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Comparative Analysis of Current International Standards for Calculations Flanges Joint with Gasket Inside the Circle Location of the Bolt Holes

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***Abstract**: This paper presents the comparative analysis of the existing international normative calculation (EN 13445-3 European, ASME-Code, Section VIII, British Standard (PD 5500: 2009)) on the flange gasket joint situated inside the circle location of the bolt holes. The comparative analysis was performed for the same type of flange. The paper presents a comparative study of American and European standard, in terms of cost of production for the pressure equipment to assess the strengths and weaknesses of these regulations.*

***Keywords**: flange joint, gasket*

1.Introduction

Why did I started comparative analysis for the standards? Because the analysis performed in specialized literature [1] it was found that there may be differences between the normative technically and economically. In the [1] study the analysis was performed for nine pressure Vessels to show technical and economic advantages or disadvantages for applying standards (EN 13445 DBF, ASME VIII Division 1 (Code stamp), ASME VIII Division 1 (CE)). The result is clear: for at least the same safety level the European standard EN 13445 enables in many cases a more cost-effective production of pressure equipment than the American ASME Boiler & Pressure Vessel Code [1].

In ASME the costs of materials and production are higher than those resulting from the use of EN 13445 because of higher wall thickness, heat treatment applied after welding [1].

These results are presented in

Table **1** where the abbreviation DBF means design according to formulas, the abbreviation DBA means design according to stress analysis.

Also, an analysis of the current norms, in which their shortcomings were highlighted, was carried out in the specialized publications [2 - 5].

Table 1
Comparing results with international standard in terms of production costs [1]

Vessel /Code	EN 13445 DBF (CE)	EN 13445 DBA (CE)	ASME VIII Division 1 (Code stamp)	ASME VIII Division 1 (CE)	ASME VIII Division 2 (Code stamp)	ASME VIII Division 2 (CE)
Natural gas storage tank	100,0 %	95,6 %	130,4%	138,5 %	118,1 %	117,9 %
Hydrogen reactor (welded course)	100,0 %	-	115,9 %	122,8 %	106,5 %	110,5 %

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Vessel /Code	EN 13445 DBF (CE)	EN 13445 DBA (CE)	ASME VIII Division 1 (Code stamp)	ASME VIII Division 1 (CE)	ASME VIII Division 2 (Code stamp)	ASME VIII Division 2 (CE)
Hydrogen reactor (forged course)	100.0 %	-	94.3 %	94.9 %	84.9 %	85.3 %
Autoclave	100.0 %	-	97.9 %	98.6 %	-	-
Stirring vessel (Impeller type mixer)	100.0 %	-	110.6 %	110.6 %	-	-
AES heat exchanger	100.0 %	-	100.3 %	101.8 %	-	-
BEM heat exchanger	100.0 %	-	99.0 %	101.9 %	-	-
NEN heat exchanger	100.0 %	-	108.2 %	106.9 %	-	-
Water separator	100.0 %	-	105.6 %	110.1 %	-	-
Header of an air-cooler	100.0 %	88.1 %	106.7 %	108.2 %	-	-

The comparative analysis of current international normative calculation was made for joint flanges with gasket located inside the circle on which are located tensioning bolts for the gasket (**Error! Reference source not found.**).inside the circle on which are located tensioning bolts for the gasket (Figure 1).

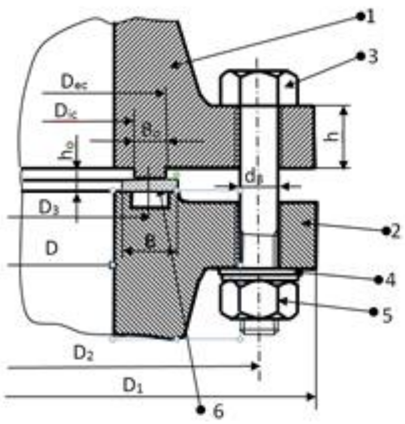


Figure 1. The flange gasket inside the circle location holes for bolts [6 -8]:1,2 – flanges; 3 – bolt; 4 – washer; 5 – nut; 6 – gasket

The current official calculation methods for calculation of flanged joints of pressure vessels [9- 11] contain the strength calculation which includes the choice of gasket; it intervenes only in the calculation of resistance. No references are made to the necessary rigidity of the flange joint or the issue of their sealing.

The calculations are made for:

– strain static state in operating conditions at calculation parameters under the pressure test of the assembly. In all the calculations are considered that flanges and bolts temperatures are equal, but inferior to the creep temperature;

– the fatigue loading with consideration of linear-elastic behavior of the material of which the flanges and bolts are constructed.

Flanges joint calculation for situations covered by the official standards

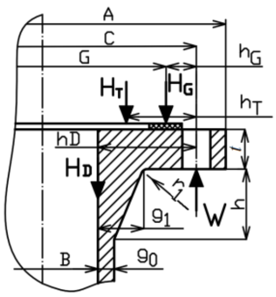
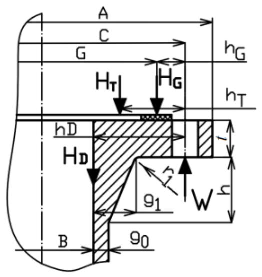
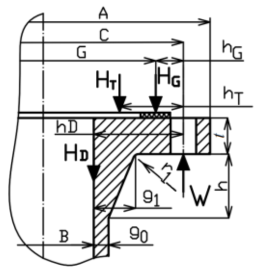
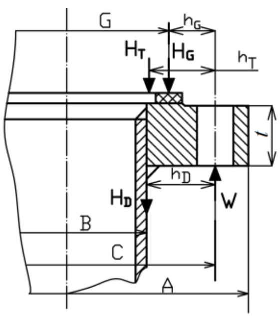
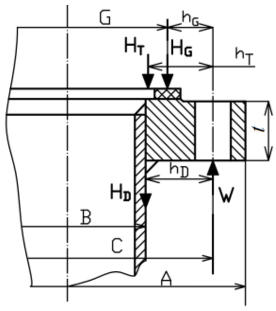
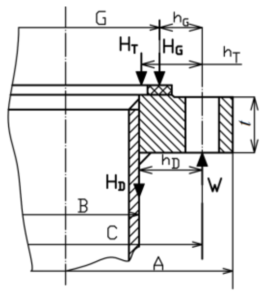
The calculation according to the standards [9-11] is based on the following assumptions:

– treats the flanges joint for gasket are arranged inside bolt circle ($d_{g,ext} < D_2$);

– the flanges bend under the influence of moments with flanges according to normative refers on only two cases, namely: - at environment condition, $T=T_0$ and $P=P_0$; - in operating condition at $T = T_s$ and $P=P_c$ (P_c - the calculation pressure, T_s - the bolt temperature), but at temperatures below the creep temperature.

In terms of the calculation both European and BS norms, flanges are divided into integral type, optional type flanges, loose hubbed type and loose flanges according to Table 2, and the ASME Code [0] classification is more detailed. They are assigned to either type integral flanges or loose flange type.

Table 2
Types of flanges foreseen in the normatives [0 – 0]

Flanges Type	ASME Code, Section VIII	EN 13445-3:2002	British Standard (PD 5500:2009)
Integral	 Fig. 1. Integral Type Flange with a Hub	Idem  Fig. 1 $e \equiv t$ where: *e* – flange thickness	Idem  Fig. 1 $N \equiv w$ where: *N* – gasket contact width, N = 0.0 for self-energizing gaskets
Optional	 Fig. 2. Loose Type Flange Without a Hub.	Idem  Fig. 2 $e \equiv t$ where: *e* – flange thickness	Idem  Fig. 2 $N \equiv w$ where: *N* – gasket contact width, N = 0.0 for self-energizing gaskets

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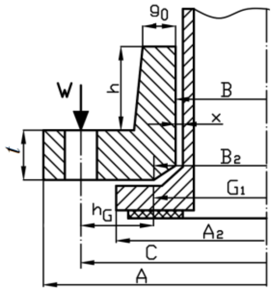
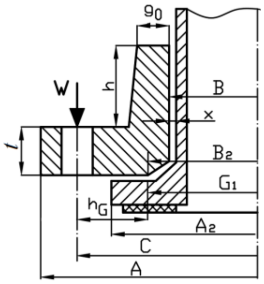
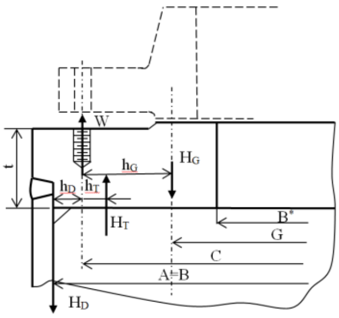
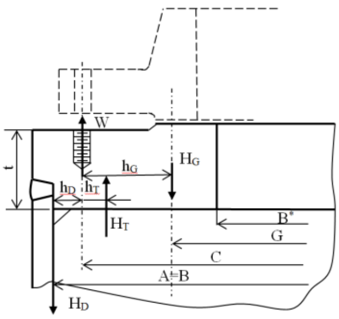
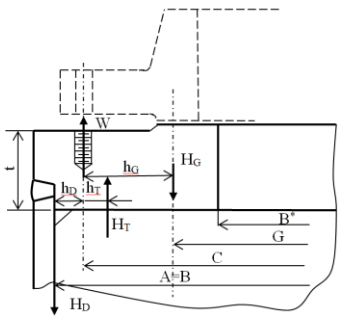
Flanges Type	ASME Code, Section VIII	EN 13445-3:2002	British Standard (PD 5500:2009)
Loose	Not included in the normative but are treated as loose type flange	 Fig. 3. Lap Type Joint; Loose Flange with Hub.	Idem  Fig. 3. where:, *t* – flange thickness
Reverse	 Fig. 4. Reverse Loose Type Flange	Idem  Fig. 4 $e \equiv t$ where: *e* – flange thickness	Idem  Fig. 4 $N \equiv w$ where: *N* – gasket contact width, N = 0.0 for self-energizing gaskets

Table 3 contains calculation relations for forces, bending moments and meridional stress according to the norms EN 13445-3, ASME–Code, Section VIII, and British Standard (PD 5500:2009).

Table 3
Table for relations calculation according to Applicable standards [0 – 0]

Codes / Causes	ASME Code, Section VIII	EN 13445-3:2002	British Standard (PD 5500:2009)
Forces	The minimum required bolt load for gasket seating condition: $W_{gs} = \pi \cdot b \cdot G \cdot (C_{us} \cdot y)$ (1) Force in operating condition: -total hydrostatic end force on the area inside of the flange: $H_D = 0{,}785 \cdot (B^2 \cdot P)$ (2) -compression load on gasket: $H_G = 2 \cdot \pi \cdot b \cdot G \cdot m \cdot P$ (3) -hydrostatic end force due to pressure on flange face:	Idem ASME Code, Section VIII, $W_A \equiv W_{gs}$ where: W_A – the minimum required bolt load for gasket seating	Idem ASME Code, Section VIII, $W_{m2} \equiv W_{gs}$ where: W_{m2} – the minimum required bolt load for gasket seating

Codes / Causes	ASME Code, Section VIII	EN 13445-3:2002	British Standard (PD 5500:2009)
	$\mathrm{H_T} = H - H_D$ (4) - total hydrostatic end force: $H = 0{,}785 \cdot \left(G^2 \cdot P\right)$ (5)		
Bending moments meridional	The moment of the flange at initial load: $M = \mathrm{M_A} \cdot \frac{C_{us}}{B_{sc}}$ (6) $\mathrm{M_A} = W \cdot h_G$ (7) The moment in operating condition for Integral Type Flanges: -displacement: $\mathrm{M_D} = H_D \cdot \left[\frac{C-(B+g_1)}{2}\right]$ (8) -sealing: $\mathrm{M_G} = H_G \cdot \left(\frac{C-G}{2}\right)$ (9) - additional due to the pressure on the flange surface, $\mathrm{M_T} = H_T \cdot \left[\frac{C}{2} - \frac{(B+G)}{4}\right]$ (10) -the total moment acting upon flange for operating condition, $M = M_o \cdot \frac{C_{us}}{B_{sc}}$ (11) $M_o = M_D + M_G + M_T$ (12)	Idem ASME Code, Section VIII, $\mathrm{M_{op}} \equiv \mathrm{M_o}$ where: $\mathrm{M_{op}}$ – total moment acting upon flange for operating condition	Idem ASME Code, Section VIII, $\mathrm{M_{op}} \equiv \mathrm{M_o}$ where: $\mathrm{M_{op}}$ – total moment acting upon flange for operating condition
Stress	1. Integral Type Flange or Loose Type Flange with a Hub: a) Flange hub stress: $S_H = \frac{f \cdot M_o}{L \cdot g_1^2 \cdot B}$ (13) b) Flange radial stress: $S_R = \frac{(1{,}333 \cdot t \cdot e + 1)}{L \cdot t^2 \cdot B}$ (14) c) Flange tangential stress: $S_T = \frac{Y \cdot M_o}{t^2 \cdot B} - Z \cdot S_R$ (15) 2. Loose Type Flange without a Hub: a) Longitudinal and radial flange hub stress: $S_R = S_H = 0$ (16) c) Flange tangential stress: $S_T = \frac{Y \cdot M_o}{t^2 \cdot B}$ (17) 3. Reverse Integral Type Flange or Reverse Loose Type Flange with a Hub: a) Longitudinal hub stress: $S_H = \frac{f \cdot M_o}{L_r \cdot t^2 \cdot B}$ (18)	Idem ASME Code, Section VIII, $\sigma_z \equiv S_H$ $\sigma_r \equiv S_R$ $\sigma_\theta \equiv S_T$ where: σ_z – flange hub stress; σ_r – flange radial stress; σ_θ – flange tangential stress;	Idem ASME Code, Section VIII.

Codes / Causes	ASME Code, Section VIII	EN 13445-3:2002	British Standard (PD 5500:2009)
	b) Flange radial stress: $$S_R = \frac{(1{,}333 \cdot t \cdot e_r + l) \cdot M_o}{L_r \cdot g_1^2 \cdot B^*} \quad (19)$$ c) Flange tangential stress: $$S_{T1} = \frac{Y_r \cdot M_o}{t^2 \cdot B^*} - \frac{Z \cdot S_R \cdot (0{,}67 \cdot t \cdot e_r + 1)}{(1{,}33 \cdot t \cdot e_r + 1)} \quad (20)$$ $$S_{T2} = \left[Y - \frac{2 \cdot K^2 (0{,}67 \cdot t \cdot e_r + l)}{(K^2 - l) \cdot L_r}\right] \cdot \frac{M_o}{t^2 \cdot B^*} \quad (21)$$ 4. Reverse Loose Type Flanges without a Hub: a) Longitudinal and radial flange hub stress: $$S_R = S_H = 0 \quad (22)$$ c) Flange tangential stress: $$S_T = \frac{Y \cdot M_o}{t^2 \cdot B^*} \quad (23)$$		

The nomenclature for
Table **2** and Table 3:

A – outside diameter of the flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots;
b – effective gasket contact width;
B – inside diameter of the flange;
B^* – inside diameter of the reverse flange;
B_{sc} – bolt spacing correction factor;
b_0 – basic gasket seating width;
C – bolt circle diameter;
e, *L*, – flange stress factor;
e_r, L_r – flange stress factor *e* for a reverse type flange;
G – diameter at the location of the gasket load reaction;
g_1 – thickness of the hub at the large end;
g_o – thickness of the hub at the small end;
h – hub length;
h_D – moment arm for load H_D;
h_G – moment arm for load H_G;
h_T – moment arm for load H_T;
K – ratio of the flange outside diameter to the flange inside diameter;
P – design pressure;
t – flange thickness;
w – width of the nubbin;
Z, *Y* – flange stress factor;
$C_{us} \equiv C_F$; β_T ; β_U ; β_Y ; β_Z ; λ and d are corrections factors according to EN 13445-3 [11].

Conclusions regarding the calculation of flanged joints based on current normative

In Table 3, we centralized the formulas for flanged joints according to current standards [9-11], it is found that:

– there are no calculation differences between relationship: forces and bending moments; calculating the stress; calculating allowable stress;

– but there are the following differences: standards using different symbols regarding forces, bending moments, stress, length, correction factors, etc. ASME norm also do not treat loose flanges type but only those integral type and optional;

The current normative provide calculations for flanges, bolts and gaskets only for $\Delta T_{fs} = 0$.

When calculating fatigue life currently in normative, the Palmgren – Miner rule is recomanded,

$$\sum_i \left(\frac{n}{N}\right)_i = C \qquad (24)$$

where *C* has values as follows:

- in ASME Code, *C=1*; in EN 13445, *C=0.8*; *0.4* or *0.3*; in BS 5500, *C=0.8*; *0.4* or *0.3*.

In this relationship is not taken into consideration:
– mean stress influence but only the amplitude stress;
– residual stress influence;
– the damage caused by the influence of various external factors.

● **Proposals:**

1 – it is necessary to supplement existing normative by introducing:
– the effect of temperature difference $\Delta T_{fs} \neq 0$, to the strength and tightness;
– influence of flange rigidity regarding tightness.

In papers [7; 12] the problem of the influence of ΔT_{fs} in normal operating and transitory regime was solved and can be used;

2 – it is suggested to use the V. V. JINESCU relationship [13], for calculation the fatigue life:

$$\sum_i \left(\frac{n}{N}\right)_i^{\frac{\alpha+1}{m}} = C_\sigma, \qquad (25)$$

where;

$$C_\sigma = 1 - \left(\frac{\sigma_m}{\sigma_u}\right)^{\alpha+1} \cdot \delta_{\sigma_u} - \left(\frac{\sigma_{res}}{\sigma_u}\right)^2 \cdot \delta_{\sigma_{res}} - D_T,$$

with k=0.16…1.0 and *m*=3…5 – for various steels [13]. D_T is the deterioration, dimensionless with values $D_T \in (0;1]$.

α, *m* – constants of material; $\alpha = 1/k$, becomes from nonlinear material behavior law,

$$\sigma = \mathrm{M}_\sigma \cdot \varepsilon^k, \qquad (26)$$

where M_σ; *k* – constants; σ – normal stress; ε – strain.

Generally, $\alpha = 0$ – through the shock loading and $\alpha = 1$ for linear-elastic behavior under monotonic loading;

σ_m – mean stress; σ_u – ultimate stress; σ_{res} – residual stress; D_T – deterioration or damage.

In the case of a sample without residual stresses $(\sigma_{res} = 0)$ and undeteriorated $D_T = 0$ [13]:

$C_\sigma = 1$, for alternating symmetrical loading $(R = -1)$ when $\sigma_m = 0$;

$C_\sigma < 1$, if $\sigma_m > 0$ and $C_\sigma > 1$, if $\sigma_m < 0$ $(R \neq -1)$.

- in the case of a sample without residual stresses (σ_{res}=0) but cracked and pre-loaded sample (when $D_T \neq 0$): $C_\sigma = C_\sigma(n) < 1$, for alternating symmetrical loading $(R = -1)$;

$C_\sigma < 1$ or $C_\sigma > 1$ in the general case $(R \neq -1)$.

Calculation of fatigue flanges according to the standards is incomplete and is necessary to be taken into consideration the effect of mean stress, residual stresses and deterioration. This is found only in the relationship proposed in the paper [13].

Deterioration and residual stresses

The deterioration of weldements due to flaws or cracks, as well as the residual stresses in the heat affected zone may reduce the strength of the flange joint.

As to take into consideration the:

-deterioration and the residual stresses one calculates the critical stresses, σ_{cr}, with the relations proposed in the papers [14-18];

-superposition of different loads one may use the results reported in the papers [19-25];

-fatigue of flanges joint due to pressure or/and temperature fluctuations is usefull to consider the relationships proposed in the papers [26-30].

2.Conclusions

In the current international normative [9-11], the calculation of the flange joints with gasket situated inside the circle location of the bolts holes refers only to static load, operating condition and pressure test.

It is necessary to complete the current normative by introducing: the effect of the temperature difference between the flange and the bolts, $\Delta T fs \neq 0$, on the strength and sealing and the influence of the rigidity of the flange on the sealing. A new method of evaluating the strength of flanged joints on the basis of the principal of critical energy is proposed, in which – unlike everything that has been proposed so far – the influence of deterioration and residual stresses are introduced.

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