Comparison of pressure vessel codes

Why do the codes differ and How do they differ

Presented by: Ray Delaforce

COMPARISON of the various pressure vessel codes

These are the codes we are going to compare:

- **ASME Section VIII, Division 1**
- **ASME Section VIII, Division 2**
- **PD 5500**
- **EN 13445 Part 3**

But first we look at the most fundamental requirement

What is the ALLOWABLE STRESS ?

This is the primary stress we must not exceed

A PRIMARY stress results from internal pressure

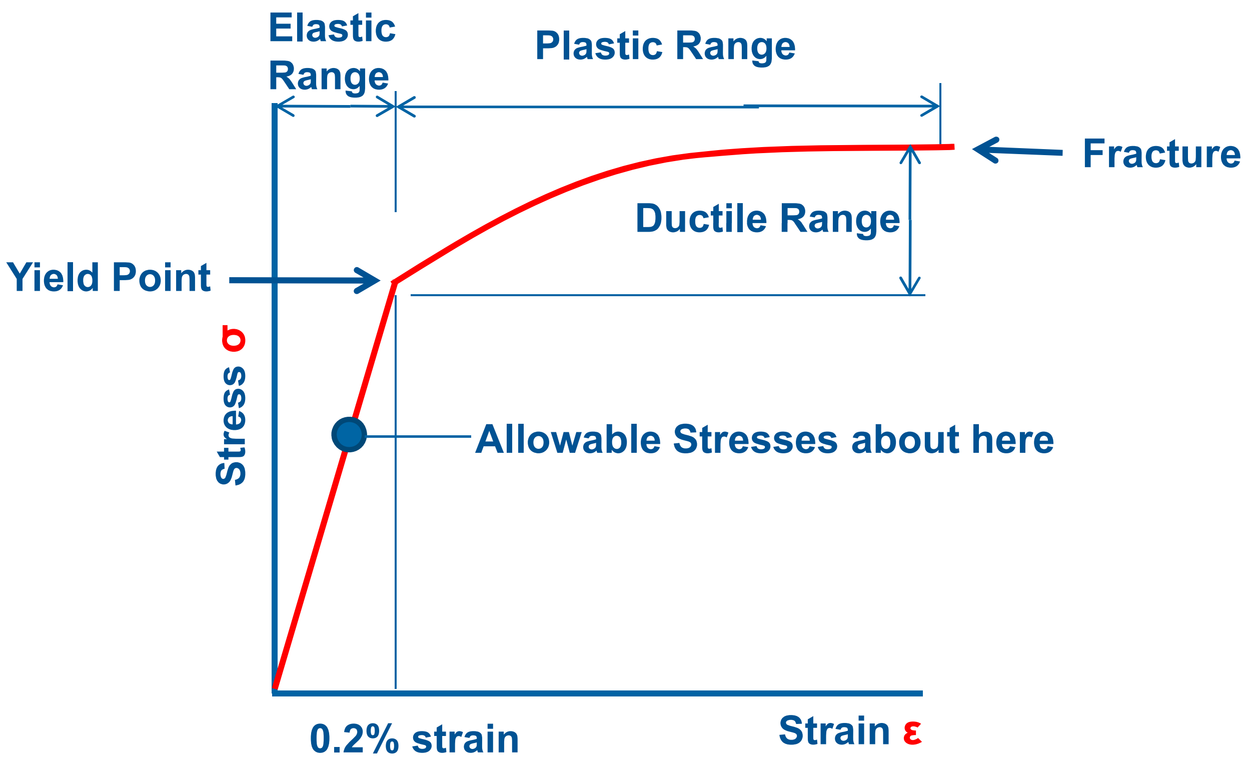
There are SECONDARY stresses – we do not discuss them

We first look at a couple of important material properties

Let us look at the Stress-Strain diagram – we get a lot of information

Collapse can occur when we reach the yield point

Let us look at the important features of our steel



Consider steel: UTS = 70 000 psi (482 MPa) Yield 38000 psi (262 MPa)

Let us look at the Stress-Strain diagram – we get a lot of information

Collapse can occur when we reach the yield point

Let us look at the important features of our steel

There are three important features we must consider

1. There is the limit of proportionality — Yield Point 0.2% strain
2. The Ultimate Tensile Strength (UTS) — When fracture occurs
3. The Ductility = Yield / UTS — Must be less than 1.0

There is a 4^{th} one – Creep which occurs at higher temperatures

Allowable stress is base on these characteristics of the metal

ASME Section VIII Division 1

S = smaller of: UTS / 3.5 or Yield / 1.5 = 20 000 psi (138 MPa)

ASME Section VIII Division 2

Sm = smaller of: UTS / 2.4 or Yield / 1.5 = 25 300 psi (174 MPa)

Both based on PED European requirements

EN 13445

f = smaller of: UTS / 2.4 or Yield / 1.5 = 25 300 psi (174 MPa)

PD 5500

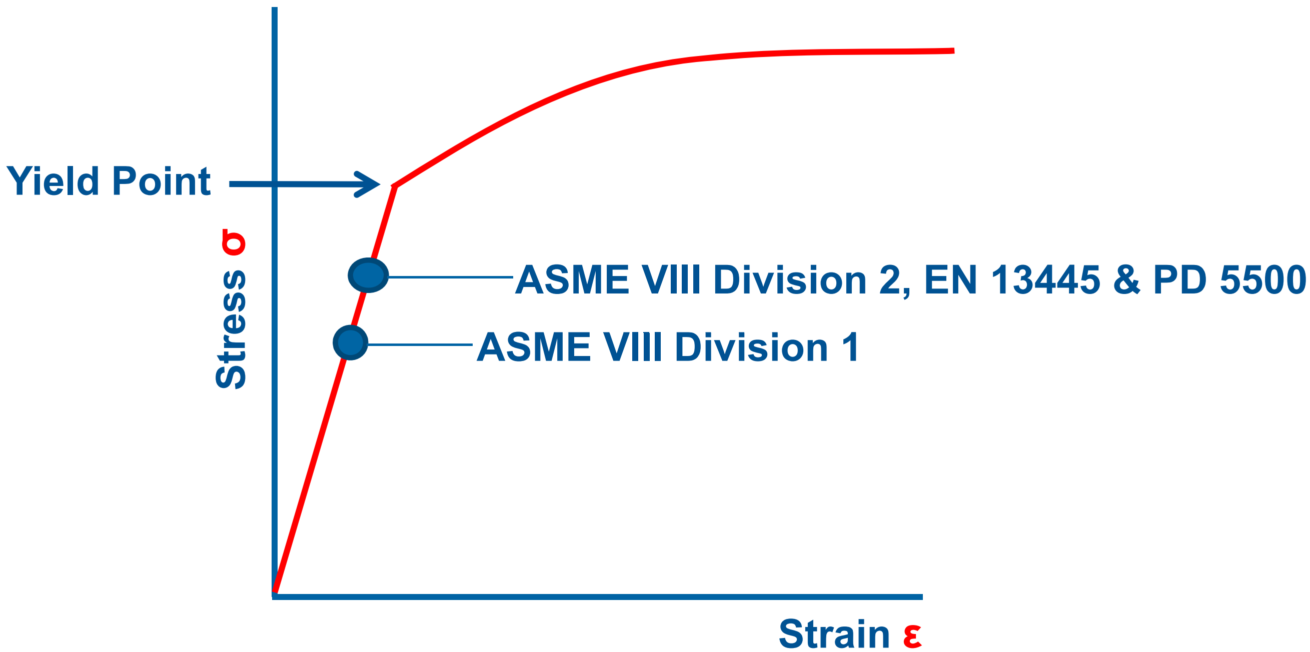
f = smaller of: UTS / 2.35 or Yield / 1.5 = 25 300 psi (174 MPa)

We consider Carbon Steel for simplicity

We look at this on the Stress Strain diagram

ASME VIII, Division 1 has a larger safety margin – safer

This code is still the favoured code throughout the World



Let us now look at a typical calculation – the cylindrical shell

Here are the basic dimensions

We shall ignore joint efficiency E (z)

We now do the calculation for the cylinder:

P = 300 psi (207 MPa)

D = 60 ins (1 524 mm)

S(f) = 20 000 psi (174 MPa)

By ASME VIII Division 1

$$t = \frac{P \cdot D}{2 \cdot S - 1.2P}$$

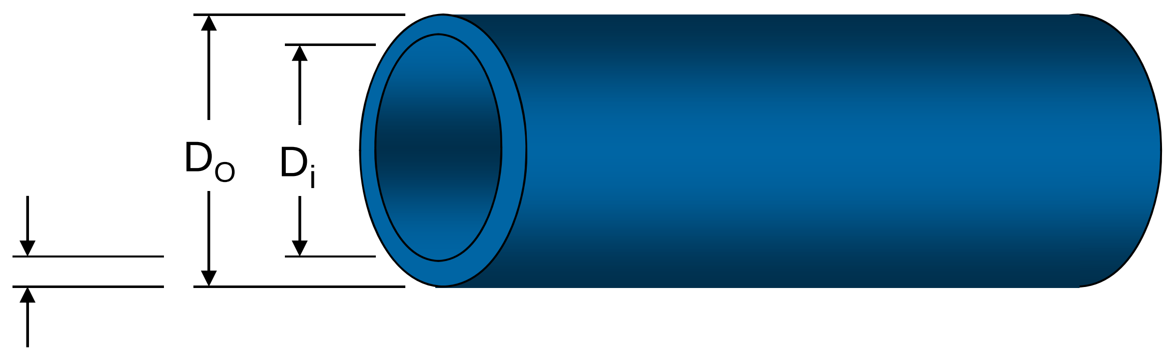
t = 0.454 in (11.534 mm)

By ASME VIII Division 2

$$t = \frac{D}{2} \cdot \left(\exp\left(\frac{P}{S} \right) - 1 \right)$$

t = 0.453 in (11.516 mm)

By EN 13445



Let us now look at a typical calculation – the Cylindrical shell Elliptical Head

Here are the basic dimensions

We shall ignore joint efficiency **E**

We now do the calculation for the cylinder:

P = 300 psi (207 MPa)

D = 60 ins (1 524 mm)

S(f) = 20 000 psi (174 MPa)

$$t = \frac{P \cdot D}{2 \cdot S}$$

That is why the differences are so small – the formulae are nearly the same !

This formula looks odd, but is actually just about the same as the others

By ASME VIII Division 1

$$t = \frac{P \cdot D}{2 \cdot S - 1.2P}$$

t = 0.454 in (11.534 mm)

By ASME VIII Division 2

$$t = \frac{D}{2} \cdot \left(\exp\left(\frac{P}{S} \right) - 1 \right)$$

t = 0.453 in (11.516 mm)

By EN 13445

$$e = \frac{P \cdot D}{2 \cdot f - P}$$

t = 0.453 in (11.516 mm)

By PD 5500

$$e = \frac{P \cdot D}{2 \cdot f - P}$$

t = 0.453 in (11.516 mm)

Let us now look at a typical calculation – the Cylindrical shell Elliptical Head

Here are the basic dimensions

We shall ignore joint efficiency E

We now do the calculation for the cylinder:

Cylinder based on the equilibrium equation

$$t = \frac{P \cdot D}{2 \cdot S}$$

That is why the differences are so small – the formulae are nearly the same !

This formula looks odd, but is actually just about the same as the others

By ASME VIII Division 1

$$t = \frac{P \cdot D}{2 \cdot S - 1.2P}$$

t = 0.454 in (11.534 mm)

By ASME VIII Division 2

$$t = \frac{D}{2} \cdot \left(\exp\left(\frac{P}{S} \right) - 1 \right)$$

t = 0.453 in (11.516 mm)

By EN 13445

$$e = \frac{P \cdot D}{2 \cdot f - P}$$

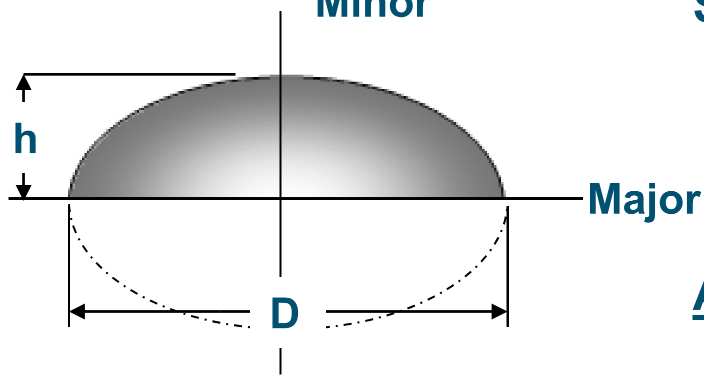
t = 0.453 in (11.516 mm)

By PD 5500

$$e = \frac{P \cdot D}{2 \cdot f - P}$$

t = 0.453 in (11.516 mm)

Let us now look at a typical calculation – the Elliptical Head



Shape is based on true ellipse

D/2h = 2

P = 300 psi (207 MPa)

D = 60 ins (1 524 mm)

S(f) = 20 000 psi (138 MPa)

ASME Division 1 – simple calculation
ASME Division 2 – complicated calculation

$$t = \frac{P \cdot D}{2 \cdot S - 0.2 \cdot P}$$

t = 0.451 in **t = 11.447 mm**

Head formula almost identical to the cylinder formula:

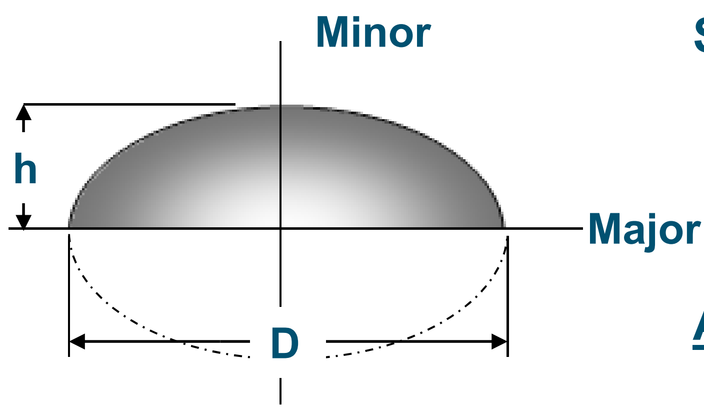
Cylinder:

$$t = \frac{P \cdot D}{2 \cdot S - 1.2P}$$

Elliptical head:

$$t = \frac{P \cdot D}{2 \cdot S - 0.2 \cdot P}$$

Let us now look at a typical calculation – the Elliptical Head



P = 300 psi (207 MPa)

D = 60 ins (1 524 mm)

S(f) = 25 300 psi (174 MPa)

Division 2 allows higher stress

Shape is based on true ellipse

D/2h = 2

ASME Division 2 – complicated calc.

1 There are many steps to do

2 Cannot calculate **t** directly only **P**

On the next slide we show the calculation per *PV Elite*

This is the calculation using *PV Elite* - ASME Division 2

The Elliptical head is transformed in equivalent Torispherical Head

Elliptical Head From 10 To 20 SA-516 70 , Fig. 3.7M Curve B at 20 C

2:1 Elliptical head

Head Calculation - Section 4.3.6 (and Section 4.3.7 for Ellipsoidal Head)

Computed Minimum Required Thickness [t]:

```
tr = 2.7226 mm
t  = tr + ci + co
   = 2.7226 + 3.1750 + 0.0000
   = 5.8976 mm - see below for the derivation
```

For and Elliptical heads: Compute [h, k, r,] per 4.3.7.1

```
h = D/(2*Aspect Ratio)+c = 1524.000 /(2*2.000 )+3.175 = 384.1750 mm
k = Dcorr/(2*h) = 1530.350 /(2*384.175 ) = 1.9917
r = Dcorr*(0.5/k-0.08) = 1530.350 *(0.5/1.992 -0.08) = 261.7470 mm
L = Dcorr*(0.44 * k + 0.02) = 1530.350 * (0.44 *1.992 + 0.02) = 1371.75
```

Crown radius

Knuckle radius

This is the calculation using *PV Elite* - ASME Division 2

Next we must calculate some geometry factors

```
BetaTh = acos[(0.5*D - r)/(L-r) = acos[(0.5*1530.3 - 261.75 )/(1371.75
       = 1.1001 radians - Equation 4.3.8

PhiTh  = sqrt( L * t ) / r
       = sqrt( 1371.750 * 2.723 ) / 261.747
       = 0.2335 radians - Equation 4.3.9

PhiTh < BetaTh, Thus:
Rth    = ( 0.5 * D - r ) / cos( BetaTh - PhiTh ) + r
       = ( 0.5 * 1530.350 - 261.747 ) / cos( 1.100 - 0.233 ) + 261.747
       = 1039.3262 mm - Equation 4.3.10
```

Step 4: Compute Coefficients: Equations 4.3.12 to 4.3.15 [C1 and C2]:

```
r/D > 0.08, Thus:
C1 = 0.692 * ( r / D ) + 0.605
   = 0.692 * ( 261.747 / 1530.350 ) - 0.605
   = 0.7234 - Equation 4.3.13

C2 = 1.46 - 2.6 * ( r/D ) = 1.46 - 2.6 * ( 261.747 /1530.350 )
   = 1.0153 - Equation 4.3.15
```

This is the calculation using *PV Elite* - ASME Division 2

Even more geometry and other factors.and.more – lots of factors

Step 5: Compute pressure to produce knuckle elastic buckling[Peth]:

```
 = C1 * Et * t² / ( C2 * Rth * ( 0.5 * Rth - r ))
 = 0.723 * 202786.203 * 8.177² /
 ( 1.018 * 912.771 * ( 0.5 * 912.771 - 259.080 ))
 = 53.4404 MPa - Equation 4.3.16
```

Step 6: Compute Internal Pressure for max stress in knuckle equals yield [Py]:

```
 C3 = Sy = 262.010 MPa (Not in the creep range - 4.3.6.1(f) )

 Py = C3 * t / ( C2 * Rth * ( 0.5 * Rth / r - 1 ) )
    = 262.010 *8.177 /(1.018 * 912.771 * ( 0.5 * 912.771 /259.080 - 1 ))
    = 3.0275 MPa - Equation 4.3.17
```

Step 7: Value of Internal Press. resulting in buckling of knuckle [G and Pck]:

```
 G   = Peth / Pv = 53.440 / 3.027 = 17.6518
```

This is the calculation using *PV Elite* - ASME Division 2

Even more geometry and other factors and more – lots of factors

```
G > 1, Thus:
Pck = ( ( 0.77508G - 0.20354G^2  + 0.019274*G^3 ) /
        ( 1 + 0.19014G - 0.089534G^2 + 0.0093965*G^3 ) )Py
    = ( ( 0.77508*17.652 -0.20354*17.652^2+0.019274*17.652^3) /
       ( 1+0.19014*17.652 -0.089534*17.652^2+0.0093965*17.652^3))*3.027
    = 6.0530 MPa   Equation 4.3.19
```

Step 8: Compute Allowable Pressure base on buckling of knuckle [Pak]:

```
= Pck/1.5 = 6.054 / 1.5 = 4.0359 MPa - Equation 4.3.21
```

Step 9: Compute allowable pressure based on rupture of crown [Pac]:

```
= 2 * S * E / (L / t + 0.5 )
= 2 * 174.000 * 1.000 / (1371.600 / 8.177 + 0.5 )
= 2.0684 MPa - Equation 4.3.22
```

Step 10: Compute final pressure [Pa]:

```
= Min( Pak, Pac ) = Min(4.036 , 2.068 )
= 2.0684 MPa - Equation 4.3.23: (same as original design pressure)
```

Finally we end up with our starting pressure

PV Elite does an iterative calculation to end up with the pressure

This is the calculation using *PV Elite* - ASME Division 2

We had to start the calculate with a 'guess' thickness t

Computed Minimum Required Thickness [t]:

```
tr = 8.1767 mm
t  = tr + ci + co
   = 8.1767 + 0.0000 + 0.0000
   = 8.1767 mm - see below for the derivation
```

And we ended up with our starting pressure

Step 9: Compute allowable pressure based on rupture of crown [Pac]:

```
= 2 * S * E / (L / t + 0.5 )
= 2 * 174.000 * 1.000 / (1371.600 / 8.177 + 0.5 )
= 2.0684 MPa - Equation 4.3.22
```

We have to use a computer to do this calculation !

The computed thickness is t = 0.3219 in t = 8.1767 mm

This is the calculation using *PV Elite* - ASME Division 2

EN 13445 has a similar method – slightly less complicated than ASME

Required Crown Thickness due to Internal Pressure, see Figure 7.5-3 [es]:

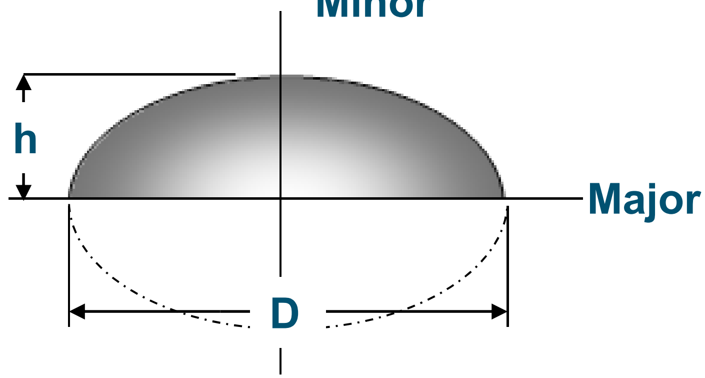
```
 = P * R/( 2 * f * z - 0.5 * P )
 = 2.068 * 1371.6 /( 2 * 174.000 * 1.00 - 0.5 * 2.068 )
 = 8.1767 mm

  ey = Beta*P(0.75*R+0.2*Di)/f =0.62*2.1(0.75*1371.6+0.2*1524.0)/
       174.0000 = 9.8619 mm
  eb = (0.75R+0.2Di)((P/111*fb)*(Di/r)^(0.825))^(1/1.5)
     = (0.75*1371.60+0.2*1524.00) *
       (2.07/111 *156.67)(1524.00/259.08)^(0.825))^(1/1.5)
     = 8.5466 mm
```

Computed Head Thickness per EN13445 - 7.5.4:

```
= Max(es,eb,ey)+c+cext = Max(8.1767,8.5466,9.8619) + 0.0000 + 0.0000
= 9.8619 + 0.0000 +0.0000 = 9.8619 mm
```

The final computed thickness is: t = 0.3886 in   t = 9.8619 mm

The method of computing the head by PD 5500 is very different



1 Calculate h / D = 0.25

2 Calculate P / f = 0.119

P = 300 psi (207 MPa)

D = 60 ins (1 524 mm)

f = 25 300 psi (174 MPa)

PD 5500 uses a graphical solutions – like this

Here is the Graph used to compute this head thickness

e / D

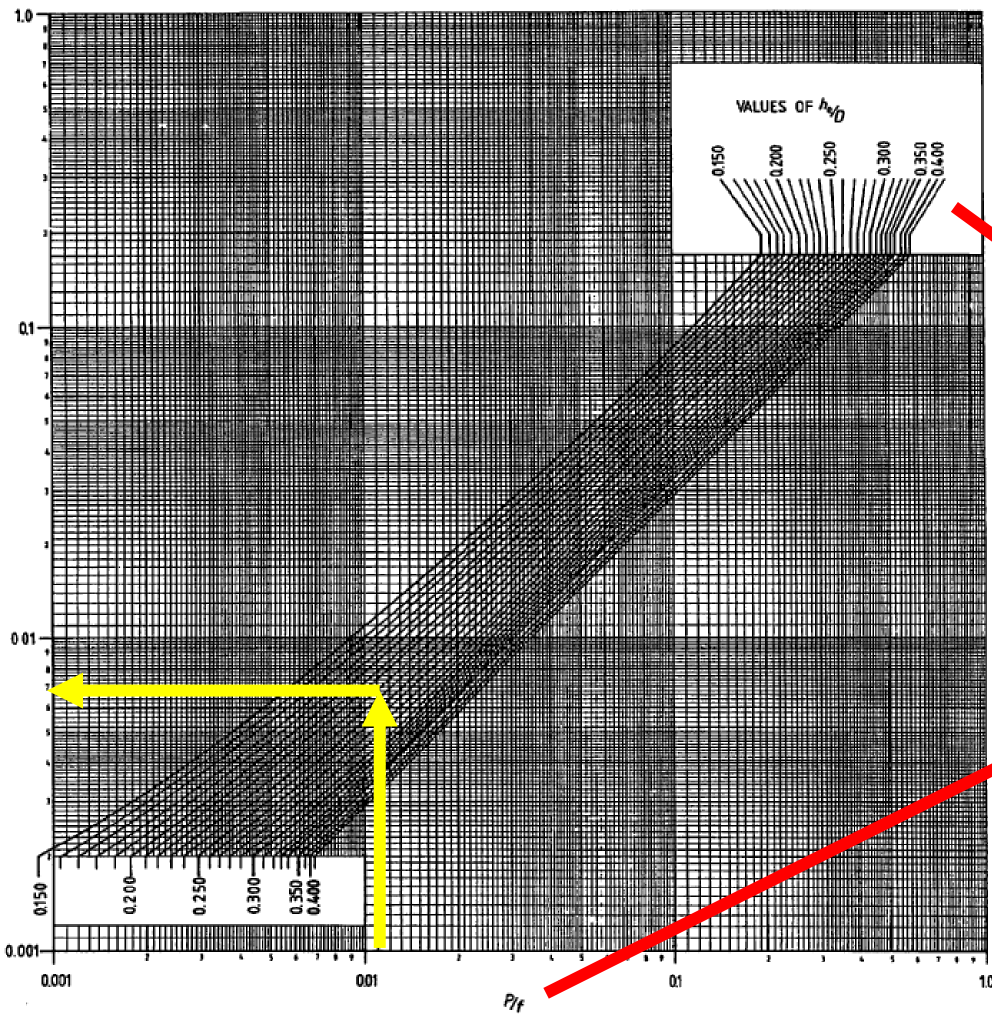
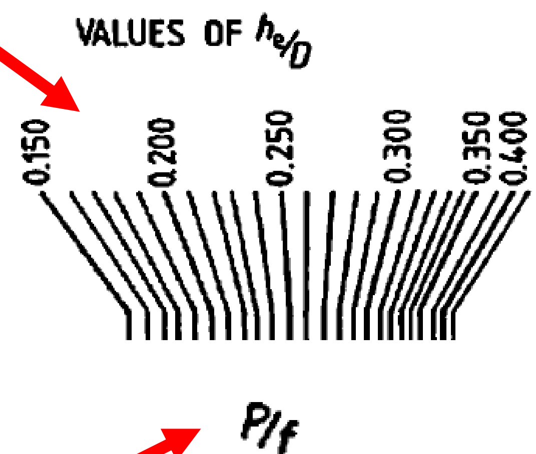


Figure 3.5-2 — Design curves for unpierced dished ends

1 Calculate **h / D = 0.25**

2 Calculate **P / f = 0.119**



e = D x (e/D)

This is the calculation using *PV Elite*

Internal Pressure Calculation Results :

British Standard PD 5500:2009

Elliptical Head From 10 To 20 BS1501-151,430A at 20 F

Thickness Due to Internal Pressure (TR):

```
 = Values_Used_Min( HT, HE1, HE2 ), P/f, he/D, D  Per 3.5.2 Domed Ends
 = Values_Used_Min(15.40,99999.00,99999.00),0.01186,0.2533,60.80
 = 0.3792 + 0.0000 = 0.3792 in
```

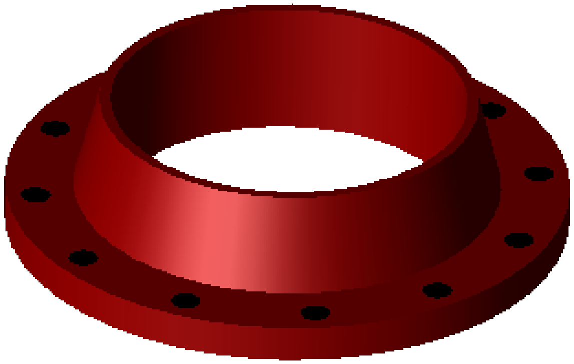
t = 0.3792 in   t = 9.6317 mm

Each code has its own way of computing a head – and other parts

But, where do codes ‘borrow’ procedures from other codes ?

COMPARISON of the various pressure vessel codes

Codes ‘Copy’ codes – some examples   Flange analysis

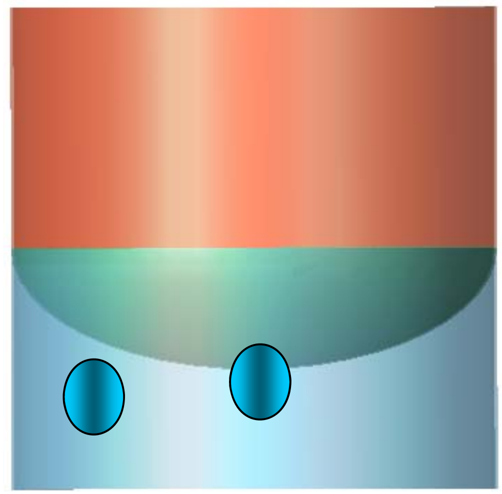


ASME Division 1

ASME Division 2 | **EN 13445-3** | **PD 5500**

Codes 'Copy' codes – some examples

Access openings in skirt



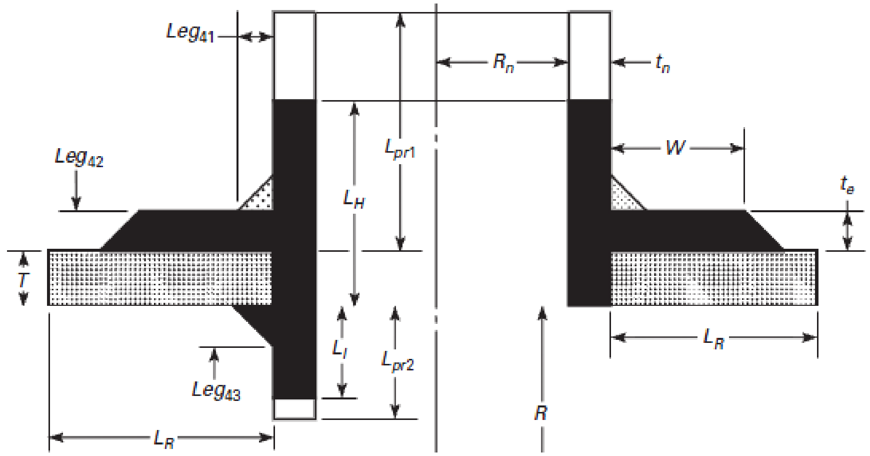
AD Merblatter (AD 2000)

↓

EN 13445-3

Codes 'Copy' codes – some examples

Pressure – Area method



PD 5500

ASME Division 1

ASME Division 2

EN 13445-3

Each of the codes has modified the method – same principle

COMPARISON of the various pressure vessel codes

We have looked at various codes of construction

We have learned some important issues

1. ASME VIII Division 1 requires thicker metal – high safety factor
2. The other codes we discussed use thinner metal, but the allowable stresses are nearer the yield point – less safety
3. Some procedure in the codes have been ‘borrowed’ from other codes
4. ASME VIII Division 2 and EN 13445 are based on the PED (European Pressure Equipment Directive)

It is hoped you got some value out of this webinar