Design of Bolted Flange Connections – Status of EN Standards*

Dr. Ing. Fernando Lidonnici

SANT'AMBROGIO SERVIZI INDUSTRIALI SRL

Piazza Carlo Donegani 8 – 20133 MILANO, Italy

Phone: +39.02.70603113 Fax: +39.02.2663546

E-mail: santambrogio@sant-ambrogio.it

www.sant-ambrogio.it

*** Rev.1 – 25/03/2022**

Which are the conditions to be fulfilled in order to seal a gasketed joint ?

A gasket is made of a **relatively soft and plastic material** (soft and plastic at least if compared to the material of the gasket seats). When this material is compressed against the seats by the tightening load it has **to fill all the possible channels through which the internal fluid, subject to its pressure, may reach the outside of the pressure chamber**.

Therefore there are **two conditions** in order to get a tight joint:

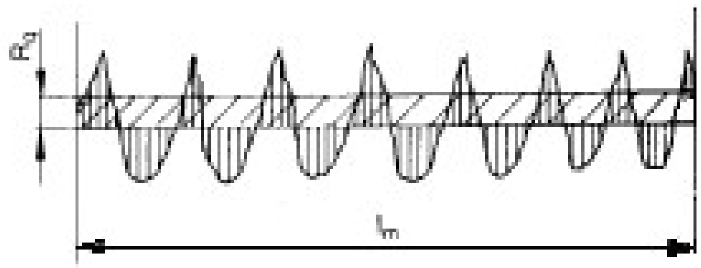
- During the **assembly** of the connection **the gasket must be subject to a pressure *y* (MPa) sufficient to get its yield limit in the local areas in contact with the seats**, so that all the possible leak channels are closed (this situation is generally known as **gasket seating).**

- During **test** and **operation** the **residual pressure on the gasket must be *m* times greater than the internal pressure** in order to avoid the loss of contact between gasket and seats.

Parameters that influence the correct gasket seating (1)

• **Physical state of the fluid to be sealed.** While it is theoretically possible to have a perfectly tight joint when the fluid to be sealed is a liquid, it is virtually impossible to get a tight joint when the fluid is a gas or a vapour. In fact liquids have a much higher viscosity than gases, so that the pressure drop of a possible flow through the microscopic leak channels at the surface between the gasket and the seats is so high that a leak cannot occur. Gases and vapours, on the contrary, tend to fill all the available space, whichever is the volume and the available flow area, so that their leak tightness can be evaluated only on the basis of a maximum allowable flow (generally given as mg/s/cm of gasket perimeter)

• **Surface roughness**. It has to be noted that the metallic surface of the seats (generally obtained by mechanical tooling) is very far from being perfectly smooth. Its appearance, if examined by means of a microscope, is like in the following figure:



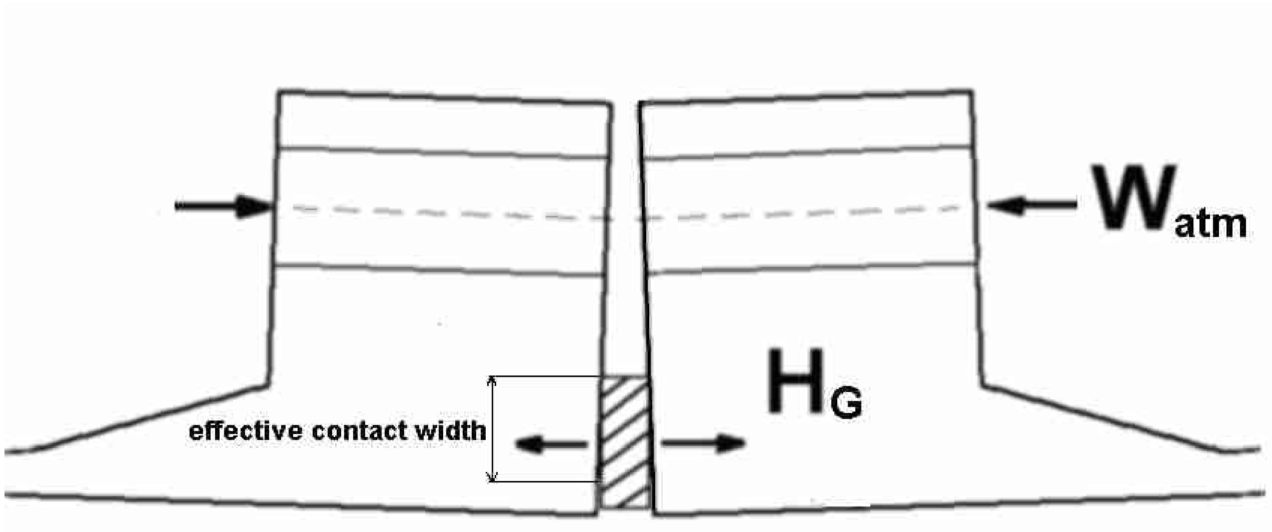
$$R_q = \sqrt{\frac{1}{l_m} \int_{x=0}^{x=l_m} y^2(x)\, dx}$$

The formula above gives the "**roughness**" of the surface, expressed in **RMS degrees** (= Root Mean Square), generally given in **microinches**, but can also be given in μ (1 microinch = 0,0254 μ). On the construction drawings the roughness is generally indicated by means of reversed triangles: 1 triangle up to 1000 microinches RMS, 2 triangles up to 400, 3 triangles up to 60, 4 triangles up to 8).

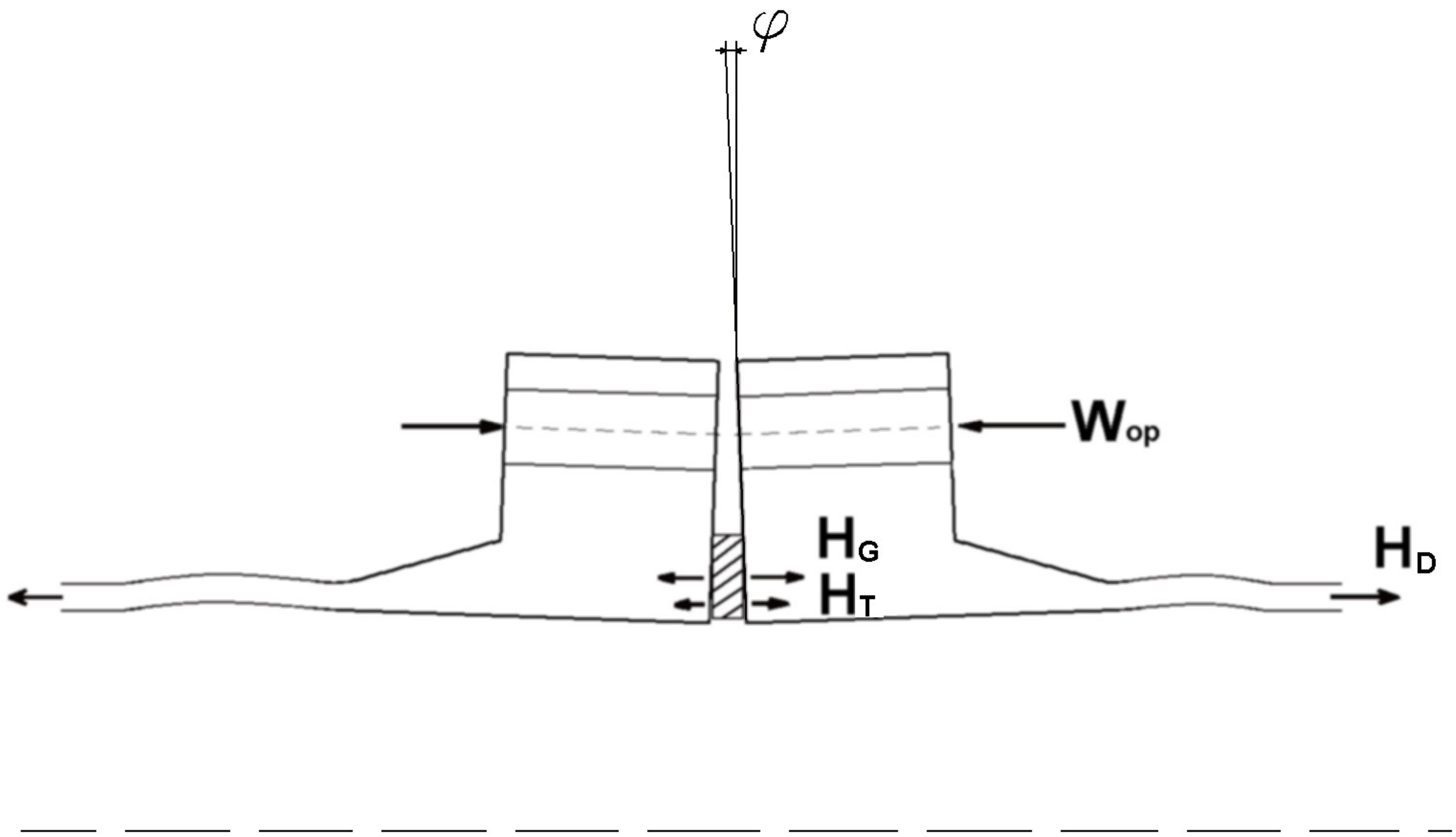
Parameters that influence the correct gasket seating (2)

- **Difference between the hardness of the gasket and the hardness of the seats.** In order to fill the gaps between the gasket and the seats the compressive load acting on the gasket must cause a plastic deformation of the same. This load will of course be higher if the gasket is hard. Note also that in the case of metallic gasket this condition is difficult to be achieved, particularly in the case of strain hardening materials (30 points of difference in the Brinell hardness are generally recommended). Gaskets harder than the seats will cause a permanent print on the seats which can compromise leak tightness in case of gasket replacement.

- **Effective area in contact with the seats**. This area is influenced by the amount of compressive load, which is causing an overall bending of the flanges during bolt tightening and subsequent application of pressure. The contact surface will be therefore reduced and displaced towards the outside because of the flange bending:



Loads and deformations on a flange assembly in service conditions



Equilibrium of a flange assembly in gasket seating and service (or test) conditions

Gasket seating (Bolt load = gasket Load)

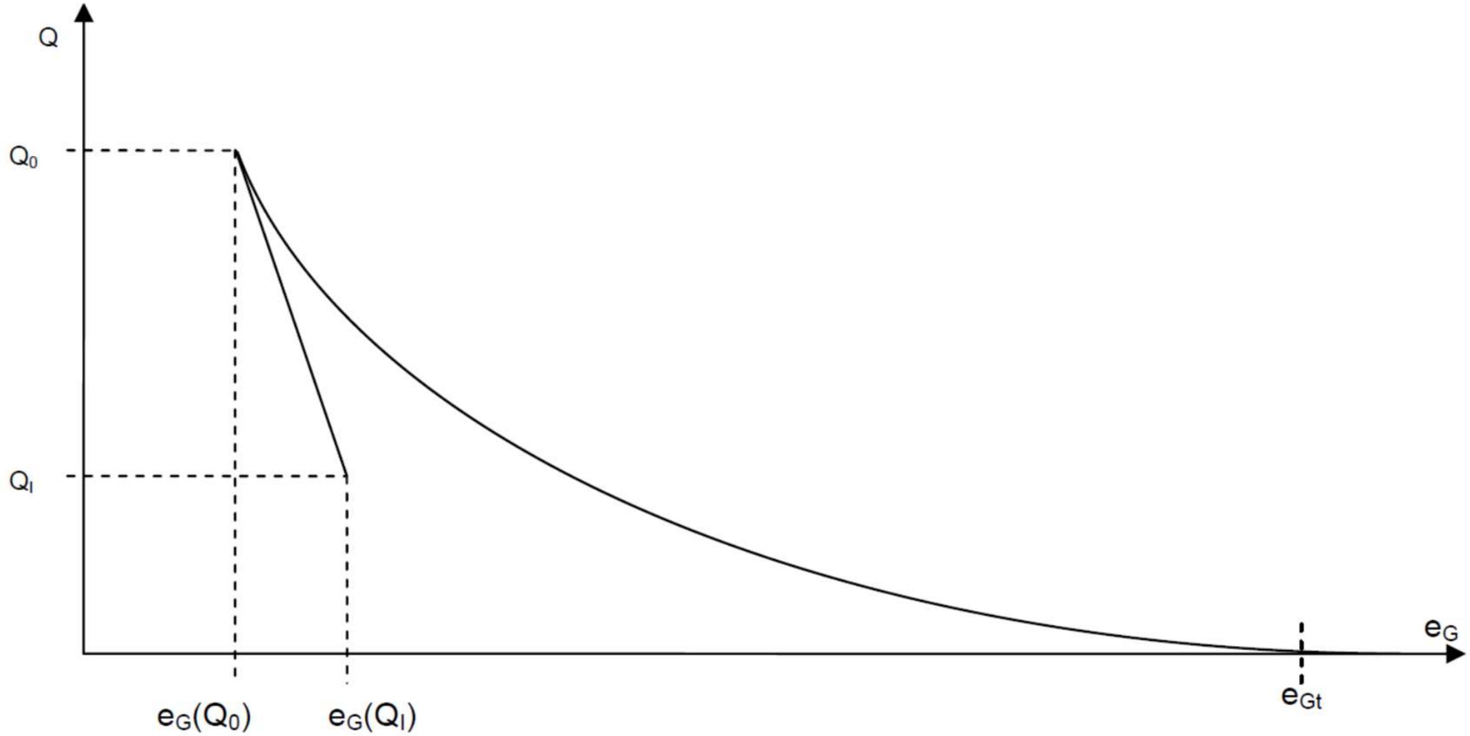
$$W_{at} = H_G$$

Service / Test (Bolt load = hydrostatic end load on flange ID + hydrostatic load on flange ring + gasket load)

$$W_{op} = H_D + H_T + H_G$$

What is the relationship among the bolt load applied during the assembly ("Bolting up") and the bolt loads existing in the subsequent operating and test conditions?

A bolted joint is a complex system composed by two flanges (or by a flange and a cover), a gasket and a series of bolts. **Bolts and flanges are metallic components whose behaviour under stress is generally elastic, while the behaviour of the gasket is only partially elastic, may be different in first time loading and subsequent unloading / reloading, and may also be subject to creep under constant load.** The figure shows the variation of the original gasket thickness e_{Gt} under the bolting up load Q_o and subsequent unloading to the bolt load Q_I. The **inelastic behaviour of the gasket** and the **variation of the gasket surface** under the bolt load are **the main reasons why it is nearly impossible to establish such a relationship.**



Is it really possible to control the bolt load in order to have the reasonable assurance that the gasket (at least at bolting up) has been subject to a predefined load?

The reality is that **all the devices used to tighten a gasketed joint have a positive and a negative tolerance**. If bolts are tightened by a manual wrench (or even a torque wrench) the relationship between the torque and the load is strongly dependent on the **friction factor** (the one between the bolt threads and the nuts and the one between the nut and the flange rear face). Supposing that the two factors have the same value μ, the following formula gives the torque M_o (Nmm) needed in order to develop a load F_o (N) on a bolt having a nominal diameter d_{BO} (mm) :

$$M_o = 1{,}2\,\mu\,d_{BO}\,F_o$$

Note that **μ may have values between 0,15 and 0,25**, so that the range is not negligible, also in the cases where you are able to measure the real value of the torque. But **also those devices** (generally hydraulic) **which put the bolts under direct load have a positive or negative tolerance**. Although **the scatter value of the load on a single bolt will decrease if you consider a bolted assembly made of several bolts**, you should consider a **minus tolerance** on the applied bolt load for the purpose of **gasket calculation** (a gasket not sufficiently compressed will leak), and a **plus tolerance** for the purpose of determining the bolt stress.

The Taylor Forge method – Conventional gasket width

The Taylor Forge method **was developed in U.S.A. in 1930**, but it is **still used in the great majority of Pressure Vessel standards** in the world (**ASME Section VIII division 1 and 2, PD 5500, CODAP, ISPESL VSR and EN 13445.3 - Clause 11**). It starts from the simplified assumption that **the minimum required bolt load has to be calculated separately for all the possible service conditions** (Gasket seating, Operation, Test), thus ignoring any relationship among them. The calculation starts from the **determination of a conventional effective gasket contact surface**. The loads on such surface have to be calculated using **conventional gasket factors *m* and *y* (MPa).** The table of the gasket factors is still basically the same developed by Taylor Forge in 1930. In the following equations the symbols and the equations of EN 13445.3 Clause 11 have been used.

- Basic conventional gasket width for flat gaskets $$b_o = w/2$$

  (*w*= effective geometric gasket width)

- Basic conventional gasket width for ring joint gaskets $$b_o = w/8$$

- Gasket width to be used for calculations:

$b = b_o$ when $b_o \leq 6{,}3\ mm$ else $b = 2{,}52\sqrt{b_o}$

Note: this formula is valid for S.I. units only

- Effective gasket sealing diameter for flat gaskets: $$G = Gasket\ OD - 2b$$
  (for ring joint gasket it is the average gasket diameter)

The Taylor Forge method – Calculation of the minimum required bolt area

- **Bolt load for Gasket seating**

$$W_{atm} = \pi bGy$$

- **Bolt load for Service**

$$W_{op} = \pi \frac{G^2}{4} P + 2\pi bGmP$$

- **Bolt load for Test**

$$W_{test} = \pi \frac{G^2}{4} P_{test} + 2\pi bGmP_{test}$$

- **Minimum required bolt area**

$$A_b \geq \max\left(\frac{W_{atm}}{f_{bo}}, \frac{W_{op}}{f_b}, \frac{W_{test}}{f_{b\,test}} \right)$$

- **Maximum recommended bolt area** (in order to avoid gasket crush) **Note**: for flat gasket without compression stop

De and ***Di*** **=** Gasket OD and ID

$$A_b \leq \frac{y\pi\left(D_e^2 - D_i^2\right)}{2 f_{bo}}$$

The Taylor Forge Method – Calculation of flange moments

- **Bolt load for flange Bolting-up:** $W'_{atm} = \left(\dfrac{A_b + A_{b\min}}{2} \right) f_{bo}$

Note: this load is greater than the bolt load W_{atm} used to determine the bolt area in gasket seating

- **Bolt loads for Service/ Test:**

$$H_G + H_T + H_D = W_{op} \qquad H_T + H_D = \pi \frac{G^2}{4} P \qquad H_G = 2\pi bGmP$$

- **Flange moment for Bolting-up:** $M_{atm} = W'_{atm} h_G$

- **Flange moment for Service/Test:** $M_{op} = H_G h_G + H_T h_T + H_D h_D$

Note: use design pressure for service, test pressure for test conditions

The Taylor Forge Method – Calculation of flange stresses

- **Correction factor for bolt spacing δ_B:**

$$C_F = \max\left(\sqrt{\frac{\delta_b}{2\,d_b + \frac{6e}{m+0{,}5}}};1\right)$$

Note: d_B is the bolt diameter, e is the flange thickness, B is the flange ID

- **Reduced calculation moments:**

$$M = \frac{M_{atm/op/test}}{B}$$

- **Stresses for integral flanges (H=Hub, r = radial, θ = tangential):**

$$\sigma_H = \frac{\varphi M}{\lambda g_1^2} \qquad \sigma_r = \frac{(1{,}333e\;\beta_F + l_0)M}{\lambda e^2 l_0} \qquad \sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2+1}{K^2-1}$$

$$\sigma_H \le 1{,}5\min\left(f;f_H\right) \qquad \sigma_r \le f \qquad \sigma_\theta \le f \qquad 0{,}5\;(\sigma_H + \sigma_r) \le f \qquad 0{,}5\;(\sigma_H + \sigma_\theta) \le f$$

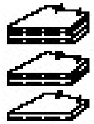
Note: for factors ***φ, β_F, β_Y, λ, K, l_o*** or equivalent (different standards use different symbols) see the relevant formulae or graphs. In PD 5500 and EN 13445.3 Clause 11 the nominal flange design stresses are reduced by a factor ***k*=2/3(1+B/2000)**, however not smaller than 0,75 and not greater than 1. f_H is the nominal design stress of the shell.

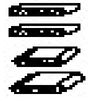
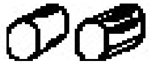
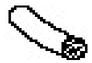
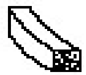
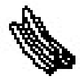
- **Stresses for a loose flange:**

$$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2}$$

Note: in a loose flange $\sigma_r = \sigma_H = 0$

Gasket factors *m* and *y* for different gasket types (source: EN 13445.3 clause 11)

Gasket material		Gasket factor *m*	Minimum design seating stress *y* N/mm²	Sketches	Dimension *w* (minimum) mm
Rubber without fabric or a high percentage of asbestos[1] fibre:					-
- below 75° IRH (International Rubber Hardness Degrees);		0,50	0		-
- 75° IRH or higher.		1,00	1,4		-
Asbestos[1] with a suitable binder for the operating conditions					
(3,2 mm thick		2,0	11,0		-
(1,6 mm thick		2,75	25,5		-
(0,8 mm thick		3,50	44,8		-
Rubber with cotton fabric insertion		1,25	2,8		
Rubber with asbestos[1] fabric insertion, with or without wire reinforcement (3-ply		2,25	15,2		-
(2-ply		2,50	20,0		-
(1-ply		2,75	25,5		-
Vegetable fibre		1,75	7,6		10
Spiral-wound metal asbestos[1] filled	(Carbon	2,50			-
	(Stainless or Monel	3,00	69,0		-
Corrugated metal, asbestos[1] inserted or Corrugated metal, jacketed asbestos[1] filled	Soft aluminium	2,50	20,0		-
	Soft copper or brass	2,75	25,5		-
	Iron or soft steel	3,00	31,0		-
	Monel or 4 to 6 % chrome	3,25	37,9		-
	Stainless steels	3,50	44,8		-
Corrugated metal	Soft aluminium	2,75	25,5		-
	Soft copper or brass	3,00	31,0		-
	Iron or soft steel	3,25	37,9		-
	Monel or 4 to 6 % chrome	3,50	44,8		-
	Stainless steels	3,75	52,4		-

Gasket material		Gasket factor *m*	Minimum design seating stress *y* N/mm²	Sketches	Dimension *w* (minimum) mm
Flat metal jacketed asbestos[1] filled	Soft aluminium	3,25	37,9		-
	Soft copper or brass	3,50	44,8		-
	Iron or soft steel	3,75	52,4		-
	Monel	3,50	55,1		-
	4 to 6 % chrome	3,75	62,0		-
	Stainless steels	3,75	62,0		-
Grooved metal	Soft aluminium	3,25	37,9		-
	Soft copper or brass	3,50	44,8		-
	Iron or soft steel	3,75	52,4		-
	Monel or 4 to 6 % chrome	3,75	62,0		-
	Stainless steels	4,25	69,5		10
Solid flat metal	Soft aluminium	4,00	60,6		-
	Soft copper or brass	4,75	89,5		6
	Iron or soft steel	5,50	124		-
	Monel or 4 to 6 % chrome	6,00	150		-
	Stainless steels	6,50	179		-
Ring joint[2]	Iron or soft steel	5,50	124		-
	Monel or 4 to 6 %chrome	6,00	150		-
	Stainless steels	6,50	179		-
Rubber O-rings:			0,7		-
below 75° IRH		0 to 0,25	1,4		-
between 75° and 85° IRH					-
Rubber square section rings:			1,0		-
below 75° IRH		0 to 0,25	2,8		-
between 75° and 85° IRH					-
Rubber T-section rings:					-
below 75° IRH		0 to 0,25	1,0		-
between 75° and 85° IRH			2,8		

1) New non-asbestos bonded fibre sheet gaskets are not necessarily direct substitutes for asbestos based materials. In particular, pressure, temperature and bolt load limitations may be applied. Use within the manufacturer's current recommendations.

2) $b = w/8$.

Nominal design stresses for bolting in the Taylor Forge method

Nominal Design Stresses for Bolting in the Taylor Forge Method ($\bar{f}$ is the nominal design stress for all other product types)			
	ASME VIII div.1 and 2		**EN 13445.3 Clause 11**
Steels not subject to heat treatment	$\min\left(\frac{R_m}{4}, \frac{R_t}{4}, \bar{f}\right)$	**Carbon Steel**	$\min\left(\frac{R_m}{4}, \frac{R_{p0,2\,t}}{3}, \bar{f}\right)$
Steels subject to heat treatment	$\min\left(\frac{R_m}{5}, \frac{R_{p0,2}}{4}, \frac{R_t}{4}, \bar{f}\right)$	**Low Alloy Steel**	$\min\left(\frac{R_m}{4}, \frac{R_{p0,2\,t}}{3}, \bar{f}\right)$
Austenitic Stainless Steel	$\min\left(\frac{R_m}{4}, \frac{R_t}{4}, \bar{f}\right)$		$\min\left(\frac{R_t}{4}, \bar{f}\right)$

In the Taylor Forge method **extremely low nominal design bolt stresses** compensate the **very low gasket seating stresses**. In reality **the actual bolt loads are much higher than the loads predicted by the method**. ASME VIII division 1 recognizes that the actual bolt stresses obtained in a normal tightening procedure is about

$$f_{bo} = \frac{1564}{\sqrt{d_{bo}}}$$

(for a bolt M20 made of SA 193 B7 it gives 349 MPa against a nominal design stress of 172 MPa). However there is no obligation to apply bolt loads using a controlled torque wrench. **Pay attention in case torque or load values must be prescribed!**

The flange design method of EN 1591.1:2001 and EN 13445.3 Annex G

The harmonised Pressure Vessel standard **EN 13445.3**, in addition to the standard design method of Clause 11 (based on Taylor Forge) gives also in its Annex G an **alternative flange design method, based on EN 1591:2001** (now superseded by the 2013 Edition). The intention of this alternative method is to overcome all the simplified assumptions made in the preceding Taylor Forge and DIN methods using **a more detailed model of a bolted assembly**. The main features of the method are the following.

1. A stress analysis of a more complex model is performed. **The model is composed by a gasket, a series of bolts, and by two (possibly different) flanges, or by flange and a cover**.
2. A **more precise study on the elastic-plastic behaviour of the gasket** is performed. This behaviour is now described by means of **6 different gasket parameters** instead of the 2 parameters considered in the preceding methods.
3. Through these gasket parameters it is possible to **study the actual deformations of the entire assembly, thus establishing a correlation among the bolt load created at bolting-up and the bolt loads in all the subsequent pressure situations** (test and service).
4. **Analysis of the tightening device used**, so that its characteristic plus or minus tolerances **(scatter values)** may be considered in gasket and bolt calculations.
5. **Additional loads** are also considered.
6. **Different temperatures for the various components** (flange 1, flange 2, bolts and gasket) may also be considered.
7. **All** the components are checked using the rules of **limit analysis.**

Gasket parameters in EN 13445.3 Annex G

- Q_{omin} (MPa) is the **minimum seating stress** (equivalent to y in the Taylor Forge method) to be put on the gasket in order to achieve the initial plasticization;

- Q_{max} (MPa) is the **maximum allowable compressive stress on the gasket** (as a function of the operating temperature);

- E_o (MPa) is the **gasket elastic modulus** (as a function of the operating temperature) **at zero compressive load** (see further explanation);

- K_1 (plain number, function of the operating temperature) is the **coefficient of variation of the gasket elastic modulus;** to understand the meaning of this coefficient and of the preceding parameter E_o, we can say that each gasket which has been compressed up to a specified load Q_m, when compression is released behaves as its elastic modulus were given by the formula $E_G = E_o + K_1 Q_m$; in other words, the more is the compressive load on the gasket, the more is its rigidity; this phenomenon doesn't take place on metallic gaskets where it is always $K_1 = 0$;

- m (plain number, function of the operating temperature) is the same as in the Taylor Forge method (however it is not possible to make a direct comparison, because **in the Taylor Forge method *m* is referred to a conventional fixed contact surface**, while **in Annex G the effective gasket contact width is calculated each time iteratively**);

- g_c (plain number, function of the operating temperature) is the "**creep factor**", that is the correction factor to be applied to the gasket elastic modulus E_G in order to take into account the creep that takes place in some gasket materials even at room temperature; use of this factor avoids further complications given by a hypothetic time dependent stress situation.

Example of tabulated gasket parameters to be used in the alternative flange design method of EN 13445.3 Annex G (source: Table G.9.6 of EN 13445.3)

Gasket type and material	T °C	$Q_{0,min}$ MPa	Q_{max} MPa	E_0 MPa	K_1	m_I	g_C
Aluminium (soft) jacket with graphite filler	0...20	50	135	500	25	1,6	1,0
	100		120	800	25	1,6	1,0
	200		90	1 100	25	1,6	1,0
	(300)		60	1 400	25	1,6	1,0
Copper or brass (soft) jacket with graphite filler	0...20	60	150	600	25	1,8	1,0
	100		140	900	25	1,8	1,0
	200		130	1 200	25	1,8	1,0
	300		120	1 500	25	1,8	1,0
	(400)		100	1 800	25	1,8	1,0
Soft iron or soft steel jacket with graphite filler	0...20	80	180	800	25	2,0	1,0
	100		170	1 100	25	2,0	1,0
	200		160	1 400	25	2,0	1,0
	300		150	1 700	25	2,0	1,0
	400		140	2 000	25	2,0	1,0
	(500)		120	2 300	25	2,0	1,0
Low alloy steel (4 % to 6 % chrome) or stainless steel jacket with graphite filler	0...20	100	250	800	25	2,2	1,0
	100		240	1 100	25	2,2	1,0
	200		220	1 400	25	2,2	1,0
	300		200	1 700	25	2,2	1,0
	400		180	2 000	25	2,2	1,0
	500		140	2 300	25	2,2	1,0

Advantages and disadvantages of the alternative flange design method of EN 13445.3 Annex G

- More **reliable calculation of the bolt loads**, which are generally higher than in any one of the other methods

- The **bolt loads are strictly dependent on the tightening device used**: if an alternative device having different scatter values has to be used, a new checking of the assembly is needed

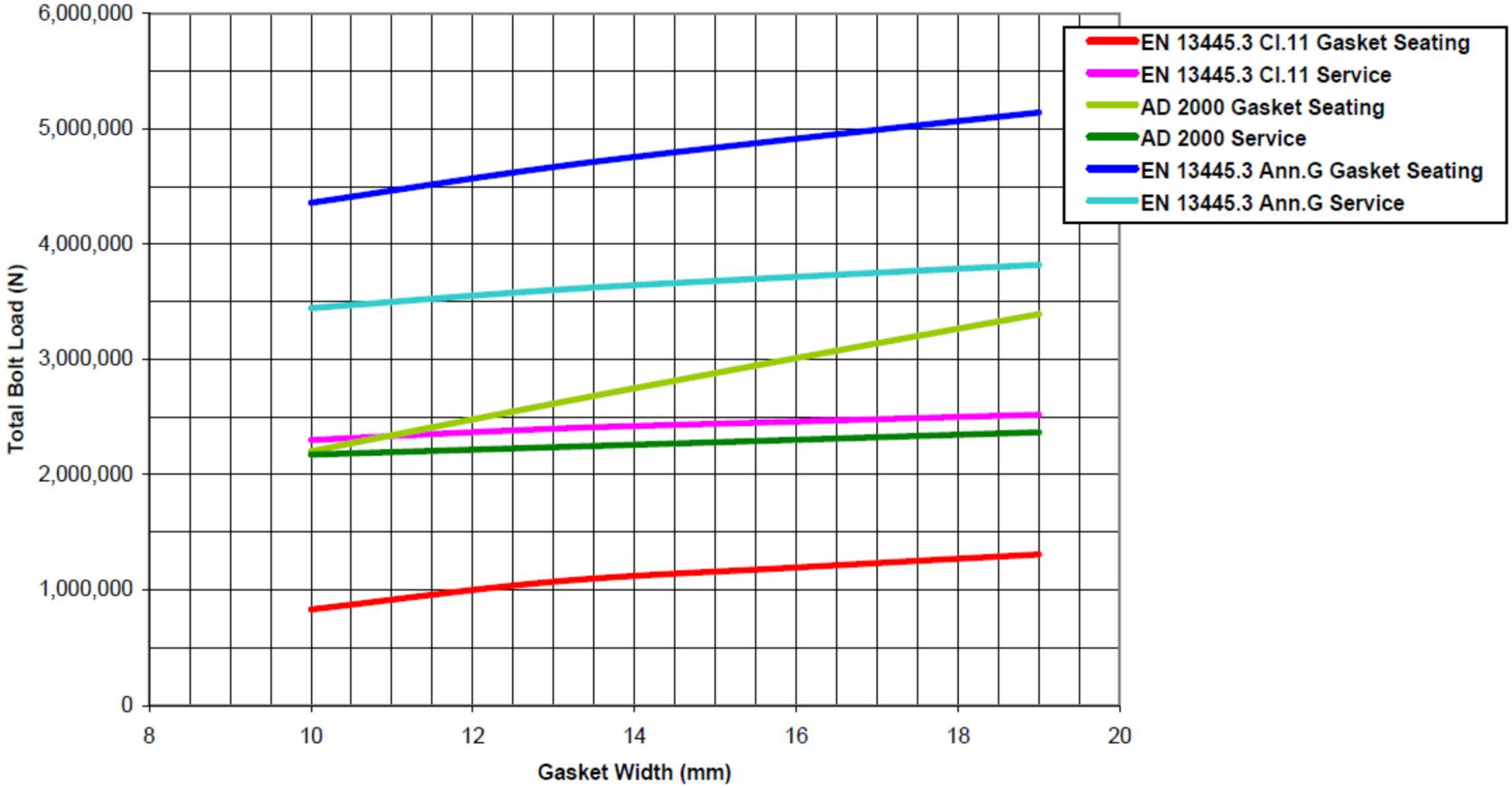
- Tightening load of any flange connection has always to be specified and **the user is not allowed to use higher loads**

- The **nominal design bolt stress is the same of the other product types:** therefore the bolts have no additional safety margin against excessive tightening (note also that **the safety factor for bolting-up is 1,1 against the yield point** as in the DIN method)

- It is **not possible to design the bolts independently from the design of the other parts of the connection** (gasket and flanges): any modification made on gasket and flanges will have an influence on the bolt loads

- The 6 **gasket parameters** recommended in the tables **should be confirmed by the gasket manufacturers**

Cross comparison of bolt load values for gasket seating and service conditions as a function of gasket width in different calculation standards (Gasket: Mineral fibre soft iron jacketed ID=1000 mm - PS=25 bar)



The new approach in EN 1591.1:2013 considering tightness classes (= guaranteed leakage rates) for the case of gases and vapours

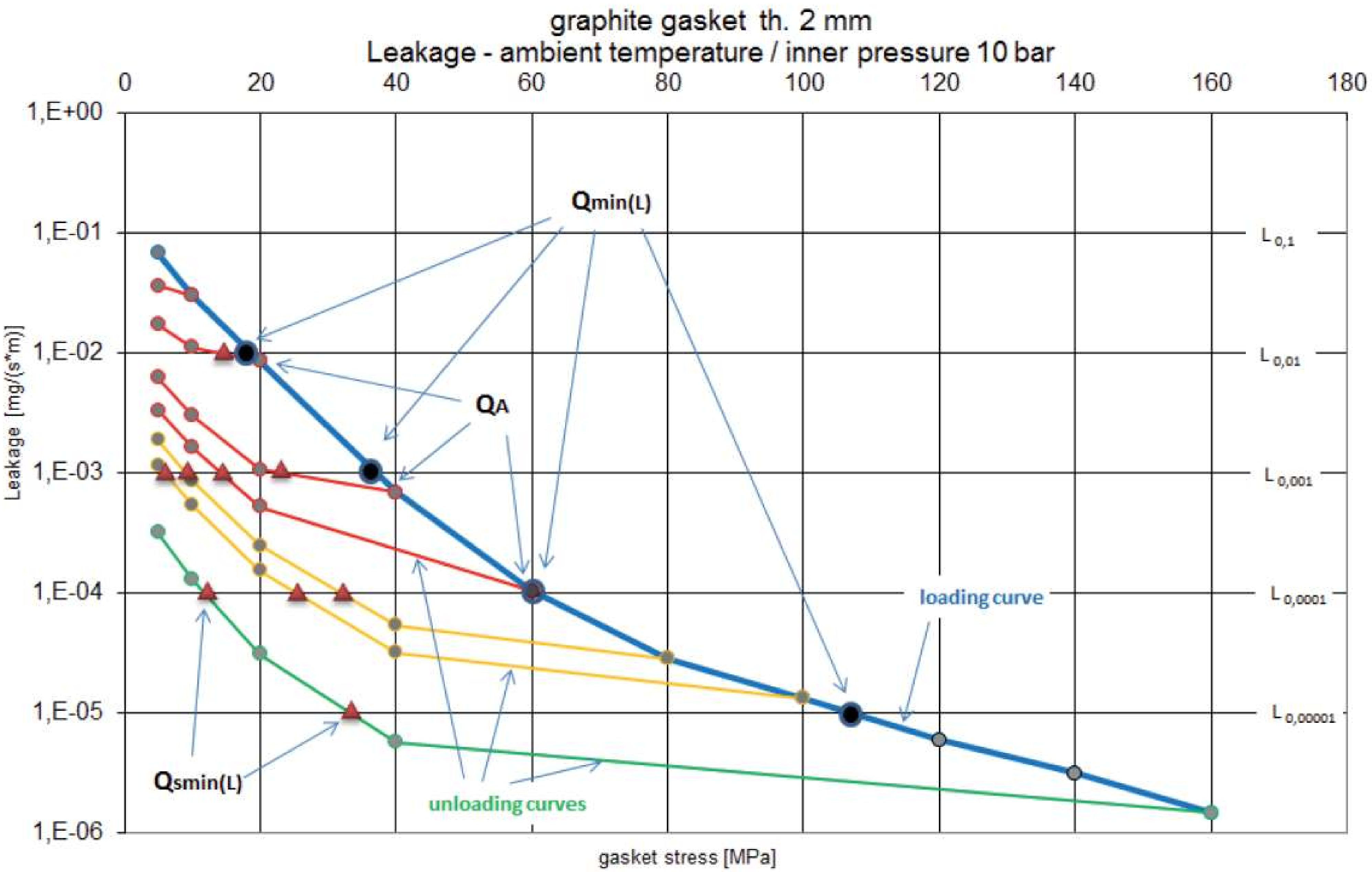
Tightness classes	$L_{1,0}$	$L_{0,1}$	$L_{0,01}$
Specific leak rates [mg s^{-1} m^{-1}]	$\leq 1,0$	$\leq 0,1$	$\leq 0,01$

- $Q_{min(L)}$ (MPa) is the **minimum required seating stress** (equivalent to ***y*** in the Taylor Forge method) to be put on the gasket in order to achieve leak tightness **for a specific Tightness Class L**;
- $Q_{Smin(L)}$ (MPa) is the **minimum required gasket surface pressure for tightness class L in service** (this characteristic depends on the **gasket surface pressure Q_A applied during assembly** – this is the equivalent of the **m x p** value of the Taylor Forge method);
- $Q_{smax}(T_o)$ (MPa) is the **maximum allowable compressive stress on the gasket** (at room temperature);
- $Q_{smax}(T)$ (MPa) is the **maximum allowable compressive stress on the gasket** (at service temperature);
- Δe_{GC} (mm) is the creep relaxation of the gasket
- E_G (Mpa) is the modulus of elasticity of the gasket for unloading/reloading
- μ_G is the friction factor between the gasket and the flange facing

Sealing gasket parameter when no leakage rate is specified (EN 1591.1:2013)

Gasket type and material	Q_{0min} MPa	m
Non-metallic flat gaskets (soft) and flat gaskets with metal insertion		
Rubber	0,5	0,9
PTFE	10	1,3
Expanded PTFE (ePTFE)	12	1,3
Expanded graphite without metal insertion	10	1,3
Expanded graphite with perforated metal insertion	15	1,3
Expanded graphite with adhesive flat metal insertion	10	1,3
Expanded graphite with metallic sheet laminated in thin layers with standing high stresses	15	1,3
Non asbestos fibre with binder (thickness < 1mm)	40	1,6
Non asbestos fibre with binder (thickness >= 1mm)	35	1,6
Grooved steel gaskets with soft layers on both sides		
PTFE layers on soft steel	10	1,3
PTFE layers on stainless steel	10	1,3
Graphite layers on soft steel	15	1,3
Graphite layers on low alloy heat resistant steel	15	1,3
Graphite layers on stainless steel	15	1,3
Silver layers on heat resistant stainless steel	125	1,8
Spiral wound gaskets with soft filler		
Spiral wound gaskets with PTFE filler, outer support-ring only	20	1,6
Spiral wound gaskets with PTFE filler, inner and outer support-rings	20	1,6
Spiral wound gaskets with graphite filler, outer support-ring only	20	1,6
Spiral wound gaskets with graphite filler, inner and outer support-rings	50	1,6
Solid metal gaskets		
Aluminium (Al) (soft)	50	2,0
Copper (cu) or brass (soft))	100	2,0
Iron (Fe) (soft)	175	2,0
Steel (soft)	200	2,0
Steel, low alloy, heat resistant	225	2,0
Stainless steel	250	2,0
Stainless steel, heat resistant	300	2,0
Covered metal-jacketed gaskets		
Soft iron or steel jacket with graphite filler and covering	20	1,3
Low alloy steel (4 % to 6 % chrome) or stainless steel jacket with graphite filler and covering	20	1,3
Stainless steel jacket with expanded PTFE filler and covering	10	1,3
Nickel alloy jacket with expanded PTFE filler and covering	10	1,3
Metal-jacketed gaskets		
Aluminium (soft) jacket with graphite filler	50	1,6
Copper or brass (soft) jacket with graphite filler	60	1,8
Soft iron or steel jacket with graphite filler	80	2,0
Low alloy steel (4 % to 6 % chrome) or stainless steel jacket with graphite filler	100	2,2

Leakage rate during unloading of a gasket tested at 10 bar Helium pressure



Scatter of initial bolt load on a single bolt (Table B.1 of EN 1591.1:2013)

Bolting up (tightening) method; Measuring method	Factors affecting scatter	Scatter value [a, b, c, d]	
		ε_{1-}	ε_{1+}
Wrench: operator feel or uncontrolled	Friction, Stiffness, Qualification of operator	0,3 + 0,5 x μ	0,3 + 0,5 x μ
Impact wrench	Friction, Stiffness, Calibration	0,2 + 0,5 x μ	0,2 + 0,5 x μ
Torque wrench = Wrench with measuring of torque (only)	Friction, Calibration, Lubrification	0,1 + 0,5 x μ	0,1 + 0,5 x μ
Hydraulic tensioner; Measuring of hydraulic pressure	Stiffness, Bolt length, Calibration	0,2	0,4
Wrench or hydraulic tensioner; Measuring of bolt elongation	Stiffness, Bolt length, Calibration	0,15	0,15
Wrench, Measuring of turn of nut (nearly to bolt yield)	Stiffness, Friction, Calibration	0,10	0,10
Wrench, Measuring of torque and turn of nut (nearly to bolt yield)	Calibration	0,07	0,07

[a] Very experienced operators can achieve scatter less than given values (e.g. ε = 0,2 instead of ε = 0,3 with torque wrench); for inexperienced operators scatter can be greater than shown.

[b] Tabulated scatter values are for a single bolt, the scatter of the total bolt load will be less, for statistical reasons, see B.2.

[c] With hydraulic tensioner, ε_{1+} et ε_{1-} are not equal, due to the fact that an additional load is supplied to the bolt while turning the unit to contact, prior to load transfer to the nut.

[d] μ is the friction coefficient which can be assumed between bolt and nut.

Positive and negative scatter values for n_B bolts starting from the scatter values of a single bolt (EN 1591.1:2013 Annex B.2)

$$\varepsilon_{n+} = \varepsilon_{1+} \frac{\left(1 + \frac{3}{\sqrt{n_B}}\right)}{4} \qquad \varepsilon_{n-} = \varepsilon_{1-} \frac{\left(1 + \frac{3}{\sqrt{n_B}}\right)}{4}$$

Nominal load to be specified for bolt tightening

$$W_A = \frac{W}{(1 - \varepsilon_{n-})}$$

Total load to be used for bolt checking

$$W_B = W_A \, (1 + \varepsilon_{n+})$$

(W = theoretical bolt load for proper gasket compression)

Future developments of the new design method of EN 1591.1:2013

- The method (in practice an evolution of Annex G EN 13445.3, prepared by CEN TC54) was developed by a different CEN TC **(TC74 - Flanges)**. It gives the possibility of designing a bolted assembly **containing a gas or a vapour on the basis of a specified leak rate** (leak rates are expressed in **mg/s/m,** with reference to a test procedure provided by **EN 13555** with Helium at 40 bar and with a standard test gasket having specified dimensions).

- **EN 1591.1:2013** contains a table of **gasket characteristics to be used with liquids having no leak rate**, however in case of **gases or vapours** for which a **specific tightness class** is specified, gasket parameters shall be **either supplied by the gasket manufacturer or resulting from tests** in accordance with **EN 13555**.

- **CEN TR 1591.2:2020** contains only gasket parameters of the gaskets used in the **PERL project**. However similar gasket types of different manufacturers may have different characteristics. Therefore the method can only be applied using the data contained in the **European gasket data base** (e.g. **http://www.gasketdata.org**)

- **Amendment 6 of EN13445.3:2014**, published in April 2019, **permits calculation according to EN 1591.1** as an alternative either to Clause 11 or to Annex G **when leakage rates are required** and when sufficient gasket data are available. This amendment is now part of **EN 13445.3:2021.**

- German experts of CEN TC54 are now asking for a more general amendment of Annex G, in order to bring it in compliance with EN 1591.1:2013: this because the German law **(TA-Luft),** which is the practical application of many European directives on air pollution, and also the **German VDI standard 2290** are specifying **particular leak tightness levels for bolted assemblies of vessels and piping containing specific pollutants,** which is only possible using EN 1591.1:2013. This new amendment should also give a method for the calculation of **flanges bolted with heat exchanger tubesheets,** which for the time being is only possible using Clause 11.

- Due to the above mentioned situation, **further evolutions are expected shortly** in the European flange standards.

THANK YOU VERY MUCH FOR YOUR ATTENTION!