Stress failure analysis of neck breakage of cathode ray tube

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Abstract

A breakage occurred in the neck area of a cathode ray tube (CRT) during the process of yoke clamping. While previous studies have shown that the BEM can successfully predict the stress and deformation due to the internal vacuum and shrink-band loading, these loading conditions require a three dimensional analysis of the complex geometry. The neck breakage, however, can be analyzed using a simplified model. The neck area of the CRT is essentially cylindrical and we are able to make huge savings in both data preparation time and computational resources by using an axisymmetric model. This has been shown to be acceptable since the vacuum and shrink band conditions have little effect on the local peak stresses in the neck due to yoke clamping.

The neck failure has been analyzed using the boundary element method, for reasons of both the ease and speed of use and also because of the method’s ability to predict sharp stress concentrations with accuracy. The crack direction and location predicted by the BEM were well matched with those observed in the real case.

BEM technology can quickly and easily point to required design changes. An analysis can be done in response to observed cracking and can quickly identify the primary cause of the problem. Further analysis has been done to determine the changes in geometry and load profile which can be made in the design to prevent failure of the type observed. This provides a good example of the power of the boundary element method in an industrial setting.
1 Introduction

A cathode ray tube (CRT) is constructed as a continuous envelope of glass. The main features of a CRT are shown in the boundary element model (figure 1). With the potentially lethal consequences of a rapid implosive failure of CRT in a home television, it is essential that a detailed stress analysis of the component be performed at the design stage. Such an analysis will need to consider two major contributors to the stress profiles experienced by the product under normal operating conditions.

Firstly, because the interior of a CRT is a vacuum, the atmospheric pressure acting on the outside of the tube will tend to compress it and will include a state of stress.

Secondly, a pretensioned metallic ‘shrink-band’ is applied to the outside of the tube inducing a state of stress which acts to prevent a potentially dangerous implosive failure under impact. Techniques for the static stress analysis of engineering components have developed rapidly over the last few years, and today the two major alternatives are finite element method (FEM) and the boundary element method (BEM). Both techniques have been used for a 27-inch cathode ray tube stress analysis. The boundary element results have achieved a high degree of convergence even for a single element in the thickness direction. Although it is found to be unnecessary to use a more refined mesh, it is also found that the remeshing presents no difficulty and can be done in extremely short space of time. The finite elements results appear to require a minimum of two elements in the thickness direction before the results approach convergence. The difficulty of modifying the model to consider this refinement is an obstacle to a rapid convergence analysis. The simplicity of producing a refined analysis model in the BEM allows a more detailed convergency study and therefore promotes confidence in the stress. In consequence, the boundary element model was easier and quicker to build, more accurate and also has tremendous shorter modeling time than the finite element model.

2 The Problem

The beams from the gun at the neck end of the CRT land on the face panel. In order to land in the right place of the face panel, an electrical yoke is installed which makes the beam deflect. The electrical yoke is installed on the glass yoke by clamping at the neck. This is called yoke clamping.

A crack was observed to have occurred circumferentially .205" below the yoke clamp area for a 32 inch CRT newly developed (shown in Figure 2).

In order to analyze the mechanism which caused the crack, the BEM was
chosen in this case because of its ease of use and its well known suitability for problems involving cracks and stress concentrations. This crack problem can be analyzed with both 3D and axisymmetric models. The 3D boundary element model's stress results matched well with those of axisymmetric boundary element models. Therefore, the analysis was performed using the axisymmetric assumption, and even though the CRT is plainly not an axisymmetric structure the geometry, loading and fracture mechanics indicate that an axisymmetric analysis is justified.

A preliminary analysis showed that the cracking is related not to the global vacuum and shrink band loading but that it is a purely local effect of the neck region caused by the clamp. Therefore, the model was confined to the neck area. This enabled:

- great simplification of the model by using the axisymmetric assumption
- speed up of the analysis so more test cases could be run
- an ability to use far more elements in the area of interest to capture more accurately the complex stress distribution

Boundary element models were run using the BEASY program. Material constants input of the glass for boundary element model follow:

- Young's modulus = 10.07E+6 psi
- Poisson's ratio = 0.23
- The thermal conductivity = 1.4E-5 BTU/in-s F
- The thermal expansion coefficient = 5.5 E-6 in/in F
- The reference temperature = 70 F

3 The results

The first analysis considered the present design with the operating clamp torque 1N-m (see the yoke clamp location on Figure 2). The applied yoke total stress consists of the yoke clamp pressure and the yoke residual stress.

First, the yoke clamp pressure has been evaluated by the following theoretical method:

The screw for the yoke clamp used is an ISO thread which has major diameter of 4 mm (0.1575") and pitch of 7mm (0.2756"). After some lengthy algebraic process using the coefficient of thread friction with value of 0.12, the relationship equation between the screw force W and clamp torque T reduces to the following simplified form:

\[ W = T / 0.0148 \] (1)
where $W$ is the screw force and $T$ is the clamp torque. By substituting our present torque of 8.851 lb·in (1 N·m) in Eq (1), the screw force, $W$, will be 598 lb. Equilibrating the screw force to the tension of yoke clamp, the yoke clamp pressure will be expressed by the formula:

$$p = \frac{W}{r_0 l}$$

(2)

Where:

- $p =$ yoke clamp pressure
- $r_0 =$ outer radius of the neck
- $l =$ yoke clamp width
- $W =$ tension of yoke clamp

By substituting $r_0 = .576''$, $l = .354''$ and $W = 598$ lb to eq (2), the yoke clamp pressure, $p$ will be 2932 psi.

The normal force to the neck will be evaluated by the following equation,

$$F_n = 2 \pi r_0 l p$$

(3)

Where $F_n =$ The normal force to the neck

By substituting known values $r_0$, $l$ and $p$ above to equation (3), the normal force to the neck, $F_n$ will be 3756 lb.

Secondly, we calculate the yoke residual stress, since the present neck failure comes from not only the yoke clamp stress, but also the residual stress. This is because the actual stress failure occurs at point $A$ in figure 2. If the stress failure were to come from the yoke clamp stress only, the failure location would be more likely at point $B$ (figure 2). Table 1 shows how the clamp stress is higher at point $B$ than at point $A$.

After heating up to $900^\circ$C the neck end for neck end sealing for 10 seconds, the neck will be cooled slowly by air. Residual stresses of considerable magnitude can result from contraction of the sealing upon cooling. The outer glass surface is in tension while inner glass surface is in compression. Through the polariscope, a significant residual stress has been detected on the region around point $A$, below the yoke clamp, while this stress is negligible around the point $B$ region, above the yoke clamp.

Considering that the residual stress comes from the temperature difference between the outer and inner glass with the polariscope’s result, the boundary element axisymmetric analysis can use a temperature difference of $20^\circ F$ and $0^\circ F$ at points $A$ and $B$ respectively.
The stress analysis results for different locations for the various axisymmetric boundary element models are shown in Table 1.

The glass breaking strength depends upon the flaw size of existing defects. Therefore, generally failure comes from the outer glass due to the existing defects. The inner neck breaking strength has been evaluated as 8500 psi by a Friedel polarimeter. As shown in Table 1, the yoke total stress of 7687.8 psi at point C, the inner center of neck clamp width, is below the glass breaking strength of 8500 psi. (all stresses in this paper is the maximum tensile principal stress). Therefore, the stress failure does not occur on the inner neck. All maximum principal stresses of the three types above occur in the (axial) z-direction. The crack is perpendicular to the principal stress direction, and thus it is shown that the boundary element results predict the crack growth direction accurately.

The stress value 3034 psi found at point B was then adopted as the design limit for the outer neck stress for the new, modified, design options 1 and 2. No stress failure occurred at B, so it can be concluded that 3034 psi represents a permissible stress value which can be applied to both the inner and outer glass surfaces.

Three different types of stress contours are shown in Figures 3, 4 and 5. Three different types of outer stress graphs are shown on Figures 6, 7 and 8. On these graphs, the left end of the horizontal axis represents the neck end and the right end is the interface to the yoke. Therefore the left peak stress location is the point A and the right peak stress is at point B on Figures 6 and 8.

The present design recommends a clamp torque of 8181 N-m (7.24 lbf-in). In order to reduce the stress, a yoke clamp pressure p was chosen to be 2400 psi. The yoke total stress at point A was calculated as 2905 psi (as shown in Table 1), which is below the 3034 psi design limit on the outer neck stress.

The yoke total stress at point C was calculated as 6246.6 psi, which is below the 8500 psi inner neck breaking strength. Therefore, no stress failure will occur for option 1.

However, to reduce the applied pressure on the neck, the normal force to the neck had to be reduced to 3074 lb from the 3756 lb required by the original design. So the new design does not guarantee to hold the yoke firmly.

Two different types of stress contours and graphs for design option 1 are shown in Figures 9 to 12.
This led to the proposal of a design option 2, in which the width of the clamp is increased. The new design recommended a yoke clamp torque of 1N-m. The width of yoke clamp increased from 9mm (0.354") to 12.6mm (0.498"). See the yoke clamp location on Figure 13. Since the yoke clamp pressure is inversely proportional to the width of the yoke clamp for the same torque, option 2 increases the width of yoke clamp from 0.354" to 0.498" in order to reduce the stress.

In order to hold the yoke firmly enough, we use the present normal force to the neck $F_n$ as 3756 lb substituting known values $r_p$, $l$ and $F_n$ into equation (3), the yoke clamp pressure, $p$ will be 2084 psi. Table 2 summarizes this design option along with the present design and design option 1.

Therefore the new design option 2 will hold the yoke securely since it offers the same normal force as the present design.

The yoke total stress at point A with a value of 3032 psi is below the 3034 psi design limit for outer neck stress. The yoke total stress at point C with value of 4595.4 psi is below the 8500 psi inner neck breaking strength. Therefore it is concluded that no stress failure will occur for design option 2.

Two different type of stress contours and graphs for option 2 are shown in Figures 14 to 17.

From Table 2, design option 2 appears to be the better design. However, experimental tests have shown that the design option 1 gives no stress failure while still holding the yoke firmly. Consequently, the option 1 has been chosen for the operation of the yoke clamp since it uses the same clamp assembly and does not induce a stress failure.

4 Conclusions

With a minimum of effort the BEM was able to provide rapid engineering solutions of high quality to point the way to a required design change. Speed and accuracy are both critical in an industrial situation where design changes become very costly as the product development processes to the later stages.

References

8. BEASY, Computational Mechanics, Southampton, UK and Billerica, MA.

<table>
<thead>
<tr>
<th>DESIGN</th>
<th>VARIOUS STRESS</th>
<th>POINT A</th>
<th>POINT B</th>
<th>POINT C</th>
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<tbody>
<tr>
<td>Present Design</td>
<td>Yoke clamp stress (psi)</td>
<td>2993</td>
<td>3040</td>
<td>7942.8</td>
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<td></td>
<td>Yoke residual stress (psi)</td>
<td>609.4</td>
<td>13.74</td>
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<td>Yoke total stress (psi)</td>
<td>3448</td>
<td>3034</td>
<td>7687.8</td>
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<tr>
<td>Option 1</td>
<td>Yoke clamp stress (psi)</td>
<td>2450</td>
<td>2489</td>
<td>6501.6</td>
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<td></td>
<td>Yoke total stress (psi)</td>
<td>2905</td>
<td>2482</td>
<td>6246.6</td>
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<tr>
<td>Option 2</td>
<td>Yoke clamp stress (psi)</td>
<td>2602</td>
<td>2465</td>
<td>4767.4</td>
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<td></td>
<td>Yoke total stress (psi)</td>
<td>3032</td>
<td>2454</td>
<td>4595.4</td>
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**TABLE 1. Comparisons of stresses for the three designs at point locations A, B and C.**
<table>
<thead>
<tr>
<th>ITEMS</th>
<th>Present Design</th>
<th>Design Option 1</th>
<th>Design Option 2</th>
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<tbody>
<tr>
<td>Yoke Clamp Torque</td>
<td>1 N-m (8.851lbf-in)</td>
<td>0.8181 N-m (7.24lbf-in)</td>
<td>1 N-m (8.851lbf-in)</td>
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<td>Width of Yoke Clamp</td>
<td>9 mm (0.354&quot;)</td>
<td>9 mm (0.354&quot;)</td>
<td>12.6 mm (0.498&quot;)</td>
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<tr>
<td>Yoke Total Stress at Point A</td>
<td>3448 psi</td>
<td>2905 psi</td>
<td>3032 psi</td>
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<td>Normal force to the neck</td>
<td>3756 lb</td>
<td>3074 lb</td>
<td>3756 lb</td>
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<td>Yoke Clamp Pressure</td>
<td>2932 psi</td>
<td>2400 psi</td>
<td>2084 psi</td>
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<tr>
<td>Condition</td>
<td>Present Operating Condition</td>
<td>Reduce Yoke Clamp Torque to 0.818 N-m on present condition</td>
<td>New design with increased yoke clamp width of 12.6 mm</td>
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<tr>
<td>Result</td>
<td>Stress failure occurred at point A</td>
<td>No stress failure expected but not guaranteed to hold yoke firmly</td>
<td>No stress failure expected at point A. Will hold yoke firmly.</td>
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**TABLE 2. Summary of present design, option 1 and option 2**
FIGURE 1. Main Features of 32V Tube with 3D Boundary Element Model

FIGURE 2. Yoke Clamp Location for Present Design and Option 1 with Boundary Element Model with 32V Neck
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FIGURE 3. Yoke Clamp Stress Contours (present design)

FIGURE 4. Yoke Residual Stress (present design)
FIGURE 5. Yoke Total Stress Contours (present design)

FIGURE 6. Yoke Clamp Stress Graph (present design)
FIGURE 7. Yoke Residual Stress Graph (present design)

FIGURE 8. Yoke Total Stress Graph (present design)
FIGURE 9. Yoke Clamp Stress Contours (Option 1)

FIGURE 10. Yoke Total Stress Contours
FIGURE 11. Yoke Clamp Stress Graph (Option 1)

FIGURE 12. Yoke Total Stress Graph (Option 1)
FIGURE 13. Yoke Clamp Location for Option 2 with Boundary Element Model of 32V Neck

FIGURE 14. Yoke Clamp Stress Contours (Option 2)
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FIGURE 15. Yoke Total Stress Contours (Option 2)

FIGURE 16. Yoke Clamp Stress Graph (Option 2)
FIGURE 17. Yoke Total Stress Graph (Option 2)