Preliminary thoughts & studies for a 'simplified' APT.

How many times is such an instr. likely to be used? – probably something like 50 to 60 times/annum.

If it is to be left unattended, does it need some automatic shutdown procedure & equipment to protect it against the elements? Almost certainly yes.

Should also consider protection against cats, hedgehogs & foxes!

Any electrics will need good waterproofing.

Should it be designed for a max. observing time of say 6 hrs. or should it be given a 24 hours capability. 6 hrs would appear to be O.K. since it could be reset & left to run from say midnight onwards.

To make the system so accurate that it could be made to find its own starfield does not seem a worthwhile objective. If the observer has to open up the obsy & start up the system he may as well do the initial locating of the object & set the system running.

Photo-metric side. SEEP style system with fluid light guide. Will only be used for diff. photometry – most likely on EB's. – initially single filter work – later UBV.

Telescope optics:

8" f4 Schmidt-Newtonian — will consider possibility of removing corrector plate from front of tube & replacing this with conventional spider. This will undoubtedly downgrade the star images away from the centre of the field of view but the centre image quality should be perfectly satisfactory for photometry.

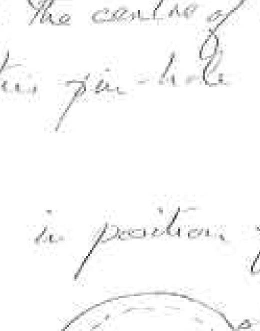
Principles

If we make the assumption that the drive will be smooth & good (but by no means perfect) then the control system will need to check & re-centre the object, say every 2 photometric cycles - obviously could be modified after obtaining experience.

N.B. Non-perfection would include mis-alignments & imperfections (cyclic errors?) & incorrect drive rate.

Initial work should be carried out using a 100 arcsec pin-hole. Assuming a star is placed in the centre of this pin-hole it must still be somewhere in this pin-hole when returning to it after a 20 min-period.

∴ allowance of 40 arcsecs for maximum error in position from all causes after a 20 min. period.



i.e Star must be within this O after 20 min.

min.

— " —

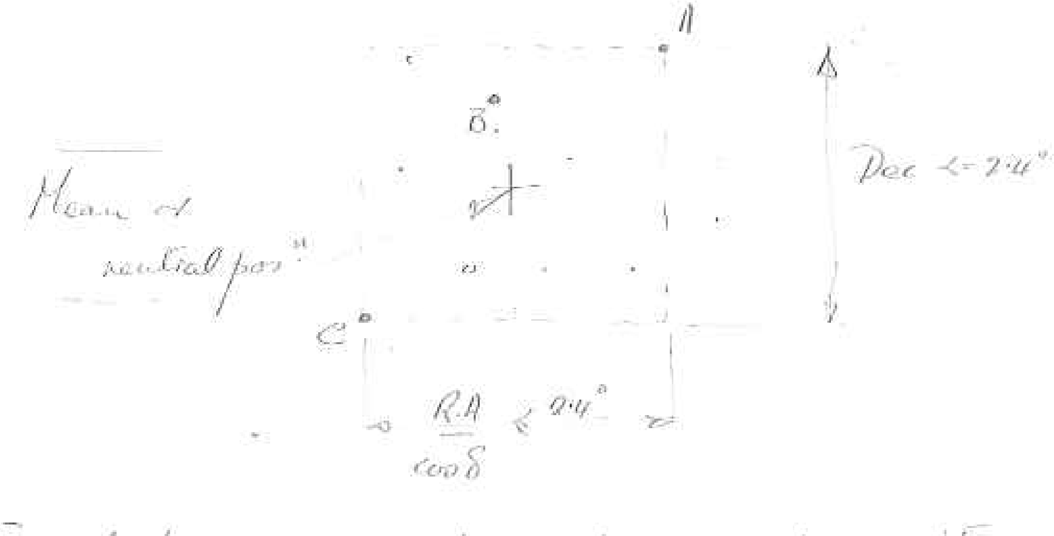
Depending on final quality, this could mean a re-alignment correction needed every 20 mins.

N.B. Length of dew-cap will need to be s.t. no water vapour is deposited on optical surfaces.

The dewcap could be made long & robust since it will almost be certainly necessary to have the tube balanced about the declination axis.

Both RA & Dec to have capability of v. fine adjustment over say $\pm 2\frac{1}{2}^\circ$

$\therefore$ if 3 stars are shot in most PEP work then max^m sep^n in dec between any two of the 3 stars must not exceed 2.4°; likewise 2.4° in R.A.



Speed of movement - when changing star positions.
6 secs. would not seem unreasonable for making the 'full' movement of 2.4°. Both changes could be made 'simultaneously' since there is unlikely to be any gyroscopic effects!!!

Time for 1 rev. at this rate $= 6\,secs \times \frac{360}{2.4} = 6 \times 150 = 900$ secs.
$= \underline{15}$ mins

or $\omega = \frac{2\pi}{900}$ rads/sec.

If the 2.4° of movement is achieved by a rad. arm say 12" long then movement of end of rad. arm
$= 12 \times \sin 2.4° \approx 0.5"$

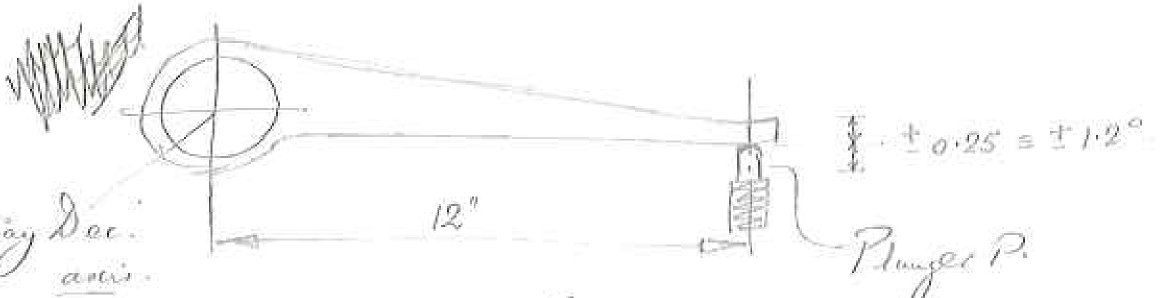


Fig 1.

N.B. 3/8" B.S.F. – 20 t.p.i.
Check with Mike Seymour re. 8, 10, & 12 mm threads.

Size of 100 arc sec pinhole for 8" f.4
$= 8 \times 4 \times \sin\left(\frac{100}{3600}\right)° = 32 \times \sin \frac{1}{36}° = 0.0155"$ or 0.390 mm.
say 0.4 mm

$x \approx \sin x$. for small x, x in rads
$= x \times 0.01745$ if x in degrees.

∴ to move star in p.h. by 5 arc secs means moving plunger P, fig 1.
by $12 \times \sin \frac{5}{3600} \approx 12 \times \frac{5}{3600} \times .01745 = \frac{.01745}{60} \approx .0003"$
i.e 3/10 of a thou!! (1/5 of a human hair!)

Just about feasible with very careful workmanship!!
It is apparent that both axes will need to be carefully balanced & as frictionless as possible.

1986 Dec? (5)

Refer to page 3.

Slew rate from star to star is 1 rev/15 mins. this is approx 900 times faster than the polar rate.

It would be more sensible is it to seperate the slewing from the driving — is it? YES.

If the drive for the polar axis was a stepper motor going at a slow speed then it would have to speed up by a factor of 100 times to catch up 2.4° in 6 secs.

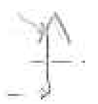
Looking at the torque vs steps/sec curves for stepper motors this is far too big a turn down (or increase) in speed.

N.B. Movement ½" of ⌀10 screw (if 20 t.p.i.) — 2000 steps ≡ 2.4 × 3,600 arc secs; each step ≡ 4.3 arc secs.

10 revs in 6 secs. $\equiv \frac{10}{6} \times 200 = \frac{2000}{6} \approx 333$ full steps/sec.

Assume axial loading on thread = 10 lbs.

Load on thread $\approx \frac{10}{.866} = 11.55$ lbs.



Torque reqd to turn screw $\approx 11.55 \times .2 \times .15$

$= 0.35$ lbs ins

100 mNm $\approx 100 \cdot \frac{1}{1000} \cdot 0.24$ (lbs) $\cdot 39.4$ (ins) $= 0.24 \times 3.94$

$= 0.9456$ lbs ins

Using tech. spec for 332-082 Working torque (max)

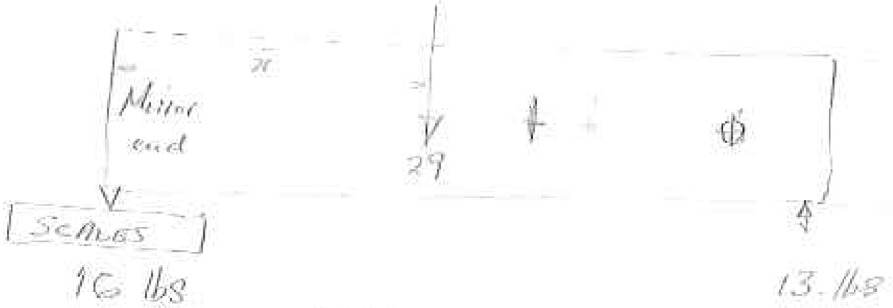
$= 320$ mNm $\approx 3.2 \times 0.9456 \approx 3$ lbs. ins.

There would appear to be an adequate margin of safety.

Existing tube, as is, complete with mirror, correction plate etc.

~10" square external, height 32"

Weight 29 lbs (bathroom scales)

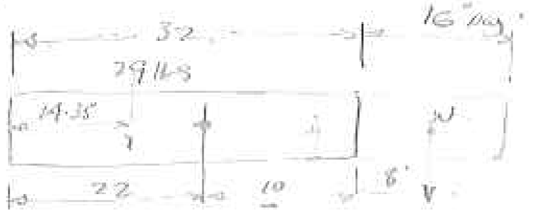


[illegible] $13 \times 32 = 29 \times \bar{x}$

$$\bar{x} = \frac{13 \times 32}{29} \sim \underline{14.35"}$$

It is preferable that there should be a long dewcap — say 18"

Assume initially that pivot point of tube is 22" from lower end



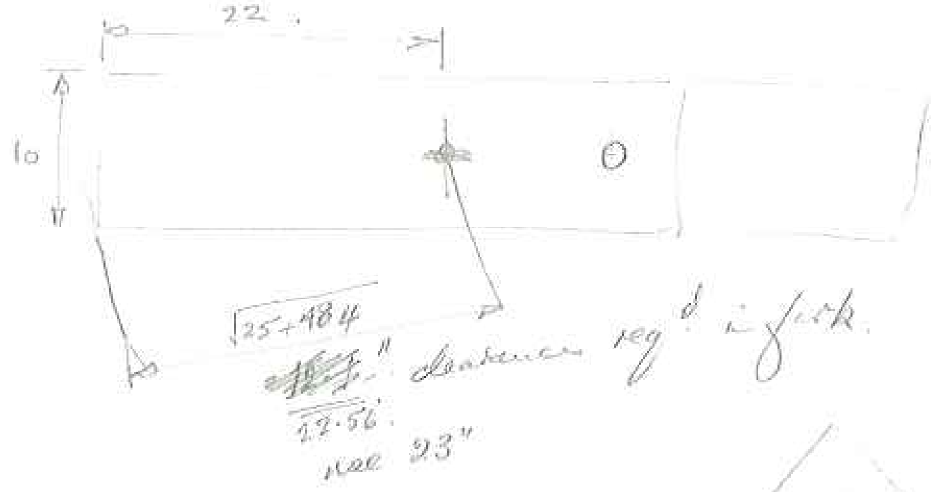
$29(22 - 14.35) = W \times 18.$ $\quad W = \frac{29 \cdot 7.65}{18}$

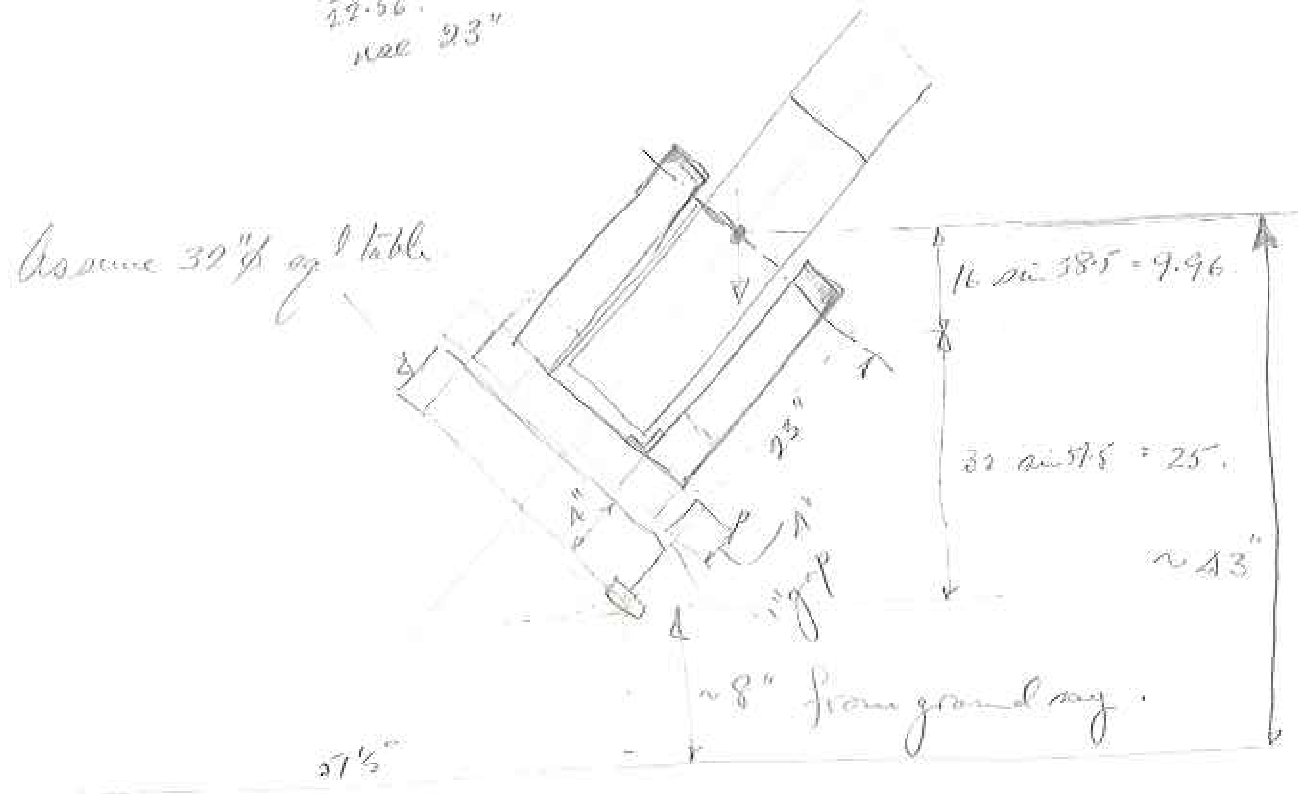
$= \underline{12.32 \text{ lbs}}.$

In addition to extended dewcap there could be a few lbs of weight in finder, bearings clamps etc. — estimate total wt. of tube = 50 lbs. With finder & PMH this may even increase to 60 lbs

Tube will need to be well balanced on roller bearings of min^m friction involved to ensure accuracy of re-positioning with repeated movements

In addition to being automatically moved by use of stepper motor it will need a manual form of adjustment for initial setting.





Might pay to raise higher on a brick pedestal.

Use of piano wire as a drive?

$E = 30{,}000{,}000$.

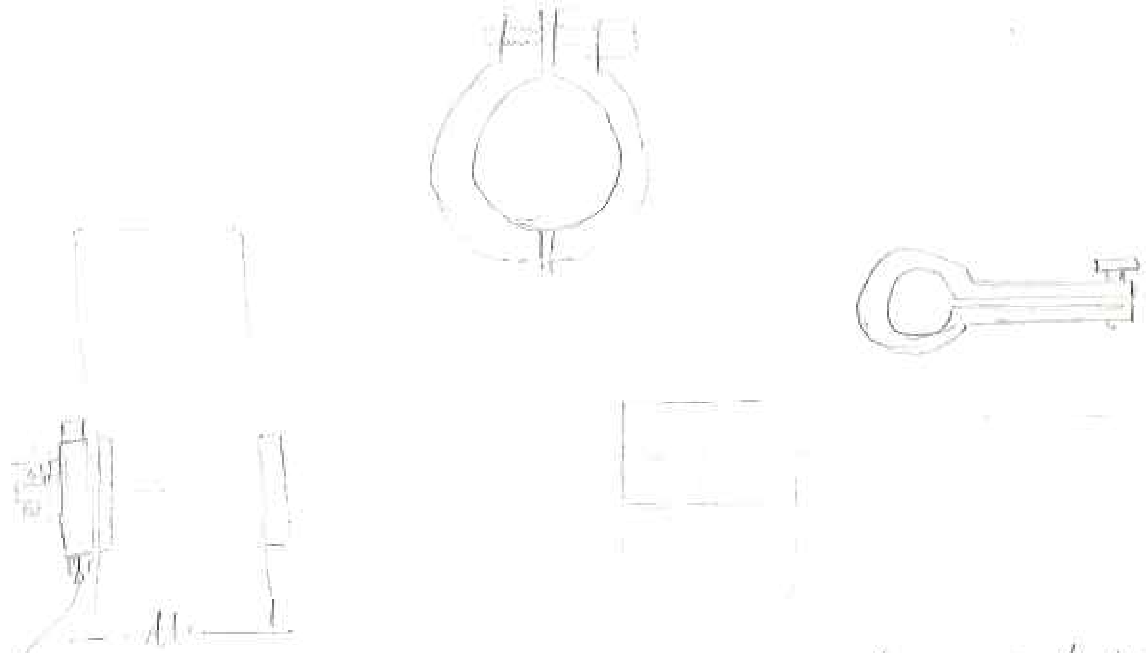
$f_{su} = 350{,}000$. $0{\cdot}020\ \phi$ Breaking load $= 350{,}000 \times \frac{\pi}{4} \times 4 \times 10^{-4}$

$= 35 \times \pi = \underline{110 \text{ lbs}}$.

If we assume yield stress $= f_y = 60\%\ f_{su} = 350{,}000 \times 0{\cdot}6 = 210{,}000$

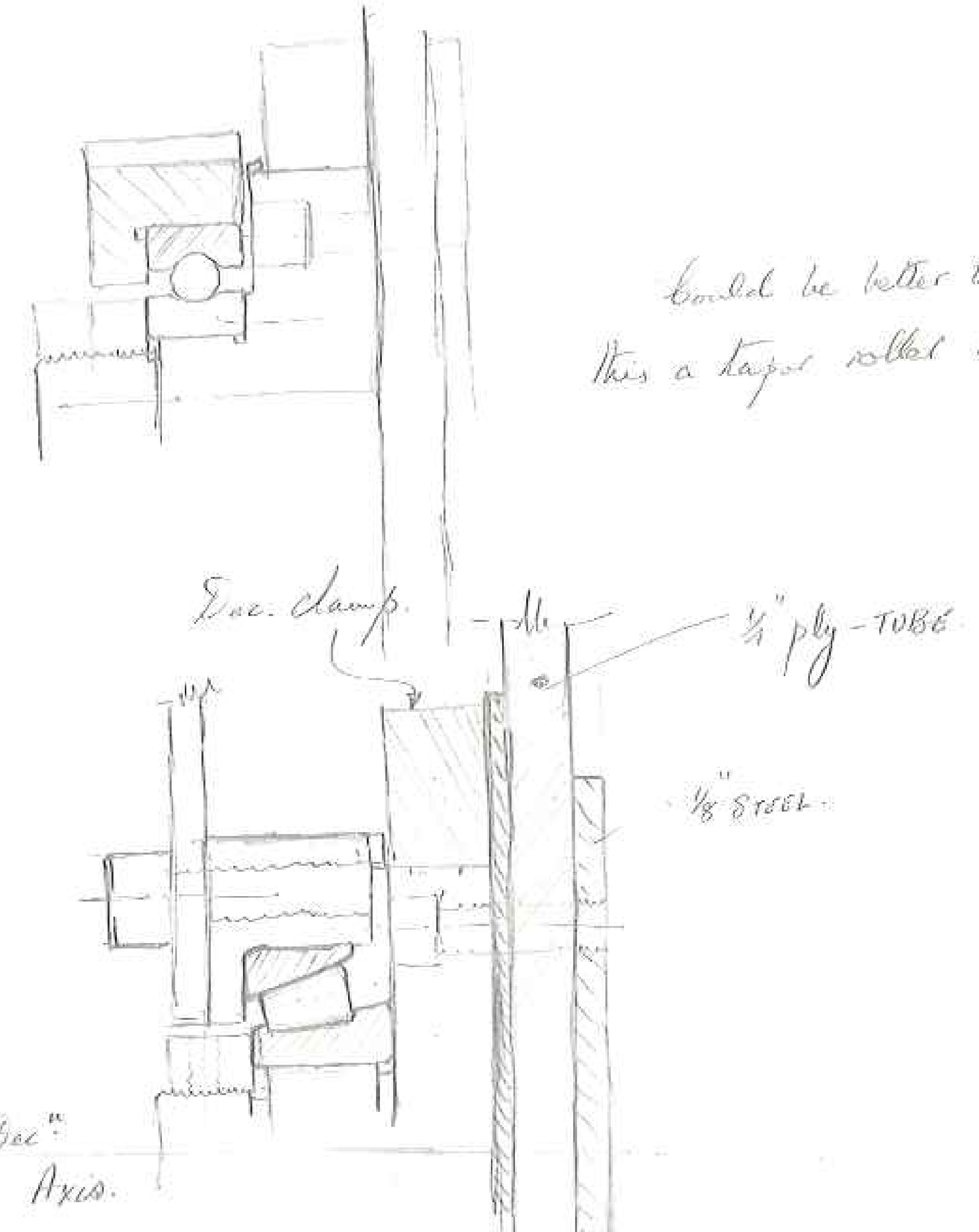
$\frac{f}{y} = \frac{E}{R}$ $\quad R = \frac{Ey}{f} = \frac{3 \times 10^7 \times 10^{-2}}{21 \times 10^5} = \frac{3}{2{\cdot}1} = 1{\cdot}43"$

i.e. min^m drum dia say 3" – not really worth serious consideration.



Manual locking ring on one side of tube — Automatic locking ring on t'other side. — no reason why they should be identical.

Could be better to make this a taper roller race.



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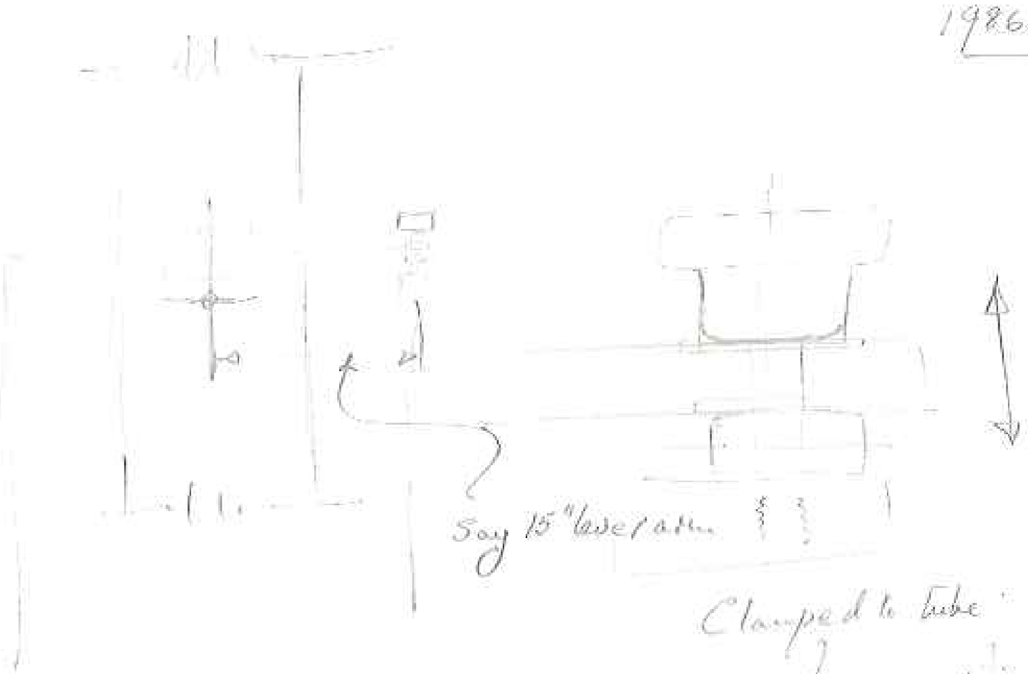
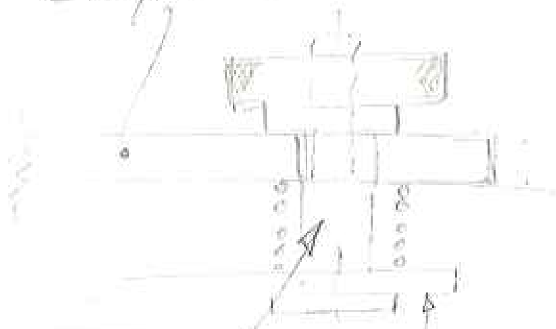
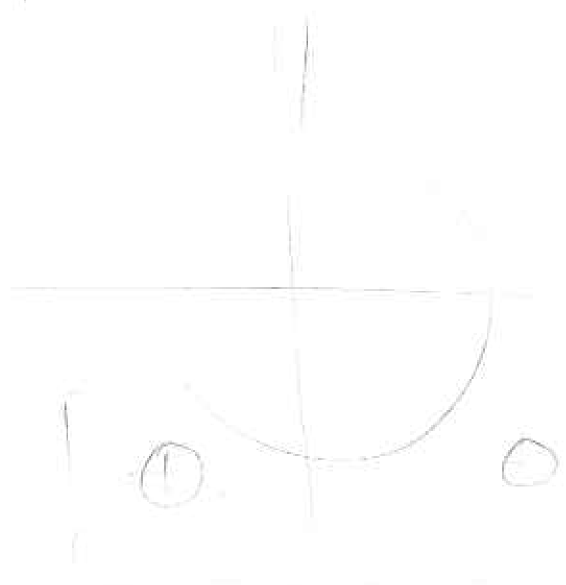
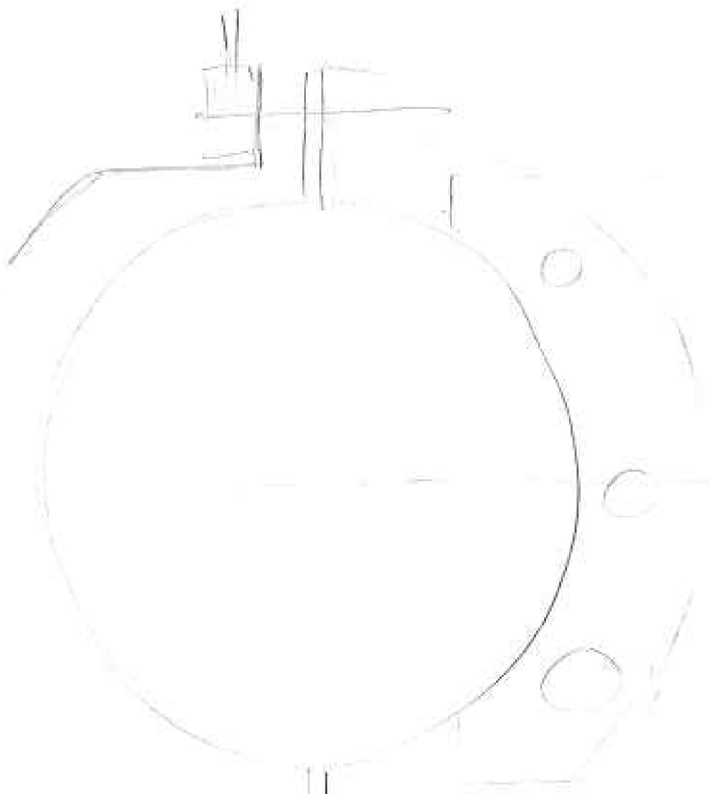
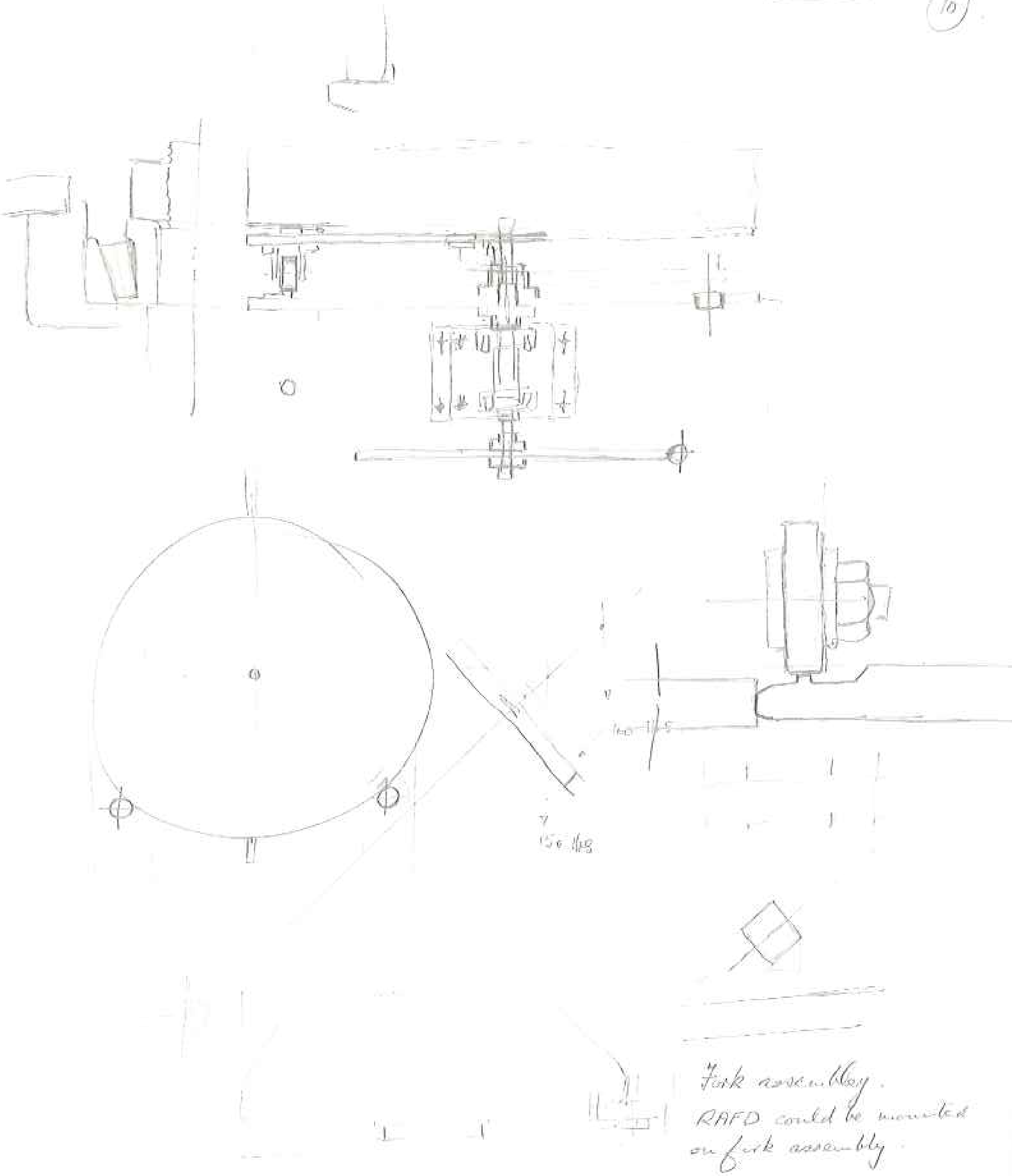




Manual adjustment for Decⁿ







Fork assembly.
RAFD could be mounted on fork assembly.

P.A Drive system

Basic to all types Synchronous motor with frequency controller system for fine tuning.

1/ Conventional worm & worm wheel - will need to be of a fair size - say between 12" & 24" ϕ. Probably need careful lapping-in to achieve necessary quality. If obtainable could be best & simplest.

2/ Friction wheel drive

~32" ϕ - outer edge accurately round & smooth Supported on two roller or ball races - say 1½" o.d. - outer race driven by synch.s motor via a reduction gear box.

Accuracy of this system will depend on the the 32" o.d. being accurately round & the o.d. & the o.d. of the ball races being kept very clean.

See article by B. Knight "An efficient gearless telescope Drive" Vol. 90, No 1. ; Session 1979-1980. p. 45

3/ Lead screw thread type. - similar to that currently used on obs.y except it would be a lot longer, to give 4/5 hours of drive & would most likely need a computer controlled stepper motor to give cond any.t speed of P.A.

1986 Dec 8.

(12).

Some further limitions / requirements of system.

1/ Due to generally poor photometric skies there seems little point in designing the system to operate below an alt. of 20°.

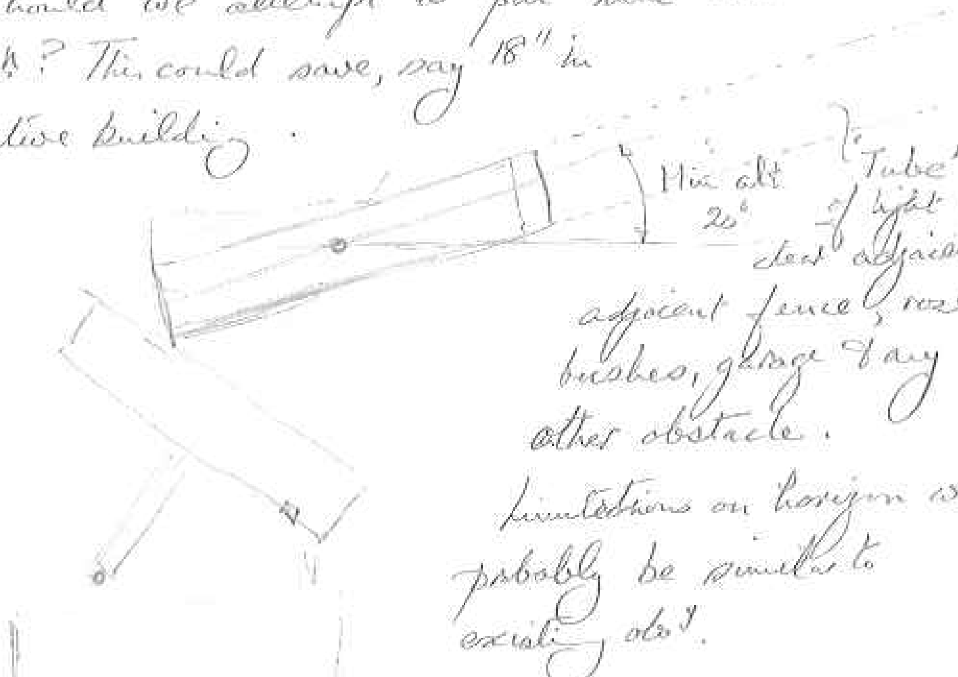
2/. In the event of poor weather arising after an observing session has be initiated there will be a need for :-

a/. deciding whether to accept or reject a reading

b/ repeating a reading

c/ when to shut down system.

d/ most important - automatic shut-down (at least provision of protection) in event of rain. For emergency closedown should we attempt to put tube into horizontal posn? This could save, say 18" in height of protective building.



'Tube' of light to clear adjacent adjacent fence, rose bushes, garage & any other obstacle.

Limitations on horizon will probably be similar to existing obs.

e, All electrical & control gear will need to be waterproof - not forgetting photometer system.

1986 Dec 8

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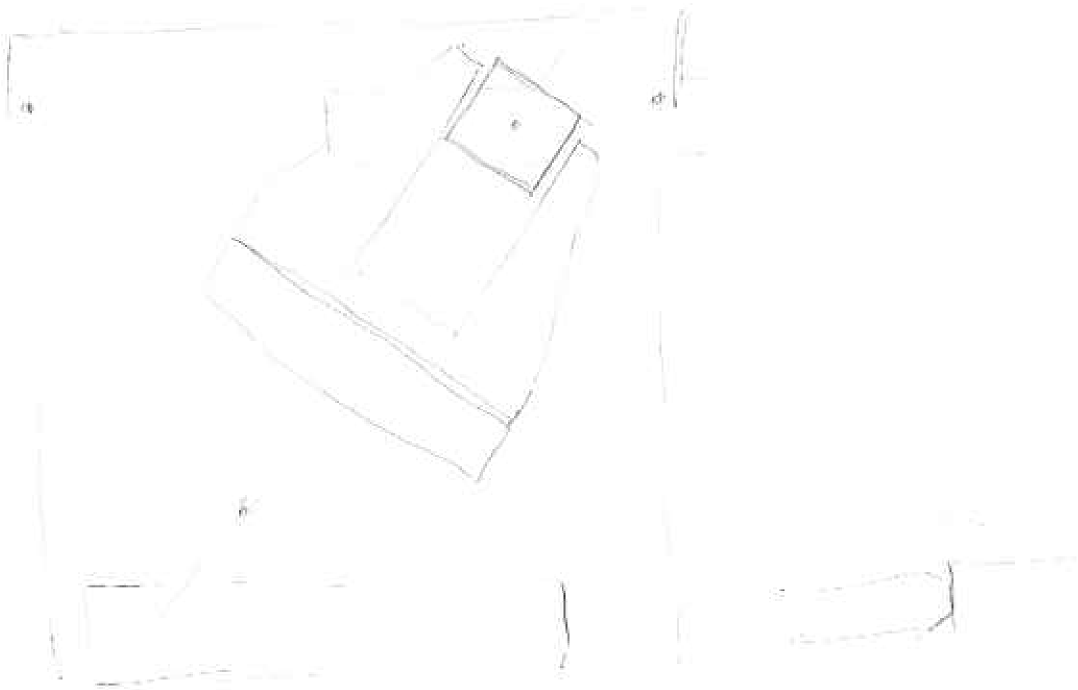
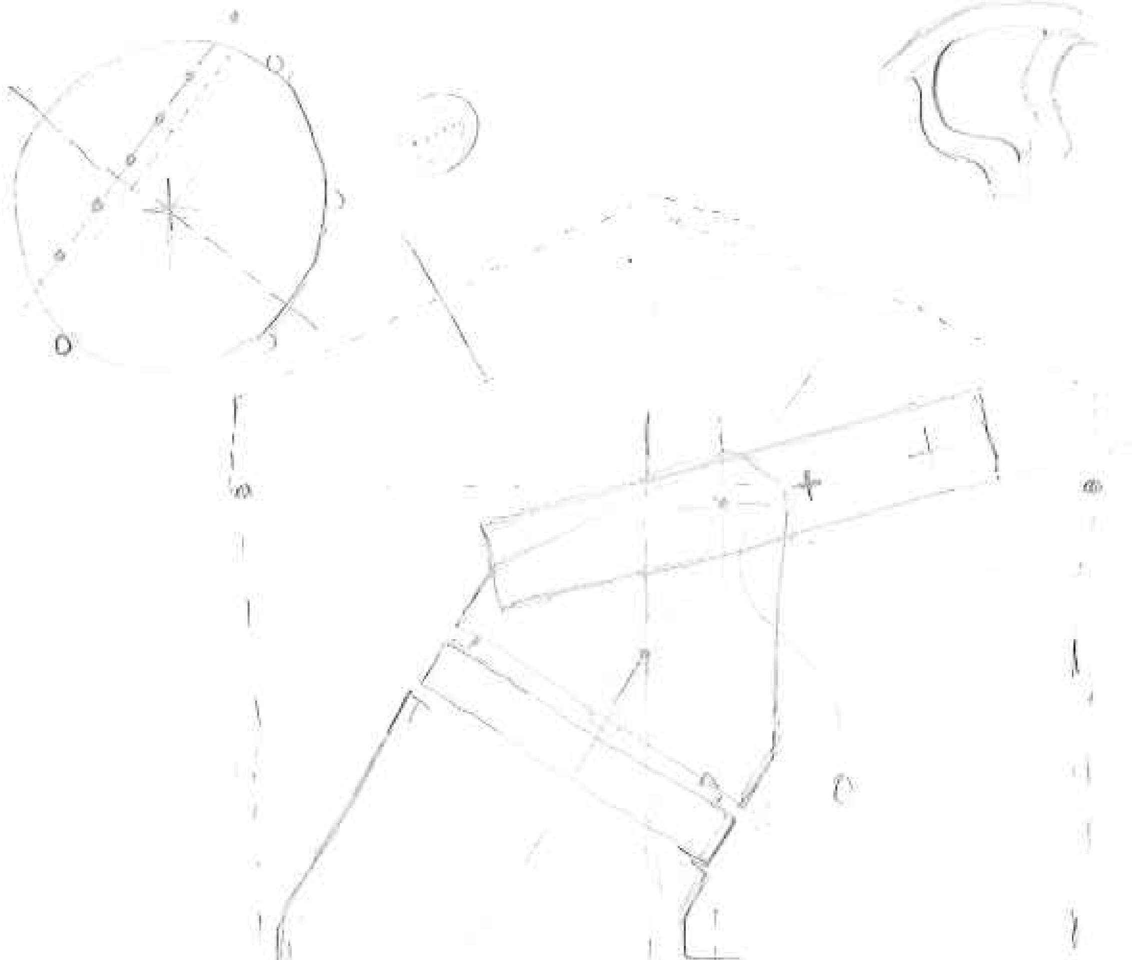




1986 Dec. 12

Frictionless drive system. – Questions for Ron Arbour. **13A**

Size of drive wheel, O.D.? 24"

Width of face. 1/4"

Type of machining – turning or grinding?

Mat?. Steel.

How was wheel centred on P.A.?

How was accuracy measured?

Drift in arc secs / 15 mins of time?

Pin-hole?

X-hair?

Graticule?

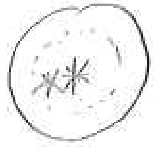
Eye-piece power?

Is it a synchronous motor?

with frequency controller?

Size of telescope? 16"

Wt. of telescope? Ton

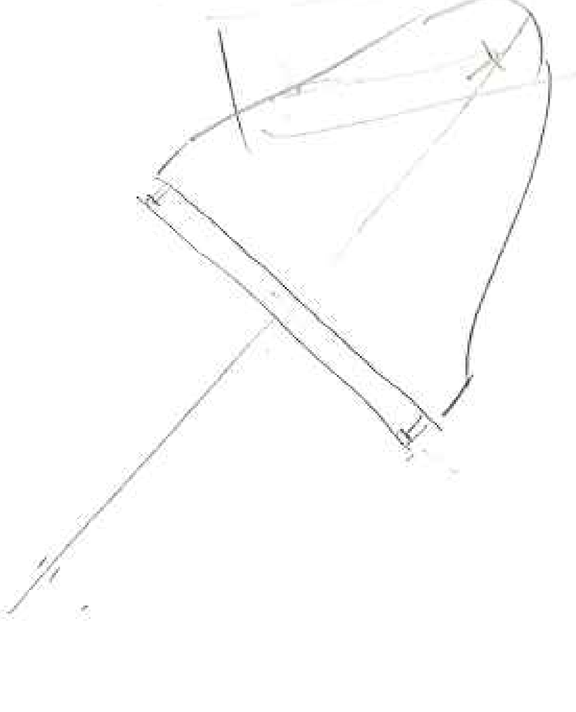


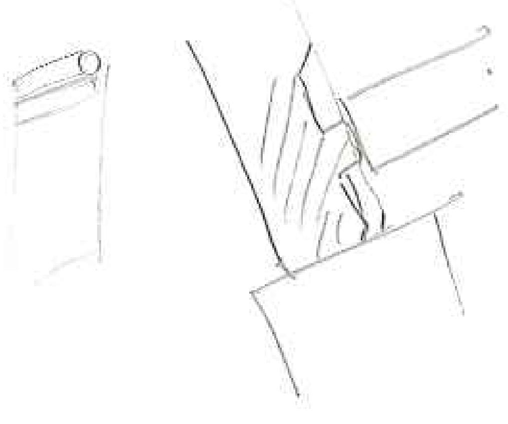
Is it protected from the wind.

Size of driving disc? 1/2"/5/8"

Force between driving disc & driving wheel?

SBVSBC





1986. Dec 15.

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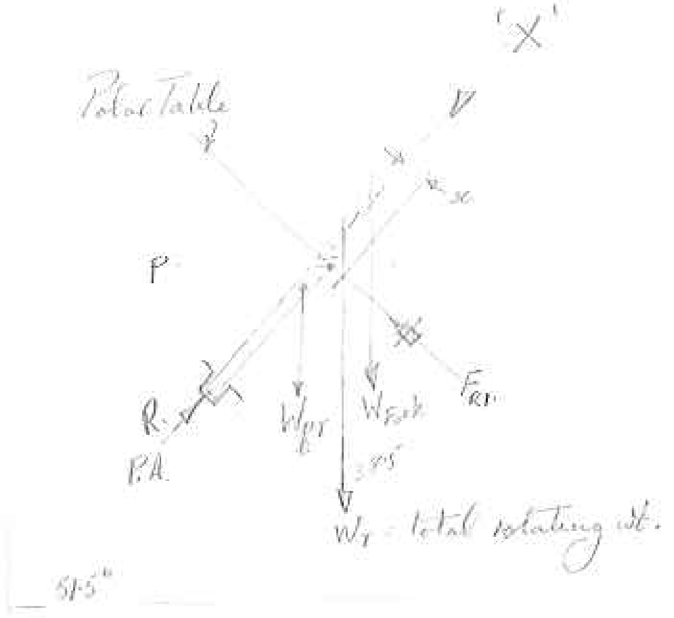
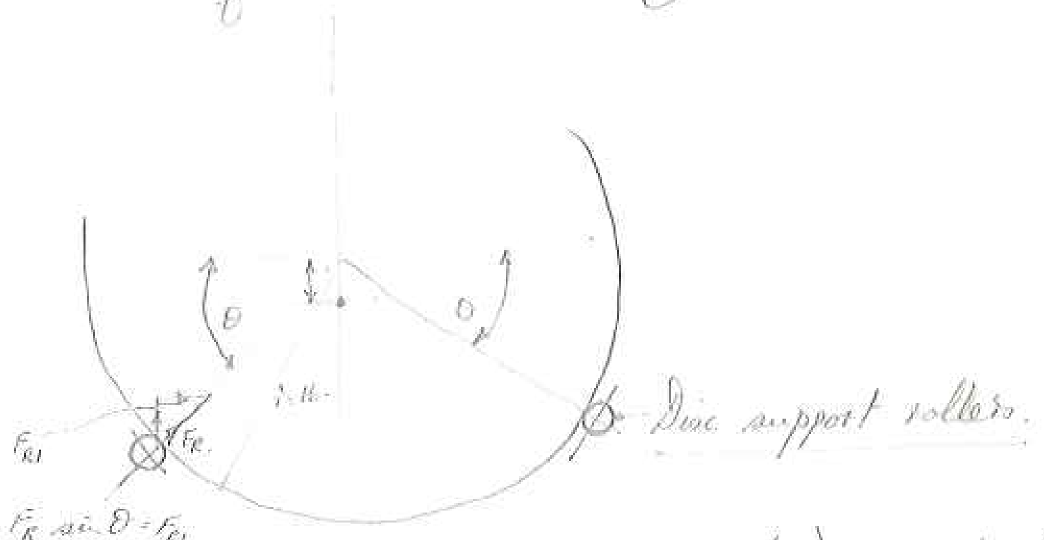


View in direction of arrow 'X' i.e. looking down P.A.



Q. Is there a preferred posn for the two disc (P.T.) support rollers?

$$2 \times F_{R1} \approx W_T \sin 38.5$$

Assume $W_T = 400$ lbs $\qquad F_{R1} = 200 \sin 38.5° = 124.5$

If $\theta = 45°$ then $F_R = \frac{F_{R1}}{\sin 45°} = \frac{124.5}{\sin 45} = \underline{176 \text{ lbs}}$

If $\theta = 60°$ then $F_R = 144$ lbs. ; % reduction $= \frac{32}{176} \times 100 = \underline{18.2\%}$

Assume that F_R = near enough 200 lbs then tangential force required to steady move P.T. < 4 lbs (i.e. < 1% of 2 × 200).

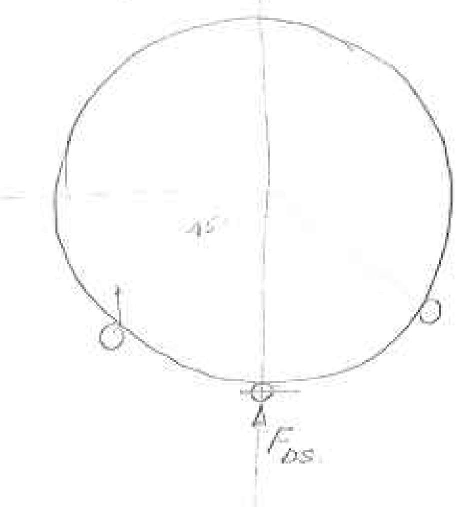
1986 Dec 15

p. 15

If $\mu = 0.03$ between driving shaft & P.T.

then min. force reqd between driving shaft & P.T.

$\simeq \frac{4}{\mu} = \frac{400}{3} = \underline{\underline{133 \text{ lbs}}}$



~~$F_{R1} = \frac{2(400 - F_{DS})}{2}$~~ $2 F_{R1} = 400 . \sin 38.5^\circ - F_{DS}$

Tang^l force req^d $= \frac{2 F_{R1}}{\sin 45} \times 1\% = \frac{.01 (400 \sin 38.5^\circ - F_{DS})}{\sin 45^\circ}$

$= F_{DS} \times \mu$

$4 \sin 38.5 - .01 F_{DS} = F_{DS} \times .03$

$\therefore F_{DS} = \frac{4 \sin 38.5}{.04} = 100 \sin 38.5$

$= 62 \text{ lbs}$

use $\mu = 0.03$ for an oiled surface

$4 \sin 38.5 - .01 F_{DS} = F_{DS} \times .03 \times \sin 45$.

$F_{DS} = \frac{4 \sin 38.5}{.02121 + .01} = 128 \sin 38.5$

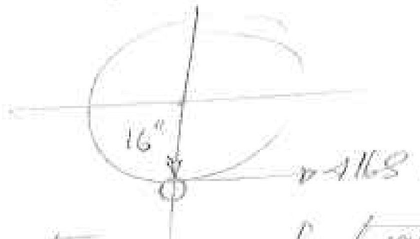
$= \sim \underline{\underline{80 \text{ lbs}}}$.

therefore for estimation purposes use 100 lbs loading on drive shaft

1986 Dec 15

To estimate power requirement. p.(16)

say 4 lb tangential load reqd at 16" rad. of P.T.



W.D = torque × angle turned thro' in rads

= 1.33 × 4 × 2π ft. lbs in 24 hours.

= 33.426 ft lbs in 1440 mins

$\equiv \frac{33.426}{33,000 \times 1440}$ so even allowing a factor of 100 !!! for inefficiency power reqt very low.

$\equiv \frac{33.426 \times 100 \times 784}{33000 \times 1440}$ Watts = 0.055 Watt

The power to drive the P.T. is negligible!

All of this is very extreme since tangential force reqd to move 400 lbs is likely to be nearer 2 lbs than 4 lbs and the static coeff. of friction between driving shaft & P.T is likely to be nearer 0.3 rather than 0.03 !!

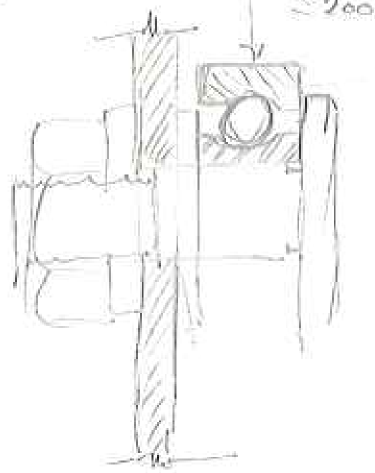
If P.T. is 32" & driving disc is 1" ϕ then torque reqd to turn driving ~~disc~~ shaft ~ 4 (48) × ½" + torque reqd to turn driving disc shaft (on roller bearings)

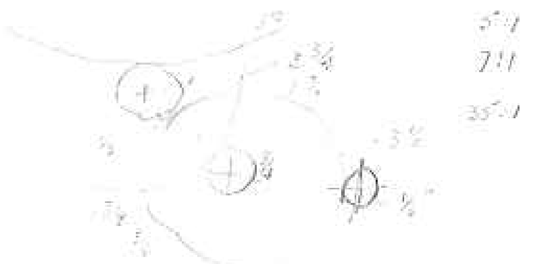
say ∴ ~ 2.5 lbs ins

Also P.T. rotates once in (1440 − 4) = 1436 mins.

& ∴ drive shaft will rotate once in $\frac{1436}{32}$ = 44.875 mins.

Use 1½" O.D. ball races to support polar table — then bore should be adequate to have a simple type of fixture:—



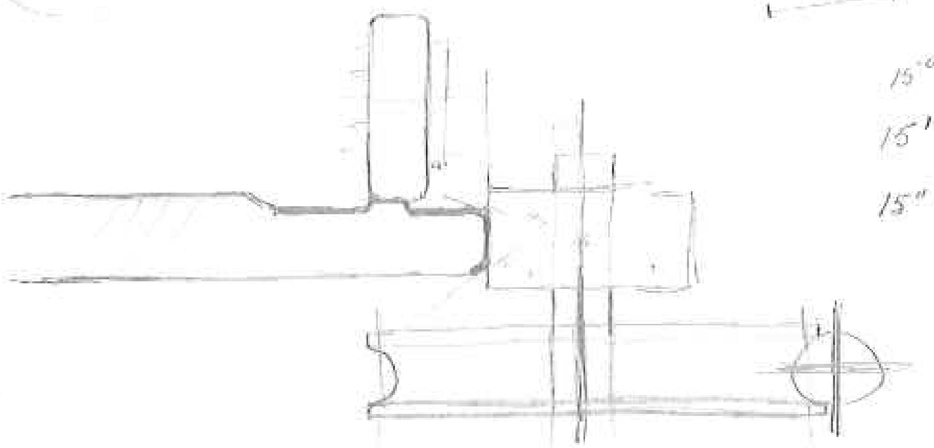


5:1
7:1
35:1

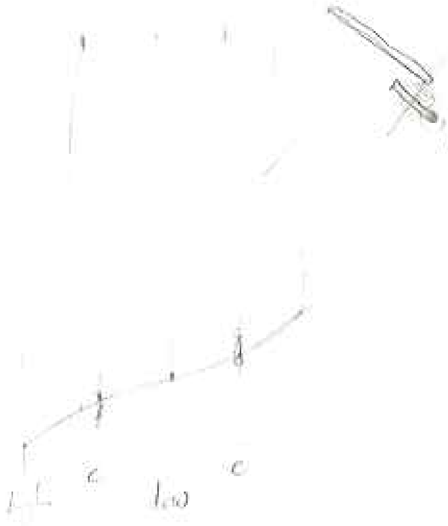
32 3/4 = 42.67

1436

33.66 : 1.



15° / hour
15' / min
15" / sec.



If drive shaft is 1"Ø then 1 rev corresponds to $\frac{360^\circ}{32} \times 3600 = 40{,}500$ secs.

in $\frac{1436}{32} \approx 45$ mins.

∴ a ½ thou, i.e 0.0005" in the drive shaft would give a max error of ≈ 16" arcsecs in half a rev i.e 22.5 mins

1986 Dec 30.

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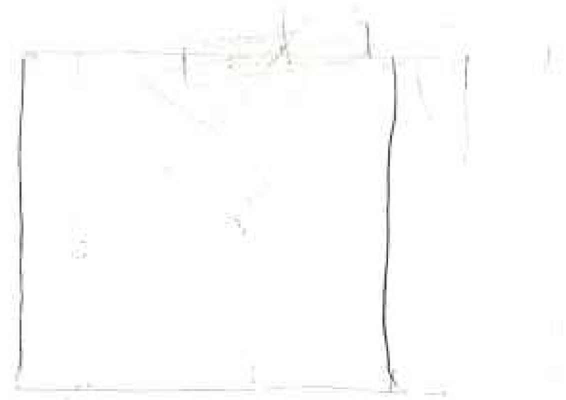
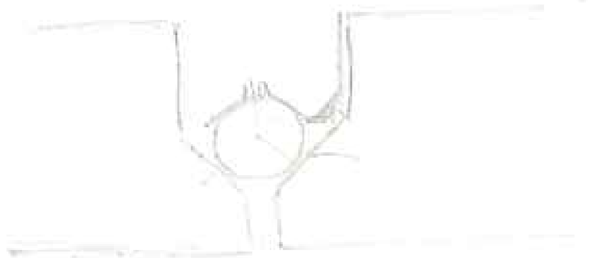
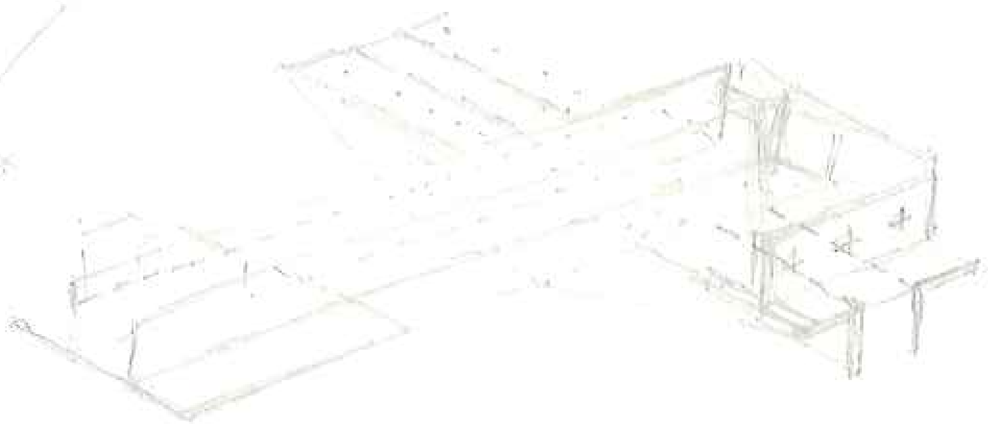
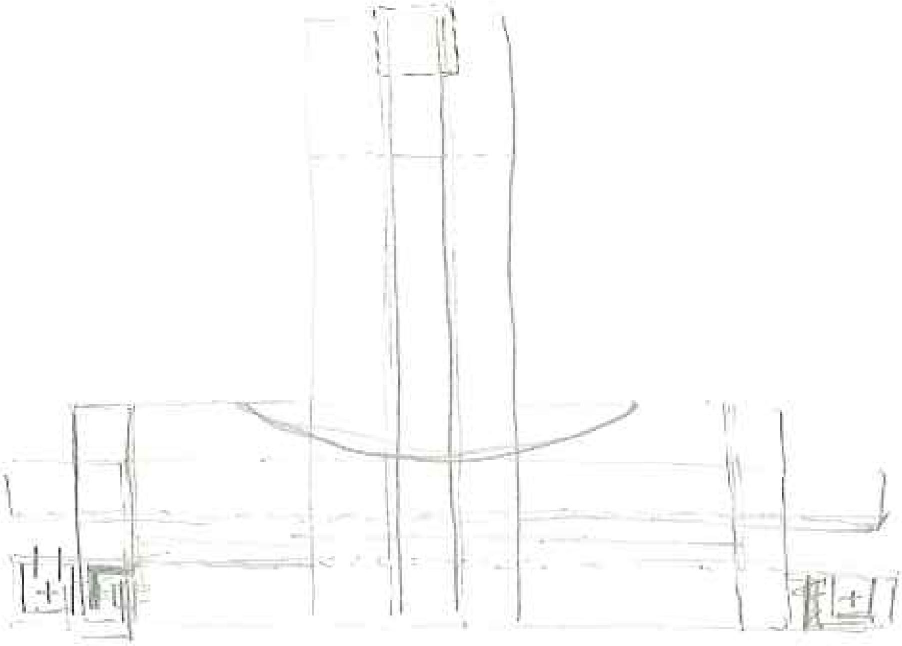


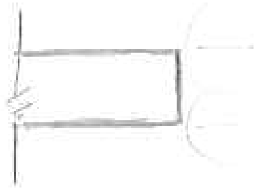


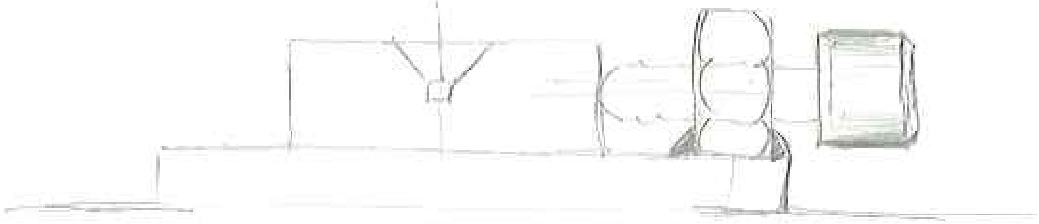


1987 Jan 1.

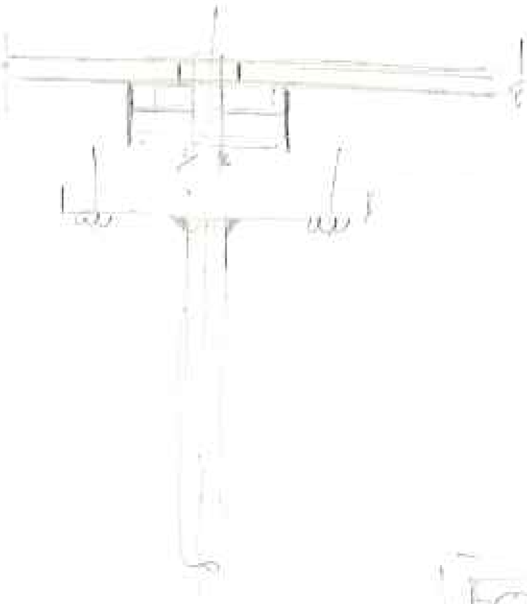
(19)

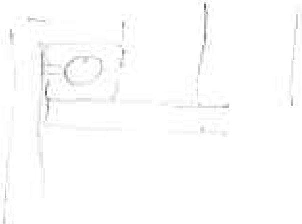






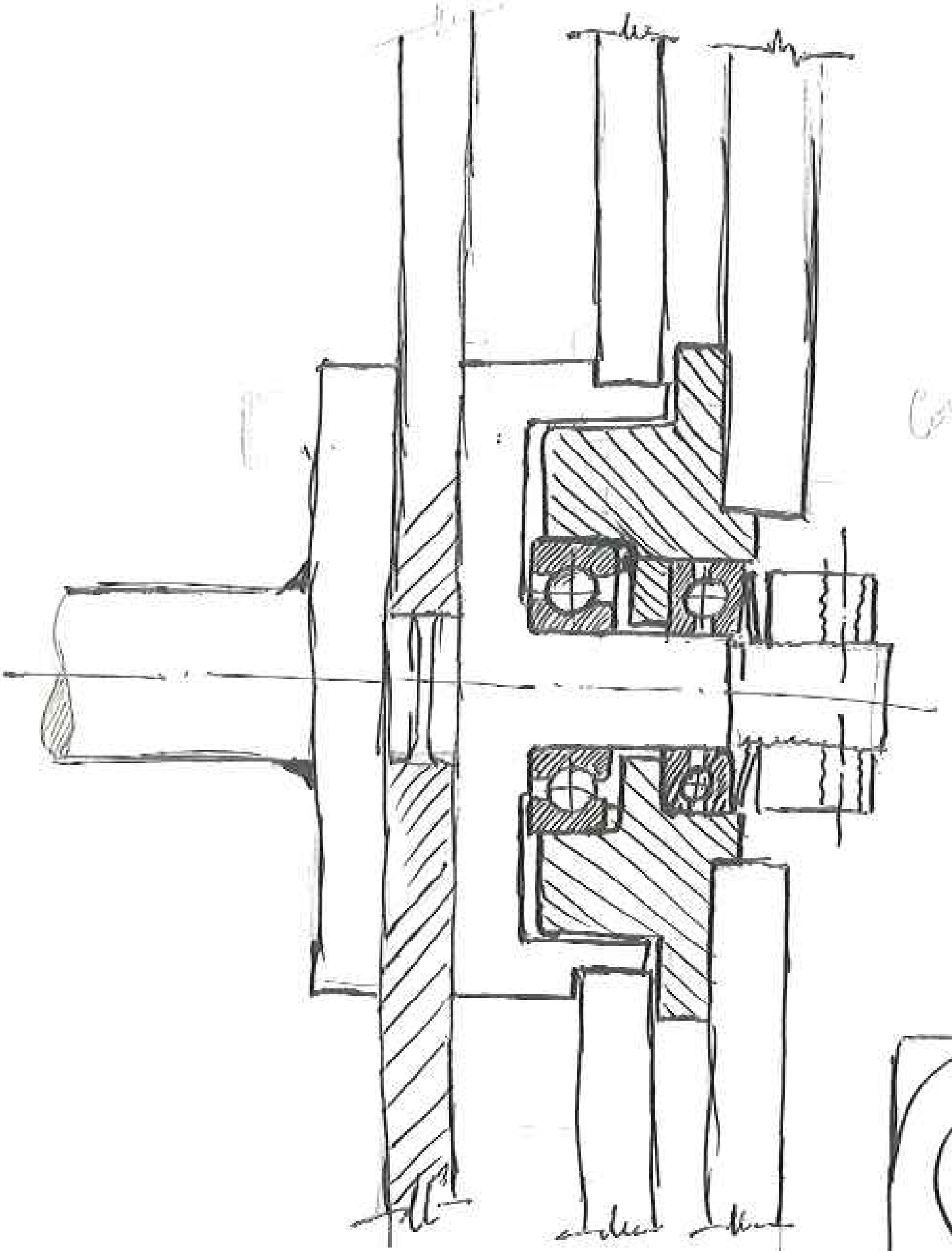
1987 Jan 1.

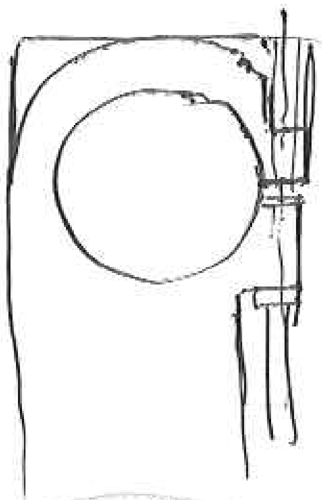


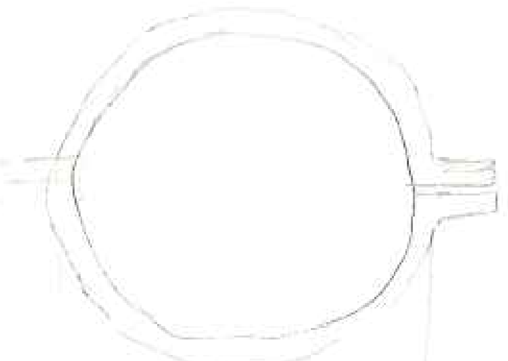


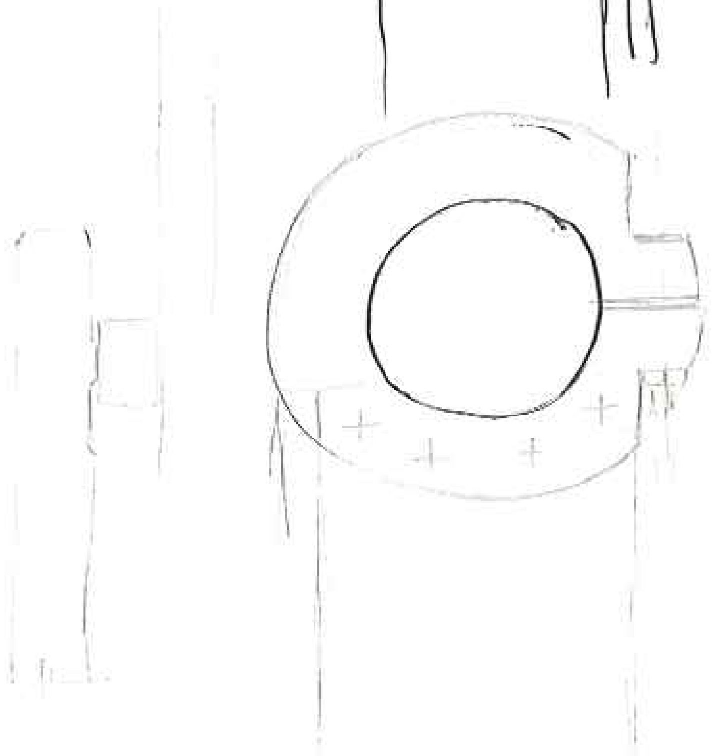
1957 Jan 5.

(21)









For purposes of drawing best to divide into major parts

A Base for Polar Axis (table)

B Polar axis including connection to fork.

C ~~R.A.~~ Drive for 'small' RA changes.

D Main synchronous motor drive for P.A.

E Fork details

F Drive for small Dec changes.

G ~~[illegible]~~ Gimbal bearings & modifications

H General layout including electronics etc.

I Observatory building.

J Foundations.

K Wiring diagrams.

L Linear Drive details ~~Bad weather – close down provision~~

M Photometer head – N. Walkers?

N Bad weather close down provision

T Telescope tube.

1987 Jan 16.

(23.)

To estimate defⁿ of 30" ⌀, 3/8" th. polar table plate due to wt of fork mounting & telescope.

Assume $r = 14"$

Replace 30" ⌀ plate by 20" wide strip.

Assume 100 lbs normal load acting.

$$\delta = \frac{Wl^3}{3EI}$$

$l = 14.$

$$I = \frac{bd^3}{12} = \frac{20 \cdot .375^3}{12}$$

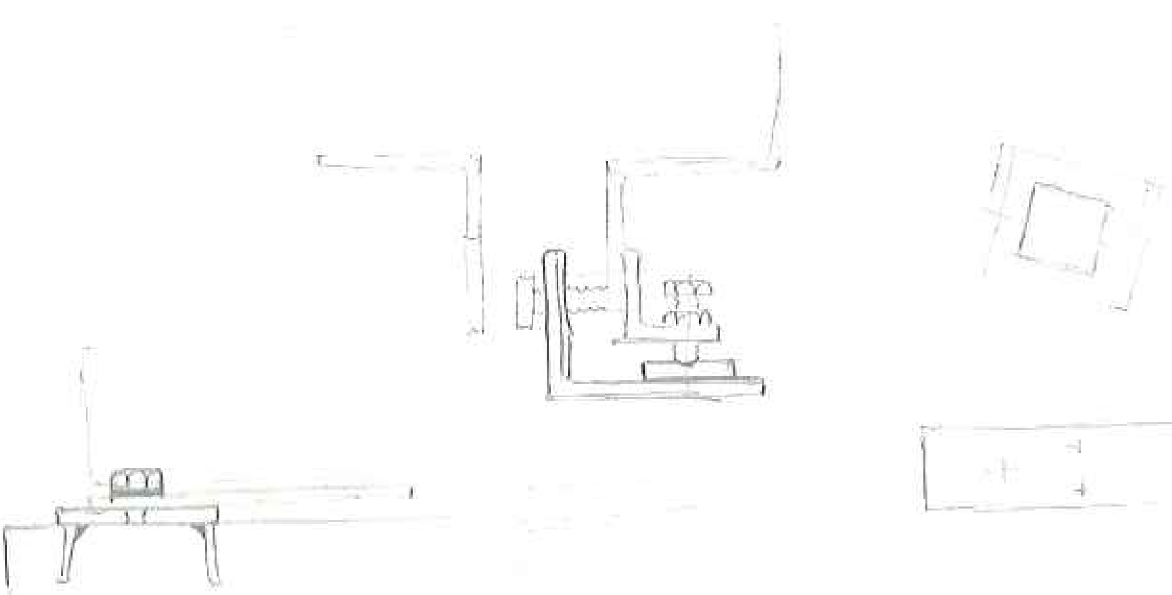
$= 0.0879 \text{ in}^4$

$$= \frac{100 \times 14^3}{3 \times 3 \times 10^7 \times .0879}$$

$$= \frac{1.4^3}{3 \times 3 \times 8.79} = 0.0347"$$

.375 100 sin 51.5° = 78.3.

As Under operating conditions the max^m actual deflection is likely to be ~ .020" (≡ 5 arc mins if considered over a 14" rad.). Any changes in declination angle due to changes in the deflection of the edge of the plate (which may arise due to opposing frictional loads at the rollers) is likely to occur only very slowly as the telescope rotates about the P.A.



(A). Base.

1987 Jan 10.

(24).

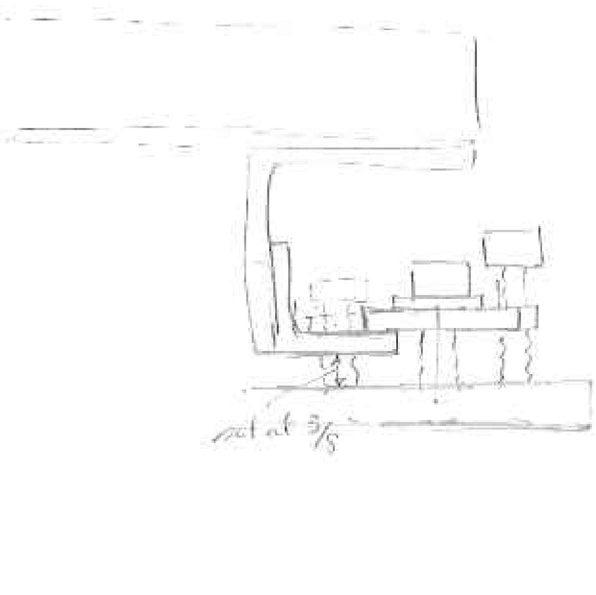
$R = (28^2 + 12^2)^{\frac{1}{2}} = \underline{30.5"}$

$30.5 \tan 1.5° = \underline{0.7"}$

Assume base can be placed in correct angular, looking on plan to an accuracy of say ± 1.5°

Assume 'lat.' angle, 57.5° can be initially set to an accuracy of ± 0.25°, ∴ vertical give & take reqd = 28 × 2 × sin 0.25

= 0.24" total.



In art. by Brian Knight Vol 90, 1. p 45.

Drive given by 1 r.p.m., output Crouzet motor

& reduction ratios of 10:1 , 12:1 & 12.033:1

125 : 12.5 150 : 12.5 600 : 49.86.

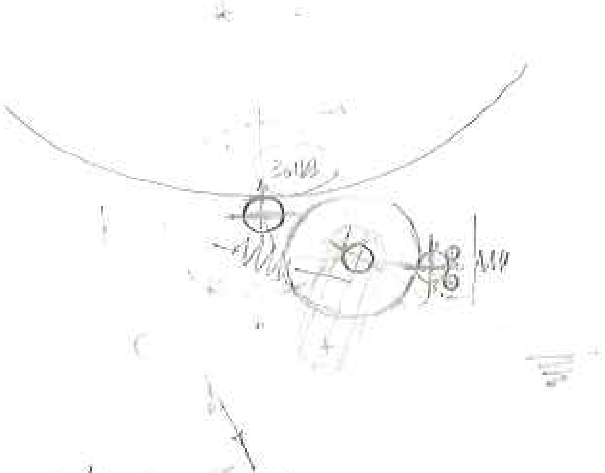
wheel ϕ's

600 : 50.14 Corrected values.

10 × 12 × 12.033 = 1443.96 mins for 1 rev of P.A.

This ratio appears to be in 'wrong direction'

10 × 12 × 11.967 = 1436.04 mins for 1 rev of P.A. ✓



The three ratios have to multiply up to approx. 1440

If P.T. disc is 30" & driving roller is 1½" O.D. then ratio = 20

& remaining two ratios together must equal $\frac{1440}{20} = 72$.

Use 9:1 & 8:1.

If we would like to achieve a final accuracy of say 15" arc secs

in 20 mins of driving, then,

20 mins of time ≡ 5 × 60 × 60 arc secs.

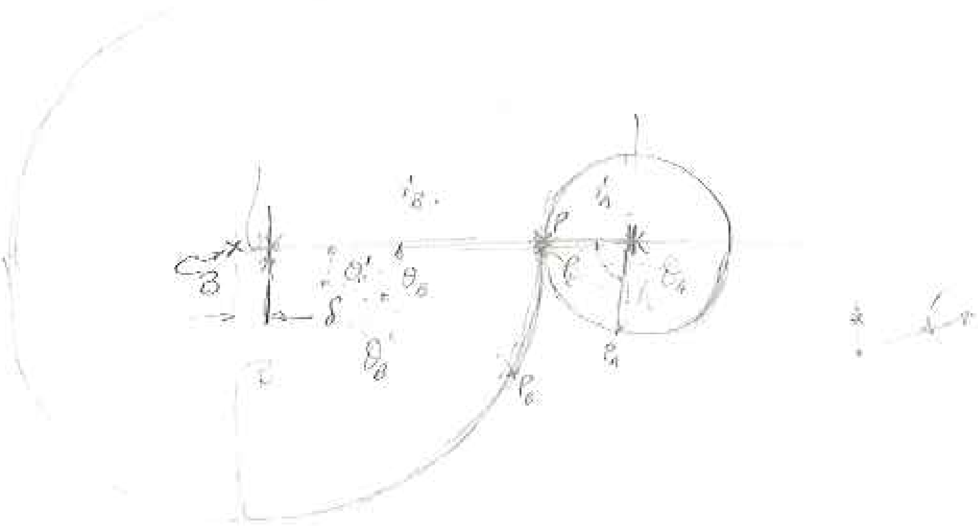
so accuracy reqd is 1 part in $\frac{5 \times 60 \times 60}{15}$ = 1200 i.e 1:1200

Contact Ron Arbour & arrange for visit to see his drive, which has very small drive roller.

Ring Crouzet motor & ask for catalogue.

1987 Jan 12.

(26)



Problem Two rollers, A & B each 'perfectly' round; A drives B around by friction only, no slipping occurs.

A is spring loaded onto B & (A) is constrained to move along original its original line of centres.

A is 'perfectly' centred to rotate about its centre.

B is not perfectly centred & has to rotate about an offset centre.

If A rotates with uniform angular vel. w.r.t. time how does B behave?

Let P = initial pt. of contact

After time t. wheel A has rotated from P to P_A

" B " " P " P_B.

new pt. of contact = P_t (P_t is on line joining centres of the two discs — and not their centres of rotation)

$\theta_A = \omega t.$

$\theta_B = \theta_B' + \theta_B''$

To a first a" $\theta_B = \frac{r_A}{r_B} \cdot \theta_A.$

$r_B \sin \theta_B'' = \delta \sin \theta_B' \approx \delta \sin \theta_B$

θ_B'' is small $\therefore \theta_B'' = \frac{\delta}{r_B} \sin \theta_B.$

will be a max" when $\theta_B = \frac{\pi}{2}$.

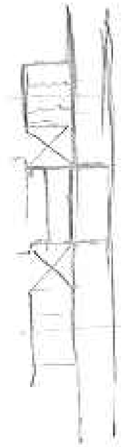
This represents the max" cumulative error in rads

In the ex. shown (as drawn) the error would be zero at the initial posⁿ & increase to a maxᵐ of $\frac{\delta}{r_B}$ rads after wheel B had rotated thro' $\pi/2$ rads & decreasing to zero again after B " " " π " from start.

If $\frac{\delta}{r} = \frac{\cdot 0005}{3} = \cdot 00017$ rads $\equiv \cdot 00017 \times 57.3 \times 3{,}600$ arc secs

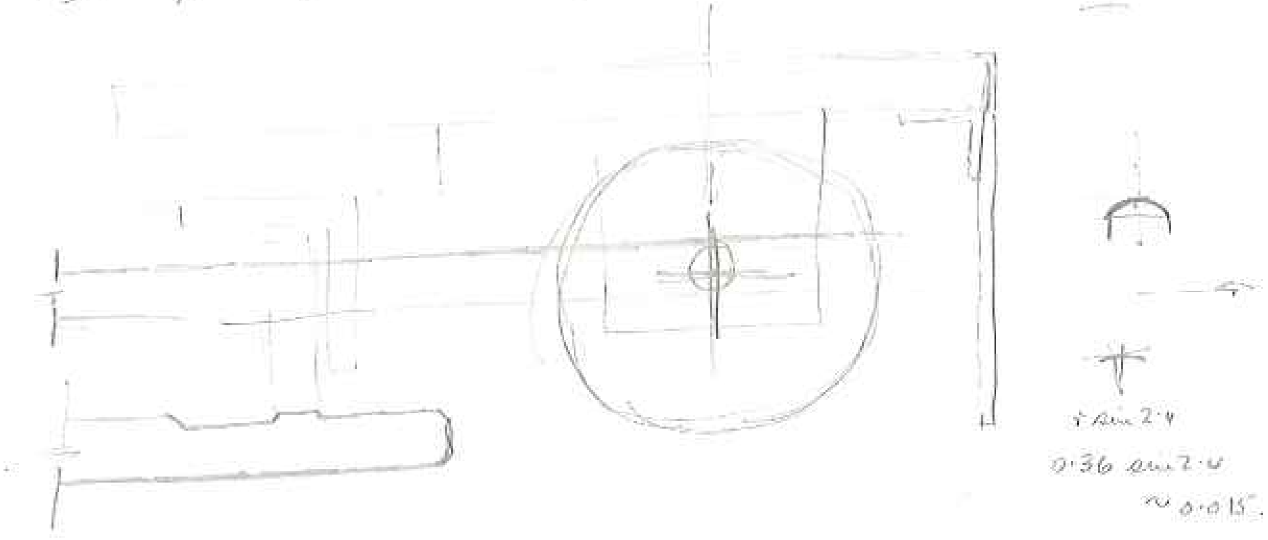
$= 35$ arc secs

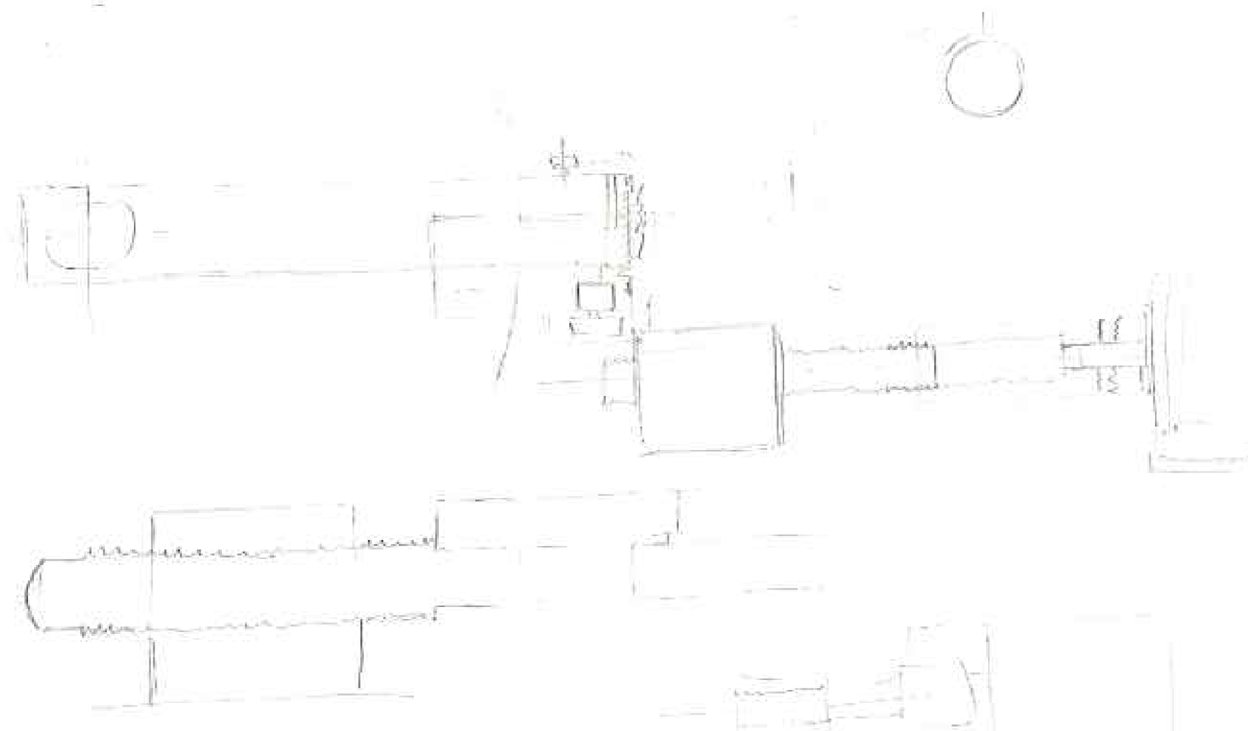
If 'beyond' this pair of wheels there was, for example, a further 10:1 reduction then the maxᵐ error of 35 secs would be reduced by a further factor of 10, i.e positional error would be 3.5 arc secs.



Alternative ideas for central clamp on polar table for imparting small changes in R.A. to fork.

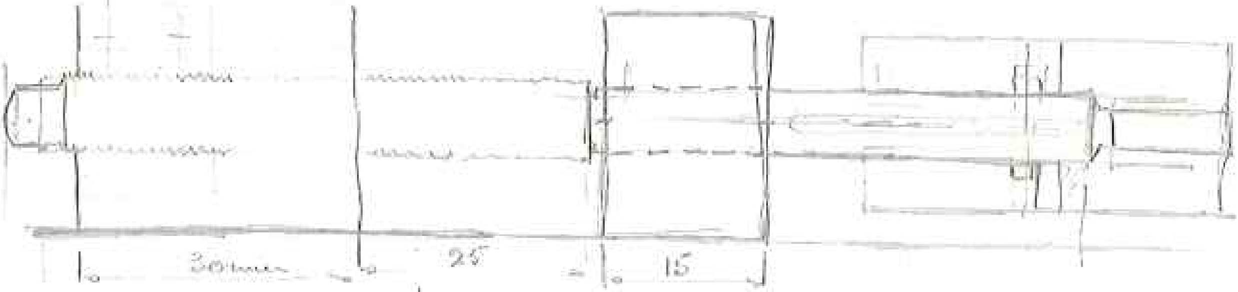
i.e. $17'' \times (1 - \cos 2.4°) = 0.015''$!!

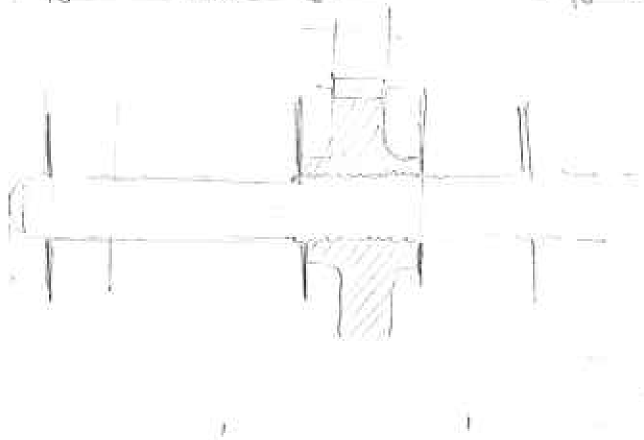


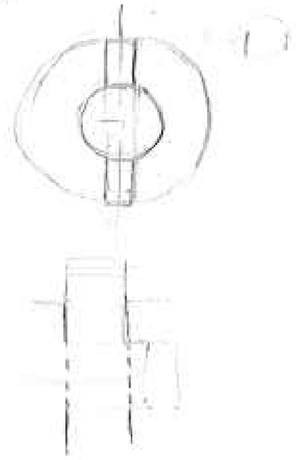


When moving an axis to change from star $\overset{*}{A}$ to $\overset{*}{B}$ the force between plunger and arm will be (Spring loading + Friction).
On returning from $\overset{*}{B}$ to $\overset{*}{A}$ the friction force will reverse & the force between plunger & arm will be (Spring loading − Friction).

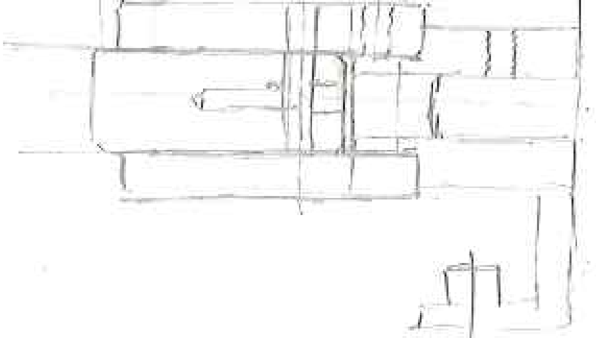
On 'returning' it might pay to overrun $\overset{*}{A}$ at first & then return so that the frictional force between plunger & arm stay more nearly constant. This may also be of some importance when re-centering on a chosen star following a complete cycle. It will also assist in taking up any 'slack' always in the same direction.

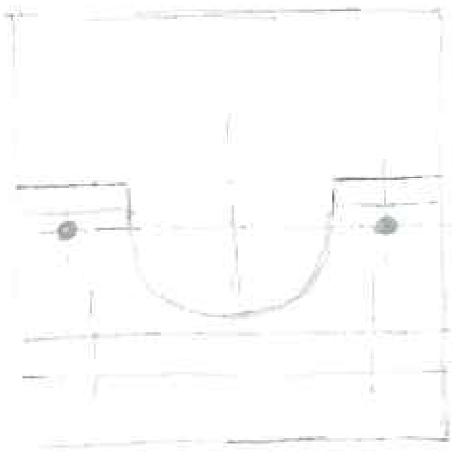






2" → 10 lbs
2½ → 12.5 lbs
1½" → 7.5 lbs



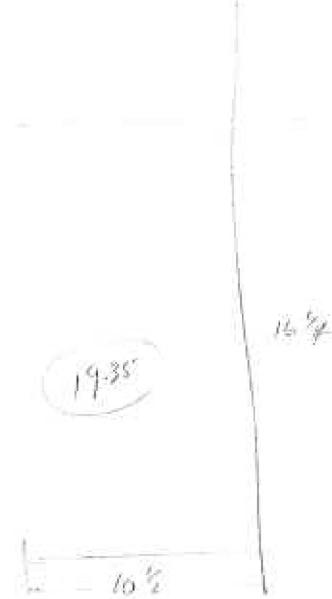


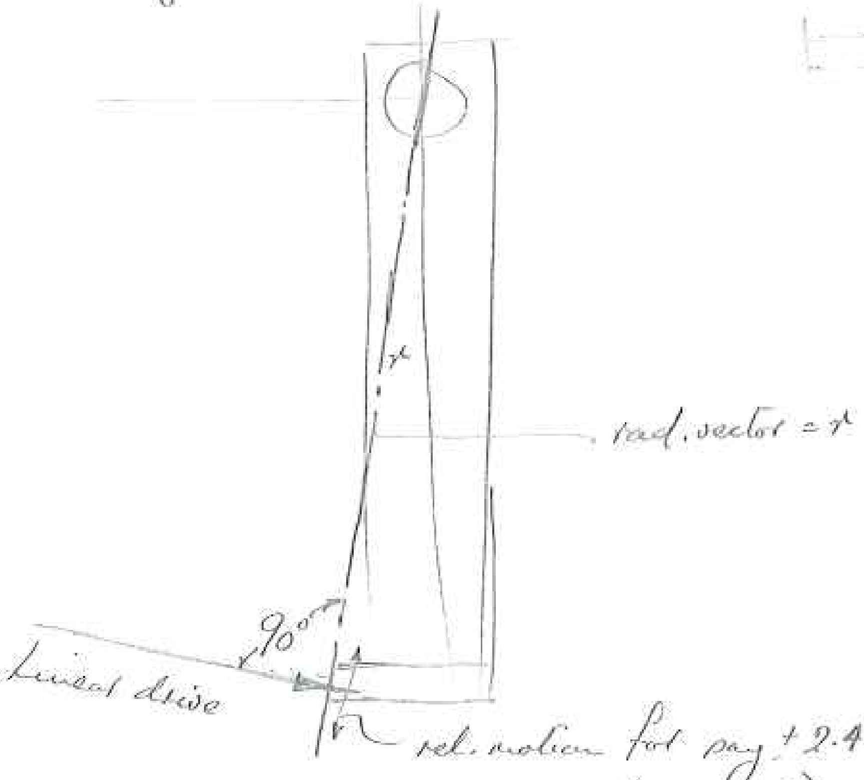
— Cable outlet on side.

1987 Jan 28.

Suggest changes to linear drive.

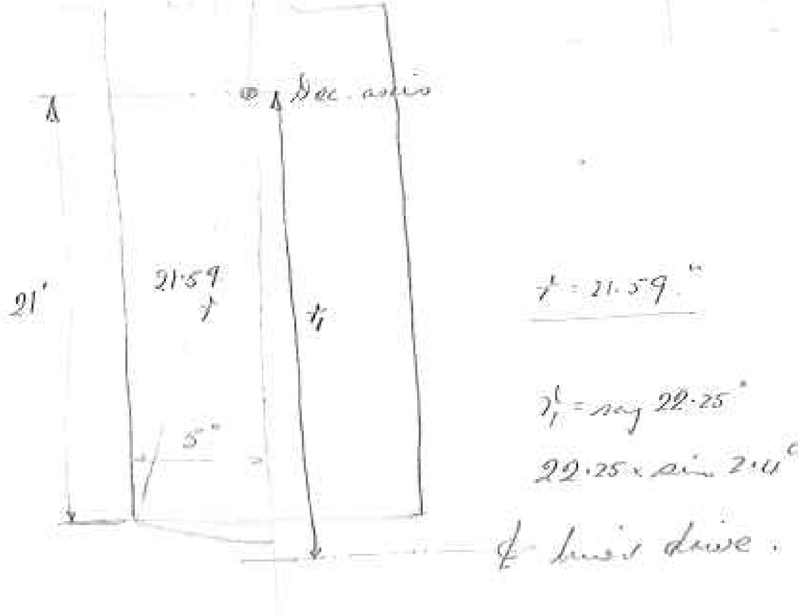
N.B. To ~~prevent~~ minimise relative motion between plunger at end of linear drive mechanism & 'arm' the point of action of the plunger must be ⊥'r to a rad. vector passing thro' the centre of rotation.





$= r(1 - \cos 2.4^\circ) \approx 0.015''$ for $r = 17''$

Posn of linear drive for changes in Dec.



N.B. for the changes in R.A. the max change in R.A. will be the same irrespective of the dec

i.e on the equator where Dec = 0° if max ch. in RA = ±2°

then at Dec = 60 " = ±1°

= 75 = ±0.5°.

It should be noted that the variable does not have to be at the mean posn

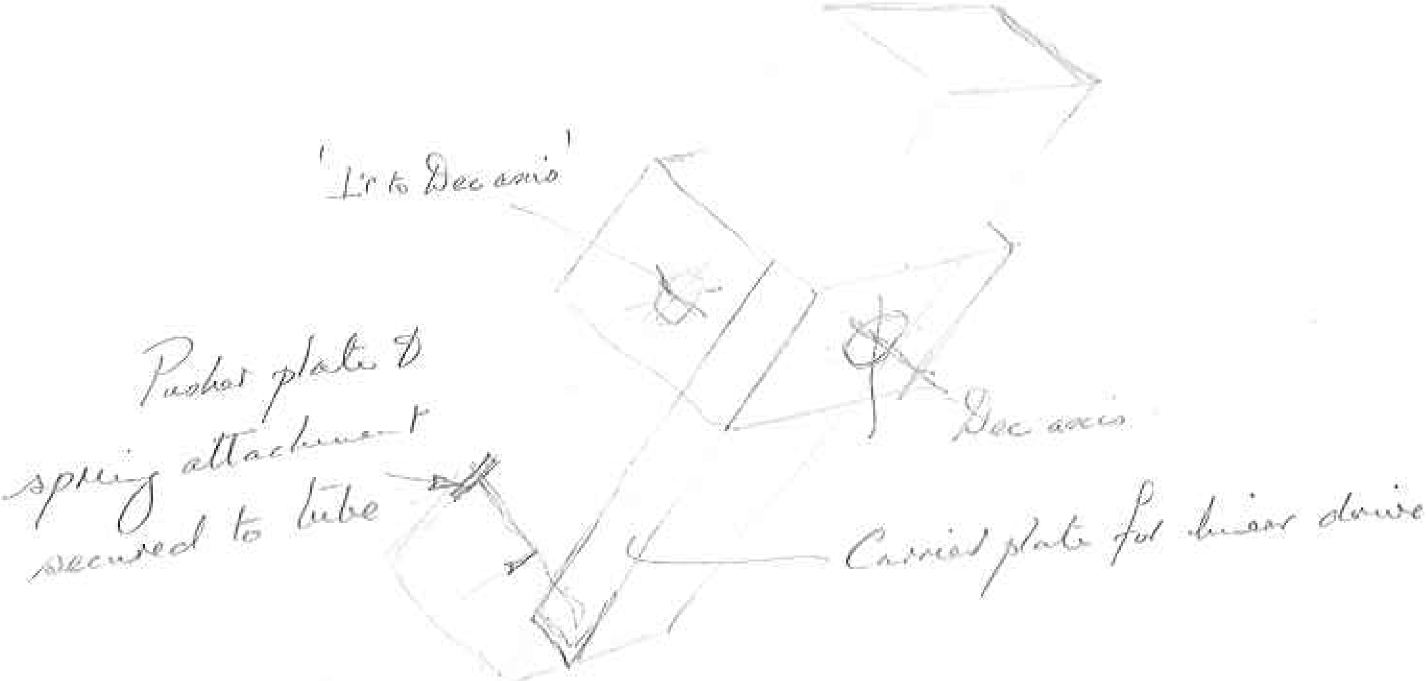
The mean posn of the linear drive could be set to the average of the two extreme stars.

It might be better to consider an alternate system where instead of small changes in (i) 'R.A' & 'Dec'. we can make small changes in (ii) 'Dec' & '⊥ to Dec'.

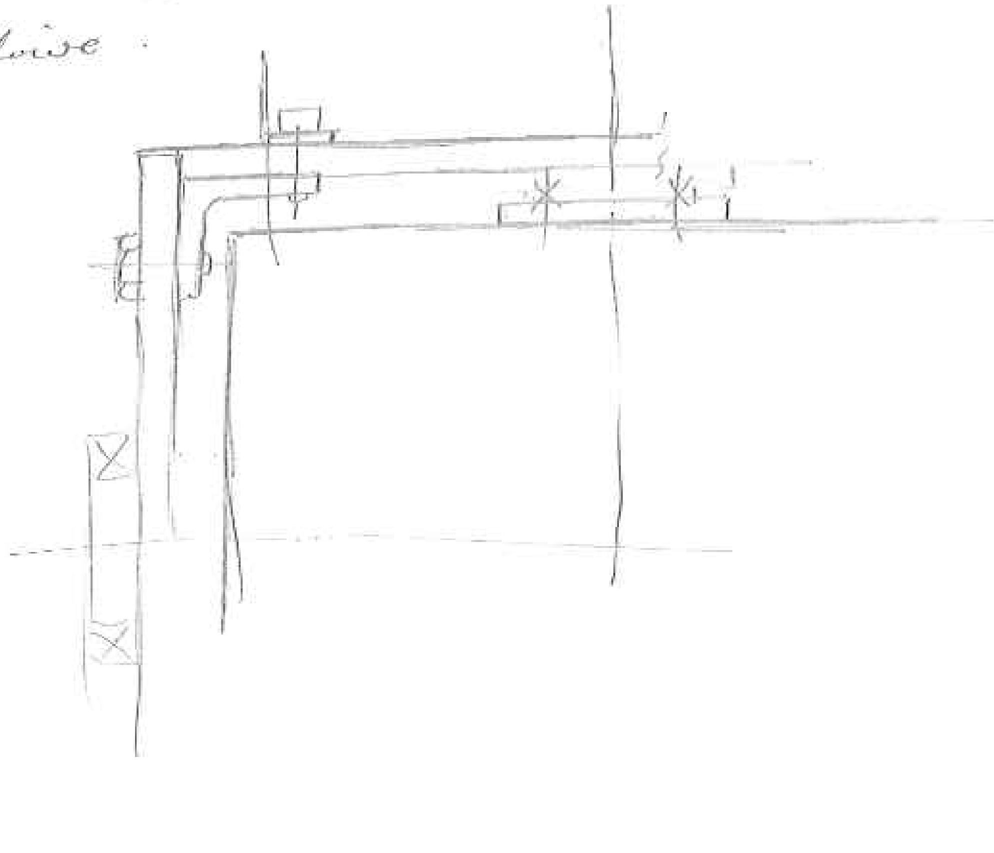
In this way say ±2.5° capability could be maintained over the whole sky.

This can probably be best achieved by a gimble type support for the telescope tube.

It might be worth considering a completely new tube for the telescope, made of dural. & incorporating a rotating flat holder.



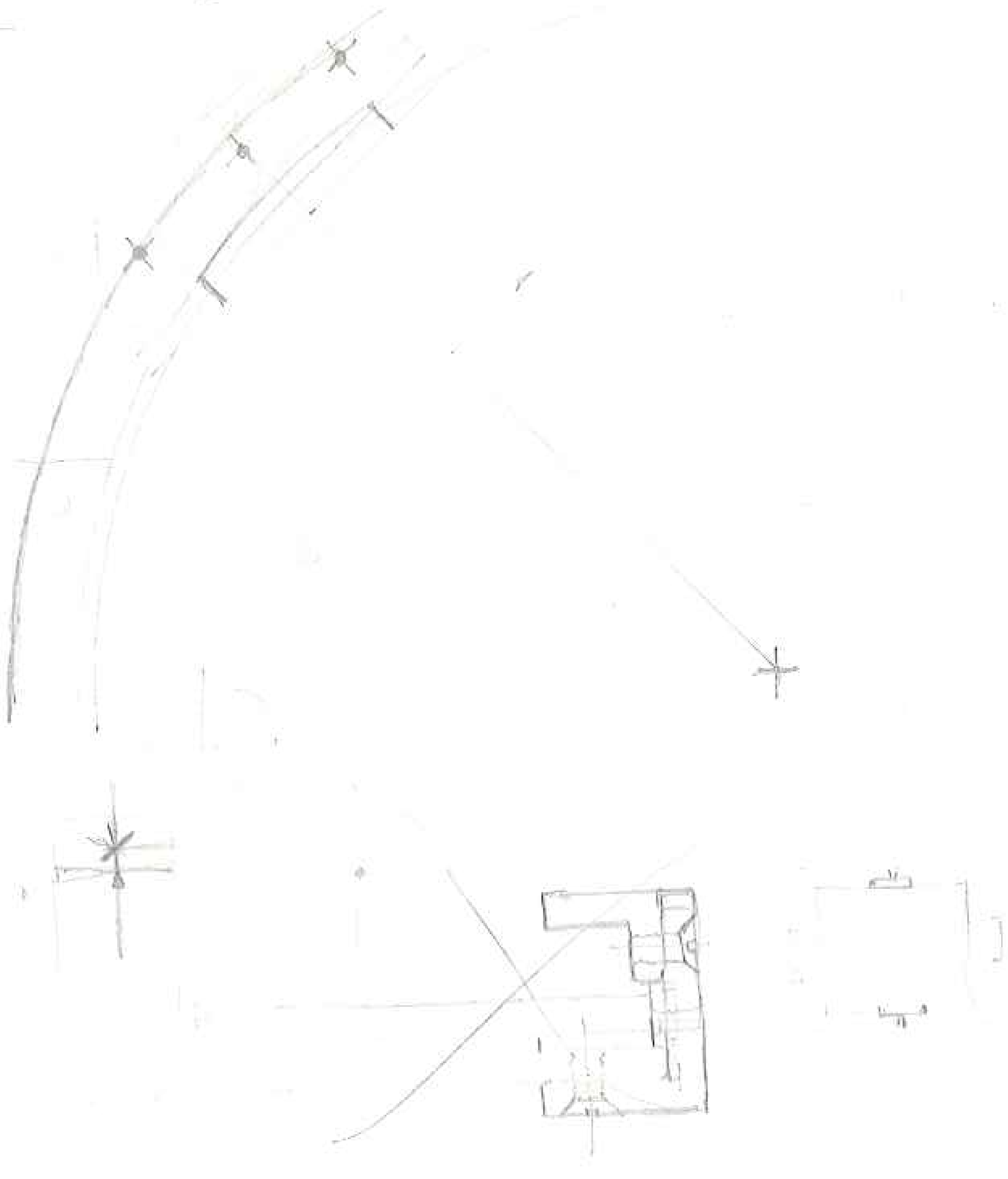
Linear drive.

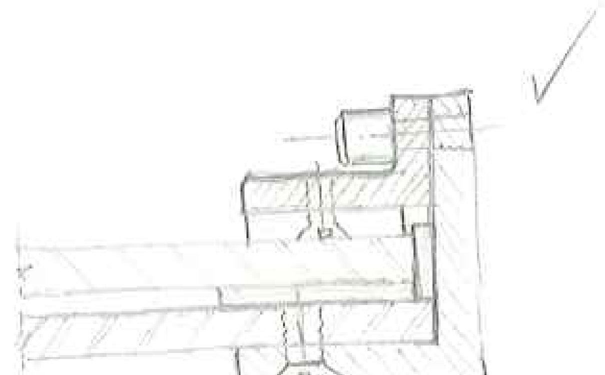


1987 Feb 2

(33)

New Tube - with rotating head





1987 Feb. 3

(34)

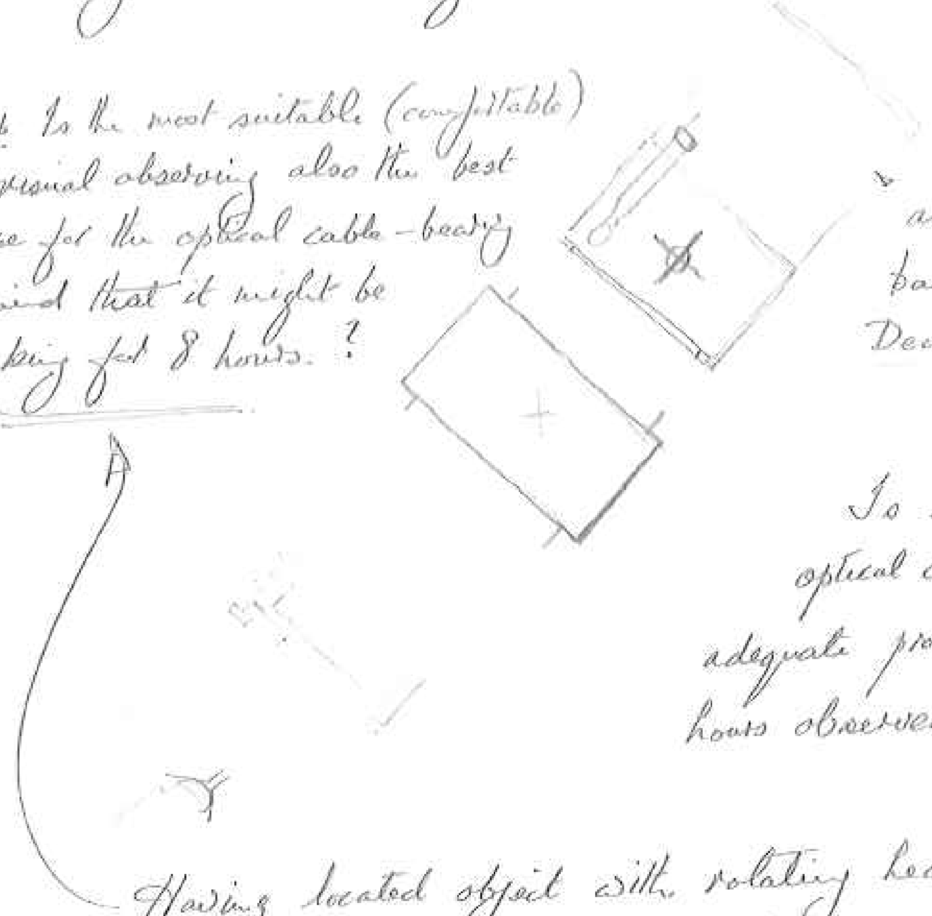
New telescope tube with rotating head

The rotating head will carry the diagonal, eyepiece, pin-hole & optical cable — also finder telescope.

Normally the rotation-axis of the head needs an adjustment so that it can be aligned with the optical axis of the main mirror.

Provided the tube can be initially made reasonably square (within say 5' arc) & accurate then the alignment of the two axes can be achieved by offsetting the main mirror a small amount (< 1/10") & tilting the mirror to align the two axes

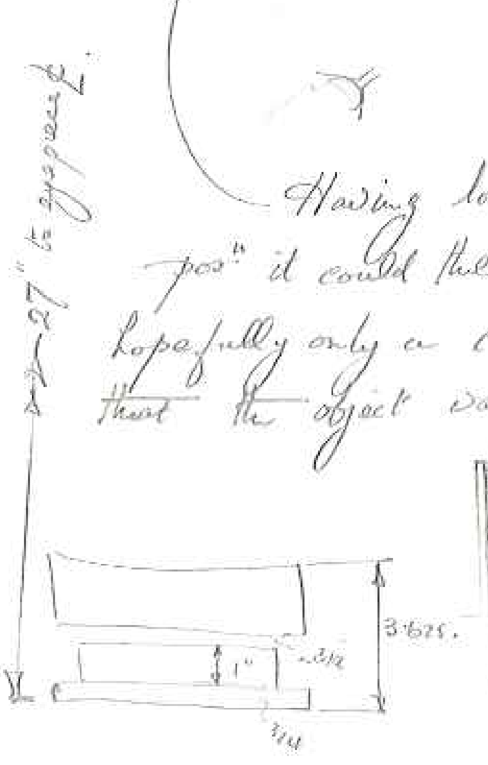
NB Is the most suitable (comfortable) for visual observing also the best place for the optical cable - bearing in mind that it might be tracking for 8 hours. ?

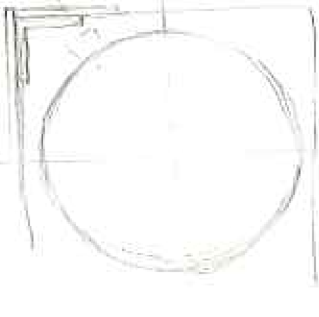


Dew-cap, may as well also be used to effect balancing about the Dec axis.

Is there a need for an optical cable support to ensure adequate protection during six hours observing?

Having located object with rotating head in most convenient posn it could then be rotated to the best position for the cable & hopefully only a cursory visual inspection would be reqd to ensure ~~that the object was still~~ in its correct position in the field of view.





s.g. glass 2.6
~ 8 lbs.

Mirror
8 15/32" ø
1 15/32" thick.

Wt. of wood + metal ~ 3 lbs
Wt. on bottom scales ~ 10 lbs

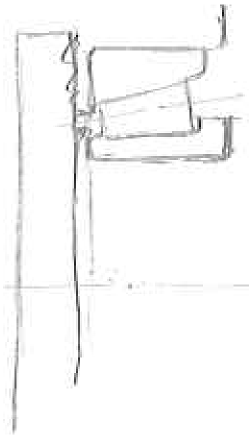
1987 Feb. 3.

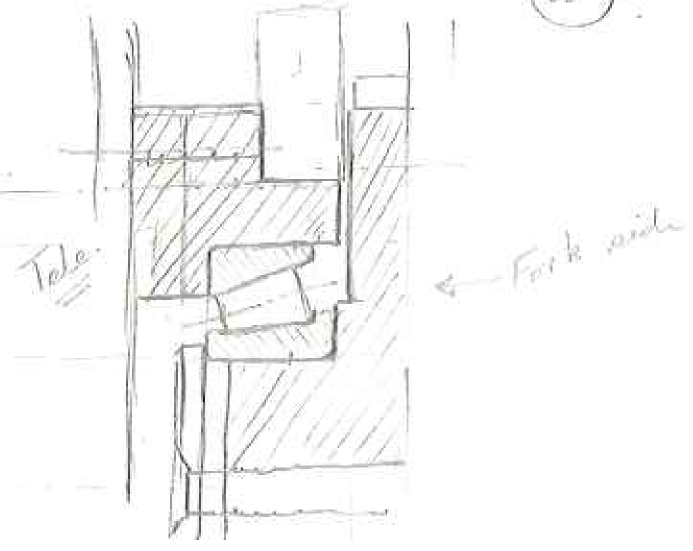
(35)

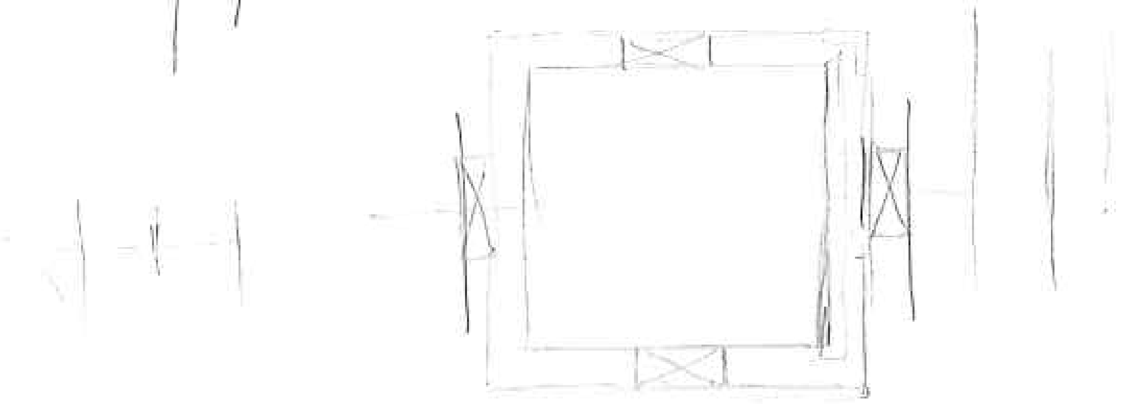
See Machinery Handbook p. 557

Taper roller bearing.

1.1875 bore 2.441 O.D. x width 0.63.







$r = \sqrt{16+25} = 6.403.$

$6.403 \cos 36^\circ = 5.1801"$

$-5"$

$= .18"$

∴ 0.25" clearance in gimbal is adequate.

If 1/4 th.L is used then 0.75 d. will need to be increased to 3/8"

$\delta = \frac{Wl^3}{3EI}$

$= \frac{10 \times 12^3}{3 \times 11 \times 10^6 \times 0.25 \frac{.5^3}{12}}$

$= \frac{1440}{81 \times 11 \times 10^6 \times 0.25}$

$= \frac{5760}{81 \times 11 \times 10^6} = \frac{5.760 \times 10^3}{.81 \times 1.1 \times 10^7}$

$= 6.46 \times 10^{-4}$

$= 7.3 \times 10^{-4}$

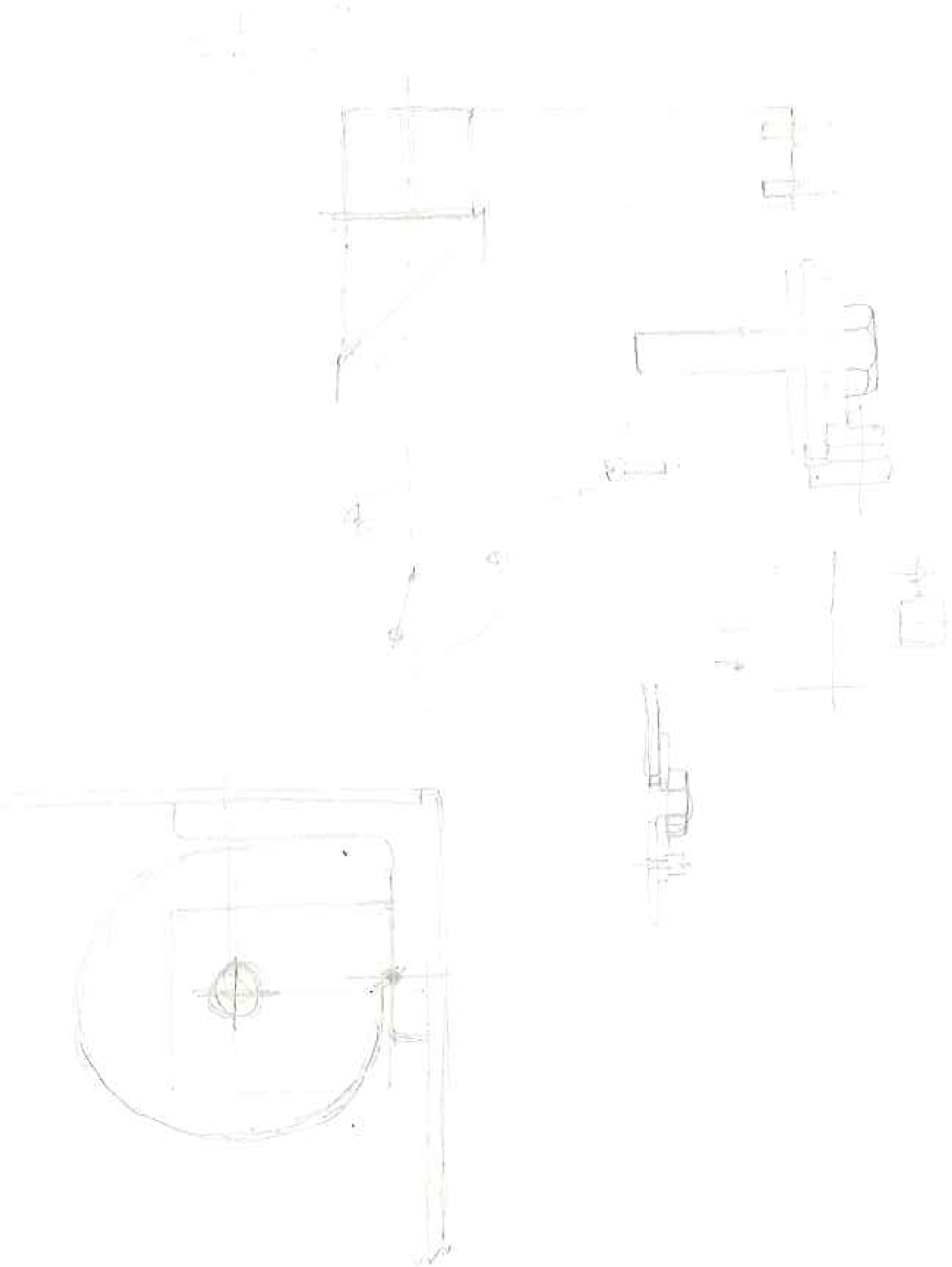
Using lighter tube — alternative

taper roller race

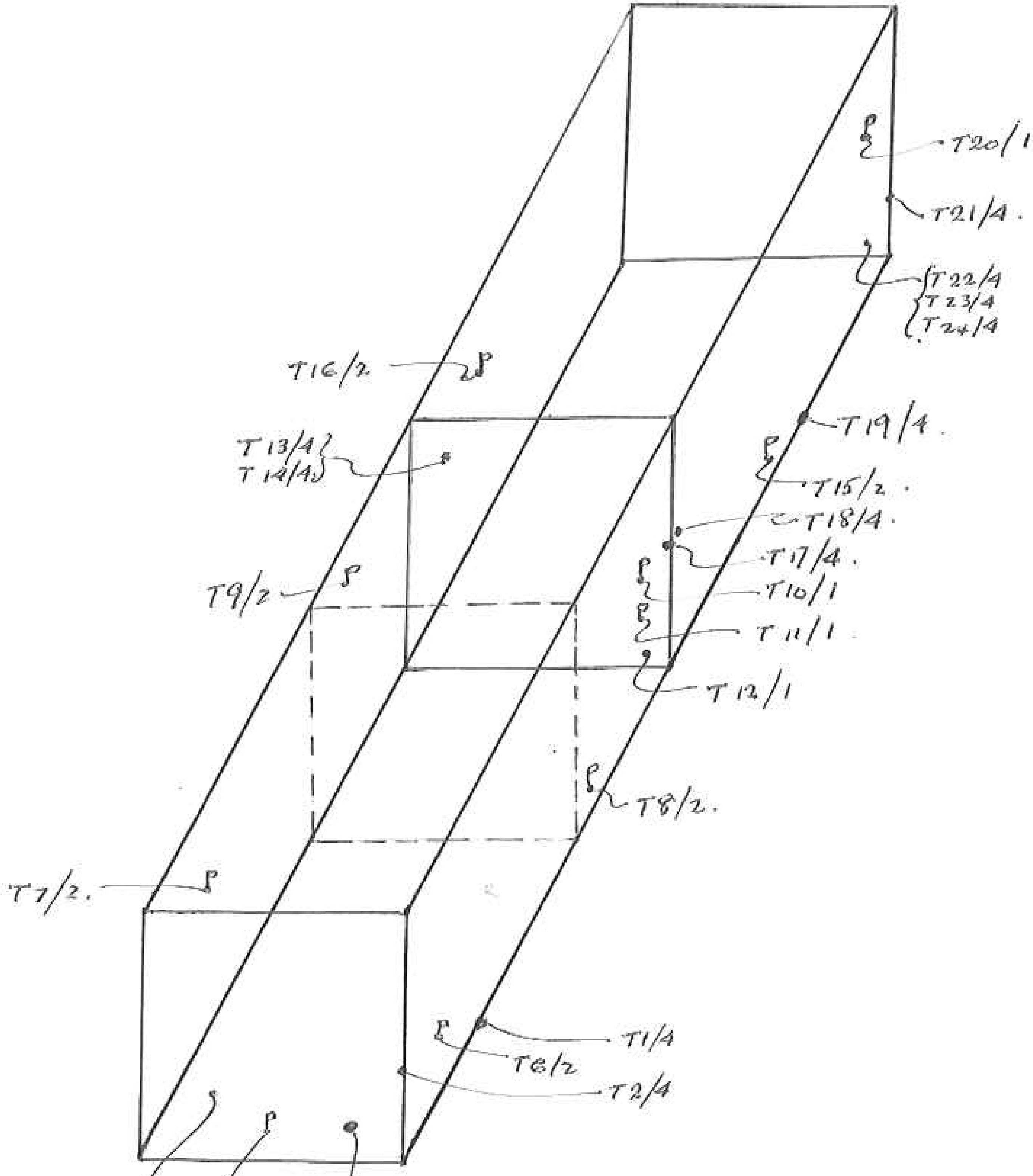
0.75" bore 1.8504 O.D. x width .5662

1987 Feb. 8.

36



1987 Feb 8.
(37)

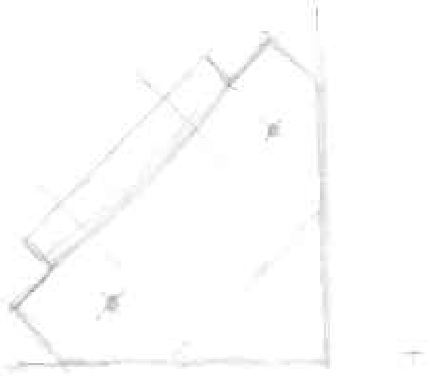


Extras → not yet listed.
- Spider
- Flat Holder.
- Eyepiece holder.
- PMH.
- Finder
- Counterweight

No.	No off.	Material	Approx size.	
T1	4	AA. 1½ × 1½ × ¼ L	1'-11" long.	
2	4	1¼ × 1¼ × 3/16 L	7½" "	
3	1	¼" plate	10" × 10"	
4	4	1½" × 1½" × ¼ L	1½" (clip.).	
5	4	"	" (clip.).	
6	2	1/10" plate	9 7/8" × 14"	
7	2	1/16" "	10" × 14"	
8	2	3/16" "	9 7/8" × 9	
9	2	3/16 "	10¼ × 9	
10	1	¼" "	[illegible] 10" × 10"	
11	1	¼ "	10 × 10	
12	1	1/8" P.T.F.E. (to make 8 small parts.		
13	4	1½ × 1½ × ¼ (CLIP) L	2"	
14	4	1½ × 1½ × ¼ (CLIP) L	1½.	
15	2	1/10" plate	9 7/8" × 13	
16	2	1/10" plate	10" × 13"	
17	4	1¼" × 1¼" × 3/16" L	7½"	
18	4	1¼ × 1¼ × 3/16 L	7½".	
19	4	1½ × 1½" × ¼"	13"	
20	1	¼" plate	10" × 10"	
21	4.	1¼" × 1¼" × 3/16" L.	7½".	
22	4	10 mm screwed rod steel	× 5"	Counterweight for mirror & other symmetrical loading about dec. axis.
23	4	2" Ø B.H.S.	2.87"	
24	4	1½ × 1½ × ¼"	2".	
25.	3	1" Ø × ¼" Thick	—	Mirror support washers.

1987 Feb. 8

Detail of retaining clip for rotating head.

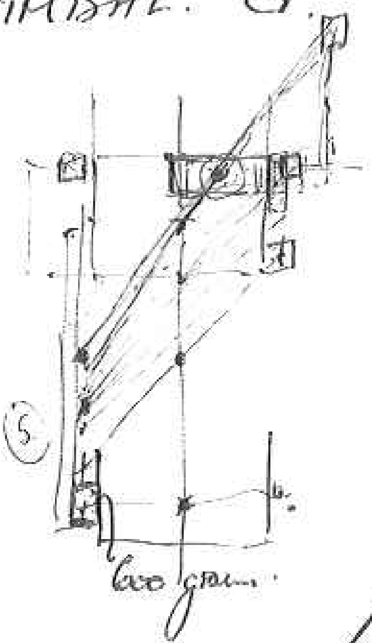


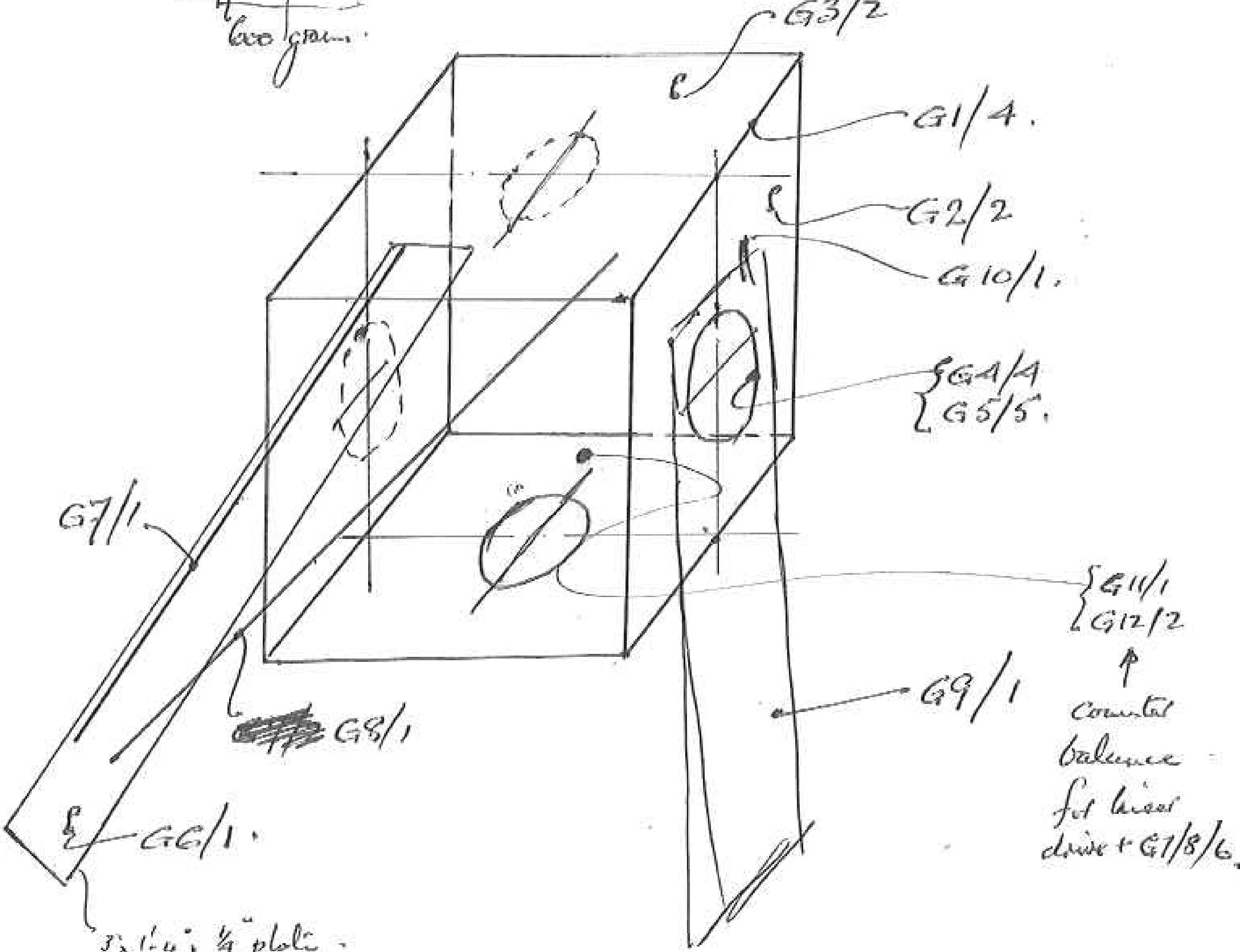
35A

1987 Feb. 9.

~~38~~ 39

GIMBAL. G.





$\frac{1}{2} \cdot \frac{2}{3} \times 4 = 1\frac{1}{3}$ lbs.

Timpken tapered roller race - 4 off.

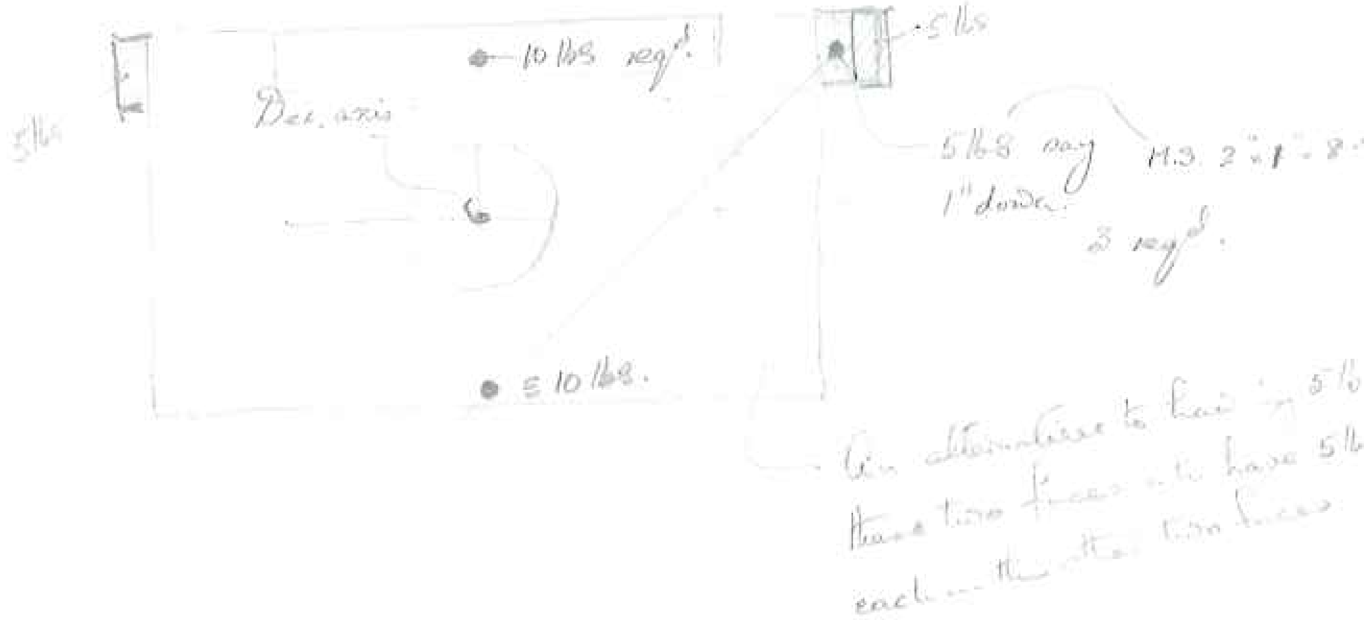
1.3
.9
.9
3.1.

1987 Feb 9

(10)

Problem.

How to c/balance linear drive (for the Dec axis) & its supporting system w.r.t. the dec. axis?



"pos" of linear drive + supporting system

~ wt. = 5 lbs

To make estimate of wt of tube (& counterweight reqd.).

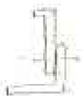
Spider

Assemblage 0.1
Mirror 0.12
Total say 0.3 lbs.
including nuts, fixing screws etc.

{ Eye piece holder. — say 0.2 lbs.
{ P. M. H. — say 2.0
2.2 + 2.2 for cwt = 4.4 lbs.

Finder 3 lbs + say 1 lb. for support = 4 lbs + 4 lbs for cwt. = 8.0 lbs.

1987 Feb 10 (42)

Tube weights:		lbs	ins		lbs in.		lbs ins
Item.		wt	$\bar{x}$		C.W.		A.C.W.
Spider + flat + flat holder.	✓	0.8	8.5	✓	15.3 ×		
Eyepiece holder + PMH + c.w.	? {	4.4	7.0	✓	30.8		
Finder + support + c.w.		8.0	10.5	✓	84.0		
T1/4.	×	0.86	6.75			✓	5.8
T2/4	×	1.5	18.2			✓	27.2
T3/1 	×	2.78	18.2			✓	50.6
T4/4 & T5/4	×	·8	17.8			✓	14.2
T6/2 & T7/2.	✓	3.89	11.5			✓	44.7
T8/2 & T9/2 ?		1.88	0				
T10/1 & T11/1	×	2.22	4.6	✓	10.2		
T12/PTFE	✓	·1	4.6.	✓	·5		
T13/4 & T14/4 	×	0.8	5.0	✓	4.0		
T15/2 & T16/2 Add stiffener?	×	5.78	11.5	✓	66.5		
T17/2 & T18/2	×	2.8	5.6.	✓	15.7		
T19/4	×	3.9	12.5	✓	48.8		
T20/1	×	1.11	17.8	✓	19.8		
T21/4	×	1.4	18.0	✓	25.2		
T22/4		·75.	16.0	✓	12.0		
T24/4		·6	14.0	✓	8.4		
Mirror	·	8.0	17.0.			✓	136
					341.2.		278.5
							62.7

T23 ?

∴ an ACW of 62.7 is reqd. for balance.
if $\bar{x}$ = 16.5" wt reqd = 3.8 lbs.
if two wts in diagonally opposite corners. each = 1.9 lbs.
1.75" ϕ (steel) × 2.82" long.

Wt = 54.3 lbs.
+ say 4 lbs for nuts, bolts etc

say 59/60 lbs total.

NB 3" ϕ steel = 1.98 lbs/in ~ 32 lbs Al. alloy

GIMBAL.

1987 Feb 10.

(45).

To estimate wt. of gimbal.

Assume each gimbal bearing is ≡ to 3¼" ϕ disc die × 1½" thick Al All.
= 0.92. say 1.0 lb with bolts etc.

Wt of linear drive. Motor (stepper). 1.3 lbs.

	Matl			
L1	Steel	·53	(2" × ¾" × 1¼") use 1¼" × 5/8 × 1" (1.875 / 0.78)	
L2	Brass	·26.	" " 3/8" ϕ × 5"	·15.
L3	Steel	·53	(2 × ¾" × 1¼") " 1¼ × 5/8 × 1"	
L4	Brass	·38		
L5	Steel	·01		
L6	Brass	·2		
L7	Al All	·5	(2 × 3/8" × 6¼) use (1¼ × ¼ × 6.)	·2.
L8	"	·1		
L9.	"	·1		
		2·6.		

Total 3·9 lbs With return spring say 4·1 lbs

Support system G6/7/8. 3·1 lbs. Total 7·2 lbs !!

2 × 2 × ¼ A.A. = 1·24 lbs/Pr.

G1/4 = 4 × 1·24 = 5 lbs.

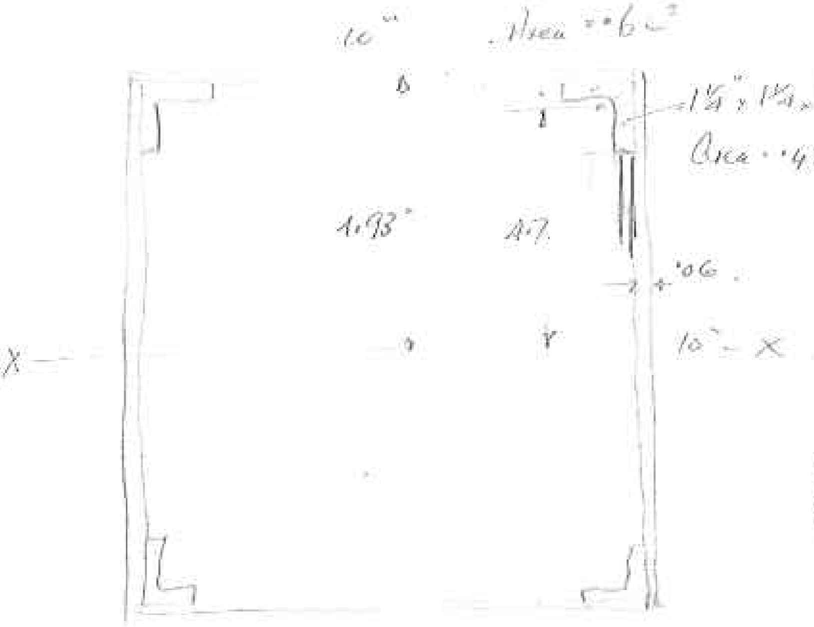
G2/2 & G3/2 = 18·8 lbs

C.W. for linear drive & support system ~ 22 lbs.

Total wt gimbal = 7·2
5·0
18·8
22·
Bolting 4
Bearing 4
61·0

At this stage a review is necessary in an endeavour to reduce weights. !!!

To assess bending of tube due to wt. of mirror etc.



$I_{xx} = 2 \times 0.6 \times 4.93^2 +$

$4 \times 0.4336 \times 4.7^2 +$

$2 \times \frac{0.06 \times 9.875^3}{12}$

$= 29.17 + 38.31 + 9.63$

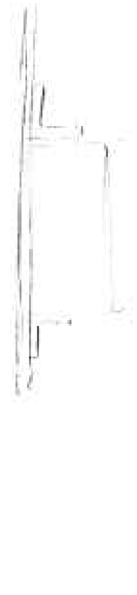
$= 77.11 \text{ in}^4$

$$\delta = \frac{WL^3}{3EI} = \frac{10 \times 1.5^3 \times 10^3}{3 \times 10^7 \times 7.711 \times 10} = \frac{2.25 \times 10^{-4}}{2 \times 7.711} = 1.46 \times 10^{-5}.$$

1" x 1" x 3/16"   ·182 Kg/ft = 0.4 lbs/ft.

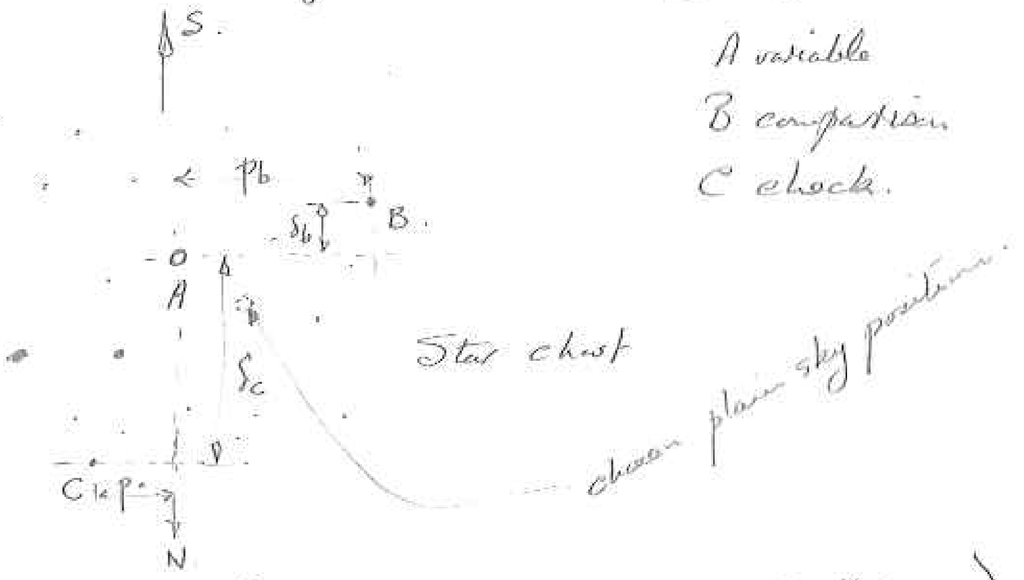
1¼ x 1¼ x 3/16.   = 0.51 lbs/ft.

Top end of tube could be re-designed to make it lighter – this will make slightly more tedious & complicated to build but will reduce wt. If total wt of tube can be reduced to 10 lbs then size of tapered roller races could also be reduced slightly



Modus operandi

(B). Rotating head carrying finder & P.M.H.
Finder & P.M.H. on opposite sides of telescope tube in order to reduce amount of counterbalancing reqd.



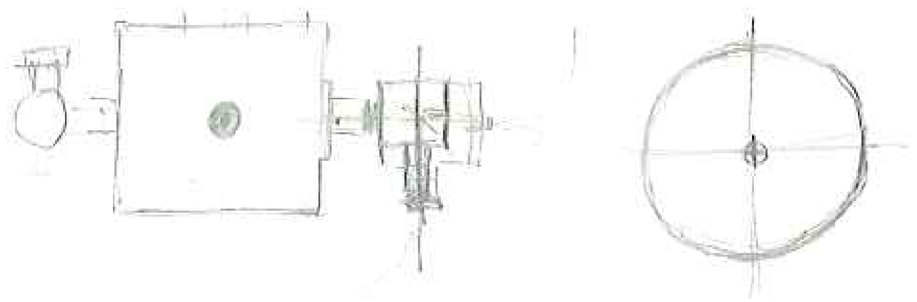
Assume in this case - (which will likely be true 95% of cases) that we will put A in centre of field with linear drives in mean posn.

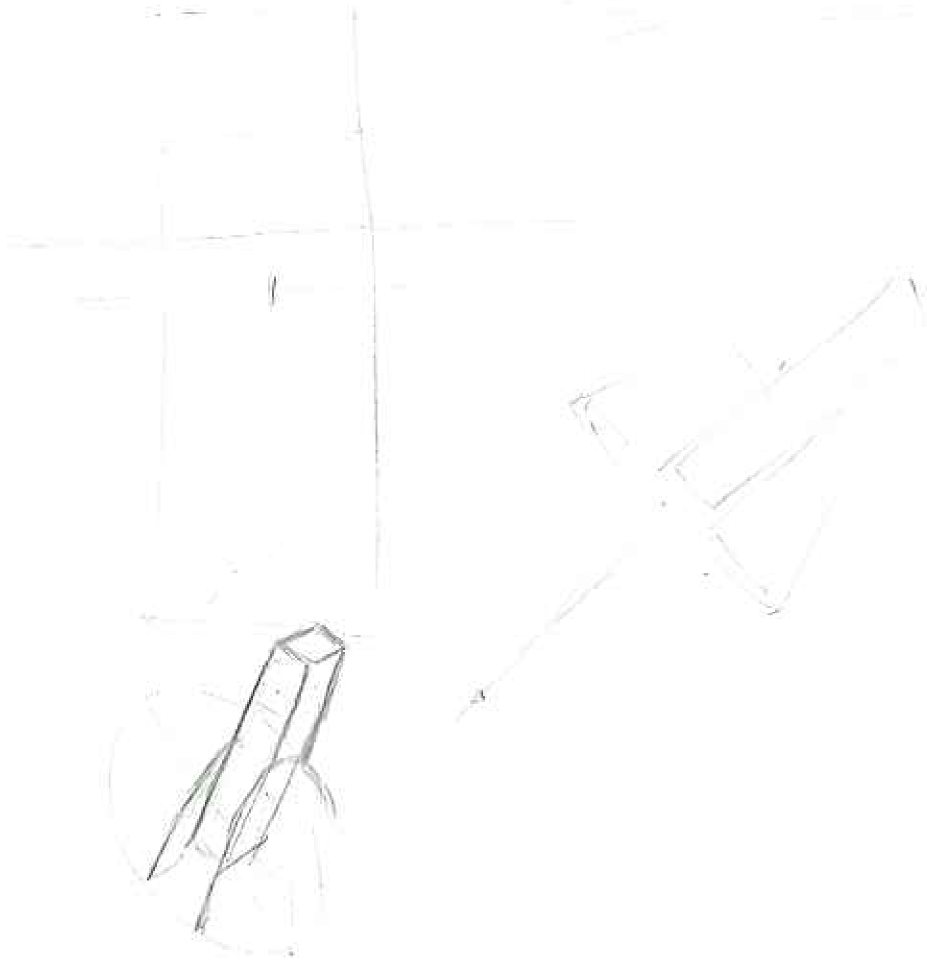
Sequence of setting up

(a) P.A. in 'free' posn, i.e. not being driven.
(b) Dec. axis clamp in 'free' position.
(c) Will most likely need a Dec axis friction clamp in action to prevent ~~Dec~~ telescope moving too freely in dec.
(d) Rotate head to put finder in most convenient posn for locating Star A.
(e) δ_b, δ_c, p_b, & p_c will be known to a reasonable degree of accuracy
(f) Locate A in finder
(g) Start P.A drive., Lock dec & release friction clamp
(h) Rotate head to give desired posn for PMH bearing in mind the need to drive for 6 hrs automatically without fouling cables or the optical light guide
(i) Check & finalise steps reqd. for B, C & sky posn

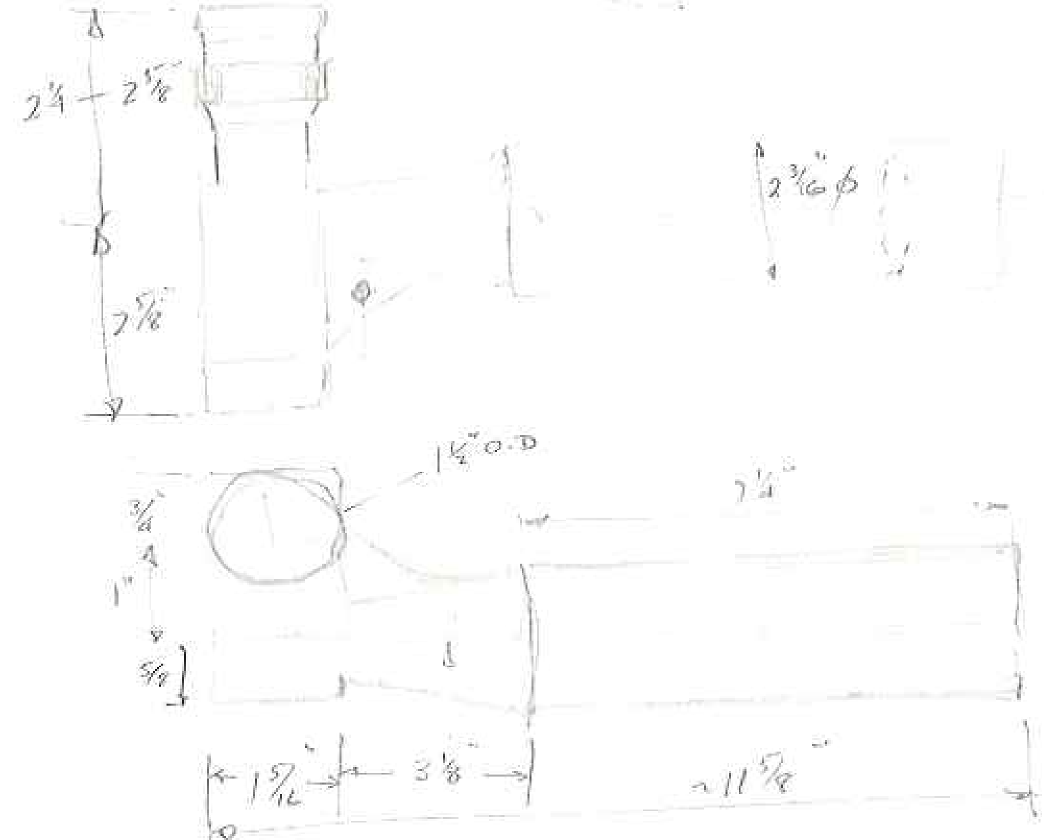
1987 Feb. 13

(46)





Elbow telescope



1987 Feb. 14

(47)

Suppose we use 12" lever arm instead 14".

Travel reqd. for $\pm 2.5°$ $= \pm 12 \sin 2.5°$ $\approx \pm = 0.5234$
± 0.625"

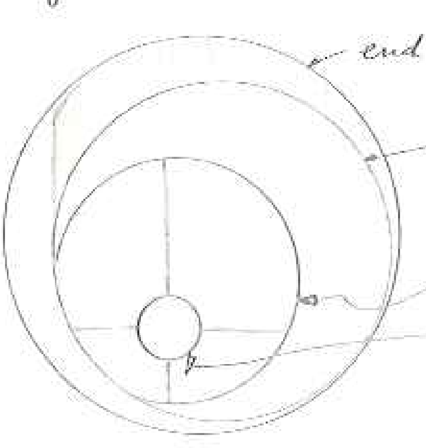
If we still fallow on the basis drive the $12 \sin x° = 0.625$
$x = 2.986°$

i.e. allows $0.486°$ for any initial misalignment or any drive inaccuracies plus P.A. misalignment.

Feb. 18.

With a rotating tube head, which carries the flat, eyepiece, finder of P.M.H how do we carry out the optical collimation so that when the head is rotated we still have a good stellar image that stays in the centre of the field.

"Amateur Astronomers Handbook" by J.B. Sidgwick, p 182 gives the necessary procedure for a simple (non-rotating head) Newtonian.



If the elements of head can be collimated separately from the main mirror then this would clearly be a simplification.

Let axis of primary mirror be A_p

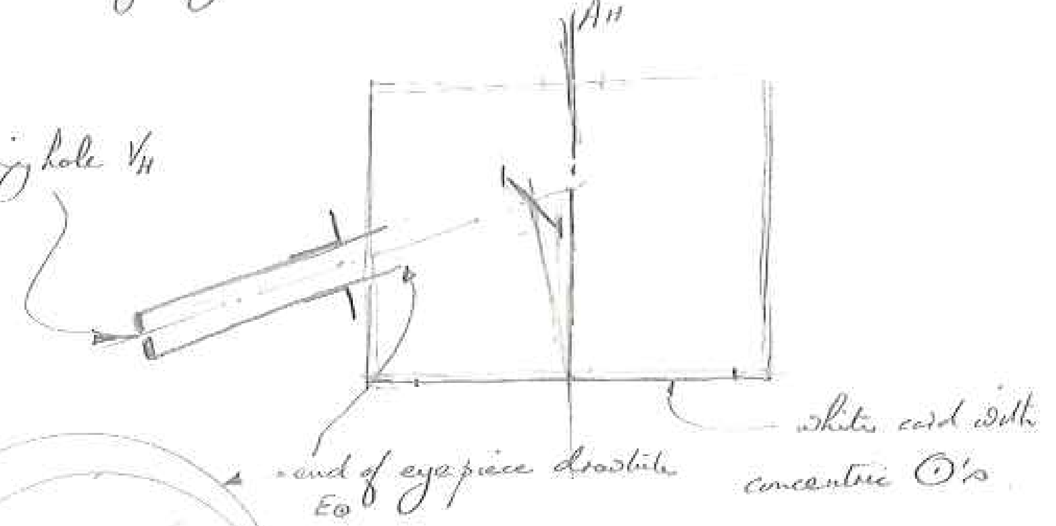
If axis

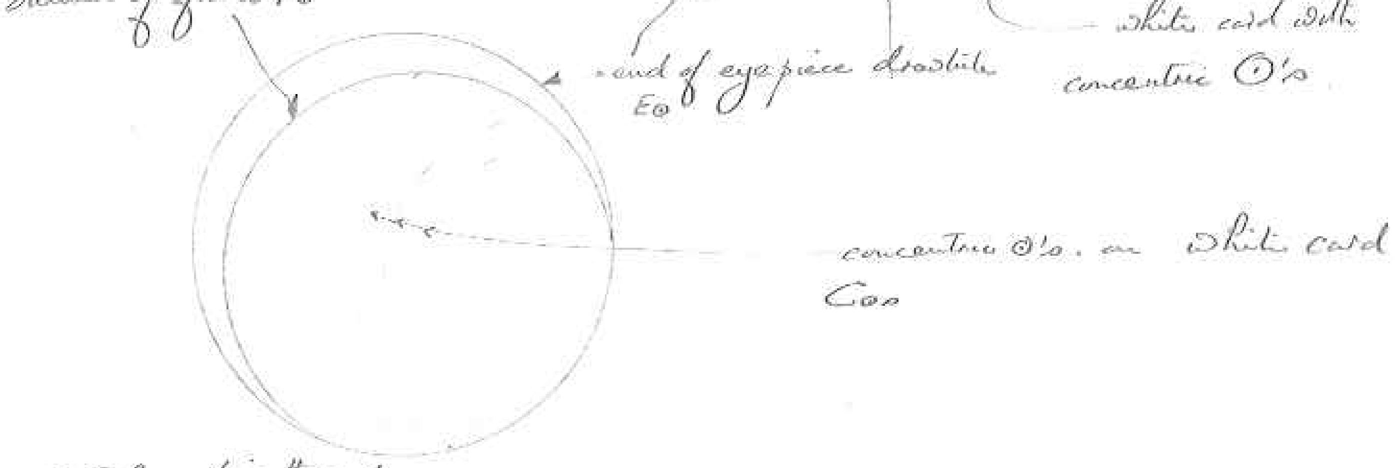
A_H & A_p need to coincide

A_D needs to intersect A_H — but since we are primarily concerned with the central star image this does not have to be at right angles. A small amount of field rotation is of no concern.

The 'base' of the rotating head will have a truly ⊙r opening ~ 1/8" ϕ greater than the primary mirror

Suppose we make up a piece of white card with a set of concentric rings carefully drawn on it





initial View thro' V_H

Options

1/. Move spider to make $E_\odot$ & $F_\odot$ concentric
(in general $C_{\odot s}$ will move but not be concentric.
2/. Rock spider to make $C_{\odot s}$ concentric ($E_\odot$ & $F_\odot$ should remain concentric.)

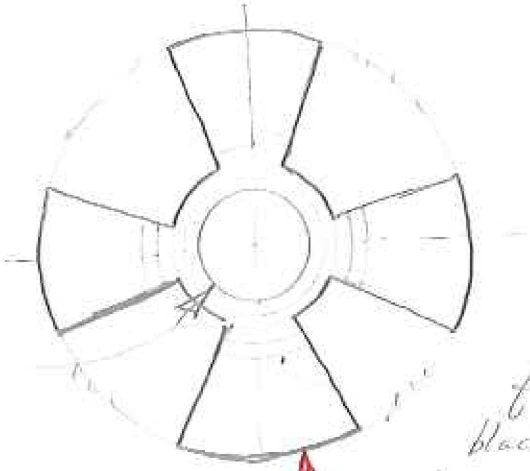
Feb 19.

This will not achieve very much. — try again"

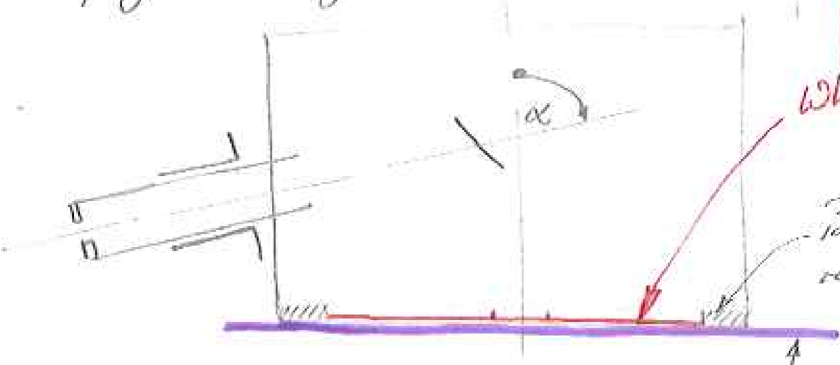
By its method of construction the axis of the rotating head - A_{11} - will be ⊥r to the bottom plate & will also pass exactly thro' the centre of the bottom hole.

White card cut-out reqd to fit bottom plate of rotating head.

Central hole ~~slit~~ slightly larger than projected ⊙ of flat.

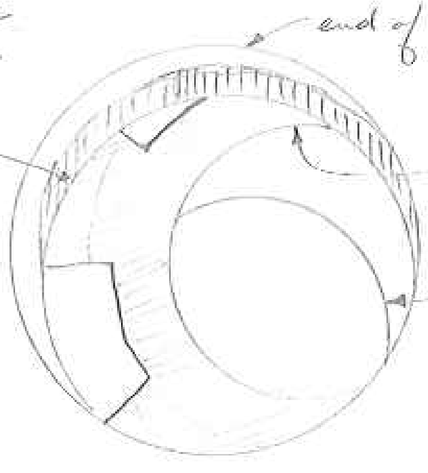


Set-up.



Remember - there is no real need for α to be exactly 90°. The quality of manufacture will probably ensure that it is better than ±0.5°.

Before adjustment

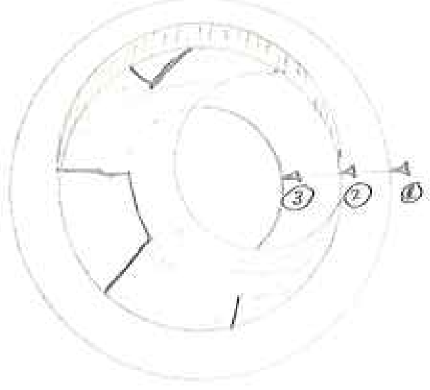


1/. Ignore. C⊙ and F⊙
Use spider adjustments to make R⊙ & E⊙ concentric, F⊙ should automatically become concentric! It might be easier at this stage not to have the card in posn

1987 Feb. 22

(50)

View after stage 1.



Concentric

① End of eye-piece drawtube E⊙
② Projected outline of elliptical flat F⊙
③ reflection of F⊙ from household mirror R⊙

2/. Use adjustment to 'rock' the flat & make the central hole of the white card concentric with the other 3 ⊙'s.

N.B. If the position (concentricity) of F⊙ becomes slightly shifted it may be necessary to repeat stage 1 & then stage 2.

It is felt that two iterations will be ~~so~~ sufficient to achieve adequate alignment.

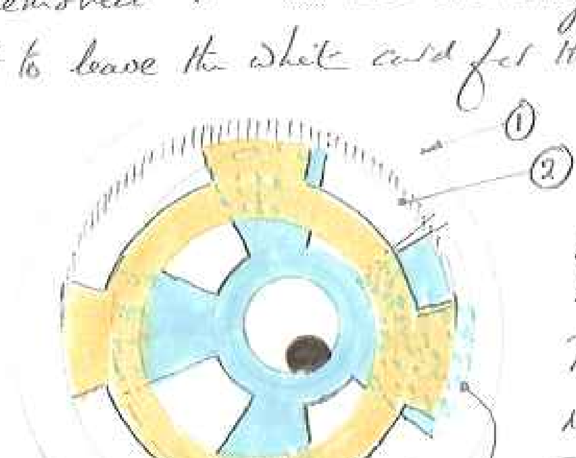
At this stage the flat ~~sh~~ will be on the axis of rotation of the head & in correct alignment with the eye-piece drawtube.

Prepare similar white card, again with central hole slightly larger than projected ⊙ of flat.

Mount the rotating head on telescope tube. N.b. household has obviously been removed & ∴ R⊙ ~~has~~ no longer exists!.

It is probably best to leave the white card for the rotating head in posn.

Before adjustment of main mirror.



Yellow is white card on rotating head
Blue is white card on main mirror
Small black ⊙ is image of flat as seen reflected from main mirror

④ is outside ⌀ of main mirror.

Step 3

[N.b. Outside of ~~main mirror, (4), shot~~

Use lateral positional adjustments on main mirror to make (4) concentric with (1) & (2).

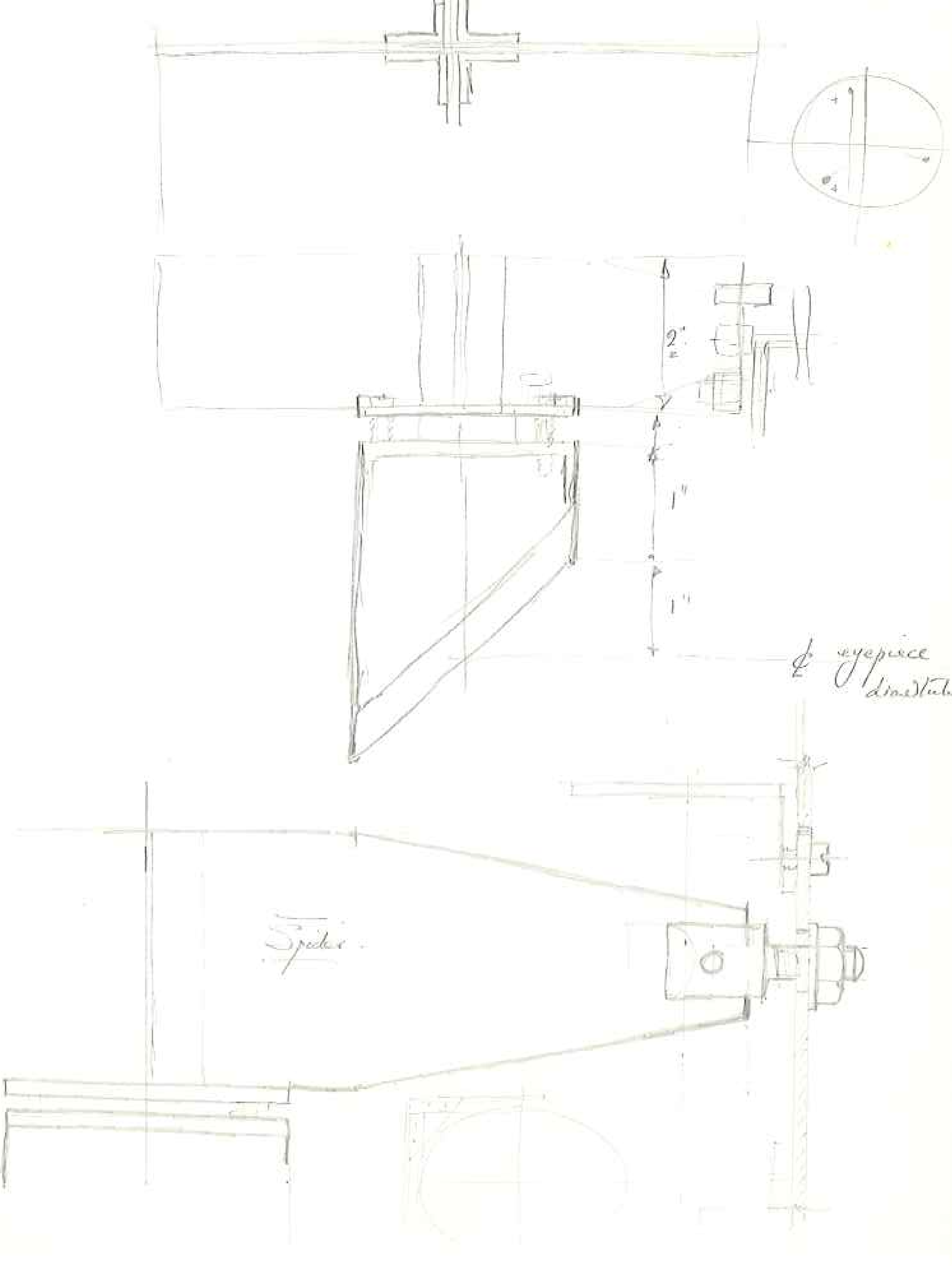
[N.b. (4) should be entirely within (2) to ensure that the whole cone of light from a star reflected by the main mirror is collected by the eyepiece. If (4) > (2) then the flat isn't big enough !!!

Step 4

Use 'rocking' adjustment on main mirror to bring small black ○ (5) into centre of hole in (blue) card for main mirror.

The tube & its optical elements should then be correctly aligned.

1987 Feb 23.



1987 April 2 (53).

If a stepper motor could be used to drive the P.A. in a fairly conventional way what are the pros. & cons. when compared with current scheme of using a synchronous motor for the P.A. & two stepper motors for each of the axes of a trunnion bearing?

Seq. of shots Sky B A ~~Sky B~~ C Sky B.

20 5 ½ 10
40 40

S B S A S B S C S B S.

30" ø disc

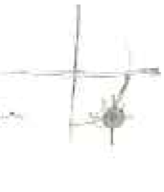
Using stepper motor type 23 (with RS drive board 332-098)

Use say 10 full steps/sec for sidereal tracking rate.

Sidereal rate = 15 arc secs × $\frac{1440}{1436}$ arc secs/sect.

10 steps/sect = 18°/sec ≡ 18 × 3600 arc secs/sect.

∴ total turn ratio = $\frac{18 \times 3600}{15 \times} \times \frac{1436}{1440}$ = 4332.0333.



If final drive roller is 1" ø then gear reduction reqd from ~~[illegible]~~ motor to drive roller = $\frac{4332.0333}{30}$ = 144.40111

Estimate of inertia about P.A.

≈ say 100 lbs at 10" rad.

= 100 × 454 × $(25.4)^2$ = ~~1.1532 × 10 gm cm~~ 29.29×10^6 gm cm^2.

11560
110..17"

(4.5-1) . 1.75.

If η of drive is say 80%

then $I_{equiv} = \frac{29.29 \times 10^6}{.8 \times 4332^2}$ = 1.95 gm cm^2

When driving at 100× sidereal rate $I_e = 1.95 \times 10^4$!!!

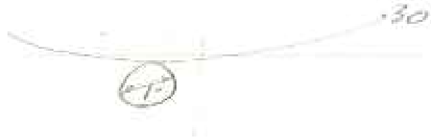
1987 April 21.

~~53~~. 53A

144·4011

Alternate gear ratios

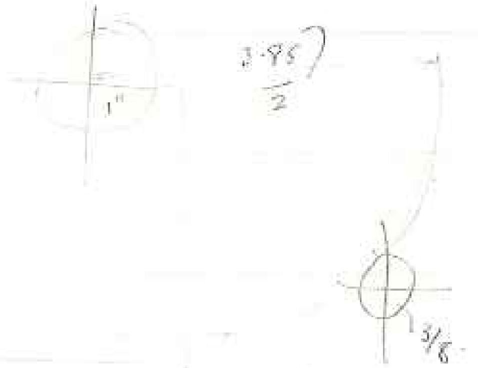
$\frac{1}{4}$: 3½.



$\frac{144 \cdot 401''}{14}$ × ·375 = 3·857.

——— " ———

$\frac{3 \cdot 857}{2}$ − 0·5 = 1·4285.



1·4285 + ·1875 = 1·616
× 2 = 3·232

No good.

Say 4" ϕ. $\frac{4}{3/8}$ = 10·66666. = 13·5. = ⅛
= 1·6876" rad.

$\frac{4}{2}$ − 0·5 = 1·5
+ ·1875. = 1·6875.

Say 4·25. $\frac{4 \cdot 25}{3/8}$ = 11·333.

$\frac{144 \cdot 011}{11 \cdot 3333}$ = 12·706862
× ⅛ = 1·5883577
3·177" ϕ.

$\frac{4 \cdot 25}{2}$ − 0·5 = 2·125 − ·5
= 1·625 + ·1875 =
1875
1·8125

Problem. It appears that the limitation on inertia,

i.e Equivalent inertia ≤ 5 × rotor inertia is required to ensure that the motor doesn't lose step. [This statement needs to be checked for the larger motor].

What steps can be taken to ensure this condition is satisfied?

1/. Use largest motor Type 34 (344-631). Rotor inertia 640 gcm^2

2/. Make Telescope light to reduce its inertia – use 'thin' mirror, say 8½" O.D × ¾" thick. Keep P.A. & fork light.

3/. Instead of 10 steps/sec. for sidereal rate – use 15 steps/sec. – this will need a larger reduction ratio ~~needed~~ on the 'gear-box'.

4/. Reduce the slewing speed requirement – i.e instead of 100 times sidereal rate, use say 70 times. * This does NOT help meet inertia requirement

———— " ————

All these steps combined might just make the size of stepper motor feasible.

———— " ————

The inertia of the instrument about the dec. axis & also about the P.A needs to be reasonably accurately determined in order to correctly choose the reqd. stepper motors.

————————

~~F~~ focal pt of mirror to be 9½" from optical axis, i.e ~ 4½" from outside of tube.

1987 April 4. (55)

All components made from dural (except P.A. shaft) in order to minimise Inertia.

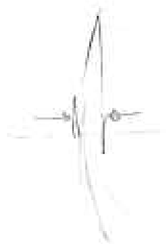
Dural 0·1 lbs/cu.in. Steel 0·28 lbs/cu.in.

S.G. of glass 2·6 = 0·095 lbs/cu in.

Mirror. $8\frac{1}{2}$" ϕ × $\frac{3}{4}$" thick plate glass. f/4.

F.L. = 34" inc.

Rad = 68".



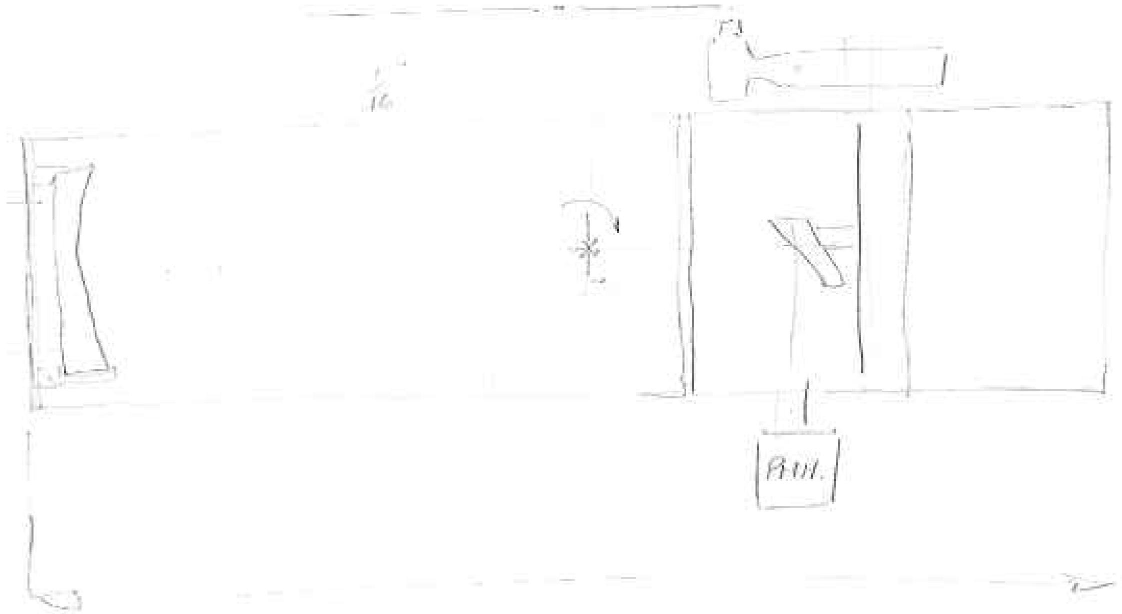
$x = \frac{4.25^2}{2 \times 68} = .1328$

Use 0·7" as mirror thickness.

Wt of mirror $= \pi \times \frac{8.5^2}{4} \times 0.7 \times 0.095 = \underline{3.78 \text{ lbs}}$

Flat. $2\frac{1}{2}$" minor axis × $\frac{1}{4}$" thick

$Wt. = \pi \frac{2.5^2}{4} \cdot \sqrt{2} \times 0.25 \times 0.095 = 0.17$ lbs



Wt/in. 1" × 1" × 3/16" D.L = 0·034 lbs/in. ·3179

$1\frac{1}{4} \times 1\frac{1}{4} \times \frac{3}{16}$" DL = 0·044 lbs/in. ·4173

$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{3}{16}$ DL = 0·053 lbs/in. 0·54

$\frac{a^2 + at - t^2}{2(a-t)}$

$(a-t)t\left(\frac{t}{2} + \frac{a-t}{2}\right)$

$t(a + a - t)$

$\frac{at^2}{2} + \frac{t(a-t)(a+t)}{2}$

$t(2a-t)$

To find c.g. of 'revised' tube.

Taking C.W. moments abt LH. end   D = dural

T1. Mirror 1 off glass $8\frac{1}{2}" \phi \times 0.7"$

wt = 3.78 lbs. $\bar{x}_1 = \frac{0.7}{2} + .187 + .187 + 0.175 = 0.724 . 0.849$

m_1 = 3.21 lbs in

T2 End plate. $9.875^2 \times 3/16"$. D 1 off

wt = 1.83 lbs $\bar{x}_2 = .0938"$ $m_2 = 0.18$ "

T3. End angles $1" \times 1" \times 3/16"$ D x = 8.875" 4 off

$_4\omega_3 = 4 \times 8.875 \times .034 = 1.21$ lbs; $\bar{x}_3 = .1875 + .318$

= 0.505 $_4m_3 = 0.61$

T4. End cleats $3" \times 1\frac{3}{4}" \times 1/10"$ 8 off.

$_8\omega_4 = 8 \times 3 \times 1.75 \times .1 \times .1 = 0.42$ lbs.

$\bar{x}_4 \approx .375 + .875 = 1.25"$ $_8m_4 = 0.53$

T5 Clip assemblies for retaining mirror = $(1\frac{1}{4} \times 1\frac{1}{4} \times 3/16) \times 2"$ - 4 off.

$_4\omega_5 = 4 \times .044 \times 2" = .352$ lbs

$\bar{x}_5 = 1.3"$ $_4m_5 = 0.46$

T6. Mirror support pads $\equiv 7/8" \phi \times 3/8"$ thick - 3 off.

(Allow for screw & nut).

$\omega_6 = Dt. \frac{\pi .875^2}{4} \times .375 \times .1 \times 3 = .07$ lbs. $\bar{x}_6 = 0.3$ $_3m_5 = 0.02$

N.B. Each 6mm. steel screw + washer will be ~ .01 lbs

T7 Corner angles for lower section $(1 \times 1 \times 3/16)$ D. $\times 23"$ long - 4 off

$_4\omega_7 = 4 \times .034 \times 23 = 3.13$ lbs $\bar{x} = 11.5$ $_4m_7 = 35.98$ lbs in

T8 Cladding for lower section $(10" \times 23" \times 1/16")$ - 4 off.

$_4\omega_8 = 4 \times 10 \times 23 \times \frac{1}{16} \times .1 = 5.75$ lbs. $_4m_8 = 66.13$

2.875

1987 April 5

(57)

T_9. Top of lower section – 3/16" pl. 1 off
9 7/8" sq, hole 8 3/4" ϕ.

1.83 lbs
1.13

1.87

$w_9 = Wt = \left(9.875^2 - \pi \frac{8.75^2}{4}\right) \times .1875 \times .1 = 0.7$ lbs

moment $0.7 \times (23 - .09)$ $m_9 = 16.06$ lbs in

T_{10} Support L's for T_9. $(1 \times 1 \times 3/16\ L) \times 7.875$ – 4 off.

$Wt = w_{10} = 4 \times .034 \times 7.875 = 1.07$ lbs.

$\bar{x} = 23 - .32 - .1875 \approx 22.5$. $4 m_{10} = 24.1$ lbs in

---

T_{11} Bottom of upper section – 1/4" pl 1 off
10" sq, hole 8 3/4" ϕ

7.5 lbs
1.5 lbs

20.93

$w_{11} = \left(10^2 - \pi \frac{8.75^2}{4}\right) \times .25 \times .1 = 1.0$ lb.

$\bar{x} = 23 + 1/8 + 1/8 = 23.25$ m_{11} 23.20

T_{12} Angle clips for T_{11} — 8 off from $(1 \times 1 \times 3/16\ L)$.

Regard as 2 1/4" long

$8 w_{12}$ $Wt = 8 \times .034 \times 2.25 = 0.61$ lbs.

$\bar{x}_{12} = 23 + 1/8 + 1/4 + .32 = 23.695$ $8 m_{12} = 14.50$

T_{13}. Special retaining clips between upper & lower sections 4 off.
i.e clipping T_9 & T_{11} together. see drawing.
regard as $(1\frac{1}{4} \times 1\frac{1}{4} \times 3/16\ L) \times 3\frac{1}{2}$" long

$Wt = 4 \times .044 \times 3.5 = 0.62$ lbs

$\bar{x} = 23 + 1/8 + 1/2 = 23.625$ $4 m_{13} = 14.55$

T_{14} Side plates for rotating head 1/10" D – 2 off
$10.25 \times 7.5 \times 1/10$ $Wt = 2 \times 10.25 \times 7.5 \times .1 \times .1$
$= 1.54$ lbs.

$\bar{x} = 23\frac{1}{8} + 3\frac{3}{4} = 26.875$ $2 m_{14} = 41.32$

T_{15} Side plates for rotating head 1/16" D. – 2 off

$Wt = 0.96$ $2 m_{15} = 25.83$

$\bar{x} = 26.875$.

1987 April 5

(58)

T_{16}. Spider & flat support — see drawing
regard as $9\frac{1}{2} \times 8" \times \frac{1}{16}"$

$$\omega_{16} = 9.5 \times \frac{8}{16} \times [illegible] \times .1 = .475 \text{ lbs.}$$

$$\bar{x}_{16} = 23 + 5 = 28$$ $\therefore m_{16}$ 13.3

T_{17} Eye-piece draw-tube D 1 off.
regard as. $1\frac{1}{4}$ I.D × 1.6" O.D × 2.75" long

$$wt. = \frac{\pi}{4} \times .1 \times 2.75 (1.6^2 - 1.25^2) = 0.34$$

$$\bar{x}_{17} = 23\frac{1}{8} + 2\frac{1}{2} = 25.625$$ $\therefore m_{17}$ 8.65.

T_{18} Angle stiffening of eye-piece section $1 \times 1 \times \frac{1}{8}"$ L — 4 off
= 9" long.

$$Wt = 4 \times .024 \times 9 = 0.87$$

$$\bar{x} = 23 + \frac{1}{8} + 7\frac{3}{4} = 30\frac{7}{8}$$ $m_{18} = 26.68$

T_{19}. FLAT - GLASS wt. = 0.17 lbs.

$\bar{x} = 23 + 2\frac{3}{8}$ m_{19}. 4.36.

T_{20} Corner brackets for T_{18}. — 4 off $\frac{1}{10}"$ t.

$$wt = 2 \times 3.75^2 \times .1 \times .1 = 0.28 \text{ lbs.}$$

$\bar{x} = 31"$ $\therefore m_{20} = 8.72.$

T_{21} Light shield (dew-cap?) extension 6" long. 4 off.
$10" \times 6" \times \frac{1}{16}$

$$wt = 4 \times 10 \times 6 \times \frac{1}{16} \times .1 = 1.5 \text{ lbs.}$$

$\bar{x}$

$\bar{x} = 33.5$ $\therefore m_{21} = 50.25.$

T_{22} Corner L's for T_{21}. $(1 \times 1 \times \frac{1}{8}" L) \times 5"$ long. 4 off

$$Wt. = 4 \times 5 \times .024 = 0.48.$$

$\bar{x} = 33.5$ $\therefore m_{22}$ 16.08

T_{23} Finder – Wt. 3 lbs.

$\bar{x} = 24\frac{1}{8} + 3\frac{3}{8} = 27.5$ $m_{23} = 82.5$.

T_{24} PMH – wt. = say 2.2 lbs allowing for eye-piece & partial wt. of fluid light guide.

$\bar{x} = 23\frac{1}{8} + 2\frac{1}{2} = 25.625$ $m_{24} = 56.4$.

T_{25} Support & system for finder say ·35 lbs

$\bar{x} = 28$ $m_{25} = 9.8$.

Total bolting mat! say <u>2 lbs.</u>

<u>N.B</u> If we have a decⁿ ⊙ & an H.A. ⊙ then we could use the ⊙'s to find the object as is done with the existing system.
∴ Delete T_{23} & T_{25}.

T_{26} Decⁿ Quadrant

5/32" D sheet
15" rad

$Wt = \pi\, 225 \times \frac{5}{32} \times \frac{115}{360} \times ·1 = 3.53$

$- 6 \quad 4"\phi$ holes for lightening $= 6 . \pi . \frac{4^2}{4} \times \frac{5}{32} \times ·1 = 1.18$.

$\rightarrow 2.35$ lbs.

for time being assume c.g. 10" from centre.

$m_{26} \sim 2.35 \times 7 =$

T_{27} Taper roller races & supports – say ≡ 3"φ × 1¼" thick D. 2 off

$m = 2 \times ·88 = 1.77$ lbs.

$I = 1.99$.

1987 April 5

(60)

Est. of wt. of tube (complete).

	wt. lbs.	m lbs"	k.	Mk^2	I_{xx}	I_T
1	3.78	3.21	14.05	746.2	17.1	763.3.
2	1.83	.18	14.72	396.5	14.9	411.4.
* 3	1.21	.61	11.41	281.0 ~~250.9~~	4.1	285.0
* 4	.42	.53	13.65+	86.8	.1	86.9
* 5	.35	.46	13.6	67.9	.1	68.0
* 6	.07	.02	14.6	14.9	—	15.0
7	3.13	35.98	5.88	108.3	138.0	246.3
8	5.75	66.13	3.4	105.4 / 33.2	126.7 / 150.7	231.8 } 183.9 }.
9	.7	16.06	8.01	46.0	9.5	55.5
10	1.07	24.1	9.1 / 7.6	44.3 / 30.9	— / 2.8	44.3 } 33.7 }.
11	1.0	23.2	8.35	69.7	13.7	83.4.
12	.61	14.5	(8.8+)	60.6	—	60.6
13	.62	14.55	(8.7+)	54.4	—	54.4.
14	1.54	41.32	(11.97+)	259.4	7.2	266.6
15	.96	25.83	11.97	137.7	12.9	150.6.
16	.48	13.3	13.1	81.5	say 5.0.	86.5
17	.34	8.65	10.72	39.1	say .4	39.5
18	.87	26.68	15.97+	121.9 / 110.9	3	122.0 } 114.05 }.
19	.17	4.36	10.63.	19.3	—	19.3.
20	.28	8.72	16.1	78.2	—	78.2
21	1.5	50.25	18.6+	556.4	—	556.4
22	.48	16.08	18.6+	175.7		175.7
T27 × ~~23~~	~~3.0~~	82.5				2.0.
24	2.2	56.4	(10.625+)	426.6		426.6
Screws × 25	.35	9.8.				234.4 } Screws etc.
✓+ 26.	2.35	16.45.		281.25		281.25
	31.71.	467.57				

with no allowance for screws $\bar{x} = 14.75$.

If we allow 2 lbs for screws, washers etc. @ $\bar{x} = 18"$
then $W = 33.71$ $m = 503.57$. $\bar{x}_T = \frac{503.57}{33.71} = 14.94$.

If we allow 1 lb for screws etc. @ $\bar{x} = 18$. $= 14.84$
$w = 32.71$ $m = 485.57$ $\bar{x}_T =$

Use 14.9 for 1st est. of 1½ lbs. @ 12.5" rad.

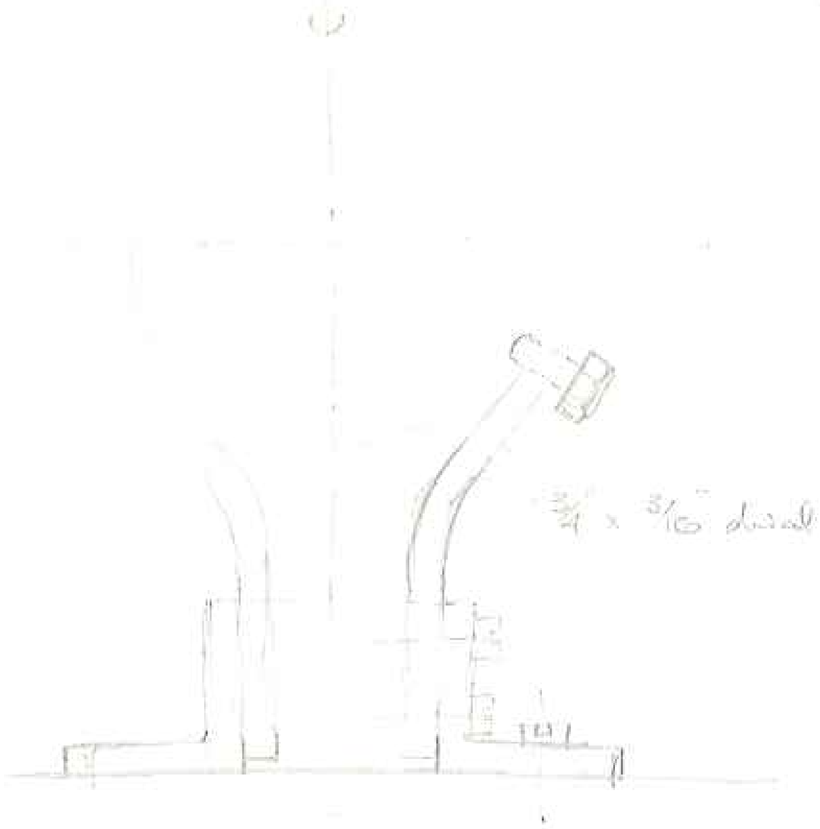
[5177]

Σ 5176.55

×2929 for gm²

1987 Feb. 24

(61)



Light cone from primary

⊙ - projected flat.

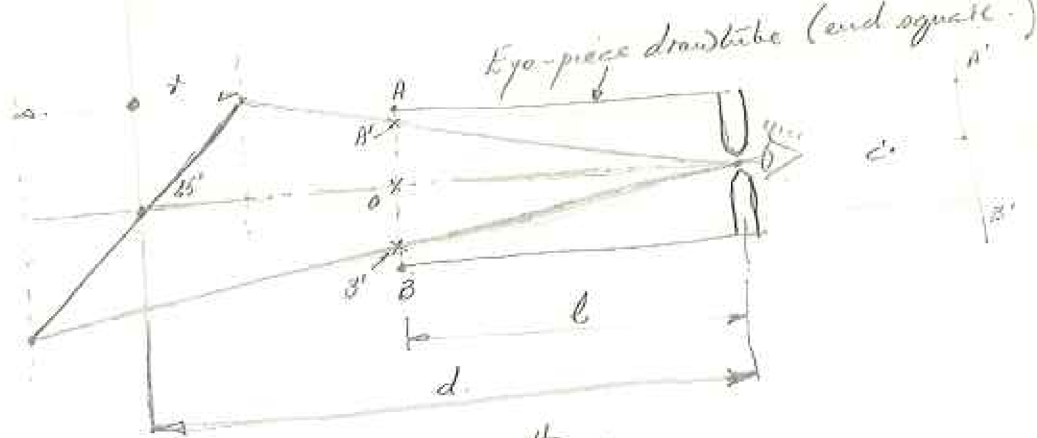
Loss of light = … %

= [illegible]

~0.25 %

1987 Feb 24.

If we have a standard Newtonian flat, i.e. cut from a cylinder of rad r what do we see when this is viewed thro' a small pin-hole as in setting up a Newtonian? (62)



If diag^l were set up as shown then :-

$OA' = r.\frac{l}{(d-r)}$ & $OB' = r.\frac{l}{(d+r)}$; $OC' = r\frac{l}{d}$.

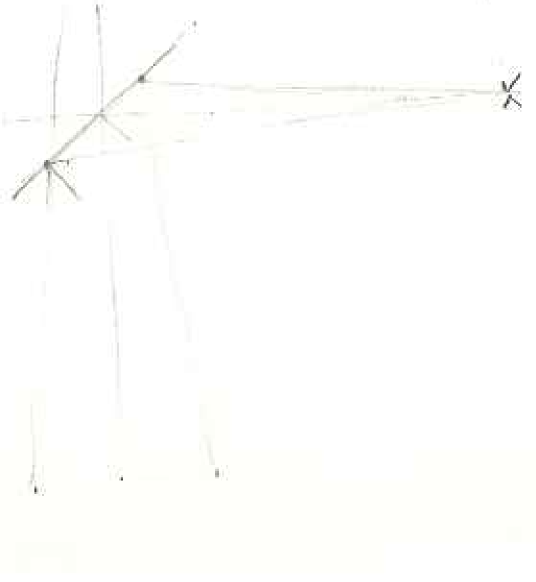
$2OC' = 2r\frac{l}{d}$ & $OA' + OB' = rl\left(\frac{1}{d-r} + \frac{1}{d+r}\right) = 2rl\left(\frac{d}{d^2-r^2}\right)$

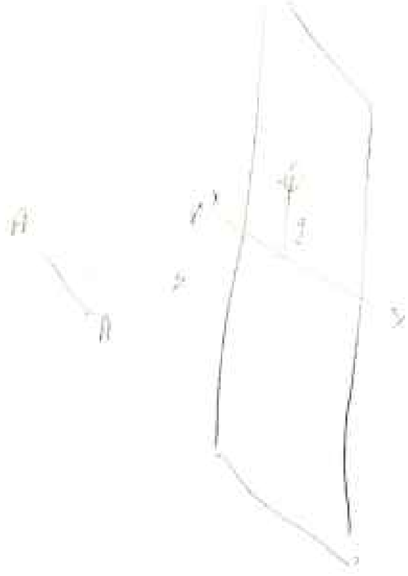
Suppose $r = 1.125$ $l = 4$ & $d = 8.5$ (taken from full-scale drawing for APT.).

$2OC' = \frac{2.25 \times 4}{8.5} = \frac{9.0}{8.5} = 1.059$ &

$OA' + OB' = 2.25 \times 4 \left(\frac{8.5}{72.25 - 1.125^2}\right) = 9 \times \frac{8.5}{} = 1.078$

Hence it is highly likely we imagine we see a ⊙.





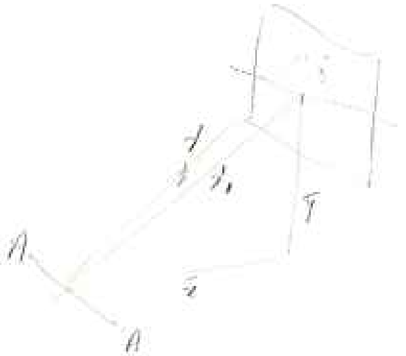
$I_{AA} = \sum m r^2$

$= \sum m (\bar{x} + y)^2$

$= \sum m (\bar{x}^2 + 2\bar{x}y + y^2)$

$= M\bar{x}^2 + I_{xx} = \ldots \frac{bd^3}{12}$

$= M\bar{x}^2 + M\frac{d^2}{12}$

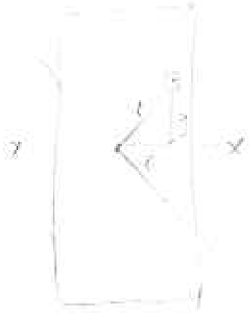


$\sum m r^2$

$= \sum m (\bar{x}^2 + (\bar{y} + y)^2)$

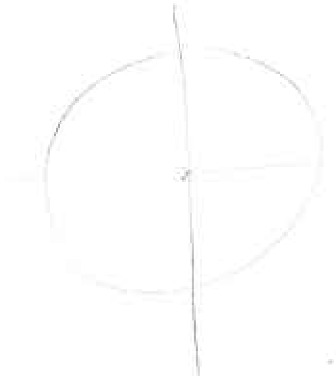
$= \sum m (\bar{x}^2 + \bar{y}^2 + 2\bar{y}y + y^2)$

$= M r^2 + M\frac{d^2}{12}$



$I = \sum m r^2 = \sum m(x^2 + y^2)$

$= M\left(\frac{d^2}{12} + \frac{b^2}{12}\right)$



$J = \frac{\pi d^4}{32} = \frac{\pi d^2}{4} \cdot \frac{d^2}{8} = \frac{Md^2}{8}$

$J_{xx} = \frac{Md^2}{16}$

1 Newton = a mass of 1 Kg × accⁿ of 1 m/sec² (3.28'/sec.) ; 1(mN) = a mass of 1 gm. × accⁿ of 1 m/sec²

(62B)

$\frac{1 \text{ lbf}}{4.448}$ = a mass of 1 Kg × " $\frac{32.2}{3.28} = 9.814$.

1 lbf = [1 lb mass] × g.

= $\left[\frac{W}{g}\right]$ × g.

a mass of 1 Kg exerts 9.814 N. at earth's surface.

4.448 × 1 lbf = 1 N = 4.448 lbs mass × g.

= 2 Kg

9.814 N = a mass of 1 Kg × accⁿ of 9.814 m/sec²

= mass of Kg × g.

∴ 1N ≡ mass of $\frac{1 \text{ Kg}}{9.814}$ × g. = 101.9 gm × g.

Torque (mN) × m = I × α

= g. m² × $\frac{\text{rads}}{\text{sec}^2}$

a mass of 1 gm will exert a force of 9.80665 mN at earth's surface.

Hence a torque of 200 (mN) m ≡ $\frac{200}{9.80665}$ × 100 ~~$\frac{20}{9.80665}$~~

≡ 2040 gm. cm.

≡ $\frac{2040}{454}$ × $\frac{1}{2.54}$ = 1.77 lbs ins.

1987 April 22

(63)

To find c.g. of tube using 'new' mirror (by EJ Ayson) & reverting back to slightly heavier sections

[See also pp 37, 56, 57, 58 & 59 for more details of items.]

Wt. of mirror — use 8 lbs.

Flat $= 0.17 \times \frac{7/16}{1/4} = 0.3$ lbs.

T1. Mirror 8 lbs ; $\bar{x} = 1.25"$ $m_1 = 10$ lbs ins

T2. End plate $9.875^2 \times 3/16"$ D 1.83 lbs $m_2 = 0.18$

T3. End angles $1 \times 1 \times 3/16 \times 8\frac{7}{8}"$ - 4 off ; $_4\omega_3 = 1.21$ lbs $m_3 = 0.61$

T4 End cleats $3" \times 1\frac{3}{4}" \times 1/10"$ 8 off $_8\omega_4 = .42$ lb $m_4 = 0.53$.

T5. Clip assemblies for retaining mirror - 4 off

$_4\omega_5 = .35$ $m_5 = 0.46$

T6. Mirror support pads. $7/8"\phi \times 3/16$.

$_3\omega_6 = .03$ $m_6 = 0.01$

T7 Corner L's for lower section $1\frac{1}{4} \times 1\frac{1}{4} \times 3/16$.

$_4\omega_7 = 4 \times .044 \times 24.75 = 4.27$ lb. $m_7 = 52.61$

$\bar{x}_7 = 12.32$

T8 Cladding $(10 \times 24\ 5/8 \times 1/16)$

$_4\omega_8 = 4 \times 10 \times 24\frac{5}{8} \times \frac{1}{16} \times .1 = 6.16$ lbs $m_8 = 75.85$

T9. Top of lower section [square with circle] $9\frac{7}{8}"^2$ $8\frac{3}{4}"\phi$ hole $\times 3/16"$

wt 0.7 lbs. $\bar{x} = 24.53$ $m_9 = 17.2$

1987 April 22

(64).

Support L's for T_9.

T_{10}. 1"×1"×3/16" L × 7 7/8" – 4 off

$_4\omega_{10} = 1.07$ lbs
$\bar{x}$ 24 1/4
m_{10} 25.81

T_{11} bottom of upper sect 10" sq × 1/4" (8 3/4 ϕ hole)

$\omega_{11} = 1.0$ lb.
$\bar{x}_{11} = 24.875$
$m_{11} = 24.88$

T_{12}
$_8 wt_{12}$ 0.61 lbs
$\bar{x}_{12}$
m_{12} 15.45

T_{13}.
$_4\omega_{13} = 0.62$ lbs
m_{13} 15.66

T_{14}.
$_2\omega_{14} = 1.54$
m_{14} 43.9

T_{15}.
$\omega_{15} = 0.96$
m_{15} 27.36

T_{16}
$\omega_{16} = .48$
$\bar{x} = 30.5$
m_{16} 14.6

T_{17}
.3
8.18

T_{18} L stiffening 1×1×3/16.
$wt = 4 \times .034 \times 9 = 1.23$
m_{18} 34.00

T_{19} FLAT Glass
ω_{19} 0.3 lbs.
m_{19} 8.2

T_{20}
$_4\omega_{20} = 0.28$ lbs
m_{20} 9.13

T_{21}
$\omega_{21} = 1.5$ lbs
m_{21} 52.61

T_{22}
$\omega_{22} = 0.5$
m_{22} 17.56

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(65)

T_{23}. Finder ω_{23} 3 lbs

$m_{23} = 87.4$

T_{24} PMH. ω_{24} 2.2 lbs. m_{24} 59.95

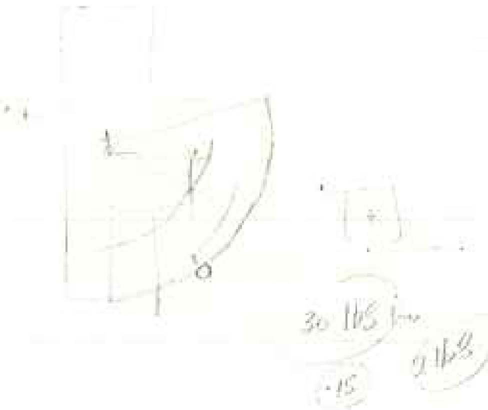
T_{25} ω_{25} .35 lbs m_{25} 10.37

T_{26} Dec Quadrant. 3/16" D. wt. ~ 5 lbs
$\bar{x} \sim 7''$ 35 lbs in

1987 April 22

(66)

Est. of wt. & c.g. of tube



1	8.0	10.0
2	1.83	0.18
3	1.21	0.61
4	0.42	0.53
5	0.35	0.46
6	0.03	0.01
7	4.27	52.61
8	6.16	75.85
9	0.7	17.2
10	1.07	25.81
11	1.0	24.88
12	0.61	15.45
13	0.62	15.66
14	1.54	43.9
15	0.96	27.36
16	0.48	14.6
17	0.3	8.18
18	1.23	34.00
19	0.3	8.20
20	0.28	9.13
21	1.5	52.61
22	0.5	17.56
23	3.0	87.4
24	2.2	59.95
25	0.35	10.37
26	5.0	35.00
	Σ 43.91	647.51

25

14.746

14.79