FEM CALCULATIONS OF UHV ALL-METAL DEMOUNTABLE JOINTS FOR LEP

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ABSTRACT

The 27 km vacuum chamber of the Large Electron Positron Collider (LEP) at CERN includes some 15,000 mixed aluminium/stainless steel flange pairs. The overall dimensions of these all-metal demountable joints were determined by the LEP design parameters and a bolted version (instead of segmented V-clamps) was chosen with a view to the space available and a sufficient sealing force as well as cost. The joints are sealed with aluminium gaskets (copper could introduce electrochemical corrosion), and the system is baked at 150°C in order to achieve the static base pressure of $2.5 \times 10^{-9}$ Pa ($2 \times 10^{-11}$ Torr). In addition to extensive testing of the LEP vacuum joints, and the more recent field experience, calculations of this sealing system have been carried out by means of the Finite Element Method (FEM). The results are summarized and illustrated with a load/deflection model showing the functioning principles and the effective resiliency, i.e. degree of quality, of this type of vacuum joint.

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1. **Introduction**

The sealing effect in all-metal demountable vacuum joints is normally obtained by the interface contacts when a gasket is compressed between a pair of flanges. The leak tightness, defined by a given sensitivity of a leak detector, will be preserved as long as a certain minimum sealing force can be maintained. The sealing force originates from the tightening of the joint by means of bolts or clamps which produces elastic as well as plastic deformation of the seal components. The plastic flow, which is practically indispensable for the functioning of any UHV joint, is usually provided by the ductility of the gasket material. Most of the elastic deformation normally occurs in the flanges and/or the fasteners (bolts, clamps, etc.), but some gasket designs offer a high degree of resilience which may compensate for rigid (inelastic) flanges.

UHV joints which are baked have to maintain their leak tightness also during and despite repeated heating/cooling cycles. These bake-outs often modify the conditions reached after the tightening of the joints, whereby a softening of the gasket material and differential expansion of the seal components are the important detrimental effects. The overall result may be a temporary or permanent loss of sealing force, in particular during and/or after the first bake-out cycle.

The processes of tightening, loading and/or baking flanged joints, including vacuum seals, can be described and analyzed by means of relevant strain versus load curves\(^1\). In such load/deflection diagrams, the "working point" of the joint is expressed by the intersection between the elastic part of the load/compression curve of the gasket and the load/deflection curve of the flanges together with given fasteners. Unfortunately, current standards for the design of flanged joints do not seem to include guide-lines for the calculation of the elastic deformations in joints tightened "flange-to-flange". Moreover, in some preliminary standards one may find statements claiming that "in joints designed to be tightened flange-to-flange no noticeable deflections are to be expected"\(^2\) or, to the contrary, "with metallic gaskets of all kinds the change of thickness (elastic compression) is so small in comparison with the deflection of the flanges that it can be ignored"\(^3\).

In reality, however, there must be some elastic deformation in one or several of the components of the joint to provide the minimum sealing force, or stored energy, required to maintain a leak tight seal. In the following, the results of the FEM calculations of the elastic deformations of the LEP type vacuum flanges with their bolts are presented together with the empirical load/compression curve of the gaskets, thus giving the load/deflection diagrams with the "working points" of the LEP vacuum joints.
2. **Design**

The design principle of the 3 basic types of LEP joints is shown in Fig. 1. The essential overall dimensions are given in Table 1. Obviously, the inner diameters or apertures of the flanges may vary, depending on the type of vacuum chamber or component, i.e. valve, pump, feedthrough etc. The dimensions of the medium and small sized LEP joints could be adapted to the ISO Standard 3669 (1986), Secondary Range for Bakeable Flanges. Although it was recognized that this attempt of standardization was not optimal from a functional point of view, because of the unusual heterogeneous joints with mixed aluminium/stainless steel flange pairs, it certainly offered some attractive features. The essential advantages seemed to be the possible use of standard size fasteners throughout, i.e. identical to those used with common types of bakeable UHV stainless steel/stainless steel joints, as well as the possibility of easily transforming standard stainless steel flanges into the LEP design.

The dimensional tolerances of the LEP flanges and gaskets are similar to those normally used with the ISO Standard 3669 (1986). However, the surface finish of the sealing area of the LEP flanges is considered to be of prime importance for the functioning of these joints and, accordingly, its total surface roughness Ra has been specified to 6-10 μm, obtained by turning and with a feed of 0.15 mm.

### Table 1

<table>
<thead>
<tr>
<th></th>
<th>H (mm)</th>
<th>A1 st. st. (mm)</th>
<th>A2 Al (mm)</th>
<th>B (mm)</th>
<th>Seal diameter (mm)</th>
<th>Gasket height (mm)</th>
<th>Groove depth (mm)</th>
<th>M (mm)</th>
<th>Size of bolts or studs</th>
<th>Number of bolts (studs)</th>
<th>Bolt force $F_{2\ max}$ (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>225</td>
<td>160</td>
<td>131 x 70 b)</td>
<td>205</td>
<td>170.5</td>
<td>6</td>
<td>5</td>
<td>22</td>
<td>M8</td>
<td>12</td>
<td>23000</td>
</tr>
<tr>
<td></td>
<td>114</td>
<td>63.5</td>
<td>60</td>
<td>92.2</td>
<td>74.7</td>
<td>5</td>
<td>4</td>
<td>17.5</td>
<td>M8</td>
<td>8</td>
<td>23000</td>
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<tr>
<td></td>
<td>70 a)</td>
<td>36</td>
<td>30 c)</td>
<td>58.7</td>
<td>40.5 d)</td>
<td>5</td>
<td>4</td>
<td>13</td>
<td>N6 e)</td>
<td>6</td>
<td>14000</td>
</tr>
</tbody>
</table>

a) Flange integrated with feedthrough
b) Elliptical aperture
c) In 44 mm thick block
d) Later changed to HELICOFLEX R type HN.200 with mean diameter 43 mm
e) Studs
The gaskets used in the LEP joints have a reinforced "diamond" section with a 90° "knife edge" made by turning from 3/4 work hardened sheet material of the creep-resistant alloy A199.85Mg0.5. The yield strength of this material at ambient temperature is σ₀.2/20 = 138 N mm⁻², and it may increase by further cold working to maximum σ₀.2/20 = 180 N mm⁻² ⁴). At 150°C (after 168 hours) the yield strength of the gasket material drops to σ₀.2/150 = 123 N mm⁻², or about σ₀.2/150 = 150 N mm⁻² in case of maximum cold working⁵). After bake-outs the yield strength reverts to practically its initial value at ambient temperature. The reduction of Young's modulus by some 5% at 150°C is not taken into account in the following.

The fasteners used in the LEP joints are of the quality A4-100 according to the ISO specification 3506 (1979), i.e. with the steel grade A4 equivalent to AISI 316 and the property class 100 corresponding to a minimum tensile strength of Rₘ = 1000 N/mm².

3. Modelling

The complete joint assembly is considered to be axisymmetric, thus permitting the calculation of the flange deformations by means of the INCA programme in the CASTEM ⁶) finite element software package for structural calculations.

During tightening of the joint, the gasket undergoes relatively heavy plastic deformations which changes its cross-section and makes a modelling of this component rather complicated. In order to overcome this difficulty, the sealing force transmitted by the gasket to the flanges is modelled by a line force along the original seal circumference. Likewise, the forces of the fasteners, which are acting as point loads on the flanges, are taken to be an average line force around the bolt circle.

The intrinsic load/compression curve for each type of LEP gasket was determined empirically in a hydraulic press by recording the compression rates as a function of the applied force, see Fig. 2. On the other side, the maximum effective tightening force in the bolts, as a function of a given applied torque, was checked in an appropriate test set-up and found to correspond well with the nominal values generally given for fasteners in the ISO property class 100 and with a friction coefficient of μ = 0.10 ⁷). However, the net sealing force transmitted from the bolt tightening force via the flanges to the gasket depends, indeed, on the elastic behaviour of the flanges when these are getting in contact. The "working point" of the joint after tightening "flange-to-flange" with a given torque, therefore, can only be determined by iteration. In a simple approach, with the given effective tightening force P₂ max in the bolts and two arbitrary values of the seal force F₁' and F₁" in the vicinity of the intrinsic load/compression curve for the gasket, two fictive "working points" can be calculated, see Fig. 2.
Assuming that the simulated load/deflection curve for the flanges between these two fictive positions be linear, the true "working point" for the joint just after tightening can then be determined graphically by its intersection with the intrinsic load/compression curve of the gasket.

The fasteners are modelled by springs, which permit the calculation of the true load/deflection curve for the flanges as a function of a reduced sealing force, for instance due to gasket flow during baking, see Fig. 3. The spring constant K for a given bolt is derived at by taking the mean value of the pitch and the minor diameters to be the effective diameter of the bolt cross-section.

Whilst it remains relatively difficult to calculate the heavy plastic deformations of the gasket, in particular during tightening, use was made once more of empirical results. The flattening W of the 90° "knife edge" of the gasket after tightening as well as after bake-out, as a function of the reduction of the gasket height \( \Delta G \) (see Fig. 1), was carefully measured. A linear relation of \( W = 1.22 \Delta G \) was found which could, of course, vary by up to plus/minus 25% with the relevant dimensional tolerances, i.e. the initial gasket height, the quality of the "knife edge" tip and the groove depth$^6$.

With the above relationship reasonably good models of the gasket cross-sections could be created for given working points of the assemblies. The modelling of the gasket cross-sections, as for the flanges, was limited to the nominal dimensions.

4. Results

The result of the calculations of the elastic deformations of the two flanges with outside diameters \( H = 225 \) mm (see also Table 1) is given in Fig. 4. These calculations were made with two axisymmetric models, one for each axis of the elliptical aperture of the Al flange, i.e. assuming decoupling of the two axis. In the following, only the load/deflection curve corresponding to the minor axis will be considered, as this represents the more rigid and, hence, the most pessimistic case for this joint.

In Fig. 5 the load/deflection curve of the above-mentioned flanges and the elastic part of the intrinsic load/compression curve of the relevant gasket (see also Fig. 2) are presented together, thus forming the load/deflection diagram of the joint. It should be noted that the flange deflections, which are of course calculated for the seal diameter, are given with reference to the nominal groove depth.

Figure 6 shows the calculated bolt force remaining in the joint with an outside diameter \( H = 225 \) mm as a function of a possible reduction of the flange deflections, for example if the force transmitted by the gasket to the flanges decreases because of a bake-out.
For the frequently occurring LEP joints with outside diameters \( H = 114 \text{ mm} \) and \( H = 70 \text{ mm} \) (mounted with the so-called pick-up block) the results, as already described for \( H = 225 \text{ mm} \) in Figs. 4-6, are given in Figs. 7-10.

Extensive tests of the various types of the LEP joints showed that the relatively rigid configuration comprising a ceramic feedthrough with a stainless steel flange \((H = 70 \text{ mm})\) mounted on a solid 44 mm thick Al block by means of studs did not offer a completely reliable joint with respect to leak tightness after bake-out. Therefore, it was decided to try to seal this type of joint with the HELICOFLEX\textsuperscript{R} HN.200 type of gasket with an Al covering which, indeed, cured the problem. The results of the calculations of this version of the joint with \( H = 70 \text{ mm} \) are given in Figs. 11-12.

Finally, the result of the FEM calculations of one of the gasket cross-sections is given in Fig. 13, where the von Mises iso-stresses at the "working point" after tightening are shown. Except for some local stress concentrations in the gasket, which can be neglected in view of the ductile gasket material, the stresses do not exceed the possible maximum yield strength of about \( \sigma_{0.2} = 180 \text{ N mm}^{-2} \). (Micro hardness measurements of used gaskets confirm that the above yield strength may be reached by further cold working of the gasket during the tightening of the joint.)

5. Discussion

As can be seen from the load/deflection diagrams in Figs. 5, 7 and 9, the elastic strain energy areas vary considerably from one type of LEP joint to another, i.e. by a factor of about ten between the largest \((H = 225 \text{ mm})\) and the two smaller sized joints. (The specific loads are practically identical in the three diagrams whilst the scale of the elastic deformation, or strain, is ten times larger in Fig. 5.) This is not so much of a surprise since already the ISO Standard 3669 (1986), Secondary Range for Bakeable Flanges, is not optimized in this respect, (smaller flanges are more rigid than larger ones). In addition, the LEP type joint with \( H = 225 \text{ mm} \) and \( M = 22 \text{ mm} \) has only 12 bolts M6\textsuperscript{b} compared to the ISO Standard with \( H = 202 \text{ mm} \) and \( M = 22 \text{ mm} \) but 20 bolts M8. More important, however, for the functioning of the two truly flange joints of the LEP type \((H = 225 \text{ and } 114 \text{ mm})\) is the fact that the elastic deflections of the flanges are roughly from 4 to 40 times larger than the relatively modest elastic compression of the LEP gaskets. The shape of the gasket cross-section and the toughness of the gasket material, i.e. its combination of strength and ductility, are important parameters for this storage of elastic strain energy in the joints, and notably in the flanges. The dimensional tolerances of the gaskets and the flange grooves, which are not taken into account in the calculations, may of course affect the working point after tightening and, hence, the elastic strain energy area. In particular, if the gasket height is on the positive side and/or the flange grooves are on the negative side, not only will the seal force increase because of a bigger "bite" \( \Delta G \) in the gasket but also due to an ultimate work hardening of the gasket material.
(In general, if a net increase of the gasket height improves the seal performance, but this is done together with a reduction of the flange section, the improvement is, indeed, likely to come more from the gain of resilience in the flanges than from that of the gasket.)

The elastic strain energy area for the LEP joint with \( H = 70 \) mm (Fig. 9), is the smallest, as could be expected from its rigid design. This in itself could explain the failure of this joint to maintain its leak tightness after bake-out, i.e. it would be beyond the acceptable limit for this type of joint. However, in view of the fact that the joint with \( H = 114 \) mm only has about 50\% more resilience, it cannot be excluded that the use of studs which are threaded into aluminium contributed to the failure.

The calculation for the LEP joint with \( H = 70 \) mm when using a HELICO-FLEXR HN.200 type gasket shows an increase of the elastic strain energy area by a factor of about 4 (Fig. 11) compared with the same for the LEP gasket (Fig. 9). Evidently, in this case, it is essentially the gasket which contributes to the resilience of the joint, even if the specific seal force is 30\% lower.

No attempt was made to try to calculate the evolution of the load/deflection diagrams of the LEP joints during bake-out. Nevertheless, a rough estimate of the effects on the seal force when the LEP gaskets are exposed to 150°C can be made. Obviously, when the yield strength of the gasket material drops with temperature the seal configuration of the LEP type gaskets, in principle without capturing, can no longer resist the initial tightening force, and the gaskets will undergo some additional plastic deformation. In addition, baking of the LEP type joint will result in differential thermal expansions of the components of the joints, including the fasteners. However, as long as the elastic deformations of the flanges are large compared to any of the possible effects of the relative thermal expansions, these effects are considered to be negligible. In fact, even if the differential expansion between the flanges in the radial direction is up to one order of magnitude larger than the elastic compression of the gasket, and as a result the shear stresses will partly lead to some plastic deformation of the gasket, the resulting loss of gasket height in the range of a few \( \mu m \) should have little influence on the present load/deflection diagrams\(^9\). The predominant factor for maintaining a sufficiently high sealing force \( F_1 \) during (and after) bake-out is then the gasket ability to withstand at 150°C a good fraction of the original tightening force, i.e. minimum 40\%\(^1\). Therefore, in a first approximation, one may expect that the gasket during bakeout will be able to provide a sealing force \( F_1 \) which does not produce stresses in the gasket higher than the equivalent of its yield strength \( \sigma_{0.2}/150 = 123 \) N mm\(^{-2}\). With the gasket cross-section given in Fig. 13 and neglecting the minor deformations due to baking but admitting von Mises stresses up to 123 N mm\(^{-2}\), the calculated specific sealing force becomes \( F_1 = 190 \) N mm\(^{-1}\), or about 55\% of its initial value.
Using the simplest possible approach, the specific sealing pressure at the seal interface during bakeout should be equivalent to the maximum normal stress, i.e. the maximum yield strength of the gasket material at 150 °C, namely \( \sigma_{0.2}/150 = 150 \text{ N mm}^{-2} \), (during tightening the gasket material has been exposed to the maximum work hardening at the seal interface). Taking the maximum nominal seal width \( W = 1.22 \text{ mm} \), the specific sealing force then becomes \( F_1 = 150 \times 1.22 = 183 \text{ N mm}^{-1} \) which is very close to the value calculated above.

During cool-down the relative "contractions" of the joint components, in particular between the aluminium flange and the stainless steel bolts, will result in a further decrease of the specific sealing force \( F_1 \). This effect will depend on the resilience of the joint, but for the LEP joints with \( H = 225 \text{ mm} \) and \( H = 114 \text{ mm} \) a rough estimation gives up to a 10% reduction of \( F_1 \) from 150°C to ambient temperature. (In fact, the above assumptions and estimates are in good agreement with phenomenological studies which were carried out with the LEP joints.)

Finally, the estimated effect of a bake-out on the seal-force can also be applied to the remaining bolt force calculated for each type of LEP joint. As can be deduced from Figs. 5 and 6, the bolt force in the joint with \( H = 225 \text{ mm} \) may drop from 23,000 N after tightening to about 15,000 N after bake-out, or some 65% of its initial value. For the LEP joints with \( H = 114 \text{ mm} \) and \( H = 70 \text{ mm} \), the loss of the tightening force after bake-out is negligible (Figs. 8, 10 and 12). Generally, this means that a high degree of resilience in the LEP type of joint is obtained at the expense of losing part of the tightening force after, for instance, a bake-out. Obviously, such a loss of bolt force reduces the resistance of the joint to external tensile loads, including bending moments. Joints in which the gasket provides a larger proportion of resilience will be more tolerant to such effects.

6. Conclusions

The excellent performance of more than 15,000 joints of the LEP type is reported in another paper at this conference\(^{10}\). The calculations herein rather confirm the working principle of these joints and give some indication to their possibilities and limitations. The calculations also show that the use of existing standards will not always give optimum design with respect to the expected performance, i.e. leak tightness combined with resistance to external loads.

Acknowledgements

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References


5. Internal test report AUSTRIA METALL A.G., Ranshofen-Braunau (Austria), ref. 1/53956/5 of 14.06.85.


FIGURE 1. DESIGN PRINCIPLE OF THE LEP JOINTS
(FLANGE DEFLECTIONS NOT SHOWN HERE)
Figure 2  Determination of the working point after tightening
\[ F_1 = \text{Seal force} \]
\[ F_2 = F_{2 \text{ max}} - K(\Delta L) \]
\[ F_{2 \text{ max}} = \text{Bolt force after tightening} \]
\[ \Delta L = \text{Deflection (including compression) of flanges at the bolt circle} \]
\[ = \text{Bolt elongation} \]

Figure 3 Bolt modelling

**Figure 4 Load /deflection curve of the flanges**
Figure 5 Load /deflection diagram of the joint

Figure 6 Residual bolt force
LEP joint with H=114 mm

Figure 7 Load/deflection diagram of the joint

LEP joint with H=114 mm

Figure 8 Residual bolt force
Figure 9 Load /deflection diagram of the joint

Figure 10 Residual bolt force
Figure 11 Load /deflection diagram of the joint

Figure 12 Residual bolt force
LEP joint with $H = 225$ mm

Figure 13: von Mises iso-stresses in gasket cross-section with seal force $F_1 = 325$ N mm$^{-1}$