

## Tutorial Session 12B: Residual Stresses

Last Update: 28/5/14

Relates to K&S question 1.19

*What are “residual stresses” and how are they generated? How can mechanically induced residual stresses be estimated? How can welding residual stresses be calculated? How can residual stresses be measured? When are residual stresses deleterious? What are the welding residual stresses for a circumferential butt weld, and how do these vary with thickness and heat input? What advice is available for welding residual stresses in other geometries? What residual stress distributions might apply for weld repairs, and how do these vary with repair length? What controls the “yield” magnitude of welding residual stresses?*

**Qu.:** What are “residual stresses”?

Residual stresses are the stresses left behind in a structure when the original source of loading has been removed but stresses remain as a result of the yield stress having been exceeded when under load. Residual stresses occur only because plasticity is irreversible.

Since residual stresses apply when the load has been removed (by definition) they are self-equilibrating.

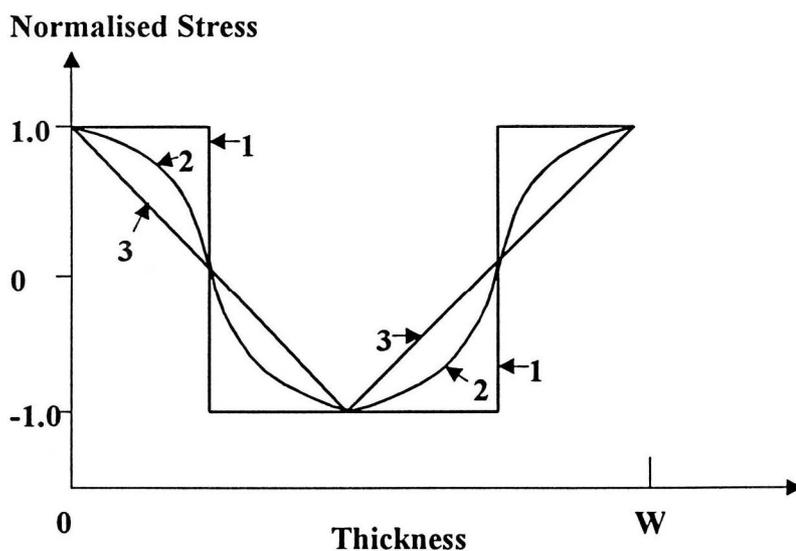
**Qu.:** What does “self equilibrating” mean?

In this context, a “self-equilibrating” stress distribution,  $\sigma_{ij}(\bar{r})$ , is one which obeys  $\sigma_{ij,j} = 0$  at every point within the structure, including on its surface – i.e., residual stresses correspond to zero applied load.

**Qu.:** Does this mean that the load resultants,  $N$  and  $M$ , are zero?

No, not necessarily.

R6 §II.6.7 refers to secondary stress distributions through a shell wall for which  $N$  and  $M$  are zero as “completely self-balancing”, examples being shown in R6 Fig. II.6.3.....



However, residual stresses, e.g., those arising from welding, need not have either zero  $N$  or  $M$ . On the contrary, as we shall see, non-zero net membrane or bending load resultants can be the source of onerous stress intensities in fracture assessments.

**Qu.: How can  $N$  and  $M$  be non-zero?**

Consider a circumferential butt weld, for example. There can be a non-zero  $N$  at a given circumferential position even though there is no net axial load. Equilibrium requires that the integral of  $N$  around the circumference be zero, not that it be zero everywhere. Non-zero  $N$  can arise due to, (a) weld pass start-stop effects, or, (b) weld repairs.

Even if perfect axisymmetry is assumed, so that  $N$  is zero everywhere, the bending load resultant,  $M$ , will still in general be non-zero. This does not violate equilibrium because the vectorial integral around the circumference of any axisymmetric  $M$  is always zero. (This is another way of saying that axisymmetric wall-bending is always secondary).

**Qu.: Are residual stresses caused *only* by welding?**

No.

Any loading which causes yielding will, in general, leave behind a distribution of residual stresses when unloaded.

**Qu.: What about a bar loaded in uniaxial tension?**

That's the one exception!

The uniform stressing means that the stresses return to zero on unloading, even if the yield stress has been exceeded. This can cause confusion. To envisage how residual stresses arise due to mechanical loading always consider a case of non-uniform stressing, e.g., a bar in bending or a bar with a notch.

**Qu.: How can residual stresses be calculated?**

That's a big question!

Because irreversible plasticity with non-proportional loading is involved in an essential way, exact analytic calculations will rarely be possible. Accurate calculations will usually require finite element analysis, especially if work hardening is involved. The modelling of mechanically induced residual stresses using FEA is relatively simple. The modelling of welding induced residual stresses is far more complicated and is discussed further below.

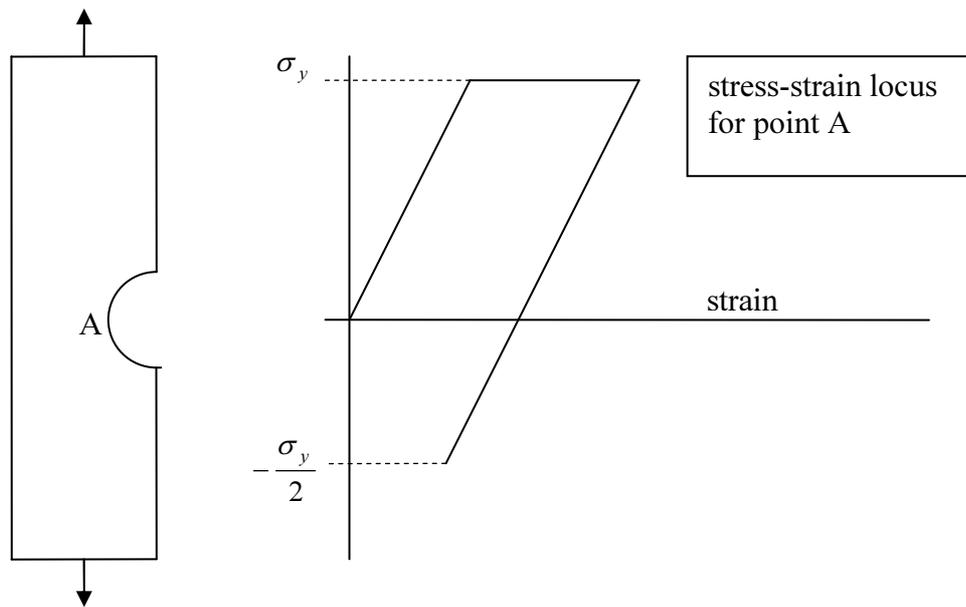
However, mechanically induced residual stresses can sometimes be estimated simply (see next item) and compendia exist offering advice on welding residual stresses (see later).

**Qu.: How might mechanically-induced residual stresses be estimated simply?**

The following illustrates a method which can give a rough indication of residual stresses. Your structure or loading may not always be simple enough for this to work, and the accuracy is variable. Nevertheless the following is instructive as to the mechanism by which residual stresses arise.

Consider a bar with a notch under uniaxial tension. The stress concentration factor of the notch is (say)  $SCF = 3$  with respect to the nominal applied tension, which is (say) half yield,  $\sigma_y / 2$ . The material is assumed elastic-perfectly-plastic.

Figure 1



The method is,

- Start with the elastic distribution of stress under load;
- Replace all stresses exceeding yield (in magnitude) by yield (possibly compressive yield);
- Model the effect of unloading by subtracting the initial elastic stress distribution;
- Again replace any stresses exceeding yield (in magnitude) by yield (possibly compressive yield).

Of course this is just a crude estimate because it makes no allowance for stress redistribution from the yielded regions to elsewhere. It is only intended to give a rough qualitative guide.

For the above problem it indicates that the residual stress at the notch root will be

$$\sigma_y - \frac{3}{2}\sigma_y = -\frac{1}{2}\sigma_y \text{ (i.e., compressive).}$$

[Aside: If Neuber applies, the reader may like to show that the strain at point A after unloading is  $\frac{3}{4}\epsilon_y$ , where  $\epsilon_y = \frac{\sigma_y}{E}$ . Note that the strain is positive despite the stress being negative].

The above very crude estimation process can be made more accurate (or less inaccurate) by increasing the stresses in the non-yielded regions to account for stress redistribution away from the yielded regions. However there is no general, unique way of doing this simply (as far as I know).

Qu.: What is the classic “bent bar” problem?

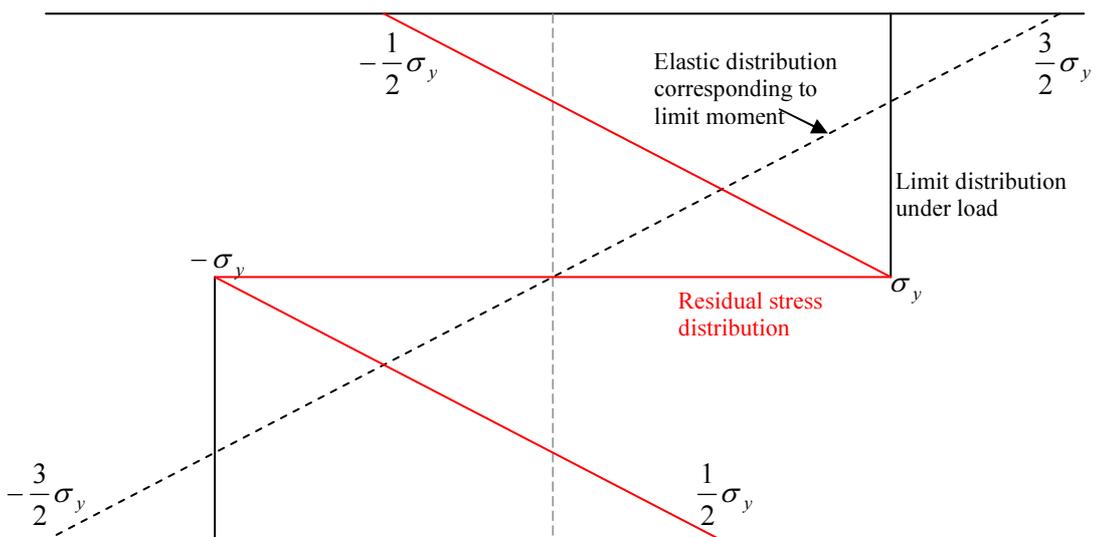
This is a problem which may be familiar to many of you from technical interviews. A solid bar of rectangular section is initially straight. The material is elastic-perfectly-plastic. A bending moment is applied to it and increased until the bar will take no more load, causing the bar to become permanently bent. The bar is then unloaded. What is the resulting residual stress distribution?

Qu.: How is the classic “bent bar” problem solved?

The following method provides a simple, but approximate, estimate of the residual stress distribution.

The moment applied to the bar is its limit moment,  $M_{\max}$ . The assumption of a “square wave” stress distribution under load (see Figure 2) leads to the lower bound estimate of the limit load  $M_{\max} = 1.5M_y$ , where  $M_y$  is the moment needed to just cause yielding of the outer fibre. If the bar had remained elastic up to moment  $M_{\max}$  the stress distribution would have been as shown in Figure 2 as the dashed line. The residual stresses are therefore obtained by subtracting this elastic distribution from the actual plastic distribution under load, leaving the “saw-tooth” residual stress distribution shown in Figure 2 as the red lines.

Figure 2



The simplicity of this problem leads to a residual stress with zero net load resultants,  $N = M = 0$ , as the reader can readily check. However this is not necessarily the case in more complicated (multiaxial) problems.

Qu.: Can residual stresses be deduced from the distortion of a body?

No.

**Qu.: How can FEA be used to calculate welding residual stresses?**

This is a big subject on its own. For austenitic materials there is a whole Section in R6 on this, i.e., R6 Chapter III, Section III.15. Here we just summarise some salient features.

Welds are usually made of many ‘passes’, each laying down an additional bead of weld. Thin sections will require only a few passes (e.g., typically 2 or 3 passes for 4mm thickness, or perhaps 7 or 8 passes for ~10mm thickness, but perhaps just one pass for ~2mm thickness). Thick sections can require in excess of 100 passes (say for >60mm thickness), but obviously this depends upon the weld prep, the size of welding rods, or wire, and how much the arc is ‘weaved’, not just the thickness.

A welding simulation ideally consists of modelling each weld pass. For each bead this consists broadly of two steps,

- (a) Modelling the heat input from the welding arc into the substrate material, and solving the subsequent heat transfer problem;
- (b) Elastic-plastic analysis of the transient thermal loading problem resulting from (a).

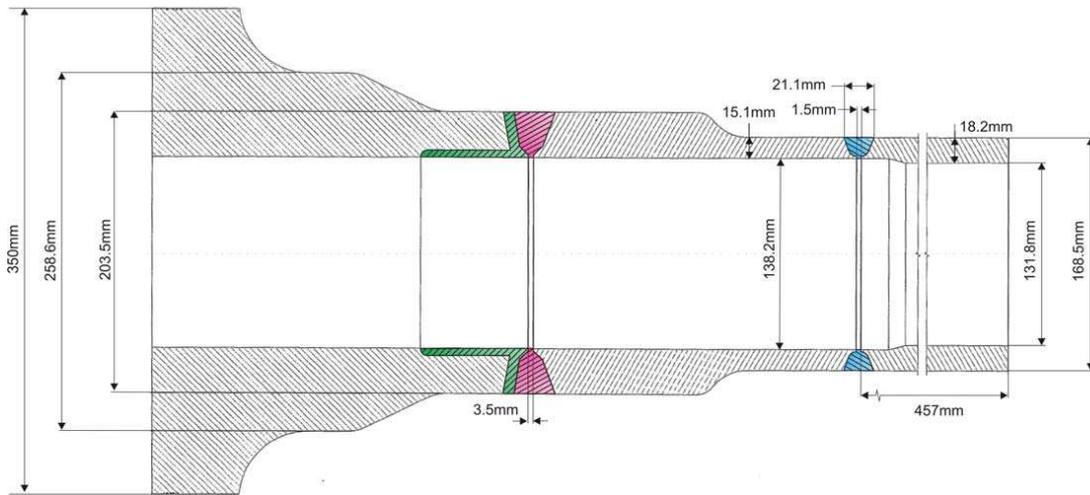
Some of the issues which arise include,

- (i) Unlike other FE analyses, welding involves gradually increasing the amount of material. Consequently the analysis must model the physical addition of material at every pass. Roughly speaking, the structure becomes stiffer as more weld is laid down.
- (ii) The heat transfer problem strictly involves conduction, radiation and convection. The convective aspect refers to the heat transfer occurring within the molten weld pool prior to solidification. In practice, however, the molten phase (and hence convection) is not usually modelled (though some sophisticated analyses do so). Moreover, radiation, although very significant, can often be addressed simply by factoring the heat input by some welding efficiency factor. The analysis can therefore be reduced in most cases to a heat conduction problem.
- (iii) Even for geometries which are 2D or axisymmetric, the welding process is *always* 3D. This is because welding involves a moving arc which melts material at a ‘point’ as it passes. The best simulations will therefore be 3D and will model this moving arc. Experience has shown that the start and stop positions of a bead produce the largest residual stresses. Whilst 2D/axisymmetric models are sometimes used, and can be adequate, they cannot model such start/stop effects and hence can only produce some sort of circumferentially averaged stress distribution. Such 2D/axisymmetric simulations are referred to as “block dumped” analyses since all the heat is assumed dumped into the whole circumference at once.
- (iv) A welding simulation requires material properties at all temperatures up to the melting point. This is unusual, of course, and it may be tricky to obtain data at the highest temperatures. However it is important to get reasonable strength data at the highest temperatures in order to model correctly when, during the cooling process, the structure starts to develop significant stress.

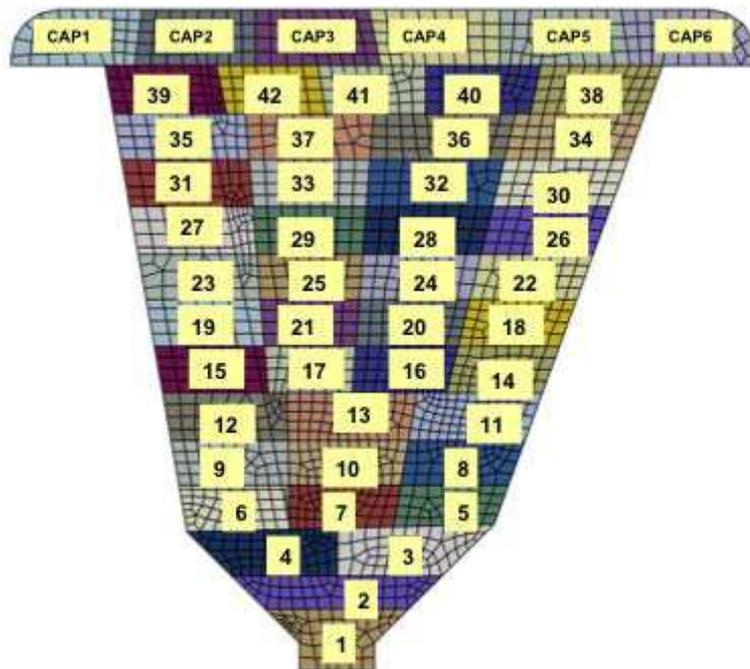
- (v) An elastic-plastic analysis step is required separately for each time point during the thermal transient caused by the weld pass in question, and this must be repeated for every weld pass. Combined with the requirement to model in 3D, and the requirement for a high degree of refinement in order to model all beads, the result is that welding simulations can be very computation-intensive, taking very long run times and generating large output files.
- (vi) Each weld bead causes a complete stress cycle, which, sufficiently close to the weld, will cause an elastic-plastic stress-strain hysteresis cycle, in general involving both tensile and compressive plasticity. Consequently all aspects of the material hardening are important in determining the residual stresses: the monotonic hardening, the cyclic hardening, and the reversing hardening (isotropic versus kinematic, etc).
- (vii) To produce results for the absolute magnitudes of the residual stresses which are in decent agreement with measurements, neither kinematic nor isotropic hardening will do (at least for the materials of interest to us). Isotropic hardening will over-estimate the residual stress magnitude, whereas kinematic hardening under-estimates. Good results for austenitic materials have been obtained using mixed hardening models (e.g., Lemaitre-Chaboche).

**Example: SZB Safe-End Weld: Ferritic/Inconel/316ss Transition Joint  
(Taken from Mike Smith, E/REP/BBGB/0070/SXB/10)**

**Figure 3a: GA**



**Figure 3b: Individual Weld Passes Modelled**



**Qu.: How is welding heat input specified?**

Welding kit specifies the voltage,  $V$ , and arc current,  $I$ , and hence the power input is known, namely  $VI$ . Whilst all this electrical power emerges as heat, not all of this heat ends up in the metal. Some of it is radiated away (welding arcs are pretty bright!). The fraction of the heat which enters the substrate metal is the welding efficiency,  $\eta$ , which varies considerably according to the welding technique. Typical values are,

<i>Process</i>	$\eta$
Manual metal arc	0.7 — 0.85
Tungsten–inert gas (argon)	0.22—0.48
Metal–inert gas (argon)	0.66—0.75
Submerged arc	0.90—0.99

However, the total power entering the metal,  $\eta VI$ , is still not a good measure of the intensity of the heat input because, depending upon the speed,  $S$ , at which the arc is moved over the workpiece this energy may be dumped into a larger or smaller volume of metal. A more useful measure is therefore the energy deposited by a single pass per unit length of bead,  $\eta VI/S$ , (units J/mm). This is probably the most commonly used measure of welding heat input.

Another commonly used heat input parameter, denoted  $q$ , is obtained by dividing the heat input per unit bead length by the thickness,  $w$ , of the section being welded,

$$q = \frac{\eta VI}{wS} \quad (1)$$

(units J/mm<sup>2</sup>). This is the heat deposited by a single pass per unit cross sectional area in the direction of travel.

Neither of these measures give the *total* heat deposited by the welding process, however, since they relate to just a single pass.

**Qu.: Is it necessary to model every bead individually?**

Where the number of weld passes is very large, say  $>100$ , it can be cumbersome and very time consuming to model every bead. Consequently analyses sometimes lump a number of beads together to reduce the effort required. However this is rather unsatisfactory for a number of reasons,

- It is not clear how heat input should be adjusted to be representative in such approximate simulations. For example, if a single modelled ‘pass’ represents (say) 4 physical passes each with heat input parameter  $q$ , it would not be appropriate to assume a modelled heat input of  $4q$  since this heat is not all applied at once. Some experience exists as to how to fiddle the heat input most appropriately (e.g., based on the correct heat flux per unit area of fusion surface), but this is less than fully satisfactory.
- Moreover, in reality each pass causes a stress-strain cycle within the substrate (HAZ) which contributes to the final stress state. Reducing the number of modelled passes means modelling too few cycles.

- Experience has shown that the last pass has a disproportionate influence on the final stress state. Using over-large beads risks exaggerating this effect.

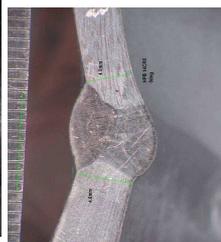
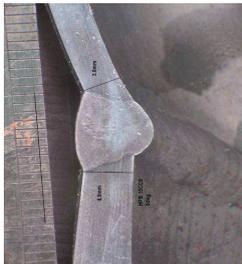
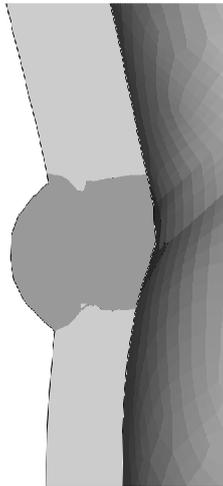
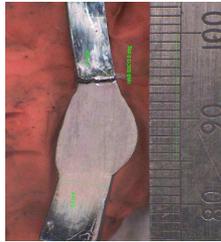
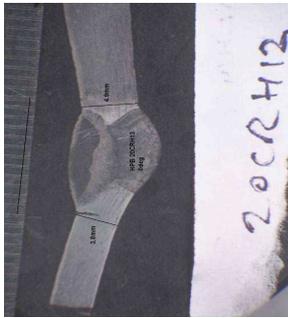
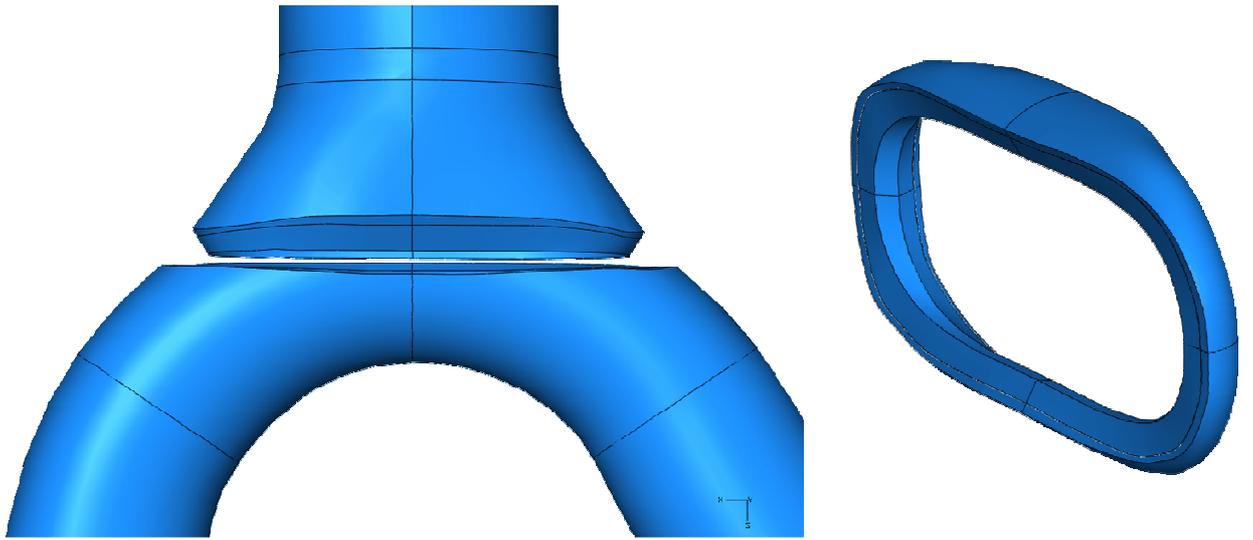
Qu.: Can the simulation be validated somehow?

A useful way to check that an FE welding simulation is roughly right is to compare the predicted bead fusion boundary profiles with a macro of a sectioned weld.

An example is shown in Figure 4 for the HPB/HNB superheater bifurcations, which involve a two-pass TIG weld simulated in 3D using a moving arc simulation.

A further example is shown in Figure 5 for the HPB/HNB superheater tailpipe to pintle welds. These are manual argon arc welds (i.e., metal inert gas, MIG), modelled with 3-passes as suggested by the macro. The weld procedure did not specify the welding current and so the heat input was estimated based on similar welds. The resulting fusion boundaries predicted by the FE model then provide a check that this estimated heat input is reasonable (see Figure 5). This was an axisymmetric model using the 'block dumped' method.

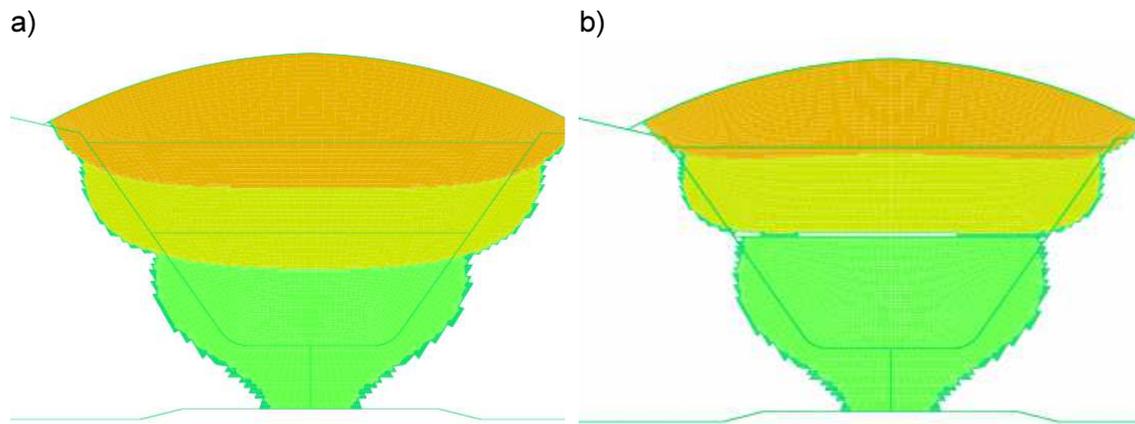
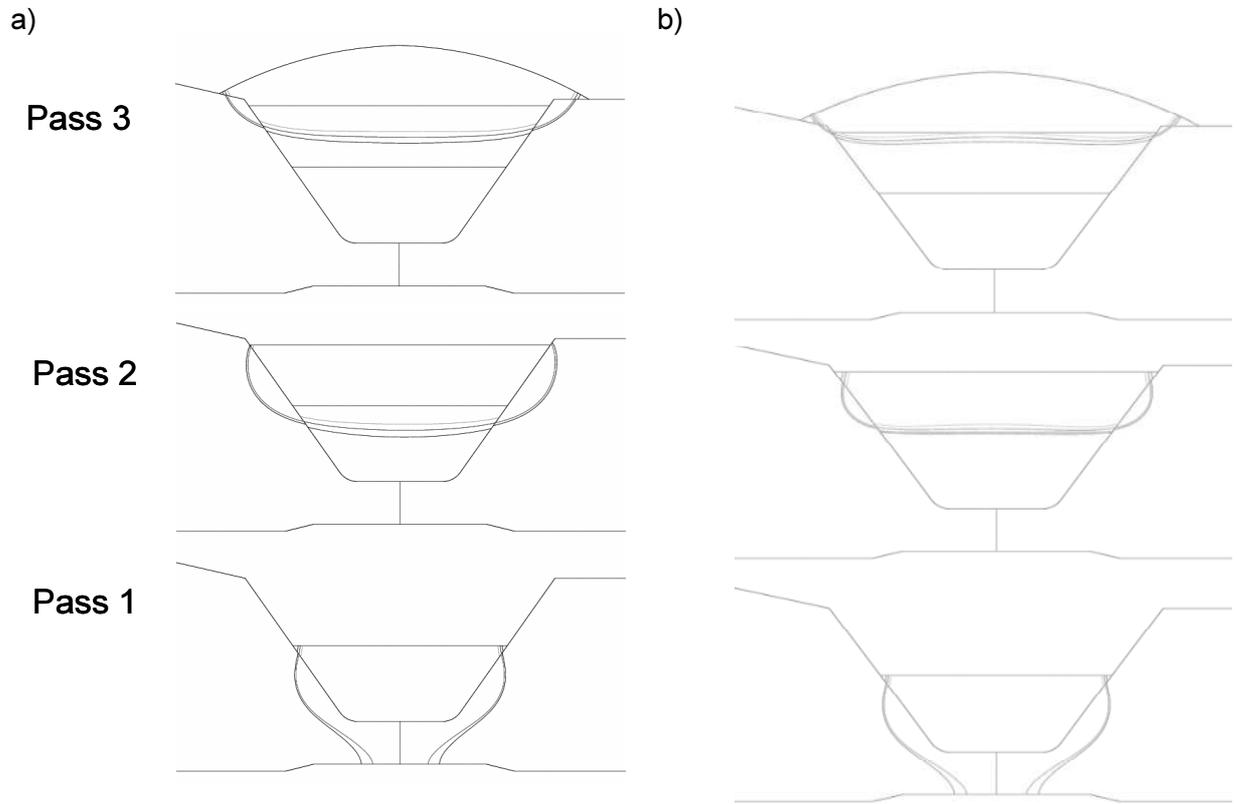
**Figure 4 HPB/HNB S/H Boiler Bifurcations**



**(macros at 0° crotch position)**



**Figure 5 (c): Finite element simulation – Position of elements representing the beads and predicted position of the fusion boundaries**



**a) Optimal Heat Input; b) Heat Sensitivity**

Qu.: Is there an analytic solution for the heat conduction from a moving arc?

Yes.

It is applicable for a point heat source moving at uniform velocity,  $v$ , over the surface of a semi-infinite slab (i.e., of infinite extent and infinitely thick). It is known as the Rosenthal solution, Ref.[1],

$$T = T_0 + \frac{P}{2\pi K \sqrt{d^2 + (vt)^2}} \exp\left\{-\frac{v\left(\sqrt{d^2 + (vt)^2} - vt\right)}{2\kappa}\right\} \quad (2)$$

where,

$T_0$  is the initial, or inter-pass, temperature;

$P$  is the welding power,  $\eta VI$

$d$  is the perpendicular depth of the point in question from the surface;

$t$  is the time (datum time zero being when the arc is directly above the point in question);

$K$  is the thermal conductivity;

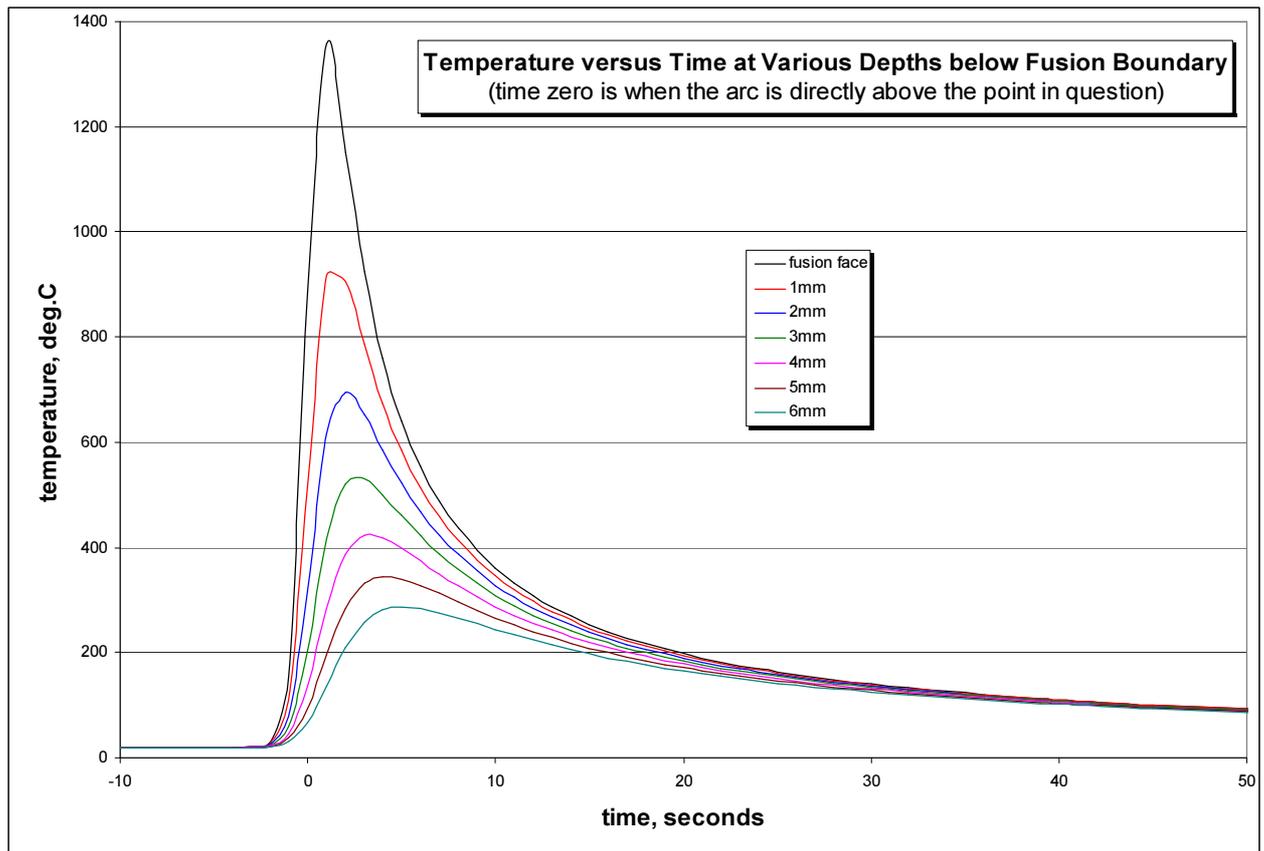
$\kappa$  is the thermal diffusivity =  $K / C_p \rho$

For mathematicians, the Rosenthal solution is a Green's function of the time-dependent heat conduction equation.

In practice the finite thickness of the substrate will mean that the workpiece will remain hot for substantially longer than implied by the Rosenthal solution. This is because, once the heat has penetrated the thickness it can then only disperse in two dimensions (in-plane) rather than in three directions.

**Figure 6: Illustration of Rosenthal Solution for Point Heat Source Moving Over a Semi-Infinite Slab.** (Note that the depth specified in the Figure is the depth from the plate surface minus 4mm, since 4mm was found to be the maximum depth at which the temperature reached the melting point,  $\sim 1400^{\circ}\text{C}$ )

$P = \eta VI =$	2025	J/s	(welding power)
$K =$	24.9	J/(m.s.K)	(thermal conductivity)
$v =$	0.0035	m/s	(speed of arc)
$\kappa =$	5.30E-06	$\text{m}^2/\text{s}$	(thermal diffusivity)
$T_0 =$	20	$^{\circ}\text{C}$	(inter-pass temperature)



**Qu.: Why does R6 III.15 not cover ferritic weldments?**

Ferritic weldments undergo a phase change as they cool down from the molten state. The phase change involves a volume change (due to the different FCC and BCC packing fractions). This volume change implies straining and hence is a separate source of residual stress. Special FE codes are required to model the strain due to the phase change.

**Qu.: Can residual stresses be measured?**

No!

Of course I'm being pedantic. My point is that all the standard methods of "residual stress measurement" are really measurements of strain or strain change – which must then be converted to stress by assuming some appropriate elastic moduli. As far as I am aware no one has ever strictly measured residual stresses.

**Qu.: OK, how can residual strains – hence stresses – be measured?**

This is a huge subject in its own right. Some of the methods available are,

**X-ray diffraction:** Non-destructive providing that you can get your specimen to the source of X-rays and the spectrometer. I think X-ray diffraction only gives near-surface stresses due to limited penetration, but I could be wrong. This method seems not to have been used much by us (EDF Energy / BE) recently, though I believe it is widely used by others. Many years ago the results I saw using X-ray diffraction were poor, but things have probably moved on.

**Neutron Diffraction:** Of late this has become the method of choice where possible. It has the advantage of providing results quite deep into a specimen, and also resolving the different stress components. Hence it provides information on stress triaxiality, for example, which is crucial in reheat cracking. Nominally non-destructive but the size of sample which can be analysed is restricted by the physical size of the diffractometer, which may require chopping down a larger item. Also you need to take your sample to the neutron source (i.e., an accelerator / spallation source). And you need to provide a piece of the same material in the unstressed state to permit the datum lattice parameter to be measured.

**Shallow Hole Drilling:** A small diameter, shallow hole is drilled at the centre of a strain gauge rosette. The measured change in strain allows the initial stress to be determined. It is basically non-destructive, the holes being very small. I suspect this method is not now regarded as accurate (but I may be wrong). One of the problems with it is that the surface will require some dressing for the attachment of the strain gauge – and this can alter the near-surface residual stress, which is all this method can measure.

**Deep Hole Drilling:** A small diameter reference hole is drilled, and its diameter measured at various orientations and depths. Residual stresses in the vicinity of this hole are caused to relax by trepanning at a larger diameter about the reference hole as centre. The diameter of the reference hole is again measured at the same orientations and depths, and the changes in these measurements are used to reconstruct the initial stresses. Destructive.

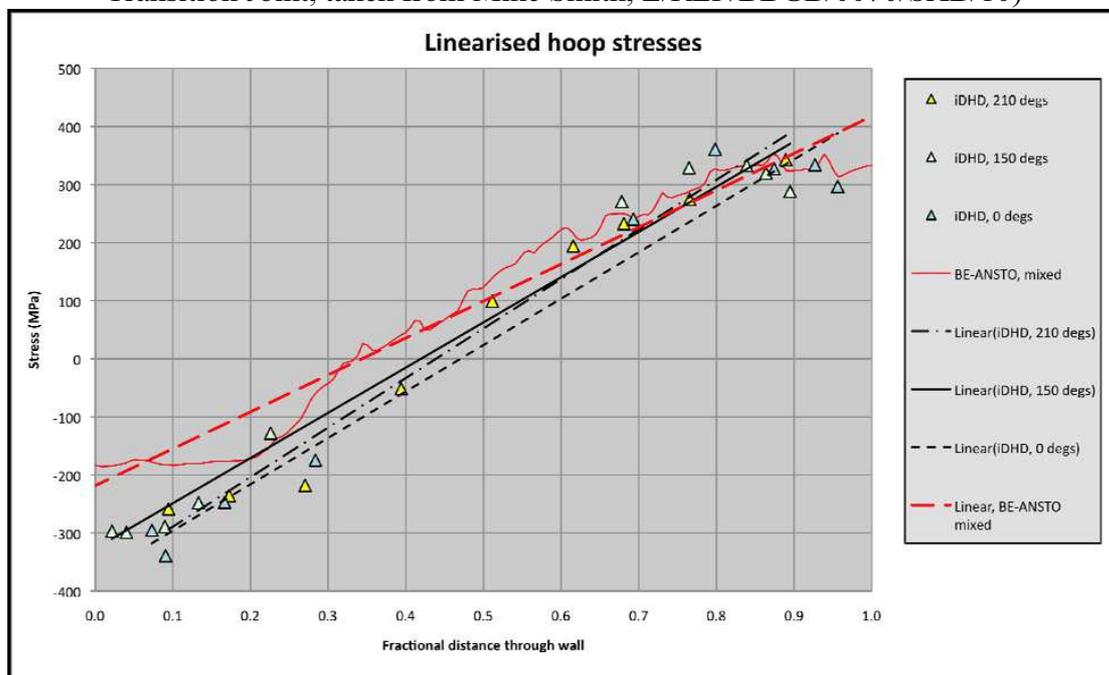
**Slotting:** Destructive. The concept here is similar to that of hole drilling. Residual stresses are relaxed incrementally by gradually increasing the depth of a slot machined across the specimen and this is detected using strain gauges. Requires FEA to convert the strain gauge data to initial stresses and the ‘calibration’ will be geometry specific. Has produced results in fairly good agreement with neutron diffraction for austenitic CT specimens.

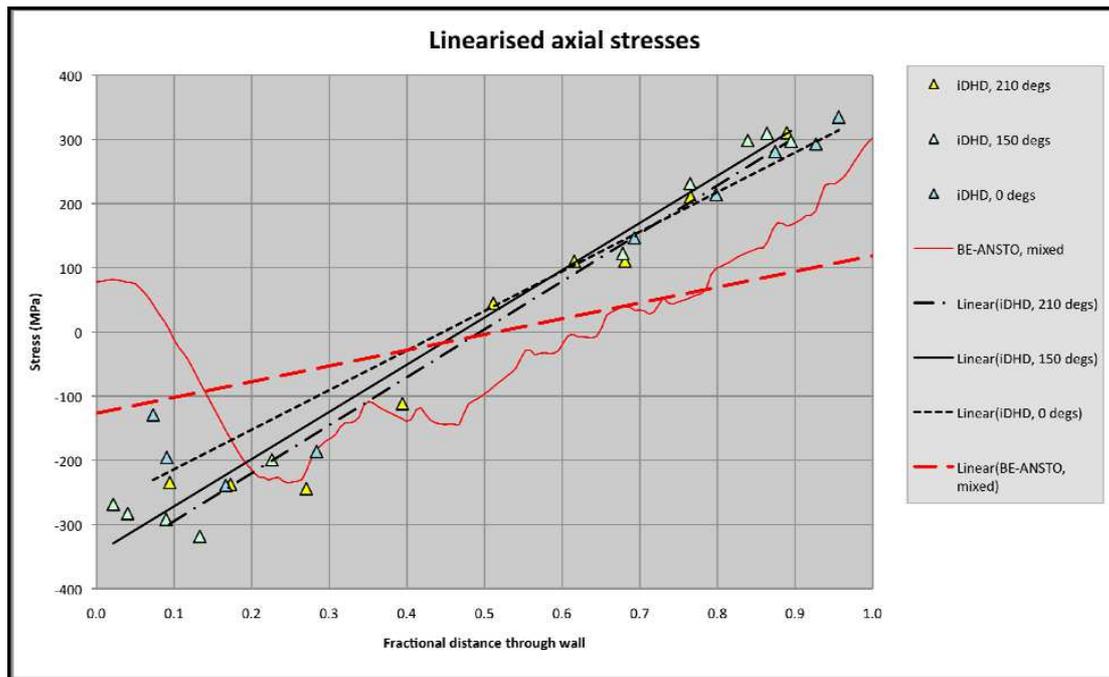
**Barkhausen Noise:** This is applicable only to ferromagnetic materials (e.g., ferritic steels). It is one of the few techniques which is fully non-destructive and also portable and hence can be deployed on-plant. Barkhausen noise is the discontinuous change in magnetisation when the magnetic field applied to a ferromagnetic material is changed (arising from sudden magnetic domain re-alignment). The amount of Barkhausen noise varies with stress state, hence it provides a means of measuring stress (but only indirectly, by correlation, because it is the magnetisation which is directly measured). I don’t know how accurate it might be.

**Qu.:** How well do FE simulations agree with residual stress measurements?

This is another big topic in its own right. I will only say this: fifteen years ago my view was that FE simulations of residual stresses would only ever give a rough indication of the general shape of the distribution. I expected that any close agreement of residual stress magnitudes would forever elude us. This has turned out to be unduly pessimistic. In fact, modern analyses and measurements are converging nicely – at least in those cases in which the welding is sufficiently well controlled, and well characterised, to permit a truly representative model. Example...

**Figure 7: Comparison of measured and predicted hoop residual stresses on SZB Safe-End Weld centre-line after removal of weld cap (Ferritic/Inconel/316ss Transition Joint, taken from Mike Smith, E/REP/BBGB/0070/SXB/10)**





**Qu.: What compendia of welding residual stresses are available?**

The obvious compendium of welding residual stress profiles is that in R6 (Chapter IV, Section IV.4), Ref.[2]. The rather old compendia of Barthelemy, Ref.[3], and Bate, Ref.[4], will have been taken into account in compiling the R6 advice, but it is possible they include material not assimilated into R6. There is also welding residual stress advice in Annex Q of BS7910, Ref.[5], but this is also likely to be very similar to that of R6 (I've not checked though).

Ref.[6] provides a summary of various finite element welding simulations of austenitic AGR components carried out under the generic reheat cracking programme in the mid-1990s. Because of the intended application this source is unusual in containing information on stress triaxiality and elastic follow-up factors,  $Z$ , for the residual field.

Refs.[7,8] present parametric fits to the residual stresses obtained from finite element models for a range of butt weld geometries. This is discussed in more detail below.

**Qu.: What are the salient features of the R6 IV.4 Compendium?**

One of the salient features of the R6 IV.4 residual stress distributions is that they violate equilibrium. For example, the transverse stress distributions for the plate butt weld and cylinder butt weld cases have a non-zero net transverse load when integrated across the section. Since these are 2D/axisymmetric cases (assuming no repairs) and there is no net load applied, equilibrium really requires zero load resultant,  $N$ .

It is unfortunate, in my opinion, that the profiles recommended in R6 were not constrained to respect equilibrium. One of the few things we can be fully confident about is equilibrium. In my view this will lead to a substantial over-estimate of stress intensity factors for longitudinal cracks (opened by transverse stresses).

**Qu.: What controls the magnitude of the residual stresses?**

The residual stress profiles give the shape of the distribution. The maximum stress is controlled by a “yield stress” of some sort, but exactly what “yield stress”? R6 defines the following for the purposes of setting residual stress magnitudes,

- For ferritic materials the “yield” stress is defined as the 0.2% proof stress, but for austenitic materials the “yield” stress is defined as the 1% proof stress;
- Best estimate proof stresses should be used to define residual stresses (despite the use of lower bound yield stresses to define  $L_r$ );
- The yield strength of both the parent,  $\sigma_{yp}$ , and the weld,  $\sigma_{yw}$ , materials are required;
- The smaller of the two yield stresses is  $\sigma_y^* = \text{MIN}(\sigma_{yp}, \sigma_{yw})$ ;
- The larger of the two yield stresses is  $\sigma_y^+ = \text{MAX}(\sigma_{yp}, \sigma_{yw})$ .

The R6 profiles use the above quantities to define the absolute magnitudes (see Figures IV.4.1-8). In general the guidance is,

- The peak of the longitudinal welding residual stress is set to  $\sigma_{yw}$ ;
- The peak of the transverse welding residual stress is set to  $\sigma_y^* = \text{MIN}(\sigma_{yp}, \sigma_{yw})$  for unrepaired welds;
- The peak of the transverse welding residual stress is set to  $\sigma_y^+ = \text{MAX}(\sigma_{yp}, \sigma_{yw})$  for repaired welds.

**Qu.: What controls the residual stress profile for cylindrical butt welds?**

Butt welds in tubes or pipes are particularly common and are sufficiently simple that some general remarks can be made regarding their residual stress profiles. The remarks below go beyond the R6 advice and apply for a weld made from the outside (a single-sided V-prep).

Ask yourself what you would expect the axial (i.e., transverse) residual stresses to be in a butt weld. You can argue as follows. The last beads tend to dominate and the axial contraction of these will cause an axial tension near the outer surface, but compression at the inner surface. On this basis one might expect the axial stress to have a bending profile with tension on the outside.

On the other hand you can also argue as follows. The last beads will tend to dominate and the hoop contraction of these will cause the weld to be pulled in to a smaller radius than the surrounding tube. This “tourniquet effect” will result in an axial stress with a bending profile but with *compression* on the outside surface. So this argument comes to the opposite conclusion.

Actually both these situations can be realised depending upon the thickness of the cylinder. More precisely it is the heat input per unit thickness,  $q$ , given by Equ.(1), which controls the nature of the residual stress distribution. The first argument is correct in the limit of thick sections (small  $q$ ), whereas the second (the tourniquet effect) applies in the case of thin sections (large  $q$ ).

For intermediate values of  $q$  the residual stress profile evolves continuously to morph one sign of bending into the other. This has been parameterised in Refs.[7,8].

**Qu.: Is there more realistic advice for welding residual stresses?**

The R6 profiles are intended to be conservative for general application in the context of failure avoidance. However you may wish to refine an assessment if you believe it to be overly conservative. To do so you can go back to the source references, either analyses or measurements – but the burden of proof will lie with the User.

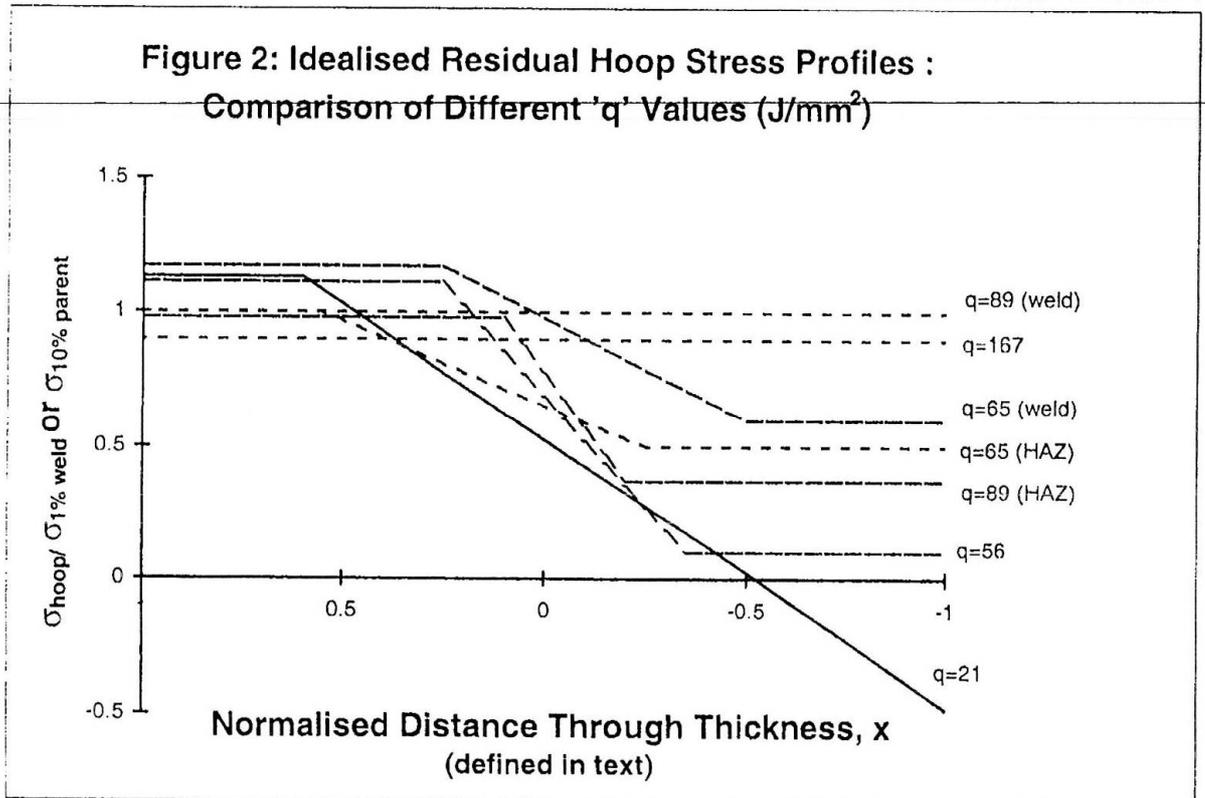
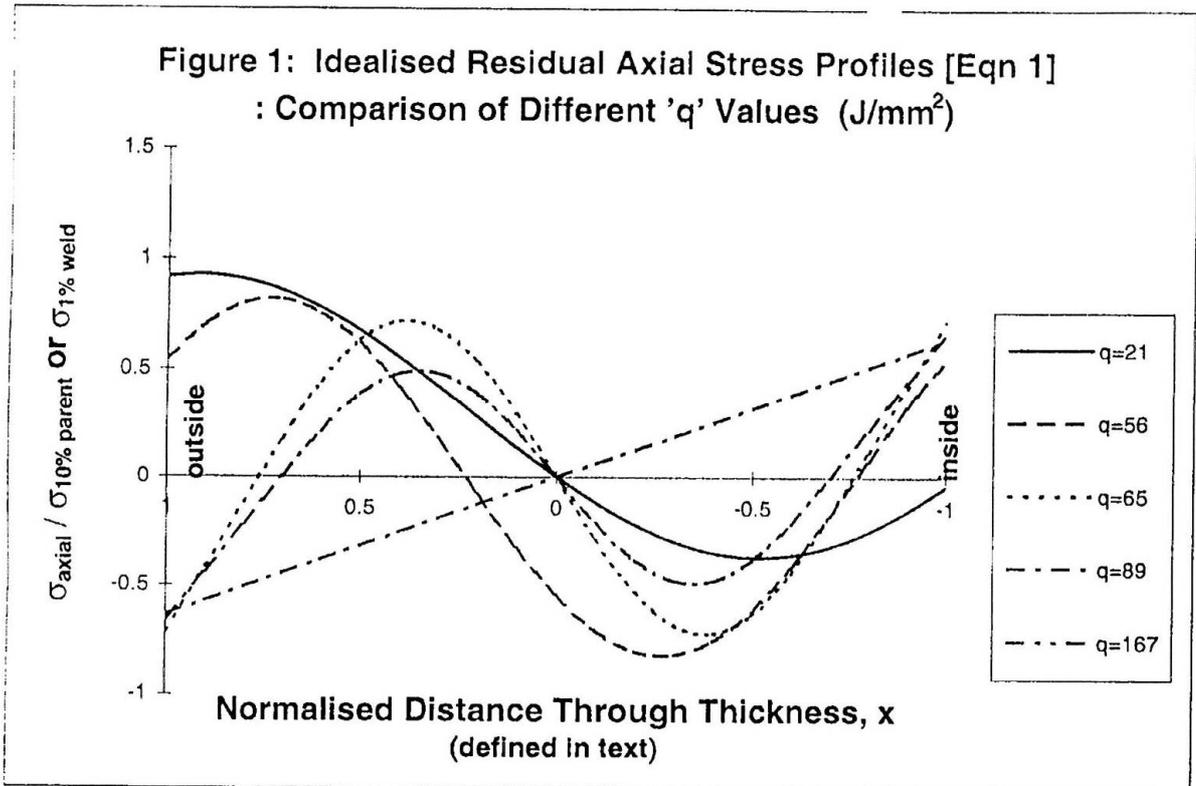
In my view there is a particular pessimism in the R6 recommendations for the axial residual stresses acting on circumferential butt welds, as noted above. In the case of austenitic circumferential butt welds one source of alternative, more detailed, advice for welds made from the outside (single-sided V-prep) is Ref.[7]. This is not the latest information by any means, but it has the advantage of displaying clearly the transition from thin to thick shell behaviour alluded to above. Ref.[7] is available from <http://rickbradford.co.uk/MyPublications.html>.

Ref.[7] gives an algebraic expression for the residual stresses, parameterised in terms of the heat input parameter  $q$ , defined by Equ.(1). The result is shown in graphical form in Figure 8. For small  $q$  ( $\sim 20 \text{ J/mm}^2$ ) the axial residual stress approximates to a bending distribution with tension on the outer surface. For large  $q$  ( $\sim 160 \text{ J/mm}^2$ ) the axial residual stress approximates to a bending distribution with tension on the inner surface (the tourniquet effect). However the formulation of Ref.[7] also provides the intermediate cases in which the distribution passes through a sequence of “sine wave” shapes. These intermediate cases may be relatively benign since they may correspond to a small net bending moment.

There is a refinement to Ref.[7] in Ref.[8], also available from <http://rickbradford.co.uk/MyPublications.html>.

In Ref.[7] the parent 10% proof stress can be identified with the weld 1% proof stress.

**Figure 8: Welding Residual Stresses for Single-Sided Austenitic Butt Welds**  
(from Ref.[7])



**Qu.: How do weld repairs affect the residual stresses?**

Weld repairs can dramatically change the residual stresses in their vicinity. In applications, the presence of weld repairs can be a problem for the assessor because the residual stress can be more onerous both as regards their distribution and their peak magnitude.

**Qu.: What is the R6 advice for repairs?**

R6 advises that both the longitudinal and transverse residual stresses be assumed to be of uniform yield magnitude over the depth of the repair (then tapering off over some 'yield zone'). If the repair is deep, this implies a distribution which is close to membrane, and hence onerous.

The longitudinal yield strength is taken to be that of the weld,  $\sigma_{yw}$ . The peak transverse welding residual stress is set to  $\sigma_y^+ = \text{MAX}(\sigma_{yp}, \sigma_{yw})$  for repaired welds, and hence generally exceeds the peak for the unrepaired weld.

**Qu.: What is the effect of repair length on the residual stress?**

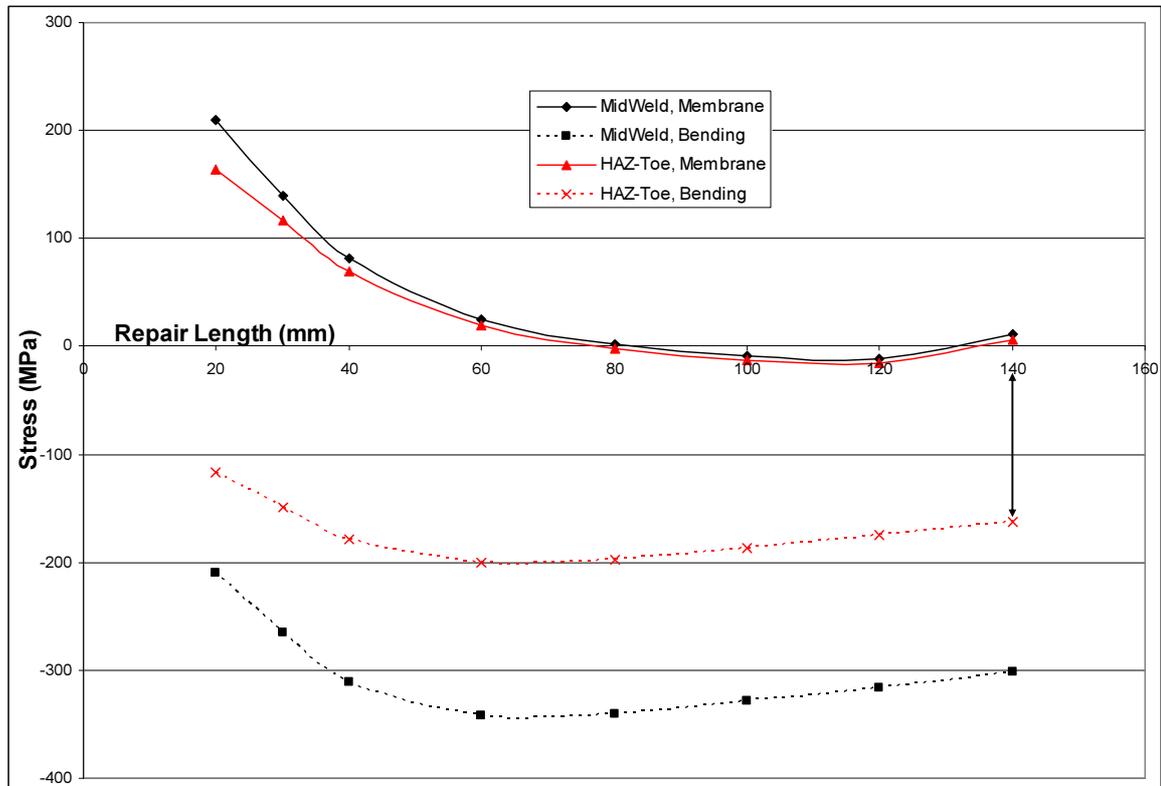
In reality the length of a repair weld is crucial in determining the local residual stresses which it causes. This is obvious because, if the repair is sufficiently long (e.g., the whole circumference!) then there is no distinction between a repair and a complete new weld. Hence, sufficiently long repairs will produce the same residual stress distribution as an unrepaired weld.

The advice in R6, referred to above, actually assumes the limit of a short repair – which is the most onerous case. R6 gives no advice regarding more realistic, i.e., less conservative, distributions for longer repairs.

Ref.[9] has addressed this question for the HPB/HNB tailpipe-pintle welds via an extremely simple FE analysis. The outer diameter of the tailpipe is ~67mm. Figure 9 shows that the membrane component of the repair weld residual stress is virtually zero when the repair length is comparable to, or greater than, the diameter. This establishes, for this geometry at least, how long a repair must be for the residual stress to approximate that for an unrepaired weld. Assuming the R6 advice, i.e., a membrane residual stress of yield magnitude, would be far too conservative for the actual repairs carried out on these welds.

**Figure 9: HPB/HNB Tailpipe-Pintle Welds, Linearised Axial Residual Stresses at Top Dead Centre versus Repair Length**

The points at 140mm (arrowed) are actually for the unrepaired weld



**Qu.: Why should we care about residual stresses?**

The question is not facetious. There are some situations where residual stresses are not significantly deleterious as regards structural integrity. In such cases it makes no sense to expend a great deal of effort in their quantification. Do not do so unless you have a concern regarding their structural implications.

However, there are several ways in which residual stresses may be seriously deleterious...

**Qu.: By what mechanisms are residual stresses deleterious?**

Residual stresses are deleterious to structural integrity because...

- They contribute to fracture;
- They contribute to crack initiation via creep;
- They reduce tolerance to fatigue by raising the mean stress;
- They contribute to creep crack growth.

Reheat cracking needs special mention since it is caused entirely by welding residual stress (as regards the loading).

Qu.: Are residual stresses always deleterious?

No.

Residual stresses can actually be beneficial. Examples of beneficial residual stress effects include,

- Warm pre-stressing;
- Proof testing;
- Peening;
- Weld overlay techniques.

Qu.: Do residual stresses *really* influence fracture?

They certainly *can*. Whether they *do* depends primarily on the ductility of the material. As a rule of thumb, if the material is very ductile then residual stresses may have little effect on the load bearing capacity since this will be determined mostly by plastic collapse. But for low toughness material residual stresses really can degrade resistance to fracture.

Qu.: Can residual stresses *really* cause cracks to initiate?

Yes, most definitely.

Reheat cracking, for example, was a very real problem for AGRs.

Creep-fatigue is also a very real threat which has been realised at HPB/HNB in tailpipe-pintle weld cracking, in the early-life liner roof cracking, and in the reheat muff cracking. The dwell stress which leads to creep damage is rightly regarded (at least in part) as a type of mechanically-induced residual stress.

Qu.: Do residual stresses *really* influence crack growth rates?

Probably. Reheat cracks can be deep (up to 35mm has been known) which must be regarded as growth not just initiation. In practice, though, it is difficult to separate residual stress driven growth from service stress driven growth.

However, it is likely that residual stresses contribute to the creep crack growth in the laboratory in some thick-section HAZ CT specimens. Watch that space.

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- [9] R.A.W.Bradford, “HPB/HNB Superheater Tailpipe-Pintle Welds: The Variation of Repair Weld Residual Stresses with Repair Length”, E/EAN/BBAB/0019/AGR/09, November 2009.