Circular Flange Joints of Pressure Vessel

Jiri Zacal(✉), Lukas Hruzik, and Vaclav Mosler

Department of Machine Parts and Mechanism, Faculty of Mechanical Engineering, VSB-Technical University of Ostrava, Ostrava, Czech Republic
university@vsb.cz

Abstract. For flange joints of pressure vessels, bolts are most commonly used. Even though they may seem to be only simple mechanical elements, the reality is ever so different. In some fields of engineering practice, these connections must meet particularly high technical requirements. The article discusses the issue of critical circular flange joints with gaskets during their assembly and subsequent operation conditions. The aim of the article is to describe characteristics of stress flow and deformations of two basic types of flange joints, which differ in the way of power flow. Subsequently, the methodology of EN 1591-1 standard is verified using the Final Element Method. The article also presents a description of the most common flange joints defects in technical practice and causes thereof. The ways to improve the methodology of flange joints design are discussed in the conclusion.

Keywords: Gasket · Tightness · Bolt · Pretension · Working Load

1 Introduction

A pressure vessel, a common device with application ranging from households through the chemical and petrochemical industries to primary or secondary loops of a nuclear power plant components. Depending on the area of application, different requirements are also applied to pressure vessels. These are, firstly, safety requirements. Pressure vessels are often working under critical operation conditions in areas where they come into direct contact with humans. A pressure chamber may contain a medium such as high temperature steam, flammable substances, acids and other dangerous substances, and even small leakage could pose a significant risk to the environment. Yet, bolt connection, arguably the most commonly used type of mechanical connection, is used for these purposes, even though it is often looked at sceptically. There is common and simplified view, that bolts are no more than simple machine elements. This can be accepted as sufficient in applications where their failure is not hazardous. However, in many other cases, it is necessary to remember that bolted joints are highly complex in terms of tensity. Underestimating importance of bolt connection could result in a fatal error.

This article discusses the issue of circular flange joints with gaskets that are both exposed to critical operation conditions and subject to high leak rate requirements at the same time. The first part provides a basic description of flange loads from assembly condition to the subsequent conditions. The next section deals with the basic classification

J. Homišin et al. (Eds.): ICMD 2020, AHE 17, pp. 92–104, 2023.
https://doi.org/10.2991/978-94-6463-182-1_12

of constructions of sealing flange joints in terms of working load flow. For the purpose of clarity, finite element method (FEM) results are used. It is also shown, what effect the type of flange construction has on the bolt working load in operation condition. Using FEM, these changes are analysed and compared with methodology of standard CSN EN 1591-1. The last part of the article describes problems that are often encountered in practice due to negligence of some basic principles of elasticity and strength of materials, selection of poor construction of joints, etc. Based on the research conducted so far, possible solutions to these problems are outlined. All calculations and generated FEM images are created by ANSYS Workbench 18.2 Academic Research.

2 Materials and Methods

Tension of flange connections with gaskets is classified according to specific load conditions (CSN EN 1591-1 2015). Initial load occurs when a flange couple is tightening. This is an assembly condition, which is designated I = 0. After that, the so-called subsequent condition is defined. These are states occur during a pressure test, in operation, or exceptional conditions (e.g. start up, cleaning, shut down or maintenance). They are marked I = 1, 2, 3 etc. For example, Fig. 1 shows an intercooler with the illustrated operating cycle.

Figure 1 is taken from (Zacal 2016). Below the figure is Table 1, with appropriate load conditions.

From Table 1 it is clear that the working chambers are not only affected by pressure (P = MPa), but also by temperature (T °C). For some flange joints, operating temperatures are at very high level (e.g. up to 500 °C). Computations regarding bolts and flanges that are exposed to long-term high level of stress and temperatures, are still very complex even nowadays. Due to temperature, the mechanical properties of steel are changing. In order to make a reliable calculation, it is necessary to start testing for so called creep deformation. However, this kind of testing is demanding in terms of time and costs. The measurement results are then determined only for simple strains and the number of

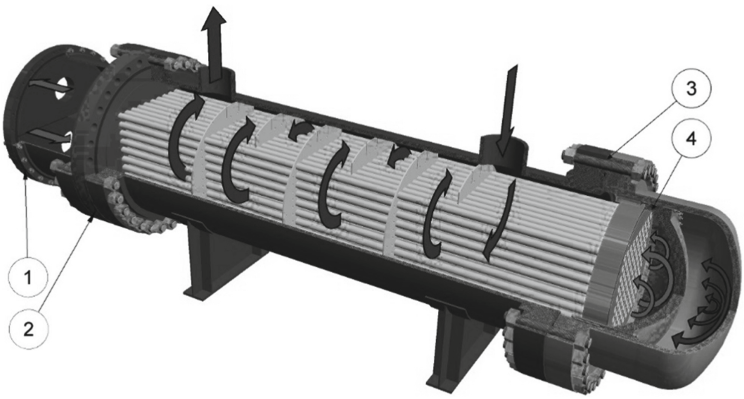

Fig. 1. Example of intercooler.

Table 1. Table with specification of load conditions.

	Cooling Chamber			Hot Chamber		
Load Conditions	I=0	I=1	I=2	I=0	I=1	I=2
P [MPa]	0	0.7	0.91	0	8.45	10.9
T [C°]	20	85	25	20	137	25

completed tests is not high. Often, there is no more to do than to use empirical knowledge, which is not always desirable for critical connections. Elevated temperature of partial components of a joint does not only affect the mechanical properties (stiffness of the components), but also introduces additional load. The increase in load can be attributed to a different coefficient of thermal linear expansion or a different temperature of the components (Pospisil 1968). The temperature differences between bolt and flange is often a momentary state when approaching the working temperature, but even then, a serious failure of the joint can occur, such as a failure of tensile yield strength and ultimate strength of the material. Another challenging aspect is the aggressiveness of the environment to which the flange joints are exposed. Unsuitable materials then disintegrate. A list of all the mentioned aspects consists of so called critical flange joints.

2.1 Problematics of Construction of Flange Joints with Gaskets

Design of flanges fundamentally influences deformation and tension of joints. Since very demanding requirements are placed on critical flange joints, there is a wide range of dimensioning, strength and tightness proofing. Regulations and standards clearly specify for what types of connections they are defined. This chapter presents some basic classification of flange joints according to force transmission signal, and influence of a flange connection on bolt working load. The analysis is only made for the most common flange type which is the integral flange with hub.

2.2 Basic Classification of Construction of Sealing Flange Joints

In terms of power transfer, flange joints are classified into floating type joints (FLT) and metal-to-metal contact joints (MMC) outside the sealing area in engineering practice.

Force Transmission of Floating Type Joints (FLT)

In a simplified way, FLT joints do not allow the flanges to contact metal around sealing surface, thus the whole gasket pressure is exerted on effective sealing surface. For principle of force transmission of FLT joints, see Fig. 2.

Advantages of FLT joints include greater sealing pressures (better sealing properties) and existence of a wide range of standards for calculations and design of a sealed flange joint. The best-known are, for example, CSN EN 1591-1, ASME Boiler and Pressure Vessel Code-Section VIII, KTA 3201.2, KTA 3211.2, and VDI 2200. All of these standards include a section for scaling of joints. The KTA, VDI and CSN EN standards also offer possibility of validation of strength and tightness of the joint.

A disadvantage of FLT joints is that they exhibit lower tolerance of external and dynamic loads and are more prone to thermal effects. This results in alteration of sealing force in operation condition. Therefore, it is necessary to account for all load conditions for given joint. It is also crucial to monitor magnitude of gasket force, considering minimum and maximum permissible gasket pressure. Flanges are affected by bending moment, resulting in decrease of their stiffness. Thus, all changes of load conditions are exhibited to greater extent in bolt tension, as angular inclination of flanges causes substantial increase of bolt bending tension. Surface layers of bolts are exposed to high strain. Due to angular inclination of flanges, effective gasket surface is decreased (see Fig. 11). Figure 3 depicts flow of stress throughout cross section of a flange in operation conditions, when steel gasket is used. This type of gasket is used in the demonstration as module 'Gasket' does not enable joint graphical representation of nonlinear materials in this type of analysis. However, the nature of stress flow and deformation is identical.

Figure 4 depicts the nature of deformation. For better representation, Fig. 4 uses 30-fold zoom.

Force Transmission in Metal-to-Metal Contact Joints (MMC)

When fully tightened by bolt connection, MMC type of flanges achieve a contact of metal-to-metal around the sealing surface. The main sealing forces are transferred not only by gasket but also by surrounding flange material (see Fig. 5).

Since not all MMC joints have the same appearance as Fig. 5, it is also possible to encounter a structure that has a gap between the flanges. Metal-to-metal contact occurs only close around the gasket (see Fig. 6) and flanges are affected by bending moment as in the FLT joints. Figure 6 shows the stress flow in the assembly condition using an expanded graphite gasket, which cannot be graphically displayed in this kind of analysis.

Advantages of MMC type of flanges include the fact that external stresses do not have much influence on joint tightness, hence it can be used to transfer larger loads (M. Schaaf 2003). Higher bolt pretension values can also be used without seriously damaging the gasket. This joint will also be able to withstand higher operating temperatures. As flange leaves (Figs. 5, 7 and 8) are not affected by bending moment, but only by pressure, the effective sealing surface in the operation load does not change dramatically. Disadvantages include the absence of standards for flange and seal scaling, due to the gasket relaxation and the sealing force disappearance. A proof of tightness of the joint more complex. The calculation is covered only by KTA 3201.2, KTA 3211.2 and VDI

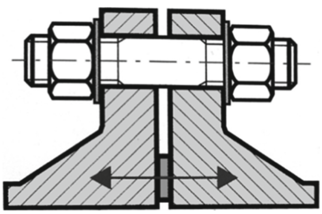

Fig. 2. Construction of FLT joint.

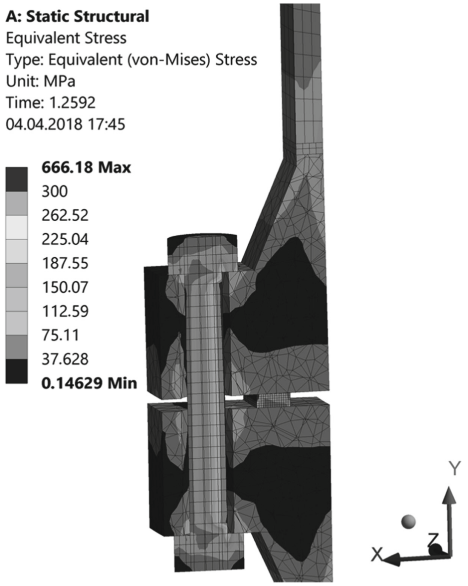

Fig. 3. Character of equivalent stress flow for FLT joints.

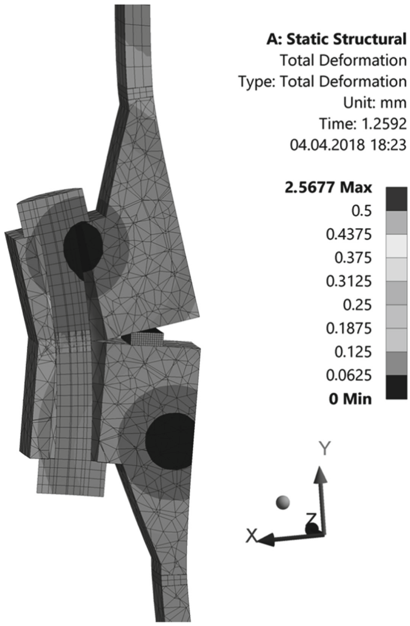

Fig. 4. Character of total deformation for FLT joints.

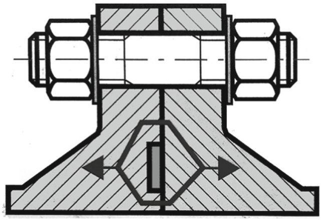

Fig. 5. Metal to metal contact joints (MMC).

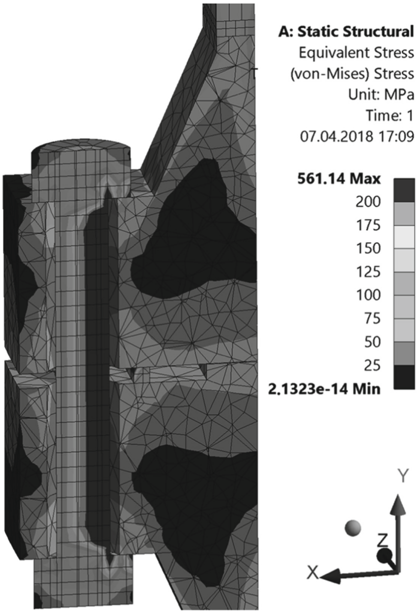

Fig. 6. Character of equivalent stress flow for MMC joints with gap.

2200. Figure 7 and Fig. 8 represent a typical stress flow in an MMC joint without a gap. Due to bolt pretension, the flange deforms only in the closest proximity around of bolt. The 'Bach Theory' states that the stress from frictional contact of nut or bolt head throughout cross section extends into the flange at an angle $\alpha = 45°$ and forms a so called pressure double-cone (Fig. 7, Fig. 8). See (Pospisil 1968) p. 169.

Figure 8 depicts the extent of effect of material around of bolt due to the 'Bach Double-Con'.

2.3 Effect of Construction Flange Joint on the Bolt Working Load

Knowledge of magnitude of bolt working load is crucial for design of sealed flange connection of a pressure vessel. For pressure vessels, bolt pretension connections are used exclusively. The magnitude of bolt pretension can be determined utilising existing standards. This is the magnitude of force in a bolt that guarantees tightness and strength of

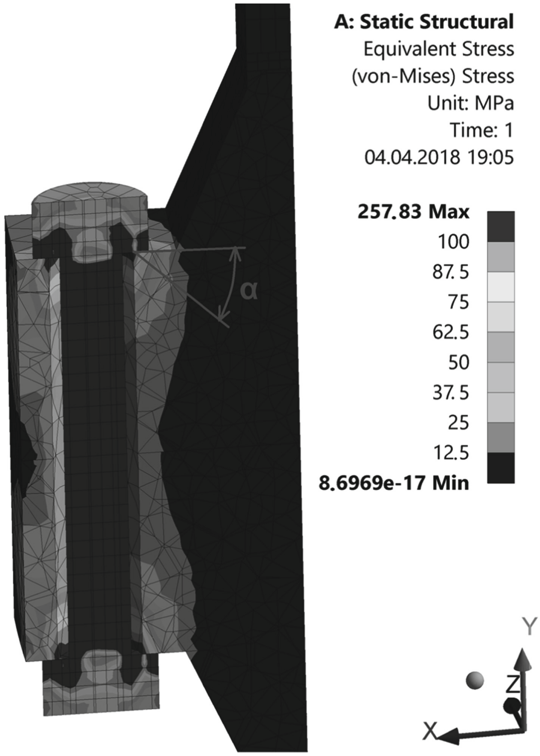

Fig. 7. Character of equivalent stress flow for MMC joints.

the joint under all load conditions (including the assembly condition). Using controlled tightening, required preload can be achieved with certain precision. Tightening accuracy is strongly dependent on selected tightening method (e.g. hydraulic, pneumatic, torque wrench, angle tightening, etc.) as well as on the type of lubricant used. For this purpose, it is proper to use an experiment for verification of friction (Kanaval 2017). These experimental measurements are usually performed by strain gauges. The specific information about the issue of strain gages are included in article (Jancar 2017). However, the flow profile of bolt working load in operation condition is no longer straightforward to determine. In this case, the pretension bolts are almost always exposed to high tension. The nominal stress approaches the tensile yield strength already in the assembly condition. An increase or decrease of bolt working load in operation condition is not only dependent on stiffness of the flanges and the bolt, but also on the location at which the operation force is exerted (Pospisil 1968). In some cases, determination of stiffness is either very complex or even not possible. Simplified computational methods often give different values from experimental measurements (Pospisil 1968). However, contemporary computations can also be verified using the FEM, improving accuracy of the computational approach. In this study, the problem is discussed only for integral flange with hub, which is one of the most commonly used flange geometries.

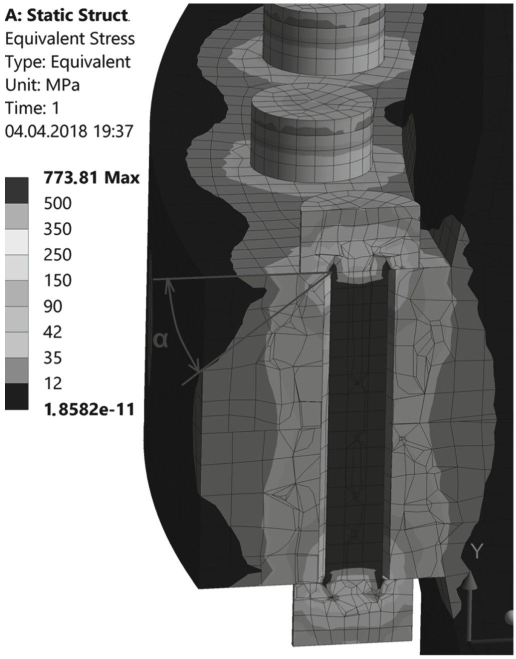

Fig. 8. Typical stress flow for double-cone theory.

A plot in Fig. 9 shows the dependence of bolt working load on both time and magnitude of the internal pressure in the vessel. This issue is addressed in more depth in article (Zacal 2016), however, this is an improved FEM model with more types of gaskets and load condition. The time axis is primarily used for better representation of the nature of FEM simulation. At time equal to 1 s, assembly condition is achieved, at which the bolt pretension is 50 000 N. After 3 s, the chamber is pressurized.

All of the depicted dependences in Fig. 9 describe construction of FLT joints, except for the red curve which describes an MMC joint without gap. From Fig. 9, it can be concluded that bolt working load in FLT joints decreases after working cycle is commenced. The same phenomenon is also proposed using calculation in CSN EN 1591-1 standard. It can be attributed to deformation of shell in the area of flange hub and subsequent inclination of flange blades (see Fig. 4). In bolts bending tension dominates over bolts simple tension, a phenomenon typical for FLT joints. Rate of decrease of working load depends on gasket material. It was found that the rigid gasket, defined in the FEM as “Flexible”, leads to lower values of flange rotation (approx. 0.1°), and thus to less additional bending of the bolts. Less additional bending results in a lower value of the bolt working load. However, the greater rigidity of the bearing surface paradoxically leads to an earlier loss of tightness of the joint (see Fig. 9). Even a relatively small angular rotation of the flanges is reflected in the rigid seal by a sharp decline of the normal sealing pressure from the inside of the seal (see Fig. 11). For gaskets from expanded PTFE, increase of working load in the first phase of working cycle can be observed. Further increase of working load occurs in the final phase of pressurising for all types of examined gaskets.

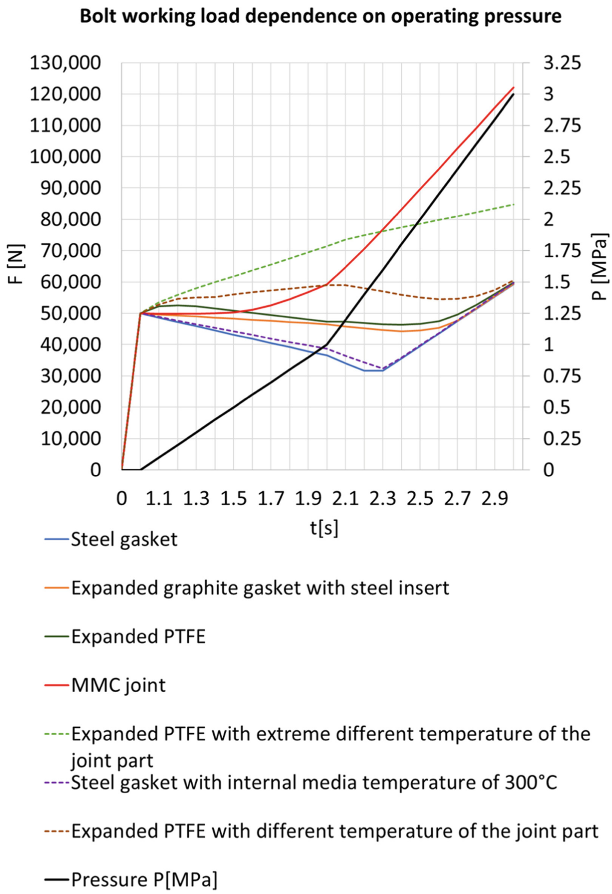

Fig. 9. Bolt working load dependence on operating pressure.

These "break points" of the graph (e.g., for steel gasket in 2.3 s in Fig. 9) show a state where the tightness of the joint has been lost (ie. The normal pressure on the seal has been lost over the entire width of the seal).

When MMC joints are implemented (see Figs. 7 and 8), only rise of working load occurs during pressurising of the internal parts of the vessel. It has to be noted that this result is only valid for MMC joints, which does not have a gap between flange blades. FEM calculation showed that flow profile of working load (Fig. 6) is almost identical with that of FLT with steel gasket.

The flow profile of working load is affected not only by the gasket material and joint construction type (FLT or MMC), but also by characteristic dimensions of the flange joint (e.g. thickness of shell, hub, flange blades, etc.), and most importantly by operation temperature. Examples of the effect of temperature on working load are shown in dashed lines. The first example depicts the maximum difference between bolt and flange temperatures of 170 °C using a PTFE gasket. The second example depicts the profile of working load affected by temperature of internal medium of 300 °C. In this case, a steel gasket was used. Temperature of other joint parts is increased by the virtue of heat transfer, where it was to simulate approaching the working temperature of the medium without following the recommended heating rate (5 °C per minute). Even though the temperature difference between a bolt and flange can reach up to 130 °C, it does not have a significant impact on the working load profile (see Fig. 9). The cause thereof is likely the fact that while the thermal deformation of flanges results in increase of working load, deformation by load largely reduces this effect through inclination of flanges. In the last case (in Fig. 9), a PTFE gasket is again implemented, this time the joint parts have slightly different coefficient of thermal expansion.

Results in Fig. 9 suggest that the flow of working load affects a number of variables. Thus, even knowing all the operation conditions, a reliable data for bolt loading can only be obtained experimentally or using FEM. A standard (e.g. CSN 1591-1) can be utilised to design and verify a flange joint of a pressure vessel, however, it is necessary to be aware of the limitations of the standards. Moreover, many of the operation factors cannot be accounted for in the standards-based computations.

3 Results and Discussing

Despite a wide range of advanced technologies (materials, tightening mechanisms, special bolts, etc.) and a number of standards for calculations concerning flange joints of pressure vessels, occasional occurrence of defects (many of which are trivial) still has not been eliminated. This chapter lists just a minor sample of the most common ones.

An issue arises with a gasket of a flange joint. In spite of prior intact operation, following a purchase of new gasket it stops to deliver proper tightness. The problem occurred even though a verified approach had been followed. After analysis of the problem, it was established that the new gasket has entirely different properties. In order for the new gasket to function properly, a significantly higher gasket pressure has to be achieved. Hence, a new calculation is conducted. Considerably higher values of bolt pretension are obtained. Given financial considerations, it is preferred to keep the original bolting. As a consequence, already in the assembly condition, the shear limit is used up to 90%. Subsequently, testing overpressure is applied and even though the vessel appears to be sealed, a leakage is discovered upon repeated pressurizing. This is due the fact, that with using an FLT joint, working load did in fact decrease in the pressure test, however, at the same time, bolt bending load rose. Added bend caused permanent deformation of bolts. Therefore, in the second test, the bolted joint was unable to be leak-proof. Subsequently, a replacement of the bolts follows. A better option is a choice of bolts with higher shear limit, rather than with higher rigidity. In some cases, the problem is solved by the replacement of bolts. If not, the cause might be the flanges themselves. The existing pressure vessels were designed several decades ago and were not constructed to such high pressures. Gaskets for which the joints were designed are no longer manufactures or their use might be prohibited. Connected shell is too thin, rigidity of flanges is, hence, too small. As soon as in assembly condition, a substantial inclination of flanges occurs. In further loading states, the situation might be further severed. Due to the inclination of the flanges (Fig. 4), not only rise of added bending of bolts is generated, but also effective gasket area is reduced. Figure 10 depicts compression of gasket from expanded graphite in assembly condition. Inclination of flanges results in higher load exerted on gaskets from the outer side.

Upon rise in pressure, further drop in loading of gaskets occurs, predominantly from the inner side (see Fig. 11). In assembly condition, inclination of the upper flange blade equals 0.35°, the maximum inclination thereof in operation condition is then 0.72°. For some types of gaskets, the manufacturer provides maximum permissible value of flange inclination. In such case, its calculation and verification is needed (CSN EN 1591-1 2015) Appendix C, pg. 42. During such verification, data from a test of leakage need to be used as basis according to (CSN EN 13 555 2005). Figure 11 depicts a gasket which is loaded due to inclination load only from one side. Thus, effective gasket area is very low.

The discussed defects of flange joints are related to a false design. In most cases, these designs are outdated and are not founded to any valid standard. It was proven, that flanges scaled according to ASME (ASME Code, Section VIII 2013), exhibit very good properties of joint tightness.

A number of malfunctions occurs in assembly of joints. Even a properly designed flange joint, including parameters of permissible loading and bolt pretension, and minimum friction force, fails to perform, unless their assembly procedure is followed, as an improper procedure can render it malfunctioning. The causes are for example uncontrolled tightening, not using washers, choice of improper lubricant, poor knowledge of properties of bolt joints etc.

Furthermore, it is worth mentioning that a crash of a flange joints of pressure vessels often occurs because of poor knowledge of operation temperatures. Faults on thermal

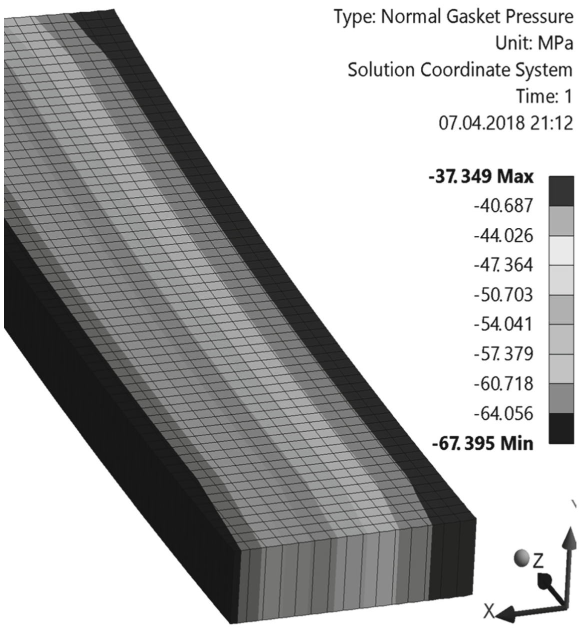

Fig. 10. Normal gasket pressure in assembly condition.

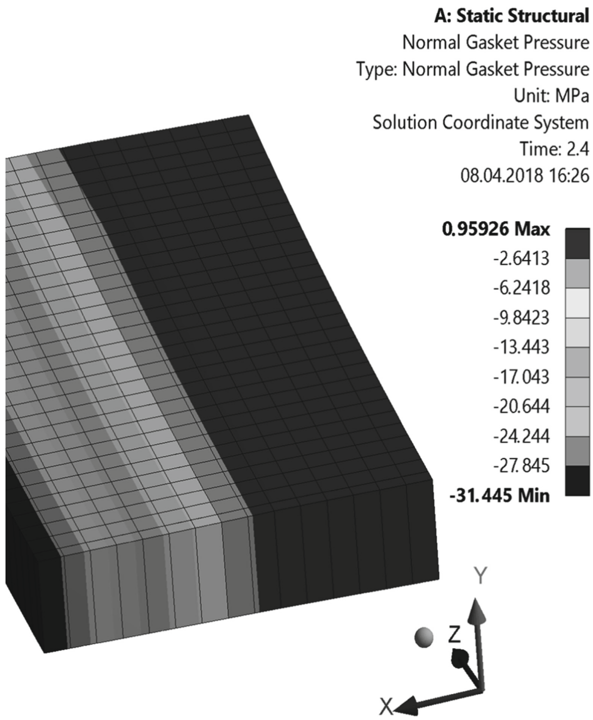

Fig. 11. Normal gasket pressure in subsequent condition.

insulation lead to high temperature differences in joining parts (see Fig. 9). A malfunction of compensators of thermal expansion can also have severe consequences.

4 Conclusions

Based on calculation using FEM, standardised methods and empirical knowledge, the article addresses the issue of critical joints of pressure vessels. The first part summarises fundamental information concerning flange joints. The information serves as introduction to the topic. Subsequently an analysis of flange joints using FEM is presented, in which data of working load flow is evaluated in relation to the internal pressure of medium. The results of the research suggest that a reliable description of working load flow and load of other machine elements can only be obtained experimentally or using FEM. This is given by the fact that technical practice includes many difficult or undefined load conditions. At the same time, there is a high number of constructionally atypical flange joints, for which methodology of standards is not defined. It is highly recommended to combine all the listed methods (FEM, experiment, standards). In the final section, the most commonly observed defects of flange joints are described. It was shown that the problems are most frequently caused by the human factor, whereby a poor knowledge of fundamental properties of flange joints or outdated methodology is manifested. Solution of the majority of the problems could be achieved by using valid standards in design. It is apparent, that progress in the field of critical flange joints has not reached the maximum. In order to achieve an improvement, an often negligent approach of managing personnel needs to be altered.

References

Pospisil, F. Threaded and bolt connections (in Czech). Praha: SNTL, 1968.

Kanaval, J. and Cezova, E. and Klima, V. and Stary, F. Testing Equipment for Complex Analysis of Screw Fasteners. In: D. Herak, ed. Proceeding of the 58th ICMD, Prague, September, 2017. Prague: Czech University Life Sciences, Dept Systems Eng, Kamycka 129, Prague 6 165 21, Czech Republic, pp 140-143. ISBN 978-80-213-2769-6

Jancar, L. and Mosler, V. and Folta, Z. Experimental Methods with Strain Gauges for Study Purposes. In: D. Herak, ed. Proceeding of the 58th ICMD, Prague, September, 2017. Prague: Czech University Life Sciences, Dept Systems Eng, Kamycka 129, Prague 6 165 21, Czech Republic, pp 124-127.ISBN 978-80-213-2769-6

Zacal, J. Problem solving of Bolted Connection of a Pressure Vessel. OSD002. Ostrava: VŠB – Technical University of Ostrava, Department of Machine Parts and Mechanism, 2016.

CSN EN 1591–1, Flange and their joints – Design rules for gasket circular flange connections – Part 1. Praha: UNMZ, May 2015.

CSN EN 13 555, Flanges and their joints – Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connections. Praha: CNI (Czech Standards Institute), August 2005.

ASME Boiler and Pressure Vessel Code, Section VIII, Rules for Construction of Pressure Vessels. Division 1. New York: ASME, July 2013.

Schaaf, M., Bartonicek, J. Calculation of Bolted Flange Connection – State of the Art (F533) [online]. August 17–22, 2003, date of last revision [2003–08–17]. Available from <http://www.lib.ncsu.edu/resolver/1840.20/27149>. F06 – Bolted connections and gaskets