cap and a metallic wedge should be used instead (Fig. 7). The following steps of the repair procedure are exactly the same if the wedge or caps are used.

3.2.3 Application of a composite sleeve

The repair procedure is considered safe, even without a composite sleeve when

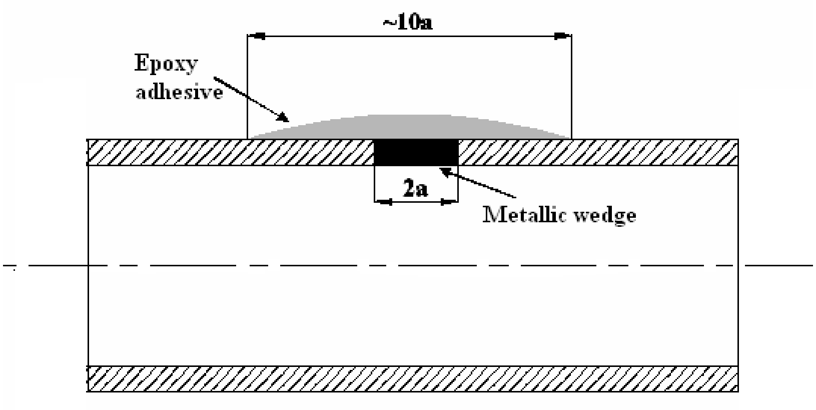

Figure 7: Metallic wedge for smaller defects.

$$\left(1+2\frac{a}{b}\right)\left(\frac{P_i r_i}{t}\right) \geqslant \sigma_y \tag{24}$$

where a and b are, respectively, the semi major and the semi minor axis of the ellipse (see section 3.1). r_i is the inner radius of the pipe, t the wall thickness and σ_y the yielding stress of the pipe material.

If constraint (24) is not verified, the suggestion is to apply the epoxy resin as described on this paper and then apply a composite material sleeve to restrain the plastic strain and to assure a satisfactory level of structural integrity. The adequate thickness of the sleeve is the biggest value between the thickness recommended by ISO/TS 24817 and the one obtained using the procedure presented in section 2 with

$$\sigma_{\max} = \frac{\sigma_y}{\left(1+2\frac{a}{b}\right)} \tag{25}$$

The stress distribution in a general through-thickness corrosion defect is very complex, but, if the size of the defect is limited, a rough estimative of the magnitude of the permanent deformations close to the defect can be performed. The term in the left side of Eq. (24) is the maximum stress in a thin-walled infinite plate with an elliptical defect with semi axis a and b subjected to traction of a uniform force per unit area $S = (P_i r_i/t)$ (see Fig. 8). The stress concentration factor in this case is $K_t = \left(1+2\frac{a}{b}\right)$. Criterion (24) states that the permanent deformation close to the defect in the pipe can be neglected when $K_t S$ is smaller than the yielding stress σ_y.

If (24) is verified, immediately after the application of the second epoxy layer, a rubber sheet should be applied over the repair around the perimeter and a simple metallic clamp, similar to those used for garden hoses, is attached (Fig. 9).

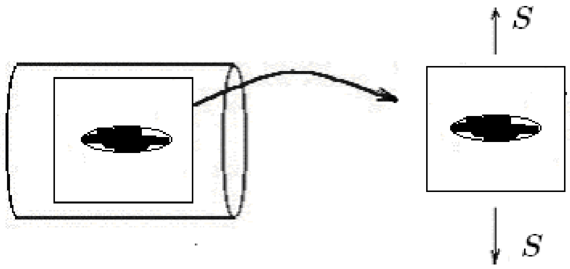

Figure 8: Equivalent system.

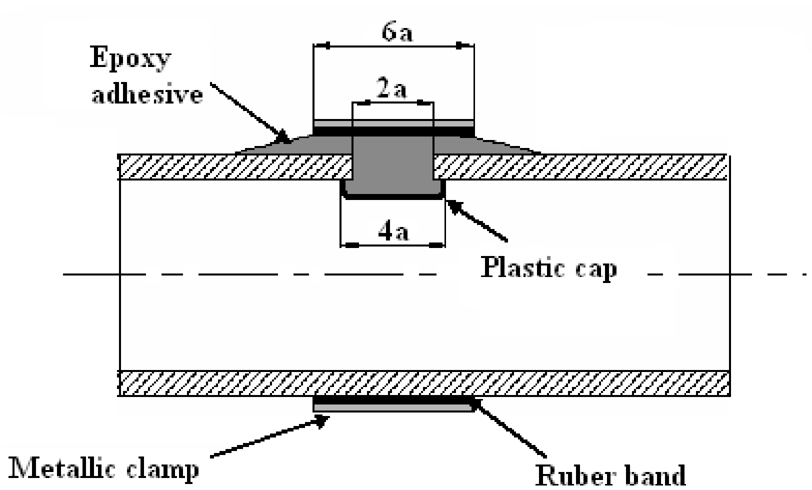

Figure 9: Complete repair system.

The clamp is not used to improve the level of structural integrity of the pipe, but to avoid the two major failure mechanisms of the adhesive repair shown in Fig. 10, mainly at the beginning of operation when the resin may not be fully cured.

In the following sections it is presented an analysis of the performance of two different commercial epoxy resins using the repair procedure defined in the previous sections.

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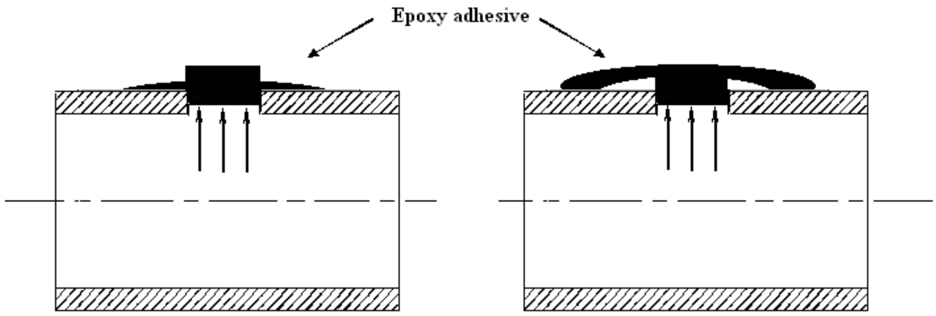

Figure 10: Types of brutal failures of the adhesive layer.

4 Materials and methods

Two different commercial fast curing epoxy resins were considered to perform the repair of through-thickness damages in metallic pipes using the methodology proposed in the previous section. Both are two-component systems consisting of a base and solidifier. The first one (System I) is conceived for leak repair on tanks and pipes as well as for other emergency applications. The product is based on a silicon steel alloy blended within high molecular weight polymers and oligomers and it is partly cured (machining and/or light loading) after 35 minutes at 25°C and it is fully cured after 1 hour at this temperature. Further technical data for System I is presented in table 1.

The second one (System II) is also a polymer-based system especially developed for repair consisting of a mixture of epoxy resin and aluminum powder and it is partly cured (machining and/or light loading) after 18 minutes at 25°C and it is fully cured after 40 minutes at this temperature. Further technical data for System II is presented in table 2.

The hydrostatic tests were performed at two different temperatures – room temperature and 80°C. An experimental set up was conceived to check the effectiveness of the methodology, trying to approximate a real repair operation. API 5L grade B steel pipes, normally used in offshore platform for produced water, were used to build the specimens for hydrostatic tests. Five different specimens were used:

1: 2"diameter schedule 80 pipe with a circular hole of 3 mm;
2: 2" diameter schedule 80 pipe with a circular hole of 10 mm;
3: 12" diameter, schedule 20, 1300 mm length pipe with a circular hole of 10 mm;
4: 12" diameter, schedule 20, 1300 mm length pipe with a circular hole of 30 mm;
5: 3,5" diameter, schedule 20, 1000 mm length pipes taken from the field with real corrosion defects (see Fig 4).

Initially all the repaired specimens (no composites sleeves were used) with the two systems were submitted to a classical hydrostatic test at room temperature to evaluate its strength and effectiveness. The maximum allowable time for each repair was 1 hour and all tests began exactly 1 hour and 15 minutes after the beginning of the procedure. In such tests, the pipe pressure is raised up to 30 kg/cm^2

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and held at this level for an hour. After five cycles, if the repair did not fail brutally, the specimen is unloaded and inspected to check eventual small leaks or reinforcement rebounding.

As a second step, the repaired specimens (no sleeves were used, only the clamp) were submitted to 5 pressure cycles (1 hour at 30 kg/cm^2) but the water temperature inside the specimen was 80ºC. This temperature level was chosen in order to simulate average offshore fluid condition. The fluid temperature of 80ºC is reached at atmospheric pressure. Only after temperature stabilization, the internal pressure is increased. After each pressure cycle the specimen is cooled to room temperature. Hence, each specimen is also submitted to 5 temperature cycles during testing. Once again the maximum allowable time for each repair was 1 hour and all tests began exactly 1 hour and 15 minutes after the beginning of the repair.

5 Results and discussion

All repairs performed with Systems I and II using the proposed methodology resisted the 5 pressure cycles with water at room temperature - 5 different tests/repairs for each specimen (1), (2), (3), (4) and (5). The repairs resisted a so high pressure level that it was not possible to obtain the failure pressure – the pipe caps were not conceived to burst tests and they deform plastically and fail before the repairs. Fig. 11 shows a type 5 specimen before and after the repair procedure (internal pressure of 30 kg/cm^2).

If the proposed procedure is not adopted (mainly the use of the plastic cap), the repair may not be able to resist the loading. Table 3 shows the failure pressure obtained for specimen (2) – 2" diameter schedule 80 pipe with a circular hole of 10 mm – repaired using system I if no cap and clamp are used.

All repaired pipes with system II at 80ºC resisted to 5 cycles. In order to decide whether a given epoxy system can be used at higher temperatures we suggest to use the same conditions adopted in the ISO Technical Specification 24817 for composite sleeves - For a design temperature greater than 40ºC the repair system shall not be used at a temperature higher than the glass transition (***Tg***) less 30ºC. For repair systems where a ***Tg*** cannot be measured, the repair system shall not be used above a heat distortion temperature (HDT) less 20ºC. For repair systems, which do not exhibit a clear transition point, i.e. a significant reduction in mechanical properties at elevated temperatures, then an upper temperature limit, ***Tm***, shall be defined (or quoted) by the repair supplier.

It is interesting to remark that the adhesive system A behaves surprisingly well if the proposed repair procedure is adopted, even at temperatures above the heat distortion temperature. All repairs resisted to 5 cycles at 80P^{oP}C in tests performed with the specimens (1), (2) and (3). Nevertheless, all repairs performed in pipes taken from the field with real corrosion defects (specimen type 5) failed.

6 Concluding remarks

The present work is a first step toward the definition of safer and more reliable procedures to apply epoxy repair systems in metallic pipelines with localized corrosion damage.

In the case of part-through corrosion damage, it is proposed a simplified methodology that can

Figure 11: 3,5" diameter, schedule 20, 1000 mm length pipe with real corrosion defect before and after repair.

be used as an auxiliary tool in the design of epoxy repair systems. This methodology can be helpful to define the pressure of application and sleeve thickness necessary to assure safer and more reliable repair systems.

In the case of through-thickness damage, the idea is to perform an adequate application of the epoxy filler in a way the pipe won't leak and to use eventually the composite sleeve as a complementary procedure that will assure a satisfactory level of structural integrity. The internal pressure of the fluid acting in the bottom of the repair and the external pressure done by the clamp (or the sleeve) dilate the epoxy system in the interior of the hole enhancing sealing. The suggestion is to apply the epoxy resin as described on this paper and then apply a composite material sleeve, with the normalized thickness, to restrain the plastic strain and to assure a satisfactory level of structural integrity. The main requirements to the epoxy resins to be used as repair systems for such kind of localized corrosion damage are: fast curing and high heat distortion temperature. The full validation of this simplified repair methodology still requires an extensive program of experimental investigation, mainly concerning repair lifetime: fatigue, creep, ageing, resistance to UV degradation and weathering.

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Appendix: Thin-walled elastic orthotropic and thin-walled elasto-plastic cylinders under pressure – closed-form expressions for stress, strain and displacements

A.1 Thick-walled elastic orthotropic cylinder under pressure

In the present study it is considered an elastic cylinder with inner radius r_i and external radius r_e submitted, respectively, to an internal pressure P_o and to an external pressure P_1 as shown in Fig. 12.

The model equations for this problem, using a cylindrical coordinates system are:

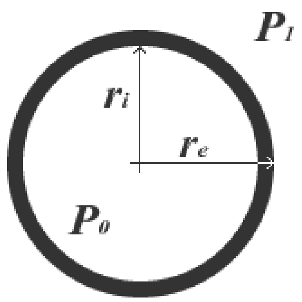

Figure 12: Pipe under external and internal pressure.

Balance of Linear momentum: Under the hypothesis of a plane state of stress and neglecting body forces, the balance of linear momentum for a pipe in static equilibrium can be expressed as

$$\frac{\partial \sigma_r}{\partial r}+\frac{1}{r}\frac{\partial \sigma_{r\theta}}{\partial \theta}+\frac{\sigma_r-\sigma_\theta}{r}=0 \quad ; \quad \frac{1}{r}\frac{\partial \sigma_\theta}{\partial \theta}+\frac{\partial \sigma_{r\theta}}{\partial r}+\frac{2\sigma_{r\theta}}{r}=0 \tag{A.1}$$

where σ_r and σ_θ are, respectively, the radial and tangential components of the stress tensor.

Constitutive equations: Assuming a linear orthotropic elastic behavior, the constitutive equations can be expressed as follows

$$\varepsilon_r=\frac{1}{E_r}\sigma_r-\frac{\nu_{r\theta}}{E_r}\sigma_\theta \quad ; \quad \varepsilon_\theta=-\frac{\nu_{r\theta}}{E_r}\sigma_r+\frac{1}{E_\theta}\sigma_\theta \quad ; \quad \varepsilon_{r\theta}=\frac{1}{2G_{r\theta}}\sigma_{r\theta} \tag{A.2}$$

where ε_r is the radial strain and ε_θ the tangential strain. E_θ the extensional modulus in the tangential direction and E_r the extensional modulus in the radial direction and $\nu_{r\theta}$ the coefficient relating contraction in the circumferential direction to extension in the radial direction. $G_{r\theta}$ is the shear modulus.

Geometric relations:

$$\varepsilon_\theta=\frac{u_r}{r}+\frac{1}{r}\frac{\partial u_\theta}{\partial \theta} \quad ; \quad \varepsilon_r=\frac{\partial u_r}{\partial r} \quad ; \quad \varepsilon_{r\theta}=\frac{1}{2}\left(\frac{1}{r}\frac{\partial u_r}{\partial \theta}+\frac{\partial u_\theta}{\partial r}-\frac{u_\theta}{r}\right) \tag{A.3}$$

where u_r is the radial displacement and u_θ the tangencial displacement.

In order to adequately model the problem, besides eqs. (A.1) - (A.3), it is necessary to provide the adequate set of boundary conditions:

$$\sigma_r|_{r=r_i}=-P_0 \quad ; \quad \sigma_r|_{r=r_e}=-P_1 \tag{A.4}$$

Equations (A.1) - (A.4) govern the behavior of an elastic orthotropic pipe. Such problem can be solved using the Airy stress function method. In this method, it is supposed to exist a differentiable function $\phi(r,\theta)$, called Airy function, such that

$$\sigma_r=\frac{1}{r}\frac{\partial \phi}{\partial r}+\frac{1}{r^2}\frac{\partial^2 \phi}{\partial \theta^2} \quad ; \quad \sigma_\theta=\frac{\partial^2 \phi}{\partial r^2} \quad ; \quad \sigma_{r\theta}=-\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial \phi}{\partial \theta}\right) \tag{A.5}$$

It is easy to verify that, if all relations in (A.5) are valid, then eqs. (A.1) are automatically satisfied. Assuming radial symmetry ($\phi(r,\theta)=\varphi(r)$), eqs. (A.5) can be reduced to:

$$\sigma_r = \frac{1}{r}\frac{d\varphi}{dr} \quad ; \quad \sigma_\theta = \frac{d^2\varphi}{dr^2} \quad ; \quad \sigma_{r\theta} = 0 \tag{A.6}$$

Introducing (A.6) in the constitutive equations (A.2), it is possible to obtain:

$$\varepsilon_r = \frac{1}{E_r}\frac{1}{r}\frac{d\varphi}{dr} - \frac{\nu_{r\theta}}{E_r}\frac{d^2\varphi}{dr^2} \quad ; \quad \varepsilon_\theta = -\frac{\nu_{r\theta}}{E_r}\frac{1}{r}\frac{d\varphi}{dr} + \frac{1}{E_\theta}\frac{d^2\varphi}{dr^2} \quad ; \quad \varepsilon_{r\theta} = 0 \tag{A.7}$$

The choice of a function $\varphi(r)$ that satisfies the boundary conditions (A.4) is not enough to assure the existence of strain fields u_r, u_θ that satisfy the geometric relations (A.3). Assuming that, due to the symmetry, $u_\theta = 0$, the geometric relations can be reduced to

$$\varepsilon_\theta = \frac{u_r}{r} \quad ; \quad \varepsilon_r = \frac{\partial u_r}{\partial r} \tag{A.8}$$

Once ε_θ is obtained from the second equation in (A.7), it is possible to determine u_r from the first geometric relation in (A.8). Using the geometric relations (A.8) it is simple to verify that the strain components ε_r and ε_θ must satisfy the following relation, called the compatibility equation of the problem

$$\frac{\partial \varepsilon_\theta}{\partial r} = \frac{1}{r}\left(\varepsilon_r - \varepsilon_\theta\right) \tag{A.9}$$

Hence, to make an adequate choice of φ and to satisfy the geometric relations, it is necessary to verify the compatibility equation (A.9). Introducing (A.7) in (A.9), it is possible to obtain

$$\frac{1}{E_\theta}\left[\frac{d^3\varphi}{dr^3} + \frac{1}{r}\frac{d^2\varphi}{dr^2}\right] - \frac{1}{E_r}\frac{1}{r^2}\frac{d\varphi}{dr} = 0 \tag{A.10}$$

The function φ that correspond to the solution of the problem (A.1)-(A.4) must be a solution of the ordinary differential equation (A.10). To solve this equation, it is interesting to perform the following change of variables: $r = e^t$. In this case, it is possible to obtain

$$\frac{d\varphi}{dr} = \frac{d\varphi}{dt}\frac{dt}{dr} = \frac{d\varphi}{dt}e^t \tag{A.11}$$

$$\frac{d^2\varphi}{dr^2} = \frac{d}{dr}\left(\frac{d\varphi}{dr}\right) = \frac{d}{dt}\left(\frac{d\varphi}{dt}e^t\right) = e^t\left(\frac{d^2\varphi}{dt^2} - \frac{d\varphi}{dt}\right) \tag{A.12}$$

$$\frac{d^3\varphi}{dr^3} = \frac{d}{dr}\left(\frac{d^2\varphi}{dr^2}\right) = \frac{d}{dt}\left[\left(\frac{d^2\varphi}{dt^2} - \frac{d\varphi}{dt}\right)e^t\right] = e^{-3t}\left(\frac{d^3\varphi}{dt^3} - 3\frac{d^2\varphi}{dt^2} + 2\frac{d\varphi}{dt}\right) \tag{A.13}$$

Using (A.11) - (A.13) and (A.10) it is possible to obtain the following linear ordinary differential equation

$$\frac{1}{E_\theta}\left[\frac{d^3\varphi}{dt^3} - 2\frac{d^2\varphi}{dt^2} + \frac{d\varphi}{dt}\right] - \frac{1}{E_r}\frac{d\varphi}{dt} = 0 \tag{A.14}$$

The general solution for Eq. (A.14) is

$$\varphi = \hat{A} + \hat{B}r^{\left(1-\sqrt{\frac{E_\theta}{E_r}}\right)} + \hat{C}r^{\left(1+\sqrt{\frac{E_\theta}{E_r}}\right)} \tag{A.15}$$

where $\hat{A}$, $\hat{B}$ and $\hat{C}$ are constants to be obtained from the boundary conditions. Finally, using (A.6) and (A.15), it is easy to verify that the stress components corresponding to that Airy function are

$$\sigma_r(r) = \frac{1}{r}\frac{d\varphi}{dr} = \hat{B}\left(1-\sqrt{\frac{E_\theta}{E_r}}\right) r^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} + \hat{C}\left(1+\sqrt{\frac{E_\theta}{E_r}}\right) r^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} \tag{A.16}$$

$$\sigma_\theta(r) = \frac{d^2\varphi}{dr^2} = -\hat{B}\left(1-\sqrt{\frac{E_\theta}{E_r}}\right)\sqrt{\frac{E_\theta}{E_r}} r^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} + \hat{C}\left(1+\sqrt{\frac{E_\theta}{E_r}}\right)\sqrt{\frac{E_\theta}{E_r}} r^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} \tag{A.17}$$

And, from the boundary conditions, it comes that

$$\hat{B} = \frac{P_1 r_i^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} - P_0 r_e^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)}}{\left(1-\sqrt{\frac{E_\theta}{E_r}}\right)\left[r_i^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} r_e^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} - r_i^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} r_e^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)}\right]} \tag{A.18}$$

$$\hat{C} = \frac{P_1 r_i^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} - P_0 r_e^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)}}{\left(1-\sqrt{\frac{E_\theta}{E_r}}\right)\left[r_i^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)} r_e^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} - r_i^{-\left(\sqrt{\frac{E_\theta}{E_r}}+1\right)} r_e^{\left(\sqrt{\frac{E_\theta}{E_r}}-1\right)}\right]} \tag{A.19}$$

The radial displacement u_r is obtained from the second constitutive relation in (A.2)

$$u_r(r) = \left(1-\sqrt{\frac{E_\theta}{E_r}}\right)\left[-\hat{B}\sqrt{\frac{E_\theta}{E_r}} r^{-\sqrt{\frac{E_\theta}{E_r}}}\left(\frac{1}{E_\theta}+\frac{\nu_{r\theta}}{E_r}\right) + \hat{C}\sqrt{\frac{E_\theta}{E_r}} r^{\sqrt{\frac{E_\theta}{E_r}}}\left(\frac{1}{E_\theta}-\frac{\nu_{r\theta}}{E_r}\right)\right] \tag{A.20}$$

A.2 Thin-walled elasto-plastic cylinder under pressure

The study of a composite sleeve – metallic pipe system under an internal pressure P_i, as it is shown if Fig. 3 can be performed using the above equations, considering the pipe as an isotropic material, the glove as an anisotropic material and using the following compatibility relation to determine the contact pressure

$$[u_r(r=r_o)]_{pipe} - \hat{u} = [u_r(r=r_o)]_{sleeve} \tag{A.21}$$

Nevertheless, it is possible to show that, if the wall thickness t of the metallic pipe is less than about $1/10$ of the internal radius ($t < r_i/10$, the simplifying hypothesis of thin-walled structures can be adopted, since the computed contact pressure between pipe and sleeve are the same. In this case, it is considered only the tangential (hoop) stress component σ_θ, which has a very simple expression

$$\sigma_\theta = \frac{P_0 r_i - P_1 r_e}{r_e - r_i} \tag{A.22}$$

The advantage of using the thin-wall theory is that an inelastic behavior of the pipe can be easily included in the analysis. For a thin-walled metallic pipe undergoing inelastic deformations at room temperature, the elastic stress-strain relation is given by

$$\sigma_\theta = E(\varepsilon_\theta - \varepsilon_\theta^p) \tag{A.23}$$

with ε_θ^p being the tangential component of the plastic deformation. Besides this classic relation, it is necessary to give additional information about the relation between the stress the plastic deformation and. The following relation is is adequate to model monotonic loading histories in metallic materials

$$\sigma_\theta = \sigma_y + K\left(\varepsilon_\theta^p\right)^N, \text{ if } \quad \sigma_\theta > \sigma_y \quad \Rightarrow \quad \varepsilon_\theta^p = \left\langle \frac{\sigma_\theta - \sigma_y}{K} \right\rangle^{\frac{1}{N}} \tag{A.24}$$

$\langle x \rangle = Max\{0, x\}$. σ_y is the yielding stress. K and n are positive constants that characterize the plastic behavior of the material. From (A.24) it is easy to verify that $\varepsilon_\theta^p = 0$ if $\sigma_\theta < \sigma_y$. It also follows that $\varepsilon_\theta = \frac{\sigma_\theta}{E} + \varepsilon_\theta^p$. Assuming that $\varepsilon_\theta \approx \frac{u_r}{r_i}$, it is possible to obtain the following expression for the radial displacement

$$u_r = r_i \left[\frac{\sigma_\theta}{E} + \left\langle \frac{\sigma_\theta - \sigma_y}{K} \right\rangle^{\frac{1}{N}} \right], \ \forall r \tag{A.25}$$

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