This solution is based on the equations appearing in the Notes - in practice the relevant equations from AS1210 would have to be used.

(**a**)

Design pressure, $p = 1.1$ (say) x $2.5 = 2.75$ MPa   design stress, $S = 108$ MPa

Assume longitudinal joint is critical, so design on basis of longitudinal then check circumferential. Use (**1**) based on mean diameter D_m (= $D_i + t$) rather than on internal diameter D_i as better representing stresses (AS1210 formulae also are based on this). The minimum or *calculation* thickness is :

$t = p (D_i + t) / 2\eta_l S$ and solving for t

$= D_i / (2\eta_l S / p - 1)$ $= 920 / (2 \times 0.85 \times 108 / 2.75 - 1) = 14.0$ mm

[check circumferential similarly :

$t = D_i / (4\eta_c S / p - 1)$ $= 7.9$ mm - less demanding than longitudinal so 14.0 mm stands]

Minimum nominal plate size is thus (14.0 + c) = 16.0 mm. Use commercially available 16 mm plate.

(**b**)

For a seamless semi-ellipsoidal head ($\eta = 1$), from (**vii**) with mean rather than internal dimensions :

$S \geq K p (D_i + t) / 2t\eta$ and solving for t

$t = D_i / (2\eta S / K p - 1)$ not quite the same formula as AS1210

$= 920 / (2 \times 1 \times 108 / 1 \times 2.75 - 1) = 11.9$ mm   as K = 1 for a / b = 2

So minimum nominal thickness is (11.9 + c) = 13.9 mm - use next standard thickness of 16 mm

A 920 x 16 head is not routinely available off-the-shelf, so would have to be manufactured specially.

(**c**)

Find the diametral strain for each component from membrane stresses

<u>cylinder</u>

$\sigma_t = pD / 2t = 2.5 (920 + 16) / 2 \times 16 = 73.1$ MPa

$\sigma_a = \sigma_t / 2 = 36.6$ MPa

$\varepsilon_D = \varepsilon_t = (\sigma_t - \nu \sigma_a) / E = (73.1 - 0.3 \times 36.6) / 207\text{ E3} = 3.00\text{ E-4}$

$\delta_D = \varepsilon_D D_i = 3.00\text{ E-4} \times 920 = 0.28$ mm

<u>head</u> (a / b =2) - from (**vi**) and the resulting graph of stresses

$\sigma_\theta = -pD / 2t = -73.1$ MPa   } ie. same magnitudes

$\sigma_\varphi = pD / 4t = 36.6$ MPa   } as for the cylinder

$\varepsilon_D = \varepsilon_\theta = (\sigma_\theta - \nu \sigma_\varphi) / E = (-73.1 - 0.3 \times 36.6) / 207\text{ E3} = -4.06\text{ E-4}$

$\delta_D = \varepsilon_D D_i = -4.06\text{ E-4} \times 920 = -0.37$ mm   the head bore would decrease if left to its own devices ie. if it were not connected to the cylinder.

(**d**)

For want of manufacturer's actual data use the proportions cited in the Notes :- R / r = 12, R = 0.95D

From (**viii**) $M = (3 + \sqrt{12}) / 4 = 1.62$

$S \geq M p (R_i + t / 2) / 2t\eta$ and solving for t

$t = R_i / (2\eta S / M p - 0.5)$ not quite the same formula as AS1210

$= 0.95 \times 920 / (2 \times 1 \times 108 / 1.62 \times 2.75 - 0.5) = 18.2$ mm

So minimum nominal thickness is (18.2 + c) = 20.2 mm - use next standard thickness of 25 mm ***BUT*** note that if the calculation thickness could be reduced to 18.0 mm by dropping the design pressure from 2.75 MPa to 2.72 (a drop of 1%) then a 20 mm head could be used - a **20%** material saving !!

(e)

Assume that the branch is made from bent and welded plate in a manner similar to the main shell. From above :

$t_b = D_b / (2\eta_l S / p - 1) = 250 / (2 \times 0.85 \times 108/2.75 - 1) = 3.8$ mm

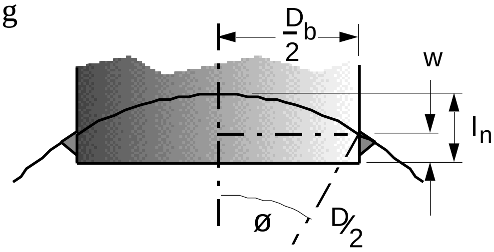
Minimum nominal plate size is thus (3.8 + c) = 5.8 mm. Use commercially available 6 mm plate.

Two extreme cases will be considered :

A the thinnest branch possible (6 mm) which would require an additional reinforcing ring

B the thinnest branch possible which doesn't need extra reinforcement.

First get an idea of the minimum set-in necessary, l_n, for a welding allowance, w, of say 15 mm



$l_n = (D/2)[1 - \cos\{\arcsin(D_b/D)\}] + w = 32$ mm

Assume that the longitudinal welds of both shell and branch are positioned outside the critical longitudinal plane, then $\eta_l = 1$.
In this plane then, from (**a**) and above :

$t = 920 / (2 \times 1 \times 108/2.75 - 1) = 11.9$ mm and $T = 16 - c = 14$ mm

$t_b = 250 / (2 \times 1 \times 108/2.75 - 1) = 3.3$ mm and $T_b = 6 - c = 4$ mm - at end of life.

$A = D_b t = 250 \times 11.9 = 2980$ mm^2

$A_1 = (2L_p - D_b - 2t_b)(T - t)$ and, assuming that $L_p = D_b$ - check this via (**x**) later -

$= (250 - 2 \times 3.3)(14 - 11.9) = 510$ mm^2

A From (**ix**) $L_n = \max[0.8\sqrt{(250 \times 4)} + T_r, \min\{2.5 \times 14, 2.5 \times 4 + T_r\}] = 25 + T_r$

which may be proved by plotting the three terms against likely T_r.

from (**xii**) $A_2 = 2(4 - 3.3)(25 + T_r) = 35 + 1.4 T_r$

$A_3 = 2(4 - 2)(25 + T_r)$ *corroded both sides* $= 100 + 4 T_r$

$A_5 = T_r(2L_p - D_b - 2T_b) = 242 T_r$

from (**xi**) $510 + (35 + 1.4 T_r) + (100 + 4 T_r) + 242 T_r \geq 2980$ ie. $T_r \geq 9.4$ mm

So use a 6 mm thick branch with a 10 mm thick reinforcing ring (no corrosion outside) right out to the $2L_p$ limits. The L_p assumption (**x**) is clearly justified.

B Let unknown branch thickness at end of life be T_b . No reinforcement so $T_r = 0$.

assume (**ix**) $L_n = 0.8\sqrt{(250 \times T_b)} = 12.65\sqrt{T_b}$ mm . . . then proceeding as above for A

$A_2 = 2 \times 12.65\sqrt{T_b}(T_b - 3.3)$ mm^2

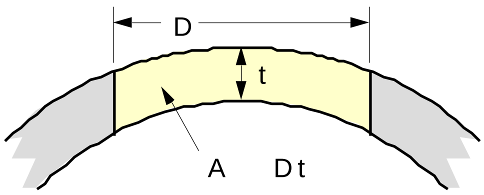
$A_3 = 2 \times 12.65\sqrt{T_b}(T_b - 2)$ mm^2

Solving (**xi**) by trial-and-error yields $T_b = 15.1$ mm and this satisfies the above L_n assumption.

The minimum current branch thickness is (15.1+c) = 17.1 mm and would have to use 20 mm.

The inward protrusion must exceed $L_n = 12.65\sqrt{(20 - 2)} = 54$ mm to comply with reinforcement necessities - easily exceeding the l_n =32 mm curvature requirement.

(f)



Curvature effects are negligible in compensation calculations.

The branch is "set on" so there is no inward protrusion. The hole lies in the inner spherical portion of the head whose inside radius R_i is 0.95 x920 = 874 mm (see (**d**) above) and whose calculation thickness, from (**v**) is :

$S \geq pD/4t$ (seamless η =1) $= p(2R_i + t)/4t$ ie. the calculation thickness is

$t = 2R_i / (4S/p - 1) = 2 \times 874 / (4 \times 108/2.75 - 1) = 11.2$ mm

while from (**d**) $T = 25 - 2 = 23$ mm at critical end of life.

For the branch, from (**e**) $t_b = 3.9$ mm, noting that since there is no “axial plane” in the sphere, the critical plane will be that which contains the branch’s longitudinal weld.

The region at the centre of the heads which can contribute towards reinforcement lies within the 80% limits, that is $L_{p\,max} = 0.8\,(920/2) = 368$ mm - provided the hole lies at the head’s centre - while from (**x**) $L_{p\,max} = D_b$ (say, to be checked) = 250 mm. The latter figure is overiding.

Proceeding as above :

$$A = D_b t = 250 \times 11.2 = 2800 \text{ mm}^2$$

$$A_1 = (2L_p - D_b)(T - t) = (2 \times 250 - 250)(23 - 11.2) = 2950 \text{ mm}^2$$

The thickness of the torispherical end is so large (being dictated by knuckle stress concentration outside the 80% region) that the area removed for this hole within the 80% region is compensated automatically by the excess material in the region - no reinforcement contribution is required from the branch. The branch thickness need only be the minimum necessary to withstand pressure in the branch - 6 mm - see (**d**) above, part **A**.

(**g**)

For T = 350 ^oC, using (**xiii**)

$$p^{FT} \geq 2.75 \times \max(1,\ 390/(622\text{-}350),\ 210/(532\text{-}350)) = 3.94 \text{ MPa} \quad \text{so need flange table J}$$

For T = 450 ^oC, using (**xiii**)

$$p^{FT} \geq 2.75 \times \max(1,\ 390/(622\text{-}450),\ 210/(532\text{-}450)) = 7.04 \text{ MPa} \quad \text{so need flange table R}$$

(**h**)

The periphery of the gasket is approximately $\pi(337 + 237)/2 = 900$ mm, the effective contact area of the gasket is therefore $900 \times 7 = 6300$ mm^2, and the total load necessary to generate 10 MPa over this area is $6300 \times 10 = 63$ kN.

The total stud area required with 100 MPa design stress is $63000/100 = 630$ mm^2, so can use either a single M36 stud (as 759 >630) which will be rather cumbersome, or two M24 studs (as $2 \times 324 > 630$) - leave this decision until after selection of door thicknesss via (**xiv**) :

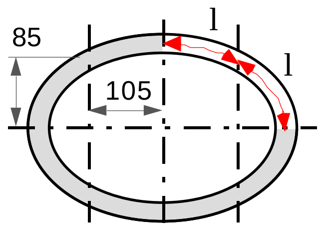
one stud $\quad t = \sqrt{[(0.4 \times 2.75 \times \pi/4 \times 330 \times 230 + 0.8 \times 100 \times 759)/108]} = 34.2$ mm

two studs $\quad t = \sqrt{[(0.4 \times 2.75 \times \pi/4 \times 330 \times 230 + 0.8 \times 100 \times 324)/108]} = 27.9$ mm

Allowing for corrosion, the thinnest commercial plate which could be used is 40 mm for the door with one stud and 32 mm for the door with two. The latter will be lighter, cheaper and probably less difficult to seal as two studs distribute the initial sealing force more uniformly than one (see below).

Select a 32 mm thick door with two M24 studs.

(**j**)

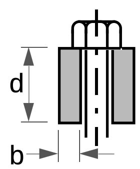


One end of each of the two bridges pressurises a quadrant of the gasket. A bridge should be positioned centrally in the area it pressurises - ie. the lengths ***l*** around the periphery are about equal, as sketched. From approximately scaled sketch the span of each bridge is 2(85+10) = 200 mm say.

If bridge fails at same load as studs and door, then central load on each bridge is $100 \times 324 = 32.4$ kN

Assume bridge is made from two bars each b x d welded to 25 mm spacers, allowing studs to pass between them. For each simply supported centrally loaded bar :

$$\sigma = My/I = (16200 \times 200/4)\, 6/bd^2 \leq 108 \text{ MPa}$$



which with d/b = 3.5 say, suggests bars of 16 x55 mm cross-section in way of the stud. Note that rounding dimensions up may result in bridge being stronger than bolt and dooor - this may not be desirable as a visibly yielding bridge may act as an overload warning while tightening the studs.