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Guideline on the Assessment of the Lid Systems and Load Attachment Systems of Transport Packages for Radioactive Materials

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This guideline is 36 pages long.

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1 Introduction

1.1 Use and content of the guideline

This is a guideline to the design of bolted lids and load attachment systems on containers for the transport of radioactive materials. It sets out requirements for load assumptions, the use of calculation methods and the assessment criteria. The guideline relates primarily to type B(U) packages specified in [20] but can also be applied to other containers used for the transport of radioactive materials.

In this guideline the isolated analysis of single structural components is extended by a system consideration, in order to include interactions between components. Based on the package (in the following also container), the closure system with several lid systems and the load attachment system are considered. Additional investigations are made for individual parts of these systems. This modelling hierarchy enables the consideration of many different kinds of interaction between subsystems of the package, between subsystems and their corresponding components, and between single components.

A load attachment system, for example, consists of a load attachment point (trunnion, gripping appliance of lid, etc.) and the corresponding bolted joint. A lid system comprises the actual lids (primary lid, secondary lid, small lids, etc.) and the corresponding bolted joints and seals.

The legal terms and requirements of dangerous goods transport regulations in respect of routine, normal and accident conditions of transport of packages for radioactive materials are taken as a basis for the design assessment of the lid systems. The criteria relevant to the assembly of the bolted joints are also explained in this guideline.

As regards the load attachment systems, the assembly state of the bolted joints as well as the requirements on the system components in terms of structural strength and fatigue strength are also covered in this guideline. The requirements consider the loading conditions on the load attachment systems, both during operation by crane and transport on public routes (routine conditions of transport [20]). A distinction is drawn between crane operations within and outside the scope of the safety regulations defined by the Nuclear Safety Standards Commission (KTA).

This guideline only covers the issues of general stress analysis and of fatigue strength of defect-free components. Additional evaluations may also be required for fracture behaviour, based, for example, on the *BAM-GGR 007* (*Guidelines for the Application of Ductile Cast Iron for Transport and Storage Casks for Radioactive Materials*) [4] or on other applicable regulations, such as the FKM Guideline on Fracture Mechanics Proof of Strength for Engineering Components [17].

The safety factors in this guideline assume a realistic or conservative modelling basis. The load assumptions, material properties and geometric simplifications has to be justified.

This guideline was compiled by Division 3.3 of the Bundesanstalt für Materialforschung und -prüfung (BAM) in the course of the earlier project on safety assessments and type testing of transport containers for radioactive materials (‘0207-3.32-0090: Sicherheitstechnische Begutachtung und Bauartprüfung von Transportbehältern für radioaktive Stoffe’) in order to summarise the main points regarding the design assessment of lid and load attachment systems on containers for the transport of radioactive materials. It is the responsibility of the applicant to check and if necessary to justify whether the stipulations in this guideline need to be supplemented or amended before they are applied. Alternative approaches can be used instead of the procedure set out in this guideline as long as compliance with the safety objectives of dangerous goods transport regulations can be guaranteed.

1.2 Legal basis

The applicable regulations for the transport of radioactive materials are based on the recommendations of the International Atomic Energy Agency (IAEA) [20]. The aforementioned recommendations have been transferred into binding national and international law by way of the dangerous goods regulations for the modes of transport of road, rail, sea and air.
The primary aims of the dangerous goods regulations in respect of the functions to be fulfilled by the design of containers for the transport of radioactive materials are as follows:

- Containment of the radioactive contents (both integrity and tightness),
- Shielding of ionising radiation from the content,
- Prevention of criticality of the content (in the case of fissile materials) and
- Protection from damage caused by heat.

Besides of shielding, the lid systems of the transport containers must also guarantee the safe containment under consideration of specific tightness requirements of the containers. Load attachment systems should guarantee the safe handling of the containers and their parts, e.g. lids, and, where applicable, the secure retention of the containers during transport.
The requirements for routine, normal and accident conditions of transport set out in the relevant regulations enable the calculation of mechanical and thermal stress or design loads to make a case for the safety of the corresponding systems or components.
Routine conditions of transport take into account loads on the container during incident free transport. Normal conditions of transport do take account of minor incidents. Accident conditions of transport cover serious accidents during transport. They are covered by a series of cumulative tests, such as a free drop from a height of 9 m onto an unyielding target and a drop from a height of 1 m onto a steel puncture bar followed by a half-hour fully engulfing fire at an average temperature of 800° C. Furthermore an immersion test in water 15 m deep for at least 8 hours (a depth of 200 m and at least 1 hour for type B(U) packages containing over $10^5 A_2$) shall be considered [20].
Approved assessment procedures in the dangerous goods regulations include experiments with prototypes or models, reference to earlier tests with containers of a similar design, tests with scale models, calculations or justified assumptions, as well as a combination of several of the options mentioned before [20]. This guideline refers to specific methods of numeric design assessment based on conservative assumptions or an experimental validation of the underlying model parameters.

1.3 Other standards and guidelines

Reference is made in this guideline to other relevant guidelines for various components used in containers for the transport of radioactive materials. VDI Guideline 2230 [28] sets the design standard for bolted joints, for example, while KTA 3905 [24] and ISO 10276 [22]

set out the industry standards for load attachment points in nuclear power plants and for trunnion systems for packages used to transport of radioactive materials.
However, the different levels of dangerous goods regulations (routine, normal and accident conditions of transport) require differentiated assessment criteria which are not covered in this form by existing guidelines and standards.
Moreover, the lid area and the load attachment systems need to be regarded as a whole, save for exceptional cases, in order to take account of the interaction between the components of these systems and potential interplay with other parts (e.g. interaction of lid with shock absorber). A numerical analysis is generally necessary here to enable a realistic evaluation of the lid and load attachment system.
The recommendations set out in the *Guidelines for Numerical Safety Verifications within the Scope of the Design Assessment of Transport and Storage Casks for Radioactive Materials (BAM-GGR 008)* [5] are relevant to numerical analyses in order to guarantee the appropriate application of numeric codes with certainty of results. The scope of this guideline includes the processing of these numerical analyses and the evaluation of the results with regard to the applicable rules and regulations.

2 Load attachment systems

The load attachment system (LAS) is used for the handling of the container and its parts and for the restraint of the container on the means of transport. Bolted LASs are the subject of this guideline.
The main component of an LAS is the load attachment point (LAP) which is defined in [24] as the connecting element between the load and the load suspension devices (cf. [13] for definitions of load and load suspension devices). Other components of an LAS include bolts for the fastening of the LAP and the corresponding nut threads or tapped blind holes. So, for example, a bolted trunnion system used as an LAS as defined by this guideline consists of the trunnion, the trunnion bolts and the part of the container body which has the tapped blind hole to fit the trunnion bolts. Other examples of LASs as defined by this guideline – other than the aforementioned bolted trunnion systems for the handling and, where required, for the retention of the container during transport – include gripping appliance for the safe handling of a lid or a basket.
This guideline includes an analysis of the loads to which the LASs are subjected during handling and transport on public routes. A distinction is drawn between crane operations within and outside the scope of KTA 3905 when analysing the stresses incurred during the handling. One example of crane operations exceeding the scope of KTA 3905 is the transhipping of the package during transport. Transport-induced stresses also occur when the LAPs are used to restrain the package on the means of transport.
The LAS design must therefore meet the statutory dangerous goods requirements relating to transhipping and transport-induced stresses on the one hand but also the requirements set out for the handling within the scope of KTA 3905 on the other hand. These overlapping requirements must be considered conjointly in the design of the LAS. Generally the proof of a correct assembly state, following by a general stress and a fatigue strength analysis must be provided for the LAS while due account must be taken of load assumptions according to both the dangerous good transport regulations and KTA 3905. Furthermore, the design of the LAS must exclude with sufficient safety the loosening of the bolted connections under operating loads.
The assessment procedure presented here is based on the evaluation of local stresses taken from a finite element (FE) analysis and is an extension of the nominal stress concept set out in KTA 3905.
If LAPs on lids are used for the handling of the container or for the retention of the container during transport, due consideration must also be given to the effects of the additional stresses on the lid bolts and the sealing system. The requirements in this regard are discussed in Section 3 of this guideline.

2.1 Calculation methods and modelling

The claimed parts can be reduced to the zones affected by the LAS. Areas which are evidently not affected by the LAS need not to be included in the modelling. The interaction between the bearing shell and trunnion is to be taken into account either by modelling the bearing

shell directly or by making sufficiently conservative assumptions in respect of the looping angle and the distribution of force over the contact surface.
The assembly conditions must be taken into consideration for the bolts. This analysis can be based on VDI 2230 [28].
In addition to the assessment of static strength (general stress analysis [24]) an evaluation of fatigue strength[1] is required for crane operations and routine conditions of transport.
The evaluation of fatigue strength can be performed analytically, preferably using the Palmgren-Miner linear cumulative damage hypothesis (2.1) [18].

$$D = \sum_i \frac{n_i}{N_i} \tag{2.1}$$

The individual stress collectives are approximated by step curves with frequency h_i and relative stress σ_{ai} so that holds $n_i = K \cdot h_i$ after K collective passes. The number of operating cycles N_i which can be endured in the collective step i is a function of the respective stress amplitude σ_{ai} and the respective mean stress σ_{mi} and is calculated with the help of the corresponding fatigue stress-number curve. Hence the total damage D of the component can be calculated.
If LAPs are operated within the scope of KTA 3905, the elementary Miner's rule must be applied (linear accumulation of damage at continuous stress number curve in a double logarithmic plot) [24]. If the critical cumulative damage Miner's sum is the same, the resulting evaluation applying the elementary Miner's rule is conservative [18]. This approach is also to be recommended if there are no mandatory KTA requirements to be met.

2.2 Design loads

2.2.1 Assembly state

VDI 2230[28] is the applicable guideline to calculate the pre-tension of bolts in LAS. The possible range of the pre-tension should be determined either by specifying an appropriate tightening factor in accordance with [28] or by taking the torque tolerance of the tightening method in conjunction with the scatter of the coefficients of friction for the used lubricant. The maximum tightening torque (nominal tightening torque plus the torque tolerance of the tightening method) coupled with the minimum coefficients of friction has to be taken to determine the maximum bolt pre-tension force. The minimum tightening torque (nominal tightening torque minus the torque tolerance of the tightening method) coupled with the maximum coefficients of friction has to be taken for the minimum bolt pre-tension force.[2]
The embedding effects in the bolted joint and the possible reduction in response to temperature changes must also be taken into account when defining the minimum pre-tension force. A potential increase in the maximum pre-tension depending on temperature must also be taken into account. The temperatures need to be checked against the requirements under dangerous goods regulations. The temperature can range from -40°C ([20], §639) to the highest conceivable operating temperature. The resulting pre-tension forces should be taken into account in the load assumptions below, e.g. in the FE analysis of the LAS. The

[1]According to [16], a distinction is drawn between endurance strength (fatigue limit) assessment and fatigue strength assessment depending on the load spectrum under service conditions. The term fatigue strength is used hereinafter because fatigue strength assessment for complex load collectives is needed in general for LAS.

[2]The calculation of the range of the bolt pre-tension force, illustrated here by taking the torque-controlled tightening as an example, is to be applied analogously if an alternative tightening method is used.

temperature-induced effects can be directly incorporated in this analysis if the modelling process allows (primarily in respect of the pre-tension force).

2.2.2 General stress analysis

A specification of the load, which includes the various transport and handling situations encountered by the container or the relevant component, is required to determine the relevant load for the general stress analysis. There are additional considerations in connection with crane operations in nuclear power stations, for example, such as the load induced by the flooding of the container (underwater loading). Different loads can be imposed, however, by changes in the weight of the container itself (handling operations without shock absorber, partial loading with radioactive material, etc.).
The load calculated in this way is to be multiplied by a live load coefficient in the case of crane operations. The live load coefficient depends on the classification of the handling area. Under KTA 3905, for example, specific live load coefficients may apply (increased and additional requirements) which include additional safety factors on the one hand and higher demands on crane systems on the other hand [24]. If the LAPs are not only used for crane operations but also for the attachment of package on the means of transport then it is also necessary to define a load coefficient for transport on public routes. The load coefficients are listed in Table 2.1.

Scope of application	*Load coefficient*
Increased requirements under KTA 3905 [24]	1,8
Additional requirements under KTA 3905 [24]	1,35
General requirements for crane operations[3]	1,45
Transport on public routes[4]	2,0

Table 2.1: Load coefficients

The live load coefficient of 1.45 for the general requirements for crane operations covers all cranes of classes H1 to H4[5] under DIN 15018[6], Part 1 [14].
The load coefficient to be taken into account for transport on public routes depends on the intended modes of transport (road, rail, sea or air). The load coefficients are generally defined in the longitudinal, transverse and vertical direction of the vehicle. The acceleration components relevant for the transport containers are given in [21] Appendix IV, Table IV.1. The calculations can be carried out also with an acceleration factor that encompasses the components and their combinations, for example with the factor of 2.0 as in Table 2.1. This acceleration is then considered to be acting in any direction. Other relevant national and international norms and standards include [7, 8, 22, 26, 29]. The applicant is required to give reasons for the load coefficient used in the design assessment. Arrangements may also need to be made which further qualify the mode of carriage (e.g. approved labelling).
The load is generally held by several LASs therefore due account must be taken of the distribution of the load on the individual LASs [24].

[3] Also includes crane transport during transport on public routes, e.g. transhipping.
[4] Includes loads imposed during transport on public routes. Does not include crane operations.
[5] For H4 the live load coefficient of 1.45 corresponds to a lifting speed of $5,5\,\mathrm{m/min}$ [14], Table 2, line 3. The live load coefficient needs to be adjusted for higher lifting speeds.
[6] The standard has been withdrawn and replaced by EN 13001-1 [9], EN 13001-2 [10] and EN 13001-3-1 [11]. The withdrawn standard is used in this guide to maintain consistency with KTA 3905 [24].

2.2.3 Fatigue strength analysis

Service conditions generally comprise several stress collectives. A stress collective is made up of the stress values and the number of corresponding stress cycles. Stress values of a collective step are maximum stress and minimum stress which fluctuate around a mean level of stress. A stress cycle is one cycle of the stress-time graph, e.g. from maximum stress and back up to maximum stress. The stress amplitude is the distance between maximum and mean stress or the distance between mean stress and minimum stress.

Crane operations

According to KTA 3905 [24], LAPs which do not fall within the scope of application of DIN 15018 [14] require fatigue strength analysis in the case of more than 6,000 stress cycles.[7] The number of stress cycles is calculated from the total number of stress cycles over the number of operational load cycles to be carried out ([24], Equation 5.1-1). An operational load cycle is the process between taking up and setting down of the load. The number of stress cycles during one operational load cycle according to [24] comes to 30 (for converter drives and cable drives with creep speed) or 60 (for other drives). Therefore 200 or 100 operational load cycles respectively are allowed without fatigue strength analysis.
If such analysis is required, a single-step collective can be drawn up according to [24] for one operational load cycle with the associated number of stress cycles. The maximum stress corresponds to the maximum value of the load after the coupling of the container and is calculated in consideration of the live load coefficients specified in [24] for the respective requirement. The minimum stress is equal to zero for the LAP. The minimum stress for the bolt is equal to the pre-tension.[8]
In accordance with [24], this guideline stipulates the need for fatigue strength analysis after more than 200 operational load cycles (for converter drives and cable drives with creep speed) or 100 operational load cycles (for other drives).[9] The use of a two-step collective is recommended in this case, however. This allows an uniform treatment for crane operations irrespective of the handling area and, where necessary, combination with transport load collectives. The first step (once per operational load cycle) is for taking up and setting down of the load. The other stress cycles represent the loading which is caused by oscillations during the load movement between taking up and setting down. For drives with creep speed 99 stress cycles must be considered at this step. For other drives this number increases to 199.
Table 2.2 shows the load coefficients to be used in combination with the load according to Section 2.2.2 of this guideline for the calculation of the stress collectives. The load collectives listed in Table 2.2 take conservative account of the results of internal tests conducted by BAM (e.g. [3]) and empirical data from past approval procedures. According to Section 5.1.3 (4) of [24], the use of such collectives is permissible within the scope of KTA 3905. The additional safety factors, by which the live load coefficients are to be multiplied in accordance with Section 5.3.1.1 (3) and 5.3.2.1 (3) of [24], are also reflected in Table 2.2. For increased

[7] The requirements set out in Section 5.1.2 (4) of [24] additionally apply to the bolts which are retightened after disassembly.

[8] These stress collectives correspond to the single-step collective method recommended in [24] following conservative deduction of single-step collectives (stress in combination with the corresponding number of cycles) from the loads typically applied in scope of KTA 3905, taking due account of the damage equivalence. They do not constitute real load conditions. Alternatively, the assessment is also possible in [24] using real load conditions.

[9] The requirements set out in section 5.1.2 (4) of [24] additionally apply to the bolts which are retightened after disassembly[24]. Cf.[7]

Collective step	I	II
Number of stress cycles	1	99 resp. 199
Scope of application	*Load coefficient*	
Increased requirements under KTA 3905 [24]	$0 \ldots 1,45$	$0,55 \ldots 1,45$
Additional requirements under KTA 3905 [24]	$0 \ldots 1,35$	$0,65 \ldots 1,35$
General requirements for crane operations	$0 \ldots 1,45$	$0,55 \ldots 1,45$

Table 2.2: Load coefficients for an operating cycle during crane operations

requirements in accordance with Section 4.3 of [24], an additional redundancy factor of 1.25 is required on the load side in case the LAS fails and there is no redundant part available to support the load. This is the case, for example, when handling a container by two trunnions on the lid side. When applying the load coefficients listed in Table 2.2, the redundancy factor may need to be taken into account as an additional safety factor when evaluating the stress.

Transport on public routes

It is not possible to define universally valid load collectives for a transport on public routes therefore these must be specified both on the basis of the requested modes of transport (road, rail, sea or air) and on the basis of the length and number of anticipated transport cycles. Corresponding fixings are to be included in the instructions for use of the package. New or additional fatigue strength analysis will be required if transports are carried out under conditions which are not covered by the safety analysis.
In addition to experimental determination of the transport collectives, reference may also be made to published measurements [6, 15, 25, 26]. The transfer to other packages or transport routes and the allowance for errors of measurement may necessitate the use of additional safety factors in the fatigue strength analysis.

2.3 Material properties

The information and advices in [24, 22] should be heeded when selecting the materials for the components of the LAS. The material properties set out in the material specifications of the safety report should basically be incorporated in the design assessment. These material properties are to be verified by way of a material evaluation or by way of recourse to the minimum values specified in the applicable standards. Some of the material properties must also be verified in the quality control process during production.
A distinction is to be drawn between material properties at room temperature T_0 and at design temperature. The stipulations set out in § 639 [20] and the results of the thermal analysis of the container must be heeded when specifying the temperature range relevant to the design.
The material properties at room temperature (e.g. $R_{\mathrm{p0.2}}(T_0)$) can be consulted when taking the assembly state into consideration. Under operating conditions, it is a generally conservative approach to use the maximum operating temperature calculated in the thermal analysis $T_{\max}$ as a basis, e. g. $R_{\mathrm{p0.2}}(T_{\max})$.

2.3.1 Mechanical properties

Modulus of elasticity, yield point and tensile strength

A realistic material model should be used to calculate the effective stresses, especially for FE analyses. A key element of any such model is the modulus of elasticity $E(T)$ which is required for all the relevant parts. Appropriate proof must also be furnished of other material properties which are used to calculate the effective stresses.

The yield point[10] at maximum operating temperature $R_{\mathrm{p0.2}}(T)$ is essential for the assessment of both the LAP and the bolts. It must be specified for both parts. The tensile strengths of the bolt material $R_{\mathrm{mB}}(T_{\mathrm{max}})$ and of the bolted parts (nut or blind hole) $R_{\mathrm{mM}}(T_{\mathrm{max}})$ are primarily needed to determine sufficient length of thread engagement.

The temperature-dependent change in pre-tension force is calculated in VDI 2230 [28] based on the pre-tension force at room temperature. In addition to the moduli of elasticity at room temperature $E(T_0)$ and at maximum operating temperature $E(T_{\mathrm{max}})$, the modulus of elasticity at minimum design temperature is also needed and taken into account according to the requirements for the package design.

Limiting surface pressure

The limiting surface pressure p_{G} which is needed to evaluate the surface pressure in the bolted joint can be calculated on the basis of VDI 2230, Table A9 [28] for the sake of simplicity if there is an absence of values which are more suitable, and can be substantiated by literature references or experimental data. There must also be sufficient verification of the admissible values for the maximum surface pressure between trunnions and bearing shells (for the load conditions during crane operations and transport).

Parameters for fatigue strength analysis

It is generally very complex to define the stress-number curves required for the fatigue strength analysis of the component in due consideration of the material and the notch cases. Therefore synthetically generated stress-number curves can also be used in the design assessment. The synthetically generated stress-number curves suggested in various guidelines are not interchangeable, however, as they are often linked to the respective calculation process and, most importantly, to the safety factors used in that approach. The synthetically generated stress-number curves recommended in this guideline are explicitly named in connection with the relevant analysis. The synthetically generated stress-number curves for the fatigue strength analysis of the trunnions and the trunnion bolts are taken from the FKM Guideline [16] and from VDI 2230 [28] respectively.

2.3.2 Thermal properties

In the particular case of containers with contents which generate appreciable amounts of heat, the coefficient of thermal expansion α_{T} is needed for the individual parts of the system in any given case for the calculation of additional deformation and stress as well as for the calculation of the temperature-dependent changes in the pre-tension. The coefficient of thermal expansion α_{T} in turn is to be suitably verified, again depending on temperature.

[10] No distinction is drawn in this guideline between the yield point R_{e} and the 0.2% proof stress $R_{\mathrm{p0.2}}$. The proof stress typically describes materials which have no definitive yield point and is used as a substitute for the yield point in the material specification.

2.3.3 Tribological properties

Minimum and maximum coefficients of friction for the corresponding material combinations and lubricants should be taken into account to determine the range of the pre-tension. A distinction is to be drawn between the coefficients of friction under the bolt head (μ_{Kmin} and μ_{Kmax}) and the coefficients of friction in the thread (μ_{Gmin} and μ_{Gmax}). The coefficients of friction should preferably be determined by experiment with the necessary guarantee that the test results can be transferred to joints differing from the tested configuration. Sufficiently verified published values may also be used if to do so is to guarantee the conservative nature of the approach [2]. The same goes for the coefficient of friction at the interface between the claimed parts which is required to verify the safety of the LAP and prevention of its lateral shifting.

2.4 Determination and assessment of stress conditions

2.4.1 General strength of load attachment points

Determination of effective stress

The general stress analysis of the LAP has to be carried out on basis of local stresses if necessitated by the complexity of the load application, by the geometry or by the interactions which need to be factored in (e.g. for trunnions). A consideration of nominal stress is not sufficient in this case due to the fact that the underlying hypotheses are no longer valid. The local distribution of the effective stress in the LAP is then to be calculated by means of FE analyses or other suitable methods of calculation.
The stress evaluation is to be based on the equivalent stress according to the maximum distortion energy theory (von Mises equivalent stress) at the point of maximum stress.

Stress evaluation

The equivalent stress is evaluated on the basis of the nominal stress concept set out in KTA 3905 [24] and in due consideration of the requirements according to [12]. If Equation (2.2) is satisfied for the maximum notch stress (or for the maximum nominal stress in nominal stress approach), the general stress analysis is completed [24].

$$\sigma_{\mathrm{v}} \leq \frac{R_{\mathrm{p0,2}}(T_{\mathrm{max}})}{1,5} \tag{2.2}$$

If the maximum notch stress exceeds $R_{\mathrm{p0.2}}(T_{\mathrm{max}})/1.5$ in the local stress approach but is still below $R_{\mathrm{p0.2}}(T_{\mathrm{max}})$, the limit load analysis can also be carried out with the load inflated 2.25 times over. The load coefficients listed in Table 2.3 must be used in this case. The load coefficient specified there for the additional requirements of KTA 3905 corresponds to the requirements in [12] whereby load handling equipment which is subjected to service weight with a coefficient of 3.0 must not be fully plasticised in the stressed cross section.[11]

[11] Cf. [12], section 5.1.1.1, sentence (1)

Scope of application	*Load coefficient*[12]
Increased requirements under KTA 3905 [24]	$4,0$
Additional requirements under KTA 3905 [24]	$3,0$
General requirements for crane operations	$3,25$
Transport on public routes	$4,5$

Table 2.3: Load coefficients for the limit analysis

The limit analysis for the LAP must be based on localised strains. A perfectly elastic-plastic material model with $R_{p0.2}(T_{max})$ for the yield point must be taken as a basis for the LAS including the bolts. The criterion which must be met to comply with the safety requirements is that there must be no full plasticisation of the cross section relevant for the load-bearing capacity in that there must be at least one area in which $\epsilon_{pl} = 0$ applies through the cross section of the LAP subject to the greatest stress. ϵ_{pl} is the equivalent plastic strain. The bolts are not evaluated in the limit load analysis. They must satisfy the requirements set out in Section 2.4.4, Table 2.5.

2.4.2 Surface pressure between trunnions and bearing shell

Service loads are generally transferred to the trunnions over bearing shells. There is usually a layer of relatively soft metal, e.g. a copper alloy, between the bearing shell and the trunnion. This guideline is only concerned with adherence to the limiting surface pressure $p_G(T_{max})$ at the trunnion. The surface pressure can be calculated either by way of conservative analyses or by taking a numerical approach. It is necessary to ensure that the contact pressure at the trunnion is properly calculated if modelling the interaction between the bearing shell and trunnion directly or if using alternative mathematical models. For example, if using a FE model, a hardening material with an upper yield point should be selected for the bearing shell model. The effective contact pressure p_{max} must be below the limiting interface pressure.

2.4.3 Fatigue strength of load attachment points

Calculation of effective stress

The effective stress evaluations can be taken from the analyses for the different load assumptions set out in Section 2.2.3. The calculation of the point of highest stress must take due consideration of all assumed loads.
When designing the LAS with the aid of an FE analysis, the resulting stresses are often not proportional to the load carried in any given case due to the inherent non-linearities in the model (contact conditions). The results of the calculation for various load coefficients are therefore generally not accessible through linear interpolation of the results of a FE analysis.

Evaluation of the effective operating stress

The effective operating stress should be evaluated according to the FKM Guideline [16] with reference to the synthetically generated stress-number curves, safety factors and critical Miner's sums recommended therein.
It is very time-consuming to conduct tests and experiments on parts to work out stress-number curves therefore the FKM Guideline sanctions the use of synthetically generated

[12] 2.25 times the values in Table 2.1.

Scope of application	*Safety factor*
Increased requirements under [24]	2,1 (2,6) [13]
Additional requirements under KTA 3905 [24]	1,7
General requirements for crane operations	1,35
Transport on public routes	1,35

Table 2.4: Safety factors for the fatigue strength analysis of the LAP

stress-number curves (cf. also Section 2.3.1). Number of cycles N_D and inclination exponent k for the construction of synthetically generated stress-number curves can be found in Table 4.4.3 in the FKM Guideline [16].
The safety factor j_erf required for the final setting of the synthetically generated stress-number curve is based on the factors of 2.0 for additional requirements and 2.5 for increased requirements under KTA 3905 [24]. However, these safety factors are based on the survival probability of 50 % as compared to 97.5 % for the values in the FKM Guideline. Therefore the safety factors proposed in KTA 3905 can be reduced if calculating according to the FKM Guideline. Assuming a mean logarithmic standard deviation of $\sigma_\mathrm{lgS} = 0.04$, Table 5.6.1 in the FKM Guideline [16] would give a statistical conversion factor of 1.2 therefore the attachment points safety factors can be adjusted with the Equation (2.3).

$$j_\mathrm{erf} = \frac{j_\mathrm{KTA}}{1,2} \tag{2.3}$$

The safety factors arrived at in this way from Table 2.4 fall within the scope of KTA 3905. If the redundancy factor for increased requirements under KTA 3905 needs to be taken into account then a safety factor of $2.1 \cdot 1.25 = 2.6$ must be applied for non-redundant LAPs. The safety factor in Table 4.5.1 in the FKM Guideline [16] should be selected in case of regular inspection and harmful consequences for crane operations outside the scope of KTA 3905 and for transport on public routes. If the LAP is subject to combined operational demands, e.g. from crane operations within the scope of KTA 3905 and from the transport-induced load, the higher of the corresponding safety factors is to be taken as a basis for the design.
The Miner's sum D_M needed for the actual fatigue strength analysis varies depending on the production process and material group and must be selected as required by the FKM Guideline [16], Table 4.4.4.

2.4.4 General strength of bolts in load attachment systems

Calculation of effective stress

VDI 2230 [28] is used to calculate the tensile stress $\sigma_\mathrm{z,Mon}$ and the torsional stress $\tau_\mathrm{G,Mon}$ during the assembly of the connection. The equivalent stress required to assess the strength of the bolts in the assembled state is formed according to Equation (2.4)

$$\sigma_\mathrm{v} = \sqrt{\sigma^2_\mathrm{z,Mon} + 3\tau^2_\mathrm{G,Mon}} \tag{2.4}$$

.

For crane operations and routine conditions of transport, the effective tensile stress and bending stress over the screw axis are calculated in the respective cross sections with the

[13] Value in brackets in case there is a need for a redundancy factor.

stress distribution σ obtained in the FE analysis. To do this, the contributions of the axial force N and of the bending moment M_b are calculated first through integration of the stress distribution σ_n over the corresponding intersection or through summation over the nodal forces[14] of the FE model, as shown in Equations (2.5) to (2.8),

$$\mathbf{N} = \iint_{\mathcal{A}} \sigma_\mathrm{n}\,\mathrm{d}\mathbf{A} \qquad N = |\mathbf{N}| = \sum_i F_{1,i} \tag{2.5}$$

$$\mathbf{M}_\mathrm{b} = \iint_{\mathcal{A}} \mathbf{r} \times \sigma_\mathrm{n}\,\mathrm{d}\mathbf{A} \qquad M_\mathrm{b} = |\mathbf{M}_\mathrm{b}| = \sqrt{M_2^2 + M_3^2} \tag{2.6}$$

$$M_2 = \sum_i r_{3,i}\,F_{1,i} \tag{2.7}$$

$$M_3 = -\sum_i r_{2,i}\,F_{1,i} \tag{2.8}$$

where $\mathrm{d}\mathbf{A} = \mathbf{n}\,\mathrm{d}A$ stands for the surface element and σ_n for the normal stress. $F_{1,i}$ is the axial force on a discrete node i and $\mathbf{r} = [0, r_{2,i}, r_{3,i}]$ is the distance vector of the node in a Cartesian coordinate system positioned arbitrarily in the respective cross-sectional area. The coordinate origin is on the screw axis and the coordinate x_1 is pointing in the direction of the screw axis. The setting is illustrated in Fig. 2.1. The mesh of the accordingly modelled bolt should be regular. Having determined the normal force N and the bending moment

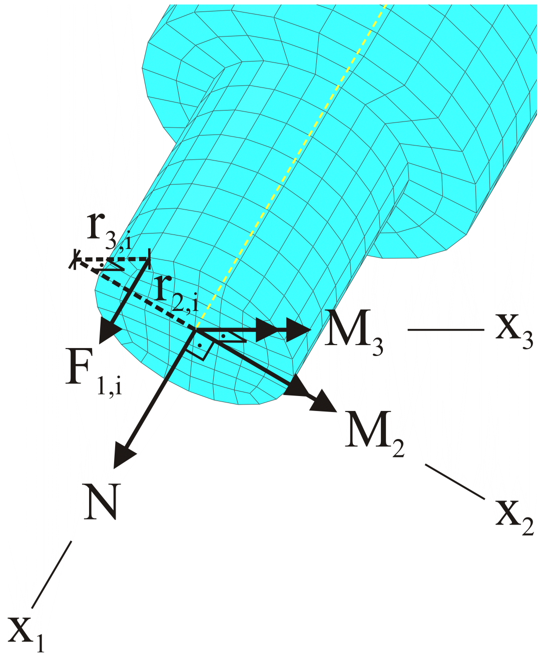

Figure 2.1: Calculation of normal force and moments of force on bolt

M_b based on Equations (2.5) to (2.8), it is now possible to determine the tensile stress and bending stress σ_z and σ_b as per Equation (2.9) with the help of cross section A and section modulus W. These stresses have the character of nominal stresses.

$$\sigma_\mathrm{z} = \frac{N}{A} \qquad \sigma_\mathrm{b} = \frac{M_\mathrm{b}}{W} \tag{2.9}$$

[14] The summation of the nodal forces corresponds to the integration of the stresses and constitutes the approach which is the easiest to adopt in practice.

The effective stress for LAS bolts in operating loads are calculated according to equation (2.10) with the reduction coefficient $k_\tau = 0,5$ [28].

$$\sigma_\mathrm{v} = \sqrt{(\sigma_\mathrm{z} + \sigma_\mathrm{b})^2 + 3(k_\tau\, \tau_\mathrm{G,Mon})^2} \tag{2.10}$$

The FE modelling of the bolts is generally greatly idealised (e.g. simplified modelling of the thread) therefore these nominal stresses are taken for the following evaluation instead of the local stress results from FE analyses. This approach allows an evaluation based on the criteria set out in KTA 3905 [24] and in VDI 2230 [28].

Stress evaluation

The criteria for the assembly state and the operating conditions are summarised in Table 2.5.

	Allowable stress	
Scope of application	*Assembly*	*Operation*
Increased requirements under KTA 3905[24]	$\sigma_\mathrm{v} \leq 0,7\, R_\mathrm{p0,2}(T_0)$	$\sigma_\mathrm{v} \leq R_\mathrm{p0,2}(T_\mathrm{max})$
Additional requirements under KTA 3905[24]	$\sigma_\mathrm{v} \leq 0,7\, R_\mathrm{p0,2}(T_0)$	$\sigma_\mathrm{v} \leq R_\mathrm{p0,2}(T_\mathrm{max})$
General requirements for crane operations	$\sigma_\mathrm{v} \leq 0,9\, R_\mathrm{p0,2}(T_0)$	$\sigma_\mathrm{v} \leq R_\mathrm{p0,2}(T_\mathrm{max})$
Transport on public routes	$\sigma_\mathrm{v} \leq 0,9\, R_\mathrm{p0,2}(T_0)$	$\sigma_\mathrm{v} \leq R_\mathrm{p0,2}(T_\mathrm{max})$

Table 2.5: Criteria for the stress evaluation of the bolts of load attachment systems

Outside the scope of KTA 3905, the stipulation in VDI 2230 [28] can be applied regarding the assembly conditions. Within the scope of KTA 3905, due account must also be taken of Equation (2.11) under the handling loads [24].

$$\sigma_\mathrm{z} - \sigma_\mathrm{z,Mon} \leq 0,1\, R_\mathrm{p0,2}(T_\mathrm{max}) \tag{2.11}$$

If a limit analysis is required for the LAP pursuant to section 2.4.1, the bolts are modelled with a perfectly elastic-plastic material law in order to ensure a realistic load distribution. $R_\mathrm{p0,2}(T_\mathrm{max})$ is to be used as the upper yield point. The limit load analysis is only needed for additional validation of the design of the LAP. The bolts are not evaluated in this analysis.

2.4.5 Surface pressure in bolted joints

Calculation of effective surface pressure

Another method which goes further than VDI 2230 [28] is applied to calculate the effective surface pressure because of the combined bending stress on the bolts. Based on the calculated stresses σ_z and σ_b derived from the evaluation of the analyses (covered in Section 2.4.4), the effective surface pressure is calculated using Equation (2.12).[15]

$$p_\mathrm{max} = \sigma_\mathrm{z}\, \frac{A}{A_\mathrm{K}} + \sigma_\mathrm{b}\, \frac{W}{W_\mathrm{K}} \tag{2.12}$$

The stress σ_z multiplied by cross section A of the bolt gives the axial bolt force, and the bending stress σ_b multiplied by the section modulus W of the bolt gives the bending moment. The surface pressure is from division taking the cross section A_K relevant for the surface pressure and the section modulus W_K which constitutes the connecting surface.

[15] A conservative estimate can also be made of the surface pressure with $p_\mathrm{max} = \frac{(\sigma_\mathrm{z}+\sigma_\mathrm{b})\,A}{A_\mathrm{K}}$.

Evaluation of surface pressure

The surface pressure is appraised using Equation (2.13). Reference may be made to Section 2.3.1 for the calculation of the limiting surrface pressure $p_G(T_{max})$.

$$p_{max} \leq p_G(T_{max}) \tag{2.13}$$

2.4.6 Length of thread engagement

Calculating the required length of engagement

A bolted joint should be designed in such a way that any failure through being subjected to excessive load occurs by way of a fracture in the free loaded thread or in the shank and does not incur any stripping of the thread where the bolt engages with the nut or tapped blind hole. To this end, the required length of thread engagement l_{erf} must be calculated during the design process by systematically matching the load-bearing capacities of the individual areas of the bolt and nut. The methods of calculation can be looked up in VDI 2230 [28], for example, or KTA 3201.2 [23].The required length of thread engagement can also be defined based on sufficiently representative test results.

Assessment of the length of engagement

Equation (2.14) is applicable for the assessment of the length of thread engagement l_{Gew} calculated from the specifications on the drawings. The thread countersinks must be taken into account when calculating l_{erf} or when determining l_{Gew} depending on the method of calculation.

$$l_{Gew} \geq l_{erf} \tag{2.14}$$

When verifying the load-bearing capacities, it is necessary to show that the lowest bearing strength is in the free loaded thread or in the shank.

2.4.7 Fatigue strength of bolts in load attachment systems

Calculation of effective stress

The effective stress evaluations can be taken from the FE analyses for the different load assumptions set out in Section 2.2.3. The stresses in the bolts linearised over the cross sections as shown in Section 2.4.4 are to be taken into account.
It may also be necessary to factor in the stress imposed on the bolts during assembly, disassembly and retightening [24].

Evaluation of the effective operating stress

The fatigue strength analysis of the bolts in LASs can be based on the linear cumulative damage hypothesis (2.1) in conjunction with VDI 2230 [28].
The synthetically generated stress-number curve needs to be calculated in this process based on VDI 2230 [28]. The number of cycles N_D specified therein is to be used for this, and the stress amplitude of the endurance limit σ_{ASG} and σ_{ASV} for bolts rolled after and before heat treatment respectively is to be calculated accordingly.
The required safety factor S_D must be selected for the relevant scope of application and is listed in Table 2.6. If the redundancy factor for increased requirements under KTA 3905 [24]

needs to be taken into account, a safety factor of $2.5 \cdot 1.25 = 3.1$ must be applied to the bolts on non-redundant LASs. If the LAS bolts are subject to combined operational demands, e.g. from crane operations within the scope of KTA 3905 and from the transport-induced load, the higher of the corresponding safety factors is to be taken as a basis for the design.

Scope of application	*Safety factor*
Increased requirements under KTA 3905 [24]	2,5 (3,1) [16]
Additional requirements under KTA 3905 [24]	2,0
General requirements for crane operations	1,5 [17]
Transport on public routes	1,5

Table 2.6: Safety factors for fatigue strength analysis of the bolts of load attachment systems

Adequate reasons are to be given for the critical Miner's sum which is to be applied to the evaluation in any given case.

[16]Value in brackets in case there is a need for a reliable redundancy factor.

[17]According to VDI 2230 [28], element (R9/4) has to be established by the user.In this guideline a safety factor of 1.5 is allowed for the design assessment outside the scope of KTA 3905.

3 Lid systems

Containers for the transport of radioactive materials are generally closed by lids with metal or elastomer seals. So, for example, the closure system of a container for spent fuel intended for long-term interim storage in Germany is designed as a double barrier system consisting of a primary lid, a secondary lid, small lids which close openings in the primary and secondary lids, bolts with tapped blind holes, and metal and elastomer seals. The individual lids and their components (bolts, tapped blind holes and seals) each form a lid system.
The closure system is primarily required to perform a sealing function. As a general rule, therefore, all its components contribute to the tightness of the package. In a double lid system, the sealing function is guaranteed by the primary lid system and also alternatively or additionally by the secondary lid system, depending on the design. The primary lid in particular also has a protective shielding function. The small lids integrated in the primary lid provide access to the inside of the container during the handling and service after loading. The secondary lid and the small lids integrated in it generally form the part of the system which monitors the tightness of the containers while they are in long-term storage in interim storage facilities.

3.1 Calculation methods and modelling

In order to enable an assessment of the entire closure system as realistic as possible under routine, normal and accident conditions of transport, the aim should be to conduct a numerical strength analysis, preferably with the FE method, even after tests. This approach makes it possible to simulate the interaction between the individual parts of the closure system with sufficient accuracy. Quasi-static FE analyses of the impact loading on the closure system under normal and accident conditions of transport can also be carried out if it can be proven that dynamic effects only have a negligible influence on the mechanical loads to which the parts are subjected or are covered by static assumptions.
However, should analytical methods of calculation be adopted in exceptional cases, it must be possible to guarantee that the aforementioned interaction is adequately taken into account and that especially the superposed bending stress on the bolts is also negligible. In this case the system need not be regarded as a whole. The parts then need to be examined in isolation. The assembly state also needs to be taken into consideration for the bolts in particular as the basis of the safety assessment for routine, normal and accident conditions of transport. The analysis of assembly state can be carried out using analytical methods, preferably by means of VDI 2230 [28].
A FE model of a lid system – either separate or integrated in the model of the entire closure system – should contain appropriate individual models of the lid itself, the lid bolts and the base body[1], to which the lid is bolted. If the stress and strain imposed on the respective base have no influence on the stress inflicted on the components of the lid system and if other interactions are negligible then the lid systems or even individual components can be

[1] In the case of the primary and secondary lid this is the container body, and in the case of small lids it is the primary resp. secondary lid.

modelled separately. In this case, the bolted parts can be reduced to the parts affected by the lid system. Areas which are irrelevant to the calculation of the stress on the lid system need not be modelled. The analysis can be limited to a section (e.g. sector of a circle) if the shape and symmetry of the lid and the load permit.
Further appropriate FE models are to be generated to study the effects of a non-uniform distribution of operating temperature on the integrity and sealing function of the closure system and to analyse the closure system during and after the heat test.

3.2 Design loads

3.2.1 Assembly state

VDI 2230 [28] is the recommended reference for the calculation of the required bolt pre-tension for the lid bolts. The possible range of the pre-tension should be determined either by specifying an appropriate tightening factor in accordance with [28] or by taking the torque tolerance of the tightening method in conjunction with the scatter of the coefficients of friction for the lubricant used. The maximum tightening torque (nominal tightening torque plus the torque tolerance of the tightening method) coupled with the minimum coefficients of friction is to be taken to determine the maximum bolt pre-tension force. The minimum tightening torque (nominal tightening torque minus the torque tolerance of the tightening method) coupled with the maximum coefficients of friction is to be taken for the minimum bolt pre-tension force.[2]
The embedding effects in the bolt joints and the possible reduction in response to temperature changes must also be taken into account when defining the minimum pre-tension force. A potential increase in the maximum pre-tension depending on temperature must also be taken into account. The temperatures need to be checked against the requirements under dangerous goods regulations. The temperature can range from -40°C ([20], §639) to the highest conceivable operating temperature. The resulting pre-tension forces should be taken into account in the load assumptions below, e.g. in the FE analysis of the lid system. The temperature-induced effects can be directly incorporated in this analysis if the modelling process allows (primarily in respect of the pre-tension force).

3.2.2 Conditions of transport under dangerous goods regulations

An analysis of the closure system for routine, normal and accident conditions of transport should take account in particular of the sealing forces, the bolt pre-tension in consideration of its change after assembly (embedding effects and effects of heat), the internal and external pressure (immersion test) and, in the case of a quasi-static analysis, the inertial forces impacting on the closure system. When analysing a lid system, the inertial forces come about as a result of the mass of the respective lid and the mass of the content if a direct interaction with the lid is possible. Additional loads can occur through interaction between lid systems or with other parts of the package. Particular account must be taken of the interaction between the outer parts of the containment (lid or secondary lid if the closure system is a double barrier design) and the shock absorber which can lead to appreciable stress on the components of the relevant lid system. In the case of a dynamic analysis, the modelling must guarantee the adequate simulation of the aforementioned interactions.

[2] The calculation of the range of the bolt pre-tension force, illustrated here by taking the torque-controlled tightening as an example, is to be applied analogously if an alternative tightening method is used.

When analysing the thermal effects, it is necessary to factor into the load assumptions either non-uniform stationary distribution of the operating temperature or non-stationary temperature fields as a result of the thermal test, depending on the case.
If the LAPs for handling of the container are fitted on lids, e.g. eye bolts, the additional stress imposed as a result of the crane operations must be taken into account in the design assessment.

3.3 Material properties

The material properties set out in the material specifications of the safety report should basically be incorporated in the design assessment. These material properties are to be verified by way of a material evaluation or by way of recourse to the minimum values specified in the applicable standards. Some of the material properties must also be verified in the quality control process during production.
A distinction is to be drawn between material properties at room temperature T_0 and at design temperature. The stipulations set out in § 639 [20] and the results of the thermal analysis of the container must be heeded when specifying the temperature range relevant to the design.
The material properties at room temperature (e.g. $R_{\mathrm{p0.2}}(T_0)$) can be consulted when taking the assembly state into consideration. Under operating conditions, it is a generally conservative approach to use the maximum operating temperature calculated in the thermal analysis T_{max} as a basis, e.g. $R_{\mathrm{p0.2}}(T_{\mathrm{max}})$.

3.3.1 Mechanical properties

Modulus of elasticity, yield point and tensile strength

A realistic material model should be used to calculate the effective stresses, especially for FE analyses. A key element of any such model is the modulus of elasticity $E(T)$ which is required for all the relevant parts. Appropriate proof must also be furnished of other material properties which are used to calculate the effective stresses.
The yield point at maximum operating temperature $R_{\mathrm{p0.2}}(T_{\mathrm{max}})$ is essential for the assessment of both the lid and the bolts. It must be specified for both parts. The tensile strengths of the respective bolt material $R_{\mathrm{mB}}(T_{\mathrm{max}})$ and of the bolted parts (nut or tapped blind holes) $R_{\mathrm{mM}}(T_{\mathrm{max}})$ are primarily needed to determine a sufficient length of thread engagement.
The temperature-dependent change in pre-tension force is calculated in VDI 2230 [28] based on the pre-tension force at room temperature. In addition to the moduli of elasticity at room temperature $E(T_0)$ and at maximum operating temperature $E(T_{\mathrm{max}})$, the modulus of elasticity at minimum design temperature is also needed and taken into account according to the requirements for the package design [20].

Limiting surface pressure

The limiting surface pressure p_{G} which is needed to evaluate the surface pressure in the bolted joint can be calculated on the basis of VDI 2230, Table A9 [28] for the sake of simplicity if there is an absence of values which are more suitable, and can be substantiated by literature references or experimental data.

3.3.2 Thermal properties

In the particular case of containers with contents which generate heat and for the calculation of the stress imposed in the thermal test, the coefficient of thermal expansion α_T is needed for the individual parts of the system in any given case for the necessary calculation of additional deformation and stress and for the calculation of the temperature-dependent changes in the pre-tension. The coefficient of thermal expansion in turn is to be suitably verified, again depending on temperature.

3.3.3 Tribological properties

Minimum and maximum coefficients of friction for the corresponding material combinations and lubricants should be taken into account to determine the range of the pre-tension. A distinction is to be drawn between the coefficients of friction under the bolt head (μ_Kmin and μ_Kmax) and the coefficients of friction in the thread (μ_Gmin and μ_Gmax). The coefficients of friction should preferably be determined by experiment with the necessary guarantee that the test results can be transferred to joints differing from the tested configuration. Sufficiently verified published values may also be used if to do so is to guarantee the conservative nature of the approach [2].
This paragraph also applies to the coefficient of friction μ_Dmin in the interface between the claimed parts which is required to verify the safety of the lids and the prevention of their lateral shifting. If computational models are generated to look at the interaction of the closure system with other parts of the container (e.g. lids with shock absorber), the coefficients of friction chosen are to be adequately substantiated in the contact definitions of the FE models.

3.3.4 Metal seals

The relevant sealing diagram is the basis for the calculation of the deformation of the seal after the assembly of the lids, under routine conditions of transport and after the tests required for normal and accident conditions of transport [20]. In the case of metal seal (metal jacket with spring core), the characteristic curve differentiates between a compression and a decompression [27]. This cycle, plotted to show the compression force related to the length of the seal over the deformation, is represented by the black line in Fig. 3.1 in the left vertical axis. The right vertical axis shows the leakage rate corresponding to the degree of deformation and represented by the blue line
During the compression, the seal's leakage rate falls below the standard helium leakage rate specified as the sealing criterion at the force Y_0 and the deformation e_0. There is a risk that the seal will damage above the critical deformation e_c. The chosen operating point of the seal is between these two limits at the force Y_2 and the deformation e_2.
In the decompression the sealing function (not exceed of the specified standard helium leakage rate) is maintained above the force Y_1 and the deformation e_1. A fall below the force Y_1 causes the specified leakage rate to be exceeded. The applicable specified standard helium leakage rate is the rate which is reached taking account of the quality criteria specified by manufacturers for seals and sealing surfaces. It is generally $10^{-8}\,\mathrm{Pa\,m^3/s}$ [27] for the above seal type. The optimum operating point of the seal which is relevant for the design is at point (e_2, Y_2). If using metal seals of other types, their specific characteristics are to be taken as the basis for the design.
Starting from the characteristic curve (Fig. 3.1), the resilience value (useful elastic recovery) r_u is defined (3.1).

$$r_\mathrm{u} = e_2 - e_1 \tag{3.1}$$

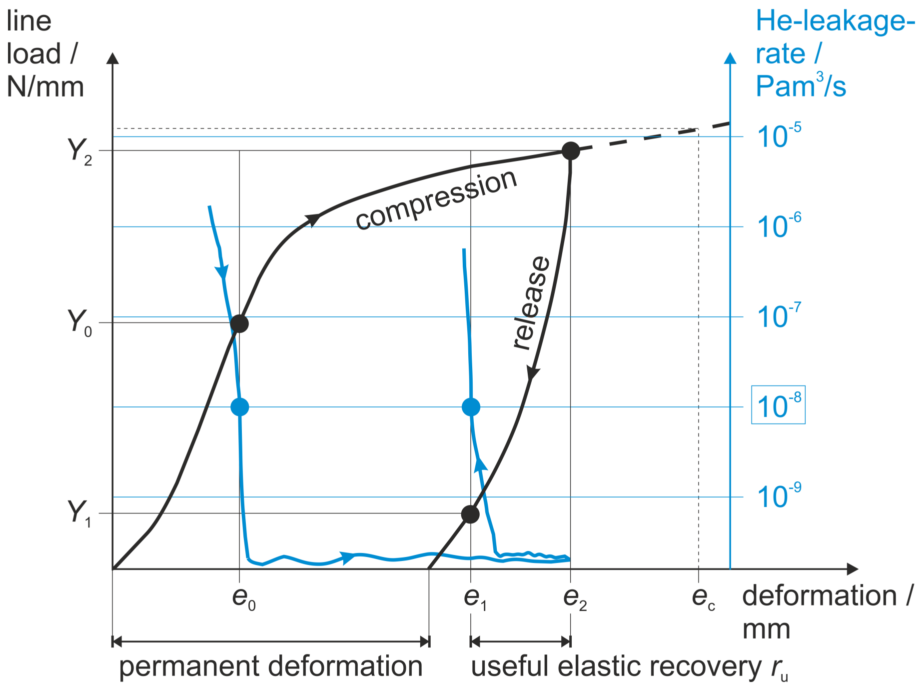

Figure 3.1: Characteristic sealing curve, partly in accordance with [27]

The value r_u is fixed on the basis of experimental appraisals of the characteristic sealing curve. These tests are generally carried out at room temperature and with negligible holding time. The resilience of the seal decreases due to ageing processes related to temperature and service life. These effects are to be taken into account when evaluating the sealing function of the closure system.

3.3.5 Elastomer seals

If elastomer seals belong to the containment, the resilience capacity of these materials must also be defined. The necessary assessments are to be made in the course of the seal validation process.
The resilience of the seal during operation under pressure reduces due to the impairment of the rubber-like elasticity resulting from physical and chemical changes of the material. Temperature, surrounding fluids and ionising radiation are all factors which influence this process. These variables which affect the resilience of the elastomer seal are to be taken into account in the design of the closure system. Consideration must also be given to the fact that the lowest temperature required under dangerous goods regulations can also adversely affect the rubber-like elasticity of the material. This can result in the reversible reduction or the irrecoverable loss of the resilience of the seal.

3.4 Determination and assessment of stress conditions

3.4.1 Strength of lid bolts

Calculation of effective stress

VDI 2230 [28] is used to calculate the tensile stress $\sigma_{\mathrm{z,Mon}}$ and the torsional stress $\tau_{\mathrm{G,Mon}}$ during the assembly of the connection. The equivalent stress σ_v required to assess the strength of the bolts in the assembly state is calculated according to Equation (2.4).

The effective tensile and bending stresses (nominal stresses) from the stress distribution obtained in the FE analysis must first be derived for routine, normal and accident conditions of transport using the same procedure as outlined in Section 2.4.4. The nominal equivalent stress is then calculated according to Equation (2.4) with a reduction coefficient of $k_\tau = 0.5$ [28]. In justified exceptional cases, e.g. low levels of load, the bolt tensions can be calculated analytically, preferably in accordance with [28].

Stress evaluation

The criteria for the stress evaluation are set out in Table 3.1. Deviations from the specified limits for assembly state are permissible in justified cases if the criteria for routine, normal and accident conditions of transport are met. If appropriate reasons are given, lower safety factors may also be applied for the bolts on the small lids which are not subject to the additional load imposed by the container internals. Under accident conditions of transport, the maximum equivalent stress should essentially meet the criterion $\sigma_\mathrm{v} \leq R_{\mathrm{p}0.2}(T_\mathrm{max})$ (Table 3.1). If this criterion is not met in the justified exceptional case, whereby the limit may only be exceeded to a minor extent, a residual plastic deformation of the bolts after the accident loading must be assumed. Therefore additional evidence is required in this case to verify that the bolts are still imposing sufficient pressure on the seal after the accident to guarantee helium-tightness despite the localised plastic elongation. In this case, an elastic-plastic behaviour of the bolt material has to be applied for the additional calculations of the impact phase of an accident. The subsequent simulation of the state after the impact loading takes account of the sealing force and the maximum operating pressure as the external load on the lid. The sealing function is evaluated in compliance with Section 3.4.3. Bolts with a high ratio of yield point to tensile strength (yield ratio) and, at the same time, a limited capacity for plastic deformation, equivalent to strength class 10.9, are exempt from this rule and must always meet the criteria set out in Table 3.1.

	Assembly	*Routine-*	*normal and accident*
		conditions	*of transport*
primary lid, secondary lid, small lids with loads caused by content	$\sigma_\mathrm{v} \leq \dfrac{R_{\mathrm{p}0.2}(T_0)}{1.5}$	$\sigma_\mathrm{v} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.1}$ $\sigma_\mathrm{z} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.5}$	$\sigma_\mathrm{v} \leq R_{\mathrm{p}0.2}(T_\mathrm{max})$ $\sigma_\mathrm{z} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.1}$ $\sigma_\mathrm{z} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.5}$ [3]
small lids without loads caused by content	$\sigma_\mathrm{v} \leq \dfrac{R_{\mathrm{p}0.2}(T_0)}{1.1}$	$\sigma_\mathrm{v} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.1}$ $\sigma_\mathrm{z} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.1}$	$\sigma_\mathrm{v} \leq R_{\mathrm{p}0.2}(T_\mathrm{max})$ $\sigma_\mathrm{z} \leq \dfrac{R_{\mathrm{p}0.2}(T_\mathrm{max})}{1.1}$

Table 3.1: Criteria for the stress evaluation of the lid bolts

[3] This criterion applies to bolts with a high yield ratio and also a limited capacity for plastic deformation, equivalent to strength class 10.9. Therefore, when using such bolts, a higher safety factor is to be taken as a basis for the average tensile stress for normal and accident conditions of transport. Reasons must be given if this criterion cannot be met.

3.4.2 Surface pressure and length of thread engagement

The effective surface pressure in the bolted lid joints is calculated according to Equation (2.12). The surface pressure is relevant in the case of creep processes and continuous load [28]. An evaluation in accordance with Equation (2.13) is therefore only reasonable for assembly and routine conditions of transport. Reference may be made to Section 3.3.1 for the calculation of the limiting surface pressure $p_{\mathrm{G}}(T_{\mathrm{max}})$.
The stipulations set out in Section 2.4.6 apply in turn to the calculation and evaluation of the length of thread engagement of the lid bolts.

3.4.3 Compression of seals

Even taking account of embedding effects and heat-induced changes, the minimum bolt pre-tension force $F_{\mathrm{M,min}}$ calculated as instructed in Section 3.2.1 must guarantee sufficient pressure on the seal, i.e. the full-contact seating of the lid. The full-contact assembly must guarantee in geometric terms the optimum operating point of the respective seal (e_2, Y_2, Fig. 3.1) specified by the seal manufacturer.
The seals are often simulated in the FE models as linear forces. Therefore the FE analyses can show local expansion between claimed parts in the sealing area under assembly conditions. Possible changes in distance as a result of the arrangement of the seal and the bolts on different pitch circles are also included in the FE analysis. The total initial expansion u_{M} can be deduced from the node spacing in the seal area in the FE analysis of the assembly state.

Calculating the resilience of the seal under load

The resilience Δs under routine, normal and accident conditions of transport can be determined geometrically as the distance u_{D} of the sealing surfaces under load related to the calculated initial gap u_{M} of the sealing surfaces after full-contact lid assembly as in Equation (3.2) if u_{M} is provable shown to be a modelling effect. If this is not the case, the resilience Δs can only be calculated conservatively with u_{D}.

$$\Delta s = u_{\mathrm{D}} - u_{\mathrm{M}} \tag{3.2}$$

Evaluating the resilience of the seal under load

The resilience Δs under load is evaluated with the help of the Criterion (3.3) which limits the permissible resilience of the seal $r_{\mathrm{u,zul}}$ in due consideration of the minimum bolt pre-tension force $F_{\mathrm{M,min}}$. The safety factor 2.0 of Equation (3.3) refers to the in tests determined useful elastic recovery until the sealing criterion is transgressed (resilience value r_{u}, cf. Section 3.3.4). It can be lower if there is sufficient statistically verified data, including the aspect of the ageing effects relating to temperature and service life.

$$\Delta s \leq r_{\mathrm{u,zul}} \text{ with } r_{\mathrm{u,zul}} = \frac{r_{\mathrm{u}}}{2.0} \tag{3.3}$$

The Criterion (3.3) applies during the loading phase under routine conditions of transport and also under normal and accident conditions of transport. After the short-term decompression during the impact phase under normal or accident conditions of transport, the assumption can then be made that sufficient recompression will be applied to the seal if the bolts were only subject to elastic deformation. If the criterion (3.3) was transgressed for a short period

in the impact phase, the analysis of activity release must take account of the fact that leakage rates may be higher than the standard helium leakage rate discussed in Section 3.3.4.
If the bolt is subject to a slight plastic deformation in an exceptional case under accident conditions of transport during the impact phase, an additional calculation is needed to take account of the elastic-plastic material properties (cf. Section 3.4.1). It must be possible to guarantee that the Criterion 3.3) will also be met again after the impact loading. In this case the analysis of activity release must also take account of increased leakage rates.
The fulfilment of the resilience Criterion (3.3) is a necessary condition to comply with the sealing function, which is often verified in practice by confirmation of the standard helium leakage rate, as explained in Section 3.3.4. The sealing effect and therefore the applicable leakage rate for the analysis of the activity release can be subjected to yet more influences, however, which cannot be computed, e.g. the surface condition of the parts of the sealing system. Therefore, results from drop tests and, where applicable, from component tests must also be consulted for the determination of the leakage rates for the activity release analysis. The potential impairment of the sealing function if the lid slips out of place (e.g. for the lateral drop position, cf. Section 3.4.5) must be evaluated separately and must also be adequately taken into account in the analyses of activity release. The leakage rates which are to be assumed in this load situation must also be established on the basis of drop tests or component tests.
In contrast to the impact loads encountered in the mechanical drop tests, the closure system is subject to load over a longer period in the thermal test. The criterion according to Equation (3.3) must not be transgressed at any time during this test and in the subsequent cooling phase. The deformation in seal areas of the closure system due to the thermal test are to be superposed with any changes in shape (plastic deformation, displacement) which may be identified after the drop impact tests.

3.4.4 Strength of lids

Determination of effective stress

The distribution of stress in the lid can also be evaluated in the FE analysis which is generally required for the lid system. In justified exceptional cases, e.g. with simple lid shapes, it is possible to conduct a separate calculation with reference to analytical methods like the plate theory [1, 30].

Stress evaluation

The criteria for the evaluation of the stress on the lid depend on the material from which the lid is made.
For example, the integrity of lids made of ductile cast iron must be proven on the basis of *BAM-GGR 007* (*Guidelines for the Application of Ductile Cast Iron for Transport and Storage Casks for Radioactive Materials*) [4].
The protection against inadmissible plastic deformations for the lids made of steel is sufficient if the condition (3.4) is met for any point of the lid.

$$\sigma_{\mathrm{v}} \leq R_{\mathrm{p0.2}}(T_{\mathrm{max}}) \tag{3.4}$$

If the criterion contained in Equation (3.4) is not fulfilled, further explanatory statements will be required to show that the safety objectives are met, making due reference to the material properties and the design of the lid system.

3.4.5 Slippage of lids

Calculation of effective forces

If the lids slip to the side, it must generally be assumed that the sealing function will be impaired and that the leakage rate will have changed.
The internal pressure and also the corresponding acceleration have to be considered for the calculation of the effective forces under routine conditions of transport. Appendix IV of the *IAEA* SSG-26 [21] [4]gives an initial overview of the relevant acceleration values applicable to transport by road, rail, sea and air. Other relevant national and international norms and standards include [7, 8, 22, 26, 29].
If the effects are not assessed by way of a dynamic analysis, the relevant accelerations during the impact phase under normal and accident conditions of transport must be used. Consideration must also be given to any interaction arising in this case between the lid and other parts of the package.
The accelerations for the aforementioned load cases under routine, normal and accident conditions of transport are the basis for the calculation of the inertial force F_{T} of the lid, taking account of the acceleration components transversal to the longitudinal axis of the container and their possible combination.
The axial component of acceleration and the internal pressure act as an operating force on the bolts. The resulting clamping force F_{N} can be obtained from the FE analysis, for example, by evaluating the nodal forces at the interface. Together with the minimum coefficient of friction μ between the lid and base body, this gives the frictional force F_{R} (3.5).

$$F_{\mathrm{R}} = \mu \, F_{\mathrm{N}} \tag{3.5}$$

Evaluation of forces

There must be no possibility of the lids slipping out of place under routine conditions of transport. Proof of adequate prevention of slippage S_{G} shall be deemed to have been provided if the frictional force arising with minimum pre-tension of the bolts, taking account of embedding effects and axial forces (internal pressure, axial acceleration), is greater than the inertial force (3.6).

$$F_{\mathrm{R}}(F_{\mathrm{M,min}}) \geq S_{\mathrm{G}} \, F_{\mathrm{T}} \tag{3.6}$$

A value of $S_{\mathrm{G}} = 1.8$ based on [28] (dynamic stress) is recommended for the safety factor under routine conditions of transport. If Condition (3.6) is not met for normal and accident conditions of transport with a safety factor of $S_{\mathrm{G}} = 1.1$, slippage of the lid is possible. This slippage is to be taken into account in the activity release analyses in view of a potential change in the leakage rate arising as a result and with respect to additional loading on the bolts.

3.4.6 Other analyses

IAEA SSR-6, §613 [20] dictates that the package must be verified to be capable of withstanding the effects of any acceleration, vibration or vibration resonance which may arise under routine conditions of transport without any deterioration in the effectiveness of the closing devices. If routine conditions of transport cause stresses which are relevant to safety in this context,

[4]The calculations can also be carried out in a simplified manner with covering values, see note after Table 2.1.

further analyses must be provided but this is beyond the scope of this guideline. The evidence also required to show that fastenings cannot unintentionally loose is provided with respect to the lid connections addressed in this guideline by the proof of prevention of slippage (Section 3.4.5) and may be ommitted [19].
A fatigue strength analysis may be required for the bolts on lid systems if the LAPs intended for the crane operations of the container are fitted on lids. The definitions in Sections 2.2.3 and 2.4.7 of this guideline then correspondingly apply in respect of the load assumptions, the analysis procedures and the assessment criteria. The effects of the repeated assembly and disassembly operations are to be taken into account in the fatigue strength analysis of the lid bolts with reference to [24].

List of symbols

A	Cross-section or area
A_{K}	Bolt head bearing area
A	Stress cross section of a bolt
D	Total damage of a component according to Palmgren-Miner rule
D_{M}	Critical cumulative damage Miner's sum
$E(T_0)$	Modulus of elasticity at room temperature
$E(T_{\mathrm{max}})$	Modulus of elasticity at maximum operating temperature
$F_{\mathrm{M,min}}$	Minimum bolt pretension force
F_{N}	Clamping force or clamp load
F_{R}	Frictional force
F_{T}	Inertial force
K	Number of load collective passes
M_{b}	Bending moment
N	Axial force
N_{D}	Number of cycles associated with endurance strength
N_{i}	Number of endurance stress cycles for $j_{\mathrm{erf}} \cdot \sigma_{\mathrm{ai}}$
R	Stress ratio
$R_{\mathrm{mB}}(T_{\mathrm{max}})$	Tensile strength of the bolt material at maximum operating temperature
$R_{\mathrm{mM}}(T_{\mathrm{max}})$	Tensile strength of the nut or blind hole material at maximum operating temperature
$R_{\mathrm{p0.2}}(T_0)$	0.2-% yield point at room temperature
$R_{\mathrm{p0.2}}(T_{\mathrm{max}})$	0.2-% yield point at maximum operating temperature
S_{D}	Safety factor for the fatigue analysis of bolts
T_0	Room temperature during installation
T_{max}	Maximum operating temperature
W	Section modulus of the bolt
W_{K}	Section modulus under the bolt head
Y_0	Compression load (line load) from where the required sealing level (standard helium leakage rate) is obtained when load is first applied
Y_1	Compression load (line load) under which the required sealing level (standard helium leakage rate) is not maintained when load is dropped
Y_2	Force of pressure (line load) at the optimum operating point
e_0	Deformation (compression) at line load Y_0
e_1	Deformation (compression) at line load Y_1
e_2	Optimum deformation (compression) at line load Y_2
e_{c}	Deformation (compression) above which the seal can be damaged
e_{R}	Useful elastic recovery

h_i	Number of cycles for step i (stage frequency) of a stress collective
j_{erf}	Required safety factor
j_{KTA}	Safety factor according to KTA
k	Inclination exponent in the stress-number curve
k_τ	Factor for torsional stress
l_{erf}	Required length of thread engagement
l_{Gew}	Effective length of thread engagement
n_i	Total number of stress cycles for step i
p_{G}	Limiting surface pressure
$p_{\mathrm{G}}(T_{\mathrm{max}})$	Limiting surface pressure at maximum operating temperature
p_{max}	Effective surface pressure
s	Lever arm for reference point
Δs	Elastic recovery
u_{D}	Distance of sealing surfaces under load
u_{M}	Distance of sealing surfaces in assembly state
α_{T}	Coefficient of thermal expansion
ϵ_{pl}	Equivalent plastic strain
μ_{Kmin}	Minimum coefficient of friction under bolt head
μ_{Kmax}	Maximum coefficient of friction under bolt head
μ_{Gmin}	Minimum coefficient of friction in thread
μ_{Gmax}	Maximum coefficient of friction in thread
μ	Coefficient of friction at the interface
σ	Normal stress
σ_{ai}	Stress amplitude of step i
σ_{ASG}	Endurance strength of bolts rolled after heat treatment
σ_{ASV}	Endurance strength of bolts rolled before heat treatment
σ_{b}	Bending stress
$\sigma_{\mathrm{b,max}}$	Maximum bending stress in lid
σ_{lgS}	Mean logarithmic standard deviation
σ_{mi}	Mean stress of step i
σ_{o}	Maximum stress
σ_{u}	Minimum stress
σ_{v}	Equivalent stress according to the von Mises theory (von Mises maximum distortion energy criterion)
σ_{z}	Tensile stress
$\sigma_{\mathrm{z,Mon}}$	Tensile stress of bolt in assembly state
$\tau_{\mathrm{G,Mon}}$	Torsional stress induced by tightening of bolt

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