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A COMPARISON OF EXPERIMENTAL RESULTS AND COMPUTATIONS FOR CRACKED TUBES SUBJECTED TO INTERNAL PRESSURE

PRIMERJAVA EKSPERIMENTALNIH REZULTATOV IN IZRAČUNA ZA CEVI Z RAZPOKO, KI SO OBREMENJENI Z NOTRANJO RAZPOKO

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A cylindrical pipe that is used for gas transportation is mainly submitted to stresses originating from internal pressure. Other stresses are due to weight and the unexpected movements of supports and/or the ground. The first give rise to circumferential stresses, the second to longitudinal bending stresses. Here, we study the case of pipes that are joined by welding. This weld is an eventual source of defects, where cracks can originate. But there are other kinds of defects that can come from corrosion pits or accidental notches caused by diving machines, or in the case of work on the pipe. In this last case, the notch corresponds to a reduction of the section, which is enhanced by the stress-concentration factor. The objective of this work is to compare the prediction of finite-element computations with fracture experiments on such notched pipes. The results are given in terms of stress-concentration factors and show some discrepancy with the experimental results. As the mechanical properties have been measured on standard plane specimens, a transfer problem to curved structures is suspected to be at the origin of the difference, because with the dimensions of the pipe it is not possible to have standard and curved specimens in the same direction of rolling, and the mechanical properties of this pipe are different in the circumferential and transverse directions.

Key words: pressure pipe, notch fracture mechanics, transferability

V valjasti cevi za pretok plina so napetosti zaradi notranjega pritiska. Druge napetosti nastanejo zaradi gravitacije in nepričakovanih premikov podpor ali zemljišča. Prve napetosti so obodne, druge pa podolžne in upogibne. V tem delu obravnavamo primer cevi, ki so povezane z varjenjem. Zvar je eventualni vir napak, na katerih nastanejo razpoke. Druge vrste napak lahko nastanejo zaradi korozijskih zajed ali odrgnin, ki jih povzročijo gradbeni stroji ali nastanejo že pri izdelavi cevi. V tem primeru zareza ustreza zmanjšanju prereza, napetost pa je povečana zaradi zarezne koncentracije napetosti. Cilj tega dela je primerjava napovedi na podlagi izračuna z metodo končnih elementov s preizkusi preloma cevi z napako. Rezultati izračunov, ki so predstavljeni v obliki faktorja intenzitete napetosti, se ne ujemajo popolnoma z rezultati preizkusov. Ker se mehanske lastnosti določijo pri standardnih ravninskih preizkušancih, predpostavljamo, da je vzrok za razliko prenos na ukrivljeno strukturo. Zaradi izmer cevi ni mogoče izdelati standardnih in ukrivljenih preizkušancev v enaki smeri valjanja in so mehanske lastnosti cevi drugačne v obodni kot v prečni smeri.

Ključne besede: cev pod pritiskom, mehanika loma, prenos

1 INTRODUCTION

The computation of circumferential stress, also called hoop stress, in cylindrical pipes subjected to internal pressure is well known, and the general relationship is given as follows:

$$\sigma_\theta = \frac{p_i \cdot r}{t} \tag{1}$$

where σ_θ is the hoop stress, p_i is the internal pressure, r is the radius of the tube and t is the thickness of the tube. This relationship is valid if $t \le r/10$, in other cases the relationship (1) is corrected as:

$$\sigma_\theta = \frac{a^2 \cdot p_i}{(b^2 - a^2)} \cdot \left[1 + \left(\frac{b^2}{r^2}\right)\right] \tag{2}$$

where a is the internal radius, b is the external radius, r is the radius of the computed point in the cylinder wall and p_i is the internal pressure.

These relationships are accurate enough to design pressure pipes if no defect is present in the tube. But, if the tube is realized by welding, misalignment or lack of penetration can occur, or if there are corrosion pits, or if the diving equipment causes damage to the external envelope of the tube, a reduction of section occurs and the stress is increased as a consequence. However, the reduction of section is not the only effect that causes a stress increase, and the notch-concentration-factor effect brings the majority of the increase. The objective of this work was to compute the actual hoop stress when an external notch is present and to compare with a fracture test under gas pressure that was made with the three coded methods, i.e., the ASME B31 G, the modified ASME B31 G and the DNV RP-F101. This work is based on a study of the pipeline and has resulted in several publications and presentations at international congresses [1,2].

2 CODED METHODS

2.1 ASME B31 G [3]

This is a code for evaluating the remnant strength of corroded pipelines. It is a supplement of the ASME B31 code for pressure piping. The code was developed in the late nineteen-sixties and early seventies at the Battelle Memorial Institute and provides a semi-empirical procedure for the assessment of corroded pipes. Based on an extensive series of full-scale tests on corroded pipe sections, it was concluded that line pipe steels have an adequate toughness and that the toughness is not a significant factor. The failure of blunt corrosion flaws is controlled by their size and the flow stress or yield stress of the material. The input parameters include the pipe's outer diameter (D_{ext}) and the wall thickness (t), the specified minimum yield strength (SMYS), the maximum allowable operating pressure (MAOP), the longitudinal extent of corrosion (2c) and the defect depth (a). According to this code, a failure equation for the corroded pipelines is proposed by means of the data from a bursting experiment and expressed by considering the two conditions below:

- First, the maximum hoop stress cannot exceed the yield strength of the material.
- Second, short corrosion defects are projected with a parabolic shape, and long corrosion defects are projected with a rectangular shape on the shape of a rectangular one (**Figure 1 and 2**).

The failure pressure equation for a corroded pipeline with a parabolic defect is:

$$\sqrt{0.8\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right)} \leq 4 \tag{3}$$

$$P_{ult} = \frac{2\cdot(1.1\sigma_y)\cdot t}{D_{ext}}\cdot\left[\frac{1-0.66(a/t)}{1-0.66(a/t)/M}\right] \tag{4}$$

with

$$M = \sqrt{1+0.8\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right)} \tag{5}$$

M is the so-called bulging factor.

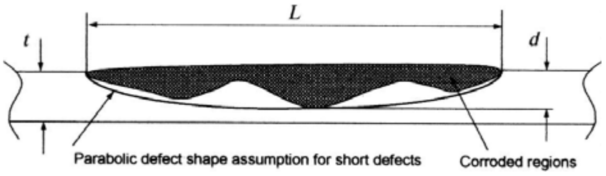

Figure 1: Corrosion defects projected with a parabolic shape
Slika 1: Korozijske poškodbe, predstavljene s parabolično obliko

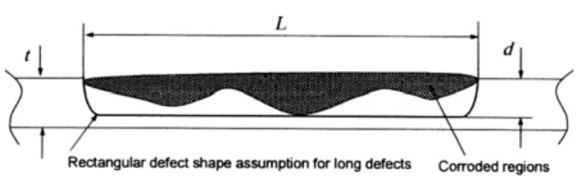

Figure 2: Corrosion defects projected with a rectangular shape
Slika 2: Korozijske poškodbe, predstavljene s pravokotno obliko

2.2 Modified ASME B31 G

This includes the modified flow stress and the bulging factor. The flow stress σ_{ult} is taken as:

$$\sigma_{ult}\ /\ \text{MPa} = 1.1\sigma_y + 69 \tag{6}$$

where σ_y is the yield strength.

Two cases are considered:

Case 1
$$\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right) \leq 50 \tag{7}$$

$$P_{ult} = \frac{2\cdot(1.1\sigma_y + 69)\cdot t}{D_{ext}}\cdot\left[\frac{1-0.85(a/t)}{1-0.85(a/t)/M}\right] \tag{8}$$

The bulging factor is equal to:

$$M = \sqrt{1+0.6275\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right) - 0.003375\left(\frac{2c}{D_{ext}}\right)^4\left(\frac{D_{ext}}{t}\right)^2} \tag{9}$$

Case 2
$$\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right) > 50 \tag{10}$$

$$P_{ult} = \frac{2\cdot(\sigma_{ult})\cdot t}{D_{ext}-t}\cdot\left[\frac{1-(a/t)}{1-(a/t)/Q}\right] \tag{11}$$

The bulging factor is equal to:

$$M = 3.3 + 0.032\left(\frac{2c}{D_{ext}}\right)^2\left(\frac{D_{ext}}{t}\right) \tag{12}$$

It is necessary to recall that the ASME B31G is limited to low stress-concentration factors and internal pressure loading conditions.

The assessment procedure considers the maximum depth and the longitudinal extent of the corroded area, but ignores the circumferential extent and the actual profile. If the corroded region is found to be unacceptable, B31G allows the use of a more rigorous analysis or a hydrostatic pressure test in order to determine the pipe's remaining strength. Alternatively, a lower maximum allowable operating pressure may be imposed.

2.3 DNV RP-F101 [4]

This is the first comprehensive and extensive code on pipeline-corrosion defect assessment. It prepares guidance on the pipeline under internal pressure and combined loading. Furthermore, it provides a codified formulation for pressure and bending and area depth. DNV RP-101 proposes two methods to find the failure pressure.

The first method is named the partial safety factor, and the second is classified as the allowable stress design. The allowable-stress-design method that considers non-interacting defects is discussed here. To pursue the design procedure via DNV RP-101 it is necessary to determine the loading type (pressure only and combined loading), and consequently the failure pressure can be obtained as:

$$P_{\mathrm{ult}} = \frac{2\sigma_{\mathrm{ult}} t}{D_{\mathrm{ext}} - t} \cdot \left[\frac{1-(a/t)}{1-(a/t)/Q} \right] \tag{13}$$

where Q is the correction factor:

$$Q = \sqrt{1 + 0.31 \left(\frac{1}{\sqrt{D_{\mathrm{ext}} t}} \right)} \tag{14}$$

According to this code, the failure pressure should not exceed the maximum allowable stress design operating pressure (MAOP); otherwise, the corroded pipe will have to be repaired or replaced before returning to service.

3 FINITE-ELEMENT COMPUTATION

3.1 Meshing the model

The model tube is a rolled steel cylindrical pipe with an external diameter of 219.1 mm and a thickness of 6.1 mm. A quasi-semi-ellipsoidal notch represents an accidental external defect. The major axis of the ellipsoid is parallel to the axis of the tube and the length of the notch is 30.5 mm.

The depth of the notch is 3.05 mm and the width is also 3.05 mm. The radius of the notch tip is 0.15 mm, as shown in **Figures 3 and 4**.

As the defect has two perpendicular planes of symmetry, only a quarter of the whole structure was meshed. A schematic view of the notched tube is given in **Figure 4**. For our computations, the CAST3M programme was used. Such a structure is not very easy to model exactly, as the very small notch-tip radius implies very strong geometrical constraints.

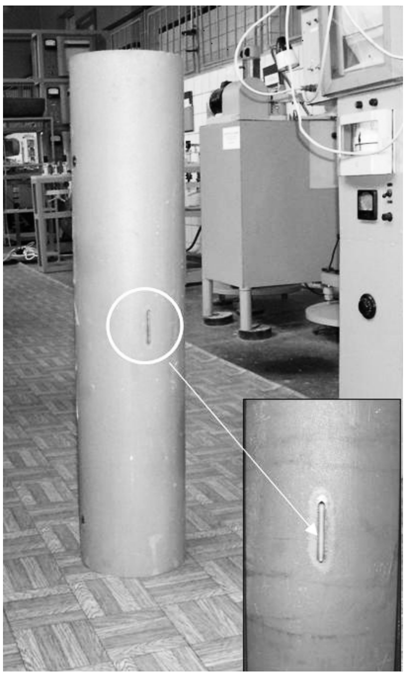

Figure 3: Notch on the pipe
Slika 3: Zareza na cevi

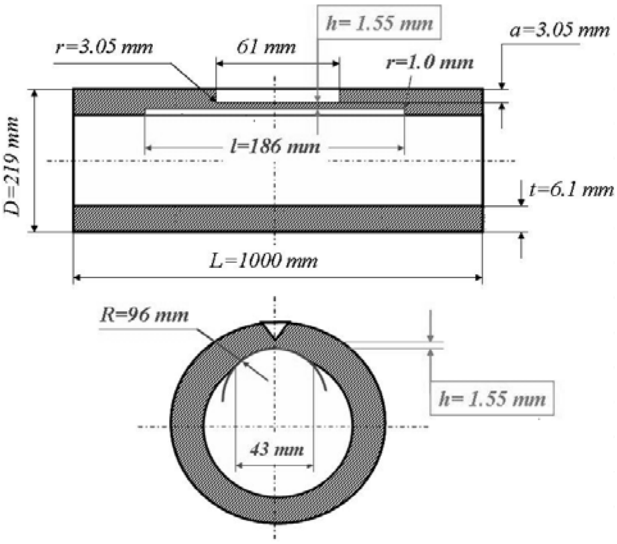

Figure 4: Plan of the grinding part
Slika 4: Načrt brušenja

Figure 5 shows the mesh of an ellipsoidal notch with the approximate notch radius. Due to the program's constraints, simple linear elements were used.

3.2 Boundary conditions and loading

As the model represents a quarter of the whole structure, and no pressure effect on the ends is assumed, symmetrical boundary conditions were applied on each section of the models. The loading is applied as a 15 MPa uniform pressure on the internal face of the tube.

3.3 Material behaviour and properties

The material constituting the tube is rolled steel. A Young's modulus of 203000 MPa was deduced from previous tests on a "Roman-tiles-type" specimen [5].

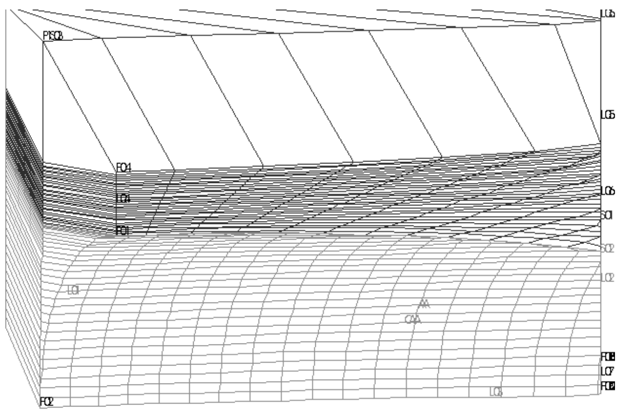

Figure 5: Details of the mesh at the notch
Slika 5: Detajl mreže pri zarezi

The yield strength was estimated to be 528 MPa. The Ludwik's law will make it possible to introduce the actual behaviour of the steel into its plastic range, where the hardening parameter is n = 0.0446, and the resistance coefficient is K = 587.3 MPa.

$$\sigma = K\varepsilon_{\mathrm{p}}^{n} \tag{15}$$

3.4 Post-processing of results

The results of the computations were given in terms of the hoop stress distribution on the ligament surface between the notch tip and the inner surface of the tube. The figures were obtained with the help of the home-made SCILAB® programme.

4 RESULTS

4.1 Finite-element computations

The results are shown in **Figure 6**. The hoop stress field is represented over the mesh where it is computed. The hottest point is on the small axis of the ellipsoid. But the hoop stress does not take into account the degree of triaxiality, when the ligament is reduced, just under the small axis of the ellipse. Consequently, the Von Mises stress was computed and represented in **Figure 7**. It is clear that if the stress level far from the notch is about the same as the hoop stress at the same place, the stress along the notch tip and, particularly at the deepest point of the notch, is much higher.

For comparison, the hoop stress is computed using the relationships (1) and (2). The results are the following:

- (1): hoop stress at mid thickness: 174.6 MPa
- (2): hoop stress at inner face: 174.7 MPa
- (2): hoop stress at outer face: 164.7 MPa

From the FE computation (**Figure 8**) we can extract the values of the hoop stress on the outer face, far from the notch, 148.5 MPa, and on the inner face, 189.9 MPa. The value at the mid thickness is, therefore, 169.2 MPa. This value is quite similar to that obtained for the usual analytical relationships of cylindrical tubes (174.6 MPa). This validates the FE computation.

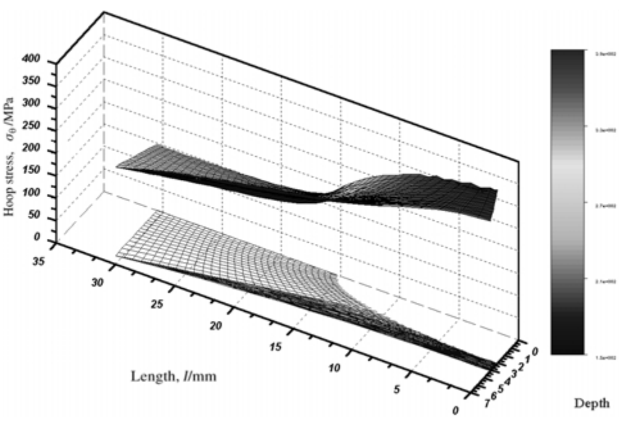

Figure 6: Variation of the hoop stress on the surface of the ligament
Slika 6: Variacija hoop-napetosti na površini ligamenta

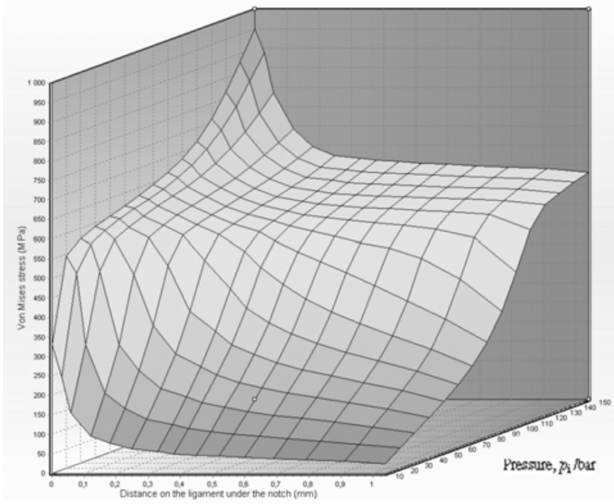

Figure 7: Von Mises stress on the ligament under the notch for different pressures
Slika 7: Von Misesova napetost v ligamentu pod zarezo pri različnem pritisku

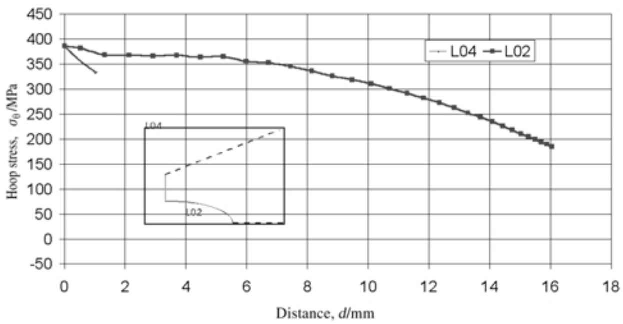

Figure 8: The variation of the hoop stress around the notch. L04 is the ligament line prolonging the small axis of the ellipse, L02 is the notch front line
Slika 8: Variacija hoop-napetosti okoli zareze. L04 je črta ligamenta podaljšek male osi elipse, L02 je čelna črta zareze

For this particular notch we have estimated the stress-concentration factor using the relationship:

$$K_{\mathrm{ep}} = \frac{\sigma_{\mathrm{max,\,pl}}}{\sigma_{\mathrm{G,\,pl}}} \tag{16}$$

In this case, we obtain:

$$K_{\mathrm{ep}} = \frac{386}{186} = 2.07 \tag{17}$$

4.2. Experimental results

Some tubes containing a notch with the dimensions given here were loaded with a gas up to failure. With the ground tubes the fracture occurs under a pressure of about 12 MPa. Only two results are available, but they give fracture pressures that are close together. These results seem to be in agreement with the tests carried out by a South Korean team[6], although these tests used a pipe in X65 with different dimensions. Moreover, this type of burst test with gas is rare, as are papers on this subject. So it is very difficult to properly compare our results.

5 COMPARISON OF RESULTS

If we consider the hoop stress, it appears that the computation gives a maximum value that is less than the yield strength. The hoop stress acts along the circular direction of the tube, but the stress in a perpendicular direction acts in the longitudinal direction of the tube, and the strains in this direction are constrained by the great length of the tube. Consequently, a triaxiality effect is induced, and the stress to be observed is the Von Mises stress (**Figure 7**). This triaxiality effect induces an over stress to obtain the yield of the material. Here, the overstress factor is $t = \sigma_{VM}/\sigma_Y = 760/528 = 1.44$. The maximum value is about 760 MPa for an internal pressure of 12 MPa.

Calculated codes give other results:

Table 1: Different ultimate pressure
Tabela 1: Različne končne napetosti

	P_{ult}/MPa	Error compared between experimental result (%)
ASME B31 G	11.3	5.8
Modified ASME B31 G	10.8	10
DNV RP-F101	6.6	45

To take account of the grinding in these calculations, we reduced the value of the ligament under the notch to 2 mm. According to the experimental result, it seems that the ASME B31G code is the closest. The DNV RP-F101 is the most conservative code.

6 CONCLUSION

It has been shown in this paper that the construction of a meshed structure that conveniently represents a notched pressure pipe is quite difficult. However, the results of the FE computations give consistent results with respect to the experimental data.

To improve the prediction of fracture of the notched pressure pipes it is necessary now to improve the meshing of the notch. Moreover, experiments provided different data that will be compared with further FE computation predictions.

7 REFERENCES

[1] J. Capelle, J. Gilgert, G. Pluvinage, Ch. Schmitt: Calcul du facteur de sécurité associé à la ténacité d'un tuyau de faible epaisseur, Paper presented at the conference IFCAM01, **(2006)**

[2] J. Capelle, M. Lebienvenu, G. Pluvinage: Hydrogen effect on fatigue life of a pipe steel, Paper presented at the conference ICMFM XIII, **(2006)**

[3] ASME B31G-1991: Manual for determining the remaining strength of corroded pipelines, The American Society of Mechanical Engineers, New York, USA, **(1991)**

[4] DNV-RP-F101: Corroded pipelines, Det Norske Veritas, 1999

[5] G. Pluvinage, J. Capelle: Etude d'un dimensionnement de conduite de gaz basée sur la mécanique de rupture et l'analyse limite, Paper presented at the 5ème Journée de Mécanique de l'Ecole Militaire Polytechnique, 2006

[6] J. B. Choi, B. K. Goo, J. C. Kim, Y. J. Kim, W. S. Kim: Development of limit load solutions for corroded gas pipelines, International Journal of Pressure Vessels and Piping 80 (**2003**), 121–128