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USE OF INELASTIC ANALYSIS IN CASK DESIGN

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ABSTRACT

In this paper, the advantages and disadvantages of inelastic analysis are discussed. Example calculations and designs showing the implications and significance of factors affecting inelastic analysis are given. From the results described in this paper it can be seen that inelastic analysis provides an improved method for the design of casks. It can also be seen that additional code and standards work is needed to give designers guidance in the use of inelastic analysis. Development of these codes and standards is an area where there is a definite need for additional work. The authors hope that this paper will help to define the areas where that need is most acute.

INTRODUCTION

The use of inelastic analysis for radioactive material transportation cask design has several advantages. Inelastic analysis allows the actual behavior of the cask subjected to impact or puncture to be determined. Because the true behavior of the cask is used in the design, it is possible for the designer to achieve more uniform factors of safety throughout the cask than is possible with other analysis techniques, and to quantify what those factors of safety are. Using a system-level approach to design leads to safer and more efficient casks. Another advantage of using inelastic analysis is that it allows the system of the impact limiter and the cask to be analyzed together. This gives a more accurate assessment of the loads imparted to the cask by the impact limiter, and eliminates the need for many of the assumptions about how the impact forces are transmitted to the cask body. Analyzing the impact limiter and cask together is more technically defensible and results in analysis results that more closely match any testing that is performed.

The use of inelastic analysis also has some disadvantages. Accurate inelastic analysis requires more detailed material properties than the traditional analysis methods. Specifically, inelastic analysis requires a complete stress-strain curve to failure, while traditional analyses can be conducted using only a few discrete points on that curve. Even if a stress-strain curve could be developed from the few discrete points, for inelastic analysis it is not always conservative to use the lower-bound estimates of material properties currently available for those discrete points. These two factors make the availability of design-code acceptable material data difficult to obtain during the design phase of a cask. Another problem stems from the lack of suitable acceptance criteria for inelastic analysis. The energy-based nature of the design-basis accidents for casks suggests that an energy-based acceptance criterion would be the best choice. However, the only inelastic analysis acceptance criterion currently available for cask designers is stress based, which is more appropriate for accidents that are load limited rather than energy limited.

ADVANTAGES OF INELASTIC ANALYSIS

A study performed at Sandia National Laboratories in the early 1990s assessed the advantages of inelastic analysis for impact accident events.[1] In this study a representative steel-lead-steel walled rail cask and a representative steel-DU-steel walled truck cask were designed to resist a center-of-gravity-over-corner impact scenario using both elastic and inelastic analysis techniques. The elastic analyses in the

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DISCLAIMER

Sandia study used a transient dynamic technique with nonlinear material behavior for all elements except the stainless steel, thereby modelling the impact event as accurately as possible while retaining the requirement for elastic behavior of the stainless steel. Generally, elastic analyses are performed in a quasi-static fashion ignoring all sources of nonlinear behavior such as shielding response. This commonly utilized technique was not used in the Sandia study so that the comparisons would be based entirely on differences in the stainless steel shell response instead of modelling technique. The study found that the use of an inelastic analysis technique for radioactive material transportation container design has an advantage over elastic analysis. Based on the impact scenarios of the rail and truck RAM packages studied, an improved knowledge of the cask behavior is obtained by using inelastic analysis. This can lead to a better overall design in the following ways.

First, elastic analysis may underpredict maximum stress at a particular location, resulting in inappropriately sized wall sections. Elastic analysis does not properly account for the decrease in stiffness that occurs when a part of the structure yields, and does not show the redistribution of load caused by this yielding. For the rail cask, the elastic analysis underpredicted the stress in the inner shell. The underprediction was a result of the outer shell yielding and redistributing the load to the gamma shielding and inner shell. It was also observed in the inelastic analysis that significant plastic straining can occur through the thickness in several areas. This may indicate that the elastic analysis is neglecting significant physical features of the impact scenario.

Conversely, elastic analysis may overpredict the maximum stress. The inelastic shells can yield and redistribute the loading to other less loaded parts of the structure, whereas the elastic shells cannot predict this behavior. This was shown in both of the 30-foot center-of-gravity-over-corner drop analyses. For the rail cask, the elastic analysis of the impact event would require an outer shell thickness of over 3.5 inches to meet the design criteria. The inelastic analysis suggested that the loading on the outer shell caused it to yield and redistribute the load to the gamma shielding and inner shell, leading to a required outer shell thickness of only 0.6 inches. Therefore, the inelastic analysis may allow for a better distribution of structural material, which can lead to weight savings. The weight savings can increase the capacity of the package, thereby decreasing the number of shipments required to transport a given quantity of material, which increases the overall shipping program safety. The use of inelastic analysis may also decrease the overall cost of a transportation package, especially for designs where multiple packages will be constructed.

DISADVANTAGES OF INELASTIC ANALYSIS

The benefits of inelastic analysis do not come without a price. In order to calculate a more accurate prediction of package response, a more accurate depiction of material response is needed. To perform accurate inelastic analysis a complete representation of the stress-strain curve is necessary. For many materials a complete representation of the stress-strain curve is not readily available or is not accepted by a standards group. In addition, material properties which account for the strain rate and temperature dependence of the stress-strain curves may be required. During impact testing the strain rates typically range from $10^{-1}\ s^{-1}$ to $10^{3}\ s^{-1}$. The fact that radioactive material contents generally have a temperature higher than the outside ambient means there will be a temperature gradient through the wall of the cask. For certain materials, especially the lead shielding used in some casks, the effects of temperature and strain rate on the material behavior can be significant and should be considered in the analysis. For elastic analysis the effects of strain rate and temperature on yield stress are well known. For an inelastic analysis it is necessary to consider the entire material response (often much past the strain corresponding to yield) where the effects of temperature and strain rate are not well known for a wide selection of materials.

An improved understanding of the response of the container depends on how accurately the loading history is predicted. The transient dynamic analysis technique can more accurately predict the load history if all sources of nonlinearity are considered. That includes the nonlinear thermomechanical behavior of the shielding material, contents, and impact limiters, and the nonlinearity arising from fabrication initial stresses, geometric imperfections, and fastener details. In elastic analysis the stresses from these sources are typically superimposed on the results of the elastic impact analysis. For nonlinear behavior to be considered in a transient dynamic analysis, it is no longer appropriate to superimpose stresses, as initial stresses may influence the load history.

If adequate material models and all sources of stresses are taken into account, there is still a problem with conducting inelastic analyses for energy limited events. To determine if the proposed design is adequate, there must be an acceptance criterion to measure the analysis results against. NRC regulations require the post-accident condition of the package to meet shielding, containment, and criticality conditions, but there is no regulation pertaining to inelastic analysis acceptance criteria. The only guidance from the NRC on structural analysis acceptance criteria is given in Regulatory Guide 7.6, Design Criteria for the Structural

Analysis of Shipping Cask Containment Vessels,[2] for elastic analysis.

PROBLEM STATEMENT

In the time since the Sandia study, two new documents have been accepted by ASME. The first of these was the adoption by ASME of the Boiler and Pressure Vessel Code Section III, Division 3, Containment Systems and Transport Packagings for Spent Nuclear Fuel and High Level Radioactive Waste,[3] commonly known as the NUPACK code, in 1997. This code provides rules for the structural analysis of casks, and establishes the elastic design method and acceptance criteria formally included in NRC Regulatory Guide 7.6. Then, in 1999, ASME Code Case N-626, Use of Plastic Analysis for the Design of Type B Containment Components for Nuclear Material Transportation Casks,[4] provided an alternative method of analysis for the puncture accident scenario.

Using these two new codes, the authors attempted to determine the minimum outer shell thickness required to avoid puncture due to a 40-inch drop onto a 6-inch diameter punch, as described in 10CFR71. The requirements from Code Case N-626 state that the maximum stress in the inner shell shall not exceed 90% of the ultimate stress. The requirements from NUPACK state that material data should be taken from the ASME Pressure Vessel and Piping Code, Section II, Part D for Division 1, Class 1 or Division 3 Class TP, which lists an ultimate stress of 70,000 psi for the stainless steel material used in the example cask.

APPROACH

Dimensions for the initial analysis are given in Table 1, and a view of the finite element mesh is shown in Figure 1 Only a portion of the cask is represented with a layered wall construction. The remainder of the cask is lumped into the end masses, which are used to achieve a total cask weight of 135 tons.

Table 1: Initial dimensions

mass	135 tons
inner shell thickness	0.98 in. (2.5 cm)
lead shielding thickness	5.95 in. (15.1 cm)
outer shell thickness	1.54 in. (3.9 cm)
inner radius	71.73 in. (182.2 cm)
punch diameter	6 in. (15.2 cm)
drop height	40 in. (101.6 cm)

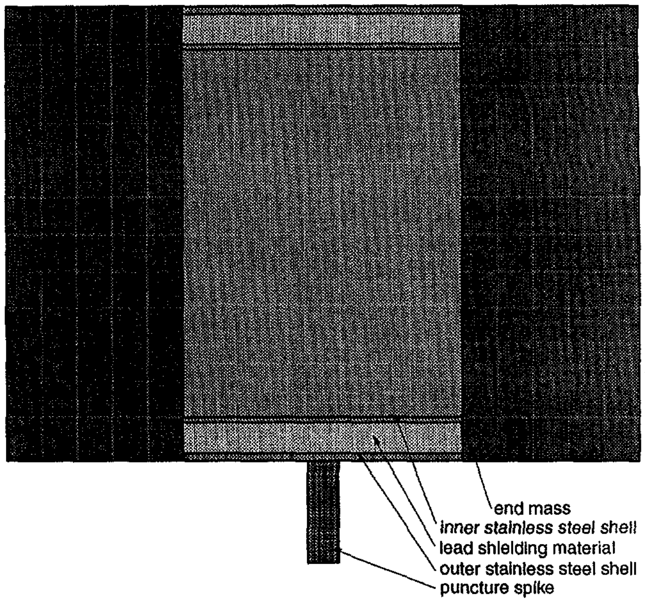

Figure 1 - Initial finite element mesh

Analyses were performed using PRONTO-3D, a transient dynamic finite element code.[5] Assumed material properties are given in Table 2. An elastic/plastic power law hardening model was used to represent the steel and lead materials that make up the layered shell wall. The description of this material model is given by Equation 1,

$$\sigma = E\varepsilon \qquad \text{for } \sigma < \sigma_y \qquad (1)$$

$$\sigma = \sigma_y + A\varepsilon_p^{\ n} \qquad \text{for } \sigma > \sigma_y$$

where σ is the stress, ε is the elastic strain, and ε_p is the plastic strain.

Table 2: Material properties

Property	304L Stainless Steel (inner, outer shells)	Lead (shield material)	Mild Steel (punch)
Density (lb/in^3)	0.2859	0.4134	0.2859
Young's Modulus E (psi)	28 x 10^6	2 x 10^6	30 x 10^6
Poissons' Ratio	0.27	0.27	0.27
Yield Stress σ_y (psi)	28000	1000	42000
Hardening Constant A (psi)	192746	800	160000
Hardening Exponent n	0.74819	0.5	

The punch was modeled using an elastic/plastic material model, as described by Equation 2.

$$\sigma = E\varepsilon \qquad \text{for } \sigma < \sigma_y \qquad (2)$$
$$\sigma = \sigma_y + A\varepsilon_p \qquad \text{for } \sigma > \sigma_y$$

FAILURE PREDICTION

To avoid puncture of a lead-shielded, multiple-layered steel containment vessel, the minimum outer shell thickness can be calculated using Nelm's equation as recommended in the NUPACK code. Nelm's equation can be written in the form shown in Equation 3.

$$t_{req} = \left(\frac{aW}{S_u}\right)^{0.71} \qquad (3)$$

where;

t_{req} is the required thickness of the outer shell,

W is the total weight of the loaded vessel, 270,000 lbs.,

S_u is the ultimate (engineering) tensile strength of the outer shell, 70,000 psi, and

a is 1.0 for vessels with a diameter greater than 30 inches.

According to this equation, the thickness of the outer shell should be 2.6 inches to avoid puncture.

Alternately, Code Case N-626 allows use of a plastic analysis to determine outer shell thickness. Using this approach for a stainless steel shell, the stress in the inner containment shell is not to exceed 0.9 S_u. This number is believed to be quite conservative based on experimental coupon testing of 304L stainless steel, as reported by Wellman and Salzbrenner.[6] A plot of stress vs. strain for 304L stainless steel is shown in Figure 2. Because the finite element code gives results in terms of true stress and true strain, the failure criteria of 0.9 S_u is shown on the true stress/true strain curve. The true stress corresponding to an engineering stress of 0.9 S_u = 63,000 psi is 72,700 psi. The value for the point of peak load in a tension test was taken from Reference 6.

ANALYTICAL RESULTS

The initial analysis used the cask dimensions as given in Table 1. The deformation results are shown in Figure 3. An inward deflection of 4.28 inches is predicted for the inner shell. Von Mises stresses in the inner shell are shown in Figure 4. All true stresses are below the 72,700 psi failure limit imposed by Code Case N-626. Von Mises stresses in the outer shell are shown in Figure 5. The maximum stress

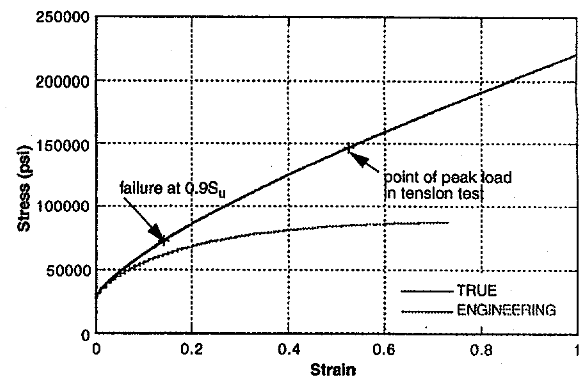

Figure 2 - Stress-strain curve for 304L stainless steel

exceeds 72,700 psi, but is less than the true stress at maximum load of 147,000 psi obtained experimentally in Reference 6. At this stress level the outer skin would not fail, and the inner containment boundary would not fail even by the most conservative estimate. The baseline case was compared to a case with outer shell thickness determined according to Nelm's equation.

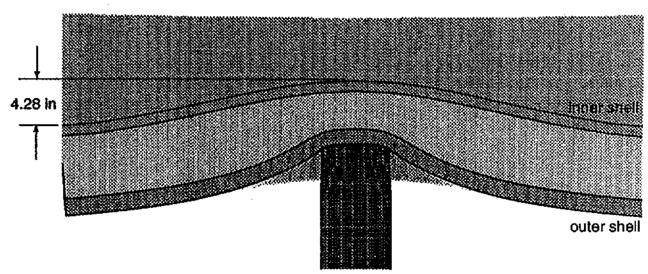

Figure 3 - Deflection of interior containment boundary for outer shell thickness = 1.54 inches

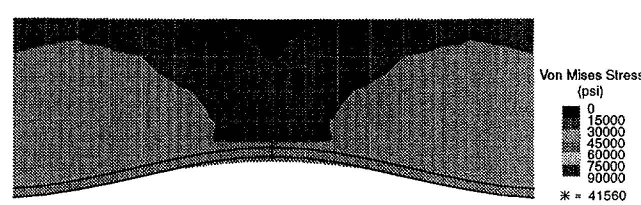

Figure 4 - Von Mises stress in inner shell for outer shell thickness = 1.54 inches

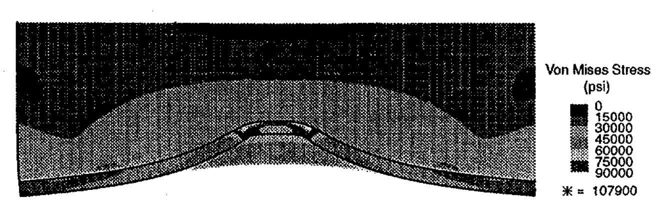

Figure 5 - Von Mises stress in outer shell for outer shell thickness = 1.54 inches

For an outer shell thickness of 2.6 in, as suggested by Nelm's equation, the deformations are as shown in Figure 6. The inward deflection of the inner shell was reduced to 2.8 inches. Von Mises stresses in the inner and outer shells are shown in Figures 7 and 8. The inner shell peak stress was reduced to 37,700 psi and the outer shell peak stress was reduced to 85,800 psi. The stress level in the outer shell is still higher than the 0.9 S_u (72,700 psi) level given by Code Case N-626.

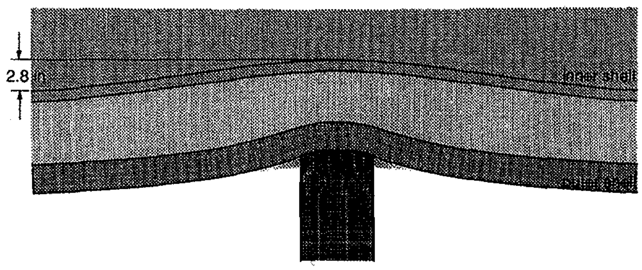

Figure 6 - Deflection of interior containment boundary for outer shell thickness = 2.6 inches

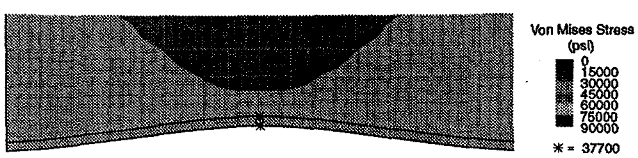

Figure 7 - Von Mises stress in inner shell for outer shell thickness = 2.6 inches

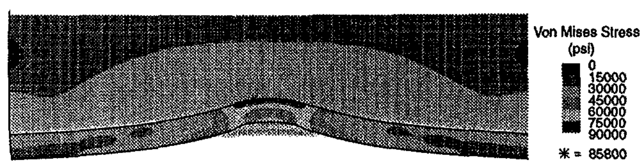

Figure 8 - Von Mises stress in outer shell for outer shell thickness = 2.6 inches

DISCUSSION

This design against puncture for a sample cask has shown several problems with using the recently developed NUPACK code and Code Case N-626. The puncture resistance requirement in the NUPACK code was developed only for lead-backed steel shells. However, this code does not give the designer any guidance on response of the containment boundary (usually the inner shell for this type of cask), but is designed to assure the outer shell is not perforated by the puncture test. On the other hand, Code Case N-626 provides guidance on containment boundary stresses, but does not provide guidance on stresses in the outer shell. Typically, a cask designer does not want material failure in either the containment boundary or the outer shell. If the plastic analysis stress requirements of Code Case N-626 for the containment boundary are applied to the outer shell, designs that are acceptable using Nelm's equation from the NUPACK code do not meet the stress requirement.

The material data from the ASME Pressure Vessel and Piping Code, Section II, Part D for Division 1, Class 1 or Division 3 Class TP referenced for Code Case N-626 was developed for elastic analysis of load limited structures. Applying the requirements for limit stresses from this code to non-linear transient dynamic energy-limited events such as puncture and impact tests results in designs that are extremely conservative. The energy absorbing capacity of a material is proportional to the area under the stress-strain curve. Examination of Figure 2 shows that the area under the stress-strain curve up to the point that corresponds to an engineering stress of 0.9 S_u is only a small percentage of the area under the curve up to the point of maximum load in a tension test. Ductile materials such as stainless steel have a large energy absorbing capacity beyond the point of maximum load in a tension test. Typical values for true strain at failure for 304L stainless steel are around 120%, compared to 55% at the point of maximum load.

SUMMARY AND RECOMMENDATIONS

The inelastic analysis technique provides an accurate assessment of the behavior of a cask to the hypothetical impact and puncture events. Using this technique, the designer can assess the survivability of the package to these events. In addition, this analysis technique provides information about the final geometry of the package after these events. Information about the final geometry is not available using other analysis techniques. Using inelastic analysis techniques results in improved knowledge about the response of the package, which leads to improved designs. The improved designs are safer, more efficient, and less expensive to fabricate than those developed using older, more traditional, analysis techniques.

The application of inelastic analysis has been demonstrated for a steel-lead-steel wall spent fuel rail cask subjected to the puncture event. Two outer wall thicknesses were considered. Both cases passed the maximum stress failure criteria imposed on the inner shell by the recent ASME Code Case N-626. If this stress criterion is applied to the outer shell, neither design would be acceptable. Inward deflection of the inner shell was 4.28 inches for a 1.54 inch thick outer shell, and 2.8 inches for a 2.6 inch thick outer shell. Using the outer shell thickness required by the NUPACK code (2.6 inches) results in lower stresses in both shells and a smaller inward deflection, but the inelastic analysis presented here suggests that a package with an outer shell thickness of 1.54 inches will also survive the puncture event without loss of containment or tearing of the outer shell that could result in a loss of shielding.

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