J CALCULATIONS UNDER COMBINED LOADINGS FOR CIRCUMFERENTIALLY CRACKED PIPES AND THEIR IMPACT ON STRESS CLASSIFICATION FOR FRACTURE ASSESSMENT

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SUMMARY

The work presented here deals with elastic and elastoplastic finite element calculations of a circumferentially cracked pipe section. The crack depth to thickness ratio a/t is 1/8, 1/4 and 1/2 and the inner radius to thickness ratio Ri/t is 35.6. The material is 316L stainless steel at 450°C. The loadings considered are:

- axial force, axial displacement, internal pressure and combined loadings,
- through the wall thermal gradient combined with pressure.

The various values of J obtained are compared to the evaluations derived from a simplified method using the elastic J calculation as a basic input. The conclusions give some guidelines on stress classification for the analysis of circumferentially cracked pipes submitted to complex combined loadings such as pressure peak, thermal expansion and through the wall bending residual stresses.

1 INTRODUCTION

The elastoplastic fracture criterion J is today very often involved for flaw assessment or for Leak Before Break arguments. In the case of piping systems, various loadings are combined such as pressure peak, thermal expansion and through the wall bending residual stresses when circumferential welded joints are concerned. Since it is not possible to perform detailed finite elements analysis for each industrial case met, validated simplified methods for J estimations are of great interest. It is clear that the assumption of all types of loadings as load controlled situations would lead to undue conservatism in J approach. That is why simplified methods don't take into account in the same way load-controlled and strain controlled situations and need stress classification prior to analysis. The aim of the present work is to bring some clarification on stress classification (or severity of loading) by comparing results of J calculations on a relatively simple geometry under various representative loadings that are relevant of piping systems: axial force, axial displacement, both combined, through the wall temperature gradient combined with pressure.

2 GEOMETRY AND MODELIZATION

2.1 Geometry

The geometry of the model is a cylindrical shell with an internal axisymmetrical crack (see fig. 1) with the following notations:

- a : crack depth ; Ri : internal radius ; Rm : mean radius ;
- t : shell thickness ; 2L : total length of the model.

All the calculations were done with Ri = 249 mm and t = 7 mm. The different a/t ratios considered are: 1/8, 1/4 and 1/2. In some cases, the influence of the length L on the results was also studied, the minimum value being L = 170 mm.
2.2 Material constitutive laws

The material considered is 316L(N) at 450°C. The material properties were derived from Appendix A3-1S of the French RCC-MR construction code (1).

For elasticity, we used a Young’s modulus:

\[ E = 157 \, 000 \, \text{MPa} \] and Poisson’s ratio \( \nu = 0.3 \).

For plasticity, we used isotropic hardening with the true stress strain relation illustrated by fig. 2 (the mean value of the 0.2 % yield stress was \( R_y = 147.5 \, \text{MPa} \)).

For thermal loading the thermal expansion coefficient was \( \alpha = 18.1 \times 10^{-6} \, \text{C}^{-1} \).

The calculations were performed using CEA Castem finite element system (2). Due to symmetry conditions, only half of the meridian plane is represented. Fig. 3 shows an example of mesh used. It includes 221 quadratic six-node triangles and 520 nodes giving therefore 1040 degrees of freedom. We can see the concentric arrangement of the elements around the crack tip.

The different types of loading applied were axial uniform tension, axial end of shell displacement, internal pressure acting also on crack faces with corresponding axial force \( P_{RM/2t} \), through the wall radial temperature gradient. These loadings were applied alone, and were eventually mixed according to the following cross-table:

<table>
<thead>
<tr>
<th>Axial uniform tension</th>
<th>Axial displacement</th>
<th>Internal pressure</th>
<th>Radial temperat. gradient</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial uniform tension</td>
<td>X</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Axial displacement</td>
<td></td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Internal pressure</td>
<td></td>
<td></td>
<td>X</td>
</tr>
<tr>
<td>Radial temperature gradient</td>
<td></td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>

3 COMPILENCE OF THE MAIN RESULTS OBTAINED

Concerning axial tension, axial displacement and combination of theses two loadings, the effect of the model length \( L \) was, as expected, very important and was quantified.

The results obtained in terms of \( J \) versus \( L \) for the different types of loads applied are shown at fig. 4 (for \( a/t = 1/4 \)) and fig. 5 (for \( a/t = 1/2 \)).

The definitions of \( P, Q \), and \( P+Q \) are as follows:

- \( P \): applied axial stress at shell end,
- \( Q = 150 \, \text{MPa} \) means an axial displacement giving the same value of elastic \( J \) as for \( P = 150 \, \text{MPa} \)
- \( P+Q \) (54 + 96 MPa) means that \( P = 54 \, \text{MPa} \) and \( Q = 96 \, \text{MPa} \) but in that case, the axial stress is applied at \( z = 300 \, \text{mm} \) while the axial displacement is applied at \( z = L > 300 \, \text{mm} \).

It appears clearly from fig. 4 and 5 that the loading severity increases with pipe length even with pure controlled displacement.

The same type of effect of the model length was found for the combination of axial displacement and internal pressure.

For the particular type of loading which is combined internal pressure and through wall temperature gradient, the length of the model was fixed at \( L = 170 \, \text{mm} \) and was proved sufficient to avoid edge effects.

The maximum temperature gradient is 200°C (100°C inner wall, 300°C outer wall) giving therefore a nominal elastic through the wall bending stress:

\[ \sigma_{bl} = E \alpha \Delta T/2 (1-\nu) \sim 406 \, \text{MPa} \].
Two levels of internal pressure were combined with this loading (and were applied at first): \( P_I = 3 \text{ MPa} \) (30 bars) and \( P_I = 6 \text{ MPa} \) (60 bars, out of the design range for this component) giving therefore an axial nominal stress \( P_I \frac{Rm}{2t} = 54 \text{ MPa} \) and circumferential stress \( P_I \frac{Rm}{t} = 108 \text{ MPa} \) (for \( P_I = 3 \text{ MPa} \)).

The results can be illustrated by fig. 6 summarizing the values of \( J \) obtained at maximum combined load versus \( a/t \).

It can be seen that the ratio elastoplastic over elastic \( J_p/J_e \) keeps close to 1 for \( a/t \leq 0.25 \) and \( P_I = 3 \text{ MPa} \) (A design acceptable pressure) while plasticity effects lead to very high values of \( J_p \) for the highest pressure \( P_I = 6 \text{ MPa} \).

4 TEST OF A SIMPLIFIED METHOD FOR \( J \) ESTIMATIONS

4.1 Presentation of the method

The purpose of the method is to give an upper bound of \( J \), called \( J_S \), starting from the elastic \( J \) value, \( J_e \). Based on the notion of reference stress and reference strain (3), it can be written as follows in the general case of pure mechanical load \( P \) and other types of loadings \( S \) called further secondary loadings:

If \( \sigma = f(\varepsilon) \) is the monotonic true stress-strain relation with yield stress \( \sigma_y \):

\[
J_S = J_e \left( \frac{\varepsilon_{ref}}{\varepsilon_e} \right) + \frac{1}{2} \left( \frac{(\sigma_{ref}/\sigma_y)^2}{(\varepsilon_{ref}/\varepsilon_e)} \right)
\]

where:

a) If the secondary loading is purely kinematically determined:

\[
\varepsilon_{ref} = f^{-1}(\sigma_{ref} P) + (\sigma_{ref} P + Q \cdot \sigma_{ref} P)/E
\]

b) or, if not:

\[
\varepsilon_{ref} = f^{-1}(\sigma_{ref} P) + \lambda (\sigma_{ref} P + Q \cdot \sigma_{ref} P)/E
\]

\( \lambda > 1 \)

and:

- \( \sigma_y \): yield stress
- \( \varepsilon_e = f(\varepsilon_{ref})/E, \sigma_{ref} = f(\varepsilon_{ref}) \)
- \( \sigma_{ref} P = \sigma_y \times P/P_{L}(a, \sigma_y) \)
- \( \sigma_{ref} P + Q = \sigma_y \times (P + Q)/(P + Q)_{L}(a, \sigma_y) \)

where: \( P_{L} \) is the limit load of the flawed structure with crack depth a with perfectly plastic behaviour and \( P \) type mechanical load.

\( (P + Q)_{L} \) is also the limit load with \( P + Q \) type loading assumed to be fully mechanical.

A number of limit load solutions can be found in (4).

Fig. 7 illustrates the different parameters involved in the method. The \( \lambda \) parameter characterizes the severity of the secondary loading or degree of "elastic follow up".

This \( J_S \) method is discussed within a working group of the french RCC-MR committee working on a project of Appendix devoted to flaw evaluation called Appendix A16 (see also paper(5) of the present SMIRT 12).

4.2 Application of the method

The method was applied to the cylindrical shell already presented and several values of \( \lambda \) where secondary loadings are concerned were determined to give conservative values of \( J_S \) when compared to finite element results.

Table I, for example, gathers the results obtained for \( a/t = 1/4 \), axial force, axial displacement and combination.


\[
\begin{array}{|c|c|c|c|c|}
\hline
\text{Load type and value} & L & J_e (kJ/m^2) & J_p (kJ/m^2) & \lambda & J_s (kJ/m^2) \\
\hline
P = 150 \, MPa & 300 & 1.36 & 13.3 & 21.7 & \\
\hline
Q = 150 \, MPa & 300 & 1.36 & 3.69 & 1 & 3.34 \\
& 900 & 1.36 & 3.89 & 2 & 4.46 \\
& 1800 & 1.36 & 4.02 & 3 & 5.85 \\
\hline
P = 54 \, MPa & 600 & 1.36 & 4.26 & 1 & 3.34 \\
+ & 1800 & 1.36 & 5.56 & 4 & 5.73 \\
Q = 96 \, MPa & 30300 & 1.36 & 11.8 & 11 & 12.1 \\
\hline
\end{array}
\]

**TABLE I**

Js method compared to F.E results : axial force, axial displacement and combination

The high values of \( \lambda \) necessary to give conservative values of \( J \) show the high degree of "elastic follow up" in the problem when long straight pipe sections are concerned. The method was also applied to combined internal pressure and temperature gradient (see table II).

\[
\begin{array}{|c|c|c|c|c|c|}
\hline
\text{Load} & a/t & J_e (kJ/m^2) & J_p (kJ/m^2) & \lambda & J_s (kJ/m^2) \\
\hline
P_i = 3 \, MPa & 1/8 & 2.78 & 2.33 & 1 & 7.89 \\
& 1/4 & 5.45 & 5.65 & 1 & 20.3 \\
\Delta T = 200^\circ C & 1/2 & 12.98 & 24.4 & 1 & 102.8 \\
\hline
P_i = 6 \, MPa & 1/8 & 2.99 & 7.35 & 1 & 9.69 \\
& 1/4 & 6.02 & 19.1 & 1 & 33.25 \\
\Delta T = 200^\circ C & 1/2 & 15.7 & 99.21 & 1 & 353.04 \\
\hline
\end{array}
\]

**TABLE II**

J_s method compared to F.E results : combined internal pressure and temperature gradient

The \( J_s \) method appears to be conservative with large margins. It could be refined by using a more relevant limit load solution (a flat plate solution was used there). The \( \lambda \) value \( \lambda = 1 \) considered here shows that this type of loading is less severe than the preceding one, as expected, because thermal through wall bending is purely kinematically determined on the defect free structure.

5 CONCLUSIONS

J calculations were performed on a circumferentially cracked pipe section with various types of loadings eventually combined. It appeared through the study that axial displacement leading to membrane strains could be a severe loading when long straight pipe sections are concerned. Therefore, care should be taken when considering thermal expansion loads although a further task will aim at evaluating the effect of this type of loading in more realistic conditions (global bending by way of imposed end of shell rotations with part-circumferential surface defects). Through the wall temperature gradient is a less severe type of loading although it can lead to high values of \( J \) when combined with high values of internal pressure.

First comparisons between a simplified J approach, \( J_{sp} \) (useful for engineering applications) with finite elements results are satisfactory but show also that the topic remains difficult.

It appeared particularly that the limit load solutions data-base should be increased for combined loadings such as internal pressure with through wall bending.

Acknowledgments : this work was sponsored by EDF.
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Figure 1: GEOMETRY OF THE CRACKED PIPE

Figure 2: TRUE MONOTONIC STRESS-STRAIN
RELATION FOR 316L (N) AT 450°C

Figure 3: EXAMPLE OF F.E MESH USED
FOR THE PARAMETRIC STUDY
Figure 4: MAXIMUM J VALUES FOR a/t = 1/4

Figure 5: MAXIMUM J VALUES FOR a/t = 1/2

Figure 6: MAXIMUM J VALUES FOR COMBINED INTERNAL PRESSURE P_i AND RADIAL ΔT = 200°C

Figure 7: ILLUSTRATION OF THE PARAMETERS USED IN THE JS METHOD