CAU2014

Advantages of using Code Case 2695 and the Comparison between ASME Division 1 and Division 2

Technology to help you

CC 2695 uses Division 2 technology for Division 1 - Saves:

- Weight
- Material (mainly heads, not cylinders or cones)
- Reduce nozzle re-pads (in some cases)
- Reduce welding time and consumables

Why:

- Uses ASME Section VIII Division 2 technology
- But: with Division 1 lower allowable stresses

Presented by: Ray Delaforce

INTERGRAPH

PV Elite demonstration - Calculation of a Head

Here are the details:

- Elliptical Head
- P = Internal pressure 1,75 MPa
- D = Internal diameter 1 500 mm
- S = Allowable stress 138 MPa

Division 1 calculation (demo first)

Required Thickness due to Internal Pressure [tr]:
= (P*D*Kcor)/(2*S*E-0.2*P) Appendix 1-4(c)
= (1.750*1500.0000*1.000)/(2*138.00*1.00-0.2*1.750)
= 9.5231 + 0.0000 = 9.5231 mm

Use a 12 mm plate

Division 2 calculation (demo first)

Computed Minimum Required Thickness [t]:
tr = 8.5872 mm
t = tr + ci + co
= 8.5872 + 0.0000 + 0.0000
= 8.5872 mm - see below for the derivation

Use a 10 mm plate

This might be a small saving but:

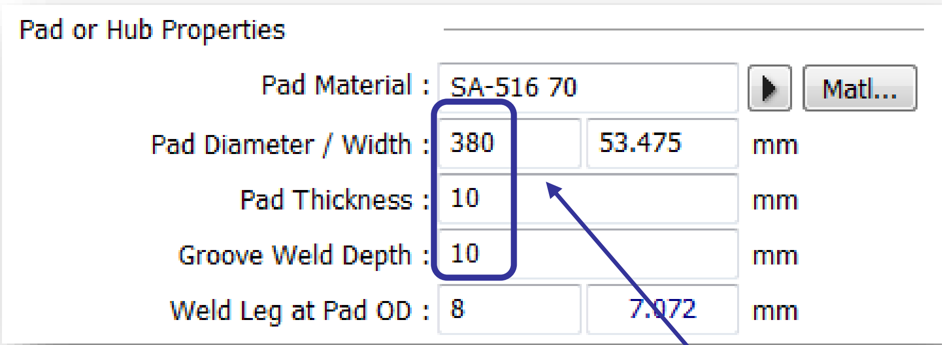
INTERGRAPH

PV Elite demonstration - Nozzle Reinforcement

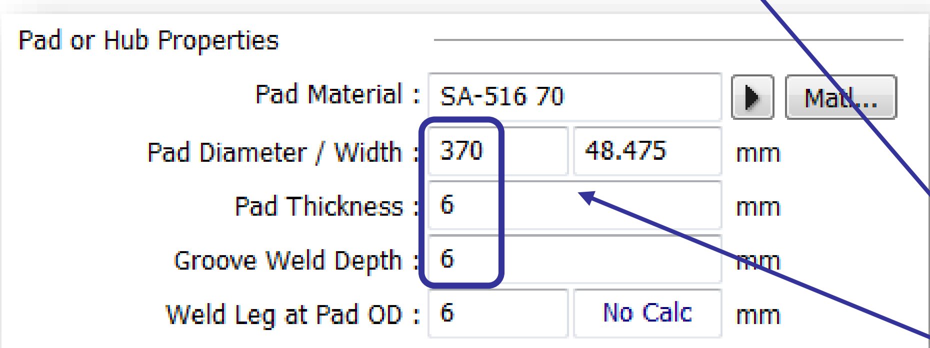
Here are the details:

- First a 10 mm thick pad x 380 mm Outside diameter

Division 1 calculation (demo first)



Division 2 calculation (demo first)

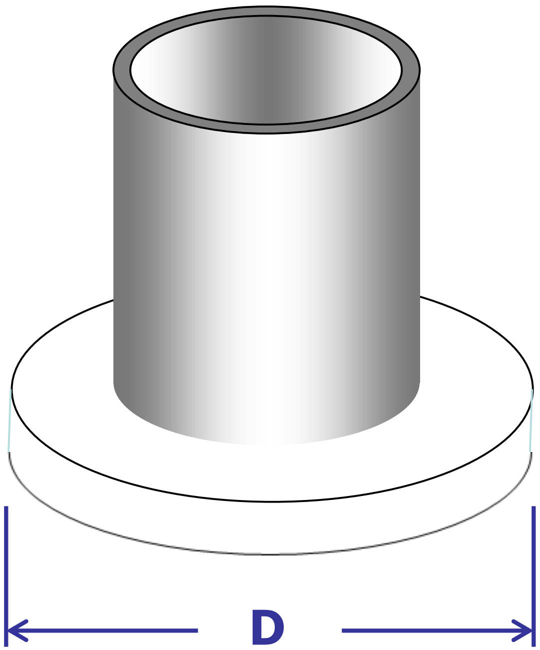


This is the saving:

INTERGRAPH

Consider a nozzle reinforcement pad using CC 2695

The re-pad size could be reduced perhaps



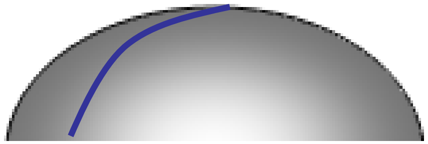
Length of welding:

Approximately 3 x D

If there are many re-pads – big saving

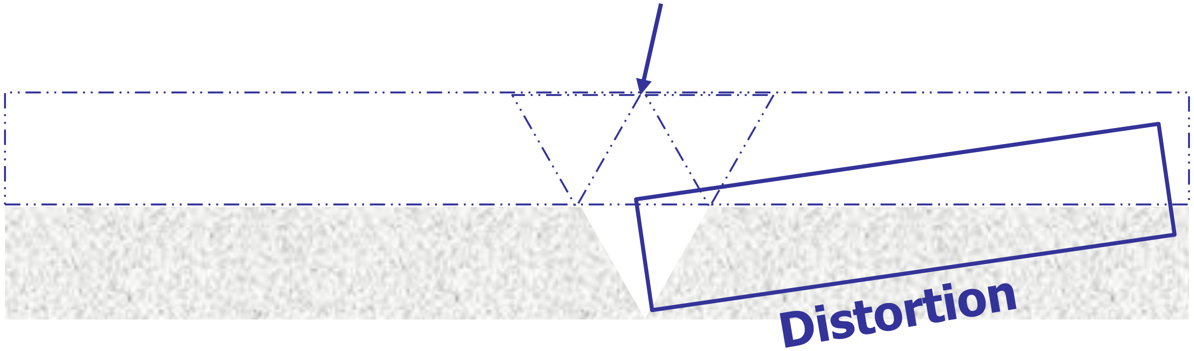
If a disc is welded to make a head

If the thickness is doubled, welding is **4** times as much welding



After forming

Metal shrinks as it cools



Save:

- ❑ Welding time
- ❑ Welding consumables
- ❑ Welding distortion

The greater the metal, the greater the distortion

INTERGRAPH

CC 2695: There are other advantages – especially over Division 2

Requirements for Division 2

- More Radiography
- More rigorous inspection
- Calculations signed of by a Professional Engineer
- An operator’s manual is required
- Restriction of some materials over Division 1
- A very comprehensive data package required
- A Fatigue analysis is required (more of this later)

Requirements for Division 1

- Less radiography
- Less inspection
- Simple U-1 form required only

Only drawback – lower stresses allowed

Comparison of Division 1 and Division 2 allowable stresses

This is the stated allowable stress for Division 1

$$S = \text{The less of: } \frac{UTS}{3,5} \text{ or } \frac{Yield}{1,5}$$

$$= \text{The less of: } \frac{483}{3,5} \text{ or } \frac{263}{1,5}$$

$$= \text{The less of: } 138\ MPa \text{ or } 174\ MPa \quad = 138\ MPa$$

This is the stated allowable stress for Division 2

$$S = \text{The less of: } \frac{UTS}{2,4} \text{ or } \frac{Yield}{1,5}$$

$$= \text{The less of: } \frac{483}{2,4} \text{ or } \frac{263}{1,5}$$

$$= \text{The less of: } 201\ MPa \text{ or } 174\ MPa \quad = 174\ MPa$$

For SA 516 70 material: UTS = 483 MPa and Yield = 263 MPa

Division 1 controlled by UTS, Division 2 controlled by Yield

INTERGRAPH

Comparison of Division 1 and Division 2 allowable stresses

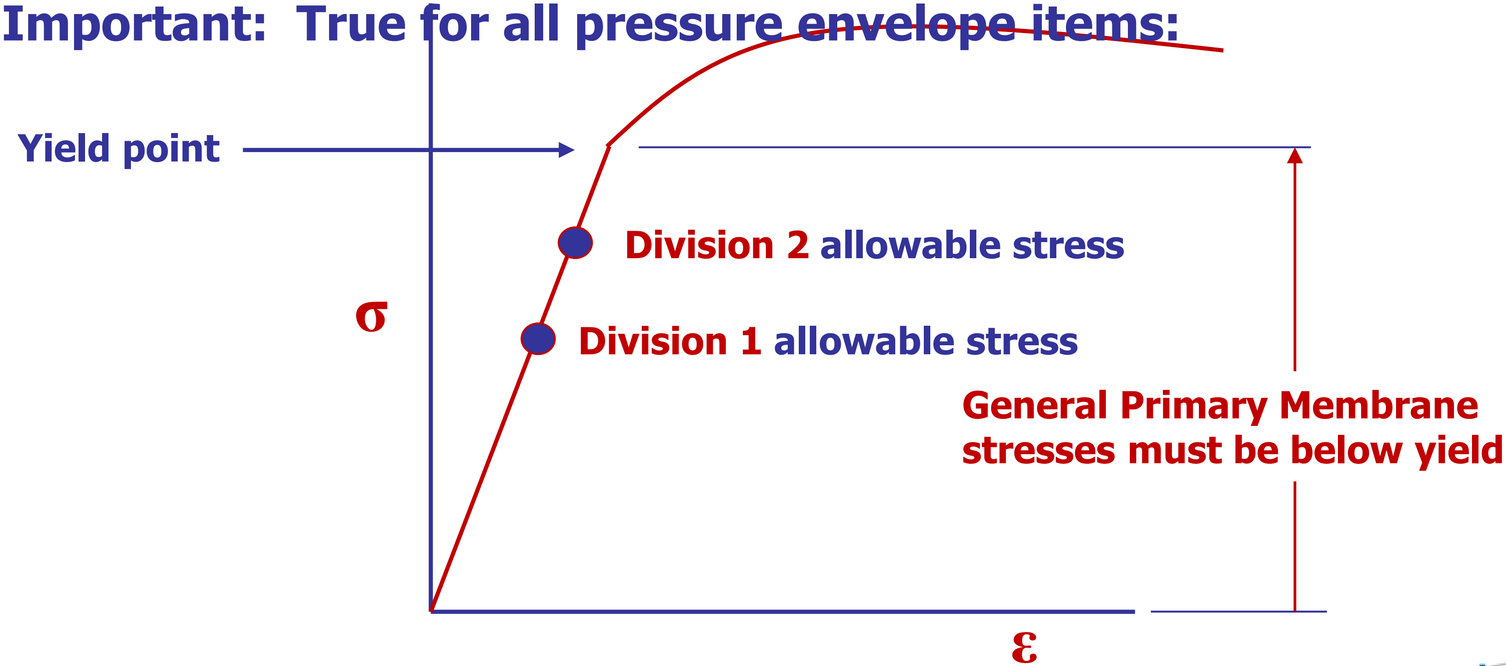
Consider the stress – strain diagram, familiar to engineers

Division 1 has a greater safety margin

This is the stress in the cylinder wall from internal pressure

This is very important for safety !

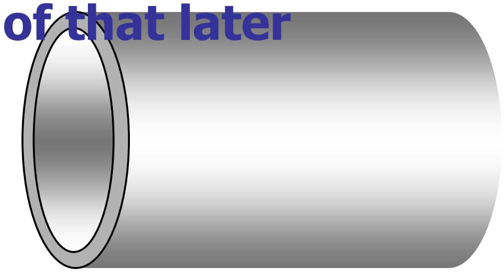
Important: True for all pressure envelope items:



Comparison of Division 1 and Division 2 allowable stresses

Consider the stress – strain diagram, familiar to engineers

Certain stresses can exist here, but more of that later



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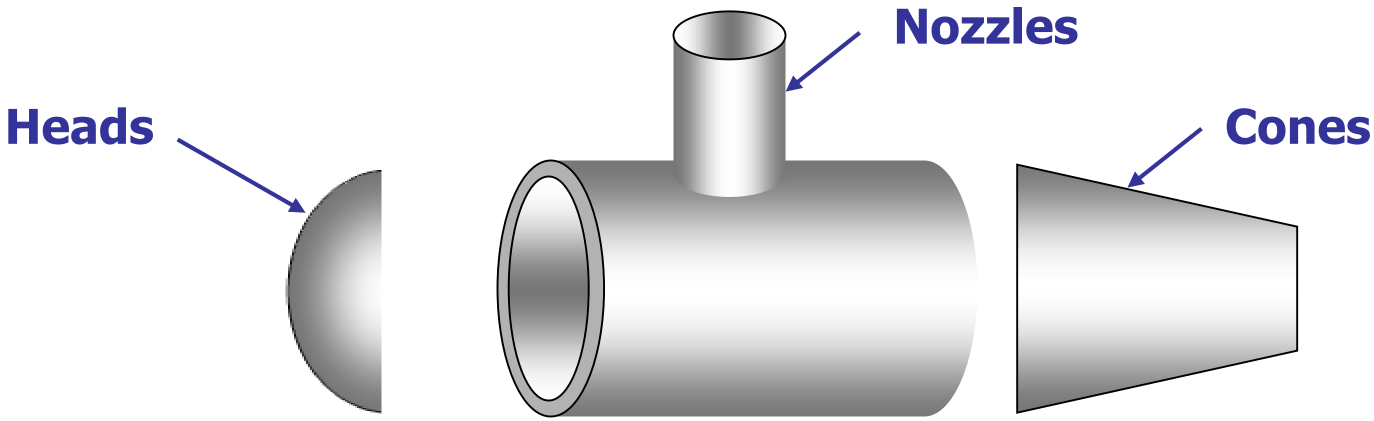
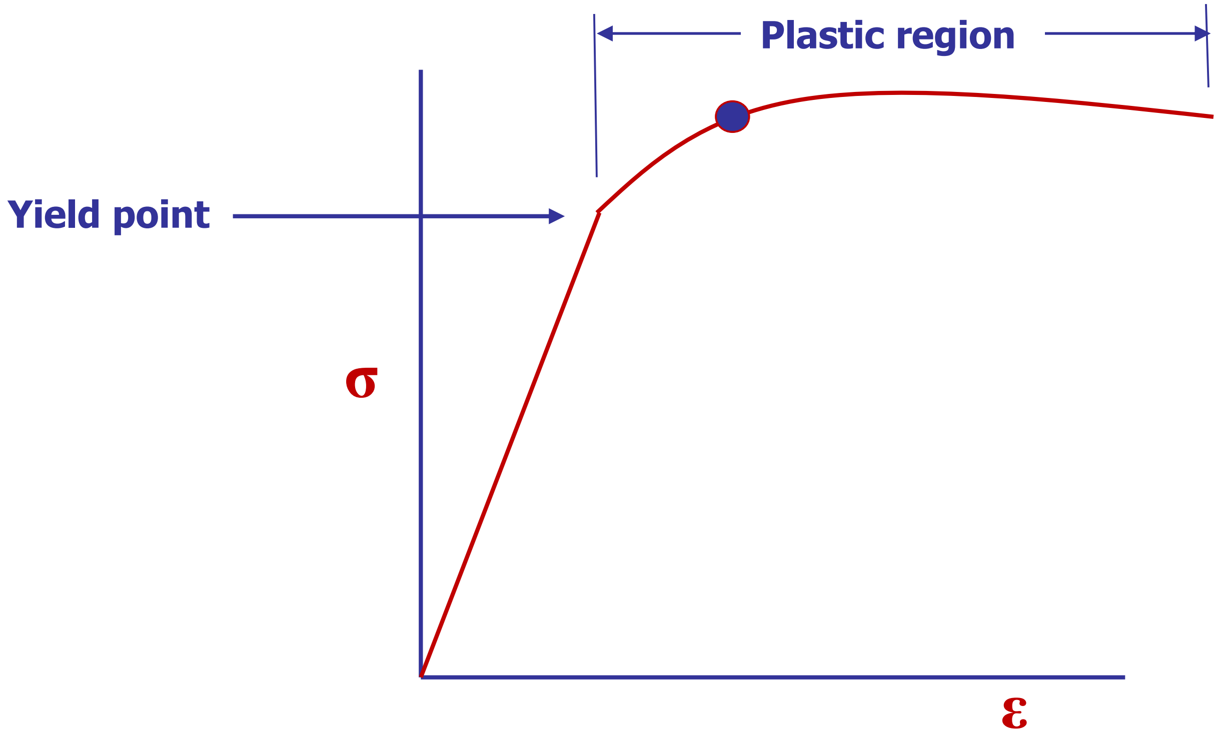


Comparison of Division 1 and Division 2 allowable stresses

Consider the stress – strain diagram, familiar to engineers

Certain stresses can exist here, but more of that later

These are known as Secondary Stresses – treated differently



Comparison of Division 1 and Division 2 allowable stresses

Division 1 and Division 2 use **different** theories of failure

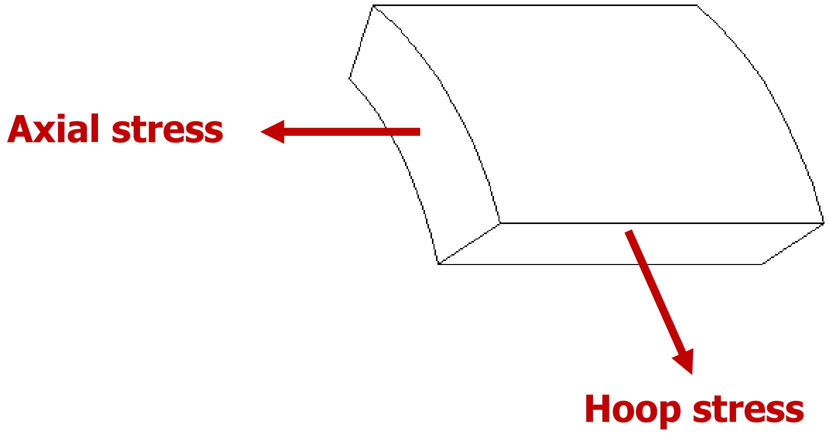
Consider a cylinder element subject to internal pressure

Generally the hoop stress is **twice** the axial stress

Also known as **Principe Stresses** – because there are **no shear** stresses

Division 1 **only considers the Hoop Stress**

So, Division 1 uses the **Maximum Principle Stress** one – **ignores axial**



Comparison of Division 1 and Division 2 allowable stresses

Division 1 and Division 2 use different theories of failure

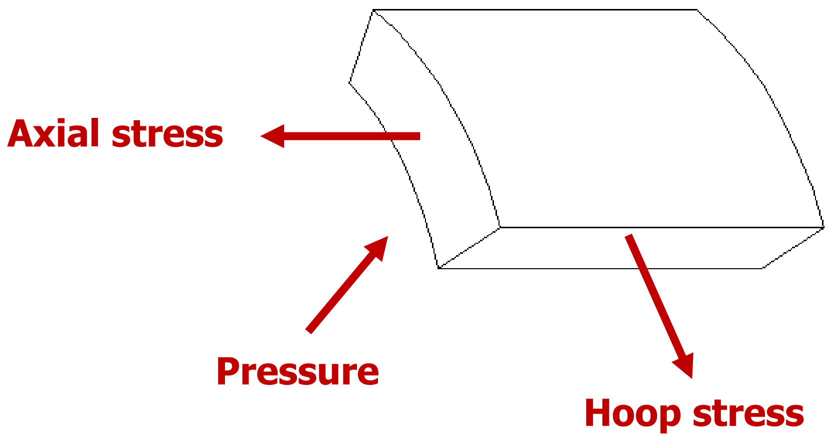
There is actually a third stress we have ignored - Pressure

Division 2 considers all three stresses in its analysis

Maximum principle stress assumes the component fails in tension

This is NOT the case

Even though Division 1 is based on the assumption



Comparison of Division 1 and Division 2 allowable stresses

Look at this test piece that has been tensile tested

The mechanism of failure is fracture at 45^{o}

This highlights an important principle

INTERGRAPH

Comparison of Division 1 and Division 2 allowable stresses

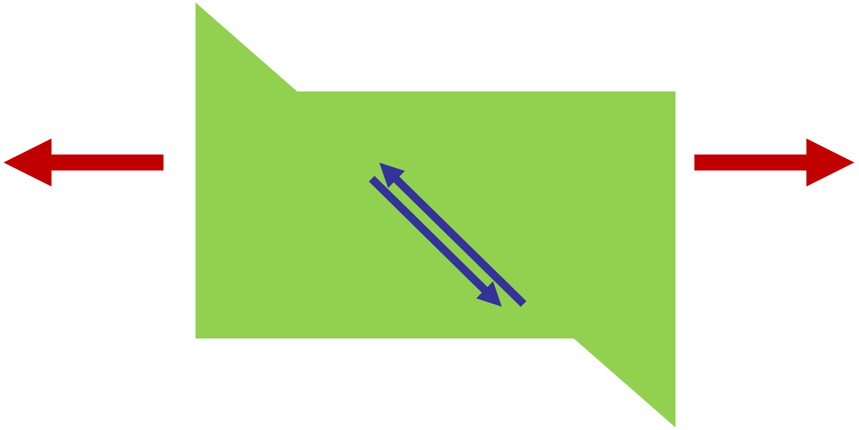
Consider a block of metal that fails at 45^{o} in a tensile test

This what happens , there are shear forces on the fracture planes

Normal forces exist also – but we shall ignore them for now

So, generally fracture takes place in shear, not tension

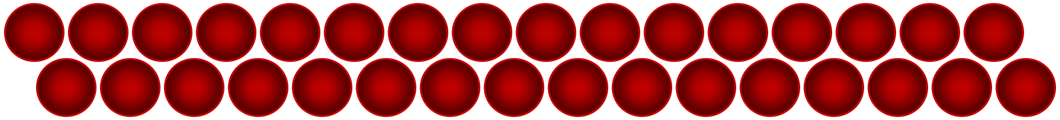
This leads is to an important concept



Comparison of Division 1 and Division 2 allowable stresses

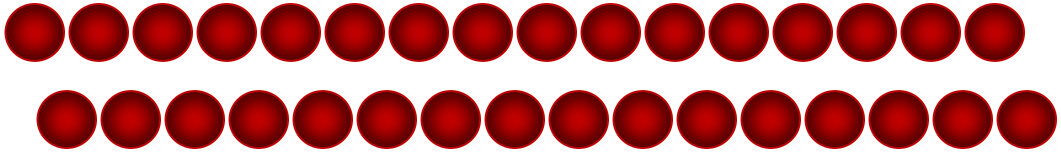
Our first theory of failure – based upon failure in shear

Atoms lie in sheets like this , and sliding takes place which is shear

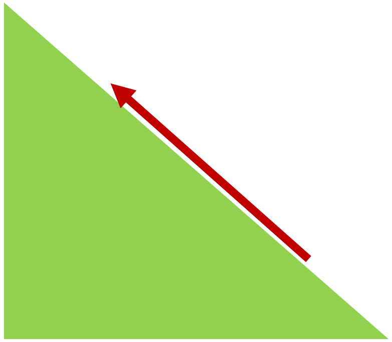


This gives rise to the shear stress on the fracture plane

Division 1 assumes the fracture occurs in tension – not quite correct



This is according to the Maximum Principle Stress theory



INTERGRAPH

Comparison of Division 1 and Division 2 allowable stresses

Logically, the failure would occur when shear stress is maximu

Suppose we have two orthogonal stresses, plotted on Mohr diagram

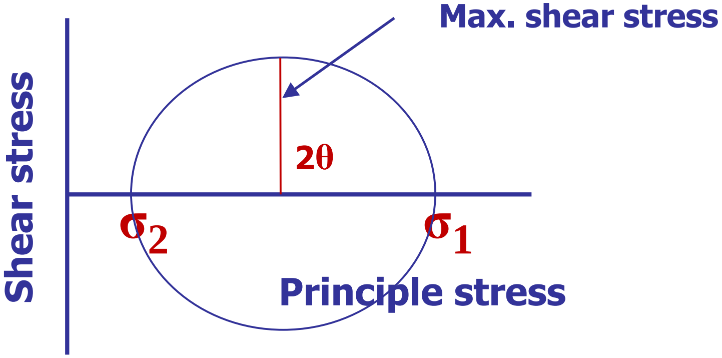
Draw the Mohr Circle

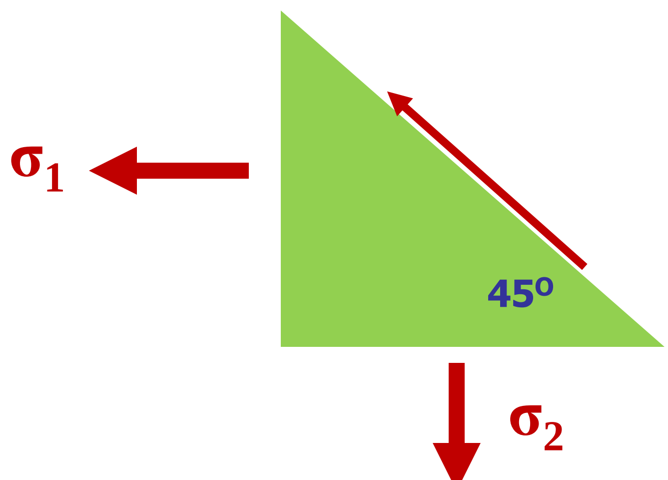
Maximum shear stress = $(\sigma_1 - \sigma_2)/2$

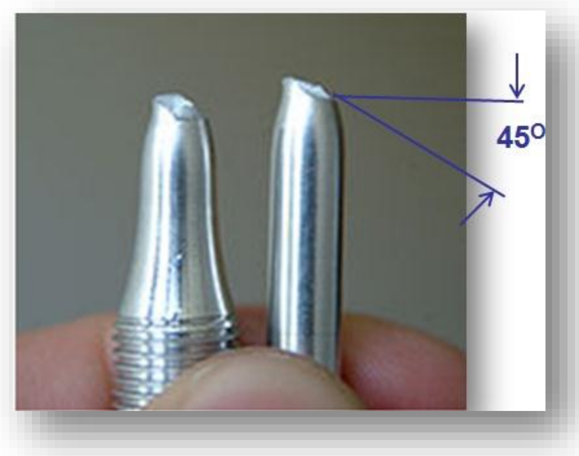
The Mohr angle is $2\theta = 90^{o}$

Thus $\theta = 45^{o}$

This gives rise to a theory of failure







Comparison of Division 1 and Division 2 allowable stresses

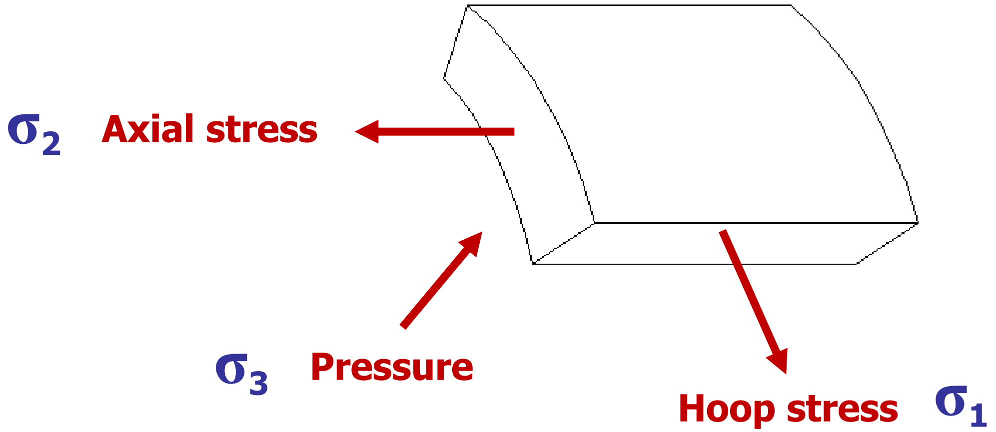
This known as the **TRESCA** or maximum shear stress theory

Collapse occurs when:

$$\sigma_Y = \max(|\sigma_1 - \sigma_2|, |\sigma_2 - \sigma_3|, |\sigma_3 - \sigma_1|)$$

And here are the stresses:

This was the situation up until the **2004** edition of Division 2

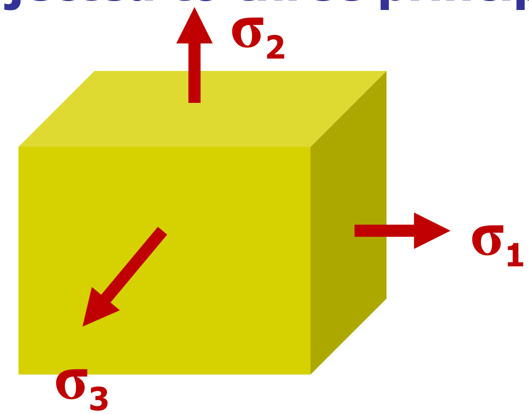


Comparison of Division 1 and Division 2 allowable stresses

The 2007 version of Division 2 changed in technology

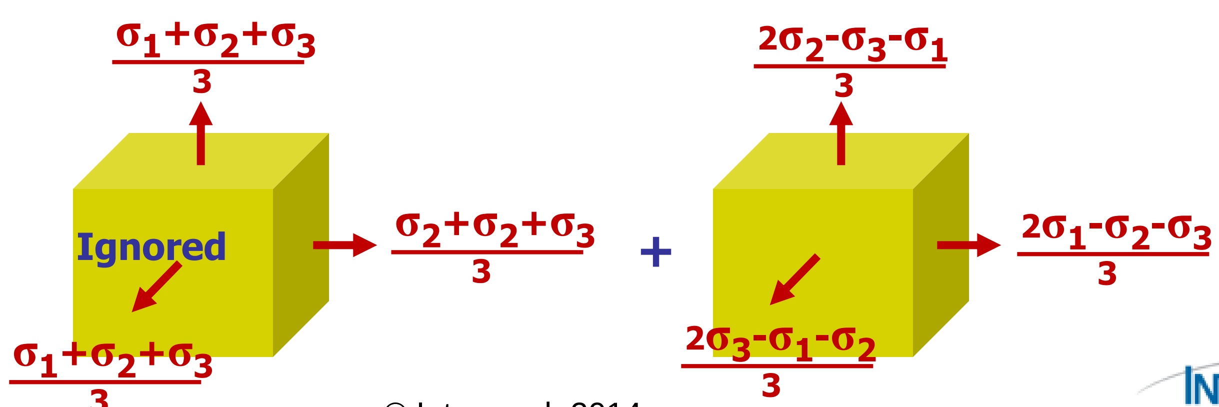
It used the Maximum Shear Strain Energy of von Mises Theory

Consider a block subjected to three principle stresses



This is divided into two components like this:

Volume change | **Shear strain energy – von Mises**



INTERGRAPH

Comparison of Division 1 and Division 2 allowable stresses

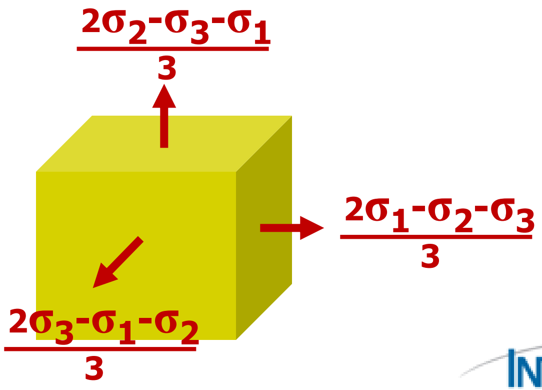
This is the von Mises Equation

$$\sigma_Y = \frac{1}{\sqrt{2}}\left[(\sigma_1 - \sigma_2)^2, (\sigma_2 - \sigma_3)^2, (\sigma_3 - \sigma_1)^2\right]^{0,5}$$

Shear strain energy – von Mises

It yields a result close to the Tresca equation

We now have three theories of failure

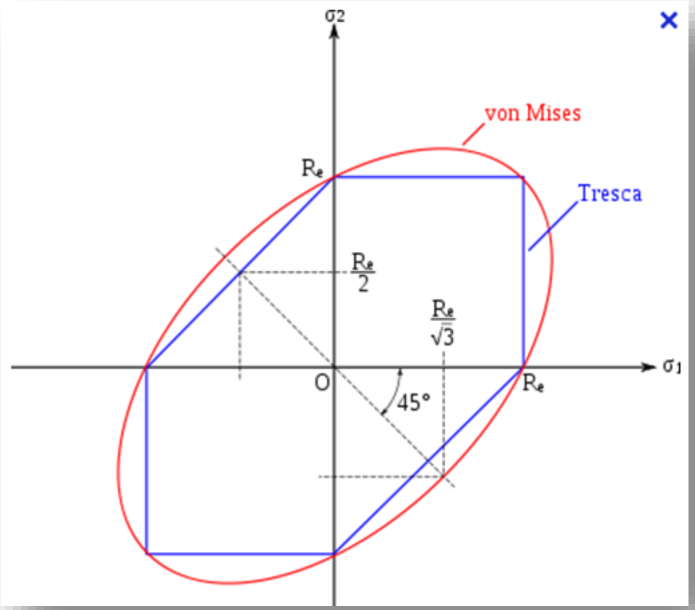


Comparison of Division 1 and Division 2 allowable stresses

The three theories of failure compared

- Rankin – Maximum Principle stress
  - The basis for ASME VIII, Division 1
- Tresca - Maximum Shear stress
  - The basis for ASME VIII, Division 2 up to 2004
- von Mises – Maximum Shear strain energy
  - The basis for ASME VIII, Division 2 from 2007

Here is a comparison between Tresca and von Mises

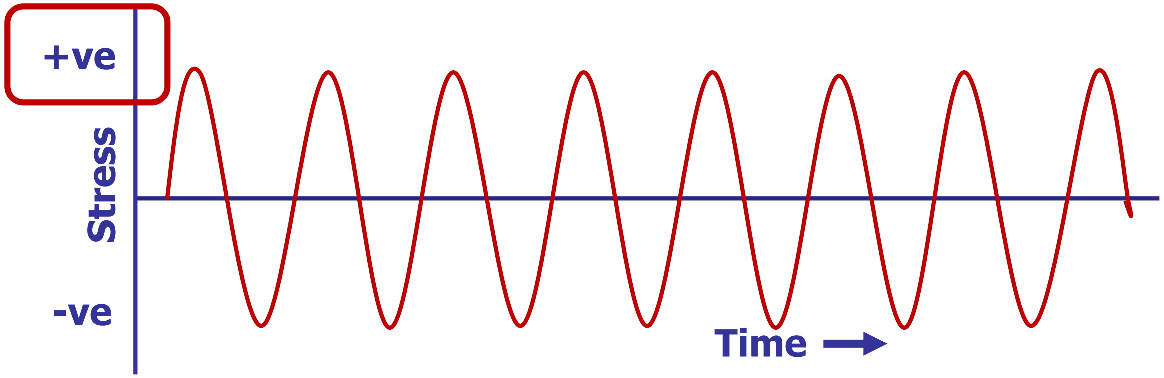


INTERGRAPH

Unlike Division 1, Division 2 requires a fatigue evaluation

First we need a definition of fatigue (not getting tired!)

Fatigue is incremental crack growth under cyclic loading



When the component is in tension, the crack grows each time

Unlike Division 1, Division 2 requires a fatigue evaluation

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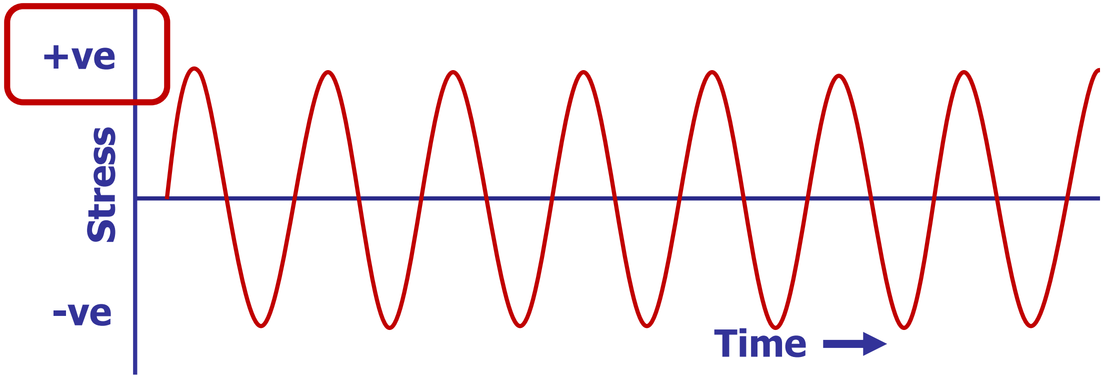

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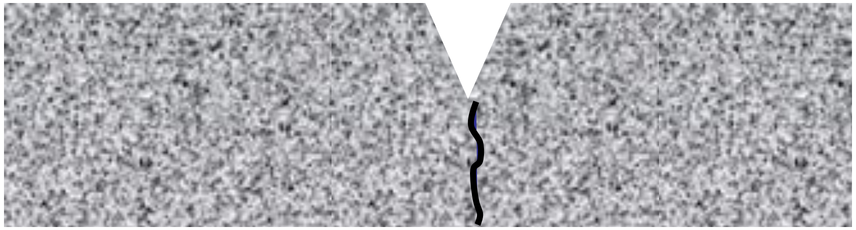
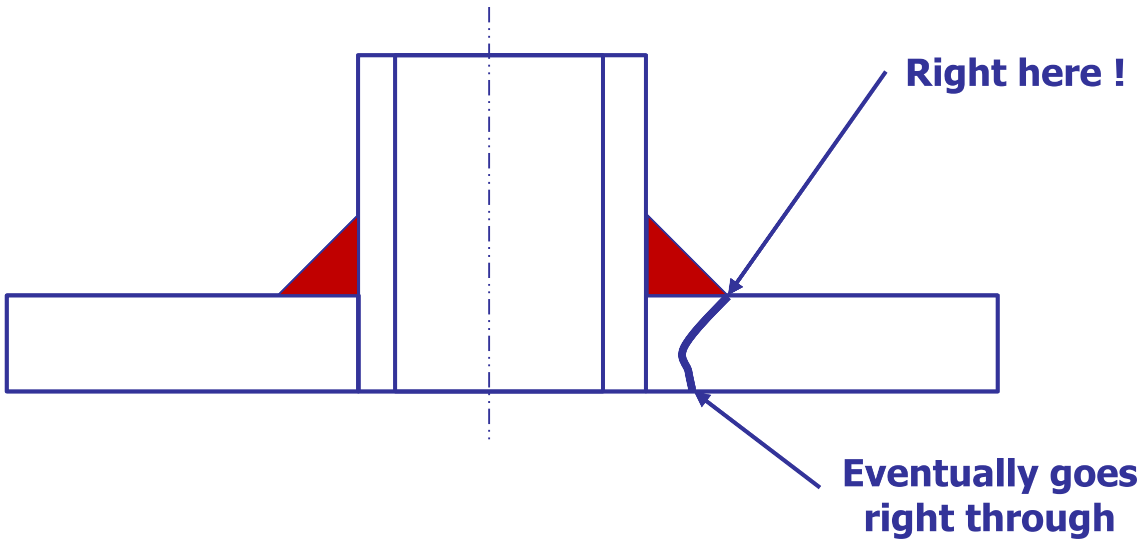


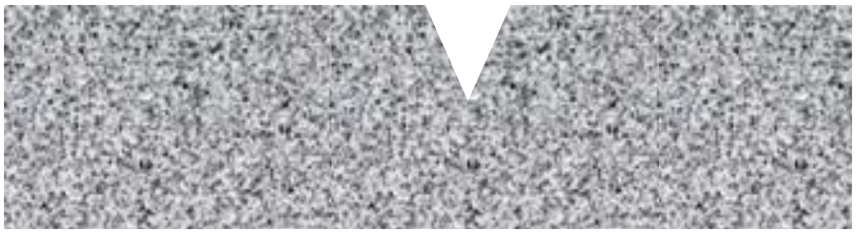
Eventually the crack deepens until it is a through crack

Unlike Division 1, Division 2 requires a fatigue evaluation

What is the source of this starter crack ? **Consider a Nozzle**

The good news – it is not a catastrophic failure



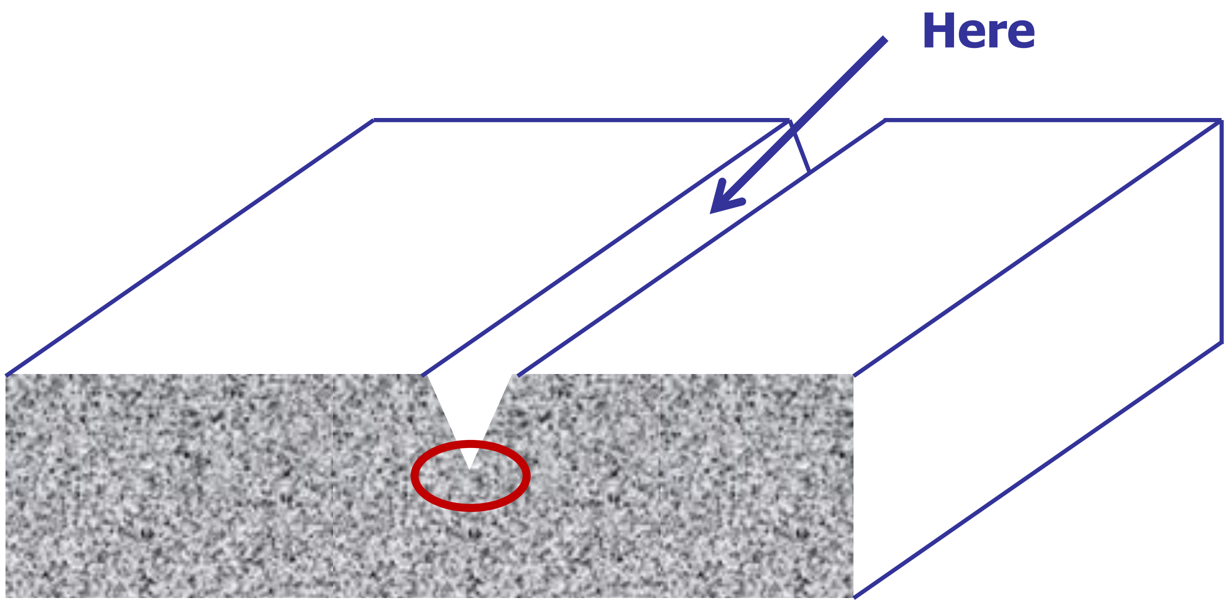


Unlike Division 1, Division 2 requires a fatigue evaluation

The internal strain energy from tension promotes 2 free surfaces

In this region, the stresses are very high

Consider those stresses on the Stress-Strain diagram



Unlike Division 1, Division 2 requires a fatigue evaluation

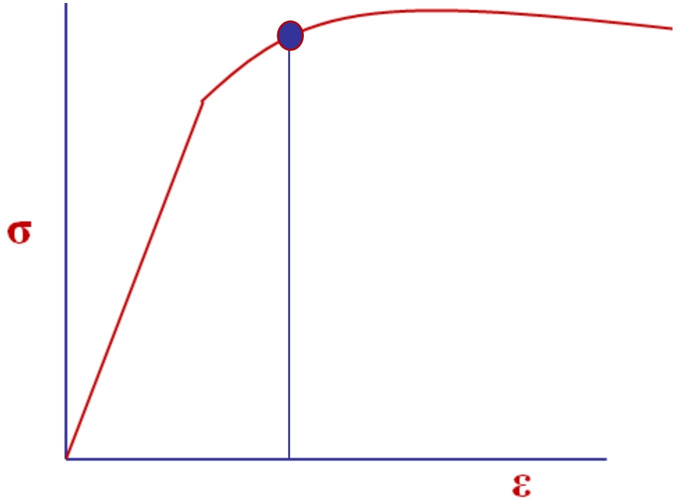
The internal strain energy from tension promotes 2 free surfaces

In this region, the stresses are very high

Consider those stresses on the Stress-Strain diagram

The fatigue stress is in the PLASTIC region, with a large strain

Note the strain at the bottom of the diagram



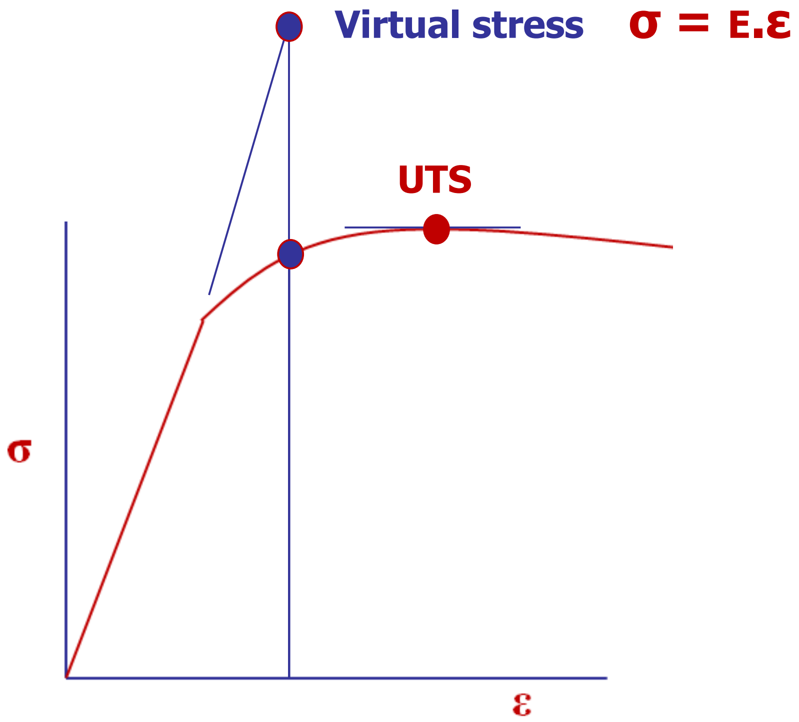
Unlike Division 1, Division 2 requires a fatigue evaluation

We can project lines to find a Virtual Stress based on the strain

This is a computed stress based on the Elastic Modulus – not real

Note: The virtual stress is higher than the UTS of the metal

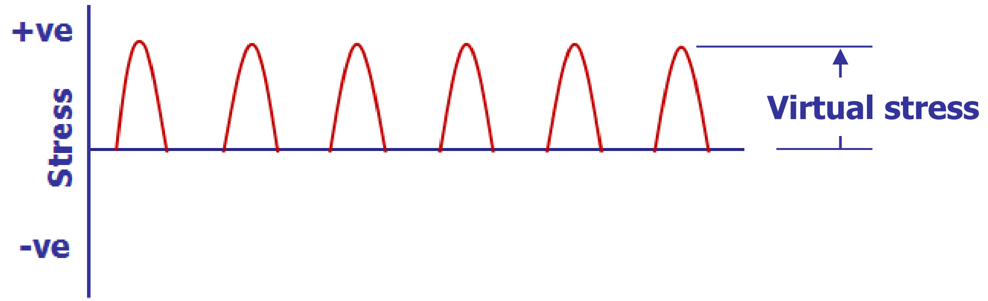
Actual stresses CANNOT really exist above the curve

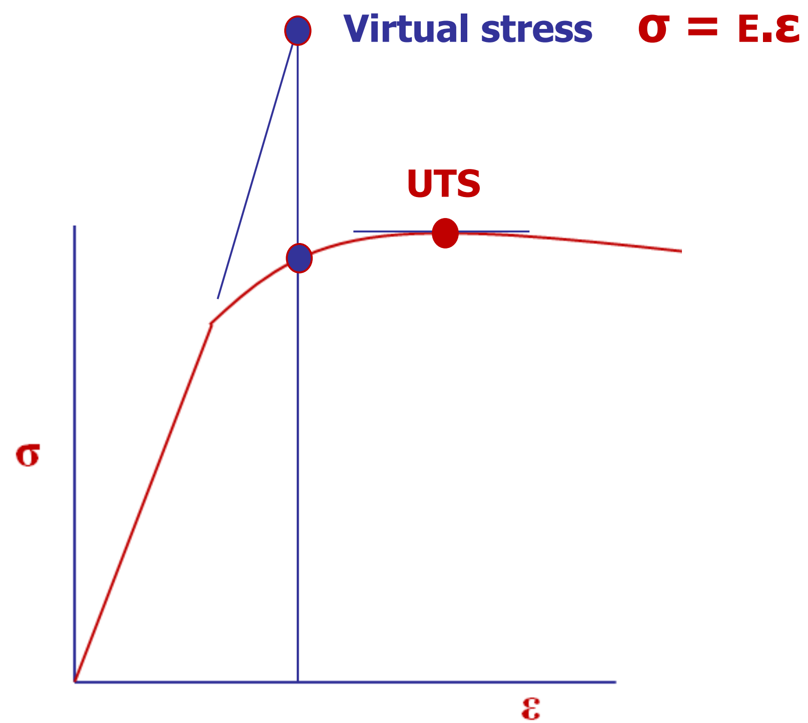


INTERGRAPH

Unlike Division 1, Division 2 requires a fatigue evaluation

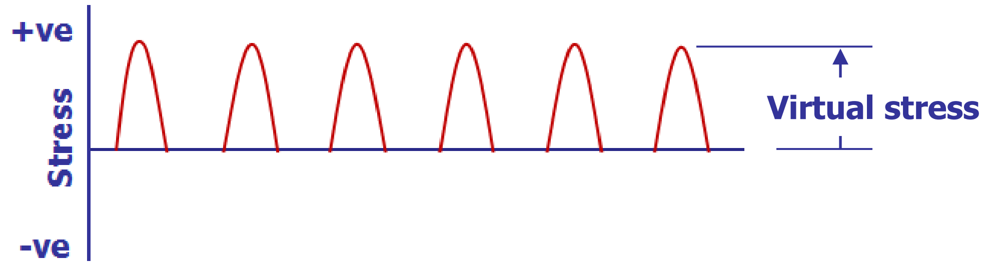
The number of cycles to failure depends in the tensile magnitude





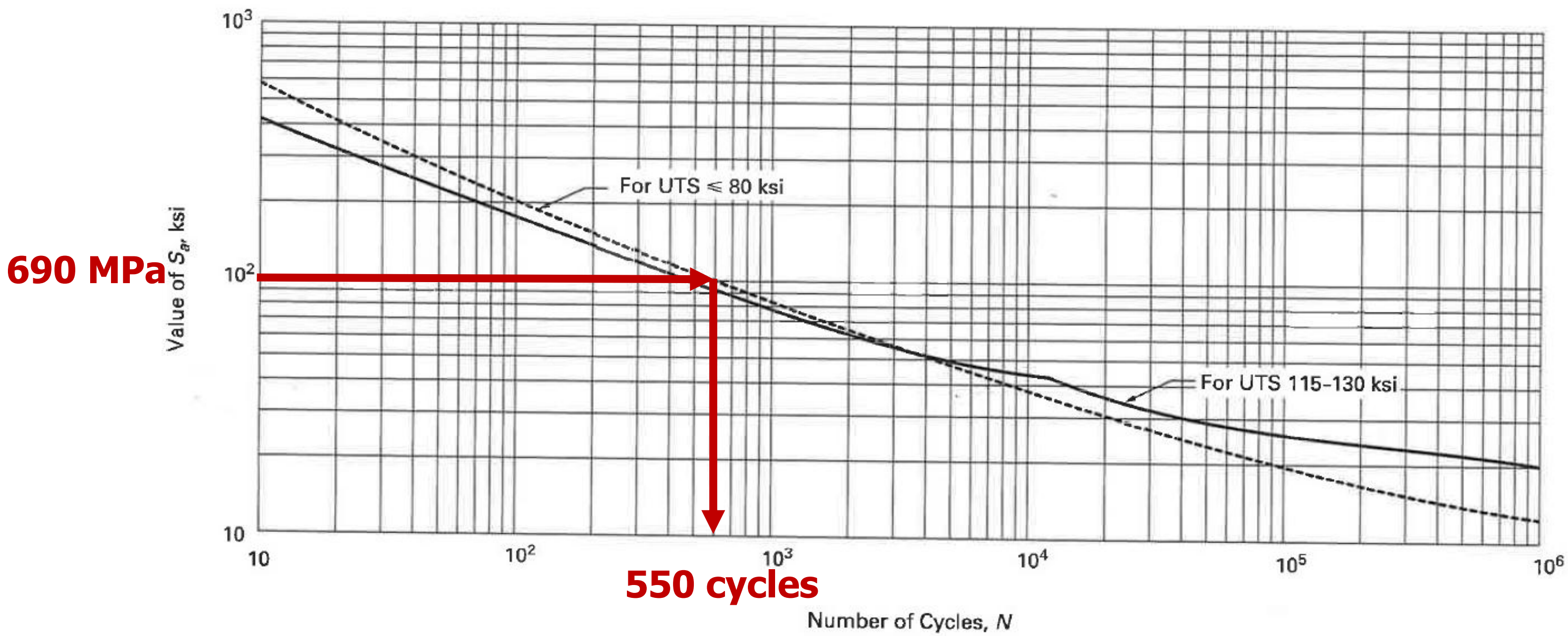
Unlike Division 1, Division 2 requires a fatigue evaluation

Consider a typical fatigue curve from the 2004 Division 2



Here is a virtual stress of 690 MPa (well above UTS)

We can read off the number of cycles allowed by the code



Unlike Division 1, Division 2 requires a fatigue evaluation

PV Elite example (demo first)

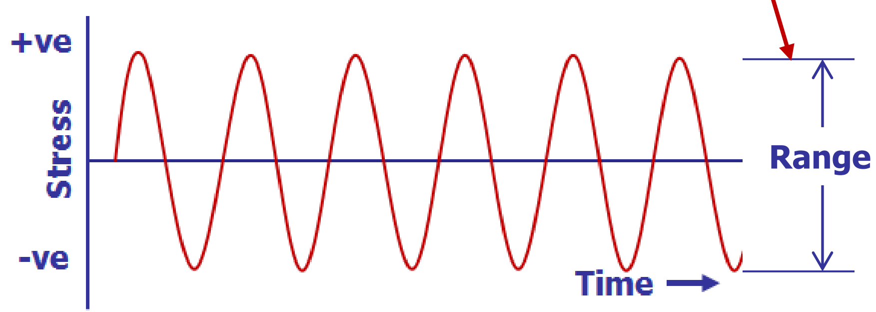
Input Values: Pressure in: MPa

Case	Pressure 1	Pressure 2	Range	Number of Cycles
1	0.000	1.750	1.750	30000.
2	0.500	1.250	0.750	15000.
3	0.750	1.300	0.550	200000.

Pressure indices per Table 5.D.1 for Internal Pressure Loading:

Stress	Inside Corner	Outside Corner
sn	2.0000	2.0000
st	-.2000	2.0000
sr	-.0267	0.0000
s	2.2000	2.0000

We get the Stress Concentration Factor (scf) from Division 2



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PV Elite example (demo first)

Input Values: Pressure in: MPa

```
Case              Pressure 1     Pressure 2          Range        Number of Cycles
------------------------------------------------------------------------------------
 1                  0.000          1.750            1.750              30000.
 2                  0.500          1.250            0.750              15000.
 3                  0.750          1.300            0.550             200000.
```

Pressure indices per Table 5.D.1 for Internal Pressure Loading:

```
         Inside     Outside
Stress   Corner     Corner
  sn     2.0000     2.0000
  st     -.2000     2.0000
  sr     -.0267     0.0000
  s      2.2000     2.0000
```

We get the Stress Concentration Factor (scf) from Division 2

Table 5.D.1 – Stress Indices For Nozzles In Spherical Shells And Portions Of Formed Heads

Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	–0.2	2.0
σ_r	$-\frac{2t}{R}$	0.0
σ	2.2	2.0

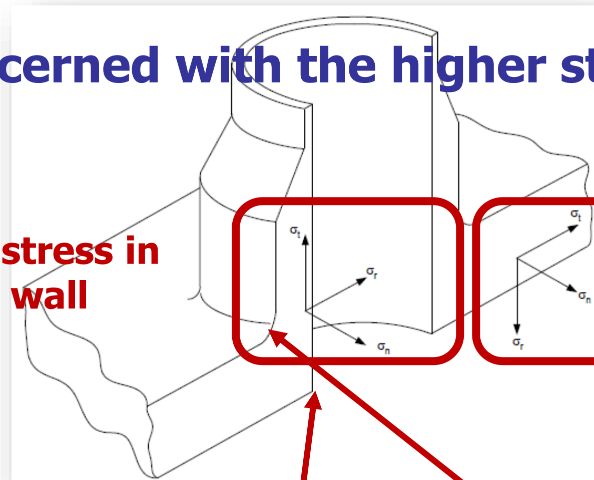
Unlike Division 1, Division 2 requires a fatigue evaluation

PV Elite example (demo first)

Here are the locations for the nozzle per Division 2

We are mainly concerned with the higher stresses in the shell

These are the stress in the nozzle wall



These are the stress in the shell (head)

We get the Stress Concentration Factor (scf) from Division 2

Table 5.D.1 – Stress Indices For Nozzles In Spherical Shells And Portions Of Formed Heads

Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	−0.2	2.0
σ_r	$-\frac{2t}{R}$	0.0
σ	2.2	2.0

INTERGRAPH

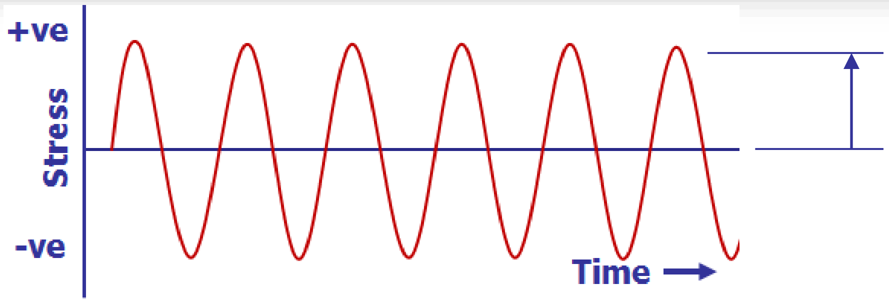
Unlike Division 1, Division 2 requires a fatigue evaluation

PV Elite example (demo first)

Continuing with the output from PV Elite

This is the stress in the head x the scf for the amplitude (not range)

Stress Intensity after applying the Pressure Index (amplitude) [Sa]:
= 130.4187 MPa



Amplitude is half the range

Case 1 Membrane Stress: Adjusted below per above Index table:

	Stress	Inside Corner	Outside Corner
sn	59.281	118.562	118.562
st	59.281	-11.856	118.562
sr	59.281	-1.581	0.000
sint	59.281	130.419	118.562

INTERGRAPH

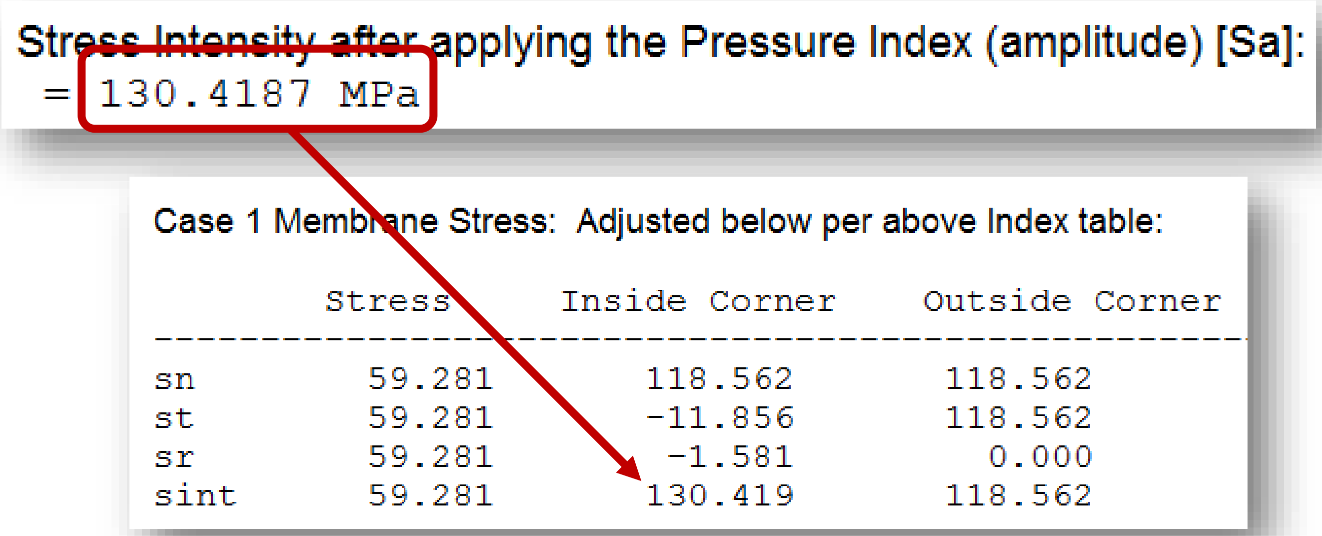
Unlike Division 1, Division 2 requires a fatigue evaluation

PV Elite example (demo first)

Continuing with the output from PV Elite

This is the S-N graph in the 2013 Division 2 – N is calculated

Here is the stress in the head for the amplitude (not range)



Stress		Inside Corner	Outside Corner
sn	59.281	118.562	118.562
st	59.281	-11.856	118.562
sr	59.281	-1.581	0.000
sint	59.281	130.419	118.562

The greatest stress

Unlike Division 1, Division 2 requires a fatigue evaluation

PV Elite example (demo first)

Continuing with the output from PV Elite

There is no S-N graph in the 2013 Division 2 – N is calculated

C Factors used in the above equation:

```
C1 =  2.25451       C2 = -.464224       C3 = -.831275      C4 = 0.863466E-01
C5 = 0.202083       C6 = -.694053E-02   C7 = -.207973E-01  C8 = 0.201024E-03
C9 = 0.713772E-03  C10 =  0.00000      C11 =  0.00000
```

From the table, EFc = 195128 MPa

Compute the Number of Cycles from Equation 3.F.1 [N]:

```
 = 10^(X)
 = 10^(5.102 )
 = 126603 Cycles
```

Case	StressIntens	N cycles	Nmax cycles	Damage Factor
1	130.419	30000.	0.1266E+06	0.237
2	55.894	15000.	0.1095E+11	0.000
3	40.989	200000.	0.1000E+12	0.000
Total:	Damage Factor:			0.237

Fatigue Analysis Passed: Damage Factor < 1.00

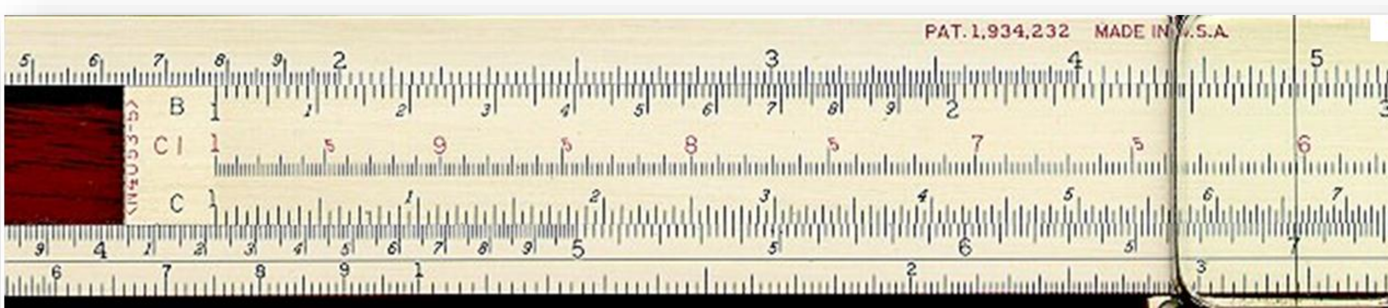
INTERGRAPH

Does Division 1 have any advantages ?

Division 1 was first published in 1925

- ❑ There were no computers
- ❑ There was no convenient software
- ❑ There were no calculators
- ❑ Engineers had:
  - A sliderule
  - Logarithm tables (for accurate work)
  - Erasures
  - Patience

Here is a typical sliderule, which very few can use today !



It doesn’t tell you where to place the decimal point (or comma)

Does Division 1 have any advantages ?

Division 1 was first published in 1925

`

Just look at this simple formula for an elliptical head

$$t = \frac{PD}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{D + 0.2t} \qquad (1)$$

That calculation can be done in a couple of minutes

INTERGRAPH

Does **Division 1** have any advantages ?

Now look at a **Division 2** equation for the head, compare the two !

$$t = \frac{PD}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{D + 0.2t} \qquad (1)$$

b) STEP 2 – Compute the head L/D, r/D, and L/t ratios and determine if the following equations are satisfied. If the equations are satisfied, then proceed to Step 3; otherwise, the head shall be designed in accordance with Part 5.

$$0.7 \le \frac{L}{D} \le 1.0 \qquad (4.3.5)$$

$$\frac{r}{D} \ge 0.06 \qquad (4.3.6)$$

$$20 \le \frac{L}{t} \le 2000 \qquad (4.3.7)$$

c) STEP 3 – Calculate the following geometric

$$\beta_{th} = \arccos\left[\frac{0.5D - r}{L - r}\right], \ radians$$

$$\phi_{th} = \frac{\sqrt{Lt}}{r}, \ radians \qquad (4.3.9)$$

$$R_{th} = \frac{0.5D - r}{\cos[\beta_{th} - \phi_{th}]} + r \qquad for \quad \phi_{th} < \beta_{th} \qquad (4.3.10)$$

$$R_{th} = 0.5D \qquad for \quad \phi_{th} \ge \beta_{th} \qquad (4.3.11)$$

d) STEP 4 – Compute the coefficients C_1 and C_2 using the following equations.

$$C_1 = 9.31\left(\frac{r}{D}\right) - 0.086 \qquad for \ \frac{r}{D} \le 0.08 \qquad (4.3.12)$$

$$C_1 = 0.692\left(\frac{r}{D}\right) + 0.605 \qquad for \ \frac{r}{D} > 0.08 \qquad (4.3.13)$$

$$C_2 = 1.25 \qquad for \ \frac{r}{D} \le 0.08 \qquad (4.3.14)$$

$$C_2 = 1.46 - 2.6\left(\frac{r}{D}\right) \qquad for \ \frac{r}{D} > 0.08 \qquad (4.3.15)$$

f) STEP 6 – Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength.

$$P_y = \frac{C_3 t}{C_2 R_{th}\left(\frac{R_{th}}{2r} - 1\right)} \qquad (4.3.17)$$

If the allowable stress at the design temperature is governed by time-independent properties, then C_3 is the material yield strength at the design temperature, or $C_3 = S_y$. If the allowable stress at the design temperature is governed by time-dependent properties, then C_3 is determined as follows.

1) If the allowable stress is established based on 90% yield criterion, then C_3 is the material allowable … y 1.1, or $C_3 = 1.1S$.

… n 67% yield criterion, then C_3 is the material allowable … y 1.5, or $C_3 = 1.5S$.

g) STEP 7 – Calculate the value of internal pressure expected to result in a buckling failure of the knuckle.

$$P_{ck} = 0.6 P_{eth} \qquad for \quad G \le 1.0 \qquad (4.3.18)$$

$$P_{ck} = \left(\frac{0.77508G - 0.20354G^2 + 0.019274G^3}{1 + 0.19014G - 0.089534G^2 + 0.0093965G^3}\right) P_y \qquad for \quad G > 1.0 \qquad (4.3.19)$$

where

$$G = \frac{P_{eth}}{P_y} \qquad (4.3.20)$$

h) STEP 8 – Calculate the allowable pressure based on a buckling failure of the knuckle.

$$P_{ak} = \frac{P_{ck}}{1.5} \qquad (4.3.21)$$

i) STEP 9 – Calculate the allowable pressure based on rupture of the crown.

$$P_{ac} = \frac{2SE}{\frac{L}{t} + 0.5} \qquad (4.3.22)$$

j) STEP 10 – Calculate the maximum allowable internal pressure.

$$P_a = \min[P_{ak}, P_{ac}] \qquad (4.3.23)$$

Does Division 1 have any advantages ?

Now look at a Division 2 equation for the head, compare the two !

`

The Division 1 calculation is very simple

The Division 2 has a number of complexities

- It is difficult to do the calculation by hand
- The required thickness cannot be computed by hand
  - You have to start with the thickness to derive the pressure
  - Can only be done by a computer
  - It is lengthy to be checked by hand

However, using CC 2695 have the advantage of the latest technology

Does Division 1 have any advantages ?

Consider look at a Division 2 procedure for cone junction analysis

It is exceedingly complex – a computer has to be used – no other way

It is impossible to do by hand

Table 4.3.1 – Large End Junction

Cylinder	Cone
Stress Resultant Calculation	**Stress Resultant Calculation**
$M_{sP} = Pt_L^2 M_{sN}$, see Table 4.3.3	$M_{csP} = M_{sP}$
$M_{sX} = X_L t_L M_{sN}$, see Table 4.3.4	$M_{csX} = M_{sX}$
$M_s = M_{sP} + M_{sX}$	$M_{cs} = M_{csP} + M_{csX}$
$Q_P = Pt_L Q_N$, see Table 4.3.3	$Q_c = Q\cos[\alpha] + N_s \sin[\alpha]$ **(1)**
$Q_X = X_L Q_N$, see Table 4.3.4	$R_C = \dfrac{R_L}{\cos[\alpha]}$
$Q = Q_P + Q_X$	
$\beta_{cy} = \left[\dfrac{3(1-\nu^2)}{R_L^2 t_L^2}\right]^{0.25}$	$\beta_{co} = \left[\dfrac{3(1-\nu^2)}{R_C^2 t_C^2}\right]^{0.25}$
$N_s = \dfrac{PR_L}{2} + X_L$	$N_{cs} = N_s \cos[\alpha] - Q\sin[\alpha]$ **(2)**
$N_\theta = PR_L + 2\beta_{cy} R_L \left(-M_s \beta_{cy} + Q\right)$	$N_{c\theta} = \dfrac{PR_L}{\cos[\alpha]} + 2\beta_{co} R_C \left(-M_{cs}\beta_{co} - Q_c\right)$
$K_{pc} = 1.0$	$K_{cpc} = 1.0$
Stress Calculation	**Stress Calculation**
$\sigma_{sm} = \dfrac{N_s}{t_L}$	$\sigma_{sm} = \dfrac{N_{cs}}{t_C}$
$\sigma_{sb} = \dfrac{6M_s}{t_L^2 K_{pc}}$	$\sigma_{sb} = \dfrac{6M_{cs}}{t_C^2 K_{cpc}}$
$\sigma_{\theta m} = \dfrac{N_\theta}{t_L}$	$\sigma_{\theta m} = \dfrac{N_{c\theta}}{t_C}$
$\sigma_{\theta b} = \dfrac{6\nu M_s}{t_L^2 K_{pc}}$	$\sigma_{\theta b} = \dfrac{6\nu M_{cs}}{t_C^2 K_{cpc}}$
Acceptance Criteria	**Acceptance Criteria**
$\sigma_{sm} \le 1.5S$	$\sigma_{sm} \le 1.5S$
$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$	$\sigma_{sm} \pm \sigma_{sb} \le S_{PS}$
$\sigma_{\theta m} \le 1.5S$	$\sigma_{\theta m} \le 1.5S$
$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$	$\sigma_{\theta m} \pm \sigma_{\theta b} \le S_{PS}$

Notes:

1. The Q and N_s values used to determine the resultant shear force in the cone, Q_c, are the same as those defined for the cylinder.
2. The Q and N_s values used to determine the resultant meridional membrane force in the

In Division 1, the calculation can be done by hand

This is a just a small sample of the analysis

INTERGRAPH

Consider the philosophy behind Division 2

To a large degree followed the PED:

- **European Pressure Equipment Directive**

The derivation of the allowable stresses, for Carbon Steel:

The PED: $$f = \min\left(\frac{UTS}{2{,}4}\ ;\ \frac{Yield}{1{,}5}\right)$$

Division 1:

TABLE 10-100
CRITERIA FOR ESTABLISHING ALLOWABLE STRESS VALUES FOR TABLES 5A AND 5B

Product/Material	Below Room Temperature		Room Temperature and Above			
	Tensile Strength	Yield Strength	Tensile Strength	Yield Strength	Stress Rupture	Creep Rate
All wrought or cast ferrous and nonferrous product forms except bolting	$\frac{S_T}{2.4}$	$\frac{S_y}{1.5}$	$\frac{S_T}{2.4}$	$\frac{R_y S_y}{1.5}$	Min. $\left(F_{avg} S_{R\,avg},\ 0.8 S_{R\,min}\right)$	$1.0 S_{C\,avg}$
All wrought or cast austenitic and similar nonferrous product forms except bolting [Note (1)]	$\frac{S_T}{2.4}$	$\frac{S_y}{1.5}$	$\frac{S_T}{2.4}$	Min. $\left(\frac{S_y}{1.5},\ \frac{0.9 S_y R_y}{1.0}\right)$	Min. $\left(F_{avg} S_{R\,avg},\ 0.8 S_{R\,min}\right)$	$1.0 S_{C\,avg}$

Consider the philosophy behind Division 2

To a large degree followed the PED:

- **European Pressure Equipment Directive**

With regard to the hydrotest pressure

The PED: **max(1,43 x MAWP; 1,25 x MAWP x Sa/S)**

Division 2:

8.2.1 Test Pressure

a) Except as noted for vessels of specific construction identified in paragraph 8.1.3, the minimum hydrostatic test pressure shall be the greater of:

$$P_T = 1.43 \cdot MAWP \quad (8.1)$$

or

$$P_T = 1.25 \cdot MAWP \cdot \left(\frac{S_T}{S} \right) \quad (8.2)$$

Consider the philosophy behind Division 2

ASME (Division 1/2) destined Europe have problems

- **Europe uses EN normative material**
- **The USA use ASME approved materials – mandatory**
- **Approvals are awarded by European based Inspection Bodies**

Division 1 is a safer code with larger safety margins than the PED

Europe does not seem inclined to approve ASME materials

This is the end of the presentation

INTERGRAPH