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Comparative research into the load-bearing capacity of horizontal pressure vessels supported by saddles

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ABSTRACT: *This article evaluates and compares permissible saddle reactions for horizontal pressure vessels resting on two symmetrically placed saddles. Successively allowable saddle loads have been determined for a pre-selected typical horizontal pressure vessel according to various recognized design codes. Control of the circumferential compressive membrane plus bending stress at the horn of the saddles is central in the consideration because it determines often the allowable support load. Limiting these stresses prevents so-called "bulging" of the cylindrical shell over the saddle ends. Significant differences were found by comparing the mutually results. Remarkable differences occur in particular between methods based on "Zick" compared to methods based on "limit loads". The primary aim of this article is to provide engineers involved in the design of pressure vessels with new insights into this matter in order to arrive at a sound and well - considered vessel support design while ensuring structural integrity requirements.*

KEYWORDS: *saddle reactions, horizontal pressure vessel, circumferential compressive membrane plus bending stress, bulging, "Zick" method, "limit load" method, support design, structural integrity.*

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I. INTRODUCTION

Horizontal pressure vessels are usually symmetrically supported with two saddle supports. More saddles would result in static indeterminacy and difficulty in predicting the load distribution in the event of foundation settlement. The methodology for calculating stresses in the vicinity of the support saddles was developed by L.P.Zick [1] in the early 1950s and is still widely used by designers of horizontal pressure vessels. Zick's analysis was based on the assumption that the supports are rigid and not connected to the vessel shell. In reality most vessels have flexible supports that are welded onto the vessel shell. This means that Zick's analysis is conservative for traditional saddle constructions. The L.P. Zick method has been adopted by many recognized design codes including Rules for Pressure Vessels (RfPV) [2], ASME Section VIII-Division 2[3] and PD 5500 [4]. The current practice is to use the semi-empirical method developed by Zick which is based on beam theory and various assumptions to simplify the problem. Due to these assumptions Zick´s method may not yield accurate results but has proved to be sufficiently reliable in practice already for a considerably long period. However vigilance is required in order not to underestimate the load carrying capacity of the vessel, by realizing that the stresses are strongly localized in the area of the saddle horns, while the rest of the vessel is only moderately stressed. This article focuses on the circumferential stresses at the horn of the saddle and at the end of the wear plate since these are often the most important stresses. Moreover, in most cases those stresses determine the allowable support load. In addition to the method developed by L.P.Zick, a limit load analysis method was developed in a later period in the former DDR that was included in the so-called TGL standards [5]. This method is now in slightly modified form included in both AD 2000 [6] and EN 13445-3[7] and will also be addressed in this article.

II. OBSERVATIONS CONSERNING WEAR PLATES

When calculating the circumferential stresses at the horn of the saddle, it is important to recognize the influence of the wear plate that may be present between the cylindrical shell and the saddle. It is claimed that such a wear plate has a stress-reducing effect if it is of sufficient dimensions. However, it appears that the

various codes apply different criteria for this which make it difficult to obtain a reliable result. Clarity leaves something to be desired. In practice, different interpretations are attributed to the influence of a wear plate.
In summary, the following conditions generally apply to both the saddles and the wear plates:

- Wear plate welded continuously around the cylindrical shell
- Width of the wear plate ≥ b + 1.56 $\sqrt{R.t}$ [1][3] respectively w + 1.6 $\sqrt{R.t}$ [2] or b + 10t [4] with b = width of saddle, R = mean radius of cylindrical shell and t = shell thickness
- Circumferential span of wear plate ≥ θ + 10° [2] or θ + 12° [3] with θ = Subtended angle or saddle contact angle
- Thickness of wear plate, maximum 2 times shell thickness (depending on applicable code) Corners of wear plate rounded off with radius ≥ 3 times the wear plate thickness [2]
- Each support should extend at least 120° around and approximately $\sqrt{30 \text{ x vessel diameter}}$ [4] along the vessel in order to transmit the reaction gradually into the shell wall
- One vessel support is fixed while the other support has slotted holes in the base plate for axial movement when thermal strains occur
- Diameter to thickness ratios up to the order of 250 [4] (depending on applicable code)
- Material of wear plate preferably identical to shell material. In case that the wear plate material has a lower allowable stress, the wear plate thickness must be corrected with the allowable stress ratio of both materials. (depending on applicable code)

Factors Affecting Stress Distribution At Horn Of Saddles

Saddle horn area is the area in the vessel where wear plate just adjoins the vessel. In this area high stresses are produced compared to the other vessel area. The following elements affect the magnitude of these stresses:

- Saddle wrap (embracing) angle
- Saddle width
- Wear plate width (wear plate is synonymous with reinforcing plate and saddle plate)
- Distance of saddle from head
- Wear plate extension
- Wear plate thickness

Exemptions Regarding Wear Plates

The wear plate according to APPENDIX 2 does not satisfy the conditions of the relevant design code for a vessel with an outside diameter of 2000 mm and a shell thickness of 10 mm. Therefore the value zero must be entered for the wear plate thickness in the formulas for the determination of the circumferential compressive membrane plus bending stress at the horn of the saddle.

Extended Wear Plate Dimensions (For "Zick" Based Analysis)

The wear plate according to APPENDIX 2 may be taken into account when calculating the circumferential compressive membrane plus bending stress if it satisfy the following dimensions:
Width of the wear plate ≥ w + 1.56 $\sqrt{R_m.t}$ = 250 + 1.56 $\sqrt{995 \text{ x } 10}$ = 405.6 mm ⇒ Take: 410 mm
Subtended angle or saddle contact angle: θ + 12° = 132°

Scope Of "Zick" Based Analysis

In determining the allowable saddle support reaction, it will be assumed that the allowable circumferential compressive membrane plus bending stress is achieved at the horn of the saddle while the vessel is under atmospheric pressure. The following cases will be calculated:

- A/R ≤ 0.5 w/o wear plate @ operating and test temperature
- A/R ≤ 0.5 with wear plate @ operating and test temperature
- A/R ≥ 1.0 w/o wear plate @ operating and test temperature
- A/R ≥ 1.0 with wear plate @ operating and test temperature

The above cases will be calculated according to the following design codes:

- RfPV- Sheet D 1105 § 3.2
- PD 5500 - Clause G.3.3.2.6.2
- ASME BPVC Section VIII-Division 2; Clause 4.15.3.5

Formula Overview For Determining The Allowable Saddle Support Reaction

The table below shows the equations to determine the allowable saddle reactions that are derived from the relevant design codes.

Design Code	Equations for allowable saddle reactions
RfPV- Sheet D 1105 § 3.2	$F = \dfrac{1.5\,f}{\left[\dfrac{1}{4(d+d_1)(b+1.6\sqrt{r.d})} + \dfrac{6\,k_3.r.C_\sigma}{b_e \cdot (d^2+d_1^2)}\right]}$
PD 5500 - Clause G.3.3.2.6.2	$W_1 = \dfrac{1.25\,f}{\left[\dfrac{1}{4(t+t_1)(b_2+10t)} + \dfrac{6\,k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}\right]}$
ASME BPVC Section VIII-Division 2; Clause 4.15.3.5	$Q = \dfrac{1.25\,S}{\left[\dfrac{1}{4(t+t_r)(b+1.56\sqrt{R_m.t})} + \dfrac{6\,k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}\right]}$

Where:

$F = W_1 = Q$ = saddle reaction (N)

$r = R_m$ = the mean vessel radius (mm)

$t = d$ = the wall thickness of the vessel (mm)

$t_1 = d_1$ = thickness of wear plate (mm)

$b = b_2$ = the width of the saddle (mm)

b_e = the smaller value of 4R and half the length between the tangent lines of the vessel (mm)

$f = S$ = design strength (MPa) respectively f_{oper} @ operating condition and f_{test} @ hydrostatic test condition i.e. f is respectively: $f_{oper} = R_{e;\vartheta m}/1.5$ (MPa) and $f_{test} = R_e/1.5$ (MPa)

$k_3 = k_6$ = factor depending on saddle angle and the distance of saddle to the tangent line (see the table below) (-)

$C_\sigma = 1 - \frac{b - 300\ mm}{A}$ with minimum 0.4 and maximum 1.0 (-)

A = distance from saddle support to adjacent end of cylindrical part (mm)

R_e = yield strength @ test temperature (MPa)

$R_{e;\vartheta m}$ = yield strength @ operating temperature (MPa)

Saddle angle θ	$k_3 = k_6$	$k_3 = k_6$
	$A/R \leq 0.5$	$A/R \geq 1.0$
120°	0.0132	0.0528
150°	0.0079	0.0316

NOTE: For $0.50 < A/R < 1.00$ values for k_3 and k_6 should be obtained by linear interpolation of the values in this table.

If the wear plate does not satisfy the conditions of the applicable design code, then the value 0 (zero) must be entered for t_1 or t_r in the formulas for F , W_1 and Q.

A notable difference has been observed in the equations with respect to F, W_1 and Q, namely that the term relating to the circumferential bending stress in particular:

$\frac{F\,.6\,k_3.r.C_\sigma}{b_e \cdot (d^2+d_1^2)}$ *respectively* $\frac{W_1\,.\,6\,k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}$ *and* $\frac{Q\,.\,6\,k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}$ differ from each other in the term between brackets!

Moreover a paragraph has been included in [8] concerning "circumferential bending". This includes specific conditions that relate to the shell - and wear plate thickness as well as the dimensions of the wear plate i.e. if $A/R \leq 0.5$(stiffened by the adjacent heads) and the wear plate extends R/10 above the horn of saddle then: $t_s = t_s + t_w$ and $ts^2 = t_s^2 + t_w^2$, where $t_s = d = t$ and $t_w = d_1 = t_r$. Note that if the saddle support is near the vessel end, then the shell remains circular and the full section is available to resist bending. If not, then only a partial section is available. The highest stress is often at the saddle horn, where the shell "bends" over the support and has little or no resistance to radial deformation. More information on this topic can be found in [9].

Unfortunately, a solid explanation for all these differences could not be found.

Numerical Elaborations "Zick" Based Analysis

Vessel and saddle data are included in APPENDIX 1.Typical saddle configuration is included in APPENDIX 2.

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

RfPV- Sheet D 1105 § 3.2 Input data: f = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_3 = 0.0132; r = 995 mm; C_σ = 1.0; d = 10 mm; d_1 = 0 mm; b = 250 mm; b_e = 3980 mm	$F = \frac{1.5\ f}{\left[\frac{1}{4(d+d_1)(b+1.6\sqrt{r.d})} + \frac{6\ k_3.r.C_\sigma}{b_e \cdot (d^2 + d_1^2)}\right]}$
$F_{operating}$ = 988302 N	**F_{test} = 988302 x 176.67/170.67 = 1023046 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

RfPV- Sheet D 1105 § 3.2 Input data: f = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_3 = 0.0132; r = 995 mm; C_σ = 1.0; d = 10 mm; d_1 = 13 mm; b = 250 mm; b_e = 3980 mm	$F = \frac{1.5\ f}{\left[\frac{1}{4(d+d_1)(b+1.6\sqrt{r.d})} + \frac{6\ k_3.r.C_\sigma}{b_e \cdot (d^2 + d_1^2)}\right]}$
$F_{operating}$ = 2556394 N	**F_{test} = 2556394 x 176.67/170.67 = 2646266 N**

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

PD 5500 - Clause G.3.3.2.6.2 Input data: f = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_6 = 0.0132; r = 995 mm; C_σ = 1.0; t = 10 mm; t_1 = 0 mm; b_2 = 250 mm; b_e = 3980 mm	$W_1 = \frac{1.25\ f}{\left[\frac{1}{4(t+t_1)(b_2+10t)} + \frac{6\ k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}\right]}$
$W_{operating}$ = 791815 N	**W_{test} = 791815 x 176.67/170.67 = 819651 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

PD 5500 - Clause G.3.3.2.6.2 Input data: f = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_6 = 0.0132; r = 995 mm; C_σ = 1.0; t = 10 mm; t_1 =13 mm; b_2 = 250 mm; b_e = 3980 mm	$W_1 = \frac{1.25\ f}{\left[\frac{1}{4(t+t_1)(b_2+10t)} + \frac{6\ k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}\right]}$
$W_{operating}$ = 3115098 N	**W_{test} = 3115098 x 176.67/170.67 = 3224611 N**

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

ASME Section VIII-Division 2; Clause 4.15.3.5 Input data: S = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_6 = 0.0132; R_m = 995 mm; C_σ = 1.0; t = 10 mm; t_r = 0 mm; b = 250 mm; b_e = 3980 mm	$Q = \frac{1.25\ S}{\left[\frac{1}{4(t+t_r)(b+1.56\sqrt{R_m.t})} + \frac{6\ k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}\right]}$
$Q_{operating}$ = 821680 N	**Q_{test} = 821680 x 176.67/170.67 = 850567 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

ASME Section VIII-Division 2; Clause 4.15.3.5 Input data: S = 170.67 MPa @ 50°C respectively: 176.67MPa @ 20°C; k_6 = 0.0132; R_m = 995 mm; C_σ = 1.0; t = 10 mm; t_r = 13 mm; b = 250 mm; b_e = 3980 mm	$Q = \frac{1.25\ S}{\left[\frac{1}{4(t+t_r)(b+1.56\sqrt{R_m.t})} + \frac{6\ k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}\right]}$
$Q_{operating}$ = 3321606 N	**Q_{test} = 3321606 x 176.67/170.67 = 3438379 N**

CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

RfPV- Sheet D 1105 § 3.2 Input data: $f = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_3 = 0.0528$; $r = 995$ mm; $C_\sigma = 1.0$; $d = 10$ mm; $d_1 = 0$ mm; $b = 250$ mm; $b_e = 3980$ mm	$F = \dfrac{1.5\ f}{\left[\dfrac{1}{4(d+d_1)(b+1.6\sqrt{r.d})} + \dfrac{6\ k_3.r.C_\sigma}{b_e \cdot (d^2 + d_1^2)}\right]}$
$F_{operating}$ = 300111 N	**F_{test} = 300111 x 176.67/170.67 = 310661 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

RfPV- Sheet D 1105 § 3.2 Input data: $f = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_3 = 0.0528$; $r = 995$ mm; $C_\sigma = 1.0$; $d = 10$ mm; $d_1 = 13$ mm; $b = 250$ mm; $b_e = 3980$ mm	$F = \dfrac{1.5\ f}{\left[\dfrac{1}{4(d+d_1)(b+1.6\sqrt{r.d})} + \dfrac{6\ k_3.r.C_\sigma}{b_e \cdot (d^2 + d_1^2)}\right]}$
$F_{operating}$ = 797621 N	**F_{test} = 797621 x 176.67/170.67 = 825661 N**

CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

PD 5500 - Clause G.3.3.2.6.2 Input data: $f = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_6 = 0.0528$; $r = 995$ mm; $C_\sigma = 1.0$; $t = 10$ mm; $t_1 = 0$ mm; $b_2 = 250$ mm; $b_e = 3980$ mm	$W_1 = \dfrac{1.25\ f}{\left[\dfrac{1}{4(t+t_1)(b_2+10t)} + \dfrac{6\ k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}\right]}$
$W_{operating}$ = 247082 N	**W_{test} = 247082 x 176.67/170.67 = 255768 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

PD 5500 - Clause G.3.3.2.6.2 Input data: $f = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_6 = 0.0528$; $r = 995$ mm; $C_\sigma = 1.0$; $t = 10$ mm; $t_1 = 13$ mm; $b_2 = 250$ mm; $b_e = 3980$ mm	$W_1 = \dfrac{1.25\ f}{\left[\dfrac{1}{4(t+t_1)(b_2+10t)} + \dfrac{6\ k_6.r.C_\sigma}{b_e \cdot (t+t_1)^2}\right]}$
$W_{operating}$ = 1180145 N	**W_{test} = 1180145 x 176.67/170.67 = 1221633 N**

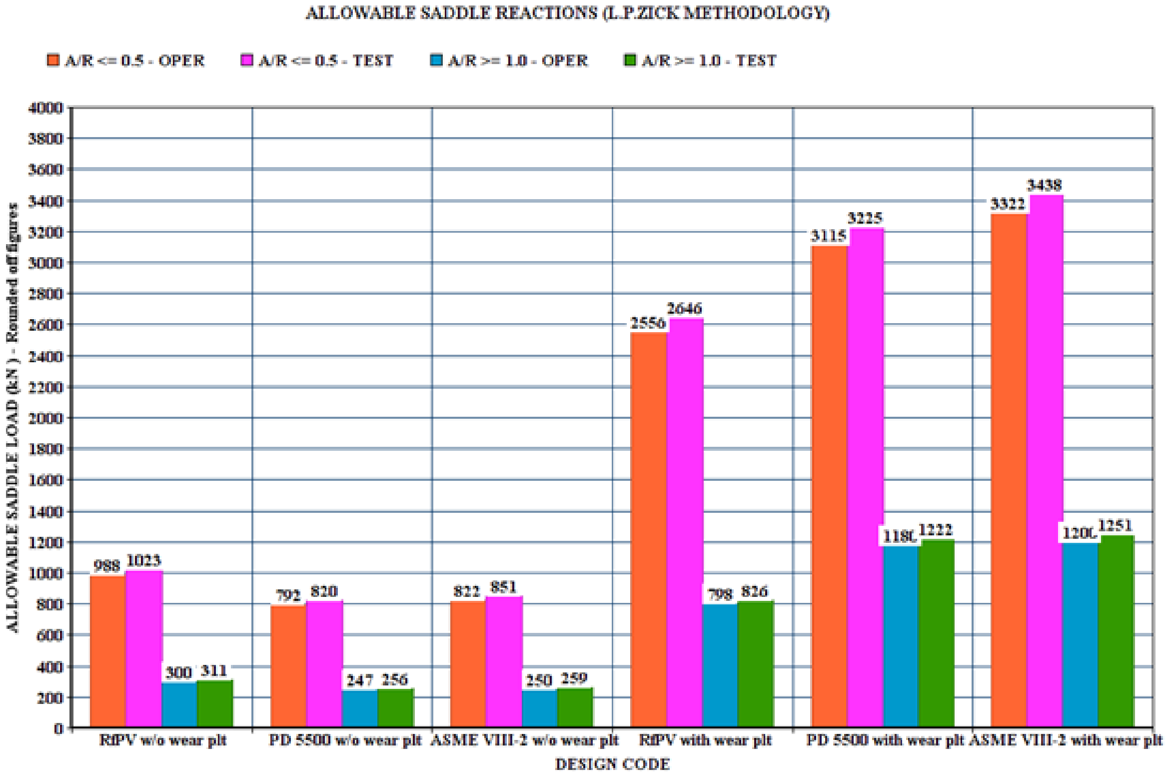
CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

ASME Section VIII-Division 2; Clause 4.15.3.5 Input data: $S = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_6 = 0.0528$; $R_m = 995$ mm; $C_\sigma = 1.0$; $t = 10$ mm; $t_r = 0$ mm; $b = 250$ mm; $b_e = 3980$ mm	$Q = \dfrac{1.25\ S}{\left[\dfrac{1}{4(t+t_r)(b+1.56\sqrt{R_m.t})} + \dfrac{6\ k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}\right]}$
$Q_{operating}$ = 249916 N	**Q_{test} = 249916 x 176.67/170.67 = 258702 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

ASME Section VIII-Division 2; Clause 4.15.3.5 Input data: $S = 170.67$ MPa @ 50°C respectively: 176.67MPa @ 20°C; $k_6 = 0.0528$; $R_m = 995$ mm; $C_\sigma = 1.0$; $t = 10$ mm; $t_r = 13$ mm; $b = 250$ mm; $b_e = 3980$ mm	$Q = \dfrac{1.25\ S}{\left[\dfrac{1}{4(t+t_r)(b+1.56\sqrt{R_m.t})} + \dfrac{6\ k_6.R_m.C_\sigma}{b_e \cdot (t+t_r)^2}\right]}$
$Q_{operating}$ = 1208611 N	**Q_{test} = 1208611 x 176.67/170.67 = 1251101 N**

In the graph below the allowable saddle loads expressed in kN are summarized.



The elaborated cases above shows that there is a considerable difference between the two cases dealt with. In the case where the saddles are placed closer to the heads (A / R ≤ 0.5), the allowable saddle load varies a factor of 2.64 to 3.3 than in the case where the saddles are more removed from the heads (A / R ≥ 1.0). Thus the load-bearing capacity can be significantly increased by placing the saddles near the ends. Moreover with respect to the influence of a wear plate that meets the requirements of the relevant code, the numerical elaboration shows that this results in a factor 2.6 to 4.8 higher allowable saddle load.

Summary of saddle load capacities (lowest calculated values)

Allowable saddle load (N) during operating / hydrostatic test	**Saddle location**
791815 / 819651 (PD 5500)	A/R ≤ 0.5 i.e. near the ends w/o wear plate
2556394/2646266 (RfPV)	A/R ≤ 0.5 i.e. near the ends with wear plate
247082 / 255768 (PD 5500)	A/R > 1.0 i.e. remote from the ends w/o wear plate
797621 / 825661 (RfPV)	A/R > 1.0 i.e. remote from the ends with wear plate
Saddle reactions of selected base case horizontal pressure vessel	**Weight divided by two**
Saddle reaction during operation	93654 N
Saddle reaction during hydro test	157397 N
Situations	**Usage factors pertaining to the considered condition**
A/R ≤ 0.5 i.e. near the ends w/o wear plate	Operation: 791815 / 93654 = 8.45 Hydro test: 819651 / 157397 = 5.2
A/R ≤ 0.5 i.e. near the ends with wear plate	Operation: 2556394 / 93654 = 27.3 Hydro test: 2646266 / 157397 = 16.8
A/R ≥ 1.0 i.e. near the ends w/o wear plate	Operation: 247082 / 93654 = 2.64 Hydro test: 255768 / 157397 = 1.62
A/R ≥ 1.0 i.e. near the ends with wear plate	Operation: 797621 / 93654 = 8.52 Hydro test: 825661 / 157397 = 5.25

Note: Saddle data are obtained from APPENDIX 2

Supplementary calculations according to ASME Section VIII - Division 1

ASME BPVC Section VIII - Division 1[10] refers in G-6 which relates to horizontal vessel supports to the publication by L.P. Zick "Stresses in Large Cylindrical Pressure Vessels on Two Saddle Supports" which corresponds to reference [1].

The following cases are further elaborated:
Case # 1: A/R ≤ 0.5 ; wear plate width = 410 mm; wear plate thickness = 13 mm; wear plate contact angle = 132°
Case # 2: A/R ≥ 1.0 ; wear plate width = 410 mm; wear plate thickness = 13 mm; wear plate contact angle = 132°
Case # 3: A/R ≤ 0.5 ; wear plate width = 340 mm; wear plate thickness = 13 mm; wear plate contact angle = 132°
Case # 4: A/R ≥ 1.0 ; wear plate width = 340 mm; wear plate thickness = 13 mm; wear plate contact angle = 132°

Formula overview

Allowable saddle support reaction Q for operating respectively hydrostatic test condition at saddle horns

$$Q = \frac{1.5\, S_a\,(resp.)\; 0.9\, S_y}{\left[\frac{1}{4(t+t_p)(b+1.56\sqrt{R_o.t})} + \frac{3\,k_3}{2(t^2+\,t_p^2)}\right]}$$

Allowable saddle support reaction Q for operating respectively hydrostatic test condition at wear plate horns

$$Q = \frac{1.5\, S_a\;(resp.)\; 0.9\, S_y}{\left[\frac{1}{4\,t\,(b+1.56\sqrt{R_o.t})} + \frac{3\,k_3}{2(t^2)}\right]}$$

Where:
S_a = Allowable stress at operating temperature = 117 MPa
S_y = Yield strength at test temperature = 265 MPa
t = Shell thickness = 10 mm ; t_p = wear plate thickness = 13 mm
R_o = Outside radius of cylindrical shell = 1000 mm ; b = saddle width = 250 mm
k_3 = Design factor depending on saddle angle and A/R ratio

Design factor	A/R ≤ 0.5	A/R ≥ 1.0
k_3 @ θ = 120°	0.0132	0.0525
k_3 @ θ = 132°	0.0109	0.0431

Summary of calculation results

Cases	Q - operating weight (w/o pressure) [N] saddle horns / wear plate horns	Q - test weight (w/o pressure) [N] saddle horns / wear plate horns	Q – Ratios saddle horns / wear plate horns (-)
Case #1	2004500 / 779735	2724064 / 1059640	2.571
Case #2	549256 / 206695	746425 / 280893	2.657
Case #3	676102 / - (*)	918805 / - (*)	n.a
Case #4	206695 / - (*)	280993 / - (*)	n.a

Note that the "COMPRESS" Pressure Vessel Design Software has been used to obtain the above results.
(*) Insufficient wear plate dimensions

Computations Based On Limit Load Analysis

AD 2000 - Merkblatt S3/2 [6] and EN 13445-3 Clause 16.8 [7] are almost identical and both stem from the TGL standard [5]. However the main focus is on EN 13445-3 rather than on AD - Merkblatt S3/2 [6]. APPENDIX 3 shows a typical saddle support arrangement for a horizontal vessel.

Calculation of maximum allowable saddle load as per EN 13445- Part 3; Clause 16.8

The horn of the saddle is considered the most critical location for determining the permissible saddle support reaction, therefore the following formula applies:

$$F_{max,all.} = \frac{0.9\,\sigma_{b,all,}\sqrt{D_i e_a}\cdot e_a}{K_7 K_9 K_{10}}$$

Where:
$F_{max\,,all.}$ allowable support reaction force resulting from loading in circumferential direction at the horn of the saddle (N)
$\sigma_{b,all,}$ the bending limit stress of shell (MPa)
D_i inside diameter of cylindrical shell (mm)
e_a wall thickness of cylindrical shell (mm)
K_7, K_9, K_{10} coefficients (-)

The following load cases will be considered:

Parameters	Case # 1	Case #2	Case # 3	Case # 4
	with wear plate according Appendix 2	with wear plate according Appendix 2	with extended wear plate	with extended wear plate
a_1 (mm)	495	995	495	995
l_1 (mm)	6970	5970	6970	5970
δ (°)	120	120	120	120
δ_2 (°)	132	132	143	143
b_2 (mm)	340	340	511	511
e_2 (mm)	13	13	13	13

Formula overview

$\sigma_{b,all,} = K_1 . K_2 . f$ $K_2 = 1.25$ for design conditions respectively 1.05 for test condition f = design strength for the considered condition	$K_4 = \frac{(1 - 2.718282^{-\beta} \ cos\, \beta)}{\beta}$
$F_{max,all.} = \frac{0.9\, \sigma_{b,all,} \sqrt{D_i e_a} \cdot e_a}{K_7 K_9 K_{10}}$	$K_7 = \frac{1.45 - 0.007505\, \delta}{sin(0.5\, \delta)}$
$\gamma = 2.83 \left(\frac{a_1}{D_i}\right) \sqrt{\frac{e_a}{D_i}}$; $\beta = \frac{0.91 b_1}{\sqrt{D_i . e_a}}$	$K_9 = 1 - \frac{0.65}{1+(6\gamma)^2} \sqrt{\frac{60}{\delta}}$
$K_1 = \frac{1 - \upsilon_2^2}{\left(\frac{1}{3} + \upsilon_1 \upsilon_2\right) + \sqrt{\left(\frac{1}{3} + \upsilon_1 \upsilon_2\right)^2 + (1 - \upsilon_2^2)\upsilon_1^2}}$	$K_{10} = \frac{1}{1 + \left(\frac{b_1}{D_i} \delta\right) 0.010472 \sqrt[3]{\frac{D_i}{e_a}}}$
$\upsilon_1 = -0.53 \frac{K_4}{K_7\, K_9 K_{10}\, sin(0.5\, \delta)}$; $\upsilon_2 = \frac{P . D_i}{2 e_a} \frac{1}{K_2 . f}$	$K_{11} = \frac{5}{(\, 0.10472 . \delta \sqrt[3]{\frac{D_i}{e_a}}}$
Nomenclature: See Appendix 1 & 3	

Detailed elaboration of the above formulas for the various cases falls outside the scope of this article and is therefore intentionally omitted. Hence, only the computation results are displayed in the next section .

Results Of Computations Obtained With The Aid Of "VES" Software Package From P3 Engineering

The table below shows the load limits of the saddles for the various load cases and the associated conditions

LOAD CASES (EN 13445-3; Clause 16.8)	CASE #1	CASE #2	CASE #3	CASE #4
Operating incl. pressure (N)	1005627	978152	1545752	1496701
Operating w/o pressure (N)	599286	584997	1091445	1059374
Hydrostatic test incl. pressure (N)	1324199	1287131	2004718	1940286
Hydrostatic test w/o pressure (N)	744426	726676	1355780	1315942

LOAD CASES (AD 2000 ; Merkblatt S 3/2)	CASE #1	CASE #2	CASE #3	CASE #4
Operating incl. pressure (N)	1271970	1233413	1858083	1796316
Operating w/o pressure (N)	759773	739961	1242866	1213535
Hydrostatic test incl. pressure (N)	1674168	1622011	2409559	2328379
Hydrostatic test w/o pressure (N)	943781	919170	1543873	1507438

The table below shows the allowable saddle loads that are ranked according to A / R ratio, with and without effective wear plate and design code.

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

RfPV	
$F_{operating}$ = 988302 N	**F_{test} = 988302 x 176.67/170.67 = 1023046 N**

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

PD 5500	
$W_{operating}$ = 791815 N	**W_{test} = 791815 x 176.67/170.67 = 819651 N**

CASE: A/R ≤ 0.5 w/o wear plate @ operating and test temperature

ASME Section VIII - 2	
$Q_{operating}$ = 821680 N	**Q_{test} = 821680 x 176.67/170.67 = 850567 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

RfPV	
$F_{operating}$ = 2556394 N	**F_{test} = 2556394 x 176.67/170.67 = 2646266 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

PD 5500	
$W_{operating}$ = 3115098 N	**W_{test} = 3115098 x 176.67/170.67 = 3224611 N**

CASE: A/R ≤ 0.5 with wear plate @ operating and test temperature

ASME Section VIII - 2	
$Q_{operating}$ = 3321606 N	**Q_{test} = 3321606 x 176.67/170.67 = 3438379 N**

CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

RfPV	
$F_{operating}$ = 300111 N	**F_{test} = 300111 x 176.67/170.67 = 310661 N**

CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

PD 5500	
$W_{operating}$ = 247082 N	**W_{test} =247082x 176.67/170.67 = 255768 N**

CASE: A/R ≥ 1.0 w/o wear plate @ operating and test temperature

ASME Section VIII - 2	
$Q_{operating}$ = 249916 N	**Q_{test} = 249916 x 176.67/170.67 = 258702 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

RfPV	
$F_{operating}$ = 797621 N	**F_{test} = 797621 x 176.67/170.67 = 825661 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

PD 5500	
$W_{operating}$ = 1180145 N	**W_{test} = 1180145 x 176.67/170.67 =1221633 N**

CASE: A/R ≥ 1.0 with wear plate @ operating and test temperature

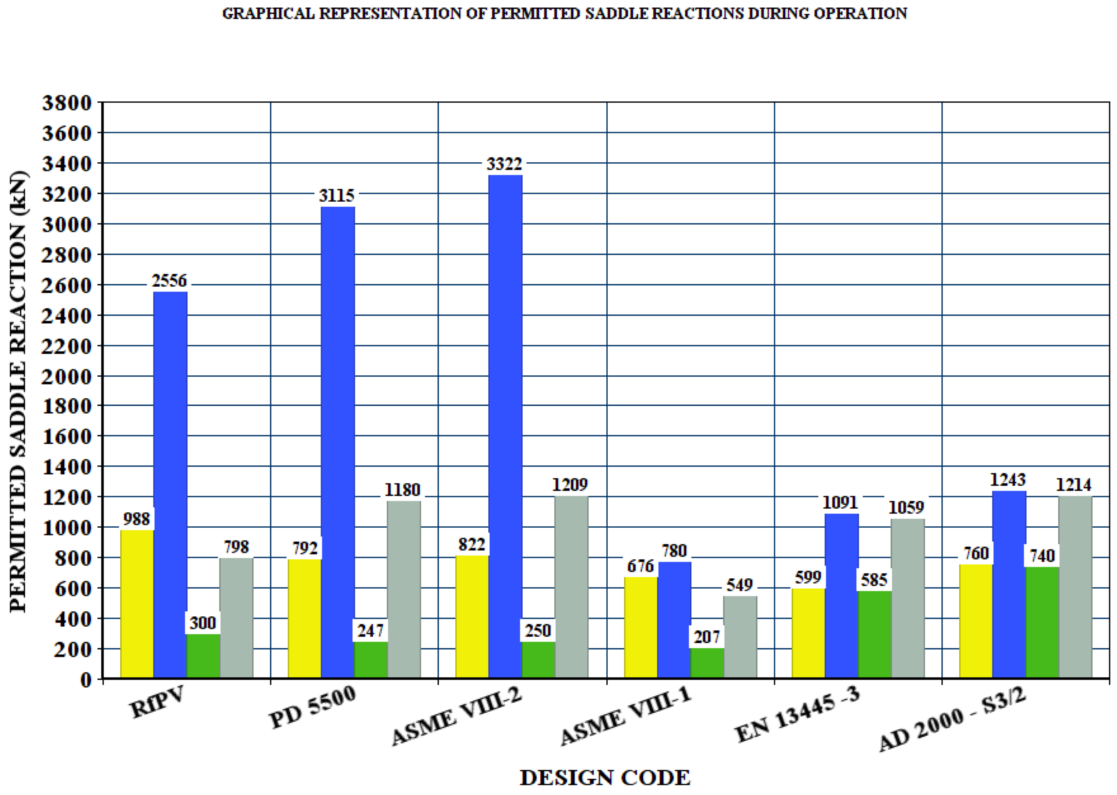
ASME Section VIII - 2	
$Q_{operating}$ = 1208611 N	**Q_{test} = 1208611 x 176.67/170.67 = 1251101 N**

The allowable saddle support reactions for four different cases calculated according to the indicated design code are presented in the table below.

Overview of allowable saddle support reactions (N) @ operating vs. test temperature

DESIGN CODE	CASE YELLOW	CASE BLUE	CASE GREEN	CASE GREY	Mutual ratios of the different cases
RfPV (NL)	988302 1023046	2556394 2646266	300111 310661	797621 825661	0.213:0.551:0.065:0.172
PD 5500 (UK)	791815 819651	3115098 3224611	247082 255768	1180145 1221633	0.148:0.584:0.046:0.221
ASME VIII-2 (USA)	821680 850567	3321606 3438379	249916 258702	1208611 1251101	0.147:0.593:0.045:0.216
ASME VIII-1 (USA)	676102 918805	779735 1059640	206695 280993	549256 746425	0.306:0.353:0.093:0.248
EN 13445 (EU)	599286 744426	1091445 1355780	584997 726676	1059374 1315942	0.180:0.327:0.175:0.318
AD 2000 - S3/2 (D)	759773 943781	1242866 1543873	73996 919170	1213535 1507438	0.192:0.314:0.187:0.307

KEY CASES:
YELLOW : A/R ≤ 0.5 w/o wear plate @ operating and test temperature or insufficient wear plate dimensions
BLUE : A/R ≤ 0.5 with wear plate @ operating and test temperature
GREEN : A/R ≥ 1.0 w/o wear plate @ operating and test temperature or insufficient wear plate dimensions
GREY : A/R ≥ 1.0 with wear plate @ operating and test temperature



Although the graph above relates to the situation during operation (saddle support reaction is half the weight during operation), it can be assumed that the mutual relationships during the hydrostatic test temperature correspond to those during the operation.

Mutual Ratios of Permitted Saddle Reactions During Operation

CASE	RfPV (NL)	PD 5500 (UK)	ASME VIII - 2 (USA)	ASME VIII - 1 (USA)	EN 13445 (EU)	AD 2000 – S 3/2 (D)
YELLOW	0.213	0.171	0.177	0.146	0.129	0.164
BLUE	0.211	0.257	0.274	0.064	0.091	0.103
GREEN	0.129	0.106	0.107	0.089	0.251	0.318
GREY	0.133	0.196	0.201	0.092	0.176	0.202
Lowest				Highest		

III. DISCUSSION

In practice, saddle supports are usually provided with a wear plate (see APPENDIX 2) that is continuous-ly welded to the cylindrical shell. In order to be able to take into account the stress-reducing effect of such a plate, the plate must meet certain dimensional requirements. When applying standardized saddle supports it often appears in practice that the dimensions do not meet the specified code requirements and their thickness is therefore generally left out of consideration. Of course, in practice, if there is a need for this, the wear plate can be given such dimensions that it can be taken into account in the saddle calculations. It also appears that in the various design codes there are different views on the interpretation of incorporating the wear plate into the calculations. This can give rise to significant differences in occurring stresses in the vicinity of the saddles. Particular in EN 13445-3 and AD 2000-Merkblatt S 3 / 2, there is potential for confusion because of the assumption that a wear plate (reinforcing plate) and also a so-called saddle plate are present, although AD 2000 – S 3/2 states that the procedure is also valid without reinforcing plate. It would be desirable for the relevant code committee to pay attention to this crucial aspect, which should lead to an adjustment of the relevant design code, which will aim to remove any ambiguity. The information presented will assist in the evaluation of the load acting on the saddle support, based on the assumption that the circumferential compressive membrane plus bending stress at the horn of the saddle is the most limiting factor.

IV. CONCLUSIONS

The key findings of this research are as follows:

- The allowable saddle loads (blue bars) calculated according to RfPV, PD 5500 or ASME VIII-2 which method initially has been developed by L.P. Zick and where the saddles are placed close to the heads (A / R ≤ 0.5) are substantially higher than calculated according to the limit load-based method as included in EN 13445-3 and AD 2000 S3/2. The prerequisite for this is however that the saddles are provided with a continuously welded wear plate to the cylindrical shell with sufficient dimensions according to the applicable code. The factor between the extremes is between 2.06 and 4.26. The case calculated according to ASME VIII-1 is an exception to this which is mainly caused by a considerably lower (approx. 31%) allowable stress.

- ❖ For the situation with the saddles in the vicinity of the heads ($A/R \leq 0.5$) without a wear plate or a wear plate of insufficient dimensions (yellow bars) , the mutual differences in permissible support reactions are less extreme. If we conveniently ignore the case calculated according to ASME VIII-1, then there is a factor of 1.65 between the values calculated according to the L.P. Zick method and those according to the limit-load method.
- ❖ In the case of saddles not placed close to the heads, i.e. $A/R \geq 1.0$ without wear plate or insufficient wear plate dimensions (green bars) , it is noticeable that the calculated allowable saddle load according to the L.P. Zick method is considerably lower than that according to the limit-load method. The difference amounts a factor of 2.47 to 3.58.
- ❖ In the case where $A/R \geq 1.0$ and a wear plate of sufficient dimensions according to the applicable code (grey bars) is applied, it is noticeable that with the exception of the calculated value according to ASME VIII-1 the values according to PD 5500 and ASME VIII -2 almost corresponds to the calculated values according to EN 13445-3 and AD 2000 S3/2. The mutual difference here varies between approx. 2.8 to 11.4 %. The allowable saddle support loads calculated according to RfPV and ASME VIII-1 differ considerably from each other, i.e. about 30%. The differences with the other calculated values are even more significant. The maximum difference amounts a factor of 2.21.
- ❖ In general it can be observed that the differences in the calculated allowable saddle support reactions are quite substantial. In particular the differences between the ones on L.P. Zick - based method and the limit-load based method (blue bars) are quite striking. Furthermore, we can conclude that a correct interpretation with regard to incorporating a wear plate in the calculation is crucial and that doubt about it must be removed. The relevant codes must provide more clarity on this.
- ❖ It is inexplicable that substantial differences exist in "saddle load capacity" for the case of saddles placed near the heads which are provided with a wear plate of sufficient dimensions between "Zick" based or "limit load" based analysis. Follow-up studies are desirable to provide clarity and insight into this matter. Numerical analyzes (FEA) offer the possibility to verify the methods.
- ❖ It appears that AD 2000 and EN 13445 (Limit Load Method) does not distinguish between $A/R \leq 0.5$ or $A/R \geq 1.0$, while this is clearly the case with the "Zick" methodology. In other words, saddles placed near the ends do not lead to a substantial increase in their load capacity in the case of AD 2000 and EN 13445.

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APPENDIX 1

Vessel and saddle data summary

Design conditions:

Quantity	Symbol	Value	Unit
Calculation temperature	$T=\vartheta_m$	50	°C
Internal pressure	P	1.5	MPa
Hydrostatic test pressure	P_T	2.145	MPa
Internal fluid density operating	ρ	510	Kg/m^3
Internal fluid density hydrotest	ρ	1000	Kg/m^3
Weld joint efficiency	z	1	-
Corrosion allowance	c	0	mm
Wall tolerance	-	0.0	mm

Materials:

Quantity	Symbol	Value	Unit	Remark
Nominal design stress of shell and wear plate	f = S	170.67	MPa	P265GH

Shell and Head dimensions:

Quantity	Symbol	Value	Unit	OD head: 2000 mm; Depth: 503.65 mm
Outside diameter	D_e	2000	mm	Dimensions acc. DIN 28013 ; Type Korbbogen
Inside diameter	D_i	1980	mm	Material: P265GH acc. EN 10028-2
Corrosion allowance	c	0	mm	Tensile strength: 410 MPa
Length of cylindrical part	$L= l_1$	7960	mm	Yield strength @20°C: 265 MPa respectively 256 MPa@ 50°C
Actual thickness	$e_n = t$	10	mm	Thickness korbbogen head: 10 mm AF
Analysis thickness	$e_a = t$	10	mm	Nominal design stress: 170.67 MPa @ 50°C and 176.67 @20°C

Saddle data:

Quantity	Symbol	Value	Unit
Included angle of saddle	$\delta = \theta$	120	degree (°)
Saddle width	$b_1 = b = w$	250	mm
Distance to adjacent head	$a_1 = A$	495 resp. 995	mm

Wear plate design data:

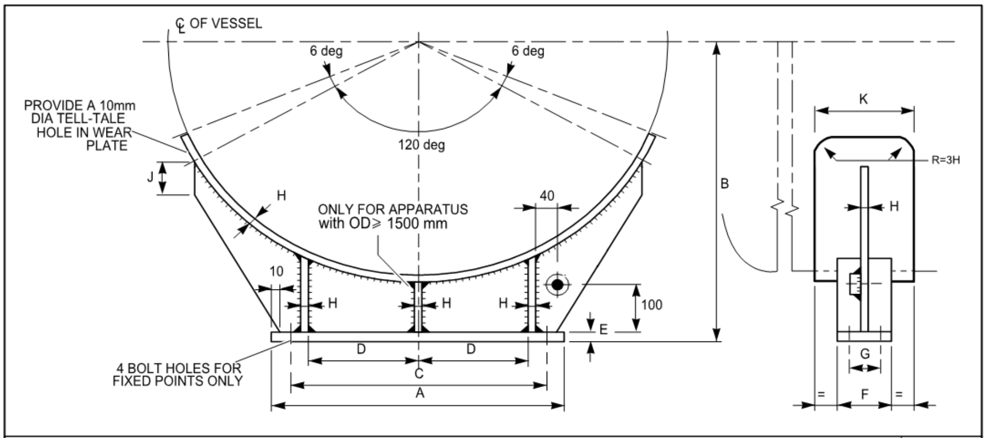
Quantity	Symbol	Value	Unit	Equation
Width of wear plate	$b_2 = w_p$	340	mm	
Critical width wear plate	$K_{11} D_i + 1.5 b_1$ $b_1 = w$	510.2	mm	$K_{11} = \frac{5}{(0.10472.\delta \sqrt[3]{\frac{D_i}{e_a}}}$
Distance from saddle horn to reinforcing (wear) plate	a_2	105 resp. 198	mm	
Included angle of wear plate	δ_2	132 resp. 143	(°)	
Thickness of wear plate	$e_2 = t_w$	13	mm	
Combined effective thickness	e_c	16.4	mm	$e_c = \sqrt{e_a^2 + e_2^2}$
Material identical to shell material				

Note: The wear plate may only be considered as a reinforcement plate if it satisfy the required conditions

Weights of vessel and contents:

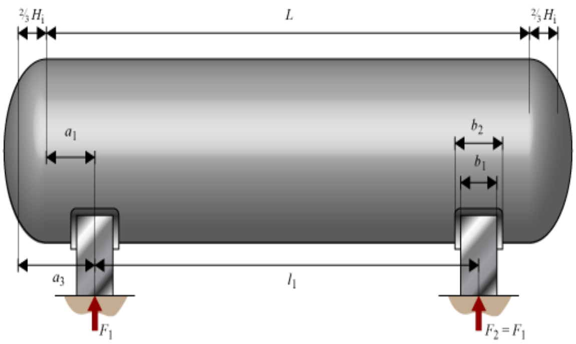
Quantity	Symbol	Value	Unit
Total weight of empty vessel	W_E	53937 / 5500	N / kg
Weight of content during operating	$W_{C,OP}$	133370/13600	N / kg
Total weight of vessel during operating	W_{OP}	187307/ 19100	N / kg
Weight of content during Hydrotest	$W_{C,T}$	260857 / 26600	N / kg
Total weight of vessel during Hydrotest	W_{HT}	314793 / 32100	N / kg
Maximum vertical force at saddle	F=Q	157397 / 16050	N / kg

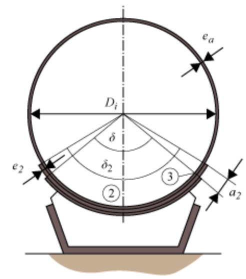
APPENDIX 2
Saddle configuration



	DIMENSIONS IN MILLIMETRES												APPROX WEIGHT
SHELL OD	A	B	C	D	E	F	G	H	J	K	BOLT DIA	BOLT HOLE DIA	TWO SADD IN N
2000	1220	1225	1070	460	19	250	150	13	125	340	M24	28	3460

APPENDIX 3 - Saddle support arrangement
(Source: The informative images below are derived from the VES user manual from P3 Engineering, Delft - The Netherlands)





a_1 Distance from saddle support to adjacent end of cylindrical part
a_3 Length of equivalent cylindrical shell
b_1 Axial width of saddle support
b_2 Width of reinforcement plate
l_1 Distance between two successive saddles
L Length of cylindrical part (including cylindrical part of heads)
H_i Internal head height

Saddle cross-section with reinforcement (wear) plate
D_i Inner shell diameter
e_n Nominal shell wall thickness
a_2 Distance from horn of saddle support to end of reinforcement plate
δ Included angle of saddle support (in degrees)
δ_2 Included angle of reinforcement plate (in degrees)

Walther Stikvoort "Comparative research into the load-bearing capacity of horizontal pressure vessels supported by saddles" American Journal of Engineering Research (AJER), vol. 8, no. 11, 2019, pp 62-74