Documents

Calculation Bases of EN 1591-1 method

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Foreword

In design, the strength calculations corresponds to check that the applied loads are admissible :

Applied loads <= Allowable loads

Classically, the verification is based on the theory of linear elasticity and the associated criteria are often found as following :

Applied stresses <= Allowable stresses

Besides, to achieve an appropriate leak-tightness the surface pressure applied on the gasket Q must be higher than a given value. In the ASME Boiler and Pressure Vessel code, the following criterion is given :

$Q >= m \times | P |$

where m is the gasket factor and P the fluid pressure.

Generally, a leak-tightness criterion can be written as follow :

Applied loads >= Required loads

As a consequence lower and upper limits must be respected by the loads applied on gasket and bolts :

Required loads <= Applied loads <= Allowable loads

(In ASME code it is assumed that applied load = required load)

In the framework of the Pressure Equipment Directive PED 97/23/EC "*New approach directive*", the CEN Technical Committee TC74 "*Flanges and their joints*" prepared a new calculation standard : EN1591 : "*Flanges and their joints - Design rules for gasketed circular flange connections - Part 1 : calculation method , Part 2 : gasket parameters*".

The aim of the EN 1591 calculation method is to verify both leak-tightness and strength criteria. The method does not only consider basic calculation parameters such as :

- fluid pressure,
- mechanical strength values of flanges, bolts and gaskets,
- gasket compression factors;
- nominal bolt load,

but also :

- possible scatter due to bolting-up procedure,
- changes in gasket force due to deformation of all components of the joint,
- influence of connected shell or pipe,
- effect of external axial forces and bending moments,
- differential axial thermal expansion between the flanges and the bolts.

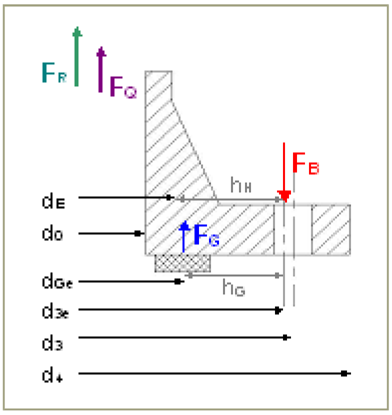
Mechanical model

In the calculation method of the EN1591-1, the behaviour of the whole Flanges-bolts-gasket system is considered in an axisymmetric mechanical model. The calculation is not only based on a forces balance, it also considers a deformation balance and rheological laws of the bolted flanged connection components. The calculation is organized in calculation conditions numbered with the letter I. The calculation is performed using the forces and deformations balances between the initial calculation condition : assembly condition (I=0), which is the reference state, and a subsequent condition I. The forces and deformations are determined at subsequent calculation conditions (typically : test condition (I=1), design condition (I=2), operating condition n°1 (I=3), ...).

Flanges

3 different flange configurations are treated in the EN1591 :

- Integral flanges
- Loose flanges and collars
- Blank flanges

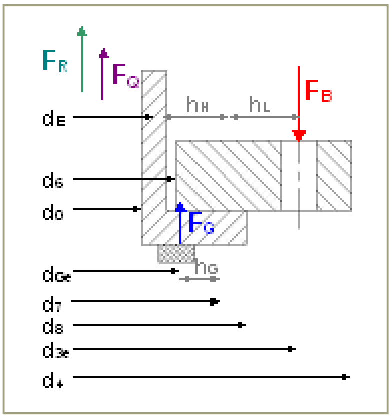


Integral flange

The flange is treated like a rectangular ring cross section. The ring cross section remains undeformed.

For integral flanges and collar, the ring cross section is considered to be connected to an equivalent cylindrical shell. A tapered hub is treated as being an equivalent cylindrical shell whose thickness is calculated. For flanges without hub, the dimension of the equivalent cylindrical shell are those of the connected shell.

Effective dimensions of the flanges :



Loose flange

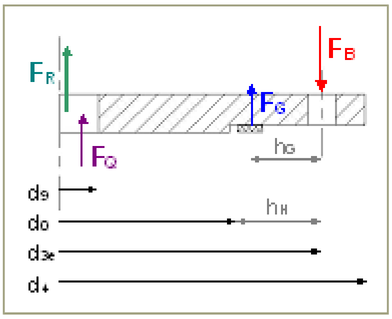
In the calculation of the width of the rectangular ring cross section, the bolts holes are partially subtracted :

$$b_F = (d_4 - d_0)/2 - d_{5e}$$

When the pitch between bolts is small, d_{5e} is close to d_5, when the pitch between bolts is high, d_{5e} is close to 0.

The effective thickness of the rectangular ring cross section can be obtained by dividing the cross section area of the ring A_F or A_L by the calculated radial width of this section.

An effective bolt circle diameter is also considered in order to take into account the discrepancy between the arc of a circle and the string.



Blank flange

$$d_{3e} = d_3 \cdot \left(1 - \frac{2}{n_B^{\,2}}\right)$$

The relation between the flange deformation and the load applied on the flange is :

$$\theta_F = \frac{Z_F}{E_F} \cdot M_F$$

$$M_F = F_G \cdot h_G + F_Q \cdot (h_H - h_P + h_Q) + F_R \cdot (h_H + h_R)$$

where Θ_F is the rotation angle of the flange, Z_F is the flexibility modulus of the flange, E_F is the Young modulus of the flange and M_F is the rotational moment applied on the flange.

In the case of loose flange :

$$\theta_L = \frac{Z_L}{E_L} \cdot M_L$$

$$M_L = F_B \cdot h_L$$

where Θ_L is the rotation angle of the loose flange, Z_L is the flexibility modulus of the loose flange, E_L is the Young modulus of the loose flange and M_L is the rotational moment applied on the loose flange.

These rotation angles can be determined at every calculation situation. If a maximum acceptable value of flange rotation is specified, the calculated values can be checked to ensure that they are below the maximum acceptable value.

Bolts

The relation between the bolt elongation and the bolt load is :

$$U_B = \frac{X_B}{E_B} \cdot F_B$$

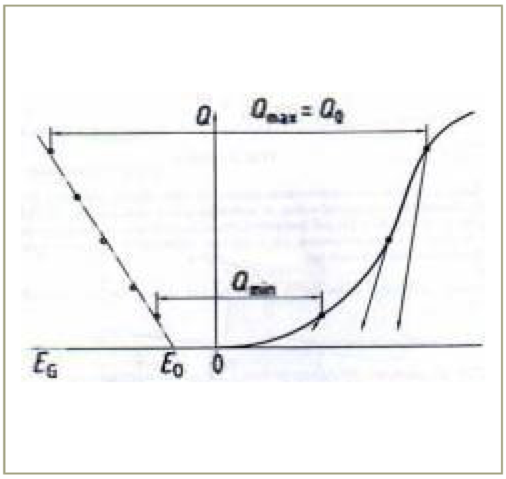
where U_B is the bolts elongation, X_B is the flexibility modulus of the bolts, E_B is the Young modulus of the bolts and F_B is the bolts load.

Gasket

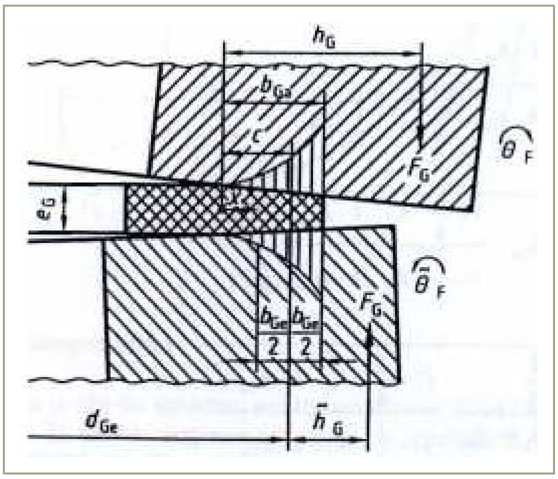
The relation between the gasket compression and the load on the gasket is :

$$\Delta U_{G0\to I} = -\frac{X_G}{E_G} \cdot \Delta F_{G0\to I}$$

where $\Delta U_{G0\to 1}$ is the variation of gasket thickness between the tightening situation and the situation I. A negative value corresponds to a compression and a positive one means a recovery of the gasket thickness. X_G is the flexibility modulus of the gasket, E_G is the elasticity modulus in decompression of the gasket and $\Delta F_{G0\to 1}$is the load variation on the gasket between the tightening situation and the situation I.



elasticity modulus



effective gasket width

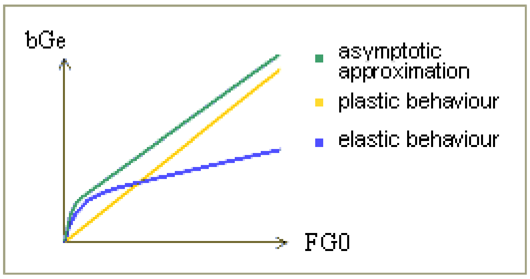
E_G is determined with the gasket parameters E_0 & K_1 and depends on the gasket surface pressure Q applied at bolting-up :

$$E_G = E_0 + K_1 \cdot Q$$

The gasket contacts the flange faces over a calculated annular area. The effective gasket width varies with the flange rotation. This rotation also leads to a non homogenous radial gasket stress.

The effective gasket width b_{Ge} is calculated for the assembly condition (I=0) and is assumed to be unchanged for all subsequent load conditions.

The calculation of b_{Ge} includes the elatic rotation of the flanges as well as the elastic and plastic deformations of the gasket. For gaskets with elastic behaviour, the evolution of the effective width of the gasket is a square root curve. For gaskets with plastic behaviour, the evolution of the effective width of the gasket is a straight line.



The expression of the calculated gasket width is an approximation which enable to consider both elastic and plastic behaviour.

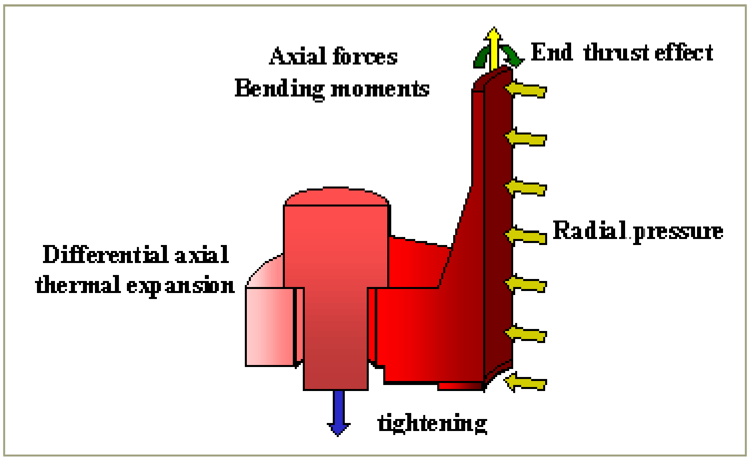
4 different types of gaskets are considered in the calculation of the effective dimensions :

- Flat gasket of low hardness, composite or pure metallic materials
- Metal gaskets with curved surfaces, simple contact
- Metal octogonal section gaskets
- Metal oval or circular section gaskets, double contact

Under compression and (or) at elevated temperature, the gasket may creep and gasket relaxation may occur.
In the EN 1591 calculation method, the creep relaxation phenomenon of the gasket is approximated by the factor g_c applied to E_G. It leads to a higher required tightening force in order to compensate the loose of reaction on the gasket due to the creep relaxation phenomenon.

Loads

The following loads are considered in the calculation, in each condition I:



Fluid pressure

internal ($P_I < 0$) or external ($P_I > 0$) pressure, resulting in a force:

$$F_{QI} = (\pi/4) \times d_{Ge}^2 \times P_I$$

d_{Ge} is the location of the forces acting on the gasket and not the location where the leak tightness is achieved. This is conservative, overestimating the load coming from the pressure of the fluid for large gasket width.

Radial effect of the internal pressure

The radial effect of the internal pressure is considered in the loading of the flange ring cross section with the term :

$$P \cdot \int_0^{e_P} r(z) \cdot dz$$

where eP is the thickness of the flange ring cross section submitted to internal pressure. We find this effect in the expression of lever arms correction such as hP or hQ.

External loads

axial tensile ($F_{AI}>0$) or compressive ($F_{AI}<0$) forces, and bending moments M_{AI}, resulting in a force:

$$F_{RI} = F_{AI} \text{ +or- } (4/d_{3e}) \times M_{AI}$$

In the presence of external moment, both sides of the joint are considered:

- on the side where the moment induces an additional tensile load (sign '+'), load limits of flanges or bolts may govern, as well as minimum gasket compression
- on the side where the moment induces an additional compression load (sign '-'), load limit of gasket may be decisive.

Differential axial thermal expansion between the bolts and the flanges

The differential axial thermal expansion between the bolts and the flanges is given by the following expression :

$$\Delta U_{\mathrm{I}} = l_{\mathrm{B}} \times \alpha_{\mathrm{BI}} \times \left(T_{\mathrm{BI}} - T_0\right) - e_{\mathrm{Ft}} \times \alpha_{\mathrm{FI}} \times \left(T_{\mathrm{FI}} - T_0\right) - e_{\mathrm{L}} \times \alpha_{\mathrm{LI}} \times \left(T_{\mathrm{LI}} - T_0\right)$$
$$- e_{\mathrm{G}} \times \alpha_{\mathrm{GI}} \times \left(T_{\mathrm{GI}} - T_0\right) - \tilde{e}_{\mathrm{Ft}} \times \tilde{\alpha}_{\mathrm{FI}} \times \left(\tilde{T}_{\mathrm{Ft}} - T_0\right) - \tilde{e}_{\mathrm{L}} \times \tilde{\alpha}_{\mathrm{LI}} \times \left(\tilde{T}_{\mathrm{LI}} - T_0\right)$$

where $T_{B,G,F,L}$ and $\alpha_{B,G,F,L}$ are respectively the temperature and the thermal expansion factor of the corresponding elements.

e_{Ft} is the width of the flange to be considered in thermal expansion, possibly including the thickness of the washers (it is assumed that they have the same temperature and thermal expansion coefficient as the flange)

Forces and Deformation balances

At every calculation condition I a forces balance is established between the bolt load, the gasket reaction, the resulting force due to the external loads, the resulting force due to the internal pressure :

$$F_{BI} = F_{GI} + F_{QI} + F_{RI}$$

At assembly condition as well as for all the subsequent calculation conditions, the bolted flange connection components are joint together by the internal forces. It leads to the following geometrical relation between the component displacements :

$$\left\{\theta_F \cdot h_G + \tilde{\theta}_F \cdot \tilde{h}_G + \theta_L \cdot h_L + \tilde{\theta}_L \cdot \tilde{h}_L + U_B + U_G\right\}_{I=0} = \left\{\theta_F \cdot h_G + \tilde{\theta}_F \cdot \tilde{h}_G + \theta_L \cdot h_L + \tilde{\theta}_L \cdot \tilde{h}_L + U_B + U\right.$$

From these 2 balances, the fundamental equation which links the forces variation in a bolted flange connection is obtained :

$$F_{G0} \cdot Y_{G0} + F_{Q0} \cdot Y_{Q0} + F_{R0} \cdot Y_{R0} = F_{GI} \cdot Y_{GI} + F_{QI} \cdot Y_{QI} + F_{RI} \cdot Y_{RI} + \Delta U_I$$

Required tightening force : $F_{B0\ req}$

The EN 1591 calculation is based on integrity criteria.

A minimum tightening force is determined by considering both seating and leak-tightness criteria.
At assembly condition (I=0) : the gasket surface pressure must be higher than Qmin (seating criterion).
At all the subsequent conditions (I>0) : the gasket surface pressure must be higher than Q_I (leak-tightness criterion).

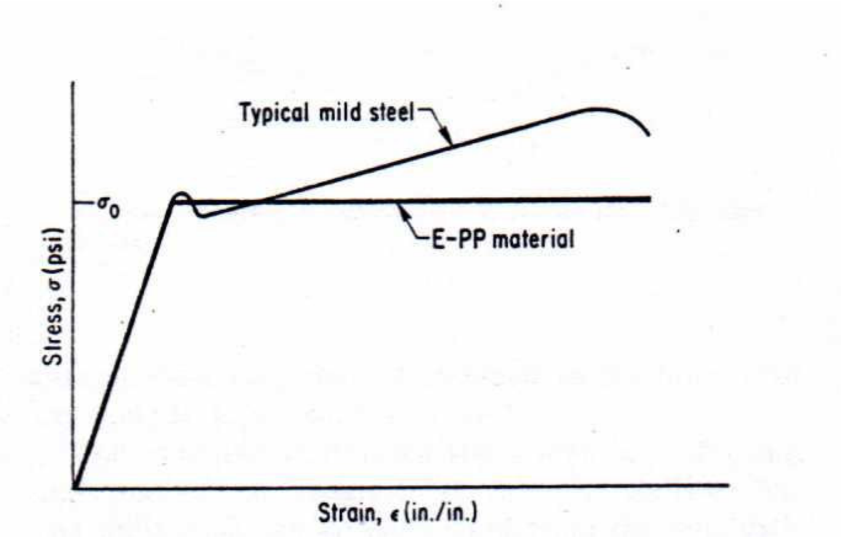
If leak-tightness test results are not available, Q_I can be determined with P_I and the m value.
When leak-tightness test results are available, Q_I can be determined in order to maintain the required leak-rate for the given pressure, temperature and maximum gasket surface pressure applied previously.

From the Q_{min} and Q_I values, we determine the required tightening force $F_{B0\ req}$.
An iterative calculation must be performed until the assumed tightening force in the calculation of the effective dimensions corresponds to the calculated required tightening force.

Load rates

Generally speaking, several types of damage can affect components : excessive deformations, creep, erosion/corrosion, fatigue…

In the EN 1591 the strength criteria are based on the limitation of excessive deformations. The creep of flanges and bolts as well as fatigue proof (usually not taken into account in such code) are not considered in EN1591.



In pressure equipment and static structures, deformation becomes excessive when the equipment dimensions increase much more rapidly than the load does. It leads to the definition of an excessive deformation threshold. Limit analysis theory defines a mathematical approach of the excessive deformation threshold. In this view, the material is considered elastic – perfectly plastic. The material is assumed to have an elastic behaviour until it reaches yield stress Sy. Then the stress remains constant at Sy regardless of the strain imposed.

In the EN1591, the strength criteria correspond to the verification that load rates are acceptable. Load rate in EN1591 can be defined as the ratio between the load applied on the considered component and the strength of the component. Since the load influences the strength of the component, there is no exact proportionality following :

Allowable load = (applied load) / (load rate)

This relation is verified only when the load rate is equal to 1.

The nominal design stresses to be used in the calculation of bolts and flanges load rates are not specified in EN 1591. They depend on other codes which are applied, for example these values are given in EN13445 and EN13480.

At tightening the load rates are calculated with F_{B0max} which is the tightening force taking into account the scattering due to the bolting up method.

For the subsequent calculation conditions, the forces to consider in the calculation of load rates are obtained from an assembly gasket force F_{G0d} which guarantee that the required gasket surface pressure is applied at all the calculation conditions. In the case of frequent re-assembly, accumulation of plastic deformations is limited.

Gasket load rate

The strength criterion on the gasket corresponds to a limitation of the gasket compression. The condition on the load rate given below must be verified :

$$\Phi_G = \frac{F_G / A_{Gt}}{Q_{max}} \leq 1$$

F_G:	reaction on the gasket
A_{Gt}:	theoretical gasket contact area
Q_{max}:	maximum allowable compressive stress on the gasket

Bolts load rate

The strength criterion on the bolts corresponds to a limitation of the bolts traction. The limit load equation for the bolts is the following :

$$\left(\frac{F_B}{A_B}\right)^2 + 3\cdot\left(\frac{M_{t,B}}{I_B}\right)^2 \leq f_B{}^2$$

F_B:	bolt load
A_B:	bolt cross section
C:	coefficient to take into account of the twisting moment in bolts
$M_{t,B}$:	twisting moment acting on bolt shanks (depending on the tightening device)
I_B:	plastic torsion modulus of bolt shanks
f_B:	nominal design stress of the bolts as defined and used in pressure vessel codes

The condition on the load rate given below must be verified :

$$\Phi_B = \frac{1}{f_B}\cdot\sqrt{\left(\frac{F_B}{A_B}\right)^2 + 3\cdot\left(C\cdot\frac{M_{t,B}}{I_B}\right)^2} \leq 1$$

The value C = 1 is based on a plastic limit criterion. Due to this criterion, some limited plastic strains may occur at periphery of the bolts in assembly condition.
The value C = 4/3 is based on an elastic limit criterion. It may be selected in the expression of the load rate if a strict elastic behaviour of the bolts is wished at bolting up.

Flanges load rates

The strength criterion on the flanges corresponds to a limitation of the flanges rotation. The radial cross section of the flange ring is considered undeformed. Only circumferential stresses and strains in the ring are treated; radial and axial stresses and strains are neglected.

For the flanges, the load ratios are calculated for the section of the flange ring or collar, of the loose flange (if there is one), and in some cases, for particularly critical sections.

Example of determination of flange with connected shell load rate :

If we consider S_T : circumferential stress in the ring and S_F the yield stress or the nominal design stress for the flange ring.

From the elasticity theory : we determine the resulting force and moment in the flange ring due to the deformation.

$$R_F = +\iint S_T\cdot dA$$
$$M_F = -\iint S_T\cdot z\cdot dA$$

At the limit load :

$$S_T = \pm S_F \quad for\ -e_Q \leq z < z_0$$
$$S_T = \pm S_F \quad for\ z_0 < z \leq +e_P$$

For a rectangular cross section :

$$R_F = \pm S_F\cdot b_F\cdot\left(2\cdot z_0 + e_Q - e_P\right)$$
$$M_F = \pm S_F\cdot b_F\cdot\left(\frac{1}{2}\cdot\left(e_Q{}^2 + e_P{}^2\right) - z_0{}^2\right)$$

The limit load equation for the flange ring is then :

$$\frac{4}{S_F \cdot b_F \cdot e_F^{\,2}} \cdot \left| M_F + R_F \cdot \left(e_P - \frac{e_F}{2} \right) \right| + \left(\frac{R_F}{S_F \cdot b_F \cdot e_F} \right)^2 = 1$$

We use the expression of RF and MF determined with the elasticity theory applied to the flange ring. We also use the force and moment expressions applied on the flange ring by the connected shell and we obtain the load rate expression of the flange with a connected shell.

Each load rate shall be less than or equal to unity for all calculation conditions.

For wide flanges, a more stringent requirement applies to integral flanges and loose flanges : the load rate shall be less than or equal to Φ max < 1.

Tightening recommendation

The bolting up method generates some degree of inaccuracy. That is why the targeted tightening force must be higher than the required tightening force. The EN 1591 considers the negative ε - and positive ε + scattering due to the bolting up method.

As a consequence, the actual bolt tightening force FB0 is limited as follow :

$$F_{B0\min} \le F_{B0} \le F_{B0\max}$$

with:

$$F_{B0\min} = F_{B0av} \cdot (1 - \varepsilon_-)$$

$$F_{B0\max} = F_{B0av} \cdot (1 + \varepsilon_+)$$

The nominal bolt assembly force must verify the following condition :

$$F_{B0nom} \ge \frac{F_{B0\min}}{(1 - \varepsilon_-)}$$

In the same way, the load rates at assembly condition are calculated with the following bolting up force.

$$F_{B0\max} = F_{B0nom} \cdot (1 + \varepsilon_+)$$

Tightening torque

To obtain the target bolt assembly force F_{B0nom}, the value of the torque to apply at tightening is given by the expression below :

$$M_{t,nom} = \frac{F_{B0nom}}{n_B} \cdot \left(\frac{p_t}{2\pi} + \frac{\mu_t \cdot d_t}{2\cos\alpha} + \frac{\mu_n \cdot d_n}{2} \right)$$

n_B:	number of bolts
d_n:	mean contact diameter under nut or bolt head
d_t:	mean contact diameter on thread
μ_n:	friction coefficient under nut or bolt head
μ_t:	friction coefficient on thread
p_t:	thread pitch
α:	half thread-angle

-- *Last update : 19 Feb. 2002*