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Failure of pressurized reactor vessels due to severe pressure transients

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ABSTRACT: We will consider a particular family of severe accidents, namely uncontrolled pressure transients, to assess whether it is possible to guarantee that the incident sequence leads to limited structural damage or to the loss of sealing of the vessel. With a detailed study of the deformation behavior of flanges and O-Rings, based on both finite element analysis and simple model, the sealing at the flange/O-Ring interface of an AP600-like vessel is investigated. The theoretical pressure level at which the sealing is lost depends on both bolt tightening and O-Ring design. The obtained results demonstrate that, under some circumstances, the sealing is lost and, consequently, the primary circuit is depressurized, before that a serious structural damage of the vessel occurs. A simple methodology that permits rapid and reasonably accurate analysis for assessing the sealing of AP600-like vessels was developed. This methodology is easily applicable to other plant-types.

1. INTRODUCTION

The reliability of nuclear power plants needs thorough attention to minimize the potential risk of radiological exposure of the public by every feasible mean. Therefore in the last few years, the safety analysis of nuclear plants has also been performed under severe accident conditions, which include both the non-functioning of the safety-relief valve that can cause an uncontrolled overpressurization of the primary circuit, and the non-functioning of the emergency cooling system with the possibility of subsequent core melting. While core melting has been investigated in numerous recent studies, uncontrolled overpressurization still lacks a methodical investigation.

Ideally, during an uncontrolled overpressurization, stresses in the vessel or in the primary circuit can rise to the ultimate limit and cause a catastrophic rupture with the consequent dynamic loading on the containment building. However if loss of sealing or limited structural damage occurs, we can have a depressurization of the primary circuit that considerably modifies the accident evolution. This observation results in a substantial change for the safety requirements in pressure vessel sealing assessments. In fact the purpose of this research is not to unconditionally guarantee sealing, even in the case of uncontrolled pressure transients, but to assure leakage before the beginning of other, limited or catastrophic, failure modes.

2. CONSIDERED INCIDENTS

Uncontrolled pressure transients belong to a family of incidents that have little probability of occurring, but can produce dynamic loading on the containment building. The analysis is conducted at service temperature with the assumption that no significant thermal stresses, due for example to hot spots or high thermal gradients, are present in the structure. We will not consider any incident that implies core damage and that could give raise to dynamic loads on the vessel or primary circuit. Brittle fracture of the vessel under service and upset conditions are likewise excluded.

3. SEALING ANALYSIS

In this work the pressure vessel sealing is analyzed at the flange/O-Ring interface, mainly by means of a complex finite element model, but also by means of a simplified "engineering" model of the O-Ring. The analysis is based on the accurate description of all the components considered, briefly outlined below.

3.1 O-Ring

Metallic O-Rings for nuclear power plants are made of metal tubing which is formed into circular shapes and the two ends welded together. Under service conditions this type of O-Ring presents a nonlinear load-displacement curve, mainly due to localized plasticity and to geometrical nonlinearities (large deformation and displacements).

The load-displacement curve of the considered O-Rings, similar to those used in AP600-like pressure vessels, was obtained both with a simple toroidal shell model solved by finite differences and two different three dimensional axisymmetric finite element models. In the finite element analysis the flange/O-Ring interaction was explicitly considered as a contact problem in large deflection/deformation conditions. Two different models were developed:

- a detailed model (see figure 1) in which both the O-Ring and the flanges are considered deformable contacting bodies,
- a simplified model where the groove is replaced by rigid surfaces.

The analysis was repeated for three types of O-Ring, with same geometry but different internal pressurization: Plain (not Self-Energizing nor Pressure-Filled), Self-Energizing (the inner periphery of the O-Ring is vented by small holes or a slot which account for a pressure inside the O-Ring equal to the vessel pressure), Pressure-Filled (the ring is filled with an inert gas at about 4 MPa). The analysis is carried out accounting for the elasto-plastic behavior of the O-Ring material (Inconel 718).

This non linear analysis, carried out with the ABAQUS F.E. code, was able to correctly predict the wall instability that leads the O-Ring to assume a characteristic eight shape. From the detailed model the load-displacement curve, the critical load, the radial displacement on loading and recovery on unloading for every O-Ring have been calculated. In figure 2 is shown the load-displacement curve for Pressure-Filled O-Rings. The same analysis carried out with the simplified model agrees well with these results, showing that the approximation of rigid grooves is acceptable for modeling the contact between O-Ring and flanges. The detailed model however becomes necessary if we want to investigate the effects of the deformation of the flanges.

The "engineering" finite-difference model leaves out of consideration the contact between the O-Ring and the flanges: the load-displacement curve for a toroidal shell, loaded by two opposite systems of axisymmetric forces acting on the contact circumferences, is computed under the large displacement assumption. In spite of its simplicity, the predicted load-displacement curve is in well agreement with the finite element results.

3.2 Flanges

The stresses and deformations of the vessel flanges, when the internal pressure increases, were determined with reference to a three-dimensional finite element model of a vessel sector (figure 3) and an "equivalent" simplified axisymmetric model (figure 4). The three-dimensional model describes the rotation of the flanges with much greater accuracy. Frictionless contact between the flanges is considered, along with a fairly accurate three-dimensional model of the bolts: tightening is simulated imposing a displacement discontinuity between bolt and nut. The internal pressure, applied on the inner surface of the vessel, is supposed to be uniform throughout the analysis.

The pressure transient considered is a ramp between 0 and 28 MPa and the opening displacement of the flanges, for varying the bolt tightness, was computed. In figure 5 the flanges opening displacement, for different internal pressure levels, is depicted as a function of the radius in the cross-section relative to the bolt centerline; the same data are represented in figure 6 for a median cross-section between two bolts. Both figures correspond to a bolt-tightness of $224\ 10^6$ N.

The results of figure 5 and 6 indicate that an axisymmetric model can be adequate to compute the flanges opening displacement. In fact, the axisymmetric model in figure 4 is a very simple discretization in which the bolts are replaced by three springs of suitable stiffness and the presence of holes in the flanges is simulated by modifying the elastic material properties of the bored ring. In the axisymmetric model the bolts cannot be replaced by a single equivalent spring because this element exhibits only axial stiffness, while the bolt bending stiffness is not negligible. Using three springs in parallel (see fig. 4) with stiffness K_1, K_2 e K_3, it is possible to simulate more accurately the bending of the bolts by choosing the following values:

$$K_1 = 0.9K_2, \quad K_2 = \tfrac{1}{3}K, \quad K_3 = 1.1K_2,$$

where $K = K_1 + K_2 + K_3$ is the total axial stiffness of all the bolts. The material of the bored ring was considered anisotropic in the axisymmetric model: all the elastic moduli coincide with the corresponding (isotropic) moduli of the base material (SA533) with exception of Young modulus in circumferential direction $E_\theta = RE$, where R is the "void ratio" for the bored ring (defined as the holes volume over the unbored ring volume) and E the Young modulus of the isotropic material.

For the axisymmetric model both rough and frictionless contact were considered. The results for a bolt tightening of $F = 224\ 10^6$ [N] are depicted in figure 7, account taken of the friction between flanges. Up to a pressure of 20 MPa we have linear behavior, with subsequent rapid separation. We have a high level of agreement 3

	$F = 210 \cdot 10^6$ N		$F = 224 \cdot 10^6$ N		$F = 260 \cdot 10^6$ N	
	with friction	without friction	with friction	without friction	with friction	without friction
Plain	22,3	22,5	22,9	23,2	26,0	24,9
Pressure-Filled	23,9	24,3	24,4	24,72	26,2	25,4
Self-Energizing	24,9	25,3	25,4	25,8	> 28,0	28,0

Table 1: Pressure value at which the sealing is lost, [MPa], for different bolt tightening F.

with the corresponding values of figures 5 and 6.

If C and D are the contact points between O-Ring and flanges, figure 2, the distance $\overline{CD}$ is depicted in figure 8, as a function of internal pressure, for three different bolt tightening, $F = 210\,10^6$ N, $F = 224\,10^6$ N, $F = 260\,10^6$ N, respectively. These curve are obtained with a 0.1 friction coefficient between the flanges. The corresponding curves in absence of friction always present a slightly smaller flange opening.

4. CONCLUSIONS

The pressure level at which the groove opening displacement is greater than the elastic spring-back of the O-Ring is presented in table 1, for different bolt-tightening and with or without friction between the flanges: this value can be considered the predicted leakage pressure. While with plain O-Rings and low bolt-tightening this pressure does not endanger the structural integrity of the vessel, the case with self-energizing O-Rings and high bolt-tightening requires further investigation.

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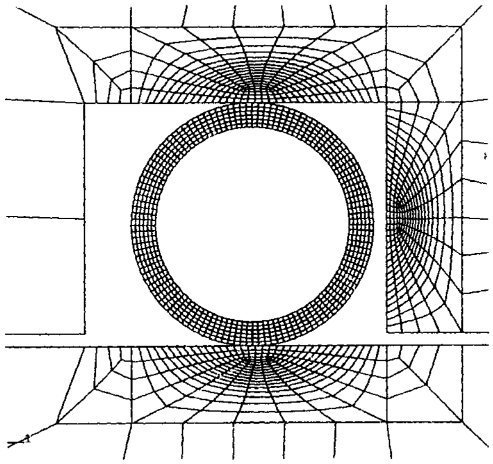

Figure 1: Detailed model of O-Ring

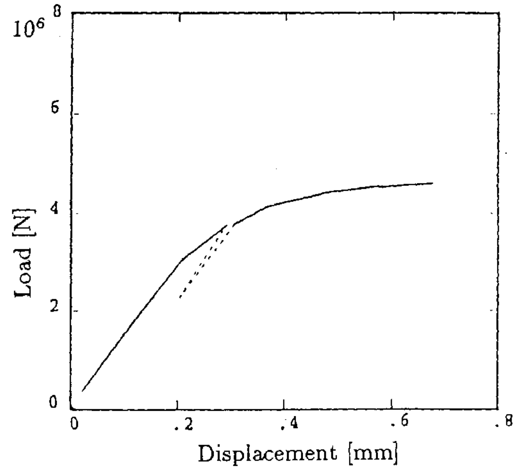

Figure 2: O-Ring Load-Displacement curve when tightening bolts

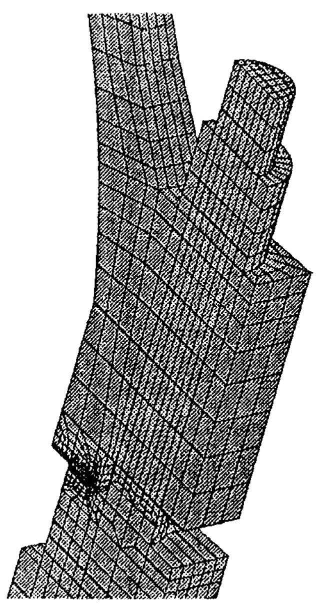

Figure 3: Three-dimensional model of the vessel

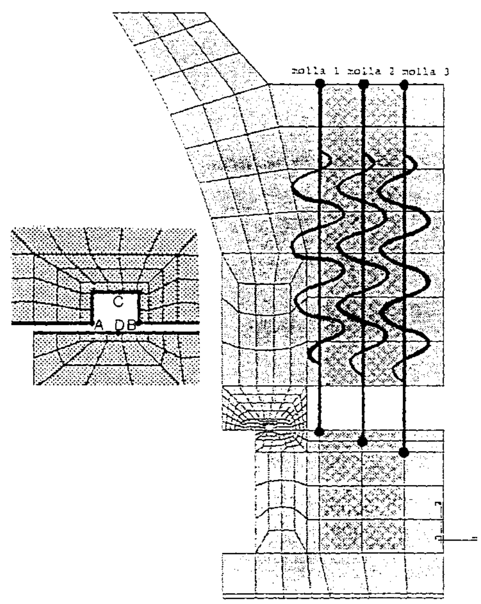

Figure 4: Axisymmetric model of the vessel

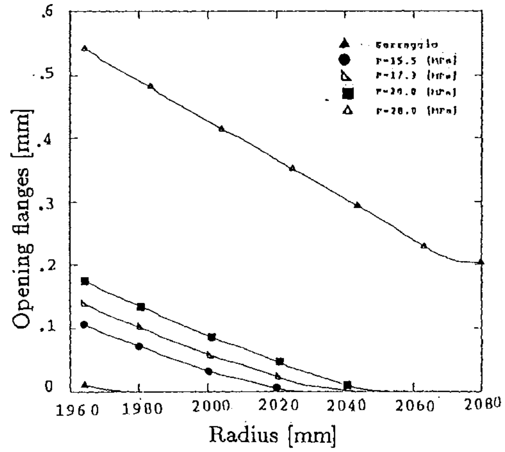

Figure 5: Opening displacement of the flanges in the cross-section relative to the bolt centerline, bolt-tightness of $224 \cdot 10^6$ N

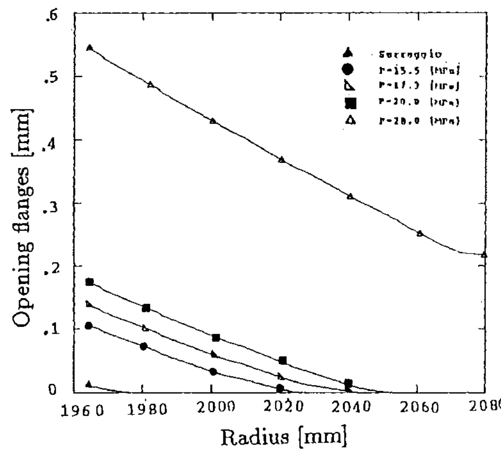

Figure 6: Opening displacement of the flanges in the cross-section relative for a median cross-section between two bolts, bolt-tightness of $224 \cdot 10^6$ N

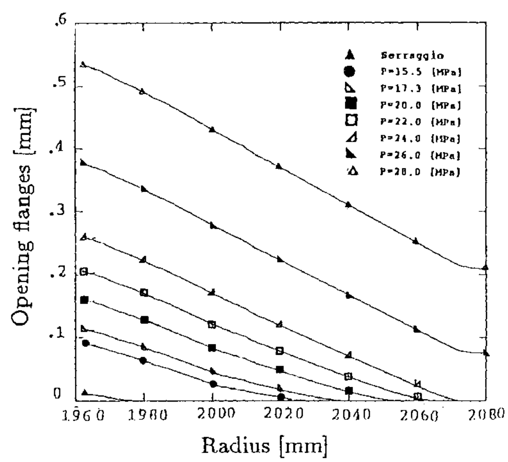

Figure 7: Opening displacement of the flanges, axisymmetric model, bolt-tightness $224 \cdot 10^6$ N, with friction

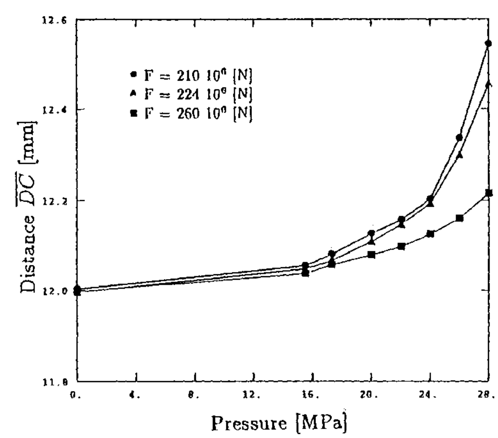

Figure 8: Distance $\overline{CD}$ as a function of internal pressure, bolt tightening $F = 210 \cdot 10^6$ N, $224 \cdot 10^6$ N, $260 \cdot 10^6$ N, axisymmetric model, with friction.