270 stroke → 0.000387 m³ + 0.0006 = 0.000957 m³ total

which 1 kg argon will occupy at around 123 MPa, requiring
0.67 MPa air + 10 MPa gas
Intensifier Design

Haskell gas booster takes pressure to 170 MPa at about 130 psi air, 2000 psi gas (0.7 Mpa air, 14 Mpa gas).
With 0.7 Mpa air and 10 Mpa gas, gas booster will go to 130 MPa.

From previous calculations, the bomb will accommodate 0.0006 m$^3$ = 1 kg argon at 700 MPa.
This amount of gas occupies 0.0009 m$^3$ at 150 MPa
= 0.001050 m$^3$ at 130 MPa (0.0008 at 220 MPa).
Thus, for one-stroke operation from 130 MPa, we need an intensifier volume of 0.0034 m$^3$ = 3400 cc = 20.7 in$^3$.
With an intensifier piston diameter of 41 mm, this corresponds to 258 mm of stroke — say 275-300 strokes.

For 30 mm diam, stroke = 480 mm, a bit long.
The 41 mm is chosen as having the same O-ring (225) as the compensating piston (42.4 diam).

If we choose 41 mm diam and 10:1 ratio, then old piston is 129.65, say 130 mm diameter
and the oil pressure is 70 MPa for 700 MPa pressure (100 MPa for 16 MPa pressure) [10,000 psi; 14,500 psi].
Thus we need a Haskell AW-100 or AW-122 pump with 100 psi air supply (or AW-150 if 16 MPa is required).

For oil cylinder, use 212/125 (or 180/125) hollow bar of En-19

For gas cylinders, need O/d of at least 3.5, i.e. $D = 144$ mm

[cont'd on p.13]
Barricade Design

The PVT data for argon at room temperature are set out in the 1981 Servo-Cryop Standpipe calculations book. The perfect gas law can be applied (within a few percent) up to 10 MPa and above that integration under a PVT plot is used to obtain the available energy in isothermal expansion from \( f \) to \( dp \).

The total pressure vessel volume is 0.0012 m\(^3\) and normally the load cell and furnace occupy about half of this. However, in case other types of experiment are set up, that occupy less space, we shall assume that three-quarters of the space is gas-filled, that is 0.0009 m\(^3\).

In practice, for design of a barricade we need the available energy in an adiabatic expansion. In the perfect gas region we have \( pv = C \) where \( \gamma = \frac{C_p}{C_v} \).

If we start at a pressure \( p_0 \) and known volume \( v_i \) (from the room temp PVT data), then \( C = p_1 v_i \) and so

\[
\frac{1}{\gamma} = \frac{1}{v_i} \frac{p_1}{p_0} \frac{1}{\gamma} \frac{v}{v_i} - 1 \nabla p_1
\]

\[
\frac{1}{\gamma} \frac{v}{v_i} \frac{p_1}{p_0} \frac{1}{\gamma} \frac{v}{v_i} - 1 \nabla p_1
\]

\[
E = \frac{\gamma}{\gamma - 1} v_i p_1 \frac{1}{\gamma - 1} \left( \frac{p_1}{p_0} \right)^{\gamma - 1} - 1 \nabla p_1
\]

From room temperature PVT data, \( v_i = 0.00588 \text{ m}^3/\text{kg} \) at \( p_0 = 10 \text{ MPa} \) and \( \gamma = 5/3 \) (Lieberberg et al. JAP 1974 Fig. 7.15), so

\[
E = \frac{\gamma}{\gamma - 1} v_i p_1 \frac{1}{\gamma - 1} \left( \frac{p_1}{p_0} \right)^{\gamma - 1} - 1 \nabla p_1
\]

\[
E = \frac{\gamma}{\gamma - 1} v_i p_1 \frac{1}{\gamma - 1} \left( \frac{p_1}{p_0} \right)^{\gamma - 1} - 1 \nabla p_1
\]

\[
= 0.124 \text{ MJ/kg}
\]

In fact, if the pressure vessel has burst at a higher pressure, the gas will be considerably colder than room temperature by the stage that it has the pressure has dropped to 10 MPa, certainly below 273 K by observation on controlled pressure release; we could conservatively assume around -10°C, i.e. around 0.9 times room temperature and hence room volume, reducing \( E \) to 0.110 MJ or less.
This quantity can be compared with the isothermal energy release of
RT \ln \left( \frac{p_1}{p_0} \right) = 8.314 \cdot 295 \cdot \ln \frac{10}{10^5} \text{ per mole}

\approx \frac{8.314 \cdot 295 \ln 100}{0.003995} J/kg
= 0.283 MJ/kg

We now consider the 100 MPa to 10 MPa expansion. At 100 MPa,
the perfect gas would occupy \( \frac{8.314 \cdot 295}{10 \cdot 0.003995} = 0.000614 \) m³/kg
whereas the real gas occupies about 0.00104 m³/kg, i.e.
1.69 times the perfect gas volume, so on average through
this interval the real gas has about 35% more volume than
the ideal and so for psi could \( S_{vp} \) could be expected to
be of the order of 35% greater than calculated for perfect
gas expansion. \( RT \ln \left( \frac{p_1}{p_0} \right) \) for this interval is 0.141
MJ/kg, which would become 0.191 MJ/kg if increased
35 percent. The actual isothermal energy release from
the pVT curve at room temperature is 0.157 MJ/kg,
suggesting that the 35% is an overcorrection. If we
take the real gas at 100 MPa, occupying 0.00104 m³/kg at
room temperature and expand this adiabatically to 10 MPa
pressure as if it were perfect gas but using the observed
heat capacity ratio \( \gamma \approx 2 \) (Lichtenberg et al., JAP 1974
45 745 Fig 5) we have

\[ E = \frac{2}{2-1} \cdot 0.00104 \cdot 10 \left( \frac{8.314 \cdot 295}{10} - 10 \right) \]
\[ = 0.142 \text{ MJ/kg} \]

Taking into account that the temperature may already be
near to 0°C, say a few °C, we can reduce the above value
by say 5% to \( E = 0.135 \text{ MJ/kg} \).

Note that this compares with 0.157 MJ/kg real isothermal,
whereas in the 10 to 0.1 MPa range, the adiabatic figure was
0.110 MJ/kg compared with 0.283 MJ/kg isothermal; that is,
The factor adiabatic/isoenthalpic has increased from 0.39 to 0.86 as we go from 10-0.1 MPa to 100-10 MPa interval. Therefore at the higher pressures we expect a less difference and so will take the full isothermal expansion energy as a conservative figure for design. Therefore we have (in 1981 book) as energies to be allowed for:

<table>
<thead>
<tr>
<th>Pressure Interval</th>
<th>Work (MJ)/kg</th>
<th>Cumulative Energy (MJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1 - 10 MPa</td>
<td>0.110</td>
<td>0.245</td>
</tr>
<tr>
<td>10 - 100 MPa</td>
<td>0.135</td>
<td>0.412</td>
</tr>
<tr>
<td>100 - 500 MPa</td>
<td>0.167</td>
<td>0.549</td>
</tr>
<tr>
<td>500 - 700 MPa</td>
<td>0.137</td>
<td>0.675</td>
</tr>
<tr>
<td>700 - 1000 MPa</td>
<td>0.126</td>
<td>0.849</td>
</tr>
</tbody>
</table>

Note that the figures for the lower pressure intervals are only valid when the expansion has begun from substantially greater than 100 MPa because of the allowance for some earlier cooling.

Now assuming three-quarters free volume in the pressure vessel, the available release energies are:

<table>
<thead>
<tr>
<th>Maximum Pressure</th>
<th>Work (MJ)/kg</th>
<th>Mass (kg)</th>
<th>Total available energy (MJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>300 MPa</td>
<td>0.412</td>
<td>1.22</td>
<td>0.50</td>
</tr>
<tr>
<td>500 &quot;</td>
<td>0.549</td>
<td>1.38</td>
<td>0.76</td>
</tr>
<tr>
<td>700 &quot;</td>
<td>0.675</td>
<td>1.49</td>
<td>1.00</td>
</tr>
<tr>
<td>1000 &quot;</td>
<td>0.849</td>
<td>1.61</td>
<td>1.37</td>
</tr>
</tbody>
</table>

Thus we shall design on the basis of 1 MJ of released energy, which should be valid for three-quarters free volume at 700 MPa or a little more than half free volume at 1500 MPa.

This compares with about 7 MJ in a full gas bottle (8 m³ at STP) at 17 MPa or 0.22 kg TNT (4.56 kJ/g) assuming 1 MJ at 15 MPa would be available.

This calculation is done under isothermal conditions.
Filling the cavity with 1.5 kg argon (0.84 m$^3$ at NTP) would raise the pressure about 0.08 MPa.

Ignoring the walls, cross-section of four 90x150x8 angles = 0.0827 m$^2$, and so stress is 78 MPa in near-horizontal cross-section, and 132 MPa in vertical cross-section.

Assuming 8 mm plate & 89x89x8 angles (1.134 x 1.484)

A pair of additional 90x90x8 angles across the middle of the two side walls would reduce the latter stress to 111 MPa — probably not needed. The duration of the blast burst will be very short — so important that welds must be strong.
Blast effects

The first question is how much of the available energy goes into the blast or shockwave and how much is dissipated otherwise, as in fragments. The HPTA code code suggests 0.8E in shockwave energy and 0.2E in fragment kinetic energy for complete brittle fracture of the pressure vessel. In our case there is a constraining outer sleeve that will absorb some energy (calculated later) and lead to considerable mottling of the blast, thereby reducing its shockwave effect. In view of this consideration, 0.8E is probably a reasonably conservative figure to take, i.e. 0.8MJ available for blast at 700 MPa.

To calculate the equivalent static pressure on the barricade, two formulae are given in the literature:

1. \[ p = \frac{E}{v} \] in Frye (ASME PVPP0148, 1981, p145)
2. \[ p = 0.76 \left( \frac{E}{v} \right)^{0.72} \text{ MPa} \] when \( E \) in MJ & \( v \) in m³ in HPTA Code

(1) gives \[ p = \frac{0.8 \times 10^6}{1.3} = 0.62 \text{ MPa} \] suggesting that we take 0.6 MPa as the equivalent static pressure.

(2) gives \[ p = 0.76 \left( \frac{0.8}{1.3} \right)^{0.72} = 0.54 \text{ MPa} \]

The above volume of 1.3 m³ is based on the dimensions of 1.15 x 0.90 x 1.52, less the volume of the pressure head and other items.

In horizontal cross-section, there is 0.025 m² of steel in the walls & corners to support the blast on a cross-section of 0.979 m², leading to a mean stress of \( \left( 0.979/0.025 \right) \times 0.6 = 23.5 \text{ MPa} \).

In vertical cross-section (one-half; in each case the sliding doors are not load bearing), there is 0.0186 m² to support blast on 0.814 m², leading to a mean stress of 26.3 MPa.

Taking a tensile strength for mild steel of over 600 MPa, there is an adequate factor of safety against splitting open.
Treating the walls as freely supported plates, Marks Metal Engineering Handbook (1952 p. 449) gives
\[ \sigma_{\text{max}} = 0.72 \left( \frac{4 \tau}{E} \right) \frac{2P}{1 + \left( \frac{h}{d} \right)^2} \]
for stress on diagonal,
\[ \sigma = 6 \left( \frac{0.36}{1 + \left( \frac{h}{d} \right)^2} \right)^{1/2} \]

For front wall, \( b = 722 \), \( a = 1132 \text{ mm} \), so taking \( \sigma = 250 \text{ MPa} \) as yield stress,
\[ \tau = 722 \left( \frac{0.36}{1 + \left( \frac{722}{1132} \right)^2} \right)^{1/2} = 17.9 \text{ mm} \]

For side wall, \( b = 722 \), \( a = 882 \)
\[ \tau = 722 \left( \frac{0.36}{1 + \left( \frac{722}{882} \right)^2} \right)^{1/2} = 15.7 \text{ mm} \]

This plate of the order of 20 mm would be called for to support such a blast elastically unless the plate is reinforced. Heavy ribs on the centre lines vertically & horizontally would effectively reduce the plate by a factor of two in both \( a \) & \( b \) dimensions and so halve the required plate sizes. Then 8 mm would probably suffice if going into the plastic range is allowed as the safety factor.

However, it is worth considering whether the above estimate of 0.6 MPa as the design blast pressure is not too conservative. It makes very little allowance for the use of a ductile steel plate around the pressure vessel in absorbing some energy and thumping the gas so that as to reduce the blast effect. Some sort of calibration on this factor is obtainable from the experience of a brittle pressure vessel failure in 1967. Here the pressure was 500 MPa at failure but due to the use of fireproofing as insulation, the free volume in the vessel may have been only about \( \frac{3}{4} \) instead of the \( \frac{3}{4} \) assumed above, that is, available energy about 0.76/3, say 0.25 MJ.

The enclosure was about 3\( \times \)3 m, say 27 m\(^3\), & there was a window on one side. This was not shattered & a visit to the blip suggests vol. \( \sim 50 \text{ m}^3 \).
HPTA Code 73 seems to indicate that this value of I should be multiplied by 5. Is this because of possible directional nature of the blast?
so we can assume that the blast pressure was less than 0.003 MPa (HTPA code; Fryer, 1981). Then, using formula (1) on p 6, we have

\[ E < 0.003 \times 10^6 J = 0.08 MJ \]

or using formula (2),

\[ E < \frac{0.003}{0.76} MJ = 0.012 MJ \]

If we take the latter figure as the more accurate, then 5% of the available energy is in the blast wave, not 80% as taken above. Even if we take the other figure of 0.08, it corresponds to 32% of the available energy being in the blast wave.

It therefore seems to me that, in this situation, that is, with a ductile safety sleeve around the pressure vessel, it would be conservative and possibly substantially so, to assume that not more than 20% of the energy will go into the blast wave, for which the formula (2) on p 6 gives

\[ p = 0.76 \left( \frac{0.2}{1.3} \right)^{0.72} = 0.20 \text{ MPa} \]

In this case, the required thickness of plate for elastic constraint is 10.3 mm for front wall and 9.5 mm for the side walls. This is probably a conservative estimate and there is also a considerable factor of safety in being able to go into the plastic range.

For ear damage from the shock wave, we take the consideration in HTPA p73 according to which the impulse on the wall is

\[ I = 0.008 E^{\frac{1}{3}} (R/E)^{-1.2} \text{ Nm}^{-2} \]

and \[ V = \frac{I}{m} \] is the velocity imparted to the air external to the barriers \((m = \text{mass/unit area of wall})\).

In our case, with \( E = 0.2 \text{MJ} \) and \( R = 0.5 \text{m} \) (with respect
a steel wall 0.010 m thick

The HPTA code allows up to $3 \text{ m s}^{-1}$ before rear damage, so this is OK. However, the situation at the back where $R$ may be only 0.2 m is less favourable. $I = 425.8 \text{ g V} = 5.46 \text{ m s}^{-1}$

A thicker rear door would alleviate this; 12 mm would lead to 4.5 m s$^{-1}$ but 18 mm would be needed to bring it right down to 3 m s$^{-1}$

An alternative approach is given by Fryer 1981 (p147) who gives

$$\text{db} = 20 \log \left( \frac{P}{1\text{MPa}} \right)$$

where $P$ is the shockwave pressure. For $P = 0.2 \text{ MPa}$ we get 200 db, compared with an allowable 170 db. He recommends 20 db as a conservative figure for attenuation of a steel plate, so a double steel plate with wood sandwiched between should be adequate.
Missile Effects

(1) Complete brittle fracture of pressure vessel. HPFA Code p67. It is proposed that fragment kinetic energy is 0.2 of total available energy, calculated above to be 1075 J in worst case of only 25% filling.

\[ t = \frac{(2E)^{1/2}}{m_{\text{fracture}}} \times \frac{0.25 \times 10^4}{49 \text{ ms}^{-1}} \times \frac{1 + 5 \times 10^{-5}}{0.23 \times 0.6} = 0.0014 \text{ m} \]

\[ = 1.4 \text{ mm} \]

Even with 0.8 of energy, \( t = 4.4 \text{ mm} \).

(2) Major failure e.g. transverse fracture at bottom of land plug threads. \( m = 16 \text{ kg} \), \( A = \frac{\pi}{4} (0.23)^2 \)

Code suggests 0.6 total energy \( = 0.6 \text{ MJ} \)

\[ t = 6.6 \text{ mm} \]

However, code also suggests calculating energy from work done by full pressure over underside until it clears one diameter. Since the bore opening at the plug is 77 mm, we have

\[ E = \left(\pi \times 0.077\right) \times 700 \times 10^6 \times 0.077 = 0.25 \text{ MJ} \]

Which leads to \( t = 3.9 \text{ mm} \).

In fact, a portion of the top of the plug is 120 mm diameter, giving \( A = \frac{\pi}{4} (0.12)^2 \). In this case with \( E = 0.25 \text{ MJ} \), we get \( t = 14.5 \text{ mm} \). This calculation is a bit conservative because the pressure will drop to around 400 MPa or lower with this amount of gas expansion.
G.I. Taylor (Collected works vol 3 p363) gives another approach from exploding bombs, which would appear to lead to larger velocities but it is not easy to relate—he talks coupling to continue to about 10x original explosive diameter.

Ballistic Research Lab formula in Bornem (Appl Mech Rev 1986 39 186) 42 mm for penetration or 52mm suggested thickness to prevent perforation. Applies for any shape s so is presumably conservative for blunt pieces.
If only the end plug itself is ejected, \( m = 9 \text{ kg} \), we have \( t = 0.010 \), \( \approx 10 \text{ mm} \).

The worst case is the small inner end plug, with 65 mm of thread, 40 mm dia, which we assume to travel \( 65 + 40 = 105 \text{ mm} \) before decoupling.

Energy = \( 700 \cdot 10^6 \cdot \frac{\pi}{4} \cdot (0.040)^2 \cdot 0.105 = 0.092 \text{ J} \)

\( m = 0.83 \text{ kg} \) \( A = 0.0013 \)

Penetration \( t = 17.9 \text{ mm} \) by large fragment formula.

Using the small fragment formula

\[ t = 3.8 \cdot 10^{-5} m^{0.33} \n \]

\[ = 3.8 \cdot 10^{-5} m^{0.33} \left( \frac{2E}{m} \right)^{0.5} = 5.37 \cdot 10 \text{ m} \]

\[ \text{gives} \ t = 16.8 \text{ mm} \]

Using the rod-shaped missile formula (p.71)

\[ t = 3.1 \cdot 10^{-7} \frac{E}{d^{0.71}} \]

\[ \text{we get} \ t = 37 \text{ mm} \]

Since this plug is only marginally rod-shaped, this figure is probably exaggerated a bit. Also the gas will be slightly throttled by passing through the 30 mm hole in the plug — but not much.

So 40 mm of plate above the pressure vessel should be adequate protection. But maybe better make it 50 mm.
Ballistic Research Lab formula (see p10a) leads to $e = 22\text{ mm}$
Intensifier Case
The maximum energy case will be that of filling with a Haskel gasjetting to 25,000 psi = 170 MPa (will need 150 psi air) and then raising to 700 MPa with the valve closed.

If stroke is 300 mm, piston diameter 41 mm, and overall length of intensifier cylinder 450 mm, we have gas volume
\[ V = \frac{\pi}{4} (0.041)^2 \times 0.3 \]
\[ = 0.00040 \text{ m}^3 \text{ initially} \]

At 170 MPa, this corresponds to 0.46 kg of argon. Thus if this is taken to 700 MPa and then failure occurs, the energy available on release to about 100 MPa is (from p. 5)
\[ (0.126 + 0.137 + 0.167) \times 0.46 = 0.198 \text{ MJ} \]

With mass = 62 kg, \( \frac{E}{A} = 80 \text{ ms}^{-1} \) may be \( 0.2 \text{ MJ} \).

The large fragment formula (p 10) gives penetration in mild steel of
\[ t = 10.6 \text{ mm} \]

If a safety shell of 212 mm o.d. is shrunk on,
- mass = 124 kg
- \( A = 0.0353 \text{ m}^2 \)
- Then \( t = 5.7 \text{ mm} \)

The above case refer to failure at the bottom end of the intensifier, sometimes 9 the whole body being accelerated by the piston.

Alternatively if the top end came off at the end of the bore say 100 mm length, then the energy
\[ E = \frac{\pi}{4} (0.041)^2 \times 0.04 \times 700 \times 10^6 \times 0.03 = 0.0120 \text{ MJ} \]

This case mass = 14 kg, \( A = 0.0177 \) for 150 mm dia, and \( t = 2.2 \text{ mm} \) — no problem.

So 20 mm above the intensifier should be adequate protection. Other aspects are covered by the present calculations.
Intensifier Design (contain from P 2)

Probably should make intensifier wall ratio ~ 4:1, i.e. 160 mm o.d. for 41 mm bore. With 6 mm pitch thread, root dia of external thread is ~ 152 mm. We shall design wall basis of 15 Pa pressure in case it is desired in the future to go to that pressure & in any case to give some margin at 700 MPa

\[ \frac{P_d}{P_a} \times 1000 = 1141 \text{ MPa as max hoop stress} \]

in the wall. This slightly exceeds the yield stress of steel at 42 Rc (around 1150 MPa). So if it is intended to go to 15 Pa then the steel probably should be heat treated to 44 Rc (y. stress a bit over 1150 MPa) to get 1200 MPa would require 45-46 Rc.

The max axial force on 41 mm dia is 1320 kN, so with a shear stress of 100 MPa in the threads, length of thread would be

\[ \frac{1320000}{\pi \times 0.152 \times 100} = 0.0276 \text{ m} \]

or 28 mm.

If we use En19 steel for oil cylinder, this has a y. stress around 870 MPa (25-33 Rc) maybe a bit higher. So perhaps 50-70 MPa is a better stress for threads, leading to a thread length of 40-55 mm — take 50 including remant.

For 41 mm piston, use a 223 o’ring (fnd 41ID, 47O), section 3.53 φ. We need 15-20% squeeze on the O-ring (squeezes of ~ 10% seem to be marginal for high pressure work, maybe 15% is a minimum and 20% is starting to get difficult to seal; 18-20% may be a good range to aim at). So O-ring gap should be \( \frac{3.99}{2.82} \) to 2.82 (3.00 gap gives 15% squeeze), leading to 35 \pm 0.25 to 35.25 -- 46.79 to 46.65 for O.D. of O-ring groove, say 46.7 ± 0.1 or 47.05 -- 47.75 46.65 -- 46.75.
Assumes tolerance bar 212/130
The seal nut then has to support the pressure acting on 46.7 OD, 41.0 ID, which at 14 kPa gives a force of 0.393 MN
For 100 MPa on the threads of say 56 OD, the length of thread is
\[
\frac{0.372}{\pi \times 0.058 \times 100} = 0.027 \text{ m} = 22 \text{ mm} \text{ at } \frac{35}{30} \text{ mm}
\]
Length of engagement, say 30 total should be OK revised p17

For the 130 mm piston, use a 426 O-ring \((116 \text{ ID}, 130 \text{ OD})\) of 6.99 mm section. 15% squeeze = 5.94 groove, 20% = 5.59 groove, say 5.7 groove.

Expansion of oil cylinder:
\[
\frac{\Delta r}{r} = \frac{P}{E} \left(1 + \frac{R^2}{R_t^2} \right) + \frac{V}{E}
\]

Put \(P = \frac{1000}{1000} \text{ MPa}, E = 200 \times 10^3 \text{ MPa}, R_t = 210/130, V = 0.27\)

\[
\frac{\Delta r}{r} = 0.060 \times \left(0.361 + 0.27 \right) = 2.51 \times 10^{-3}
\]

\[
\Delta r = 0.08 \text{ mm.}
\]

At the seal it will be about half this (unpressured one side pressurized the other) ie about 0.04 mm.

Poisson's expansion of piston:
\[
\frac{\Delta r}{r} = 10^{-3} \times 0.27 = 0.001 \text{ mm.}
\]

So net increase in clearance \(= 0.06 \text{ mm}\).

Aguire booklet allows radial clearance up to 0.1 mm at this O-ring size but probably does not envisage pressures beyond about 50 MPa. However, this would be without middle ring.

So it is doubtful whether we need provision for compensation as in No 1 explantret intensifier.

Oil closure nut: With 160 x 6 thread, minor diameter is about 156. Length of thread:
\[
\frac{\pi \left(130^2 - 100^2 \right)}{\pi \times 156 \times 0.05} = 54 \text{ mm if assume}
\]
50 MPa shear stress

High pressure piston seal:

Diametral clearance = \(4d\)

Then pressure to close the clearance is obtained from
\[
\Delta F = \text{strain} = \frac{P \times d}{E \times 25}
\]
Use 203 mm premachined 718 bar

\[ \frac{\Delta T}{T} = 1.37 \times 10^{-3} \]

so \( D_d = 0.178 \text{ mm} \)

Capacity: stroke 302, dia 41 \( \rightarrow 398.7 \text{ cm}^3 (24.3 \text{ cm}) \)

(Hartwood DA14 100,000 psi has 28.8 cm
150,000 psi 17.8 cm
SA14 100,000 33.13 cm
150,000 21.20 cm

Ratio in \( \left( \frac{130}{41} \right)^2 = 10 : 1 \)
that is, \( \phi = B \frac{2.75 \cdot d}{d} \)

Taking \( E = 800,000 \text{ MPa}, \ d = 41 \), \( \phi = 240 \ t \cdot d \).

With \( t = 0.5 \), \( d = 0.025 \), \( \phi = 3 \text{ MPa} \)

\( t = 1.0 \), \( d = 0.05 \), \( \phi = 12 \text{ MPa} \)

Thus the gap will be closed at approximately double pressure, even if a thickness of 1 mm is used.

Possibility of 190/32 Hollow Bar for oil cylinder (see p.14)

\[
\frac{\Delta r}{r} = 2 \cdot 10^{-3} \left( \frac{3.07}{1.07} + 0.27 \right) = \frac{1}{2} \times 3.13 \times 10^{-3}
\]

\( \Delta r = 0.21 \text{ mm} \)

so at seal, the radial stretch will be \( \approx 0.1 \text{ mm} \), compared with Poisson expansion of piston of \( \approx 0.02 \text{ mm} \), net increase in radial clearance \( \approx 0.08 \text{ mm} \). This could probably be lived with just as well as 0.06 mm with a wire ring.

The hoop tension at 100 MPa oil pressure (15% gas pressure) would be

\[
100 \cdot \frac{3.07}{1.07} = 287 \text{ MPa}
\]

compared with a yield stress of 850 - 1000 MPa, a factor of \( \approx 3 \) or more. At 700 MPa gas pressure, the factor is 4. Should be OK. Would save \$200 - 300 (t) compared with using a solid piece of 705 bar (4340) & drilling.

If we finished at 132 mm, the high pressure cylinder piston would need to be 41.7 mm

133 \( \rightarrow \) 42.06

so maybe aim at 133, 42 ???

At the root of the attachment thread of 160 mm major dia.

the stress will be \( \sqrt{(190^2 - 160^2)} \) times the oil pressure = \( \frac{1}{1.66} \) times oil pressure or max of 166 MPa — OK, although some effect from the internal pressure will add some financial effect.
Expansion of cylinder at 700 kPa = 0.0189 — say 0.010

Poisson expansion of piston at 700 kPa = 0.0387

Difference = 0.029 mm, so 0.030 should suffice, just

\[
\begin{align*}
\text{cylinder} & \quad 41.080/41.025 \\
\text{piston} & \quad 40.960/40.945 \\
& \quad \text{or } 40.970/40.955
\end{align*}
\]
Clearances on high pressure piston seal:

Expansion of the inner cylinder is

\[
\Delta d = \frac{2 \times 10^{-3}}{d} \left( \frac{160}{42} \right) + 0.27
\]

\[
= 0.71 \times 10^{-3}
\]

so at \( d = 42 \) mm, \( \Delta d = 0.0030 \) mm.

The actual expansion around the seal area will be of the order of half this, i.e. \( \Delta d \approx 0.0015 \) mm.

Piston expansion of piston is

\[
\frac{1000}{200000} \times 0.27 \times 42 = 0.037 \text{ mm}
\]

As we need a minimum clearance of about 0.030 mm, we should drop the min. clearance by \((0.030 - 0.009)/2 = 0.010 \) mm in order to ensure the min. clearance. Thus we arrive at the following fits for 42 mm piston:

- D1: Piston 41.970 - 41.985
  - Bore 42.025 - 42.000

- D2: "Piston" 48.485 - 48.470
  - Bore 48.525 - 48.500

The max. clearance combined for the two surfaces is thus 0.110 mm and the min. combined clearance is 0.030 mm.

For 41 mm piston, we have:

- D1: Piston 40.970 - 40.985
  - Bore 41.025 - 41.000

- D2: Ringgroove 48.485 - 48.470
  - Bore 48.525 - 48.500

OD for 223 ring-groove is 55.45 if ring thickness is 1 mm, giving a squeeze of 22% (which allows for some re-polishing). OD for 225 ring-groove is 54.0 with similar squeeze.
For seal nut, area to be supported is 48.5 OD 41 ID which at 16 Pa gives a force of 0.527 MN. For 80 MPa on 56 x 5.5 ≈ 46 mm dia. we would need length:

\[
\frac{0.527}{\pi \times 0.046}\text{.80} = 45\text{ mm}
\]

For 64 x 6 ≈ 54 mm dia, length = 39 mm say 50 mm with clearance. For in Rc 42 material (yield ≈ 1100 MPa) we could easily allow 100 MPa shear stress in the threads, leading to length of 64 x 6 threads of 31 mm, say 40-45 mm with clearance.

For high pressure cylinder for internal use: If we use 160 od. 41 id.,

\[
\frac{D}{d} = 3.90 \quad \text{and} \quad \frac{(D/d)^2 + 1}{(D/d)^2 - 1} = 1.141
\]

so at 16 Pa the circum tension on i.d. = 1141 MPa, just above the yield stress of ~1100 MPa for 42 Re.

Radius of plastic front (lab book 9 p10) is given by

\[
p = \frac{5}{\sqrt{3}} \left[ 2 \ln \frac{a}{d} + 1 - \left( \frac{d}{a} \right)^2 \right] \text{ for plane strain} \quad a = \frac{1}{2}d
\]

or diameter of plastic front is given by

\[
p = \frac{5}{\sqrt{3}} \left[ 2 \ln \frac{5}{d} + 1 - \left( \frac{5}{d} \right)^2 \right]
\]

With σ₀ = 1100, d = 41, D = 160 we have:

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<tr>
<th>σ (MPa)</th>
<th>P (MPa)</th>
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<td>42</td>
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</table>

(2000 MPa for full plasticity if D = 200)
As the plastic front advances, the tangential stress rises at the outer surface according to

$$
\sigma_t = \frac{25}{13} \left( \frac{S}{D} \right)^2
$$

At \( S = 41 \text{ mm} \),

<table>
<thead>
<tr>
<th>( \sigma ) (MPa)</th>
<th>( D = 160 \text{ mm} )</th>
<th>( D = 200 \text{ mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>700</td>
<td>45</td>
<td>53</td>
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<td>160</td>
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<td>200</td>
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</table>

\( S = D \)

1270 MPa

Increase in diameter elastically is

$$
\Delta d = \text{as calculated} \quad \Delta d = 0.06 \text{ mm}
$$

In plastic case,

$$
\Delta d = \frac{\sigma_y}{2\sqrt{3} \cdot 80 \cdot 10^3} \cdot \frac{S^2}{d}
$$

\( \sigma_y = 80 \text{ GPa} \)

\( S = 41 \text{ mm} \)

\( \Delta d = 0.16 \text{ mm} \)

<table>
<thead>
<tr>
<th>( \sigma ) (MPa)</th>
<th>( \Delta d ) (mm)</th>
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<tbody>
<tr>
<td>45</td>
<td>0.20</td>
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<tr>
<td>50</td>
<td>0.24</td>
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</table>

ie permanent stretch \( \approx 0.14 \text{ mm} \)
Example in Machinery Design gives 0.001 Ad/d for interference & 0.004 for the clearance, i.e. 160µm interference and 640µm clearance on 160 mm, so total 0.8 mm, requiring \( \Delta T = 417 K \) i.e. \( T \approx 460^\circ C \).

Rule of thumb: 6 mm per mm interference corresponds to \( \sim 200 \) MPa stress if shell is thin compared to inner? — maybe when inner distortion is also considered, the interference stress comes down to about 100 MPa.

So use around 150^\circ C preheat of the cylinder to shrink it.
Shrink fit of protective sleeve on intensifier

Suppose we aim at 160 id, 200 od.

Slightest clearance for easy assembly H11/f11 listed is
+250/0: -210/-460. It a clearance of minimum
0.21 mm and maximum 0.71 mm

A loose fit is a clearance of 0.145 to 0.405 mm
0.083 to 0.146

For a differential thermal expansion at 100°C we have
160 x 12 x 10^-6 x 100 = 0.19 mm, almost achievable with dry ice, but the mass of dry ice needed would be of the same order as the mass of steel to be cooled.

(Cp for steel ~ 0.46 J kg^-1 K^-1 i.e. 46 J/kg per 100 K, while latent heat of CO2 ~ 300-400 J/g), so would need around 10.1 to 0.2 kg dry ice per kg steel if the heat transfer were 100% efficient.

In a thin-walled cylinder, the stress after shrinking is E (Thermal expansion - clearance during fitting) / d.

If on 160 diam, we have 0.19 thermal expansion and aim at 0.10 mm clearance during fitting, then
Circum. stress = 200000 x 0.09 = 112 MPa, well within the elastic range, yet quite a firm fit.

Desired final fit: Minimum requirement would be 0.18-0.28 in order not to slip off, and to be effective as constraint. Maximum stress desirable is probably around 100 MPa. Ratio: 4 strain of 0.0005 or diametral interference of 0.0005. 160 = 0.080 mm. Thus: 160.000 to 160.040

This would require minimum least of 0.180 mm
160 x 12 x 10^-6 = 9.4 K for a clearance of 0.1 mm during assembly (an easy "loose" fit is min clearance 0.084 mm, max 0.285 mm; a "loose" fit has min clearance 0.145 mm).
Integrating cylinder - outlet end:

We need a such that

\[
p \times \frac{\pi (y/2)^2}{\pi/4} = \text{say 100MPa, about one-tenth of stress, i.e. one-fifth of shear strength}
\]

\[
l = \frac{100 \times 41}{4 \times 110} = 93 \text{mm}
\]

100 mm would be plenty

50 mm would do

High pressure fitting: Is the Nova Swiss fitting for 3/8” tubing OK for 1.4 GPa?

Strength of thread at 80 MPa shear stress = \( \pi (0.020 \times 0.0175) \cdot 80 \)

= 0.088 MN

Strength of shoulders at 200 MPa bearing stress = \( \pi (0.0132 - 0.0098)^2 \cdot 200 \)

= 0.0156 MN

Force on tubing assuming seal at \( \phi = 3.5 \) = \( \frac{\pi}{4} (0.0035)^2 \cdot 1000 \)

= 0.0096 MN

at \( \phi = 7.0 \) = 0.0385

\( \phi = 4.5 \) = 0.0159 MN

So the fitting should be OK although not nearly as conservative as the Harwood type fitting & doesn’t allow much margin.

Is there any point in such a fitting rather than the normal \( 1/4 \)” fitting? It might be useful for proving to 800 MPa (116,000 psi, say 120,000 psi), for which the \( 1/4 \)” fitting is pretty marginal (given as proof pressure by Harwood)

For the present, may be best to stick to 700 MPa limit &
to use standard ¼" for simplicity and to avoid temptations to indulge in explorations up the pressure scale. Raising the pressure scale might be better done more deliberately by starting with harder pressure vessels (and thicker sleeves) and special flanging.

The above dimensions would appear still acceptable, especially for 700 MPa pressure range.

Thread in met:

- Force on piston at 16 MPa = 0.707 MN
- Shear stress 16 MPa = 0.70 MPa
- Stress in steel, 16 MPa x 0.5 = 8.6 MPa

Actually 16 is not a recommended usage, but 42 MPa, with 0.5 x 3 pitch.

Also we need to address for the met to be not screwed right in, in case of extra long access. If screwed back 20 mm, 16 thread = 125 MPa, 40 -> 135 MPa, still OK.
For M160x6 thread, pitch dia = 156, length = 50, area = 0.0245 m²

For 4.66 MN, \( \tau = 190 \text{ MPa} \)

At thread roots in cycl, \( \phi = 160 \), length 50, area = 0.0251

\( \tau \) for 4.66 MN, \( T = 185 \text{ MPa at } 1 \text{ GPa} \)

\( 130 \text{ MPa at } 0.7 \text{ GPa} \)

FS = 2.4

FS = 4.9

This calculation neglects the axial load of ~0.1 to 0.15 MN, but this is not serious.

Assumes ~1100 MPa y stress = 40 Re or 370 HRC

If steel comes down to ~34 HRC (320 HB), y stress ~850 MPa

\( \tau = 185 \text{ MPa}, \text{ comp to } \sigma = 185.13 = 320 \)

or \( \sigma = 2.7 \times 5 \) \( \text{FS} = 2.7 \)

\( \tau = 130 \text{ MPa}, \text{ comp to } \sigma = 130.13 = 245 \)

FS = 3.8
Review of Main Pressure Vessel Design

The prototype pressure vessel has a larger and shorter closure nut than would otherwise have been chosen because of the length constraint of the existing piece of steel. Is this optimum, as well as convenient for access and for having the same diameter thread as the intensified cylinder (M160)?

\[ \text{Force on } 77\bar{A} = 4.66 \text{ MN at 16 Pa} \]
\[ \text{Shear stress on threads } = 185 \text{ MPa at 14 Pa} \]
\[ \text{Stress in root of thread in cylinder } = 217 \text{ MPa at 16 Pa} \]
\[ \text{or 130 MPa at 0.74 Pa} \]
\[ \text{Stress in root of thread in cylinder } = 152 \text{ MPa at 0.74 Pa} \]

Comparing figures in No1/2 pressure vessel:

\[ \text{Force on } 73\bar{A} = 4.19 \text{ MN at 16 Pa} \]
\[ \text{Shear stress on threads } = 187 \text{ MPa at 14 Pa} \]
\[ \text{Stress in root of thread in cylinder } = 127 \text{ MPa at 14 Pa} \]

ie closely comparable in stress on the threads & more conservative in the root thread in cylinder.

The above dimensions would appear still acceptable, especially for 700 MPa pressure range.

\[ \text{Thread in nut:} \]
\[ \text{Force on piston at 16 Pa } = 0.707 \text{ MN} \]
\[ \text{So shear stress on } 40 \times 65 \text{ thread } = 86.5 \text{ MPa} \]

Actually 40 is not a recommended size but 42 is, with 4.5 as 3 pitch.

Also we need to allow for the nut to be not screwed right in, in case of extra long assembly. If screwed back 20mm, 45 thread \[ \text{Shear stress } 125 \text{ MPa} \]
\[ \text{40 } \Rightarrow 134 \text{ MPa} \]

- still OK.
Furnace & Top Plug Design

The top plug design does not seem to need modifying except that the furnace & attachment & dust-retaining portion could usefully be extended downwards in place of the upward extension of the top section of the furnace core, in order to reduce the temperature at the O-ring seal a little. It may be desirable to allow for the furnace windings to be lowered 10 mm also to help this factor. This does not necessarily mean lengthening the insulation but it wouldn’t hurt to allow for it, say 27 mm to 185 mm.

Below the furnace, the steel parts should again be brought up as close to the furnace as possible.

\[ \text{(Equation or diagram content)} \]
### Relative Thermal Resistances $x$:

<table>
<thead>
<tr>
<th>Material</th>
<th>$x$</th>
<th>$\Delta x$</th>
<th>$\frac{\Delta T}{\Delta x}$</th>
<th>Resistance $R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>12</td>
<td>0.36</td>
<td></td>
<td>0.36%</td>
</tr>
<tr>
<td>ZrO₂</td>
<td>45</td>
<td>22.5</td>
<td></td>
<td>71%</td>
</tr>
<tr>
<td>Al₂O₃</td>
<td>45</td>
<td>9</td>
<td></td>
<td>28%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>31.86</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Heat flow up Al₂O₃ core of 150 length above upper winding:

$$Q = xA \frac{dT}{dx} = 9 \cdot 4\pi \left(\frac{0.0312^2}{0.021}\right) \frac{1200}{0.05} = 49$$

So reducing the section to half would save ~0.02 W/K.

Maybe should put $k = 15$ rather than $9$, giving ~82 W/K or 0.06 W/K. This is a significant effect.

**Important:**

- Probably better to stick to PS2 for the upper core rather than use a longer Al₂O₃ core.

However, if the OD of the core tube is reduced from 27 to 24, reducing the cross-section to half, then the heat loss is reduced to one half of the original, which is probably still not important.
Heat flux in pistons

Lower piston: Max gradient 22.6 K mm⁻¹ in all ZrO₂ at 1000°C
   (run 6013) suggesting 31 K mm⁻¹ at 1400°C
   max grad 12 K mm⁻¹ in Al₂O₃ at 700°C
   suggesting 24 K mm⁻¹ at 1400°C

 Flux = \( x_i A_i \frac{\Delta T}{\Delta x} = -k \frac{4}{\pi} \frac{\Delta T}{\Delta x} \)

For ZrO₂ \( k = 9 \text{ W m}^{-1} \text{ K}^{-1} \), so Flux = \( 9 \times (0.015)^3 \frac{24}{31} = 38 \text{ W} \)

For \( x = 2 \) Flux = \( 2 \times (0.015)^3 = 11 \text{ W} \)

In the jacket, \( x = 40 \)

 Flux = \( 40 \times (0.015)^3 (0.015)^3 \frac{24}{31} = 11.5 \text{ W} \)

So total heat flux = 50 W for Al₂O₃
   or 26 W for ZrO₂
   at 1400°C

This is presumably mainly conductive transfer.
50 W along the steel piston at the top (\( x = 40 \), say) gives a large difference along 20 mm of piston of 141 K (7 K mm⁻¹). The observed gradient is much greater than this, indicating other forms of heat transfer at the ends, maybe radiation (??).

Upper piston: Observed max grad (6013) = 16 K mm⁻¹ in all ZrO₂ at 1000°C
   \( \rightarrow 22 \text{ K mm}^{-1} \text{ at } 1400°C \)
   max grad (6021) = 10 K mm⁻¹ in Al₂O₃ at 700°C
   \( \rightarrow 20 \text{ K mm}^{-1} \text{ at } 1400°C \)

So for Al₂O₃, heat flux = 32 W + 18 W mon = 32 W total
   for ZrO₂ = 11 W + 11 W = 22 W

Then lower gradients presumably a set by an overlay of convective transport tending to swamp the conductive by a substantial factor, (\( \sim \) 2??).

Upper stainless steel tube: Say \( \Delta T/\Delta x \sim 10 \text{ K mm}^{-1} = 10^4 \text{ K m}^{-1} \)
   \( x = 20 \text{ m} \)

 Flux = \( 20 \times (0.025)^3 (0.021)^3 \frac{10^4}{31} = 37 \text{ W} \)

so could be a useful saving in going to zirconia with \( x = 2 \).
Radial heat flow in heat sink under furnace.

In an infinite cylinder, radial flux per unit length is

\[
\frac{2\pi x \Delta T}{\ln D/d}
\]

For heat flux, \[
= \frac{2\pi x \Delta T}{\ln D/d}
\]

For mild steel, \( x = 50 \), \( D = 65 \), \( d = 16 \)

\[\text{so flux} = 224 \Delta T \text{ l}.\]

For \( l = 10 \text{ mm} \), \( \Delta T = 100 \text{ K} \), \[\text{flux} = 224 \text{ W compared with} 25 - 50 \text{ coming down the piston. So if } l = 10 \text{ mm and all the heat flux of say 50 W were extracted this way, the} \Delta T \text{ would be 22 K, quite small.}\]

The main impediments to heat removal in this way would be the \( \Delta T \) between piston & heat sink, and heat sink & bomb wall, which could easily be 50 K each. Nevertheless, such a heat sink is probably useful.

It could occupy space which could be made available for other transducers in the future in relation to strain measurement, e.g. capacitor plates or the mounting plates for special LVDT's to be located in holes in the end cover block. Two plates would be required in the latter case, separated by the amount of displacement to be measured, say 10 mm + 2 x 2 mm plates = 14 mm.

For capacitor plates, one could possibly get away with two, but response would be non-linear; better to have three, then 10 mm displ requires \( 2 \times 10 + 3 \times 2 = 26 \text{ mm} \) and 5 mm displacement 16 mm.

So we could allow say 15 mm.

Temp drop in argon layer: \( \Delta T \) for argon at atmospheric pressure is about \( 1.9 \times 10^{-5} \) at atmospheric pressure and about \( 0.24 \times 10^{-5} \) at 700 mmHg at a temperature of around 50°C (323 K) (Zeman, 1972)

(19.79 AER 1979)
Thus \( \Delta T = \frac{Q}{T \cdot \ln \frac{D_A}{D}} \). 

For \( Q = 100 = 10^4 \) and \( K = 0.24 \) we get \( \Delta T = 6631 \cdot \ln \frac{D_A}{D} \).

For \( D = 65 \), \( d = 64.9 \), \( \ln \frac{D_A}{D} = 0.0015 \) we get \( \Delta T = 10 \) K. This increases approximately with the gap. 

For \( D = 16 \), \( d = 15.5 \), \( \ln \frac{D_A}{D} = 0.032 \), \( \Delta T = 210 \) K. So the temp. diff. in still argon between piston & heat sink could be considerable. 

E.g. \( D = 16 \), \( d = 15.7 \) gives \( \Delta T = 125 \) K. Both these figures are excessive because the total heat flux down the piston cannot be expected to exceed 25-50 W, i.e. the above \( \Delta T \) are in excess by at least 2-4 x, leading to 50 to 100 K of \( \Delta T \) with \( d = 15.5 \), or 30-60 K with \( d = 15.7 \).

\( D = 15.7 \), \( d = 15.5 \), \( \ln \frac{D_A}{D} = 0 \). Leads to \( \Delta T = 85 \) K.

The effects at 300 kPa are 1.66 times greater & at 100 kPa 3.3 times greater. So minimizing the argon gaps is much more important than selecting special materials for heat sinks.

Heat flows through the \( \text{Al}_2\text{O}_3 \) core beneath the lower winding:

\( \Delta T = \frac{Q}{T \cdot \ln \frac{D_A}{D}} \)

Suppose \( Q = 400 \) W (an upper limit, perhaps exaggerated)

\( N = 0.025 \)

\( K = 5 \) say for hot \( \text{Al}_2\text{O}_3 \)

\( D = 27 \), \( d = 21 \) mm

Then \( \Delta T = 125 \) K

We might expect ~100 K temp. difference across the \( \text{Al}_2\text{O}_3 \), not too much to worry about.
Heat flow at top end of pressure vessel

Suppose we flow the heat up the 30 mm piston for 55 mm. Then radially out over 80 mm length.

Suppose this heat flux to be 300 W

\[ \frac{\text{heat flux}}{\text{area}} = \frac{300}{30 \times \pi(0.065)^2} = 45 \text{ K} \]

Then \( \Delta T \) in piston is \( \frac{300 \ln \frac{270}{30}}{0.080 \ln 2.1} \) = 707 K

The figure of \( \Delta T \) in the piston is obviously artificial. If the heat conducted up is 50 W, \( \Delta T \) becomes 118 K, still quite appreciable, while if we now consider only 15 mm distance it is 32 K.

If one conducted 300 W up the plug of 65 \( \phi \) for a length of 40 mm,

\[ \Delta T = \frac{300 \times 0.040}{33 \times \pi(0.065)^2} = 110 \text{ K} \]

or halfway up and run out would involve 55 K + 45 K = 100 K.

These calculations, although crude, indicate that if there is 300 W of heat conducted & converted up to about the face of the closure, the temp flux could fairly readily get up to 100°C or so at the face & possibly a bit higher. Thus convective loss from the furnace is of paramount importance.

A value of around 100°C at the face of the top plug seems more or less consistent with what the temperature profiles seem to extrapolate to.
Bottom Plug design

Force on section A in plug will be a bit less than on section B, so section B is thick enough if we extend the 77 mm diameter section down a bit compared with previously. Bring it down far enough so that bleed hole can go straight in. Section C strong enough.

If length in 16 + 6 = 22, it will be about the same as for the top plug.

Shear stress at section A:
\[
\tau = \frac{p}{\pi \cdot 30 \cdot 22}
\]

\[
= 0.19p \quad (\text{at 1 GPa, 136 MPa at 0.7 GPa})
\]

Should be OK.

Compensating piston: Force in over whole 420 area the cause hole is closed at top. So shear stress at section D is

\[
\tau = \frac{p}{\pi \cdot 30 \cdot 22}
\]

With \( l = 40 \), shear stress = 187 MPa at 1 GPa
or 131 MPa at 0.7 GPa

Should be OK.

With \( l = 50 \), which is probably needed for adequate bearing area (6 mm length) in lowest position. On further thought, seem to need 50.

With \( l = 50 \), shear stress = 150 MPa at 1 GPa
or 105 MPa at 0.7 GPa

More comment, but still OK at 50 when shear stress = 166 MPa at 1 GPa
or 116 MPa at 0.7 GPa.
For double wall,
\[ \Delta T = \frac{Q}{L} \left( \frac{1}{\ln D_1} + \frac{1}{\ln D_2} \right) \]

If we take
\[ \Delta T = 900, \quad k_{\text{outer}} = 2, \quad k_{\text{inner}} = 0.8 \]

Then
\[ \frac{Q}{L} = \frac{2 \pi \times 900}{\ln 59/40 + \ln 40/27} = \frac{8250}{2} = 4125 \]

ie \( Q = 825 \)

so total power consumption = 975 on assumption of partition opposite or efficiency = 0.75 \( \text{W/K} \), an increase of around 60\% or about 2\% more current to say 16A.

Temp drop across the first layer is
\[ \Delta T = \frac{8250 \ln 40/27}{2 \pi 0.8} = 645, \text{say 650} \]

\[ T \text{ at } \phi = 40 \text{ would be } 1300 - 650 = 650 \text{K.} \]
Radial heat flow in furnace
At 0.7 W/K, power at 1300 K is 700 W. Suppose that
this amount or at least 600 W if it travels out radially
along 100 mm of winding, i.e. \( W = 6000 \)

Then \( \Delta T = \frac{W}{l} \cdot \frac{\ln(D)}{2\pi} \)

\[ \Delta T = \frac{6000}{2\pi} \frac{\ln(59/27)}{K} \]

\[ \Delta T = \frac{746}{K} \]

Thus if \( \Delta T = 900 K \), \( \Delta T = 800 K \)

So the effective thermal conductivity of the insulation is
presumably a bit under 1 compared with 2 of zirconia, and with 0.14 of argon at 300 KPa. So
there must be convective loss as well as conductive,
unless the solid parts of the insulation are conducting
quite a lot (X for insulite \( \approx 4 \) W/mK)

17/7/92

If we take 0.5 W/K, power at 1300 K = 650 W. Suppose \( \frac{3}{4} \) of this flows
radially from 100 mm of winding, i.e. \( \frac{500}{W} \). \( W = 5000 \)

\( \Delta T = \frac{5000}{2\pi} \frac{\ln(59/27)}{K} = 622 \)

Actual \( \Delta T \) is probably at least 700 K because winding will run
bitter than 1300 K and temp at \( \phi = 59 \) probably not more than 400K
\( \Delta T \). Thus \( \frac{622}{900} = \frac{K}{\Delta T} \approx 0.7 \) W/mK

But see p61: this seems too high: 0.4

Thus if we change to \( K = 2 \), the 500 W \( \rightarrow 500 \cdot \frac{2}{0.7} = 1430 W \), making
\( \frac{1600 W}{K} \) total or around 1.2 W/K power consumption with a
solid zirconia insulation.
Safety & cooling sleeve

This sleeve is to fit over the whole length of the pressure vessel except where the support overlaps at the bottom, i.e. over a length of about 620 mm. The i.d. is determined by pressure vessel o.d. i.e. 230 mm.

For a shrink fit, Hall, Holowinks & Haugklin "Machine design" 1980 p23 designate an H7-56 fit, i.e. equivalent to a heavy press fit (interference on 230 mm = 0.004 mm min. or 0.019 mm max.), i.e. interference strain of 0.0004 to 0.0007.

They also mention a diametral clearance during assembly of 0.05 mm for 100 mm dia (p24) and 0.045 mm for 25 mm diameter (p25), so we could conservatively select 0.1 mm assembly clearance for 230 mm diameter i.e. clearance strain of 0.00043, giving a total relative thermal expansion required of 0.0008 to 0.0016, or with \( x = 12 \times 10^{-6} \text{ K}^{-1} \), \( \Delta T = 70 \text{ to } 100 \text{ K} \).

However, this does not take into account the heat transfer during assembly. In fact, the sleeve will lose its heat to the pressure vessel on a scale of \( x = \sqrt{DE} \)

where \( x \) can be taken as roughly equal to the thickness of the sleeve (heat from half-thickness of sleeve into similar thickness of metal of blank), and

\[ D = \frac{2}{\rho c} \]  

the thermal diffusivity. The thermal conductivity can be taken an average of mild steel (\( \approx 50 \)) and H13 (\( \approx 25 \text{ Wm}^{-1}\text{K}^{-1} \)), i.e. 37 Wm^{-1}K^{-1} and \( c \) (heat capacity) = 460 Jkg^{-1}K^{-1}

\[ D = \frac{37}{7800 \times 460} = 1.03 \times 10^{-5} \text{ m}^2\text{ s}^{-1} \]

is enough to \( 10^{-5} \text{ m}^2\text{ s}^{-1} \)

\[ x^2 = DT = 10^{-5} \text{ s} \]  

so for \( x = 0.02 \text{ (20 mm)} \), \( t = 40 \text{ s} \) or \( x = 0.01 \text{ (10 mm)} \), \( t = 10 \text{ seconds} \). So the piece by crane may well take more than 10 seconds, so the heat transfer will be important at the leading edge.
If \( t = 205 \), \( x = 14 \) mm, so the temperature will drop by something of the order of a factor of 2 through a substantial part of the thickness. Thus we need extra heating to at least twice the calculation viz. \( T_1 \approx 140 - 200 \) K ie to \( 160 - 220 \)°C. This is for an interference of 0.09 to 0.17 mm.

The minimum interference is perhaps a heavy push fit, 1/7 - 1/6, for which the interference range is \(-0.015 \) mm to \( 0.060 \) mm ie \(-0.0005 \) to 0.00026 interference strain. If we take 0.0001 interference strain as the maximum acceptable, this amounts to an interference of 0.023 mm actual interference. Then an assembly clearance of 0.1 mm, total expansion of 0.012 - 0.123 mm (0.00053 strain), would require a min \( \Delta T = 45 \) K, or say 120° allowing for the heat transfer during about 20 seconds.

If we choose selective fit and allowed between 0.02 mm min interference and 0.07 mm interference (0.0003, interference strain), then the latter limit would require a \( \Delta T = 61 \) K, or say 150° for 20 second fit. One might specify an interference of 0.050 ± 0.025 mm.

Wall thickness. If we use Assab grade 2C heavy wall hollow bar (0.2 C, 0.65 Mn steel, elongation \( \approx 28\% \) according to Smithells, p.561), then there is a choice of o.d. of 273 mm and 298 mm, i.e. wall thickness of 21 or 34 mm (one could even go to 324 o.d., wt. 47). The main requirement is to hold pressure vessel fragments together while the gas escapes after a brittle failure. By the time a crack opened up a few tenths mm and would expect the gas pressure to dissipate. Take 1 mm as a conservative upper limit, so the safety sleeve has to stretch 1 mm. If this is limited to ±30° of the failure site due to friction (\( \tan \phi = \tan 30° = 0.6 \) ), then gauge length \( L = \frac{110}{0.230} \) and so
Strain = \[ \frac{21.6}{11.230} = 0.0085 \text{, it is less than } 1\% \text{. Even allowing for shock loading, although it is not quite clear how to do this (will require going into the dynamics of the gas escape?), it seems intuitively that the mild steel will 25\% of elongation ought to be able to cope and that the thickness of 21 mm wall ought to be enough.}

The cooling can be with 12.7 mm copper tubing, for which a groove of say 13 mm diameter \& 6.5 mm radius would be suitable (leave 6 mm w 0.15 mm radial clearance for heat conducting glue & fitting) — could be a little closer, say 6.4 to 6.5 R. Could sink the grooves 7 mm deep to give max contact. A pitch of 25 mm is probably OK (alternatives probably available on RS. Would be 20 and 30 mm)

Increase in O.D. with pressure in elastic case (Ref Blk 7 p 20) is given by:

\[ \frac{AD}{d^2} = \frac{2P}{E(d^2 - 1)} \]

\[ d = \text{O.D.} \]
\[ d = \text{i.d.} \]

\[ \frac{2.700}{(230)^2 - 1} \cdot \frac{1}{200000} = 0.0000576 \text{ (8 = 121 MPa)} \]

\[ AD = 0.140 \text{mm} \]

So conclude: cylinder O.D. = 230.000 / 229.970
jacket I.D. = 229.930 / 229.900

Heat to min 175-200°C before assembly.

compared with extensions in others
**Bearing Stress of Actuating Piston on Stirrup**

Bearing over \(28\frac{1}{2}\) \(\text{o.d.}, 10\text{i.d.}\)

\[
\text{area} = 638\text{mm}^2 - 78.5\text{mm}^2 = 559\text{mm}^2
\]

Divided by two because of push-pull requirement \(\rightarrow 280\text{mm}^2\)

Which, for 100 KN max. load gives a bearing stress of 358 MPa. Just tolerable on \(E=25\) with proof stress of 1800 MPa.

\(280\text{mm}^2\) in the threaded part corresponds to a pitch diam of thread of 21.4 mm

A 24\# threaded has pitch diam of 22.05

\[24 \times 3.0\]

Fine thread: \(M22 \times 1.5\) has pitch \(P = 21.03\) \(\rightarrow 269\text{mm}^2 \rightarrow 372\text{MPa}\)

Minor \(P = 20.4\) \(\rightarrow 248\text{mm}^2 \rightarrow 403\text{MPa}\) stress.

\(28.5\text{mm} 22\text{id} \rightarrow 258\text{mm}^2 \rightarrow 388\text{MPa}\)

**Bending of Top of Stirrup (approx.)**

Deflection \(s = \frac{FL^3}{384EI}\)

\[
\frac{Wl^3}{192EI} \sim \frac{WE^3}{48EI}
\]

designed and freely supported

So take \(s = \frac{Wl^3}{100EI}\)

\[
\frac{10^5 \cdot 0.13}{100 \cdot 200 \cdot 10^6 \cdot 0.13} \approx 1.014 \cdot 10^{-9}
\]

\[
\frac{h^3}{h^3} = \frac{31.6 \times 10^{-3}}{6} \text{ for } h = 0.030\text{m}
\]

\(0.028 \text{mm} \text{ for } h = 30\text{mm}\)

\(0.016 \text{mm} \Rightarrow \frac{T(10^{-5})}{40} \Rightarrow 0.0081\text{mm} \Rightarrow 0.151\text{mm}\)

Compared with extension in sides, \(s_1 = \frac{Wl^5}{2 \cdot 0.13 \cdot 0.02 \cdot 200 \cdot 10^6} \Rightarrow 0.0087\text{mm}\)

and in the piston: \(s_1 = \frac{10^{-5}}{l(0.03)^2 \cdot 200 \cdot 10^6 \cdot 0.2} = 0.142\text{mm}\)
Thus assumed 30 to 35 mm seems stiff enough, say 35.

Length of thread on piston:

M22 × 1½ thread has pitch \( \phi = 21 \)

Taking shear stress in threads as \( 100 \text{ MPa} \), then length of thread needed is

\[
L = \frac{10^5}{\pi (0.021) \cdot 100 \cdot 10^5} = 0.015 \text{ m} = 15 \text{ mm}
\]

say 18.

Structure for fixing head to plate in housing:

The fixing is either to 12 × 12 strip stacked in fibre or to 22 × 22 strip attached to the inner angle. Original planned to have M18 screws on 220 spacing. However, it would cross bolts. So use M8 screws from tang and space them at 150 or frame holes at 150.
Eyebolts for Lifting Housing

First, standard 2317-1984 eyebolt. A suitable one is that of M22 shank, rated for 1.6 tonnes single lift, 0.8 tonne 90° string, and 0.4 tonne side lift. Fit four of them, one at each corner, so that if two are used for a side lift, this should still be adequate.

Screws for Fixing Liner Plates in Housing

The fixing is either to 12x12 strip attached to the 8mm wall, or to 20x20 strip attached to the corner angles. Originally planned to have M8 screws on 200 spacing. However, it would seem better to use M6 screws, 16mm long, and space them at 150 or even better at 120.
Lifting tackle for intension: simplest to use a loder hoist, of which Blackwood catalogue lists several for 0.75 tonne capacity, with min. distance between hooks of 25.3 to 380 mm. May be able to reduce this a little by substituting a bolt for the upper hook for fixing.

I don't need much headroom above the bracket, say 50 - 100 mm.

Dimensions of bent boiler plate: Calculated on centre line of plate.

\[ 575\ -\ (36 \ +\ 12) = 575 - 48 = 527 \]
\[ 900\ -\ (50\ +\ 36\ +\ 12) = 900 - 98 = 802 \]
\[ \frac{1}{4} (2\pi \times 36) = \pi \times 18 = 56.5 \]
\[ \frac{1385.5}{2} = 2771 \]  
\[ \text{say } 2770 \]

Cutout width: \[ (527 + 56.5) \times 2 = 1167 \]

Cutout height: \[ 360 + 40 = 400 \]

Height = 1500 - 220 = 1280
6. So

\[ \text{Diagram with numbers and calculations.} \]

\[ \text{Mathematical calculations and notes.} \]

\[ \text{Additional notes and diagrams.} \]
Possible Hydraulic Actuator Arrangement

The Instron attachment adaptor could be used as the cylinder of a hydraulic actuator. Its present internal diameter is 65 mm but there would seem to be no reason why it could not be opened up to larger bore—at least to 80 mm. Since the restrictor at the top end could serve to support the linear ball bearing.

If we use 65 and step down to 40 for the piston extension, then pressure required for $100 \text{ kN}$ force is given by

$$\frac{\pi}{4} \cdot p \cdot (0.065^2 - 0.040^2) = 10$$

$$p = 48.5 \text{ MPa (7000 psi)}$$

which is within the range of Portopower type equipment.

If we went up to 80 mm x 50 mm and $200 \text{ kN}$ force,

$$\frac{\pi}{4} \cdot p \cdot (0.080^2 - 0.050^2) = 20$$

$$p = 65.3 \text{ MPa (9500 psi)}$$

which is just beyond ordinary Portopower pressures. At 8000 psi, 170 kN could be reached, which could be quite useful.

Vol. of oil for 30 mm stroke: vol = $\frac{\pi}{4} (0.065^2 - 0.040^2) \times 0.030$

= $6.185 \times 10^{-6} \text{ m}^3$ or 62 cc.

If we used a screw press of 300, then the stroke required would be

$$l = \frac{61.85 \times 10^{-6}}{0.0875} = 0.0875 \text{ m} = 88 \text{ mm}$$

so 100 mm stroke would be plenty. Max force would be

$$\frac{\pi}{4} (0.030^2) \times \frac{(0.030)^2}{(0.065)^2 - (0.040)^2} = 48.5$$

$$= 16.6 \text{ MPa} \times \frac{\pi}{4} (0.030)^2$$

$$= 11.75 \text{ kN} (\approx 2600 \text{kN})$$

This is within the range of the thrust bearing on porous fluid volumeters (300 MPa on 4.78 $\phi$ = 5.4 kN).
If the screw press has a thread of 3.5 mm pitch (coarse; 2 mm is fine) and worm wheel is 100:1, then low of worm advance is given by

\[ 0.035 \text{ mm} = 35 \mu \text{m} \]  
so torque is given by

\[ 2\pi T = 11750 \times 35 \times 10^{-6} \]

or \[ T = 0.0655 \text{ Nm} \] neglecting friction

Efficiency of screw: Wallis steel machine design \( \eta = \frac{0.1 \times \pi/27.78}{0.1} \) gives:

\[ \eta = \frac{0.057}{0.057 + \frac{0.1}{\cos 30^\circ}} \]

Worm has about 2.5 pitch on 14 mean diam

\[ g = \frac{0.057}{0.057 + \frac{0.1}{\cos 30^\circ}} = 0.33 \]

Giving an overall efficiency of 0.085

No motor needs to apply a torque of 0.77 Nm (\( \equiv 109 \text{ oz in} \))

For 300 psia poor pressure on present volumeter, need 0.376 Nm of torque, whereas the G9124 motor gives only 0.177 Nm continuous rating. It can provide 2.9 Nm for 50 ms every 5 sec, ie 1% duty cycle; maybe get away with it if the servo does not operate more than about 1/3 of the time.

Compliance: There could be up to 100 mL of oil in the system. At 50 MPa pressure, the density increases about 3%, ie volume decreases about 3%, ie by 3 mL, corresponding to a displacement of

\[ \frac{3000}{4(65^2 - 40^2)} \]

Main piston shortening is \( 1.46 \text{ mm} \) or about 0.05 mm/100 kN.

\[ \text{Main piston shortening is } \approx 170 \times \frac{\pi}{4(0.030)^2} \times 0.10 \]

\[ = 0.12 \text{ mm per 100 kN or } 0.012 \text{ mm/10 kN} \] With yoke etc
If we use 100/75, pressure required for 100 kN is
\[
\frac{10^5}{\pi (0.1^2 - 0.075^2)} = 29.1 \text{ MPa (4220 psi)}
\]

Vol of stroke of same dia screw press would be increased by a factor of \[
\left(\frac{0.1^2 - 0.08^2}{0.1^2 - 0.075^2}\right)^{-1} = 1.22
\]
considered, there may be two or three times this amount of compliance in the whole system, but still perhaps a factor of up to 5 or more less than the oil compliance. The total compliance for an oil system would correspond to roughly about 0.6% strain per 100 MPa stress drop or a bit more.

Revision

Have chosen 100 & step down to 80, using O-rings 3.41 and 3.38. Then pressure required for 100 kN in

\[ \frac{\pi}{4} (0.1^2 - 0.08^2) \times 10^5 = 35.4 \text{ MPa} \] (5100 psi),

leaving quite a margin for safety (could go to 150 kN or so).

Stroke for \( \phi 30 \) screw press for 30 mm travel would then be

\[ \frac{0.1^2 - 0.08^2}{0.03^2} \times 30 = 120 \text{ mm} \]

and max. force in screw press is

\[ \frac{0.03^2}{0.1^2 - 0.08^2} \times 35.4 \times 10^6 \times \frac{\pi}{4} (0.030)^2 = 6.25 \text{ kN} \] (638 lbf)

which is not much more than for 300 kPa in present screw press. Could go to 150 kN.

Compliance would be a bit more than calculated above.
Length of Pressure Vessel Support

**Instron Actuator**

Max Extension of Instron = 410 above face A (POS = -100)
Min ... = 310 ... (POS = +100)

(both according to Instron drawing and direct measurement)

Which means 540 between main face of actuator attachment and face of pressure vessel.

2 mm margin in stroke.
Position of Pressure Vessel Top

180 mm travel for hermocoupl.
if attachment occupies

level of top of
height of

previous page

bottom row of lining-fixing
screws
The raised pressure is 39.

24.1 MPa on φ30 piston ≈ 20.6 kN force.
Screw Drive for Hydraulic Actuator.

As calculated earlier (3pp), the force requirement for a screw with $\phi 30$ piston to drive the $\phi 100/80$ hydraulic actuator is within the capability of the screw drive for the fluid volume monitor currently in use. This is based on the value of the embedment angle of 1/8" centres (47.625) and 100:1 ratio, and the pitch of the screw is 3 mm. (Pitch on INSTRON = 22 mm)

Thus, the piston advances 0.03 mm per rev. of drive shaft.

A Pintle Motor 912 M 4 motor has rated speed of 3650 rpm with torque of 0.39 Nm.

G 12 M 4 .... speed 2660 rpm, torque 1.0 Nm

(bore 140 mm diameter)

So at 2660 rpm, piston advances 80 mm per minute

3650

110

Thus is a reduction ratio of 4:1 hydraulically between screw press and actuator, so the respective maximum actuator speeds would be 20 and 27 mm per minute

0.33

0.46 mm per second

[Max speed of INSTRON actuator is 350 mm/min.]

1.7 x 10^{-2} s^{-1} strain rate on 20 mm

Minimum speed will depend on transducer & motor resolution. If the motor advances in steps of about 1/200 rev, this corresponds to an actuator advance of 0.03 x 0.005 x $\pi$ = 37.5 mm or strain of 1.9 x 10^{-6} on 20 mm.

Actually, the motion of the motor is probably much smoother than this because of offsetting of brushes & so one may have a resolution of a few nm, or 1 x 10^{-7} strain on 20 mm. INSTRON claim, a resolution of 50 nm from their system or minimum velocity of 0.3 mm s^{-1}, equivalent to a strain rate of 2.1 x 10^{-8} s^{-1} on 20 mm.

INSTRON quoted minimum velocity of 0.000 016 mm/min corresponds to

2.50

= 0.000137 mm/min

= 2 x 10^{-6} mm/s

So a Pintle Motor servo motor on a hydraulic actuator could probably be used down to 10^{-8} s^{-1} strain rate.
Efficiency of Screw Drive & Motor Requirement.


Approximate efficiency of worm gear unit is:

\[ y = \frac{1 - f \tan \alpha}{1 + f \tan \alpha} \]

where \( f \) is friction coeff. and \( \alpha \) is lead angle.

For Albew gears (80, 19° & 100:1 reduction) take \( f = 0.17 \) (grassy oil on phosph. bronze; Marks p.218).

\[ y = \frac{1 - 0.17 \tan 2.4°}{1 + 0.17 \tan 2.4°} = 0.196 \]

Efficiency of a screw unit (bid p.146) is:

\[ y = \frac{\tan \alpha}{\tan \alpha + \frac{f}{\cos \alpha} + \frac{f_c R_c}{T_m}} \]

Taking \( \alpha = 30° \) so \( T_m = 0.866 \), \( \tan \alpha = 0.517 \), \( f_c R_c \approx f \), \( \alpha = 18° \)

\[ y = \frac{0.0317}{0.0317 + 0.17/0.866 + 0.17} \]

\[ = 0.571 \]

The overall efficiency is \( 0.571 \times 0.196 = 0.11 \).

The coefficient of friction taken may be a bit high, but there may be other aspects (friction in non-rotating key, etc.) that are neglected. It is probably safer to take overall efficiency as 10%, i.e., need 10x as much torque as for frictionless calculation.

With a maximum actual load of 180 kN, the screw press load is 25 kN, so work per pitch = \( 25.10 \times \frac{0.023}{0.005} \) = 0.75 \( \mu \) torr.

\[ = 77 T \text{ in Nm per turn} \]

\[ = 212 T \text{ in Nm per revolution} \]

So max torque (frictionless) = \( 0.75 \times 211 = 0.12 \text{ Nm} \)

So with friction, need 1.2 Nm. Painted motor Motor G 12M44t produces 1.00 Nm at rated speed = 83 kN. At lower speeds, it could presumably do a bit better (see Torque vs speed curve).
Overpressure Protection for Pressure Vessel

The intensifiers can be protected by a relief valve in the oil line. This would also protect the pressure vessel if there is no isolate valve between intensifier valve and pressure vessel. Perhaps it would be best to omit the isolate valve if there is enough capacity in the intensifier to make re-shocking necessary. The isolate valve becomes necessary when the intensifiers is to be re-shocked (the intensifier pressure gauge becomes necessary then too). So intensifier re-shocking requires:

1. Isolate valve
2. Intensifier pressure gauge (oil pressure gauge might do)
3. Overpressure protection for main pressure vessel

Preferably in the form of a rupture disc. Only Harwood offer rupture discs above 700 MPa. Min. margin would be 25%, or 875, say 900 MPa (125,000 psi) - which means a C-4257 Safety Assembly.

Calculation on next two pages indicates that this over-pressure may not do irreparable damage to the pressure vessel and end closures.
With data:

\[ D = 265 \text{ mph} \]

587
615
633
651
666
685

702
718
746
811
867
924
975
1000
1023

Residual sum of squares: 22.0 M"
High Pressure Vessel - Plastic Yielding Calculations

As for the intensification body (p.17) we calculate the plastic front diameter \( S \) from

\[
p = \frac{\sigma_y}{\sqrt{3}} \left( 2 \ln \frac{S}{D} + 1 - \left( \frac{S}{D} \right)^{2} \right)
\]

with \( d = 65 \text{ mm} \)
\( D = 230 \text{ mm} \)
\( \sigma_y = 1100 \text{ MPa} \)

(for 42 RC)

<table>
<thead>
<tr>
<th>( S ) (mm)</th>
<th>( P ) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>65</td>
<td>584</td>
</tr>
<tr>
<td>66</td>
<td>602</td>
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<tr>
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<td>620</td>
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<td>90</td>
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</tr>
<tr>
<td>94</td>
<td>998</td>
</tr>
</tbody>
</table>

As the plastic front advances, the tangential tensile stress at the outer surface increases, being given by (lab bk 9 p 11)

\[
\sigma_t = \frac{2\sigma_y}{\sqrt{3}} \left( \frac{S}{D} \right)^{2}
\]

As per (4.1) in HC, \( r = 6 \)

<table>
<thead>
<tr>
<th>( S ) (mm)</th>
<th>( \sigma_t ) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>65</td>
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</tr>
<tr>
<td>120</td>
<td>346</td>
</tr>
<tr>
<td>230</td>
<td>1270</td>
</tr>
</tbody>
</table>
Deformation of bore: From Higman's Table p89, eq. (8.50), we have
\[ \Delta d = \frac{d}{2\sqrt{3} \cdot 9d} \]
where \( d \) is the bore diameter and \( S \) is the strain of the platen.

Initially, using the values on previous page, we have (for \( S_0 = 1100 \) MPa
\[ (\frac{4\pi}{3})d^3 \text{(plisقر)} \]
\[ S_0 = 82000 \text{''} \]
\( \Delta d = \frac{d}{2\sqrt{3} \cdot 9d} \)

| \( P \)  | \( \Delta d \)  |  |  |  |
|--------|-----------------|  |  |  |
| 584 MPa| 0.252           |  |  |  |
| 654    | 0.284           |  |  |  |
| 703    | 0.309           |  |  |  |
| 794    | 0.363           |  |  |  |
| 902    | 0.441           |  |  |  |
| 998    | 0.526           |  |  |  |
| ~1200  | 0.788           |  |  |  |
| 1605   | 3.15            |  |  |  |

All the above calculations are based on perfect plasticity, and so will overestimate the yielding effects due to strain-hardening.

Pressuring to 980 MPa may stretch the pressur and a few tens of micrometres, 0.050 \( \mu \)m on 65 mm = 0.0008 strain = 150 MPa stress extra on an expanding plug. Should still be able to disassemble & reassemble end plugs.
Bottom Plug and Nut — Revision

Assuming that a miter ring can be got into the groove at left, so as to seal on D by putting O-ring. The max. shear of the plug at the seal becomes 167mm instead of 77. This reduces the force on the nut from 0.66 MN to 3.53 MN at 16Pa pressure or from 3.26 MN to 2.47 MN at 700 MPa pressure.

If nut is M160x6, pitch \( \phi = 156 \) (for 60 thickness of nut), then shear stress on thread is:

\[
\frac{3.53 \times 10^6}{77 \times 0.156 \times 0.054} = 133 \text{ MPa at 16Pa, quite reasonable.}
\]

700 MPa

The thickness of the nut against bending can be reduced by \( (3.53 \times 0.66)^{0.5} = 0.91 \). However, by spreading the force out to 145\( \phi \), the need to thicken the nut to against bending is largely eliminated.

The thickness below the O-ring, \( L \), needs to be given by:

\[ L = \frac{\pi (75.5^2 - 67^2)}{4 \times (0.0755)^2 - (0.067)^2} \times (0.0755) \times 2 \times 1000 \text{ MPa} = 211 \text{ MPa with } L = 19 \text{ at 16Pa or 148 MPa at 700 MPa.}
\]

Should also be OK.
Top plug and nut

Again, with the same sealing arrangement as proposed for the bottom plug (p 47), the pressure is effective over an area of 67 ft and the threads on a nut 60 mm thick (engaging over 54 mm) should be adequate. However, the loading is now over a smaller area nearer the centre of the nut and so bending may be important. Also we have to provide adequate thread length at the specimen access nut. The force on the nut is

\[ 10 \pi (0.030)^2 = 707 \text{ N} + 100 \text{ KN} = 807 \text{ KN} \]

We assume a M40 X 3 thread (38 mm pitch diameter) and take then the shear stress is

\[ \frac{6.76 \text{ MPa}}{2} \text{ for } l \text{ in length of nut}. \]

For 120 MPa shear stress,

\[ l = \frac{6.76}{0.068} = 0.068 \text{ or } 68 \text{ mm}. \]

If 15 mm is needed below the lowest position of the nut, then total nut thickness = 68 + 15 = 83 mm. If we reduce it to 70 mm and leave the access nut screwed out 10 mm, then length of engagement = 45 and shear stress = 150 MPa at 420 KPa or 105 at 700 KPa.

This could still be marginally acceptable; 75 mm would be a bit better.

Actually, 40 is a 3rd choice size, and 42 X 4.5 would be the 1st choice (39 pitch diameter), which would very slightly reduce the shear stresses again.

Next choice up would be 48 X 5 (pitch 4.5). Then for a nut 70 thick, length of engagement of access nut when screwed out 10 mm is about 70 -18 -10 = 42. Shear stress = 807000

\[ \frac{10 \pi (0.045)^2}{0.042} = 137 \text{ MPa at full load at 1572 (9.6 MPa at 780 KPa).} \]

This may be even better. It reduces the loading at central part of the big nut and reduces the need for stiffening in bending. The force on the plug is mainly at around 50 mm. Total force on plug

\[ 18 \times 0.066 \times 100 \times 1000 = 363 \text{ KN}. \]

If this is born in shear at 67 MPa at 70 mm thick, shear stress = 246 MPa at 14 KPa or 172 MPa at 780 KPa, just acceptable at 42 RC (yield stress - 1100 MPa = \[ \frac{110}{0.53} = 635 \text{ MPa in shear} \]

FS = 2.5
If we use an 80 mm thick nut and M42 × 45 for the acc nut, then with acc nut screwed out 10 mm, length of engagement is 50 mm and so shear stress in acc nut thread is \[ \frac{807000}{\pi \cdot 0.037 \cdot 0.020} = 132 \text{ MPa} \] or 926 Pa at 100 Pa.

If we concentrate the plug force of 3.63 MN at a diameter of 57 mm, the mean shear stress in the nut would be \[ \frac{3630000}{\pi \cdot 0.067 \cdot 0.080} = 215 \text{ MPa at 14 Pa}, \quad 151 \text{ MPa at 700 Pa} \]—looks rather better than for 70 mm thick nut.

This is a bit of an arbitrary number but seems OK as the force will be distributed more than this.

Shear stress above O ring groove: if the depth of groove is \( t \) and the length from groove to surface is \( L \), then shear stress is \( \frac{4t}{L} \times \text{pressure} \). For 15 Pa and a shear stress of say, 200 MPa (SF = 3), then we need \( L \approx 5t \), i.e. about 25 mm for a 5.3\#4 wing.

**Fit of plugs in cylinders**

For Mark III I specified a bore of 77.00 ± 0.02 for the cylinder and diameter 76.97 ± 0.01 for the plug, equivalent to a fit of 77.040/77.000 for cyl, 77.000/76.980 for plug, which is very near to the fit H7-h6 (average location), not quite as close. The outside limits are the same as for H7-96 (precision running location), +19 -29, but the inside limits are closer, in fact touching, whereas the 10 µm minimum clearance of the H7-96 fit is probably better. So fix on H7/96 for plug in bore fit.

Thickness of flange plug

\[ \pi \cdot 30 \cdot t = \pi \cdot (30) \cdot p \]

\[ t = 7.5 \frac{p}{\pi} \quad \Rightarrow \quad t = \frac{7.5 \cdot 1000}{\pi} = 150 \text{ MPa} \quad \text{OK} \]

Better not to make less than 50 in order to spread load over nut face better.
...
Top O-ring Seal on Jacket

The O-ring has a section of φ 1.78
ie cross-sectional area = 0.22489

To fill a 45° triangle of side d
would require \( \frac{d}{2} = 2.489 \)

\[ d = 2.23 \text{ mm}. \]

If we require contact along \( \frac{3}{8} \) of the
90° faces and half of the diagonal face.

The area of the triangle would be

\[ \frac{d^2}{2} = 2 \left( \frac{1}{2} \cdot \frac{d}{2} \cdot \frac{\sqrt{2}d}{4} \cdot \sin 45° \right) \]

\[ = \frac{d^2}{2} \left( 1 - \frac{\sqrt{2}}{8} \sin 45° \right) = \frac{d^2}{2} \cdot 0.875 \]

Then we would require \( d = 2.39 \text{ mm}, \) say 2.4 mm.

The earlier design for ink III called for \( d = 2.7 \text{ mm} \)
ie an area of 3.648 mm², with the O-ring only filling
68% of the area. In actual fact, the retained has the
following dimensions:

\[ \frac{3}{8} 90° \]

so \( d = 3 \text{ mm} \) seems to work ok. It has to be done up
not too tightly in order to avoid pushing down the
jacket (which leads to a leak).
Specimen assembly sizes

Previously used dimensions have been as follows:

- Steel piston: $15.00 \pm 0.01$
- Ceramic pistons: $15.00 \pm 0.02 \text{ OD}$, $2.0 \pm 0.01 \text{ ID}$, $1.921$" diameter
- Iron jacket: $0.543^* \pm 0.002 \text{ ID} \times 0.010 \pm 0.001 \text{ WT}$, $15.06 \text{ (15.11/15.01)}$, $0.25 \text{ (0.28/0.23)}$
- Inner Hylz sleeve: $15.64 \pm 0.01 \text{ ID} \times 20.94 \pm 0.02 \text{ OD}$
- Furnace tube: $21.00 \pm 0.02 \text{ ID} \times 275 \pm 0.1 \text{ OD}$
- Spacer rod: $9.9 \pm 0.1 \text{ OD}$

It now seems to me to be better to aim at a dead fit of tube on piston & allows tube ID tolerance above this, & piston OD tolerance below this, such as:

- Jacket tubing: $15.000 / 15.025$ ID
- Piston: $14.989 / 15.000$

For the fit of inner sleeve over jacket, we can choose $15.60$ as the nominal size and so arrive at

- Jacket tubing: $15.584 / 15.596$
- Inner sleeve: $15.600 / 15.618$

May be better to go to $15.50$ as nominal size, arriving at

- Jacket tubing: OD $15.484 / 15.466$
- Inner sleeve: ID $15.500 / 15.518$
Furnace Design

Now go to an outer can (64.8 OD 63 ID) and an inner double can (58.59.60 φ), giving 15 mm lateral deflection.

Try and load the inner can with a thin plate, 24 OD and 58 OD

Love p. 490 gives for force F and deflection S at the radius b of an outer radius a

\[ S = \frac{1}{8 \pi D} \left( -26 \ln \frac{a}{b} + \frac{1}{2} \frac{a^4 - b^4}{a^2} \right) \]

\[ = \frac{26^2}{8 \pi D} \left( \frac{1}{4} \left( \frac{a^2}{b^2} - \frac{b^2}{a^2} \right) - \frac{\ln \frac{a}{b}}{b} \right) \]

\[ = \frac{d^2}{16 \pi D} \left( \frac{1}{4} \left( \frac{a^2}{b^2} - \frac{b^2}{a^2} \right) - \frac{\ln \frac{a}{b}}{b} \right) \text{ where } d = 2b \]

\[ \frac{a}{b} = \frac{58}{24} \quad d = 24 \]

\[ = 0.613 \times 10^{-5} \text{ in SI base units} \]

\[ D \text{ = flexural rigidity} = \frac{E}{3} \left( \frac{h^3}{2} \right) \]

\[ = \frac{t^3 E}{12 (1 - \nu^2)} \]

\[ \nu = \frac{0.613 \times 10^{-5}}{0.1} = \frac{0.613 \times 10^{-4}}{1.46 \times 10^{-5}} \]

\[ t = \frac{10^9}{90 \times 10^9} \text{ for something like Be copper} \]

\[ t = 0.02 \text{ mm} = 20 \mu \text{m} \]

So not feasible. This calculation is for a full plate and a bit more deflection will occur in a plate with the centre cut out. But not a lot.

Consider alternatively a segmented plate, giving a cantilever beam with thickness at the end bending moment of about 13 mm and length 15 mm, or effectively 13 mm to loading point.
If \( W \) = load, deflection \( s = \frac{Wl^3}{3EI} \) \( 10N \)

Suppose we want \( s = 1\) mm for a total load of 1kg over six segments, i.e. 1/6 kg or 1/6 N per segment. Therefore we need

\[
I = \frac{10 \cdot (0.013)^3}{3 \cdot 0.001} = 1.221 \times 10^{-4}
\]

substituting with \( E = 100 GPa \)

\[
\frac{6h^3}{12} = \frac{0.013h^3}{12}
\]

\[
h^3 = 1.127 \times 10^{-11}
\]

\[
h = 0.000224 \text{ m} = 0.224 \text{ mm}
\]

Increasing the thickness to 0.5 mm will increase the force per mm by about 11 times, and even using two plates well double this, giving overall 440 N

A convenient thickness might be 0.25 mm. Then for 1mm deflection we would have

\[
W = \frac{3ES}{l^3} = \frac{3ES \cdot 0.25^3}{l^3}
\]

\[
= \frac{10 \cdot 0.001 \cdot 0.013 \cdot (0.00025)^3}{4 \cdot (0.013)^3}
\]

\[
= 2.3 N \text{ per segment}
\]

or total force of 28 N for two decks of 6 segments.

This is probably OK.
Electrical Feed-throughs

The following modification, using PZT has worked so far:

It is possible that this could be further modified and simplified by going to a cone construction, or some modified version of this which avoids OD sealing on the O-ring and this has not been tried.

The present question is to stress the metal feed through part. In shear, not taking into account any contribution from the stem on the high pressure end, we have at 1000 MPa pressure:

\[ \frac{1000 \times \pi (0.015)^2}{4} = \tau \times \pi \times 1.6 \times 4 \]

\[ 1000 \times 1.6 \times 4 \]

\[ \tau = \frac{1000 \times 1.6}{16} = 100 \text{ MPa} \]

This should be well within the strength of En25 or Be copper.

Electrical conductor: The stem is about 1.5 mm dia., with a length of about 25 mm, so resistance = \( \frac{\rho L}{A} \)

For copper, \( \rho = 2.1 \times 10^{-8} \text{ m}^2/\text{N} \cdot \text{m} \)

NiCr steel, \( 30 \times 10^{-8} \text{ m}^2/\text{N} \cdot \text{m} \)

\[ 20 \times R = 0.00028 \text{ ohms} \text{ for copper} \]

\[ w = 0.0042 \text{ ohms} \text{ for NiCr steel} \]

So at current 15 amperes, power = \( 1^2R = 0.064 \text{ W} \text{ for copper} \)

\( 0.955 \text{ W} \text{ for NiCr steel} \)

This will give rise to heat temp gradient = \( \frac{1^2R}{kA} \)

\( K = 400 \text{ W/m}^\circ \text{C} \text{ for Cu} \)

\( 35 \text{ W/m}^\circ \text{C} \text{ for NiCr steel} \)

\[ 0.064 / 400 \times (0.0015)^2 = 90 \text{ K/m for Cu} \]

\[ 0.455 / 35 \times (0.0015)^2 = 1540 \text{ K/m for steel} \]

So over 15mm length, \( \Delta T = \frac{1.4 \text{ K for Cu, } 230 \text{ K for steel} \)}
More generally, power \( P = \frac{I^2 R}{A} \) and \( \Delta T = \frac{P}{k A} = \frac{I^2 R}{k A} \).

For copper, \( \rho = 5 \times 10^{-8} \), so \( \Delta T = 0.010 I^2 = 0.5 \text{K} \) for \( I = 7 \text{amp} \)
\( \) or \( 2.3 \text{K} \) for \( I = 15 \text{amp} \).

For NiCr steel, \( \rho = 8.6 \times 10^{-9} \), so \( \Delta T = 1.715 I^2 = 84 \text{K} \) for \( I = 7 \text{amp} \)
\( \) or \( 386 \text{K} \) for \( I = 15 \text{amp} \).

This is an overestimate because it assumes that the heat only escapes from one end, whereas some will escape from the other end, through the insulation sideways, but it may not be overestimated by more than a factor of 3. Therefore for the power heaters (in the event of not being able to use Be copper) we need to silver solder a copper lead into the end of the NiCr steel mushroom; also it assumes that all the heat is generated at the point of maximum temperature whereas its production will be distributed over the whole length. So the length factors are probably around 3-4\( \times \) and this can be increased a bit for the losses of heat sideways, so probably one can reduce \( \Delta T \) by a factor of 4 from the above overestimate, bringing it down to less than 100 K even for steel. So we can probably get away with steel, certainly for the centre winding through the top as well, especially if the current comes down to around 12 A.

If we run at 110 V instead of 40 and need 500 W in the lower winding only then the rms current is only 5 A — then there is certainly no problem. But this doesn’t make sense — we still need the same 1\( ^2 \)R; higher voltage simply means higher instantaneous current with longer "off" fraction to give same rms current.
Oil Reservoir for Intensifiers

Intensifier stroke is 302 mm of diameter 130 mm, giving a swept volume of $4009 \times 10^{-6} \text{ m}^3 = 4.0 \text{ litres}$.

Max. height for right glass would be 180 mm, but 80 mm would be better, to allow some latitude. So cross-section of tank needs to be

$$\frac{4009 \times 10^{-6}}{0.080} = 0.050 \text{ m}^2 \text{ or } 0.22 \times 0.22 \text{ m}^2$$

If we allow 100 mm rise, then cross-section = 0.040 m^2

$$= 0.110 \times 0.365$$

or 110 x 365 mm

Pressure capacity of $\frac{1}{4} \times 0.375 \times \frac{1}{4} \text{ ID} \text{ mild steel tubing}$

$$P = \frac{200 \times 3}{D^2 + d^2} = \frac{200 \times 3}{5} = 120 \text{ MPa}$$

assuming normalised condition with 200 MPa yield stress. In practice, the tube is probably cold drawn.

In testing No 3 machine, a piece was accidentally put in the high pressure line and burst – my recollection is at about 200 MPa, Graeme's is that it was certainly above 100 MPa.

So $\frac{1}{4} \times 0.375 \text{ ID} \text{ tubing is certainly OK for all bottle pressure lines for the gas.}$ The intensifier oil pressure is 100 MPa maximum at 150 MPa gas pressure, so the use of the MS tubing is marginally adequate, with a factor of perhaps 2 in hand. The stainless steel $\frac{1}{4} \times 0.375 \text{ ID} \text{ tubing has a rating of 450 MPa and would be an alternative choice}$

So 200 MPa should be adequate.
\[ D = \frac{K}{pc} \approx \frac{40}{7500 \times 500} = 1.1 \times 10^{-5} \]
Check Calculation on Shrinkage Sleeve on Intensifier Cylinder

No. 1 Cylinder 160.10 max. od. No. 1 Jacket 160.05 Max. i.d.
Therefore minimum expansion for equal diameters = 0.05.
If we allow 0.1 mm clearance during assembly, the total expansion needed at end of assembly is 0.15 mm or a strain of 0.15/160 = 0.00094. This corresponds to a temperature rise of
\[
\frac{0.00094}{11 \times 10^{-6}} = 85 \text{ K} + 25 \text{ room temp}
\]
= 110°C temp at end of assembly.

Characteristic distance \( x = \sqrt{DE} = \sqrt{10 \times 10} = 10 \text{ mm for } t = 10 \text{ sec.} \)
So with perfect thermal contact, the mean temperature might be halved in 10 sec by heat transfer to the cylinders. This may be an exaggerated scenario because of the lack of perfect thermal contact, but to be safe it may be advisable to have about 100% superheat, i.e., start at 200°C.

Check Calculation on Shrinkage Sleeve on Pressure Vessel

No. 1 pressure vessel 230.100 max. od. No. 1 Sleeve 230.050 min. i.d.
Therefore minimum expansion for equal diameters = 0.05.
Allowing 0.1 mm clearance during assembly, total expansion required at end of assembly is 0.15 mm or a strain of 1/230 = 0.00065. This corresponds to a temperature rise of
\[
\frac{0.00065}{11 \times 10^{-6}} = 59 \text{ K} + 25 \text{ room temp}
\]
= 84°C at end of assembly.

Superheat requirement same as above. So 200°C should be adequate.
A Flexible Inner Sleeve for Furnaces

If we go to a furnace that fills the diameter of the pressure vessel fairly nearly in order to minimize the upward transport of heat by convection, the outside annulus between furnace and vessel wall, we lose the ability to accommodate sideways movement of the upper piston by moving the furnace, so we need to accommodate such movement with a flexible inner sleeve which will also restrict convective circulation. It is worth trying a sleeve of alumina paper for this purpose. Tests on the Zircon soft alumina paper indicate that it squashes down to about 0.55 mm with 0.117 MPa pressure, 0.4 mm with 0.24 MPa and 0.3 mm with about 0.6 MPa. Thus sideways forces should not be too bad if the paper is initially compressed to between 0.6 and 0.5 mm, and the paper is given a further 0.1 mm compression in accommodating side movement — this would give ±0.4 mm movement if we had 4 layers of initially 0.55 mm, ie 4 layers packed into 2.2 mm, for the 15.6 mm OD of the piston, assembly packed up to 15.6 + 2 x 2.2 = 20 mm.

The alumina paper needs to be constrained by a stainless steel tube during assembly, a split arrangement would be convenient.

The retaining rings need to be a little less than the furnace bore because of greater thermal expansion, by about 10 x 10^-6, ie Δd = 10.10^-6 x 1280 x 0.21 = 0.25 mm for 1280 °C, and they need to have a wall thickness of say 0.2 mm min. For ease of assembly, the seat of the retaining ring should have 0.1 mm interference (This will disappear with the thermal expansion), and there should be at least 0.3 mm thick inner under the seat, ie ID = 20.4 - 2 x 0.3 = 19.8
Insulation Analysis of Furnace

We consider a model furnace as follows:

For radial heat flow, we assume $1000^\circ C \to 10^\circ C$ over $100\text{mm}$ length.

For axial heat flow, we assume $1000^\circ C \to 10^\circ C$ over $70\text{ mm}$ at top and $55\text{ mm}$ at bottom.

Assume thermal conductivities:

- Zn: about $0.2$ at $1000^\circ C$
- ZrO$_2$:
  - Fibre insulation: $0.2$ [0.1 furnace + 0.1 argon] downwards
  - $0.4$ upwards (extra 0.2 for convection, radiation)

- Indecape etc. of zirconia: $2$
- Pyrophillite: $1.3$
- Macor: $1.7$

- Alumina: 5-30; say average 15 over end core of alumina.
- Iron: 45
Best figure for effective $k$ of ASTH insulation seems to be about 0.14 Wm$^{-1}$K$^{-1}$, compared with 0.14 for still argon.

Manufacturers of ASTH give $k = 0.07$ at $-250^\circ$C, rising to 0.20 at $1100^\circ$C. Thus there must be substantial heat transfer by convection—at least half 0.1 maybe more.

If we take 0.15 as the mean $k$ for ASTH at atmospheric pressure and add 0.14 for still argon, the effective $k$ for ASTH without convection is 0.29; say 0.3 Wm$^{-1}$K$^{-1}$

But from Younglove & Hanley (1986 table) $k$ for argon is around 0.05 to 0.1 at 300 kPa and high temp. by s/rap from 200 kPa & lower T.
Radial Heat Flow:

\[ Q = \frac{2\pi K \Delta T}{\ln \frac{R}{r}} \]

or \[ Q = \frac{2\pi \Delta T}{\ln \frac{R}{r_1} + \ln \frac{R}{r_2}} \]

if two layers \( K_1 \) & \( K_2 \)

\[ Q = \frac{2\pi \cdot 0.2 \cdot 900}{\ln \frac{R}{0.1}} \cdot 0.100 = 142 \text{ W} \]  
\( \text{or} \ 0.14 \text{ W/K} \)

(or if \( K = 0.4 \) \( Q = 284 \text{ W} \))

In case we put pyrophylite outside \( \phi = 50 \),

\[ Q = \frac{2\pi \cdot 900 \cdot 0.1}{\ln \frac{0.5}{0.2} + \ln \frac{0.5}{0.1}} = 176 \text{ W for } K = 0.2 \]

or 336 W for \( K = 0.4 \).

Axial Heat Flow—Top End.

\[ Q = \frac{\pi (D^2 - d^2) K \Delta T}{L} \]

or \[ Q = \frac{\pi (D^2 - d^2) \Delta T}{L} \frac{1}{K_1 + \frac{1}{K_2}} \]

for two materials.

Insulation:

\[ Q = \frac{\pi}{4} (0.06^2 - 0.027^2) \times (0.45/0.4 + \frac{0.02}{2}) \times 900 = 15 \text{ W} \]

Core:

\[ Q = \frac{\pi}{4} (0.027^2 - 0.021^2) \times 0.15, 900 = 44 \text{ W if all aluminin} \]

or \[ Q = \frac{\pi}{4} (0.027^2 - 0.021^2) \times 900 \left( \frac{0.025 + 0.005}{2} \right) \]

Inner sleeve:

\[ Q = \frac{\pi}{4} (0.027^2 - 0.015^2) \times 900 \times 0.15 = 30 \text{ W if all aluminin} \]

much less if SS + fibre paper say 5.

Piston:

\[ Q = \frac{\pi}{4} \times 0.015^2 \times 900 \times 0.15 = 34 \text{ W if all aluminin} \]

or 9 if aluminin + Zn.

Jacket:

\[ Q = \frac{\pi}{4} (0.015^2 - 0.015^2) \times 900 \times 0.15 = 7 \text{ W} \]

Total: 130 W in max aluminin case, 44 W in min aluminin case. 

0.14 W/K

0.05 W/K
See 1981 Servo creep notebook p. 217 (day) for analysis of convective fluid transfer. Also, 1085 p. 82.
Axial Heat Flow - Bottom End.

Insulation:
\[ Q = \frac{\pi}{4} \left( 0.06^2 - 0.02^2 \right) \frac{900}{0.05} = 9 \text{ W} \]

Core:
\[ Q = \frac{\pi}{4} \left( 0.025^2 - 0.015^2 \right) \frac{900}{0.05} = 56 \text{ W} \text{ if all alumina} \]
\[ Q = \frac{\pi}{4} \left( 0.025^2 - 0.020^2 \right) \frac{900}{0.05} = 12 \text{ W} \text{ if Al}_2\text{O}_3 + 12\% \]

Piston:
\[ Q = \frac{\pi}{4} \left( 0.015^2 - 0.012^2 \right) \frac{900}{0.05} = 43 \text{ W} \text{ if all alumina} \]
\[ Q = \frac{\pi}{4} \left( 0.015^2 - 0.010^2 \right) \frac{900}{0.05} = 10 \text{ W} \text{ if Al}_2\text{O}_3 + 15\% \]

Jacket:
\[ Q = \frac{\pi}{4} \left( 0.015^2 - 0.012^2 \right) \frac{900}{0.05} = 9 \text{ W} \]

Total: 117 W if all alumina, 140 W if min. alumina.

Iron tube 0.25 wall thickness:
\[ Q = \frac{\pi}{4} \left( 0.021^2 - 0.020^2 \right) \frac{900}{0.05} = 12 \text{ W} \text{ outwards} \]

If we take \( k = 0.4 \) for fibre insulation upwards, we have a total heat loss of 0.53 W/K for max. alumina case (compared with 0.39 W/K for \( k = 0.2 \)). This increases to 0.58 W/K if an outer pyrophyllite sleeve is added, and can be compared with around 0.7 W/K measured on a furnace with a bit more pyrophyllite at the ends and 5 mm shorter at the ends (run 6110). Thus the analysis seems to be a reasonable approximation to observations, and in turn suggests that the figures taken for \( k \) of the fibre insulation are reasonable.

The figures also suggest that we should be able to get down to 0.37 W/K by reducing use of alumina to a minimum, say around 0.4 W/K, a figure that on earlier furnaces came up as perhaps a minimum when all major convective losses are eliminated.
Digital Meters for Internal Load & Displacement.

Could use a Quantum meter with LVDT conditioning (BSCL 2.2 V 3.5 kHz) for both internal load cell (ILC) and a displacement transducer in the hydraulic actuator (or an extra one attached to connecting bridge in case of failure.)

In case of load with indication to 99.99 kN, the last digit 0.01 kN = 10 N corresponds to 0.5 MPa on a 5 mm diam specimen, and in case of displacement, 0.01% strain on 10 mm = 0.001 mm or $4 \times 10^{-5}$ of a 25 mm range ($\pm 25$ mm transducer), which operating at $\pm 10$ V output, corresponds to 0.4 mV or 0.001% on a range of 9.999 V, so we need 9999 indication in both meters. The appropriate meter is

Q 9999 indication

- 8
- 0
- 1
- L

Q 9801L, current price $1305.
Hydraulic Actuator - Fits

With max load 100 kN on φ75, stress = 22.6 MPa on shaft, ie strain 0.00011 or axial diameteral strain 3.8 x 10⁻⁵
ie diameter increase 0.003 mm, so no risk of binding on standard precision fit -0.010/-0.029

O-rings: For shaft φ75, optimum O-ring is 619 (\(1\mathrm{D} = 74.6\)) ±0.13
although could use 337 (\(1\mathrm{D} = 75.6\)) probably. Section = 5.33 mm,
40° depth of groove with 15% squeeze = 4.62 mm, so should take
groove depth = 4 mm, ie diameter of groove = 75 + 8.8 + 75 + 8.6
ie = 83.80/83.60

For cylinder φ100, optimum O-ring is 621 (\(1\mathrm{D} = 89.7, 2\mathrm{D} = 100.3\)).
Section = 5.33 ±0.13, 40° min depth of groove with 15% squeeze = 4.42,
so should take groove depth 4.4 x 4.3 mm, ie diameter of groove = 100 - 8.8 to 100 - 8.6 = 91.2 to 91.4

Plug to fit into 100 bore, standard precision fit
bore 100.000/100.035 +0 +35
plug 99.988/99.966 -12 -34

Bull seal:

For r = 3, need a ball of φ6.
Design of Loading Pistons

Question of what diameter hole to select for the piston. There are three considerations:

1. Strength of piston against collapse under pressure.
   If the spec. diam. is 10, this means that the diametral could be up to $\phi 2.5$ or 3.0. But if we reduce the spec. to 7, or in the extreme to 5, such a value for d is a bit high — better not more than about 2 in case of 7 spec., or perhaps 1.7 in case of 5.

2. Support of ends of specimen, normally 3 times:
   \[ d = \frac{4T}{P} \quad \text{or} \quad T = \frac{d^2}{4P} \]

   \[ \text{If} \ d = \frac{1}{2} l, \quad T = \frac{8}{3} P, \quad \text{OK for 700/720A} \]

   \[ \text{If} \ d = l, \quad T = \frac{4}{3} P, \quad \text{getting rather high.} \]

   Inertia, \( d > 2 \) seems to be getting uncomfortable & \( d = 1.5 \) would be a good conservative figure.

3. Clearance over thermocouple insulator:
   If using metal sheathed thermocouple, \( \frac{1}{16} = 1.6 \) is a fairly standard diameter, with the next size \( \frac{1}{8} = 2.4 \), getting a bit big. There is at least ±0.1 tolerance on hole in thermocouple tubing, so with 0.1 clearance & 0.1 tolerance, the nominal size \( \frac{1}{16} \) would be 1.8 — leads to 1.7/20 as specification.

If using alumina 2-bore or 4-bore insulation, here are a number of options:

<table>
<thead>
<tr>
<th>OD</th>
<th>1.6</th>
<th>1.6</th>
<th>1.55</th>
<th>1.2</th>
<th>1.25</th>
<th>1.2972</th>
<th>1.7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>0.4</td>
<td>0.4</td>
<td>0.4</td>
<td>0.5</td>
<td>0.2</td>
<td>0.5</td>
<td>0.6</td>
</tr>
<tr>
<td>suit wire</td>
<td>0.3</td>
<td>0.3</td>
<td>0.3</td>
<td>0.3</td>
<td>0.15</td>
<td>0.4</td>
<td>0.5</td>
</tr>
</tbody>
</table>

4. Matthys, but standard 0.5 φ wire requires too big an insulator.

So best compromise seems to be a hole of 1.7/20 φ to take a TC tube of φ 1.6 using 0.3 φ wire, although at the upper end of this range we may be able to use 2 bore Walden-Holm tube, φ 1.8.
Insulators for Internal Load Cell Capacitor Plates

Total length of plate support = 25 - 3.75 = 21.25

Let \( x \) = length \( \text{Al}_2\text{O}_3 \)
\( y \) = length \( \text{Zr}_2\text{O}_2 \)
\( z \) = 21.25 - \( x \) - \( y \) = length \( 18/8 \)

Then we require a compressibility & thermal expansion match to the 25mm of Stavanx (420 type) stabilised stellite.

<table>
<thead>
<tr>
<th>Material</th>
<th>( E )/GPa</th>
<th>( \alpha \times 10^{-6}/K )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{Al}_2\text{O}_3 )</td>
<td>230</td>
<td>6.5</td>
</tr>
<tr>
<td>( Z_{23} ) zirconia</td>
<td>130</td>
<td>10</td>
</tr>
<tr>
<td>18/8</td>
<td>163</td>
<td>17</td>
</tr>
<tr>
<td>Stavanx (420 55)</td>
<td>165</td>
<td>11</td>
</tr>
</tbody>
</table>

For compressibility match:
\[
\frac{x}{230} + \frac{2}{130} + \frac{3.75}{165} + \frac{21.25 - x - y}{163} = \frac{25}{165}
\]

For thermal expansion match:
\[
x \cdot 6.5 + y \cdot 10 + 3.75 \cdot 11 + (21.25 - x - y) \cdot 17 = 25.11
\]

These are solved if:
\[
\begin{align*}
x &= 7.3 & \text{Al}_2\text{O}_3 \\
y &= 7.2 & \text{Zr}_2\text{O}_2 \\
z &= 6.7 & 18/8
\end{align*}
\]

Finally used:

\[
\begin{align*}
\text{Al}_2\text{O}_3 &= 7.3 \\
\text{Zr}_2\text{O}_2 &= 5.7 \\
18/8 &= 3.3
\end{align*}
\]
Wiring Connections for Capacitance Load Cell

Teflon spray was put on centre plate A, B in order to reduce chance of dirt shorting out A, B and other plates.
Pressure vessel distortion due to shrinking on cooling sleeve.
The effect will be strongest at the bottom end.
1\1/4\60 thread has pitch \( \approx 156 \) \( \text{mm} \), and beyond nut, bore is 165, so mean bore at bottom end \( \approx 150 \)

\[ \frac{40}{20} = \left( \frac{\Delta r}{r} \right) \frac{40}{20} \]

or \( \frac{\Delta r}{X S} - 2 \left( \frac{\Delta r}{X} \right)^2 = 0 \) approx

but \( \frac{\Delta r}{X S} + \left( \frac{\Delta r}{X} \right)^2 = 0.080 \) interference (case of no 3 vessel)

\[ 3 \Delta r V = 0.080 \]

\[ \Delta r V = \frac{0.080}{3} = 0.027 \text{ \( \mu \text{m} \)} \approx 27 \text{ \( \mu \text{m} \)} \]

Of the tolerance range, interference varies from 0.020 to 0.100, so \( \Delta r V = 7 \text{ \( \mu \text{m} \)} \) to 33 \text{ \( \mu \text{m} \)}, mean 20 \text{ \( \mu \text{m} \)}

Thus we may expect the bottom thread in vessel to close in about 20 \text{ \( \mu \text{m} \)} on average and up to 30 \text{ \( \mu \text{m} \)} in extreme case.
So in cutting the thread on the vessel, should aim at about 0.080 oversize.

The effect at the top end will be about half that at the bottom due to mass support at inner side.
High-Temperature Furnace Insulation

The standard ASTM insulation used in the 1200°C furnace was 59.5 OD and 27 ID, to fit in an inner can dia. of 61 and with core diam. 27. There are three layers of alumina paper on the radius, i.e. 6 layers per diameter. If we allow 0.4 compressed thickness per layer that gives 2.4 total, so 27 + 2.4 = 30 to fit into 61.0 - 27.0 = 34.0, i.e. about 0.9 squeeze on the AST insulation, thickness 32.5", or 27%.

The first HT furnace supplied to Minneapolis simply had FBC zirconia substituted for AST & zirconia paper for alumina paper. The zirconia paper turned out to pulverize on compression & the FBC was stiff enough that assembly was very difficult.

I now propose to use a composite insulation, with SALI (80% Al2O3 20% SiO2) next to the core, backed up with AST because of the resilience of the latter. The AST should give less to make up for lack of rigidity in the SALI. Possible dimensions would be as follows:

SALI: OD 40.0 ID 27.5 (-2)
ASH: OD 59.5 ID 40.0

It may be possible to increase the 59.5 by a few tenths mm & to decrease the 27.5 similarly, but we need to try one first.
Materials to think about:
Aluminium nitride - high $x$ at room temp
Boron nitride - high $x$, a bit weak. Very hard.
Yb/Zr/zirconia (McDanel) for outer core tubes
Silicon nitride ??
Sialons? (McDanel)

Shrinkage will tend to exacerbate the effects of longitudinal temp. gradient, being probably greater in the outer layers, although greater shrinkage in both parts up the longitudinal gradient could have a mitigating effect.
Shrinkage will also mitigate the effects of radial temp. gradient
Furnace Core Design.

There has been a persistent problem with fracturing of the alumina core at the bottom of the bottom winding. In a few cases (1380°C furnaces in Minneapolis) where there has also been a fracture in the middle of the bottom winding & some longitudinal fracturing in the same region (the latter probably results from contamination under pressure). In all cases, the alumina has been Ceramic Oxide Fabricators material, which appears from light transmission, to have significantly lower density than Duramic alumina. The latter rarely fractured at the bottom of the bottom winding or showed longitudinal cracking, although the conditions may not have been as searching.

The first step to overcoming the above problem will be to obtain some higher density alumina so as to eliminate or reduce effects due to shrinkage.

The second step is to think of ways of ameliorating effects due to thermal gradients. These are most severe at the lower winding due to the higher T gradient downwards. There will also be a radial T gradient associated with the inward flow of heat, which might be substantial. The longitudinal gradient will give the equivalent of a bending stress because successive circumferences longitudinally will become smaller downwards, leading to a longitudinal tensile stress on the outer layers. The radial temp. gradient will also tend to give a circumferential tensile stress in the outer layers and a radial tensile stress.

Some mitigation may result from joining the core at the lower end, similarly as at the upper end, especially if low conductivity upper material could be introduced. May have to put in a liner to cover the join.

The Manchester & Fountain Hill furnaces seem to have been reasonably successful with a join in the core (although running conditions are less severe than at Minneapolis).
A further possibility (suggested by Graeme) is that the fracture at the bottom of the bottom winding may have something to do with the current way of holding the winding in place by looping it around and tightening up by twisting the triple tail. Also, that a slightly higher coefficient of expansion than Mo & so this loop would tend to get thicker. The max tensile stress axially will be a little outside the bearing point of the wire & this seems at least in the last furnace to correspond to where the fracture is. So we should go back to the device of using a separate wire to hold the winding in place, with a minimum tie-twist that can give under tension.

[signature]

Note: The handwriting is clear and legible, with the text discussing a technical scenario involving winding and its potential issues.
Effect of Intermediate Sleeve on Furnace Insulation

A further development from the use of SAAI + ASH insulations would be to put an intermediate sleeve between these two insulation layers with a view to cutting down conduction between the two.

Radial Heat Flow = \( \frac{2\pi CTl}{X} \) where \( X = \frac{X_1}{K_1} + \frac{X_2}{K_2} + \frac{X_3}{K_3} \)

Taking \( K_1 = 0.4 = K_3 \) (6) and \( K_2 = 4 \) for mullite in the intermediate sleeve, we have

\[
X = \frac{\ln 27}{0.4} + \frac{\ln 40}{4} + \frac{\ln 43}{0.4}
\]

\[
= 0.98 + 0.02 + 0.83 = 1.83
\]

compared with

\[
X = \frac{\ln 30}{0.4} = 2.00 \]

If without the intermediate sleeve is about 8\% increase in conductive heat flow with intermediate sleeve.

If we used stainless steel with \( K = 10 \), the effect would be almost the same & could be reduced by using thinner steel – have to have a generous cut-out around the power leads, though.

Putting a mullite sleeve over the windings would give

\[
X = \frac{\ln 27}{4} + \frac{\ln 40}{0.4} + \frac{\ln 43}{0.4}
\]

\[
= 0.92 + 1.73
\]

\[
= 1.76
\]

an increase of 12\% in conductive heat flow. Then would not be much temperature drop across this sleeve \( \frac{0.92}{1.76} \times 130 \approx 20\text{ max} \).

However, there would still be some gap over & a potential crack in the main core & so it may not be very effective in countering the effect of cracking in the core.
4) Incorporation of torque sensing
5) Introduction of splines to serve as anchor for torsion tests.

In finite case, \( \Delta r = \frac{dA}{V} - \frac{dA}{dA + dA + S A} \) for \( \frac{dA}{dA} = 1 \),

\[
\frac{dA}{dA + dA + S A} = \frac{1}{1 + \frac{dA}{dA} + \frac{S}{d} \frac{dA}{dA}}
\]

\[
\frac{dA}{dA + dA + S A} = \frac{1}{1 + \frac{S}{d} \frac{dA}{dA}}
\]

If \( \frac{S}{d} \ll 1 \),

\[
\frac{dA}{dA} \approx -\frac{\frac{S}{d}}{2 \left(2 + \frac{S}{d}\right)} = -\frac{\frac{S}{d}}{2 \left(1 + \frac{5}{2d}\right)}
\]

ie not quite linear when \( S \) becomes appreciable relative to \( d \) with only one active capacitor.

See p78 for case of two active capacitors.

For effect of plastic (e.g., film between plates, see fax to Mainprice May 9, 1974.
Revision of Internal Load Cell

The design of the ILC has been extensively modified in the following respects:

1) The elastic element is again on the outside (as for the strain gauge load cells) with a view to increased bending stiffness and better protection for the capacitor plates.

2) The plates are now full-circle, and the use of a full bridge of active capacitors has been given up, so as to avoid the bother of balancing the bridge during assembly. Further, for mechanical simplicity, of the remaining two capacitors in the high pressure space, one is fixed as a reference, and only one varies with the load. (This arrangement also saves one feed line).

3) The fluid port and connection is modified for better fitting of the stem carrying out the electrical leads.

The electrical circuit is now as follows (apart from any phasing needs):

\[ C = \frac{2A}{d} + Z = \frac{1}{\omega C} = \frac{d}{\omega A} \]

If apply voltage \( V \), then voltage \( V \):

\[ V = \frac{2ZkV}{Zk+Z} = \frac{1}{1 + \frac{dA}{A}} \frac{dA}{dA} \]

If \( A, A_R, d_R \) are fixed & d varies by amount \( S \), we have

\[ \frac{dV}{V} = -\frac{1}{1 + \frac{dA}{A}} \frac{dA}{A} S \]

Therefore, minimum value of the denominator is \( 2d \) when \( \frac{dA}{A} = 1 \)

So max value of \( \frac{dV}{V} = -\frac{1}{4d} S \) when \( \frac{dA}{A} = 1 \)
It is notable that the sensitivity in this configuration
\( \left( \frac{dV}{dU} = \frac{4}{a^2} \right) \) is independent of the dielectric and depends
only on the displacement relative to the active plate
spacing, \( S \), so maximizing the output requires a
maximum elastic distortion \( S \) and a minimum
plate spacing \( d \). The only effect of increasing
the permittivity \( \varepsilon \) is to increase the actual
capacitances (as given in Fig. 1d), which is advantageous
from the point of view of minimizing lead effects.

If plate spacing = 0.25 mm, and \( S = \frac{0.001 \times 0.030}{30 \text{ mm elastic gage length}} = \frac{30 \times 10^{-3}}{30 \times 10^{-3}} = \frac{30}{30} \text{ mV per volt excitation,}

With 18 V excitation, the output should therefore be
300 mV full scale with maximum strain 10^{-3} in
the elastic element at 30 mm length. This is
the figure given to George on 14/10/82.

If we get a sensitivity of \( 1 \times 10^{-4} \) (~ 14 bit), we can detect
\( 10^{-4} \times 100 \text{ KN} = 10^{-3} \text{ N} \), corresponding to \( \approx 10 \text{ MPa} \)
on 10 mm diameter or 10\% sensitivity on 1 MPa stress
(= 10 bar, about the strength of Solnhofen at 900°C).

If the plates are \( \sim 1000 \text{ mm}^2 \) less cut-out, say 800 mm^2 area,
the \( C = \frac{\varepsilon - \varepsilon_0 A}{\varepsilon_0} = \frac{1.3 \times 85 \times 10^{-12} \times 800 \times 10^{-6}}{0.25 \times 10^{-3}} = \frac{37 \times 10^{-12}}{37 \text{ pF}} \)
\( = 37 \text{ pF} \).

1981 book day 166 This may be increased a bit by the
presence of milar film between the plates.
If we used O-217, 
\[ \phi_{40} \times 6 = \phi_{30} \]
\[ \phi_{48} = \phi_{30} \]
\[ \text{3 support} \]
- would give greater stability in bending

O-ring O-223, 3.53 section, is 41 ID, 48 OD, would be suitable - too big. Needs about 5 wide groove at least, better?

If we came down to 217, a standard size, ID = 30, OD = 37; groove width = \( \frac{1}{2} (48-30) = 9 \), more than enough - too small.

O-ring O-222 is 45 OD x 38 ID, makes groove 5 wide.

Bearing these on splines:
Torque Measurement

If the ILC is going to be used to measure torque as well, it will have to be constrained from rotating by means of splines attached to the lower end. This will necessitate a re-arrangement of the attachment to the first piston: instead of the ILC screwing into the top of the piston, the piston will have to be screwed into the bottom of the ILC base.

If we have a spline arrangement as sketched opposite, the mean diameter at the splines is 57 mm. Then the mean diameter at the threads to support 150 kN in tension on the load cell body is given by

\[ \frac{\pi}{4} \left( \frac{0.057^2 - D^2}{D} \right) \sigma = 150,000 \]

or \[ D^2 = \frac{4 \cdot 150,000}{\pi \sigma} - 0.057^2 \]

With \( \sigma = 200 \text{ MPa} \), \( D = 0.0479 \) i.e. 48 mm

Using 1750 x 1.5 has pitch dia = 49 \( \times \) minor dia = 48,

so \( \sigma = 225 \text{ MPa} \), probably acceptable (FS at least 5).

For length of thread at \( \tau = 100 \text{ MPa} \),

\[ \frac{\pi d \tau}{E} \tau = 150,000 \]

\[ L = \frac{150,000}{\pi \cdot 0.049 \cdot 100 \cdot 10^{-6}} = 0.0097 \text{ i.e. } 10 \text{ mm}. \]

An actual length of 8 mm would be adequate, about 5 threads.

Revision: It would be desirable to have only one standard thread M52 x 2 for which there is already a master. In this case, with mean diameter of spline = 58, as before, and max diam of thread = 52, the tensile stress in the root of the attachment thread of the ILC is

\[ \frac{\pi}{4} \left( \frac{0.058^2 - 0.052^2}{0.058 - 0.052} \right) \sigma = 289 \text{ MPa}, \]

or about under 200 MPa at design nominal load 100 kN. This is OK. The shear stress on 8 mm of thread is

\[ \frac{150,000}{\pi \cdot 0.051 \cdot 0.008} = 117 \text{ MPa} \] also OK.

Drilling groove is now #01D. Can still use 0.223 but have to be careful it doesn't slip off (assemble upside down)
Splines for top end: pitch 30° (15° + 15° land)
   ie 12 lugs
   - 3 for plate support
   - 9 for supporting piston assembly.

In case of four cut-outs of 30°, remaining circum. = 240° = \( \frac{2}{3} \)

So \( \frac{1}{3} \left( \frac{\pi}{4} \left( 5^2 - d^2 \right) \right) = 500 \) \( d = 41.8 \), say 42

So wall thickness = 5.

To accommodate \#173 hold-down screws for the capacitor plates, have to turn down the 5.5\( \phi \) heads to 5\( \phi \). Still gives the same contact width (1mm) as a 1/12\( \phi \) screw with 45\( \phi \) head & is more robust (hie. is 2.5 AF w 2.9 across edges)
Shear stress in attachment nut is given by
\[ T = \frac{150000}{\pi \times 0.052 \times L} \]

With \( T = 100 \text{ MPa} \)
\[ L = \frac{150000}{\pi \times 0.052 \times 100 \times 10^{-6}} = 0.0092 \text{ in} \]
\( L = 9 \text{ mm} \).

So 10mm is plenty, 8mm would do.

The normal stress on the bearing area is
\[ \frac{150000}{\pi \times 0.050 - 0.047} = 339 \text{ MPa} \]
and if we introduce a splined attachment for torsion, this rises to 677 MPa, or higher, when some grooves are used up for attaching torque capacitor plate, say to \( \approx 800 \) MPa. This is still below the yield stress, so should be OK, esp. for nominal load of 100 kN.

For the top torsion attachment, mean dia = 47, so if

Torsional bearing area is \( 3 \times 6 \times 6 \) bearing stress \( \sigma \) is given by
\[ 0.003 \times 0.006 \times \frac{1}{2} \times 0.47 \times 12 = 1000 \text{ Nm} \]
\[ \sigma = 197 \text{ MPa} \] OK.

Shear stress in root of lug is given by
\[ \frac{2 \times 2 \times 47}{\pi \times 0.047 \times 0.003 \times \frac{1}{2} \times 12} = 1000 \text{ Nm} \]
\[ T = 192 \text{ MPa} \] a bit high but tolerable.

Cross-section of elastic element
Suppose we have the strain to 0.001 at 100kN, i.e. 200 MPa stress.

Then cross-section = \( \frac{100 \times 10^3}{200 \times 10^{-6}} = 0.5 \times 10^{-3} = 500 \text{ mm}^2 \) required (25φ)
520, 4410 = 603 mm^2, i.e. can only affect to cut out \( \frac{2\pi}{3} \) of circumference, i.e. 61°, or 20° per opening (0.06 fraction), or 7mm. Not enough, used at least 25° × 25° = 0.208 (diameter \( \frac{38}{2} \text{ mm} \) or 0256 of the circumference, i.e. 3 × 30°, i.e. \( \frac{3}{4} \left( \frac{2\pi}{4} \right) = 500 \), \( d = 43 \). With \( d = 44 \), we have support area = 451 mm^2, or stress 220 MPa, strain 1.8 × 10^{-3}. This would still be acceptable.
Specimen assembly avoid support. This is a question whether it would be simpler to make this in one piece rather than having a removable bottom (dotted left). Considerations:

1. Much stronger in presence of cut-outs for torque transmitting plate, etc.
2. Simple for pressure connection (coming to seal on side rather than on bottom — but could seal on bottom).
3. EDM could produce a flat bottom with undercut.
4. More difficult to clean bottom face (need a special cap).

Pressure Compensation

From Kay & Baby, the compressibility of most steels is 165 GPa & thermal expansion ~11 ×10^-6 K^-1 (actually ~10 ×10^-6 K^-1 for 12% C steel). For PSZ, Nilaco catalogue gives E = 205, ν = 0.31, hence k = 180 GPa and thermal expansion is 10 ×10^-6 K^-1. Thus introducing some PSZ insulating spacers does not affect temperature sensitivity. For pressure, the effect at 300 MPa is

\[
\frac{300 - 300}{165000} \times \frac{3}{180000}
\]

= 0.0005 mm = 0.5 μm in a total displacement of \(\frac{300}{165000} = 18.6 μm\). Thus the effect of the zero should be around 1% of a 300 MPa, not worth worrying about. If we did do something, could substitute some glass for steel in the plate support, but this would introduce a bit of temperature sensitivity on account of \(α ≈ 19 \times 10^{-6} K^{-1}\).
Case of one active capacitor with $A_1 = A_2$, $d_1 = d_2$

\[
\frac{V}{V} = \frac{d}{d + d} = \frac{1}{2}
\]

\[
\frac{V + Au}{V} = \frac{d + \frac{d}{2}}{d + d} = \frac{d + \frac{d}{2}}{2d} = \frac{1 + \frac{d}{2}}{2} = \frac{1}{2} + \frac{d}{4}
\]

\[
\Delta U = \frac{1}{2} \left( \frac{1 + \frac{d}{2}}{1 + \frac{d}{2}} \right) = \frac{1}{2} \left( 1 + \frac{d}{2} \right)
\]

\[
= \frac{d}{2d} \left( 1 + \frac{d}{2d} \right)
\]

The output is non-linear when $A_1 = A_2 = 0$. Therefore for linearity we should use two active capacitors in the half-wedge and keep the areas exactly equal.
In the case of two active capacitors:

\[
\frac{V}{V} = \frac{Z_1}{Z_1 + Z_2} = \frac{d_1 + d_2}{\varepsilon_1 + \varepsilon_2}
\]

\[
\frac{V + \Delta V}{V} = \frac{d_1 + \Delta d}{\varepsilon_1} + \frac{d_2 - \Delta d}{\varepsilon_2}
\]

If \( \Delta \) = displacement

\[
\Delta V = \frac{d_1 + \Delta d}{\varepsilon_1} - \frac{d_1}{\varepsilon_1}
\]

\[
= \frac{1 + \frac{\Delta d}{\varepsilon_1}}{1 + \frac{\Delta d}{\varepsilon_1}} - \frac{1}{1 + \frac{\Delta d}{\varepsilon_1}}
\]

If we put \( d_2 = d_1 + \Delta d \) and drop the suffix, we have

\[
\frac{d_2}{d_1} = \frac{d_1 + \Delta d}{d_1} = \frac{1 + \Delta d}{d_1} \quad \text{and} \quad 1 - \frac{\Delta d}{d_1} = 1 - \frac{\Delta d}{d_1 + \Delta d}
\]

Then \( \Delta V/V = \frac{1 + \frac{\Delta d}{\varepsilon_1}}{1 + \frac{\Delta d}{\varepsilon_1}} - \frac{1}{1 + \frac{\Delta d}{\varepsilon_1}} \)

Now assume \( \frac{\Delta d}{d_1} \ll 1 \) and neglect higher powers, so

\[
\Delta V = \frac{(1 + \frac{\Delta d}{d_1})}{(1 + \frac{\Delta d}{d_1})} - \frac{1}{1 + \frac{\Delta d}{d_1}}
\]

\[
= \frac{1 + \frac{\Delta d}{d_1}}{1 + \frac{\Delta d}{d_1}} - \frac{1}{1 + \frac{\Delta d}{d_1}}
\]

\[
= \frac{\Delta d}{2 + \frac{\Delta d}{d_1}}
\]

\[
= \frac{\frac{\Delta d}{2} + \frac{\Delta d}{d_1} + \frac{\Delta d}{d_1}}{2 + \frac{\Delta d}{d_1} + \frac{\Delta d}{d_1}}
\]

\[
= \frac{\frac{\Delta d}{2}}{2 + \frac{\Delta d}{d_1} + \frac{\Delta d}{d_1}}
\]

\[
= \frac{S}{2d} \left(1 - \frac{1}{2d} - \frac{1}{2d} \right)
\]

The effect of non-centrality of the middle plate (\( \Delta d \approx 0 \)) is to degrade the sensitivity slightly but not affect the linearity.
The effect of having the two areas different ($\Delta A = 0$) is to have a smaller degrade the sensitivity to a smaller degree but also to introduce a slight non-linearity. Thus care should be taken to have perfect symmetry of the two outer plates with respect to the inner, and to have the inner plate reasonably centred. In this case we have

$$v = \frac{S}{2d} V$$

If $d = 0.25 \text{ cm}$ and $S = 0.030 \text{ cm}$, $V = 10V$,

then $v = 600 \text{ mV full scale}$. Should we relax $d$ to 0.5 mm to give 300 mV F.S.? — main disadvantage is reducing the capacitance relative to that to earth; may not matter much if we could put a buffer amplifies close in.

With only one active capacitor, the sensitivity halves again, i.e. to $\approx 300 \text{ mV full scale}$ for 0.25 mm spacing.

[Contd p 92]
Redesign of HP connection to intensifiers in order to avoid blind-ended bore of cylinders. Thus, use an inserted plug in same bore diameter. 

Box = 401 mm, force on plug at 14Pa = 1.32 MN
If we use a nut with M100x6 thread, pitch dia = 96,

d = 0.096, L = 1.32 x 10^6

For shear stress of 100 MPa in thread, i.e. L = 0.0438, or 44 mm
If we make L = 35 mm, T = 125 MPa
L = 30 mm, T = 146 MPa

From p 22, the comp. shear stress in main pressure vessel closure nut in 185 MPa, so 30 mm long nut should be OK. The nut should be nearer to 40 mm long for good proportions, so we could make the thread 40 deep, including the run-out—maybe 14 deep is better, to give space for an extracting thread on the plug.

M100 x 6 thread has minor diameter = 93.5 mm.

The top end of the plug can be φ40; if the central stem is φ20, the bearing stress will be reasonably well distributed, & only 1/6 higher than without the stem.

The stress at the root of the 1/16 pressure connection thread with 14Pa transiently across the full φ5 cone would be

\[
\frac{10^9 \times (0.005)^2 \times 14}{\pi \times (0.020)^2 - (0.016)^2} = 174 \text{ MPa}
\]

Which should be OK, & unlikely to be reached.
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