



standard for the solution of a cylindrical or conical shape of shell, or what differences can be expected from the point of view of tightness and bolt working load. So, in this case, using the Final Element Method (FEM) is appropriate.

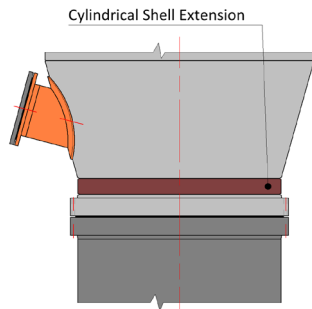


Figure 3. Design modification of the pressure vessel [Zacal 2020]

For comparison purpose it is proper to analyse variant without cylindrical shell extension and simple cylindrical shape of shell too. Tab. 1 shows the loads parameters. The load conditions are designated by numbers. The first column characterizes the assembly condition, the other one depicts subsequent conditions. The parameters marked  $M_A$  and  $F_A$  define external loads of pressure vessel, affect from bending moment and axial force (see Fig. 2). These external additional loads (e.g. from the thermal expansion of the connected pipe) were defined according to standard [CSN EN ISO 1591-1 2015] page 17, even for FEM analysis. Depicted values in Tab. 1 was designed for purpose of analysing.

Load Conditions	I=0	I=1	I=2
P [MPa]	0	1.014	1.014
T [C°]	22	22	200
$M_A$ [kNm]	-	-	143
$F_A$ [kN]	-	-	3.8

Table 1. Table with specification of load conditions

First, the influence of pressure vessel shape of shell to bolt working load was analysed. Very often is, an opinion widely spread among laymen, that bolt working load always increases in case of pressure vessels which are loads by a positive inner pressure. However, the reality differs from this opinion. Behaviour of bolt working load in operation condition is dependent on stiffness (i.e. design) of the joint parts (flanges, gasket, shell, and the bolts) and on the nature of the deformation due to operating load (i.e. position of operating forces). In this case, the nature of deformation of pressure vessel shell is affected by own shape too. Determination of these aspects in engineering practice is either very complex or even not possible. Due to this reality the FEM was used.

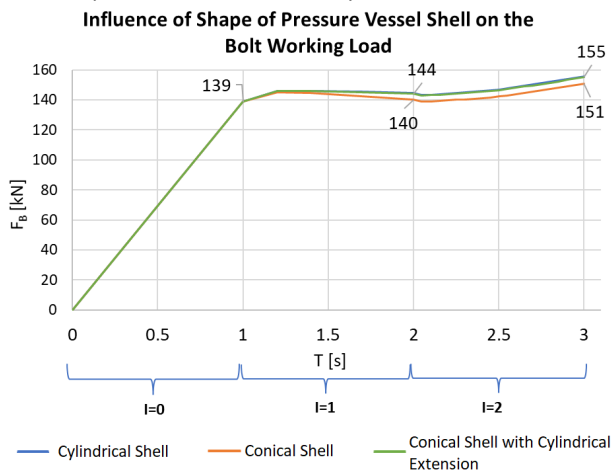


Figure 4. Influence of shape of pressure vessel shell on the bolts load

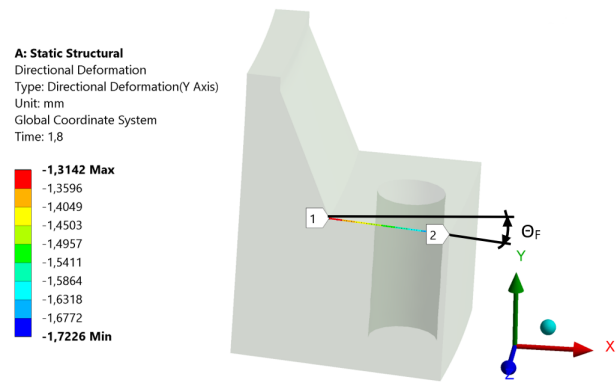


Figure 5. Angular flange inclination in FEM [Zacal 2020]

Figure 4, taken from [Zacal 2020], depicts the behaviour of bolt working load  $F_B$  in operation conditions by FEM. Decreasing trend of bolt working load in first load condition (I=1), in case of circular flange joints, is not uncommon. This behaviour is typical for floating type of joint (FLT) and metal to metal connection (MMC) with gap between flange blades [Schaaf 2003]. This decreasing trend is caused by nature of shell deformation in area of the flange hub and subsequent inclination of flanges (see Fig.5), during chamber pressurization.

Influence of Shape of Pressure Vessel Shell on the Tightness

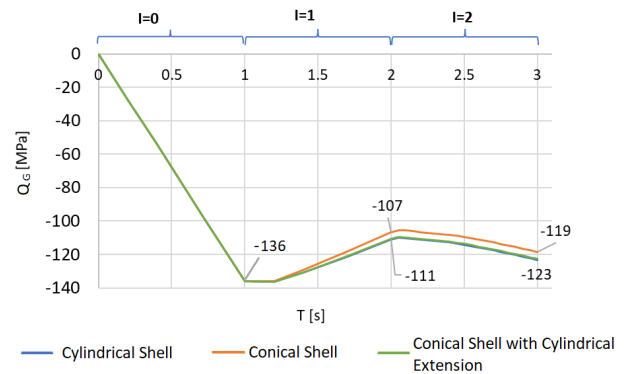


Figure 6. Influence of shape of pressure vessel shell on the tightness

Next, the influence of pressure vessel shape of shell to joint tightness was analysed. Figure 6 shows the behaviour of mean normal gasket pressure  $Q_G$  in operating conditions by FEM. The values of normal gasket pressure in operating conditions correspond to leak rate  $L=10^{-5} \text{ mg}/(\text{s}\cdot\text{m})$ , which was defined by the standard [CSN EN 13 555 2005]. Such a high tightness was by achieved sufficient preload in assembly conditions. The results of behaviour of the bolt working load and joint tightness are proper to complete by inclination of flange blades (see Fig. 5). This parameter is very important from view of leak rate requirement. The FEM result of angle of flanges inclination is depicted in Fig. 7.

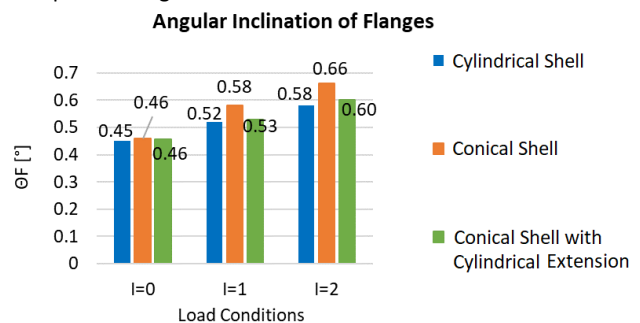


Figure 7. Results of angular inclination of flanges by FEM [Zacal 2020]

Due to the fact, that these analysed pressure vessels were symmetrical, it was possible to use symmetry in the FEM calculation. The APDL CPCYC command (1) was used for this purpose.

$$cpcyc, Lab, Toler, Kcn, dx, dy, dz, Knorrot \quad (1)$$

Where *cpcyc* couples of nodes of the two side faces of cyclically symmetric model, *Lab* is degree of freedom label for coupled nodes, *Toler* is tolerance of coincidence set to value 0.0001 by default, *Kcn* defines number of the coordinate system, (*dx*, *dy*, *dz*) defines degree of displacement to the *Kcn* (11.739°) and value of *Knorrot* was set to zero (i.e. the nodes of coupled sets are rotated into *Kcn*).

The FEM results were further extended by a calculation defined by standard CSN EN 1591-1. The results of the standard calculation are shown in Tab. 2 [Zacal 2020].

	Cylindrical Shell			Conical Shell		
	I=0	I=1	I=2	I=0	I=1	I=2
<b>CSN EN 1591-1</b>						
$F_B$ [kN]	139	130	161	139	126	156
$Q_G$ [MPa]	131	93	118	135	91	116
$\theta_F$ [°]	0.36	0.41	0.52	0.40	0.49	0.62
<b>FEM results</b>						
$F_B$ [kN]	139	144	155	139	140	151
$Q_G$ [MPa]	136	111	123	136	107	119
$\theta_F$ [°]	0.45	0.52	0.58	0.46	0.58	0.66

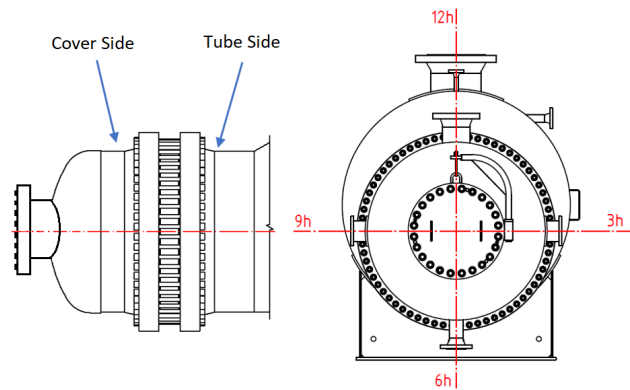
**Table 2.** Table with a comparison of analysis results

From the comparison of the results of FEM (Fig. 4, Fig. 6 and Fig. 7) and results of the standard (see Tab. 2), it is possible to see certain differences in the values of bolt working load in the state (I=1). While by standard there was a decrease of bolt working load after pressurizing, the FEM results indicate a slight increase of this value. However, both the standard calculation and the FEM approach show a greater susceptibility of the conical shape of shell to causing more decrease of bolt working load under pressure from the inner medium. This decrease also affects the final load in the bolts in the state (I = 2). Next, the results of normal gasket pressure (see Fig. 6) and the results of the mean value of angular inclination of the flanges (see Fig. 7) nearly correspond to the values in tab. 2, which was given by the standard. It can therefore be stated that the methodology of CSN EN 1591-1 is able to capture relatively accurately the effect of a symmetrically shaped shell connected to a flange, whether it is a cylindrical or conical connection.

The overlap curve (see Fig. 4 and Fig. 6) also confirms that a flange with an added cylindrical shell extension (see Fig. 3) can be considered as a standard connection of a cylindrical shell, without the influence of the conical shell connected above.

### 3 ASYMMETRICAL SHAPE OF PRESSURE VESSEL SHELL

Next FEM analysis was aimed to the pressure vessel with asymmetrical conical shape of shell from tube side (see Fig. 8).

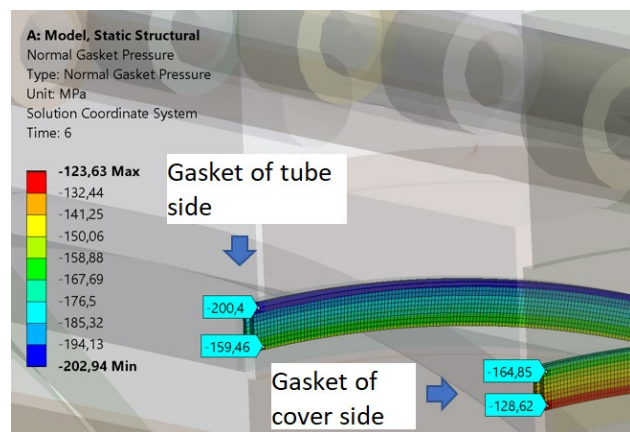


**Figure 8.** Asymmetrical conical shape of the pressure vessel shell

In this case, the volume of symmetry simplification in the FEM was considerably limited to half.

This research was based on real situation in practice, where the circular flange joint of ethylene condenser repeatedly leaked out during the pressure test in position 3h and 9h (see Fig. 8).

A detailed analysis proved a negative influence of asymmetrical shape of conical shell in problematic position 9h and 3h. The deformation character of the asymmetrical conical shell also caused an asymmetrical deformation of flange. This behaviour was watched only on the flange of tube side. Asymmetrical deformation leads to uneven sealing gasket pressure. The normal gasket pressure difference between position 12h and 3h was always approximately 20 MPa, see Fig. 9 and Fig. 10 taken from [Zacal 2020]. From view of gasket tightness, it may seem that this difference is not very significant. But it is important to remember, that in some causes in technical practice, much lower sealing pressures can be defined, differences even less than 20 MPa can be critical for tightness or required leak rate of joint. Furthermore, analysis by FEM have own limits compared to reality (e.g. geometrical accuracy of 3D model, accuracy of pretension of the bolts, some simplify operating conditions etc.).



**Figure 9.** Normal gasket pressure in pos. 12h in operating conditions

It can be simply imagined that the flange of tube side is deformed around the imaginary axis 3h and 9h (see Fig. 8), due to the asymmetrical conical shape of shell, which leads to the problem of the joint tightness close to this axis.

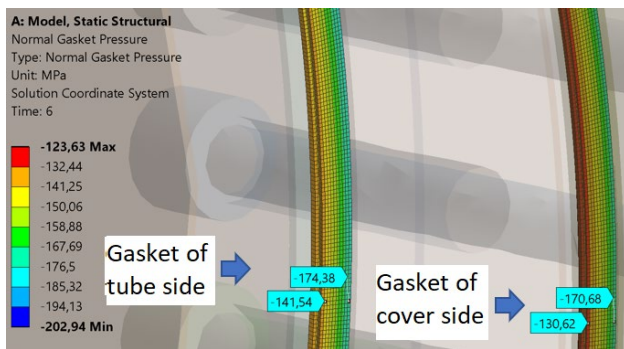


Figure 10. Normal gasket pressure in pos. 3h in operating conditions

The values of normal gasket pressure for tube side flange in position 6h are almost identical to values in position 12h. The same is true for position 3h and 9h.

The influence of asymmetrical shell depends also character of bolt working load. The bolts in position 3h and 9h show a significant increase in load.

From Fig. 9 and Fig. 10 is also clear, that gasket of cover side with symmetric shell is compressed evenly in all position. That confirm again the negative influence of asymmetry shape of shell.

#### 4 CONCLUSION

The results obtained in the research of influence of shape of pressure vessel shell on bolt working load and joint tightness shows the possible effect these aspects on the function of flange connections in technical practice.

From current research it is obvious, that the methodology of CSN EN 1591-1 is able to capture relatively accurately the effect of a symmetrically shaped shell connected to a flange, whether it is a cylindrical or conical connection. In this case, the use of FEM is not necessary.

The problem only occurs in case of asymmetrical shape of pressure vessel shell. Standard definitions of CSN EN 1591-1 does not provide a calculation methodology for an asymmetrical shell. Due to this reality the FEM was used. A detailed analysis proved a negative influence of asymmetrical shape of conical shell in problematic position 9h and 3h (see Fig. 8). This negative effect is reflected in decrease of normal gasket pressure and significant increase in bolt working load. Using FEM was designed possible solution for elimination of negative influence of the asymmetrical shape of pressure vessel shell. New design contained improvement from view of flanges stiffness. In practice, the use of a higher class of neck flanges according to the ASME Boiler and Pressure Vessel Code [ASME Code, Section VIII 2013] has proved successful. The possibility of reinforcing the conical shell at a critical area of deformation was also tested. However, it is necessary to remember that this is a nonstandard, therefore unsubstantiated modification. The next step in the future is to verify these results by experimentally using strain gauges [Jancar 2017 et al.].

#### ACKNOWLEDGMENTS

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