

## **Prediction of the burst pressure for metallic pipelines with localized corrosion defects**

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### **ABSTRACT**

The goal of the present paper is to propose a simple methodology to estimate the burst pressure of thin-walled metallic pipelines with arbitrary localized corrosion damage. This methodology is conceived as a preliminary tool for a quick analysis of the structural integrity of real corroded pipelines. Due to the different possible geometries of the corroded region, the exact analysis of this kind of problem can be very complex (in general using an elasto-plastic finite element simulation). The idea is to obtain an approximate exact analytical solution of the problem for any arbitrary geometry of the corroded region considering elasto-plastic constitutive equations and a factor that accounts for the stress concentration due to the metal loss caused by corrosion. Different expressions are considered for such factor, and they all depend on the general average geometry of the corroded region. With this simple expression, a reasonable lower limit for the burst pressure can be obtained. The predicted burst pressures for different corroded geometries are compared with experimental results, showing a good agreement.

**Keywords:** Corrosion, metallic pipelines, elasto-plasticity, burst tests

### **1 INTRODUCTION**

The goal of this paper is propose a simple methodology to predict the failure pressure of metallic pipelines with localized corrosion defects. The study is an extension of the methodology presented in [1] for undamaged pipelines. The analysis of this problem, due to the variety of possible corrosion conditions, is quite complex (and expensive) when a finite element simulation is adopted. The idea of the proposed methodology is to use elasto-plasticity constitutive equations obtained in [2] and to solve analytically the resulting problem, including a factor that takes into account the stress concentration due to loss of material caused by corrosion. This factor is based on expressions found in classical criteria for corroded pipelines (see [3]), but the analysis is extended to situations where these criteria are not satisfied and plastic deformation occurs. Thus, it is expected to be possible to obtain a lower limit for the failure pressure of a metallic pipeline with arbitrary localized corrosion defect only taking into account a few data concerning its geometry and the ultimate stress of the material obtained in a simple tensile test.

Generally the standards for corroded pipelines try to approximate the corroded region through a rectangle or an ellipse with depth corresponding to the greater corrosion depth measured along the

pipe axis (see figure 1).

The most widely used criteria for structural integrity evaluation of corroded pipelines under internal pressure constitute a family of criteria known as "effective area methods" and are described in [3]. This family includes the ASME B31G criterion [4] and the criterion RSTENG 0.85 (also known as modified B31G criterion presented in [5]). These criteria were developed in the late 1960s and early 1970s to assess the serviceability conditions of corroded gas transmission lines. The basic empirical hypothesis is that the loss of strength due to corrosion is proportional to the amount of material loss measured axially along the pipe. Other approaches may be found in the literature but in all of them consider part-wall metal loss defects obtained in [6].

Hydrostatic burst tests are generally recommended for assessing the structural integrity of these pipelines. For experimental studies performed in laboratory, rectangular regions with reduced wall thickness are artificially created in the specimens. In a burst test, the axial stress induced by the pressure applied at the extremities of the specimen can be important. The particular nature of the specimens may lead to mistaken conclusions. Real pipelines are long and the effect of axial stresses in straight lines is almost negligible (all criteria for corroded pipelines mentioned before neglect the effect of axial stresses), what is not the case of the specimens for hydrostatic testing. Hence, such a difference must be taken into account or the strength of the pipeline is overestimated. In order to identify and eventually "correct" or even eliminate the perturbation caused by the closed ends of the specimen on experimental results, a theoretical analysis of closed-ended pipelines is also performed in the present paper.

Model predictions are compared with experimental results obtained in [7] showing a good agreement.

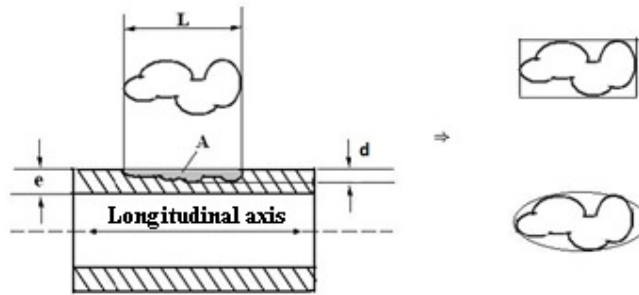


Figure 1: Metal loss in the pipeline

## 2 MODELING

### 2.1 Summary of the elasto-plastic constitutive equations

The following set of elasto-plastic constitutive equations is a particular case of the constitutive equations discussed in [2] but restricted to isotropic hardening. These equations are adequate to model the monotonic inelastic behaviour of metallic material undergoing a quasi-static and isothermal process at room temperature.

In the framework of small deformations and isothermal processes, besides the stress tensor  $\underline{\underline{\sigma}}$  and the strain tensor  $\underline{\underline{\epsilon}} = 1/2[\nabla \underline{u} + (\nabla \underline{u})^T]$  ( $\underline{u}$  is the displacement at given material point), it is considered the following auxiliary variables: the plastic strain tensor  $\underline{\underline{\epsilon}}^p$ , the cumulated plastic strain  $p$  and another variable  $Y$ , related to the isotropic hardening. A complete set of elasto-plastic constitutive equations is given by:

$$\underline{\underline{\sigma}} = \frac{\nu E}{(1+\nu)(1-2\nu)} \text{tr}(\underline{\underline{\varepsilon}} - \underline{\underline{\varepsilon}}^p) \underline{\underline{1}} + \frac{E}{(1+\nu)} (\underline{\underline{\varepsilon}} - \underline{\underline{\varepsilon}}^p) \quad (1)$$

$$(\underline{\underline{\varepsilon}} - \underline{\underline{\varepsilon}}^p) = \frac{(1+\nu)}{E} \underline{\underline{\sigma}} - \frac{\nu}{E} \text{tr}(\underline{\underline{\sigma}}) \underline{\underline{1}} \quad (2)$$

$$\underline{\underline{\dot{\varepsilon}}}^p = \frac{3}{2J} \underline{\underline{S}} \dot{p} \quad (3)$$

$$Y = \sigma_y + \nu_1 [1 - \exp(-\nu_2 p)] \quad (4)$$

$$\dot{p} \geq 0; F = (J - Y) \leq 0; \dot{p}F = 0 \quad (5)$$

with,

$$J = \sqrt{\frac{3}{2} (\underline{\underline{S}} : \underline{\underline{S}})} = \sqrt{\frac{3}{2} \sum_{i=1}^3 \sum_{j=1}^3 (S_{ij})^2} \quad (6)$$

Where  $E$  is the Young Modulus,  $\nu$  the Poisson's ratio and  $\sigma_y, \nu_1, \nu_2$  are positive constants that characterize the plastic behaviour of the material.  $\underline{\underline{1}}$  is the identity tensor, and  $\text{tr}(\bullet)$  is the trace of a tensor ( $\bullet$ ).  $\underline{\underline{\sigma}}$  is the stress tensor and  $\underline{\underline{S}}$  is the deviatoric stress tensor given by the following expression:

$$\underline{\underline{S}} = \left[ \underline{\underline{\sigma}} - \left( \frac{1}{3} \right) \text{tr}(\underline{\underline{\sigma}}) \underline{\underline{1}} \right] \quad (7)$$

$J$  is the Von Mises equivalent stress.  $Y$  is an auxiliary variable related with the isotropic hardening.  $p$  is usually called the accumulated plastic strain and  $\dot{p}$  can be interpreted as Lagrange multiplier associated to the constraint  $F < 0$ . Function  $F$  characterizes the elasticity domain and the plastic yielding surface. From the constraint  $\dot{p}F = 0$ , it is possible to conclude that  $\dot{p} = 0$  if  $F < 0$  and hence  $\underline{\underline{\dot{\varepsilon}}}^p = 0$  (see equation (3)). If  $\dot{p} \neq 0$ , from the constraint  $\dot{p}F = 0$  it comes that necessarily  $F = 0$ . Besides, from equations (3) and (4) it is possible to verify that, in this case,  $\underline{\underline{\dot{\varepsilon}}}^p \neq 0$  and  $\dot{Y} \neq 0$ . Therefore, the elasto-plastic material is characterized by an elastic domain in the stress space where yielding doesn't occur ( $\underline{\underline{\dot{\varepsilon}}}^p = 0, \dot{p} = \dot{Y} = 0$  if  $F < 0$ ).

Using expressions (3) and (6) it is possible to obtain the following relations:

$$\dot{p} = \sqrt{\frac{2}{3} \underline{\underline{\dot{\varepsilon}}}^p : \underline{\underline{\dot{\varepsilon}}}^p} \Rightarrow p(t) = p(t=0) + \int_{t=0}^t \left( \sqrt{\frac{2}{3} \underline{\underline{\dot{\varepsilon}}}^p(\zeta) : \underline{\underline{\dot{\varepsilon}}}^p(\zeta)} \right) d\zeta \quad (8)$$

Generally the following initial conditions are used for a "virgin" material

$$p(t=0) = 0, \underline{\underline{\varepsilon}}^p(t=0) = \underline{\underline{0}} \quad (9)$$

From now on, initial conditions (9) are assumed to hold in the analysis. It is also important to remark that the evolution law (3) with initial condition (9) and definition (7) imply that the principal directions the stress tensor, of the deviatoric stress tensor and of the plastic strain tensor are the same. From evolution law (3) and considering initial conditions (9), it is possible to verify that the following relation always holds

$$\frac{S_i}{S_j} = \frac{\varepsilon_i^p}{\varepsilon_j^p} \quad \forall (i,j=1,2 \text{ or } 3) \quad (10)$$

With  $S_i$  ( $i = 1, 2$  or  $3$ ) and  $\varepsilon_i^p$  ( $i = 1, 2$  or  $3$ ) being the principal components (eigenvalues) respectively of  $\underline{\underline{S}}$  and  $\underline{\underline{\varepsilon}}^p$ .

## 2.2 Thin-walled elasto-plastic cylinder under internal pressure

This section it is considered an elasto-plastic cylinder with internal radius  $R$ , thickness  $e$  submitted to an internal pressure  $P$ . The internal radius  $R$  and the thickness  $e$  are such that

$$\frac{R}{e} > 10 \quad (11)$$

The components of the stress tensor  $\underline{\underline{\sigma}}$  and of the deviatoric stress tensor  $\underline{\underline{S}}$  in cylindrical coordinates for a thin-walled cylinder are supposed to be reasonably approximated in the framework of membranes theory by the following expressions

$$\underline{\underline{\sigma}} = \begin{bmatrix} \sigma_r = 0 & 0 & 0 \\ 0 & \sigma_\theta = \alpha_\theta \sigma & 0 \\ 0 & 0 & \sigma_z = \alpha_z \frac{\sigma}{2} \end{bmatrix} \quad \underline{\underline{S}} = \begin{bmatrix} S_r = A_r \sigma & 0 & 0 \\ 0 & S_\theta = A_\theta \sigma & 0 \\ 0 & 0 & S_z = A_z \sigma \end{bmatrix} \quad (12)$$

with,

$$\sigma = \frac{PR}{e}; \quad A_r = -\frac{2\alpha_\theta + \alpha_z}{6}; \quad A_\theta = \frac{4\alpha_\theta - \alpha_z}{6}; \quad A_z = \frac{\alpha_z - \alpha_\theta}{3} \quad (13)$$

$\sigma_r$  is the radial stress component,  $\sigma_\theta$  the circumferential stress component and  $\sigma_z$  the axial stress component. All other stress components are considered to be equal to zero.  $\alpha_\theta$  and  $\alpha_z$  are parameters that take into account the corrosion damage and that, in principle, will be treated as constants.

From equation (6) it is possible to found the following expression to Mises equivalent stress gives by

$$J = A |\sigma| \quad (14)$$

where,

$$A = \left[ \frac{3}{2} (A_r^2 + A_\theta^2 + A_z^2) \right]^{1/2} \quad (15)$$

Introducing the last result and the expression for circumferential component of deviatoric stress in equation (3) it is possible to obtain the following expressions in the case of a monotonically increasing loading (for instance,  $P = \alpha t$ ,  $\alpha > 0$ ),

$$P = \frac{2}{3} A \varepsilon^p, \text{ where } \varepsilon^p = \frac{\varepsilon_\theta^p}{A_\theta} \quad (16)$$

Therefore, from equation (5) it comes that

$$Y = A\sigma \text{ if } F = 0 \quad (17)$$

Hence, it is possible to obtain the following expression relating the pressure  $P$  with  $\varepsilon^p$  combining equations (14), (16), (17) and (4) in the case of a monotonic loading gives by

$$P = \frac{e}{AR} \left\{ \sigma_y + \nu_1 \left[ 1 - \exp \left( -\nu_2 \frac{2}{3} A \varepsilon^p \right) \right] \right\} \text{ if } \left( A \frac{PR}{e} \right) > \sigma_y \quad (18)$$

The yield pressure  $P_y$  is obtained taking  $\varepsilon^p = 0$  in equation (18)

$$P_y = \frac{e}{AR} \quad (19)$$

Therefore, once the geometric parameters of the cylinder are known, it can be easily verified that the yield pressure  $P_y$  can be obtained from the axial yield stress  $\sigma_y$ . The maximum pressure  $P_{\max}$  is obtained by taking the limit of  $P$  as  $\varepsilon^p \rightarrow \infty$ . Hence,

$$P_{\max} = \frac{e}{AR} (\sigma_y + \nu_1) \quad (20)$$

It can be verified that the maximum pressure  $P_{\max}$  can be related with the ultimate stress obtained in a tensile test  $\sigma_{\max} = (\sigma_y + \nu_1)$ . Besides, the following analytic expression can be obtained

$$\varepsilon_\theta^p = \frac{3A_\theta}{2\nu_2 A} \left\langle -\ln \left( \frac{\sigma_y + \nu_1 - \frac{PR}{e} A}{\nu_1} \right) \right\rangle \quad (21)$$

With  $\langle x \rangle = \max\{0, x\}$ . Finally, with this last result it is possible to obtain the strain components in the case of monotonic loading history

$$\varepsilon_{\theta} = \frac{PR}{e} \left( \frac{2\alpha_{\theta} - \nu\alpha_z}{2E} \right) + \frac{3A_{\theta}}{2\nu_2 A} \left\langle -\ln \left( \frac{\sigma_y + \nu_1 - \frac{PR}{e} A}{\nu_1} \right) \right\rangle \quad (22)$$

$$\varepsilon_z = \frac{PR}{e} \left( \frac{\alpha_z - 2\nu\alpha_{\theta}}{2E} \right) + \frac{3A_z}{2\nu_2 A} \left\langle -\ln \left( \frac{\sigma_y + \nu_1 - \frac{PR}{e} A}{\nu_1} \right) \right\rangle \quad (23)$$

It is important to remark that  $P_{\max}$  is the limit pressure  $P_{\max}$  beyond which the hypothesis of quasi-static process is invalid and the dynamic must be accounted, since the acceleration field is no longer negligible. From the engineering point of view such pressure can be taken as the limit pressure (or failure pressure), beyond which there is not enough time to make any repair procedure: the rupture process is considered brutal and instantaneous after this pressure level is attained (see fig. 2). Such reasoning is very similar to the one adopted in fracture mechanics in order to define the critical load in a cracked medium. The proof of this fact can be obtained within a thermodynamic framework summarized in [1].

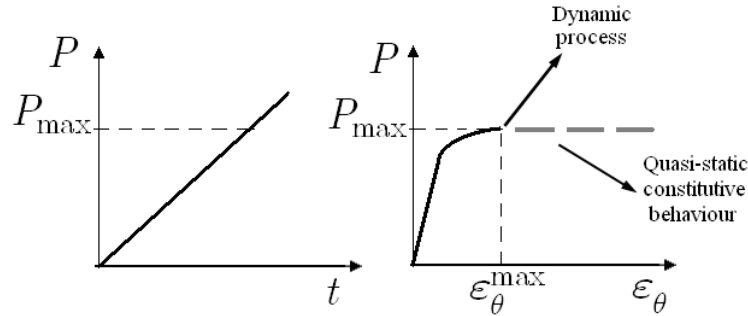


Figure 2: Dynamic rupture in a hydrostatic test with monotonically increasing pressure.

### 2.3 Remaining strength criteria for corrosion defects

Possible expressions for  $\alpha_{\theta}$  can be obtained from the criteria presented in [3], generally called *remaining strength criteria for corrosion defects*. It can be verified that these criteria can always be expressed as follows

$$\alpha_{\theta} \frac{PR}{e} < \sigma_{\max} \quad (24)$$

Where  $\alpha_{\theta}$  is a function of geometry and  $\sigma_{\max}$  a maximum admissible tensile strength before failure that varies according to the criterion. The term  $(1/\alpha_{\theta})$  is usually called the remaining strength factor. In these criteria, the component in the axial direction is not taken into account because for long lines, is reasonable consider  $\sigma_z$  negligible in comparison with  $\sigma_{\theta}$ . The following expressions are found for  $\alpha_{\theta}$

- ASME B31G Criterion

Firstly, it is necessary to calculate the nondimensional factor ( $A_f$ ) given by:

$$A_f = 0.893 \left( \frac{L}{\sqrt{De}} \right) \quad (25)$$

$$\text{If } A_f \leq 4 \text{ then } \alpha_\theta = \frac{1 - \frac{2}{3} \left( \frac{d}{e \sqrt{A_f^2 + 1}} \right)}{1 - \frac{2}{3} \left( \frac{d}{e} \right)} \quad (26)$$

$$\text{If } A_f > 4 \text{ then } \alpha_\theta = \left( \frac{e}{e - d} \right) \quad (27)$$

$$\sigma_{\max} = 1.1 \sigma_y \quad (28)$$

Where  $d$  is the maximum depth of the defect,  $L$  the total axial extent of the defect,  $e$  the wall thickness of the pipe and  $\sigma_y$  is the yield stress of the pipe (0.5% criterion).

- RSTRENG 0.85 or Modified B31G Criterion

$$\text{If } \frac{L^2}{De} \leq 50 \text{ then } M_t = \sqrt{1 + 0.6275 \left( \frac{L^2}{De} \right) - 0.003375 \left( \frac{L^2}{De} \right)^2} \quad (29)$$

$$\text{If } \frac{L^2}{De} > 50 \text{ then } M_t = 3.3 + 0.032 \left( \frac{L^2}{De} \right) \quad (30)$$

$$\alpha_\theta = \frac{1 - 0.85 \left( \frac{d}{e} \right) \left( \frac{1}{M_t} \right)}{1 - 0.85 \left( \frac{d}{e} \right)} \quad (31)$$

$$\sigma_{\max} = \sigma_y + 69 \text{ MPa} \quad (32)$$

Where  $M_t$  is the bulging factor in RSTRENG's criterion.

- Chell Limit Load Analysis

$$\alpha_\theta = \left[ 1 - \left( \frac{d}{e} \right) + \left( \frac{d}{e} \right) \left( \frac{1}{M_c} \right) \right]^{-1} \quad M_c = \sqrt{1 + 1,61 \left( \frac{\pi}{8} \right)^2 \frac{L^2}{Rd}} \quad \sigma_{\max} = 1.1 \sigma_y \quad (33)$$

$R$  is the inner radius of the pipe and  $M_c$  is the bulging factor in Chell's criterion.

- Kanninen Shell Theory Criterion

$$\alpha_{\theta} = \left[ \frac{1 - \left(\frac{d}{e}\right) \left(\frac{1}{B}\right)}{1 - \left(\frac{d}{e}\right)} \right] \quad \text{and} \quad \sigma_{\max} = \sigma_{ult} \quad (34)$$

$$B = B_1 B_2$$

$$B_1 = (1 + \eta^4) (\cosh \alpha \cdot \sinh \alpha + \sin \alpha \cdot \cos \alpha) + 2\eta^{3/2} (\cosh^2 \alpha - \cos^2 \alpha) + 2\eta^2 (\cosh \alpha \cdot \sinh \alpha - \sin \alpha \cdot \cos \alpha) + 2\eta^{5/2} (\cosh^2 \alpha - \sin^2 \alpha) \quad (35)$$

$$B_2 = \{ \cosh \alpha \cdot \sin \alpha + \sinh \alpha \cdot \cos \alpha + 2\eta^{5/2} \cosh \alpha \cdot \cos \alpha + \eta^2 (\sinh \alpha \cdot \cos \alpha - \cosh \alpha \cdot \sin \alpha) \}^{-1}$$

$$\alpha = 0,9306 \frac{L}{\sqrt{D(e-d)}} \quad \text{and} \quad \eta = 1 - \frac{d}{e} \quad (36)$$

Kanninen's criterion uses the pipe ultimate tensile strength ( $\sigma_{ult}$ ) with a maximum admissible tensile strength.

- Sims Pressure Vessel Criteria

$$\text{If } w > 6d + 0.1D \quad \text{then} \quad M_s = \sqrt{1 + 0,8 \left(\frac{D}{e}\right) \left(\frac{L}{D}\right)^2} \quad (37)$$

$$\text{If } w \leq 6d + 0.1D \quad \text{then} \quad M_s = \sqrt{1 + 2,5 \left(\frac{D}{e}\right) \left(\frac{L}{D}\right)^2} \quad (38)$$

$$\alpha_{\theta} = \left[ \frac{1 - (1 - R_t)(M_s)^{-1}}{R_t} \right] \quad R_t = 1 - \frac{d}{e} \quad \sigma_{\max} = \frac{\sigma_y}{0,9} \quad (39)$$

Where  $R_t$  is the thickness ratio,  $M_s$  is the bulging facto in Sims's criterion and  $w$  is the minimum defect width.

- Ritchie and Last Criterion

$$\alpha_{\theta} = \left[ \frac{1 - \left(\frac{d}{e}\right) (M_{t1})^{-1}}{1 - \left(\frac{d}{e}\right)} \right] \quad M_{t1} = \sqrt{1 + 0,8 \frac{L^2}{De}} \quad \sigma_{\max} = 0,9 \sigma_{ult} \quad (40)$$

Where  $M_{t1}$  is the bulging factor in Ritchie and Last criterion.



- PRC//Battelle PCORRC Plastic Collapse Criterion

$$\alpha_{\theta} = \left\{ 1 - \frac{d}{e} \left[ 1 - \exp \left( -0,157 \frac{L}{\sqrt{R(e-d)}} \right) \right] \right\}^{-1} \quad \sigma_{\max} = \sigma_{ult} \quad (41)$$

- BG/DNV Level 1 Criterion

$$\alpha_{\theta} = \left[ \frac{1 - \left( \frac{d}{e} \right) \left( \frac{1}{Q} \right)}{1 - \left( \frac{d}{e} \right)} \right] \quad Q = \sqrt{1 + 0,31 \left( \frac{L^2}{De} \right)} \quad \sigma_{\max} = \sigma_{ult} \quad (42)$$

### 3 MATERIALS AND EXPERIMENTAL PROCEDURES

In order to analyse the adequacy of the proposed methodology, tensile and hydrostatic tests were performed in API 5L X60 steel specimens. Table 1 presents the material properties obtained in tensile tests.

Table 1: Mechanical properties. API 5L X60 steel

$E$ (GPa)	$\sigma_y$ (MPa)	$\nu_1$ (MPa)	$\nu_2$
182	478	172	44.3

Hydrostatic tests were performed in thin-walled pipes with a rectangular localized defect as it can be seen in figure 2.

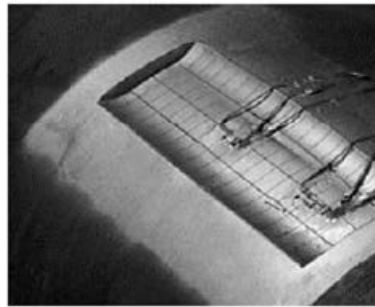


Figure 2: Rectangular damage

The specimen dimensions are presented in table 2.

Table 2. Specimen dimensions

Inner Radius - $R$	249.2 mm
Wall thickness - $e$	14.3 mm
Maximum depth of defect - $d$	10.0 mm

Length of the defect - $L$	500.0 mm
Width of the defect - $w$	95.0 mm

In the test, a strain gage was installed in a central region of the defect (at point "C") as it can be seen in figure 3, in order to measure the circumferential strain and the longitudinal strain of the pipe.

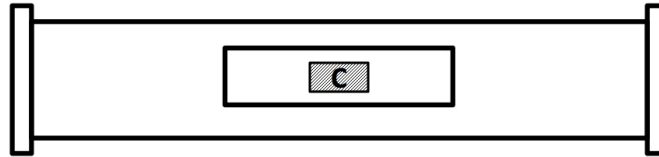


Figure 3: Specimen with rectangular damage

The different values of  $\alpha_\theta$  computed in this case are presented in table 3.

Table 3. Determination of  $\alpha_\theta$

Criterion	$\alpha_\theta$
ASME B31G	3.32
RSTRENG 0.85	2.12
Chell	2.28
Kanninen Chell	3.33
Sims	3.08
Ritchie and Last	2.88
Battelle	2.75
BG/DNV	2.63

Fig. 4 shows, respectively, the curves pressure versus circumferential strain and pressure versus longitudinal strain obtained in a hydrostatic test. The failure pressure (beginning of the plastic necking at the final stage of the loading process. Rupture process is considered brutal and instantaneous after this pressure level is attained) is approximately 11.6 MPa.

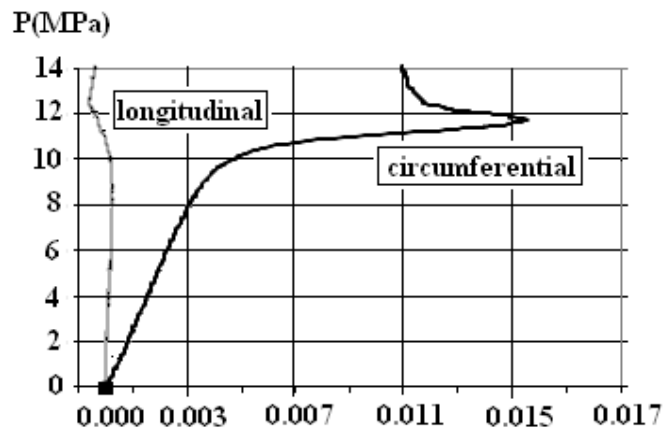


Figure 4: Curves pressure - strain

#### 4 RESULTS AND DISCUSSION

Fig. 5 presents both theoretical and experimental stress-strain curves in a tensile test. In order to obtain the experimentally the coefficients  $\alpha_\theta$  and  $\alpha_z$  it is necessary to determine the slope of the curve press-strain in the elastic region using equations (21) and (22). If  $\sigma < \sigma_y$ , it is possible to obtain

$$\varepsilon_\theta = \frac{P}{K_\theta} = \frac{PR}{e} \left( \frac{2\alpha_\theta - \nu\alpha_z}{2E} \right) \quad (43)$$

$$\varepsilon_z = \frac{P}{K_z} = \frac{PR}{e} \left( \frac{\alpha_z - 2\nu\alpha_\theta}{2E} \right) \quad (44)$$

Where  $K_\theta$  and  $K_z$  are the slopes of the press-strain curves in the elastic region (see fig. 4). Solving this linear system the following values are obtained:  $\alpha_\theta = 4.48$  and  $\alpha_z = 3.31$ . Figure 6 shows a comparison between model prediction and experimental results for the uniaxial tensile test and the figures 7.a and 7.b show comparisons between model predictions and experimental results using equations (22) and (23). It can be seen that the model gives a reasonable (but conservative) prediction of the failure pressure.

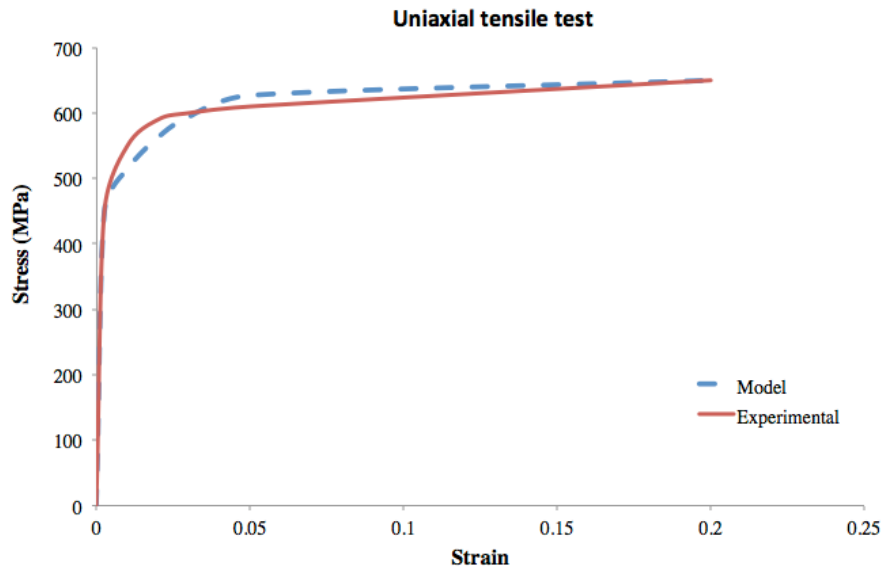


Figure 6: Tensile test – Experiment versus mode

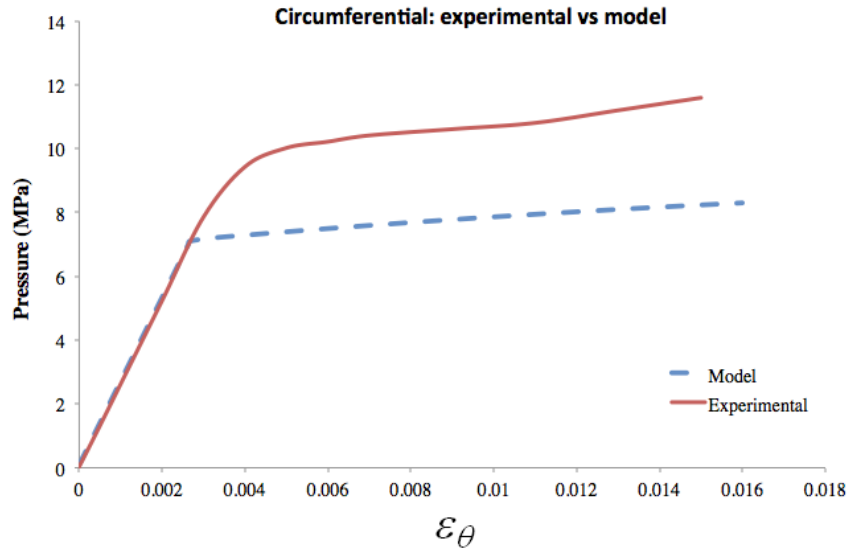


Figure 7.a: Hydrostatic test – Experiment versus model

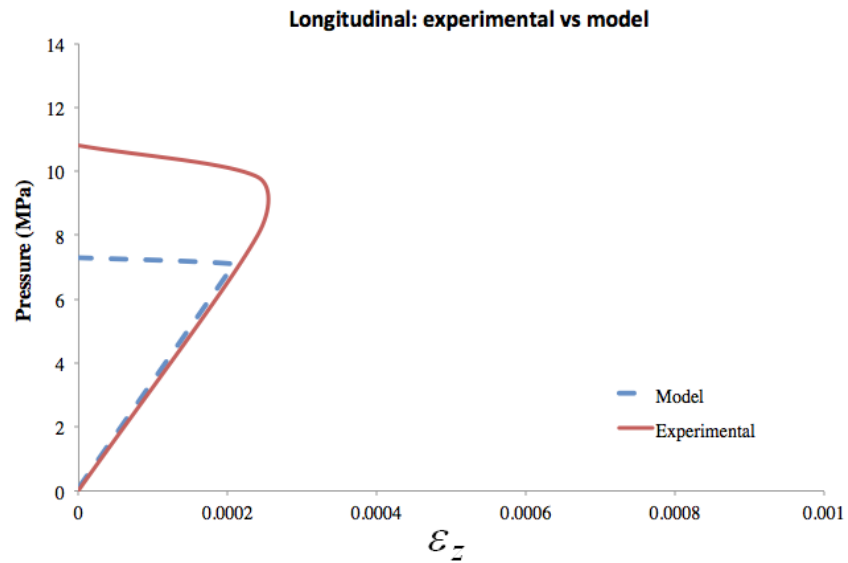


Figure 7.b: Hydrostatic test – Experiment versus model

The criteria presented in the previous section aims at giving a conservative estimation of the factor  $\alpha_\theta$  (they do not consider the axial stresses and strains). Table 4 presents the failure pressure obtained using all criteria presented in the previous section ( $P_{\max} = (e\sigma_{\max}) / (R\alpha_\theta)$ , see equations (24)-(42) and table 3) and the predicted failure pressure obtained using the proposed methodology ( $\alpha_\theta = 4.48$ ,  $\alpha_z = 3.31$  and  $P_{\max} = (e(v_1 + \sigma_y)) / RA$ ).

Table 4: Failure pressure obtained using different criteria

Criterion	$P_{\max}$ (MPa)	$(P_{\max} / P_{\max}^{\text{exp}})$
Experimental	11.6	1.00
Model ( $\sigma_{\max} = 650$ , $\alpha_{\theta} = 4.48$ and $\alpha_z = 3.31$ )	9.50	0.82
ASME B31G ( $\sigma_{\max} = 525.8$ , $\alpha_{\theta} = 3.32$ )	9.07	0.78
RSTRENG 0.85 ( $\sigma_{\max} = 547$ , $\alpha_{\theta} = 2.12$ )	<b>14.80</b>	<b>1.28</b>
Chell ( $\sigma_{\max} = 525.8$ , $\alpha_{\theta} = 2.28$ )	<b>13.22</b>	<b>1.14</b>
Kanninen Chell ( $\sigma_{\max} = 650$ , $\alpha_{\theta} = 3.33$ )	11.22	0.98
Sims ( $\sigma_{\max} = 531.11$ , $\alpha_{\theta} = 3.08$ )	9.90	0.85
Ritchie and Last ( $\sigma_{\max} = 585$ , $\alpha_{\theta} = 2.88$ )	11.64	1.00
Battelle ( $\sigma_{\max} = 650$ , $\alpha_{\theta} = 2.75$ )	<b>13.59</b>	<b>1.17</b>
BG/DNV ( $\sigma_{\max} = 650$ , $\alpha_{\theta} = 2.63$ )	<b>14.16</b>	<b>1.22</b>

The proposed methodology, ASME B31G, Kanninen Chell, Sims and Ritchie and Last criteria give conservative results while the predicted failure pressures using RSTRENG 0.85, Chell, Battelle and BG/DNV criteria are above the experimental failure pressure. Ritchie and Last and Kanninen Chell criteria give the best predictions. It is important to remark that pressure at the beginning of the plastic necking was taken as the failure pressure. Most of these criteria are more suitable for long axial flaws and when the axial stress component is negligible in comparison with the circumferential stress component. Therefore, the preliminary results show that the proposed methodology allows obtaining a conservative, but reasonable, prediction of the failure pressure when the axial component is also important. In the case it is necessary to perform an approximate prediction, it is suggested to use a corrected version of the ASME B31G criterion, replacing  $\alpha_{\theta}$  by  $A$ , obtained considering  $\alpha_z \approx \alpha_{\theta}/1.35 \Rightarrow A \approx 0.87\alpha_{\theta}$

$$P_{\max} \approx \left( \frac{1.1}{0.87} \right) \left( \frac{e\sigma_y}{R\alpha_{\theta}} \right) \approx 1.26 \frac{e\sigma_y}{R\alpha_{\theta}} \quad (45)$$

In this case we have  $P_{\max} = 11.34$  MPa ( $P_{\max} / P_{\max}^{\text{exp}} = 0.98$ ) for the specimen defined in table 1 and table 2.

## 5 CONCLUSION

Assessment methods are needed to determine the severity of corrosion defects when they are detected in pipelines. The idea of the proposed methodology is to obtain a preliminary, but adequate estimate of the failure pressure of a pipe with arbitrary corrosion defect when the effects of closed ends are important. It aims at providing tools to allow deciding whether operation must be immediately stopped or if it is safe to apply a provisory reinforcement system [8] and to wait until the next scheduled maintenance stop. Therefore, the proposed methodology can be a valuable auxiliary tool for assessing the integrity of corroded pipelines, since it does not require the use of numerical codes and not even hydrostatic testing. To compute the failure pressure, besides the pipe geometry and the average geometry of the flaw, it is only necessary to know the elastic properties and the ultimate stress obtained in a tensile test.

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