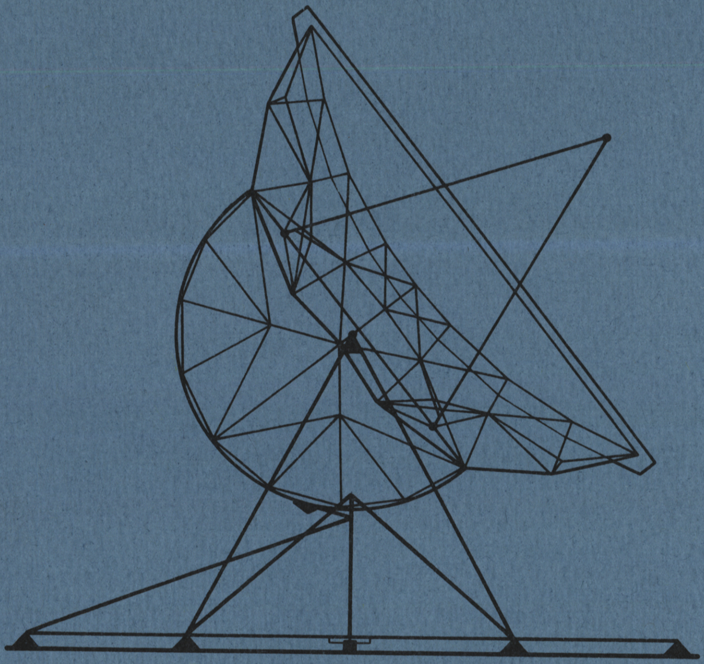
A 300 FOOT HIGH-PRECISION RADIO TELESCOPE

Appendix 2a



May, 1969

National Radio Astronomy Observatory *
Green Bank, West Virginia

* OPERATED BY ASSOCIATED UNIVERSITIES, INC., UNDER CONTRACT WITH THE NATIONAL SCIENCE FOUNDATION.

DESIGN CONCEPT STUDY

OF

300 FT. DIAMETER HOMOLOGY RADIO TELESCOPE

FOR

NATIONAL RADIO ASTRONOMY OBSERVATORY *

GREEN BANK, WEST VIRGINIA

BY

SYSTEMS DEVELOPMENT LABORATORY

SUBCONTRACT : RAP-79

DATE : JUNE 1969

S.D.L. REPORT : H79-8

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PROJECT: 300 FT. DIA. HOMOLOGY RADIO TELESCOPE

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ABSTRACT

This report represents the initial efforts by S.D.L. on the conceptual design of the 300 ft. diameter homology radio telescope which have been performed in conjunction with Dr. V. Hoerner's work on the overall structural configuration of the instrument, as a part of a general feasibility/development study conducted by N.R.A.O.

The study contained herein covers those major subsystems of the telescope for which specific design solutions had to be found in order to define a feasible and practical system which could achieve the desired performance characteristics.

S.D.L. performed this work in direct consultation with the originator of the homology concept, Dr. v. Hoerner, and with Dr. H. Hvatum and Mr. W. Horne of N.R.A.O.

Applicable weight and cost figures along with outline drawings for all major subsystems were prepared as required and are part of this report.

The dynamic behavior of the overall system including the effects of soil, foundation and other contributing areas, was determined in detail through performance of a computer analysis by utilizing a program developed by the space and re-entry systems division of Philco-Ford Corp.

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SUMMARY AND CONCLUSION

Based on this study effort, which includes a rigorous dynamic analysis of the system and a preliminary servo analysis, it is concluded that the design objectives set forth by Dr. S. v. Hoerner are within reach and that a desired pointing accuracy of ±5 arc. sec. is within the state of art providing the reference system can be applied in practice.

It is estimated that the moving weight of the homology telescope will be around 2300 tons and that the total cost (including engineering, control buildings, etc.) of the complete 300 ft. dia. telescope will be below $10,000,000.00.

No major design, fabrication or erection problems are anticipated as it was the purpose of this effort to develop a practical, feasible, low cost concept. Emphasis has been put in particular on design simplicity in the adoption of articulating - anti-backlash drives and proposed use of "zero play" - self-aligning bearing assemblies which would minimize structural distortions.

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DISCUSSION OF SUBJECTS STUDIED

A. STRUCTURAL ITEMS

1. Built-Up Members

A study of the sizing of members for the built-up reflector beams per original concept by Dr. v. Hoerner revealed that a majority are prone to fail due to wind induced, self-excited vibrations through "V. Karman" vortex shedding. Most members are too long in respect to their diameter and exceed the maximum permissible length, which was calculated to be $l_{MAX} = 13.92\,D^{1.5}$

Another criteria was found to be the minimum wall thickness, which for practical purpose in order to assure economy of fabrication and welding, must be limited to 3/32 in.

Finally, it was indicated by manufacturers that for reason of economy a straight member with tapered ends with parallel lacing arrangement would be preferable over the original curved member concept having an irregular lacing arrangement.

Consideration of these three factors will undoubtedly result in a weight increase for many reflector members over those weights calculated by Dr. v. Hoerner and used in his homology analysis (present estimate is 20%); however, it is believed that a final - optimized - reflector design will result in a weight figure of less than the 1150 tons presently used.

The proposed built-up member concept is shown on Drawing 109-D-003.

The built-up members for the tower structure as shown on Drawing 109-D-004 were designed to be safe against wind induced vibrations and practical from the viewpoint of fabrication.

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2. Member Joints

Drawing 109-D-005 shows the proposed concept of a typical member joint and detail of the built-up member end connection. As can be seen, it is proposed that the joints be prefabricated and that the members be fitted and welded during erection at the site.

The design of the joint is such that maximum joint flexibility is assured while sufficiently low stress levels in the member ends can be maintained. Each joint must be individually analyzed and designed to deal with the imposed moments and loads and the joint stiffness must be determined for inclusion in the final homology analysis.

At the present time, the effect of joint stiffnesses were not included in the homology analysis and dynamic analysis of the telescope. Several joints such as for instance reflector joint No. 45 or those used as bearing, truck or gear train mounts, will have to be of a special design to facilitate the desired function or to avoid structural interference.

3. Foundation and Track

Dr. v. Hoerner initially proposed a double, 4-rail standard railroad track bed with gravel ballast supporting four individually driven truck assemblies utilizing standard heavy duty gondolas, for the azimuth turntable. This concept was found to be impractical, primarily due to the length of the gondolas which required unusually long and heavy walking beams for the formation of a suitable truck assembly.

Instead, it was proposed to only use standard railroad truck assemblies with springs removed and to place these units as closely as possible together. This resulted in a more manageable overall truck assembly size with relative short walking beams, which could be made sufficiently stiff. This concept is shown on Drawing No. 109-D-010.

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An analysis of the obtainable foundation and truck stiffness indicated suitability for the intended purpose and a dynamic analysis of the complete system showed that the flexibility of this turntable arrangement would affect the lowest resonance of the system only to a reasonable degree.

At the same time, a single (two rails) 175 lbs/yd. crane rail track running on a 9 ft. wide by 2 ft. thick concrete slab was studied. This configuration permits the application of fewer wheels per truck assembly (only 8 - 36 in. dia. crane wheels are required vs. 32 - 38 inch dia. wheels for the double railroad track), resulting in lower cost truck assemblies due to simplification of the truck design.

It was found that in comparison to the railroad track turntable, the concrete slab - crane rail turntable would be less costly while at the same time having improved stiffness characteristics which would increase the lowest resonant frequency of the system by about 10% over the resonance obtainable with the railroad track.

Furthermore, there are a number of additional reasons why the concrete slab - crane rail concept would be preferable over the railroad track concept. They are as follows:

a. Reduced truck maintenance.
b. More accurate rail surface contour.
c. Fewer mechanical components.
d. Shorter truck dimensions.
e. Higher wheel loads improve friction drive traction and eliminate need for counterweight.
f. Lower friction and reduced surface wear.
g. Stow anchors can easily be incorporated.

Because of the improved performance characteristics and reduced cost the concrete slab - crane rail concept was finally selected as the turntable configuration for the homology telescope. This concept is shown on Drawing 109-D-009.

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B. MECHANICAL ITEMS

1. Elevation Bearing Arrangement

The basic requirements for the two elevation bearing arrangements of the homology telescope are that they be able to nearly equally share any loads imposed on the reflector while producing a minimum of strain which could transfer into the reflector structure and thereby affect the homogeneous deformation characteristic of the reflector surface.

These requirements are met by the features of the concept shown on Drawing 109-D-007, which employs a preloaded, self-aligning, dual bearing arrangement.

The proposed scheme is somewhat unusual because of the application of spherical roller thrust bearings as primary radial bearings, but since this arrangement requires axial preloading, equal load sharing as well as elimination of play is essentially assured, while a self-aligning capability is achieved.

2. Pintle Bearing Arrangement

The structural concept of the telescope tower requires that the pintle bearing must be able to take loads simultaneously in both vertical and radial directions. This bearing should also be a self-aligning type, as the turntable may inherently have a significant wobble (which will, however, not affect the system's pointing accuracy since the reference system will eliminate this error source) and since again it is desired to minimize the strain in the tower structure due to misalignments.

In addition, the pintle bearing arrangement must be designed in such way that a predetermined amount of deadload can be imposed on it.

The arrangement shown on Drawing 109-D-006 fulfills these above requirements as it consists of a combination of a spherical roller thrust bearing, which can simultaneously take radial loads, with a self-aligning cylindrical roller thrust bearing, which is used for preloading purpose only.

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The preloading of the spherical roller thrust bearing eliminates any radial play and assures a minimum radial load capacity regardless of the magnitude of applied thrust loads. The bearings must be selected so that their origin of alignment curvature coincide. The bearings are housed around a king pin located in a suitable flange mounted housing, which is located on top of a circular pintle foundation. The flange mounting of the housing makes it feasible to jack the bearing assembly against the tower structure until (by strain measurement) the desired dead load has been applied and also makes it possible to replace the bearing in case of damage.

3. Truck Assemblies and Azimuth Drive

The turntable as described before consists of 175 lbs/yd. heat treated crane rails of 5 ft. spur width mounted on a reinforced concrete slab.

The proposed typical truck assembly shown on Drawing 109-D-009 consists of eight double flanged 36 in. dia. "rim toughened" crane wheels arranged in pairs in pivot mounted bogies. The bogies are mounted in suitable side frames which are connected by a transom structure containing a torque tube, thereby providing for effective load equalization. The load is transferred into the transom structure by means of a spherically shaped saddle.

The load is nearly evenly divided among all eight wheels, as a result of this structural arrangement, even if there should be an opposing waviness pattern in the parallel rails (the application of the torque tube permits controlled flexibility - load pattern). Four wheels of the truck assembly are independently driven by means of spur gear trains connected to the axles and powered by DC motors; thus azimuth locomotion is friction type.

Each bogie contains a pneumatically released, spring applied fail safe braking arrangement, exerting a brake foce on each of the eight truck wheels during parking or emergency stop conditions.

The truck assembly is designed to accept vertical loads in both up and downward direction; however, upward load can only be taken in designated stow positions containing an anchor arrangement.

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The four drive wheels on each truck will have sufficiently high contact loads even under "drive to stow" conditions so as to be capable to generate the required traction without slippage.

Four such truck assemblies are required to support the weight of the telescope and the loads generated by wind forces and moments. Two additional trucks having reduced load capacity will furthermore be required to provide for auxiliary support.

The gear trains are connected to the truck frame by means of a link, so that the gear box is only rotationally restrained, to counteract the tangential friction drive forces acting on the wheel radii and the wheels are therefore free to follow the rail contour. The effect of this arrangement is that only tangential drive forces are introduced at the respective points of load transfer.

4. Elevation Drive Arrangement

The conceptual idea for the elevation drive arrangement is basically to apply an articulating drive mount which can follow the contour of the elevation gear in an unrestrained fashion and which is simply connected to the tower structure in such way as to eliminate moments or large vertical loads at this point.

This is achieved by means of a guided drive housing containing four flange mounted drive trains engaging in the elevation gear. This housing floats on the elevation gear support ring and is linked to the tower structure at point 6.

The elevation drive arrangement is thus unaffected by pitch radius tolerances of the elevation gear or dimensional changes due to temperature variations. Backlash in the link is eliminated by preloading of the linkage bearings. The concept is shown on Drawing 109-D-008.

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C. DRIVE AND CONTROL SYSTEM

DC motors with tachometer feedback controlled by solid state amplifiers have been found to be the best suitable drive power source. Application of other types of prime movers such as variable frequency, voltage controlled AC motors, eddy current clutches, DC torque motors and amplidyne controlled DC motors have also been considered, but were found to be less practical.

The use of 16 drive units in azimuth and 4 drive units in elevation was considered most desirable as it results in good symmetry in azimuth (4 drive units per truck, one on each outer wheel) and in minimum mounting space in elevation.

The recommended drive method is to control the power units so that during tracking operation, when maximum accuracy is required, 50% of the units will generate the required drive torque while the other 50% will be torqued in reverse in order to eliminate play or backlash in the truck assemblies or elevation gear. During periods of maximum torque demand as, for instance, during "drive to stow" operation, all units are to be controlled to operate in parallel as "bucking" or high position accuracy is no longer required.

15 HP - 2500 RPM totally enclosed, fan cooled DC motors geared down so that a slew speed of 18°/min. can be achieved at maximum motor speed, were considered in azimuth, while 40 HP - 1750 RPM units and a maximum slewing speed of 12°/min. were considered in elevation.

The control system requirements were studied in sufficient detail to assure that stable operation under the effect of disturbance due to environmental conditions (wind gusts, etc.) will be feasible and that the desired high pointing accuracy will be attainable with the application of the optical reference system, assuming that the lowest locked rotor natural frequency of the entire system will be above 1.0 cycles per second.

Interlock requirements for protection of the parallel operating servo motors and amplifiers have been studied along with reference system interface problems, generation of position error signals and mechanical feedback path, including feedback circuitry for correction of torque and position differentials between the trucks.

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D. DYNAMIC ANALYSIS

A study was made of the free vibration mode shapes and frequencies of the entire telescope assembly. Analyses were accomplished by use of a digital computer structural dynamics program developed by the Re-entry Systems Division of the Philco-Ford Corporation.

Two basic structural models were considered: a) a "truss" model where all masses were concentrated at the joints, and b) a "frame" model, where some masses were concentrated at the midpoint of various members and member bending flexibilities were part of the input. All joints were considered to be pinned and the reflector was modeled as a concentrated mass and inertia supported on springs representative of its own stiffness, in both cases. The truss model had 42 degrees of freedom and the frame model had 76.

It was found by comparing the results obtained with the consideration of these models that only the higher modes appear to be influenced by member flexibilities, whereas the fundamental mode was not affected. Several different combinations of structural configuration and flexibilities were thus studied and the final considered reflector and mount concept resulted in a lowest computed resonance of 1.08 C.P.S.

It was also found that the application of a standard gravel ballast railroad track as the azimuth turntable base as suggested by Dr. V. Hoerner would reduce the lowest resonant frequency of the system by about 10% compared to the resonance of 1.08 C.P.S. of the telescope system with a concrete crane rail turntable base.

"Simpson, Gumpertz and Heger" have also computed the lowest resonances of the telescope system by considering a "truss" model for both tower and reflector structures and have found the lowest resonance to be at 1.05 C.P.S., which appears to be in good agreement with the 1.08 C.P.S. computed by Philco. However, the reflector weights and masses used in their analysis were about 9% lower than those used in the Philco analysis. A comparison of equivalent reflector stiffnesses computed from the results of a dynamic analysis of the reflector "only" by Simpson, Gumpertz and Heger with the inputs used in the frame model analysis showed that a 20% lower X-direction stiffness should have been applied, which would have resulted in a lower resonance.

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Altogether, it appears most likely that the lowest resonance of the presently considered system occurs at around or slightly below 1.0 C.P.S. considering the lower stiffness that should have been used in the Philco program and the higher reflector weight that should have been applied in the S, G and H computations. The higher mode frequencies were found to be 1.49, 1.58, 1.60 and 1.62 C.P.S.; all due to excitation of single members in the tower structure.

It will be relatively easy to improve the dynamics of the telescope during the course of the final design effort by selectively improving the stiffness of major contributing areas as the mode shapes are now fairly well-known.

This should not result in a significant cost increase as it only involves additional material as the structural configuration remains unchanged. It is considered realistic to assume that a lowest systems resonance of 1.2 C.P.S. may be achieved in the final configuration.

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E.) COST ESTIMATE SUMMARY

DESCRIPTION	ESTIMATED COST	REFERENCE SECTION	REFERENCE PAGE NO.
Elevation bearings assemblies	$ 100,000	2.0	
Azimuth trucks	300,000	3.0	76
Foundation and track	135,000	3.0	75
Elevation gear and gear trains	190,000		
Pintle bearing arrangement	60,000	5.0	9
Drive control system	300,000	8.0	70
Reflector and tower structure	4,140,000)	See estimate	
Surface panels	700,000)	of	
Feed mount	50,000)	March 18, 1969	
Optical position reference system	350,000)		
Control computer	100,000)		
TOTAL BASIC TELESCOPE	$6,425,000		
TOTAL ANCILLARIES	470,000		
TOTAL ENGINEERING	650,000		
TOTAL COMPLETE TELESCOPE	$7,545,000		
CONTINGENCY (10%)	755,000		
GRAND TOTAL	$8,300,000		

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