Proceedings of the 15th International Symposium on the Packaging and Transportation of Radioactive Materials
PATRAM 2007
October 21-26, 2007, Miami, Florida, USA

SIMILARITY ASPECTS FOR CLOSURE SYSTEMS IN SMALL SCALE PACKAGE DROP TESTING

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ABSTRACT

An exact scaling of all structural components of a package for radioactive materials and their mechanical characteristics is not always possible in drop tests with small-scale models. This has to be especially considered for bolted closure systems. On the one hand, the sizes of the bolts (diameter, pitch of the thread, stress cross section) cannot be scaled with the same geometrical scale factor. On the other hand, the possibilities of an accurate scaling of seal characteristics are very limited.

Due to nonlinearity of the force-compression relationship of a gasket it is, for instance, impossible to simulate the maximum compression force and permissible elastic decompression of a metallic gasket simultaneously on the same small-scale model. Additional problems can also result from a dispersion of friction conditions at the contact areas of the bolted joints (in threads and under bolt heads). This dispersion as well as an imprecision of the bolt tightening technique lead to more or less considerable scatter of the bolt pre-tension. The minimum pre-tension creates more severe conditions in a drop test with regard to the seal function (higher probability of the lid opening and sliding). The maximum pre-tension is usually conservative for the total bolt stress (the sum of the initial tension and additional load due to the drop test). These circumstances should be considered in planning drop tests as well as regarding the interpretation and transfer of test results to the original package design. For instance, in some cases a correction of the theoretical similarity relation between tightening torques for the bolts of the original cask and its tested small-scale model may be necessary to make this transfer correct.

In this paper some recommendations and examples are described concerning the modelling of closure systems based on BAM experience in the approval assessment of transport casks for radioactive materials.

INTRODUCTION

According to IAEA-Regulations [1], tests with the model of the package can be used as a constituent part of a demonstration of design safety under normal transport and hypothetical accident conditions. In general, the appliance of geometrical scale models is recommended [2]. An exact scaling of all structural components of a package and their mechanical characteristics is not always possible for reduced scale models. Especially for bolted closure systems, there are some technical limitations on the accurate fulfilment of similarity laws. Therefore, it has to be ensured that at least a "functional" similarity to the closure system of the original package is implemented in the model [3]. Potential failure modes of the sealing function in the original closure system should be identified first to achieve this "functional" similarity. Calibration of the closure system of the scale model before the drop test has to exclude the model having higher resistance against these failure modes than the original package. This is a crucial condition for transferability of test results regarding the sealing function to the original package. In this paper some recommendations and examples concerning modelling of closure systems of transport and storage casks for radioactive materials will be presented.

DESIGN CRITERIA AND CHARACTERISTIC LOADINGS OF CLOSURE SYSTEMS

The closure system of transport and storage casks usually consists of two lids (primary and secondary lids), covers for opening in these lids, bolts and metallic or elastomeric gaskets. The tightness of a closure system is usually maintained by metal O-rings in dual purpose transport and storage casks. Elastomeric gaskets can also be used to perform this function in pure transport casks. The gaskets are usually put in the special grooves on the lid. During the lid assembly, the gasket is compressed until contact of flange surfaces is achieved. Further tightening leads to additional pre-load (pre-tension) in the bolts, which is balanced by the clamp load on the lid flange. Increasing the pre-load after the contact in the flange is achieved does not change the gasket compression practically.

The pre-tension in the lid bolts has an essential influence on the sealing function. There is a considerable scatter of a large percent in the bolt pre-tension by an appointed torque because of the dispersion of friction conditions at the threads and under the bolt heads as well as an imprecision of the bolt tightening technique. Thus, it must be ensured that the necessary compression of the gasket is achieved under this dispersion, but the bolts also have sufficient strength reserves to resist all external loads from accident conditions. According to the BAM guideline for numerical analyses of bolted joints [4], the minimum pre-tension in the lid bolts has to prevent impairment of the sealing function under routine transport conditions. The criteria for the maximum pre-tension are:

$$\sigma_V \le \frac{R_{p0.2}(RT)}{1.5} \text{ (cask lid bolts);} \qquad \sigma_V \le \frac{R_{p0.2}(RT)}{1.1} \text{ (opening cover bolts)} \qquad \textbf{(1)}$$

where σ_V is the effective (Mises) stress and $R_{p0.2}(RT)$ is the 0.2% proof stress of the bolts at room temperature. Specific load situations under accident conditions for large and small lids are the reason for these different criteria. While the interaction with the contents or impact limiter has to be taken into account for the large cask lids, the opening covers are generally only loaded by their own inertia forces.

Depending on the accident situation, the forces acting on the lid are balanced by additional bolt loads in combination with a change of the clamp load or friction force on the lid flange. The gasket can be impaired if either contact on the flange is reduced significantly by the axial outside force and/or if the friction grip is overcome by the transverse one. An impairment of tightness due to outside forces can be caused by the following effects:

- Change of the gasket position in the flange so that the gasket leaves the sealing track. Such relative sliding displacements of the gasket are caused by transverse (lateral) outside forces on the lid or transformation of the cask cross section near the lid flange into an elliptical shape.
- Gasket decompression (lid opening) beyond useful elastic recovery due to axial outside forces on the lid.
- Loss of pre-tension in the lid bolts as a result of their plastic deformation or complete failure due to outside forces.

LIMITATIONS FOR MODELING CLOSURE SYSTEM COMPONENTS

Lid bolts

Scaling can be performed for clamping length, bolt head or stress cross section of shank bolts, (e.g. by additional mechanical treatment of the model bolts), but it is not possible for the bolt pitch because of its standardized stepwise change. This effect is estimated below for the bolts M48 and M12 with reference to the standard DIN EN ISO 4762 [5]. These bolt sizes have been selected because they are typical for the lids and opening covers of casks for spent nuclear fuel. On the other hand, bolts with nominal diameters scaled by factors λ = 2, 3, 4 are standardized in [5] as well. The estimation should only illustrate how geometrical deviations from accurate scaling can affect the assembly state of the joint.

The total tightening torque required to produce the pre-load is composed of the effective thread moment M_{Gst}, thread friction moment M_{GR} and head friction moment M_{KR} [6]:

$$M = M_{Gst} + M_{GR} + M_{KR} = F_M \left(0.16p + 0.58\mu_G d_2 + \frac{D_{km}}{2}\mu_K \right)\Big|_{Original} = F_M \Psi , \qquad \textbf{(2)}$$

$$M_\lambda = F_{M,\lambda} \left(0.16p + 0.58\mu_G d_2 + \frac{D_{km}}{2}\mu_K \right)\Big|_{Model} = F_{M,\lambda} \Psi_\lambda . \qquad \textbf{(3)}$$

Here, F_M and $F_{M,\lambda}$ are the assembly pre-loads in the bolts where index λ stands for a (1:λ)-model. p, d_2 and D_{km} are the pitch of the thread, pitch diameter of the bolt and effective diameter for the friction moment at the bolt head. μ_G and μ_K are the friction coefficients in the thread and head bearing area respectively. If the theoretical relation $M = \lambda^3 M_\lambda$ of the tightening torques in the original and model cask is kept, a relative deviation in the assembly pre-load of the model bolts to the theoretical value $F_{M,\lambda} = F_M / \lambda^2$ is

$$\Delta_F = \frac{F_M - \lambda^2 F_{M,\lambda}}{F_M} = \frac{M/\Psi - \lambda^2 (M_\lambda / \Psi_\lambda)}{M/\Psi} = 1 - \frac{\Psi}{\lambda \Psi_\lambda} \qquad \textbf{(4)}$$

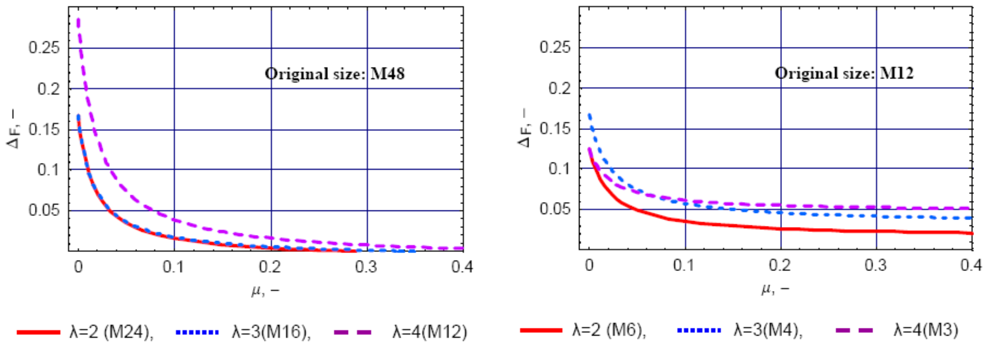

Figure 1. Relative deviation in the assembly pre-load of the model bolts Δ_F; Eq.(4)

These deviations are shown in Fig. 1 in relation to scale factor and friction coefficient for the bolt sizes under consideration (M48 and M12). The same friction coefficients were assumed for the thread and head bearing area. The analysis shows that the assembly pre-loads are smaller in the model bolts than the values derived theoretically from scaling laws. The relative error becomes higher with decreasing friction coefficients because deviation in the thread pitch scaling is dominant under the small friction.

Concerning the pre-load situation after assembly, the embedding effects should also be considered. These effects are caused by plastic flattening of surface roughness at the bearing areas and loaded flanks of the threads and finally lead to reduction of the assembly pre-load. Since they occur during operation they should only be taken into account for the original cask. Therefore, effective preloads are $F_V = F_M - F_Z$ for the bolts of the original closure system and $F_{V,\lambda} = F_{M,\lambda}$ for those of the model.

<u>Gaskets</u>

The metallic gasket characteristic shown in Fig. 2 focuses on a special Helicoflex design [7]. These gaskets consist of an inner spring with one or more metallic jackets, which can be deformed elastically or plastically. The nonlinear characteristic (Fig. 2) results from interaction between these elements.

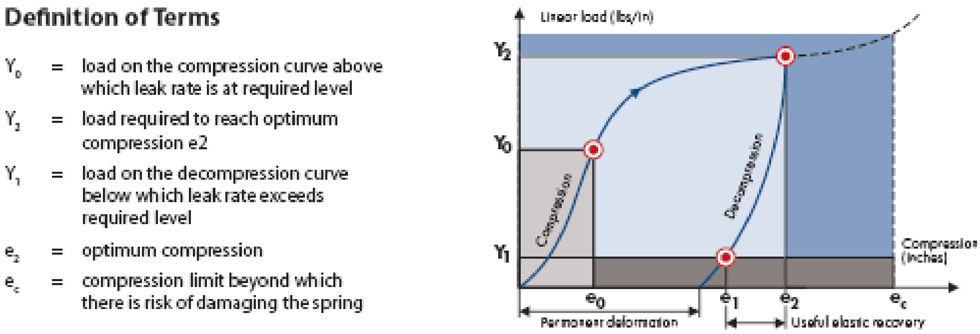

Figure 2. Metallic gasket characteristic [7]

The position of the gasket on the flange and, consequently, its total length can be scaled in the model. However, no accurate scaling of other gasket sizes and force-compression behaviour is possible, because metallic gaskets are only manufactured with certain torus diameters. Potential modes of tightness failure should be taken into account in deciding on model gaskets. As noted above, a change of the sealing function is always expected if decompression of the gasket beyond the useful elastic recovery $r_u = e_2 - e_1$ (Fig. 2) takes place under the axial load acting on the lid. Therefore, it makes sense to choose a model gasket which approximately allows scaling of the useful elastic recovery:

$$r_{u,\lambda} \leq \frac{r_u}{\lambda} . \qquad \textbf{(5)}$$

Since stiffness of the metallic gasket characterised by Y_2 does not decrease in the same relation as torus diameter, we usually have for the gasket reaction F_D

$$F_{D,\lambda} \geq \frac{F_D}{\lambda^2} . \qquad \textbf{(6)}$$

Lid sliding or flange deformation to an oval shape can lead to gasket displacement out of the sealing track. The width of the sealing track cannot be scaled as well. However, the sealing track is more narrow for the model gasket and the probability of leakage paths occurring in a narrow sealing track is higher. In this sense, behaviour of the model gasket is conservative compared to that of the original gasket. In general, gasket dislocations of such dimensions have to be prevented by defining appropriate gaps in the lid flange region.

EFFECTS OF SCALING DEVIATION ON TEST CONDITIONS FOR CLOSURE SYSTEMS

Pre-load dispersion and unfavourable conditions for the bolts

A specific problem for bolted joints is dispersion of the friction coefficients, which is finally embodied in dispersion of the bolt pre-load.

Before the contact interface in the lid flange is opened, the total load in the bolt results from the pre-tension and additional load from outside forces acting on the lid. Although higher additional loads can occur for the minimum pre-tension in the bolts (e.g. due to additional bending), the maximum total loads are generally reached for the maximum pre-tension. Only if the outside force leads to complete contact separation for the whole pre-tension range, the maximum total load will not depend on the pre-tension. Hence, simulation of the maximum pre-load before the drop test is conservative concerning tightness loss due to plastic deformation or failure of the bolts. On the other hand, the minimum pre-load is always unfavourable for the gaskets. In this case, the outside force, which would lead to lid displacement, is lower and gasket impairment probability increases. Both situations cannot be simultaneously set as the initial state before the drop test.

For additional considerations, it is sufficiently accurate to discuss the problem for a concentrically clamped and loaded connection.

The equilibrium conditions for the cask lid in an assembly state are:

$$N_S \cdot F_V = F_K + F_D \qquad \text{or} \qquad F_K = N_S \cdot F_V - F_D \tag{7}$$

where N_S is the number of bolts in the lid connection.

The residual clamp load on the flange under the outside axial force F_A is

$$F_{KR} = F_K - (1-\Phi)\cdot F_A = N_S \cdot F_V - F_D - (1-\Phi)\cdot F_A \tag{8}$$

where Φ is the relative resilience factor (or load factor).

In the case of a minimum pre-load, a complete opening at the flange ($F_{KR} = 0$) occurs when

$$F_{A0,\min} = \frac{N_S \cdot (F_{M,\min} - F_Z) - F_D}{(1-\Phi)} \tag{9}$$

In the case of a maximum pre-load, the force $F_{A0,min}$ does not lead to contact opening and the total load in the bolt is

$$F_S = (F_{M,\max} - F_Z) + \Phi \cdot \frac{F_{A0,\min}}{N_S} = (F_{M,\max} - F_Z) + \frac{\Phi}{(1-\Phi)} \cdot \left((F_{M,\min} - F_Z) - \frac{F_D}{N_S} \right) \tag{10}$$

or $$F_S < F_{M,\max} \cdot \left(1 + \frac{\Phi}{(1-\Phi)} \cdot \frac{1}{\alpha} \right) \tag{11}$$

where $\alpha = \dfrac{F_{M,max}}{F_{M,min}}$ is the tightening factor. With the usual values of $\alpha \in [1.4, 2.0]$ and $\Phi < 0.1$ for the cask lids, the additional bolt load of a few percent (relating to the pre-tension) results from the outside force $F_{A0,min}$. Taking into account the limitation of the assembly pre-load, given by Eq.(1), we can see that no plastic deformation capable of impairing the sealing function is expected under this outside force even with the maximum pre-tension in the bolt.

In the above estimation some effects –for example, lid deflection or additional bending tensions in the bolts- are not taken into account. Nevertheless, the conclusion remains valid that the sealing function impairment in the case of minimum pre-load occurs under lower outside forces than in the case of maximum pre-load. The case of a compressed lid flange (e.g. due to a 1 m

drop onto a bar) is one of the few counter-examples. In general, setting the pre-load in the model connections equal to the minimum values (for the defined tightening torques) creates the conservative initial state for the sealing function under accident conditions. Whether the estimation presented here is valid in an individual case, should always be checked additionally.
Calibration of the assembly state for the most severe conditions for the gasket is also preferential since computation of gasket behaviour is very problematical. On the other hand, the loading state of the bolts can be investigated reliably by analytical or numerical methods. Hence, the calculation for the bolts can be performed additionally to complete test results with information concerning the behaviour of the bolts in the pre-load situation, which is not directly simulated by the test.

Simulation of unfavourable conditions for the gasket under axial forces

If the axial outside force F_A acts on the lid, the following gasket loading conditions can be identified

1. $0 < F_A < F_{A0}$ up to the complete opening of the clamping interface on the lid flange;
2. $F_{A0} \le F_A < F_{A1}$ up to gasket decompression beyond the useful elastic recovery;
3. $F_A \ge F_{A1}$ gasket decompression beyond useful elastic recovery.

Due to the lid deflection and generally eccentric loading and clamping conditions of the lid, gasket decompression can also occur before the complete opening of the clamping interface. These effects are not considered here to keep the derivation from becoming too involved. Although the force range defined above will be slightly changed, the analysis presented retains its validity in principle.
The objective of the scale model test is a possibly exact, but at least conservative, simulation of these gasket loading phases. Some specific details will be discussed in the following:
For 1: The maximum outside total force on the lid results from the accident conditions (F_T) and pressure in the inside of the cask (F_p). The internal pressure is usually not modelled in drop tests. The residual clamp load under these forces is

$$F_{KR} = F_K - (1-\Phi)\cdot F_A = N_S \cdot F_V - F_D - (1-\Phi)\cdot(F_T + F_p)\,, \tag{12}$$

$$F_{KR,\lambda} = F_{K,\lambda} - (1-\Phi_\lambda)\cdot F_{A,\lambda} = n_S \cdot F_{V,\lambda} - F_{D,\lambda} - (1-\Phi_\lambda)\cdot F_{T,\lambda}\,, \tag{13}$$

with the number of bolts in the model $n_S \neq N_S$ for the general case.
Complete opening of clamping interfaces occurs if $F_{KR} = 0$ and $F_{KR,\lambda} = 0$. Conservative conditions must be attained for the gasket in the scale model test. If complete opening of the lid flange is possible under accident conditions, it must occur in the model at latest under the outside force which is in a similarity relation to the appropriate outside force in the original cask: $F_{T0,\lambda} \le F_{T0}/\lambda^2$. Thus,

$$F_{V,\lambda} \le \frac{1}{\lambda^2}\cdot\frac{1-\Phi_\lambda}{1-\Phi}\cdot\left(\frac{N_S}{n_S}\cdot F_V - \frac{F_D}{n_S} - (1-\Phi)\cdot\frac{F_p}{n_S}\right) + \frac{F_{D,\lambda}}{n_S} \tag{14}$$

The relative resilience factor Φ_λ for the scaled bolted joint deviates slightly from that for the original cask Φ because the geometrical scaling is not detailed. Since these factors for cask lid connections are normally relatively small (<0.1), $\frac{1-\Phi_\lambda}{1-\Phi} \approx 1$ can be assumed.
Tendencies in the pre-load, embedding effects in the connection as well as in the gasket compression force shown above permit the statement that the condition Eq.(14) is always fulfilled, if

$$F_{M,\lambda} \le \frac{1}{\lambda^2} \cdot \left(\frac{N_S}{n_S} \cdot (F_M - F_Z) - (1-\Phi) \cdot \frac{F_p}{n_S} \right) \quad \textbf{(15)}$$

If the same lubricant is used for model and original cask bolts and the theoretical relation between tightening torques is kept, a non-conservative effect will occur. The quantity of this effect depends on the proportion of the pre-tension in the original bolts to the pressure force and the pre-tension loss due to embedding.

For 2: This condition cannot be fulfilled due to a complex, nonlinear gasket characteristic. Since the gasket is not loaded directly by the outside force and its decompression equals the gap in the flange, it is sufficient to fulfil the condition Eq.(5). Gap formation in the lid flange almost keeps the similarity laws.

For 3: Changes in the leakage rate are not governed by similarity laws. Adequate safety factors always have to be included in the transfer of the test results regarding an increasing leakage rate to the original cask.

Simulation of unfavourable conditions for the gasket under transverse forces

Impairment of the gasket is only possible after a sliding of the lid under the transverse component of outside force. Outside forces overcoming the friction grip at the flange are

$$Q = \mu \cdot (N_S \cdot F_V - F_D - (1-\Phi) \cdot F_p) \quad \textbf{(16)}$$

$$Q_\lambda = \mu \cdot (n_S \cdot F_{V,\lambda} - F_{D,\lambda}) \quad \textbf{(17)}$$

where μ is the friction coefficient at the flange. The conservative condition $Q_\lambda \le Q / \lambda^2$ leads finally to the same relation as for axial forces - namely, to Eq.(15). If Eq. (15) is fulfilled, the sliding of the model lid, if possible, begins at latest under a similar lateral force as in the original cask. Thus, the same conclusion can be drawn for the transverse force as for the axial force. A reduction of the pre-load in the model bolted connections below the theoretical scaling value is necessary for a conservative simulation of the initial conditions for the gasket.

Practical realisation

If the pre-load F_M on the right sides of Eq.(15) is interpreted as the minimum assembly pre-load $F_{M,min}$, these equations will define the conservative assembly conditions for the gaskets in the scale model test. However, practical realization of this requirement with close control of the pre-load is very difficult due to the dispersion of the friction coefficients discussed above. Since lid displacements mainly depend on the total clamping load on the flange and the number of lid bolts is usually large, it is acceptable to formulate these conditions for the average pre-load. In this respect the following steps are necessary:

1. Average pre-load in each lid flange of the scale model has to be controlled by recording the pre-tension in the bolts equipped with strain gauges. Preferably the condition

$$F_{Mmit,\lambda} = \frac{1}{n_i} \sum_{i=1}^{n_i} F_i \le \frac{1}{\lambda^2} \left(\frac{N_S}{n_S} \cdot (F_{Mmit} - F_Z) - (1-\Phi) \cdot \frac{F_p}{n_S} \right) \quad \textbf{(18)}$$

is fulfilled. $F_{M,mit}$ is the average pre-load in the original bolts; F_i and n_i are the measured pre-tension and number of the model bolts with strain gauges.

2. During the drop test strains in the lids and bolts have to be measured. Changes in the lid positions and cask deformation in the lid region have to be registered after the test. This information combined with the closure system leakage rate before and after the drop test is the basis for estimating whether it is acceptable to transfer the test results for the sealing function to the original design or not.

3. It has to be demonstrated by additional computations that no problems will also occur in the closure system (e.g.: failure of the bolts under the maximum preload) under different assembly conditions from those as simulated in the test. In general, tests with scale models always have to be supported by additional calculations.
4. If the model closure system tightness meets the requirements after the drop test, but a non-negligible increase in leakage rate occurs, transferability of the test results to the original design will be difficult. Firstly, reasons for changes in the leakage rate have to be detected and necessary corrections to improve the sealing function then have to be defined (e.g. a change in the tightening torque or the sizes and number of bolts). If, for example, a change in the leakage rate along with a change in the lid position is detected after the test, higher pre-load in the flange by increasing of the tightening torques in the original cask could improve containment performance. Whether this step is possible or not depends on the stress level measured in the bolts. In some cases the solution can be found without repeating the test. However, this statement is not valid for cases with some plastic deformation in the closure region of the scale model (cask body, lids or lid bolts) after the drop test. In such cases design modification and additional drop tests are necessary.

CONCLUSIONS

The objective of the drop tests is, amongst others, demonstration of the containment performance of the cask. Conservative loading conditions for the different components of the cask bolted closure system cannot be created simultaneously by the drop tests and, therefore, a direct safety demonstration without additional analysis is usually not possible. When using scale cask models, the problem becomes more difficult due to deviations in an accurate geometrical and physical scaling of closure system components. In this case, it is important to ensure a “functional similarity” between the sealing functions of the test model and original cask. On the basis of a preliminary investigation of each lid, conditions for the assembly state should be specified which could be detrimental to the original closure system sealing function under the accident conditions. Similar, or at least conservative, assembly conditions in the model closure system must be provided by the drop test program for this failure mode. In this article some requirements for “functional modelling” in regard to the closure system of the transport and storage cask for radioactive materials have been discussed.

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