Structural analysis

for a cylindrical flat bottomed tank

made from polyethylene PE 100

Project title:	Test Project qwertz asdfgh yxcvbn miscellaneous information
Medium:	Hydrochloric acid, 37 %
Installation:	inside
Order number:	123
Tank number:	007
Customer:	Client ABC 123 XXXX
Operating company:	ABC
Installation location:	0000 town, Great Britain

These structural analysis was prepared by:

April 25, 2016

Date Signature

This structural analysis was prepared by using Software PROFITank, Release 6.3

Contents

1 Preliminary note

1.1 Scope

This structural analysis is based on guideline DVS 2205-2, issued by the Deutscher Verband für Schweißtechnik (German Association for Welding and Allied Processes).

The abbreviations used in this structural analysis correspond with the Guideline DVS 2205-2. In order to avoid recurrent calculation processes and to increase clarity, assumed values ϕ are described in section 6 (proofs). These assumed values are not contained in the guideline DVS 2205-2.

Contrary to the information provided in DVS 2205-2, in this structural analysis the lowest cylinder tier is not referenced with the index 'F', but the index '2', as in this instance there are 2 cylinder tiers.

It is assumed that the tank is manufactured according to standard technical regulations, transported with due care and properly installed. Of particular importance is an even surface under the tank, the bottom of which must be completely flush with the foundation. The structural analysis of the foundation is not part of this calculation.

Underlying drawings:

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1.2 Design, operation

PE 100 will be used as material for the tank. An adequate resistance of the tank material to the storage liquid is expected for the intended design life and operational temperature.

The estimated service life of the storage tank is 25 Years.

The tank is ventilated. As a result there can be no inner build up of long-term overpressure or underpressure.

The tank roof will be in the form of a cone. The connection between roof and cylinder corresponds to DVS 2205-2, figure 13.

The cylinder shell is constructed of tiered wall thickness out of a wound cylinder or an extruded pipe.

The connection between cylinder and bottom corresponds to DVS 2205-2, figure 12.

The tank must have a residual filling height of $h_{RF} \geq 24$ mm (see section 6.3).

1.3 Installation, anchorage

The tank is to be installed in an enclosed building. Pressure from wind and snow will therefore not be assessed. Angaben zur Verankerung sind dem Abschnitt 6.4 zu entnehmen.

1.4 Storage liquid

The tank is to be used to store Hydrochloric acid, 37 %.

Density $\rho_F = 1.500$ g/cm^3
Specific weight $\gamma_F = 14.71$ kN/m^3

Filling volume $V_F = 12.01$ m^3
Filling height $h_F = 2\,654$ mm

Weight of the storage liquid: $G_F = V_F \times \gamma_F = 12.01 \times 14.71 = 176.6$ kN

1.5 Temperatures

1.5.1 Operating and ambient temperature

The temperature of the storage liquid is:
long-term: $T_M = 20\,°\mathrm{C}$ (= middle temperature)
short-term: $T_{MK} = 30\,°\mathrm{C}$

The average Temperature T_M is the temperature, which causes the same damage of the tank material, like changing temperatures in real operation.

The environment temperature for indoor installation is:
long-term: $T_A = 30\,°\mathrm{C}$
short-term: $T_{AK} = 40\,°\mathrm{C}$

1.5.2 Design Temperatures

The temperatures which can be set for the stress and stability proofs are computed according to DVS 2205-2.

a) Conical Roof

long-term: $T_D = (T_M + T_A) \times 0.5 = (20 + 30) \times 0.5 = 25.0\,°\mathrm{C}$
short-term: $T_{DK} = (T_{MK} + T_{AK}) \times 0.5 = (30 + 40) \times 0.5 = 35.0\,°\mathrm{C}$

b) Cylinder, above the liquid level

long-term: $T_{Z,o} = (T_M + T_A) \times 0.5 = (20 + 30) \times 0.5 = 25.0\,°\mathrm{C}$
short-term: $T_{ZK,o} = (T_{MK} + T_{AK}) \times 0.5 = (30 + 40) \times 0.5 = 35.0\,°\mathrm{C}$

c) Cylinder, below the liquid level

long-term: $T_Z = T_M = 20.0\,°\mathrm{C}$
short-term: $T_{ZK} = T_{MK} = 30.0\,°\mathrm{C}$

1.6 Safety factors

For proof of stress and proof of stability (see section 6) the following partial safety factors apply:

$\gamma_{F1} = 1.35$ for effects from dead load, filling and mountings
$\gamma_{F2} = 1.50$ for effects from overpressure, underpressure, wind and snow
$\gamma_{F3} = 0.90$ for dead weight decreasing stress

$\gamma_I = 1.20$ Importance factor
$\gamma_M = 1.10$ Partial safety factor of the resistance or, alternatively, of the load capacity

1.7 Miscellaneous

miscellaneous information

2 Dimensions

2.1 Tank

2.1.1 Main dimensions

Internal diameter	$d = 2\,400$ mm		
Cylindrical height	$h_Z = 3\,210$ mm	(up to the lower edge of the conical roof)
Height of conical roof	$h_D = 322$ mm	
Total height	$h = 3\,532$ mm	

Roof slope $\alpha_D = 15°$
$\kappa = 75°$

2.1.2 Wall thickness

Table 1: Wall thickness and height of tiers

Component		Wall thickness	Height of tier
Roof		$s_D = 15.0$ mm	
Cylinder	Tier 1	$s_{Z,1} = 10.0$ mm	$h_{Z,1} = 2\,710$ mm
	Tier 2	$s_{Z,2} = 15.0$ mm	$h_{Z,2} = 500$ mm
Bottom		$s_B = 15.0$ mm	
			$h_Z = 3\,210$ mm

2.1.3 Volume

Volume of the cylinder	$V_Z = 14.52$ m^3	
Volume of the conical roof	$V_D = 0.48$ m^3	
Total volume	$V_{tot} = 15.01$ m^3	
Maximum usable volume	$V_{95\%} = 13.80$ m^3	$(= 0.95 \times V_Z)$

2.2 Nozzles

2.2.1 Nozzles in the roof

The largest roof nozzle has an external diameter of $d_A = 160$ mm. Additional roof nozzles have no effect on the structural analysis.

The necessary weakening coefficient required for the proof of stress of the conical roof (see section 6.1.2) is according to DVS 2205, equation (34):

$$v_A = \frac{0.75}{1 + \dfrac{d_A}{2 \times \sqrt{(d + s_D) \times s_D}}} = \frac{0.75}{1 + \dfrac{160}{2 \times \sqrt{(2\,400 + 15.0) \times 15.0}}} = 0.528 \tag{1}$$

For nozzles in the roof the following minimum wall thickness applies:

min s_N = SDR 17.6   for liquid-conveying pipe systems
min s_N = SDR 51   for none liquid-conveying pipe systems

2.2.2 Nozzles in the cylinder

Dimensions

It means:

d_A = Outer diameter of the nozzle
s_N = Nozzle wall thickness
s_Z = Cylinder wall thickness near the opening
h_N = For distance of nozzle center from upper edge of tank bottom
a_N = Welding thickness connection cylinder-nozzle, inside and outside
l_1 = Nozzle projection outside
l_2 = Nozzle projection inside

Minimum value for the nozzle projection:

$$l_1 \geq \sqrt{(d_A + s_N) \times s_N}$$
$$l_2 \geq s_N + a_N$$

The dimensions are given in Table 2 on the following page.

The distance from the edge of the opening to a weld must be at least 100 mm.

weakening coefficient

The weakening coefficient v_A is required for the proof of stress of the cylinder (see section .6.2.1).

According to DVS 2205 for nozzles SDR 11 ($s_N \geq d_A/11$) :

$$v_{A\ N,i} = \frac{0.75}{1 + \dfrac{d_A}{2 \times \sqrt{(d + s_Z) \times s_Z}}} \qquad \text{with } d = 2\,400 \text{ mm} \tag{2}$$

Evaluation of the equation (1) is tabulary executed.

Table 2: Nozzles in the cylinder - dimensions and weakening coefficient
All dimensions in millimeters

No.	Designation	d_A	s_N	s_Z	h_N	a_N	l_1	l_2	$v_{A\ N,i}$
N1	D50	63	≥ 5.8	15.0	200	4.1	≥ 20	≥ 10	0.643 *)

*) Calculation for SDR 11

2.3 Overview drawing

The main dimensions of the tank are given in figure 1.

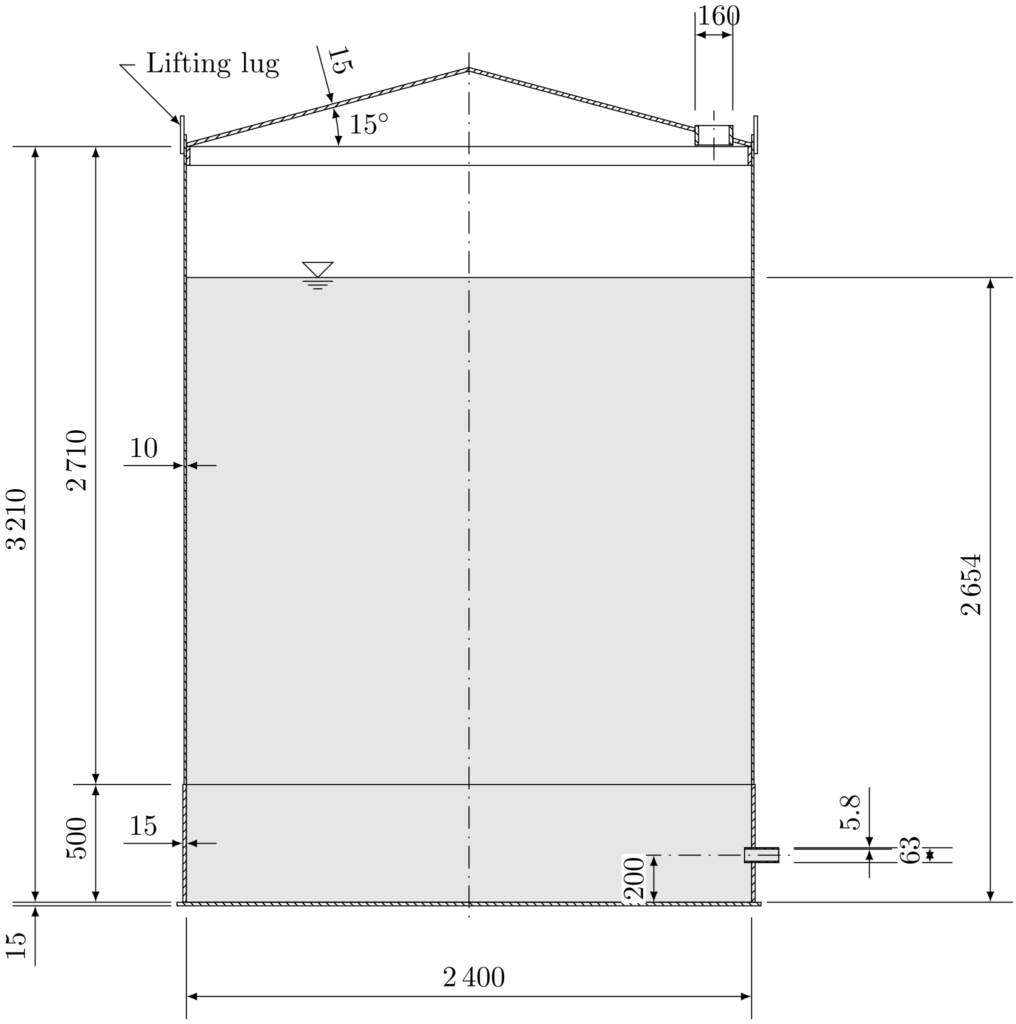

Figure 1: Tank, all dimensions in millimetres

3 Material

3.1 Density

For PE 100 the following material density applies: $\rho = 0.960$ g/cm^3

3.2 Values depending on temperature

The material specific values depending on temperature are supplied in the following table. Moreover, the strength K and the E-Modulus E are dependent on load duration. It means:

K_K^* and E_K	$\Rightarrow$ short-term	= 6 minutes =	0.1 hours	(e.g. wind load)
K_M^*	$\Rightarrow$ medium-term	= 3 month =	2 190 hours	(e.g. snow load)
K_L^* and E_L	$\Rightarrow$ long-term	= 25 years =	219 000 hours	(e.g. dead load)

For the various temperatures and load durations the given creep strengths K_K^*, K_M^* and K_L^* can be derived from Guideline DVS 2205-1, supplement 8

The design value of the resistance is obtained from the value of the characteristic strength divided by γ_M. With $\gamma_M = 1.1$:

$$K_{K,d}^* = \frac{K_K^*}{1.1} \qquad K_{M,d}^* = \frac{K_M^*}{1.1} \qquad K_{L,d}^* = \frac{K_L^*}{1.1} \tag{3}$$

The short-term E-Modulus E_K conforms to guideline DVS 2205-2, table 8.
The long-term E-Modulus E_L at 20 °C conforms to guideline DVS 2205-2, table 9.

The reduction factor A_1 to allow for viscosity conforms to DIN EN 1778 , or alternatively, guideline DVS 2205-1 , table 2 (here referred to as A_4).

Table 3: Values depending on temperature

°C	K_K^*	K_M^*	K_L^*	$K_{K,d}^*$	$K_{M,d}^*$	$K_{L,d}^*$	E_K	E_L	A_1
20.0	14.77	11.43	10.15	13.42	10.39	9.23	800	235	1.00
25.0	13.59	10.52	9.35	12.36	9.56	8.50	675		1.00
30.0	12.55	9.71	8.63	11.41	8.83	7.84	550		1.00
35.0	11.61	8.99	7.99	10.56	8.17	7.26	470		1.00

3.3 Reduction factor A2

The reduction factor for the influence of the storage liquid on the material, at a operating temperature of $T_M = 20\,°C$:

$A_2 = 1.20$ (for proof of stress, in the DIBt-List designated as A_{2B}).
$A_{2I} = 1.00$ (for proof of stability)

3.4 Welding factors

3.4.1 Roof

Welding process for the radial weld in the conical roof: Hot gas extrusion welding

In accordance with DVS 2205-1: Welding factor, long term: $f_S = 0.60$
Welding factor, short term: $f_Z = 0.80$

3.4.2 Cylinder

The cylinder shell is constructed out of a wound cylinder or an extruded pipe.

The 'welding factor' therefore applies: Welding factor, long term: $f_S = 1.00$
Welding factor, short term: $f_Z = 1.00$

4 Loadings

4.1 Dead weight

4.1.1 Weight of the roof

The following equivalent area load is allowed (e.g. for nozzles or similar) in the roof:

$$g_A = 0.00 \text{ kN/m}^2$$

Area load:

$$g_D = \frac{s_D \times \rho \times g \times 10^{-6}}{\sin\kappa} + g_A \times 10^{-3}$$

$$g_D = \frac{15.0 \times 0.960 \times 9.81 \times 10^{-6}}{\sin 75.0} + 0.00 \times 10^{-3} = 0.000\,15 \text{ N/mm}^2 \tag{4}$$

Total load:

$$G_D = g_D \times \frac{\pi \times d^2}{4} = 0.000\,15 \times \frac{\pi \times 2\,400^2}{4} = 661 \text{ N} \tag{5}$$

4.1.2 Weight of the cylinder

It means:

$$G_{Z,i} = \text{Dead weight of tier } i$$

$$G_{Z,i} = \pi \times (d + s_{Z,i}) \times h_{Z,i} \times s_{Z,i} \times \rho \times g \times 10^{-6}$$

$$G_{Z,i} = \pi \times (2\,400 + s_{Z,i}) \times h_{Z,i} \times s_{Z,i} \times 0.96 \times 9.81 \times 10^{-6}$$

$$\Sigma G_{Z,i} = \text{Dead weight of cylinder down to lower edge of cylinder tier } i$$

$$\Sigma G_{Z,i} + G_D = \text{Dead weight of tank down to lower edge of cylinder tier } i$$

Table 4: Weight of the cylinder

tier i	Wall thickn. $s_{Z,i}$	Height tier $h_{Z,i}$	$G_{Z,i}$	$\Sigma G_{Z,i}$	$\Sigma G_{Z,i} + G_D$
1	10.0 mm	2 710 mm	1 932 N	1 932 N	2 593 N
2	15.0 mm	500 mm	536 N	2 467 N	3 129 N

The dead load of the cylinder is: $G_Z = 2\,467$ N

4.1.3 Weight of the bottom

For an approximate allowance of the bottom diameter, the dead weight calculated with the cylinder interior diameter is increased by 5 % (factor 1.05).

$$G_B = \frac{\pi \times d^2 \times s_B \times \rho \times g \times 10^{-6}}{4} \times 1.05$$

$$G_B = \frac{\pi \times 2\,400^2 \times 15.0 \times 0.960 \times 9.81 \times 10^{-6}}{4} \times 1.05 = 639 \text{ N} \quad (6)$$

4.1.4 Weight of the tank

$$G_E = G_D + G_Z + G_B = 661 + 2\,467 + 639 = 3\,768 \text{ N} \quad (7)$$

4.2 Loadings from attachments

There are no loads from attachments (e.g. from tank-mounted platforms or agitators).

4.3 Overpressure and underpressure

Long-term overpressure and underpressure cannot occur since the tank is externally ventilated.

When a tank is mounted indoors with external ventilation inside the building, underpressurep_{uS} from wind suction cannot occur.

Table 5: Overpressure and underpressure

	internal overpressure	internal underpressure
long-term	$p_{ü} = 0.000\,00$ N/mm^2	$p_u = 0.000\,00$ N/mm^2
short-term	$p_{üK} = 0.000\,50$ N/mm^2 (5.0 mbar)	$p_{uK} = 0.000\,30$ N/mm^2 (3.0 mbar)
		$p_{uS} = 0.000\,00$ N/mm^2

4.4 Snow load

For indoor installation: $p_S = 0.0$ N/mm^2

4.5 Wind load

For indoor installation: $p_{eu} = 0.0$ N/mm^2

5 Axial stresses in the cylinder

For proof of the axial stability of the cylinder (see section 6.2.4), the axial compressive stresses at the lower edge of each cylinder tier are required.

5.1 Axial stresses from the individual load cases

from dead weight: $\sigma_{G,i} = \dfrac{G_D + \Sigma G_{Z,i}}{\pi \times d \times s_{Z,i}} = \dfrac{G_D + \Sigma G_{Z,i}}{\pi \times 2\,400 \times s_{Z,i}}$

from attachment-load: $\sigma_{A,i}$ = not applicable

from underpressure long-term: $\sigma_{pu,i}$ = not applicable

from underpressure short-term: $\sigma_{puK,i} = \dfrac{p_{uK} \times d}{4 \times s_{Z,i}} = \dfrac{0.000\,30 \times 2\,400}{4 \times s_{Z,i}}$

from underpressure caused by wind suction: $\sigma_{puK,i}$ = not applicable

from snow load on the roof: $\sigma_{S,i} = \dfrac{p_S \times d}{4 \times s_{Z,i}} = \dfrac{0.000\,68 \times 2\,400}{4 \times s_{Z,i}}$

from wind load: $\sigma_{W,i}$ = not applicable

from earthquake: $\sigma_{E,i}$ = not applicable

The Analysis of the equations is tabulary executed.

Table 6: Axial stresses from the individual load cases

Tier	$s_{Z,i}$	Axial stresses [N/mm^2]							
i	[mm]	$\sigma_{G,i}$	$\sigma_{A,i}$	$\sigma_{pu,i}$	$\sigma_{puk,i}$	$\sigma_{puS,i}$	$\sigma_{S,i}$	$\sigma_{W,i}$	$\sigma_{E,i}$
1	10.0	0.034	0.000	0.000	0.018	0.000	0.000	0.000	0.000
2	15.0	0.027	0.000	0.000	0.012	0.000	0.000	0.000	0.000

5.2 Axial stresses in the area of nozzles

In the area of nozzles the proof of the axial stability of the cylinder is to perform with the residual cross section values (taking into account the opening but excluding the nozzle).

See the DIBt Berechnungsempfehlung 40-B5.

5.3 Axial stresses from load combinations

5.3.1 Fundamental combinations of actions

LC 1: With underpressure

- LF 1.1: $$\Sigma\sigma_{i,d\,1.1} = \gamma_{F1} \times \sigma_{G,i} + \gamma_{F2} \times \left(\max(\sigma_{puK,i}, \sigma_{puS,i}) + 0.7 \times \sigma_{S,i} + \frac{\sigma_{W,i}}{1.2}\right) \tag{8a}$$

- LF 1.2: $$\Sigma\sigma_{i,d\,1.2} = \gamma_{F1} \times \sigma_{G,i} + \gamma_{F2} \times (\sigma_{puK,i} + \sigma_{S,i}) \tag{8b}$$

LC 2: Without underpressure

- LF 2.1: $$\Sigma\sigma_{i,d\,2.1} = \gamma_{F1} \times \sigma_{G,i} + \gamma_{F2} \times \left(0.7 \times \sigma_{S,i} + \frac{\sigma_{W,i}}{1.2}\right) \tag{9a}$$

- LF 2.2: $$\Sigma\sigma_{i,d\,2.2} = \gamma_{F1} \times \sigma_{G,i} + \gamma_{F2} \times \sigma_{S,i} \tag{9b}$$

5.3.2 Combinations of actions for seismic design situations

LC 3: Earthquake not applicable

5.3.3 Analysis

Table 7: Axial stresses in the cylinder tiers, load combinations

Tier	Axial stresses $\Sigma\sigma_{i,d}$ $\left[\text{N/mm}^2\right]$				
i	LF 1.1	LF 1.2	LF 2.1	LF 2.2	LF 3
1	0.073	0.073	0.046	0.046	-
2	0.055	0.055	0.037	0.037	-

6 Proofs

6.1 Proof of the conical roof

6.1.1 Minimum wall thickness

$$\min s_D = \frac{\delta_D}{1000} \times d = \frac{5.5}{1000} \times 2\,400 = 13.2 \text{ mm} \tag{10}$$

$s_D = 15.0$ mm $\geq \min s_D$ $\Rightarrow$ **Proof supplied**

This proof considers a load of 1 kN/m^2 at 20 °C short-term acting.

6.1.2 Proof of stresses (inward loadings)

Assumed values:

$$\begin{aligned} A &= -0.000\,103 \times \alpha_D^2 + 0.007\,825 \times \alpha_D - 1.777\,1 \\ &= -0.000\,103 \times 15^2 + 0.007\,825 \times 15 - 1.777\,1 = -1.682\,9 \end{aligned} \tag{11}$$

$$B = -0.000\,433 \times \alpha_D^2 + 0.008\,115 \times \alpha_D - 0.187\,0$$
$$= -0.000\,433 \times 15^2 + 0.008\,115 \times 15 - 0.187\,0 = -0.162\,7 \quad (12)$$

$$Exp1 = A \times \ln\left(\frac{2 \times s_D}{d}\right) + B = -1.682\,9 \times \ln\left(\frac{2 \times 15.0}{2\,400}\right) -0.162\,7 = 7.211\,8 \quad (13)$$

$$\phi D_1 = \mathrm{e}^{Exp1} \times A_2 \times \gamma_I = \mathrm{e}^{7.211\,8} \times 1.20 \times 1.20 = 1\,952 \quad (14)$$

$$\phi D_2 = 0.5 \times \frac{d}{s_D} \times \frac{A_2 \times \gamma_I}{\cos\kappa} = 0.5 \times \frac{2\,400}{15.0} \times \frac{1.20 \times 1.20}{\cos 75} = 445 \quad (15)$$

Calculation of the stresses in the following sections applies to the...

...welding seam: According to DVS 2205-2, section 4.1.6.1, equations (27) to (29)
...opening: According to DVS 2205-2, section 4.1.7, equation (35)

Long-term loading

$$p_{DL,d} = \gamma_{F1} \times g_D + \gamma_{F2} \times p_u = 1.35 \times 0.000\,15 + 1.50 \times 0.000\,00 = 0.000\,20 \text{ N/mm}^2 \quad (16)$$

The effective temperature is $T_D = 25.0\,°\text{C} \Rightarrow A_1 = 1.00$

Stresses in the area of the welding seam:

$$K_{L,d} = \frac{p_{DL,d} \times A_1 \times \phi D_1}{f_{sD}} = \frac{0.000\,20 \times 1.00 \times 1\,952}{0.60} = 0.64 \text{ N/mm}^2 \quad (17)$$

Stresses in the area of the opening:

$$K_{L,d} = \frac{p_{DL,d} \times A_1 \times \phi D_2}{v_A} = \frac{0.000\,20 \times 1.00 \times 445}{0.528} = 0.17 \text{ N/mm}^2 \quad (18)$$

Medium-term loading A medium-term strain is not present when the tank is set up indoors.

Short-term loading

$$\begin{aligned} p_{uK} &= 0.000\,30 \text{ N/mm}^2 \\ p_{uS} &= 0.000\,00 \text{ N/mm}^2 \\ p_{DK} &= \max(p_{uK}, p_{uS}) = 0.000\,30 \text{ N/mm}^2 \end{aligned} \quad (19)$$

$$\Sigma p_{DK,d} = \gamma_{F1} \times g_D + \gamma_{F2} \times p_{DK} = 1.35 \times 0.000\,15 + 1.50 \times 0.000\,30 = 0.000\,65 \text{ N/mm}^2 \quad (20)$$

The effective temperature is $T_{DK} = 35.0\,°\mathrm{C} \quad \Rightarrow \quad A_1 = 1.00$

Stresses in the area of the welding seam:

$$\Sigma K_{K,d} = \frac{\Sigma p_{DK,d} \times A_1 \times \phi D_1}{f_{zD}} = \frac{0.000\,65 \times 1.00 \times 1\,952}{0.80} = 1.58\ \mathrm{N/mm^2} \tag{21}$$

Stresses in the area of the opening:

$$\Sigma K_{K,d} = \frac{\Sigma p_{DK,d} \times A_1 \times \phi D_2}{v_A} = \frac{0.000\,65 \times 1.00 \times 445}{0.528} = 0.55\ \mathrm{N/mm^2} \tag{22}$$

Proofs for the conical roof in the area around the welding seam:

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \underbrace{\frac{K_{L,d}}{K^*_{L,d}}}_{25.0\,°\mathrm{C}} + \frac{K_{M,d}}{K^*_{M,d}} = \frac{0.64}{8.50} + 0.00 = 0.076 \quad < 1,0$$

$\Rightarrow$ **Proof supplied**

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \underbrace{\frac{K_{K,d}}{K^*_{K,d}}}_{35.0\,°\mathrm{C}} = \frac{1.58}{10.56} = 0.150 \quad < 1,0$$

$\Rightarrow$ **Proof supplied**

Proofs for the conical roof in the area around the opening:

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \underbrace{\frac{K_{L,d}}{K^*_{L,d}}}_{25.0\,°\mathrm{C}} + \frac{K_{M,d}}{K^*_{M,d}} = \frac{0.17}{8.50} + 0.00 = 0.020 \quad < 1,0$$

$\Rightarrow$ **Proof supplied**

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \underbrace{\frac{K_{K,d}}{K^*_{K,d}}}_{35.0\,°\mathrm{C}} = \frac{0.55}{10.56} = 0.052 \quad < 1,0$$

$\Rightarrow$ **Proof supplied**

6.1.3 Proof of stresses (outward loadings)

Assumed values:

$$\begin{aligned} C &= 1.30 \times 10^{-5} \times \alpha_D^2 - 0.000\,97 \times \alpha_D - 1.405\,4 \\ &= 1.30 \times 10^{-5} \times 15^2 - 0.000\,97 \times 15 - 1.405\,4 = -1.417\,0 \end{aligned} \tag{23}$$

$$\begin{aligned} D &= 0.000\,265 \times \alpha_D^2 - 0.045\,74 \times \alpha_D + 1.562\,2 \\ &= 0.000\,265 \times 15^2 - 0.045\,74 \times 15 + 1.562\,2 = 0.935\,7 \end{aligned} \tag{24}$$

$$Exp2 = C \times \ln\left(\frac{2 \times s_D}{d}\right) + D = -1.4170 \times \ln\left(\frac{2 \times 15.0}{2\,400}\right) + 0.9357 = 7.1452 \quad (25)$$

$$\phi D_2 = 445 \qquad \text{(see section 6.1.1)} \quad (26)$$

$$\phi D_3 = \mathrm{e}^{Exp2} \times A_2 \times \gamma_I = \mathrm{e}^{7.1452} \times 1.20 \times 1.20 = 1\,826 \quad (27)$$

Long-term loading

$$p_{DL,d} = \gamma_{F2} \times p_{ü} - \gamma_{F3} \times g_D = 1.50 \times 0.000\,00 - 0.90 \times 0.000\,15 < 0.0 \text{ N/mm}^2 \quad (28)$$

The effective temperature is $T_D = 25.0\,°\text{C} \Rightarrow A_1 = 1.00$

Stresses in the area of the conical roof:

$$K_{L,d} = p_{DL,d} \times A_1 \times \phi D_3 = 0.000\,00 \times 1.00 \times 1\,826 = 0.00 \text{ N/mm}^2 \quad (29)$$

Stresses in the area of the opening:

$$K_{L,d} = \frac{p_{DL,d} \times A_1 \times \phi D_2}{v_A} = \frac{-0.000\,13 \times 1.00 \times 445}{0.528} = 0.00 \text{ N/mm}^2 \quad (30)$$

Medium-term loading A medium-term strain is not present when the tank is set up indoors.

Short-term loading

$$\Sigma p_{DK,d} = \gamma_{F2} \times p_{ük} - \gamma_{F3} \times g_D = 1.50 \times 0.000\,50 - 0.90 \times 0.000\,15 = 0.000\,62 \text{ N/mm}^2 \quad (31)$$

The effective temperature is $T_{DK} = 35.0\,°\text{C} \Rightarrow A_1 = 1.00$

Stresses in the area of the conical roof:

$$\Sigma K_{K,d} = \Sigma p_{DK,d} \times A_1 \times \phi D_3 = 0.000\,62 \times 1.00 \times 1\,826 = 1.13 \text{ N/mm}^2 \quad (32)$$

Stresses in the area of the opening:

$$\Sigma K_{K,d} = \frac{\Sigma p_{DK,d} \times A_1 \times \phi D_2}{v_A} = \frac{0.000\,62 \times 1.00 \times 445}{0.528} = 0.52 \text{ N/mm}^2 \quad (33)$$

<u>Proofs for the conical roof:</u>

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \frac{K_{L,d}}{K^*_{L,d}} + \frac{K_{M,d}}{K^*_{M,d}} = 0.00 + 0.00 = 0.000 \quad < 1,0 \qquad \Rightarrow \textbf{Proof supplied}$$

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \frac{K_{K,d}}{\underbrace{K^*_{K,d}}_{35.0\,°\text{C}}} = \frac{1.13}{10.56} = 0.107 \quad < 1,0 \qquad \Rightarrow \textbf{Proof supplied}$$

<u>Proofs for the conical roof in the area around the opening:</u>

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \frac{K_{L,d}}{K^*_{L,d}} + \frac{K_{M,d}}{K^*_{M,d}} = 0.00 + 0.00 = 0.000 \quad < 1,0 \qquad \Rightarrow \text{Proof supplied}$$

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \underbrace{\frac{K_{K,d}}{K^*_{K,d}}}_{35.0\,°\mathrm{C}} = \frac{0.52}{10.56} = 0.049 \quad < 1,0 \qquad \Rightarrow \text{Proof supplied}$$

6.1.4 Proof of stability

Assumed values:

$$\phi D_3 = \frac{d}{4 \times \cos\kappa \times s_D} = \frac{2\,400}{4 \times \cos 75 \times 15.0} = 154.5 \tag{34}$$

$$\phi D_4 = \frac{2.68}{\gamma_M} \times \sin\kappa \times \sqrt{\cos\kappa} \times \left(\frac{s_D}{d}\right)^{1.5}$$

$$\phi D_4 = \frac{2.68}{1.10} \times \sin 75 \times \sqrt{\cos 75} \times \left(\frac{15.0}{2\,400}\right)^{1.5} = 0.000\,592 \tag{35}$$

The following applies for indoor installation:

The effective temperature is $T_{DK} = 35.0\,°\mathrm{C}$
Short-term E-Modulus: $E_K = 470\ \mathrm{N/mm^2}$
Loading: $\Sigma p_{DK,d} = 0.000\,65\ \mathrm{N/mm^2}$ see equation (20)
Existing stress: $\Sigma\sigma_d = \Sigma p_{DK,d} \times \phi D_3 = 0.000\,65 \times 154.5 = 0.100\,0\ \mathrm{N/mm^2}$
Critical stress: $\sigma_{k,d} = E_K \times \phi D_4 = 470 \times 0.000\,592 = 0.278\,0\ \mathrm{N/mm^2}$

Proof according to DVS 2205-2, equation (56):

$$\eta = \frac{A_{2I} \times \gamma_I \times \Sigma\sigma_d}{\sigma_{k,d}} = \frac{1.00 \times 1.20 \times 0.100\,0}{0.278\,0} = 0.432 \quad < 1,0 \qquad \Rightarrow \text{Proof supplied}$$

6.2 Proof of the cylindrical shell

6.2.1 Stress in the circumference direction

The proof of stress of the cylinder in the circumference direction is carried out according to DVS 2205-2, section 4.1.3.1. The cylinder shell is constructed out of a wound cylinder or an extruded pipe. A proof of the outer fibre expansion according to DVS 2205-2, table 1, is therefore not necessary. Assumed value:

$$\phi Z_1 = 0.5 \times d \times A_2 \times \gamma_I = 0.5 \times 2\,400 \times 1.20 \times 1.20 = 1\,728\ \mathrm{mm} \tag{36}$$

The existing circumference tension in the cylinder at the bottom edge of a cylinder tier i is:

$$\text{long-term:}\quad K_{L,d,i} = \frac{\gamma_{F1} \times p_{\text{stat},i} + \gamma_{F2} \times p_{\ddot{u}}}{s_{Z,i} \times f_s} \times A_1 \times \phi Z_1$$

$$= \frac{1.35 \times p_{\text{stat},i} + 1.50 \times 0.000\,00}{s_{Z,i} \times 1.0} \times A_1 \times 1\,728 \quad (37)$$

$$\text{short-term:}\quad \Sigma K_{K,d,i} = \frac{\gamma_{F1} \times p_{\text{stat},i} + \gamma_{F2} \times p_{\ddot{u}K}}{s_{Z,i} \times f_z} \times A_1 \times \phi Z_1$$

$$= \frac{1.35 \times p_{\text{stat},i} + 1.50 \times 0.000\,50}{s_{Z,i} \times 1.0} \times A_1 \times 1\,728 \quad (38)$$

The existing circumference tension in the cylinder at the bottom edge of a nozzle is:

$$\text{long-term:}\quad K_{L,d,i} = \frac{\gamma_{F1} \times p_{\text{stat},i} + \gamma_{F2} \times p_{\ddot{u}}}{s_{Z,i} \times v_{A,\text{N}i}} \times A_1 \times \phi Z_1$$

$$= \frac{1.35 \times p_{\text{stat},i} + 1.50 \times 0.000\,00}{s_{Z,i} \times v_{A,\text{N}i}} \times A_1 \times 1\,728 \quad (39)$$

$$\text{short-term:}\quad \Sigma K_{K,d,i} = \frac{\gamma_{F1} \times p_{\text{stat},i} + \gamma_{F2} \times p_{\ddot{u}K}}{s_{Z,i} \times v_{A,\text{N}i}} \times A_1 \times \phi Z_1$$

$$= \frac{1.35 \times p_{\text{stat},i} + 1.50 \times 0.000\,50}{s_{Z,i} \times v_{A,\text{N}i}} \times A_1 \times 1\,728 \quad (40)$$

Strains due to influences of medium effect duration do not arise for this proof.

The overpressure from the storage liquid is determined as follows:

$p_{stat,i} = \rho_F \times g \times h_{F,i} \times 10^{-6} = 1.500 \times 9.81 \times h_{F,i} \times 10^{-6}$

$h_{F,i}$ = Height of the liquid level above the bottom edge of cylinder tier i or alternatively above the bottom edge of the nozzle.

All cylinder tiers bottom edges and nozzle bottom edges are located below the maximum liquid level.

For these cylinder tiers the effective temperature is

long-term: $T_Z = 20.0\,°\text{C} \Rightarrow A_1 = 1.00$

short-term: $T_{ZK} = 30.0\,°\text{C} \Rightarrow A_1 = 1.00$

Evaluation of the equations follows in table form.

tier i	$s_{Z,i}$ mm	Region	$v_{A\,N,i}$	$h_{F,i}$ mm	$p_{stat,i}$ N/mm²	$K_{L,d,i}$ N/mm²	$\Sigma K_{K,d,i}$ N/mm²
1	10.0	Bottom edge	—	2 154	0.031 68	7.39	7.52
2	15.0	Nozzle N1	0.643	2 485	0.036 56	8.84	8.97
		Bottom edge	—	2 654	0.039 04	6.07	6.16

The two following proofs are accomplished with the maximum value according to the table.

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \underbrace{\frac{K_{L,d}}{K^*_{L,d}}}_{20.0\,°\mathrm{C}} + \frac{K_{M,d}}{K^*_{M,d}} = \frac{8.84}{9.23} + 0.00 = 0.957 \quad < 1,0 \qquad \Rightarrow \textbf{Proof supplied}$$

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \underbrace{\frac{K_{K,d}}{K^*_{K,d}}}_{30.0\,°\mathrm{C}} = \frac{8.97}{11.41} = 0.786 \quad < 1,0 \qquad \Rightarrow \textbf{Proof supplied}$$

6.2.2 Proof of stress in the longitudinal direction

Proof of stress of the cylinder in the longitudinal direction according to DVS 2205-2, section 4.1.3.2 only for the transition point between cylinder and bottom.

The factor C according to DVS 2205-2, table 4, is:
$C = 1.20$

The effective temperature is

$$\begin{array}{llcl} \text{long-term:} & T_Z = 20.0\,°\mathrm{C} & \Rightarrow & A_1 = 1.00 \\ \text{short-term:} & T_{ZK} = 30.0\,°\mathrm{C} & \Rightarrow & A_1 = 1.00 \end{array}$$

The existing long-term tensile stress at the lower edge of the cylinder is equal to

$$\begin{aligned} K_{L,d} &= \left(C \times \left(\gamma_{F1} \times p_{stat} + \gamma_{F2} \times p_{ü} \right) \times \frac{d}{2} + \gamma_{F2} \times p_{ü} \times \frac{d}{4} \right. \\ &\quad \left. - \frac{\gamma_{F3} \times (G_D + G_Z)}{\pi \times d} \right) \times \frac{A_1 \times A_2 \times \gamma_I}{s_{ZF}} \\ &= \left(1.20 \times \left(1.35 \times 0.039\,04 + 1.50 \times 0.00 \right) \times \frac{2\,400}{2} + 1.50 \times 0.00 \times \frac{2\,400}{4} \right. \\ &\quad \left. - \frac{0.90 \times (661 + 2\,467)}{\pi \times 2\,400} \right) \times \frac{1.00 \times 1.20 \times 1.20}{15.0} \\ &= \left(75.89 + 0.00 - 0.37 \right) \times 0.096 = 7.25\ \mathrm{N/mm^2} \end{aligned} \tag{41}$$

Strains due to influences of medium effect duration do not arise for this proof.

$$K_{M,d} = 0.00\ \mathrm{N/mm^2} \tag{42}$$

The existing short-term tensile stress at the lower edge of the cylinder is equal to

$$\Sigma K_{K,d} = \left(C \times \left(\gamma_{F1} \times p_{stat} + \gamma_{F2} \times p_{\ddot{u}k}\right) \times \frac{d}{2} + \gamma_{F2} \times p_{\ddot{u}k} \times \frac{d}{4} + \frac{4 \times \gamma_{F2} \times M_W \times 10^3}{\pi \times d^2} - \frac{\gamma_{F3} \times (G_D + G_Z)}{\pi \times d} \right) \times \frac{A_1 \times A_2 \times \gamma_I}{s_{ZF}} \tag{43}$$

$$= \left(1.20 \times \left(1.35 \times 0.039\,04 + 1.50 \times 0.000\,50\right) \times \frac{2\,400}{2} + 1.50 \times 0.000\,50 \times \frac{2\,400}{4} + \frac{4 \times 1.50 \times 0 \times 10^3}{\pi \times 2\,400^2} - \frac{0.90 \times (661 + 2\,467)}{\pi \times 2\,400} \right) \times \frac{1.00 \times 1.20 \times 1.20}{15.0}$$

$$= \left(76.97 + 0.45 + 0.00 - 0.37\right) \times 0.096 = 7.40 \text{ N/mm}^2$$

Proof 1 according to DVS 2205-2, equation (13):

$$\eta_1 = \underbrace{\frac{K_{L,d}}{K^*_{L,d}}}_{20.0\,°\text{C}} + \frac{K_{M,d}}{K^*_{M,d}} = \frac{7.25}{9.23} + 0.00 = 0.785 \quad < 1,0 \qquad \Rightarrow$$ **Proof supplied**

Proof 2 according to DVS 2205-2, equation (15):

$$\eta_2 = \underbrace{\frac{K_{K,d}}{K^*_{K,d}}}_{30.0\,°\text{C}} = \frac{7.40}{11.41} = 0.648 \quad < 1,0 \qquad \Rightarrow$$ **Proof supplied**

For calculating the minimum wall thickness of the bottom plate (see section 6.3.1), the minimum required cylinder wall thickness at the transition point with the bottom plate is needed:

By converting the DVS Guideline equation (22a) and (22b) we obtain:

$$s^*_{ZF,L} = \frac{K_{L,d}}{K^*_{L,d}} \times s_{ZF} = \frac{7.25}{9.23} \times 15.0 = 11.8 \text{ mm}$$

$$s^*_{ZF,K} = \frac{K_{K,d}}{K^*_{K,d}} \times s_{ZF} = \frac{7.40}{11.41} \times 15.0 = 9.7 \text{ mm}$$

$$s^*_{ZF} = \max(s_{ZF,L}, s_{ZF,K}) = 11.8 \text{ mm} \tag{44}$$

6.2.3 Casing compressive stability

The proof of stability in the circumference direction for the cylinder with tired walls thickness is performed acc. DIN 18800-4 on an equivalent cylinder with three tiers.

For the equivalent cylinder with three tiers applies:

	Fictitious tier length	Fictitious wall thickness
above	l_o = 1 605 mm	s_o = 10.0 mm
mean	l_m = 802 mm	s_m = 10.0 mm
bottom	l_u = 802 mm	s_u = 13.1 mm

Ratio values:

$$\frac{l_o}{l} = \frac{1\,605}{3\,210} = 0.50 \tag{45a}$$

$$\frac{s_m}{s_o} = \frac{10.0}{10.0} = 1.00 \tag{45b}$$

$$\frac{s_u}{s_o} = \frac{13.1}{10.0} = 1.31 \tag{45c}$$

The following can be read from diagram 20 of DIN 18800-4:

$$\beta = 0.54 \tag{46}$$

The relevant underpressure can be determined from the following underpressures:

$$\begin{aligned} p_{uK} &= 0.000\,30 \text{ N/mm}^2 \\ p_u + p_{uS} + p_{eu} &= 0.000\,00 + 0.000\,00 + 0.000\,00 = 0.000\,00 \text{ N/mm}^2 \\ p_{u,\max} &= \max(p_{uK},\, p_u + p_{uS} + p_{eu}) = 0.000\,30 \text{ N/mm}^2 \end{aligned} \tag{47}$$

$$\Sigma p_d = \gamma_{F2} \times p_{u,\max} = 1.50 \times 0.000\,30 = 0.000\,45 \text{ N/mm}^2 \tag{48}$$

The critical shell pressure of the equivalent cylinder with three tiers is for $T_{ZK,o} = 35.0\,°C$

$$\begin{aligned} p_{kM,d} &= 0.67 \times \beta \times C^* \times \frac{E_K \times r}{\gamma_M \times l_o} \times \left(\frac{s_o}{r}\right)^{2.5} \\ &= 0.67 \times 0.54 \times 1.0 \times \frac{470 \times 1\,200}{1.10 \times 1\,605} \times \left(\frac{10.0}{1\,200}\right)^{2.5} = 0.000\,73 \text{ N/mm}^2 \end{aligned} \tag{49}$$

$C^* = 1.0$ applies to the tank with roof

Proof according to DVS 2205-2, equation (50):

$$\eta_M = \frac{A_{2I} \times \gamma_I \times \Sigma p_d}{p_{kM,d}} = \frac{1.00 \times 1.20 \times 0.000\,45}{0.000\,73} = 0.739 \quad < 1,0 \;\Rightarrow$$ **Proof supplied**

6.2.4 Axial stability of the shell

The proof of stability in axial direction is performed according to DVS 2205-2 section 4.2.2.1 at the bottom edge of each cylinder tier.

Auxiliary value per DVS 2205-2, equation (48):

$$\alpha_i = \frac{0.7}{\sqrt{\frac{E_K^{20\,°C}}{E_L^{20\,°C}} \times \left(1 + \frac{r}{100 \times s_{Z,i}}\right)}} = \frac{0.7}{\sqrt{\frac{800}{235} \times \left(1 + \frac{1\,200}{100 \times s_{Z,i}}\right)}} \tag{50}$$

The buckling stress is calculated for the temperature $T_{ZK,o} = 35.0\,°\text{C}$

$$\sigma_{k,i,d} = \alpha_i \times 0.62 \times \frac{E_K \times s_{Z,i}}{\gamma_M \times r} = \alpha_i \times 0.62 \times \frac{470 \times s_{Z,i}}{1.10 \times 1\,200} \quad \leq K^*_{K,d} \tag{51}$$

The following condition must be observed for each tier:

$$\eta_{A,i} = \frac{A_{2I} \times \gamma_I \times \Sigma\sigma_{i,d}}{\sigma_{k,i,d}} = \frac{1.00 \times 1.20 \times \sigma_{i,d}}{\sigma_{k,i,d}} \leq 1.0 \tag{52}$$

with $\Sigma\sigma_{i,d}$ from section 5.3.3 , load case 1 (max. value from LC 1.1 and LC 1.2)

i	$s_{Z,i}$ mm	α_i	$\sigma_{k,i,d}$ N/mm²	$\sigma_{i,d}$ N/mm²	Proof according to DVS 2205-2, equation (49)
1	10.0	0.256	0.565	0.073	$\eta_{A,1} =$ 0.155 < 1,0 ⇒ **Proof supplied**
2	15.0	0.283	0.936	0.055	$\eta_{A,2} =$ 0.070 < 1,0 ⇒ **Proof supplied**

6.2.5 Axial stability in the area of nozzles

The following calculation is performed for the nozzle N1. Outer diameter of the nozzle: $d_A = 63$ mm

This nozzle is located in cylinder tier 2. The wall thickness of the cylinder is here $s_{Z,2} =$ 15 mm.

$$\text{Ratio value} = \frac{d_A}{\sqrt{r \times s_{Z,2}}} = \frac{63}{\sqrt{1\,200 \times 15}} = 0.47 \tag{53}$$

For a ratio value ≤ 3.50 the reduction factor is:

$$\alpha_{N1} = \frac{0.65}{\sqrt{\frac{E_K^{20\,°\text{C}}}{E_L^{20\,°\text{C}}} \times \left(1 + \frac{r}{100 \times s_{Z,2}}\right)}} = \frac{0.65}{\sqrt{\frac{800}{235} \times \left(1 + \frac{1\,200}{100 \times 15}\right)}} = 0.263 \tag{54}$$

The buckling stress is calculated for the temperature $T_{ZK,o} = 35.0\,°\text{C}$

$$\sigma_{k,N1,d} = \alpha \times 0.62 \times \frac{E_K \times s_{Z,2}}{\gamma_M \times r} = 0.263 \times 0.62 \times \frac{470 \times 15.0}{1.10 \times 1\,200} = 0.870\ \text{N/mm}^2 \leq K^*_{K,d} \tag{55}$$

Proof according to DVS 2205-2 equation (49):

$$\eta_A = \frac{A_{2I} \times \gamma_I \times \Sigma\sigma_{i,d}}{\sigma_{k,d}} = \frac{1.00 \times 1.20 \times 0.056}{0.870} = 0.078 \quad < 1,0 \quad \Rightarrow$$ **Proof supplied**

with $\Sigma\sigma_{i,d}$ from section 5.3.3 , load case 1 (max. value from LC 1.1 and LC 1.2)

6.2.6 Interaction casing compressive stability / axial stability

For the proof of interaction the utilization without consideration of the axial stresses resulting from underpressure (i.e. without consideration of p_u, p_{uK}, p_{uS} and p_{eu}) is required.

$$\eta_{A,i} = \frac{A_{2I} \times \gamma_I \times \Sigma\sigma_{i,d}}{\sigma_{k,i,d}} = \frac{1.00 \times 1.20 \times \sigma_{i,d}}{\sigma_{k,i,d}} \tag{56}$$

with $\Sigma\sigma_d$ from section 6.2.1 , load case 2 (max. value from LC 2.1 and LC 2.2)

The following condition must be observed for each tier:

$$\eta_i = \eta_{A,i}^{1.25} + \eta_M^{1.25} \leq 1.0 \quad \text{with } \eta_M \text{ from section 6.2.2} \tag{57}$$

Evaluation of the equations follows in table form ($\sigma_{k,i,d}$ see section 6.2.4).

i	$\sigma_{k,i,d}$ N/mm^2	$\Sigma\sigma_{i,d}$ N/mm^2	$\eta_{A,i}$	η_M	Proof according to DVS 2205-2, equation (53)
1	0.565	0.046	0.098	0.739	$\eta_1 =$ 0.740 $<$ 1,0 $\Rightarrow$ **Proof supplied**
2	0.936	0.037	0.048	0.739	$\eta_2 =$ 0.708 $<$ 1,0 $\Rightarrow$ **Proof supplied**

6.3 Proof of the bottom plate

6.3.1 Proof for the load case filling

The bottom plate and the cylinder are connected with fillet welds. The proof of the bottom for this load case is performed according to DVS 2205 section 4.1.4.1.

The relation between cylinder radius and minimum required cylinder wall thickness is:

$$\frac{d}{s^*_{ZF}} = \frac{2\,400}{11.8} = 204 \qquad (s^*_{ZF} \text{ see section 6.2.2}) \tag{58}$$

The diagram (DVS 2205-2, figure 7) can be interpreted as follows:

$$\delta_B = 0.80$$

Proof according to DVS 2205-2, section 4.1.4.1: (with exist $s_B =$ 15.0 mm)

min s_B	$= \delta_B \times s^*_{ZF} = 0.80 \times 11.8 =$	9.4 mm	$\leq$ vorh s_B	$\Rightarrow$	**Proof supplied**
max $s_B =$	s_{ZF} $=$	15.0 mm	$\geq$ vorh s_B	$\Rightarrow$	**Proof supplied**

6.3.2 Proof of non-anchored tanks with overpressure

This proof according to DVS 2205-2, section 4.1.4.2.

The effective overpressure for this proof is:

$$p_{ü} = 0.000\,00 \text{ N/mm}^2$$
$$p_{üK} = 0.000\,50 \text{ N/mm}^2$$

$$p_1 = \max\,(p_{ü}, p_{üK}) = 0.000\,50 \text{ N/mm}^2$$

Proof of stress (equations (24) to (24c) of the Guideline DVS 2205-2)

$$\begin{aligned}
\delta_\sigma &= 1.50 \qquad \text{(Indoor installation)}\\
n_{Z,d} &= 0.077\ \text{N/mm}\\
l_B &= 13\,771\ \text{mm}\\
p_{B,k} &= -1.252\,193 \times 10^{-5}\ \text{N/mm}^2\\
h_{RF,\sigma} &= -44\ \text{mm}
\end{aligned}$$

Limitation of hoisting (equations (25) to (25c) of the Guideline DVS 2205-2)

$$\begin{aligned}
\delta_w &= 0.56 \qquad \text{(Indoor installation)}\\
n_Z &= -0.073\ \text{N/mm} \Rightarrow n_Z \text{ is negative. Further calculation is not necessary.}\\
l_B &= 0\ \text{mm}\\
p_B &= 0.000\,000\ \text{N/mm}^2\\
h_{RF,w} &= 0\ \text{mm}
\end{aligned}$$

The residual height of the filling level results for this proof:

$$h_{RF} = \max\left(h_{RF,\sigma},\, h_{RF,w}\right) = 0\ \text{mm}$$

The deciding factor for determination of the residual height of the filling level is the proof of inner underpressure.

6.3.3 Proof for internal underpressure

The effective underpressure for this proof is according to equation (47):

$$p_{u,\max} = 0.000\,30\ \text{N/mm}^2$$

The bottom dead weight is:

$$g_B = s_B \times \rho \times g \times 10^{-6} = 15.0 \times 0.960 \times 9.81 \times 10^{-6} = 0.000\,14\ \text{N/mm}^2 \qquad (59)$$

The residual height of the filling level in the tank is calculated as follows:

$$\begin{aligned}
h_{RF} &= \frac{\gamma_{F2} \times p_{u,\max} - \gamma_{F3} \times g_B}{\gamma_{F3} \times \rho \times g \times 10^{-6}} \qquad (60)\\
&= \frac{1.50 \times 0.000\,30 - 0.90 \times 0.000\,14}{0.90 \times 1.500 \times 9.81 \times 10^{-6}} = 24\ \text{mm} \qquad \Rightarrow \textbf{Proof supplied}
\end{aligned}$$

The proof is assumed to have been provided when the above-mentioned value for h_{RF} is not exceeded.

6.4 Proof of the anchorage

Eine zusätzliche Verankerung des Behälters ist nicht erforderlich, da ...

a) the hoisting of the cylinder resulting from overpressure is not higher than the limiting value $w_{limit} =$ 10 mm (proof of deformation),

b) the bending load of the bottom resulting from overpressure can be absorbed safely (proof of stress),

c) there can be no overturning moment from wind loads when the tank is installed indoors.

The proofs mentioned in a) and b) see section 6.3.2.

6.5 Proof of the lifting lugs

2 lifting lugs are mounted on the tank in accordance with DVS 2205-2, figure 11. A parallel lifter (tie-bar) will be used to lift the tank.

The 1.5x load (impact coefficient) on each lifting lug amounts to

$$F = \frac{1.5 \times \gamma_{F1} \times G_E}{2} = \frac{1.5 \times 1.35 \times 3\,768}{2} = 3\,815 \text{ N} \tag{61}$$

It can be demonstrated that this load is temporarily sustainable up to 20 °C.In this case $\gamma_I =$ 1.20 is applicable, regardless of later load case.

The thickness of the welding-seam between the lifting lugs and the cylinder is:

$$a = 0.7 \times s_{Z,1} = 0.7 \times 10.0 = 7.0 \text{ mm} \quad \text{(umlaufend)}$$

Diameter of the shackle:

$$d_{Sch} = 20.0 \text{ mm}$$

Diameter of the hole in the lifting lugs:

$$d_L = 22.0 \text{ mm } (\leq 1.1 \times d_{Sch} = 22.0 \text{ mm see DVS 2205-2, equation (40))}$$

Wall thickness of the lifting lugs (requ. s_{Oe} = minimum required thickness):

$$erf s_{Oe} = \frac{F \times A_1 \times \gamma_I}{2 \times d_{Sch} \times K^*_{K,d}} = \frac{3\,815 \times 1.00 \times 1.20}{2 \times 20.0 \times 13.42} = 8.5 \text{ mm} \tag{62}$$

$\min s_{Oe} = s_{Z,1} = 10.0$ mm
$\max s_{Oe} = 3 \times s_{Z,1} = 30.0$ mm

For a selected thickness $s_{Oe} = 15.0$ mm is ⇒ **Proof supplied**

Width of the lifting lugs:

Shearing stress of the cross weld when lifting the lying tank

$$b_{Oe,1} = \frac{F \times A_1 \times \gamma_I}{a \times f_Z \times K^*_{K,d}} = \frac{3\,815 \times 1.00 \times 1.20}{7.0 \times 0.8 \times 13.42} = 60.9 \text{ mm} \tag{63}$$

Eye bar

$$b_{Oe,2} = \frac{F \times A_1 \times \gamma_I}{s_{Oe} \times K^*_{K,d}} + \frac{7}{3} \times d_L = \frac{3\,815 \times 1.00 \times 1.20}{15.0 \times 13.42} + \frac{7}{3} \times 22.0 = 74.1 \text{ mm} \tag{64}$$

For a selected width $b_{Oe} = 80.0$ mm is ⇒ **Proof supplied**

Minimum height of the lifting lugs:

$h_{Oe} = 2.5 \times b_{Oe} = 2.5 \times 80.0 = 200.0$ mm for lug with curve base
$h_{Oe} = 2.0 \times b_{Oe} = 2.0 \times 80.0 = 160.0$ mm for lug with cornered base

7 Summary

This structural analysis supply the proofs described in the guideline DVS 2205-2.