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Prirobnice in prirobnični spoji - Pravila za izračun okroglih prirobnic s tesnili - Temeljne informacije

Flanges and their joints - Design rules for gasketed circular flange connections - Background information

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Flanges and their joints - Design rules for gasketed circular flange connections - Background information

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EUROPEAN COMMITTEE FOR STANDARDIZATION
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Contents

Foreword

This CEN report has been prepared by Technical Committee CEN/TC 74 "Flanges and their joints", the secretariat of which is held by DIN.

This text should be read alongside prEN 1591-1, the same notation is used here. References to numbered equations in prEN 1591-1 are indicated by integer numbers, those in the present document have decimal format.

1 Introduction

Strength assessments in design calculations generally involve a 'proof of load' in the form:

actual loads ≤ allowable loads (1.1a)

and similarly when determining wall thicknesses, etc. The classical basis is linear elasticity theory, for which proofs are often written:

actual stresses ≤ allowable stresses (1.1b)

However, elasticity theory can lead to illogical results such as stress increasing (strength decreasing) with increasing wall thickness. On the other hand plasticity theory avoids such inconsistencies and Limit Load Analysis gives reliable results. Therefore in prEN 1591-1 Limit Load Analysis is used as the basis for proof of load.

To ensure adequate leak-tightness, gasket compressive stress Q must not fall below a certain value. For example in the ASME Boiler and Pressure Vessel Code, SectionVIII ('ASME' hereafter) the proof for leak-tightness is:

$$Q \geq m \cdot |P| \tag{1.2}$$

where m is a 'gasket factor' and P is fluid pressure. Note that the general form of a leak-tightness proof is equivalent to a load proof of the form:

actual load ≥ required load (1.3)

It follows that both upper and lower limits have to be imposed on gasket and bolt loads, as in:

required load ≤ actual load ≤ allowable load (1.4)

Various commonly used design codes (e.g. ASME and related codes) for gasketed joints do not apply this condition but assume:

required load = actual load (1.5)

and neglect interactions between assembly and subsequent test or service conditions, or additional assumptions are introduced (e.g. 'gasket force is constant'). This is a poor model of real joints and leakage problems result.

A code which properly treats **all** load conditions (assembly, test, and all service conditions) is TGL 32903/13 (1983), a national standard of the former German Democratic Republic. Variants of this have been in use since 1973 and it has been applied to the design of thousands of gasketed joints without leakage problems. Therefore when CEN/TC/74/WG 10 was requested to produce a design procedure for gasketed joints, the TGL method was chosen as the basis. At the request of CEN/TC 54/WG C the scope of the TGL version was extended, with basic principles unchanged. The established validity was unaffected by this extension but behaviour in the new domain has yet to be verified. Examples of validation tests for the original domain are given in Annex B.

1 Introduction (Francais)

En conception, les calculs de résistance reviennent dans leur principe à faire la preuve que les sollicitations exercées sont admissibles, c'est à dire à vérifier que l'on a:

sollicitations appliquées ≤ sollicitations admissibles (1.1a)

ceci restant vrai même lorsque les calculs se réduisent à l'application de formules de détermination d'épaisseurs. Classiquement, la vérification est fondée sur la théorie de l'élasticité linéaire, les critères associés étant souvent écrits:

contraintes appliquées ≤ contraintes admissibles (1.1b)

Cependant, la théorie de l'élasticité peut conduire à des résultats aberrants, par exemple des contraintes qui croissent (donc une résistance qui diminue) quand on augmente l'épaisseur de paroi.

La théorie de la plasticité, quand à elle, évite de telles incohérences, et la théorie de l'Analyse Limite, qui en découle, permet d'obtenir des résultats pertinents. C'est la raison pour laquelle l'Analyse Limite est la base utilisée pour établir l'admissibilité des charges dans le projet prEN 1591-1.

Par ailleurs, pour assurer une étanchéité appropriée, la contrainte de compression Q du joint d'étanchéité ne doit pas descendre sous une certaine valeur. Par exemple, dans le Code ASME des Générateurs de Vapeur et Appareils à Pression, Section VIII, le critère d'étanchéité s'écrit:

$$Q \geq m \cdot |P| \quad (1.2)$$

où m est le coefficient de joint et P la pression du fluide. On notera qu'un critère d'étanchéité a une forme générale équivalente à celle d'un critère d'admissibilité de charge, c'est à dire:

charge appliquée ≥ charge requise (1.3)

Il en ressort que des limites supérieures aussi bien qu'inférieures doivent être imposées aux efforts sur le joint et dans la boulonnerie, ce qui s'écrit:

effort requis ≤ effort appliqué ≤ effort admissible (1.4)

Divers codes de construction d'usage courant (par exemple l'ASME et les codes qui en dérivent) n'appliquent pas cette condition, mais supposent implicitement que l'on a:

effort appliqué = effort requis (1.5)

et ils négligent les interactions entre la situation d'assise et les situations de service ou d'épreuve, ou introduisent des hypothèses supplémentaires (par exemple: "l'effort sur le joint est constant"). Cela aboutit à une modélisation grossière de la réalité des assemblages à brides, d'où des problèmes d'étanchéité.

La TGL 32903/13 (1983), norme nationale de l'ex-République Démocratique Allemande, est un code qui traite correctement **toutes** les situations (assise, épreuve, ensemble des situations de service). Des variantes de cette norme sont utilisées depuis 1973 et ont été appliquées à la conception de milliers d'assemblages à brides sans problèmes d'étanchéité. Aussi, lorsque le CEN/TC 74/WG 10 a été chargé d'élaborer une méthode de calcul des assemblages à brides, la norme TGL a-t-elle été choisie comme base.
A la demande du CEN/TC 74/WG C, le domaine d'application de la version TGL a été étendu, les principes de base demeurant inchangés. La validité déjà établie n'est pas affectée par cette extension, mais ce que l'on obtient dans le nouveau domaine reste cependant à vérifier. L'Annexe A donne des exemples d'essais de validation relatifs au domaine d'application initial.

1 Einleitung (Deutsch)

Im allgemeinen beinhaltet die Festigkeitsanalyse in der Berechnung den Nachweis, daß die Belastungen zulässig sind in der Form:

$$\text{vorhandene Belastungen} \leq \text{zulässige Belastungen} \tag{1.1a}$$

Dies gilt auch für die Berechnung der Wanddicken etc. Die klassische Basis ist die lineare Elastizitätstheorie, deren Nachweis gegeben ist durch das Kriterium

$$\text{vorhandene Spannungen} \leq \text{zulässige Spannungen} \tag{1.1b}$$

Die Elastizitätstheorie kann jedoch zu unlogischen Resultaten wie zunehmender Spannung (abfallende Festigkeit) mit zunehmender Wandstärke führen. Andererseits vermeidet die Theorie der plastischen Verformungen solche Inkonsistenzen und die Traglast-Berechnung führt zu annehmbaren Resultaten. Daher wird in der prEN 1591-1 die Traglast-Berechnung als Grundlage für den Spannungsnachweis verwendet.

Um die erforderliche Dichtheit sicherzustellen, darf die Druckspannung der Dichtung Q nicht unter einen bestimmten Wert fallen. Der Nachweis der Dichtheit wird beispielsweise im ASME Druckbehältercode, Section VIII ("ASME" im folgenden genannt) wie folgt geführt:

$$Q \geq m \cdot |P| \tag{1.2}$$

wobei m ein Dichtungsfaktor und P der Mediendruck ist. Die allgemeine Form des Dichtheitsnachweises ist äquivalent zu einem Nachweis, daß die Belastungen ausreichend sind:

$$\text{vorhandene Belastung} \geq \text{erforderliche Belastung} \tag{1.3}$$

Hieraus folgt, daß sowohl obere als auch untere Schranken für die Belastung der Dichtung und der Schrauben wie folgt gelten:

$$\text{erforderliche Belastung} \leq \text{vorhandene Belastung} \leq \text{zulässige Belastung} \tag{1.4}$$

Unterschiedliche allgemein verwendete Regelwerke für die Berechnung (z.B. ASME und ähnliche) für Verbindungen mit Dichtungen verwenden diese Relation nicht, sondern nehmen an:

$$\text{erforderliche Belastung} = \text{vorhandene Belastung} \tag{1.5}$$

und vernachlässigen Wechselwirkungen zwischen Montagezustand und nachfolgenden Prüf- oder Betriebszuständen, oder es werden zusätzliche Annahmen gemacht (z.B. konstante Dichtungskraft). Dies ergibt kein gutes Modell für tatsächliche Verbindungen und Leckageprobleme sind die Folge.

Ein Regelwerk, daß alle Belastungszustände richtigt behandelt (Montage, Prüfung und alle Betriebsbedingungen), ist TGL 32903/13 (1983), eine nationale Norm der früheren Deutschen Demokratischen Republik. Diese wird in unterschiedlichen Versionen seit 1973 verwendet und wurde für die Auslegung tausender Flanschverbindungen ohne Leckageprobleme angewendet. Als daher CEN/TC 74/WG 10 aufgefordert wurde, eine Berechnungsregel für Flanschverbindungen mit Dichtungen zu erarbeiten, wurde die TGL-Methode als Basis gewählt. Auf Anforderung von CEN/TC 54/WG C wurde der Anwendungsbereich der TGL-Version erweitert, wobei die Methode im Grundsatz unverändert blieb. Die erprobte Gültigkeit wurde durch diese Erweiterung nicht berührt, jedoch steht das Verhalten im neuen Anwendungsgebiet noch aus. Beispiele für Auswertungstests im ursprünglichen Anwendungsbereich sind in Anhang B angegeben.

2 Forces in gasketed joints

2.1 Definition of active and passive forces

Active forces (and moments) are those which can, in principle, cause unlimited deformation. Examples in joints are the axial fluid-pressure force F_Q and an external dead-weight force F_A. If limited plastic deformation is possible **without** loss of function (as in dished heads) only active forces are required in the proof of load.

Passive forces (and moments) are due to limited elastic deformation and can only cause limited deformation (they do affect fatigue). An example, is a force due to differential thermal expansion. If limited plastic deformations **can** cause loss of function, passive forces must also be included in the proof of load.

NOTE: This is partially true in the case of leak-tightness. Differential thermal expansion ΔU_I can cause plastic deformation, with loss of bolt and gasket force, and hence of leak-tightness. Therefore ΔU_I is included in the calculation of F_{GI}, F_{BI} in eqs. (51) ff and in the proof of load (Section 5). On the other hand, scatter of assembly bolt-load is not taken into account in eqs. (67) ff. This is because if assembly bolt-load exceeds the minimum (to ensure subsequent leak-tightness) and limited plastic deformation occurs in a subsequent condition, the forces can fall to the minimum required without loss of leak-tightness.

2.2 Coupling of internal forces

Under both assembly conditions and subsequent load conditions the component parts of a gasketed joint are coupled by internal forces. Therefore the following geometric relation must hold between displacements of parts:

$$\{\Theta_F \cdot h_G + \tilde{\Theta}_F \cdot \tilde{h}_G + \Theta_L \cdot h_L + \tilde{\Theta}_L \cdot \tilde{h}_L + U_B + U_G\}_{(I=0)} = \\ = \{\Theta_F \cdot h_G + \tilde{\Theta}_F \cdot \tilde{h}_G + \Theta_L \cdot h_L + \tilde{\Theta}_L \cdot \tilde{h}_L + U_B + U_G + \Delta U_I\}_{(I \neq 0)} \tag{2.1}$$

Substituting θ_F, θ_L (see eqs. (E.1), (E.2) and U_B, U_G (eq. (5.1, 4.27, 4.29)) and the equilibrium condition

$$F_B = F_G + F_Q + F_R, \text{ for all } I \tag{2.2}$$

gives:

$$F_{G0} \cdot Y_{G0} + F_{Q0} \cdot Y_{Q0} + F_{R0} \cdot Y_{R0} = F_{GI} \cdot Y_{GI} + F_{QI} \cdot Y_{QI} + F_{RI} \cdot Y_{RI} + \Delta U_I \tag{2.3}$$

This is the fundamental equation relating force changes in a joint, subject to $F_{Q0} = 0$ ($P_{I=0} = 0$). The flexibility parameters Y_G, Y_Q, Y_R are given in eqs. (46) to (48).

If the required gasket force (eq. 50) is known for subsequent conditions (e.g.from pressure P_I and gasket factor m_I), then from eq. (2.3) the assembly force to ensure leak-tightness is:

$$F_{G0} \geq \{F_{GI} \cdot Y_{GI} + F_{QI} \cdot Y_{QI} + F_{RI} \cdot Y_{RI} - F_{R0} \cdot Y_{R0} + \Delta U_I\} / Y_{G0} \tag{2.4}$$

which is the basis of eq. (51).

If there is more than one subsequent condition, the assembly force must be adequate for all; therefore use the largest. But in the other subsequent conditions the gasket force is greater than required (eq. (52)) and gaskets, flanges and bolts must be able to withstand this additional force.

The required gasket force, F_{G0} in eq. (2.4) (or $F_{G\Delta}$) in eq. (51)), may be very small, e.g. if $\Delta U_I < 0$ or $F_{QI} < 0$.

The assembly force must ensure (eq. (50)) a gasket stress of at least Q_{min} (this parameter is explained in prEN V 1591-2). Subsequently the condition $F_{G0} > F_{G\Delta}$, is unnecessary if the gasket stress is sufficient to prevent the gasket slipping radially. In fact the calculation ensures this for all conditions, therefore subsequent forces are calculated using the minimum required force $F_{G\Delta}$ rather than the possibly enlarged force F_{G0} (eq. (52)).

2.3 Assembly conditions

The bolt-tightening method must produce a bolt load not less than the required minimum thus, due to scatter, the target bolt load must be greater than this minimum. These effects are taken into account in prEN 1591-1, subclause 4.4.2. The scatter parameter ε can be defined in various ways. In prEN 1591-1 the following 'linear definition' is used (as in VDI 2230):

$$\check{F}_{B0} = \overline{F}_{B0} \cdot (1 - \varepsilon); \qquad \hat{F}_{B0} = \overline{F}_{B0} \cdot (1+\varepsilon) \tag{2.5}$$

$$\overline{F}_{B0} = 1/2 \cdot (\check{F}_{B0} + \hat{F}_{B0}); \qquad 0<\varepsilon<1 \tag{2.6}$$

An alternative possibility is the 'geometric definition':

$$\check{F}_{B0} = \overline{F}_{B0}/(1 + \varepsilon); \qquad \hat{F}_{B0} = \overline{F}_{B0} \cdot (1 + \varepsilon) \tag{2.7}$$

$$\overline{F}_{B0} = \sqrt{\check{F}_{B0} \cdot \hat{F}_{B0}}; \qquad 0 < \varepsilon < \infty \tag{2.8}$$

These definitions are nearly the same for 0 < ε < 0,1 ... 0,2, but the alternative definitions account for disagreements between published data. In prEN 1591-1, examples of ε are given based on the linear definition.

3 Gasket characteristics

3.1 Mechanical behavior

When structual deformation were not taken into account, the details of gasket behaviour were of less concern, but with the present more comprehensive approach they need further consideration. The load-deformation behaviour of most types of gasket is very nonlinear, as indicated schematically in figure 1.

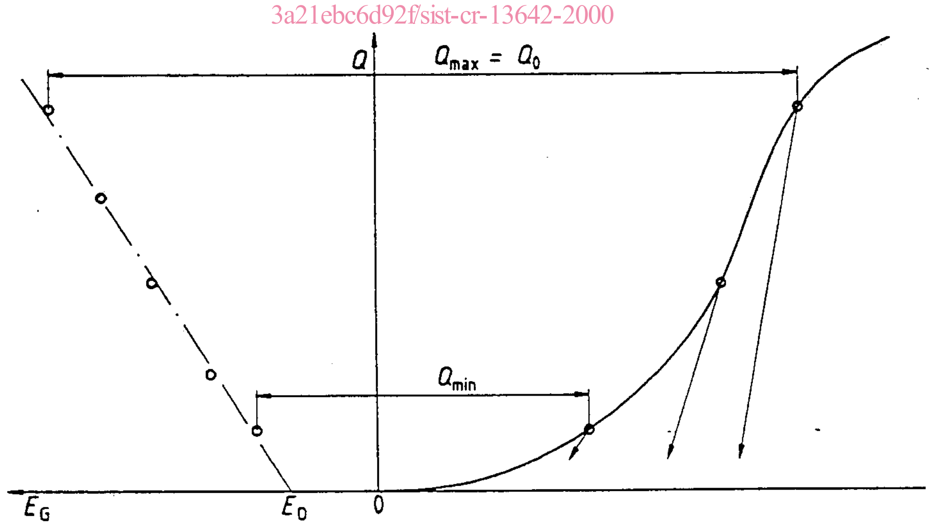


Figure 1

3.1.1 Unloading modulus

The initial-loading line (right of diagram) is strongly non-linear, and unloading-lines are approximately linear with slope increasing with assembly stress. The corresponding values of unloading modulus E_G, are plotted on the left and vary almost linearly with assembly stress:

$$E_G \approx E_0 + K_1 \cdot Q_0 \tag{3.1}$$

In reality the unloading line is not strictly linear, its slope reduces at lower stresses. Therefore, to measure E_G, a more complete definition is necessary and is dealt with separately in measurement procedures being drawn-up by CEN/TC 74/WG 8.

Some values of E_0, and K_1 for bonded compressed asbestos fibre gaskets ('It' or 'caf') have been measured by VEB KCA Dresden and other values can be obtained from published literature, nevertheless available data is inadequate. [For some composite gaskets it may be possible to calculate E_G from values for the component parts.]

3.1.2 Creep

When a gasket is subjected to compressive stress an immediate elastic (or elastic-plastic) deformation U_G occurs followed by creep, increasing deformation with time at constant load. The figure 2 shows this diagrammatically.

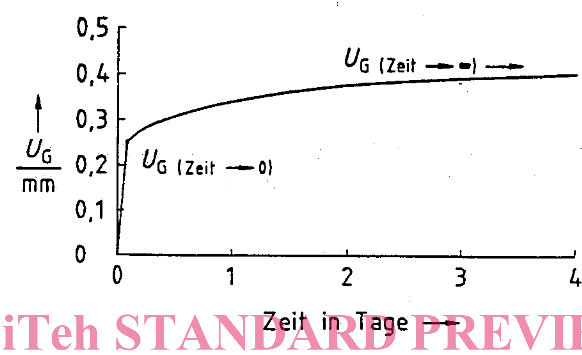


Figure 2

A simple means of modelling creep is to treat it as effectively reducing the compression modulus:

$$E_{G(\text{including creep})} = g_C \cdot E_{G(\text{excluding creep})} \tag{4.29}$$

$$g_C = U_{G(\text{time} \to 0)} / U_{G(\text{time} \to \infty)} \qquad \text{(for } Q = \text{const.)} \tag{4.30}$$

The creep factor g_C gives simple information about the time-dependent behaviour of gaskets and in the absence of detailed information it is assumed that both stress relaxation and creep are covered by g_C. In reality there is unlikely to be an asymptotic value of g_C, therefore the effective value depends on service time. Again, more detailed definition and measurement of g_C is dealt with in measurement procedures being drawn-up by CEN/TC 74/WG 8.

Some measurements of g_C have been made by VEB KCA Dresden. Values in prEN V 1591-2 are very approximate.

3.1.3 Maximum compressive stress

The measured value of maximum allowable nominal gasket compressive stress Q_{max} is modified by a factor c_G in eq. (72), this makes allowance for the gasket width in relation to thickness. Hence the designation "nominal" and this is also the reason for reducing some Q_{max} values relative to DIN 2505-2. For more details see 4.4.

Some values of Q_{max} for compressed asbestos fiber with binder ('It' or 'caf') have been measured by VEB KCA Dresden. Again, measurement of Q_{max} is dealt with separately in procedures being drawn-up by CEN/TC 74/WG 8.

3.2 Sealing criteria

The minimum required gasket working stress Q_{Imin} and minimum gasket assembly stress Q_{min} are basically familiar parameters relating to leakage control. Nevertheless values differ considerably for different design codes (e.g. ASME and DIN 2505-2). Some differences are due to different assumptions for effective gasket-width and others due to insufficient experimental verification. Furthermore, it is now known that the required working stress depends on the actual assembly stress, although this is not taken into account in traditional approaches. The use of such unsatisfactory design parameters must continue until acceptable validated alternatives are available, hence 'traditional' m values are listed in prEN V 1591-2 as an interim measure.

An alternative design approach to leakage control is outlined in prEN V 1591-2. This method is likely to be preferred in future because it involves a quantifiable leak-tightness, the main calculation procedure in prEN 1591-1 remains unchanged. Since experience with this method is limited at present, it is presented as an optional alternative. As for the traditional approach, availability of validated data is limited.

NOTE: Further measurements of all gasket parameters are necessary and planned by CEN/TC 74.

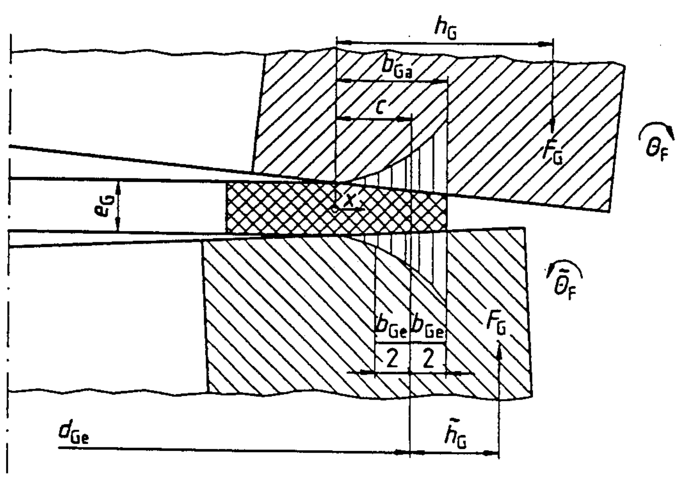
3.3 Effective width

The effective width of a gasket varies with flange rotation, which also causes a radial variation of compressive stress. Strictly, an iterative calculation is required to reconcile the changing width, gasket stresses and bolt load. However the approach adopted in prEN 1591-1 is to calculate gasket-width for the assembly condition and then assumed this to be unchanged for subsequent conditions. This simplifying assumption is strictly correct only if gasket-force F_G and flange rotations do not change. However, the assumption is conservative if the effective width for subsequent conditions is actually smaller than in the assembly condition, which is often the case.

4 Calculations for gaskets

4.1 Effective width of gaskets

4.1.1 Flat gaskets



Notation:
b_{Ga} = contact width
b_{Ge} = effective width
b_{Gc} = calculated width
Q(x) = compressive stress

Figure 3

Elastic rotation of flanges (see eq.(7.36)):

$$\Theta_F + \tilde{\Theta}_F = F_G \cdot (Z_F \cdot h_G / E_F + \tilde{Z}_F \cdot \tilde{h}_G / \tilde{E}_F) \qquad (4.1)$$

Elastic deformation of gasket (for $0 \le x \le b_{Ga}$) :

$$\varepsilon = (\Theta_F + \tilde{\Theta}_F) \cdot \frac{x}{e_\varepsilon} = k \cdot x \qquad (4.2)$$

$$E_G = E_0 + K_1 \cdot Q = \frac{dQ}{d\varepsilon} \qquad (4.3)$$

$$Q = \frac{E_0}{K_1} \cdot \{\exp(K_1 \cdot \varepsilon) - 1\} \approx E_0 \cdot \varepsilon \cdot \left[1 + \tfrac{1}{2} \cdot K_1 \cdot \varepsilon\right] \qquad (4.4)$$

The resultant gasket force F_G is:

$$F_G = \pi \cdot d_{Ge} \cdot \int_0^{b_{Ga}} Q(x) \cdot dx \qquad (4.5a)$$

acting at $x = c$, given by:

$$c \cdot \int_0^{b_{Ga}} Q(x) \cdot dx = \int_0^{b_{Ga}} Q(x) \cdot x \cdot dx \qquad (4.5b)$$

From this follows, step by step:

$$F_G = \pi \cdot d_{Ge} \cdot E_0 \cdot k \cdot \tfrac{1}{2} \cdot b_{Ga}^2 \cdot \left[1 + \tfrac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga}\right] \qquad (4.6)$$

$$b_{Ga} = \sqrt{\frac{F_G \cdot e_G \cdot 2}{\pi \cdot d_{Ge} \cdot E_0 \cdot (\Theta_F + \tilde{\Theta}_F) \cdot \left[1 + \tfrac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga}\right]}} \qquad (4.7)$$

$$c = \frac{2}{3} \cdot b_{Ga} \cdot \frac{1 + \frac{3}{8} \cdot K_1 \cdot k \cdot b_{Ga}}{1 + \frac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga}} \qquad (4.8)$$

$$b_{Gc} = 2 \cdot (b_{Ga} - c) = \frac{2}{3} \cdot b_{Ga} \cdot \frac{1 + \frac{1}{4} \cdot K_1 \cdot k \cdot b_{Ga}}{1 + \frac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga}} \qquad (4.9)$$

$$b_{Gc} = \sqrt{\frac{F_G \cdot e_G \cdot \frac{8}{9}}{\pi \cdot d_{Ge} \cdot (\Theta_F + \tilde{\Theta}_F) \cdot E_0} \cdot \frac{(1 + \frac{1}{4} \cdot K_1 \cdot k \cdot b_{Ga})^2}{(1 + \frac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga})^3}} \qquad (4.10)$$

The remaining elimination of $k \cdot b_{Ga}$ is simplified by assuming $K_1 \cdot k \cdot b_{Ga} \cdot \frac{1}{3} << 1$, then eq. (4.10) becomes:

$$b_{Gc} = \sqrt{\frac{F_G \cdot e_G \cdot \frac{8}{9}}{\pi \cdot d_{Ge} \cdot (\Theta_F + \tilde{\Theta}_F) \cdot E_0 \cdot (1 + \frac{1}{2} \cdot K_1 \cdot k \cdot b_{Ga})}} \qquad (4.11)$$

[This approximation gives an error on $b_{Gc} \leq 5\ \%$ for $K_1 \cdot k \cdot b_{Ga} \leq 10$, almost always true!]

F_Q will be at least:

$$F_Q = \pi \cdot d_{Ge} \cdot b_{Ge} \cdot Q, \text{ where } Q \text{ is an average value,} \tag{4.12}$$

and with eqs. (4.6) and (4.9) this gives:

$$E_0 \cdot k \cdot b_{Ga} = Q \cdot \frac{2 \cdot b_{Ge}}{b_{Ga} \cdot (1 + \frac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga})} = Q \cdot \frac{4 \cdot b_{Ge}}{3 \cdot b_{Gc}} \cdot \frac{1 + \frac{1}{4} \cdot K_1 \cdot k \cdot b_{Ga}}{(1 + \frac{1}{3} \cdot K_1 \cdot k \cdot b_{Ga})^2} \tag{4.13}$$

With the approximation: $E_0 \cdot k \cdot b_{Ga} \approx Q$ (4.14)

and omitting the factor 8/9, eq.(4.11) becomes:

$$b_{Gc} = \sqrt{\frac{F_G \cdot e_G}{\pi \cdot d_{Ge} \cdot (\Theta_F + \tilde{\Theta}_F) \cdot \left[E_0 + K_1 \cdot \frac{Q}{2}\right]}} = b_{Gc(el)} \tag{4.15}$$

for elastic behaviour of a gasket.

For plastic behaviour:

$$b_{Gc} = \frac{F_G}{\pi \cdot d_{Ge} \cdot Q_{max}} \tag{4.16}$$

True elasto-plastic deformation gives an effective-width greater than for pure-elastic and pure-plastic deformation, approximately:

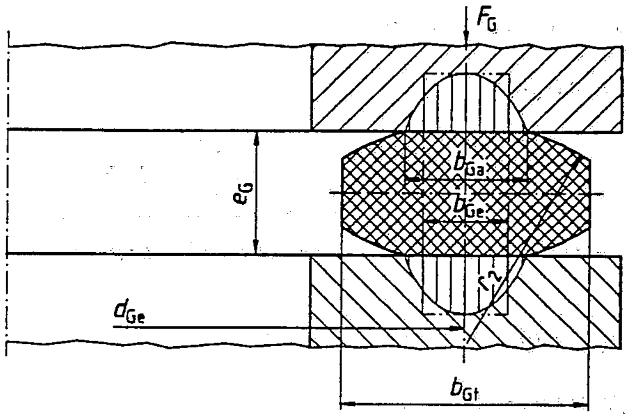
$$b_{Gc} \approx \sqrt{b_{Gc(el)}^2 + b_{Gc(pl)}^2} \tag{4.17}$$

From eqs. (4.1), (4.15) and (4.16):

$$b_{Gc} = \sqrt{\frac{e_G}{\pi \cdot d_{Ge} \cdot \left[h_g \cdot \frac{Z_F}{E_F} + \tilde{h}_G \cdot \frac{\tilde{Z}_F}{\tilde{E}_G}\right] \cdot \left[E_0 + K_1 \cdot \frac{Q}{2}\right]} + \left(\frac{F_G}{\pi \cdot d_{Ge} \cdot Q_{max}}\right)^2} \tag{4.18}$$

This is the equation in Table 1 of prEN 1591-1 ('Type 1'). The auxiliary notation b_{Gc} is introduced, because eq. (4.18) may give $b_{Gc} > b_{Gt}$, but it is necessary that $b_{Ge} \leq b_{Gt}$. (Later the denomination b_{Gc} was changed into b_{Gi}, because b_{Gc} and b_{Ge} are too similar.)

4.1.2 Effective width of gaskets with curved surfaces



Notation:

b_{Ga} = contact width
b_{Ge} = effective width
b_{Gc} = calculated width

Figure 4

For elastic deformation of the gasket (Hertzian contact) the contact width is :

$$b_{Ga}^2 = \frac{32}{\pi} \cdot \frac{F_G \cdot r_2 \cdot (1 - \nu^2)}{E_G \cdot \pi \cdot d_{Ge}} \qquad (4.19)$$

and the maximum contact pressure, assumed equal to the mean pressure in b_{Ge} , is:

$$Q_{max} = \frac{4}{\pi} \cdot \frac{F_G}{\pi \cdot d_{Ge} \cdot b_{Ga}} \qquad (4.20a)$$

and $Q_{max} = Q$ where:

$$Q = \frac{F_G}{\pi \cdot d_{Ge} \cdot b_{Gc}} \qquad (4.20b)$$

therefore

$$b_{Gc} = b_{Ga} \cdot \frac{\pi}{4} \qquad (4.21a)$$

i.e.

$$b_{Gc} = \sqrt{\frac{F_G \cdot r_2 \cdot 2 \cdot \pi \cdot (1 - \nu^2)}{E_G \cdot \pi \cdot d_{Ge}}} \qquad (4.21b)$$

If the contacting surfaces are inclined at angle φ_G to the reference plane as drawn, the normal forces on the curved surfaces are $F_G / \cos \varphi_G$ (instead of F_G), the real contact width increases with $\sqrt{1/\cos \varphi_G}$, but the calculated width is the projection on the reference plane (projection factor $\cos \varphi_G$). Therefore:

$$b_{Gc} = \sqrt{\frac{F_G \cdot \cos \varphi_G \cdot r_2 \cdot 2 \cdot \pi \cdot (1 - \nu^2)}{E_G \cdot \pi \cdot d_{Ge}}} = b_{Gc(el)} \tag{4.22}$$

The plastic width $b_{Gc(pl)}$, and approximation for elasto-plastic behaviour, correspond to eqs. (4.16) and (4.17). Table 2 in prEN 1591-1 is simplified by assuming $2 \cdot \pi \cdot (1 - \nu^2) \approx 6$ (similar to $\frac{8}{9} \approx 1$ for flat gaskets).

Double-contact:
Each surface sees half the force hence the calculated width is double that for one surface with half the force, which is the basis of Table 2 in prEN 1591-1.

4.2 Elastic stiffness of gaskets

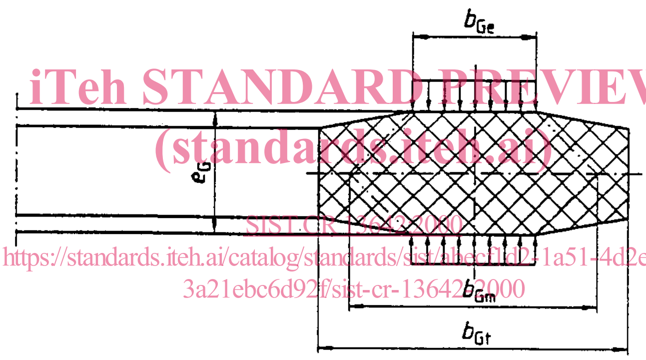


Figure 5

A simple model gives:

$$\Delta U_G = \frac{\Delta F_G \cdot e_G}{E_G \cdot \pi \cdot d_{Ge} \cdot \bar{b}_G} \; ; \; b_{Ge} \le \bar{b}_G \le b_{Gt} \tag{4.23}$$

The active (deformed) width assumed is indicated by the dashed line in the diagram (defined by the angle $\pi/4$) which gives a maximum 'active width' (mid-plane):

$$b_{Gm} = b_{Ge} + 2 \cdot \frac{e_G}{2} \; , \; (b_{Gm} \le b_{Gt}) \tag{4.24}$$

and average active width:

$$\bar{b}_G = b_{Ge} + e_G/2 \; , \; (\bar{b}_G \le b_{Gt}) \tag{4.25}$$