

Static Equipment Awareness Session

Part 2

Stresses and mechanical design of static equipment



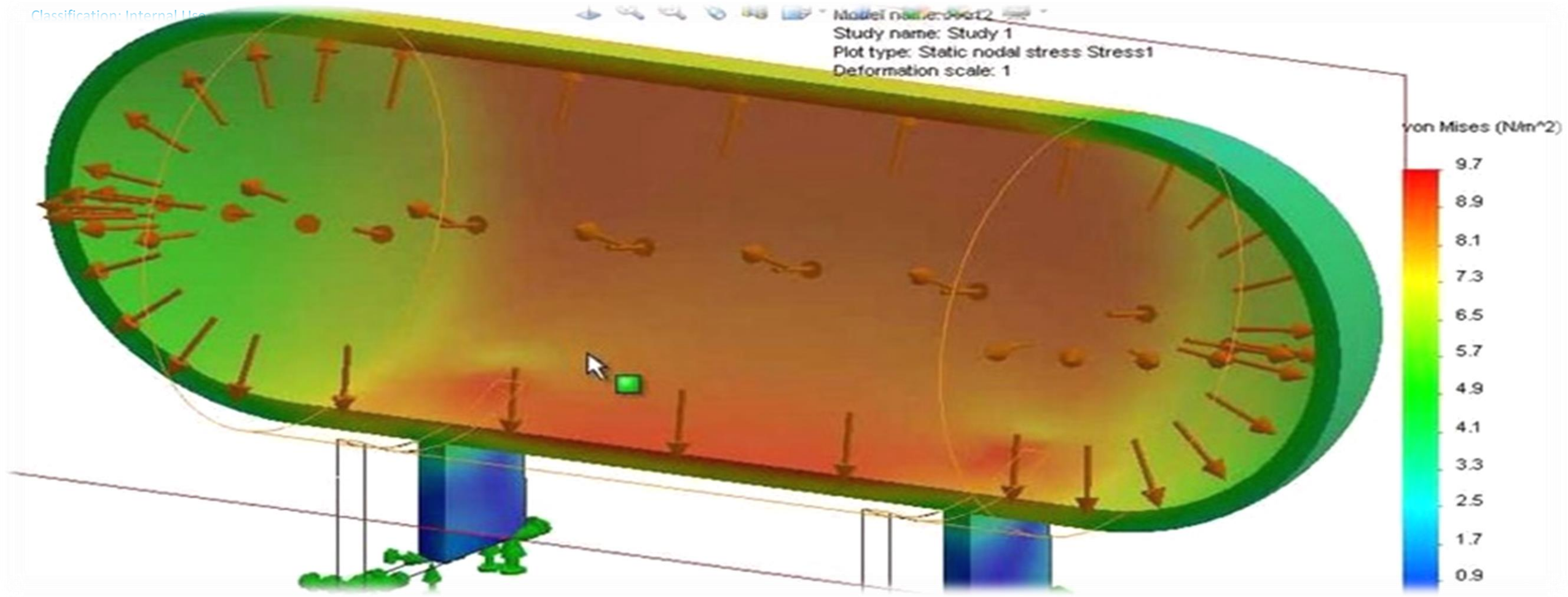
INSPECTION ACADEMY



BAHER ELSHEIKH

Contents

- ✓ Static Equipment Applications in Oil and Gas Industry
- ✓ Main static equipment
- ✓ Codes and standards
- ➡ Stresses and mechanical design of static equipment
- ⌚ Overview of common materials
- ⌚ Corrosion and damage mechanisms
- ⌚ Shell and tube heat exchanger as a practical example of static equipment



Stresses in Thin Cylindrical Pressure Vessel

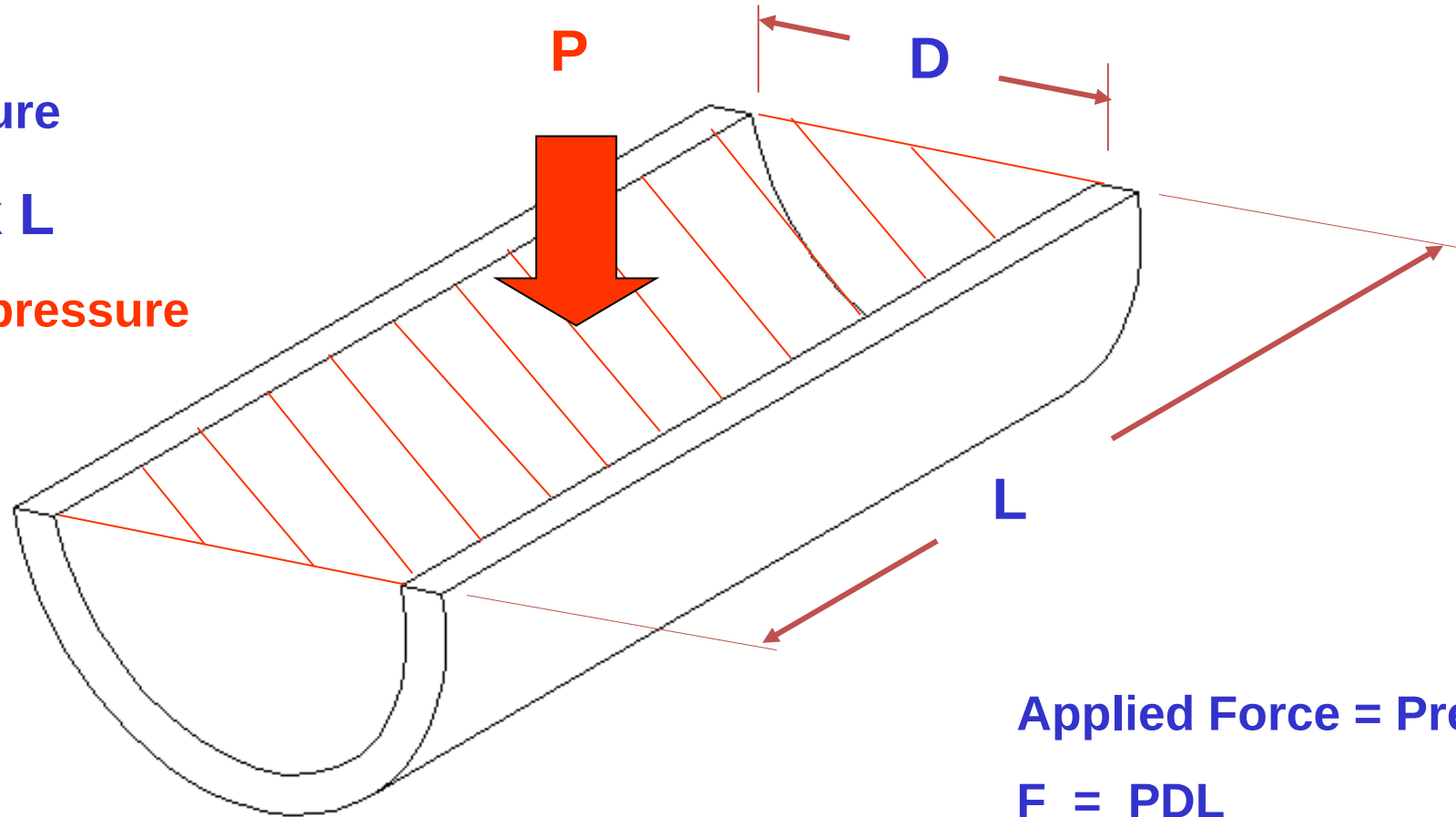
Internal Pressure stresses on cylindrical shell

Consider the forces acting on the Shell from Pressure

From pressure

Area = $D \times L$

Here is the pressure



Applied Force = Pressure X Area

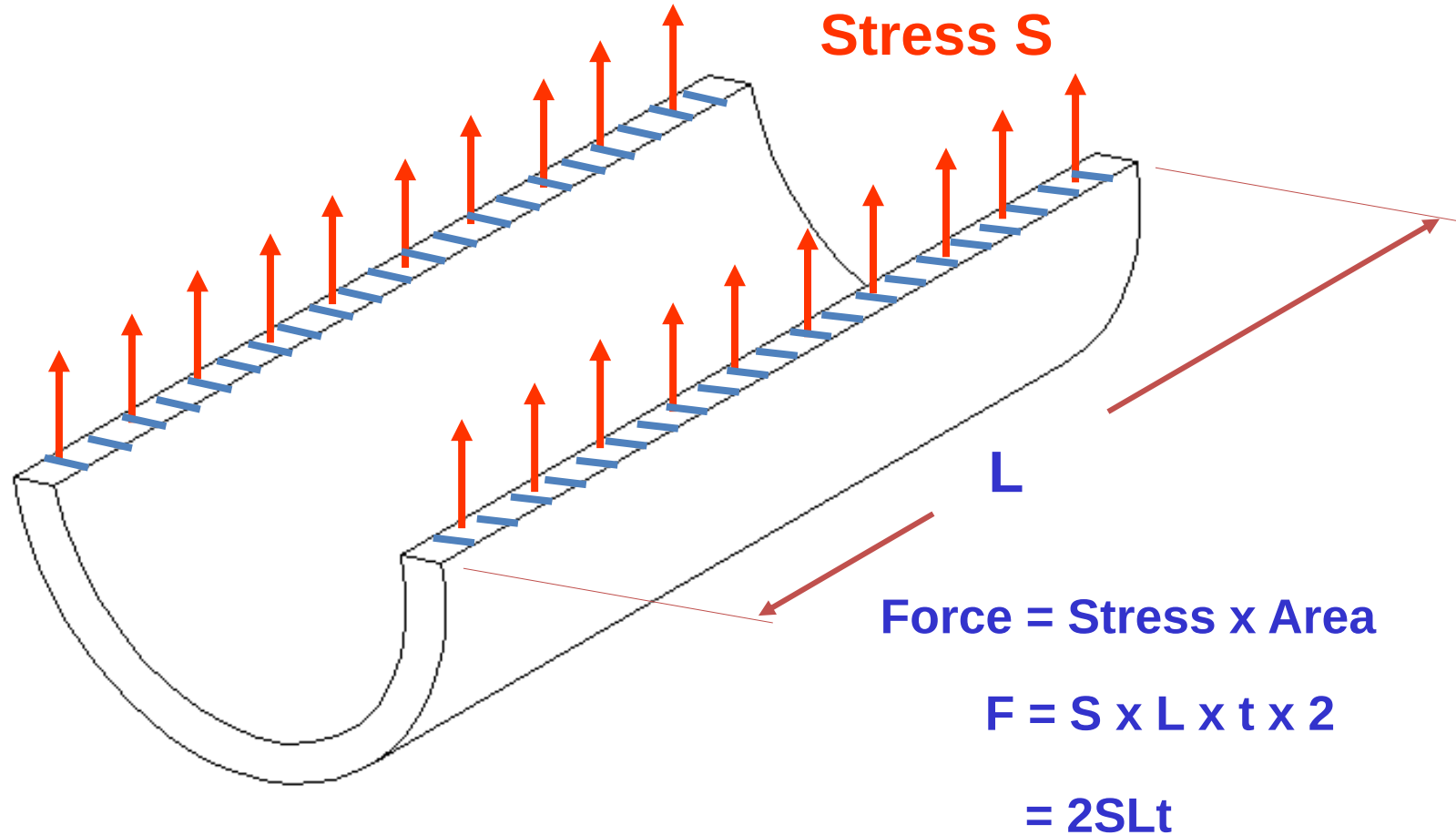
$$F = PDL$$

Internal Pressure stresses on cylindrical shell

This is resisted by the internal stress

$$\text{Area} = 2 \times t \times L$$

Stress S



Internal Pressure stresses on cylindrical shell

For equilibrium - Forces must be Equal

From pressure : $F = PDL$

From internal stress: $F = 2SLt$

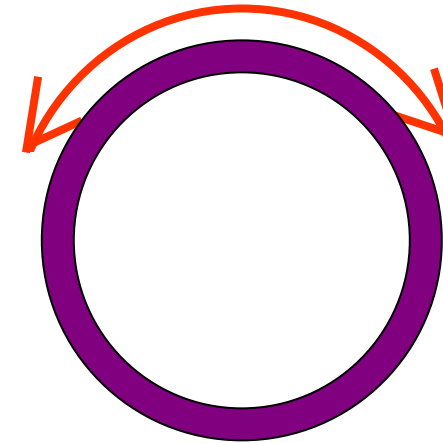
Equating therefore : $PDL = 2SLt$

Finally : $S_h = \frac{PD}{2t}$

Rearrange : $t = \frac{PD}{2 S_h}$

$$t = \frac{PR}{S_h}$$

This is known as the HOOP STRESS S_h



Simple General form

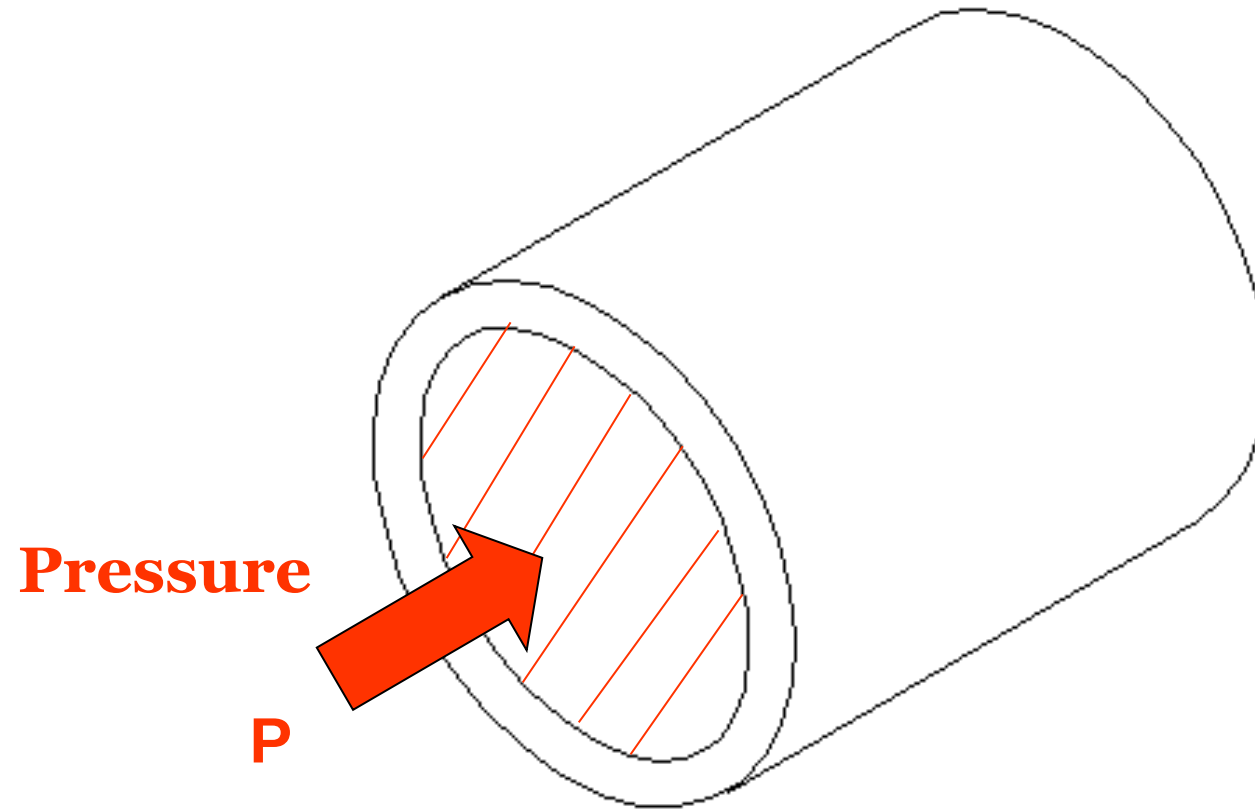
Internal Pressure stresses on cylindrical shell

Consider now the Axial or Longitudinal Stress

Force = Pressure x Area

$$\text{Area} = \frac{\pi.D^2}{4}$$

$$F = \frac{P.\pi.D^2}{4}$$



Internal Pressure stresses on cylindrical shell

Consider now the Axial or Longitudinal Stress

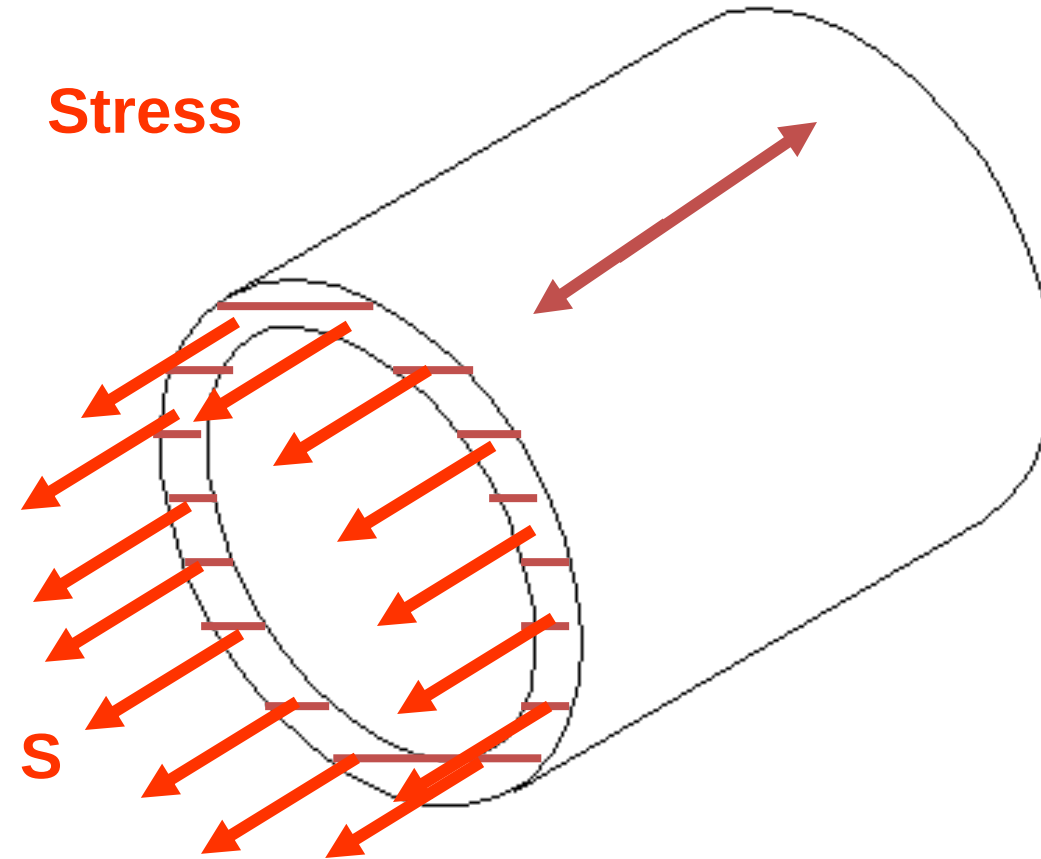
Force = Stress x Area

Area = $\pi.D.t$ (approx)

F = $S.\pi.D.t$

Equate F = $S.\pi.D.t = \frac{P.\pi.D^2}{4}$

Thus $S_L = \frac{P.D}{4t}$

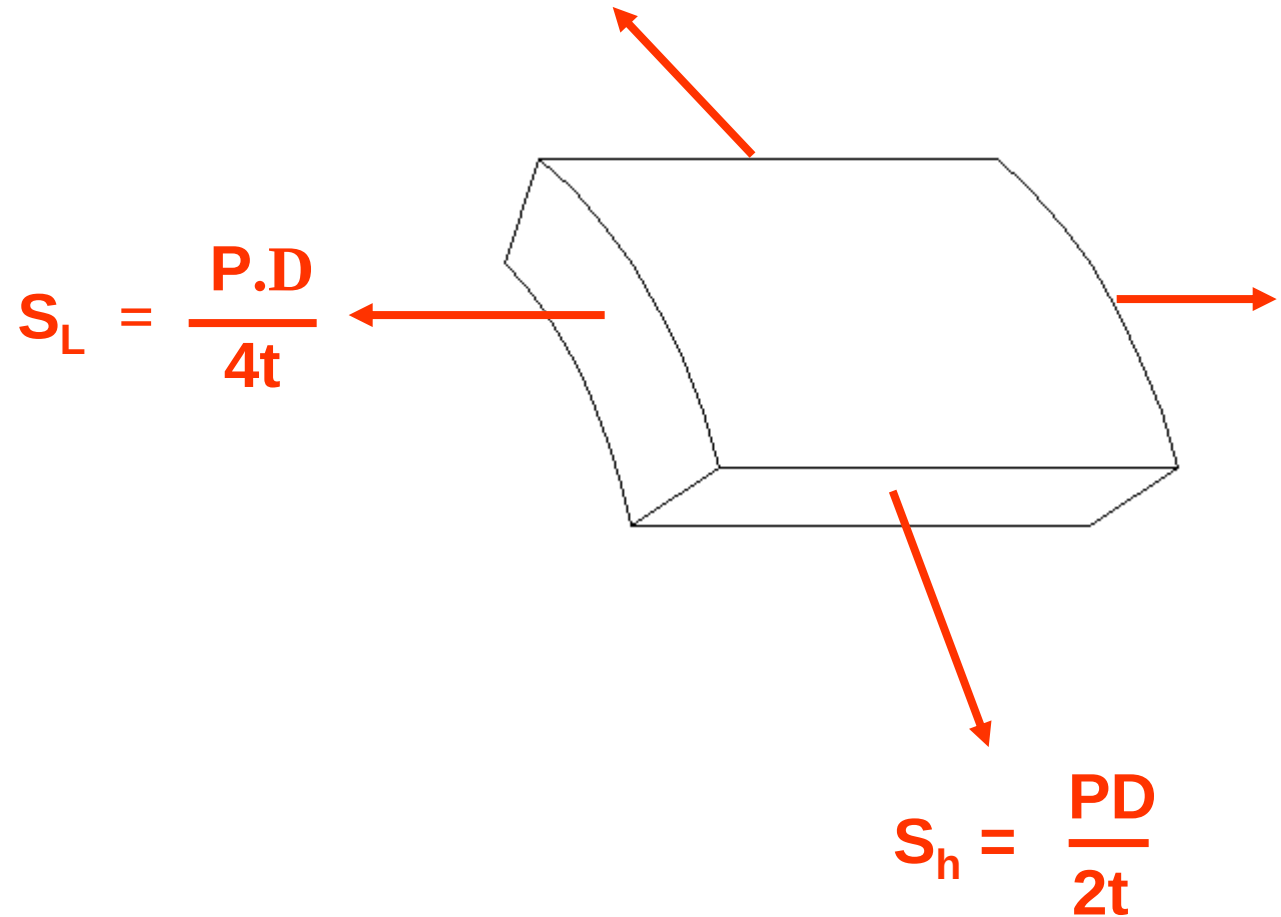


This is known as the Axial or Longitudinal Stress

Internal Pressure stresses on cylindrical shell

S_h is twice S_L

$$t = \frac{P R}{S}$$



Internal Pressure stresses on cylindrical shell

This is the formula per UG-27 in ASME BPVC Sec. VIII div.1 :

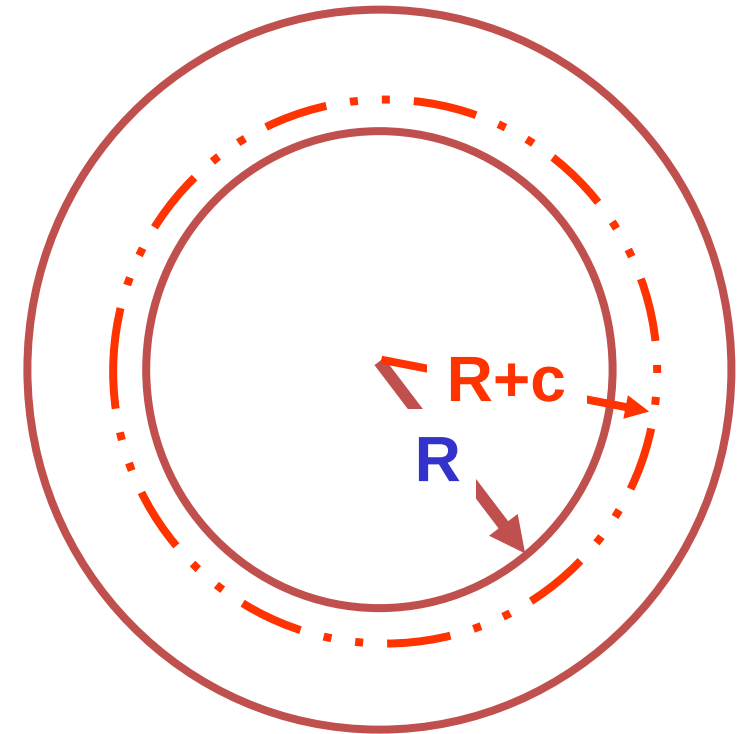
$$t = \frac{P.R}{S.E - 0.6.P}$$

P = Pressure psi

R = Radius inches

S = Design Stress psi

E = Welded Joint Efficiency



Calculations are done the **CORRODED** condition

Shell and Cylindrical Parts Under Internal Pressure

Internal Pressure Calculations – ASME BPVC Sec. VIII Div.1 UG-27

$$t = \frac{PR}{SE - 0.6 P} + CA + UT$$

Internal Pressure Calculations – ASME B31.3 [304.1.2]

$$t = \frac{PD}{2(SEW + PY)}$$

Internal Pressure Calculations – ASME B31.1 [104.1.2]

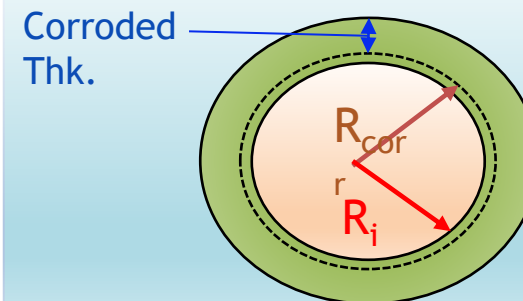
$$t_m = \frac{PD_o}{2(SEW + Py)} + A$$

Internal Pressure Calculations – ASME BPVC Sec.1 PG.27.2

$$t = \frac{PD}{2SE + 2yP} + C$$

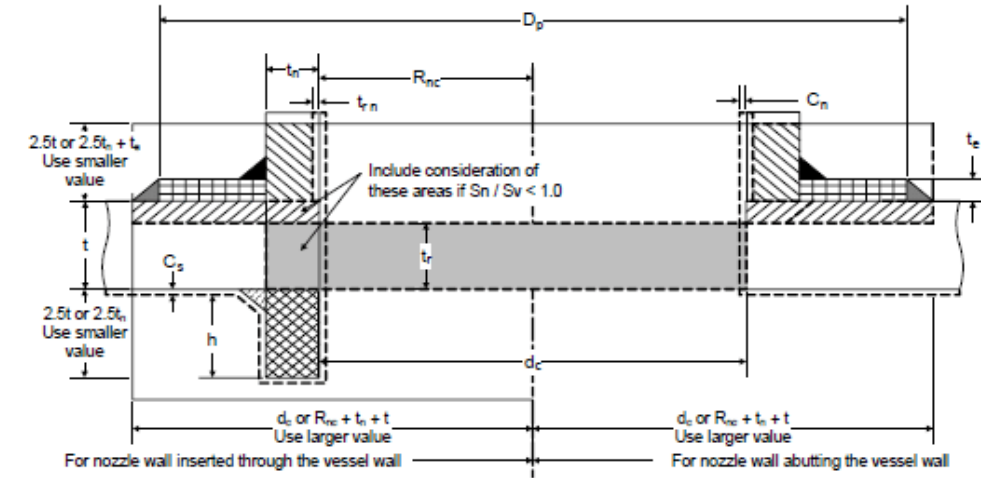
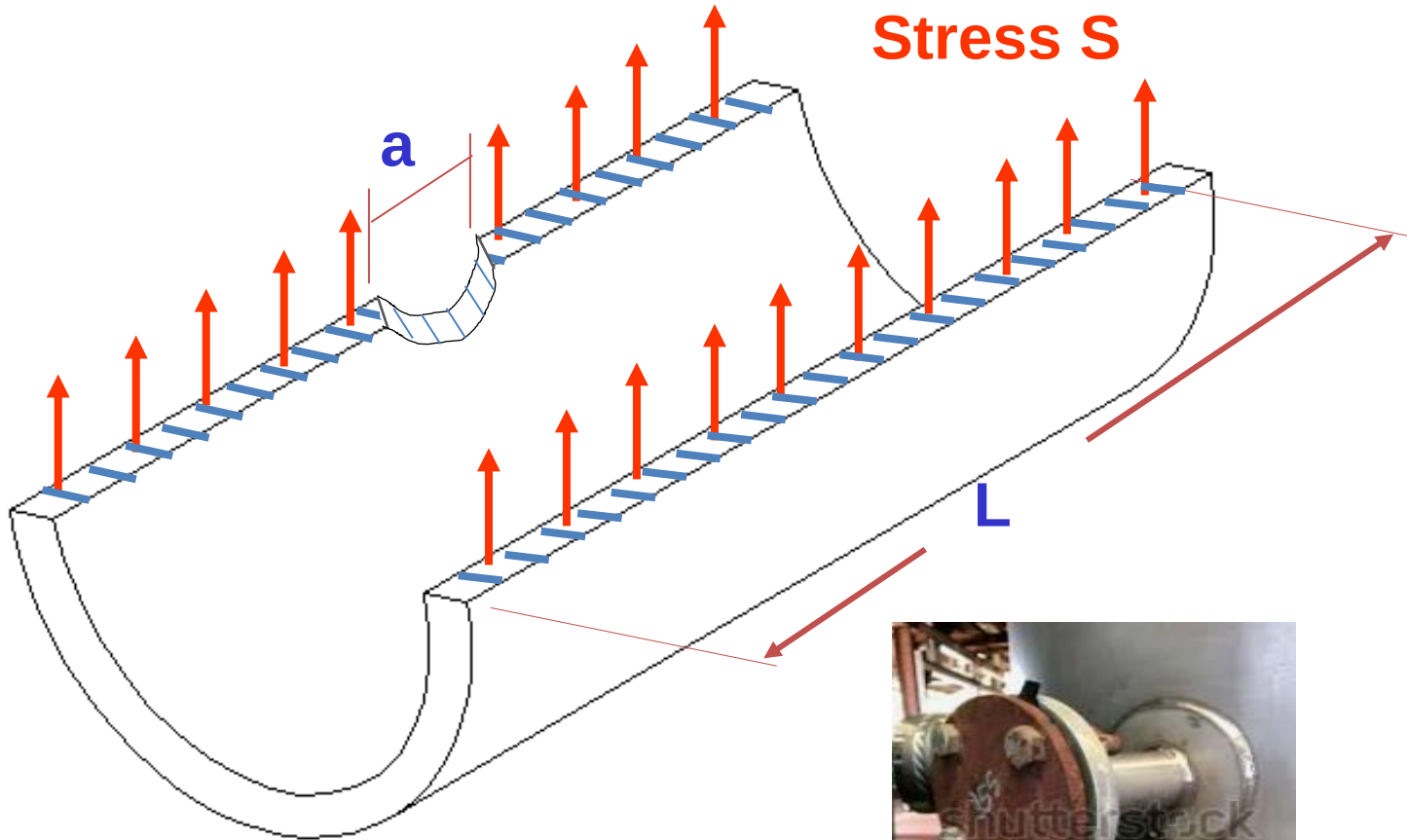
P: Internal Design Pressure
R: Radius (in corroded conditions)
S: Allowable stresses from tables in sec. II Part D
E: Joint Efficiency
CA: Corrosion Allowance
UT: Under tolerance (12.5% for standard pipes)

Calculations shall be in corroded conditions



Internal Pressure stresses on cylindrical shell - Shell Openings

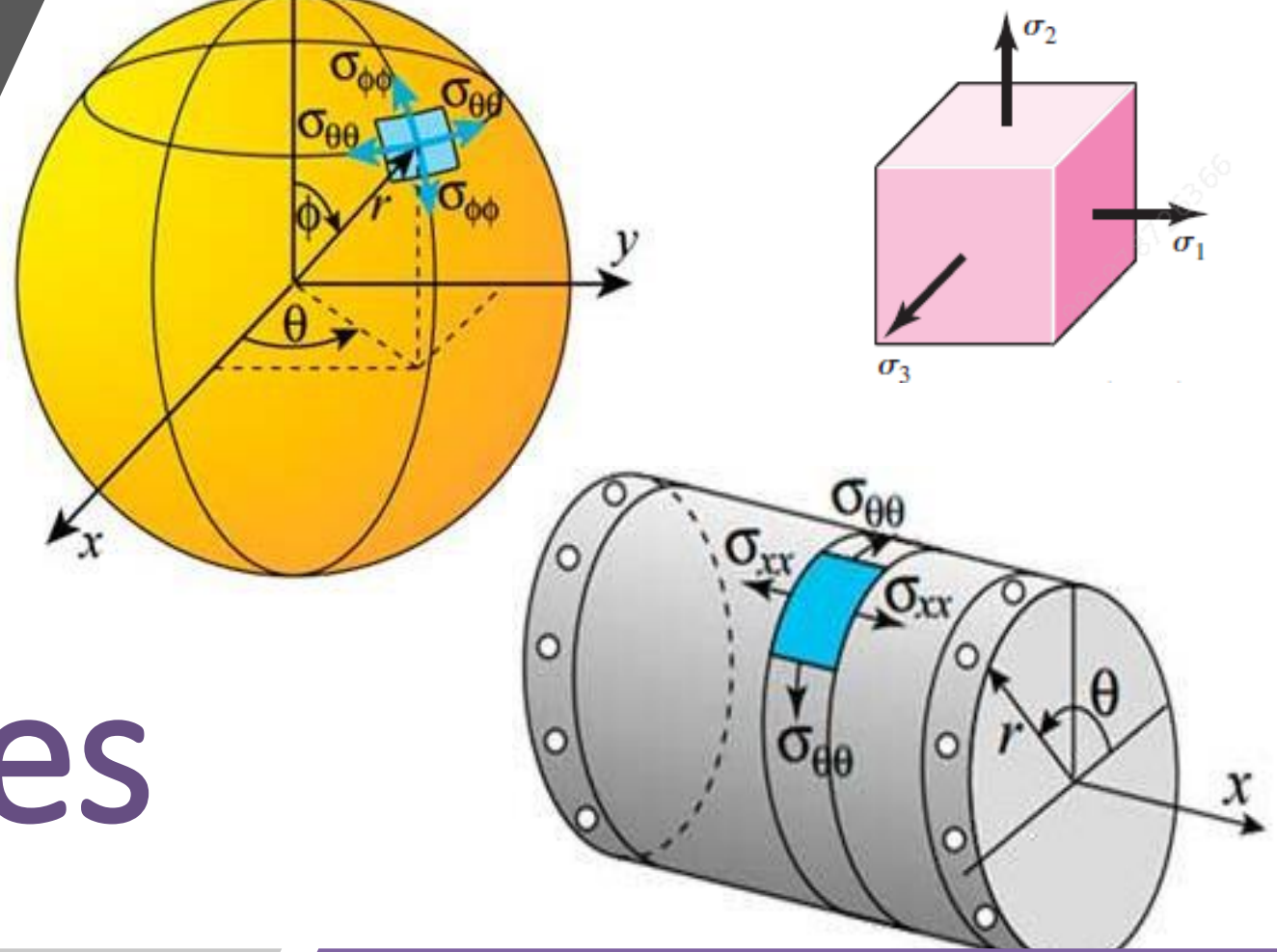
$$\text{Area} = 2 \times t \times L - a \times t$$



- A_r Required reinforcement area
- A_1 Available reinforcement area in the shell
- A_2 Available reinforcement area in the nozzle
- A_3 Available reinforcement area, inside nozzle projection
- $A_{4.1}$ Available reinforcement area in the nozzle to pad or vessel weld
- $A_{4.2}$ Available reinforcement area in the nozzle to weld, inside surface
- $A_{4.3}$ Available reinforcement area in the reinforcing pad attachment weld
- A_5 Available reinforcement area in the reinforcing pad

Area Replacement Calculations
ASME BPVC Sec. VIII div. 1 - UG 37

Failure Theories



Static Equipment Awareness Session

Reference: Shigley's Mechanical Engineering Design

Baher Elsheikh

Component Failure

Why do parts fail?

If we keep increasing the magnitude of load applied to a part, at some point it will fail.

How can we predict when the failure will occur?

What level does the stresses in the part need to reach for it to fail?

Failure theories help to answer

Failure

A part or equipment is said to be failed when it become unable to preform its design function. Which in pressure equipment is mainly to safely contain the pressure

Safe Design

Please remember that ultimate goal of design **codes** is to have **Safe** design and avoid **Failure**

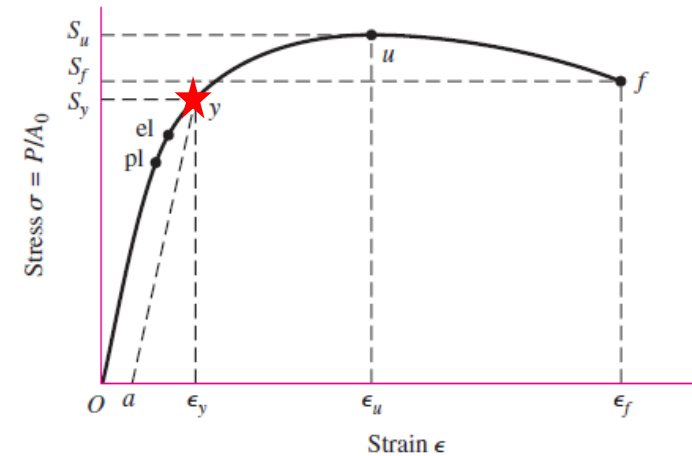


Ductile Material: Failure occurs at the onset of plastic deformation

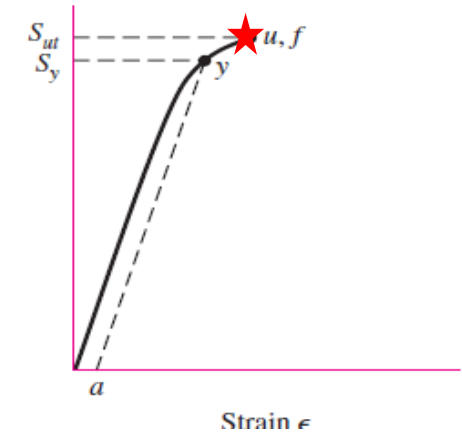
Brittle Material: Failure occurs at fracture

Structural metal behavior is typically classified as being **ductile or brittle**,). Ductile materials are normally classified such that $\epsilon_f \geq 0.05$ and have an identifiable yield strength that is often the same in compression as in tension ($S_{yt} = S_{yc} = S_y$).

Brittle materials, $\epsilon_f < 0.05$, do not exhibit an identifiable yield strength, and are typically classified by ultimate tensile and compressive strengths, S_{ut} and S_{uc} .



Ductile Material



Brittle Material



Failure Theories

Failure Prediction

There is no universal method to predict failure. Different **failure theories** developed using different approaches to account for the applied stress and to predict when the components will fail. Each working well under different circumstances

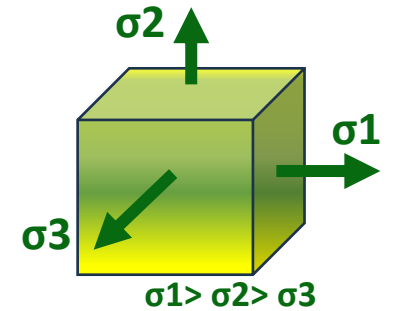
Failure theories defines stress at a point as function of normal stresses σ_1 , σ_2 , σ_3 and compare to material properties σ_{yield} or σ_{ultimate}

Most Common Failure Theories

Maximum Normal stress theory

Von Mises Theory

Maximum Shear Stress (Tresca) Theory



The appropriate equivalent stress calculation, whether on basis of the max. principle stress, Von Mises, maximum shear, maximum principal, or other possible governing stress criterion, may **depend on the material, temperature, stress, and stress state**.

Different codes adopt different failure theories which reflected on the applied rules and the factor of safety applied for σ_{yield} or σ_{ultimate} to get the allowable stress

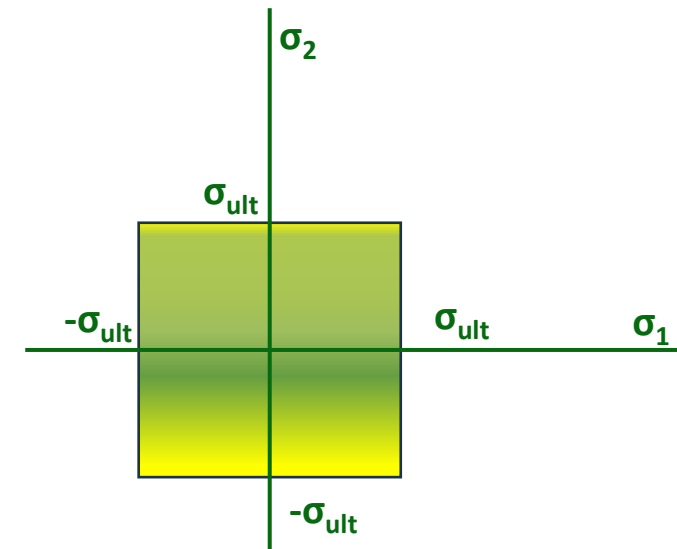
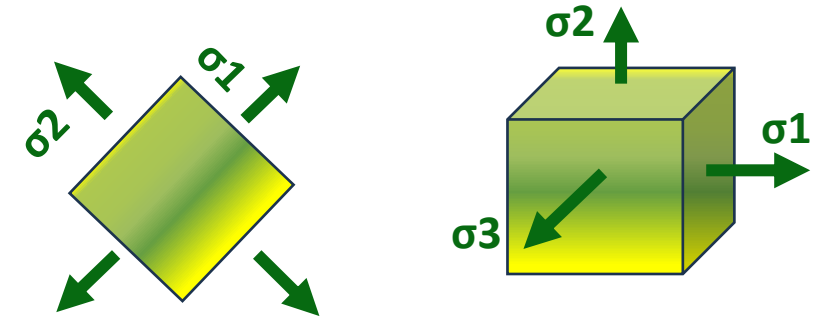
Maximum Normal Stress Theory - [Rankine Theory]

Maximum normal Stress theory (**Rankine Theory**) is the simplest theory where maximum principal stress σ_1 , σ_2 , σ_3 compared to material property (σ yield or σ Ultimate)

When the **Maximum Normal Stress** in any direction of reaches the **Strength** of the material - the part fails. Thus, finding the **Principal Stresses** at critical locations is important.

Theory is simple in calculations but not consistently matching with the experiments results, hence when used high safety margin should be considered

Basis of ASME BPVC sec. VIII Div1 With incorporation of safety factor



Theory concept: Failure will occur when stress lie outside the safe shaded area

Maximum Shear stress Theory (MSS) - [Tresca or Guest theory]

The maximum-shear-stress theory predicts that yielding begins whenever the maximum shear stress in any element **equals or exceeds the maximum shear stress** in a tension test specimen of the same material when that specimen begins to yield

As a strip of a ductile material is subjected to tension, slip lines (called **Lüder lines**) form at approximately 45° with the axis of the strip. These slip lines are the beginning of yield fracture lines are also seen at angles approximately 45°

Normal Stress $\sigma = F/A$,
Maximum shear stress $\tau_{\max} = \sigma/2$ occurs @ 45° from the tensile surface.

General state of stress $\sigma_1 \geq \sigma_2 \geq \sigma_3 = P/A$,
Maximum shear stress $\tau_{\max} = (\sigma_1 - \sigma_3) / 2 \geq \sigma_y / 2$
 $(\sigma_1 - \sigma_3) / 2 \geq \sigma_y / 2$... note yield strength in shear is $0.5 \sigma_y$

In plane stress mode, one principle stress is Zero and the others σ_A and σ_B
Three cases:

[Case-1] $\sigma_A \geq \sigma_B \geq 0$ $\sigma_A \geq \sigma_y$

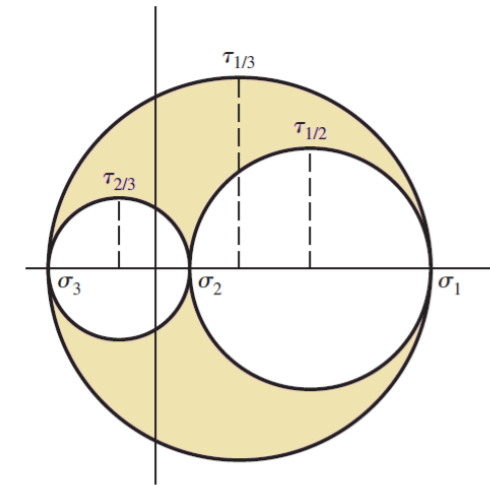
[Case-2] $\sigma_A \geq 0 \geq \sigma_B$ $\sigma_A - \sigma_B \geq \sigma_y$

[Case-3] $0 \geq \sigma_A \geq \sigma_B$ $\sigma_B \leq -\sigma_y$

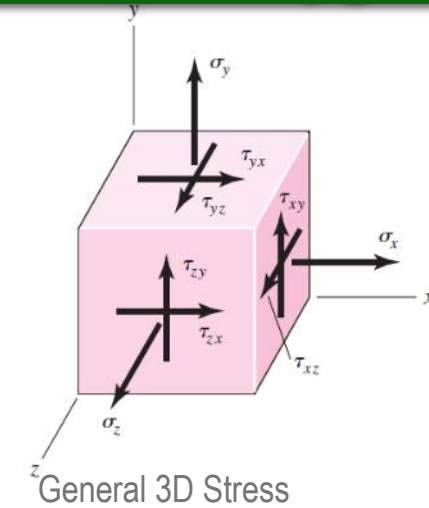
$(\sigma_1 - \sigma_3) / 2 \geq \sigma_y / 2$... note yield strength in shear is $0.5 \sigma_y$



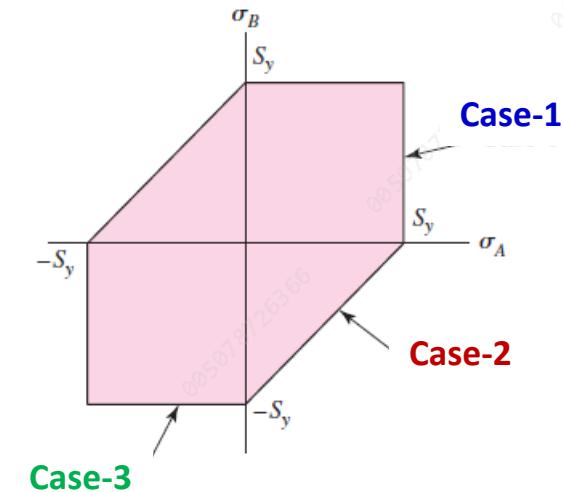
Lüders lines on steel specimen



Mohr's circles for 3D stress.



General 3D Stress



Von Mises - Distortion Energy Theory (DE)

The distortion-energy theory predicts that yielding occurs when the distortion strain energy per unit volume reaches or exceeds the distortion strain energy per unit volume for yield in simple tension or compression of the same material

The distortion-energy (DE) theory originated from the observation that ductile materials stressed hydrostatically exhibited yield strengths greatly in excess of the values given by the simple tension test.

Strain energy “U” is a form of potential energy stored in a structural part due to elastic deformation

$$U = \frac{1}{2} V \sigma \epsilon \quad \dots = \frac{1}{2} \sigma \epsilon \text{ per unit volume}$$

Distortion energy **U_d** is the portion of strain energy corresponding to the effect of deviatoric stresses which cause shape deflection (Principle stress – hydrostatic stress (σ_{avg}))

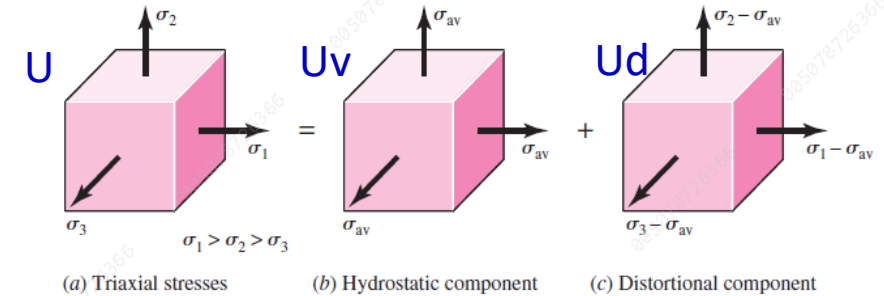
$$U_d = U - U_v$$

U_d: distortion energy

U: Strain energy

U_v: Energy result on only volume change

$$u = \frac{1}{2E} [\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\nu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)]$$



(a) Element with triaxial stresses; this element undergoes both volume change and angular distortion. (b) Element under hydrostatic tension undergoes only volume change. (c) Element has angular distortion without volume change.

Distortion energy general equation

$$u_d = u - u_v = \frac{1 + \nu}{3E} \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2} \right]$$

Yielding during simple tensile test, (uniaxial stress applied = yield strength) and other two stress = 0 $\sigma_1 = \sigma_y$ & $\sigma_2 = \sigma_3 = 0$

Distortion energy at yielding

$$u_d = \frac{1 + \nu}{3E} S_y^2$$

Equivalent Von Mises Stresses

So yield is predicted if

$$\left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2} \right]^{1/2} \geq S_y$$

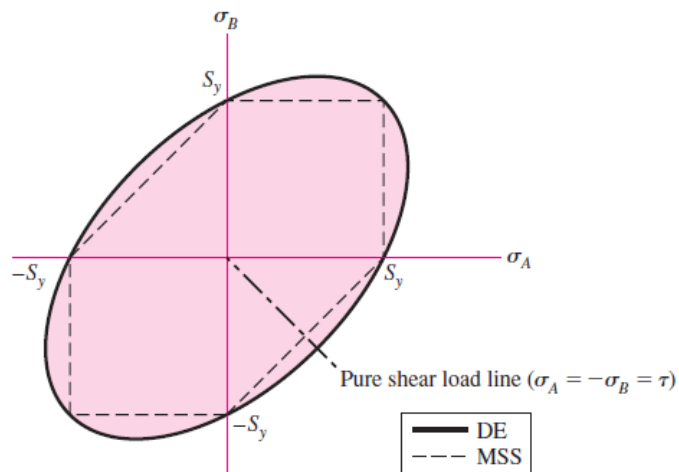
Von Mises - Distortion Energy Theory (DE)

Equivalent Von Mises stress

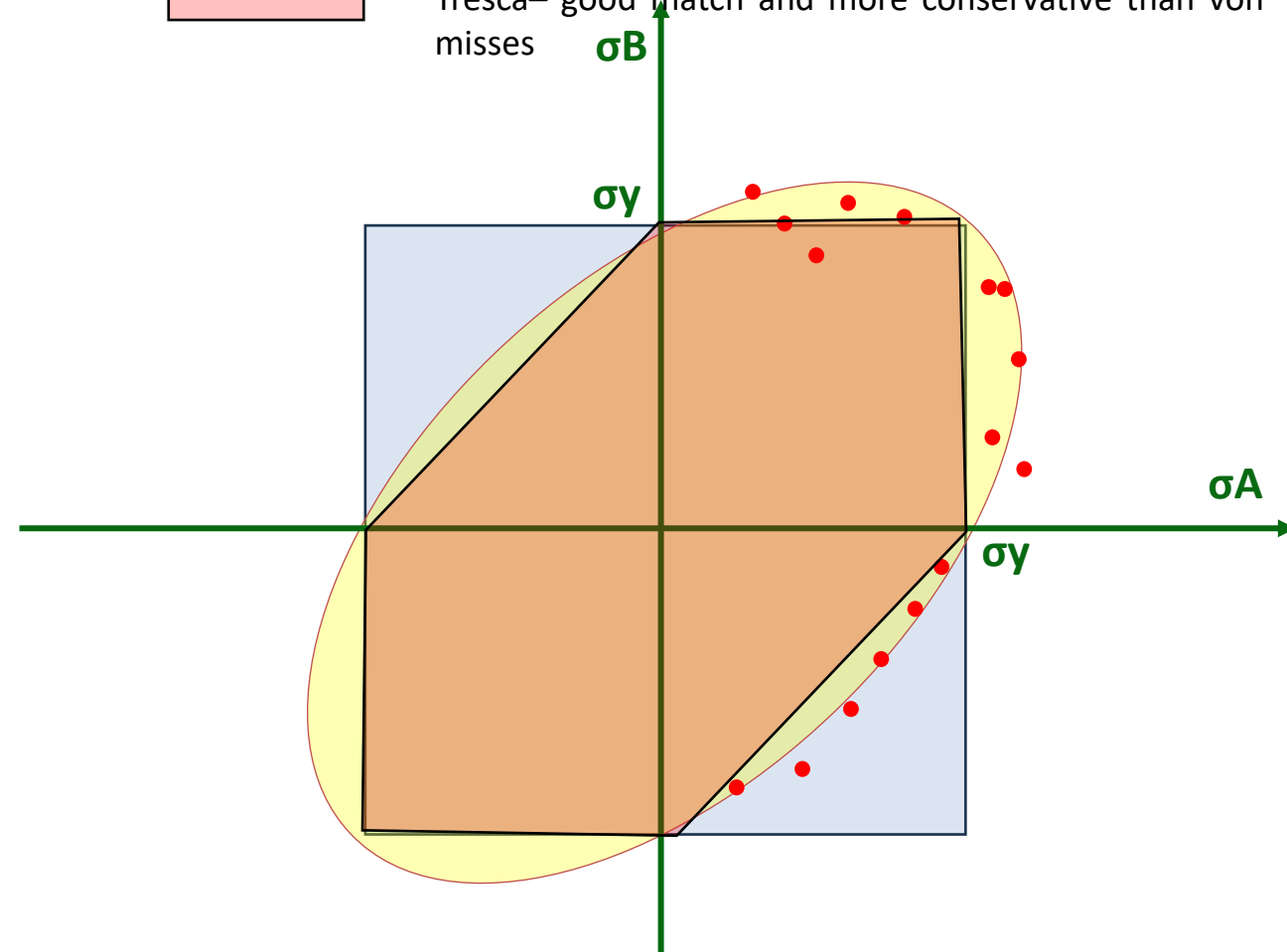
$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2} \right]^{1/2}$$

Equivalent Von Mises stress for plane stress condition

$$\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2}$$



- maximum Principal Stress – potentially unsafe
- Von mises –Best match
- Tresca– good match and more conservative than von mises



Comparison of 3 theories against experimental data

Shell and Cylindrical Parts Under Internal Pressure

Internal Pressure Calculations – ASME BPVC Sec. VIII Div.1 UG-27

$$t = \frac{PR}{SE - 0.6 P} + CA + UT$$

Internal Pressure Calculations – ASME B31.3 [304.1.2]

$$t = \frac{PD}{2(SEW + PY)}$$

Internal Pressure Calculations – ASME B31.1 [104.1.2]

$$t_m = \frac{PD_o}{2(SEW + Py)} + A$$

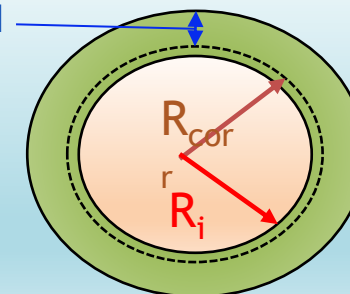
Internal Pressure Calculations – ASME BPVC Sec.1 PG.27.2

$$t = \frac{PD}{2SE + 2yP} + C$$

P: Internal Design Pressure
R: Radius (in corroded conditions)
S: Allowable stresses from tables in sec. II Part D
E: Joint Efficiency
CA: Corrosion Allowance
UT: Under tolerance (12.5% for standard pipes)

Calculations shall be in corroded conditions

Corroded
Thk.



Codes Design Margin / Safety Factor [SF]

ASME BPVC Sec. I (Power Boiler) : **SF 3.5**

ASME BPVC Sec. III (Nuclear) : **SF 3**

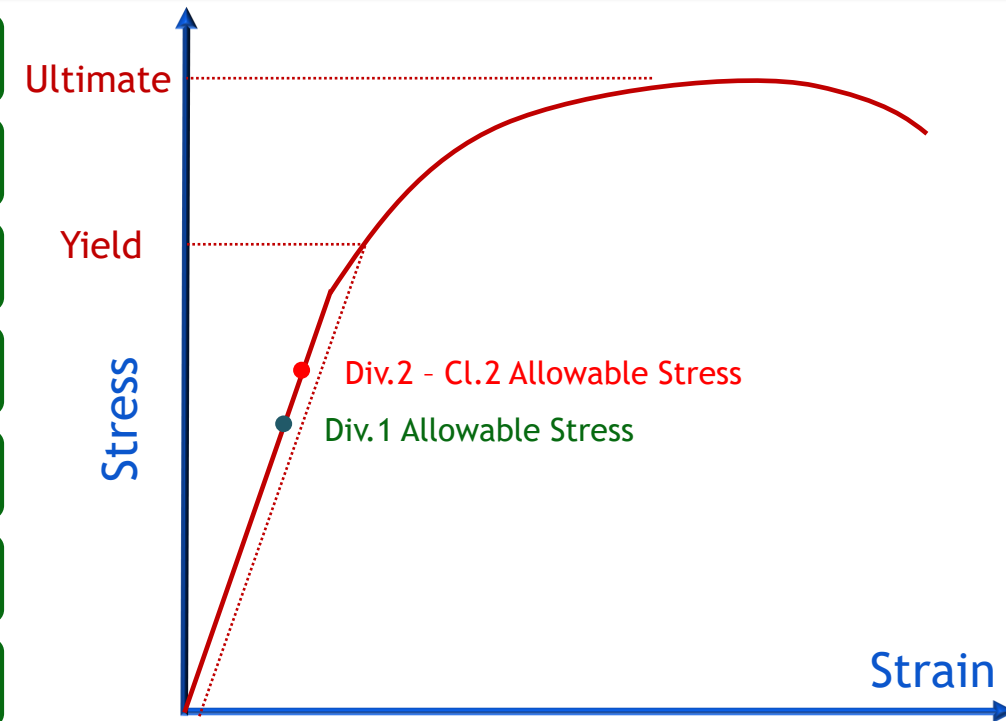
ASME BPVC Sec. VIII Div.1 (Pressure Vessel) : **SF 3.5**

ASME BPVC Sec. VIII Div. 2 Class 1 : **SF 3**

ASME BPVC Sec. VIII Div.2 Class 2 : **SF 2.4**

ASME B 31.1 (Power Piping): **SF 3.5**

ASME B31.3 (Process Piping): **SF 3**



Changes History of SF of ASME BPVC

In 1915

SF = 5

In 1943

SF = 4

World War II
Period

1950

SF = 5

In 1951

SF = 4

In 1999

SF = 3.5

Note: Above comparison are based on the SF as a function of ultimate strength for illustration purpose only Allowable stresses are established based on multiple criteria as a percent of ultimate strength, yield strength or based on accumulated strain in creep conditions

$$SF = \frac{\text{Tensile Strength}}{\text{Allowable Stress}}$$

Q 5

It is generally accepted to use tabulated values of allowable stresses in one code and design rules of other code

a. YES

b. NO

Current Criteria for ASME BPVC Sec. VIII Div.1

- Room-temperature tensile strength / 3.5
- Room-temperature yield strength / 1.5
- The stress required to cause a creep rate of 0.0001%/1000 hours
- The average stress to cause rupture at 100,000 hours / 1.5
- The minimum stress to cause rupture at 100,000 hours / 1.25

**Design Code is
not a
Handbook**



Working Example

ASME BPVC Sec VIII div.1 – UG-27

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Shell and Cylindrical Parts Under Internal Pressure

Internal Pressure Calculations - ASME BPVC Sec. VIII Div.1 UG-27

$$t = \frac{PR}{SE - 0.6 P} + CA + UT$$

P: Internal Design Pressure

R: Radius (in corroded conditions)

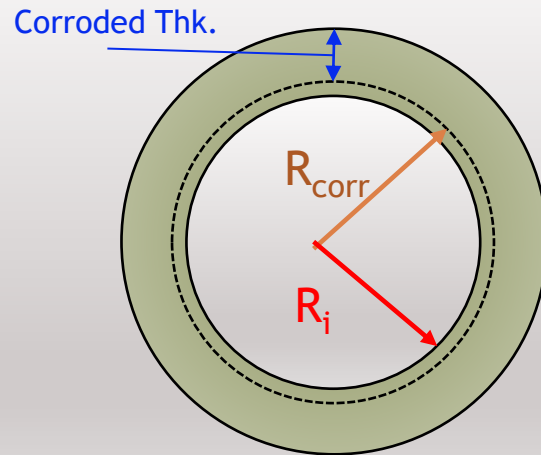
S: Allowable stresses from tables in sec. II Part D

E: Joint Efficiency

CA: Corrosion Allowance

UT: Under tolerance (12.5% for standard pipes)

Calculations shall be in corroded conditions



Overview of Internal Pressure Calculations per ASME BPVC Sec. VIII div.1

The required thickness of a cylindrical shell based on circumferential stress given a pressure (UG-27(c)(1)).

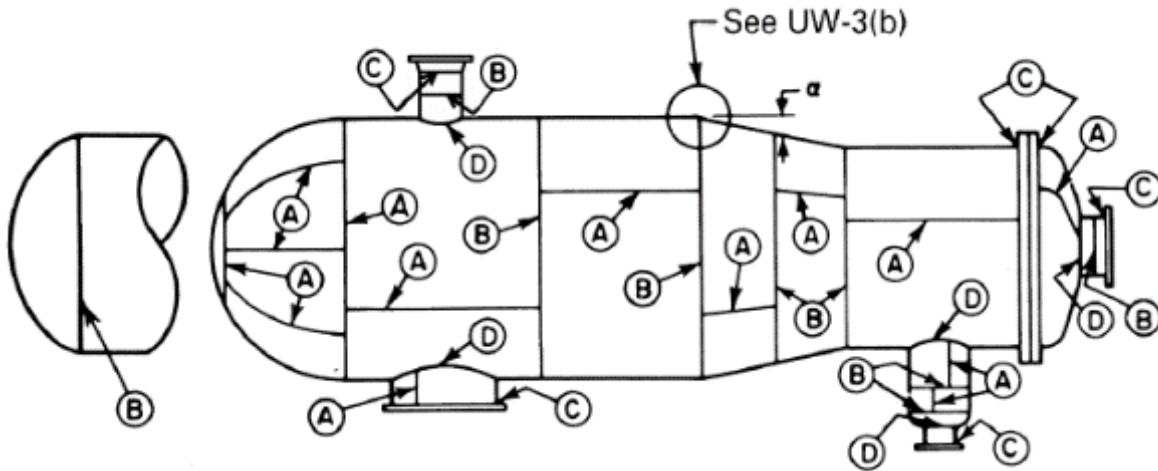
The vessel part **Maximum Allowable Working Pressure (MAWP)** for a cylindrical shell based on circumferential stress given a metal thickness (UG-27(c)(1)).

The required thickness of a head (ellipsoidal, Torispherical and hemispherical) given a pressure. (UG-32 (d), (e), & (f)).

The vessel part **MAWP** for a head (ellipsoidal, Torispherical and hemispherical) given a metal thickness using paragraphs UG-32 (d), (e), & (f).

Weld Joint Efficiency

Illustration of Welded Joint Locations Typical of Categories A, B, C, and D



The term “Category” as used herein defines the location of a joint in a vessel, but not the type of joint.

Category A Longitudinal welded joints within the main shell, communicating chambers, transitions in diameter, or nozzles; any welded joint within a sphere, within a formed or flat head, or within the side plates of a flat-sided vessel; circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameters, to nozzles, or to communicating chambers.

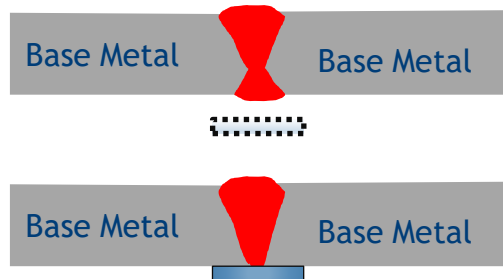
Category B Circumferential welded joints within the main shell, communicating chambers, nozzles, or transitions in diameter including joints between the transition and a cylinder at either the large or small end; circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers

Category C Welded joints connecting flanges, Van Stone laps, tubesheets, or flat heads to main shell, to formed heads, to transitions in diameter, to nozzles, or to communicating chambers any welded joint connecting one side plate to another side plate of a flat sided vessel

Category D Welded joints connecting communicating chambers or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels, and those joints connecting nozzles to communicating chambers (for nozzles at the small end of a transition in diameter, see Category B)

Weld Joint Efficiency

Type 1



Double Welded butt joint or equivalent. Backing strip if used must be removed.

Type 2



Single Welded butt joint with backing strip remains in place

Type 3



Single Welded butt joint without use of backing strip

Type 4

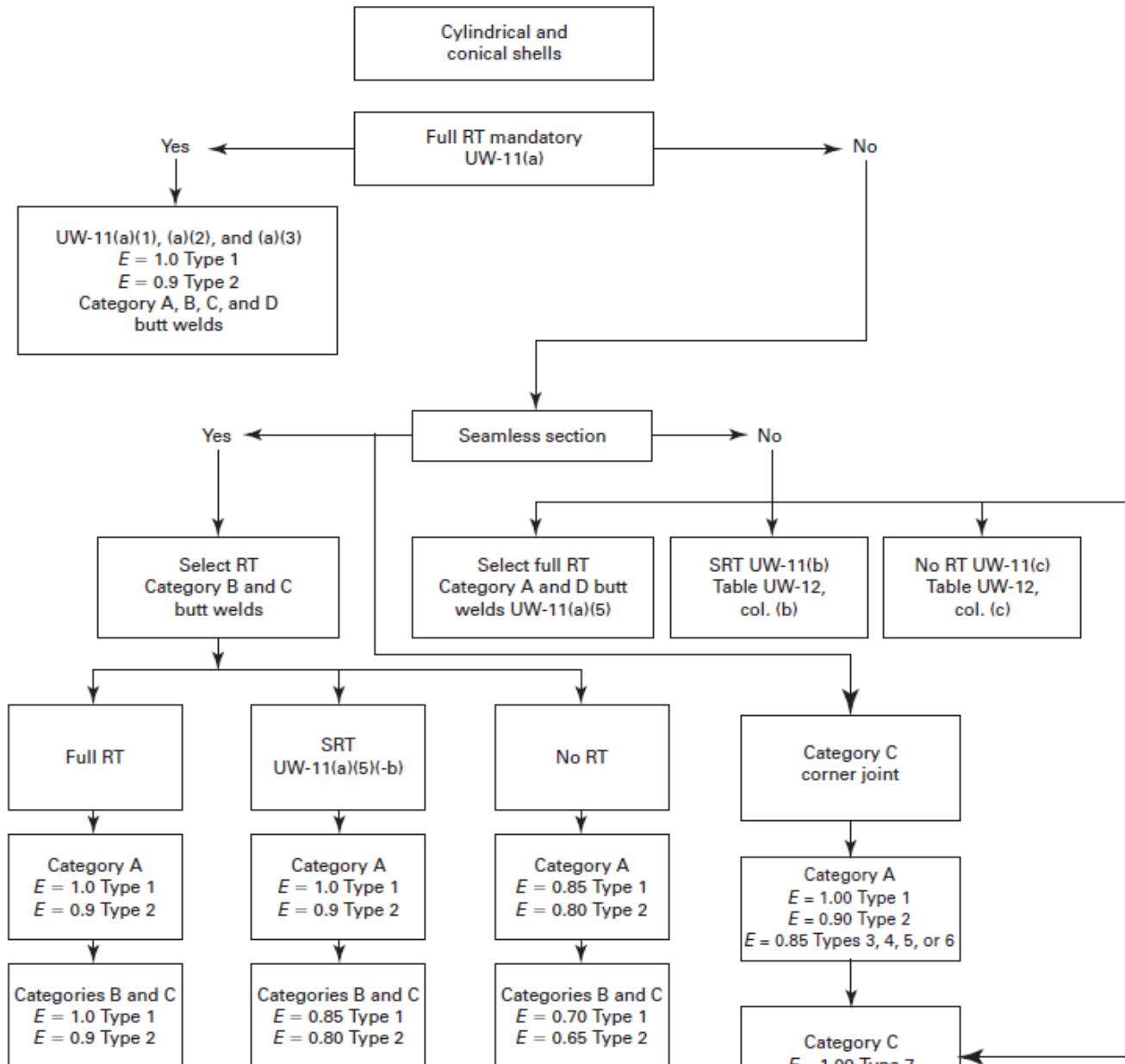


Double full fillet lap joint

Table UW-12
Maximum Allowable Joint Efficiencies for Welded Joints

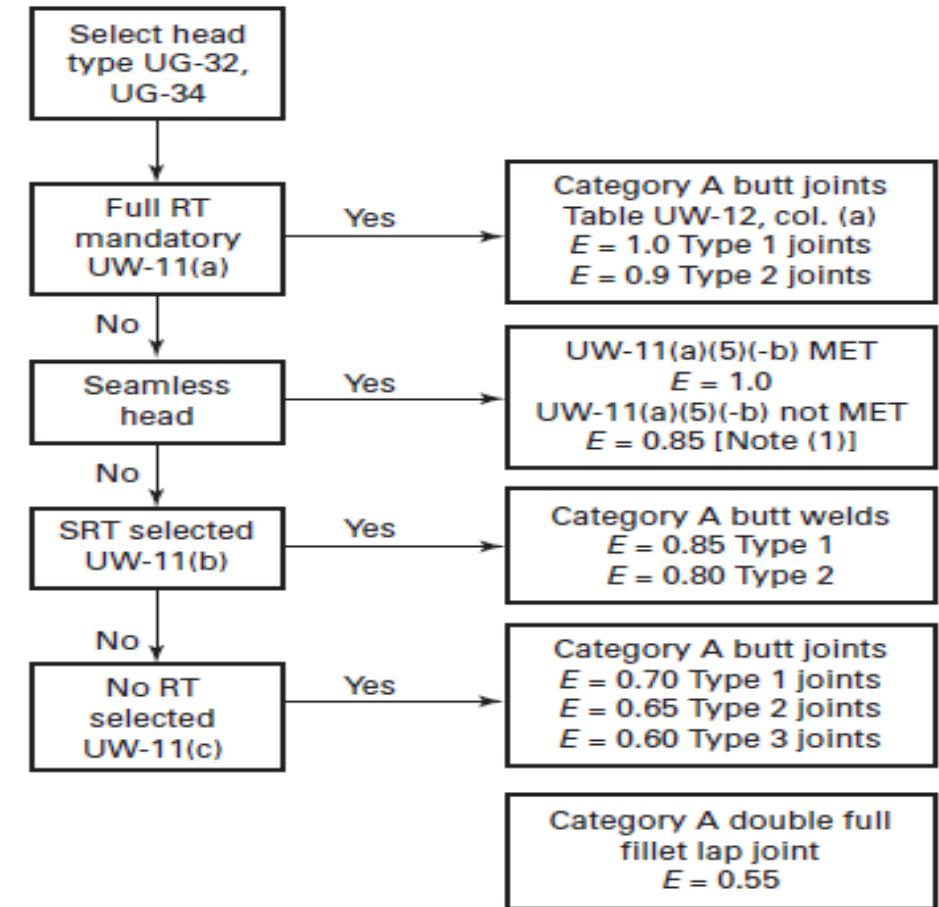
Type No.	Joint Description	Limitations	Joint Category	Extent of Radiographic or Ultrasonic Examination [Note (1), Note (2), Note (3)]		
				(a) Full [Note (4)]	(b) Spot [Note (5)]	(c) None
(1)	Butt joints as attained by double-welding or by other means that will obtain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips that remain in place are excluded.	None	A, B, C, and D	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	(a) None except as in (b) below (b) Circumferential butt joints with one plate offset; see UW-13(b)(4) and Figure UW-13.1, sketch (i)	A, B, C, and D A, B, and C	0.90 0.90	0.80 0.80	0.65 0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential butt joints only, not over $\frac{3}{8}$ in. (16 mm) thick and not over 24 in. (600 mm) outside diameter	A, B, and C	NA	NA	0.60
(4)	Double full fillet lap joint	(a) Longitudinal joints not over $\frac{5}{16}$ in. (10 mm) thick (b) Circumferential joints not over $\frac{5}{16}$ in. (16 mm) thick	A B and C [Note (6)]	NA NA	NA NA	0.55 0.55
(5)	Single full fillet lap joints with plug welds conforming to UW-17	(a) Circumferential joints [Note (7)] for attachment of heads not over 24 in. (600 mm) outside diameter to shells not over $\frac{1}{2}$ in. (13 mm) thick (b) Circumferential joints for the attachment to shells of jackets not over $\frac{5}{8}$ in. (16 mm) in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than $1\frac{1}{2}$ times the diameter of the hole for the plug.	B C	NA NA	NA NA	0.50 0.50
(6)	Single full fillet lap joints without plug welds	(a) For the attachment of heads convex to pressure to shells not over $\frac{5}{16}$ in. (16 mm) required thickness, only with use of fillet weld on inside of shell; or (b) for attachment of heads having pressure on either side, to shells not over 24 in. (600 mm) inside diameter and not over $\frac{1}{4}$ in. (6 mm) required thickness with fillet weld on outside of head flange only	A and B A and B	NA NA	NA NA	0.45 0.45
(7)	Corner joints, full penetration, partial penetration, and/or fillet welded	As limited by Figure UW-13.2 and Figure UW-16.1	C and D [Note (8)]	NA	NA	NA
(8)	Angle joints	Design per U-2(g) for Category B and C joints	B, C, and D	NA	NA	NA

Figure L-1.4-1
Joint Efficiency and Weld Joint Type — Cylinders and Cones



Summary for Joint Efficiency

Figure L-1.4-2
Joint Efficiency and Weld Joint Type — Heads



Summery for Joint Efficiency

Type	Category	Full	Spot	None
1	A, B, C, D	1	0.85	0.7
2	A, B, C, D	0.9	0.8	0.65
3	A, B, C	N/A	N/A	0.6

Sample Calculations

Internal Pressure Calculations – ASME BPVC Sec. VIII Div.1 UG-27

$$t = \frac{PR}{SE - 0.6 P} + CA + UT$$

P: Internal design pressure: 2.5 Mpa

T: design Temperature: 90 °C

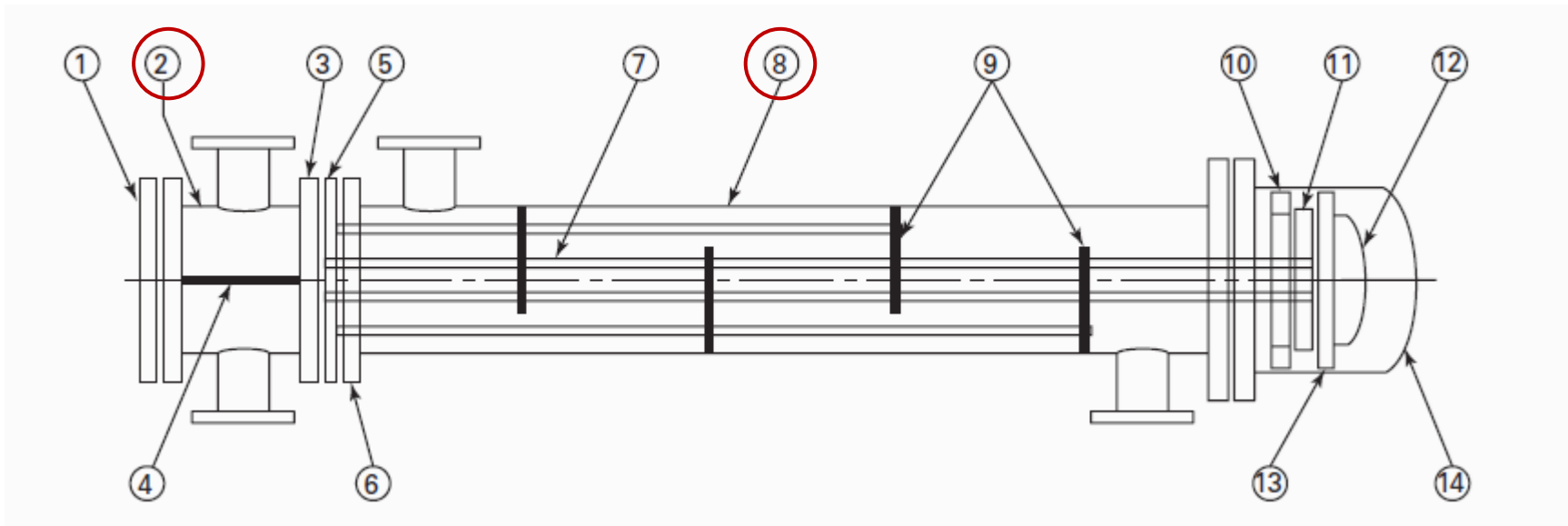
R: Radius (in corroded conditions) : 360 mm

Shell Material: Carbon Steel SA-516 Gr.70 N

E: Joint Efficiency: RT-3 Spot

CA: Corrosion Allowance: 3 mm

UT: NA



Sample Calculations

Table 1A (Cont'd)
Section I; Section III, Classes 2 and 3;* Section VIII, Division 1; and Section XII
Maximum Allowable Stress Values, S , for Ferrous Materials
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Nominal Composition	Product Form	Spec. No.	Type/Grade	Alloy Desig./UNS No.	Class/Condition/ Temper	Size/ Thickness, mm	P-No.	Group No.
1	Carbon steel	Sheet	SA-414	E	K02704	...	...	1	1
2	Carbon steel	Plate	SA-662	B	K02203	...	...	1	1
3	Carbon steel	Plate	SA-537	...	K12437	1	$65 < t \leq 100$	1	2
4	Carbon steel	Wld. pipe	SA-691	CMSH-70	K12437	...	$65 < t \leq 100$	1	2
5	Carbon steel	Plate, bar	SA/CSA-G40.21	44W	...	...	$t \leq 200$	1	1
6	Carbon steel	Plate, bar	SA/CSA-G40.21	50W	...	...	$t \leq 150$	1	1
7	Carbon steel	Plate	SA/AS 1548	PT460N	...	Normalized	≤ 150	1	1
8	Carbon steel	Plate	SA/AS 1548	PT460NR	...	Norm. rld.	≤ 150	1	1
9	Carbon steel	Forgings	SA/EN 10222-2	P280GH	...	NT or QT	$35 < t \leq 160$	1	1
10	Carbon steel	Plate	SA/EN 10028-2	P295GH	...	...	$60 < t \leq 100$	1	1
11	Carbon steel	Plate	SA/EN 10028-2	P295GH	...	...	≤ 60	1	1
12	Carbon steel	Forgings	SA/EN 10222-2	P280GH	...	Normalized	$t \leq 35$	1	1
13	Carbon steel	Plate	SA/GB 713	Q345R	...	...	$150 < t \leq 200$	1	2
14	Carbon steel	Plate	SA/EN 10028-2	P355GH	...	...	$150 < t \leq 250$	1	2
15	Carbon steel	Plate	SA/GB 713	Q345R	...	...	$100 < t \leq 150$	1	2
16	Carbon steel	Plate	SA/EN 10028-2	P355GH	...	...	$100 < t \leq 150$	1	2
17	Carbon steel	Plate	SA-455	...	K03300	...	$15 < t \leq 20$	1	2
18	Carbon steel	Bar	SA-675	70	...	...	...	1	2
19	Carbon steel	Forgings	SA-105	...	K03504	...	...	1	2
20	Carbon steel	Forgings	SA-181	...	K03502	70	...	1	2
21	Carbon steel	Castings	SA-216	WCB	J03002	...	...	1	2
22	Carbon steel	Forgings	SA-266	2	K03506	...	...	1	2
23	Carbon steel	Forgings	SA-266	4	K03017	...	...	1	2
24	Carbon steel	Forgings	SA-350	LF2	K03011	1	...	1	2
25	Carbon steel	Forgings	SA-350	LF2	K03011	2	...	1	2
26	Carbon steel	Forgings	SA-508	1	K13502	...	...	1	2
27	Carbon steel	Forgings	SA-508	1A	K13502	...	...	1	2
28	Carbon steel	Forgings	SA-541	1	K03506	...	...	1	2
29	Carbon steel	Forgings	SA-541	1A	K03020	...	...	1	2
30	Carbon steel	Cast pipe	SA-660	WCB	J03003	...	...	1	2
31	Carbon steel	Forgings	SA-765	II	K03047	...	...	1	2
32	Carbon steel	Plate	SA-515	70	K03101	...	...	1	2
33	Carbon steel	Plate	SA-516	70	K02700	...	...	1	2
34	Carbon steel	Wld. pipe	SA-671	CB70	K03101	...	...	1	2
35	Carbon steel	Wld. pipe	SA-671	CC70	K02700	...	...	1	2



Table 1A (Cont'd)
Section I; Section III, Classes 2 and 3;* Section VIII, Division 1; and Section XII
Maximum Allowable Stress Values, S , for Ferrous Materials
(*See Maximum Temperature Limits for Restrictions on Class)

Line	Maximum Allowable Stress, MPa (Multiply by 1000 to Obtain kPa), for Metal Temperature, °C, Not Exceeding														
No.	40	65	100	125	150	200	250	300	325	350	375	400	425	450	475
1	128	128	128	128	128	128	128	128	128	127	114	95.1	79.6	63.1	45.7
2	128	128	128	128	128	128	128	128	128	127	114	...	...	...	...
3	128	128	128	128	128	128	128	128	128	127	114	...	...	...	...
4	128	128	128	128	128	128	128	128	128	127	114	...	...	...	...
5	128	128	128	128	128	128	128	128	128	128	...	...	...	...	...
6	128	128	128	128	128	128	128	128	128	128	...	...	...	...	...
7	131	131	131	131	131	131	131	131	131	129	114	95.1	79.6	63.2	45.3
8	131	131	131	131	131	131	131	131	131	129	114	95.1	79.6	63.2	45.3
9	131	131	131	131	131	131	131	131	128	124	120	101	83.9	67.0	51.1
10	131	131	131	131	131	131	131	131	131	127	112	96.2	79.1	62.1	46.0
11	131	131	131	131	131	131	131	131	131	127	112	96.2	79.1	62.1	46.0
12	131	131	131	131	131	131	131	131	131	131	131	101	83.9	67.0	51.1
13	134	134	134	134	134	134	134	134	134	130	123	101	83.8	67.1	...
14	134	134	134	134	134	134	134	134	134	134	123	101	83.8	67.1	51.0
15	137	137	137	137	137	137	137	137	137	137	123	101	83.8	67.1	...
16	137	137	137	137	137	137	137	137	137	137	123	101	83.8	67.1	51.0
17	138	138	138	138	138	137	132	126	122	119	...	...	...	...	...
18	138	138	138	138	138	137	132	126	122	119	114	101	83.9	67.0	51.1
19	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
20	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
21	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
22	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
23	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
24	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
25	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
26	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
27	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
28	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
29	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
30	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
31	138	138	138	138	138	138	136	129	125	122	117	101	83.9	67.0	51.1
32	138	138	138	138	138	138	138	136	132	128	123	101	83.8	67.1	51.0
33	138	138	138	138	138	138	138	136	132	128	123	101	83.8	67.1	51.0
34	138	138	138	138	138	138	138	136	132	128	123	...	...	...	...
35	138	138	138	138	138	138	138	136	132	128	123	...	...	...	...

Get the allowable stress from ASME Sec. II Part D , table-
at the design temperature value (70C) for SA-516 Gr.70 N

$S = 138 \text{ Mpa}$

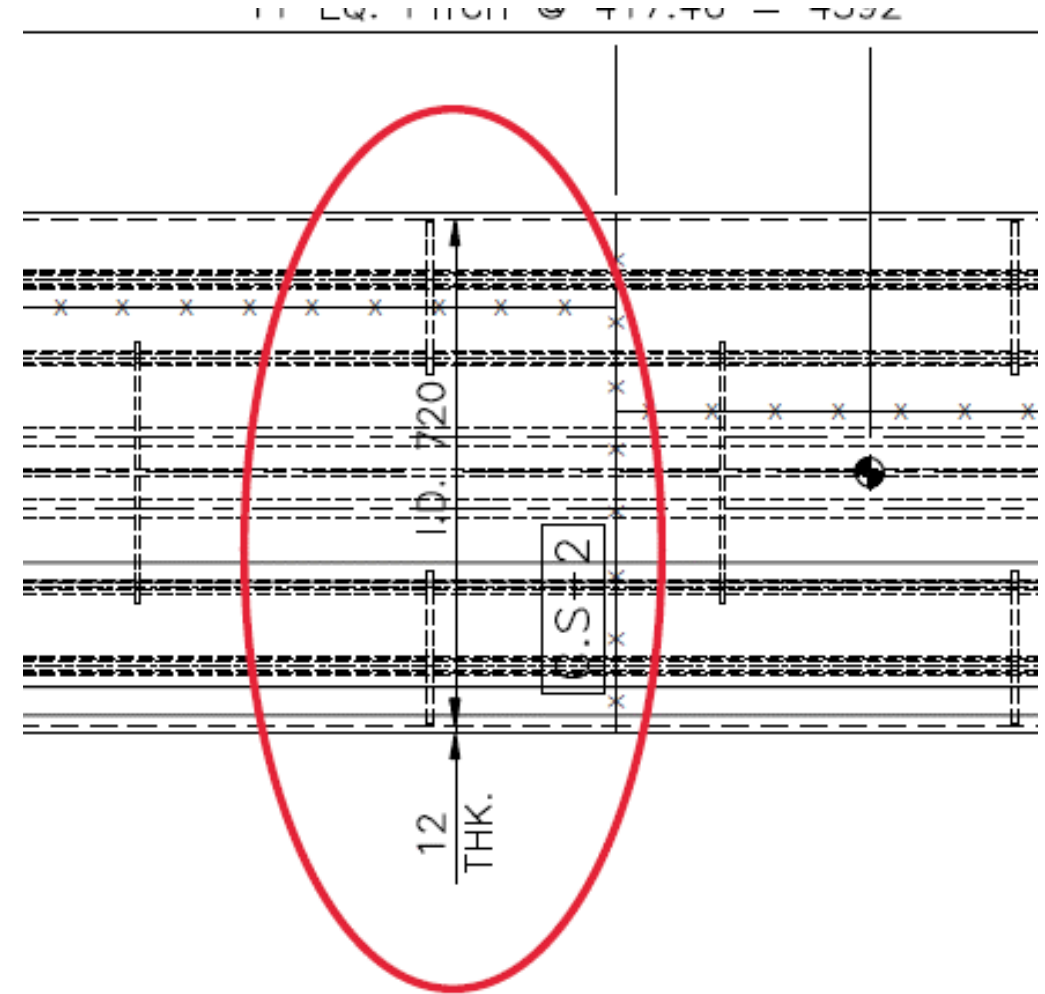
Sample Calculations

Shell Thickness Calculations - Under Internal Pressure

$$t = \frac{PR}{SE - 0.6 P} + CA + UT$$

$$t = \frac{2.5 * 363}{138 * 0.85 - 0.6 * 2.5} + 3 + 0 = 10.3 \text{ mm}$$

⇒ Used Thk 12 mm



Static Equipment Awareness Session

Thank you for Listening

Baher Elsheikh

Baher Elsheikh @

