DESIGN CALCULATIONS FOR HPT SERVOCONTROL MACHINE (1988 et seq)
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Book 4 of HPT Design Calculations
High Sensitivity Load Cell for Orleans

This will be a development of the LVDT load cell in which, instead of measuring the elastic deflection of the body of the load cell, a special insert will be put in the load cell instead of the present LVDT carrier (PI 4504).

For the standard load cell, the LVDT deflection is $\pm 0.32 \text{ mm}$ (p. 235) for 100 kN range. This range can be reduced to 10 kN by using 10X gain in RS card, corresponding to deflection $\pm 0.032 \text{ mm}$. So we could readily measure 1 kN force with a sensitivity of order of 1/1000.

The Orleans requirement is to measure viscosity of $10^{-10^7} \text{ Pa s at } T = 10^{-4}$, i.e. $T = 10^{-7} \cdot 10^{-4} = 10^{-11}$ on a 15 mm specimen (317 mm²), so $F = 0.018 \text{ N}$. Thus they require a load range of $\approx 0.1 \text{ N}$.

We could use an insert in the LVDT load cell as at left, using two flexure plates of thickness $t$ and length $L$ with breadth (normal to page) of $b$. Roughly $L = 18$ and $b = 40$ mm would be available.

From Marks p. 426, we have:

$$S = \frac{W L^3}{192 EI}$$

For a case in which half the rigid ended beam (at left), so $L = \frac{L}{2}$ or $L = 2L$

For rectangular section, $I = \frac{b t^3}{12}$

Also Salmon p. 103.

Salmon p. 79.
Note: The bending configuration on previous page can also be viewed as

\[ S = \frac{W(L^3)}{192EI} \]

Put \( W = 2F \), \( L = 2l \)

\[ S = \frac{2F \cdot 8l^3}{192EI} = \frac{Fl^3}{12EI} \]

Putting \( I = \frac{b^4}{12} \)

\[ S = \frac{Fl^3}{12EI} \]

or \( S = \frac{FL^3}{Ebh^3} \)
Therefore \( S = \frac{Wl^3}{24Et^3} = \frac{Wl^3}{25Et^3} \).

With \( l = 18 \) and \( t = 40 \), we have

\[ S = 2E40t^3 \] and \( E = 210,000 \text{ N mm}^{-2} \)

\[ S = \frac{W}{2881t^3} \]

If we work at the same sensitivity as for the standard load cell, we have \( S = 0.03 \text{ mm} \) for max load range.

\( W = 2881 \times 0.03 \times t^3 = 922 \times t^3 \text{ N} \).

or 922, say 1000 N at \( t = 1 \text{ mm} \).

115, say 100 N at \( t = 0.5 \text{ mm} \).

There are two flexure plates, so range = 1800 N for \( t = 1 \text{ mm} \),

230 N for \( t = 0.5 \text{ mm} \).

So we could measure \( \approx 2 \text{ N} \) at 1% sensitivity with \( t = 0.5 \text{ mm} \),

0.2 N at 10%. 

...
RS Card Ranges: 250 - 750 mV/V

<table>
<thead>
<tr>
<th>Range</th>
<th>Conversion Factor</th>
<th>Multiplier</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 - 300</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>50 - 150</td>
<td></td>
<td>5</td>
</tr>
<tr>
<td>25 - 75</td>
<td></td>
<td>2.5</td>
</tr>
<tr>
<td>10 - 30</td>
<td></td>
<td>5</td>
</tr>
<tr>
<td>5 - 15</td>
<td></td>
<td>25</td>
</tr>
<tr>
<td>Plus x10</td>
<td></td>
<td>50</td>
</tr>
</tbody>
</table>

PCD 37 for position LVDT

PCD 41

Load cell

500

200

100

50

20

10
In order to gain more sensitivity, we could make a special load cell body without splines, to run on the inner radius of the splines (Φ 55) and having minimal wall thickness, say 1 mm, so ID is 53 mm. Then the elastic element of the LVDT carrier can have OD 52.5 x 52. Then the force LVDT's can be colinear with the pivot, LVDT and we can have about 25 mm length of flexure plate.

Now $S = \frac{W}{2EIC^3}$. With $L = 25$, $b = 40$, $E = 210,000$,

$S = \frac{W}{1075E^3}$

One pair of LVDT's gives full range on 25/75 mV/V setting for deflection of 0.32 mm. If we put $L = 1$ mm, $S = \frac{W}{1075}$ or $W = 3.44 \times 10^5$ N per blade, or $688$ N for two blades.

At this level we should have 0.1% sensitivity. Thus at about 100 N, we have ~ 1% sensitivity.

If we take $L = 0.8$ mm, then $W = 1.1075 \times (0.8)^3 = 330$ N for 2 LVDT's or 1100 N for 4 LVDT's.

Taking into account that there are two blades to support the load, we should take

$S = \frac{W}{2150E^3}$

Experience with the standard LVDT load cell indicates that with two LVDT's we get output in the 25/75 mV/V range (for 100 kN full scale) at $S = 0.32$ mm — say 570 mV/V.

Then for $S = 1$ mm, output is 150 mV/V.

If we take $S = 1$ and $L = 0.8$ mm, then $W = 1100$ N.
Max moment \( = \frac{WL}{8} \)

\[ = \frac{2F \cdot 2l}{8} = \frac{FL}{2} \]

\[ M = \frac{FL}{2} \]

Stress relation to stress is

\[ M = \sigma I \]

\[ = \frac{\sigma I}{C} = \frac{\sigma I}{12} = \frac{2\sigma I}{L} \]

\[ I = \frac{6t^3}{12} \]

\[ \therefore M = \frac{2\sigma}{L} \cdot \frac{6t^3}{12} = \frac{\sigma \cdot 6t^2}{6} \]

\[ \therefore \sigma = \frac{6M}{6t^2} = \frac{6}{6t^2} \cdot \frac{FL}{2} \]

\[ \therefore \sigma = \frac{3FL}{6t^2} \]
Stresses: From opposite, max stress in beam is
\[ \sigma = \frac{3FL}{4BE^2} \Rightarrow F = \frac{\sigma E^2}{3L} \]

For present design, we have \( L = 24 \) and \( b = 40 \) approximately. We could also put a limit on \( \sigma \) of about 500 MPa if we use ready-heat-treated steel with yield stress 800 MPa (Assat 718). Then we have
\[ F = \frac{500 \times 40 \times E^2}{3 \times 24} \]

or
\[ F = 278 E^2 \tag{1} \]

Reflection: From opposite p. 246, we have
\[ S = \frac{F}{E t^3} \]

For \( L = 24 \) and \( b = 40 \) and \( E = 210,000 \), this gives
\[ F = \frac{210,000 \times 40 \times E t^3}{24^3} = 608 E t^3 \]

\[ F = 608 E t^3 \tag{2} \]

Equating (1) and (2) gives
\[ 8 t = 0.457 \]

<table>
<thead>
<tr>
<th>( t ) in mm</th>
<th>( S ) in mm</th>
<th>( F ) in N</th>
<th>2F for double plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>1.5</td>
<td>25</td>
<td>50</td>
</tr>
<tr>
<td>0.5</td>
<td>0.91</td>
<td>69</td>
<td>138</td>
</tr>
<tr>
<td>0.7</td>
<td>0.65</td>
<td>136</td>
<td>272</td>
</tr>
<tr>
<td>0.8</td>
<td>0.57</td>
<td>178</td>
<td>356</td>
</tr>
<tr>
<td>1.0</td>
<td>0.46</td>
<td>278</td>
<td>556</td>
</tr>
<tr>
<td>1.2</td>
<td>0.38</td>
<td>400</td>
<td>800</td>
</tr>
<tr>
<td>1.5</td>
<td>0.30</td>
<td>625</td>
<td>1250</td>
</tr>
<tr>
<td>2.0</td>
<td>0.23</td>
<td>1100</td>
<td>2220</td>
</tr>
<tr>
<td>3.0</td>
<td>0.15</td>
<td>2500</td>
<td>5000</td>
</tr>
</tbody>
</table>

(Add further detail p. 261)
Measurement sensitivity with $S = 0.30$, $t = 1.5\, \text{mm}$

From previous page, if we use $t = 1.5\, \text{mm}$, the deflection $S$ will be $0.30\, \text{mm}$ when the applied load $2F$ is $1250\, \text{N}$.

For the standard load cell with mechanical gain $4K$, we have a deflection of $\pm 0.32\, \text{mm}$, giving an amplified signal of $\pm 10V$ on the $25-75\, \text{mV/V}$ scale. This is with $2.25\, \text{LVDT's}$ in series. If we put $4\, \text{LVDT's}$ in series, the full-scale signal of $\pm 10V$ would be obtained on the $50-150\, \text{mV/V}$ scale.

Thus the $0.30\, \text{mm}$ deflection for $t = 1.5\, \text{mm}$ at $1250\, \text{N}$ load would give a $\pm 10V$ amplified signal on the $50-150\, \text{mV/V}$ scale.

<table>
<thead>
<tr>
<th>Scale</th>
<th>Max Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>25-75</td>
<td>1250 N</td>
</tr>
<tr>
<td>25-15</td>
<td>125 N</td>
</tr>
<tr>
<td>5-15</td>
<td>250 N</td>
</tr>
<tr>
<td>5-15 x10</td>
<td>12.5 N</td>
</tr>
<tr>
<td>5-15 x10</td>
<td>25 N</td>
</tr>
</tbody>
</table>

So we could probably get $\approx 10 \, \text{N}$ full scale with max amplification with $t = 1.5\, \text{mm}$. Max load = $1250\, \text{N}$.

If we use $t = 1\, \text{mm}$, $S = 0.46\, \text{mm}$ when load = $556\, \text{N}$.

Full scale signal would require somewhere between $50-150$ and $100-300\, \text{mV/V}$ scale. Thus the use a sensitivity range $5-15\, \text{mV/V} \times 10$ gain would give max load $\approx 8\, \text{N}$.
Max torque is set by friction at the ends. For rigid-plastic bar, torque = $\frac{\pi d^3 (\mu p)}{12}$.

If $\mu = 0.5$, $p = 500$, torque = $\frac{\pi \times 250 \times 10^6}{654} \times \frac{12}{d^3}$ if $d$ in m.

So for $d = 15$ mm, $M = 221$ Nm.

If $\mu = 1$, this value is doubled.

If $\mu = 0.5$, $p = 300$ kN, $M = 133$ Nm.
Torque calibrating bars for higher sensitivity.

Instead of the bar with dogclutch at one end and splines at the other, we could use a bar in place of the normal specimen assembly with the specimen assembly dogclutch termination at one end and splined end at the other end.

The effective length will be 160 instead of 226/240. If the bar were made of DF2 at HRC 60, 0.2% yield stress is 2200 MPa, so $\sigma_{0.2} \approx 1270$ MPa, say 1000 MPa $\approx 10^9$ Pa although this may cause a lot of yielding at the corners.

$$ T = \frac{16M}{11d^3} \rightarrow d^3 = \frac{16}{11} \cdot \frac{M}{T} = \frac{16}{11} \cdot \frac{1000}{15} = \frac{16}{11} \cdot 33.33 \text{ mm}^3 $$

Thus, for $M = 1000$ Nm,

<table>
<thead>
<tr>
<th>$d$ (mm)</th>
<th>$T$ (Nmm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>17</td>
<td>20</td>
</tr>
<tr>
<td>12</td>
<td>15</td>
</tr>
<tr>
<td>8</td>
<td>10</td>
</tr>
<tr>
<td>6</td>
<td>7</td>
</tr>
<tr>
<td>4</td>
<td>5</td>
</tr>
</tbody>
</table>

$$ M = \frac{\pi G d^4}{32L} \rightarrow \theta = \frac{32 \cdot M}{\pi \cdot 80 \cdot 10^9 \cdot d^4} = \frac{32 \cdot 0.16 \cdot 10^3}{\pi \cdot 80 \cdot 10^9 \cdot 33.33} = 0.0537 \text{ radian} $$

$$ \theta = \frac{0.0537 \cdot 10}{d} $$

For $M = 1000$ Nm, $\theta = 0.127$ radians

<table>
<thead>
<tr>
<th>$d$ (mm)</th>
<th>$\theta$ (radian)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>0.121</td>
</tr>
<tr>
<td>10</td>
<td>0.204</td>
</tr>
<tr>
<td>7</td>
<td>0.255</td>
</tr>
<tr>
<td>5</td>
<td>0.326</td>
</tr>
</tbody>
</table>

Alternatively, if we made the bar of Assab 718, $\sigma_{0.2} \approx 400$ MPa, then for $M = 1000$ Nm, $d = 0.023$, say 200 mm, and $\theta = 0.068$

<table>
<thead>
<tr>
<th>$d$ (mm)</th>
<th>$\theta$ (radian)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>0.103</td>
</tr>
<tr>
<td>11</td>
<td>0.129</td>
</tr>
<tr>
<td>7.5</td>
<td>0.193</td>
</tr>
<tr>
<td>5</td>
<td>0.326</td>
</tr>
</tbody>
</table>
Timoshenko & Goodier p 308

Their $\Theta$ = twist per unit length. But we take $\Theta$ as total twist, therefore

$$\Theta = \frac{3 M l}{bc^3}$$

$$T_{\text{max}} = \frac{3 M}{b c^2}$$

For a thicker plate, a factor somewhat greater than 3 is needed (Tim. 84. p312,313), ~ 3.2 for 10:1

3.7 for 3:1

4.0 for 2.5:1

4.8/7.1 for 1:1

Also Salmon p253
High Sensitivity Torsion Load Cell

If we want to measure viscosities $10^{-7}$ to $10^{-8}$ Pa s with strain rates $10^{-4}$ to $10^{-6}$ i.e. shear strain $= 10^3$ to $10^7$ Pa.

For $T = 10^3$ Pa, and specimen $d = 15$ mm,
\[
M = \frac{\pi d^3 T}{16} = \frac{\pi (0.015)^3 \cdot 10^3}{16} = 0.0007 \text{ Nm}
\]

For $T = 10^7$ Pa,
\[
M = 7 \text{ Nm}.
\]

If we wish to measure the 0.0007 Nm to 1%, then a range of 0.007 Nm would be OK, say 0.01 Nm.

The normal torque cell will measure $\sim 1000 / 50$ say around 20 Nm full scale. So 10 Nm full scale on the high sensitivity torque cell would give some overlap. Thus, tentatively, take 0.01 to 10 Nm as the range 2 or 100:1. So a displacement of about 0.050 mm would give the maximum range of 10 Nm (with 4:1 mechanical gain), this means 0.1 mm movement of the RVDT core.

If the 0.050 mm displacement occurs at a radius of 15 mm, then max $\theta = 0.050 / 15 = 0.003$ radians.

From $0.003 = \frac{3 \cdot 10 \cdot 0.02}{(0.06)^2 \cdot 80 \cdot 10^{-4}}$, we get $c = 0.0035$ i.e. 3.5 mm.

If we take two webs, $M = 5$ each, which would reduce $c$ to 2.8 mm.

For $C = 1$ mm, we have $M = \frac{0.6 c^3}{3b} \times 2$.

For $C = 1.5$ mm, $M = 1.6$ Nm.
For $c = 2 \text{ mm}$, \( M = 4.8 \text{ Nm} \) for double web.

Thus with a web thickness of $1-2 \text{ mm}$, we can get a max torque range of $0.5 - 5 \text{ Nm}$ roughly.

In finest setting, assuming the gain can be increased about $50\times$, we get a torque range of $0.01 - 0.1 \text{ Nm}$ so we should be able to measure a torque of about $0.001 \text{ Nm}$ to an accuracy of $1\%$ or so, meeting the requirements, more or less.

Max stress \( \tau = \frac{3T}{bc^3} = \frac{3 \cdot 0.5}{0.06 \cdot (0.001)^2} \approx 25 \text{ MPa} \), OK.

For 1 mm web: \( \approx 25 \text{ MPa} \), OK.
The spheres are at 150 to meet in ordinary LLC.
HS Combined H/C

The LVDT's both require a hole 9.5 mm diameter. If they are put on PCD = 245, then the outer diameter is 45 + 9.5 = 54.5 cm, comfortably inside the spline minor diameter, with plenty of heat under the spline major diameter.

The splines are at 30° and the PI LVDTs can also be located at 30°. The arc length between centres is

\[ \frac{30 \times \pi \times 45}{360} = 11.78 \]

leaving 11.78 - 9.5 = 2.28 mm land between the LVDTs. If we put PI LVDTs both sides, the distance between them is 29.5 mm. So the slots can be up to 28 wide (or even wider if no LVDTs are put on the RTH side for IF, but only 31, so hardly worth going asymmetric).

The dimensions at lower left lead to a PCD 18 for M3 screws to hold down the HS H/C LVDT block. Also the maximum circle not exceeding the full through in 37.6 mm.

From the dimensions on next page, the optimum dimensions for the O-ring groove are 330 x 26 ID, which will suit a 0-121 O-ring (2.6 mm section), 26.65 ID, 31.89 OD.

This will give plenty of meat over the M3 screws on PCD 18, 12.25 mm for 4 mm over the 0.008 of the screw. So screws on PCD 18 and O-ring groove ID 33 and 0.126 ID 26

(see over page also)
φ26 to give 3.5 width to O-ring groove.
φ33 for 0.5mm wall 0-ring.
φ34 to give clearance between LD70 and φ55.

φ19.5

φ55

φ61

φ20.5

φ37.5 cleaned through hole
φ38.5 O-ring groove, width 3.5
Extension & tension piston fixing:

Diagram showing inner and outer dimensions of the piston fixing.

Mathematical calculations:

\[ \theta = \arcsin \left( \frac{225}{5} \right) = 26.7^\circ \]

To we can make \( \alpha = 53.5^\circ \)

2000N on 5x6 area x2

= 50 MPa stress

probably OK on ready heat-treated steel with YS yield of ~800 MPa but it is pushing to limit.
SPECIMEN ASSEMBLY ARRAYS FOR ORLEANS.

From Foot & Ashley p.60, the flow stress for pure iron is around 600-2000 MPa, shear stress 1000-4000 MPa, and normal stress 200-1000 MPa. Normal stress in a jacket of 0.15 in. and wall thickness 0.25 in. (cross-section 2 in.²) is around 3000 to 4000 N, or 1 to 4 kN.

This would be comparable to the strongest specimen, totally swamping the weaker specimen strengths of the Orleans specimen.

Foot & Ashley do not list strengths for gold, but for Ag the normal shear stress at 900°C is around 3-7 MPa at 10^-7 to 10^-4 strain rate, ie axial force on 1.5 in. jacket is around 30-80 N. Gold is probably comparable, so jacketing with gold would require a jacket cross-section of order 50 N, which may be tolerable.

Another jacketing possibility is weak glass, and there is the further possibility of using unjacketed specimens.

So there is probably not much point in designing a specimen assembly with full-length iron jacket. The alternatives are:

1. Short Cu, Al etc. jacket (too much thermal conductivity & too expensive for full length)
2. Iron jacket with welded insert of weak material
3. No jacket.

So these three options cover the specimen itself & its immediate locality.

The next two issues concern the upper end of the specimen assembly and the lower end.
minimum thread M6 x 0.82
could be M4 x 0.65 or Inconel

If 463 piston were to be used, may need a dimple in the end to avoid cracking at the M6 hole.

Increases bearing stress on φ15 piston about 20%.
Top end of spec assembly:

The leading piston could be attached to the steel $\#30$ piston as it left using the O-ring retainers to center the piston. A central peg could also be used, possibly threaded for positive attachment (but minimum threaded for PSZ in 86).

This piston could be PSZ, Inconel, Al+Z, etc. It could have an end for a mechanical seal or a plain end for unjacketed specimens.

Possibility that PSZ pistons will deintegrate with oxygen loss, but maybe this will not happen until above 1200 K?

Some destruction occurred in the split inner sealing sleeves. The temperature may have been well above 1200 K at the ZrO$_2$ probably in the tetragonal phase.
Bottom end of specimen assembly:

There are two possibilities:

1. No venting or pore fluid to access from the bottom. In this case, there can be a short close-off which contacts an anvil set up on the load cell suitably for compression only. Alternatively, the lower piston can be centred down to the top of the load cell with a bypass connection for extension as well as compression.

2. With venting of the bottom piston, allowing compression, extension, and torsion. There is some compromise due to the force supported by the vent tube itself.

Force on spring is:

\[ F = \frac{Gd^4 s}{8 D^3} \]

Where \( s \) is the deflection per coil.

For 3 coils & total deflection 1 mm, \( s = 0.33 \text{ mm} \)

\[ F = \frac{80,000 \cdot 0.33}{8 \cdot 25^3} = 3300 \frac{d^4}{D^3} \]

With \( d = 16 \), \( D = 25 \) when \( F = 1.4 \) Newton. So this method would give a pre-load for a load cell of 1N or so but the connection to the measured load after zeroing would be 1 part in 1000 if the full scale deflection were 1000N. This applies to extension only; in compression, this preload will be applied to the specimen itself or not detected by the load cell.

The alternatives of venting may, as in normal LVDT load cells, be a non-vented arrangement as shown on next page.
If this anvil piece (either) is to be held down by screws, the strength of the screws is as follows:

<table>
<thead>
<tr>
<th>Screw size</th>
<th>Min. area</th>
<th>Stress (MPa)</th>
<th>Strength of 8 screws</th>
</tr>
</thead>
<tbody>
<tr>
<td>M8</td>
<td>3.24 mm²</td>
<td>820 N</td>
<td>6.6 kN</td>
</tr>
<tr>
<td>M10</td>
<td>4.14 mm²</td>
<td>1350 N</td>
<td>10.8 kN</td>
</tr>
<tr>
<td>M12</td>
<td>4.92 mm²</td>
<td>1900 N</td>
<td>15.2 kN</td>
</tr>
</tbody>
</table>

So a set of 6-8 M12 screws should be sufficient for up to 2 kN extensible force.

Advantages of separate top piece:
1. can add torsion relief
2. can have an access from the bottom for proton yacht, 
   port holes, bottom thermocouple
3. can have alternative anvil arrangements

Disadvantages:
1. Extra fixing screws
2. Have to incorporate torsion initially if likely to be required
3. Cannot easily have cutting etc.
Notes on Jacket Sealing using push-on rings.

The following are notes made on 11/6/76 (Lab Book p203)

K165 TC pistons 9 rings — some problems with ring tightness, welding probably by 1100°C

Al63 pistons with T217 rings OK up to 1000°C, then tend to weld. Some considerations about supporting Al63 rings.

There were also problems with blow outs with unjacketed Al63 pistons.

The following dimensions are given in Apr. 1976 drawing:

![Diagram]

Coefficients of expansion:

- High-speed tool steel (X12CrNiMo17-12-1): 12.10⁻⁶
- Ti alloys: 9.10⁻⁶
- Mo: 5.10⁻⁶
- Inconel 625: 13.10⁻⁶
- Al63: 7.10⁻⁶
- PSZ: 10.10⁻⁶

Best high temp Ti alloy (TIMETAL 1100, 6.6Al, 2.7Zr, 4.2Mn, 0.47Fe, 0.05Fe) is said to be OK to 600°C (873 K).

Use of iron/copper or iron/gold overlapping jacket:

- Cu-Fe has a minimum about 20 K or so below Au 1083 (1336 K, 1063°C)
- Cu-Fe does not have a min below the Cu 1083 (1357 K, 1180°C)
From Frost & Ashby, the creep stresses \( \tau \) (shear) at around 1000°C (1273 K) at 10^{-4} and 10^{-5} strain rates are as follows (based on RT shear modulus, \( T_0 \)):

<table>
<thead>
<tr>
<th>Material</th>
<th>( \tau ) (MPa) at 10^{-4} ( s^{-1} )</th>
<th>( \tau ) (MPa) at 10^{-5} ( s^{-1} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fe</td>
<td>190</td>
<td>70</td>
</tr>
<tr>
<td>Cu</td>
<td>4</td>
<td>1.4</td>
</tr>
</tbody>
</table>

Thus, the load suggested by an iron jacket above 1200K is of the order of 1kN, while that suggested by a copper jacket is of the order of tens of Newtons. The latter is still high compared with the 1-10 N desired by Bolew, but not too bad at 10-100 N range.
Reconsideration of loads & deflections for HS LLC.

From p248, we have \( 2F = \frac{2EBst^3}{3L^2} \) for the double beam and \( 2F = \frac{5b}{3L} \).

\[
\text{whence } \frac{St}{3E} = \frac{5b}{3L} \text{ regardless of } b! \]

With \( \sigma = 500 \text{ MPa}, E = 210,000 \text{ MPa}, \]
\[
l = 27 \text{ mm}, \quad b = 36 \text{ mm}, \quad S = \text{ in mm}
\]

we have \( 2F = \frac{2 \times 500 \times 36 \times E^2}{3 \times 27^2} \) or \( 2F = 444 \text{ N}^2 \).

\[
\text{and } \frac{St}{3E} = \frac{500 \times 27}{3 \times 210000} \Rightarrow S = 0.579 \text{ in mm}
\]

<table>
<thead>
<tr>
<th>( t/\text{mm} )</th>
<th>( 2F/\text{N} )</th>
<th>( S/\text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.30</td>
<td>40</td>
<td>1.93</td>
</tr>
<tr>
<td>0.56</td>
<td>111</td>
<td>1.16</td>
</tr>
<tr>
<td>0.70</td>
<td>218</td>
<td>1.03</td>
</tr>
<tr>
<td>1.00</td>
<td>444</td>
<td>0.58</td>
</tr>
<tr>
<td>1.20</td>
<td>640</td>
<td>0.48</td>
</tr>
<tr>
<td>1.50</td>
<td>1000</td>
<td>0.39</td>
</tr>
<tr>
<td>2.00</td>
<td>1778</td>
<td>0.29</td>
</tr>
<tr>
<td>2.60</td>
<td>3001</td>
<td>0.22</td>
</tr>
</tbody>
</table>

With a maximum deflection of 1 mm and a thickness of 0.5 mm,
\[
2F = \frac{2EBst^3}{3L^2} = 768 \times 36 = 96 \text{ N}.
\]

Thus at \( t = 0.5 \text{ mm} \), we have a max force of approx 100 N at a maximum deflection of 1 mm.
MULTI-RANGE LVDT INTERNAL LOAD CELL

The following considerations have arisen in connection with 1kC's:

1. Orleans are interested in a high sensitivity load cell, for which preliminary considerations have already been made, from 0.235 upwards.

2. It would be useful to have easily-interchangeable load cell "capsules" that could be mounted on a basic load cell support incorporating the shears and the electrical feed-through, common to all load cells.

3. The LVDT 1kC concept lends itself to multi-ranging with each "capsule".

4. Experience in Himequlens reveals that the method of attaching the LVDT unit to the elastic element can give rise to hysteresis due to local distortions at the attachment.

5. It would be desirable to get away from the mechanical amplification arrangement in the most recent LVDT load cell by using a more compliant elastic element. This step has the disadvantage of "softening" the weigh measurement system, but if it still has a compliance comparable to that of the specimen assembly, this could be acceptable. The machine is not so much used for brittle fracture studies and, even then, servo control may be sufficient to compensate for machine softness.

6. So it would be desirable to integrate the high sensitivity requirement with the modified high load requirements, with a new rationalization, as in the following page.
General concept:
1. Separate the elastic element and its measurement from the spline and feed-through base, so that it can be readily exchanged.
2. Use flexure-shear type elastic element to give somewhat more compliance and so a greater scope for multi-ranging.
3. Have elastic element/LVDT measuring assembly in one piece to avoid screw connections.
4. Keep the same venting etc. arrangements and the same overall (push-pull-torsion) arrangements.
ANVIL DESIGN

We now fit the anvil assembly inside the position LVDT b. Sticking to LVDT diameter = 9.5
this has to be accommodated inside the top-bottom flag
spline ID of 55, so the LVDT b can be put on
PCL 45.

Thus the support ring A (opposite) can have threads
with a maximum diameter of 45-9.5 = 35.5—say
M35 x 2
(not conventional, but M35 x 2 is too small
and M36 x 2 is too big).

The minor diameter of the M35 x 2 will be
or nearly 33. The OD of the dog-teeth could then be
38.5. The ID of the dog-teeth is fixed by the 426
of the load cell body to accommodate the venting
assembly, etc.

Loading on dog-teeth: projected area of OD 32.5, ID 26
is 276 mm², so with an axial load of 100 kN the
bearing stress is 32.85 MPa. This should be OK
if we run OFFER UPPER at HRC 46 (0.2% YS = 1300 MPa)
In torsion, with mean diameter of 29 mm and shear stress
we have torque T = T. 248. 0.0145 or T = 0.278 M,
that is, at 1000 Nm, T = 278 MPa
and 500 T = 139 MPa.

This is adequate.

Loading on anvil should be L 2. The dog-teeth have a
wavelength of π/26/16 = 5.1 mm on the 26 φ, so the
height of the dog-teeth is 2.5 mm, or the mean height
of root area is 1.3 mm. Therefore L 2 = 1.3 means to
support 100 kN. If we allow 300 MPa shear stress T,
then π/26. (L 2 = 1.3). 300 = 100 000

or L 2 = 4.1

Thus if we make L 2 = 7, probably OK (T = 215 MPa).
L 2 = 8. T = 183 MPa

Loading on thread. The threads only have to support elastic
load, a maximum of 500 MPa on φ15 = 88.4 kN
so max stress in threads = 88.4/400 = 879

π/32 L 1
147 ... L 4 = 6
176 ... L 4 = 5

So optimum L 4 = 6.
Elastic Element in Compression / Tension

After considering many configurations, it now seems to me that, at least for high loads, a bellows-like config. would be the best compromise, giving maximum deflection for given load. It would be desirable to have, say 0.5 ± 0.4 mm deflection at 180 kN. The specimen column is assumed 200 mm effective length of 15 mm φ, so if steel, the deflection at 100 kN would be

\[
\frac{200}{100,000} = 0.57 \text{ mm},
\]

so there is no point in having the load cell much stiffer than this.

The basic element of the elastic element is then:

If we treat it as a thin plate of thickness \( t \), we can apply the formulae given in Marks, p 448:

\[
\sigma = \frac{K F t}{(E t^3)} \quad \text{deflect } S = \frac{M F t}{(4 E t^4)}.
\]

We have \( \frac{R}{D} = \frac{25}{26} = 2.1 \), from which the chart gives

\( K_e = 0.15 \quad M_e = 0.7 \) for freely supported at one edge, free at inner
\( K_b = 0.4 \quad M_b = 0.02 \) for built-in inner edge, non-rotating one edge.

Our case is probably somewhere in between, say \( K = 1, \, M = 0.3 \)

Then \( \sigma = \frac{22,500}{(E t^3)} \) or \( t = \sqrt[3]{\frac{22,500}{\sigma}} \)

For \( \sigma = 500 \text{ MPa} = 72,500 \text{ psi} \), \( t = \sqrt[3]{\frac{22,500}{500 \text{ psi}}} \)

\[
= 0.56 \text{ in} = 14 \text{ mm}
\]

and deflection \( S = \frac{0.3 \times 22,500 \times (2.17)^2}{4 \times 30,000 \times (0.56)^3} \)

\[
= 0.0015'' = 0.038 \text{ mm}
\]

The formulae are said to be only valid for \( t \leq 0.15 (\beta - \varphi) \)

ie \( t \leq 0.15 (55 - 26) = 44 \text{ mm} \). Thus with \( t = 14 \text{ mm} \), the
The calculation is well outside the validity range.

The effective D & d are also likely to be more like 52 and 29, giving $R/T = 1.8$. This could reduce $t$ to ~10 mm and $s$ to around 0.030 mm.

If $x$ in the figure on previous page in the inner diameter is 3 mm, the supporting area is $32 \times 2.6 \times 26 = 278 \text{ mm}^2$. For mean stress under 100 kN load of 366 MPa. For $x = 2$, the mean stress here becomes 568 MPa. For $x = 2.5$, the mean stress is 447 MPa. There will be a stress concentration of at least 2x at the radius and some additional stress due to bending effect in the supporting area, so we need $x$ to be at least 25 mm and maybe 3 mm.

At the outer diameter, the supporting area is $25 \times 49 \times 49$ for $x = 3$, giving a mean stress of 804 MPa; for $x = 2$, we have $560 \times 51 \times 51$ mean stress 300 MPa. So 2 mm would be OK for $x$ at the outer.

In the case of $x = 3$ inside & 2 outside, we have for plate formula $D = 51$ and $d = 32$, $12 \times D/d = 1.6$

and so for pinned ends $K_d = 0.3$, $M_e = 0.02$

free

$t = 0.15 (51 - 32) = 2.85 \text{ mm}$

For fixed ends

$t = \sqrt{0.3 \times 22500 \times 0.305 \times 175} = 7.75 \text{ mm}$

$s = 0.02 \times 22500 \times (51/25)^2 \times 0.305^2 = 0.0075 \text{ mm}$

For free ends, $t = \sqrt{0.3 \times 22500 \times 0.635 \times 16.1} = 16.1 \text{ mm}$

$s = 0.57 \times 22500 \times (51/25)^2 \times 0.635^2 = 0.0017 \text{ mm}$

Since 0.15 (51 - 32) = 2.85 mm, this calculation is again far outside the validity range.

If we neglect bending altogether & consider only shear, the supporting area is least at the (1D = 32), where for thickness $t$ of

the area is $\pi \times 32 \times t$, $9 \times$ shear stress = $\frac{100000}{\pi \times 32} = 300 \text{ MPa}$, say

$giving \ t = \frac{100000}{\pi \times 32 \times 300} = 3.3 \text{ mm}$
The actual situation is presumably somewhere between the shear only and the free-end bending cases, i.e. it between 3.3 and 16.1 mm. A geometric mean is 7.3 mm.

Two possibilities to try:

LATERAL FLEXIBILITY.

In seeking greater compliance in the load cell in order to a greater range in a multi-range load cell, we are shortening rigidity in bending, compared say to the simple cylindrical elastic element used so far. Does this matter? The load cell is still stiff with regard to shearing normal to its axis but will give less constraint against buckling or shearing off on an inclined plane. However, previous experience shows that buckling tends to occur with direction constraint on the end of the specimen assembly, so shearing loss of constraint may not matter much. The shears measurement becomes relatively meaningless when buckling or shearing occurs. If so no real loss is incurred there. Specimen alignment is the critical factor.
Torsion

In order to gain more torsional compliance in the 26 ID 55 OD seed section, we can split it lengthwise;

E.g. split it into 24 segments of 15° each,

which will be 7.2 mm wide at the OD
and 3.4 mm wide at the ID.

The effective width as a slab is then around 6 mm.

From the formulae opposite p 257,

\[ \tau_{\text{max}} = \frac{3 \times 300 \text{ MPa}}{\left(\frac{62}{12}\right) \times 0.035} = 120 \text{ MPa} \]

12 double segments, 500 Nm = 500 000 Nmm

Then \( G = \frac{\tau_{\text{max}} l}{c} = \frac{120 \times 40}{6.8 \times 10^3} = 0.0100 \text{ rad} \)

For \( l = 40 \text{ mm} \),

For an LVDT at 20 mm radius,
displacement \( \delta = 20 \times 0 \times 0.0100 \)

= 0.2 mm.

For a thicker plate the \( \tau_{\text{max}} \) is a bit higher

than in the above formula, but \( \delta \) is much the same.

Since the change in angle within the 38 mm long LVDT

will also be 0.01 radians, the lateral movement of the

LVDT core will also be around 0.2 mm. as much as could be accommodated comfortably (see p 271).

With 16 segments of 22.5° each, the widths will be 10.85 mm, effectively around 8 mm and \( \tau_{\text{max}} = 101 \text{ MPa} \),

\[ \delta = \frac{191.40}{8.8 \times 10^3} = 0.0013 \text{ mm} \]

\( \delta = 0.13 \text{ mm} \)

Because of the grooving for axial compliance, this may be as much as is needed, since the above formulae will underestimate the torsional compliance.

But it needs to be in 30° segments to fit around the LVDTs. These will weaken the segments substantially, so there may be only the equivalent of about 4 double segments.
TORSION REACTION

The reacted area under the load cell is 26 mm diameter.
From P & O for the rigid-plastic case we have

\[ M = \frac{\pi d^3 \tau_y}{12} \]

where now \( \tau_y = \mu_p \), i.e.

\[ M = \frac{\pi d^3 \mu_p}{12} = 2.3 \, \text{Nm} \]

in Nm with \( \mu_p = 0.8 \)

i.e \( M = 230 \, \text{Nm} \) at \( \mu = 100 \, \text{MPa} \).

or \( 690 \, \text{Nm} \)

Note this is OK but rules out strong torsion below
100 MPa confining pressure.

If we put in 8 pins at PCD 55 of say \( \phi_3 \), then

8 pins with shear stress of \( 100 \, \text{MPa} \). The torque

required would be

\[ M = \frac{\pi d^3 \phi_3 \mu_p}{12} = \frac{\pi d^3 \cdot 8 \cdot 100}{12} \]

\[ = 154 \, \text{Nm} \]

If we allow \( \tau = 300 \, \text{MPa} \) (a high tensile steel pin), then

\[ M = 46.2 \, \text{Nm} \]

For 4 mm pin, these values would be 274 and 821 Nm, resp.

Conclusion: better put in some pins but \( \phi_3 \) high
tensile should be OK.

With 6 pins, \( M = \frac{\pi d^3 \phi_3 \mu_p}{12} = \frac{\pi d^3 \cdot 6 \cdot 50}{12} = 1.17 \, \text{T} \)

ie \( 117 \, \text{Nm} \) for \( \tau = 100 \, \text{MPa} \)

350 \( \therefore \) 300

583 \( \therefore \) 500

So with 6 pins could support \( \approx 300-500 \, \text{Nm} \) at gas pressure

500-750 \( \therefore \) 100 MPa.
LVDT's for Multispan 16 C

Lining to the sloping in the elastic element & the fact that the LVDT's have to be located in the free end, there will not be a continuous iron path, or at least not a very direct one & so we must need to provide an iron sleeve on the LVDT's. This should be as thin as possible so as not to require serious reduction in the number of windings.

So Eq PI 4506 needs to be modified:

1) Change OD 9.5 to OD 9.0/8.95 for the former  
2) Thicken 1 to draw a tube 9.5 OD x 9.0/9.1 ID to fit over the former.

Can be held in place by slight keening at the open end.

3) Change terminal arrow to φ1.0 on PCD 7.0 for pins  
   M16 C/S screws on PCD 7.0 or M2
   125° or 112.5°
   45°  95°  56.25°  5°

and put two M1.6 holes in flange of former as shown.

All this assumes staying with SS Former, which has the advantage of being able to use φ4 for the ID of the coil. If the test with secondaries wound over the top of the primaries works, then we can also eliminate the intermediate flanges.

For the torsion, we need a shorter LVDT, maybe overall 34 or less.
Sideways displacement of core of torsion LVDT:

Suppose the torsional displacement is 0.5 mm, then \( \alpha = \frac{0.5}{25.5} \approx 0.0196 \) radian.

\[
\beta = \arctan \left( \frac{22.5}{12} \right) = 61.93^\circ
\]

\[
\gamma = \frac{1}{2} (180^\circ - 112.3^\circ) = 89.4^\circ
\]

\[
\frac{x}{0.5} = \sin (\gamma - \beta) \quad \text{or} \quad x = 0.5 \sin 27.5^\circ = 0.23 \text{ mm}
\]

Thus we can only afford about 0.5 mm torsional displacement if the clearance between LVDT core and former is about 0.25 mm. When LVDT is 24 mm long it positioned at PCD 45.

Clearance over splines:

Since we cannot wirecut, the clearance over the splines on a limited length of the load cell, the clearance is best milled in with a rectangular slot. As the calculation opposite shows, the slot can be 8.0 mm wide and 3.0 mm deep. That is, 54.4° or better, 54.3 mm across the flats.

The biggest circle that can be recommended at 10.4 mm is therefore \( 27.2 - 22.5 = 4.7 \) to allow a 9.6°. Thus a \( \phi 9.5 \) hole will break through.

The biggest circle \( X \) that can be fitted in on PCD 45 & clipping the corners of the slots will be 11.5 φ.

However, we need to make the slots wider then 9.0 wide, with 54.0 across the flats to allow for twist. A re-calculation (top of page opposite) gives a maximum diameter of 10.86 mm. Thus we should limit the circle \( X \) to about 10.5 mm.
mV/V ranges on RS card are 0.5/1.5 mV/V to 250/350 mV/V
Load Range Possible

In the case of the Porteus/Minneapolis machines, the axial displacement of the elastic element was amplified mechanically about 4x, from about 0.380 mm to about 0.320 mm. The range setting on the RS card was 25/75 mV from memory.

In the multirange load cell, the maximum displacement at full load range needs to be limited to ±0.5 mm for reasons of accommodating both axial & torsional movements. Thus, the least sensitive range in the RS card that could be used would be about 50 mV/V. The most sensitive range is 0.5/1.5 mV/V, so there is the possibility of, at most, increasing the sensitivity 100X, or possibly only 50X. That is, for the normal range of 100 kN, the most sensitive range would be either 1 kN or 2 kN full scale.

Thus, the most sensitive range required on the high-sensitivity load cell insert would be 1 kN F/S. If this is also based on ±0.5 mm displacement, then the most sensitive range would be 10 N F/S.

In the case that a force of 0.2 N is to be measured (p. 245), this would be registered at about 2% full scale. If the sensitivity of measurement is 0.1%, i.e. 0.01 N on 10 N scale, then we have 5% sensitivity in measuring 0.2 N. This may be marginally acceptable. Otherwise, we need to go to a third load cell insert.
Torsion compliance (contd. from p. 268) Assume 610D 261D 

The effective width of segment is \( n = \frac{z}{12} \) m.

From Eqns. p. 251,

\[
\theta = \frac{3Ml}{bc^3} \quad \text{and} \quad \theta_{\text{max}} = \frac{3M}{bc^2}
\]

Put \( M = 500 \text{ Nm} \), \( b = 61-26 = 35 \)

\[
\frac{c^2}{l} = \frac{3M}{bc^2} \quad \text{Segment}
\]

Neglecting the three sectors containing the LVDT's of torsion 
support, there are about 4 effective double segments.

\[
\frac{c^2}{l} = \frac{(61-26) \times 200 \times 10^6}{4} \quad \text{or} \quad c = 0.0073 \text{ m} = 7.3 \text{ mm}
\]

For a shear stress of 200 MPa.

Then \( \theta_{\text{max}} = \frac{200 \times 40}{120000} = 0.0143 \text{ rad} \approx 0.8^\circ \)

This gives 0.29 mm at 20 mm radius.

If we wish to double this displacement (we need to halve \( c \) with 
the same shear stress) or double the shear stress with the same 
width.

\[
\text{For width which is 200 \times 10^6}{\text{N/m}^2} = \frac{200 \times 40}{120000} \text{ rad} = 0.8^\circ \]

\[
\text{For width which is 200 \times 10^6}{\text{N/m}^2} = \frac{200 \times 40}{120000} \text{ rad} = 0.8^\circ \]
Tool for Load Cell Cap, PL 4505-4

Concentrational area of each stud = 6.5 × 5, Σ Ad = 
four studs, \( A = 6.5 \times 5 \times 4 \) = 130 mm²

Lever arm = \( \frac{1}{2} \times \frac{61 + 48}{2} = 27.25 \)

So to apply a torque of 1800 Nm
(max allowable), shear stress \( \tau \) is given by

\( 180 \cdot 10^6 \cdot 0.02725 = 1000 \)

\( \tau = 282 \) MPa.

Requires HT steel, Assab 708.

If we make \( L = 400 \) mm,
then for 1000 Nm we need a

force \( F \) of \( 2F L = 1000 \), \( F = \frac{1000}{2L} = \frac{1000}{2 \times 4} = 1250 \) N

This is more than a person can reasonably apply, so it will serve as upper design limit.

Then bending moment (max) on the bar is 500 Nm

\( \sigma = \frac{M}{I} = \frac{500 \cdot D^3}{\pi D^4/64} = \frac{500 \cdot 32}{\pi D^3} \)

or \( D^3 = \frac{16000}{\pi} \)

For mild steel, \( \sigma \sim 300 \cdot 10^6 \), \( D^3 = \frac{16000}{\pi \cdot 300 \cdot 10^6} \)

\( D = 0.026, \sigma = 26 \) mm

For \( D = 20 \) mm, \( M = 300 \cdot 10^6 \cdot \pi (0.020)^3 = 236 \) Nm per side

\( \sigma = 472 \) Nm total.
For 30°, area of segment minus LDDT hole is ~6000 mm². Normal stress ~240 MPa at 189 kN.

5" sheath saw, pitch: 1/32", 3/64", 1/16".

Thickness: 0.8, 1.2, 1.6 mm.
Another possible elastic element configuration

From opp. p. 248, \( M = \frac{Fl}{2} \) and \( S = \frac{3Ft}{6t^2} \)

For \( 6 = 1 \) deflection \( S = \frac{Ft^3}{12EI} = \frac{Ft^3}{Elt^3} \)

Hence: \( F = \frac{6Et^2}{3l} \) and \( 8t = \frac{6Et^2}{3E} \).

For \( t = 500 \text{ MPa}, \varepsilon = 210,000 \text{ MPa} \), \( l = 30 \), \( t = 55 \)

\( F = 194 t^2 \quad 8t = 0.714 (S = 0.714) \)

<table>
<thead>
<tr>
<th>( t/\text{mm} )</th>
<th>( F/N )</th>
<th>( 8/\text{mm} ) for one beam</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>48</td>
<td>1.43</td>
</tr>
<tr>
<td>1</td>
<td>194</td>
<td>0.71</td>
</tr>
<tr>
<td>2</td>
<td>776</td>
<td>0.36</td>
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<tr>
<td>5</td>
<td>4850</td>
<td>0.143</td>
</tr>
<tr>
<td>8</td>
<td>12,416</td>
<td>0.089</td>
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<tr>
<td>10</td>
<td>19,400</td>
<td>0.076</td>
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<tr>
<td>13</td>
<td>43,650</td>
<td>0.048</td>
</tr>
<tr>
<td>17.5</td>
<td>77,600</td>
<td>0.036</td>
</tr>
<tr>
<td>20</td>
<td>121,350</td>
<td>0.029</td>
</tr>
</tbody>
</table>

cf. p. 261 for a flexural line constrained

For 100 kN, we would need to decrease \( t \) somewhat to get a better balance of bending stresses in bar & anchors so if \( l = 27 \), \( F \) becomes \( 77,600 \left( \frac{30}{27} \right) = 95,800 \text{ N} \) & \( 8 \) will be somewhat greater than the 0.036 mm, maybe almost double that, or comparable to the cylindrical elastic element.
To support 100,000 N at $\sigma = 315$ MPa, we need $317 \text{ mm}^2$, almost the area surrounding the holes. The shaded area, $\phi = 57 \text{ mm}$ is another $300 \text{ mm}^2$ if $10 \text{ mm}$ wide. So twice as much support area as we need.
ELASTIC ELEMENT CONFIGURATION NO 4

Effective beam has $l = 27, b = 28, t$ to be determined.

From Moment's Eq 26, $M_{max} = \frac{F*l}{4}$ pin jointed.

Any $M = \frac{F*6}{6} = \frac{F}{8}$ quad ends.

$s = \frac{F}{1050E} \times \frac{F*3}{192EI}$

Also we have $I = \frac{bt^3}{12}$ as $\sigma = \frac{Mn}{I} = \frac{M}{12} \times \frac{b*t^3}{2}$

With $F = \frac{F}{6}$,

Then $s = \frac{F}{bt^2}$.

With $F = 100000, l = 27, b = 28, \sigma = 500 \text{ MPa}$,

$t = 13.9 \text{ mm}$.

$s = \frac{F*3}{96EI} = \frac{F}{32*bt^3}$

$s = \frac{F*3}{96EI} = \frac{F*12}{96Ebt^3} = \frac{l^2}{8Et} = s$

So $s = 0.0156 \text{ mm}$ for one leaf of spring.

We can fit in almost three leaves, so total $s \approx 0.047 \text{ mm}$.

However, with the simple cylindrical elastic element at 315 MPa stress (0.015 strain), and 50 mm long 12, we get $s = 0.075 \text{ mm}$, which can be increased to 0.130 with 4x mechanical magnification. Thus, it is very questionable whether we can do better than the PI4500 load cell for 100 kN load range. It has great lateral stability & the problems associated with distortion from the dogteeth affecting hysteresis seem to be solved by tightening the cap on the dogteeth enough.
Area = \((54 - 26) \times 2\) \\
= 56 \text{ cm}^2
ELASTIC ELEMENT CONFIGURATION No. 5

If strain limited to 0.0015, i.e. $\varepsilon = 314 \text{ MPa}$, then

$$56t \cdot 314 = 100,000$$

so $t = 5.7$ mm, say 6 mm.

In torsion (p. 251)

$$M = \frac{T_{\text{max}} \cdot b c^2}{3}$$

$$= \frac{200 \cdot 28 \cdot 6^2}{3} = 67,200 \text{ Nmm}$$

Shear stress

or 200 Nm at 607 MPa shear stress

This config may be worth considering if no too large torsional load is envisaged.
area ~ 68 mm² by mm² counting

made up by extra shear deformation each end.

with 0.25 mm wire, cut is ~ 0.32 mm

shear area ~ 200 mm²

under each leg 10

\[ T = \frac{25,000}{200} = 125 \text{ kN} \]

see p. 264
ELASTIC ELEMENT CONFIGURATION No. 6, cm at left

The shaded areas are somewhere around 70 mm to 80 mm, far
75 mm × 4 = 300 mm² total.

At 100 kN, the mean compressive stress is therefore about
330 N/mm², corresponding to a strain of 0.0016, or a
displacement in a 40 mm length of 0.064 mm, 64 μm.

This compares with about 75 μm for the simple cylinder
element (5208 481D) used so far, so we lose about 12% signal,
so the signal will be ~ 88% of previous.

In torsion, with a mean radius of about 27 mm, we have a
shear stress \( \tau \) given by
\[ \tau = \frac{300 \text{ N/mm}^2 \times 0.027 = 1000} {T = 123 \text{ Nm}}, \]
or shear strain of 0.0015, shear displacement 0.062 mm at
27 mm or 0.051 mm at 22.5 mm (the location of CVDT).

There are some complications at each end. At the
bottom only two of the "pillars" bear directly on the
piston head; even then will only about half area,
the direct contact area is about 75 mm², giving a
contact stress of 1300 N/mm², around the yield stress of the
steel at 45 HRC. However, bending of the lower part will
bring further support and so the load will probably be fully
supported elastically, although the stress distribution
will be complex.

At the top end, the total area of the "dog-tooth" contact
is 224 mm², compy. to 337 N/mm² at 100 kN, and the shear
area over 6 x 7 mm depth is 582-583 mm², compy. to shear
stress 4200 N/mm². The load will be non-uniformly distributed
over these regions, at least doubling these stresses, but we
probably get away with the stress concentration, again.
Further Reconsideration of broad-light reflections for HS 16 C (Ecliptic Standard, No. 7).

From p. 246, 248, 261, we have for each half of the config. opposite, 

\[ F = \frac{2.6E^2}{3l} \quad \text{or} \quad 4F = \frac{4.6E^2}{3l} \]

\[ S = \frac{5.5}{3EL} \]

in terms of the maximum allowable stress \( \sigma \). As the total force on the load cell is \( 2F \) for a single plate and \( 4F \) for two plates. We now measure \( d \) around the arc, probably a better approximation.

Take \( E = 500 \, \text{MPa} \), \( E = 210,000 \, \text{MPa} \), \( l = 40 \), \( b = 15 \)

\[ 4F = 250 \, E^2 \quad \text{and} \quad S = \frac{1.27}{E} \]

<table>
<thead>
<tr>
<th>( t, \text{mm} )</th>
<th>( 4F, \text{N} )</th>
<th>( S, \text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>63</td>
<td>2.54</td>
</tr>
<tr>
<td>0.8</td>
<td>81/80</td>
<td>1.59</td>
</tr>
<tr>
<td>1.0</td>
<td>100</td>
<td>1.00</td>
</tr>
<tr>
<td>1.0</td>
<td>250</td>
<td>1.27</td>
</tr>
<tr>
<td>2.0</td>
<td>1000</td>
<td>0.64</td>
</tr>
<tr>
<td>3.5</td>
<td>3060</td>
<td>0.36</td>
</tr>
<tr>
<td>6.3</td>
<td>10,080</td>
<td>0.20</td>
</tr>
<tr>
<td>10</td>
<td>25,000</td>
<td>0.13</td>
</tr>
<tr>
<td>20</td>
<td>100,000</td>
<td>0.064</td>
</tr>
</tbody>
</table>

100 N range

3000 N range

Contd p. 298
Note re $\Theta = \frac{Tl}{CG}$: This $I$ is by membrane analogy, similar along most of the length $b$, if so is not affected by cutting out the piece $b_2$, although $I$ is reduced by the cutting.

Note re $M = \frac{2Tc^2(b_1 - b_2)}{3}$: It seems strange that the torque depends only on the width $(b_1 - b_2)$ and not on its absolute radius $b_1$, but this may again be reflecting the fact that the stress is the same along most of the length of $c$. The torque really is determined by the arms $c$, not by the stress (the shear stress in direction $b$).
High Sensitivity Torque Measurement

For the configuration B, we can calculate the torque \( M \) from configuration A minus C, multiplied by 2.

For A, \( p = 5 \),
\[
M_A = \frac{B_a b_c G}{3l}
\]

For B, \( M_B = \frac{B_b b_c G}{3l} \).

We put \( B_A = B_B = B \).

Then
\[
M = 2(M_A - M_B) = \frac{2B c^3 G (b_1 - b_2)}{3l}
\]

The displacement \( \delta \) at the end of a diameter is \( \frac{b \theta}{2} \) - (2)

But (p. 251) \( \theta = \frac{\tau l}{cG} \) or \( \tau = \frac{bG \delta}{c} \).

So (1) becomes
\[
M = \frac{2\tau c^2 (b_1 - b_2)}{3}
\]

(2) becomes
\[
\delta = \frac{\tau l}{2CG}
\]

Take \( \tau = 250 \text{ MPa} \), \( c = 80,000 \text{ MPa} \), \( l = 15 \), \( b = 45 \) \( \begin{cases} M = 0.1667c^2 (b_1 - b_2) \\ \delta = 1.0547 \text{ mm} \end{cases} \)

<table>
<thead>
<tr>
<th>( c )</th>
<th>( b_1 - b_2 = 6 )</th>
<th>( b_1 - b_2 = 10 )</th>
<th>( b_1 - b_2 = 35 )</th>
<th>( \delta )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>0.25</td>
<td>0.42</td>
<td>1.5</td>
<td>2.1</td>
</tr>
<tr>
<td>0.8</td>
<td>0.64</td>
<td>1.07</td>
<td>3.7</td>
<td>1.3</td>
</tr>
<tr>
<td>1.0</td>
<td>1.00</td>
<td>1.67</td>
<td>5.8</td>
<td>1.05</td>
</tr>
<tr>
<td>1.5</td>
<td>2.25</td>
<td>2.8</td>
<td>13.1</td>
<td>0.70</td>
</tr>
<tr>
<td>2</td>
<td>4.0</td>
<td>6.7</td>
<td>23</td>
<td>0.53</td>
</tr>
<tr>
<td>2.3</td>
<td>2.35</td>
<td>3.1</td>
<td>5.3</td>
<td>0.46</td>
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<tr>
<td>3</td>
<td>9</td>
<td>15</td>
<td>5.3</td>
<td>0.35</td>
</tr>
<tr>
<td>5</td>
<td>25</td>
<td>4.2</td>
<td>14.6</td>
<td>0.21</td>
</tr>
</tbody>
</table>

1.05 range

20N\( _u \) range
LVDT Design with Secondary wound over Primary

Tests have just been carried out with a new design of LVDT. This was wound by Stan-Delta on a plastic former made by Karl at 451.

![Diagram of LVDT design]

- **Primary**: 1000 turns
- **Secondary**: 4000 turns

Primary resistance: 103.2
Secondary:

<table>
<thead>
<tr>
<th>Core Length</th>
<th>Primary (mV)</th>
<th>Secondary (mV)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>7.56</td>
<td>76.5</td>
</tr>
<tr>
<td>18</td>
<td>8.19</td>
<td>81.1</td>
</tr>
<tr>
<td>16</td>
<td>8.57</td>
<td>84.3</td>
</tr>
<tr>
<td>14</td>
<td>8.38</td>
<td>90.2</td>
</tr>
<tr>
<td>12</td>
<td>8.20</td>
<td>80</td>
</tr>
</tbody>
</table>

Current = 14 mA with 20 mm core.

The output was linear (within 1% over 5 mm), so optimum core length is 16 ± 15 mm.

The windings were tapered down to the centre over a length of about 6 mm, as shown above. It may be better to have a divide of 1 mm, thereby saving 5 mm in length.

A problem is that, although the output is linear over the 250/750 range, there is a fluctuation assumed this linearity trend, suggesting an unevenness in density of windings. Although it shows up in LVDTs the 250/750 and 25/75 mV/V ranges.

On the AC output before amplification it is about 0.2 V/mil, i.e. about 22 mV/V/mm, much higher than for the commercial LVDTs.
LVDT Re-design

In the light of the above experience (p281) & the requirements of the new multirange load cell, the following factors are to be taken into account:

1. For a transversely located torsion LVDT, the maximum overall length at 9.5 mmφ that can be accommodated is 24-25 mm. The terminals could protrude a little further if at a smaller P.D.

2. The maximum linear range needed is probably ±1 mm, and we need maximum sensitivity to cope with smaller ranges.

3. In the load-measuring applications, the full 9.5 mmφ can be used, but for a displacement LVDT, a shear would be needed.

In the accident on p281, an optimum core length was about 16 mm, so at zero position there were 5 mm of windings beyond each end. With ±2.5 mm linear range, this leaves 2.5 mm free extra windings beyond the extreme position.

If we want ±1 mm linear range, there would be ±3.5 mm beyond.

The core length. With 5 mm taken up with the ends, a total length of 24-5 = 19 mm is available for windings but including the central division of 1 mm, group 9 + 1 + 9 = 19, or 9 mm of windings. If we take 19 mm total less 3.5 mm each end, this gives 12 mm as core length. However, if the core goes from centre to centre of the windings, this would suggest a core length of 10 mm and a linear range of ±2 mm. It seems worth making & determining the optimum core length for sensitivity & linear range.

If we go from 24 total length to 72 total length, the winding length goes from 9 to 33 & so an extra (33-9)/2 = 12 should be added to the linear range, giving ±14 mm linear range, probably OK for the displacement LVDTs, with core length 34.
Tightening torque for dog-tooth cap on MN 14 C
Marks p.275 gives a formula for tension \( F \) in a bolt due to tightening with torque \( M \):

\[
F = \frac{2 \pi M}{p + \mu \rho \cos \left( \arctan \frac{p}{2} \right) \sin \left( \arctan \frac{p}{2} \right) \sec \left( \frac{\pi}{4} \right) + d' \frac{3 \pi \rho}{2}}
\]

where:
- \( p \) = pitch
- \( \mu \) = coeff. friction on thread
- \( \mu' = \frac{\mu}{\sin \varphi} \) (under load)
- \( d \) = pitch dia of thread
- \( \rho \) = half-angle of thread form - take 30°

So with pitch = 2mm = 0.002, \( d = 55mm = 0.055 \), we have

\[
F = \frac{2 \pi M}{0.002 + 0.1 \times 0.002 \times 86.4 \times 1.155 + 0.1 \times 15 \pi \times 0.055}
\]

\( d \ F = 131 \ M \) Newtons

Thus with \( M = 100 \ Nm \), \( F = 13 \ kN \).
Review of Incompressible Oil Piston Design

The question is whether the clearance at the seal opens up significantly under pressure.

Increase in bore

$$\frac{A_d}{d} = \frac{\frac{(D^2)}{d} + 1}{E}$$

$$D = 290$$
$$d = 130$$

$$\frac{A_d}{d} = \frac{700000}{210000} \left( \frac{2.46 + 0.29}{2.46} \right) = 0.00092$$

or $$A_d = 0.119 \text{ mm}$$

say $$0.12 \text{ mm}$$.

So radial expansion = 0.06 mm

Poisson expansion of the piston is

$$\frac{A_d}{d} = \frac{\nu P}{E} = \frac{0.29 \times 700000}{210000} = 0.00097$$

$$A_d = 0.0126$$

or radial expansion = 0.0063 mm.

Thus, the clearance over the seal will tend to open with pressure. Actually, due to the absence of pressure behind the seal, the increase in diameter resulting will be effectively halved to 0.03 mm, but if the piston goes off centre, this could still be 0.06 mm.

Further, this clearance should be compensated for by the nitre ring expanding.

So there should be no serious case for undercutting the seal region as shown opposite.
Bottom Electrical Connections for Load Cell.

There has always been a rather unsatisfactory arrangement for getting the 14 connections out of the compensating piston. Having a floating connector on the end of leads attached to a tube from the load cell is very unsatisfactory, vulnerable to damage, and lacks an easy way for lead detection.

I tried last year to develop a system (Dox PI 4501E) using bore-in-tube connectors but there was not enough space in the $\phi 10$ bore of the pistons to make this work. However, if we use a $\phi 12$ bore in the pistons, there is a better chance.

With the $\phi 12$ bore and an M16x1.5 connecting piece, the bearing stress is $\sigma = \frac{28}{15} = 1485 \text{ MPa}$ or $742 \text{ MPa}$ for $P = 500 \text{ MPa}$, $1070 \text{ MPa}$ for $P = 720 \text{ MPa}$.

At HRC 55, the yield stress of the steel in the pistons will be of order 1600 MPa, so there should be no problem with bearing stress.

Similarly, the max circumferential compression in the M30 piston is $\frac{30 + 12}{30 - 12} = \frac{1.38P}{P} = 640 \text{ MPa}$ for $P = 500 \text{ MPa}$ and $994 \text{ MPa}$ for $P = 720 \text{ MPa}$.

This is well in the elastic range so it allows for some complex stress state effects.

If we use an M16x1.5 thread on the connecting piece (same thread as for Nova/Swiss HP scales), then $f = 14.376$. So there is $(14.376 - 12)/2 = 1.19 \text{ mm of "meat" at the bottom of the threads.}$. This should give a strength of at least 10 kN and probably much more (theoretically 30 kN at 800 MPa).

Previous experience indicates that the connector from a dense 3B·312 can be bored out to 6.4 (to clean 1/4" HP tubing, $\phi 6.35$) and still not expose the metal part.
In order to position the "bracing" (or use a split nut), the hole needs to be larger, which decreases the field size. Minor of M10 x 0.75 is 9/16.

If tube OD = 7.5, then ID = 7.5 + 0.14 = 7.64. About 0.75" mean without flange.

OK.

Lens insert
EGG 3B 3122L

These parts are on PCD 8-2
However, it is very close & it may be necessary to bore out further & then back-fill with epoxy. But this is difficult since the metal parts tend to form under the tool & so should be avoided. In an extreme situation, it may be necessary to reduce the tube to say 6.0 or 5.5 mm & cut an epoxy layer on it to insulate it. This would degrade the strength of the pipe by

\[
\frac{\frac{6.35^2 - 1.6^2}{6.0^2 - 1.6^2}}{\frac{6.35^2 - 1.6^2}{6.5^2 - 1.6^2}} = 1.13 \text{ times}
\]

or

\[
\frac{\frac{6.35^2 - 1.6^2}{5.5^2 - 1.6^2}}{\frac{6.35^2 - 1.6^2}{6.5^2 - 1.6^2}} = 1.36 \text{ times}
\]

so the 700 MPa rating would be reduced to 620 or 513 MPa, respectively — still okay for harm pressure of 590 MPa max. The coating needs only its area, the region under the metal parts.

A possible configuration is as at left, which is a development from the design in Fig 61 45-06(A). The same plug arrangement as on Fig 6507, using a nerve insert FG 35 312 24A, could be used. This could be adapted for attaching to the 1722 nut by means 17 screws.

According to British 150 (BS 3692: 1967) standard for nuts, an M22 nut is 32 AF & 36.9\% average, thickness 18 (Mack. Helix p 1342)

If the 17 screws are on PCD 31, then the centre is 2.6 from the face — OK. The diameter inside the screw head (0.55) is then 25.5 whereas the body supporting the nerve plug is 0.28.

If we put the 17 screws on PCD 32, then their centres will be 2.1 mm from the face, i.e. only 0.6 mm of metal over the 17 screw thread or even less (0.4 if the nut is at min 31.4 AF). Better to stick with PCD 31 and accommodate the heads with flats.
Internal Load Cell - Elastic element config no 6

PCD of displacement LVDT:

\[ x = 22.5^2 + 27.23^2 - 2 \times 22.5 \times 27.23 \cos 7^\circ \]

\[ 2x = 26.35 \]

So clearance \( y = 27.5 - 26.35 \) = 1.15 mm.

Alternatively, \( z = 22.5^2 + 27.5^2 - 2 \times 22.5 \times 27.5 \cos 7^\circ \)

\[ z = 5.97 \text{ mm} \]

So, still essentially 1 mm clearance.

So PCD = 45 for the displacement LVDT is OK.

Actually, not an arc but a flat (54 AF)

Now \( z = 22.5^2 + 27.23^2 - 2 \times 22.5 \times 27.23 \cos 7^\circ \)

\[ z = 5.73 \text{ mm} \] instead of 5.97 — still OK but less margin. This applies to the ILCD body, the actual 5.97 applies to the ILCD base.
33 = minimum length of clearance required on load cell.
32 = length of load cell base.
Load cell in bottom position.
Pins to support torque

When the pressure is relatively low and a torque bar is used to apply torque, the reaction has to be taken on some pins. If we have 6 pins of φ3 on PCD 49 and we allow 308 N/m in shear on each pin, then the torque

\[ M = \frac{\pi \times 3^2 \times 6 \times 200 \times 24.5}{1000} = 208 \text{ Nm} \]

**Torque at pressure:** The area of contact with normal pressure across it is φ26. For rigid plastic case (Pr.0 p.1344)

\[ M = \frac{\pi}{12} d^3 \tau_y = \frac{\pi}{12} d^3 \mu P \]

\[ = \frac{\pi}{12} (0.026)^3 \times 0.5 \times 300 \times 10^6 \text{ at } 300 \text{ MPa} \]

\[ = 690 \text{ Nm} \]

to which we can add the above 208 Nm, giving a total of 900 Nm. At 100 MPa, the total is \( \approx 430 \text{ Nm} \), still quite useful.

**Length of spline**

The sketch opposite. If we make the cut-outs that clear the splines 35 mm long, there will be adequate clearance when the load cell is in bottom position.

**Wire cut**

Typical 0.25 mm wire would give a cut of \( \approx 0.32 \text{ mm} \)

Furthermore 0.2 \( \approx 0.27 \text{ mm} \)

So torsional displacement of \( \approx 0.07 \text{ mm} \) (p.278) would be adequately accommodated within a single cut.
Water in ALTI furnaces

Both ETH and Darmouth have recently had refurbished furnaces of ALTI construction that have produced a "lot" of water and had the hot spot move up high. No such problem has arisen with a similarly designed furnace in Minneapolis. Differences may be:

1) Minneapolis dry the furnace with a vacuum oven
2) "furnace may have less "rigidity" used on the fiber ceramic (not sure of this) & may be SAE was used instead of AL30.

The volume of fiber ceramic is 6103 x 2512 x 1082 L = 0.185 L

At 450 K, this would accommodate

$\frac{300 \times 0.15}{400} = 0.18$ L water = 0.09 g water

of less than 1% of liquid water. Only a fraction of this, although not a substantial fraction, would be expelled on heating the furnaces to ~ 1200 K. Suppose 0.05 L was expelled in this way = 50 cm$^3$

Total area of steel is of order of $\pi \times 310.65$ cm$^2$, total area and bond cell is of order of 100 000 cm$^2$. Therefore thickness of water film would be of order $\frac{\pi}{50 000} = 0.0005$ mm = 0.5 micrometer.

This is 0.0005 cm$^2$ water per cm$^2$ or around 0.5 $10^{-6}$ g water per cm$^2$.

This would produce a layer of water of perhaps up to ten times the volume of water, i.e. of order of a few micrometers of rust.
Infrared load cell - elastic element No 6

On p 278, an attempt was made, on the basis of counting mm squares on a drawing, to estimate the support area in the elastic element. We now try to do an analytical estimate first without the cut-outs for the spline clearance. Let $x =$ distance of wire-cut from centre.

$\alpha =$ angle subtended by the cut (chord) at the centre.

Then

$$
\text{area } A = \frac{1}{2} \pi r^2 (\alpha - \sin \alpha)
$$

$$
\text{area } B = \frac{1}{2} \pi r^2 - \left\{ \frac{1}{2} \left( \alpha - \sin \alpha \right) \right\} = \frac{r^2}{2} \left( \pi - \alpha + \sin \alpha \right)
$$

$$
\text{area } C = \frac{r^2}{4} \left( \pi - \alpha + \sin \alpha \right) - \pi r^2
$$

$$
\text{area } D = \frac{1}{2} \text{area } A - \text{area } C
$$

$$
= \frac{r^2}{4} \left( \alpha - \sin \alpha \right) - \frac{r^2}{4} \left( \pi - \left( \alpha - \sin \alpha \right) \right) + \pi r^2
$$

$$
= \frac{r^2}{2} \left( \alpha - \sin \alpha \right) - \frac{\pi r^2}{4} + \pi r^2
$$

But $\alpha = 2 \arccos \frac{x}{r}$, so

$$
\text{area } D = \frac{r^2}{2} \left( 2 \arccos \frac{x}{r} - \sin \left( 2 \arccos \frac{x}{r} \right) \right) - \frac{\pi r^2}{4} + \pi r^2
$$

For $x = 30.25 \text{ mm}$,

$$
\text{area } D = \frac{30.25^2}{2 \times 457.33} \left( 2 \arccos \frac{30.25}{57.296} - \sin \left( 2 \arccos \frac{30.25}{57.296} \right) \right) + \pi r^2
$$

For $x = 13$, area $D = 126.11$

<table>
<thead>
<tr>
<th>$x$</th>
<th>$\text{area } D$</th>
<th>$4D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.5</td>
<td>112.17</td>
<td>448.7</td>
</tr>
<tr>
<td>14</td>
<td>98.98</td>
<td>395.9</td>
</tr>
<tr>
<td>14.5</td>
<td>86.54</td>
<td>346.2</td>
</tr>
<tr>
<td>15</td>
<td>74.88</td>
<td>299.5</td>
</tr>
</tbody>
</table>

At $2x = 28 \text{ mm}$, area $4D = 396 \text{ mm}^2$, strain for 100 kN = 253 MPa, strain = 0.00120.
\[ a^2 = 22.5^2 + 27.542^2 - 2 \cdot 22.5 \cdot 27.542 \cdot \cos 6.662^\circ \]
\[ a = 5.86 \text{ mm} \]
Area cut out for sheath pass: area $E +$ area $F$

Area $E = (4 + y)^2$ 

ignoring curvature

$\frac{y}{\cos 30^0} = \frac{2}{3} \frac{z}{y}$

But $y = \frac{27.25}{\sin 30^0} - x = 13.625 - x$

$\therefore y = \frac{2}{3} (13.625 - x) = 15.733 - \frac{2}{3} x$

So area $E = (19.733 - \frac{2x}{15})^2 = 59.198 - 2\sqrt{3} x$

Area $F = \frac{3}{2\sqrt{3}} = 2.598$

$\therefore$ area $E +$ area $F = 61.797 - 2\sqrt{3} x$. for other side

For $x = 13$

<table>
<thead>
<tr>
<th>$y$</th>
<th>$E$</th>
<th>$F$</th>
<th>$E+F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.5</td>
<td>15.03</td>
<td>1.299</td>
<td>16.329</td>
</tr>
<tr>
<td>14</td>
<td>13.29</td>
<td>1.067</td>
<td>14.356</td>
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<tr>
<td>14.5</td>
<td>11.57</td>
<td>0.835</td>
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</tr>
<tr>
<td>15</td>
<td>9.835</td>
<td></td>
<td>9.835</td>
</tr>
</tbody>
</table>

So total tentative remaining area = area $D - (\text{area } E + F)$

For $x = 13$

<table>
<thead>
<tr>
<th>$y$</th>
<th>$E$</th>
<th>$F$</th>
<th>$E+F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.5</td>
<td>82.11</td>
<td>97.91</td>
<td>179.02</td>
</tr>
<tr>
<td>14</td>
<td>72.38</td>
<td>85.68</td>
<td>158.06</td>
</tr>
<tr>
<td>14.5</td>
<td>63.41</td>
<td>74.97</td>
<td>138.38</td>
</tr>
<tr>
<td>15</td>
<td>55.2</td>
<td>65.05</td>
<td>120.25</td>
</tr>
</tbody>
</table>

$2x = 27$ leads to $10000 / 328.4 = 305$ MPa, strain = $0.0015$ / 0.0015

$2x = 28$ leads to $345$ MPa, strain = $0.00164$

A wire cut at $28\,\text{mm}$ would give $2x = 27.7$, so this should be OK (area ~ 301 mm$^2$, 332 MPa, strain = 0.0015).

Displacement = $0.00158 \times 36 = 0.057\,\text{mm}$ plus a lot of distortion, i.e. $0.060\,\text{mm}$
1000 primary returns requested in each case.

Impedance = \( 5/0.068 = 73.5 \ \Omega \) so inductance = \( \sqrt{73.5^2 \ - \ 28.5^2} = 67.8 \ \Omega \).

Inductance \( \propto N^2 / L \) (Handbook Physics p.491)

Primary 2700 turns would give \( (2.7)^2 \) times the inductance or 2.7 times resistance.

So \( Z = ((2.7 \times 67.8)^2 + (2.7 \times 28.5)^2)^{1/2} = 500.2 \ \Omega \)

or current 10 mA.

Secondary 2300 turns then gives output of \( 5.8 \times 15 \times 1082 \times \frac{2300}{2700} = 9.9 \ V \) full scale for 15 mm.
Performance of LVDT with secondary over primary

As wound by Star-Delta after putting cages on the SS former (all other attempts failed) The resistances were:

| Primary    | 28.5 Ω | 1000 turns of 0.112 mm wire |
| Secondary | 185.3 Ω | 800 turns of 0.060 mm |
| Secondary | 106.1 Ω |                 |

The current was 22.4 A with 16 mm core and 26.7 A with 12 mm.

With 16 mm core, linear range was approx 2.5 mm (± 0.25) and the output on 250/750 mA scale range was 1.93 V/mm

With 12 mm core, linear range still ± 2.5 mm and 250/750 mA range output was 2.67 V/mm.

These compare with about 8 Vdc/mm for the plastic former prototype (p 281). Now we have about four times the primary windings (28.5 vs. 103.25) and about 5½ times the secondary windings (211 vs 115). I don't have the number of turns - actually I'm number of turns so expect \( \frac{5}{3} \times 1.6 \) Vdc/mm.

Long LVDT (p 282): The linear range with 35 mm core was only about 10 mm by eyeball test, but a 20 mm range was within ± 1% of linearity.

With 5 V excitation, the output was about 68 mA. This was reduced to 8.9 A with 510 Ω resistor in series.

The output was about 0.76 Vdc/mm with this reduced excitation, corresponding to about 5.8 Vdc/mm with full 5V excitation. Don't have the number of turns yet.

| Primary    | 28.5 Ω | 1000 turns of 0.112 mm wire |
| Secondary | 954 Ω | 7500 turns of 0.060 mm |
| Secondary | 951 Ω |                 |
Further LVDT Considerations

On looking back to the first secondary-over-primary trial (p 281) without a central partition, and comparing with the later trials with central partitions (p 282 and 292), there does not seem to be much gained by adding the central partition. The RDP unit does not use one. So I propose to abandon this item and go back to a single bobbin for both the possible POSITION LVDT and the IF & IT LVDT's. Then I have accordingly been drawn up with respectively, 20 mm and 68 mm of total winding length. Based on maximum sensitivity.

The optimum core length for the 26 mm winding of p 281 was 16 or 15 mm, or about 60% of the winding length. On this basis, 20 mm winding length would require 12 mm core length.

A 10 mm core length is one-half the winding length, which would seem to optimize the linear range (equal space each side), although it may compromise the sensitivity slightly.

Looking at Solatron & Saelog figures, the cores are always more than half the winding length, maybe 0.7 is nearer, suggesting a core length for 20 mm winding & 48 mm for the 68 mm winding.

Taking a compromise, perhaps the best guess is 12 and 40 mm respectively. That may be more empirical tests are needed. Also there is some indication that shorter cores give more sensitivity e.g. p 292: 16 mm → 1.93 V/mm
12 → 2.67

Although for long LVDT, sensitivity the same for 40 x 55 mm core, or for PS2, 6.2 mm core gave little output than 19.5 mm core.

For no - fringe 26mm long LVDT, 20 mm → 7.56 V/mm
18 → 8.19
16 → 8.6
14 → 8.4

Intuitively, one would guess that the core should be half the total winding length (ie midpoint one secondary to mid point of the other, maybe +1 x 2 mm for the proximity of windings between the two) for 26 mm windings, \( \frac{26 + 2}{2} = 15 \) core length
20 → 12
68 → \( \frac{68 + 2}{2} = 36 \)
Position LVDT core stem:

[Diagram of LVDT core stem with dimensions and annotations]
\[
\frac{A}{ld} = \sin(29-\alpha)
\]
\[
\frac{S}{ld} = \cos(29-\alpha)
\]

\[
\therefore \Delta = S \tan(29-\alpha)
\]

\[
\alpha \text{ order of } 1^\circ
\]

\[
\therefore \Delta \approx 0.538
\]
Lateral movement of LVDT core in torsion

For small displacements $S$ of the torsion LVDT core, its lateral displacement is of order of 0.58 from approx calculation, opps, so a 12.5 mm core & stem in a $\phi 3$ bore should be OK provided it is reasonably centred. Probably a maximum $S$ of 0.4 mm is good to aim at.

cross-talk to axial IF LVDT from torsion

Provided that the core is fixed to the main body (not the torsion top), there is no lateral motion of the core and any core displacement can be accommodated.

cross-talk to axial position LVDT from torsion

The body of the LVDT should be attached to the axial section of the ILC, in which case it will undergo the same displacement as the torsion LVDT core, say 0.4 mm. Thus there has to be a clearance of ~1 mm min. in diameter where the body of the position LVDT passes through the main body of the ILC. Since making a bore of $\phi 10.5$ or more will tend to break out into the slots for clearance of the splines, we need to make the clearance hole elongated by 1x2 mm on a basic $\phi 10.0$. 
Optimum max displacements for LVDT cores

From Sunday notebook p 87-89, the test on the short LVDT with centre partition gave an approx. linear range of about 1-25 mm to within ~ 2%. However, over 0.7 mm (±0.35) the linearity was better than 0.5% (mean deviation from fitted line), maybe ±0.25%. So we can probably use a range of up to ± 0.4 mm satisfactorily, or even double this.

The short LVDT gave 2.7 Vdc/mm on 250/750 scale range with 800 secondary windings, so 4000 windings should give ~13 Vdc/mm. This compares with 7.7 Vdc/mm for the experimental LVDT with no partition & 4000 windings (same primary), but this was 30 mm long instead of 24. So we can probably reckon on ~10 Vdc/mm per LVDT, i.e. 20 Vdc/mm for two LVDT's or 18 Vdc for ± 0.4 mm displacement on 250/750 range. So we could expect to use the 250/750 or 100/300 range for the 0.4 mm displacement. This leaves the possibility of certainly going to 10x this sensitivity and probably 100x.

The 0.4 mm displacement would only apply to the cantilever-type load cell insert, not the pillar-type for 100 kN. In this case, the displacement is of order 0.060 mm, giving only ~ ±0.6 V, requiring ~ 17x gain, 850 for 25/75 or 10/30 mV/V range, on the amplifier card. Then we would require the 2.5/7.5 w 1/3 mV/V range (on the 10x scale) for a 18 kN range of loads.
<table>
<thead>
<tr>
<th>Type</th>
<th>Primary turns</th>
<th>Secondary turns</th>
<th>Primary current</th>
<th>Sensitivity on 250/750 mV/N range</th>
<th>slope of 2 LVDTs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main book</td>
<td>1300</td>
<td>3750</td>
<td>1.6 DC/mm</td>
<td>3.2 DC/mm</td>
<td></td>
</tr>
<tr>
<td>p18</td>
<td></td>
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<td></td>
</tr>
<tr>
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<td>1200</td>
<td>2800</td>
<td>1.25 DC/mm</td>
<td>2.5 DC/mm</td>
<td></td>
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<tr>
<td>Book</td>
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<td></td>
<td></td>
<td></td>
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<td>p17</td>
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<td>Experimental</td>
<td>1000</td>
<td>4000</td>
<td>8.4 DC/mm</td>
<td>-16.8 DC/mm</td>
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<td>p68</td>
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<td></td>
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<tr>
<td>Short</td>
<td>1000</td>
<td>800</td>
<td>2.7 DC/mm</td>
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<td>p87</td>
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<tr>
<td>Long</td>
<td>1000</td>
<td>7500</td>
<td>0.8 DC/mm</td>
<td>12 DC/mm</td>
<td></td>
</tr>
<tr>
<td>p91</td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
</tbody>
</table>
\[ F^* = \frac{4 \sigma b t^2}{3 l} \]
\[ S = \frac{0.1 t^2}{3 E t} \]
\[ \sigma = \frac{3 E t S}{l^2} \]
\[ F^* = \frac{4 E t^3 S}{l^3} \]
Elastic Element $F^*$ - Another check (following p.279)
We now designate the total axial force as $F^* = 4F$ (from previous $F$).

$$F^* = \frac{4.5\ell t^2}{3L}$$
$$S = \frac{0.00432 \sigma}{t^2}$$

We take $L = 15.75$, $\ell = 4.75$, $\ell = 1.175$, $E = 210000$

$$S = 0.636, \sigma = 0.001729\delta$$

For $S = 0.4$, $\sigma = 0.00432 \delta$ or $\sigma = 231.4 \delta$

For $F^* = 1000$,
$$1000 = 0.636 \cdot 231.4 \cdot t^3$$
$$t^3 = 6.79$$
$$t = 1.89\text{mm}$$

If we make $t = 2$, $F^* = \frac{4.210000 \cdot 1575.88}{33} 8 = 29458$

So for $F^* = 1000$, $\sigma = 0.343 \text{ MPa}$
$$\sigma = 0.396, \delta = 435 \text{ MPa}$$

For $F^* = 1$, $t = 1.9$ $F^* = 4.210000 \cdot 1575.88 (1.9)^3 = 2525 \delta$

For $F^* = 1000$, $S = 0.396$, $\sigma = 435 \text{ MPa}$

Thus, for a range of 1000N, we could choose $t = 2$ to have $S = 0.34$

for a range of 10N, $t = 0.4$ $S = 0.41$

The 1000N element could be used for 100N F/S on 10/30 mV/V scale
and for 10N F/S on 1030 mV/V plus 10X scale.

The 10N element, could be used on 1N and 0.1N F/S ranges
with respectively, 10/30 and 10/30 + 10X scale.
Element no 7 in torsion (following p. 280) - recalculation:

From p 280, total torque \( M = \frac{2 \pi c^2}{3} (b - b_0) \), \( S = \frac{\pi c}{2c} \)

\[ M = 23.00 \text{ Nm}^2 \]
\[ S = 0.005 \frac{\pi c}{2c} \]

\[ M = 4.543 c^2 S \text{ Nmm} \]

\[ T = 197.5 \text{ Nc MPA} \]

For \( M = 10 \text{ Nmm} \), we have \( c^2 S = 2.201 \)

<table>
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<tr>
<th>( S )</th>
<th>( c )</th>
<th>( T )</th>
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<td>0.4</td>
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<td>0.3</td>
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<td>174</td>
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<tr>
<td>0.25</td>
<td>3.00</td>
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</table>

For \( M = 0.1 \text{ Nmm} \), we have \( c^2 S = 0.0220 \)

<table>
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<th>( S )</th>
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<td>0.5</td>
<td>0.21</td>
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<td>19</td>
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<td>0.27</td>
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<td>0.05</td>
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<td>0.245</td>
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<td>0.14</td>
<td>0.40</td>
<td>11</td>
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<tr>
<td>0.09</td>
<td>0.50</td>
<td>8.7</td>
</tr>
</tbody>
</table>
\[ \frac{d}{ds} \left( \frac{d}{dM} \right) = 2 \]

First used 31/8/4.3

P. Adams

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<tr>
<th>0.05</th>
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15/11/03 - ordered ATT1-2 parts
Some Furnace History

The first commercial furnaces, up to 1995, were ASH furnaces. These were of the design developed in the late 1980's with single alumina core below an upper core extension, and 3mm ASH alumina silica insulation, segmented in three, and preloaded with compressed alumina paper on the OD. These were used up to about no. 8 x 10. There were problems with core cracking leading to lower efficiency. This was the incentive for an effort to make a furnace less prone to change in connection pattern as the core cracks. This led to:

Sept 94 PSZ furnace. Outer insulation PSZ, inner insulation SAI all over core & PSZ at each end. Furnaces 11-25.
Alumina paper was used as pre-loading on the OD of the outer PSZ & there was alumina paper in the segments. Lots of problems with loss of efficiency when the cores cracked. Introduced SiAlON cores in May 1997. The Vesuvius cores expanded & cracked the wires. This was in part overcome by winding loosely & in part by a change to AKUPRO SiAlON and later to British SiAlON (via Pogem). Then we started getting meltdown of SiAlON & % wire. This only became a problem in about 2000 & caused many failures from early 2001. At first it was thought to correlate with a new batch of % work (electropolished) but this was disproved. In Minnesota in July 2002 it became obvious that it was a chemical reaction effect. The meltdown problem did not appear at first. Was the Vesuvius SiAlON less prone to this??

Problems with electrical shorts at the bottom end - cured by redesign at new part here. Problems with low efficiency, leading to various uses of SS core segments to preload the SAI insulation but not at first.

Late 2000. Introduced alumina titanate instead of NZP (p 224).
Still problems with efficiency, notably related to sealing around the SAI.

Aug 2002. ATH 2 design, gradually introduced in refurbishment of NZP furnaces. These developments were mainly concentrated in furnaces 34 and 35.
PSZ
N2P
ALTI
SS segments and/or can
Later rigidized SAT1

Core 178 long
21 D 2500D
Furnace 34 (Potier)

This was initially built to the ALT1 design of Nov 00, shown opposite
16/8/01 MN 1020 p44. First test - custom up
22/8/01 50 Meltdown in Section after DP 100
7/9/01 56 New core 9619 N2P. Small round, instability
68 Pot-2, OP60 ~ 1600K, 13A. Eventually meltdown
25/1/01 84 New core (Section). SS can one SS segment to lead
88 92 S/A. Instability
18/10/01 98 Regarded a lower section of S/A (shown opposite)

This seems to be the last run until the furnace was used in the
Potier machine in a run on a steel specimen just before shipping
in May 02 (p112 in Potier's book).

Tests in Section on 7/5/02 (p112) gave 950K 8PS2 11.3A
in Potier on 21/6/02 (p131) 820K 8PS3 11.3A 0.35 W/K
15/10/02 (p135) 1050K 8PS7 11.7A 0.46 W/K

This is presumably still with the Section core. No news since then.
20.5.02: ALTI. SAHI one piece, reduced in size at bottom to fit tightly into SS piece, grooved on ID to take tails. SAHI a firm push fit into the ALTI, no AL paper used. ALTI undistorted & had to be machined oversize on ID. Re-drilled lead holes because not correct PCD.

25.9.02: New core (double ALDs), SiAlON extension pieces to complete length.

One-piece SAHI with grooves on ID for tails, 2mm gap on OD within ID of ALTI, rammed with AL paper. SAHI made in 3 pieces to suit core: upper one 241D X 440D X 5 long; centre 271D X 440D X 118 long; lower 241D X 440D X 65.

Would have to be 11 to give 2mm gap.
Furnace 35 (Minnesota)
Also built initially to the ALT1 No.00 design (previous page)
There is no record of before shipping, or of construction details, unless
Frank has some, except an indication that a pyrophylite piece was used in lieu
of the rigidized ALT1 in the lower part.
First run in Minneapolis on 9/7/02 or 12/7/02 (book 2, p.14-28). Generally
\( \approx 9 \text{ A} \times 35-400 \text{ O.P.} \), \( \approx 0.4 \text{ W/K} \) (there is some question about the amp readings)
Then there was a meltdown after going to \( \approx 1500 \text{ K} \).

Refurbished to the ALT1-2 (Aug 02) design shown opposite with some
modifications. Probably using Al2O3 core with Al2O3 ended core to
insulate the wires coming down. I think the original ALT1 may
have been cut off at the level A.

First run in refurbished form in Minneapolis on 25/01/02 (p.45)
and later on 29/10/02, 9/11 I have a record from another run on 4/5/05
In these runs the indicated current was \( \approx 8 \text{ A} \) in lowest winding,
OP settings around 30, \& consumption \( \approx 0.35 \text{ W/K} \) at \( \approx 1500 \text{ K} \).
As of this date, I have heard no more \& presume that the
furnace is still functioning.

17/6/05: Furnace damaged - details in furnace log.
8/7/05: Back from Minneapolis - core extensions renewed, heavy winding gone.
After "upgrade":
Dec 03, furnace 26 tested in ETH. 7 water problem arose. Also hot to top.

- AL30
- ALTI

23/4/04 test: -134\(\text{A}\)  OP 42.5  505\(\text{°C}\)  0.77 W/\(\text{K}\)

<table>
<thead>
<tr>
<th>Value</th>
<th>OP</th>
<th>Temp</th>
<th>1/K</th>
</tr>
</thead>
<tbody>
<tr>
<td>14.9</td>
<td>50</td>
<td>640</td>
<td>0.86</td>
</tr>
<tr>
<td>14.8</td>
<td>51</td>
<td>1020</td>
<td>0.88</td>
</tr>
<tr>
<td>14.3</td>
<td>56.1</td>
<td>1045</td>
<td>0.93</td>
</tr>
<tr>
<td>15.0</td>
<td>50</td>
<td>665</td>
<td>0.95</td>
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<td>13.5</td>
<td>42.5</td>
<td>427</td>
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</tr>
<tr>
<td>14.8</td>
<td>50</td>
<td>665</td>
<td>0.88</td>
</tr>
<tr>
<td>14.3</td>
<td>51</td>
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<tr>
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</tr>
<tr>
<td>14.9</td>
<td>50</td>
<td>587</td>
<td>0.99</td>
</tr>
</tbody>
</table>

Test 31/4/04 after taking out AL paper & string loading bottom. ALTI - essentially the same went to 1300\(\text{°C}\), 0.74 W/\(\text{K}\).
Furnace 26 refurbishment.

Furnace 26 is the first of the N2P furnaces (24/9/99).

First run in tests in Reg 7 at AS1 (p.15) on 28/9/99. At 880 K, bottom current 10.5, OP 41, 0.64 W/K. Went to 1500 K (0.5 W/K) OP 11, thin bottom winding meltdown in SiAlON core; much carbon.

Due to 45V being required for 1500K, the winding spacing was increased from 1.25 to 1.5mm in refurbishment. No note on new core; presumably fitted with SiAlON again.

Next run was in Zurich 8/11/99. At 786 K, 10.3 A, OP 415 & 0.42 W/K. Bottom short. Then ran again. 1107 K 12 A OP 54, 0.49 W/K. Came back to same.

Repeat calibration on 30/11/99 up to 1400 K, 0.6 W/K, 13 A OP 66.

Furnace kept back for refurbishment & upgrade as opposite. ALSO 2nd rigidized; sent to ETH on 5/11/03. Filled machine with water; it was not upgraded.

The original furnace has a complete N2P top 9 outer cylinder (leads up the outside), with stainless indentation on 3 ways. Al paper, and enclosed in a SS can with a ring around the top. This was reasonably efficient before the meltdown and after curing shorting problems, although it may have lost efficiency as time went on. The immediate behaviour after the first refurbishment (following meltdown) was the same as originally, but the degeneration presumably led to it being returned to an "upgrade" agreed to.

Mark 04: dismantled & also part rigidized at 1000C reassembled with additional rigidize AL 30. Other parts the same.

2/3/04 test. 16 A, OP 46, 1.2 W/K - reasonably inefficient due to high bottom current. Test in ES 2 machine.

New rigidized the AL 30 (cf. no. 28).

2/4/04 test. 16 A, OP 60, 380°C 1.3 W/K - worst:

<table>
<thead>
<tr>
<th>OP</th>
<th>67</th>
<th>72°C</th>
<th>1.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>15A</td>
<td>625</td>
<td>190°C</td>
<td>0.96 W/K</td>
</tr>
<tr>
<td>666</td>
<td>1218°C</td>
<td>0.92 W/K</td>
<td></td>
</tr>
<tr>
<td>2555</td>
<td>353°C</td>
<td>1.5 W/K</td>
<td></td>
</tr>
</tbody>
</table>

Initial performance similar before rigidizing except OP. Marked instability above 700°C collapse at OP 55 on coming down (~1000°C). Test carried out under same conditions in green machine. Don't understand OP.

25/4/04 up to 1216°C, after putting large section leads through the ALT. Definite improvement in efficiency, but still using rather too much power & instability still there.
At 8 ft:

Pyrophylite

PSZ

AL layer

Remum AL layer

"3 slots forming seal, metal leads in aluminum tube against core"

"SAL had 42 slots, only one slot all length"

"Reputal SAL, split but no gap"

26/4/92
Formae 30 (ETH)

It came back for refurnishment, tested 8/102 (MNp 128), with wider spacing of lower winding & harder V. Insufficient due to cool bottom winding (OP 100 for 890k).

Ref. 2: Lower section of SAF1 below bottom winding rigidized, segmented but no Al paper. Used 55 wires instead of 55 segments. In 55 can??

P.18 Again OP 100 but went to 1160k (6A).

P.18 Ref. 3: New core, back to 525 pitch; no comment on other changes. The bottom & top windings somewhat interchanged. At 1100k, OP 71, 81/4 in run 1.

Run 2: Rather variable behavior, 6.8 to 0.70 W/k, OP up to 100.

P.142 Run 3: Well down on top. New OP set up to 100 for ~1200k (15.7A) 0.6 W/k.

P.178 Run 4: "Model" of bottom, covered by PS 485, shifted around core. Mixture of carbon. OP 70, T 930, 12.4 A; OP 100 for ~130k, 15.5 A.

P.84 Run 5: Same, to do another cycle. OP 70, T 1200k, 0.4 W/k. Meltdown (max T 1310k) still mention of 55 wires -- some staining opposite split?

Ref. 4: New core, new bottom closure piece of rigidized SAF1 above bottom winding (don't care?)

P.90 Run 6: OP 83 for 1000k, 14.6 A. Loads of carbon.

P.100 Run 7: Probably after cleaning out carbon. Again to OP 100, 1170k, 15.9 A.

P.151 Run 8: OP 100 to set to 890k, 0.6 W/k, 15.8 A.

What happened with this? To be continued next day. Still waiting? Further wait.

Not clear the thermal measurement is even different from 34 (p301)
Furnace 28 (GT#)
9/12/99 first completed; N2P design
16/6/03 Republished to "new design" presumably ALT(2) with double Al2O3 core, new Al3O2 insulation rigidized & "baked". Al3O2 reduced by 3mm, 9 gap between it and outer SS can packed with Al paper, crimped with tubular tool. ALT1 upper, 9 lower pieces.

An email of 24/3/04 says no. 28 can only get to 900°C (1173K) if the hot spot is too high (so 1073K with a better profile).

So this furnace is similar to 26 prior to the water episode.

(4/6/03 We ran at RSES on two days: Went to 1285°C (1558K), OP45, 14A, 0.76 w/K, much moisture in afterwards.)
Should get AL30 to finish at Ø58 OD 9 24 ID
Fibre-ceramic insulation for AL1-2 furnace—heat flow

Recently we have changed from using Zircal SAL1 to AL30. Their respective properties of special interest are:

<table>
<thead>
<tr>
<th></th>
<th>SAL1</th>
<th>AL30</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shrinkage at 1650°C</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>Thermal conduct* at 800°C</td>
<td>0.34</td>
<td>0.19</td>
</tr>
<tr>
<td>Linear expansion RT-150°C</td>
<td>8.0</td>
<td>5.0</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>15</td>
</tr>
</tbody>
</table>

SAL1 has an alumina "matrix" and AL30 a silica "matrix". The latter is inerpatible under reducing conditions (the atmosphere). But I think whether this is a problem at Fe/FeO or Fe/Fe₂O₃ conditions their porosity is 84 95% respectively; no indications on permeability.

So on the whole AL30 looks the better specification except for shrinkage. But if we can spring load the insulation against the core, this may not matter. See opposite. The introduction of the AL paper in the 120°C splits may encourage more convection but this is hopefully less than presently occurs in the core/AL30 interface.

Radial heat flow. From p 61 we have

\[ \frac{q}{\Delta T} = \frac{2\pi l K}{\ln(D/l)} \]

where \( l = 0.108 \) m

\( D/l = 61/28 \)

\( K = \) Thermal conductivity

\[ = 0.872 \text{ W/K}. \]

<table>
<thead>
<tr>
<th></th>
<th>( \frac{q}{\Delta T} )</th>
<th>( \frac{q}{\Delta T} + 0.15 \text{ ends} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.19</td>
<td>0.17</td>
<td>0.32</td>
</tr>
<tr>
<td>0.34</td>
<td>0.30</td>
<td>0.50</td>
</tr>
<tr>
<td>0.4</td>
<td>0.35</td>
<td>0.50</td>
</tr>
<tr>
<td>AL1</td>
<td>0.87</td>
<td>1.02</td>
</tr>
<tr>
<td>PSZ</td>
<td>1.74</td>
<td>1.89</td>
</tr>
</tbody>
</table>

From \( q = \frac{KAT}{L} \), and heat flow would not be much more than 0.05 to 0.1 W/K, so \( \frac{q}{\Delta T} \) should be not more than 0.1 greater than the above figures. But this does not include conduction along the pistons which may be another 0.05 W/K.
Thus for AL30 we should probably be using no more than 0.3 to 0.4 W/K. Actually in recent ALTI-2 furnaces we have been using of order of 0.9 W/K, i.e. excessive 0.5 W/K due to connections.

If we take the specific heat of argon as about 1/5 that of water (density is similar), i.e. \( \frac{1}{5} \times 4.5 \text{ J/cm}^3 \text{ K} = \frac{0.8}{1000} \text{ J/m}^3 \text{ K} \), or at 1200 K, then, \( \text{specific heat} = 1.5 \text{ J/m}^3 \).

If total heat loss = 0.5 x 1200 = 600 W = 600 J/s, then there must be a heat flux of 600 mm$^3$/s or about 70 m/s velocity of flow through a 0.1 mm gap around the core, i.e. almost the full length of the core per second.

600 m/s coming back down the outside AL paper (0.5 mm thickness, area 64 mm$^2$) would involve a velocity of \( \approx 2 \text{ m/s} \). Maybe this is conceivable. It suggests that convection through small gaps is very important to control.
Coefficient of static friction for steel on steel:

- Rheodeville: 0.58
- Kent's Habit: 0.15
- Habit of Physics: 0.15 - 0.3
- Martin Habit: 0.78
- Smithells: 0.8

- Web, Concrete: 0.8
- School for Engineers: 0.6
- Servoy: 0.74

Average of all above: 0.58
" I expect 0.15": 0.72

Elastic: \( M = \frac{\pi d^3 e}{16} \)
Simple torque calibrating bars

The torque calibration bars used so far (P1 6012; AS1 7533 & 7534) have dogteeth at one end and splines at the other. The splines engage at atmospheric pressure but the dogteeth need confining pressure to engage. However, there are flats on the dogteeth and whereby torque can be applied at atmospheric pressure & so a torque-wrench used. Experience in Minneapolis & at AS1 (with Manchester) is that it is not very easy to use a torque-wrench.

So the conclusion is that there is not much point in attempting calibration at atmospheric pressure. At high confining pressure we could use a dummy specimen arrangement: This is limited to the plugging torque on the attachment to the pistons, but that limit applies to specimen testing also, so the limitation is unimportant.

From p.250, the friction-limited torque is:

\[ \begin{array}{ccc}
\text{At 500 MPa} & \mu = 0.5 & M = 221 \text{ Nm} \\
\mu = 1.0 & M = 442 \text{ Nm} \\
\end{array} \]

From numbers opposite \( \mu = 0.6 \) seems a likely value, leading to

\[ M = 265 \text{ Nm} \] at 500 MPa confining pressure

Thus there seems little point in selecting a calibrating bar that is stronger than 300 Nm.

Then for such a bar of 15 mm diameter, the shear stress is

\[ \tau = \frac{16M}{\pi d^3} = \frac{16 \times 300}{\pi (0.015)^3} \times 10^6 = 453 \text{ MPa} \]

or a tensile elastic range of \( 0.09 \times 453 \) MPa.

Assist 718 is quoted as yield stress (0.2\%) of \( \approx 800 \text{ MPa} \)
but Assist 6126

If we put a limit on \( \tau \) of 400 MPa, then from \( M = \frac{\pi d^4}{16} \times 10^6 \) we have for \( d = 15 \text{ mm} \):

\[ M = 265 \text{ Nm} \]

\[ d = 10 \text{ mm} \]

\[ M = 78 \text{ Nm} \]

\[ d = 5 \text{ mm} \]

\[ M = 10 \text{ Nm} \]
EMO Wave Washers: George Emmett

<table>
<thead>
<tr>
<th>Model</th>
<th>OD</th>
<th>ID</th>
<th>Wall Thickness</th>
<th>Water Thickness</th>
<th>Force per 15mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>EPL 19</td>
<td>28.0</td>
<td>27.5</td>
<td>0.30</td>
<td>3.00</td>
<td>54 N</td>
</tr>
<tr>
<td>EPL 45</td>
<td>55.0</td>
<td>54.5</td>
<td>0.45</td>
<td>5.5</td>
<td>132 N</td>
</tr>
<tr>
<td>EPL 48</td>
<td>62.0</td>
<td>61.5</td>
<td>0.55</td>
<td>5.5</td>
<td>132 N</td>
</tr>
</tbody>
</table>
Design Requirements of ALTI-2 Furnace

Introduction of additional EM0 wave washers.
We already use an EPL 48 washer under the furnace, in the outer cam. This is for 62 housing diameter.

Wave washer made core. This would be an EPL 19, designed for a housing of 28 mm OD. Our groove is only 23.25 mm OD, so the washer would have to be reduced in OD to about 24.5 mm from 27.5 mm. The ID is 21.0, so the width is reduced from 6.5 to 3.5 mm. The original spring rating is 54 N at 1.5 mm compression, so after machining, it would be 29 N at 1.5 mm.

If we set it up with 1 mm compression from its original 3 mm thickness, then it will start with 2 mm end load and after 1 mm core expansion, the end load would be 232 N. This is probably sufficient to ensure that cracks are kept closed. This we need to allow a 2 mm gap for initially locating the washer, same as now.

Wave washer made core with ALTI insulation - EPL 45.
This is the largest that will fit inside the 26 ID cam. A compression of 15 mm gives a load of 132 N; 2 mm → 176 N; 2.5 mm → 220 N; 3 mm → 264 N.

This is probably not more than 0.10-0.2 mm thermal expansion in the main part of the furnace, so the spring can be set for a final load, say, 200 N at 2 mm compression. Then the depth of groove (opposite) is 5.5 - 2 = 3.5 mm; would be 3 mm, giving 220 N force.

The external spring is compressed to 3 mm, similarly 220 N.
Choice of insulation:

<table>
<thead>
<tr>
<th></th>
<th>SA11</th>
<th>AL30</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>80% Al₂O₃ 20 SiC</td>
<td>85 Al₂O₃ 15 SiC₂</td>
</tr>
<tr>
<td>Max temperature</td>
<td>1700°C</td>
<td>Max continuous 1600°C, intermittent 1700°C</td>
</tr>
<tr>
<td>Temp. 1500°C</td>
<td>1570°C</td>
<td>1700°C</td>
</tr>
</tbody>
</table>

| Open porosity | 84% | 85% |
| Shrinkage     | 1% /day 1650°C | 1500 |
|               | 3%  1700 |
| Thermal conductivity | 0.20 280°C | 0.09 250°C |
|                | 0.31 800°C | 0.16 800°C |
|                | 0.39 1400°C | 0.27 1650°C |
| Coefficient of expansion | 8 x 10⁻⁶ | 5 x 10⁻⁶ |

If we use a double walled can, the temp. on the OD is unlikely to be more than the specimen temp., say, 1400°C, so the shrinkage of the AL30 may not be an issue, especially if the insulation is effectively spring loaded. The thermal conductivity is favourable for AL30, although the heat loss may be mainly by convection, not much to choose on open porosity.

Generally, the choice seems to point to AL30.

We should also try splitting the AL30 in 3 segments, with cut of 0.810 mm thickness & insert one layer of APA2 paper in cut. Then use tamped APA3 paper APA2 paper on the OD of the AL30.
Multirange Internal Torque Cell — Dogtooth on amid-piece

The dogtooth for transmitting torque to the load cell are now positioned in-board and involve 16 teeth. The pitch diameter for the teeth is \( \frac{33+26}{2} = 29.5 \) so tooth spacing is \( \pi \cdot 29.5/16 = 5.8 \) mm. Therefore the theoretical depth of the tooth is 2.9 mm.

If a 0.5 mm flat is put on the profile, then the tooth depth becomes 2.4 mm for engagement.

If the pitch surface is located at \( \frac{12.5}{16} \) from the top of the body, then the teeth will be cut in a bit from the face 1\( \frac{12}{16} \) from the top.

With 100 kN external force, the shear stress on the M35 thread, say effective length 2.5, will be

\[
\frac{100}{\pi \cdot 33.5} = 193 \text{ MPa}
\]

This should be OK, although a bit above the usual 100% design figure.

The shear stress in the shoulder at bottom of nut (\( \phi 25.8 \)) and mean depth (to pitch plane) of 6.5 will be

\[
\frac{100,000}{\pi \cdot 25.8 \cdot 6.5} = 190 \text{ MPa}. \quad \text{Still OK, yield stress in shear for Al 718 is } 460 \text{ MPa.}
\]
Multi-range Internal Load Cell - IT Cross-talk Problem

It turned out that when axial load was applied there was an internal torque (IT) signal that was similar in magnitude to the internal force (IF) signal. After much fruitless chasing, it became evident from sliming experiments that the effect must be associated with imperfect matching between the bottom surface of the elastic element section (Fig 4602) and the top of the load cell base (Fig 4601).

Bending of base of elastic element:

From Parks p 447, the deflection of a plate loaded at the centre is
\[ s = \frac{F}{2Et^3} \]
where \( r = \) radius
\( t = \) thickness. This formula probably does not apply very well to the base of the elastic element but it will serve to give an idea of magnitude.

Assuming that the circumference of the "plate" corresponds to the diameter of the load support pillars, \( r = 26 \text{ mm} \) and \( t = 12 \text{ mm} \),

Then \( s = \left( \frac{0.026}{2 \times 0.012} \right)^3 \times \frac{F}{210} \cdot 10^{-6} \text{ m} \) when \( F \) is in N.

For \( F = 10 \text{ kN} \), \( s = 9.3 \cdot 10^{-6} \text{ m} = 9.3 \mu\text{m} \)
For \( F = 100 \text{ kN} \), \( s = 93 \cdot 10^{-6} \text{ m} = 93 \mu\text{m} \)

Due to the relative thickness of the "plate", there may be an overestimate?

Shear of base of elastic element

If we suppose a shear force \( F \) is applied at the radius 26 mm and is supported at the radius 15, an average radius is about 20 mm. 217. 0.020 0.012 so with 10 kN applied, the shear strain is
\[ \frac{0.020}{0.012} \approx 1.66 \text{ for } 10 \text{ kN} \]
or \( s = 0.00016 \cdot 11 \text{ mV} = 0.00018 \text{ mm} \) for 10 kN
or \( s = 0.000083 \cdot 11 \text{ mV} = 0.000091 \text{ mm} \) for 100 kN

About an order of magnitude less than calculated from bending.
The deflection of the base if supported at the edges is estimated somewhere between these two estimates, maybe 30 mm to 50 mm. This could apply if there were no lack of register between the centre part of the elastic element and the base, i.e., the base is exerting an upward force on the centre and the axial loading is exerting a downward force around the perimeter. This lack of register seems to be taken up with about 50 KN or so, which gives a deflection of maybe 15 mm - this seems feasible, and would be within the 25 mm tolerance on flatness.

Use of confining pressure to ensure complete contact.

The confining pressure acts over the area inside the ring, viz.

\[ q_{26} \times (26)^2 = 531 \text{ mm}^2 \]

so the force from 100 MPa is 53 KN. This 53 KN would then give, on the formula above, a deflection of

\[ \frac{(0.026)^2}{2 (0.012)^3} \times 53000 \times \frac{1}{210.10^5} \text{ m} = 0.049 \text{ mm} \]

or 49 \mu m.

If this overestimates by a factor of 2 to 3, then \( S \approx 24 \text{ to } 36 \text{ mm}. \)

Thus, if we had a recess of say, 0.020 mm, we might expect it to be pushed into contact with the base with a confining pressure of around 100 MPa or maybe a bit more.
<table>
<thead>
<tr>
<th></th>
<th>Inside seal</th>
<th>Centre of vessel</th>
<th>Outside seal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston</td>
<td>0.045</td>
<td>0.045</td>
<td>-0.026</td>
</tr>
<tr>
<td>Cyl.</td>
<td>0.070</td>
<td>0.138</td>
<td>+0.070</td>
</tr>
<tr>
<td>Initial clearance</td>
<td>0.040</td>
<td>0.040</td>
<td>+0.040</td>
</tr>
<tr>
<td></td>
<td>0.155</td>
<td>0.223</td>
<td>0.084</td>
</tr>
</tbody>
</table>

per 500 MPa

|                  | 0.063       | 0.077            | 0.049        |
|                  |            |                  | per 100 MPa  |

smallest value and largest value...
Re-examination of Intensifier Gas Cylinder Design.

Following the seizure of the piston in the cylinder of the Brown HPT, I checked all the dimensions and they were within spec, often on the lower limit, and the hardness of the piston was ~54 RC, also within spec.

Dimension changes under pressure at 500 MPa

Poisson expansion of piston outside seal:

\[ \Delta d = \frac{pdV}{E} = \frac{500 \times 41 \times 0.14}{210000} = 0.026 \text{ mm} \]

Compression of piston inside vessel:

Fixed compressibility = \( E \left( 1 - \nu^2 \right) \) so

\[ \Delta d = \frac{P(1-\nu) d}{E} = \frac{500 \times 0.46 \times 41}{210000} = 0.045 \text{ mm} \]

Increase of bore under pressure:

Plan form elastic case (Eq. 1.9 p.12)

\[ \Delta d = \frac{pd}{E} \left( \frac{D^2 + d^2 + y}{D^2 - d^2} \right) = \frac{500 \times 41}{210000} \left( \frac{160^2 + 41^2}{160^2 - 41^2} + 0.17 \right) \]

\[ = 0.138 \text{ mm} \]

Near the seal area, it is probably about one half of this, ~ 0.070 mm.

Then is also a minimum clearance over the piston of ~0.040, ~0.155

So the total clearance over the piston will vary from about 0.23 mm

The seal to 0.070 mm in the central section at 500 MPa.

At 100 MPa, these figures are 0.063 and 0.077 mm resp. This is

the situation at the beginning of the stroke. So the seizure could hardly

be attributed to lack of clearance. This is also borne out by the

damage being confined to two fairly narrow sections of the circumference

about 180° apart (pick-up on one side presumably pushed the piston across

until pick-up occurred on the other side). Note: Inside the seal, the

figures are a bit less, 0.049 and 0.084 at 100 and 500 MPa resp.

(see opposite).
\[ \Phi D = \Phi 4.9 \]

\[ \Phi 4.67 \text{ increase to } \Phi 4.7 \]

\[ M60 \times 4 \text{ (fine)} \]

Pitch \( \Phi = 57.4 \)
Seal Repair in Intensifiers

With 223 O-ring, present squeeze is 5.7/2 = 2.85 mm or 3.57 \% squeeze.
Maybe 15\% is enough, making O-ring groove 3.00 deep so incer. 0.407 to 0.470. If we make D = 49.0, the flange is 1mm thick - should be OK. Then OD of wrote - O-ring groove becomes 49 + 6 = 55.0.

If we put \( y = 15 \), the length compressed against the vessel is \( y - 5 = 10 \text{mm} \) should be ample

Force on unit at 730 MPa is 730 \((47 - 41) \frac{2}{4}) = 302,700 \text{ N}

If we make \( x = 20 \) of flanges 3 run-out for thread, area of thread = 3066 mm\(^2\), giving shear stress = 99 MPa - OK.
From Clark

<table>
<thead>
<tr>
<th></th>
<th>E (GPa)</th>
<th>S (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>granite</td>
<td>40 E+00</td>
<td>15 E+00</td>
</tr>
<tr>
<td>limestone</td>
<td>60 E+00</td>
<td>20 E+00</td>
</tr>
<tr>
<td>marble</td>
<td>50 E+00</td>
<td>20 E+00</td>
</tr>
<tr>
<td>sandstone</td>
<td>20 E+00</td>
<td>10 E+00</td>
</tr>
<tr>
<td>schist</td>
<td>50 E+00</td>
<td>20 E+00</td>
</tr>
<tr>
<td>shale</td>
<td>20 E+00</td>
<td>10 E+00</td>
</tr>
</tbody>
</table>

Typical: 50,000 MPa

Points from Tom Jackson:

- General indication: 10^-5 strain is upper limit for linear behavior.
- Sub-Hz frequencies will be very limiting in exploring the physics of attenuation, e.g., in fluid-saturated conditions. Probably want to go up to kHz frequencies.
- Doing both P and S important in fluid-saturated rocks.
Attenuation Module for HPIT

The general concept is set out in PI drawing 9000. It requires a high sensitivity, internal load cell and a porous fluid volumometer to operate the loading actuators for the cyclic loading. The main axial actuator is used for initial positioning.

Assume a specimen 25 mm long with Young's modulus 50,000 MPa and shear modulus 20,000 MPa.

If we work at strain amplitudes $10^{-5}$, then $0 = 10^{-5} \cdot 50,000 \cdot 0.5 = 0.5$ MPa.

Then force on 25 mm = $\frac{F}{21} = 0.5 = 245$ N - no problem

and displacement on 50 length = $10^{-5} \cdot 50 = 0.0005$ mm or 0.5 mm.

Displacement requirements.

With our present LVDT's in the HPIT, the maximum displacement at full load is of order of 50 mm, which gives full scale (10 V DC) signal on a relatively coarse gain in the isolation card, of order of 50 mV per V setting. This could be increased by 10x without the problem, probably, giving 5 mm full scale. Then in still a 10x gain possibility, leading to 0.5 mm full scale with maybe 10% sensitivity. At $Q = 100$, we need to detect ~ 0.005 mm displacements, so the accuracy of $Q$ determination is then only about 10%. Perhaps this is useful.

As determination of $Q$ at 10% strain amplitude may be marginally possible. At $10^{-4}$ strain amplitude, it would be easy. $10^{-6}$ is out of the question.

Drive requirements.

At $10^{-5}$ strain amplitude, we have to move the actuator 0.5 mm in loading the specimen. If this is done by pumping gas into a bellows of 130 mm effective cross-section, then the fluid volume (neglecting compressibility) is $\frac{0.0005 \times 130}{38} = 0.065$ mm$^3$.

The cross-section of the volumeteric piston is $\frac{\pi}{2} \times 0.065 = 38$ mm$^2$, so the piston displacement required is $0.065 / 38 = 0.0017$ mm (or 1.7 mm).

Assuming a lead of 5 mm in the volumeteric screw and 160:1 reduction, this corresponds to $\frac{0.0017 \times 160}{5} = 0.054$ motor turn or 19° motor rotation.
It may be better to make up a device with compressed pistons driven by an electromagnetic actuator which could give quite high frequencies, maybe up to the kHz level. Good control on amplitude would be important.

A force of 245 N acting on a bore of 130 mm effective section would require a pressure \( \frac{245 \times 130}{51,850} = 31 \text{ N/m}^2 \) or \( 1.9 \text{ MPa} \), and if this is generated with a piston of \( 5 \phi = 19.6 \text{ mm}^2 \), the piston force required would be \( \frac{245 	imes 19.6}{130} = 37 \text{ N} \).

Thus the electromagnetic actuator would need only apply 37 N (plus friction), as a rating of order 100 N would be required.

Requirements of actuator:
- Apply force up to 50-100 N.
- Displacement 1.7 mm.
- Control for sinusoidal oscillation, based on either force or displacement.
- Operation over frequency range \( 10^{-3} \) to \( 10^3 \) Hz at least 0.1 to \( 10^3 \) Hz.

These requirements assume a strain of amplitude of 10^{-5}.

Torque Requirements:

Again consider strain amplitude 10^{-5}. Then target = \( \frac{22.5}{c} = 2.5 \times 10^{-5} = 4 \times 10^{-5} \) radians.

Displacement at radius 7.6 mm will be \( \frac{22.5 \times 4 \times 10^{-5}}{90 \times 10^{-5}} = 0.8 \text{ mm} \) or \( 0.8 \times 10^{-3} \text{ mm} \)

Similarly for axial displacement so similar signal from LVDT is somewhat larger.

Torque = \( M = \frac{\pi G \cdot 4 \times 10^{-5}}{32} \approx \frac{20,000 \times 25 \times 4 \times 10^{-5}}{32.5} = 0.61 \text{ Nm} \).

This requires a force of 613.6 = 38 N at radius 16 mm, or 19 N in each of the bellows, or 0.15 MPa pressure in bellows of 130 mm cross-section.
Do with the same differential pressure in the bellows we can go to somewhat higher strains in torsion than in axial loading.

**LVDT's**

We use the same short LVDT's as for the 1LC, i.e. 24mm long (not clear why there are not 25 but we have standardized on 24).
The same core can also be used as for the torsion LVDT's in the 1LC, as per drawing 4607-4.

Bellows

Max OD ~ 20mm
Max length ~ 25mm (30 including welded ends)
Max height ~ 1mm
Pressure – max ~ 10 MPa (gives force ~1300 N) or 20 MPa (e = 0.5 x 10^-5)

Compliance of bellows not an issue.

**Alternative to Bellows – a straight tube (sketch opposite)**

For axial case, the length L could conveniently be 50mm and the diameter D = 20mm.

If max strain in y = 10^-4, then stress in y = 20 MPa and the force on y = 9 = 2454 N, say 2500 N.

If cylinder ID = 18, then pressure in cylinder = \( \frac{2500}{\frac{\pi}{4} 18^2} = 9.8 \), say 10 MPa.

Circumferential stress in cylinder at 10 MPa = \( \frac{10 \cdot 18}{20 - 18} = 90 \text{ MPa} \). Could make the wall thickness less than 1mm, easily 0.5 mm or less.

Volume of fluid: strain 10^-4 in 50mm long specimen gives \( \Delta L = 5 \cdot 10^{-3} \) mm plus about a quarter of this in the pistons, i.e. \( \Delta L = 6.3 \cdot 10^{-3} \) for \( L = 10^{-4} \).

So length of fluid column in cylinder changes by \( 6.3 \cdot 10^{-3} \frac{\pi}{4} 18^2 = 28 \text{ mm} \).

Thus the volumometer supplying the drive has to provide \( 1.6 \text{ mm}^3 \) of fluid. The 0.7 mm piston of volumometer has area 38 mm^2 so the displacement of the piston is \( 1.6/38 = 0.042 \text{ mm} \), or 0.0083 of a turn of the ball screw (lead 5 mm) or 1.3 revs of the motor (160:1).

At 1° strain amplitude, the motor will only turn 0.13 rev or 48° for 0.16 mm^3.

Thus the fluid pumping will constitute a problem for
Even with a diaphragm of 20 μ, a force of 3000 N is needed for $10^{-4}$ strain amplitude. This is much in excess of the linear motor "strength" capacity for Lego which goes to 1365 N with 15 x 10 dimension. Seems hardly feasible to make an electromagnet that is strong enough in the dimensions available.
The following reasons:

1) At strain amplitudes ~10^{-5} when there is only a fraction of a turn of the motor involved — this may present control problems.

2) It is not clear how high one can go in frequency, might be difficult above 1 Hz.

So we should probably consider alternative ways of driving the fluctuating strain. I think the hydraulic drive inside the pressure vessel is still a good idea but we need a fluctuating pressure generator outside which is more flexible than the cycloconverter.

One possibility is sketched opposite, using a balanced φ3 piston arranged. Area of φ3 piston = 7 mm² so to provide 0.16 m³ fluid at ε = 10⁻⁵ the stroke is 0.4/7 = 0.223 mm or 22.3 μm. If we used a lever arm of 400:1, the displacement would be 9 mm.

The force of 300.19 Pa (continuous pressure on φ3 is 2100 N), and if the friction were 2%, the frictional force on one piston would be 420 N, or ~800 N for the two pistons. So we are looking at an alternating force of the order of 1 kN = 1000 N on the φ3 piston. With a 400:1 lever arm, this requires a force of 2.5 N at the actuator, well within the range of electromagnetic actuators.

The smallest Cogley "servo tube", 204, provides a continuous force of 51 N, while their smallest "thrust tube" provides continuous 4 N (mini-series) or 51 N (TT 25 series).

These actuators can be controlled with an Accelmer or Accelus amplifier which takes a ±10V DC command signal.

The movement required from the actuator with 400:1 lever arm is 9 mm for 10⁻⁵ strain amplitude. There are all small compared to the max travel of the Thrust Tube linear motors [377 - 1214 mm].

An alternative might be an electromagnetically loaded diaphragm pump. Suppose the diaphragm is 0.5 mm (0.020") thick and 2" (50 mm) diameter. The deflection ε is given by

\[ \varepsilon = \frac{0.5 P}{0.02(30.10)^2} \]

If P = 100 lb/in², \varepsilon = 0.0021 in or ε = 0.005 mm for 45 N force.

But only 0.003 mm deflection is needed to provide 1\frac{1}{2} mm³ of volume.

The force for this is \( \frac{0.05}{2} \times 10 \text{ N} \); pressure differential is 101.3 kPa. Need to compromise on diaphragm diameter, deflection & 45 N force.
Summary of Requirements for strain 10^{-4}

Stress in specimen 5 MPa, force in specimen 2450 N.
Pressure in bellows 9.6 MPa
Displacement "bellows" ~ 0.006 mm
Strain in "bellows" (β=60) 10^{-4}
Stress ... 21.1 MPa
Circum. stress in " 173 MPa (0.5m WT) ; inner diameter = 15 mm
ΔV in "bellows" 1.5 m^3

LVDT connection

EM90 waved washer EPL 34
OD 39.5
ID 33.0

Unloaded height = 4.0
Loaded height = 1.5
Material thickness = 0.43

differential pressure for axial loading of specimen up to 9.2 MPa for 10^{-4} strain.
Spring-loading of Axial Driver unit

The thickness of the waved washer is initially 6.0 mm, but it seems to be designed to be compressed to 1.5 mm at which the load is 77 N (the spring rate is 33 N/mm). If we leave a [0.5 mm] gap between the two parts when the spring loaded, the forces on the component A are as follows:

\[
F_s = \text{force from load in specimen}
\]

\[
F_f = \text{force from pressure in driving fluid (actually differential pressure)}
\]

\[
F = \text{force due to tightening down on the ENO waved washer - spring}
\]

\[
\Delta F = \text{additional force in springs due to movement in specimen}
\]

Then for static balance we have

\[
-F_s + F_f + (F + \Delta F) - (F - \Delta F) = 0
\]

\[
i.e. \quad F_s = F_f + 2\Delta F
\]

The value of \( \Delta F \) derives from the displacement \( s \) associated with the strain in the specimen. If spring constant of ENO waved washer is \( C = 31 \, \text{N/mm} \), then \( \Delta F = 5.0 \, \text{kN} \).

\( s \approx 0.006 \, \text{mm} \) (linear aproximate)\( \Rightarrow \Delta F = 0.006 \times 31 = 0.19 \, \text{N} \) when strain in specimen is \( 10^{-5} \) or 0.02 N at strain \( s = 10^{-5} \). These values are independent of the initial loading on the waved washers. Also the displacements in the springs/washers is very small, equal to the displacement in the specimen, so the gaps can be 0.5 mm or less between the two components A & B.

If the fluid pressure differential \( F_f = 0 \), then \( F_s = 2\Delta F \). This is the situation when bringing up the actuator to make contact with the specimen assembly. Then \( F_s = 2\Delta F \) when \( \Delta F \) is the displacement after contact. If \( s = 0.010 \, \text{mm} \) and \( C = 31 \, \text{N/mm} \), then \( F_s = 0.6 \, \text{N} \). This force can be monitored with the UCC and any desired initial stress established in the specimen by virtue of the spring washers. From this basis, stress can then be added to the specimen by applying differential pressure in the "fifiers".
Dimensions of Torsion Drive Knob

\[ \theta = \frac{4}{100} \]

\[ \alpha = \frac{l}{r} \]
Flexure Pivot Design

The primary blade X has to support the force to apply pressure to the actuation unit, which is small (3.6 N/m x 7 m^2 = 25.2 N), and the friction on the two of 3 pistons. At 500 N/s, the force on the pistons is 500 x 7 = 3500 N or 7000 N for two pistons. The friction is likely to be of the order of 10% of the force, i.e., ~700 N, say 1000 N.

If the stress in the blade is to be ~100 MPa, then 10 mm are needed for supporting 1000 N, say 10 x 1 mm.

If we had a component to actuate the continuous force ratio for, say, the ST25 - 2506 model is 70 N, needing a lever amplification of 1000/70 = 14, say 20x.

So if the fulcrum is 5 mm from the piston axis, the actuator has to be 100 mm from the fulcrum.

Thus for a volume amplitude of 1.5 mm (strain 10^-4), the piston displacement amplitude (phi3 piston, area 7 mm^2) is 1.5/7 = 0.2 mm. Then the actuator amplitude will be 0.2 x 20 = 4 mm for 10^-4 strain.

Flexure strip driving the phi3 piston:
The strain in the strip is
\[ \frac{2\pi r}{2\pi r} = \frac{r}{2r} = \frac{r}{2r} \]
where r is the radius of curvature of the strip. But r = \( \frac{2t}{\pi} \) when \( t \) is the length of the strip and \( \pi \) is the angle through which the strip is bent.

\[ E = \frac{k}{2l} = \frac{6E}{2E} = \frac{E}{2E} = \frac{E}{2E} \]

For t = 1 and l = 10, E = 1 / 500 so \( \sigma = \frac{200 E t}{500} = 400 \text{ MPa} \).

This would still be within the elastic limit of spring steel but may be near its fatigue limit. Using the copper would halve this stress.

Also it is based on 10^-4 strain amplitude, so 10^-5 strain amplitude is

Buckling: For fixed ends, Euler buckled load is
\[ F = \frac{4\pi^2 E/L^2}{4L^2} = \frac{4\pi^2 E L^2}{16 L^2} = \frac{E L^3}{4L^2} \]

For t = 10, t = 1, l = 10, \[ F = \frac{4\pi^2 E L^2}{16 L^2} = \frac{\pi^2 E L^2}{4L^2} = \frac{E L^3}{4L^2} \]

For t = 10, t = 1, l = 10, F = \[ \frac{\pi^2 E L^2}{16 L^2} = \frac{10,000 N}{400} = 25,000 N \]

No problem.
Pivot Strips. The force in the strip is half the resolved force along the strip ie 
\[ F' = \frac{F}{\sqrt{2}} = \frac{707 \text{ N}}{\sqrt{2}} = 1000 \text{ N}. \]

Length of strip \( l' = 50 \text{ mm} \), to buckling load
\[ \frac{l'^2E\epsilon^3}{4l'^2} = \frac{l'^2E\epsilon^3}{42500} = \frac{200000 \times 10.1}{4 \times 2500} = 2000 \text{ N}, \]
which is nearly three times the estimated force shown. Probably get away with it with steel strips but marginal with brass.

**CHOICE OF MOTOR**

After considerable consideration of Copley Thrust Tube motors (linear actuators) they said they were not suitable. However, a similar product from Lincoln Technologies seemed a possibility. Again it was a linear actuator, ideally more suited to some rather than just instruments. They could provide both actuators & amplifiers.

An alternative type of actuator would be a "voice coil" type. The following are possibilities:

**Motion:** 40N, \( \varnothing 99 \times 70 \pm 5 \text{ mm} \) \$125.50D; Willard says he drives them with a function generator & good quality stereo amplifiers.

**BEI Kimerco Magnetics:** \( \varnothing 80 \times 50 \) \$80.00D; also have motor drivers.

**Sourcourt Magnetics Inc:** \( \varnothing 60 - 0.25 \) 75N, \( \pm 6 \text{ mm} \) only one listed on web.

These look to be the most suitable

Motor is chosen, together with a special Copley amplifier (412/22).

See p331 for pressure connection to the internal attenuation cell.
Furnace 40 (Minneapolis)

After refugishment with new bottom windup & aluminizing cement fillup groove over windup; old core shortened a little to lessen waved wasting, bearing on inner core; stl. trade at both ends of windup in inner core & some cracks in outer core near upper end of bottom windup. New Al3O4A11 insulation. Bottom face of Al3O4A11 covered with aluminizing cement. Running on MANUAL.

<table>
<thead>
<tr>
<th>Resistance</th>
<th>T 0.5</th>
<th>£ 10</th>
<th>B 0.5</th>
<th>(leads 0.2 inl)</th>
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<tbody>
<tr>
<td>A</td>
<td>270</td>
<td>8.0</td>
<td>10.0</td>
<td>128</td>
</tr>
<tr>
<td>B</td>
<td>1000</td>
<td>14.0</td>
<td>29.8</td>
<td>417</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>310</td>
<td>8.8</td>
<td>17.0</td>
<td>150</td>
</tr>
<tr>
<td>B</td>
<td>1000</td>
<td>14.0</td>
<td>29.2</td>
<td>409</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

At ~1200°C, had to back off OP while increasing T & C to get flat profile at around 1200°C (+), but it proved to be unstable — bottom cooled off & top got hot, so backed off T & C & increased OP. Efficiency came down to 0.8 W/K at OP ~ 43.

So increased OP again & backed off T & C. Bottom V had come down a lot, i.e. bottom windup cooling.

Settled for a while at OP 47. T gradient downwards, $y = 0.93$ W/K.

Slightly increased OP to 49; bottom V built up very slowly.

As OP is increased, a point is reached at which T climbs while OP is reduced — an instability. $y$ can vary from 0.8 to 0.94. Two examples:

<table>
<thead>
<tr>
<th></th>
<th>A</th>
<th>V</th>
<th>W</th>
<th>OP</th>
<th>Spec T 1155°C</th>
<th>1126°C - 1186°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>T</td>
<td>340</td>
<td>9.1</td>
<td>28.2</td>
<td>274</td>
<td>OP = 49</td>
<td>Spec T 316/111°C</td>
</tr>
<tr>
<td>C</td>
<td>400</td>
<td>5.0</td>
<td>32.2</td>
<td>167</td>
<td>Spec T 122°C</td>
<td>120.5°C - 123.4°C</td>
</tr>
<tr>
<td>B</td>
<td>1000</td>
<td>14.0</td>
<td>42.0</td>
<td>623/1055</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

17-7-06
Left in OP 48 for reduced T/C/3 to 310/0/1000

After ~ 40 mins, temp at cut face at bottom winding:
- T 310 8.6 14.0 120 238 MPa
- C 0 - 0 0 0
- B 1000 13.5 20.0 270 123 W/K.

Down to zero and left over night.

18/7/07

188% H₂. Started up on manual. Needs about OP 30 to start off at reasonable rate (10A 12V). Up to OP 48

- T 310 8.7 14.0 122 233 MPa
- C 0 0 0
- B 1000 13.5 20.2 273 1.22 W/K.

Tamped up to 340 / 400 / 1000 across at OP 48.0
Temp going up rapidly, ~ 2°C/sec, then 1°C/sec.
- T 340 9.4 23.5 221 279 MPa
- C 400 6.6 31.7 209 OP 48.0 spec T = ~ 860°C
- B 1000 14.1 30.0 423 \[=853 \]

Put on OP = 50.0 for more than 1 hour

- T 340 9.8 27.8 272 296 MPa
- C 400 5.8 32.4 188 OP 50.0 spec T = ~ 1070°C
- B 1000 14.6 38.0 555 \[=1015 \]

Put on OP = 52.0; after a while, cut back T to C; at ~ 1200°C, cut back C

- T 340 8.4 20.0 168 310 MPa
- C 340 4.7 30.3 142 OP 51.0 spec T = 1250°C
- B 1000 14.7 48.7 716 \[=1026 \]

~ 0.88 W/K.

Up to OP 52 again and back to 51-6

- T 260 8.4 24.9 209 314 MPa
- C 340 4.6 30.5 140 OP 51.6 spec T = 1280°C
- B 1000 14.4 51.4 \[=740 \]

\[=1090 \] ~ 0.85 W/K.
West up to 1325°C, then 1333°C (1610K). Cut off power.
About 1 hr in vicinity of 1300°C

So far, it is very inefficient, improving a bit at very high T. Presumably, this behavior is all related to the cracked core. The filling of the grooves does not appear to have had any beneficial effect, from which I conclude that there is no significant role of convection into the bottom crack, up the helical groove & out of the top crack, back down the inside.

Rather, the convection must be in through the crack & down through the insulation, & back up the ID of the core. Maybe this halves the normal convection resistance of a circulation entirely within the insulation. The effectiveness in the absence of the crack is shown by the first runs by Mark on NS 40 in Minneapolis. However, the occurrence of the crack may have more dramatic effect with porous rather than solid insulated at the bottom of the furnace (Mark Zimmerman has also speculated on this possibility).

So it is about time to try to make a furnace core element without grooves in the core. This requires winding the wire tightly, with regular spacing (either by using the draw-cutting facility on the lathe or winding several wires together & later removing the spacers). Then cement has to be put over the wires & dried. Mark uses Ceramabond 569, the same cement as we have been using for patching, applying it in thin layers & drying between each application. One winding is done at a time. However, in the PSZ furnace, where the wires come radially out to the split in the groove, the cement can be smoothed off to a well defined radius & also aluminizing paper is used.

In our AL 30 furnace, there will be a problem with getting a good fit between the cement coating & the ID of the AL 30. Because the length of trial is the requirement to come down axially before going out radially, with the tails making it impossible to machine the cement to a good OD. One possibility is to use a teflon mold to determine this OD.
Core without Grooves

There are two other possible approaches that come to mind:

1. After covering the windings, bend the tube down with a half-
   cylinder of alumina tube under it to insulate, & plastic
   cement over the while assembly, building up to
   a uniform thickness.

   OD of core tube currently is 25

   if 4mm wire is 0.5mm, so OD our wire = 26.

   height of alumina tube = 15mm will
   have to be taken, the thickness will be
   175, ie add 3.5 to the OD to core, this
   ie OD will finally be 26 + 3.5 = 29.5, so we will need
   to build up to 40, ie an additional rated thickness of 25mm.
   This may be pushing things a bit with the 569 cement.

2) Go back to taking the windings out radially.

   This becomes very complicated for getting the
   Al2O3 in place - hence to use split cylinders
   again, ideally split 6-ways. Not a viable
   option?

   Winding:

   Centre should be in same location as before,
   so total length covered = 52 and located 33 from
   the top of core. Can keep the pitch 1.5mm. Length of
   winding is increased by factor 24/35 = 1.11, ~10%

   Will voltages ~ 30. This is not a problem & V is only
   increased ~5% for same power V^2R, 3A decreased 5%

   Top: Need a space & 4 between windings, so bottom of top winding should
   be 29 (or 28) from top of core. Previously we had 29.335, 29 = 98.4mm

   To keep the same length we now need 29.335 / 2.5 = 18.4mm length wound.

   Therefore, maybe previous pitch of 1.5 could be reduced to 1.0, then
   we only need 12.3mm length of core wound for same number of turns.

   or 18.4  at pitch 1.5

   Bottom: Previously we had 29.335, 29 = 135.8mm. However we want to
   reduce the voltage by about a factor of 45/40, ie V by 1.26 x & so R by 1.26 x
   ie reduce length of wire to 1673 mm, so number of turns = 13.4, 2

   or length wound = 13.4 mm at pitch 1, 20 mm at pitch 1.5.
The text is not clearly legible due to the handwriting. It appears to be a page of notes or a record of some sort, possibly related to a scientific or technical topic. The handwriting is cursive and contains some mathematical or technical terms. The boxed text in the corner reads "Results in SUNDAY 1 book 1746 at ASI."
Tests run with non-ground core in furnace no. 41
Frank refurbished furnace 41 with an alumina core without grooves,
using the configuration shown on p. 324 for the windings. He
commented on the very hard setting of the Ceramabond 587 cement.
The furnace had AL2O3AT1 insulation taken to the bottom of
the can (no AL2O3 bottom piece). The diameter over
the Ceramabond was 32 mm.

In the first run up with OP 38, the temperature was 1090°C and
power 0.71 W/K with still around 18 K temperature gradient upwards.
Then in attempt to get a better gradient some instability was experienced,
as may be the core cracked at this stage. When settled down again,
at 1080°C & temp gradient 20 K upwards, power was 0.77 W/K & OP 38.0.
The resistance of the top winding went 2.49 → 2.37 Ω & bottom 2.72 → 3.25 Ω,
showing slight relative cooling of the bottom winding. Presumably the gas
convection is up past the bottom winding & down the inside of the core.
In reducing temp again, the temp gradient greatly increased &
power went up somewhat.

Second roundup. At same OP 38, temp went up similarly to 1016°C
but with more upward temp gradient. At this stage, the top winding was
much hotter (3212°C) than the bottom (1855°C). By reducing the top
power slightly (570/350/1500 to 540/300/1050), the relative temperatures
flattened out to top 204°C, bottom 2.82°C, with slight increase
0.80 → 0.82 W/K, with somewhat reduced gradient. OP 38 → 39.

Up to 1235°C, 0.78 W/K with 2.82°C top & 1.87°C bottom. On cooling
down, OP 42 → 238, temp dropped substantially to 857°C, low gradient
with top 1.94°C bottom 2.54°C, when I changed settings unit & the
relative temperatures flipped again to 3.9°C, 1.78°C, power 0.78 → 0.80 W/K
& increased T gradient.

Third run up. Still OP 38, up to 1058°C again. Finished up at
1287°C, large T gradient upwards, OP 46.0, 0.80 W/K.
On dropping back to ~620°C & large T gradient at 3 OP 38, power 0.84 W/K
with top winding (172°C) hotter than bottom (166°C).

When tried to increase T again, temp dropped strongly at first &
then went to ~580°C at 0.97 W/K × OP 44.0 — become grossly
inefficient.
On removal, no moisture. Crack in core at top of bottom winding.
Layer 3 to come back & give smooth wall to AL30

Layer 2 to here to cover wires before bending down tails.

Layer 1 to here. Then cut grooves.

Layer expansion here

Dun expansion here

Dun expansion here

10 0.5 0.5 0.5 0.5 0.5

ID φ21 φ23 φ25 φ28 φ25

22 24 27

Notes:
- Layer 3 should be smooth
- Layer 2 to cover wires
- Layer 1 to cut grooves
Metal-lined Core Design

Evidently eliminating the 30 grooves in the core does not eliminate the tendency to crack at the top of the bottom winding, which must be due simply to thermal stress independent of any stress concentration. Indeed, taking the tensile strength at 1200°C as 12.5 MPa (CRC Haz. Sci. 7th ed. Table 4.06) & E as 275,000 MPa, the strain at failure is 0.004 corresponding to 55°C temperature differential. It is hard to imagine such a temperature differential along the core. There will be a general expansion axially of the core under the windings but this will be less in the gap between the windings so there will be a tendency for tensile stress to develop there.

The high hardness developed by the Ceramabond 369 cement suggests the possibility of using it to cast a metal core, or which contain the windings could be cast, as shown at left. There would be an advantage to have single-ply tails instead of 3-ply. This could possibly be achieved by justing temporarily tied at the ends of the coil, removing them after cementing & patching up the gap or pre-winding the wire and collapsing it into pre-machined grooves. The cementing stages would be:

1. Layer of cement built up to about 1 mm thick & 0.5 mm deep grooved in it — or build up to 0.7 mm thickness & cut shallower grooves, in each case have 0.5 mm of cement between groove bottom & metal.

2. Layer of cement over the windings, up to about 1.5 mm total thickness to insulate the windings before bending down the tails.

3. Final layer of cement to cover tails, etc. & Thermocouples (Thermocouples could be bare)

(The wouldn’t work because of acting.)
\[ WIR \ 260 \ \text{D} \ 21 \ \text{ID}, \ \sigma = 185 \ MPa \]

2300 \ \text{D} \ 80 \ \text{ID} \ 22 \ 41 \ \text{ID} \]
Furnace core design - cracking considerations

The cracking almost invariably occurs at the top of the bottom winding where the heat generation is a maximum. The heat generated in the bottom winding is around 500-700 W, say 600 W at 1600 K. Part of this heat will be propagated outwards, part of it through the core wall to the specimen assembly.

The amount of heat conducted down the piston assembly if all H2O is calculated (p 24) to be ~ 50W at 1700 K. This is small compared with the heat generated in the bottom winding. There is probably some heat convected upwards in the gas also. But it is hard to imagine more than 100 W being required to flow upwards from the bottom winding.

If we assume 100 W, then the average flux \( \frac{q}{A} = \frac{100}{20} = 4.8 \text{ W/m}^2 \) or 14300 W/m. However, there will be a gradient upwards in the heat generation due to temperature effect of on the H2 resistive. If this is linear with \( T \), then the heat flux density at the top will be double the above, say 10 W/m².

\[
\frac{d}{h} = \frac{2\pi K \Delta T}{\ln \frac{D}{d}} \quad \text{or} \quad \Delta T = \frac{2\pi K d}{D}
\]

For \( D = 25 \), \( d = 2 \) and \( K = 5 \text{ J m}^{-1} \text{K}^{-1} \), \( \Delta T = \frac{10 \text{ J m}^{-1} \text{K}^{-1}}{10 \pi} = 3.2 \text{K} \)

This will give rise to a stress difference between OD \& ID of the core tube of \( E \Delta T \Delta d \) where \( E = 350 \text{ GPa} \) at 1300 K \& \( \alpha = 7.9 \times 10^{-6} \text{ K}^{-1} \).

\( \sigma = 350 \times 10^9 \times 3.2 \times 7.9 \times 10^{-6} = 152 \text{ MPa} \)

The tensile strength is ~ 200 MPa or so, so will slightly higher.

The cracking is quite feasible.

Reducing the wall thickness would reduce the thermal stress, e.g. reducing \( 25 \rightarrow 23 \text{ mm} \) would result in approximately halve the stress.

Another possible factor reducing the chance of cracking is the use of alumina cement to embed the windings. This cement probably shrinks as it sinters \& so applies a compressive longitudinal stress in the core tube, to some extent counteracting the tensile thermal stress.
Questions of outer ALTi sleeve

Recently we have given up the ALTi sleeve between the AL30 insulation and the SS can. What effect?

Total heat dissipation in the furnace has been assumed to be 0.5 W/K measured in machine PI10 i.e. 0.5 W/K when corrected to free surface (info from Mark on test on furnace no. 40) 10 at ~ 1470 K, power is 750 W. If we assume 50 W is conducted each way in the piston assembly, the total outward heat flow is ~ 550 W and an effective furnace length of, say, 70 mm, i.e. \( t = \frac{630}{0.07} = 9300 \) kW m s⁻¹

So effective thermal conductivity \( K = \frac{\frac{31}{28} \ln \frac{61}{28}}{2 \pi \cdot 0.96} = 0.96 \) W m K⁻¹

so the effective thermal conductivity of the AL30 insulation is around 1 W m K⁻¹.

For comparison, Roger lists \( K \) as 1.0 at 20°C & 2.0 at 1100°C so the effective \( K \) of AL30 is about the same as ALTi at the cool outer part of the furnace. Therefore we should not increase the heat loss by substituting ALTi for the outer layers of the AL30 & there may even be some gain if the convective losses in the AL30 were reduced.

\( \Delta T = \frac{9300 \ ln \frac{61}{28}}{2 \pi \cdot 0.96} = 925 \) instead of 1200 at diameter 51.

The (D) of the ALTi sleeve is so the \( \Delta T \) driving convection in the ALTi is reduced to \( \frac{925}{1200} \) of what it was = 0.77. If the heat loss were half convection, half conduction, then the total heat loss would be reduced by 10% compared with no ALTi sleeve. Actually, the \( K \) for AL30.4 ATT is listed as around 0.2, so there could be a reduction of somewhere 20% in heat loss.
Convection Considerations

The convection within the A330 is likely to occur mainly in the zone EF to BG. It is not clear to me whether this would form a single convection cell or break up into a number—probably doesn't matter much. However, the strength of the convection will be sensitive to whether there is a chimney along the AE or HT interfaces.

The HT interface should probably be a light push fit. When installing the ALT, slip over the AL30, after the latter has been installed over the core.

The core should be finished with a smooth OD, as shown opposite p.326. Then the AL30 is fitted over the core, perhaps applying some thin cement to lubricate the assembly & give a bonded fit between the AL30 & the core cement. After this, the heads are fitted (possibly attaching them tubing as an continuation, with 0.6 ID, 16 OD) & bent into shape to come up the 0D of the AL30 in grooves of pitch diamet 50.

The grooves could be filled with cement & everything smoothed off to an OD of 511 to be a light push fit in the AL31.

Then the ALT sleeve is pushed on to the AL30. Finally the SS casing is shrink on to the complete assembly.

If the core is made of AlD3, it can be a snug fit in the ALT for closure piece. The difference in coefficient of thermal expansion $\Delta \kappa = 6 \times 10^{-6}$. The temperature along the core-AL31 interface probably varies from 300 to 100°C, so the differential expansion between aluminum core & ALT will vary from around 0.11 to 0.014" (3 strain from $\pm 0.005$ to 0.001). We could probably give the core a convenient assembly clearance of 0.03 to 0.05 mm & depend on the shrinkage going into compression on expanding. Thus if we have a SS inner sealing sleeve, it will have a $\Delta \kappa$ of $38.15^6$ & so a new differential expansion of about 0.13 mm & we would need to allow a clearance on the OD of the SS sleeve of say 0.1 mm in order not to fracture the aluminum core sleeve. Maybe it could be 0.08 clearance if some support from the AL31 is effective. Or make it with two or three OD's, eg. 20.97, 20.93, 20.88 (or tapered)
For large-bore furnace with standard wind core 271D 30 OD, if we wind in shallow grooves with pitch \( \phi \), then the OD over the cement will be \( \phi + 5 = 35 \).
Core design for thin-walled alumina core tube

If we get alumina tubes from Regain with 0.5 mm WT, the OD will be 22 mm.
- Degree of tube:
- Apply 0.5 mm thickness of cement to machine 0.25 mm deep grooves in this to locate the windings.
- Apply the windings & secure the ends with 1/0 wire.
- Cover the windings & build up to 26 (This will give about 1.5 mm insulation over the windings).
- Bind on the tails and attach the thermocouple.
- Cover the whole assembly to 27.5 - 28.0

The cement we have used is Aerocem Ceramotic 569. Another brand may be a possibility is Ceramotic 671. Comparison is as follows:

<table>
<thead>
<tr>
<th>Brand</th>
<th>Yes</th>
<th>No</th>
</tr>
</thead>
<tbody>
<tr>
<td>569</td>
<td>Set @ RT; good filler</td>
<td>cc, cm, probes, sensors</td>
</tr>
<tr>
<td>671</td>
<td>High adhesive strength</td>
<td>cc, cm, mm, textile, thread locking</td>
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<table>
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<th>671</th>
<th>503</th>
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<td>7.2 x 10^-6</td>
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<td>1.0 x 10^5</td>
<td>1.0 x 10^5</td>
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<td>240</td>
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<tr>
<td>Toughness, ft lb</td>
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<td>240</td>
<td>5.6</td>
</tr>
<tr>
<td>Viscosity, cps</td>
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<td>44,000</td>
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<td>SG</td>
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<td>2.24</td>
<td>2.50</td>
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<tr>
<td>Air set (hours)</td>
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<td>1.4</td>
<td>&lt;1</td>
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<td>Heat cure °C, hrs</td>
<td>93°C, 2 hrs</td>
<td>93°C, 2 hrs</td>
<td>93°C, 24 hrs</td>
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<tr>
<td>Shelf life</td>
<td>6 months</td>
<td>6 months</td>
<td>6 months</td>
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</tbody>
</table>

There is also cement 503 (above) for CC use, mentioned for bonding Pt/40 Rh heater wire on an alumina core or for providing oxidation & corrosion protection at 1700°C. The 503 has higher dielectric strength than the others, while the volume resistance of the 671 is lower. The 671 seems to have the highest strength. 503 will not dry at RT & has to be skip cured at 93, 260 & 375°C for 1-2 hrs at each temperature.

671 or 503 are preferred over 569 for bonding alumina.
Case where AP is below confining pressure.

Case where AP is above CP.
Drive Pressure Connection to Internal Alternation Cell

There is a problem with using the normal venting assembly in the internal load cell for operating the internal alternation cell.

Case 1: The pressure differential loading the specimen is below confining pressure (CP). Here the force applied to the specimen is extensible & is proportional to AP, \( (7)^2 \) where AP is the pressure differential. But the force transmissible across the seal on the vent assembly is proportional to \( AD^2 \). Therefore, the loading assembly will tend to lift off the vent assembly.

Case 2: The pressure differential that loads the specimen \( > CP \): Now the force in the specimen is compressive load is proportional to \( AP \), which is supported by the internal load cell. However, the pressure across the interface to the vent assembly is now greater than \( CP \) & so this interface will blow open.

So we need a double connection through the vent assembly, one for atmospheric venting & one for the high pressure operating the alternation cell. If we introduce a second small diameter O-ring seal as at left, the pressure in this seal area is \( > CP \). But if we vent the area surrounding it to atmosphere & this area is greater than the small seal area, then the pressure on this area will have a force across it from the confining pressure which is greater than the force from the AP fluid from the alternation driver. Such a system should allow operation either above or below the confining pressure.
Revised design for thin-walled alumina core tube furnace.

In a first run on the design described on p. 330, a failure occurred at a little above 800°C (1100 K), as described in SUNDAY BOOK D P158. The reason for the failure is presumed to be arising between the centre top winding leads & the bottom winding due to accumulation of carbon in the potshaped cement. The design depended on retaining insulation in a layer of cement about 1 mm thick, which seems to be inadequate. Also, the efficiency of the furnace is poor. So I now propose that we try again with a 0.5 mm wall thickness tube and only use cement to cover the windings minimally. Then bring down the tools & thermocouples in alumina tubes.

For insulation, we should revert to the arrangement in the original furnace 40, viz. segmented AL30 A111 with alumina paper in the segments, bonded against the core with tamped alumina paper.
High-sensitivity IF - IF core design & hysteresis

The 10N 12C turned out to have very serious hysteresis, of the order of 20-30% of FS. We tracked this down to being mainly associated with the cores in the IF LVDT's.

As shown in drawing PI 4603, the retaining screw has an $\phi 2.9$ in a hole of $\phi 3.0$, so there is a possibility of the core assembly tilting over & contacting the box of the LVDT if there is a close fit on the M3 thread.

So the core assembly needs to be re-designed for a $\phi 3.0$ shaft that can be fitted to the $\phi 3.0$ hole.

This is shown in drawing PI 4607A.
For ideal gas, $PV = RT$, $\nabla V + \rho dV = 0$, \( \rho \beta = \frac{-1}{V} \frac{dV}{dp} = \frac{1}{p} \)

ie $\beta = 10^{-5}$ at $p = 10^5$ Pa (strength pressure)

$= 10^{-8}$ at $p = 10^8$ Pa $= 100$ MPa, ie compressibility is down to $\frac{1}{3}$ of ideal gas at $\sim 100$ MPa.

$80^\circ_{V}$ $\ldots$ $50^\circ_{V}$

$\Delta V = \rho \nabla V \Delta p$
Attenuation Noise Performance

There was no signal (or highly small) in the S X E readings when operating the attenuation module at (320, 321). After removing the 2:01 fixture first mechanical advantage, still no signal. It is evident that:

1) The gas at ~100 MPa is still too compliant
2) The flow volume in the pipe connections exacerbates this situation.

At a very rough estimate, there is at least 3000 m<sup>3</sup> volume in the pipe connections.

Compressibility: \( \frac{\Delta V}{V} = \frac{P}{K} \) where \( \beta = \) compressibility
\[ K = \text{bulk modulus} \]

For argon, \( \beta = 8 \times 10^{-9} \text{ Pa}^{-1} \) at \( P = 50 \text{ MPa} \)
\[ 8.5 \times 10^{-9} \text{ Pa}^{-1} \]
\[ 1.5 \times 10^{-9} \text{ Pa}^{-1} \]

Thus if we have a volume \( V = 3000 \text{ m}^3 \)
then \( \Delta V = 1200 \text{ m}^3 \) at 50 MPa
\[ 1050 \text{ m}^3 \]
\[ 900 \text{ m}^3 \]

However, we are only concerned with the \( \Delta V \) associated with the pressure amplitude needed to drive the attenuation actuators. At \( 10^{-4} \) strain amplitude, strain amplitude = \( 10^{-4} \times 100 \times 10^{-9} \text{ Pa} \text{ i.e. } \Delta S = 10^{-7} \text{ Pa} \text{ (10 MPa)} \). The diameter of the driven piston is the actuator is \( 18 \text{ mm} \). The diameter of the specimen is \( 25 \text{ mm} \) so the pressure amplitude in the driver is
\[ 10 \times \frac{\Delta S}{\Delta S} = 193 \times 10 \text{ MPa} \approx 19 \text{ MPa for } 10^{-4} \text{ strain amplitude.} \]

Thus at \( 10^{-4} \) strain amplitude, the \( \Delta V \) amplitude in the driver is
\[ 8.1 \times 10^{-9} \times 19.3 \times 10 = 456 \text{ mm}^3 \text{ at 50 MPa conf. pressure} \]
\[ 3.5 \times 10^{-9} \times 19.3 \times 10 = 86 \text{ mm}^3 \]

However, the maximum displacement amplitude of the driver unit (\( \Phi 3 \text{ piston x } 10 \text{ stroke} \)) is 70 mm.

So at 100 MPa, the maximum strain amplitude achievable would be \( 70/200 \times 10^{-4} = 3.5 \times 10^{-5} \approx 3.5 \times 10^{-5} \text{ m} \)

At \( 3.5 \times 10^{-5} \) strain amplitude on 100mm Al should give \( \Delta L = 0.0035 \text{ mm} \text{ or } 3.5 \mu \text{m} \) about 1/20 of the range of the LVDT. This should be discernible.

Similarly at \( 3.5 \times 10^{-5} \text{ str } \Delta L = 2.8 \text{ MPa on } \Phi 25 = 1.374 \text{ N(1.37 kN)} \text{ also measurable.} \)
It is therefore a little surprising that we did not see a signal.

Kerosine vs water:

For kerosine, $K = 1.62 \times 10^9 \text{ Pa}^{-1}$, $\beta = 0.62 \times 10^{-9} \text{ Pa}^{-1}$ (Key Algebra p55)

water, $K = 2.0 \quad \beta = 0.5 \times 10^{-9} \text{ Pa}^{-1}$ (""")

The compressibility will be a little less at high pressures. Thus compared with argon at 100 MPa, kerosene is 5.6 x less compressible. (It has about 100 x the compressibility of iron). This would only contribute 5.6 x the strain amplitude in the specimen for the same displacement amplitude in the drive, not a spectacular gain. (The factor would be 7 x with water).

However, at 50 MPa confining pressure, kerosine is about 13 x less compressible than argon, so this gives an order of magnitude advantage to kerosine.

So it is strongly worthwhile to use kerosine in the drive. However, it will still be very important to keep the free volume to a minimum.
\[ Q^{-1} = \tan \delta \approx \sin \delta = \frac{\Delta E}{2 \varepsilon_0} \]
when \( \varepsilon \) is strain.
\( \varepsilon_0 \) is strain amplitude.
\[ wt = 0: \quad \sigma(t) = 0, \quad \varepsilon(t) = -\varepsilon_0 \sin \delta - \varepsilon_0 \delta \]
\[ wt = \pi/2: \quad \sigma(t) = \sigma_0, \quad \varepsilon(t) = \varepsilon_0 \cos \delta - \varepsilon_0 (1 - \delta^2/2) \sigma_0 \]
\[ wt = \pi: \quad \sigma(t) = 0, \quad \varepsilon(t) = \varepsilon_0 \sin \delta \]
\[ wt = 3\pi/2: \quad \sigma(t) = -\sigma_0, \quad \varepsilon(t) = -\varepsilon_0 \cos \delta \]

Width measured along \( y \) axis is \( 2\varepsilon_0 \sin \delta \)

Width measured along \( x \) axis is \( 2\varepsilon_0 \sin \delta \)

Applied Stress \( \sigma(t) = \sigma_0 \sin \omega t \)

Resulting strain \( \varepsilon(t) = \varepsilon_0 \sin (\omega t - \delta) = \varepsilon_0 (\sin \omega t \cos \delta - \cos \omega t \sin \delta) \)

Work done \( W = \int \sigma \, d\varepsilon \)

So energy maximized per cycle \( \Delta E \) is

\[ \Delta E = \frac{\varepsilon_0 \sigma_0}{2} \int_0^{2\pi} \sin \omega t \cos (\omega t - \delta) \, d(\omega t) \]
\[ = \frac{\varepsilon_0 \sigma_0}{2} \int_0^{2\pi} \sin \omega t \cos \delta + \sin \omega t \sin \delta \, d(\omega t) \]
\[ = \frac{\varepsilon_0 \sigma_0}{2} \int_0^{\pi/2} \cos \delta \sin 2\omega t + \sin \delta (1 - \cos 2\omega t) \, d(\omega t) \]
\[ = \pi \varepsilon_0 \sigma_0 \sin \delta \]

Max energy stored \( E \) is

\[ wt = \pi/2 \]

\[ E = \int_0^{\pi/2} \sigma \, d\varepsilon \text{ (in phase)} \]
\[ = \frac{\varepsilon_0 \sigma_0}{2} \int_0^{\pi/2} \sin \omega t \cos \omega t \cos \delta \, d(\omega t) \]
\[ = \frac{\varepsilon_0 \sigma_0}{2} \int_0^{\pi/2} \sin 2\omega t \, d(\omega t) \]
\[ = \frac{\varepsilon_0 \sigma_0 \cos \delta}{2} \]

\[ \delta = \frac{\Delta E}{2\pi E} = \frac{\pi \varepsilon_0 \sigma_0 \sin \delta}{\varepsilon_0 \sigma_0 \cos \delta} = \tan \delta \Rightarrow \sin \delta = \frac{\delta}{2\varepsilon_0} \]
Re-design of axial actuator in internal attenuation cell indicates that in cyclic loading using the external axial actuator there is a substantial phase lag in the force registration in the ICC relative to the micro-displacement measured in the ICC. It therefore seems desirable to get rid of all potential friction sources in the internal axial actuator. For O-rings etc. which suggests going to a ballon type such as shown at left.

Adapting we allow for 10% strain amplitude in a specimen 50 mm long and 25 mm diameter.

Displacement = 10% * 50 = 0.005 mm

Force will be 10% * 100,000 = 1000 N

Displacement in microns = \( \frac{0.005}{50 \times 10^{-6}} = 100,000 \) µm

So total displacement = 0.0075 mm if the specimen modulus is 100,000 MPa (100 GPa)

If the specimen modulus is 50,000 MPa, total displacement is 0.0125 mm = 12.5 µm.

If the diagonal gap in the internal axial actuator is 54 µm, then for a 5kN force in the specimen, we need a driving pressure \( p \) for which \( \frac{\pi}{4} \times 54^2 \times \frac{5kN}{2500} \times \frac{1}{2} = 2.2 \text{ MPa} \).

If we treat the diagonal as a plate under uniform pressure, then from Eqs. p 47 (case 1) we have

\[ \sigma = \frac{0.5 \times (54/2)^2 \times 2.2}{t^2} = \frac{802}{t^2} \]

or \( t = \sqrt{ \frac{802}{\sigma} } = 1.4 \text{ mm} \)

\( \sigma = \frac{200 \text{ MPa}}{t} \)

\( t = 2.5 \text{ mm} \) for \( \sigma = 120 \text{ MPa} \).

These figures would apply if then using no specimen and \( \Delta \) internal axial actuator were given a driving pressure of 2.2 MPa enough driving pressure to load a specimen to 5000 N. However, in the presence of a specimen, the assembly will be much stiffer but the force would still be transmitted to the specimen.
Further note 23/6/04 on hydraulic internal actuators.

The force transmitted to the specimen will consist of the pressure load. In mining, the force need to deflect the “tunnel.”

As noted p.336, the force generated by the pressure differential of 2.2 MPa (for 15° strain) should be 5 kN. This is to be offset by the force to deflect the 2 mm thick plate by, say, 0.0125 mm. There are two plates, so deflection is 0.00625 mm each. They can be regarded as built-in at the centre (no change in slope) at the outer edge prevented from rotating (case 6, Exercise 14.48). So

\[ \text{deflection } y = \frac{0.1 \times P \times 2.7}{200,000 \times 2^2} = 0.00625 \]

\[ P = 137 \text{ N} \]

This will not be serious relative to the 5000 N force generated by the differential pressure.
Treatment of hollow plate, case 19.4 p. 448 (Smith): If we take a ratio of 5:1 (0.5 mm 0, 10 mm i.d.) then $K_4 = 1.5$, $M_y = 0.5$ and the deflection $S$ will be the same as above but the plate stresses will be three times higher. So if $P = 500$ kPa we have $t = 2$ mm and $S = 0.37$ mm (total deflections with two plates is $0.74$ mm).

So far we have considered the case of loading by the differential driving pressure. There are two other cases:

1. The differential driving pressure but an axial load applied by the external axial actuator of $F$. This constitutes the case 3 p. 447

   \[ S = \frac{1.365}{\pi} \frac{F}{E} b \frac{t^2}{G} \]

   Taking $t_0 = 5$ mm, $d = 10$ mm we have
   \[ \sigma = \frac{0.7327}{F} \]
   \[ S = 0.000602 \frac{F}{t^3} \]

   For $F = 10$ kN, $\sigma = 1832$ MPa, $S = 1002$ mm

   It will close up when $0.00081 F = 0.5$ i.e. $F = 5000$ N = 5 kN. Then the axial force will be supported by the 0, 10 middle piece. If the stress in FEA is 400 MPa, the extra load supported will be 31 kN.

2. Confining pressure applied with zero driving pressure. Now the 0.5 mm gap will close up progressively from the inner diameter until the remaining gap can support the pressure. This again constitutes case $K_4 = 1.5$ p. 448 but with $P_g$ very near to 1.

   The formulae are probably no longer applicable and the pressure load will be supported mainly by shear in the plate, i.e.

   \[ P \sqrt{\frac{y^2 - d^2}{d}} = \frac{C}{t_0} \]

   With $P = 300$ MPa and $t = 2$ mm, $C = 400$ MPa, we get $d = 49$ mm, i.e. only an annulus of 2.5 mm left not closed. The shear strain will be $0.525 = 0.2$ and so the elastic strain would be 0.29 = 16.4 kPa, way beyond the yield stress. Therefore application of 300 MPa without driving pressure will permanently deform the ring-gear. However, the residual elasticity may cause enough rebound so that it will still be usable.

Conclusion: Make $t = 2$ and $d = 10$
From Wikipedia, in a closed magnetic circuit the force is

\[ F = \frac{\mu^2 N^2 I^2 A}{2 \mu_0 L^2} \]

- \( \mu \) = permeability of magnetic material \( 6.28 \times 10^{-3} \)
- \( \mu_0 \) = permeability of free space \( 4\pi \times 10^{-7} \)
- \( N \) = no. of turns
- \( I \) = electrical current
- \( A \) = area of pole face
- \( L \) = length of magnetic circuit \( 0.1 \text{ m} \)
LVDT - optimum displacement of cores

- summary of behaviour

- Multi-range LC - further check on elastic element no. 11
  - element no. 7 in tension

- Some furnace history
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- Fiber-ceramic insulation for ALTI-2 furnaces (see also 310)

- Simple force calibrating bars

- Washed washers for ALTI-2 furnaces

- Choice of insulation

- Multi-range LC - dog-ears on anvil piece

- 1" 1/2 core-tube problem

- Re-examination of Intensive Gas Cylinder design

- Seal region in Brown Intensifice

- Attenuation module for HT PT

- Furnace 160 (Minneapolis) test

- Furnace cores without grooves

- Metal-lined core design

- Furnace core design - cracking considerations

- Question of outer ALTI sleeve

- Convex considerations

- Design for thin-walled aluminum core tube & winding

- Torque pressure connection to internal attenuation cell

- Revised design for thin-walled aluminum core tube furnace

- High sensitivity LC - IF core design & hysteresis

- Attenuation drive performance - fluid compatibility issues

- Re-design of internal axial actuator for attenuation module

- Electromagnetic actuator for attenuation module
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Small LVDT Output with 5Vac RMS @ 10kHz Applied to Primary Winding
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</tr>
</tbody>
</table>
RMS @ 10kHz Applied to Primary Winding

3.1 VAC/mm per 5VAC
or 0.625 V/V/mm
6.24 mV/V/mm