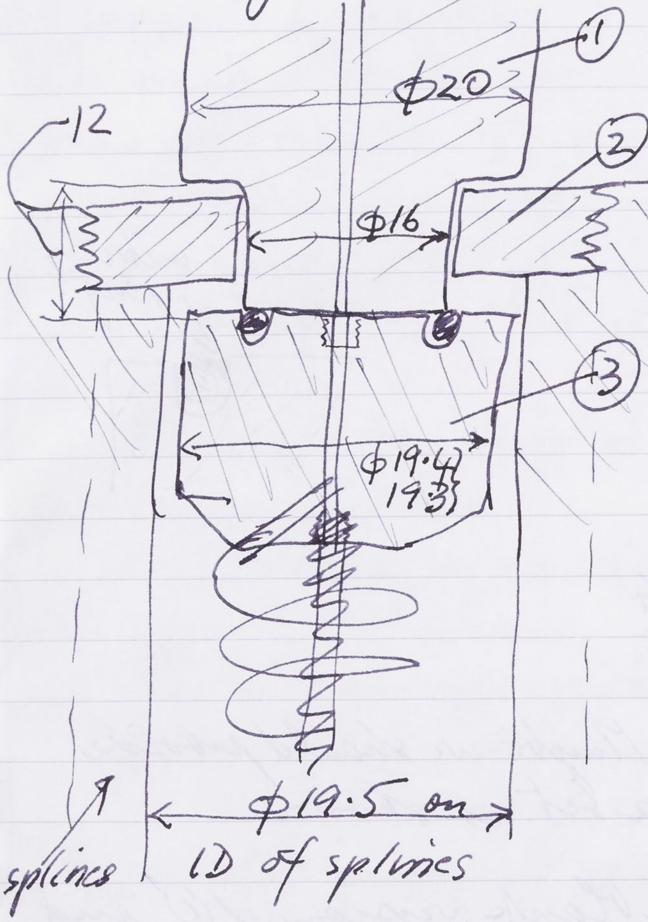
DESIGN CALCULATIONS FOR 1988 SERVOCREEP (HPT) MACHINE

③

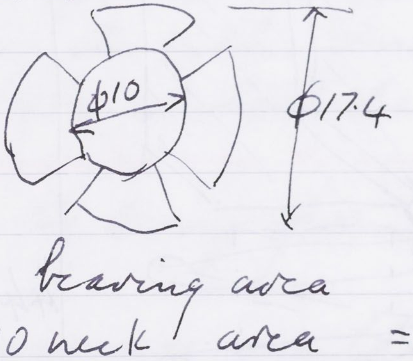
Victory
NOTE BOOK

Jan 1998

Re-design of bottom piston extension of specimen assembly, and extension arrangement, with venting



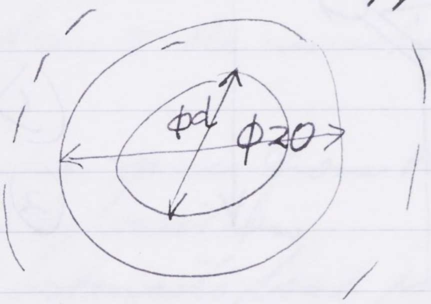
Instead of the bayonet arrangement for extension in which



bearing area 79·6 mm²

φ10 neck area = 78·5 mm²,

we would now have an annular support of area $\frac{\pi}{4}(20^2-d^2)$



If part ② is of H13 at HRC46, then $\sigma_y \sim$ 1300 - 1400 MPa.

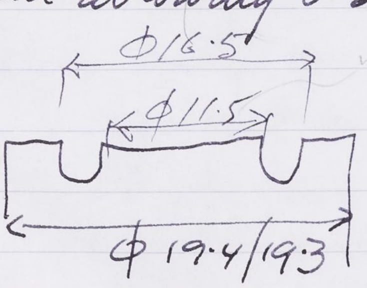
So allowing 1000 MPa at 100 kN, bearing stress is

17?

$\frac{100\,000}{\frac{\pi}{4}(20^2-d^2)} = 1000$ or d = 16·5 mm,

1260 MPa with 3mm slots for tool.

and allowing 0·5 clearance, the neck of ① can be φ16.



why not 12

Allowing 2·5 mm O-ring groove width for an 014 O-ring (id 12·4, od 16·0) section φ 1·78

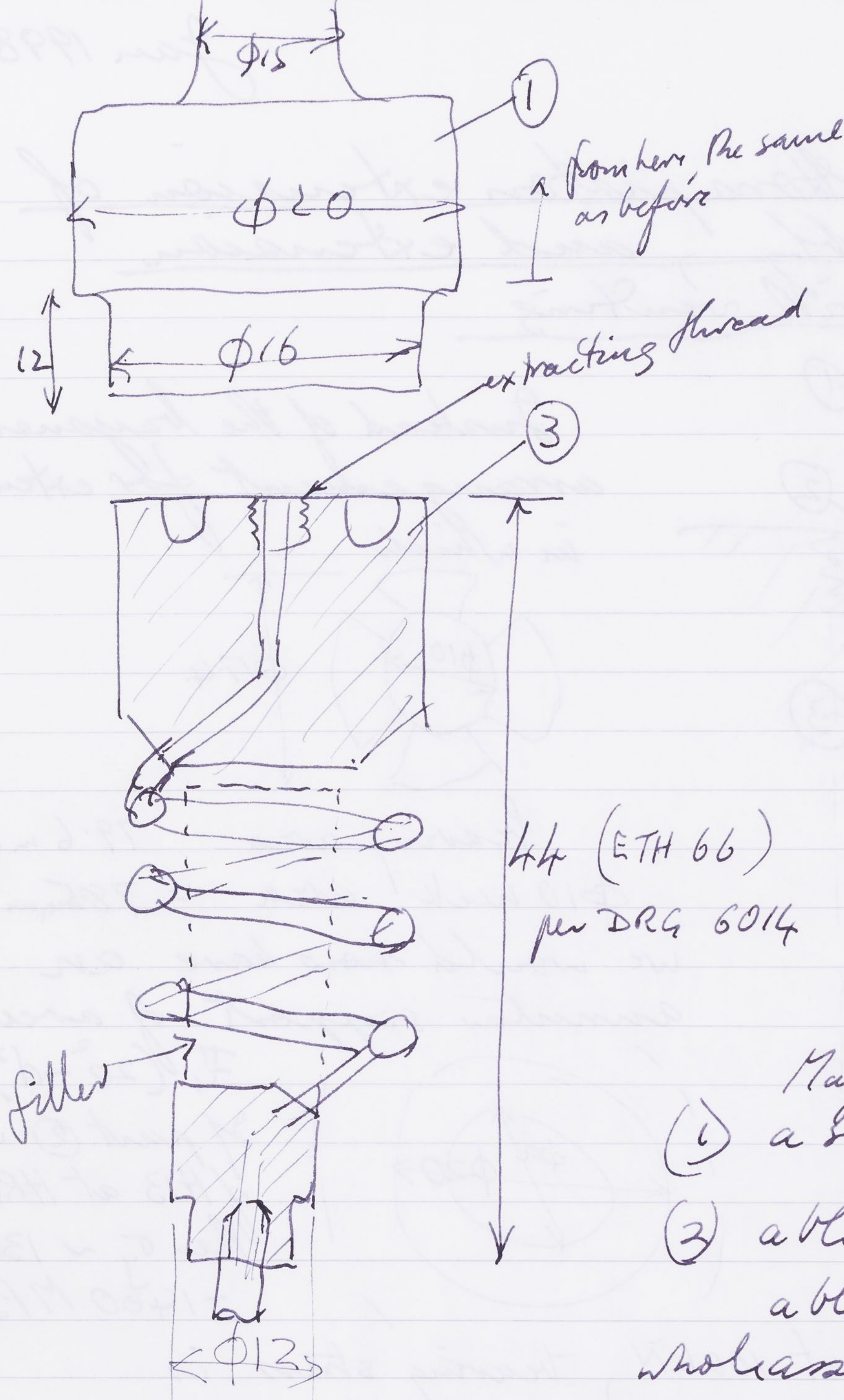
we could have a bearing area of

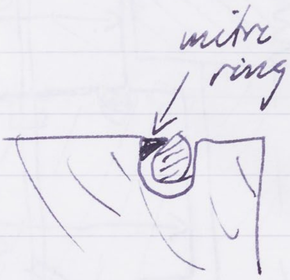
area as 2·42

but 1·3 × 2·5 = 3·25, +34% 21%

φ19·3 − φ16·5 = 78 mm² so

allowing 1000 MPa bearing stress, max axial extension load is 78000 N = 78 kN, or approx 1000 MPa on a φ10 specimen, which would need > 1 GPa gas pressure to apply. Therefore, plenty of bearing area.





Maybe we should provide
① a set as shown
② a blank version of ① and a blank substitute for the whole assembly ③

The assembly ③ could be replaced through the bore of the furnace.

The main remaining question is how much stress we can apply to a ϕ10 specimen. This will be limited to something less than the confining pressure in all cases of gripping by suction, unless dog-bone shaped specimens are used. If the contact area between ① & ③ is $\overset{12\cdot0}{\phi 11\cdot5}$, then the suction stress across it is always $\left(\frac{1}{1.15}\right)^2$ times that on a ϕ specimen end, ie about $\overset{69\%}{75\%}$. So the specimen should always pull off first, unless the rubber O-ring intrudes very prematurely & this could be controlled with a mitre ring. also by having an unsupported area to increase σ_n.

It should be easy to change the O-ring from the specimen access.

The main precaution necessary would be to keep the top of ③ clean.

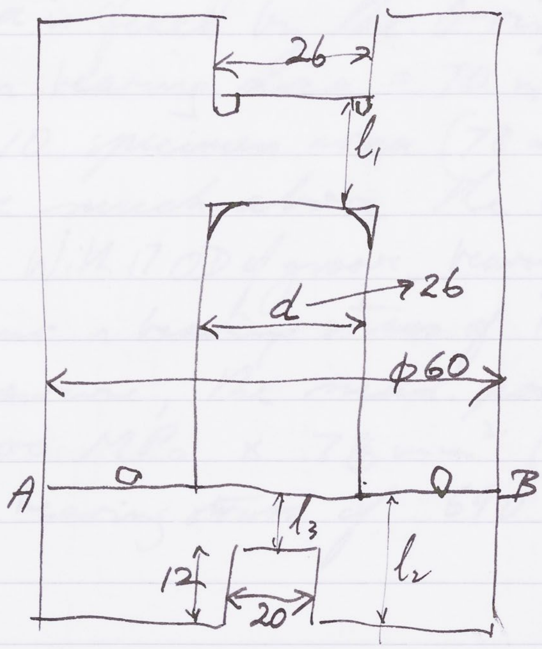
The advantages of this system would be:

1) get rid of bayonet connection & the uncertainty of getting it into the correct azimuth
2) positive venting of specimen space from both sides for best extension gripping by suction
3) even though the absolute extensile load possible is not as great as with the bayonet end, the suction load is always 33% greater than can be supported by suction on the end of a ϕ10 specimen.
4) no extra parts are needed for pore fluid connection
5) a bubbler on a tube attached to ③ will give positive information about jacket leaks in the lower half of the specimen assembly.
6) A shortened version could be used in torsion with the ϕ22 specimen assembly, but the ϕ15 spec. assembly would need a smaller top end.

This type of seal has proved to be reliable in previous pore fluid experiments.

CONTD p 164

Dimensions of possible pre-amplifier space in PPM bottom plug



Make body out of CALMAX at about 58 HRC.

0.2% σ_y in compression = 2000 MPa

so $\tau_y \sim 1100$ MPa or so.

Take $\tau_{max} = 500$ MPa.

For d:

$$\sigma_t = p\,\frac{60^2+d^2}{60^2-d^2} = 1000\cdot\frac{60^2+d^2}{60^2-d^2}$$

If we allow $\sigma_t = 1600$ MPa,

$$\frac{60^2+d^2}{60^2-d^2} = 1.6$$

or $d = 28.8$

With $d=26$, $\sigma_t = 1460$ MPa. This seems a reasonable figure, so take $d = 26$.

For shear failure at ends, $\cancel{4}\ \tau\,\pi d l = p\,\frac{\pi d^2}{4}$, $l = \frac{p}{\tau}\cdot\frac{d}{4}$

For $\tau = 500$ $p = 1000$ MPa, $l = \cancel{d/4}\cdot d/2$

so l_1 = ~~6.5 mm~~ <u>13</u> mm

l_2 = ~~6.5~~ 13 "

l_3 = ~~5~~ 10 " $\therefore l_2 = l_3 + 12 = \underline{22}$ mm

At surface AB, with 135 O-ring groove 56OD 48ID, the net bearing surface is 1643 mm² & pressure acts over φ60 = 2827 mm², ie $\sigma_n = 1721$ MPa.

With $\sigma_t = 1460$ MPa, effective stress $= \frac{1}{\sqrt{2}}\left(1721^2 + 1460^2 + 261^2\right)^{\frac{1}{2}}$

$= 1606$ MPa

probably just tolerable with everything in compression & some ductility to relieve the stress concentrations.

CONTD from p162.

Bearing area on extension piece ③ :

The OD of this piece will be 19 & the ID of the bearing area is fixed by the O-ring groove. If this is Φ16·5 (p161), then bearing area = 70 mm^2, just a bit larger than the Φ10 specimen area (78 mm^2) so the bearing stress cannot be much above the confining pressure

With 17 OD of groove, bearing area = 56·5 mm^2, which would incur a bearing stress of 1768 MPa at 100,000 N. However, the max. possible extensile load would be 500 MPa × 78 mm^2 (10 Φ specimen) = 39 kN, giving a bearing stress of 690 MPa — no problem.

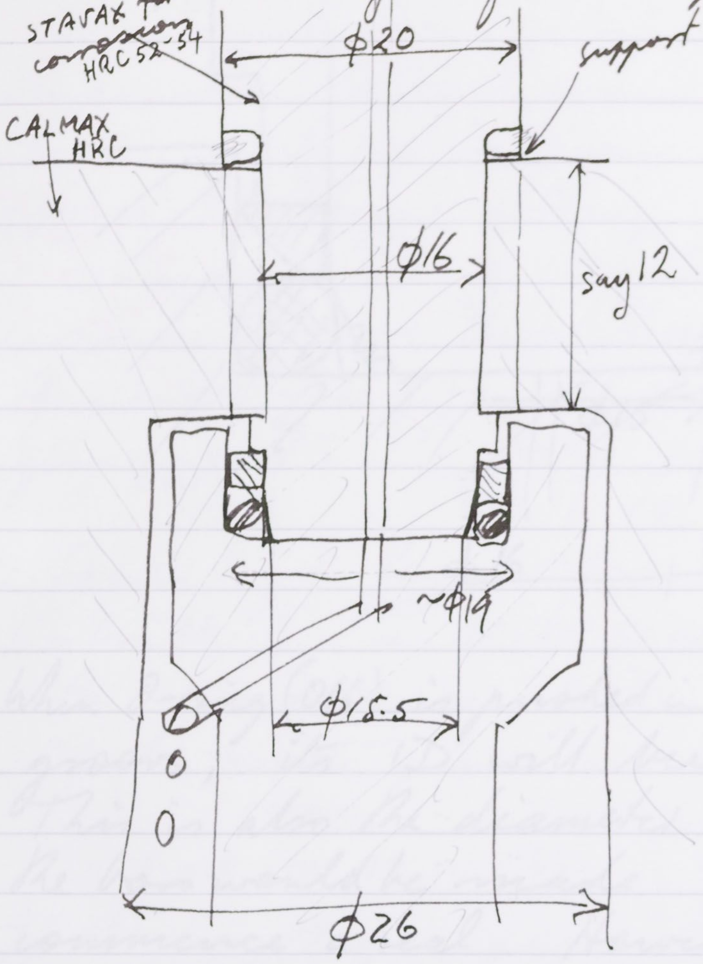
It would be possible to get greater bearing area by making the male splines narrower & the female splines wider; there is some reserve torsional strength to allow this

If we make the splines 19 OD, 16 ID, compression support area is $\frac{\pi}{4}\left(20^2-19^2\right) + \frac{1}{2}\,\frac{\pi}{4}\left(19^2-16^2\right) = 71.9\,\text{mm}^2$
At 100 kN, bearing stress = 1392 MPa, much better.

The splines would be 1.5 deep,

March 98

Further redesign of bottom piston extension for torsion



Compression support area

$= \frac{1}{2} \cdot \frac{\pi}{4}(20^2 - 16^2) = 56.5 \text{ mm}^2$

At 100 kN, bearing stress = 1768 MPa.

80 " " " = 1415 "

0.2YPt of STAVAX at HRC52 ≈ 1520 MPa so the absolute limit is about 85 kN.

Torsional strength in splines is

$\frac{1}{2} \cdot \pi \cdot 16 \cdot 0.008 \cdot \frac{\tau}{\text{MPa}} \cdot 12$ Nm

$= \overset{2.41}{\cancel{4.8}}\, \tau$ Nm

$= 965$ Nm at $\overset{400}{\cancel{200}}$ MPa stress — OK.

Extension support area $\approx \frac{\pi}{4}\left((25.5)^2 - 19^2\right) = 227 \text{ mm}^2$

so at ~~80kN~~ 100 kN, bearing stress = 440 MPa — OK.

Extension suction limit $= \frac{\pi}{4}(15.5)^2 \times 500$

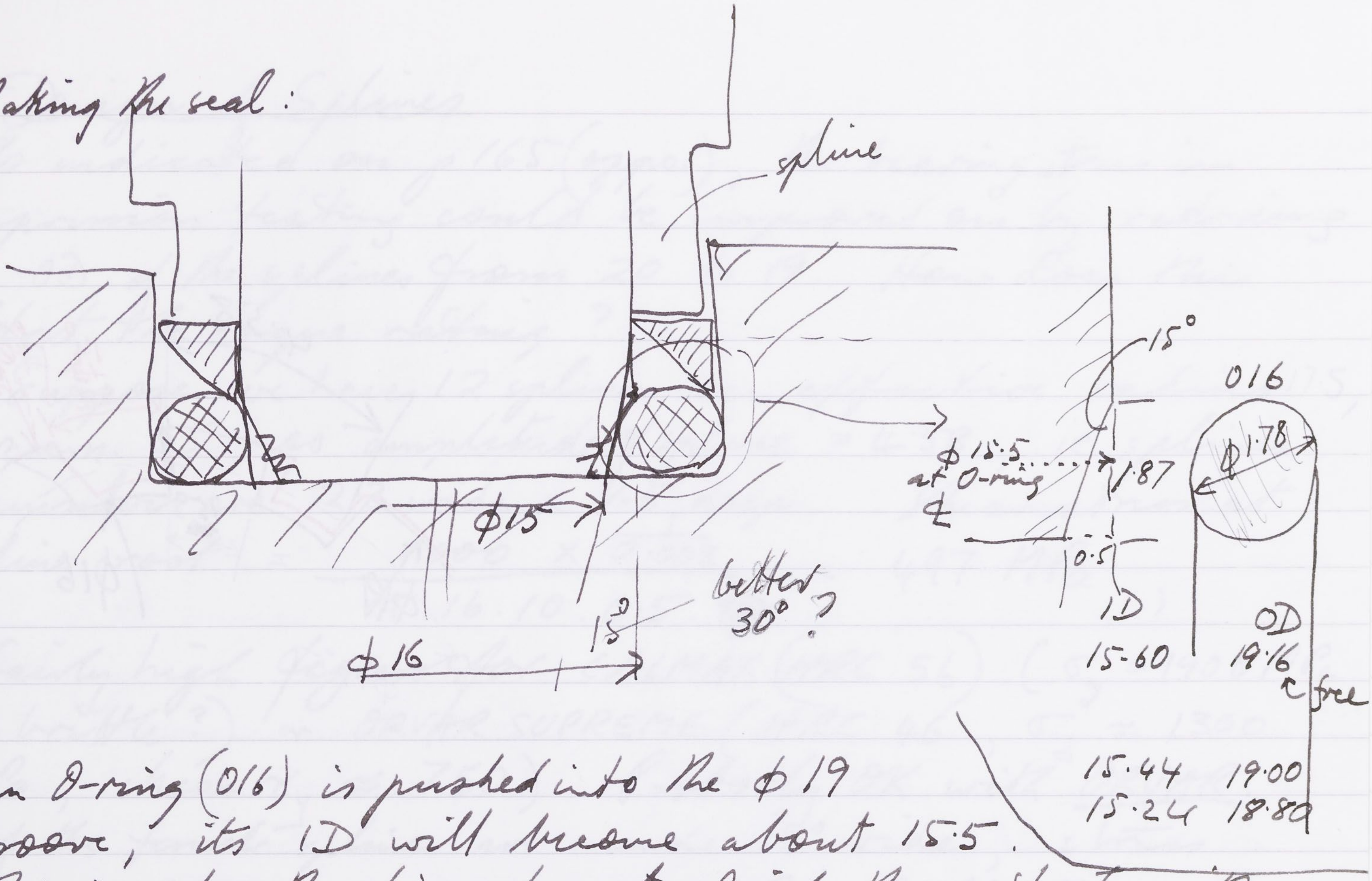
$= \underline{94,350 \text{ N}} \approx 94$ kN.

This should be ample. If we settled for 85kN, diameter could be $\left(\frac{85000}{\frac{\pi}{4} \cdot 500}\right)^{\frac{1}{2}} = 14.7$ mm.

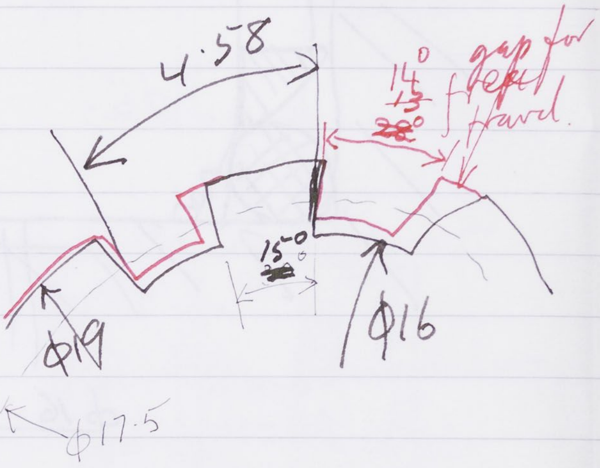
So at 15 mm, the limit is 88 kN.

If we ~~have some unsupported area under the contact area, the contact stress will be higher & so the chances of intrusion might be reduced~~ — this argument is not valid because at lift-off point, there is no contact stress to magnify.

Making the seal:



When O-ring (016) is pushed into the ϕ19 groove, its ID will become about 15·5. This is also the diameter at which the contact with the boss would be made. This may or may not commence a seal. However, the start of sealing could be assisted by preload on the O-ring from the floating mitre ring being pushed down by the bottom end of the splines as the boss comes down to seating.


4·58
14°
~~13~~
~~20°~~
gap for ~~reft~~ free travel.
15·0°
~~20~~
Φ19
Φ16
Φ17·5

Design of Splines

As indicated on p 165 (oppo.), the bearing stress in compression testing could be improved on by reducing the OD of the splines from 20 to 19. How does this affect the torque rating?

Suppose we have 12 splines on effective radius 17·5, circum. 55 so amplitude of spline = 4·58, ie splines themselves are 2·3 wide × 1·5 deep. Shear stress at spline root $= \dfrac{1000 \times 0{\cdot}008}{\pi . 16 . 10 . 0{\cdot}5 . [illegible]} = 497$ MPa
)

a fairly high figure for CALMAX (HRC 56) ($\sigma_y = 1900$ MPa, too brittle?) or ORVAR SUPREME (HRC 46, $\sigma_y \approx 1300$ MPa, shear $\tau_y \approx 750$). Probably OK with ORVAR.

If the female splines are made 13° wide, stress $= \dfrac{1000}{0{\cdot}008 . \pi . 19 . 10 . ^{13}/_{30}} = 483$ MPa. Still OK.

If the length of engagement is increased to 12, the 497 MPa comes down to 414 MPa; may help a bit although the highest stress will be at the top as the splines distort.

In practice, this level of torque can only be reached with special specimen gripping arrangements.

Bearing area on splines = 3 × 10 × 12 at r = ~~0·088~~ 0·00875

so bearing stress $= \dfrac{1000}{0{\cdot}00875 . 3 . 10 . 12} = 317$ MPa — no problem

19/Mar 98 (168)

LVDT Load Cell

Looking at the possibility of an LVDT-based ILC. Smallest RDP pressure-resistant LVDT is the D5/25K , ϕ9·5 and length 19·3 mm.

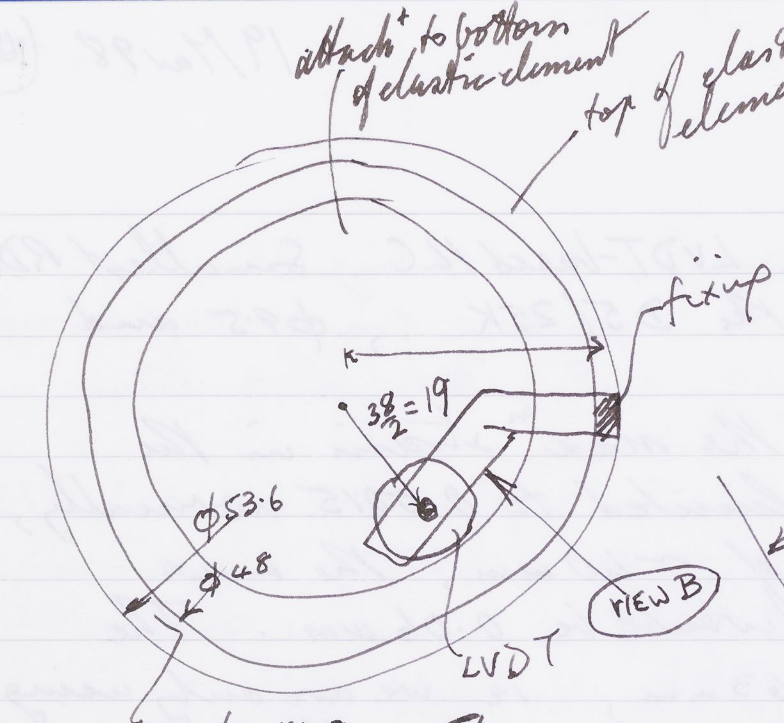
For axial deformation, the max^m strain in the elastic element should be limited to 0·0015 normally, so for an effective length of ~40 mm, the max displacement at 100 kN would be 0·06 mm. The range of the D5/25K is 0·63 mm, ie we are only using 10% of the range, from which we need finally to get a 10V signal.

The nominal sensitivity of the D5/25K is 43 mV/V at FS. so for 5V excitation the output is 215 mV at FS or ~~21~~ 21 mV at 0·06 mm amplitude. We wish to amplify this to 10V ie 476× amplification. RDP's S7AC amplifier has gain up to 500×(although its supply is only 1Vrms), so maybe this is not an unreasonable amount of gain to seek. The RS conditioner has gain up to 460×.

A slightly better choice is the D5/100K , range ± 2·5 mm and 190 mV/V at FS ie $190 \cdot \frac{0.06}{2.5} \cdot 5 = 23$ mV output for 5V excitation at full load. Need 439× gain to get 10V; within the range of the RS unit.

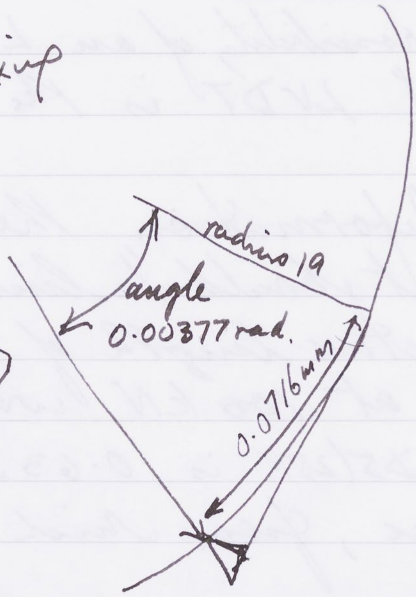
The axial load measurement is thus fairly straightforward.

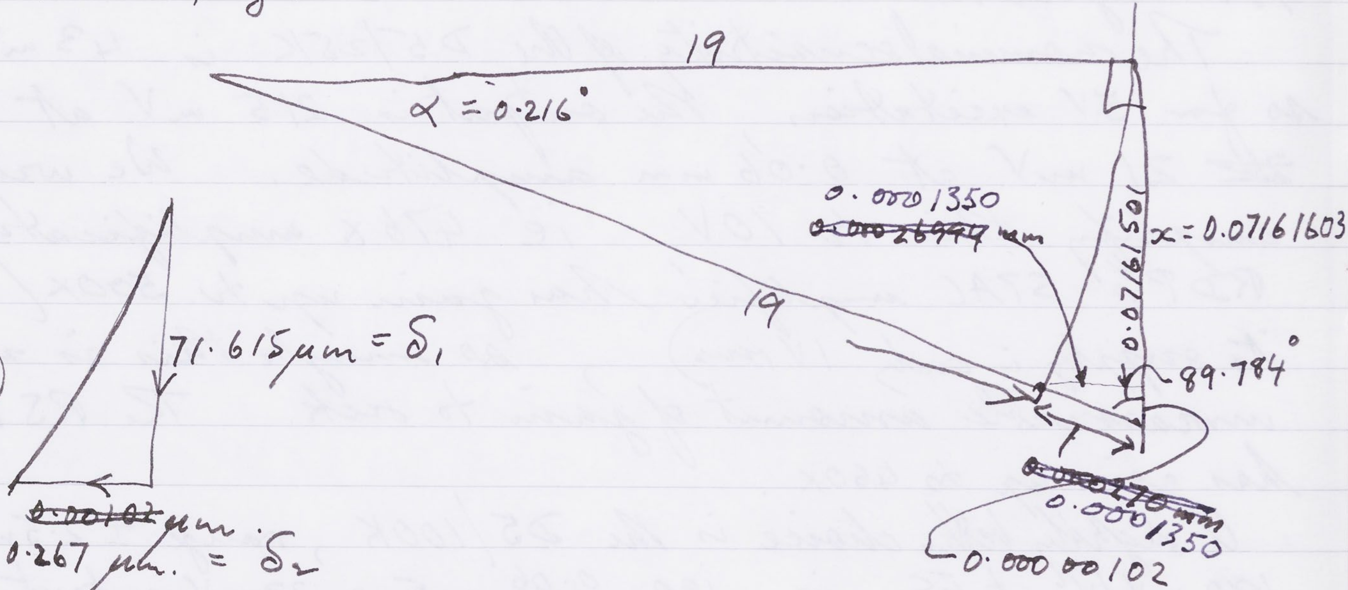
However, the ~~torsional~~ torque is more tricky. It is difficult to fit an LVDT transversely, so the torsional ~~strain~~ displacement needs to be transferred into longitudinal displacement so that the unit can be mounted longitudinally. A scheme for doing this will now be examined.



mean φ = 50.8, ~~say 51~~.

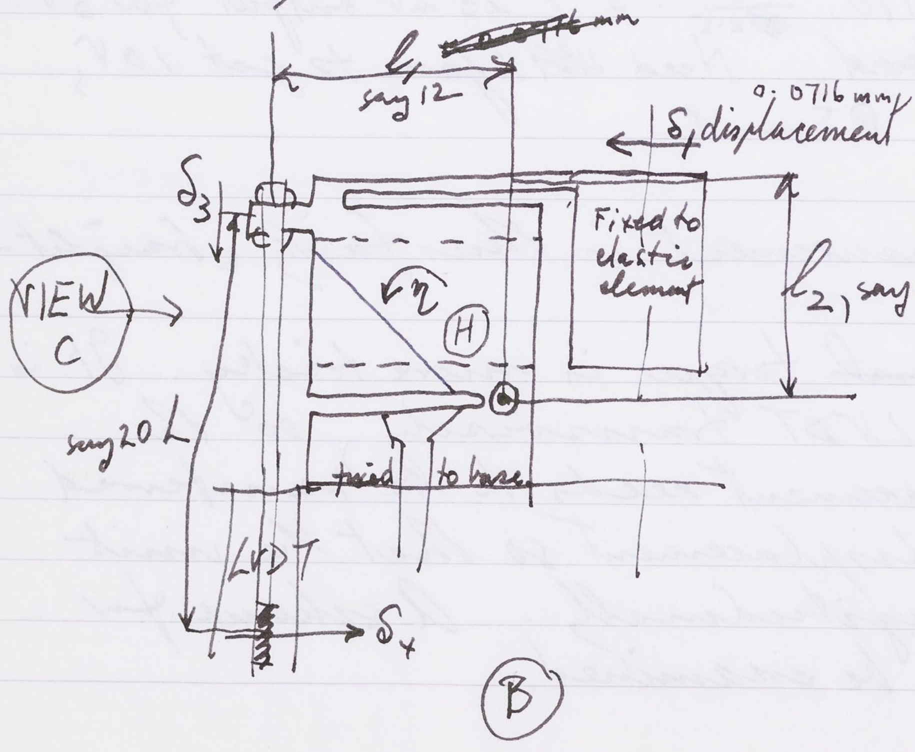
radius 25.4, say 25

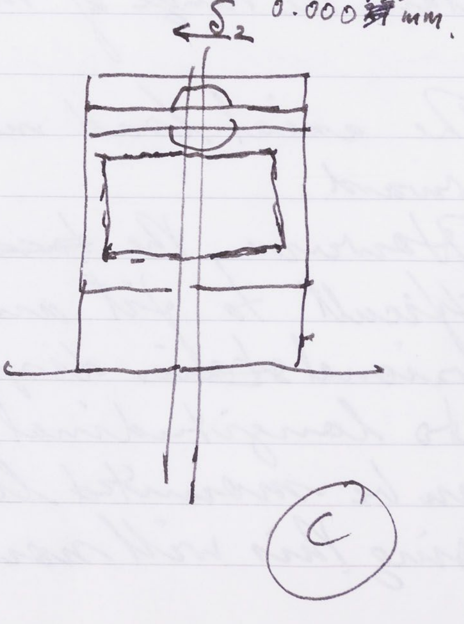




~~0.00102~~ μm.

0.267 μm. = δ₂





For torsion, $M = \frac{\pi G \theta (d_0^4 - d_i^4)}{32 l}$

so $\theta = \frac{32 l M}{\pi G (d_0^4 - d_i^4)}$

$l = 0{\cdot}040$ m $M = 1000$ N m
$G = 80.10^9$, $d_0 = 0{\cdot}0508$
$d_i = 0{\cdot}048$

$= 0{\cdot}00377$ radian
or $0{\cdot}216°$

, so $\gamma = \frac{\theta d}{2l} = \frac{\theta . 50{\cdot}8}{2 . 40}$

$= 0{\cdot}00236$ ← 60% above 0·0015

Therefore at radius 25, the ~~tangential~~ circumferential displacement at full scale is 0·0942 mm, and at radius 19, it is 0·0716 mm.

Therefore the ~~tangential~~ circumferential displacement at the LVDT on PCD 38 is 0·0716 mm. This resolves into the displacements shown in diagrams Ⓐ representing normal to & parallel to the original radius to the LVDT.

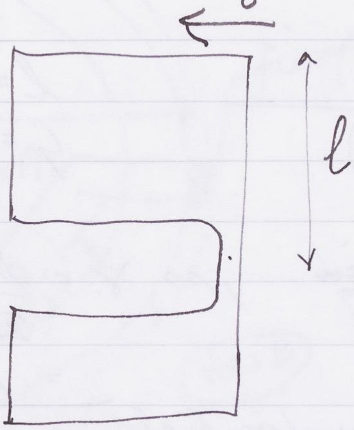
The device of diagram Ⓑ converts the ~~arc~~ tangential displacement δ_1 into an axial displacement δ_3 by rotating the block H through an angle η $\quad \eta = \frac{\delta_1}{l_2} = \frac{\delta_3}{l_1} \quad \therefore \delta_3 = \frac{l_1}{l_2} \delta_1$

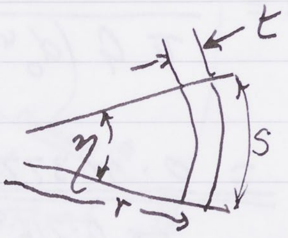
The sideways displacement of the LVDT core at support length L is given by $\quad \eta = \frac{\delta_1}{l_2} = \frac{\delta_4}{(L - l_2)} \quad \therefore \delta_4 = \frac{L - l_2}{l_2} \delta_1$

If $l_1 = 12$ $\quad l_2 = 10$ $\quad L = 20$, say, then for $\delta_1 = 71{\cdot}6\,\mu m$, we have $\delta_3 = 85{\cdot}9\,\mu m$ and $\delta_4 = 71{\cdot}6\,\mu m$.
$\delta_2 = 0{\cdot}27\,\mu m$ still.

The output of the LVDT at 5V excitation is then $\frac{0{\cdot}0716}{2{\cdot}5} . 190.5 = 27$ mV

I expect that the lateral movements δ_2 and δ_4 of the LVDT will not significantly affect its reading & will be accommodated in the clearance in the bore.





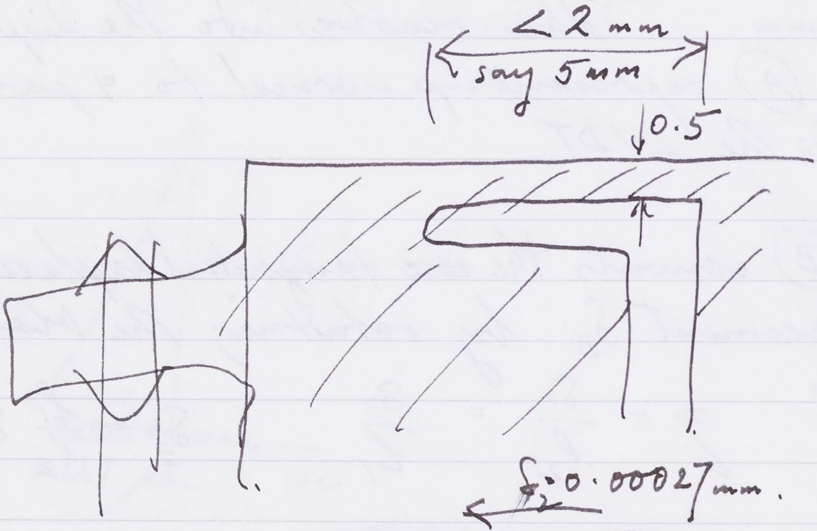
$s = r\eta$

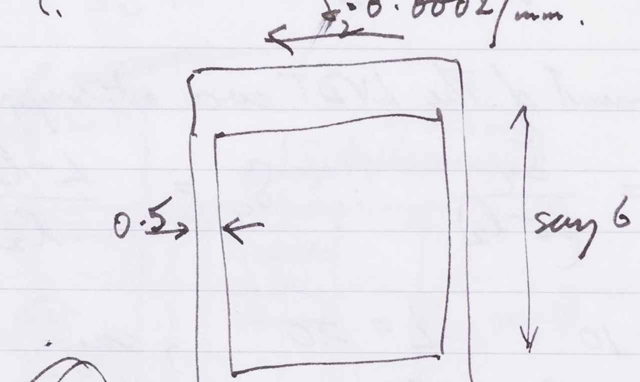
$$s + \Delta s = \left(r + \frac{t}{2}\right)\eta$$

$$\therefore\ 1 + \frac{\Delta s}{s} = \frac{r + t/2}{s}\cancel{\eta} = \frac{r + \frac{t}{2}}{r\eta}\eta$$

$$= 1 + \frac{t}{2r}$$

$$\therefore\ \overset{\varepsilon}{\cancel{\beta}} = \frac{\Delta s}{s} = \frac{t}{2r}$$ is the strain





(170)

Dimensions of the flexure pivots:

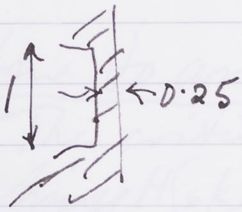
The max strain in the pivot, as a beam, is $\frac{t}{2r}$ (opposite)

But $\eta = \frac{\delta}{l}$ and $r = \frac{s}{\eta} = \frac{sl}{\delta}$, so $\overset{\varepsilon}{\cancel{\varepsilon}} = \frac{t}{2r} = \frac{t\delta}{2sl}$

or $\frac{t}{s} = \frac{2l\varepsilon}{\delta} < 0.002\,\frac{l}{\delta}$ for $\varepsilon < 0.001$.

In our case $\frac{l}{\delta} = \frac{10}{0.07}$, so $\frac{t}{s} < 0.28$

ie thickness has to be around ~~0.03~~ 0.25 of the length, or less.



is probably OK.

The upper flexure pivot would be 5 mm × 0.5 mm (see left).

The radial displacement of 0.00027 mm has to be absorbed in shear strain in the view Ⓒ. The shear force would be:

$$\underset{\text{(strain)}}{\frac{0.000\overset{135\,mm}{\cancel{27}}}{6\,mm}} \times \underset{\text{area}}{0.5 \times 2 \times 8(\text{say})} \times \underset{G}{80\,000} = \overset{14}{\cancel{21}}\,N$$

This seems a bit too much. So reduce the sides to 0.2 and the length from 8 to say 5 mm, then shear force $= \frac{\overset{0.000135}{\cancel{0.00027}}}{6} . 80000 . 0.2 \times 2 \times 5$

$= 3.6\cancel{0}\,N$,

somewhat less than 1 kg force, still quite significant if the fixing to the elastic element is not very firm. (See p171)

Bending moment: $\sigma = \frac{Mu}{I}$ so $M = \frac{\sigma I}{u} = \sigma Z = E\overset{\varepsilon}{\cancel{\varepsilon}}Z$

$Z = bt^2/6$ so $M = \frac{E\varepsilon bt^2}{6}$.

For $b = 1.5$ mm, $t = 0.25$ mm, $\varepsilon = 0.001$, $E = 210$ GPa,

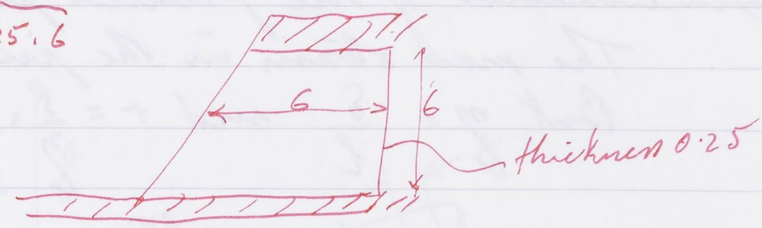
$M = 210.10^9 . 0.001 . 1.5.10^{-3} . (0.25.10^{-3})^2/6 = 0.00328$ N m

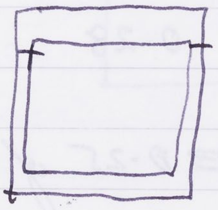
so at 6 mm lever arm, $F = \frac{0.00328}{0.006} = 0.547$ N

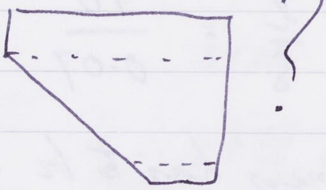
OK.

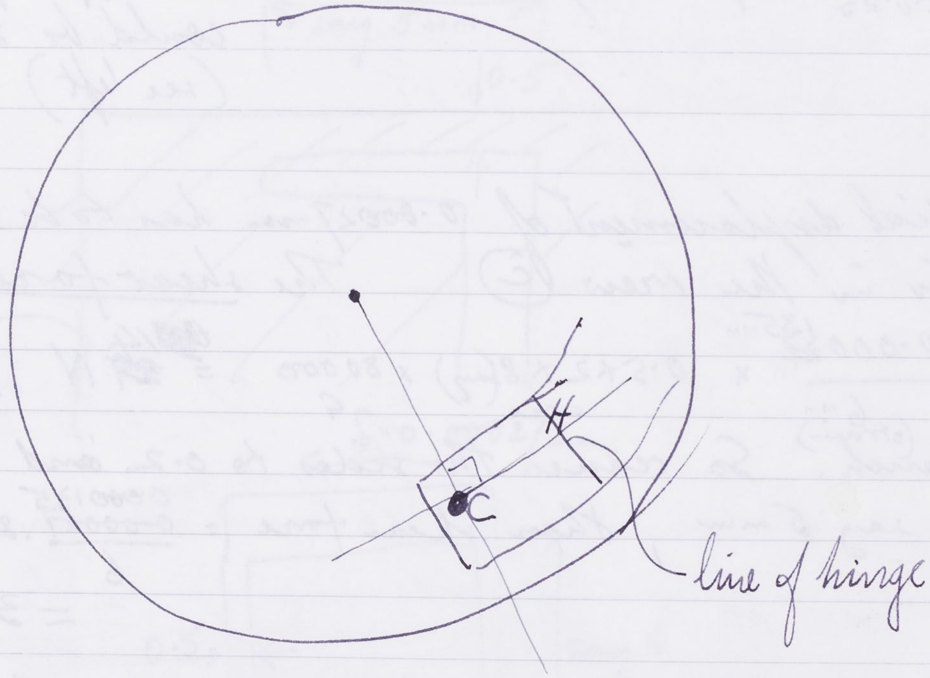
$$\text{shear force} = \frac{\overset{\gamma}{0.000135}}{6} \times \overset{G}{80,000} \times \overbrace{2 \cdot 0{\cdot}25 \cdot 6}^{area}$$

$$= 5{\cdot}4\ N$$









Radial shear force

The calculation on p170 leads to a radial shear force of order of 10N or somewhat less, arising from the radial component of the twist motion of the 1TC. This neglects any twist in the ~~[illegible]~~ aspect shown in ©. But the load is applied eccentrically & so there is likely to be some twist, alleviating some of this radial force. The effect could be accentuated by tapering the part as shown at left.

~~But this effect will be resisted by the top leaf spring, as will the~~

How to arrange configuration B (p169).

The critical issue is: how to orient the line of the hinge H (sketch opposite). Since we want the core C to move vertically downwards (ie not radially), the hinge H has to be normal to a line at right angles to the radius through C, ie the hinge has to be parallel to this radius. Ie, we want the motions to occur in a plane that ~~is~~ contains both the LVDT axis and the direction of motion of the load cell driving point, which is at C. But the motion is not exactly tangential; there is also the radial component considered at top of this page, which has to be accommodated by the shearing discussed there.

Contd p 184

Mar/98

Maximum Capacity of HPT Testing Machine

in view of Chinese enquiry.

Pressure Vessel Support:

This is supported on six M14 screws, at the bottom into Assab 2C Hollow bar, 0.2% C (UTS ~ 450 MPa, yield stress ~ 350 MPa) — Machinery Hbk p 398).

M14 screw has root diameter = 11.8, root area = 109 mm²
so tensile yield nominal = 109 . 350 = 38 kN ×6 = 230 kN
(shear ~~strength~~ strength on 10 mm length (370 mm²) @ 150 MPa = 370.150 = 56 kN)
So at 200 kN, the tensile stress in root of screw is about 300 MPa, OK for high tensile screw, and the shear stress in the hollow bar is about 90 MPa, also OK.

So the pressure vessel support could take up to 200 kN as an absolute maximum with very little F of S.

Stirrup: cross-sectional area ~ 5000 mm², stress 400 MPa - OK

Piston φ30: 200 kN gives stress = 280 MPa — OK.
Shoulder of piston 30OD 22ID : 200 kN gives stress = 612 MPa — OK on piston itself. Stirrup in 718 has y stress ~800 MPa, just OK.

Ballscrew: This has a static rating of 167 kN for 0.0001 × ball diameter of indentation; ball φ ~ 5, so indentation ~ 0.0005 mm, ie 0.5 μm. Maybe it could be pushed a little over this but not much. Have to go to a 63×10 ballscrew to get 200 kN — this would require minor re-design of the actuator.

51220 Thrust Bearing. Static load capacity is 75000 lbs ≡ 334 kN — no problem.

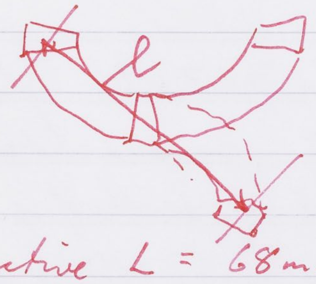
So the maximum rating of the loading system (apart from the internal load cell) would be 160 kN, maybe pushed to 200 kN with minor re-design of the actuator. This would allow stresses as follows (neglecting friction):

Specimen Dia/mm	160 kN	200 kN
20	509 MPa	637 MPa
25	326	407
30	226	283
40	~~[illegible]~~ 127	~~[illegible]~~ 159
50	81	102

30° : support area $= 2\left\{\frac{30}{360}\cdot\frac{\pi}{4}\left(64^2-30^2\right) - \frac{34}{2}\cdot 3\right\} = 316\ \text{mm}^2$

bearing stress = 316 MPa

– more comfortable.



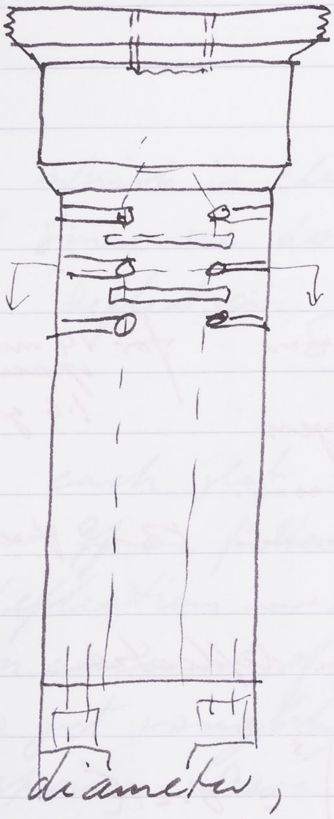
Effective L = 68 mm, so for 10·4 mm depth d,

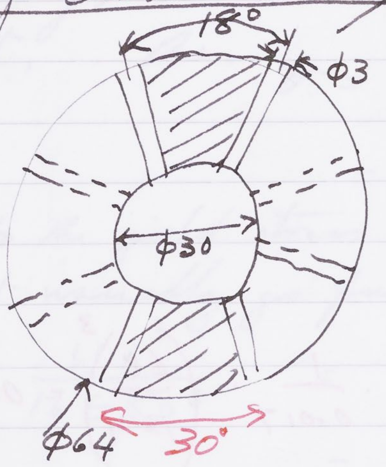
$\sigma = \frac{3WL}{4Bd^2}$ = 1387 MPa 10·4 mm.

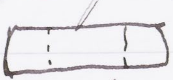
976 " 12·4 mm

835 MPa for ~~D.~~ 13·4 mm

Re-design of calibrating spring for KC





Instead of the flat spring  we could make more efficient use of the space with a circular section, and it could all be machined on a milling machine, by drilling ϕ3 holes and making a 1·6 cut (maybe it could be 1·2).

With the drill holes 18° apart & 3 mm diameter, the remaining area to support the load is $2 \cdot \left\{ \frac{18}{360} \cdot \frac{\pi}{4}\left(64^2 - 30^2\right) - \frac{64-30}{2} \cdot 3 \right\} = \overset{149\cdot0}{\cancel{74\cdot5}}\ mm^2$

so at 100 kN load, stress = 670 MPa. With Assab 8407 (H13) at HRC 48, yield stress ≃ 1400 MPa, so this is OK & allows for stress conc. factor of 2 at the ϕ3 radii.

The bottom end can be supported by 8 × M10 screws on PCD 46. Stress area of screw (M10) = 58 mm^2 × 8 = 464 mm^2, so stress in screws at 100 kN = 216 MPa — OK.

The spring works approx as a pair of beams D thick X 17 wide and effectively ~ 45 mm long less ~5 thickness of centre loading support, say a fixed end beam with centre loading, 40 mm long.

From earlier calculations (1981 Servocreep book, Apr 22 page) $D = \sqrt{\frac{3WL}{4B\sigma}}$ where $W = \overset{50\,000}{\cancel{100\,000}}$ N (one side only)

$L = 0\cdot040$ mm

$B = 0\cdot017$ mm

σ = say 500 MPa

Then $D = \overset{0\cdot0133}{\cancel{0\cdot0188m}} = \overset{13\cdot3mm}{\cancel{18\cdot8mm}}$

~~or~~

10·5 mm for σ = 800 MPa.

with $l = 68\text{mm}$,

$$\delta = \frac{50\,000}{16.210.10^9} \cdot \frac{1}{0.017} \cdot \left(\frac{68}{12.4}\right)^3 = 0.144\text{ mm}$$ for 14 mm spacing 1.6 gap

$\times$ 12 layers

$= 1.73\text{ mm}$.

We seem to observe deflection nearer to free ends than fixed ends, ie $\times 4$ $\longrightarrow 6.93\text{ mm}$

so may get ~ 5–6 mm deflection.

Or if we use 12mm spacing $$\delta = \frac{50\,000}{16.210.10^9} \cdot \frac{1}{0.017} \cdot \left(\frac{68}{10.4}\right)^3 = 0.245$$

$\times 13$

$= 3.18$

free ends $\times 4 = 12.7$

so may get ~ 9–10 mm deflection.

If we made the spacings 10 mm (ie 8.4 or so when cut is taken into account, then

$$\sigma = \frac{3 . 50000 . 0.040}{4 . 0.017 \; (0.0084)^2}$$

$$= 1250 \text{ MPa}$$

which is close to the yield stress.
Since we do not usually go full scale, this may be OK.
Deflection

$$\delta = \frac{WL^3}{16 EBD^3} = \frac{50\,000 . (0.040)^3}{16 . 210 . 10^9 . 0.017 . (0.0084)^3}$$

$$= 0.0001 \text{ m} = 0.0945 \text{ mm per}$$

each slab.

If we fitted in 15 slots, this would give 1.5 mm deflection — seems too small. May be underestimating on account of fixed end assumption. We would expect to get roughly the same deflections as with the flat spring where the spacing is 9 mm (with 1 mm cuts).

If the spring proves only usable (or linear) up to ~50kN, we could still calibrate to 100kN using the external load cell.

$$\sigma_\theta = \sigma_r = \frac{\alpha E T_i}{2(1-\nu)\ln b/a}\left(1 - \frac{2\left(b/a\right)^2}{\left(\frac{b}{a}\right)^2 - 1}\ln\frac{b}{a}\right)$$

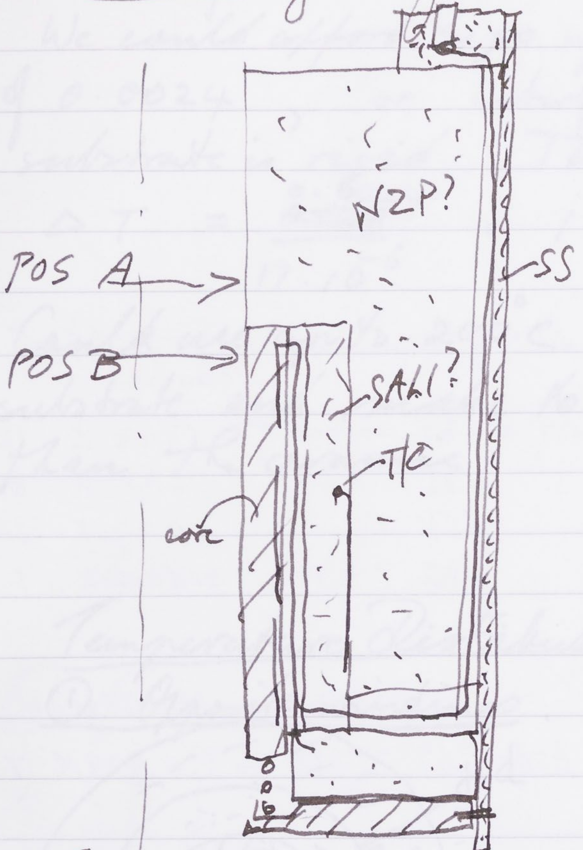
at ID ~~at A~~

$$\frac{\alpha E T_i}{2(1-\nu)\ln b/a}\left(1 - \frac{2}{\left(\frac{b}{a}\right)^2 - 1}\ln\frac{b}{a}\right)$$

at OD ~~at B~~

May 1998

Re-design of Furnace — NZP furnace



~~Ø~~ This would use a single cylinder of ceramic such as NZP, encased in a shrunk-on SS sleeve, with leads taken down the outside to the bottom for connection to the replaceable core. Over the core the ceramic could be lined with fibre ceramic such as SALI in which the thermocouple are incorporated.

First question is what happens about thermal stress.

Timoshenko & Goodier p449 give

~~$$\sigma_\theta = \sigma_r = \frac{\alpha E T_i}{2(1-\nu)\ln b/a}\left(1 - \frac{2b^2}{b^2-a^2}\ln\frac{b}{a}\right)$$~~

a = inner rad
b = outer rad
T_i = temp diff

At position A, take $T_i = 900$, $b/a = 61/27$, $\nu = 1/3$

$$\text{so } \sigma_\theta = \frac{\alpha E\, 1000}{4/3 \ln 61/27}\left(1 - \frac{2}{\left(\frac{61}{27}\right)^2-1}\ln\frac{61}{27}\right)$$

$= 555\,\alpha E$ on the O.D. [~~1134~~ ~~769~~ $-851\,\alpha E$ on ID]

& At position B, take $T_i = 1200$ $b/a = 61/40$, $\nu = \frac{1}{3}$

$$\sigma_\theta = \frac{\alpha E\, 1200}{4/3 \ln 61/40}\left(1 - \frac{2}{\left(\frac{61}{40}\right)^2-1}\ln\frac{61}{40}\right)$$

$= 775\,\alpha E$ on the O.D. [~~[illegible]~~ $-1025\,\alpha E$ on ID]

For NZP, take say $E = 100$ GPa (actually 50 GPa.), $\alpha = 1.10^{-6}$

then $\sigma_\theta = -55$ MPa. at A } on the OD.
$= -78$ MPa at B }
$\sigma_\theta = 85$ MPa at A } on the ID
$= 103$ MPa at B }

Shrinking SS sleeve on ceramic:

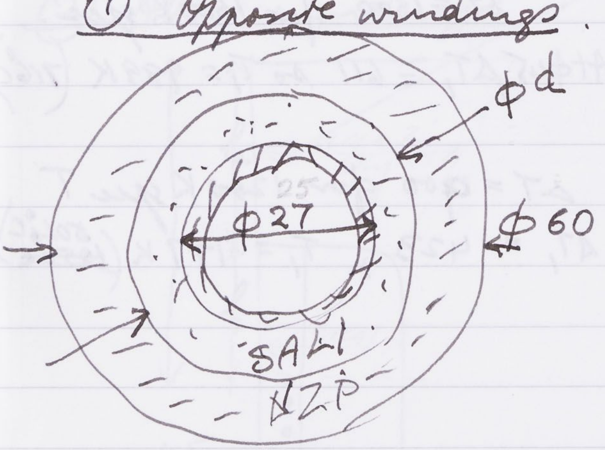
We could afford to go up to 500 MPa probably, ie a strain of 0.0024, or interference of 0.148 mm on $\phi 61$ if the substrate is rigid. This corresponds to a heating temp

$$\Delta T = \frac{\overset{0.0024}{\cancel{0.148}}}{17.10^{-6}} = 140\,K \quad \text{ie heating to} \sim 160°C.$$

Could use up to 200°C in view of the resilience of the substrate and make the SS ID about 0.15 mm smaller than the ceramic.

Temperature Distribution

① Opposite windings.



$$\text{heat flow } q = \frac{2\pi \Delta T l}{\dfrac{\ln \frac{d}{27}}{K_{SAL}} + \dfrac{\ln \frac{60}{d}}{K_{NZP}}}$$

Take $K_{NZP} = 1$, $K_{SAL} = 0.5$ (may be lower than this)

0.3

$$\therefore q = \frac{2\pi \Delta T l}{\ln d - 2\ln 27 + \ln 60}$$

Put $l = 100$ mm, $\Delta T = 1400$, $q = \dfrac{2\pi\, 1400 \cdot 0.1}{\ln d - 2\ln \underset{25}{27} + \ln 60} = \dfrac{879.6}{\ln d - 2.497_3}$ 2.343₄

∴ q =	738 W	for d =	40 mm	0.53 W/K
"	709	"	42	0.51 W/K
"	672	"	45	0.48 W/K
"	622	"	50	0.44 W/K
	550		60	0.39 "
	1102		27	0.79 "

Intermediate temperatures:

$$q = \frac{2\pi\, K_{1_{SAL}} \Delta T_1\, l}{\ln \frac{d}{27}} = \frac{2\pi\, K_{2_{NZP}} \Delta T_2\, l}{\ln \frac{60}{d}}$$

0.5

$$\therefore \frac{\Delta T_2}{\Delta T_1} = \frac{K_{1_{SAL}}}{K_{2_{NZP}}} \cdot \frac{\ln \frac{60}{d}}{\ln \frac{d}{27}} = 2\, \frac{\ln \frac{60}{d}}{\ln \frac{d}{27}}$$

$\Delta T = 700$ or 1000K spec T

$\Delta T_1 = 329$, $T_1 = 671$ K (398°C) at d=45

$\Delta T = 400$, spec T = 700 K

$\Delta T_1 = 188$, $T_1 =$ ~~[illegible]~~ 512 K (~~290~~ 239°C) at d=45

$\Delta T = 200$, spec T = 500 K

$\Delta T_1 =$ ~~[illegible]~~ 94, $T_1 =$ ~~[illegible]~~ 406 K (130°C) at d=45

$\Delta T = 1400$ means $T = \sim 1700$ K at centre.

$\Delta T_1 = 457$ means $T_1 = 1243$ K 970°C

535 1165 K 892°C

658 1042 K 769°C

680 820 K 547°C

$\Delta T = 1300$ for 1600K spec.

At φ45, $\Delta T_1 = 611$ so $T_1 = 989$ K (716°C)

$\Delta T = 900$ for 1200K spec T

$\Delta T_1 = 423$, $T_1 = 777$ K (~~[illegible]~~ 504°C)

~~Δ~~ $\Delta T_1 + \Delta T_2 = \Delta T$

$$\Delta T_1 \left(1 + 2 \frac{\ln \frac{60}{d}}{\ln d/27} \right) = \Delta T$$ See p 220

$$\text{or} \quad \Delta T_1 = \frac{\Delta T}{1 + 2\frac{\ln 60/d}{\ln d/27}}$$

K above wall temp ~300 K

So with $\Delta T = 1400$, $\Delta T_1 = 457$, $T_1 = 943$ for $d = 40$

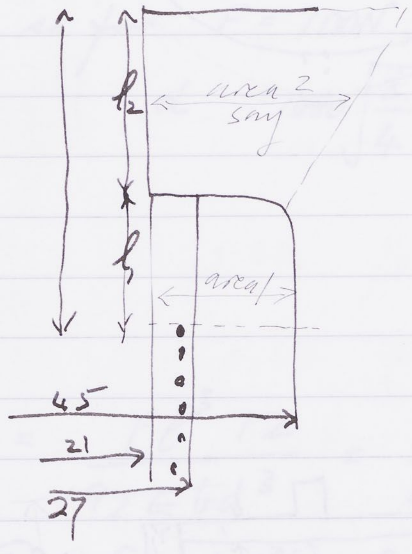
$\Delta T_1 = 535$ $T_1 = 865$ 42

" 658 $T_1 : 742$ 45

" 680 $T_1 = 520$ 50

Choose $d = 45$ mm

② Above windings :



$$q = \frac{2 . K_2 . \Delta T_2}{l_2} = \frac{1 . K_1 . \Delta T_1}{l_1} \text{, say}$$

$$\text{ie} \quad \frac{\Delta T_2}{\Delta T_1} = \frac{K_1 \, l_2}{2 K_2 \, l_1} =$$

$$\text{mean } K_1 = \frac{27^2 - 21^2}{45^2 - 21^2} . \underset{6,\text{ say}}{K_{Al_2O_3}} + \frac{45^2 - 27^2}{45^2 - 21^2} . \underset{0.5,\text{ say}}{K_{SALI}}$$

$$= 0.18 . 6 + 0.818 . 0.5$$

$$= 1.50$$

$$\therefore \frac{\Delta T_2}{\Delta T_1} = \frac{1.5 \, l_2}{2 \, l_1} = 0.75 \frac{l_2}{l_1}$$

~~$\Delta T_1 = 1.333 \frac{l_1}{l_2} . \Delta T_2$~~

$l_1 + l_2 = 50$ approx

$l_2 = 50 - l_1$

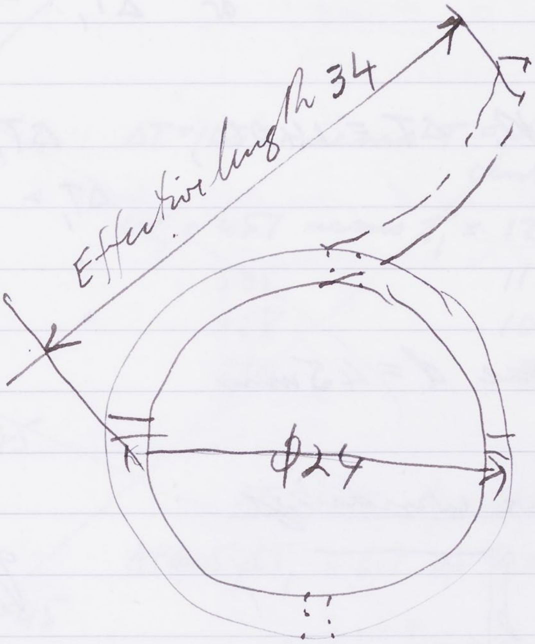
$$\Delta T_1 + \Delta T_2 = \Delta T$$

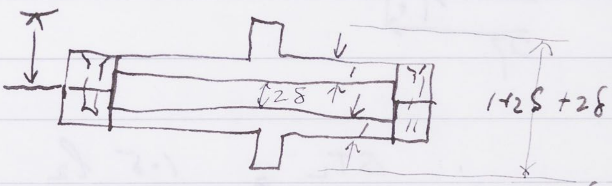
$$\Delta T_1 + 0.75 \frac{l_2}{l_1} \Delta T_1 = \Delta T$$

$$\Delta T_1 = \frac{\Delta T}{1 + 0.75 \frac{l_2}{l_1}}$$

$$= \frac{\Delta T}{1 + 0.75 \left(\frac{50 - l_1}{l_1} \right)}$$

$$= \frac{\Delta T}{0.25 + \frac{37.5}{l_1}}$$





$= 1+4\delta$

$= 1+4.0.4$

$= 2.6$

$+ \underline{1.3}$

3.9

$$\Delta T_1 = \frac{1}{0{\cdot}25 + \frac{37{\cdot}5}{l_1}} \Delta T$$

Take $\Delta T = 1300$ at top of top winding, then

$\Delta T_1 = 300$	$T_1 = 1000$	for $l = 10$
355	944	12
436	= 863	" 15
565	= 735	" 20

Choose $l = 15$.

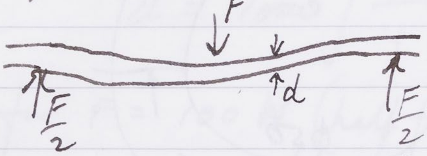
Spring loading for wire

Spooner p607

From 1981 Servocreep notebook p "April 22",

$$M = \frac{Fl}{8} = Z\sigma \qquad \delta = \frac{Fl^3}{192EI}$$

$$\text{when } Z = \frac{bd^2}{6} \qquad I = \frac{bd^3}{12}$$



$$\text{so} \quad M = \frac{bd^2\sigma}{6} = \frac{Fl}{8} \quad \therefore \quad d = \sqrt{\frac{3Fl}{4b\sigma}}$$

so for $F = 100N$, say, $l = 34mm$, $b = 3mm$, we have

$$d = 1000\sqrt{\frac{3.100.\ 0{\cdot}034}{4\ 0{\cdot}003.\ \sigma}}\ mm = \frac{29154}{\sqrt{\sigma/Pa}}\ mm$$

~~1.683~~

$$= 2{\cdot}92 \ mm \text{ for } 100 \ MPa$$

~~0.168~~

$$= 1{\cdot}46 \ mm \text{ for } 400 \ MPa$$

$$= 1{\cdot}00 \ mm \text{ for } 850 \ MPa$$

$$\delta = \frac{Fl^3\ 12}{192\ Ebd^3} = \frac{F}{E}\left(\frac{l}{d}\right)^3 \cdot \frac{1}{b} \cdot \frac{1}{16}$$

For $F = 100\ N$, $E = 200.10^9$, $b = 0{\cdot}003\ m$, $l = 34$

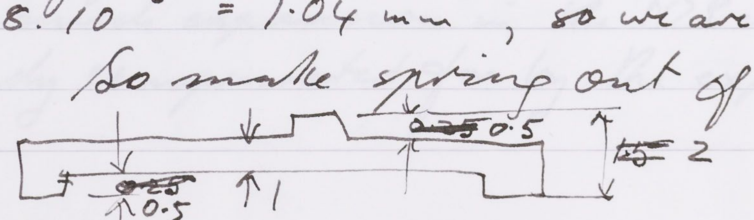
$$\delta = \frac{100}{16.200.10^9.0{\cdot}003}\left(\frac{34}{d/mm}\right)^3 \cdot 1000.\ mm. = 0{\cdot}41\ mm \text{ for } d = 1mm$$

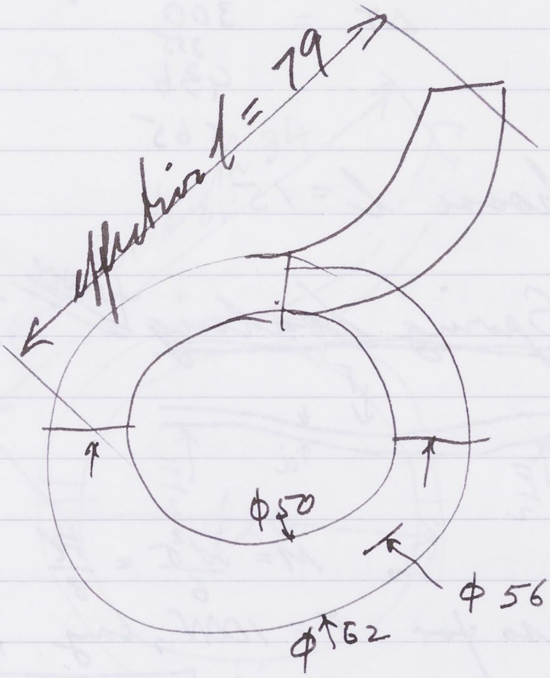
$$= 0{\cdot}13\ mm \text{ for } d = 1{\cdot}46mm$$

If we stack up three such springs, total deflection is 1·2 mm.

Expansion of ~100 mm alumina at 1300°C is $100.\ 1300.\ 8.10^{-6} = 1{\cdot}04$ mm, so we are looking at ~1mm expansion. So make spring out of three elements

four




effective l = 79
φ50
φ56
φ62

The space available for spring = 7 mm so if we use four elements of 2.0 each , initial deflection = 1 mm and final deflection = 2 mm , so force rises from ~100N total to ~200 N total — OK

Spring rate required is about ~~200~~ 125 N/mm. 100 N/mm would do, with initial compression of ~1mm.

Spring loading for whole furnace in SS can

$$d = 1000\sqrt{\frac{3 . 100 . 0.079}{4 . 0.006 . \sigma}} = \frac{31425}{\sqrt{\sigma}}$$

for F = 100 N (half force)

$= 3.14$ mm for 100 MPa

$= 1.57$ " 400 MPa

$= 1.11$ 800 MPa.

$$\delta = \frac{100}{16 . 200 . 10^9 . 0.006}\left(\frac{79}{d}\right)^3 1000 \text{ mm}$$

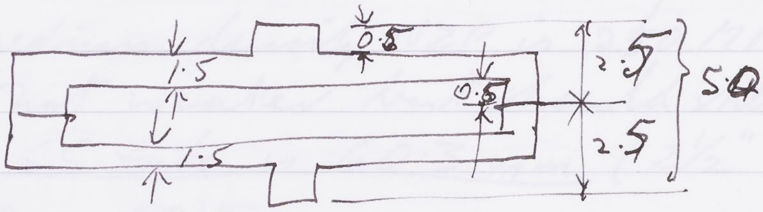
$= 0.08_3$ for $d = 3.14$

$= 0.66$ for $d = 1.57$

$= 1.88$ $d = 1.11$

Again need ~ 125 N/mm for whole spring, with precompression of ~ 1 mm or more.

Suitable spring:



squeezed into 4 gap. Spring will close up. But there is not much expansion in the NZP with temp & it will be largely compensated for by the expansion of the MS can.

Interference on cylinders.

If we calculate an interference between the outer SS and the NZP cylinder assuming the latter to be rigid & the SS to be taken up to say 500 MPa stress, then the stress will be somewhat below this level & allow a margin for thermal expansion of the NZP (although this may be compensated for by thermal expansion of the SS, since its α is some 18× that of the NZP; if mean temp in NZP is, say, 400°C, then any temp rise above ~ 20°C in the SS will cause it to expand more). So we could design for at least 500 MPa stress in the SS, ie a strain of ~ 0·0025, corresp to a temperature rise of about 140K; so we would need to heat the SS at least to 200°C & preferably more for easy assembly.

0·0025 strain ≡ 0·15 mm interference on ϕ60.

If we make it <u>0·2 interference</u>, ≡ 0·0033 strain ≡ 700 MPa stress, we need 183 K temp rise, say at least 250°C.

The stress in the SS has to be supported by stress in the NZP, the wall thickness of which is 15/2 mm compared with ~ 2·8/2 mm of SS, ie stress in NZP = $\frac{2·8}{15}$ × that in SS

$$\sigma_{NZP} = 0·187\ \sigma_{SS}$$

$$= 93\ \text{MPa for}\ \sigma_{SS} = 500\ \text{MPa}.$$

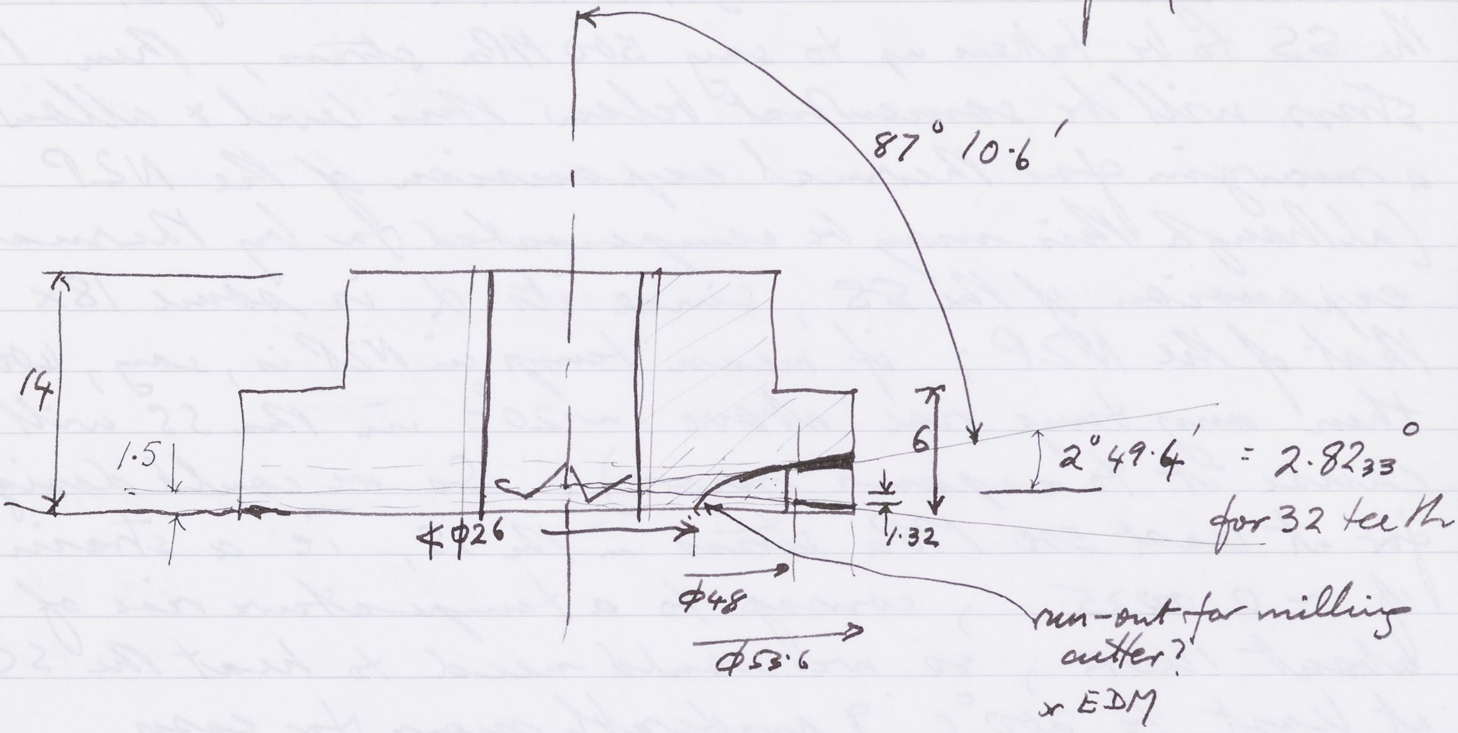
Compression strength of medium density NZP is 250 MPa; low density will be somewhat weaker but should still be OK.

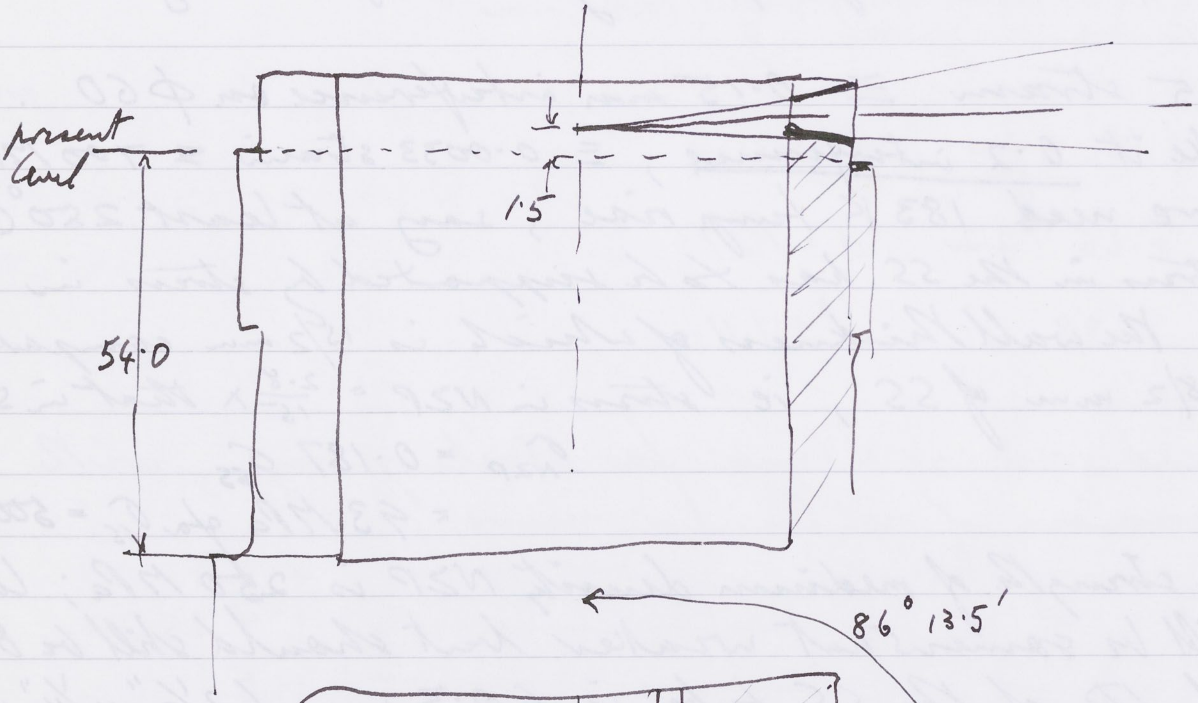
So if <u>ID of the SS tube is 60·3 mm</u> (2½" × 1/16" WT), we make the <u>NZP OD = 60·5</u>.

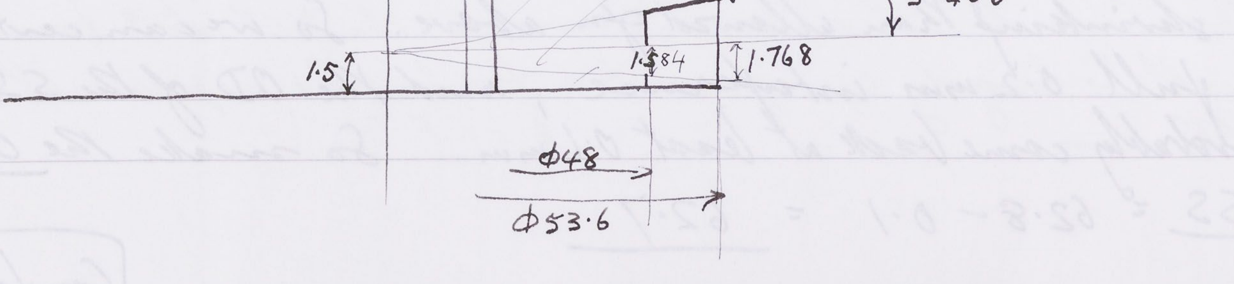
Note that E for NZP is ~50 GPa, so there will be more relaxation on shrinking than allowed for above. So we can certainly use a full 0·2 mm interference, and the OD of the SS will probably come back at least 0·1 mm. So make the <u>OD of the SS</u> ≐ 62·8 − 0·1 = <u>62·7</u>

Contd p186

Machinery Hdbk
p 2240







6/10/98

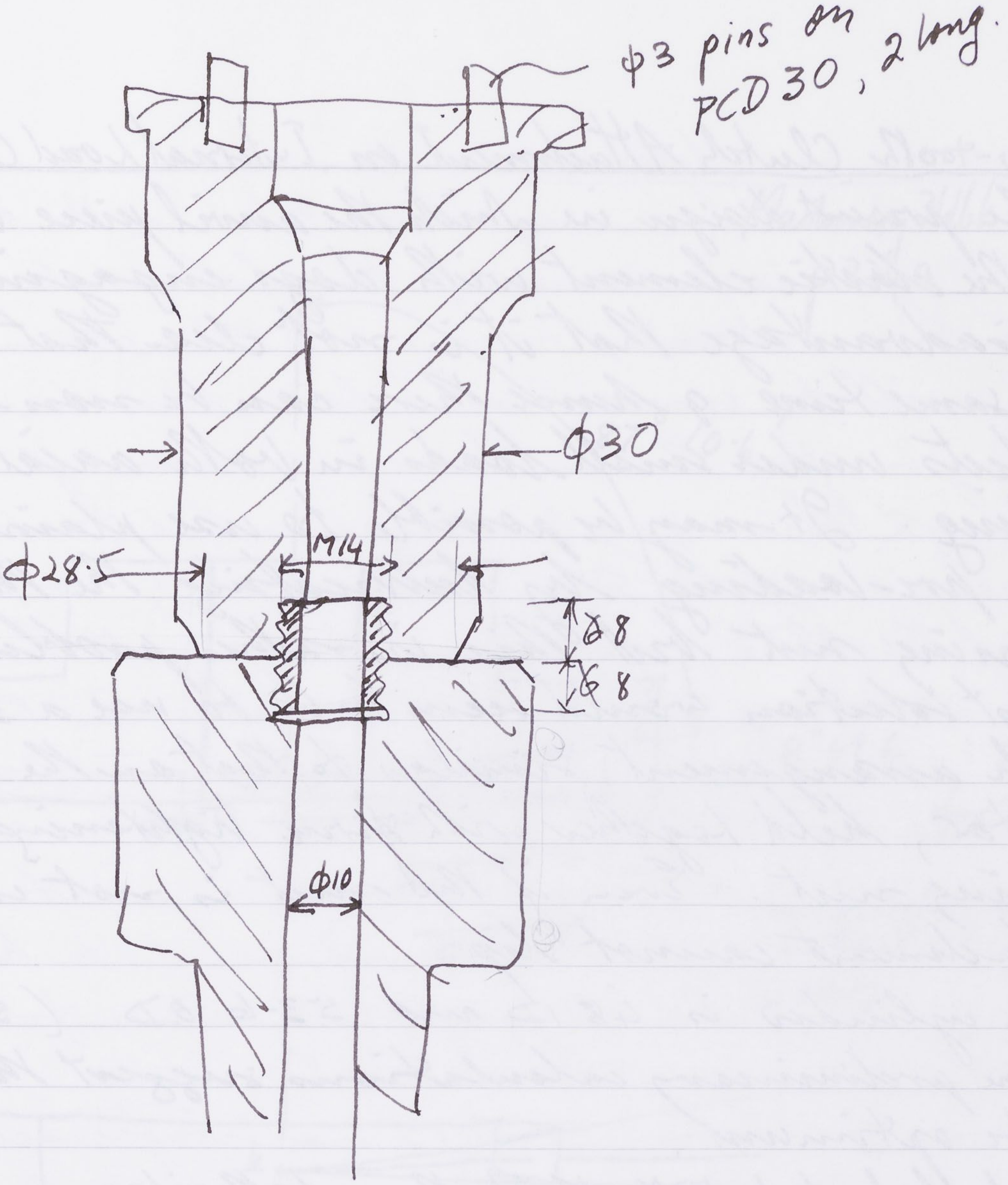
Saw-tooth Clutch Attachment on Internal Load Cell

The present design in which the anvil piece is fixed to the top of the elastic element with dogs engaging in slots has the disadvantage that it is not clear that all dogs seat at the same time & hence there can be non-linear take-up effects under small loads in both axial & torsion loading. It may be possible to use plain faces with high pre-loading by lubricating the threads of the retaining nut but this is rather problematical. The best solution would seem to be to use a saw-tooth clutch arrangement similar to that on the driver of the actuator, held together with firm tightening of the retaining nut. Even if the nut is not well tightened, the attachment cannot slip.

The cylinder is 48 ID and 53·6 OD (50·8 mean ϕ) and some preliminary calculations suggest that 32 teeth would be optimum.

Probably best to EDM the teeth on both sides.

→ In view of the part being already made with 12 lugs, modification would be simplest on a 24 teeth basis, and further calculation indicates that this would be also a suitable long-term proposition. Can still stay with the ℄ 1·5 above present ~~top~~ support level. Theoretically, there will be 0·268 mm cut into this but this cut will be reduced when a radius is introduced in the bottom of the sawtooth profile.


ϕ3 pins on PCD 30, 2 long.

Alternative Interconnection for Pistons

At present the load cell & compensating pistons are held together, against friction in the plug, by a nut on the load cell stem that carries the leads of the load cell. This has the serious disadvantage that the lead wires have to come out in grooves under the nut & are very liable to damage.

An alternative arrangement is shown opposite. The pistons are held together with a short (11mm) piece of M14 × 10·2 ID steel (4140), which can be tightened up by turning the comp. piston while the load cell piston is restrained by two ϕ3 pins.

The max possible bearing stress on the interface between the two pistons (ϕ28·5 OD, 14 ID = 484 mm^2) occurs when 100 kN is applied at 700 MPa, ie total force $= \frac{\pi}{4}(30)^2 \cdot 700 + 100\,000 = 595{,}000$ N, so max bearing stress = 1230 MPa, well within the strength of the HRC 55 pistons (>1500 MPa).

M14×2 has ID = 11·8, so with bore 10·2, area = 28 mm^2.

Previous stem has M10×0·75 thread, min ϕ 9·2, with bore 6·4, area = 34 mm^2, so with same (4140) steel, the new arrangement would be marginally weaker — could be compensated for if necessary by making of 8407 at HRC 42, or with a stronger ready-heat-treated steel (En26).

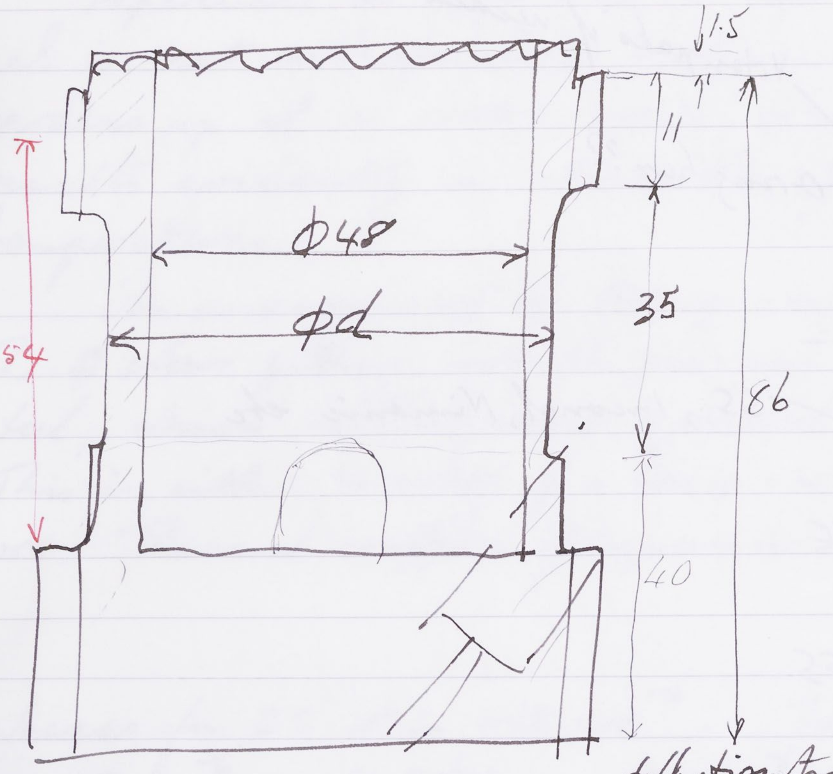
Then the stem can be made with simple reduced OD & the leads simply trapped under heatshrink — no need for grooves.

Effective length ℓ ~50 mm,
so max displacement at strain 0·0015 is 75 μm.

[Drawing of June 98 shows ϕd as 55.
but modified later]

Contd from 171 (Oct 98)

LVDT Load Cell – Elastic element.



Diameter d?
If total load range is 100 kN, then

$$\sigma = \frac{100\,000}{\pi \frac{(d^2-48^2)}{4}} = \frac{400\,000}{\pi(d^2-48^2)}$$

For max strain of 0.0015,

$$\sigma = 0.0015 \cdot 210{,}000$$
$$= 315\ \text{MPa}.$$

$$\therefore d^2 = 48^2 + \frac{400\,000}{\pi . 315} \cancel{= 404}$$

$$d = 52.04$$

ie $\underline{\underline{d = 52}}$

0.0015 × 54 = 0.081 mm

deflection ~~strain~~ = 0.0015 × 35 = 0.053 mm.

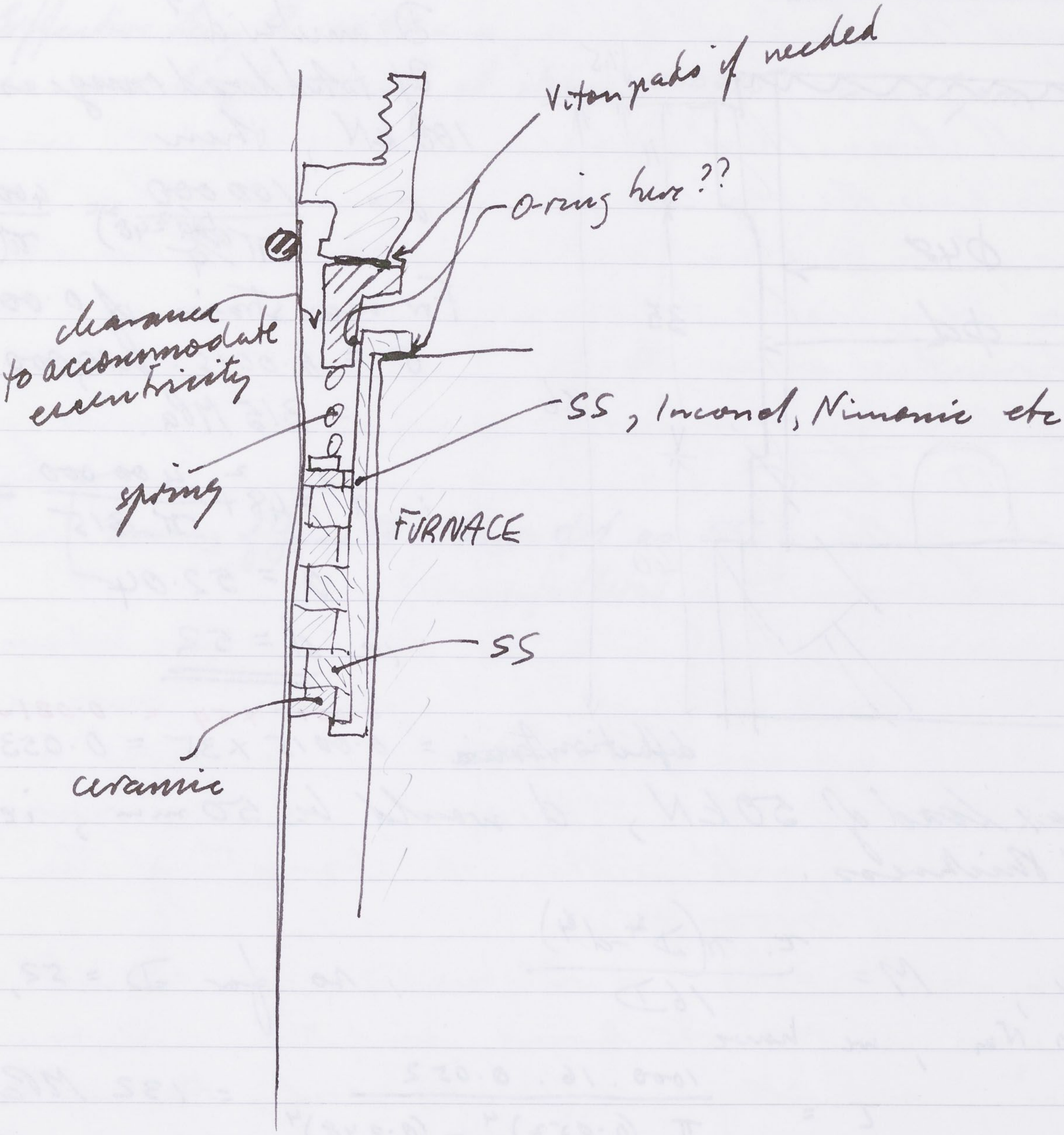
For a max load of 50 kN, d would be 50 mm, ie 1 mm wall thickness.

In torsion, $M = \frac{\tau . \pi(D^4-d^4)}{16D}$, so for D = 52, d = 48, M = 1000 Nm, we have

$$\tau = \frac{1000 . 16 . 0.052}{\pi\ (0.052)^4 - (0.048)^4} = 132\ \text{MPa}$$

OK.

Contd p 194


Viton pads if needed
O-ring here??
clearance to accommodate eccentricity
SS, Inconel, Nimonic etc
spring
FURNACE
SS
ceramic

9/10/98

Furnace Inner Seal for Torsion

Experience at GFZ shows that the alumina-paper inner seal is not satisfactory, at least for torsion. It hardens up at ~ 1000–1200 K and rotating the piston results evidently in variable opening and fluctuating temperatures.

An arrangement of sliding rings as at left may be better. The sleeve fitting into the furnace could be of stainless steel, which will expand against the furnace insulation. This is either SiAlON if a long core, or PSZ or NZP if short core. Thermal coeffic of expansion α is 3×10^{-6} for SiAlON
10×10^{-6} for PSZ
or $< 1 \times 10^{-6}$ for NZP

whereas for SS it is $\sim 18 \times 10^{-6}$. Thus the differential is $\sim 15 \times 10^{-6}$ for SiAlON or 8×10^{-6} for PSZ.

If we allow for 1000 K rise in temperature, we need an initial clearance of $15 . 10^{-3} . 21$ = 0.31 mm for SiAlON
0.17 " " PSZ

If we settle on a clearance of 0.2 mm, this will ensure tightness for the PSZ but still 0.1 clearance on SiAlON or NZP. Maybe at 1000°C the SS will be sufficiently weak that if there were less clearance, the interference would be taken up by plastic deform^{n} in the SS. So maybe we shoot for 0.1 mm clearance.

The outer rings should be of SS just sliding inside the tube, or maybe with 0.05 clearance. It is undesirable that they become too tight in running because they have to be free axially.

The inner rings could be of mullite, $\alpha = 5 \times 10^{-6}$, relative to 8×10^{-6} for Al_2O_3 pistons or 10×10^{-6} for PSZ pistons. There could be up to 0.04 clearance between jacket & piston, which will be taken up by pressure. At 1000°C, the differential expansion is $5. \times 10^{-3} . 15 = 0.075$. So an initial clearance of ~ 0.02 or 0.03 mm over the jacket should be OK.

Contd from p 181)

Dec 1998

Test on NZP furnace (see p 176)

The NZP furnace with bare Al_2O_3 core was tested in the "Green Machine" at ANU.

On first heating to about 830K the power consumption with 0·74W/K (291 + 70 + 50 W in B, C & T), then with a re-adjustment of power settings it became 0·63W/K at ~ 890K (276 + 63 + 48 W).

On heating to ~1200K, consumption became 0·77 W/K (510 + 119 + 95) and the T profile was peaked to POS12, appro. top winding.

On raising power further, the temperature did not go up much, peaking at POS 12 at 1300K, but power consumption rose to 0·95 W/K (677 + 159 + 120).

On second day & same settings, the first day was roughly repeated at ~860K and 0·72 W/K (300 + 75 + 60 W). Now the inner seal had been re-fitted to seat better & new Al_2O_3 paper inserted. Did not explain yesterday's behaviour. On attempting to go higher, the performance became dismal. At a limit of 18A in bottom winding, max temp was just under 1000K, peaked at POS12, and power consumption 1·07 W/K (534 + 139 + 98W).

POSTMORTEM with Frank, 29/12/98:

1. Alumina core cracked off below bottom winding – normal.
2. Windings & core pulled out easily – fair bit of clearance, especially around the tails. Presumably the expansion of the core had enlarged the bore of the SALI a bit.
3. Removed outer sleeve & T/C's, then pulled out SALI sleeve. It came out easily (Frank said it was neat on way in) + had clearance esp. on upper half, where there was also some discolouration. Evidently circulation up the ID of the SALI & down the OD (may leak through the SALI ~~B~~; but it had to get to bottom somehow so maybe all the way down the OD).

Conclusion: The SALI seems to have opened up on ID (due to expansion of Al_2O_3 core?) and been closed down on the OD (due to expansion against the NZP on previous run??); but this problem was already becoming evident on first run up; maybe the OD had already been "reamed" down a bit on the way in).

The following modifications suggest themselves:

1. Split the SALI & back it with alumina paper and stainless steel segments to allow for thermal expansion.

Expansion of core $= 8.10^{-6}.1300.24 = 0·25$ mm at max.
∴ SALI probably half this at interface, and less outside. If OD at 400°C & $\alpha = 4.10^{-6}$ (??)
then $\Delta d = 400.4.10^{-6}.40 = 0·06$ mm.

2. Bring the thermocouples in the same way as the windings

~~2~~3. Make a SS bottom plate & introduce a plug & socket connection — use 24/21 core.

4. Possibly introduce cementing between core and SALI.

5. If there is a problem still with Al_2O_3 cores, try SiAlON again — but it will have to be without cement so as to have the wires loosely wound.

For 4: Possible cementing procedure:

1) cement over windings & smooth off to OD of core

2) cement fillets by tails

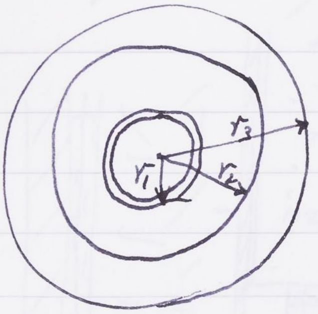
3) coat inside of SALI with cement and smooth off to correct Φ.

4) do same with grooves for tails.

5) fit SALI over core, perhaps first re-moistening the cement slightly. (or use very dilute cement?)

Either fit alumina paper between SALI segments or fill gaps with cement; maybe both.

Predicted heat loss : $\frac{q}{\Delta T} = \frac{2\pi l}{\frac{\ln \frac{r_2}{r_1}}{K_1} + \frac{\ln \frac{r_3}{r_2}}{K_2}}$



Take $K_1 = 0{\cdot}4$ for SALI & $1{\cdot}0$ for NZP

$l = 0{\cdot}1$, $r_3 = 61$, $r_2 = 45$, $r_1 = 24$ or 27

$$\frac{q}{\Delta T} = \frac{0{\cdot}2\pi}{\frac{\ln \frac{45}{r_1}}{0{\cdot}4} + \frac{\ln \frac{61}{45}}{1}} \quad \left[= \frac{0{\cdot}2\pi}{1{\cdot}57 + \underset{\text{for } r=24}{0{\cdot}30}} \right]$$

$r = 24$ $q/\Delta T = 0{\cdot}335$ W/K

$r = 27$ $q/\Delta T = 0{\cdot}397$ "

If we take $K_2 = 2$ (for PSZ), $r = 24$ $q/\Delta T = 0{\cdot}365$

$r = 27$ $q/\Delta T = 0{\cdot}440$

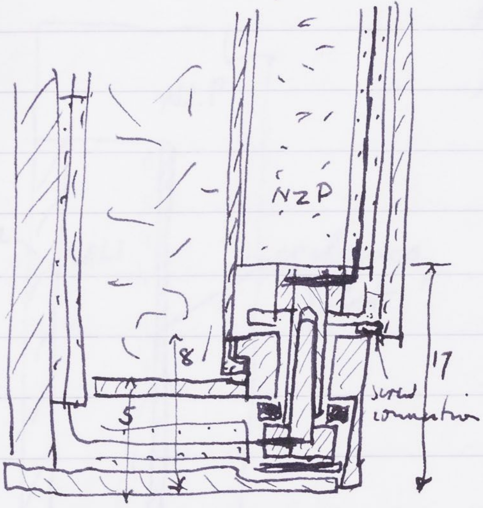
Thus reducing the OD of the core from 27 to 24 reduces heat loss 0·84 of that for 27 diameter.

Substituting NZP for PSZ reduces heat ~~of~~ loss to 0.90 or 0·92 of that for PSZ.

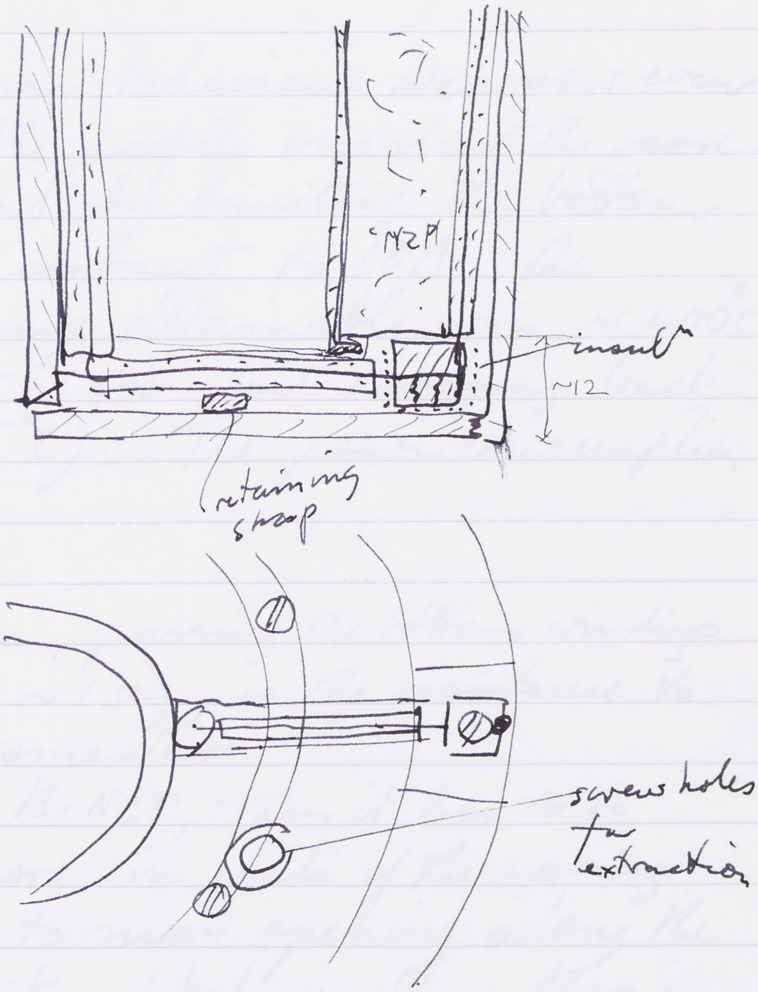
The combination of putting ~~B~~ NZP in place of PSZ and reducing the core OD to 24 should reduce heat loss 0·75 that for 27 core & PSZ.

Connection arrangements:

One possibility is to have plug & socket arrangements such as:

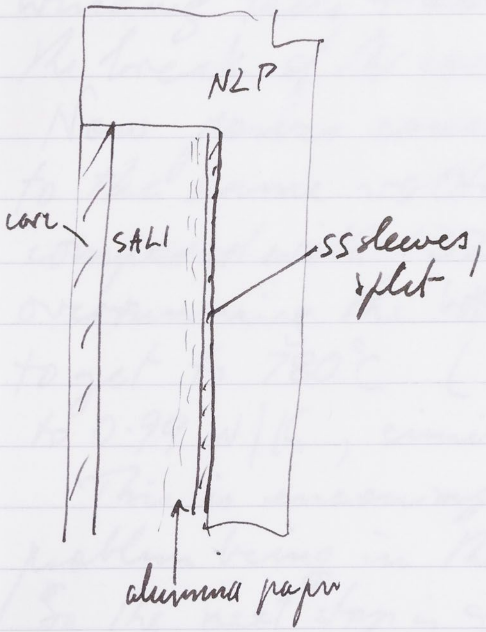


or a screw attachment such as :



Further test on NZP furnace

8/1/99



Frank introduced spring loading of the SALI insulation by using SS split sleeves & two layers of alumina paper. The SALI was split but otherwise the same as before.

The furnace behaviour was now even worse. The profile was about the same shape but by limiting the bottom winding current to 18A the max-temp attainable was ~ 400°C (at first ~ 430, but dropping back to 415°C). The power consumption was then 1·23 W/K!.

Conclusions:

1. There has to be extra convection, passing the bottom winding & keeping it cool (resistance is ~ 1Ω), so the resistance to convection has been reduced somewhere.

2. If the convection is within the NZP, then it has to be heating the I.D of the NZP more in spite of the spring loading, ie there would have to more opening along the core than before — seems a bit unlikely unless there is arching on the alumina paper in the splits of the SALI, or greater clearance around the tails, or a circulation ~~is~~ in the alumina paper (not noticed in earlier furnaces).

3. If the convection is around the whole furnace, the NZP must have become very permeable

Two things that could be tried:

1. Try applying cement at the bottom end to plug up the gaps between SALI & core — tests case 2 above.

2. Try putting the "insert" parts in a steel can to block gas flow into the NZP — tests case 3 above.

Next test on NZP furnace 11-1-99

Frank applied Ceramabond cement over the bottom ends of the winding tails & across the SALI insulation, as well as in the break of the core. Dried for an hour or so at 80°C.

Now power consumption was 0·92 W/K on going up to the same voltage level (110V on Variac) as last time, compared with 1·32, so there is some improvement. By overrunning the bottom winding current (18·5A) I was able to get to 780°C (1050K) but power consumption was up to 0·99 W/K, similar to earlier runs.

This is encouraging, probably pointing to the convection problem being in the core/SALI area rather than in the NZP. So the next step is for Frank to try cementing the SALI on to the core all the way along. For simplicity, still keep ~~to~~ the alumina tubes on the tails.

Test with full cementing 18-1-99

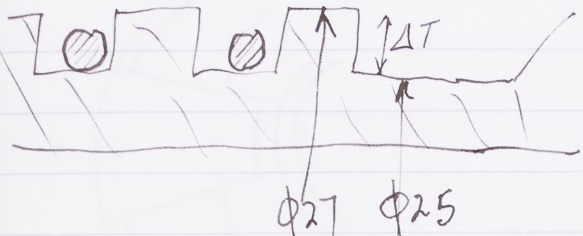
Considerably better:	0·76 W/K	& 565°C	on variac 110V, gradually
changing to	0·81 "	543°C	"
Then	0·92 W/K	786°C	" 160V
	0·97 W/K	1028°C	" 207

When I went to full voltage on Variac, current just over 18A, the feedthrough sparked over & bottom winding current went to zero; however, temperature was around 1200°C.

No water in furnace afterwards (it had been baked for a couple hours at ~120°C & then a couple hours at 200°C).

Cont^d p 203

Core cracking



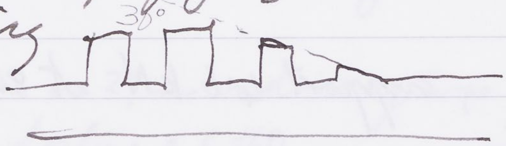
If there is a substantial temperature gradient in the lands between the grooves, the cooler outer part will constrain the expansion of this section relative to the section adjacent without a land, so giving rise to an axial tensile stress on the ID.

$$\Delta T = \frac{q \ln \frac{r_2}{r_1}}{2\pi l K}$$

$$\approx \frac{800 \ln \frac{27}{25}}{2\pi\ 0.15} = 20\,K$$

$$\equiv 20 \cdot 8 \cdot 10^{-6} = 0.0002 \text{ strain}$$

$$\text{or } \times 300 \cdot 10^{9} \equiv 47 \text{ MPa}.$$

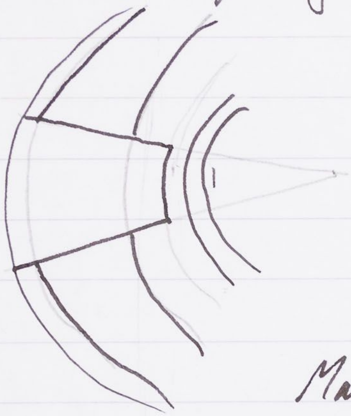
This amount of stress may well be enough to start a crack at grinding marks in the bore. However, it is also probably a marked overestimate because of the smaller heat production at this end of the core & various other simplifying assumptions. Still, may be worth trying  tapering off the grooves.

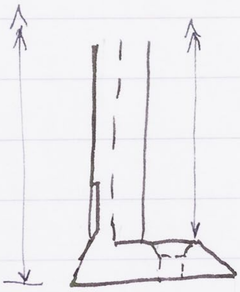
M3 pitch ϕ = ~ ~~2·8~~ 2·7 mm ≡ 5·7 mm^2 , support 4·6 kN at 800 MPa stress

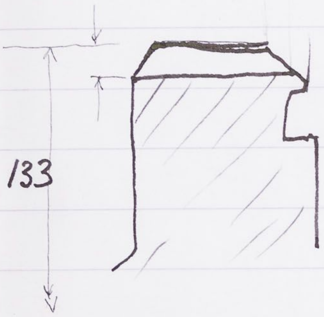
∴ 12 screws 55 kN .

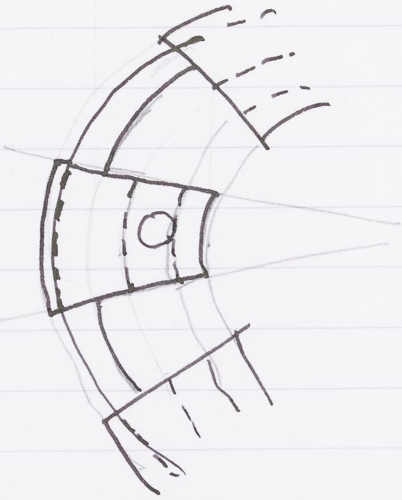
13/12/99

Bottom plug re-design for detachable upper section







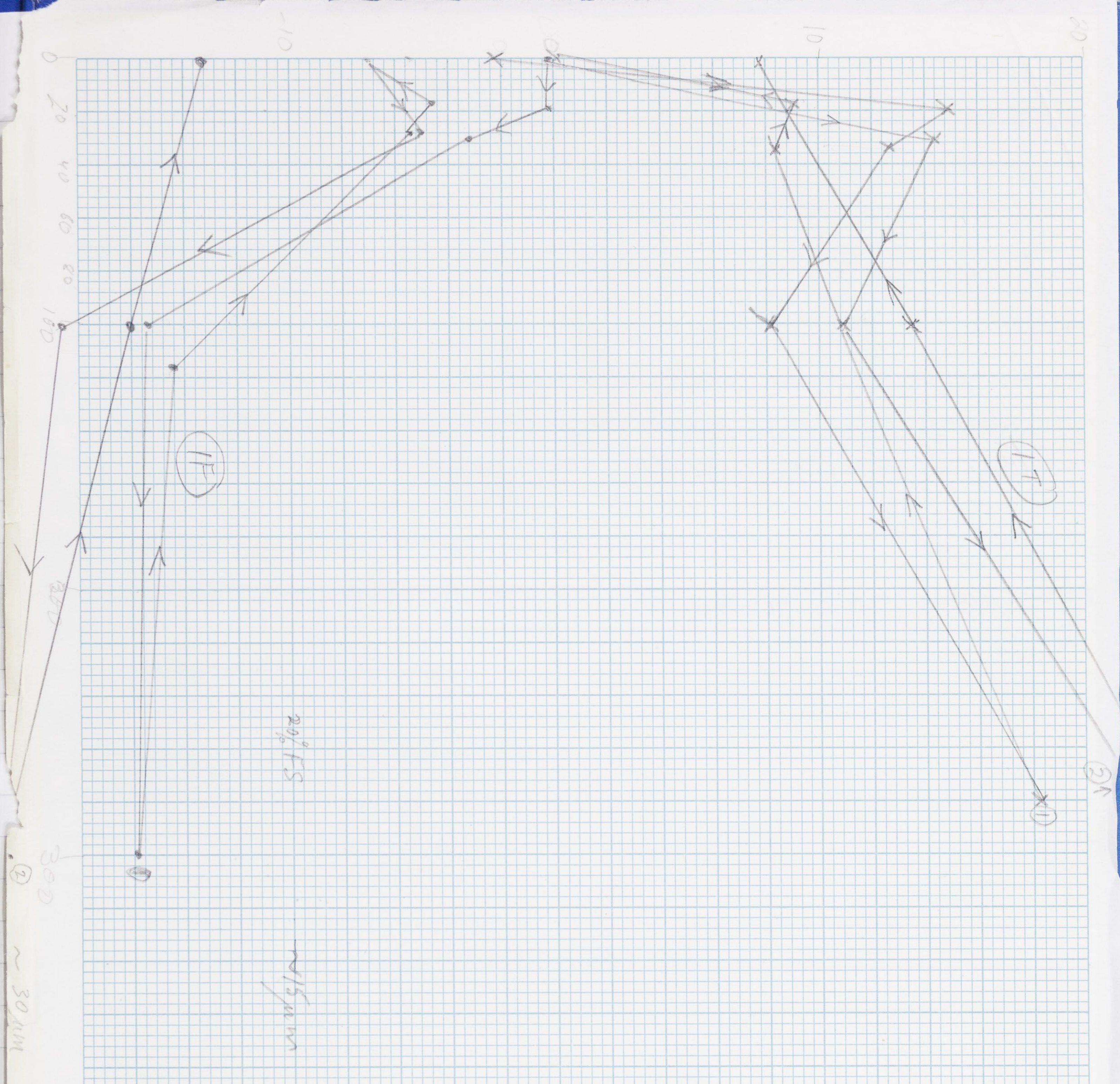


~~Th~~

See Drawing 4301E.

1000 Nm on 29mm radius gives force on splines = 34,500 N or 2870 N per spline.
Area of spline = 6×60 mm, so bearing stress = 8.0 MPa.

The 34.5 kN would give axial force approx the same if $\mu \approx 1$, but there should be substantial support for this from the transverse splines & the M3 screws would also probably support that much.



AARQUE A4 2mm.

Contd from p 184 (194)

LVDT load cell

12/1/00

The tests in Bayreuth (p142) showed that:

				Final reading
changing cooling water temperature 20-22°C,				
	1F shifted	−0·39 kN	run back somewhat	−0·3
	1T "	9·2 Nm	" "	−22
On letting in 19 MPa				
	1F shifted	+0·31 kN	& finally −0·20 kN	−0·2
	1T "	+132 Nm	unidirectionally.	150
Going 18 MPa to 33 MPa				
	1F shifted	+0·49 kN	& settled at −2·45 kN	−2·6
	1T "	−22 Nm	unidirectionally	127
Going 33 to 101 MPa				
	1F shifted	−14·3 kN	mostly unidirectionally	−15·3
	1T "	−95 Nm	− "	81
Going 101 to 300 MPa				
	1F shifted	−0·7 kN	after −6 kN excursion	~~−15·9~~
	1T "	+100 Nm	after +124 Nm "	183
Back to 108 MPa				
	1F shifted	+1·7 kN	mostly unidirectionally	−14·2
	1T "	−160 Nm	" "	109
Back from 108 to 34 MPa				
	1F shifted	+9·2 kN	unidirectionally	−5·4
	1T "	−24 Nm	after −32 Nm excursion	84
Back from 34 to 17 MPa				
	1F shifted	0·97 kN	after −1 kN excursion	−4·5
	1T "	+10 Nm	unidirectional	93
Back to zero pressure				
	1F shifted	−2·4 kN	after −4 kN excursion	−6·9
	1T "	−90 Nm	unidirectional	3

$$K = \frac{E}{3(1-2\nu)}$$

Repeated cycle:				Final reading
0 to 30 MPa :	1T	+1.7	after +3 excursion	−5
	1F	+141	" +163 "	144
30 − 100 MPa :	1T	−13.6	mostly unidirectional	−18.7
	1F	−136	after +63 excursion	108
100 − 300 MPa :	1T	− 1.9	after −7 excursion	−20.5
	1F	+109	after +130 "	217
300 − 100 MPa :	1T	+4.5	unidirectional	−16.0
	1F	−81	"	136
100 − zero MPa	1T	+ 2.3	after −7 excursion	−12.7
	1F	−57	after −106 "	79

Thus we are getting up to ± 40% of full scale displacements (this load cell being of 50 kN capacity), which are ~75 μm, ie equivalent displacements of 30 μm.

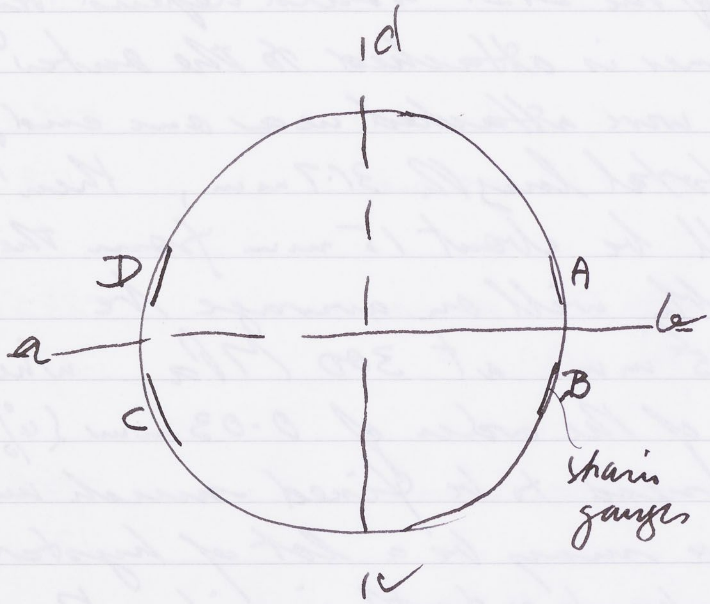
The above zeroshifts refer to settled situations of at least an hour or so, so they presumably do not reflect effects of temperature but are directly effects of pressure. A calculation of the effect of 10 GPa difference in a linear rigidity of ~400 GPa (eg between steel of load cell body & SS core stem) gives ~ 0.4 μm. This does not explain the large excursions noted above by a factor of 100×, which suggests a role of the plastic core tube. ~~(actually steel but plastic supported.~~

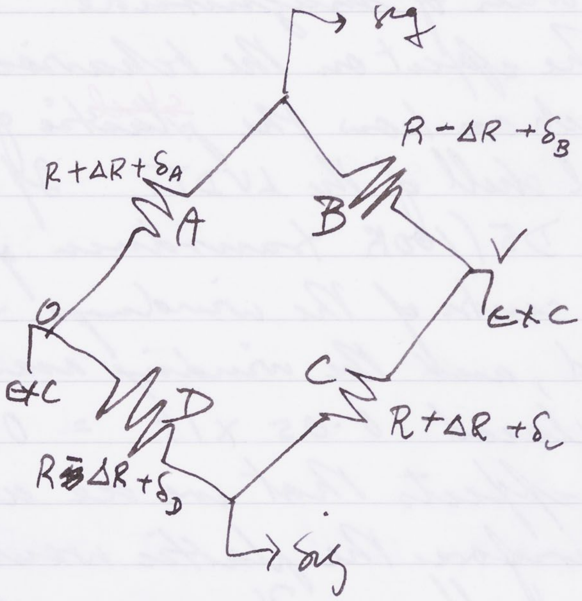
Most plastics have a Young's modulus & therefore approx bulk modulus of order of 1 or 2 GPa (100× less than steel). If we take K = 2 GPa, then applying 300 MPa = 0.3 GPa pressure would decrease linear dimensions by ~~4~~ $\frac{0.3}{3K}$ = $\frac{0.3}{6}$ = 0.05 , 5% as an order of magnitude.

The effect on the behaviour of the LVDT would depend very much on how the ~~plastic~~ steel former is attached to the outer steel shell of the LVDT. If it were attached near one end of the D5/100R transducer of total length 31.7 mm, then the centre of the windings will be about 15 mm from the end, and the winding assembly will on average be displaced 0.05 × 15 = 0.75 mm at 300 MPa, whereas the effects that we see are of the order of 0.03 mm (4% of 0.75) Therefore the plastic would need to be fixed much more centrally. However, there may be a lot of hysteresis associated with it sliding back & forth inside the steel outer tube, corresponding to non-reproducibility of the observations.

The coefficient of expansion of plastics is also around $50-100 \times 10^{-6} K^{-1}$, so if the plastic former is attached 15 mm from the centre, then 10 K of temp. change will produce $15 . 10 . 50 . 10^{-6} = 0.0075$ mm displacement. This is only $\frac{1}{100}$ of the compressibility effect. However, there seem to be up to 10% full scale drifts after changing pressure, so maybe these reflect elastic aftereffects in the plastic rather than temperature effects. Changing water temperature 2°C gives an effect of 0.17 kN in 50 kN FS, ie 0.34%, equivalent to 0.0034 × 0.075 = 0.3 μm of ~~couple~~ elastic element displace^t. — but this is also about 4% of predicted, as for the effect of pressure above. So it seems that the plastic former for the coil has an effective fixing position about 4% off centre

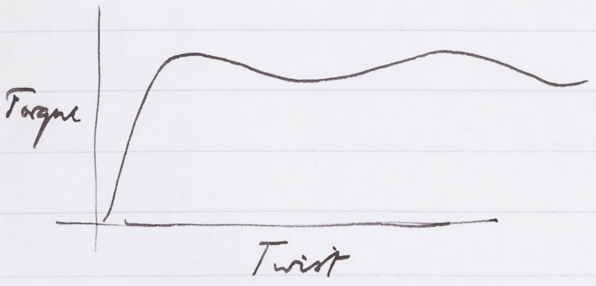
Cont^d p 201


d
D
A
a
b
C
B
strain gauges
c


sig
R+ΔR+δ_A
R − ΔR + δ_B
A
B
V
0
exc
exc
D
C
R+ΔR+δ_C
R−ΔR+δ_D
sig

Bending Effects in SG Load Cell

It is observed (Potsdam, Bayreuth) that torque twist curves



often are wavy, with a wavelength that is synchronous with the rotation of the driver. One possible cause is temperature fluctuation due to variation in loss of gas past the inner furnace seal with rotation.

Another is that there is a bending applied to the ILC which is not compensated.

If a pure torque is applied & the strain gauges are accurately at 45°, the changes in resistance are ±ΔR. Suppose $\delta_{A,B,C,D}$ are departures from this situation.

Then $$\frac{\Delta V}{V} = \frac{R + \Delta R + \delta_A}{2R + \delta_A + \delta_B} - \frac{R - \Delta R + \delta_D}{2R + \delta_c + \delta_D}$$

$$= \frac{(R+\Delta R+\delta_A)(2R+\delta_c+\delta_D) - (R-\Delta R+\delta_D)(2R+\delta_A+\delta_B)}{(2R+\delta_A+\delta_B)(2R+\delta_c+\delta_D)}$$

$$\approx \frac{4R\Delta R + R(\delta_c+\delta_D) + 2R\delta_A - R(\delta_A+\delta_B) - 2R\delta_D}{4R^2 + 2R(\delta_c+\delta_D) + 2R(\delta_A+\delta_B)}$$

$$= \frac{4\Delta R + \delta_A - \delta_B + \delta_c - \delta_D}{4R + \delta_A + \delta_B + \delta_c + \delta_D}$$

$$\approx \frac{4\Delta R + \delta_A - \delta_B + \delta_c - \delta_D}{4R}\left(1 - \frac{\delta_A + \delta_B + \delta_c + \delta_D}{4R}\right)$$

$$\approx \frac{4\Delta R + \delta_A - \delta_B + \delta_c - \delta_D}{4R} - \frac{4\Delta R(\delta_A + \delta_B + \delta_c + \delta_D)}{16R^2}$$

$$\approx \frac{\Delta R}{R} + \frac{\delta_A - \delta_B + \delta_c - \delta_D}{4R}$$ ~~4R~~

so the error signal $$\frac{\delta V}{V} = \frac{(\delta_A - \delta_B) + (\delta_c - \delta_D)}{4R}$$

For bending about ab, $\delta_A = -\delta_B = -\delta_c = \delta_D$, so $\frac{\delta V}{V} = \frac{2(\delta_A + \delta_c)}{2R} = 0$

For bending about cd, $\delta_A = \delta_B = -\delta_c = -\delta_D$, $\frac{\delta V}{V} = 0$.

If the bending is about an axis inclined to ab & cd, it can be resolved into ab and cd components, & so the compensation still holds.

Suppose the bending were about an axis a'b' or c'd' that does not pass through the load cell axis. Then we would have:

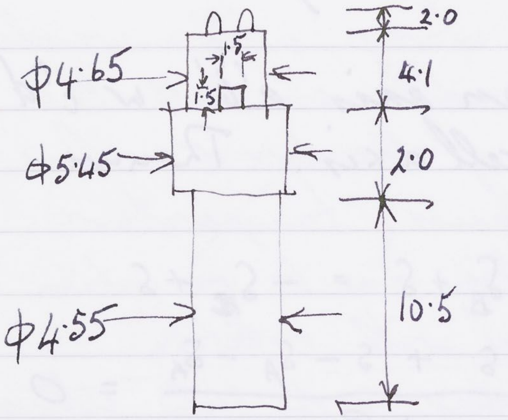
For bending about axis a'b', $\delta_A = \delta_D = -\delta_B + \delta = -\delta_C + \delta$

so $$\frac{\delta V}{V} = \frac{(\delta_A - \delta_B) + (\delta_C - \delta_D)}{4R} = \frac{\delta_A + \delta_A - \delta + \delta - \delta_A - \delta_A}{4R} = 0$$

For bending about axis c'd', $\delta_A = \delta_B = -\delta_D + \delta = -\delta_C + \delta$

so $$\frac{\delta V}{V} = \frac{\delta_A - \delta_A + \delta - \delta_A + \delta_A - \delta}{4R} = 0$$

So the load cell should also be insensitive to off-centre bending as well.



hole 4.7

hole 5.5

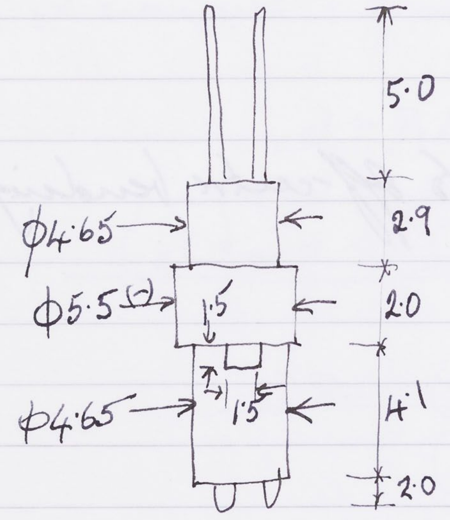
EGG 0B 302 2LL

Supporting ③ on ②:

$$\sigma = p\frac{d^2}{D^2-d^2} = p\cdot\frac{(6.5)^2}{(9.2)^2-(6.5)^2}\,2 = 0.997p$$

$$= 498\ \mathrm{MPa}$$

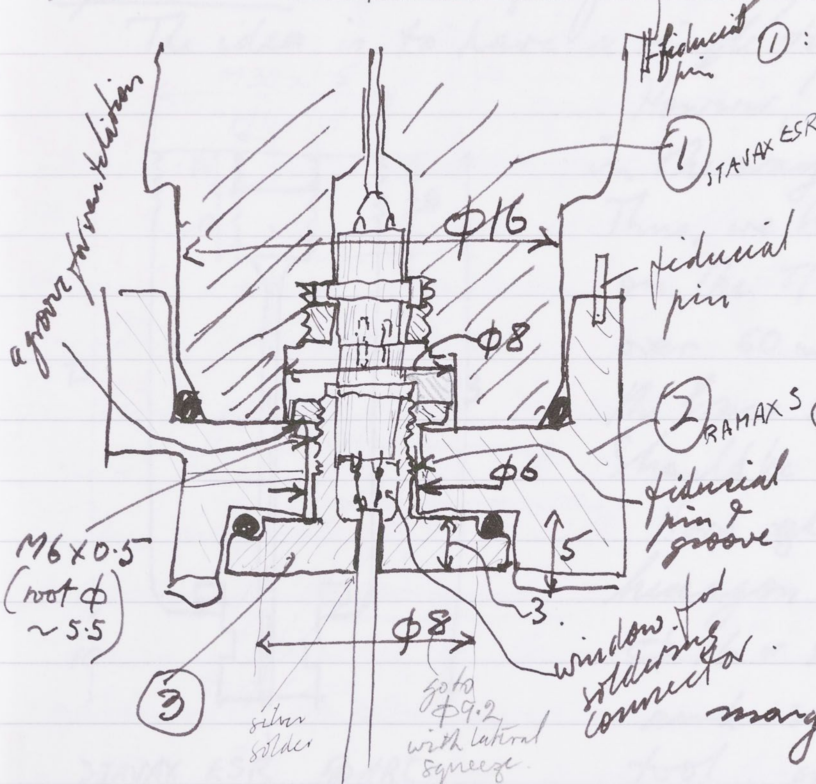
still OK for RamaxS or Assab 718



FGG 0B302 2LA

April '00 199

Detachable Bottom Thermocouple



①: Bottom piston: Φ8 hole in Φ16

gives $\sigma = p\frac{D^2+d^2}{D^2-d^2} = p\cdot\frac{5}{3}$

$= 830\,MPa$,

OK for Stavax ESR at HRC46 1470UTS.

②: Support of boss ③ against ②,

$\sigma = p\cdot\frac{d^2}{D^2-d^2} = p\cdot\frac{6^2}{\underset{(8.2)^2}{8^2}-6^2}$

$= \underset{0.74}{1.29}p = \underset{370}{640}\,MPa$,

marginally OK for RamaxS at 340HB 37HRC 1150MPa UTS

③: Shear strength of ③ over hole in ②.

$$\tau = \frac{\pi p d^2}{4\pi d t} = p\frac{d}{4t} = p\frac{6}{\cancel{3}12} = \overset{0.5p}{\cancel{2p}} = \overset{250}{\cancel{1000}}\,MPa,$$

~~totally inadequate~~. OK for RamaxS

④: Bearing pressure of ① on ②:

$$\sigma = \frac{p\frac{\pi}{4}15^2}{\cancel{p}\frac{\pi}{4}(15^2-8^2)} = \frac{15^2}{15^2-8^2} = 1.40p = 700\,MPa$$

OK on Ramax S & Stavax.

When we add axial load, extra $\sigma = \frac{100\,000}{\frac{\pi}{4}(15^2-8^2)} = 791\,MPa$.

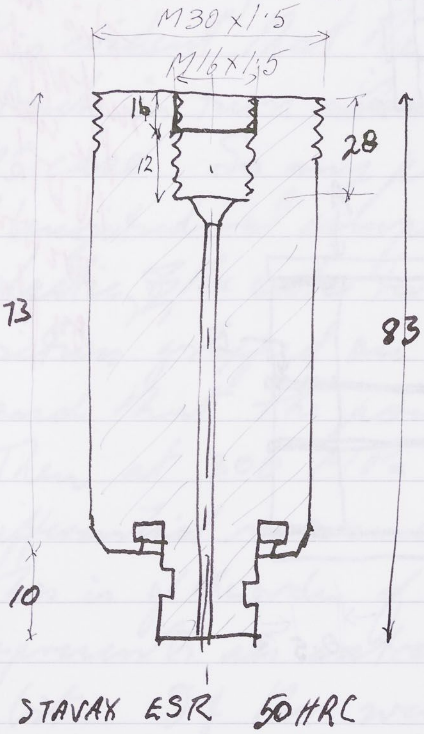
Total max σ = 1491 MPa at 100kN load.

This will exceed the flow stress of the RamaxS, so axial load is limited to about 50kN at 500 MPa. Can go to near 100kN at 300MPa.

Maybe should make ② out of Stavax ESR

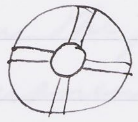
Top Piston – Redesign for O-ring seal & por fluid attachment.

The idea is to have a single top piston design.



However, the design at the left would get in the way of the traversing thermocouple. Thus, we have to shorten the brass sleeve on the T/C by 40 mm – still leaves over 60 mm thermocouple traverse before the brass sleeve comes out of the O-ring seal. Should be OK.

Nova gland nut is 24 mm long. The hexagon can be machined off to $\phi 16$ or slightly under, say $\phi 15{\cdot}9$ and slots put in for a tightening tool, say 4 slots



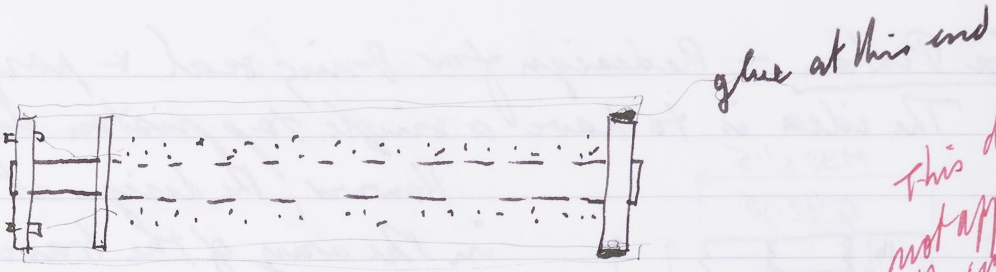
Tightening torque is 80 Nm, so at mean $\phi = 14$ mm, force is 5710 N, so if lugs are 2 × 4 mm, total shear area $= 8 \times 4 = 32$ mm^2, so shear stress on lugs = 179 MPa, OK for NiCrMo En25 steel.

Bearing area on piston $= \frac{\pi}{4}\left(28.376^2 - 16^2\right) = 431$ mm^2.

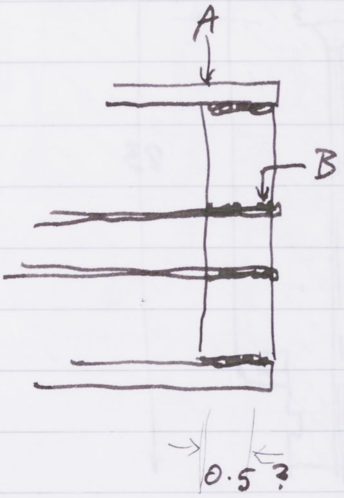
Max force at 500 MPa pressure and 100 kN load is

$$\frac{\pi}{4}(30)^2 \cdot 500 \quad + 100{,}000 \quad = 453{,}430\ \text{N}$$

so bearing pressure = 1052 MPa, close to the yield stress of the nut, which is 1100 MPa or so.



This does not apply to the smaller LVDT's which are wound on plastic



Cont^d from p 196

LVDT Load Cell – P & T effects analysed further.

On receiving the ~~St~~ failed & cut-open MD5/500K LVDT from Bayreuth, it is evident that the windings are on a stainless steel former, which in turn ~~are~~ is supported on plastic rings within the outer MS case. So any pressure effects must arise from some sort of longitudinal component of differential movement between the plastic, ~~&~~ the outer tube & the inner tube. Suppose the glue or friction gripped on the outer tube at A & inner tube at B and that the points A & B were 0·5 mm apart longitudinally. Then at 300 MPa with bulk modulus K = 2 GPa = 2,000 MPa, differential movement = $\frac{300}{2000} \cdot \frac{1}{3} \cdot 0{\cdot}5 = 0{\cdot}025$ mm = 25 μm. This is of the order of the effects estimated on p 195. But it represents an extreme case, so it is pushing the explanation a bit. If there were any inhomogeneity in the plastic, maybe there could be a plastic bowing component as well.

The coeff of expansion of the plastic is ~ 50 to $100 \cdot 10^{-6}$, ie at least ~~30~~ 40 $\cdot 10^{-6}$ differential relative to the steels. So a ΔT of 10K would add or subtract $40 \cdot 10^{-6} \cdot 10 \cdot 0{\cdot}5 = 0{\cdot}0002$ mm = 0·2 μm, a relatively small effect. So the temperature effects must be due to temperature differentials between ~~coil~~ winding and outer supports. If we take the ~~core~~ winding tube to be centred 15 mm from its support & we have a 10K differential in temperature between ~~the~~ winding & support, effect equals $15 \times 10 \times 40 \times 10^{-6} = 0{\cdot}006$ mm = 6 μm, still only about 10% of full scale.

For 2K ΔT, the effect is 0·001 mm ≈ 1 μm or $\frac{1}{75}$ FS of 50kN ie 0·6 kN — 0·4 kN was observed for a change in cooling water temperature of 2K — but this is not a ΔT effect, only a T effect, & so has to be a Δα effect of SS vs MS. To get a 0·6 μm effect, $0{\cdot}0006 = 15 \cdot \Delta\alpha \cdot 2$ so $\Delta\alpha = \frac{0{\cdot}0006}{30} = 20 \times 10^{-6}$. This is more than double the difference between SS & MS = $18 - 11 . 10^{-6}$
$= 7 . 10^{-6}$

but it is ~~the~~ almost the right order of magnitude

The manufacturers spec. is ± 0·01% FS/K , so for FS of 2·5 mm, the ~~span~~ zero temp ~~coef~~ change for 2K is
2 · 0·0001 · 2·5 = 0·0005 mm = 0·5 μm , so this is close to that observed, see p 194 : 0·4 kN shift in 50 kN range of ~75 μm ≡ $\frac{0{\cdot}4}{50} \cdot 75$ = 0·6 μm.

– Contd p210

Contd from p 191

May 2000.

NZP Furnace - Development & Problems.

Last November, I did some running of the NZP furnace in Zürich (no 26).

On the first run, it used 0·42 W/K getting to 786K with settings 360/0/1000 & wattages 37/0/177, controlling on bottom T/C. Then a short at a connection at bottom. Resolved this (same problem later in Bayreuth furnace - see below).

In subsequent runs, the performance was variable, as follows:

Set T at Bottom	Spec T	OP	W	~~W/K~~ W/K	settings
800 K	789 - 889 K	41-59	227-357	0·43 - 0·58	~ 300/0/1000
900	1039 - 1100	56-64	401 - 491	0·52 - 0·59	
1000	1107-1246	54 - 66	405-602	0·49 - 0·62	
1100	1269 - 1402	57-66	515-704	0·52 - 0·62	~ 270/210/1000
1200	1444	59	653	0·56	

No similar information from Bayreuth, but running another NZP furnace at RSES gave very inefficient behaviour - still to be checked up on.

In view of shorting problems in ETH & BGI furnaces, we re-designed the connector arrangements at the bottom & put in thicker nylon insulation in place of the mylar film used earlier.

Furnace 31 in Bayreuth (BGI) was of this revised design. It was first run up in May & failed. Profiles (±5K) were run at 1000 - 1300K with settings 300/300/1000. At 1400K, the bottom winding initially took 40V, 14 A, ie 560 W, so ~~bottom~~ total power was probably ~~< 420 W~~ 750 W, ie ~ 0·65 W/K, so definitely less efficient than above. Then voltage went down & current went up & OP changed from 75 to 85, with 410/340/1000 settings.

On opening up here, we observed the following:

1. Nylon insulation was discoloured, yellowish, & connectors looked heated.
2. Bottom turn of centre winding, and several turns near top & centre of lower winding had melted & the SiAlON also "melted" (?).

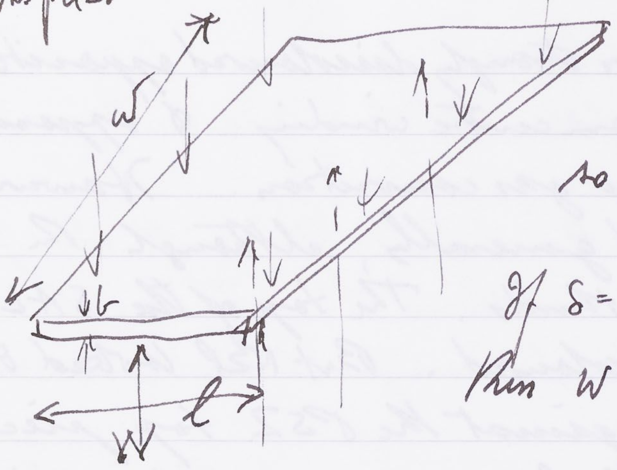
3. The SS segments around the SAL1 were strongly discoloured opposite the winding coming out through Al_2O_3 tube from centre winding & opposite the winding end at the top, indicating gas convection. However the alumina paper was not discoloured generally, although the SAL1 was darkened in the upper two-thirds. The top of the SAL1 was not contacting the PSZ & was discoloured. But NZP looked OK inside, & appeared to be snugly up against the PSZ top piece.

So it would appear that (a) the furnace was not operating very efficiently and (b) the bottom winding (and possibly centre?) was short-circuited by something conducting along the SiAlON. However, there was not a great deal of blackness normally associated with carbon. Is it possible that the SiAlON becomes conducting? If so, something irreversible happened to it since the resistance was 58 Ω at room temperature (and it is unlikely to have been as low as that initially or the furnace would never have worked). So a thin carbon film seems the most likely. From where? This was a new furnace.

The discolouration of the nylon is unlikely to be decomposition of it according to Zbigniew Stachurski (it might turn yellowish with oxidation but the atmosphere is more likely to be reducing). However, it could take up oil, [Nylon would also be fragile if degraded] as from machining — should not use a lubricant & run lathe slowly. Could decontaminate by heating in vacuum over 100-150°C.

$I = \frac{1}{12} w b^3$

Marks p425:



$$\delta_{\text{deflection}} = \frac{Wl^3}{48EI} = \frac{Wl^3}{4Ewb^3}$$

$$\text{so } W = \frac{4\delta E w b^3}{l^3}$$

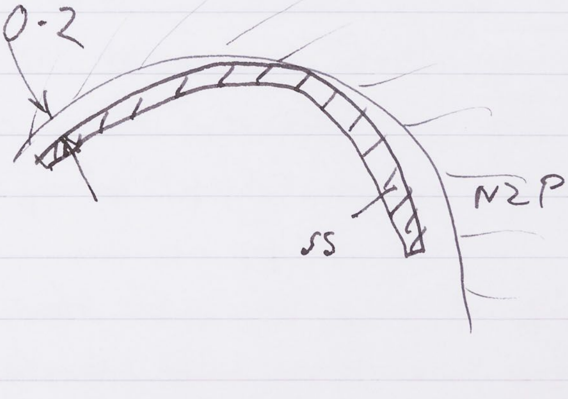
If $\delta = \overset{0.5\text{mm}}{0.0005}$, $E = 200.10^9$, $w = \overset{125\text{mm}}{0.125}$, ~~ilm~~ $l = \overset{50\text{mm}}{\cancel{50}}$ /m 0.05

$$\text{then } W = \frac{4 . 0.0005 . 200 . 10^9 . 0.125}{(0.050)^3} b^3$$

$$= 4 . 10^{11} \cancel{} b^3 \text{ kg}$$

For $b = \overset{1\text{mm}}{0.001}$, W = 400 kg – seems too high
$b = 0.0005$ 50 "
$b = 0.0002$ 3.2" – factor of 16 better than 0.0005 m.

} for 0.5 mm deflection



Tests on NZP Furnace no 29

This furnace had a slight flaw at the top end of the SiAlON & so was not delivered.

Ran it in RSES in "Green Machine" on 9/5/00 (notes in ETH Machine 9 notebook).

It used ~ 0·85 W/K at ~ 500°C, 800 K.

Ran again on 1/6/00 with the additions of the O-ring at top and single wave spring at bottom.

Now used ~ 0·72 W/K at ~ 500°C, 800 K. Got to 900K using 0·81 W/K. On going back to 700K, consumption was 0·72 W/K (compared with 0·67 on way up).

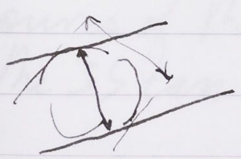
So there is probably a saving of ~ 0·1 W/K in putting in the O-ring & spring.

On dismantling the furnace, the following features were noted:

1) Dismantling would be facilitated by a better connector arrgt, to avoid flattening & catching of wires, and not putting stress on the core locating groove (do we need this?).

2) There were signs of gas staining opposite the splits in the SS segments (opp.), especially in one case where the gap was not well covered with alumina paper; the staining was esp. marked at the bottom end of the join. So there may have been a chimney effect.

3) The SS segments were only bearing on the NZP near their centres, ie they had higher curvature than the NZP. Measurements with calipers gave ~~0·60~~ 44·60 to 44·95 at the top and 44·58 to 44·90 at the bottom, ie a gap of ~ 0·2 mm, or maybe ~ 0·4 mm because of the threefold symmetry.



see summary
ETH book
p 62.

(206

Re-assembled furnace with the SS segments rotated 90° in azimuth so as to cover the joins in the SALI segments.

Ran furnace again on 9/6/00.

Worse than before ; 0·83 to 0·88 W/K (ETH9 notebook).

Bottom winding runs very cool.

Suspect circulation up outside of SS segments owing to them being of different curvature than NZP bore because of springing when cut. Also may be pinched in a bit to too small a diameter in lower part due to constraint in the SS bottom piece.

27/6/00

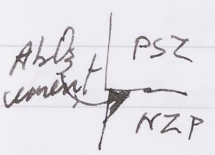
Ran again with SS segments straightened to same curvature as NZP. No difference in performance.

The notable feature is that the power consumption in the bottom winding is relatively high but the indicated temp. is lower. (see comparison in ETH Book p62).

Maybe there is a severe circulation around the bottom end somehow but the outward flow would have to be out through the gaps between the segments, no longer over the gaps in the SALI.

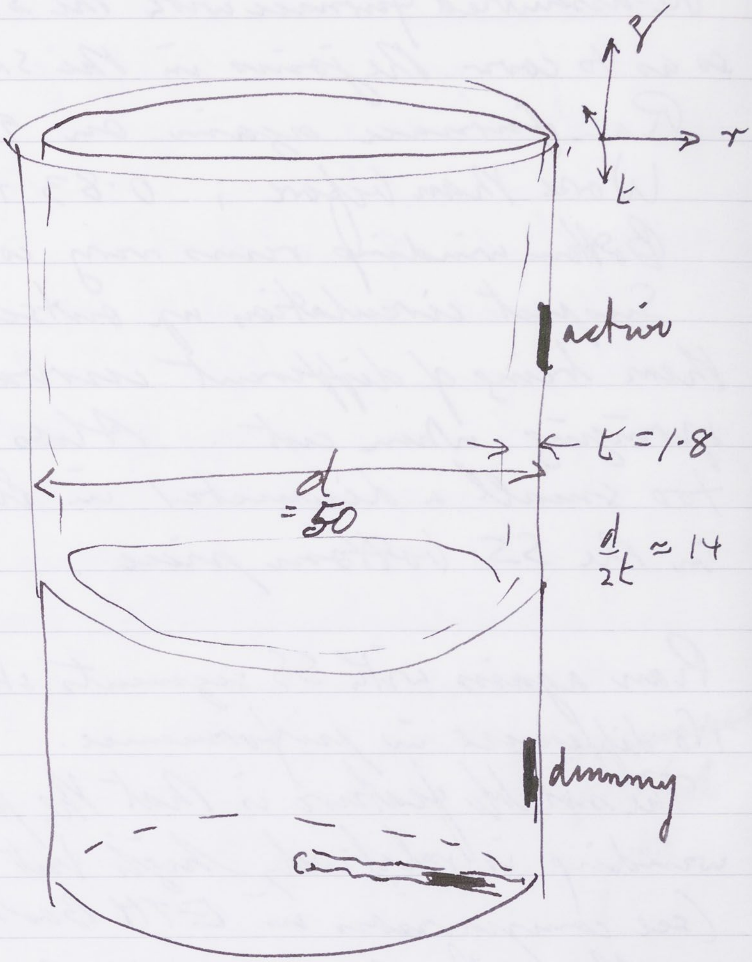
30/6/00

Found that one can blow air through the furnace with the opening in the top PSZ blocked off – ie it is getting through the PSZ-NZP interface (or through the NZP itself). Tried putting alumina cement in the interface. Power consumption now higher & deteriorated with time, so maybe the cement did not seal properly ; the change suggests that we are close to the source of the problem. Next can try an 035 O'ring between the SS can & the top plug.



Cont^d p 211

$$E\varepsilon_r = \sigma_r - \nu(\sigma_t + \sigma_z)$$
$$E\varepsilon_t = \sigma_t - \nu(\sigma_z + \sigma_r)$$
$$E\varepsilon_z = \sigma_z - \nu(\sigma_r + \sigma_t)$$



most of this thus comes from the Poisson effect from the radial expansion under internal pressure.

June 2000

Analysis of Pressure Effects on Strain Gauge Load Cell

Considering the inner unit on which the strain gauges are stuck, we first look at the strains experienced by the active gauge as pressure p is applied.

Before closure of the gap over the inner unit, $\sigma_r = 0$, $\sigma_z = p$ and $\sigma_t = -pd/2t$

$$\therefore E\varepsilon_z = p + \nu\left(\frac{pd}{2t}\right)$$

$$\text{or } \varepsilon_z = \frac{p}{E}\left(1 + \frac{\nu d}{2t}\right) \approx \underline{\frac{p}{E}(1+14\nu)}$$

Also $E\varepsilon_r = 0 - \nu\left(-\frac{pd}{2t} + p\right) = +\nu\left(\frac{pd}{2t} - p\right) = \nu p\left(\frac{d}{2t} - 1\right)$

$$\text{or } \varepsilon_r \approx \frac{13\nu p}{E} \quad \& \quad \Delta d = d.\varepsilon_r = \frac{13.0{\cdot}3.\ p\ .50}{200\ 000}\ \text{mm.}$$

If clearance is 0·020 mm, then closure at $p = \frac{200\ 000}{13.0{\cdot}3.50} \cdot \frac{0{\cdot}020}{2}$

$$\underline{= 10\ \text{MPa}}$$

→ Thus $\varepsilon_z \approx \frac{10}{200\ 000}(1+14\nu) = 0.00025$ strain
or 0.17 of full load strain 0.0015

In the solid section below the hollow cylinder, the value of ε_z is given by $\sigma_r = 0$, $\sigma_z = p$, $\sigma_t = 0$

$$E\varepsilon_z = p \ \cancel{- \nu(0 + p) = - \nu p}$$

$$\text{or } \varepsilon_z = \frac{p}{E}$$

until the gap closes, which will be at substantially higher pressure than the gap closure in the thin wall section.

The load cell response arises from the difference in these two strains, ie

$$\Delta\varepsilon_z = \frac{p}{E}(1+14\nu) - \frac{p}{E} = 14\frac{\nu p}{E} = 42\frac{p}{E}$$

so up to 10 MPa, $\Delta\varepsilon_r = 0{\cdot}0002$ or 14 % of full load strain.

If the dummy gauge is placed radially in the bottom; it detects ε_r, so

$E\varepsilon_r = 0 - \nu(0 + p)$ or $\varepsilon_r = -\frac{\nu}{E}p$

so measured $\Delta\varepsilon = \frac{p}{E}(1+14\nu) + \frac{\nu}{E}p = (1+15\nu)\frac{p}{E} \approx 5{\cdot}5\frac{p}{E}$

worse than axial position, 18 % FS

-0·0003 for another 100 MPa.

If closes at 100 MPa, $\Delta\varepsilon = -0{\cdot}0003$ or $+0{\cdot}00035$

$\equiv -20\%$ FS $\equiv +23\%$ FS

Total -6% FS Total 40% FS

<u>After thin-walled cylinder gap closure</u> & before closure of the gap over the solid cylinder:

In thin cylinder, $\sigma_r = \sigma_t = \sigma_z = p$

$$\text{so } E\varepsilon_z = p - \nu(p+p) = p(1-2\nu)$$

$$\text{or } \varepsilon_z = \frac{1-2\nu}{E}p.$$

In solid cylinder, we still have $\varepsilon_z = \frac{p}{E}$ or $-\frac{\nu}{E}p$

$$\text{so } \Delta\varepsilon_z = \frac{p}{E}\left(1-2\nu\cancel{3}-1\right) = \underline{-2\nu\frac{p}{E} = \frac{-0.6p}{E}}$$

$$\text{or } \frac{p}{E}\left(1-2\nu+\nu\right) = \underline{+\frac{0.7p}{E}}$$

<u>After closure of the gap over the solid cylinder</u>, we now have in the solid cylinder also $\sigma_r = \sigma_t = \sigma_z = p$, and so

$$\Delta\varepsilon_z = 0,$$

or if in the radial position, $\cancel{\Delta}\varepsilon_r = \cancel{\Delta}\varepsilon_z$ so $\Delta\varepsilon = 0$; doesn't really matter what orientation.

Thus the zero shift effects are

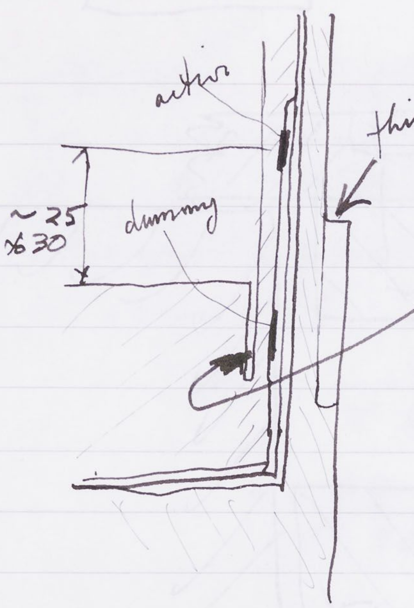
1) Poisson effect from radial expansion up to ~10 MPa, which gives +14% FS zero shift for axial dummy
+18% FS " " radial dummy.

2) Continued contribution from the solid cylinder up to higher pressures, maybe ~100 MPa, in which case we get an extra -20% FS zero shift for axial dummy
+23% " " for radial dummy

giving total shift = −6% FS for axial dummy
+41% " radial "

The latter may be the case in Bayreuth (memory??)

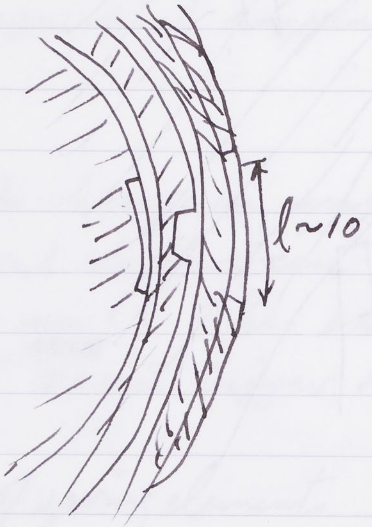
For best pressure compensation, we need to have the cylinder gap taken up as early as possible & the dummy in a vertical orientation & under as nearly similar pressure conditions as the active gauge as possible.



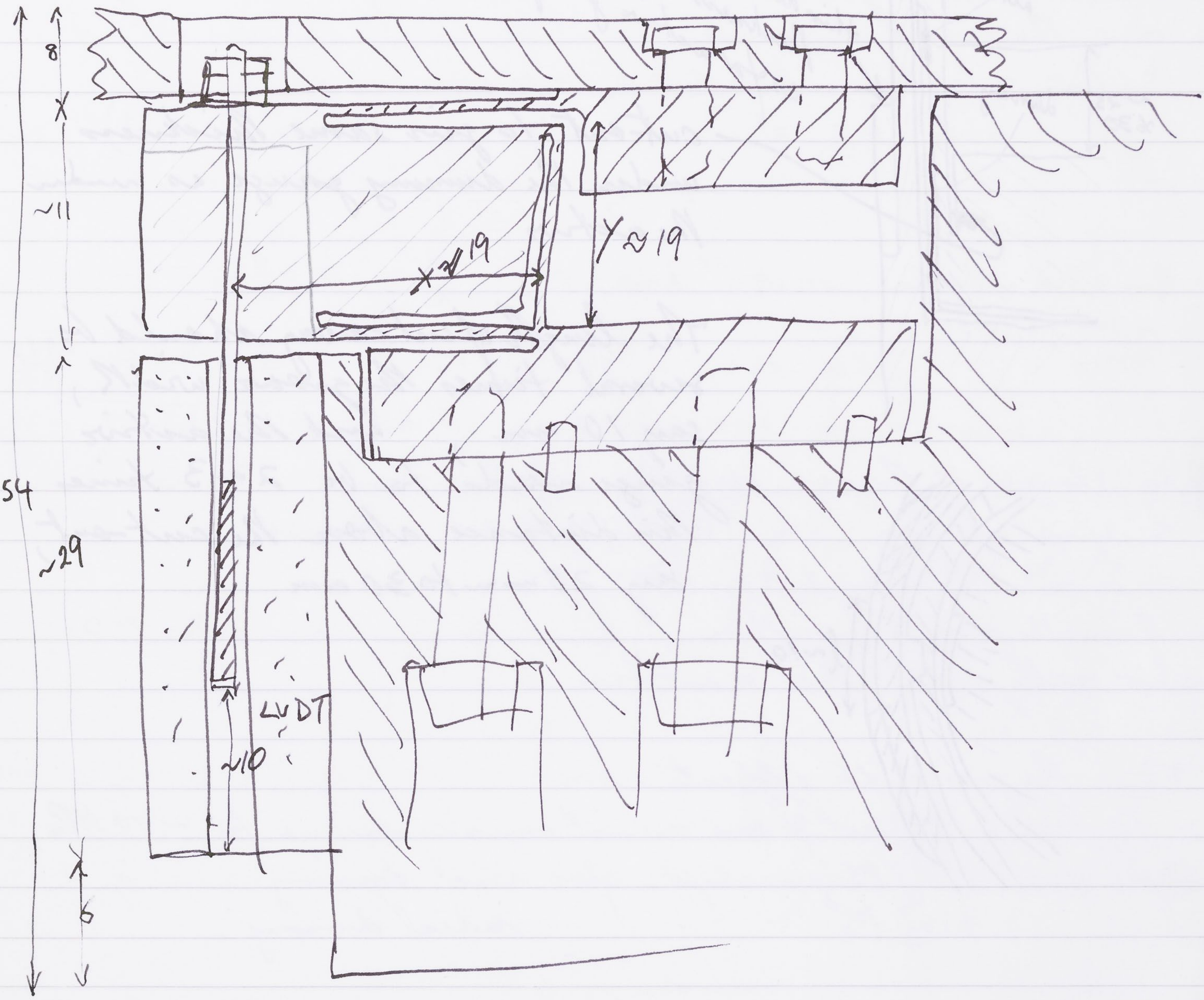
thin down here to get same thickness over gauge groove as for active gauge.

cut-out to give same thickness under the dummy gauge as under the active

The length of cut-away should be several times the groove width, say 10 mm, and the active gauge needs to be 2 ~ 3 times this distance above the cut-out, say 25 mm to 30 mm



cf p169



l = 10 mm

Contd. from p 202

LVDT loadcell – re-arrangement for axially-mounted torsion LVDT

For no magnification, need $X \simeq Y$, probably ~10 mm.

From p 184, if elastic element is 52OD 48ID,

From Paterson & Olgaard (2000) equation (7), angular twist at torque M is

$$\theta = \frac{32 M l}{\pi G \left(D^4 - d^4\right)}$$

where
$M = 1000$ Nm
$l = 54$ mm
$D = 52$
$d = 48$
$G = 80$ GPa

$$= \frac{32 . 1000 . 0{\cdot}054}{\pi . 80.10^9 \left(0{\cdot}052^4 - 0{\cdot}048^4\right)}$$

$= 0{\cdot}0034$ radians

so tangential displacement at radius 25 $= 0{\cdot}086$ mm.
" " " 18·5 (PCD 37) $= 0{\cdot}064$ mm

Therefore the change in angle of the LVDT core is $\sim \frac{0{\cdot}064}{19}$ rad.
so lateral deflection at end of core is about $\frac{0{\cdot}064}{\cancel{10}\,19} . 20$
$\simeq \overset{0{\cdot}07}{\cancel{0{\cdot}128}}$ mm! This should be no problem if the LVDT bore is at least 0·5 mm bigger than the core.

Deflection of spring elements $\delta = \frac{Wl^3}{3EI} = \frac{Wl^3 12}{3Ebh^3}$ for cantilever $I = \frac{bh^3}{12}$

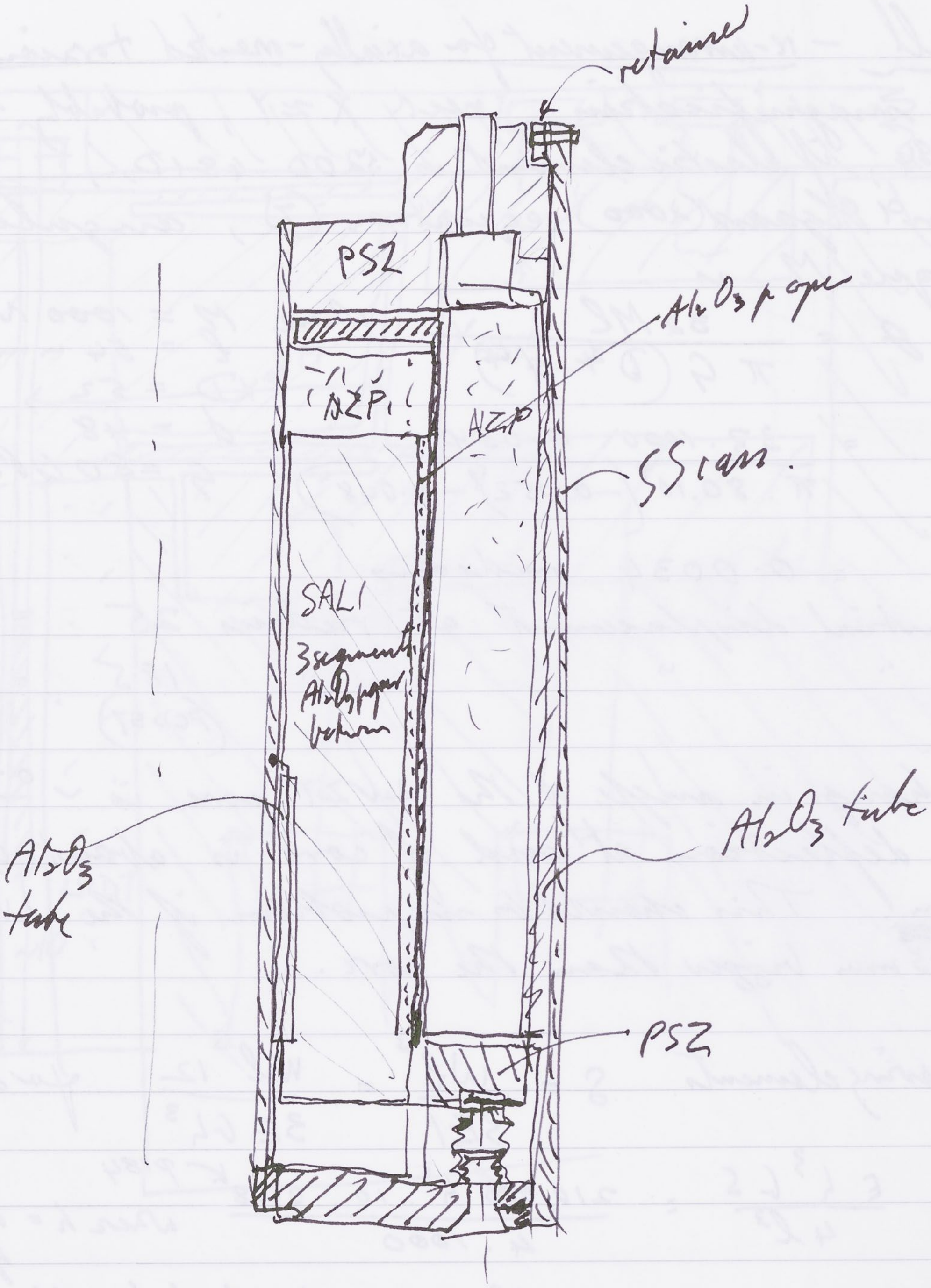
so $W = \frac{E h^3 b \delta}{4 l^3} = \frac{210{,}000\, h^3 . 10 . 0{\cdot}08}{4 . 1000}$ K p184 where h = thickness of spring in mm

for $h = \cancel{0{\cdot}0}\,0{\cdot}5$ $= 5{\cdot}3$ N probably tolerable
$h = 0{\cdot}3$ $= 1{\cdot}1$ N better.

Contd p214?
219)

With $\delta = 0{\cdot}2$ mm,

$h = 0{\cdot}5$	$W = 13$ N
0·4	7
0·3	3



See recalculation p 220 et seq.

Cont^d from p 206

NZP Furnace – inner can modification

Test 18/7/00: With 036 O-ring on top of furnace as well as inner one there. Still ~ 0.9 W/K (ETH-9 notebook). Concluded problem is nearer the bottom of the furnace. On inspection:

- still gaps between SS segments & NZP
- some chimneys in Al_2O_3 paper where earth windings go.

This gives rise to the thought of a complete SS can inside the NZP. Need a jig to compress the alumina paper / SALI assembly for pushing in — would probably work (tested on a flat plate — alumina paper did not pucker up).

Bore of NZP = 45. From p 178, temperature at bore of NZP (d = 45) will be ~ 1000 K for 1700 K spec temp — temp rise of ~ 700 K

1000 K ~ 670 K for 1300 ~~1000~~ K spec temp " " 530 370 K

~ 510 K for 700 K " " " 210 ~~270~~ K.

~ 405 ~~360~~ K for 500 K " 110 ~~140~~ K.

Coeffic of expansion of NZP ~ 1.10^{-6}!, SS ~ 16.10^{-6}, so differential 15.

At diameter 45, relative expansion will be strain

at 1700 K spec temp : $700 . 15 . 10^{-6} . 45$ = 0.47 mm 0.010_4

1300 1000 K " " 530 370 = 0.36 0.25 mm 0.008 0.005_6

700 K " " 210 ~~270~~ = 0.14 ~~[illegible]~~ mm 0.003_1

500 K " " 110 ~~140~~ = 0.07 ~~[illegible]~~ mm 0.001_6

If the SS can has a clearance initially of 0.1 mm, then it will make contact with the NZP in the central region when the specimen temperature is around 600 K (300°C).

With 0.1 mm clearance, the interference at 1700 K spec. temp will be 0.37 mm, or a strain of 0.0082. This will be partitioned between expansion of the NZP and compression of the SS.

Diametral strain for hollow cylinder under internal pressure (Lab Bk 9 p12) is

$$\varepsilon = \frac{\Delta d}{d} = \frac{p}{E}\left(\frac{D^2+d^2}{D^2-d^2} + \nu\right)$$

For the NZP, $\varepsilon = \frac{p}{50000}\left(\frac{61^2+45^2}{61^2-45^2} + 0\cdot3\right)$ with $E = 50$ GPa for NZP & $\nu = 0\cdot3$ (guess)

$$= 74 \times 10^{-6}\, p/\text{MPa}.$$

p is limited by yielding of the SS inner can, say ~~[illegible]~~ 0·3 wall thickness, 45 ϕ and y.str. of 600 MPa (~¼ hard)

$$p = \frac{0\cdot6}{45} . 600 = 8 \text{ MPa}$$

so ε = 0·00059 or Δd = 0·027 mm, say 0·03 mm at most. The rest of the differential expansion has to be taken by distortion of the inner can, as follows

Spec temp	1700K	differential strain = 0·0098	elastic stress =	2,060 MPa
	1300K	= 0·007$_4$	=	1,560
	1000K	= 0·005$_6$	=	1,050
	700K	= 0·0025	=	530
	500K	= 0·0010	=	200 MPa

So above around 800K specimen temperature, the inner SS can will yield, giving rise to permanent strain 0·002$_6$, Δd = 0·12 mm at 1000K
0·004$_4$ 0·20 " at 1300K
0·0068 0·31 " at 1700K

So even if we give the inner can 0·1 mm clearance to begin with, this clearance will become about 0·2 mm after going to 1000K, 0·3 mm after 1300K and 0·4 mm after 1700K

An alternative is to use Ti or a Ti alloy. For commercially pure Ti, $\kappa = 17$ W/mK, $\alpha = 9.10^{-6}$/K & $\sigma_y = 275-450$ MPa, typical 300 min, $E = 115$ GPa at RT
for 6Al-4V, $K = 12$ @ 400°C, $\alpha = 10.10^{-6}$ @ 400°C, $E = 107$ GPa @ 400°C, $\sigma_y \sim 1000$ MPa at RT
700 MPa at 400°C

So at 1700K specT, relative ~~α~~ $\varepsilon = 9.10^{-6}.700$ = 0·0063 or 0·28 mm.
1300K " 530 = 0·0048, 0·21 mm
1000K 370 = 0·0033 0·15 mm.

but how much of this strength is retained at 1000K

this assumes no initial clearance

So differential strain on can = 0·0063 − 0·0006 = 0·0057, elastic stress = 627 MPa. Thus the 6Al4V titanium alloy would not yield if there was ~~an~~ no initial ~~0·1mm~~ clearance but pure Ti would.

If we have an initial clearance of 0·05 mm, then differential strain in can at 1700K = 0·23/45 = 0·0051 − 0·0006 = 0·0045, elastic stress = 500 MPa. 430 MPa with $\alpha = 9.10^{-6}$ This would give rise to a permanent strain corresp to $\overset{130}{200}$ MPa non-elastic, ie non-elastic strain of $\overset{0\cdot0012}{0\cdot0018}$ or $\overset{0\cdot053}{0\cdot082}$ mm of permanent distortion, making total clearance $\overset{0\cdot10}{0\cdot132}$ mm at RT.

To check again, consider a specimen temperature of 1600K, ie 1300K above ambient, and an initial clearance of 0·05 mm. The differential expansion between NZP & Ti is $\sim 9.10^{-6} \cdot (\overset{(P178)}{989}\ \cancel{45} - 300).45$ = 0·28 mm, so differential interference = 0·28 − 0·05 = 0·23 mm or 0·0051 strain without allowing for can strain, and is 0·0051 − 0·0003 = 0·0048 when yielding of the can at 300 MPa is taken into account ~~in~~ for strain in the NZP. The elastic limit of pure Ti is 300/115000 = 0·0026, leading to a permanent strain in the can of 0·0022, or 0·10 mm, leading to a clearance of 0·15 mm after the run. However, for ~~For spec. temp~~ 6Al4V the elastic limit is $\frac{700}{105000}$ = 0·0067 so there would be no permanent set at any specimen temp. but it would creep.

In order that the initial clearance be closed up by the stage at which the specimen temp. reaches 700K, ie temp. rise of 210 K at d = 45 mm, the initial clearance Δd is given by $\Delta d = 9.10^{-6} \cdot 210 \cdot 45 = 0\cdot085$ mm. So an initial diameter of 44·95/44·90 should be OK.

See p220

$$e = K_1 x \left(1 - \frac{x^2}{K_2}\right)$$

$$= K_1 x - \frac{K_1}{K_2} x^3$$

2/12/00

LVDT Load Cell - Design of LVDT.

The small LVDT's from RDP are actually wound on plastic (in contrast to the ±½" travel LVDT used for position measurement, which is wound on metal & does not show pronounced pressure effects, although the gain is a couple of orders of magnitude less). No other manufacturer of similar range LVDT's supplies with metal former that I can discover. So this leads us to trying to make our own LVDT's. A metal former was devised for tests at Bayreuth but to date they have not been successful in getting a short-free winding.

It may be worth thinking about how better to tailor the LVDT design to our purposes. The basic design considerations are set out in a paper by P D Atkinson & R W Hynes "Analysis & design of a linear differential transformer" The Elliott Journal 1954 2 144-151, of which I have a photocopy, & this is reproduced in H K P Neubert "Instrument Transducers" Oxford 1963 almost verbatim. This shows that the sensitivity of the transducer (voltages per unit displacement) is given by

$$K_1 = \frac{16\pi^{3/2}}{10^9} \cdot \frac{f I_p N_p N_s}{\ln(r_o/r_i)} \cdot \frac{(b+2d)x_0 + x_0^2}{m L_a} \quad ①$$

where f = frequency, I_p = primary current, $\left.\begin{matrix}N_s\\N_p\end{matrix}\right\}$ = number of $\left.\begin{matrix}\text{secondary}\\\text{primary}\end{matrix}\right\}$ turns, r_o, r_i radii of outer & inner surface of coil, $x_0 = \frac{x_1 + x_2}{2}$ where x_1, x_2 are the depths to which the core penetrates the secondary coils, L_a is total length of the core, m is length of secondary coil, d is spacing between cores, & b is the length of the primary coil.

The coefficient of linearity is given by

$$K_2 = (b + 2d)x_0 + x_0^2$$

so increasing the length of the primary improves linearity. So we need to make the primary relatively long & the secondary coils relatively short.

$N = 860, X = 82{\cdot}5$

$R = 34{\cdot}4$

$Z = \sqrt{82{\cdot}5^2 + 34{\cdot}4^2}$

$= 89{\cdot}4\,\Omega$

$\frac{V}{I} = \frac{5}{0{\cdot}06} = 83$

In our case, we want to drive the LVDT's with a RS card, which gives a total excitation current of 50 mA per card (I thought this was per channel but Paul thinks per card). If we have two cards for the six LVDTS (POS, IF & IT) & the POS LVDT's are excited in series & IF & IT in parallel, we have in effect $4\frac{1}{2}$ transducers taking up to 100 mA ie 22 mA per transducer. Thus the primary current should be perhaps 15 mA as an aim.

The inductive impedance of a coil Giancoli (p 553 & 556) is $X = 2\pi f L = 2\pi f \frac{\mu_0 N^2 A}{l}$ where f = freq, $\mu_0 = 4\pi.10^{-7}$ in air, N = number of turns, A = cross-section of coil $= \frac{\pi d^2}{4}$, l = length

$$\text{so } X = \frac{2\pi f \mu_0 N^2 \pi d^2}{4l} = 2\pi^3 10^{-7} f . \frac{N^2 d^2}{l} \quad \text{———} \quad ②$$

$$= 0{\cdot}0310\, \frac{N^2 d^2}{l} \text{ for } f = 5\,kHz$$

For a coil of avge diameter ~6 mm and length 10 mm,

$$X = 111{\cdot}6.10^{-6} N^2$$

$$= 112\ \Omega \text{ for } N = 1000 \text{ turns}$$

$$\text{or } \overset{251}{446}\ \Omega \text{ for } \overset{1500}{2000} \text{ turns}$$

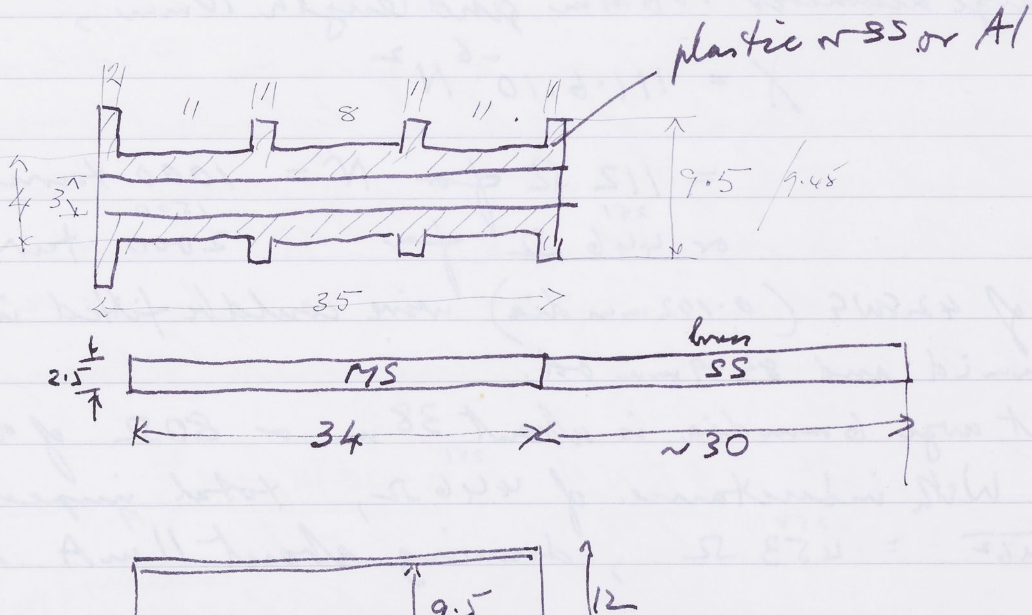
2000 turns of 42SWG (0·102 mm dia) wire could be fitted in 10 mm with a coil 4 mm id and $8\frac{1}{2}$-9 mm od.

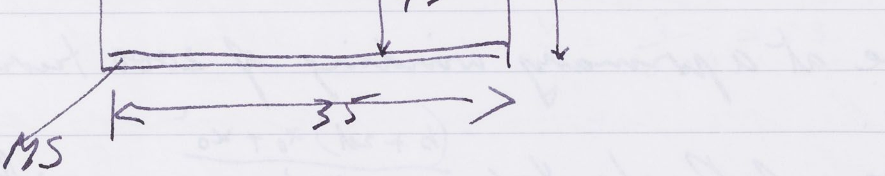
2000 turns at avge 6 mm dia is about 38 m or 80 Ω of d.c. resistance. With inductance of $\overset{251}{446}\ \Omega$, total impedance is $\sqrt{80^2 + 446^2} = \overset{258}{453}\ \Omega$, drawing about $\overset{19}{11}$ mA at 5V 5 kHz.

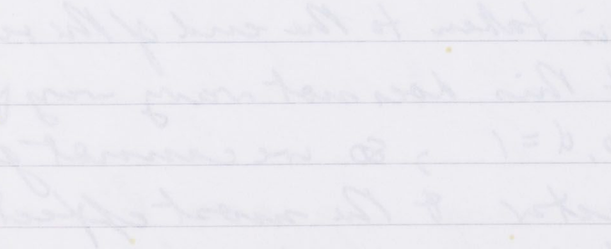
Thus we arrive at a primary winding of 2000 turns in a length of 10 mm.

Consideration of the factor $\frac{(b+2d)\,x_0 + x_0^2}{m L_a}$, which is approx $\frac{b+2d+m}{b+2d+2m}$ when the core is taken to the end of the secondary for maximum sensitivity, & this does not vary very fast with $m \sim 5$ to 10 when $b = 10$, $d = 1$, so we cannot gain much by playing with this factor & the most effective way of increasing sensitivity is to increase N_s. Space limitations

not true


plastic or SS or Al
12
11
11
8
11
11
11
4
3
9.5
9.45
35


brass
2.5
MS
SS
34
~30


9.5
12
35
MS

dictate a length not much greater than 10 mm, so again we are looking at 2000 turns. With $m = 10 = x$ & $L_a = 10 + 2 + 20$ the first factor in ① is $\sim 0{\cdot}69$ and $\ln\left(\frac{r_0}{r_i}\right) \sim 0{\cdot}69$ for $\begin{matrix} r_0 = 8 \\ r_i = 4 \end{matrix}$ so ① becomes:

$$K_1 = \frac{16\pi^2}{10^9} \cdot f \, I_p N_p N_s$$

$$= \frac{16\pi^2}{10^9} \cdot 5000 \cdot 0{\cdot}011 \cdot (2000)^2 \qquad \text{for } N_p = N_s = 2000$$

$$= 34{\cdot}7$$

ie a sensitivity of 35 V/m or 35 mV/mm (5mV/V/mm)

a typical RDP transducer gives 70 mV/V/mm so we may be down on this by a factor of 2.

At a displacement of 75 μm for fullscale load, we would have ~~35~~ $0{\cdot}035 \times 0{\cdot}075 \times 5 = 0{\cdot}013$ V, needing a gain of 762 × in order to get an output of 10V. The RS card gives 460 : 1 switched, 3 : 1 variable, so presumably up to 1380 ×, or twice as much as needed?? Getting close to the limit.

Actually by putting two transducers in series, we double this output & so half the gain needed.

To improve performance, we may want to shorten the primary a bit if we can still get 2000 turns on, or reduce the number of turns if the current was lower than necessary. Since $X_p \propto N_p^2$, we have

$$I_p = \frac{V}{X_p} \propto \frac{V}{N_p^2}, \text{ and } K_1 \propto I_p N_p \propto \frac{V}{N_p^2} \cdot N_p \text{ ie } \propto \frac{1}{N_p}.$$

So we gain some sensitivity by reducing N_p provided I_p does not become too large.

With $\overset{8\,mm}{1600}$ turns on primary, $I_p = 19$ mA, and $\overset{11\,mm}{2200}$ turns on the secondary, we would get 53 V/m sensitivity, an improvement of 50% — quite useful.

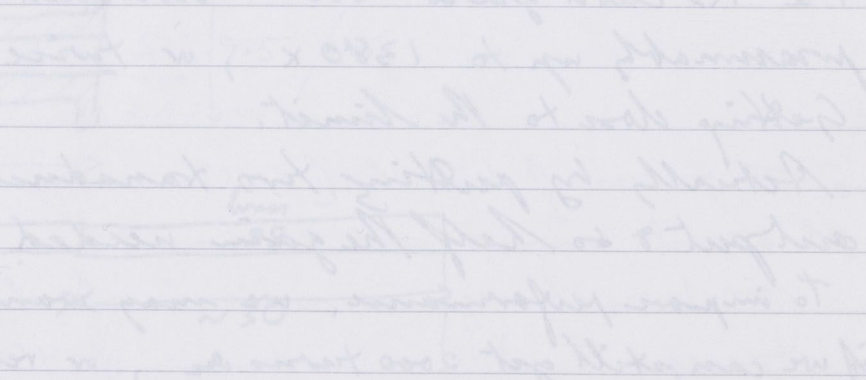
Should do a trial as opposite and check — current, sensitivity, linearity & effect of length of core.

Increasing f will be ineffective because its effect in K_1 will be offset by the effect in I_p through X_p — unless the current is too high for the RS card.

$\frac{N_p}{b}$ is number of turns per mm of primary, so this has to be as small as possible, & N_s as big as possible.

And d', the effective diameter of primary winding has to be as small as possible.

$\boxed{1.4 \text{ mV/V full scale}}$



From ②, we have $I_p = \frac{V}{X_{\cancel{L}}} = \frac{V l_p}{2\pi^3 . 10^{-7} f N_p^2 d^2}$ — neglecting the resistance component.

& substituting this in ① gives

spacing of coils

$$K_1 = \frac{16\pi^2}{10^9} . \frac{V l_p}{2\pi^3 . 10^{-7}} . \frac{1}{f N_p^2 d^2} . \frac{f N_p N_s}{\ln\left(\frac{d_o}{d_i}\right)} \quad \frac{(b + 2d')x_o + x_o^2}{m L_a}$$

dia of coil

$l_p = b$

$$= \frac{Vb}{12.5\pi d^2} \frac{N_s}{N_p} . \underbrace{\frac{(b + 2d')x_o + x_o^2}{m L_a \ln\left(\frac{d_o}{d_c}\right)}}_{\text{around unity from above}}$$

ie $\boxed{K_1 \simeq \frac{Vb}{12.5\pi d^2} \frac{N_s}{N_p}.}$ — ③ where b is length of primary & d is ~~dan~~ its diameter.

Putting $b = 0.008$, $d = 0.006$, $V = 5$, $N_s = 2200$, $N_p = 1600$,

we have $K_1 = \cancel{29\ V/m}\ 28.3 . \frac{N_s}{N_p}$ ~~≈ 29 mV/mm~~ $= 39$ mV/mm

a bit less than calculated before. or 47 mV/mm for $N_s = 2500$, $N_p = 1500$.

Note that K_1 is independent of frequency and is proportional to the length of the primary winding & inversely proportional to the square of its diameter. The limitation on this quantities is determined by the current magnitude.

ie $\boxed{K_1 = 28.3 \frac{N_s}{N_p}}$ for $b = 0.008$, $d = 0.006$, $V = 5$.

So we can get up to ~~10~~50 mV/mm out of one transducer & hence about 100 mV/mm out of the two in series.

For 70μm full scale, output voltage = 7 mV, which has to be amplified up to 10V, ie gain of 1429x.

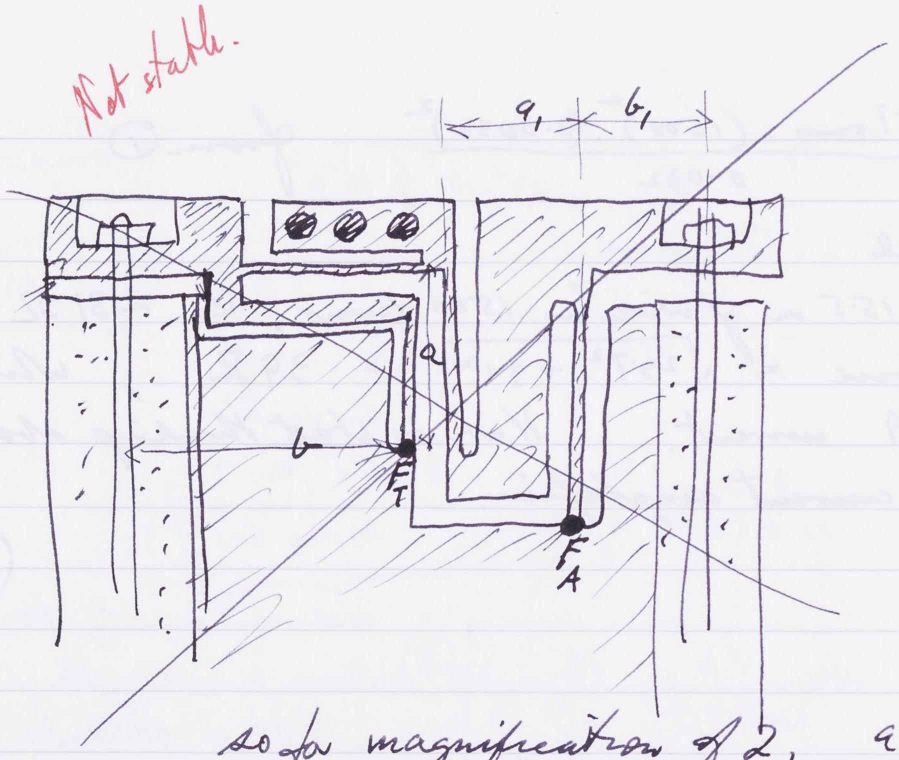
Further gain could be got by spreading the primary winding over the full 32 mm length & reducing the former diameter so that the ~~size~~ average coil diameter is 3.3 mm, we get $K = 374 \frac{N_s}{N_p}$ or of the order of 500 mV/mm, ie an order of magnitude better. However, the catch is that the primary impedance is reduced & so the current goes up. We would need 4800 turns to keep the ~~inductance up to~~ inductive impedance up to 250Ω. At 1500 turns, $X = \frac{2\pi^3 10^{-7} 5000 . (1500)^2 . (0.0033)^2}{0.032}$ from ①

$= 23.7\,\Omega$

There will be about 15.5 m of wire in 1500 turns, ie $\sim 31\,\Omega$
so total impedance $= \sqrt{23.7^2 + 31^2} = 39\,\Omega$, which will lead to 128 mA current. Have to start thinking about heating, or putting in a current amplifier.

Contd p 227

Not stable.

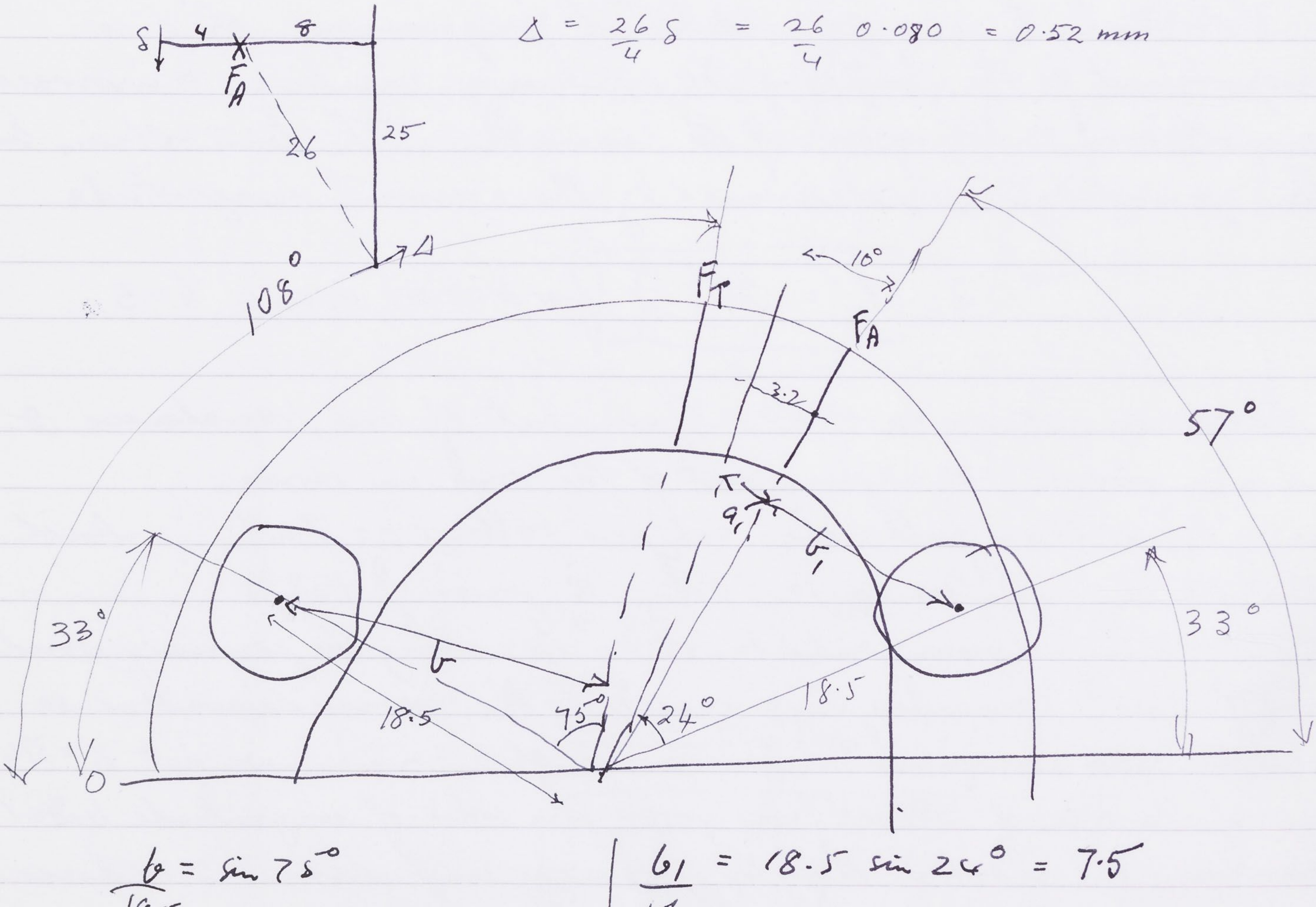


Typical values for LVDT's spaced 114° (37 mm on PCD 37) are $b = 24$, $a_1 + b_1 = 12$ approx.

so for magnification of 2, $a = 12$, $b = 24$

$a_1 = 4$, $b_1 = 8$

$\Delta = \frac{26}{4}\delta = \frac{26}{4}\,0{\cdot}080 = 0{\cdot}52$ mm



$\frac{b}{18{\cdot}5} = \sin 75°$

$b = 17{\cdot}9$

say 18 so need $a \sim 9$ but $a = 12$, so $m = 1{\cdot}5\times$

$b_1 = 18{\cdot}5 \sin 24° = 7{\cdot}5$

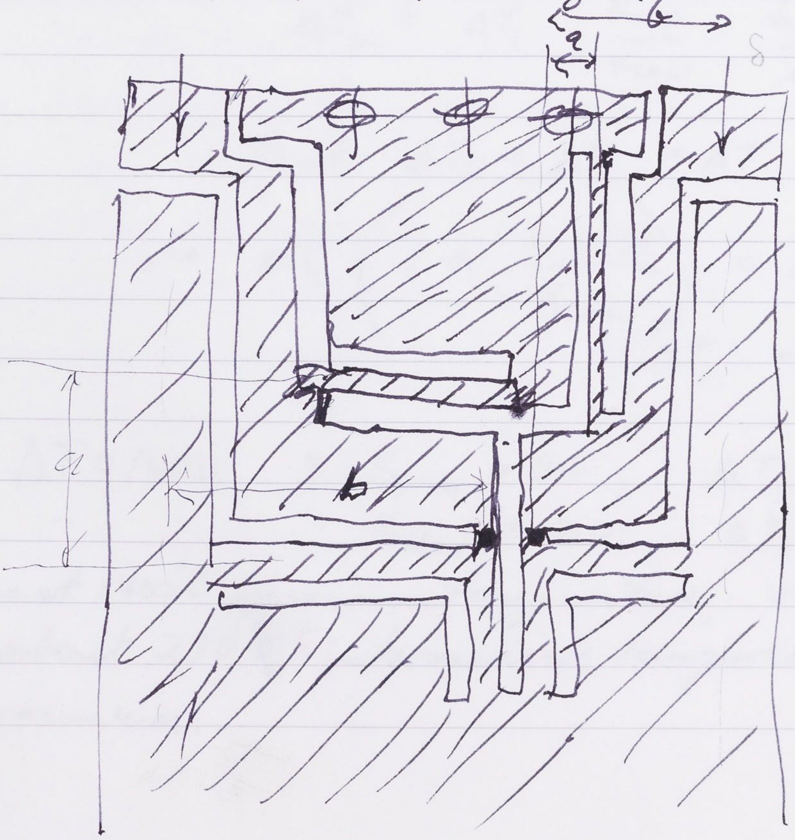
Then $a_1 = 3{\cdot}2$ $m = 2{\cdot}3_4\times$

LVDT Load Cell – Mechanical Amplification

The scheme opposite would permit mechanical amplification by a factor $m = \frac{b}{a} = \frac{b_1}{a_1}$ for both axial & torsion motion. The motions consist of rotation about the fulcrum points F_A and F_B. This will produce the most rotation of the LVDT core in the axial case, which will give a lateral movement at the bottom end of the core ~~of~~ approaching 0·5 mm. So we need a core of Φ2 in a bore of LVDT coil former of Φ3 if we have a mechanical amplification of around 2×.

With the positions in the drawing (which seem a fairly natural selection), the magnification for axial comes out to be around 2·3×, and for torsion around 1·5×. It is difficult to make the torsion figure higher without making the vertical flexure member (in direction of a) unduly short. For m=1·5, this length is about 8 mm; to get m = 2·0, we would need to shorten it to 5 mm – may be acceptable if the flexure member is thin enough.

Contd p 226



Recalculation of NZP Furnace Temperatures

Cf. p177. Those calculations were for Φ27 core and variable d for bore of NZP. We have now fixed on Φ25 core & d = 45. Also the intermediate temp. calculation is wrong due to interchange of thermal conductivities.

Now ~~q~~ $$q = \frac{2\pi l \Delta T}{\frac{\ln\frac{45}{25}}{K_{SALI}} + \frac{\ln\frac{60}{45}}{K_{NZP}}}$$

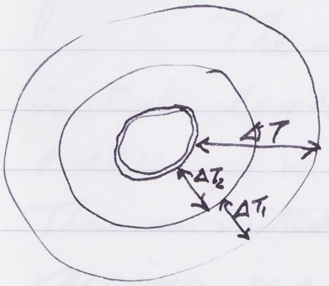
$$= \frac{879\cdot6\ \mathrm{W}}{\frac{0\cdot5878\,\text{~~[illegible]~~}}{K_{SALI}} + \frac{0\cdot2877}{K_{NZP}}}$$

for $\boxed{\Delta T = 1400\ K}$
$l = 100$ mm

With $K_{NZP} = 1$, $$q = \frac{879\cdot6}{0\cdot2877 + \frac{0\cdot5878}{K_{SALI}}} = 601\ \mathrm{W} \text{ for } K_{SALI} = 0\cdot5$$

$$= 501\ \mathrm{W} \text{ for } K_{SALI} = 0\cdot4$$ (≡ 0.36 W/K.

For intermediate temperature,



$$\frac{q}{2\pi l} = \frac{\Delta T_2 \cdot K_{SALI}}{\ln\frac{45}{25}} = \frac{\Delta T_1 \cdot K_{NZP}}{\ln\frac{60}{45}}$$

$$\therefore \Delta T_2 = \Delta T_1 \cdot \frac{K_{NZP}}{K_{SALI}} \cdot \frac{\ln\frac{45}{25}}{\ln\frac{60}{45}} = \Delta T_1 \cdot \frac{1}{K_{SALI}} \cdot \frac{0\cdot5878}{0\cdot2877}$$

But $\Delta T_2 + \Delta T_1 = \Delta T$

so $$\Delta T_1 \left(1 + \frac{2\cdot0431}{K_{SALI}}\right) = \Delta T$$

$$\Delta T_1 = \frac{\Delta T}{1 + \frac{2\cdot0431}{K_{SALI}}}$$

For $\Delta T = 1400$ & $K_{SALI} = 0\cdot5$, $\Delta T_1 = 275$ K
" " $K_{SALI} = 0\cdot4$ $\Delta T_1 = 229$ K

Thus at 1400°C specimen temperature, we can expect not more than about 270 ~~K~~K intermediate temperature above wall temp., say 300 K elevation as a maximum.

With mild steel can, $\Delta\alpha \sim 9.10^{-6}$, so expansion on 45mm at $\Delta T = 275K$ is $9.10^{-6}.\ 275.45 = 0.111\ m$ $- 0.050$mm initial clearance $\rightarrow$ 0.061mm interference or 285 MPa, within the strength of hard-drawn tubing.

Gap closure occurs for $9.10^{-6}.\Delta T.45 = 0.050$, $\Delta T = 123K$, ie $T = \frac{123}{275}.1400 = 628K$ (356°C). Below this temperature, the gap probably wouldn't matter much.

So a mild steel can from cold-drawn tubing should be OK.

Reverting to p211, we now expect a can temperature rise of ~~25~~ 275 K or even less for specimen temperature of ~1400°C

With differential thermal expansion of 15.10^{-6}, the expansion at 45 mm ϕ will be $275 . 15 . 10^{-6} . 45 = 0{\cdot}0041 . 45 = 0{\cdot}186$ mm. If the initial clearance is 0·050 mm, then the interference will be $0{\cdot}186 - 0{\cdot}050 = 0{\cdot}136$ mm, or an interference strain of 0·0030, corresponding to a stress of 633 MPa. This is more or less in the elastic range of the SS, especially since there will be some elastic take-up in the NZP not allowed for here. So there is unlikely to be any permanent set.

If the specimen temperature is ~700°C (~1000K), then the can wall temperature will be halved & so the expansion will be halved to ~0·093 mm. This is only 0·043 mm above an initial clearance of 0·050, giving an ~~interfe~~ interference strain of 0·0010 or stress of 210 MPa, still in good contact.

The minimum specimen temp. for taking up an initial clearance of 0·050 mm, ie thermal expansion strain of 0·0011, is that which gives $\Delta T_1 = 74$ K, ie $\frac{74}{275} . 1400 = 377$ K. Thus an initial clearance of 0·050 mm is taken up before the specimen reaches 400 K.

If the initial clearance were 0·150 mm (adding 0·050 tolerance both ways), the thermal strain would be 0·0033, requiring $\Delta T_1 = 222$ K, which is reached at $\frac{222}{275} . 1400 = 1131$ K specimen temperature rise ΔT, ie ~1130°C. Clearly it would be desirable to keep the initial clearance nearer to 0·050 mm & it would not matter if it were closer.

SALI $\alpha = 6 . 10^{-6}$ K^{-1} $\begin{cases} \rho = 480 \\ k = 0{\cdot}34 \, W m^{-1} K^{-1} \text{ at } 1100^\circ C \end{cases}$. good for $\pm 0{\cdot}001''$ machining

AL30 $\alpha = 5 . 10^{-6}$ K^{-1} $\rho = 480$ $k = 0{\cdot}19$ @ $1100^\circ C$

ZAL45 $\alpha = 5 . 10^{-6}$ K^{-1} no cylinders. For HIP. $\rho = 720$ $k = 0{\cdot}29$ @ $1100^\circ c$

SALI Moldable : Dried, 400 - 480 $kg \, m^{-3}$.

Shrinkage on drying : length & width 5% Thickness 18%.

Alternatives to NZP for outer insulation

If we used SALI insulation also on the outside, the expected heat loss radially (neglecting the presence of the SS can would be

$$q = \frac{2\pi l \; K_{SALI} \; \Delta T}{\ln \frac{60}{25}} = \frac{2\pi \; 0.1 \; 1400 \, . \, K_{SALI}}{\ln \frac{60}{25}} = 917 . K_{SALI}$$

$$= 459 \text{ W if } K_{SALI} = 0.5 \quad (0.33 \text{ W/K})$$

$$367 \text{ W if } K_{SALI} = 0.4 \quad (0.26 \text{ W/K})$$

Intermediate temperature rise ΔT_1 is

$$\Delta T_1 = \frac{\Delta T}{1 + \frac{\ln 45/25}{\ln 60/45}} = \frac{\Delta T}{3.043} = 460 \text{K}$$

for $\Delta T = 1400$

If we used SALI for outer insulation, its $\alpha = 6.10^{-6}$, so differential from SS is 10.10^{-6} and so the expansion ~~than~~ will be $460.10.10^{-6}.45 = 0.207$ mm. If the initial clearance is 0.050, then the interference at temperature will be 0.157 mm, a strain of 0.0035 or max stress in SS of 730 MPa. Actually quite a bit of this strain may be taken up, permanently, in the SALI.

The chief objection to SALI is that it is about 90% porous, compared with 30% porosity in the NZP. However, we have lived with this amount of gas in the past & should be able to again. The housing has been designed conservatively enough to contain any problem.

Copper :	$k \approx 400\ \mathrm{W m^{-1} K^{-1}}$	$\rho = 2 . 10^{-8}\ \Omega\mathrm{m}$	$k\rho = 800$
Brass :	$k \sim 120$	$\rho = 6 . 10^{-8}$ "	$k\rho = 720$
SS	$k \sim 16$	$\rho = 66 . 10^{-8}$ "	$k\rho = 1056$
MS	$k \sim 50$	$\rho > 17 . 10^{-8}$	$k\rho = 850$

With $\phi 3$ mildsteel, $R = 0{\cdot}0038\ \Omega$, generating $0{\cdot}87$ W for 15 A, or total wattage loss of around 4 W — no problem

Heat flow $q = \dfrac{50 . \frac{\pi}{4}(0{\cdot}003)^2\, 180}{0{\cdot}05} = \overset{1{\cdot}27}{\cancel{0{\cdot}0[illegible]}}$ W or total heat loss $= \overset{36}{\cancel{25}}$ W.

The best choice seems to be stainless steel $\phi 3$ – but see p224.

$\phi 1{\cdot}5$ mildsteel : Heat generated, around ~~[illegible]~~ 17 W

Heat flow, around 4 W.

<u>Thermal losses in furnace connections in NZP furnace.</u>

If we have 1·5 ϕ copper wires running from top to bottom, these will generate heat and will conduct heat out.

<u>Resistance</u> $= \frac{\rho l}{A}$ where ρ = resistivity, $\sim 2.10^{-8}$ Ωm, l is length & A the area. $l = 0.16$ m & $A = \frac{\pi}{4}(0.0015)^2$.

$\therefore R = \frac{2.10^{-8}.0.16}{\frac{\pi}{4}(0.0015)^2} = 0.0018\,\Omega$, so for 15 A, heat = 0.4 W

& for all wires ~40A max total, double length, heat = ~2 W.
$(0.0018.15^2.\times 3\times 2)$. This is of no great consequence.

<u>Heat flow</u> along wires $q = k.\frac{A}{l}.\Delta T = \frac{400.\frac{\pi}{4}(0.0015)^2}{0.05}\Delta T = 0.014\Delta T.$

for roughly 50 mm away from hot region in each direction.

If the wires are near the outside, ΔT will be small, say 50 K at most, ~~so~~ so loss ≃ 1.4 W per wire ≃ 8 W total.

If the wires are at 50 PCD, temp may be ~ 300 K, so total loss is then about 51 W, which is beginning to be serious relative to the radial conduction loss of ~400 W.

So it is better to keep the wires at max^m^ radius.

If we put the wires through the NZP at PCD 50, the temperature will be:

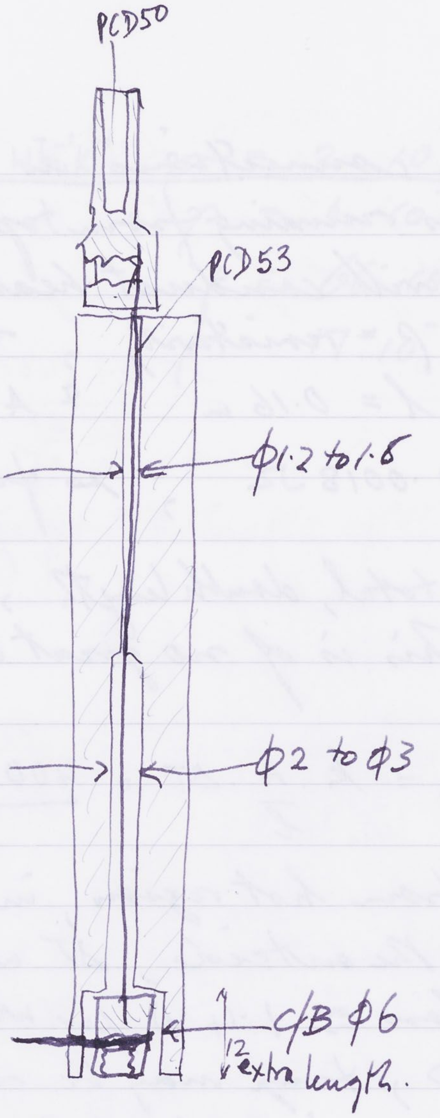
$$\Delta T_1 = \frac{275}{1 + \frac{\ln 50/45}{\ln 61/50}} = 180K.$$

<u>If we use ϕ3 SS</u>, $R = \frac{66.10^{-8}.0.16}{\frac{\pi}{4}(0.003)^2} = 0.0149\,\Omega$, generating 3·36 W for 15 A × 2 = 6·7 W so total wattage about 15 W, or an extra 3-4% power consumption.

Heat flow along ϕ3 SS $q \approx \frac{20.\frac{\pi}{4}(0.003)^2}{0.05}.180 = 0.5$ W each way, or about 12 W <u>heat loss altogether.</u>

<u>With ϕ2 brass</u>, $R = 0.003\,\Omega$, and total extra power ~ 3·4 W.

Heat flow $q \sim \frac{120.\frac{\pi}{4}(0.002)^2.180}{0.05} = 1.36$ W or total heat loss = 33 W.



At PCD 53, $\Delta T_{53} = \dfrac{\Delta T_1}{1 + \dfrac{\ln 53/45}{\ln 61/53}} = \dfrac{\Delta T_1}{2.164}$

For N2P insulation, $\Delta T_1 \sim 275\,K$, $\Delta T_{53} = 127\,K$

For SALI insulation, $\Delta T_1 = 460\,K$, $\Delta T_{53} = 213\,K$

Copper $\rho = 2.10^{-8}\,\Omega m$ $K = 400\,Wm^{-1}K^{-1}$
Mildsteel $\rho = 20.10^{-8}\,\Omega m$ $K = 50$ "
Brass $\rho = 7.10^{-8}$ $K = 120$ "
Stainless S $\rho = 70.10^{-8}$ $K = 16$ "

Further considerations.

After talking to Roger Wing of Rojan, the possibility of having holes supplied in the NZP insulation piece (using Al_2TiO_5 instead of NZP) arises, & it now seems to me that the optimum position is at PCD 53, halfway between 45 & 61, since we can then have a connector in a counterbore of $\phi 6$ at the bottom end & eliminate the PSZ parts. It is then probably best to use $\phi 1.0$ to $\phi 1.5$ copper wire if we can get a suitable hole drilled. The small bore need only go down about 60 mm to the hotzone. Below this the temp profile is flat or negative, so convection should not be a problem.

For $\phi 1$ copper:

Resistance $R = \dfrac{2.10^{-8} \cdot 0.16}{\frac{\pi}{4}(0.001)^2} = 0.0041\Omega$, generating 0.9 W at 15A

so total heat production will be around 4-5 W — OK.

Heat flow $q = \dfrac{400 \cdot \frac{\pi}{4}(0.001)^2 \Delta T}{0.05} = 0.0063 \Delta T$ ~~= 1.7 W for 275K ΔT~~
~~2.9 W " 460K~~

so total heat loss will be 0.80 W per wire X24 = 19W total loss for NZP insulation ($\Delta T_{53} = 127$ K)

and 1.34 X 24 = 32 W total loss for SALI insulation.

For $\phi 1.5$ copper:

Heat production is about 2W
Total heat loss = 43 W for NZP-type insulation
or 72 W for SALI-type insulation.

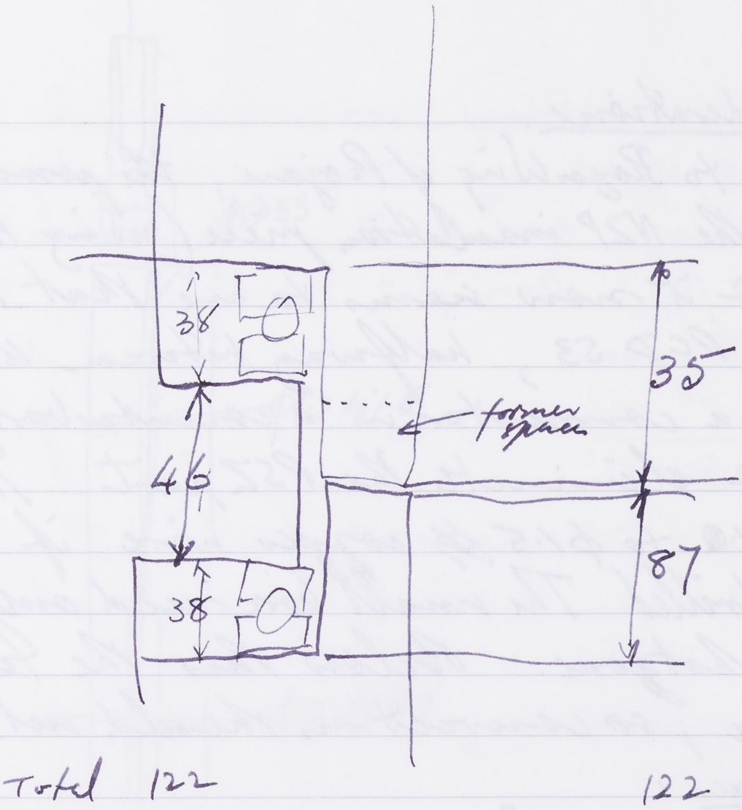
Clearly we are better off with 1.0 mm copper wire.

The two choices are still between $\phi 3$ S.S. and $\phi 1$ copper. The SS is better for heat loss (~9W vs 19W) but the copper will be more convenient.

	$\phi 15$ brass
For $\phi 1.5$ mildsteel, heat production is about 20W and	7W
heat loss around 5 W for NZP-type	12W
" 9 W for SALI-type.	22W

Since heat production figure may be rather inflated, mildsteel may be the best choice.

Contd. p234


38
46
38
35
former space
87
Total 122
122

Eliminating Spacer in Axial Actuator

In past machines, we have put in adjustable spacer in the frame of the axial actuator in order to ensure optimum pre-load of the thrust bearings, by assembling, measuring the gap and then making the spacer 0.010 bigger than the gap.

If we make the recesses 35.000 / 34.990
and 87.000 / 86.990
122.000 / 121.980
and the flange 45.900 / 49.970

assuming bearings exact, gives minimum clearance 0.000 and maximum clearance 0.030 mm. 0.030 mm is equivalent to six turns of the motor on going through hysteresis at zero load.

(from p 219)

Effects of elastic distortions in LVDT load cell

For 304 or 316 SS, E = 193 GPa, ν = 0.30 } Callister p793, 796
4140 or 4340 steel E = 207 GPa ν = 0.30 }

so linear incompressibility $= \frac{\cancel{3}}{\cancel{3}} \cdot \frac{E}{3(1-2\nu)} = \frac{E}{1-2\nu}$ = 482.5 GPa for SS
517.5 " for alloy steel

so strain at 300 MPa = 0.000622 for SS
0.000580 for alloy steel
differential = 0.000042

So if the LVDT former of length 38 is fixed at one end, the centre point will move 0.000042 × 19 = 0.0008 mm or 0.8 μm.
This will give an effect of 0.8 / 75 = 1.1% of fullscale range.
To avoid this effect, one would need to attach the LVDT former at its centre point, which complicates construction.
If it were attached 5 mm from centre, effect = $\frac{0.000042 \times 5}{75}$ = 0.3% fullscale

If we use an aluminium former (4% Cu), E = 72.4 GPa, ν = 0.33
so linear incompressibility = 213.2 GPa
strain at 300 MPa = 0.001407
so differential to alloy steel = 0.000827
Fixed at one end, centre point moves 0.0157 mm, or 15.7 μm ie 21% FS
" " 5 mm " " 5.5% "
" " 1 mm " " 1.1%.

Better to stay with stainless steel if possible and attach at the ends of the primary winding.

Contd. p235

60 mA at 5V → $Z = 83\,\Omega$, giving inductive impedance of 73 Ω. 73 → 146 if we double the frequency, leading to an overall impedance of 152 Ω instead of 83, which should have reduced the 60 mA to 33 mA. No note was made of this.

$\frac{5}{0.00388} = 1288\,\Omega$ total Z, or $\underset{\text{inductive}}{1252\,\Omega}$ when R = 304 — ie primary has 304 Ω resistance and 1252 Ω inductance

From ② on p 215, inductive impedance $= 0.0310\,\frac{N^2d^2}{l}$ for f = 5 kHz
With N = 2630 and l = 0.008 m

$$\frac{0.0310\,(2630)^2\,d^2}{0.008} = 1252$$ ∴ d = 0.0068 effective diameter of coil.

On test with 2630 primary turns, current = 3.88 mA at 5 kHz } factor
2.57 " 10 kHz } of 1.51

However, if R = 304 Ω + inductance of 1252 goes to 2504 Ω at 10 kHz, we would expect a current of 1.98 mA. Maybe there is a substantial departure from 90° phase shift between resistive + inductive parts? Ie, a big change in phase angle when frequency is changed.

Contd from p218.

LVDT Test and Further Design

A transducer was made up on the former shown opposite p216. Paul put on some windings in RSPhysS but they did not go on very evenly. There were 860 primary windings and 2 × 1720 secondary windings. Exciting from an RS card, I have a note that the current was 60mA but my recollection is that it was ~40 mA. A calculation suggests around 38 mA. The primary resistance was 40 Ω (& secondary 170 Ω) The original excitation frequency was 5 kHz but this seems to have been ~~reduced to~~ increased to 10 kHz, which should have reduced the current to 33 mA. This transducer gave a somewhat irregular V versus displacement curve, which we attributed to unevenness of winding. Various lengths of core were tested & it appeared that the optimum was about 19 mm length, ie approx from centre to centre of the secondary windings.

with or without core?

!

maybe in series.

A new set of windings were now put on by Star Delta (courtesy Neville McElroy) as follows:

Primary 2630 turns of 0·063 wire, R = 304 Ω

Secondaries 3750 " each of 0·06 wire, R = 517 and 509 Ω.

without core

On connection to the RS card at 5 kHz, current = 3·88 mA. This set-up gave a reasonably linear (better than 2%, maybe 1%) behaviour, over ± 0·2 mm travel. So linearity is OK, & it looks OK up to ± 1·5 mm. The output was ± 10 V for ± 0·2 mm travel, with the RS card on the 1 to 3 mV/V range — somewhere in the middle. We need ± 10 V for ± 0·075 mm travel, that is, 2·7 times the sensitivity — probably just attainable on the max. gain setting.

according to ③ p217

So it is desirable to increase the sensitivity. Increasing the frequency does not achieve anything because the effect is countered by the decrease in current for given windings. But it would enable us to use fewer windings on the primary ~~and so can~~ within the same current limits, and so increase the transformer ratio.

In fact there may be something to be gained from increasing the frequency.

1300 turns 6·5

~ 5·1 mm

2·52

6·48

goes as $\sqrt{f}$

Measurements by Paul show that increasing f from 5kHz to 10kHz decreases the current from 3·88 mA to 2·57 mA, ie it does not halve the current as would be expected on a simple formula

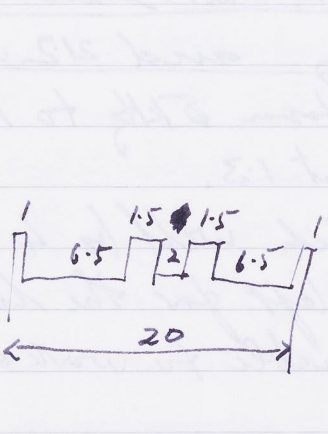
In further design changes, it appears simplest to keep the length of primary the same and reduce the number of turns. With the 2630 turns on the test unit, the o.d. of windings was 9·0 mm the i.d. 4·0, ie winding thickness was 5/2 mm. Effective diameter for determining inductance is probably something like $(9^2 - 4^2)^{1/2}$ = 8·1 mm. A preliminary calculation suggests that with around 1500 turns the od will be around 6·8 and the effective diameter around 5·5 mm. Reducing the effective diameter from 8·1 to 5·5 will reduce the inductance by a factor $(8{\cdot}1/5{\cdot}5)^2$ = 2·17, so increasing the current from 2·57 to 5·57 mA.

<u>The RDP position transducer takes 27·4 mA excitation at 5kHz without a core and 8 mA with a core, and 21·2 without, 6mA with core at 10 kHz</u>, so going from 5kHz to 10 kHz reduces the current by a factor of about 1·3.
If we allow for the 8 mA at 5 kHz, which will be 4 mA for two in series, then we have 46 mA left for the load cell LVDT's, ie 23 mA each. It seems reasonable to work on 20mA

The 2·57 mA was measured without core, so it may have been not more than 2 mA with core at 10 kHz or 3 mA at 5 kHz. So reducing the effective diameter as above would increase the current to, say, 4·3 mA at 10 kHz or 6·5 mA at 5 kHz. To get the current up to 20 mA we need to reduce the number of turns by $(20/4{\cdot}3)^{\frac{1}{2}}$ = 2·16x at 10 kHz or 1·75x at 5 kHz, leading to 1220 or 1500 turns respectively instead of 2630.

5.58 for 2.57

It would seem a safe bet to <u>choose 1300 turns</u>, ie half as many as the 2630, which may be unstable at 5kHz but should be OK at 10 kHz if not.


1
1·5
1·5
1
6·5
2
6·5
20

Since the sensitivity of the LVDT is proportional to $f I_p N_p N_s$, decreasing the primary windings to 1300 will increase the sensitivity by a factor $\frac{10 . 20 . 1300}{5 . \sim 3 . 2630} \simeq 6{\cdot}6$, depending on the exact figure for the current drawn at 5kHz for the 2630 winding with core in (the figure with core out is 3·88mA at 5 kHz but may be lower than 3 with core in if the observations on the RDP are any guide; they would predict 1·1mA).

The factor ~6·6 is doubled when two transducers are put in series, so we gain at least an order of magnitude compared to the Star Delta prototype measurement.

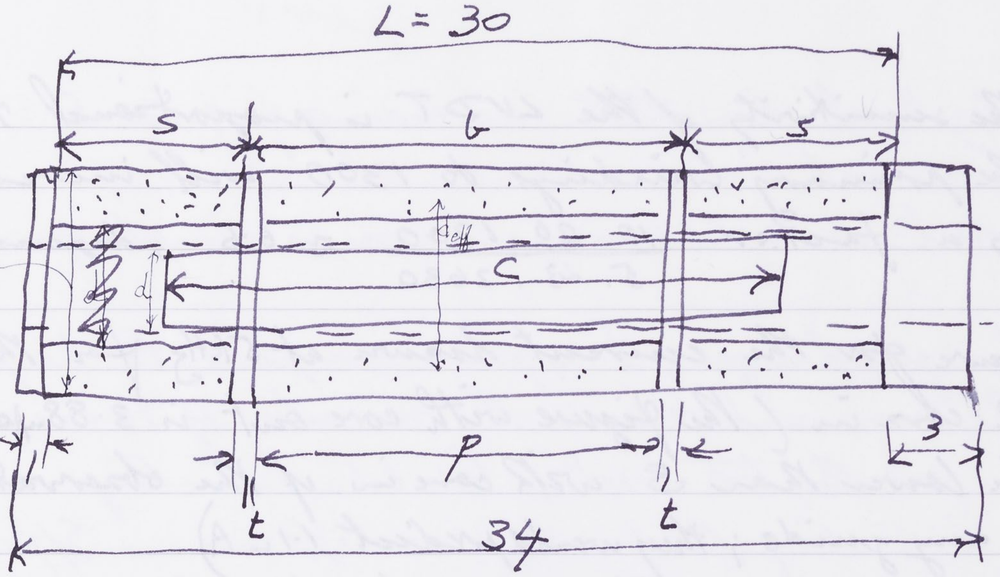
For torsion load in Bayreuth, the total length is 20.

If we reduce the 8 mm length of primary to 2mm, the 2 mA assumed for 8mm length goes up to 8mm. Reducing the effective diameter of windings gives another factor of around 1·5, to give 12mA. If we allow 20mA, then the number of turns is reduced by $(20/12)^{1/2}$ ~~by~~ to $\frac{2630}{4} \times \left(\frac{12}{20}\right)^{\frac{1}{2}} = 509$. Thus we can opt for <u>500 primary turns</u>.

There will be space for 6·5 mm of secondary windings instead of 11mm, ~~or relative to~~ ie 2216 windings. So relative to the Star Delta prototype, the gain in sensitivity (using 10 kHz) will be

$$\frac{10 . 20 . 500 . 2216}{5 . \sim 3 . 2630 . 3750} = 1{\cdot}50$$

or a factor of 3 when two transducers are used. This should be just workable, with perhaps a factor of 2 to spare, assuming that the wall thickness is 2 mm. If it is 1mm, there is plenty.



μ_0 in $\frac{\Omega s}{m}$

LVDT Design Revisited

Adapting the formulae from p 214 to the nomenclature opposite, we have the basic formulae:

① $\Delta V = K_1 \delta \left(1 - \frac{\delta^2}{K_2}\right)$ where δ = displacement

② $K_1 = \frac{16\pi^{\cancel{2}3} f I_p N_p N_s}{10^9 \ln\frac{D}{d}} \quad \frac{b\left(\frac{c-b}{2}\right) + \left(\frac{c-b}{2}\right)^2}{cs}$

f = freq ; I_p = primary current
N_p = no. of primary turns
N_s = " secondary "

③ $K_2 = b\left(\frac{c-b}{2}\right) + \left(\frac{c-b}{2}\right)^2 = \frac{c^2-b^2}{4}$

Design steps: 1) given constraint of over-all length, determine the basic dimensions to give required non-linearity

2) given the basic dimensions, determine the number of windings for max. sensitivity

Basic dimensions.

We take the diameters D, d as given and $b + 2s = \overset{30}{\cancel{28}}$ mm.

For 0·5% non-linearity at full displacement $\delta = \Delta$, we have

$\frac{\Delta^2}{K_2} = 0{\cdot}005$, ie $K_2 = \frac{\Delta^2}{0{\cdot}005} = \overset{50}{\cancel{800}}$ for $\Delta = 0{\cdot}5$ mm

$\therefore c^2 - b^2 = \cancel{50}\,200$

For symmetry we put $c = b + s$ $\quad\therefore b + 2s = b + 2c - 2b = 2c - b = 30$

or $b^2 = (2c-30)^2$

$\therefore c^2 - (2c-30)^2 = \overset{200}{\cancel{50}}$ or $c = \overset{25{\cdot}8}{\cancel{22{\cdot}1}}$ & $b = \overset{21{\cdot}5}{\cancel{28}}$, $s = 4{\cdot}3$

~~This is too narrow for convenience~~.

If we take s = 4, then b = 22 and c = 26 for symmetry

~~so $c^2 - b^2 = 192$ & $\frac{\delta^2}{192} = 0{\cdot}0013$ or 0·13% linearity. This is better than necessary.~~

Primary windings. We are limited to 24 mA if we assume 4 mA to the position LVDT's in series (p 228) and assume four LVDT's are connected to 2RS cards giving 50 mA each. Better to assume 20 mA for a margin.

Inductance $L = \frac{\mu_0 N_p^2 \cdot \cancel{3}\pi d_{eff}^2}{(b-2t)4}$ so ~~my~~ inductive impedance is $2\pi f L = \frac{\cancel{2}\pi^2 \mu_0 N_p^2 d_{eff}^2 f}{\cancel{4}(b-2t)}$. If we assume that the resistive part of the total impedance X_p is negligible, then

$$④ \quad I_p = \frac{V}{X_p} = \frac{2(b-2t)V}{\pi^2 \mu_0 N_p^2 d_{eff}^2 f}$$

For no core $\mu_0 = 4\pi 10^{-7}$ (p 215). With an iron core, μ_0 is about 4000 times greater, but the check on the RDP position LVDT (p 228) suggests that the effective μ_0 is only $\frac{27.4}{8} = 3.4$ times greater than without core, ie $\mu_{0eff} \approx 43.10^{-7}$. With $I_p = 0.02A$ and $f =$ ~~10~~ 5 kHz.

$$N_p^2 = \frac{2 \cdot \overset{b-2t}{0.020} \cdot 5}{\pi^2 . 43.10^{-7} . (0.005)^2 . 5.10^3 . 0.020}$$

$$N_p = 1373$$

Previous ~~experience~~ suggests that around 600 turns would be better (p 227)

From ② on p 230 and ④ above, we have sensitivity

$$K_1 = \frac{32 V \pi}{10^9 \mu_0 \ln\frac{D}{d} . d_{eff}^3} \cdot \frac{b-2t}{N_p} \cdot \frac{N_s}{s} \cdot \frac{c^2-b^2}{4c}$$

1) The $\frac{b-2t}{N_p}$ term is fixed by the current requirement.

2). $\frac{N_s}{s}$ is fixed by the density of windings on the secondary, and so is maximized by using the smallest possible wire.

3). $\frac{c^2-b^2}{c}$ is maximized by making c & b as large as possible, ie keeping s as small as possible.

~~With $b = c-s$, $\frac{c^2-b^2}{c} = 2s - \frac{s^2}{c} = s(2 - \frac{s}{c})$ which is maximized for given s by making c as large as possible~~

Putting $c = 30 - s$ and $b = 30 - 2s$,

$$\frac{c^2-b^2}{c} = \frac{(30-s)^2 - (30-2s)^2}{30-s} = \frac{60s - 3s^2}{30-s} = X$$

This is a max/min at $\frac{dX}{ds} = 0$, ie $\frac{60-6s}{30-s} - \frac{60s-3s^2}{(30-s)^2} = \frac{1800 - 300s + 9s^2}{(30-s)^2} = 0$

or $S = \frac{150}{9} \pm \sqrt{\left(\frac{150}{9}\right)^2 - 200} = 25.5$ or 7.85

$$\frac{d^2X}{ds^2} = -10{,}800 + 1140s - 27s^2 \quad \begin{matrix} < 0 \text{ for } s = 7.85 \therefore \text{max}^m \\ > 0 \text{ for } s = 25.5 \therefore \text{min}^m \end{matrix}$$

N_p and I_p are not independent variables for a given V.

When N_p is fixed & I_p is varied, $K \propto I_p$.

When I_p is fixed & N_p is varied, $K \propto \frac{1}{N_p}$

so the term $\frac{c^2-b^2}{4c}$ is a max^m when $s=8$, $b=14$, $c=22$; then $b-2t = 12$ for length of primary winding. Going back to relation ④ for the primary current, we now get

$$N_p^{\,2} = \frac{2 \cdot 0{\cdot}012 \cdot 5}{\pi^2 \; \underset{\uparrow ?}{43.10^{-7}} (0{\cdot}005)^2 \cdot 0{\cdot}020 \cdot 5.10^3}, \text{ or } N_p = 1063$$

instead of 1373, ie less by factor of 0·775

<u>Lets start again</u>.

From ② on p 230, the sensitivity

$$K_x = \frac{16\pi^{\overset{3}{2}}}{10^9 \ln\frac{D}{d}} \cdot f I_p N_p \cdot \frac{N_s}{s} \cdot \frac{c^2-b^2}{4c} \qquad \text{checked} \qquad b = p+2t$$

But $I_p = \frac{2V(b-2t)}{\pi^2 \mu_0 N_p^{\,2} d_{eff}^{\,2} f}$ $\rightarrow$ ~~$f I_p N_p = \frac{2V}{\pi^2 \mu_0 d_{eff}^2} \cdot \frac{b-2t}{N_p}$~~ is a fixed quantity, set by the RS card.

$$\therefore N_p = \sqrt{\frac{2V(b-2t)}{\pi^2 \mu_0 I_p d_{eff}^{\,2} f}}$$

$$\text{and } f I_p N_p = \sqrt{\frac{2V\overset{f}{(I_p f)}^{3}}{\pi^2 \mu_0 d_{eff}^{\,2}}} \cdot \sqrt{b-2t} \quad \text{for the second term in } K_2$$

We can then write K_x as

$$K_x = \frac{16\pi}{10^9 \ln\frac{D}{d}} \sqrt{\frac{2V\overset{f}{(I_p f)}^{3}}{\mu_0 d_{eff}^{\,2}}} \cdot \frac{N_s}{s} \cdot \frac{\sqrt{b-2t}\,(c^2-b^2)}{4c} \quad\text{———} ⑤$$

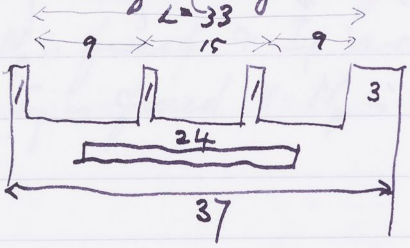
Of the quantities in the <u>first term</u>, we want to make d (the diameter of the core) as big as possible, d_{eff} (the effective diameter of the primary coil) as small as possible, and the frequency f as big as possible. The role of μ_0 is problematic — it would seem to require μ_0 to be as small as possible, ~~or~~ which for given permeability of iron would mean making d as small as possible, so these factors play off against each other.

The <u>second term</u> is the ~~reciprocal of~~ number of turns per unit length of secondary winding, so it depends only on the diameter of wire when other dimensions are fixed — we want as fine wire as possible on the secondary.

	F
L=33 S=6	11·6
7	12·5
8	13·0
9	13·2
10	13·0
11	12·4
12	11·3
13	9·8
14	7·7

For $L = 33$, $s = 11$ we get $F = 12{\cdot}38$ or 9% better than opposite.
(optimum is $s = 9$, $F = 13{\cdot}2$) 17% better

Earlier design (Bayreuth test) has $t = 1{\cdot}5$, $b = 11$, $s = 11$, $L = 33$, $F = 11{\cdot}7$



For $L = 72$, $s = 20$, get max $F = 44{\cdot}2$ $b = 32$ $c = 52$ $K_2 = 420$
$s = 30$ gives $F = 30{\cdot}5$ OK $b = 12$ $c = 42$ $K_2 = 405$

but $\dfrac{4\delta^2}{c^2 - b^2} = 0{\cdot}005$ gives $\delta = 1{\cdot}4$ mm for 0·5% linearity with $s = 30$
— doesn't make much sense?

assumption

The third term determines the geometric proportions. We have $b = L - 2s$ and $c = L - s$, and we can assume $t = 1$, so this term becomes $F = \frac{\left((L-s)^2 - (L-2s)^2\right)\sqrt{L-2s-2}}{4(L-s)} = \frac{\left(2Ls - 3s^2\right)\sqrt{L-2s-2}}{4(L-s)}$

For $L = 30$, and $s = 2$, $F = 4.72$
4 8.26
6 10.50
8 11.34 → max value so $s = 8$, $b = 14$, $c = 22$.
10 10.61
12 8.00

→ $s = 9$
$b = 15$
$c = 24$
optimum for $L = 33$

For $L = 16$, $s = 2$, $F = 2.94$
$s = 3$, 3.75
4 4.08 → max value, so $s = 4$, $b = 8$, $c = 12$
5 3.86
6 2.97

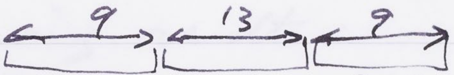
Note that the $L = 30$ case is 2.8 times more sensitive than $L = 16$, for the same I_p and N_s/s.

Linearity check:

For the $L = 30$, $b = 14$, $c = 22$ case, $\frac{\delta^2}{K_2} = 0.005$ gives $\delta = \sqrt{0.005 \frac{c^2 - b^2}{4}}$
$= 0.6$ mm

For $L = 16$, $b = 8$, $c = 12$, $\delta = \sqrt{0.005 \frac{12^2 - 8^2}{4}} = 0.3$ mm as linear ranges within 0.5%.

9/8/01



ie $L = 33$, $s = 9$, $b = 15$

Current Test on New Former

Wound with 1300 turns on primary, 1348 on secondary (~0.1 mm wire?).
Primary current = 11 mA with core in.
So for 1000 turns, we expect about 18.6 mA, maybe nearer 20 mA when smaller effective diameter of coil is taken into account.

Contd p235

maybe due to SS segment relaxing →

Melting probably occurred at this point →

(Contd from p 224)

Apr 2001

Furnace Problems – Melting Windings.

Three NZP-insulated furnaces (nos. 27, 31 and 32) have come back from Bayreuth with failures by melting of the Mo winding near the top of the bottom winding. Also furnace 24, PSZ-insulated, has failed by melting in the top winding. These failures are discussed in an email of 24/4/01 to Mackwell but further ~~to~~ thought suggests a correlation with putting the windings on more loosely than formerly, a practice adopted because of failures associated with permanent expansion of the Vesuvius cores.

The thermal conductivity of argon at 300 MPa and high temperature (Hamann p90 and Kay & Laby) is probably close to 0·3 W/mK. So ~~putts~~ burying the wires in alumina cement (order of 3 – 4 W/mK) should help in getting the heat out better & reducing the wire temperature. Just winding the wires tight again may be enough but we are probably risking melting still if higher temperatures are used and furnaces lose efficiency with age.

So in refurbishing no 27, we propose winding the wires tightly again & filling the grooves with alumina cement.

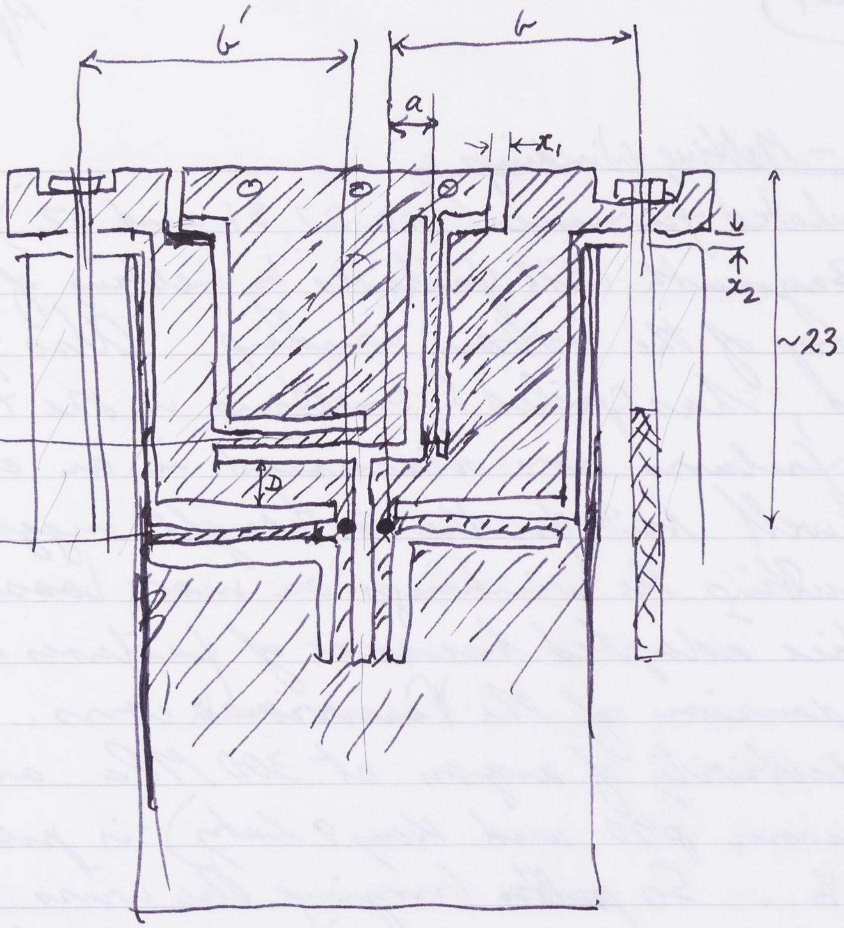
June 2001

Furnace 27, re-numbered 33, was assembled with alumina cement over the windings, as above, & sent to Bayreuth. It was run three times in calibration runs (details are in furnace reports folder) with the following results:

Run 1. From 800 to 1300K there was a uniform progression with power consumption 0·44 to 0·53 W/K, bottom current 11 to 13·5A and resistance 1·79 to 2·15 Ω. Then at 1400 K, & approx the same power consumption, the bottom current dropped to 11·5 A & resistance jumped to 3·35 Ω, with corresp jump up in V. 1500K was similar.

Run 2. Now at 1300K, the power consumption is the same as before but the current is 11 A instead of 13·5 and the resistance is 3·32 instead of 2·15 Ω. On going up to 1400K the

Contd p236



$\text{Mech gain} = \dfrac{b}{a}$

$b \simeq 14{\cdot}5$

In final design $a = a' = 3{\cdot}5$
$b = b' = 15{\cdot}05$

$\therefore$ Mech. gain $= \dfrac{15{\cdot}05}{3{\cdot}5} = 4{\cdot}3$

From p 184, max displacement on elastic element $\approx 0{\cdot}075$ mm

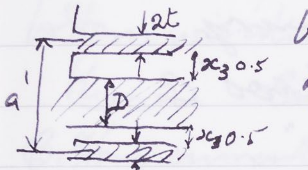
Therefore range for LVDT $= \pm 4{\cdot}3 \times 0{\cdot}075$
$= \pm 0{\cdot}32_5$ mm.

or linear range of LVDT $= 0{\cdot}65$ mm.

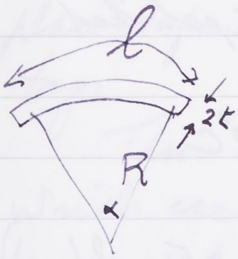
From p ~~226~~ 233

LVDT Load Cell – Critical Dimensions

After many preliminary trials, it seems that about 4× is the maximum mechanical gain that is feasible. It is set mainly by the dimension a'. The dimension D needs to be not too small so that it does not bend significantly under the torque needed to rotate the flexure pivots.



Flexure pivots undergo a double bend, so half their length is the length of arc of the bend. The angle of bend α is given by $\alpha = \frac{\delta}{b}$ where δ = deflection $= 0.081 \cdot \frac{b}{a}$ ≈ 0.32 mm.



$$\approx \frac{0.32}{14.5} = 0.022 \quad (\sim 1.25°)$$

The gap x_1 has to be such that $\frac{x_1}{25} > 0.022$ ie $x_1 > 0.55$ mm.
So we can choose $x_1 = 0.8$. x_2 can also be 0.8 mm.
The gaps x_3 need to be $> 0.022 \cdot \sim 10 = 0.22$ mm, say 0.5 mm is plenty
If thickness of flexure pivot is 2t, strain $= \frac{t}{R} = \frac{\sigma}{E}$
If $\frac{l}{R} = \alpha = 0.022$, then $\frac{1}{R} = \frac{0.022}{4}$ if flexure pivots are 8 long.

$$\therefore t = \frac{\sigma}{E} \Big/ \frac{1}{R} = \frac{\sigma}{E} \cdot \frac{4}{0.022} = \frac{\overset{468}{500}}{210\,000} \cdot \frac{4}{0.022} = \overset{0.50}{0.43} \text{ mm for}$$

a peak stress of 500 MPa. So 0.4 mm would be a maximum and maybe 0.3 mm would be optimum if the wire cutting allows.
If we want the section D to be at least two orders of magnitude stiffer than the flexure blades, D has to be $(100)^{1/3}$ times the thickness of the blades, ie 4.6×, so if blade is 0.4 mm, D ~ 1.86, say 2 mm.
With blades 0.4, gaps 0.5 & D = 2, we have $a' = 3.4$ mm, so gain $= 14.5/3.4 = 4.26$. With gaps 0.4, D = 2, gain $= \frac{14.5}{3.2} = 4.5_3$

All this presumes that the screw holes connecting to the top of the elastic element are precisely located. If they are mis-located by 0.08 mm, this could double the deflections to be allowed for. Thus gaps x_1 & x_2 probably should be at least 1 mm, & gaps x_3 kept at 0.5.

Contd. p239

Emissivity Values from Marks p382 : Wavelength unspecified

Brass	250 - 375°C	0·033 — 0·037	highly polished
..	22°C	0·06	natural, rolled
..	..	0·20	rubbed with emery
..	200 - 600°C	0·61 - 0·59	oxidized at 600°C
Mo	725 - 2600°C	0·096 → 0·292	"filament"
Ni	20 - 370°C	0·045 - 0·087	pure, polished
..	20°C	0·11	electroplated, not polished
..	200 - 1000°C	0·096 - 0·186	wire
..	200 - 600°C	0·37 - 0·48	oxidized at 600°C

Values from Kaye & Laby p50 at 0·65 μm (values at 1 & 5 μm sl. lower)

Ni	1100 K	0·45	
"	1300 K	0·88	oxidized
Nichrome	1200 K	0·4	
"	"	0·9	oxidized
Mo	2000 K	0·4	(0·3 at 1 μm ; 0·15 at 5 μm)

Contd. from p234

Melting Furnace Windings - contd

resistance rose to 4·1 Ω. ~~at which point~~ On trying to go to 1500 K, the hot spot suddenly moved upwards (top T = 1554 K) & the bottom winding temperature dropped by ~ 300 K.

Run 3. Could only get to 1010 K with OP = 100 in Eurotherm. Bottom current was 14·5 A but V = 24 & R = 1·66 Ω.

It turned out that the bottom winding had melted at a place a few turns down from the top, & also on the "tie wire" at the top.

Thus we have a persistent problem with melting of the windings, which seems to date mainly from when a new batch of Mo wire began to be used. This wire was electropolished & is shiny, in contrast to the previous batch, of dull appearance. I therefore now have the impression that the wires are getting very hot due to the radiation of heat from them being the limiting factor in heat transfer (making the use of alumina cement less useful). Figures on emissivity (opposite) indicate that this may possibly be roughly halved by polishing, corresponding to a 0·84 times output of radiant heat by Stefan-Boltzmann law, ie a 1·19 times increase in temperature would be required to transfer the same amount of heat, e.g. raising T from 2200°C to 2620°C (melting point). So the polishing of the wire may have caused several hundred K increase in wire temperature.

Resistance of winding is ~ 0·3 Ω at room temperature (which corresponds well to the resistivity of Mo). The ~~resistivity~~ resistance increase to 4 Ω corresponds to a rise in mean temperature to around 2200°C?, so rising above this resistance would take the upper part of the winding easily to the m.pt.

Realization that the ~~to~~ radiative transfer of heat from the wires ~~is~~ maybe the limiting factor also calls for a ~~new~~ review of the wire diameter, since increasing the surface area will increase the heat transfer. We are limited by the maximum current that can be drawn from the system, so increasing the diameter also tends to need a longer winding to ~~too~~ avoid too low a resistance.

Increasing 0.5 to $\overset{0.635}{0.6}$ mm diam changes R by $\overset{0.62}{0.69}$, thus increasing the current by $\overset{1.27}{1.20}$, eg. $\overset{13}{12A} \rightarrow \overset{15.7\ \ 15.2}{14.4A}$. $\left(I \propto \frac{1}{\sqrt{R}}\right)$

Increasing 0.5 to 0.7 mm decrease R by 0.51, increasing I by 1.40, eg. $\overset{13}{12A} \rightarrow \overset{18.2}{16.8A}$

To get back to 14 A in this case we need to increase the length of wire 16.8/14 = $\overset{1.4}{1.20}$ times, so the pitch of the winding has to be reduced by 1/1.20, ie 1.5 → 1.25mm or 1.25 → $\overset{0.9}{1.04}$ mm.

The increases in current with going to thicker wire will probably be greater than above due to the temperature, & hence resistivity, of the wire decreasing also. So increase to 0.6 mm or 0.635mm may be as far as we can go unless the basic current (12A) assumed above also comes down with greater furnace efficiency (thus 9.5A → 11.4 with 0.6 wire & → 13.3 with 0.7). Maybe 0.7mm is worth a try, but the 0.635mm (0.025") listed by Ed Fagan in USA sounds optimum.

Thermal Stability of Heating Wire

The local melting of the Mo wire may be an instability effect associated with the positive coefficient of resistivity, since a local increase in temperature will lead to a positive feed-back effect due to increase in resistance & hence heat production. Consideration of this (Thermal Stability in Heating Wires - ~~to~~ in computer) leads to the stability condition:

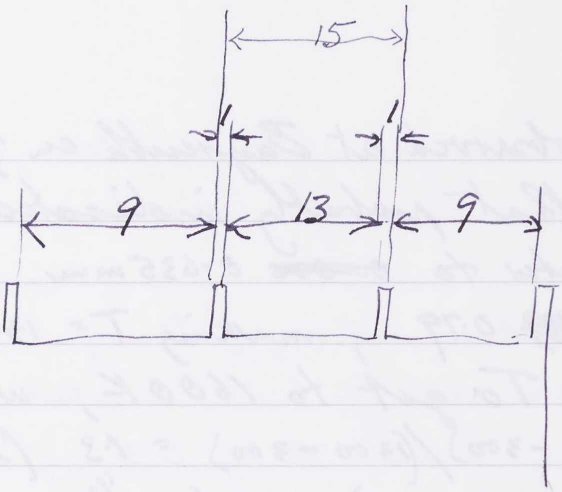
$$I^2 < \frac{\pi^2 d^3 \sigma e (T-T_e)^3}{\partial\rho/\partial T}$$

Where T = wire temp, T_e = environment temp, e = emissivity, σ = Stefan-Boltzmann constant ($5.670 \cdot 10^{-8}\ \text{W m}^{-2}\text{K}^{-4}$), d = wire diam. and $\partial\rho/\partial T$ = coeffic of resistivity ($0.027 \cdot 10^{-8}\ \Omega\text{m K}^{-1}$)

If we arbitrarily take e = 0.17, then for d = 0.5mm, T = 2900K (mp^t of Mo^x) and T_e = 1300K (spec. temperature), we get $I = 13\cdot4$ A, as observed at Bayreuth on furnace 33 for the sudden change in I that probably indicated melting.

If we increase diameter to ~~[illegible]~~ 0·635 mm, T-T_e is decreased by factor $\left(\frac{0\cdot5}{0\cdot635}\right)^{\cancel{3}}$ = ~~[illegible]~~ 0·79, making T = 1300 + 1260 = 2560 K, well under the m.pt. To get to 1600 K, we have to increase I^2 ~~[illegible]~~ by ~~1600/1300~~ (600 - 300)/(1300 - 300) = 1·3 (neglecting increase in resistivity etc), ie d(T-T_e) by $(1\cdot3)^{1/3}$ = 1·09, so T-T_e increases ~~from~~ 1260 to 1375, & T to 1375+1600 = 2975, again just over the m.pt. Actually the melting in furnace 33 occurred somewhere between 1300 & 1400, so may there is a bit of margin. So it would appear that increasing the wire diameter to 0·635 mm may get one up to 1600 K but just marginally. Less polished wire with higher emissivity may help. Thus if e is doubled, T will be decreased ~~[illegible]~~ by 0·84, ~~so~~ e.g. 2975 → 2500 K, giving 400 K headroom, in addition to the nearly 300 K gained by increasing the wire diameter.

Contd p 240


15
1
1
9
13
9

contd. from p 235

LVDT Tests for Core Length & Linearity

Core length/mm	Output on 10/30 mV/V range/V	Primary current/mA
12	8.0	23.7
14	9.8	20.7
→ 16	10.5	19.1
20	6.7	18.3
24	~0.9	18.8

run with RS card at 10kHz osc freq and 1kΩ demod. input impedⁿ

So the optimum core length is about 16mm, or about 3mm longer than the primary winding. The earlier theory seemed to be based on the assumption that the flux connection was directly from the end of the core to the sheath & that the optimum core length was from midpoint of secondary to midpoint of secondary ie 24mm, with the present dimensions. However, it seems that there is important flux flow further out from this & that by the time 24mm core length is reached, this is being lost outside the LVDT. So our calculation of optimum primary : secondary lengths is in doubt. However, we seem to get good performance as it is, so that is OK. At 16 mm core length, the primary current of 19 mA is OK — 19 + 19 + 4 = 42 mA, within the 50mA rating of the RS card.

On 16mm core length & 10.5 V/mm, we get (p 235) 3.4 V per 0.32 mm
ie we need 10/3.4 = 2.9_3 times more gain for one LVDT
or 1.47 " more gain for two LVDT's in series
ie instead of 10/30 mV/V range, we need 7/20 mV/V range. So depending on where in the range the above measurements were made, we expect to use 10/30 or 5/15 mV/V ranges. This still leaves a factor of 10 available for greater sensitivity!

contd.

Contd from p.238 5/8/02

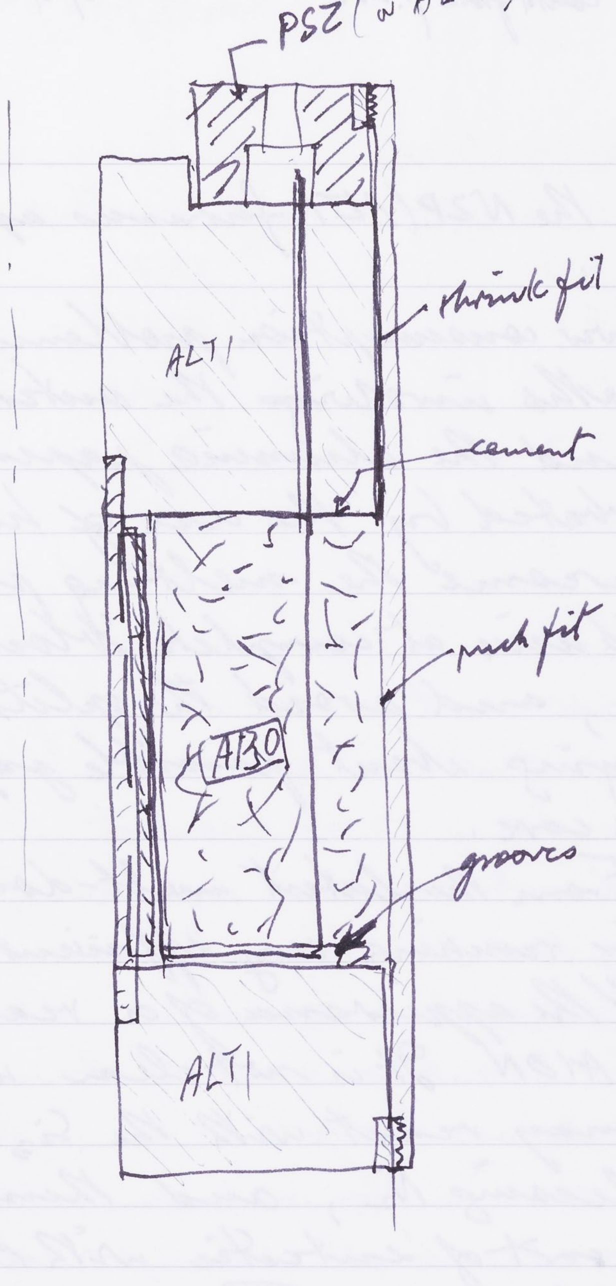
Furnace Re-Design

Two persistent problems with the NZP/ALTI furnaces appear to be approaching a resolution:

1) Inefficiency. The power consumption problems seem to be related to convection paths involving the outer surface of the SALI fibre insulation and the alumina paper in splits in the SALI, exacerbated by the use of heavier Mo wire introduced to overcome the melting problem. It now seems that one should aim at complete ~~b~~ blockage of space at the OD of the SALI, and avoid the splits with alumina paper, not worrying about possible gaps between the ID of the SALI and the core.

2) Melting of winding: From the latest melt-down in Minneapolis of no. 35, after running very efficiently (~ 0·35-0·4 W/K), has all the appearance of a reaction between the Mo and the SiAlON. It is not clear what the reaction is but the Mo may react with the Si_3N_4 component of the SiAlON, releasing N_2, and then the Mo silicide may have some sort of eutectic with Al_2O_3 — speculation, but something happens. The previous correlation with polished Mo may be spurious because there is also a rough correlation with the change from Al_2O_3 to SiAlON cores (the first SiAlON cores were bought in Oct 97 & May 98; the first NZP furnace was run in Dec 98 but with Al_2O_3 core; I don't have notes on the early SiAlON furnaces but we had problems with expansion & breaking the windings, which went on before we met the melting problems around May 2000. ~~As~~ The polished Mo wire was bought in Nov 99).

So we have to abandon SiAlON cores & go back to alumina. At the same time, the convection might be sufficiently controlled to make cracking of the core less serious. The insulation arrangements can probably also be simplified, by omitting


PSZ (or ALTI?)
shrink fit
ALTI
cement
push fit
AlBO
grooves
ALTI

the outer ALTI sleeve over the SALI & taking the leads up through a full ~~[illegible]~~ diameter ALTI top section, as shown opposite. To bring the winding tails down, it may be better to use an outer Al_2O_3 sleeve rather than alumina tubes. Stick to 25OD for the main core but put a 28OD 25ID sleeve over it, in sections, as earlier.

Radial heat loss (p 220) is

$$q = \frac{2\pi l \Delta T}{\dfrac{\ln\frac{28}{25}}{K_{Al_2O_3}} + \dfrac{\ln\frac{60}{28}}{K_{SALI}}}$$

Using $K_{SALI} = 0{\cdot}4$ + $K_{Al_2O_3} = 5$ at 1200°C(+), $\Delta T = 1400$ $l = 100$ mm, we have

$$q = \frac{2\pi . 0{\cdot}1 . 1400}{\dfrac{\ln 28/25}{5} + \dfrac{\ln\frac{60}{28}}{0{\cdot}4}} = 456 \text{ W}$$

compared with 500 W calculated with ALTI outer sleeve (p 220). If we use AL30 with possibly $K = 0{\cdot}3$, q would be 343 W. AL30 is listed as having lower thermal conductivity than SALI, & its shrinkage is still only 1% at 1500°C (SALI: 1% at 1650°C), which should be acceptable.

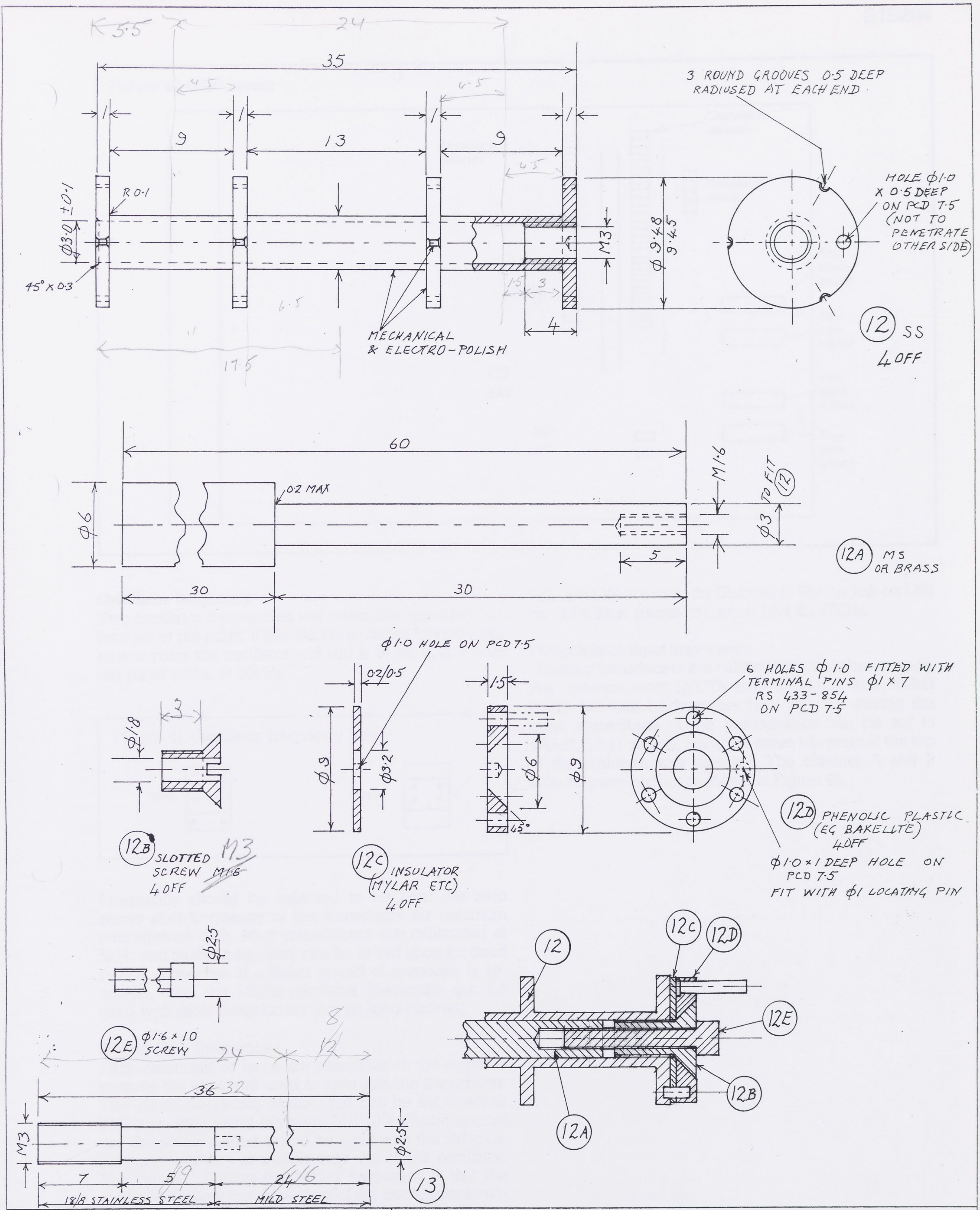
<u>Shrink fit</u>: α for stainless steel $= 16 \times 10^{-6}$

Thus expansion on 60ϕ at 200°C ΔT $\cdot$ $16.10^{-6}.200.60 = 0{\cdot}19$ mm.

Min ϕ of SS sleeve $= 61{\cdot}00$, so max dia of ALTI $= 61{\cdot}00 + 0{\cdot}19$
61·00/61·05 $= 61{\cdot}19$

If we make the ALTI $\phi = 61{\cdot}15 / 61{\cdot}10$ then interference is 0·05/0·15 or 050 to 150 µm. Light ~~press~~ fit is -028/021, & press fit = -002/051 (ferrous) or 023/072 (nonferrous).

227	LVDT test & further design
230	LVDT design revisited
234	Furnace problems – melting windings
235	LVDT load cell – critical dimensions
236	Melting furnace windings – contd
239	LVDT linearity etc tests
240	Furnace re-design.



DIMENSIONS IN MILLIMETRES
DO NOT SCALE

ꞁOLERANCES ± UNLESS STATED

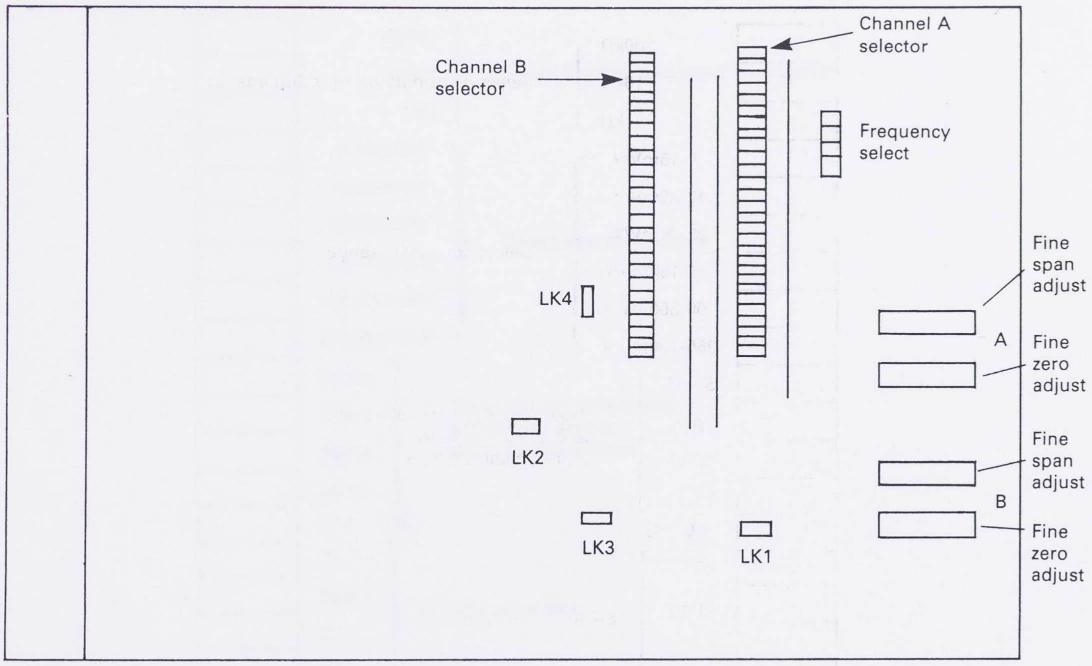
ACN 008 644 273

PATERSON INSTRUMENTS .

LVDT FORMER (12) FOR
LVDT LOAD CELL

A3	SCALE 4:1	DATE MAY 2001	DRAWING NO 4506

Figure 40 **Card layout**



Oscillator frequency

Two oscillator frequencies are selectable using the four way set of pin pairs. If the links are placed over the two centre pairs the oscillator will run at 5kHz, and, if over the outer pairs, at 10kHz.

Figure 41 **Oscillator frequency links**

Frequency should be selected to be near the zero phase shift frequency of the transducer for minimum temperature drift. Most transducers are calibrated at 5kHz and so this frequency can be relied upon for good results. However, if a faster speed of response is required, then the 10kHz oscillator frequency can be used with most transducers (not ac long strokes).

Output filter frequency

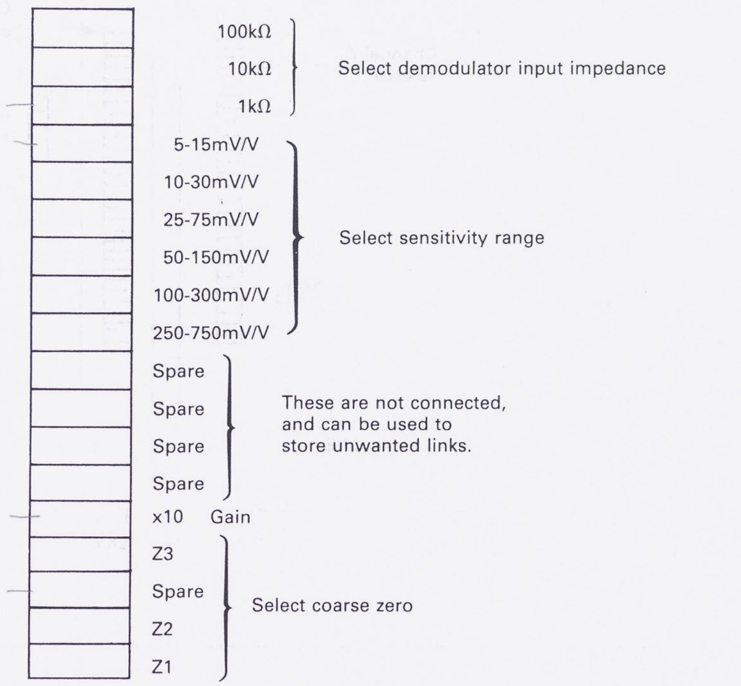
Each demodulator has a low pass filter on the output to remove the ac signal used to energise the transducer. The cut off frequency of this filter can be set to either 500Hz or 1kHz using pin pairs LK1-LK4. Under normal use the 500Hz setting would be used with the 5kHz oscillator frequency and 1kHz with the 10kHz oscillator. The benefit of using the higher frequency is that the output will follow the movement of the transducer armature more quickly, but use of the lower frequency results in less ripple on the dc output. The best compromise is with the settings above, but if, for example, a fast response with a long stroke is required, it will be necessary to use a 5kHz oscillator and a 1kHz filter. The disadvantage is an increase in ripple. LK1 and LK3 are used for Channel A. Put the link on LK1 for 1kHz filter frequency, or on LK3 for 500Hz.

LK2 and LK4 are used for Channel B. Put the link on LK2 for 1kHz filter frequency, or on LK4 for 500Hz.

Demodulator input impedance

Different transducers are calibrated into different loads. For instance most LVDTs are calibrated with 100kΩ loads, but half bridges use 1kΩ. For this reason the input impedance of the demodulator can be set to 100kΩ, 10kΩ or 1kΩ, using the three pin pairs at the top of the eighteen way selector. The channel A and B selectors are laid out as shown in Figure 42.

Figure 42 **Channel A and Channel B selectable links**



A link should be placed over one of the 100kΩ, 10kΩ or 1kΩ pin pairs to select the correct impedance. If unsure, use the 10kΩ setting.

Span and zero

To set up the span and zero controls, examine the output that will finally be required, ie: the voltage or current output, to avoid errors in the current drivers. If the A±B/2 facility is being used, then it is possible to calibrate on (for instance) the A+B output, again for maximum accuracy.

Note that it is necessary in this case to zero the unwanted transducer, or remove the secondary and short the demodulator input so that one transducer is examined at a time.

With transducers such as load cells that have an obvious centre point (ie. no load) then it is merely necessary to set the card span and zero as described below. However, for LVDTs and half bridges it is necessary to find the mechanical zero (ie. centre of linear stroke) first. To accomplish this:

(a) Remove transducer connections from card input to demodulator.

(b) Short the demodulator input, to simulate a transducer at centre of stroke.

(c) Read output from card.

(d) Remove short and reconnect transducer.

(e) Adjust transducer to give same output reading as at step (c). The transducer is now set to the middle of its linear stroke.

To set the card span and zero it is necessary to set some links and then use the fine span and zero controls for final adjustment. There are nine coarse span ranges, in two overlapping ranges of six each:

Range	Transducer	Sensitivity	Select	Select
	Minimum	Maximum	Pin Pair	×10 Link
1	250mV/v	750mV/v	250-750	No
2	100	300	100-300	No
3	50	150	50-150	No
4	25	75	25- 75	No
5	10	30	10- 30	No
6	5	15	5- 15	No
4′	25	75	250-750	Yes
5′	10	30	100-300	Yes
6′	5	15	50-150	Yes
7	2.5	7.5	25- 75	Yes
8	1	3	10- 30	Yes
9	0.5	1.5	5- 15	Yes

Selecting the ×10 link increases the gain of the amplifier and so reduces the necessary sensitivity of the transducer. The span control is used to set the span in the range between minimum and maximum. The above sensitivity ranges are for a standard ±5V or ±10mA output (10V or 20mA total range).

If a different output range is required (say V Volts) then the necessary transducer sensitivities shown should be multiplied by V/10. For example, if an output of ±3V is required (total range 6V) then range 1 becomes 250 × 0.6 to 750 × 0.6 which is 150 to 450mV/V.

Eight coarse zero ranges are provided and selected by linking up to three off pin pairs Z1 to Z3. Again a fine control is used to set the zero anywhere required. On minimum gain, the amount of zero offset provided by the links is:

Z1 −1.5V = −30% of 5V
Z2 −3V = −60% of 5V
Z3 5.5V = 110% of 5V
Potentiometer −2V to 0V = −40% to 0% of 5V

MADE IN CHINA