## Calculation report

**Asme VIII Div. 1 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 2 674.0 psi</td>
<td>14.22 MPa 2 062.4 psi</td>
</tr>
<tr>
<td>Internal design temperature</td>
<td>454.00 °C 849.20 °F</td>
<td>420.00 °C 788.00 °F</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>0 mm 0 in</td>
<td>3.00 mm 0.118 in</td>
</tr>
<tr>
<td>Vacuum?</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Minimum design temperature</td>
<td>-4.00 °C 24.80 °F</td>
<td></td>
</tr>
</tbody>
</table>
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### 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>
<th>Max test pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>21.38</td>
<td>18.45</td>
<td>1,158</td>
<td></td>
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<tr>
<td>Left channel</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>24.93</td>
<td>21.52</td>
<td>1,158</td>
<td></td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.01</td>
<td>21.61</td>
<td>18.66</td>
<td>1,158</td>
<td></td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot;</td>
<td>18.44</td>
<td>0</td>
<td>0.004</td>
<td>21.72</td>
<td>18.94</td>
<td>1,158</td>
<td></td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>18.44</td>
<td>0</td>
<td>0.03</td>
<td>21.76</td>
<td>18.79</td>
<td>1,158</td>
<td></td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot;</td>
<td>18.44</td>
<td>0</td>
<td>0.03</td>
<td>21.72</td>
<td>18.94</td>
<td>1,158</td>
<td></td>
</tr>
<tr>
<td>Tubesheet</td>
<td>18.44</td>
<td>0</td>
<td>0.02</td>
<td>26.08</td>
<td>22.24</td>
<td>1,158</td>
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<tr>
<td>Tubes bundle</td>
<td>18.54</td>
<td>0</td>
<td>0.02</td>
<td>54.31</td>
<td>54.31</td>
<td>1</td>
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</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) = 18.45 MPa (limited by Flat Head)**

**Tubes side MAP (New & Cold conditions) = 21.38 MPa (limited by Flat Head)**

**Tubes side Lowest Stress Ratio = 1.000**

**Tubes side test pressure = $Pt=1.3\cdot MAWP H&C (Item) \cdot St/S (Item) = 23.99$ MPa**

**Tubes side maximum test pressure = (limited by Tubes bundle)**

### 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>21.38</td>
<td>18.45</td>
<td>21.38</td>
<td>21.38</td>
</tr>
<tr>
<td>Left channel</td>
<td>24.93</td>
<td>21.52</td>
<td>15.37</td>
<td>15.37</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>21.61</td>
<td>18.66</td>
<td>19.91</td>
<td>19.91</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot;</td>
<td>21.72</td>
<td>18.94</td>
<td>18.94</td>
<td>18.94</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>21.76</td>
<td>18.79</td>
<td>19.91</td>
<td>19.91</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot;</td>
<td>21.72</td>
<td>18.94</td>
<td>18.94</td>
<td>18.94</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>26.08</td>
<td>22.24</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>54.31</td>
<td>54.31</td>
<td>27.98</td>
<td>20.93</td>
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</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>
<th>Max test pressure</th>
</tr>
</thead>
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<tr>
<td>Shell Flange</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>14.58</td>
<td>14.52</td>
<td>1,108</td>
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<tr>
<td>Main shell</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>16.47</td>
<td>14.25</td>
<td>1,108</td>
<td></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>1,108</td>
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</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>1,108</td>
<td></td>
</tr>
<tr>
<td>Tubesheet</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>16.90</td>
<td>14.36</td>
<td>1,158</td>
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</tr>
<tr>
<td>Tubes bundle</td>
<td>14.32</td>
<td>0</td>
<td>0.01</td>
<td>27.98</td>
<td>20.93</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Head</td>
<td>14.22</td>
<td>0</td>
<td>0.02</td>
<td>17.39</td>
<td>14.44</td>
<td>1,108</td>
<td></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) = 14.25 MPa (limited by Main shell)**

**Shell side MAP (New & Cold conditions) = 14.58 MPa (limited by Shell Flange)**

**Shell side Lowest Stress Ratio = 1.000**

**Shell side test pressure = Pt=1.3·MAWP H&C (Item)·St/S (Item) = 18.53 MPa**

**Shell side maximum test pressure = (limited by Head)**

### 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>14.58</td>
<td>14.52</td>
<td>14.52</td>
<td>14.52</td>
</tr>
<tr>
<td>Main shell</td>
<td>16.47</td>
<td>14.25</td>
<td>9.51</td>
<td>9.11</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>Out 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>16.90</td>
<td>14.36</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>27.98</td>
<td>20.93</td>
<td>7.60</td>
<td>7.02</td>
</tr>
<tr>
<td>Head</td>
<td>17.39</td>
<td>14.44</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

All pressures in MPa
### Weights

<table>
<thead>
<tr>
<th>Component</th>
<th>Dead</th>
<th>Live</th>
<th>Liquid</th>
<th>Full of water</th>
<th>Operating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>4,025 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>4,444 kg</td>
<td>4,025 kg</td>
</tr>
<tr>
<td>Left channel</td>
<td>4,622 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>6,428 kg</td>
<td>4,622 kg</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>1,244 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1,428 kg</td>
<td>1,244 kg</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot;</td>
<td>2,819 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>2,957 kg</td>
<td>2,819 kg</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>1,244 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1,428 kg</td>
<td>1,244 kg</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot;</td>
<td>2,819 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>2,957 kg</td>
<td>2,819 kg</td>
</tr>
<tr>
<td>Shell Flange</td>
<td>4,027 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>4,576 kg</td>
<td>4,027 kg</td>
</tr>
<tr>
<td>Main shell</td>
<td>8,985 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>11,763 kg</td>
<td>8,985 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>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>Tubesheet</td>
<td>5,016 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>5,016 kg</td>
<td>5,016 kg</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>15,284 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>17,082 kg</td>
<td>15,284 kg</td>
</tr>
<tr>
<td>Head</td>
<td>867 kg</td>
<td>0 kg</td>
<td>0 kg</td>
<td>1,458 kg</td>
<td>867 kg</td>
</tr>
</tbody>
</table>

**Totals:**

- 53,826 kg | 0 kg | 0 kg | 62,463 kg | 53,826 kg

**Total shell side volume:** 3.97127 m³

**Total tube side volume:** 4.66602 m³

**Total volume:** 8.63729 m³
<table>
<thead>
<tr>
<th>Component</th>
<th>Dimensions</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Head</td>
<td>Id = 1,238.00 mm, Od = 1,470.00 mm, Tk = 306.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Left channel</td>
<td>Id = 1,238.00 mm, Od = 1,470.00 mm, Tk = 116.00 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 = 787.40 mm, Tk = 76.20 mm, L = 581.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot; - Flange</td>
<td>Id = 635.00 mm, Od = 1,420.00 mm, Tk = 310.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot; - Gasket</td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
<td></td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot; - Bolts</td>
<td>20 x ANSI_TEMA 3-3/4&quot;</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>Id = 635.00 mm, Od = 787.40 mm, Tk = 76.20 mm, L = 581.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot; - Flange</td>
<td>Id = 635.00 mm, Od = 1,420.00 mm, Tk = 310.00 mm</td>
<td>SA-182 F22 3 - Forging</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot; - Gasket</td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
<td></td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot; - Bolts</td>
<td>20 x ANSI_TEMA 3-3/4&quot;</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>Shell Flange - Flange</td>
<td>Id = 1,275.00 mm, Od = 2,010.00 mm, Tk = 275.00 mm</td>
<td>SA-387 22 2 - Plate</td>
</tr>
<tr>
<td>Shell Flange - Gasket</td>
<td>Grooved Metal - Stainless steels and nickel-base alloys</td>
<td></td>
</tr>
<tr>
<td>Shell Flange - Bolts</td>
<td>24 x ANSI_TEMA 4-1/2&quot;</td>
<td>SA-193 B16 - Bolting</td>
</tr>
<tr>
<td>Main shell</td>
<td>Id = 1,275.00 mm, Od = 1,427.00 mm, Tk = 76.00 mm, L = 3,594.00 mm</td>
<td>SA-387 22 2 - Plate</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>Grooved Metal - Stainless steels and nickel-base alloys</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>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>Grooved Metal - Stainless steels and nickel-base alloys</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>Tubesheet - Flange</td>
<td>Od = 2,010.00 mm, Tk = 265.00 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-444 2 Solution ann. N06625 (high allowable stresses) - Pipe / tube</td>
</tr>
<tr>
<td>Head</td>
<td>Id = 1,312.00 mm, Od = 1,390.00 mm, Tk = 39.00 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>787.40 mm</td>
<td>76.20 mm</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>WN NotStandard</td>
<td>SA-182 F22 3</td>
<td>787.40 mm</td>
<td>76.20 mm</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>18&quot; LWN 1500 ANSI</td>
<td>SA-182 F22 3</td>
<td>597.00 mm</td>
<td>69.90 mm</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>18&quot; 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>600.00 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>In Shell Side</td>
<td>Main shell</td>
<td>Radial/ Reinforced</td>
<td>420.00 mm</td>
<td>0 °</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>
</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>In Shell 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>
</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>48.95 °C / 120.11 °F</td>
<td>No</td>
</tr>
<tr>
<td>Left channel</td>
<td>40.28 °C / 104.50 °F</td>
<td>No</td>
</tr>
<tr>
<td>Bocchello In Tube Side</td>
<td>25.78 °C / 78.40 °F</td>
<td>No</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot;</td>
<td>24.53 °C / 76.15 °F</td>
<td>No</td>
</tr>
<tr>
<td>Flange In Tube Side 26&quot; (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Bocchello Out Tube Side</td>
<td>25.78 °C / 78.40 °F</td>
<td>No</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot;</td>
<td>24.53 °C / 76.15 °F</td>
<td>No</td>
</tr>
<tr>
<td>Flange Out Tube Side 26&quot; (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Shell Flange</td>
<td>41.84 °C / 107.30 °F</td>
<td>No</td>
</tr>
<tr>
<td>Shell Flange (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Main shell</td>
<td>42.86 °C / 109.16 °F</td>
<td>No</td>
</tr>
<tr>
<td>In Shell Side</td>
<td>25.76 °C / 78.38 °F</td>
<td>No</td>
</tr>
<tr>
<td>In Shell Side (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Out Shell Side</td>
<td>25.76 °C / 78.38 °F</td>
<td>No</td>
</tr>
<tr>
<td>Out Shell Side (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Tubesheet</td>
<td>41.52 °C / 106.74 °F</td>
<td>No</td>
</tr>
<tr>
<td>Tubesheet (bolting)</td>
<td>-110.00 °C / -166.00 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Tubes bundle</td>
<td>-198.00 °C / -332.40 °F</td>
<td>Yes</td>
</tr>
<tr>
<td>Head</td>
<td>30.75 °C / 87.34 °F</td>
<td>No</td>
</tr>
</tbody>
</table>

**Item minimum design temperature Tmin:** -4.00 °C / 24.80 °F  
**Item MDMT:** 48.95 °C / 120.11 °F  
**One or more components have a MDMT higher than item minimum design temperature.**
**Welded flat cover - Flat Head**  
*According to: Asme VIII Div. 1 Ed. 2015, UG-34 - Metric Units*

### Design data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal design temperature</td>
<td>454.00 °C</td>
<td>849.20 °F</td>
</tr>
<tr>
<td>Internal design pressure</td>
<td>18.44 MPa</td>
<td>2674.0 psi</td>
</tr>
<tr>
<td>External design temperature</td>
<td>20.00 °C</td>
<td>68.00 °F</td>
</tr>
<tr>
<td>External design pressure</td>
<td>0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>1.00</td>
<td></td>
</tr>
</tbody>
</table>

### Material: SA-387 22 2 - Plate

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allowable stress</td>
<td>127.76 MPa</td>
<td>18530.0 psi</td>
</tr>
<tr>
<td>Allowable stress at room temperature</td>
<td>148.00 MPa</td>
<td>21465.6 psi</td>
</tr>
</tbody>
</table>

### Geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside diameter</td>
<td>1238.00 mm</td>
<td>48.740 in</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>1470.00 mm</td>
<td>57.874 in</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>306.00 mm</td>
<td>12.047 in</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Sketch</td>
<td>Sketch o</td>
<td></td>
</tr>
<tr>
<td>Factor C</td>
<td>0.30000</td>
<td></td>
</tr>
<tr>
<td>Outside diameter</td>
<td>1470.00 mm</td>
<td>57.874 in</td>
</tr>
</tbody>
</table>

### Internal pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
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</tr>
<tr>
<td>Internal pressure</td>
<td>18.44 MPa</td>
<td>2674.0 psi</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>18.44 MPa</td>
<td>2674.0 psi</td>
</tr>
<tr>
<td>Required thickness</td>
<td>305.86 mm</td>
<td>12.042 in</td>
</tr>
<tr>
<td>Item service</td>
<td>NotSpecified</td>
<td></td>
</tr>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>1.50 mm</td>
<td>0.059 in</td>
</tr>
</tbody>
</table>

\[
t \geq t_{r \text{ UG-16(b)}}: \text{ Ok}
\]

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

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>21.38 MPa</td>
<td>3100.5 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>18.45 MPa</td>
<td>2676.5 psi</td>
</tr>
</tbody>
</table>

### External pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>External design temperature</td>
<td>20.00 °C</td>
<td>68.00 °F</td>
</tr>
<tr>
<td>External design pressure</td>
<td>0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>Required thickness</td>
<td>21.25 mm</td>
<td>0.837 in</td>
</tr>
</tbody>
</table>

\[
t \geq t_{r}: \text{ Ok}
\]

### Maximum allowable external pressures

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
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<td>3100.5 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>21.38 MPa</td>
<td>3100.5 psi</td>
</tr>
</tbody>
</table>
Minimum Design Metal Temperature (MDMT)

**Welded flat cover - Flat Head**

Material  =  SA-387 22 2
Curve of fig. UCS-66  =  A
Governing Thickness  =  116.00 mm 4.567 in
Reduction in MDMT based on available excess thickness  =  0.05 °C 32.09 °F
Unadjusted MDMT from table UCS-66  =  49.00 °C 120.20 °F
Design pressure  =  18.44 MPa 2674.0 psi
Maximum allowable working pressure  =  18.45 MPa 2676.5 psi
Coincident ratio  =  0.99907
Adjusted MDMT from fig. UCS-66  =  48.95 °C 120.11 °F

**Welded flat cover - Flat Head**

Material  =  SA-387 22 2
Curve of fig. UCS-66  =  A
Governing Thickness  =  116.00 mm 4.567 in
Reduction in MDMT based on available excess thickness  =  8.72 °C 47.70 °F
Unadjusted MDMT from table UCS-66  =  49.00 °C 120.20 °F
Joint efficiency  =  1.00
Corrected joint efficiency  =  1.00000
Corrosion allowance  =  0 mm 0 in
Minimum required thickness in corroded condition  =  97.79 mm 3.850 in
Nominal noncorroded thickness  =  116.00 mm 4.567 in
Coincident ratio  =  0.84304
Adjusted MDMT from fig. UCS-66  =  40.28 °C 104.50 °F

**External pressure**

**Welded flat cover - Flat Head**

Material  =  SA-387 22 2
Curve of fig. UCS-66  =  A
Governing Thickness  =  116.00 mm 4.567 in
Reduction in MDMT based on available excess thickness  =  80.00 °C 176.00 °F
Unadjusted MDMT from table UCS-66  =  49.00 °C 120.20 °F
Design pressure  =  0.10 MPa 14.9 psi
Maximum allowable working pressure  =  18.45 MPa 2676.5 psi
Coincident ratio  =  0.00558
Adjusted MDMT from fig. UCS-66  =  -31.00 °C -23.80 °F
**Welded flat cover - Flat Head**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-387 22 2</td>
</tr>
<tr>
<td>Curve of fig. UCS-66</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 49.00 °C</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E' = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 4.77 mm</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t_E = t_n - \sigma = 0.04112$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= -31.00 °C</td>
</tr>
<tr>
<td>Design data</td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Internal design temperature</td>
<td>$T = 454.00 \degree C = 849.20 \degree F$</td>
</tr>
<tr>
<td>Internal design pressure</td>
<td>$P = 18.44 \text{ MPa} = 2674.0 \text{ psi}$</td>
</tr>
<tr>
<td>External design temperature</td>
<td>$T_e = 20.00 \degree C = 68.00 \degree F$</td>
</tr>
<tr>
<td>External design pressure</td>
<td>$P_e = 0.10 \text{ MPa} = 14.9 \text{ psi}$</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>$E = 1.00$</td>
</tr>
</tbody>
</table>

| Material: SA-387 22 2 - Plate |
|---|---|
| Allowable stress | $S = 127.76 \text{ MPa} = 18530.0 \text{ psi}$ |
| Allowable stress at room temperature | $S_T = 148.00 \text{ MPa} = 21465.6 \text{ psi}$ |

| Geometry |
|---|---|
| Inside diameter | $D = 1238.00 \text{ mm} = 48.740 \text{ in}$ |
| Outside diameter | $D_o = 1470.00 \text{ mm} = 57.874 \text{ in}$ |
| Length | $L = 1500.00 \text{ mm} = 59.055 \text{ in}$ |
| Adopted thickness | $t = 116.00 \text{ mm} = 4.567 \text{ in}$ |
| Corrosion allowance | $c = 0 \text{ mm} = 0 \text{ in}$ |
| External corrosion allowance | $c_e = 0 \text{ mm} = 0 \text{ in}$ |
| Wall undertolerance | $c' = 0 \text{ mm} = 0 \text{ in}$ |
| Forming strain (cylinders formed from plate) | $\varepsilon_f = \frac{50 \cdot t}{(R + t/2)} = 8.57\%$ |

| Ligament Efficiency |
|---|---|
| Reference figure | None |
| Diameter of tube holes | $d = 0 \text{ mm} = 0 \text{ in}$ |

| Internal pressure |
|---|---|
| Allowable stress | $S = 127.76 \text{ MPa} = 18530.0 \text{ psi}$ |
| Internal pressure | $P_i = 18.44 \text{ MPa} = 2674.0 \text{ psi}$ |
| Overpressure due to static head | $P_h = 0 \text{ MPa} = 0 \text{ psi}$ |
| Calculation pressure | $P = P_i + P_h = 18.44 \text{ MPa} = 2674.0 \text{ psi}$ |
| Reference diameter | Inside |
| Calculation radius (inside) | $R = 619.00 \text{ mm} = 24.370 \text{ in}$ |
| Required thickness for circumferential stress, UG-27(c)(1) | $t_r = \frac{P(R + c + c')}{2SE + 0.4R} + c + c_e + c' = 97.79 \text{ mm} = 3.850 \text{ in}$ |
| Required thickness for longitudinal stress, UG-27(c)(2) | $t_l = \frac{P(R + c + c')}{2SE + 0.4R} + c + c_e + c' = 43.41 \text{ mm} = 1.709 \text{ in}$ |
| Minimum required thickness | $t_r = \max\{t_r(circ),t_l(long)\} = 97.79 \text{ mm} = 3.850 \text{ in}$ |
| Item service | Service |
| Minimum required thickness as per UG-16(b), including corrosion | $t_r \text{ UG-16(b)} = 1.50 \text{ mm} = 0.059 \text{ in}$ |
| $t \geq t_r: \text{ Ok}$ |
| $t \geq t_r \text{ UG-16(b)}: \text{ Ok}$ |

| Maximum allowable pressures (at the top of the vessel) |
|---|---|
| New & cold | $= 24.93 \text{ MPa} = 3616.0 \text{ psi}$ |
| Hot & corroded | $= 21.52 \text{ MPa} = 3121.5 \text{ psi}$ |
External pressure

External design temperature
\[ T_e = 20.00 \, ^\circ{C} \quad 68.00 \, ^\circ{F} \]

External pressure
\[ P_e = 0.10 \, MPa \quad 14.9 \, psi \]

External static head
\[ P_h = 0 \, MPa \quad 0 \, psi \]

Calculation pressure
\[ P = P_e + P_h = 0.10 \, MPa \quad 14.9 \, psi \]

Outside diameter
\[ D_o = 1470.00 \, mm \quad 57.874 \, in \]

Modulus of elasticity
\[ E = 200,000.00 \, MPa \quad 29,007,547.6 \, psi \]

Axial length between reinforcements
\[ L = 1500.00 \, mm \quad 59.055 \, in \]

L / Do ratio
\[ = 1.02041 \]

Do / t ratio
\[ = 12.67241 \]

Factor A
\[ = 0.03111 \]

Factor B
\[ = 146.07 \, MPa \quad 21,185.3 \, psi \]

External pressure
\[ P_a = \frac{4B}{3[D_o/(t-e-c_e-e)]} = 15.37 \, MPa \quad 2229.0 \, psi \]

Required thickness
\[ t_r = 4.77 \, mm \quad 0.188 \, in \]

Maximum allowable external pressures

New & cold
\[ = 15.37 \, MPa \quad 2229.0 \, psi \]

Hot & corroded
\[ = 15.37 \, MPa \quad 2229.0 \, psi \]

Customer
Riccardo Petrelli

Drawing
U_150

Revision

---

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**

**Cylindrical shell - Left channel**

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-387 22 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. UCS-66</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 8.72 °C 47.70 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 49.00 °C 120.20 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm 0 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 97.79 mm 3.850 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t,E^* \times \frac{t,E^*}{t_n - c} = 0.84304$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 40.28 °C 104.50 °F</td>
</tr>
</tbody>
</table>

**External pressure**

**Cylindrical shell - Left channel**

<table>
<thead>
<tr>
<th>Material</th>
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<tr>
<td>Curve of fig. UCS-66</td>
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<tr>
<td>Governing Thickness</td>
<td>116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 49.00 °C 120.20 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
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<td>$E^* = \max(E, 0.8) = 1.00000$</td>
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<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm 0 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 4.77 mm 0.188 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t,E^* \times \frac{t,E^*}{t_n - c} = 0.04112$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= -31.00 °C -23.80 °F</td>
</tr>
</tbody>
</table>

**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]
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
\[ E = 1.00 \]

Nozzle material: SA-182 F22 3 - Forgings
Allowable stress in nozzle
\[ S_n = 127.76 \, \text{MPa} = 18,530.0 \, \text{psi} \]

Shell material: SA-387 22 2 - Plate
Allowable stress in vessel
\[ S_v = 127.76 \, \text{MPa} = 18,530.0 \, \text{psi} \]

Nozzle geometry

Outside diameter
\[ d_o = 787.40 \, \text{mm} = 31.000 \, \text{in} \]

Inside diameter
\[ d_i = 635.00 \, \text{mm} = 25.000 \, \text{in} \]

Nozzle thickness
\[ t_n = 76.20 \, \text{mm} = 3.000 \, \text{in} \]

Nozzle reinforcement thickness
\[ t_x = 165.00 \, \text{mm} = 6.496 \, \text{in} \]

Length of projection defining the thickened portion of integral reinforcement
\[ L = 172.00 \, \text{mm} = 6.772 \, \text{in} \]

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

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

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

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

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

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

Offset from shell border
\[ = 600.00 \, \text{mm} = 23.622 \, \text{in} \]

Angular offset
\[ = 0 ^\circ \]

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

Minimum weld leg length of the outside nozzle fillet weld
\[ = 8.49 \, \text{mm} = 0.334 \, \text{in} \]

\[ r \geq \text{Min}[1/4t, 1/8 \text{in.} \text{(3mm)}]: \text{Ok} \]

Opening geometry

Finished diameter of circular opening
\[ d = 635.00 \, \text{mm} = 25.000 \, \text{in} \]

Finished radius of circular opening
\[ R_n = 317.50 \, \text{mm} = 12.500 \, \text{in} \]

Nearest opening
\[ = \text{Bocchello Out Tube Side} \]

Distance to nearest opening
\[ = 2311.23 \, \text{mm} = 90.994 \, \text{in} \]

Maximum distance before multiple openings calculation occurs
\[ = 1333.15 \, \text{mm} = 52.486 \, \text{in} \]

Shell Geometry

Inside shell diameter
\[ D = 1238.00 \, \text{mm} = 48.740 \, \text{in} \]

Inside radius of shell course under consideration
\[ R = 619.00 \, \text{mm} = 24.370 \, \text{in} \]

Shell thickness
\[ t = 116.00 \, \text{mm} = 4.567 \, \text{in} \]

Shell corrosion allowance
\[ c_s = 0 \, \text{mm} = 0 \, \text{in} \]

Shell undertolerance
\[ c's = 0 \, \text{mm} = 0 \, \text{in} \]
**Internal pressure**

**Net thicknesses**
- Net shell thickness: \( t' = t - c_s - c_s' = 116.00 \text{ mm} = 4.567 \text{ in} \)
- Net required thickness of a seamless shell or formed head: \( t_r = 97.79 \text{ mm} = 3.850 \text{ in} \)
- Net nozzle thickness: \( t_n' = t_n - c_n - c_n' = 76.20 \text{ mm} = 3.000 \text{ in} \)
- Net required thickness of a seamless nozzle wall: \( t_rn = 50.16 \text{ mm} = 1.975 \text{ in} \)
- Nozzle reinforcement thickness: \( t_x' = t_x - c_n = 165.00 \text{ mm} = 6.496 \text{ in} \)

**Limit of reinforcement parallel to the vessel wall**: \( L_R = \max(d, R_n + t_n + t) = 635.00 \text{ mm} = 25.000 \text{ in} \)

**Useful limit of reinforcement in vessel wall**: \( L_o = \text{distance from border} = 117.50 \text{ mm} = 4.626 \text{ in} \)

**Useful limit of reinforcement normal to the vessel wall**: \( h_o = \min(2.5 \cdot t, 2.5 \cdot t_n) = 290.00 \text{ mm} = 11.417 \text{ in} \)

**Internal pressure**
- Strength reduction factor: \( f_r = 1.00000 \)
- fr1 factor: \( f_{r1} = \max(S_n/S_v; 1) = 1.00000 \)
- fr2 factor: \( f_{r2} = \max(S_n/S_v; 1) = 1.00000 \)
- Lenght of projection defining the thickened portion of integral reinforcement: \( L = 172.00 \text{ mm} = 6.772 \text{ in} \)
- Correction factor: \( F = 1.00000 \)
- Internal pressure: \( P = 18.44 \text{ MPa} = 2674.0 \text{ psi} \)
- Static head internal: \( Ph = 0 \text{ MPa} = 0 \text{ psi} \)
- Area available in shell: \( A_1 = \frac{d(E_p F_t) - 2t_x'(E_p F_t' - t_x') (1 - f_{r1})}{2(t + t_x') (E_p F_t' - t_x') - 2t_x'(E_p F_t' - t_x') (1 - f_{r1})} = 10287.3 \text{ mm}^2 = 15945 \text{ in}^2 \)
- Area available in outward nozzle: \( A_2 = \frac{(r^2 - \pi r^2/4) f_{r1}}{A_3} = 21.5 \text{ mm}^2 = 0.333 \text{ in}^2 \)
- Area available in outward weld: \( A_41 = (\text{leg})^2 \cdot f_{r2} = 225.0 \text{ mm}^2 = 0.349 \text{ in}^2 \)
- Area available in inward nozzle: \( A_3 = 0 \text{ mm}^2 = 0 \text{ in}^2 \)
- Area available in inward weld: \( A_43 = (\text{leg})^2 \cdot f_{r2} = 0 \text{ mm}^2 = 0 \text{ in}^2 \)
- Area required: \( A = d - t_r F + 2t_x' \cdot F_r (1 - f_{r1}) = 62098.2 \text{ mm}^2 = 96252 \text{ in}^2 \)
- Total available area: \( A_t = A_1 + A_2 + A_3 + A_41 + A_43 = 64005.1 \text{ mm}^2 = 99208 \text{ in}^2 \)

**Appendix 1-7(a)**

Area available in shell, within Appendix 1-7 limits: \( A_1 \text{ (1-7(a))} = 6999.0 \text{ mm}^2 = 10848 \text{ in}^2 \)

Area available in outward nozzle: \( A_2 = 53492.9 \text{ mm}^2 = 82914 \text{ in}^2 \)

Area available in inward nozzle: \( A_3 = 0 \text{ mm}^2 = 0 \text{ in}^2 \)

Area available in outward weld: \( A_41 = (\text{leg})^2 \cdot f_{r2} = 225.0 \text{ mm}^2 = 0.349 \text{ in}^2 \)

Area available in inward weld: \( A_43 = (\text{leg})^2 \cdot f_{r2} = 0 \text{ mm}^2 = 0 \text{ in}^2 \)

Total area available within Appendix 1-7 limits: \( A_t (1-7) = 60716.8 \text{ mm}^2 = 94111 \text{ in}^2 \)

Area required as per Appendix 1-7(a): \( A_t (1/3) = 41398.8 \text{ mm}^2 = 64168 \text{ in}^2 \)

**Opening maximum allowable pressure**
- Opening maximum allowable pressure: \( P_{max} = 18.66 \text{ MPa} = 2705.9 \text{ psi} \)
- Total pressure: \( P_t = 18.44 \text{ MPa} = 2674.0 \text{ psi} \)

\( P_t \leq P_{max}: \text{ Ok} \)
Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion:
\[ t(UG-16) = 1.50 \text{ mm} \quad 0.059 \text{ in} \]

Minimum neck thickness required for internal and external pressure using UG-27 and UG-28:
\[ t_a = 50.16 \text{ mm} \quad 1.975 \text{ in} \]

Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance:
\[ t_{b1} = t_{(P \text{ int})} - c' = 97.79 \text{ mm} \quad 3.850 \text{ in} \]

\( t_{b1} \) no less than minimum thickness specified in UG-16(b):
\[ t_{b1^*} = \max[t_{b1}, t(UG-16)] = 97.79 \text{ mm} \quad 3.850 \text{ in} \]

Min. thickness required using external pressure as internal (with E=1) for the shell or head:
\[ t_{b2} = 0.50 \text{ mm} \quad 0.020 \text{ in} \]

\( t_{b2} \) no less than minimum thickness specified in UG-16(b):
\[ t_{b2^*} = \max[t_{b2}, t(UG-16)] = 1.50 \text{ mm} \quad 0.059 \text{ in} \]

Minimum thickness of a standard wall pipe:
\[ t_b = \min[t_b, \max(t_{b1^*}, t_{b2^*})] = 8.34 \text{ mm} \quad 0.328 \text{ in} \]

Minimum required nozzle neck thickness:
\[ t(UG-45) = \max(t_a, t_b) = 50.16 \text{ mm} \quad 1.975 \text{ in} \]

Nozzle thickness:
\[ t_n = 76.20 \text{ mm} \quad 3.000 \text{ in} \]

\( t_n \geq t(UG-45) \): Ok
**External pressure**

**Net thicknesses**
- **Net shell thickness**
  \[ t' = t - c_s - c_s' = 116.00 \text{ mm} = 4.567 \text{ in} \]
- **Net required thickness of a seamless shell or formed head**
  \[ t_{rr} = 4.77 \text{ mm} = 0.188 \text{ in} \]
- **Net nozzle thickness**
  \[ t_{tn} = 1.77 \text{ mm} = 0.070 \text{ in} \]
- **Net required thickness of a seamless nozzle wall**
  \[ t_{tn} = 76.20 \text{ mm} = 3.000 \text{ in} \]
- **Net required thickness of a formed nozzle wall**
  \[ t_{tn} = 1.77 \text{ mm} = 0.070 \text{ in} \]
- **Nozzle reinforcement thickness**
  \[ t_x = 165.00 \text{ mm} = 6.496 \text{ in} \]

**Useful limit of reinforcement parallel to the vessel wall**
\[ L_o = (\text{distance from border}) = 117.50 \text{ mm} = 4.626 \text{ in} \]

**Useful limit of reinforcement normal to the vessel wall**
\[ h_o = \min(2.5 \cdot t, 2.5 \cdot t_{tn}) = 290.00 \text{ mm} = 11.417 \text{ in} \]

**Strength reduction factor**
\[ f_r = 1.00000 \]

**fr1 factor**
\[ f_{r1} = \max(\frac{S_n}{S_v}; 1) = 1.00000 \]

**fr2 factor**
\[ f_{r2} = \max(\frac{S_n}{S_v}; 1) = 1.00000 \]

**Useful limit of reinforcement in vessel wall**
\[ L_o = (\text{distance from border}) = 117.50 \text{ mm} = 4.626 \text{ in} \]

**Limit of reinforcement parallel to the vessel wall**
\[ L_{fr} = \max(d, R_e + t_{tn} + t) = 635.00 \text{ mm} = 25.000 \text{ in} \]

**Static head external**
\[ P_h = 0 \text{ MPa} = 0 \text{ psi} \]

**Area required**
\[ A = \left[ d \cdot t_{rr} \cdot f + 2 \cdot t_x \cdot t_{rr} \cdot f \left( 1 - f_{r1} \right) \right] / 2 = 1514.5 \text{ mm}^2 = 2.347 \text{ in}^2 \]

**Appendix 1-7(a)**

**Area available in shell, within Appendix 1-7 limits**
\[ A_{1-7} = 42756.8 \text{ mm}^2 = 66.273 \text{ in}^2 \]

**Area available in outward nozzle**
\[ A_1 = (\text{leg})^2 \cdot f_r = 225.0 \text{ mm}^2 = 0.349 \text{ in}^2 \]

**Area required**
\[ A = [\text{d-tr} + 2 \cdot t_x \cdot t_{(1-fr1)}] / 2 = 1514.5 \text{ mm}^2 = 2.347 \text{ in}^2 \]

**Total available area**
\[ A_t = A_{1-7} + A_{1-7(a)} + A_3 + A_4 + A_4 = 144629.1 \text{ mm}^2 = 224.175 \text{ in}^2 \]

**Opening maximum allowable pressure**
\[ P_{max} = 17.98 \text{ MPa} = 2607.4 \text{ psi} \]
\[ P_T = 0.10 \text{ MPa} = 14.9 \text{ psi} \]

\[ P_T \leq P_{max}: \text{ Ok} \]
### Nozzle neck thickness (according to UG-45)

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Minimum Thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>t(UG-16) = 1.50 mm, 0.059 in</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27 and UG-28</td>
<td>ta = 1.77 mm, 0.070 in</td>
</tr>
<tr>
<td>Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance</td>
<td>tb1 = tr(P,int) = c' = 97.79 mm, 3.850 in</td>
</tr>
<tr>
<td>tb1 no less than minimum thickness specified in UG-16(b)</td>
<td>tb1* = max[tb1,t(UG-16)] = 97.79 mm, 3.850 in</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with E=1) for the shell or head</td>
<td>tb2 = 0.50 mm, 0.020 in</td>
</tr>
<tr>
<td>tb2 no less than minimum thickness specified in UG-16(b)</td>
<td>tb2* = max[tb2,t(UG-16)] = 1.50 mm, 0.059 in</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>tb3 = 8.34 mm, 0.328 in</td>
</tr>
<tr>
<td>tb</td>
<td>tb = min[tb3,max(tb1*,tb2*)] = 8.34 mm, 0.328 in</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>t(UG-45) = max(ta,tb) = 8.34 mm, 0.328 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>tn = 76.20 mm, 3.000 in</td>
</tr>
</tbody>
</table>

\[ t_{n} \geq t(UG-45): \text{Ok} \]

### Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]
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
\[ Te = 20.00 \, ^\circ C = 68.00 \, ^\circ F \]

External design pressure
\[ Pe = 0.10 \, MPa = 14.9 \, psi \]

Joint efficiency
\[ E = 1.00 \]

**Nozzle material:** SA-182 F22 3 - Forgings

Allowable stress in nozzle
\[ S_{n} = 127.76 \, MPa = 18530.0 \, psi \]

**Shell material:** SA-387 22 2 - Plate

Allowable stress in vessel
\[ S_{v} = 127.76 \, MPa = 18530.0 \, psi \]

**Nozzle geometry**

Outside diameter
\[ do = 787.40 \, mm = 31.000 \, in \]

Inside diameter
\[ di = 635.00 \, mm = 25.000 \, in \]

Nozzle thickness
\[ tn = 76.20 \, mm = 3.000 \, in \]

Nozzle reinforcement thickness
\[ tx = 165.00 \, mm = 6.496 \, in \]

Length of projection defining the thickened portion of integral reinforcement
\[ L = 172.00 \, mm = 6.772 \, in \]

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

Corner radius
\[ r = 10.00 \, mm = 0.394 \, in \]

Nozzle corrosion allowance
\[ cn = 0 \, mm = 0 \, in \]

Nozzle undertolerance
\[ c'n = 0 \, mm = 0 \, in \]

Nozzle connection = Integrially reinforced

Nozzle position = Radial

Offset from shell border
\[ = 700.00 \, mm = 27.559 \, in \]

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

Weld leg length of the outside nozzle fillet weld
\[ two = 15.00 \, mm = 0.591 \, in \]

Minimum weld leg length of the outside nozzle fillet weld
\[ = 8.49 \, mm = 0.334 \, in \]

\[ r \geq \text{Min}[1/4t, 1/8 \text{ in. (3mm)}]: \) Ok

**Opening geometry**

Finished diameter of circular opening
\[ d = 635.00 \, mm = 25.000 \, in \]

Finished radius of circular opening
\[ Rn = 317.50 \, mm = 12.500 \, in \]

Nearest opening
\[ = \text{Bocchello In Tube Side} \]

Distance to nearest opening
\[ = 2311.23 \, mm = 90.994 \, in \]

Maximum distance before multiple openings calculation occurs
\[ = 1333.15 \, mm = 52.486 \, in \]

**Shell Geometry**

Inside shell diameter
\[ D = 1238.00 \, mm = 48.740 \, in \]

Inside radius of shell course under consideration
\[ R = 619.00 \, mm = 24.370 \, in \]

Shell thickness
\[ t = 116.00 \, mm = 4.567 \, in \]

Shell corrosion allowance
\[ cs = 0 \, mm = 0 \, in \]

Shell undertolerance
\[ c's = 0 \, mm = 0 \, in \]
### Internal pressure

#### Net thicknesses

- **Net shell thickness**
  \[ t' = t - c_s - c's = 116.00 \text{ mm} \quad 4.567 \text{ in} \]
- **Net required thickness of a seamless shell or formed head**
  \[ t_r = 97.79 \text{ mm} \quad 3.850 \text{ in} \]
- **Net nozzle thickness**
  \[ t'n = t'n - c'n = 76.20 \text{ mm} \quad 3.000 \text{ in} \]
- **Net required thickness of a seamless nozzle wall**
  \[ t'n = 76.20 \text{ mm} \quad 3.000 \text{ in} \]
- **Nozzle reinforcement thickness**
  \[ t'x = t'x - c'n = 165.00 \text{ mm} \quad 6.496 \text{ in} \]

#### Internal pressure

- **Strength reduction factor**
  \[ f_r = 1.00000 \]
- **fr1 factor**
  \[ f_{r1} = \max(Sn/Sv; 1) = 1.00000 \]
- **fr2 factor**
  \[ f_{r2} = \max(Sn/Sv; 1) = 1.00000 \]

#### Area calculations

- **Area available in shell**
  \[ A_1 = \max \left\{ \frac{d(E_F' - F_{rt}) - 2t_r'(E_F' - F_{rt}) (1 - f_{r})}{2(t + t'_n) (E_F' - F_{rt}) - 2t_r'(E_F' - F_{rt}) (1 - f_{r})} \right\} = 11,561.8 \text{ mm}^2 \quad 17.921 \text{ in}^2 \]
- **Corner area to be subtracted from nozzle area**
  \[ A_2 = \left( \frac{r^2}{2} \right) f_{r} A_1 = 21.5 \text{ mm}^2 \quad 0.033 \text{ in}^2 \]

#### Appendix 1-7(a)

- **Area available in shell, within Appendix 1-7 limits**
  \[ A_1 = 6,999.0 \text{ mm}^2 \quad 10.848 \text{ in}^2 \]
- **Area available in outward nozzle**
  \[ A_2 = 53,429.9 \text{ mm}^2 \quad 82.914 \text{ in}^2 \]
- **Area available in inward nozzle**
  \[ A_3 = 0 \text{ mm}^2 \quad 0 \text{ in}^2 \]
- **Area available in outward weld**
  \[ A_{41} = (\text{leg})^2 \cdot f_{r2} = 225.0 \text{ mm}^2 \quad 0.349 \text{ in}^2 \]
- **Area available in inward weld**
  \[ A_{43} = (\text{leg})^2 \cdot f_{r2} = 0 \text{ mm}^2 \quad 0 \text{ in}^2 \]
- **Area required**
  \[ A = d \cdot t_r + 2t'_x \cdot t_r \cdot f_{r1} = 62,098.2 \text{ mm}^2 \quad 96.252 \text{ in}^2 \]

#### Opening maximum allowable pressure

- **Opening maximum allowable pressure**
  \[ P_{max} = 18.79 \text{ MPa} \quad 2724.7 \text{ psi} \]
- **Total pressure**
  \[ P_t = 18.44 \text{ MPa} \quad 2674.0 \text{ psi} \]
**Nozzle neck thickness (according to UG-45)**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>t(UG-16)</td>
<td>1.50 mm</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27</td>
<td>ta</td>
<td>50.16 mm</td>
</tr>
<tr>
<td>Minimum thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance</td>
<td>tb1 = tr(P,int) - c'</td>
<td>97.79 mm</td>
</tr>
<tr>
<td>tb1 no less than minimum thickness specified in UG-16(b)</td>
<td>tb1* = max[tb1, t(UG-16)]</td>
<td>97.79 mm</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with E=1) for the shell or head</td>
<td>tb2</td>
<td>0.50 mm</td>
</tr>
<tr>
<td>tb2 no less than minimum thickness specified in UG-16(b)</td>
<td>tb2* = max[tb2, t(UG-16)]</td>
<td>1.50 mm</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>tb3</td>
<td>8.34 mm</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>t(UG-45) = max(ta, tb)</td>
<td>50.16 mm</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>tn</td>
<td>76.20 mm</td>
</tr>
</tbody>
</table>

\[ tn \geq t(UG-45) \]: Ok
## Net thicknesses

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Net shell thickness</td>
<td>116.00</td>
<td>mm</td>
</tr>
<tr>
<td>Net required thickness of a seamless shell or formed head</td>
<td>4.77</td>
<td>mm</td>
</tr>
<tr>
<td>Net nozzle thickness</td>
<td>76.20</td>
<td>mm</td>
</tr>
<tr>
<td>Net required thickness of a seamless nozzle wall</td>
<td>1.77</td>
<td>mm</td>
</tr>
<tr>
<td>Nozzle reinforcement thickness</td>
<td>165.00</td>
<td>mm</td>
</tr>
<tr>
<td>Limit of reinforcement parallel to the vessel wall</td>
<td>635.00</td>
<td>mm</td>
</tr>
<tr>
<td>Useless limit of reinforcement in vessel wall</td>
<td>152.50</td>
<td>mm</td>
</tr>
<tr>
<td>Useless limit of reinforcement normal to the vessel wall</td>
<td>290.00</td>
<td>mm</td>
</tr>
</tbody>
</table>

## External pressure

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>External pressure</td>
<td>0.10</td>
<td>MPa, psi</td>
</tr>
<tr>
<td>Static head external</td>
<td>0</td>
<td>MPa, psi</td>
</tr>
<tr>
<td>Length of projection defining the thickened portion of integral reinforcement</td>
<td>172.00</td>
<td>mm</td>
</tr>
<tr>
<td>Correction factor</td>
<td>1.00000</td>
<td></td>
</tr>
<tr>
<td>External pressure</td>
<td>0.10</td>
<td>MPa, psi</td>
</tr>
</tbody>
</table>

## Appendix 1-7(a)

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area available in shell, within Appendix 1-7 limits</td>
<td>42 756.8</td>
<td>mm²</td>
</tr>
<tr>
<td>Area available in outward nozzle</td>
<td>225.0</td>
<td>mm²</td>
</tr>
<tr>
<td>Area required</td>
<td>1 514.5</td>
<td>mm²</td>
</tr>
<tr>
<td>Total available area</td>
<td>152 415.2</td>
<td>mm²</td>
</tr>
</tbody>
</table>
### Nozzle neck thickness (according to UG-45)

<table>
<thead>
<tr>
<th>Calculation</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>1.50 mm</td>
<td>0.059 in</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27 and UG-28</td>
<td>1.77 mm</td>
<td>0.070 in</td>
</tr>
<tr>
<td>Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance</td>
<td>97.79 mm</td>
<td>3.850 in</td>
</tr>
<tr>
<td>tb1 no less than minimum thickness specified in UG-16(b)</td>
<td>97.79 mm</td>
<td>3.850 in</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with E=1) for the shell or head</td>
<td>0.50 mm</td>
<td>0.020 in</td>
</tr>
<tr>
<td>tb2 no less than minimum thickness specified in UG-16(b)</td>
<td>1.50 mm</td>
<td>0.059 in</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>8.34 mm</td>
<td>0.328 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>8.34 mm</td>
<td>0.328 in</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>76.20 mm</td>
<td>3.000 in</td>
</tr>
<tr>
<td>tn ≥ t(UG-45): Ok</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Validation warnings:

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]
Design data

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

Internal design pressure
\[ P = 18.44 \, MPa \quad 2674.0 \, psi \]

External design temperature
\[ T_e = 20.00 \, ^\circ C \quad 68.00 \, ^\circ F \]

External design pressure
\[ P_e = 0.10 \, MPa \quad 14.9 \, psi \]

Joint efficiency
\[ E = 1.00 \]

Material: SA-182 F22 3 - Forgings

Allowable stress
\[ S = 127.76 \, MPa \quad 18530.0 \, psi \]

Allowable stress at room temperature
\[ S_T = 148.00 \, MPa \quad 21465.6 \, psi \]

Geometry

Inside diameter
\[ D = 635.00 \, mm \quad 25.000 \, in \]

Outside diameter
\[ D_o = 787.40 \, mm \quad 31.000 \, in \]

Length
\[ L = 581.00 \, mm \quad 22.874 \, in \]

Adopted thickness
\[ t = 76.20 \, mm \quad 3.000 \, in \]

Corrosion allowance
\[ c = 0 \, mm \quad 0 \, in \]

External corrosion allowance
\[ c_e = 0 \, mm \quad 0 \, in \]

Wall undertolerance
\[ c' = 0 \, mm \quad 0 \, in \]

Forming strain (cylinders formed from plate)
\[ \varepsilon_f = \frac{50 \cdot t}{(R + t/2)} = 10.71\% \]

Internal pressure

Allowable stress
\[ S = 127.76 \, MPa \quad 18530.0 \, psi \]

Internal pressure
\[ P_i = 18.44 \, MPa \quad 2674.0 \, psi \]

Overpressure due to static head
\[ P_h = 0 \, MPa \quad 0 \, psi \]

Calculation pressure
\[ P = P_i + P_h = 18.44 \, MPa \quad 2674.0 \, psi \]

Reference diameter
\[ = \text{Inside} \]

Calculation radius (inside)
\[ R = 317.50 \, mm \quad 12.500 \, in \]

Required thickness for circumferential stress, UG-27(c)(1)
\[ t_c = \frac{P(R + c + c')}{6E(R + c + c')} = 50.16 \, mm \quad 1.975 \, in \]

Required thickness for longitudinal stress, UG-27(c)(2)
\[ t_l = \frac{2SE + 0.4P + c + c_e + c' }{2SE + 0.4P} = 22.27 \, mm \quad 0.877 \, in \]

Minimum required thickness
\[ t_r = \max[t_r(\text{circ}),t_r(\text{long})] = 50.16 \, mm \quad 1.975 \, in \]

Maximum allowable pressures (at the top of the vessel)

New & cold (opening)
\[ = 21.61 \, MPa \quad 3134.6 \, psi \]

Hot & corroded (opening)
\[ = 18.66 \, MPa \quad 2705.9 \, psi \]

New & cold (cylinder)
\[ = 26.80 \, MPa \quad 3887.4 \, psi \]

Hot & corroded (cylinder)
\[ = 26.80 \, MPa \quad 3887.4 \, psi \]
### External pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>External design temperature</td>
<td>$T_e = 20.00 , ^\circ C$</td>
<td>68.00 °F</td>
</tr>
<tr>
<td>External pressure</td>
<td>$P_e = 0.10 , \text{MPa}$</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>External static head</td>
<td>$P_h = 0 , \text{MPa}$</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>$P = P_e + P_h = 0.10 , \text{MPa}$</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>$D_o = 787.40 , \text{mm}$</td>
<td>31.000 in</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>$E = 200 \times 10^3 , \text{MPa}$</td>
<td>29,007,547.6 psi</td>
</tr>
<tr>
<td>Axial length between reinforcements</td>
<td>$L = 331.00 , \text{mm}$</td>
<td>13.031 in</td>
</tr>
<tr>
<td>L / $D_o$ ratio</td>
<td>$= 0.42037$</td>
<td></td>
</tr>
<tr>
<td>Do / t ratio</td>
<td>$= 10.3333$</td>
<td></td>
</tr>
<tr>
<td>Factor A</td>
<td>$= 0.09136$</td>
<td></td>
</tr>
<tr>
<td>Factor B</td>
<td>$= 154.29 , \text{MPa}$</td>
<td>22,377.8 psi</td>
</tr>
</tbody>
</table>

$$P_a = \frac{4B}{3[D_o/(t - c_e - c_t) + 1] + L}$$

$$t = 1.77 \, \text{mm}$$

$$P_a \geq P : \text{Ok}$$

$$t \geq tr : \text{Ok}$$

### 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>$P = 19.91 , \text{MPa}$</td>
<td>2,887.5 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>$P = 19.91 , \text{MPa}$</td>
<td>2,887.5 psi</td>
</tr>
</tbody>
</table>
### Minimum Design Metal Temperature (MDMT)

**Internal pressure**

**Nozzle - Bocchello In 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 8.72 °C 47.70 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 34.50 °C 94.10 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8)$ = 1.00000</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm 0 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 97.79 mm 3.850 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$\frac{t_n E^*}{t_n - c} = 0.84304$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 25.78 °C 78.40 °F</td>
</tr>
</tbody>
</table>

**External pressure**

**Nozzle - Bocchello In 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 34.50 °C 94.10 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8)$ = 1.00000</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm 0 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 4.77 mm 0.188 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm 4.567 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$\frac{t_n E^*}{t_n - c} = 0.04112$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= -45.50 °C -49.90 °F</td>
</tr>
</tbody>
</table>

**Validation warnings:**

- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]
### Design data

**Internal design temperature**

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

**Internal design pressure**

\[ P = 18.44 \, \text{MPa} \quad 2674.0 \, \text{psi} \]

**External design temperature**

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

**External design pressure**

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

**Joint efficiency**

\[ E = 1.00 \]

### Flange material

SA-182 F22 3 - Forgings

### Shell material

SA-182 F22 3 - Forgings

### Bolting material

SA-193 B16 - 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>Sfo=127.76 MPa / 18 530.0 psi</td>
<td>Sno=127.76 MPa / 18 530.0 psi</td>
<td>Sbo=144.76 MPa / 20 995.7 psi</td>
</tr>
<tr>
<td>Seating</td>
<td>Sfg=148.00 MPa / 21 465.6 psi</td>
<td>Sng=148.00 MPa / 21 465.6 psi</td>
<td>Sbg=152.00 MPa / 22 045.7 psi</td>
</tr>
</tbody>
</table>

### Internal pressure

\[ P_d = 18.44 \, \text{MPa} \quad 2674.0 \, \text{psi} \]

### Overpressure due to static head

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

### Calculation pressure

\[ P = 18.44 \, \text{MPa} \quad 2674.0 \, \text{psi} \]

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

\[ E_{yo} = 179,600.00 \, \text{MPa} \quad 26,048,877.7 \, \text{psi} \]

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

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

### Corrosion allowance

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

### Flange external diameter

\[ A = 1420.00 \, \text{mm} \quad 55.906 \, \text{in} \]

### Inside diameter

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

### Bolt circle

\[ C = 1235.00 \, \text{mm} \quad 48.622 \, \text{in} \]

### Flange thickness

\[ t = 310.00 \, \text{mm} \quad 12.205 \, \text{in} \]

### Mean gasket diameter

\[ G_{\text{mean}} = 742.00 \, \text{mm} \quad 29.213 \, \text{in} \]

### Thickness of the hub at the small end

\[ g_0 = 76.20 \, \text{mm} \quad 3.000 \, \text{in} \]

### Thickness of the hub at the large end

\[ g_1 = 90.00 \, \text{mm} \quad 3.543 \, \text{in} \]

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

\[ g_0' = g_0 - c = 76.20 \, \text{mm} \quad 3.000 \, \text{in} \]

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

\[ g_1' = g_1 - c = 90.00 \, \text{mm} \quad 3.543 \, \text{in} \]

### Hub length

\[ h = 115.00 \, \text{mm} \quad 4.528 \, \text{in} \]

### Gasket parameters

**Gasket factor**

\[ m = 4.25 \]

**Gasket factor**

\[ y = 70.00 \, \text{MPa} \quad 10152.6 \, \text{psi} \]

**Gasket contact width**

\[ N = 106.40 \, \text{mm} \quad 4.189 \, \text{in} \]

**Basic gasket seating width**

\[ b_0 = \frac{N}{2} = 53.20 \, \text{mm} \quad 2.094 \, \text{in} \]

**Conversion factor for length**

\[ C_b = 2.5000 \]

**Effective gasket contact width**

\[ b = C_b \sqrt{N} = 18.23 \, \text{mm} \quad 0.718 \, \text{in} \]

**Outside diameter of the gasket contact area**

\[ G_c = G_{\text{mean}} + N = 848.40 \, \text{mm} \quad 33.402 \, \text{in} \]

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

\[ G = G_c - 2b = 811.93 \, \text{mm} \quad 31.966 \, \text{in} \]
### Bolt loads

- **Number of bolts**: \( n = 20 \)
- **Bolt type**: ANSI_TEMA 3-3/4"
- **Bolt spacing**: \( B_s = 193.99 \text{ mm} = 7.638 \text{ in} \)
- **Nominal bolt diameter**: \( a = 95.25 \text{ mm} = 3.750 \text{ in} \)
- **Design bolt load for the operating condition**: \( W_o = 0.785G^2P + 2\pi\sigma_{Gm}P = 16829735 \text{ N} = 3783474.67 \text{ lbf} \)
- **Design bolt load for the gasket seating (Bolt)**: \( W_p = \pi^2\sigma_{Gm} = 3255838 \text{ N} = 731941.41 \text{ lbf} \)
- **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**: \( A_{m} = \max \left( \sqrt[4]{\frac{N}{G}}, \sqrt[6]{\frac{P}{G}} \right) = 116259.6 \text{ mm}^2 = 180.203 \text{ in}^2 \)
- **Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion**: \( A_b = 130425.5 \text{ mm}^2 = 202.160 \text{ in}^2 \)
- **Maximum bolts area for gasket crush**: \( A_{b,max} = \left( \frac{A_m + A_b}{2} \right)S_{bg} = 249974.1 \text{ mm}^2 = 387.461 \text{ in}^2 \)
- **Design bolt load for the gasket seating (Flange)**: \( W_g = \left( \frac{A_m + A_b}{2} \right)S_{bg} = 18748069 \text{ N} = 4214733.16 \text{ lbf} \)

### Flange constants

- **Ratio of the flange outside diameter to the inside diameter**: \( K = A/B' = 2.23622 \)
- **Stress factor Y**: \( Y = \frac{1}{K - 1} \left[ \frac{0.66845 + 571690}{K^{2} - 1} \right] = 2.56106 \)
- **Stress factor T**: \( T = \frac{K^2(1 + 855246\log(K) - 1)}{(104720 + 194448K^2)(K - 1)} = 1.42288 \)
- **Stress factor U**: \( U = \frac{136136(K^2-1)(K - 1)}{K^2(1 + 855246\log(K) - 1)} = 2.81435 \)
- **Stress factor Z**: \( Z = \frac{(K^2 + 1)}{(K^2 - 1)} = 1.49991 \)
- **Hub length parameter**: \( h_o = \sqrt[4]{\frac{a}{S_{bg}}} = 219.97 \text{ mm} = 8.660 \text{ in} \)
- **Hub thickness ratio**: \( X_o = \frac{S_{bg}}{h_o} = 1.18110 \)
- **Hub length ratio**: \( X_h = \frac{h_o}{h_o} = 0.52280 \)
- **Flange stress factor for integral type flanges**: \( F = \text{fig. 2-7.2} = 0.87835 \)
- **Flange stress factor for integral type flanges**: \( V = \text{fig. 2-7.3} = 0.42577 \)
- **Hub stress correction factor for integral flanges**: \( f = \text{fig. 2-7.6} = 1.00000 \)
- **Stress factor e**: \( e = \frac{F}{V} = 0.00399 \)
- **Stress factor L**: \( L = \frac{h_o + 1}{F - 1} = 8407169.84802 \)
- **Gasket load for the operating condition**: \( H_G = W_o - H = 7288903 \text{ N} = 1638610.48 \text{ lbf} \)
- **Total hydrostatic end force on the area inside of the flange**: \( H_D = 0.785G^2P = 5835736 \text{ N} = 1311925.51 \text{ lbf} \)
- **Total hydrostatic end force**: \( H = 0.785G^2P = 9540832 \text{ N} = 2144864.19 \text{ lbf} \)
- **Difference**: \( H_d = H - H_D = 3705096 \text{ N} = 832938.67 \text{ lbf} \)
- **Moment arm for load HD**: \( h_D = \frac{1}{2} \left[ \frac{C - B - S_h}{h_o} \right] = 255.00 \text{ mm} = 10.039 \text{ in} \)
- **Moment arm for load HG**: \( h_G = \frac{1}{2} \left[ \frac{C - B - S_h}{h_o} \right] = 211.53 \text{ mm} = 8.328 \text{ in} \)
- **Moment arm for load HT**: \( h_T = \frac{1}{2} \left[ \frac{C - B - S_h}{h_o} \right] = 255.77 \text{ mm} = 10.070 \text{ in} \)
Flange moments
Nominal bolt diameter
\[ dB = 95.25 \text{ mm} = 3.750 \text{ in} \]
TEMA Load concentration factor
\[ c_F = \text{MAX} \left[ \frac{\frac{n^2}{f^2}}{2dB} \right] = 1.00000 \]
Moment factor used to design split rings
\[ F_s = 1.00 \]
Flange design moment for the operating condition
\[ M_c = c_F \cdot \frac{W_c}{2} \left[ H_1 + H_2 + H_o \right] = 3977610.2 \text{ N·m} = 35204813.0 \text{ lbf·in} \]
Flange design moment for the gasket seating condition
\[ M_g = c_F \cdot \frac{W_g}{2} \left[ (C - G) \right] = 3965864.9 \text{ N·m} = 35100858.6 \text{ lbf·in} \]

Flange stresses - operating condition
Corrected inside diameter of the flange
\[ B_1' = B' + g_1' = 725.00 \text{ mm} = 28.543 \text{ in} \]
Flange hub stress - operating condition
\[ S_H = \frac{fM_c}{L_0g_2B_1} = 132.39 \text{ MPa} = 19201.1 \text{ psi} \]
Flange radial stress - operating condition
\[ S_R = \left( \frac{133e+1}{M_c} \right) M_c = 33.71 \text{ MPa} = 4889.9 \text{ psi} \]
Flange tangential stress - operating condition
\[ S_T = \frac{Y M_c R}{r B} - Z S_R = 116.37 \text{ MPa} = 16877.4 \text{ psi} \]
\[ S_{Ho} \leq \min[1.5Sfo, 2.5Sno]: \text{ Ok} \]
\[ S_{Ro} \leq Sfo: \text{ Ok} \]
\[ S_{To} \leq Sfo: \text{ Ok} \]
\[ \frac{(S_{Ho} + S_{Ro})}{2} \leq Sfo: \text{ Ok} \]
\[ \frac{(S_{Ho} + S_{To})}{2} \leq Sfo: \text{ Ok} \]

Flange stresses - seating condition
Flange hub stress - gasket seating condition
\[ S_H = \frac{fM_g}{L_0g_2B_1} = 132.00 \text{ MPa} = 19144.4 \text{ psi} \]
Flange radial stress - gasket seating condition
\[ S_R = \left( \frac{133e+1}{M_g} \right) M_g = 33.61 \text{ MPa} = 4875.4 \text{ psi} \]
Flange tangential stress - gasket seating condition
\[ S_T = \frac{Y M_g R}{r B} - Z S_R = 116.02 \text{ MPa} = 16827.6 \text{ psi} \]
\[ S_{Hg} \leq \min[1.5Sfg, 2.5Sng]: \text{ Ok} \]
\[ S_{Rg} \leq Sfg: \text{ Ok} \]
\[ S_{Tg} \leq Sfg: \text{ Ok} \]
\[ \frac{(S_{Hg} + S_{Rg})}{2} \leq Sfg: \text{ Ok} \]
\[ \frac{(S_{Hg} + S_{Tg})}{2} \leq Sfg: \text{ Ok} \]

Flange rigidity - operating condition
Rigidity index factor
\[ K_R = 0.30 \]
Flange rigidity index
\[ J = \frac{5214V M_c}{L E y g_2 B_1} = 0.25185 \]
\[ J_{o} \leq 1: \text{ Ok} \]
Flange rigidity - seating condition
Flange rigidity index
\[ J = \frac{5214V M_g}{L E y g_2 B_1} = 0.21440 \]
\[ J_{g} \leq 1: \text{ Ok} \]
Hub thickness
Minimum hub thickness as cylindrical shell
\[ th_{min} = 50.16 \text{ mm} = 1.975 \text{ in} \]
\[ g_0 \geq th_{min}: \text{ Ok} \]
<table>
<thead>
<tr>
<th></th>
<th>New &amp; cold (flange)</th>
<th>Hot &amp; corroded (flange)</th>
<th>New &amp; cold (bolts)</th>
<th>Hot &amp; corroded (bolts)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>21.94 MPa</td>
<td>18.94 MPa</td>
<td>21.72 MPa</td>
<td>20.68 MPa</td>
</tr>
<tr>
<td></td>
<td>3181.9 psi</td>
<td>2746.7 psi</td>
<td>3149.8 psi</td>
<td>2999.8 psi</td>
</tr>
</tbody>
</table>
### 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=148.00 MPa / 21 465.6 psi</td>
<td>Sno=148.00 MPa / 21 465.6 psi</td>
<td>Sbo=152.00 MPa / 22 045.7 psi</td>
</tr>
<tr>
<td>Seating condition</td>
<td>Sfg=148.00 MPa / 21 465.6 psi</td>
<td>Sng=148.00 MPa / 21 465.6 psi</td>
<td>Sbg=152.00 MPa / 22 045.7 psi</td>
</tr>
</tbody>
</table>

- External design pressure: \( Pe = 0.10 \text{ MPa} \) (14.9 psi)
- Overpressure due to static head: \( Ph = 0 \text{ MPa} \) (0 psi)
- Calculation pressure: \( P = Pe + Ph = 0.10 \text{ MPa} \) (14.9 psi)
- External design temperature: \( Te = 20.00 ^\circ\text{C} \) (68.00 °F)
- Modulus of Elasticity at the operating load case temperature: \( Eyo = 210 350.00 \text{ MPa} \) (30 508 688.2 psi)
- Modulus of Elasticity at the gasket seating load case temperature: \( Eyg = 210 350.00 \text{ MPa} \) (30 508 688.2 psi)

### Bolt loads

- Number of bolts: \( n = 20 \)
- Bolt type: ANSI_TEMA 3-3/4"
- Bolt spacing: \( Bs = 193.99 \text{ mm} \) (7.638 in)
- Nominal bolt diameter: \( a = 95.25 \text{ mm} \) (3.750 in)
- Design bolt load for the operating condition: \( W_o = 0.785G^2P + 2b\pi GmP = 94 023 \text{ N} \) (21 137.30 lbf)
- Design bolt load for the gasket seating (Bolt): \( W_f = \pi bGy = 3 255 838 \text{ N} \) (731 941.41 lbf)
- External tensile net-section axial force: \( FA = 0 \text{ N} \) (0 lbf)
- Absolute value of the external net-section bending moment: \( ME = 0 \text{ N m} \) (0 lbf in)
- External tensile net-section axial force (test condition): \( FA_t = 0 \text{ N} \) (0 lbf)
- External tensile net-section bending moment (test condition): \( ME_t = 0 \text{ N m} \) (0 lbf in)
- Total minimum required cross-sectional area of the bolts: \( A_{min} = \max \left( \frac{W_o + F_d + 4W_f}{S_{bo}}, \frac{W_f}{S_{bg}} \right) = 21 420.0 \text{ mm}^2 \) (33.201 in²)
- Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion: \( Ab = 130 425.5 \text{ mm}^2 \) (202.160 in²)
- Maximum bolts area for gasket crush: \( A_{m,\text{max}} = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bt}} = 249 974.1 \text{ mm}^2 \) (387.461 in²)
- Design bolt load for the gasket seating (Flange): \( W_g = \left( \frac{A_m + Ab}{2} \right)S_{bg} = 11 540 260 \text{ N} \) (2 594 353.52 lbf)
Flange constants

Ratio of the flange outside diameter to the inside diameter

\[ K = \frac{A}{B'} = 2.23622 \]

Stress factor Y

\[ Y = 0.66845 + 5.71690 \left( \frac{K^2 - 1}{K^2} \log_{10}(K) \right) = 2.56106 \]

Stress factor T

\[ T = \frac{10.4720 + 194.48K^2}{K^2(1 + 855246\log_{10}(K) - 1)} = 1.42288 \]

Stress factor U

\[ U = \frac{136136(K^2 - 1)(K - 1)}{K^2 + 1} = 2.81435 \]

Stress factor Z

\[ Z = \frac{1}{K^2 - 1} = 1.49991 \]

Hub length parameter

\[ h_o = \sqrt{\frac{E_{so}}{E_{ho}}} = 219.97 \text{ mm} \quad 8.660 \text{ in} \]

Hub thickness ratio

\[ X_o = \frac{1}{2} \frac{E_{so}}{E_{ho}} = 1.18110 \]

Hub length ratio

\[ X_h = \frac{h_o}{h_0} = 0.52280 \]

Flange stress factor for integral type flanges

\[ F = \text{fig. 2-7.2} = 0.87835 \]

Flange stress factor for integral type flanges

\[ V = \text{fig. 2-7.3} = 0.42757 \]

Hub stress correction factor for integral flanges

\[ f = \text{fig. 2-7.6} = 1.00000 \]

Stress factor e

\[ e = \frac{C - B}{2} = 0.00399 \]

Stress factor d

\[ d = \frac{Ug_{2}h_{2}}{V} = 8,407,169.84802 \]

Stress factor L

\[ L = \frac{te + 1}{t} = 5,11628 \]

Gasket load for the operating condition

\[ H_G = W_o - H \quad 40,721 \text{ N} \quad 9,154.50 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange

\[ H_D = 0.7856H_G^2 \quad 32,603 \text{ N} \quad 7,329.39 \text{ lbf} \]

Total hydrostatic end force

\[ H = 0.7856H_D^2 \quad 53,302 \text{ N} \quad 11,982.81 \text{ lbf} \]

Difference

\[ H_T = H - H_G \quad 20,699 \text{ N} \quad 4,653.41 \text{ lbf} \]

Moment arm for load HD

\[ h_D = \frac{C - B}{2} \quad 255.00 \text{ mm} \quad 10.039 \text{ in} \]

Moment arm for load HG

\[ h_G = \frac{C - B}{2} \quad 211.53 \text{ mm} \quad 8.328 \text{ in} \]

Moment arm for load HT

\[ h_T = \frac{1}{2} \left( \frac{C - B}{2} + h_G \right) \quad 255.77 \text{ mm} \quad 10.070 \text{ in} \]

Flange moments

Nominal bolt diameter

\[ d_B = 95.25 \text{ mm} \quad 3.750 \text{ in} \]

TEMA Load concentration factor

\[ c_T = \text{MAX} \left( \sqrt{\frac{r^2}{2d_B + \frac{r^2}{m_{t}}}}, 1 \right) = 1.00000 \]

Moment factor used to design split rings

\[ M = F_s = 1.00 \]

Flange design moment for the operating condition

\[ M_o = \text{abs} \left( H_f(h_D - h_G) + H_f(h_T - h_G)F_s \right) \quad 2,332.7 \text{ N·m} \quad 20,646.0 \text{ lbf·in} \]

Flange design moment for the gasket seating condition

\[ M_s = W_g h_o F_s = 2,441,164.2 \text{ N·m} \quad 21,606.121.4 \text{ lbf·in} \]
Flange stresses - operating condition

Corrected inside diameter of the flange

\[ B_1' = B' + g_1' = 725.00 \text{ mm} \]

Flange hub stress - operating condition

\[ S_H = \frac{fM_z}{L_g^2B_1'} = 0.08 \text{ MPa} \quad 11.3 \text{ psi} \]

Flange radial stress - operating condition

\[ S_R = \frac{(133e + 1)M_z}{L_g^2B_1'} = 0.02 \text{ MPa} \quad 2.9 \text{ psi} \]

Flange tangential stress - operating condition

\[ S_T = \frac{YM_R}{R^2} - ZS_R = 0.07 \text{ MPa} \quad 9.9 \text{ psi} \]

\[ \frac{S_H}{0.6} \leq \min[1.5S_f, 2.5S_{no}] : \text{ Ok} \]

\[ S_R \leq S_f : \text{ Ok} \]

\[ S_T \leq S_f : \text{ Ok} \]

\[ \frac{(S_H + S_R)}{2} \leq S_f : \text{ Ok} \]

\[ \frac{(S_H + S_T)}{2} \leq S_f : \text{ Ok} \]

Flange stresses - seating condition

Flange hub stress - gasket seating condition

\[ S_H = \frac{fM_z}{L_g^2B_1'} = 81.25 \text{ MPa} \quad 11,784.2 \text{ psi} \]

Flange radial stress - gasket seating condition

\[ S_R = \frac{(133e + 1)M_z}{L_g^2B_1'} = 20.69 \text{ MPa} \quad 3,001.0 \text{ psi} \]

Flange tangential stress - gasket seating condition

\[ S_T = \frac{YM_R}{R^2} - ZS_R = 71.42 \text{ MPa} \quad 10,358.1 \text{ psi} \]

\[ \frac{S_H}{0.6} \leq \min[1.5S_{fg}, 2.5S_{ng}] : \text{ Ok} \]

\[ S_R \leq S_{fg} : \text{ Ok} \]

\[ S_T \leq S_{fg} : \text{ Ok} \]

\[ \frac{(S_H + S_R)}{2} \leq S_{fg} : \text{ Ok} \]

\[ \frac{(S_H + S_T)}{2} \leq S_{fg} : \text{ Ok} \]

Flange rigidity - operating condition

Rigidity index factor

\[ K_R = 0.30 \]

Flange rigidity index

\[ J = \frac{5214VM_R}{LE_f \gamma g_{k_f}K_f h_0} = 0.00013 \]

\[ J_0 \leq 1 : \text{ Ok} \]

Flange rigidity - seating condition

Flange rigidity index

\[ J = \frac{5214VM_R}{LE_f \gamma g_{k_f}K_f h_0} = 0.13197 \]

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

### Internal pressure

**Welding neck flange - Flange In Tube Side 26” (Bolting)**

| Material | = | SA-193 B16 |
| Governing Thickness | = | 95.25 mm 3.750 in |
| Reduction in MDMT based on available excess thickness | TR | 80.00 °C 176.00 °F |
| Unadjusted MDMT from table UCS-66 | = | -30.00 °C -22.00 °F |
| Adjusted MDMT from fig. UCS-66.1 | = | -110.00 °C -166.00 °F |

**Welding neck flange - Flange In Tube Side 26”, Flange**

| Material | = | SA-182 F22 3 |
| Curve of fig. UCS-66 | = | B |
| Governing Thickness | = | 76.20 mm 3.000 in |
| Reduction in MDMT based on available excess thickness | TR | 1.47 °C 34.65 °F |
| Unadjusted MDMT from table UCS-66 | = | 26.00 °C 78.80 °F |
| Maximum allowable working pressure | MAWP H&C | = | 18.94 MPa 2746.7 psi |
| Coincident ratio | MAWP H & C | = | 0.97351 |
| Adjusted MDMT from fig. UCS-66.1 | = | 24.53 °C 76.15 °F |

### External pressure

**Welding neck flange - Flange In Tube Side 26” (Bolting)**

| Material | = | SA-193 B16 |
| Governing Thickness | = | 95.25 mm 3.750 in |
| Reduction in MDMT based on available excess thickness | TR | 80.00 °C 176.00 °F |
| Unadjusted MDMT from table UCS-66 | = | -30.00 °C -22.00 °F |
| Adjusted MDMT from fig. UCS-66.1 | = | -110.00 °C -166.00 °F |

**Welding neck flange - Flange In Tube Side 26”, Flange**

| Material | = | SA-182 F22 3 |
| Curve of fig. UCS-66 | = | B |
| Governing Thickness | = | 76.20 mm 3.000 in |
| Reduction in MDMT based on available excess thickness | TR | 80.00 °C 176.00 °F |
| Unadjusted MDMT from table UCS-66 | = | 26.00 °C 78.80 °F |
| Maximum allowable working pressure | MAWP H&C | = | 18.94 MPa 2746.7 psi |
| Coincident ratio | MAWP H & C | = | 0.00544 |
| Adjusted MDMT from fig. UCS-66.1 | = | -54.00 °C -65.20 °F |
Nozzle - Bocchello Out Tube Side

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - Metric Units

Design data

Internal design temperature \( T = 454.00 \, ^\circ C \quad 849.20 \, ^\circ F \)
Internal design pressure \( P = 18.44 \, MPa \quad 2674.0 \, psi \)
External design temperature \( T_e = 20.00 \, ^\circ C \quad 68.00 \, ^\circ F \)
External design pressure \( P_e = 0.10 \, MPa \quad 14.9 \, psi \)
Joint efficiency \( E = 1.00 \)

Material: SA-182 F22 3 - Forgings

Allowable stress \( S = 127.76 \, MPa \quad 18530.0 \, psi \)
Allowable stress at room temperature \( ST = 148.00 \, MPa \quad 21465.6 \, psi \)

Geometry

Inside diameter \( D = 635.00 \, mm \quad 25.000 \, in \)
Outside diameter \( Do = 787.40 \, mm \quad 31.000 \, in \)
Length \( L = 581.00 \, mm \quad 22.874 \, in \)
Adopted thickness \( t = 76.20 \, mm \quad 3.000 \, in \)
Corrosion allowance \( c = 0 \, mm \quad 0 \, in \)
External corrosion allowance \( ce = 0 \, mm \quad 0 \, in \)
Wall undertolerance \( c' = 0 \, mm \quad 0 \, in \)
Forming strain (cylinders formed from plate) \( \epsilon_f = 50 \cdot t/(R + t/2) = 10.71\% \)

Internal pressure

Allowable stress \( S = 127.76 \, MPa \quad 18530.0 \, psi \)
Internal pressure \( P_i = 18.44 \, MPa \quad 2674.0 \, psi \)
Overpressure due to static head \( Ph = 0 \, MPa \quad 0 \, psi \)
Calculation pressure \( P = P_i + Ph = 18.44 \, MPa \quad 2674.0 \, psi \)
Reference diameter \( = \) Inside
Calculation radius (inside) \( R = 317.50 \, mm \quad 12.500 \, in \)
Required thickness for circumferential stress, UG-27(c)(1) \( t_c = \frac{P(R + c + c')}{6SE(R + c + c')} + c_0 + c' + c'' = 50.16 \, mm \quad 1.975 \, in \)
Required thickness for longitudinal stress, UG-27(c)(2) \( t_l = \frac{2SE + 0.4P}{2SE + 0.4P + c + c_0 + c'} = 22.27 \, mm \quad 0.877 \, in \)
Minimum required thickness \( tr = \max[tr(circ),tr(long)] = 50.16 \, mm \quad 1.975 \, in \)

Maximum allowable pressures (at the top of the vessel)

New & cold (opening) \( = 21.76 \, MPa \quad 3156.3 \, psi \)
Hot & corroded (opening) \( = 18.79 \, MPa \quad 2724.7 \, psi \)
New & cold (cylinder) \( = 26.80 \, MPa \quad 3887.4 \, psi \)
Hot & corroded (cylinder) \( = 26.80 \, MPa \quad 3887.4 \, psi \)
### External pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Conversion</th>
</tr>
</thead>
<tbody>
<tr>
<td>External design temperature</td>
<td>Te = 20.00 °C</td>
<td>68.00 °F</td>
</tr>
<tr>
<td>External pressure</td>
<td>Pe = 0.10 MPa</td>
<td>14.9 psi</td>
</tr>
<tr>
<td>External static head</td>
<td>Ph = 0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>P = Pe + Ph</td>
<td>0.10 MPa</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>Do = 787.40 mm</td>
<td>31.000 in</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>E = 200,000.00 MPa</td>
<td>29,007,547.6 psi</td>
</tr>
<tr>
<td>Axial length between reinforcements</td>
<td>L = 331.00 mm</td>
<td>13.031 in</td>
</tr>
<tr>
<td>L / Do ratio</td>
<td></td>
<td>0.42037</td>
</tr>
<tr>
<td>Do / t ratio</td>
<td></td>
<td>10.33333</td>
</tr>
<tr>
<td>Factor A</td>
<td></td>
<td>0.09136</td>
</tr>
<tr>
<td>Factor B</td>
<td></td>
<td>154.29 MPa</td>
</tr>
<tr>
<td></td>
<td></td>
<td>22,377.8 psi</td>
</tr>
<tr>
<td>External pressure</td>
<td>Pa = $\frac{4B}{3[D_o/(t-c_e-c_t)]}$</td>
<td>19.91 MPa</td>
</tr>
<tr>
<td>Required thickness</td>
<td>tr = 1.77 mm</td>
<td>0.070 in</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pa ≥ P: Ok</td>
</tr>
<tr>
<td></td>
<td></td>
<td>t ≥ tr: Ok</td>
</tr>
</tbody>
</table>

### Maximum allowable external pressures

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
<th>Conversion</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>19.91 MPa</td>
<td>2,887.5 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>19.91 MPa</td>
<td>2,887.5 psi</td>
</tr>
</tbody>
</table>
## 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 8.72 °C</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 34.50 °C</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8)$ = 1.00000</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 97.79 mm</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm</td>
</tr>
</tbody>
</table>
| Coincident ratio                                                          | \[ \frac{t_n E^*}{t_n - c} = 0.84304 \]
| Adjusted MDMT from fig. UCS-66.1                                         | = 25.78 °C | 78.40 °F |

### 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 34.50 °C</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8)$ = 1.00000</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 0 mm</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 4.77 mm</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 116.00 mm</td>
</tr>
</tbody>
</table>
| Coincident ratio                                                          | \[ \frac{t_n E^*}{t_n - c} = 0.04112 \]
| Adjusted MDMT from fig. UCS-66.1                                         | = -45.50 °C | -49.90 °F |

### Validation warnings:
- Since shell-nozzle diameters are outside limits given in UG-36(b)(1), Appendix 1-7 is applied: in vessels with 60 in. (1500 mm) inside diameter or less, nozzle diameter shall not be greater than one-half the vessel diameter or 20 in. (500 mm). Vessel diameter is 1 238.00 mm with a nozzle diameter of 635.00 mm. Alternatively, rules of Appendix 1-10 can be applied. Code reference: UG-36(b)(1) [Required value: 500.00 mm]
Design data

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

Internal design pressure
\[ P = 18.44 \, \text{MPa} \quad 2674.0 \, \text{psi} \]

External design temperature
\[ T_e = 20.00 \, ^\circ\text{C} \quad 68.00 \, ^\circ\text{F} \]

External design pressure
\[ P_e = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]

Joint efficiency
\[ E = 1.00 \]

Flange material
SA-182 F22 3 - Forgings

Shell material
SA-182 F22 3 - Forgings

Bolting material
SA-193 B16 - 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} = 127.76 , \text{MPa} / 18530.0 , \text{psi} )</td>
<td>( S_{no} = 127.76 , \text{MPa} / 18530.0 , \text{psi} )</td>
<td>( S_{bo} = 144.76 , \text{MPa} / 20995.7 , \text{psi} )</td>
</tr>
<tr>
<td>Seating</td>
<td>( S_{fg} = 148.00 , \text{MPa} / 21465.6 , \text{psi} )</td>
<td>( S_{ng} = 148.00 , \text{MPa} / 21465.6 , \text{psi} )</td>
<td>( S_{bg} = 152.00 , \text{MPa} / 22045.7 , \text{psi} )</td>
</tr>
</tbody>
</table>

Gasket parameters

- Gasket factor: \( m = 4.25 \)
- Gasket factor: \( y = 70.00 \, \text{MPa} \quad 10152.6 \, \text{psi} \)
- Gasket contact width: \( N = 106.40 \, \text{mm} \quad 4.189 \, \text{in} \)
- Basic gasket seating width: \( b_b = 53.20 \, \text{mm} \quad 2.094 \, \text{in} \)
- Conversion factor for length: \( C_b = 2.5000 \)
- Effective gasket contact width: \( b = C_b \sqrt{N} = 18.23 \, \text{mm} \quad 0.718 \, \text{in} \)
- Outside diameter of the gasket contact area: \( G_c = G_{mean} + N = 848.40 \, \text{mm} \quad 33.402 \, \text{in} \)
- Diameter at the location of the gasket load reaction: \( G = G_c - 2b = 811.93 \, \text{mm} \quad 31.966 \, \text{in} \)
Bolt loads

Number of bolts \( n = 20 \)

Bolt type = ANSI_TEMA 3-3/4"

Bolt spacing \( B_s = 193.99 \text{ mm} \) \( 7.638 \text{ in} \)

Nominal bolt diameter \( a = 95.25 \text{ mm} \) \( 3.750 \text{ in} \)

Design bolt load for the operating condition \( W_o = 0.785G^2P + 2\pi\theta\text{GMP} \)

\( W_o \) = 16 829 735 N \( 3 783 474.67 \text{ lbf} \)

Design bolt load for the gasket seating (Bolt) \( W_g = \pi b G_y \)

\( W_g \) = 3 255 838 N \( 731 941.41 \text{ lbf} \)

External tensile net-section axial force \( FA = 0 \text{ N} \) \( 0 \text{ lbf} \)

Absolute value of the external net-section bending moment \( ME = 0 \text{ N·m} \) \( 0 \text{ lbf·in} \)

External tensile net-section axial force (test condition) \( FAt = 0 \text{ N} \) \( 0 \text{ lbf} \)

External tensile net-section bending moment (test condition) \( MEt = 0 \text{ N·m} \) \( 0 \text{ lbf·in} \)

Total minimum required cross-sectional area of the bolts \( A_m = \max \left( \frac{W_o + F_d + \frac{4\pi \sqrt{y}}{S_{bo}}}{S_{bo}} \right) \left( \frac{W_g}{S_{bg}} \right) \)

\( A_m \) = 116 259.6 mm² \( 180.203 \text{ in}² \)

Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion \( A_B = 130 425.5 \text{ mm²} \) \( 202.160 \text{ in}² \)

Maximum bolts area for gasket crush \( A_m = \frac{2\pi\sqrt{y}G+N}{S_{bh}} \)

\( A_m \) = 249 974.1 mm² \( 387.461 \text{ in}² \)

Design bolt load for the gasket seating (Flange) \( W_g = \left( \frac{A_m + A_B}{2} \right) S_{bg} \)

\( W_g \) = 18 748 069 N \( 4 214 733.16 \text{ lbf} \)

\( Ab \geq Am: Ok \)

Flange constants

Ratio of the flange outside diameter to the inside diameter \( K = A/B' \)

\( K \) = 2.23622

Stress factor \( Y \)

\( Y = \left( \frac{K^2 \log_{10}(K)}{K^2 - 1} \right) \left( \frac{0.66845 + 571690}{K^2 - 1} \right) \)

\( Y \) = 2.56106

Stress factor \( T \)

\( T = \frac{K^2 (1 + 855246 \log(104720 + 19448K^2)) (K - 1)}{(104720 + 19448K^2) (K - 1) - 1} \)

\( T \) = 1.42288

Stress factor \( U \)

\( U = \frac{136136(K^2 - 1)(K - 1)}{136136(K^2 - 1)(K - 1) - 1} \)

\( U \) = 2.81435

Stress factor \( Z \)

\( Z = \frac{(K^2 + 1)}{(K^2 - 1)} \)

\( Z \) = 1.49991

Hub length parameter \( h_0 = \sqrt{\frac{I}{G_{bh}}} \)

\( h_0 \) = 219.97 mm \( 8.660 \text{ in} \)

Hub thickness ratio \( X_g = \frac{G_c}{G_b} \)

\( X_g \) = 1.18110

Hub length ratio \( X_h = \frac{h_0}{h} \)

\( X_h \) = 0.52280

Flange stress factor for integral type flanges \( F (\text{fig. 2-7.2}) \)

\( F \) = 0.87835

Flange stress factor for integral type flanges \( V (\text{fig. 2-7.3}) \)

\( V \) = 0.42757

Hub stress correction factor for integral flanges \( f (\text{fig. 2-7.6}) \)

\( f \) = 1.00000

Stress factor \( e \)

\( e = \frac{1}{h} \)

\( e \) = 0.00399

Stress factor \( d \)

\( d = \frac{U_g h_0}{P} \)

\( d \) = 8 407 169.84802

Stress factor \( L \)

\( L = \frac{K^2 + 1}{K^2 - 1} \)

\( L \) = 5.11628

Gasket load for the operating condition \( H_G = W_o - H \)

\( H_G \) = 7 288 903 N \( 1 638 610.48 \text{ lbf} \)

Total hydrostatic end force on the area inside of the flange \( H_d = 0.785G^2P \)

\( H_d \) = 5 835 736 N \( 1 311 925.51 \text{ lbf} \)

Total hydrostatic end force \( H = 0.785G^2P \)

\( H \) = 9 540 832 N \( 2 144 864.19 \text{ lbf} \)

Difference \( H = H - H_G \)

\( H \) = 3 705 096 N \( 832 938.67 \text{ lbf} \)

Moment arm for load HD \( hD = \frac{C - B}{2} \)

\( hD \) = 255.00 mm \( 10.039 \text{ in} \)

Moment arm for load HG \( hG = \frac{C - B}{2} \)

\( hG \) = 211.53 mm \( 8.328 \text{ in} \)

Moment arm for load HT \( hT = \frac{1}{2} \left( C - B + h_G \right) \)

\( hT \) = 255.77 mm \( 10.070 \text{ in} \)
Flange moments

Nominal bolt diameter

\[ d_B = 95.25 \text{ mm} \]
\[ 3.750 \text{ in} \]

TEMA Load concentration factor

\[ c_F = \text{MAX} \left\{ \frac{d_B^2}{2d_B + \frac{d_B}{e_{65}}} , 1 \right\} = 1.00000 \]

Moment factor used to design split rings

\[ F_s = 1.00 \]

Flange design moment for the operating condition

\[ M_s = c_F \cdot \text{abs} \left[ (H_p h_D + H_p h_T + H_p h_C) F_s \right] = 3977610.2 \text{ N·m} \]
\[ 35204813.0 \text{ lbf·in} \]

Flange design moment for the gasket seating condition

\[ M_g = c_F \cdot \frac{W_s (C - G)}{2} F_s = 3965864.9 \text{ N·m} \]
\[ 35100858.6 \text{ lbf·in} \]

Flange stresses - operating condition

Corrected inside diameter of the flange

\[ B_1' = B' + g_1' = 725.00 \text{ mm} \]
\[ 28.543 \text{ in} \]

Flange hub stress - operating condition

\[ S_H = \frac{f M_s}{L_g B_1} = 132.39 \text{ MPa} \]
\[ 19201.1 \text{ psi} \]

Flange radial stress - operating condition

\[ S_R = (\frac{133e + 1}{M_s}) M_s = 33.71 \text{ MPa} \]
\[ 4889.9 \text{ psi} \]

Flange tangential stress - operating condition

\[ S_T = \frac{Y M_s^2}{P B} - Z S_R = 116.37 \text{ MPa} \]
\[ 16877.4 \text{ psi} \]

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

Flange stresses - seating condition

Flange hub stress - gasket seating condition

\[ S_H = \frac{f M_s}{L_g B_1} = 132.00 \text{ MPa} \]
\[ 19144.4 \text{ psi} \]

Flange radial stress - gasket seating condition

\[ S_R = (\frac{133e + 1}{M_s}) M_s = 33.61 \text{ MPa} \]
\[ 4875.4 \text{ psi} \]

Flange tangential stress - gasket seating condition

\[ S_T = \frac{Y M_s^2}{P B} - Z S_R = 116.02 \text{ MPa} \]
\[ 16827.6 \text{ psi} \]

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

Flange rigidity - operating condition

Rigidity index factor

\[ K_R = 0.30 \]

Flange rigidity index

\[ J = \frac{5214VM_s}{LE_y g_8 k_R} = 0.25185 \]

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

Flange rigidity - seating condition

Flange rigidity index

\[ J = \frac{5214VM_s}{LE_y g_8 k_R} = 0.21440 \]

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

Hub thickness

Minimum hub thickness as cylindrical shell

\[ t_h_{min} = 50.16 \text{ mm} \]
\[ 1.975 \text{ in} \]

\[ g_0 \geq t_h_{min} : \text{Ok} \]
**Maximum allowable pressures (at the top of the vessel)**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold (flange)</td>
<td>= 21.94 MPa</td>
<td>= 3181.9 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded (flange)</td>
<td>= 18.94 MPa</td>
<td>= 2746.7 psi</td>
</tr>
<tr>
<td>New &amp; cold (bolts)</td>
<td>= 21.72 MPa</td>
<td>= 3149.8 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded (bolts)</td>
<td>= 20.68 MPa</td>
<td>= 2999.8 psi</td>
</tr>
</tbody>
</table>
# 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=148.00 MPa / 21 465.6 psi</td>
<td>Sno=148.00 MPa / 21 465.6 psi</td>
<td>Sbo=152.00 MPa / 22 045.7 psi</td>
</tr>
<tr>
<td><strong>Seating condition</strong></td>
<td>Sfg=148.00 MPa / 21 465.6 psi</td>
<td>Sng=148.00 MPa / 21 465.6 psi</td>
<td>Sbg=152.00 MPa / 22 045.7 psi</td>
</tr>
</tbody>
</table>

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

## Bolt loads

- **Number of bolts** \( n \) = 20
- **Bolt type** = ANSI_TEMA 3-3/4"
- **Bolt spacing** \( Bs \) = 193.99 mm / 7.638 in
- **Nominal bolt diameter** \( a \) = 95.25 mm / 3.750 in

### Design bolt load for the operating condition

\[
W_o = \frac{0.785G^2P + 2b\pi GmP}{S_{bo}}
\]

- **Design bolt load for the gasket seating (Bolt)** \( W_{bg} = \pi bGy \) = 3 255 838 N / 731 941.41 lbf
- **External tensile net-section axial force** \( F_A \) = 0 N / 0 lbf
- **Absolute value of the external net-section bending moment** \( M_E \) = 0 N m / 0 lbf in
- **External tensile net-section axial force (test condition)** \( F_{At} \) = 0 N / 0 lbf
- **External tensile net-section bending moment (test condition)** \( M_{Et} \) = 0 N m / 0 lbf in

### Total minimum required cross-sectional area of the bolts

\[
A_{in} = \max \left( \frac{W_o + F_A + 4W_{bg}}{S_{bo}}, \frac{W_{bg}}{S_{bg}} \right)
\]

- **Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion** \( Ab \) = 130 425.5 mm² / 202.160 in²

### Maximum bolts area for gasket crush

\[
A_{bg,\text{max}} = \frac{2\pi \cdot y \cdot G \cdot N}{S_{bg}}
\]

- **Design bolt load for the gasket seating (Flange)**

\[
W_g = \left( \frac{A_{in} + A_{b}}{2} \right) S_{bg}
\]

### Total minimum required cross-sectional area of the bolts

\[
A_{in} = \max \left( \frac{W_o + F_A + 4W_{bg}}{S_{bo}}, \frac{W_{bg}}{S_{bg}} \right)
\]

- **Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion** \( Ab \) = 130 425.5 mm² / 202.160 in²
**Flange constants**

Ratio of the flange outside diameter to the inside diameter
\[ K = \frac{A}{B'} = 2.23622 \]

Stress factor \( Y \)
\[ Y = \frac{1}{K - 1} \left[ \frac{K(1 + 855246\log K)}{K^2 - 1} \right] = 2.56106 \]

Stress factor \( T \)
\[ T = \frac{104720 + 19448K^2}{K^2(1 + 855246\log K) - 1} = 1.42288 \]

Stress factor \( U \)
\[ U = \frac{136136(K^2 - 1)(K - 1)}{K^2 + 1} = 2.81435 \]

Stress factor \( Z \)
\[ Z = \frac{1}{(K^2 - 1)} = 1.49991 \]

Hub length parameter
\[ h_o = \sqrt{B_9} = 219.97 \text{ mm} \quad 8.660 \text{ in} \]

Hub thickness ratio
\[ X_o = \frac{g_5}{g_9} = 1.18110 \]

Hub length ratio
\[ X_h = \frac{h_e}{h_o} = 0.52280 \]

Flange stress factor for integral type flanges
\( F \) (fig. 2-7.2)
\[ F = 0.87835 \]

Flange stress factor for integral type flanges
\( V \) (fig. 2-7.3)
\[ V = 0.42757 \]

Hub stress correction factor for integral flanges
\( f \) (fig. 2-7.6)
\[ f = 1.00000 \]

Stress factor \( e \)
\[ e = \frac{L}{d} = 0.00399 \]

Stress factor \( d \)
\[ d = \frac{Ug_9h_e}{T} = 8407169.84802 \]

Stress factor \( L \)
\[ L = \frac{Te + 1}{T} = 5.11628 \]

Gasket load for the operating condition
\[ H_G = W_o - H = 40721 \text{ N} \quad 9154.50 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange
\[ H_D = 0.78547F = 32603 \text{ N} \quad 7329.39 \text{ lbf} \]

Total hydrostatic end force
\[ H = 0.78547F = 53302 \text{ N} \quad 11982.81 \text{ lbf} \]

Difference
\[ H_T = H - H_G = 20699 \text{ N} \quad 4653.41 \text{ lbf} \]

Moment arm for load HD
\[ h_D = \frac{C - B - g_4}{2} = 255.00 \text{ mm} \quad 10.039 \text{ in} \]

Moment arm for load HG
\[ h_G = \frac{C - G}{2} = 211.53 \text{ mm} \quad 8.328 \text{ in} \]

Moment arm for load HT
\[ h_T = \frac{1}{2} \left[ \frac{C - B}{2} + h_3 \right] = 255.77 \text{ mm} \quad 10.070 \text{ in} \]

**Flange moments**

Nominal bolt diameter
\( d_B = 95.25 \text{ mm} \quad 3.750 \text{ in} \)

TEMA Load concentration factor
\[ c_F = \text{MAX} \left[ \sqrt{\frac{n_d^2}{2d_B^2} + \frac{n_e^2}{n_{e5}}}, 1 \right] = 1.00000 \]

Moment factor used to design split rings
\( F_s = 1.00 \)

Flange design moment for the operating condition
\[ M_o = \text{abs} \left[ (H_D(h_D - h_G) + H_T(h_T - h_3))F_s \right] = 2332.7 \text{ N·m} \quad 20064.6 \text{ lbf·in} \]

Flange design moment for the gasket seating condition
\[ M_s = W_s g_9 F_s = 2441164.2 \text{ N·m} \quad 2160612.4 \text{ lbf·in} \]
Flange stresses - operating condition

Corrected inside diameter of the flange

\[ B_1' = B' + g_1' \]

\[ S_H = \frac{fM_s}{Lg_2B_1} \]

\[ S_R = \frac{(133e+1)M_s}{R^2} \]

\[ S_T = \frac{YM_s}{R^2} - ZS_R \]

Flange hub stress - operating condition

\[ S_H = \frac{0.08}{0.07} \text{ MPa} \]

Flange radial stress - operating condition

\[ S_R = \frac{0.02}{0.07} \text{ MPa} \]

Flange tangential stress - operating condition

\[ S_T = \frac{0.07}{0.07} \text{ MPa} \]

\[ \begin{align*}
SHo & \leq \min[1.5Sfo, 2.5Sno] : \text{ Ok} \\
SRo & \leq Sfo : \text{ Ok} \\
STo & \leq Sfo : \text{ Ok} \\
(\text{SHo} + \text{SRo}) / 2 & \leq \text{Sfo} : \text{ Ok} \\
(\text{SHo} + \text{STo}) / 2 & \leq \text{Sfo} : \text{ Ok}
\end{align*} \]

Flange stresses - seating condition

Flange hub stress - gasket seating condition

\[ S_H = \frac{fM_s}{Lg_2B_1} \]

\[ S_R = \frac{(133e+1)M_s}{R^2} \]

Flange tangential stress - gasket seating condition

\[ S_T = \frac{YM_s}{R^2} - ZS_R \]

\[ \begin{align*}
SHg & \leq \min[1.5Sfg, 2.5Sng] : \text{ Ok} \\
SRg & \leq Sfg : \text{ Ok} \\
STg & \leq Sfg : \text{ Ok} \\
(\text{SHg} + \text{SRg}) / 2 & \leq \text{Sfg} : \text{ Ok} \\
(\text{SHg} + \text{STg}) / 2 & \leq \text{Sfg} : \text{ Ok}
\end{align*} \]

Flange rigidity - operating condition

Rigidity index factor

\[ KR = 0.30 \]

Flange rigidity index

\[ J = \frac{5214VM_s}{LEg^2K^2h_0} = 0.00013 \]

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

Flange rigidity - seating condition

Flange rigidity index

\[ J = \frac{5214VM_s}{LEg^2K^2h_0} = 0.13197 \]

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

### Internal pressure

**Welding neck flange - Flange Out Tube Side 26° (Bolting)**

- **Material** = SA-193 B16
- **Governing Thickness** = 95.25 mm (3.750 in)
- **Reduction in MDMT based on available excess thickness** = 80.00 °C (176.00 °F)
- **Unadjusted MDMT from table UCS-66** = -30.00 °C (-22.00 °F)
- **Adjusted MDMT from fig. UCS-66.1** = -110.00 °C (-166.00 °F)

### Welding neck flange - Flange Out Tube Side 26°, Flange

- **Material** = SA-182 F22 3
- **Curve of fig. UCS-66** = B
- **Governing Thickness** = 76.20 mm (3.000 in)
- **Reduction in MDMT based on available excess thickness** = 1.47 °C (34.65 °F)
- **Design pressure** = 18.44 MPa (2674.0 psi)
- **Maximum allowable working pressure** = 18.94 MPa (2746.7 psi)
- **Coincident ratio** = 0.97351
- **Adjusted MDMT from fig. UCS-66.1** = 24.53 °C (76.15 °F)

### External pressure

**Welding neck flange - Flange Out Tube Side 26° (Bolting)**

- **Material** = SA-193 B16
- **Governing Thickness** = 95.25 mm (3.750 in)
- **Reduction in MDMT based on available excess thickness** = 80.00 °C (176.00 °F)
- **Unadjusted MDMT from table UCS-66** = -30.00 °C (-22.00 °F)
- **Adjusted MDMT from fig. UCS-66.1** = -110.00 °C (-166.00 °F)

**Welding neck flange - Flange Out Tube Side 26°, Flange**

- **Material** = SA-182 F22 3
- **Curve of fig. UCS-66** = B
- **Governing Thickness** = 76.20 mm (3.000 in)
- **Reduction in MDMT based on available excess thickness** = 1.47 °C (34.65 °F)
- **Design pressure** = 0.10 MPa (14.9 psi)
- **Maximum allowable working pressure** = 18.94 MPa (2746.7 psi)
- **Coincident ratio** = 0.00544
- **Adjusted MDMT from fig. UCS-66.1** = -54.00 °C (-65.20 °F)
**Welding neck flange - Shell Flange**

*According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - Metric Units*

### Design data

#### Internal design temperature

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

#### Internal design pressure

\[ P = 14.22 \, \text{MPa} = 2062.4 \, \text{psi} \]

#### 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-387 22 2 - Plate

### Shell material

SA-387 22 2 - Plate

### Bolting material

SA-193 B16 - 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>Sfo=133.60 MPa</td>
<td>Sno=133.60 MPa</td>
<td>Sbo=138.00 MPa</td>
</tr>
<tr>
<td></td>
<td>/ 19 377.0 psi</td>
<td>/ 19 377.0 psi</td>
<td>/ 20 015.2 psi</td>
</tr>
<tr>
<td>Seating condition</td>
<td>Sfg=148.00 MPa</td>
<td>Sng=148.00 MPa</td>
<td>Sbg=138.00 MPa</td>
</tr>
<tr>
<td></td>
<td>/ 21 465.6 psi</td>
<td>/ 21 465.6 psi</td>
<td>/ 20 015.2 psi</td>
</tr>
</tbody>
</table>

- **Internal pressure**
  \[ P_d = 14.22 \, \text{MPa} = 2062.4 \, \text{psi} \]
- **Overpressure due to static head**
  \[ P_h = 0 \, \text{MPa} = 0 \, \text{psi} \]
- **Calculation pressure**
  \[ P = 14.22 \, \text{MPa} = 2062.4 \, \text{psi} \]
- **Modulus of Elasticity at the operating load case temperature**
  \[ E_{yo} = 182 400.00 \, \text{MPa} = 26 454 883.4 \, \text{psi} \]
- **Modulus of Elasticity at the gasket seating load case temperature**
  \[ E_{yg} = 210 350.00 \, \text{MPa} = 30 508 688.2 \, \text{psi} \]
- **Corrosion allowance**
  \[ c = 3.00 \, \text{mm} = 0.118 \, \text{in} \]
- **Flange external diameter**
  \[ A = 2010.00 \, \text{mm} = 79.134 \, \text{in} \]
- **Inside diameter**
  \[ B = 1275.00 \, \text{mm} = 50.197 \, \text{in} \]
- **Bolt circle**
  \[ C = 1810.00 \, \text{mm} = 71.260 \, \text{in} \]
- **Flange thickness**
  \[ t = 275.00 \, \text{mm} = 10.827 \, \text{in} \]
- **Mean gasket diameter**
  \[ G_{mean} = 1352.85 \, \text{mm} = 53.262 \, \text{in} \]
- **Thickness of the hub at the small end**
  \[ g_0 = 76.00 \, \text{mm} = 2.992 \, \text{in} \]
- **Thickness of the hub at the large end**
  \[ g_1 = 133.00 \, \text{mm} = 5.236 \, \text{in} \]
- **Thickness of the hub at the small end (Corroded)**
  \[ g_0' = g_0 - c = 73.00 \, \text{mm} = 2.874 \, \text{in} \]
- **Thickness of the hub at the large end (Corroded)**
  \[ g_1' = g_1 - c = 130.00 \, \text{mm} = 5.118 \, \text{in} \]
- **Hub length**
  \[ h = 155.00 \, \text{mm} = 6.102 \, \text{in} \]

### Gasket parameters

- **Gasket factor**
  \[ m = 4.25 \]
- **Gasket contact width**
  \[ N = 77.85 \, \text{mm} = 3.065 \, \text{in} \]
- **Basic gasket seating width**
  \[ \frac{b_0 - N}{2} = 38.93 \, \text{mm} = 1.532 \, \text{in} \]
- **Conversion factor for length**
  \[ C_b = 2.50000 \]
- **Effective gasket contact width**
  \[ b = C_b \sqrt{b_0} = 15.60 \, \text{mm} = 0.614 \, \text{in} \]
- **Outside diameter of the gasket contact area**
  \[ G_c = G_{mean} + N = 1430.70 \, \text{mm} = 56.327 \, \text{in} \]
- **Diameter at the location of the gasket load reaction**
  \[ G = G_c - 2b = 1399.51 \, \text{mm} = 55.099 \, \text{in} \]
Bolt loads

Number of bolts \( n = 24 \)

Bolt spacing \( B_s = 236.93 \text{ mm} = 9.328 \text{ in} \)

Nominal bolt diameter \( a = 114.30 \text{ mm} = 4.500 \text{ in} \)

Maximum bolt spacing \( B_{s\text{max}} = 2a + \frac{6r}{m + 0.5} = 575.97 \text{ mm} = 22.676 \text{ in} \)

Design bolt load for the operating condition \( W_o = 0.785G^2P + 2\pi G m G \)

Design bolt load for the gasket seating (Bolt) \( W_{gB} = \pi G Y \)

External tensile net-section axial force \( F_A = 0 \text{ N} = 0 \text{ lbf} \)

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

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

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

Total minimum required cross-sectional area of the bolts \( A_{n min} = \max \left( \frac{W_o + F_A + \frac{4M_A}{L}}{S_{bb}}, \frac{W_{gB}}{S_{bb}} \right) = 218489.7 \text{ mm}^2 = 338.660 \text{ in}^2 \)

Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion \( A_{b\text{max}} = \frac{2\pi \cdot \gamma \cdot G \cdot N}{S_{bb}} = 347241.7 \text{ mm}^2 = 538.226 \text{ in}^2 \)

Maximum bolts area for gasket crush \( A_{bb} = \frac{A_{nmax} + A_{b}}{2} S_{bb} = 30535316 \text{ N} = 6864611.62 \text{ lbf} \)

Design bolt load for the gasket seating (Flange) \( W_{gF} = \frac{(A_{nmax} + A_{b})}{2} S_{bb} = 218489.7 \text{ mm}^2 = 338.660 \text{ in}^2 \)

Flange constants

Ratio of the flange outside diameter to the inside diameter \( K = A/B' = 1.56909 \)

Stress factor Y \( Y = K - 1 \) \( F = 1.00000 \)

Stress factor T \( T = \left( \frac{1 + 855246\log(K)}{K^{2} - 1} \right) \left( \frac{104720 + 19448K^{2}}{K^{2} - 1} \right) \left( \frac{1 + 855246\log(K)}{K^{2} - 1} \right) \) \( = 1.68080 \)

Stress factor U \( U = \left( \frac{136136(K^{2} - 1)}{K^{2} - 1} \right) \left( \frac{1 + 855246\log(K)}{K^{2} - 1} \right) \) \( = 4.92782 \)

Stress factor Z \( Z = \left( \frac{136136(K^{2} - 1)}{K^{2} - 1} \right) \) \( = 2.36796 \)

Hub length parameter \( h_k = \sqrt{\frac{S_{gb}}{E_{gb}}} = 305.80 \text{ mm} = 12.039 \text{ in} \)

Hub thickness ratio \( X_k = \frac{h_k}{h_{gb}} = 0.50687 \)

Flange stress factor for integral type flanges \( F \) \( = 0.83184 \)

Flange stress factor for integral type flanges \( V \) \( = 0.25473 \)

Hub stress correction factor for integral flanges \( f \) \( = 1.00000 \)

Stress factor e \( e = \frac{F}{Y} = 0.00272 \)

Stress factor d \( d = \frac{V f}{Y} = 0.83184 \)

Stress factor L \( L = \frac{e + 1}{d} = 1.69971 \)

Gasket load for the operating condition \( H_G = W_o - H = 8288697 \text{ N} = 1863373.06 \text{ lbf} \)

Total hydrostatic end force on the area inside of the flange \( H_{D} = 0.785G^2P = 18317095 \text{ N} = 4117846.38 \text{ lbf} \)

Total hydrostatic end force \( H = 0.785G^2P = 21862875 \text{ N} = 4914969.42 \text{ lbf} \)

Difference \( H_p = H - H_D = 3545780 \text{ N} = 797123.04 \text{ lbf} \)

Moment arm for load HD \( h_d = \frac{C - B - g_1}{2} = 199.50 \text{ mm} = 7.854 \text{ in} \)
Moment arm for load HG
\[ h_G = \frac{C - G}{2} = 205.25 \text{ mm} \]
\[ 8.081 \text{ in} \]

Moment arm for load HT
\[ h_T = \frac{1}{2} \left( \frac{C - B}{2} + h_G \right) = 234.87 \text{ mm} \]
\[ 9.247 \text{ in} \]

Flange moments
Nominal bolt diameter
\[ d_B = 114.30 \text{ mm} \]
\[ 4.500 \text{ in} \]

TEMA Load concentration factor
\[ c_F = \text{MAX} \left[ \frac{B}{2d_B + \frac{G}{0.85}}, 1 \right] = 1.00000 \]

Bolt spacing correction factor
\[ B_{ns} = \text{max} \left[ \frac{B}{2a + t}; 1 \right] = 1.00000 \]

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

Flange design moment for the operating condition
\[ M_o = c_F \abs \left( H_2 h_2 + H_T h_T + H_{12} h_{12} \right) B_{ns} F_S = 6188 \text{ 305.2 N·m} \]
\[ 6 \text{ 267 296.6 lbf·in} \]

Flange design moment for the gasket seating condition
\[ M_o = c_F \frac{W_o (C - G) B_{ns} F_S}{2} = 6267 \text{ 296.6 N·m} \]
\[ 55 \text{ 470 243.8 lbf·in} \]

Flange stresses - operating condition
Corrected inside diameter of the flange
\[ B_1' = B' + g_1' = 1411.00 \text{ mm} \]
\[ 55.551 \text{ in} \]

Flange hub stress - operating condition
\[ S_H = \frac{f M_o}{L_6 B_1'^2} = 152.68 \text{ MPa} \]
\[ 22 \text{ 144.4 psi} \]

Flange radial stress - operating condition
\[ S_R = \frac{1}{2} \frac{M_o}{L_6 B_1'^2} = 74.97 \text{ MPa} \]
\[ 10 \text{ 874.0 psi} \]

Flange tangential stress - operating condition
\[ S_T = \frac{1}{2} \frac{M_o}{L_6 B_1'^2} - Z S_R = 108.92 \text{ MPa} \]
\[ 15 \text{ 797.5 psi} \]

\( SH_o \leq \min\{1.5 S_fo, 2.5 S_no\}: \text{ Ok} \)
\( SR_o \leq S_fo: \text{ Ok} \)
\( ST_o \leq S_fo: \text{ Ok} \)
\( (SH_o + SR_o) / 2 \leq S_fo: \text{ Ok} \)
\( (SH_o + ST_o) / 2 \leq S_fo: \text{ Ok} \)

Flange stresses - seating condition
Flange hub stress - gasket seating condition
\[ S_H = \frac{f M_o}{L_6 B_1'^2} = 154.63 \text{ MPa} \]
\[ 22 \text{ 427.0 psi} \]

Flange radial stress - gasket seating condition
\[ S_R = \frac{1}{2} \frac{M_o}{L_6 B_1'^2} = 75.93 \text{ MPa} \]
\[ 11 \text{ 012.8 psi} \]

Flange tangential stress - gasket seating condition
\[ S_T = \frac{1}{2} \frac{M_o}{L_6 B_1'^2} - Z S_R = 110.31 \text{ MPa} \]
\[ 15 \text{ 999.1 psi} \]

\( SH_g \leq \min\{1.5 S_fg, 2.5 S_ng\}: \text{ Ok} \)
\( SR_g \leq S_fg: \text{ Ok} \)
\( ST_g \leq S_fg: \text{ Ok} \)
\( (SH_g + SR_g) / 2 \leq S_fg: \text{ Ok} \)
\( (SH_g + ST_g) / 2 \leq S_fg: \text{ Ok} \)

Flange rigidity - operating condition
Rigidity index factor
\[ K_R = 0.30 \]

Flange rigidity index
\[ J = \frac{5214 V M_o}{L E y x^2 K f o} = 0.54228 \]
\[ Jo \leq 1: \text{ Ok} \]

Flange rigidity - seating condition
Flange rigidity index
\[ J = \frac{5214 V M_o}{L E y x^2 K f o} = 0.47623 \]
\[ Jg \leq 1: \text{ Ok} \]
### Hub thickness

- **Minimum hub thickness as cylindrical shell**
  - $th_{min} = 75.82 \text{ mm} = 2.985 \text{ in}$
  - $g \geq th_{min}: \text{ Ok}$

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

- **New & cold (flange)**
  - $= 16.41 \text{ MPa} = 2380.4 \text{ psi}$
- **Hot & corroded (flange)**
  - $= 14.52 \text{ MPa} = 2106.5 \text{ psi}$
- **New & cold (bolts)**
  - $= 14.58 \text{ MPa} = 2114.9 \text{ psi}$
- **Hot & corroded (bolts)**
  - $= 14.58 \text{ MPa} = 2114.9 \text{ psi}$
### 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>$S_{fo}=148.00 \text{ MPa} / 21,465.6 \text{ psi}$</td>
<td>$S_{no}=148.00 \text{ MPa} / 21,465.6 \text{ psi}$</td>
<td>$S_{bo}=138.00 \text{ MPa} / 20,015.2 \text{ psi}$</td>
</tr>
<tr>
<td><strong>Seating condition</strong></td>
<td>$S_{fg}=148.00 \text{ MPa} / 21,465.6 \text{ psi}$</td>
<td>$S_{ng}=148.00 \text{ MPa} / 21,465.6 \text{ psi}$</td>
<td>$S_{bg}=138.00 \text{ MPa} / 20,015.2 \text{ psi}$</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**
- $T_e = 20.00 \degree C / 68.00 \degree F$

**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** $n = 24$
- **Bolt type** = ANSI_TEMA 4-1/2"
- **Bolt spacing** $B_s = 236.93 \text{ mm} / 9.328 \text{ in}$
- **Nominal bolt diameter** $a = 114.30 \text{ mm} / 4.500 \text{ in}$

**Maximum bolt spacing**
- $B_{s\text{max}} = 2a + \frac{6t}{n} + 0.5 = 575.97 \text{ mm} / 22.676 \text{ in}$

**Design bolt load for the operating condition**
- $W_o = 0.785G^2P + 2b\pi GmF = 218\,403 \text{ N} / 49\,098.93 \text{ lbf}$

**Design bolt load for the gasket seating (Bolt)**
- $W_{yo} = \pi b G y = 4\,800\,392 \text{ N} / 1\,079\,170.95 \text{ lbf}$

**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)**
- $F_A = 0 \text{ N} / 0 \text{ lbf}$

**External tensile net-section bending moment (test condition)**
- $M_{E\text{t}} = 0 \text{ N.m} / 0 \text{ lbf.in}$

**Total minimum required cross-sectional area of the bolts**
- $A_m = \max \left( \frac{W_o + F_A + \frac{4\pi y G^2}{S_{bo}}}{\frac{W_o}{S_{bo}}}, \frac{W_{yo}}{S_{bo}} \right) = 34\,785.4 \text{ mm}^2 / 53.918 \text{ in}^2$

**Cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion**
- $A_b = 224\,051.2 \text{ mm}^2 / 347.280 \text{ in}^2$

**Maximum bolts area for gasket crush**
- $A_{b\text{max}} = 2\pi y G N \sqrt{S_{bg}} = 347\,241.7 \text{ mm}^2 / 538.226 \text{ in}^2$

**Design bolt load for the gasket seating (Flange)**
- $W_{g} = \left( \frac{A_m + A_b}{2} \right) S_{bg} = 17\,859\,726 \text{ N} / 4\,015\,025.86 \text{ lbf}$

 Bs <= BsMax: Ok
**Flange constants**

Ratio of the flange outside diameter to the inside diameter

\[ K = \frac{A}{B'} = 1.56909 \]

Stress factor \( Y \)

\[ Y = 0.66845 + 571690 \left( \frac{K^2 \log \frac{A}{B'}}{K^2 - 1} \right) = 4.48433 \]

Stress factor \( T \)

\[ T = \frac{104720 + 194.48K^2}{K^2(1 + 855246 \log \frac{A}{B'}) - 1} = 1.68080 \]

Stress factor \( U \)

\[ U = \frac{136136(K^2 - 1)(K - 1)}{K^2 + 1} = 4.92782 \]

Stress factor \( Z \)

\[ Z = \frac{36136(K^2 - 1)}{(K^2 - 1) + 1} = 2.36796 \]

Hub length parameter

\[ h_o = \sqrt{\frac{B_o}{2}} = 305.80 \text{ mm} \quad 12.039 \text{ in} \]

Hub thickness ratio

\[ X_o = \frac{g_o}{2h_o} = 1.78082 \]

Hub length ratio

\[ X_k = \frac{h_k}{h_o} = 0.50687 \]

Flange stress factor for integral type flanges

\( F \) (fig. 2-7.2)

\[ F = 0.83184 \]

Flange stress factor for integral type flanges

\( V \) (fig. 2-7.3)

\[ V = 0.25473 \]

Hub stress correction factor for integral flanges

\( f \) (fig. 2-7.6)

\[ f = 1.00000 \]

Stress factor \( e \)

\[ e = \frac{L}{P} = 0.00272 \]

Stress factor \( d \)

\[ d = \frac{Ug_2h_c}{P} = 31.524 \text{ 790.28339} \]

Stress factor \( L \)

\[ L = \frac{L_e + 1}{T} \cdot \frac{P}{d} = 1.69971 \]

Gasket load for the operating condition

\[ H_G = W_o - H = 60039 \text{ N} \quad 13497.35 \text{ lbf} \]

Total hydrostatic end force on the area inside of the flange

\[ H_D = 0.0785C^2P = 132680 \text{ N} \quad 29827.63 \text{ lbf} \]

Total hydrostatic end force

\[ H = 0.0785C^2P = 158364 \text{ N} \quad 35601.59 \text{ lbf} \]

Difference

\[ H_B = H - H_G = 25684 \text{ N} \quad 5773.96 \text{ lbf} \]

Moment arm for load HD

\[ h_D = \frac{C - B - G}{2} = 199.50 \text{ mm} \quad 7.854 \text{ in} \]

Moment arm for load HG

\[ h_G = \frac{C - G}{2} = 205.25 \text{ mm} \quad 8.081 \text{ in} \]

Moment arm for load HT

\[ h_T = \frac{1}{2} \left( \frac{C - B}{2} + h_o \right) = 234.87 \text{ mm} \quad 9.247 \text{ in} \]

**Flange moments**

Nominal bolt diameter

\[ d = 114.30 \text{ mm} \quad 4.500 \text{ in} \]

TEMA Load concentration factor

\[ C_P = \text{MAX} \left[ \frac{n^2}{d^2} + \frac{4}{a + b} \right] = 1.00000 \]

Bolt spacing correction factor

\[ B_s = \text{MAX} \left[ \frac{1}{2d^2 + 1} \right] = 1.00000 \]

Moment factor used to design split rings

\( F_s = 1.00 \)

Flange design moment for the operating condition

\[ M_o = \text{ABS} \left[ \left( H_D(h_D - h_o) + H_T(h_T - h_o) \right) F_s \right] = 1.7 \text{ Nm} \quad 14.7 \text{ lbf-in} \]

Flange design moment for the gasket seating condition

\[ M_g = W_g h_F F_s = 3665663.8 \text{ Nm} \quad 32443854.9 \text{ lbf-in} \]
Flange stresses - operating condition

Corrected inside diameter of the flange

\[ B1' = B' + g1' = \]

1 411.00 mm 55.551 in

Flange hub stress - operating condition

\[ S_H = \frac{fM_o}{Lg2B1} = \]

0.00004 MPa 0.006 psi

Flange radial stress - operating condition

\[ S_R = \frac{(133e+1)M_o}{r^2B} = \]

0.00002 MPa 0.003 psi

Flange tangential stress - operating condition

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

0.00003 MPa 0.004 psi

\[ SHo \leq \min[1.5Sfo, 2.5Sno]: Ok \]

\[ SRo \leq Sfo: Ok \]

\[ STo \leq Sfo: Ok \]

\[ \frac{SHo + SRo}{2} \leq Sfo: Ok \]

\[ \frac{SHo + STo}{2} \leq Sfo: Ok \]

Flange stresses - seating condition

Flange hub stress - gasket seating condition

\[ S_H = \frac{fM_o}{Lg2B1} = \]

90.44 MPa 13 117.3 psi

Flange radial stress - gasket seating condition

\[ S_R = \frac{(133e+1)M_o}{r^2B} = \]

44.41 MPa 6 441.2 psi

Flange tangential stress - gasket seating condition

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

64.52 MPa 9 357.7 psi

\[ SHg \leq \min[1.5Sfg, 2.5Sng]: Ok \]

\[ SRg \leq Sfg: Ok \]

\[ STg \leq Sfg: Ok \]

\[ \frac{SHg + SRg}{2} \leq Sfg: Ok \]

\[ \frac{SHg + STg}{2} \leq Sfg: Ok \]

Flange rigidity - operating condition

Rigidity index factor

\[ KR = 0.30 \]

Flange rigidity index

\[ J = \frac{5214VM_o}{LEg2g_{K}h_{o}} = \]

0.00000

\[ Jo \leq 1: Ok \]

Flange rigidity - seating condition

Flange rigidity index

\[ J = \frac{5214VM_o}{LEg2g_{K}h_{o}} = \]

0.27854

\[ Jg \leq 1: Ok \]
Minimum Design Metal Temperature (MDMT)

**Internal pressure**

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

| Material | = | SA-193 B16 |
| Governing Thickness | = | 114.30 mm | 4.500 in |
| Reduction in MDMT based on available excess thickness | TR | 80.00 °C | 176.00 °F |
| Unadjusted MDMT from table UCS-66 | = | -30.00 °C | -22.00 °F |
| Adjusted MDMT from fig. UCS-66.1 | = | -110.00 °C | -166.00 °F |

**Welding neck flange - Shell Flange, Flange**

| Material | = | SA-387 22 2 |
| Governing Thickness | = | 76.00 mm | 2.992 in |
| Reduction in MDMT based on available excess thickness | TR | 1.16 °C | 34.10 °F |
| Design pressure | P | 14.22 MPa | 2062.4 psi |
| Maximum allowable working pressure | MAWP H&C | 14.52 MPa | 2106.5 psi |
| Coincident ratio | | P | MAWP H & C | = | 0.97904 |
| Adjusted MDMT from fig. UCS-66.1 | = | 41.84 °C | 107.30 °F |

**External pressure**

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

| Material | = | SA-193 B16 |
| Governing Thickness | = | 114.30 mm | 4.500 in |
| Reduction in MDMT based on available excess thickness | TR | 80.00 °C | 176.00 °F |
| Unadjusted MDMT from table UCS-66 | = | -30.00 °C | -22.00 °F |
| Adjusted MDMT from fig. UCS-66.1 | = | -110.00 °C | -166.00 °F |

**Welding neck flange - Shell Flange, Flange**

| Material | = | SA-387 22 2 |
| Governing Thickness | = | 76.00 mm | 2.992 in |
| Reduction in MDMT based on available excess thickness | TR | 1.16 °C | 34.10 °F |
| Design pressure | P | 0.10 MPa | 14.9 psi |
| Maximum allowable working pressure | MAWP H&C | 14.52 MPa | 2106.5 psi |
| Coincident ratio | | P | MAWP H & C | = | 0.00709 |
| Adjusted MDMT from fig. UCS-66.1 | = | -37.00 °C | -34.60 °F |
Cylindrical shell - Main shell

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28 - Metric Units

Design data

- **Internal design temperature**: $T = 420.00 \, ^\circ\text{C} = 788.00 \, ^\circ\text{F}$
- **Internal design pressure**: $P = 14.22 \, \text{MPa} = 2062.4 \, \text{psi}$
- **External design temperature**: $Te = 20.00 \, ^\circ\text{C} = 68.00 \, ^\circ\text{F}$
- **External design pressure**: $Pe = 0.10 \, \text{MPa} = 14.9 \, \text{psi}$
- **Joint efficiency**: $E = 1.00$

**Material: SA-387 22 2 - Plate**

- **Allowable stress**: $S = 133.60 \, \text{MPa} = 19377.0 \, \text{psi}$
- **Allowable stress at room temperature**: $ST = 148.00 \, \text{MPa} = 21465.6 \, \text{psi}$

**Geometry**

- **Inside diameter**: $D = 1,275.00 \, \text{mm} = 50.197 \, \text{in}$
- **Outside diameter**: $Do = 1,427.00 \, \text{mm} = 56.181 \, \text{in}$
- **Length**: $L = 3,594.00 \, \text{mm} = 141.496 \, \text{in}$
- **Adopted thickness**: $t = 76.00 \, \text{mm} = 2.992 \, \text{in}$
- **Corrosion allowance**: $c = 3.00 \, \text{mm} = 0.118 \, \text{in}$
- **External corrosion allowance**: $ce = 0 \, \text{mm} = 0 \, \text{in}$
- **Wall undertolerance**: $c' = 0 \, \text{mm} = 0 \, \text{in}$

**Forming strain (cylinders formed from plate)**

\[ \varepsilon_f = \frac{50 \cdot t}{R + \frac{t}{2}} = 5.63\% \]

**Ligament Efficiency**

- **Reference figure**: None
- **Diameter of tube holes**: $d = 0 \, \text{mm} = 0 \, \text{in}$

**Internal pressure**

- **Allowable stress**: $S = 133.60 \, \text{MPa} = 19377.0 \, \text{psi}$
- **Internal pressure**: $Pi = 14.22 \, \text{MPa} = 2062.4 \, \text{psi}$
- **Overpressure due to static head**: $Ph = 0 \, \text{MPa} = 0 \, \text{psi}$
- **Calculation pressure**: $P = Pi + Ph = 14.22 \, \text{MPa} = 2062.4 \, \text{psi}$
- **Reference diameter**: Inside
- **Calculation radius (inside)**: $R = 637.50 \, \text{mm} = 25.098 \, \text{in}$

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

- **New & cold**: $16.47 \, \text{MPa} = 2388.2 \, \text{psi}$
- **Hot & corroded**: $14.25 \, \text{MPa} = 2067.1 \, \text{psi}$
<table>
<thead>
<tr>
<th><strong>External pressure</strong></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>External design temperature</td>
<td>$T_e = 20.00 ^\circ C \quad 68.00 ^\circ F$</td>
</tr>
<tr>
<td>External pressure</td>
<td>$P_e = 0.10 \text{ MPa} \quad 14.9 \text{ psi}$</td>
</tr>
<tr>
<td>External static head</td>
<td>$P_h = 0 \text{ MPa} \quad 0 \text{ psi}$</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>$P = P_e + P_h = 0.10 \text{ MPa} \quad 14.9 \text{ psi}$</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>$D_o = 1427.00 \text{ mm} \quad 56.181 \text{ in}$</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>$E = 200 \text{0} \text{0} \text{0} \text{.00} \text{ MPa} \quad 29007 \text{547.6 psi}$</td>
</tr>
<tr>
<td>Axial length between reinforcements</td>
<td>$L = 3771.04 \text{ mm} \quad 148.466 \text{ in}$</td>
</tr>
<tr>
<td>$L / D_o$ ratio</td>
<td>$= 2.64263$</td>
</tr>
<tr>
<td>$D_o / t$ ratio</td>
<td>$= 19.54795$</td>
</tr>
<tr>
<td>Factor A</td>
<td>$= 0.00535$</td>
</tr>
<tr>
<td>Factor B</td>
<td>$= 133.56 \text{ MPa} \quad 19371.3 \text{ psi}$</td>
</tr>
<tr>
<td>External pressure</td>
<td>$P_a = \frac{4B}{3[D_o/((t-c_e)-c^2)]} = 9.11 \text{ MPa} \quad 1321.3 \text{ psi}$</td>
</tr>
<tr>
<td>Required thickness</td>
<td>$t = 9.88 \text{ mm} \quad 0.389 \text{ in}$</td>
</tr>
</tbody>
</table>

$P_a \geq P: \text{ Ok}$
$t \geq t_r: \text{ Ok}$

<table>
<thead>
<tr>
<th><strong>Maximum allowable external pressures</strong></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>$= 9.51 \text{ MPa} \quad 1379.8 \text{ psi}$</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>$= 9.11 \text{ MPa} \quad 1321.3 \text{ psi}$</td>
</tr>
</tbody>
</table>
## Minimum Design Metal Temperature (MDMT)

### Internal pressure

**Cylindrical shell - Main shell**

- **Material**:
  - SA-387 22 2

- **Curve of fig. UCS-66**:
  - A

- **Governing Thickness**:
  - 76.00 mm (2.992 in)

- **Reduction in MDMT based on available excess thickness**:
  - TR = 0.14 °C (32.24 °F)

- **Unadjusted MDMT from table UCS-66**:
  - 43.00 °C (109.40 °F)

- **Joint efficiency**:
  - E = 1.00

- **Corrected joint efficiency**:
  - $E^* = \max(E, 0.8) = 1.00000$

- **Corrosion allowance**:
  - $c = 3.00 \text{ mm} (0.118 \text{ in})$

- **Minimum required thickness in corroded condition**:
  - $Tr = 72.82 \text{ mm} (2.867 \text{ in})$

- **Nominal noncorroded thickness**:
  - $Tn = 76.00 \text{ mm} (2.992 \text{ in})$

- **Coincident ratio**:
  - $t_nE^* / t_n - c = 0.99756$

- **Adjusted MDMT from fig. UCS-66.1**:
  - 42.86 °C (109.16 °F)

### External pressure

**Cylindrical shell - Main shell**

- **Material**:
  - SA-387 22 2

- **Curve of fig. UCS-66**:
  - A

- **Governing Thickness**:
  - 76.00 mm (2.992 in)

- **Reduction in MDMT based on available excess thickness**:
  - TR = 80.00 °C (176.00 °F)

- **Unadjusted MDMT from table UCS-66**:
  - 43.00 °C (109.40 °F)

- **Joint efficiency**:
  - E = 1.00

- **Corrected joint efficiency**:
  - $E^* = \max(E, 0.8) = 1.00000$

- **Corrosion allowance**:
  - $c = 3.00 \text{ mm} (0.118 \text{ in})$

- **Minimum required thickness in corroded condition**:
  - $Tr = 6.88 \text{ mm} (0.271 \text{ in})$

- **Nominal noncorroded thickness**:
  - $Tn = 76.00 \text{ mm} (2.992 \text{ in})$

- **Coincident ratio**:
  - $t_nE^* / t_n - c = 0.09425$

- **Adjusted MDMT from fig. UCS-66.1**:
  - -37.00 °C (-34.60 °F)

### 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]
### Design data

- **Internal design temperature**
  \[ T = 420.00 \, ^\circ\text{C} \quad 788.00 \, ^\circ\text{F} \]
- **Internal design pressure**
  \[ P = 14.22 \, \text{MPa} \quad 2062.4 \, \text{psi} \]
- **External design temperature**
  \[ T_e = 20.00 \, ^\circ\text{C} \quad 68.00 \, ^\circ\text{F} \]
- **External design pressure**
  \[ P_e = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \]
- **Joint efficiency**
  \[ E = 1.00 \]

### Nozzle material: SA-182 F22 3 - Forgings

- **Allowable stress in nozzle**
  \[ S_n = 133.60 \, \text{MPa} \quad 19377.0 \, \text{psi} \]

### Shell material: SA-387 22 2 - Plate

- **Allowable stress in vessel**
  \[ S_v = 133.60 \, \text{MPa} \quad 19377.0 \, \text{psi} \]

### Nozzle geometry

- **Outside diameter**
  \[ d_o = 597.00 \, \text{mm} \quad 23.504 \, \text{in} \]
- **Inside diameter**
  \[ d_i = 457.20 \, \text{mm} \quad 18.000 \, \text{in} \]
- **Nozzle thickness**
  \[ t_n = 69.90 \, \text{mm} \quad 2.752 \, \text{in} \]
- **Nozzle reinforcement thickness**
  \[ t_x = 130.00 \, \text{mm} \quad 5.118 \, \text{in} \]
- **Length of projection defining the thickened portion of integral reinforcement**
  \[ L = 200.00 \, \text{mm} \quad 7.874 \, \text{in} \]
- **Tapering angle**
  \[ \alpha = 45.00 ^\circ \]
- **Corner radius**
  \[ r = 10.00 \, \text{mm} \quad 0.394 \, \text{in} \]
- **Nozzle corrosion allowance**
  \[ c_n = 3.00 \, \text{mm} \quad 0.118 \, \text{in} \]
- **Nozzle undertolerance**
  \[ c'n = 0 \, \text{mm} \quad 0 \, \text{in} \]

- **Nozzle connection**
  = Integrimly reinforced
- **Nozzle position**
  = Radial
- **Offset from shell border**
  = 420.00 mm  
  16.535 in
- **Angular offset**
  = 0 °
- **Weld leg length of the outside nozzle fillet weld**
  \[ t_2 = 15.00 \, \text{mm} \quad 0.591 \, \text{in} \]
- **Minimum weld leg length of the outside nozzle fillet weld**
  \[ = 8.49 \, \text{mm} \quad 0.334 \, \text{in} \]

  \[ r \geq \min[1/4t,1/8 \text{ in. (3mm)}]: \text{Ok} \]

### Opening geometry

- **Finished diameter of circular opening**
  \[ d = 463.20 \, \text{mm} \quad 18.236 \, \text{in} \]
- **Finished radius of circular opening**
  \[ R_n = 231.60 \, \text{mm} \quad 9.118 \, \text{in} \]
- **Nearest opening**
  = Out Shell Side
- **Distance to nearest opening**
  = 3,532.33 mm  
  139.068 in
- **Maximum distance before multiple openings calculation occurs**
  = 928.91 mm  
  36.571 in

### Shell Geometry

- **Inside shell diameter**
  \[ D = 1,275.00 \, \text{mm} \quad 50.197 \, \text{in} \]
- **Inside radius of shell course under consideration**
  \[ R = 637.50 \, \text{mm} \quad 25.998 \, \text{in} \]
- **Shell thickness**
  \[ t = 76.00 \, \text{mm} \quad 2.999 \, \text{in} \]
- **Shell corrosion allowance**
  \[ c_s = 3.00 \, \text{mm} \quad 0.118 \, \text{in} \]
- **Shell undertolerance**
  \[ c's = 0 \, \text{mm} \quad 0 \, \text{in} \]
### Internal pressure

**Net thicknesses**
- Net shell thickness: \( t' = t - c_s - c's = 73.00 \text{ mm} = 2.874 \text{ in} \)
- Net required thickness of a seamless shell or formed head: \( t_r = 72.82 \text{ mm} = 2.867 \text{ in} \)
- Net nozzle thickness: \( t'n = t_n - c_n - c'n = 66.90 \text{ mm} = 2.634 \text{ in} \)
- Net required thickness of a seamless nozzle wall: \( t'n = 26.33 \text{ mm} = 1.037 \text{ in} \)
- Nozzle reinforcement thickness: \( t'x = t_n - c_n = 127.00 \text{ mm} = 5.000 \text{ in} \)
- Limit of reinforcement parallel to the vessel wall: \( L_r = \max(d, R_v + t_n + t) = 463.20 \text{ mm} = 18.236 \text{ in} \)
- Useless limit of reinforcement in vessel wall: \( L_0 = \text{(distance from border)} = 61.40 \text{ mm} = 2.417 \text{ in} \)
- Useless limit of reinforcement normal to the vessel wall: \( h_0 = \min(2.5 \cdot t, 2.5 \cdot t_n) = 182.50 \text{ mm} = 7.185 \text{ in} \)

**Internal pressure**
- Strength reduction factor: \( f_r = 1.00000 \)
- Fr1 factor: \( f_{r1} = \max(S_n/S_v; 1) = 1.00000 \)
- Fr2 factor: \( f_{r2} = \max(S_n/S_v; 1) = 1.00000 \)
- Length of projection defining the thickened portion of integral reinforcement: \( L = 200.00 \text{ mm} = 7.874 \text{ in} \)
- Correction factor: \( F = 1.00000 \)
- Internal pressure: \( P = 14.22 \text{ MPa} = 2062.4 \text{ psi} \)
- Static head internal: \( P_h = 0 \text{ MPa} = 0 \text{ psi} \)
- Area available in shell: \( A_1 = \max \left\{ \frac{d(E_f - Ft_r) - 2 t_n(E_f - Ft_r) \left(1 - f_{r1}\right)}{2(t + t_n')(E_f - Ft_r) - 2 t_n'(E_f - Ft_r) \left(1 - f_{r1}\right)} \right\} = 67.2 \text{ mm}^2 = 0.104 \text{ in}^2 \)
- Corner area to be subtracted from nozzle area: \( A_2 = (r^2 - \pi r^2 f_{r1}^2) / 2 = 21.5 \text{ mm}^2 = 0.033 \text{ in}^2 \)
- Area available in outward nozzle: \( A_3 = 36701.0 \text{ mm}^2 = 56.887 \text{ in}^2 \)
- Area available in inward nozzle: \( A_4 = 0 \text{ mm}^2 = 0 \text{ in}^2 \)
- Area available in outward weld: \( A_{41} = (\text{leg})^2 \cdot f_{r2} = 225.0 \text{ mm}^2 = 0.349 \text{ in}^2 \)
- Area available in inward weld: \( A_{43} = (\text{leg})^2 \cdot f_{r2} = 0 \text{ mm}^2 = 0 \text{ in}^2 \)
- Area required: \( A = d \cdot t + 2 \cdot t_n' - t \cdot f_{r1} = 33731.0 \text{ mm}^2 = 52.283 \text{ in}^2 \)
- Total available area: \( A_t = A_1 + A_2 + A_3 + A_{41} + A_{43} = 36993.2 \text{ mm}^2 = 57.340 \text{ in}^2 \)

**Opening maximum allowable pressure**
- Opening maximum allowable pressure: \( P_{\text{max}} = 15.20 \text{ MPa} = 2204.2 \text{ psi} \)
- Total pressure: \( P_t = 14.22 \text{ MPa} = 2062.4 \text{ psi} \)

\( P_t \leq P_{\text{max}}: \text{ Ok} \)
### Nozzle neck thickness (according to UG-45)

<table>
<thead>
<tr>
<th>Condition</th>
<th>Formula</th>
<th>t (UG-45)</th>
<th>t (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>t(UG-16) =</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27 and UG-28</td>
<td>ta =</td>
<td>29.33 mm</td>
<td>1.155 in</td>
</tr>
<tr>
<td>Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance</td>
<td>tb1 = tr(P_int) - c' =</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>tb1 no less than minimum thickness specified in UG-16(b)</td>
<td>tb1* = max[tb1, t(UG-16)] =</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with E=1) for the shell or head</td>
<td>tb2 =</td>
<td>3.49 mm</td>
<td>0.138 in</td>
</tr>
<tr>
<td>tb2 no less than minimum thickness specified in UG-16(b)</td>
<td>tb2* = max[tb2, t(UG-16)] =</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>tb3 =</td>
<td>11.34 mm</td>
<td>0.446 in</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>t(UG-45) = max(ta, tb) =</td>
<td>29.33 mm</td>
<td>1.155 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>tn =</td>
<td>69.90 mm</td>
<td>2.752 in</td>
</tr>
</tbody>
</table>

\[ \text{tn} \geq \text{t(UG-45)}: \text{Ok} \]
**External pressure**

**Net thicknesses**
- Net shell thickness \( t' = t - c_s - c's \) = 73.00 mm \( 2.874 \) in
- Net required thickness of a seamless shell or formed head \( tr \) = 6.88 mm \( 0.271 \) in
- Net nozzle thickness \( tn' = tn - cn - c'n \) = 66.90 mm \( 2.634 \) in
- Net required thickness of a seamless nozzle wall \( trn \) = 3.08 mm \( 0.121 \) in
- Nozzle reinforcement thickness \( tx' = tx - cn \) = 127.00 mm \( 5.000 \) in
- Limit of reinforcement parallel to the vessel wall \( L_{R} = \max(d, R_v + t_n + t) \) = 463.20 mm \( 18.236 \) in
- Useful limit of reinforcement in vessel wall \( L_o = \text{(distance from border)} \) = 61.40 mm \( 2.417 \) in
- Useful limit of reinforcement normal to the vessel wall \( h_o = \min(2.5 \cdot t, 2.5 \cdot t_n) \) = 182.50 mm \( 7.185 \) in

**External pressure**
- Strength reduction factor \( fr \) = 1.00000
- fr1 factor \( fr1 = \max(S_n/S_v; 1) \) = 1.00000
- fr2 factor \( fr2 = \max(S_n/S_v; 1) \) = 1.00000
- Lenght of projection defining the thickened portion of integral reinforcement \( L \) = 200.00 mm \( 7.874 \) in
- Correction factor \( F \) = 1.00000
- External pressure \( P \) = 0.10 MPa \( 14.9 \) psi
- Static head external \( Ph \) = 0 MPa \( 0 \) psi

**Area available**
- Area available in shell \( d_1 = \frac{d (E_y' f - F_{tr}) - 2 t' \left( E_y' f - F_{tr} \right) \left( 1 - f_{st} \right)}{2 \left( t + t'_n \right) \left( E_y' f - F_{tr} \right) - 2 t'_n \left( E_y' f - F_{tr} \right) \left( 1 - f_{st} \right)} \) = 24,914.0 mm² \( 38.617 \) in²
- Area available in outward nozzle \( d_2 = \min \left\{ \frac{5 \left( t'_n - t_m \right) f_{st} f'}{5 \left( t'_n - t_m \right) f_{st} f'} \right\} \cdot A_3 \) = 45,187.9 mm² \( 70.041 \) in²
- Area available in inward nozzle \( A_3 \) = 0 mm² \( 0 \) in²
- Area available in outward weld \( A_{41} = (\text{leg})^2 \cdot f_{r2} \) = 225.0 mm² \( 0.349 \) in²
- Area available in inward weld \( A_{43} = (\text{leg})^2 \cdot f_{r2} \) = 0 mm² \( 0 \) in²
- Area required \( A = [d \cdot t_r + 2 \cdot t_x' \cdot t_r (1 - f_{r1})] / 2 \) = 1,593.4 mm² \( 2.470 \) in²
- Total available area \( A_{t} = A_{1} + A_{2} + A_{3} + A_{41} + A_{43} \) = 70,326.9 mm² \( 109.007 \) in²
- \( A_{t} \geq A \): Ok

**Opening maximum allowable pressure**
- Opening maximum allowable pressure \( P_{max} \) = 13.11 MPa \( 1,900.8 \) psi
- Total pressure \( P_t \) = 0.10 MPa \( 14.9 \) psi
- \( P_t \leq P_{max} \): Ok
### Nozzle neck thickness (according to UG-45)

<table>
<thead>
<tr>
<th>Description</th>
<th>Formula</th>
<th>Value</th>
<th>Conversion to inches</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>$t_{(UG-16)}$</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27 and UG-28</td>
<td>$t_a$</td>
<td>6.08 mm</td>
<td>0.239 in</td>
</tr>
<tr>
<td>Min. thickness required at internal pressure (with $E=1$) for the shell or head, plus corrosion allowance</td>
<td>$t_{b1} = t_{(P, int) - c'}$</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>$t_{b1}$ no less than minimum thickness specified in UG-16(b)</td>
<td>$t_{b1*} = \max{t_{b1}, t_{(UG-16)}}$</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with $E=1$) for the shell or head</td>
<td>$t_{b2}$</td>
<td>3.49 mm</td>
<td>0.138 in</td>
</tr>
<tr>
<td>$t_{b2}$ no less than minimum thickness specified in UG-16(b)</td>
<td>$t_{b2*} = \max{t_{b2}, t_{(UG-16)}}$</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>$t_b = \min{t_b, \max{t_{b1*}, t_{b2*}}}$</td>
<td>11.34 mm</td>
<td>0.446 in</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>$t_{(UG-45)} = \max{t_a, t_b}$</td>
<td>11.34 mm</td>
<td>0.446 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>$t_n$</td>
<td>69.90 mm</td>
<td>2.752 in</td>
</tr>
</tbody>
</table>

$t_n \geq t_{(UG-45)}$: Ok
Reinforcement of opening - Out Shell Side

According to: Asme VIII Div. 1 Ed. 2015, UG-36 - Metric Units

Design data

**Internal design temperature**
- \( T = 420.00 \, ^\circ\text{C} \quad 788.00 \, ^\circ\text{F} \)

**Internal design pressure**
- \( P = 14.22 \, \text{MPa} \quad 2062.4 \, \text{psi} \)

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

**External design pressure**
- \( P_e = 0.10 \, \text{MPa} \quad 14.9 \, \text{psi} \)

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

**Nozzle material: SA-182 F22 3 - Forgings**
- Allowable stress in nozzle \( \text{Sn} = 133.60 \, \text{MPa} \quad 19377.0 \, \text{psi} \)

**Shell material: SA-387 22 2 - Plate**
- Allowable stress in vessel \( \text{Sv} = 133.60 \, \text{MPa} \quad 19377.0 \, \text{psi} \)

**Nozzle geometry**

- **Outside diameter** \( \text{do} = 597.00 \, \text{mm} \quad 23.504 \, \text{in} \)
- **Inside diameter** \( \text{di} = 457.20 \, \text{mm} \quad 18.000 \, \text{in} \)
- **Nozzle thickness** \( \text{tn} = 69.90 \, \text{mm} \quad 2.752 \, \text{in} \)
- **Nozzle reinforcement thickness** \( \text{tx} = 130.00 \, \text{mm} \quad 5.118 \, \text{in} \)
- **Length of projection defining the thickened portion of integral reinforcement** \( \text{L} = 200.00 \, \text{mm} \quad 7.874 \, \text{in} \)
- **Tapering angle** \( \text{ta} = 45.00 ^\circ \)
- **Corner radius** \( \text{r} = 10.00 \, \text{mm} \quad 0.394 \, \text{in} \)
- **Nozzle corrosion allowance** \( \text{cn} = 3.00 \, \text{mm} \quad 0.118 \, \text{in} \)
- **Nozzle undertolerance** \( \text{c'n} = 0 \, \text{mm} \quad 0 \, \text{in} \)
- **Nozzle connection** \( = \quad \text{Integrally reinforced} \)
- **Nozzle position** \( = \quad \text{Radial} \)
- **Offset from shell border** \( = 3150.00 \, \text{mm} \quad 124.016 \, \text{in} \)
- **Angular offset** \( = 180.00 ^\circ \)
- **Weld leg length of the outside nozzle fillet weld** \( \text{two} = 15.00 \, \text{mm} \quad 0.591 \, \text{in} \)
- **Minimum weld leg length of the outside nozzle fillet weld** \( \text{two} = 8.49 \, \text{mm} \quad 0.334 \, \text{in} \)
- \( r = \text{Min}[\frac{1}{4}t, 1/8 \text{in. (3mm)}] = \text{Ok} \)

**Opening geometry**

- **Finished diameter of circular opening** \( \text{d} = 463.20 \, \text{mm} \quad 18.236 \, \text{in} \)
- **Finished radius of circular opening** \( \text{Rn} = 231.60 \, \text{mm} \quad 9.118 \, \text{in} \)
- **Nearest opening** \( = \quad \text{In Shell Side} \)
- **Distance to nearest opening** \( = 3532.33 \, \text{mm} \quad 139.068 \, \text{in} \)
- **Maximum distance before multiple openings calculation occurs** \( = 928.91 \, \text{mm} \quad 36.571 \, \text{in} \)

**Shell Geometry**

- **Inside shell diameter** \( \text{D} = 1275.00 \, \text{mm} \quad 50.197 \, \text{in} \)
- **Inside radius of shell course under consideration** \( \text{R} = 637.50 \, \text{mm} \quad 25.098 \, \text{in} \)
- **Shell thickness** \( \text{t} = 76.00 \, \text{mm} \quad 2.992 \, \text{in} \)
- **Shell corrosion allowance** \( \text{cs} = 3.00 \, \text{mm} \quad 0.118 \, \text{in} \)
- **Shell undertolerance** \( \text{c's} = 0 \, \text{mm} \quad 0 \, \text{in} \)
Internal pressure

Net thicknesses
Net shell thickness \( t' = t - c_s - c's \) = 73.00 mm 2.874 in
Net required thickness of a seamless shell or formed head \( t_r \) = 72.82 mm 2.867 in
Net nozzle thickness \( t'n = t_n - c_n' \) = 66.90 mm 2.634 in
Net required thickness of a seamless nozzle wall \( t'n = t_n - c_n' \) = 66.90 mm 2.634 in
Nozzle reinforcement thickness \( t'x = t_x - c_n \) = 127.00 mm 5.000 in
Limit of reinforcement parallel to the vessel wall
\( L_R = \max(d, R_e + t_n + t) \) = 463.20 mm 18.236 in
Useful limit of reinforcement in vessel wall
\( L_o = \text{distance from border} \) = -78.49 mm -3.090 in
Useful limit of reinforcement normal to the vessel wall
\( h_o = \min(2.5 \cdot t, 2.5 \cdot t_n) \) = 182.50 mm 7.185 in

Internal pressure
Strength reduction factor \( f_r \) = 1.0000
fr1 factor \( f_r1 = \max(S_n/S_v; 1) \) = 1.0000
fr2 factor \( f_r2 = \max(S_n/S_v; 1) \) = 1.0000
Lenght of projection defining the thickened portion of integral reinforcement \( L \) = 200.00 mm 7.874 in
Correction factor \( F \) = 1.0000
Internal pressure \( P \) = 14.22 MPa 2062.4 psi
Static head internal \( P_h \) = 0 MPa 0 psi

Area available in shell \( A_1 = \max\left\{ \frac{d(E_{g'} - F_{t_r}) - 2t'_c(E_{g'} - F_{t_r}) \left( 1 - f_{r1} \right)}{2(t + t'_c)(E_{g'} - F_{t_r}) - 2t'_c(E_{g'} - F_{t_r}) \left( 1 - f_{r1} \right)} \right\} = 17.3 \text{ mm}^2 0.027 \text{ in}^2
Corner area to be subtracted from nozzle area \( A_2 = \frac{(r^2 - \pi \cdot r^2 \cdot f_{r1})}{4} = 21.5 \text{ mm}^2 0.033 \text{ in}^2
Area available in outward nozzle \( A_3 = \min\left\{ \frac{5(t'_c - t_m)\cdot t'_x}{5(t'_c - t_m)\cdot t'_x - A_3} \right\} = 36701.0 \text{ mm}^2 56.887 \text{ in}^2
Area available in inward nozzle \( A_4 = 0 \text{ mm}^2 0 \text{ in}^2
Area available in outward weld \( A_{41} = (\text{leg})^2 \cdot f_{r2} = 225.0 \text{ mm}^2 0.349 \text{ in}^2
Area available in inward weld \( A_{43} = (\text{leg})^2 \cdot f_{r2} = 0 \text{ mm}^2 0 \text{ in}^2
Area required \( A = d \cdot tr + 2 \cdot tx' \cdot tr (1-f_{r1}) \) = 33731.0 mm² 52.283 in²
Total available area \( A_t = A_1 + A_2 + A_3 + A_{41} + A_{43} = 36943.3 \text{ mm}^2 57.262 \text{ in}^2
At ≥ A: OK

Opening maximum allowable pressure
Opening maximum allowable pressure \( P_{max} \) = 15.20 MPa 2204.2 psi
Total pressure \( P_t \) = 14.22 MPa 2062.4 psi
Pt ≤ Pmax: OK
### Nozzle neck thickness (according to UG-45)

<table>
<thead>
<tr>
<th>Description</th>
<th>Formula</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum required thickness as per UG-16(b), including corrosion</td>
<td>$t_{\text{UG-16}}$</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum neck thickness required for internal and external pressure using UG-27 and UG-28</td>
<td>$t_a$</td>
<td>29.33 mm</td>
<td>1.155 in</td>
</tr>
<tr>
<td>Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance</td>
<td>$t_{b1} = t_{\text{r(int)-c'}}$</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>$t_{b1}$ no less than minimum thickness specified in UG-16(b)</td>
<td>$t_{b1*} = \max(t_{b1}, t_{\text{UG-16}})$</td>
<td>75.82 mm</td>
<td>2.985 in</td>
</tr>
<tr>
<td>Min. thickness required using external pressure as internal (with E=1) for the shell or head</td>
<td>$t_{b2}$</td>
<td>3.49 mm</td>
<td>0.138 in</td>
</tr>
<tr>
<td>$t_{b2}$ no less than minimum thickness specified in UG-16(b)</td>
<td>$t_{b2*} = \max(t_{b2}, t_{\text{UG-16}})$</td>
<td>4.50 mm</td>
<td>0.177 in</td>
</tr>
<tr>
<td>Minimum thickness of a standard wall pipe</td>
<td>$t_b$</td>
<td>11.34 mm</td>
<td>0.446 in</td>
</tr>
<tr>
<td>$t_b = \min[t_{b3}, \max(t_{b1*}, t_{b2*})]$</td>
<td></td>
<td>11.34 mm</td>
<td>0.446 in</td>
</tr>
<tr>
<td>Minimum required nozzle neck thickness</td>
<td>$t_{\text{UG-45}} = \max(t_a, t_b)$</td>
<td>29.33 mm</td>
<td>1.155 in</td>
</tr>
<tr>
<td>Nozzle thickness</td>
<td>$t_n$</td>
<td>69.90 mm</td>
<td>2.752 in</td>
</tr>
</tbody>
</table>

$t_n \geq t_{\text{UG-45}}$: Ok
### External pressure

**Net thicknesses**

- Net shell thickness: \( t' = t - c_s - c'_s \) = 73.00 mm = 2.874 in
- Net required thickness of a seamless shell or formed head: \( t_r \) = 6.88 mm = 0.271 in
- Net nozzle thickness: \( t_n' = t_n - c_n - c'_n \) = 66.90 mm = 2.634 in
- Net required thickness of a seamless nozzle wall: \( t_{nr} \) = 3.08 mm = 0.121 in
- Nozzle reinforcement thickness: \( t_x = t_x - c_n \) = 127.00 mm = 5.000 in
- Limit of reinforcement parallel to the vessel wall: \( L_x = \max(d, R_e + t_n + t) \) = 463.20 mm = 18.236 in
- Useful limit of reinforcement in vessel wall: \( L_0 = \text{(distance from border)} \) = -78.49 mm = -3.090 in
- Useful limit of reinforcement normal to the vessel wall: \( h_0 = \min(2.5 \cdot t, 2.5 \cdot t_n) \) = 182.50 mm = 7.185 in

**External pressure**

- Strength reduction factor: \( f_r = 1.00000 \)
- \( f_{r1} \) factor: \( f_{r1} = \max(S_{n}/S_{v}; 1) = 1.00000 \)
- \( f_{r2} \) factor: \( f_{r2} = \max(S_{n}/S_{v}; 1) = 1.00000 \)
- Length of projection defining the thickened portion of integral reinforcement: \( L = 200.00 \text{ mm} = 7.874 \text{ in} \)
- Correction factor: \( F = 1.00000 \)
- External pressure: \( P = 0.10 \text{ MPa} = 14.9 \text{ psi} \)
- Static head external: \( P_h = 0 \text{ MPa} = 0 \text{ psi} \)

Area available in shell:

\[
A_1 = \max \left\{ \frac{d(E_{y}F_{tr} - 2t_x(E_{y}F_{tr} - (1 - f_{r1})))}{2(t + t_x') (E_{y}F_{tr} - 2t_x(E_{y}F_{tr} - (1 - f_{r1})))} \right\} = 6415.0 \text{ mm}^2 = 9.943 \text{ in}^2
\]

Area available in outward nozzle:

\[
A_2 = \min \left\{ \frac{5(t_x' - t_{m})(F_{tr}F_{tr} - t_{m})}{2(t + t_x') (E_{y}F_{tr} - 2t_x(E_{y}F_{tr} - (1 - f_{r1})))} \right\} A_3 = 45187.9 \text{ mm}^2 = 70.041 \text{ in}^2
\]

Area available in inward nozzle:

\[A_3 = 0 \text{ mm}^2 = 0 \text{ in}^2\]

Area available in outward weld:

\[A_{41} = \text{(leg)}^2 \cdot f_{r2} = 225.0 \text{ mm}^2 = 0.349 \text{ in}^2\]

Area available in inward weld:

\[A_{43} = \text{(leg)}^2 \cdot f_{r2} = 0 \text{ mm}^2 = 0 \text{ in}^2\]

Area required:

\[A = \frac{d \cdot t_r + 2 \cdot t_x' \cdot t_r (1 - f_{r1})}{2} = 1593.4 \text{ mm}^2 = 2.470 \text{ in}^2\]

Total available area:

\[A_{t} = A_1 + A_2 + A_3 + A_{41} + A_{43} = 51827.8 \text{ mm}^2 = 80.333 \text{ in}^2\]

\[A_{t} \geq A: \text{ Ok}\]

### Opening maximum allowable pressure

- Opening maximum allowable pressure: \( P_{max} = 13.11 \text{ MPa} = 1900.8 \text{ psi} \)
- Total pressure: \( P_t = 0.10 \text{ MPa} = 14.9 \text{ psi} \)

\[P_t \leq P_{max}: \text{ Ok}\]
Nozzle neck thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion
\[ t_{\text{UG-16}} = 4.50 \text{ mm} \quad \text{0.177 in} \]

Minimum neck thickness required for internal and external pressure using UG-27 and UG-28
\[ t_a = 6.08 \text{ mm} \quad \text{0.239 in} \]

Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance
\[ t_{b1} = \min(t_{\text{int}}, t_{\text{UG-16}}) = 75.82 \text{ mm} \quad \text{2.985 in} \]

\[ t_{b1}^* = \max(t_{b1}, t_{\text{UG-16}}) = 75.82 \text{ mm} \quad \text{2.985 in} \]

Min. thickness required using external pressure as internal (with E=1) for the shell or head
\[ t_{b2} = 3.49 \text{ mm} \quad \text{0.138 in} \]

\[ t_{b2}^* = \max(t_{b2}, t_{\text{UG-16}}) = 4.50 \text{ mm} \quad \text{0.177 in} \]

Minimum thickness of a standard wall pipe
\[ t_3 = 11.34 \text{ mm} \quad 0.446 \text{ in} \]

Minimum required nozzle neck thickness
\[ t_{\text{UG-45}} = \max(t_a, t_b) = 11.34 \text{ mm} \quad 0.446 \text{ in} \]

Nozzle thickness
\[ t_n = 69.90 \text{ mm} \quad 2.752 \text{ in} \]

\[ t_n \geq t_{\text{UG-45}}: \text{Ok} \]

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

Customer
Riccardo Petrelli
Drawing
U_150
Revision

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Nozzle thickness (according to UG-45)

Minimum required thickness as per UG-16(b), including corrosion
\[ t_{\text{UG-16}} = 4.50 \text{ mm} \quad 0.177 \text{ in} \]

Minimum neck thickness required for internal and external pressure using UG-27
\[ t_a = 6.08 \text{ mm} \quad 0.239 \text{ in} \]

Min. thickness required at internal pressure (with E=1) for the shell or head, plus corrosion allowance
\[ t_{b1} = \min(t_{\text{int}}, t_{\text{UG-16}}) = 75.82 \text{ mm} \quad 2.985 \text{ in} \]

\[ t_{b1}^* = \max(t_{b1}, t_{\text{UG-16}}) = 75.82 \text{ mm} \quad 2.985 \text{ in} \]

Min. thickness required using external pressure as internal (with E=1) for the shell or head
\[ t_{b2} = 3.49 \text{ mm} \quad 0.138 \text{ in} \]

\[ t_{b2}^* = \max(t_{b2}, t_{\text{UG-16}}) = 4.50 \text{ mm} \quad 0.177 \text{ in} \]

Minimum thickness of a standard wall pipe
\[ t_3 = 11.34 \text{ mm} \quad 0.446 \text{ in} \]

Minimum required nozzle neck thickness
\[ t_{\text{UG-45}} = \max(t_a, t_b) = 11.34 \text{ mm} \quad 0.446 \text{ in} \]

Nozzle thickness
\[ t_n = 69.90 \text{ mm} \quad 2.752 \text{ in} \]

\[ t_n \geq t_{\text{UG-45}}: \text{Ok} \]
Standard Long Welding Neck flange - In Shell Side

According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - 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>

Flange material: SA-182 F22 3 - Forgings
Nozzle material: SA-182 F22 3 - Forgings
Bolting material: SA-193 B16 - Bolting
Gasket: Grooved Metal - Stainless steels and nickel-base alloys

**Flange standard / specification**

| Flange standard / specification | ASME B16.5 2013 |

**Flange rating**

| Flange rating | 1 500 |

**Nominal size**

| Nominal size | 18" |

**Number of bolts**

| Number of bolts | 16 |

**Bolt type**

| Bolt type | ANSI_TEMA 2-1/2" |

**Material group**

| Material group | 1.10 |

**Calculation temperature**

| Calculation temperature | T = 420.00 °C 788.00 °F |

**Internal pressure**

| Internal pressure | Pd = 14.22 MPa 2 062.4 psi |

**Overpressure due to static head**

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

**Calculation pressure**

| Calculation pressure | P = 14.22 MPa 2 062.4 psi |

**Maximum pressure at temperature allowed by the specifications**

| Maximum pressure at temperature allowed by the specifications | Pmax = 17.67 MPa 2 562.8 psi |

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

| New & cold (flange) | = 25.86 MPa 3 750.7 psi |
| Hot & corroded (flange) | = 17.67 MPa 2 562.8 psi |
| New & cold (bolts) | = 25.86 MPa 3 750.7 psi |
| Hot & corroded (bolts) | = 17.67 MPa 2 562.8 psi |
| New & cold (opening) | = 17.98 MPa 2 608.3 psi |
| Hot & corroded (opening) | = 15.20 MPa 2 204.2 psi |
| New & cold (cylinder) | = 34.52 MPa 5 006.5 psi |
| Hot & corroded (cylinder) | = 32.89 MPa 4 770.5 psi |

**External pressure**

| External design pressure | Pe = 0.10 MPa 14.9 psi |
| Overpressure due to static head | Ph = 0 MPa 0 psi |

**Calculation pressure**

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

**Maximum pressure at temperature allowed by the specifications**

| Maximum pressure at temperature allowed by the specifications | Pmax = 17.67 MPa 2 562.8 psi |
Cylindrical shell

- Inside diameter: \( D = 457.20 \text{ mm} = 18.000 \text{ in} \)
- Outside diameter: \( D_o = 597.00 \text{ mm} = 23.504 \text{ in} \)
- Length: \( L = 1016.00 \text{ mm} = 40.000 \text{ in} \)
- Adopted thickness: \( t = 69.90 \text{ mm} = 2.752 \text{ in} \)
- Corrosion allowance: \( c = 3.00 \text{ mm} = 0.118 \text{ in} \)
- External corrosion allowance: \( c_e = 0 \text{ mm} = 0 \text{ in} \)
- Wall undertolerance: \( c' = 0 \text{ mm} = 0 \text{ in} \)
- Forming strain (cylinders formed from plate): \( \varepsilon_f = 50\cdot t/(R + t/2) = 7,10\% \)
- Allowable stress: \( S = 133.60 \text{ MPa} = 19\,377.0 \text{ psi} \)
- Internal pressure: \( P_i = 14.22 \text{ MPa} = 2\,062.4 \text{ psi} \)
- Overpressure due to static head: \( P_h = 0 \text{ MPa} = 0 \text{ psi} \)
- Calculation pressure: \( P = P_i + P_h = 14.22 \text{ MPa} = 2\,062.4 \text{ psi} \)
- Reference diameter: \( = \) Inside
- Calculation radius (inside): \( R = 228.60 \text{ mm} = 9.000 \text{ in} \)

Required thickness for circumferential stress, UG-27(c)(1)
\[
 t_{r,c} = \frac{P(R + c + c_e)}{P[R + c + c_e]} + c_e + c_t \]
\( = 29.33 \text{ mm} = 1.155 \text{ in} \)

Required thickness for longitudinal stress, UG-27(c)(2)
\[
 t_{r,l} = \frac{4E}{2SE + 0.45} + c_e + c_t \]
\( = 15.07 \text{ mm} = 0.593 \text{ in} \)

Minimum required thickness
\[
 t_{r} = \max[t_{r,c}, t_{r,l}] = 29.33 \text{ mm} = 1.155 \text{ in} \]
\( t \geq t_r: \text{ Ok} \)

External pressure
- External design temperature: \( T_e = 20.00 \text{ °C} = 68.00 \text{ °F} \)
- External pressure: \( P_e = 0.10 \text{ MPa} = 14.9 \text{ psi} \)
- External static head: \( P_h = 0 \text{ MPa} = 0 \text{ psi} \)
- Calculation pressure: \( P = P_e + P_h = 0.10 \text{ MPa} = 14.9 \text{ psi} \)
- Outside diameter: \( D_o = 915.00 \text{ mm} = 36.024 \text{ in} \)
- Modulus of elasticity: \( E = 200\,000.00 \text{ MPa} = 29\,007\,547.6 \text{ psi} \)
- Axial length between reinforcements: \( L = 1016.00 \text{ mm} = 40.000 \text{ in} \)
- \( L / D_o \) ratio: \( = 1.11038 \)
- \( D_o / t \) ratio: \( = 13.67713 \)
- Factor A: \( = 0.02495 \)
- Factor B: \( = 144.44 \text{ MPa} = 20\,949.3 \text{ psi} \)
- External pressure: \( P_a = \frac{4B}{3[D_o/\left(t-c-e_c-c_e\right)]} = 14.08 \text{ MPa} = 2\,042.3 \text{ psi} \)
- Required thickness: \( t_r = 6.08 \text{ mm} = 0.239 \text{ in} \)
\( P_a \geq P: \text{ Ok} \)
\( t \geq t_r: \text{ Ok} \)

Minimum Design Metal Temperature (MDMT)

Internal pressure
- Long Welding Neck flange - In Shell Side (Bolting)
  - Material: \( = \) SA-193 B16
  - Governing Thickness: \( = 63.50 \text{ mm} = 2.500 \text{ in} \)
  - Reduction in MDMT based on available excess thickness: \( TR = 80.00 \text{ °C} = 176.00 \text{ °F} \)
  - Unadjusted MDMT from table UCS-66: \( = -30.00 \text{ °C} = -22.00 \text{ °F} \)
  - Adjusted MDMT from fig. UCS-66.1: \( = -110.00 \text{ °C} = -166.00 \text{ °F} \)
Long Welding Neck flange - In 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>69.90 mm (2.752 in)</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 10.85 °C 51.53 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>23.00 °C 73.40 °F</td>
</tr>
<tr>
<td>Design pressure</td>
<td>14.22 MPa 2062.4 psi</td>
</tr>
<tr>
<td>Maximum allowable working pressure</td>
<td>MAWP H&amp;C = 17.67 MPa 2562.8 psi</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>TR = 10.85 °C 51.53 °F</td>
</tr>
<tr>
<td>Impact test exemption temperature</td>
<td>= 12.15 °C 53.87 °F</td>
</tr>
</tbody>
</table>

Note: No impact testing is required for the ASME B16-5 and ASME B16-47 flanges when used at minimum design metal temperatures no colder than -20°F (-29°C)

Long Welding Neck flange - In Shell 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>76.00 mm (2.992 in)</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 0.14 °C 32.24 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>25.90 °C 78.62 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 3.00 mm 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 72.82 mm 2.867 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 76.00 mm 2.992 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t_E^* = 0.99756$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 25.76 °C 78.38 °F</td>
</tr>
</tbody>
</table>

External pressure

Long Welding Neck flange - In 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>Governing Thickness</td>
<td>63.50 mm (2.500 in)</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>-30.00 °C -22.00 °F</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 110.00 °C -166.00 °F</td>
</tr>
</tbody>
</table>

Long Welding Neck flange - In 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. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>69.90 mm (2.752 in)</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>23.00 °C 73.40 °F</td>
</tr>
<tr>
<td>Design pressure</td>
<td>MAWP H&amp;C = 17.67 MPa 2562.8 psi</td>
</tr>
<tr>
<td>Maximum allowable working pressure</td>
<td>$P = 0.10 \text{ MPa} 14.9 \text{ psi}$</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>TR = 10.85 °C 51.53 °F</td>
</tr>
<tr>
<td>Impact test exemption temperature</td>
<td>= -57.00 °C -70.60 °F</td>
</tr>
</tbody>
</table>
Long Welding Neck flange - In Shell Side (Nozzle to wall - shell)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>76.00 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness TR</td>
<td>80.00 °C</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>25.90 °C</td>
</tr>
<tr>
<td>Joint efficiency E</td>
<td>1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E' = \max(E, 0.8)$</td>
</tr>
<tr>
<td>Corrosion allowance c</td>
<td>3.00 mm</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition Tr</td>
<td>6.88 mm</td>
</tr>
<tr>
<td>Nominal noncorroded thickness Tn</td>
<td>76.00 mm</td>
</tr>
<tr>
<td>Coincident ratio $\frac{t_E}{t_{n-c}}$</td>
<td>0.09425</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>$-54.10 ^\circ C$</td>
</tr>
</tbody>
</table>
Standard Long Welding Neck flange - Out Shell Side
According to: Asme VIII Div. 1 Ed. 2015, Appendix 2 - Metric Units

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flange material</td>
<td>SA-182 F22 3 - Forgings</td>
</tr>
<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>
<tr>
<td>Flange standard / specification =</td>
<td>ASME B16.5 2013</td>
</tr>
<tr>
<td>Flange rating</td>
<td>1500</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 

\[ T = 420.00 \degree C \quad 788.00 \degree F \]

Internal pressure 

\[ P_{d} = 14.22 \text{ MPa} \quad 2062.4 \text{ psi} \]

Overpressure due to static head 

\[ P_{h} = 0 \text{ MPa} \quad 0 \text{ psi} \]

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)**

<table>
<thead>
<tr>
<th>Condition</th>
<th>Pressure (MPa)</th>
<th>PSI</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold (flange)</td>
<td>25.86</td>
<td>3750.7</td>
</tr>
<tr>
<td>Hot &amp; corroded (flange)</td>
<td>17.67</td>
<td>2562.8</td>
</tr>
<tr>
<td>New &amp; cold (bolts)</td>
<td>25.86</td>
<td>3750.7</td>
</tr>
<tr>
<td>Hot &amp; corroded (bolts)</td>
<td>17.67</td>
<td>2562.8</td>
</tr>
<tr>
<td>New &amp; cold (opening)</td>
<td>17.98</td>
<td>2608.3</td>
</tr>
<tr>
<td>Hot &amp; corroded (opening)</td>
<td>15.20</td>
<td>2204.2</td>
</tr>
<tr>
<td>New &amp; cold (cylinder)</td>
<td>34.52</td>
<td>5006.5</td>
</tr>
<tr>
<td>Hot &amp; corroded (cylinder)</td>
<td>32.89</td>
<td>4770.5</td>
</tr>
</tbody>
</table>

**External pressure**

<table>
<thead>
<tr>
<th>Condition</th>
<th>Pressure (MPa)</th>
<th>PSI</th>
</tr>
</thead>
<tbody>
<tr>
<td>External design pressure</td>
<td>0.10</td>
<td>14.9</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>( P = P_{e} + P_{h} ) = 0.10 MPa \quad 14.9 psi</td>
<td></td>
</tr>
<tr>
<td>Maximum pressure at temperature allowed by the specifications</td>
<td>( P_{\text{max}} = 17.67 \text{ MPa} \quad 2562.8 \text{ psi} )</td>
<td></td>
</tr>
</tbody>
</table>
### Cylindrical shell

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Conversion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside diameter</td>
<td>457.20 mm</td>
<td>18.000 in</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>597.00 mm</td>
<td>23.504 in</td>
</tr>
<tr>
<td>Length</td>
<td>1 016.00 mm</td>
<td>40.000 in</td>
</tr>
<tr>
<td>Adopted thickness</td>
<td>69.90 mm</td>
<td>2.752 in</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>3.00 mm</td>
<td>0.118 in</td>
</tr>
<tr>
<td>External corrosion allowance</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Wall undertolerance</td>
<td>0 mm</td>
<td>0 in</td>
</tr>
<tr>
<td>Forming strain (cylinders formed from plate)</td>
<td>$\varepsilon_f = \frac{50 \cdot t}{R + t/2}$</td>
<td>7.10%</td>
</tr>
<tr>
<td>Allowable stress</td>
<td>133.60 MPa</td>
<td>19 377.0 psi</td>
</tr>
<tr>
<td>Internal pressure</td>
<td>14.22 MPa</td>
<td>2 062.4 psi</td>
</tr>
<tr>
<td>Overpressure due to static head</td>
<td>0 MPa</td>
<td>0 psi</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>14.22 MPa</td>
<td>2 062.4 psi</td>
</tr>
<tr>
<td>Reference diameter</td>
<td>Inside</td>
<td></td>
</tr>
<tr>
<td>Calculation radius (inside)</td>
<td>228.60 mm</td>
<td>9.000 in</td>
</tr>
<tr>
<td>Required thickness for circumferential stress, UG-27(c)(1)</td>
<td>( t_r = \frac{P(R + c + c_e)}{P(R + c + c_e)} + c + c_e + c_e' )</td>
<td>29.33 mm  1.151 in</td>
</tr>
<tr>
<td>Required thickness for longitudinal stress, UG-27(c)(2)</td>
<td>( t_r = \frac{2SE + 0.4P}{R + c + c_e + c_e'} )</td>
<td>15.07 mm  0.593 in</td>
</tr>
<tr>
<td>Minimum required thickness</td>
<td>( t_{min} = \max{t_{circ}, t_{long}} )</td>
<td>29.33 mm  1.151 in</td>
</tr>
</tbody>
</table>

### External pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>External design temperature</td>
<td>20.00 °C</td>
</tr>
<tr>
<td>External pressure</td>
<td>0.10 MPa</td>
</tr>
<tr>
<td>External static head</td>
<td>0 MPa</td>
</tr>
<tr>
<td>Calculation pressure</td>
<td>0.10 MPa</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>915.00 mm</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>200 000.00 MPa</td>
</tr>
<tr>
<td>Axial length between reinforcements</td>
<td>1 016.00 mm</td>
</tr>
<tr>
<td>L / Do ratio</td>
<td>1.11038</td>
</tr>
<tr>
<td>Do / t ratio</td>
<td>13.67713</td>
</tr>
<tr>
<td>Factor A</td>
<td>0.02495</td>
</tr>
<tr>
<td>Factor B</td>
<td>144.44 MPa</td>
</tr>
<tr>
<td>External pressure</td>
<td>14.08 MPa</td>
</tr>
<tr>
<td>Required thickness</td>
<td>6.08 mm</td>
</tr>
<tr>
<td>( P_a \geq P ): Ok</td>
<td></td>
</tr>
<tr>
<td>( t \geq t_r ): Ok</td>
<td></td>
</tr>
</tbody>
</table>

### Internal pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-193 B16</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>63.50 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>80.00 °C  176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>-30.00 °C</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>-110.00 °C</td>
</tr>
</tbody>
</table>

### Minimum Design Metal Temperature (MDMT)

**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>Governing Thickness</td>
<td>63.50 mm</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>80.00 °C  176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>-30.00 °C</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>-110.00 °C</td>
</tr>
</tbody>
</table>
### Long Welding Neck flange - Out Shell Side (Flange)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>69.90 mm 2.752 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 10.85 °C 51.53 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 23.00 °C 73.40 °F</td>
</tr>
<tr>
<td>Design pressure</td>
<td>P = 14.22 MPa 2062.4 psi</td>
</tr>
<tr>
<td>Maximum allowable working pressure</td>
<td>MAWP H&amp;C = 17.67 MPa 2562.8 psi</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$\frac{P}{\text{MAWP H &amp; C}} = 0.80473$</td>
</tr>
<tr>
<td>Impact test exemption temperature</td>
<td>= 12.15 °C 53.87 °F</td>
</tr>
</tbody>
</table>

**Note:** No impact testing is required for the ASME B16-5 and ASME B16-47 flanges when used at minimum design metal temperatures no colder than -20°F (-29°C)

### Long Welding Neck flange - Out Shell Side (Nozzle to wall - shell)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>76.00 mm 2.992 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 0.14 °C 32.24 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 25.90 °C 78.62 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>$E = 1.00$</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 3.00 mm 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 72.82 mm 2.867 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 76.00 mm 2.992 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$\frac{t_E^*}{t_n - c} = 0.99756$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 25.76 °C 78.38 °F</td>
</tr>
</tbody>
</table>

### External pressure

### Long Welding Neck flange - Out Shell Side (Bolting)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-193 B16</th>
</tr>
</thead>
<tbody>
<tr>
<td>Governing Thickness</td>
<td>63.50 mm 2.500 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= -30.00 °C -22.00 °F</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= -110.00 °C -166.00 °F</td>
</tr>
</tbody>
</table>

### Long Welding Neck flange - Out Shell Side (Flange)

<table>
<thead>
<tr>
<th>Material</th>
<th>SA-182 F22 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>69.90 mm 2.752 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 23.00 °C 73.40 °F</td>
</tr>
<tr>
<td>Design pressure</td>
<td>P = 0.10 MPa 14.9 psi</td>
</tr>
<tr>
<td>Maximum allowable working pressure</td>
<td>MAWP H&amp;C = 17.67 MPa 2562.8 psi</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$\frac{P}{\text{MAWP H &amp; C}} = 0.00583$</td>
</tr>
<tr>
<td>Impact test exemption temperature</td>
<td>= -57.00 °C -70.60 °F</td>
</tr>
</tbody>
</table>
### Long Welding Neck flange - Out Shell Side (Nozzle to wall - shell)

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
</tr>
<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>76.00 mm / 2.992 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>80.00 °C / 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>25.90 °C / 78.62 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8)$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>3.00 mm / 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>6.88 mm / 0.271 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>76.00 mm / 2.992 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>0.09425</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>-54.10 °C / -65.38 °F</td>
</tr>
</tbody>
</table>

**Customer**

Riccardo Petrelli

**Drawing**

U_150

**Revision**

Walter Tosto S.p.A.

via Erasmo Piaggio

Chieti

Telephone, Fax

Website, Email address

Date ________ Calc. ________ Contr._______ Appr._______

01/07/2016

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2015.2
<table>
<thead>
<tr>
<th>Component</th>
<th>Operating conditions</th>
<th>Design temperature</th>
<th>Design pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell</td>
<td>Ts = 420.00 °C / 788.00 °F</td>
<td>Ps = 14.22 MPa / 2062.4 psi</td>
<td></td>
</tr>
<tr>
<td>Channel</td>
<td>Tc = 454.00 °C / 849.20 °F</td>
<td>Pt = 18.44 MPa / 2674.0 psi</td>
<td></td>
</tr>
<tr>
<td>Tubesheet</td>
<td>T = 454.00 °C / 849.20 °F</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tubes</td>
<td>Tt = 454.00 °C / 849.20 °F</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Tubesheet material
**SA-182 F22 3 - Forgings**
- Tubersheet design temperature: T = 454.00 °C / 849.20 °F
- Modulus of elasticity for tubesheet material at T: E = 179 600.00 MPa / 26 048 777.7 psi
- Allowable stress for tubesheet material at T: S = 127.76 MPa / 18 530.0 psi
- Allowable primary plus secondary stress for tubesheet material: SPS = 2·Sy = 461.76 MPa / 66 972.6 psi

### Tube material
**SB-444 2 Solution ann. N06625 (high allowable stresses) - Smls. pipe & tube**
- Tube design temperature: Tt = 454.00 °C / 849.20 °F
- Modulus of elasticity for tube material at Tt: Et = 182 760.00 MPa / 26 507 097.0 psi
- Allowable stress for tube material at tubesheet design temperature: StT = 184.00 MPa / 26 686.9 psi

### Channel material
**SA-387 22 2 - Plate**
- Channel design temperature: Tc = 454.00 °C / 849.20 °F
- Modulus of elasticity for channel material at Tc: Ec = 179 600.00 MPa / 26 048 777.7 psi
- Poisson's ratio of channel material: νc = 0.30
- Allowable stress for channel material at Tc: Sc = 127.76 MPa / 18 530.0 psi
- Allowable primary plus secondary stress for channel material at Tc: SPSc = 2·Sy = 461.76 MPa / 66 972.6 psi

### Shell material
**SA-387 22 2 - Plate**
- Shell design temperature: Ts = 420.00 °C / 788.00 °F
- Modulus of elasticity for shell material at Ts: Es = 182 400.00 MPa / 26 454 883.4 psi
- Poisson's ratio of shell material: νs = 0.30
- Allowable stress for shell material at Ts: Ss = 133.60 MPa / 19 377.0 psi
- Allowable primary plus secondary stress for shell material at Ts: SPSs = 2·Sy = 475.60 MPa / 68 979.9 psi

### Bolting material
**SA-193 B16 - Bolting**
- Allowable stress for the bolt evaluated at the design temperature: Sbo = 129.92 MPa / 18 843.3 psi
- Allowable stress for the bolt evaluated at the gasket seating temperature: Sbg = 138.00 MPa / 20 015.2 psi
<table>
<thead>
<tr>
<th>Geometric data</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter of tubesheet</td>
<td>A</td>
<td>2010.00 mm</td>
</tr>
<tr>
<td>Bolt circle diameter</td>
<td>C</td>
<td>1810.00 mm</td>
</tr>
<tr>
<td>Shell corrosion allowance</td>
<td>cs</td>
<td>3.00 mm</td>
</tr>
<tr>
<td>Shell undertolerance</td>
<td>cs'</td>
<td>0 mm</td>
</tr>
<tr>
<td>Shell thickness</td>
<td>t_s</td>
<td>73.00 mm</td>
</tr>
<tr>
<td>Channel corrosion allowance</td>
<td>cc</td>
<td>0 mm</td>
</tr>
<tr>
<td>Channel undertolerance</td>
<td>cc'</td>
<td>0 mm</td>
</tr>
<tr>
<td>Channel thickness</td>
<td>t_c</td>
<td>116.00 mm</td>
</tr>
<tr>
<td>Perimeter of the tube layout</td>
<td>C_p</td>
<td>3764.42 mm</td>
</tr>
<tr>
<td>Area enclosed by perimeter C_p</td>
<td>A_p</td>
<td>1127677.1 mm²</td>
</tr>
<tr>
<td>Tubeside corrosion allowance</td>
<td>c_ts</td>
<td>0 mm</td>
</tr>
<tr>
<td>Shellside corrosion allowance</td>
<td>c_ss</td>
<td>3.00 mm</td>
</tr>
<tr>
<td>Tubesheet undertolerance</td>
<td>c'</td>
<td>0 mm</td>
</tr>
<tr>
<td>Tubesheet thickness</td>
<td>t_tubesheet</td>
<td>265.00 mm</td>
</tr>
<tr>
<td>Tubesheet thickness for calculation</td>
<td>h_t</td>
<td>262.00 mm</td>
</tr>
<tr>
<td>Nominal outside diameter of tubes</td>
<td>d_t</td>
<td>31.75 mm</td>
</tr>
<tr>
<td>Radius to outermost tube hole center</td>
<td>r_o</td>
<td>599.13 mm</td>
</tr>
<tr>
<td>Equivalent diameter of outer tube limit circle</td>
<td>D_o</td>
<td>1230.00 mm</td>
</tr>
<tr>
<td>Square tube pitch</td>
<td>p</td>
<td>42.33 mm</td>
</tr>
<tr>
<td>Nominal tube wall thickness</td>
<td>t_t</td>
<td>4.19 mm</td>
</tr>
<tr>
<td>Total area of untubed lanes</td>
<td>AL</td>
<td>0 mm²</td>
</tr>
<tr>
<td>Expanded length of tube in tubesheet</td>
<td>l_t</td>
<td>262.00 mm</td>
</tr>
<tr>
<td>Tube side pass partition groove depth</td>
<td>h_g</td>
<td>0 mm</td>
</tr>
<tr>
<td>Effective tube side pass partition groove depth</td>
<td>h_g'</td>
<td>0 mm</td>
</tr>
<tr>
<td>Diameter of shell gasket load reaction</td>
<td>G_s</td>
<td>1399.51 mm</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>ρ_c</td>
<td>1.13781</td>
</tr>
<tr>
<td>Inside channel diameter</td>
<td>D_c</td>
<td>1238.00 mm</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>ρ_c</td>
<td>1.00650</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>k_c</td>
<td>0.0046 mm⁻¹</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>λ_c</td>
<td>235 507 634 N</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>δ_ε</td>
<td>0.01563</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>ω_c</td>
<td>37 423.5 mm²</td>
</tr>
<tr>
<td>Diameter ratio</td>
<td>F</td>
<td>2.75916</td>
</tr>
</tbody>
</table>

### Minimum RCB 7.11 thickness

**TEMA Class**

\[ t - c_t - c_s \geq d_o: \text{ Ok} \]
\[ t \geq 19.10 \text{ mm}: \text{ Ok} \]
### Flanged extension

- **Design bolt load for the operating condition**
  \[ W_o = 0.785G^2 + 2\pi GmP = 30151572 \text{ N} \quad 6778342.47 \text{ lbf} \]
- **Design bolt load for the gasket seating (Flange)**
  \[ W_g = \left( \frac{A_m + A_b}{2} \right) S_{bg} = 31472911 \text{ N} \quad 7075391.36 \text{ lbf} \]
- **Moment arm for load HG**
  \[ h_G = \frac{(C-G)}{2} = 205.25 \text{ mm} \quad 8.081 \text{ in} \]
- **Allowable stress for the tubesheet extension at design temperature**
  \[ S_o = 127.76 \text{ MPa} \quad 18530.0 \text{ psi} \]
- **Allowable stress for the tubesheet extension at gasket seating temperature**
  \[ S_g = 148.00 \text{ MPa} \quad 21465.6 \text{ psi} \]
- **Flanged extension thickness**
  \[ t_{fe} = \frac{19W}{S_o h_G} = 773.75 \text{ mm} \quad 30.463 \text{ in} \]
- **Minimum required thickness of the tubesheet flanged extension (operating)**
  \[ h_r = \frac{19W}{S_o G} = 256.44 \text{ mm} \quad 10.096 \text{ in} \]
- **Minimum required thickness of the tubesheet flanged extension (gasket seating)**
  \[ h_r = \frac{19W}{S_g G} = 243.43 \text{ mm} \quad 9.584 \text{ in} \]
- **Minimum required thickness of the tubesheet flanged extension**
  \[ t_{fe} = \frac{19W}{S_o G} = 256.44 \text{ mm} \quad 10.096 \text{ in} \]

### Tube to tubesheet joints

- **Fillet weld leg**
  \[ a_f = 9.00 \text{ mm} \quad 0.354 \text{ in} \]
- **Min. required length of the weld leg(s)**
  \[ a_f = \sqrt{\left(0.75d_e\right)^2 + 2.73d_o \left(d_o - t\right)} / f_w f_d - 0.75 d_e = 8.14 \text{ mm} \quad 0.321 \text{ in} \]
- **Allowable stress of the tube**
  \[ S_a = 184.00 \text{ MPa} \quad 26686.9 \text{ psi} \]
- **Allowable stress of the material to which the tube is welded**
  \[ S_t = 127.76 \text{ MPa} \quad 18530.0 \text{ psi} \]
- **Allowable stress in weld**
  \[ S_w = \min\{S_o, S_t\} = 127.76 \text{ MPa} \quad 18530.0 \text{ psi} \]
- **Groove weld strength**
  \[ F_g = 0.85 \pi a_f (d_e + 0.67a_f) S_w = 75061 \text{ N} \quad 16874.29 \text{ lbf} \]
- **Axial tube strength**
  \[ F_i = \pi t (d_e - t) S_o = 66765 \text{ N} \quad 15009.37 \text{ lbf} \]
- **Ratio of the design strength to the tube strength**
  \[ f_d = 1.00000 \]
- **Ratio of the fillet weld strength to the design strength**
  \[ f_w = \frac{S_o}{S_w} = 1.44020 \]
- **Max axial load (pressure only)**
  \[ L_{max} = F_t = 66765 \text{ N} \quad 15009.37 \text{ lbf} \]
- **Max axial load (pressure or thermally induced)**
  \[ L_{max} = 2F_t = 133530 \text{ N} \quad 30018.73 \text{ lbf} \]

\[ a_f \geq max\{ar, t\}: \text{ Ok} \]
Loading case Design 1: Pt = 18.44 MPa, Ps = -0.10 MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear
\[ \mu = \frac{p - d_t}{p} \]
\[ \mu = 0.24998 \]

Effective tube pitch
\[ p = p \left[ 1 - 4 \cdot \min \left\{ \frac{A_0}{4D_p^2} \right\} \right] \]
\[ p = 42.33 \text{ mm} \]
\[ p = 1.667 \text{ in} \]

Tube expansion depth ratio
\[ \rho = \frac{b_t}{b} \]
\[ \rho = 1.00000 \]

Effective tube hole diameter
\[ d' = \max \left\{ d_t - 2t, \left( \frac{E_p}{E} \right) \left\{ \frac{S_t}{S} \right\} \rho, \left( d_t - 2t \right) \right\} \]
\[ d' = 23.37 \text{ mm} \]
\[ d' = 0.920 \text{ in} \]

Effective ligament efficiency for bending
\[ \mu' = \frac{F_p - d'}{p'} \]
\[ \mu' = 0.44798 \]

A0 = 0.03720
A1 = 1.03140
A2 = -0.64020
A3 = 2.62010
A4 = -2.19290
B0 = 0.33410
B1 = 0.12600
B2 = -0.69200
B3 = 0.68770
B4 = -0.06000

Effective Poisson ratio in perforated region
\[ \nu = B_0 + B_1 \mu' + B_2 (\mu')^2 + B_3 (\mu')^3 + B_4 (\mu')^4 \]
\[ \nu = 0.51801 \]

Effective Poisson ratio in perforated region
\[ \nu = 0.31108 \]

Tubesheet bending stress

Tubesheet design bold load
\[ W^* = 0 \text{ N} \]
\[ W^* = 0 \text{ lb} \]

Moment acting on the unperforated tubesheet rim
\[ M_{TS} = \frac{D_p^2}{16} (\rho_2 - 1) (\rho_2 + 1) P_s - (\rho_2 - 1) (\rho_2 + 1) P_t \]
\[ M_{TS} = -25905 \text{ N} \]
\[ M_{TS} = -5823.59 \text{ lb} \]

Moment acting on the unperforated tubesheet rim
\[ M' = M_{TS} + \omega \rho P_t + \left( \frac{C - G}{2} \right) W^* \]
\[ M' = 664054 \text{ N} \]
\[ M' = 149285.16 \text{ lb} \]

Maximum bending moments acting on the tubesheet at the periphery
\[ M_p = \frac{M^* - \frac{32}{3} D_p}{1 + E} \]
\[ M_p = 819995 \text{ N} \]
\[ M_p = 184342.21 \text{ lb} \]

Maximum bending moments acting on the tubesheet at the center
\[ M_c = \frac{D_p^2}{64} (3 + \nu) (P_s - P_t) \]
\[ M_c = -361107 \text{ N} \]
\[ M_c = \text{141878.52 lb} \]

Maximum bending moments acting on the tubesheet
\[ M = \max \left\{ M_p, M_c \right\} \]
\[ M = 819995 \text{ N} \]
\[ M = 184342.21 \text{ lb} \]

Bending stress
\[ \sigma = \frac{M_{TS}}{6M} \]
\[ \sigma = 159.99 \text{ MPa} \]
\[ \sigma = 23204.9 \text{ psi} \]
\[ \sigma \leq 2S: \text{Ok} \]

Tubesheet shear stress

|Ps - Pt| ≤ 3.2·S·μ·h/Do: Shear stress is not required to be calculated

Shell and channel stresses

Axial membr. stress in the channel at tubesheet junction
\[ \sigma_{cm} = \frac{D_{p}^2 P_t}{4A(D_t + t)} \]
\[ \sigma_{cm} = 44.98 \text{ MPa} \]
\[ \sigma_{cm} = 6523.3 \text{ psi} \]

Bending stress in the channel at its junction to the tubesheet
\[ \sigma_{cb} = \frac{6D_p}{E} \left( \frac{1}{D_p^2} P_t - \frac{6(1 - \nu^2)}{E} \left( \frac{D_p}{D_t} \right) \left( \left( \frac{h_D}{2} \right)^2 \right) \right) \]
\[ \sigma_{cb} = 167.70 \text{ MPa} \]
\[ \sigma_{cb} = 24322.5 \text{ psi} \]

Channel axial stress
\[ \sigma = \left| \sigma_{cm} \right| + \left| \sigma_{cb} \right| \]
\[ \sigma = 212.67 \text{ MPa} \]
\[ \sigma = 30845.7 \text{ psi} \]
Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation

\[ E_c = E_s \frac{k_s}{k_c} \]

\[ E_c = 170748.48 \text{ MPa} = 24727118.4 \text{ psi} \]

\[ k_c = \frac{\beta E_s^2}{6(1-\nu^2)} \]

\[ k_c = 223558480 \text{ N} = 50257941.34 \text{ lbf} \]

\[ \lambda_c = \frac{6D_k k_c}{R^2} \left(1 + \frac{h_k}{R} + \frac{h_k^2}{2} \right) \]

\[ \lambda_c = 269971.89 \text{ MPa} = 39156112.7 \text{ psi} \]

\[ \omega_c = \rho \beta_0 \beta \delta_c \left(1 + h_k \beta_0 \right) \]

\[ \omega_c = 35524.7 \text{ mm}^2 = 55.063 \text{ in}^2 \]

Diameter ratio

\[ F = \frac{1 - \nu}{\beta_c + B \cdot \ln(K_0)} \]

\[ F = 1.63415 \]

Coefficient

\[ f = \frac{1 - \nu}{\beta_c + B \cdot \ln(K_0)} \]

\[ f = 2.65231 \]

Maximum bending moments acting on the tubesheet at the periphery

\[ M_p = \frac{M - \frac{\rho q Z_2}{32} \frac{Z_2}{Z_1}}{1 + F} \]

\[ M_p = 818342 \text{ N} = 183970.49 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the center

\[ M_c = M_p + \frac{\rho_q Z_2 (3 + \nu) (P_2 - P_1)}{64} \]

\[ M_c = -632761 \text{ N} = -142250.24 \text{ lbf} \]

Maximum bending moments acting on the tubesheet

\[ M = \max \left[ |M_p| \right] \]

\[ M = 818342 \text{ N} = 183970.49 \text{ lbf} \]

Bending stress (simplified Elastic-plastic calculation performed)

\[ \sigma = \frac{M}{\mu^2 (h - h_k)} \]

\[ \sigma = 159.67 \text{ MPa} = 23158.1 \text{ psi} \]

\[ \sigma \leq 2S: \text{ Ok} \]
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

$$\rho = \frac{4.5}{\pi D_t} = 42.33\, \text{mm} = 1.667\, \text{in}$$

Tube expansion depth ratio

$$\rho = \frac{h}{L} = 1.00000$$

Effective tube hole diameter

$$d'' = \max \left\{ d_i - \left( \frac{S_t}{S} \right) P_t, \left( d_i - 2h_t \right) \right\} = 23.37\, \text{mm} = 0.920\, \text{in}$$

Effective ligament efficiency for bending

$$\mu' = \frac{p - d''}{\rho} = 0.44798$$

$A_0 = 0.03720$

$A_1 = 1.03140$

$A_2 = -0.64020$

$A_3 = 2.62010$

$A_4 = -2.19290$

$B_0 = 0.33410$

$B_1 = 0.12600$

$B_2 = -0.69200$

$B_3 = 0.68770$

$B_4 = -0.06000$

$$\frac{E''}{E} = A_0 + A_1\mu' + A_2(\mu')^2 + A_3(\mu')^3 + A_4(\mu')^4 = 0.51801$$

Effective Poisson ratio in perforated region

$$\nu = B_0 + B_1\mu' + B_2(\mu')^2 + B_3(\mu')^3 + B_4(\mu')^4 = 0.31108$$

**Tubesheet bending stress**

Shell flange design bolt load for the operating condition

$$W_{ds} = 30\,151\,572\, N = 6\,778\,342.47\, \text{lbf}$$

Tubesheet design bold load

$$W^* = W_{ds} = 30\,151\,572\, N = 6\,778\,342.47\, \text{lbf}$$

Moment acting on the unperforated tubesheet rim

$$M_{TS} = \frac{D_t^2}{16} \left( \rho - 1 \right) \left( \rho + 1 \right) P_t - \left( \rho - 1 \right) \left( \rho + 1 \right) P_1 \right) = 425\,300\, N = 95\,611.24\, \text{lbf}$$

Moment acting on the unperforated tubesheet rim

$$M'' = M_{TS} + \omega \dot{P}_t + \frac{C - G_s}{2\pi D_t} \frac{W^*}{2\pi (P_2 - P_1)} = 2\,022\,967\, N = 454\,781.04\, \text{lbf}$$

Maximum bending moments acting on the tubesheet at the periphery

$$M_p = \frac{M''}{1 + \mu' \rho^2} = 41\,128\, N = 9\,245.95\, \text{lbf}$$

Maximum bending moments acting on the tubesheet at the center

$$M_\rho = M_p + \frac{D_t^2}{64} \left( 3 + \nu' \right) \left( P_2 - P_1 \right) = 1\,162\,173\, N = 261\,266.91\, \text{lbf}$$

Maximum bending moments acting on the tubesheet

$$M = \max \left\{ M_p, M_\rho \right\} = 1\,162\,173\, N = 261\,266.91\, \text{lbf}$$

Bending stress

$$\sigma = \frac{6M}{\mu' (h - \rho C)^3} = 226.76\, \text{MPa} = 32\,888.1\, \text{psi}$$

$$\sigma \leq 2S: \text{Ok}$$

**Tubesheet shear stress**

$$|Ps - Pt| \leq 3.2 \cdot S \cdot \mu \cdot h/Do: \text{Shear stress is not required to be calculated}$$

**Shell and channel stresses**

Axial membr. stress in the channel at tubesheet junction

$$\sigma_{\omega n} = \frac{D_t^2 P_1}{4(D_t + t)} = -0.25\, \text{MPa} = -36.4\, \text{psi}$$

Bending stress in the channel at its junction to the tubesheet

$$\sigma_{\phi b} = \frac{6t}{r} \left( \beta \delta P_1 - \frac{6(1 - \nu')}{E''} \frac{D_t}{r} \left( 1 + \beta \right) \left( M_p + \frac{D_t^2}{32} \left( P_2 - P_1 \right) \right) \right) = -367.69\, \text{MPa} = -53\,329.5\, \text{psi}$$

Channel axial stress

$$\sigma_c = \sigma_{\phi b} = 367.94\, \text{MPa} = 53\,365.9\, \text{psi}$$

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2015.2
Elastic plastic calculation performed

Modulus of elasticity for channel material for simplified elastic-plastic calculation

\[ E'_c = E_c \sqrt{\frac{1}{5} S_c / \sigma_c} = 129,615.81 \text{ MPa} \]
\[ 18,799,184.2 \text{ psi} \]
\[ k_c = \frac{\beta E'_c}{6(1-\nu_c^2)} = 169,963,883 \text{ N} \]
\[ 38,209,397.55 \text{ lbf} \]
\[ \lambda_c = \frac{6D \cdot \kappa_c}{R^2 \left(1 + h_2 \beta_c + \frac{2}{2}\right)} = 205,250.42 \text{ MPa} \]
\[ 29,769,056.2 \text{ psi} \]
\[ \omega_c = \rho_c \cdot k_c \cdot \beta_c (1 + h_2 \beta_c) = 27,008.2 \text{ mm}^2 \]
\[ 41.863 \text{ in}^2 \]

Diameter ratio

\[ K = \frac{A_D}{A} = 1.63415 \]

Coefficient

\[ F = \frac{1 - \nu}{R^2} \frac{\lambda_c + E \cdot \ln(K)}{\sigma} = 2.17304 \]

Maximum bending moments acting on the tubesheet at the periphery

\[ M_P = \frac{M'}{1 + F} = 173,806 \text{ N} \]
\[ 39,073.21 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the center

\[ M_0 = M + \frac{D^2 (3 + \nu) (P_2 - P_1)}{64} = 1,294,852 \text{ N} \]
\[ 291,094.18 \text{ lbf} \]

Maximum bending moments acting on the tubesheet

\[ M = \max \left( |M_1|, |M_2| \right) = 1,294,852 \text{ N} \]
\[ 291,094.18 \text{ lbf} \]

Bending stress (simplified Elastic-plastic calculation performed)

\[ \sigma = \frac{\mu^2 (h - h_E)}{R^2} = 252.64 \text{ MPa} \]
\[ 36,642.8 \text{ psi} \]

\( \sigma \leq 2S: \text{ Ok} \)
Loading case Design 3: Pt = 18.44 MPa, Ps = 14.22 MPa, thermal exp.: N, corr.: Y, vacuum: N

Basic ligament efficiency for shear

\[ \mu = \frac{p - d_i}{p} \]

Effective tube pitch

\[ p^* = p \left( 1 - 4 \cdot \min \left[ A_p \left( \frac{D_p}{D} \right) \right] \right)^{1/4} \]

Tube expansion depth ratio

\[ \rho = \frac{D}{D'} \]

Effective tube hole diameter

\[ d^* = \max \left[ \left\{ d_i - 2t_i \left( \frac{E_p}{E} \right) \left( \frac{S_E}{S} \right) \rho \right\} \right] \]

Effective ligament efficiency for bending

\[ \mu^* = \frac{E_p - d^*}{p^*} \]

---

**Basic Ligament Efficiency**

- **Shear**: 0.24998
- **Bending**: 0.44798

**Effective Tube Pitch**: 42.33 mm (1.667 in)

**Tube Expansion Depth Ratio**: 1.00000

**Effective Tube Hole Diameter**: 23.37 mm (0.920 in)

**Effective Poisson Ratio in Perforated Region**: 0.31108

**Shell Flange Design Bolt Load for the Operating Condition**: Wds = 30 151 572 N (6 778 342.47 lbf)

**Tubesheet Design Bolt Load**: W* = Wds = 30 151 572 N (6 778 342.47 lbf)

**Moment Acting on the Unperforated Tubesheet Rim**

\[ M_{TS} = \frac{D_p^2}{16} \left[ (\rho - 1) + (\rho^2 + 1)P_2 - (\rho - 1) + (\rho^2 + 1)P_1 \right] \]

\[ = 402 348 N \] (90 451.34 lbf)

**Moment Acting on the Unperforated Tubesheet Rim**

\[ M^* = M_{TS} + \omega \rho i \frac{(C - G_e)W^*}{2\pi D_p} \]

\[ = 2 693 827 N \] (605 596.44 lbf)

**Maximum Bending Moments Acting on the Tubesheet at the Periphery**

\[ M_p = M^* + \frac{D_p^2}{32} \] (1 + \theta)

\[ = 862 934 N \] (193 995.24 lbf)

**Maximum Bending Moments Acting on the Tubesheet at the Center**

\[ M_o = M_p + \frac{D_p^2}{64} (3 + \nu) (P_2 - P_3) \]

\[ = 532 877 N \] (119 795.46 lbf)

**Maximum Bending Moments Acting on the Tubesheet**

\[ M = \max \left[ M_1, M_2 \right] \]

\[ = 862 934 N \] (193 995.24 lbf)

**Bending Stress**

\[ \sigma = \frac{M}{h \cdot h^2} \]

\[ = 168.37 \text{ MPa} \] (24 420.0 psi)

\[ \sigma \leq 2S: \text{ Ok} \]

---

**Tubesheet Shear Stress**

**Shell and Channel Stresses**

**Axial Membrane Stress in the Channel at the Tubesheet Junction**

\[ \sigma_{cb} = \frac{D_p^2}{4A(D_p + t)} \]

\[ = 44.98 \text{ MPa} \] (6 523.3 psi)

**Bending Stress in the Channel at its Junction to the Tubesheet**

\[ \sigma_{cb} = -\frac{D_p^2}{4A(D_p + t)} \]

\[ = -200.15 \text{ MPa} \] (-29 028.7 psi)

**Channel Axial Stress**

\[ \sigma_c = \sqrt{\sigma_{cb}^2 + \sigma_{cb}^2} \]

\[ = 245.12 \text{ MPa} \] (35 551.9 psi)

---

**Customer**

Riccardo Petrelli

**Drawing**

U_150

**Revision**

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

01/07/2016

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2015.2
Modulus of elasticity for channel material for simplified elastic-plastic calculation

\[ E' = E \sqrt{5S'/\sigma_c} = 158802.89 \text{ MPa} \quad 23032411.5 \text{ psi} \]

\[ k_e = \frac{\beta E' R^2}{6(1-\nu^2)} = 208236594 \text{ N} \quad 46813444.59 \text{ lbf} \]

\[ \lambda_e = \frac{6D_e k_e}{R^2} \left(1 + h\beta_e + \frac{h^2}{2}\right) = 251469.00 \text{ MPa} \quad 36472495.0 \text{ psi} \]

\[ \alpha_e = \rho D_e \beta_e \lambda_c (1 + h\beta_e) = 33090.0 \text{ mm}^2 \quad 51290 \text{ in}^2 \]

Diameter ratio

\[ K = \frac{D_e}{D} = \frac{(1-\nu) (\lambda_e + \beta \ln(K_e))}{F} = 2.51529 \]

Coefficient

Maximum bending moments acting on the tubesheet at the periphery

\[ M_p = M - \frac{D_e}{2} = 908968 \text{ N} \quad 204344.12 \text{ lbf} \]

Maximum bending moments acting on the tubesheet at the center

\[ M_0 = M + \frac{D_e}{64} (3 + \nu) (P_2 - P_1) = 578911 \text{ N} \quad 130144.33 \text{ lbf} \]

Maximum bending moments acting on the tubesheet

\[ M = \max \{ |M_p|, |M_0| \} = 908968 \text{ N} \quad 204344.12 \text{ lbf} \]

Bending stress (simplified Elastic-plastic calculation performed)

\[ \sigma = \frac{M}{\rho'^2 (h - h_e)^2} = 177.35 \text{ MPa} \quad 25722.7 \text{ psi} \]

\[ \sigma \leq 2S: \text{ Ok} \]
Loading case Design 4: \( Pt = -0.10 \) MPa, \( Ps = -0.10 \) MPa, thermal exp.: N, corr.: Y, vacuum: Y

Basic ligament efficiency for shear

\[
\mu = \frac{p - d_t}{p} = \frac{0.24998}{4.5} = 0.04336
\]

Effective tube pitch

\[
p = \frac{d_t - 2\varepsilon}{P - \rho} = \frac{42.33 \text{ mm}}{1.667 \text{ in}} = 1.00000
\]

Tube expansion depth ratio

\[
r = \frac{d_t - 2\varepsilon}{d_t - 2\varepsilon} = 23.37 \text{ mm} = 0.920 \text{ in}
\]

Effective tube hole diameter

\[
d^* = \max \left\{ d_t - 2\varepsilon \left( \frac{S_t}{S} \rho_1 \right), \left( d_t - 2\varepsilon \right) \right\}
\]

Effective ligament efficiency for bending

\[
\mu = \frac{p - d_t}{p} = 0.44798
\]

\[
A_0 = 0.03720
A_1 = 1.03140
A_2 = -0.64020
A_3 = 2.62010
A_4 = -2.19290
B_0 = 0.33410
B_1 = 0.12600
B_2 = -0.69200
B_3 = 0.68770
B_4 = -0.06000
\]

Effective Poisson ratio in perforated region

\[
\nu = B_0 + B_1\mu + B_2\mu^2 + B_3\mu^3 + B_4\mu^4 = 0.51801
\]

Tube sheet bending stress

Shell flange design bolt load for the operating condition

\[
W_{ds} = -218,403 \text{ N} = -49,098.93 \text{ lbf}
\]

Tubesheet design bold load

\[
W^* = W_{ds} = -218,403 \text{ N} = -49,098.93 \text{ lbf}
\]

Moment acting on the unperforated tube sheet rim

\[
M_{TS} = \frac{D_t^2}{16} \left( \rho_2 - 1 \right) \left( \rho_2^2 + 1 \right) P_t - \left( \rho - 1 \right) \left( \rho^2 + 1 \right) P_s \right]
\]

Moment acting on the unperforated tube sheet rim

\[
M^t = M_{TS} + \omega_\rho P_t + \left( \frac{C - G_\rho}{2\pi D_t} \right) W^* = -18,407 \text{ N} = -41,383.16 \text{ lbf}
\]

Maximum bending moments acting on the tube sheet at the periphery

\[
M_p = M_{TS} + \frac{D_t^2}{64} (3 + \nu) \left( P_s - P_t \right)
\]

Maximum bending moments acting on the tube sheet at the center

\[
M_c = max \left\{ \left| M_{TS} \right| \right\} = 4,897 \text{ N} = 1,100.82 \text{ lbf}
\]

Moment acting on the unperforated tube sheet rim

\[
M = \frac{D_t^2}{64} \left( \rho_2 - 1 \right) \left( \rho_2^2 + 1 \right) P_t - \left( \rho - 1 \right) \left( \rho^2 + 1 \right) P_s \right]
\]

Bending stress

\[
\sigma = \frac{M}{\mu (h - h_c)} = 0.96 \text{ MPa} = 138.6 \text{ psi}
\]

\( \sigma \leq 2S: \text{ Ok} \)

Tubesheet shear stress

| \( \frac{Ps - Pt}{S} \times 3.2 \times \mu \cdot h/Do | \text{ Shear stress is not required to be calculated} |

Shell and channel stresses

Axial membrit. stress in the channel at tube sheet junction

\[
\sigma_{\text{cl}} = \frac{D_t^2 P_t}{4 (D_t - t_f)} = -0.25 \text{ MPa} = -36.4 \text{ psi}
\]

Bending stress in the channel at its junction to the tube sheet

\[
\sigma_{\text{cb}} = \frac{D_t^2}{4d_t^2} \left( \frac{P_t}{E_t} \right) \left( 1 + \frac{h_t \rho_t}{2} \right) \left( M_p + \frac{D_t^2}{32} (P_s - P_t) \right)
\]

\[
\sigma_c = |\sigma_{\text{cl}}| + |\sigma_{\text{cb}}| = 1.98 \text{ MPa} = 286.7 \text{ psi}
\]

\( L_c \geq 1.8 \sqrt{D_c \cdot t_c} \): \text{ Ok}

\( \sigma_c \leq 1.5 S_c \): \text{ Ok}
### Minimum Design Metal Temperature (MDMT)

#### U-Tube tubesheet - Tubesheet (Bolting)

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SA-193 B16</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>114.30 mm, 4.500 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>80.00 °C, 176.00 °F</td>
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<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>-30.00 °C, -22.00 °F</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>-110.00 °C, -166.00 °F</td>
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#### U-Tube tubesheet - Tubesheet, Flange

<table>
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<tr>
<th>Description</th>
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<tbody>
<tr>
<td>Material</td>
<td>SA-182 F22 3</td>
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<tr>
<td>Curve of fig. UCS-66</td>
<td>B</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm, 4.567 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>0.54 °C, 32.98 °F</td>
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<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>34.50 °C, 94.10 °F</td>
</tr>
<tr>
<td>Design pressure</td>
<td>14.22 MPa, 2062.4 psi</td>
</tr>
<tr>
<td>Maximum allowable working pressure</td>
<td>14.36 MPa, 2082.7 psi</td>
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<tr>
<td>Coincident ratio</td>
<td>0.99025</td>
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<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>33.96 °C, 93.12 °F</td>
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#### U-Tube tubesheet - Tubesheet, Shell

<table>
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<th>Description</th>
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</thead>
<tbody>
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<td>Material</td>
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<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>116.00 mm, 4.567 in</td>
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<tr>
<td>Reduction in MDMT based on available excess thickness</td>
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<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>49.00 °C, 120.20 °F</td>
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<tr>
<td>Joint efficiency</td>
<td>1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>3.00 mm, 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>97.79 mm, 3.850 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>116.00 mm, 4.567 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>0.86542</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>41.52 °C, 106.74 °F</td>
</tr>
</tbody>
</table>
Tube bundle - Tubes bundle

According to: Asme VIII Div. 1 Ed. 2015, UG-27/28, Appendix 1.1 - Metric Units

Design data

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

Internal design pressure \( P = 18.54 \, \text{MPa} = 2688.9 \, \text{psi} \)

External design temperature \( T_e = 420.00 \, ^\circ C = 788.00 \, ^\circ F \)

External design pressure \( P_e = 14.32 \, \text{MPa} = 2077.3 \, \text{psi} \)

Joint efficiency \( E = 1.00 \)

Material: SB-444 2 Solution ann. N06625 (high allowable stresses) - Smls. pipe & tube

Allowable stress \( S = 184.00 \, \text{MPa} = 26868.9 \, \text{psi} \)

Allowable stress at room temperature \( S_{\text{RT}} = 184.00 \, \text{MPa} = 26868.9 \, \text{psi} \)

Geometry

Inside diameter \( D = 23.37 \, \text{mm} = 0.920 \, \text{in} \)

Outside diameter \( D_o = 31.75 \, \text{mm} = 1.250 \, \text{in} \)

Length \( L = 3658.00 \, \text{mm} = 144.016 \, \text{in} \)

Adopted thickness \( t = 4.19 \, \text{mm} = 0.165 \, \text{in} \)

Corrosion allowance \( c = 0 \, \text{mm} = 0 \, \text{in} \)

External corrosion allowance \( c_e = 0 \, \text{mm} = 0 \, \text{in} \)

Wall undertolerance \( c' = 0 \, \text{mm} = 0 \, \text{in} \)

Forming strain (tubes and pipe bends) \( \varepsilon_f = 100 \cdot r/R = 33.33\% \)

Internal pressure

Allowable stress \( S = 184.00 \, \text{MPa} = 26868.9 \, \text{psi} \)

Internal pressure \( P_i = 18.54 \, \text{MPa} = 2688.9 \, \text{psi} \)

Overpressure due to static head \( P_h = 0 \, \text{MPa} = 0 \, \text{psi} \)

Calculation pressure \( P = P_i + P_h = 18.54 \, \text{MPa} = 2688.9 \, \text{psi} \)

Reference diameter \( = \) Outside

Calculation radius (outside) \( R_o = 15.88 \, \text{mm} = 0.625 \, \text{in} \)

Required thickness for circumferential stress, Appendix 1.1(a) \( t_r = \frac{P R_o}{S F + 0.4 P} = 1.54 \, \text{mm} = 0.061 \, \text{in} \)

Required thickness for longitudinal stress, UG-27(c)(2) \( t_l = \frac{P (R + c + c^{'})}{2S E + 0.4 P} = 0.58 \, \text{mm} = 0.023 \, \text{in} \)

Minimum required thickness \( t_r = \max [t_r(\text{circ}), t_r(\text{long})] = 1.54 \, \text{mm} = 0.061 \, \text{in} \)

Maximum allowable pressures (at the top of the vessel)

New & cold \( = 54.31 \, \text{MPa} = 7877.2 \, \text{psi} \)

Hot & corroded \( = 54.31 \, \text{MPa} = 7877.2 \, \text{psi} \)
External pressure

- External design temperature: $T_e = 420.00 \, ^\circ C = 788.00 \, ^\circ F$
- External pressure: $P_e = 14.32 \, MPa = 2077.3 \, psi$
- External static head: $P_h = 0 \, MPa = 0 \, psi$
- Calculation pressure: $P = P_e + P_h = 14.32 \, MPa = 2077.3 \, psi$
- Outside diameter: $D_o = 31.75 \, mm = 1.250 \, in$
- Axial length between reinforcements: $L = 3393.00 \, mm = 133.583 \, in$
- $L / D_o = 106.86614$
- $D_o / t = 7.57576$
- Factor A: $A = 0.01943$
- Factor B: $B = 103.21 \, MPa = 14969.8 \, psi$
- Yield strength at design temperature: $S_y = 206.45450$
- Allowable stress: $S = 184.00 \, MPa = 26686.9 \, psi$
- $S = \min(2S, 0.9S_y)$
- $\frac{P_1}{2S} = \frac{2167}{D_o / (t - c - c_v - c_v')} - 0.0833 \cdot B = 20.93 \, MPa = 3035.0 \, psi$
- $\frac{P_2}{2S} = \frac{1}{D_o / (t - c - c_v - c_v')} - 1 \cdot \frac{1}{D_o / (t - c - c_v - c_v')} = 42.58 \, MPa = 6175.5 \, psi$
- $\min(P_{a1}, P_{a2}) = 20.93 \, MPa = 3035.0 \, psi$
- Required thickness: $t_r = 3.26 \, mm = 0.128 \, in$
- $P_a \geq P: \, Ok$
- $t \geq t_r: \, Ok$

TEMA RCB Requirements

- Outside diameter: $d_o = 31.75 \, mm = 1.250 \, in$
- Mean radius of bend: $R = 47.63 \, mm = 1.875 \, in$
- Required tube wall thickness prior to bending: $t_0 = t_r \left[ 1 + \frac{d_o}{4R} \right] = 3.80 \, mm = 0.150 \, in$
- $t \geq t_0: \, Ok$

Maximum allowable external pressures

- New & cold: $P_a = 27.98 \, MPa = 4058.0 \, psi$
- Hot & corroded: $P_a = 20.93 \, MPa = 3035.0 \, psi$
### Minimum Design Metal Temperature (MDMT)

#### Internal pressure
**Tube bundle - Tubes bundle**

| Material | = | SB-444 2 Solution annealed N06625 (high allowable stresses) |
| Governing Thickness | = | 4.19 mm | 0.165 in |
| Impact test exemption temperature | = | -198.00 °C | -324.40 °F |

*Note: MDMT according to UNF-65*

#### External pressure
**Tube bundle - Tubes bundle**

| Material | = | SB-444 2 Solution annealed N06625 (high allowable stresses) |
| Governing Thickness | = | 4.19 mm | 0.165 in |
| Impact test exemption temperature | = | -198.00 °C | -324.40 °F |

*Note: MDMT according to UNF-65*
**Hemispherical head - Head**

According to: ASME VIII Div. 1 Ed. 2015, UG-32 / UG-33 - Metric Units

**Design data**

Internal design temperature  
\[ T = 420.00 ^\circ C \quad 788.00 ^\circ F \]

Internal design pressure  
\[ P = 14.22 \text{ MPa} \quad 2062.4 \text{ psi} \]

External design temperature  
\[ T_e = 20.00 ^\circ C \quad 68.00 ^\circ F \]

External design pressure  
\[ P_e = 0.10 \text{ MPa} \quad 14.9 \text{ psi} \]

Joint efficiency  
\[ E = 1.00 \]

**Material: SA-387 22 2 - Plate**

Allowable stress  
\[ S = 133.60 \text{ MPa} \quad 19377.0 \text{ psi} \]

Allowable stress at room temperature  
\[ S_{T} = 148.00 \text{ MPa} \quad 21465.6 \text{ psi} \]

**Geometry**

Inside diameter  
\[ D = 1312.00 \text{ mm} \quad 51.654 \text{ in} \]

Outside diameter  
\[ D_o = 1390.00 \text{ mm} \quad 54.724 \text{ in} \]

Inside crown radius (corroded)  
\[ L = 659.00 \text{ mm} \quad 25.945 \text{ in} \]

Adopted thickness  
\[ t = 39.00 \text{ mm} \quad 1.535 \text{ in} \]

Corrosion allowance  
\[ c = 3.00 \text{ mm} \quad 0.118 \text{ in} \]

External corrosion allowance  
\[ c_e = 0 \text{ mm} \quad 0 \text{ in} \]

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

Straight flange length  
\[ FL = 0 \text{ mm} \quad 0 \text{ in} \]

Forming strain (double curvature, e.g., heads)  
\[ \varepsilon_f = \frac{75 \cdot t}{(L + t/2)} = 4.33\% \]

**Internal pressure**

Allowable stress  
\[ S = 133.60 \text{ MPa} \quad 19377.0 \text{ psi} \]

Internal pressure  
\[ P_i = 14.22 \text{ MPa} \quad 2062.4 \text{ psi} \]

Overpressure due to static head  
\[ Ph = 0 \text{ MPa} \quad 0 \text{ psi} \]

Calculation pressure  
\[ P = P_i + Ph = 14.22 \text{ MPa} \quad 2062.4 \text{ psi} \]

Minimum required thickness  
\[ t_r = \frac{P L}{2S E - 0.2P} + c + c_e + c' = 38.45 \text{ mm} \quad 1.514 \text{ in} \]

Item service  
Service = NotSpecified

Minimum required thickness as per UG-16(b), including corrosion  
\[ t_{r \text{ UG-16(b)}} = 4.50 \text{ mm} \quad 0.177 \text{ in} \]

\[ t \geq t_{r \text{ UG-16(b)}}: \text{Ok} \]

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

New & cold  
\[ = 17.39 \text{ MPa} \quad 2522.3 \text{ psi} \]

Hot & corroded  
\[ = 14.44 \text{ MPa} \quad 2094.2 \text{ psi} \]
### External pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Te (External design temperature)</td>
<td>20.00 °C 68.00 °F</td>
</tr>
<tr>
<td>Factor A</td>
<td>A = 0.125/(R0/t) = 0.00650</td>
</tr>
<tr>
<td>Factor B</td>
<td>B = 134.90 MPa 19,565.2 psi</td>
</tr>
<tr>
<td>External pressure Pe</td>
<td>0.10 MPa 14.9 psi</td>
</tr>
<tr>
<td>External static head Ph</td>
<td>0 MPa 0 psi</td>
</tr>
<tr>
<td>Calculation pressure P</td>
<td>P = Pe + Ph = 0.10 MPa 14.9 psi</td>
</tr>
<tr>
<td>Outside radius Ro</td>
<td>692.00 mm 27.244 in</td>
</tr>
<tr>
<td>Maximum allowable external working pressure Pa</td>
<td>7.02 MPa 1017.8 psi</td>
</tr>
<tr>
<td>Minimum required thickness tr</td>
<td>4.89 mm 0.193 in</td>
</tr>
</tbody>
</table>

**Calculation:**

\[ P = P_e + P_h \]

\[ P = 0.10 \text{ MPa} + 0 \text{ MPa} = 0.10 \text{ MPa} \]

**Results:**

\[ P_a \geq P: \text{ Ok} \]

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

### Maximum allowable external pressures

<table>
<thead>
<tr>
<th>Type</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>New &amp; cold</td>
<td>7.60 MPa 1102.1 psi</td>
</tr>
<tr>
<td>Hot &amp; corroded</td>
<td>7.02 MPa 1017.8 psi</td>
</tr>
</tbody>
</table>
### 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. UCS-66</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>39.00 mm 1.535 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 0.85 °C 33.54 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 31.60 °C 88.88 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 3.00 mm 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 35.45 mm 1.396 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 39.00 mm 1.535 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t_e E^* = 0.98465$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= 30.75 °C 87.34 °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. UCS-66</td>
<td>A</td>
</tr>
<tr>
<td>Governing Thickness</td>
<td>39.00 mm 1.535 in</td>
</tr>
<tr>
<td>Reduction in MDMT based on available excess thickness</td>
<td>TR = 80.00 °C 176.00 °F</td>
</tr>
<tr>
<td>Unadjusted MDMT from table UCS-66</td>
<td>= 31.60 °C 88.88 °F</td>
</tr>
<tr>
<td>Joint efficiency</td>
<td>E = 1.00</td>
</tr>
<tr>
<td>Corrected joint efficiency</td>
<td>$E^* = \max(E, 0.8) = 1.00000$</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>c = 3.00 mm 0.118 in</td>
</tr>
<tr>
<td>Minimum required thickness in corroded condition</td>
<td>Tr = 1.89 mm 0.075 in</td>
</tr>
<tr>
<td>Nominal noncorroded thickness</td>
<td>Tn = 39.00 mm 1.535 in</td>
</tr>
<tr>
<td>Coincident ratio</td>
<td>$t_e E^* = 0.05263$</td>
</tr>
<tr>
<td>Adjusted MDMT from fig. UCS-66.1</td>
<td>= -48.40 °C -55.12 °F</td>
</tr>
</tbody>
</table>

### 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]