Q&A Session for CCOPPS: Fatigue of Welded Pressure Vessels

Session number: 661284552

Date: May 27, 2008

Q: I would be interested in an example showing an illustrative comparison of the 4 weld modeling approaches: Continuum, shell (unreinforced), shell (angle), shell with thickness increase.

Jim Wood

A: There is one in the FE module. Send me an e-mail and I will send you a summary if interested. The work has not been published.

Q: Do we have access to the FE Module?

Jim Wood

A: The module is not finished yet. It will be offered to industry as a web-based course early next year.

Q: Steve mentioned that welds usually exhibit fatigue crack growth, why is a stress based (S-N) curve being used instead of a fatigue crack growth (da/dN) approach used in BS PD 5500?

Steve Maddox

A: The point I made was that because welds contain sharp imperfections, notably at the weld toe in the region of already high stress concentration due to the weld geometry, fatigue cracks initiate very readily and the resulting fatigue life consists mainly of crack propagation. It is very difficult to detect the point at which a fatigue crack is definitely present but my own early work in this area produced cases of weld toe fatigue failure where there was definite evidence of fatigue cracking after 6% of the total life. Without the weld it could take up to 95% of the life to initiate a crack, depending on the level of stress concentration present. In view of this situation, it is possible to calculate the fatigue lives of welded joints using fracture mechanics fatigue crack growth analysis (i.e. integrating the fatigue crack growth law relating the applied stress intensity factor range (ΔK) and rate of crack propagation (da/dN) between the limits of initial flaw size and crack size at failure. However, apart from the case of fatigue failure from identifiable crack-like flaws (like the incomplete penetration region of a cruciform fillet welded joint that could then fail by crack growth through the weld throat) fatigue cracks in welded joints initiate at minute flaws, typically 0.1 to 0.4mm deep in the case of weld toe imperfections of the kind I showed in one of my early slides. These are not detectable by non-destructive means (and actually rather difficult to find from metallurgical cross-sections) and so it is necessary to assume their sizes when using fracture mechanics crack growth analysis to calculate the fatigue life. So, testing the actual welded joint is the only way to determine its fatigue life accurately and such test data are the most reliable basis for calculating the service life of a similar welded joint in a real structure.

Q: It looks to me that ASME VIII is based on a strain based approach which would require strain based testing not stress based. Is that correct?

Steve Maddox

A: There is no doubt that the 'old' ASME fatigue rules, using polished specimen S-N curves with stress concentration factors, was based on low-cycle fatigue considerations and hence fatigue data obtained under strain cycling conditions. However, the new structural stress-based approach relies most heavily on high-cycle fatigue data obtained under load control since nominal stresses are all elastic. This is also the case with the PD 5500 and EN 13445 rules. The assumption has been made, which is supported by low-cycle fatigue data, that as long as shakedown occurs in the pressure vessel the high-cycle S-N curves simply extrapolated back to the low-cycle regime on the basis of the pseudo-elastic stress range, strain range x E, will still be applicable.

Q: How does the ASME consider the mean stress to construct their fatigue curve? They say that the curve is based on the mean stress.

Steve Maddox

A:

I have not been able to find a clear background document explaining the new structural stress -based ASME VIII fatigue rules and my understanding is based partly on technical papers produced by Pingsha Dong and colleagues or deductions based on my own knowledge. As far as the mean stress correction is concerned my guess is that the 'master curve' was derived directly from fatigue test data obtained mainly under approximately zero-tension loading from small-scale welded specimens that could not be relied upon to contain high tensile residual stresses, as we would expect to find in real structures. Thus, no correction is needed for $R \leq 0$ but a penalty is required for $R>0$.

In contrast, if use is made of the polished specimen S-N curves, as in the previous editions of ASME VIII the design curves already include the worse effects of mean stress and so no further correction is needed.

Q: has DBA manual been released?

Jim Wood

A: Do you mean DBA module or manual? I will assume that you mean module (last time I tried to find the DBA manual on-line all I could find was an errata). The DBA module is not finished yet. It may be offered to industry as a web-based course early next year.
Companies selected to help with the evaluation of the 2 modules and the Educational Base will get a copy of these free for internal non-commercial use.____

Q: How does this compare with RCC-M in France?

Steve Maddox

I am still investigating this and hope to provide an answer later.

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Q: can you show Fatigue life improvement methods please

Steve Maddox

A. One of my slides noted that all three Standards allow the use of weld toe improvement methods, only toe grinding in the case of PD 5500 and EN 13445, but also hammer peening and TIG dressing in ASME VIII. I added that a good feature of the credit given by ASME was that it increased with decrease in applied stress (i.e. it leads to a rotation of the S-N curve for the as-welded joint), which is consistent with actual test data. The European Standards are more cautious and just allow a parallel shift upwards of the as-welded S-N curve. However, I added that care was needed in the case of residual stress-based techniques, like hammer peening, in that they are unlikely to produce any benefit at all in the low-cycle fatigue regime.

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Q: Is there any basis for post processing the stress in shell elements used to represent a fillet weld? Can fatigue be predicted from this stress?

Jim Wood

A: The methods used for the modelling of shell intersections in general and with welds in particular are many and varied. The new version of ASME VIII has come off the wall and includes details of how to use shell element stresses. There is even a diagram of a shell model where an attempt has been made to try and include the weld profile. However there are variations on this theme depending on whether you have full penetration or not (details not given). The recommendation is simply made that you ensure that the joint stiffness is correct and that’s about the extent of the modelling advice.

There are quite a few different approaches to using shells out there and I think the key is validation of any technique you use with fatigue test data for your actual configuration if possible.

Steve Maddox

A. TWI are comparing methods of calculating stresses calculated from a wide range of mesh types, including brick and shell models, and linear and quadratic elements. Three methods of calculating the appropriate stresses have been investigated: Surface Stress Extrapolation (SSE), Through Thickness Integration (TTI) and Nodal Force (NF). SSE uses stresses at two locations near the weld and extrapolation is used to get the weld toe value. TTI and NF are used to calculate the membrane load and bending moment at the weld and then a stress is calculated from these values. TTI uses the through thickness the stress distribution and NF uses the FE calculated forces at nodes for these calculations. The study includes examination of the effect of excluding fillet welds, modeling them as inclined elements or simulating their effect by increasing the plate thickness locally. An extensive comparison of these methods has been made some complex geometries, taking the fine brick mesh as a benchmark. As noted in the presentation, the outcome of this project will form the basis of recommendations to be included in a revision of BS 7608.

Q: did you make reference to 2007 revision of ASME VIII div2 in this presentation?

Steve Maddox

A. Yes, all the information I presented about the ASME rules was based on this version of the Standard.

Q: Can u suggest some references to understand fatigue in weld. Something which has some numerical examples

Jim Wood

A: Steve's book is excellent for background and understanding, but doesn't cover the simulation (*Fatigue Strength of Welded Structures*; Woodhead Publishing; Second Edition; ISBN 1 85573 013 8; 1991). John Draper's new book on Modern Metal Fatigue Analysis contains a chapter on FEA, but not in the context of pressure vessels (EMAS Publishing Ltd. ISBN 0947817794); 2007. There is also a chapter on Fatigue of Welded Steel Joints with a section on hot-spot stresses. Another book worth having in your library is Fatigue Analysis of Welded Components. Design Guide to the Structural Hot-spot Stress Approach; *Woodhead Publishing Ltd.*; ISBN-13: 978-1-84569-124-0, 2006.

There are numerous papers out there on modeling ideas for welds, but I think it is true to say that last word has yet to be said on the matter.

Steve Maddox

A: In addition, you should still be able to find copies of the standard textbook (out of print) produced by my now retired mentor Dr T R Gurney 'Fatigue of welded structures, 2nd Edition, Cambridge University Press. He also has a new book, 'Cumulative damage of welded joints', ISBN-13: 978-1-85573-938-3, Woodhead publishing Ltd, Cambridge, UK, which includes some of the basic material from the earlier one.

With regard to worked examples, see Annex W of BS PD 5500 and the Case Studies in Niemi, Fricke and Maddox 'Fatigue Analysis of Welded Components. Design Guide to the Structural Hot-spot Stress Approach' mentioned by Jim above.

Q: Is there any reliable tool available to simulate the fatigue life in welded joints in pressure vessels

Jim Wood

A: The basic tool is an FE system. These are also pre- and post- processors such FE-Safe (also has Battelle's Verity) and n-Code that you can buy as add-ons. These can make life a lot easier if you do a lot of fatigue analysis. Some FE systems already have fatigue facilities built-in so have a look at these before assessing the need for one of the specialist systems. Getting internal nodal forces for Dong's method may be problematic with some FE systems but all will have tools for through-thickness linearization.

Q: What are the softwares (FEM) which can allow fatigue in weld calculation?

Jim Wood

A: All finite element systems will allow you to model and analyze welds (see previous question). When it comes to simulating cyclic plasticity in the joint then some systems are better than others however.

Q: Is the ASME approach along the lines of Verity?

Jim Wood

A: One of the ASME Structural Stress methods is (nodal forces). As Steve mentioned, this requires agreement from all parties. There is also through-thickness integration, which is a long-standing technique for removing peak stresses as part of linearization (2 variations on this). The use of Fatigue Strength Reduction Factors is also still there as well.