

Thermal Stress Analysis and Assessment Criteria of Water Cooled Stainless Steel and Copper Vacuum Vessels in the new Storage Ring DDBA Cell

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Abstract - The objective of introducing a double-double bend achromat (DDBA) cell into the existing storage ring double bend achromat lattice requires the design of new reduced aperture vessels in both copper alloy and stainless steel. The methods of analysis and the failure criteria used to assess the compliance of new vessels with the operating conditions are presented.

Keywords: Vacuum, Vessels, Thermal, Stress, Assessment

1 Introduction

The concept of converting individual cells of the existing Double Bend Achromat (DBA) lattice into a modified double-double bend achromat (DDBA) with a new straight section for Insertion Devices (IDs) in the middle, grew out of earlier studies of low emittance multi-bend achromat lattices. It was motivated by the need for additional ID straight sections. This need has arisen because all of the 22 ID straight sections in the Diamond Light Source (DLS) storage ring are either occupied or have been allocated to future Beamlines. Such a modification effectively replaces a single DBA cell with two new DBA cells, as shown in Fig. 1. It can be seen that the existing tangent point for the bending magnet Beamline lies close to the projected light from the new ID therefore the same ratchet wall ports and areas on the experimental hall can be used.

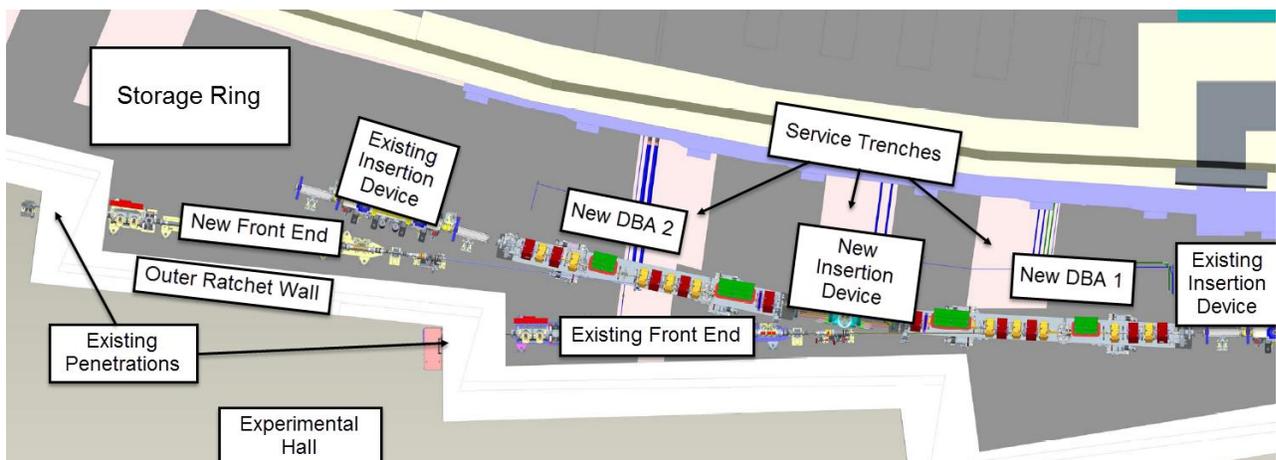


Figure 1: Layout of new DDBA cell

2 Vessel profile

One of the drivers for the DDBA project is to trial solutions for future increased storage ring brightness; this will require the multipole magnetic flux strengths to be increased. In order to achieve the required quadrupole gradient of 70 T/m the magnet pole tip radius must be reduced from the existing 39mm to 15 mm. Therefore the apertures of the storage ring vacuum vessels have to be correspondingly reduced. The conflicting requirement is to maximise the vessel aperture to allow the maximum amount of Undulator ID light to pass through unimpeded. This has driven the need to minimize the vessel wall thickness to 1mm.

To allow flexibility of location, both horizontally canted, and straight ahead, ID light trajectories had to be catered for. This led to the choice of an elliptical vacuum vessel with nominal internal dimensions of 27mm (H) x 18.4mm (V) with 1mm wall thickness.

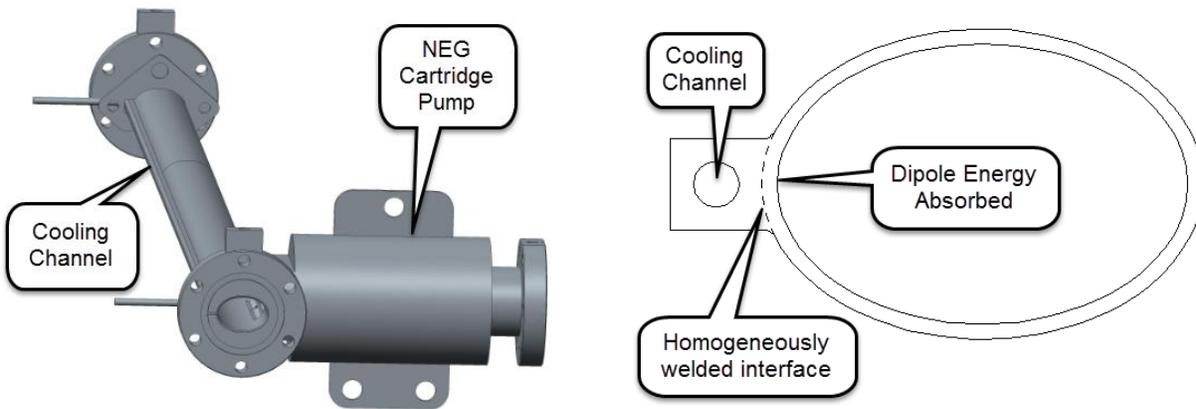


Figure 2: Stainless steel Storage Ring Vessel Assembly and Profile

3 Previous assessment criteria

The present design of the storage ring only has OFE copper material (C101) intercepting significant amounts of energy. There are a few aluminium alloy, and stainless steel components absorbing small quantities of energy, but the temperature and stress values are very low. In (Huang, 2005) the maximum strain limits failure criteria are described for OFE copper. The criterion for assessing compliance of the vessels is strain based, i.e. using results from non-linear finite element analysis (NLFEA). Limits are set at 0.5% strain peak surface value and 0.2% global limit for copper and copper alloys. From a heat load capacity and thermal conductivity aspect it would have been ideal for all the DDBA vessels to have been made of copper. However, the corrector coils within the sextupole magnets have to transmit small variations in field at a sufficiently high frequency to stabilise the electron orbit to at least 100Hz. The screening effect of a highly conducting wall material would limit and delay the penetration of the magnetic flux variations to the vessel interior.

No formal assessment criteria previously existed within DLS for stainless steel vacuum vessels apart from requiring that stress intensity values for mechanical loads should not exceed the 0.2% yield stress of the material grade.

4 Proposed criteria for stainless steel vessels subjected to high thermal loads

When considering the acceptable thermal stress levels in the wall of a stainless steel vessel two approaches can be taken:

1. To limit the peak stress intensity (Von Mises), derived from the finite element analyses, to the yield stress at the coincident surface temperature of the component. This approach is very conservative.
2. That typically adopted in (ASME, 2013) and (BS EN 13445, 2007). There is an option in these codes for the ‘design by analysis’ method which divides the stresses into different categories: membrane, bending and peak. Membrane stresses are constant through thickness stresses. Bending stress is the linearly varying through thickness stress. Peak stresses are localised thermal surface stresses. The limits for the categories differ.

The highest stress values are those resulting from thermal stresses where the vessels have to directly absorb synchrotron light.

A suitable range of design stresses across a range of temperatures is required. To produce this range the values for yield stress contained in (BS EN 10272, 2007) are used as summarised in Table 1.

Table 1: Minimum values of proof stress for austenitic stainless steels at elevated temperatures in the solution annealed condition

Name	Number	Maximum 0.2% Yield Stress (MPa) at temperature			Maximum 1.0% Yield Stress (MPa)		
		100°C	200°C	300°C	100°C	200°C	300°C
304L	1.4306	145	118	100	180	145	127
316L	1.4435	165	137	119	200	165	145

These values are used as the basis for generating Figure 3 by applying the reduction factor from section 6.5.1 of (BS EN 13445, 2007) shown in equation 1:

$$Design\ Stress\ (f) = \frac{\sigma_{1.0\%yield}}{1.2} \quad (1)$$

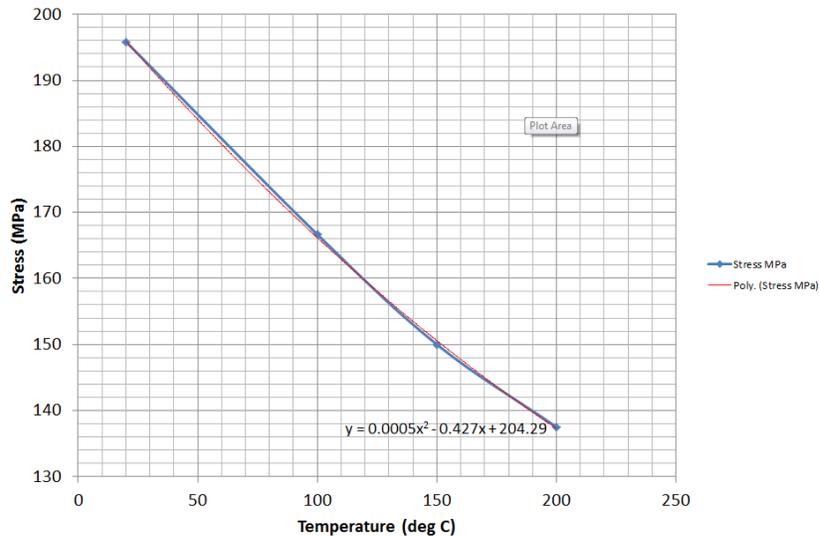


Figure 3: Variation of design stress (f) with temperature for 316L stainless steel.

When the temperature through the vessel wall varies greatly, then a nominal temperature is calculated using equation 2 which is from (BS EN 13445, 2007). This design stress ‘f’ corresponding to this nominal value is used.

$$T = 0.75T_{max} + 0.25T_{min} \quad (2)$$

In the above equation T_{max} and T_{min} represent the extreme values of temperature through the vessel wall.

4.1 Assessment in accordance with (BS EN 13445, 2007)

(BS EN 13445, 2007) contains three approaches to the design of pressure vessels made from steel, stainless steel, and aluminium alloy. The three methods are:

1. Design by formula
2. Design by analysis: direct route (Annex B)
3. Design by analysis: stress categorisation (Annex C).

Method 3 is the method under consideration here. The unconventional shapes of vessel exclude method 1. Method 2 seems superficially to be more applicable as it uses plastic failure limits and would require NLFEA to produce results which is the same method used previously at DLS for OFE copper absorbers. However, the 5% allowable strain defined in section B.8.2.1 of (BS EN 13445, 2007) is considered to be an order of magnitude too high for this application where maintaining geometric tolerances is critical to the function of the vessels.

It should be noted that vessel temperatures during operation will be limited to 100°C and therefore none of the failure criteria relating to creep have been considered.

Method 3 has been criticised for being over simplistic and too open to (mis)interpretation (Staat, Heitzer, Lang, & Wirtz, 2005). However the use of extrapolated elastic stresses above the material yield stress has been used in synchrotron component design before (Zhang, Biasci, & Plan, 2002) and has long been used in the nuclear industry (ASME, 2013) to assess code compliance of pressure vessels.

In using stress categorisation and linearisation the assessment principle is to prove that three conditions are satisfied.

1. The membrane stress levels will not cause failure by gross plastic deformation (yielding) of the structure.
2. The combined membrane and bending stress levels indicate that the structure will ‘shakedown’ to purely elastic behaviour after a few load cycles and progressive plastic ratcheting of the structure does not occur.
3. The peak surface stresses are within the fatigue limits of the material.

The factors applied to ‘f’, the design stress, for various classifications are shown in Table 2

Table 2: Stress Classifications and Limits

Stress Classification (BS EN 13445, 2007), Annex C	Design Stress factor
General Primary Membrane P_m	f
Local Primary Membrane P_L (near discontinuities)	1.5f
Membrane + Bending (general or local)	1.5f
Secondary Membrane + Bending (thermal)	3f

4.2 Example Calculation

A typical ray trace of dipole light impinging on a stainless steel vacuum vessel is shown in Figure 4. A series of internal tapered protrusions serve to cast shadows across uncooled portions of the vessel assembly, such as flanges, beam position monitor buttons and bellows.

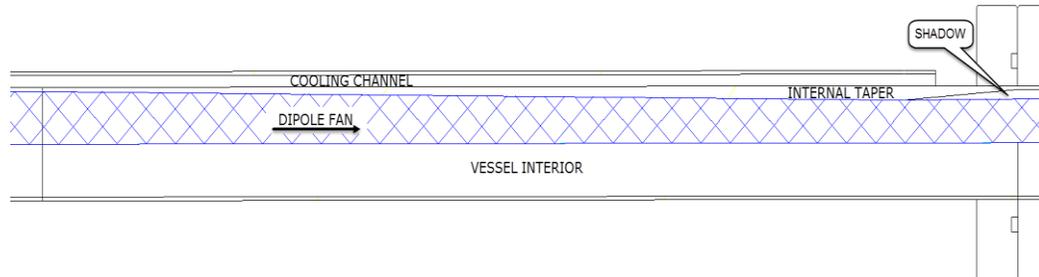


Figure 4: Detail in horizontal mid plane of vacuum vessel and internal features.

The absorption of dipole light on the wall of the cooled stainless steel tube produces the stress contour pattern illustrated in Figure 5 with the maximum values on the taper leading edge. σ_z (axial) is the most significant stress component for a thermal stress wall loading of this type. $T_{\max}=166^\circ\text{C}$ and $T_{\min}=22^\circ\text{C}$. It should be noted that the dipole light power densities are calculated for a beam current of 550mA (500mA +10%) which is the maximum design storage ring current. At present the maximum operating current is 350mA.

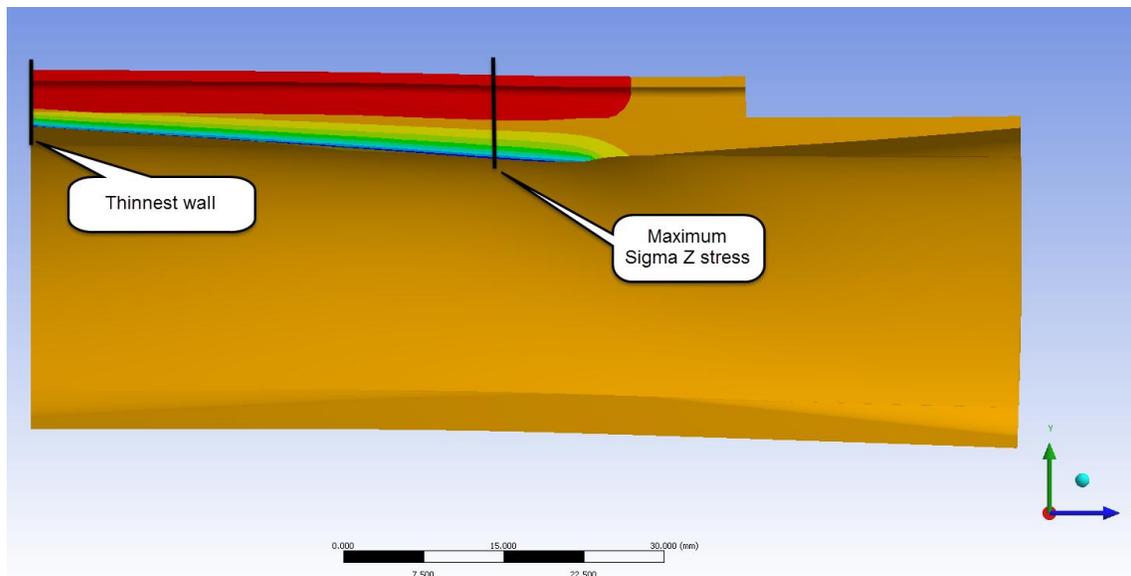


Figure 5: Horizontal cooled vessel section showing stress contours resulting from dipole heating

Stress linearisation was performed along two lines as shown in Figure 5. The resulting stress vs. location graph through the thicker section is shown in Figure 6. It should be noted that a sufficient number of through wall thickness elements are required in order to calculate the stress distributions with sufficient accuracy.

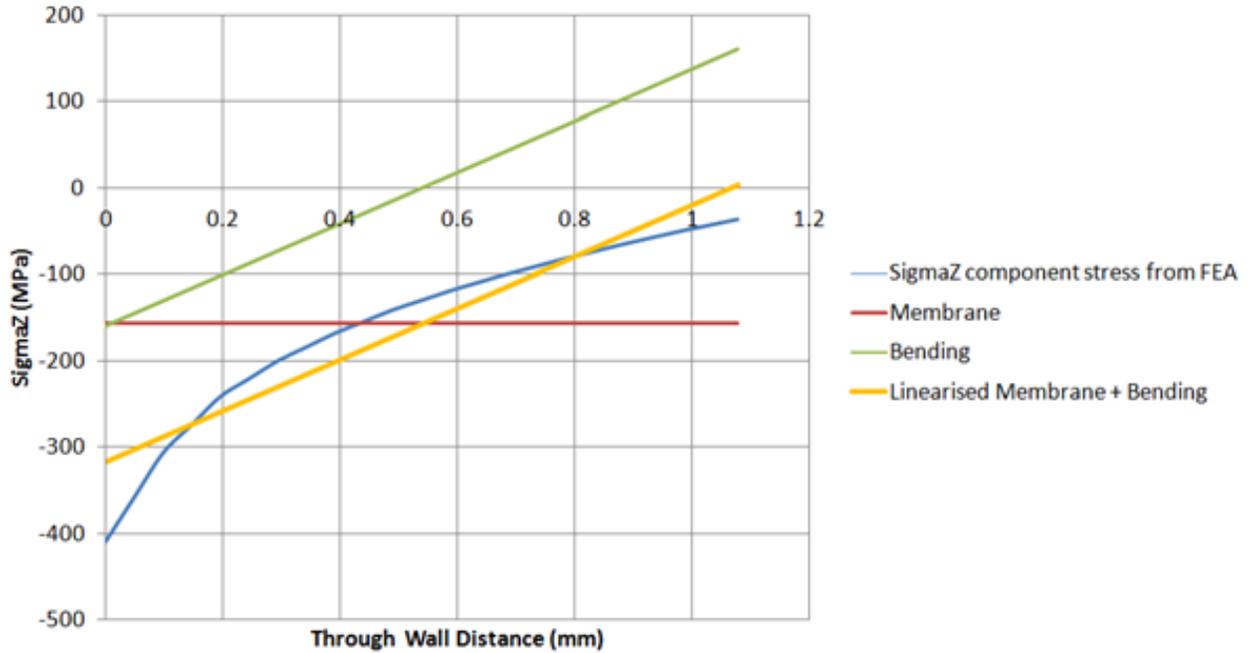


Figure 6: Illustration of longitudinal stress and categorised stresses

According to the assessment criteria laid out in (BS EN 13445, 2007), and using the value obtained from Figure 3 at $T = 130^{\circ}\text{C}$, where T is calculated using equation (2). The design stress $f = 157\text{MPa}$, the limit for secondary membrane stress $1.5f$ is 236MPa , and the limit for secondary bending + membrane stresses is $3f = 471\text{MPa}$.

Table 3: Comparison of stresses with code limits (MPa)

Location	Secondary Membrane	Code Limit ($1.5f$)	Secondary Membrane + Bending	Code Limit ($3f$)
Thick	-85	236	-248	471
Thin	-160	236	-317	471

These results indicate that the design stresses are within the allowable limits for secondary membrane and secondary membrane plus bending.

When considering the allowable fatigue life for both copper and austenitic stainless steel there are several conflicting considerations. These require care when choosing a data source which is valid for synchrotron applications.

1. Both materials can become annealed through joining processes.
2. Both significantly work harden through the first few hundred cycles.
3. The high surface thermal stresses on the interior surface are compressive.
4. The interior surface faces a vacuum.
5. The type of loading experienced during operation sustains the high stress for long periods of time, which is important if sustained temperatures are in the creep range.

(Wang, Nian, Ryder, & Kuzay, 1994) provides information for copper and (Khairul, Edi, Mohd, Nur, & Aidy, 2013) provides some relevant information for 316L stainless steel. There is also a

calculation procedure in clause 18 of (BS EN 10272, 2007) including weld strength reduction factors.

The information from the above sources, combined with the variation of elastic modulus with temperature, produced Figure 7. This determines the limits of 316L for a fatigue life of 10^4 cycles. The value of 10^4 is the expected number of cycles for operation of the storage ring over a 30 year lifetime to the nearest factor of ten.

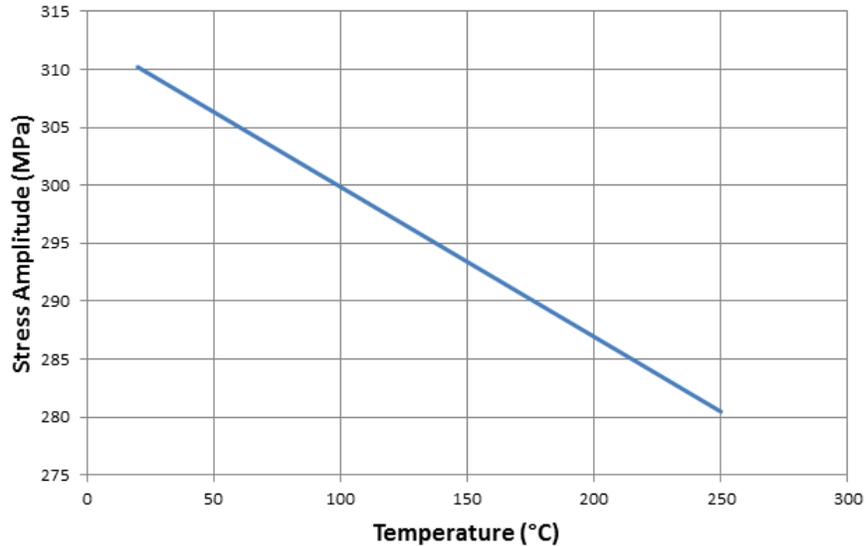


Figure 7: Maximum Stress Range vs. Temperature for 316L Stainless Steel for 10^4 cycles

In the example given the maximum allowable stress amplitude for 10^4 cycles at 130°C is 296MPa using Figure 7. In Figure 6 the peak stress is 409MPa . The design of this component was therefore modified by increasing the length of the internally protruding taper until the peak stresses reduced to values well below the estimated limit.

5 Conclusions

The strain based assessment criteria for copper vessels previously used at DLS (Huang, 2005) are still valid and can be used for these vessels in assessing the thermal and pressure loads.

The method used in Annex C of (BS EN 13445, 2007) lends itself well to stainless steel vessels. The method and assessment criteria used in Annex B could be used, but the allowable strains are considered to be excessive for these storage ring vacuum vessels.

(BS EN 10272, 2007) does not include copper vessels within its scope, but the methodology used in annex C would be applicable to copper vessels. Conversely the strain based assessment of (Huang, 2005) may well be applicable to stainless steel vessels and these two approaches need further comparison.

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