Dimensioning a
Heat Exchanger Flange

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In addition, we have been providing engineering services such as the calculation of columns, heat exchangers and other types of pressure vessels for more than 20 years. We use this practical experience and the interaction with our clients to further optimize the software.

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1. Introduction

This article explains how to dimension a custom flange for a heat exchanger. From the internal diameter B the flange dimensions are determined step by step on the basis of a number of simple rules. The end result is a feasible flange with dimensions that can be used to do a stress analysis with, for example, the P3 Engineering VES software. The VES software includes implementations for norm calculations for flanges according to ASME VIII Division 1 [A], EN 13445-3 [B], AD 2000 Regelwerk [C], and Dutch Rules for Pressure Vessels (RTOD) [D].

2. Design data

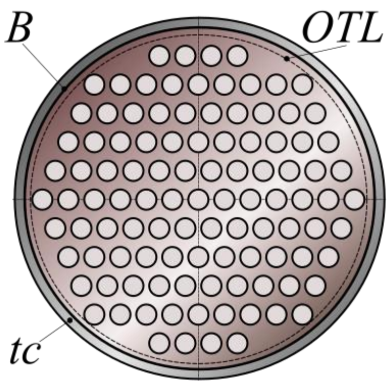
The following information is required to begin flange design.

a) The design conditions:

P_d = *Design pressure*	*[MPa]*	(1)
T_d = *Design temperature*	*[°C]*	(2)
C_a = *Corrosion allowance*	*[mm]*	(3)

b) The Outer Tube Limit:

OTL = *The smallest diameter that includes all tubes in the bundle.*	*[mm]*	(4)



B	*Internal diameter of the enveloping cylinder*	*[mm]*
OTL	*Smallest diameter around all tubes in the bundle*	*[mm]*
t_c	*Wall thickness of the cylinder*	*[mm]*

Figure 1. Cross sections of a cylinder containing a tube bundle.

c) The bolt dimension:

d_b = *start with ¾" or M20*	*[mm]*	(5)

For large flange diameters or high design pressure, a larger bolt size d_b can be chosen to start the design.

In a flange norm calculation, the total bolt area is important for a leak-proof seal. The total bolt area A_b is equal to the number of bolts n_b times the bolt area per bolt A_k:

$$A_b = n_b A_k \qquad [mm] \qquad (6)$$

With the permissible tension of the bolt material equal to

$$f_{ba} \qquad [MPa] \qquad (7)$$

the total maximum bolt load can be calculated:

$$F_{nb} = f_{ba} A_b \qquad [N] \qquad (8)$$

For a leak-free flange connection, the maximum bolt load F_{nb} must be sufficient to apply the required gasket pressure. For more information, see Section 5 on bolt dimensions.

Bolts with a larger diameter d_b have a cross section with a larger area A_k. A larger diameter therefore results in a smaller number of bolts n_b needed to reach the same bolt area A_b. In Section 4 about the flange dimensions, it will be shown that with a larger bolt diameter d_b, the external diameter A of the flange will also increase.

d) The compression factor m of the gasket

$$m = 2\text{, for camprofile gaskets} \qquad (9)$$

$$m = 3\text{, for spiral wound gaskets}$$

$$m = 5.5\text{, for steel flat rings}$$

To prevent leakage the gasket requires a minimum gasket pressure under operational conditions. This minimum gasket pressure is proportional to the size of the compression factor m. The size of the compression factor depends on the type of gasket and the materials used (see Table 4. *Gasket properties* m *and* y). A higher compression factor m results in a higher required operational gasket pressure. To guarantee this gasket pressure, a higher bolt load F_{nb} is required, i.e. a larger number of bolts n_b or a larger bolt diameter d_b.

3. Cylinder dimensions

The flange dimensions are constructed from the internal diameter B (see Figure 1). The flange dimensions are determined from the inside out. Afterwards, when the external diameter A has been calculated, the gasket dimensions can also be determined.

First, this internal diameter B is calculated using the following formula:

$$B \geq OTL + 12 \qquad [mm] \qquad (10)$$

The internal diameter B may have to be larger depending on the function or construction of the heat exchanger. For example in case of a boiler type heat exchanger or when space is required for sliding strips.

Secondly, the required wall thickness t_{cr} of the connecting cylinder is determined. Figure 2 shows the required wall thickness of t_{cr} for a carbon steel cylinder with an external diameter of $D_o = 1000$ *[mm]* as a function of the design pressure P_d and the design temperature T_d. The wall thickness t_{cr} is calculated using the formulas of ASME VIII Division 1 - Appendix 1 [A] and the material properties from ASME II - Part D [E].

$$t_{cr} = \frac{P_d R_o}{S E_i + 0.4 P_d} \qquad [mm] \qquad (11)$$

R_o = External radius of the cylinder *[mm]*

S = Permissible stress of the material *[MPa]*

E_i = Weld factor of the longitudinal seam of the cylinder *[-]*

See Appendix A for tabulated values of t_{cr} as a function of design pressure P_d and design temperature T_d.

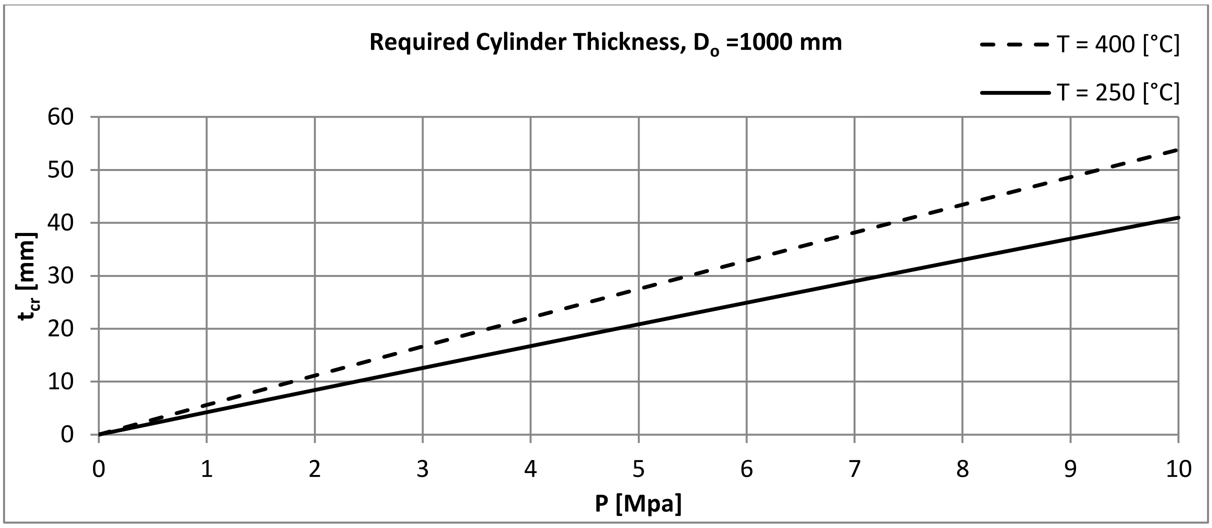


Figure 2. Required cylinder wall thickness t_{cr}.

For diameters other than Do = 1000 mm, an estimate of the cylinder wall thickness t_c can be made using Figure 2 and the formula below.

$$t_c = \frac{B\, t_{cr}}{1000 - 2\, t_{cr}} + C_a \qquad [mm] \qquad (12)$$

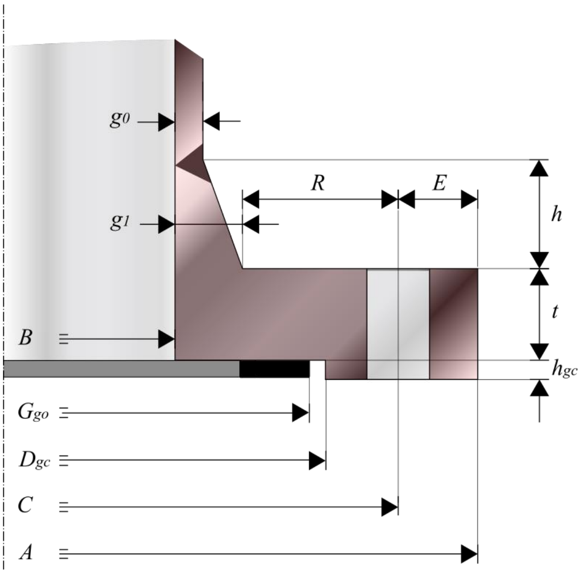
In addition a manufacturing tolerance t_{tol} on the wall thickness t_c must be taken into account in case a standard pipe is used.

The tolerance for pipes according to ASME B36.10M [F] and ASME B36.19M [G] is:

$$t_{tol} = 12.5 \qquad [\%] \qquad (13)$$

For pipes made of materials from the European standards EN 10216 [H] and EN 10217 [I] there are different tolerances depending on the type of material, carbon steel or stainless steel, the diameter and the wall thickness.

4. Flange dimensions



t	*Flange thickness*	
h_{gc}	*Confining height*	
B	*Internal diameter of the flange*	*[mm]*
g_0	*Cylindrical wall thickness at welding end*	*[mm]*
g_1	*Cylindrical wall thickness at flange blade*	*[mm]*
h	*Hub height*	*[mm]*
R	*Radial distance from hub to bolt circle C*	*[mm]*
E	*Radial distance from bolt circle C to external diameter A of the flange*	*[mm]*
A	*External diameter of the flange*	*[mm]*
C	*Diameter of the bolt circle*	*[mm]*
D_{gc}	*Confining diameter of the gasket*	*[mm]*
G_{go}	*External diameter of the gasket*	*[mm]*

Figure 3. Flange nomenclatures for ASME VIII Division 1.

The hub thickness at the welding end is equal to the wall thickness t_c of the cylinder:

$$g_0 = t_c \qquad [mm] \qquad (14)$$

The hub has a minimum height that depends on the machines and tools used. In this case a minimum hub height of 15 mm has been chosen. Increase the minimum hub height if more space is needed around the hub. For example as a result of the manufacturing process.

$$h_{min} = 15 \qquad [mm] \qquad (15)$$

The height of the hub also depends on the thickness g_0 at the welding end. This dependency is given by the hub height coefficient:

$$C_{hh} = 2 \qquad (16)$$

From this follows the formula for the height of the hub:

$$h = C_{hh} g_0 \geq h_{min} \qquad [mm] \qquad (17)$$

For a conical hub a slope coefficient is used:

$$C_{hs} = 3 \tag{18}$$

The thickness of the hub at the flange blade can be calculated using C_{hs} using the following formula:

$$g_1 = \frac{h}{C_{hs}} + g_0 \quad [mm] \tag{19}$$

The external diameter of the hub is then calculated as follows:

$$D_h = B + 2g_1 \quad [mm] \tag{20}$$

Depending on the application or manufacturing method, different values for h_{min}, C_{hh}, and C_{hs} can be chosen. Please make sure that the slope of the hub still meets the requirements of the selected flange calculation norm. The slope coefficient C_{hs} should not be chosen too small.

Appendix B contains the minimum values for radial flange sizes R and E. These minimum sizes ensure that there is enough space for mounting tools and that the nuts do not protrude beyond the flange. The radial distance from the hub to the bolt circle has to satisfy:

$$R \geq S_{Hmin} \quad [mm] \tag{21}$$

The size S_{Hmin} is based on a socket wrench or a torque wrench. Other values are available for hydraulic tightening equipment that depends on the diameter of the hydraulic head. Each tool supplier has its own datasheet with minimum radial distances for R and E.

There is also a minimum value for the radial distance E from bolt circle C to the external diameter of the flange blade A:

$$E \geq S_{Emin} \quad [mm] \tag{22}$$

Now the diameter of the bolt circle C can be calculated as follows:

$$C = D_h + 2R \quad [mm] \tag{23}$$

And for the external diameter A of the flange blade the following rule applies:

$$A = C + 2E \quad [mm] \tag{24}$$

Because the minimum radial distance E increases for larger bolt diameters, the external diameter A will also increase when choosing a larger bolt diameter d_b.

The exact flange thickness cannot yet be determined, because for a norm calculation all dimensions of the flange must first be known. Therefore an estimate for the flange thickness t is used as a starting point. The estimate is a function of the previously determined cylinder thickness t_c and a thickness coefficient C_{ft}. The initial estimate for the flange thickness is calculated as follows:

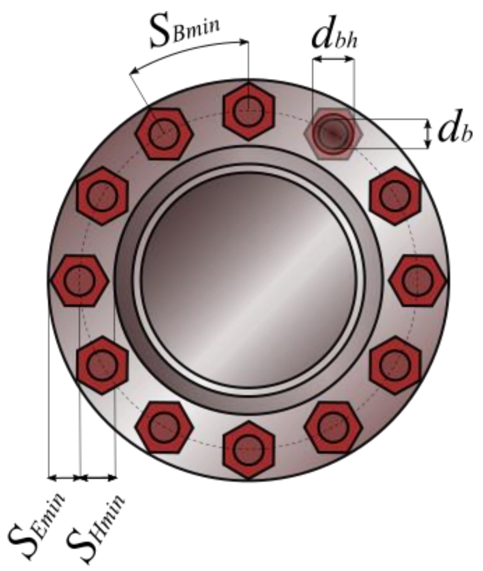
$$C_{ft} \in [5,6,7] \tag{25}$$

$$t = C_{ft}\, t_c \qquad [mm] \qquad (26)$$

With an increasing cylinder thickness t_c, which depends on the design pressure P_d, the estimated flange thickness t will increase proportionally. The final norm calculation will show whether this flange thickness t is sufficient. If this is not the case, the optimum flange thickness must be found in an iterative way. Different flange norms will result in a different flange thickness.

5. Bolt dimensions

To determine the bolt dimensions, use the tables in *Appendix B* . In these tables, the bolt dimensions and fitting dimensions are given as a function of the bolt diameter.



d_b	*Bolt diameter*	*[mm]*
d_{bh}	*Hole diameter for the bolts*	*[mm]*
S_{Bmin}	*Minimum center-to-center distance between the bolts measured over bolt circle C*[1]	*[mm]*
S_{Hmin}	*Minimum radial distance from hub to bolt circle C*[1]	*[mm]*
S_{Emin}	*Minimum radial distance from bolt circle C*[1] *to the outer diameter A*[1] *of the flange*	*[mm]*

[1]*For A and C see Figure 3.*

Figure 4. Bolt configuration.

The nominal bolt diameter is:

$$d_b \qquad (27)$$

The following rules apply to the diameter of the bolt hole:

$$d_{bh} \geq d_b + 2.5\ for\ M12 \qquad [mm] \qquad (28)$$

$$d_{bh} \geq d_b + 3\ for\ 3/4\,'' \qquad [mm]$$

…

$$d_{bh} \geq d_b + 10\ mm\ for\ M56 \qquad [mm]$$

A smaller bolt hole diameter d_{bh} is possible if the misalignment of the flange is limited during installation. In this case the following applies to all bolt holes:

$$d_{bh} \geq d_b + 3 \qquad [mm] \qquad (29)$$

The maximum distance between the bolts S_{bmax} can be determined in two ways. The first way is according to TEMA *RCB-11.23- Load Concentration Factor* [J] in which S_{bmax} is determined as follows:

$$S_{bmax} = 2d_b + \frac{6t}{m + 0.5} \qquad [mm] \qquad (30)$$

The second way is to determine the maximum distance S_{bmax} between the bolts using ASME VIII Division 1 - Appendix 2 [A]:

$$S_{bmax} = 2d_b + t \qquad [mm] \qquad (31)$$

Limiting the distance between the bolts should prevent the seal from leaking. If the bolts are too far apart, there may not be sufficient gasket pressure in the area between the bolts. Leakage may occur in this area. If the bolt distance is still greater, the calculated flange moment in the norm calculation must be increased by a factor of C_{Mb}:

$$C_{Mb} = \sqrt{\frac{S_b}{S_{bmax}}} \geq 1.0 \qquad (32)$$

The minimum distance between the bolts can be determined using the tables in *Appendix B* :

S_{bmin} = see *Appendix B Bolt dimensions* *[mm]* (33)

The minimum number of bolts n_{bmin} can now be determined with:

$$n_{bmin} = \pi \frac{C}{S_{bmax}} \qquad (34)$$

Similarly the maximum number of bolts can be determined with:

$$n_{bmax} = \pi \frac{C}{S_{bmin}} \qquad (35)$$

The arithmetic mean is used to estimate the required number of bolts n'_b:

$$n_b \geq n'_b = \frac{(n_{bmin} + n_{bmax})}{2}\,;\; n_b \in [4,8,12,\ldots] \qquad (36)$$

The number of bolts n_b must be a multiple of 4. For n_b, choose the next multiple of 4 bigger or equal to n'_b.

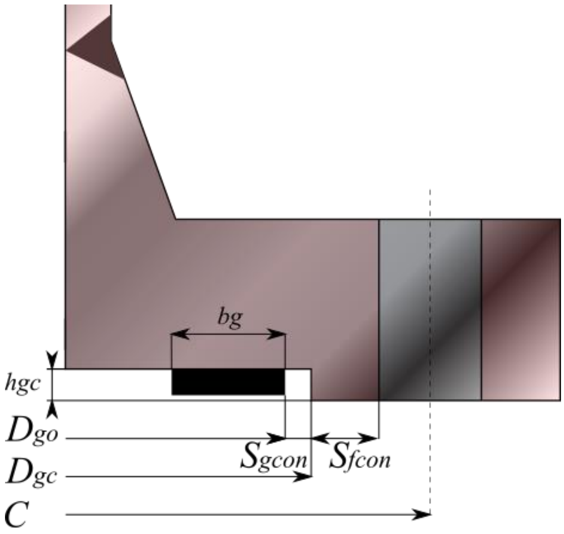
The actual number of bolts n_b in the flange design can only be determined using a norm calculation such as ASME VIII Division1 - Appendix 2 [A]. The calculation will show if there are enough bolts to keep the gasket closed in the following states:

1) When mounting with a minimum gasket pressure $P_g = y$.
2) During operational conditions with a minimum gasket pressure $P_g = mP_d$.

Norm calculations for flanges have the total bolt area as input. To get to the desired value of the total bolt area $A_b = n_b A_k$ it is possible to vary both the number of bolts n_b and the bolt diameter d_b.

6. Gasket dimensions

When designing a flange for a heat exchanger, a narrow gasket is assumed to lie entirely within the bolt circle. In addition, there is a confining diameter D_{gc} within which the gasket lies.



b_g	*Gasket width*	*[mm]*
h_{gc}	*Confining depth for the gasket*	*[mm]*
S_{fcon}	*Radial distance between the edge of the bolt hole and the confining depth D_{gc}.*	*[mm]*
S_{gcon}	*Radial distance between the confining diameter D_{gc} and the external gasket diameter D_{go}.*	*[mm]*
C	*Diameter of the bolt circle*	*[mm]*
D_{gc}	*Confining diameter for the gasket*	*[mm]*
D_{go}	*External diameter of the gasket.*	*[mm]*

Figure 5. Gasket configuration.

The minimum radial distance from the edge of the bolt hole to the gasket confining diameter is at least equal to:

$$S_{fcon} = 6 \qquad \textit{[mm]} \qquad (37)$$

For large flange diameters, the value for S_{fcon} can be selected larger. Particularly for high design pressures, it can be advantageous to keep the gasket diameter as small as possible. This reduces the internal pressure surface of the flange. In that case S_{fcon} must be selected in such a way that the internal gasket diameter is close to the internal diameter of the flange.

The following formula applies to the confining diameter D_{gc} of the gasket:

$$D_{gc} = C - d_{bh} - 2S_{fcon} \qquad \textit{[mm]} \qquad (38)$$

Where the confining depth h_{gc} for the gasket is at least 6 mm:

$$h_{gc} = 6 \qquad \textit{[mm]} \qquad (39)$$

The confining depth must be greater than the gasket thickness.

The minimum radial distance between the confining diameter D_{gc} and the external diameter of the gasket is as follows:

$$S_{gcon} = 3 \qquad \textit{[mm]} \qquad (40)$$

This can be used to calculate the external diameter D_{go} of the gasket:

$$D_{go} = D_{gc} - 2S_{gcon} \qquad [mm] \qquad (41)$$

For the gasket width applies:

$$b_g = \in [13,16,19, \dots] \qquad [mm] \qquad (42)$$

If the gasket width b_g is too narrow, the surface pressure of the gasket for seating can be too high for large flange diameters or high design pressures. A larger gasket width b_g should then be selected (16 mm or larger). However, if the gasket width is too large, the bolt loads will be unnecessarily high during assembly. The final check of the gasket width b_g takes place during the norm calculation.

The gasket must stay completely within the confining diameter. For this reason, the following applies to the maximum gasket thickness:

$$t_{gmax} = h_{gc} - 2 \qquad [mm] \qquad (43)$$

7. Example

In this section an example will be given which demonstrates the effectiveness of the above approach as an initial step to quickly get to a flange design which is norm compliant. The initial design data are estimated using the procedure as described in this article and are given in the tables below. Numbers in the table refer to equation numbers in this article.

This initial flange design is then used as input for the VES software in order to do ASME VIII Division 1 flange norm calculations. We will show that after the initial design only a few iterations are needed in order to get to a norm compliant design.

1) Design Data

(1)	*Design pressure*	P_d	= 2.9	*[MPa]*
(2)	*Design temperature*	T_d	= 200	*[°C]*
(3)	*Corrosion allowance*	C_a	= 3	*[mm]*
(4)	*Smallest diameter around the pipes*	*OTL*	= 962	*[mm]*
(5)	*Bolt diameter*	d_b	= UNC 1”	*[in.]*
(9)	*Compression factor camprofile gasket*	*m*	= 2	*[-]*

2) *Flange dimensions*

(10)	$B \geq OTL + 12$	B	= 974	*[mm]*
(11)	*Estimated wall thickness* D_o *= 1000 [mm]*	t_{cr}	= 13	*[mm]*
(12)	$t_c = \frac{B\ t_{cr}}{1000 - 2\ t_{cr}} + C_a$	t_c	= 16	*[mm]*
(14)	$g_0 = t_c$	g_o	= 16	*[mm]*
(16)	*Height coefficient of the hub*	C_{hh}	= 2	-
(17)	$h = C_{hh} g_0 \geq h_{min}$	h	= 32	*[mm]*
(18)	*Slope coefficient of the hub*	C_{hs}	= 3	*[-]*
(19)	$g_1 = \frac{h}{C_{hs}} + g_0$	g_1	= 27	*[mm]*
(20)	$D_h = B + 2g_1$	D_h	= 1028	*[mm]*
(21)	$R \geq S_{Hmin} = 29$	R	= 35	*[mm]*
(23)	$C = D_h + 2R$	C	= 1098	*[mm]*
(22)	$E \geq S_{Emin} = 27$	E	= 27	*[mm]*
(24)	$A = C + 2E$	A	= 1152	*[mm]*
(25)	Estimated thickness coefficient $C_{ft} \in [5,6,7]$	C_{ft}	= 6	*[-]*
(26)	$t = C_{ft}\ g_0$	t	= 96	*[mm]*

3) *Bolt dimensions*

(27)	*Selected bolt diameter*	d_b	= 25.4	*[mm]*
(29)	$d_{bh} \geq d_b + 3$	d_{bh}	= 29	*[mm]*
(31)	$S_{bmax} = 2d_b + t$	S_{bmax}	= 146.8	*[mm]*
(34)	$n_{bmin} = \pi \frac{C}{S_{bmax}}$	*nbmin*	= 23.5	
(33)	S_{bmin} = see *Appendix B Bolt dimensions*	S_{bmin}	= 57	*[mm]*

(35)	$n_{bmax} = \pi \frac{C}{S_{bmin}}$	n_{bmax}	= 60.52	
(36)	$n_b \geq n'_b = \frac{(n_{bmin} + n_{bmax})}{2}$	n'_b	= 42.01	
(36)	*Number of bolts*	n_b	= 44	

4) Gasket dimensions

(37)	*Minimal radial distance bolt hole to* D_{gc}	S_{fcon}	= 6	*[mm]*
(38)	$D_{gc} = C - d_{bh} - 2S_{fcon}$	D_{gc}	= 1057	*[mm]*
(39)	*Confining height*	h_{con}	= 6	*[mm]*
(40)	*Minimal radial distance* D_{go} *to* D_{gc}	S_{gcon}	= 3	*[mm]*
(41)	$D_{go} = D_{gc} - 2S_{gcon}$	D_{go}	= 1051	*[mm]*
(42)	*Gasket width* $b_g = \in [13,16,19,\ldots]$	b_g	= 13	*[mm]*
(43)	$t_{gmax} = h_{gc} - 2$	t_{gmax}	= 4	*[mm]*

The above values are used as input for the VES software to perform an ASME VIII Division 1 flange norm calculation. The results are given in Appendix D. On pages 2 and 3 of the output from VES you can see that the initial design does not yet meet the requirements of ASME VIII Division 1.

On page 2, the total bolt area A_b is marked in red because the requirement $A_b \geq A_m$ has not yet been met. Here A_m is the bolt area which is required to ensure that the minimum gasket pressure is guaranteed during assembly and operational conditions.

$$A_b = n\, A_k = \mathbf{15642.00} \quad [mm^2] \qquad (44)$$

$$A_b \geq A_m = 16721.83 \quad [mm^2] \qquad (45)$$

On page 3 of the calculation, the stress S_{Co} is marked in red because the requirement $S_{Co} \leq S_{fo}$ has not been met.

$$S_{Co} = Max\left[\frac{(S_{Ho} + S_{Ro})}{2}, \frac{(S_{Ho} + S_{To})}{2}\right] = \mathbf{146.14} \quad [N/mm^2] \qquad (46)$$

$$S_{Co} \leq S_{fo} = 137.93 \quad [N/mm^2] \qquad (47)$$

In order for the flange to comply with the norm, the following two modifications are made.

First, the flange thickness t is increased by 4 mm, which will reduce the stress S_{Co}.

$$t = 96 + 4 = 100 \qquad [mm]$$

Secondly, the number of bolts n_b is increased by 4 in order to increase the total bolt area A_b.

$$n_b = 44 + 4 = 48$$

After these minor adjustments, the flange is recalculated using the VES software. The results now show that the bolt surface A_b and the stress S_{Co} also meet the requirements of ASME VIII Division 1.

$$A_b = n\,A_k = \mathbf{17064.00} \qquad [mm^2] \qquad (48)$$

$$A_b \geq A_m = 16721.83 \qquad [mm^2] \qquad (49)$$

$$S_{Co} = Max\left[\frac{(S_{Ho} + S_{Ro})}{2}, \frac{(S_{Ho} + S_{To})}{2}\right] = \mathbf{137.34} \qquad [N/mm^2] \qquad (50)$$

$$S_{Co} \leq S_{fo} = 137.93 \qquad [N/mm^2] \qquad (51)$$

8. Conclusion

Using a limited number of input data, it is possible to design a custom flange for a heat exchanger. By means of a number of estimated coefficients, a practical design can be realized that is reasonably close to the requirements for a flange norm calculation. The following parameters are available to guide the initial flange design:

Parameter		**Standard value**	
Bolt diameter	d_b	¾”, M20	$d_b \geq 3/4"$
Height coefficient of the hub	C_{hh}	= 2	$C_{hh} \geq 2.0$
Slope coefficient of the hub	C_{hs}	= 3	$C_{hs} \geq 3.0$
Estimated thickness coefficient	C_{ft}	= 6	$C_{ft} \in [5,6,7]$
The compression factor	m	= 2	$m \in [0 \ldots 6.5]$

In order to comply with the norm calculation, it will be necessary to do one or more iterations in which the estimation coefficients may have to be changed slightly in order to get closer to the desired result. Eventually the last optimizations can take place during the norm calculations, for example by changing the flange thickness t or the number of bolts n_b.

Literature

[A] ASME Boiler and Pressure Vessel Code, Section VIII Rules for Construction of Pressure Vessels, Division 1,release 2015
[B] EN 13445 Unfired pressure vessels – Part 3: Design, Issue 3 2016
[C] AD 2000 Regelwerk, Arbeitsgemeinschaft Druckbehälter
[D] Dutch Rules for Pressure Vessels (RTOD), Technical Committee for “Toestellen onder Druk”
[E] ASME Boiler and Pressure Vessel Code, Section II Materials, Part D, Properties, release 2015
[F] ASME B36.10M, Welded and Seamless Wrought Steel Pipe
[G] ASME B36.19M, Stainless Steel Pipe
[H] EN 10216, Seamless steel tubes for pressure purposes.
[I] EN 10217, Welded steel tubes for pressure purposes.
[J] TEMA: Standard of the Tubular Exchanger Manufacturers Association, Ninth Edition

Appendix A Cylinder wall thickness

Required cylinder thickness t_{cr} according to ASME VIII Division 1 [A] Appendix 1-1:

$$t_{cr} = \frac{P_d R_o}{SE + 0.4P_d} \qquad [mm] \qquad (52)$$

$$D_o = 1000 \qquad [mm] \qquad (53)$$

$$R_o = 500 \qquad [mm] \qquad (54)$$

$$E = 1.0 \qquad (55)$$

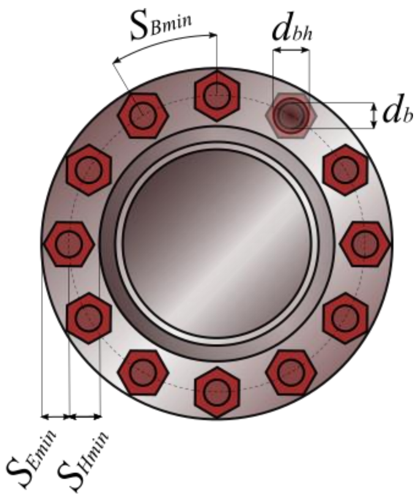
$$S = Allowable\ stress\ \boldsymbol{SA\ 516\ GR.65}, ASME\ II\ D \qquad [\text{MPa}] \qquad (56)$$

Table 1. Required wall thickness t_{cr} for a carbon steel cylinder with $D_o = 1000\ [mm]$.

Allowable stress S ASME II part D SA-516-Gr.60	T	S	P [MPa]										
	[°C]	[MPa]	0	1	2	3	4	5	6	7	8	9	10
	up to 250	118.00	0.00	4.22	8.42	12.58	16.72	20.83	24.92	28.97	33.00	37.01	40.98
	300	115.00	0.00	4.33	8.64	12.91	17.15	21.37	25.55	29.71	33.84	37.94	42.02
	325	112.00	0.00	4.45	8.87	13.25	17.61	21.93	26.22	30.49	34.72	38.93	43.10
	350	108.00	0.00	4.61	9.19	13.74	18.25	22.73	27.17	31.59	35.97	40.32	44.64
	375	104.00	0.00	4.79	9.54	14.26	18.94	23.58	28.20	32.77	37.31	41.82	46.30
	400	88.90	0.00	5.60	11.15	16.65	22.10	27.50	32.86	38.17	43.43	48.65	53.82

Appendix B Bolt dimensions

The bolt size tables contain bolt sizes and the minimum distances required to design a mountable flange (see TEMA [J] Tabel D-5).



d_b	*Bolt diameter*	*[mm]*
d_{bh}	*Bolt hole diameter*	*[mm]*
S_{bmin}	*Minimum circular distance between bolts*	*[mm]*
S_{Hmin}	*Minimum radial distance between outer hub diameter and the bolt circle*	*[mm]*
S_{Emin}	*Minimum radial distance between the bolt circle and the external flange diameter*	*[mm]*
r_b	*Hub to flange transition radius*	*[mm]*

Figure 6. Bold dimensions.

Table 2. Unified Inch thread.

Nominal diameter	Hole diameter	Minimum distance	Radial distance hub to bolt ciclel	Minimum corner radius	Radial distance bolt circle to edge
Bolts for non-standard flanges: Unified Inch thread					
d_b	d_{bh}	S_{Bmin}	S_{Hmin}	r_h	S_{Emin}
5/8"	19	38	24	8	19
3/4"	22	45	29	10	21
7/8"	25	52	32	10	24
1"	28	57	35	11	27
1 1/8"	32	64	38	11	29
1 1/4"	35	72	45	14	32
1 3/8"	38	78	48	14	35
1 1/2"	41	83	51	16	38
1 5/8"	44	89	54	16	41
1 3/4"	47	95	57	16	45
1 7/8"	51	102	60	16	48
2"	54	108	64	18	51
2 1/4"	60	121	70	18	57
2 1/2"	67	133	78	21	60
2 3/4"	73	146	86	22	67
3"	79	159	92	24	73

Table 3. Metric thread.

Nominal diameter	Hole diameter	Minimum distance	Radial distance hub to bolt circle	Minimum corner radius	Radial distance bolt circle to edge
Bolts for non-standard flanges: Metric thread					
d_b	d_{bh}	S_{Bmin}	S_{Hmin}	r_h	S_{Emin}
M12	15	31.75	20.64	7	15.88
M16	19	44.45	28.58	7	20.64
M20	23	52.39	31.75	9	23.81
M22	25	53.98	33.34	9	25.40
M24	27	58.74	36.51	9	28.58
M27	30	63.50	38.10	11	29.00
M30	33	73.03	46.04	11	33.34
M33	36	79.00	51.00	12	37.00
M36	39	84.14	53.97	12	39.69
M42	45	100.00	61.91	14	49.21
M48	51	112.71	68.26	15	55.56
M56	59	127.00	76.20	17	63.50
M64	67	139.70	84.14	19	66.68
M72	75	155.58	88.90	24	69.85
M80	83	166.69	93.66	27	74.61
M90	93	188.91	107.95	28	84.14
M100	103	207.96	119.06	30	93.66

Appendix C Gasket properties

m	*Compression factor of the gasket. To calculate the required operational bolt load, the design pressure P_d is multiplied by the compression factor m. This ensures sufficient gasket pressure in the operational state to prevent leakage.*
y	*Minimum surface tension required for a leak-proof flanged connection during installation. This parameter is used to calculate the minimum total bolt load for the seating condition.*
D_{go}	*Outside diameter*
b_g	*Gasket width*
t_g	*Gasket thickness*

Tables with gasket parameters m and y can be found in the calculation norms. They can also be found in the catalogues of gasket suppliers.

All flange calculation norms determine the minimum bolt load required for a leak-free connection using parameters m and y. However, not all norms check whether the maximum gasket pressure is exceeded. The maximum gasket pressure depends on the type of gasket, the materials used and the design temperature T_d and is usually found in the manufacturer's catalogue.

Table 4. Gasket properties *m* and *y*.

ASME VIII Division 1 Table 2-5.1 Gasket compression factor *m* and minimum design seating stress *y*			
Gasket types shape and material	**m [-]**	**y [psi]**	** [MPa]**
Self-engineering types	0.00	0	0.0
Elastomers without fabric or high percentage of mineral fiber			
Below 75A Shore Durometer	0.50	0	0.0
75A or higher Shore Durometer	1.00	200	1.4
Mineral fiber with binder			
1/8 inch (3.2 mm) thick	2.00	1600	11.0
1/16 inch (1.6 mm) thick	2.75	3700	25.5
1/32 inch (0.8 mm) thick	3.50	6500	44.8
Elastomers with cotton fabric insertion	1.25	400	2.8
Elastomers with mineral fiber fabric insertion with or without wire reinforcement			
3-ply	2.25	2200	15.2
2-ply	2.50	2900	20.0
1-ply	2.75	3700	25.5
Vegetable Fiber	1.75	1100	7.6
Spiral-wound metal, mineral fiber filled			
Carbon	2.50	10000	69.0
Stainless, Monel and nickel based alloys	3.00	10000	69.0
Corrugated metal, mineral fiber insert, or corrugated metal, jacketed mineral fiber filled			
Soft aluminum	2.50	2900	20.0
Soft copper or brass	2.75	3700	25.5
Iron or soft steel	3.00	4500	31.0
Monel or 4-6% chrome	3.25	5500	37.9
Stainless steel and nickel based alloys	3.50	6500	44.8
Corrugated metal			
Soft aluminum	2.75	3700	25.5
Soft copper or brass	3.00	4500	31.0
Iron or soft steel	3.25	5500	37.9

ASME VIII Division 1 Table 2-5.1 Gasket compression factor *m* and minimum design seating stress *y*			
Gasket types	**m**	**y**	
shape and material	**[-]**	**[psi]**	**[MPa]**
Monel or 4-6% chrome	3.50	6500	44.8
Stainless steels and nickel based alloys	3.75	7600	52.4
Flat metal, jacketed mineral fiber filled			
Soft aluminum	3.25	5500	37.9
Soft copper or brass	3.50	6500	44.8
Iron or soft steel	3.75	7600	52.4
Monel	3.50	8000	55.2
4-6% chrome	3.75	9000	62.1
Stainless steels and nickel based alloys	3.75	9000	62.1
Grooved metal			
Soft aluminum	3.25	5500	37.9
Soft copper or brass	3.50	6500	44.8
Iron or soft steel	3.75	7600	52.4
Monel or 4-6% chrome	3.75	9000	62.1
Stainless steels and nickel based alloys	4.25	10100	69.7
Solid flat metal			
Soft aluminum	4.00	8800	60.7
Soft copper or brass	4.75	13000	89.7
Iron or soft steel	5.50	18000	124.1
Monel or 4-6% chrome	6.00	21800	150.3
Stainless steels and nickel based alloys	6.50	26000	179.3
Ring joint			
Iron or soft steel	5.50	18000	124.1
Monel or 4-6% chrome	6.00	21800	150.3
Stainless steels and nickel based alloys	6.50	26000	179.3
Stainless steel camprofile			
Graphite layer low pressure	2.00	3200	22.1
Graphite layer medium pressure	2.00	6600	45.5
Graphite layer high pressure	2.00	10000	69.0
Spiral wound, SS 316L, graphite filled	3.00	10000	69.0

Appendix D Norm calculation

In this appendix a complete flange calculation according to the ASME VIII Division 1 Appendix 2 norm is made with our VES software. Page 1 contains the input data with the flange dimensions taken from the example calculation in chapter 7.

ASME VIII div.1 [Issue 2017] Appendix 2 Bolted flange connections				
Type of flange		Integral hub		
Design pressure	P	3.1	N/mm²	
Design temperature	T	200	°C	
Material		A-105	-	
Allowable stress at T	Sfo	137.93	N/mm²	
Allowable stress at 20 °C	Sfa	137.93	N/mm²	
Modulus of elasticity at T	E	198000	N/mm²	
Outside diameter	A	1152	mm	
Nominal inside diameter	Bn	974	mm	
Bolt circle diameter	C	1098	mm	
Mean gasket diameter	G	1038	mm	
Nominal flange thickness	tn	96	mm	(t = tn - Caf)
Allowance on flange facing	Caf	0	mm	(Optional: default Caf = 0)
Flange-to-shell connection				
Nominal hub thickness (small end)	g0n	16	mm	
Nominal hub thickness (large)	g1n	27	mm	
Hub height	h	32	mm	
Corrosion allowance	Ca	3	mm	
Bolting material		A-193-B16<=2.5		
Allowable stress at T	Sbo	172.41	N/mm²	
Allowable stress at 20 °C	Sba	172.41	N/mm²	
Number of bolts	n	44	-	
Nominal bolt diameter	a	25.40	mm	
Root area per bolt	Ak	355.50	mm²	
Optional: only required for flange pairs with different design conditions (example: two flanges around a tubesheet)				
Min. Am high pressure side	Am'	0.00	mm²	(Optional: default=0)
Operating load high press. side	Wo'	0	N	(optional, default = 0)
Gasket material		Camprofile SS with graphite layer (high press)		
Gasket width	N	13	mm	
Basic seating width	bo	6.5	mm	
Effective width	b	6.42	mm	
Minimum seating pressure	y	68.95	N/mm²	
Gasket compression factor	m	2	-	
Gasket partition factor: default = 1.0	Gpf	1	(No. tubesheet partition lanes:1=>Gpf=1.3; 2=>Gpf=1.6)	

User	P3 Engineering, AvdV		Job	P-001	
Client	P3 Engineering		Item	Flange-1	
Location	The Netherlands		Part	Flange 1	
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ASME VIII div.1 [Issue 2017] Appendix 2 Bolted flange connections				
Calculated values				
B = Bn + 2 Ca	B	980	mm	
g0 = g0n - Ca	g0	13	mm	
g1 = g1n - Ca	g1	24	mm	
t = tn - Caf	t	96	mm	
Bs = pi C/n	Bs	78.40	mm	(Bsmin <= Bs <= Bsmax)
Minimum bolt spacing	Bsmin	58.00	mm	(center-to-center)
Bsmax = 2a + 6t / (m+0.5)	Bsmax	281.20	mm	
Max. number of bolts = pi C / Bsmin	nbmax	59.47	-	(to fit bolt circle)
R = (C - B)/2 - g1	R	35.00	mm	(R >= Rmin)
Minimum distance bolt circle to hub	Rmin	35.00	mm	
E = (A - C)/2	E	27.00	mm	(E >= Emin)
Min. distance bolt circle to flange O.D.	Emin	27.00	mm	
Flange loadings				
H = pi/4 G² P	H	2623290	N	
Hp = 2 b pi G m P Gpf	Hp	259784	N	
Wm1 = H + Hp	Wm1	2883074	N	
Wm2 = pi b G y Gpf	Wm2	1444474	N	
Am1 = Wm1 / Sbo	Am1	16721.83	mm²	
Am2 = Wm2 / Sba	Am2	8377.95	mm²	
Am = Max(Am1,Am2, Am')	Am	16721.83	mm²	
Ab = n Ak	Ab	15642.00	mm²	(Ab >= Am)
Wo = Max(Wm1,Wo')	Wo	2883074	N	
Wa = (Am+Ab) Sba/2	Wa	2789985	N	
Nmin = Ab Sba / (2 y pi G Gpf)	Nmin	6.00	mm	(N >= Nmin)
Integral hub flange				
HD = pi/4 B² P	HD	2338319	N	
HG = Wo - H	HG	259784	N	
HT = H - HD	HT	284971	N	
hD = R + 0.5 g1	hD	47.00	mm	
hG = (C-G)/2	hG	30.00	mm	
hT = (R+g1+hG)/2	hT	44.50	mm	
MD = hD HD	MD	109900985	Nmm	
MG = hG HG	MG	7793521	Nmm	
MT = hT HT	MT	12681211	Nmm	
Mo' = MD + MG + MT	Mo'	130375717	Nmm	
Ma' = Wa hG	Ma'	83699557	Nmm	
Bsc = Max(sqr(Bs / (2a + t)), 1.0)	Bsc	1.0000	-	(Bsc >= 1)
K = A/B	K	1.1755	-	
h0 = sqr(B g0)	h0	112.8716	mm	
h/h0	h/h0	0.2835	-	
Fig. 2-7.1	T	1.8487	-	
Fig. 2-7.1	Z	6.2380	-	
Fig. 2-7.1	Y	12.0870	-	
Fig. 2-7.1	U	13.2824	-	
Fig. 2-7.2	F	0.8751	-	
Fig. 2-7.3	V	0.3262	-	
Fig. 2-7.6	f	1.8380	-	
e = F/h0	e	7.75E-03	1/mm	
g1/g0	g1/g0	1.8462	-	
d = U/V h0 g0²	d	776636.1596	mm³	

User	P3 Engineering, AvdV			Job	P-001	
Client	P3 Engineering			Item	Flange-1	
Location	The Netherlands			Part	Flange 1	
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ASME VIII div.1 [Issue 2017] Appendix 2 Bolted flange connections				
L = (t e + 1)/T + t³/d	L	2.0827	-	
Stresses under operating condition				
Allowable stress	**Sfo**	137.93	**N/mm²**	
Allowable stress hub	**1.5 Sfo**	206.90	**N/mm²**	
Mo = Mo' Bsc	**Mo**	130375717	**Nmm**	
SHo = f Mo / (L g1² B)	**SHo**	203.83	**N/mm²**	(SHo <= 1.5 Sfo)
SRo = (1.33 t e + 1) Mo / (L t² B)	**SRo**	13.79	**N/mm²**	(SRo <= Sfo)
STo = Y Mo/(t² B) - Z Sro	**STo**	88.45	**N/mm²**	(STo <= Sfo)
SCo = Max[(SHo+SRo)/2, (SHo+STo)/2]	**SCo**	**146.14**	**N/mm²**	(SCo <= Sfo)
Stresses under atmospheric condition				
Allowable stress	**Sfa**	137.93	**N/mm²**	
Allowable stress hub	**1.5 Sfa**	206.90	**N/mm²**	
Ma = Ma' Bsc	**Ma**	83699557	**Nmm**	
SHa = f Ma / (L g1² B)	**SHa**	130.86	**N/mm²**	(SHa <= 1.5 Sfa)
SRa = (1.33 t e + 1) Ma / (L t² B)	**SRa**	8.85	**N/mm²**	(SRa <= Sfa)
STa = Y Ma/(t² B) - Z Sra	**STa**	56.78	**N/mm²**	(STa <= Sfa)
SCa = Max[(SHa+SRa)/2, (SHa+STa)/2]	**SCa**	93.82	**N/mm²**	(SCa <= Sfa)
J = 52.14 Mo' V / (L E g0² h0 0.3)	**J**	0.94	-	(J <= 1)

User	**P3 Engineering, AvdV**		**Job**	**P-001**		
Client	**P3 Engineering**		**Item**	**Flange-1**		
Location	**The Netherlands**		**Part**	**Flange 1**		
Prog. rev.	**VES [19]**		**Revision**	**A.0**	**18/Jun/19**	
Date	**18/Jun/19 11:29:02**	**Signed**	**Code**	**App 2**	**Page**	**3**