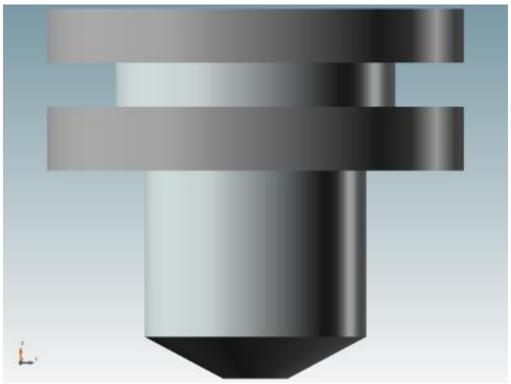
Calculation report

EN 13445 Ed. 2014 Issue 3

Project:	Laboratorní autokláv
Item:	Laboratorní autokláv 1L
Customer:	VUOS a.s.
Drawing:	400-001
Revision:	-
Date:	04.08.2018

Internal design pressure	P =	4.00 MPa
Internal design temperature	T =	200.00 °C
Internal corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Joint efficiency	z =	0.85
Minimum design temperature	=	0 °C



Test pressure (MPa)

Component	*P*	*Static head (design)*	*Static head (test)*	*Stress ratio*	*1.25·P·f0/f*	*1.43·P*
Conical shell #1	4.00	0	0.002	1,81	9.05	5.72
Cylindrical shell #1	4.00	0	0.001	1,81	9.05	5.72
Loose lap flange #1	4.00	0	0.0005	1,327	6.63	5.72
Bolted flat cover #1	4.00	0	0.00003	1,81	9.05	5.72

All pressures in MPa.

Item design pressure P = 4.00 MPa
Item maximum allowable design pressure (Pmax) = 4.26 MPa (limited by Loose lap flange #1)
Item lowest stress ratio = 1.327 (limited by Loose lap flange #1)
Item test pressure = Pt=max[1.25·Pd·(Item f0/f);1.43·Pd] = 6.63 MPa

Maximum Pressures (MPa)

Component	*Internal, test*	*Internal*
Conical shell #1	18.29	5.99
Cylindrical shell #1	28.33	8.15
Loose lap flange #1	5.00	4.26
Bolted flat cover #1	13.10	4.66

All pressures in MPa.

Weights

Component	*Dead*	*Live*	*Liquid*	*Full of water*	*Operating*
Bolted flat cover #2	5 kg	0 kg	0 kg	5 kg	5 kg
Welded flat cover #1	1 kg	0 kg	0 kg	1 kg	1 kg
Conical shell #1	1 kg	0 kg	0 kg	1 kg	1 kg
Cylindrical shell #1	2 kg	0 kg	0 kg	3 kg	2 kg
Loose lap flange #1	8 kg	0 kg	0 kg	8 kg	8 kg
Bolted flat cover #1	6 kg	0 kg	0 kg	6 kg	6 kg
Totals:	23 kg	0 kg	0 kg	25 kg	23 kg

Total volume: 0.00153 m^3
Center of gravity (erection): Cx=0 mm, Cy=0 mm, Cz=127.73 mm, W = 17 kg
Center of gravity (operating): Cx=0 mm, Cy=0 mm, Cz=127.73 mm, W = 17 kg
Center of gravity (test): Cx=0 mm, Cy=0 mm, Cz=124.12 mm, W = 19 kg

Definitions
Dead: net, uncorroded weight of component, including additional dead weight

Live: additional live weight on component

Liquid: weight of liquid contained in component in operating conditions (depending on liquid level)

Insulation: weight of insulation on component, when present

Full of water: sum of component's proper weight and contained water in test conditions

Operating: sum of component's proper weight and contained liquid in operating conditions

Bill of materials

Component	***Dimensions***	***Material***
Bolted flat cover #2 - Flange	Od = 160.00 mm, Tk = 30.00 mm	1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube (t ≤ 60.00 mm) - No.: 1.4301
Welded flat cover #1	Od = 25.00 mm, Tk = 10.00 mm	Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035
Conical shell #1	Min Id = 25.00 mm, Max Id = 115.00 mm, Tk = 6.00 mm, α = 65.00 °, L = 24.81 mm	Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035
Cylindrical shell #1	Id = 115.00 mm, Od = 127.00 mm, Tk = 6.00 mm, L = 137.00 mm	Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035
Loose lap flange #1 - Flange	Id = 142.00 mm, Od = 240.00 mm, Tk = 32.00 mm	1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75) - No.: 1.4571
Loose lap flange #1 - Gasket	Rubber O-rings - below 75° IRH	
Loose lap flange #1 - Bolts	8 x ISO M16 x 2.00	A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)
Bolted flat cover #1 - Flange	Od = 240.00 mm, Tk = 30.00 mm	Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035

Material properties summary

1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube (t ≤ 60.00 mm) - No.: 1.4301

Temp.	***Allowable***	***Yield strength***	***Tensile strength***	***Elasticity***	***Thermal expansion***
Room	166.67 MPa	195.00 MPa	500.00 MPa	195 188.00 MPa	0.000015967 1/°C
Design	103.33 MPa	127.00 MPa	0 MPa	182 323.00 MPa	0.000016802 1/°C
Test	250.00 MPa				

1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75) - No.: 1.4571

Temp.	***Allowable***	***Yield strength***	***Tensile strength***	***Elasticity***	***Thermal expansion***
Room	173.33 MPa	220.00 MPa	520.00 MPa	195 188.00 MPa	0.000016500 1/°C
Design	130.67 MPa	167.00 MPa	390.00 MPa	182 323.00 MPa	0.000017500 1/°C
Test	247.62 MPa				

A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

Temp.	***Allowable***	***Yield strength***	***Tensile strength***	***Elasticity***	***Thermal expansion***
Room	291.67 MPa	450.00 MPa	700.00 MPa	195 188.00 MPa	0.000015967 1/°C
Design	240.00 MPa	360.00 MPa	0 MPa	182 323.00 MPa	0.000016802 1/°C
Test	428.57 MPa				

Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36) - No.: 3.7035

Temp.	***Allowable***	***Yield strength***	***Tensile strength***	***Elasticity***	***Thermal expansion***
Room	175.00 MPa	300.00 MPa	420.00 MPa	22 230.75 MPa	0.000010780 1/°C
Design	96.67 MPa	145.00 MPa	420.00 MPa	97 000.00 MPa	0.000093000 1/°C
Test	285.71 MPa				

Notes
Allowable stress calculation may vary upon component type or characteristics

Yield strength shown refers to 0.2% plastic strain

Bolted flat cover - Bolted flat cover #2

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 10

Design data

Internal design temperature	Ti	=	200.00 °C
Internal design pressure	Pi	=	4.00 MPa
Joint efficiency	z	=	1.00

Material: 1.4301 bezešvá trubka (EN 10216-5:2008) - Seamless tube (t ≤ 60.00 mm)

Nominal design stress at internal design temperature	$f = (\frac{R_{p1.0/T}}{1.5})$	=	103.33 MPa
Nominal design stress at room temperature	$f = \max[\frac{R_{p1.0/T}}{1.5}; \min(\frac{R_{p1.0/T}}{1.2}; \frac{R_{m/T}}{3})]$	=	166.67 MPa
Nominal design stress in test condition	$f_{test} = \max(\frac{R_{p1.0/Ttest}}{1.05}; \frac{R_{m/Ttest}}{2})$	=	250.00 MPa

Bolts Material: A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

Nominal design stress at design temperature	$f = \min(\frac{R_{p0.2/T}}{3}; \frac{R_{m/20}}{4})$	=	120.00 MPa
Nominal design stress at room temperature	$f = \frac{R_{m/T}}{4}$	=	175.00 MPa
Nominal design stress in test condition	$f = 1.5 \cdot \frac{R_{m/T}}{4}$	=	262.50 MPa

Geometry

Nominal thickness	en	=	30.00 mm
Corrosion allowance	c	=	0 mm
External corrosion allowance	ce	=	0 mm
Undertolerance	δ	=	0 mm
Bolt circle	C	=	206.00 mm
Mean gasket diameter	Gmean	=	117.00 mm
Adopted thickness (periphery)	e1	=	24.00 mm
Gasket groove depth	g	=	0 mm
External corrosion allowance	c''	=	0 mm
Flange external diameter	A	=	160.00 mm
Inside diameter of connected flange	B	=	115.00 mm
Inside diameter of connected flange (corroded)	B* = B + 2c	=	115.00 mm
Hub thickness of connected flange	g1	=	4.00 mm
Hub thickness of connected flange (corroded)	g1*	=	4.00 mm

Gasket parameters

Gasket factor	m	=	0
Minimum gasket seating pressure	y	=	0.70 MPa
Gasket contact width	w	=	0 mm
Basic gasket seating width	b0 = w / 2	=	0 mm
Effective gasket seating width	b = b0	=	0 mm
Diameter of gasket load reaction	G = Gmean	=	117.00 mm

Bolt loads

Number of bolts		=	8
Bolt type		=	ISO M16 x 2.00
Bolt hole diameter		=	18.00 mm
Bolt spacing	tBmax	=	80.90 mm
Root area of one bolt		=	144.0 mm²
Distance between centre lines of adjacent bolts	δb	=	80.90 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$	=	43 005 N
Minimum required bolt load for operating condition	Wop = H	=	43 005 N
	$H_t = \frac{G^2 \pi P_t}{4}$	=	71 310 N
Minimum required bolt load for the test condition	Wt = Ht	=	71 310 N

Minimum required bolt load for assembly condition	WA	=	0 N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}}; \frac{W_{op}}{f_B}; \frac{W_t}{f_{B,t}}\right]$	=	358.4 mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB	=	1 152.0 mm²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot w}{f_{B,A}}$	=	0 mm²
Design bolt load for assembly condition	$W = 0,5(A_{B,\min} + A_B) f_{B,A}$	=	132 158 N

AB ≥ AB,min: Ok

Length of screws in threaded hole to 11.4.3.3

Length of screws in threaded hole	ls	=	36.00 mm
	Rp,component	=	155.00 MPa
	Rp,screw	=	360.00 MPa
	$l_{s,\min} = \max\left[0,8 \cdot d_n \cdot \frac{R_{p,\text{screw}}}{R_{p,\text{component}}};\ 0,8 \cdot d_n\right]$	=	29.73 mm

ls ≥ ls,min (36.00 mm ≥ 29.73 mm): Ok

Internal pressure

Overpressure due to static head	Ph	=	0 MPa
Calculation pressure	P=Pi+Ph	=	4.00 MPa
Minimum thickness in assembly condition (center)	$e_A = \sqrt{C_F \frac{3(C-G)}{\pi G}(\frac{W}{f_A})} + \max(c, g) + ce + \delta$	=	24.00 mm
Minimum thickness in assembly condition (periphery)	$e_{A1} = \sqrt{C_F \frac{3(C-G)}{\pi G}(\frac{W}{f_A})} + ce + \delta$	=	24.00 mm
Minimum thickness in operating condition (center)	$e_P = \sqrt{[\frac{3(3+v)}{32}G^2 + 3C_F(\frac{G}{4} + 2b \cdot m)(C-G)]\frac{P}{f}} + \max(c, g) + ce + \delta$	=	21.59 mm
Minimum thickness in operating condition (periphery)	$e_{P1} = \sqrt{3C_F(\frac{G}{4} + 2b \cdot m)(C-G)\frac{P}{f}} + ce + \delta$	=	17.39 mm
Minimum required thickness (periphery)	e1=max(eA1;eP1)	=	24.00 mm
Minimum required thickness	e=max(eA;eP)	=	24.00 mm

e1 ≥ e1(min): Ok
en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	18.68 MPa
Maximum allowable design pressure	=	4.00 MPa

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f	=	1.32653
Coincident design pressure for the maximum pressure load case	Pd	=	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f)	=	6.63 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd	=	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2)	=	6.63 MPa
Overpressure due to static head in test condition	Pht	=	0 MPa
Calculation pressure	Pc=Pt+Pht	=	6.63 MPa
Minimum thickness in operating condition (center)	$e_P = \sqrt{[\frac{3(3+v)}{32}G^2 + 3C_F(\frac{G}{4} + 2b \cdot m)(C-G)]\frac{P}{f}} + g + \delta$	=	17.88 mm
Minimum thickness in operating condition (periphery)	$e_{P1} = \sqrt{3C_F(\frac{G}{4} + 2b \cdot m)(C-G)\frac{P}{f}} + \delta$	=	14.39 mm
Minimum required thickness	e=eP	=	17.88 mm

en ≥ e: Ok

Welded flat cover - Welded flat cover #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 10

Design data

Internal design temperature	Ti	=	200.00 °C
Internal design pressure	Pi	=	4.00 MPa
Joint efficiency	z	=	0.85

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$	=	96.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$	=	175.00 MPa
Nominal design stress in test condition	$f_{\mathrm{test}} = (\frac{R_{\mathrm{p0.2/Ttest}}}{1.05})$	=	285.71 MPa

Geometry

Outside diameter	De	=	25.00 mm
Nominal thickness	en	=	10.00 mm
Minimum head thickness after forming	t-c'	=	10.00 mm
Corrosion allowance	c	=	0 mm
External corrosion allowance	ce	=	0 mm
Undertolerance	δ	=	0 mm
Type of flat end		=	Welded directly to shell
Thickness of cylindrical part connected to the end	es	=	6.00 mm
Length of cylindrical part connected to the end	lc	=	160.00 mm

Internal pressure

Overpressure due to static head	Ph	=	0 MPa
Calculation pressure	P=Pi+Ph	=	4.00 MPa
Length of cylindrical shell which contributes to the strength of the flat end	lcyl	=	160.00 mm
Length of cylindrical shell which contributes to the strength of the flat end	$l_{\mathrm{cyl}} = \sqrt{(D_i + e_s)e_s}$	=	10.68 mm
Parameter C1	$C_1 = \max\left\{[0.40825 \cdot A_1 \frac{D_i + e_s}{D_i}], [0.299(1 + 1.7\frac{e_s}{D_i})]\right\}$	=	0.53360
Minimum required thickness	$e = C_1 \cdot D_i \sqrt{\frac{P}{f}}$	=	1.41 mm
			en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure		=	100.00 MPa
Maximum allowable design pressure		=	100.00 MPa

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f	=	1.32653
Coincident design pressure for the maximum pressure load case	Pd	=	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f)	=	6.63 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd	=	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2)	=	6.63 MPa
Overpressure due to static head in test condition	Pht	=	0 MPa
Calculation pressure	Pc=Pt+Pht	=	6.63 MPa
Length of cylindrical shell which contributes to the strength of the flat end	lcyl	=	160.00 mm
Length of cylindrical shell which contributes to the strength of the flat end	$l_{\mathrm{cyl}} = \sqrt{(D_i + e_s)e_s}$	=	10.68 mm
Parameter C1	$C_1 = \max\left\{[0.40825 \cdot A_1 \frac{D_i + e_s}{D_i}], [0.299(1 + 1.7\frac{e_s}{D_i})]\right\}$	=	0.53360
Minimum required thickness	$e = C_1 \cdot D_i \sqrt{\frac{P}{f}}$	=	1.06 mm
			en ≥ e: Ok

Conical shell - Conical shell #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8

Design data

Internal design temperature	Ti =	200.00 °C
Internal design pressure	Pi =	4.00 MPa
Joint efficiency	z =	1.00

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	96.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	175.00 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	285.71 MPa

Geometry

Length	L =	24.81 mm
Nominal thickness	en =	6.00 mm
Corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Maximum Inside Diameter	Di =	115.00 mm
Maximum Outside Diameter	De =	127.00 mm
Minimum Inside Diameter	di =	25.00 mm
Minimum Outside Diameter	de =	37.00 mm
Half-apex angle	α =	65.00 °
Thickness at large end	e2nL =	6.00 mm
Thickness at small end	e2ns =	6.00 mm
Nominal thickness of cylinder at large end	e1nL =	6.00 mm
Minimum required thickness of cylinder at large end	e1L =	2.43 mm
Nominal thickness of cylinder at small end	e1ns =	6.00 mm
Minimum required thickness of cylinder at small end	e1s =	0.53 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	4.00 MPa
Mean diameter of the cone at large end	Dc=Di+e1nL+c+ce+δ =	121.00 mm
Minimum length along cone	l2L=√((Dc*e2L)/cos(α)) =	35.14 mm
Calculation diameter	DK=Dc-e1L-2r[1-cos(α)]-l2·sin(α) =	86.30 mm
Minimum required cone thickness	e2+c+ce+δ (iterative) =	4.31 mm
		en≥e2: Ok

Large end junction (without knuckle)

Minimum length along cylinder	1.4·l1L=1.4·√(Dc*e1L) =	24.00 mm
Minimum length along cone	1.4·l2L=1.4·√((Dc*e2L)/cos(α)) =	49.20 mm
β factor defined in 7.6.6	7.6.6 =	1.46651
Minimum required thickness at the junction at the large end of the cone	e2L=ej=(P·Dc·β)/2f+c+ce+δ =	3.67 mm
Maximum allowable pressure of junction at large end	Pmax(large end)=2·f·e2nL/(β·Dc) =	6.54 MPa
		e2nL≥ej: Ok

Small end junction

Mean diameter of the cone	dc=di+e1ns+c+ce+δ =	31.00 mm
Minimum length along cylinder	l1s=√(dc*e1s') =	13.64 mm
Minimum length along cone	l2s=√((dc*e2s')/cos(α)) =	20.98 mm
s factor defined in 7.6.8	s=e2ns/e1ns =	1.00000
τ factor defined in 7.6.8	7.6-24/23 =	2.53825
βH factor defined in 7.6.8	7.6-25 =	1.26817
Minimum required thickness at the junction at the small end of the cone	e2s (iterative) =	0.008 mm

Maximum allowable pressure of junction at small end	Pmax(small end)=2·f·z·e1/(dc·βH) =	29.51 MPa
		e2ns≥e2s: Ok
		en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	18.29 MPa
Maximum allowable design pressure	=	5.99 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(di/2+en/2) =	19.355 %

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.32653
Coincident design pressure for the maximum pressure load case	Pd =	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	6.63 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	6.63 MPa
Overpressure due to static head in test condition	Pht =	0.002 MPa
Calculation pressure	Pc=Pt+Pht =	6.63 MPa
Mean diameter of the cone at large end	Dc=Di+e1nL+δ =	121.00 mm
Minimum length along cone	l2L=√((Dc*e2L)/cos(α)) =	27.27 mm
Calculation diameter	DK=Dc-e1L-2r[1-cos(α)]-I2·sin(α) =	93.43 mm
Minimum required cone thickness	e2+δ (iterative) =	2.60 mm
		en≥e2: Ok

Large end junction (without knuckle)

Minimum length along cylinder	1.4·l1L=1.4·√(Dc*e1L) =	24.00 mm
Minimum length along cone	1.4·l2L=1.4·√((Dc*e2L)/cos(α)) =	38.17 mm
β factor defined in 7.6.6	7.6.6 =	1.79906
Minimum required thickness at the junction at the large end of the cone	e2L=ej=(P·Dc·β)/2f+δ =	2.53 mm
Maximum allowable pressure of junction at large end	Pmax(large end)=2·f·e2nL/(β·Dc) =	15.75 MPa
		e2nL≥ej: Ok

Small end junction

Mean diameter of the cone	dc=di+e1ns+c+ce+δ =	31.00 mm
Minimum length along cylinder	l1s=√(dc*e1s') =	13.64 mm
Minimum length along cone	l2s=√((dc*e2s')/cos(α)) =	20.98 mm
s factor defined in 7.6.8	s=e2ns/e1ns =	1.00000
τ factor defined in 7.6.8	7.6-24/23 =	2.53825
βH factor defined in 7.6.8	7.6-25 =	1.26817
Minimum required thickness at the junction at the small end of the cone	e2s (iterative) =	0.008 mm
Maximum allowable pressure of junction at small end	Pmax(small end)=2·f·z·e1/(dc·βH) =	87.21 MPa
		e2ns≥e2s: Ok
		en ≥ e: Ok

Cylindrical shell - Cylindrical shell #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 7 and 8

Design data

Internal design temperature	Ti =	200.00 °C
Internal design pressure	Pi =	4.00 MPa
Joint efficiency	z =	0.85

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	96.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	175.00 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	285.71 MPa

Geometry

Inside diameter	Di =	115.00 mm
Outside diameter	De =	127.00 mm
Length	L =	137.00 mm
Nominal thickness	en =	6.00 mm
Corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm

Internal pressure

Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P=Pi+Ph =	4.00 MPa
Inside diameter	Di'=Di+2δ+2c =	115.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + c + ce + \delta$ =	2.87 mm

e/De ≤ 0,16 (0.02300 ≤ 0.16000): Ok
en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure	=	28.33 MPa
Maximum allowable design pressure	=	8.15 MPa

Deformation according to EN13445-4 Clause 9

Ratio of deformation	F=50·en/(Di/2+en/2) =	4.959 %

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f =	1.32653
Coincident design pressure for the maximum pressure load case	Pd =	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f) =	6.63 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd =	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2) =	6.63 MPa
Overpressure due to static head in test condition	Pht =	0.001 MPa
Calculation pressure	Pc=Pt+Pht =	6.63 MPa
Inside diameter	Di'=Di+2δ =	115.00 mm
Minimum required thickness	$e = \frac{P \cdot D_i'}{2f \cdot z - P} + \delta$ =	1.35 mm

e/De ≤ 0,16 (0.01100 ≤ 0.16000): Ok
en ≥ e: Ok

Loose lap flange - Loose lap flange #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 11

Flange material	**1.4571 plech 13,5-75 mm (EN 10028-7:2008) - Plate (13,501 ≤ t ≤ 75)**
Stub flange (shell) material	**Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)**
Bolting material	**A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)**
Gasket	**Rubber O-rings - below 75° IRH**

Allowable stresses	**Flange - f**	**Hub - fH**	**Bolting - fB**
Design condition	130.67 MPa / 18 951.6 psi	96.67 MPa / 14 020.3 psi	120.00 MPa / 17 404.5 psi
Seating condition	173.33 MPa / 25 139.9 psi	175.00 MPa / 25 381.6 psi	175.00 MPa / 25 381.6 psi
Test condition	247.62 MPa / 35 914.1 psi	285.71 MPa / 41 439.4 psi	262.50 MPa / 38 072.4 psi

Internal pressure	Pd =	4.00 MPa
Overpressure due to static head	Ph =	0 MPa
Calculation pressure	P =	4.00 MPa
Calculation temperature	T =	200.00 °C

Geometry

Corrosion allowance	c =	0 mm
Flange external diameter	A =	240.00 mm
Inside diameter	B =	142.00 mm
Stub flange thickness	e' =	22.00 mm
Shell outside diameter	B' =	127.00 mm
Outside diameter of the contact face between loose and stub flanges	A2 =	160.00 mm
Inside diameter of the contact face between loose and stub flanges	B2 =	142.00 mm
Nominal gap between the shell and loose flange	δ = (B - B') / 2 =	7.50 mm
Assumed diameter of load reaction between loose and stub flanges	G1 = (A2 + B2) / 2 =	151.00 mm
Area of the contact face	$A_C = \frac{\pi}{2}\min\left[(A_2-\delta)^2 - G_1^2; G_1^2 - (B_2+\delta)^2 \right]$ =	708.0 mm²
Bolt circle	C =	260.00 mm
Flange thickness	en =	32.00 mm
Mean gasket diameter	Gmean =	140.00 mm
Hub length	h =	4.00 mm
Thickness of hub at back of flange	g1 =	4.00 mm
Thickness of hub at back of flange (corroded)	g1* =	4.00 mm
Thickness of hub at small end	g0 =	6.00 mm
Thickness of hub at small end (corroded)	g0* =	6.00 mm

Gasket parameters

Gasket factor	m =	0
Minimum gasket seating pressure	y =	0.70 MPa
Gasket contact width	w =	5.00 mm
Basic gasket seating width	b0 = w / 2 =	2.50 mm
Effective gasket seating width	b = b0 =	2.50 mm
Diameter of gasket load reaction	G = Gmean =	140.00 mm

Bolt loads

Number of bolts	n =	8
Bolt type	=	ISO M16 x 2.00
Root area of one bolt	=	144.0 mm²
Distance between centre lines of adjacent bolts	δb =	102.10 mm
Bolt outside diameter	db =	14.12 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	61 575 N
Minimum required bolt load for operating condition	Wop = H =	61 575 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	102 110 N

Minimum required bolt load for the test condition	Wt = Ht	=	102 110 N
Minimum required bolt load for assembly condition	WA	=	0 N
Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}}; \frac{W_{op}}{f_B}; \frac{W_t}{f_{Bt}}\right]$	=	513.1 mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB	=	1 152.0 mm²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot w}{f_{B,A}}$	=	17.6 mm²
Design bolt load for assembly condition	$W = 0,5(A_{B,\min} + A_B) f_{B,A}$	=	145 699 N

AB ≥ AB,min: Ok

Length of screws in threaded hole to 11.4.3.3

Length of screws in threaded hole	ls	=	25.00 mm
	Rp,component	=	196.00 MPa
	Rp,screw	=	360.00 MPa
	$l_{s,\min} = \max\left[0,8 \cdot d_n \cdot \frac{R_{p,\text{screw}}}{R_{p,\text{component}}};\ 0,8 \cdot d_n\right]$	=	23.51 mm

ls ≥ ls,min (25.00 mm ≥ 23.51 mm): Ok

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}};\ 1\right]$	=	1.00000
Ratio of the flange diameters	K = A / B	=	1.69014
Length parameter	$l_0 = \sqrt{B \cdot g_0^*}$	=	29.19 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}(B^2 P)$	=	63 347 N
Hydrostatic end force due to pressure on flange face	HT = H - HD	=	-1 772 N
Radial distance from bolt circle to circle on which load reaction acts	hL = (A - G1) / 2	=	44.50 mm
Radial distance from bolt circle to circle on which HD acts	hD = (A - B) / 2	=	49.00 mm
Radial distance from gasket load reaction to bolt circle	hG = (A - G) / 2	=	50.00 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2A - B - G) / 4	=	49.50 mm
Flange stress factor	$\beta_T = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{(1.0472 + 1.9448K^2)(K-1)}$	=	1.62942
Flange stress factor	$\beta_U = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{1.36136(K^2-1)(K-1)}$	=	4.25664
Flange stress factor	$\beta_Y = \frac{1}{K-1}\left[0.66845 + 5.7169\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=	3.87356

Flange constants - Stub flange

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0,5}}};\ 1\right]$	=	1.00000
Ratio of the flange diameters	K = A / B*	=	1.39130
Length parameter	$l_0 = \sqrt{B^* \cdot g_0^*}$	=	26.27 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}(B^{*2} P)$	=	41 548 N
Hydrostatic end force due to pressure on flange face	HT = H - HD	=	20 028 N
Radial distance from bolt circle to circle on which HD acts	hD = (C - B* - g1*) / 2	=	16.00 mm
Radial distance from gasket load reaction to bolt circle	hG = (C - G) / 2	=	5.50 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2C - B* - G) / 4	=	11.75 mm
Flange stress factor	$\beta_T = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{(1.0472 + 1.9448K^2)(K-1)}$	=	1.75801
Flange stress factor	$\beta_U = \frac{K^2(1 + 8.55246\log_{10}K) - 1}{1.36136(K^2-1)(K-1)}$	=	6.64058
Flange stress factor	$\beta_Y = \frac{1}{K-1}\left[0.66845 + 5.7169\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=	6.04294
	βF	=	0.92859
	βV	=	0.77762

Hub stress correction factor	φ =	1.00000
	$\lambda = \left(\frac{e\beta_F + l_0}{\beta_T l_0} + \frac{e^3\beta_V}{\beta_U l_0 g_0^{*2}}\right)$ =	2.32978

Flange moments

Total moment acting upon flange for assembly condition	MA = W·hL =	6 483.6 N·m
Total moment acting upon flange for operating condition	Mop = Wop·hL =	2 740.1 N·m
Moment factor used to design split rings	Fs =	1.00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op}\frac{C_F}{B}$ =	19.3 N·m
Moment exerted on the flange per unit of length (assembly)	$M = F_s \cdot M_A\frac{C_F}{B}$ =	45.7 N·m

Flange moments - Stub flange

Total moment acting upon flange for assembly condition	MA = W·hG =	801.3 N·m
Total moment acting upon flange for operating condition	$M_{op} = \lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \rvert$ =	900.1 N·m
Moment factor used to design split rings	Fs =	1.00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op}\frac{C_F}{B^*}$ =	7.8 N·m
Moment exerted on the flange per unit of length (assembly)	$M = F_s \cdot M_A\frac{C_F}{B^*}$ =	7.0 N·m

Flange stresses - operating condition

Longitudinal stress in hub	σH_o =	0 MPa
Radial stress in flange	σr_o =	0 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2}$ =	72.99 MPa
Stress factor	k =	1.00000
	k·σH ≤ 1.5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0.5k(σH + σr) ≤ f:	Ok
	0.5k(σH + σθ) ≤ f:	Ok
	σb ≤ 1.5min(f;fH):	Ok

Flange stresses - operating condition - Stub flange

Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda g_1^{*2}}$ =	209.97 MPa
Radial stress in flange	$\sigma_r = \frac{(1.333e \cdot \beta_F + l_0)M}{\lambda \cdot e^2 \cdot l_0}$ =	14.14 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r\frac{K^2+1}{K^2-1}$ =	53.37 MPa
Stress factor	k =	1.00000
	k·σH ≤ 1.5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0.5k(σH + σr) ≤ f:	Ok
	0.5k(σH + σθ) ≤ f:	Ok

Flange stresses - seating condition

Longitudinal stress in hub	σH_A =	0 MPa
Radial stress in flange	σr_A =	0 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2}$ =	172.72 MPa
	k·σH ≤ 1.5min(f;fH):	Ok
	k·σr ≤ f:	Ok
	k·σθ ≤ f:	Ok
	0.5k(σH + σr) ≤ f:	Ok
	0.5k(σH + σθ) ≤ f:	Ok
	σb ≤ 1.5min(f;fH):	Ok

Flange stresses - seating condition - Stub flange

Longitudinal stress in hub	$\sigma_H = \frac{\varphi M}{\lambda g_1^{*2}}$	=	186.93 MPa
Radial stress in flange	$\sigma_r = \frac{(1.333e \cdot \beta_F + l_0) M}{\lambda \cdot e^2 \cdot l_0}$	=	12.59 MPa
Tangential stress in flange	$\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2+1}{K^2-1}$	=	47.51 MPa

k·σH ≤ 1.5min(f;fH): Ok
k·σr ≤ f: Ok
k·σθ ≤ f: Ok
0.5k(σH + σr) ≤ f: Ok
0.5k(σH + σθ) ≤ f: Ok

Maximum allowable pressures (at the top of the vessel)

New & cold (flange)	=	5.00 MPa
Hot & corroded (flange)	=	4.26 MPa
New & cold (bolts)	=	13.10 MPa
Hot & corroded (bolts)	=	8.98 MPa

Hydrostatic test

Item or side hydrostatic test pressure	Pt	=	6.63 MPa
Overpressure due to static head	Ph	=	0.0005 MPa
Calculation pressure	P = Pt + Ph	=	6.63 MPa

Flange constants

Bolt pitch correction factor	$C_F = \max\left[\sqrt{\frac{\delta_b}{2d_b + \frac{6e}{m+0{,}5}}}\,;1\right]$	=	1.00000
Ratio of the flange diameters	K = A / B	=	1.69014
Length parameter	$l_0 = \sqrt{B \cdot g_0^*}$	=	29.19 mm
Hydrostatic end force applied via shell to flange	$H_D = \frac{\pi}{4}(B^2 P)$	=	105 048 N
Hydrostatic end force due to pressure on flange face	HT = H - HD	=	-2 938 N
Radial distance from bolt circle to circle on which load reaction acts	hL = (A - G1) / 2	=	44.50 mm
Radial distance from bolt circle to circle on which HD acts	hD = (A - B) / 2	=	49.00 mm
Radial distance from gasket load reaction to bolt circle	hG = (A - G) / 2	=	50.00 mm
Radial distance from bolt circle to circle on which HT acts	hT = (2A - B - G) / 4	=	49.50 mm
Flange stress factor	$\beta_T = \frac{K^2(1+8.55246\log_{10}K)-1}{(1.0472+1.9448K^2)(K-1)}$	=	1.62942
Flange stress factor	$\beta_U = \frac{K^2(1+8.55246\log_{10}K)-1}{1.36136(K^2-1)(K-1)}$	=	4.25664
Flange stress factor	$\beta_Y = \frac{1}{K-1}\left[0.66845+5.7169\left(\frac{K^2\log_{10}K}{K^2-1}\right)\right]$	=	3.87356

Flange moments

Total moment acting upon flange for operating condition	Mop = Wop·hL	=	2 740.1 N·m
Moment factor used to design split rings	Fs	=	1.00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op} \frac{C_F}{B}$	=	19.3 N·m

Flange moments - Stub flange

Total moment acting upon flange for operating condition	$M_{op} = \lvert H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \rvert$	=	1 492.5 N·m
Moment factor used to design split rings	Fs	=	1.00
Moment exerted on the flange per unit of length (operating)	$M = F_s \cdot M_{op} \frac{C_F}{B}$	=	13.0 N·m

Flange stresses - operating condition

Longitudinal stress in hub	σH_o	=	0 MPa
Radial stress in flange	σr_o	=	0 MPa

Tangential stress in flange $\sigma_\theta = \frac{\beta_Y \cdot M}{e^2}$ = 72.99 MPa

Stress factor k = 1.00000

k·σH ≤ 1.5min(f;fH): Ok
k·σr ≤ f: Ok
k·σθ ≤ f: Ok
0.5k(σH + σr) ≤ f: Ok
0.5k(σH + σθ) ≤ f: Ok
σb ≤ 1.5min(f;fH): Ok

Flange stresses - operating condition - Stub flange

Longitudinal stress in hub $\sigma_H = \frac{\varphi M}{\lambda g_1^{*2}}$ = 348.16 MPa

Radial stress in flange $\sigma_r = \frac{(1.333 e \cdot \beta_F + l_0) M}{\lambda \cdot e^2 \cdot l_0}$ = 23.44 MPa

Tangential stress in flange $\sigma_\theta = \frac{\beta_Y \cdot M}{e^2} - \sigma_r \frac{K^2+1}{K^2-1}$ = 88.49 MPa

Stress factor k = 1.00000

k·σH ≤ 1.5min(f;fH): Ok
k·σr ≤ f: Ok
k·σθ ≤ f: Ok
0.5k(σH + σr) ≤ f: Ok
0.5k(σH + σθ) ≤ f: Ok

Bolted flat cover - Bolted flat cover #1

According to: EN 13445 Ed. 2014 Issue 3, Part 3, Clause 10

Design data

Internal design temperature	Ti =	200.00 °C
Internal design pressure	Pi =	4.00 MPa
Joint efficiency	z =	1.00

Material: Ti grade 2 (EN 10028-7:2008) - Plate (1,5 ≤ t ≤ 36)

Nominal design stress at internal design temperature	$f = \min(\frac{R_{p0.2/T}}{1.5}; \frac{R_{m/20}}{2.4})$ =	96.67 MPa
Nominal design stress at room temperature	$f = \min(\frac{R_{p0.2/20}}{1.5}; \frac{R_{m/20}}{2.4})$ =	175.00 MPa
Nominal design stress in test condition	$f_{test} = (\frac{R_{p0.2/Ttest}}{1.05})$ =	285.71 MPa

Bolts Material: A2-70 (ISO 3506-1:2009) - Bolting (t ≤ 39.00 mm)

Nominal design stress at design temperature	$f = \min(\frac{R_{p0.2/T}}{3}; \frac{R_{m/20}}{4})$ =	120.00 MPa
Nominal design stress at room temperature	$f = \frac{R_{m/T}}{4}$ =	175.00 MPa
Nominal design stress in test condition	$f = 1.5 \cdot \frac{R_{m/T}}{4}$ =	262.50 MPa

Geometry

Nominal thickness	en =	30.00 mm
Corrosion allowance	c =	0 mm
External corrosion allowance	ce =	0 mm
Undertolerance	δ =	0 mm
Bolt circle	C =	260.00 mm
Mean gasket diameter	Gmean =	140.00 mm
Adopted thickness (periphery)	e1 =	27.00 mm
Gasket groove depth	g =	0 mm
External corrosion allowance	c'' =	0 mm
Flange external diameter	A =	240.00 mm
Inside diameter of connected flange	B =	142.00 mm
Inside diameter of connected flange (corroded)	B* = B + 2c =	142.00 mm

Gasket parameters

Gasket factor	m =	0
Minimum gasket seating pressure	y =	0.70 MPa
Gasket contact width	w =	5.00 mm
Basic gasket seating width	b0 = w / 2 =	2.50 mm
Effective gasket seating width	b = b0 =	2.50 mm
Diameter of gasket load reaction	G = Gmean =	140.00 mm

Bolt loads

Number of bolts	=	8
Bolt type	=	ISO M16 x 2.00
Bolt hole diameter	=	18.00 mm
Bolt spacing	tBmax =	102.10 mm
Root area of one bolt	=	144.0 mm²
Distance between centre lines of adjacent bolts	δb =	102.10 mm
Total hydrostatic end force	$H = \frac{G^2 \pi P}{4}$ =	61 575 N
Minimum required bolt load for operating condition	Wop = H =	61 575 N
	$H_t = \frac{G^2 \pi P_t}{4}$ =	102 102 N
Minimum required bolt load for the test condition (Loose lap flange #1)	Wt =	102 110 N
Minimum required bolt load for assembly condition	WA =	0 N

Total required cross-sectional area of bolts	$A_{B,\min} = \max\left[\frac{W_A}{f_{B,A}};\frac{W_{op}}{f_B};\frac{W_t}{f_{B,t}}\right]$	=	513.1 mm²
Total cross-sectional area of bolts at the section of least bolt diameter	AB	=	1 152.0 mm²
Maximum bolts area for gasket crush	$A_b\max = \frac{2\pi \cdot y \cdot G \cdot w}{f_{B,A}}$	=	17.6 mm²
Design bolt load for assembly condition	$W = 0{,}5(A_{B,\min} + A_B) f_{B,A}$	=	145 699 N
			AB ≥ AB,min: Ok

Internal pressure

Overpressure due to static head	Ph	=	0 MPa
Calculation pressure	P=Pi+Ph	=	4.00 MPa
Minimum thickness in assembly condition (center)	$e_A = \sqrt{C_F \frac{3(C-G)}{\pi G}(\frac{W}{f_A})} + \max(c,g) + \mathrm{ce} + \delta$	=	26.10 mm
Minimum thickness in assembly condition (periphery)	$e_{A1} = \sqrt{C_F \frac{3(C-G)}{\pi G}(\frac{W}{f_A})} + \mathrm{ce} + \delta$	=	26.10 mm
Minimum thickness in operating condition (center)	$e_P = \sqrt{[\frac{3(3+v)}{32}G^2 + 3C_F(\frac{G}{4} + 2b \cdot m)(C-G)]\frac{P}{f}} + \max(c,g) + \mathrm{ce} + \delta$	=	27.79 mm
Minimum thickness in operating condition (periphery)	$e_{P1} = \sqrt{3C_F(\frac{G}{4} + 2b \cdot m)(C-G)\frac{P}{f}} + \mathrm{ce} + \delta$	=	22.83 mm
Minimum required thickness (periphery)	e1=max(eA1;eP1)	=	26.10 mm
Minimum required thickness	e=max(eA;eP)	=	27.79 mm
			e1 ≥ e1(min): Ok
			en ≥ e: Ok

Maximum allowable pressures (at the top of the vessel)

Maximum allowable test pressure		=	13.10 MPa
Maximum allowable design pressure		=	4.66 MPa

Hydrostatic test

Item or side minimum allowables ratio	Item f0/f	=	1.32653
Coincident design pressure for the maximum pressure load case	Pd	=	4.00 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-1	Pt1 = 1.25·Pd·(Item f0/f)	=	6.63 MPa
Test pressure as per EN13445-5 formula 10.2.3.3.1-2	Pt2 = 1.43·Pd	=	5.72 MPa
Item or side hydrostatic test pressure	Pt=max(Pt1,Pt2)	=	6.63 MPa
Overpressure due to static head in test condition	Pht	=	0.00003 MPa
Calculation pressure	Pc=Pt+Pht	=	6.63 MPa
Minimum thickness in operating condition (center)	$e_P = \sqrt{[\frac{3(3+v)}{32}G^2 + 3C_F(\frac{G}{4} + 2b \cdot m)(C-G)]\frac{P}{f}} + g + \delta$	=	20.82 mm
Minimum thickness in operating condition (periphery)	$e_{P1} = \sqrt{3C_F(\frac{G}{4} + 2b \cdot m)(C-G)\frac{P}{f}} + \delta$	=	17.10 mm
Minimum required thickness	e=eP	=	20.82 mm
			en ≥ e: Ok