### Calculation report

**Asme VIII Div. 2 Ed. 2015 - Metric Units**

<table>
<thead>
<tr>
<th></th>
<th>Tube side</th>
<th>Shell side</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal design pressure</td>
<td>18.44 MPa</td>
<td>14.22 MPa</td>
</tr>
<tr>
<td>Internal design temperature</td>
<td>454.00 °C</td>
<td>420.00 °C</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>0 mm</td>
<td>3.00 mm</td>
</tr>
<tr>
<td>Vacuum?</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Minimum design temperature</td>
<td>-4.00 °C</td>
<td>24.80 °F</td>
</tr>
</tbody>
</table>
# Table of contents

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Table of contents</td>
<td>2</td>
</tr>
<tr>
<td>Test Pressure - Tube side (MPa)</td>
<td>3</td>
</tr>
<tr>
<td>Maximum Pressures - Tube side (MPa)</td>
<td>3</td>
</tr>
<tr>
<td>Test Pressure - Shell side (MPa)</td>
<td>4</td>
</tr>
<tr>
<td>Maximum Pressures - Shell side (MPa)</td>
<td>4</td>
</tr>
<tr>
<td>Weights</td>
<td>5</td>
</tr>
<tr>
<td>Bill of materials</td>
<td>6</td>
</tr>
<tr>
<td>Nozzle connections</td>
<td>7</td>
</tr>
<tr>
<td>Nozzle positions</td>
<td>7</td>
</tr>
<tr>
<td>Nozzle welds</td>
<td>7</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>8</td>
</tr>
<tr>
<td>Welded flat cover - Flat Head</td>
<td>9</td>
</tr>
<tr>
<td>Cylindrical shell - Left channel</td>
<td>11</td>
</tr>
<tr>
<td>Reinforcement of opening - Bocchello In Tube Side</td>
<td>14</td>
</tr>
<tr>
<td>Reinforcement of opening - Bocchello Out Tube Side</td>
<td>19</td>
</tr>
<tr>
<td>Nozzle - Bocchello In Tube Side</td>
<td>24</td>
</tr>
<tr>
<td>Welding neck flange - Flange In Tube Side</td>
<td>26</td>
</tr>
<tr>
<td>Nozzle - Bocchello Out Tube Side</td>
<td>34</td>
</tr>
<tr>
<td>Welding neck flange - Flange Out Tube Side</td>
<td>36</td>
</tr>
<tr>
<td>Welding neck flange - Shell Flange</td>
<td>44</td>
</tr>
<tr>
<td>Cylindrical shell - Main shell</td>
<td>52</td>
</tr>
<tr>
<td>Reinforcement of opening - Out Shell Side</td>
<td>55</td>
</tr>
<tr>
<td>Reinforcement of opening - In Shell Side</td>
<td>59</td>
</tr>
<tr>
<td>Standard Long Welding Neck flange - Out Shell Side</td>
<td>63</td>
</tr>
<tr>
<td>Standard Long Welding Neck flange - In Shell Side</td>
<td>66</td>
</tr>
<tr>
<td>U-Tube tubesheet - Tubesheet</td>
<td>69</td>
</tr>
<tr>
<td>Tube bundle - Tubes bundle</td>
<td>79</td>
</tr>
<tr>
<td>Hemispherical head - Head</td>
<td>82</td>
</tr>
</tbody>
</table>
## Test Pressure - Tube side (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>P</th>
<th>Static head (design)</th>
<th>Static head (test)</th>
<th>MAP N&amp;C</th>
<th>MAWP H&amp;C</th>
<th>Stress ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>25.91</td>
<td>18.75</td>
<td>1,381</td>
</tr>
<tr>
<td>Left channel</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>25.91</td>
<td>18.75</td>
<td>1,381</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.01</td>
<td>25.61</td>
<td>18.54</td>
<td>1,381</td>
</tr>
<tr>
<td>Flange In Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.002</td>
<td>20.77</td>
<td>19.00</td>
<td>1,381</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.03</td>
<td>25.61</td>
<td>18.54</td>
<td>1,381</td>
</tr>
<tr>
<td>Flange Out Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.03</td>
<td>20.77</td>
<td>19.00</td>
<td>1,381</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>31.26</td>
<td>22.60</td>
<td>1,381</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>18.54</td>
<td>0</td>
<td>0.02</td>
<td>41.99</td>
<td>41.99</td>
<td>1</td>
</tr>
</tbody>
</table>

All pressures in MPa

### Tubes side design pressure P = 18.54 MPa

*Design pressure used for test pressure calculation increased due to vacuum*

### Tubes side MAWP (Hot & Corroded conditions) = 18.54 MPa (limited by Bocchello In Tube Side)

### Tubes side MAP (New & Cold conditions) = 20.77 MPa (limited by Flange In Tube Side)

### Tubes side Lowest Stress Ratio = 1.000

### Tubes side test pressure = Pt = 1.43 * MAWP = 26.51 MPa

## Maximum Pressures - Tube side (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>MAP N&amp;C</th>
<th>MAWP H&amp;C</th>
<th>MAEP N&amp;C</th>
<th>MAEWP H&amp;C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>25.91</td>
<td>18.75</td>
<td>18.75</td>
<td>18.75</td>
</tr>
<tr>
<td>Left channel</td>
<td>25.91</td>
<td>18.75</td>
<td>21.88</td>
<td>21.88</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>25.61</td>
<td>18.54</td>
<td>23.09</td>
<td>23.09</td>
</tr>
<tr>
<td>Flange In Tube Side</td>
<td>20.77</td>
<td>19.00</td>
<td>19.00</td>
<td>19.00</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>25.61</td>
<td>18.54</td>
<td>23.09</td>
<td>23.09</td>
</tr>
<tr>
<td>Flange Out Tube Side</td>
<td>20.77</td>
<td>19.00</td>
<td>19.00</td>
<td>19.00</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>31.26</td>
<td>22.60</td>
<td>23.69</td>
<td>19.90</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>41.99</td>
<td>41.99</td>
<td>23.69</td>
<td>19.90</td>
</tr>
</tbody>
</table>

All pressures in MPa
### Test Pressure - Shell side (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>P</th>
<th>Static head (design)</th>
<th>Static head (test)</th>
<th>MAP N&amp;C</th>
<th>MAWP H&amp;C</th>
<th>Stress ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell Flange</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>22.47</td>
<td>21.58</td>
<td>1,305</td>
</tr>
<tr>
<td>Main shell</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>20.10</td>
<td>14.65</td>
<td>1,305</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>14.22</td>
<td>0</td>
<td>0.03</td>
<td>25.86</td>
<td>17.67</td>
<td>1,305</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>14.22</td>
<td>0</td>
<td>0.01</td>
<td>25.86</td>
<td>17.67</td>
<td>1,305</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>19.04</td>
<td>14.22</td>
<td>1,381</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>14.32</td>
<td>0</td>
<td>0.01</td>
<td>23.69</td>
<td>19.90</td>
<td>1</td>
</tr>
<tr>
<td>Head</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>21.01</td>
<td>14.65</td>
<td>1,305</td>
</tr>
</tbody>
</table>

All pressures in MPa

**Shell side design pressure P = 14.32 MPa**

*Design pressure used for test pressure calculation increased due to vacuum*

**Shell side MAWP (Hot & Corroded conditions) = 14.22 MPa (limited by Tubesheet)**

**Shell side MAP (New & Cold conditions) = 19.04 MPa (limited by Tubesheet)**

**Shell side Lowest Stress Ratio = 1.000**

**Shell side test pressure = Pt = 1.43 * MAWP = 20.34 MPa**

### Maximum Pressures - Shell side (MPa)

<table>
<thead>
<tr>
<th>Component</th>
<th>MAP N&amp;C</th>
<th>MAWP H&amp;C</th>
<th>MAEP N&amp;C</th>
<th>MAEWP H&amp;C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell Flange</td>
<td>22.47</td>
<td>21.58</td>
<td>21.58</td>
<td>21.58</td>
</tr>
<tr>
<td>Main shell</td>
<td>20.10</td>
<td>14.65</td>
<td>17.21</td>
<td>16.41</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>25.86</td>
<td>17.67</td>
<td>17.67</td>
<td>17.67</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>25.86</td>
<td>17.67</td>
<td>17.67</td>
<td>17.67</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>19.04</td>
<td>14.22</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>23.69</td>
<td>19.90</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Head</td>
<td>21.01</td>
<td>14.65</td>
<td>14.52</td>
<td>12.88</td>
</tr>
</tbody>
</table>

All pressures in MPa
<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>Flat Head</td>
<td>3 559 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>3 559 kg</td>
<td>3 559 kg</td>
</tr>
<tr>
<td>Left channel</td>
<td>3 530 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>5 445 kg</td>
<td>3 530 kg</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>937 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 183 kg</td>
<td>937 kg</td>
</tr>
<tr>
<td>Flange In Tube Side</td>
<td>635 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>710 kg</td>
<td>635 kg</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>937 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 183 kg</td>
<td>937 kg</td>
</tr>
<tr>
<td>Flange Out Tube Side</td>
<td>635 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>710 kg</td>
<td>635 kg</td>
</tr>
<tr>
<td>Shell Flange</td>
<td>3 231 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>3 769 kg</td>
<td>3 231 kg</td>
</tr>
<tr>
<td>Main shell</td>
<td>7 696 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>10 519 kg</td>
<td>7 696 kg</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>1 437 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 464 kg</td>
<td>1 437 kg</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>1 437 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 464 kg</td>
<td>1 437 kg</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>3 674 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>3 674 kg</td>
<td>3 674 kg</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>15 229 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>17 027 kg</td>
<td>15 229 kg</td>
</tr>
<tr>
<td>Head</td>
<td>744 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1 327 kg</td>
<td>744 kg</td>
</tr>
<tr>
<td>Totals:</td>
<td>43 681 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>52 035 kg</td>
<td>43 681 kg</td>
</tr>
</tbody>
</table>

Total shell side volume: 3.99688 m³
Total tube side volume: 4.35666 m³
Total volume: 8.35355 m³
<table>
<thead>
<tr>
<th>Component</th>
<th>Dimensions</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>Id = 1 275.00 mm, Od = 1 445.00 mm, Tk = 280.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Left channel</td>
<td>Id = 1 275.00 mm, Od = 1 445.00 mm, Tk = 85.00 mm, L = 1 500.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>Id = 635.00 mm, Od = 725.00 mm, Tk = 45.00 mm, L = 778.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange In Tube Side - Flange</td>
<td>Id = 635.00 mm, Od = 1 026.00 mm, Tk = 170.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange In Tube Side - Gasket</td>
<td></td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
</tr>
<tr>
<td>Flange In Tube Side - Bolts</td>
<td>20 x ANSI_TEMA 2-1/2&quot;</td>
<td>SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>Id = 635.00 mm, Od = 725.00 mm, Tk = 45.00 mm, L = 778.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange Out Tube Side - Flange</td>
<td>Id = 635.00 mm, Od = 1 026.00 mm, Tk = 170.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange Out Tube Side - Gasket</td>
<td></td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
</tr>
<tr>
<td>Flange Out Tube Side - Bolts</td>
<td>20 x ANSI_TEMA 2-1/2&quot;</td>
<td>SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting</td>
</tr>
<tr>
<td>Shell Flange - Flange</td>
<td>Id = 1 277.00 mm, Od = 1 900.00 mm, Tk = 300.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Shell Flange - Gasket</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shell Flange - Bolts</td>
<td>28 x ANSI_TEMA 4-1/2&quot;</td>
<td>SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting</td>
</tr>
<tr>
<td>Main shell</td>
<td>Id = 1 275.00 mm, Od = 1 405.00 mm, Tk = 65.00 mm, L = 3 629.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Out Shell Side - Flange</td>
<td>Id = 457.20 mm, Od = 915.00 mm, Tk = 69.90 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Out Shell Side - Gasket</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Out Shell Side - Bolts</td>
<td>16 x ANSI_TEMA 2-1/2&quot;</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>In Shell Side - Flange</td>
<td>Id = 457.20 mm, Od = 915.00 mm, Tk = 69.90 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>In Shell Side - Gasket</td>
<td></td>
<td></td>
</tr>
<tr>
<td>In Shell Side - Bolts</td>
<td>16 x ANSI_TEMA 2-1/2&quot;</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>Tubesheet - Flange</td>
<td>Od = 1 900.00 mm, Tk = 230.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>Id = 23.37 mm, Od = 31.75 mm, Tk = 4.19 mm, L = 3 658.00 mm</td>
<td>SB-517 Cold drawn/ann. N06600 - Pipe / tube</td>
</tr>
<tr>
<td>Head</td>
<td>Id = 1 306.00 mm, Od = 1 374.00 mm, Tk = 34.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
</tbody>
</table>
### Nozzle connections

<table>
<thead>
<tr>
<th>Name</th>
<th>Flange</th>
<th>Material</th>
<th>OD</th>
<th>Tk</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bocchello In Tube Side</td>
<td>WN NotStandard</td>
<td>SA-182 F22 3</td>
<td>725.00 mm</td>
<td>45.00 mm</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>WN NotStandard</td>
<td>SA-182 F22 3</td>
<td>725.00 mm</td>
<td>45.00 mm</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>18” LWN 1500 ANSI</td>
<td>SA-182 F22 3</td>
<td>597.00 mm</td>
<td>69.90 mm</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>18” LWN 1500 ANSI</td>
<td>SA-182 F22 3</td>
<td>597.00 mm</td>
<td>69.90 mm</td>
</tr>
</tbody>
</table>

### Nozzle positions

<table>
<thead>
<tr>
<th>Name</th>
<th>Placed on</th>
<th>Type</th>
<th>Distance from reference</th>
<th>Orientation</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bocchello In Tube Side</td>
<td>Left channel</td>
<td>Radial/ Reinforced</td>
<td>700.00 mm</td>
<td>0 °</td>
<td></td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>Left channel</td>
<td>Radial/ Reinforced</td>
<td>700.00 mm</td>
<td>180.00 °</td>
<td></td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>Main shell</td>
<td>Radial/ Reinforced</td>
<td>3150.00 mm</td>
<td>180.00 °</td>
<td></td>
</tr>
<tr>
<td>In Shell Side</td>
<td>Main shell</td>
<td>Radial/ Reinforced</td>
<td>420.00 mm</td>
<td>0 °</td>
<td></td>
</tr>
</tbody>
</table>

### Nozzle welds

<table>
<thead>
<tr>
<th>Name</th>
<th>Nozzle to wall</th>
<th>Pad to wall</th>
<th>Shell groove</th>
<th>Pad groove</th>
<th>Inside</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bocchello In Tube Side</td>
<td>15.00 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>15.00 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>15.00 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>In Shell Side</td>
<td>15.00 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Minimum Design Metal Temperature (MDMT)

<table>
<thead>
<tr>
<th>Component</th>
<th>MDMT</th>
<th>Tmin &gt; MDMT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Left channel</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Flange In Tube Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Flange In Tube Side (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Flange Out Tube Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Flange Out Tube Side (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Shell Flange</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Shell Flange (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Main shell</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Out Shell Side (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>In Shell Side (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>Impact tests required</td>
<td></td>
</tr>
<tr>
<td>Tubesheet (bolting)</td>
<td>-29.00 °C / -20.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>-104.00 °C / -155.20 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Head</td>
<td>39.50 °C / 103.10 °F</td>
<td>No</td>
</tr>
</tbody>
</table>

**Item minimum design temperature Tmin: -4.00 °C / 24.80 °F**

Impact tests required by Code

One or more components have a MDMT higher than item minimum design temperature.
Calculation temperature
\[ T = 454.00 \, ^\circ C = 849.20 \, ^\circ F \]

Material: SA-387 22 2 - Plate

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allowable stress</td>
<td>149.84 MPa</td>
</tr>
<tr>
<td>Allowable stress at room temp.</td>
<td>207.00 MPa</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>1275.00 mm</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>1.00</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>0 mm</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>0 mm</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>0 mm</td>
</tr>
<tr>
<td>Factor C</td>
<td>C = 0.30000</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>1445.00 mm</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>280.00 mm</td>
</tr>
<tr>
<td>Head type</td>
<td>Type9</td>
</tr>
</tbody>
</table>

Internal pressure

\[ t_r = d \sqrt{\frac{CP}{SE}} + c + c_r + c' = 277.63 \, mm = 10.930 \, in \]

Maximum allowable pressures (at the top of the vessel)

- New & cold: 25.91 MPa = 3757.6 psi
- Hot & corroded: 18.75 MPa = 2720.0 psi

External pressure

\[ t_r = d \sqrt{\frac{CP}{SE}} + c + c_r + c' = 17.66 \, mm = 0.695 \, in \]

Maximum allowable external pressures

- New & cold: 18.75 MPa = 2720.0 psi
- Hot & corroded: 18.75 MPa = 2720.0 psi
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure

**Welded flat cover - Flat Head**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-387 22 2</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>( tg = 85.00 \text{ mm} = 3.346 \text{ in} )</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

#### External pressure

**Welded flat cover - Flat Head**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-387 22 2</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>( tg = 85.00 \text{ mm} = 3.346 \text{ in} )</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*
Cylindrical shell - Left channel

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

<table>
<thead>
<tr>
<th>Calculation temperature</th>
<th>$T =$</th>
<th>454.00 °C</th>
<th>849.20 °F</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Material:</strong></td>
<td><strong>SA-387 22 2 - Plate</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Allowable stress at room temperature</td>
<td>$ST =$</td>
<td>207.00 MPa</td>
<td>30 022.8 psi</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>$E =$</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>$c =$</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>$ce =$</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>$c'$</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>$D =$</td>
<td>1 275.00 mm</td>
<td>50.197 in</td>
</tr>
<tr>
<td>Length</td>
<td>$L =$</td>
<td>1 500.00 mm</td>
<td>59.055 in</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>$t =$</td>
<td>85.00 mm</td>
<td>3.346 in</td>
</tr>
</tbody>
</table>

**Joint efficiency**

$E = 1.00$

| Corrosion allowance | $c =$ | 0 mm | 0 in |
| External corrosion allowance | $ce =$ | 0 mm | 0 in |
| Wall undertolerance | $c'$ | 0 mm | 0 in |

| Inside diameter | $D =$ | 1 275.00 mm | 50.197 in |
| Length | $L =$ | 1 500.00 mm | 59.055 in |
| Adopted thickness | $t =$ | 85.00 mm | 3.346 in |

<table>
<thead>
<tr>
<th>Material:</th>
<th><strong>SA-387 22 2 - Plate</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside diameter</td>
<td>$D =$</td>
</tr>
<tr>
<td>Length</td>
<td>$L =$</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>$t =$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Internal pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allowable stress</td>
</tr>
<tr>
<td>Internal pressure</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
</tr>
<tr>
<td>Calculation pressure</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>External pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>External pressure</td>
</tr>
<tr>
<td>Outside diameter</td>
</tr>
<tr>
<td>Max unsupported length</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
</tr>
</tbody>
</table>

**Shell parameter**

$M_x = \sqrt{R_x(t - c - c')} = 0.0579$  

<table>
<thead>
<tr>
<th>Elastic buckling stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{be} = \frac{16 \cdot C_x \cdot E_y \cdot (t - c - c')} {D_o} = 3 327.40 MPa</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Allowable external pressure in the absence of other loads</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_a = 2F_{ha} \left( \frac{t - c - c_e - c'} {D_o} \right) = 21.88 MPa</td>
</tr>
</tbody>
</table>

**Minimum required thickness**

$t = 4.31$ mm  
$0.170$ in  

$t \geq tr$:  Ok

---

Customer  
Riccardo Petrelli

Drawing  
U_150

Walter Tosto S.p.A.  
via Erasmo Piaggio  
Chieti  
Telephone, Fax  
Website, Email address  
Date _______ Calc. _______ Contr._______ Appr.________
<table>
<thead>
<tr>
<th>Condition</th>
<th>Pressure</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>21.88 MPa</td>
<td>3173.1 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>21.88 MPa</td>
<td>3173.1 psi</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>----------------------------------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Internal pressure</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Cylindrical shell - Left channel</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>= SA-387 22 2</td>
<td></td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>= A</td>
<td></td>
</tr>
<tr>
<td>Governing Thickness tg</td>
<td>= 85.00 mm 3.346 in</td>
<td></td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>= No</td>
<td></td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>=</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>= Yes</td>
<td></td>
</tr>
<tr>
<td><strong>Note:</strong> Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| **External pressure**                  |
| **Cylindrical shell - Left channel**   |
| Material                               | = SA-387 22 2 |
| Curve of fig. 3.7 / 3.8                | = A |
| Governing Thickness tg                 | = 85.00 mm 3.346 in |
| PostWeld Heat Treatment                | = No |
| Minimum Design Metal Temperature (MDMT)| = |
| Impact tests required by Code          | = Yes |
| **Note:** Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.) |

**Validation warnings:**
- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000 ]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]
**Reinforcement of opening - Bocchello In Tube Side**

*According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units*

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Calculation</th>
<th>SI Units</th>
<th>US Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calculation temperature</td>
<td>T =</td>
<td>454.00 °C</td>
<td>849.20 °F</td>
</tr>
<tr>
<td>Nozzle material</td>
<td>SA-182 F22 3 - Forgings</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shell material</td>
<td>SA-387 22 2 - Plate</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad material</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Allowable stress from Annex 3.A for the vessel at the design temperature</td>
<td>S =</td>
<td>149.84 MPa</td>
<td>21 732.5 psi</td>
</tr>
<tr>
<td>Shell allowable stress at room temperature</td>
<td>S0 =</td>
<td>207.00 MPa</td>
<td>30 022.8 psi</td>
</tr>
<tr>
<td>Allowable stress from Annex 3.A for the nozzle at the design temperature</td>
<td>Sn =</td>
<td>149.84 MPa</td>
<td>21 732.5 psi</td>
</tr>
<tr>
<td>Nozzle allowable stress at room temperature</td>
<td>Sn0 =</td>
<td>207.00 MPa</td>
<td>30 022.8 psi</td>
</tr>
<tr>
<td>Shell thickness</td>
<td>t =</td>
<td>85.00 mm</td>
<td>3.346 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>tn =</td>
<td>130.00 mm</td>
<td>5.118 in</td>
</tr>
<tr>
<td>Nominal wall thickness of the nozzle thinner portion</td>
<td>tn2 =</td>
<td>45.00 mm</td>
<td>1.772 in</td>
</tr>
<tr>
<td>Tapering angle</td>
<td>ta =</td>
<td>45.00 °</td>
<td></td>
</tr>
<tr>
<td>Nozzle inside diameter</td>
<td>d =</td>
<td>635.00 mm</td>
<td>25.000 in</td>
</tr>
<tr>
<td>Nozzle outside diameter</td>
<td>Od =</td>
<td>725.00 mm</td>
<td>28.543 in</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E =</td>
<td>1.00000</td>
<td></td>
</tr>
<tr>
<td>Nozzle internal corrosion allowance</td>
<td>cni =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Nozzle external corrosion allowance</td>
<td>cne =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Nozzle total corrosion allowance</td>
<td>cn =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Nozzle undertolerance</td>
<td>cn’ =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Nozzle position</td>
<td>=</td>
<td>Radial</td>
<td></td>
</tr>
<tr>
<td>Nozzle connection</td>
<td>=</td>
<td>Integrally reinforced</td>
<td></td>
</tr>
<tr>
<td>Weld joint type</td>
<td>=</td>
<td>7 - Full penetration welds</td>
<td></td>
</tr>
<tr>
<td>Offset from shell border</td>
<td>=</td>
<td>700.00 mm</td>
<td>27.559 in</td>
</tr>
<tr>
<td>Angular offset</td>
<td>=</td>
<td>0 °</td>
<td></td>
</tr>
<tr>
<td>Width of the reinforcing pad</td>
<td>W =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Thickness of the reinforcing pad</td>
<td>te =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Nozzle inside radius</td>
<td>Rn = d/2 + cni + cn’</td>
<td>=</td>
<td>317.50 mm</td>
</tr>
<tr>
<td>Shell inside diameter</td>
<td>Di =</td>
<td>1 275.00 mm</td>
<td>50.197 in</td>
</tr>
<tr>
<td>Effective radius of the shell</td>
<td>Reff = 0.5 \cdot D_i + \phi</td>
<td>=</td>
<td>637.50 mm</td>
</tr>
<tr>
<td>Nozzle projection from the outside of the vessel wall</td>
<td>Lpr1 =</td>
<td>693.00 mm</td>
<td>27.283 in</td>
</tr>
<tr>
<td>Nozzle projection from the inside of the vessel wall</td>
<td>Lpr2 =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Length of variable thickness from the outside of the vessel wall</td>
<td>Lpr3 =</td>
<td>170.00 mm</td>
<td>6.693 in</td>
</tr>
<tr>
<td>Nozzle projection from the outside of the vessel wall to tn2</td>
<td>Lpr4 =</td>
<td>255.00 mm</td>
<td>10.039 in</td>
</tr>
<tr>
<td>Weld leg length of the outside nozzle fillet weld</td>
<td>L41 =</td>
<td>15.00 mm</td>
<td>0.591 in</td>
</tr>
<tr>
<td>Weld leg length of the pad to vessel fillet weld</td>
<td>L42 =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Weld leg length of the inside nozzle fillet weld</td>
<td>L43 =</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Corner radius</td>
<td>r =</td>
<td>10.00 mm</td>
<td>0.394 in</td>
</tr>
<tr>
<td>Effective length along the vessel wall</td>
<td>L_H = \min\left[\sqrt{R_{eff} \cdot (t - c - d)} \cdot 2R_w\right]</td>
<td>=</td>
<td>232.78 mm</td>
</tr>
<tr>
<td>[ L_{eff} = \min[15(t - c - d), t_\phi]\sqrt{R_w(t_n - c_n - c_n')}</td>
<td>=</td>
<td>203.16 mm</td>
<td>7.999 in</td>
</tr>
<tr>
<td>[ L_{qu} = L_{pr1} ]</td>
<td>=</td>
<td>693.00 mm</td>
<td>27.283 in</td>
</tr>
<tr>
<td>[ L_{qu} = 8(t - c - d + t_d) ]</td>
<td>=</td>
<td>680.00 mm</td>
<td>26.772 in</td>
</tr>
<tr>
<td>Effective length along the nozzle wall outside the vessel</td>
<td>L_H = \min[L_{qu}, L_{qu2}, L_{qu3} + t - c - d]</td>
<td>=</td>
<td>288.16 mm</td>
</tr>
<tr>
<td>Effective thickness</td>
<td>t_{eff} = t - c - d + \left(\frac{A_{SSP}}{L_S}\right)</td>
<td>=</td>
<td>85.00 mm</td>
</tr>
<tr>
<td>L11 =</td>
<td>0 mm</td>
<td>0 in</td>
<td></td>
</tr>
<tr>
<td>L12 =</td>
<td>0 mm</td>
<td>0 in</td>
<td></td>
</tr>
</tbody>
</table>
Effective length along the nozzle wall inside the vessel
\[ L_I = \min \left\{ \begin{array}{l} L_{D1}, L_{D2}, L_{D3} \end{array} \right\} \]
\[ \lambda = \min \left\{ \left( \frac{2R_n + t_o - c_o - c'_o}{\sqrt{D_1 + t_{eff}} \cdot t_{eff}} \right) \right\} L_{D3} \]
\[ = 0 \text{ mm} \]
\[ = 0 \text{ in} \]
\[ = 2.25000 \]

Area contributed by the vessel wall
\[ A_1 = \left( (t - c - c')L_{D2} \right) \cdot \max \left( \frac{L_{D3}}{A_{eff}}, \frac{t_{eff}}{t} \right) \]
\[ = 19786.5 \text{ mm}^2 \]
\[ = 30.669 \text{ in}^2 \]

Wall thickness at the variable thickness portion of the nozzle
\[ t_{as} = \left[ 1 + \frac{R_n - t_o - c_o}{t_o - c_o} \cdot \frac{L_{D4} - L_{D1}}{L_{D4} - L_{D3}} \right] (t_o - c_o - c_o') \]
\[ = 96.84 \text{ mm} \]
\[ = 3.813 \text{ in} \]

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3
\[ A_{2a} = \frac{1}{2} (t_o - c_o - c_o') L_{D3} \]
\[ = 33150.0 \text{ mm}^2 \]
\[ = 51.383 \text{ in}^2 \]

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3
\[ A_{2b} = \frac{1}{2} (t_o - c_o - c_o') + t_{as} \cdot \min \left( \frac{0.78}{2} R_n \frac{(L_{D3} - t_o - c_o - c_o')}{t_{eff}}, L_{D3} - L_{D1} \right) \]
\[ = 3761.2 \text{ mm}^2 \]
\[ = 5.830 \text{ in}^2 \]

Nozzle outs. vessel wall area
\[ A_2 = A_{2a} + A_{2b} - r^2 \left( 1 - \frac{\pi}{4} \right) \]
\[ = 36889.8 \text{ mm}^2 \]
\[ = 57.179 \text{ in}^2 \]

Nozzle material factor
\[ f_m = \frac{F_{N}}{S_{m}} \]
\[ = 1.00000 \]

Pad material factor
\[ f_p = \frac{F_{p}}{S_{p}} \]
\[ = 0 \]

Nozzle ins. vessel wall area
\[ A_3 = (t_o - 2c_o - c_o') L_{o} \]
\[ = 0 \text{ mm}^2 \]
\[ = 0 \text{ in}^2 \]

Area contributed by the outside nozzle fillet weld
\[ A_{41} = 0.5L_{D4} \]
\[ = 112.5 \text{ mm}^2 \]
\[ = 0.174 \text{ in}^2 \]

Area contributed by the pad to vessel fillet weld
\[ A_{42} = 0.5L_{D4} \]
\[ = 0 \text{ mm}^2 \]
\[ = 0 \text{ in}^2 \]

Area contributed by the inside nozzle fillet weld
\[ A_{43} = W_{o} \]
\[ = 0 \text{ mm}^2 \]
\[ = 0 \text{ in}^2 \]

Area contributed by the reinforcing pad
\[ A_5 = \min \left\{ A_{4b}, A_{4b} \right\} \]
\[ = 0 \text{ mm}^2 \]
\[ = 0 \text{ in}^2 \]

Total area
\[ A_T = A_1 + f_m (A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_p A_5 \]
\[ = 56788.8 \text{ mm}^2 \]
\[ = 88.023 \text{ in}^2 \]

Radius of the nozzle opening
\[ R_{nc} = R_n \]
\[ = 317.50 \text{ mm} \]
\[ = 12.500 \text{ in} \]

Nozzle radius for force calculation
\[ R_n = \frac{R_{nc} - c_o - c_o'}{0.78} \]
\[ = 378.79 \text{ mm} \]
\[ = 14.913 \text{ in} \]

Shell radius for force calculation
\[ R_{ex} = \frac{R_{ns} - c_o - c_o'}{0.78} \]
\[ = 679.11 \text{ mm} \]
\[ = 26.737 \text{ in} \]

Force from internal pressure in the nozzle
\[ f_N = P_{max} R_n \]
\[ = 2012397 \text{ N} \]
\[ = 452404.82 \text{ lbf} \]

Force from internal pressure in the shell
\[ f_s = P_{max} R_{ex} \]
\[ = 4542206 \text{ N} \]
\[ = 1021128.44 \text{ lbf} \]

Discontinuity force from internal pressure
\[ f_y = P_{max} R_{ex} f_p \]
\[ = 3975253 \text{ N} \]
\[ = 893672.24 \text{ lbf} \]

Average primary membrane stress
\[ \sigma_{av} = \frac{f_N + f_s + f_y}{A_T} \]
\[ = 185.42 \text{ MPa} \]
\[ = 26893.1 \text{ psi} \]

General primary membrane stress
\[ \sigma_{pec} = \frac{P_{max}}{A_T} \]
\[ = 147.30 \text{ MPa} \]
\[ = 21364.0 \text{ psi} \]

Allowable stress
\[ S_{allow} = 1.5 S_{SE} \]
\[ = 224.76 \text{ MPa} \]
\[ = 32598.7 \text{ psi} \]

Maximum local primary membrane stress
\[ P_L = \max \left\{ \left( 2 \sigma_{av} - \sigma_{pec} \right), \sigma_{pec} \right\} \]
\[ = 223.54 \text{ MPa} \]
\[ = 32422.2 \text{ psi} \]

\[ r \geq \min \left[ 0.25t, 3 \text{ mm}(0.125 \text{ in}) \right]: \text{ Ok} \]

\[ PL \leq S_{allow}: \text{ Ok} \]

\[ P \leq P_{max}: \text{ Ok} \]

Strength of nozzle attachment welds

Nozzle maximum allowable pressure (bottom)
\[ P_{max} = \min \left( P_{max1}, P_{max2} \right) \]
\[ = 18.54 \text{ MPa} \]
\[ = 2688.5 \text{ psi} \]

Area resisting pressure
\[ A_p = \frac{f_N + f_s + f_y}{P_{max}} + r^2 \left( 1 - \frac{\pi}{4} \right) \]
\[ = 571163.2 \text{ mm}^2 \]
\[ = 88.023 \text{ in}^2 \]

Nozzle maximum allowable pressure (bottom)
\[ P \leq P_{max}: \text{ Ok} \]

\[ \text{Strength of nozzle attachment welds} \]
Throat dimension of the nozzle to shell weld \( tc = \frac{L_{41}}{\sqrt{2}} \)

Discontinuity factor \( k_d = \frac{P_{\text{max}} + t_{n} - c_{n} - c_{n}'}{P_{\text{max}}} \)

Nozzle to shell weld length \( L_{41} = \frac{\pi}{2} \left( R_{n} + t_{n} - c_{n} - c_{n}' \right) \)

Throat dimension of the outside nozzle fillet weld \( L_{42} = \frac{707L_{41}}{4P_{\text{max}}k_{d}^{2}} \)

Throat dimension for the pad to vessel fillet weld \( L_{43} = \frac{707L_{41}}{4} \)

Welds force \( f_{\text{welds}} = \min \left[ \frac{f_{\text{welds}}}{P_{\text{max}}} , 15S_{s}(A_{2} + A_{3}), \frac{P_{\text{max}}}{4} P_{\text{max}} k_{d}^{2} \right] \)

Nozzle to shell groove weld depth \( t_{\text{wall}} = \frac{L_{w1} + 0.6c_{n} + 0.49L_{41}}{L_{w1}} \)

Average effective shear stress \( \tau = \frac{2899709 \text{ N}}{651880.56 \text{ lbf}} \)

\( tc \geq \min \left[ 0.7t_n, 6 \text{ mm} \right] \) Ok

\( \tau \leq S \) Ok

External pressure

External design temperature \( T_{e} = 20.00 \text{ °C} 68.00 \text{ °F} \)

Allowable stress from Annex 3.A for the vessel at the design temperature \( S = 207.00 \text{ MPa} 30222.8 \text{ psi} \)

Allowable stress from Annex 3.A for the nozzle at the design temperature \( S_n = 207.00 \text{ MPa} 30222.8 \text{ psi} \)

Nozzle inside radius \( R_n = d/2 + c_{n} + c_{n}' \)

Shell inside diameter \( D_i = 1275.00 \text{ mm} 50.197 \text{ in} \)

Effective radius of the shell \( R_{\text{eff}} = 0.5 \cdot D_i + c_{n} \)

Nozzle projection from the outside of the vessel wall \( L_{pr1} = 693.00 \text{ mm} 27.283 \text{ in} \)

Nozzle projection from the inside of the vessel wall \( L_{pr2} = 0 \text{ mm} 0 \text{ in} \)

Length of variable thickness from the outside of the vessel wall \( L_{pr3} = 170.00 \text{ mm} 6.693 \text{ in} \)

Nozzle projection from the outside of the vessel wall to \( t_n \) \( L_{pr4} = 255.00 \text{ mm} 10.039 \text{ in} \)

Weld leg length of the outside nozzle fillet weld \( L_{41} = 15.00 \text{ mm} 0.591 \text{ in} \)

Weld leg length of the pad to vessel fillet weld \( L_{42} = 0 \text{ mm} 0 \text{ in} \)

Weld leg length of the inside nozzle fillet weld \( L_{43} = 0 \text{ mm} 0 \text{ in} \)

Effective length along the vessel wall \( L_{B1} = \min \left[ \sqrt{R_{\text{eff}} \cdot \left( t - c - c' \right) \cdot 2R_{n}} \right] \)

Effective length along the nozzle wall outside the vessel \( t_{\text{eff}} = t - c - c' + \left( \frac{A_{3}f_{\text{weld}}}{L_{B1}} \right) \)

Effective thickness \( t_{\text{eff}} = t - c - c' + \left( \frac{A_{3}f_{\text{weld}}}{L_{B1}} \right) \)

Effective length along the nozzle wall inside the vessel \( t_{\text{eff}} = t - c - c' + \left( \frac{A_{3}f_{\text{weld}}}{L_{B1}} \right) \)

Effective length along the vessel wall with \( t_{\text{wall}} \)

Area contributed by the vessel wall \( A_{1} = \left( t - c - c' \right) f_{\text{weld}} \cdot \max \left[ \frac{A_{3}}{L_{B1}}, \frac{A_{3}}{L_{B2}}, \frac{A_{3}}{L_{B3}} \right] \)

Wall thickness at the variable thickness portion of the nozzle \( t_{\text{wall}} = \left[ 1 + \left( \frac{t_{n} - t_{a2}}{t_{a2} - c_{n} - c_{n}'} \right) \cdot \frac{L_{a2} - L_{a1}}{L_{a2} - L_{a3}} \right] \)

Portion of area \( A_{2} \) for variable nozzle wall thickness, contributed by the nozzle wall within \( L_{pr3} \)

Customer
Riccardo Petrelli

Drawing
U_150

Revision

1028-1811874457
Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3

\[ A_{2b} = \frac{t_n - c_n - c_n' + t_{m}}{2} \cdot \min \left[ \frac{t_n - c_n - c_n' + t_{m}}{2}, L_{N} - L_{A3} \right] \]

Nozzle outs. vessel wall area
\[ A_2 = A_{2a} + A_{2b} - r^2 \left( 1 - \frac{t_n}{N} \right) = 36,889.8 \text{ mm}^2 = 57.179 \text{ in}^2 \]

Nozzle material factor
\[ f_m = \frac{S_{N}}{S} = 1.00000 \]

Pad material factor
\[ f_p = \frac{S_{N}}{S} = 0 \]

Nozzle ins. vessel wall area
\[ A_3 = (t_n - 2c_n - c_n')L_n = 0 \text{ mm}^2 = 0 \text{ in}^2 \]

Area contributed by the outside nozzle fillet weld
\[ A_{41} = 0.5L_{N} = 112.5 \text{ mm}^2 = 0.174 \text{ in}^2 \]

Area contributed by the pad to vessel fillet weld
\[ A_{42} = 0.5L_{1N} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]

Area contributed by the inside nozzle fillet weld
\[ A_{43} = 0.5L_{2N} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]

A5b
\[ A_{5b} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]

Area contributed by the reinforcing pad
\[ A_x = \min \left[ A_{4a}, A_{5b} \right] = 0 \text{ mm}^2 = 0 \text{ in}^2 \]

Total area
\[ A_T = A_1 + f_m(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_m A_2 = 56,788.8 \text{ mm}^2 = 88.023 \text{ in}^2 \]

Radius of the nozzle opening
\[ R_{nc} = R_n = 317.50 \text{ mm} = 12.500 \text{ in} \]

Nozzle radius for force calculation
\[ R_{nt} = \frac{t_n - c_n - c_n'}{f_{nt}} = 378.79 \text{ mm} = 14.913 \text{ in} \]

Shell radius for force calculation
\[ R_{nt} = \frac{t_n - c_n - c_n'}{f_{nt}} = 679.11 \text{ mm} = 26.737 \text{ in} \]

Force from internal pressure in the nozzle
\[ f_z = PR_{nt} = 11,243 \text{ N} = 2,527.47 \text{ lbf} \]

Force from internal pressure in the shell
\[ f_y = PR_{nt}L_n = 25,376 \text{ N} = 5,704.78 \text{ lbf} \]

Discontinuity force from internal pressure
\[ f_{z_d} = PR_{nt}R_{nc} = 22,209 \text{ N} = 4,992.72 \text{ lbf} \]

Average primary membrane stress
\[ \sigma_{avg} = \frac{f_{z} + f_{z_d} + f_y}{A_T} = 1.04 \text{ MPa} = 150.2 \text{ psi} \]

General primary membrane stress
\[ \sigma_{circ} = f_{nt} = 0.82 \text{ MPa} = 119.4 \text{ psi} \]

Allowable stress
\[ \sigma_{allow} = f_{nt} = 185.96 \text{ MPa} = 26,971.6 \text{ psi} \]

Maximum local primary membrane stress
\[ P_{L} = \max \left[ \left( 2\sigma_{avg} - \sigma_{circ} \right), \sigma_{circ} \right] = 1.25 \text{ MPa} = 181.1 \text{ psi} \]

\[ PL \leq Sallow: \text{ Ok} \]

Area resisting pressure
\[ A_P = f_n + f_{z_d} + f_y + r^2 \left( 1 - \frac{t_n}{L} \right) = 571,163.2 \text{ mm}^2 = 885,305 \text{ in}^2 \]

Nozzle maximum allowable pressure (bottom)
\[ P_{max} = \min \left[ P_{max1}, P_{max2} \right] = 15.34 \text{ MPa} = 2,224.5 \text{ psi} \]

\[ P \leq P_{max}: \text{ Ok} \]
Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld
\[ tc = \frac{L_{41}}{\sqrt{2}} \]
\[ = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

Discontinuity force factor
\[ k_y = \frac{R_{xx} + t_n - c_n - c_d}{R_{xx}} \]
\[ = 1.40945 \]

Nozzle to shell weld length
\[ L_c = \frac{\pi}{2} \left( R_{xx} + t_n - c_n - c_d \right) \]
\[ = 702.93 \text{ mm} \quad 27.674 \text{ in} \]

Throat dimension of the outside nozzle fillet weld
\[ L_{417} = 0.7071L_{41} \]
\[ = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

Throat dimension for the pad to vessel fillet weld
\[ L_{427} = 0.7071L_{42} \]
\[ = 0 \text{ mm} \quad 0 \text{ in} \]

Throat dimension for inside nozzle fillet weld
\[ L_{437} = 0.7071L_{43} \]
\[ = 0 \text{ mm} \quad 0 \text{ in} \]

Welds force
\[ \hat{f}_{\text{welds}} = \min \left[ f_{f} \cdot k_y, 1.5S_t (A_2 + A_3), \frac{P}{d} PR_2 k_y^2 \right] \]
\[ = 16,200 \text{ N} \quad 3,641.89 \text{ lbf} \]

Nozzle to shell groove weld depth
\[ t_{w1} = t_{\text{wall}} \]
\[ = 85.00 \text{ mm} \quad 3.346 \text{ in} \]

Average effective shear stress
\[ \tau = \frac{f_{\text{welds}}}{L_{c} \left( 0.49L_{417} + 0.8S_{w1} + 0.49L_{437} \right)} \]
\[ = 0.41 \text{ MPa} \quad 59.5 \text{ psi} \]

\( tc \geq \min [0.7t_n, 6\text{ mm} (0.25\text{ in})] \): Ok
\( \tau \leq S \): Ok
Reinforcement of opening - Bocchello Out Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units

Calculation temperature
\[ T = 454.00 \, ^\circ C \quad 849.20 \, ^\circ F \]

Nozzle material - SA-182 F22 3 - Forgings

Shell material - SA-387 22 2 - Plate

Pad material -

Allowable stress from Annex 3.A for the vessel at the design temperature
\[ S = 149.84 \, \text{MPa} \quad 21,732.5 \, \text{psi} \]

Shell allowable stress at room temperature
\[ S0 = 207.00 \, \text{MPa} \quad 30,022.8 \, \text{psi} \]

Allowable stress from Annex 3.A for the nozzle at the design temperature
\[ Sn = 149.84 \, \text{MPa} \quad 21,732.5 \, \text{psi} \]

Nozzle allowable stress at room temperature
\[ Sn0 = 207.00 \, \text{MPa} \quad 30,022.8 \, \text{psi} \]

Shell thickness
\[ t = 85.00 \, \text{mm} \quad 3.346 \, \text{in} \]

Nozzle thickness
\[ t_n = 130.00 \, \text{mm} \quad 5.118 \, \text{in} \]

Nominal wall thickness of the nozzle thinner portion
\[ t_{n2} = 45.00 \, \text{mm} \quad 1.772 \, \text{in} \]

Tapering angle
\[ \alpha = 45.00 \, ^\circ \]

Nozzle inside diameter
\[ d = 635.00 \, \text{mm} \quad 25.000 \, \text{in} \]

Nozzle outside diameter
\[ Od = 725.00 \, \text{mm} \quad 28.543 \, \text{in} \]

Joint efficiency
\[ E = 1.00000 \]

Nozzle internal corrosion allowance
\[ c_{ni} = 0 \, \text{mm} \quad 0 \, \text{in} \]

Nozzle external corrosion allowance
\[ c_{ne} = 0 \, \text{mm} \quad 0 \, \text{in} \]

Nozzle total corrosion allowance
\[ c_n = 0 \, \text{mm} \quad 0 \, \text{in} \]

Nozzle undertolerance
\[ c_n' = 0 \, \text{mm} \quad 0 \, \text{in} \]

Nozzle position
\[ = \text{Radial} \]

Nozzle connection
\[ = \text{Integrally reinforced} \]

Weld joint type
\[ = 7 - \text{Full penetration welds} \]

Offset from shell border
\[ = 700.00 \, \text{mm} \quad 27.559 \, \text{in} \]

Angular offset
\[ = 180.00 \, ^\circ \]

Width of the reinforcing pad
\[ W = 0 \, \text{mm} \quad 0 \, \text{in} \]

Thickness of the reinforcing pad
\[ te = 0 \, \text{mm} \quad 0 \, \text{in} \]

Nozzle inside radius
\[ R_n = d/2 + c_{ni} + c_{n}' = 317.50 \, \text{mm} \quad 12.500 \, \text{in} \]

Shell inside diameter
\[ D_1 = 1275.00 \, \text{mm} \quad 50.197 \, \text{in} \]

Effective radius of the shell
\[ R_{eff} = 0.5 \cdot D_1 + \delta = 637.50 \, \text{mm} \quad 25.098 \, \text{in} \]

Nozzle projection from the outside of the vessel wall
\[ L_{pr1} = 693.00 \, \text{mm} \quad 27.283 \, \text{in} \]

Nozzle projection from the inside of the vessel wall
\[ L_{pr2} = 0 \, \text{mm} \quad 0 \, \text{in} \]

Length of variable thickness from the outside of the vessel wall
\[ L_{pr3} = 170.00 \, \text{mm} \quad 6.693 \, \text{in} \]

Nozzle projection from the outside of the vessel wall to \( t_{n2} \)
\[ L_{pr4} = 255.00 \, \text{mm} \quad 10.039 \, \text{in} \]

Weld leg length of the outside nozzle fillet weld
\[ L_41 = 15.00 \, \text{mm} \quad 0.591 \, \text{in} \]

Weld leg length of the pad to vessel fillet weld
\[ L_{42} = 0 \, \text{mm} \quad 0 \, \text{in} \]

Weld leg length of the inside nozzle fillet weld
\[ L_43 = 0 \, \text{mm} \quad 0 \, \text{in} \]

Corner radius
\[ r = 10.00 \, \text{mm} \quad 0.394 \, \text{in} \]

Effective length along the vessel wall
\[ L_{w} = \min \left[ \sqrt{R_{eff} \cdot (t - c - \delta)} \cdot 2R_{n}, \frac{2R_{n}}{t_{eff}} \right] = 232.78 \, \text{mm} \quad 9.165 \, \text{in} \]

Effective length along the variable thickness of the vessel wall
\[ L_{HI} = \min \left[ 15 \left( t - c - \delta \right), t_{eff} \right] \sqrt{R_{n} \left( t_n - c_n - c_n' \right)} = 203.16 \, \text{mm} \quad 7.999 \, \text{in} \]

Effective length along the nozzle wall
\[ L_{III} = \frac{L_{HI} + L_{RE}}{2} = 693.00 \, \text{mm} \quad 27.283 \, \text{in} \]

Effective length along the nozzle wall outside the vessel
\[ L_{HI} = \min \left[ L_{HE}, L_{RE}, \frac{t - c - \delta}{L_{HI}} \right] = 680.00 \, \text{mm} \quad 26.772 \, \text{in} \]

Effective length along the variable thickness of the vessel wall
\[ L_{II} = \min \left[ L_{HI}, L_{RE}, L_{HI} + t - c - \delta \right] = 288.16 \, \text{mm} \quad 11.345 \, \text{in} \]

Effective thickness
\[ t_{eff} = t - c - \delta + \left( \frac{A_{eff}}{L_{HI}} \right) = 85.00 \, \text{mm} \quad 3.346 \, \text{in} \]

L11 = 0 \, \text{mm} \quad 0 \, \text{in} \]

L12 = 0 \, \text{mm} \quad 0 \, \text{in} \]
Effective length along the nozzle wall inside the vessel
\[ L_{I} = \min \left[ L_{DB}, L_{D2}, L_{D3} \right] \]
\[ \lambda = \min \left\{ \frac{2R_{n} + t_{m} - c_{w} - c_{y}'}{\sqrt{D_{h} + t_{eff}}}, \frac{a_{ss}}{10} \right\} \]
\[ = 2.25000 \]

Area contributed by the vessel wall
\[ A_{1} = \left( (t - c' - c_{w})L_{D2} \right) \max \left( \frac{L_{D2}}{t_{n2 - c_{w} - c_{y}'}} \right) \]
\[ = 19 \, 786.5 \, mm^{2} = 30.669 \, in^{2} \]

Wall thickness at the variable thickness portion of the nozzle
\[ t_{n} = \left[ 1 + \frac{t_{n2 - c_{w} - c_{y}'}}{L_{D4} - L_{pr3}} \right] \left( t_{n2 - c_{w} - c_{y}'} \right) \]
\[ = 96.84 \, mm = 3.813 \, in \]

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3
\[ A_{22} = (t_{n} - c_{w} - c_{y}')L_{D3} \]
\[ = 33 \, 150.0 \, mm^{2} = 51.383 \, in^{2} \]

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall outside of Lpr3
\[ A_{22} = \frac{t_{n} - c_{w} - c_{y}'}{2} \min \left( \frac{L_{D4} - L_{pr3}}{0.78R_{n}}, \frac{L_{D4} - L_{pr3}}{2} \right) \]
\[ = 3 \, 761.2 \, mm^{2} = 5.830 \, in^{2} \]

Nozzle outs. vessel wall area
\[ A_{2} = A_{2d} + A_{2b} = 36 \, 889.8 \, mm^{2} = 57.179 \, in^{2} \]

Nozzle material factor
\[ f_{m} = \frac{S_{y}}{S_{n}} = 1.00000 \]

Pad material factor
\[ f_{mp} = \frac{S_{m}}{S_{n}} = 0 \]

Nozzle ins. vessel wall area
\[ A_{3} = (t_{m} - 2c_{n} - c_{y}')L_{D2} \]
\[ = 0 \, mm^{2} = 0 \, in^{2} \]

Area contributed by the outside nozzle fillet weld
\[ A_{41} = 0.5f_{m}A_{2} \]
\[ = 112.5 \, mm^{2} = 0.174 \, in^{2} \]

Area contributed by the pad to vessel fillet weld
\[ A_{42} = 0.5f_{mp}A_{2} \]
\[ = 0 \, mm^{2} = 0 \, in^{2} \]

Area contributed by the inside nozzle fillet weld
\[ A_{51} = W_{n} \]
\[ = 0 \, mm^{2} = 0 \, in^{2} \]

Area contributed by the reinforcing pad
\[ A_{s} = \min \left\{ A_{s1}, A_{s2}, A_{s3} \right\} \]
\[ = 0 \, mm^{2} = 0 \, in^{2} \]

Total area
\[ A_{7} = A_{1} + f_{m}(A_{2} + A_{3}) + A_{41} + A_{42} + A_{51} + f_{mp}A_{3} \]
\[ = 56 \, 788.8 \, mm^{2} = 88.023 \, in^{2} \]

Radius of the nozzle opening
\[ R_{nc} = R_{n} \]
\[ = 317.50 \, mm = 12.500 \, in \]

Nozzle radius for force calculation
\[ R_{na} = \frac{f_{m} - c_{w} - c_{y}'}{2} \min \left( \frac{L_{D4} - L_{pr3}}{0.78R_{n}}, \frac{L_{D4} - L_{pr3}}{2} \right) \]
\[ = 378.79 \, mm = 14.913 \, in \]

Shell radius for force calculation
\[ R_{n} = \frac{f_{m} - c_{w} - c_{y}'}{2} \min \left( \frac{L_{D4} - L_{pr3}}{0.78R_{n}}, \frac{L_{D4} - L_{pr3}}{2} \right) \]
\[ = 679.11 \, mm = 26.737 \, in \]

Force from internal pressure in the nozzle
\[ f_{n} = \frac{P_{n}R_{nc}L_{D2}}{R_{n}} \]
\[ = 2 \, 012 \, 397 \, N = 452 \, 404.82 \, lbf \]

Force from internal pressure in the shell
\[ f_{y} = \frac{P_{S}R_{ex}A_{ex}}{R_{ex}} \]
\[ = 4 \, 542 \, 206 \, N = 1 \, 021 \, 128.44 \, lbf \]

Discontinuity force from internal pressure
\[ \sigma_{avg} = \frac{f_{n} + f_{y}}{A_{w}} \]
\[ = 185.42 \, MPa = 26 \, 893.1 \, psi \]

Average primary membrane stress
\[ \sigma_{p} = \frac{P_{n}R_{nc}L_{D2}}{R_{n}} \]
\[ = 147.30 \, MPa = 21 \, 364.0 \, psi \]

General primary membrane stress
\[ \sigma_{G} = \frac{P_{S}R_{ex}A_{ex}}{R_{ex}} \]
\[ = 224.76 \, MPa = 32 \, 598.7 \, psi \]

Allowable stress
\[ A_{w} = \frac{2\sigma_{avg} - \sigma_{circ}}{4} \]
\[ = 223.54 \, MPa = 32 \, 422.2 \, psi \]

Maximum local primary membrane stress
\[ P_{L} = \max \left\{ 2\sigma_{avg} - \sigma_{circ}, \sigma_{circ} \right\} \]
\[ = 18.75 \, MPa = 2720.1 \, psi \]

Area resisting pressure
\[ A_{p} = \frac{f_{n} + f_{y} + f_{y}}{P_{n}} + R_{nc} \left( 1 - \frac{r}{D_{h}} \right) \]
\[ = 571 \, 163.2 \, mm^{2} = 885.305 \, in^{2} \]

Nozzle maximum allowable pressure (bottom)
\[ P_{m} = \min \left[ P_{max1}, P_{max2} \right] \]
\[ = 18.54 \, MPa = 2688.5 \, psi \]

Strength of nozzle attachment welds
Throat dimension of the nozzle to shell weld
\[ tc = \frac{L41}{\sqrt{2}} = 10.61 \text{ mm} = 0.418 \text{ in} \]

Discontinuity force factor
\[ k_p = \frac{P_{hex} + t_n - c_n - c_i'}{P_{tan}} = 1.40945 \]

Nozzle to shell weld length
\[ L_\text{t} = \frac{\pi}{2} \left( R_k + t_n - c_n - c_i' \right) = 702.93 \text{ mm} = 27.674 \text{ in} \]

Throat dimension of the outside nozzle fillet weld
\[ L_{412} = 0.707L_{412} = 10.61 \text{ mm} = 0.418 \text{ in} \]

Throat dimension for the pad to vessel fillet weld
\[ L_{422} = 0.707L_{422} = 0 \text{ mm} = 0 \text{ in} \]

Throat dimension for inside nozzle fillet weld
\[ L_{433} = 0.707L_{433} = 0 \text{ mm} = 0 \text{ in} \]

Welds force
\[ f_{welds} = \min\left[ f_{y, k_p}, 1.5S_{y}(A_2 + A_3), \frac{\pi}{4}PR_{g}k_p^2 \right] = 2899709 \text{ N} = 651880.56 \text{ lbf} \]

Nozzle to shell groove weld depth
\[ tw_1 = tw_{wall} = 85.00 \text{ mm} = 3.346 \text{ in} \]

Average effective shear stress
\[ \tau = \frac{f_{welds}}{L_{c}(0.49L_{412} + 0.6t_{w3} + 0.49L_{412})} = 73.41 \text{ MPa} = 10646.5 \text{ psi} \]

\( tc \geq \min[0.7tn, 6\text{mm (0.25in)}]: \text{Ok} \)
\( \tau \leq S: \text{Ok} \)

**External pressure**

External design temperature
\[ Te = 20.00 \degree C = 68.00 \degree F \]

Allowable stress from Annex 3.A for the vessel at the design temperature
\[ S = 207.00 \text{ MPa} = 30022.8 \text{ psi} \]

Allowable stress from Annex 3.A for the nozzle at the design temperature
\[ S_n = 207.00 \text{ MPa} = 30022.8 \text{ psi} \]

Nozzle inside radius
\[ R_n = d/2 + cni + cn' = 317.50 \text{ mm} = 12.500 \text{ in} \]

Shell inside diameter
\[ D_i = 1275.00 \text{ mm} = 50.197 \text{ in} \]

Effective radius of the shell
\[ R_{eff} = 0.5\cdot D_i + c = 637.50 \text{ mm} = 25.098 \text{ in} \]

Nozzle projection from the outside of the vessel wall
\[ L_{pr1} = 693.00 \text{ mm} = 27.283 \text{ in} \]

Nozzle projection from the inside of the vessel wall
\[ L_{pr2} = 0 \text{ mm} = 0 \text{ in} \]

Length of variable thickness from the outside of the vessel wall
\[ L_{pr3} = 170.00 \text{ mm} = 6.693 \text{ in} \]

Nozzle projection from the outside of the vessel wall to tn2
\[ L_{pr4} = 255.00 \text{ mm} = 10.039 \text{ in} \]

Weld leg length of the outside nozzle fillet weld
\[ L_{41} = 15.00 \text{ mm} = 0.591 \text{ in} \]

Weld leg length of the pad to vessel fillet weld
\[ L_{42} = 0 \text{ mm} = 0 \text{ in} \]

Weld leg length of the inside nozzle fillet weld
\[ L_{43} = 0 \text{ mm} = 0 \text{ in} \]

Effective length along the vessel wall
\[ L_{v} = \min\left[ \sqrt{R_{eff} \cdot (t - c - d'), 2R_{n}} \right] = 232.78 \text{ mm} = 9.165 \text{ in} \]

Effective length along the vessel wall
\[ L_{\text{HV}} = \min\left[ 15(t - c - d'), t_v \right] + \sqrt{R_{n} \cdot (t_n - c_n - c_i')} = 203.16 \text{ mm} = 7.999 \text{ in} \]

Effective length along the vessel wall
\[ L_{\text{HV2}} = L_{pr1} = 693.00 \text{ mm} = 27.283 \text{ in} \]

Effective length along the vessel wall
\[ L_{\text{HV3}} = 8(t - c - d' + t_e) = 680.00 \text{ mm} = 26.772 \text{ in} \]

Effective length along the nozzle wall outside the vessel
\[ L_{R} = \min\left[ L_{HB}, L_{AH}, L_{b32} \right] + t - c - d' = 288.16 \text{ mm} = 11.345 \text{ in} \]

Effective thickness
\[ t_{eff} = t - c - d' + \left( \frac{A_{32}}{L_{R}} \right) = 85.00 \text{ mm} = 3.346 \text{ in} \]

Effective length along the nozzle wall inside the vessel
\[ L_{l} = \min\left[ L_{HB}, L_{AH}, L_{b32} \right] = 0 \text{ mm} = 0 \text{ in} \]

Effective length along the nozzle wall inside the vessel
\[ L_{l} = \min\left[ L_{HB}, L_{AH}, L_{b32} \right] = 0 \text{ mm} = 0 \text{ in} \]

Area contributed by the vessel wall
\[ A_{3} = \left( (t - c - d')L_{32} \right) \cdot \max\left[ \left( \frac{A_{32}}{L_{32}} \right), 10 \right] = 19786.5 \text{ mm}^2 = 30669 \text{ in}^2 \]

Wall thickness at the variable thickness portion of the nozzle
\[ \tau_{w3} = \left[ 1 + \frac{L_{w3} - L_{b32}}{L_{b4} - L_{w4}} \right] \cdot \left( c_n - c_i' \right) = 96.84 \text{ mm} = 3.813 \text{ in} \]

Portion of area A2 for variable nozzle wall thickness, contributed by the nozzle wall within Lpr3
\[ A_{23} = (L_{b2} - c_n - c_i')L_{b32} = 33150.0 \text{ mm}^2 = 51383 \text{ in}^2 \]
Portion of area \( A_2 \) for variable nozzle wall thickness, contributed by the nozzle wall outside of \( L_{pr3} \):

\[
A_2 = \frac{t_n - c_n - c_n'}{2} + t_{wa} \min \left[ 0.78 \sqrt{R_n \left( \frac{t_n - c_n - c_n'}{2} + t_{wa} \right)}, L_{nt} - L_{ct} \right]
\]

\[= 3761.2 \text{ mm}^2 \quad 5.830 \text{ in}^2 \]

Nozzle outs. vessel wall area:

\[
A_2 = A_2 + A_2 - r^2 \left( 1 - \frac{r}{R_n} \right)
\]

\[= 36889.8 \text{ mm}^2 \quad 57.179 \text{ in}^2 \]

Nozzle material factor:

\[
f_m = \frac{S}{S_n} = 1.00000
\]

Pad material factor:

\[
f_p = \frac{S}{S_n} = 0
\]

Nozzle ins. vessel wall area:

\[
A_3 = \left( t_n - 2c_n - c_n' \right) L_n = 0 \text{ mm}^2 \quad 0 \text{ in}^2
\]

Area contributed by the outside nozzle fillet weld:

\[
A_{A1} = 0.5L_{nt} \frac{r_n}{R_n} = 112.5 \text{ mm}^2 \quad 0.174 \text{ in}^2
\]

Area contributed by the pad to vessel fillet weld:

\[
A_{A2} = 0.5L_{nt} \frac{r_n}{R_n} = 0 \text{ mm}^2 \quad 0 \text{ in}^2
\]

Area contributed by the inside nozzle fillet weld:

\[
A_{A3} = 0.5L_{nt} \frac{r_n}{R_n} = 0 \text{ mm}^2 \quad 0 \text{ in}^2
\]

Area contributed by the reinforcing pad:

\[
A_5 = \begin{cases} 0 \text{ mm}^2 \quad 0 \text{ in}^2 \end{cases}
\]

Total area:

\[
A_T = A_1 + f_m \left( A_2 + A_3 \right) + A_{A1} + A_{A2} + A_{A3} + f_p A_5
\]

\[= 56788.8 \text{ mm}^2 \quad 88.023 \text{ in}^2 \]

Radius of the nozzle opening:

\[R_{nc} = R_n = 317.50 \text{ mm} \quad 12.500 \text{ in} \]

Nozzle radius for force calculation:

\[R_{nt} = \frac{t_n - c_n - c_n'}{f_{nt}} = 378.79 \text{ mm} \quad 14.913 \text{ in} \]

Shell radius for force calculation:

\[R_{ns} = \frac{S}{f_{nt}} = 679.11 \text{ mm} \quad 26.737 \text{ in} \]

Force from internal pressure in the nozzle:

\[f_\phi = PR_{nt} = 11243 \text{ N} \quad 2527.47 \text{ lbf} \]

Force from internal pressure in the shell:

\[f_y = PR_{nt} = 25376 \text{ N} \quad 5704.78 \text{ lbf} \]

Discontinuity force from internal pressure:

\[f_y = PR_{nt} R_{nt} = 22209 \text{ N} \quad 4992.72 \text{ lbf} \]

Average primary membrane stress:

\[
\sigma_{awg} = \frac{A_T}{A_T} \left( f_\phi + f_y + f_y \right) = 1.04 \text{ MPa} \quad 150.2 \text{ psi}
\]

General primary membrane stress:

\[
\sigma_{arc} = \frac{S}{f_{nt}} = 0.82 \text{ MPa} \quad 119.4 \text{ psi}
\]

Allowable stress:

\[
S_{allow} = S_{allow} = 185.96 \text{ MPa} \quad 26971.6 \text{ psi}
\]

Maximum local primary membrane stress:

\[
P_L = \max \left[ 2\sigma_{awg} - \sigma_{arc} \right] = 1.25 \text{ MPa} \quad 181.1 \text{ psi}
\]

\[
PL \leq S_{allow}: \text{ Ok}
\]

\[
P \leq P_{max}: \text{ Ok}
\]

Area resisting pressure:

\[
A_p = \frac{f_\phi + f_y + f_y}{P} + r^2 \left( 1 - \frac{r}{R_n} \right)
\]

\[= 571163.2 \text{ mm}^2 \quad 885.305 \text{ in}^2
\]

Nozzle maximum allowable pressure (bottom):

\[
P_{max} = \min \left[ P_{max1}, P_{max2} \right] = 15.34 \text{ MPa} \quad 2224.5 \text{ psi}
\]

\[P \leq P_{max}: \text{ Ok}\]
### Strength of nozzle attachment welds

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Throat dimension of the nozzle to shell weld</td>
<td>(tc = \frac{L_{41}}{\sqrt{2}} = 10.61 \text{ mm} 0.418 \text{ in})</td>
</tr>
<tr>
<td>Discontinuity force factor (k_y)</td>
<td>(1.40945)</td>
</tr>
<tr>
<td>Nozzle to shell weld length (L_c)</td>
<td>(702.93 \text{ mm} 27.674 \text{ in})</td>
</tr>
<tr>
<td>Throat dimension of the outside nozzle fillet weld (L_{427})</td>
<td>(0 \text{ mm} 0 \text{ in})</td>
</tr>
<tr>
<td>Throat dimension for the pad to vessel fillet weld (L_{437})</td>
<td>(0 \text{ mm} 0 \text{ in})</td>
</tr>
<tr>
<td>Welds force (f_{\text{welds}})</td>
<td>(16,200 \text{ N} 3,641.89 \text{ lbf})</td>
</tr>
<tr>
<td>Nozzle to shell groove weld depth (t_w1 = t_{\text{wall}})</td>
<td>(85.00 \text{ mm} 3.346 \text{ in})</td>
</tr>
<tr>
<td>Average effective shear stress (\tau)</td>
<td>(0.41 \text{ MPa} 59.5 \text{ psi})</td>
</tr>
<tr>
<td>(tc \geq \min[0.7t_n, 6\text{mm} (0.25\text{in})]: \text{Ok})</td>
<td>(\tau \leq S: \text{Ok})</td>
</tr>
</tbody>
</table>

---

**Customer**

Riccardo Petrelli

**Drawing**

U_150

**Revision**
Nozzle - Bocchello In Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units

Calculation temperature

\[ T = 454.00 \, ^\circ\text{C} = 849.20 \, ^\circ\text{F} \]

Material: SA-182 F22 3 - Forgings

Allowable stress at room temperature

\[ ST = 207.00 \, \text{MPa} = 30,022.8 \, \text{psi} \]

Joint efficiency

\[ E = 1.00 \]

Corrosion allowance

\[ c = 0 \, \text{mm} = 0 \, \text{in} \]

External corrosion allowance

\[ ce = 0 \, \text{mm} = 0 \, \text{in} \]

Wall undertolerance

\[ c' = 0 \, \text{mm} = 0 \, \text{in} \]

Inside diameter

\[ D = 635.00 \, \text{mm} = 25.000 \, \text{in} \]

Length

\[ L = 778.00 \, \text{mm} = 30.630 \, \text{in} \]

Adopted thickness

\[ t = 45.00 \, \text{mm} = 1.772 \, \text{in} \]

Internal pressure

Allowable stress

\[ S = 149.84 \, \text{MPa} = 21,732.5 \, \text{psi} \]

Internal pressure

\[ Pi = 18.44 \, \text{MPa} = 2,674.0 \, \text{psi} \]

Overpressure due to static head

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

Calculation pressure

\[ P = Pi + Ph = 18.44 \, \text{MPa} = 2,674.0 \, \text{psi} \]

Required thickness

\[ t_r = \frac{D + 2(c + ce) + 2(c' + ce + c')}{2 - (c/c' - 1) + c + ce + c'} = 41.57 \, \text{mm} = 1.637 \, \text{in} \]

External pressure

External pressure

\[ Pe = 0.10 \, \text{MPa} = 14.9 \, \text{psi} \]

External static head

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

Calculation pressure

\[ P = Pe + Ph = 0.10 \, \text{MPa} = 14.9 \, \text{psi} \]

External design temperature

\[ Te = 20.00 \, ^\circ\text{C} = 68.00 \, ^\circ\text{F} \]

Outside diameter

\[ Do = 725.00 \, \text{mm} = 28.543 \, \text{in} \]

Max unsupported length

\[ L = 778.00 \, \text{mm} = 30.630 \, \text{in} \]

Modulus of elasticity

\[ Ey = 210,350.00 \, \text{MPa} = 30,508,688.2 \, \text{psi} \]

Shell parameter

\[ \frac{L}{\sqrt{R_c (t - c - c')}} = 6.09143 \]

\[ C_h = \frac{M_c - 0.579}{M_c} = 0.16690 \]

Elastic buckling stress

\[ F_{he} = \frac{16 \cdot C_h \cdot Ey \cdot (t - c - ce - c')} {D_o} = 3,486.44 \, \text{MPa} = 505,664.7 \, \text{psi} \]

Buckling stress

\[ Fic = Sy = 310.00 \, \text{MPa} = 44,961.7 \, \text{psi} \]

Yield strength at design temperature

\[ Sy = 310.00 \, \text{MPa} = 44,961.7 \, \text{psi} \]

Design factor

\[ FS = 1.66700 \]

Hoop compressive membrane stress

\[ Fha = \frac{Fic}{FS} = 185.96 \, \text{MPa} = 26,971.6 \, \text{psi} \]

Allowable external pressure in the absence of other loads

\[ P_a = 2Fha \left( \frac{t - c - ce - c'} {D_o} \right) = 23.09 \, \text{MPa} = 3,348.2 \, \text{psi} \]

Minimum required thickness

\[ tr = 2.20 \, \text{mm} = 0.087 \, \text{in} \]

Maximum allowable external pressures

New & cold

\[ = 23.09 \, \text{MPa} = 3,348.2 \, \text{psi} \]

Hot & corroded

\[ = 23.09 \, \text{MPa} = 3,348.2 \, \text{psi} \]
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure

**Nozzle - Bocchello In Tube Side (Nozzle to wall - shell)**

- **Material** = SA-182 F22 3
- **Curve of fig. 3.7 / 3.8** = B
- **Governing Thickness**
  - **tg** = 85.00 mm / 3.346 in
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)**
- **Impact tests required by Code** = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

#### External pressure

**Nozzle - Bocchello In Tube Side (Nozzle to wall - shell)**

- **Material** = SA-182 F22 3
- **Curve of fig. 3.7 / 3.8** = B
- **Governing Thickness**
  - **tg** = 85.00 mm / 3.346 in
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)**
- **Impact tests required by Code** = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*
### Welding neck flange - Flange In Tube Side

According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units

#### Design data

- **Internal design temperature**
  - \( T = 454.00 \, ^\circ C \) = 849.20 \, ^\circ F \)

- **Internal design pressure**
  - \( P = 18.44 \, MPa \) = 2674.0 psi

- **External design temperature**
  - \( T_e = 20.00 \, ^\circ C \) = 68.00 \, ^\circ F \)

- **External design pressure**
  - \( P_e = 0.10 \, MPa \) = 14.9 psi

- **Joint efficiency**
  - \( E = 1.00 \)

#### Flange material

- **SA-182 F22 3 - Forgings**

#### Shell material

- **SA-182 F22 3 - Forgings**

#### Bolting material

- **SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting**

#### Gasket

- **Grooved Metal - Stainless steels and nickel-base alloys**

#### Allowable stresses

<table>
<thead>
<tr>
<th>Condition</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>( S_{fo} = 149.84 , MPa / 21,732.5 , psi )</td>
<td>( S_{no} = 149.84 , MPa / 21,732.5 , psi )</td>
<td>( S_{bo} = 161.44 , MPa / 23,414.9 , psi )</td>
</tr>
<tr>
<td>Seating</td>
<td>( S_{fg} = 207.00 , MPa / 30,022.8 , psi )</td>
<td>( S_{ng} = 207.00 , MPa / 30,022.8 , psi )</td>
<td>( S_{bg} = 172.00 , MPa / 24,946.5 , psi )</td>
</tr>
<tr>
<td>Test</td>
<td>( S_{ft} = / )</td>
<td>( S_{nt} = / )</td>
<td>( S_{bt} = / )</td>
</tr>
</tbody>
</table>

#### Internal pressure

- \( P_d = 18.44 \, MPa \) = 2674.0 psi

#### Overpressure due to static head

- \( P_h = 0 \, MPa \) = 0 psi

#### Calculation pressure

- \( P = 18.44 \, MPa \) = 2674.0 psi

#### Modulus of Elasticity at the operating load case temperature

- \( E_{yo} = 179,600.00 \, MPa \) = 26,048,777.7 psi

#### Modulus of Elasticity at the gasket seating load case temperature

- \( E_{yg} = 210,350.00 \, MPa \) = 30,508,688.2 psi

#### Corrosion allowance

- \( c = 0 \, mm \) = 0 in

#### Flange external diameter

- \( A = 1026.00 \, mm \) = 40.394 in

#### Inside diameter

- \( B = 635.00 \, mm \) = 25.000 in

#### Bolt circle

- \( C = 905.00 \, mm \) = 35.630 in

#### Flange thickness

- \( t = 170.00 \, mm \) = 6.693 in

#### Mean gasket diameter

- \( G_{mean} = 651.00 \, mm \) = 25.630 in

#### Thickness of the hub at the small end

- \( g_0 = 45.00 \, mm \) = 1.772 in

#### Thickness of the hub at the large end

- \( g_1 = 57.00 \, mm \) = 2.244 in

#### Thickness of the hub at the small end (Corroded)

- \( g_0' = g_0 - c = 45.00 \, mm \) = 1.772 in

#### Thickness of the hub at the large end (Corroded)

- \( g_1' = g_1 - c = 57.00 \, mm \) = 2.244 in

#### Hub length

- \( h = 68.00 \, mm \) = 2.677 in

#### Gasket parameters

- **Gasket factor**
  - \( m = 4.25 \)

#### Gasket contact width

- \( y = 70.00 \, MPa \) = 10 152.6 psi

#### Basic gasket seating width

- \( h_b = \frac{N}{2} = 8.00 \, mm \) = 0.315 in

#### Conversion factor for length

- \( Cul = 25.4000 \)

#### Effective gasket contact width

- \( b = 0.5C_{ul} \frac{h_b}{C_{rl}} = 7.13 \, mm \) = 0.281 in

#### Outside diameter of the gasket contact area

- \( G_c = G_{mean} + N = 667.00 \, mm \) = 26.260 in

#### Diameter at the location of the gasket load reaction

- \( G = G_c - 2b = 652.75 \, mm \) = 25.699 in
### Bolt loads

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of bolts</td>
<td>20</td>
</tr>
<tr>
<td>Bolt type</td>
<td>ANSI_TEMA 2-1/2&quot;</td>
</tr>
<tr>
<td>Bolt spacing</td>
<td>142.16 mm 5.597 in</td>
</tr>
<tr>
<td>Nominal bolt diameter</td>
<td>63.50 mm 2.500 in</td>
</tr>
<tr>
<td>Design bolt load for the operating condition</td>
<td>8 456 915 N 1 901 189.99 lbf</td>
</tr>
<tr>
<td>Design bolt load for the test condition</td>
<td>0 N 0 lbf</td>
</tr>
<tr>
<td>Design bolt load for the gasket seating (Bolt)</td>
<td>1 023 113 N 230 004.85 lbf</td>
</tr>
<tr>
<td>External tensile net-section axial force</td>
<td>0 N 0 lbf</td>
</tr>
<tr>
<td>Absolute value of the external net-section bending moment</td>
<td>0 Nm 0 lbf/in</td>
</tr>
<tr>
<td>External tensile net-section axial force (test condition)</td>
<td>0 N 0 lbf</td>
</tr>
<tr>
<td>External tensile net-section bending moment (test condition)</td>
<td>0 Nm 0 lbf/in</td>
</tr>
<tr>
<td>Total minimum required cross-sectional area of the bolts</td>
<td>52 384.3 mm² 81.196 in²</td>
</tr>
<tr>
<td>Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion</td>
<td>55 380.5 mm² 85.840 in²</td>
</tr>
<tr>
<td>Maximum bolts area for gasket crush</td>
<td>26 706.3 mm² 41.395 in²</td>
</tr>
<tr>
<td>Design bolt load for the gasket seating (Flange)</td>
<td>9 267 772 N 2 083 477.94 lbf</td>
</tr>
</tbody>
</table>

### Flange constants

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ratio of the flange outside diameter to the inside diameter</td>
<td>K = A/B' = 1.61575</td>
</tr>
<tr>
<td>Stress factor Y</td>
<td>4.22139</td>
</tr>
<tr>
<td>Stress factor T</td>
<td>1.66082</td>
</tr>
<tr>
<td>Stress factor U</td>
<td>4.63888</td>
</tr>
<tr>
<td>Stress factor Z</td>
<td>2.24174</td>
</tr>
<tr>
<td>Hub length parameter</td>
<td>169.04 mm 6.655 in</td>
</tr>
<tr>
<td>Hub thickness ratio</td>
<td>1.26667</td>
</tr>
<tr>
<td>Hub thickness ratio</td>
<td>1.26667</td>
</tr>
<tr>
<td>Hub length ratio</td>
<td>0.40227</td>
</tr>
<tr>
<td>Flange stress factor for integral type flanges</td>
<td>0.87850</td>
</tr>
<tr>
<td>Flange stress factor for integral type flanges</td>
<td>0.40681</td>
</tr>
<tr>
<td>Hub stress correction factor for integral flanges</td>
<td>1.00000</td>
</tr>
<tr>
<td>Stress factor e</td>
<td>0.00520</td>
</tr>
</tbody>
</table>

---

**Note:** The calculations for the bolt loads and flange constants are based on the given data and standard engineering practices. The values provided are for illustrative purposes and may need to be verified with specific engineering standards and codes.
Stress factor \( d \) 
\[
\frac{U g \theta h_c}{d} = 3903371.60229
\]

Stress factor \( L \) 
\[
L = \frac{b + 1}{b} \frac{d}{d} = 2.39272
\]

Gasket load for the operating condition 
\[
H_G = W_o - H = 2290461 \text{ N} \quad 514916.01 \text{ lbf}
\]

Total hydrostatic end force on the area inside of the flange 
\[
H_S = 0.785B^2P = 5835736 \text{ N} \quad 1311925.51 \text{ lbf}
\]

Total hydrostatic end force 
\[
H = 0.785Q^2P = 6166454 \text{ N} \quad 1386273.97 \text{ lbf}
\]

Difference 
\[
H_T = H - H_S = 330718 \text{ N} \quad 74348.46 \text{ lbf}
\]

Moment arm for load HD 
\[
h_D = \frac{2}{3} C - B - g = 106.50 \text{ mm} \quad 4.193 \text{ in}
\]

Moment arm for load HG 
\[
h_G = C - G = 126.13 \text{ mm} \quad 4.966 \text{ in}
\]

Moment arm for load HT 
\[
h_T = \frac{1}{2} \left( \frac{C - B}{2} + h_C \right) = 130.56 \text{ mm} \quad 5.140 \text{ in}
\]

Average of the hub thicknesses 
\[
A_H = 0.5(A_B + h_C) = 195.50 \text{ mm} \quad 7.697 \text{ in}
\]

\( \text{AA} = A_B = 195.50 \text{ mm} \quad 7.697 \text{ in} \)

\( \text{BB} = t = 170.00 \text{ mm} \quad 6.693 \text{ in} \)

\( \text{CC} = h = 68.00 \text{ mm} \quad 2.677 \text{ in} \)

\( \text{DDG} = \text{Gavg} = 51.00 \text{ mm} \quad 2.008 \text{ in} \)

Moment of inertia \( K_{AB} \) 
\[
K_{AB} = (A_B B_B^2) \left[ 1 - 0.12 \left( \frac{B_B}{A_B} \right) \left( 1 - \frac{1}{12} \left( \frac{B_B}{A_B} \right) \right) \right] = 153126579 \text{ mm}^4 \quad 367888 \text{ in}^4
\]

Moment of inertia \( K_{CD} \) 
\[
K_{CD} = (C_D D_D^2) \left[ 1 - 0.0105 \left( \frac{D_D}{C_D} \right) \left( 1 - \frac{1}{192} \left( \frac{D_D}{C_D} \right) \right) \right] = 2297581 \text{ mm}^4 \quad 5520 \text{ in}^4
\]

Bending moment of inertia of the flange cross-section 
\[
I = \frac{1}{V} \left[ \frac{h_D}{12} + (C - 2h_D) \right] = 111738738 \text{ mm}^4 \quad 268453 \text{ in}^4
\]

Cross-section polar moment of inertia
\[
I_p = K_{AB} + K_{CD} = 15542159 \text{ mm}^4 \quad 373408 \text{ in}^4
\]

Flange moments

Nominal bolt diameter 
\[
\text{dB} = 63.50 \text{ mm} \quad 2.500 \text{ in}
\]

TEMA Load concentration factor 
\[
c_F = \text{MAX} \left[ \min \left( \frac{1}{22B + \frac{m}{63}} \right) \right] = 1.00000
\]

Moment factor used to design split rings
\[
F_s = 1.00
\]

Component of the flange design moment resulting from a net section bending moment and/or axial force
\[
M_{oe} = 4M_{F} \left[ \frac{1}{0.38466} \frac{I}{B^2} + \frac{h_D}{(C - 2h_D)} \right] + F_A h_D = 0 \text{ N} \cdot \text{m} \quad 0 \text{ lbf} \cdot \text{in}
\]

Flange design moment for the operating condition
\[
M_o = c_F \text{abs} \left[ H_D h_D + H_T h_T + H_C h_C + M_{oe} \right] F_s = 953575.6 \text{ N} \cdot \text{m} \quad 8439854.3 \text{ lbf} \cdot \text{in}
\]

Flange design moment for the gasket seating condition
\[
M_g = c_F \text{abs} \left[ \frac{W_G (C - G)}{2} \right] F_s = 1168920.1 \text{ N} \cdot \text{m} \quad 10345814.1 \text{ lbf} \cdot \text{in}
\]

Flange stresses - operating condition

Corrected inside diameter of the flange
\[
B_1' = B' + g_1' = 692.00 \text{ mm} \quad 27.244 \text{ in}
\]

Flange hub stress - operating condition
\[
S_H = \frac{f M_o}{L_g B_1} = 177.26 \text{ MPa} \quad 25709.2 \text{ psi}
\]

Flange radial stress - operating condition
\[
S_R = \frac{(133\alpha + 1)M_o}{L_g B} = 47.23 \text{ MPa} \quad 6850.7 \text{ psi}
\]

Flange tangential stress - operating condition
\[
S_T = \frac{f M_o}{L_g B} = 113.46 \text{ MPa} \quad 16456.5 \text{ psi}
\]

\( S_H \leq \min[1.5S_f o, 2.5S_{no}]: \text{ Ok} \)

\( S_R \leq S_{fo}: \text{ Ok} \)

\( S_T \leq S_{fo}: \text{ Ok} \)

\( (S_H + S_R) / 2 \leq S_{fo}: \text{ Ok} \)

\( (S_H + S_T) / 2 \leq S_{fo}: \text{ Ok} \)
Flange stresses - seating condition

Flange hub stress - gasket seating condition
\[ S_{h} = \frac{f M_{g}}{t_{g} B_{h}} \]
\[ = 217.29 \text{ MPa} \quad 31515.1 \text{ psi} \]

Flange radial stress - gasket seating condition
\[ S_{r} = \frac{(133e+1) M_{g}}{B_{r} R_{g}} \]
\[ = 57.90 \text{ MPa} \quad 8397.8 \text{ psi} \]

Flange tangential stress - gasket seating condition
\[ S_{t} = \frac{M_{g} R}{P_{B}} - Z S_{h} \]
\[ = 139.09 \text{ MPa} \quad 20172.9 \text{ psi} \]

SHg ≤ min[1.5Sfg, 2.5Sng]: Ok
SRg ≤ Sfg: Ok
STg ≤ Sfg: Ok
\((SHg + SRg) / 2 ≤ Sfg: Ok\)
\((SHg + STg) / 2 ≤ Sfg: Ok\)

Flange rigidity - operating condition

Rigidity index factor
\[ KR = 0.30 \]

Flange rigidity index
\[ J = \frac{5214VM_{g}}{LEy_{g}g^{2}K_{f}K_{h}} \]
\[ = 0.45833 \]
\[ Jo ≤ 1: Ok \]

Flange rigidity - seating condition

Flange rigidity index
\[ J = \frac{5214VM_{g}}{LEy_{g}g^{2}K_{f}K_{h}} \]
\[ = 0.47970 \]
\[ Jg ≤ 1: Ok \]

Hub thickness

Minimum hub thickness as cylindrical shell
\[ th_{,min} = 41.57 \text{ mm} \quad 1.637 \text{ in} \]
\[ g_{0} ≥ th_{,min}: Ok \]

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)
\[ = 26.18 \text{ MPa} \quad 3796.5 \text{ psi} \]

Hot & corroded (flange)
\[ = 19.00 \text{ MPa} \quad 2756.4 \text{ psi} \]

New & cold (bolts)
\[ = 20.77 \text{ MPa} \quad 3011.8 \text{ psi} \]

Hot & corroded (bolts)
\[ = 19.49 \text{ MPa} \quad 2826.9 \text{ psi} \]
External pressure

<table>
<thead>
<tr>
<th>Allowable stresses</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design condition</td>
<td>Sfo = 207.00 MPa / 30 022.8 psi</td>
<td>Sno = 207.00 MPa / 30 022.8 psi</td>
<td>Sbo = 172.00 MPa / 24 946.5 psi</td>
</tr>
<tr>
<td>Seating condition</td>
<td>Sfg = 207.00 MPa / 30 022.8 psi</td>
<td>Sng = 207.00 MPa / 30 022.8 psi</td>
<td>Sbg = 172.00 MPa / 24 946.5 psi</td>
</tr>
</tbody>
</table>

External design pressure
Pe = 0.10 MPa / 14.9 psi

Overpressure due to static head
Ph = 0 MPa / 0 psi

Calculation pressure
P = Pe + Ph = 0.10 MPa / 14.9 psi

External design temperature
Te = 20.00 °C / 68.00 °F

Modulus of Elasticity at the operating load case temperature
Eyo = 210 350.00 MPa / 30 508 688.2 psi

Modulus of Elasticity at the gasket seating load case temperature
Eyg = 210 350.00 MPa / 30 508 688.2 psi

Bolt loads

Number of bolts
n = 20

Bolt type
= ANSI_TEMA 2-1/2"

Bolt spacing
Bs = 142.16 mm / 5.597 in

Nominal bolt diameter
a = 63.50 mm / 2.500 in

Design bolt load for the operating condition
\( W_o = 0.785G^2P + 2b\pi GmP \) = 47 247 N / 10 621.46 lbf

Design bolt load for the gasket seating (Bolt)
\( W_{bg} = \pi bGy \) = 1 023 113 N / 230 004.85 lbf

External tensile net-section axial force
FA = 0 N / 0 lbf

Absolute value of the external net-section bending moment
ME = 0 N·m / 0 lbf·in

External tensile net-section axial force (test condition)
FAt = 0 N / 0 lbf

External tensile net-section bending moment (test condition)
MEt = 0 N·m / 0 lbf·in

Total minimum required cross-sectional area of the bolts
\[ A_{in} = \max \left( \frac{W_o}{S_{bo}}, \frac{W_{bg}}{S_{bg}} \right) \] = 5 948.3 mm² / 9.220 in²

Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion
Ab = 55 380.5 mm² / 85.840 in²

Maximum bolts area for gasket crush
\[ A_{g,\max} = \frac{2\pi \cdot y \cdot G \cdot N}{S_{tg}} \] = 26 706.3 mm² / 41.395 in²

Design bolt load for the gasket seating (Flange)
\[ W_{bg} = \left( \frac{A_{in} + A_{bo}}{2} \right) S_{bg} \] = 5 274 282 N / 1 185 705.72 lbf
Flange constants

Ratio of the flange outside diameter to the inside diameter

\[ K = \frac{A}{B'} = 1.61575 \]

Stress factor Y

\[ Y = \frac{1}{K - 1}\left[ 0.06845 + \frac{571690}{K^2-1}\right] = 4.22139 \]

Stress factor T

\[ T = \frac{1}{K - 1}\left( 104720 + 194.4K^2 \right) = 1.66082 \]

Stress factor U

\[ U = \frac{136136(K^2-1)(K-1)}{K^2+1} = 4.63888 \]

Stress factor Z

\[ Z = \frac{2}{K^2-1} = 2.24174 \]

Hub length parameter

\[ h_o = \sqrt{\frac{Bg_h}{A}} = 169.04 \text{ mm} \]
\[ X_o = \frac{g_h}{h_o} = 1.26667 \]

Hub thickness ratio

\[ X_h = \frac{g_h}{h_o} = 0.40227 \]

Flange stress factor for integral type flanges

\[ F = \left( 0.0897697 - 0.297012X_g + 9.5257(10^3)X_g^+ \right) + \frac{0.123586(X_g)^2 + 0.0358580X_g(X_h)^2 - 0.0194422X_g(X_h)-0.0181259(X_g)^2+0.0129360(X_h)^2}{0.0377993(X_g)^2 + 0.0273791(X_h)^2 - 0.0500244 + \frac{249188X_h^2}{X_g^2} + \frac{344440}{X_h^2}} = 0.87850 \]

Flange stress factor for integral type flanges

\[ V = \left( 0.0873446 \frac{X_g}{X_h} + 0.0190993 \frac{X_h}{X_g} - 1060682X_h^3 - 149970 \frac{X_h}{X_g} \right) + \frac{0.719413 \frac{X_h}{X_g}}{X_h} = 0.40681 \]

Hub stress correction factor for integral flanges

\[ f = \max \left\{ 0, \frac{0.0927779 - 0.053633X_g + 0.016476X_g^2 + 0.0056226X_h + 0.0347076X_h^2 + 0.418699X_h^3}{1 - 0.496039(10^3)X_g + 162904X_h^2 + 34392X_h^2 + 139052X_h^3} \right\} = 1.0000 \]

Stress factor e

\[ e = \frac{f}{h_o} = 0.00520 \]

Stress factor d

\[ d = \frac{Ug_h h_o}{V} = 3903.371.60229 \]

Stress factor L

\[ L = \frac{L_0 + f \frac{1}{2} \frac{d}{d}} = 2.39272 \]

Gasket load for the operating condition

\[ H_G = W_o - H = 12796 \text{ N} \]
\[ 2876.70 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange

\[ H_o = 0.78585P = 32603 \text{ N} \]
\[ 7329.39 \text{ lbf} \]

Total hydrostatic end force

\[ H = 0.78585P = 34450 \text{ N} \]
\[ 7744.76 \text{ lbf} \]

Difference

\[ H = H_o - H_G = 1848 \text{ N} \]
\[ 415.37 \text{ lbf} \]

Moment arm for load HD

\[ hD = \frac{C-B - G}{2} = 106.50 \text{ mm} \]
\[ 4.193 \text{ in} \]

Moment arm for load HG

\[ h_G = \frac{C - G}{2} = 126.13 \text{ mm} \]
\[ 4.966 \text{ in} \]

Moment arm for load HT

\[ h_T = \frac{C - B + h_G}{2} = 130.56 \text{ mm} \]
\[ 5.140 \text{ in} \]

Average of the hub thicknesses

\[ A_A = AR = 195.50 \text{ mm} \]
\[ 79.67 \text{ in} \]

\[ BB = t = 170.00 \text{ mm} \]
\[ 6.693 \text{ in} \]

\[ CC = h = 68.00 \text{ mm} \]
\[ 2.677 \text{ in} \]

\[ DDG = Gavg = 51.00 \text{ mm} \]
\[ 2.008 \text{ in} \]

Moment of inertia KAB

\[ K_{AB} = (A_p B_p) \left[ \frac{1}{3} - 0.02 (B_p / A_p) \left( 1 - \frac{1}{12} \left( B_p / A_p \right)^2 \right) \right] = 153126579 \text{ mm}^4 \]
\[ 367.888 \text{ in}^4 \]

Moment of inertia KCD

\[ K_{CD} = (C_p D_p) \left[ \frac{1}{3} - 0.0105 (D_p / C_p) \left( 1 - \frac{1}{192} \left( D_p / C_p \right)^2 \right) \right] = 2297581 \text{ mm}^4 \]
\[ 5520 \text{ in}^4 \]
Bending moment of inertia of the flange cross-section
\[ I = \frac{0.0874 MB_{g}}{Y} = 111 \, 738 \, 738 \, \text{mm}^4 = 268.453 \, \text{in}^4 \]
Cross-section polar moment of inertia
\[ I_p = K_{AB} + K_{CD} = 155 \, 424 \, 159 \, \text{mm}^4 = 373.408 \, \text{in}^4 \]

**Flange moments**

Nominal bolt diameter
\[ d_B = 63.50 \, \text{mm} = 2.500 \, \text{in} \]

TEMA Load concentration factor
\[ c_F = \text{MAX} \left[ \frac{h_D}{2 d_B + \frac{d_B}{0.65}} - 1 \right] = 1.00000 \]

Moment factor used to design split rings
\[ F_S = 1.00 \]

Component of the flange design moment resulting from a net section bending moment and/or axial force
\[ M_{ae} = 4M_z \left[ 0.03846 I_p + I \right] \left( \frac{h_D}{(C-2h_D)} \right) + F_S h_D = 0 \, \text{N} \cdot \text{m} = 0 \, \text{lbf} \cdot \text{in} \]

Flange design moment for the operating condition
\[ M_g = W_{gph} F_S = 665 \, 231.6 \, \text{N} \cdot \text{m} = 5 \, 887 \, 795.0 \, \text{lbf} \cdot \text{in} \]

**Flange stresses - operating condition**

Corrected inside diameter of the flange
\[ B' = B' + g_1' = 692.00 \, \text{mm} = 27.244 \, \text{in} \]

Flange hub stress - operating condition
\[ S_H = \frac{f M_g}{L g_2 B_Y} = 0.12 \, \text{MPa} = 17.0 \, \text{psi} \]

Flange radial stress - operating condition
\[ S_R = \frac{(133 \times 1) M_g}{Y M_{g}^{1.5}} = 0.03 \, \text{MPa} = 4.5 \, \text{psi} \]

Flange tangential stress - operating condition
\[ S_T = \frac{Y M_{g}^{1.5}}{Y B} - Z S_R = 0.08 \, \text{MPa} = 10.9 \, \text{psi} \]

\[ \text{SHo} \leq \text{min}[1.5 S_{fo}, 2.5 S_{no}]: \text{Ok} \]
\[ \text{SRo} \leq S_{fo}: \text{Ok} \]
\[ \text{STo} \leq S_{fo}: \text{Ok} \]
\[ \frac{(\text{SHo} + \text{SRo})}{2} \leq S_{fo}: \text{Ok} \]
\[ \frac{(\text{SHo} + \text{STo})}{2} \leq S_{fo}: \text{Ok} \]

**Flange stresses - seating condition**

Flange hub stress - gasket seating condition
\[ S_H = \frac{f M_g}{L g_2 B_Y} = 123.66 \, \text{MPa} = 17935.2 \, \text{psi} \]

Flange radial stress - gasket seating condition
\[ S_R = \frac{(133 \times 1) M_g}{Y M_{g}^{1.5}} = 32.95 \, \text{MPa} = 4779.2 \, \text{psi} \]

Flange tangential stress - gasket seating condition
\[ S_T = \frac{Y M_{g}^{1.5}}{Y B} - Z S_R = 79.15 \, \text{MPa} = 11480.4 \, \text{psi} \]

\[ \text{SHg} \leq \text{min}[1.5 S_{fg}, 2.5 S_{ng}]: \text{Ok} \]
\[ \text{SRg} \leq S_{fg}: \text{Ok} \]
\[ \text{STg} \leq S_{fg}: \text{Ok} \]
\[ \frac{(\text{SHg} + \text{SRg})}{2} \leq S_{fg}: \text{Ok} \]
\[ \frac{(\text{SHg} + \text{STg})}{2} \leq S_{fg}: \text{Ok} \]

**Flange rigidity - operating condition**

Rigidity index factor
\[ K_R = 0.30 \]

Flange rigidity index
\[ J = \frac{5214VM_g}{LE_{ygh_{k}r_{g}}} = 0.00026 \]

\[ J_O \leq 1: \text{Ok} \]

**Flange rigidity - seating condition**

Flange rigidity index
\[ J = \frac{5214VM_g}{LE_{ygh_{k}r_{g}}} = 0.27300 \]

\[ J_G \leq 1: \text{Ok} \]
Minimum Design Metal Temperature (MDMT)

**Internal pressure**

**Welding neck flange - Flange In Tube Side (Bolting)**

- **Material** = SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
- **Curve of fig. 3.7 / 3.8** = None
- **Governing Thickness**
  - \( tg = 63.50 \text{ mm} \)
  - \( 2.500 \text{ in} \)
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)** = -29.00 °C
  - -20.20 °F

**Welding neck flange - Flange In Tube Side**

- **Material** = SA-182 F22 3
- **Curve of fig. 3.7 / 3.8** = B
- **Governing Thickness**
  - \( tg = 45.00 \text{ mm} \)
  - \( 1.772 \text{ in} \)
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)**
- **Impact tests required by Code** = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

**External pressure**

**Welding neck flange - Flange In Tube Side (Bolting)**

- **Material** = SA-193 B16 (Code Case 2655 - using Division 1 stress tables)
- **Curve of fig. 3.7 / 3.8** = None
- **Governing Thickness**
  - \( tg = 63.50 \text{ mm} \)
  - \( 2.500 \text{ in} \)
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)** = -29.00 °C
  - -20.20 °F

**Welding neck flange - Flange In Tube Side**

- **Material** = SA-182 F22 3
- **Curve of fig. 3.7 / 3.8** = B
- **Governing Thickness**
  - \( tg = 45.00 \text{ mm} \)
  - \( 1.772 \text{ in} \)
- **PostWeld Heat Treatment** = No
- **Minimum Design Metal Temperature (MDMT)**
- **Impact tests required by Code** = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

Validation warnings:
- Gasket overloaded: \( Ab > AbMax \)
Calculation temperature

\[ T = 454.00 \, ^\circ\text{C} \quad 849.20 \, ^\circ\text{F} \]

**Material:** SA-182 F22 3 - Forgings

Allowable stress at room temperature

\[ ST = 207.00 \, \text{MPa} \quad 30,022.8 \, \text{psi} \]

Joint efficiency

\[ E = 1.00 \]

Corrosion allowance

\[ c = 0 \, \text{mm} \quad 0 \, \text{in} \]

External corrosion allowance

\[ ce = 0 \, \text{mm} \quad 0 \, \text{in} \]

Wall undertolerance

\[ c' = 0 \, \text{mm} \quad 0 \, \text{in} \]

Inside diameter

\[ D = 635.00 \, \text{mm} \quad 25.000 \, \text{in} \]

Length

\[ L = 778.00 \, \text{mm} \quad 30.630 \, \text{in} \]

Adopted thickness

\[ t = 45.00 \, \text{mm} \quad 1.772 \, \text{in} \]

**Internal pressure**

Allowed stress

\[ S = 149.84 \, \text{MPa} \quad 21,732.5 \, \text{psi} \]

Internal pressure

\[ Pi = 18.44 \, \text{MPa} \quad 2,674.0 \, \text{psi} \]

Overpressure due to static head

\[ Ph = 0 \, \text{MPa} \quad 0 \, \text{psi} \]

Calculation pressure

\[ P = Pi + Ph = 18.44 \, \text{MPa} \quad 2,674.0 \, \text{psi} \]

Required thickness

\[ t_r = \frac{D + 2(c + ce) - (t + c + ce + c')}{2} \quad (t + c + ce + c') = 41.57 \, \text{mm} \quad 1.637 \, \text{in} \]

\[ t \geq t_r: \quad \text{Ok} \]

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

New & cold

\[ = 27.44 \, \text{MPa} \quad 3,979.4 \, \text{psi} \]

Hot & corroded

\[ = 19.86 \, \text{MPa} \quad 2,880.6 \, \text{psi} \]

**External pressure**

External pressure

\[ Pe = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]

External static head

\[ Ph = 0 \, \text{MPa} \quad 0 \, \text{psi} \]

Calculation pressure

\[ P = Pe + Ph = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]

External design temperature

\[ Te = 20.00 \, ^\circ\text{C} \quad 68.00 \, ^\circ\text{F} \]

Outside diameter

\[ Do = 725.00 \, \text{mm} \quad 28.543 \, \text{in} \]

Max unsupported length

\[ L = 778.00 \, \text{mm} \quad 30.630 \, \text{in} \]

Modulus of elasticity

\[ Ey = 210,350.00 \, \text{MPa} \quad 30,508,688.2 \, \text{psi} \]

Shell parameter

\[ M_x = \frac{L}{\sqrt{R_e(t - c - ce)}} = 6.09143 \]

\[ C_h = \frac{M_x - 0.579}{0.92} = 0.16690 \]

Elastic buckling stress

\[ F_{he} = 16 \cdot C_h \cdot Ey \cdot (t - c - ce - c') \cdot D_o = 3486.44 \, \text{MPa} \quad 505,664.7 \, \text{psi} \]

Buckling stress

\[ F_{ic} = Sy = 310.00 \, \text{MPa} \quad 44,961.7 \, \text{psi} \]

Yield strength at design temperature

\[ Sy = 310.00 \, \text{MPa} \quad 44,961.7 \, \text{psi} \]

Design factor

\[ FS = 1.66700 \]

Hoop compressive membrane stress

\[ F_{ha} = F_{ic} / FS = 185.96 \, \text{MPa} \quad 26,971.6 \, \text{psi} \]

Allowable external pressure in the absence of other loads

\[ P_o = 2F_{ha} \left( t - c - ce - c' \right) \cdot D_o = 23.09 \, \text{MPa} \quad 3,348.2 \, \text{psi} \]

Minimum required thickness

\[ t_r = 2.20 \, \text{mm} \quad 0.087 \, \text{in} \]

\[ t \geq t_r: \quad \text{Ok} \]

**Maximum allowable external pressures**

New & cold

\[ = 23.09 \, \text{MPa} \quad 3,348.2 \, \text{psi} \]

Hot & corroded

\[ = 23.09 \, \text{MPa} \quad 3,348.2 \, \text{psi} \]
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure

**Nozzle - Bocchello Out Tube Side (Nozzle to wall - shell)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>85.00 mm 3.346 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

#### External pressure

**Nozzle - Bocchello Out Tube Side (Nozzle to wall - shell)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>85.00 mm 3.346 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*
## Welding neck flange - Flange Out Tube Side

*According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units*

### Design data

- **Internal design temperature**
  \[ T = 454.00 \, ^\circ\text{C} \; 849.20 \, ^\circ\text{F} \]
- **Internal design pressure**
  \[ P = 18.44 \, \text{MPa} \; 2674.0 \, \text{psi} \]
- **External design temperature**
  \[ T_e = 20.00 \, ^\circ\text{C} \; 68.00 \, ^\circ\text{F} \]
- **External design pressure**
  \[ P_e = 0.10 \, \text{MPa} \; 14.9 \, \text{psi} \]
- **Joint efficiency**
  \[ E = 1.00 \]

### Flange material
- SA-182 F22 3 - Forgings

### Shell material
- SA-182 F22 3 - Forgings

### Bolting material
- SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting

### Gasket
- Grooved Metal - Stainless steels and nickel-base alloys

### Allowable stresses

<table>
<thead>
<tr>
<th>Condition</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Design condition</strong></td>
<td>( S_{f0} = 149.84 , \text{MPa} ; 21732.5 , \text{psi} )</td>
<td>( S_{n0} = 149.84 , \text{MPa} ; 21732.5 , \text{psi} )</td>
<td>( S_{b0} = 161.44 , \text{MPa} ; 23414.9 , \text{psi} )</td>
</tr>
<tr>
<td><strong>Seating condition</strong></td>
<td>( S_{f0} = 207.00 , \text{MPa} ; 30022.8 , \text{psi} )</td>
<td>( S_{n0} = 207.00 , \text{MPa} ; 30022.8 , \text{psi} )</td>
<td>( S_{b0} = 172.00 , \text{MPa} ; 24946.5 , \text{psi} )</td>
</tr>
<tr>
<td><strong>Test condition</strong></td>
<td>( S_{f0} = / )</td>
<td>( S_{n0} = / )</td>
<td>( S_{b0} = / )</td>
</tr>
</tbody>
</table>

### Internal pressure
\[ P_d = 18.44 \, \text{MPa} \; 2674.0 \, \text{psi} \]

### Overpressure due to static head
\[ P_h = 0 \, \text{MPa} \; 0 \, \text{psi} \]

### Calculation pressure
\[ P = 18.44 \, \text{MPa} \; 2674.0 \, \text{psi} \]

### Modulus of Elasticity at the operating load case temperature
\[ E_{yo} = 179600.00 \, \text{MPa} \; 26048777.7 \, \text{psi} \]

### Modulus of Elasticity at the gasket seating load case temperature
\[ E_{yg} = 210350.00 \, \text{MPa} \; 30508688.2 \, \text{psi} \]

### Corrosion allowance
\[ c = 0 \, \text{mm} \; 0 \, \text{in} \]

### Flange external diameter
\[ A = 1026.00 \, \text{mm} \; 40.394 \, \text{in} \]

### Inside diameter
\[ B = 635.00 \, \text{mm} \; 25.000 \, \text{in} \]

### Bolt circle
\[ C = 905.00 \, \text{mm} \; 35.630 \, \text{in} \]

### Flange thickness
\[ t = 170.00 \, \text{mm} \; 6.693 \, \text{in} \]

### Mean gasket diameter
\[ G_{\text{mean}} = 651.00 \, \text{mm} \; 25.630 \, \text{in} \]

### Thickness of the hub at the small end
\[ g_0 = 45.00 \, \text{mm} \; 1.772 \, \text{in} \]

### Thickness of the hub at the large end
\[ g_1 = 57.00 \, \text{mm} \; 2.244 \, \text{in} \]

### Thickness of the hub at the small end (Corroded)
\[ g_0' = g_0 - c = 45.00 \, \text{mm} \; 1.772 \, \text{in} \]

### Thickness of the hub at the large end (Corroded)
\[ g_1' = g_1 - c = 57.00 \, \text{mm} \; 2.244 \, \text{in} \]

### Hub length
\[ h = 68.00 \, \text{mm} \; 2.677 \, \text{in} \]

### Gasket parameters

- **Gasket factor**
  \[ m = 4.25 \]

- **Gasket factor**
  \[ y = 70.00 \, \text{MPa} \; 10152.6 \, \text{psi} \]

- **Gasket contact width**
  \[ N = 16.00 \, \text{mm} \; 0.630 \, \text{in} \]

- **Basic gasket seating width**
  \[ b_0 = \frac{N}{2} = 8.00 \, \text{mm} \; 0.315 \, \text{in} \]

- **Conversion factor for length**
  \[ C_{ul} = 25.4000 \]

- **Effective gasket contact width**
  \[ b = 0.5 C_{ul} \frac{b_0}{C_{vl}} = 7.13 \, \text{mm} \; 0.281 \, \text{in} \]

- **Outside diameter of the gasket contact area**
  \[ G_c = G_{\text{mean}} + N = 667.00 \, \text{mm} \; 26.260 \, \text{in} \]

- **Diameter at the location of the gasket load reaction**
  \[ G = G_c - 2b = 652.75 \, \text{mm} \; 25.699 \, \text{in} \]
### Bolt loads

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of bolts</td>
<td>20</td>
</tr>
<tr>
<td>Bolt spacing</td>
<td>142.16 mm</td>
</tr>
<tr>
<td>Nominal bolt diameter</td>
<td>63.50 mm</td>
</tr>
<tr>
<td>Design bolt load for the operating condition</td>
<td>8456915 N</td>
</tr>
<tr>
<td>Design bolt load for the test condition</td>
<td>0 N</td>
</tr>
<tr>
<td>Design bolt load for the gasket seating (Bolt)</td>
<td>1023113 N</td>
</tr>
<tr>
<td>External tensile net-section axial force</td>
<td>0 N</td>
</tr>
<tr>
<td>External tensile net-section bending moment</td>
<td>0 N m</td>
</tr>
<tr>
<td>Maximum bolts area for gasket crush</td>
<td>25706.3 mm²</td>
</tr>
<tr>
<td>Design bolt load for the gasket seating (Flange)</td>
<td>9267772 N</td>
</tr>
<tr>
<td>Total minimum required cross-sectional area of the bolts</td>
<td>52384.3 mm²</td>
</tr>
<tr>
<td>Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion</td>
<td>55380.5 mm²</td>
</tr>
<tr>
<td>Maximum bolts area for gasket crush</td>
<td>26706.3 mm²</td>
</tr>
</tbody>
</table>

### Flange constants

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ratio of the flange outside diameter to the inside diameter</td>
<td>1.61575</td>
</tr>
<tr>
<td>Stress factor Y</td>
<td>4.22139</td>
</tr>
<tr>
<td>Stress factor T</td>
<td>1.66082</td>
</tr>
<tr>
<td>Stress factor U</td>
<td>4.63888</td>
</tr>
<tr>
<td>Stress factor Z</td>
<td>2.24174</td>
</tr>
<tr>
<td>Hub length parameter</td>
<td>169.04 mm</td>
</tr>
<tr>
<td>Hub thickness ratio</td>
<td>1.26667</td>
</tr>
<tr>
<td>Hub length ratio</td>
<td>0.40227</td>
</tr>
<tr>
<td>Flange stress factor for integral type flanges</td>
<td>0.87580</td>
</tr>
<tr>
<td>Flange stress factor for integral type flanges</td>
<td>0.40681</td>
</tr>
<tr>
<td>Hub stress correction factor for integral flanges</td>
<td>1.00000</td>
</tr>
<tr>
<td>Stress factor e</td>
<td>0.00520</td>
</tr>
</tbody>
</table>
Stress factor \( d \)
\[
d = \frac{U g h}{P} = 3903371.60229
\]

Stress factor \( L \)
\[
L = \frac{1}{P} \left( \frac{1}{P} + \frac{1}{D} \right) = 2.39272
\]

Gasket load for the operating condition
\[
H_G = W_o - H = 2290461 \text{ N} \quad 514916.01 \text{ lbf}
\]

Total hydrostatic end force on the area inside of the flange
\[
H_G = 0.785 R_P^2 P = 5835736 \text{ N} \quad 1311925.51 \text{ lbf}
\]

Total hydrostatic end force
\[
H = 0.785 Q P = 6166454 \text{ N} \quad 1386273.97 \text{ lbf}
\]

Difference
\[
H = H - H_G = 330718 \text{ N} \quad 74348.46 \text{ lbf}
\]

Moment arm for load HD
\[
h_D = \frac{2}{7} C - B - g_1 = 106.50 \text{ mm} \quad 4.193 \text{ in}
\]

Moment arm for load HG
\[
h_G = \frac{2}{7} C - G = 126.13 \text{ mm} \quad 4.966 \text{ in}
\]

Moment arm for load HT
\[
h_T = \frac{1}{2} \left( \frac{C - B}{2} + h_3 \right) = 130.56 \text{ mm} \quad 5.140 \text{ in}
\]

Average of the hub thicknesses
\[
A = \frac{A_R - 0.5 (A - B) }{2} = 195.50 \text{ mm} \quad 7.697 \text{ in}
\]

Moment of inertia \( K_{AB} \)
\[
K_{AB} = (A_p B_p) \left( \frac{1}{3} - 0.021 \frac{B_p}{A_p} \left( \frac{1}{2} \frac{B_p}{A_p} \right)^{2} \times \frac{1}{2} \frac{B_p}{A_p} \right) = 153126579 \text{ mm}^4 \quad 367888 \text{ in}^4
\]

Moment of inertia \( K_{CD} \)
\[
K_{CD} = (C_p D_p) \left( \frac{1}{3} - 0.0105 \frac{D_p}{C_p} \left( \frac{1}{2} \frac{D_p}{C_p} \right)^{2} \times \frac{1}{2} \frac{D_p}{C_p} \right) = 2297581 \text{ mm}^4 \quad 5520 \text{ in}^4
\]

Bending moment of inertia of the flange cross-section
\[
I = \frac{1}{4} \frac{C_G}{V} = 111738738 \text{ mm}^4 \quad 268453 \text{ in}^4
\]

Cross-section polar moment of inertia
\[
I_p = K_{AB} + K_{CD} = 155424159 \text{ mm}^4 \quad 373408 \text{ in}^4
\]

Flange moments

Nominal bolt diameter
\[
D_B = 63.50 \text{ mm} \quad 2.500 \text{ in}
\]

TEMA Load concentration factor
\[
cF = \text{MAX} \left[ \frac{2C}{2dB + m/3} \right] = 1.00000
\]

Moment factor used to design split rings
\[
F_B = 0 \text{ N·m} \quad 0 \text{ lbf·in}
\]

Component of the flange design moment resulting from a net section bending moment and/or axial force
\[
M_{oe} = 4M_E \left[ \frac{I}{0.38466I_F + I} \right] \left[ \frac{h_D}{(C - 2h_D)} \right] + F_B I_D = 953575.6 \text{ N·m} \quad 8439854.3 \text{ lbf·in}
\]

Flange design moment for the operating condition
\[
M_B = cF \left( \frac{W_o (C - G) F_2}{2} \right) = 1168920.1 \text{ N·m} \quad 10345814.1 \text{ lbf·in}
\]

Flange design moment for the gasket seating condition
\[\text{Flange stresses - operating condition}\]

Corrected inside diameter of the flange
\[
B_1' = B' + q 1' = 692.00 \text{ mm} \quad 27.244 \text{ in}
\]

Flange hub stress - operating condition
\[
S_H = \frac{fM_E}{L g B_1} = 177.26 \text{ MPa} \quad 25709.2 \text{ psi}
\]

Flange radial stress - operating condition
\[
S_R = \left( \frac{1338}{1 + 1} \right) M_E = 47.23 \text{ MPa} \quad 6850.7 \text{ psi}
\]

Flange tangential stress - operating condition
\[
S_T = \frac{1338}{1 + 1} M_E = 113.46 \text{ MPa} \quad 16456.5 \text{ psi}
\]

\[\text{Sho} \leq \min[1.5 S_{fo}, 2.5 S_{no}]: \text{ Ok}\]
\[\text{Sro} \leq S_{fo}: \text{ Ok}\]
\[\text{Sto} \leq S_{fo}: \text{ Ok}\]
\[\text{(Sho + Sro) } / 2 \leq S_{fo}: \text{ Ok}\]
\[\text{(Sho + Sto) } / 2 \leq S_{fo}: \text{ Ok}\]
Flange stresses - seating condition

Flange hub stress - gasket seating condition
\[ S_H = \frac{f M_g}{L g^2 R_1} = 217.29 \text{ MPa} \quad 31,515.1 \text{ psi} \]

Flange radial stress - gasket seating condition
\[ S_R = \frac{(133e+1) M_g}{133 R^2} = 57.90 \text{ MPa} \quad 8,397.8 \text{ psi} \]

Flange tangential stress - gasket seating condition
\[ S_T = \frac{M_g}{R B} - Z S_R = 139.09 \text{ MPa} \quad 20,172.9 \text{ psi} \]

\[ \text{SHg} \leq \min[1.5 S_{fg}, 2.5 S_{ng}]: \text{ Ok} \]
\[ \text{SRg} \leq S_{fg}: \text{ Ok} \]
\[ \text{STg} \leq S_{fg}: \text{ Ok} \]
\[ (\text{SHg} + \text{SRg}) / 2 \leq S_{fg}: \text{ Ok} \]
\[ (\text{SHg} + \text{STg}) / 2 \leq S_{fg}: \text{ Ok} \]

Flange rigidity - operating condition

Rigidity index factor
\[ K_R = 0.30 \]

Flange rigidity index
\[ J = \frac{5214 V M_{\phi}}{LE_y g^2 K_R \psi} = 0.45833 \quad \text{Jo} \leq 1: \text{ Ok} \]

Flange rigidity - seating condition

Flange rigidity index
\[ J = \frac{5214 V M_{\phi}}{LE_y g^2 K_R \psi} = 0.47970 \quad \text{Jg} \leq 1: \text{ Ok} \]

Hub thickness
Minimum hub thickness as cylindrical shell
\[ \text{th,min} = 41.57 \text{ mm} \quad 1.637 \text{ in} \]

\[ g_0 \geq \text{th,min}: \text{ Ok} \]

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)
\[ = 26.18 \text{ MPa} \quad 3,796.5 \text{ psi} \]

Hot & corroded (flange)
\[ = 19.00 \text{ MPa} \quad 2,756.4 \text{ psi} \]

New & cold (bolts)
\[ = 20.77 \text{ MPa} \quad 3,011.8 \text{ psi} \]

Hot & corroded (bolts)
\[ = 19.49 \text{ MPa} \quad 2,826.9 \text{ psi} \]
### External pressure

<table>
<thead>
<tr>
<th>Allowable stresses</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Design condition</strong></td>
<td>Sfo=207.00 MPa / 30 022.8 psi</td>
<td>Sno=207.00 MPa / 30 022.8 psi</td>
<td>Sbo=172.00 MPa / 24 946.5 psi</td>
</tr>
<tr>
<td><strong>Seating condition</strong></td>
<td>Sfg=207.00 MPa / 30 022.8 psi</td>
<td>Sng=207.00 MPa / 30 022.8 psi</td>
<td>Sbg=172.00 MPa / 24 946.5 psi</td>
</tr>
</tbody>
</table>

- **External design pressure** \( P_e = 0.10 \text{ MPa} \), \( 14.9 \text{ psi} \)
- **Overpressure due to static head** \( \Phi_h = 0 \text{ MPa} \), \( 0 \text{ psi} \)
- **Calculation pressure** \( P = P_e + \Phi_h = 0.10 \text{ MPa} \), \( 14.9 \text{ psi} \)
- **External design temperature** \( T_e = 20.00 ^\circ C \), \( 68.00 ^\circ F \)
- **Modulus of Elasticity at the operating load case temperature** \( E_{yo} = 210 \text{,}350.00 \text{ MPa} \), \( 30\text{,}508 \text{,}688.2 \text{ psi} \)
- **Modulus of Elasticity at the gasket seating load case temperature** \( E_{yg} = 210 \text{,}350.00 \text{ MPa} \), \( 30\text{,}508 \text{,}688.2 \text{ psi} \)

### Bolt loads

- **Number of bolts** \( n = 20 \)
- **Bolt type** = ANSI_TEMA 2-1/2"
- **Bolt spacing** \( B_s = 142.16 \text{ mm} \), \( 5.597 \text{ in} \)
- **Nominal bolt diameter** \( a = 63.50 \text{ mm} \), \( 2.500 \text{ in} \)
- **Design bolt load for the operating condition** \( W_0 = 0.785G^2P + 2b\pi GmP^2 = 47\text{,}247 \text{ N} \), \( 10\text{,}621.46 \text{ lbf} \)
- **Design bolt load for the gasket seating (Bolt)** \( W_g = \pi b G y = 1\text{,}023\text{,}113 \text{ N} \), \( 230\text{,}004.85 \text{ lbf} \)
- **External tensile net-section axial force** \( FA = 0 \text{ N} \), \( 0 \text{ lbf} \)
- **Absolute value of the external net-section bending moment** \( ME = 0 \text{ Nm} \), \( 0 \text{ lbf.in} \)
- **External tensile net-section axial force (test condition)** \( FA_t = 0 \text{ N} \), \( 0 \text{ lbf} \)
- **External tensile net-section bending moment (test condition)** \( ME_t = 0 \text{ Nm} \), \( 0 \text{ lbf.in} \)
- **Total minimum required cross-sectional area of the bolts** \( A_{m} = \max \left[ \frac{W_0 + F_1 + \frac{4W_g}{a}}{S_{bo}}, \frac{W_g}{S_{bg}} \right] = 5\text{,}948.3 \text{ mm}^2 \), \( 9.220 \text{ in}^2 \)
- **Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion** \( A_b = 55\text{,}380.5 \text{ mm}^2 \), \( 85.840 \text{ in}^2 \)
- **Maximum bolts area for gasket crush** \( A_{mg} = \frac{2\pi \cdot y \cdot G \cdot N}{S_{pg}} = 26\text{,}706.3 \text{ mm}^2 \), \( 41.395 \text{ in}^2 \)
- **Design bolt load for the gasket seating (Flange)** \( W_g = \left( \frac{A_m + A_b}{2} \right)S_{bg} = 5\text{,}274\text{,}282 \text{ N} \), \( 1\text{,}185\text{,}705.72 \text{ lbf} \)
Flange constants

Ratio of the flange outside diameter to the inside diameter

\[ K = \frac{A}{B'} = 1.61575 \]

Stress factor \( Y \)

\[ Y = \frac{1}{K} \left[ \frac{0.66845 + 5.71690}{K^2 - 1} \right] \]

\[ Y = 4.22139 \]

Stress factor \( T \)

\[ T = \frac{104720 + 948K^2}{1655246 \log(K^2 - 1)} \]

\[ T = 1.66082 \]

Stress factor \( U \)

\[ U = \frac{13636}{1655246 \log(K^2 - 1)} \]

\[ U = 4.63888 \]

Stress factor \( Z \)

\[ Z = \frac{K^2 - 1}{K^2 + 1} \]

\[ Z = 2.24174 \]

Hub length parameter

\[ h_e = \sqrt{\frac{B_g}{K} Y} = 169.04 \text{ mm} \]

\[ h_e = 6.655 \text{ in} \]

Hub thickness ratio

\[ X_g = \frac{g_1}{g_0} \]

\[ X_g = 1.26667 \]

Hub length ratio

\[ X_h = \frac{h}{g_0} \]

\[ X_h = 0.40227 \]

Flange stress factor for integral type flanges

\[ F = \frac{0.097697 - 0.297012X_g + 9.5257(10^3)X_g + 0.123866(X_g)^2 + 0.0358580 (X_h)^2 - 0.194422(X_g)(X_h) - 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - 0.0377993(X_g)^2 + 0.0273791(X_h)^2 - 0.500244 + \frac{617791}{X_g} - 187071X_h^2 - \frac{334400}{X_h} + 249189X_h^2}{(X_g)^3 + (X_h)^3} \]

\[ F = 0.87850 \]

Flange stress factor for integral type flanges

\[ V = \frac{0.097697 - 0.297012X_g + 9.5257(10^3)X_g + 0.123866(X_g)^2 + 0.0358580 (X_h)^2 - 0.194422(X_g)(X_h) - 0.0181259(X_g)^3 + 0.0129360(X_h)^3 - 0.0377993(X_g)^2 + 0.0273791(X_h)^2 - 0.500244 + \frac{617791}{X_g} - 187071X_h^2 - \frac{334400}{X_h} + 249189X_h^2}{(X_g)^3 + (X_h)^3} \]

\[ V = 0.40681 \]

Hub stress correction factor for integral flanges

\[ f = \max \left( \frac{0.0972779 - 0.0336633X_g + 0.964176X_g^2 + 0.0566286X_h - 0.0347076X_h^2 - 0.418699X_h^2}{1 - 596093(10^3)X_g + 162904X_g^2 + 349329X_h^2 + 139052X_h^2} \right) \]

\[ f = 1.00000 \]

Stress factor \( e \)

\[ e = \frac{u}{Y} \]

\[ e = 0.00520 \]

Stress factor \( d \)

\[ d = \frac{u}{Y} \]

\[ d = 3903.3716.0229 \]

Stress factor \( L \)

\[ L = \frac{\pi l + 1}{d} \]

\[ L = 2.39272 \]

Gasket load for the operating condition

\[ H_G = W_e H \]

\[ H_G = 12.796 \text{ N} \]

\[ H_G = 2876.70 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange

\[ H = 0.07852P \]

\[ H = 32.603 \text{ N} \]

\[ H = 7329.39 \text{ lbf} \]

Total hydrostatic end force

\[ H = 34.450 \text{ N} \]

\[ H = 7744.76 \text{ lbf} \]

Difference

\[ H_T = H - H_G \]

\[ H_T = 1.848 \text{ N} \]

\[ H_T = 415.37 \text{ lbf} \]

Moment arm for load HD

\[ h_D = \frac{C - B}{2} + \frac{G}{2} \]

\[ h_D = 106.50 \text{ mm} \]

\[ h_D = 4.193 \text{ in} \]

Moment arm for load HG

\[ h_G = \frac{C - G}{2} \]

\[ h_G = 126.13 \text{ mm} \]

\[ h_G = 4.966 \text{ in} \]

Moment arm for load HT

\[ h_T = \frac{C - G}{2} + h_G \]

\[ h_T = 130.56 \text{ mm} \]

\[ h_T = 5.140 \text{ in} \]

Average of the hub thicknesses

\[ A_A = A_R = 195.50 \text{ mm} \]

\[ A_A = 769.79 \text{ in} \]

\[ B_B = b = 170.00 \text{ mm} \]

\[ B_B = 6.693 \text{ in} \]

\[ C_C = h = 68.00 \text{ mm} \]

\[ C_C = 2.677 \text{ in} \]

\[ D_D = G_{avg} = 51.00 \text{ mm} \]

\[ D_D = 2.008 \text{ in} \]

Moment of inertia \( K_{AB} \)

\[ K_{AB} = \left( A_g B_g \right) \left[ \frac{1}{3} - \frac{0.021B_g}{A_g} \left( \frac{1}{12} \left( \frac{B_g}{A_g} \right)^2 \right) \right] \]

\[ K_{AB} = 153.126.579 \text{ mm}^4 \]

\[ K_{AB} = 367.888 \text{ in}^4 \]

Moment of inertia \( K_{CD} \)

\[ K_{CD} = \left( C_D^2 D_D \right) \left[ \frac{1}{3} - 0.0105 \left( \frac{D_D}{C_D} \right) \left( \frac{1}{192} \left( \frac{D_D}{C_D} \right)^2 \right) \right] \]

\[ K_{CD} = 2297.581 \text{ mm}^4 \]

\[ K_{CD} = 5.520 \text{ in}^4 \]
Bending moment of inertia of the flange cross-section

\[ I = \frac{0.0874Mg_0}{YgB} \]

Cross-section polar moment of inertia

\[ I_p = K_{AB} + K_{CD} \]

**Flange moments**

Nominal bolt diameter

\[ dB = 63.50 \text{ mm} \quad 2.500 \text{ in} \]

TEMA Load concentration factor

\[ cF = \text{MAX} \left[ \frac{\frac{h_D}{C}}{2dB + \frac{w}{m6.5}}, 1 \right] \]

Moment factor used to design split rings

\[ F_s = 1.00 \]

Component of the flange design moment resulting from a net section bending moment and/or axial force

\[ M_{oe} = 4M_0 \left[ \frac{I}{0.03846I + I} \right] \]

\[ + F_s \frac{h_D}{(C-2h_D)} \]

\[ = 0 \text{ N·m} \quad 0 \text{ lbf·in} \]

Flange design moment for the operating condition

\[ M_g = W_gF_s \]

\[ = 665.231.6 \text{ N·m} \quad 5 \text{ 887 795.0 lbf·in} \]

Flange design moment for the gasket seating condition

\[ M_g = W_gF_s \]

**Flange stresses - operating condition**

Corrected inside diameter of the flange

\[ B1' = B + g1' = 692.00 \text{ mm} \quad 27.244 \text{ in} \]

Flange hub stress - operating condition

\[ S_H = \frac{fM_g}{Lg2b_1} \]

\[ = 0.12 \text{ MPa} \quad 17.0 \text{ psi} \]

Flange radial stress - operating condition

\[ S_R = \left( \frac{133e+1}{M_g} \right) \]

\[ = 0.03 \text{ MPa} \quad 4.5 \text{ psi} \]

Flange tangential stress - operating condition

\[ S_T = \frac{YM_g}{r^2} - ZS_R \]

\[ = 0.08 \text{ MPa} \quad 10.9 \text{ psi} \]

**Flange stresses - seating condition**

Flange hub stress - gasket seating condition

\[ S_H = \frac{fM_g}{Lg2b_1} \]

\[ = 123.66 \text{ MPa} \quad 17 \text{ 935.2 psi} \]

Flange radial stress - gasket seating condition

\[ S_R = \left( \frac{133e+1}{M_g} \right) \]

\[ = 32.95 \text{ MPa} \quad 4 \text{ 779.2 psi} \]

Flange tangential stress - gasket seating condition

\[ S_T = \frac{YM_g}{r^2} - ZS_R \]

\[ = 79.15 \text{ MPa} \quad 11 \text{ 480.4 psi} \]

**Flange rigidity - operating condition**

Rigidity index factor

\[ KR = 0.30 \]

\[ J = \frac{5214VM_g}{LEy^2g_2\frac{k}{g_{th}}} \]

\[ = 0.00026 \]

\[ Jo \leq 1: \text{ Ok} \]

**Flange rigidity - seating condition**

\[ J = \frac{5214VM_g}{LEy^2g_2\frac{k}{g_{th}}} \]

\[ = 0.27300 \]

\[ Jg \leq 1: \text{ Ok} \]
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16 (Code Case 2655 - using Division 1 stress tables)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>63.50 mm, 2.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-29.00 °C, -20.20 °F</td>
</tr>
</tbody>
</table>

#### Welding neck flange - Flange Out Tube Side (Bolting)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>45.00 mm, 1.772 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

#### External pressure

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16 (Code Case 2655 - using Division 1 stress tables)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>63.50 mm, 2.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-29.00 °C, -20.20 °F</td>
</tr>
</tbody>
</table>

#### Welding neck flange - Flange Out Tube Side (Bolting)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>45.00 mm, 1.772 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

### Validation warnings:

- Gasket overloaded: Ab > AbMax
### Welding neck flange - Shell Flange

*According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units*

#### Design data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal design temperature</td>
<td>$T = 420.00 \degree C$</td>
</tr>
<tr>
<td>Internal design pressure</td>
<td>$P = 14.22 \text{ MPa}$</td>
</tr>
<tr>
<td>External design temperature</td>
<td>$T_e = 20.00 \degree C$</td>
</tr>
<tr>
<td>External design pressure</td>
<td>$P_e = 0.10 \text{ MPa}$</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>$E = 1.00$</td>
</tr>
</tbody>
</table>

#### Flange material

- **SA-387 22 2 - Plate**

#### Shell material

- **SA-387 22 2 - Plate**

#### Bolting material

- **SA-193 B16** (Code Case 2655 - using Division 1 stress tables) - Bolting

#### Gasket

- **Grooved Metal - Stainless steels and nickel-base alloys**

#### Allowable stresses

<table>
<thead>
<tr>
<th>Condition</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design condition</td>
<td>$S_{fo} = 158.60 \text{ MPa / 23 003.0 psi}$</td>
<td>$S_{no} = 158.60 \text{ MPa / 23 003.0 psi}$</td>
<td>$S_{bo} = 138.00 \text{ MPa / 20 015.2 psi}$</td>
</tr>
<tr>
<td>Seating condition</td>
<td>$S_{fg} = 207.00 \text{ MPa / 30 022.8 psi}$</td>
<td>$S_{ng} = 207.00 \text{ MPa / 30 022.8 psi}$</td>
<td>$S_{bg} = 138.00 \text{ MPa / 20 015.2 psi}$</td>
</tr>
<tr>
<td>Test condition</td>
<td>$S_{ft} =$</td>
<td>$S_{nt} =$</td>
<td>$S_{bt} =$</td>
</tr>
</tbody>
</table>

#### Gasket parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasket factor</td>
<td>$m = 4.25$</td>
</tr>
<tr>
<td>Gasket factor</td>
<td>$y = 1 000.00 \text{ MPa}$</td>
</tr>
<tr>
<td>Gasket contact width</td>
<td>$N = 20.00 \text{ mm}$</td>
</tr>
<tr>
<td>Basic gasket seating width</td>
<td>$b_0 = 10.00 \text{ mm}$</td>
</tr>
<tr>
<td>Conversion factor for length</td>
<td>$C_{ul} = 25.40000$</td>
</tr>
<tr>
<td>Effective gasket contact width</td>
<td>$b = 0.5C_{ul} \sqrt{\frac{b_0}{C_{ul}}}$</td>
</tr>
<tr>
<td>Outside diameter of the gasket contact area</td>
<td>$G_c = G_{mean} + N$</td>
</tr>
<tr>
<td>Diameter at the location of the gasket load reaction</td>
<td>$G = G_c - 2b$</td>
</tr>
</tbody>
</table>

---

**Walter Tosto S.p.A.**  
via Erasmo Piaggio  
Chieti  
Telephone, Fax  
Website, Email address  
Date ________ Calc. ________ Contr._______ Appr. ________
### Bolt loads

- **Number of bolts**
  \[ n = 28 \]

- **Bolt spacing**
  \[ B_s = 190.74 \text{ mm} = 7.509 \text{ in} \]

- **Nominal bolt diameter**
  \[ a = 114.30 \text{ mm} = 4.500 \text{ in} \]

- **Design bolt load for the operating condition**
  \[ W_o = 0.785G^2P + 2\pi aGmP = 22,832,134 \text{ N} = 513,867.40 \text{ lbf} \]

- **Design bolt load for the test condition**
  \[ W_I = 0.785G^2P + 2\pi aGmP = 0 \text{ N} = 0 \text{ lbf} \]

- **Design bolt load for the gasket seating (Bolt)**
  \[ W_{G} = \pi bGy = 32,571,288 \text{ N} = 7,322,316.25 \text{ lbf} \]

- **External tensile net-section axial force**
  \[ F_A = 0 \text{ N} = 0 \text{ lbf} \]

- **Absolute value of the external net-section bending moment**
  \[ M_E = 0 \text{ N} \cdot \text{m} = 0 \text{ lbf} \cdot \text{in} \]

- **External tensile net-section axial force (test condition)**
  \[ F_{A_t} = 0 \text{ N} = 0 \text{ lbf} \]

- **External tensile net-section bending moment (test condition)**
  \[ M_{E_t} = 0 \text{ N} \cdot \text{m} = 0 \text{ lbf} \cdot \text{in} \]

- **Total minimum required cross-sectional area of the bolts**
  \[ A_m = \frac{2\pi \cdot G \cdot N}{S_{bt}} = 236,023.8 \text{ mm}^2 = 365,838 \text{ in}^2 \]

- **Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion**
  \[ A_{b} = 261,393.0 \text{ mm}^2 = 405,160 \text{ in}^2 \]

- **Maximum bolts area for gasket crush**
  \[ A_{b,max} = \frac{2\pi \cdot G \cdot N}{S_{bt}} = 1,184,756.2 \text{ mm}^2 = 1,836,376 \text{ in}^2 \]

- **Design bolt load for the gasket seating (Flange)**
  \[ W_{G} = \frac{(A_m + A_{b})}{2} \cdot S_{bt} = 34,321,763 \text{ N} = 7,715,838.57 \text{ lbf} \]

### Flange constants

- **Ratio of the flange outside diameter to the inside diameter**
  \[ K = \frac{A}{B'} = 1.48090 \]

- **Stress factor Y**
  \[ Y = \frac{0.66845 + 571690}{K^2 - 1} = 5.11631 \]

- **Stress factor T**
  \[ T = \frac{104,720 + 194,482K}{(1855246 + 184482K)(K - 1)} = 1.71899 \]

- **Stress factor U**
  \[ U = \frac{136136(K^2 - 1)(K - 1)}{K^2 + (K^2 - 1)} = 5.62231 \]

- **Stress factor Z**
  \[ Z = \frac{1}{K^2 - 1} = 2.67634 \]

- **Hub length parameter**
  \[ h_c = \sqrt{\frac{E_{sc}}{G_{sc}}} = 282.04 \text{ mm} = 11.104 \text{ in} \]

- **Hub thickness ratio**
  \[ X_s = \frac{g_{1}}{g_{s}} = 1.56452 \]

- **Hub length ratio**
  \[ X_h = \frac{h_c}{h_s} = 0.42547 \]

- **Flange stress factor for integral type flanges**
  \[ F = \frac{0.097697 - 0.297012X_g + 9.5257 \times 19^3X_g}{0.123368(X_g)^2 + 0.0358580(X_h)^2 - 0.194422(X_g)(X_h)} + \frac{0.0181259(X_g)^3 + 0.0129360(X_h)^3}{0.0377693(X_g)^2 + 0.0273791(X_h)^2 + 0.050244 + 0.279918 + 18707X_{K' - 804494} + 249188X_{K'}^3} = 0.85520 \]

- **Flange stress factor for integral type flanges**
  \[ F' = \frac{0.873446 + 0.14903}{X_h^2} + 106082X_h^2 - 146970(X_h^2) + 0.31733 \]

- **Hub stress correction factor for integral flanges**
  \[ f = \max \left[ 10, \frac{0.097779 - 0.0336333X_g + 0.0641767X_h^2}{0.0562862X_g + 0.0347076X_h^2 - 0.18699X_h^3} \right] = 1.00000 \]

- **Stress factor e**
  \[ e = \frac{F}{h_s} = 0.00303 \]
Stress factor $d$

\[ d = \frac{Ug_f h_0}{V} \]

\[ d = \frac{g_f h_0}{V} = 19208417.78589 \]

Stress factor $L$

\[ L = \frac{1}{T} + \frac{1}{\beta} \]

\[ L = 2.51655 \]

Gasket load for the operating condition

\[ H_G = W_o - H \]

\[ H_G = 3936793 \text{ N} \quad 885026.11 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange

\[ H = 0.785Q_p^2P \]

\[ H = 18374336 \text{ N} \quad 4130714.64 \text{ lbf} \]

Total hydrostatic end force

\[ H = 0.785Q_p^2P \]

\[ H = 18895341 \text{ N} \quad 4247841.29 \text{ lbf} \]

Difference

\[ H - H_G = 521005 \text{ N} \quad 117126.65 \text{ lbf} \]

Moment arm for load HD

\[ h_D = \frac{C - B - g_1}{\frac{r_0}{2}} \]

\[ h_D = 160.00 \text{ mm} = 6.299 \text{ in} \]

Moment arm for load HG

\[ h_G = \frac{C - G}{\frac{r_0}{2}} \]

\[ h_G = 199.47 \text{ mm} = 7.853 \text{ in} \]

Moment arm for load HT

\[ h_T = \frac{1}{2} \left( \frac{C - B}{r_0} + g_3 \right) \]

\[ h_T = 203.98 \text{ mm} = 8.031 \text{ in} \]

Average of the hub thicknesses

\[ \bar{AA} = \bar{AR} = 308.50 \text{ mm} = 12.146 \text{ in} \]

\[ \bar{BB} = t = 300.00 \text{ mm} = 11.811 \text{ in} \]

\[ \bar{CC} = h = 120.00 \text{ mm} = 4.724 \text{ in} \]

\[ \bar{DDG} = G_{avg} = 79.50 \text{ mm} = 3.130 \text{ in} \]

Moment of inertia $K_{AB}$

\[ K_{AB} = \left( \frac{A_p B_p^3}{12} \right) \left[ \frac{1}{3} - 0.021 \left( \frac{C}{A_p} \right) \right] \]

\[ K_{AB} = \frac{1202261513}{\text{mm}^4} = 2888445 \text{ in}^4 \]

Moment of inertia $K_{CD}$

\[ K_{CD} = \left( \frac{C_p D_p^3}{12} \right) \left[ \frac{1}{3} - 0.0105 \left( \frac{D_p}{C_p} \right) \right] \]

\[ K_{CD} = \frac{15908319}{\text{mm}^4} = 38220 \text{ in}^4 \]

Bending moment of inertia of the flange cross-section

\[ I = \frac{K_{AB} + K_{CD}}{2} \]

\[ I = \frac{1218169832}{\text{mm}^4} = 2926665 \text{ in}^4 \]

Flange moments

Nominal bolt diameter $d_B$

\[ d_B = 114.30 \text{ mm} = 4.500 \text{ in} \]

TEMA Load concentration factor $cF - MAX$

\[ cF - MAX = \frac{\pi t_0^2}{2 m G} \]

\[ cF - MAX = 1.00000 \]

Moment factor used to design split rings $c_F$

\[ c_F = 1.00 \]

Component of the flange design moment resulting from a net section bending moment and/or axial force $M_{ax} = 4M_{x}\left( \frac{0.38466I_{f} + 1}{C - 2h_0} \right) + FJ_0$

\[ M_{ax} = 0 \text{ N} \cdot \text{m} = 0 \text{ lbf} \cdot \text{in} \]

Flange design moment for the operating condition $M_o = cF \left( H_f^2 + H_f h_T + H_f h_G + M_{ax} \right) F_x$

\[ M_o = 3831437.5 \text{ N} \cdot \text{m} = 3391076.2 \text{ lbf} \cdot \text{in} \]

Flange design moment for the gasket seating condition $M_g = cF \left( C - G \right) F_x$

\[ M_g = 6846117.0 \text{ N} \cdot \text{m} = 60593235.7 \text{ lbf} \cdot \text{in} \]

Flange stresses - operating condition

Corrected inside diameter of the flange $B_1^* = B^* + g_1'$

\[ B_1^* = 1380.00 \text{ mm} = 54.331 \text{ in} \]

Flange hub stress - operating condition $S_H = \frac{f M_o}{Lg^2 P_e}$

\[ S_H = 117.26 \text{ MPa} = 17006.5 \text{ psi} \]

Flange radial stress - operating condition $S_R = \frac{(133e + 1)M_o}{P_e}$

\[ S_R = 29.14 \text{ MPa} = 4226.0 \text{ psi} \]

Flange tangential stress - operating condition $S_T = \frac{Y M_o}{P_e} - Z S_R$

\[ S_T = 91.78 \text{ MPa} = 13312.2 \text{ psi} \]

\[ S_{Ho} \leq \min[1.5S_{fo}, 2.5S_{no}]: \text{ Ok} \]

\[ S_{Ro} \leq S_{fo}: \text{ Ok} \]

\[ S_{To} \leq S_{fo}: \text{ Ok} \]

\[ (S_{Ho} + S_{Ro})/2 \leq S_{fo}: \text{ Ok} \]

\[ (S_{Ho} + S_{To})/2 \leq S_{fo}: \text{ Ok} \]
Flange stresses - seating condition

Flange hub stress - gasket seating condition
\[ S_H = \frac{fM_g}{Lg^2B1} = 209.52 \text{ MPa} \quad 30\,387.6 \text{ psi} \]

Flange radial stress - gasket seating condition
\[ S_R = \frac{(133e+1)M_g}{RF} = 52.06 \text{ MPa} \quad 7\,551.1 \text{ psi} \]

Flange tangential stress - gasket seating condition
\[ S_T = \frac{YM_g}{RF} - ZS_H = 164.00 \text{ MPa} \quad 23\,786.6 \text{ psi} \]

\[ (SHg + SRg) / 2 \leq Sfg: \text{ Ok} \]
\[ (SHg + STg) / 2 \leq Sfg: \text{ Ok} \]

Flange rigidity - operating condition

Rigidity index factor
\[ KR = 0.30 \]

Flange rigidity index
\[ J = \frac{5214VM_g}{LEyog_kKgJg} = 0.42462 \]

\[ Jo \leq 1: \text{ Ok} \]

Flange rigidity - seating condition

Flange rigidity index
\[ J = \frac{5214VM_g}{LEyog_kKgJg} = 0.65791 \]

\[ Jg \leq 1: \text{ Ok} \]

Hub thickness

Minimum hub thickness as cylindrical shell
\[ th_{min} = 63.17 \text{ mm} \quad 2.487 \text{ in} \]

\[ g0 \geq th_{min}: \text{ Ok} \]

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)
\[ = 26.08 \text{ MPa} \quad 3\,782.4 \text{ psi} \]

Hot & corroded (flange)
\[ = 21.58 \text{ MPa} \quad 3\,129.5 \text{ psi} \]

New & cold (bolts)
\[ = 22.47 \text{ MPa} \quad 3\,258.3 \text{ psi} \]

Hot & corroded (bolts)
\[ = 22.47 \text{ MPa} \quad 3\,258.3 \text{ psi} \]
External pressure

<table>
<thead>
<tr>
<th>Allowable stresses</th>
<th>Flange</th>
<th>Hub</th>
<th>Bolting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design condition</td>
<td>Sfo=207.00 MPa / 30 022.8 psi</td>
<td>Sno=207.00 MPa / 30 022.8 psi</td>
<td>Sbo=138.00 MPa / 20 015.2 psi</td>
</tr>
<tr>
<td>Seating condition</td>
<td>Sfg=207.00 MPa / 30 022.8 psi</td>
<td>Sng=207.00 MPa / 30 022.8 psi</td>
<td>Sbg=138.00 MPa / 20 015.2 psi</td>
</tr>
</tbody>
</table>

- External design pressure: $P_e = 0.10 \text{ MPa} = 14.9 \text{ psi}$
- Overpressure due to static head: $P_h = 0 \text{ MPa} = 0 \text{ psi}$
- Calculation pressure: $P = P_e + P_h = 0.10 \text{ MPa} = 14.9 \text{ psi}$
- External design temperature: $T_e = 20.00 ^\circ C = 68.00 ^\circ F$
- Modulus of Elasticity at the operating load case temperature: $E_{yo} = 210 350.00 \text{ MPa} = 30 508 688.2 \text{ psi}$
- Modulus of Elasticity at the gasket seating load case temperature: $E_{yg} = 210 350.00 \text{ MPa} = 30 508 688.2 \text{ psi}$

**Bolt loads**

- Number of bolts: $n = 28$
- Bolt type: ANSI_TEMA 4-1/2"
- Bolt spacing: $B_s = 190.74 \text{ mm} = 7.509 \text{ in}$
- Nominal bolt diameter: $a = 114.30 \text{ mm} = 4.500 \text{ in}$
- Design bolt load for the operating condition: $W_o = 0.785G^2P + 2\beta\pi Gm^3 = 165 385 \text{ N} = 37 179.93 \text{ lbf}$
- Design bolt load for the gasket seating (Bolt): $W_g = \pi b Gm^3 = 32 571 288 \text{ N} = 7 322 316.25 \text{ lbf}$
- External tensile net-section axial force: $F_A = 0 \text{ N} = 0 \text{ lbf}$
- Absolute value of the external net-section bending moment: $M_E = 0 \text{ N.m} = 0 \text{ lbf.in}$
- External tensile net-section axial force (test condition): $F_{At} = 0 \text{ N} = 0 \text{ lbf}$
- External tensile net-section bending moment (test condition): $M_{Et} = 0 \text{ N.m} = 0 \text{ lbf.in}$
- Total minimum required cross-sectional area of the bolts: $A_{min} = \max \left( \left( \frac{W_o + F_A + 4W_g}{E_{yo}} \right) , \left( \frac{W_g}{E_{yg}} \right) \right) = 236 023.8 \text{ mm}^2 = 365.838 \text{ in}^2$
- Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion: $A_b = 261 393.0 \text{ mm}^2 = 405.160 \text{ in}^2$
- Maximum bolts area for gasket crush: $A_{g,\text{max}} = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}} = 1 184 756.2 \text{ mm}^2 = 1 836.376 \text{ in}^2$
- Design bolt load for the gasket seating (Flange): $W_g = \left( \frac{A_{min} + A_b}{2} \right) S_{bg} = 34 321 763 \text{ N} = 7 715 838.57 \text{ lbf}$
Flange constants
Ratio of the flange outside diameter to the inside diameter
\[ K = \frac{A}{B'} = 1.48090 \]
Stress factor Y
\[ Y = \frac{1}{K} \left[ \frac{0.66845 + 5.71690}{K^2 - 1} \right] = 5.11631 \]
Stress factor T
\[ T = \frac{K^2 (1 + 8.55246 \log (K) - 1)}{10.4720 + 19.48 K^2 (1 - K) - 1} = 1.71899 \]
Stress factor U
\[ U = \frac{136136 (K^2 - 1) (K - 1)}{K^2 (1 + 8.55246 \log (K) - 1)} = 5.62231 \]
Stress factor Z
\[ Z = \frac{K^2 + 1}{(K^2 - 1)} = 2.67364 \]
Hub length parameter
\[ h_{h} = \sqrt{B_{h}g_{h}} = 282.04 \text{ mm} \]
Hub thickness ratio
\[ X_{g} = \frac{g_{h}}{g_{h}'} = 1.56452 \]
Hub length ratio
\[ X_{h} = \frac{h_{h}}{h_{h}'} = 0.42547 \]
Flange stress factor for integral type flanges
\[ F = \left( 0.0979697 - 0.297012 X_{g} + 9.5257 (10^{-3}) X_{g} \right) + \left( 0.012386 (X_{g})^2 + 0.0358858 (X_{h})^2 - 0.194422 (X_{g}) (X_{h}) - 0.0181259 (X_{g})^3 + 0.0129360 (X_{h})^3 - 0.0377693 (X_{g}) (X_{b}) + 0.0427371 (X_{g})^2 (X_{b}) - 0.502444 + 0.116791 X_{g} - 0.187071 Y_{h} - 0.104440 \right + \left( 2.49189 Y_{h} \right) \]
Ham stress correction factor for integral flanges
\[ f = \max \left( 0, \frac{0.0927779 - 0.0336633 X_{g} + 0.964176 Y_{g} + 0.056286 Y_{h} + 0.034707 Y_{h}^2 + 0.418699 Y_{h} + 0.0156692 (10^{-3}) X_{g} + 16.2904 Y_{g} + 3.49329 Y_{h}^2 + 13.9052 Y_{h}}{1 - 0.59603 (10^{-3}) X_{g} + 16.2904 Y_{g} + 3.49329 Y_{h}^2 + 13.9052 Y_{h}} \right) = 1.00000 \]
Stress factor e
\[ e = \frac{1}{f} = 0.00303 \]
Stress factor d
\[ d = \frac{U g_{h} h_{h}}{Y} = 19208417.78589 \]
Stress factor L
\[ L = \frac{28 + 1.1 D}{D} = 2.51655 \]
Gasket load for the operating condition
\[ H_{G} = W_{o} = 28516 \text{ N} = 6410.69 \text{ lbf} \]
Total hydrostatic end force on the area inside of the flange
\[ H_{G} = 0.7852 F = 133095 \text{ N} = 29290.84 \text{ lbf} \]
Total hydrostatic end force
\[ H = 0.7852 F = 136868 \text{ N} = 30769.24 \text{ lbf} \]
Difference
\[ C = H - H_{G} = 3774 \text{ N} = 848.41 \text{ lbf} \]
Moment arm for load HD
\[ hD = \frac{3}{2} C - h_{G} = 160.00 \text{ mm} = 6.299 \text{ in} \]
Moment arm for load HG
\[ h_{G} = \frac{C - G}{2} = 199.47 \text{ mm} = 7.853 \text{ in} \]
Moment arm for load HT
\[ h_{T} = \frac{1}{4} \left( \frac{C - B}{2} + h_{G} \right) = 203.98 \text{ mm} = 8.031 \text{ in} \]
Average of the hub thicknesses
\[ \text{AA} = \text{AR} = 308.50 \text{ mm} = 12.146 \text{ in} \]
\[ \text{BB} = t = 300.00 \text{ mm} = 11.811 \text{ in} \]
\[ \text{CC} = h = 120.00 \text{ mm} = 4.724 \text{ in} \]
\[ \text{DDG} = G_{avg} = 79.50 \text{ mm} = 3.130 \text{ in} \]
Moment of inertia KAB
\[ K_{AB} = (A_{B} P_{B}) \left[ \frac{1}{3} - 0.021 \left( \frac{B_{g}}{A_{A}} \right) \right] \left( 1 - \frac{1}{12} \left( \frac{B_{g}}{A_{A}} \right)^4 \right) \]
\[ = 1202261513 \text{ mm}^4 \]
Moment of inertia KCD
\[ K_{CD} = (C_{D} D_{g}) \left[ \frac{1}{3} - 0.101 \left( \frac{D_{g}}{C_{C}} \right) \right] \left( 1 - \frac{1}{192} \left( \frac{D_{g}}{C_{C}} \right)^4 \right) \]
\[ = 15908319 \text{ mm}^4 \]
<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Conversion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bending moment of inertia of the flange cross-section</td>
<td>$I = 964,097,900 \text{ mm}^4$</td>
<td>$2,316,254 \text{ in}^4$</td>
</tr>
<tr>
<td>Cross-section polar moment of inertia</td>
<td>$I_p = 1,218,169,832 \text{ mm}^4$</td>
<td>$2,926,665 \text{ in}^4$</td>
</tr>
</tbody>
</table>

**Flange moments**

- Nominal bolt diameter: $d_B = 114.30 \text{ mm}$ (4.500 in)
- Moment factor used to design split rings: $c_F = 1.00$

**Flange design moment**

- Resulting from a net section bending moment and/or axial force: $M_{so} = 0 \text{ N} \cdot \text{m}$ (0 lbf·in)
- Operating condition: $M_g = 5,236.0 \text{ N} \cdot \text{m}$ (46,342.7 lbf·in)
- Gasket seating condition: $M_g = 6,846,117.0 \text{ N} \cdot \text{m}$ (60,593,235.7 lbf·in)

**Flange stresses - operating condition**

- Corrected inside diameter of the flange: $B_1' = B' + g_1' = 1,380.00 \text{ mm}$ (54.331 in)
- Hub stress: $S_H = 0.16 \text{ MPa}$ (23.2 psi)
- Radial stress: $S_R = 0.04 \text{ MPa}$ (5.8 psi)
- Tangential stress: $S_T = 0.13 \text{ MPa}$ (18.2 psi)

**Flange stresses - seating condition**

- Hub stress: $S_H = 209.52 \text{ MPa}$ (30,387.6 psi)
- Radial stress: $S_R = 52.06 \text{ MPa}$ (7,551.1 psi)
- Tangential stress: $S_T = 164.00 \text{ MPa}$ (23,786.6 psi)

**Flange rigidity - operating condition**

- Rigidity index factor: $K_R = 0.30$
- Rigidity index: $J = 0.00050$

**Flange rigidity - seating condition**

- Rigidity index: $J = 0.65791$

---

Customer: Riccardo Petrelli
Drawing: U_150
Revision: 1028-1729603698

**Contact Information**
Walter Tosto S.p.A.
via Erasmo Piaggio
Chieti
Telephone, Fax
Website, Email address
Date ________ Calc. ________ Contr.________ Appr.________
## Minimum Design Metal Temperature (MDMT)

### Internal pressure

**Welding neck flange - Shell Flange (Bolting)**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16 (Code Case 2655 - using Division 1 stress tables)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>114.30 mm / 4.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-29.00 °C / -20.20 °F</td>
</tr>
</tbody>
</table>

**Welding neck flange - Shell Flange**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-387 22 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>65.00 mm / 2.559 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

### External pressure

**Welding neck flange - Shell Flange (Bolting)**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16 (Code Case 2655 - using Division 1 stress tables)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>114.30 mm / 4.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-29.00 °C / -20.20 °F</td>
</tr>
</tbody>
</table>

**Welding neck flange - Shell Flange**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-387 22 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>65.00 mm / 2.559 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)
**Cylindrical shell - Main shell**

*According to: Asme VIII Div. 2 Ed. 2015, 4.3.3 - Metric Units*

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calculation temperature</td>
<td>$T$ = 420.00 °C</td>
<td>788.00 °F</td>
</tr>
<tr>
<td>Material:</td>
<td>SA-387 22 2 - Plate</td>
<td></td>
</tr>
<tr>
<td>Allowable stress at room temperature</td>
<td>$S_T$ = 207.00 MPa</td>
<td>30 022.8 psi</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>$E$ = 1.00</td>
<td></td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>$c$ = 3.00 mm</td>
<td>0.118 in</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>$c_e$ = 0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>$c'$ = 0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>$D$ = 1 275.00 mm</td>
<td>50.197 in</td>
</tr>
<tr>
<td>Length</td>
<td>$L$ = 3 629.00 mm</td>
<td>142.874 in</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>$t$ = 65.00 mm</td>
<td>2.559 in</td>
</tr>
<tr>
<td>Ligament Efficiency</td>
<td>$E$ = 1.00</td>
<td></td>
</tr>
<tr>
<td>Reference figure</td>
<td>= None</td>
<td></td>
</tr>
<tr>
<td>Diameter of tube holes</td>
<td>$d$ = 0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Internal pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Allowable stress</td>
<td>$S$ = 158.60 MPa</td>
<td>23 003.0 psi</td>
</tr>
<tr>
<td>Internal pressure</td>
<td>$P_i$ = 14.22 MPa</td>
<td>2 062.4 psi</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>$P_h$ = 0 MPa</td>
<td></td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>$P = P_i + P_h$ = 14.22 MPa</td>
<td>2 062.4 psi</td>
</tr>
<tr>
<td>Required thickness</td>
<td>$t_r = \frac{D + 2(c + c_e)}{2(\phi L) - 1} + c + c_e + c' = 63.08$ mm</td>
<td>2.483 in</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Condition</th>
<th>Pressure</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>20.10 MPa</td>
<td>2 915.0 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>14.65 MPa</td>
<td>2 125.4 psi</td>
</tr>
</tbody>
</table>

**External pressure**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>External pressure</td>
<td>$P_e$ = 0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>External static head</td>
<td>$P_h$ = 0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>$P = P_e + P_h$ = 0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>External design temperature</td>
<td>$T_e$ = 20.00 °C</td>
<td>68.00 °F</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>$D_o$ = 1 405.00 mm</td>
<td>55.315 in</td>
</tr>
<tr>
<td>Max unsupported length</td>
<td>$L$ = 3 808.40 mm</td>
<td>149.937 in</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>$E_y$ = 210 350.00 MPa</td>
<td>30 508 688.2 psi</td>
</tr>
<tr>
<td>Shell parameter</td>
<td>$M_s = \frac{L}{\sqrt{R_c(t - c - c')}} = 18.24835$</td>
<td></td>
</tr>
<tr>
<td>Elatic buckling stress</td>
<td>$F_{he} = 16 \cdot C_{11} \cdot E_y \cdot (t - c - c_e - c'') / D_o = 770.23$ MPa</td>
<td>111 712.7 psi</td>
</tr>
<tr>
<td>Buckling stress</td>
<td>Fic = Sy = 310.00 MPa</td>
<td>44 961.7 psi</td>
</tr>
<tr>
<td>Yield strength at design temperature</td>
<td>Sy = 310.00 MPa</td>
<td>44 961.7 psi</td>
</tr>
<tr>
<td>Design factor</td>
<td>FS = 1.66700</td>
<td></td>
</tr>
<tr>
<td>Hoop compressive membrane stress</td>
<td>Fha = Fic / FS = 185.96 MPa</td>
<td>26 971.6 psi</td>
</tr>
<tr>
<td>Allowable external pressure in the absence of other loads</td>
<td>$P_a = 2F_{ha} \left(\frac{t - c - c_e - c'}{D_o}\right) = 16.41$ MPa</td>
<td>2 380.4 psi</td>
</tr>
<tr>
<td>Minimum required thickness</td>
<td>$tr = 9.26$ mm</td>
<td>0.365 in</td>
</tr>
</tbody>
</table>
# Maximum allowable external pressures

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>17.21 MPa</td>
<td>2495.6 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>16.41 MPa</td>
<td>2380.4 psi</td>
</tr>
</tbody>
</table>
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure

**Cylindrical shell - Main shell**

- **Material**: = SA-387 22 2
- **Curve of fig. 3.7 / 3.8**: = A
- **Governing Thickness**: \( tg = 65.00 \text{ mm} \) \( 2.559 \text{ in} \)
- **PostWeld Heat Treatment**: = No
- **Minimum Design Metal Temperature (MDMT)**: = 
- **Impact tests required by Code**: = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

#### External pressure

**Cylindrical shell - Main shell**

- **Material**: = SA-387 22 2
- **Curve of fig. 3.7 / 3.8**: = A
- **Governing Thickness**: \( tg = 65.00 \text{ mm} \) \( 2.559 \text{ in} \)
- **PostWeld Heat Treatment**: = No
- **Minimum Design Metal Temperature (MDMT)**: = 
- **Impact tests required by Code**: = Yes

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

### Validation warnings:

- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]
**Reinforcement of opening - Out Shell Side**

According to: *Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units*

**Calculation temperature**

\[ T = 420.00 \, ^\circ C = 788.00 \, ^\circ F \]

**Nozzle material**

SA-182 F22 3 - Forgings

**Shell material**

SA-387 22 2 - Plate

**Pad material**

Allowable stress from Annex 3.A for the vessel at the design temperature

\[ S = 158.60 \, MPa = 23003.0 \, psi \]

Shell allowable stress at room temperature

\[ S_0 = 207.00 \, MPa = 30022.8 \, psi \]

Allowable stress from Annex 3.A for the nozzle at the design temperature

\[ S_n = 158.60 \, MPa = 23003.0 \, psi \]

Nozzle allowable stress at room temperature

\[ S_{n0} = 207.00 \, MPa = 30022.8 \, psi \]

**Shell thickness**

\[ t = 65.00 \, mm = 2.559 \, in \]

Nozzle thickness

\[ t_n = 100.00 \, mm = 3.937 \, in \]

Nominal wall thickness of the nozzle thinner portion

\[ t_{n2} = 69.90 \, mm = 2.752 \, in \]

Tapering angle

\[ \alpha = 45.00 \, ^\circ \]

Nozzle inside diameter

\[ d = 457.20 \, mm = 18.000 \, in \]

Nozzle outside diameter

\[ D_o = 597.00 \, mm = 23.504 \, in \]

Joint efficiency

\[ E = 1.00000 \]

Nozzle internal corrosion allowance

\[ c_{ni} = 3.00 \, mm = 0.118 \, in \]

Nozzle external corrosion allowance

\[ c_{ne} = 0 \, mm = 0 \, in \]

Nozzle total corrosion allowance

\[ c_n = 3.00 \, mm = 0.118 \, in \]

Nozzle undertolerance

\[ c_n' = 0 \, mm = 0 \, in \]

Nozzle position

Radial

Nozzle connection

Integrally reinforced

Weld joint type

7 - Full penetration welds

Offset from shell border

\[ 3150.00 \, mm = 124.016 \, in \]

Angular offset

\[ \beta = 180.00 \, ^\circ \]

Width of the reinforcing pad

\[ W = 0 \, mm = 0 \, in \]

Thickness of the reinforcing pad

\[ t_e = 0 \, mm = 0 \, in \]

Nozzle inside radius

\[ R_{n} = d/2 + c_{ni} + c_{n}' = 231.60 \, mm = 9.118 \, in \]

Shell inside diameter

\[ D_i = 1275.00 \, mm = 50.197 \, in \]

Effective radius of the shell

\[ R_{ef} = 0.5 \cdot D_i + \delta = 640.50 \, mm = 25.217 \, in \]

Nozzle projection from the outside of the vessel wall

\[ L_{pr1} = 881.10 \, mm = 34.689 \, in \]

Nozzle projection from the inside of the vessel wall

\[ L_{pr2} = 0 \, mm = 0 \, in \]

Length of variable thickness from the outside of the vessel wall

\[ L_{pr3} = 150.00 \, mm = 5.906 \, in \]

Nozzle projection from the outside of the vessel wall to \( t_{n2} \)

\[ L_{pr4} = 88.10 \, mm = 3.469 \, in \]

Weld leg length of the outside nozzle fillet weld

\[ L_{41} = 15.00 \, mm = 0.591 \, in \]

Weld leg length of the pad to vessel fillet weld

\[ L_{42} = 0 \, mm = 0 \, in \]

Weld leg length of the inside nozzle fillet weld

\[ L_{43} = 0 \, mm = 0 \, in \]

Corner radius

\[ r = 10.00 \, mm = 0.394 \, in \]

Effective length along the vessel wall (limited by shell offset)

\[ L_{R} = \min \left[ \sqrt{R_{ef} \cdot (t - c - \delta)} \cdot 2R_{o} \right] = 1.60 \, mm = 0.063 \, in \]

\[ L_{R} = \min \left[ 15(t-c-\delta) \cdot t_{r} \right] + \sqrt{R_{o} \left( t_{r} - c_{o} - c_{o}' \right)} \]

\[ L_{BD} = 8(t - c - \delta + t_{r}) \]

\[ L_{BD} = 881.10 \, mm = 34.689 \, in \]

Effective length along the nozzle wall outside the vessel

\[ L_{R} = \min \left[ L_{BD} \cdot L_{R} \right] + t - c - \delta = 211.88 \, mm = 8.342 \, in \]

Effective thickness

\[ t_{ef} = t - c \quad \text{and} \quad \frac{A_{S} f_{p}}{L_{S}} \]

\[ L_{11} = 0 \, mm = 0 \, in \]
Effective length along the nozzle wall inside the vessel
\[ \lambda = \min \left\{ \frac{2R_{n} + t_{n} - c_{0} - c_{0}'}{\sqrt{(D_{t} + t_{eff})t_{eff}^{3}}}, \frac{120}{1055} \right\} = 1.94137 \]

Area contributed by the vessel wall
\[ A_{1} = (t_{n} - c_{0} - c_{0}')L_{21} \]
\[ A_{1} = 99.2 \text{ mm}^{2} \text{ 0.154 in}^{2} \]

Nozzle outs. vessel wall area
\[ A_{2} = (t_{n} - c_{0} - c_{0}')L_{22} \]
\[ A_{2} = 20,552.7 \text{ mm}^{2} \text{ 31.857 in}^{2} \]

Nozzle material factor
\[ f_{m} = \frac{S_{n}}{S_{m}} = 1.00000 \]

Pad material factor
\[ f_{tp} = \frac{S_{n}}{S_{tp}} = 0 \]

Nozzle ins. vessel wall area
\[ A_{3} = (t_{n} - 2c_{0} - c_{0}')L_{23} \]
\[ A_{3} = 0 \text{ mm}^{2} \text{ 0 in}^{2} \]

Area contributed by the outside nozzle fillet weld
\[ A_{41} = 0.5L_{21} \]
\[ A_{41} = 112.5 \text{ mm}^{2} \text{ 0.174 in}^{2} \]

Area contributed by the pad to vessel fillet weld
\[ A_{42} = 0.5L_{22} \]
\[ A_{42} = 0 \text{ mm}^{2} \text{ 0 in}^{2} \]

Area contributed by the inside nozzle fillet weld
\[ A_{43} = 0.5L_{23} \]
\[ A_{43} = 0 \text{ mm}^{2} \text{ 0 in}^{2} \]
\[ A_{5b} = 0 \text{ mm}^{2} \text{ 0 in}^{2} \]

Area contributed by the reinforcing pad
\[ A_{5} = \min \left\{ A_{4}, A_{g} \right\} \]
\[ A_{5} = 20,764.4 \text{ mm}^{2} \text{ 32.185 in}^{2} \]

Total area
\[ A_{T} = A_{1} + f_{m}(A_{2} + A_{3}) + A_{41} + A_{42} + A_{43} + f_{m}A_{5} \]
\[ A_{T} = 20,764.4 \text{ mm}^{2} \text{ 32.185 in}^{2} \]

Radius of the nozzle opening
\[ R_{nc} = R_{n} \]
\[ R_{nc} = 231.60 \text{ mm} \text{ 9.118 in} \]

Nozzle radius for force calculation
\[ R_{n} = \frac{L_{n} - 2c_{0} - c_{0}'}{\ln \frac{R_{nc}}{R_{eff}}} \]
\[ R_{n} = 277.28 \text{ mm} \text{ 10.916 in} \]

Shell radius for force calculation
\[ R_{ss} = \frac{L_{n} - 2c_{0} - c_{0}'}{\ln \frac{R_{nc}}{R_{eff}}} \]
\[ R_{ss} = 671.02 \text{ mm} \text{ 26.418 in} \]

Force from internal pressure in the nozzle
\[ f_{P} = PR_{ss}(L_{21} + t_{n} - c_{0} - c_{0}') \]
\[ f_{P} = 835,415 \text{ N} \text{ 187,808.68 lbf} \]

Force from internal pressure in the shell
\[ f_{y} = PR_{ss}(L_{22} + t_{n} - c_{0} - c_{0}') \]
\[ f_{y} = 940,812 \text{ N} \text{ 211,502.91 lbf} \]

Discontinuity force from internal pressure
\[ \sigma_{an} = \frac{f_{y}}{A_{T}} \]
\[ \sigma_{an} = 191.97 \text{ MPa} \text{ 27,842.4 psi} \]

Average primary membrane stress
\[ \sigma_{w} = \frac{f_{y}}{A_{T}} \]
\[ \sigma_{w} = 153.90 \text{ MPa} \text{ 22,321.1 psi} \]

General primary membrane stress
\[ S_{allow} = 1.5\sigma_{w} \]
\[ S_{allow} = 237.90 \text{ MPa} \text{ 34,504.5 psi} \]

Allowable stress
\[ P_{L} = \max \left\{ (2\sigma_{w} - \sigma_{d}), \sigma_{d} \right\} \]
\[ r \geq \min \left\{ 0.25t, 3 \text{ mm} \right\}: \text{ Ok} \]
\[ PL \leq S_{allow}: \text{ Ok} \]

Maximum local primary membrane stress
\[ P_{L_{max}} = \max \left\{ 0.3P_{T}, 230.04 \text{ MPa} \right\} \]
\[ PL \leq S_{allow}: \text{ Ok} \]

Area resisting pressure
\[ P_{max} = \min \left\{ P_{max1}, P_{max2} \right\} \]
\[ P_{max} = 14.65 \text{ MPa} \text{ 2,125.4 psi} \]

Nozzle maximum allowable pressure (bottom)
\[ P_{max} = \min \left\{ P_{max1}, P_{max2} \right\} \]
\[ P_{max} = 14.65 \text{ MPa} \text{ 2,125.4 psi} \]
Strength of nozzle attachment welds

- Throat dimension of the nozzle to shell weld
  \[ tc = \frac{L_{41}}{\sqrt{2}} = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

- Discontinuity force factor
  \[ k_y = \frac{R_{max} + t_n - c_{i} + c_{d}}{R_{max}} = 1.41883 \]

- Nozzle to shell weld length
  \[ L_n = \frac{R}{2} (R_n + t_n - c_{i} - c_{d}) = 516.16 \text{ mm} \quad 20.321 \text{ in} \]

- Throat dimension of the outside nozzle fillet weld
  \[ L_{41T} = 0.0707L_{41} = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

- Throat dimension for the pad to vessel fillet weld
  \[ L_{42T} = 0.0707L_{42} = 0 \text{ mm} \quad 0 \text{ in} \]

- Throat dimension for inside nozzle fillet weld
  \[ L_{43T} = 0.0707L_{43} = 0 \text{ mm} \quad 0 \text{ in} \]

- Welds force
  \[ f_{welds} = \min \left[ f_{Y}k_{y} \cdot 1.5S_{n}(A_{2} + A_{3}), \frac{\pi}{4}PR_{4}k_{y}^{2} \right] = 1,205,907 \text{ N} \quad 271,098.57 \text{ lbf} \]

- Nozzle to shell groove weld depth
  \[ t_{w1} = twall = 62.00 \text{ mm} \quad 2.441 \text{ in} \]

- Average effective shear stress
  \[ \tau = \frac{f_{welds}}{tc(0.49L_{41T} + 0.05_{w1} + 0.49L_{43T})} = 55.10 \text{ MPa} \quad 7,992.3 \text{ psi} \]

  \[ tc \geq \min[0.7tn, 6mm (0.25in)]: \text{ Ok} \]

  \[ \tau \leq S: \text{ Ok} \]

External Pressure

- External design temperature
  \[ T_{e} = 20.00 \text{ °C} \quad 68.00 \text{ °F} \]

- Allowable stress from Annex 3.A for the vessel at the design temperature
  \[ S = 207.00 \text{ MPa} \quad 30,022.8 \text{ psi} \]

- Nozzle inside radius
  \[ R_{n} = \frac{d}{2} + c_{ni} + c_{n}' = 231.60 \text{ mm} \quad 9.118 \text{ in} \]

- Shell inside diameter
  \[ D_{i} = 1,275.00 \text{ mm} \quad 50.197 \text{ in} \]

- Nozzle projection from the outside of the vessel wall
  \[ L_{pr1} = 881.10 \text{ mm} \quad 34.689 \text{ in} \]

- Nozzle projection from the inside of the vessel wall
  \[ L_{pr2} = 0 \text{ mm} \quad 0 \text{ in} \]

- Length of variable thickness from the outside of the vessel wall
  \[ L_{pr3} = 150.00 \text{ mm} \quad 5.906 \text{ in} \]

- Nozzle projection from the outside of the vessel wall to tn2
  \[ L_{pr4} = 88.10 \text{ mm} \quad 3.469 \text{ in} \]

- Weld leg length of the outside nozzle fillet weld
  \[ L_{41} = 15.00 \text{ mm} \quad 0.591 \text{ in} \]

- Weld leg length of the pad to vessel fillet weld
  \[ L_{42} = 0 \text{ mm} \quad 0 \text{ in} \]

- Weld leg length of the inside nozzle fillet weld
  \[ L_{43} = 0 \text{ mm} \quad 0 \text{ in} \]

- Effective length along the vessel wall (limited by shell offset)
  \[ L_{R} = \min \left[ \sqrt{R_{eff} \cdot (t - c - c_{d})}, 2R_{e} \right] = 1.60 \text{ mm} \quad 0.063 \text{ in} \]

- Effective length along the nozzle wall outside the vessel
  \[ L_{HI} = \min \left[ 15(t - c - c_{d}), t_{s} \right] + \sqrt{R_{n}(t_{n} - c_{i} - c_{d})} \]
  \[ L_{HI} = 149.88 \text{ mm} \quad 5.901 \text{ in} \]

- Effective length along the nozzle wall outside the vessel
  \[ L_{HI} = \min \left[ L_{HI}, L_{HD}, L_{H3} \right] + t - c - c' \]
  \[ L_{eff} = t - c - c' + \left( \frac{L_{HI}}{L_{R}} \right) \]
  \[ L_{eff} = 211.88 \text{ mm} \quad 8.342 \text{ in} \]

- Effective thickness
  \[ t_{eff} = t - c - c' + \left( \frac{L_{HI}}{L_{R}} \right) \]
  \[ t_{eff} = 62.00 \text{ mm} \quad 2.441 \text{ in} \]
  \[ L_{11} = 0 \text{ mm} \quad 0 \text{ in} \]
  \[ L_{12} = 0 \text{ mm} \quad 0 \text{ in} \]
  \[ L_{13} = 0 \text{ mm} \quad 0 \text{ in} \]

- Effective length along the nozzle wall inside the vessel
  \[ L_{I} = \min \left[ \frac{(2R_{e} + t_{e} - c_{i} - c_{d})}{\sqrt{(D_{i} + t_{eff})f_{eff}^{2}] / \left( \frac{2}{3} \right)}}, 120 \right] \]
  \[ L_{I} = 99.2 \text{ mm} \quad 0.154 \text{ in} \]

- Area contributed by the vessel wall
  \[ A_{1} = ((t - c - c')L_{I}) \cdot \max \left( \frac{2}{3}, 10 \right) \]
  \[ A_{1} = 99.2 \text{ mm}^2 \quad 0.154 \text{ in}^2 \]

- Nozzle outs. vessel wall area
  \[ A_{2} = (t_{n} - c_{n} - c_{n}')L_{n} \]
  \[ A_{2} = 20,552.7 \text{ mm}^2 \quad 31,857 \text{ in}^2 \]

- Nozzle material factor
  \[ f_{m} = \frac{S}{S_{n}} = 1.00000 \]

- Pad material factor
  \[ f_{p} = \frac{S_{p}}{S} = 0 \]

- Nozzle ins. vessel wall area
  \[ A_{3} = (t_{n} - 2c_{n} - c_{n}')L_{n} \]
  \[ A_{3} = 0 \text{ mm}^2 \quad 0 \text{ in}^2 \]

- Area contributed by the outside nozzle fillet weld
  \[ A_{41} = 0.5L_{41}^{2} \]
  \[ A_{41} = 112.5 \text{ mm}^2 \quad 0.174 \text{ in}^2 \]
Area contributed by the pad to vessel fillet weld

\[ A_{42} = 0.5L_{42}^2 \]

\[ = 0 \text{ mm}^2, 0 \text{ in}^2 \]

Area contributed by the inside nozzle fillet weld

\[ A_{43} = 0.5L_{43}^2 \]

\[ = 0 \text{ mm}^2, 0 \text{ in}^2 \]

\[ A_{5} = Wt_5 \]

\[ = 0 \text{ mm}^2, 0 \text{ in}^2 \]

Area contributed by the reinforcing pad

\[ A_{5b} = \min[A_{5a}, A_{5b}] \]

\[ = 0 \text{ mm}^2, 0 \text{ in}^2 \]

Total area

\[ A_T = A_1 + f_m(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_m A_5 = 20,764.4 \text{ mm}^2, 32,185 \text{ in}^2 \]

Radius of the nozzle opening

\[ R_{nc} = R_n = 231.60 \text{ mm}, 9.118 \text{ in} \]

Nozzle radius for force calculation

\[ R_{sa} = \frac{L_{sa} - t_n - c_n - c_s}{t_{eff}} \]

\[ = 277.28 \text{ mm}, 10.916 \text{ in} \]

Shell radius for force calculation

\[ R_{as} = \frac{L_{as}}{t_{eff}} \]

\[ = 671.02 \text{ mm}, 26.418 \text{ in} \]

Force from internal pressure in the nozzle

\[ f_M = PR_{sa}L_{si} = 6,051 \text{ N}, 1,360.39 \text{ lbf} \]

Force from internal pressure in the shell

\[ f_S = PR_{sa}(L_{si} - t_n - c_n - c_s) = 6,815 \text{ N}, 1,532.02 \text{ lbf} \]

Discontinuity force from internal pressure

\[ f_Y = PR_{sa}R_{ns} = 16,007 \text{ N}, 3,598.54 \text{ lbf} \]

Average primary membrane stress

\[ \sigma_{av} = \frac{f_M + f_S + f_Y}{A_T} \]

\[ = 1.39 \text{ MPa}, 201.7 \text{ psi} \]

General primary membrane stress

\[ \sigma_{ave} = \frac{PR_{sa}}{t_{eff}} \]

\[ = 1.11 \text{ MPa}, 161.7 \text{ psi} \]

Allowable stress

\[ S_{allow} = f_{ha \_shell} = 185.96 \text{ MPa}, 26,971.6 \text{ psi} \]

Maximum local primary membrane stress

\[ P_L = \max\left\{ 2\sigma_{av} - \sigma_{ave}, \sigma_{ave} \right\} \]

\[ = 1.67 \text{ MPa}, 241.7 \text{ psi} \]

Nozzle to shell groove weld depth

\[ t_{m} = L_{41} / v_2 \]

\[ = 10.61 \text{ mm}, 0.418 \text{ in} \]

Discontinuity force factor

\[ k_Y = R_{ns} + t_n - c_n - c_s \]

\[ = 1.41883 \]

Nozzle to shell weld length

\[ L_c = \frac{v_2}{2}(R_{ns} - t_n - c_n - c_s) \]

\[ = 516.16 \text{ mm}, 20.321 \text{ in} \]

Throat dimension of the outside nozzle fillet weld

\[ L_{41} = 0.0707L_{41} = 10.61 \text{ mm}, 0.418 \text{ in} \]

Throat dimension for the pad to vessel fillet weld

\[ L_{432} = 0.0707L_{432} = 0 \text{ mm}, 0 \text{ in} \]

Throat dimension for inside nozzle fillet weld

\[ L_{433} = 0.0707L_{433} = 0 \text{ mm}, 0 \text{ in} \]

Welds force

\[ f_{welds} = \min\left\{ f_Mk_Y, 1.5S_{allow}(A_2 + A_3), \frac{PR_{sa}k_Y^2}{4} \right\} \]

\[ = 8,735 \text{ N}, 1,963.70 \text{ lbf} \]

Nozzle to shell groove weld depth

\[ tw_1 = \text{twall} \]

\[ = 62.00 \text{ mm}, 2.441 \text{ in} \]

Average effective shear stress

\[ \tau = \frac{f_{welds}}{L_c(0.45L_{41} + 0.5t_{w1} + 0.45L_{43})} \]

\[ = 0.40 \text{ MPa}, 57.9 \text{ psi} \]

\[ tc \geq \min[0.7tn, 6mm (0.25\text{in})]: \text{Ok} \]

\[ \tau \leq S: \text{Ok} \]
**Reinforcement of opening - In Shell Side**

According to: *Asme VIII Div. 2 Ed. 2015, 4.5.5 - Metric Units*

**Calculation temperature**

\[ T = 420.00 \ ^\circ \text{C} = 788.00 \ ^\circ \text{F} \]

**Nozzle material**

SA-182 F22 3 - Forgings

**Shell material**

SA-387 22 2 - Plate

**Pad material**

-

Allowable stress from Annex 3.A for the vessel at the design temperature

\[ S = 158.60 \ \text{MPa} = 23\,003.0 \ \text{psi} \]

Shell allowable stress at room temperature

\[ S_0 = 207.00 \ \text{MPa} = 30\,022.8 \ \text{psi} \]

Allowable stress from Annex 3.A for the nozzle at the design temperature

\[ S_{n} = 158.60 \ \text{MPa} = 23\,003.0 \ \text{psi} \]

Nozzle allowable stress at room temperature

\[ S_{n0} = 207.00 \ \text{MPa} = 30\,022.8 \ \text{psi} \]

Shell thickness

\[ t = 65.00 \ \text{mm} = 2.559 \ \text{in} \]

Nozzle thickness

\[ t_{n} = 100.00 \ \text{mm} = 3.937 \ \text{in} \]

Nominal wall thickness of the nozzle thinner portion

\[ t_{n2} = 69.90 \ \text{mm} = 2.752 \ \text{in} \]

Tapering angle

\[ \alpha = 45.00 \ ^\circ \]

Nozzle inside diameter

\[ d = 457.20 \ \text{mm} = 18.000 \ \text{in} \]

Nozzle outside diameter

\[ O_{d} = 597.00 \ \text{mm} = 23.504 \ \text{in} \]

Joint efficiency

\[ E = 1.00000 \]

Nozzle internal corrosion allowance

\[ c_{ni} = 3.00 \ \text{mm} = 0.118 \ \text{in} \]

Nozzle external corrosion allowance

\[ c_{ne} = 0 \ \text{mm} = 0 \ \text{in} \]

Nozzle total corrosion allowance

\[ c_{n} = 3.00 \ \text{mm} = 0.118 \ \text{in} \]

Nozzle undertolerance

\[ c_{n}' = 0 \ \text{mm} = 0 \ \text{in} \]

Nozzle position

\[ = \text{Radial} \]

Nozzle connection

\[ = \text{Integrially reinforced} \]

Weld joint type

\[ = 7 - \text{Full penetration welds} \]

Offset from shell border

\[ = 420.00 \ \text{mm} = 16.535 \ \text{in} \]

Angular offset

\[ = 0 \ ^\circ \]

Width of the reinforcing pad

\[ W = 0 \ \text{mm} = 0 \ \text{in} \]

Thickness of the reinforcing pad

\[ t_{e} = 0 \ \text{mm} = 0 \ \text{in} \]

Nozzle inside radius

\[ R_{n} = d/2 + c_{ni} + c_{n}' = 231.60 \ \text{mm} = 9.118 \ \text{in} \]

Shell inside diameter

\[ D_{i} = 1275.00 \ \text{mm} = 50.197 \ \text{in} \]

Effective radius of the shell

\[ R_{eff} = 0.5 \cdot D_{i} + \epsilon = 640.50 \ \text{mm} = 25.217 \ \text{in} \]

Nozzle projection from the outside of the vessel wall

\[ L_{pr1} = 881.10 \ \text{mm} = 34.689 \ \text{in} \]

Nozzle projection from the inside of the vessel wall

\[ L_{pr2} = 0 \ \text{mm} = 0 \ \text{in} \]

Length of variable thickness from the outside of the vessel wall

\[ L_{pr3} = 150.00 \ \text{mm} = 5.906 \ \text{in} \]

Nozzle projection from the outside of the vessel wall to \( t_{n2} \)

\[ L_{pr4} = 88.10 \ \text{mm} = 3.469 \ \text{in} \]

Weld leg length of the outside nozzle fillet weld

\[ L_{41} = 15.00 \ \text{mm} = 0.591 \ \text{in} \]

Weld leg length of the pad to vessel fillet weld

\[ L_{42} = 0 \ \text{mm} = 0 \ \text{in} \]

Weld leg length of the inside nozzle fillet weld

\[ L_{43} = 0 \ \text{mm} = 0 \ \text{in} \]

Corner radius

\[ r = 10.00 \ \text{mm} = 0.394 \ \text{in} \]

Effective length along the vessel wall (limited by shell offset)

\[ L_{R} = \min \left[ \sqrt{R_{eff} \cdot (t - c - \epsilon)} \cdot 2R_{0} \right] = 91.40 \ \text{mm} = 3.598 \ \text{in} \]

Effective length along the vessel wall

\[ L_{R} = \min \left[ 15(t - c - \epsilon), \ t_{e} \right] + \sqrt{R_{0}(t_{n} - c_{ni} - c_{n}')} = 149.88 \ \text{mm} = 5.901 \ \text{in} \]

Effective length along the nozzle wall outside the vessel

\[ L_{R} = \min \left[ L_{R} \cdot L_{R3}, L_{R3} + t - c - \epsilon \right] = 211.88 \ \text{mm} = 8.342 \ \text{in} \]

Effective thickness

\[ t_{eff} = t - c - \epsilon + \frac{A_{f} \cdot f_{L}}{L_{R}} = 62.00 \ \text{mm} = 2.441 \ \text{in} \]
Effective length along the nozzle wall inside the vessel
\[ \lambda = \min \left( \frac{2R_n + t_o - c_o - c_i}{\sqrt{(D_1 + t_{eff})t_{eff}}}, 1.2 \right) \]
\[ = 1.94137 \]

Area contributed by the vessel wall
\[ A_1 = (t_o - c_o - c_i)L_o \cdot \max \left( \frac{3}{5}, 10 \right) \]
\[ = 5666.8 \, \text{mm}^2 \quad 8.784 \, \text{in}^2 \]

Nozzle outs. vessel wall area
\[ A_2 = (t_o - c_o - c_i)L_o \]
\[ = 20552.7 \, \text{mm}^2 \quad 31.857 \, \text{in}^2 \]

Nozzle material factor
\[ f_m = \frac{S}{S_0} = 1.00000 \]

Pad material factor
\[ f_{sp} = \frac{S}{S_0} = 0 \]

Nozzle ins. vessel wall area
\[ A_3 = (t_o - 2c_o - c_i)L_o \]
\[ = 0 \, \text{mm}^2 \quad 0 \, \text{in}^2 \]

Area contributed by the outside nozzle fillet weld
\[ A_{A_1} = 0.5L_{A_1} \]
\[ = 112.5 \, \text{mm}^2 \quad 0.174 \, \text{in}^2 \]

Area contributed by the pad to vessel fillet weld
\[ A_{A_2} = 0.5L_{A_2} \]
\[ = 0 \, \text{mm}^2 \quad 0 \, \text{in}^2 \]

Area contributed by the inside nozzle fillet weld
\[ A_{A_3} = 0.5L_{A_3} \]
\[ = 0 \, \text{mm}^2 \quad 0 \, \text{in}^2 \]

Area contributed by the reinforcing pad
\[ A_5 = \min \left( A_{A_1}, A_{A_2}, A_{A_3} \right) \]
\[ = 0 \, \text{mm}^2 \quad 0 \, \text{in}^2 \]

Total area
\[ A_T = A_1 + f_m(A_2 + A_3) + A_{A_1} + A_{A_2} + A_{A_3} + f_m A_5 \]
\[ = 26332.0 \, \text{mm}^2 \quad 40.815 \, \text{in}^2 \]

Radius of the nozzle opening
\[ R_{nc} = R_n \]
\[ = 231.60 \, \text{mm} \quad 9.118 \, \text{in} \]

Nozzle radius for force calculation
\[ R_{n} = \frac{L_{n}(c_o - c_i)}{\ln \left( \frac{2(t_o - c_o - c_i)}{t_o - c_o - c_i} \right)} \]
\[ = 277.28 \, \text{mm} \quad 10.916 \, \text{in} \]

Shell radius for force calculation
\[ R_{s} = \frac{L_{s}(c_o - c_i)}{\ln \left( \frac{2(t_o - c_o - c_i)}{t_o - c_o - c_i} \right)} \]
\[ = 671.02 \, \text{mm} \quad 26.418 \, \text{in} \]

Force from internal pressure in the nozzle
\[ f_3 = PR_{as}(L_{e} + t_o - c_o - c_i) \]
\[ = 835415 \, \text{N} \quad 187808.68 \, \text{lbf} \]

Force from internal pressure in the shell
\[ f_6 = PR_{as}R_{n} \]
\[ = 1797667 \, \text{N} \quad 404129.29 \, \text{lbf} \]

Discontinuity force from internal pressure
\[ f_{Y} = \frac{PR_{as}R_{s}}{A_{T}} \]
\[ = 2209858 \, \text{N} \quad 496795.88 \, \text{lbf} \]

Average primary membrane stress
\[ \sigma_{awg} = \frac{f_m f_6 + f_m f_Y}{A_{T}} \]
\[ = 183.92 \, \text{MPa} \quad 26675.0 \, \text{psi} \]

General primary membrane stress
\[ \sigma_{awg} = \frac{PR_{as}}{A_{T}} \]
\[ = 153.90 \, \text{MPa} \quad 22321.1 \, \text{psi} \]

Allowable stress
\[ S_{allow} = 1.5SE \]
\[ = 237.90 \, \text{MPa} \quad 34504.5 \, \text{psi} \]

Maximum local primary membrane stress
\[ P_l = \max \left( 2\sigma_{awg} - \sigma_{circ}, \sigma_{circ} \right) \]
\[ = 213.94 \, \text{MPa} \quad 31028.9 \, \text{psi} \]

Maximum allowable pressure (bottom)
\[ P_{max} = \min \left( P_{max1}, P_{max2} \right) \]
\[ = 14.65 \, \text{MPa} \quad 2125.4 \, \text{psi} \]

Nozzle maximum allowable pressure (bottom)
\[ PL \leq Sallow: \quad Ok \]
\[ r \geq \min(0.25t,3\text{mm}(0.125\text{in})): \quad Ok \]
\[ PL \leq P_{max} : \quad Ok \]
Strength of nozzle attachment welds

Throat dimension of the nozzle to shell weld
\[ t_c = \frac{L_{41}}{\sqrt{2}} = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

Discontinuity force factor
\[ k_y = \frac{R_{w}}{R_{w} + t_c - c_w - c'_f} = 1.41883 \]

Nozzle to shell weld length
\[ L_c = \frac{R_c}{2} (R_c + t_c - c_w - c'_f) = 516.16 \text{ mm} \quad 20.321 \text{ in} \]

Throat dimension of the outside nozzle fillet weld
\[ L_{41} = 0.0707L_{41} = 10.61 \text{ mm} \quad 0.418 \text{ in} \]

Throat dimension for the pad to vessel fillet weld
\[ L_{42} = 0.0707L_{42} = 0 \text{ mm} \quad 0 \text{ in} \]

Throat dimension for inside nozzle fillet weld
\[ L_{43} = 0.0707L_{43} = 0 \text{ mm} \quad 0 \text{ in} \]

Welds force
\[ f_{\text{welds}} = \min\left[ f_y, \frac{1}{2}S_n(A_2 + A_3), \frac{\pi}{4}P_{\text{eff}}k_y^2 \right] = 1205907 \text{ N} \quad 271098.57 \text{ lbf} \]

Nozzle to shell groove weld depth
\[ t_{w1} = t_{\text{wall}} = 62.00 \text{ mm} \quad 2.441 \text{ in} \]

Average effective shear stress
\[ \tau = \frac{f_{\text{welds}}}{L_c(0.49L_{43} + 0.6S_{w1} + 0.49L_{43})} = 55.10 \text{ MPa} \quad 7992.3 \text{ psi} \]

\[ tc \geq \min[0.7tn, 6\text{mm (0.25in)}]: \text{Ok} \]
\[ \tau \leq S: \text{Ok} \]

External pressure

External design temperature
\[ T_e = 20.00 ^\circ \text{C} \quad 68.00 ^\circ \text{F} \]

Allowable stress from Annex 3.A for the vessel at the design temperature
\[ S = 207.00 \text{ MPa} \quad 30222.8 \text{ psi} \]

Allowable stress from Annex 3.A for the nozzle at the design temperature
\[ S_n = 207.00 \text{ MPa} \quad 30222.8 \text{ psi} \]

Nozzle inside radius
\[ R_n = \frac{d}{2} + c_{ni} + c_n' = 231.60 \text{ mm} \quad 9.118 \text{ in} \]

Shell inside diameter
\[ D_i = 1275.00 \text{ mm} \quad 50.197 \text{ in} \]

Effective radius of the shell
\[ R_{\text{eff}} = 0.5 \cdot D_i + c' = 640.50 \text{ mm} \quad 25.217 \text{ in} \]

Nozzle projection from the outside of the vessel wall
\[ L_{pr1} = 881.10 \text{ mm} \quad 34.689 \text{ in} \]

Nozzle projection from the inside of the vessel wall
\[ L_{pr2} = 0 \text{ mm} \quad 0 \text{ in} \]

Length of variable thickness from the outside of the vessel wall
\[ L_{pr3} = 150.00 \text{ mm} \quad 5.906 \text{ in} \]

Weld leg length of the outside nozzle fillet weld
\[ L_{41} = 15.00 \text{ mm} \quad 0.591 \text{ in} \]

Weld leg length of the pad to vessel fillet weld
\[ L_{42} = 0 \text{ mm} \quad 0 \text{ in} \]

Weld leg length of the inside nozzle fillet weld
\[ L_{43} = 0 \text{ mm} \quad 0 \text{ in} \]

Effective length along the vessel wall (limited by shell offset)
\[ L_R = \min\left( R_{\text{eff}} \cdot (t - c - d') \cdot 2R_c \right) = 91.40 \text{ mm} \quad 3.598 \text{ in} \]

Effective length along the nozzle wall outside the vessel
\[ L_{HI} = \min\left( L_{HI} - L_{pr1} - t_c - t_{pr} \right) = 149.88 \text{ mm} \quad 5.901 \text{ in} \]

Effective thickness
\[ t_{\text{eff}} = t - c - d' - \left( \frac{d_{f}'}{L_{\text{eff}}} \right) = 62.00 \text{ mm} \quad 2.441 \text{ in} \]

\[ L_I = \min\left( \frac{1.25}{D_i + t_{\text{eff}}} \right) = 199.60 \text{ mm} \quad 7.842 \text{ in} \]

\[ L_I = \min\left( L_{HI} - L_{HI} + t_c + t_{pr} \right) = 211.88 \text{ mm} \quad 8.342 \text{ in} \]

Effective length along the nozzle wall inside the vessel
\[ \lambda = \min\left( \frac{2R_c + t_c - c_w - c'_f}{\sqrt{(D_i + t_{\text{eff}}) + t_{\text{eff}}}} \right) = 1.94137 \]

Area contributed by the vessel wall
\[ A_1 = (t - c - d')L_d \cdot \max\left( \frac{2}{3}, 10 \right) = 5666.8 \text{ mm}^2 \quad 878.4 \text{ in}^2 \]

Nozzle outs. vessel wall area
\[ A_2 = (t_c - c_w - c'_f)L_d = 20552.7 \text{ mm}^2 \quad 31857 \text{ in}^2 \]

Nozzle material factor
\[ f_m = \frac{S}{S_n} = 1.00000 \]

Pad material factor
\[ f_{\text{pad}} = \frac{S}{S_n} = 0 \]

Nozzle ins. vessel wall area
\[ A_3 = (t_n - 2c_n - c'_n)L_d = 0 \text{ mm}^2 \quad 0 \text{ in}^2 \]

Area contributed by the outside nozzle fillet weld
\[ A_4 = 0.5L_{43} = 112.5 \text{ mm}^2 \quad 0.174 \text{ in}^2 \]
Area contributed by the pad to vessel fillet weld
\[ A_{dP} = 0.5 L_{dP} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]
Area contributed by the inside nozzle fillet weld
\[ A_{dA} = 0.5 L_{dA} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]
\[ A_{dS} = W_{dS} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]
\[ A_{5b} = 0 \text{ mm}^2 = 0 \text{ in}^2 \]
Area contributed by the reinforcing pad
\[ A_k = \min [A_{3a}, A_{3b}] = 0 \text{ mm}^2 = 0 \text{ in}^2 \]
Total area
\[ A_T = A_1 + J_n f_m (A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_m \frac{A_5}{A_T} = 26,332.0 \text{ mm}^2 = 40,815 \text{ in}^2 \]
Radius of the nozzle opening
\[ R_{cn} = R_n = 231.60 \text{ mm} = 9.118 \text{ in} \]
Nozzle radius for force calculation
\[ R_{3a} = \frac{R_{3n} - c_n - c_0}{\sqrt{2}} = 277.28 \text{ mm} = 10.916 \text{ in} \]
Shell radius for force calculation
\[ R_{3a} = \frac{R_{3n} - c_n - c_0}{\sqrt{2}} = 277.28 \text{ mm} = 10.916 \text{ in} \]
Force from internal pressure in the nozzle
\[ f_g = PR_{3a} = 6,051 \text{ N} = 1,360.39 \text{ lbf} \]
Force from internal pressure in the shell
\[ f_y = PR_{3n} = 13,021 \text{ N} = 2,927.31 \text{ lbf} \]
Discontinuity force from internal pressure
\[ f_\tau = \left( f_m + f_g + \frac{f_y}{A_T} \right) \frac{\sigma_{avg}}{\sigma_{ae}} = 1.33 \text{ MPa} = 193.2 \text{ psi} \]
Average primary membrane stress
\[ \sigma_{ae} = \frac{PR_{3a}}{A_{3a}} = 1.11 \text{ MPa} = 161.7 \text{ psi} \]
General primary membrane stress
\[ S_{allow} = \frac{f_{ha}}{A_{7}} = 185.96 \text{ MPa} = 26,971.6 \text{ psi} \]
Allowable stress
\[ P_{max} = \max \left( 2 \sigma_{avg} - \sigma_{ae}, \sigma_{circ} \right) = 1.55 \text{ MPa} = 224.8 \text{ psi} \]
Maximum local primary membrane stress
\[ P_{max} = \max \left( S_{allow}, \sigma_{circ} \right) = 12.36 \text{ MPa} = 1,792.7 \text{ psi} \]
Nozzle maximum allowable pressure (bottom)
\[ P_{max} = \min \left( P_{max1}, P_{max2} \right) = 12.36 \text{ MPa} = 1,792.7 \text{ psi} \]
Area resisting pressure
\[ A_P = \frac{f_m + f_g + \frac{f_y}{A_T}}{P} = 340,580.3 \text{ mm}^2 = 527,900 \text{ in}^2 \]
Strength of nozzle attachment welds
Throat dimension of the nozzle to shell weld
\[ tc = \frac{L_{41}}{\sqrt{2}} = 10.61 \text{ mm} = 0.418 \text{ in} \]
Discontinuity force factor
\[ k_{\tau} = \frac{R_{3a} + t_{a} - c_n - c_0}{R_{3n}} = 1.41883 \]
Nozzle to shell weld length
\[ L_{a} = \frac{\pi}{2} \left( R_{3a} + t_{a} - c_n - c_0 \right) = 516.16 \text{ mm} = 20.321 \text{ in} \]
Throat dimension of the outside nozzle fillet weld
\[ L_{417} = 0.0707 L_{a} = 10.61 \text{ mm} = 0.418 \text{ in} \]
Throat dimension for the pad to vessel fillet weld
\[ L_{432} = 0.0707 L_{a} = 0 \text{ mm} = 0 \text{ in} \]
Throat dimension for inside nozzle fillet weld
\[ L_{437} = 0.0707 L_{a} = 0 \text{ mm} = 0 \text{ in} \]
Welds force
\[ f_{\text{welds}} = \min \left( f_{m} k_{\tau}, 1.5 S_{n}(A_2 + A_3), \frac{\pi}{4} PR_{3n} k_{\tau}^2 \right) = 8,735 \text{ N} = 1,963.70 \text{ lbf} \]
Nozzle to shell groove weld depth
\[ tw_1 = tw_{all} = 62.00 \text{ mm} = 2.441 \text{ in} \]
Average effective shear stress
\[ \tau = \frac{f_{\text{welds}}}{L_{a} \left( 0.45 L_{417} + 0.52 w_1 + 0.45 L_{437} \right)} = 0.40 \text{ MPa} = 57.9 \text{ psi} \]
\[ tc \geq \min \left( 0.7 t_{n}, 6 \text{ mm} \left( 0.25 \text{ in} \right) \right); \text{ Ok} \]
\[ \tau \leq S; \text{ Ok} \]
<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flange material</td>
<td>SA-182 F22 3 - Forgings</td>
<td></td>
</tr>
<tr>
<td>Nozzle material</td>
<td>SA-182 F22 3 - Forgings</td>
<td></td>
</tr>
<tr>
<td>Bolting material</td>
<td>SA-193 B16 - Bolting</td>
<td></td>
</tr>
<tr>
<td>Gasket</td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
<td></td>
</tr>
<tr>
<td>Flange standard / specification</td>
<td>=</td>
<td>ASME B16.5 2013</td>
</tr>
<tr>
<td>Flange rating</td>
<td>=</td>
<td>1 500</td>
</tr>
<tr>
<td>Nominal size</td>
<td>=</td>
<td>18”</td>
</tr>
<tr>
<td>Number of bolts</td>
<td>=</td>
<td>16</td>
</tr>
<tr>
<td>Bolt type</td>
<td>=</td>
<td>ANSI_TEMA 2-1/2”</td>
</tr>
<tr>
<td>Material group</td>
<td>=</td>
<td>1.10</td>
</tr>
<tr>
<td>Calculation temperature</td>
<td>T =</td>
<td>420.00 °C</td>
</tr>
<tr>
<td>Internal pressure</td>
<td>Pd =</td>
<td>14.22 MPa</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>Ph =</td>
<td>0 MPa</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>P =</td>
<td>14.22 MPa</td>
</tr>
<tr>
<td>Maximum pressure at temperature allowed by the specifications</td>
<td>Pmax =</td>
<td>17.67 MPa</td>
</tr>
<tr>
<td>Maximum allowable pressures (at the top of the vessel)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>New &amp; cold (flange)</td>
<td>=</td>
<td>25.86 MPa</td>
</tr>
<tr>
<td>Hot &amp; corroded (flange)</td>
<td>=</td>
<td>17.67 MPa</td>
</tr>
<tr>
<td>New &amp; cold (bolts)</td>
<td>=</td>
<td>25.86 MPa</td>
</tr>
<tr>
<td>Hot &amp; corroded (bolts)</td>
<td>=</td>
<td>17.67 MPa</td>
</tr>
<tr>
<td>New &amp; cold (cylinder)</td>
<td>=</td>
<td>55.23 MPa</td>
</tr>
<tr>
<td>Hot &amp; corroded (cylinder)</td>
<td>=</td>
<td>40.25 MPa</td>
</tr>
<tr>
<td>External pressure</td>
<td>Pe =</td>
<td>0.10 MPa</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>Ph =</td>
<td>0 MPa</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>P = Pe + Ph =</td>
<td>0.10 MPa</td>
</tr>
<tr>
<td>Maximum pressure at temperature allowed by the specifications</td>
<td>Pmax =</td>
<td>17.67 MPa</td>
</tr>
</tbody>
</table>
**Cylindrical shell**

Allowable stress at room temperature \( ST = 207.00 \text{ MPa} \) = 30 022.8 psi

Joint efficiency \( E = 1.00 \)

Corrosion allowance \( c = 3.00 \text{ mm} \) = 0.118 in

External corrosion allowance \( ce = 0 \text{ mm} \) = 0 in

Wall undertolerance \( c' = 0 \text{ mm} \) = 0 in

Inside diameter \( D = 457.20 \text{ mm} \) = 18.000 in

Length \( L = 1 \text{ 016.00 mm} \) = 40.000 in

Adopted thickness \( t = 69.90 \text{ mm} \) = 2.752 in

Allowable stress \( S = 158.60 \text{ MPa} \) = 23 003.0 psi

Internal pressure \( Pi = 14.22 \text{ MPa} \) = 2 062.4 psi

Overpressure due to static head \( Ph = 0 \text{ MPa} \) = 0 psi

Calculation pressure \( P = Pi + Ph = 14.22 \text{ MPa} \) = 2 062.4 psi

Required thickness \( t_r = \frac{D + 2(c + c')}{2} (e_x - 1) + c + ce + c' = 24.72 \text{ mm} \) = 0.973 in

\( t \geq t_r: \text{ Ok} \)

**External pressure**

External pressure \( Pe = 0.10 \text{ MPa} \) = 14.9 psi

External static head \( Ph = 0 \text{ MPa} \) = 0 psi

Calculation pressure \( P = Pe + Ph = 0.10 \text{ MPa} \) = 14.9 psi

External design temperature \( Te = 20.00 \text{ °C} \) = 68.00 °F

Outside diameter \( Do = 915.00 \text{ mm} \) = 36.024 in

Max unsupported length \( L = 1 \text{ 016.00 mm} \) = 40.000 in

Modulus of elasticity \( Ey = 210 \text{ 350.00 MPa} \) = 30 508 688.2 psi

Shell parameter \( M_s = \frac{L}{\sqrt{R_e(t - c - c')}} = 5.80744 \)

\( C_h = \frac{M_s - 0.579}{0.92} = 0.17596 \)

Elastic buckling stress \( F_{ha} = \frac{16 \cdot C_h \cdot E_y \cdot (t - c - ce - c')}{D_o^2} = 4329.95 \text{ MPa} \) = 628 006.1 psi

Buckling stress \( Fic = Sy = 310.00 \text{ MPa} \) = 44 961.7 psi

Yield strength at design temperature \( Sy = 310.00 \text{ MPa} \) = 44 961.7 psi

Design factor \( FS = 1.66700 \)

Hoop compressive membrane stress \( Fha = Fic / FS = 185.96 \text{ MPa} \) = 26 971.6 psi

Allowable external pressure in the absence of other loads \( P_a = 2F_{ha} \left( t - c - ce - c' \right) = 27.19 \text{ MPa} \) = 3 944.0 psi

Minimum required thickness \( tr = 5.81 \text{ mm} \) = 0.229 in

\( t \geq tr: \text{ Ok} \)

**Minimum Design Metal Temperature (MDMT)**

Long Welding Neck flange - Out Shell Side (Bolting)

Material = SA-193 B16

Curve of fig. 3.7 / 3.8 = None

Governing Thickness \( tg = 63.50 \text{ mm} \) = 2.500 in

PostWeld Heat Treatment = No

Minimum Design Metal Temperature (MDMT) = -29.00 °C = -20.20 °F
### Long Welding Neck flange - Out Shell Side (Flange)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>tg = 69.90 mm (2.752 in)</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)

### External pressure

### Long Welding Neck flange - Out Shell Side (Bolting)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-193 B16</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>tg = 63.50 mm (2.500 in)</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature</td>
<td>-29.00 °C (-20.20 °F)</td>
</tr>
</tbody>
</table>

### Long Welding Neck flange - Out Shell Side (Flange)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
</tr>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>tg = 69.90 mm (2.752 in)</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)
**Standard Long Welding Neck flange - In Shell Side**

*According to: Asme VIII Div. 2 Ed. 2015, 4.16 - Metric Units*

<table>
<thead>
<tr>
<th>Flange material</th>
<th>SA-182 F22 3 - Forgings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nozzle material</td>
<td>SA-182 F22 3 - Forgings</td>
</tr>
<tr>
<td>Bolting material</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>Gasket</td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Flange standard / specification</th>
<th>ASME B16.5 2013</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flange rating</td>
<td>1 500</td>
</tr>
<tr>
<td>Nominal size</td>
<td>18&quot;</td>
</tr>
<tr>
<td>Number of bolts</td>
<td>16</td>
</tr>
<tr>
<td>Bolt type</td>
<td>ANSI_TEMA 2-1/2&quot;</td>
</tr>
<tr>
<td>Material group</td>
<td>1.10</td>
</tr>
</tbody>
</table>

Calculation temperature

\[ T = 420.00 \, ^\circ C \quad 788.00 \, ^\circ F \]

Internal pressure

\[ P_{d} = 14.22 \, \text{MPa} \quad 2062.4 \, \text{psi} \]

Overpressure due to static head

\[ Ph = 0 \, \text{MPa} \quad 0 \, \text{psi} \]

Calculation pressure

\[ P = 14.22 \, \text{MPa} \quad 2062.4 \, \text{psi} \]

Maximum pressure at temperature allowed by the specifications

\[ P_{\text{max}} = 17.67 \, \text{MPa} \quad 2562.8 \, \text{psi} \]

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

New & cold (flange)

\[ = 25.86 \, \text{MPa} \quad 3750.7 \, \text{psi} \]

Hot & corroded (flange)

\[ = 17.67 \, \text{MPa} \quad 2562.8 \, \text{psi} \]

New & cold (bolts)

\[ = 25.86 \, \text{MPa} \quad 3750.7 \, \text{psi} \]

Hot & corroded (bolts)

\[ = 17.67 \, \text{MPa} \quad 2562.8 \, \text{psi} \]

New & cold (cylinder)

\[ = 55.23 \, \text{MPa} \quad 8010.0 \, \text{psi} \]

Hot & corroded (cylinder)

\[ = 40.25 \, \text{MPa} \quad 5837.2 \, \text{psi} \]

**External pressure**

External design pressure

\[ Pe = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]

Overpressure due to static head

\[ Ph = 0 \, \text{MPa} \quad 0 \, \text{psi} \]

Calculation pressure

\[ P = Pe + Ph = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]

Maximum pressure at temperature allowed by the specifications

\[ P_{\text{max}} = 17.67 \, \text{MPa} \quad 2562.8 \, \text{psi} \]
Cylindrical shell

Allowable stress at room temperature \( ST = 207.00 \text{ MPa} = 30 022.8 \text{ psi} \)
Joint efficiency \( E = 1.00 \)
Corrosion allowance \( c = 3.00 \text{ mm} = 0.118 \text{ in} \)
External corrosion allowance \( ce = 0 \text{ mm} = 0 \text{ in} \)
Wall undertolerance \( c' = 0 \text{ mm} = 0 \text{ in} \)
Inside diameter \( D = 457.20 \text{ mm} = 18.000 \text{ in} \)
Length \( L = 1 016.00 \text{ mm} = 40.000 \text{ in} \)
Adopted thickness \( t = 69.90 \text{ mm} = 2.752 \text{ in} \)
Allowable stress \( S = 158.60 \text{ MPa} = 23 003.0 \text{ psi} \)
Internal pressure \( Pi = 14.22 \text{ MPa} = 2 062.4 \text{ psi} \)
Overpressure due to static head \( Ph = 0 \text{ MPa} = 0 \text{ psi} \)
Calculation pressure \( P = Pi + Ph = 14.22 \text{ MPa} = 2 062.4 \text{ psi} \)
Required thickness \( t_r = \frac{D + 2(c + c')}{2} = 24.72 \text{ mm} = 0.973 \text{ in} \)
\( t \geq t_r: \text{ Ok} \)

External pressure

External pressure \( Pe = 0.10 \text{ MPa} = 14.9 \text{ psi} \)
External static head \( Ph = 0 \text{ MPa} = 0 \text{ psi} \)
Calculation pressure \( P = Pe + Ph = 0.10 \text{ MPa} = 14.9 \text{ psi} \)
External design temperature \( Te = 20.00 \, ^\circ\text{C} = 68.00 \, ^\circ\text{F} \)
Outside diameter \( Do = 915.00 \text{ mm} = 36.024 \text{ in} \)
Max unsupported length \( L = 1 016.00 \text{ mm} = 40.000 \text{ in} \)
Modulus of elasticity \( Ey = 210 350.00 \text{ MPa} = 30 508 688.2 \text{ psi} \)
Shell parameter \( M_s = \frac{L}{\sqrt{R_c(t - c - c')}} = 5.80744 \)
\( C_h = \frac{M_s - 0.579}{0.92} = 0.17596 \)
Elastic buckling stress \( F_{ha} = \frac{16}{C_h^2} \cdot Ey \cdot (t - c - c') = 4 329.95 \text{ MPa} = 628 006.1 \text{ psi} \)
Buckling stress \( Fic = Sy = 310.00 \text{ MPa} = 44 961.7 \text{ psi} \)
Yield strength at design temperature \( Sy = 310.00 \text{ MPa} = 44 961.7 \text{ psi} \)
Design factor \( FS = 1.66700 \)
Hoop compressive membrane stress \( Fha = Fic / FS = 185.96 \text{ MPa} = 26 971.6 \text{ psi} \)
Allowable external pressure in the absence of other loads \( P_x = 2F_{ha} \left( \frac{t - c - c'}{D_o} \right) = 27.19 \text{ MPa} = 3 944.0 \text{ psi} \)
Minimum required thickness \( tr = 5.81 \text{ mm} = 0.229 \text{ in} \)
\( t \geq tr: \text{ Ok} \)

Minimum Design Metal Temperature (MDMT)

Long Welding Neck flange - In Shell Side (Bolting)

Material \( = \) SA-193 B16
Curve of fig. 3.7 / 3.8 \( = \) None
Governing Thickness \( tg = 63.50 \text{ mm} = 2.500 \text{ in} \)
PostWeld Heat Treatment \( = \) No
Minimum Design Metal Temperature (MDMT) \( = -29.00 \, ^\circ\text{C} = -20.20 \, ^\circ\text{F} \)
Long Welding Neck flange - In Shell Side (Flange)

<table>
<thead>
<tr>
<th>Material</th>
<th>=</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>=</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>( tg = 69.90 ) mm</td>
<td>2.752 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>=</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>=</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>=</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

External pressure

Long Welding Neck flange - In Shell Side (Bolting)

<table>
<thead>
<tr>
<th>Material</th>
<th>=</th>
<th>SA-193 B16</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>=</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>( tg = 63.50 ) mm</td>
<td>2.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>=</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>=</td>
<td></td>
</tr>
</tbody>
</table>

Long Welding Neck flange - In Shell Side (Flange)

<table>
<thead>
<tr>
<th>Material</th>
<th>=</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>=</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>( tg = 69.90 ) mm</td>
<td>2.752 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>=</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>=</td>
<td></td>
</tr>
<tr>
<td>Impact tests required by Code</td>
<td>=</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*
U-Tube tubesheet - Tubesheet

According to: ASME VIII Div. 2 Ed. 2015, 4.18 - Metric Units

Operating conditions

<table>
<thead>
<tr>
<th>Component</th>
<th>Design temperature</th>
<th>Design pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell</td>
<td>$T_s = 420.00 \degree C / 788.00 \degree F$</td>
<td>$P_s = 14.22 \text{ MPa} / 2062.4 \text{ psi}$</td>
</tr>
<tr>
<td>Channel</td>
<td>$T_c = 454.00 \degree C / 849.20 \degree F$</td>
<td>$P_t = 18.44 \text{ MPa} / 2674.0 \text{ psi}$</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>$T = 454.00 \degree C / 849.20 \degree F$</td>
<td></td>
</tr>
<tr>
<td>Tubes</td>
<td>$T_t = 454.00 \degree C / 849.20 \degree F$</td>
<td></td>
</tr>
</tbody>
</table>

### Tubesheet material

**SA-182 F22 3 - Forgings**
- Tubesheet design temperature: $T = 454.00 \degree C / 849.20 \degree F$
- Modulus of elasticity for tubesheet material at $T$: $E = 179,600.00 \text{ MPa} / 26,048,777.7 \text{ psi}$
- Allowable stress for tubesheet material at $T$: $S = 149.84 \text{ MPa} / 21,732.5 \text{ psi}$
- Allowable primary plus secondary stress for tubesheet material: $SPS = 2 \cdot S = 461.76 \text{ MPa} / 66,972.6 \text{ psi}$

### Tubes material

**SB-517 Cold drawn/ann. N06600 - Wld. pipe**
- Tube design temperature: $T_t = 454.00 \degree C / 849.20 \degree F$
- Modulus of elasticity for tube material at $T_t$: $E_t = 188,760.00 \text{ MPa} / 27,377,323.4 \text{ psi}$
- Allowable stress for tube material at tubesheet design temperature: $S_{T_t} = 161.18 \text{ MPa} / 23,376.7 \text{ psi}$
- Allowable stress of the welded product divided by 0.85: $SP_{S_{T_t}} = 2 \cdot S_{T_t} = 461.76 \text{ MPa} / 66,972.6 \text{ psi}$

### Channel material

**SA-387 22 2 - Plate**
- Channel design temperature: $T_c = 454.00 \degree C / 849.20 \degree F$
- Modulus of elasticity for channel material at $T_c$: $E_c = 179,600.00 \text{ MPa} / 26,048,777.7 \text{ psi}$
- Poisson's ratio of channel material: $\nu_c = 0.30$
- Allowable stress for channel material at $T_c$: $S_c = 149.84 \text{ MPa} / 21,732.5 \text{ psi}$
- Allowable primary plus secondary stress for channel material at $T_c$: $SP_{S_{c}} = 2 \cdot S_c = 461.76 \text{ MPa} / 66,972.6 \text{ psi}$

### Shell material

**SA-387 22 2 - Plate**
- Shell design temperature: $T_s = 420.00 \degree C / 788.00 \degree F$
- Modulus of elasticity for shell material at $T_s$: $E_s = 182,400.00 \text{ MPa} / 26,454,883.4 \text{ psi}$
- Poisson's ratio of shell material: $\nu_s = 0.30$
- Allowable stress for shell material at $T_s$: $S_s = 158.60 \text{ MPa} / 23,003.0 \text{ psi}$
- Allowable primary plus secondary stress for shell material at $T_s$: $SP_{S_{s}} = 3 \cdot S_s = 475.80 \text{ MPa} / 69,009.0 \text{ psi}$

### Bolting material

**SA-193 B16 (Code Case 2655 - using Division 1 stress tables) - Bolting**
- Allowable stress for the bolt evaluated at the design temperature: $S_{b_o} = 129.92 \text{ MPa} / 18,843.3 \text{ psi}$
- Allowable stress for the bolt evaluated at the gasket seating temperature: $S_{b_g} = 138.00 \text{ MPa} / 20,015.2 \text{ psi}$
Geometric data

Outside diameter of tubesheet \( A = 1900.00 \text{ mm} \) \( = 74.803 \text{ in} \)

Bolt circle diameter \( C = 1700.00 \text{ mm} \) \( = 66.929 \text{ in} \)

Shell corrosion allowance \( c_s = 3.00 \text{ mm} \) \( = 0.118 \text{ in} \)

Shell undertolerance \( c_s' = 0 \text{ mm} \) \( = 0 \text{ in} \)

Shell thickness \( t_z = t_{\text{shell}} - c_s - c_s' = 62.00 \text{ mm} \) \( = 2.441 \text{ in} \)

Channel corrosion allowance \( c_c = 0 \text{ mm} \) \( = 0 \text{ in} \)

Channel undertolerance \( c_c' = 0 \text{ mm} \) \( = 0 \text{ in} \)

Channel thickness \( t_z = t_{\text{channel}} - c_c - c_c' = 85.00 \text{ mm} \) \( = 3.346 \text{ in} \)

Perimeter of the tube layout \( C_P = 3764.42 \text{ mm} \) \( = 148.206 \text{ in} \)

Area enclosed by perimeter \( C_P \) \( A_p = 1127677.1 \text{ mm}^2 \) \( = 1747.903 \text{ in}^2 \)

Tubeside corrosion allowance \( c_{ts} = 0 \text{ mm} \) \( = 0 \text{ in} \)

Shellside corrosion allowance \( c_{ss} = 3.00 \text{ mm} \) \( = 0.118 \text{ in} \)

Tubesheet undertolerance \( c'_t = 0 \text{ mm} \) \( = 0 \text{ in} \)

Tubesheet thickness \( t_{\text{tubesheet}} = 230.00 \text{ mm} \) \( = 9.055 \text{ in} \)

Nominal outside diameter of tubes \( d_t = 31.75 \text{ mm} \) \( = 1.250 \text{ in} \)

Equivalent diameter of outer tube limit circle \( \varnothing = D_0 = 2r_0 + d_t = 1230.00 \text{ mm} \) \( = 48.425 \text{ in} \)

Triangular tube pitch \( p = 42.33 \text{ mm} \) \( = 1.667 \text{ in} \)

Nominal tube wall thickness \( t_t = 4.19 \text{ mm} \) \( = 0.165 \text{ in} \)

Total area of untubed lanes \( A_L = 0 \text{ mm}^2 \) \( = 0 \text{ in}^2 \)

Effective tube side pass partition groove depth \( h_g = \max \left( (h_g - c_t), 0 \right) = 0 \text{ mm} \) \( = 0 \text{ in} \)

Diameter of shell gasket load reaction \( G_s = 1301.06 \text{ mm} \) \( = 51.223 \text{ in} \)

Diameter ratio \( \beta_c = \frac{D_c}{D_t} \) \( = 1.05777 \)

Inside channel diameter \( D_c = 1275.00 \text{ mm} \) \( = 50.197 \text{ in} \)

Diameter ratio \( \beta_c = \frac{D_c}{D_t} \) \( = 1.03659 \)

Coefficient \( \beta = \frac{12(1 - N^2)}{[(D_t + t_t)\gamma]^{1/2}} \) \( = 0.0053 \text{ mm}^-1 \) \( = 0.136 \text{ in}^-1 \)

\( k_c = \frac{E_d^2}{D_c} \) \( = 108005833 \text{ N} \) \( = 24280674.98 \text{ lbf} \)

\( \lambda_c = \frac{6D_c k_c}{\beta_c} \left( 1 + h_\beta + \frac{h_\beta^2}{2} \right) \) \( = 208391.28 \text{ MPa} \) \( = 30224599.9 \text{ psi} \)

\( \delta_c = \frac{D_c^2}{4E_d t_c} \left( 1 + \frac{N^2}{2} \right) \) \( = 0.02263 \)

\( \omega_c = \frac{D_c k_c \beta_c (1 + h_\beta)}{2} \) \( = 29984.5 \text{ mm}^2 \) \( = 46476 \text{ in}^2 \)

Diameter ratio \( K = \frac{D_c}{D_t} \) \( = 1.54472 \)

Coefficient \( \begin{align*} F &= \frac{1 - \nu}{E} \left( \lambda_c + E \cdot \ln \left( \frac{K}{\beta} \right) \right) \\ &= 2.33305 \end{align*} \)

Minimum RCB 7.11 thickness

TEMA Class \( t - c_t - c_s \geq D_0: \text{Ok} \)

\( t \geq 19.10 \text{ mm}: \text{Ok} \)
Flanged extension

Design bolt load for the operating condition
\[ W_o = 0.785 \sigma_P^2 + 2 \pi \sigma_G \rho \] = 22 832 134 N 5 132 867.40 lbf

Design bolt load for the gasket seating (Flange)
\[ W_g = \left( \frac{A_m + A_b}{2} \right) S_{bg} \] = 30 162 175 N 6 780 726.10 lbf

Moment arm for load HG
\[ h_G = \frac{(C-G)}{2} \] = 199.47 mm 7.853 in

Allowable stress for the tubesheet extension at design temperature
\( \sigma_o = 149.84 \text{ MPa} \) 21 732.5 psi

Allowable stress for the tubesheet extension at gasket seating temperature
\( \sigma_g = 207.00 \text{ MPa} \) 30 022.8 psi

Flanged extension thickness
\[ t_{fe} = \text{flanged extension} - C \] = 773.75 mm 30.463 in

Minimum required thickness of the tubesheet flanged extension (operating)
\[ h_{rg} = \sqrt{\frac{19W_hG}{S_o G}} \] = 210.68 mm 8.295 in

Minimum required thickness of the tubesheet flanged extension (gasket seating)
\[ h_{rg} = \sqrt{\frac{19W_hG}{S_g G}} \] = 206.02 mm 8.111 in

Minimum required thickness of the tubesheet flanged extension.
\[ h_c = \left( \frac{19W_hG}{S_o G} \right)^{\frac{3}{5}} \] = 210.68 mm 8.295 in

Tfe ≥ hr: Ok

Tube to tubesheet joints

Fillet weld leg
\[ \alpha_f = \sqrt{\left(0.75d_o\right)^2 + 2.73t(d_o-1)f_w}\] = 10.00 mm 0.394 in

Min. required length of the weld leg(s)
\[ a_d = \sqrt{\left(0.75d_o\right)^2 + 2.73t(d_o-1)f_w}\] = 5.89 mm 0.232 in

Allowable stress of the tube
\( \sigma_a = 137.00 \text{ MPa} \) 19 870.2 psi

Allowable stress of the material to which the tube is welded
\( \sigma_t = 149.84 \text{ MPa} \) 21 732.5 psi

Allowable stress in weld
\[ S_w = \min(S_o, S_t) \] = 137.00 MPa 19 870.2 psi

Fillet weld strength
\[ F_s = 0.55 \pi a_f (d_o + 0.67a_f)S_w \] = 91 018 N 20 461.76 lbf

Groove weld strength
\[ F_g = 0.85 \pi a_g (d_o + 0.67a_g)S_w \] = 0 N 0 lbf

Axial tube strength
\[ F_t = \pi t(d_o - r)S_o \] = 49 711 N 11 175.45 lbf

Ratio of the design strength to the tube strength
\[ \frac{F_d}{F_t} = 1 \] = 1.00000

Ratio of the fillet weld strength to the design strength
\[ \frac{F_s}{F_t} = 1 \] = 1.00000

Weld strength factor
\[ \frac{F_w}{S_w} = 1 \] = 1.00000

Max axial load (pressure only)
\[ L_{max} = F_t \] = 49 711 N 11 175.45 lbf

Max axial load (pressure or thermally induced)
\[ L_{max} = 2F_t \] = 99 422 N 22 350.90 lbf

Af ≥ max[ar, t]: Ok
Loading case Design 1: Pt = 18.44 MPa, Ps = -0.10 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear

\[ \mu = p - p_i \]

Effective tube pitch

\[ p = p_i \left( 1 - \frac{1}{4} \min \left( \frac{A_0}{\pi D_t^2} \right) \right)^{4.5} \]

= 0.24998

Effective tube pitch

\[ p = \frac{A_0}{p} \]

= 42.33 mm

1.667 in

Tube expansion depth ratio

\[ \rho = \frac{L - h}{h} \]

= 1.00000

Effective tube hole diameter

\[ d' = \max \left\{ \frac{d_i - 2h_i}{S} \left( \frac{S}{S} \right)^{\rho} \right\} \]

= 23.37 mm

0.920 in

Effective ligament efficiency for bending

\[ \mu = \frac{p - d'}{p} \]

= 0.44798

\[ A_0 = \frac{d_i}{2} \]

= -0.00290

\[ A_1 = \frac{d_i}{2} \]

= 0.21260

\[ A_2 = \frac{d_i}{2} \]

= 3.99060

\[ A_3 = \frac{d_i}{2} \]

= -6.17300

\[ A_4 = \frac{d_i}{2} \]

= 3.43070

\[ B_0 = \frac{d_i}{2} \]

= 0.99660

\[ B_1 = \frac{d_i}{2} \]

= -4.19780

\[ B_2 = \frac{d_i}{2} \]

= 9.04780

\[ B_3 = \frac{d_i}{2} \]

= -7.99550

\[ B_4 = \frac{d_i}{2} \]

= 2.23980

Effective Poisson ratio in perforated region

\[ \nu = \frac{B_3 + B_4 \mu - B_2 \mu^2 + B_1 \mu^3 + B_0 \mu^4}{E'} \]

= 0.47640

\[ E'/E = A_0 + A_1 \mu + A_2 (\mu^2)^2 + A_3 (\mu^2)^3 + A_4 (\mu^2)^4 \]

= 0.30322

**Tubesheet bending stress**

Tubesheet design bold load

\[ W^* = 0 \text{ N} \]

0 lbf

Moment acting on the unperforated tubesheet rim

\[ M_{TS} = \frac{D_t^2 (\rho - 1) (\rho^2 + 1) P_i - (\rho - 1) (\rho^2 + 1) P_e}{16} \]

= -133 502 N

-30 012.42 lbf

Moment acting on the unperforated tubesheet rim

\[ M = M_{TS} + \omega_c P_c + \frac{(C - O) W^*}{2\pi D_t} \]

= 419 308 N

94 264.11 lbf

Maximum bending moments acting on the tubesheet at the periphery

\[ M_p = \frac{M - \frac{32}{3} \frac{T}{1 + \nu}}{1 + \nu} \]

= 739 340 N

166 210.12 lbf

Maximum bending moments acting on the tubesheet at the center

\[ M_c = M_p + \frac{D_t^2 (3 + \nu) (P_e - P_i)}{64} \]

= -708 319 N

-159 236.38 lbf

Maximum bending moments acting on the tubesheet

\[ M = \max \left\{ |M_p|, |M_c| \right\} \]

= 739 340 N

166 210.12 lbf

Bending stress

\[ \sigma = \frac{M}{\mu (h - h_e)^2} \]

= 192.17 MPa

27 871.7 psi

\[ \sigma \leq 2S: \text{ Ok} \]

**Tubesheet shear stress**

\[ |P_s - P_t| \leq 3.2 \cdot S \cdot \mu \cdot h/D_o: \text{ Shear stress is not required to be calculated} \]

**Shell and channel stresses**

Axial membr. stress in the channel at tubesheet junction

\[ \sigma_{cm} = \frac{D_t^2 P_e}{4A(D_t + t_1)} \]

= 64.82 MPa

9 400.7 psi

Bending stress in the channel at its junction to the tubesheet

\[ \sigma_{cb} = \frac{D_t^2 P_e}{4(D_t - t_1)} \left( \frac{D_t}{K} \right) \left( 1 + \frac{h_0}{2} \right) \left( \frac{M_p + \frac{D_t^2}{32} (P_e - P_i)}{1 + \nu} \right) \]

= 301.64 MPa

43 749.5 psi

Channel axial stress

\[ \sigma_c = \sigma_{cm} + \sigma_{cb} \]

= 366.46 MPa

53 150.2 psi
Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation

\[ E'_c = E_A \frac{15S_A}{\sigma_c} = 140,654.57 \text{ MPa} \quad 20,400,220.3 \text{ psi} \]

\[ k_c = \frac{\beta E'_c R}{6(1-\nu)} = 84,585,265 \text{ N} \quad 19,015,522.45 \text{ lbf} \]

\[ \lambda_c = \frac{6D_k R}{R^3} \left( 1 + h \beta_c + \frac{1 - 2\nu}{2} \right) = 163,202.59 \text{ MPa} \quad 23,670,534.7 \text{ psi} \]

\[ \alpha_c = \rho k_c \beta_c \delta_c (1 + h \beta_c) = 23,482.5 \text{ mm}^2 \quad 36.398 \text{ in}^2 \]

Diameter ratio

\[ K = \frac{A}{D_k} = 1.54472 \]

Coefficient

\[ F = \frac{1 - \nu}{F} (\lambda_c + E \cdot \ln[K]) = 1.96505 \]

Maximum bending moments acting on the tubesheet at the periphery

\[ M_p = M - \frac{M}{32} \frac{1+E}{1-F} = 722,315 \text{ N} \quad 162,382.78 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the center

\[ M_0 = M_p + \frac{D_k^2 (3 + \nu)(P_s - P_t)}{64} = -725,344 \text{ N} \quad -163,063.73 \text{ lbf} \]

Maximum bending moments acting on the tubesheet

\[ M = \max \left[ \left| M_0 \right|, \left| M_p \right| \right] = 725,344 \text{ N} \quad 163,063.73 \text{ lbf} \]

Bending stress (simplified Elastic-plastic calculation performed)

\[ \sigma = \frac{M}{\mu^2(h-k_c)^3} = 188.53 \text{ MPa} \quad 27,344.1 \text{ psi} \]

\[ \sigma \leq 2S: \text{ Ok} \]
Loading case Design 2: Pt = -0.10 MPa, Ps = 14.22 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear
\[ \mu = \frac{p - d_i}{P} = 0.24998 \]

Effective tube pitch
\[ p^* = p \left[ 1 - 4 \cdot \min \left\{ \frac{A_i}{4D \times p} \right\} \right]^{-\frac{1}{4}} = 42.33 \text{ mm} \]

Tube expansion depth ratio
\[ \rho = \frac{h}{h_i} = 1.00000 \]

Effective tube hole diameter
\[ d^* = \max \left( \frac{d_i - 2\epsilon_i (E_i + \left( \frac{S_i}{S} \right)^p \rho_i), (d_i - 2h_i)}{\rho_i}, \rho \right) = 23.37 \text{ mm} \]

Effective ligament efficiency for bending
\[ \mu^* = \frac{E^*}{p^*} = 0.47498 \]

A0 = -0.00290
A1 = 0.21260
A2 = 3.99060
A3 = -6.17300
A4 = 3.43070
B0 = 0.99660
B1 = -4.19780
B2 = 9.04780
B3 = -7.99550
B4 = 2.23980

\[ \frac{E^*}{E} = A_0 + A_1 \mu + A_2 (\mu^*) + A_3 (\mu^*)^2 + A_4 (\mu^*)^3 \]

Effective Poisson ratio in perforated region
\[ \nu^* = B_0 + B_1 \mu + B_2 (\mu^*) + B_3 (\mu^*)^2 + B_4 (\mu^*)^3 = 0.30322 \]

Tubesheet bending stress

Shell flange design bolt load for the operating condition
\[ W_{ds} = 22,832,134 \text{ N} \quad 5,132,867.40 \text{ lbf} \]

Tubesheet design bold load
\[ W^* = W_{ds} = 22,832,134 \text{ N} \quad 5,132,867.40 \text{ lbf} \]

Moment acting on the unperforated tubesheet rim
\[ M_{TS} = \frac{D_i^2}{16} (\rho_2 - 1) (\rho_2 + 1)P_i - (\rho_1 - 1) (\rho_1 + 1)P_j \]

Moment acting on the unperforated tubesheet rim
\[ M' = M_{TS} + \omega_x P_i \left( \frac{C - G_x}{2\pi G_x} \right) W^* = 1,340,848 \text{ N} \quad 301,434.51 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the periphery
\[ M_p = \frac{M - \frac{D_i}{2} \frac{1 + \mu^*}{2}}{64} = -71,697 \text{ N} \quad -16,118.20 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the center
\[ M_p = \frac{\left( 3 + \nu^* \right) (P_i - P_j)}{64} = 1,046,687 \text{ N} \quad 235,304.64 \text{ lbf} \]

Maximum bending moments acting on the tubesheet
\[ M = \max \left[ \frac{M}{4M} \right] = 1,046,687 \text{ N} \quad 235,304.64 \text{ lbf} \]

Bending stress
\[ \sigma = \frac{272.05 \text{ MPa} \quad 39,458.1 \text{ psi}}{\mu^*(h - h_c)} \]

\[ \sigma \leq 2S: \text{ Ok} \]

Tubesheet shear stress

Shell and channel stresses

Axial memb. stress in the channel at tubesheet junction
\[ \sigma_{\alpha m} = \frac{D_i^2 P_i}{4(D_i + t)} = -0.36 \text{ MPa} \quad -52.5 \text{ psi} \]

Bending stress in the channel at its junction to the tubesheet
\[ \sigma_{\alpha b} = \frac{6E}{R} \left( \beta \delta P_i - \frac{6(1 - \nu^*)}{E^*} \left( \frac{D_i}{R} \right) \left( 1 + \frac{h \beta}{2} \right) \left( M_p + \frac{D_i^2}{32} (P_i - P_j) \right) \right) = -449.46 \text{ MPa} \quad -65,188.2 \text{ psi} \]

Channel axial stress
\[ \sigma_c = |\sigma_{\alpha m} + |\sigma_{\alpha b} = 449.82 \text{ MPa} \quad 65,240.7 \text{ psi} \]
Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation

\[ E' = E \sqrt{\frac{1}{\varepsilon_c}} \]

\[ k_c = \frac{\beta E' \beta_c}{6(1-\nu^2)} \]

\[ \lambda_c = \frac{6D_b k_c}{R^3} \left(1 + \frac{h_b c}{2}\right) \]

\[ \omega_c = \frac{\rho_i k_c \beta_c}{6} \left(1 + \frac{h_b c}{2}\right) \]

Diameter ratio

\[ \frac{D}{D_c} = \frac{A}{A_c} \]  

Coefficient

\[ F = \frac{(1-\nu)(\lambda_c + B \cdot \ln[K])}{F_c^2} \]

Maximum bending moments acting on the tubesheet at the periphery

\[ M_p = M - \frac{h_b c}{2} \]

\[ M_p = M + \frac{h_b c}{2} \frac{3}{2} (3 + 4 \nu) (P_2 - P_1) \]

Maximum bending moments acting on the tubesheet at the center

\[ M_0 = M_p + \frac{h_b c}{2} \frac{3}{2} (3 + 4 \nu) (P_2 - P_1) \]

Maximum bending moments acting on the tubesheet

\[ M = \max [\frac{M_p}{M}] \]

Bending stress (simplified Elastic-plastic calculation performed)

\[ \sigma = \frac{M}{\mu^2 (h - h_c)^3} \]

\[ \sigma \leq 2\sigma : \text{Ok} \]
Loading case Design 3: Pt = 18.44 MPa, Ps = 14.22 MPa, thermal exp.: N, corr.: Y, vacuum: N

Basic ligament efficiency for shear
\[ \mu = \frac{p - d_t}{P} = 0.24998 \]

Effective tube pitch
\[ P^* = \frac{1}{4.45} \min \left[ \frac{A_d}{\pi D_s^3} \right] = 42.33 \text{ mm} \]

Tube expansion depth ratio
\[ \rho^* = \frac{h}{h_t} = 1.00000 \]

Effective tube hole diameter
\[ d^* = \max \left[ \left( \frac{S_t}{S} \right)^{\frac{1}{2}}, (d_t - 2h_t) \right] = 23.37 \text{ mm} \]

Effective ligament efficiency for bending
\[ \mu^* = \frac{P^* - d^*}{P^*} = 0.47498 \]
\[ A_0 = -0.00290 \]
\[ A_1 = 0.21260 \]
\[ A_2 = 3.99060 \]
\[ A_3 = -6.17300 \]
\[ A_4 = 3.43070 \]
\[ B_0 = 0.99660 \]
\[ B_1 = -4.19780 \]
\[ B_2 = 9.04780 \]
\[ B_3 = -7.99550 \]
\[ B_4 = 2.23980 \]

Effective Poisson ratio in perforated region
\[ \nu = B_0 + B_1 \mu^* + B_2 (\mu^*)^2 + B_3 (\mu^*)^3 + B_4 (\mu^*)^4 = 0.47640 \]

**Tubesheet bending stress**

Shell flange design bolt load for the operating condition
\[ W_{ds} = 22,832,134 \text{ N} \]

Tubesheet design bold load
\[ W^* = W_{ds} = 22,832,134 \text{ N} \]

Moment acting on the unperforated tubesheet rim
\[ M_{TS} = \frac{D_s^2}{16} \left( \rho^2 - 1 \right) \left( \rho^2 + 1 \right) P_t - \left( \rho - 1 \right) \left( \rho^2 + 1 \right) P_t \] = 32,288 N

Moment acting on the unperforated tubesheet rim
\[ M' = M_{TS} + \omega_x P_t + \frac{(C - G_x) W^*}{2\pi D_s} \] = 1,763,697 N

Maximum bending moments acting on the tubesheet at the periphery
\[ M_p = \frac{M^* - \frac{D_s}{32} \left( 3 + \nu^* \right) (P_t - P_1)}{64} \] = 339,431 N

Maximum bending moments acting on the tubesheet at the center
\[ M_p = \max \left[ M_p \left[ M_p \right] \right] = 668,705 N \]

Bending stress
\[ \sigma = \frac{6M}{\mu^* (h - h_C)^3} = 173.81 \text{ MPa} \]
\[ \sigma \leq 2S: \text{ Ok} \]

**Tubesheet shear stress**

\[ |P_s - P_t| \leq 3.2 \cdot S \cdot \mu \cdot h / D_o: \text{ Shear stress is not required to be calculated} \]

**Shell and channel stresses**

Axial membr. stress in the channel at tubesheet junction
\[ \sigma_{ch} = \frac{D_s^3 P_t}{4A_{ch} + t_{ch}} \] = 64.82 MPa

Bending stress in the channel at its junction to the tubesheet
\[ \sigma_{cb} = \frac{6k_s}{h_r} \left[ \beta \delta_p P_t \left( \frac{1 - \nu^*}{E^*} \right) \left( \frac{D_s}{h_r} \right) \left( 1 + \frac{h_r^2}{2} \right) \left( M_p + \frac{D_s^3}{32} (P_t - P_1) \right) \right] \] = -147.48 MPa

Channel axial stress
\[ \sigma_c = \sigma_{cb} + |\sigma_{cb}| \] = 212.30 MPa

\[ L_c \geq 1.8(D - t_c): \text{ Ok} \]

\[ \sigma_c \leq 1.5 S_c: \text{ Ok} \]
Loading case Design 4: $Pt = -0.10$ MPa, $Ps = -0.10$ MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear
$$\mu = \frac{p - d_t}{P} = 0.24998$$

Effective tube pitch
$$p^* = p \left(1 - \frac{4 \cdot \min \left[ A_{pa} \left( \frac{D_{tp}}{P} \right) \right]}{\pi D_{tp}} \right)^{1/3} = 42.33 \text{ mm} = 1.667 \text{ in}$$

Tube expansion depth ratio
$$\rho = \frac{h_t}{h} = 1.00000$$

Effective tube hole diameter
$$d' = \max \left\{ d_t - 2\delta \left( \frac{S_t}{S} \right) \rho, (d_t - 2\delta) \right\} = 23.37 \text{ mm} = 0.920 \text{ in}$$

Effective ligament efficiency for bending
$$\mu^* = \frac{p^* - d'}{p^*} = 0.44798$$

Equation coefficients:
- $A_0 = -0.00290$
- $A_1 = 0.21260$
- $A_2 = 3.99060$
- $A_3 = -6.17300$
- $A_4 = 3.43070$
- $B_0 = 0.99660$
- $B_1 = -4.19780$
- $B_2 = 9.04780$
- $B_3 = -7.99550$
- $B_4 = 2.23980$

Effective Poisson ratio in perforated region
$$\nu = \frac{B_0 + B_1 \rho + B_2 (\rho^2) + B_3 (\rho^3) + B_4 (\rho^4)}{1 - 2\nu} = 0.30032$$

**Tubesheet bending stress**

Shell flange design bolt load for the operating condition
$$W_{ds} = 165\,385 \text{ N} = 37\,179.93 \text{ lbf}$$

Tubesheet design bolt load
$$W^* = W_{ds} = 165\,385 \text{ N} = 37\,179.93 \text{ lbf}$$

Moment acting on the unperforated tubesheet rim
$$M_{TS} = \frac{D_t^2}{16} \left[ (\rho_s - 1) (\rho_s + 1) P_s - (\rho_c - 1) (\rho_c + 1) P_c \right] = -453 \text{ N} = -101.86 \text{ lbf}$$

Moment acting on the unperforated tubesheet rim
$$M' = M_{TS} + \omega \rho P_t + \frac{(C - G_s) W^*}{2 D_t} = -12\,079 \text{ N} = -2\,715.39 \text{ lbf}$$

Maximum bending moments acting on the tubesheet at the periphery
$$M_P = \frac{M' - \frac{D_t^2}{32}}{1 + \mu^2} = -3\,624 \text{ N} = -814.69 \text{ lbf}$$

Maximum bending moments acting on the tubesheet at the center
$$M_c = \frac{D_t^2 (3 + \nu^2) (P_s - P_c)}{64} = -3\,624 \text{ N} = -814.69 \text{ lbf}$$

Maximum bending moments acting on the tubesheet
$$M = \max \left\{ |M_t|, |M_c| \right\} = 3\,624 \text{ N} = 814.69 \text{ lbf}$$

Bending stress
$$\sigma = \frac{M}{\mu (h - h_c)^2} = 0.94 \text{ MPa} = 136.6 \text{ psi}$$

$$\sigma \leq 2S: \text{ Ok}$$

**Tubesheet shear stress**

$$|P_s - P_t| \leq 3.2 \cdot S \cdot \mu \cdot h / D_o: \text{ Shear stress is not required to be calculated}$$

**Shell and channel stresses**

Axial memb. stress in the channel at tubesheet junction
$$\sigma_{\phi,\text{in}} = \frac{D_t^2 P_s}{4 (A_{pa} + t_t)} = -0.36 \text{ MPa} = -52.5 \text{ psi}$$

Bending stress in the channel at its junction to the tubesheet
$$\sigma_{\phi} = \frac{D_t^2}{4} \left[ \beta \left( P_t - \frac{6 (1 - \nu^2)}{E_s} \right) \left( \frac{D_t}{R_t} \right) \left( 1 + \frac{h \beta}{2} \right) \left( \frac{4}{M_P} + \frac{D_t^2}{32} (P_s - P_c) \right) \right] = \frac{1.57 \text{ MPa} = 227.1 \text{ psi}}{279.6 \text{ psi}}$$

Channel axial stress
$$\sigma_c = |\sigma_{\phi,\text{in}}| + |\sigma_{\phi}| = 1.93 \text{ MPa} = 279.6 \text{ psi}$$

$$L_c \geq 1.8 \sqrt{D_c \cdot t_c}: \text{ Ok}$$

$$\sigma_c \leq 1.5 S_c: \text{ Ok}$$

---

Page 77 of 83

01/07/2016

NextGen Software by Sant'Ambrogio Servizi Industriali Srl - www.sant-ambrogio.it

2015.2
Minimum Design Metal Temperature (MDMT)

**U-Tube tubesheet - Tubesheet (Bolting)**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16 (Code Case 2655 - using Division 1 stress tables)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>114.30 mm 4.500 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-29.00 °C -20.20 °F</td>
</tr>
</tbody>
</table>

**U-Tube tubesheet - Tubesheet**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>85.00 mm 3.346 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>Yes</td>
</tr>
</tbody>
</table>

*Note: Nominal governing thickness for welded parts not subject to postweld heat treatment greater than 38 mm (1-1/2 in.)*

**Validation warnings:**

- Gasket overloaded: Ab > AbMax
**Tube bundle - Tubes bundle**

According to: Asme VIII Div. 2 Ed. 2015, 4.3 & 4.18 - Metric Units

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calculation temperature</td>
<td>T</td>
<td>454.00 °C</td>
</tr>
<tr>
<td>Material:</td>
<td>SB-517 Cold drawn/ann. N06600 - Wld. pipe</td>
<td></td>
</tr>
<tr>
<td>Allowable stress at room temperature</td>
<td>ST</td>
<td>137.00 MPa</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E</td>
<td>1.00</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c</td>
<td>0 mm</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>ce</td>
<td>0 mm</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>c'</td>
<td>0 mm</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>D</td>
<td>23.37 mm</td>
</tr>
<tr>
<td>Length</td>
<td>L</td>
<td>3 658.00 mm</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>t</td>
<td>4.19 mm</td>
</tr>
<tr>
<td>Allowable stress</td>
<td>S</td>
<td>137.00 MPa</td>
</tr>
<tr>
<td>Internal pressure</td>
<td>Pi</td>
<td>18.54 MPa</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>Ph</td>
<td>0 MPa</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>P = Pi + Ph</td>
<td>18.54 MPa</td>
</tr>
<tr>
<td>Required thickness</td>
<td>t_r = ( \frac{D + 2(c + c_e)}{2(M_c - 1) + c + c_e + c'} )</td>
<td>1.69 mm</td>
</tr>
<tr>
<td>Maximum allowable pressures (at the top of the vessel)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>New &amp; cold</td>
<td>=</td>
<td>41.99 MPa</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>=</td>
<td>41.99 MPa</td>
</tr>
<tr>
<td>External pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>External pressure</td>
<td>Pe</td>
<td>14.32 MPa</td>
</tr>
<tr>
<td>External static head</td>
<td>Ph</td>
<td>0 MPa</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>P = Pe + Ph</td>
<td>14.32 MPa</td>
</tr>
<tr>
<td>External design temperature</td>
<td>Te</td>
<td>420.00 °C</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>Do</td>
<td>31.75 mm</td>
</tr>
<tr>
<td>Max unsupported length</td>
<td>L</td>
<td>3 428.00 mm</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>Ey</td>
<td>190 800.00 MPa</td>
</tr>
<tr>
<td>Shell parameter</td>
<td>M_x = ( \frac{L}{\sqrt{R_c(t - c - c_e)}} )</td>
<td>420.26683</td>
</tr>
<tr>
<td></td>
<td>C_R = 0.55 ( \frac{L}{D_o} )</td>
<td>0.07260</td>
</tr>
<tr>
<td>Elastic buckling stress</td>
<td>F_{he} = 16 \cdot C_R \cdot E_y \cdot (t - c - c_e - c) \cdot D_o</td>
<td>2 925.50 MPa</td>
</tr>
<tr>
<td>Buckling stress</td>
<td>Foc = Sy</td>
<td>198.60 MPa</td>
</tr>
<tr>
<td>Factor A</td>
<td>A</td>
<td>0.01947</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>79.78 MPa</td>
</tr>
<tr>
<td>Tangent modulus of elasticity</td>
<td>Et = 2B / A</td>
<td>8 196.75 MPa</td>
</tr>
<tr>
<td>Yield strength at design temperature</td>
<td>Sy</td>
<td>198.60 MPa</td>
</tr>
<tr>
<td>Design factor</td>
<td>FS</td>
<td>1.66700</td>
</tr>
<tr>
<td>Hoop compressive membrane stress</td>
<td>F_{he} = ( \frac{F_{he}}{FS \cdot E_y} )</td>
<td>75.39 MPa</td>
</tr>
<tr>
<td>Allowable external pressure in the absence of other loads</td>
<td>P_o = 2F_{he} ( \frac{(t - c - c_e - c')}{D_o} )</td>
<td>19.90 MPa</td>
</tr>
<tr>
<td>Minimum required thickness</td>
<td>t_r = 3.13 mm</td>
<td>0.123 in</td>
</tr>
</tbody>
</table>

\( t \geq t_r: \text{Ok} \)
### TEMA RCB Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter</td>
<td>31.75 mm</td>
<td>1.250 in</td>
</tr>
<tr>
<td>Mean radius of bend</td>
<td>47.63 mm</td>
<td>1.875 in</td>
</tr>
<tr>
<td>Required tube wall thickness prior to bending</td>
<td>3.65 mm</td>
<td>0.144 in</td>
</tr>
<tr>
<td>Outside diameter of the tube</td>
<td>31.75 mm</td>
<td>1.250 in</td>
</tr>
<tr>
<td>Tube pitch</td>
<td>42.33 mm</td>
<td>1.667 in</td>
</tr>
<tr>
<td>Minimum tube pitch</td>
<td>39.69 mm</td>
<td>1.563 in</td>
</tr>
<tr>
<td>t ≥ t₀: Ok</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pitch ≥ Pitch(TEMA): Ok</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Maximum allowable external pressures

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>23.69 MPa</td>
<td>3436.3 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>19.90 MPa</td>
<td>2886.8 psi</td>
</tr>
</tbody>
</table>
## Minimum Design Metal Temperature (MDMT)

### Internal pressure

**Tube bundle - Tubes bundle**

<table>
<thead>
<tr>
<th>Material</th>
<th>SB-517 Cold drawn/ann. N06600</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness (tg)</td>
<td>4.19 mm, 0.165 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-104.00 °C, -155.20 °F</td>
</tr>
</tbody>
</table>

*Note: Component made from DN 100 (NPS 4) pipe or smaller or equivalent size of tubes of P-No. 1 materials*

### External pressure

**Tube bundle - Tubes bundle**

<table>
<thead>
<tr>
<th>Material</th>
<th>SB-517 Cold drawn/ann. N06600</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>None</td>
</tr>
<tr>
<td>Governing Thickness (tg)</td>
<td>4.19 mm, 0.165 in</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Minimum Design Metal Temperature (MDMT)</td>
<td>-104.00 °C, -155.20 °F</td>
</tr>
</tbody>
</table>

*Note: Component made from DN 100 (NPS 4) pipe or smaller or equivalent size of tubes of P-No. 1 materials*
Hemispherical head - Head

According to: Asme VIII Div. 2 Ed. 2015, 4.3.5 - Metric Units

Calculation temperature

\[ T = 420.00 \, ^\circ C \quad 788.00 \, ^\circ F \]

Material: SA-387 22 2 - Plate

Allowable stress at room temperature

\[ ST = 207.00 \, MPa \quad 30 \, 022.8 \, psi \]

Joint efficiency

\[ E = 1.00 \]

Corrosion allowance

\[ c = 3.00 \, mm \quad 0.118 \, in \]

External corrosion allowance

\[ ce = 0 \, mm \quad 0 \, in \]

Wall undertolerance

\[ c' = 0 \, mm \quad 0 \, in \]

Inside diameter

\[ D = 1306.00 \, mm \quad 51.417 \, in \]

Length

\[ L = 538.20 \, mm \quad 21.189 \, in \]

Adopted thickness

\[ t = 34.00 \, mm \quad 1.339 \, in \]

**Internal pressure**

Allowable stress

\[ S = 158.60 \, MPa \quad 23 \, 003.0 \, psi \]

Internal pressure

\[ Pi = 14.22 \, MPa \quad 2 \, 062.4 \, psi \]

Overpressure due to static head

\[ Ph = 0 \, MPa \quad 0 \, psi \]

Calculation pressure

\[ P = Pi + Ph = 14.22 \, MPa \quad 2 \, 062.4 \, psi \]

Required thickness

\[ t_r = \frac{D + 2(c + c_e)}{2} \left( \frac{1}{e_{fr}} - 1 \right) + c + c_e + c' = 33.08 \, mm \quad 1.302 \, in \]

\[ t \geq tr: \, Ok \]

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

New & cold

\[ = 21.01 \, MPa \quad 3 \, 047.8 \, psi \]

Hot & corroded

\[ = 14.65 \, MPa \quad 2 \, 124.3 \, psi \]

**External pressure**

External pressure

\[ Pe = 0.10 \, MPa \quad 14.9 \, psi \]

External static head

\[ Ph = 0 \, MPa \quad 0 \, psi \]

Calculation pressure

\[ P = Pe + Ph = 0.10 \, MPa \quad 14.9 \, psi \]

External design temperature

\[ Te = 20.00 \, ^\circ C \quad 68.00 \, ^\circ F \]

Outside radius

\[ Ro = D/2 + t = 687.00 \, mm \quad 27.047 \, in \]

Yield stress

\[ Sy = 310.00 \, MPa \quad 44 \, 961.7 \, psi \]

Modulus of elasticity

\[ Ey = 210 \, 350.00 \, MPa \quad 30 \, 508 \, 688.2 \, psi \]

Elastic buckling stress

\[ F_{le} = 0.075B \left( \frac{t - c - c_e - c'}{R_{o}} \right)^{1.3} \left( \frac{S_y}{115 + \frac{S_y}{B}} \right) = 711.88 \, MPa \quad 103 \, 249.9 \, psi \]

Buckling stress

\[ F_{le} = \left( \frac{115 + \frac{S_y}{B}}{S_y} \right) F_{he} = 256.14 \, MPa \quad 37 \, 149.9 \, psi \]

Design factor

\[ FS = 2.407 - 0.741 \left( \frac{F_{he}}{F_{le}} \right) = 1.79474 \]

Hoop compressive membrane stress

\[ F_{la} = \frac{F_{ha}}{FS} = 142.72 \, MPa \quad 20 \, 699.3 \, psi \]

Allowable external pressure in the absence of other loads

\[ P_e = 2F_{ha} \left( \frac{t - c - c_e - c'}{R_{o}} \right) = 12.88 \, MPa \quad 1 \, 868.1 \, psi \]

Minimum required thickness

\[ tr = 4.69 \, mm \quad 0.185 \, in \]

\[ t \geq tr: \, Ok \]

**Maximum allowable external pressures**

New & cold

\[ = 14.52 \, MPa \quad 2 \, 105.3 \, psi \]

Hot & corroded

\[ = 12.88 \, MPa \quad 1 \, 868.1 \, psi \]
Minimum Design Metal Temperature (MDMT)

### Internal pressure

**Hemispherical head - Head**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-387 22 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>(tg = 34.00 \text{mm} = 1.339 \text{in})</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Unadjusted MDMT from figure 3.7</td>
<td>40.90 °C = 105.62 °F</td>
</tr>
<tr>
<td>Coincident ratio Rts</td>
<td>0.97000</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness TR</td>
<td>1.40 °C = 34.52 °F</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. 3.12</td>
<td>39.50 °C = 103.10 °F</td>
</tr>
</tbody>
</table>

**External pressure**

**Hemispherical head - Head**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-387 22 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. 3.7 / 3.8</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>(tg = 34.00 \text{mm} = 1.339 \text{in})</td>
</tr>
<tr>
<td>PostWeld Heat Treatment</td>
<td>No</td>
</tr>
<tr>
<td>Unadjusted MDMT from figure 3.7</td>
<td>40.90 °C = 105.62 °F</td>
</tr>
<tr>
<td>Coincident ratio Rts</td>
<td>0.05500</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. 3.12</td>
<td>-104.00 °C = -155.20 °F</td>
</tr>
</tbody>
</table>

**Note:** Stress ratio Rts ≤ 0.24

**Validation warnings:**

- Product of nominal diameter and design pressure exceeds maximum allowed by TEMA RCB Code reference: TEMA RCB-1.11 [Required value: 17 500 000]
- Corrosion allowance for carbon steel or cast iron parts is smaller than minimum required by TEMA RCB: upon agreement between purchaser and fabricator, exceptions to TEMA requirements are acceptable as long as the exception is documented. Code reference: TEMA RCB-1.51 [Required value: 3.20 mm]