Interoffice

# National Radio Astronomy Observatory

## Very Large Array

November 21,1983

To: J.Findlay,H.Hvatum,P.Napier,G.Peery,T.Legg

From: W.Horne

VLBA ANTENNA MEMO No. 4

Subject: Drive Design-VLBA Antennas

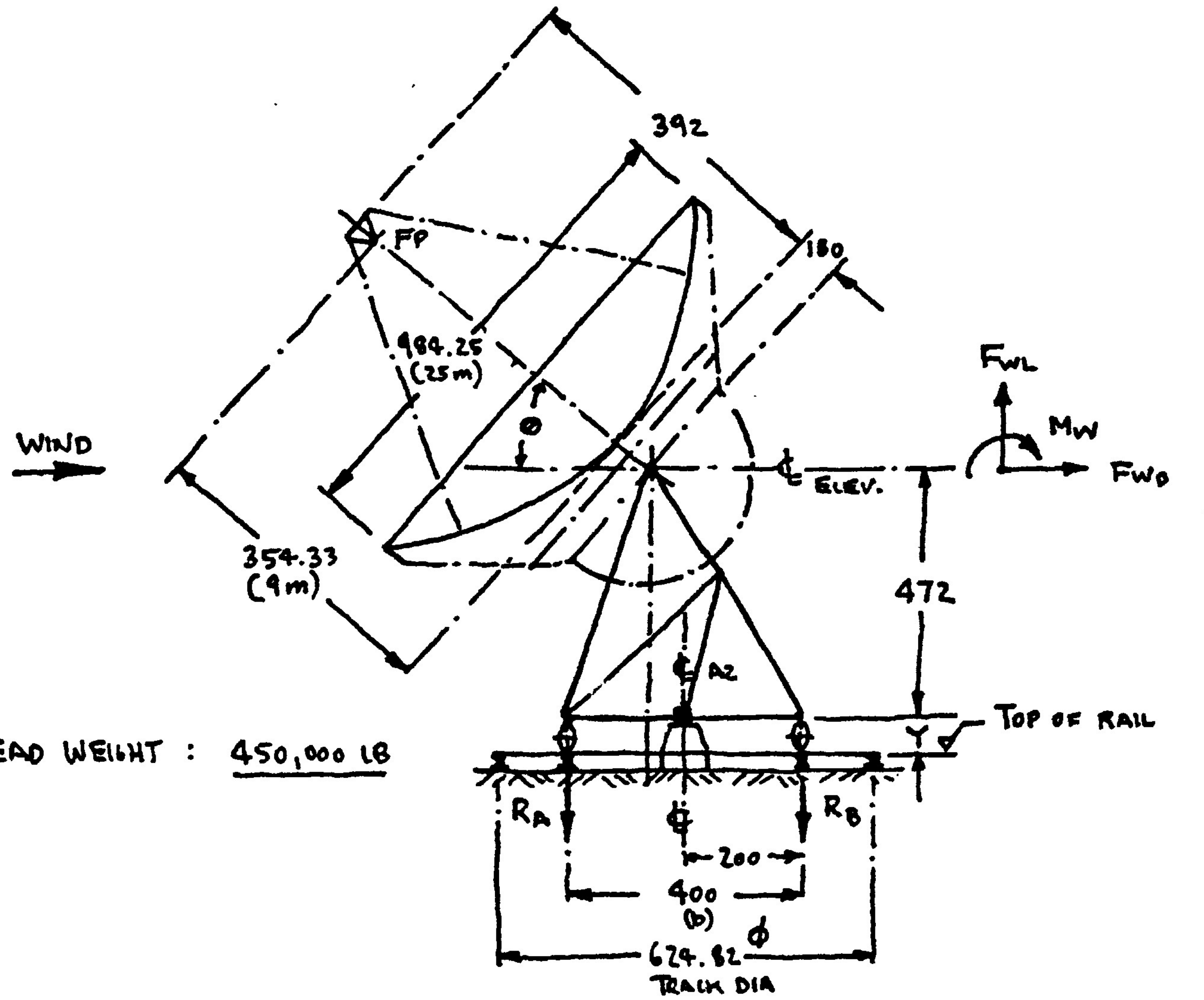
Attached for your information is VLBA Antenna Memo No. 4covering the drive design for the VLBA antenna. This design was performed by O.HEINE of Systems Development Laboratories. We will make some revision in the area of the pintle bearing and perhaps in the azimuth gearbox/support wheel interface.

ORH

# VLBA ADVANCED DESIGN

## WIND-LOADS AND MOMENTS

## VLBA Antenna Memo #4




DEAD WEIGHT : 450,000 LB

| WINDLOADS & MOMENTS @ AXIS INTERSECTION ( BASE FACTORS ONLY) * | | | | | |
|---|---|---|---|---|---|
| ANGLE OF ATTACK, θ | 0° | 60° | 90° | 120° | 180° |
| DRAG FORCE, $F_{WD}$ (LB) | 1,158,140 | 578,800 | 91,280 | 208,040 | 806,080 |
| LIFT FORCE, $F_{WL}$ (LB) | -14,450 | -1,086,200 | 59,720 | 124,680 | 21,440 |
| MOMENT, $M_W$ (IN-LB) | $14.76 \times 10^6$ | $45.1 \times 10^6$ | $90.9 \times 10^6$ | $115.76 \times 10^6$ | $21.06 \times 10^6$ |

* BASED ON DATA SUPPLIED BY L. KING

## REACTIONS AND WHEEL LOADS :

$$R_{A,B} = \tfrac{1}{2}(DL + IL) \pm \frac{F_{WD}(\mp 72 + Y) + M_W + M_I}{b} - F_{WL}/2 \quad ; \quad b = 400 \text{ IN}$$

$IL$ = 40,000 LB (ICE LOAD) ; $M_I = -9.69 \times 10^6$ INLB @ 0° ; $+9.69 \times 10^6$ INLB @ 180°
(ICE UNBALANCE MOMENT)

$DL$ = 450,000 LB (DEAD LOAD)

| VLBA FORCES & MOMENTS @ INTERSECTION OF AXIS , $V_W$ = 60 MPH , 1 cm ICE * | | | | | |
|---|---|---|---|---|---|
| ANGLE OF ATTACK, θ | 0° | 60° | 90° | 120° | 180° |
| $F_{WD}$ (LB) | 74,120 | 37,040 | 5,840 | 13,310 | 51,590 |
| $F_{WL}$ (LB) | -925 | -69,520 | 3,820 | 7,980 | 1,370 |
| $M_W$ (INLB) | $.94 \times 10^6$ | $2.89 \times 10^6$ | $5.82 \times 10^6$ | $7.41 \times 10^6$ | $1.35 \times 10^6$ |
| $M_I$ (INLB) | $-9.69 \times 10^6$ | $-4.85 \times 10^6$ | 0 | $4.85 \times 10^6$ | $9.69 \times 10^6$ |

ASSUMED : WHEEL DIA. = 30.0 IN ; Y = 36.0 IN

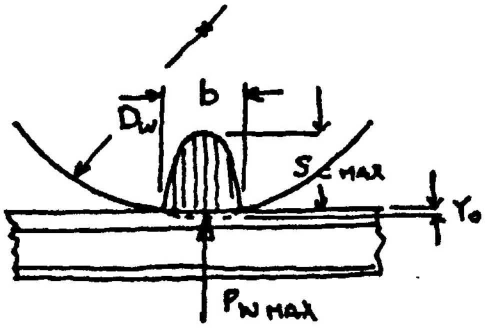
| REACTIONS @ AZ TRACK (RAIL) @ $V_W$ = 60 MPH , 1 cm ICE | | | | | |
|---|---|---|---|---|---|
| ANGLE OF ATTACK , θ | 0° | 60° | 90° | 120° | 180° |
| $R_A$ (LB) | 173,205 | 237,619 | 221,123 | 193,456 | 151,196 |
| $A_B$ (LB) | 317,720 | 321,901 | 265,057 | 288,564 | 337,434 |

$$P_{W\,MAX} = \frac{R_{B\,MAX}}{2} = \frac{337,434}{2} \cong \underline{169,000} \text{ LB (OPERATIONAL LOAD)}$$

$$P_{W\,MAX} = \frac{R_{B\,MAX}}{2} = \frac{312,393}{2} \cong \underline{156,000 \text{ LB}} \text{ (SURVIVAL LOAD @ 110 MPH, } \theta = 90°)$$

$$\text{VS.} \quad P_{ALLOWABLE} = 1600\,WD = 1600 \times \underbrace{3.5 \times 30}_{\text{WIDTH} \times \text{DIA}} = \underline{168,000} \text{ LB}$$

GIVEN BY C.M.A.A. FOR CLASS C , MODERATE SERVICE ON 175 CR RAIL




$$F_W = \frac{P_W}{W} = \frac{169{,}000}{3.5} = \underline{48{,}286 \text{ LB/IN}}$$

$$\bar{b} = 2.15\sqrt{\frac{F_W D_W}{E}} = 2.15\sqrt{\frac{169{,}000/3.5\,(30)}{30\times10^6}} = \underline{.472 \text{ IN}}$$

$$Y_0 = 2\,F_W \frac{1-\gamma^2}{\pi E} \ln \frac{\pi E W}{F_W(1-\gamma^2)}$$

$$Y_{0\,MAX} = 2\times 48{,}286 \frac{1-.25^2}{\pi\times30\times10^6} \ln \frac{\pi\, 30\times10^6\times3.5}{48{,}286(1-.25^2)} = \underline{.0085} \text{ IN}$$

$$Y_{0\,MIN} = 2\times \frac{\overset{(21{,}600)}{151{,}196}}{2\times3.5} \frac{1-.25^2}{\pi\times30\times10^6} \ln \frac{\pi\,30\times10^6\times3.5}{21{,}600\,(1-.25^2)} = \underline{.0042} \text{ IN}$$

POINTING ERROR DUE TO DIFFERENTIAL CONTACT DEFORMATION IS THEN:

$$tg\,\varepsilon = \frac{Y_{0\,MAX} - Y_{0\,MIN}}{400} = \frac{.0085 - .0042}{400} = \underline{1.075\times10^{-5}}$$

$\varepsilon = .00062^\circ \cong$ $\underline{2.2 \text{ ARC SEC}}$ @ $V_W = 60$ MPH (QUITE OK)

$$S_{c\,MAX} = 1.24\times.591\sqrt{\frac{F_W\ E}{D_W}}$$

$$S_{c\,MAX} = 1.24\times.591\sqrt{\frac{48{,}286\times30\times10^6}{30}} = \underline{161{,}000} \text{ LB/IN}^2$$

THE FACTOR 1.24 ACCOUNTS FOR THE INCREASE IN CONTACT STRESS DUE TO SLIDING FRICTION COFF. = .20. (SEE NEXT PAGE). THE MAX RECOMMENDED WHEEL LOAD OF 168,000 LB CORRESPONDS TO A CONTACT STRESS OF ABOUT 129,500 LB/IN$^2$; THUS THE OVERLOAD FACTOR IS APPROX 25% OVER RATED LOAD FOR CLASS "C", MODERATE SERVICE.

SOME HEAT TREATMENT OF THE RAIL & WHEELS MAY BE REQUIRED.

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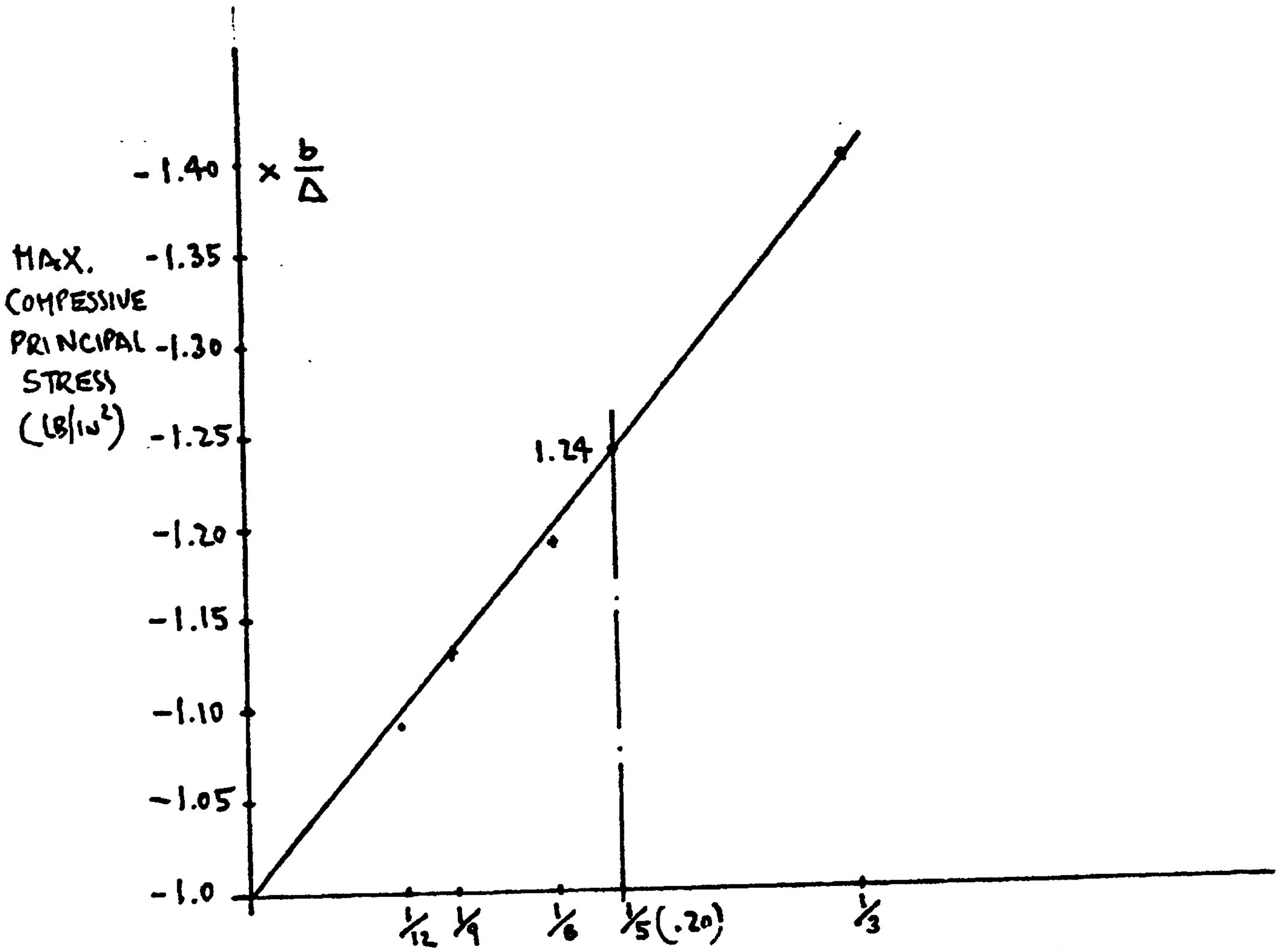
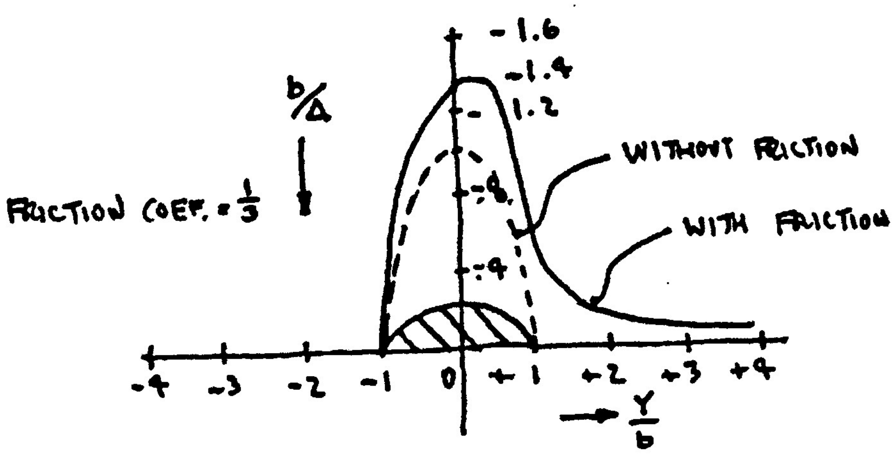
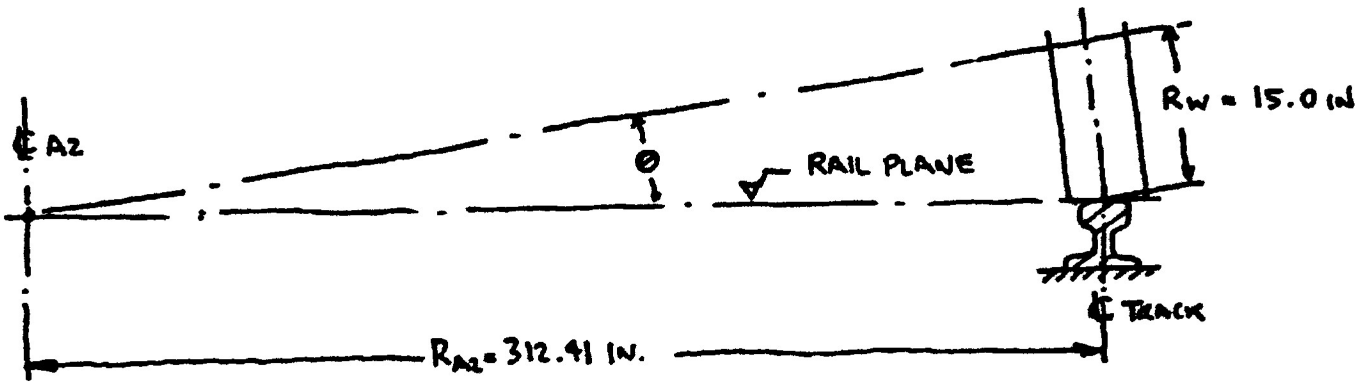


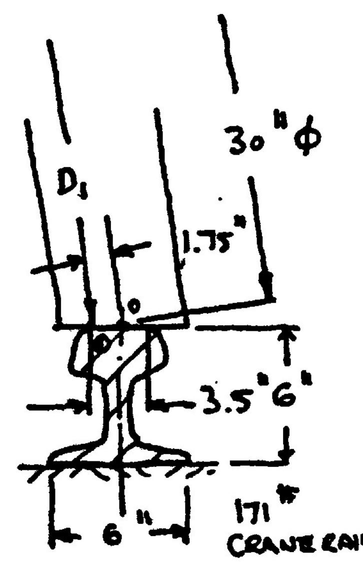

EFFECT OF TANGENTIAL FORCE DUE TO FRICTION ON CONTACT STRESS

REF: SEELY & SMITH, 2[ND] EDIT. ADVANCED MECHAN[ICS] OF MATERIAL, PG. 3[…] WILEY.

THE RAIL & WHEEL MAY EASILY BE HEAT TREATED (OIL QUENCHED & TEMPERED) SINCE THE CARBON CONTENT OF THE RAIL STEEL IS TYPICALLY .67 ÷ .82%.




INCLINE OF WHEEL REQ'D TO AVOID SPIN FRICTION :

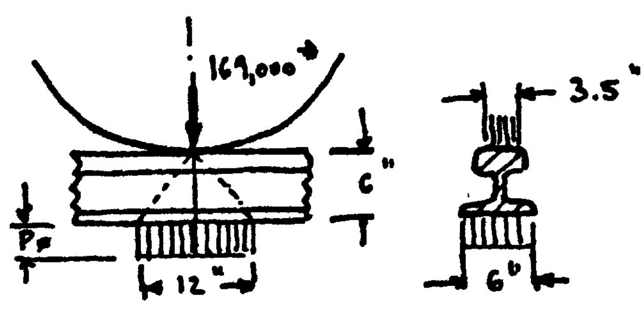



$$\sin\theta = \frac{15}{312.41} = .048 \; ; \quad \theta = \underline{2.752^\circ}$$

$$D_1 = 30 - 2(\tan 2.752^\circ \times 1.75) = \underline{29.832 \text{ IN}}$$

$$S_{c\,max\,①} = 1.24 \times .591\sqrt{\frac{48,286 \times 30 \times 10^6}{29.832}} = \underline{161,500} \text{ LB/IN}^2$$

THIS IS ONLY MODERATELY HIGHER THAN THE COMPRESSIVE STRESS AT POINT "O".




$$P_F = \frac{169,000}{6 \times 12} = \underline{2,347} \text{ LB/IN}^2$$ DISTRIBUTED PRESSURE LOAD ON FOUNDATION.

## FOUNDATION/RAIL INTERACTION:

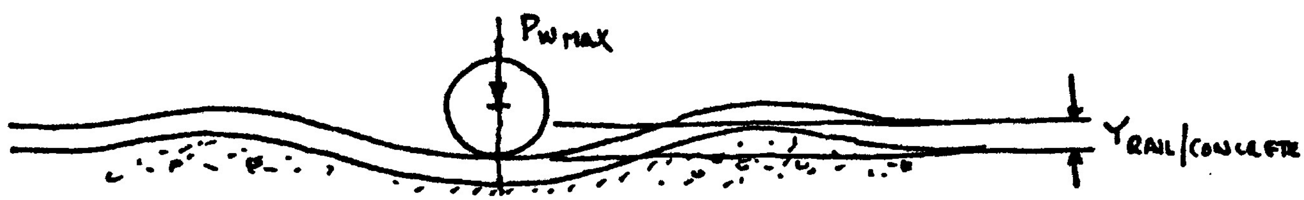
HIGH STRENGTH CONCRETE HAS FOLLOWING PROPERTIES:

| TIME ELAPSE | COMPRESSIVE STRENGTH | MODULUS OF ELASTICITY, $E_c$ (LB/IN²) |
|---|---|---|
| 3 MONTHS | 7,810 LB/IN² | $4.46 \times 10^6$ |
| 1 YEAR | 9,170 " | $5.53 \times 10^6$ |

SAFETY MARGIN AFTER 3 MONTHS: , $M_3 = \frac{7{,}810}{2{,}347} = \underline{3.3}$ OK

THUS, THE RAIL MAY BE PLACED DIRECTLY (VIA GROUTING) ON THE CONCRETE FOUNDATION, BUT THIS MAY NOT BE PRACTICAL IN THIS CASE.

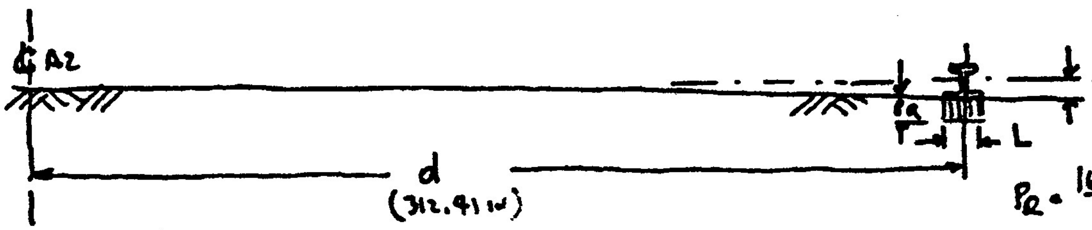
## ELASTIC DEFORMATION OF RAIL ON CONCRETE:




$\beta = \sqrt[4]{\frac{k}{4EI}}$ ; $M_{B\,max} = \frac{P_{w\,max}}{4\beta}$ , AT THE LOAD, IN RAIL

$Y_{max} = \frac{P_{w\,max}\,\beta}{2k}$ , AT THE LOAD

$I_{RAIL} = 73.4 \text{ IN}^4$ FOR 171# RAIL




$P_R = \frac{169{,}000}{6} = \underline{28{,}167} \frac{\#}{IN}$

$$Y = \frac{2P}{\pi E}\left(L \ln \frac{d}{L/2}\right) + PL\left(\frac{1-\nu_c}{\pi E_c}\right)$$

RE: ROARK, 4th EDITION, CASE 11 PG. 322

LET: $E_c = 4.46 \times 10^6 \text{ LB/IN}^2$ ; $\nu_c = .15$

$$Y = \frac{2 \times 28{,}167}{\pi\, 4.46 \times 10^6}\left(6.0 \ln \frac{312.41}{3}\right) + 28{,}167 \times 6.0\left(\frac{1-.15}{\pi\, 4.46 \times 10^6}\right) = \underline{.122 \text{ IN}}$$

$$k = \frac{P_R L}{Y} = \frac{169{,}000}{.122} = \underline{1{,}385{,}246} \text{ LB/IN}^3$$

$$\beta = \sqrt[4]{\frac{1{,}385{,}246}{4 \times 30 \times 10^6 \times 73.4}} = \underline{.112 \text{ IN}^{-1}}$$

$$M_{B\,MAX} = \frac{P_W}{4\beta} = \frac{169{,}000}{4 \times .112} = \underline{377{,}280} \text{ IN LB}$$

$$Y_{MAX} = \frac{P_W \beta}{2k} = \frac{169{,}000 \times .112}{2 \times 1{,}385{,}246} = \underline{.0068} \text{ IN}$$

(RAIL & FOUNDATION)

$$Y_{MIN} = \frac{75{,}598 \times .112}{2 \times 1{,}385{,}246} = \underline{.0031 \text{ IN}}$$

tg POINTING ERROR, $\alpha = \frac{.0035}{400} = 8.75 \times 10^{-6}$ ; $\alpha = \underline{1.80 \text{ ARC SEC}}$

THIS IS WORST CASE @ 60 MPH.

TOTAL POINTING ERROR DUE TO COMBINED EFFECTS OF CONTACT AND BENDING DEFORMATION IS AT 60 MPH (WORST CASE) :

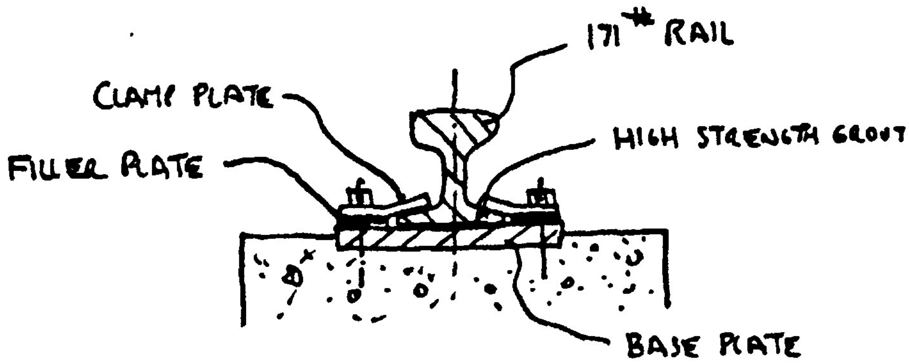
$$\gamma = \varepsilon + \alpha = 2.2 + 1.8 = \underline{4.0 \text{ ARC SEC}}$$

THIS DOES NOT INCLUDE FOUNDATION/SOIL MASS TILT !

BENDING STRESS IN RAIL :

$$S_{B\,max} = \frac{M_{B\,max}}{Z_{MIN}} = \frac{377,280}{24.42} = \underline{15,450} \ ^{LB}/_{IN^2} \ \text{OK}$$

WHILE IT APPEARS TO BE FEASIBLE TO MOUNT THE RAIL/TRACK DIRECTLY ONTO THE CONCRETE BASE (FROM A LOAD/STRESS VIEWPOINT), IT IS PROBABLY DESIRABLE TO IMBED A BASE PLATE INTO THE FOUNDATION SURFACE SO THAT THE RAIL MAY BE ATTACHED SECURELY TO THE BASE VIA ADJUSTABLE, BOLTED CLIPS.



# VLBA - AZIMUTH TRACK - CROSS SECTION

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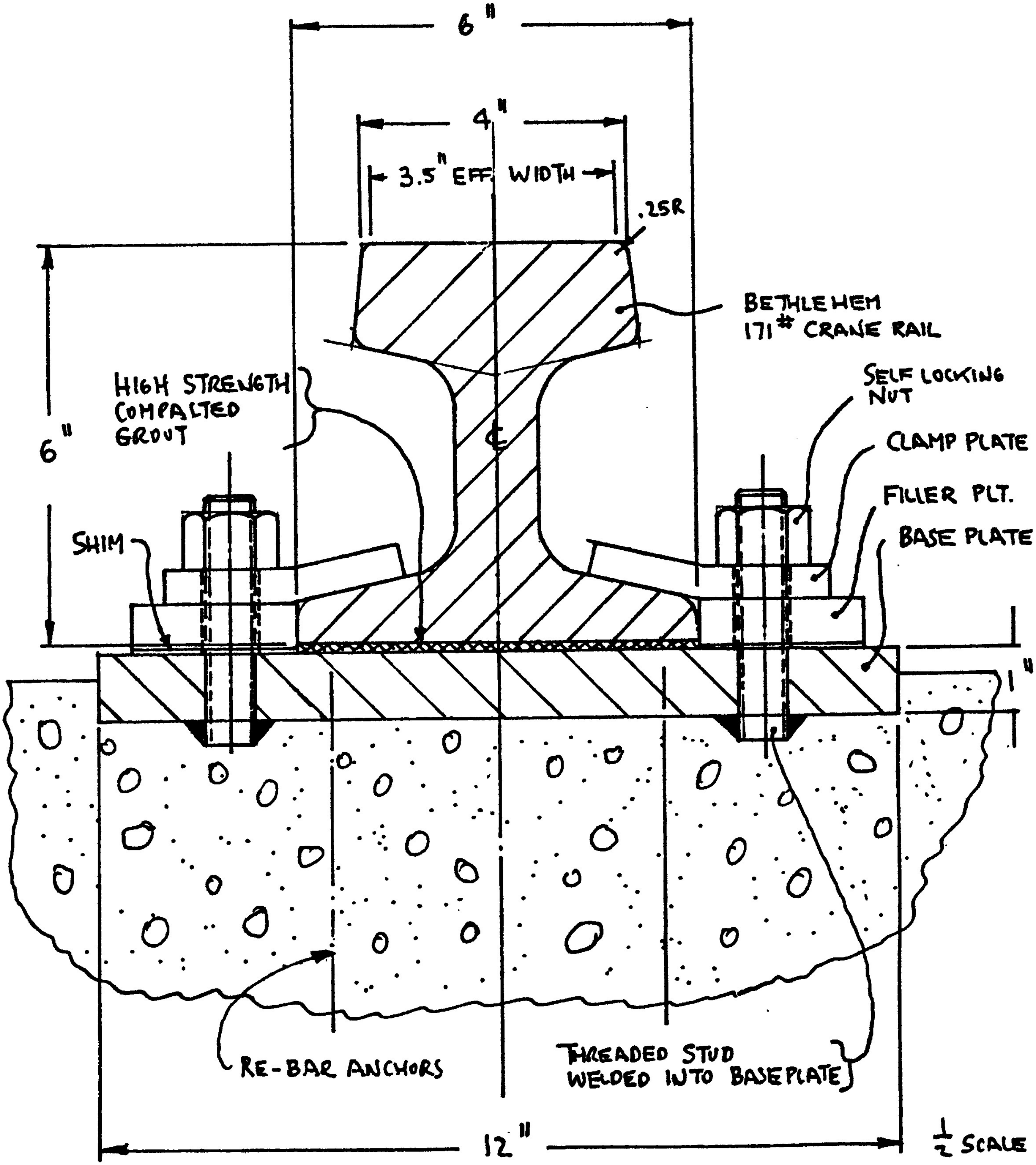


$\frac{1}{2}$ SCALE

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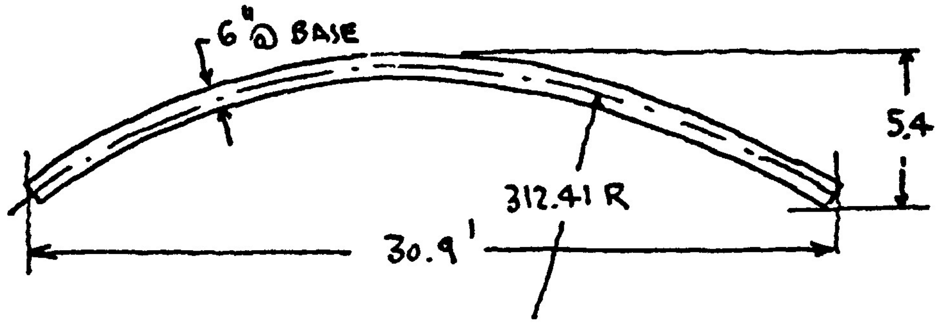
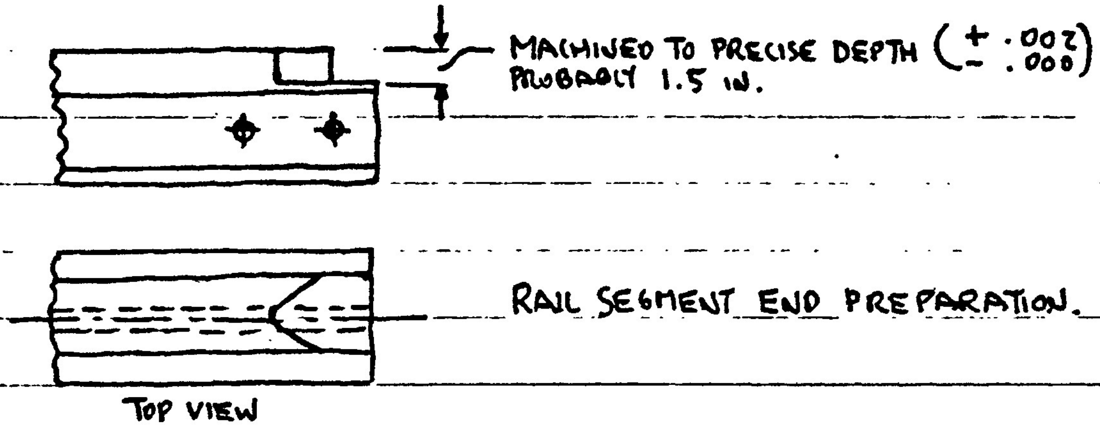
MANUFACTURING CONSIDERATIONS:

THE CIRCULAR AZ-CRANE RAIL TRACK WILL MOST LIKELY HAVE TO BE FABRICATED IN FIVE SEGMENTS SINCE THE STD. RAIL LENGTH IS 39 FT.. SPECIAL LENGTHS OF UP TO 60 FT CAN BE PROVIDED BY THE STEEL MILLS, BUT WOULD NOT BE AS EASELY SHIPPED & HANDLED.
EACH OF THE FIVE SEGMENTS WOULD BE ABOUT 32.7 FT. LONG STRAIGHT, AND 30.9 FT LONG WHEN BENT AND WILL WEIGH APPROX. 2.8 TONS.




THE FIVE SEGMENTS REQUIRED TO MAKE A FULL CIRCLE COULD BE TEMPORARY MOUNTED ON A SAY 1.0 IN THICK, FLAT STEEL PLATE. IN STAGGERED CONFIGURATION THE AREA REQUIRED WOULD BE 31 X 8 FT..
ALL FIVE SEGMENT COULD THEN BE GROUND FLAT VIA BLANCHARD GRINDER OR LARGE VERTICAL MILL PROVIDED WITH A GRINDING HEAD.
OBVIOUSLY, THE SEGMENTS WOULD HAVE TO BE ROLLED, HEATREATED (HARDENED) AND END-MACHINED PRIOR TO THIS OPERATION.

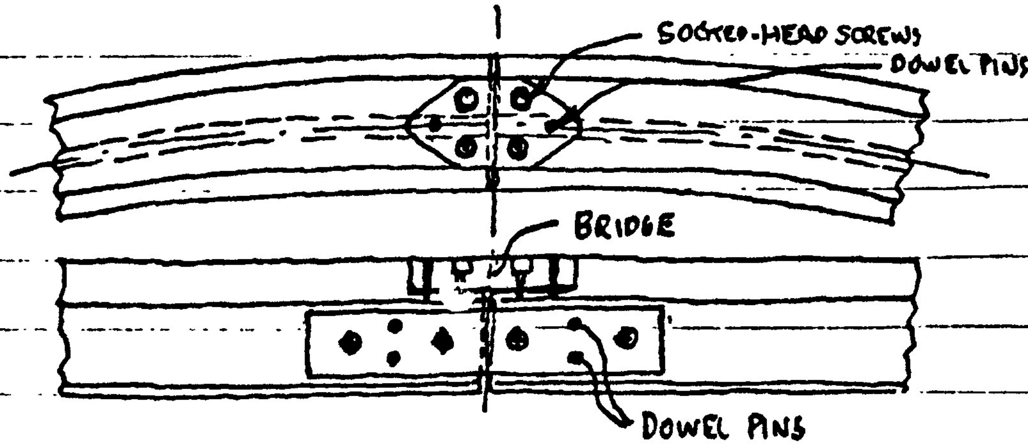
SUBSEQUENTLY, THE RAIL SEGMENTS COULD BE PREPARED FOR PRECISION SPLICING VIA. MILLING AND DRILLING OPERATION.




AFTER THIS WORK HAS BEEN ACCOMPLISHED, ALL FIVE SEGMENTS ARE TEMPORARELY PLACED AND MOUNTED ON A SUITABLE, FLAT MOUNTING PLANE AND OPTICALLY ALIGNED TO BE CIRCULAR WITHIN ± .030 AND FLAT (IN PLANE) WITHIN ± .003 IN.

THEREAFTER THE SPLICING IS APPLIED AND SECURED IN POSITION BY MEANS OF 2 DOWEL PINS ON EACH RAIL END. SPLICES & RAIL ENDS ARE THAN MATCH-MARKED SO THAT THEY MAY BE RE-ASSEMBLED WITH THE SAME PRECISION IN THE FIELD.

WHEN THIS IS ACCOMPLISHED, THE FIVE PRECISION BRIDGES REQUIRED FOR SMOOTH SURFACE TRANSITION ARE FABRICATED VIA MATCH FITTING & ASSEMBLED.



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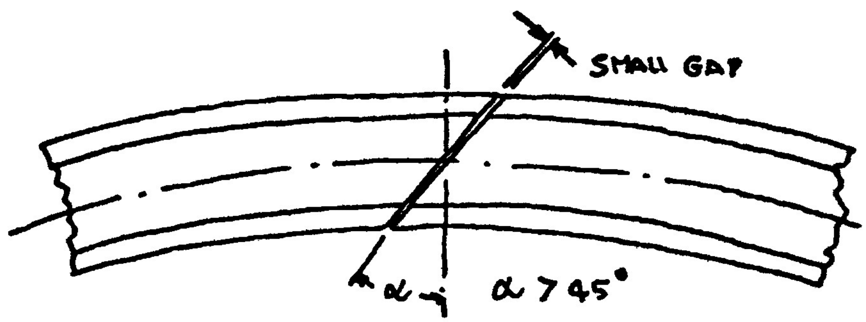
THE BRIDGES SHOULD BE MADE FROM A HIGH STRENGTH ALLOY SUCH AS SAE 4140 AND HEAT TREATED TO 58 RC SURFACE HARDNESS.

AFTER ASSEMBLING THE BRIDGES, THE SURFACE TRANSITION IS CHECKED VIA SUITABLE DIAL INDICATORS AND GROUND FLUSH VIA PORTABLE GRINDER, IF TOO HIGH, OR SHIMMED, IF TOO LOW.

AFTER MATCH-MARKING THE BRIDGES & RAIL ENDS, THE SEGMENTS ARE DISASSEMBLED AND READIED FOR SHIPMENT TO THE SITE.

ANOTHER FEASIBLE AND PERHAPS MORE COST-EFFECTIVE METHOD FOR FABRICATION OF THE AZ-TRACK WOULD BE TO MACHINE THE JOINTS AS FOLLOWS:




IN THIS CASE A HIGH PRECISION MACHINING EFFORT IS REQUIRED TO MATCH THE ANGULAR CUT ENDS OF THE MATING SEGMENTS SO AS TO ACHIEVE A VERY SMALL GAP. PRE-GRINDING & SPLICING OPERATIONS WOULD BE THE SAME AS FOR THE KEYED BRIDGE VERSION.

TO ACHIEVE MINIMAL- OR NEAR ZERO DIMENSION GAPS WITH ALL 5 JOINTS, THE PROCEDURE WOULD BE TO ASSEMBLE 4 PRE-MACHINED SEGMENTS FIRST AND MATCH-FIT THE 5th UNIT AFTER MAKING PRECISE MEASUREMENTS OF THE CORD LENGTH AND MITRE ANGLES.

AGAIN, THE SPLICES & JOINTS WOULD BE MATCHMARKED TO ASSURE EASE OF ASSEMBLY AT SITE.

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## ASSEMBLY AT SITE:

ASSEMBLY / INSTALLATION AT THE SITE INVOLVES THE FOLLOWING PROCEDURAL STEPS :

1. THE FIVE SEGMENTS ARE PLACED IN THE RIGHT ORDER PER MATCH MARKINGS ONTO THE BASE PLATE AND CENTERED WITH RESPECT TO THE PINTLE BEARING AS CLOSE AS POSSIBLE.
2. THE SPLICE PLATES ARE MOUNTED AND FASTENED AFTER INSERTION OF DOWEL PINS.
3. THE ENTIRE ASSEMBLED TRACK IS NEXT RAISED A FRACTION OF AN INCH AND SHIFTED IN ITS FINAL CONCENTRIC POSITION.
4. THE TRACK IS THEN LEVELED VIA APPLICATION OF A JIG TRANSIT FITTED WITH OPTICAL MICROMETER AND PRECISION SCALE (OR EQUIVALENT LASER INSTRUMENT) BY RAISING OR LOWERING SPECIAL SECTIONS VIA SHIMS. THE ACHIEVABLE PRECISION IS PROBABLY ON THE ORDER OF $\pm$ .005 IN; THE ACCURACY OF THE OPTICAL TOOL IS APPROX. $\pm$ .001 IN.
5. AFTER LEVELING & ALIGNMENT HAS BEEN ACCOMPLISHED, HIGH STRENGTH (IRON-CEMENT TYPE) GROUT IS PLACED BETWEEN THE BASE PLATE AND RAIL AROUND THE TRACK AND COMPACTED BY SUITABLE MEANS.
6. FOLLOWING THE APPLICATION OF GROUT AND ITS SUBSEQUENT CURING, FILLER PLATES PLUS SHIMS ARE FITTED AND THE RAIL-TRACK IS CLAMPED DOWN FIRMLY VIA APPLIED CLAMP PLATES.

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REQ'D VS. PROBABLE ACHIEVABLE AZ-ALIGNMENT ACCURACIES.

| ALIGNMENT ERROR | MAX. ALLOWABLE ERROR (ARC SEC) | EST'D ACHIEVABLE ACCURACY (ARC-SEC) |
|---|---|---|
| AZ-AXIS TILT (W. RESPECT TO GRAVITY) | 15 | 5 |
| AZ-AXIS RUN OUT | 10 | 5 |
| AZ-AXIS NON-REPEATABLE | 4 | 2 |

AZ-AXIS DRIVE REQUIREMENTS:

1. NOMIN. SPEED @ 40.3 MPH (45.9 MPH W. GUSTS) : $90°/\text{MIN} = \underline{.25 \text{ RPM}}$
2. MINIM. SPEED @ 54.0 MPH (60 MPH MAX. WIND) : $60°/\text{MIN} = \underline{.167 \text{ RPM}}$

WIND INDUCED MOMENT @ AZ-AXIS AT 45.9 MPH : (MAXIMUM VALUE)

$$M_{t\,AZ\,NOM.} = \frac{.00256\,(45.9)^2}{144} \times 57.88 \times 10^6 (2) = 4.335 \times 10^6 \text{ IN LB}$$
$$= \underline{361{,}310} \text{ FT LB}$$

WIND INDUCED MOMENT @ AZ-AXIS AT 60 MPH : (MAXIMUM VALUE)

$$M_{t\,AZ\,MAX} = \frac{.00256\,(60)^2}{144} \times 57.86 \times 10^6 \times (2) = 7.40864 \times 10^6 \text{ IN LB}$$
$$= \underline{617{,}387} \text{ FT LB}$$

SPEED REDUCTION REQ'D @ 1800 RPM INPUT SPEED OF DC-DRIVE MOTOR(S) AND .25 RPM AZ-AXIS SPEED WITH 30" DIA WHEELS ON 624.82" DIA TRACK:

$$i_{GT} = \frac{1800}{\frac{624.82}{\;} \;\underline{1}} = 345.7 : 1 \quad @\,1800 \text{ RPM INPUT SPEED}$$

$$i_{GT} = \frac{2500}{624.82/30\ 1/4} = 480.1 : 1 \quad @\ 2500 \text{ RPM INPUT SPEED}$$

CONSIDER STD. QUADRUPLE REDUCTION SPEED REDUCER WITH $i = 450:1$ REDUCTION & 97% OVERALL EFFICIENCY FOR HELICAL GEARS:

$$n_{AZ} = \frac{2500}{450}\ \frac{30}{624.82} = \underline{.27 \text{ RPM}} = 96\ ^\circ/\text{MIN. (OK)}$$

ROLLING FRICTION LOSSES:

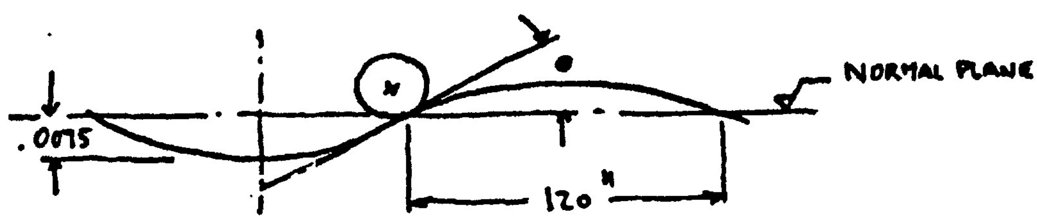
AVERAGE WHEEL LOAD IS WITH 1 cm ICE:

$$\bar{F}_W = \frac{490{,}000}{4} = \underline{122{,}500} \text{ LB}$$

ROLLING FRICTION LOSS $\bar{F}_{FR} \approx \frac{\bar{F}_W}{R_W} f_R$ ; $f_R = .010$ (FOR STEEL ON STEEL) RUSTY SURFACE

$$\bar{F}_{FR} = \frac{122{,}500}{15}\ .010 = \underline{81.67 \text{ LB}}/\text{WHEEL} = \underline{326.67} \text{ LB TOTAL (4 WHEELS)}$$

IN ADDITION, THERE COULD BE WAVYNESS IN THE TRACK WHICH WOULD REQUIRE ADDITIONAL RESISTANCE. ASSUMING THAT THIS MAY RESULT (WORST CASE ASSUMPTION) IN A MAXIMUM SLOPE OF $2 \times tg^{-1} \frac{.0075}{60} = .014^\circ$ AND THAT ALL 4 WHEELS WOULD ENCOUNTER THE SAME SLOPE SIMULTANEOUSLY.




$$\bar{F}_{FR} = 326.67 + 4 \times 122{,}500\ tg\ .014^\circ = \underline{446.4 \text{ LB}}$$

REFERRED TO AZ-AXIS: $\bar{M}_{tFR} = 446.4 \times 26.035 = \underline{11,622}$ FT LB

WHEEL BEARING FRICTION LOSS:

$\bar{M}_{tBG} = DL \times BG.RADIUS \times f_{BG}$ ; $f_{BG} = .0018$ FOR TAPERED ROLLER BG.

$\bar{M}_{tBG} = 490,000 \times .25 \times .0018 = \underline{220.5 \text{ FT LB}}$

REFERRED TO AZ-AXIS: $\bar{M}_{tBG} = 220.5 \times \frac{624.82}{30} = \underline{4,592}$ FT LB

RUNNING FRICTION @ AZ-AXIS = $F_{AZ}$ = 16,214 FT-LB

ADDING CABLE FRICTION & PINTLE BG. FRICTION, SAY $F_{AZ} = \underline{20,000}$ FT LB

AZIMUTH INERTIA: $I_{AZ} = 168.56 \times 10^6$ IN LB SEC$^2$ WITHOUT MOTORS (SEE PAGE 23)

$I_{AZ} = \underline{14.047 \times 10^6}$ SLUG-FT$^2$

ACCELERATION $= \frac{\omega}{t}$ ; $t_{ACC.}$ @ 45.9 MPH WIND = 2 SEC, $0 \div 90°$/MIN

$t_{ACC.}$ @ 60 MPH WIND = 4 SEC, $0 \div 60°$/MIN.

$a_1 = \frac{.0262 \text{ RAD/SEC}}{2} = \underline{.0131 \text{ RAD/SEC}^2} = .75°/\text{SEC}^2$

$a_2 = \frac{.0175 \text{ RAD/SEC}}{4} = \underline{.0044 \text{ RAD/SEC}^2} = .25°/\text{SEC}^2$

PEAK TORQUE REQ'D @ AZ-AXIS:

1) @ 45.9 MPH, $T_{45.9} = 361,310 + 20,000 + 184,016 = \underline{565,326}$ FT-LB

2) @ 60 MPH, $T_{60} = 617,387 + 20,000 + 61,807 = \underline{699,194}$ FT-LB

GEARBOX OUTPUT TORQUES:

A) DIRECT WHEEL DRIVE, $i = 450:1$

$$i_{AZ\ WHEEL/TRACK} = \frac{624.82}{30} = \underline{20.83:1}$$

$$M_t @ 45.9\ MPH \quad 1) = \frac{565,326}{20.83} = \underline{27,140}\ \text{FT-LB} \quad (2\ \text{UNITS})$$
$$M_t @ 60\ MPH \quad 2) = \frac{699,194}{20.83} = \underline{33,567}\ \text{FT-LB} \quad (2\ \text{UNITS})$$

INPUT TORQUE @ 80 % MINIMUM OVERALL EFFICIENCY:

$$M_{MOTOR\ 1)} = \frac{27,140}{2 \times .80}\ \frac{1}{450} = 37.69\ \text{FTLB} = \underline{452\ \text{INLB}} @ 2393\ \text{RPM}$$
$$M_{MOTOR\ 2)} = \frac{33,567}{2 \times .80}\ \frac{1}{450} = 46.62\ \text{FTLB} = \underline{559\ \text{INLB}} @ 1562\ \text{RPM}$$

B) DRIVE THROUGH BULLGEAR BUILT INTO WHEEL, $i = 100:1$

$$M_t @ 45.9\ MPH \quad 1) = \frac{565,326}{20.83}\ \frac{35}{150}\ \frac{1}{.98} = \underline{6,462}\ \text{FTLB}\ (2\ \text{UNITS})$$
$$M_t @ 60\ MPH \quad 2) = \frac{699,194}{20.83}\ \frac{35}{150}\ \frac{1}{.98} = \underline{7,992}\ \text{FT-LB}\ (2\ \text{UNITS})$$

$$M_{MOTOR\ 1)} = \frac{6,462}{2 \times .80}\ \frac{1}{100} = 40.39\ \text{FT-LB} = \underline{485\ \text{INLB}} @ 2232\ \text{RPM}$$
$$M_{MOTOR\ 2)} = \frac{7992}{2 \times .80}\ \frac{1}{100} = 49.95\ \text{FTLB} = \underline{599\ \text{INLB}} @ 1488\ \text{RPM.}$$

DRIVE INPUT POWER REQUIREMENTS: (BOTH MOTORS DRIVING, ZERO BUCKING)

$$A_{1)} = \frac{37.69 \times 2393}{5252} = \underline{16.81}\ \text{HP, MAX}$$
$$A_{2)} = \frac{46.62 \times 1562}{5252} = \underline{13.87}\ \text{HP MAX}$$
$$B_{1)} = \frac{40.39 \times 2232}{5252} = \underline{17.16}\ \text{HP, MAX}$$
$$B_{2)} = \frac{49.95 \times 1488}{5252} = \underline{14.15}\ \text{HP, MAX}$$

## HARMONIC DRIVE :

ASSUME 4 WHEELS ARE DRIVEN VIA BULL GEAR & PINION, GEARBOX RATIO IS 100:1, TOTAL AZ-RATIO IS $20.827 \frac{150}{35} 100 = 8,926:1$.

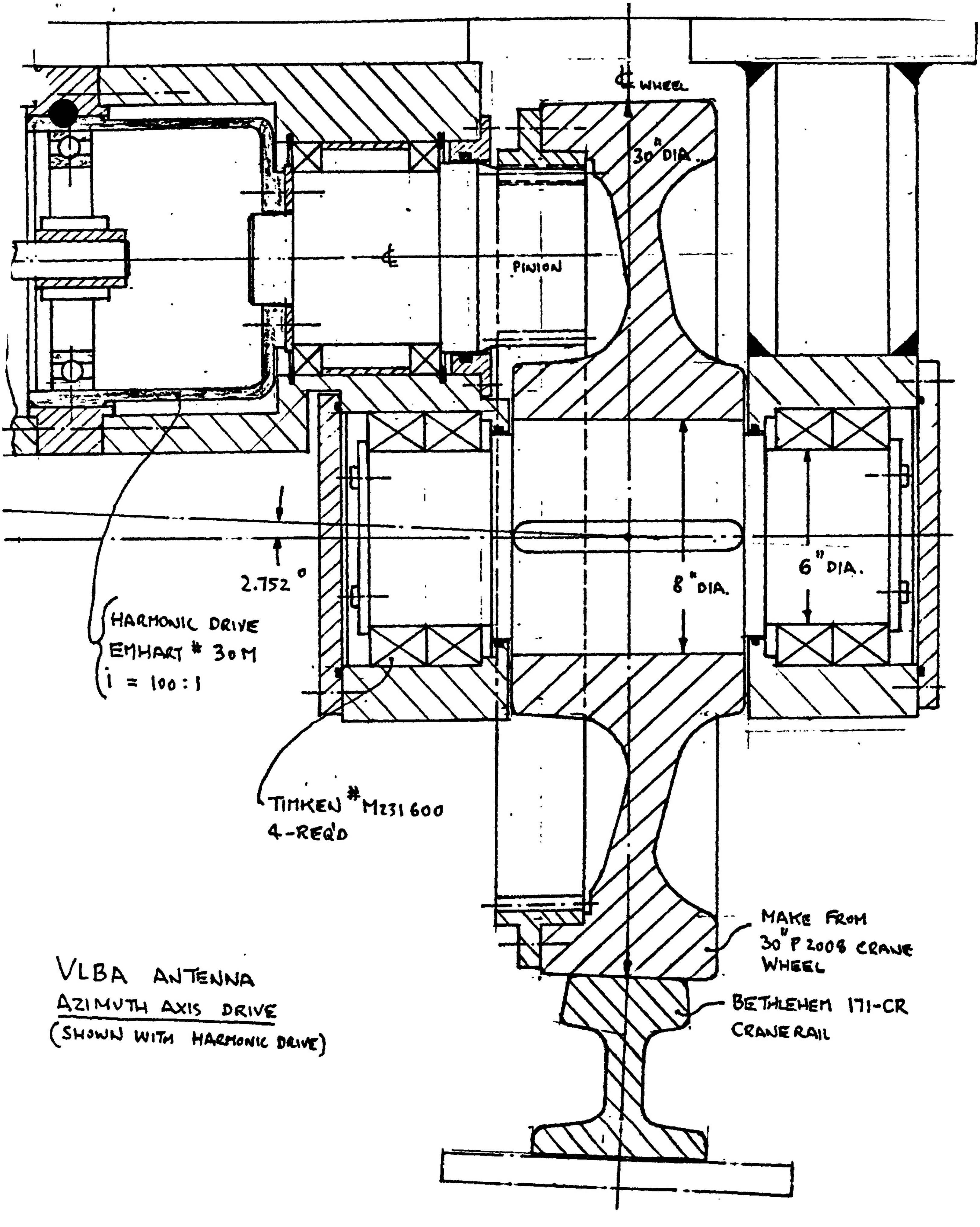
CHARACTERISTIC OF EMHART # HDC-30 M WITH 100.:1 SPEED RATIO :

| | | |
|---|---|---|
| RATED HP @ SERVICE FACTOR (1) | : | 9.6 HP |
| RATED OUTPUT TORQUE | : | 28,000 IN-LB (2,333 FT-LB) @ 1750 RPM |
| MAX. OUTPUT TORQUE | : | 33,000 IN-LB (2,750 FT-LB) @ 1750 RPM |
| FULL TORQUE & SPEED EFFICIENCY | : | 81 % |
| TORSIONAL SPRING RATE, MINIMUM (20% PRELOAD) | : | $7.5 \times 10^6$ IN-LB/RAD @ OUTPUT |
| " " , MAXIMUM | : | $15.0 \times 10^6$ " @ OUTPUT |
| INERTIA @ INPUT SHAFT | : | 158 LB-IN$^2$ |
| NO LOAD STARTING TORQUE | : | 20 IN-LB |
| B-10 LIFE | : | 5,000 HRS @ RATED SPEED & TORQUE |
| AVERAGE LIFE | : | 25,000 HRS " |
| MAXIMUM SPEED | : | 2,250 RPM |

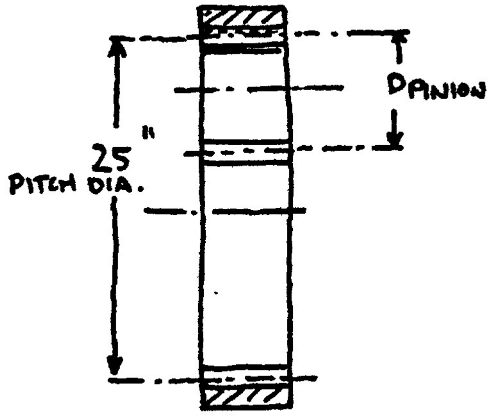
$N_{INPUT} = \frac{17.16}{2} = \underline{8.58\ HP}$ (VS. 9.6 RATED HP, OK)
@ 2232 RPM

DRIVE TO STOW TORQUE $= \frac{7,992}{4}\ 12 = \underline{24,000\ IN\text{-}LB}$ (OK) @ 1468 RPM
VS 28,000 IN-LB @ 1750 RPM (OK)

MAXIMUM INPUT SPEED IS 2250 RPM FOR THIS UNIT




VLBA ANTENNA
AZIMUTH AXIS DRIVE
(SHOWN WITH HARMONIC DRIVE)




LET : $n_T$ BULL GEAR = 150 TEETH

$n_T$ PINION = 35 "

DIAMETRAL PITCH = 6

PINION PITCH DIA = $35/6 = \underline{5.833}$ IN

$i$ BULL GEAR | PINION $= \frac{25}{5.833} = \frac{150}{35} = \underline{4.286:1}$

$n_{INPUT} = 100 \times \frac{150}{35} \times 20.627 \times .25 = \underline{2{,}232}$ RPM

$i_{AZ} = 100 \times \frac{150}{35} \times 20.627 = \underline{8925.86:1}$

$M_{t\,MAX} = \frac{i_{AZ}}{i_{DRIVE}} \frac{33{,}000}{12} 4 = \frac{8{,}925.86}{100} \frac{33{,}000}{12} 4 = \underline{981{,}945}$ FT LB

TORQUE CAPABILITY

VS. 699,194 FT LB APPLIED

USE MAX. TORQUE CAPABILITY OF 33,000 IN LB TO SIZE GEARS !

10/6/83
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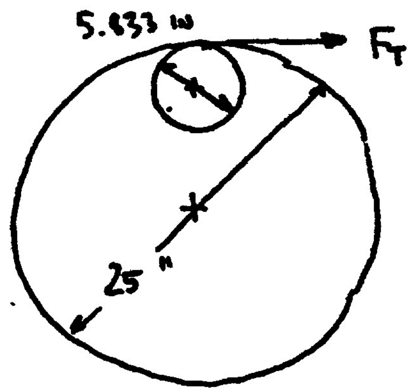
## CHECK GEARING




$$F_{T\,MAX} = 33{,}000 \; \frac{2}{5.833} = \underline{11{,}315 \text{ LB}}$$

LET $b$ = FACE WIDTH = 3.0 IN

$C$ = VELOCITY FACTOR = $\dfrac{600}{600 + V_m}$

$V_m$ = PITCH LINE VELOCITY = $\dfrac{2{,}231.5}{100} \; \dfrac{5.833\,\pi}{12} = \underline{34 \text{ FT/MIN.}}$

$$C = \frac{600}{634} = \underline{.946}$$

$$S_0 = \frac{F_t \, P_D}{C\, b\, Y} \quad ; \quad Y = \text{LEWIS FACTOR} = .373 \text{ FOR } 20^\circ$$

$$S_0 = \frac{11{,}315 \times 6}{.946 \times 3 \times .373} = \underline{64{,}133} \text{ LB/IN}^2$$

(OK FOR HIGH STRENGTH STEEL SAE 4140 HEAT TREATED TO 210 ÷ 245 BRINELL HARDNESS)

## CHECK "K" FACTOR : (RE: PRACTICAL GEAR DESIGN, DUDLEY)

$$Q = \frac{N}{RPM_{PIN.}} \; \frac{(i_{GEAR/PINION} + 1)^3}{i_{G/P}} = \frac{9.67}{22.32} \; \frac{(4.29+1)^3}{4.29} = \underline{14.95}\,^*$$

$$K = \frac{F_T}{b\, D_{PIN.}} \; \frac{(i_{G/P} + 1)}{i_{G/P}} = \frac{11{,}315}{3 \times 5.833} \left(\frac{5.29}{4.29}\right) \approx \underline{800} \text{ (OK)}$$

* [illegible] ≃ [illegible] LB FOR K = 500 (OK)

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ALLOWABLE "K" FACTOR = 1500 FOR AUTOMOTIVE GEAR (LOW GEAR) HAVING A SURFACE HARDNESS OF 58 RC.

STIFFNESS OF GEAR MESH :

$$\frac{F_T}{Y} \cong b \times 2\times10^6 = 3.0 \times 2 \times 10^6 = \underline{6\times10^6} \text{ LB/IN}$$

$$\frac{M_T}{\theta} = \frac{F_T}{Y}\left(\frac{D_{PIN}}{2}\right)^2 = 6\times10^6\left(\frac{5.833}{2}\right)^2 = \underline{5.1\times10^7} \text{ IN LB/RAD @ OUTPUT}$$

STIFFNESS OF PINION SHAFT :

EFFECTIVE LENGTH OF SHAFT : 8.5 IN (SCALED)

EFFECTIVE DIA. OF SHAFT : 6.0 IN

$$\frac{M_T}{\theta} = \frac{.5\, R^4 \pi\, G}{L_E} = \frac{.5\,(3)^4\pi\; 11.6\times10^6}{8.5} = \underline{1.74\times10^8} \text{ IN LB/RAD}$$

TOTAL STIFFNESS OF HARMONIC DRIVE & PINION|GEAR|SHAFT :

$$\frac{1}{K_{TOTAL}} = \frac{1}{K_{HD}} + \frac{1}{K_{GM}} + \frac{1}{K_{SHAFT}}$$

$$K_{TOTAL} = \left[(7.5\times10^6)^{-1} + (5.1\times10^7)^{-1} + (1.74\times10^8)^{-1}\right]^{-1}$$

$$K_{TOTAL} = 6.3\times10^6 \text{ IN LB/RAD}$$

$K_{TOTAL}$ REFERED TO ANTENNA AXIS :

$$K_{TOTAL_{AZ}} = 4\times6.3\times10^6\times\Big(\overbrace{20.83\times\frac{150}{35}}^{89.27}\Big)^2 = \underline{2.01\times10^{11}} \text{ IN LB/RAD}$$

10/17/67
ORH

MOMENT OF INERTIA REFERED TO AZ-AXIS:

$$I_{AZ} = 14.54 \times 10^9 \text{ LB-IN}^2 = \underline{37.67 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{\text{GEAR BOXES (HARM. DRIVE)}} = 154 \times 4 \times 8{,}925.86^2 \frac{1}{386} = \underline{130.45 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{\text{MOTORS}} = 4 \times \overset{\text{ASSUMED}}{(3.0)} \, 144 \frac{1}{386} (8{,}925.86)^2 = \underline{356.66 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{\text{PINIONS}} = 4 \times 414 \frac{1}{386} (89.21)^2 = \underline{.034 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{\text{WHEELS}} = 4 \left( \frac{800 \times 15^2}{2} \right) \frac{1}{386} \, 20.83^2 = \underline{.405 \times 10^6} \text{ LB-IN-SEC}^2$$

DISK + GEARBOXES + MOTORS + PINIONS + WHEELS

$$I_{AZ,\text{TOTAL}} = (37.67 + 130.45 + 356.66 + .034 + .405) 10^6 = \underline{525.22 \times 10^6} \text{ LB IN SEC}^2$$

$$f_n \cong \frac{1}{2\pi} \sqrt{\frac{K}{I}} \cong \frac{1}{2\pi} \sqrt{\frac{2.01 \times 10^{11}}{5.252 \times 10^8}} \cong \underline{3.11 \text{ CPS}}$$

VS. IN COMPARISON: (FOR VLA ANTENNA)

$$I_{AZ} = 11.25 \times 10^9 \text{ LB-IN}^2 = \underline{29.145 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{\text{GEARBOXES}} = 2 \times 1.34 \times 10^9 \frac{1}{386} = \underline{6.94 \times 10^6} \text{ LB IN SEC}^2$$

$$I_{\text{MOTORS}} = 2 \times 1.36 \times 144 \frac{1}{386} \times (10{,}342.5)^2 = \underline{108.54 \times 10^6} \text{ LB-IN-SEC}^2$$

$$I_{AZ,\text{TOTAL}} = (29.145 + 6.94 + 108.54) 10^6 = \underline{144.63 \times 10^6} \text{ LB-IN-SEC}^2$$

$$K_{AZ} = 2 \times 5 \times 10^6 \times 12 \times (10.5)^2 = \underline{1.32 \times 10^{10}} \text{ IN LB/RAD}$$

$$f_n \cong \frac{1}{2\pi} \sqrt{\frac{1.32 \times 10^{10}}{144.63 \times 10^6}} \cong 1.52 \text{ CPS}$$

$f_n$ (FOR 2 MOTOR/DRIVES DRIVING WHEEL AXLES):

$I_{AZ} = 14.54 \times 10^9$ LB-IN$^2$ $= \underline{37.67 \times 10^6}$ LB IN SEC$^2$

$I_{GEARBOXES}$ (ESTIMATED) $= \underline{10 \times 10^6}$ LB IN SEC$^2$

$I_{MOTORS} = 2(.037)12 \times 9372^2$ $= \underline{78 \times 10^6}$ LB IN SEC$^2$

15 HP @ 2500 RPM
FRAME $ 812 A

$I_{WHEELS} =$ $= \underline{.405 \times 10^6}$ LB IN SEC$^2$

$I_{AZ} = (37.67 + 10.0 + 78.0 + .405)10^6 =$ $\underline{126.1}$ LB-IN-SEC$^2$ ($\times 10^6$)

$K = 20.827^2 \times 12 \times 10^7 = \underline{5.205 \times 10^{10}}$ IN LB/RAD

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{I}} = \frac{1}{2\pi}\sqrt{\frac{5.205 \times 10^{10}}{126.1 \times 10^6}} = \underline{3.23 \text{ CPS}}$$

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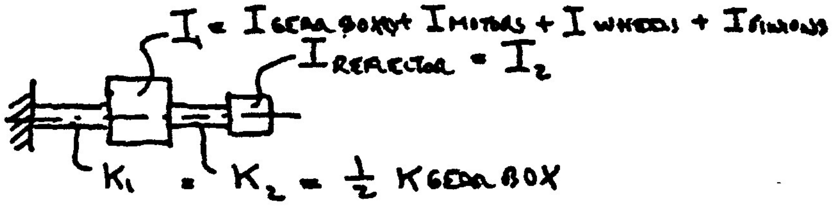
A MORE REALISTIC MODEL:



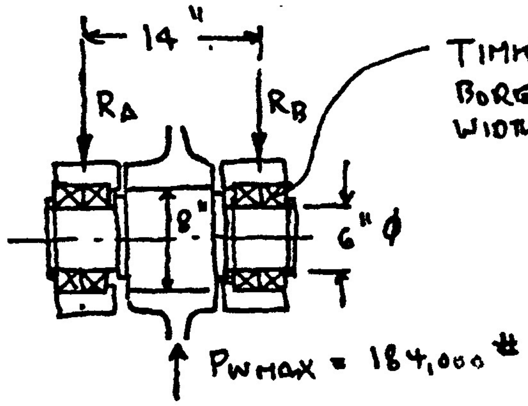

$$f_{n\,AZ} = \frac{1}{2^{1.5}\pi}\left\{\left(\frac{K_1}{I_1} + \frac{K_2}{I_1} + \frac{K_2}{I_2}\right) \pm \left[\left(\frac{K_1}{I_1} + \frac{K_2}{I_1} + \frac{K_2}{I_2}\right)^2 - \frac{4K_1K_2}{I_1I_2}\right]^{\frac{1}{2}}\right\}^{\frac{1}{2}}$$

$$f_{n\,AZ} = \frac{1}{2^{1.5}\pi}\left\{\overset{3216.09}{\left(\frac{1.05\times10^{11}}{467.55\times10^{6}} + \frac{1.05\times10^{11}}{467.55\times10^{6}} + \frac{1.05\times10^{11}}{37.67\times10^{6}}\right)} \pm \left[(3{,}216.09)^2 - \frac{4\,(1.05\times10^{11})^2}{(467.55\times37.67)\,10^{12}}\right]^{\frac{1}{2}}\right\}^{\frac{1}{2}}$$

$f_{n\,AZ}$ = <u>2.20 CPS</u> (FOR NEW DESIGN) THIS IS (OK)

VS. $f_{n\,AZ}$ = <u>2.10 CPS</u> (FOR VLA) VS. 2.15 CPS MEASURED

<u>WHEEL AXLE & BEARINGS:</u>




TIMKEN TAPERED ROLLER BEARING # M231600
BORE : 6.0 IN ; O.D : 8.750 IN
WIDTH : 1.8437 , BLR = 15,000# @ 500 RPM

$R_A = R_B = \frac{P_{W\,MAX}}{2} = \underline{92{,}000}$ LB

LOAD ON EACH OF 4 BEARINGS $= \frac{R_A}{2} = \frac{R_B}{2} = \underline{46{,}000}$ LB

SPEED FACTOR , $SF = \left(\frac{500}{n}\right)^{.3} = \left(\frac{500}{5.21}\right)^{.3} = \underline{3.93}$

BASIC LOAD RATING @ 5.21 RPM = 15,000 × 3.93 = <u>58,950</u> LB VS.
46,000 APPLIED , @ $B_{10}$ LIFE = 3000 HRS. (OK)

STRESS IN AXLE:

$$M_{B_{MAX}} = P_{W_{MAX}} \frac{\ell}{4} = 184{,}000 \frac{14}{4} = \underline{644{,}000} \text{ IN LB}$$

$$Z_B = \frac{I}{R} = \pi \frac{R^3}{4} = \pi \frac{4^3}{4} = \underline{50.27} \text{ IN}^3$$

$$S_{B_{MAX}} = \frac{644{,}000}{50.27} = \underline{12{,}810} \text{ LB/IN}^2 \quad \text{(OK)}$$

TORSIONAL STRESS IN AXLE IF DIRECTLY DRIVEN:

$$M_{t_{MAX}} = \frac{29{,}999}{2} 1.10 = 16{,}500 \text{ FT LB} \cong \underline{198{,}000 \text{ IN LB}} \text{ @ 10\% BUCKING}$$

$$\cong 18{,}000 \text{ " } \cong \underline{216{,}000 \text{ IN LB}} \text{ @ 20\% "}$$

$$S = \frac{\pi D^3}{16} = \frac{\pi 6^3}{16} = \underline{42.41 \text{ IN}^3}$$

$$S_{T_{MAX}} = \frac{216{,}000}{42.41} = \underline{5{,}093} \text{ LB/IN}^2$$ (OK) AXLE MAY BE DRIVEN DIRECTLY.

MINIMUM INPUT SPEED @ SIDEREAL TRACKING:

$$i_{TOTAL} = 8{,}925.86 : 1$$

$$n_{OUTPUT} = 1 \text{ RPD} = \frac{1}{24 \times 60} = \underline{6.94 \times 10^{-4}} \text{ RPM}$$

$$n_{INPUT} = i_{TOTAL} \times n_{OUTPUT} = 8{,}925.86 \times 6.94 \times 10^{-4} = \underline{6.2 \text{ RPM}}$$

$$\text{DRIVE SPEED RANGE} = \frac{.25}{6.94 \times 10^{-4}} = \underline{360 : 1}$$

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MINIMUM TORQUE REQUIRED @ ZERO WIND VELOCITY
(ABOUT AZ-AXIS)

$T_{1\ MIN.} = M_{t\ FR} + M_{t\ a} = 4,817 + 7,975 = \underline{12,792}$ FT LB

ASSUMING 80 % BUCKING AT THIS CONDITION TO GET HIGHER SPEED REDUCER EFFICIENCY (SEE PAGE 10 OF EMHART BROCHURE):

$T_{1\ MIN\ DRIVE} = 1.8 \times 12,792 \cong \underline{23,026}$ FT LB

REFERED TO PINION SHAFT @ i = 89.27 : 1 @ .98 % EFFICIENCY OF BULLGEAR / PINION MESH :

$T_{MIN\ PINIONS} = \dfrac{23,026 \times 12}{89.27 \times .98} = \underline{3,158}$ IN LB VS. 28,000 IN LB RATED TORQUE FOR UNITS

$\dfrac{T_{MIN\ PINION}}{T_{RATED}} = \dfrac{3,158}{2 \times 28,000} = \underline{.06}$ (6 %)

EFFICIENCY IS ONLY ~ 20 % AT THIS POINT

ASSUMING THAT BUCKING TORQUE IS MUCH HIGHER SO THAT UNIT WOULD BE OPERATED MORE OR LESS AT CONSTANT TORQUE REGARDLESS OF SPEED OR TORQUE DEMAND :

LET DRIVE TORQUE @ AZ-AXIS $= 28,000 \times 2 \times .98 \times 89.27 \times \frac{1}{12} = \underline{425,095}$ FT LB

BUCKING TORQUE $= 425,095 - 12,792 = \underline{412,303}$ FT LB
(97 %)

DRIVE EFFICIENCY WOULD BE AT LEAST 61 %.

AT 50 % RATED TORQUE = 14,000 X 2 X .96 X 69.27 X $\frac{1}{12}$ = 204,131 FT LB

BACKING TORQUE = 204 131 - 12,792 = 191,339 FTLB
(94 %)

EFFICIENCY WOULD BE ABOUT 80 % (SEE PAGE 10 OF EMHART BROCHURE)

SUMMARY :

- APPLICATION OF HARMONIC DRIVE GEAR TRAINS FOR THE AZIMUTH DRIVE OF THE VLBA ANTENNAS APPEARS TO BE FEASIBLE, PROVIDING THE SERVO LOOP CAN BE SUCCESSFULLY CLOSED. THIS REQUIRES SOME FURTHER EVALUATION BY ELECTRO SPACE.

- 4 DRIVES WOULD BE REQUIRED (ONE ON EACH WHEEL) WITH GEAR/PINNION ARRANGEMENT AS SHOWN ON PAGE 19)

- THE UNITS MAY HAVE TO BE OPERATED AT NEAR CONSTANT TORQUE TO ACHIEVE GOOD DRIVE EFFICIENCIES.

- THE COMPONENT GROUP COSTS $ 2210.-, FOR THE 30M DRIVE, IN QUANTITIES OF 25 ÷ 49. THIS RELATIVELY LOW COST MAY MAKE THE APPLICATION OF HARMONIC DRIVES ATTRACTIVE IN THIS CASE, ALTHOUGH THIS MUST BE TRADED OFF AGAINST THE INCREASE IN COST DUE TO THE ADDITIONAL SERVO DRIVE PACKAGE REQUIRED FOR 4 DRIVES WHICH WOULD COST ABOUT $ 120,000.-

Systems Development Laboratory

ENGINEERING CONSULTANTS

13372 CALLE COLINA

POWAY, CALIFORNIA 92064

~~[illegible]~~ 485-5657
619

Mr. Michael J. Hardiman
Manager, Special Products Division
PHILADELPHIA GEAR CORPORATION
King of Prussia, PA 19406

October 12, 1983

Dear Mr. Hardiman:

Attached are preliminary Azimuth Axis drive requirements for the VLBA Antenna as discussed with you during our recent telcon on the 10th of October.

Please review these requirements and indicate the option you would prefer and also provide the needed information as indicated on sheet 28 at your earliest convenience.

Very truly yours,

SYSTEMS DEVELOPMENT LABORATORY

Otto R. Heine

P.S: THE ELEV. GEARBOXES WOULD BE IDENTICAL TO THE VLA UNITS WITH EXCEPTION OF THE MAXIMUM "SLEW TO STOW" OUTPUT TORQUE REQUIREMENT WHICH WILL BE FOR THE VLBA:

MAX. WIND MOMENT @ 60MPH + 1cm ICE UNBALANCE @ 120° ELEVATION ANGLE

$$M_{t\ MAX} = \frac{1{,}021{,}137}{2(.98)\,36.71}(12) = \underline{170{,}300}\ \text{IN-LB}$$

@ $n$ = 774 RPM INPUT SPEED ($\frac{1}{36}$ RPM @ ELEV. AXIS = 10°/MIN.)

MAX. SLEW SPEED IS 2322 RPM (INPUT) FOR VLBA; TOTAL ELEV. GEAR RATIO IS 27,868 : 1

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# VLBA ANTENNA

## AZIMUTH AXIS DRIVE REQUIREMENTS :

HELICAL OR PLANETARY GEAR UNITS

4 OR 2 UNITS , (DRIVEN VIA DC-SERVO MOTORS WITH 2500 RPM MAX. INPUT SPEED)

| | | |
|---|---|---|
| AZ-AXIS SPEED RANGE | : | SIDEREAL TO .25 RPM (360 : 1) |
| MAX. TORQUE @ AZ-AXIS (TRACKING): | | 374,102 FT-LB @ .25 RPM |
| MAX. TORQUE @ AZ-AXIS (DRIVE TO STOW): | | 624,883* FT-LB @ .167 RPM |
| TYPE OF DRIVE ARRANGEMENT | : | BIASING, ANTI-BACKLASH (BUCKING) |
| WHEEL / TRACK RATIO | : | 20.827 : 1 (30 IN WHEELS ON 624.82 IN TRACK) |

## OPTION 1 :

DRIVE THROUGH INTERNAL BULL GEARS BUILT INTO 30 IN WHEELS

| | | |
|---|---|---|
| NO. OF GEAR TRAINS | : | 4 |
| PITCH DIA. OF BULL GEARS (INTERNAL GEAR, BUILT INTO WHEELS) | : | 25 IN ; $P_D$ = 6 ; $n_{TEETH}$ = 150 |
| PITCH DIA. OF PINIONS | : | 5.833 IN ; $n_{TEETH}$ = 35 |
| RATIO BULLGEAR / PINION | : | 4.286 : 1 |
| RATIO OF SPEED REDUCERS | : | 100 : 1 |
| TOTAL RATIO | : | 8,925.86 : 1 |
| MAX. INPUT SPEED | : | 2232 RPM (@ .25 RPM OUTPUT SPEED) |
| MAX. INPUT POWER (EACH UNIT) | : | 11 HP @ 2232 RPM WITH 20% BUCKING TORQUE |
| SPRING CONSTANT OF SPEED REDUCER | : | $7.5 \times 10^6$ IN LB/RAD (MINIMUM) REFERRED TO OUTPUT SHAFT |

NOTE : RATIO MAY BE CHANGED TO ACHIEVE A MAX. INPUT SPEED OF 2500 RPM.

## OPTION 2 :

SAME AS OPTION 1, WITH EXCEPTION OF THE FOLLOWING :

| | | |
|---|---|---|
| NO. OF GEAR TRAINS | : | 2 |
| MAX INPUT POWER (EACH UNIT) | : | 22 HP @ 2232 RPM WITH 20% BUCKING TORQUE |
| SPRING CONSTANT OF SPEED REDUCER | : | $15 \times 10^6$ IN LB/RAD (MINIMUM) REFERRED TO OUTPUT SHAFT |

10/11/83
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OPTION 3:

DRIVE WHEELS DIRECTLY

| | | |
|---|---|---|
| NO OF GEAR TRAINS | : | 4 |
| RATIO OF SPEED REDUCERS | : | 450 : 1 |
| TOTAL RATIO | : | 9,372.15 : 1 |
| MAX. INPUT SPEED | : | 2343 RPM |
| MAX. INPUT POWER (EACH UNIT) | : | 11 HP @ 2343 RPM W. 20% BUCKING TORQUE |
| SPRING CONSTANT OF SPEED REDUCER | : | $3.0 \times 10^7$ IN LB/RAD. (MINIMUM) REFERED TO OUTPUT SHAFT. |

NOTE: RATIO MAY BE CHANGED TO ACHIEVE A MAX. INPUT SPEED OF 2500 RPM

OPTION 4:

SAME AS OPTION 3 WITH EXCEPTION OF THE FOLLOWING:

| | | |
|---|---|---|
| NO. OF GEAR TRAINS | : | 2 |
| MAX. INPUT POWER/UNIT | : | 22 HP @ 2343 RPM W. 20% BUCKING TORQUE |
| SPRING CONSTANT OF SPEED REDUCER | : | $6.0 \times 10^7$ IN LB/RAD (MINIMUM) REFERED TO OUTPUT SHAFT |

NOTE: *

WHEN DRIVING TO STOW, BUCKING IS NOT REQUIRED, THUS ALL 4 (OR 2 RESPECTIV.) UNITS SHARE THE LOAD.

INFORMATION NEEDED:

1. OUTLINE DIMENSIONS, SHAFT DIMENSIONS, MOUNTING ARRANGEMENT.
2. WEIGHT
3. APPROX. COST FOR 40 (RESPECTIVELY 20) UNITS
4. CHECK BULLGEAR/PINION DIMENSIONS; DETERMINE FACE WIDTH REQUIRED BASED ON HIGH STRENGTH STEEL & 58 RC SURFACE HARDNESS.
5. APPROX. COST OF BULLGEAR & DRIVE PINION.

10/12/83
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# VLBA ANTENNA

## ELEVATION DRIVE REQUIREMENTS:

CONSIDER VLA - ELEVATION DRIVE :

| | | |
|---|---|---|
| GEAR BOX RATIO | : | 564 :1 (541.077) |
| BULL GEAR / PINION RATIO | : | 36.71 :1 (36.706) |
| ELEVATION GEAR RATIO | : | 20,704 :1 OVERALL (19,860.71) |
| BULL GEAR DIA. (PITCH DIA) | : | 312 IN |
| PINION PITCH DIA | : | 8.5 IN |
| DIAMETRAL PITCH, P | : | 2 |
| GEAR BOX SPRING CONSTANT | : | $3.5 \times 10^6$ FT LB / RAD @ OUTPUT |
| GEAR BOX INERTIA | : | $7.0 \times 10^9$ LB-IN$^2$ @ ELEV. AXIS |

VLBA ELEVATION DRIVE : (ASSUME VLA GEARBOX IS UTILIZED)

| | | |
|---|---|---|
| BULL GEAR PITCH DIA | : | 420 IN |
| MAX. AXIS SPEED | : | $\frac{1}{12}$ RPM (30°/MIN) |
| BULL GEAR / PINION RATIO | : | 49.41 :1 $\left(\frac{420}{8.5}\right)$ |
| ELEVATION GEAR RATIO | : | 27,868.24 :1 (49.41 x 564) 26,736 :1 |
| MAX. INPUT SPEED | : | 2,322 RPM (OK VS. 2500 ALLOWABLE) 2226 RPM |

1) MAX. WIND MOMENT @ EL-AXIS @ 45.9 MPH, $\theta = 120°$, $M_{W_1}$ = 361,310 FT LB
2) MAX. " " @ 60 MPH, $\theta = 120°$, $M_{W_{MAX}}$ = 617,387 "

ICE UNBALANCE MOMENT : $9.69 \times 10^6$ IN LB @ $\theta = 0$, $M_I$ = 403,750 " @ $\theta = 120°$

MAX. MOMENT DUE TO WIND @ 60 MPH + 1 IN ICE, $M_{MAX}$ = 1,021,137 FT LB
(UNIT IS DRIVEN VIA. BOTH DRIVE UNITS AT THIS POINT)
@ 10°/MIN.

$N_{INPUT(1)} = \frac{361,310}{5252} \frac{1}{12} \frac{1}{.95} 1.2 = \underline{7.24 \text{ HP}}$, AT NORMAL OPERATING CONDITION, ASSUMING AN OVERALL GEARTRAIN EFFICIENCY OF 95% AND BIAS TORQUE OF 20%.

$N_{INPUT(2)} = \frac{1,021,137}{2 \times 5252} \frac{1}{36} \frac{1}{.95} = \underline{2.84 \text{ HP}}$, AT DRIVE TO STOW OPERATING CONDITION (60 MPH & 10°/MIN.)

INERTIA @ ELEV. AXIS (ADV. DES.) : $I_{ELEV} = 15.44 \times 10^9$ LB-IN$^2$

$= \underline{40 \times 10^6}$ LB-IN-SEC$^2$

$I_{GEARBOXES} = 2 \times 7.0 \times 10^9 \left(\frac{27,868}{20,704}\right)^2 \frac{1}{386} = \underline{65.7 \times 10^6}$ LB-IN-SEC$^2$

$(2 \times .037 \times 12) \times 27,868^2 = \underline{689.6} \times 10^6$ LB IN SEC$^2$ W. 15 HP @ 2500 RPM, FRAME B-1612 ATZ

$I_{MOTORS} = 2 \times 1.36 (144) \frac{1}{386} (27,868)^2 = \underline{748.1} \times 10^6$ LB-IN-SEC$^2$

REFLECTOR GEARS MOTORS

$I_{ELEV.\,TOTAL} = (40.0 + 65.7 + 748.1) 10^6$ = 795.3 $\times 10^6$ / $\underline{893.8 \times 10^6}$ LB-IN-SEC$^2$

$K_{ELEV.} = 2 \times 3.5 \times 10^6 \times 12 \times 49.41^2 = \underline{2.05 \times 10^{11}}$ IN LB/RAD

$f_n \approx \frac{1}{2\pi} \sqrt{\frac{K}{I}} = \frac{1}{2\pi} \sqrt{\frac{2.05 \times 10^{11}}{893.8 \times 10^6}}$ = 2.55 CPS (OR) $\underline{2.41 \text{ CPS}}$

VS. IN COMPARISON: (FOR VLA ANTENNA)

$I_{ELEV} = 8.85 \times 10^9 \frac{1}{386} = \underline{22.93 \times 10^6}$ LB-IN-SEC$^2$

$I_{GEARBOXES} = 2 \times 7.0 \times 10^9 \frac{1}{386} = \underline{36.3 \times 10^6}$ LB-IN-SEC$^2$

$I_{MOTORS} = 2 \times 1.36 \times 144 \frac{1}{386} (20,704)^2 = 435 \times 10^6$ LB-IN-SEC$^2$

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$$I_{ELEV.\ TOTAL} = (22.93 + 36.3 + 435.0)10^6 = \underline{494.23 \times 10^6}\ \text{LB.IN.SEC}^2$$

$$K_{ELEV.} = 2 \times 3.5 \times 10^6 \times 12 \times 36.71^2 = \underline{1.13 \times 10^{11}}\ \text{INLB/RAD}$$

$$f_n \cong \frac{1}{2\pi}\sqrt{\frac{1.13 \times 10^{11}}{494.23 \times 10^6}} \cong \underline{2.41\ \text{CPS}}\ (\text{FOR VLA})$$

THIS IS IN GOOD AGREEMENT WITH $f_n \cong 2.41$ CPS CALCULATED FOR THE VLBA ADVANCED DESIGN.

HENCE, THE VLA - GEARBOXES ARE CONSIDERED SUITABLE FOR THE ELEVATION DRIVE OF THE VLBA ANTENNA.

<u>MORE ACCURATELY</u>:

$$f_n = \frac{1}{2^{1.5}\pi}\left\{\left(\overbrace{\frac{K_1}{I_1} + \frac{K_2}{I_2} + \frac{K_2}{I_2}}^{(a)}\right) \pm \left[(a)^2 - \frac{4K_1K_2}{I_1I_2}\right]^{\frac{1}{2}}\right\}^{\frac{1}{2}}$$

$$f_n = \frac{1}{2^{1.5}\pi}\left\{\left(\overbrace{\frac{1.025\times10^{11}}{853.6\times10^6} + \frac{1.025\times10^{11}}{853.6\times10^6} + \frac{1.025\times10^{11}}{40\times10^6}}^{2802.6}\right) \pm \left[2802.6^2 - \frac{4\times(1.025\times10^{11})^2}{(853.6\times40)10^{12}}\right]^{\frac{1}{2}}\right\}^{\frac{1}{2}}$$

$$f_{n_{ELEV}} = \underline{1.70\ \text{HZ}}$$

LIKEWISE, $f_{n_{ELEV}}$ FOR VLA $= \underline{1.70\ \text{HZ}}$

AGAIN, GOOD AGREEMENT (OK)

VS. 2.07 HZ ACTUALLY MEASURED FOR VLA ANTENNAS.

THE ACTUAL LOCKED ROTOR RESONANT FREQUENCY WILL PROBABLY BE AROUND 2.1 HZ FOR THE VLBA ADVANCED DESIGN.

ELECTROSPACE SYSTEMS, INC.

83-RNW-0242
18 October 1983

Mr. Otto R. Heine, Consultant
13372 Calle Colina
Coway Roway, California 92064

Attention: Mr. Otto R. Heine

Subject: VLBA

Reference: O. Heine/R. White Telecon of 10 Oct 83

Gentlemen:

The following information is enclosed in support of your VLBA study efforts:

1. Reliance Electric DC Motor Dimension Sheet
2. Stearns Brakes Series 87,000 Data Sheet
3. Proposal for A Standard A Antenna Control System with Linear DC Drives

The DC motors can be purchased for C-Face or D-Flange mounting. The 1750 RPM motors are currently optimum in terms of availability and price. We have encouraged most designs around them, but we also have recent applications with 1150 and 2500 RPM motors. In this VLBA case, the 2500 RPM base speed would increase the drive-train ratio and help on the minimum smooth velocity characteristic. As we discussed, the Reliance motors with the special options included with our orders have proven to be excellent torque sources for antenna drives.

The oil filled helical gear main speed reducers used on VLA have provided the best compliance and minimum smooth velocity characteristics of the various approaches that we have experienced. The oil shear losses are higher in these reducers at high speed, but their other advantages justify them overall.

In addition to the servo interface with the radio telescope drive train, another critical interface is the axis position pick-off design. Assuming an accurate position transducer is used, its mechanical tie-in with respect to the true antenna boresight is very critical. Making sure no or little main load paths are involved in the pick-off attachment design has historically been difficult.

The proposal enclosed addresses an INTELSAT Standard A commercial communication antenna control system (32 Meter diameter antenna operating at 4/6 GHZ). The control system hardware presented would be applicable to this VLBA requirement. Photographs and block diagrams depict our existing designs. We are an engineering house and can design to meet most requirements, but taking advantage of as much as possible of our present equipment will keep the costs minimized.

Please contact Don Heitzman o~ myself if we can be of further assistance.

Very truly yours,

ELECTROSPACE SYSTEMS, INC.

Ralph N. White
Divison Manager,
Antenna and Control Systems

RNW/djm

Enclosures

Systems Development Laboratory
ENGINEERING CONSULTANTS
13372 CALLE COLINA
POWAY, CALIFORNIA 92064
(714) 485-5657

Mr. Ralph N. White
Division Manager
Antenna and Control Systems
ELECTROSPACE SYSTEMS, INC.
Box 1359, Richardson, Texas 75080

21 October 1983

Dear Mr. White:

Thank you for the information you supplied with your letter of 18 October.

Attached are the presently anticipated VLBA drive requirements, the applicable operating conditions and environmental conditions, along with pertinent information on the Harmonic Drive (AZ option A) to be considered. Please review this material and provide the following information, needed for budgetary purpose and to prepare realistic design specifications, at your earliest convenience.

1. Recommended DC-servo motors for AZ- option A and or option B, and EL axes. Please include specific information on motor characteristics, brakes and tachometers.

2. Estimated performance of Servo/Drive system under the various operating conditions. Please provide supporting engineering calculations where applicable.

3. Estimated cost of control system hardware excluding, but interfacing with data system, considering the VLA control system hardware requirements as a guideline. Please consider that 10 Telescopes would be required, as a minimum.

A system description will not be necessary if the control system hardware described in your INTELSAT Standard A commercial communication antenna control system would indeed be applicable to the VLBA requirements.

Very truly yours,

SYSTEMS DEVELOPMENT LABORATORY

Otto R. Heine

cc. William G. Horne, NRAO

# VLBA - OPERATING CONDITIONS

| ITEM | NOMENCLATURE | CONDITION | REQUIREMENT |
|---|---|---|---|
| 1 | PRECISION I<br>FULL ACCURACY | WIND VELOCITY (NO ICE) | 13.42 ± 2.24 MPH (GUST) |
| 2 | PRECISION IIA<br>REDUCED ACCURACY | " " | 40.26 ± 5.59 MPH (GUST) |
| 3 | PRECISION IIB<br>REDUCED ACCURACY | " " | 53.69 MPH (NO GUST) |
| 4 | DRIVE TO STOW | " + 1 cm ICE | 60 MPH |
| 5 | AXIS ACCELERATION | PRECISION I, IIA, AZIMUTH | .0131 RAD/SEC² |
| 6 | AXIS ACCELERATION | " I, IIA,B, ELEVATION | .0044 RAD/SEC² |
| 7 | AXIS ACCELERATION | PRECISION IIB, AZIMUTH | .0044 RAD/SEC² |
| 8 | AXIS ACCELERATION | " , ELEVATION | .0044 RAD/SEC² |
| 9 | AXIS SPEED | PRECISION I, IIA, AZIMUTH | .25 RPM, MIN. |
| 10 | AXIS SPEED | PRECISION I, IIA,B, ELEVATION | .0833 ($\frac{1}{12}$) RPM, MIN. |
| 11 | AXIS SPEED | PRECISION IIB, AZIMUTH | .167 ($\frac{1}{6}$) RPM, MIN. |
| 12 | AXIS SPEED | DRIVE TO STOW, ELEVATION | .0278 ($\frac{1}{36}$) RPM, MIN. |
| 13 | SKY COVERAGE | TRAVEL LIMITS | ± 270° AZ, 5° - 125° ELEV. |

## ENVIRONMENTAL CONDITIONS

| | | | |
|---|---|---|---|
| a) | TEMPERATURE RANGE | PRECISION I | 0°F – 86°F |
| b) | TEMPERATURE RANGE | PRECISION IIA & B | -22°F – 104°F |
| c) | RELATIVE HUMIDITY | PRECISION I | 0 – 50% |
| d) | RELATIVE HUMIDITY | PRECISION IIA, B & DRIVE TO STOW | 0 – 98% |
| e) | RAINFALL | PRECISION IIA, B & DRIVE TO STOW | 2 IN/HR |
| f) | ICE & SNOW | " " | 1 cm, 4 LB/FT² |
| g) | ELEVATION | ALL | 0 – 10,000 FT |
| h) | SURVIVAL WIND VELOCITY | | 110 MPH, 1 cm ICE |

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# VLBA - DRIVE PARAMETERS

| | | Azimuth (A) | Azimuth (B) | Elevation |
|---|---|---|---|---|
| $K_{WT}$ | - Wind Torque Constant | | 172, EL = 0° | 172, EL = 120° |
| $V$ | - Wind Velocity, MPH | | | |
| $I_{AX}$ | - Inertia @ Axis, Slug-FT² | | $14.05 \times 10^6$ | $8.81 \times 10^6$ |
| $A$ | - Axis Level Acceler., RAD/SEC² | | | |
| $UB$ | - Unbalance, FT-LB | | | 5,000 NOM<br>399,000 (KE) |
| $F_{AX}$ | - Axis Level Running Friction, FT-LB | 20,000 | | 10,000 |
| $N_{AX}$ | - Overall Gear Ratio | 8,926 | 9,372 | 27,868 |
| $N_{MR}$ | - Main Reducer Gear Ratio | 100 | 450 | 564** |
| $F$ | - Motor Level Running Frict., FT-LB | | | |
| $I_S$ | - Motor Inertia, Slug-FT² | | | |
| $\eta$ | - Efficiency, Main Reducer | .80 | .80 | .80 |
| $f_n$ | - Locked Rotor Resonance, CPS | | 2.1 Minimum | 2.4 Minim |
| $K$ | - Axis Level Spring Constant, FT-LB/RAD | $1.68 \times 10^{10}$ | | $1.71 \times 10^{10}$ |
| $n$ | - Number of Drives | 4 | 2 | 2 |
| Type | - Type of Gear Boxes | Harmonic Drive, 30 M or Triple Reduction Helical | Quadruple Reduction Helical | Quadruple Reduction Helical |
| $N_M$ | - Max. Motor Speed, RPM | 2500 | | 2500 |
| $E_{SS}$ | - Servo System Error, Precision I, ARC-SEC | | 1.5* | |
| $D_R$ | - Data System Resolution, BIT | | 21 | |
| $E_{DS}$ | - Data System Error, ARC SEC | | 1.24* (20 BIT) | |
| $V_{MIN}$ | - Minimum Smooth Velocity, °/SEC | | < .002; < .0004*(1) | |
| $B$ | - Bias Torque Ratio @ Precision I, % | | 10 - 20 | |

Notes:

(1) Stick slip threshold velocity is est'd at .0004 °/sec @ Precision I.

* Target

** Same as used in VLA

SIZING OF DRIVE MOTORS: $T_M = \dfrac{K_{WT} V^2 + I_{AX} A + F_{AX} + U_B}{n N_{AX} \eta} + F + I_S N_{AX} A$

ASSUME: (AZIMUTH DRIVE & ELEV. DRIVE)

$F_{AZ}(A)$ ; $F_{EL}$ = .75 FT.LB ; $F_{AZ}(B)$ = 1.5 FT-LB

$I_{S\,AZ}(A)$ ; $I_{S\,EL}$ = .042 SLUG-FT$^2$ | $I_{AZ}(B)$ = .08 SLUG-FT$^2$

$$T_{M\,AZ}(A)_{IIA} = \frac{\overset{(WIND)}{172\,(45.85)^2} + \overset{(INERTIA)}{14.05\times10^6\times.0131} + \overset{(BG.\ FRICTION)}{20{,}000} + \overset{(UNBALANCE)}{0}}{4\,\underset{(i)}{(8926)}\,.80} + \overset{MOTOR\ FRICTION}{.75} + \overset{MOTOR\ INERTIA}{.042(8926).0131} = \underline{25.46\ FT\text{-}LB}$$

(PEAK VALUE) INCPL. GUST

$$T_{M\,AZ}(A)_{IIB} = \frac{172\,(57.96)^2 + 14.05\times10^6\times.0044 + 20{,}000 + 0}{4(8926)\,.80} + .75 + .042(8926).0044 = \underline{22.80\ FT\ LB}$$

VS. RATED OUTPUT TORQUE OF 22.8 FT-LB WITH SHORT TERM OVERLOAD CAPABILITY OF 34.2 FT-LB. FOR 5 HP @ 1150 RPM "VLA" MOTOR (OK)

$$PEAK\ HP = \frac{25.46 \times 2232}{5252} \cong \frac{11\ HP\ @\ 2232\ RPM}{4\ MOTORS}$$ (.25 RPM AXIS SPEED) (OK)

$$T_{M\,AZ}(B)_{IIA} = \frac{172\,(45.85)^2 + 14.05\times10^6\times.0131 + 20{,}000 + 0}{2(9372)\,.80} + 1.5 + .08(9372).0131 = \underline{49.04}\ FT\text{-}LB$$

$$T_{M\,AZ}(B)_{IIB} = \frac{172\,(57.96)^2 + 14.05\times10^6\times.0044 + 20{,}000 + 0}{2(9372)\,.80} + 1.5 + .08(9372).0044 = \underline{43.65}\ FT\text{-}LB$$

VS. RATED OUTPUT TORQUE OF 52 FT LB WITH SHORT TERM OVERLOAD CAPACITY OF 66 FT LB FOR 15 HP @ 1750 RPM INTELSAT STD. A DRIVE MOTOR (OK)

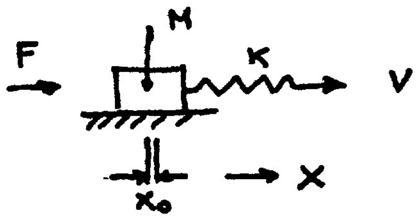
$$PEAK\ HP = \frac{49.04 \times 2343}{5252} \cong \frac{22\ HP\ @\ 2343\ RPM}{2\ MOTORS}$$ (.25 RPM AXIS SPEED) (OK)

$$T_{M\,ELEV.}(A)_{II.B} = \frac{172\,(45.85)^2 + 8.61\times10^6\times.0044 + 10{,}000 + 5{,}000}{2(27{,}868)\,.80} + .75 + .042(27{,}868).0044 = \underline{18.34\ FT\text{-}LB}$$

(OK)

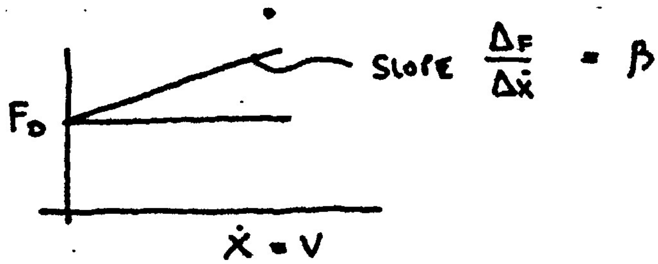
# STICK | SLIP THRESHOLD

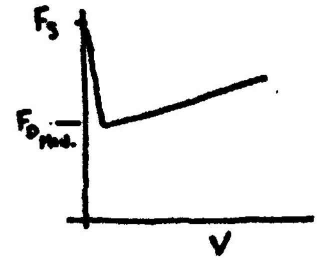
SIMPLE SPRING MASS SYSTEM




DYNAMIC FRICTION FORCE — DRIVING FORCE

$$M\ddot{X} + \beta\dot{X} + KX + F_D = F$$

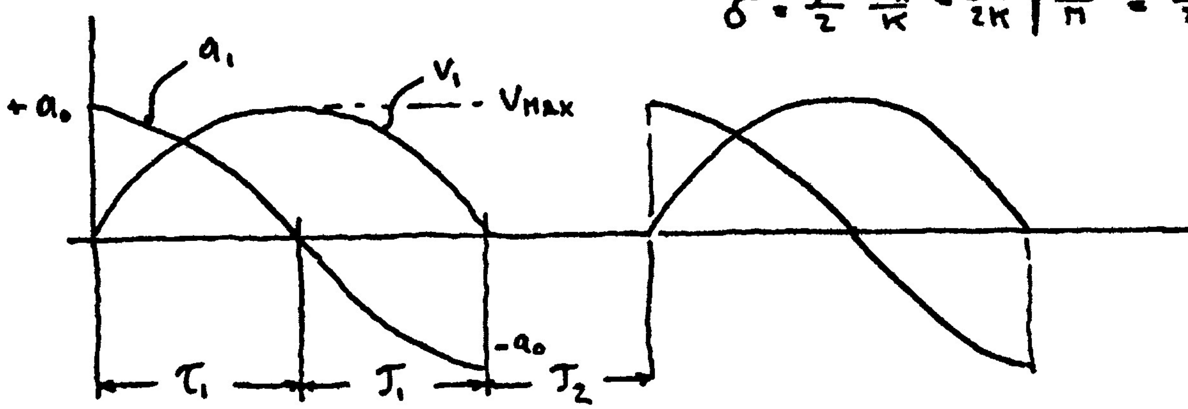



$$M\ddot{X} + \beta\dot{X} + KX = F - F_D \quad ; \quad X = \frac{F - F_D}{MS^2 + \beta S + K}$$

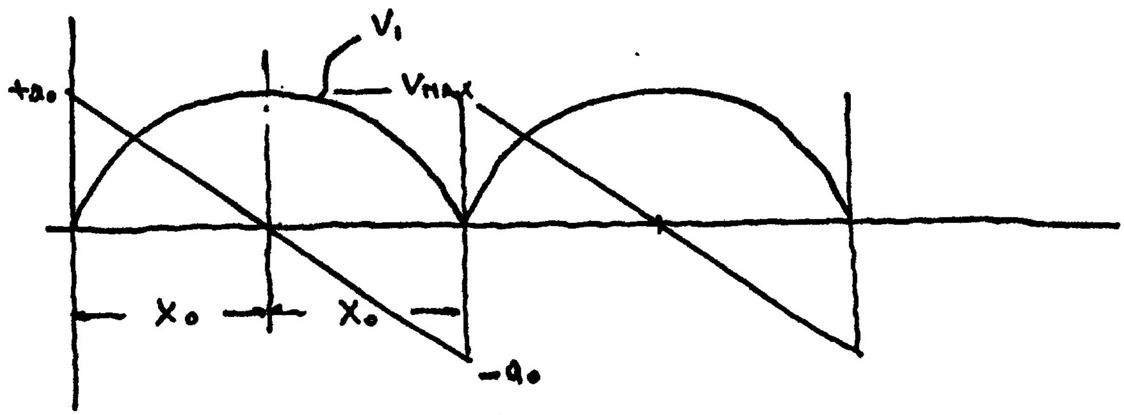
$$X = \frac{F - F_D}{K}\,\frac{1}{\frac{M}{K}S^2 + \frac{\beta}{K}S + 1} \quad ; \quad \omega_n = \sqrt{\frac{K}{M}} \quad ; \quad \frac{2\delta}{\omega_n} = \frac{\beta}{K}$$

$$\delta = \frac{\beta}{2}\,\frac{\omega_n}{K} = \frac{\beta}{2K}\sqrt{\frac{K}{M}} = \frac{\beta}{2}\sqrt{\frac{1}{KM}}$$




STICK | SLIP CONDITION

$\bar{V}_{MIN} > V$




STICK | SLIP THRESHOLD
(BOUNDARY CONDITION)

$$V_1 = \sqrt{a_o x_o} \sin\left(\sqrt{\frac{a_o}{x_o}}\,\tau\right) \; ; \; \frac{dV_1}{dt} = a_1$$

$$a_1 = a_o \cos\left(\sqrt{\frac{a_o}{x_o}}\,\tau\right) = a_o\sqrt{1-\frac{V^2}{a_o x_o}} \; ; \; \frac{da_1}{dt} = -a_o\sqrt{\frac{a_o}{x_o}} \sin\sqrt{\frac{a_o}{x_o}}\,\tau$$

$$\boxed{\tau_1 = \frac{\pi}{2}\sqrt{\frac{x_o}{a_o}}} \qquad V_1 = \sqrt{a_o x\left(2-\frac{x}{x_o}\right)} \; ; \; a_1 = a_o - \frac{a_o}{x_o}x$$

$$\frac{dv_1}{dx} = \sqrt{\frac{a_o}{x}}\,\frac{\left(1-\frac{x}{x_o}\right)}{\sqrt{2-\frac{x}{x_o}}} \; ; \; \frac{da_1}{dx} = -\frac{a_o}{x_o}$$

$$V_{MAX} = \sqrt{a_o x_o} \qquad \boxed{\tau_2 = \frac{2x_o}{\bar{V}} - 2\tau_1}$$

WHEN $\tau_2 = 0$ $\quad \bar{V} = \bar{V}_{MIN}$ (STICK SLIP THRESHOLD)

$V > \bar{V}_{MIN}$ DEFINES SMOOTH VELOCITY!

$$\bar{V}_{MIN} = \frac{2}{\pi}\sqrt{a_o x_o} \; ; \; X = x_o\left(1-\frac{a}{a_o}\right) = x_o\left(1-\sqrt{1-\frac{V^2}{a_o x_o}}\right)$$

$$X_o = \frac{F - F_D}{K} \; ;$$ LET $F = F_S$ = STATIC FRICTION FORCE

$$\boxed{X_o = \frac{F_S - F_D}{K}} \qquad \text{LET } k = \frac{F_S}{F_D} \qquad \boxed{X_o = \frac{k-1}{K}F_D}$$

$$a_o = \frac{1}{M}(F_S - F_D) \; ; \; \boxed{a_o = \frac{F_D}{M}(k-1)}$$

SINCE $F_D = \mu W = \mu M g$ AND $\frac{1}{\omega} = \sqrt{\frac{M}{K}}$

$$X_o = \frac{(k-1)\mu W}{K} = \frac{(k-1)\mu M g}{K} = \frac{\mu g(k-1)}{\omega_n^2}$$

$$\bar{V}_{MIN} = \frac{2\mu W}{\pi\sqrt{MK}}(k-1) = \frac{2\mu g}{\pi}\sqrt{\frac{M}{K}}(k-1) = \frac{2}{\pi}\frac{\mu g}{\omega_n}(k-1)$$

$$\bar{V}_{MIN.} = \frac{2}{\pi}\sqrt{\frac{F_D}{M}(k-1)\frac{(k-1)}{K}F_D} = \boxed{\frac{2}{\pi}\frac{F_D}{\sqrt{KM}}(k-1)}$$

PHYSICAL EXPLANATION :

DURING STICK/SLIP THRESHOLD CONDITIONS THE FORWARD MOVING MASS "M" OSCILLATES IN DISCRETE STEPS OF $\frac{1}{2}$ CYCLES OF THE RESONANT FREQUENCY OF THE SPRING-MASS SYSTEM, REPEATING EACH $\frac{1}{2}$ CYCLE IN SUCCESSION, SINCE $T_2 = 0$. ANY VELOCITY ABOVE THIS THRESHOLD VELOCITY HAS TO BE CONSIDERED AS SMOOTH SINCE FORWARD MOTION IS CONTINUOUS.

AS THE RESONANT FREQUENCY OF THE SIMPLE SPRING-MASS SYSTEM IS :

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

AND THE FORCE AVAILABLE DUE TO THE WINDUP OF THE SYSTEM WHEN THE APPLIED FORCE EQUALS THE STATIC FRICTION FORCE "$F_S$" AND THE STORED ENERGY IS RELEASED AS F IS REDUCED TO THE DYNAMIC FRICTION FORCE "$F_D$", IS :

$$X_0 = \frac{F_S - F_D}{K}$$

THE STICK|SLIP VELOCITY THRESHOLD BECOMES :

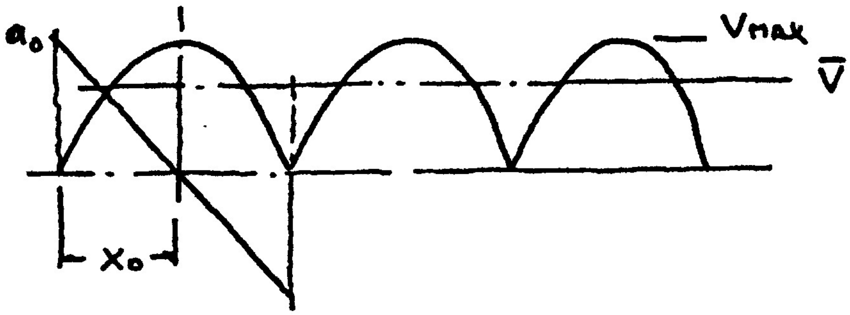
$$\bar{V}_{MIN} = 4 x_0 f_n$$

$$\bar{V}_{MIN} = \frac{4(F_S - F_D)}{K}\frac{1}{2\pi}\sqrt{\frac{K}{M}} \quad ; \text{ LET } k = \frac{F_S}{F_D}$$

$$\boxed{\bar{V}_{MIN} = \frac{2}{\pi}\frac{F_D}{\sqrt{KM}}(k-1)} \qquad V_{SMOOTH} > \bar{V}_{MIN}$$

# MINIMUM-SMOOTH VELOCITY AND POSITION ERROR:

CONSIDER STICK | SLIP MOTION THRESHOLD:




$$k = F_S/F_D = M_{TS}/M_{TD}$$

$$X_0 = \frac{k-1}{K} M_{TD}$$

$$a_0 = \frac{M_{TD}}{I}(k-1)$$

$$\bar{V}_{MIN} = \frac{2\, M_{TDYN}}{\pi\sqrt{K I}}(k-1)$$

$M_{TDYN\ AZ}$ = $\underline{20,000}$ FT-LB, MIN. ; $15.56^2(172) + 20{,}000$ = $\underline{61,644}$ FTLB, MAX @ CONDITION I

$M_{TSTAT.\ AZ}$ = $1.5 \times 20{,}000$ = $\underline{30,000}$ FT-LB, MIN ; $\underline{71,644}$ FT-LB MAX

$k_{MAX} = \dfrac{30{,}000}{20{,}000} = \underline{1.5}$ ; $k_{MIN} = \dfrac{71{,}644}{61{,}644} = \underline{1.16}$ MIN

$I_{AZ} = 43.8 \times 10^6$ SLUG-FT$^2$ (INCLUDE. MOTOR)

$K_{AZ} = 1.68 \times 10^{10}$ FTLB | RAD

$$\bar{V}_{MIN\ AZ(1)} = \frac{2\,(20{,}000)}{\pi\sqrt{1.68\times10^{10} \times 43.8\times10^6}}(1.5-1) = 7.42\times10^{-6}\ \text{RAD/SEC} = \underline{.000425}\ °/\text{SEC}$$

@ 0 WIND VELOCITY

$$\bar{V}_{MIN\ AZ(2)} = \frac{2\,(61{,}644)}{\pi\sqrt{1.68\times10^{10} \times 43.8\times10^6}}(1.16-1) = 7.32\times10^{-6}\ \text{RAD/SEC} = \underline{.000419}\ °/\text{SEC}$$

@ 15.56 MPH

$$X_{0\,(1)} = \frac{1.5-1.0}{1.68\times10^{10}}\,20{,}000 = 5.95\times10^{-7}\ \text{RAD} = \underline{.125\ \text{ARC SEC}}$$

$2X_{0(1)}$ = .25 ARC SEC (DISCRETE STEPS) = UNCONTROLLABLE DYNAMIC POINTING ERROR @ AZ

$$X_{o(2)} = \frac{1.16-1}{1.60 \times 10^{10}} 61,644 = 5.87 \times 10^{-7} \text{ RAD} = \underline{.12 \text{ ARC SEC}}$$

$$2X_{o(2)} = \underline{.24 \text{ ARC SEC}} \text{ @ 15.56 MPH , UNCONTROLLABLE DYNAMIC POINTING ERROR @ AZ}$$

$$I_{ELEV} = 41.18 \times 10^{6} \text{ SLUG-FT}^2 \text{ (INCLUD. MOTOR)}$$
$$K_{ELEV} = .942 \times 10^{10} \text{ FT-LB/RAD}$$

LET : $M_{T\,DYN\,EL} = \underline{10,000} \text{ FT-LB, MIN.} \; ; \; \underline{51,644} \text{ FT-LB, MAX}$

$M_{T\,STAT.\,EL} = \underline{15,000} \text{ FT-LB, MIN.} \; ; \; \underline{56,644} \text{ FT-LB, MAX}$

$$R_{MAX} = \frac{15,000}{10,000} = \underline{1.5} \quad ; \quad R_{MIN} = \frac{56,644}{51,644} = \underline{1.10}$$

$$\bar{V}_{MIN\,EL(1)} = \frac{2(10,000)}{\pi\sqrt{.942 \times 10^{10} \times 41.18 \times 10^{6}}}(1.5-1) = 5.11 \times 10^{-6} \text{ RAD/SEC} = \underline{.000293\ °/\text{SEC}} \text{ @ 0 MPH}$$

$$\bar{V}_{MIN\,EL(2)} = \frac{2(51,644)}{\pi\sqrt{.942 \times 10^{10} \times 41.18 \times 10^{6}}}(1.10-1) = 5.24 \times 10^{-6} \text{ RAD/SEC} = \underline{.0003\ °/\text{SEC}} \text{ @ 15.56 MPH}$$

$$2X_{o(1)} = 2\,\frac{(1.5-1.0)}{.942 \times 10^{10}} \times 10,000 = 1.06 \times 10^{-6} \text{ RAD} = \underline{.22 \text{ ARC SEC}} \text{ @ 0 MPH}$$

$$2X_{o(2)} = 2\,\frac{(1.10-1)}{.942 \times 10^{10}} \times 51,644 = 1.10 \times 10^{-6} \text{ RAD} = \underline{.23 \text{ ARC SEC}} \text{ @ 15.56 MPH}$$

THESE ESTIMATED MINIMUM SMOOTH VELOCITIES AND DISCRETE STEP MOTIONS ARE BASED ON THE ASSUMPTION THAT THE STATIC (BREAKOUT) FRICTION LOSSES ARE 50% HIGHER THAN THE DYNAMIC LOSSES RELATIVE TO THE INHERENT AXIS FRICTION LOSSES ONLY. THIS IS CONSIDERED TO BE REALISTIC SINCE THE AVERAGE DIFFERENCE BETWEEN STATIC AND DYNAMIC EFFICIENCY OF A WELL DESIGNED GEAR BOX IS AROUND 4%.

THE STICK|SLIP MOTION IS CHARACTERIZED AS $\frac{1}{2}$ CYCLE STEP FORWARD MOTION WHERE THE 2 STEP FREQUENCY SHOULD EQUAL THE LOCKED ROTOR RESONANT FREQUENCY OF THE SYSTEM :

$$f_{STICK|SLIP\,(AZIM)} = \left(\frac{1}{2}\right)\frac{7.42\times10^{-6}}{2\times5.95\times10^{-7}} = \underline{3.12\ CPS}$$ (VS. 3.11 LOCKED ROTOR FREQUENCY CALCULATED)

$$f_{STICK|SLIP\,(ELEV.)} = \frac{1}{2}\ \frac{5.11\times10^{-6}}{1.06\times10^{-6}} = \underline{2.41\ CPS}$$ (VS. 2.41 LOCKED ROTOR FREQUENCY CALCULATED)

<u>STEPPING ACCELERATIONS</u> :

@ AZ : $a_{STEP} = \frac{20{,}000}{43.8\times10^{6}}\cdot .5 = 2.28\times10^{-4}\ \frac{RAD}{SEC^2} = \underline{.013\ ^\circ/SEC^2}$

@ ELEV : $a_{STEP} = \frac{10{,}000}{41.18\times10^{6}}\cdot .5 = 1.214\times10^{-4}\ \frac{RAD}{SEC^2} = \underline{.007\ ^\circ/SEC^2}$

NORTHERN PRECISION LABORATORIES, INC.
P.O. Box 1016 • 11 Madison Rd., Fairfield, NJ 07006 • Phone: 201-227-4800 • TWX 710-734-4301

October 21, 1983

Systems Development Laboratories
13372 Talle-Colina
Poway, California 92064

Attention: Mr. Otto R. Heine

Reference: Telecon Heine/Nigro

Dear Mr. Heine:

Thank you for your interest in Northern Precision Laboratories' product line. Per our recent telephone conversation, enclosed please find copies of Northern Precision Laboratories' most recent Encoder and Drive literature. I have also enclosed a copy of a letter I previously sent to Mr. William Horn at National Radio Astronomy Observatory and some of our drawings which I feel may be of some assistance.

Should you have any questions or require additional information, please do not hesitate to contact me at your convenience.

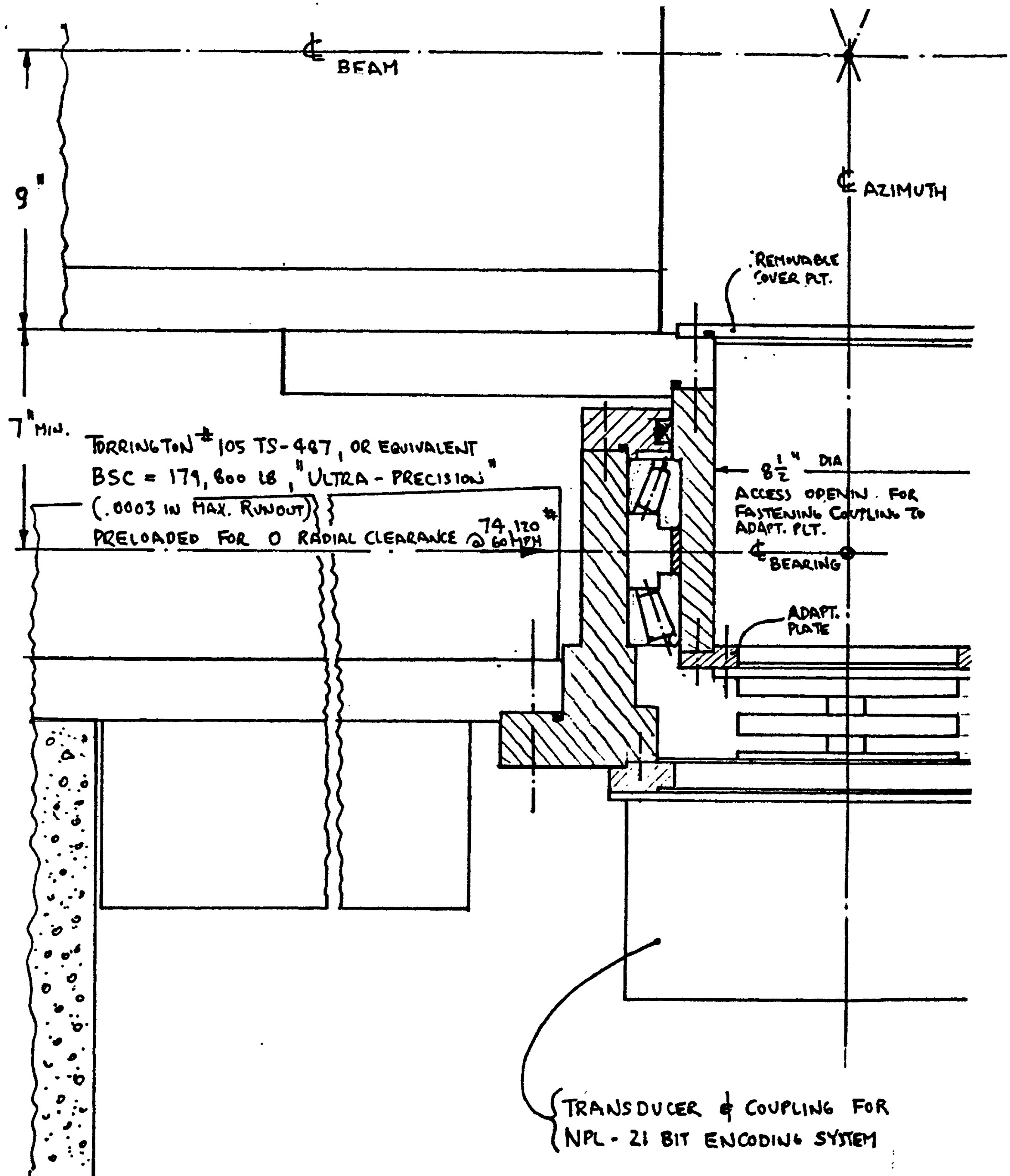
Very truly yours,

NORTHERN PRECISION LABORATORIES, INC.

Michael P. Nigro
Product Development

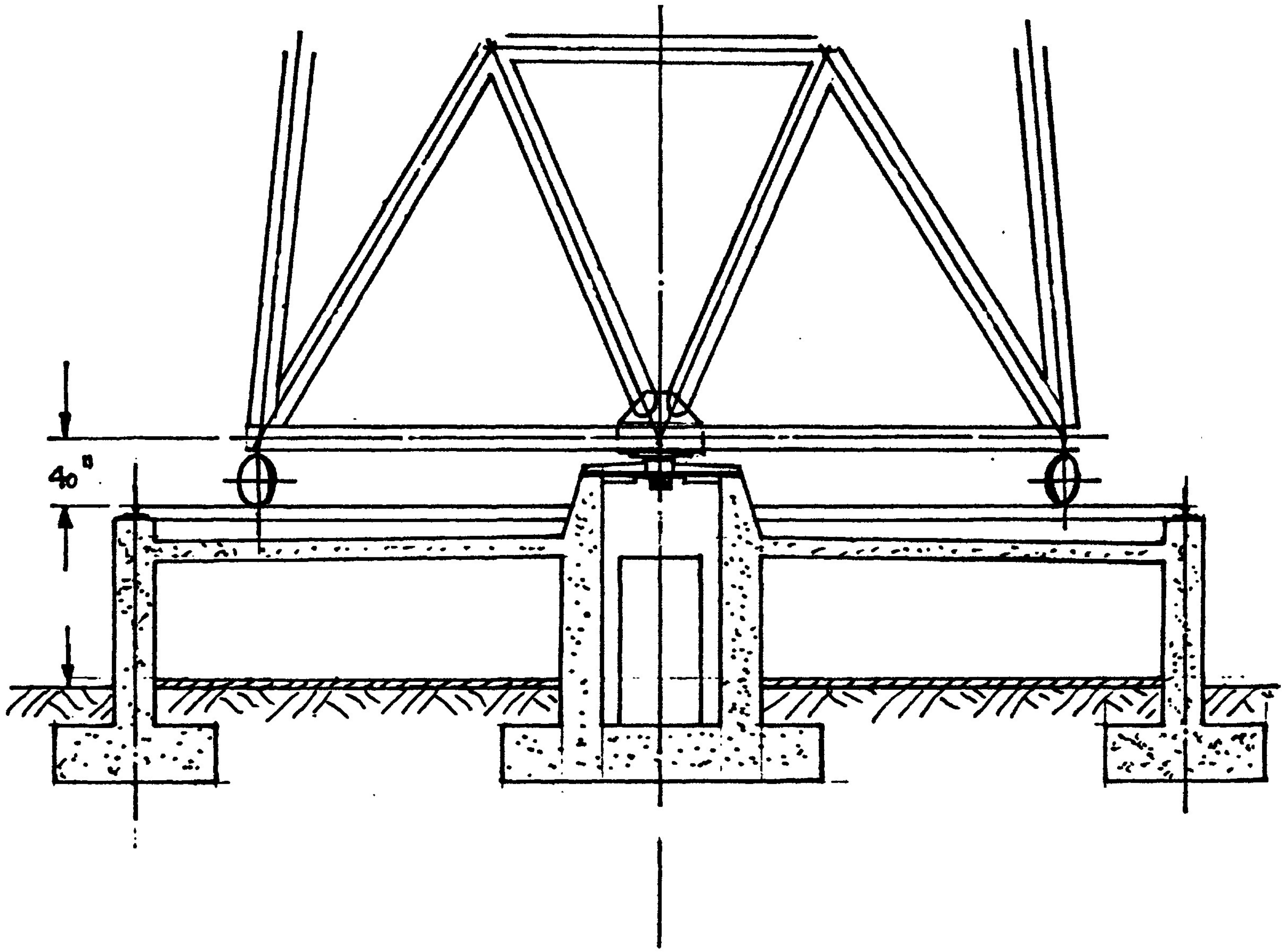
MPN:nls

Encls. Catalogs 82900, 78900,
Model 25 10HP SCR Servo Amplifier
Drawings 801404-1, 801342-1,
801342-2, 801908
Copy of NRAO Letter 8/31/83

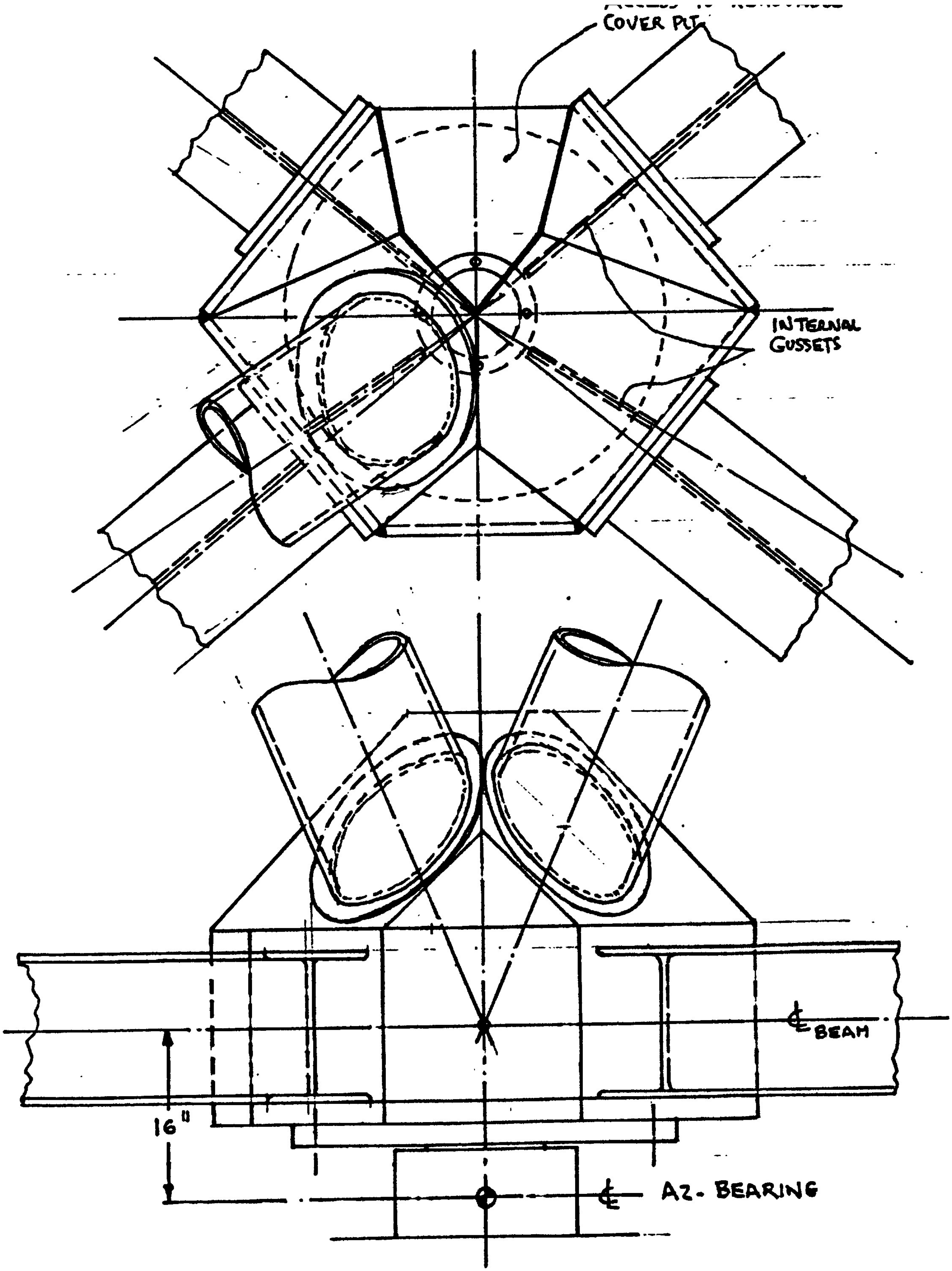



SCALE : 1/4

VLBA - AZIMUTH PINTLE BEARING




VLBA - AZIMUTH BEARING & TRACK



COVER PLT
INTERNAL GUSSETS
℄ BEAM
16"
℄ AZ. BEARING

Systems Development Laboratory
ENGINEERING CONSULTANTS
13372 CALLE COLINA
POWAY, CALIFORNIA 92064
(714) 485-5657

Mr. Anthony Tartaglio
Sales Manager, Enclosed Drives
PHILADELPHIA GEAR CORPORATION
181 South Gulph Road
King of Prussia, PA 19406

November 2, 1983

Dear Mr. Tartaglio:

Enclosed is a 1/4 scale sketch showing the prefered Azimuth Drive Arrangement for the VLBA Antenna, option # 4 as indicated in my letter of October 12. Please review this concept and let me know at your earliest convenience whether the VLA Elevation Drive Gear Box can be modified and mounted as shown on the sketch or whether the 30 inch diameter drive wheel would have be provided with its own bearings?

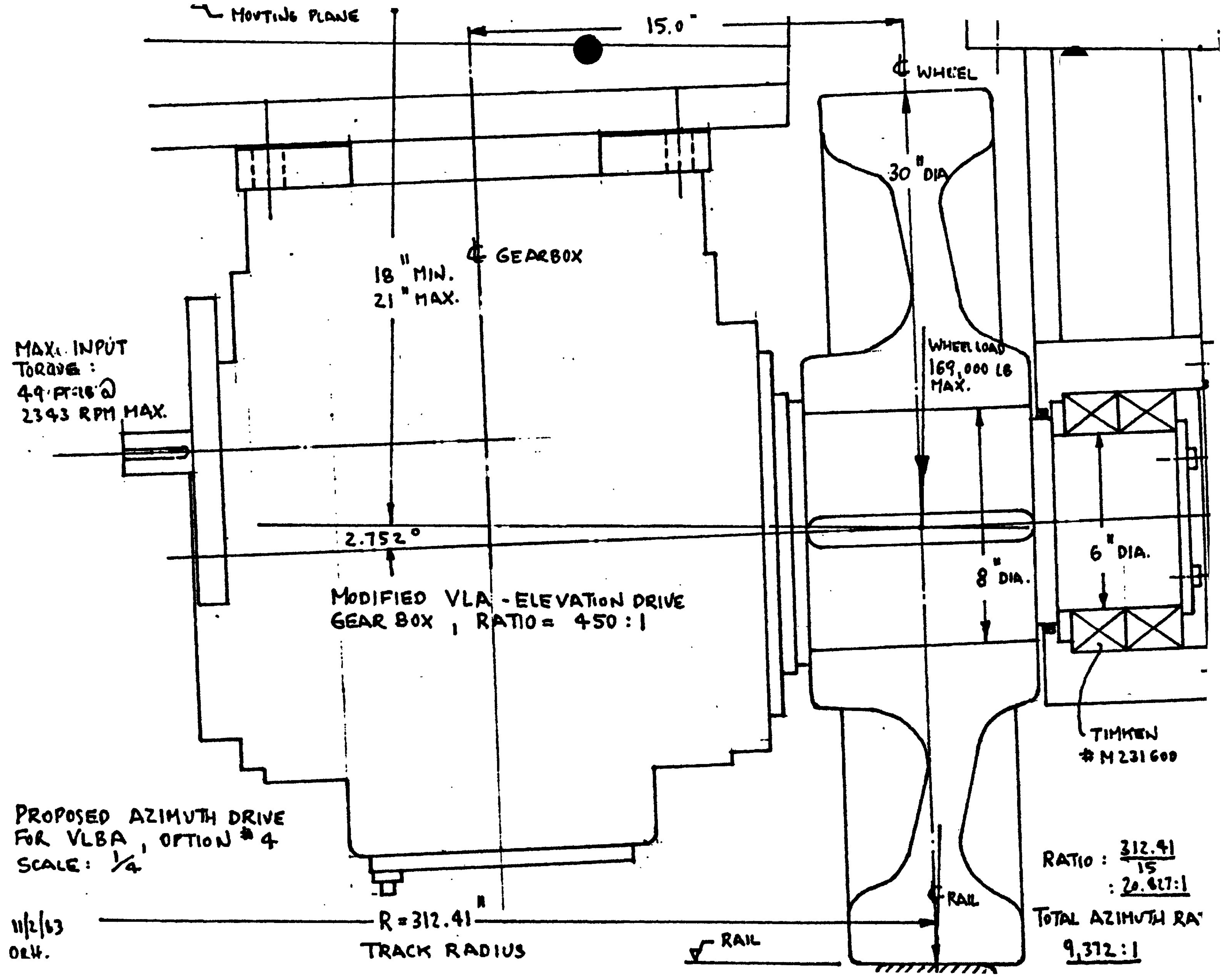
I am assuming that the VLA Elevation Drive Gear Box Ratio could easely be reduced from the present 541:1 to 450:1 by changing the input stage ratio and that mounting the reducer overhead and at a slight incline would not be problematic.
At the present time I feel that utelizing the VLA Elevation Drive Gear Box for both the Elevation- and Azimuth Drives of the VLBA Antenna would be the most cost effective and practical solution in this case. Please concentrate your efforts on this concept and provide the information on options 1,2 and 3 at a later date. As mentioned before, the VLBA Elevation Drive Gear Boxes would be identical to the VLA Units with exception that the input speed would be raised to 2322 RPM and the maximum drive torque required would be 170,300 in-lb at 774 RPM.

Very truly yours,

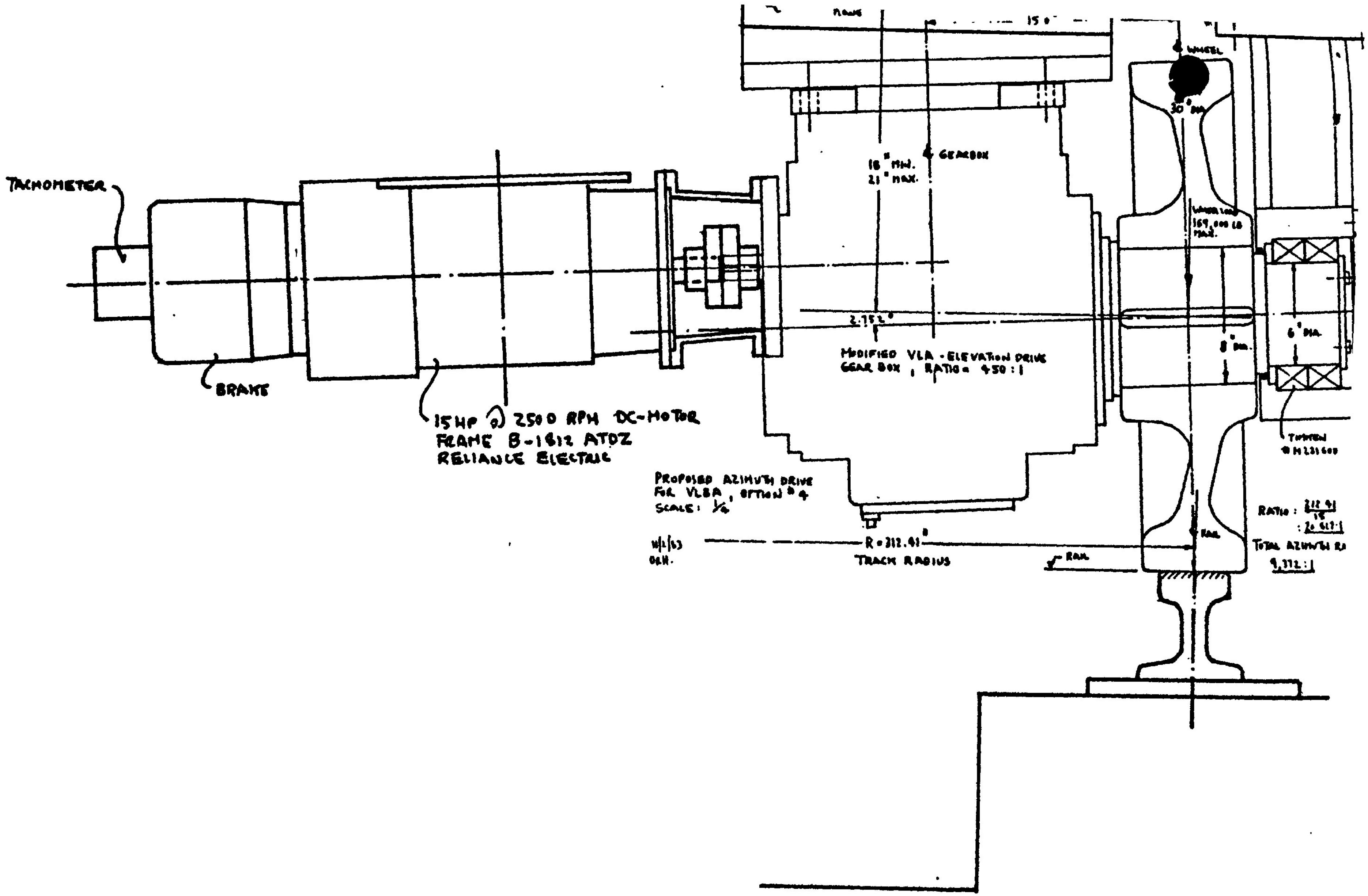
SYSTEMS DEVELOPMENT LABORATORY

Otto R. Heine

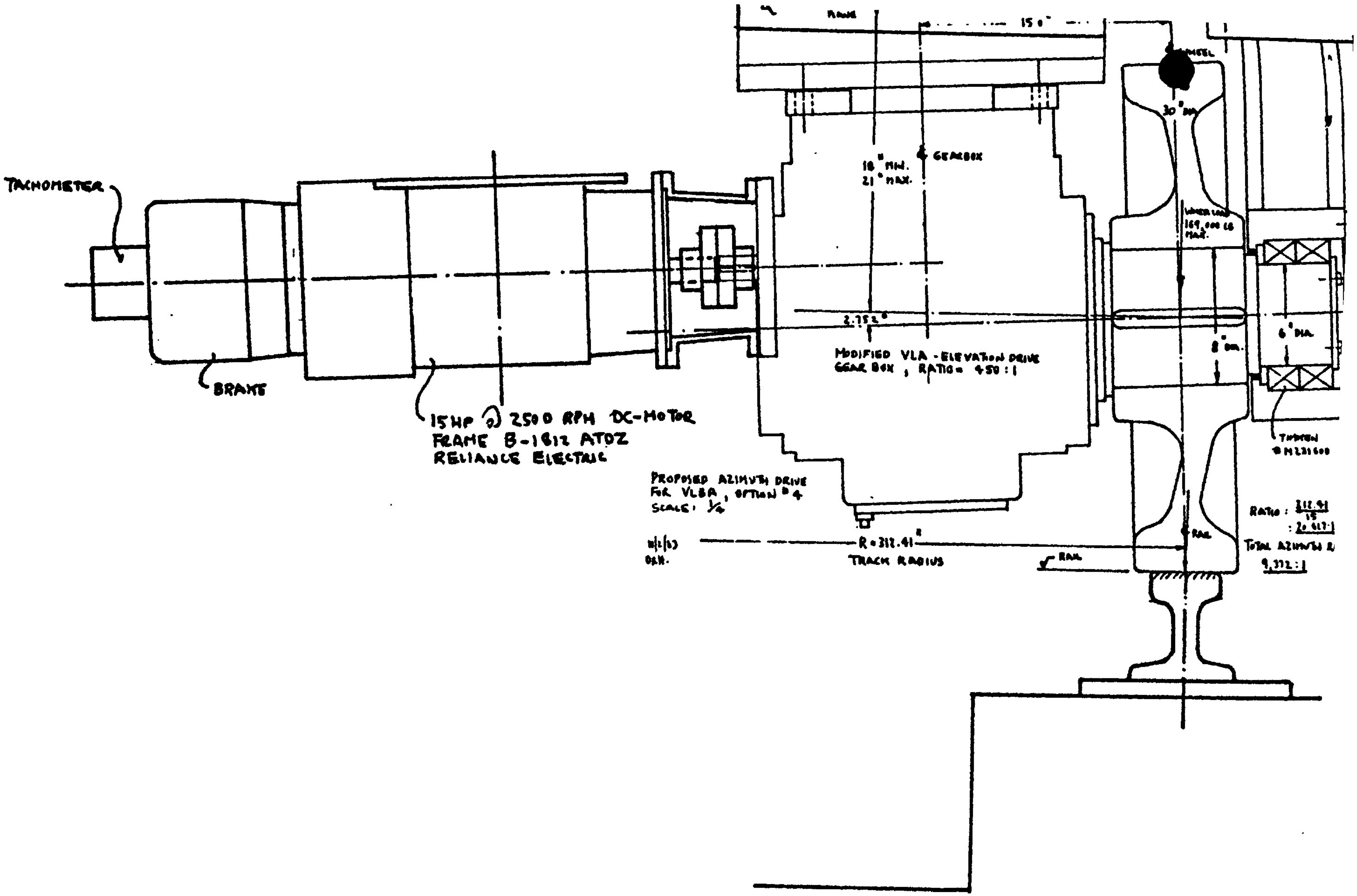
cc.: William G. Horne, NRAO



MOUTING PLANE
15.0"
℄ WHEEL
30" DIA
℄ GEARBOX
18" MIN.
21" MAX.
MAX. INPUT
TORQUE :
49 FT-LB @
2343 RPM MAX.
WHEEL LOAD
169,000 LB
MAX.
2.752°
6" DIA.
8" DIA.
MODIFIED VLA - ELEVATION DRIVE
GEAR BOX , RATIO = 450:1
TIMKEN
# M 231 600
PROPOSED AZIMUTH DRIVE
FOR VLBA , OPTION # 4
SCALE: 1/4
RATIO : 312.41 / 15
: 20.827:1
℄ RAIL
11/2/83
ORH.
R = 312.41"
TRACK RADIUS
RAIL
TOTAL AZIMUTH RA
9,372:1



TACHOMETER
BRAKE
15HP @ 2500 RPM DC-MOTOR
FRAME B-1812 ATDZ
RELIANCE ELECTRIC
PROPOSED AZIMUTH DRIVE
FOR VLBA, OPTION # 4
SCALE: 1/4
MODIFIED VLA - ELEVATION DRIVE
GEAR BOX, RATIO = 450:1
16" MIN.
21" MAX.
℄ GEARBOX
2.752"
R = 312.41"
TRACK RADIUS
℄ RAIL
℄ WHEEL
30" DIA.
8" DIA.
6" DIA.
TIMKEN
15 0"
RATIO: 312.41/15 : 20.827:1
TOTAL AZIMUTH RATIO
9,372:1



TACHOMETER
BRAKE
15 HP @ 2500 RPM DC-MOTOR
FRAME B-1812 ATDZ
RELIANCE ELECTRIC
15.0"
GEARBOX
18" MIN.
21" MAX.
2.752"
MODIFIED VLA - ELEVATION DRIVE
GEAR BOX, RATIO = 450:1
30" DIA.
8" DIA.
6" DIA.
TIMKEN
PROPOSED AZIMUTH DRIVE
FOR VLBA, OPTION #4
SCALE: 1/4
R = 312.41"
TRACK RADIUS
RAIL
TOTAL AZIMUTH R
9,372:1