PRESSURE VESSEL CALCULATION

Doc.No.	NDC-CP21-24561-01
Rev.	00
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N° de Commande Alfa Laval / Alfa Laval Order No. **CP2021-24561**	Nom du Client / Customer Name **DESMET**
N° de Série Alfa Laval / Alfa Laval Serial No. **CP40-2823**	N° de Commande Client / Customer Ref No. **-**
Type d'Equipement Alfa Laval / Alfa Laval Equipment type **CPX40 - V - 200Plates - SA 240 type 316L**	Repère Client / Customer Tag No. **681B1.1 / 19-X-753A**

PROJECT - WORPAR
ITEM NBR - 681B1.1
eB NBR - 100207388.000
DATE - 01.09.2021

Reference World Energy tag number on this sheet.

Note that design pressures exceed specification

05								
04								
03								
02								
01								
00	19/07/21	N.ROSSIGNOL	NRL	T.SINTHOMEZ	TSZ	S.BECHU		First issue
Indice *Review*	**Date *Date***	**Nom *Author***		**Verifié *Checked***		**Approuvé *Approved***		**Objet de la revision *Review subject***

Pressure vessel calculation for COMPABLOC Heat Exchangers

I - GENERAL DATA

Pressure vessel code	ASME Boiler and Pressure Vessel Code, section VIII, Division 1 in SI units (1)	
Code edition	Edition 2019	
Calculation form version	ASME VIII-1.CP.19-0	
Standard version	ASME	
Heat Exchanger type	WHE_type = "CP40-V-200Plates"	
	Side A	Side B
Design pressure	$\boxed{p_{.design} = 1.04 \cdot MPa / FV}$	$\boxed{p_{.design.} = 1.04 \cdot MPa / FV}$
	$\boxed{p_{.design} = 10.4 \cdot bar / FV}$	$\boxed{p_{.design.} = 10.4 \cdot bar / FV}$
	$p_{.design} = 151 \cdot psi / FV$	$p_{.design.} = 151 \cdot psi / FV$
Test pressure	p..test = See Appendix 2	p..test.= See Appendix 2
Max. design temperature	$\boxed{T_{maxA} = 120 \cdot °C}$ $T_{maxA} = 248 \cdot °F$	$\boxed{T_{maxB} = 120 \cdot °C}$ $T_{maxB} = 248 \cdot °F$
Min. design temperature	$\boxed{T_{min} = 0 \cdot °C}$ $T_{min} = 32 \cdot °F$	$\boxed{T_{min.} = 0 \cdot °C}$ $T_{min.} = 32 \cdot °F$

General drawing: Dwg_n = "DWG-CP21-24561-01"

Material specification: ASME section II, Edition 2019 (SI units)

Allowable stress: ASME section II, Part D, Edition 2019 (SI units)

(1) SI units are used for calculation. US customary units are converted from SI units for information.

II - Considered loads (UG 22)

LOADS	Yes/No	REMARKS
Internal design pressure	Yes	
External design pressure	Yes	Appendix 3 (Sheet 33 to 41)
Weight of the vessel	No	.
Weight of the contents	No	
Additional pressure due to static of liquids	Yes	
Static reaction from weight of attached equipments	No	
Internal attachments	No	
Vessel support attachment	No	.
Cyclic and dynamic reactions due to pressure	No	
Cyclic and dynamic reactions due to thermal variations	No	
Cyclic and dynamic reactions from equipment	No	
Mechanical loadings	No	
Wind reaction	No	.
Snow reaction	No	.
Seismic reaction	No	.
Impact reactions	No	
Differential thermal expansion	No	
Temperature gradients	No	
Deflagration loads	No	.
Nozzles loads calculations	Yes	FEA-CP21-24561-01
Test pressure checking with 90% yield stress	No	.

III - CALCULATION PARTS SIDE A :

1- **BOLT LOADS CALCULATION**. (Appendix 2)

Design pressure : $p_{.design} = 1.04 \cdot MPa$ $\quad p_{.design} = 10.4 \cdot bar$

Additional pressure due to static height of liquid : $p_{stat} = 0.02 \cdot MPa$ $\quad p_{stat} = 0.2 \cdot bar$

Calculation pressure according to UG-98 (b) : $P = p_{.design} + p_{stat}$ $\quad \boxed{P_A = 1.06 \cdot MPa}$ $\quad \boxed{P_A = 10.6 \cdot bar}$ $\quad P_A = 153.7 \cdot psi$

Gasket material : Mat_{gkt} = "Graphite PSM"

Gasket factor : $m_{gkt} = 2$

Min design seating stress : $y_{gkt} = 26 \cdot MPa$

(Characteristic gasket values are based on the manufacturer's seal recommendation for the Compabloc design and ASME table 2.5.1)

Gasket width : $N_{gkt} = 14 \cdot mm$

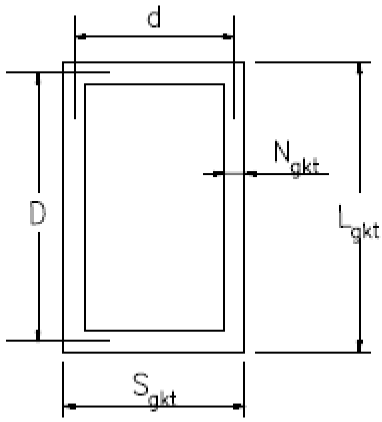
b_0 = Basic gasket seating width : $b_0 := \dfrac{N_{gkt}}{2} = 7 \cdot mm$ (From table 2.5.2 fig.(a))

b = effective gasket seating width :

$$b := \begin{cases} b_0 & \text{if } b_0 \le 6 \cdot mm \\ C_b \cdot \sqrt{b_0 \cdot mm} & \text{otherwise} \end{cases} \qquad b = 6.61438 \cdot mm$$

With C_b conversion factor = 2.5 for SI calculation.

<u>Sketch of gasket</u> :



Load reaction on line of gasket (fig 2.5.2)

Short gasket span : $\boxed{S_{gkt} = 399.7 \cdot mm}$

$S_{gkt} = 15.7362 \cdot in$

Long gasket span : $\boxed{L_{gkt} = 1190.7 \cdot mm}$

$L_{gkt} = 46.878 \cdot in$

Short gasket span on line : $\boxed{d := S_{gkt} - (2 \cdot b) = 386.5 \cdot mm}$ $\quad d = 15.2 \cdot in$

Long gasket span on line : $\boxed{D := L_{gkt} - (2.b) = 1177.5 \cdot mm}$ $\quad D = 46.4 \cdot in$

Gasket perimeter : $\boxed{P_g := (d + D) \cdot 2 = 3127.9 \cdot mm}$ $\quad P_g = 123.1 \cdot in$

1.1 OPERATING LOAD

(Appendix 2 § 2.5, equation 1 adapted for rectangular gaskets)

$W_{m1} := \left(D \cdot d \cdot P_A\right) + \left(2 \cdot b \cdot P_g \cdot m_{gkt} \cdot P_A\right)$ $\qquad W_{m1} = 570083.7 \cdot N$

1.2 INITIAL LOAD

(Appendix 2 § 2.5, equation 2 adapted for rectangular gaskets)

$W_{m2} := P_g \cdot b \cdot y_{gkt}$ $\qquad W_{m2} = 537914.4 \cdot N$

2- <u>REQUIRED BOLT AREA</u> (Appendix 2 § 2.5(d))

Bolt material : Mat_{trod} = "SA 193 gr B7"

Allowable bolt stress :

At design temperature, $Sb_{trod} = 172 \cdot MPa$ $\qquad A_{m1} := \dfrac{W_{m1}}{Sb_{trod}} = 3314.4 \cdot mm^2$

At room temperature, $Sa_{trod} = 172 \cdot MPa$ $\qquad A_{m2} := \dfrac{W_{m2}}{Sa_{trod}} = 3127.4 \cdot mm^2$

Required bolt area :

$A_m := \max\left(A_{m1}, A_{m2}\right)$ $\qquad A_m = 3314.4 \cdot mm^2$

Actual bolt area (Ab) :

Diameter of threaded rods, $d_{trod} = 24 \cdot mm$

Root area, $S_{trod} = 317 \cdot mm^2$

Number of threaded rods, $N_{trod} = 32$

Bolting area provided, $Ab := N_{trod} \cdot S_{trod}$ $\qquad Ab = 10144 \cdot mm^2$

To have adequate number and size, Ab shall be greater than 1 * A_m

(With corrective factor according mechanical strength test RGE-CP90-322)

$1 \cdot A_m = 3314.4 \cdot mm^2$ *(1=true, 0=false)* $\qquad Ab > 1 \cdot A_m = 1$

3- STUD BOLT THREAD LENGTH According to UG-43 (g)

Nominal diameter of bolt : $\boxed{d_{trod} = 24 \cdot mm}$ $d_{trod} = 0.9449 \cdot in$

Allowable stress of bolt at design temperature : $Sb_{trod} = 172 \cdot MPa$

Allowable stress of girder at design temperature : $Sb_{girder} = 118 \cdot MPa$

$$L_{threadmin} := \min\left(1.5 \cdot d_{trod}, \max\left(0.75 \cdot d_{trod} \cdot \frac{Sb_{trod}}{Sb_{girder}}, d_{trod}\right)\right)$$

$\boxed{L_{threadmin} = 26.2 \cdot mm}$

$L_{threadmin} = 1.033 \cdot in$

Thread length : $\boxed{L_{thread} = 36 \cdot mm}$ $L_{thread} = 1.417 \cdot in$

Validated criterion : $L_{threadmin} < L_{thread} = 1$

(*1=true, 0=false*)

4- PANEL CALCULATION

UNSTAYED FLAT HEAD CALCULATION WITHOUT OPENING: (UG 34)

Flange design bolt load calculation : (Appendix 2, § 2.5)

Under operating conditions : (equation 4)

$$W1 := W_{m1}$$

$\boxed{W1 = 570083.7 \cdot N}$

Under gasket seating conditions : (equation 5)

$$W2 := \frac{(A_m + Ab)Sa_{trod}}{2}$$

$\boxed{W2 = 1157425.9 \cdot N}$

Bolt load used on each thread rod:

$$F_{b.} := \frac{\max(W1, W2)}{N_{trod}} = 36170 \cdot N$$

$\boxed{F_{b.} = 36170 \cdot N}$

IV - CALCULATION PARTS SIDE B :

1- <u>BOLT LOADS CALCULATION</u>. (Appendix 2)

Design pressure : $p_{.design} = 1.04 \cdot MPa$ $\quad p_{.design} = 10.4 \cdot bar$

Additional pressure due to static height of liquid : $p_{stat} = 0.02 \cdot MPa$ $\quad p_{stat} = 0.2 \cdot bar$

Calculation pressure according to UG-98 (b) : $P = p_{.design} + p_{stat}$ $\quad \boxed{P_B = 1.06 \cdot MPa}$ $\quad \boxed{P_B = 10.6 \cdot bar}$ $\quad P_B = 153.7 \cdot psi$

Gasket material : $Mat_{gkt} = \text{"Graphite PSM"}$

Gasket factor : $m_{gkt} = 2$

Min design seating stress : $y_{gkt} = 26 \cdot MPa$

(Characteristic gasket values are based on the manufacturer's seal recommendation for the Compabloc design and ASME table 2.5.1)

Gasket width : $N_{gkt} = 14 \cdot mm$

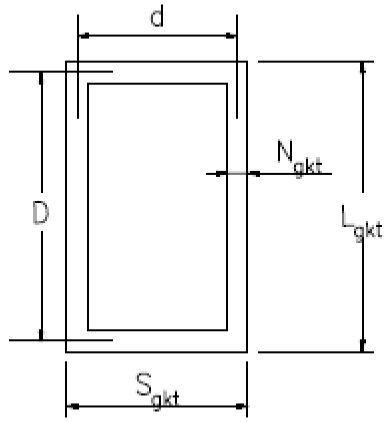
b_0 = Basic gasket seating width : $b_0 := \dfrac{N_{gkt}}{2} = 7 \cdot mm$ (From table 2.5.2 fig.(a))

b = effective gasket seating width :

$$b := \begin{cases} b_0 & \text{if } b_0 \le 6 \cdot mm \\ C_b \cdot \sqrt{b_0 \cdot mm} & \text{otherwise} \end{cases} \qquad b = 6.61438 \cdot mm$$

With C_b conversion factor = 2.5 for SI calculation.

<u>Sketch of gasket</u> :



Load reaction on line of gasket (fig 2.5.2)

Short gasket span : $\boxed{S_{gkt} = 399.7 \cdot mm}$

$S_{gkt} = 15.7362 \cdot in$

Long gasket span : $\boxed{L_{gkt} = 1190.7 \cdot mm}$

$L_{gkt} = 46.878 \cdot in$

Short gasket span on line : $\boxed{d := S_{gkt} - (2 \cdot b) = 386.5 \cdot mm}$ $\quad d = 15.2 \cdot in$

Long gasket span on line : $\boxed{D := L_{gkt} - (2.b) = 1177.5 \cdot mm}$ $\quad D = 46.4 \cdot in$

Gasket perimeter : $\boxed{P_g := (d + D) \cdot 2 = 3127.9 \cdot mm}$ $\quad P_g = 123.1 \cdot in$

1.1 OPERATING LOAD

(Appendix 2 § 2.5, equation 1 adapted for rectangular gaskets)

$W_{m1} := (D \cdot d \cdot P_B) + (2 \cdot b \cdot P_g \cdot m_{gkt} \cdot P_B)$ $\qquad W_{m1} = 570083.7 \cdot N$

1.2 INITIAL LOAD

(Appendix 2 § 2.5, equation 2 adapted for rectangular gaskets)

$W_{m2} := P_g \cdot b \cdot y_{gkt}$ $\qquad W_{m2} = 537914.4 \cdot N$

2- <u>REQUIRED BOLT AREA</u> (Appendix 2 § 2.5(d))

Bolt material : $Mat_{trod} =$ "SA 193 gr B7"

Allowable bolt stress :

At design temperature, $Sb_{trod} = 172 \cdot MPa$ $\qquad A_{m1} := \dfrac{W_{m1}}{Sb_{trod}} = 3314.4403 \cdot mm^2$

At room temperature, $Sa_{trod} = 172 \cdot MPa$ $\qquad A_{m2} := \dfrac{W_{m2}}{Sa_{trod}} = 3127.4092 \cdot mm^2$

Required bolt area :

$A_m := \max(A_{m1}, A_{m2})$ $\qquad A_m = 3314.4 \cdot mm^2$

Actual bolt area (Ab) :

Diameter of threaded rods, $d_{trod} = 24 \cdot mm$

Root area, $S_{trod} = 317 \cdot mm^2$

Number of threaded rods, $N_{trod} = 32$

Bolting area provided, $Ab := N_{trod} \cdot S_{trod}$ $\qquad Ab = 10144 \cdot mm^2$

To have adequate number and size, Ab shall be greater than 1 * A_m

(With corrective factor according mechanical strength test RGE-CP90-322)

$1 \cdot A_m = 3314.4 \cdot mm^2$ *(1=true, 0=false)* $\qquad Ab > 1 \cdot A_m = 1$

3- STUD BOLT THREAD LENGTH

According to UG-43 (g)

Nominal diameter of bolt : $\boxed{d_{trod} = 24 \cdot mm}$ $d_{trod} = 0.945 \cdot in$

Allowable stress of bolt at design temperature : $Sb_{trod} = 172 \cdot MPa$

Allowable stress of girder at design temperature : $Sb_{girder} = 118 \cdot MPa$

$$L_{threadmin} := \min\left(1.5 \cdot d_{trod}, \max\left(0.75 \cdot d_{trod} \cdot \frac{Sb_{trod}}{Sb_{girder}}, d_{trod}\right)\right)$$

$\boxed{L_{threadmin} = 26.2 \cdot mm}$

$L_{threadmin} = 1.033 \cdot in$

Thread length : $\boxed{L_{thread} = 36 \cdot mm}$ $L_{thread} = 1.417 \cdot in$

Validated criterion : $L_{threadmin} < L_{thread} = 1$

(*1=true, 0=false*)

4- PANEL CALCULATION

UNSTAYED FLAT HEAD CALCULATION WITHOUT OPENING: (UG 34 equation 5)

Flange design bolt load calculation : (Appendix 2, § 2.5)

Under operating conditions : (equation 4)

$W1b := W_{m1}$ $\boxed{W1b = 570083.7 \cdot N}$

Under gasket seating conditions : (equation 5)

$$W2b := \frac{(A_m + Ab)Sa_{trod}}{2}$$ $\boxed{W2b = 1157425.9 \cdot N}$

Bolt load used on each thread rod:

$$F_b := \frac{\max(W1b, W2b)}{N_{trod}} = 36170 \cdot N$$ $\boxed{F_b = 36170 \cdot N}$

V - PANELS AND HEADS CALCULATION

Alfalaval Compabloc ® has a design that differ from these covered in ASME VIII div.1 and the following calculation way is defined :

-Use of the chapter UG-16 for these designs with the alternative of using Mandatory Appendix 46

-Use of the chapter 4.12 of Div.2 as allowed by Mandatory Appendix 46, and if the design by rule is not possible, move to next option.

-Use of part 5 of Div.2 by using a FEA calculation.

Indeed, the design of the Compabloc (panels and heads) corresponds directly to the chapter 4.12 of Div.2: "Design rules for non circular vessels".
But in the "scope" in the paragraph 4.12.1.2 and in the "General design requirement" in the paragraph 4.12.2.8, for vessel other than described in 4.12 and for "Bolted full side or end plates…" "Part 5 shall be used".
Moreover, for openings in kind of vessel, the Part 5 is referenced again in the paragraph 4.12.2.9 (b) : "rules for specific designs are not provided and the design shall be in accordance with Part 5".

Specific FEA calculation : $FEA_{calculation}$ = "FEA-CP21-24561-01"

This document shows that the design of those parts fulfils the requirements of the division 1 of the section VIII ASME Code with the following parameters:

1- PANELS DATA

Side A

Calculation pressure side A	:	$\boxed{P_A = 1.06 \cdot MPa}$	$\boxed{P_A = 10.6 \cdot bar}$	$P_A = 154 \cdot psi$
Design Temperature side A	:	$\boxed{T_{maxA} = 120 \cdot °C}$		$T_{maxA} = 248 \cdot °F$
Panel Aa minimal thickness	:	$\boxed{Th_{Aa} = 64 \cdot mm}$		$Th_{Aa} = 2.5197 \cdot in$
Panel Ab minimal thickness	:	$\boxed{Th_{Ab} = 54 \cdot mm}$		$Th_{Ab} = 2.126 \cdot in$

Side B

Calculation pressure side B	:	$\boxed{P_B = 1.06 \cdot MPa}$	$\boxed{P_B = 10.6 \cdot bar}$	$P_B = 154 \cdot psi$
Design Temperature side B	:	$\boxed{T_{maxB} = 120 \cdot °C}$		$T_{maxB} = 248 \cdot °F$
Panel Ba minimal thickness	:	$\boxed{Th_{Ba} = 64 \cdot mm}$		$Th_{Ba} = 2.5197 \cdot in$
Panel Bb minimal thickness	:	$\boxed{Th_{Bb} = 54 \cdot mm}$		$Th_{Bb} = 2.126 \cdot in$

2- SQUARE HEAD CALCULATION

UNSTAYED STEEL SQUARE HEAD, INSERTED AT THE EXTREMITY OF LATERAL PANELS.

Square head material : Mat_{head} = "SA 516 gr 60N"

Allowable head stress (Section II part D) :

At design temperature, $S_{head} = 118 \cdot MPa$

As described above, the head calculation is done in :

Specific FEA calculation : $FEA_{calculation}$ = "FEA-CP21-24561-01"

VI - LOCKINGS CALCULATION

1- STRENGTH OF LOCKING HEAD / PANEL

1.1 Data

S_{shear} = Load carried by clamp / Strength section of the square head tongues

Calculation pressure Max. : $P_{max} = 1.06 \cdot MPa$ $P_{max} = 10.6 \cdot bar$

Load carried by clamp : $W_{head} := d_{head}^{2} \cdot P_{max} = 254506 \cdot N$

Load per link : $P_{hl} := \dfrac{W_{head}}{4} = 63626 \cdot N$

Mating height in compressed gasket condition : $\boxed{h_{b\_h} = 4 \cdot mm}$ $h_{b\_h} = 0.157 \cdot in$

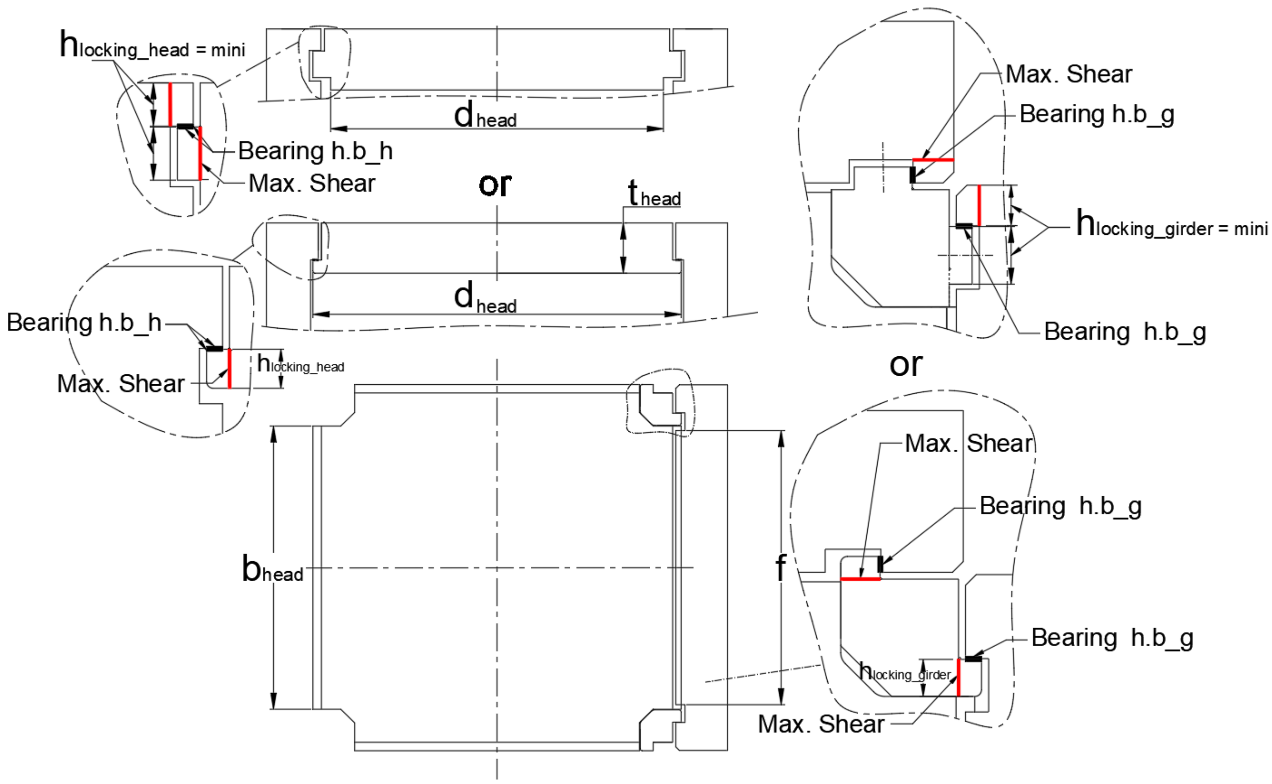
Panel locking length size : $\boxed{f = 350.5 \cdot mm}$ $f = 13.799 \cdot in$

Head locking length value : $\boxed{b_{head} = 370 \cdot mm}$ $b_{head} = 14.567 \cdot in$

Head locking height value : $\boxed{h_{locking\_head} = 15 \cdot mm}$ $h_{locking\_head} = 0.591 \cdot in$

Head square size : $\boxed{d_{head} = 490 \cdot mm}$ $d_{head} = 19.291 \cdot in$

Sketch of Locking :



1.2 CRITERIA DESIGN STRESS UG23 (f) and (g)

Maximum shear stress :

Strength section of the square head tongues :

$$S_{ht} := 4 \cdot h_{locking\_head} \cdot \min(f, b_{head}) \qquad S_{ht} = 21030 \cdot mm^2$$

Locking Shear stress :

$$S_{shear} := \frac{W_{head}}{S_{ht}} = 12.1 \cdot MPa$$

$$S_{locking\_head} := \min(S_{panel}, S_{head}) \qquad S_{locking\_head} = 118 \cdot MPa$$

S_{shear} shall be lower than 0.8x $S_{locking\_head}$

$0.8 \cdot S_{locking\_head} = 94.4 \cdot MPa$ $\qquad S_{shear} < 0.8 \cdot S_{locking\_head} = 1$

(*1=true, 0=false*)

Maximum bearing stress :

$$P_{b\_h} := \frac{P_{hl}}{\min(f, b_{head}) \cdot h_{b\_h}} \qquad \boxed{P_{b\_h} = 45.4 \cdot MPa}$$

P_{b_h} shall be lower than 1.6x $S_{locking\_head}$

$1.6 \cdot S_{locking\_head} = 188.8 \cdot MPa$ $\qquad P_{b\_h} < 1.6 \cdot S_{locking\_head} = 1$

(*1=true, 0=false*)

2- STRENGTH OF LOCKING GIRDER / PANEL

2.1 Data

S_{shear_girder} = Load (per unit length) carried by clamp / Strength length of the girder tongues

Calculation pressure Max. : $P_{max} = 1.06 \cdot MPa$

Calculation pressure Min. : $P_{min} = 1.06 \cdot MPa$

Girder material : Mat_{girder} = "SA 516 gr 60N"

Allowable girder stress (Section II part D) :

At max. design temperature, $Sb_{girder} = 118 \cdot MPa$

Square size of girder : $\boxed{d_{col} = 59.5 \cdot mm}$ $d_{col} = 2.343 \cdot in$

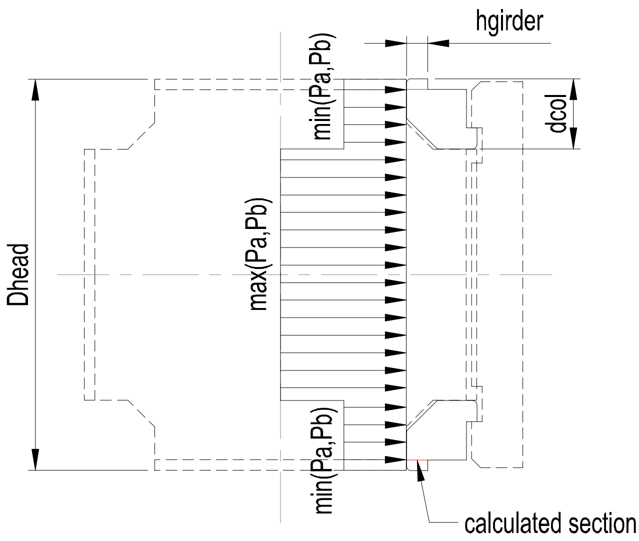
Mating height in compressed gasket condition : $\boxed{h_{b\_g} = 4 \cdot mm}$ $h_{b\_g} = 0.157 \cdot in$

Sketch of girder:

with : Girder locking length value : $\boxed{h_{locking\_girder} = 15 \cdot mm}$ $h_{locking\_girder} = 0.591 \cdot in$

$\boxed{D_{head} = 490 \cdot mm}$ $D_{head} = 19.291 \cdot in$

$\boxed{d_{col} = 59.5 \cdot mm}$ $d_{col} = 2.343 \cdot in$



2.2 CRITERIA DESIGN STRESS UG23 (f) and (g)

Load (per unit length) carried by clamp :

$$W_{girder} := (D_{head} - 2 \cdot d_{col}) \cdot \max(P_A, P_B) + 2 \cdot d_{col} \cdot \min(P_A, P_B) \qquad W_{girder} = 519.4 \cdot \frac{N}{mm}$$

Strength length of the girder tongues : $S_{gt} := 2 \cdot h_{locking\_girder}$ $\boxed{S_{gt} = 30 \cdot mm}$ $S_{gt} = 1.181 \cdot in$

$$S_{locking\_girder} := \min(S_{panel}, Sb_{girder})$$

<u>Maximum shear stress :</u>

$$S_{shear\_girder} := \frac{W_{girder}}{S_{gt}} = 17.3 \cdot MPa$$

S_{shear_girder} shall be lower than 0.8x $S_{locking\_girder}$

$S_{locking\_girder} = 118 \cdot MPa$ $\qquad$ $0.8 \cdot S_{locking\_girder} = 94.4 \cdot MPa$

$$S_{shear\_girder} < 0.8 \cdot S_{locking\_girder} = 1$$

(*1=true, 0=false*)

<u>Maximum bearing stress :</u>

$$P_{b\_g} := \frac{W_{girder}}{2 \cdot h_{b\_g}} \qquad \boxed{P_{b\_g} = 64.9 \cdot MPa}$$

P_{b_g} shall be lower than 1.6x $S_{locking\_girder}$ $\qquad$ $1.6 \cdot S_{locking\_girder} = 188.8 \cdot MPa$

$$P_{b\_g} < 1.6 \cdot S_{locking\_girder} = 1$$

(*1=true, 0=false*)

3- MAXIMUM BOLT SPACING: (B_{smax}) according to App 2-5 (d) equation (3)

Only applicable for Lethal service

with :	Nominal bolt diameter	:	$d_{trod} = 24 \cdot mm$	$d_{trod} = 0.9449 \cdot in$
	Panel thickness (Machined at min value)	:	$Th_{Aa} = 64 \cdot mm$	$Th_{Ab} = 54 \cdot mm$
			$Th_{Ba} = 64 \cdot mm$	$Th_{Bb} = 54 \cdot mm$
	Gasket factor side A & B	:	$m_{gkt.A} = 2$	$m_{gkt.B} = 2$
	Head bolt spacing	:	$B_{s.head} = 110 \cdot mm$	$B_{s.head} = 4.3307 \cdot in$
	Column bolt spacing	:	$B_{s.col} = 98.32 \cdot mm$	$B_{s.col} = 3.8709 \cdot in$
	Column/head bolt spacing :		$B_{s.c.h} = 116.8 \cdot mm$	$B_{s.c.h} = 4.6 \cdot in$

$$B_{smax} := 2 \cdot d_{trod} + \frac{6 \cdot \min\left(Th_{Aa}, Th_{Ab}, Th_{Ba}, Th_{Bb}\right)}{\max\left(m_{gkt.A}, m_{gkt.B}\right) + 0.5} \qquad B_{smax} = 177.6 \cdot mm$$

The maximum head bolt spacing $B_{s.head}$, the maximum column bolt spacing $B_{s.col}$ and the column/head bolt spacing $B_{s.c.h}$ shall not exceed B_{smax}

(*1=true, 0=false*)

$B_{smax} > B_{s.head} = 1$

$B_{smax} > B_{s.col} = 1$

$B_{smax} > B_{s.c.h} = 1$

VII - PIPE CHECKING :

Calculation pipe checking is done on the paragraph "Analytical calculation for nozzle" :

See Specific FEA calculation : $FEA_{calculation}$ = "FEA-CP21-24561-01"

VIII - WELDING NECK AND BLIND FLANGE CHECKING :

The Welding neck flange or Long Welding neck flange must be define according to the Pressure-Temperature ratings for groups materials :

- ASME B16.5 Table 2 Ed.2013 (Nominal diameter <= DN600 (24"))
- ASME B16.47a Table 2 Ed.2017 (Nominal diameter > DN600 (24"))

Check of flange design

	Side A:		Side B :	
Design temperature :	$T_{maxA} = 120 \cdot °C$		$T_{maxB} = 120 \cdot °C$	
Calculation pressure :	$P_A = 1.06 \cdot MPa$	$P_A = 10.6 \cdot bar$	$P_B = 1.06 \cdot MPa$	$P_B = 10.6 \cdot bar$

PRESSURE TEMPERATURE RATING

ASME B16.5 Ed.2013

Table 2-1.1
PressureTemperature Ratings for Group 1.1 Materials
Working Pressure by Classes, **bar**

Temp.,°C	Class 150	Class 300	Class 400	Class 600
−29 to 38	19.6	51.1	68.1	102.1
50	19.2	50.1	66.8	100.2
100	17.7	46.6	62.1	93.2
150	15.8	45.1	60.1	90.2
200	13.8	43.8	58.4	87.6
250	12.1	41.9	55.9	83.9
300	10.2	39.8	53.1	79.6
325	9.3	38.7	51.6	77.4
350	8.4	37.6	50.1	75.1
375	7.4	36.4	48.5	72.7
400	6.5	34.7	46.3	69.4
425	5.5	28.8	38.4	57.5
450	4.6	23	30.7	46
475	3.7	17.4	23.2	34.9
500	2.8	11.8	15.7	23.5
538	1.4	5.9	7.9	11.8
A350 Gr. LF2				

Table 2-2.3
PressureTemperature Ratings for Group 2.3
Working Pressure by Classes, **bar**

Temp.,°C	Class 150	Class 300	Class 400	Class 600
−29 to 38	15.9	41.4	55.2	82.7
50	15.3	40	53.4	80
100	13.3	34.8	46.4	69.6
150	12	31.4	41.9	62.8
200	11.2	29.2	38.9	58.3
250	10.5	27.5	36.6	54.9
300	10	26.1	34.8	52.1
325	9.3	25.5	34	51
350	8.4	25.1	33.4	50.1
375	7.4	24.8	33	49.5
400	6.5	24.3	32.4	48.6
425	5.5	23.9	31.8	47.7
450	4.6	23.4	31.2	46.8
A182F316L				

VII-NOZZLE CALCULATION $Item_{opening}$ = "A1-A2-B1-B2"

A- GENERAL DATA Acc. to section VIII Div.2. Ed 2019

Connection standard (DN - PN) :	Dim_{nozzle} = "DN150 Cl150"		
Calculation pressure :	$\boxed{P = 1.06 \cdot MPa}$	$\boxed{P = 10.6 \cdot bar}$	$P = 153.7 \cdot psi$
Design temperature :	$\boxed{T_{.max} = 120 \cdot °C}$	$T_{.max} = 248 \cdot °F$	
Axial load at the flange :	$\boxed{F_A = 776 \cdot N}$	$F_A = 174 \cdot lbf$	
Moment load at the flange :	$\boxed{M_y = 750 \cdot N \cdot m}$	$M_y = 6638 \cdot lbf \cdot in$	
	$\boxed{M_z = 750 \cdot N \cdot m}$	$M_z = 6638 \cdot lbf \cdot in$	
Bending moment on the flange :	$\boxed{M_E = 1061 \cdot N \cdot m}$	$M_E := \sqrt{M_y^2 + M_z^2} = 9388 \cdot lbf \cdot in$	
Outside diameter of flange :	$\boxed{A = 280 \cdot mm}$	$A = 11.0236 \cdot in$	
Inside diameter of flange :	$\boxed{B = 155.1 \cdot mm}$	$B = 6.1063 \cdot in$	
Bolt circle diameter :	$\boxed{C = 241.3 \cdot mm}$	$C = 9.5 \cdot in$	
Thick. of hub at back of flange :	$\boxed{g_1 = 18.45 \cdot mm}$	$g_1 = 0.7264 \cdot in$	
Minimum Thick. of hub flange :	$\boxed{g_0 = 6.6 \cdot mm}$	$g_0 = 0.2598 \cdot in$	
Real hub length :	$\boxed{h_r = 52.8 \cdot mm}$	$h_r = 2.0787 \cdot in$	
Small thickness of hub (pipe) :	$\boxed{g_0 = 6.6 \cdot mm}$	$g_0 = 0.2598 \cdot in$	
Flange thickness in corroded condition :	$\boxed{t = 23.9 \cdot mm}$	$t = 0.9409 \cdot in$	

Geometrical validation :

Integral Type Flanges with a Hub and a weld (geometric criteria Figure 4.16.2)

$\text{Slope} := \dfrac{(g_1 - g_0)}{h_r}$ $\text{Slope} = 0.2244$ $\text{Slope} < \dfrac{1}{3} = 1$

If slope < 1/3: the Figure 4.16.2 (a) is used wihout more condition
If slope > 1/3: the Figure 4.16.2 (b) is used with the following condtion

Length of tube part of flange $h_2 = 10.3 \cdot \text{mm}$

Minimal half of gap for the weld $\text{gap} = 0.5 \cdot \text{mm}$

$1.5 g_0 = 9.9 \cdot \text{mm}$ $h_2 + \text{gap} \geq 1.5 g_0 = 1$

B- BOLT LOADS CALCULATION. (§ 4.16.6)

Calculation pressure : $P = 1.06 \cdot MPa$

External pressure : $P_{ext} := 0.103MPa$

Considered gasket : $Mat_{gkt} = \text{"Graphite PSM"}$

Gasket factor : $m_{gkt} = 2$

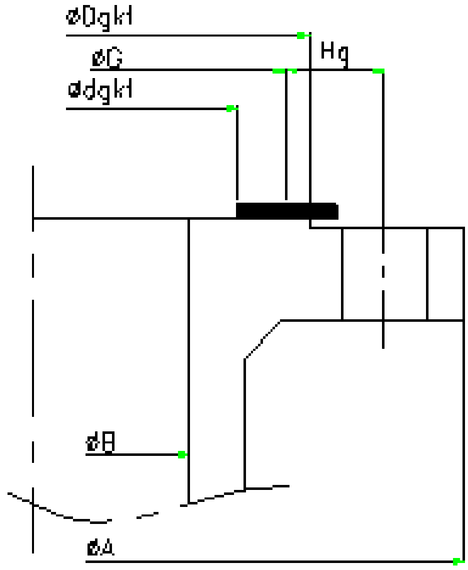
Min design seating stress : $y_{gkt} = 26 \cdot MPa$ (From table 4.16.1 or gasket manufacturers)

Sketch of gasket :

Gasket outer diameter : $D_{gkt} = 215.9 \cdot mm$

Gasket inner diameter : $d_{gkt} = 168 \cdot mm$

Gasket width : $N_{gkt} = 24 \cdot mm$



b_0 = Basic gasket seating width:

(From table 4.16.3 fig.(1a))

$$b_0 := \frac{N_{gkt}}{2} \qquad b_0 = 12 \cdot mm$$

b = effective gasket seating width :

$C_{ul} := 25.4$ conversion factor for SI calculation.

$$b := \begin{cases} b_0 & \text{if } b_0 \le 6 \cdot mm \\ 0.5 \cdot C_{ul} \cdot \sqrt{\frac{b_0}{C_{ul}} \cdot mm} & \text{otherwise} \end{cases} \qquad b = 8.7 \cdot mm$$

Gasket load reaction diameter:

$$G := D_{gkt} - (2 \cdot b) \qquad G = 198.5 \cdot mm$$

Operating load

$$W_o := 0.785 \cdot G^2 \cdot P + 2 \cdot b \cdot \pi \cdot G \cdot m_{gkt} \cdot P \qquad W_o = 55825.5 \cdot N$$

$$W_{oext} := 0.785 \cdot G^2 \cdot P_{ext} + 2 \cdot b \cdot \pi \cdot G \cdot m_{gkt} \cdot P_{ext} \qquad W_{oext} = 5424.6 \cdot N$$

Initial load

$$W_{gs} := \pi \cdot b \cdot G \cdot y_{gkt} = 31778.5224 \cdot lbf \qquad W_{gs} = 141357.9 \cdot N$$

Required bolt Area

Bolt material : $Mat_{trod} = \text{"SA 193 gr B7"}$

Allowable bolt stress :

At room temperature, $S_{bg} = 172 \cdot MPa$

At design temprature, $S_{bo} = 172 \cdot MPa$

Total required bolt area:

$$A_m := \max\left(\frac{W_o + F_A + \frac{4 \cdot M_E}{G}}{S_{bo}}, \frac{W_{gs}}{S_{bg}} \right) \qquad A_m = 821.8 \cdot mm^2$$

Actual bolt area (Ab) :

Diameter of bolts, $d_{trod} = 20 \cdot mm$

Root area, $S_{trod} = 220 \cdot mm^2$

Number of bolts, $N_{trod} = 8$

Bolting area provided, $Ab := N_{trod} \cdot S_{trod}$ $Ab = 1760 \cdot mm^2$

To have adequate number and size, Ab shall be greater than A_m

$A_m = 821.8 \cdot mm^2$ *(1=true, 0=false)* $Ab > A_m = 1$

Bolt load for gasket conditions:

$$W_g := \frac{(A_m + Ab)S_{bg}}{2} \qquad W_g = 222039 \cdot N$$

C- FLANGE DESIGN PROCEDURE. (§ 4.16.7)

C.1 Moment arms for flange loads under operating conditions, in acc. with Table 4.16.6

Radial distance from bolt-circle to point of action of Hydrostatic end force:

$$h_D := \frac{C - B - g_1}{2} \qquad h_D = 33.9 \cdot mm$$

$$h_G := \frac{C - G}{2} \qquad h_G = 21.4 \cdot mm$$

Radial distance from bolt-circle to point of action of Hydrostatic differences:

$$h_T := \frac{1}{2} \cdot \left(\frac{C - B}{2} + h_G \right) \qquad h_T = 32.3 \cdot mm$$

C.2 Forces

Total hydrostatic end force for internal pressure:

$$H := 0.785G^2 \cdot P \qquad H = 7367.7291 \cdot lbf \qquad H = 32773 \cdot N$$

Hydrostatic end force on area inside of flange for internal pressure:

$$H_D := 0.785 \cdot B^2 \cdot P \qquad H_D = 4500.0019 \cdot lbf \qquad H_D = 20017 \cdot N$$

Hydrostatic difference: $H_T := H - H_D$ $H_T = 2867.7272 \cdot lbf$ $H_T = 12756\,N$

Bolt Load at Operating Conditions: $H_G := W_o - H$ $H_G = 5182.3437 \cdot lbf$ $H_G = 23052\,N$

Total hydrostatic end force for external pressure:

$$H_{ext} := 0.785G^2 \cdot P_{ext} \qquad H_{ext} = 715.9208 \cdot lbf \qquad H_{ext} = 3185 \cdot N$$

Hydrostatic end force on area inside of flange for external pressure:

$$H_{Dext} := 0.785 \cdot B^2 \cdot P_{ext} \qquad H_{Dext} = 437.2643 \cdot lbf \qquad H_{Dext} = 1945 \cdot N$$

Hydrostatic difference: $H_{Text} := H_{ext} - H_{Dext}$ $H_{Text} = 278.6565 \cdot lbf$ $H_{Text} = 1240\,N$

Bolt Load at Operating Conditions: $H_{Gext} := W_{oext} - H_{ext}$ $H_{Gext} = 503.5674 \cdot lbf$ $H_{Gext} = 2240\,N$

C.3 Calculation of Flange Stress Factors, Table 4.16.4 and 4.16.5.

$$K := \frac{A}{B} \qquad K = 1.8053$$

$$Y := \frac{1}{K-1}\cdot\left(0.66845 + 5.71690\cdot\frac{K^2\cdot\log(K)}{K^2-1}\right) \qquad Y = 3.4576$$

$$T := \frac{K^2\cdot(1 + 8.55246\cdot\log(K)) - 1}{(1.04720 + 1.9448\cdot K^2)\cdot(K-1)} \qquad T = 1.5822$$

$$U := \frac{K^2\cdot(1 + 8.55246\cdot\log(K)) - 1}{1.36136\cdot(K^2-1)\cdot(K-1)} \qquad U = 3.7995$$

$$Z := \frac{K^2+1}{K^2-1} \qquad Z = 1.8853$$

$$h_o := \sqrt{B\cdot g_0} \qquad h_o = 32.0\cdot mm$$

Numerical decrease of hub length "h" for the F factor calculation in conservative way:

$$h := \begin{cases} h_r & \text{if } \frac{h_r}{h_o} \le 2 \\ 2\cdot h_o & \text{otherwise} \end{cases} \qquad h_r = 52.8\cdot mm \qquad h = 52.8\cdot mm$$

$$X_h := \frac{h}{h_o} \qquad X_h = 1.6503$$

$$X_g := \frac{g_1}{g_0} \qquad X_g = 2.7955$$

$$\begin{aligned} F := {} & 0.897697 - 0.297012\cdot\ln(X_g) + 9.5257\cdot10^{-3}\cdot\ln(X_h) \ldots \\ & + 0.123586\cdot(\ln(X_g))^2 + 0.0358580\cdot(\ln(X_h))^2 - 0.194422\cdot\ln(X_g)\cdot\ln(X_h) \ldots \\ & + -0.0181259\cdot(\ln(X_g))^3 + 0.0129360\cdot(\ln(X_h)^3) \ldots \\ & + -0.0377693\cdot(\ln(X_g))\cdot(\ln(X_h))^2 + 0.0273791\cdot(\ln(X_g))^2\cdot\ln(X_h) \end{aligned}$$

$$F = 0.6233$$

$$V := \begin{cases} \left(0.500244 + \frac{0.227914}{X_g} - 1.87071 \cdot X_h - \frac{0.344410}{X_g^2} + 2.49189 \cdot X_h^2 \ldots + 0.873446 \cdot \frac{X_h}{X_g} + \frac{0.189953}{X_g^3} - 1.06082 \cdot X_h^3 - 1.49970 \cdot \frac{X_h^2}{X_g} \ldots + 0.719413 \cdot \frac{X_h}{X_g^2}\right) & \text{if } 0.1 \le X_h \le 0.5 \\ \left(0.0144868 - \frac{0.135977}{X_g} - \frac{0.0461919}{X_h} + \frac{0.560718}{X_g^2} + \frac{0.0529829}{X_h^2} \ldots + \frac{0.244313}{X_g \cdot X_h} + \frac{0.113929}{X_g^3} - \frac{0.00928265}{X_h^3} - \frac{0.0266293}{X_g \cdot X_h^2} - \frac{0.217008}{X_g^2 \cdot X_h}\right) & \text{if } 0.5 < X_h \le 2 \\ \text{"ERROR"} & \text{otherwise} \end{cases}$$

$V = 0.0648$

$$f := \max\left[1, \left[\frac{\left[0.0927779 - 0.0336633 \cdot X_g + 0.964176 \cdot X_g^2 \ldots + \left(0.0566286 \cdot X_h + 0.347074 \cdot X_h^2 - 4.18699 \cdot X_h^3\right)\right]}{\left[1 - 5.96093 \cdot 10^{-3} \cdot X_g + 1.62904 \cdot X_h \ldots + \left(3.49329 \cdot X_h^2 + 1.39052 \cdot X_h^3\right)\right]}\right]\right]$$

$f = 1.0000$

$$d := \frac{U \cdot g_0^2 \cdot h_o}{V}$$

$d = 81661 \cdot mm^3$

$$e := \frac{F}{h_o}$$

$e = 19.4816 \frac{1}{m}$

$$L := \frac{t \cdot e + 1}{T} + \frac{t^3}{d}$$

$L = 1.0935$

C.4 Calculation of the Flange Moments of Inertia Table 4.16.7

$$I := \frac{0.0874 \cdot L \cdot g_0^{\,2} \cdot h_o \cdot B}{V} \qquad I = 318589.9 \cdot mm^4$$

$$G_{avg} := 0.5 \cdot (g_0 + g_1) \qquad G_{avg} = 12.5 \cdot mm$$

$$A_R := 0.5 \cdot (A - B) \qquad A_R = 62.5 \cdot mm$$

$$A_A := \begin{cases} A_R & \text{if } t \geq G_{avg} \\ (h + t) & \text{if } t < G_{avg} \end{cases} \qquad A_A = 62.5 \cdot mm$$

$$B_B := \begin{cases} t & \text{if } t \geq G_{avg} \\ G_{avg} & \text{if } t < G_{avg} \end{cases} = 0.0239 \qquad B_B = 23.9 \cdot mm$$

$$C_C := \begin{cases} h & \text{if } t \geq G_{avg} \\ A_R - G_{avg} & \text{if } t < G_{avg} \end{cases} \qquad C_C = 52.8 \cdot mm$$

$$D_{DG} := \begin{cases} G_{avg} & \text{if } t \geq G_{avg} \\ t & \text{if } t < G_{avg} \end{cases} \qquad D_{DG} = 12.5 \cdot mm$$

$$K_{AB} := A_A \cdot B_B^{\,3} \cdot \left[\frac{1}{3} - 0.21 \cdot \frac{B_B}{A_A} \cdot \left[1 - \frac{1}{12} \cdot \left(\frac{B_B}{A_A}\right)^4\right]\right]$$

$$K_{CD} := C_C \cdot D_{DG}^{\,3} \cdot \left[\frac{1}{3} - 0.105 \cdot \frac{D_{DG}}{C_C} \cdot \left[1 - \frac{1}{192} \cdot \left(\frac{D_{DG}}{C_C}\right)^4\right]\right]$$

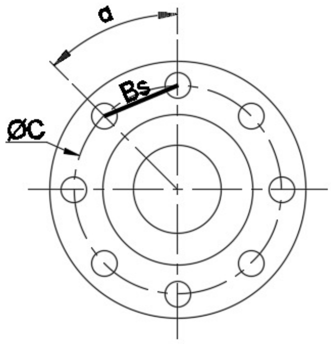
$$I_P := K_{AB} + K_{CD} \qquad I_P = 247788.6 \cdot mm^4$$

C.5 Moments

External moment

$$M_{oe} := 4 \cdot M_E \cdot \left(\frac{I}{0.3846 \cdot I_P + I}\right) \cdot \frac{h_D}{C - 2 \cdot h_D} + F_A \cdot h_D \qquad M_{oe} = 5874.5 \cdot lbf \cdot in \qquad \boxed{M_{oe} = 663726 \cdot N \cdot mm}$$

<u>Bolt Spacing Equations Table 4.16.11</u>



Bolt circle diameter : $C = 9.5 \cdot in$ $\boxed{C = 241.3 \cdot mm}$

Angle between 2 bolts : $\alpha := \frac{2 \cdot \pi}{N_{trod}}$ $\alpha = 45 \cdot deg$

Bolt spacing : $B_s := C \cdot \sin\left(\frac{\alpha}{2}\right)$ $B_s = 3.6355 \cdot in$ $\boxed{B_s = 92.3 \cdot mm}$

Nota : The bolt spacing used is the chord length between adjacent bolt locations

<u>Maximum bolt spacing according to Appendix 2 §2.5(d) eq.3 of Section VIII Div.1</u> :

MANDATORY only for lethal.service.

$$B_{smax} := 2 \cdot d_{trod} + \frac{6 \cdot t}{m_{gkt} + 0.5} = 3.8331 \cdot in \qquad \boxed{B_{smax} = 97.4 \cdot mm} \qquad B_s < B_{smax} = 1$$

with : $B_{sc} := \max\left(1, \sqrt{\frac{B_s}{2d_{trod} + t}}\right) = 1.2021$

otherwise Bolt spacing correction factor : $B_{sc} := 1$ **for non lethal service**

Moment acting on the flange due to hydrostatic end force for internal pressure

$$M_{oint} := F_s \cdot \left[\left(H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G\right) \cdot B_{sc} + M_{oe}\right] \qquad \boxed{M_{oint} = 2247103 \cdot N \cdot mm}$$

With Fs = 1.0 for non-split rings (non split loose type flanges) $M_{oint} = 19888.5 \cdot in \cdot lbf$

Moment acting on the flange due to bolting up condition for internal pressure

$$M_{gint} := \frac{W_g \cdot (C - G) \cdot B_{sc} \cdot F_s}{2} \qquad \boxed{M_{gint} = 4756111 \cdot N \cdot mm} \qquad M_{gint} = 42095.1 \cdot in \cdot lbf$$

Moment acting on the flange due to hydrostatic end force for external pressure

$$M_{oext} := \left|F_s \cdot \left[H_{Dext} \cdot \left(h_D - h_G\right) + H_{Text} \cdot \left(h_T - h_G\right) + M_{oe}\right]\right| \qquad \boxed{M_{oext} = 701388 \cdot N \cdot mm}$$

Fs = 1.0 for non-split rings (non split loose type flanges) $M_{oext} = 6207.8 \cdot in \cdot lbf$

Moment acting on the flange due to bolting up condition for external pressure

$$M_{gext} := W_g \cdot h_G \cdot F_s \qquad M_{gext} = 42095.1 \cdot in \cdot lbf \qquad \boxed{M_{gext} = 4756110.7 \cdot N \cdot mm}$$

Maximal moments :

$$M_o := \max\left(M_{oext}, M_{oint}\right) = 19888.5 \cdot in \cdot lbf \qquad M_g := \max\left(M_{gext}, M_{gint}\right) = 42095.1 \cdot in \cdot lbf$$

$$\boxed{M_o = 2.2 \times 10^6 \cdot N \cdot mm} \qquad \boxed{M_g = 4.8 \times 10^6 \cdot N \cdot mm}$$

C.6 Calculation of Flange stresses in Operating and Bolting Up conditions, Table 4.16.8

Longitudinal Hub Stress in Flange

$$S_{H.o} := \frac{f \cdot M_o}{L \cdot g_1^{\,2} \cdot B} \qquad S_{H.o} = 38.9 \cdot \text{MPa}$$

$$S_{H.g} := \frac{f \cdot M_g}{L \cdot g_1^{\,2} \cdot B} \qquad S_{H.g} = 82.4 \cdot \text{MPa}$$

Radial Stress in Flange

$$S_{R.o} := \frac{(1.33 \cdot t \cdot e + 1) \cdot M_o}{L \cdot t^2 \cdot B} \qquad S_{R.o} = 37.6 \cdot \text{MPa}$$

$$S_{R.g} := \frac{(1.33 \cdot t \cdot e + 1) \cdot M_g}{L \cdot t^2 \cdot B} \qquad S_{R.g} = 79.5 \cdot \text{MPa}$$

Tangential Stress in Flange

$$S_{T.o} := \frac{Y \cdot M_o}{t^2 \cdot B} - Z \cdot S_{R.o} \qquad S_{T.o} = 16.9 \cdot \text{MPa}$$

$$S_{T.g} := \frac{Y \cdot M_g}{t^2 \cdot B} - Z \cdot S_{R.g} \qquad S_{T.g} = 35.7 \cdot \text{MPa}$$

C.7 Allowable flange design stresses at Operating and Bolting Up condition

	Stress values	Allowable Stresses	Criteria evaluation *(1=true, 0=false)*
Longitudinal Hub Stress	$S_{H.o} = 38.9 \cdot \text{MPa}$	$\min(1.5 \cdot S_{fo}, 2.5 \cdot S_{no}) = 138.4 \cdot \text{MPa}$	$S_{H.o} \le \min(1.5 \cdot S_{fo}, 2.5 \cdot S_{no}) = 1$
	$S_{H.g} = 82.4 \cdot \text{MPa}$	$\min(1.5 \cdot S_{fg}, 2.5 \cdot S_{ng}) = 172.5 \cdot \text{MPa}$	$S_{H.g} \le \min(1.5 \cdot S_{fg}, 2.5 \cdot S_{ng}) = 1$
Radial Hub Stress	$S_{R.o} = 37.6 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$S_{R.o} \le S_{fo} = 1$
	$S_{R.g} = 79.5 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$S_{R.g} \le S_{fg} = 1$
Tangential Hub Stress	$S_{T.o} = 16.9 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$S_{T.o} \le S_{fo} = 1$
	$S_{T.g} = 35.7 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$S_{T.g} \le S_{fg} = 1$
Combined Hub Stress 1	$\frac{S_{H.o} + S_{R.o}}{2} = 38.2 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$\frac{S_{H.o} + S_{R.o}}{2} \le S_{fo} = 1$
	$\frac{S_{H.g} + S_{R.g}}{2} = 80.9 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$\frac{S_{H.g} + S_{R.g}}{2} \le S_{fg} = 1$

Combined Hub Stress 2

$$\frac{S_{H.o} + S_{T.o}}{2} = 27.9 \cdot MPa \qquad S_{fo} = 92.3 \cdot MPa \qquad \frac{S_{H.o} + S_{T.o}}{2} \le S_{fo} = 1$$

$$\frac{S_{H.g} + S_{T.g}}{2} = 59.1 \cdot MPa \qquad S_{fg} = 115 \cdot MPa \qquad \frac{S_{H.g} + S_{T.g}}{2} \le S_{fg} = 1$$

C.8 Flange rigidity criterion

Modulus of Elasticity at operating temperature: $E_{yo} = 28800000 \cdot MPa$

Modulus of Elasticity at gasket seating condition $E_{yg} = 29400000 \cdot MPa$

Rigidity index factor: $K_R := 0.3$

$$J_o := \frac{52.14 \cdot V \cdot M_o}{L \cdot E_{yo} \cdot g_0^{\,2} \cdot K_R \cdot h_o} \qquad J_o = 0.0006$$

$$J_g := \frac{52.14 \cdot V \cdot M_g}{L \cdot E_{yg} \cdot g_0^{\,2} \cdot K_R \cdot h_o} \qquad J_g = 0.0012$$

(1=true, 0=false)

J_o shall be smaller or equal than 1.0 $J_o \le 1.0 = 1$

J_g shall be smalle or equal than 1.0 $J_g \le 1.0 = 1$

APPENDIX 1

(IMPACT TEST REQUIREMENTS UG 20 (f))

Minimum design metal temperature : $T_{min} = 0 \cdot °C$

PARTS	METAL	IMPACT TEST
FLAT HEADS nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
LONGITUDINAL GIRDER nominal thick : 70 mm type : CP40	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Aa nominal thick : 80 mm	SA 516 gr 60 Normalized Governing Thick : 20 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Ab nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Ba nominal thick : 80 mm	SA 516 gr 60 Normalized Governing Thick : 20 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Bb nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D

Plate governing thick =

plate without nozzles : ¼ nominal thickness (UCS 66 (a)(3))

plate with nozzles : the larger of ¼ panel nominal thickness and nozzle pipe thickness (UCS 66 (a)(1)(c))

THREAD ROD [Dia <= 64 mm	SA 193 gr B7	IMPACT TEST EXEMPTION ACCORDING TO FIG. UCS 66M NOTE (c) ABOVE -55°F (-48°C)
NUTS	SA 194 gr 7	IMPACT TEST EXEMPTION ACCORDING TO FIG. UCS 66M NOTE (c) ABOVE -55°F (-48°C)
NON CODE EXCHANGE PLATE	SA 240 type 316L Governing Thick : 1 mm	EXEMPTED PER UHA 51
WELDS	Weld metal + Heat affected zone (Note : Mandatory in all cases if MDMT < - 29°C)	NOT REQUIRED ACCORDING TO UCS 67 (a) WHEN IMPACT TEST IS NOT REQUIRED ON BASE METAL
	Impact test on production test plate	NOT REQUIRED PER UG 84
NOZZLE FLANGE ASME B16-5	SA 182 F 316L	NOT REQUIRED ACC. TO UHA 51
NOZZLE PIPE NPS 6"	SA 312 TP 316L Nominal thick :7.11 mm (sch.40 s)	EXEMPTED PER UHA 51
LININGS	SA 240 type 316L Thickness = 3 mm	EXEMPTED PER UHA 51

APPENDIX 2

TEST PRESSURE CALCULATION ACCORDING TO ASME VIII Div.1 CODE UG-99 (b)

Test pressure = 1.3 x Design pressure x Lowest ratio (S at test temperature / S at design temperature)

According to the configuration of hydrostatic test carry out on exchanger (Manometer on the top of the CP), we use for the definition of test pressure the design pressure without the pressure static head of liquid.

Design pressure Side A : $P_{.designA} = 1.04 \cdot MPa$ $P_{.designA} = 10.4 \cdot bar$

Design pressure Side B : $P_{.designB} = 1.04 \cdot MPa$ $P_{.designB} = 10.4 \cdot bar$

Bolting Side A (Tie rod) : $Mat_{trod} = \text{"SA 193 gr B7"}$ — Ratio

Allowable stress at test temperature : $Sa_{trod} = 172 \cdot MPa$

Allowable stress at design temperature : $Sb_{trod} = 172 \cdot MPa$

$$\frac{Sa_{trod}}{Sb_{trod}} = 1$$

Bolting Side B (Tie rod) : $Mat_{trod} = \text{"SA 193 gr B7"}$ — Ratio

Allowable stress at test temperature : $Sa_{trod.B} = 172 \cdot MPa$

Allowable stress at design temperature : $Sb_{trod.B} = 172 \cdot MPa$

$$\frac{Sa_{trod.B}}{Sb_{trod.B}} = 1$$

Bolting ratio is not included in the test pressure calculation.

Panels side A : $Mat_{panel} = \text{"SA 516 gr 60N"}$

Allowable stress at test temperature : $Sa_{panel} = 118 \cdot MPa$

Allowable stress at design temperature : $Sb_{panel} = 118 \cdot MPa$

$$\frac{Sa_{panel}}{Sb_{panel}} = 1$$

Panels side B : $Mat_{panel} = \text{"SA 516 gr 60N"}$

Allowable stress at test temperature : $Sa_{panel.B} = 118 \cdot MPa$

Allowable stress at design temperature : $Sb_{panel.B} = 118 \cdot MPa$

$$\frac{Sa_{panel.B}}{Sb_{panel.B}} = 1$$

Head : $Mat_{head} = \text{"SA 516 gr 60N"}$

Allowable stress at test temperature : $Sa_{head} = 118 \cdot MPa$

Allowable stress at design temperature : $Sb_{head} = 118 \cdot MPa$

$$\frac{Sa_{head}}{Sb_{head}} = 1$$

Columns : $Mat_{columns} = \text{"SA 516 gr 60N"}$

Allowable stress at test temperature : $Sa_{col} = 118 \cdot MPa$

Allowable stress at design temperature : $Sb_{col} = 118 \cdot MPa$

$$\frac{Sa_{col}}{Sb_{col}} = 1$$

Pipe A1 : $Mat_{PipeA1} = "SA\ 312\ TP\ 316L"$

Allowable stress at test temperature : $Sa_{PipeA1} = 115 \cdot MPa$ $\quad \frac{Sa_{PipeA1}}{Sb_{PipeA1}} = 1$

Allowable stress at design temperature : $Sb_{PipeA1} = 115 \cdot MPa$

Flange A1 : $Mat_{FlangeA1} = "SA\ 182\ F\ 316L"$

Allowable stress at test temperature : $Sa_{FlangeA1} = 115 \cdot MPa$ $\quad \frac{Sa_{FlangeA1}}{Sb_{FlangeA1}} = 1.2459$

Allowable stress at design temperature : $Sb_{FlangeA1} = 92.3 \cdot MPa$

Pipe A2 : $Mat_{PipeA2} = "SA\ 312\ TP\ 316L"$ $\quad \frac{Sa_{PipeA2}}{Sb_{PipeA2}} = 1$

Allowable stress at test temperature : $Sa_{PipeA2} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{PipeA2} = 115 \cdot MPa$

Flange A2 : $Mat_{FlangeA2} = "SA\ 182\ F\ 316L"$ $\quad \frac{Sa_{FlangeA2}}{Sb_{FlangeA2}} = 1.2459$

Allowable stress at test temperature : $Sa_{FlangeA2} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{FlangeA2} = 92.3 \cdot MPa$

Pipe B1 : $Mat_{PipeB1} = \text{"SA 312 TP 316L"}$ $\frac{Sa_{PipeB1}}{Sb_{PipeB1}} = 1$

Allowable stress at test temperature : $Sa_{PipeB1} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{PipeB1} = 115 \cdot MPa$

Flange B1 : $Mat_{FlangeB1} = \text{"SA 182 F 316L"}$ $\frac{Sa_{FlangeB1}}{Sb_{FlangeB1}} = 1.2459$

Allowable stress at test temperature : $Sa_{FlangeB1} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{FlangeB1} = 92.3 \cdot MPa$

Pipe B2 : $Mat_{PipeB2} = \text{"SA 312 TP 316L"}$

Allowable stress at test temperature : $Sa_{PipeB2} = 115 \cdot MPa$ $\frac{Sa_{PipeB2}}{Sb_{PipeB2}} = 1$

Allowable stress at design temperature : $Sb_{PipeB2} = 115 \cdot MPa$

Flange B2 : $Mat_{FlangeB2} = \text{"SA 182 F 316L"}$

Allowable stress at test temperature : $Sa_{FlangeB2} = 115 \cdot MPa$ $\frac{Sa_{FlangeB2}}{Sb_{FlangeB2}} = 1.2459$

Allowable stress at design temperature : $Sb_{FlangeB2} = 92.3 \cdot MPa$

Plates or Linings : $Mat_{plates} = \text{"SA 240 type 316L"}$

Allowable stress at test temperature : $Sa_{plates} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{plates} = 92.3 \cdot MPa$

$$\frac{Sa_{plates}}{Sb_{plates}} = 1.2459$$

Linings side A : $Mat_{linings.A} = \text{"SA 240 type 316L"}$

Allowable stress at test temperature : $Sa_{linings.A} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{linings.A} = 92.3 \cdot MPa$

$$\frac{Sa_{linings.A}}{Sb_{linings.A}} = 1.2459$$

Linings side B : $Mat_{linings.B} = \text{"SA 240 type 316L"}$

Allowable stress at test temperature : $Sa_{linings.B} = 115 \cdot MPa$

Allowable stress at design temperature : $Sb_{linings.B} = 92.3 \cdot MPa$

$$\frac{Sa_{linings.B}}{Sb_{linings.B}} = 1.2459$$

$Lowest_{ratio.A} = 1$ $Lowest_{ratio.B} = 1$

Nota : If Test pressure calculated below is greater than $1.3 \cdot Lowest_{ratio} \cdot P_{design}$ check the calculation done with 90% of yield constraints according to ASME VIII div.1 CODE UG-99(c)

$Coef_{maxA} = 1.3$ $Coef_{maxB} = 1.3$

Adopted Test Pressure

$Test_{pressure.A} = Coef_{maxA} \cdot P_{.designA}$ $Test_{pressure.B} = Coef_{maxB} \cdot P_{.designB}$

$\boxed{Test_{pressure.A} = 1.36 \cdot MPa}$ $\boxed{Test_{pressure.B} = 1.36 \cdot MPa}$

$\boxed{Test_{pressure.A} = 13.6 \cdot bar}$ $\boxed{Test_{pressure.B} = 13.6 \cdot bar}$

$Test_{pressure.A} = 197 \cdot psi$ $Test_{pressure.B} = 197 \cdot psi$

APPENDIX 3

FULL VACUUM CALCULATION NOTE

1. SQUARE HEADS

Since these parts are unstayed flat heads designed previously at $P = 1.06 \cdot MPa$ (for internal pressure), no new calculation is necessary for full vacuum (15 psi, 1.03 bar, 0.103MPa).

2. STEEL PANELS (applicable if analytical calculation done)

Since these parts are unstayed flat heads designed previously at $P = 1.06 \cdot MPa$ (for internal pressure on side A and on side B), no new calculation is necessary for full vacuum (15 psi, 1.03 bar 0.103MPa).

3. NOZZLES PIPES LINING

Calculations according to ASME VIII Edition 2019, UG 28

3.1.1 - Pipe size :

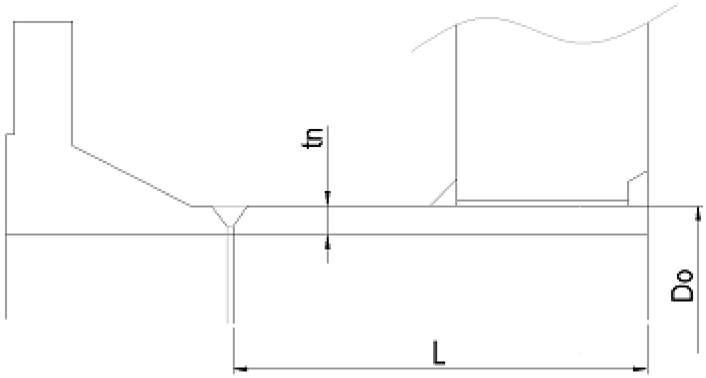
Item : $Item_{opening}$ = "A1-A2-B1-B2"   Pipe material : Mat_{pipe} = "SA 312 TP 316L"

Nominal diameter : N_{Dia} = "NPS 6""

External diameter : $D_o = 168.3 \cdot mm$
(in corroded condition)

Length of pipe : $L = 165 \cdot mm$

Thickness of pipe : $t_n = 6.6 \cdot mm$
(in corroded condition)



Step 1 - Ratio determination :

$$\frac{L}{D_o} = 0.9804 \qquad \frac{D_o}{t_n} = 25.5$$

Step 2, 3 - Factor A determination according to ASME II Part D - Subpart 3, Fig.G

$$A = 0.011134$$

Step 4, 5 - Factor B determination according to ASME II Part D - Subpart 3,   Fig = "HA-4"

$$B = 82 \cdot MPa$$

Step 6 - Calculation of Maximun allowable external pressure.

- For pipe having value of $\frac{D_o}{t_n} \geq 10$

Value of P_a

$$P_a := 4 \cdot \frac{B}{3\left(\frac{D_o}{t_n}\right)} \qquad P_a = 4.3 \cdot MPa$$

- For pipe having value of $\frac{D_o}{t_n} < 10$

Value of P_{a1}

$$P_{.a1} := \left(\frac{2.167}{\frac{D_o}{t_n}} - 0.0833\right) \cdot B \qquad P_{a1} = 0.1 \cdot MPa$$

(P_{a1} value can not be lower than 0 MPa)

<u>Value of</u> P_{a2}

Allowable stress (Pipe) $S_m = 115 \cdot MPa$ (ASME II part D - Table 1-A)

Yield strength value (Pipe) $S_y = 139 \cdot MPa$ (ASME II part D - Table Y-1)

$S := \min\left(2 \cdot S_m, 0.9 \cdot S_y\right)$ $S = 125.1 \cdot MPa$

$$P_{a2} := \frac{2 \cdot S}{\frac{D_o}{t_n}} \cdot \left(1 - \frac{1}{\frac{D_o}{t_n}}\right)$$ $P_{a2} = 9.4 \cdot MPa$

- Result of the calculation:

$$P_{ext} := \begin{cases} P_a & \text{if } \frac{D_o}{t_n} \geq 10 \\ \min\left(P_{a1}, P_{a2}\right) & \text{otherwise} \end{cases}$$ $P_{ext} = 4.3 \cdot MPa$

External pressure P_{ext} shall be upper or equal than required external working pressure $P_{.UG28.f}$

$P_{.UG28.f} = 0.103 \cdot MPa$ (*1=true, 0=false*) $P_{ext} \geq P_{.UG28.f} = 1$

$P_{ext} = 4.3 \cdot MPa$

4. PANEL LINING SIDE A

Calculation of panels linings in full vacuum condition according to ASME VIII Div.1, Appendix 17.

Lining material : $Mat_{lining} = \text{"SA 240 type 316L"}$

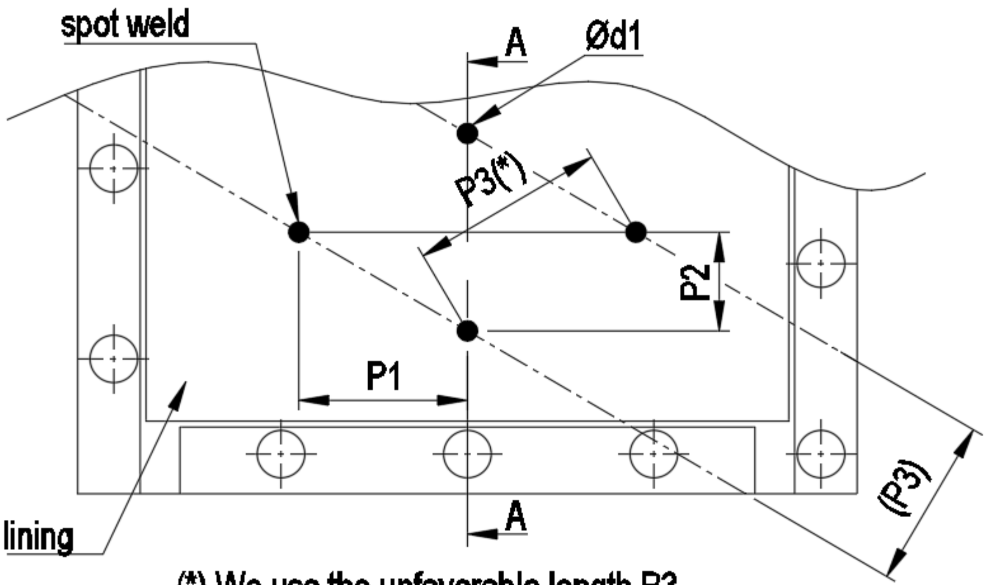
Lining thickness : $Th_{lining} = 3 \cdot mm$

Maximum allowable stress at design Temperature : $S_{lining} = 92.3 \cdot MPa$

Thickness limitations (Appendix 17, App. 17-4) : $Th_{min} := 0.8mm$ *(1=true, 0=false)*

The nominal thickness Th_{lining} shall be not less than Th_{min} $Th_{lining} \geq Th_{min} = 1$

Lining attachment :
Lining composed of one sheet attached to side panels by spot welds symmetrically spaced on a rectangular pitch (Appendix 17, Fig. 17-4)



(*) We use the unfavorable length P3 instead of (P3) to be more conservative

$p_1 = 85 \cdot mm$ $p_2 = 50 \cdot mm$ $p_3 = 98.6 \cdot mm$ $p := \max(p_1, p_2, p_3)$ $p = 98.6 \cdot mm$

Maximum allowable working pressure App. 17-5 (b)(2)

Calculation of maximum allowable working pressure P according to UG-47 equation (2) :

$$P_{fv} := \frac{Th_{lining}^{2} \cdot S_{lining} \cdot C_{.lining}}{p^2}$$ $P_{fv} = 0.179 \cdot MPa$

with : $C_{.lining} = 2.1$

$p = 98.6 \cdot mm$ maximum pitch = greater of p_1, p_2, p_3

$Th_{lining} = 3 \cdot mm$

$S_{lining} = 92.3 \cdot MPa$

External pressure P_{fv} shall be upper than required external working pressure $P_{.UG28.f}$

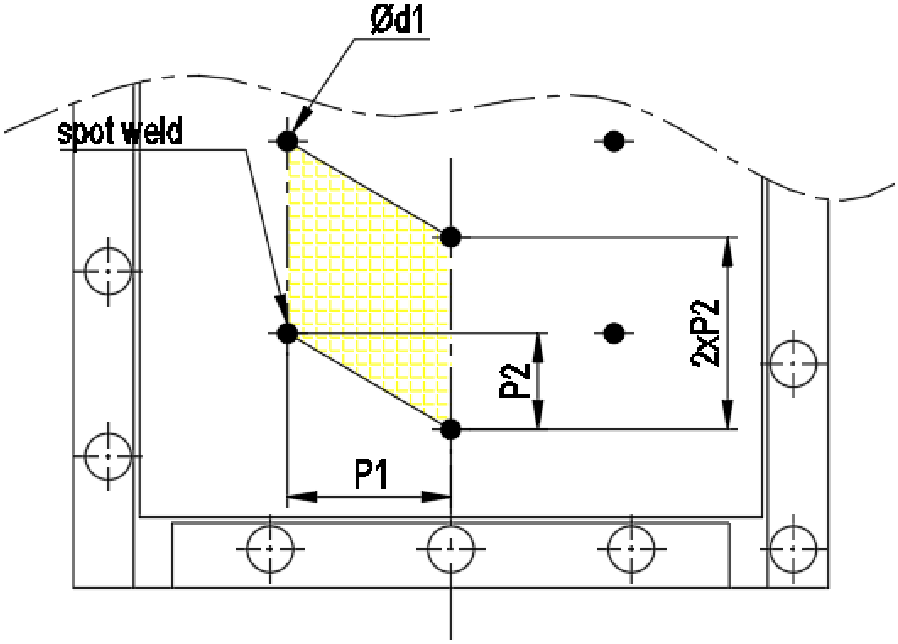
$P_{.UG28.f} = 0.103 \cdot MPa$ (*1=true, 0=false*) $P_{fv} \geq P_{.UG28.f} = 1$

$P_{fv} = 0.179 \cdot MPa$

Spot welds verification :

$p_1 = 85 \cdot mm$

$p_2 = 50 \cdot mm$



Considered an area : $S_{fv} := 2 \cdot p_2 \cdot p_1$ $S_{fv} = 8500 \cdot mm^2$

Design pressure : $p_{.fv} := P_{.UG28.f}$ $p_{.fv} = 0.103 \cdot MPa$

Load due to pressure on this area : $F_{fv} := S_{fv} \cdot p_{.fv}$ $F_{fv} = 875.5 \cdot N$

Strength section of spot/screw welds supporting this area
:

1 spot weld with a diameter of
: $d_1 = 6 \cdot mm$ $\boxed{d_1 = 6 \cdot mm}$

Allowable stress at design Temperature $S_{lining} = 92.3 \cdot MPa$ $\boxed{S_{lining} = 92.3 \cdot MPa}$

Tension area of spot weld $S_T := \frac{d_1^{\,2} \cdot \pi}{4}$ $\boxed{S_T = 28.3 \cdot mm^2}$

Shear area in lining around the
screw head / spot weld $S_s := \pi \cdot d_1 \cdot Th_{lining} = 56.5 \cdot mm^2$ $\boxed{S_s = 56.5 \cdot mm^2}$

Remark: in conservative way, in both case, the minimal diameter d1 is choosen for the shear calculation.

tension stress in spot/screw weld section
: $\sigma := \frac{S_{fv} \cdot p_{.fv}}{S_T} = 31 \cdot MPa$ $\sigma = 31 \cdot MPa$

$\sigma < S_{lining} = 1$

Shear stress on lining : $\tau := \frac{S_{fv} \cdot p_{.fv}}{S_s} = 15.5 \cdot MPa$ $\tau = 15.5 \cdot MPa$

$\tau < 0.8\, S_{lining} = 1$

Screw verification :

Load by screw : $F_{fv} = 875.5 \cdot N$

Filet shear surface in 1 screw : $Sf_{filet} := \pi \cdot d_{filet} \cdot \frac{L_{filet}}{2}$ $Sf_{filet} = 106 \cdot mm^2$

with : Diameter of thread pitch $d_{filet} = 9 \cdot mm$

Length screwed in the panel $L_{filet} = 7.5 \cdot mm$

Actual shear stress in filets : $Sh_{filet} := \frac{F_{fv}}{Sf_{filet}}$ $Sh_{filet} = 8.3 \cdot MPa$

Sh_{filet} value shall be smaller than $0.8x\,S_{screw}$ and $0.8x\,S_{panel}$ according to UG-23(f)

Allowable stress value of panel : $S_{panel} = 118 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{panel} = 1$

(1=true, 0=false)

Allowable stress value of screw : $S_{screw} = 92.3 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{screw} = 1$

5. PANEL LINING SIDE B

Calculation of panels linings in full vacuum condition according to ASME VIII Div.1, Appendix 17.

Lining material : $Mat_{lining} = \text{"SA 240 type 316L"}$

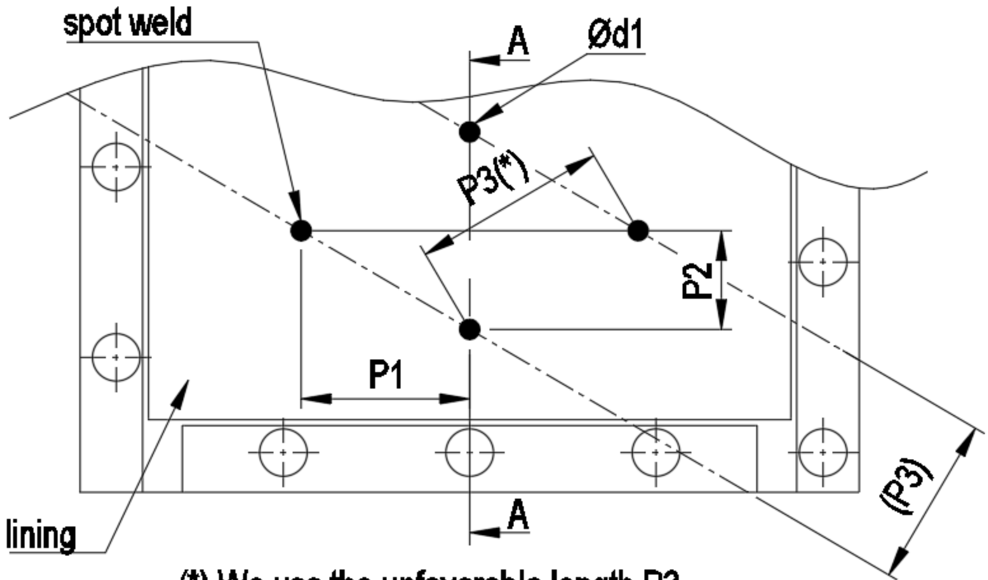
Lining thickness : $Th_{lining} = 3 \cdot mm$

Maximum allowable stress at design Temperature : $S_{lining} = 92.3 \cdot MPa$

Thickness limitations (Appendix 17, App. 17-4) : $Th_{min} := 0.8mm$ *(1=true, 0=false)*

The nominal thickness Th_{lining} shall be not less than Th_{min} $Th_{lining} \geq Th_{min} = 1$

Lining attachment :
Lining composed of one sheet attached to side panels by spot welds symmetrically spaced on a rectangular pitch (Appendix 17, Fig. 17-4)



$p_1 = 85 \cdot mm$ $p_2 = 50 \cdot mm$ $p_3 = 98.6 \cdot mm$ $p := \max(p_1, p_2, p_3)$ $p = 98.6 \cdot mm$

Maximum allowable working pressure App. 17-5 (b)(2)

Calculation of maximum allowable working pressure P according to UG-47 equation (2) :

$$P_{fv} := \frac{Th_{lining}^{\,2} \cdot S_{lining} \cdot C_{.lining}}{p^2} \qquad P_{fv} = 0.179 \cdot MPa$$

with : $C_{.lining} = 2.1$

$p = 98.6 \cdot mm$ maximum pitch = greater of $p_1, 2p_2, p_3$

$Th_{lining} = 3 \cdot mm$

$S_{lining} = 92.3 \cdot MPa$

External pressure P_{fv} shall be upper than required external working pressure $P_{.UG28.f}$

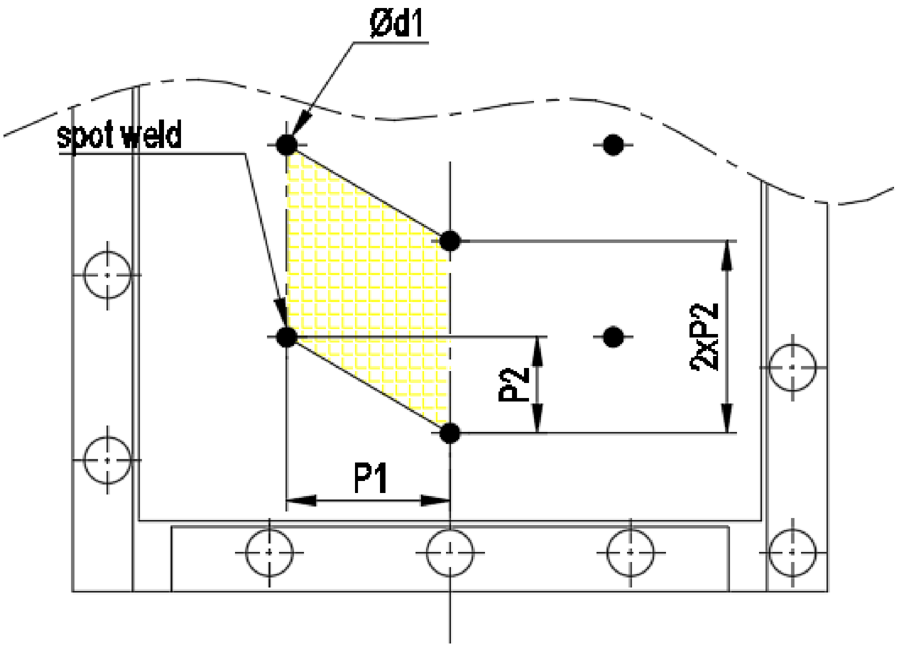
$P_{.UG28.f} = 0.103 \cdot MPa$ (*1=true, 0=false*) $P_{fv} \geq P_{.UG28.f} = 1$

$P_{fv} = 0.179 \cdot MPa$

Spot welds verification :

$p_1 = 85 \cdot mm$

$p_2 = 50 \cdot mm$



Considered an area : $S_{fv} := 2 \cdot p_2 \cdot p_1$ $S_{fv} = 8500 \cdot mm^2$

Design pressure : $p_{.fv} := P_{.UG28.f}$ $p_{.fv} = 0.103 \cdot MPa$

Load due to pressure on this area : $F_{fv} := S_{fv} \cdot p_{.fv}$ $F_{fv} = 875.5 \cdot N$

Strength section of spot/screw welds supporting this area :

1 spot weld with a diameter of : $d_1 = 6 \cdot mm$ $\boxed{d_1 = 6 \cdot mm}$

Allowable stress at design Temperature $S_{lining} = 92.3 \cdot MPa$ $\boxed{S_{lining} = 92.3 \cdot MPa}$

Tension area of spot weld $S_T := \dfrac{d_1^{\,2} \cdot \pi}{4}$ $\boxed{S_T = 28.2743 \cdot mm^2}$

Shear area in lining around the screw head / spot weld $S_s := \pi \cdot d_1 \cdot Th_{lining} = 56.5487 \cdot mm^2$ $\boxed{S_s = 56.5487 \cdot mm^2}$

Remark: in conservative way, in both case, the minimal diameter d1 is choosen for the shear calculation.

tension stress in spot/screw weld section : $\sigma := \dfrac{F_{fv}}{S_T} = 31 \cdot MPa$ $\sigma = 31 \cdot MPa$

$\sigma < S_{lining} = 1$

Shear stress on lining : $\tau := \dfrac{F_{fv}}{S_s} = 15.5 \cdot MPa$ $\tau = 15.5 \cdot MPa$

$\tau < 0.8\, S_{lining} = 1$

Screw verification :

Load by screw : $F_{fv} = 875.5 \cdot N$

Filet shear surface in 1 screw : $Sf_{filet} := \pi \cdot d_{filet} \cdot \frac{L_{filet}}{2}$ $Sf_{filet} = 106 \cdot mm^2$

with : Diameter of thread pitch $d_{filet} = 9 \cdot mm$

Length screwed in the panel $L_{filet} = 7.5 \cdot mm$

Actual shear stress in filets : $Sh_{filet} := \frac{F_{fv}}{Sf_{filet}}$ $Sh_{filet} = 8.3 \cdot MPa$

Sh_{filet} value shall be smaller than $0.8x\,S_{screw}$ and $0.8x\,S_{panel}$ according to UG-23(f)

Allowable stress value of panel : $S_{panel} = 118 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{panel} = 1$

(1=true, 0=false)

Allowable stress value of screw : $S_{screw} = 92.3 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{screw} = 1$