Calculation report
EN 13445 Ed. 2014 Issue 3

Project: Laboratorní autokláv
Item: Laboratorní autokláv 1L
Customer: VUOS a.s.
Drawing: 400-001
Revision: -
Date: 04.08.2018

Internal design pressure \( P = 4.00 \text{ MPa} \)
Internal design temperature \( T = 200.00 \text{ °C} \)
Internal corrosion allowance \( c = 0 \text{ mm} \)
External corrosion allowance \( ce = 0 \text{ mm} \)
Joint efficiency \( z = 0.85 \)
Minimum design temperature \( = 0 \text{ °C} \)
Test pressure (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>P</th>
<th>Static head (design)</th>
<th>Static head (test)</th>
<th>Stress ratio</th>
<th>1.25·P·f₀/f</th>
<th>1.43·P</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conical shell #1</td>
<td>4.00</td>
<td>0</td>
<td>0.002</td>
<td>1.81</td>
<td>9.05</td>
<td>5.72</td>
</tr>
<tr>
<td>Cylindrical shell #1</td>
<td>4.00</td>
<td>0</td>
<td>0.001</td>
<td>1.81</td>
<td>9.05</td>
<td>5.72</td>
</tr>
<tr>
<td>Loose lap flange #1</td>
<td>4.00</td>
<td>0</td>
<td>0.0005</td>
<td>1.327</td>
<td>6.63</td>
<td>5.72</td>
</tr>
<tr>
<td>Bolted flat cover #1</td>
<td>4.00</td>
<td>0</td>
<td>0.00003</td>
<td>1.81</td>
<td>9.05</td>
<td>5.72</td>
</tr>
</tbody>
</table>

All pressures in MPa.

Item design pressure P = 4.00 MPa
Item maximum allowable design pressure (Pmax) = 4.26 MPa (limited by Loose lap flange #1)
Item lowest stress ratio = 1.327 (limited by Loose lap flange #1)
Item test pressure = \( P_t = \max[1.25 \cdot P_d \cdot (\text{Item } f_0/f); 1.43 \cdot P_d] = 6.63 \) MPa

Maximum Pressures (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>Internal, test</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conical shell #1</td>
<td>18.29</td>
<td>5.99</td>
</tr>
<tr>
<td>Cylindrical shell #1</td>
<td>28.33</td>
<td>8.15</td>
</tr>
<tr>
<td>Loose lap flange #1</td>
<td>5.00</td>
<td>4.26</td>
</tr>
<tr>
<td>Bolted flat cover #1</td>
<td>13.10</td>
<td>4.66</td>
</tr>
</tbody>
</table>

All pressures in MPa.

Weights

<table>
<thead>
<tr>
<th>Component</th>
<th>Dead</th>
<th>Live</th>
<th>Liquid</th>
<th>Full of water</th>
<th>Operating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bolted flat cover #2</td>
<td>5 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>5 kg</td>
<td>5 kg</td>
</tr>
<tr>
<td>Welded flat cover #1</td>
<td>1 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 kg</td>
<td>1 kg</td>
</tr>
<tr>
<td>Conical shell #1</td>
<td>1 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 kg</td>
<td>1 kg</td>
</tr>
<tr>
<td>Cylindrical shell #1</td>
<td>2 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>3 kg</td>
<td>2 kg</td>
</tr>
<tr>
<td>Loose lap flange #1</td>
<td>8 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>8 kg</td>
<td>8 kg</td>
</tr>
<tr>
<td>Bolted flat cover #1</td>
<td>6 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>6 kg</td>
<td>6 kg</td>
</tr>
</tbody>
</table>

Totals: 23 kg  0 kg  0 kg  25 kg  23 kg

Total volume: 0.00153 m³
Center of gravity (erection): Cₓ=0 mm, Cᵧ=0 mm, Cₗ=127.73 mm, W = 17 kg
Center of gravity (operating): Cₓ=0 mm, Cᵧ=0 mm, Cₗ=127.73 mm, W = 17 kg
Center of gravity (test): Cₓ=0 mm, Cᵧ=0 mm, Cₗ=124.12 mm, W = 19 kg

Definitions
Dead: net, uncorroded weight of component, including additional dead weight
Live: additional live weight on component
Liquid: weight of liquid contained in component in operating conditions (depending on liquid level)
Insulation: weight of insulation on component, when present
Full of water: sum of component's proper weight and contained water in test conditions
Operating: sum of component's proper weight and contained liquid in operating conditions
Bill of materials

<table>
<thead>
<tr>
<th>Component</th>
<th>Dimensions</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bolted flat cover #2 - Flange</td>
<td>Od = 160.00 mm, Tk = 30.00 mm</td>
<td>1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube (t ≤ 60.00 mm) - No.: 1.4301</td>
</tr>
<tr>
<td>Welded flat cover #1</td>
<td>Od = 25.00 mm, Tk = 10.00 mm</td>
<td>Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035</td>
</tr>
<tr>
<td>Conical shell #1</td>
<td>Min Id = 25.00 mm, Max Id = 115.00 mm, Tk = 6.00 mm, α = 65.00 °, L = 24.81 mm</td>
<td>Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035</td>
</tr>
<tr>
<td>Cylindrical shell #1</td>
<td>Id = 115.00 mm, Od = 127.00 mm, Tk = 6.00 mm, L = 137.00 mm</td>
<td>Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035</td>
</tr>
<tr>
<td>Loose lap flange #1 - Flange</td>
<td>Id = 142.00 mm, Od = 240.00 mm, Tk = 32.00 mm</td>
<td>1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75) - No.: 1.4571</td>
</tr>
<tr>
<td>Loose lap flange #1 - Gasket</td>
<td></td>
<td>Rubber O-rings - below 75° IRH</td>
</tr>
<tr>
<td>Loose lap flange #1 - Bolts</td>
<td></td>
<td>8 x ISO M16 x 2.00 A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)</td>
</tr>
<tr>
<td>Bolted flat cover #1 - Flange</td>
<td>Od = 240.00 mm, Tk = 30.00 mm</td>
<td>Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035</td>
</tr>
</tbody>
</table>

Material properties summary

1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube (t ≤ 60.00 mm) - No.: 1.4301

<table>
<thead>
<tr>
<th>Temp.</th>
<th>Allowable</th>
<th>Yield strength</th>
<th>Tensile strength</th>
<th>Elasticity</th>
<th>Thermal expansion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room</td>
<td>166.67 MPa</td>
<td>195.00 MPa</td>
<td>500.00 MPa</td>
<td>195 188.00 MPa</td>
<td>0.000015967 1/°C</td>
</tr>
<tr>
<td>Design</td>
<td>103.33 MPa</td>
<td>127.00 MPa</td>
<td>0 MPa</td>
<td>182 323.00 MPa</td>
<td>0.000016802 1/°C</td>
</tr>
<tr>
<td>Test</td>
<td>250.00 MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75) - No.: 1.4571

<table>
<thead>
<tr>
<th>Temp.</th>
<th>Allowable</th>
<th>Yield strength</th>
<th>Tensile strength</th>
<th>Elasticity</th>
<th>Thermal expansion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room</td>
<td>173.33 MPa</td>
<td>220.00 MPa</td>
<td>520.00 MPa</td>
<td>195 188.00 MPa</td>
<td>0.000016500 1/°C</td>
</tr>
<tr>
<td>Design</td>
<td>130.67 MPa</td>
<td>167.00 MPa</td>
<td>390.00 MPa</td>
<td>182 323.00 MPa</td>
<td>0.000017500 1/°C</td>
</tr>
<tr>
<td>Test</td>
<td>247.62 MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

<table>
<thead>
<tr>
<th>Temp.</th>
<th>Allowable</th>
<th>Yield strength</th>
<th>Tensile strength</th>
<th>Elasticity</th>
<th>Thermal expansion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room</td>
<td>291.67 MPa</td>
<td>450.00 MPa</td>
<td>700.00 MPa</td>
<td>195 188.00 MPa</td>
<td>0.000015967 1/°C</td>
</tr>
<tr>
<td>Design</td>
<td>240.00 MPa</td>
<td>360.00 MPa</td>
<td>0 MPa</td>
<td>182 323.00 MPa</td>
<td>0.000016802 1/°C</td>
</tr>
<tr>
<td>Test</td>
<td>428.57 MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035

<table>
<thead>
<tr>
<th>Temp.</th>
<th>Allowable</th>
<th>Yield strength</th>
<th>Tensile strength</th>
<th>Elasticity</th>
<th>Thermal expansion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room</td>
<td>175.00 MPa</td>
<td>300.00 MPa</td>
<td>420.00 MPa</td>
<td>22 230.75 MPa</td>
<td>0.000010780 1/°C</td>
</tr>
<tr>
<td>Design</td>
<td>96.67 MPa</td>
<td>145.00 MPa</td>
<td>420.00 MPa</td>
<td>97 000.00 MPa</td>
<td>0.000093000 1/°C</td>
</tr>
<tr>
<td>Test</td>
<td>285.71 MPa</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Notes
Allowable stress calculation may vary upon component type or characteristics
Yield strength shown refers to 0.2% plastic strain
### Design data

- **Internal design temperature** \( Ti = 200.00 \, ^\circ\text{C} \)
- **Internal design pressure** \( Pi = 4.00 \, \text{MPa} \)
- **Joint efficiency** \( z = 1.00 \)

**Material: 1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube \( t \leq 60.00 \, \text{mm} \)**

- **Nominal design stress at internal design temperature**
  \[
  f = \left( \frac{R_{p0.2\,T_i}}{1.5} \right) = 103.33 \, \text{MPa}
  \]
- **Nominal design stress at room temperature**
  \[
  f = \min \left( \frac{R_{p0.2\,T}}{1.5}, \min \left( \frac{R_{p0.2\,T}}{1.2}, \frac{R_{p0.2\,T}}{3} \right) \right) = 166.67 \, \text{MPa}
  \]
- **Nominal design stress in test condition**
  \[
  f_{\text{test}} = \max \left( \frac{R_{p0.2\,T_{\text{test}}}}{1.05}, \frac{R_{\sigma_{0.2}+T_{\text{test}}}}{2} \right) = 250.00 \, \text{MPa}
  \]

**Bolts Material: A2-70 (ISO 3506-1:2009) - Bolting \( t \leq 39.00 \, \text{mm} \)**

- **Nominal design stress at design temperature**
  \[
  f = \min \left( \frac{R_{p0.2\,T}}{3}, \frac{R_{R_{\sigma_{0.2}+T}}}{4} \right) = 120.00 \, \text{MPa}
  \]
- **Nominal design stress at room temperature**
  \[
  f = \frac{R_{R_{\sigma_{0.2}+T}}}{4} = 175.00 \, \text{MPa}
  \]
- **Nominal design stress in test condition**
  \[
  f = 1.5 \frac{R_{R_{\sigma_{0.2}+T_{\text{test}}}}}{4} = 262.50 \, \text{MPa}
  \]

### Geometry

- **Nominal thickness** \( en = 30.00 \, \text{mm} \)
- **Corrosion allowance** \( c = 0 \, \text{mm} \)
- **External corrosion allowance** \( ce = 0 \, \text{mm} \)
- **Undertolerance** \( \delta = 0 \, \text{mm} \)
- **Bolt circle** \( C = 206.00 \, \text{mm} \)
- **Mean gasket diameter** \( G_{\text{mean}} = 117.00 \, \text{mm} \)
- **Adopted thickness (periphery)** \( e_1 = 24.00 \, \text{mm} \)
- **Gasket groove depth** \( g = 0 \, \text{mm} \)
- **External corrosion allowance** \( c' = 0 \, \text{mm} \)
- **Flange external diameter** \( A = 160.00 \, \text{mm} \)
- **Inside diameter of connected flange** \( B = 115.00 \, \text{mm} \)
- **Inside diameter of connected flange (corroded)** \( B^* = B + 2c = 115.00 \, \text{mm} \)
- **Hub thickness of connected flange** \( g_1 = 4.00 \, \text{mm} \)
- **Hub thickness of connected flange (corroded)** \( g_1^* = 4.00 \, \text{mm} \)

### Gasket parameters

- **Gasket factor** \( m = 0 \, \text{mm} \)
- **Minimum gasket seating pressure** \( y = 0.70 \, \text{MPa} \)
- **Gasket contact width** \( w = 0 \, \text{mm} \)
- **Basic gasket seating width** \( b_0 = w / 2 = 0 \, \text{mm} \)
- **Effective gasket seating width** \( b = b_0 = 0 \, \text{mm} \)
- **Diameter of gasket load reaction** \( G = G_{\text{mean}} = 117.00 \, \text{mm} \)

### Bolt loads

- **Number of bolts** \( = 8 \)
- **Bolt type** \( = \) ISO M16 x 2.00
- **Bolt hole diameter** \( = 18.00 \, \text{mm} \)
- **Bolt spacing** \( \text{t} \text{bmax} = 80.90 \, \text{mm} \)
- **Root area of one bolt** \( = 144.0 \, \text{mm}^2 \)
- **Distance between centre lines of adjacent bolts** \( \text{d} \text{b} = 80.90 \, \text{mm} \)
- **Total hydrostatic end force**
  \[
  H = \frac{G^2 \pi P}{4} = 43 \, 005 \, \text{N}
  \]
- **Minimum required bolt load for operating condition**
  \[
  W_{\text{op}} = H = 43 \, 005 \, \text{N}
  \]
- **Minimum required bolt load for the test condition**
  \[
  H_{1} = \frac{G^2 \pi P_{1}}{4} = 71 \, 310 \, \text{N}
  \]
  \[
  W_{t} = H_{1} = 71 \, 310 \, \text{N}
  \]
Minimum required bolt load for assembly condition

\[
W_{A_{\text{min}}} = \max \left[ \frac{W_{d}}{f_{E_d}}, \frac{W_{B}}{f_{E_B}} \right]
\]

\[
W_{A_{\text{min}}} = 0 \text{ N}
\]

Total required cross-sectional area of bolts

\[
A_{B} = 1152.0 \text{ mm}^2
\]

Total cross-sectional area of bolts at the section of least bolt diameter

\[
A_{B_{\text{max}}} = \frac{2\pi \cdot y \cdot G \cdot w}{f_{E_d}}
\]

\[
W = 0.5(A_{B_{\text{min}}} + A_{B})f_{E_d} = 132 158 \text{ N}
\]

Design bolt load for assembly condition

\[
AB = AB_{\text{min}}: \text{ Ok}
\]

Length of screws in threaded hole to 11.4.3.3

Length of screws in threaded hole

\[
l_s = 36.00 \text{ mm}
\]

\[
R_p,\text{component} = 155.00 \text{ MPa}
\]

\[
R_p,\text{screw} = 360.00 \text{ MPa}
\]

\[
l_s = 29.73 \text{ mm}
\]

\[
l_s \geq l_s,\text{min} (36.00 \text{ mm} \geq 29.73 \text{ mm}): \text{ Ok}
\]

Internal pressure

Overpressure due to static head

\[
\nu = 0 \text{ MPa}
\]

Calculation pressure

\[
P_s = P_s + \nu = 4.00 \text{ MPa}
\]

Minimum thickness in assembly condition (center)

\[
\epsilon_A = \sqrt{\frac{3C - G}{\pi G}} W_f + \max(e, c_G) + ce + \delta
\]

\[
\epsilon_A = 24.00 \text{ mm}
\]

Minimum thickness in assembly condition (periphery)

\[
\epsilon_{A1} = \sqrt{\frac{3C - G}{\pi G}} W_f + \max(e, c_G) + ce + \delta
\]

\[
\epsilon_{A1} = 24.00 \text{ mm}
\]

Minimum thickness in operating condition (center)

\[
\epsilon_P = \sqrt{\frac{3C + v}{32}} G^2 + 3C \cdot \frac{G}{A} + 2b \cdot m(C - G)\frac{P}{f} + \max(e, c_G) + ce + \delta
\]

\[
\epsilon_P = 21.59 \text{ mm}
\]

Minimum thickness in operating condition (periphery)

\[
\epsilon_{P1} = \sqrt{3C \cdot \frac{G}{A} + 2b \cdot m(C - G)\frac{P}{f} + \max(e, c_G) + ce + \delta}
\]

Minimum required thickness (periphery)

\[
e_1 = \max(e_{A1}; e_{P1}) = 24.00 \text{ mm}
\]

\[
e_1 = \max(e_A; e_P) = 24.00 \text{ mm}
\]

\[
e_1 \geq e_1(\text{min}): \text{ Ok}
\]

\[
en \geq e: \text{ Ok}
\]

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure

\[
= 18.68 \text{ MPa}
\]

Maximum allowable design pressure

\[
= 4.00 \text{ MPa}
\]

Hydrostatic test

Item or side minimum allowables ratio

\[
\text{Item f0/f} = 1.32653
\]

Coincident design pressure for the maximum pressure load case

\[
P_d = 4.00 \text{ MPa}
\]

Test pressure as per EN13445-5 formula 10.2.3.1-1

\[
Pt_1 = 1.25 \cdot P_d + \nu = 6.63 \text{ MPa}
\]

Test pressure as per EN13445-5 formula 10.2.3.1-2

\[
Pt_2 = 1.43 \cdot P_d = 5.72 \text{ MPa}
\]

Item or side hydrostatic test pressure

\[
P_t = \max(Pt_1, Pt_2) = 6.63 \text{ MPa}
\]

Overpressure due to static head in test condition

\[
P_{\text{ht}} = 0 \text{ MPa}
\]

Calculation pressure

\[
P_c = P_t + P_{\text{ht}} = 6.63 \text{ MPa}
\]

Minimum thickness in operating condition (center)

\[
\epsilon_f = \sqrt{\frac{3C + v}{32}} G^2 + 3C \cdot \frac{G}{A} + 2b \cdot m(C - G)\frac{P}{f} + \max(e, c_G) + ce + \delta
\]

\[
\epsilon_f = 17.88 \text{ mm}
\]

Minimum thickness in operating condition (periphery)

\[
\epsilon_{P1} = \sqrt{3C \cdot \frac{G}{A} + 2b \cdot m(C - G)\frac{P}{f} + \max(e, c_G) + ce + \delta}
\]

Minimum required thickness

\[
e = e_P = 17.88 \text{ mm}
\]

\[
en \geq e: \text{ Ok}
\]
Welded flat cover - Welded flat cover #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 10

Design data

Internal design temperature

\[ Ti = 200.00 \, ^\circ C \]

Internal design pressure

\[ Pi = 4.00 \, MPa \]

Joint efficiency

\[ z = 0.85 \]

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature

\[ f = \max \left( \frac{R_{p0.2}}{\sigma_T}, \frac{R_{p0.2}}{2.4} \right) = 96.67 \, MPa \]

Nominal design stress at room temperature

\[ f = \min \left( \frac{R_{p0.2}}{\sigma_T}, \frac{R_{p0.2}}{2.4} \right) = 175.00 \, MPa \]

Nominal design stress in test condition

\[ f_{\text{test}} = \left( \frac{R_{p02}\text{Test}}{105} \right) = 285.71 \, MPa \]

Geometry

Outside diameter

\[ De = 25.00 \, mm \]

Nominal thickness

\[ en = 10.00 \, mm \]

Minimum head thickness after forming

\[ t-c' = 10.00 \, mm \]

Corrosion allowance

\[ e = 0 \, mm \]

External corrosion allowance

\[ ce = 0 \, mm \]

Undertolerance

\[ \delta = 0 \, mm \]

Type of flat end

= Welded directly to shell

Thickness of cylindrical part connected to the end

\[ es = 6.00 \, mm \]

Length of cylindrical part connected to the end

\[ le = 160.00 \, mm \]

Internal pressure

Overpressure due to static head

\[ Ph = 0 \, MPa \]

Calculation pressure

\[ P = Pi + Ph = 4.00 \, MPa \]

Length of cylindrical shell which contributes to the strength of the flat end

\[ l_{cyl} = 160.00 \, mm \]

Length of cylindrical shell which contributes to the strength of the flat end

\[ l_{cyl} = 10.68 \, mm \]

Parameter C1

\[ C_1 = \frac{0.40825\cdot A_1}{P_{\text{d}}} \left( \frac{D_i + e_i}{D_i} \right)^{1/2} = 0.53360 \]

Minimum required thickness

\[ e = C_1 \cdot \frac{P}{f} \]

\[ en \geq e: \text{Ok} \]

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure

\[ = 100.00 \, MPa \]

Maximum allowable design pressure

\[ = 100.00 \, MPa \]

Hydrostatic test

Item or side minimum allowable ratio

\[ f_{0f} = 1.32653 \]

Coincident design pressure for the maximum pressure load case

\[ P_d = 4.00 \, MPa \]

Test pressure as per EN13445-5 formula 10.2.3.3.1-1

\[ Pt1 = 1.25 \cdot P_d \cdot (\text{Item} \, f_0/f) = 6.63 \, MPa \]

Test pressure as per EN13445-5 formula 10.2.3.3.1-2

\[ Pt2 = 1.43 \cdot P_d = 5.72 \, MPa \]

Item or side hydrostatic test pressure

\[ P_{\text{h}} = \max (Pt1, Pt2) = 6.63 \, MPa \]

Overpressure due to static head in test condition

\[ Ph = 0 \, MPa \]

Calculation pressure

\[ P_c = P_{\text{h}} + Ph = 6.63 \, MPa \]

Length of cylindrical shell which contributes to the strength of the flat end

\[ l_{cyl} = 160.00 \, mm \]

Length of cylindrical shell which contributes to the strength of the flat end

\[ l_{cyl} = 10.68 \, mm \]

Parameter C1

\[ C_1 = \frac{0.40825\cdot A_1}{P_{\text{d}}} \left( \frac{D_i + e_i}{D_i} \right)^{1/2} = 0.53360 \]

Minimum required thickness

\[ e = C_1 \cdot \frac{P}{f} \]

\[ en \geq e: \text{Ok} \]
Conical shell - Conical shell #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8

Design data

Internal design temperature \( T_i = 200.00 \, ^\circ\text{C} \)
Internal design pressure \( P_i = 4.00 \, \text{MPa} \)
Joint efficiency \( z = 1.00 \)

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1.5 ≤ t ≤ 36)

Nominal design stress at internal design temperature
\[
f = \max \left( \frac{R_{p0.2}}{1.5}, \frac{R_{p0.20}}{2.4} \right) = 96.67 \, \text{MPa}
\]
Nominal design stress at room temperature
\[
f = \min \left( \frac{R_{p0.2}}{1.5}, \frac{R_{p0.20}}{2.4} \right) = 175.00 \, \text{MPa}
\]
Nominal design stress in test condition
\[
f_{\text{test}} = \frac{R_{p0.2}}{105} = 285.71 \, \text{MPa}
\]

Geometry

Length \( L = 24.81 \, \text{mm} \)
Nominal thickness \( e_{nL} = 6.00 \, \text{mm} \)
Corrosion allowance \( c = 0 \, \text{mm} \)
External corrosion allowance \( cc = 0 \, \text{mm} \)
Undertolerance \( \delta = 0 \, \text{mm} \)
Maximum Inside Diameter \( D_i = 115.00 \, \text{mm} \)
Maximum Outside Diameter \( D_e = 127.00 \, \text{mm} \)
Minimum Inside Diameter \( d_i = 25.00 \, \text{mm} \)
Minimum Outside Diameter \( d_e = 37.00 \, \text{mm} \)
Half-apex angle \( \alpha = 65.00 \, ^\circ \)
Thickness at large end \( e_{2L} = 6.00 \, \text{mm} \)
Thickness at small end \( e_{2s} = 6.00 \, \text{mm} \)
Nominal thickness of cylinder at large end \( e_{1L} = 6.00 \, \text{mm} \)
Minimum required thickness of cylinder at large end \( e_{1L} = 2.43 \, \text{mm} \)
Nominal thickness of cylinder at small end \( e_{1s} = 6.00 \, \text{mm} \)
Minimum required thickness of cylinder at small end \( e_{1s} = 0.53 \, \text{mm} \)

Internal pressure

Overpressure due to static head \( Ph = 0 \, \text{MPa} \)
Calculation pressure \( P = P_i + Ph = 4.00 \, \text{MPa} \)
Mean diameter of the cone at large end \( D_c = D_i + e_{1L} + c + cc + \delta = 121.00 \, \text{mm} \)
Minimum length along cone \( l_2L = \left( \frac{D_c \cdot e_{2L}}{\cos(\alpha)} \right) = 35.14 \, \text{mm} \)
Calculation diameter \( D_K = D_c - e_{2L} - 2r[1 - \cos(\alpha)] - l_2 \cdot \sin(\alpha) = 86.30 \, \text{mm} \)
Minimum required cone thickness \( e_{2L} + c + cc + \delta \) (iterative) \( = 4.31 \, \text{mm} \)

Large end junction (without knuckle)

Minimum length along cylinder \( 1.4 \cdot l_1L = 1.4 \cdot \sqrt{(D_c \cdot e_{1L})} = 24.00 \, \text{mm} \)
Minimum length along cone \( 1.4 \cdot l_2L = 1.4 \cdot \sqrt{(D_c \cdot e_{2L}) \cdot \cos(\alpha)} = 49.20 \, \text{mm} \)
\( \beta \) factor defined in 7.6.6 \( \beta = 7.66 = 1.46651 \)
Minimum required thickness at the junction at the large end of the cone \( e_{2L} = (P \cdot D_c \cdot \beta)/2f + c + cc + \delta = 3.67 \, \text{mm} \)
Maximum allowable pressure of junction at large end \( P_{\text{max}}(\text{large end}) = 2 \cdot f \cdot e_{2L}/(\beta \cdot D_c) = 6.54 \, \text{MPa} \)

Small end junction

Mean diameter of the cone \( dc = di + e_{1s} + c + cc + \delta = 31.00 \, \text{mm} \)
Minimum length along cylinder \( l_1s = \sqrt{(dc \cdot e_{1s})} = 13.64 \, \text{mm} \)
Minimum length along cone \( l_2s = \sqrt{(dc \cdot e_{2s} \cdot \cos(\alpha))} = 20.98 \, \text{mm} \)
\( s \) factor defined in 7.6.8 \( s = 2ns/e_{1ns} = 1.00000 \)
\( t \) factor defined in 7.6.8 \( t = 7.624/23 = 2.53825 \)
\( \beta H \) factor defined in 7.6.8 \( \beta H = 7.625 = 1.26817 \)
Minimum required thickness at the junction at the small end of the cone \( e_{2s} \) (iterative) \( = 0.008 \, \text{mm} \)
Maximum allowable pressure of junction at small end

\[ P_{\text{max(small end)}} = 2 \cdot f \cdot z \cdot e_1 / (d_{c} \cdot \beta_H) = 29.51 \text{ MPa} \]

\[ e_{2n}\geq e_2: \text{ Ok} \]
\[ e_n \geq e: \text{ Ok} \]

Maximum allowable pressures (at the top of the vessel)

- Maximum allowable test pressure = 18.29 MPa
- Maximum allowable design pressure = 5.99 MPa

Deformation according to EN13445-4 Clause 9

- Ratio of deformation \( F = 50 \cdot e_n / (d_i / 2 + e_n / 2) = 19.355 \% \)

Hydrostatic test

- Item or side minimum allowables ratio \( f_0 / f = 1.32653 \)
- Coincident design pressure for the maximum pressure load case \( P_d = 4.00 \text{ MPa} \)
- Test pressure as per EN13445-5 formula 10.2.3.3.1-1
  \[ P_{t1} = 1.25 \cdot P_d \cdot (f_0 / f) = 6.63 \text{ MPa} \]
  \[ P_{t2} = 1.43 \cdot P_d = 7.52 \text{ MPa} \]
- Item or side hydrostatic test pressure \( P_{t} = \text{max}(P_{t1}, P_{t2}) = 6.63 \text{ MPa} \)
- Overpressure due to static head in test condition \( P_{ht} = 0.002 \text{ MPa} \)
- Calculation pressure \( P_c = P_t + P_{ht} = 6.63 \text{ MPa} \)
- Maximum allowable pressure of junction at large end
  \[ P_{\text{max(large end)}} = 2 \cdot f \cdot z \cdot e_1 / (d_{c} \cdot \beta_H) = 15.75 \text{ MPa} \]

Large end junction (without knuckle)

- Minimum length along cylinder \( l_{1L} = 1.4 \cdot \sqrt{d_{c} \cdot e_{1L}} = 24.00 \text{ mm} \)
- Minimum length along cone \( l_{2L} = 1.4 \cdot \sqrt{(d_{c} \cdot e_{2L}) / \cos(\alpha)} = 38.17 \text{ mm} \)
- Minimum required thickness at the junction at the large end of the cone \( e_{2+\delta} (\text{iterative}) = 2.60 \text{ mm} \)
- \( e_{2nL} \geq e_j: \text{ Ok} \)

Small end junction

- Mean diameter of the cone \( d_{c} = d_{i} + e_{1l} + c + e + \delta = 31.00 \text{ mm} \)
- Minimum length along cylinder \( l_{1s} = 1.4 \cdot \sqrt{(d_{c} \cdot e_{1s})} = 13.64 \text{ mm} \)
- Minimum length along cone \( l_{2s} = 1.4 \cdot \sqrt{(d_{c} \cdot e_{2s}) / \cos(\alpha)} = 20.98 \text{ mm} \)
- \( s = 2\pi/e_{1ns} = 1.00000 \)
- \( r = 2\pi/e_{1ns} = 7.62423 = 2.53285 \)
- \( \beta_{H} = 7.62423 = 1.26817 \)
- Minimum required thickness at the junction at the small end of the cone \( e_{2s} (\text{iterative}) = 0.008 \text{ mm} \)
- Maximum allowable pressure of junction at small end

\[ P_{\text{max(small end)}} = 2 \cdot f \cdot z \cdot e_1 / (d_{c} \cdot \beta_H) = 87.21 \text{ MPa} \]
\[ e_{2n}\geq e_2: \text{ Ok} \]
\[ e_n \geq e: \text{ Ok} \]
Cylindrical shell - Cylindrical shell #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8

Design data
Internal design temperature \( T_i = 200.00 \, ^\circ\text{C} \)
Internal design pressure \( P_i = 4.00 \, \text{MPa} \)
Joint efficiency \( \zeta = 0.85 \)

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1.5 ≤ \( t \) ≤ 36)

Nominal design stress at internal design temperature
\[
f = \max \left( \frac{R_{p02/T}}{15}, \frac{R_{m20}}{2.4} \right) = 96.67 \, \text{MPa}
\]
Nominal design stress at room temperature
\[
f = \min \left( \frac{R_{p02/20}}{15}, \frac{R_{m20}}{2.4} \right) = 175.00 \, \text{MPa}
\]
Nominal design stress in test condition
\[
f_{\text{test}} = \left( \frac{R_{p02/Test}}{105} \right) = 285.71 \, \text{MPa}
\]

Geometry
Inside diameter \( D_i = 115.00 \, \text{mm} \)
Outside diameter \( D_e = 127.00 \, \text{mm} \)
Length \( L = 137.00 \, \text{mm} \)
Nominal thickness \( e_n = 6.00 \, \text{mm} \)
Corrosion allowance \( c = 0.00 \, \text{mm} \)
External corrosion allowance \( c_e = 0.00 \, \text{mm} \)
Undertolerance \( \delta = 0.00 \, \text{mm} \)

Internal pressure
Overpressure due to static head \( \Phi_h = 0 \, \text{MPa} \)
Calculation pressure \( P_c = P_i + \Phi_h = 4.00 \, \text{MPa} \)
Inside diameter \( D_i' = D_i + 2\delta + 2c = 115.00 \, \text{mm} \)
Minimum required thickness
\[
\frac{e}{D_e} \leq 0.16 \quad (0.02300 \leq 0.16000) : \text{Ok}
\]
\[
en \geq e : \text{Ok}
\]

Maximum allowable pressures (at the top of the vessel)
Maximum allowable test pressure
\[
= 28.33 \, \text{MPa}
\]
Maximum allowable design pressure
\[
= 8.15 \, \text{MPa}
\]

Deformation according to EN13445-4 Clause 9

Ratio of deformation
\[
F = 50 \cdot e_n / (D_i/2 + e_n/2) = 4.959 \, \%
\]

Hydrostatic test

Item or side minimum allowables ratio
\[
\text{Item f0/f} = 1.32653
\]
Coincident design pressure for the maximum pressure load case
\[
P_d = 4.00 \, \text{MPa}
\]
Test pressure as per EN13445-5 formula 10.2.3.3.1-1
\[
Pt_1 = 1.25 \cdot P_d \cdot (\text{Item f0/f}) = 6.63 \, \text{MPa}
\]
Test pressure as per EN13445-5 formula 10.2.3.3.1-2
\[
Pt_2 = 1.43 \cdot P_d = 5.72 \, \text{MPa}
\]
Item or side hydrostatic test pressure
\[
Pt = \max (Pt_1, Pt_2) = 6.63 \, \text{MPa}
\]
Overpressure due to static head in test condition
\[
P_h = 0.001 \, \text{MPa}
\]
Calculation pressure
\[
P_c = P_t + P_h = 6.63 \, \text{MPa}
\]
Inside diameter \( D_i' = D_i + 2\delta = 115.00 \, \text{mm} \)
Minimum required thickness
\[
\frac{P \cdot D_i'}{2f \cdot z - P} + \delta = 1.35 \, \text{mm}
\]
\[
\frac{e}{D_e} \leq 0.16 \quad (0.01100 \leq 0.16000) : \text{Ok}
\]
\[
en \geq e : \text{Ok}
\]
Loose lap flange - Loose lap flange #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11

<table>
<thead>
<tr>
<th>Allowable stresses</th>
<th>Flange - f</th>
<th>Hub - fH</th>
<th>Bolting - fB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design condition</td>
<td>130.67 MPa / 18 951.6 psi</td>
<td>96.67 MPa / 14 020.3 psi</td>
<td>120.00 MPa / 17 404.5 psi</td>
</tr>
<tr>
<td>Seating condition</td>
<td>173.33 MPa / 25 139.9 psi</td>
<td>175.00 MPa / 25 381.6 psi</td>
<td>175.00 MPa / 25 381.6 psi</td>
</tr>
<tr>
<td>Test condition</td>
<td>247.62 MPa / 35 914.1 psi</td>
<td>285.71 MPa / 41 439.4 psi</td>
<td>262.50 MPa / 38 072.4 psi</td>
</tr>
</tbody>
</table>

### Flange material
1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75)

### Stub flange (shell) material
Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

### Bolting material
A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

### Gasket
Rubber O-rings - below 75° IRH

#### Internal pressure
Pd = 4.00 MPa

#### Overpressure due to static head
Ph = 0 MPa

#### Calculation pressure
P = 4.00 MPa

#### Calculation temperature
T = 200.00 °C

### Geometry

**Corrosion allowance**
c = 0 mm

**Flange external diameter**
A = 240.00 mm

**Inside diameter**
B = 142.00 mm

e' = 22.00 mm

**Shell outside diameter**
B' = 127.00 mm

**Outside diameter of the contact face between loose and stub flanges**
A2 = 160.00 mm

**Inside diameter of the contact face between loose and stub flanges**
B2 = 142.00 mm

**Nominal gap between the shell and loose flange**
\[ \delta = \frac{(B - B')}{2} = 7.50 \text{ mm} \]

**Assumed diameter of load reaction between loose and stub flanges**
\[ G1 = \frac{(A2 + B2)}{2} = 151.00 \text{ mm} \]

**Area of the contact face**
\[ C = \frac{\pi}{2} \min \left( (A2 - \delta)^2 - G_1^2, G_1^2 - (B2 + \delta)^2 \right) = 708.0 \text{ mm}^2 \]

**Bolt circle**
C = 260.00 mm

**Flange thickness**
en = 32.00 mm

**Mean gasket diameter**
Gmean = 140.00 mm

**Hub length**
h = 4.00 mm

**Thickness of hub at back of flange**
g1 = 4.00 mm

g1* = 4.00 mm

**Thickness of hub at small end**
g0 = 6.00 mm

g0* = 6.00 mm

### Gasket parameters

**Gasket factor**
m = 0

**Minimum gasket seating pressure**
y = 0.70 MPa

**Gasket contact width**
w = 5.00 mm

**Basic gasket seating width**
b0 = w / 2 = 2.50 mm

**Effective gasket seating width**
b = b0 = 2.50 mm

**Diameter of gasket load reaction**
G = Gmean = 140.00 mm

### Bolt loads

**Number of bolts**
n = 8

**Bolt type**
= ISO M16 x 2.00

**Root area of one bolt**
= 144.0 mm²

**Distance between centre lines of adjacent bolts**
\[ \delta b = 102.10 \text{ mm} \]

**Bolt outside diameter**
db = 14.12 mm

**Total hydrostatic end force**
\[ H = \frac{G^2 \pi P}{4} = 61 575 \text{ N} \]

**Minimum required bolt load for operating condition**
\[ W_{op} = H = 61 575 \text{ N} \]

\[ H_1 = \frac{G^2 \pi P}{4} = 102 110 \text{ N} \]
Minimum required bolt load for the test condition

\[ W_t = H_t = 102\,110 \text{ N} \]

Minimum required bolt load for assembly condition

\[ W_A = 0 \text{ N} \]

Total required cross-sectional area of bolts

\[
A_{\text{Bmin}} = \max\left[ \frac{W_A}{f_{BA}}, \frac{W_B}{f_{BB}} \right]
\]

Total cross-sectional area of bolts at the section of least bolt diameter

\[ A_B = 1\,152.0 \text{ mm}^2 \]

Maximum bolts area for gasket crush

\[ W = 0.5(A_{\text{Bmin}} + A_B)f_{BA} = 145\,699 \text{ N} \]

Design bolt load for assembly condition

\[ \text{AB} \geq \text{AB}_{\text{min}}: \text{ Ok} \]

**Length of screws in threaded hole to 11.4.3.3**

Length of screws in threaded hole

\[
l_s = \max\left[ 0, 0.8d_h, \frac{R_{\text{D, component}}}{R_{\text{D, screw}}} ; 0, 0.8d_h \right]
\]

Flange constants

**Bolt pitch correction factor**

\[
C_p = \max\left[ \frac{\delta_b}{2d_h + \frac{60}{\pi}} , 1 \right]
\]

**Ratio of the flange diameters**

\[ K = A / B = 1.69014 \]

**Length parameter**

\[ l_s = \sqrt{B \cdot 90^*} = 29.19 \text{ mm} \]

**Hydrostatic end force applied via shell to flange**

\[ H_D = \frac{\pi}{4} (B^2 P) = 63\,347 \text{ N} \]

**Hydrostatic end force due to pressure on flange face**

\[ H_T = H - HD \]

**Radial distance from bolt circle to circle on which load reaction acts**

\[ h_L = (A - G1) / 2 = 44.50 \text{ mm} \]

**Radial distance from bolt circle to circle on which HD acts**

\[ h_D = (A - B) / 2 = 49.00 \text{ mm} \]

**Radial distance from bolt circle to circle on which HT acts**

\[ h_T = (2A - B - G) / 4 = 49.50 \text{ mm} \]

**Flange stress factor**

\[
\beta_F = \frac{K^2 \left( 1 + 8.55246 \log_{10} K \right) - 1}{(10472 + 19448K^2) (K - 1)} = 1.62942
\]

**Flange stress factor**

\[
\beta_V = \frac{136136(K^2 - 1) (K - 1)}{69845 + 5.7169 \left( K \log_{10} K \right)} = 3.87356
\]

**Flange stress factor - Stub flange**

**Bolt pitch correction factor**

\[
C_p = \max\left[ \frac{\delta_b}{2d_h + \frac{60}{\pi}} , 1 \right]
\]

**Ratio of the flange diameters**

\[ K = A / B^* = 1.39130 \]

**Length parameter**

\[ l_s = \sqrt{B \cdot 90^*} = 26.27 \text{ mm} \]

**Hydrostatic end force applied via shell to flange**

\[ H_D = \frac{\pi}{4} (B^2 P) = 41\,548 \text{ N} \]

**Hydrostatic end force due to pressure on flange face**

\[ H_T = H - HD \]

**Radial distance from bolt circle to circle on which load reaction acts**

\[ h_L = (C - B^* - g1^*) / 2 = 5.50 \text{ mm} \]

**Radial distance from bolt circle to circle on which HD acts**

\[ h_D = (C - G) / 2 = 5.50 \text{ mm} \]

**Radial distance from bolt circle to circle on which HT acts**

\[ h_T = (2C - B^* - G) / 4 = 11.75 \text{ mm} \]

**Flange stress factor**

\[
\beta_F = \frac{K^2 \left( 1 + 8.55246 \log_{10} K \right) - 1}{(10472 + 19448K^2) (K - 1)} = 1.75801
\]

**Flange stress factor**

\[
\beta_V = \frac{136136(K^2 - 1)(K - 1)}{69845 + 5.7169 \left( K \log_{10} K \right)} = 6.46058
\]

**Flange stress factor**

\[
\beta_F = \frac{1}{K - 1} \left[ 0.69845 + 5.7169 \left( K \log_{10} K \right) \right] = 6.04294
\]

\[
\beta_V = \frac{0.92859}{0.77762}
\]
Hub stress correction factor
\[ \lambda = \left( \frac{\phi + l_0}{\beta_1} + \frac{\phi^2}{\beta_1^2} \right) \]
\[ \phi = 1.00000 \]
\[ \lambda = 2.32978 \]

Flange moments
Total moment acting upon flange for assembly condition
\[ MA = W \cdot h_L = 6483.6 \text{ Nm} \]
Total moment acting upon flange for operating condition
\[ M_{op} = W_{op} \cdot h_L = 2740.1 \text{ Nm} \]
Moment factor used to design split rings
\[ F_x = 1.00 \]
Moment exerted on the flange per unit of length (operating)
\[ M = F_x \cdot M_{op} \cdot C_y \]
\[ M = 19.3 \text{ Nm} \]
Moment exerted on the flange per unit of length (assembly)
\[ M = F_x \cdot M_{A} \cdot C_y \]
\[ M = 45.7 \text{ Nm} \]

Flange moments - Stub flange
Total moment acting upon flange for assembly condition
\[ MA = W \cdot h_G = 801.3 \text{ Nm} \]
Total moment acting upon flange for operating condition
\[ M_{op} = |H_G \cdot h_G + H_{G} \cdot h_{G} + H_{G} \cdot h_{G}| = 900.1 \text{ Nm} \]
Moment factor used to design split rings
\[ F_x = 1.00 \]
Moment exerted on the flange per unit of length (operating)
\[ M = F_x \cdot M_{op} \cdot C_y \]
\[ M = 7.8 \text{ Nm} \]
Moment exerted on the flange per unit of length (assembly)
\[ M = F_x \cdot M_{A} \cdot C_y \]
\[ M = 7.0 \text{ Nm} \]

Flange stresses - operating condition
Longitudinal stress in hub
\[ \sigma_{H,0} = 0 \text{ MPa} \]
Radial stress in flange
\[ \sigma_{r,0} = 0 \text{ MPa} \]
Tangential stress in flange
\[ \sigma_r = \frac{\beta_r \cdot M}{\sigma_k^2} = 72.99 \text{ MPa} \]
Stress factor
\[ k = 1.00000 \]
\[ k \cdot \sigma_H \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]
\[ k \cdot \sigma_r \leq f: \text{ Ok} \]
\[ k \cdot \sigma_\theta \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok} \]
\[ \sigma_b \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]

Flange stresses - operating condition - Stub flange
Longitudinal stress in hub
\[ \sigma_H = \frac{\phi M}{\lambda q_k^2} = 209.97 \text{ MPa} \]
Radial stress in flange
\[ \sigma_r = \frac{(1333\phi \beta_r + l_0)M}{K \cdot \phi^2 l_0} = 14.14 \text{ MPa} \]
Tangential stress in flange
\[ \sigma_\theta = \frac{\beta_r \cdot M}{\sigma_k^2} - \sigma_r \cdot \frac{K^2 + 1}{K^2 - 1} = 53.37 \text{ MPa} \]
Stress factor
\[ k = 1.00000 \]
\[ k \cdot \sigma_H \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]
\[ k \cdot \sigma_r \leq f: \text{ Ok} \]
\[ k \cdot \sigma_\theta \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok} \]
\[ \sigma_b \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]

Flange stresses - seating condition
Longitudinal stress in hub
\[ \sigma_{H,A} = 0 \text{ MPa} \]
Radial stress in flange
\[ \sigma_{r,A} = 0 \text{ MPa} \]
Tangential stress in flange
\[ \sigma_\theta = \frac{\beta_r \cdot M}{\sigma_k^2} = 172.72 \text{ MPa} \]
\[ k \cdot \sigma_H \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]
\[ k \cdot \sigma_r \leq f: \text{ Ok} \]
\[ k \cdot \sigma_\theta \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok} \]
\[ 0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok} \]
\[ \sigma_b \leq 1.5 \text{ min(f;f_H)}: \text{ Ok} \]
Flange stresses - seating condition - Stub flange

Longitudinal stress in hub

$$\sigma_H = \frac{6M}{d^2} = 186.93 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = \frac{1333\epsilon \beta_r + h_0^2}{K^2} \frac{M}{h_0^2} = 12.59 \text{ MPa}$$

Tangential stress in flange

$$\sigma_\theta = \frac{1}{d^2} \left( \beta_r M - \sigma_r \frac{K^2 + 1}{K^2 - 1} \right) = 47.51 \text{ MPa}$$

$$k \sigma_H \leq 1.5 \min(f; f_H): \text{ Ok}$$

$$k \sigma_r \leq f: \text{ Ok}$$

$$k \sigma_\theta \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_r) \leq f: \text{ Ok}$$

$$0.5k(\sigma_H + \sigma_\theta) \leq f: \text{ Ok}$$

Maximum allowable pressures (at the top of the vessel)

New & cold (flange) = 5.00 MPa

Hot & corroded (flange) = 4.26 MPa

New & cold (bolts) = 13.10 MPa

Hot & corroded (bolts) = 8.98 MPa

Hydrostatic test

Item or side hydrostatic test pressure

$$P_t = 6.63 \text{ MPa}$$

Overpressure due to static head

$$P_h = 0.0005 \text{ MPa}$$

Calculation pressure

$$P = P_t + P_h = 6.63 \text{ MPa}$$

Flange constants

Bolt pitch correction factor

$$C_p = \max \left[ \frac{\delta_t}{2d_o + \delta_t}, 1 \right] = 1.00000$$

Ratio of the flange diameters

$$K = A / B = 1.69014$$

Length parameter

$$l_0 = \sqrt{B_0} P_t = 29.19 \text{ mm}$$

Hydrostatic end force applied via shell to flange

$$H_D = \frac{P_t}{4} \left( B^2 P \right) = 105.048 \text{ N}$$

Hydrostatic end force due to pressure on flange face

$$H_T = H - HD = -2938 \text{ N}$$

Radial distance from bolt circle to circle on which load reaction acts

$$h_L = (A - G1) / 2 = 44.50 \text{ mm}$$

Radial distance from bolt circle to circle on which HD acts

$$h_D = (A - B) / 2 = 49.00 \text{ mm}$$

Radial distance from gasket load reaction to bolt circle

$$h_G = (A - G) / 2 = 50.00 \text{ mm}$$

Radial distance from bolt circle to circle on which HT acts

$$h_T = (2A - B - G) / 4 = 49.50 \text{ mm}$$

Flange stress factor

$$\beta_T = \frac{K^2 (1 + 85524\log_{10} K) - 1}{(10472 + 19448K^2) (K - 1)} = 1.62942$$

Flange stress factor

$$\beta_Y = \frac{K^2 (1 + 85524\log_{10} K) - 1}{136136 (K^2 - 1) (K - 1)} = 4.25664$$

Flange stress factor

$$\beta_Y = \frac{K^2 (1 + 85524\log_{10} K) - 1}{0.66845 + 5.7169 \left( K^3 \log_{10} K \right)} = 3.87356$$

Flange moments

Total moment acting upon flange for operating condition

$$M_{op} = W_{op} h_L = 2740.1 \text{ N} \cdot \text{m}$$

Moment factor used to design split rings

$$F_X = 1.00$$

Moment exerted on the flange per unit of length (operating)

$$M = F_X M_{op} C_p = 19.3 \text{ N} \cdot \text{m}$$

Flange moments - Stub flange

Total moment acting upon flange for operating condition

$$M_{op} = |H_D h_D + H_T h_T + H_G h_G| = 1492.5 \text{ N} \cdot \text{m}$$

Moment factor used to design split rings

$$F_X = 1.00$$

Moment exerted on the flange per unit of length (operating)

$$M = F_X M_{op} C_p = 13.0 \text{ N} \cdot \text{m}$$

Flange stresses - operating condition

Longitudinal stress in hub

$$\sigma_H = 0 \text{ MPa}$$

Radial stress in flange

$$\sigma_r = 0 \text{ MPa}$$
Tangential stress in flange

\[ \sigma_\theta = \frac{\beta \cdot M}{r^2} = 72.99 \text{ MPa} \]

Stress factor

\[ k = 1.00000 \]

\[ k \cdot \sigma_H \leq 1.5 \min(f; f_H) : \text{Ok} \]
\[ k \cdot \sigma_r \leq f : \text{Ok} \]
\[ k \cdot \sigma_\theta \leq f : \text{Ok} \]
\[ 0.5k(\sigma_H + \sigma_r) \leq f : \text{Ok} \]
\[ 0.5k(\sigma_H + \sigma_\theta) \leq f : \text{Ok} \]
\[ \sigma_b \leq 1.5 \min(f; f_H) : \text{Ok} \]

Flange stresses - operating condition - Stub flange

Longitudinal stress in hub

\[ \sigma_H = \frac{\varphi M}{\lambda r^2} = 348.16 \text{ MPa} \]

Radial stress in flange

\[ \sigma_r = \frac{1333 \varphi \beta_r + \beta_l}{k \varphi r^2 \lambda} M = 23.44 \text{ MPa} \]

Tangential stress in flange

\[ \sigma_\theta = \frac{\beta \cdot M}{r^2} - \frac{k^2 + 1}{K^2} \sigma_r = 88.49 \text{ MPa} \]

Stress factor

\[ k = 1.00000 \]

\[ k \cdot \sigma_H \leq 1.5 \min(f; f_H) : \text{Ok} \]
\[ k \cdot \sigma_r \leq f : \text{Ok} \]
\[ k \cdot \sigma_\theta \leq f : \text{Ok} \]
\[ 0.5k(\sigma_H + \sigma_r) \leq f : \text{Ok} \]
\[ 0.5k(\sigma_H + \sigma_\theta) \leq f : \text{Ok} \]
Bolted flat cover - Bolted flat cover #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 10

Design data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal design temperature</td>
<td>Ti = 200.00 °C</td>
</tr>
<tr>
<td>Internal design pressure</td>
<td>Pi = 4.00 MPa</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>z = 1.00</td>
</tr>
</tbody>
</table>

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature

\[ f = \min \left( \frac{R_{\text{p02/T}}}{1.5}, \frac{R_{\text{m20}}}{2.4} \right) = 96.67 \text{ MPa} \]

Nominal design stress at room temperature

\[ f = \min \left( \frac{R_{\text{p02/20}}}{1.5}, \frac{R_{\text{m20}}}{2.4} \right) = 175.00 \text{ MPa} \]

Nominal design stress in test condition

\[ f_{\text{test}} = \frac{R_{\text{p02/Test}}}{105} = 285.71 \text{ MPa} \]

Bolts Material: A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

Nominal design stress at design temperature

\[ f = \min \left( \frac{R_{\text{p02/T}}}{3}, \frac{R_{\text{m20}}}{4} \right) = 120.00 \text{ MPa} \]

Nominal design stress at room temperature

\[ f = \frac{R_{\text{p02/T}}}{4} = 175.00 \text{ MPa} \]

Nominal design stress in test condition

\[ f = 1.5 \cdot \frac{R_{\text{m20}}}{4} = 262.50 \text{ MPa} \]

Geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal thickness</td>
<td>( e_n = 30.00 \text{ mm} )</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>( e = 0 \text{ mm} )</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>( e_{ce} = 0 \text{ mm} )</td>
</tr>
<tr>
<td>Undertolerance</td>
<td>( \delta = 0 \text{ mm} )</td>
</tr>
<tr>
<td>Bolt circle</td>
<td>( C = 260.00 \text{ mm} )</td>
</tr>
<tr>
<td>Mean gasket diameter</td>
<td>( \text{Gmean} = 140.00 \text{ mm} )</td>
</tr>
<tr>
<td>Adopted thickness (periphery)</td>
<td>( e_l = 27.00 \text{ mm} )</td>
</tr>
<tr>
<td>Gasket groove depth</td>
<td>( g = 0 \text{ mm} )</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>( c^* = 0 \text{ mm} )</td>
</tr>
<tr>
<td>Flange external diameter</td>
<td>( A = 240.00 \text{ mm} )</td>
</tr>
<tr>
<td>Inside diameter of connected flange</td>
<td>( B = 142.00 \text{ mm} )</td>
</tr>
<tr>
<td>Inside diameter of connected flange (corroded)</td>
<td>( B^* = B + 2c = 142.00 \text{ mm} )</td>
</tr>
</tbody>
</table>

Gasket parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasket factor</td>
<td>( m = 0 )</td>
</tr>
<tr>
<td>Minimum gasket seating pressure</td>
<td>( y = 0.70 \text{ MPa} )</td>
</tr>
<tr>
<td>Gasket contact width</td>
<td>( w = 5.00 \text{ mm} )</td>
</tr>
<tr>
<td>Basic gasket seating width</td>
<td>( b_0 = \frac{w}{2} = 2.50 \text{ mm} )</td>
</tr>
<tr>
<td>Effective gasket seating width</td>
<td>( b = b_0 = 2.50 \text{ mm} )</td>
</tr>
<tr>
<td>Diameter of gasket load reaction</td>
<td>( G = \text{Gmean} = 140.00 \text{ mm} )</td>
</tr>
</tbody>
</table>

Bolt loads

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of bolts</td>
<td>( = 8 )</td>
</tr>
<tr>
<td>Bolt type</td>
<td>ISO M16 x 2.00</td>
</tr>
<tr>
<td>Bolt hole diameter</td>
<td>( = 18.00 \text{ mm} )</td>
</tr>
<tr>
<td>Bolt spacing</td>
<td>( t_{\text{bmax}} = 102.10 \text{ mm} )</td>
</tr>
<tr>
<td>Root area of one bolt</td>
<td>( = 144.0 \text{ mm}^2 )</td>
</tr>
<tr>
<td>Distance between centre lines of adjacent bolts</td>
<td>( \delta_b = 102.10 \text{ mm} )</td>
</tr>
<tr>
<td>Total hydrostatic end force</td>
<td>( H = \frac{G^2 r d P^4}{4} = 61 \text{ 575 N} )</td>
</tr>
<tr>
<td>Minimum required bolt load for operating condition</td>
<td>( W_{\text{op}} = H = 61 \text{ 575 N} )</td>
</tr>
<tr>
<td>Minimum required bolt load for the test condition (Loose lap flange #1)</td>
<td>( W_{t} = H_{1} = \frac{G^2 r d P^4}{4} = 102 \text{ 102 N} )</td>
</tr>
<tr>
<td>Minimum required bolt load for assembly condition</td>
<td>( W_{A} = 0 \text{ N} )</td>
</tr>
</tbody>
</table>
Total required cross-sectional area of bolts

\[
A_{\text{min}} \geq \max \left[ \frac{W_A}{f_{EA}}, \frac{W_B}{f_B}, \frac{W}{f_{EA}} \right] = 513.1 \text{ mm}^2
\]

Total cross-sectional area of bolts at the section of least bolt diameter

\[
A_{\text{max}} = \frac{2\pi \cdot y \cdot G \cdot w}{f_{EA}} = 17.6 \text{ mm}^2
\]

Maximum bolts area for gasket crush

\[
W = 0.5 \left( A_{\text{min}} + A_B \right) f_{EA} = 145,699 \text{ N}
\]

Design bolt load for assembly condition

\[
AB \geq A_{\text{min}}: \text{ Ok}
\]

**Internal pressure**

Overpressure due to static head

\[
\text{Ph} = 0 \text{ MPa}
\]

Calculation pressure

\[
P = P_i + \text{Ph} = 4.00 \text{ MPa}
\]

Minimum thickness in assembly condition (center)

\[
e_A = \sqrt{\frac{3C - G}{32} + 3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 26.10 \text{ mm}
\]

Minimum thickness in assembly condition (periphery)

\[
e_A = \sqrt{\frac{3C - G}{32} + 3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 26.10 \text{ mm}
\]

Minimum thickness in operating condition (center)

\[
e_P = \sqrt{\frac{3(C + G)}{32} - 3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 27.79 \text{ mm}
\]

Minimum thickness in operating condition (periphery)

\[
e_P = \sqrt{3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 22.83 \text{ mm}
\]

Minimum required thickness (periphery)

\[
e_{\text{min}} = \max(e_A, e_P) = 26.10 \text{ mm}
\]

Minimum required thickness

\[
e = \max(e_A, e_P) = 27.79 \text{ mm}
\]

**Maximum allowable pressures (at the top of the vessel)**

Maximum allowable test pressure

\[
P_t = 13.10 \text{ MPa}
\]

Maximum allowable design pressure

\[
P_d = 4.66 \text{ MPa}
\]

**Hydrostatic test**

Item or side minimum allowables ratio

\[
I_{f0/f} = 1.32653
\]

Coincident design pressure for the maximum pressure load case

\[
P_d = 4.00 \text{ MPa}
\]

Test pressure as per EN13445-5 formula 10.2.3.1.1-1

\[
P_{1} = 1.25 \cdot P_d (\text{Item } f0/f) = 6.63 \text{ MPa}
\]

Test pressure as per EN13445-5 formula 10.2.3.1.2-1

\[
P_{2} = 1.43 \cdot P_d = 5.72 \text{ MPa}
\]

Item or side hydrostatic test pressure

\[
P_{\text{ht}} = \max(P_1, P_2) = 6.63 \text{ MPa}
\]

Overpressure due to static head in test condition

\[
P_{\text{ht}} = 0.00003 \text{ MPa}
\]

Calculation pressure

\[
P_{\text{calc}} = P_d + P_{\text{ht}} = 6.63 \text{ MPa}
\]

Minimum thickness in operating condition (center)

\[
e_P = \sqrt{\frac{3(C + G)}{32} - 3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 20.82 \text{ mm}
\]

Minimum thickness in operating condition (periphery)

\[
e_P = \sqrt{3C \cdot \frac{G}{A} + 2h \cdot m} (C - G)^2 + \max(c, g) + ce + \delta = 17.10 \text{ mm}
\]

Minimum required thickness

\[
e = e_P = 20.82 \text{ mm}
\]

\[en \geq e: \text{ Ok}\]