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# **Metallic industrial piping**

Part 3: Design and calculation

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English Version

## Metallic industrial piping - Part 3: Design and calculation

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et calcul

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## Contents

Page

|                                                                         |    |
|-------------------------------------------------------------------------|----|
| Foreword.....                                                           | 9  |
| 1 Scope .....                                                           | 11 |
| 2 Normative references .....                                            | 11 |
| 3 Terms, definitions, symbols and units .....                           | 12 |
| 3.1 Terms and definitions .....                                         | 12 |
| 3.2 Symbols and units .....                                             | 12 |
| 4 Basic design criteria.....                                            | 14 |
| 4.1 General.....                                                        | 14 |
| 4.2 Loadings .....                                                      | 14 |
| 4.2.1 General.....                                                      | 14 |
| 4.2.2 Combination of loads .....                                        | 15 |
| 4.2.3 Loads for dimensioning .....                                      | 15 |
| 4.2.4 Other loads to be taken into account .....                        | 16 |
| 4.2.5 Design conditions .....                                           | 18 |
| 4.3 Thickness .....                                                     | 20 |
| 4.4 Tolerances .....                                                    | 22 |
| 4.5 Joint coefficient .....                                             | 22 |
| 4.6 Dimensioning of piping components subject to pressure .....         | 22 |
| 5 Design stresses .....                                                 | 23 |
| 5.1 General.....                                                        | 23 |
| 5.2 Time-independent nominal design stress.....                         | 23 |
| 5.2.1 Steels other than austenitic steels.....                          | 23 |
| 5.2.2 Austenitic steels .....                                           | 23 |
| 5.2.3 Nickel and / or chromium alloy steels .....                       | 24 |
| 5.2.4 Steels castings.....                                              | 24 |
| 5.2.5 Additional requirements for steels with no specific control ..... | 24 |
| 5.3 Time-dependent nominal design stress .....                          | 25 |
| 5.3.1 General.....                                                      | 25 |
| 5.3.2 Steels .....                                                      | 25 |
| 5.3.3 Nickel and/or chromium alloy steels .....                         | 26 |
| 6 Design of piping components under internal pressure.....              | 26 |
| 6.1 Straight pipes.....                                                 | 26 |
| 6.2 Pipe bends and elbows .....                                         | 27 |
| 6.2.1 General.....                                                      | 27 |
| 6.2.2 Symbols .....                                                     | 27 |
| 6.2.3 Required wall thicknesses .....                                   | 27 |
| 6.3 Mitre bends .....                                                   | 29 |
| 6.3.1 General.....                                                      | 29 |
| 6.3.2 Symbols .....                                                     | 29 |
| 6.3.3 Effective radius of mitre bend .....                              | 30 |
| 6.3.4 Multiple mitre bends.....                                         | 31 |
| 6.3.5 Single mitre bends.....                                           | 31 |
| 6.3.6 Adjacent straight pipe sections of mitre bends .....              | 31 |
| 6.4 Reducers.....                                                       | 31 |
| 6.4.1 Conditions of applicability .....                                 | 31 |
| 6.4.2 Specific definitions .....                                        | 32 |
| 6.4.3 Specific symbols and abbreviations.....                           | 32 |
| 6.4.4 Conical shells.....                                               | 33 |

|        |                                                                                 |    |
|--------|---------------------------------------------------------------------------------|----|
| 6.4.5  | Junctions - general.....                                                        | 34 |
| 6.4.6  | Junction between the large end of a cone and a cylinder without a knuckle ..... | 34 |
| 6.4.7  | Junction between the large end of a cone and a cylinder with a knuckle.....     | 37 |
| 6.4.8  | Junction between the small end of a cone and a cylinder.....                    | 38 |
| 6.4.9  | Offset reducers .....                                                           | 40 |
| 6.4.10 | Special forged reducers.....                                                    | 40 |
| 6.5    | Flexible piping components .....                                                | 41 |
| 6.5.1  | General .....                                                                   | 41 |
| 6.5.2  | Expansion joints.....                                                           | 41 |
| 6.5.3  | Corrugated metal hose assemblies.....                                           | 42 |
| 6.6    | Bolted flange connections.....                                                  | 44 |
| 6.6.1  | General .....                                                                   | 44 |
| 6.6.2  | Symbols.....                                                                    | 44 |
| 6.6.3  | Standard flange .....                                                           | 44 |
| 6.6.4  | Non-standard flange.....                                                        | 45 |
| 7      | Design of ends under internal pressure .....                                    | 45 |
| 7.1    | Dished ends .....                                                               | 45 |
| 7.1.1  | Symbols.....                                                                    | 45 |
| 7.1.2  | Hemispherical ends.....                                                         | 46 |
| 7.1.3  | Torispherical ends.....                                                         | 46 |
| 7.1.4  | Ellipsoidal ends .....                                                          | 48 |
| 7.1.5  | Calculation of $\beta$ .....                                                    | 49 |
| 7.2    | Circular flat ends .....                                                        | 52 |
| 7.2.1  | General .....                                                                   | 52 |
| 7.2.2  | Symbols.....                                                                    | 52 |
| 7.2.3  | Unstayed flat circular ends welded to cylindrical shells/pipes .....            | 54 |
| 7.2.4  | Unstayed flat circular bolted ends.....                                         | 61 |
| 7.2.5  | Reinforcements of openings in unstayed flat ends.....                           | 67 |
| 8      | Openings and branch connections .....                                           | 69 |
| 8.1    | General .....                                                                   | 69 |
| 8.2    | Symbols.....                                                                    | 70 |
| 8.3    | Limitations .....                                                               | 71 |
| 8.3.1  | Thickness ratio .....                                                           | 71 |
| 8.3.2  | Openings in the vicinity of discontinuities .....                               | 72 |
| 8.3.3  | Types of reinforcement.....                                                     | 74 |
| 8.3.4  | Calculation method .....                                                        | 75 |
| 8.3.5  | Elliptical openings and oblique branch connections .....                        | 75 |
| 8.3.6  | Reinforcing pads .....                                                          | 77 |
| 8.3.7  | Dissimilar material of shell and reinforcements .....                           | 77 |
| 8.3.8  | Extruded outlets .....                                                          | 77 |
| 8.3.9  | Branches in bends or elbows .....                                               | 77 |
| 8.3.10 | Screwed-in branches .....                                                       | 77 |
| 8.4    | Isolated openings .....                                                         | 78 |
| 8.4.1  | General .....                                                                   | 78 |
| 8.4.2  | Unreinforced openings .....                                                     | 81 |
| 8.4.3  | Reinforced openings with $d_i/D_i < 0,8$ .....                                  | 81 |
| 8.4.4  | Reinforced single openings with $0,8 < d/D \leq 1,0$ .....                      | 87 |
| 8.5    | Adjacent openings .....                                                         | 87 |
| 8.5.1  | Unreinforced openings .....                                                     | 87 |
| 8.5.2  | Reinforced openings with $d/D \leq 0,8$ .....                                   | 87 |
| 8.6    | Design of special piping components.....                                        | 88 |
| 8.6.1  | Cylindrical Y-pieces .....                                                      | 88 |
| 8.6.2  | Spherical Y-pieces.....                                                         | 89 |
| 8.6.3  | Triform reinforced branches .....                                               | 90 |
| 9      | Design of piping components under external pressure.....                        | 91 |
| 9.1    | General .....                                                                   | 91 |
| 9.2    | Symbols and elastic stress limits.....                                          | 93 |

|        |                                                                                      |     |
|--------|--------------------------------------------------------------------------------------|-----|
| 9.2.1  | Symbols .....                                                                        | 93  |
| 9.2.2  | Elastic stress limits .....                                                          | 94  |
| 9.3    | Cylindrical pipes, elbows and mitre bends .....                                      | 94  |
| 9.3.1  | Determination of lengths .....                                                       | 94  |
| 9.3.2  | Interstiffener collapse .....                                                        | 96  |
| 9.3.3  | Overall collapse of stiffened pipes .....                                            | 98  |
| 9.3.4  | Stiffener stability .....                                                            | 99  |
| 9.3.5  | Heating/cooling channels .....                                                       | 102 |
| 9.4    | Reducers (conical shells) .....                                                      | 103 |
| 9.5    | Dished ends .....                                                                    | 104 |
| 9.5.1  | Hemispherical ends .....                                                             | 104 |
| 9.5.2  | Torispherical ends .....                                                             | 105 |
| 9.5.3  | Ellipsoidal ends .....                                                               | 105 |
| 10     | Design for cyclic loading .....                                                      | 106 |
| 10.1   | General .....                                                                        | 106 |
| 10.2   | Exemption from detailed fatigue analysis .....                                       | 106 |
| 10.3   | Fatigue design for cyclic pressure .....                                             | 107 |
| 10.3.1 | Equivalent full load cycles .....                                                    | 107 |
| 10.3.2 | Simplified fatigue analysis .....                                                    | 107 |
| 10.4   | Fatigue design for thermal gradients .....                                           | 121 |
| 10.4.1 | General .....                                                                        | 121 |
| 10.4.2 | Design guidance .....                                                                | 121 |
| 10.5   | Fatigue design for combined loads .....                                              | 121 |
| 11     | Integral attachments .....                                                           | 122 |
| 11.1   | General .....                                                                        | 122 |
| 11.2   | Allowable stresses .....                                                             | 122 |
| 11.3   | Symbols .....                                                                        | 123 |
| 11.4   | Hollow circular attachments .....                                                    | 125 |
| 11.4.1 | Limitations .....                                                                    | 125 |
| 11.4.2 | Preliminary calculations .....                                                       | 125 |
| 11.4.3 | Analysis of attachments welded to pipe with a full penetration weld .....            | 127 |
| 11.4.4 | Analysis of attachments welded to pipe with fillet or partial penetration weld ..... | 128 |
| 11.5   | Rectangular attachments .....                                                        | 128 |
| 11.5.1 | Limitations .....                                                                    | 128 |
| 11.5.2 | Preliminary calculations .....                                                       | 128 |
| 11.5.3 | Analysis of attachments welded to pipe with a full penetration weld .....            | 130 |
| 11.5.4 | Analysis of attachments welded to pipe with fillet or partial penetration weld ..... | 130 |
| 11.6   | Stress analysis of the run pipe .....                                                | 131 |
| 11.7   | Shear stress analysis in attachment .....                                            | 132 |
| 11.7.1 | Hollow circular attachments .....                                                    | 132 |
| 11.7.2 | Rectangular attachments .....                                                        | 133 |
| 11.8   | Alternative calculation methods .....                                                | 133 |
| 12     | Flexibility analysis and acceptance criteria .....                                   | 133 |
| 12.1   | Basic conditions .....                                                               | 133 |
| 12.1.1 | General .....                                                                        | 133 |
| 12.1.2 | Loading conditions .....                                                             | 133 |
| 12.1.3 | Allowable stresses .....                                                             | 133 |
| 12.2   | Piping flexibility .....                                                             | 135 |
| 12.2.1 | General .....                                                                        | 135 |
| 12.2.2 | Basic conditions .....                                                               | 135 |
| 12.2.3 | Displacement strains .....                                                           | 136 |
| 12.2.4 | Displacement stresses .....                                                          | 137 |
| 12.2.5 | Stress range .....                                                                   | 137 |
| 12.2.6 | Cold pull .....                                                                      | 138 |
| 12.2.7 | Properties for flexibility analysis .....                                            | 138 |
| 12.2.8 | Supporting conditions .....                                                          | 139 |
| 12.2.9 | Expansion joints .....                                                               | 140 |

|         |                                                                   |     |
|---------|-------------------------------------------------------------------|-----|
| 12.2.10 | Flexibility analysis .....                                        | 140 |
| 12.3    | Flexibility analysis .....                                        | 142 |
| 12.3.1  | General .....                                                     | 142 |
| 12.3.2  | Stress due to sustained loads .....                               | 142 |
| 12.3.3  | Stress due to sustained and occasional or exceptional loads ..... | 143 |
| 12.3.4  | Stress range due to thermal expansion and alternating loads ..... | 144 |
| 12.3.5  | Additional conditions for the creep range .....                   | 145 |
| 12.3.6  | Stresses due to a single non-repeated support movement .....      | 145 |
| 12.3.7  | Determination of resultant moments .....                          | 145 |
| 12.3.8  | Reactions .....                                                   | 148 |
| 12.4    | Fatigue analysis .....                                            | 148 |
| 12.5    | Vibration .....                                                   | 148 |
| 13      | Supports .....                                                    | 148 |
| 13.1    | General requirements .....                                        | 148 |
| 13.1.1  | General .....                                                     | 148 |
| 13.1.2  | Classification of supports .....                                  | 149 |
| 13.1.3  | Additional definitions .....                                      | 149 |
| 13.1.4  | Boundaries .....                                                  | 150 |
| 13.1.5  | Welded support attachments .....                                  | 152 |
| 13.2    | Material requirements .....                                       | 154 |
| 13.3    | Design .....                                                      | 154 |
| 13.3.1  | General .....                                                     | 154 |
| 13.3.2  | Design temperatures for support components .....                  | 155 |
| 13.3.3  | Detail design .....                                               | 157 |
| 13.3.4  | Buckling .....                                                    | 158 |
| 13.3.5  | Support location .....                                            | 158 |
| 13.3.6  | Determination of component sizes .....                            | 158 |
| 13.4    | Connections .....                                                 | 159 |
| 13.4.1  | Welded connections .....                                          | 159 |
| 13.4.2  | Bolted connections .....                                          | 160 |
| 13.5    | Design requirements for special components .....                  | 160 |
| 13.5.1  | Constant load hangers and supports .....                          | 160 |
| 13.5.2  | Variable load hangers and supports .....                          | 161 |
| 13.5.3  | Rigid struts .....                                                | 162 |
| 13.5.4  | Shock arrestors (shock absorber, snubber) .....                   | 163 |
| 13.5.5  | Sliding supports .....                                            | 164 |
| 13.5.6  | Anchors .....                                                     | 164 |
| 13.6    | Documentation of supports .....                                   | 164 |
| 13.7    | Marking of supports .....                                         | 164 |
| Annex A | (informative) Dynamic analysis .....                              | 165 |
| A.1     | General .....                                                     | 165 |
| A.2     | Analysis by calculation .....                                     | 165 |
| A.2.1   | Seismic events .....                                              | 165 |
| A.2.2   | Rapid valve closure .....                                         | 169 |
| A.2.3   | Flow induced vibration .....                                      | 172 |
| A.2.4   | Safety valve discharge .....                                      | 174 |
| A.2.5   | Allowable stresses .....                                          | 176 |
| A.3     | Alternative means of design verification .....                    | 176 |
| A.3.1   | Comparative studies .....                                         | 176 |
| A.3.2   | Full scale testing .....                                          | 177 |
| A.3.3   | Reduced scale testing .....                                       | 177 |
| Annex B | (normative) More accurate calculation of bends and elbows .....   | 178 |
| B.1     | General .....                                                     | 178 |
| B.2     | Symbols and units .....                                           | 178 |
| B.3     | Required wall thickness .....                                     | 179 |
| B.4     | Calculation .....                                                 | 180 |
| B.4.1   | Calculation of wall thickness .....                               | 180 |

|                |                                                                                 |            |
|----------------|---------------------------------------------------------------------------------|------------|
| <b>B.4.2</b>   | <b>Stress calculation .....</b>                                                 | <b>182</b> |
| <b>Annex C</b> | <b>(informative) Expansion joints .....</b>                                     | <b>186</b> |
| <b>C.1</b>     | <b>Incorporation of expansion joints into piping systems .....</b>              | <b>186</b> |
| <b>C.1.1</b>   | <b>General.....</b>                                                             | <b>186</b> |
| <b>C.1.2</b>   | <b>Types of expansion joints.....</b>                                           | <b>186</b> |
| <b>C.1.3</b>   | <b>Design of expansion joints .....</b>                                         | <b>187</b> |
| <b>C.1.4</b>   | <b>Designing with expansion joints.....</b>                                     | <b>188</b> |
| <b>C.1.5</b>   | <b>Analyses and calculation.....</b>                                            | <b>189</b> |
| <b>C.1.6</b>   | <b>Cold pull.....</b>                                                           | <b>190</b> |
| <b>C.2</b>     | <b>Maximum spacing for unrestrained axially compensated straight runs .....</b> | <b>190</b> |
| <b>C.2.1</b>   | <b>General.....</b>                                                             | <b>190</b> |
| <b>C.2.2</b>   | <b>Calculation rules .....</b>                                                  | <b>190</b> |
| <b>C.2.3</b>   | <b>Maximum spacing for defined conditions.....</b>                              | <b>192</b> |
| <b>C.3</b>     | <b>Indication for the design of expansion joints .....</b>                      | <b>193</b> |
| <b>C.3.1</b>   | <b>General.....</b>                                                             | <b>193</b> |
| <b>C.3.2</b>   | <b>Design data, Symbols .....</b>                                               | <b>194</b> |
| <b>C.3.3</b>   | <b>Design and calculation.....</b>                                              | <b>195</b> |
| <b>C.3.4</b>   | <b>Information for the system analyst.....</b>                                  | <b>197</b> |
| <b>Annex D</b> | <b>(normative) Flanges .....</b>                                                | <b>198</b> |
| <b>D.1</b>     | <b>Purpose.....</b>                                                             | <b>198</b> |
| <b>D.2</b>     | <b>Specific terms and definitions.....</b>                                      | <b>198</b> |
| <b>D.3</b>     | <b>Specific symbols and abbreviations.....</b>                                  | <b>199</b> |
| <b>D.4</b>     | <b>General.....</b>                                                             | <b>200</b> |
| <b>D.4.1</b>   | <b>Introduction .....</b>                                                       | <b>200</b> |
| <b>D.4.2</b>   | <b>Use of standard flanges without calculation .....</b>                        | <b>201</b> |
| <b>D.4.3</b>   | <b>Bolting.....</b>                                                             | <b>201</b> |
| <b>D.4.4</b>   | <b>Flange construction.....</b>                                                 | <b>203</b> |
| <b>D.4.5</b>   | <b>Machining .....</b>                                                          | <b>203</b> |
| <b>D.4.6</b>   | <b>Gaskets .....</b>                                                            | <b>203</b> |
| <b>D.5</b>     | <b>Narrow face gasketed flanges .....</b>                                       | <b>204</b> |
| <b>D.5.1</b>   | <b>General.....</b>                                                             | <b>204</b> |
| <b>D.5.2</b>   | <b>Bolt loads and areas.....</b>                                                | <b>207</b> |
| <b>D.5.3</b>   | <b>Flange moments .....</b>                                                     | <b>208</b> |
| <b>D.5.4</b>   | <b>Flange stresses and stress limits .....</b>                                  | <b>209</b> |
| <b>D.5.5</b>   | <b>Narrow face flanges subject to external pressure .....</b>                   | <b>215</b> |
| <b>D.5.6</b>   | <b>Lap joints .....</b>                                                         | <b>215</b> |
| <b>D.5.7</b>   | <b>Split ring flanges.....</b>                                                  | <b>218</b> |
| <b>D.6</b>     | <b>Full face flanges with soft ring type gaskets .....</b>                      | <b>219</b> |
| <b>D.6.1</b>   | <b>Specific symbols and abbreviations.....</b>                                  | <b>220</b> |
| <b>D.6.2</b>   | <b>Bolt loads and areas.....</b>                                                | <b>220</b> |
| <b>D.6.3</b>   | <b>Flange design.....</b>                                                       | <b>221</b> |
| <b>D.6.4</b>   | <b>Full face flanges subject to external pressure .....</b>                     | <b>222</b> |
| <b>D.7</b>     | <b>Seal welded flanges.....</b>                                                 | <b>222</b> |
| <b>D.8</b>     | <b>Reverse narrow face flanges .....</b>                                        | <b>223</b> |
| <b>D.8.1</b>   | <b>Internal pressure.....</b>                                                   | <b>223</b> |
| <b>D.8.2</b>   | <b>External pressure.....</b>                                                   | <b>225</b> |
| <b>D.9</b>     | <b>Reverse full face flanges.....</b>                                           | <b>225</b> |
| <b>D.9.1</b>   | <b>General.....</b>                                                             | <b>225</b> |
| <b>D.9.2</b>   | <b>Design following method of D.5.....</b>                                      | <b>225</b> |
| <b>D.9.3</b>   | <b>Design following method of D.6.....</b>                                      | <b>227</b> |
| <b>D.10</b>    | <b>Full face flanges with metal to metal contact .....</b>                      | <b>229</b> |
| <b>D.10.1</b>  | <b>General.....</b>                                                             | <b>229</b> |
| <b>D.10.2</b>  | <b>Specific symbols and abbreviations.....</b>                                  | <b>229</b> |
| <b>D.10.3</b>  | <b>Design .....</b>                                                             | <b>230</b> |
| <b>Annex E</b> | <b>(normative) Design of branch connections in piping accessories.....</b>      | <b>232</b> |
| <b>E.1</b>     | <b>Scope .....</b>                                                              | <b>232</b> |
| <b>E.1.1</b>   | <b>General.....</b>                                                             | <b>232</b> |

|         |                                                                                                                                                                       |     |
|---------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----|
| E.2     | Reinforcement .....                                                                                                                                                   | 234 |
| E.2.1   | Angles and areas .....                                                                                                                                                | 234 |
| E.2.2   | The following condition shall be satisfied: .....                                                                                                                     | 234 |
| E.3     | Flexibility analysis .....                                                                                                                                            | 236 |
| Annex F | (informative) Testing during operation in the case of cyclic loading .....                                                                                            | 237 |
| F.1     | Testing during operation .....                                                                                                                                        | 237 |
| F.2     | Measures to be taken when the calculated fatigue life has been reached .....                                                                                          | 237 |
| Annex G | (informative) Physical properties of steels .....                                                                                                                     | 238 |
| G.1     | General .....                                                                                                                                                         | 238 |
| G.2     | Physical properties .....                                                                                                                                             | 238 |
| G.2.1   | Density .....                                                                                                                                                         | 238 |
| G.2.2   | Differential coefficient of linear expansion .....                                                                                                                    | 239 |
| G.2.3   | Specific thermal capacity .....                                                                                                                                       | 239 |
| G.2.4   | Thermal diffusivity .....                                                                                                                                             | 239 |
| G.2.5   | Poisson's ratio .....                                                                                                                                                 | 239 |
| G.3     | Physical properties of steels .....                                                                                                                                   | 239 |
| Annex H | (normative) Flexibility characteristics, flexibility and stress intensification factors and section moduli of piping components and geometrical discontinuities ..... | 245 |
| Annex I | (informative) Production testing of spring supports and shock arrestors (shock absorbers) .....                                                                       | 255 |
| I.1     | Constant load supports .....                                                                                                                                          | 255 |
| I.2     | Variable spring supports .....                                                                                                                                        | 255 |
| I.3     | Shock arrestors .....                                                                                                                                                 | 255 |
| Annex J | (normative) Type testing of support components .....                                                                                                                  | 260 |
| Annex K | (informative) Attachment of supports to structures .....                                                                                                              | 261 |
| K.1     | Attachment of supports to concrete structures .....                                                                                                                   | 261 |
| K.2     | Attachment to metallic structures .....                                                                                                                               | 262 |
| K.2.1   | Standard bolts .....                                                                                                                                                  | 262 |
| K.2.2   | Friction grip bolts .....                                                                                                                                             | 262 |
| K.2.3   | Welding .....                                                                                                                                                         | 262 |
| Annex L | (informative) Buckling of linear type supports .....                                                                                                                  | 263 |
| L.1     | General .....                                                                                                                                                         | 263 |
| L.2     | Symbols .....                                                                                                                                                         | 263 |
| L.3     | Basic formulae .....                                                                                                                                                  | 264 |
| L.4     | Allowable compressive stress .....                                                                                                                                    | 264 |
| L.5     | Buckling length .....                                                                                                                                                 | 265 |
| Annex M | (informative) Design guidance for structural components .....                                                                                                         | 267 |
| M.1     | Linear type components subjected to bending .....                                                                                                                     | 267 |
| M.1.1   | General .....                                                                                                                                                         | 267 |
| M.1.2   | Supplementary verifications for linear type supports .....                                                                                                            | 267 |
| M.2     | Stability of plate type supports .....                                                                                                                                | 269 |
| M.3     | Anchorage plates or equivalent anchorage components .....                                                                                                             | 269 |
| M.3.1   | General .....                                                                                                                                                         | 269 |
| M.3.2   | Design of simple anchorage plates .....                                                                                                                               | 269 |
| M.3.3   | Fixing plates with stiffening gussets .....                                                                                                                           | 270 |
| M.3.4   | Load calculations for anchorages fixed in concrete .....                                                                                                              | 270 |
| Annex N | (normative) Documentation of supports .....                                                                                                                           | 271 |
| Annex O | (normative) Alternative method for checking branch connections .....                                                                                                  | 273 |
| O.1     | Scope .....                                                                                                                                                           | 273 |
| O.2     | Symbols .....                                                                                                                                                         | 273 |

|              |                                                                                                                         |     |
|--------------|-------------------------------------------------------------------------------------------------------------------------|-----|
| O.3          | Design and checking of the branch connection.....                                                                       | 275 |
| O.3.1        | Limit value for the load due to pressure only for straight pipes without opening .....                                  | 275 |
| O.3.2        | Determination of the minimum thicknesses under loading due to pressure only .....                                       | 276 |
| O.3.3        | Checking of the thicknesses selected for the combination of pressure loading and loadings due to external loads .....   | 276 |
| Annex P      | (informative) Bolted flange connections — Application of EN 1591.....                                                   | 327 |
| P.1          | Introduction .....                                                                                                      | 327 |
| P.2          | Scope .....                                                                                                             | 328 |
| P.2.1        | General.....                                                                                                            | 328 |
| P.2.2        | Materials .....                                                                                                         | 328 |
| P.2.3        | Loadings .....                                                                                                          | 328 |
| P.2.4        | Assumptions .....                                                                                                       | 328 |
| P.3          | Application of EN 1591 .....                                                                                            | 329 |
| P.3.1        | Calculations.....                                                                                                       | 329 |
| P.3.2        | Gasket coefficients .....                                                                                               | 329 |
| P.3.3        | Tightening.....                                                                                                         | 330 |
| Annex Q      | (informative) Simplified pipe stress analysis.....                                                                      | 389 |
| Q.1          | General.....                                                                                                            | 389 |
| Q.2          | Simplified procedure .....                                                                                              | 389 |
| Q.2.1        | General.....                                                                                                            | 389 |
| Q.2.2        | Specification of allowable spacing of supports .....                                                                    | 389 |
| Q.2.3        | Check of elasticity .....                                                                                               | 389 |
| Q.3          | Explanatory notes for Table Q.1.....                                                                                    | 391 |
| Q.4          | Symbols .....                                                                                                           | 393 |
| Q.5          | Indices $f_L$ .....                                                                                                     | 393 |
| Q.6          | Explanatory notes to Q.2.2 .....                                                                                        | 394 |
| Q.6.1        | Specification of allowable spacing of supports .....                                                                    | 394 |
| Q.7          | Conversion of the allowable lengths .....                                                                               | 395 |
| Q.7.1        | Other support conditions.....                                                                                           | 395 |
| Q.7.2        | Other parameters .....                                                                                                  | 395 |
| Q.8          | Additional single loads .....                                                                                           | 396 |
| Q.8.1        | General.....                                                                                                            | 396 |
| Q.9          | Explanatory note on Figure Q.2 .....                                                                                    | 399 |
| Q.9.1        | General.....                                                                                                            | 399 |
| Q.9.2        | Required pipe leg length $L_1$ , for $f_1$ from the nomogram .....                                                      | 401 |
| Q.9.3        | Required pipe leg length $L_2$ , for $f_2$ from the nomogram .....                                                      | 401 |
| Annex Y      | (informative) History of EN 13480-3 .....                                                                               | 406 |
| Annex ZA     | (informative) Relationship between this European Standard and the essential requirements of EU Directive 97/23/EC ..... | 407 |
| Bibliography | .....                                                                                                                   | 408 |

## Foreword

This document (EN 13480-3:2012) has been prepared by Technical Committee CEN/TC 267 “Industrial piping and pipelines”, the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by December 2012, and conflicting national standards shall be withdrawn at the latest by December 2012.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EC Directive(s).

For relationship with EC Directive(s), see informative Annex ZA, which is an integral part of this document.

In this European Standard the Annexes A, C, F, G, I, K, L, M, P and Q are informative and the Annexes B, D, E, H, J, N and O are normative.

This European Standard EN 13480 for metallic industrial piping consists of eight interdependent and not dissociable Parts which are:

- *Part 1: General;*
- *Part 2: Materials;*
- *Part 3: Design and calculation;*
- *Part 4: Fabrication and installation;*
- *Part 5: Inspection and testing;*
- *Part 6: Additional requirements for buried piping;*
- *CEN/TR 13480-7: Guidance on the use of conformity assessment procedures;*
- *Part 8: Additional requirements for aluminium and aluminium alloy piping.*

Although these Parts may be obtained separately, it should be recognised that the Parts are interdependent. As such the manufacture of metallic industrial piping requires the application of all the relevant Parts in order for the requirements of the Standard to be satisfactorily fulfilled.

This European Standard will be maintained by a Maintenance MHD working group whose scope of working is limited to corrections and interpretations related to EN 13480.

The contact to submit queries can be found at <http://www.unm.fr> ([en13480@unm.fr](mailto:en13480@unm.fr)). A form for submitting questions can be downloaded from the link to the MHD website. After subject experts have agreed an answer, the answer will be communicated to the questioner. Corrected pages will be given specific issue number and issued by CEN according to CEN Rules. Interpretation sheets will be posted on the website of the MHD.

This document supersedes EN 13480-3:2002+A1:2005+A2:2006+A3:2009+A4:2010+A5:2012. This new edition incorporates the Amendments/the corrigenda which have been approved previously by CEN members, the corrected pages up to Issue 17 without any further technical change. Annex Y provides details of significant technical changes between this European Standard and the previous edition.

Amendments to this new edition may be issued from time to time and then used immediately as alternatives to rules contained herein. It is intended to deliver a new Issue of EN 13480:2012 each year, consolidating these Amendments and including other identified corrections. Issue 4 (2016-07) includes the corrected pages listed in Annex Y.

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, Former Yugoslav Republic of Macedonia, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and the United Kingdom.

## 1 Scope

This Part of this European Standard specifies the design and calculation of industrial metallic piping systems, including supports, covered by EN 13480.

## 2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

EN 287-1:2004+A2:2006, *Qualification test of welders — Fusion welding — Part 1: Steels*

EN 1515-2:2001, *Flanges and their joints — Bolting — Part 2: Combination of flange and bolting materials for steel flanges PN designated*

EN 1515-3:2005, *Flanges and their joints — Bolting — Part 3: Classification of bolt materials for steel flanges, Class designated*

EN 1515-4:2010, *Flanges and their joints — Bolting — Part 4: Selection of bolting for equipment subject to the Pressure Equipment Directive 97/23/EC*

EN 1591-1:2001+A1:2009+AC:2011, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*

EN 1591-2:2008, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 2: Gasket parameters*

EN 1993 (all parts), *Eurocode 3: Design of steel structures*

EN 10204:2004, *Metallic products — Types of inspection documents*

EN 12953-3:2002, *Shell boilers — Part 3: Design and calculation for pressure parts*

EN 13445-3:2009, *Unfired pressure vessels — Part 3: Design*

EN 13480-1:2012, *Metallic industrial piping — Part 1: General*

EN 13480-2:2012, *Metallic industrial piping — Part 2: Materials*

EN 13480-4:2012, *Metallic industrial piping — Part 4: Fabrication and installation*

EN 13480-5:2012, *Metallic industrial piping — Part 5: Inspection and testing*

EN ISO 15614-1:2004, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 1: Arc and gas welding of steels and arc welding of nickel and nickel alloys* (ISO 15614-1:2004)

EN ISO 5817:2007, *Welding — Fusion-welded joints in steel, nickel, titanium and their alloys (beam welding excluded) — Quality levels for imperfections* (ISO 5817:2003, corrected version:2005, including Technical Corrigendum 1:2006)

### **3 Terms, definitions, symbols and units**

#### **3.1 Terms and definitions**

For the purposes of this Part of this European Standard, the terms and definitions given in EN 13480-1 apply.

#### **3.2 Symbols and units**

For the purposes of this Part of this European Standard, the symbols and units given in EN 13480-1 and in Table 3.2-1 apply.

Specific symbols are defined in the relevant sub-clauses.

Table 3.2-1 — General symbols and units

| Symbol        | Description                                                                                                                                                                                                                                                                             | Unit                     |
|---------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|--------------------------|
| $PS^a$        | maximum allowable pressure                                                                                                                                                                                                                                                              | bar                      |
| $R, r^b$      | radii                                                                                                                                                                                                                                                                                   | mm                       |
| $R_{eH\ t}$   | minimum specified value of upper yield strength at calculation temperature when this temperature is greater than the room temperature                                                                                                                                                   | MPa (N/mm <sup>2</sup> ) |
| $S_1$         | mean value of the stress which leads to a 1 % creep elongation in 100 000 h                                                                                                                                                                                                             | MPa (N/mm <sup>2</sup> ) |
| $S_2$         | mean value of the stress which leads to a 1 % creep elongation in 200 000 h                                                                                                                                                                                                             | MPa (N/mm <sup>2</sup> ) |
| $S_{R\ T\ t}$ | mean value of creep rupture strength as indicated by the standards, for the material in question at the considered temperature, $t$ , and for the considered lifetime $T$ (in hours) whereby the dispersion band of the results does not deviate by more than 20 % from the mean value. | MPa (N/mm <sup>2</sup> ) |
| $TS$          | maximum allowable temperature                                                                                                                                                                                                                                                           | °C                       |
| $Z$           | section modulus for a pipe                                                                                                                                                                                                                                                              | mm <sup>3</sup>          |
| $c_0$         | corrosion or erosion allowance (see Figure 4.3-1)                                                                                                                                                                                                                                       | mm                       |
| $c_1$         | absolute value of the negative tolerance taken from the material standard (see Figure 4.3-1)                                                                                                                                                                                            | mm                       |
| $c_2$         | thinning allowance for possible thinning during manufacturing process (see Figure 4.3-1)                                                                                                                                                                                                | mm                       |
| $e_a$         | analysis thickness of a component used for the check of the strength (see Figure 4.3-1)                                                                                                                                                                                                 | mm                       |
| $e_n$         | nominal thickness on drawings (see Figure 4.3-1)                                                                                                                                                                                                                                        | mm                       |
| $e_{ord}$     | ordered thickness (see Figure 4.3-1)                                                                                                                                                                                                                                                    | mm                       |
| $e_r$         | minimum required thickness with allowances and tolerances (see Figure 4.3-1)                                                                                                                                                                                                            | mm                       |
| $f$           | design stress (see clause 5)                                                                                                                                                                                                                                                            | MPa (N/mm <sup>2</sup> ) |
| $f_{cr}$      | Design stress in the creep range                                                                                                                                                                                                                                                        | MPa (N/mm <sup>2</sup> ) |
| $f_f$         | Design stress for flexibility analysis                                                                                                                                                                                                                                                  | MPa (N/mm <sup>2</sup> ) |
| $p_c$         | calculation pressure (see 4.2.3.4)                                                                                                                                                                                                                                                      | MPa (N/mm <sup>2</sup> ) |
| $p_o$         | operating pressure (see 4.2.3.1)                                                                                                                                                                                                                                                        | MPa (N/mm <sup>2</sup> ) |
| $t_c$         | calculation temperature (see 4.2.3.5)                                                                                                                                                                                                                                                   | °C                       |
| $t_o$         | operating temperature (see 4.2.3.2)                                                                                                                                                                                                                                                     | °C                       |
| $z$           | joint coefficient (see 4.5)                                                                                                                                                                                                                                                             | -                        |
| $\varepsilon$ | additional thickness resulting from the selection of the ordered thickness (see Figure 4.3-1)                                                                                                                                                                                           | mm                       |

<sup>a</sup> All pressures for calculation purposes are in MPa (N/mm<sup>2</sup>) and  $PS$  is in bar.

<sup>b</sup> The following subscripts apply :

i inside

m mean

o outside

## **4 Basic design criteria**

### **4.1 General**

The calculation rules in this Part shall apply for operating and testing conditions as well as preset, cold pull conditions, flushing and cleaning conditions.

The scope of each of the calculation rules is limited on a case by case basis by geometrical characteristics which shall be taken into account for each component, loading, failure mode and material property.

**NOTE** Where this European Standard does not indicate a calculation rule, it is the designer's responsibility to use widely acknowledged calculation rules, or experimental methods to justify the dimensions and thicknesses selected.

Elastic calculation methods shall be used in this Part, although some components might exhibit plastic behaviour.

If shaping and assembly techniques specific to the pressure vessel industry are used for large diameter pipes, the rules applicable to such techniques shall be those relating to the design of shells for pressure vessels. For the general stability, the requirements of EN 13480 remain applicable if the structure, as a whole, behaves according to the beam theory.

For temporary piping, e.g. flushing, cleaning and blow-through systems, the nominal design stress for the design conditions shall be used.

### **4.2 Loadings**

#### **4.2.1 General**

Any piping system is subjected to a number of loadings during its lifetime. These loadings can be one or a combination of the following loads:

- internal and/or external pressure;
- temperature;
- weight of piping and contents;
- climatic loads;
- dynamic effects of the fluid;
- movements of the ground and buildings;
- vibrations;
- earthquakes.

**NOTE 1** This list is not exhaustive.

**NOTE 2** Explanations of these loads are given in 4.2.3 and 4.2.4.

## 4.2.2 Combination of loads

The loads and their possible combinations, given in 4.2.5.1 to 4.2.5.4, shall be taken into account at the design stage of the piping system and its supports. Some unlikely combinations may be ruled out following a study considering both the likelihood of their occurrence, failure of the fluid containment boundary and the health and safety consequences.

Where a piping system is subjected to more than one pressure/temperature condition, the greatest calculated thickness determined for these conditions shall be used.

## 4.2.3 Loads for dimensioning

### 4.2.3.1 Operating pressure

The operating pressure,  $p_o$ , shall be below the maximum allowable pressure,  $PS$ , specified for the piping system.

### 4.2.3.2 Operating temperature

The operating temperature,  $t_o$ , shall be below the maximum allowable temperature,  $TS$ , specified for the piping system.

### 4.2.3.3 Sets

The set ( $p_o$ ,  $t_o$ ) to be considered for the dimensioning of the elements of a piping system shall correspond to the most severe conditions of pressure and temperature which prevail simultaneously over a long time in the piping section under consideration. Thus for the thickness calculation of a component, the simultaneous conditions of pressure and temperature to be considered are the conditions which lead to the greatest thickness.

For all piping system elements, an allowable maximum pressure, based on

- a) specified material (mechanical properties),
- b) a given temperature,

can be easily determined by taking into account the applicable safety factors.

Temporary deviations e.g. due to pressure surge or operation of control release valve (safety valve) shall not be taken into account if the calculated stresses from such variations do not exceed the allowable stress by more than 10 % for less than 10 % of any 24 h operating period.

### 4.2.3.4 Calculation pressure

For all pressure temperature conditions ( $p_o$ ,  $t_o$ ) specified in 4.2.3.3 calculation pressures  $p_c$  shall be determined.

The calculation pressure  $p_c$  shall be not less than the associated operating pressure  $p_o$ , taking into account the adjustments of the safety devices. The conditions ( $p_o$ ,  $t_o$ ) resulting in the greatest wall thickness shall be considered with both of the following minimum conditions:

- 1)  $p_c = p_o = PS$  with the associated  $t_c$  as defined in 4.2.3.5;
- 2)  $t_c$  as defined in 4.2.3.5 for  $t_o = TS$  with the associated  $p_c = p_o$ .

NOTE If there is a condition where  $p_o = PS$  and  $t_o = TS$  only this condition has to be calculated.

When the calculation temperature  $t_c$  is such that the creep rupture strength characteristics are relevant for the determination of the nominal design stress, the calculation pressure shall be considered equal to the operating pressure ( $p_o$ ) which is associated with the corresponding temperature ( $t_o$ ).

#### **4.2.3.5 Calculation temperature**

The calculation temperature,  $t_c$ , shall be the maximum temperature likely to be reached at the mid-thickness of the piping, under normal operating conditions, at the calculation pressure  $p_c$ . The calculation temperature shall be determined as indicated below. Any heat transfer calculation shall be performed on the assumption that there is no heat loss due to wind.

- a) For externally uninsulated and internally unlined piping components, the calculation temperature shall be as follows :
- 1) For fluid temperatures below 40 °C, the calculation temperature for the component shall be taken as the fluid temperature;
  - 2) For fluid temperatures of 40 °C and above, unless a lower average wall temperature is determined by test or heat transfer calculation, the calculation temperature for uninsulated components shall be not less than the following values, but not less than 40°C:
    - i) 95 % of the fluid temperature for valves, pipes, ends, welding fittings, and other components having wall thickness comparable to that of the pipe;
    - ii) 90 % of the fluid temperature for flanges (except lap joint flanges) including those on fittings and valves;
    - iii) 85 % of the fluid temperature for lap joint flanges;
    - iv) 80 % of the fluid temperature for bolting.
- b) For externally insulated piping components, the component calculation temperature shall be the fluid temperature unless calculations, tests, or service experience based on measurements, support the use of another temperature. Where piping is heated or cooled by tracing or jacketing, this effect shall be considered in establishing component calculation temperatures;
- c) For lined piping components, the component calculation temperature shall be specified or based on heat transfer calculations or tests taking into account the fluid temperature and the characteristics of the lining;

NOTE The lining may be used for insulation.

- d) When the calculated thickness is determined by the creep strength properties, the operating pressure ( $p_o$ ) and the operating temperature ( $t_o$ ) can exceed temporarily the values used for this calculation, see 4.2.5.2.1 and 12.3.3.

#### **4.2.4 Other loads to be taken into account**

##### **4.2.4.1 Weight of piping and contents**

Gravity loads acting on the piping system shall be taken into account in the design and include:

- the weight of the pipe, fittings, valves and insulation;

- the weight of the conveyed fluid;
- the weight of the test fluid.

#### **4.2.4.2 Climatic loads**

When the piping system is exposed to climatic loads, these shall be taken into account. The maximum loads shall be determined in accordance with the actual local climatic and piping system exposure conditions.

#### **4.2.4.3 Dynamic effects of the fluid**

The piping system shall be designed in order to avoid the accidental dynamic effects due to the fluid. Where such effects are impossible to avoid, they shall be taken into account. Where these effects result from a direct consequence of the process, or the use of the equipment supplied by the purchaser, they shall be defined quantitatively in the purchase specification.

The reaction forces of safety valves during their operation shall be taken into account. If these valves are not supplied by the piping installer, the purchaser shall define the reaction forces and their directions.

For the assessment of the dynamic effects of the fluid, see Annex A.

#### **4.2.4.4 Movements of the ground and of the buildings**

When such movements (movements of the ground, differential settlements of the buildings) are likely to occur during the lifetime of the piping system, the values to be used in the design shall be provided in the purchase specification.

#### **4.2.4.5 Vibrations**

The piping system shall be designed and supported so as to eliminate or minimize excessive and harmful effects of vibration which may arise from such sources as impact, pressure pulsations, resonance due to compressors, and wind loads.

Where vibrations can occur during operation, the piping layout shall be studied and supports, dampers, restraints, anchors etc. shall be provided in accordance with clause 13 to remove these effects. If such studies are insufficient, it shall be necessary to carry out a specific vibration analysis to ensure the piping system is not over-stressed.

NOTE It is not necessary to provide written proof of the vibration analysis.

#### **4.2.4.6 Earthquake**

If required in a specification, the piping system shall be designed to withstand seismic loads. The specification shall give details relating to the characteristics of the seismic conditions to be taken into account.

NOTE Guidance for assessment of the earthquake effects is given in Annex A.

## **4.2.5 Design conditions**

### **4.2.5.1 Normal operating conditions**

Normal operating conditions shall be the conditions met in steady state at constant power and in transient working conditions corresponding to normal operating processes. Full load, partial load and shut-down/shut-down conditions shall be examined together with the associated start-up, switching and shut-down operations.

For the calculation under normal operating conditions, each of the following conditions shall be examined:

- internal pressure and/or external pressure, including static head, where relevant;
- dead weight of piping, including internal structures and attached equipment;
- weight of insulation;
- weight of fluid;
- thermal expansion;
- supporting conditions;
- reaction of spring and constant load supports;
- displacements and rotations of anchor points, supports and connected equipment;
- cold pull;
- settlement of buildings.

### **4.2.5.2 Occasional operating conditions**

#### **4.2.5.2.1 General loadings**

Occasional operating conditions shall be normal operation incidents, for example the reaction effects due to operation of safety and relief devices, turbine load rejection, pump failure, opening and closing of shut-off valves.

For the calculation under occasional operating conditions, in addition to normal loadings given in 4.2.5.1, each of the following conditions shall be examined:

- safety valve operations;
- dynamic shock forces e.g. steam, water hammer;
- discharge reactions;
- temperatures deviating from normal operating conditions;
- restraining effects of dampers and snubbers;
- loading conditions of spring and constant effort supports;
- possible normal climatic effects e.g. normal snow load and wind load corresponding to the actual local conditions;

- seismic conditions (Design Basis Earthquake).

#### 4.2.5.2.2 Cleaning loading

For the calculation under cleaning (acid-cleaning and rinsing) loading conditions, all static, dynamic and kinematic boundary conditions shall be examined.

Each of the following conditions shall be examined:

- internal pressure;
- dead weight of piping, including internal structures and attached equipment;
- weight of insulation (partially or completely applied);
- weight of cleaning fluid;
- thermal expansion at cleaning temperature;
- supporting conditions (including temporary supports);
- locked and unlocked spring and constant effort supports;
- displacements and rotations of anchor points, supports and connected equipment;
- cold pull.

#### 4.2.5.2.3 Steam purge/blow loading

For the calculation under steam purge/blow loading conditions, all static and dynamic boundary conditions shall be taken into account.

Each of the following conditions shall be examined:

- changed system geometry;
- dead weight of piping system, including internal structure and attachments;
- weight of insulation (partially or completely applied);
- blow-through pressures;
- thermal expansion at blow-through temperature;
- supporting conditions (including temporary supports);
- locked and unlocked spring and constant effort supports;
- displacements and rotations of anchor points, supports and connected equipment;
- cold pull;
- forces at exit (discharge reaction force).

#### **4.2.5.3 Exceptional operating conditions**

Exceptional operating conditions shall be rarely occurring events.

For the calculation under exceptional operating conditions, in addition to normal loadings given in 4.2.5.1, each of the following conditions shall be examined:

- possible exceptional climatic effects, e.g. exceptional snow and wind corresponding to the local conditions;
- seismic conditions (Safe Shut-down Earthquake).

#### **4.2.5.4 Test conditions**

For the calculation under test conditions, all static, dynamic and kinematic boundary conditions shall be taken into account.

Each of the following conditions shall be examined:

- internal pressure (test gauge pressure and static head);
- dead weight of piping, including internal structures and attached equipment;
- weight of insulation (partially or completely applied);
- weight of test fluid;
- thermal expansion;
- supporting conditions (including temporary supports);
- blocked and unblocked spring and constant effort supports;
- displacements and rotations of anchor points, supports and connected equipment;
- cold pull.

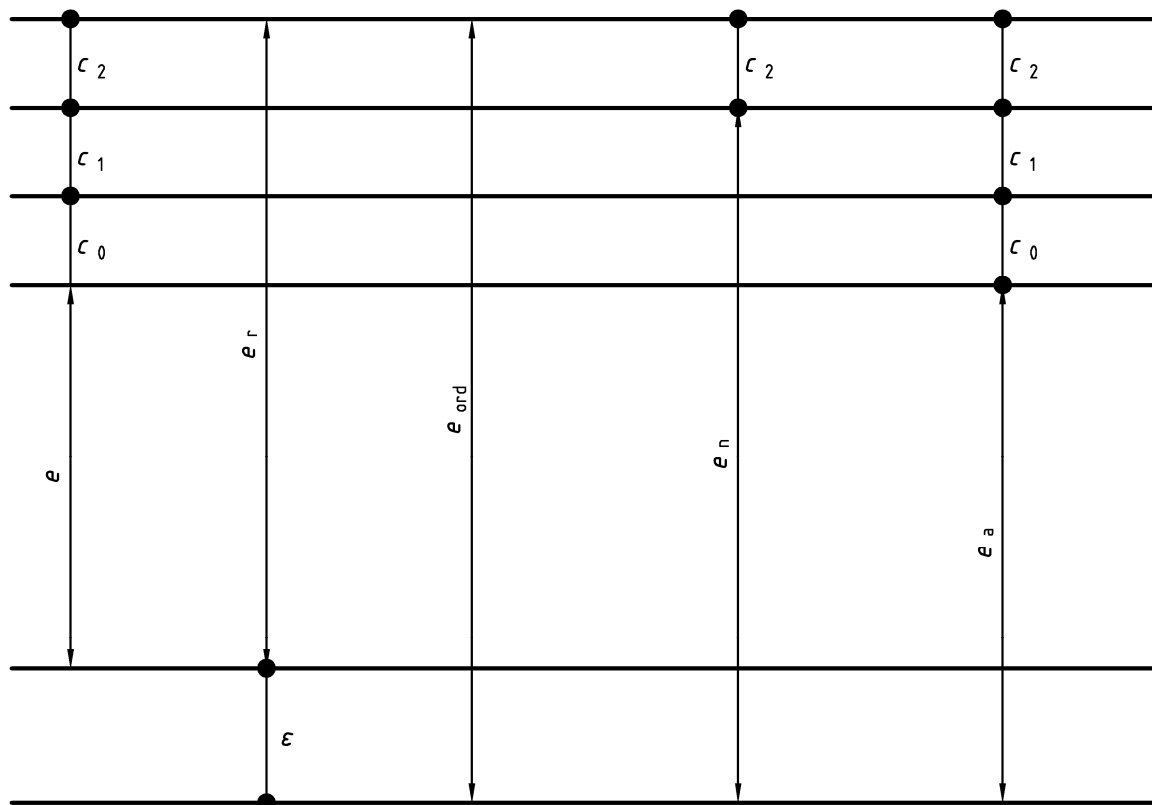
### **4.3 Thickness**

The minimum thickness shall be determined with regard to the manufacturing process for pipes and fittings.

Corrosion can be internal or external or both at the same time (the term corrosion includes erosion).

The value of the corrosion allowance  $c_0$  (which may be zero if no corrosion is to be expected) shall be determined by the manufacturer in accordance with the nature, temperature, velocity etc. of the products in contact with the wall.

All thicknesses, the corrosion allowance  $c_0$ , the tolerance  $c_1$  and the thinning  $c_2$  are shown in Figure 4.3-1.



Where

- $e$  is the minimum required thickness without allowances and tolerances to withstand pressure, calculated by the appropriate equations given in this standard;
- $c_0$  is the corrosion or erosion allowance;
- $c_1$  is the absolute value of the negative tolerance taken from the material standards or as provided by the pipe manufacturer;
- $c_2$  is the thinning allowance for possible thinning during manufacturing process (e.g. due to bending, straving, threading, grooving, etc);
- $e_r$  is the minimum required thickness with allowances and tolerances;
- $\varepsilon$  is the additional thickness resulting from the selection of the ordered thickness  $e_{ord}$ ;
- $e_{ord}$  is the ordered thickness (where  $c_2$  is often equal to 0; e.g. straight pipe);
- $e_n$  is the nominal thickness (on drawings);
- $e_a$  is the analysis thickness of a component, used for the check of the strength.

**Figure 4.3-1 — Thickness (applicable to straight pipes as well as bends)**

The analysis thickness  $e_a$  shall be the lowest thickness after corrosion and shall be given by:

$$e_a = e + \varepsilon \quad (4.3-1)$$

or

$$e_a = e_{ord} - c_0 - c_1 - c_2 \quad (4.3-2)$$

The ordered thickness  $e_{ord}$  of a component (pipe or fitting) shall be at least equal to:

- if the value of tolerance  $c_1$  is expressed in unit of length

$$e_{ord} \geq e + c_0 + c_1 + c_2 \quad (4.3-3)$$

- if the value of tolerance  $c_1$  is expressed as a percentage  $x$  of the ordered thickness  $e_{ord}$  :

$$e_{ord} \geq (e + c_0 + c_2) 100 / (100 - x) \quad (4.3-4)$$

#### **4.4 Tolerances**

The nominal dimensions shall be used in the calculations and the tolerances regarding thickness shall be fulfilled.

#### **4.5 Joint coefficient**

The joint coefficient  $z$  shall be used in the calculation of the thicknesses of components which include one or several butt welds, other than circumferential, and shall not exceed the following values:

- for equipment subject to destructive and non-destructive testing which confirms that the whole series of joints show no significant imperfections: 1;
- for equipment subject to random non-destructive testing: 0,85;
- for equipment not subject to non-destructive testing other than visual inspection: 0,7.

For the calculation of the strength of butt welded assemblies under exceptional operating conditions or under test conditions, it shall not be necessary to take a joint coefficient into account.

NOTE See EN 13480-5, Table 8.3-1.

#### **4.6 Dimensioning of piping components subject to pressure**

Clauses 6, 7, 8, 9, 10 and 11 describe the "design by rules" of piping components under static and cyclic loadings. This may be completed or replaced by a "design by analysis" as described in EN 13445-3, Annexes B and C. The requirements of clauses 6, 7, 8, 9 and 11 shall apply for loads of predominantly non-cyclic nature. It is assumed that a number of 1 000 full load pressure cycles do not lead to fatigue failure of the piping component under consideration. Where high strength materials with  $f$  greater than 250 N/mm<sup>2</sup> are used, a detailed load cycle analysis shall be carried out in accordance with clause 10.

Clauses 6, 7, 8, 9, 10 and 11 describe the "design by rules" of piping components under static and cyclic loadings. The « design by rule » can be completed or replaced by a « design by analysis » as described in EN 13445-3, Annex B and Annex C, where applicable.

In the case of cyclic loading (see clause 10), the geometry of the component under consideration shall be designed to avoid high stress concentrations. Where a higher number of pressure cycles shall be considered, the evaluation procedure outlined in 10.3 shall apply. For significant through-wall temperature gradients with cyclic nature in combination with cyclic pressure, the evaluation procedure outlined in 10.4 shall apply.

If the component under consideration is subjected to significant section moments resulting from connected piping, this shall be in accordance with 12.4.

The stress limits of components in accordance with European Standards with P/T ratings, e.g. flanges and components with wall thickness related to standard pipes, e.g. fittings, need not be recalculated.

## 5 Design stresses

### 5.1 General

The design stress shall be the lower of the time-independent stress value as calculated in 5.2 and the time-dependent stress value as calculated in 5.3 and shall be determined for each design and test condition.

The values of the design stress shall be determined from the material properties as calculated in accordance with the material standards and specifications given in EN 13480-2. These minimum values, specified for the delivery conditions, shall be used for design purpose, unless fabrication and/or heat treatment is known to lead to lower values. In such cases, the values to be used shall be determined by the manufacturer on the basis of the data given in the specification.

For steels used at low temperature (i.e. less than  $-10\text{ }^{\circ}\text{C}$ ), the design stress shall be determined at room temperature and they shall have an impact energy at design reference temperature in accordance with EN 13480-2.

NOTE 1 For intermediate temperatures, linear interpolation may be used.

NOTE 2 For use of the specified value at room temperature for temperatures less than or equal to  $50\text{ }^{\circ}\text{C}$ , see EN 13480-2, 4.2.2.1.

NOTE 3 For temporary piping, see 4.1.

For bolts, see additional requirements in 6.6.

### 5.2 Time-independent nominal design stress

#### 5.2.1 Steels other than austenitic steels

##### 5.2.1.1 Design conditions

The design stress shall be in accordance with the following:

$$f = \min \left\{ \frac{R_{eH\ t}}{1,5} \text{ or } \frac{R_{p0,2\ t}}{1,5}, \frac{R_m}{2,4} \right\} \quad (5.2.1-1)$$

##### 5.2.1.2 Test conditions

The designer shall ensure that the nominal design stress  $f_{\text{test}}$  under the proof test conditions, given in EN 13480-5, does not exceed 95 %  $R_{eH}$  at specified test temperature.

#### 5.2.2 Austenitic steels

##### 5.2.2.1 Design conditions

The design stress shall be in accordance with the following:

— for  $A \geq 35\text{ }^{\circ}\text{C}$

$$f = \frac{R_{p1,0\ t}}{1,5} \quad (5.2.2-1)$$

or  $f = \min\left(\frac{R_{m t}}{3}; \frac{R_{p1.0 t}}{1,2}\right)$  if  $R_{m t}$  is available

— for 35 % >  $A \geq 30$  %

$$f = \min\left(\frac{R_{p1.0 t}}{1,5}; \frac{R_m}{2,4}\right) \quad (5.2.2-2)$$

— for  $A < 30$  %, see 5.2.1.1.

#### **5.2.2.2 Test conditions**

For  $A \geq 25$  %, the designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed the greater of the two following values:

- 95 %  $R_{p1.0}$  at specified test temperature;
- 45 %  $R_m$  at specified test temperature.

For  $A < 25$  %, see 5.2.1.2.

#### **5.2.3 Nickel and / or chromium alloy steels**

Time-independent nominal design stress for Nickel and/or chromium alloy steels depend on the specified minimum elongation after rupture at room temperature.

#### **5.2.4 Steels castings**

##### **5.2.4.1 Design conditions**

The design stress shall be in accordance with the following:

$$f = \min\left(\frac{R_{eH t}}{1,9} \text{ or } \frac{R_{p0,2 t}}{1,9}; \frac{R_m}{3,0}\right) \quad (5.2.4-1)$$

##### **5.2.4.2 5.2.4.2 Test conditions**

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed  $R_{eH}$  or  $R_{p0,2}$  at the specified test temperature divided by a safety factor of 1,4.

#### **5.2.5 Additional requirements for steels with no specific control**

##### **5.2.5.1 General**

Steels with no specific control are those not possessing at least a test report 2.2 in accordance with EN 10204, and shall only be used if permitted in the technical specification.

##### **5.2.5.2 Design conditions**

The design stress given in 5.2.1.1 shall be divided by an additional safety factor which shall not be less than 1,2.

Where the yield strength values at elevated temperatures for unalloyed steels are not specified in the material standards, the following equation may be used:

$$R_{p\,0,2} = R_m \frac{720 - t}{1400} \quad (5.2.5-1)$$

where  $t$  is between 20 °C and 150 °C.

### 5.2.5.3 Test conditions

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed 95 %  $R_{eH}$  at the specified test temperature.

## 5.3 Time-dependent nominal design stress

### 5.3.1 General

For welds other than circumferential welds in welded pipes and fittings, the creep strength values of the base material shall be reduced by 20 %, except where ensured creep strength values have been determined for these pipes and fittings. This reduction is valid only for dimensioning.

NOTE If the creep characteristics determine the thickness, additional creep tests (based on extrapolations using the Larson-Miller formula) for instance on the welding consumables and the complete weld, should be performed.

### 5.3.2 Steels

#### 5.3.2.1 Design conditions

The design stress in the creep range  $f_{cr}$  to be used for design under static loading shall be:

$$f_{cr} = \frac{S_{RTt}}{Sf_{cr}} \quad (5.3.2-1)$$

where

$Sf_{cr}$  is a safety factor which depends on the time and shall be in accordance with Table 5.3.2-1.

**Table 5.3.2-1 — Safety factor as a function of mean creep rupture strength related to time**

| Time<br>$T$<br>h | Safety factor<br>$Sf_{cr}$ |
|------------------|----------------------------|
| 200 000          | 1,25                       |
| 150 000          | 1,35                       |
| 100 000          | 1,5                        |

If the design lifetime is not specified, the mean creep rupture strength at 200 000 h shall be used.

In cases where the 200 000 h values are not specified in the material Standards, the mean creep rupture strength at 150 000 h or 100 000 h shall be used.

If a design lifetime between 100 000 h and 200 000 h is specified, and a lifetime monitoring system is provided, divergent from Table 5.3.2-1, a safety factor  $S_{f_{cr}} = 1,25$  may be used.

In cases where design lifetimes shorter than 100 000 h are specified, one of the following methods shall be used:

- a) If a lifetime monitoring System is not provided, the safety factor  $S_{f_{cr}}$  shall be equal to 1,5 and shall be applied to the mean creep rupture strength at the relevant lifetime of at least 10 000 h;
- b) If a lifetime monitoring system is provided, a safety factor of  $S_{f_{cr}} = 1,25$  may be specified with regard to the mean creep rupture strength at the relevant lifetime of at least 10 000 h. In no case shall the 1 % creep strain limit (mean value) at 100 000 h be exceeded.

The creep rupture strength associated to the specified lifetime shall be interpolated based on a logarithmic time axis as well as a logarithmic stress axis (double logarithmic interpolation scheme).

### 5.3.2.2 Test conditions

The designer shall ensure that the stress under the proof test conditions, given in EN 13480-5, shall not exceed 95 %  $R_{eH}$  or 95 %  $R_{p1,0}$  or 95 %  $R_{p0,2}$  as applicable, at the specified test temperature .

### 5.3.3 Nickel and/or chromium alloy steels

The requirements of 5.3.2 shall apply unless otherwise stated in the specification.

## 6 Design of piping components under internal pressure

### 6.1 Straight pipes

The minimum required wall thickness for a straight pipe without allowances and tolerances,  $e$ , shall be calculated as follows:

— where  $D_o/D_i \leq 1,7$ :

$$e = \frac{p_c D_o}{2fz + p_c} \quad (6.1-1)$$

or

$$e = \frac{p_c D_i}{2fz - p_c} \quad (6.1-2)$$

— where  $D_o/D_i > 1,7$ :

$$e = \frac{D_o}{2} \left( 1 - \sqrt{\frac{fz - p_c}{fz + p_c}} \right) \quad (6.1-3)$$

or

$$e = \frac{D_i}{2} \left( \sqrt{\frac{fz + p_c}{fz - p_c}} - 1 \right) \quad (6.1-4)$$

NOTE This is Lamé's equation.

## 6.2 Pipe bends and elbows

### 6.2.1 General

There are two methods for calculating the wall thickness of elbows (see 6.2.3.1 and Annex B) and three methods for the wall thickness of pipe bends (see 6.2.3.1, 6.2.3.2 and Annex B). The chosen method shall be used in its entirety.

The equations given in 6.2.3 shall be applicable only if the out-of-roundness of the pipes bends are within the tolerances given in EN 13480-4.

NOTE These calculation rules take into account [1] and [2] that upon applying internal pressure to a pipe bend, higher stresses occur on the intrados of the bend (and lower stresses on the extrados of the bend) than on a straight pipe with identical wall thickness.

### 6.2.2 Symbols

For the purposes of 6.2, the symbols given in Table 6.2.2-1 shall apply in addition to those given in Table 3.2-1.

**Table 6.2.2-1 — Additional symbols for the purposes of 6.2**

| Symbol           | Description                                                                             | Unit |
|------------------|-----------------------------------------------------------------------------------------|------|
| $e_{\text{int}}$ | minimum required thickness without allowances and tolerances for a bend on the intrados | mm   |
| $e_{\text{ext}}$ | minimum required thickness without allowances and tolerances for a bend on the extrados | mm   |
| $R$              | radius of bend or elbow                                                                 | mm   |
| $r$              | mean radius of the pipe                                                                 | mm   |

### 6.2.3 Required wall thicknesses

#### 6.2.3.1 Normal route

The minimum required wall thickness without allowances and tolerances shall be calculated by:

— on the intrados

$$e_{\text{int}} = e \frac{(R/D_0) - 0,25}{(R/D_0) - 0,5} \quad (6.2.3-1)$$

— on the extrados

$$e_{\text{ext}} = e \frac{(R/D_0) + 0,25}{(R/D_0) + 0,5} \quad (6.2.3-2)$$

where

$e$  is calculated in accordance with 6.1 for straight pipe.

#### 6.2.3.2 Alternative route

Pipes to be bent, by any manufacturing process, shall have a sufficient thickness such that it can be demonstrated that the following requirements are met after bending.

- a) The minimum thickness at any point in a pipe bend (including the extrados) shall be not less than that required for the equivalent straight pipe.
- b) Where the design stress is time-dependent, and the bend radius is less than six times the outside diameter of the pipe, the intrados thickness shall be not less than that calculated from the following equation:

$$e_{\text{int}} = e \frac{2R - r}{2R - 2r} \quad (6.2.3-3)$$

where

$$r = \frac{D_0 - e}{2} \quad (6.2.3-4)$$

- c) Where the design stress is time-independent, and the bend radius is less than three times the outside diameter, the intrados thickness shall be not less than that calculated from the following equation:

$$e_{\text{int}} = \max \left( e; \frac{e}{1,25} \frac{2R - r}{2R - 2r} \right) \quad (6.2.3-5)$$

where

$r$  is calculated in accordance with equation (6.2.3-4).

NOTE Manufacturing processes and experience should be taken into account for the determination of the thickness before bending. Table 6.2.3-1 gives guidance on the pipe wall thickness needed to meet the requirements of 6.2.3.2.

### 6.2.3.3 More accurate route

The more accurate method of calculating the wall thickness of pipe bends and elbows shall be in accordance with Annex B.

**Table 6.2.3-1 — Minimum pipe wall thickness before bending by induction**

| Radius    | Normal route 6.2.3.1 | Alternative route 6.2.3.2 |
|-----------|----------------------|---------------------------|
| 10 $D_0$  | 1,02 $e$             | 1,04 $e$                  |
| 8 $D_0$   | 1,03 $e$             | 1,05 $e$                  |
| 6 $D_0$   | 1,04 $e$             | 1,06 $e$                  |
| 5 $D_0$   | 1,04 $e$             | 1,08 $e$                  |
| 4 $D_0$   | 1,05 $e$             | 1,10 $e$                  |
| 3 $D_0$   | 1,06 $e$             | 1,13 $e$                  |
| 2,5 $D_0$ | 1,08 $e$             | 1,16 $e$                  |
| 2 $D_0$   | 1,10 $e$             | 1,20 $e$                  |
| 1,5 $D_0$ | 1,15 $e$             | 1,25 $e$                  |

## 6.3 Mitre bends

### 6.3.1 General

The following rules for mitre bends (see Figure 6.3.2-1) shall only be used if the following conditions are met:

a) Time independent design stress:

- the calculation pressure  $p_c$  is less or equal to 20 bar (2,0 MPa );

b) Time dependent design stress:

- the piping system is fully compensated by expansion joints;
- the internal pressure is limited to 4 bar (0,4 MPa );
- the number of full pressure cycles is limited to 100;
- consideration is given to high temperatures cycling.

A mitre bend with an angle of change in direction at a single joint greater than  $22,5^\circ$  (see angle  $\alpha$  in Figure 6.3.2-1) shall not be used under cyclic loadings ( $> 7\,000$  cycles).

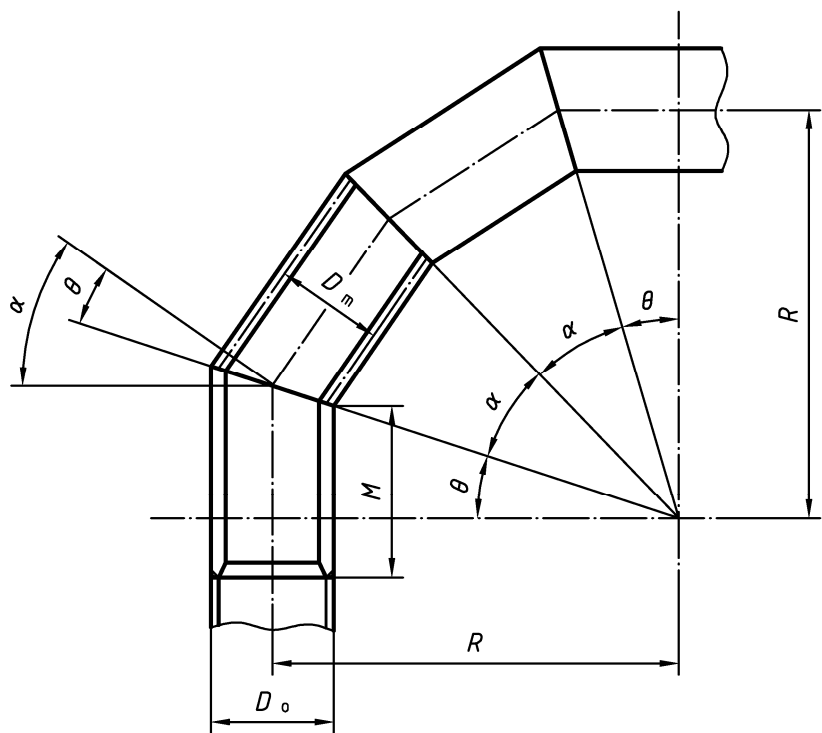
NOTE For an angle of change in direction of  $3^\circ$  or less at a single joint, the calculation method given in 6.1 may be used.

### 6.3.2 Symbols

For the purposes of 6.3, the symbols given in Table 6.3.2-1 shall apply in addition to those given in Table 3.2-1.

**Table 6.3.2-1 — Additional symbols for the purposes of 6.3**

| Symbol   | Description                                                               | Unit |
|----------|---------------------------------------------------------------------------|------|
| $R$      | Effective radius of mitre bend (see Figure 6.3.2-1)                       | mm   |
| $l_a$    | Empirical value as a function of wall thickness $e_a$ (see Table 6.3.3-1) | mm   |
| $\theta$ | Angle (see Figure 6.3.2-1)                                                | °    |
| $\alpha$ | Angle of change in direction at mitre joint                               | °    |



NOTE  $\alpha = 2 \theta$

Table 6.3.2-1 — Scheme for a mitre bend

6.3.3 Effective radius of mitre bend

The value of  $R$  shall not be less than the following

$$R = \frac{l_a}{\tan \theta} + \frac{D_o}{2} \tag{6.3.3-1}$$

where

$l_a$  is given in Table 6.3.3-1.

Table 6.3.3-1 — Empirical values of  $l_a$  for given values of  $e_a$

| $e_a$<br>mm     | $l_a$<br>mm    |
|-----------------|----------------|
| $e_a \leq 13$   | 25             |
| $13 < e_a < 22$ | $2 e_a$        |
| $e_a \geq 22$   | $2/3 e_a + 30$ |

### 6.3.4 Multiple mitre bends

The maximum allowable internal pressure,  $p_a$ , for multiple mitre bends (see Figure 6.3.2-1) shall be the lesser of the values calculated from equations (6.3.4-1) and (6.3.4-2) which shall only apply for an angle  $\theta \leq 22,5^\circ$ .

$$p_a = \frac{2fze_a}{D_m} \left( \frac{e_a}{e_a + 0,643 \tan \theta \sqrt{0,5D_m e_a}} \right) \quad (6.3.4-1)$$

$$p_a = \frac{2fze_a}{D_m} \left( \frac{R - 0,5D_m}{R - 0,25D_m} \right) \quad (6.3.4-2)$$

### 6.3.5 Single mitre bends

A single mitre bend is a mitre bend with one angular offset only.

The maximum allowable internal pressure,  $p_a$ , for a single mitre bend with an angle  $\theta$  not greater than  $22,5^\circ$  shall be calculated in accordance with 6.3.4.

The maximum allowable internal pressure,  $p_a$ , for a single mitre bend with an angle  $\theta$  greater than  $22,5^\circ$  shall be calculated from equation (6.3.5-1).

$$p_a = \frac{2fze_a}{D_m} \left( \frac{e_a}{e_a + 1,25 \tan \theta \sqrt{0,5D_m e_a}} \right) \quad (6.3.5-1)$$

### 6.3.6 Adjacent straight pipe sections of mitre bends

The wall thickness shall extend a distance not less than  $M$  (see Figure 6.3.2-1) from the inside crotch of the end mitre welds where

$$M = \max \left\{ 2,5 \sqrt{0,5D_m e_a} ; \left( R - \frac{D_m}{2} \right) \tan \theta \right\} \quad (6.3.6-1)$$

## 6.4 Reducers

### 6.4.1 Conditions of applicability

Requirements are given in 6.4.4 to 6.4.8 for right circular cones and cone/cylinder intersections where the cone and the cylinder are on the same axis of rotation. Requirements for offset cones are given in 6.4.9.

The requirements do not apply to:

- cones for which the half angle at the apex of the cone is greater than  $75^\circ$ ;
- cones for which;

$$\frac{e_a \cos \alpha}{D_c} \leq 0,001; \quad (6.4.1-1)$$

- short cones joining a jacket to a shell.

Limits on the minimum distance from other major discontinuities are given in individual clauses.

## 6.4.2 Specific definitions

### 6.4.2.1

#### junction between the cylinder and the cone

intersection of the mid-thickness lines of cylinder and cone, extended if necessary where there is a knuckle (see Figure 6.4.2-1 and Figure 6.4.2-2 for examples at the large end)

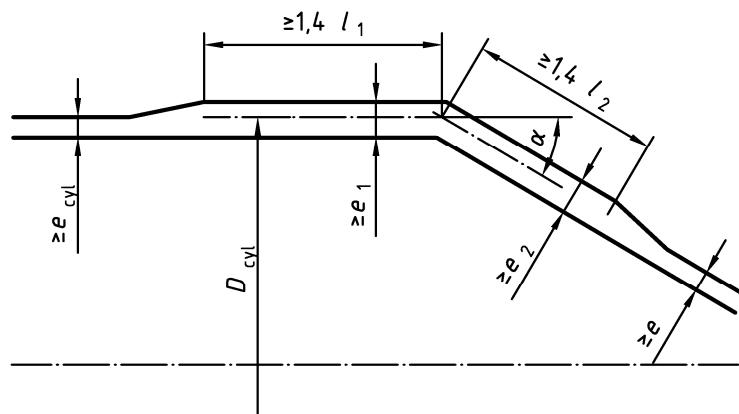


Figure 6.4.2-1 — Geometry of cone/cylinder intersection without knuckle – Large end

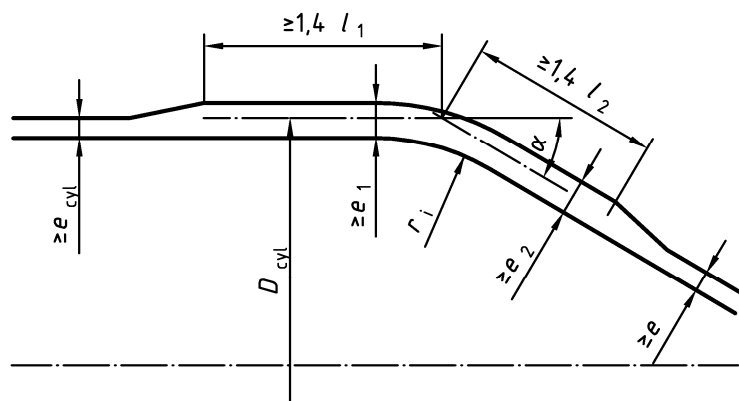


Figure 6.4.2-2 — Geometry of cone/cylinder intersection with knuckle – Large end

## 6.4.3 Specific symbols and abbreviations

For the purposes of 6.4, the symbols given in Table 6.4.3-1 shall apply in addition to those given in Table 3.2-1.

**Table 6.4.3-1 — Additional symbols for the purposes of 6.4**

|           |                                                                                                                                            |
|-----------|--------------------------------------------------------------------------------------------------------------------------------------------|
| $D_c$     | the mean diameter of the cylinder at the junction with the cone;                                                                           |
| $D_e$     | the outside diameter of the cone;                                                                                                          |
| $D_i$     | the inside diameter of the cone;                                                                                                           |
| $D_K$     | a diameter given by equation (6.4.4-7);                                                                                                    |
| $D_m$     | the mean diameter of the cone;                                                                                                             |
| $e_{con}$ | required thickness of cone as determined in 6.4.4;                                                                                         |
| $e_{cyl}$ | required thickness of cylinder as determined in 6.1;                                                                                       |
| $e_j$     | a required or analysis thickness at a junction at the large end of a cone;                                                                 |
| $e_1$     | required thickness of cylinder at junction;                                                                                                |
| $e_{1a}$  | analysis reinforcing thickness in cylinder;                                                                                                |
| $e_2$     | required thickness of cone and knuckle at junction;                                                                                        |
| $e_{2a}$  | analysis reinforcing thickness in cone;                                                                                                    |
| $f$       | the nominal design stress. In the design of junctions to 6.4.6 to 6.4.9 it is the lowest of the values for the individual component parts; |
| $l_1$     | length along cylinder;                                                                                                                     |
| $l_2$     | length along cone at large or small end;                                                                                                   |
| $r_i$     | inside radius of knuckle                                                                                                                   |
| $\alpha$  | the semi angle of cone at apex (degrees);                                                                                                  |
| $\beta$   | a factor defined in 6.4.6;                                                                                                                 |
| $\beta_H$ | a factor defined in 6.4.8;                                                                                                                 |
| $\gamma$  | a factor defined in 6.4.7;                                                                                                                 |
| $\rho$    | a factor defined in 6.4.7;                                                                                                                 |
| $\tau$    | a factor defined in 6.4.8.                                                                                                                 |

#### 6.4.4 Conical shells

The required thickness at any point along the length of a cone shall be calculated from one of the following two equations:

$$e_{con} = \frac{p_c D_i}{2f z - P} \frac{1}{\cos \alpha} \quad (6.4.4-1)$$

or

$$e_{con} = \frac{p_c D_e}{2f z + P} \frac{1}{\cos \alpha} \quad (6.4.4-2)$$

where  $D_i$  and  $D_e$  are at the point under consideration.

For a given geometry:

$$PS = \frac{2f z e_{con} \cos \alpha}{D_m} \quad (6.4.4-3)$$

where  $D_m$  is at the point under consideration.

At the large end of a cone attached to a cylinder it is permissible to make the following substitutions:

$$D_i = D_k \quad (6.4.4-4)$$

$$D_e = D_k + 2e_2 \cos \alpha \quad (6.4.4-5)$$

$$D_m = (D_i + D_e)/2 \quad (6.4.4-6)$$

where

$$D_K = D_c - e_1 - 2r(1 - \cos \alpha) - l_2 \sin \alpha \quad (6.4.4-7)$$

NOTE 1 The thickness given by this section is a minimum. Thickness may have to be increased at junctions with other components, or to provide reinforcement at nozzles or openings, or to carry non-pressure loads.

NOTE 2 Since the thickness calculated above is the minimum allowable at that point along the cone, it is permissible to build a cone from plates of different thickness provided that at every point the minimum is achieved.

### 6.4.5 Junctions - general

The requirements of 6.4.6, 6.4.7 and 6.4.8 apply when the junction is more than  $2l_1$  along the cylinder and  $2l_2$  along the cone from any other junction or major discontinuity, such as another cone/cylinder junction or a flange, where:

$$l_1 = \sqrt{D_c e_1} \quad (6.4.5-1)$$

$$l_2 = \sqrt{\frac{D_c e_2}{\cos \alpha}} \quad (6.4.5-2)$$

The length of the cone can be reduced to less than  $2l_2$  if both of the following conditions are fulfilled:

- the wall thickness  $e_2$ , calculated in accordance with 6.4.6 or 6.4.7, is existent along the whole length of the cone;
- the junction at the small end of the cone is sufficiently dimensioned according to 6.4.8.

### 6.4.6 Junction between the large end of a cone and a cylinder without a knuckle

#### 6.4.6.1 Conditions of applicability

The requirements of 6.4.6.2 and 6.4.6.3 apply provided that all the following conditions are satisfied:

- 1) the joint is a butt weld where the inside and outside surfaces merge smoothly with the adjacent cone and cylinder without local reduction in thickness; and
- 2) the weld at the junction shall be subject to 100 % non-destructive examination, either by radiography or ultrasonic techniques, unless the design is such that the thickness at the weld exceeds  $1,4e_j$ , in which case the normal rules for the relevant design shall be applied.

#### 6.4.6.2 Design

The required thickness  $e_1$  of the cylinder adjacent to the junction is the greater of  $e_{cyl}$  and  $e_j$  where  $e_j$  shall be determined by the following procedure:

$$\beta = \frac{1}{3} \sqrt{\frac{D_c}{e_j}} \cdot \frac{\tan \alpha}{1 + 1/\sqrt{\cos \alpha}} - 0,15 \quad (6.4.6-1)$$

$$e_j = \frac{p_c D_c \beta}{2f} \quad (6.4.6-2)$$

The answer is acceptable if the value given by equation (6.4.6-2) is not less than assumed in equation (6.4.6-1).

$\beta$  can also be read from the graph in Figure 6.4.6-1.

This thickness shall be maintained for a distance of at least  $1,4l_1$  from the junction along the cylinder.

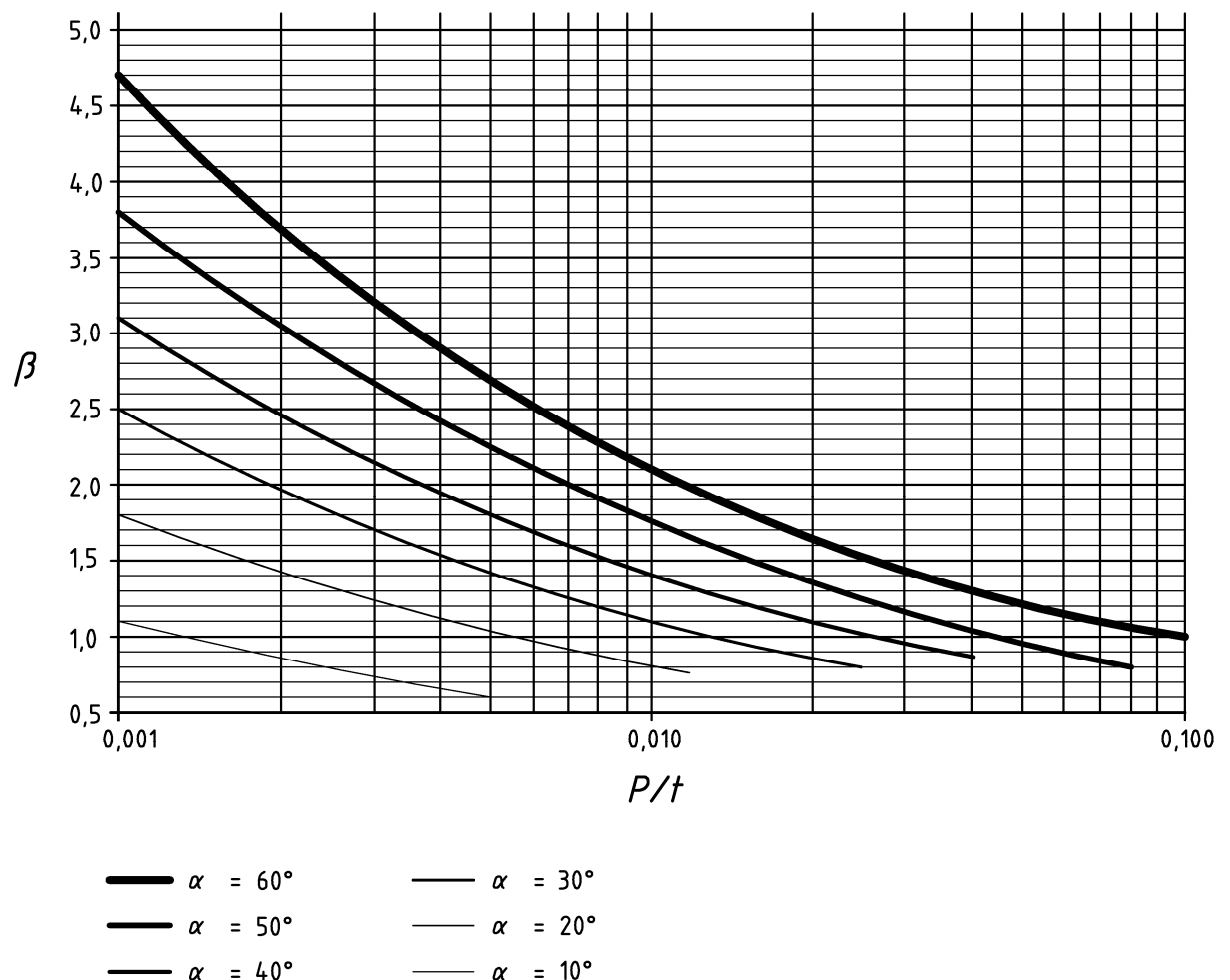


**Figure 6.4.6-1 — Values of coefficient  $\beta$  for cone/cylinder intersection without knuckle**

The required thickness  $e_2$  of the cone adjacent to the junction is the greater of  $e_{con}$  and  $e_j$ . This thickness shall be maintained for a distance of at least  $1,4l_2$  from the junction along the cone, see Figure 6.4.6-1.

It is permissible to redistribute the reinforcement in the following way, provided that the minimum thicknesses given by 6.1 and 6.4.4 continue to be met.

The thickness for the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance  $1,4l_1$  from the junction is not less than  $1,4e_{1a}$ . In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance  $1,4l_2$  from the junction is not less than  $1,4e_{2a}$ .

### 6.4.6.3 Rating

The maximum permissible pressure for a given geometry shall be determined as follows:

- a) apply equation (6.4.4-3) to cylinder;

$$P_S = \frac{2f z e_a}{D_c} \quad (6.4.6-3)$$

- b) apply equation (6.4.4-3) to the cone;  
c) determine the analysis reinforcing thickness  $e_{1a}$  of the cylinder at the junction;  
d) determine the analysis reinforcing thickness  $e_{2a}$  of the cone at the junction;  
e) apply equation (6.4.4-3) with thickness  $e_{2a}$  and diameter  $D_m$ ;  
f) find  $e_j$ , the lesser of  $e_{1a}$  and  $e_{2a}$ ;  
g) calculate  $\beta$  from equation (6.4.6-1), then,

$$P_{\max} = \frac{2f e_j}{\beta D_c} \quad (6.4.6-4)$$

- h) the maximum permissible pressure is the lowest of the pressures determined in a), b), e) and g).

NOTE The following procedure may be used to find the analysis reinforcing thickness at c) or d) above:

- 1) Assume  $e_{1a}$  (the initial choice should be the thickness at the junction).
- 2) Calculate

$$l_1 = 1,4\sqrt{D_C e_{1a}} \quad (6.4.6-5)$$

- 3) If the thickness is constant within the distance  $l_1$  then  $e_{1a}$  is confirmed.
- 4) If not, calculate the metal area  $A_1$  within the distance  $l_1$  from the junction.
- 5) Obtain a better estimate by.

$$e_{1a} = A_1 / l_1 \quad (6.4.6-6)$$

The answer is acceptable if it is not greater than  $e_{1a}$  assumed in 1).

- 6) If the answer is unacceptable, return to 1).
- 7) Use a similar procedure to find  $e_{2a}$  making.

$$l_2 = 1,4 \sqrt{\frac{D_c e_{2a}}{\cos \alpha}} \quad (6.4.6-7)$$

## 6.4.7 Junction between the large end of a cone and a cylinder with a knuckle

### 6.4.7.1 Conditions of applicability

This sub-clause applies provided that all the following conditions are satisfied:

- a) the knuckle is of toroidal form and merges smoothly with the adjacent cone and cylinder; and
- b) the inside radius of curvature of the knuckle,  $r_i < 0,3 D_c$ .

NOTE This clause does not prescribe a lower limit to the radius of curvature of the knuckle.

### 6.4.7.2 Design

The value of  $e_j$  shall be determined by the following procedure:

Assume a value of  $e_j$  and calculate:

$$\beta = \frac{1}{3} \sqrt{\frac{D_c}{e_j}} \cdot \frac{\tan \alpha}{1 + 1/\sqrt{\cos \alpha}} - 0,15 \quad (6.4.7-1)$$

$$\rho = \frac{0,028 r_i}{\sqrt{D_c e_j}} \cdot \frac{\alpha}{1 + 1/\sqrt{\cos \alpha}} \quad (6.4.7-2)$$

$$\gamma = 1 + \frac{\rho}{1,2 \left( 1 + \frac{0,2}{\rho} \right)} \quad (6.4.7-3)$$

$$e_j = \frac{p_c D_c \beta}{2f \gamma} \quad (6.4.7-4)$$

The answer is acceptable if the value given by equation (6.4.7-4) is not less than that assumed.

The required thickness  $e_1$  of the cylinder adjacent to the junction is the greater of  $e_{cyl}$  and  $e_j$ .

This thickness shall be maintained for a distance of at least  $1,4l_1$  from the junction and  $0,5l_1$  from the knuckle/cylinder tangent line along the cylinder.

The required thickness  $e_2$  of the knuckle and the cone adjacent to the junction is the greater of  $e_{con}$  and  $e_j$ . This thickness shall be maintained for a distance of at least  $1,4l_2$  from the junction and  $0,7l_2$  from the cone/knuckle tangent line along the cone.

### 6.4.7.3 Rating

The maximum permissible pressure for a given geometry shall be found as follows.

- Determine  $e_{1a}$ , the analysis thicknesses for the cylinder next to the knuckle, and  $e_{2a}$ , the analysis thickness for the knuckle and the adjacent part of the cone;
- Check that the limitations of 6.4.7.1 are met;
- Apply equation (6.4.6-3) to the cylinder with  $e_a = e_{1a}$ ;
- Apply equation (6.4.4-3) to the cone with  $e_{con} = e_{2a}$ ;
- Find  $e_j$ , the lesser of  $e_{1a}$  and  $e_{2a}$ ;
- Find  $\beta$  and  $\gamma$  from equations (6.4.7-1) and (6.4.7-3), then

$$PS = \frac{2f \gamma e_j}{\beta D_c} \quad (6.4.7-5)$$

- The maximum permissible pressure is the lowest of the pressures determined in c), d) and f).

### 6.4.8 Junction between the small end of a cone and a cylinder

#### 6.4.8.1 Conditions of applicability

The requirements of 6.4.8.2 and 6.4.8.3 apply provided that all the following conditions are satisfied:

- the required thickness of the cylinder  $e_1$  is maintained for a distance  $l_1$  and that of the cone  $e_2$  is maintained for a distance  $l_2$  from the junction (see Figure 6.4.8-1); and
- the thicknesses meet the requirements of 6.1 and 6.4.4;

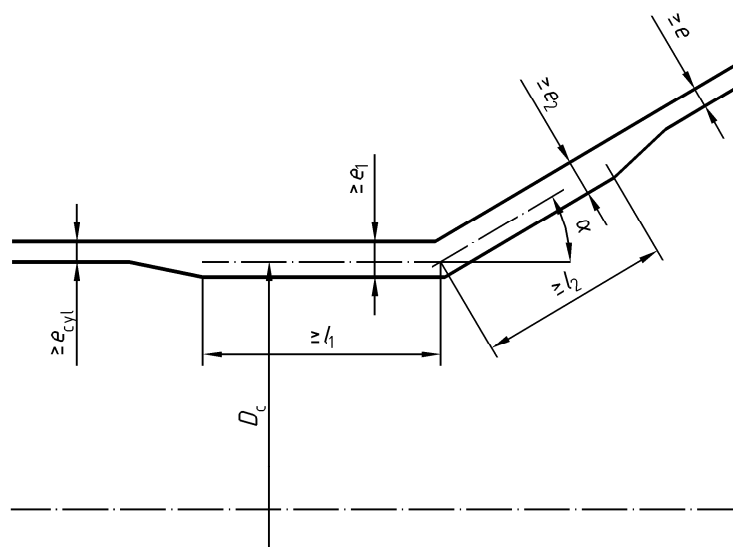


Figure 6.4.8.1-1 — Geometry of cone/cylinder intersection: small end

#### 6.4.8.2 Design

Required thicknesses  $e_1$  and  $e_2$  shall be found by the following procedure:

Assume values of  $e_1$  and  $e_2$ :

$$s = \frac{e_2}{e_1} \quad (6.4.8-1)$$

when  $s < 1$

$$\tau = s \sqrt{\frac{s}{\cos \alpha}} + \sqrt{\frac{1+s^2}{2}} \quad (6.4.8-2)$$

when  $s \geq 1$

$$\tau = 1 + \sqrt{s \left\{ \frac{1+s^2}{2 \cos \alpha} \right\}} \quad (6.4.8-3)$$

$$\beta_H = 0,4 \sqrt{\frac{D_c}{e_1} \frac{\tan \alpha}{\tau}} + 0,5 \quad (6.4.8-4)$$

If

$$p_c \leq \frac{2f z e_1}{D_c \beta_H} \quad (6.4.8-5)$$

then  $e_1$  and  $e_2$  are acceptable. If not, repeat with increased values of  $e_1$  and/or  $e_2$ .

NOTE The above procedure does not provide values for  $e_1$  and  $e_2$  independently. Any values may be selected to suit the needs of the design, for example to obtain a favourable value of  $l_1$  or  $l_2$ .

Provided that the requirements of 6.1 and 6.4.4 continue to be met, it is permissible to modify a design according to the above rule in one of the following ways:

- Where  $e_1 = e_2$  a knuckle of the same thickness may be included.  $l_1$  and  $l_2$  continue to be measured from the junction (i.e. the point where the centre lines of cone and cylinder meet);
- The thickness of the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance  $l_1$  from the junction is not less than  $l_1 e_1$ . In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance  $l_2$  from the junction is not less than  $l_2 e_2$ .

#### 6.4.8.3 Rating

The maximum permissible pressure for a given geometry shall be:

$$PS = \frac{2f z e_1}{D_c \beta_H} \quad (6.4.8-6)$$

$\beta_H$  is found from equations (6.4.8-1) to (6.4.8-4) using  $e_{1a}$  and  $e_{2a}$  in place of  $e_1$  and  $e_2$ .

NOTE 1 The procedure for finding  $e_{1a}$  and  $e_{2a}$  is as provided in the note to 6.4.6.3.

NOTE 2 Analysis thicknesses may exceed the required thickness without leading to any increase in  $l_1$  or  $l_2$ .

#### 6.4.9 Offset reducers

This sub-clause shall apply to offset reducers (see Figure 6.4.9-1). The cylindrical parts shall have parallel centre lines offset from each other by a distance  $d_{\text{off}}$  not greater than the difference of their radius. The thicknesses shall be calculated in accordance with 6.4.6 or 6.4.7 for the junction at the large end. The minimum thicknesses shall be calculated in accordance with 6.4.8 for the junction at the small end. The greater of these shall apply to the whole reducer. The angle  $\alpha$  shall be taken as the greatest angle between the conical and cylindrical parts.

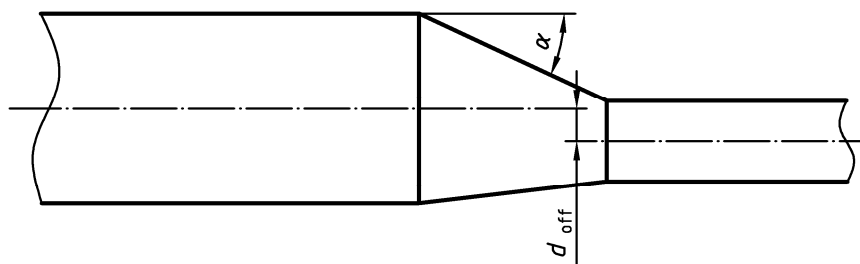


Figure 6.4.9-1 — Offset reducer

#### 6.4.10 Special forged reducers

Special forged reducers, for example for very high temperature and/or very high internal pressure, which are not covered by the product standards may be designed as shown in Figure 6.4.10-1 where:

$$l_l \geq \sqrt{\frac{D_l e_l}{\cos \alpha}} \quad (6.4.10-1)$$

$$l_s \geq \sqrt{\frac{D_s e_s}{\cos \alpha}} \quad (6.4.10-2)$$

$$r \geq 10 \text{ mm} \quad (6.4.10-3)$$

$$r' \geq 100 \text{ mm} \quad (6.4.10-4)$$

$$e_{r=\max} \{e_{\text{cyl}}; e_j\} \quad (6.4.10-5)$$

With  $e_{\text{cyl}}$  according to 6.1 and  $e_j$  according to equation (6.4.7-4).

In such cases additional design calculations are not required.

Where reducers of other design are supplied, their suitability shall be demonstrated.

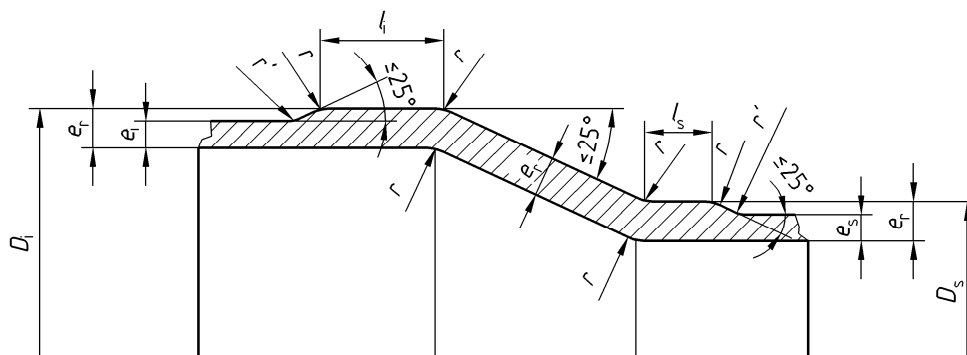


Figure 6.4.10-1 — Special forged reducer

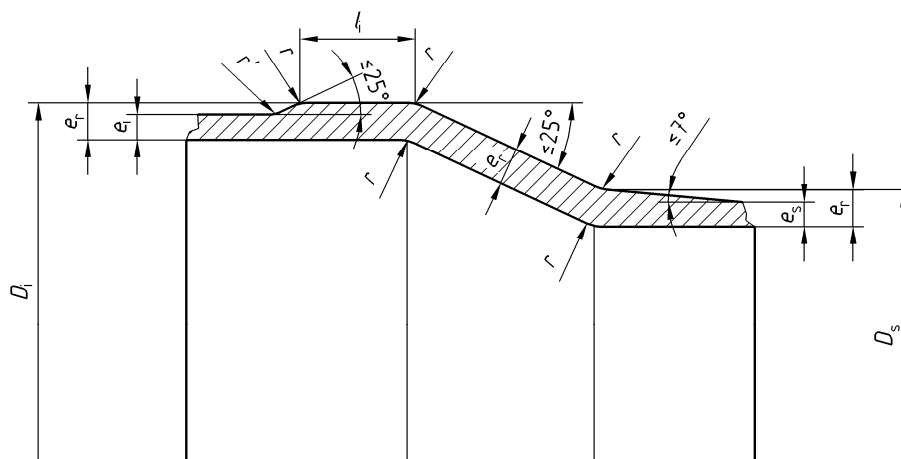


Figure 6.4.10-2 — Special forged reducer (alternative solution)

## 6.5 Flexible piping components

### 6.5.1 General

This sub-clause deals with bellows expansion joints and braided corrugated hose assemblies which are used to absorb relative movements and misalignments of piping or plant components, as well as to reduce forces and moments, i.e. stresses, in the pipes and their connections.

The manufacturer of such components has the responsibility for their correct design and manufacture.

### 6.5.2 Expansion joints

A piping system containing a bellows expansion joint is critically dependent upon a proper combination of each and every component. This requires that detailed consideration shall be given to the system, its supports and anchors and their interaction with the bellows expansion joints. The bellows expansion joints should not be considered as a commodity item.

For design and incorporation of expansion joints, see Annex C.

In order to design the expansion joints the following data shall be supplied:

- type of expansion joint (axial, angular, lateral, universal);
- end fitting (weld end, flange);
- nominal diameter DN or dimensions of end fittings (e.g. diameter, wall thickness);
- operating pressure or calculation pressure;
- operating temperature or calculation temperature;
- movements of the expansion joint:
  - total axial movement;
  - total angular rotation;
  - total lateral movement;
- presetting (magnitude, direction);
- number of load cycles (movement, pressure);
- fluid (type, density, additives);
- fluid velocity;
- material requirements;
- additional loads (see 4.2.4).

The individual movements may also be specified as a combined movement from which the relevant design shall be determined by the manufacturer.

**NOTE** The corrugations of metallic expansion joints generally have a wall thickness substantially less than that of the equipment in conjunction with which they are used. It is therefore essential to select material for their manufacture having adequate resistance to all the corrosive agents likely to be encountered in any particular application.

The manufacturer of the expansion joint shall provide, when requested, the following information for system analysis:

- reaction force rates for all directions of movement;
- reaction moment rates for all rotation axes;
- forces and moments resulting from the friction in the bearings of the hinges;
- pressure thrust, the pressure-induced axial force of unrestrained expansion joints, acting on anchors.

### **6.5.3 Corrugated metal hose assemblies**

Braided pressurized corrugated hose assemblies are self restrained axially and do not exert pressure loads on adjacent pipework.

Straight hose assemblies shall not be used for axial movement. Movement should be accommodated laterally or by use of bends or loops (see Figures 6.5.3-1, 6.5.3-2, 6.5.3-3)

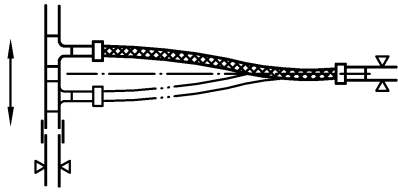


Figure 6.5.3-1

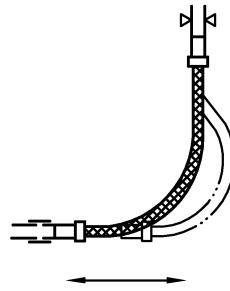


Figure 6.5.3-2

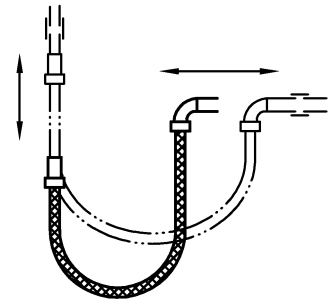


Figure 6.5.3-3

To design a corrugated hose assembly the following design data shall be supplied to and taken into consideration by the manufacturer:

- description of intended application;
- nominal size of the hose assembly;
- maximum operating pressure (internal and/or external);
- vacuum conditions where applicable;
- ambient and maximum operating temperature;
- materials specification;
- fluid to be conveyed;
- Information about possible corrosion, erosion, abrasion;
- fluid velocity;
- movements and/or vibrations (including torsion if any);
- specified life (e.g. number of cycles);
- type of fitting for hose assembly;
- external or internal protection if required;
- special requirements for heat treatment and/or cleaning;
- any other parameter or loadings which can influence design and life expectancy (e.g. water hammer).

Where corrugated metal hose assemblies can be subjected to different working conditions (e.g. for normal and cleaning operation) all these conditions shall be made known to and taken into consideration by the manufacturer.

## 6.6 Bolted flange connections

### 6.6.1 General

The rules of this sub-clause are to check the mechanical resistance of the flange connection subjected to static loads. It is also in the responsibility of the designer to ensure the adequacy of the flange connection (gasket type and characteristics, etc) with the operating conditions, in particular with regards to any specific required tightness.

If there is a specific requirement on tightness for the flange connection, this shall be calculated in accordance with EN 1591-1, using Annex P.

The designer shall consider section loadings caused by the connected piping system.

The classification of material for flanges, bolts and nuts is given by EN 1515-2 (PN flanges) and EN 1515-3 (Class flanges). The selection of bolting shall comply with Annex D or Annex P and EN 1515-4.

### 6.6.2 Symbols

For the purposes of 6.6, the symbols given in Table 6.6.2-1 shall apply in addition to those given in Table 3.2-1.

**Table 6.6.2-1 — Additional symbols for the purposes of 6.6**

| Symbol   | Description                                              | Unit                     |
|----------|----------------------------------------------------------|--------------------------|
| $P_{eq}$ | Equivalent design pressure                               | MPa (N/mm <sup>2</sup> ) |
| $P$      | Internal calculation pressure                            | MPa (N/mm <sup>2</sup> ) |
| $F$      | Pulling axial force (to be a positive value in equation) | N                        |
| $M$      | External bending moment                                  | N mm                     |
| $G$      | Diameter of gasket load reaction                         | mm                       |

### 6.6.3 Standard flange

A standard steel flange connection in accordance with defined material requirements, giving the maximum allowable pressure with regards to the flange materials and the design temperature, may be used within the construction of piping subjected to internal pressure, without the necessity of carrying out a calculation to verify its resistance when the following conditions are met:

- For each normal working condition, the design pressure shall not exceed the maximum allowable pressure specified.
- For conditions where the flange connection is simultaneously subjected to internal pressure, axial load and bending moment, the equivalent design pressure,  $P_{eq}$ , according to equation (6.6.2-1) shall not exceed the limits specified in a).

$$P_{eq} = P + \frac{4F}{\pi G^2} + \frac{16|M|}{\pi G^3} \quad (6.6.2-1)$$

Where:

- $G$  is the diameter of circle on which applies the compression load of the gasket (normally the mean diameter of the gasket).

- c) The gasket types, for each PN, are specified in EN 1514-1 to EN 1514-8.
- d) The strength of the bolting for the flange connection, for each PN, shall be as indicated in EN 1515-1 to 4.
- e) The difference of temperature between the flanges and the bolting shall not exceed, 50 °C in any case.
- f) If the design temperature is  $\geq 120$  °C, the thermal expansion coefficient of the flange material shall not exceed the thermal expansion coefficient of the bolt material by more than 10 %.

#### 6.6.4 Non-standard flange

If a non-standard flange is used, the design shall be done by applying the calculation method in EN 1591-1, using for example Annex P, or by applying the algorithm shown in the Taylor-Forge method, using for example Annex D.

NOTE 1 The Taylor-Forge method does not ensure tightness.

NOTE 2 The algorithm given in EN 1591-1 includes a consideration of section loadings.

NOTE 3 The bolt torque should be specified by the designer. Attention should be paid in such cases to the method of tightening. Guidance of scatter band when applying the different methods of tightening are given in EN 1591-1.

## 7 Design of ends under internal pressure

### 7.1 Dished ends

#### 7.1.1 Symbols

For the purposes of 7.1, the symbols given in Table 7.1.1-1 shall apply in addition to those given in Table 3.2-1:

**Table 7.1.1-1 — Additional symbols for the purposes of 7.1**

| Symbol      | Description                                                         | Unit                     |
|-------------|---------------------------------------------------------------------|--------------------------|
| $e_{kn}$    | thickness of knuckle                                                | mm                       |
| $e_s$       | minimum thickness of end to limit membrane stress in spherical part | mm                       |
| $e_{kn\ y}$ | minimum thickness of knuckle to avoid axisymmetric yielding         | mm                       |
| $e_{kn\ b}$ | minimum thickness of knuckle to avoid plastic buckling              | mm                       |
| $D_o$       | outside diameter of end                                             | mm                       |
| $D_i$       | inside diameter of end                                              | mm                       |
| $h_i$       | inside height of ellipsoidal end                                    | mm                       |
| $K$         | shape factor for ellipsoidal end                                    | -                        |
| $R_i$       | inside spherical radius for torispherical end                       | mm                       |
| $r_i$       | inside knuckle radius                                               | mm                       |
| $p_s$       | maximum pressure of end to limit membrane stress in spherical part  | MPa (N/mm <sup>2</sup> ) |
| $p_{kn\ y}$ | maximum pressure of knuckle to avoid axisymmetric yielding          | MPa (N/mm <sup>2</sup> ) |
| $p_{kn\ b}$ | maximum pressure of knuckle to avoid plastic buckling               | MPa (N/mm <sup>2</sup> ) |

### 7.1.2 Hemispherical ends

The minimum required thickness of a hemispherical end shall be given by the following equation:

$$e = \frac{p_c D_i}{4fz - p_c} \quad (7.1.2-1)$$

The thickness of the cylindrical part,  $e_{cyl}$ , shall be not less than the minimum thickness of the connected pipe calculated in accordance with 6.1 up to the point A shown in Figure 7.1.2-1 a) and b).

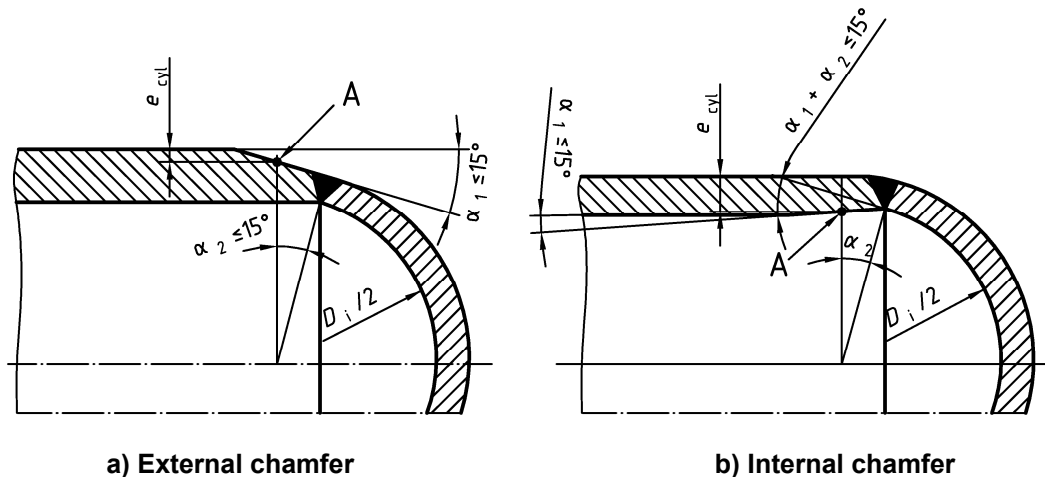


Figure 7.1.2-1 — Hemispherical ends

### 7.1.3 Torispherical ends

This sub-clause shall apply provided that the following conditions are simultaneously fulfilled:

$$r_i \leq 0,2D_i$$

$$r_i \geq 0,06D_i$$

$$r_i \geq 2e$$

$$0,001D_i \leq e \leq 0,08D_i$$

$$R_i \leq D_o$$

If  $e < 0,003D_i$  then the method shall only be applicable:

- for carbon steel and austenitic stainless steel ends; and
- with a calculation temperature  $t_c \leq 100$  °C.

The minimum required thickness  $e$  shall be the greatest of  $e_s$ ,  $e_{kn y}$  and  $e_{kn b}$ , where

$$e_s = \frac{p_c R_i}{2fz - 0,5p_c} \quad (7.1.3-1)$$

$$e_{kn y} = \frac{\beta p_c (0,75R_i + 0,2D_i)}{f} \quad (7.1.3-2)$$

where

$\beta$  is calculated in accordance with 7.1.5

and

$$e_{kn b} = (0,75R_i + 0,2D_i) \left\{ \frac{p_c}{111f_b} \left( \frac{D_i}{r_i} \right)^{0,825} \right\}^{\frac{1}{1,5}} \quad (7.1.3-3)$$

where

$f_b$  is the design stress to prevent buckling:

— for all materials except cold formed austenitic stainless steel:

$$f_b = f \quad (7.1.3-4)$$

— for cold formed austenitic stainless steel:

$$f_b = 1,6f \quad (7.1.3-5)$$

NOTE The factor of 1,6 takes account of the benefit of strain hardening.

For a given geometry, the maximum pressure  $PS$  shall be the minimum of  $p_s$ ,  $p_{kn y}$  and  $p_{kn b}$ ,

where

$$p_s = \frac{2fe_a z}{R_i + 0,5e_a} \quad (7.1.3-6)$$

$$p_{kn y} = \frac{fe_a}{\beta(0,75R_i + 0,2D_i)} \quad (7.1.3-7)$$

where  $\beta$  is calculated in accordance with 7.1.5

$$p_{kn b} = 111f_b \left( \frac{e_a}{0,75R_i + 0,2D_i} \right)^{1,5} \left( \frac{r_i}{D_i} \right)^{0,825} \quad (7.1.3-8)$$

NOTE Where  $e_{kn y} > 0,005 D_i$ , it is not necessary to calculate  $e_{kn b}$  or  $p_{kn b}$ .

It shall be permissible to reduce the thickness of the spherical part of the end to the value of  $e_s$  over a circular area that is not closer to the knuckle than the distance  $\sqrt{R_i e_{kn}}$  as shown in Figure 7.1.3-1.

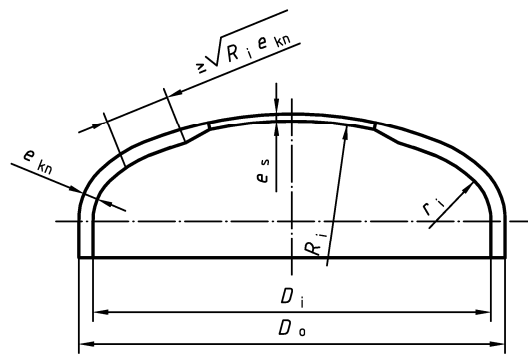


Figure 7.1.3-1 — Torispherical ends

Any cylinder skirt shall meet the requirements of 6.1 for a straight pipe, unless its length is not greater than  $0,2\sqrt{D_i e_{kn}}$ , in which case its thickness and that of the knuckle may be the same.

#### 7.1.4 Ellipsoidal ends

This sub-clause shall apply to ends for which  $1,7 < K < 2,2$  and  $z = 1$ ,

where

$$K = \frac{D_i}{2 h_i} \text{ (see Figure 7.1.4-1).}$$

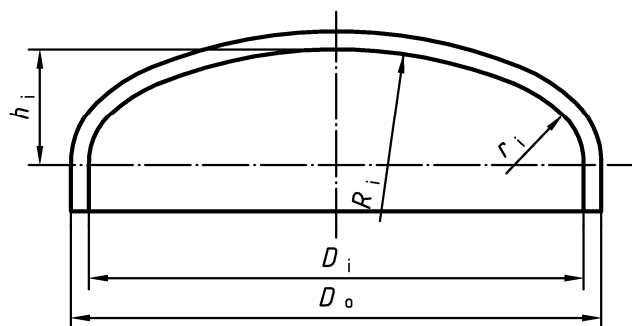


Figure 7.1.4-1 — Ellipsoidal ends

Ellipsoidal ends shall be designed as nominally equivalent torispherical ends with:

$$r_i = D_i (0,5/K - 0,08) \quad (7.1.4-1)$$

and

$$R_i = D_i (0,44K + 0,02) \quad (7.1.4-2)$$