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Deckel für Einsteigeöffnungen nach DIN EN 14025(6.3.6):2008-08

Literatur : www.beuth.de

Mit Modul Deck können Einsteigeöffnungen mit flachen oder gewölbten Deckeln für Tankfahrzeuge nach EN 14025 berechnet werden. Das Programm berechnet die erforderliche Bodendicke und den erforderlichen Schraubenquerschnitt. Nach Programmstart kann eine flache oder gewölbte Bauform gewählt werden. Anschließend erscheint ein entsprechendes Eingabefenster mit den erforderlichen Angaben für die Berechnung. Das Eingabefenster wird dynamisch an die jeweiligen Vorgaben angepasst und blendet nur die erforderlichen Angaben und Berechnungsgleichungen ein. Bei einer durchgehenden Dichtung ist die erforderliche Bodendicke beispielsweise unabhängig von der Schraubenkraft. Bei einer innenliegenden Dichtung werden weitere Eingabewerte und Gleichungen für die statischen Kräfte eingeblendet.

Manche Parameter werden vom Programm automatisch eingetragen. Beispielsweise kann ein Dichtungstyp nach Tabelle 3 in EN 14025 ausgewählt werden und das Programm ermittelt die zugehörigen Dichtungsparameter. Die Schraubenabmessungen für metrische DIN-Schrauben werden ebenfalls automatisch eingetragen.

1 Kreisrunde oder elliptische flache Deckel

Bei Einsteigeöffnungen mit flachen Deckeln können kreisrunde oder elliptische mit innenliegenden oder durchgehenden Dichtungen ausgewählt werden. Die wichtigsten Eingabegrößen sind in Bild 8 und Bild 9 dargestellt.

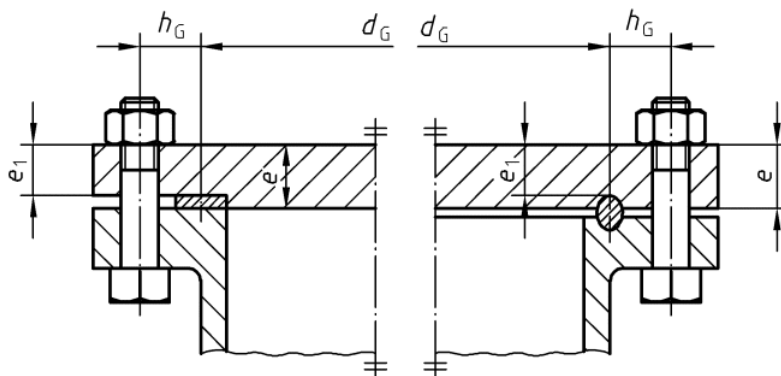


Bild 8: Kreisrunde oder elliptische Deckel mit innenliegender Dichtung

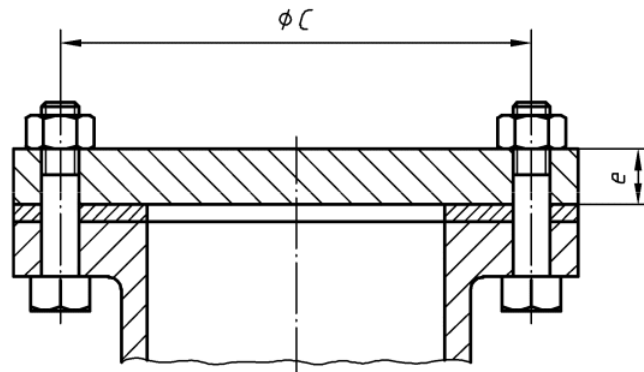


Bild 9: Kreisrunde oder elliptische Deckel mit durchgehender Dichtung
 (nur für Flüssigkeiten)

2 Kreisrunde gewölbte Böden

Bei gewölbten Böden können innenliegende oder durchgehende Dichtungen gewählt werden. Bei einer innenliegenden Dichtung muss der Versatz hr zwischen der Schalenmitte und Flanschmitte an der Schweißnaht 1 in Bild 11 eingegeben werden.

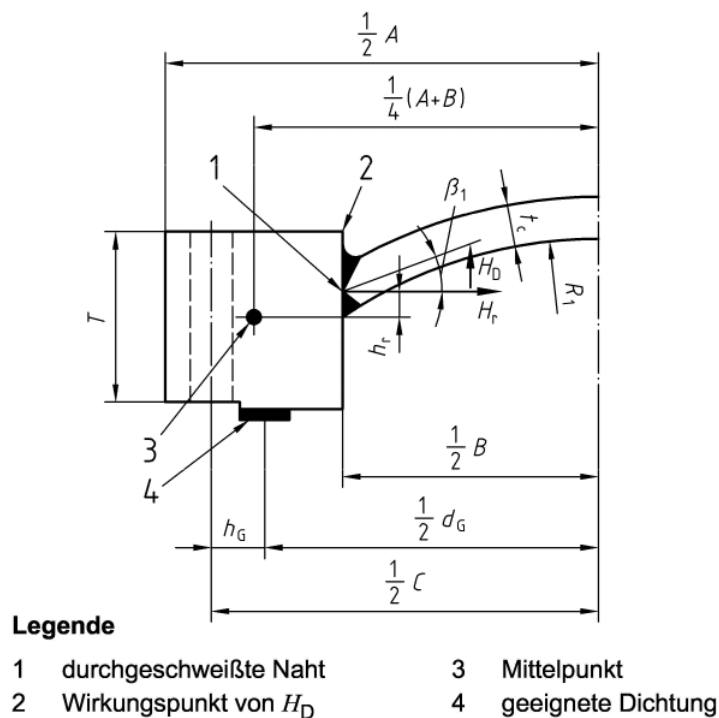


Bild 11: Gewölbte Böden



Covers for manholes acc. DIN EN 14025(6.3.6):2008-08

Literature : www.beuth.com

Flat or dished covers for manholes can be calculated with module DECK according to EN 14025. The program calculates the required cover thickness and the required bolt area. A flat or dished design type can be selected at program start and an appropriate input window follows with the required input values for the calculation. The input window is dynamically adapted according to the specifications and displays only the required input data and equations. For example, the required cover thickness for a full face gasket is independent from the bolt force. For a narrow face gasket additional input data and equations for the static forces are displayed.

Some parameters are automatically inserted by the program. For example, a gasket type according table 3 in EN 14025 can be selected and the program determines the required gasket parameters. The bolt dimensions for metric DIN-bolts are also automatically evaluated.

1 Circular or elliptical flat ends

For man holes with flat covers circular or elliptical and narrow face or full face gaskets can be selected. The most important input values are illustrated in Fig. 8 and Fig.9.

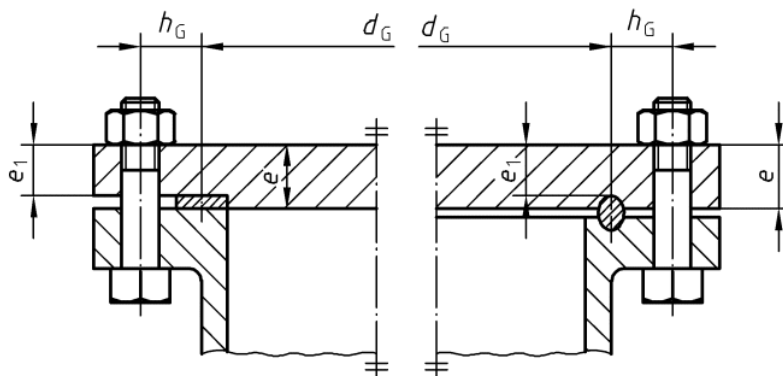


Fig. 8: Circular or elliptical flat ends with narrow face gasket

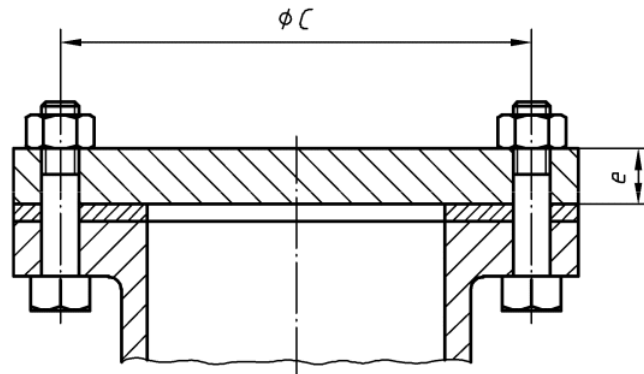
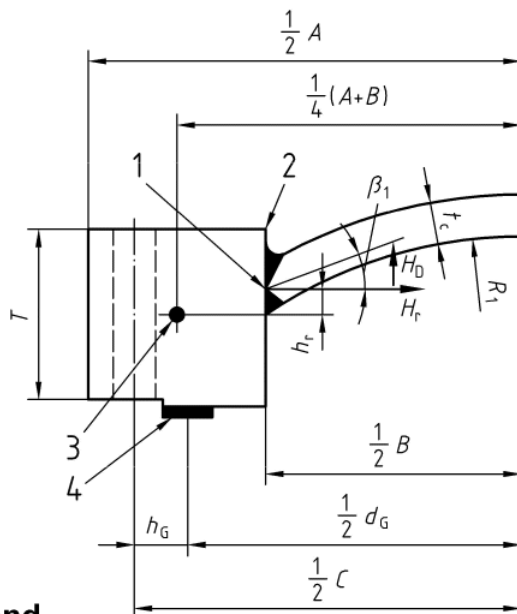


Fig. 9: Circular or elliptical flat ends with full face gasket
 (Only for liquid media)

2 Dished ends

For dished heads narrow or full face gaskets can be selected. For narrow face gaskets the offset hr between shell center and flange center at the welding 1 in Fig. 11 must be specified.



Legend

- | | |
|---------------------------|----------------------|
| 1 Penetration welding | 3 Center |
| 2 Application point of HD | 4 Appropriate gasket |

Fig. 11: Dished heads



Example EN 14025 Appendix B

Example EN 14025 Appendix B

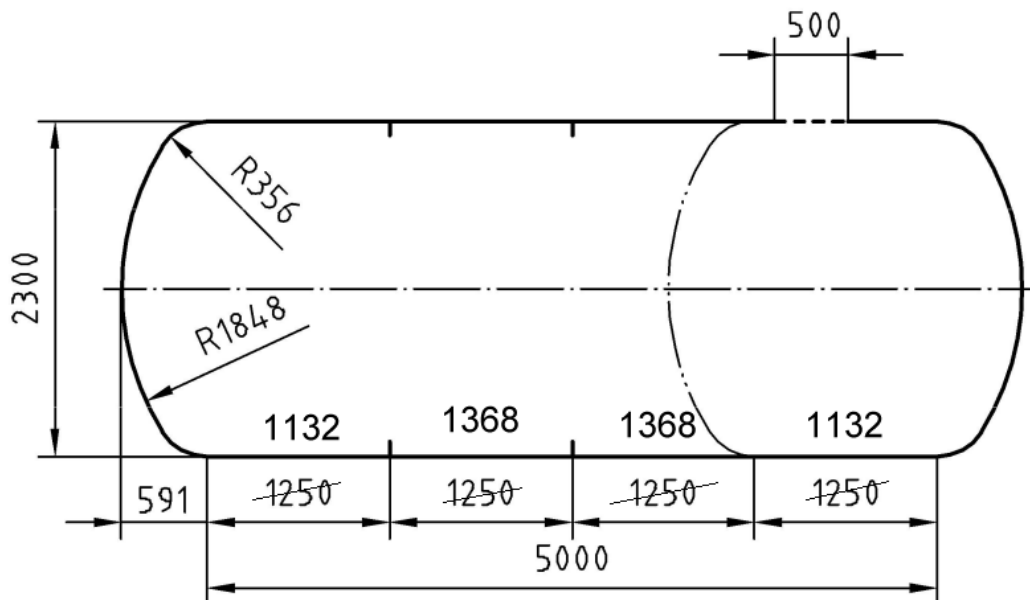


Fig.: Calculation example with dimensions in mmm acc. En 14025

Appearance

Input values:	1.234	or	1.234
Calculated values:	1.234	or	1.234
Critical values:	1.234	or	1.234
Estimated values:	1.234	or	1.234



Example EN 14025 Appendix B

1 EN07 B.5.1

Shells under internal pressure DIN EN 13445-3/7:2003-11 (State Nov.2005) and EN14025:2008-08

7.4.2 Cylindrical shells under internal pressure

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods		

Load case: Operation = 1 / Test = 2	lc	2 1,2
Calculation temperature	t	100 °C
Calculation pressure	P	4 bar
Final wall thickness acc. drawing	en	5 mm
Outside diameter	De	2310 mm
Weld factor ($=\lambda$ acc. EN 14025)	Z	0.8

Material:

Material designation	Number 1.4404 (H)	
Wall thinning allowance	δ_e	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δ_m	0 mm
Sum of allowances	$\Sigma(\delta)$	0 mm
Material strength (R_e, R_p, R_m)	K	260 MPa
Safety factor acc. EN 13445	S	1.05
Allowable stress	f	195 MPa

Result

Inside diameter	Di	2300 mm
Mean diameter	Dm	2305 mm
Geometrical ratio	e/De	0.001278
Analysis thickness $e_n - \Sigma(\delta)$	ea	5 mm
Required thickness	e	2.953 mm
Required thickness with allowances	e_δ	2.953 mm
Maximum permissible pressure	Pmax	0.6768 MPa

Load case	Test
The strength condition is	valid
The geometrical condition is	valid



Example EN 14025 Appendix B

— Allowable stress for testing acc. EN 14025 —

Tensile strength at 20°C	Rm20	530	MPa
Yield stress Re at 20°C (Rp02 or Rp1)	Re	260	MPa
Allowable stress = Min[0.5*Rm20, 0.75*Re]	f	195	MPa

$$f = \text{Min}[0.5 * \mathbf{530} \text{ MPa} , 0.75 * \mathbf{260} \text{ MPa}] = \mathbf{195} \text{ MPa}$$

— Equations —

$$e = P * Di / (2 * f * Z - P) = \mathbf{2.953} \text{ mm} \quad (7.4-1), \text{ EN 14025 (1)}$$

$$e = \frac{0.4 * 2300}{2 * \mathbf{195} * 0.8 - 0.4}$$

The strength condition is **valid** :

$$ea = \mathbf{5} \geq \mathbf{2.953} = e$$

The geometrical condition is **valid** :

$$e/De = \mathbf{0.001278} = \mathbf{2.953} / \mathbf{2310} \leq 0.16$$

$$P_{\max} = P * f * Z * ea / Dm = \mathbf{0.6768} \text{ MPa} \quad (7.4-3)$$

$$P_{\max} = \frac{0.4 * \mathbf{195} * 0.8 * \mathbf{5}}{\mathbf{2305}}$$



Example EN 14025 Appendix B

2 EN07 B.5.2

Shells under internal pressure DIN EN 13445-3/7:2003-11 (State Nov.2005) and EN14025:2008-08

7.5.3 Torispherical ends under internal pressure - Korbbogenboden type

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods		
Load case: Operation = 1 / Test = 2		2 1,2
Calculation temperature	t	100 °C
Calculation pressure (ext. pres. acc.8.8.2: P<0)	P	4 bar
Final wall thickness according to drawing	en	5 mm
Outside diameter	De	2310 mm
Weld factor (=λ acc. EN 14025)	Z	1

Material:

Material designation	Number	1.4404 (H)
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Material strength (Re, Rp, Rm)	K	260 MPa
0.2% proof stress, operation	Rp02T	166 MPa
0.2% proof stress, room temperature	Rp02R	220 MPa
Safety factor	S	1.05
Allowable stress (f=fd acc EN14025)	f	195 MPa
Allowable buckling stress fb=fd=f acc. EN14025	fb	195 MPa

— Results: —

Maximum permissible pressure Min(Ps, Py, Pb)	Pmax	0.6542	MPa
Calculation thickness without allowance	ea	5	mm
Required wall thickness Max(es, ey, eb)	e	3.602	mm
Required thickness with allowances	eδ	3.602	mm
Required thickness of spherical part, membrane	es	1.896	mm
Required thickness of knuckle, yield 7.5.3.5	ey	2.988	mm
Required thickness of knuckle, plastic buckling	eb	3.602	mm
Inside diameter	Di	2300	mm
Inside radius of spherical part	R	1848	mm
Inside radius of knuckle	r	355.7	mm
Geometrical ratio	R/De	0.8	
Geometrical ratio	r/De	0.154	
Calculation coefficient	β	0.7891	
Thickness reduction distance √(R*e) acc. 7.5.3.4	l	81.59	mm
Limit skirt length 0.2*√(Di*e) acc. 7.5.3.4	h	18.2	mm

Loading case: **Test**
 The strength condition is **valid**
 The geometrical conditions are **valid**

— Allowable stress for testing acc. EN 14025 —

Tensile strength at 20°C	Rm20	530	MPa
Yield stress Re at 20°C (Rp02 or Rp1)	Re	260	MPa
Allowable stress = Min[0.5*Rm20, 0.75*Rppe]	f	195	MPa
f = Min[0.5* 530 MPa , 0.75* 260 MPa] = 195 MPa			



Example EN 14025 Appendix B

— Equations

$$e_s = P * R / (2 * f * Z - 0.5 * P) = 1.896 \text{ mm}$$

$$e_s = 0.4 * 1848 / (2 * 195 * (1 - 0.5 * 0.4))$$

$$e_y = \beta * P * (0.75 * R + 0.2 * D_i) / f = 2.988 \text{ mm}$$

$$e_y = 0.7891 * 0.4 * (0.75 * 1848 + 0.2 * 2300) / 195$$

The auxiliary values for β (X, Y, Z, N at the end of mask) have been calculated iteratively with the required thickness e.

$$e_b = (0.75 * R + 0.2 * D_i) * \left| \frac{P * (D_i/r)^{0.825}}{111 * f_b} \right|^{(2/3)} = 3.602$$

$$e_b = (0.75 * 1848 + 0.2 * 2300) * \left| \frac{0.4 * (2300 / 355.7)^{0.825}}{111 * 195} \right|^{(2/3)}$$

$$e = \text{Max}(e_b, e_y, e_s) = \text{Max}(3.602, 2.988, 1.896) = 3.602$$

The strength condition is **valid** :

$$e_a = 5 \geq 3.602 = e$$

The strength condition is **valid** :

$$\begin{aligned} \text{Max}(0.06 * D_i, 2 * e_a) &\leq r \leq D_i / 5 \\ \text{Max}(138, 10) &\leq 355.7 \leq 460 \end{aligned}$$

$$D_e / 1000 \leq e_a \leq 0.08 * D_e, \quad 2.31 \leq 5 \leq 184.8$$

$$R = 355.7 \leq 2310 = D_e$$



Example EN 14025 Appendix B

— Rating of the permissible pressure

$$P_s = 2 * f * Z * e_a / (R + 0.5 * e_a) = 1.054$$

$$P_s = 2 * 195 * 1 * 5 / (1848 + 0.5 * 5)$$

$$P_y = f * e_a / [\beta * (0.75 * R + 0.2 * D_i)] = 0.6693$$

$$P_y = 195 * 5 / [0.7891 * (0.75 * 1848 + 0.2 * 2300)]$$

$$P_b = 111 * f_b * [e_a / (0.75 * R + D_i / 5)]^{1.5} * (r / D_i)^{0.825} = 0.6542$$

$$P_b = 111 * 195 * [5 / (3 * 1848 / 4 + 2300 / 5)]^{1.5} * (355.7 / D_i)^{0.825}$$

$$P_{max} = \min(P_b, P_y, P_s) = \max(0.6542, 0.6693, 1.054) = 0.6542$$

$$\text{Parameter: } Y = 0.001949, Z = 2.71, X = 0.1547, N = 0.8447$$

$$\text{Parameter: } \beta_{0.06} = 1.668, \beta_{0.1} = 1.104, \beta_{0.2} = 0.5281$$

$$\beta = 0.7891$$



Example EN 14025 Appendix B

3 EN08 B.5.3

Shells under external pressure DIN EN 13445-3/7 edition 2003-11 (state Nov.2005)

Spherical shells acc. to 8.7.1

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods, section 6.4		
Load case: Operation = 1 / Test = 2	lc	2
Safety factor acc. to section 8.4.4	S	1.1
Calculation temperature	t	20 °C
Calculation pressure	P	4 bar
Final wall thickness with allowances	en	8 mm
Mean radius of the shell	R	1848 mm

Material properties of the spherical shell:

Material designation	Number 1.4404 (H)	
Poisson's ratio	nu	0.3
Austenitic steel (1=yes, 2=no)		1
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Strength acc. to specification (Re, Rp, Rm)	K	260
Safety factor according to section 8.4.4	S	1.1
Modulus of elasticity	E	200000 MPa
0.2% proof stress at operation temperature	Rp02	220 N/mm ²
0.2% proof stress at room temperature	Rp02p	220 MPa
allowable elastic limit (Rpx, Rpx/1.25)	σe	176 N/mm ²
Curvature deviation greater than 30%? 1=yes, 2=no, Abw		2
based on a max. arc length of measuring range	Bog	
maximum radius of curvature	RKmax	mm

Results:

Calculation thickness without allowances	ea	8 mm
Limit pressure for circumferential yield (8.7.1-1)	py	1.524 MPa
Theoretical instability pressure (8.7.1-2)	pm	4.535 MPa
Ratio pm/py	pm/py	2.976
Ratio pr/py (Fig. 8.5-5)	pr/py	0.4327
allowable pressure (pr/S)	pzul	0.5994 MPa

Condition: $P = 0.4 < \mathbf{0.5994} = p_{zul}$

The strength condition is **valid**
for load case **Test**



Example EN 14025 Appendix B

$$p_y = 1.524 = 2 * \sigma_e * e_a / R = 2 * 176 * 8 / 1848 -$$

$$p_m = 4.535 = 1.21 * E * e_a^2 / R^2 = 1.21 * 200000 * 8^2 / 1848^2$$

$$p_{zul} = \begin{cases} p_r/S = 0.6593 / 1.1, & \text{shape deviation} < 30\% \\ p_r/S * (1.3 * R / R_{Kmax})^2 = 0.6593 / 1.1 * (1.3 * 1848 /)^2 \end{cases}$$

Max. measuring length of curvature deviation acc. to 8.7.2:

$$B_{og} = = 2.4 * \sqrt{(e_a * R_{Kmax})} = 2.4 * \sqrt{(8 *)}$$

Vessel ends acc. to section 8.8:

Semi-spherical ends shall be designed acc. to the rules for spheres.
 The mean sphere radius for torispherical shells is R=crown outside radius
 and for the stress calculation acc. 7.5.2 (inside pressure P<0 with
 module EN07) holds N=1.

For semi-ellipsoidal ends the mean sphere radius is $R=D^2/(4h)$.



Example EN 14025 Appendix B

4 EN07 B.6.1

Shells under internal pressure DIN EN 13445-3/7:2003-11 (State Nov.2005) and EN14025:2008-08

7.4.2 Cylindrical shells under internal pressure

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods		

Load case: Operation = 1 / Test = 2	lc	1 1,2
Calculation temperature	t	100 °C
Calculation pressure	P	3 bar
Final wall thickness acc. drawing	en	3.02 mm
Outside diameter	De	2306 mm
Weld factor (=λ acc. EN 14025)	Z	0.8

Material:

Material designation	Number	1.4404 (H)
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Material strength (Re, Rp, Rm)	K	430 MPa
Safety factor acc. EN 13445	S	3
Allowable stress	f	143.3 MPa

Result

Inside diameter	Di	2300 mm
Mean diameter	Dm	2303 mm
Geometrical ratio	e/De	0.001306
Analysis thickness en - Σ(δ)	ea	3.02 mm
Required thickness	e	3.013 mm
Required thickness with allowances	eδ	3.013 mm
Maximum permissible pressure	Pmax	0.3007 MPa

Load case **Operation**
 The strength condition is **valid**
 The geometrical condition is **valid**

Equations

Allowable stress for the selected load case:

$$f = K/S = \frac{430 \text{ MPa}}{3} = 143.3 \text{ MPa}$$

$$e = \frac{P * Di}{(2 * f * Z - P)} = \frac{3 * 2300}{(2 * 143.3 * 0.8 - 3)} = 3.013 \text{ mm} \quad (7.4-1), \text{ EN 14025(1)}$$

The strength condition is **valid** :

$$ea = 3.02 \geq 3.013 = e$$

The geometrical condition is **valid** :

$$e/De = 0.001306 = \frac{3.013}{2306} \leq 0.16$$

$$P_{max} = \frac{P * f * Z * ea}{Dm} = \frac{3 * 143.3 * 0.8 * 3.02}{2303} = 0.3007 \text{ MPa} \quad (7.4-3)$$



Example EN 14025 Appendix B

5 EN07 B.6.2

Shells under internal pressure DIN EN 13445-3/7:2003-11 (State Nov.2005) and EN14025:2008-08

7.5.3 Torispherical ends under internal pressure - Korbboogenboden type

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods		

Load case: Operation = 1 / Test = 2		1 1,2
Calculation temperature	t	100 °C
Calculation pressure (ext. pres. acc.8.8.2: P<0)	P	3 bar
Final wall thickness according to drawing	en	3.654 mm
Outside diameter	De	2310 mm
Weld factor (=λ acc. EN 14025)	Z	1

Material:

Material designation	Number 1.4404 (H)	
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Material strength (Re, Rp, Rm)	K	430 MPa
0.2% proof stress, operation	Rp02T	166 MPa
0.2% proof stress, room temperature	Rp02R	220 MPa
Safety factor	S	3
Allowable stress (f=fd acc EN14025)	f	143.3 MPa
Allowable buckling stress fb=fd=f acc. EN14025	fb	143.3 MPa

— Results: —

Maximum permissible pressure Min(Ps, Py, Pb)	Pmax	0.3 MPa
Calculation thickness without allowance	ea	3.654 mm
Required wall thickness Max(es, ey, eb)	e	3.654 mm
Required thickness with allowances	eδ	3.654 mm
Required thickness of spherical part, membrane	es	1.935 mm
Required thickness of knuckle, yield 7.5.3.5	ey	3.051 mm
Required thickness of knuckle, plastic buckling	eb	3.654 mm
Inside diameter	Di	2303 mm
Inside radius of spherical part	R	1848 mm
Inside radius of knuckle	r	355.7 mm
Geometrical ratio	R/De	0.8
Geometrical ratio	r/De	0.154
Calculation coefficient	β	0.7894
Thickness reduction distance √(R*e) acc. 7.5.3.4	l	82.17 mm
Limit skirt length 0.2*√(Di*e) acc. 7.5.3.4	h	18.35 mm

Loading case:	Operation
The strength condition is	valid
The geometrical conditions are	valid



Example EN 14025 Appendix B

— Equations

Allowable stress for the selected load case:

$$f = K/S = 430 \text{ MPa} / 3 = 143.3 \text{ MPa}$$

$$e_s = P * R / (2 * f * Z - 0.5 * P) = 1.935 \text{ mm}$$

$$e_s = 0.3 * 1848 / (2 * 143.3 * 1 - 0.5 * 0.3)$$

$$e_y = \beta * P * (0.75 * R + 0.2 * D_i) / f = 3.051 \text{ mm}$$

$$e_y = 0.7894 * 0.3 * (0.75 * 1848 + 0.2 * 2303) / 143.3$$

The auxiliary values for β (X, Y, Z, N at the end of mask) have been calculated iteratively with the required thickness e.

$$e_b = (0.75 * R + 0.2 * D_i) * \left| \frac{P * (D_i/r)^{0.825}}{111 * f_b} \right|^{(2/3)} = 3.654$$

$$e_b = (0.75 * 1848 + 0.2 * 2303) * \left| \frac{0.3 * (2303/355.7)^{0.825}}{111 * 143.3} \right|^{(2/3)}$$

$$e = \text{Max}(e_b, e_y, e_s) = \text{Max}(3.654, 3.051, 1.935) = 3.654$$

The strength condition is **valid** :

$$e_a = 3.654 \geq 3.654 = e$$

The strength condition is **valid** :

$$\text{Max}(0.06 * D_i, 2 * e_a) \leq r \leq D_i/5$$

$$\text{Max}(138.2, 7.308) \leq 355.7 \leq 460.5$$

$$D_e/1000 \leq e_a \leq 0.08 * D_e, \quad 2.31 \leq 3.654 \leq 184.8$$

$$R = 355.7 \leq 2310 = D_e$$



Example EN 14025 Appendix B

— Rating of the permissible pressure

$$P_s = 2 * f * Z * e_a / (R + 0.5 * e_a) = \mathbf{0.5663}$$

$$P_s = 2 * \mathbf{143.3} * 1 * \mathbf{3.654} / (\mathbf{1848} + 0.5 * \mathbf{3.654})$$

$$P_y = f * e_a / [\beta * (0.75 * R + 0.2 * D_i)] = \mathbf{0.3593}$$

$$P_y = \mathbf{143.3} * \mathbf{3.654} / [\mathbf{0.7894} * (0.75 * \mathbf{1848} + 0.2 * \mathbf{2303})]$$

$$P_b = 111 * f_b * [e_a / (0.75 * R + D_i / 5)]^{1.5} * (r / D_i)^{0.825} = \mathbf{0.3}$$

$$P_b = 111 * \mathbf{143.3} * [\mathbf{3.654} / (3 * \mathbf{1848} / 4 + \mathbf{2303} / 5)]^{1.5} * (\mathbf{355.7} / D_i)^{0.825}$$

$$P_{\max} = \min(P_b, P_y, P_s) = \max(\mathbf{0.3}, \mathbf{0.3593}, \mathbf{0.5663}) = \mathbf{0.3}$$

$$\text{Parameter: } Y = \mathbf{0.001977}, \quad Z = \mathbf{2.704}, \quad X = \mathbf{0.1545}, \quad N = \mathbf{0.8447}$$

$$\text{Parameter: } \beta_{0.06} = \mathbf{1.665}, \quad \beta_{0.1} = \mathbf{1.102}, \quad \beta_{0.2} = \mathbf{0.528}$$

$$\beta = \mathbf{0.7894}$$



Example EN 14025 Appendix B

6 EN08 B.6.3

Shells under external pressure DIN EN 13445-3/7 edition 2003-11 (state Nov.2005)

Spherical shells acc. to 8.7.1

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods, section 6.4		
Load case: Operation = 1 / Test = 2	lc	1
Safety factor acc. to section 8.4.4	S	1.5
Calculation temperature	t	100 °C
Calculation pressure	P	3 bar
Final wall thickness with allowances	en	8 mm
Mean radius of the shell	R	1848 mm

Material properties of the spherical shell:

Material designation	Number	1.4404 (H)
Poisson's ratio	nu	0.3
Austenitic steel (1=yes, 2=no)		1
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Strength acc. to specification (Re, Rp, Rm)	K	430
Safety factor according to section 8.4.4	S	1.5
Modulus of elasticity	E	194000 MPa
0.2% proof stress at operation temperature	Rp02	166 N/mm ²
0.2% proof stress at room temperature	Rp02p	220 MPa
allowable elastic limit (Rpx, Rpx/1.25)	σe	132.8 N/mm ²
Curvature deviation greater than 30%? 1=yes, 2=no,	Abw	2
based on a max. arc length of measuring range	Bog	
maximum radius of curvature	RKmax	mm

Results:

Calculation thickness without allowances	ea	8 mm
Limit pressure for circumferential yield (8.7.1-1)	py	1.15 MPa
Theoretical instability pressure (8.7.1-2)	pm	4.399 MPa
Ratio pm/py	pm/py	3.826
Ratio pr/py (Fig. 8.5-5)	pr/py	0.4992
allowable pressure (pr/S)	pzul	0.3827 MPa

Condition: $P = 0.3 < 0.3827 = p_{zul}$

The strength condition is **valid**
for load case **Design**



Example EN 14025 Appendix B

$$p_y = 1.15 = 2 * \sigma_e * e_a / R = 2 * 132.8 * 8 / 1848 -$$

$$p_m = 4.399 = 1.21 * E * e_a^2 / R^2 = 1.21 * 194000 * 8^2 / 1848^2$$

$$p_{zul} = \begin{cases} p_r/S = 0.574 / 1.5, & \text{shape deviation} < 30\% \\ p_r/S * (1.3 * R / R_{Kmax})^2 = 0.574 / 1.5 * (1.3 * 1848 /)^2 \end{cases}$$

Max. measuring length of curvature deviation acc. to 8.7.2:

$$B_{og} = = 2.4 * \sqrt{(e_a * R_{Kmax})} = 2.4 * \sqrt{(8 *)}$$

Vessel ends acc. to section 8.8:

Semi-spherical ends shall be designed acc. to the rules for spheres.

The mean sphere radius for torispherical shells is R=crown outside radius
and for the stress calculation acc. 7.5.2 (inside pressure P<0 with
module EN07) holds N=1.

For semi-ellipsoidal ends the mean sphere radius is $R=D^2/(4h)$.



Example EN 14025 Appendix B

7 EN08 B.6.4.2

Shells under external pressure DIN EN 13445-3/7 edition 2003-11 (state Nov.2005)

Cylindrical shells with light stiffeners according to section 8.5.3.6

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods, section 6.4		
Load case: Operation = 1 / Test = 2		1
Safety factor according to section 8.4.4 (1.5, 1.1)S		1.1
Calculation temperature	t	100 °C
Calculation pressure	P	0.4 bar
Final wall thickness of cylinder with allowances	en	3.65 mm
Web thickness of a stiffener without allowances	ew	8 mm

Material properties of the cylinder:

Material designation	Number 1.4404 (H)	
Austenitic steel? yes=1, no=2		1
Poisson's ratio (e.g. =0.3)	nu	0.3
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Total allowance	Σ(δ)	0 mm
Strength acc. specification (Re, Rp1, Rm)	RK	430
Modulus of elasticity	E	195000 MPa
0.2% proof stress at operation temperature	Rp02	166 N/mm ²
0.2% proof stress at room temperature	Rp02p	220 MPa
Allowable elastic limit (Rp02, Rp02/1.25)	σe	132.8 N/mm ²

Material properties of the stiffener:

Material designation	Number 1.4404 (H)	
Austenitic steel? yes=1, no=2		1
Modulus of elasticity	ES	195000 MPa
Poisson's ratio (e.g. =0.3)	nuS	0.3
0.2% proof stress at operation temperature	Rp02S	166 MPa
0.2% proof stress at room temperature	Rp02Sp	220 MPa
Allowable elastic limit (Rp02, Rp02/1.25)	σeL	132.8 N/mm ²
Additional safety factor for cold or hot rolled steel (1,33 or 1,20)	Sf	1.33

Rem.:

Definition of cylinder length according to Table 8.5-1:

a) Analysis of failure of a cylinder section according to 8.5.3.4:

- (1) between stiffener and cylinder end,
- (2) between two stiffeners.

Please enter (1, 2) for the concerned section 1

For a stiffener at the cylinder end, please specify the distance L_s' from the end, height h' of dished end, and profile centroid w_1'' , F. 8.5-6, 8.5-8

$$L = (L_s' - w_1'') + 0.4 * h' \quad (8.5.3-1)$$

$$L = (1132 - 0) + 0.4 * 591$$

Result for the unsupported shell length: L = 1368 mm



Example EN 14025 Appendix B

Please specify also the distance L_s'' of two stiffeners, and the appropriate profile centroids w_2' and w_2'' acc. to Fig. 8.5-8

$$L = L_s'' - w_2' - w_2'' \quad (8.5.3-2)$$

$$L = 1368 - 0 - 0$$

Result for the unsupported shell length: $L = 1368$ mm

b) Verification of the elastic stability of single stiffeners (8.5.3.6.2):

- (1) for the first or last stiffener
 - (2) for an intermediate stiffener
- Please specify the type (1 or 2) of stiffener

For stiffeners at the cylinder end with values above

$$L_s = (L_s' + 0.4 \cdot h' + L_s'')/2 \quad (8.5.3-6)$$

$$L_s = (1132 + 0.4 \cdot 591 + 1368)/2$$

Result for the mean distance: $L_s = 1368$ mm

Please specify the distance L_s''' of adjacent stiffeners

$$L_s = (L_s'' + L_s''')/2 \quad (8.5.3-7)$$

$$L_s = (1368 + 1368)/2$$

The mean distance results in: $L_s = 1368$ mm

c) Additional specifications for the whole cylinder (8.5.3.6.2):

Please specify the cylinder length L_{cyl} and end height h'' acc. Fig. 8.5-6

$$LH = (L_{cyl} + 0.4 \cdot h' + 0.4 \cdot h'') \quad (8.5.3-10)$$

$$LH = (5000 + 0.4 \cdot 591 + 0.4 \cdot 591)$$

Result for the total length of the cylinder: $LH = 5473$ mm

Examination of failure between stiffeners acc. to 8.5.3.4

Specification:

Calculation thickness of cylinder without allowances	ea	3.65	mm
Mean radius of shell	R	1152	mm
Radius of stiffener centroid	R _s	1127	mm
Cross sectional area of stiffener	AS	368	mm ²
Width of a stiffener in contact with shell	w	8	mm

Results:

Wave number of buckling shape	ncyl	10	
Factor acc. to Eq. (8.5.3-19)	δ	0.01982	
Calculation parameter of Eq. (8.5.3-22)	G	0	
Calculation parameter of Eq. (8.5.3-21)	N	1	
Factor Gamma according to Eq. (8.5.3-16)	γ	0.4179	
Modified area of stiffener, (8.5.3-17)	A _m	384.4	mm ²
Calculation parameter acc. to (8.5.3-18)	B	0.8903	
Limit pressure for circumferential yield (8.5.3-15)	p _y	0.4208	MPa
Unsupported shell length between stiffeners	L	1368	mm
Parameter π^*R/L	Z	2.644	
Circumferential strain at collapse	ε	0.000143	
Theoretical elastic buckling pressure (8.5.2-5)	p _m	0.08804	MPa
Real buckling pressure (Fig. 8.5-5)	p _r	0.04402	MPa
Allowable pressure (p _r /S)	p _{zul}	0.04002	MPa

Condition (8.5.2-8) $P < p_r/S$ is: **valid**
 Load case: **Design**



Example EN 14025 Appendix B

Stability of a stiffened cylinder according to 8.5.3.6.2

a) Elastic instability, Specification of the stiffener

+1=internal, -1=external stiffener, please enter +1,-1	1	
Manufacturing safety factor of stiffener	Sf	1.33
Cross sectional area of stiffener	AS	368 mm ²
Area moment at stiffener centroid	IS	64891 mm ⁴

Results:

Mean distance between the stiffeners (Table 8.5-1)	LS	1368	mm
Total length of the cylinder (Table 8.5-1)	LH	5473	mm
Number of circumferential buckling waves	n	4	
Factor according to Eq. (8.5.3-25)	β	0.000046	
Combined cross sectional area of stiffened shell	Ae	731.4	mm ²
Calculation parameter	Xe	14.32	
Effective Length acc. to (8.5.3-34) (u=	Le	99.57	mm
x= 0.0507, Y1= 1.565, Y2= 1.2, Y3= 0.7211)			
Area moment of shell and stiffener	Ie	178029	mm ⁴
Theoretical elastic buckling pressure	pg	0.2778	MPa

Condition (8.5.3-31): $P < pg / (S \cdot Sf)$ is: **valid**

b) Maximum stress in the stiffener, additional specifications:

Contact width of the stiffener	wi	8	mm
Outside radius of the stiffener	Rf	1104	mm

Results

Modified area of stiffener (8.5.3-17)	Am	384.4	mm ²
Factor (8.5.3-19)	δ	0.01982	
Factor (8.5.3-49)	dquer	35.34	
Limit pressure for circumferential yield	pys	0.9335	MPa
Maximum stress in the stiffener (8.5.3-46)	σs	128.1	MPa

Condition (8.5.3-50) $0 < \sigma_s < \sigma_{eL} = 132.8$ is: **valid**

Sideways tripping of stiffeners

a) Stiffener with rectangular profile? (1=yes, 2=no)

Radial height of the profile	d	46	mm
Width of the profile	ew	8	mm

b) Specifications for non-rectangular profile, Figs. 8.5-14 to 8.5-17

Angle profile acc. to Fig. 8.5-16 (yes=1, no=2)			
Radial height of the web plate between flanges	d	46	mm
Web thickness of the stiffener	ew	8	mm
Thickness of the profile flange	ef		mm
Leg length of the profile flange	wf		mm
Shell outside radius to the web plate	ri		mm
Coefficient for non-rectangular profile	C		

Instability stress for sideways tripping	σi	765.3	MPa
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Condition: $\sigma_i > \sigma_{eL}$ is **valid**

For stiffeners with flange at the remote edge of the web plate:

Limit dimensions according to (8.5.3-63/64)



Example EN 14025 Appendix B

8 EN08 B.6.4.4

Shells under external pressure DIN EN 13445-3/7 edition 2003-11 (state Nov.2005)

Cylindrical shells with light stiffeners according to section 8.5.3.6

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods, section 6.4		
Load case: Operation = 1 / Test = 2		1
Safety factor according to section 8.4.4 (1.5, 1.1)S		1.1
Calculation temperature	t	100 °C
Calculation pressure	P	0.4 bar
Final wall thickness of cylinder with allowances	en	3.65 mm
Web thickness of a stiffener without allowances	ew	8 mm

Material properties of the cylinder:

Material designation	Number 1.4404 (H)	
Austenitic steel? yes=1, no=2		1
Poisson's ratio (e.g. =0.3)	nu	0.3
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Total allowance	Σ(δ)	0 mm
Strength acc. specification (Re, Rp1, Rm)	RK	430
Modulus of elasticity	E	195000 MPa
0.2% proof stress at operation temperature	Rp02	166 N/mm ²
0.2% proof stress at room temperature	Rp02p	220 MPa
Allowable elastic limit (Rp02, Rp02/1.25)	σe	132.8 N/mm ²

Material properties of the stiffener:

Material designation	Number 1.4404 (H)	
Austenitic steel? yes=1, no=2		1
Modulus of elasticity	ES	195000 MPa
Poisson's ratio (e.g. =0.3)	nuS	0.3
0.2% proof stress at operation temperature	Rp02S	166 MPa
0.2% proof stress at room temperature	Rp02Sp	220 MPa
Allowable elastic limit (Rp02, Rp02/1.25)	σeL	132.8 N/mm ²
Additional safety factor for cold or hot rolled steel (1,33 or 1,20)	Sf	1.33

Rem.:

Definition of cylinder length according to Table 8.5-1:

a) Analysis of failure of a cylinder section according to 8.5.3.4:

- (1) between stiffener and cylinder end,
- (2) between two stiffeners.

Please enter (1, 2) for the concerned section 1

For a stiffener at the cylinder end, please specify the distance L_s' from the end, height h' of dished end, and profile centroid w_1'' , F. 8.5-6, 8.5-8

$$L = (L_s' - w_1'') + 0.4 * h' \quad (8.5.3-1)$$

$$L = (\quad - \quad) + 0.4 * 591$$

Result for the unsupported shell length: L = 1368 mm



Example EN 14025 Appendix B

Please specify also the distance L_s'' of two stiffeners, and the appropriate profile centroids w_2' and w_2'' acc. to Fig. 8.5-8

$$L = L_s'' - w_2' - w_2'' \quad (8.5.3-2)$$

$$L = - -$$

Result for the unsupported shell length: $L = 1368 \text{ mm}$

b) Verification of the elastic stability of single stiffeners (8.5.3.6.2):

(1) for the first or last stiffener

(2) for an intermediate stiffener

Please specify the type (1 or 2) of stiffener

For stiffeners at the cylinder end with values above

$$L_s = (L_s' + 0.4 \cdot h' + L_s'')/2 \quad (8.5.3-6)$$

$$L_s = (\quad + 0.4 \cdot 591 + \quad)/2$$

Result for the mean distance: $L_s = \quad \text{mm}$

Please specify the distance L_s''' of adjacent stiffeners

$$L_s = (L_s'' + L_s''')/2 \quad (8.5.3-7)$$

$$L_s = (\quad + \quad)/2$$

The mean distance results in: $L_s = \quad \text{mm}$

c) Additional specifications for the whole cylinder (8.5.3.6.2):

Please specify the cylinder length L_{cyl} and end height h'' acc. Fig. 8.5-6

$$LH = (L_{cyl} + 0.4 \cdot h' + 0.4 \cdot h'') \quad (8.5.3-10)$$

$$LH = (5000 + 0.4 \cdot 591 + 0.4 \cdot 591)$$

Result for the total length of the cylinder: $LH = 5473 \text{ mm}$

Examination of failure between stiffeners acc. to 8.5.3.4

Specification:

Calculation thickness of cylinder without allowances	ea	3.65	mm
Mean radius of shell	R	1152	mm
Radius of stiffener centroid	R _s	1127	mm
Cross sectional area of stiffener	AS	368	mm ²
Width of a stiffener in contact with shell	w	8	mm

Results:

Wave number of buckling shape	ncyl	10	
Factor acc. to Eq. (8.5.3-19)	δ	0.01982	
Calculation parameter of Eq. (8.5.3-22)	G	0	
Calculation parameter of Eq. (8.5.3-21)	N	1	
Factor Gamma according to Eq. (8.5.3-16)	γ	0.4179	
Modified area of stiffener, (8.5.3-17)	A _m	384.4	mm ²
Calculation parameter acc. to (8.5.3-18)	B	0.8903	
Limit pressure for circumferential yield (8.5.3-15)	p _y	0.4208	MPa
Unsupported shell length between stiffeners	L	1368	mm
Parameter π^*R/L	Z	2.645	
Circumferential strain at collapse	ε	0.000143	
Theoretical elastic buckling pressure (8.5.2-5)	p _m	0.08807	MPa
Real buckling pressure (Fig. 8.5-5)	p _r	0.04403	MPa
Allowable pressure (p _r /S)	p _{zul}	0.04003	MPa

Condition (8.5.2-8) $P < p_r/S$ is: **valid**

Load case: **Design**



Example EN 14025 Appendix B

Stability of a stiffened cylinder according to 8.5.3.6.2

a) Elastic instability, Specification of the stiffener

+1=internal, -1=external stiffener, please enter +1,-1		1	
Manufacturing safety factor of stiffener	Sf	1.33	
Cross sectional area of stiffener	AS	368	mm ²
Area moment at stiffener centroid	IS	64891	mm ⁴

Results:

Mean distance between the stiffeners (Table 8.5-1)	LS	1368	mm
Total length of the cylinder (Table 8.5-1)	LH	5473	mm
Number of circumferential buckling waves	n	4	
Factor according to Eq. (8.5.3-25)	β	0.000046	
Combined cross sectional area of stiffened shell	Ae	731.4	mm ²
Calculation parameter	Xe	14.32	
Effective Length acc. to (8.5.3-34) (u=	Le	99.57	mm
x= 0.0507 , Y1= 1.565 , Y2= 1.2 , Y3= 0.7211)			
Area moment of shell and stiffener	Ie	178029	mm ⁴
Theoretical elastic buckling pressure	pg	0.2778	MPa

Condition (8.5.3-31): $P < pg / (S \cdot Sf)$ is: **valid**

b) Maximum stress in the stiffener, additional specifications:

Contact width of the stiffener	wi	8	mm
Outside radius of the stiffener	Rf	1104	mm

Results

Modified area of stiffener (8.5.3-17)	Am	384.4	mm ²
Factor (8.5.3-19)	δ	0.01982	
Factor (8.5.3-49)	dquer	35.34	
Limit pressure for circumferential yield	pys	0.9335	MPa
Maximum stress in the stiffener (8.5.3-46)	σs	128.1	MPa

Condition (8.5.3-50) $0 < \sigma_s < \sigma_{eL} =$ **132.8** is: **valid**

Sideways tripping of stiffeners

a) Stiffener with rectangular profile? (1=yes, 2=no)

Radial height of the profile	d	46	mm
Width of the profile	ew	8	mm

b) Specifications for non-rectangular profile, Figs. 8.5-14 to 8.5-17

Angle profile acc. to Fig. 8.5-16 (yes=1, no=2)			
Radial height of the web plate between flanges	d	46	mm
Web thickness of the stiffener	ew	8	mm
Thickness of the profile flange	ef		mm
Leg length of the profile flange	wf		mm
Shell outside radius to the web plate	ri		mm
Coefficient for non-rectangular profile	C		

Instability stress for sideways tripping σ_i **765.3** MPa

Condition: $\sigma_i > \sigma_{eL}$ is **valid**

For stiffeners with flange at the remote edge of the web plate:

Limit dimensions according to (8.5.3-63/64)



Example EN 14025 Appendix B

9 EN08 B.6.5

Shells under external pressure DIN EN 13445-3/7 edition 2003-11 (state Nov.2005)

Spherical shells acc. to 8.7.1

Regulation (0=EN13445-3, 1=EN14025)	TFZ	1 (0,1)
EN 14025: Tanks for the transport of dangerous goods, section 6.4		
Load case: Operation = 1 / Test = 2	lc	1
Safety factor acc. to section 8.4.4	S	1.1
Calculation temperature	t	100 °C
Calculation pressure	P	0.4 bar
Final wall thickness with allowances	en	3.65 mm
Mean radius of the shell	R	1848 mm

Material properties of the spherical shell:

Material designation	Number	1.4404 (H)
Poisson's ratio	nu	0.3
Austenitic steel (1=yes, 2=no)		1
Wall thinning allowance	δe	0 mm
Corrosion allowance	c	0 mm
Thinning allowance during manufacturing	δm	0 mm
Sum of allowances	Σ(δ)	0 mm
Strength acc. to specification (Re, Rp, Rm)	K	430
Safety factor according to section 8.4.4	S	1.1
Modulus of elasticity	E	194000 MPa
0.2% proof stress at operation temperature	Rp02	166 N/mm ²
0.2% proof stress at room temperature	Rp02p	220 MPa
allowable elastic limit (Rpx, Rpx/1.25)	σe	132.8 N/mm ²
Curvature deviation greater than 30%? 1=yes, 2=no,	Abw	2
based on a max. arc length of measuring range	Bog	
maximum radius of curvature	RKmax	mm

Results:

Calculation thickness without allowances	ea	3.65 mm
Limit pressure for circumferential yield (8.7.1-1)	py	0.5246 MPa
Theoretical instability pressure (8.7.1-2)	pm	0.9157 MPa
Ratio pm/py	pm/py	1.746
Ratio pr/py (Fig. 8.5-5)	pr/py	0.2889
allowable pressure (pr/S)	pzul	0.1378 MPa

Condition: $P = 0.04 < \mathbf{0.1378} = p_{zul}$

The strength condition is **valid**
for load case **Design**



Example EN 14025 Appendix B

$$p_y = 0.5246 = 2 * \sigma_e * e_a / R = 2 * 132.8 * 3.65 / 1848 -$$

$$p_m = 0.9157 = 1.21 * E * e_a^2 / R^2 = 1.21 * 194000 * 3.65^2 / 1848^2$$

$$p_{zul} = \begin{cases} p_r/S = 0.1516 / 1.1, & \text{shape deviation} < 30\% \\ p_r/S * (1.3 * R / R_{Kmax})^2 = 0.1516 / 1.1 * (1.3 * 1848 /)^2 \end{cases}$$

Max. measuring length of curvature deviation acc. to 8.7.2:

$$B_{og} = = 2.4 * \sqrt{(e_a * R_{Kmax})} = 2.4 * \sqrt{(3.65 *)}$$

Vessel ends acc. to section 8.8:

Semi-spherical ends shall be designed acc. to the rules for spheres.

The mean sphere radius for torispherical shells is R=crown outside radius
and for the stress calculation acc. 7.5.2 (inside pressure P<0 with
module EN07) holds N=1.

For semi-ellipsoidal ends the mean sphere radius is $R=D^2/(4h)$.



Example EN 14025 Appendix B

10 EN09 B6.6 Betr.

Openings in spherical and cylindrical shells EN 13445-3/9, A5:2006D + EN14025:2008-08

9. Isolated openings in spherical and cylindrical shells

Regulation (0=EN13445-3, 1=EN14025)	TFZ	0 (0,1)
EN 13445-3: Unfired pressure vessels		
Load case: Operation = 1, test = 2		1
Calculation temperature	t	100 °C
Calculation pressure	P	0.3 MPa

Material properties	shell	nozzle	reinforcement
Material designation	1.4404 (H)	1.4404 (H)	
Thickness allowance δ_e	0 mm	0 mm	mm
Corrosion allowance c	0 mm	0 mm	mm
Manufacturing allowance	0 mm	0 mm	0 mm
Total allowance $\Sigma(\delta)$	0 mm	0 mm	mm
Strength Re, Rp, Rm K	430 MPa	430 MPa	MPa
Safety factor S	3	3	
Allowable stress f	143.3 MPa	143.3 MPa	MPa
Weld factor 9.5.2.3	1	1	
Remark:			

Geometry of shell:

Shell: Cylinder=1; sphere+semi-sphere+torispherical=2;			
elliptical=3; cone=4			
Final wall thickness acc. drawing	ens	3.64	mm
Calculation thickness without allowance	eas	3.64	mm
Reduced calculation thickness without all.*)	ecs	3.64	mm
Outside diameter of shell	De	2307	mm
Actual length	ls	500	mm
Conical shell: semi-apex angle	α	0	°
Elliptical shell: internal height	h		mm

*)eas=ecs must satisfy the strength condition: ecs>required thickness

Geometry of opening:

Type of nozzle: without=1, set-in=2, set-on=3			
reinforcement ring=4, extruded=5			
Orientation of sectional cut: axial=1 (cylinder)			
lateral=2 (sphere, cylinder)			
Outside diameter of opening, nozzle or ring	d,deb	510	mm
Final nozzle (ring) thickness acc. drawing	ev,lr	5	mm
Application acc. 9.4.6: Fatigue=1,			
creep range=2, other=3			
External cover (1=yes, 2=no)			
Length of nozzle extension outside the shell	lb1	150	mm
Length of nozzle extension inside the shell	lbi	50	mm
Insertion length for partial penetration	lbp	3.64	mm
Reinforcement ring: Final axial length	ear	0	mm
Reinforcement ring: Radial thickness	lr		mm
Oblique nozzle: axial inclination angle	φ_a	0	°
Oblique nozzle: circumferent. inclination angle	φ_u	0	°

Geometry of reinforcement:

Final thickness	enp	0	mm
Width of reinforcement	lp	0	mm



Example EN 14025 Appendix B

Results

Allowable pressure ($P = 0.3 \leq P_{max}$)	P_{max}	0.4367	MPa
Pressure loaded area of shell	A_{ps}	398557	mm ²
Pressure loaded area of nozzle	A_{pb}	38410	mm ²
Additional pressure loaded area of oblique nozzle	$A_{p\phi}$	0	mm ²
Cross-sectional area of fillet weld	A_{fw}	0	mm ²

Shell:

Inside radius	r_{is}	1150	mm
Maximum supporting length	l_{so}	91.57	mm
Supporting length of cylindrical connection	l_{cyl}	91.57	mm
Effective supporting length Min(l_s, l_{so})	$l's$	91.57	mm
Cross-sectional area of shell	A_{fs}	333.3	mm ²

Nozzle:

Accountable wall thickness acc. 9.4.6	e_b	5	mm
Calculation thickness without allowances	e_{ab}	5	mm
Max. supporting length of nozzle outside shell	l_{bo}	50.25	mm
Max. supporting length of nozzle inside shell	$l_{bo}/2$	25.12	mm
Reduced nozzle length outside shell Min(l_b, l_{bo})	$l'b$	150	mm
Red. nozzle length inside shell Min($l_{bi}, l_{bo}/2$)	$l'bi$	46.36	mm
Reduced stress, Min(f_s, f_b)	f_{ob}	143.3	MPa
Cross-sectional area of nozzle	A_{fb}	1000	mm ²
Adapted penetration length Min(l_{bp}, e_{as})	$e's$	3.64	mm
Inside diameter of nozzle	d_{ib}	500	mm
Outside radius of nozzle or opening	a	255	mm
Mean thickness of ring reinforcem. (iterative)	e_{ams}	0	mm
Accountable reinforcement width of ring	l_0		mm
Accountable axial ring length (9.5-44)	e_r		mm

Reinforcement:

Final thickness without allowances	e_{ap}	0	mm
Reduced thickness Min. (e_{ap}, e_{as})	e_p	0	mm
Reduced width Min. (l_p, l_{so})	$l'p$	0	mm
Cross-sectional area	A_{fp}	0	mm ²
Reduced allowable stress Min(f_s, f_p)	f_{op}	0	MPa

Geometrical condition for openings:

9.4.5 Shell without nozzle:	$d/(2*r_{is}) =$	0.2174	\leq	0.5
9.4.5 Spheres+domed heads:	$d/D_e, d_{ib}/D_e$ or $d_{ir}/D_e =$	0.2174	\leq	0.6
9.4.5 Cylinder with nozzle:	$d/(2*r_{is}) =$	0.2174	\leq	1
9.4.6 Reduced nozzle thickness:*)	$(e_v - c)/e_{as} =$	1.374	\leq	2
9.4.6 Max. nozzle thickness (fatigue):	$e_{ab}/e_{as} =$	1.374	\leq	3

*) Only for creep range or fatigue

Load case	Operation
Application case	Fatigue
Shell type	Cylinder
Type of nozzle	Set-in nozzle
Orientation of section	longitudinal
The strength condition is	valid
The geometrical conditions are	valid



Example EN 14025 Appendix B

— Additional results for openings close to discontinuities acc. 9.7 —

9.7.2.1 Permissible distance w_{min} between opening in cylindrical shell and:

- | | | |
|--|-----------|-----------------|
| a) dished/flat end, reducer, flange | w_{min} | 18.31 mm |
| b) small conical end, convex shell, branch | w_{min} | 91.57 mm |
| c) expansion joint | w_{min} | 45.79 mm |

9.7.2.2 Permissible distance w_{min} between opening in conical shell and:

- | | | |
|-----------------------------------|-----------|-----------------|
| a) Cylindrical shell at wide end | w_{min} | 18.31 mm |
| b) Cylindrical shell at small end | w_{min} | 91.57 mm |

with a connection diameter $D_c =$ **2304** mm

and a thickness e_1 (or e_2 acc. Fig. 9.7-10) **3.64** mm

9.7.2.3 Permissible distance w_{min} between opening in domed and bolted head and a flange

w_{min} **18.31** mm

9.7.2.4 Permissible distance w_{min} between opening in elliptical and torispherical heads and the knuckle ($D_e/10$ acc. Fig.9.5-4) $w_{min} = 0$

Available distance (please specify a value) w mm

9.5.2.2:

Reinforcement dispensable if: $d =$ **510** \leq **13.74** $= 0.15 * l_{so}$ and $w > w_{min}$

9.7.3 Reduced distance from discontinuity:

For $w =$ $< w_p$, the available support length of the reinforcement for the calculation $l_s =$ 500 of the shell must be reduced.

The minimum distance w_p of the opening from discontinuities without influence on l_s and the reduced available support length $Max(l_s)$

acc. 9.7.3 amount to:

	w_p	$Max(l_s)$
a) acc. to 9.7.2.1a), 9.7.2.2a), 9.7.2.3, 9.7.2.4	91.57	
b) acc. to 9.7.2.1b)	183.1	
b) acc. to 9.7.2.1c)	137.4	
c) acc. to 9.7.2.2b)	183.1	

— Equations —

$$l_{so} = \sqrt{(2 * r_{is} + e_{cs}) * e_{cs}} = \sqrt{(2 * 1150 + 3.64) * 3.64}$$

$$l_{so} = \mathbf{91.57} \text{ mm}$$

Cross-sectional area A_{fb} of the nozzle for type = 2: (Type)

$$A_{fb} = e_b * (l'_b + l'_{bi} + e'_s) = 5 * (150 + 46.36 + 3.64) \quad (2)$$

$$= e_b * l'_b = 5 * 150 \quad (3)$$

$$= e_r * l_r = * \quad (4)$$

$$= e_b * l'_b = 5 * 150 \text{ (Approximation)} \quad (5)$$

Cross-sectional area A_{fs} of the shell:

$$A_{fs} = e_{cs} * l'_s = 3.64 * 91.57 \quad (1, 2, 4, 5)$$

$$= e_{cs} * (e_b + l'_s) = 3.64 * (5 + 91.57) \quad (3)$$

$$A_{fs} = \mathbf{333.3}, \quad A_{fb} = \mathbf{1000}, \quad A_{fw} = 0$$

$$A_{fp} = 0 = e_p * l'_p = 0 * 0$$



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Pressure loaded area of nozzle

$$\begin{aligned}
 A_{pb} &= 0.5 * d_i * (l'b + e_{as}) = \mathbf{38410} \text{ mm}^2 \\
 &= 0.5 * \mathbf{510} * (150 + \mathbf{3.64}) \\
 &= 0 \text{ for Typ 1 and 4}
 \end{aligned}$$

Additional pressure loaded area for oblique direction **longitudinal**

$$\begin{aligned}
 A_{p\varphi} &= (d_{ib}^2 * \tan(\varphi)) / 2 = \mathbf{0} \\
 &= (\mathbf{500}^2 * \tan(0)) / 2 \quad \text{(axial)} \\
 &= (\mathbf{500}^2 * \tan(0)) / 2 \quad \text{(lateral)}
 \end{aligned}$$

$$P \leq \frac{(A_{fs} + A_{fw}) * (f_s - 0.5 * P) + A_{fp} * (f_{op} - 0.5 * P) + A_{fb} * (f_{ob} - 0.5 * P)}{(A_{ps} + A_{pb})}$$

$$P \leq \frac{\left[\begin{aligned} &(\mathbf{333.3} + 0) * (\mathbf{143.3} - 0.5 * \mathbf{0.3}) + \\ &+ \mathbf{0} * (\mathbf{0} - 0.5 * \mathbf{0.3}) + \mathbf{1000} * (\mathbf{143.3} - 0.5 * \mathbf{0.3}) \end{aligned} \right]}{(\mathbf{398557} + \mathbf{38410})}$$

The strength condition is **valid** : Efficiency = $\mathbf{0.687} \leq 1$

— Maximum permissible pressure —

$$P_{max} = \mathbf{0.4367} = \frac{(A_{fs} + A_{fw}) * f_s + A_{fb} * f_{ob} + A_{fp} * f_{op}}{(A_{ps} + A_{pb}) + 0.5 * (A_{fs} + A_{fw} + A_{fb} + A_{fp})}$$

$$P_{max} = \frac{(\mathbf{333.3} + 0) * \mathbf{143.3} + \mathbf{1000} * \mathbf{143.3} + \mathbf{0} * \mathbf{0}}{(\mathbf{398557} + \mathbf{38410}) + 0.5 * (\mathbf{333.3} + 0 + \mathbf{1000} + 0)}$$



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11 EN09 B6.6 Prf.

Openings in spherical and cylindrical shells EN 13445-3/9, A5:2006D + EN14025:2008-08

9. Isolated openings in spherical and cylindrical shells

Regulation (0=EN13445-3, 1=EN14025)	TFZ	0 (0,1)
EN 13445-3: Unfired pressure vessels		
Load case: Operation = 1, test = 2		2
Calculation temperature	t	100 °C
Calculation pressure	P	0.4 MPa

Material properties	shell	nozzle	reinforcement
Material designation	1.4404 (H)	1.4404 (H)	
Thickness allowance δ_e	0 mm	0 mm	mm
Corrosion allowance c	0 mm	0 mm	mm
Manufacturing allowance	0 mm	0 mm	0 mm
Total allowance $\Sigma(\delta)$	0 mm	0 mm	mm
Tensile stress 20°C Rm	530 MPa	530 MPa	MPa
Proof stress 20°C Rp	260 MPa	260 MPa	MPa
f=Min[Rm20/2, Rpe*3/4]=	195 MPa	195 MPa	MPa
Weld factor 9.5.2.3	1	1	
Remark:			

Geometry of shell:

Shell: Cylinder=1; sphere+semi-sphere+torispherical=2;			
elliptical=3; cone=4			
Final wall thickness acc. drawing	ens	3.64	mm
Calculation thickness without allowance	eas	3.64	mm
Reduced calculation thickness without all.*)	ecs	3.64	mm
Outside diameter of shell	De	2307	mm
Actual length	ls	500	mm
Conical shell: semi-apex angle	α	0	°
Elliptical shell: internal height	h		mm

*)eas=ecs must satisfy the strength condition: ecs>required thickness

Geometry of opening:

Type of nozzle: without=1, set-in=2, set-on=3			
reinforcement ring=4, extruded=5			
Orientation of sectional cut: axial=1 (cylinder)			
lateral=2 (sphere, cylinder)			
Outside diameter of opening, nozzle or ring	d, deb	510	mm
Final nozzle (ring) thickness acc. drawing	ev, lr	5	mm
Application acc. 9.4.6: Fatigue=1,			
creep range=2, other=3			
External cover (1=yes, 2=no)		1	(1,2)
Length of nozzle extension outside the shell	lb1	150	mm
Length of nozzle extension inside the shell	lbi	50	mm
Insertion length for partial penetration	lbp	3.64	mm
Reinforcement ring: Final axial length	ear	0	mm
Reinforcement ring: Radial thickness	lr		mm
Oblique nozzle: axial inclination angle	φ_a	0	°
Oblique nozzle: circumferent. inclination angle	φ_u	0	°

Geometry of reinforcement:

Final thickness	enp	0	mm
Width of reinforcement	lp	0	mm



Example EN 14025 Appendix B

Results

Allowable pressure ($P = 0.4 \leq P_{max}$)	P_{max}	0.5941	MPa
Pressure loaded area of shell	A_{ps}	398557	mm ²
Pressure loaded area of nozzle	A_{pb}	38410	mm ²
Additional pressure loaded area of oblique nozzle	$A_{p\phi}$	0	mm ²
Cross-sectional area of fillet weld	A_{fw}	0	mm ²

Shell:

Inside radius	r_{is}	1150	mm
Maximum supporting length	l_{so}	91.57	mm
Supporting length of cylindrical connection	l_{cyl}	91.57	mm
Effective supporting length $\text{Min}(l_s, l_{so})$	$l's$	91.57	mm
Cross-sectional area of shell	A_{fs}	333.3	mm ²

Nozzle:

Accountable wall thickness acc. 9.4.6	e_b	5	mm
Calculation thickness without allowances	e_{ab}	5	mm
Max. supporting length of nozzle outside shell	l_{bo}	50.25	mm
Max. supporting length of nozzle inside shell	$l_{bo}/2$	25.12	mm
Reduced nozzle length outside shell $\text{Min}(l_b, l_{bo})$	$l'b$	150	mm
Red. nozzle length inside shell $\text{Min}(l_{bi}, l_{bo}/2)$	$l'bi$	46.36	mm
Reduced stress, $\text{Min}(f_s, f_b)$	f_{ob}	195	MPa
Cross-sectional area of nozzle	A_{fb}	1000	mm ²
Adapted penetration length $\text{Min}(l_{bp}, e_{as})$	$e's$	3.64	mm
Inside diameter of nozzle	d_{ib}	500	mm
Outside radius of nozzle or opening	a	255	mm
Mean thickness of ring reinforcem. (iterative)	e_{ams}	0	mm
Accountable reinforcement width of ring	l_0		mm
Accountable axial ring length (9.5-44)	e_r		mm

Reinforcement:

Final thickness without allowances	e_{ap}	0	mm
Reduced thickness $\text{Min.}(e_{ap}, e_{as})$	e_p	0	mm
Reduced width $\text{Min.}(l_p, l_{so})$	$l'p$	0	mm
Cross-sectional area	A_{fp}	0	mm ²
Reduced allowable stress $\text{Min}(f_s, f_p)$	f_{op}	0	MPa

Geometrical condition for openings:

9.4.5 Shell without nozzle:	$d/(2 \cdot r_{is}) =$	0.2174	\leq	0.5
9.4.5 Spheres+domed heads:	$d/D_e, d_{ib}/D_e$ or $d_{ir}/D_e =$	0.2174	\leq	0.6
9.4.5 Cylinder with nozzle:	$d/(2 \cdot r_{is}) =$	0.2174	\leq	1
9.4.6 Reduced nozzle thickness:*)	$(e_v - c)/e_{as} =$	1.374	\leq	2
9.4.6 Max. nozzle thickness (fatigue):	$e_{ab}/e_{as} =$	1.374	\leq	3

*) Only for creep range or fatigue

Load case	Test
Application case	Fatigue
Shell type	Cylinder
Type of nozzle	Set-in nozzle
Orientation of section	longitudinal
The strength condition is	valid
The geometrical conditions are	valid



Example EN 14025 Appendix B

— Additional results for openings close to discontinuities acc. 9.7 —

9.7.2.1 Permissible distance w_{min} between opening in cylindrical shell and:

- | | | |
|--|-----------|-----------------|
| a) dished/flat end, reducer, flange | w_{min} | 18.31 mm |
| b) small conical end, convex shell, branch | w_{min} | 91.57 mm |
| c) expansion joint | w_{min} | 45.79 mm |

9.7.2.2 Permissible distance w_{min} between opening in conical shell and:

- | | | |
|-----------------------------------|-----------|-----------------|
| a) Cylindrical shell at wide end | w_{min} | 18.31 mm |
| b) Cylindrical shell at small end | w_{min} | 91.57 mm |

with a connection diameter $D_c =$ **2304** mm

and a thickness e_1 (or e_2 acc. Fig. 9.7-10) **3.64** mm

9.7.2.3 Permissible distance w_{min} between opening in domed and bolted head and a flange

w_{min} **18.31** mm

9.7.2.4 Permissible distance w_{min} between opening in elliptical and torispherical heads and the knuckle ($D_e/10$ acc. Fig.9.5-4) $w_{min} = 0$

Available distance (please specify a value) w mm

9.5.2.2:

Reinforcement dispensable if: $d =$ **510** \leq **13.74** $= 0.15 * l_{so}$ and $w > w_{min}$

9.7.3 Reduced distance from discontinuity:

For $w =$ $< w_p$, the available support length of the reinforcement for the calculation $l_s =$ 500 of the shell must be reduced.

The minimum distance w_p of the opening from discontinuities without influence on l_s and the reduced available support length $Max(l_s)$

acc. 9.7.3 amount to:

	w_p	$Max(l_s)$
a) acc. to 9.7.2.1a), 9.7.2.2a), 9.7.2.3, 9.7.2.4	91.57	
b) acc. to 9.7.2.1b)	183.1	
b) acc. to 9.7.2.1c)	137.4	
c) acc. to 9.7.2.2b)	183.1	

— Equations —

$$l_{so} = \sqrt{(2 * r_{is} + e_{cs}) * e_{cs}} = \sqrt{(2 * 1150 + 3.64) * 3.64}$$

$$l_{so} = \mathbf{91.57} \text{ mm}$$

Cross-sectional area A_{fb} of the nozzle for type = 2: (Type)

$$A_{fb} = e_b * (l'_b + l'_{bi} + e'_s) = \mathbf{5} * (150 + 46.36 + \mathbf{3.64}) \quad (2)$$

$$= e_b * l'_b = \mathbf{5} * 150 \quad (3)$$

$$= e_r * l_r = * \quad (4)$$

$$= e_b * l'_b = \mathbf{5} * 150 \text{ (Approximation)} \quad (5)$$

Cross-sectional area A_{fs} of the shell:

$$A_{fs} = e_{cs} * l'_s = \mathbf{3.64} * \mathbf{91.57} \quad (1, 2, 4, 5)$$

$$= e_{cs} * (e_b + l'_s) = \mathbf{3.64} * (5 + \mathbf{91.57}) \quad (3)$$

$$A_{fs} = \mathbf{333.3}, \quad A_{fb} = \mathbf{1000}, \quad A_{fw} = 0$$

$$A_{fp} = 0 = e_p * l'_p = 0 * 0$$



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Pressure loaded area of nozzle

$$\begin{aligned}
 A_{pb} &= 0.5 * d_i * (l'b + eas) = \mathbf{38410} \text{ mm}^2 \\
 &= 0.5 * \mathbf{510} * (150 + \mathbf{3.64}) \\
 &= 0 \text{ for Typ 1 and 4}
 \end{aligned}$$

Additional pressure loaded area for oblique direction **longitudinal**

$$\begin{aligned}
 A_{p\varphi} &= (d_{ib}^2 * \tan(\varphi)) / 2 = \mathbf{0} \\
 &= (\mathbf{500}^2 * \tan(0)) / 2 \quad \text{(axial)} \\
 &= (\mathbf{500}^2 * \tan(0)) / 2 \quad \text{(lateral)}
 \end{aligned}$$

$$P \leq \frac{(A_{fs} + A_{fw}) * (f_s - 0.5 * P) + A_{fp} * (f_{op} - 0.5 * P) + A_{fb} * (f_{ob} - 0.5 * P)}{(A_{ps} + A_{pb})}$$

$$P \leq \frac{\left[\begin{aligned} & (\mathbf{333.3} + 0) * (\mathbf{195} - 0.5 * 0.4) + \\ & + 0 * (\mathbf{0} - 0.5 * 0.4) + \mathbf{1000} * (\mathbf{195} - 0.5 * 0.4) \end{aligned} \right]}{(\mathbf{398557} + \mathbf{38410})}$$

The strength condition is **valid** : Efficiency = $\mathbf{0.6733} \leq 1$

— Maximum permissible pressure —

$$P_{max} = \mathbf{0.5941} = \frac{(A_{fs} + A_{fw}) * f_s + A_{fb} * f_{ob} + A_{fp} * f_{op}}{(A_{ps} + A_{pb}) + 0.5 * (A_{fs} + A_{fw} + A_{fb} + A_{fp})}$$

$$P_{max} = \frac{(\mathbf{333.3} + 0) * \mathbf{195} + \mathbf{1000} * \mathbf{195} + 0 * 0}{(\mathbf{398557} + \mathbf{38410}) + 0.5 * (\mathbf{333.3} + 0 + \mathbf{1000} + 0)}$$



Example EN 14025 Appendix B

12 Zusammenfassung

Documentation

Beispiel EN 14025 Anhang B, Vergleich Lauterbach-Berechnung

Berechnung nach EN14025 Abschnitt	Lauterbach Verfahrenstechnik
B.5.1: Prüfdruck Zylinder erforderliche Dicke $e = 2.95$ mm	$e = 2.953$ mm
B.5.2: Prüfdruck Korbbogen erforderliche Dicke $e = 3.6$ mm $e_y = 2.91$ (mit $e_v=5$ mm)	$e = 3.602$ mm $e_y = 2.988$ (iterativ mit e_y)
B.5.3: Prüfdruck Trennwand $e=8$ mm zulässiger Prüfdruck $P = 0.6$ MPa	$P_{zul} = 0.5994$ MPa
B.6.1: Betriebsdruck 3bar Zylinder erforderliche Dicke $e = 3.02$ mm	$e = 3.013$ mm
B.6.2: Betriebsdruck 3bar Korbbogen erforderliche Dicke $e_{min} = 3.65$ mm	$e_{min} = 3.654$ mm
B.6.3: Betriebsdruck 3bar Trennwand $e=8$ mm zulässiger Betriebsdruck $p = 0.383$ MPa	$P_{zul} = 0.3827$ MPa
B.6.4.2 wie B.6.4.3	
B.6.4.3: Versagen zwischen Versteifungen zulässiger Außendruck $p=0.04$ MPa	$p_{zul} = 0.04002$ MPa
B.6.4.4: Versteifter Zylinder zulässiger Außendruck $p=0.190$ MPa Spannung in der Versteifung $SigS = 129.1$ Seitliche Auslenkung: $Sigi/4 = 766.72/4 > 0.04*132.8/0.932$	$p_{zul} = 0.1899$ MPa (8.5.3-31) $SigS = 128.1$ $765.3/4 > 0.04*132.8/0.9335$
B.6.5: Außendruck 0.4 bar Kugelschale $p_{zul} = 0.138$ MPa	$p_{zul} = 0.1378$ MPa
B.6.6 Betriebsdruck Ausschnitt $131\ 200 / 190\ 663 = 0.688 < 1$	Auslastung = $0.687 < 1$
B.6.6: Prüfdruck Ausschnitt $174\ 933 / 259\ 995 = 0.673 < 1$	Auslastung = $0.6733 < 1$



Example EN 14025 Appendix B

13 Summary

Documentation

Example EN 14025 Appendix B, Comparison with Calculation acc. EN14025 section	calculation by Lauterbach Lauterbach Verfahrenstechnik
B.5.1: Testing pressure cylinder required thickness $e = 2.95$ mm	$e = 2.953$ mm
B.5.2: Test pressure Korbbogen required thickness $e = 3.6$ mm $e_y = 2.91$ (with $e_v=5$ mm)	$e = 3.602$ mm $e_y = 2.988$ (iter. with e_y)
B.5.3: Test pressure baffle $e=8$ mm allowable pressure $P = 0.6$ MPa	$P_{zul} = 0.5994$ MPa
B.6.1: Operation pressure 3bar Cylinder required thickness $e = 3.02$ mm	$e = 3.013$ mm
B.6.2: Operation pressure 3bar Korbbogen required thickness $e_{min} = 3.65$ mm	$e_{min} = 3.654$ mm
B.6.3: Operation pressure 3bar baffle $e=8$ mm allowable pressure $p = 0.383$ MPa	$P_{zul} = 0.3827$ MPa
B.6.4.2 as B.6.4.3	
B.6.4.3: Failure between stiffeners allowable external pressure $p=0.04$ MPa	$p_{zul} = 0.04002$ MPa
B.6.4.4: Stiffened cylinder allowable external pressure $p=0.190$ MPa Stress in stiffener $\text{SigS} = 129.1$	$p_{zul} = 0.1899$ MPa (8.5.3-31) $\text{SigS} = 128.1$
Sideways tripping: $\text{Sigi}/4 = 766.72/4 > 0.04 \cdot 132.8/0.932$	$765.3/4 > 0.04 \cdot 132.8/0.9335$
B.6.5: External pressure 0.4bar sphere $p_{zul} = 0.138$ MPa	$p_{zul} = 0.1378$ MPa
B.6.6 Operation pressure opening $131\ 200 / 190\ 663 = 0.688 < 1$	efficiency = $0.687 < 1$
B.6.6: Test pressure opening $174\ 933 / 259\ 995 = 0.673 < 1$	efficiency = $0.6733 < 1$