Seal Mismatch During Postbuckling of an Equipment Hatch

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Abstract

There are at least two models used to characterize the possible leakage of a containment during a severe accident: (1) the threshold model in which a very large rupture occurs. and (2) the leak-before-break model in which small leak paths develop at levels below the threshold. This work investigates the leak-before-break potential of a typical equipment hatch seal. The relative deformations of the sealing surfaces during pressurization are predicted with a three-dimensional finite element model of the equipment hatch assembly. The dominate observable behavior in the prebuckling range was "ovaling" of the penetration sleeve relative to the hatch cover. In the buckling and postbuckling regions, seal deformations increased but not enough to cause leakage.

1. Introduction

Nuclear containments are designed to prevent leakage of radioactive material. As the pressure and temperature increase during a severe accident, which is beyond the design level, the containment may reach a point where leakage begins to occur. There are at least two models used to characterize this leakage--the threshold model and the leak-before-break model.

In the threshold model, the containment is assured to be leak-tight until certain pressure/temperature conditions are reached. At this point, a very large rupture or burst area is postulated and large quantities of fission products are released. Typically, the predicted threshold pressure is based upon a model of the containment shell which only permits gross failure modes. Local discontinuities, e.g., penetrations, hatches, seals, are usually omitted from such models [1].

As an alternative model, the leak-before-break model is probably more realistic. In this model, it is hypothesized that small leak paths will develop as the containment is being pressurized at levels below the threshold. Hence, pressure is being released at earlier stages and the threshold pressure may never be reached. Typically, these small leaks are visualized as occurring at local details, e.g., penetrations, hatches, and seals which are often omitted from the threshold model. The objective of this work is to investigate the leak-before-break potential of a typical equipment hatch seal. Large displacements due to ovaling of the penetration sleeve and to postbuckling of the hatch shell are considered to be possible causes of leakage as local deformations distort the sealing surface.

2. Equipment Hatch Description

There are many different equipment hatch configurations in existing containments. Since postbuckling was a consideration, a hatch under compression was selected (pressure seating) as opposed to a hatch in tension (pressure unseating). Again, since buckling was important, it was decided to select a hatch with the most likely possibility of buckling, i.e., a large r/t value.

The equipment hatch is located on the containment meridian as shown in Fig. 1. Of particular interest in Figures 1, 2, and 3 are:

(1) The containment shell is 1/2 inch thick just below the springline. Previous studies [2] using axisymmetric (threshold type) models of the containment, which neglect penetrations and other nonsymmetric discontinuities, predict shell failure in the 1/2 inch plate at a static pressure near 60 psig.

(2) There is 5/8 inch shell plate in the immediate vicinity of the equipment hatch penetration. The local plate thickness of 1-1/2 inch is presumably used to satisfy the ASME area replacement rule.

(3) The sleeve length between the containment wall and the sealing surfaces is only 14.5 inches at the horizontal diameter of the sleeve. This distance is quite short and, as will be seen, couples the containment and hatch shell deformations.

(4) The actual containment in the upper right quadrant of the hatch region is shown in Fig. 1, except a ring actually located at Elevation 740' 6 5/8" was moved to the hatch centerline to introduce symmetry. The containment in the remaining three quadrants is not symmetrical, e.g., the 5/8 inch plate does not exist in the same locations in the other quadrants.

Shell geometric imperfections have a significant effect on buckling strength. For the subsequent three-dimentional analysis, a single cosine imperfection lobe was used with a length of 47 inches, which is the critical buckling length for elastic buckling of a perfect sphere [3]. The ASME maximum [4] deviation from perfect circularity of 3/4 inch was used. The imperfection was also assumed to be 47 inches in the circumferential direction, on the basis that this shape will be sympathetic with the elastic buckling mode in which eight lobes form around the circumference. The imperfection was located near the flange to maximize its effect on possible distortion at the sealing surface. In anticipation of the possible ovaling effect, compression in the hatch will be largest in the vertical direction. Hence, the imperfection was placed on a vertical diameter (12 o'clock position).

The containment steel is A516, Grade 60 with a specified minimum yield of 32 ksi. Actual mill test yield strengths for typical plates with the various thicknesses were used. Residual stresses have a significant affect on the buckling strength of steel components. When regions with residual stress yield, the material tangent modulus is reduced and the structural tangent stiffness is, thereby, reduced. In lieu of knowing the actual initial residual stress pattern, this effect can be approximated by using an effective stress-strain curve with a proportional limit below the yield strength [5,6]. For the present case, the proportional limit was taken as one-half yield which approximates residual stresses of the order of one-half yield. The stress-strain curve was approximated by a piecewise linear curve.

3. Three-Dimensional Analysis

3.1 Model

A three-dimensional finite element model of the equipment hatch was constructed for the ANSYS [7] computer program. One-quarter symmetry is assumed about the hatch vertical and horizontal diameters, as suggested in Fig. 2, even though the containment is only approximately symmetrical. Note that this forced symmetry implies an idealized imperfection at the bottom of the hatch (6 o'clock). Nonsymmetric buckling is also precluded. The model is extended only 20 degrees circumferentially from the meridional plane through the center of the hatch (Fig. 2). (This certainty represents a minimum.) The top of the model is constrained to move uniformly vertically, i.e., the top of the model remains a plane section. Flat triangular shell elements were used (STIF48 in ANSYS). Figure 4 is a plot of the finite element mesh from inside the containment (915 elements, 541 nodes). The swing bolts which are pretensioned to hold the hatch against the sealing surface (Fig. 4) were included in the model as uniaxial, pretensioned bar elements (STIF8). The material model follows the von Mises yield surface with the Prandtl-Reuss flow rule with isotropic strain hardening.

The interface between the flange and the sleeve is modeled using interface elements (STIF 52) which have friction and opening capability. Normal to the interface, the element is very stiff in compression and very flexible in tension. Tangential to the interface, the element permits relative sliding if the tangential force exceeds the normal (compressive) force multiplied by the coefficient of friction. Two interface elements are used at each nodal meridian of the sleeve--one on the inside and one on the outside of the sleeve. Such an arrangement permits rotational and sliding discontinuities at the seal interface. Two different coefficients of friction, 0.3 and 0.6, were used in the analysis of the hatch model.

3.2 Loads and Execution

The swing bolts were first prestressed to 25 kip/bolt (about 1/4 yield). The finite element model was then loaded to correspond to internal pressure on the containment, i.e., pressure on the inside face of the appropriate elements and a vertical load at the top of the model corresponding to the meridional membrane stress resultant in the containment.

In an attempt to track the postbuckling behavior with a physically, realistic analysis, the "slow dynamic" analysis suggested in [7] was used for the 0.3 coefficient of friction case. In principle the method is attractive because the actual buckling process is dynamic. A high damping ratio is used to minimize the vibration response. Physically, this would correspond to placing the hatch in a viscous fluid during pressurization. The pressure was increased linearly over a rise time larger than several times the structure natural period (about 1 psi per second). The integration time increments were initially quite large (1 second) as the structure behaved linearly and the high damping produced an essentially static response. As material and geometric nonlinearities began to dominate the problem, the time steps were successively cut in half. For example, a time step of 0.125 seconds was used at 82 psi. At this point, the pressure was increased to 85 psi as a step function and 24 time increments of 0.01 second were run. Since no buckling occurred and since the postbuckling shape and not necessarily the buckling pressure was of interest, the pressure was stepped to 90 psi and the solution was performed for 100 time steps. During these steps, the time increment size varied between 0.01 to 0.1 seconds. A "traditional" static analysis was performed for the a 0.6 coefficient of friction case.

3.3 Results

Figure 5 is a plot of the radial displacement before buckling at the center of the hatch and the center of the imperfection. In the prebuckling regime, ovaling of the sleeve occurred. That is, the circumferential tensile forces in the containment cause the sleeve to deform into an elliptical shape with a horizontal major axis. The hatch remains essentially circular, although the frictional forces tend to ovalize it also. The net result is that, at 82 psi and with a 0.3 coefficient of friction, the sleeve moves outward 0.5 inch farther than the hatch flange on the horizontal diameter (3 and 9 o'clock) and the sleeve moves inward 0.9 inch farther than the hatch flange on the vertical diameter (6 and 12 o'clock).

The solution was continued to track the postbuckling behavior of the hatch, especially the seal surface. During the postbuckling process, the radial displacement in the imperfection region continued to grow rapidly compared to the rest of the hatch model. Figure 6 illustrates the deformed shape of the hatch as the buckling process initiated at 90 psi and continued to snap-through. The displacements are plotted to the same scale as the original structure. Imposed also in Fig. 6 is an inset figure, drawn to scale, illustrating the relative translation and rotation of the sealing surfaces. With these small amounts of displacements of the seals, leakage is unlikely [8]--especially in the prebuckled state.

4. Summary and Conclusions

This study was undertaken to investigate the leak-before-break potential of a typical equipment hatch seal. Buckling of the hatch door, large deformations and ovaling of the hatch sleeve are potential causes of mismatch at the sealing surface were investigated with a three-dimensional finite element analysis of the containment/sleeve/hatch assembly. A critical imperfection shape was incorporated into the hatch model and was selected to be used in the model. The model was analyzed twice, using coefficients of friction of 0.6 and 0.3. The results of the finite element analysis indicated that neither pre- nor postbuckling displacements caused excessive relative motions at the seal surface.

In conclusion, the equipment hatch should not leak before very large strains develop in the 1/2-inch containment shell plate near the springline, which occurs near 60 psi. In the unlikely event of hatch buckling, postbuckling deformations would not introduce leakage.

Some of the items which were not included in this study which could be looked at in the future include other hatch imperfections, different friction coefficients and other pretensions in the swing bolts. The analysis methods should continue to be calibrated with existing experimental results, many of which have been published, e.g., shell buckling, pressure vessel experiments. The behavior of the seals with flange separation and rotation needs more study. The effect of temperature gradients on displacements at the seal surface could be important.

High strains in the sleeve, near the containment, should be investigated further. As pressure is increased, leakage will most likely begin at these high strain locations. The potential size of such a path should be studied.

5. References

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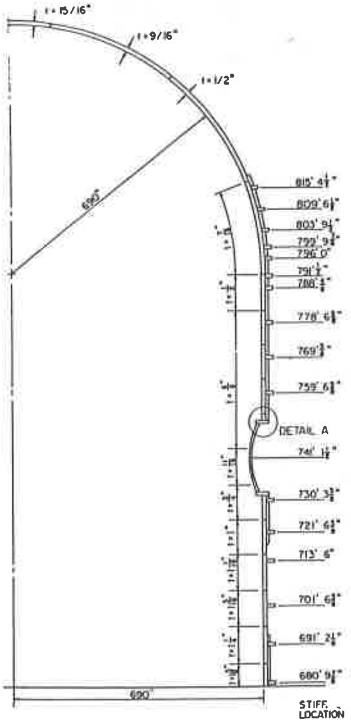


Figure 1 Containment Meridian

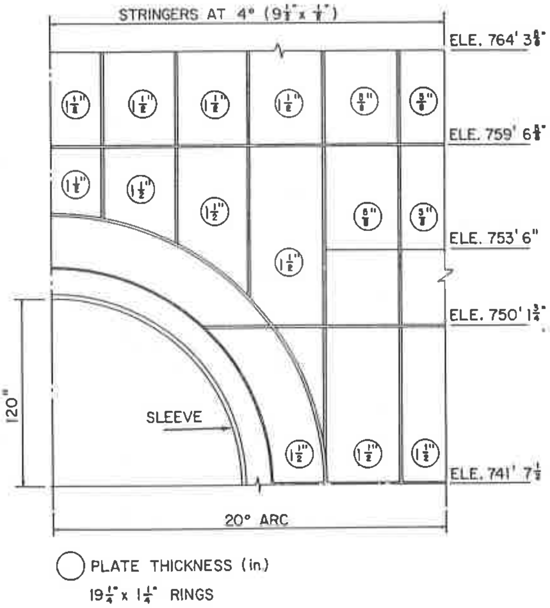


Figure 2 Shell Thickness

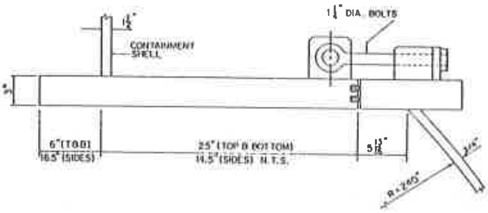


Figure 3 Detail A - Equipment Hatch

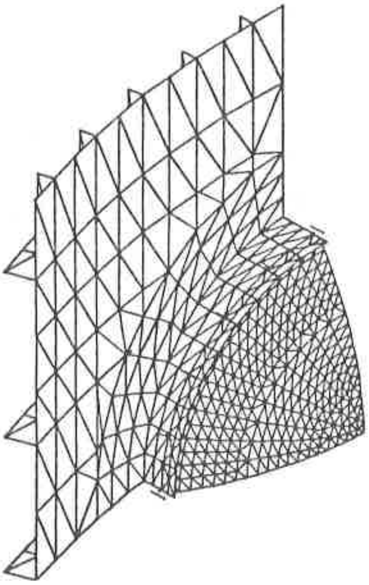


Figure 4 Plot of Finite Element Mesh

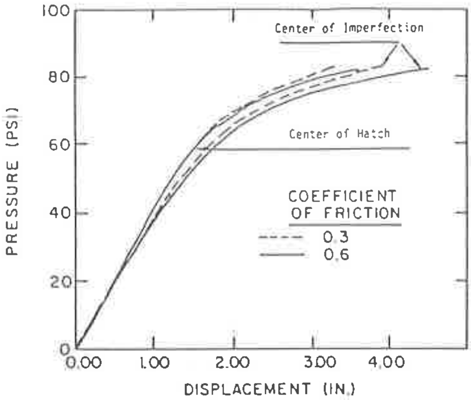
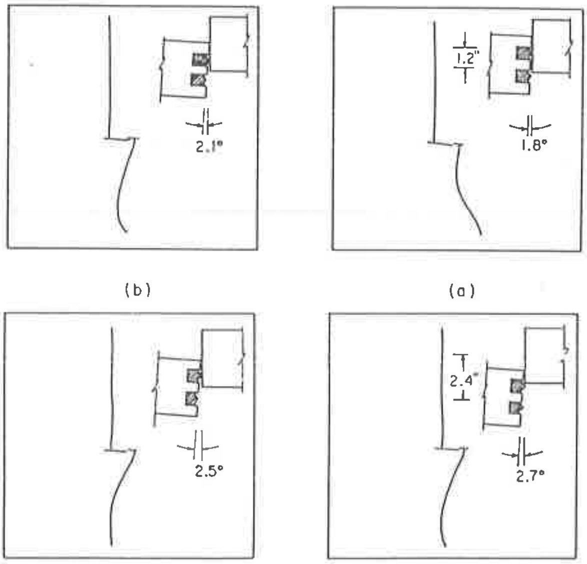


Figure 5 Radial Displacement



(b) (a)

(c) (d)

Figure 6 Successive Stages of Post-Buckling