Abstract

The procedure for absolute calibration of acoustic emission (AE) sensors is provided in ASTM E1106. In that test method, a glass capillary is fractured against a large steel cylinder in order to generate an acoustic waveform used for calibration of the sensor. In this work the finite element method was used to investigate the viability of the point source load assumption made in the analytic calculation of the resultant waveform for the glass capillary fracture procedure. It was found that the stress field on the steel cylinder due to elastic deformation was not singular, but approximately ellipsoidal in nature. Part II of this work will highlight the impact that the shape of the stress field had on the resultant waveform, leading to the necessity for revision of ASTM E1106.

Keywords: Absolute AE sensor calibration, ASTM E1106, finite element modeling, glass capillary fracture, NIST steel block

1. Introduction

Recent developments in acoustic emission (AE) techniques based upon the spectral content of AE waveforms have proven to be effective in damage detection and identification [1-4]. These advances have primarily been enabled through ever increasing computational power. However, these analysis techniques necessitate properly, and absolutely calibrated transducers, as an improperly calibrated transducer could potentially distort the waveform of the AE signal and lead to an improper analysis. The ASTM International test method that governs the absolute calibration of AE transducers is E1106. In this test method, a loading screw applies a force to a glass rod, which is in contact and oriented in a perpendicular fashion to a glass capillary. The capillary sits on a thin glass slide, which rests on top of a large steel cylinder as shown in Fig. 1 [5]. The fracture of the glass capillary is intended to simulate a point load on an infinite half-space. Such a set-up for absolutely calibrating AE transducers exists at the National Institute of Standards and Technology in Boulder, CO.

In ASTM E1106-07 two methods are mentioned for determining the resultant waveform that the transducer under test should register upon fracture of the glass capillary. The first method discussed relies on a high fidelity capacitive transducer that measures the absolute out-of-plane surface displacement on the surface of the steel cylinder. The second method mentioned is an analytic calculation of the surface displacement of an infinite half-space subjected to a step load point force; the analytic solution was first derived by Pekeris [6]. For either method mentioned, the waveform recorded by the transducer under test is compared to the waveform determined by either method.

In Fig. 1, it is clear that initially a line contact condition exists between the glass capillary and the glass slide. Considering Hertz’s work on bodies in contact [7], as the capillary deforms into the glass slide, the resulting stress field within the glass slide would be expected to be
elliptical in nature. Then, as the load from the glass slide is transferred to the steel cylinder, the expected loading most certainly would not be a point source, but more likely ellipsoidal in nature. Thus, assuming a point source as the loading source to analytically predict the surface displacement of an infinite half space appears debatable.

To analytically solve the multiple bodies in contact problem that the set-up requires is certainly not trivial and would be quite laborious in nature. Hence, the approach adopted in this work was to use a numeric technique. Specifically, the finite element (FE) method was used to approximate the resultant surface stress field on the surface of the steel cylinder as a result of performing the procedure described in ASTM E1106. Additionally, the state of stress within the glass capillary was evaluated, to provide insight on the fracture initiation site within the glass capillary.

![Fig. 1 Schematic of the set-up used for the absolute sensor calibration described in ASTM E1106.](image)

2. Methods

Table 1 summarizes pertinent dimensions of all parts shown in Fig. 1. The large discrepancy in dimensions between the steel cylinder, and the dimensions of the other components presents an issue in developing a numeric model. Attempting to model the entire set-up would result in an extremely large numerical model, requiring an excessive amount of time to solve. For computational efficiency only the near field portion of the steel cylinder was modeled, as stresses are expected to decay to zero in a natural logarithmic fashion in the far field [8]. Thus, for the numeric model the radius of the steel cylinder was modeled as 1.25 mm with a thickness of 1 mm. For further computational efficiency a quarter-volume model was developed as shown in
Fig. 2, with appropriate symmetric boundary conditions applied to represent the full 3D volume. The entire mesh consisted of linearly interpolated three dimensional hexahedral volume elements; both full and reduced integration formulations were considered\(^1\). A mesh biasing scheme was used for the steel cylinder to reduce the characteristic element size and capture the stress field directly under the applied load, but allowed the element size to increase further away from the applied load where little variation in the stress field was present.

Table 1 Summary of pertinent dimensions from the NIST Boulder AE absolute sensor calibration set-up.

<table>
<thead>
<tr>
<th>Component</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass Rod</td>
<td>Diameter = 2 mm</td>
</tr>
<tr>
<td></td>
<td>L = 70 mm</td>
</tr>
<tr>
<td>Glass Capillary</td>
<td>Outer Diameter = 0.2 mm</td>
</tr>
<tr>
<td></td>
<td>Inner Diameter = 0.1 mm</td>
</tr>
<tr>
<td></td>
<td>L = 50.0 mm</td>
</tr>
<tr>
<td>Glass Slide</td>
<td>W = 1.5 mm</td>
</tr>
<tr>
<td></td>
<td>L = 1.5 mm</td>
</tr>
<tr>
<td></td>
<td>t = 0.08 mm</td>
</tr>
<tr>
<td>Steel Cylinder</td>
<td>Diameter = 900 mm</td>
</tr>
<tr>
<td>Loading Screw</td>
<td>Minimum Diameter = 1.6 mm</td>
</tr>
</tbody>
</table>

All materials were assigned isotropic, linear elastic material properties. The glass rod, capillary, and slide were modeled as having a Young’s modulus of 72 GPa and a Poisson’s ratio of 0.22, while the steel cylinder had a Young’s modulus of 200 GPa and a Poisson’s ratio of 0.29. Plane-strain boundary conditions were assigned to both the glass rod and capillary. All nodes on the bottom of the steel cylinder were constrained in the out-of-plane direction to maintain static equilibrium from the force to be applied. The degrees of freedom for the in-plane directions of the center node on the bottom of the steel cylinder were additionally constrained to prevent rigid body modes.

The numeric model was run in force control, with loads of 10 N and 20 N being applied; these loads were selected as they are the common bounds of experimentally reported fracture loads [5]. A concentrated load was applied to a single node, which was rigidly beamed to nodes on the top edge of the glass rod (rigid beam elements are identified in red in Fig. 2). The length of the beamed nodes on the glass rod matched the outer diameter of the loading screw (OD = 1.6 mm) in Fig. 1. By applying the load in this manner “hot-spotting,” of the mesh was avoided. A surface-to-surface, finite sliding contact definition was used to optimize the stress field calculation due to bodies in contact [9]. A tie constraint was assigned to all surfaces that were in contact with one another, before loads were applied. The tie constraint fixes the associated degrees of freedom to the surface that they were initially assigned in contact with, expediting convergence of the non-linear contact solution.

\(^1\) Full and reduced integration refers to the number of evaluation points used for the Gaussian quadrature integration. In order to fully integrate a linearly interpolated hexahedral volume element a 2x2x2 integration rule is necessary, whereas for a reduced integration formulation only a 1x1x1 integration rule is needed. A reduced integration element is less computationally expensive than a fully integrated element.
Fig. 2 Quarter volume finite element representation of the absolute calibration set-up of [5]. With appropriate boundary conditions applied, the solution is identical to what would be obtained if the entire volume were modeled.

Fig. 3 Out-of-plane stress ($\sigma_{33}$) field contour for the 20 N load case. Stresses in the legend are presented in MPa.

3. Results and Discussion

The out-of-plane stress field of the surface of the steel block was found to be predominantly compressive in nature and the stress field contour is shown in Fig. 3. Clearly the stress field was not singular, but was found to be roughly elliptical in nature with a major radius of 1.053 mm, and a minor radius of 0.140 mm. A mesh convergence study was carried out, and the solution was considered converged when the out-of-plane stress did not differ by more than 1% from the previous mesh, while using regular refinement of the near-field elements (the so-called h refinement of [10]).
Fig. 4 Normal stresses as a function of distance on the surface of the steel cylinder along a line oriented in a parallel fashion to the glass capillary.

Fig. 5 Normal stresses as a function of distance on the surface of the steel cylinder along a line oriented in a perpendicular fashion to the glass capillary.
Stress as a function of radius for the three normal stress components on the surface of the steel block along lines parallel (in the global 1 direction) and perpendicular (in the global 2 direction) to the axial direction of the glass capillary are shown in Figs. 4 and 5 for an applied load of 20 N, respectively. For the former, the three principal stresses were found to be compressive in nature and maximum at the center of the steel block, decaying to zero with increasing distance from the center. The corner of the glass slide resulted in a geometric stress concentration, which slightly raised the local stress state (Fig. 4). As for the latter, the $\sigma_{11}$ and $\sigma_{33}$ stresses were found to again be maximum and compressive at the center of the steel block. The $\sigma_{22}$ component exhibited a lobed feature that was maximum 60 $\mu$m from the center of the top of the steel block, and became slightly tensile in nature before decaying to zero (Fig. 5).

In Part II of this work, Hamstad used a dynamic axi-symmetric finite element code to calculate the surface wave displacement of the steel block subjected to the loading condition determined from this work. The out-of-plane stress field distribution determined in this work was used as input into the axi-symmetric dynamic finite element code. To convert the stress field for the axi-symmetric analysis, the centroidal locations of each element on the surface of the entire steel block were “binned” into 26 consecutive annular discs, and the corresponding stress values were averaged over the area. Figure 6 shows the averaged out-of-plane stress ($\sigma_{AVE}$) as a function of the distance from the center of the steel cylinder for both full and reduced integration element formulations. It is clear from Fig. 6 that using a reduced integration element formulation had a negligible effect on the numeric solution, and the analysis run-time was reduced by 18% in comparison to the full integration analysis. Thus, the reduced integration hexahedral element formulation proved to be the more valuable analysis technique for this work.

Stress contours for the two in-plane principal stresses of the glass capillary are shown for the 20 N load case. A state of bi-axial tension was found on the inner diameter of the glass capillary nearest the glass rod (Fig. 7). Considering the brittle nature of silicate glass (which performs relatively poorly in tension), the inner diameter nearest the glass rod was found to be the most probable point of fracture initiation. Additionally, the maximum values shown in Fig. 7 ($\sigma_{11} = 536.1$ MPa and $\sigma_{22} = 2515.0$ MPa) indicate the sensitivity that this experimental set-up has to

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\[^2\] The $\sigma_i$ and $s_i$ notations are used interchangeably as it is the convention of the FE software Abaqus; the two are identical in meaning, and are to be understood as the $i^{th}$ component of stress.
variability. The value of $\sigma_{22}$ was found to be on the order of the tensile strength of silicate glass \[11\], thus fracture loads, as well as potentially the actual fracture location, would be significantly affected by surface flaws on the inner diameter of the glass capillary, as well as dimensional variability on the inner and outer diameters of the glass capillary. The variability in typical glass capillaries, as discussed in \[5\], coupled with this work point out the reason that such large variability is observed in fracture loads (~50%).

4. Conclusion

In this work, it was found that the procedure described in ASTM test method E1106 does not result in a point source load. Rather the out-of-plane stress field on the surface of the steel
cylinder as a result of the applied loading was found to be roughly elliptical in nature (with a major radius of 1.053 mm and a minor radius of 0.140 mm). The solution of this multiple bodies in contact problem was achieved through the use of the finite element method. The significance of this finding will be elaborated upon in Part II of this work. In addition, it was found that the most probable point of glass capillary fracture would initiate on the inner diameter, nearest the glass rod.

In light of the primary findings of this work, and the significance that it has (elaborated upon in Part II), a revision of ASTM E1106 may be in order. A revision of the alternative analytic approach to calibration is necessary, as it was shown in this work that the approximation of a point source is not sufficiently accurate for the loading condition that occurs on the surface of the steel cylinder.

Acknowledgements

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References