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Tightness of flange joints: A case study

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Abstract. One of the negative technological factors that often have a significant impact on the environment is the leakage of operating media (so-called fugitive emissions), which most often occurs at different separable joints. Therefore, high demands are placed on the joints – they must have sufficient strength to maintain pressure and other loads and also be tight enough to avoid undesired leaks. For a large part of such joints, standard flanges or flanges designed according to standards are used, in the European Union in particular according to EN 1092-1, EN 1591-1 and EN 13445-3. Although the strength of such flanges is sufficient for most applications, the basic design methods do not cover the tightness of flange joints and their emissions completely. A case inspired by industrial practice with a history of insufficient tightness resulting in leaks is investigated using FEA and EN 1591. The flange joint is modelled in software ANSYS 2019 R2 with parts of adjacent shells to take their effect into account. The goal of the analyses is estimation of gasket contact pressures resulting from load history. The pressures are directly related to the seal tightness. Lastly, effect of shell shape modification on the gasket pressures is also investigated and the results are compared with those from the original configuration. Effects of resulting pressures on seal tightness are discussed.

1. Introduction

Flange joints are an integral part of the equipment and differ in a number of factors that depend on the design method used [1–5]. The differences are in the design and choice of the flange connection. (for example, with the force transmitted directly or indirectly through the gasket), as well as in the choice of gaskets, which are characterized by different designs with a wide range of sealing parameters. Last but not least, the flange connection is closely coupled to the specified bolt force, which ensures sufficient contact pressure on the sealing surface while not exceeding the limit stresses and deformations of all parts of the flange connection.

In the design phase, appropriate tightness of the flange joint must be achieved by a combination of force in the bolts (that is influenced by the tightening torque, which is connected with the friction in the threads, the bolt head and on the strip), the stiffness of the flange, and the working range of the tightness of the selected gasket (each gasket has a minimum and maximum pressure – under minimum pressure undesired leakage may occur, after maximum pressure the seal may be destroyed). Problematic cases can occur, for example, when the flange joint is designed at the limit of the stiffness of the flange and at the same time at the lower limit of the seal tightness. In these cases, it is very difficult to achieve the tightness of the flange connection during assembly.

Modern calculation procedures such as EN 1591-1 [2] allow making a detailed design of flange joints. The basic problem is that tightness curves are generated for one set of parameters (medium, temperature and pressure) and for uniform pressure on the gasket. In the real case, however, the pressure



on the gasket is practically always uneven and differences in other parameters are very frequent. When designing or examining a particular flange joint, there is the possibility of using finite element method (FEM) analysis, where we can perform a very detailed evaluation of the joint according to the selected level of model accuracy. In the analyses, the non-linear behavior of gasket and other materials may be taken into account together with effects from connected pipes and standard or non-standard operating conditions.

The present article aims to investigate the flange joint of two heat exchangers placed on top of each other inspired by a case from industrial practice where problems with leakage of the joint occurred. Possibilities of leakage of such joints are discussed and FEM analyses investigating the influence of individual factors on contact pressure are performed and evaluated. Firstly, tightening force in the bolts and its transfer to the sealing surfaces and the resulting sealing pressure. Then, the influence of pipe force on sealing pressure and determination of force influence on leakage. Lastly, the geometry of the shell attached to the weld neck flange and the evaluation of the effect of the different stiffness of the shells on the load transfer from the pipe to the gasket and contact pressure respectively.

2. Description of investigated flange joint

The flange connection can be seen in figure 1. The diameter of the upper part shell is 2470 mm, the lower chamber has a diameter of 2190 mm. The length of the cone is 1346 mm, the dished shell has a smaller radius of 1000 mm and a larger radius of 1800 mm. The shell thicknesses are 15 mm. The flange joint includes 92 M30x3 bolts and camprofile gasket with graphite layers. The case 1 has a conical shell connected to the flange connection, case 2 then has a dished shell connected to the flange connection. The pressure chambers operate at pressures of 0.6 and 0.78 MPa.

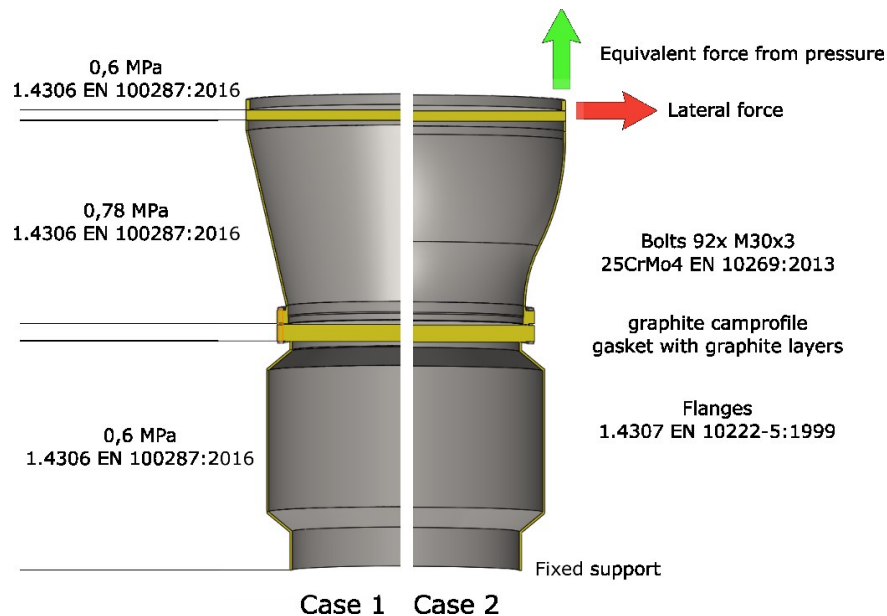


Figure 1. Geometry of the flange connection and boundary conditions.

3. FEM model, boundary conditions

The FEM simulations were performed in ANSYS 2019 R2 [6] as static analyses with small deformations. Symmetry was applied, so only half of the geometry was modelled. The boundary conditions were applied in three steps and can be seen in figure 1. The exact sequence of the entered values can be seen in table 1. The analyzed flange connection took into account the forces in the bolts determined by the calculation of the flange connection according to ČSN EN 13445-3 (chapter 11) [3]. To investigate the effect of the connected piping the lateral force $2 \cdot 10^5$ N was used. The force was

increased gradually and its effect on the sealing pressures is visible from the following plots. Bolt connection was simulated by beam connection. Gasket and its connection to contact surfaces was simulated using special gasket elements with frictional connection with friction coefficient 0.22 [7]. The mesh used for the simulation is shown in figure 2. The effect of temperature is not investigated here.

Table 1. Boundary conditions by load steps.

Load step	Bolt pretension (N)	Pressures (MPa)	Equivalent force from pressure (N)	Force from piping (N)
1	56099	0	0	0
2	Lock	0.6; 0.78	1.44E+06	0
3	Lock	0.6; 0.78	1.44E+06	2E+05

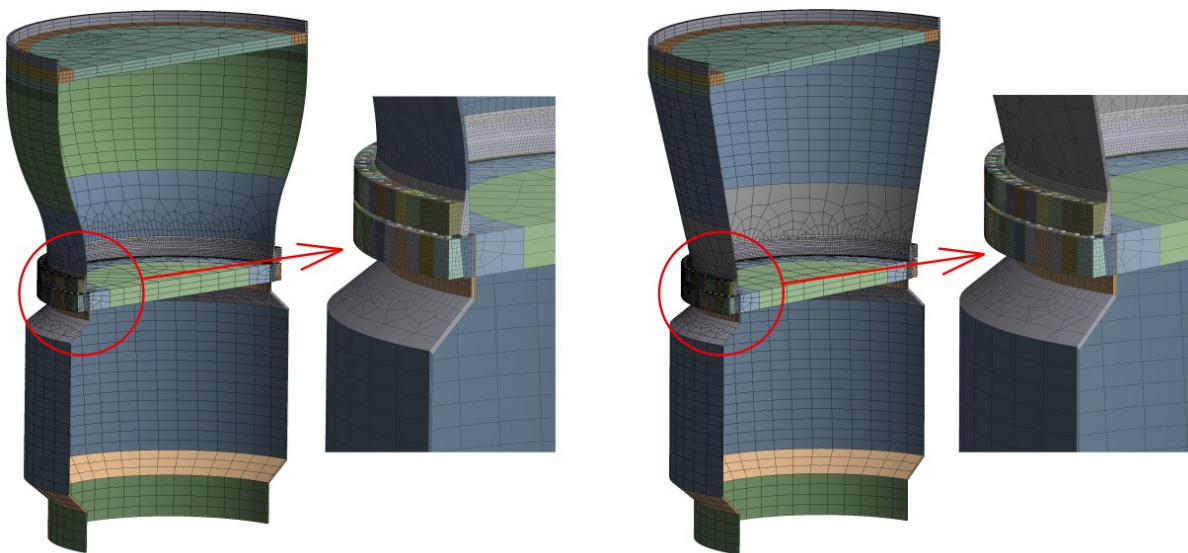


Figure 2. Mesh of investigated geometry.

The effects of different shell geometry on the leak rate of the flange joint were investigated. Therefore, goal of FEA model was to accurately estimate pressures on the gasket during whole load history, however, there were some noteworthy simplifications:

- FEA models were simplified – the tube sheet connected to the flange was modelled without tubes.
- The effect of temperature was neglected as it was not subject of interest for the case study. Temperature nevertheless is one of the parameters, which significantly influence the distribution of contact pressures and tightness of flange connection respectively [8].
- Analyses with small deformations were used due to good trade-off between computational time and results accuracy. In practice, analyses with large deformation may be more accurate.
- The gasket was modelled using special gasket elements with nonlinear behaviour, however, the rest of the structure was modelled using linear elastic behaviour as it provided adequate results with reasonable computation time. For more precise analysis, plasticity should be introduced to the whole model as it may influence the results.
- Bolts were modelled using beam connections that were in fixed contact with flange areas, which would be below bolt nuts or washers. In a case without lateral force, this should not be a problem, however, when the lateral force is present, the modelled bolt connections may have greater resistance to this force than real bolts. The current version of EN 1591-1 [2] allows to include effects of lateral force (and also axial moment) in flange joint calculations and the assumption used there is that the

joint resists the lateral force through friction between flange faces and gasket. In practice, friction between nuts/washers and flanges would also contribute to lateral force resistance.

- Mechanical behaviour of the gasket is captured extensively in normal direction using testing procedures from EN 13555 [8]. However, in presence of lateral force, other directions may be important, but the elastic and shear moduli are usually not available.

4. Results

FEM analyses were performed on two geometries (case 1 and case 2). For a general overview, the overall deformation of both models is shown in figure 3.

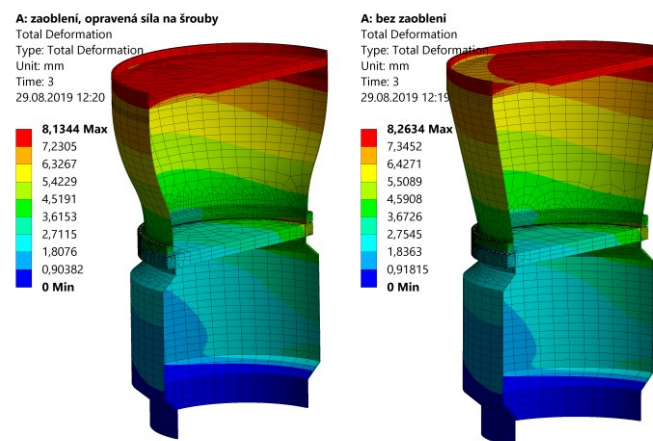


Figure 3. FEM results – total deformation.

4.1. Leak rate estimation

Postprocessing of FEA results was focused on gasket pressures and their influence on gasket sealing properties, especially the leak rate. According to EN 13555 [8], the leak rate of flange joint with gasket strongly depends on contact pressure. During assembly, contact pressure leads to seating of the gasket, so the higher assembly pressure leads to lower leak rate (up to the point where the gasket is crushed by excessive pressure). The assembly contact pressure usually determines behaviour of the gasket, because contact pressure in subsequent operation is often lower than the assembly pressure. Lowering the contact pressure leads to increase in leak rate. Therefore, the resulting leak rate is affected not only by actual contact pressure but also by the highest contact pressure in the gasket-flange face history. This behaviour can be captured by loading-unloading curves, that can be created from measurements of gasket under certain conditions according to EN 13555 [8].

Leak rate measurements according to EN 13555 [8] are represented in a semi-logarithmic plot (y-axis) by single loading curve and multiple unloading curves representing leak rate as a function of contact pressure. Significant values from the plot are also supplied in a table. The values in the table are considered just at the points, where a curve intersects border between tightness classes, so the information in the table is somewhat reduced in comparison to the plot. In order to get better precision, leak rate information for the article was taken from digitized leak rate plot.

4.2. Leak rate for maximum and current gasket pressure

For every loading pressure, an unloading curve can be constructed using semi-logarithmic interpolation between 2 adjacent unloading curves. The new unloading curve represents all leak rates for all the pressures lower than previously reached maximum loading pressure. Example of leak rate interpolation for maximum loading pressure 52 MPa and unloading pressure 35 MPa is shown in figure 4.

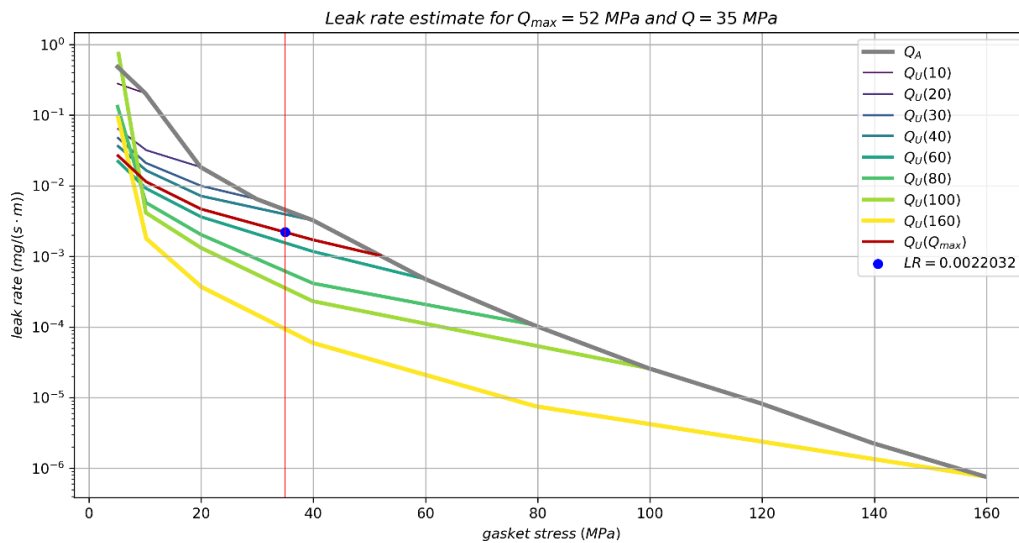


Figure 4. Leak rate plot and interpolation; new unloading curve for maximum loading pressure 52 MPa is marked by dot.

4.3. Overall leak rate at a particular point in time

Conditions under which the curves describing the gasket leak rate are obtained include uniform contact pressure at every moment. However, in practice, this would rarely be the case. Contact pressures in the investigated gasket are also far from uniform at every calculated time point. As an example, pressure distributions at the end of load history for both cases are shown in Figure 5. The distributions are clearly not uniform, with case 2 exhibiting slightly higher pressures.

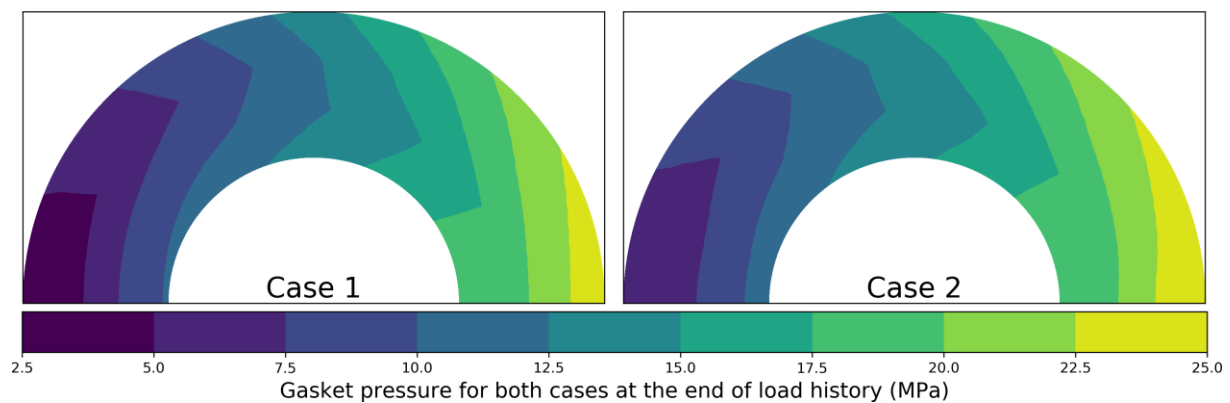


Figure 5. Gasket pressure distribution for case 1 (left) and case 2 (right) at the end of load history (gasket width has been enhanced in the image for better clarity).

The resulting leak rate of the flange joint as a whole cannot be directly estimated as one value for the whole gasket. Instead, it will be different for every point on the gasket seal face as shown in figure 6. Highlighted area in the figure represents leak rate distribution at the end of the load history for the case 2. In case of axisymmetric gasket pressure distributions, the area in the figure would be reduced to a curve.

Although a single leak rate value for the flange joint cannot be estimated using current understanding, the leak rate value should be within the bounds in the figure 6, i.e. $[6.597, 35.233] \mu\text{g m}^{-1} \text{s}^{-1}$ for the case 2. In the case 1 the bounds are $[6.8086, 49.266] \mu\text{g m}^{-1} \text{s}^{-1}$.

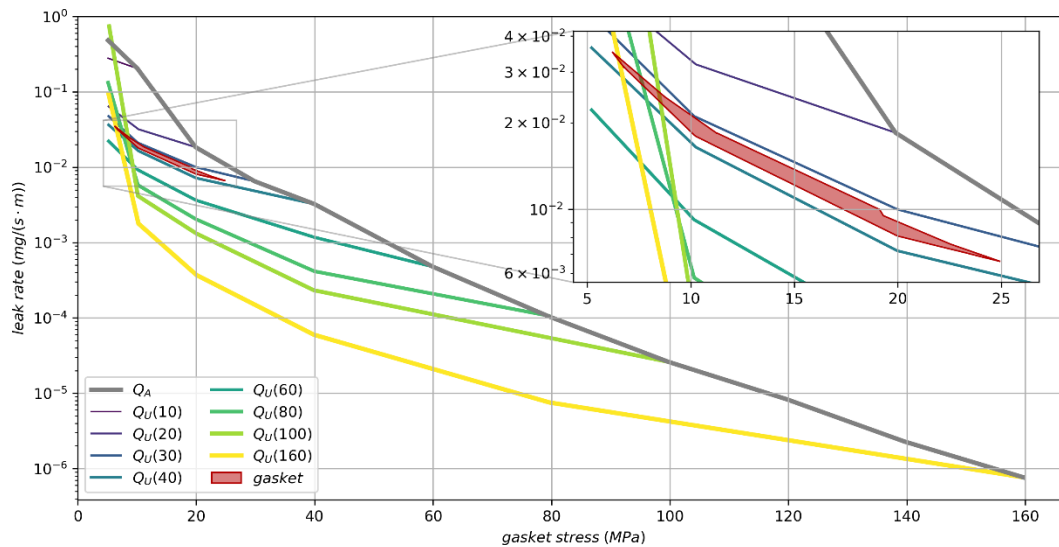


Figure 6. Leak rate "distribution" at the end of the load history for the case 2.

4.4. Corrections

Since the leak rate values were estimated from test data under different conditions than those in analysed cases, corrections had to be applied. EN 1591-1 [2] offers guidelines for corrections, but also emphasizes, that they may be inaccurate. Gasket leak rate test was performed with helium under room temperature with pressure difference ($\Delta p_{(ref)}$) 4 MPa. Pressure correction is calculated according to formula 1 from [2]:

$$L_{p(act)} = L_{p(ref)} \cdot \frac{\Delta p_{(act)}}{\Delta p_{(ref)}} \quad (1)$$

where $L_{p(act)}$ denotes actual leak rate and $L_{p(ref)}$ denotes reference leak rate from test data. Reference pressure difference $\Delta p_{(ref)}$ was 4 MPa and actual pressure difference $\Delta p_{(act)}$ changes during the second load step from 0 to 0.78 MPa. Corrected leak rate results in new bounds; [1.3277, 9.6068] $\mu\text{g m}^{-1} \text{s}^{-1}$ for case 1 and [1.2864, 6.8951] $\mu\text{g m}^{-1} \text{s}^{-1}$ for case 2. Lower expected leak rate bounds in case 2 can be partially attributed to higher gasket contact pressure at the end of the load history, which was shown in Figure 5.

4.5. Leak rate bounds during load history

If the contact pressures vary during operation, it may be also beneficial to calculate leak rate bounds of the flange joint as a function of time. For cases and load histories investigated in this paper, the above-mentioned plot is shown in Figure 7. As a conservative estimate of the actual leak rate, upper bounds could be used.

Since zero pressure difference means zero leak rate, which cannot be shown in the logarithmic plot, the corrected bounds start in the second load step (time 1 s in figure 7), where the internal pressure is introduced. The figure shows, that the leak rate bounds of both investigated cases are initially very similar. However, after internal pressure application and subsequent application of lateral force, the difference of the upper leak rate bounds rises to 28 %. In some cases, this difference may be significant, however, in terms of European tightness class, both cases would fit into $L_{0.01}$.

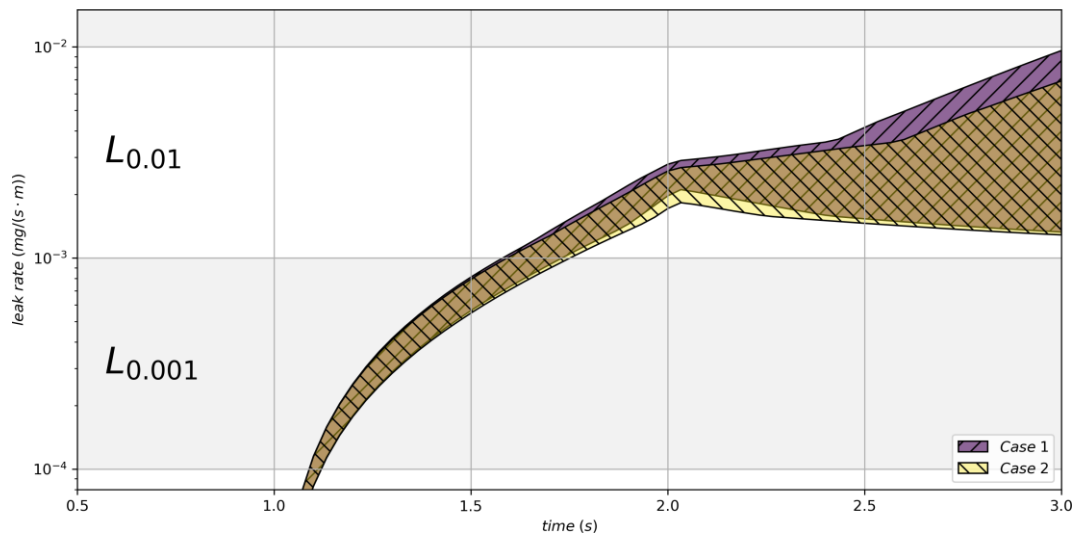


Figure 7. Comparison of leak rate bounds throughout load history (letters ‘L’ with subscripts denote tightness classes).

5. Discussion

Precise computational leak rate evaluation poses many problems. The most important ones are discussed here:

- Gasket leak rates after loading and subsequent unloading are measured for conditions that may significantly deviate from practice:
 - gasket geometry – not addressed
 - reference fluid (usually helium) – not addressed
 - fluid temperature – not addressed (temperature was not considered in the study)
 - fluid pressure – addressed
 - uniform gasket contact pressure – addressed by introducing leak rate bounds
 - smooth test face finish – not addressed
- Formulas for leak rate estimation in cases with different fluid, temperature and pressure than tested are suggested in EN 1591-1 [2], but they are not very precise.
- Effect of non-uniform gasket contact pressure (especially in combination with serrated flange face finish) on leak rate is not yet sufficiently described in the literature. Therefore, the approach used in this paper estimates leak rate bounds as a minimum and maximum from leak rates at every node on gasket contact face.
- According to EN 1591-1 [2], recommendations for leak rate interpolations should be in the next edition. Although the interpolation will be probably semi-logarithmic as used in this paper, some additional rules would be also helpful. For example, in the case 1 geometry in this paper, gasket contact pressure at the end dropped below 5 MPa, which is the lowest value in unloading curves. It can be expected, that simple extrapolation to lower pressures would, in this case, lead to underestimation of the actual leak rate.

6. Conclusion

This work was focused on the investigation of contact pressure in bolted flange joint and estimation of resulting leak rates for 2 different geometries. This topic was inspired by a real case with history of insufficient joint tightness. Original joint (case 1) included flanged tube sheet at one end and flange welded to conical shell at the other. In the second geometry (case 2), the conical shell was replaced by dished shell with the same thickness. The question was, how would this change affect tightness of the joint.

Since the tightness of the joint is strongly influenced by contact pressures in the gasket and the geometry of interest was not fully covered by standard calculations, FEA was used for estimation of contact pressures. For greater accuracy, special gasket elements with nonlinear Young modulus were used for modelling behaviour of the gasket. Boundary conditions included fluid pressures and their effects and also lateral force. Tightness evaluations require not only gasket contact pressures at the end, but also their history up to the evaluated time point, therefore, loads in the simulations were introduced in 3 consecutive steps: bolting, fluid pressure, lateral force.

Gasket contact pressure histories were evaluated using gasket test data and standard EN 1591-1 [2]. The major issue in the evaluation was, that gasket test data is obtained with uniform gasket contact pressure, which was not uniform in investigated cases. To tackle this issue, leak rate evaluation was performed for every node on gasket face separately and the expected leak rate for the whole flange joint at a particular moment was represented by interval and conservatively with its upper bound.

The results have shown, that the expected leak rate intervals for both cases were mostly overlapping. However, there was an observable difference, especially between the upper bounds of expected leak rates, which was greatly influenced by the lateral force. At the end of the load history the difference of 28 % in favour of modified geometry – case 2 was observed.

7. References

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