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CALCULATION OF A KORBBOGEN FRONT ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
EN 13445 Issue 8 (2003-11)

Object name : Korbogen
Order n° : 12345
Description : Test Horizontaal Vat

Material
Code : EN 10088-2 (5-1995)
Name : X2CrNiMo17132 (1.4404)1%

Temp. (°C)	R_p 20°C (N/mm²)	R^t_p (N/mm²)	R_m 20°C (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
120.000	260.000	191.800	520.000	0.000	0.000

Min (Rpt0.2 / 1.5 ; Rm / 2.4) f,d = 127.867 N/mm²
Rp0.2 / 1.05 f,test = 247.619 N/mm²
Rpt0.2 / 1.05 f,exc = 182.667 N/mm²

Cold spun seamless austenic stainless steel = No

Dimensions
External diameter D_e = 1500.00 mm Base Diameter
Internal diameter D_i = 1486.00 mm
Thickness e_n = 7.00 mm

Allowance: Tol = 0.00 mm
Fab = 0.00 mm
CAInt = 0.00 mm
CAExt = 0.00 mm

Weld joint coefficient z = Test group 1 (z = 1.0)

Inside knuckle radius r = 231.00 mm
Inside dish radius R = 1200.00 mm

Mass front M = 0.00 Kg
Volume front (content) V = 450.17 dm³

CALCULATION OF A NOZZLE ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
ISSUE 8 (2003-11)

Object name : Nozzle 2
Order n° : 12345
Description : Test Horizontaal Vat

General data

Shell type : Head
Nozzle type : Set in

Design pressure	P_d	=	1.000 MPa
Design temperature	t	=	120.00 °C
Weld joint coefficient	z	=	Test group 1 (z = 1.0)

Materials

	Nozzle	**Shell**
Code	EN 10088-2 (5-1995)	EN 10088-2 (5-1995)
Name	X2CrNiMo17132 (1.4404)1%	X2CrNiMo17132 (1.4404)1%

	Temp. (°C)	**R_e 20°C (N/mm²)**	**R^t_e (N/mm²)**	**R_m 20°C (N/mm²)**	**R^t_m (N/mm²)**	**R_{mg} (N/mm²)**
Nozzle	120.0	260.000	191.800	520.000	0.000	0.000
Shell	120.0	260.000	191.800	520.000	0.000	0.000

Nozzle		**N/mm²**	**Shell**		**N/mm²**
f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87	f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87
f,test	Rp0.2 / 1.05	247.62	f,test	Rp0.2 / 1.05	247.62
f,exc	Rpt0.2 / 1.05	182.67	f,exc	Rpt0.2 / 1.05	182.67

Nozzle dimensions

External diameter	d_e	=	350.00 mm	Base diameter
Internal diameter	d_i	=	330.95 mm	
External length	l_{bo}	=	250.00 mm	
Internal length	l_{bi}	=	0.00 mm	
Nominal thickness	e_{nom}	=	9.52 mm	
Allowance	Tol	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	

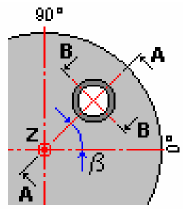
Shell data

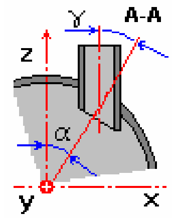
Name : Korbogen
Calculation used : Internal Case 1

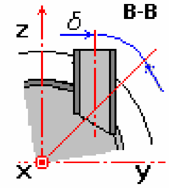
Design pressure	$P_{d,shell}$	=	1.00 MPa	
Design temperature	t_{shell}	=	120.00 °C	
Internal radius	R_{is}	=	1200.00 mm	(9.5-3/4/5/6)
Thickness	e_{shell}	=	7.00 mm	
Allowance	Tol	=	0.00 mm	
	Tol_{fab}	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	

Position and orientation of nozzle

Orientation : Radial.







β = 45.00 °

γ = 0.00 °
α = 7.18 °
x = 150.00 mm

δ = 0.00 °

Calculations

Minimum C7.4.2 thickness	$e_{n,C7.4.2}$	=	1.36 mm	{ = P•D$_e$/(2•f•z + P)}
Assumed shell thickness	e_{cs}	=	7.00 mm	
Design stress	f_{ob}	=	127.87 N/mm²	
Design stress	f_{op}	=	0.00 N/mm²	

			Contrib.	Maximum		
Effective nozzle thickness	e_b	=	9.52	14.00	mm	(F9.4-14)
Actual Nozzle thickness	e_{ab}	=	9.52	21.00	mm	(F9.4-15)
Internal nozzle length	l'_{bi}	=	0.00	28.47	mm	(9.4-44)
External nozzle length	l'_{bo}	=	56.95	56.95	mm	(9.4-43)
Shell reinforcement length	l'_s	=	129.80	129.80	mm	(9.5-30)
Plate width	l'_p	=	0.00	129.80	mm	(9.4-31)
Plate thickness	$e_{a,p}$	=	0.00	7.00	mm	

Contributing shell area	Af_s	=	908.63 mm²	(9.5-20/21)
Contributing nozzle area	Af_b	=	609.07 mm²	(9.5-41/42)

			Long.	Trans.		
Shell pressure area	Ap_s	=	0.00	182722.86	mm²	(9.5-22)
Nozzle pressure area	Ap_b	=	10581.60	10581.60	mm²	(9.5-45)
Oblique pressure area	Ap_φ	=	0.00	0.00	mm²	(9.5-50/52)
Total pressure area	Ap	=	10581.60	193304.46	mm²	

Maximum pressure area	Ap_{max}	=	193304.46 mm²	
Reinforcement	Reinf.	=	1.00	(9.5-7/11/16)
Maximum Pressure	P_{max}	=	1.00 MPa	(9.5-10/12/17)

Conclusion
OK: Nozzle is sufficiently reinforced!
P_{max} (= 1.00 MPa) >= P (= 1.00 MPa)
Miscellaneous results

Minimum distance to shell butt-weld	l_n	=	189.00 mm	(9.4-4)
Maximum distance to shell butt-weld	$l_{n,max}$	=	55.16 mm	
Minimum value for w	w_{min1}	=	150.00 mm	

External pressure Calculation (C8.7.1 & C8.8)

Description	: External Case 1			
Design condition	: Operating			
Design pressure	P_d	=	0.10 MPa	
Design temperature	T	=	120.00 °C	
Safety Factor	S	=	1.50	(8.4.4-1/2)
Modulus of elasticity	E	=	192400.00 N/mm²	
Allowance:	Tol	=	0.00 mm	
	Fab	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	
Nominal elastic limit	σ_{es}	=	153.44 N/mm²	(8.4.2-2/8.4.3-2)
Yield point circumferential stress	P_y	=	1.78 MPa	(8.7.1-1)
Elastic inst. pressure for collapse	P_m	=	7.83 MPa	(8.7.1-2)
Lower bound collapse pressure	P_r	=	0.94 MPa	(F8.5-5)
Design stress (t_d,e_{min})	$f_{e,min}$	=	127.867 N/mm²	
Design stress (t_d,e_a)	$f_{e,nom}$	=	127.867 N/mm²	
Reduced minimum thickness	$e_{min,a}$	=	2.34 mm	
Minimum thickness	e_{min}	=	2.34 mm	
Reduced thickness	e_a	=	7.00 mm	
Thickness	e_n	=	7.00 mm	
Maximum pressure	P_{max}	=	0.63 MPa	

Conclusion: **OK:** $P_d <= P_r/S$ (8.7.1-3)

Miscellaneous results

Value used for R in Calculation	R_{used}	=	1207.00 mm
Minimum e_n using internal calculation	e_{int}	=	1.23 mm

Internal pressure Calculation (C7.4.3)

Description	: Internal Case 1			
Design condition	: Operating			
Design pressure	P_d	=	1.00 MPa	
Design temperature	t_d	=	120.00 °C	

Minimum thickness calculation

Design stress (t_d,e_{min})	$f_{e,min}$	=	127.867 N/mm²	
Required thickness of:				
- end to limit memebrane stress in central part	e_s	=	4.70 mm	(7.5-1)
- knuckle to avoid axisymmetric yielding	e_y	=	6.58 mm	(7.5-2)
- knuckle to avoid plastic buckling	e_b	=	5.69 mm	(7.5-3)
Factor β from procedure 7.5.3.5	β	=	0.70	
Design stress for buckling equation	f_b	=	127.87 N/mm²	
Reduced minimum thickness	$e_{min,a}$	=	6.58 mm	
Minimum thickness	e_{min}	=	6.58 mm	

Conclusion: **OK:** $e_a >= e_{min,a}$

Maximum Pressure calculation

Design stress (t_d,e_n)	$f_{e,nom}$	=	127.867 N/mm²
Reduced thickness	e_a	=	7.00 mm
Thickness	e_{nom}	=	7.00 mm

Maximum pressure at given reduced thickness of:				
- end to limit membrane stress in central part	P_s	=	1.49 MPa	(7.5-6)
- knuckle to avoid axisymmetric yielding	P_y	=	1.07 MPa	(7.5-7)
- knuckle to avoid plastic buckling	P_b	=	1.37 MPa	(7.5-8)
Factor β from procedure 7.5.3.5	β	=	0.70	
Design stress for buckling equation	f_b	=	127.87 N/mm²	
Maximum pressure	P_{max}	=	1.07 MPa	

Conclusion: **OK: Pd <= Pmax**

Miscellaneous results

Reduced mean diameter	D_m	=	1493.00 mm
Maximum diameter opening in knuckle	$d_{i,max}$	=	51.18 mm
Minimum distance knuckle to reduced dish	$\sqrt{(R*e)}$	=	91.65 mm
Maximum length straight cylindrical flange	l_{flange}	=	20.40 mm

CALCULATION OF A NOZZLE ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
ISSUE 8 (2003-11)

Object name : Nozzle 3
Order n° : 12345
Description : Test Horizontaal Vat

General data

Shell type	: Head		
Nozzle type	: Set in		
Design pressure	P_d	=	1.000 MPa
Design temperature	t	=	120.00 °C
Weld joint coefficient	z	=	Test group 1 (z = 1.0)

Materials

	Nozzle	**Shell**	**Plate**
Code **Name**	EN 10088-2 (5-1995) X2CrNiMo17132 (1.4404)1%	EN 10088-2 (5-1995) X2CrNiMo17132 (1.4404)1%	EN 10088-2 (5-1995) X2CrNiMo17132 (1.4404)1%

	Temp. (°C)	**R_e 20°C (N/mm²)**	**R^t_e (N/mm²)**	**R_m 20°C (N/mm²)**	**R^t_m (N/mm²)**	**R_{mg} (N/mm²)**
Nozzle	120.0	260.000	191.800	520.000	0.000	0.000
Shell	120.0	260.000	191.800	520.000	0.000	0.000
Plate	120.0	260.000	191.800	520.000	0.000	0.000

Nozzle		**N/mm²**	**Shell**		**N/mm²**
f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87	f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87
f,test	Rp0.2 / 1.05	247.62	f,test	Rp0.2 / 1.05	247.62
f,exc	Rpt0.2 / 1.05	182.67	f,exc	Rpt0.2 / 1.05	182.67

Plate		**N/mm²**
f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87
f,test	Rp0.2 / 1.05	247.62
f,exc	Rpt0.2 / 1.05	182.67

Nozzle dimensions

External diameter	d_e	=	88.90 mm	Base diameter
Internal diameter	d_i	=	78.90 mm	
External length	l_{bo}	=	20.00 mm	
Internal length	l_{bi}	=	0.00 mm	
Nominal thickness	e_{nom}	=	5.00 mm	
Allowance	Tol	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	

Reinforcing plate dimensions

Plate width	l_p	=	60.00 mm
Plate thickness	$e_{p,nom}$	=	4.00 mm
Allowance	Tol	=	0.00 mm
	CA_{Int}	=	0.00 mm

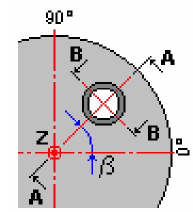
Shell data

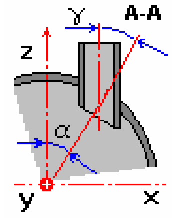
Name	: Korbogen			
Calculation used	: Internal Case 1			
Design pressure	$P_{d,shell}$	=	1.00 MPa	
Design temperature	t_{shell}	=	120.00 °C	
Internal radius	R_{is}	=	1200.00 mm	(9.5-3/4/5/6)
Thickness	e_{shell}	=	7.00 mm	
Allowance	Tol	=	0.00 mm	

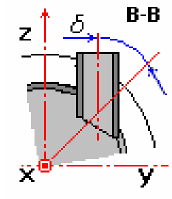
Tol_{fab}	=	0.00	mm
CA_{Int}	=	0.00	mm
CA_{Ext}	=	0.00	mm

Position and orientation of nozzle

Orientation : Radial.







β = 235.00 °

γ = 0.00 °
α = 7.18 °
x = 150.00 mm

δ = 0.00 °

Calculations

Minimum C7.4.2 thickness	$e_{n,C7.4.2}$	=	0.35 mm	$\{ = P \bullet D_e / (2 \bullet f \bullet z + P) \}$
Assumed shell thickness	e_{cs}	=	7.00 mm	
Design stress	f_{ob}	=	127.87 N/mm²	
Design stress	f_{op}	=	127.87 N/mm²	

			Contrib.	Maximum		
Effective nozzle thickness	e_b	=	5.00	14.00	mm	(F9.4-14)
Actual Nozzle thickness	e_{ab}	=	5.00	21.00	mm	(F9.4-15)
Internal nozzle length	l'_{bi}	=	0.00	10.24	mm	(9.4-44)
External nozzle length	l'_{bo}	=	20.00	20.48	mm	(9.4-43)
Shell reinforcement length	l'_s	=	129.80	129.80	mm	(9.5-30)
Plate width	l'_p	=	60.00	129.80	mm	(9.4-31)
Plate thickness	$e_{a,p}$	=	4.00	7.00	mm	

Contributing shell area	Af_s	=	908.63 mm²	(9.5-20/21)
Contributing plate area	Af_p	=	240.00 mm²	
Contributing nozzle area	Af_b	=	135.00 mm²	(9.5-41/42)

			Long.	Trans.		
Shell pressure area	Ap_s	=	0.00	104254.21	mm²	(9.5-22)
Nozzle pressure area	Ap_b	=	1065.15	1065.15	mm²	(9.5-45)
Oblique pressure area	Ap_φ	=	0.00	0.00	mm²	(9.5-50/52)
Total pressure area	Ap	=	1065.15	105319.36	mm²	

Maximum pressure area	Ap_{max}	=	105319.36 mm²	
Reinforcement	Reinf.	=	1.55	(9.5-7/11/16)
Maximum Pressure	P_{max}	=	1.55 MPa	(9.5-10/12/17)

Conclusion

OK: Nozzle is sufficiently reinforced!

P_{max} (= 1.55 MPa) >= P (= 1.00 MPa)

Miscellaneous results

Minimum distance to shell butt-weld	l_n	=	58.45 mm	(9.4-4)
Maximum distance to shell butt-weld	$l_{n,max}$	=	13.15 mm	
Minimum value for w	w_{min1}	=	150.00 mm	

OVERVIEW OF COMBINED OPENINGS

Calculations made with EuroQuip Windows
EN 13445-3 Issue 8 (2003-11)

Nozzle 1	Nozzle 2	Lb	Lb,max	Reinf.	Conclusion	Ok
Nozzle 2	Nozzle 3	301.38	479.69	1.031	Reinf >= 1	Y

CALCULATION OF TWO COMBINED OPENINGS ACCORDING TO EN 13445 Part 3

Calculations made with EuroQuip Windows
ISSUE 8 (2003-11)

General data

Nozzle 1 : Nozzle 2
Nozzle 2 : Nozzle 3

Calculations

Distance between the openings centres	L_b			= 301.38 mm
Maximum distance between openings	$L_{b,max}$	=	479.69 mm	{lso1+lso2+a1+a2}
Distance between openings	x	=	81.30 mm	{Lb-a1-a2}
Angle between pitch and generatrix	Φ	=	0.000°	
Leaning angle nozzle 1	φ_{e1}	=	0.000 rad	
Real ext. radius nozzle 1	a_1	=	175.62 mm	
Leaning angle nozzle 1	φ_{e2}	=	0.00 rad	
Real ext. radius nozzle 2	a_2	=	44.46 mm	

Dimensions

Contributing extern. nozz. height 1	l_{bo1}	=	56.95 mm	
Contributing extern. nozz. height 2	l_{bo2}	=	20.00 mm	

Areas

Shell area	Af_{1s}	=	569.11 mm²	(9.6-7)
Nozzle area 1	Af_{b1}	=	609.07 mm²	
Nozzle area 2	Af_{b2}	=	135.00 mm²	
Plate area 2	Af_{p2}	=	240.00 mm²	
Pressure area in shell	Ap_{1s}	=	180304.57 mm²	(9.6-5/6)
Pressure area in nozzle 1	Ap_{b1}	=	10581.60 mm²	
Pressure area in nozzle 2	Ap_{b2}	=	1065.15 mm²	
Oblique pressure area nozzle 1	Ap_{phi1}	=	0.00 mm²	
Oblique pressure area nozzle 2	Ap_{phi2}	=	0.00 mm²	
Total pressure area	Ap_{tot}	=	191951.32 mm²	
Reinforcement	Reinf.	=	1.031	(9.6-16)

Conclusion

The reinforcement of the combined nozzles Nozzle 2 and Nozzle 3 is satisfactory.

CALCULATION OF A KLOPPER FRONT ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
EN 13445 Issue 8 (2003-11)

Object name : Klopper
Order n° : 12345
Description : Test Horizontaal Vat

Material
Code : EN 10088-2 (5-1995)
Name : X2CrNiMo17132 (1.4404)1%

Temp. (°C)	R_p **20°C** (N/mm²)	R^t_p (N/mm²)	R_m **20°C** (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
120.000	260.000	191.800	520.000	0.000	0.000

Min (Rpt0.2 / 1.5 ; Rm / 2.4)	f,d	=	127.867 N/mm²
Rp0.2 / 1.05	f,test	=	247.619 N/mm²
Rpt0.2 / 1.05	f,exc	=	182.667 N/mm²

Cold spun seamless austenic stainless steel = No

Dimensions

External diameter	D_e	=	1500.00 mm Base Diameter
Internal diameter	D_i	=	1480.28 mm
Thickness	e_n	=	9.86 mm
Allowance:	Tol	=	0.00 mm
	Fab	=	0.00 mm
	CAInt	=	0.00 mm
	CAExt	=	0.00 mm
Weld joint coefficient	z	=	Test group 1 (z = 1.0)
Inside knuckle radius	r	=	150.00 mm
Inside dish radius	R	=	1500.00 mm
Mass front	M	=	0.00 Kg
Volume front (content)	V	=	350.37 dm³

External pressure Calculation (C8.7.1 & C8.8)

Description	: External Case 1			
Design condition	: Operating			
Design pressure	P_d	=	0.10 MPa	
Design temperature	T	=	120.00 °C	
Safety Factor	S	=	1.50	(8.4.4-1/2)
Modulus of elasticity	E	=	192400.00 N/mm²	
Allowance:	Tol	=	0.00 mm	
	Fab	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	
Nominal elastic limit	σ_{es}	=	153.44 N/mm²	(8.4.2-2/8.4.3-2)
Yield point circumferential stress	P_y	=	2.00 MPa	(8.7.1-1)
Elastic inst. pressure for collapse	P_m	=	9.93 MPa	(8.7.1-2)
Lower bound collapse pressure	P_r	=	1.10 MPa	(F8.5-5)
Design stress (t_d,e_{min})	$f_{e,min}$	=	127.867 N/mm²	
Design stress (t_d,e_a)	$f_{e,nom}$	=	127.867 N/mm²	
Reduced minimum thickness	$e_{min,a}$	=	2.92 mm	
Minimum thickness	e_{min}	=	2.92 mm	
Reduced thickness	e_a	=	9.86 mm	
Thickness	e_n	=	9.86 mm	
Maximum pressure	P_{max}	=	0.73 MPa	

Conclusion: **OK:** $P_d \le P_r/S$ (8.7.1-3)

Miscellaneous results

Value used for R in Calculation	R_{used}	=	1509.86 mm
Minimum e_n using internal calculation	e_{int}	=	1.85 mm

Internal pressure Calculation (C7.4.3)

Description	: Internal Case 1			
Design condition	: Operating			
Design pressure	P_d	=	1.00 MPa	
Design temperature	t_d	=	120.00 °C	

Minimum thickness calculation

Design stress (t_d,e_{min})	$f_{e,min}$	=	127.867 N/mm²	
Required thickness of:				
- end to limit memebrane stress in central part	e_s	=	5.88 mm	(7.5-1)
- knuckle to avoid axisymmetric yielding	e_y	=	9.86 mm	(7.5-2)
- knuckle to avoid plastic buckling	e_b	=	8.54 mm	(7.5-3)
Factor β from procedure 7.5.3.5	β	=	0.89	
Design stress for buckling equation	f_b	=	127.87 N/mm²	
Reduced minimum thickness	$e_{min,a}$	=	9.86 mm	
Minimum thickness	e_{min}	=	9.86 mm	

Conclusion: **OK:** $e_a \ge e_{min,a}$

Maximum Pressure calculation

Design stress (t_d,e_n)	$f_{e,nom}$	=	127.867 N/mm²
Reduced thickness	e_a	=	9.86 mm
Thickness	e_{nom}	=	9.86 mm

Maximum pressure at given reduced thickness of:				
- end to limit membrane stress in central part	P_s	=	1.68 MPa	(7.5-6)
- knuckle to avoid axisymmetric yielding	P_y	=	1.00 MPa	(7.5-7)
- knuckle to avoid plastic buckling	P_b	=	1.24 MPa	(7.5-8)
Factor β from procedure 7.5.3.5	β	=	0.89	
Design stress for buckling equation	f_b	=	127.87 N/mm²	
Maximum pressure	P_{max}	=	1.00 MPa	

Conclusion: **OK: Pd <= Pmax**

Miscellaneous results

Reduced mean diameter	D_m	=	1490.14 mm
Maximum diameter opening in knuckle	$d_{i,max}$	=	0.00 mm
Minimum distance knuckle to reduced dish	$\sqrt{(R*e)}$	=	121.61 mm
Maximum length straight cylindrical flange	l_{flange}	=	24.16 mm

CALCULATION OF A CYLINDRICAL SHELL ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
EN 13445 Issue 8 (2003-11)

Object name : Shell
Order n° : 12345
Description : Test Horizontaal Vat

Material
Code : EN 10088-2 (5-1995)
Name : X2CrNiMo17132 (1.4404)1%

Temp. (°C)	**R_p 20°C (N/mm²)**	**R^t_p (N/mm²)**	**R_m 20°C (N/mm²)**	**R^t_m (N/mm²)**	**R_{mg} (N/mm²)**
120.000	260.000	191.800	520.000	0.000	0.000

Min (Rpt0.2 / 1.5 ; Rm / 2.4)	f,d	=	127.867 N/mm²
Rp0.2 / 1.05	f,test	=	247.619 N/mm²
Rpt0.2 / 1.05	f,exc	=	182.667 N/mm²

Dimensions

External diameter	D_e	=	1500.00 mm Base Diameter
Internal diameter	D_i	=	1489.19 mm
Length	Length	=	5000.00 mm
Thickness	e_n	=	5.40 mm
Allowance:	Tol	=	0.00 mm
	Tol_{fab}	=	0.00 mm
	CA_{Int}	=	0.00 mm
	CA_{Ext}	=	0.00 mm
Weld joint coefficient	z	=	Test group 1 (z = 1.0)
Mass cylinder	M	=	0.00 Kg
Volume cylinder (content)	V	=	8708.89 dm³

CALCULATION OF A NOZZLE ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
ISSUE 8 (2003-11)

Object name : Nozzle 1
Order n° : 12345
Description : Test Horizontaal Vat

General data

Shell type	: Cylinder		
Nozzle type	: Set in		
Design pressure	P_d	=	1.000 MPa
Design temperature	t	=	120.00 °C
Weld joint coefficient	z	=	Test group 1 (z = 1.0)

Materials

	Nozzle	Shell
Code **Name**	EN 10088-2 (5-1995) X2CrNiMo17132 (1.4404)1%	EN 10088-2 (5-1995) X2CrNiMo17132 (1.4404)1%

	Temp. (°C)	R_e 20°C (N/mm²)	R^t_e (N/mm²)	R_m 20°C (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
Nozzle	120.0	260.000	191.800	520.000	0.000	0.000
Shell	120.0	260.000	191.800	520.000	0.000	0.000

Nozzle		N/mm²	**Shell**		N/mm²
f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87	f,d	Min (Rpt0.2 / 1.5 ; Rm / 2.	127.87
f,test	Rp0.2 / 1.05	247.62	f,test	Rp0.2 / 1.05	247.62
f,exc	Rpt0.2 / 1.05	182.67	f,exc	Rpt0.2 / 1.05	182.67

Nozzle dimensions

External diameter	d_e	=	150.00 mm	Base diameter
Internal diameter	d_i	=	0.00 mm	
External length	l_{bo}	=	38.78 mm	
Internal length	l_{bi}	=	0.00 mm	
Nominal thickness	e_{nom}	=	0.00 mm	
Allowance	Tol	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	

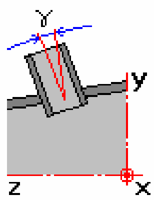
Shell data

Name	: Shell			
Calculation used	: Internal Case 1			
Design pressure	$P_{d,shell}$	=	1.00 MPa	
Design temperature	t_{shell}	=	120.00 °C	
Internal radius	R_{is}	=	744.60 mm	(9.5-3/4/5/6)
Thickness	e_{shell}	=	5.40 mm	
Allowance	Tol	=	0.00 mm	
	Tol_{fab}	=	0.00 mm	
	CA_{Int}	=	0.00 mm	
	CA_{Ext}	=	0.00 mm	

Position and orientation of nozzle

Orientation : Radial.

XX:

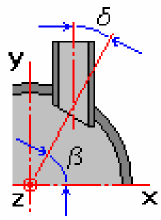


γ = 0.00 °

y = 0.00 mm

z = 2500.00 mm

YY:



x = 100.00 mm

β = 82.28 °

δ = 0.00 °

Calculations

Minimum C7.4.2 thickness	$e_{n,C7.4.2}$	=	0.58	mm	{ $= P \bullet D_e / (2 \bullet f \bullet z + P)$ }
Assumed shell thickness	e_{cs}	=	5.40	mm	
Design stress	f_{ob}	=	127.87	N/mm²	
Design stress	f_{op}	=	0.00	N/mm²	

			Contrib.	Maximum		
Effective nozzle thickness	e_b	=	10.81	10.81	mm	(F9.4-14)
Actual Nozzle thickness	e_{ab}	=	74.00	16.21	mm	(F9.4-15)
Internal nozzle length	l'_{bi}	=	0.00	19.39	mm	(9.4-44)
External nozzle length	l'_{bo}	=	38.78	38.78	mm	(9.4-43)
Shell reinforcement length	l'_s	=	89.86	89.86	mm	(9.5-30)
Plate width	l'_p	=	0.00	89.86	mm	(9.4-31)
Plate thickness	$e_{a,p}$	=	0.00	5.40	mm	

Contributing shell area	Af_s	=	485.47	mm²	(9.5-20/21)
Contributing nozzle area	Af_b	=	477.42	mm²	(9.5-41/42)

			Long.	Trans.		
Shell pressure area	Ap_s	=	122753.61	61201.86	mm²	(9.5-22)
Nozzle pressure area	Ap_b	=	44.19	44.19	mm²	(9.5-45)
Oblique pressure area	Ap_φ	=	0.00	0.00	mm²	(9.5-50/52)
Total pressure area	Ap	=	122797.79	61246.05	mm²	

Maximum pressure area	Ap_{max}	=	122797.79	mm²	
Reinforcement	Reinf.	=	1.00		(9.5-7/11/16)
Maximum Pressure	P_{max}	=	1.00	MPa	(9.5-10/12/17)

Conclusion

WARNING: Nozzle is NOT sufficiently reinforced!

MAWP = 1.00 MPa < P (= 1.00 MPa)

Miscellaneous results

Minimum distance to shell butt-weld	l_n	=	0.00	mm	(9.4-4)
Maximum distance to shell butt-weld	$l_{n,max}$	=	0.00	mm	
Minimum value for w	w_{min1}	=	17.97	mm	
Minimum value for w	w_{min2}	=	0.00	mm	
Minimum value for w	w_{min3}	=	0.00	mm	

External pressure (EN 13445 Part 3 Chapter 8.5)

Description	: External Case 1				
Design condition	: Operating				
Design pressure	P_d	=	0.10	MPa	
Design temperature	t_d	=	120	°C	
Safety factor	S	=	1.50		(8.4.4-1/2)
Nominal thickness	e_n	=	5.40	mm	
Reduced Thickness	e_a	=	5.40	mm	
Mean shell radius	R	=	747.30	mm	
Unsupported length left side of shell	h1	=	0.00	mm	
Unsupported length right side of shell	h2	=	0.00	mm	
Unsupported length cylinder	L	=	2000.00	mm	
Modulus of Elasticity shell	E	=	192400.00	N/mm²	
Nominal elastic limit shell	σ_e	=	153.44	N/mm²	(8.4.2-1/8.4.3-1)

Ring Material

Code : EN 10088-2 (5-1995)
Name : X2CrNiMo17132 (1.4404)1%

Temp. (°C)	**R_p 20°C (N/mm²)**	**R^t_p (N/mm²)**	**R_m 20°C (N/mm²)**	**R^t_m (N/mm²)**	**R_{mg} (N/mm²)**
120.000	260.000	191.800	520.000	0.000	0.000

Ring variables

Number of stiffener rings		=	2		
Offset of first ring		=	1495.00	mm	
Spacing between rings		=	2010.00	mm	
Ring height	h_r	=	50.00	mm	
Ring width at shell	w_i	=	10.00	mm	
Ring Area	A_s	=	500.00	mm	
Average panellength	L_s	=	1752.50	mm	(8.5.3-6/7/8/9)
Distance between heavy stiffeners	L_H	=	5000.00	mm	(8.5.3-10/11/12)
Radius centroid ring	R_s	=	775.00	mm	
Radius top ring	R_f	=	800.00	mm	
Modulus of Elasticity ring	E_r	=	192400.00	N/mm²	
Nominal elastic limit ring	σ_{es}	=	153.44	N/mm²	(8.4.2-2/8.4.3-2)

Cylinder thickness according to C8.5.2.2

Calculation variable	Z	=	1.174		(8.5.2-7)
No. of Circumferential waves	n_{cyl}	=	6		(F8.5-4)
Mean elas. circ. strain at collapse	ε	=	0.000216		(8.5.2-6)
Calculation variable	δ	=	0.020		(8.5.3-20)
Calculation variable	N	=	1.000		(8.5.3-21)
Calculation variable	G = 0 : See C8.5.3.4 note 2 (L > 3 * sqrt(R*e_a))!				
Calculation variable	B	=	1.029		(8.5.3-18)
Modified area of stiffener	A_m	=	464.895	mm²	(8.5.3-17)
Calculation variable	γ	=	0.375		(8.5.3-16)
Yield point circ. stress	P_y	=	1.109	MPa	(8.5.2-4/8.5.3-15)
Elastic instability P for collapse	P_m	=	0.300	MPa	(8.5.2-5)
Lower bound collapse pressure	P_r	=	0.150	MPa	(F8.5-5)
Maximum pressure (P_r/S)	P_{max}	=	0.100	MPa	(8.5.2-8)
Minimum thickness	e_{min}	=	5.403	mm	

Reduced minimum thickness	$e_{min,a}$	=	5.403 mm	
Minimum e_n using internal calculation	e_{int}	=	0.59 mm	(7.4-1/2)

Conclusion: OK: $P_d \leq P_{max}$

Maximum theoretical elastic instability of a light stiffener (C8.5.3.6.2)

Second Moment of Area Stiffener I_s = 104166.667 mm^4

	n=2	n=3	n=4	n=5	n=6		
x	0.03	0.07	0.13	0.20	0.28		(8.5.3-35)
u	26.53	26.53	26.53	26.53	26.53		(8.5.3-36)
Y1	1.56	1.56	1.56	1.56	1.56		(T8.5-3)
Y2	1.20	1.20	1.20	1.20	1.20		(T8.5-3)
Y3	0.71	0.71	0.71	0.71	0.71		(T8.5-3)
Le	102.10	100.64	98.55	95.84	92.53	mm	(8.5.3-34)
Ae	1096.56	1088.00	1075.80	1059.94	1040.60	mm^2	(8.5.3-30)
Xe	15.65	15.75	15.90	16.09	16.34		(8.5.3-27)
Ie	317926.41	316502.91	314437.59	311682.02	308207.93	mm^4	(8.5.3-26)
β	8.764E-004	7.041E-005	1.221E-005	3.166E-006	1.054E-006		(8.5.3-25)
Pg	1.57	0.77	1.26	1.97	2.84	MPa	(8.5.3-24)
Pg/Sf*S	0.87	0.43	0.70	1.10	1.58	MPa	(8.5.3-31)

Critical instability pressure $P_{g,crit}$ = 0.429 MPa

Conclusion: OK: $P_d \leq P_{g,crit}$ (8.5.3-31)

Maximum stresses in the light stiffeners (C8.5.3.6.4)

Circumferential yield pressure ring P_{ys} = 3.222 MPa (8.5.3-47)

	n=2	n=3	n=4	n=5	n=6	
max_d	40.19	40.09	39.94	39.75	39.51	(8.5.3-49)
σ_s	28.68	134.01	137.13	131.81	128.97	(8.5.3-46)

Critical stiffener stress $\sigma_{s,crit}$ = 137.133 N/mm^2

Conclusion: OK: $\sigma_{s,crit} \leq \sigma_{es}$ (8.5.3-50)

Stiffener tripping for an external flat bar stiffener (C8.5.3.8.2)

Radial height ring / shell radius	d/R	=	0.067	
Value for $(\sigma_i/E)/(d/e_w)^2$		=	1.1400	
Instability Stress sideways tripping	σ_i	=	8773.44 N/mm^2	(T8.5-5)

Conclusion: OK: $\sigma_i/4 > (P*\sigma_{es}/P_{ys})$ (8.5.3-65)

Internal pressure (EN 13445 Part 3 Chapter 7.4.2)

Description	: Internal Case 1				
Design condition	: Operating				
Design pressure	P_d	=	1.00	MPa	
Design temperature	T	=	120	°C	
Weld Joint coefficient	z	=	Test group 1 (z = 1.0)		
Allowance:	Tol	=	0.00	mm	
	CA_{Int}	=	0.00	mm	
	CA_{Ext}	=	0.00	mm	
Design stress (T,d_{min})	f_{dmin}	=	127.867	N/mm²	
Design stress (T,d_d)	f	=	127.867	N/mm²	
Reduced minimum thickness	$e_{min,a}$	=	5.84	mm	(7.4-1/2)
Minimum thickness	e_{min}	=	5.84	mm	
Reduced thickness	e_a	=	5.40	mm	
Thickness	e	=	5.40	mm	
Maximum pressure	P_{max}	=	0.92	MPa	(7.4-6)

Conclusion: **NOT OK:** $e < e_{min}$

Miscellaneous results

Reduced mean diameter	D_m	=	1494.60	mm

CALCULATION OF A FLANGE ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
EN 13445 Issue 8 (2003-11)

Object name : DIN Flange
Order n° : 12345
Description : Test Horizontaal Vat

Flange type : **Integral Smooth Bore Hub**

Flange material :
15 NiCuMoNb 5 (1.6368) VdTUeV - 377/1 06.86

Temp. (°C)	R_p 20°C (N/mm²)	R^t_p (N/mm²)	R_m 20°C (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
80.000	430.000	409.000	580.000	0.000	0.000

Min (Rpt0.2 / 1.5 ; Rm / 2.4) f,d = 241.667 N/mm²
Rp0.2 / 1.05 f,test = 409.524 N/mm²
Rpt0.2 / 1.05 f,exc = 389.524 N/mm²

Dimensions of the flange

Calculation method	: Integral method			
Nominal outside diameter	A_{nom}	=	265.000	mm
Nominal inside diameter	B_{nom}	=	104.300	mm
Thickness of the flange	e_n	=	33.000	mm
Nominal thickness of hub at small end	$g_{0,nom}$	=	5.000	mm
Nominal thickness of hub at flange side	$g_{1,nom}$	=	22.850	mm
Height of hub	h	=	42.000	mm
Minimum radius	r_{min}	=	5.000	mm
Length of cylindrical part	l	=	0.000	mm

Allowance (Inside diameter/g_0/g_1)

Tolerance	δ_e	=	0.000	mm
Internal corrosion	c_{int}	=	0.000	mm
External corrosion	c_{ext}	=	0.000	mm

Allowance (Outside diameter)

Tolerance	δ_e	=	0.000	mm
Corrosion	c_{ext}	=	0.000	mm

Gasket data

Group	: User defined gasket				
Seating/Facing	: Both flat				
Gaket factor	m	=	4.000		
Seating pressure	y	=	60.600	N/mm²	
Contact width	w	=	20.000	mm	
Basic seating width	b_o	=	10.000	mm	(11.5-1/2)
Effective seating width	b	=	7.969	mm	(11.5-3)
Outside diameter	G_o	=	162.000	mm	
Diameter reaction point	G	=	146.062	mm	(C11.5-2/11.6-3)

Bolts material :
X22CrMoV121VRm900(1.4923) DIN 17240 (1976-07)

Temp. (°C)	R_p 20°C (N/mm²)	R^t_p (N/mm²)	R_m 20°C (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
120.000	700.000	646.111	900.000	0.000	0.000

$f_{b,A}$ = Rp0.2 / 1.05 = 375.000 N/mm²
$f_{b,P}$ = Min (Rpt0.2 / 1.5 ; Rm / 2.4) = 375.000
N/mm²

Design temperature	T	=	120.000 °C
Pitch circle diameter	C	=	210.000 mm
Number of bolts	n	=	12
Diameter	d_b	=	20.000 mm
Root diameter	d_r	=	17.654 mm
Bolt surface	a_B	=	244.794 mm²
Total bolt surface ($n \cdot a_b$)	A_B	=	2937.528 mm²
Minimum seating load	W_A	=	221595.348 N
Seating load	$W_{A,used}$	=	221595.348 N
Minimum operating load	W_{op}	=	18403.723 N
Operating load	$W_{op,used}$	=	18403.723 N
Minimum design load	W	=	661584.120 N
Design load	W_{used}	=	661584.120 N

Shell material :
X2CrNiMo17132 (1.4404)1% EN 10088-2 (5-1995)

Temp. (°C)	R_p 20°C (N/mm²)	R^t_p (N/mm²)	R_m 20°C (N/mm²)	R^t_m (N/mm²)	R_{mg} (N/mm²)
120.000	260.000	191.800	520.000	0.000	0.000

$f_{s,A}$ = Rp0.2 / 1.05 = 127.867 N/mm²
$f_{s,P}$ = Min (Rpt0.2 / 1.5 ; Rm / 2.4) = 127.867 N/mm²

Shell data

Design temperature	T	=	120.000 °C
Outside diameter	$D_{e.nom}$	=	7390.000 mm
Thickness	$e_{s.n}$	=	10.000 mm
Inside diameter	D_{nom}	=	7370.000 mm

Allowance

Tolerance	δ_e	=	0.000 mm
Internal corrosion	c_{int}	=	0.000 mm
External corrosion	c_{ext}	=	0.000 mm

DesignCondition : **Operating**
Description : **Internal 1**

Design pressure	P_d	=	0.400 MPa
Design temperature	T	=	80.000 °C

InternalCalculation in **corroded** condition.

Flange

Reduced outside diameter	A	=	265.000 mm
Reduced inside diameter	B	=	104.300 mm
Ratio flange diameters	K	=	2.541
Reduced inside diameter of shell	D	=	7370.000 mm
Reduced hubthickness at small end	g_0	=	5.000 mm
Reduced hubthickness at flange	g_1	=	22.850 mm
$\sqrt{(B \cdot g_0)}$	l_0	=	22.836 mm
Minimum thickness seating	$e_{min,A}$	=	30.366 mm
Minimum thickness operating	$e_{min,P}$	=	1.078 mm
Minimum thickness	e_{min}	=	30.366 mm

Bolts

Bolt pitch correction factor	C_F	=	1.000 -
Bolt design stress seating	$f_{B,A}$	=	375.000 N/mm²
Bolt design stress operating	f_B	=	375.000 N/mm²
Minimum number of bolts	n_{min}	=	4
Number of bolts	n	=	12
Minimum total bolt surface	$A_{B,min}$	=	590.921 mm²
Total bolt surface $(n \cdot a_B)$	A_B	=	2937.528 mm²

Sufficient number of bolts !

Loads on bolts

Minimum bolt seating load	W_A	=	221595.348 N
Seating load	$W_{A,used}$	=	221595.348 N
Minimum bolt operating load	W_{op}	=	18403.723 N
Operating load	$W_{op,used}$	=	18403.723 N
Minimum bolt design load	W	=	661584.120 N
Design load	W_{used}	=	661584.120 N

Loads on flange				**Distance**		
Gasket compressionload	H_G	=	11701.405 N	h_G	=	31.969 mm
Force by pressure across gasket (at (B+G)/2)	H_T	=	3284.740 N	h_T	=	42.409 mm
Force by pressure on flange face (at B)	H_D	=	3417.578 N	h_D	=	41.425 mm

Bending moments on flange

Momentum gasket seating	M_A	=	21150142.827 Nmm
Momentum operating	M_{op}	=	654958.776 Nmm

Stress(es) in gasket seating condition

Tangential flange stress	$\sigma_{\theta,A}$	=	147.751 N/mm²
Radial flange stress	$\sigma_{r,A}$	=	193.268 N/mm²
Longitudinal hub stress	$\sigma_{H,A}$	=	199.281 N/mm²
Allowable longitudinal hub stress		=	260.000 N/mm²
Flange allowable stress in seating condition	f_A	=	241.667 N/mm²
Shell/flange allowable stress seat. condition	$f_{H,A}$	=	173.333 N/mm²
Stress factor	k	=	1.000

Conclusion

$k \cdot |\sigma_{\theta,A}|$ (147.751 N/mm²) <= f_A (241.667 N/mm²) : **Stress OK !**
$k \cdot |\sigma_{r,A}|$ (193.268 N/mm²) <= f_A (241.667 N/mm²) : **Stress OK !**
$k \cdot |\sigma_{H,A}|$ (199.281 N/mm²) <= $1.5 \cdot \min(f_A, f_{H,A})$ (260.000 N/mm²) : **Stress OK !**
$k \cdot (|\sigma_{H,A}|+|\sigma_{\theta,A}|)/2$ (173.516 N/mm²) <= f_A (241.667 N/mm²) : **Stress OK !**
$k \cdot (|\sigma_{H,A}|+|\sigma_{r,A}|)/2$ (196.274 N/mm²) <= f_A (241.667 N/mm²) : **Stress OK !**

Stress(es) in operating condition

Tangential flange stress	$\sigma_{\theta,P}$	=	4.575 N/mm²
Radial flange stress	$\sigma_{r,P}$	=	5.985 N/mm²
Longitudinal hub stress	$\sigma_{H,P}$	=	6.171 N/mm²
Allowable longitudinal hub stress		=	191.800 N/mm²
Flange allowable stress in operating condition	f_P	=	241.667 N/mm²
Shell/flange allowable stress op. condition	$f_{H,p}$	=	127.867 N/mm²
Stress factor	k	=	1.000

Conclusion

$k \cdot |\sigma_{\theta,P}|$ (4.575 N/mm²) <= f_P (241.667 N/mm²) : Stress OK !

$k \cdot |\sigma_{r,P}|$ (5.985 N/mm²) <= f_P (241.667 N/mm²) : Stress OK !

$k \cdot |\sigma_{H,P}|$ (6.171 N/mm²) <= $1.5 \cdot \min(f_P, f_{H,P})$ (191.800 N/mm²) : Stress OK !

$k \cdot (|\sigma_{H,P}|+|\sigma_{\theta,P}|)/2$ (5.373 N/mm²) <= f_P (241.667 N/mm²) : Stress OK !

$k \cdot (|\sigma_{H,P}|+|\sigma_{r,P}|)/2$ (6.078 N/mm²) <= f_P (241.667 N/mm²) : Stress OK !

Factors

β_T	=	1.327173	β_U	=	2.430636	β_Y	=	2.211879
β_F	=	0.530966	β_V	=	0.023836	λ	=	1.948910

Hub stress correction factor for "Integral" flanges φ = 1.000000

Conclusion

The design according to EN 13445-3 Clause 11 is satisfactory.

CALCULATION OF SADDLES ACCORDING TO EN 13445 Part 3

Calculation made with EuroQuip Windows
EN 13445-3 Issue 8 (2003-11)

Object name : Saddles 1
Order n° :
Description : Saddle Suports

Type of saddle supports: Type A
Warning: Vessel not symmetrical!

Saddle Dimensions

Axial width of saddle	b_1	=	20.00 mm
Included angle of saddle support	δ	=	120.00°

Saddle	No.	x (mm)	l_1 (mm)	l_2 (mm)	Plate	δ_2 (°)	b_2 (mm)	e_2 (mm)
LeftSaddle	1	250.00	440.55	4500.00	No			
RightSaddle	2	4750.00	4500.00	501.54	No			

Cylinder Data

Name		=	Shell
Thickness	e_{nom}	=	5.40 mm
Internal Diameter	D_i	=	1489.19 mm
Allowance	Tol	=	0.00 mm
	Tol_{fab}	=	0.00 mm
	CA_{Int}	=	0.00 mm
	CA_{Ext}	=	0.00 mm
Length	L	=	5000.00 mm

Critical cylinder design condition

Design condition	: Operating		
Design pressure	P_d	=	1.00 MPa
Design temperature	t	=	120 °C

Maximum external cylinder pressure

Maximum External Pressure	P_{max}	=	0.10 MPa
External Pressure	P_{ext}	=	0.10 MPa

Left Front

Name		=	Klopper
Type		=	Klopper
Thickness	e_{nom}	=	9.86 mm
Internal height	H_i	=	285.82 mm
Allowance	Tol	=	0.00 mm
	Tol_{fab}	=	0.00 mm
	CA_{Int}	=	0.00 mm
	CA_{Ext}	=	0.00 mm

Right Front

Name		=	Korbogen
Type		=	Korbbogen
Thickness	e_{nom}	=	7.00 mm
Internal height	H_i	=	377.31 mm
Allowance	Tol	=	0.00 mm
	Tol_{fab}	=	0.00 mm
	CA_{Int}	=	0.00 mm
	CA_{Ext}	=	0.00 mm

Saddles Design Condition (EN 13445-3 C16.8)

Description : External
Design condition : Operating

Design pressure	P_d	=	-0.10 MPa
Design temperature	t	=	120 °C
Mamimum external pressure acc. C8	P_{max}	=	0.10 MPa
Vessel weight	W_{ves}	=	5000.00 kg
Fluid weight	W_F	=	5000.00 kg
Total weight	W	=	10000.00 kg

Global loads (Chapter 16.14)

Modulus of elasticity	E	=	192400.000 N/mm²
Calculation factor	K	=	5.933
Calculation factor	α	=	0.550
Calculation factor	Δ	=	0.531
Max. permitted compr. long. stress	$\sigma_{c,all}$	=	81.546 N/mm²
Maximum tensile force	$F_{t,max}$	=	3.507E+006 N
Maximum compressive force	$F_{c,max}$	=	2.236E+006 N
Maximum bending moment	M_{max}	=	8.354E+008 Nmm

Saddle Forces, Moments and Shear Forces

Load per unit vessel	q	=	18.03 N/mm
Moment at edge	M_0	=	1247794.54 Nmm

Saddle	F_i (N)	Q_i (mm)	M_i (Nmm)	M_{ij} (Nmm)
2	4.91E+004	4.01E+004	1.02E+006	4.34E+007
1	4.91E+004	4.10E+004	5.01E+005	4.64E+007

Load limits between the saddles

Saddle	Instability	
2	0.000	OK!
1	0.000	OK!

Load limit at the saddles

Saddle	e_a (mm)	b_1 (mm)	δ_1 (°)
2	5.84	20.00	120.00
1	5.84	20.00	120.00

Saddle	γ	β	K_3	K_4	K_5	K_6	K_7	K_8	K_9	K_{10}
2	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903
1	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903

Saddle	$K_{1,2}$	$K_{1,3}$	K_2	$v_{1,2}$	$v_{1,3}$	$v_{2,2}$	$v_{2,3}$
2	1.494	0.407	1.250	-0.015	-1.903	-0.040	-0.080
1	1.494	0.407	1.250	-0.015	-1.903	-0.040	-0.080

Saddle	$\sigma_{b,all,2}$ (N/mm²)	$\sigma_{b,all,3}$ (N/mm²)	$F_{2,max}$ (N)	$F_{3,max}$ (N)	
2	238.80	65.00	113509.03	100283.37	Ok: F_i <= $\min(F_{2,max}; F_{3,max})$
1	238.81	65.00	113513.58	100283.37	Ok: F_i <= $\min(F_{2,max}; F_{3,max})$

Saddle	M_{max} (Nmm)	F_{max} (N)	Q_{max} (N)	F_{eq} (N)
2	8.35E+008	2.24E+006	1.19E+006	3.22E+004
1	8.35E+008	2.24E+006	1.19E+006	3.22E+004

Saddle	Instability (<= 1.0)	
2	0.828	OK!
1	0.828	OK!

Saddles Design Condition (EN 13445-3 C16.8)

Description : ConditionB
Design condition : Operating

Design pressure	P_d	=	0.00 MPa
Design temperature	t	=	120 °C
Mamimum external pressure acc. C8	P_{max}	=	0.10 MPa
Vessel weight	W_{ves}	=	5000.00 kg
Fluid weight	W_F	=	5000.00 kg
Total weight	W	=	10000.00 kg

Global loads (Chapter 16.14)

Modulus of elasticity	E	=	192400.000 N/mm²
Calculation factor	K	=	5.933
Calculation factor	α	=	0.550
Calculation factor	Δ	=	0.531
Max. permitted compr. long. stress	$\sigma_{c,all}$	=	81.546 N/mm²
Maximum tensile force	$F_{t,max}$	=	3.507E+006 N
Maximum compressive force	$F_{c,max}$	=	2.236E+006 N
Maximum bending moment	M_{max}	=	8.354E+008 Nmm

Saddle Forces, Moments and Shear Forces

Load per unit vessel	q	=	18.03 N/mm
Moment at edge	M_0	=	1247794.54 Nmm

Saddle	F_i (N)	Q_i (mm)	M_i (Nmm)	M_{ij} (Nmm)
2	4.91E+004	4.01E+004	1.02E+006	4.34E+007
1	4.91E+004	4.10E+004	5.01E+005	4.64E+007

Load limits between the saddles

Saddle	K12	Strength (N/mm²)	f_{max}	Instability	
2	0.00	0.00		0.000	OK!
1	0.00	0.00		0.000	OK!

Load limit at the saddles

Saddle	e_a (mm)	b_1 (mm)	δ_1 (°)
2	5.84	20.00	120.00
1	5.84	20.00	120.00

Saddle	γ	β	K_3	K_4	K_5	K_6	K_7	K_8	K_9	K_{10}
2	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903
1	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903

Saddle	$K_{1,2}$	$K_{1,3}$	K_2	$v_{1,2}$	$v_{1,3}$	$v_{2,2}$	$v_{2,3}$
2	1.499	0.441	1.250	-0.015	-1.903	-0.001	0.000
1	1.499	0.441	1.250	-0.015	-1.903	-0.000	0.000

Saddle	$\sigma_{b,all,2}$ (N/mm²)	$\sigma_{b,all,3}$ (N/mm²)	$F_{2,max}$ (N)	$F_{3,max}$ (N)	
2	239.62	70.54	113898.99	108838.27	Ok: F_i <= $\min(F_{2,max}; F_{3,max})$
1	239.63	70.54	113900.66	108838.27	Ok: F_i <= $\min(F_{2,max}; F_{3,max})$

Saddle	M_{max} (Nmm)	F_{max} (N)	Q_{max} (N)	F_{eq} (N)
2	8.35E+008	2.24E+006	1.19E+006	3.22E+004
1	8.35E+008	2.24E+006	1.19E+006	3.22E+004

Saddle	Instability (<= 1.0)	
2	0.828	OK!
1	0.828	OK!

Saddles Design Condition (EN 13445-3 C16.8)

Description	: ConditionA			
Design condition	: Operating			

Design pressure	P_d	=	1.00	MPa
Design temperature	t	=	120	°C
Mamimum external pressure acc. C8	P_{max}	=	0.10	MPa
Vessel weight	W_{ves}	=	2500.00	kg
Fluid weight	W_F	=	2500.00	kg
Total weight	W	=	5000.00	kg

Global loads (Chapter 16.14)

Modulus of elasticity	E	=	192400.000	N/mm²
Calculation factor	K	=	5.933	
Calculation factor	α	=	0.550	
Calculation factor	Δ	=	0.531	
Max. permitted compr. long. stress	$\sigma_{c,all}$	=	81.546	N/mm²
Maximum tensile force	$F_{t,max}$	=	3.507E+006	N
Maximum compressive force	$F_{c,max}$	=	2.236E+006	N
Maximum bending moment	M_{max}	=	8.354E+008	Nmm

Saddle Forces, Moments and Shear Forces

Load per unit vessel	q	=	9.01	N/mm
Moment at edge	M_0	=	623897.27	Nmm

Saddle	F_i (N)	Q_i (mm)	M_i (Nmm)	M_{ij} (Nmm)
2	2.45E+004	2.01E+004	5.10E+005	2.17E+007
1	2.45E+004	2.05E+004	2.51E+005	2.32E+007

Load limits between the saddles

Saddle	K12	Strength (N/mm²)	f_{max}	Instability	
2	0.00	0.00		0.000	OK!
1	0.00	0.00		0.000	OK!

Load limit at the saddles

Saddle	e_a (mm)	b_1 (mm)	δ_1 (°)
2	5.84	20.00	120.00
1	5.84	20.00	120.00

Saddle	γ	β	K_3	K_4	K_5	K_6	K_7	K_8	K_9	K_{10}
2	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903
1	0.030	0.195	0.817	0.988	0.981	0.346	0.634	0.151	0.555	0.903

Saddle	$K_{1,2}$	$K_{1,3}$	K_2	$v_{1,2}$	$v_{1,3}$	$v_{2,2}$	$v_{2,3}$
2	1.285	0.782	1.250	-0.015	-1.903	0.398	0.797
1	1.285	0.782	1.250	-0.015	-1.903	0.398	0.797

Saddle	$\sigma_{b,all,2}$ (N/mm²)	$\sigma_{b,all,3}$ (N/mm²)	$F_{2,max}$ (N)	$F_{3,max}$ (N)	
2	205.34	125.00	97605.45	192865.36	Ok: F_i <= min($F_{2,max}$; $F_{3,max}$)
1	205.31	125.00	97591.45	192865.36	Ok: F_i <= min($F_{2,max}$; $F_{3,max}$)

Saddle	M_{max} (Nmm)	F_{max} (N)	Q_{max} (N)	F_{eq} (N)
2	8.35E+008	2.24E+006	1.19E+006	1.61E+004
1	8.35E+008	2.24E+006	1.19E+006	1.61E+004

Saddle	Instability (<= 1.0)	
2	0.820	OK!
1	0.819	OK!