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The Mechanical Design of the Horn-Reflector Antenna and Radome

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(Manuscript received March 15, 1963) 10883

This paper describes the mechanical design of the horn-reflector antenna and the associated radome. The mechanical considerations dictating the final configuration of the antenna structure are discussed, along with the engineering aspects of the design, fabrication, and erection of the structure. The mechanical features of the data take-off and antenna drive systems are given in detail. The final section presents an account of the requirements, manufacture, and installation of the radome and its accessory equipment.

AUTHOR

I. INTRODUCTION

The ground-based communication antenna of the Telstar project must figuratively project a needle of energy toward a 3-foot satellite at a distance of several thousand miles and listen for a whisper in return. It must perform in this fashion while moving about both azimuth and elevation axes. Furthermore, it must provide service in all weather conditions.

Many weeks of study preceded the determination of a general configuration for the structure, its support, and the method of control. The mechanical problems involved in achieving the required antenna performance are discussed in this article.*

The first part (Section II) describes the system requirements and the early design concepts. It also outlines the major factors governing the over-all design and points out the necessary compromises. The next part (Section III) describes the mechanical structure which evolved as the final design and tells of a number of the problems encountered during manufacture and erection.

* The information on concept consideration was provided by Mr. Dolling; that on the mechanical structure by Mr. Blackmore. Mr. Kindermann was concerned with the position data devices and the power drives, and Mr. Woodard with the radome.

Ltr Telstar 1, Vol. 2 Jun. 1963 p 1137-
56 refs (See N64-10882 02-01)
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The angles which define the pointing of the antenna must be accurately known at all times. The position data devices are among the most precise components in the entire Telstar system. The accuracy requirements for the data take-offs and their mechanical construction are given in Section IV, which also explains the means for driving the antenna about the two axes of motion.

An air-supported construction shelter provided a controlled environment during erection and alignment of the antenna. Later this was replaced by an air-supported radome which assures all-weather operation of the completed antenna. Section V presents a detailed account of the requirements, manufacture, and installation of the radome and its accessory equipment.

II. CONCEPT CONSIDERATION

2.1 *System Requirements*

The horn-reflector type of antenna was selected for the Telstar project because of its broadband and low-noise properties. It provides an offset feed arrangement with very good shielding between focus and reflector. This configuration also permits placement of the receiver at the focal point, thus minimizing line loss.

The velocities and accelerations necessary for tracking the Telstar satellite imposed the need for azimuth and elevation drives capable of operating over a very wide range of velocities, starting from zero. Fig. 1 shows the azimuth velocity requirements for two angles of peak elevation, φ. Accuracy, rigidity and mass inertia are the three basic parameters which determine the quality of the antenna:

2.1.1. A high order of accuracy is called for in the manufacture of all reflecting, rolling, or sliding surfaces. Accuracy of the reflecting surfaces determines the electrical efficiency of the antenna. Furthermore, provision must be made to let fixed and moving parts of the antenna rotate against one another about perpendicular axes and at very accurately controlled tracking speeds.

2.1.2. The antenna structure must present considerable rigidity against both external and internal forces. It is necessary to maintain true focal point-reflector orientation for all elevation and azimuth angles under the influences of gravity, driving acceleration, and thermal expansion forces. The structure also must exhibit high-frequency response to servo control.

2.1.3. The problem of mass inertia about the rotational axes not only calls for a compact design of short over-all dimensions and small radii

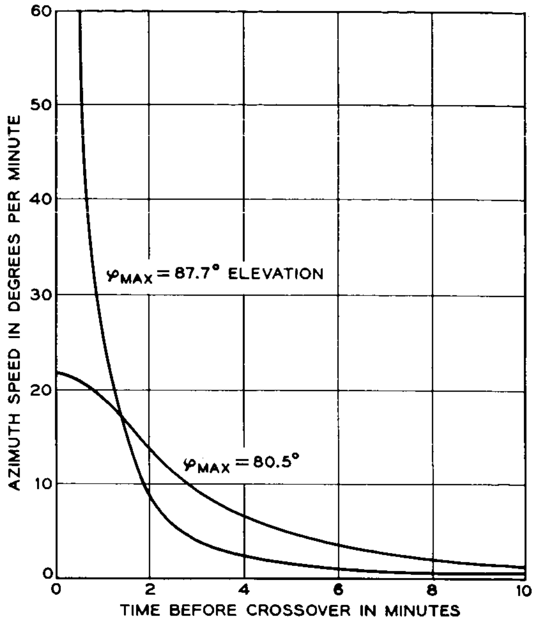


Fig. 1 — Antenna rotational speed for 2200-mile-high satellite in circular orbit.

about the axes but also imposes a very stringent requirement of light weight for every piece that moves with respect to the ground.

Finally, a reliable communication system requires that the antenna be operable under all weather conditions, and that provisions be made for survival in the event of loss of the radome.

2.2 *Investigation of Concepts*

Several antenna design concepts were investigated to find the one design that would best meet the stated requirements. Although the investigation narrowed down to studies of horn-reflector type antennas, early studies included paraboloidal and hemispherical dishes, and sectional sphere-type antennas. Consideration was given to the designs of the hemispherical dish and sectional sphere-type antennas because of the small mass inertia of those parts which require the most precise positioning, the high structural rigidity and compact design. Some of the major reasons for rejecting these types were feed shadow and long transmission lines, each of which contribute excessive noise.

Of the many horn-reflector antenna concepts investigated, the six

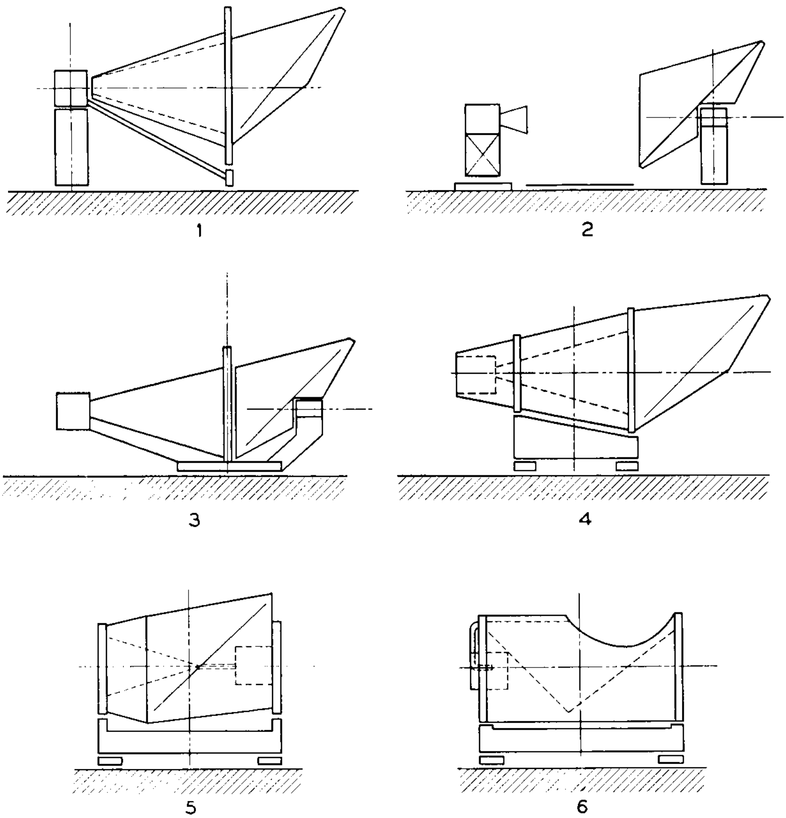


Fig. 2 — Horn-reflector antenna concepts.

that exhibited characteristics falling within system requirements are shown in Fig. 2. The main features of these concepts, the motives that warranted their consideration, and the major reasons for eliminating five of them, are discussed briefly in the following paragraphs.

Concept 1 is a four-sided pyramidal horn-reflector antenna with the horn's symmetry line horizontal and coinciding with the elevation axis. Rotation in azimuth takes place about the horn apex, which is identical with the paraboloid's focal point, so as to avoid translational motions of the equipment room. After study, this concept was abandoned, mainly because of the large turning circles in azimuth and elevation which would result in excessive wind torques and preclude the possibility of a radome because of the very large size.

Concept 2 does away with most of the horn while maintaining the geometry of the paraboloid. A metallic ground plane substitutes in part for the shielding effect of the horn. The axis of azimuth rotation is coincident with the reflector center. Both measures aim at minimizing wind and mass inertia torques. Uncertainties about effective noise temperature and synchronization of azimuth motions of reflector and feed excluded this scheme from further consideration.

Concept 3 is an attempt to reduce turning circles by using a conical horn and placing the azimuth axis at the largest diameter of the cone. Only the reflector rotates in elevation, but the entire antenna turns in azimuth on a small-diameter platform. Reasons for not following this layout were the excessive weight of the shell structure, lack of rigidity in the reflector structure, and restriction in the elevation motion.

Concept 4 maintains the advantages of a conical horn and a mid-way location of the azimuth axis but adds full elevating rotation, highest stiffness by one-piece design of the horn and reflector plus equipment room, optimum support against gravitational deflection by a cradle-like azimuth carriage, and lightweight space frame design. This concept incorporated most of the desired features.

Concept 5 essentially takes the elements of No. 4, introduces a plane mirror perpendicular to the horn axis and places equipment room and cradle under the reflector, thereby reducing the antenna's length by 40 per cent and improving the contour stability of the sectional paraboloid reflector. Unknowns about the noise temperature of this shortened horn arrangement precluded its adoption.

Concept 6, for the sake of minimum size and ultimate rigidness, opens the flare angle of the conical horn and tilts its symmetry axis, enclosing the horn-reflector antenna in an elevating structure of cylindrical outer shape. The length reduces by about 30 per cent, height by 35 per cent, compared with the antenna actually built. The necessity of a transmission line between horn apex and equipment room was the reason for rejection of this concept.

In the selection of the final antenna concept, an evaluation was made of each on the basis of its ability to meet system and design requirements. Table I illustrates the method employed of comparing and determining the compatibility of the above antenna concepts with the requirements. The concept numbers correspond to those described above, above, and the check marks indicate where the stated requirement is fully met.

It can be seen by examining the columns of Table I that only concept 4 appears to meet all of the basic requirements. Thus concept 4 was selected as the most suitable design for present use in the Telstar program.

Table I—Evaluation of Antenna Concepts

Basic Requirements	Concept					
	1	2	3	4	5	6
System:						
Rotating equipment rooms	√	√	√	√	√	√
Minimum of 180° elevation coverage	√	√	—	√	√	√
Tracking velocity and acceleration	—	√	√	√	√	√
Pointing accuracy	—	—	—	√	√	√
Beam spread stability	—	—	√	√	√	√
Electrical shielding	√	—	√	√	—	√
Minimum wind torques	—	√	√	√	√	√
Thermal differential expansion:						
Ambient temperature	—	√	√	√	√	√
Radiation heat	—	√	—	√	√	√
Operation in extreme precipitation	√	√	√	√	√	√
Survival (hurricane winds & ice)	√	√	√	√	√	√
Life expectancy	√	√	√	√	√	√
Design:						
No feed shadow	√	√	√	√	√	√
Horizontal RF feed	√	√	√	√	√	√
Short, straight transmission line	√	√	√	√	√	—
Minimum mass inertia about azimuth axis	—	√	√	√	√	√
Torsional and mounting rigidity of drive	√	—	—	√	√	√
Rigid structure	—	—	—	√	√	√
Thermal expansion restraint	—	√	√	√	√	√
Minimum deflection suspension	√	√	—	√	√	√
Stable foundation	√	√	—	√	√	√
Stow position	√	—	—	√	√	√
Accessibility of equipment room	√	√	√	√	√	√
Protective radome adaptability	—	—	√	√	√	√

2.3 *Basic Design Considerations*

With the selection of concept 4 as the most desirable antenna, several fundamental questions had to be answered before detailed design could start. They are discussed in the following paragraphs.

2.3.1. What is the best length-to-height ratio of the entire antenna structure? It had been decided that the effective aperture area for the radio beam should be 3600 square feet,[1,2] that the reflector should be a section of a paraboloid, and that the cone axis should coincide with the horizontal elevation axis. Consideration of the location of the center of gravity, the azimuth turning diameter, and the preferred flare angle influenced the choice of the length-to-diameter ratio.

2.3.2. What kind of structural pattern should be chosen for the skeleton of the horn enclosure? A polygonal pyramid of approximately round cross section requires an elevation turning circle 30 per cent smaller than the elevation circle of a horn with a square cross section. Furthermore, it can be made much more rigid against bending, shear, torsion, differential thermal expansion, and cross-sectional deformation. Also, deflection of the polygonal structure in the vertical plane remains constant with all elevation angles, and finally a round horn is more suitable electrically. The spiral pattern of the structural members has no parts that are idle at any time and thereby provides the most efficient configuration for a high rigidity-to-weight ratio. The lengths of the members were chosen in accordance with resonance and buckling requirements. A shell design would necessarily become prohibitively heavy to prevent skin buckling.

2.3.3. What is the most suitable type of configuration for the cradle structure? To achieve the highest rigidity-to-weight ratio of the entire structure, lattice-box girders are used for the cradle. The structure has the shortest and most direct load-carrying connections between the elevation assembly and the foundation. Minimum deflection support is provided for the elevation structure in two planes. Stability against overturning moments is achieved by use of a sufficiently large base frame. The least amount of material is required and a high natural frequency of structural members is obtained.

2.3.4. Which structural shapes should be used for building the horn enclosure? Structural tubes are three times lighter than any other structural shape for the same static deflection, and their resonant frequency between joints is about 70 per cent higher than that of other shapes of the same weight.

2.3.5. What material is appropriate as structural building material? Light weight and high rigidity are desirable in view of deflections due to gravity and high-frequency response to servo forces in the structure. Steel was selected because of its high modulus of elasticity. The rigidity of an aluminum structure of the same volume would be intolerably low when attached dead loads are considered, even though the resonant frequencies of the load-bearing skeletons may be equal. Increasing the volume of aluminum for stiffening would cancel the effects of savings in weight and mass inertia. Under a radome, an aluminum horn enclosure could at best save 20 per cent by weight of the present antenna and increase the natural frequency by about 10 per cent. Differential thermal expansions, due to sunlight in the open or because of air stratification in a radome, would double with aluminum. An all-aluminum structure of equal static and dynamic but worse thermal properties would cost about 10 per cent more than a steel structure.

2.3.6. Is damping feasible and needed to limit the amplitude of vibrations? Since the structure is designed to have no play in any of the joints, damping can be expected only from the material. This damping decrement is known to be no more than 0.05. With servo impulses applied directly, the structure may resonate, causing erroneous reactions of the servo. Many of the devices investigated for increasing internal damping proved to be either excessively heavy or ineffective because the permissible vibrational amplitudes were too small. External damping devices were also considered but were found to add drag to the rotational torques, thereby increasing the pointing error. Due to the autotrack's minimizing effect on directional corrections, and due to the need for converting digital information into analog, the torque changes originated at the drives are small. These changes are smoothed by gear and wheel friction so that little of this energy is converted to structural vibration. With the antenna in a radome enclosure, no wind forces are present to excite structural resonance. The relatively high natural frequency of the antenna, 2.2 cps, and the coupling of several antenna components with different higher frequencies, eliminate oscillations at resonance.

2.3.7. How should manufacturing and deflection tolerances be distributed among the structural components and foundation? Tolerance accumulation in an antenna system starts deep in the ground and is traced all the way up through the foundation, rails, trucks, azimuth carriage, elevation structure, and reflector suspension to the last panel surface. The distribution of the total allowable antenna pointing error for each part of the mechanical system has to be accomplished in accordance with the limitations of structural geometry, physical properties of materials, resonant frequency requirements, and currently available manufacturing techniques. Consideration also must be given to directions of errors and the probability of their occurrence.

Equations (1) and (2) below represent the total pointing errors about azimuth and elevation axes. Angular errors about the third axis, perpendicular to both azimuth and elevation axes, can be expressed in azimuth measure. Rotations about the nominal pointing vector result in equivalent elevation errors.

$$
\begin{aligned}
A = {} & 2\left\{\left[\beta_{GE} + \beta_{TE} - \left(\frac{c}{l}\right)_{GE} + \left(\frac{c}{l}\right)_{TE}\right]\tan\varphi \pm \beta_{ME}/\cos\varphi\right\} \\
& + (\delta_{GA} - \delta_{TE})\sin\varphi + (\epsilon_{GA} + \epsilon_{TE})\cos\varphi \pm \delta_{MA}\tan\varphi \\
& + \delta_{GE}\sin 2\varphi\tan\varphi + \epsilon_{GE}(1 - \cos 2\varphi) \\
& \pm \left\{\epsilon_{MF}^2 + \epsilon_{MA}^2 + \epsilon_{ME}^2 + \epsilon_{GF}^2 + \epsilon_{TF}^2 + \epsilon_{TA}^2\right. \\
& + (\delta_{MF}^2 + \delta_{ME}^2 + \delta_{GF}^2 + \delta_{TA}^2 + \delta_{TF}^2)\tan^2\varphi \\
& \left. + 4\left(\beta_{MEr}^2 + \left(\frac{c}{l}\right)_{ME}^2 \Big/ \cos^2\varphi\right)\right\}^{1/2}
\end{aligned}
\tag{1}
$$

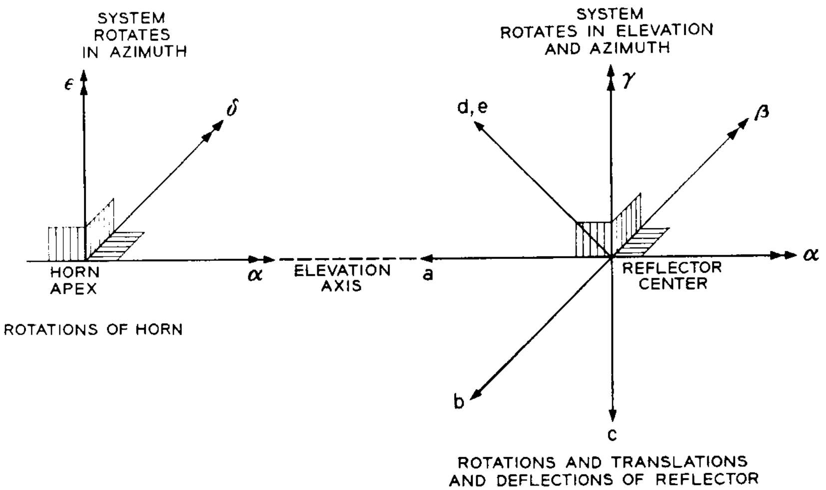
$$E = \left[\alpha_{GA} + \alpha_{GE} + \alpha_{TE} - \gamma_{GE} - \gamma_{TE} + \left(\frac{b}{l}\right)_{GE}\right] \cos \varphi$$
$$\pm \left[\alpha_{MF}^2 + \alpha_{MA}^2 + \alpha_{ME}^2 + \alpha_{GF}^2 + \alpha_{TF}^2 + \alpha_{TA}^2 \right. \tag{2}$$
$$\left. + \gamma_{ME}^2 + \left(\frac{b}{l}\right)_{ME}^2 + \left(\frac{b}{l}\right)_{TE}^2 \cos^2 \varphi\right]^{1/2}.$$

In these expressions, note that (*i*) the equations describe electrical pointing deviations with respect to ground due to structural imperfections and varying deformations; (*ii*) β_{ME} is usually chosen negative to compensate for β_{GE} at the zenith position, which is readily measurable at the time of alignment; (*iii*) δ_{MA}, even though preferably a residual value (r) after adjustment, has a permanent great influence so that its direction must be known for + or −; (*iv*) the equations contain signs for 0 to 90° elevation φ only; and (*v*) the temperature is assumed stratified in the radome for the elevation structure, random for the remaining structure. It is also assumed that there are no wind forces and that operational forces are negligible.

Fig. 3 shows the coordinate systems on which the above equations are based. With the help of this theory, it became possible to generate a tolerance budget for designers, manufacturers and erectors, as presented in Table II. Data for deflections due to servo or wind forces were left out as being negligibly small and zero, respectively. After the antenna design had been completed, the expected total pointing errors were calculated, in part compensated for by mechanical bias against gravitational deflection, and later compared with measurements from radio star tracking, Fig. 4. Agreement between calculations and measurements of variable errors was found to be excellent. The shaded areas in Fig. 4 indicate data scatter, due essentially to the influence of the rolling surfaces. The variable errors then were counteracted by data corrections as functions of elevation angle, so that predictable errors are entirely eliminated. Reflector panels were to be manufactured to within 0.03 inch in contour, and total accumulated deviation from the desired paraboloid was to be kept under $\pm\frac{1}{8}$ inch, so as to insure the least distortion to the beam.

2.4 *Additions to Basic Concept*

After the basic design concept had been selected, some features were added to the antenna to increase its performance and reliability. A radome excludes all influences by the elements such as wind, ice, snow and rain. To obtain the smallest possible azimuth turning clearance, the azimuth axis was shifted closer to the reflector. The four cradle sup-



α = ROTATION ABOUT ELEVATION AXIS
β = ROTATION IN PLANE OF ELEVATION AXIS AND POINTING VECTOR
γ = ROTATION ABOUT POINTING VECTOR
δ = ROTATION IN PLANE OF ELEVATION AXIS AND AZIMUTH AXIS
ϵ = ROTATION ABOUT AZIMUTH AXIS
a = DISPLACEMENT ALONG ELEVATION AXIS
b = DISPLACEMENT PERPENDICULAR TO PLANE OF ELEVATION AXIS AND POINTING VECTOR
c = DISPLACEMENT ALONG POINTING VECTOR
d = CORRUGATION HEIGHT OF REFLECTOR SURFACE
e = MEMBRANE DISPLACEMENT OF REFLECTOR

POINTING ERRORS:
A = AZIMUTH
E = ELEVATION
P = PITCH

DIRECTIONS:
+E = DOWN
+A = COUNTERCLOCKWISE

POINTING ANGLES:
φ = ELEVATION ABOVE HORIZON
ψ = AZIMUTH ROTATION

SUBLETTERS:
A = AZIMUTH STRUCTURE
E = ELEVATION STRUCTURE
F = FOUNDATION AND TRACK
G = GRAVITY
M = MANUFACTURING
T = TEMPERATURE

Fig. 3 — Tolerance coordinate systems.

port trucks no longer ride on one common double track but on two individual rails, each mounted on one of two concentric ring foundations. The trucks became two-wheeled and non-self-aligning. Therefore, both a pintle bearing on a central foundation and an extra structure inside the four structural planes of the azimuth carriage had to be added to keep the trucks on the rails against considerable friction forces. More struc-

TABLE II—STRUCTURAL TOLERANCES FOR POINTING ACCURACY

Symbol of Rotation	Tolerances[a] for			Influences[b] on pointing					
				Azimuth			Elevation		
	Mfg. (M)	Grav. (G)	Temp. (T)	A_M	A_G	A_T	E_M	E_G	E_T
				Foundation and track (F)					
α	0.10	0.05	0	0	0	0	± 1[c]	± 1	± 1
δ	0.10	0.05	0	$\pm 1u$[d]	$\pm 1u$	$\pm 1u$	0	0	0
ϵ	0.10	0.05	0	± 1	± 1	± 1	0	0	0
				Azimuth structure (A)					
α	0	0.05	0.05	0	0	0	± 1	$\pm 1w$	± 1
δ	0.05	0.10	0.05	$\pm 1u$	$+1v$[e]	$\pm 1u$	0	0	0
ϵ	0	0.10	0	± 1	$+1w$[f]	± 1	0	0	0
				Elevation structure (E)					
α	0	0.15	0	0	0	0	± 1	$+1w$	$+1w$
β	0.05	0.30	0.15	$-2x$[g]	$+2y$[h]	$+2y$	0	0	0
γ	0.15	0.50	0.15	0	0	0	± 1	$-1w$	$-1w$
δ	0.15	0.05	0.05	$\pm 1u$	$\pm 1u$	$-1w$	0	0	0
ϵ	0.10	0.05	0	± 1	± 1	$+1w$	0	0	0
b/l[i]	0.15	0.30	0	0	0	0	± 1	$+1w$	$\pm 1w$
c/l	0.05	0.10	0	$\pm 2x$	$-2y$	$+2y$	0	0	0

(a) Angular deviations in milliradians; (b) factors of magnification; (c) $+A$ counterclockwise, $+E$ down; (d) $u = \tan \varphi$; (e) $v = \sin \varphi$; (f) $w = \cos \varphi$; (g) $x = 1/\cos \varphi$; (h) $y = (1 - \sin \varphi)/\cos \varphi$; (i) l = horizontal distance between focus and reflector.

ture was added for the transmission of horizontal forces from the rear elevation bearing to the pintle bearing, because the elevation bearing was fixed axially. To allow for expansion at a later date, the equipment room at the apex of the horn was not enclosed within the elevation structure but was placed directly on the azimuth cradle. A lower equipment room located on the azimuth cradle had not been anticipated in the original design but was found to be a necessary addition. Both equipment rooms represent dead loads and therefore impose considerable penalties to the antenna in terms of weight and inertia. Structure-mounted positive drives were employed instead of truck-mounted friction drives to insure slip-free operation under all conceivable conditions, thus imposing high accuracy requirements on the location of the center of azimuth rotation. A view of the antenna model, showing most of the above features, is seen as Fig. 5.

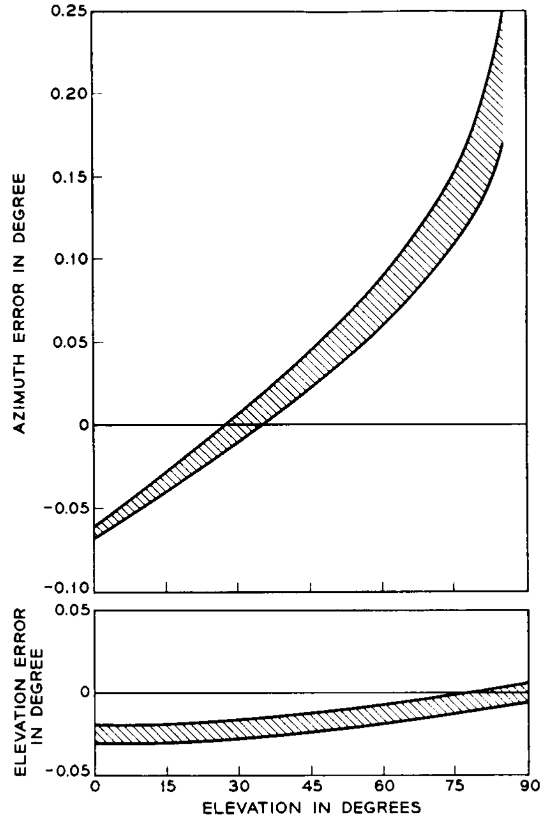


Fig. 4 — Calculated structural variable pointing errors.

III. MECHANICAL STRUCTURE

3.1 *General Configuration*

Fig. 6 is an outline drawing of the horn-reflector antenna. This antenna, which is a greatly enlarged version of the horn-reflector antennas employed in the Bell System's transcontinental microwave relay network, can best be visualized as two separate structures — an elevation structure rotating about a horizontal axis and an azimuth structure rotating about a vertical axis. The elevation structure is an extremely rigid framework which houses a conical horn some 90 feet long and supports a reflecting surface measuring approximately 100 by 70 feet. The horn terminates at its apex in an enclosed room housing the transmitting

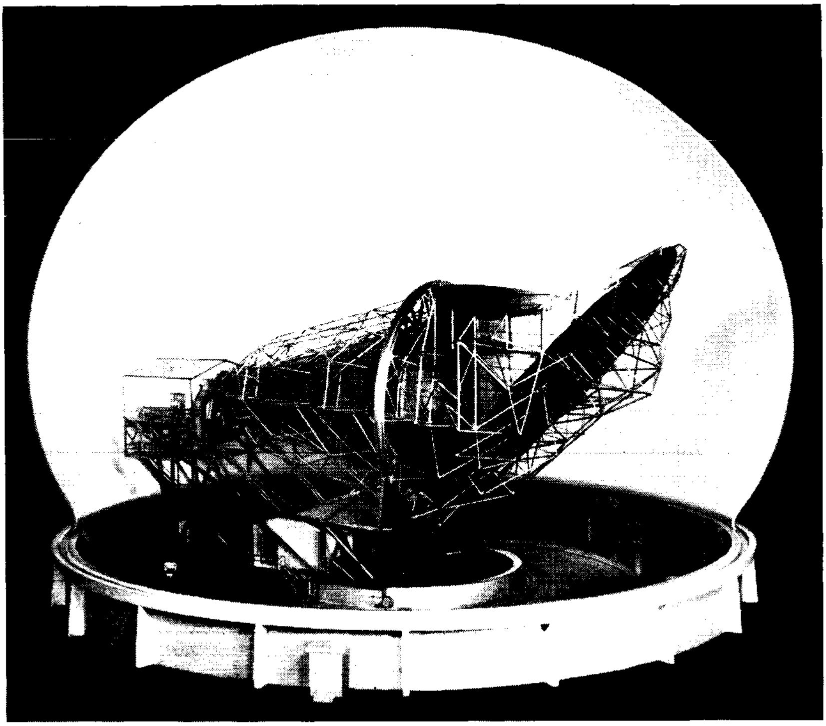


Fig. 5 — Model of Telstar horn-reflector antenna.

and low-noise receiving equipment. At its large end the horn terminates in the reflector, which is a sector of a paraboloid of revolution set at an angle of 45 degrees with the cone axis. The space between horn and reflector is occupied by a cylindrical structure which serves as a shield for the 67.7-foot diameter aperture of the antenna. Fig. 7 illustrates the relation between the conical, cylindrical, and paraboloidal surfaces comprising the horn, shield, and reflector.

The entire antenna structure is 177 feet long, stands some 95 feet above the azimuth rails at maximum elevation angle, and weighs about 380 tons with all equipment installed. The McKiernan-Terry Corporation of Dover and Harrison, New Jersey, were the prime contractors responsible for the detailed design, fabrication, and erection of the structure, guided by the basic design developed by Bell Laboratories. Subcontractors responsible to McKiernan-Terry, and working in close cooperation with Bell Laboratories engineers, included Burns & Roe, Inc., of New York City, responsible for the antenna and radome founda-

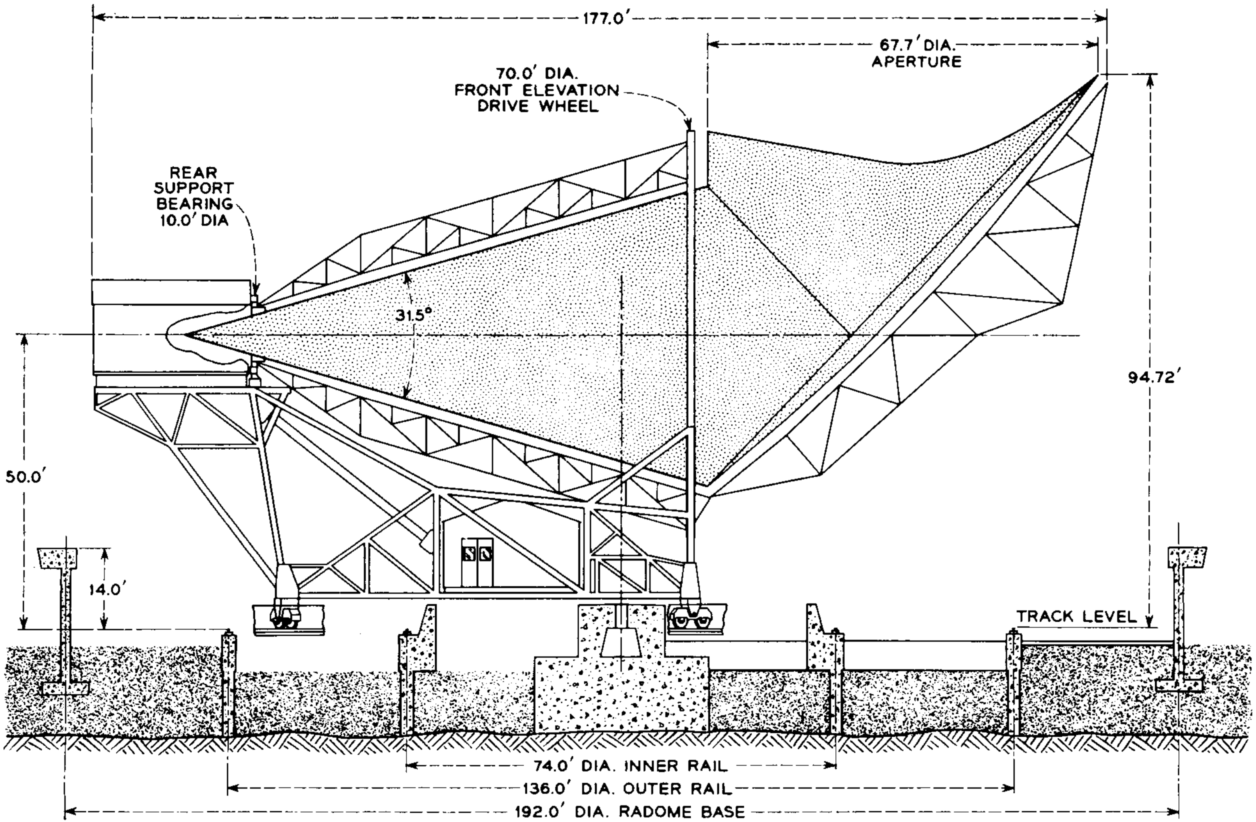


Fig. 6 — Outline drawing of horn-reflector antenna.

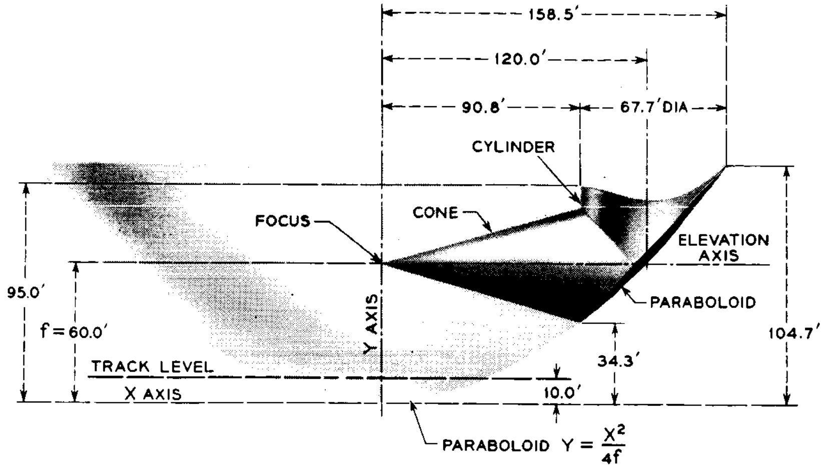


Fig. 7 — Relationship between horn, shield, and reflector surfaces.

tions; Radio Construction Corporation of Pittsburgh, Pennsylvania, erectors of the structure; and Advanced Structures of La Mesa, California, who fabricated and aligned the cone, shield, and reflector surfaces of the elevation structure.

3.1.1 *Elevation Structure*

The elevation structure is supported at two diameters and is provided with a choke joint near the cone apex to permit continuous rotation about the horizontal axis of the conical horn. At the large end of the cone, this support is provided by a circular track ring, integral with the structure. Near the apex, the support, as well as the axial restraint, is provided by a self-aligning ball bearing 10 feet in diameter. The wheels of the two trucks supporting the 70-foot diameter elevation track ring are provided with ball splines to accommodate any breathing of the elevation structure brought about by temperature differences between it and the supporting structure.

3.1.2 *Azimuth Structure*

The azimuth structure serves as the cradle to support the elevation structure. Rotation about a vertical axis is provided for by four trucks supporting the entire weight of both structures and running on two circular and concentric tracks. All horizontal reactions associated with azimuth rotation are assumed by a self-aligning roller bearing three feet in diameter and located at the center of the concentric tracks. This pintle bearing is isolated from vertical reactions by means of a flexible diaphragm that forms the connection between the outer race of the bearing and the rotating structure. Continuous azimuth rotation is afforded by a slip ring assembly for electrical power and signal circuits, and by a rotary joint for the chilled water required for cooling equipment on the structure. The slip ring assembly, which is an adaptation of a previous design, accommodates 295 electrical circuits: 28 for heavy power, 21 for signal circuits and 246 for general purpose.

3.1.3 *Equipment Rooms*

Two equipment rooms are mounted on the azimuth structure. One, commonly called the upper room, is at the cone apex. Its floor area is 900 square feet, and it houses the Telstar communication transmitters and receivers as well as associated electronic gear and the elevation angle indicating devices. The other room, commonly called the lower room, is located beneath the horn section of the antenna and just to the rear of the pintle bearing. Its floor area is 1600 square feet, and it houses the alarm and signaling equipment, the antenna digital control system, the antenna servo system, the station monitor system, the autotrack system, a transmitter for use in the NASA project Relay experiment, and various items of auxiliary equipment.

3.1.4 *Antenna Foundation*

The antenna foundation consists of three main elements. At the center of azimuth rotation is the massive reinforced concrete pintle block keyed into granite bedrock some 15 feet below grade to enable it to withstand a horizontal reaction of 300,000 pounds which could develop if, in the event of a hurricane, the protective radome were lost and the structure were subjected directly to the full force of a 100-mph wind. The two concentric reinforced concrete rings for the azimuth tracks are also supported directly on bedrock to minimize localized vertical deflections that would result in antenna pointing errors. These foundations are designed so as to contribute no more than 0.005-inch differential vertical deflection between any two azimuth trucks as the 380-ton antenna rotates in azimuth. The two track rings are tied to each other and to the pintle block by eight radial concrete beams to further increase the stability of the entire foundation. The inner ring has associated with it a wall supporting the azimuth bull gear. Fig. 8 shows the completed antenna foundation. The outer wall, which the photograph shows still under construction, is the foundation to which the radome was attached.

3.1.5 *Azimuth Rail System*

The outer azimuth track, 136 feet in diameter, is made up of 24 segments of heavy-duty crane rail weighing 171 pounds per yard, and the 74-foot diameter inner track contains twelve segments of the same weight rail. These segments were rolled to the proper curvature by the fabricator, Bethlehem Steel, ground to produce a flat and horizontal upper surface in another shop, and then butt-welded to produce a continuous ring after installation at the site. Jacks were interposed between the rails and the foundation sole plates to permit leveling of the entire track system to within $\frac{1}{32}$ inch. The excess weld material at the butt joints was faired into the adjacent ground surface of the rail head with a rail grinder developed specifically for this task. After this operation, each rail weld was checked for proper hardness with a portable hardness tester, and was subjected to a dye penetrant test to disclose cracks and porosity.

Fig. 8 — Completed antenna foundation and partially constructed foundation for attachment of radome.

The inner rail is the most heavily loaded and deflects approximately 0.018 inch between jacks when subjected to the wheel load. However, because of the relative spacing between wheels and jacks and the equalizing action of the truck, a total vertical movement of only half this amount is realized by the corner of the cradle structure.

3.2 *Azimuth Structure Considerations*

3.2.1 *Structural Details*

The principal members comprising the azimuth structure are steel box girders having a cross section one foot square. These girders were fabricated of four common angle sections laced together by welding smaller angle sections between them in a diagonal pattern. The girders were shop-assembled, where practicable, into subassemblies, but these shop subassemblies had to be limited in size so they could be transported by truck. On-site assembly of these members, as well as those of the elevation structure, was accomplished by bolted connections. High tensile strength bolts, which develop highly efficient and rigid joints by virtue of the clamping action on the bolted members, were used for all critical connections. Every high tensile strength bolt was hand-torqued to a predetermined value to insure that the required clamping forces were developed.

3.2.2 *Truck Alignment*

The design and alignment of the azimuth trucks and wheels were engineered to provide, as nearly as possible, pure rolling contact between wheel and rail. Because the only geometric shape capable of providing pure rolling contact along a circular path on a flat surface is a cone, the azimuth wheels were designed as frustrums of cones, and each wheel was aligned at assembly so that the projected cone apex coincided with the point of intersection of the azimuth axis and the plane formed by the top of the azimuth rails. To accomplish this alignment, a mirror mounted on the inboard end of the wheel axle was used to collimate a transit located at the center of azimuth rotation and at the level of the rail heads. The wheel being adjusted was raised off the track by a few thousands of an inch so that it could be rotated during the alignment procedure to verify orthogonality of the mirror and the wheel axis.

3.3 *Elevation Structure Considerations*

3.3.1 *Elevation Wheel Fabrication*

With the installation of the large elevation bearing assembly at the rear and the elevation trucks at the front, the azimuth structure was essentially complete and ready to receive the elevation structure. The first member of the elevation structure to be erected was the 70-foot diameter elevation wheel. This wheel had been fabricated in six separate segments to facilitate truck shipment to the site. Upon arrival at the site, the segments were placed on wooden cribbing some ten feet above the radome arena floor and assembled to form a complete wheel, the plane of which was horizontal. The elevation track, consisting of twelve sections of 104-pound crane rail previously rolled to the proper curvature, was next installed and the twelve rail joints butt-welded to form a continuous track ring. The entire 220-foot circumference of this ring was then ground to produce a smooth and concentric supporting track for the elevation structure. For this operation a grinding rig was devised, consisting of a precision grinding head mounted on a radius arm pivoted about the wheel center from a pintle anchored to the floor. The radius arm was supported near the wheel circumference by a track provided on the upper face of the wheel. The arm was motor-driven to make one circuit of the wheel in about four minutes, and carried a platform at the outer end to accommodate the operator. After some 136 hours of grinding, a continuous track having the required surface finish and width was attained. With the installation of the elevation bull gear segments and a "strongback" to reinforce the wheel during the lifting operation, the complete assembly, weighing close to 50 tons, was then ready for erection.

3.3.2 *Elevation Wheel Erection*

The lifting of this huge assembly, first to a vertical attitude and then up onto the elevation trucks, was the most spectacular and critical phase of the antenna erection sequence. Because of space limitations imposed by the presence of the radome wall and a temporary construction shelter (see below), the cranes involved in this lift could not be deployed to their best advantage and, consequently, the combined lifting capacity of three large cranes, one 90 ton and two 60 ton, had to be employed. Extreme caution was necessary throughout the entire 8 hours required for the two operations to insure that the lifting efforts of the three cranes were carefully synchronized. Any tendency for one of the cranes

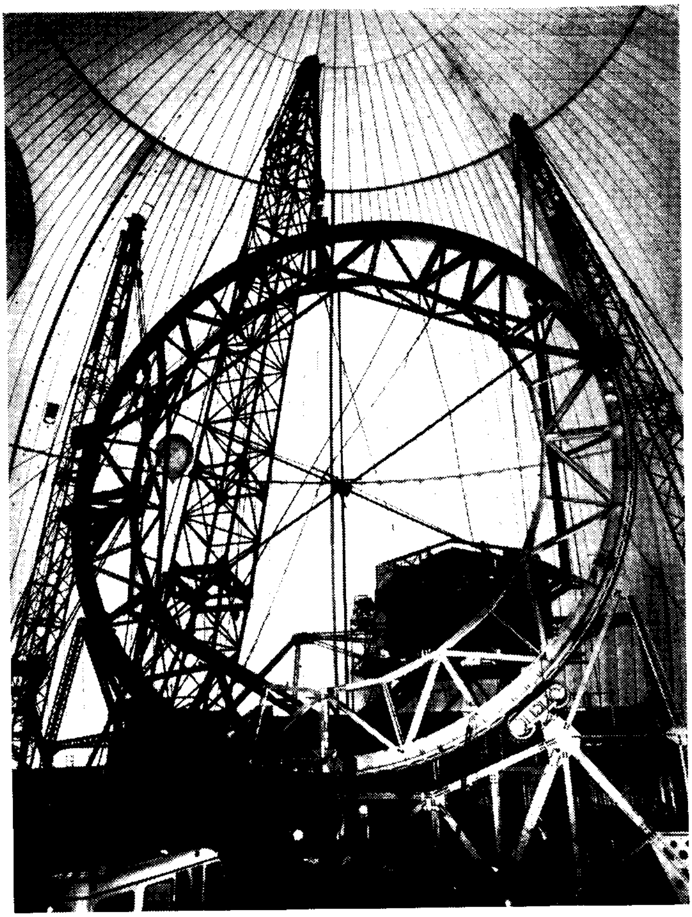


Fig. 9 — Final phase of installation of elevation wheel.

to assume more or less of its proper share of the total load could cause an imbalance, which would have resulted in an overloaded crane and would have led to disaster. Fig. 9 shows the final phase of the operation. The rear elevation bearing can be seen in the background.

3.3.3 *Cone Structure*

The next operation was the installation of the twelve cone rafters connecting the rotating race of the elevation bearing with the elevation wheel and forming the primary support structure for the cone as well as the means for keeping the wheel erect. These rafters, some 75 feet long by 8 feet deep, had been fabricated as weldments of steel tubing and had been jig-assembled at the shop to insure uniformity of all critical dimensions. Each rafter was made in two sections with a field-bolted joint between the two units to facilitate shipment by truck and passage through the vehicular air lock of the radome.

3.3.4 *Cone Structure Alignment and Panel Installation*

When the cone structure was complete, but before the panels comprising the cone surface could be installed, it was necessary to rotate the cone structure about the elevation axis to establish the center of rotation of the elevation wheel. Because of size and weight considerations it was not feasible to design this wheel to have negligible deflections. Furthermore, the cross section of the wheel had to vary from sector to sector to accommodate the nonsymmetrical loading the reflector structure would impose on it. Consequently, it was necessary to find the actual center of the wheel as it rotated in order to establish the nominal axis for the cone surface. To accomplish this, a theodolite mounted at the theoretical location of the cone apex was used to observe a target suspended in the plane of the wheel. The position of the target was then adjusted to produce a minimum orbit as the wheel rotated. The centroid of this irregular orbit was the desired center of rotation, and the required cone surface was centered about a line connecting this point and the cone apex. For adjusting the cone panels to conform to the desired surface, an optical instrument located at the cone apex and aligned to this centerline was provided with an angle attachment enabling it to sweep the required cone angle. The tolerance on the cone surface is $\pm\frac{1}{32}$ inch at the rear panels and increases linearly to ± 1 inch at the large end of the horn.

3.3.5 *RF Surface Considerations*

The panels comprising the cone, shield, and reflector surfaces were fabricated of aluminum honeycomb. Averaging 3 by 12 feet in area and weighing about 1 pound per square foot, these panels could be readily handled by two men but were strong enough to support safely the weight of the workmen during the installation procedure. To make these panels, two 0.020-inch sheets of aluminum separated by a 1- to $1\frac{1}{2}$-inch thick lightweight aluminum honeycomb core were form molded to conform to the desired curvature to within 0.030 inch. With the curing of the adhesive used to bond the three elements together, this curvature was "locked in," producing a panel having the required curvature plus an excellent compromise between weight and rigidity.

3.3.6 *Reflector Back-Up Structure*

Erection of the reflector back-up structure was performed with the structure oriented in elevation to point upward toward the zenith because, for this attitude, the nominal 45-degree angle between the reflector plane and elevation axis could be readily monitored by vertical measurements to a horizontal reference plane. Since this structure rotates about the elevation axis, and consequently with respect to the gravitational vector, its asymmetry results in a departure from the normal 45-degree relationship. This departure had been calculated and found to be within allowable limits. Pointing errors resulting from it could be compensated for by azimuth and elevation motions for any pointing angle except the zenith, where no amount of azimuth or elevation correction could overcome an outward or inward sag of the reflector surface. Hence, the zenith orientation was the most critical (as well as the most convenient) for monitoring the erection of this structure.

3.3.7 *Reflector Panel Installation and Alignment*

Upon completion of the reflector back-up structure, the entire elevation assembly was again rotated about the elevation axis, this time to determine the average center of rotation of the back-up structure. The centroid of the minimum orbit thus determined would become the starting point for the paneling of the reflector surface.

These panels, like the cone and shield panels, were provided with six threaded studs for attachment to the secondary support structure. The mounting nuts were appropriately positioned on these studs, and slotted clearance holes were used in the supporting members so that adjustment of the panel in three coordinates was possible. The panels, already curved in two directions by the fabrication process, were positioned to conform to the theoretical paraboloid of revolution to within 0.060 inch on a one-sigma basis by an alignment method developed jointly by Advanced Structures and Bell Telephone Laboratories. Paper targets, cemented to the front surface of each panel in locations corresponding to the support points, were observed from two theodolites located on a predetermined base line established on the radome floor. For this procedure the elevation structure had been rotated to the nadir position and the entire reflector was therefore readily observable from the floor-mounted instruments. The angles observed from the two theodolite positions were fed to a computer located within the construction shelter and programmed to produce numerical values indicating the departure, in a direction normal to the surface, of each observed target from the theoretical paraboloidal surface. These values could be used directly to move the target points of the panels in or out until they coincided with the desired reflector surface. Corrections for gravitationally induced deflections of the back-up structure and for dimensional changes due to temperature variations were incorporated in the computer program. Fig. 10 is a contour plot of the

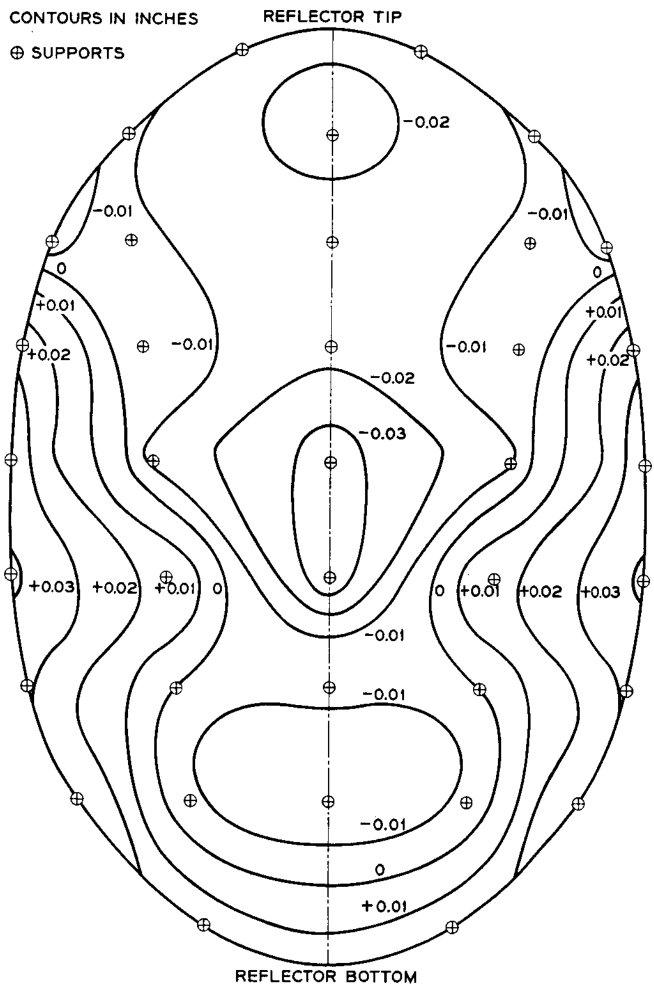


Fig. 10 — Elastic reflector contours.

local deflections compensated in this manner. The downward sag of the plane of the entire reflector frame from the nominal 45° angle also had to be included in the correction, since the effect of the gravitational vector was reversed as the structure was rotated from the zenith position in which it had been aligned to the nadir position employed for panel alignment. Actually, only three-fourths of the full correction was applied. This resulted in a nominally correct reflector surface at 30° elevation angle instead of at 90°. This represented a departure from the original intention to produce an optimum reflector surface at the zenith pointing angle. The compromise was dictated by the consideration that the antenna would never be required to track through the zenith and would spend comparatively little time tracking at the higher elevation angles. Only every other panel was observed, computed, and adjusted. The intervening panels were then faired to the adjusted panels. Shield panel adjustment was a much more simple procedure, the only requirement being that the cylindrical shield not encroach on the radiating aperture of the reflector surface.

With the adjustment of the shield panels and the taping of the joints between horn, shield, and reflector panels with aluminum-faced tape, the antenna structure was essentially complete.

IV. POSITION DATA DEVICES AND ANTENNA DRIVES

Important contributors to the successful performance of the horn-reflector antenna are the precise data take-off devices and the highly responsive hydraulic transmissions with associated low-compliance gearing in the servo drives. This equipment, operating under control of the antenna pointing system, must meet the antenna beam pointing tolerance of ± 0.019 degree under satellite tracking conditions.

4.1 *Data Take-Off Devices*

The narrow beamwidth and stringent pointing error requirements for the antenna dictated the use of extremely accurate data take-off devices on both axes of rotation.

The large primary reference data gears were produced by the Westinghouse Corporation, Lester, Pennsylvania. Type 410 stainless steel $\frac{3}{4}$-inch thick plate stock was used for the gear blanks, and the standard $14\frac{1}{2}$-degree pressure angle, full depth involute tooth forms were produced by hobbing. Table III lists pertinent data applicable to these gears as mounted on the hobbing machine after the final finishing cuts were made.

In addition, very precise reference diameters were machined on the gears just below the root diameter of the teeth. The reference diameters

TABLE III—GEAR DATA

	Azimuth	Elevation
Pitch diameter	40 inches	132 inches
Diametral pitch	32	16
Number of teeth	1280	2112
Tolerance, tooth to any other tooth	10 seconds	12 seconds
Tolerance, tooth-to-tooth	4 seconds	$1\frac{1}{2}$ seconds

were later used to accurately center the gears on the antenna axes for final installation.

The final inspection of the gears was conducted by Bell Telephone Laboratories personnel at the manufacturing plant with the aid of rather specialized instrumentation. Precise measurement of angles subtended by groups of teeth were made by means of autocollimation techniques used in conjunction with an optical polygon mounted on the hobber table axis. At each test position, a mechanical probe engaged a gear tooth at the pitch line. Errors in tooth flank angular position as evidenced by probe displacement were detected by an electronic indicator mounted on the probe support.

Gear tooth-to-tooth error and pitch circle eccentricity checks were made with a certified master gear mounted on a precision slide and rolling in engagement with the gear under test. An electronic indicator sensed radial movement of the master gear slide resulting from variations in pitch circle or tooth contours, and the indicator output was connected to a rectilinear recorder to produce a permanent chart record of the test. In correlation with this test, electronic indicator checks were made on the datum circle runout. Figs. 11 and 12 are photographs of the setups for these tests.

On the horn-reflector antenna, the azimuth data gear is mounted and centered on the stationary pintle post, and the data take-off devices are located on the adjacent rotating structure. The elevation data gear is secured to the rotating inner race of the ten-foot elevation bearing, and the data units are mounted on the face of the large pillow block weldment that secures the bearing to the azimuth structure. Thus the synchros and resolvers, comprising the data take-off units, are part of and revolve with the azimuth cradle of the horn. The associated signal leads can therefore run directly to terminal bays in the lower equipment room without passing through slip rings.

The elevation data take-off devices, driven by the rotating gear through anti-backlash pinions, are as follows:

(a) *Precision resolvers* at 1:1 and 64:1 antenna speed. These outputs

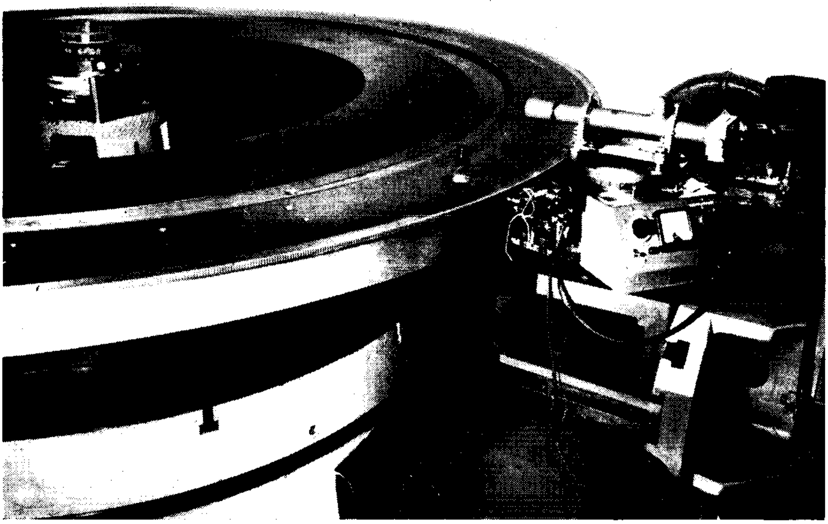


Fig. 11 — Test equipment for checking errors in gear tooth flank angular position.

are time-sequence sampled and translated to digital form for antenna position data input to the antenna pointing system.[3,4]

(b) *Synchros*: torque transmitters at 1:1 and 36:1 speeds are used for remote operation of slaved antenna position dials; control transformers at 1:1 and 64:1 speeds are used for manual servo loop control in position or velocity modes. These are used for off-line operation of the antenna.[5]

(c) *Boresight synchro* at 360:1 for initial boresighting and antenna pattern measurements.[2]

(d) *Autotrack coordinate converter*. This is a 1:1 speed synchro and potentiometer assembly used in conjunction with the autotrack system.[6]

Except for the omission of item (d), a similar arrangement is used for the azimuth data devices.

Fig. 13 is a photograph of the horn apex showing four data units clustered around one side of the 132-inch data gear. Fig. 14 shows two typical data units.

4.2 *Antenna Gear Drives and Hydraulic Transmission Systems*

The antenna utilizes hydraulic servo transmissions made by Vickers, Incorporated, and intermediate speed gear boxes made by the Western Gear Co. Two 25-horsepower and two 10-horsepower hydraulic trans-

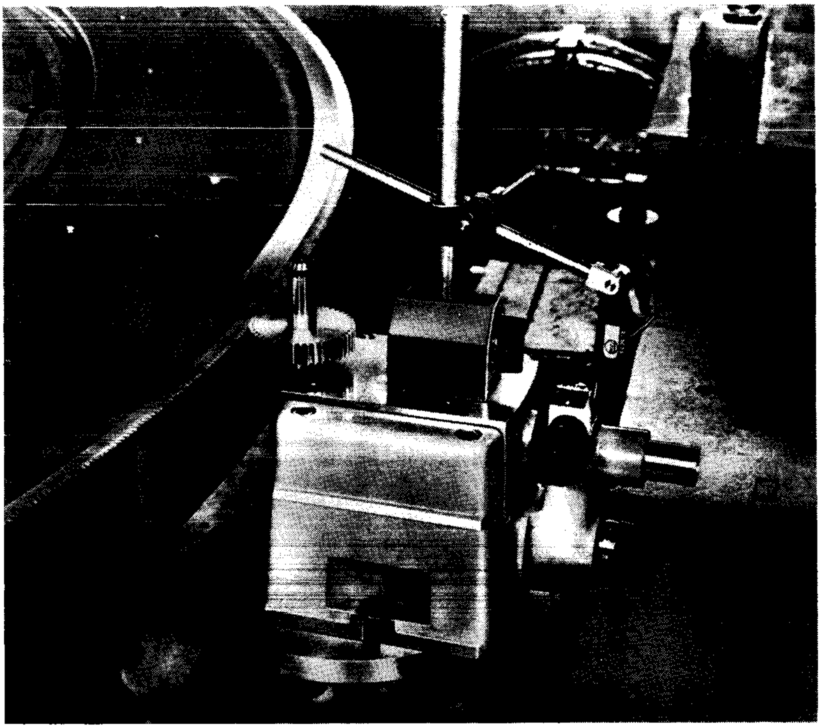


Fig. 12 — Test equipment for checking gear tooth-to-tooth error and pitch circle runout.

missions are used for driving, respectively, in azimuth and in elevation. Each transmission has two motor output shafts — one to drive and the other to provide a reverse torque for anti-backlash purposes as described below. Consequently, four complete gear trains are required for the drives on each axis.

4.2.1 *Drive Gearing*

The two azimuth drives are mounted at diametrically opposite locations on the horizontal framework of the azimuth structure. Fig. 15 depicts the arrangement of one azimuth drive. Fig. 16 shows the schematic of the gear trains.

It is noted that each transmission output shaft drives into the 11.32:1 input gear box which, in turn, drives the 22.59:1 output gear box. The 13-inch pitch diameter pinions engage the 64-foot pitch diameter bull

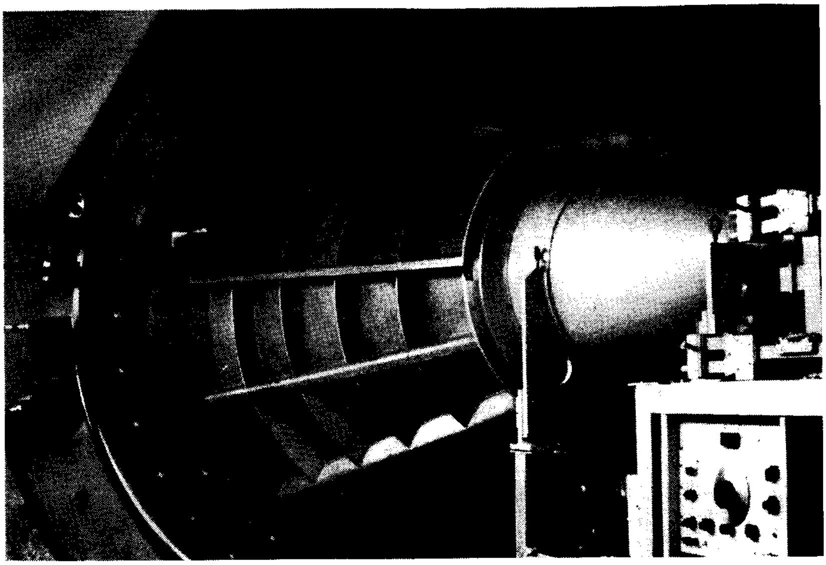


Fig. 13 — Horn apex showing four data units clustered on periphery of 132-inch data gear.

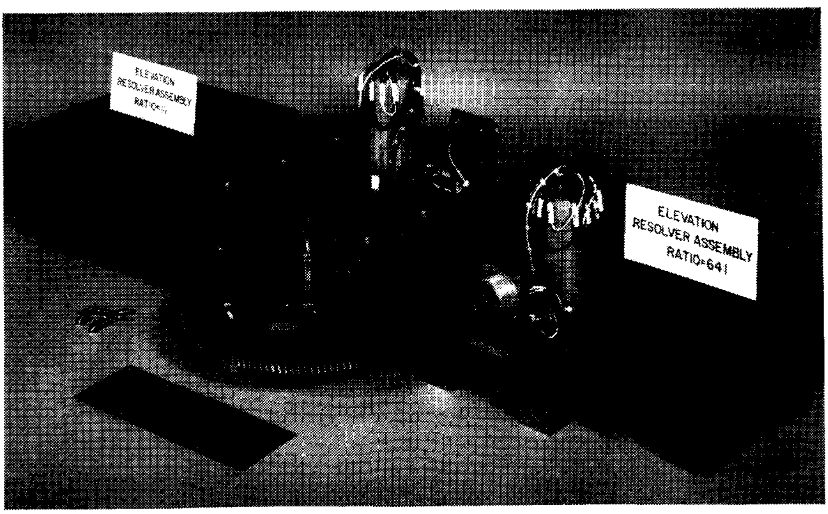


Fig. 14 — Two typical data units.

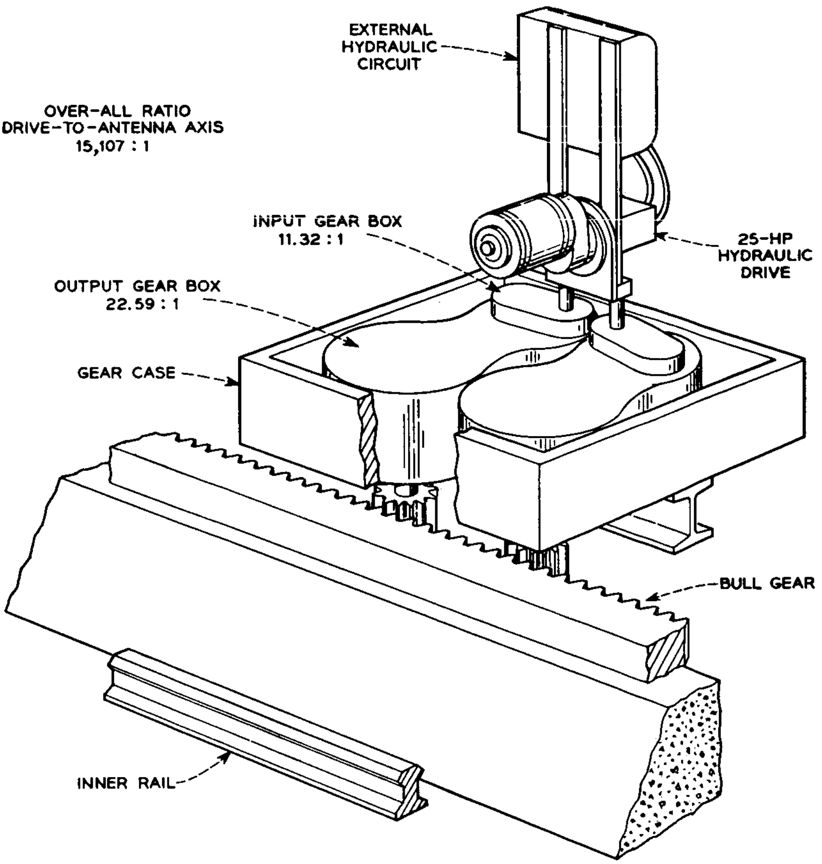


Fig. 15 — Schematic showing arrangement of one azimuth drive.

gear for an over-all drive ratio of 15,107:1; the 13-tooth pinions literally "walk" around the 768-tooth bull gear in rotating the structure.

The azimuth gearing is rather large. The output gear box is 54 inches long, 35 inches wide, and 22 inches high. A pair of boxes mount in a gear case that is 70 by 80 by 28 inches. A complete drive assembly, including the gear boxes and hydraulic unit, weighs $5\frac{1}{2}$ tons. Figs. 17 and 18 are photographs of an azimuth drive mounted on the antenna structure.

The azimuth bull gear is an internal gear of 64-foot pitch diameter and 6-inch face width, and is composed of 32 equal segments. To bring the gear as close as possible to the underside of the horn azimuth framework, the segments are bolted to the top of a five-foot high parapet that is part

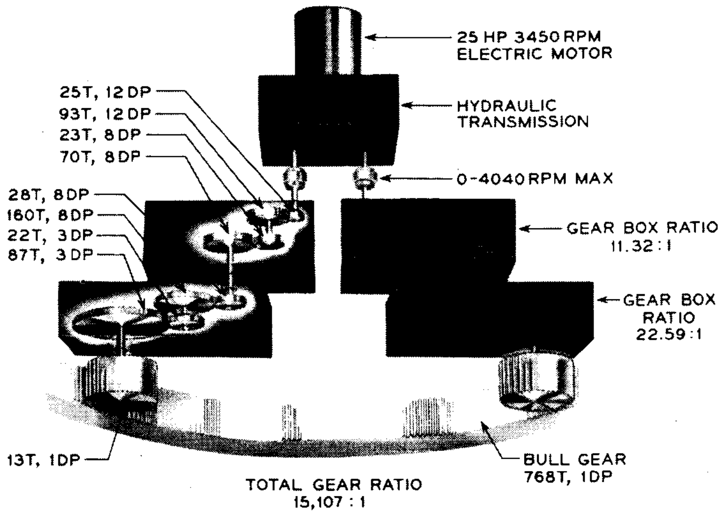


Fig. 16 — Schematic of azimuth gear train.

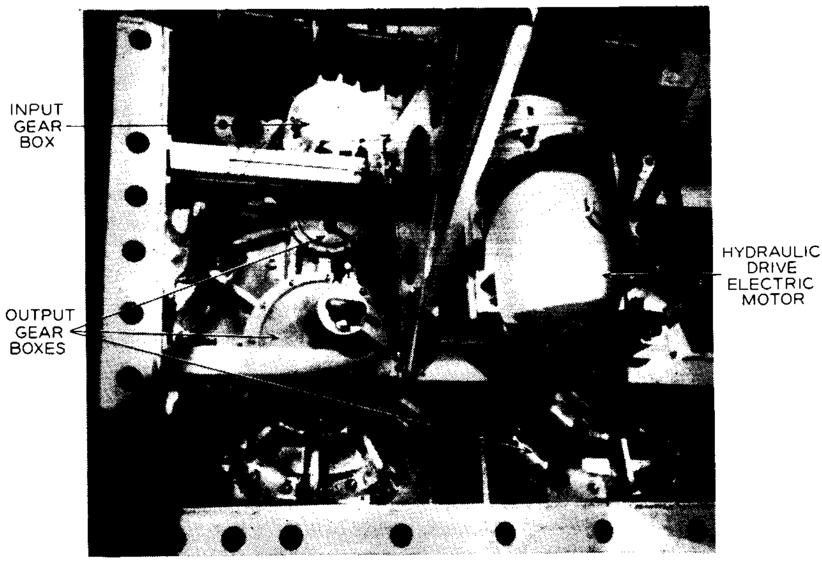


Fig. 17 — Azimuth drive mounted on antenna structure (top view).

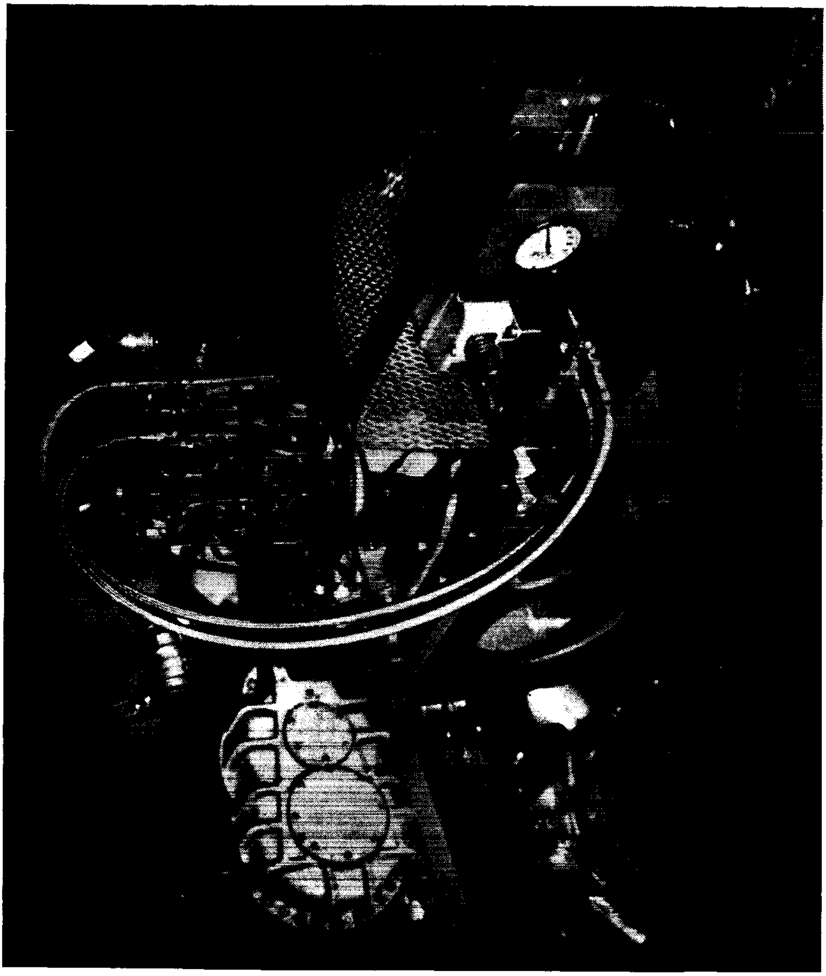


Fig. 18 — Azimuth drive mounted on antenna structure (side view).

of the foundation structure. By this means, the cantilever moment of the gear box and pinion and the resulting azimuth structure compliance are minimized.

Fig. 19 is a photograph of the azimuth drive pinions and a section of the bull gear.

The two elevation drives are mounted on the forward truss of the azimuth structure adjacent to the trucks that support the elevation wheel. Each assembly of the pair consists of the 10-horsepower hydraulic

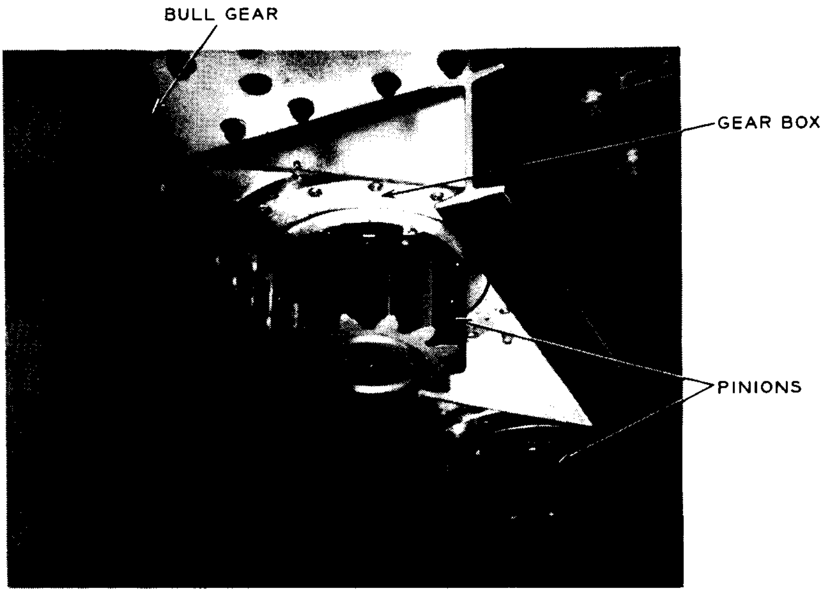


Fig. 19 — Azimuth drive pinions and section of bull gear.

unit which drives the 9.28:1 input gear box. This is the input to the final gear drive box of 23.52:1 ratio. The fifteen-tooth, 10-inch pitch diameter output pinions engage the $1\frac{1}{2}$ diametral pitch, 70-foot pitch diameter bull gear to yield an over-all elevation drive ratio of 18,334:1. The bull gear consists of 36 segments bolted and doweled to the periphery of the elevation wheel. The elevation drive gear schematic is shown in Fig. 20.

Fig. 21 is a photograph of the left-hand elevation drive mounted on the structure. Each gear box measures 48 by 38 by 18 inches, and the pair of boxes are mounted in the gear case of 88 by 90 inches. One complete drive assembly, including the hydraulic package, weighs approximately 3 tons.

4.2.2 *Hydraulic Drives*

The hydraulic drives are the prime movers for the antenna.

The 25-horsepower azimuth drive unit is shown in Fig. 22. The 10-horsepower elevation unit is quite similar in appearance and functioning but is slightly smaller.

The basic hydraulic circuit in somewhat simplified form is shown in Fig. 23. The power input to the system is a three-phase induction motor

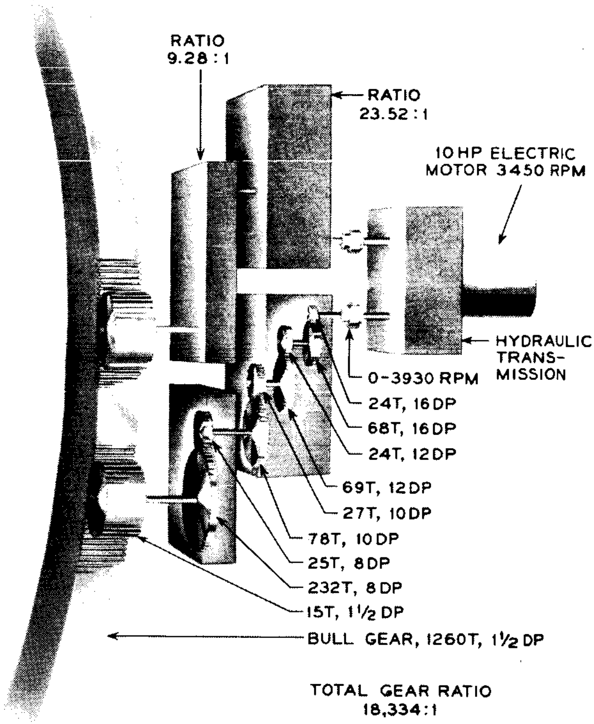


Fig. 20 — Elevation drive gear schematic.

that drives the servo-controlled main pump and two gear-type pumps for control and replenishing fluid pressure. The main pump is a nine-cylinder, variable flow, positive displacement type. As shown in Fig. 24, fluid is displaced when the shaft drives the pistons and cylinder barrel contained within the yoke housing. The yoke is mounted to swing 30 degrees either side of center about a pintle bearing, under control of the stroking cylinder. The flow is directed through ports in the stationary valve plate, and thence through passages in the yoke to the hollow pintle, where connections are made to the hydraulic motor circuit (see Fig. 23). The yoke angular position determines the displacement of the pistons and the direction of output flow.

The pump is connected in a series circuit with the two nine-cylinder hydraulic motors with intermediate connections through relief or di-

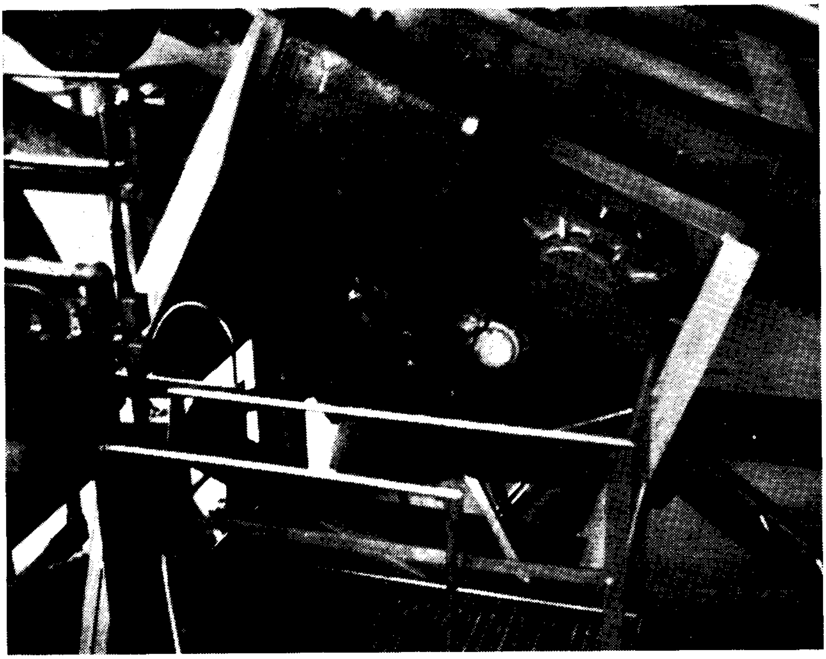


Fig. 21 — Left-hand elevation drive mounted on antenna structure.

rectional valves to the replenishing pump and sump. The hydraulic motors are similar in construction to the pump except that the stroke angle is fixed at 25 degrees. Fluid under pressure acting upon the pistons causes the cylinder barrel assembly to revolve and drive the output shaft. With the fixed angle between the cylinder barrel and shaft, the displacement per stroke is constant. Hence, motor speed is related to fluid flow rate.

For zero input signal, the servo valve, stroking cylinder, and main-pump yoke self adjust to the null or neutral position, and no flow occurs. However, the replenishing pump produces a differential pressure of approximately 60 psi across the two motors, tending to drive them in opposite directions with equal torque. The effect is that clearances in the gear trains are taken up in the direction that the motors normally drive. Because the pressure acting on both motors is equal, net driving torque applied to the bull gear is zero. This arrangement of the replenishing pump in the circuit effectively eliminates backlash in the gearing.

When an input signal actuates the torque motor, the directly coupled servo valve meters fluid at control-pump pressure to the stroking cylinder, causing it to move in magnitude proportional to the signal ampli-

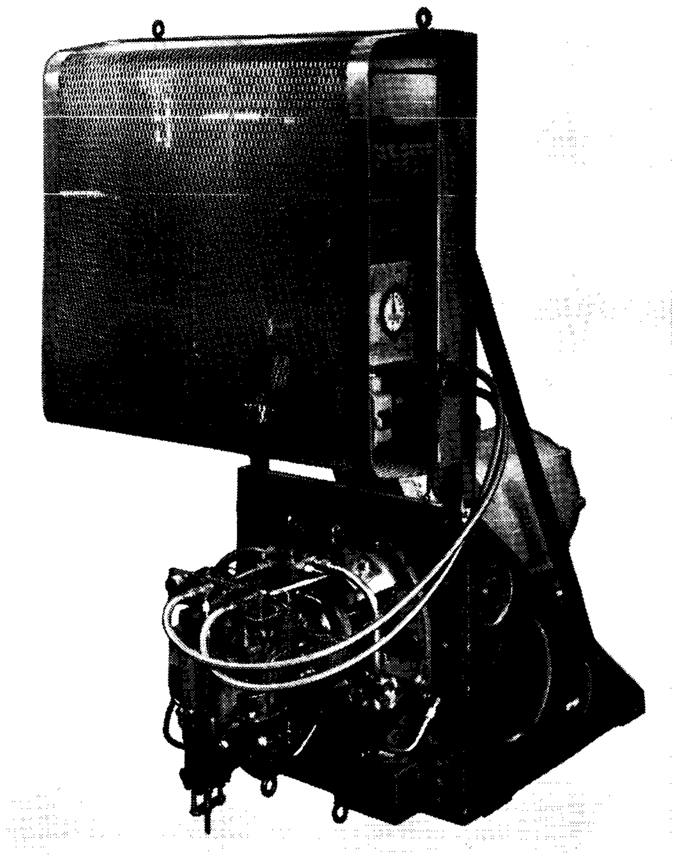


Fig. 22 — Vickers 25-horsepower azimuth drive unit.

tude and in direction in accord with signal polarity. By mechanical connection, the stroking-cylinder piston rod adjusts the position of the main-pump yoke to control a proportional pump output rate in the desired direction of flow.

If it is assumed that the flow is directed to motor 1, the increased pressure closes the directional check valve on the upper side of the replenishing pump connection and acts in the motor to produce driving torque. As antenna rotation occurs, the spent fluid passes to motor 2, which is driven by its gear train as a pump discharging fluid against the 60-psi replenishing pump pressure differential. The resulting light load on the gear train eliminates backlash.

When the input signal amplitude is increased, the pump yoke angle

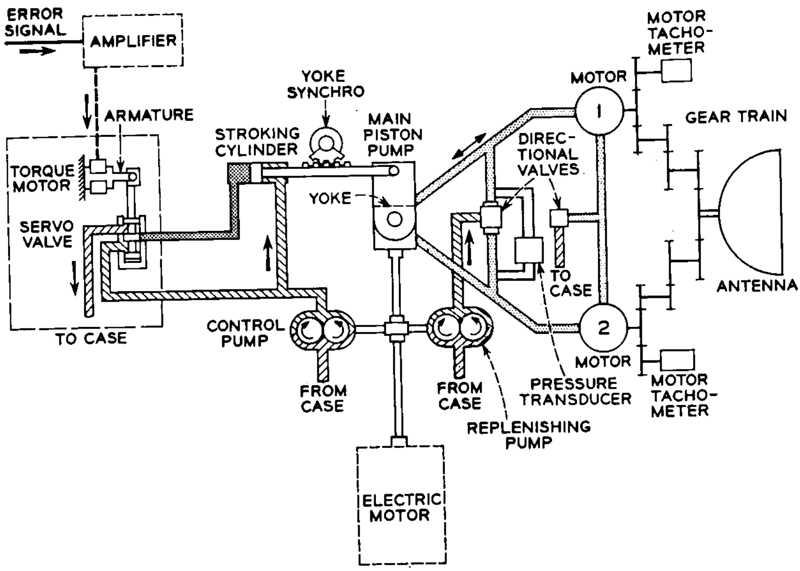


Fig. 23 — Simplified schematic of basic hydraulic drive circuit.

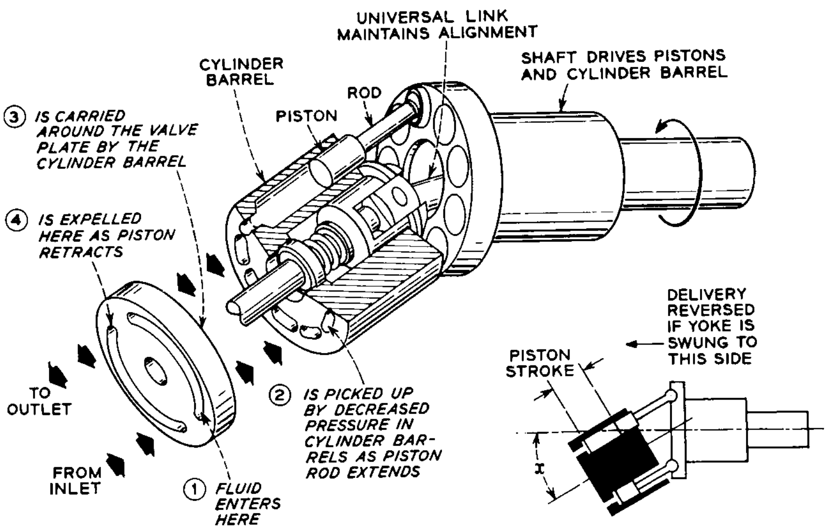


Fig. 24 — Mechanical schematic of piston pump.

increases correspondingly. Motor 1 operates at a higher output, but the retarding torque of motor 2 remains constant.

If a signal of opposite polarity is applied to the torque motor, the same sequence of operations drives the pump yoke to a position on the opposite side of center, direction of flow is reversed, and the two motors interchange their roles, reversing antenna rotation. Because the retarding torque is maintained at all times, reversal is smooth.

The pressure transducer (see Fig. 23) provides an electrical signal proportional to the pressure drop across the main pump, which is proportional to the work load on the hydraulic motor. This signal, as an input to the servo system, is used to equalize the output of the two hydraulic transmissions and to provide system damping. The yoke synchro and motor tachometers are also feedback inputs to the servo system for operational control.

The combination of the Vickers transmission and drive gearing provides remarkable responsiveness and antenna tracking capability. Speeds may vary from zero to maximum antenna slew rates of 1.5 degrees per second in azimuth and 1.4 degrees per second in elevation. As an indication of the acceleration capabilities, it should be noted that at slew speed the drives complete a full reverse command in an interval of less than three teeth on the bull gear. At the other extreme, as in tracking radio stars or under certain conditions during satellite tracking, rotation of one axis may be imperceptible. Particularly impressive is the response to sinusoidal servo input signals as high as 2 cps in frequency.

V. RADOME

5.1 *Need for Radome Coverage*

Early studies at Bell Laboratories and past experience based on tests of military antenna systems have proven that, despite the added system noise, a radome is essential to protect the horn-reflector antenna from the effects of enviornmental conditions. Some of the considerations and the advantages afforded by radome protection are discussed herein.

Without radome coverage, the antenna would be subject to the effects of the common elemental conditions of the area, such as solar heat, wind, snow, ice, rain, and dust. These conditions can seriously affect antenna performance.

(a) Direct solar heat, for example, can disturb the antenna reflecting surface contour accuracy so that the required patterns and gain are not achieved. Furthermore, differential solar thermal distortion can adversely affect beam pointing accuracy.

(b) The influences of wind on an exposed antenna are even more pronounced than those of solar heat. An antenna structure and high-gain antenna servo drive system can become unstable under buffeting winds as low as 30 miles per hour. Excitation or distortion under high wind loading also can jeopardize its tracking performance.

(c) Accumulations of ice, snow, and possibly rain, if suitable drainage is not provided, can substantially interfere with antenna performance. Such accumulations sometimes necessitate limiting the range of antenna motion about its axis; otherwise, there is the risk of introducing too great a turning moment for the servo drives to counteract. This hazard makes disposal of the accumulation difficult, as indicated by a recent experience at a large installation in New England. An attempt to dump a heavy snowfall by tilting the dish caused an excessive moment, resulting in complete lack of control of the moved axis. The snow had to be removed by hand-shoveling and sweeping—a hazardous and time-consuming task that encroached on vital test time.

Under radome protection, however, the antenna is nearly independent of ambient weather conditions and can be scheduled on an optimum basis. Not only does this protection avert the direct adverse effects of the elements on antenna performance, but it also retards physical deterioration. Maintenance requirements are drastically reduced when an antenna is housed in a radome. Moreover, within a radome, antenna maintenance can be conducted with a higher degree of confidence and reliability.

Other less obvious advantages are also derived. If the antenna is to be radome-housed, its structure and servo systems can be built for lighter duty than if it is to be exposed. When no direct provisions are needed to counter the effects of exposure, a greater latitude in design approach is permissible.

5.2 *Selection of Structure*

When the need for a radome was established, it became necessary to choose the type of protective structure that would meet all requirements. Rigid-foam, space-frame, and air-supported structures were considered on the basis of: (a) state of the art, (b) electrical performance, (c) cost, and (d) effect on the schedule. The air-supported structure won by a sizable margin on all counts.

At that time, the largest radome built, either rigid or air-supported, was 150 feet in diameter. The diameter of the protective radome structure for the horn-reflector antenna is 210 feet, which obviously called

TABLE IV — PERTINENT LOAD FIGURES FOR 210-FT DIAMETER TRUNCATED SPHERE

Inflation lift at 0.175 lb/in²	= 730,000 lbs
Aerodynamic lift in a 100-mph wind	= 844,000 lbs
Total lift	= 1,574,000 lbs
Aerodynamic drag	= 221,000 lbs
Aerodynamic moment	= 8,700,000 lb-ft
Fabric tension due to above loads	= 463 lbs/linear in

for a major step beyond the state of the art at that time for any kind of radome structure.

5.3 *Radome Design*

The principal loads imposed on the radome fabric under the most severe weather conditions are due to the internal pressure of the supporting air and the effects of the wind. These loads in a 100-mph wind, which requires an internal pressure of 0.175 lb/in^2, are listed in Table IV.

Fig. 25 is a good illustration of the aerodynamic load factors encountered on a truncated sphere supported on a base and placed in a free wind stream. The contour lines are on the surface of the sphere and

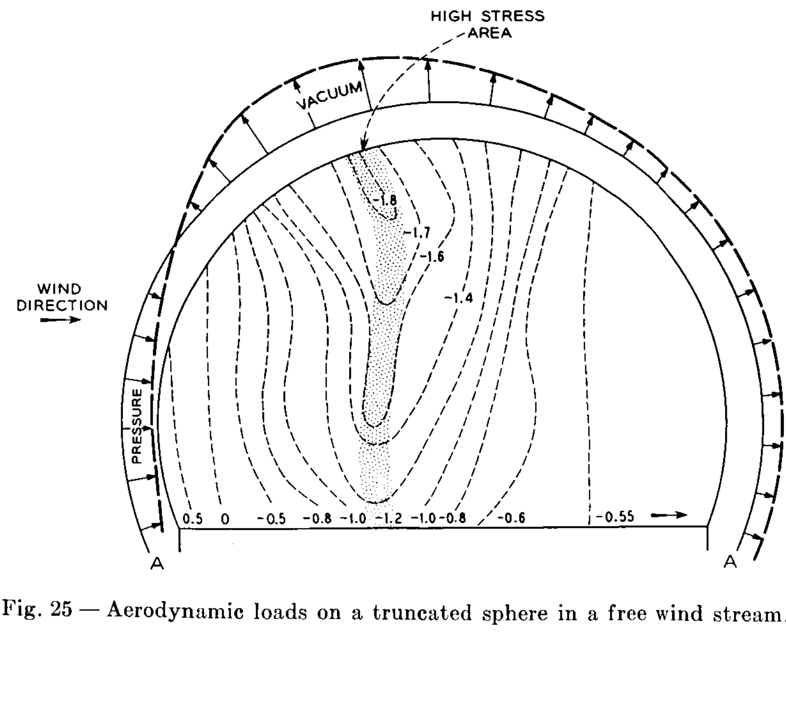


Fig. 25 — Aerodynamic loads on a truncated sphere in a free wind stream.

represent equal aerodynamic pressure of the values shown in the area contoured. A value of 1.0 is equal to the dynamic pressure of the wind for a given air velocity. Plus figures represent inward forces or pressure; minus values stand for outward forces or vacuum.

Fig. 25 reveals that, for a moderately high wind velocity, the aerodynamic suction forces about the sphere add to the internal pressure loading, becoming highest in a circumferential band perpendicular to the wind direction, as shown by the shaded area of high stress. The contour numbers in the figure also reveal that the particular area of highest stress is near the pole or crown. For the sake of clarity, the pressure values in a plane through the center of the radome and parallel to the wind have been plotted from an arc of a circle (A-A) used as a baseline (see Fig. 25). The directions of the arrows from the baseline indicate the direction of the reaction, depending on whether it is pressure or vacuum. The figure also pictorially shows that the aerodynamic drag force and moment about the base are high because of the additive impact pressure on the windward side and the vacuum on the leeward side.

The computations of aerodynamic and inflation pressure loads are straightforward and are all contained in a report by Cornell Aeronautical Laboratory.[7] Therefore, they are not repeated in this paper.

5.4 *Material Development*

The common pneumatic automobile tire could be considered a very distant relative of a large inflatable radome. Since it is a pressure-rigidized structure, it can serve to illustrate just one of the many difficulties in the design of a 210-foot diameter air-supported structure.

At an inflation of 28 pounds per square inch, a passenger automobile tire must withstand a tensile loading of approximately 100 pounds per inch. The tire carcass, of course, is designed to withstand puncturing bruises and road shock; therefore, it is made with a large safety margin of thickness above that required for the tensile load.

Unlike the tire, however, the radome must pass RF energy with minimum loss and pattern distortion. In this case, an undue safety margin of thickness would cause serious degradation of electrical performance. For a given pressure, as the diameter of a spherical pressure vessel increases, the tensile loading per inch of skin increases in direct proportion to the diameter. In the case of the radome, its required inflation pressure is a function of its wind environment. For the Telstar radome, the maximum wind environment capability is 100 miles per hour, and the corresponding inflation pressure is 0.175 pound per square inch (roughly $\frac{1}{100}$ of the tire pressure).

This low pressure might appear to be an insignificant problem in comparison to the 28 pounds per square inch required for the automobile tire. However, with very little calculation it can be shown that, when the 210-foot diameter radome is inflated, its skin is under a tensile loading of 150 pounds per inch, which is 50 per cent higher than the tensile load on the tire. To this load must be added the aerodynamic force of 300 pounds per inch, bringing the total radome skin tension to 450 pounds per inch (based on a 100-mph wind), or about $4\frac{1}{2}$ times the tensile load on the tire.

This load in itself does not appear to be a formidable problem; however, it becomes more drastic in view of the other requirements that are placed on the material. Besides sustaining the tension load, the material must also have the following characteristics:

(a) The base fabric must be stable, little affected by constant exposure to solar ultraviolet radiation, and must be highly rip resistant, with a sufficient margin of safety to allow for factory handling, erection, and anticipated life expectancy under load.

(b) Any protective coating for the base material must be tough, durable, and resistant to both abrasion and erosion from weathering. It must have good adherence to the base material and be capable of being bonded or cemented into joints that will exhibit characteristics equal to or better than the base material. For RF transmission, its moisture absorption must be low.

(c) For electrical reasons, the finished material must have a dielectric constant as low as possible (3.0 in the case of the Telstar system) and a thickness of no more than 0.071 inch, and must be held to an over-all thickness tolerance of ± 0.002 inch. The material must be light in weight to permit transportation and erection with a minimum of difficulty. To equalize stress and to preserve the aerodynamic shape, strength and stretch properties of the material in the warp and fill directions must be as close to equal as possible.

With these specifications to be met, the selection and development of the material obviously was a formidable problem. Because no off-the-shelf commercial material was available, the problem was solved by producing a special radome skin from the best raw materials obtainable. The base yarn, selected for high strength and ultraviolet resistance, was Dacron polyester fiber, hot-stretched and heat-stabilized. Investigation of loom capacity revealed that, to meet strength, rip resistance, and elongation requirements, the radome fabric had to be specially woven to the heaviest gauge within loom capacity. After coating, two such fabric layers were laminated. To distribute loads equally, the two

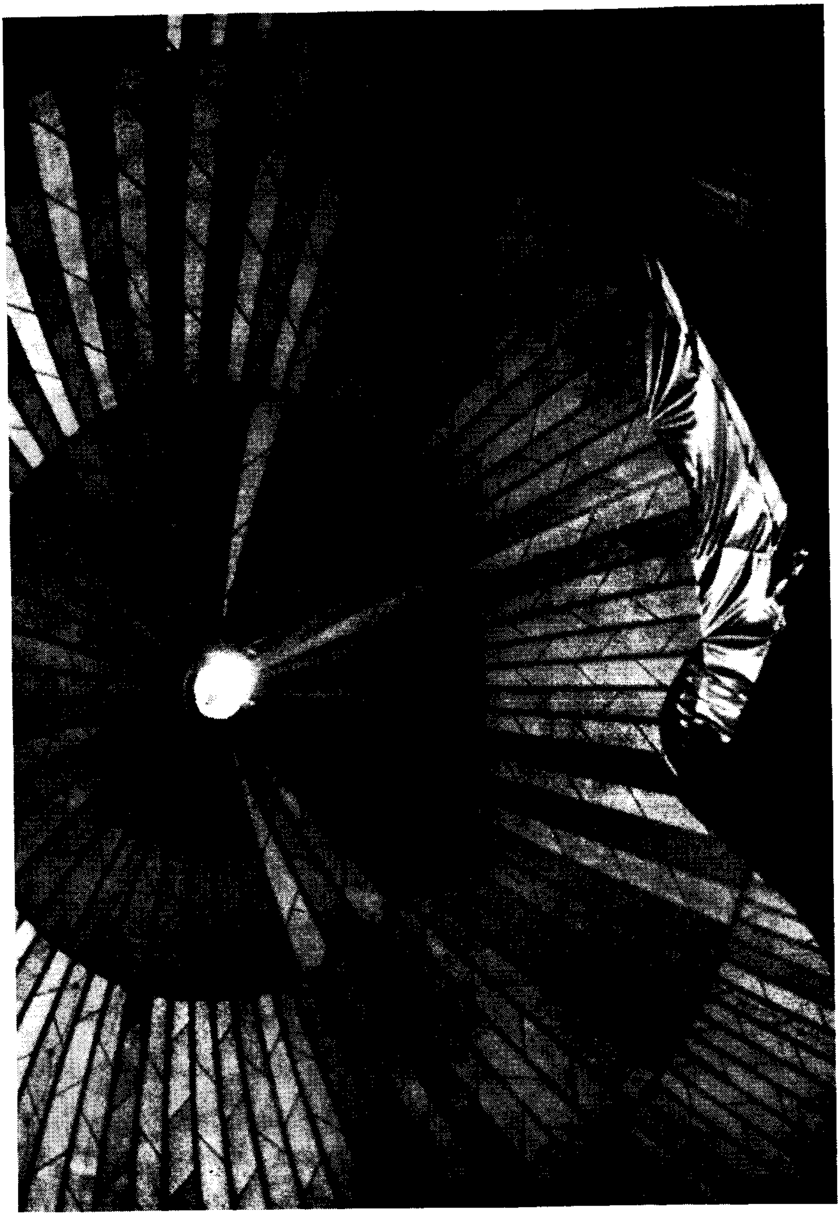


Fig. 26 — Upward view showing herringbone pattern inside Andover radome.

plies were bias-laminated to form a herringbone pattern (see Fig. 26). This laminate was then bonded, piece by piece, to make the full shaped radome shell.

The coating material selected was a synthetic rubber (based on Hypalon, a DuPont proprietary material) specially formulated to minimize RF loss and moisture permeability. This coating, together with commercially available Hypalon air-curing cements, was capable of developing the necessary inter-laminar and joint-bonded strengths, in addition to meeting all the requirements stated above.

To assure quality and uniformity in the finished product, checking was required throughout the entire process — i.e., checks of the yarn, the woven material, the coating material, the coating process, the thickness and quality control of the coated material, the ply laminating process, and the joints. All had to be continuously checked and double checked.

Quality control was a major undertaking because the inflated structure is composed of 6 acres of single-ply material laminated to make 3 acres of 2-ply patterned panels and contains 31,600 feet of hand-cemented joints which bond the patterned panels one to another to produce the final single-piece radome.

5.5 *Radome Fabrication*

The double-ply material, coated and laminated in the materials processing plant in Beacon, New York, was shipped to the radome fabricating plant of Birdair Structures, Inc., Buffalo, New York. The coated fabric was processed so that a single batch operation produced a continuous roll of a length in excess of that required to fabricate two of the 168 major gores, each 154 feet long. The length which was selected permitted sampling of material for quality-control purposes from each end and the middle of the roll. The samples thus represented the full range of curing-heat history. Quality control included warp-and-fill strength tests, coating-adhesion tests, and tests for elongation under load.

When the material was received at the radome fabricating plant, test samples were cut from the prescribed locations on the roll and forwarded to the receiving inspection laboratory for testing to confirm the prior tests and to check the weight, gauge, tear resistance, ply adhesion, and water permeability. Lap-joint specimens were also made and subjected to dead loads at 160 degrees Fahrenheit to determine the hot-ply adhesion and cementability of the coatings.

Material passing all tests, including 100 per cent visual inspection, was then ready for the next step, the cutting of gores according to templates. Patterning the gores took place on a flat table. The templates, therefore, included compensation factors to adjust the shape of the gore for transition from a plane surface to a spherical sector. Other factors took into account the differential elongation between the straight and alternating bias plies and that between the warp and fill of each ply under load. All of these compensations were made to insure the sphericity, under the required inflation pressure, which must be attained in the radome to realize adequate stress distribution and utilization of fabric strength.

The gore edges were next processed for stress relief at lap-joint terminations by subjecting them to a proprietary slitting process.

A sufficient number of gores was produced to cover the entire range of gauge and weight within the tolerance specified. To maintain uniformity of electrical characteristics, each gore produced was selectively assigned to various sections of the radome.

Bonding of gores commenced when an optimum number of gores was available for suitable dispersal in the radome and for consecutive cementing of adjacent panels sequenced in accordance with the folding scheme, which anticipated the eventual unfolding and deployment of the radome about the foundation at the erection site.

The edges to be bonded were buffed with a power buffer to expose a virgin surface of the Hypalon coating for manual application of two separate precoats of Hypalon cement. Mating edges of the precoated gores were then laid up on a form especially designed for holding and aligning the adjacent panel edges for application of bonding cement and curing in the desired panel-to-panel relationship to produce sphericity in the finished radome under inflation pressure.

The bonding cement was applied manually and the joints were rolled with a power-roller, the weight and speed of which were designed specifically for the Telstar radome. Joints remained in tension on the form for 3 to 4 hours for final room-temperature cure. Each long meridianal joint required 3 moves or positions on the form to complete bonding of the entire length.

Each move on and off the form had to be made without jeopardizing the anticipated sequence of folding the finished radome for placement in the 8 × 10 × 40-foot crate for delivery. This latter point was important since the complete radome weighs 60,000 pounds and is difficult to move when bonded into one piece. The upper edge of the main panels was bonded to the crown, the lower edge to the roped clamping bead. The crown, like the main gores, was made up of numerous bonded pieces.

Sample joints were made daily with each batch of cement, using production materials, and processed by production cleaning and curing methods. Samples were identified with each joint made and with each day's production. After cure, the samples were subjected to strength and peel tests for assurance that no unknown factors had inadvertently affected the uniformly high quality of the unit. To meet acceptance requirements, the strength of the joint had to exceed that of the material.

After final cure of each cemented joint, every inch of each side was visually inspected and "prodded" with a peel tester to ascertain that it was properly and securely bonded. The minutest area exhibiting the slightest indication of being questionable for any reason whatsoever was immediately reprocessed.

A precoated tape was cemented over the outer seam of each finished lap joint. The tape adds nothing to the joint strength, but it protects the exposed edge from invasion of the elements and improves the appearance.

A final inspection of all seams was made again while the radome was being folded into the shipping crate.

5.6 *Erection*

Program schedules did not allow sufficient time for the material development and manufacture of the radome prior to assembly of the antenna structure. To provide environmental protection during the antenna construction period and while development and manufacture of the radome material proceeded, a temporary air-supported structure (construction shelter), also 210 feet in diameter, was designed and fabricated. This structure could be provided relatively quickly and inexpensively because the wind requirements were lower than for the radome, and electrical specifications and long life span were not required. This intermediate structure provided protection for antenna construction and also was used in erecting the final radome.

Use of the construction shelter as an erection aid necessitated the addition of a bulge or blister to the fabric envelope, as seen in Fig. 27. This blister provided clearance for the upper equipment room when the construction shelter was lowered to the position required for radome installation, shown in Fig. 28. Erection of the one-piece, 60,000-pound inflatable radome over the pre-erected 95-foot high antenna and con-

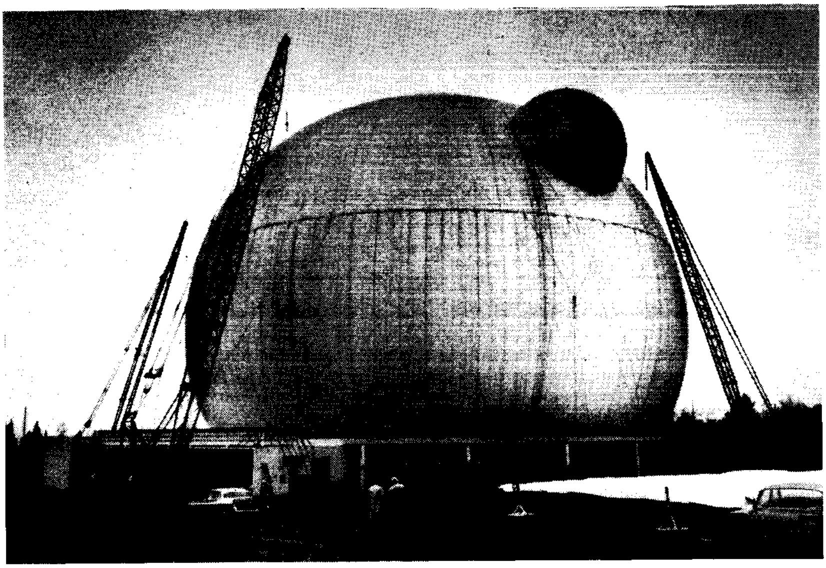


Fig. 27 — Preparing the construction shelter for deflation; permanent radome being unfolded at lower right.

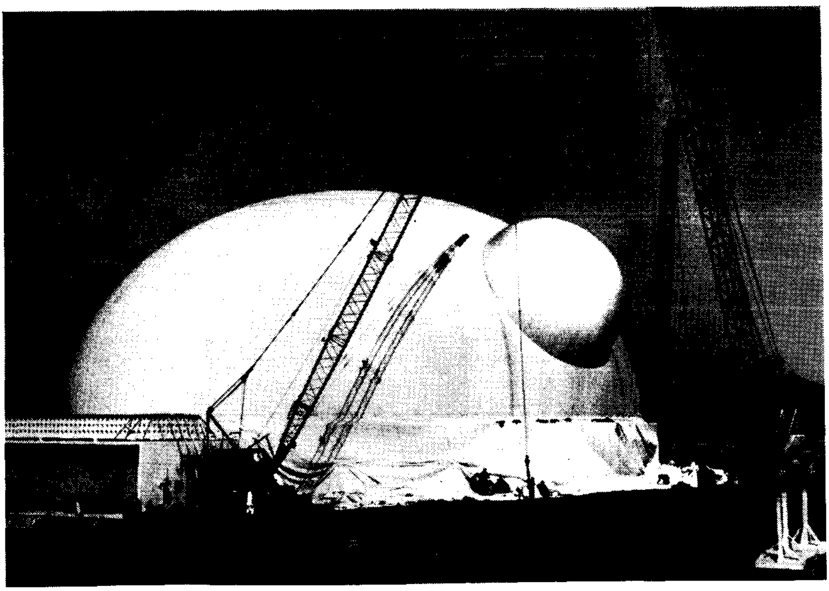


Fig. 28 — Construction shelter lowered and reattached to foundation ring; 160° of the radome is attached to the clamping ring.

struction shelter was a herculean task. Many schemes were considered; some of them included the use of towers, or helicopters or helium bags. Each method considered appeared to be hazardous and expensive or, in many cases, impractical. Strangely enough, the final method utilized air pressure as the main motive force to place the radome over the antenna and construction shelter. The method used is shown in Figs. 27 through 30.

Fig. 27 shows the air-supported construction shelter. The completed antenna has been placed in the stow position inside the construction shelter. In Fig. 28 the temporary construction shelter has been lowered, reattached to its foundation, and reinflated. The radome has been attached to the foundation around 160 degrees of its periphery and has been folded in preparation for lifting the leading edge. In Fig. 29 the leading edge has been lifted over the "blister," and tension is exerted on the leading edge bead by winches at the foundation. Fig. 30 shows

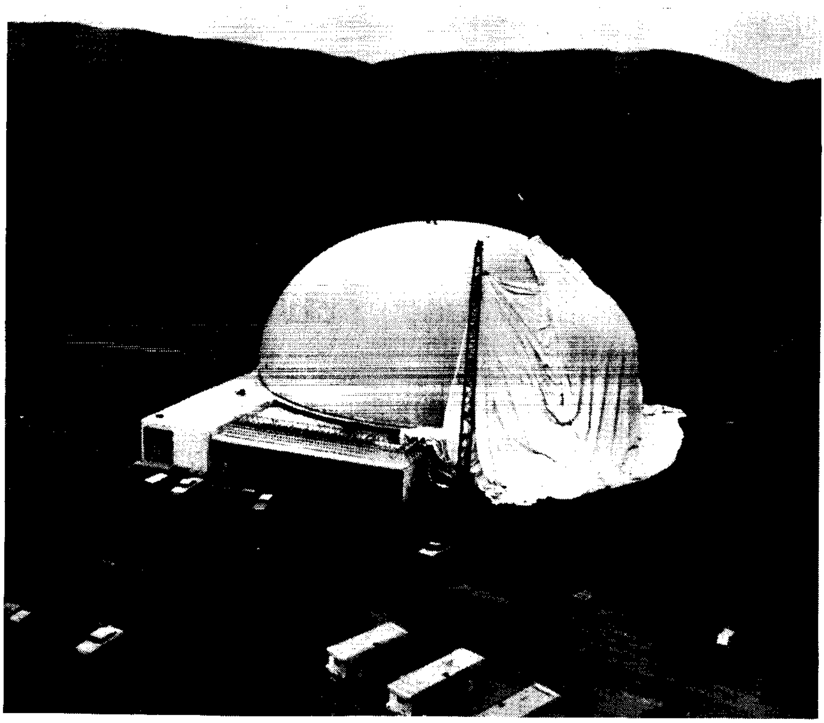


Fig. 29 — Initial raising of the radome over the blister of the construction shelter.

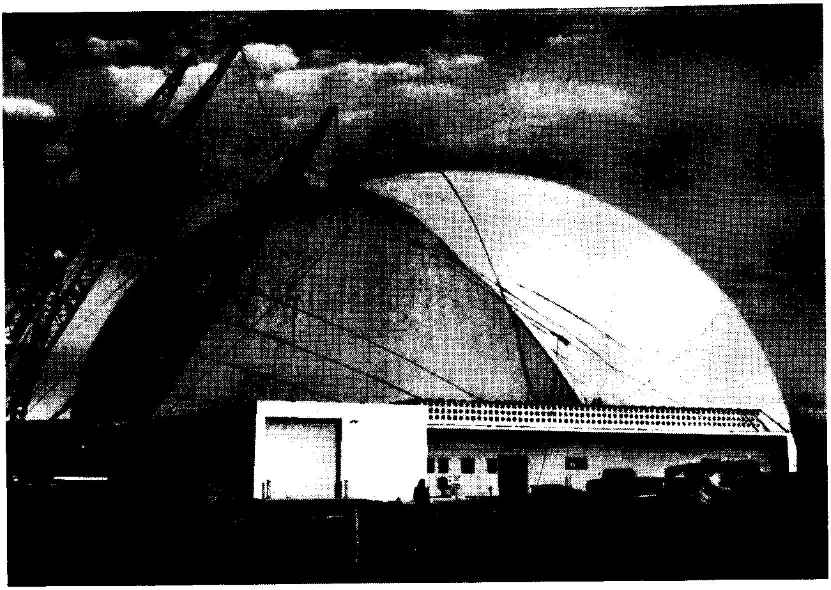


Fig. 30 — Radome being pressurized to allow sliding it over the construction shelter.

crane lines attached to the leading edge of the radome and the radome being pressurized. The pressure reaction on the upper free portion of the radome delivers a net force to the radome toward the cranes and slides the leading edge up and over the inflated construction shelter to engulf it. The radome bead is attached completely and the radome is inflated; the shelter is then removed from within. Fig. 31 shows the final installation with the radome fully inflated.

5.7 *Pressurization System*

The pressurization system is completely automatic and functions always to keep the radome internally pressurized to a level in excess of the dynamic pressure of the wind.

The system is designed to maintain any one of three pressure levels: $1\frac{1}{2}$ inches of water (0.054 psi) by use of two low-pressure blowers for winds from 0 to 45 miles per hour, 3 inches of water (0.108 psi) by use of one intermediate-pressure blower for winds from 45 to 70 miles per hour, and $5\frac{1}{2}$ inches of water (0.20 psi) by use of two high-pressure blowers for winds from 70 to 100 miles per hour. All blowers operate as

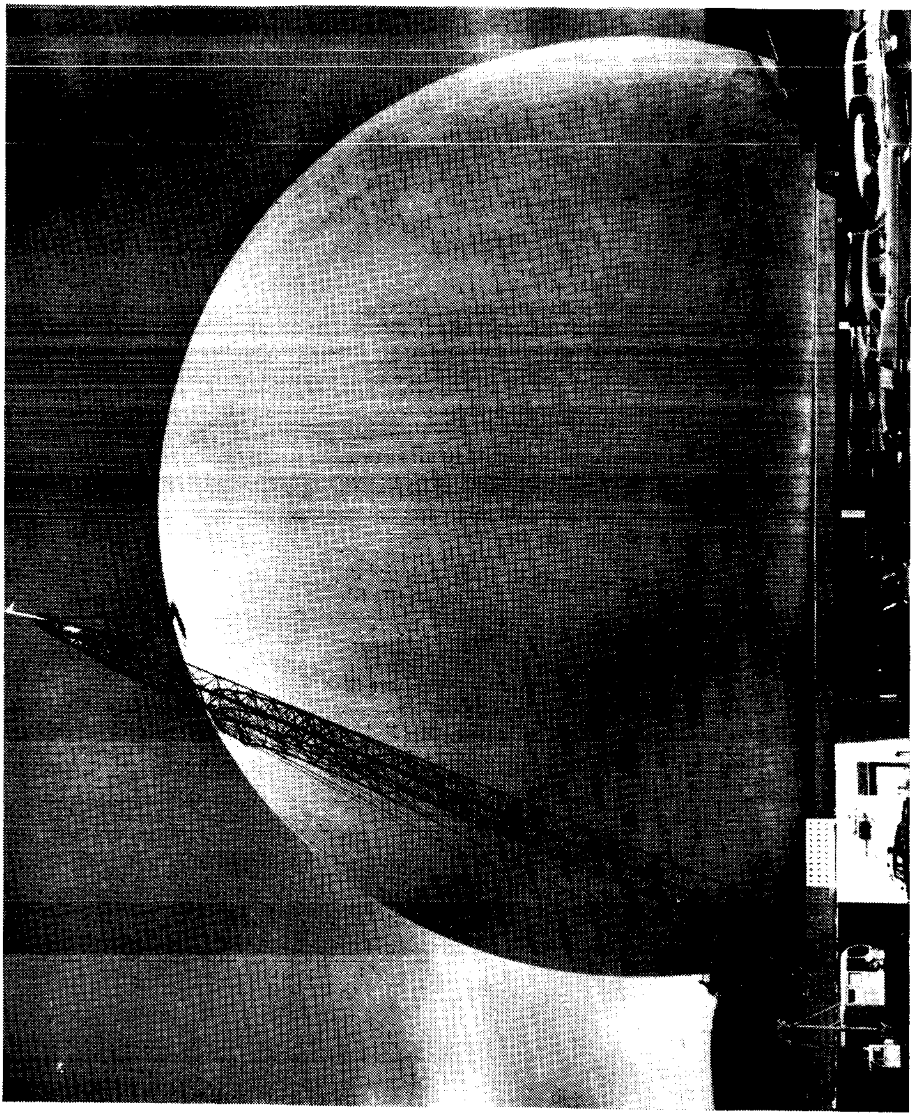


Fig. 31 — Radome installation complete and fully inflated.

a function of the output of an external anemometer and the internal radome pressure. Should a blower fail for any reason, the system is designed so that the next pressure-stage blowers will automatically come on to maintain pressure. For emergency situations, manual remotely controlled vents are provided. Also, high-pressure valves are installed, which will prevent the pressure from ever exceeding the maximum design pressure. All blowers, with the exception of the low-pressure blower, will function automatically from an emergency generator provided at the site.

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