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Report No. 10075

DEVELOPMENT OF ANTENNA GROUP OA-1227(XE-1)/TPS

Final Technical Report
25 June 1957 to 1 May 1961

Contract DA-36-039 SC-74866
Dept. of the Army Project
3D20-01-001

U.S. Army Signal Research and
Development Laboratory
Fort Monmouth, New Jersey

HAZELTINE ELECTRONICS DIVISION

HAZELTINE CORPORATION

Little Neck, New York

AD-____________________ Div. 8

Hazeltine Corp., Little Neck, N.Y. DEVELOPMENT OF ANTENNA GROUP OA-1227(XE-1)/TPS by Gerold W. Bugen and Paul L. Dominique. Final Technical Report, June 1957 to May 1961, 151 p incl. illus. Contract DA-36-039 SC-74866 Unclassified Report.

This report describes the development of the antenna group over an approximate 3-1/2 year period. The over-all program was broken into two tasks. Task A encompassing the antenna and Task B the required modification to the shelter. The separate tasks are further subdivided into phases of work. Problems and solutions of a nature peculiar to this program, are discussed in detail. The final design for the antenna essentially conformed to Signal Corps Technical Requirements SCL-5296 as amended.

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HAZELTINE CORPORATION

Report No. 10075

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Final Technical Report
25 June 1957 to 1 May 1961

Contract DA-36-039 SC-74866
Signal Corps Technical Requirements SCL-5296
Dept. of the Army Project 3D20-01-001

United States Signal Supply Agency
Laboratory Procurement Support Office
Fort Monmouth, New Jersey

Prepared by

Gerold W. Bugen

and

Paul L. Dominique

Approved: V. J. Young

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ANTENNA GROUP
OA-1227(XE-1)/TPS

SECTION 1. PURPOSE

The original purpose of Contract DA-36-039 SC-74866 was the development of a high power, L-band antenna to provide increased range capability and high angle coverage for Radio Set AN/TPS-1D. An IFF channel and provisions for operation with high power radars (up to two megawatts) were to be included.

The equipment to be transportable in two transit cases, each designed to fit on an M-35 truck or equivalent. The transit cases to provide stowage for the antenna reflector sections, feedhorn assembly, antenna pedestal, cables, tools and mounting components.

The equipment to be designed in accordance with Signal Corps Technical Requirements SCL-5296. Service conditions, in accordance with the specification, to include operation in high winds, under severe icing conditions and in extremely low temperatures.

Primary power for the antenna drive to be furnished by an external, diesel driven engine generator (GFE) delivering three-phase 60-cycle ac power. Synchro reference voltages (115-volt, 60-cycle and 115-volt, 400-cycle) to be obtained from the associated radar set.

Initially the contract and specification only covered development of Antenna Group OA-1227(XE-1)/TPS. The contract and specification were subsequently amended to provide coverage of antenna operation with Radar Surveillance Shelter AN/GSS-1.

Development of Antenna Group OA-1227(XE-1)/TPS is covered in task A of this report and the necessary modifications to the shelter in task B. The two tasks were further subdivided into the following phases and parts, details of which will be found in Section 4, Factual Data.

TASK A - ANTENNA GROUP

PHASE 1.	RF SYSTEM
Part a.	Reflector
Part b.	Feedhorn
Part c.	Rotary Joint
Part d.	Coax-to-Waveguide Transition
Part e.	RF Cabling

TASK A - ANTENNA GROUP (Cont'd)

PHASE 2.	CIRCUITRY
Part a.	Contactor Box
Part b.	Antenna Control Box
Part c.	Antenna Drive Motor
Part d.	Synchro Assembly
Part e.	Lubrication
Part f.	Pressurization
Part g.	Cabling
PHASE 3.	TESTING
PHASE 4.	MECHANICAL
Part a.	Antenna Pedestal
Part b.	Transit Cases
Part c.	Installation Accessories
Part d.	Miscellaneous

TASK B - SHELTER MODIFICATION

PHASE 1.	CIRCUITRY
Part a.	Interconnection Box
Part b.	Cabling
PHASE 2.	MECHANICAL

SECTION 2. ABSTRACT

The development of Antenna Group OA-1227(XE-1)/TPS (task A) consumed a total time of approximately three years. Development of the shelter modification (task B) was much simpler and added approximately six months, making a total development time of approximately 3-1/2 years.

The first two test models are alike and employ fuses for protection in the three-phase, 60-cycle power lines. To improve reliability and provide simultaneous opening of the three power lines, the subsequent five test models (serial numbers 3 through 7) are provided with a circuit breaker in place of the above fuses. In addition, minor control circuit modifications were effected on the last five equipments delivered.

The antenna reflector contour and size, is based on an existing fixed station antenna (Antenna OA-1124/FPS). The sectionalization of the reflector and design of support and feedhorn assemblies (Task A, Phase 1, Parts a. and b.) were subcontracted to the Donald S. Kennedy Co., Cohasset, Massachusetts.

Design of transit cases was also subcontracted to the Donald S. Kennedy Co., since the bulk of the items to be stowed (i.e., reflector sections, feedhorn assembly and reflector and feedhorn support assemblies) were being designed by them.

The antenna pedestal was designed by Hazeltine for transportability and to comply with the other specification requirements.

The balance of the components, with the exception of the coax-to-waveguide transition (subcontracted to Diamond Microwave Corp.), were designed by Hazeltine and its subsidiary, Suffolk Products Corp.

Problems encountered in the design of the reflector, feedhorn, rotary joint, coax-to-waveguide transition and rf cabling are discussed in Phase 1 of Task A. Control circuitry problems, for the antenna group, and their solutions are discussed under Phase 2 of Task A, and shelter modification circuitry under Phase 1 of Task B.

Problems which were strictly of a mechanical nature, such as transit cases and soil anchors, are discussed in detail in Phase 4 of Task A and Phase 2 of Task B.

SECTION 3. PUBLICATIONS, LECTURES, REPORTS, CONFERENCES

PUBLICATIONS.

(1) Contract DA-36-039 SC-74866.

(2) Signal Corps Technical Requirements SCL-5296.

(3) Design Plan for Antenna Group OA-1227(XE-1)/TPS, Hazeltine Report no. 5227.

(4) Operating and Performance Tests on Antenna Group OA-1227(XE-1)/TPS, Hazeltine Report no. 5122.

(5) Technical Manual for Antenna Group OA-1227(XE-1)/TPS.

(6) Supplement to Technical Manual for Antenna Group OA-1227(XE-1)/TPS.

(7) Revised Coverage for Antenna OA-1124/FPS( ) dated 19 September 1956.

(8) Production Tests for Antenna Group OA-1227( )/TPS, Hazeltine Electronics Division Test Procedure 654-A.

(9) Technical Proposal for Design and Construction of Antenna Group OA-1227( )/TPS, dated 4 January 1957, Report no. 5207.

CONFERENCES.

(1) Evans Signal Laboratory, Belmar, New Jersey, 3 October 1957. Attended by Signal Corps and Hazeltine.

Resume

a. Clarification of items in specification.

b. Discussion of design philosophy.

(2) Hazeltine Electronics Division, Little Neck, New York, 10 October 1957. Attended by Signal Corps and Hazeltine.

Resume

a. Interpretation of specification on method of input to rotary joint.

b. Type of lubricating oil to be used in the pedestal.

(3) D. S. Kennedy & Co., Cohasset, Massachusetts, 23 October 1957. Attended by Signal Corps, Hazeltine and Kennedy.

Resumé

a. Construction and size of transit cases.

b. Design of feedhorn and support.

c. Erection of the reflector assembly.

d. Calculation of torque requirements.

(4) Hazeltine Electronics Division, Little Neck, New York, 21 November 1957. Attended by Signal Corps and Hazeltine.

Resumé

a. Method of unloading transit cases from trucks.

b. Erection assembly sequence.

c. Availability of rf cable.

(5) Evans Signal Laboratory, Belmar, New Jersey, 9 January 1958. Attended by Signal Corps and Hazeltine.

Resumé

a. Discussion of Design Plan for Antenna Group OA-1227(XE-1)/TPS, Hazeltine Report no. 5122.

b. Amendment of specification specifying 1/2 megawatt transmission line to pedestal.

c. Signal Corps intent to change specification to specify operation with shelter AN/GSS-1 instead of AN/TPS-1D ground radar.

(6) Evans Signal Laboratory, Belmar, New Jersey, 22 January 1957. Attended by Signal Corps and Hazeltine.

Resumé

a. Presentation of list of exceptions to the specification, by Hazeltine, due to additional items required by the Signal Corps, prior to approval of the Design Plan.

b. Further discussion concerning use of Antenna Group OA-1227(XE-1)/TPS and shelter AN/GSS-1.

(7) Hazeltine Electronics Division, Little Neck, New York, 27 February 1958. Attended by Signal Corps and Hazeltine.

Resumé

a. Discussion of Design Plan Approval.

b. Limitation on use of M-35 trucks in transporting and unloading of transit cases.

(8) Hexagon Building, Fort Monmouth, New Jersey, 13 March 1958. Attended by Signal Corps and Hazeltine.

Resumé

Use of truck M-35 for transporting transit cases.

a. Possible drop test modification due to provisions for top lift only.

b. Use of special jacks under front bumper during unloading to limit strain on front axel.

c. Attachment of "A" frame to front bumper.

(9) Evans Signal Labs, Belmar, New Jersey, 26 March 1958. Attended by Signal Corps and Hazeltine.

Resumé

Requirements for modification of shelter AN/GSS-1 for use with Antenna Group OA-1227(XE-1)/TPS.

(10) D. S. Kennedy Co., Cohasset, Massachusetts, 22 April 1958. Attended by Signal Corps, Hazeltine and Kennedy.

Resumé

a. Location of pedestal in transit cases.

b. Discussion on various tests to be performed on transit cases.

c. Method of raising reflector into position.

(11) Evans Signal Laboratory, Belmar, New Jersey, 29 April 1958. Attended by Signal Corps and Hazeltine.

Resumé

a. Discussion on Spare parts for Antenna Group OA-1227(XE-1)/TPS.

b. Discussion on various equipment tests to be performed.

c. Specifications for possible lubricating oils were given to Hazeltine by Signal Corps.

(12) Signal Corps, Fort Monmouth, New Jersey, 11 and 23 June 1958. Attended by Signal Corps and Hazeltine.

Resumé

Request for contract changes on: anchoring methods; vibration, drop and bounce tests; lubricating oil and modifications for M-35 trucks.

(13) Fort Bliss, Texas, 30 March 1959 through 16 April 1959. Attended by Signal Corps and Hazeltine.

Resumé

Field Evaluations of Antenna Group OA-1227(XE-1)/TPS and shelter AN/GSS-1 modification kit.

(14) Evans Signal Laboratory, Belmar, New Jersey, 28 April 1959. Attended by Signal Corps and Hazeltine.

Resumé

a. RF cable difficulties

b. Delivery schedules

(15) Fort Bliss, Texas, 15 June 1959 through 18 June 1959. Attended by Signal Corps and Hazeltine.

Resumé

a. New rf cable (RG-19A/U) assembly evaluation.

b. Observation of disassembly, erection of Antenna Group OA-1227(XE-1)/TPS and operational tests.

c. Discussion of mechanical difficulties encountered during evaluation tests.

d. Target Acquisition data and blip-scan ratio recordings.

(16) Evans Signal Laboratory, Belmar, New Jersey, 7 July 1959. Attended by Signal Corps and Hazeltine.

Resumé

Discussion of field evaluation tests at Fort Bliss, Texas.

(17) Fort Monmouth Procurement Office, Fort Monmouth, New Jersey, 22 September 1959. Attended by Signal Corps and Hazeltine.

Resumé

Discussion of design and contractual problems.

a. Compatability with Radar Set AN/UPS-1.

b. Packaging of running spares, and shelter AN/GSS-1 modification kit.

c. Modification of Antenna Group OA-1227(XE-1)/TPS in accordance with findings at Fort Bliss, Texas.

d. Addendum to instruction books to reflect modifications.

(18) Fort Monmouth, Fort Monmouth, New Jersey, 27 September 1960. Attended by Signal Corps and Hazeltine.

Resumé

Review of corrosion on Antenna Group OA-1227(XE-1)/TPS.

SECTION 4. FACTUAL DATA

INTRODUCTION

As outlined in Section 1, Purpose, this report divides the development of Antenna Group OA-1227(XE-1)/TPS into two tasks; A - Antenna Group, and B - Modification of Shelter. Task A is subdivided into five phases, with phases one and three being further divided into two parts each and phase two into seven parts. Task B is subdivided into two phases, with phase one being further divided into four parts.

The following factual data is presented in the same order as outlined in Section 1. The overall schematic diagrams (figures 28 and 29) are supplied for reference to show the electrical ties between the pedestal and shelter AN/GSS-1. Figure 1 shows Antenna Group OA-1227(XE-1)/TPS erected and ready for service.

TASK A. ANTENNA GROUP
(See figure 1)

Antenna Group OA-1227(XE-1)/TPS is a high gain transportable antenna system designed for use with Radar Set AN/TPS-1D as installed in Radar Surveillance Center AN/GSS-1.

PHASE 1. RF SYSTEM
(See figure 2)

The rf system, for the antenna group, transmits and receives both search radar and IFF signals. The radar pulses, at 1250 to 1350 megacycles, from the AN/TPS-1D receiver-transmitter units (RT-212/TPS-1D) are fed through coaxial cable, to the coax to waveguide transition. Then, through the rotary joint to the feed-horn.

The IFF pulses, at 1020 ± 18 megacycles or 1100 ± 18 megacycles are fed from Receiver-Transmitter RT-211/TPX through the shelter interconnecting box and, via coaxial cable, to a coaxial connection on the bottom of the rotary joint. From the top of the rotary joint, another coaxial cable feeds the IFF signals to the feedhorn coaxial connection.

Difficulties were encountered in meeting the specification requirements for the voltage standing wave ratio (VSWR), of 1.3 to 1, at the input to the rotary joint, and 1.5 to 1 from the output of Radio Set AN/TPS-1D. The difficulties were overcome by holding the coax-to-waveguide transition, rotary joint and feedhorn to very strict tolerances. The VSWR measurements of the first test model are provided in addendum A of this report.

Part a. REFLECTOR

1. General
(See figure 3)

The reflector is approximately 40 feet wide and 11 feet high, and produces a search radar horizontally polarized beam, approximately 1.7 degrees wide and 6.0 degrees high. The reflector has a parabolic trough-type configuration and produces a cosecant squared pattern. The IFF beam produced is vertically polarized and approximately 1.8 degrees wide and 7.5 degrees high. Antenna coverage tests are given in addendum B of this report.

The reflector is constructed in eight sections, designed for easy assembly and disassembly (see figure 4).

2. Design

The contour and size of the reflector is based on fixed station antenna reflector OA-1124/FPS( ). The primary design consideration was suitable sectioning of the above reflector and supports, so it could be quickly erected, and easily stowed in a small volume transit case.

In order to satisfy the requirements of a quick disconnect fastener, that would support the large forces involved, it was necessary to design a new fastener. A cast fastener of 356-T6 aluminum alloy was designed to support a static load of 20,000 lbs. with no appreciable motion. The final configuration arrived at (after tests at Northeastern University, Boston, Massachusetts) is a tapered connection, with a 20° included angle on the male and female half. The line of force was designed to pass directly through the center of the contact area. The angle was chosen, as it closely approximated spherical surface at the diameter used and had sufficient friction to prevent slippage. The final design held together under 20,000 lbs. of tension or compression. This was emphasized during the static load test when it was discovered that two connections were not secured in the reflector support, at points of maximum stress, after the reflector was fully loaded. (The deflections recorded during static load test showed that the reflector and support could withstand about 50% greater wind loadings without permanent set.)

The next problem on the reflector was how to provide a structural dissection that would allow stowing all sections in one transit case. As 14 or even 10 sections presented too great a tooling and attachment problem, eight sections were decided upon. Because of the width available, a thickness of eight inches was the maximum depth possible for each section. This depth would not support the load (imposed by the wind and ice loading) adequately, so an additional bracing means was devised. A main horizontal truss of four sections was provided (of the proper section), and braces were attached by means of rod-end bearings to provide the necessary pivot arrangement. Four braces were provided for each section. The braces stowed onto the main truss by means of brackets and Air-Lock 1/4 turn fastener. During assembly of the sections the cross pin in the Kennedy designed quick fastener repeatedly came loose. The Spirol pins used were found unsatisfactory and were replaced by solid pins (Groove).

Tooling presented many problems. In order that the reflector tolerance could be held after assembly, it was necessary to make elaborate welding and machining jigs. All tubing was precut to finished size prior to welding. The major part of the panel back-up structure was welded. The welded section was transferred to machining jigs, and all connections were machined. The section was then transferred to a screening jig and the 5/8 squarex screen attached. The screen is attached to the panel back up structure by tack welding approximately 7 inches on center to all face tubes, and secured to the tube at the perimeter by means of a cap strip, cherry riveted in place over the raw edge. The section was then transferred to the mold along with adjacent sections, fastened together and final adjustment made to fit the surface of the section to the mold surface. After fitting, the remaining welding was done on the mold. This welding consisted of a seam on telescoping members that were clamped in a predetermined stressed position; so that, when relaxed, the proper surface contour was maintained. The section was then removed and the screen tightened, to eliminate the excessive bag between face tubes.

The horizontal truss and braces were assembled and welded on one jig, and machined on another jig.

The assembled reflector was checked for proper alignment by means of cross wires.

The reflector support had to be designed at a great structural disadvantage, due to the desired location of the reflector to the pedestal. As it was much too large to be stowed in one piece it was broken down into five parts, left and right supports, rear support, front support and center support. Due to large stresses, the majority of tubes were 3 inch 1/4 wall or 3 inch 3/8 wall 6061-T6 aluminum alloy. Where castings of 356-T6 aluminum alloy were used, the minimum wall thickness had to be made double the tube wall, due to axial and bending loads. A complete stress analysis and several models were made to satisfy the requirements of the loads encountered in each member. As the quick disconnect

Kennedy fastener did not lend itself to the joints of the support, high strength bolts, and tapered bosses, were used to transmit the biggest loads. Pivot points with Heim bearings were provided at the rear of the left and right support to facilitate erection. Welding and machining jigs were used to fabricate all sections.

Two inner and two outer legs (actually trusses) were used to connect the reflector to the reflector support. They were simply constructed in the form of an "N" and attached by means of the Kennedy quick connect fastener. A welding and machining jig was used to fabricate both.

Because of the heavy tube wall thicknesses in the supports and odd joints of tube to tube connections, many special welding techniques were required to limit distortion and provide sound welds.

The Fort Bliss, Texas evaluation tests revealed minor mechanical troubles as follows:

(1) The large one inch structural bolts used for assembling the reflector supports became clogged during the tests and tended to bind and cross-thread.

A cleaning groove has been milled across the threads of the structural bolts, this allows any dirt on the bolt, or in the mating threaded hole, to roll into the groove; thus avoiding jamming of the threads.

(2) Identification tags were decals secured with glue and fell off during the tests.

The decal tags were changed to metal and are now securely fastened with drive screws.

Inspection of the reflector and support assemblies after the evaluation tests revealed other minor problems, which were corrected as follows:

(1) The clevis arms, connecting with the auxiliary "A" frame were extended to prevent the frame from rubbing against the lower center reflector panels during erection.

(2) A slot was milled into the clevis to prevent damage to the "A" frame, if hooked up incorrectly during erection.

(3) Two set screws were added to hold the self aligning bearings in their mountings.

(4) Pins in the Kennedy quick disconnect fasteners, used on the reflector, were changed from groove to straight pins brazed in place.

part b. FEEDHORN

1. General
(See figure 5)

The feedhorn (antenna feed) is a dual polarized horn. Inputs from the search radar and IFF radar are both sections of R6-69/U waveguide, with the search radar input rotated 90° (with respect to the IFF radar input) to insure a high degree of isolation. The energy, from either channel, enters the flared section of the feedhorn, propagates to the aperture and from there illuminates the reflector. The search radar energy is horizontally polarized and the IFF energy vertically polarized.

The dual polarized horn has several advantages over the normal dipole feed, placed in front of a single polarized feedhorn. (1) Either input is capable of handling a power level equal to the maximum waveguide power. (2) The radiated patterns are controlled by the aperture dimensions only, and does not affect the other radar feed, as do dipoles and reflecting flaps placed in front of the feedhorn mouth. (3) The dual polarized horn is simpler in construction, than the combined feeds of the horn and dipoles.

The feedhorn and associated waveguide are assembled together to facilitate stowing and antenna erection. When erected, the feedhorn is situated at the focal point of the reflector, which concentrates the energy into a narrow beam (see antenna patterns in figure 6).

2. Design

The feedhorn presented many problems. The primary problem was to develop a horn that would give the electrical characteristics that were required by specification. Development from start of design, to a final design, took place over a period of one year and four months, approximately. About 1800 hours of work were spent in development and testing prior to shipment of first unit.

A Tchebyscheff feed was originally proposed but did not appear suitable early in the design. The major part of the effort was spent in development of a flared feedhorn with Radar and IFF connections. For ease of fabrication a sheet metal horn was selected. The major drawback to this type of construction was holding the tolerances, which were found necessary after many tests, to ±0.010 inches of the desired configuration. A cast horn could have been used but, by the time a final design was determined, there was insufficient time to get a suitable casting with the desired tolerances. All parts of the horn were machined to exact size (±0.003 inches) before welding. The edge joints were machined in a manner so that no inside welding would be needed, and thereby require no internal hand finishing. A special holding fixture was necessary to maintain the desired tolerance. Many trials were necessary to obtain the desired matching. Because of the inherent critical design of the horn each one must be matched and checked in free space and finally checked with a reflector on an antenna range.

The feedhorn support had to place the horn accurately in place without adjustment. The non-adjustable feature placed the feed system at an extreme disadvantage. Most critical was the vertical angle of inclination to the reflector. Also critical, was the focus which was finally determined to be 145.6 inches from the reflector surface point of origin. In order to keep a maximum area free of interference from feed supports, and to simplify the feed support, the waveguide was used as a structural member of the feed support system. In order to gain maximum rigidity and positive placement of the horn, three "A" frames were attached to the horn, and the two lower attached to one point at the bend of the waveguide run. Another problem was the critical length of the horizontal run of waveguide. The waveguide flange had to terminate within 0.06 inches of the pedestal waveguide connection. A very slight error in the bend would cause a large misalignment. The waveguide run from the horn to the bend used a cast twist, available from a previous Kennedy job and simplified that transition considerably.

Many electrical tests were necessary to determine the correct placement of the horn, as the theoretical placement did not give the best location for all pattern requirements.

Meeting the original IFF vertical beamwidth specification requirement of 6.2 ± 0.5 degrees presented a major problem. The solution was a complicated feedhorn employing 10 beam broadening probes and four impedance matching irises. This horn produced the high edge illumination required for a vertical beamwidth of 6.2°, but also produced a ripple of at least 3db in the secondary vertical pattern. Further modifications produced the feedhorn of final design, which, while not meeting the original requirement of 6.2 degrees, has the following advantages:

(1) Increased high angle coverage. - The upper one-half power angle will be 6.3 degrees above the horizon, 0.7 degree increase in vertical coverage.

(2) Increased gain. - An IFF gain increase of at least 1.2db was realized.

(3) Lower VSWR. - The IFF VSWR was much improved, enabling meeting the specification requirements in this respect.

(4) Smoother secondary vertical pattern. - Vertical patterns show secondary ripples of 1db.

(5) Mechanically simpler. - The new horn is of much simpler mechanical construction, therefore easier to produce.

Pointing out the above advantages to the Signal Corps, resulted in a specification requirement change in the IFF vertical beamwidth to 7.8 ± 0.5 degrees, and enabled use of the improved feedhorn design.

Part c. ROTARY JOINT
(See figure 7)

1. General

The rotary joint for Antenna Group OA-1227(XE-1)/TPS is pressurized and transmits the radar and IFF signals from the coax-to-waveguide transition to the feedhorn.

The rotary joint has a waveguide flange input, enabling direct connection to waveguide, for coupling a high power radar (up to two megawatts).

The coaxial section of the rotary joint consists of a center conductor which rotates with respect to the lower waveguide section of the joint, and an outer conductor which is divided into an upper and lower section. The two outer conductor sections are connected by a choke joint, which permits the upper section to rotate. The lower section is fixed to the lower waveguide and the upper section to the upper waveguide sections of the rotary joint. The upper waveguide section consists of a doorknob-type transition which feeds the rf power from the coaxial section into the upper waveguide. The lower waveguide section consists of a similar transition. Matching irises, in each of the upper and lower waveguide sections, match the rotary joint to the waveguide sections and thus minimize reflections within the rotary joint.

The IFF energy is also coupled to the feedhorn through the rotary joint. A center coaxial conductor with slip rings (fingers) permitting the rotation. Coax connectors on the upper and lower transistions permit connecting the IFF cables.

In the event the antenna is used with a high power radar (1.5 to 2 megawatts) facilities for pressurizing the rotary joint are provided, with pressure windows located in each of the upper and lower waveguide sections. A pressure switch is also provided to indicate normal and subnormal pressures.

2. Design

The rotary joint is an original Hazeltine design. The door knob upper and lower transitions are basically a scaling of door knob transitions used in other rotary joints at higher frequencies. The location of the short circuit and elevation of the doorknob were varied to achieve an optimum transition. A matching iris was then introduced in the waveguide to complete the matching of the transition. Location of the several variables was made during low power VSWR tests. A more ruggedized unit was then fabricated and tested at higher power. No breakdown resulted in this transition during the test. Refer to addendum C, of this report, for high power test calculations of the rotary joint.

The IFF channel is also an adaptation of a previous rotary joint design. The design employs finger contacts for both the inner and outer conductors of the coaxial line. The major problem was a mismatch, resulting in a high VSWR. The HN connectors at both ends of the channel were found to be the cause of the mismatch. The match of the connectors was improved by modifying the dimensions of several of the insulators. The modified connectors lowered the IFF VSWR to within requirements.

Pressurizing the rotary joint presented by far the largest problems. The first problem was obtaining pressure windows, at the required two megawatt rating. The original windows were a commercial product, made by Airtron, Inc., constructed of a fiberglas material. When tested under pressure, with two megawatts peak power, the material buckled outward toward the lower pressure side, and subsequent inspection revealed burn marks at points of maximum E-field. A new pressure window was designed by Hazeltine using Ruby Muskavite (mica) material. The new window handled the pressure adequately and showed no evidence of breaking down when subjected to 2.15 megawatts rf power.

The main pressurization problem was trying to construct the rotary joint to meet the specification requirement of a maximum pressure loss of 0.4 psi in 24 hours when subjected to 30 psia. To meet this requirement, oversize "O" rings (see addendum D for design) were placed in the pressure window seals. A pressure loss of slightly over one psi per day still occurred. Further modifications could have been effected, as follows, to reduce the leakage.

(1) Increasing bellows strength and therefore the sealing pressure. However, there is a maximum limiting surface pressure at which seal wear life is sacrificed for a little less pressure. Present design enables the rotary seal bellows joint to function safely below this level.

(2) The fine finish on the sealing surfaces, could conceivably be made finer, but a harder closer grained material would be required. A compromise was effected here by using a high grade graphite ring, with superior qualities of long wear and natural lubrication.

Rather than impair the effectiveness of the rotary joint by requiring design changes in critical areas, Hazeltine discussed with the Signal Corps the selection of a pressure tank with longer service life, and a substantial factor of safety for pressure loss The air tank of final design has a capacity of 386 cubic inches and a service pressure of 2100 psig. At a pressure loss of over 1.6 psi per day the tank provides over 15,000 hours (almost 2 years) of steady life without refilling the tank, thus far exceeding the

the 5,000 hour reliability specification requirement. On the basis of the above, the Signal Corps waived the specification requirement of a maximum pressure loss of 0.4 psi per day and permitted a loss of 1.35 to 1.675 psi in 24 hours. Refer to addendum E for calculations supporting the pressurization specification change.

part d. COAX-TO-WAVEGUIDE TRANSITION
(See figure 7.)

1. General

The coax-to-waveguide transition is required to couple Antenna Group OA-1227(XE-1)/TPS to the coaxial cable from the duplexer of Radio Set AN/TPS-1D. If the antenna group is used with higher powered radars (with waveguide output connections) this part will not be used.

Energy from the coaxial cable is coupled into the waveguide portion by a short radiator which is part of the coaxial connector.

2. Design

The design of the transition is similar to a commercial unit, which was not available at the 600 kw power, required by the specification. The design and production of the transition was subcontracted to the Diamond Microwave Corp.

During the electrical tests of the first two units the VSWR was found satisfactory, but arcing was heard with less than 500 kw of power. It was determined that the matching probe was not making adequate contact with the wall of the waveguide. Welding the probe to the waveguide was attempted, however, this distorted the waveguide and was discarded. A threaded probe in the waveguide (and then soldered) was tried, this provided the desired contact, but was very difficult to adjust. The final solution was a probe that was threaded into a special seat in the waveguide.

The electrical tests, on the transition of final design, proved the VSWR to be satisfactory, with a power handling capacity in excess of 600 kw.

Part e. RF CABLING

1. General

The rf cabling interconnects Radar Set AN/TPS-1D and the coax-to-waveguide transition. The original design plan was to use Amphenol cable 421-103 to provide the flexibility for storage on the cable reel. However, during evaluation testing the rf cable heated up and power to the antenna was greatly attenuated.

To overcome the heating problem, RG-19A/U cable was tried and found satisfactory. However, RG-19A/U cable has a solid core with a larger safe bending radius and other storage facilities had to be provided.

The original requirement had specified that cables were to be 50 feet long. Amending the contract to have the antenna operate with shelter AN/GSS-1 permitted shortening the cable. Storage for the shorter cable, with the necessarily large bend radius, was obtained by clamping to the top inside of the reflector transit case.

PHASE 2. CIRCUITRY
(See figures 28 and 29)

Circuitry for Antenna Group OA-1227(XE-1)/TPS encompasses the three phase a-c control circuits, antenna control system, synchro system, lubrication and pressure controls, antenna drive motor and cabling to Radar Surveillance Center AN/GSS-1.

Safety switch S108 and interlock switches S101, S802, and S803 are provided for personnel safety, removing power from the antenna drive motor when switch S108 or a compartment (i.e., contactor box S101 or stow lock access door S803) is opened or the manual crank S802) is in its operating position. Interlock switch S804 opens the antenna drive motor circuit when the antenna is locked in stow position, to prevent the motor from being accidentally energized and damaging the equipment. Interlock switches S803 and S101 may be cheated if visual inspection is required of the components while the equipment is in operation.

Part a. CONTACTOR BOX
(See figures 8 through 11)

1. General

The contactor box is mounted externally on the antenna pedestal; it operates and protects the antenna drive motor. The box is remotely controlled from the antenna control box (Part b. below). The contactor boxes supplied with antenna groups serial numbers 1 and 2 (see figures 8 and 9) differ from those bearing serial numbers 3 through 7 (see figures 10 and 11). The two earlier units employ fuses and individual blown fuse indicators, in lieu of the circuit breaker with one indicator lamp, for protection in the three-phase power to the antenna drive motor. The above change and other minor design changes described below, were made to improve the electrical and mechanical reliability of the equipment. In addition, synchro information and the Antenna safety switch are provided in/on the Contactor Box. See "CONTACTOR BOX" in figures 28 and 29 for schematics of the delivered units.

2. Design

The original design of the contactor box was made by Hazeltine in accordance with the specification requirements.

The original design employed an a-c contactor (K101) and a thermal overload relay for the motor.

During the vibration tests (see Hazeltine report 5122, page 15) small particles were rubbed off the main contacts by their holders. The contacts rattled and appeared loose, although the operation of the contactor was not impaired. However, to improve reliability and to provide a cover over the main contacts, it was decided to change the contactor. The new contactor selected is a d-c aircraft type which is vibration resistant, hermetically sealed and lighter than the original contactor.

The new contactor was mounted in its normal operating position on the vibration table and subjected to the vibration tests. There were no mechanical failures during the test and after an electrical test, following the vibration test, the relay functioned as required.

The addition of the new d-c contactor necessitated adding a full-wave rectifier to the contactor box to convert the a-c to the 120 volt d-c required for the contactor coil. The rectifying bridge consists of four 1N540 diodes mounted on a component board near the contactor. The contactor was electrically tested with the rectifying bridge and found to operate with 65 volts a-c input and drop out with an a-c input of 25 volts. The circuit was operated for 25 minutes with an a-c input of 125 volts with no detrimental effects to the contactor or diodes.

The original (a-c) contactor had two sets of normally closed auxiliary contacts, and the d-c replacement has but one set. To provide control for the ANTENNA ON indicator lamp, in the control box, an auxiliary relay (K102) was added.

The contactor box as originally designed included a current sensitive relay to prevent overloading of the motor. However, this relay was set to operate at currents much higher than the motor protection fuses (i.e., 80 amperes for the relay with 30 ampere fuses). This would mean that the relay would never operate since the fuses would blow first. This was confirmed during the temperature test (see Hazeltine report 5122, page 25) when the fuses were replaced with steel slugs and the relay never operated. The thermal overload relay was, therefore, deleted from the contactor box circuitry.

Heater relay (K103) failed to operate properly during the environmental tests. Even though it was established that this was due to a defective relay, it was decided to change to a more reliable type. The new type relay used is hermetically sealed and successfully passed the vibration tests. The new relay is larger than the relay used in the original design and therefore is mounted in a different location, but the accessibility is improved and with a tube pin base can be more easily replaced.

The original design of the motor circuit employed fuses (F101, F102 and F103) as protection in the three phase line. Antenna Group OA-1227(XE-1)/TPS serial numbers 1 and 2 were delivered with this circuitry. To provide more reliable operation and simultaneous opening of all lines to the motor, a three-pole trip free circuit breaker (CB101) was substituted for the three individual fuses. Replacing the individual blown fuse indicators (DS101, DS102 and DS103), a single indicator lamp (DS101) was placed across one of the three phases. The replacement indicator operates the same as a blown fuse indicator, i.e., illuminates with power applied and the circuit breaker tripped.

Other minor circuit changes were effected, these are discussed under the individual Parts as follows:

c. Antenna Drive Motor

e. Lubrication

f. Pressurization

The final design of the contactor box conformed to the specification SCL-5296 and passed all environmental tests successfully.

Part b. ANTENNA CONTROL BOX
(See figure 12)

1. General

The antenna control box is mounted on the wall of Radar Surveillance Shelter AN/GSS-1 and provides remote control and monitoring of the antenna operation. ANTENNA DRIVE MOTOR START-STOP switch S201 turns the motor on or off. OIL HEATER switch (S203) controls heating of the lubricating oil in the oil reservoir. Switch S203 operates independently of START-STOP switch S201 in order that the oil may be pre-heated before starting antenna operation. Switch S202, mounted behind the front panel, is in the air pressure system and is normally set to the OFF position when used with the AN/TPS-1D. However, during high power operation (1.5 to 2 megawatts) where pressurization of the rotary joint is required, the switch is placed to ON, enabling the use of AIR PRESSURE OFF indicator DS204.

2. Design

The original design of the antenna control box was made by Hazeltine to comply with the specification requirements.

The control boxes supplied with the first and secont test models of Antenna Group OA-1227(XE-1)/TPS (Serial numbers 1 and 2) were manufactured in accordance with the original design. The contactor box (Part a) modifications made on serial numbers 3 through 7 necessitated minor lamp rewiring of the control box.

In the original design (and serial numbers 1 and 2) the START-STOP control for the contactor was provided by breaking the neutral lead of the three phase ac. To comply with good design practice, control of the contactor is now provided by breaking one of the three phases.

Schematic diagrams of the delivered models can be found under "ANTENNA CONTROL" in figures 28 and 29.

Part c. ANTENNA DRIVE MOTOR
(See figure 13.)

1. General

The antenna drive motor is mounted beneath the pedestal and drives the antenna in azimuth, through suitable gearing, at a rotation rate of six rpm in a clockwise direction only. The motor is rated at 7.5 hp and requires an input of 120/208 volts, 60 cycle - 3 phase a-c. The motor used is a commercial type with a standard NEMA mounting. To lighten the overall weight of the equipment, Hazeltine requested that the motor manufacturer provide the motor in an aluminum frame.

The motor control circuitry was modified after two test models of Antenna Group OA-1227(XE-1)/TPS were delivered. Simplified schematic diagrams for each model are shown in figures 14 and 15.

2. Design

The size of the antenna drive motor and design of motor control circuitry were made by Hazeltine in accordance with the specification requirements.

To compute the size of motore required, the weight of the reflector and feedhorn were considered with the extreme environmental operating conditions indicated in the specification. The 7.5 hp motor decided upon, proved more than satisfactory under all conditions. Addendum F, of this report, provides calculations supporting the choice of a 7.5 hp motor.

The design changes in the motor control circuitry, i.e. circuit breaker replacing fuses are discussed under Part a. CONTACTOR BOX above.

Part d. SYNCHRO ASSEMBLY
(See figure 16)

1. General

The synchro assembly consists of a 5G, 60-cycle synchro transmitter (B501) which provides one-speed azimuth information, and a 31Tx4, 400-cycle synchro transmitter (B502) which provides 24-speed azimuth information. The antenna drive is geared 1:1 with the 60-cycle synchro, and 1:24 with the 400-cycle synchro. Refer to addendum G for calculations on gearing. A cam driven switch (S501) actuates relay K104 to provide a 400-cycle zeroing voltage to the 400-cycle 24-speed synchro motor (B602) in Azimuth-Range Indicator IP-141/TPS-1D, until the antenna reaches zero azimuth. The 60 and 400-cycle synchro reference voltages and the 27-volt dc for relay K104 are taken from Power Supply PP-674/TPS-1D. The 60-cycle one-speed azimuth information is supplied to terminals in Power Supply PP-674/TPS-1D for use with a PPI repeater.

2. Design

Design of the synchro assembly as made by Hazeltine in accordance with the specification requirements. The original design met all requirements and no changes were necessary.

Part e. LUBRICATION
(See figures 17 and 18)

1. General

The antenna drive gear train is force-lubricated by means of an oil pump geared to the antenna drive.

An oil flow switch (S301) is provided to interrupt power to contactor K101, thus stopping the antenna, if oil flow is missing for two minutes or more. If flow switch S301 is closed, OIL FLOW-LOW indicating lamp on the antenna control box is illuminated. When S301 switch opens, with normal oil flow, OIL FLOW-NORMAL indicator lights. Should S301 fail to open within 2 minutes, thermal relay K105 will open; breaking the circuit to the bridge rectifiers (supplying the coil of contactor K101) thus removing power from the antenna drive motor.

Turning OIL HEATER switch S203 in the antenna control box to the ON position will provide heating of the oil in the oil reservoir if the oil temperature drops below 12.2°C (+10°F). Antenna control indicator OIL TEMP-NORMAL will light if the temperature of the oil is about -27.3°C (-10°F) and be extinguished below this temperature, thus providing an indication that pre-heating of the oil is required, before starting the antenna drive motor.

2. Design

The oil heating system as originally designed by Hazeltine essentially conformed to the system contained in the first two antenna groups delivered (see figure 17) except that the inclusion of a heating element on the oil feed line was planned for in the original design.

Difficulties encountered in installing the circular heating element in the prototype model indicated the inclusion of the element would not be feasible for production. Calculations indicated the element was not necessary and it was decided to omit the oil line heater pending results of the environmental tests. The single heater in the oil reservoir proved more than adequate at -65°F, and the oil line heating element was deleted from the design plan.

The original design plan was for an oil conforming to specification MIL-L-7808 for lubrication of the drive system. Exception to this particular oil was taken by the buyer (Signal Corps) as it is not readily available in the field. Hazeltine tried to accommodate the Signal Corps and investigated all oils, readily available to Signal Corps maintenance personnel. The investigation proved that all available oils were not suitable for this application and reversion to the MIL-L-7808 oil was necessary.

Minor modifications in the oil heating control circuits were instituted in Antenna Group OA-1227(XE-1)/TPS serial numbers 3 through 7; a simplified schematic of this circuit is given in figure 18.

Part f. PRESSURIZATION
(See figures 19 and 20)

1. General

Pressurization of the rotary joint for use with high (1.5 to 2 megawatts) power is a design requirement in accordance with the specification. RF and mechanical difficulties experienced meeting this requirement are discussed under rotary joint. Task A, Phase 1, Part c. and mechanical phase 4 respectively. AIR PRESSURE ON and OFF indicators (DS205 and DS204), energized by air pressure switch S805 in the rotary joint, pressure line, are provided as indications of correct air pressure. At pressures below 12 lbs. per square inch, OFF indicator DS204 is illuminated, indicating below normal air pressure. Pressures above 13 lbs. per square inch actuate switch S805, which illuminate ON indicator DS205, indicating that the rotary joint is correctly pressurized.

Antenna Group OA-1227(XE-1)/TPS is primarily designed to operate with Radar Set AN/TPS-1D at comparatively low (0.5 megawatt) power, which does not require pressurization of the rotary joint. Switch S202 (mounted behind the front panel of the antenna control box) is provided to disconnect OFF indicator DS204 when operating with low power, to avoid presenting a false impression of incorrect operation.

2. Design

Hazeltine's original design plan for the pressurization control system proved to be quite satisfactory. However, to simplify the wiring, minor circuit changes were effected after delivery of Antenna Group OA-1227(XE-1)/TPS. Simplified schematics (figures 19 and 20) indicate the control circuit changes between the earlier equipments and serial numbers 3 through 7.

Part g. CABLING
(See figure 21.)

Cabling for Antenna Group OA-1227(XE-1)/TPS is completely different than that shown in the original design plan. The changes were necessitated by the contract change calling for operation of the antenna group with Radar Surveillance Shelter AN/GSS-1.

All cables except the one to the engine generator were shortened from the original 50 feet specified, since the antenna would be located about 15 feet from the shelter.

All cabling to the shelter was terminated by connectors to facilitate quick assembly and disassembly. Additional cables inside the shelter were required to complete the connections to the units from the interconnecting box.

The interconnecting box is discussed under Task B, Phase 1. RF cabling difficulties and design changes are discussed in Task A, Phase 1, Part e.

PHASE 3. TESTING

Part a. STATIC AND WIND LOAD DEFLECTION TEST

The reflector assembly is required to withstand, without damage or impaired performance, the loading effects of wind conditions as described in paragraphs 3.7.1(e) and 4.3.10 of the Technical Requirements SCL-5296. The tests were performed by D.S. Kennedy & Co. at their Hingham Plant on November 12-13, 1958, using a preproduction model of the OA-1227(XE-1)/TPS forty-foot antenna.

Whereas the maximum loading under the conditions described in the Technical Requirements is 14.7 lbs/sq. ft. (see calculations in addendum H), the reflector was loaded up to 15.9 lbs/sq. ft. during the tests. No unusual deflections were recorded during the test, and no appreciable permanent set was measured at its conclusion. Radiation pattern tests were conducted before and after the deflection test. No pattern changes were brought about by the deflection of the reflector assembly. Thus, on the basis of these test results, the antenna is considered to have easily surpassed its technical requirements. The procedures, conditions and results of this test are provided in addendum I.

Part b. SURVIVAL TEST

For this test the pedestal was fully erected on its legs with ground anchors and tie rods in place, and the reflector assembly mounted upon it. Using an M-35 truck winch and an "A" frame, a static load was applied to the back of the reflector. The 5500 pound load, equal to the maximum survival wind loading, was measured by a 10,000 pound dynamometer.

The results of this test were positive. The entire structure remained stable, with no damage sustained by any of the structural members. A comparison of radiation pattern tests conducted before and after the survival test revealed no pattern changes. A visual inspection of the entire structure, made immediately after the survival test, disclosed that the pedestal was inclined 0° 43' from the vertical due to settling of the ground plates, and there were slight deflections in the ground anchor rods. The antenna assembly is considered to have successfully met its specified requirements in this area.

Part c. MOISTURE RESISTANCE TEST

The moisture resistance test was performed in the temperature altitude humidity chamber of the Burgoyne Test Laboratory Inc., Westbury, N. Y. in the presence of USASRDL personnel. A 1600 lb. load was attached to the rotating portion of the pedestal, such that the dynamic effects of the reflector assembly would be simulated within the confines of the test chamber. The equipment was then subjected to a 48 hour initial conditioning test followed by a variable temperature cycle of 48 hours duration, the latter being repeated 5 times. The equipment was non-operating except for 15 minute test periods during which measurements were taken, all within 5 minutes after the power was applied.

During each of the 5 cycles, readings of the operating voltage and current, starting current, and the power of each of the three phases were recorded. Thermocouples, placed at nine separate locations within the pedestal, monitored its temperature conditions. This test was conducted in accordance with the conditions and procedures set forth in paragraph 4.3.7 of Signal Corps Technical Requirements No. SCL-5296, 8 May 1956. The procedures followed and the entabulated results are given in addendum J.

It was observed that throughout each cycle, both the electrical and temperature readings remained consistent by within ± 10% of the initial conditions. In all cases, starting time was instantaneous, the pedestal speed remaining constant at 6.2 RPM.

No defects of any nature were observed. Since the performance of the equipment was not adversely affected during test conditions, it is concluded that the equipment qualifies under this section of the Signal Corps technical requirements.

Part d. TEMPERATURE TEST

The temperature test was performed in the same location as the moisture resistance test (part c) and conducted in accordance with paragraph 4.3.6 of the Technical Requirements. The procedures, conditions and results of the test are shown in addendum K.

During the cold temperature test certain malfunctions were observed. The reasons for these malfunctions were determined and corrective measures taken as follows:

(1) Malfunction

Initial failure of the pedestal to rotate at -65°F.

Reason

From observations made during the test, and subsequent conversation with the motor manufacturer, it is know that moisture probably entered the motor case through the normal breathing process during the 10 day moisture resistance test. It is believed that this moisture froze, thereby locking the rotor. In all probability, high current surges eventually melted the ice, whereupon the pedestal began to function in a normal manner. Had the temperature test been performed prior to the moisture test, this situation would probably not have occurred.

Solution

Following the recommendation of the motor manufacturer a note will be included in the maintenance manual that the motor drain plug be removed and any moisture present be allowed to drain out before shipment of the equipment to a frigid environment.

(2) Malfunction

Almost instantaneous blowing of fuses at locked rotor condition.

Reason

Poor choice of fuses.

Solution

Improper fuses have been replaced with Military Standard 90085-70 fuses. These are slow-blow fuses with a minimum delay time of 6 seconds for locked rotor conditions. This will aid in starting the motor should it ever inadvertently freeze again. Assurances have been received from the motor manufacturer that 6 seconds of locked rotor current will not damage the motor in any way. (Note: at -65°F a 30 second delay is harmless.)

(3) Malfunction

Heater relay failed to close.

Reason

Failure of the particular relay used during the test. Temperature is not believed to be a factor.

Solution

A second relay has been tested at -65°F and found to function satisfactorily. This relay has been substituted for the defective one.

(4) Malfunction

Motor control thermostat prevented pedestal from rotating once it started.

Reason

As the pedestal started to rotate, the oil began to circulate and cold oil came into contact with the thermostat thereby opening it and shutting the motor off. A check of the thermostat after the test showed it to be set at +20.3°F instead of -10°F as called for.

Solution

Thermostat S303 will be set to -10°F and will be used for the sole purpose of operating a green light on the control panel. This light will be off until the oil temperature reaches -10°F. Above -10°F the light will be on and all oil will be filtered. The Heater Control Thermostat will be set so as to turn on the heater when the temperature falls below +10°F.

(5) With the above modifications, and the observations made during the cold temperature test, it is felt that the pedestal can be operated instantaneously, if need be, at any of the environmental conditions specified in SCL-5296, thereby bettering the requirements specified in paragraph 3.7.7.

Part e. VIBRATION TEST

In accordance with paragraph 4.3.1 of the Technical Requirements, the pedestal subassemblies containing sensitive electrical and mechanical components were subjected to vibration testing.

The tests described below were performed on a vibration table located in the test laboratory of Hazeltine Electronics Division, Little Neck, New York. These tests were witnessed by V. Tristan, a resident Naval Inspector.

(1) Control Box

The control box cover was removed and attached to the back of the control box. The box was then mounted by means of four 1/4 - 20 screws to a 1" aluminum plate. This plate was then fastened to a 150 lb. 1/2" thick steel plate which was in turn bolted to a vibration table.

The equipment was vibrated at 1/64" amplitude in three planes at frequencies of 5 to 55 cycles/second changing at a rate of 1 cycle/second. The equipment was cycled 4 times. A strobe light as well as visual inspection was used to determine any mechanical resonances, none were noted. The unit passed satisfactorily.

(2) Synchro Assembly

The synchro assembly was mounted to a fixture which was bolted directly to the vibration table. Test conditions were the same as described in paragraph(1). There were no mechanical resonances noted. The assembly passed satisfactorily.

(3) Contactor

The contactor was fastened to a 1" thick aluminum plate and vibrated on two separate occasions. The first was under the conditions described in paragraph (1). Loose washers and other parts rattled during this test due to the floating contacts but no damage was visible.

The sécond vibration test was performed at an amplitude of 1/16" and a frequency of 55 cycles/second for a duration of 15 minutes in each plane. No damage was noted outside of some particles which were rubbed off the contacts by their holders.

(4) Contactor Box

The contactor box was mounted on a 1" aluminum plate. The plate was bolted to the vibration platform and held approximately 7/8" above it to allow the cable conduit to clear. Test conditions were the same as described in paragraph(1). The loose contactor parts rattled and the core of the thermal relay moved in and out. There was no damage noted after the test.

PHASE 4. MECHANICAL

Mechanical design problems encountered during development of Antenna Group OA-1227(XE-1)/TPS are discussed in parts as outlined in Section 1.

The mechanical design changes for the rf components (reflector, feedhorn, rotary joint and coax-to-waveguide transition) are discussed in Task A, Phase 1.

Detailed final design procedures for antenna assembly, disassembly and component stowage in transit cases are given in the Technical Manual for Antenna Group OA-1227(XE-1)/TPS.

Calculations providing the overturn moment, of the assembly, during erection are given in addendum L.

Part a. ANTENNA PEDESTAL
(See figure 13)

1. General

Mechanical design of the antenna pedestal was generated to fulfill the electrical, mechanical and environmental requirements of the equipment specification. The Design Plan was used as a guide for development of the Pedestal.

2. Design

The original design plan of the antenna pedestal component parts is the same as the final design except as given below.

(a) ANTENNA DRIVE MOTOR. - Lightened by use of cast aluminum frame instead of cast iron.

(b) CONTACTOR BOX. - Component locations were rearranged to accommodate inclusion of circuit breaker in place of fuses. (Serial numbers 3 through 7 only.)

(c) REFLECTOR SUPPORT. - The lifting points on the originally designed pedestal required one short and two long legs of the sling for lifting. To avoid possible mistakes in attaching the sling, the lifting points were relocated to use a sling with three short legs.

(d) PEDESTAL WEIGHT

Before any official drawings existed, a preliminary estimate was made, predicting the pedestal weight to be approximately 1500 pounds. With later developments and the availability of more accurate information, the necessity for a reappraisal of this estimate was recognized. Listed below are areas where such information influenced this evolution:

(1) Hardware and Internal Components

Little was known of the weights and quantities of hardware and components which were eventually incorporated in the final design.

(2) Rotary Joint

All of the electrical and mechanical requirements of the rotary joint were not known in sufficient detail to accurately predict its weight. It eventually proved necessary to employ heavy-wall brass in its construction in order to provide the ideal electrical properties and sufficient strength to contain the required pressure. Consequently, its final weigth was over four times as great as the original estimate.

(3) Main Bearing

The original estimate of the weight of the main bearing was evolved from other information of a preliminary nature. It was later found that the weight of the reflector assembly was higher than the preliminary figure, and that its center of gravity was further from the center of the pedestal than originally assumed. Thus the offset load moment was much higher than anticipated. This, in turn necessitated an increase in the load-carrying capacity of the main bearing with a resultant increase in its size, and therefore, its weight.

(4) Castings

(a) The wall thicknesses of certain portions of the castings, such as strength webs and stiffeners, became greater as dictated by optimum foundry practices. The relationship between adjacent molten masses of various sizes cooling in a mold, must be such that distortion is held to a minimum.

(b) Solid cores were used in making the sand molds for the castings. In removing the cores from the molds, it was necessary to "rap" them in order to work them free. This "rapping" caused the mold cavities to become slightly enlarged, with the consequence that a greater amount of molten metal flowed into each mold. The result was heavier castings.

The circumstances listed above had the cumulative effect of increasing the pedestal weight to approximately 2600 pounds.

Part b. TRANSIT CASES
(See figures 21 and 22)

1. General

Two transit cases for transport stowage of the antenna group components are provided. In accordance with the revised specification requirements, the maximum height of the loaded truck can not exceed 11 feet and the case's center of gravity shall be low. To comply with these requirements required considerable changes from the original design plan (Hazeltine Report No. 5227).

The antenna reflector case of final design is 146 inches long x 80 inches high x 86-1/2 inches wide. The antenna pedestal case of final design is 146 inches long x 81 inches high x 86-1/2 inches wide.

2. Design

After the design of the reflector was established, the reflector transit case design was started. Preliminary layouts showed that only two possible arrangements of the reflector sections could be used to fit the eight sections in one case. Assembly sequence was considered but could not be maintained.

As two connections on each horizontal joint of the section were available the opposite half of the connection was secured to two 4 x 4 beams in the transit case bottom for the main support. The top side of each section was held by two light aluminum braces, which at first were rubber mounted, but were later revised to a solid mount because of excessive transmissibility during the package test. As the reflector sections used up about 85% of the available space in one transit case, only a limited number of additional parts could be stowed with it. The auxiliary "A" frame, used for erection, was stowed on the outside as there was no room in either case. As the Signal Corps wanted to unload the reflector case on the truck, a swing out step was provided in front. The tail gate could be used as a step in the rear. It was not possible to get between the case and the reflector sections. Access to the reflector sections was by means of three hinged gates secured by the Kennedy designed connector (see Phase 1, Part a).

The pedestal transit case presented many more difficult stowing problems than the reflector case. Most parts were of a shape that would not allow economical placement. Most difficult were the pedestal, pedestal support legs, feedhorn assembly, and center reflector support. The great majority of parts were of tubular sections, so a clamp was devised using a 1/4 turn Air-Loc fastener foreclosure. As the clamp had to clamp tightly, the Air-Loc fastener was found unsuitable as it could not pull the hinged clamp tight enough. In place of this fastener a Simmons Link Lock #2 was used with good results. As the pedestal weighed over 2500 lbs. it presented a large problem in itself. The case could not be made strong enough to take 10G shocks for three hours on an item of this weight, without some attenuation of the shock. Twelve (12) Lord compression mounts that allowed 1" of motion were mounted on a 3-1/2" aluminum tube at the six main leg mounting points. As these were calculated to be on a plane in line with the center of gravity of the pedestal, there should be a little rocking motion caused by vibration and shocks. During the package test it was noted that there was some rocking, indicating the mounting was not exactly on the plane of the center of gravity. However, the shock-mounts, pedestal and case showed no signs of damage. In order that the pedestal could be removed first, a gate in the top of the case (over the pedestal) was provided.

Many of the problems that occurred were not apparent during the initial proposal. Solution of the problems was arrived at only after many times the allotted hours. Because of the urgency of delivery, design of the pedestal transit case suffered greatly. Final design was accomplished only after one was built. As a matter of fact, final placement of the stowed parts was arrived at only by actual placement of the full size pieces, and engineering personnel providing on the spot supervision.

In spite of the many difficulties, a sound adequate design was obtained and few changes would be necessary to optimize the parts. In general the overall design of both transit cases was accomplished maintaining a complete structure at all sides.

At various stages in the development and testing of this equipment, difficulties have arisen in the fastening of rubber chafing pads to the inner surfaces of clamps on the transit cases. Some rubber pads, seemingly securely attached, could be lifted and peeled off by hand. Others would vibrate free during a test, and still others would fall off by themselves, after having been securely attached for a considerable period. Continuous investigations and testing were conducted until the causes and remedies for this problem were discovered.

Among the causes for this deficiency are the following:

(1) Certain adhesives recommended for bonding rubber to metal surfaces, although showing excellent strength in tension,

proved to be weak in shear. As a result, there was little resistance to scuffing and peeling, the most common disturbances to which the pads are exposed.

(2) Some of the rubber samples tested exhibited a relatively high wax content. The presence of wax has a retardant affect on adhesive bonding.

(3) Unclean surfaces:

(a) It was found that dirt, grime or paint were present under some of the pads which had a bonding deficiency.

(b) The rubber itself has molding powder on its surfaces. This powder acts as a retardant to adhesive bonding.

(4) Improper application:

(a) In some cases, it was evident that an insufficient amount of adhesive had been applied, parts of the rubber pad surface being without adhesive.

(b) Certain adhesives used, required intermittent stages of application and air-curing prior to final contact. Doubt exists that these procedures were strictly followed in all cases.

(c) Some adhesives required the application of pressure after contact. In some instances this may not have been done for the required period.

Accidental mechanical abrasion accounted for some of the deficiencies. There is no adhesive known that can withstand abrasive forces in the magnitude possible during the handling of heavy equipment.

The eventual solution to this problem lay in the elimination of all of the above factors, which alone, are causes of adhesive bonding deficiencies. This was accomplished by:

(1) Using Eastman 9-10 adhesive, which has a relatively high shear strength, and presents no particular problems in application or curing.

(2) Thorough cleaning of all surfaces.

(a) Metal surfaces cleaned by scraping to bare metal, removing all paint, grime, and oxide films. Metallic dust is wiped off and the surface treated with MEK solution.

(b) Rubber surfaces to be cemented are prepared by abrading.

(c) Molding dust is removed from rubber surface by brushing with MEK solution.

(3) Application of adhesive is administered carefully, so entire pad surface is covered.

(4) Pressure applied to the rubber pads by closing the clamp tightly about a pipe.

(5) Rubber having a lower wax content was used, samples being tested intermittently.

In view of the findings determined by the field evaluation tests and Hazeltine inspection on their return from Fort Bliss, Texas, the following minor changes were made:

(a) To both transit cases:

(1) The bottom portions of both ends were reinforced to make the case capable of being fork lifted.

(2) Jam nuts were added to turnbuckles on hold down cables.

(3) Holes in the case skids were enlarged so they could accept the no. 30 hook supplied with the M-35 truck.

(4) All door holding catches were changed to the quick disconnect type used on the antenna reflector.

(5) All Airlock Fasteners were replaced by a Simmons Link-Lock type fastener.

(b) To the antenna reflector case:

(1) Clamps were added to the auxiliary "A" frame stowage to prevent end movement.

(2) The top member of the door side of the case was strengthened.

(c) To the antenna pedestal case:

(1) Brackets were added on the top of the tool box to secure the antenna control box during transit.

(2) A reinforcing gusset was added to the upright near the tool box.

(3) The access door in the top was redesigned to eliminate brackets protruding in the pedestal area.

(4) Brackets that support the bridges were set further apart to provide more room in the contactor box area. The resultant long bridges were reinforced to prevent bending.

(5) Lifting holes were reinforced.

(6) The method of securing the cable reel was changed from "J" bolts to a clamp through the center hub.

(7) The rear access door was reinforced.

(8) The lead of pedestal bridges were chamfered to facilitate loading.

(9) Bracket stops were added to prevent the jack screws, on the tripod legs, from unscrewing during transit.

(10) The antenna rear support stowage was redesigned; from a stud that fastened to the threaded hole in the support, to a socket type mount.

(11) The stowage of the front antenna support was relocated to prevent the captive screws from rubbing on the left inner truss.

3. Bounce and Shock Tests

The transit cases were subjected to bounce and shock tests in accordance with the provisions of paragraphs 3.8 and 4.3 of the Technical Requirements No. SCL-5296. These tests were conducted for Hazeltine Electronics Division under the direction of USASRDL personnel for the purpose of qualifying the transit case as a shipping container for military radar equipment.

(a) BOUNCE TEST

Each transit case, with contents in place, was mounted on a 4600 pound steel structure simulating a truck bed. The transit cases were secured to the carriage with four chains equipped with load binders and restricted from lateral motion by corner fences attached to the carriage. At Package Research Laboratory, Rockaway, New Jersey, the carriage was placed upon the vibrating platform of an L.A.B. Corp. package tester of 10,000 lbs. capacity, 1" amplitude, operating in a synchronous circular orbit in the vertical plane. Vertical forces were measured by an accelerometer affixed to the carriage, its amplified output being displayed on a CRT oscilloscope. The results of the test are as follows:

(1) Reflector Case

One guy rod clamp slipped allowing the free end to whip against parts below. The test was interrupted and it was seen that improper tightening of clamp bolts allowed this to occur. After the clamp was relocated and tightened, no further loosening occurred.

The top "A" frame clamp broke, adjacent to the weld which attached it to the case. The test was interrupted and it was found that a rubber liner in the clamp had been lost, and the "A" frame cable was causing abrasion to the "A" frame. Several changes were made on the reflector case and the "A" frame as as result of this test. A clamp was used to replace the yoke which held the apex of the "A" frame to the transit case. The upper swiveling guide of the "A" frame was made secure with a fastener and a storage bin was installed on the "A" frame for stowing the "A" frame bridle.

An anchor bolt sheared at the frame end of the rear stabilizing tube. The test was interrupted, and the bolt replaced. The bolt was probably faulty since all other anchor bolts held. Peaks of 10g and 7g were subsequently reached without further failure.

(2) Pedestal Case

After peaks up to 10g, a clamp attached to the pedestal mount, holding the "Right Inner Truss" broke near the attachment weld. The test was not interrupted.

After 30 minutes of testing, it became evident that the concentration of the pedestal weight at one end of the case caused an imbalance, leading to higher values of vertical acceleration than would normally be encountered. The test was interrupted and 150 pounds of iron added to the end of the carriage opposite the pedestal. The 150 pound weight was later increased to 436 pounds.

A captive pin backed out, releasing one of the leveling jack screws. The test was interrupted, and the jack screw was removed and left out for the remainder of the test. A positive locking device was used to replace the captivating pin for the jack screws.

All other broken clamps were repaired, reinforced and relocated so as to prevent further failure.

(3) Conclusion

After these minor failures had been discovered and corrected, the tests proceeded without further difficulty. The transit cases were considered suitable for their intended purpose in cognizance of paragraphs 3.8 and 4.3 of the Technical Requirements. A complete description of the tests and recorded data is included in Section F of Hazeltine Report No. 5122 "Operating and Performance Tests."

(b) SHOCK TEST

In accordance with paragraph 4.3.5 of the Technical Requirements, both transit cases were subjected to a free fall drop test.

Each case was supported at one corner of its base by a block 5 inches in height. A 12-inch block was placed under the other corner on the same end of the transit case. By means of a quick-release hook, the opposite end of the case was raised 6 inches and allowed to fall freely. This operation was repeated until each of the four corners had been dropped.

(1) Reflector Case

The reflector case was tested at the Package Research Laboratory, Rockaway, New Jersey. An inspection revealed no damage as a result of this test.

(2) Pedestal Case

The pedestal case was tested at Suffolk Products Corp., Greenlawn, New York, with no damage resulting from this test.

A description of the shock tests is included as section H of Hazeltine Report No. 5122, "Operating and Performance Tests."

Part c. INSTALLATION ACCESSORIES
(See figures 24 through 27.)

1. General

In accordance with the contract requirements accessory components are included, to aid in the assembly of Antenna Group OA-1227(XE-1)/TPS. The contract was modified to include the truck bumper support jacks (figure 25), when investigation determined that the front axle of the M-35 truck was only rated for an additional 500 lb. load.

2. Design

The auxiliary "A" frame as originally designed was thought to be foolproof. However, during the field evaluation tests at Fort Bliss, Texas, a tearing of the frame occurred during erection of the antenna. Hazeltine engineers examined the damaged frame and concluded the failure had occurred because the erection instructions had not been followed. However, in order to prevent other such occurrences, Hazeltine redesigned the auxiliary "A" frame. Extensive tests and use in numerous erections produced no further failures.

The ground nail originally planned for use, included a lifting eye for easy removal. However, during the field evaluation tests a lifting eye broke at the weld while the nail was being driven into the ground by a sledge hammer. In view of this failure, ground nails being supplied (see figure 26) have a straight pin, below the nail head, in lieu of the lifting eye. The nail head shoulder, extending below the pin, prevents the pin from contacting the support pad. This allows the hook of the "A" frame cable to engage the pin, permitting easy removal from the ground.

One basic lift sling is provided (A - figure 27), by removal of the various extensions, as indicated, the one sling is used to perform all lifting functions.

The guy rods as provided are adjustable, with a minimum length of 7 feet 3 inches; which length, calculations have proven, is the minimum acceptable guy length.

Two types of soil anchors are provided for soft and hard soils. In general the screw type anchor is used whenever possible. The arrowhead type anchor is used when the soil is too hard to accommodate the screw type. A very comprehensive investigation of soil types, and the anchors required for each type, was conducted. In collaboration with the Signal Corps it was decided not to provide anchors for rocky conditions as special drilling tools would be required, or swampy conditions as the area would probably be inaccessible by truck. Different diameters (8 inches to 16 inches) of screw anchors were investigated; the 10 inch diameter anchor supplied, was found to hold best in most soil conditions. Refer to addendum M for information compiled on all suitable types of ground anchors.

Part d. MISCELLANEOUS

1. Corrosion of some steel parts was observed during the field evaluation tests; where permissible, these parts will be given an application of grease.

2. All rubber surfaces will not be painted.

3. Inspection of the first service test model, by Hazeltine, after the field evaluation tests revealed loose gaskets and loose rubber bumpers. To prevent future loosening, Eastman 910, a rapid setting, high strength adhesive is now used to secure all rubber items.

4. The name and number tags identifying the assembly parts originally were decals, these became loose and fell off during the evaluation tests. The decals were changed to metal tags and are now securely fastened with drive screws.

TASK B. MODIFICATION OF SHELTER

The shelter modifications were necessitated by the contractual change requiring Antenna Group OA-1227(XE-1)/TPS to operate with Radar Surveillance Shelter AN/GSS-1.

This task is divided into two phases - circuitry and mechanical. The antenna control box (while mounted in the shelter) is supplied as part of the antenna group. Reference to the control box under Task B is therefore limited to mounting requirements and cabling for the box.

PHASE 1. CIRCUITRY

Circuitry for the shelter modification is discussed in two parts, part a interconnecting box and part b cabling.

Part a. INTERCONNECTING BOX

1. <u>General</u>

To comply with the specification requirement (to filter all voltage lines entering and leaving the shelter) the interconnecting box is provided as part of the shelter modification. In addition to filtering the voltage lines, the box provides a convenient termination for all antenna to shelter cabling except the rf cable (RF30), thus facilitating assembly and disassembly.

2. Design

To provide the correct filtering for the interconnecting box, four different types of filters are employed. The filters are purchased parts from Sprague Electric Co., the four Sprague types and ratings are as follows:

2JX44A1 - 2 amp 115-volt 60-cycle

S-68316 - 0.1 amp 115-volt 60-cycle

5JX94 - 5 amp 115-volt 400-cycle

5JX27 - 5 amp 50-volt dc

To accommodate the control wiring changes between antenna groups serial numbers 1, 2 and 3 through 7, filters FL909, FL910 and FL925 were changed from the 0.1 amp, 115-volt, 60-cycle type to the 2 amp, 115-volt, 60-cycle type.

Part b. CABLING
(See figure 21)

1. General

Cables are not supplied as part of the modification kit. However, cable W29, supplied with Antenna Group OA-1227(XE-1)/TPS, is required for connection of the antenna control box within the shelter. The balance of the cables within the shelter (W10, W11, W701 and RF854) required for the shelter modification are supplied with Radar Surveillance Shelter AN/GSS-1.

2. Design

All cabling design changes in the shelter were in the rf cable. Originally a stranded conductor (Amphenol type 421-103) rf cable was used. The Fort Bliss, Texas evaluation test proved this cable to be inferior to the solid conductor RG-19/U type. The final design utilizes a section of RG-19/U (RF854) supplied with shelter AN/GSS-1.

The original design for rf connector (RF901) was for a straight through connection, with both connections at right angles to the shelter wall. Changing rf cable RF30 (See Task A, Phase 1, Part e.) to type RG-19/U, necessitated a right angle connector at the outside of the shelter. Changing this connector permitted the cable to parallel the shelter wall, thus allowing regulation of the solid conductor cable bend radius.

PHASE 2. MECHANICAL

1. General

The mechanical phase of Task B consists of (1) the physical modifications required on the shelter wall to accommodate rf connector (RF901) and the interconnecting box, and (2) installation of the antenna control box mounting.

All hardware and templates to complete the modification are supplied as part of the modification kit. In addition cover plates are provided for the wall openings made to accommodate the interconnecting box and rf connector. These cover plates are for use if the components are removed from the shelter.

Refer to addendum to Technical Manual for Antenna Group OA-1227(XE-1)/TPS for final design details of AN/GSS-1 modification for antenna group.

2. Design

In the original design of the shelter modification (as shipped to Fort Bliss, Texas) the interconnecting box and rf connector adapter plates were to be attached to the outer wall of the shelter with rivnuts. The Signal Corps objected to this type of construction because a special tool would be required for installation. To comply with the Signal Corps objection, Hazeltine redesigned the installation utilizing back-up plates with threaded inserts for attachment of the adaptor plates to the shelter.

SECTION 5. CONCLUSIONS

The following conclusions presented are the result of evaluating the factual data presented in Section 4.

1. REFLECTOR AND FEEDHORN

The design and construction of the reflector and feedhorn was carried out by the Donald S. Kennedy Co.

Overall, the performance of the reflector and structural design is adequate. An amendment to the specification was required to finalize the feedhorn design. The specification change for increased IFF vertical beamwidth actually increased the antenna coverage and enabled meeting the low VSWR requirements.

2. ROTARY JOINT

The original specification requirement of 0.4 lbs. permitted loss of pressure per 24 hour period, would have resulted in a rotary joint that would have been operating in critical areas. Amending the specification to permit losses of from 1.35 psi to 1.675 psi per day enabled the final design of a very satisfactory rotary joint.

SECTION 6. OVERALL CONCLUSIONS

Antenna Group OA-1227(XE-1)/TPS has undergone a series of tests that prove the equipment is acceptable for military use. The tests described in Section 4 and Hazeltine Report No. 5122, have subjected the equipment to its operating extremes. No major faults or failures have developed during this comprehensive program. The minor failures that developed, have been corrected so as to preclude any further failure.

The service tests (at Fort Bliss, Texas) performed on the antenna group, tested it in actual operation for a period of over three months. Preliminary information, on the results of these tests, indicate that equipment performance was satisfactory.

Antenna Group OA-1227(XE-1)/TPS should therefore be considered as an equipment providing satisfactory operation.

SECTION 7. RECOMMENDATIONS

While the final design was adequate, certain areas lend themselves to further improvement. Therefore, the following recommendations should be considered for the future:

a. To reduce the pedestal weight it is recommended that:

(1) A rotary joint be fabricated using more aluminum and less brass.

(2) Further tests be conducted with an assembled antenna, simulating extreme specification loading conditions (e.g., ice and wind) to determine if a lower horsepower antenna drive motor is feasible.

b. Redesign of the auxiliary "A" frame to enable the truck winch operator to complete erection of assembled antenna without use of tag line.

c. For production in quantity it is recommended that the feedhorn be cast to insure uniformity of patterns and VSWR losses.

d. To forestall breakage of indicating lights on the contactor box, it is recommended that the pedestal be indented to receive the box, and a protective cover supplied.

SECTION 8. IDENTIFICATION OF KEY PERSONNEL

The following key personnel of Hazeltine Electronics Division participated in the project described in this report:

John Sinnott	Design and Development Supervisor
Gerold W. Bugen	Supervisory Electrical Engineer
Ira Krupen	Mechanical Engineering Section Head
Paul L. Dominique	Supervisory Mechanical Engineer
Frank J. Delany	Project Administrator
William F. Gerold	Project Administrator
Donald T. Geiger	Ass't Project Administrator

HAZELTINE ELECTRONICS DIVISION
Hazeltine Corporation

P. Dominique
Supervisory Mechanical Engineer

John Sinnott
Design Supervisor

Job No. 208

REPORT 10075

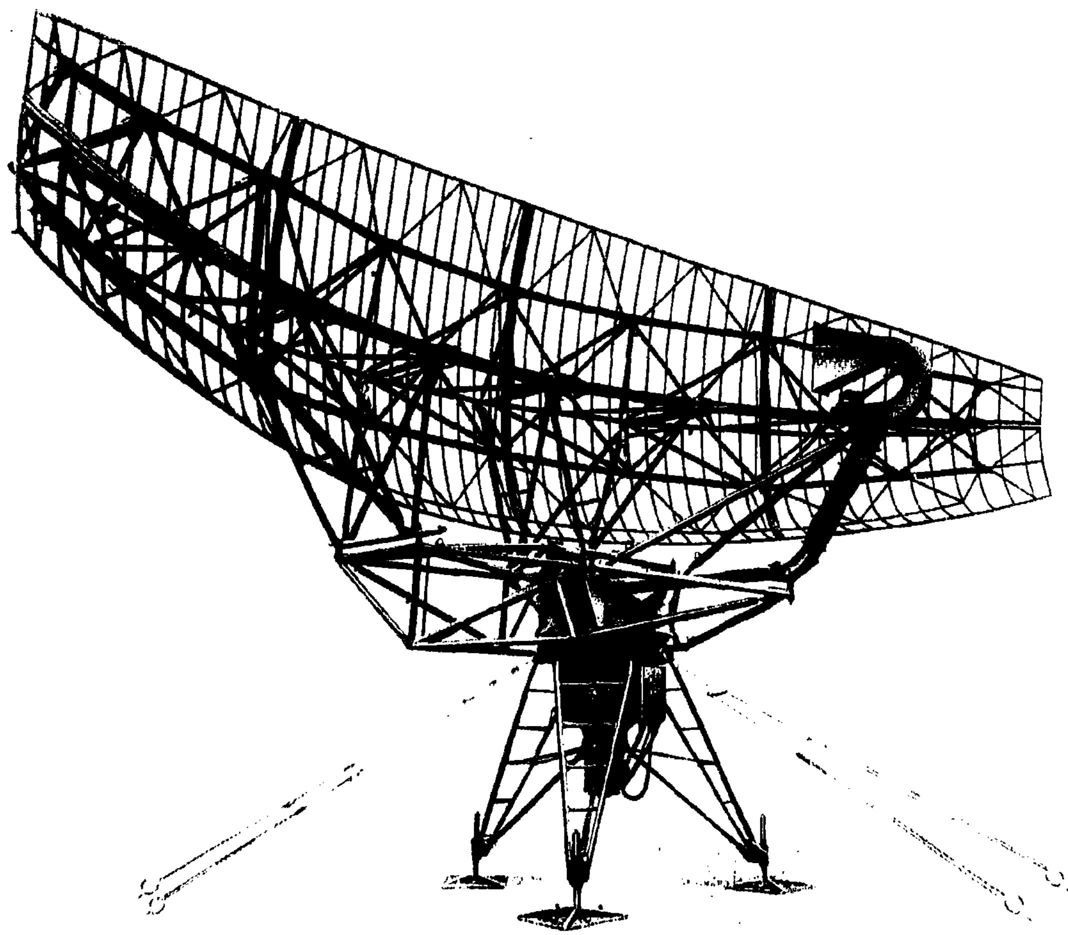

Figure 1. Antenna Group OA-1227(XE-1)/TPS

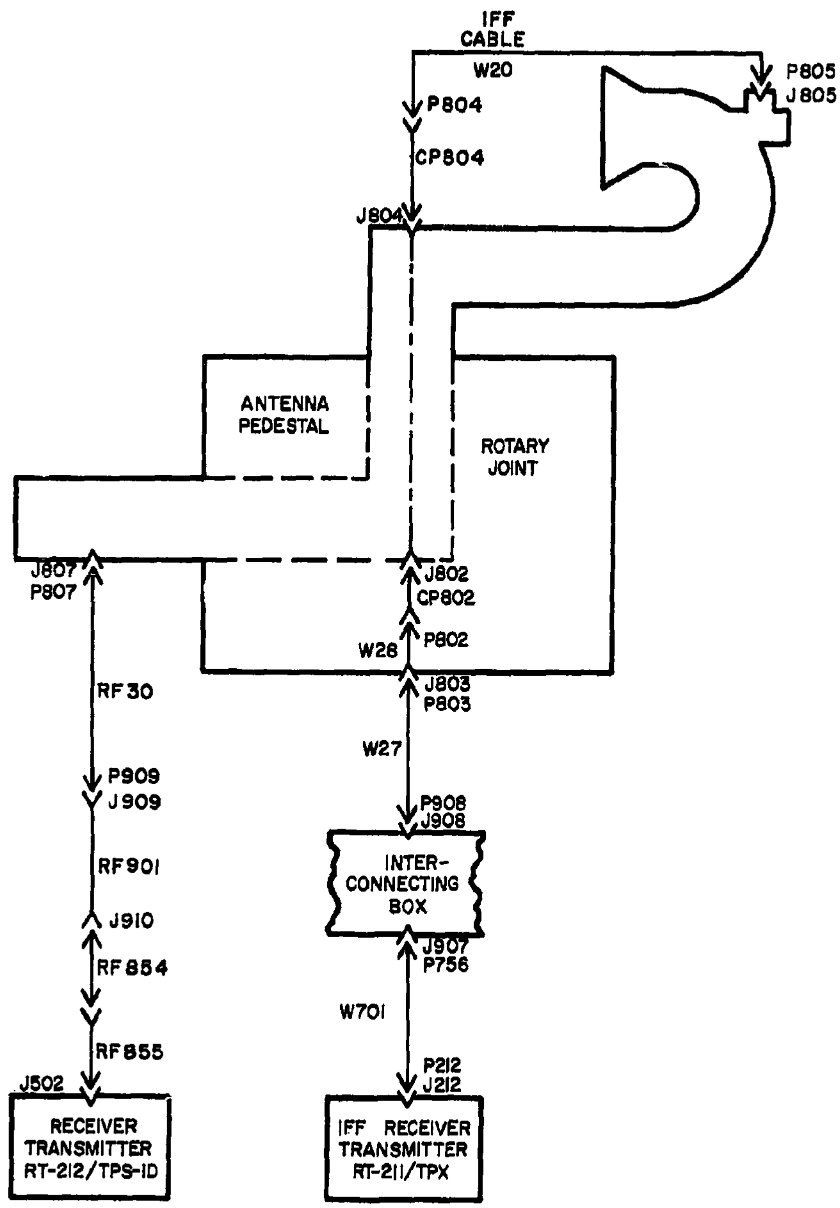

Figure 2. RF Transmission Paths

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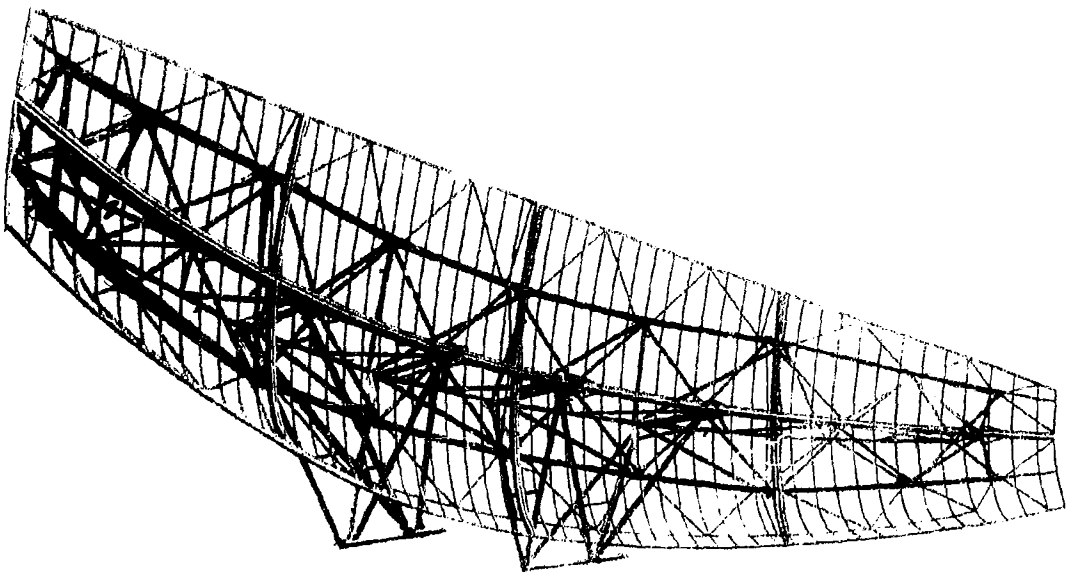

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Figure 3. Assembled Antenna Reflector

P 22 AV 2 R

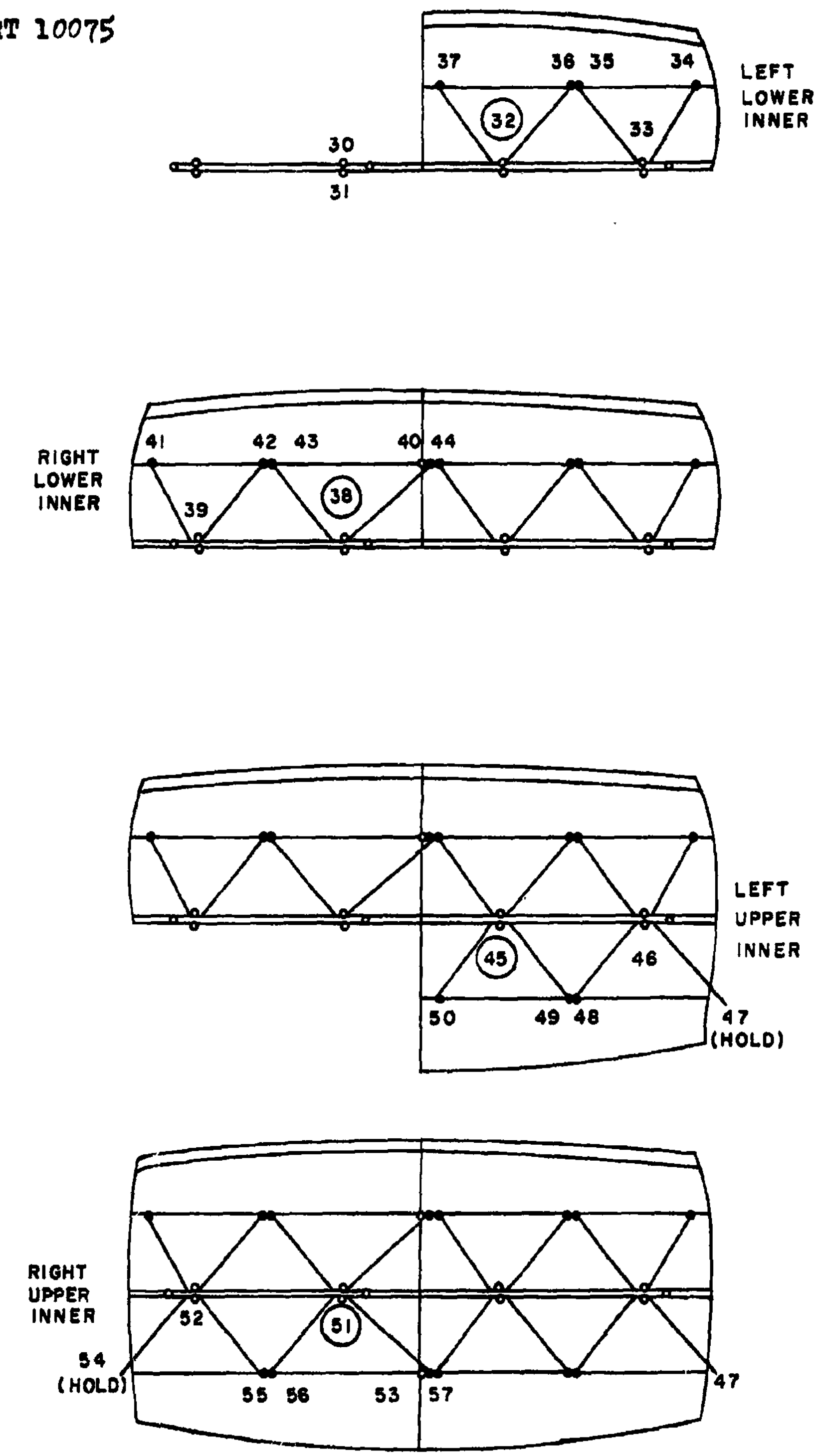

Figure 4. Assembling Antenna Reflector (Sheet 1 of 2)

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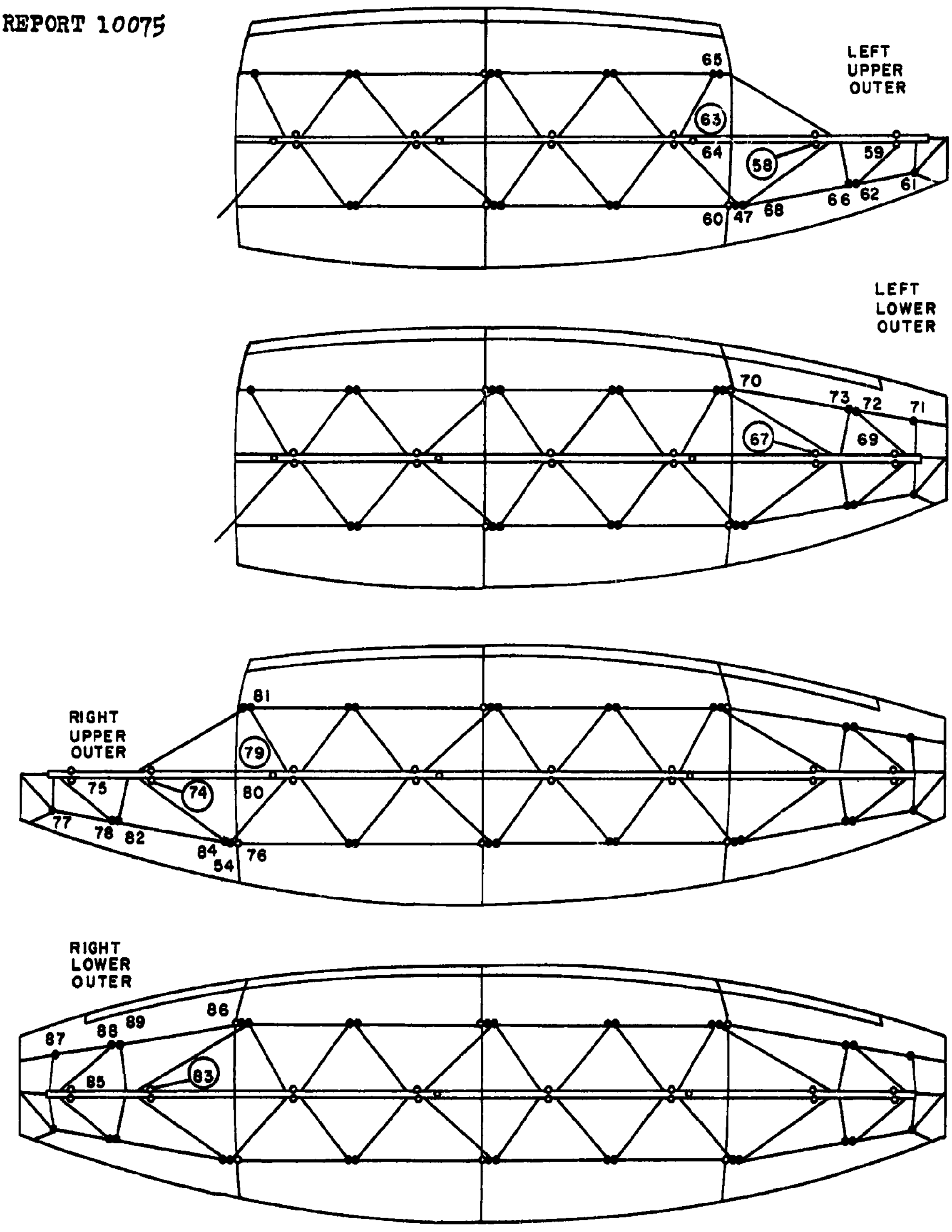

Figure 4. Assembling Antenna Reflector (Sheet 2 of 2)

P 22 AV 2 R

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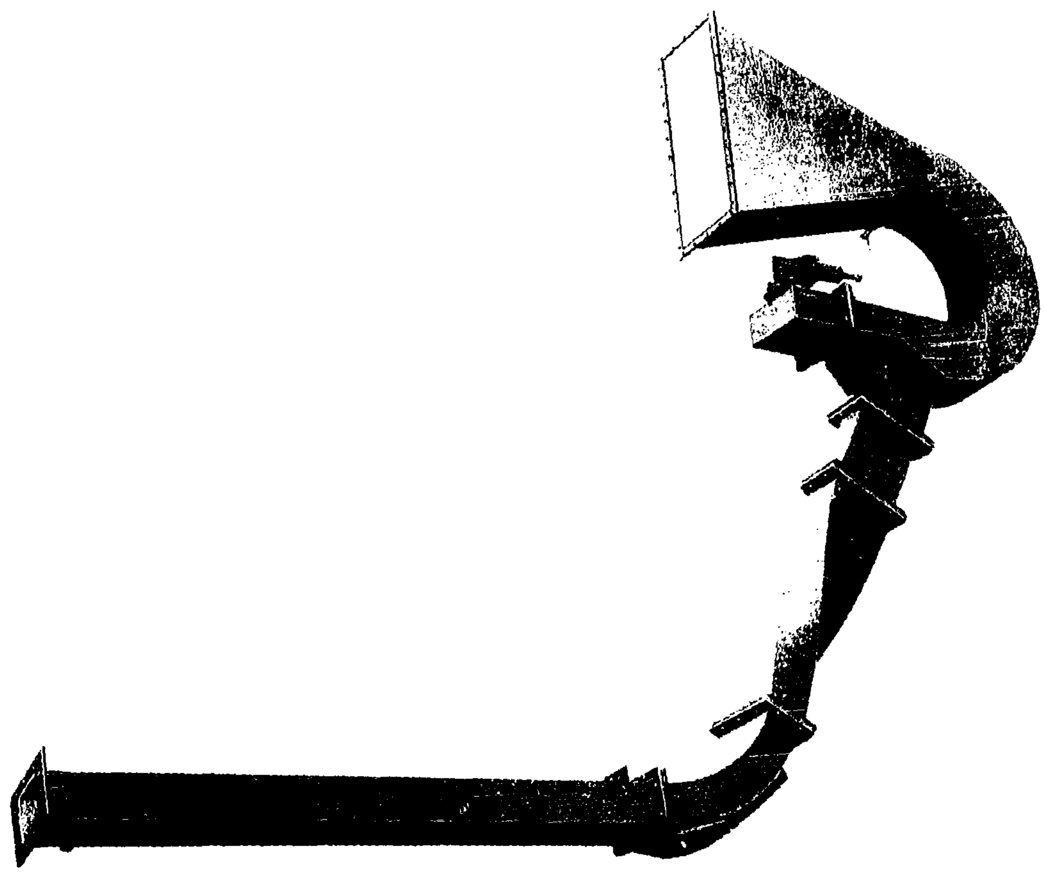

Figure 5. Feedhorn Assembly

59B1416

P 22 AV 2 R

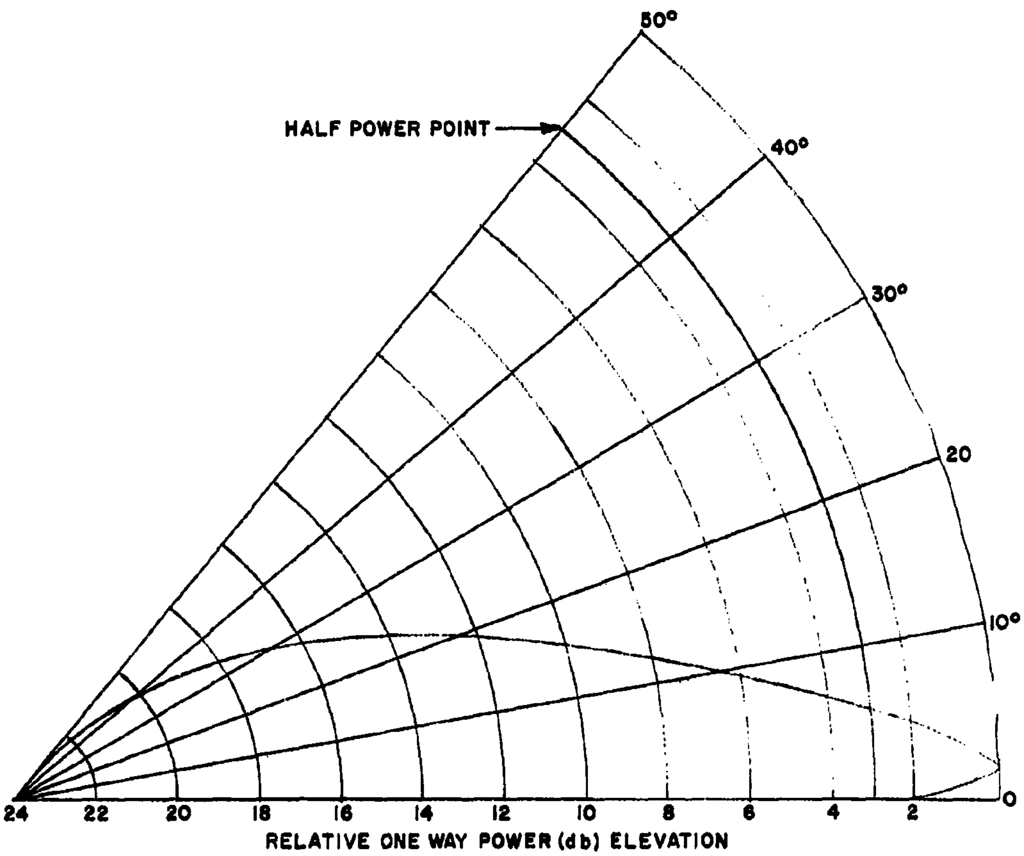

Figure 6. Antenna Radiation Patterns (Sheet 1 of 2)

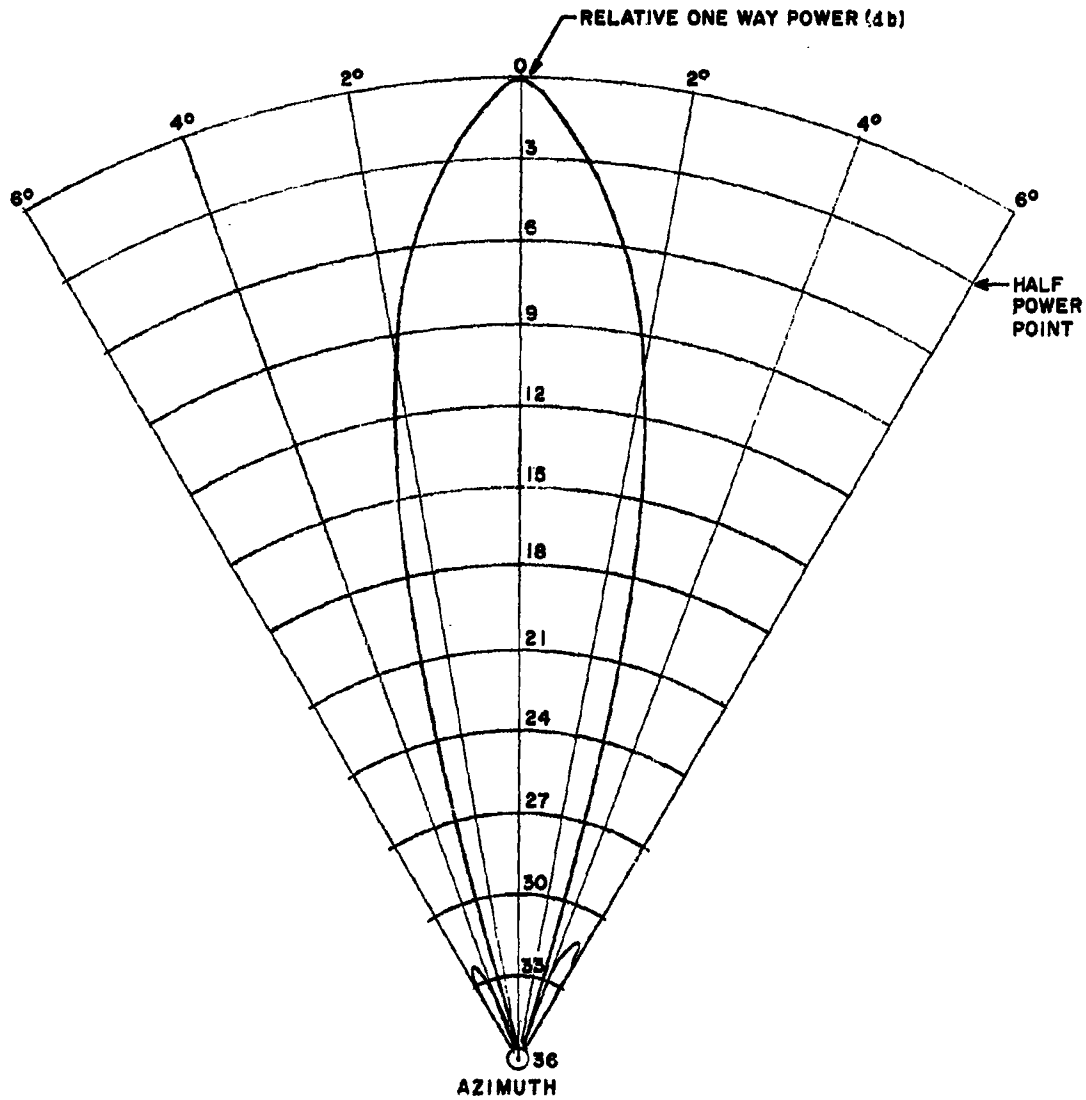

Figure 6. Antenna Radiation Patterns (Sheet 2 of 2)

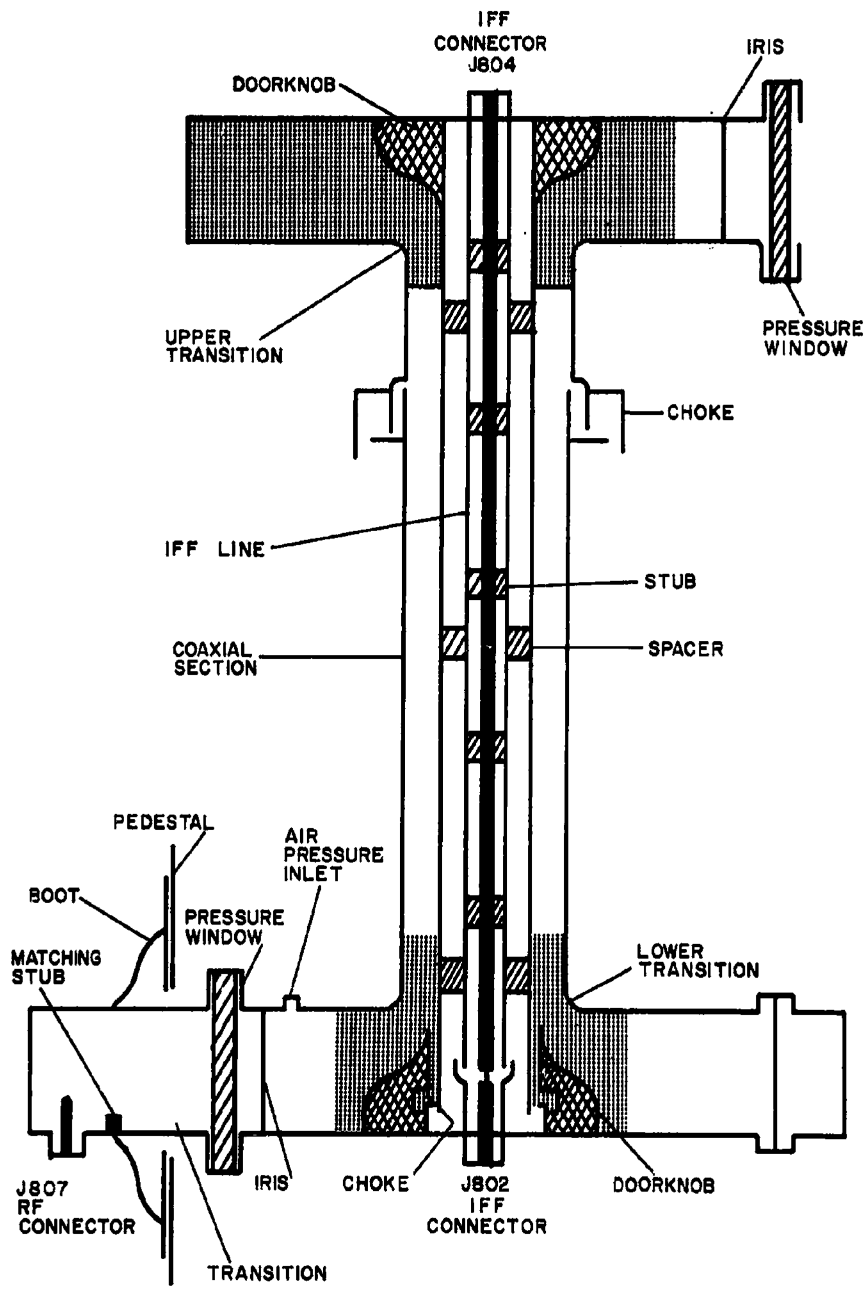

Figure 7. Rotary Joint Assembly

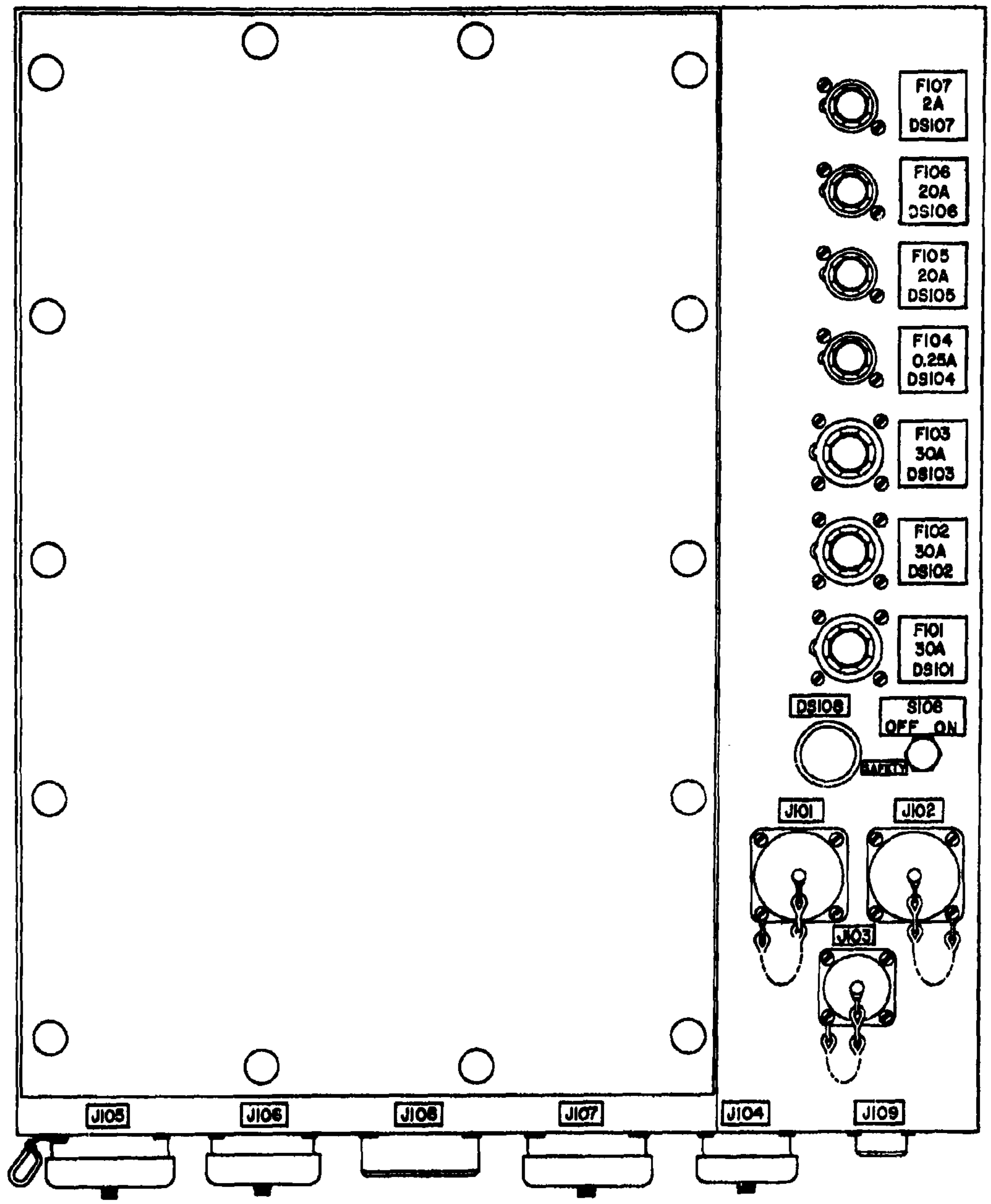

Figure 8. Front Panel of Contactor Box
(equipment serial Nos. 1 and 2)

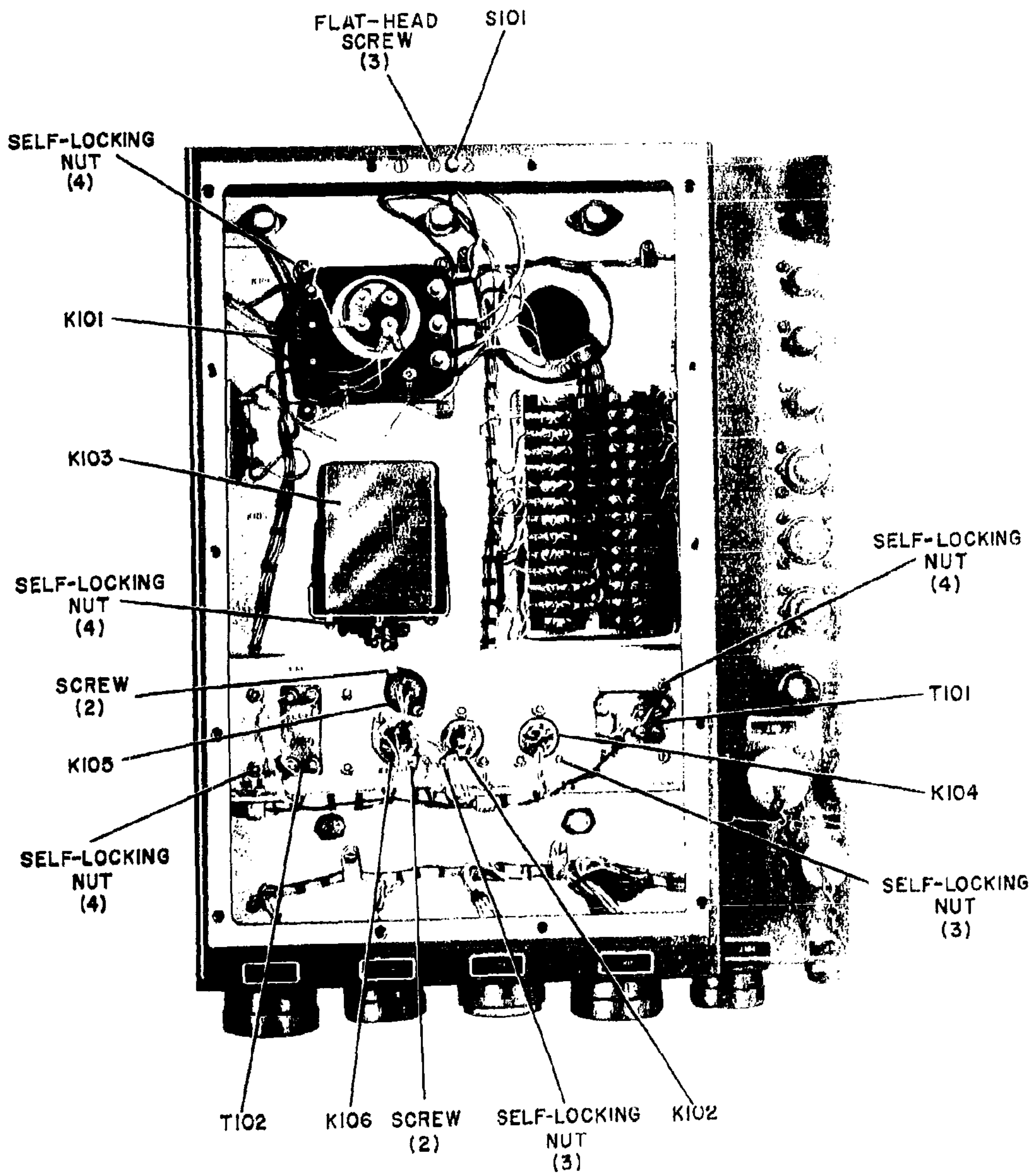

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Figure 9. Contactor Box (equipment serial Nos.1 and 2)

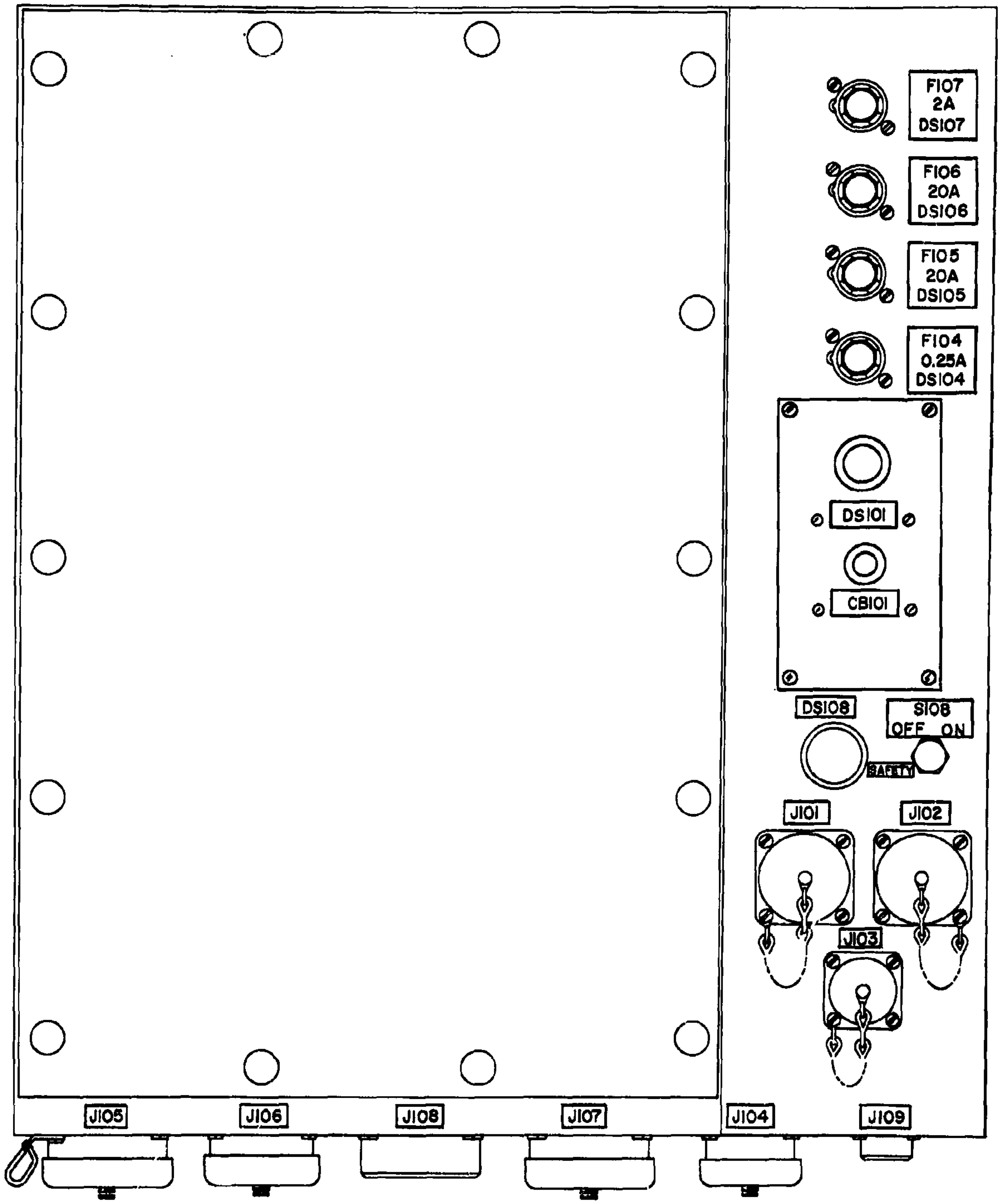

Figure 10. Front Panel of Contactor Box
(equipment serial Nos. 3 through 7)

66J1901

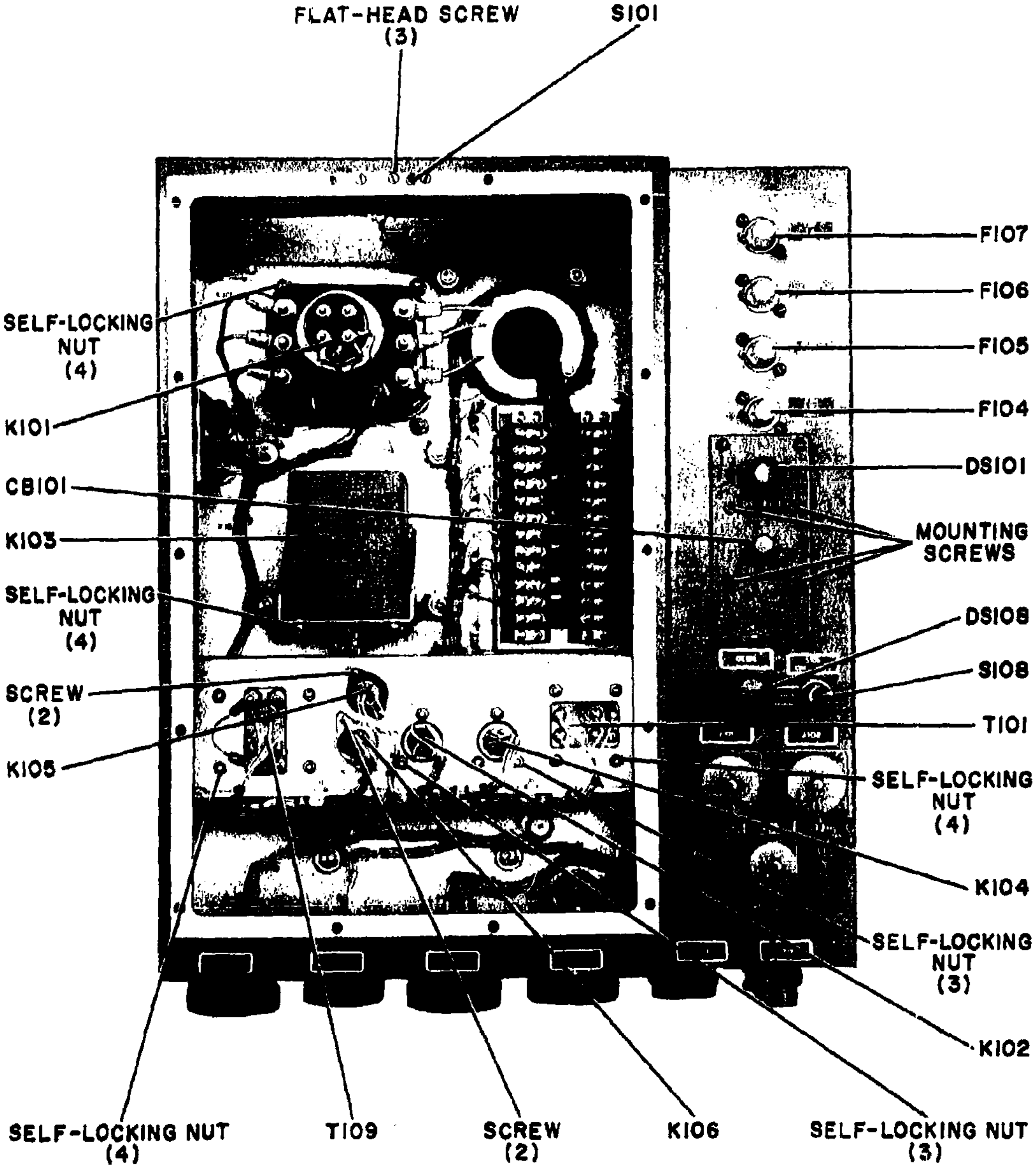

Figure 11. Contactor Box (equipment serial Nos. 3 through 7)

59B0910

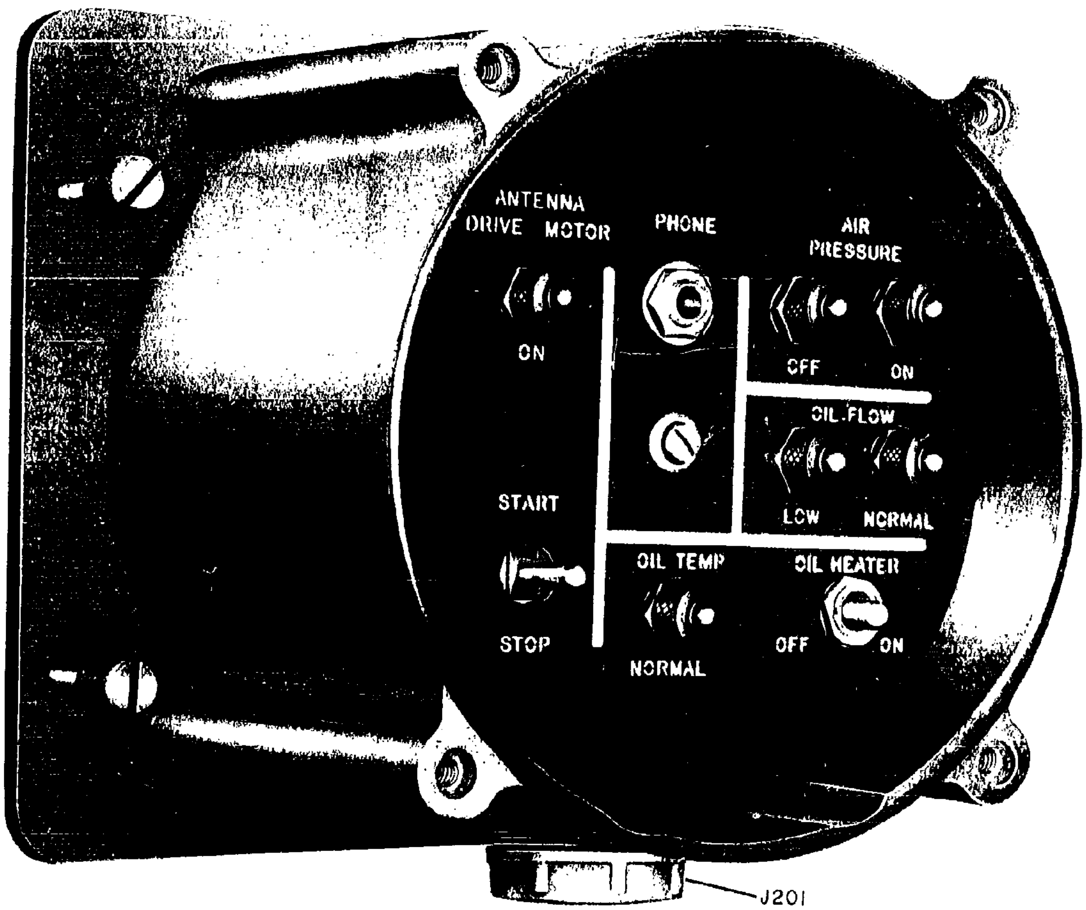

Figure 12. Antenna Control Box

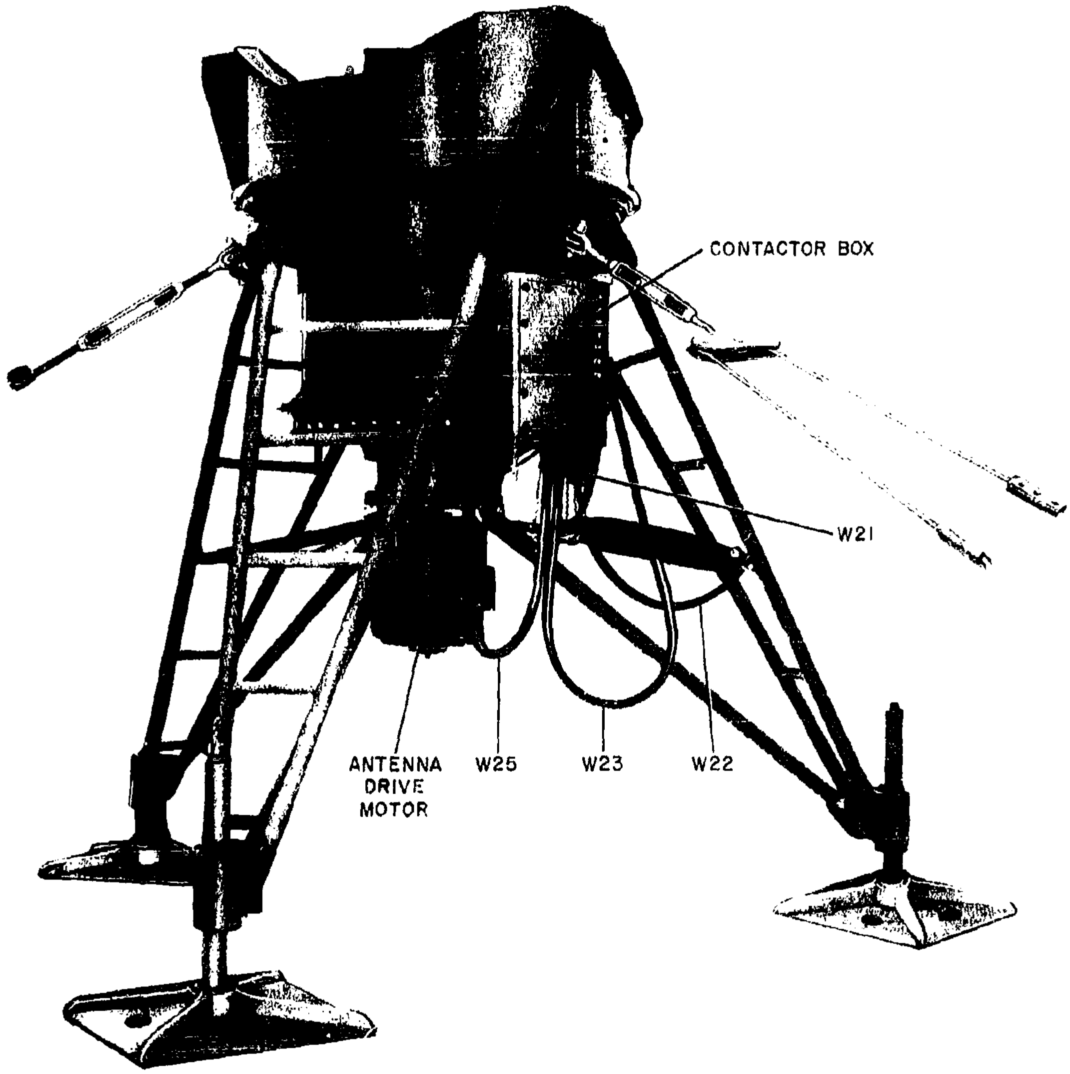

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Figure 13. Antenna Pedestal

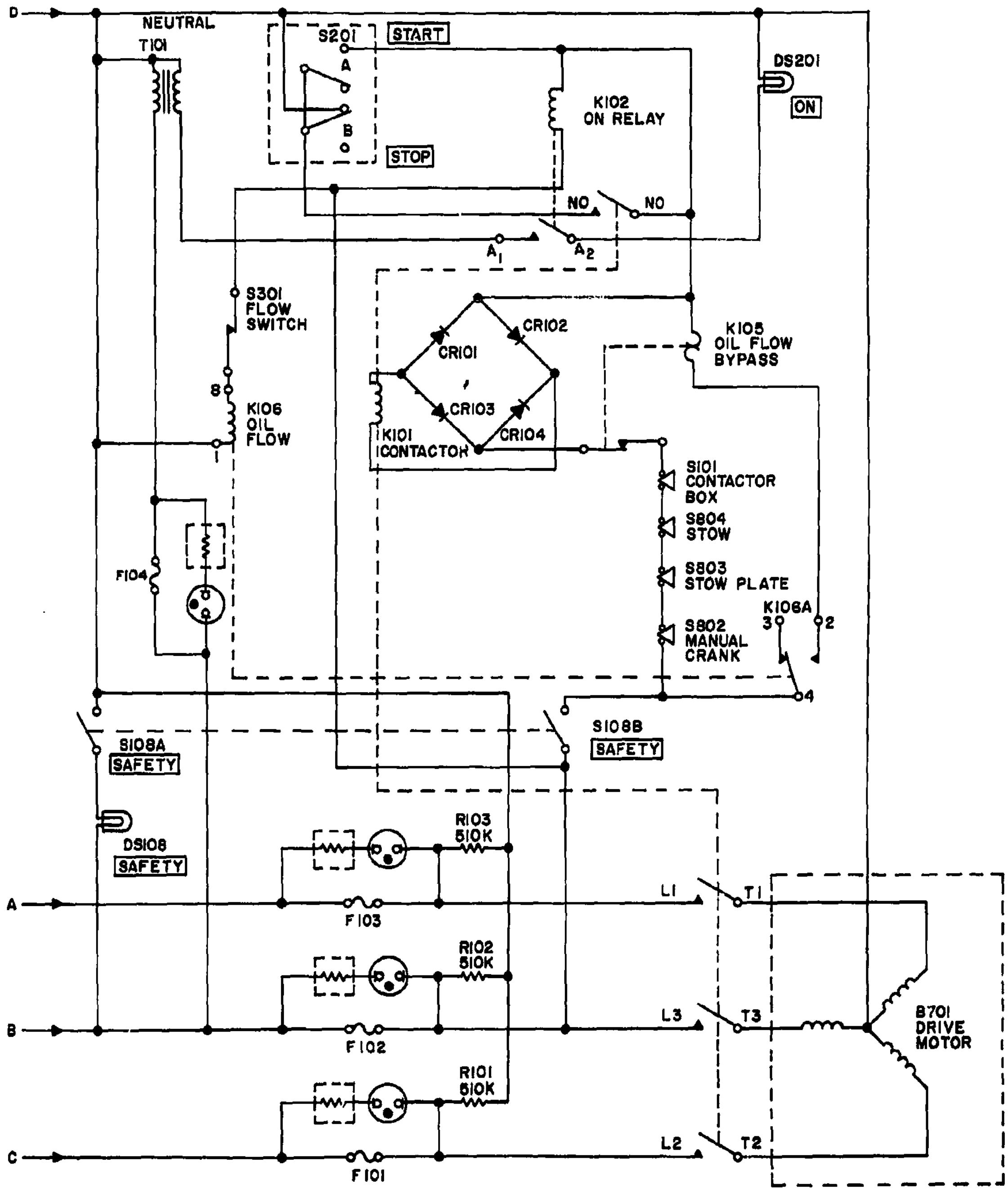


Figure 14. Control Circuits, Simplified Schematic (equipment serial Nos. 1 and 2)

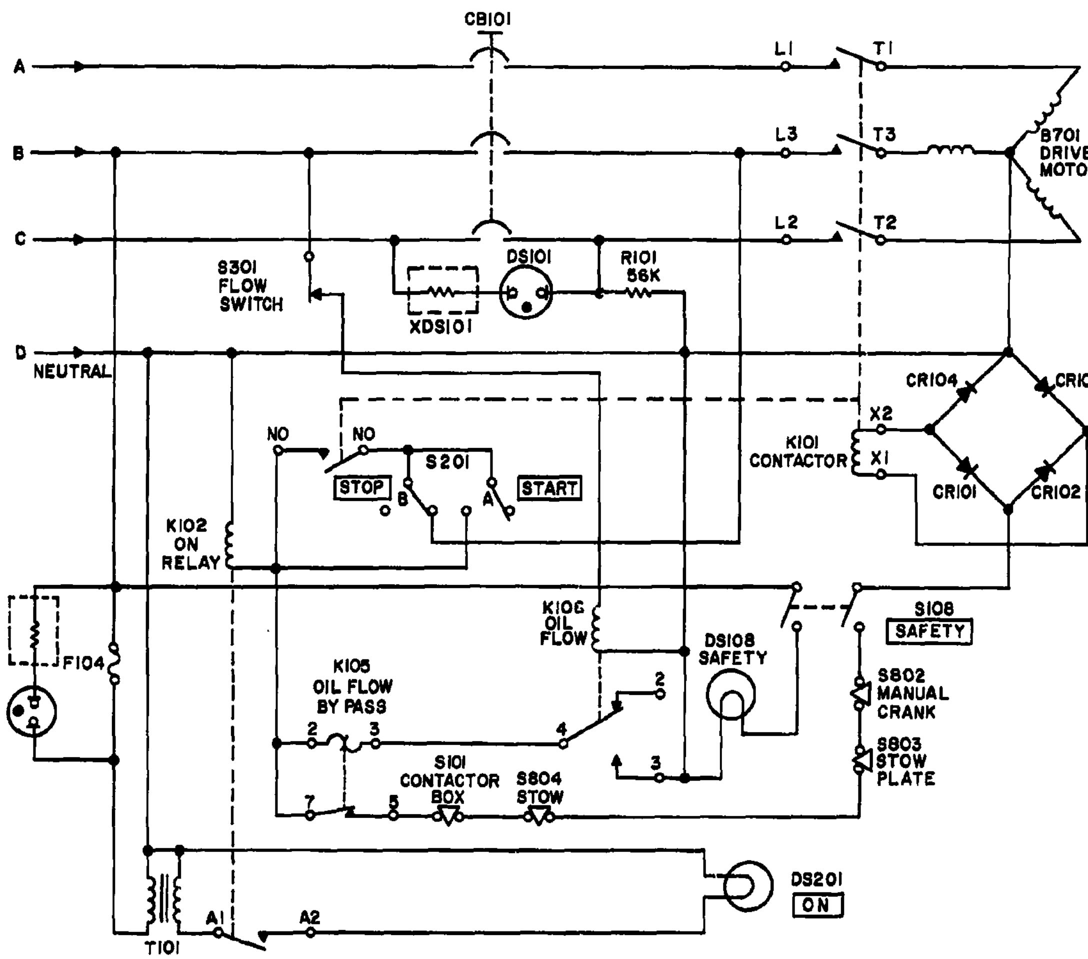


Figure 15. Control Circuits, Simplified Schematic (equipment serial Nos. 3 through 7)

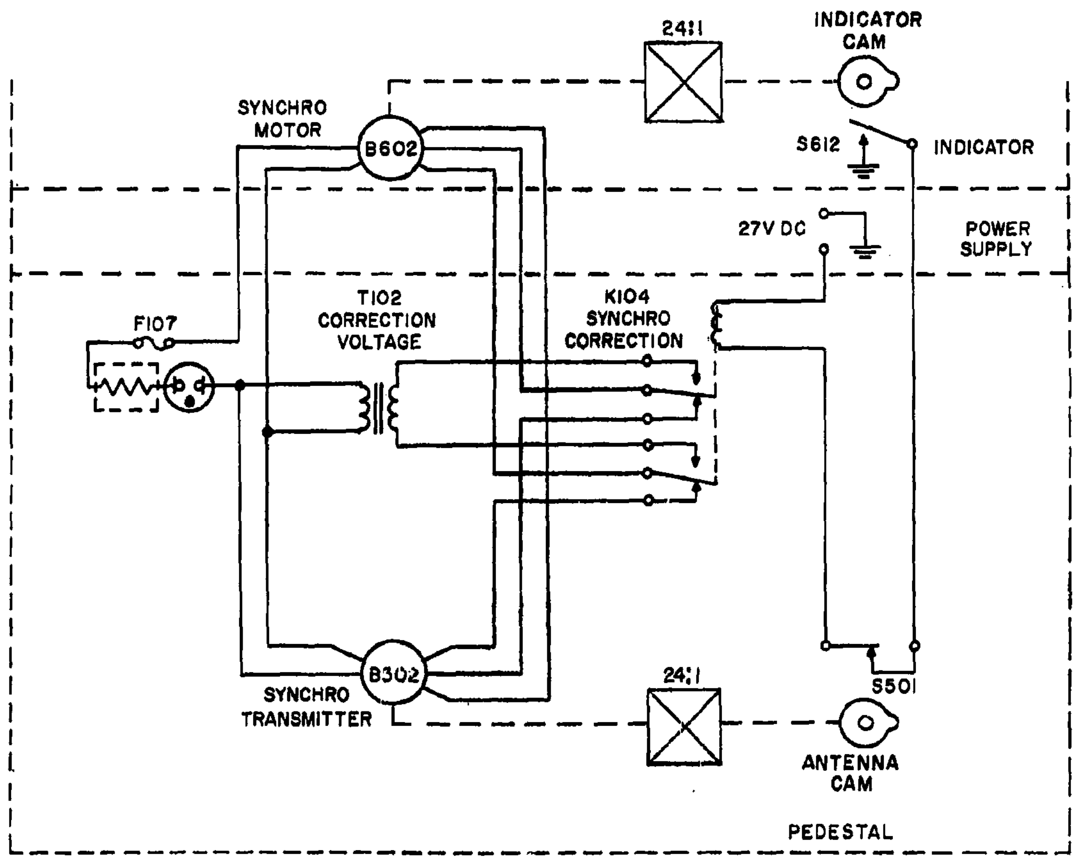


Figure 16. Synchro System, Simplified Schematic

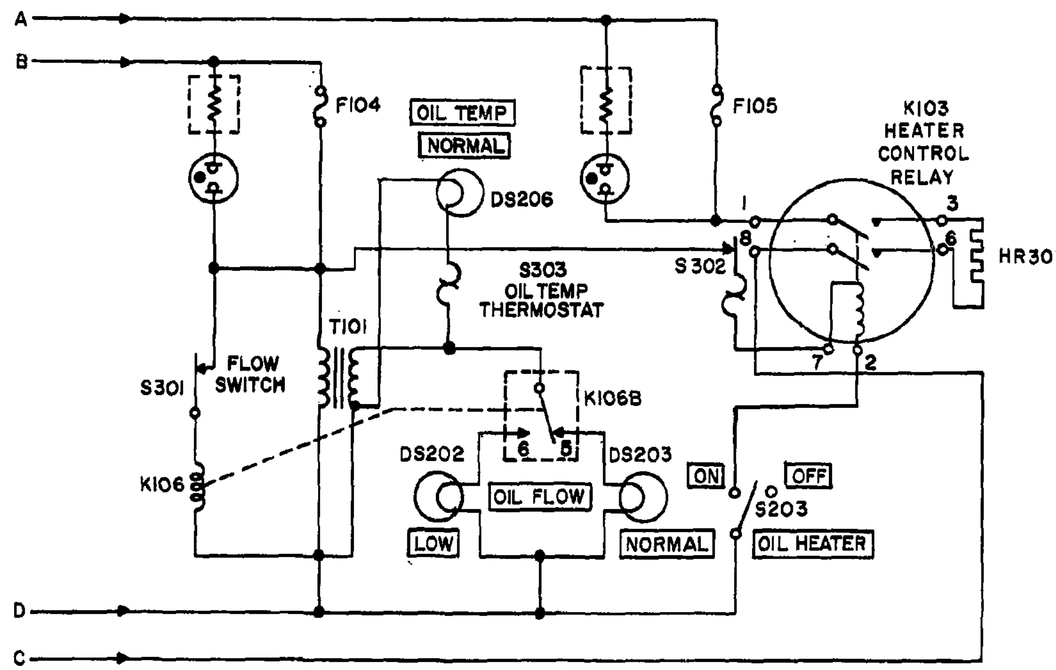


Figure 17. Lubrication Heating Control Circuits, Simplified Schematic (equipment serial Nos. 1 and 2)

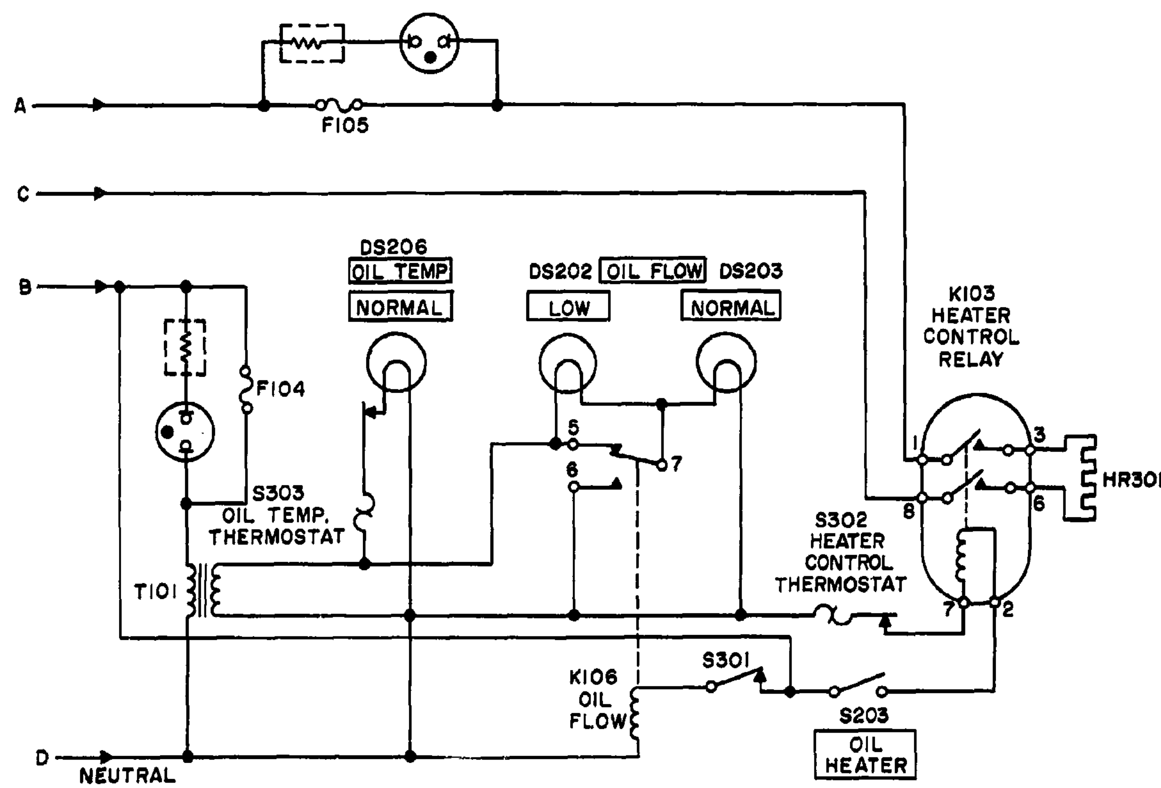


Figure 18. Lubrication Heating Control Circuits, Simplified Schematic (equipment serial Nos. 3 through 7)

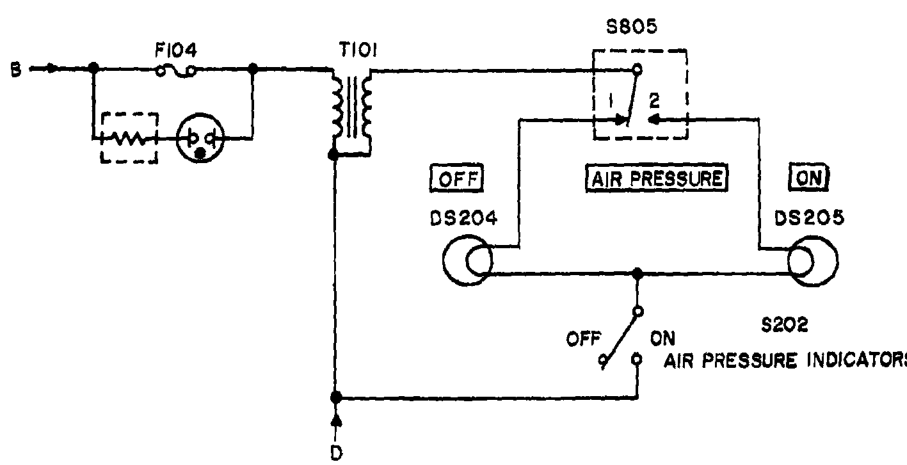


Figure 19. Pressure Control Circuit, Simplified Schematic (equipment serial Nos. 1 and 2)

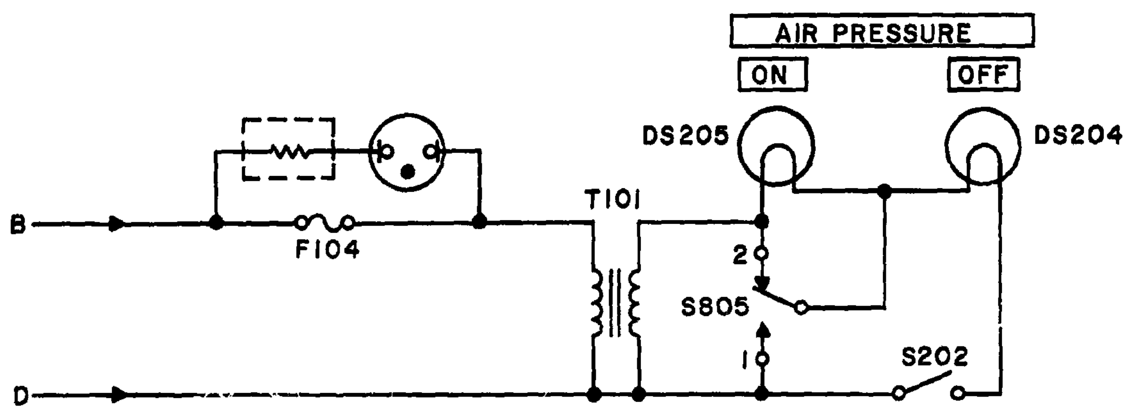


Figure 20. Pressure Control Circuit, Simplified Schematic (equipment serial Nos. 3 through 7)

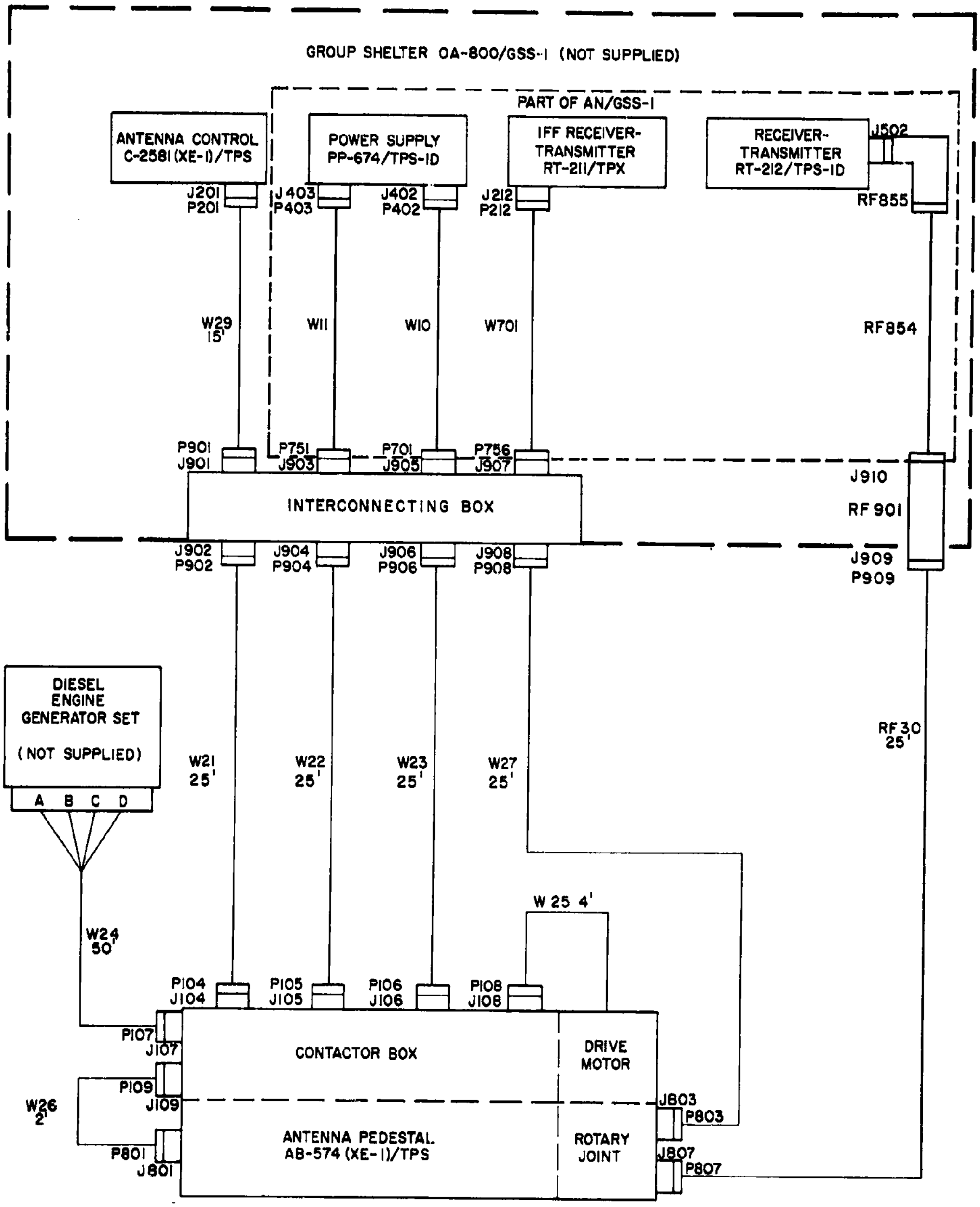
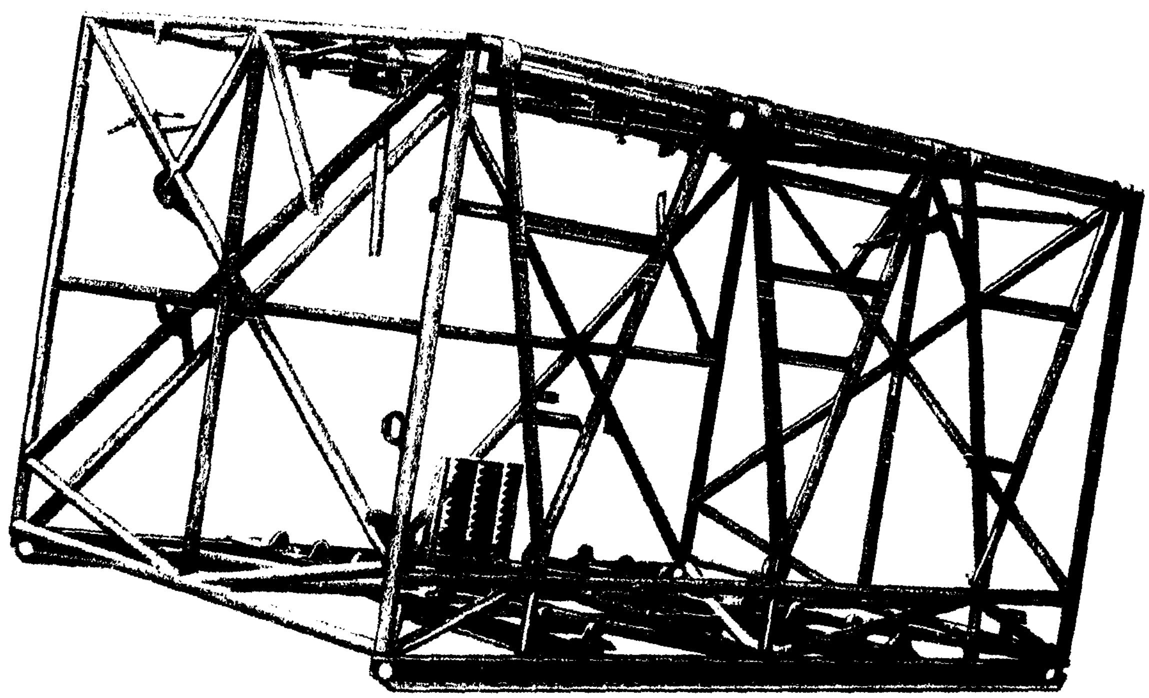
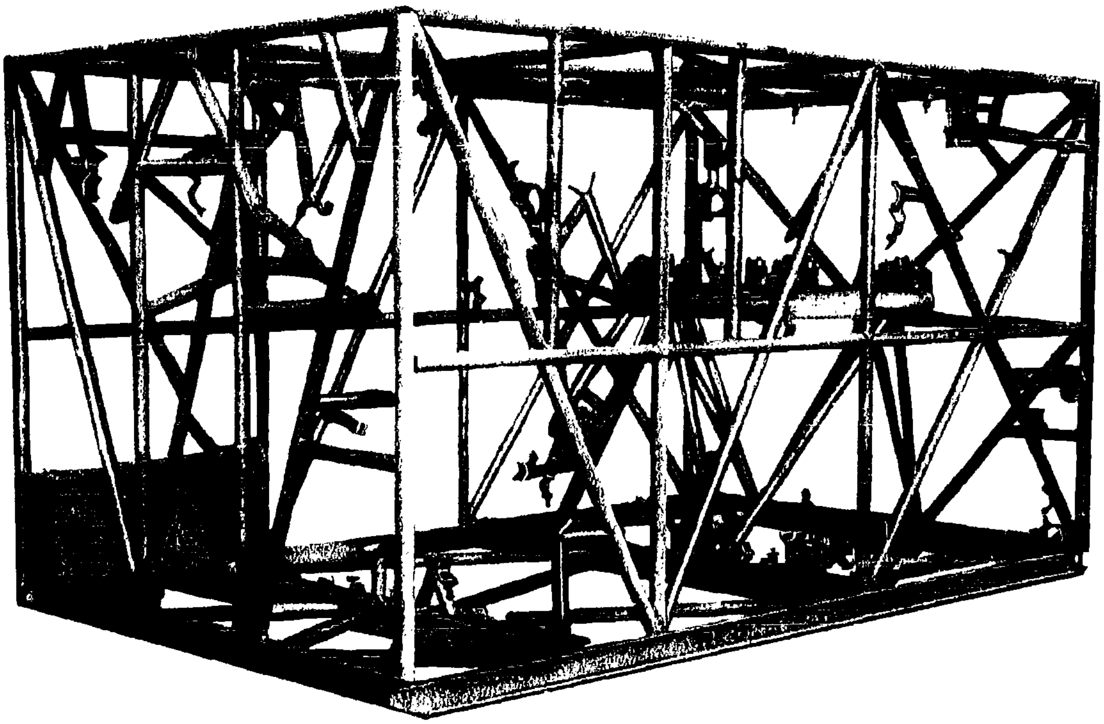


Figure 21. Interconnection Cabling Diagram



59K1309

Figure 22. Antenna Reflector Transit Case



59K1315

Figure 23. Antenna Pedestal Transit Case

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59L2601

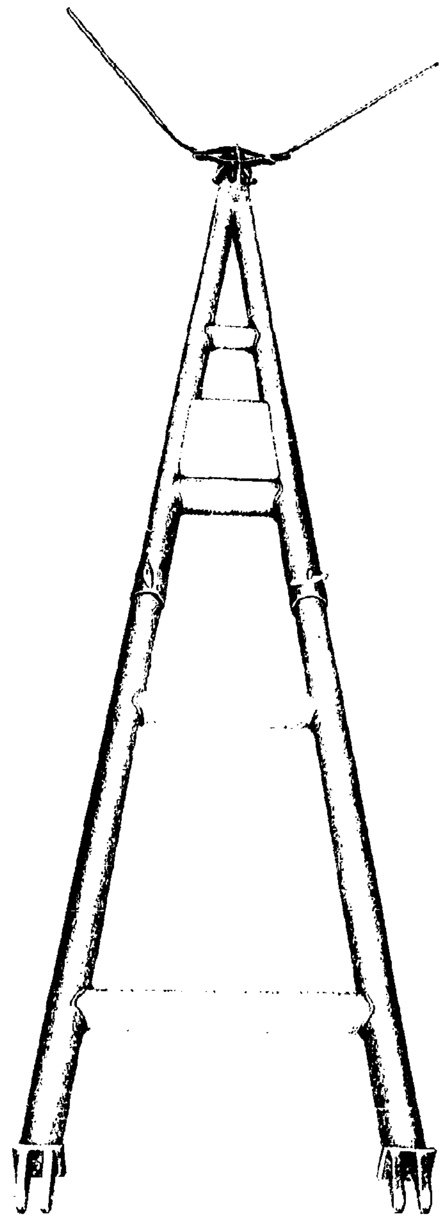


Figure 24. Auxiliary "A" Frame

P 22 AV 2 R

REPORT 10075

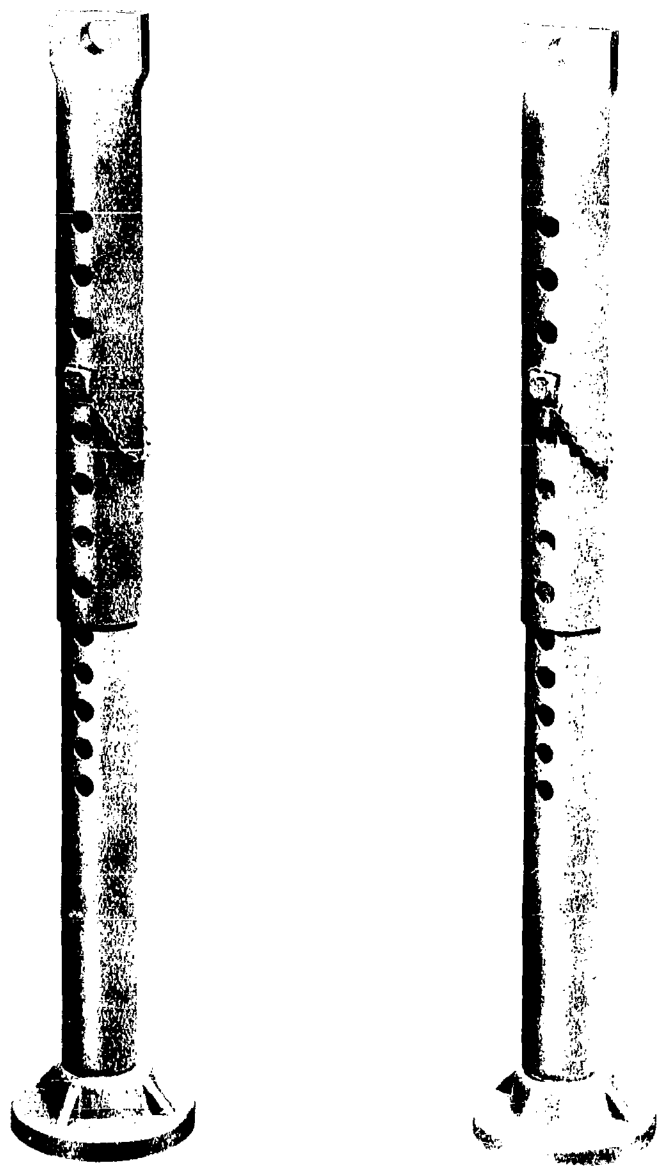


Figure 25. Bumper Jacks

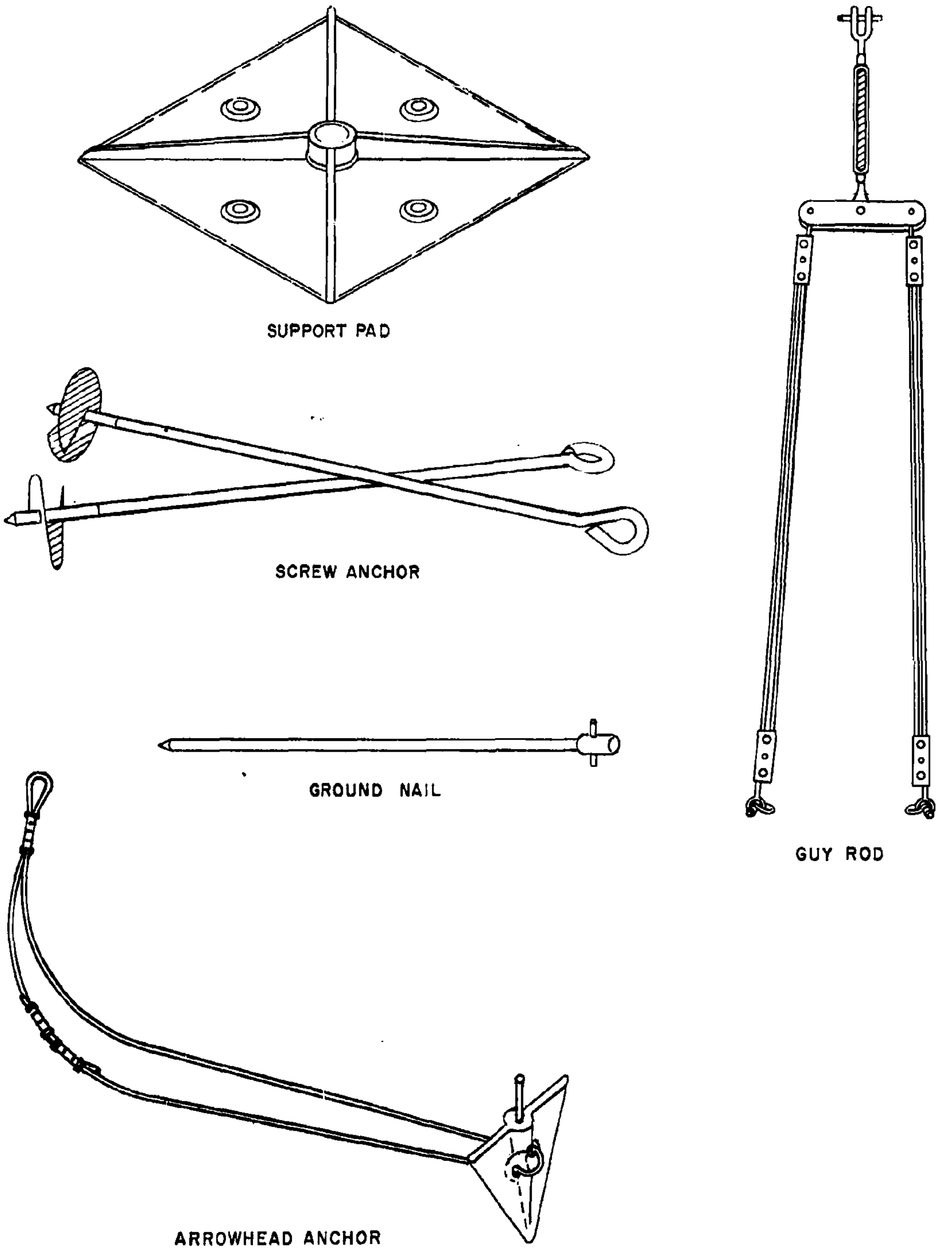
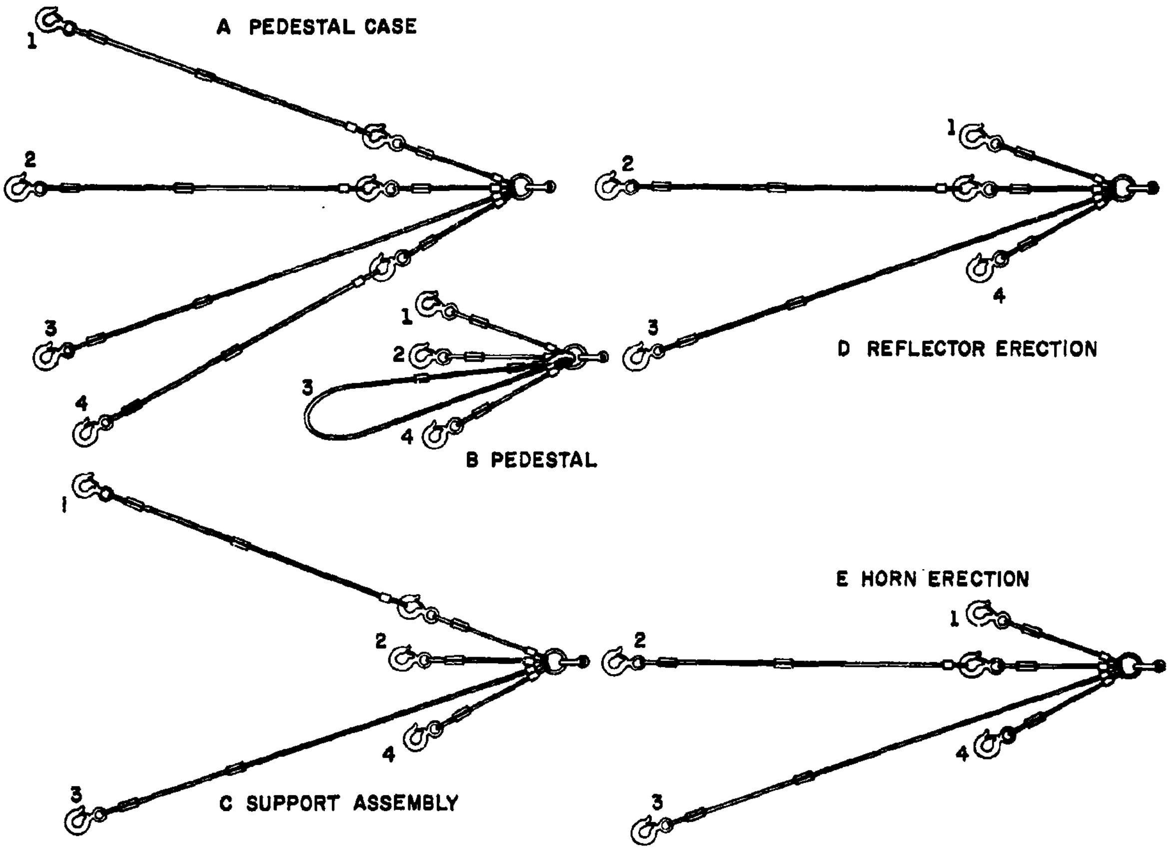


Figure 26. Miscellaneous Mounting Components



NOTES

1. NUMERALS ON ILLUSTRATION ARE USED FOR REFERENCE ONLY AND DO NOT APPEAR ON LIFT SLING

2. A LEGS 1,2,3 AND 4; ONE LEG TO EACH PICK-UP POINT ON PEDESTAL CASE.

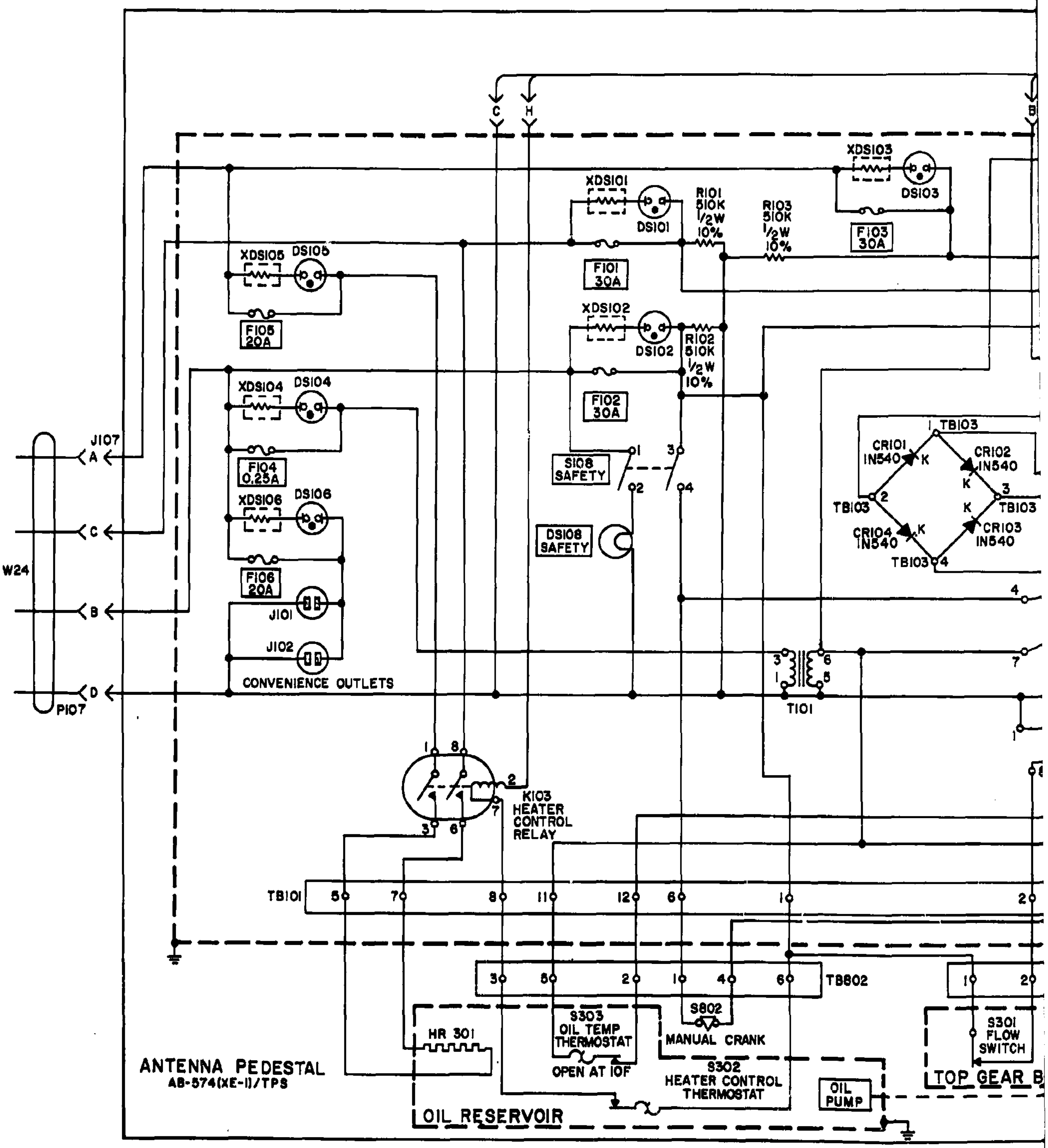
   B LEG 3 FOLDED TO "O" RING; LEGS 1 AND 4 TO PEDESTAL LIFTING EYES LEG 2 TO BAIL IRON BEHIND ROTARY JOINT.

   C LEGS 2 AND 4 NOT USED; LEGS 1 AND 3 TO PICK-UP POINTS ON SUPPORT ASSEMBLY.

   D LEGS 1 AND 4 TO EYES ON AUXILIARY "A" FRAME; LEGS 2 AND 3 TO PICK-UP POINTS ON FACE OF REFLECTOR.

   E LEGS 2 AND 3 TO PICK-UP POINTS AT WIDE END OF FEED SUPPORT; LEG 1 TO PICK-UP POINT AT APEX OF FEED SUPPORT; LEG 4 NOT USED.

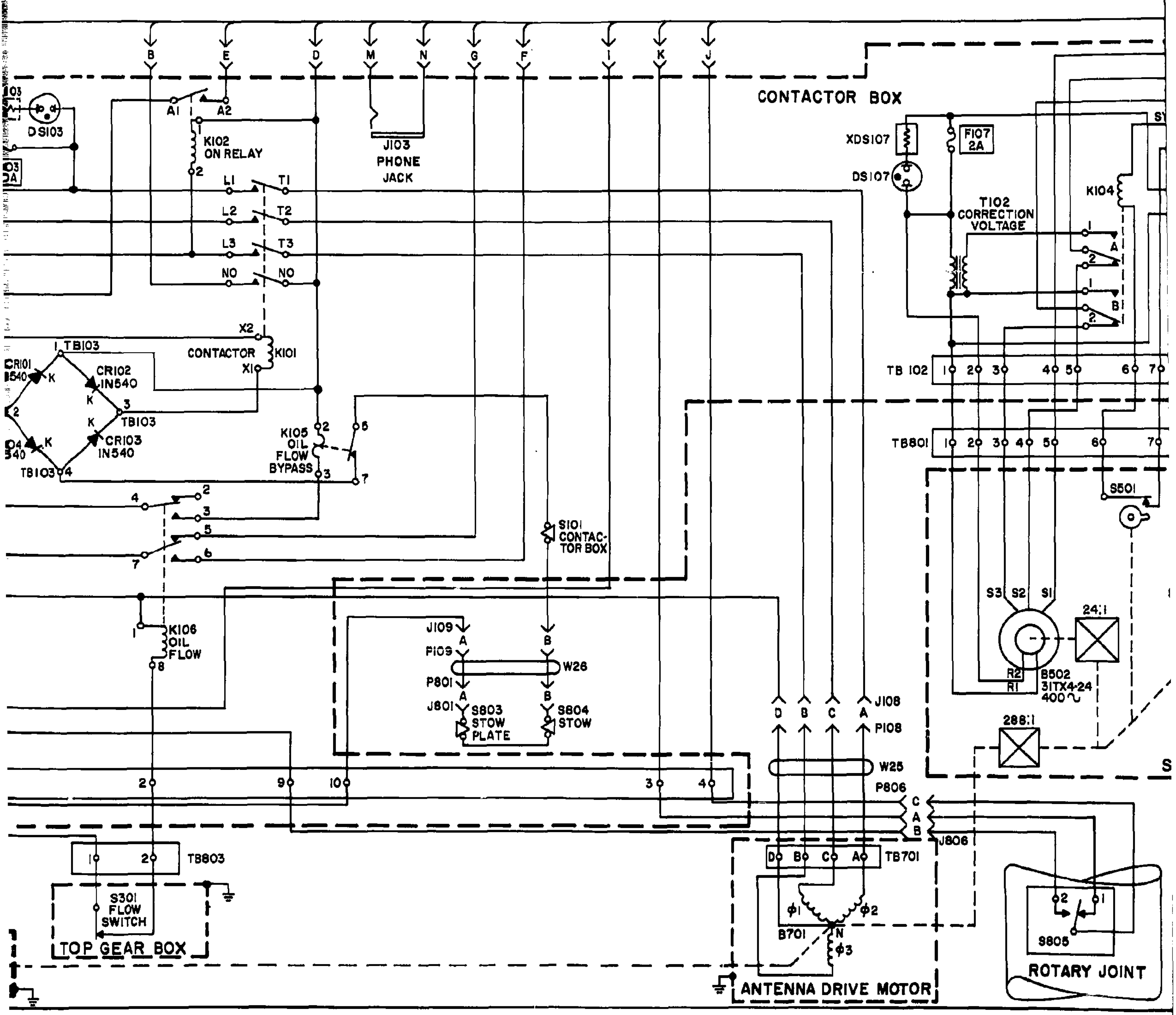
Figure 27. Lift Sling Configurations



W24
J107
P107
A
B
C
D
H
XDS101
DS101
XDS102
DS102
XDS103
DS103
XDS104
DS104
XDS105
DS105
XDS106
DS106
R101 510K 1/2W 10%
R102 510K 1/2W 10%
R103 510K 1/2W 10%
F101 30A
F102 30A
F103 30A
F104 0.25A
F105 20A
F106 20A
J101
J102
CONVENIENCE OUTLETS
S108 SAFETY
DS108 SAFETY
TB103
CR101 IN540
CR102 IN540
CR103 IN540
CR104 IN540
K
T101
K103 HEATER CONTROL RELAY
TB101
TB802
HR 301
S303 OIL TEMP THERMOSTAT
OPEN AT 10F
S802
MANUAL CRANK
S302 HEATER CONTROL THERMOSTAT
OIL PUMP
OIL RESERVOIR
S301 FLOW SWITCH
TOP GEAR B
ANTENNA PEDESTAL
AB-574(XE-1)/TPS

1

NOTE: CABLES W21, W22, W23 HAVE IDENTICAL CONNECTIONS ON EACH END



Figure

2

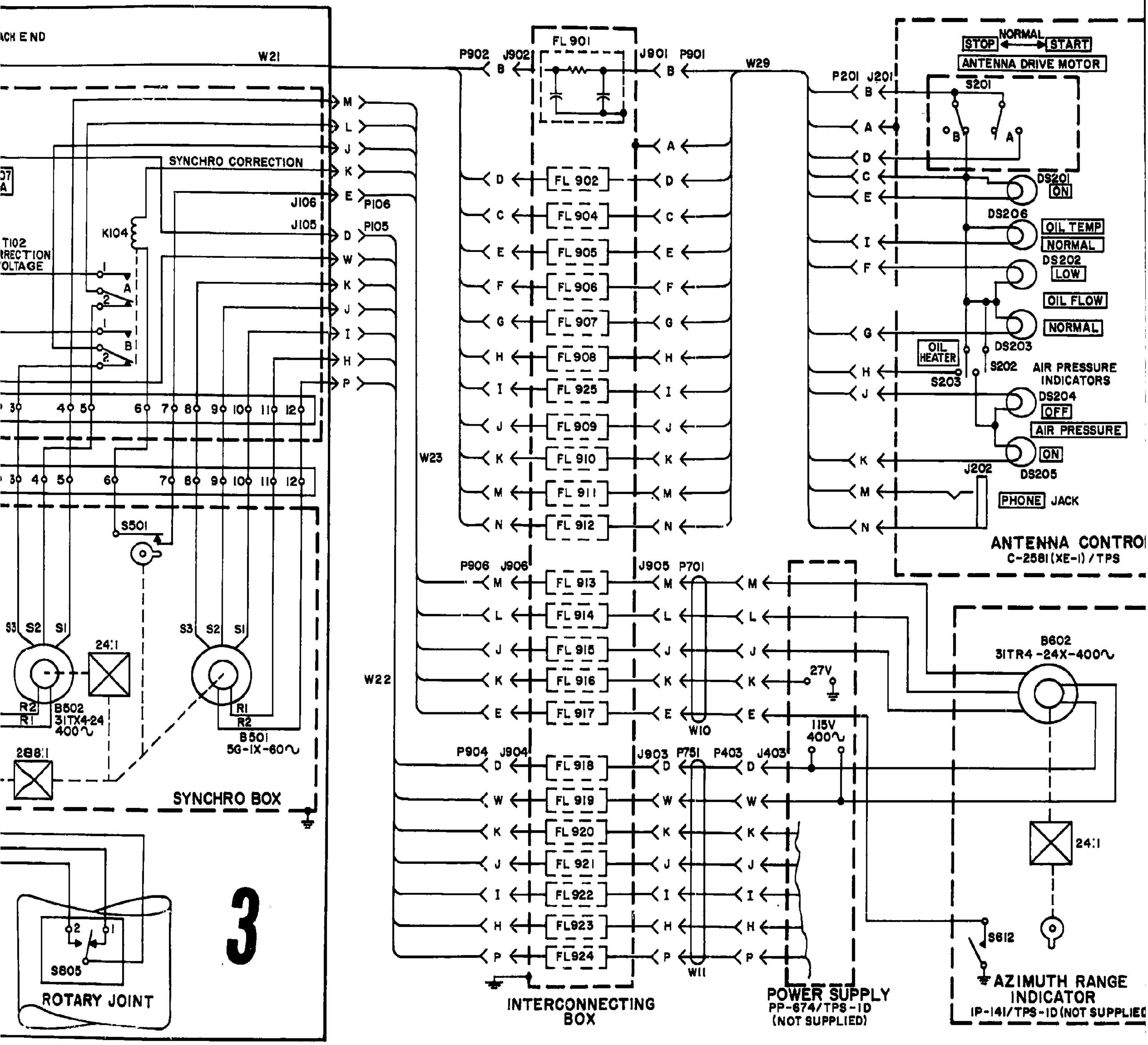
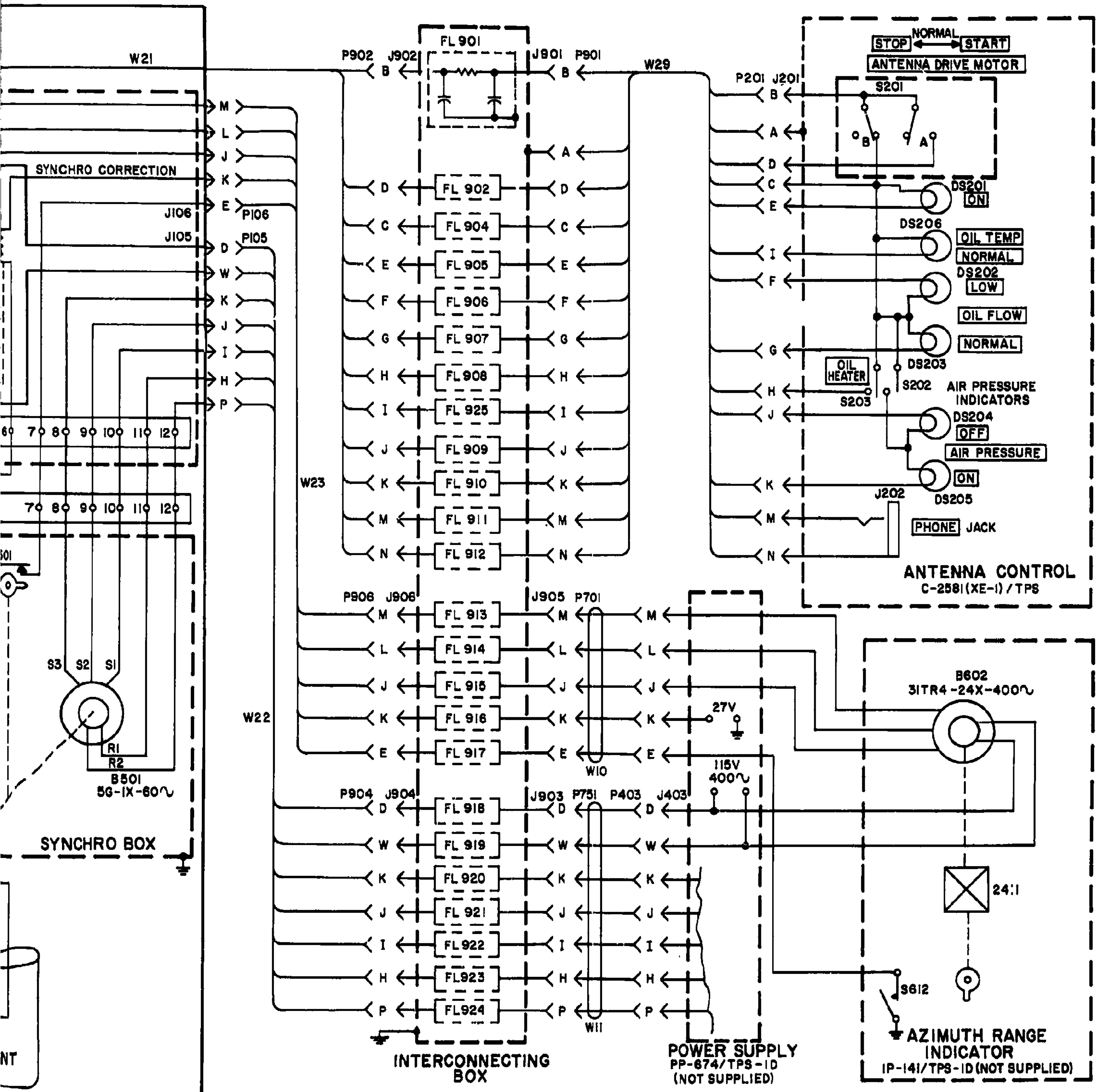
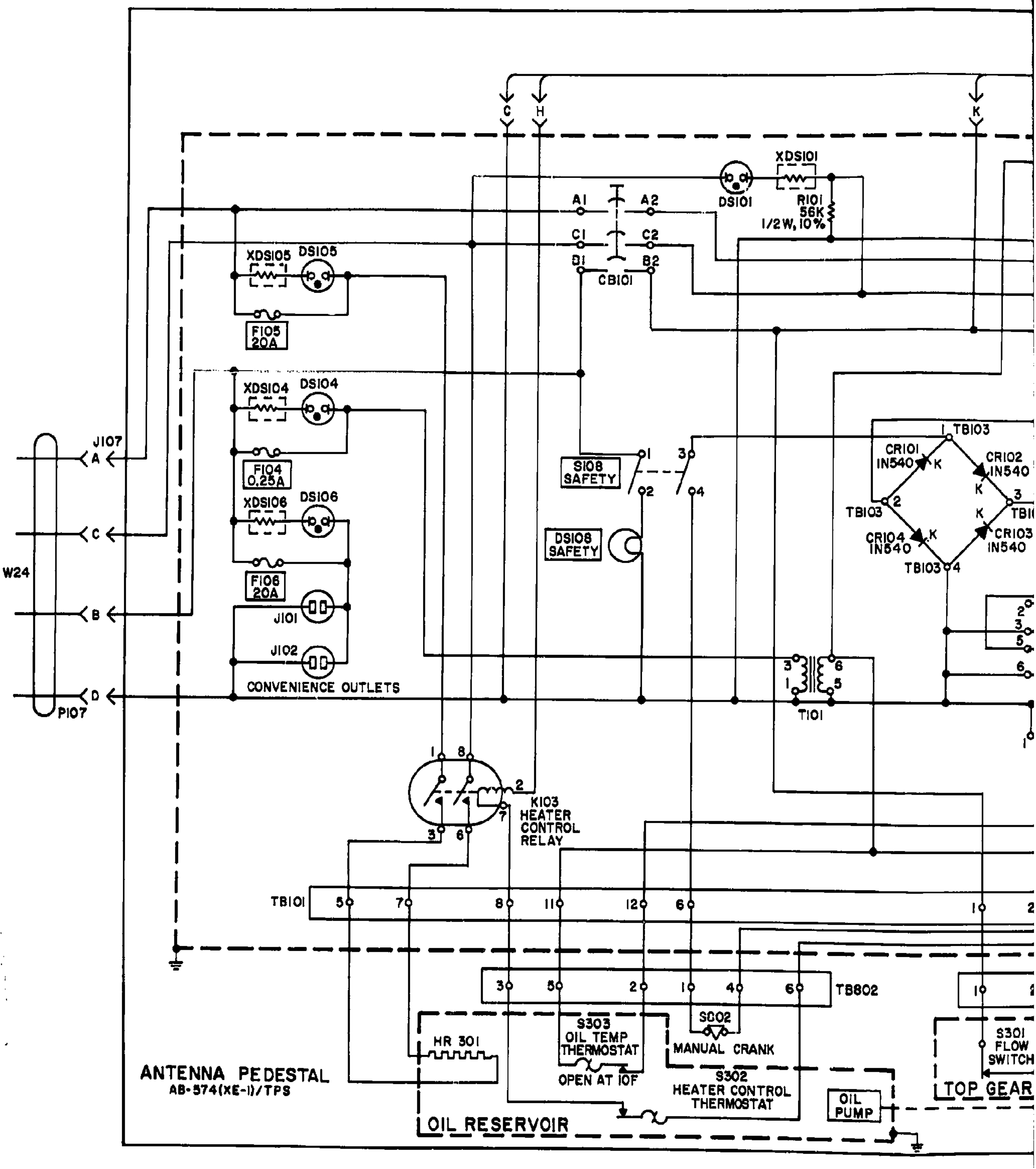


Figure 28. Overall Electrical Schematic Diagram
(equipment serial Nos. 1 and 2)



gure 28. Overall Electrical Schematic Diagram
(equipment serial Nos. 1 and 2)

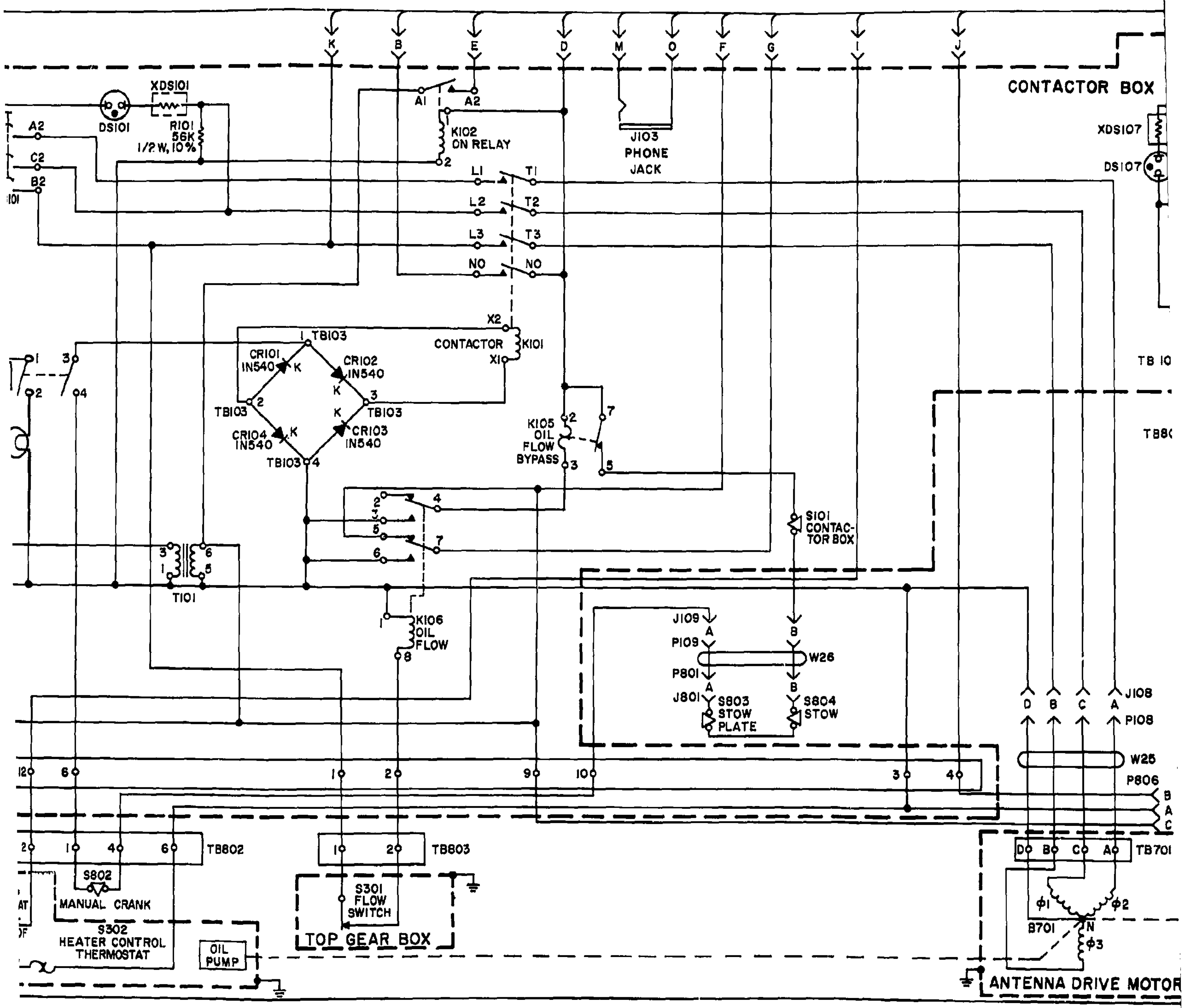
4



C
H
K
XDS101
DS101
R101
56K
1/2W, 10%
A1
A2
C1
C2
D1
B2
CB101
XDS105
DS105
F105
20A
XDS104
DS104
F104
0.25A
XDS106
DS106
F106
20A
J101
J102
CONVENIENCE OUTLETS
J107
W24
P107
S108
SAFETY
DS108
SAFETY
TB103
CR101
IN540
CR102
IN540
CR104
IN540
CR103
IN540
T101
K103
HEATER
CONTROL
RELAY
TB101
TB802
S303
OIL TEMP
THERMOSTAT
OPEN AT 10F
S002
MANUAL CRANK
HR 301
S302
HEATER CONTROL
THERMOSTAT
OIL PUMP
OIL RESERVOIR
S301
FLOW
SWITCH
TOP GEAR
ANTENNA PEDESTAL
AB-574(XE-1)/TPS

1

NOTE: CABLES W21, W22, W23 HAVE IDENTICAL CONNECT



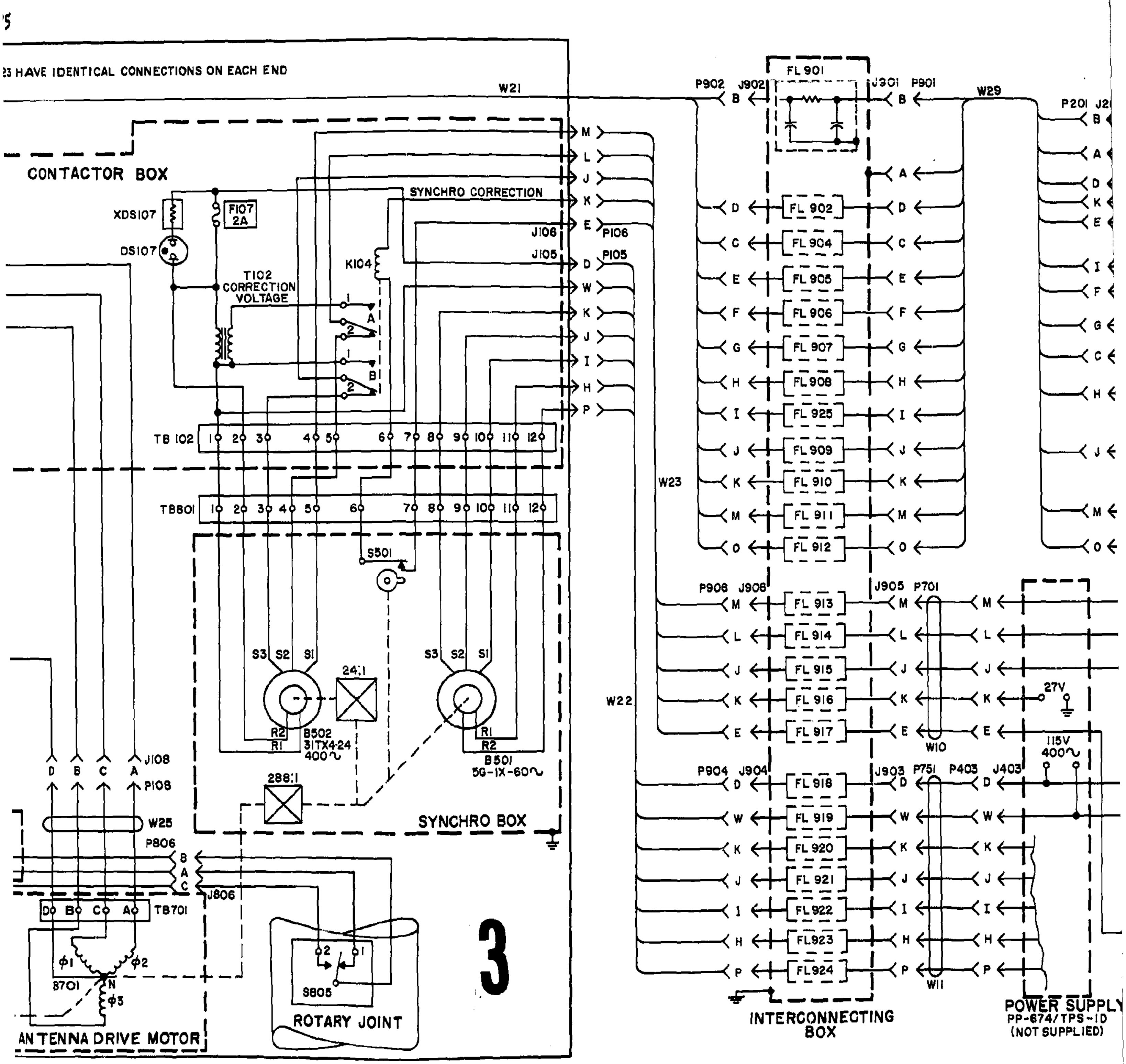


Figure 29. Overall Electrical Schematic Diagram
(equipment serial Nos. 3 through 7)

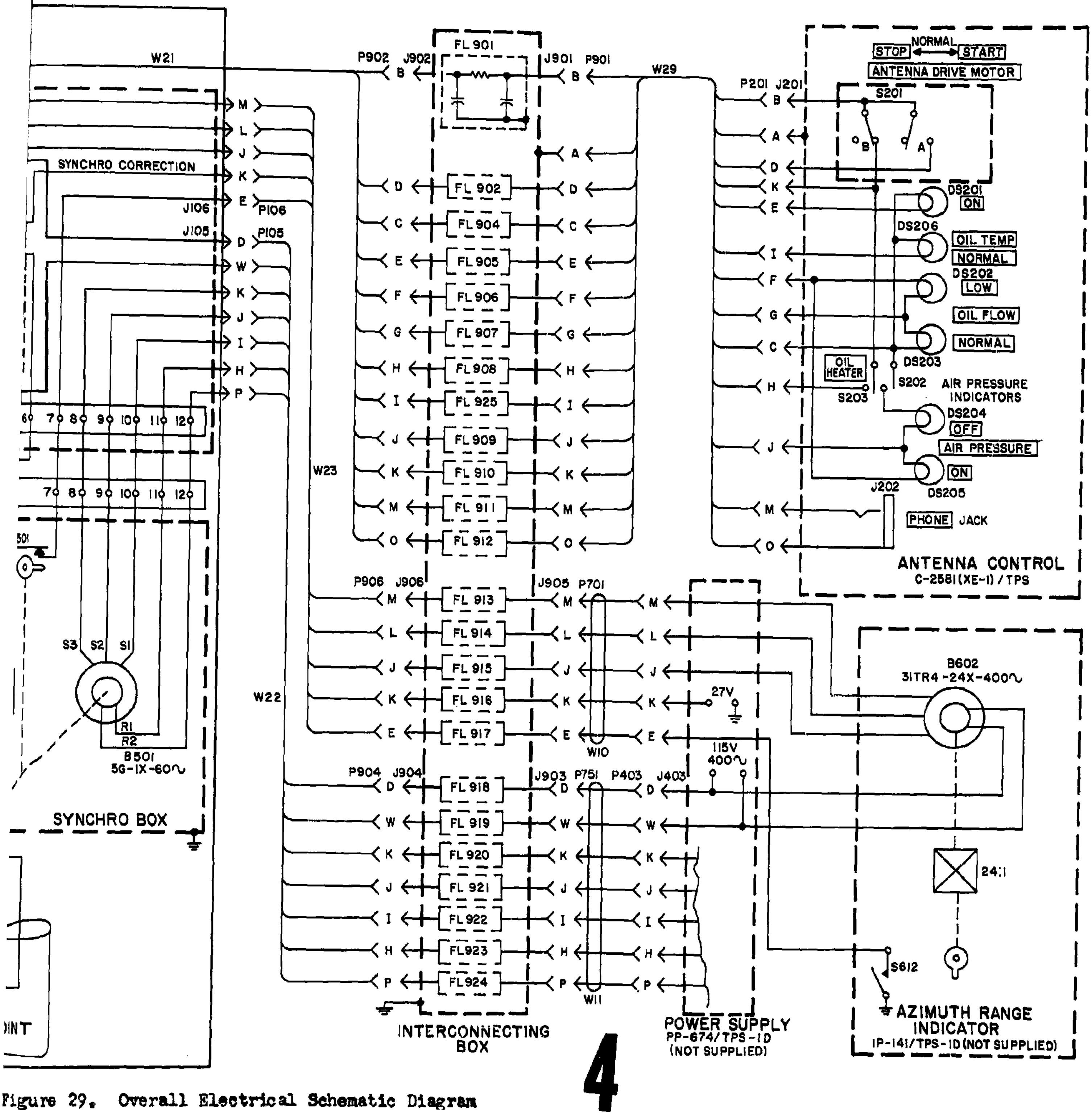


Figure 29. Overall Electrical Schematic Diagram (equipment serial Nos. 3 through 7)

ADDENDUM A

VOLTAGE STANDING WAVE RATIO MEASUREMENTS

Figures A1 and A2 show the VSWR results of the first model of final design, figure A3 (block diagram) illustrates the test hook-up for the VSWR measurements in 2, 3, and 4 below. The search radar VSWR measurements (figure A1, paragraphs 2 and 3 below) were taken over a frequency range of 1250 to 1350 megacycles. The IFF bands VSWR measurements (figure A2, paragraph 4 below) were taken over frequency ranges of 1000 to 1040 megacycles and 1080 to 1120 megacycles.

1. TEST EQUIPMENT

The signal generator used for the VSWR measurements is a Hazeltine Model 1050B, modified to generate frequencies to 1350 megacycles. A Hewlett Packard Standing Wave Meter, supplied from the slotted line (see figure A3) by a crystal diode detector, provided the indication.

2. ROTARY JOINT INPUT

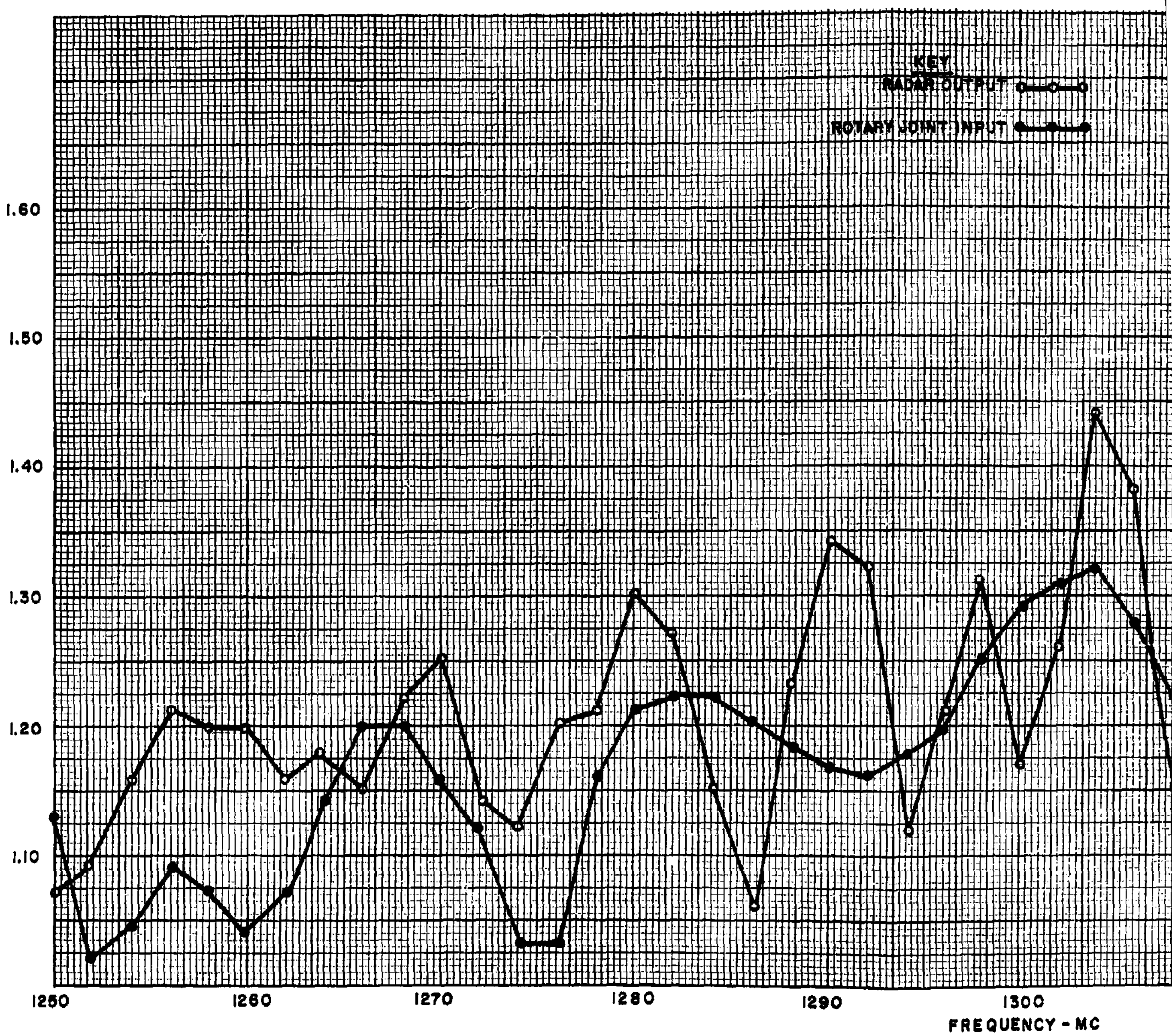
The VSWR measured at the input to the rotary joint is shown in figure A1. The maximum VSWR measured was 1.32 at 1304 megacycles.

3. RADAR TRANSMITTER OUTPUT

The VSWR as seen at the output of the radar transmitter (see figure A1) was found by measuring the VSWR into radar cable RF30 (see figure 2). The maximum VSWR measured was 1.45 at 1338 megacycles.

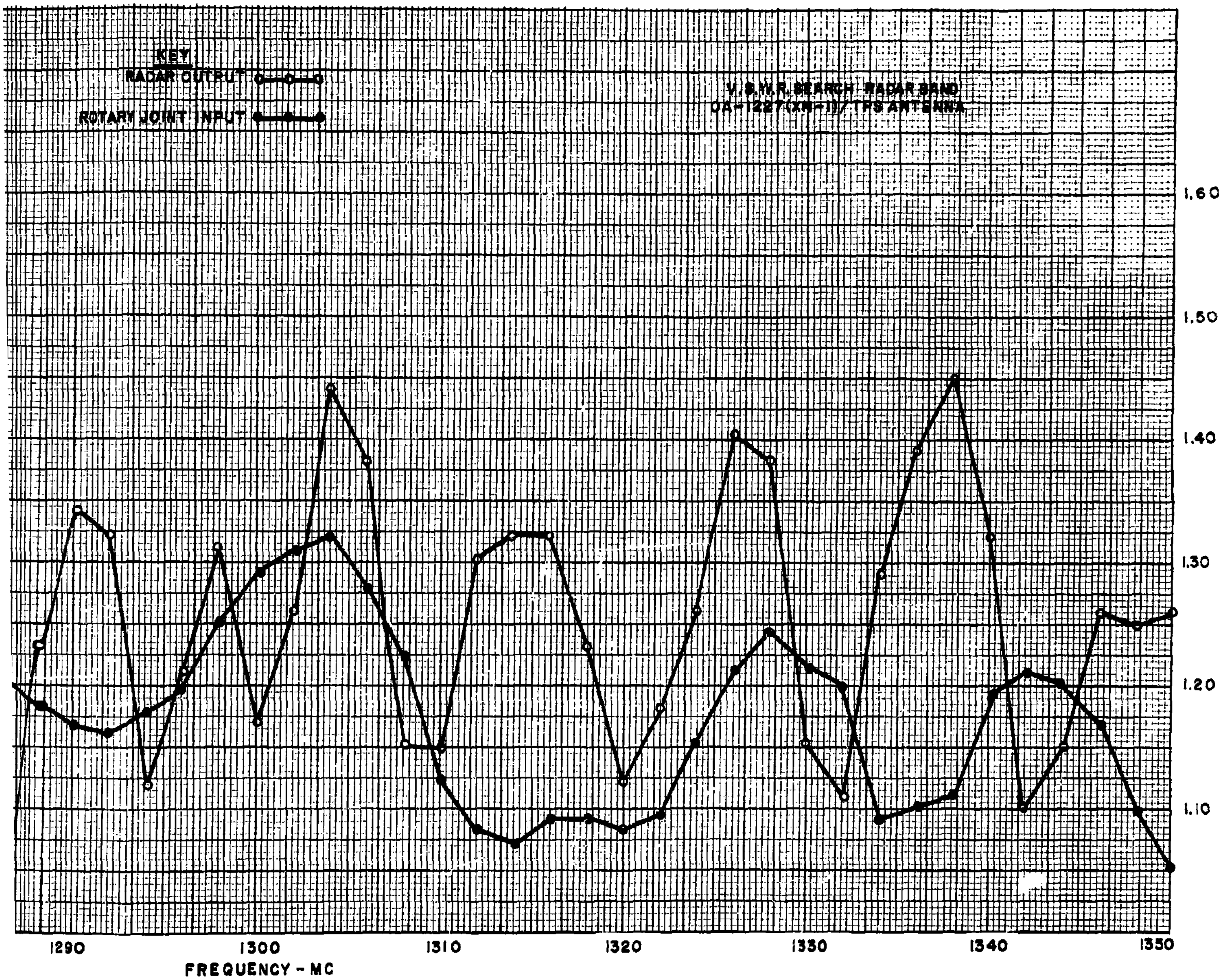
4. IFF OUTPUT

The VSWR measurements taken at the output of the IFF system, are shown in figure A2. The maximum VSWR readings were 1.55 at 1024 megacycles and 1.73 at 1110 megacycles.



1

Figure A1. Search Ra



2

Figure A1. Search Radar System, VSWR Results

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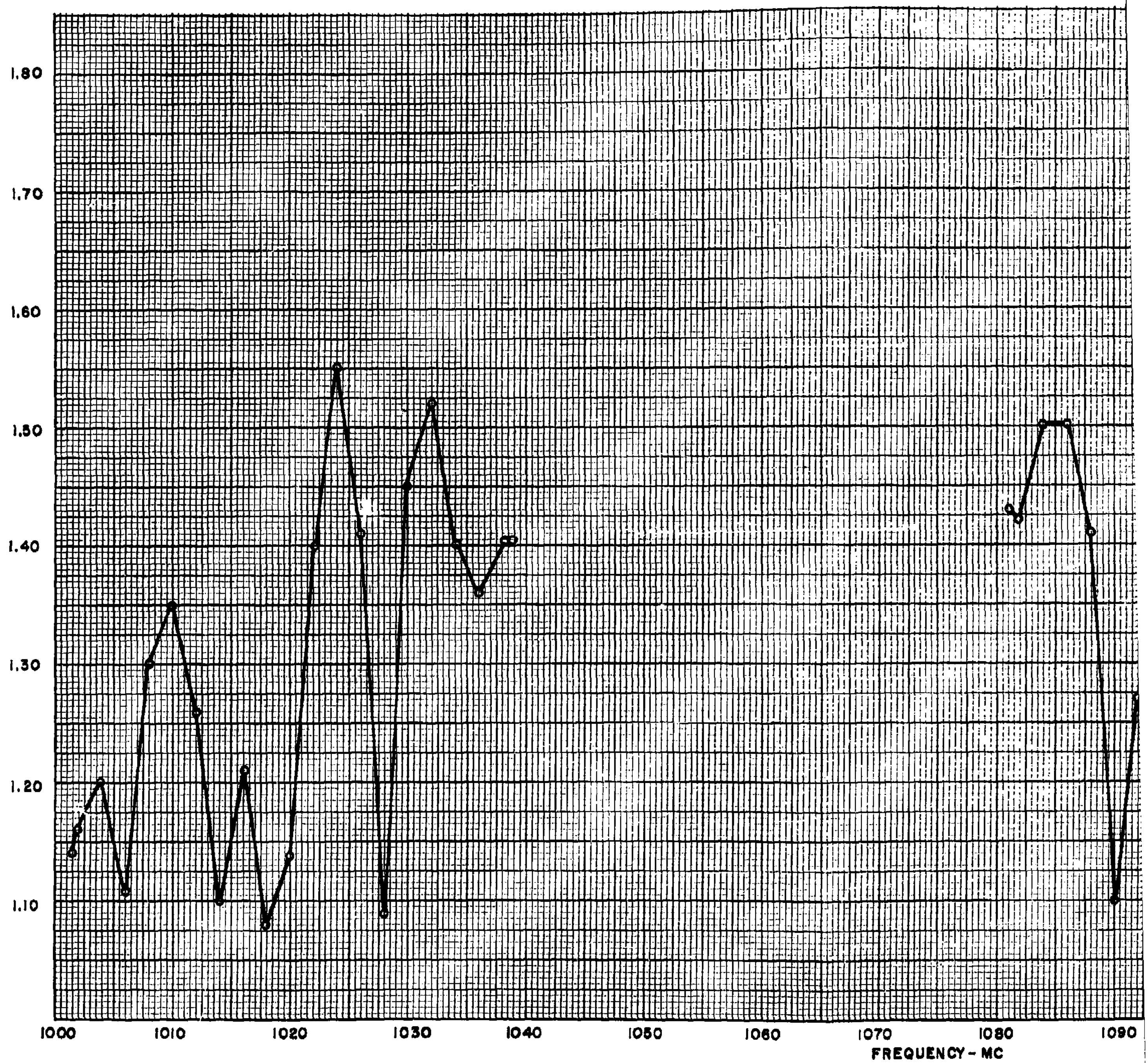

Figure A2. IFF System, VSW

HAZELTINE ELECTRONICS DIVISION
HAZELTINE CORPORATION

Appendix A

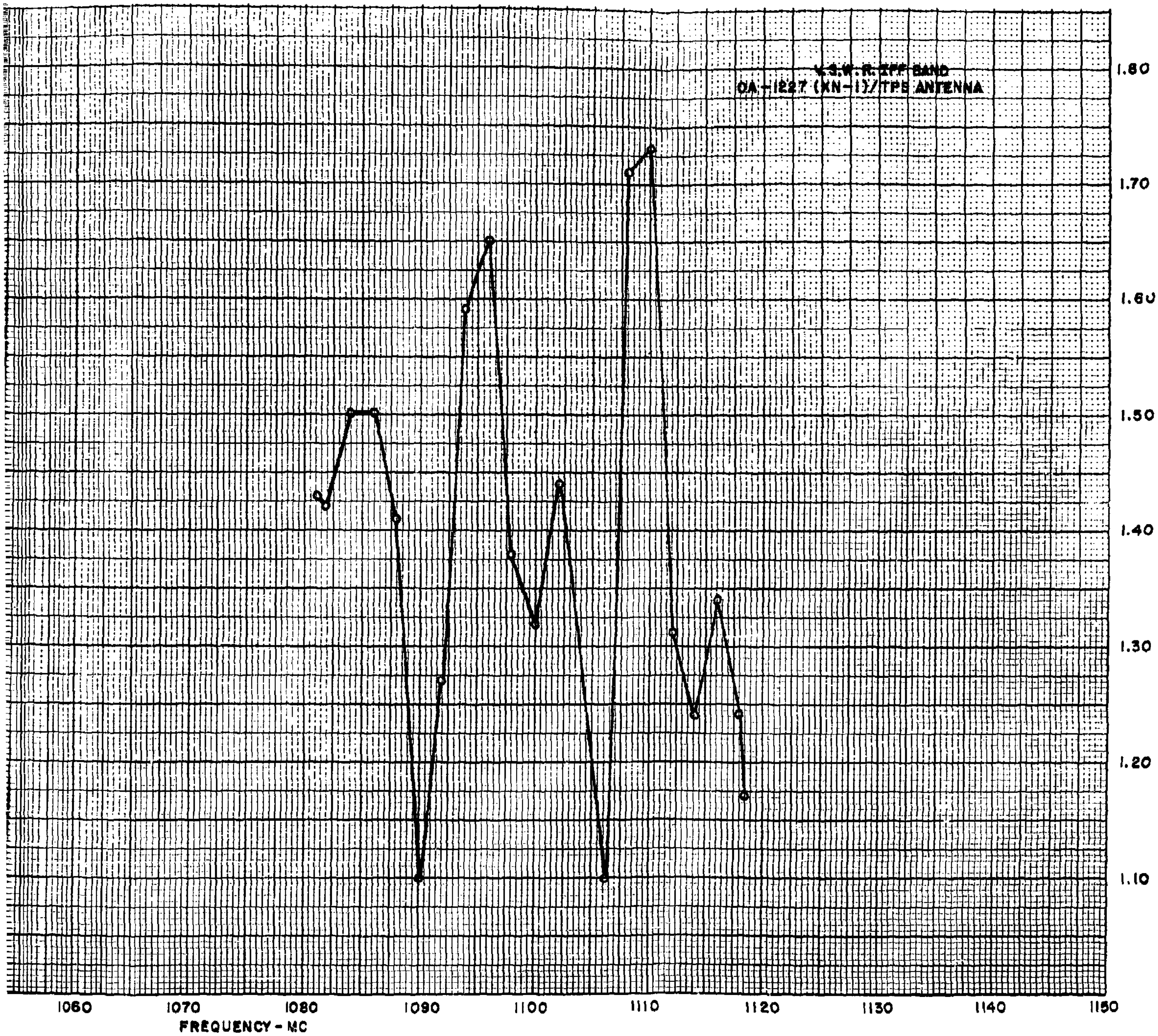

Figure A2. IFF System, VSWR Results

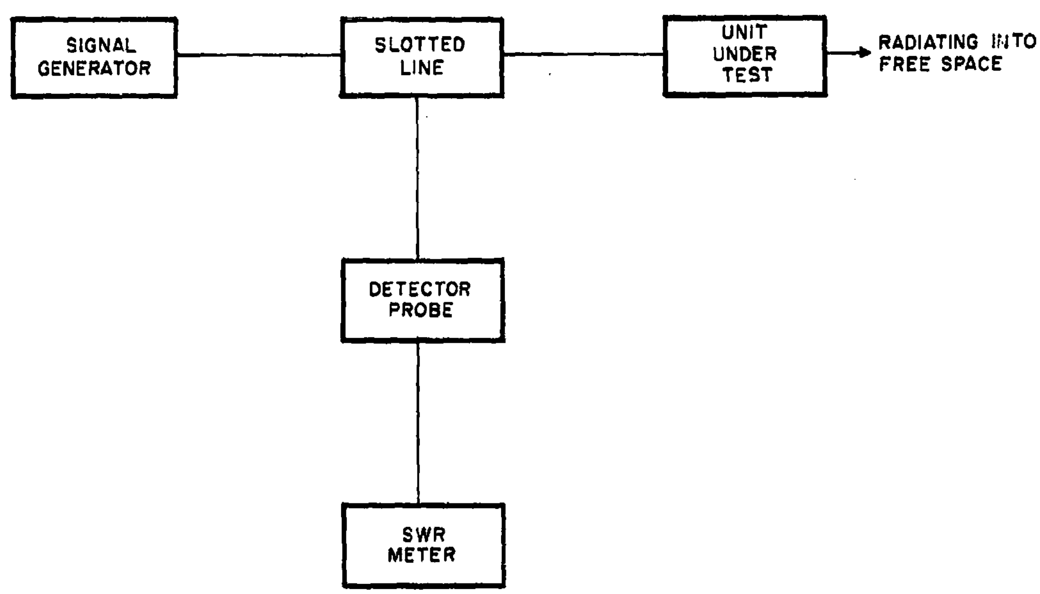

Figure A3. VSWR Measurements, Block Diagram

ADDENDUM B

ANTENNA COVERAGE TESTS

The antenna coverage tests were performed by the D. S. Kennedy Co., at Cohasset, Massachusetts. The test range is as shown in figure B1. Polarization of the transmitting antenna was adjusted in accordance with the test performed. The antenna under test was mounted on a tower approximately 30 feet high in its normal attitude for azimuth patterns and endwise for elevation patterns. Signals received by the antenna, being tested, were bolometer detected and recorded, see figures B2 through B21.

Paragraphs 1 through 6 below describe the antenna coverage tests, and give the results, for the first test model. Table B1 tabulates the half-power beamwidths at the mid-point of each band (in both E and H planes) for test models, serial numbers 2 through 7.

1. ANTENNA GAIN

The gain of the antenna was measured at 1300 megacycles by the substitution method. That is, a standard horn of known absolute value was substituted for the antenna under test, both receiving the same maximum signal and the same detecting system being employed. The gain of the antenna was calculated as follows:

Gain of antenna above standard	20.10 db
Gain of standard above isotropic	14.11 db
	34.21 db

2. BEAM ANGLE ABOVE HORIZON

The beam angles above the horizon in the two IFF frequency bands and at the mid-point of the search radar band were taken as follows:

a. The antenna was leveled in the normal position.

b. A transit was mounted to the reflector support, so that its azimuth plane was parallel to the reference plane. The transit elevation scale was adjusted to zero degrees with the telescope level.

c. The reflector was turned 90 degrees, so that it "stood" on end.

d. The antenna was tilted and trained for maximum received signal and a reading recorded.

e. The antenna was then trained for a half-power reading.

f. The telescope on the transit was adjusted in the elevation plane until the edge of the transmitting antenna was sighted. The scale reading of the transit then recorded.

g. The antenna was rotated in its elevation plane until the opposite half-power reading was obtained.

h. The transit was again adjusted as in f above.

i. The upper and lower 3 db points (half-power) were added. One half of this angle was then subtracted from the upper 3 db point to obtain the beam angle above the horizon.

j. The results of the tests are tabulated below:

Frequency	Beam angle above horizon
1300 megacycles	2 degrees 45 minutes
1001.5 megacycles	3 degrees 46 minutes
1118.5 megacycles	3 degrees 37 minutes

3. AZIMUTH PATTERNS

Figures B2 through B8 show the principal plane horizontal patterns measured in both the search radar and IFF bands. The following is a summary of the half-power beamwidths and side lobe levels:

	Frequency (megacycles)	Half-power beamwidths (degrees)	Side Lobe level (db down)
Search Radar	1250	1.6	28.0
	1300	1.6	29.7
	1350	1.5	28.4
IFF	1001.5	1.8	25.9
	1035.5	1.8	26.9
	1081.5	1.6	27.9
	1118.5	1.6	26.9

4. BACK RADIATION

Back radiation (figure B9) was measured by recording a 360 degree azimuth pattern. The radiation in the vicinity of 180 degrees from the main lobe is at least 33 db below the peak.

5. AZIMUTH SIDE LOBES ABOVE AND BELOW 2.5 DEGREES

The azimuth side lobes off the principal plane were checked by measuring horizontal patterns at specific tilt angles. The antenna was tilted sufficiently to determine the greatest intensity of side lobes whose maximums do not lie in the principal plane of the main lobe.

The azimuth side lobe patterns are given in figures B10 through B14. The power of the highest side lobe with respect to the reference power level of the main lobe is noted on the patterns.

6. ELEVATION PATTERNS

The elevation patterns (figures B15 through B21) were taken with the antenna oriented to stand on end. The principal plane vertical patterns were measured for the search radar and IFF frequency bands. The specification limit for vertical shaping is shown overlaid on figure B16.

TABLE B1. HALF-POWER BEAMWIDTHS

TEST MODEL (serial number)	H PLANE (frequency in megacycles)			E PLANE (frequency in megacycles)		
	1020	1100	1300	1020	1100	1300
2	1.83°	1.7°	1.67°	7.9°	8.2°	5.8°
3	1.87°	1.77°	1.67°	8.13°	8.27°	5.83°
4	1.83°	1.73°	1.7°	8.13°	8.2°	5.8°
5	1.83°	1.73°	1.67°	8.47°	8.2°	6.0°
6	1.80°	1.77°	1.63°	8.2°	8.2°	5.83°
7	1.87°	1.7°	1.67°	8.07°	8.1°	5.97°

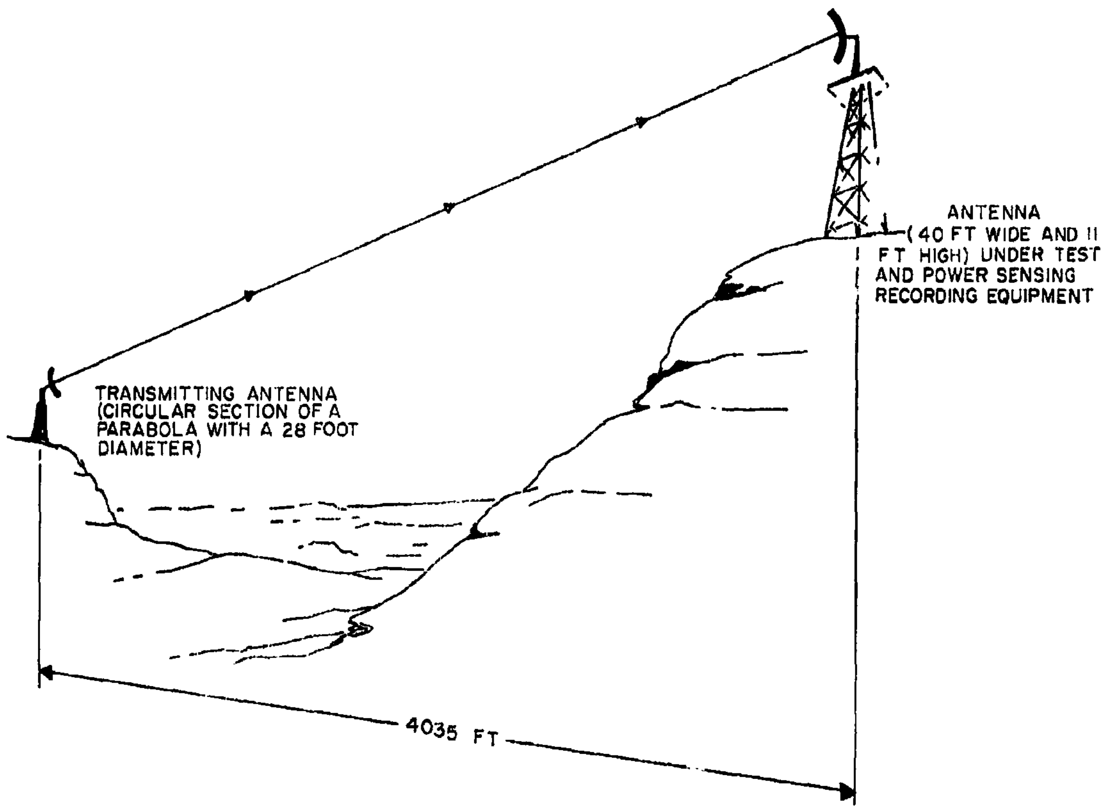

Figure B1. Antenna Test Range

Figure B2. Horizontal P

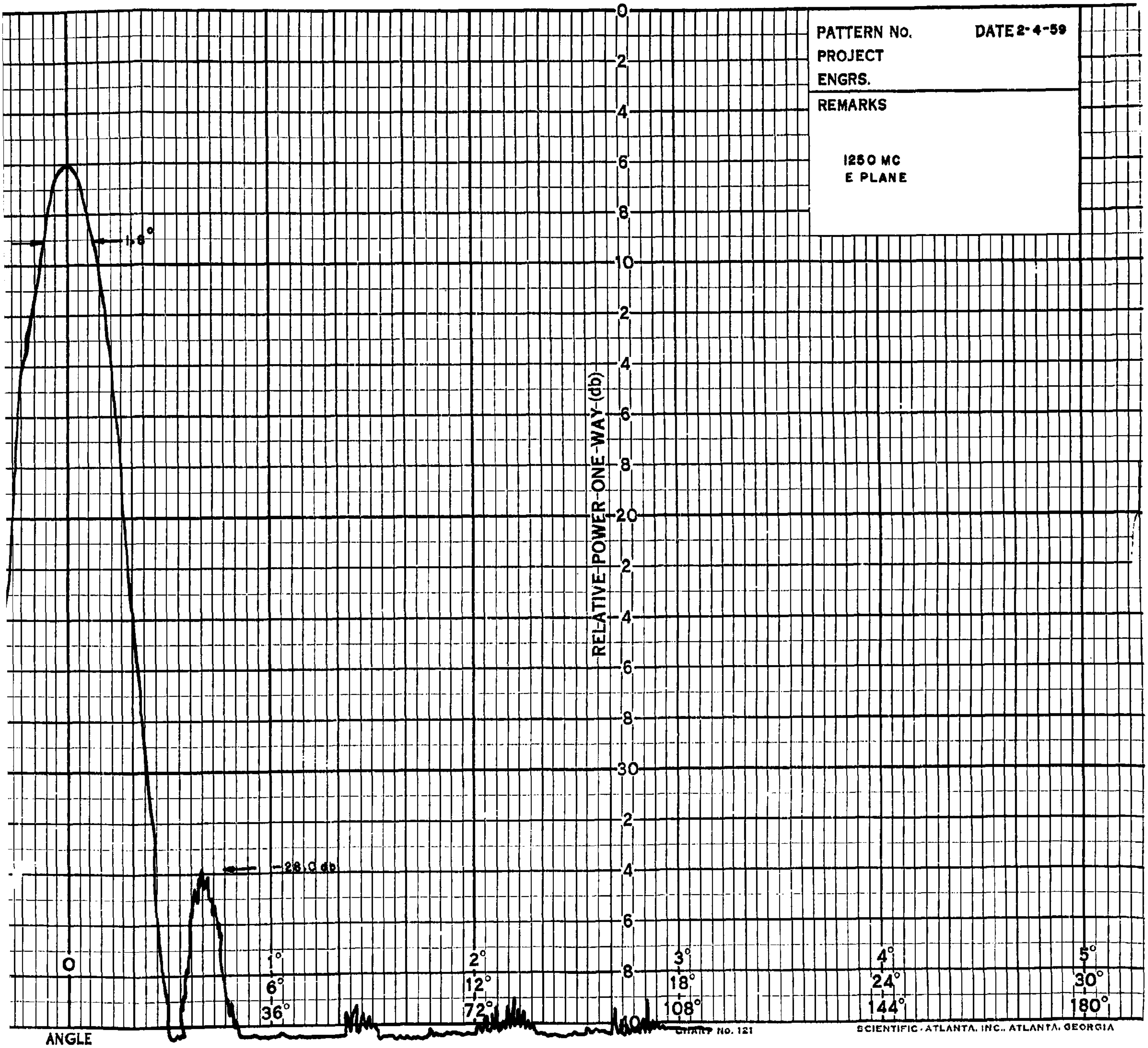

Figure B2. Horizontal Pattern, Search Radar, 1250 Megacycles

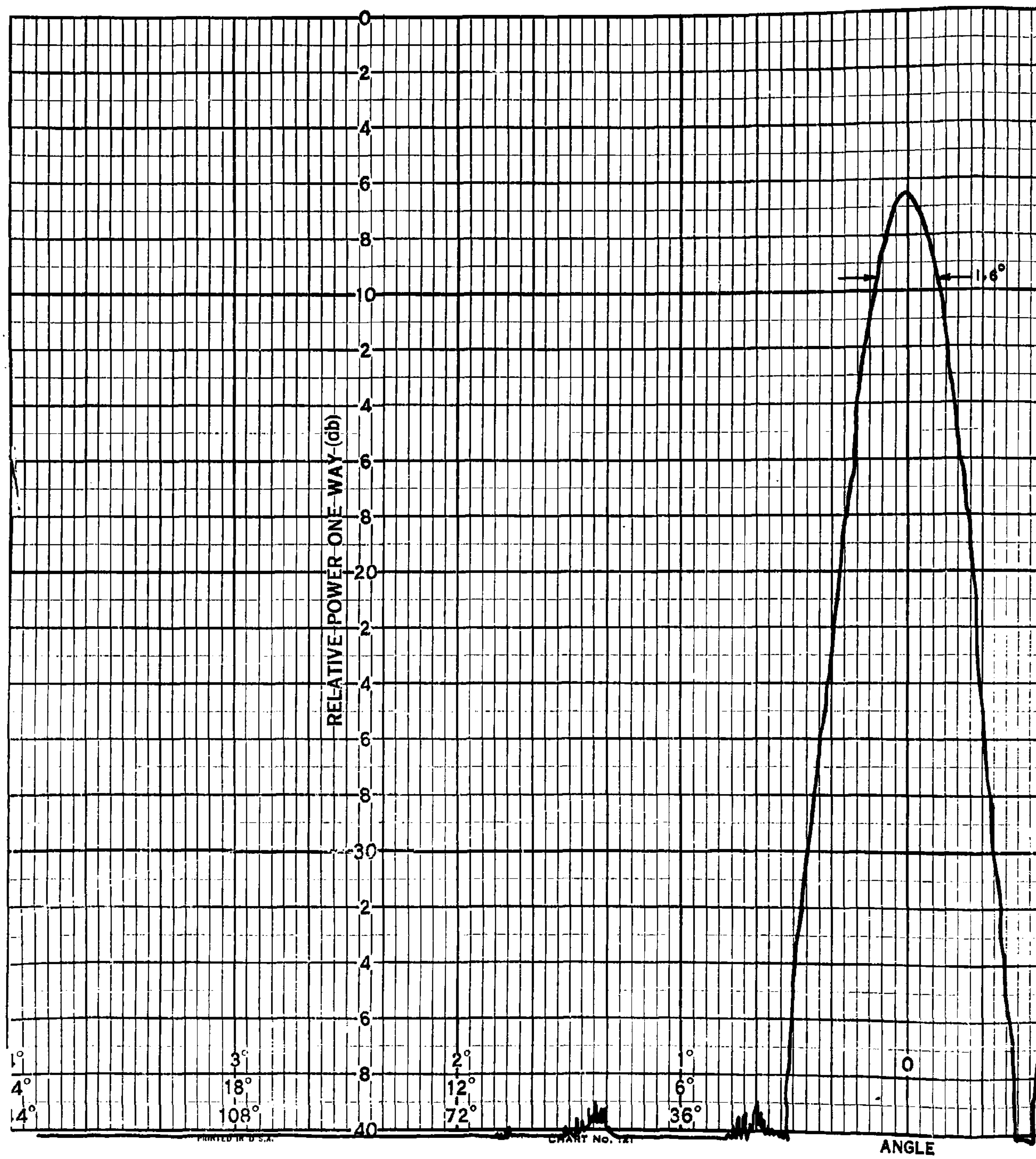

Figure B 3. Hori

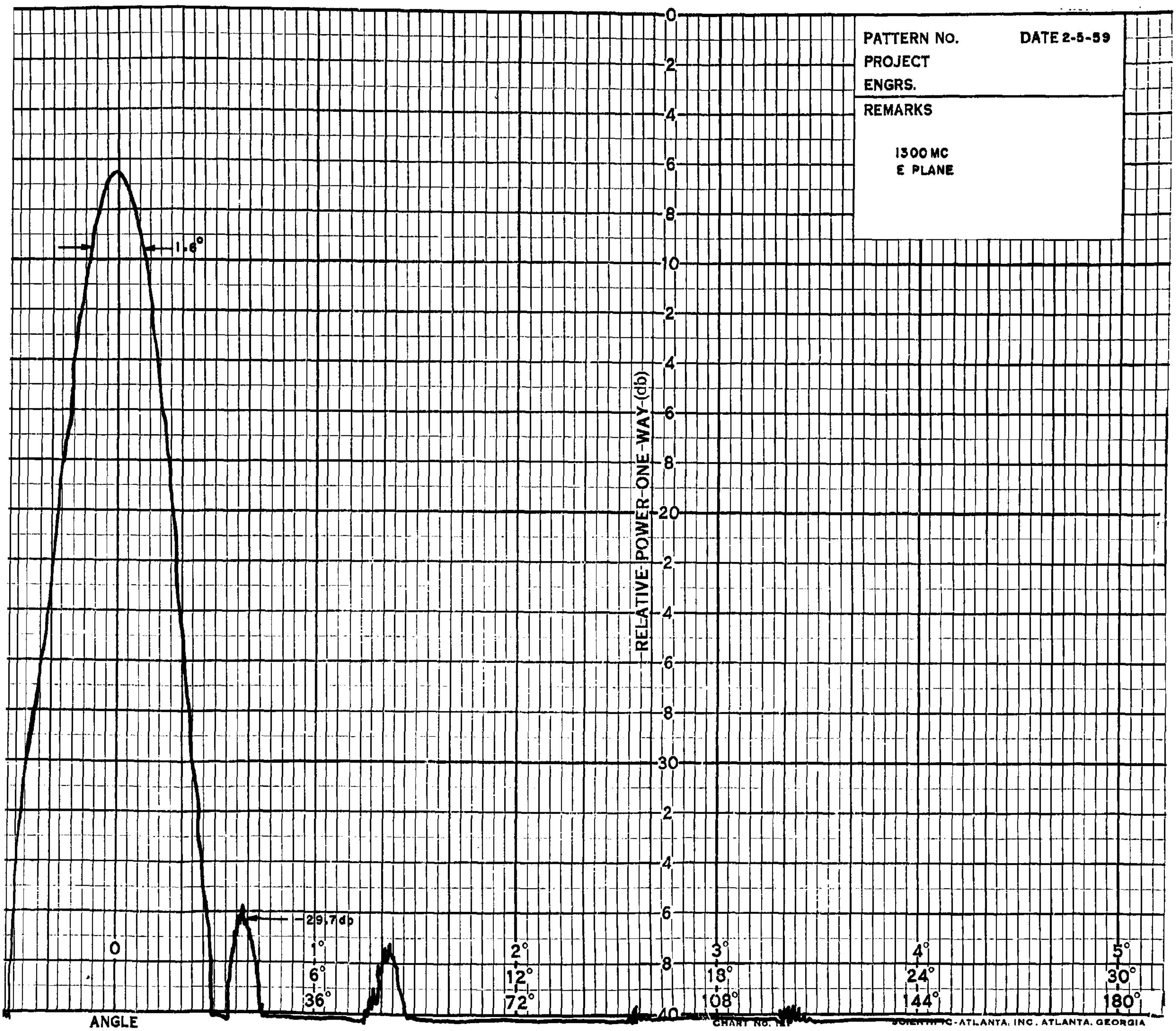

Figure B 3. Horizontal Pattern, Search Radar, 1300 Megacycles

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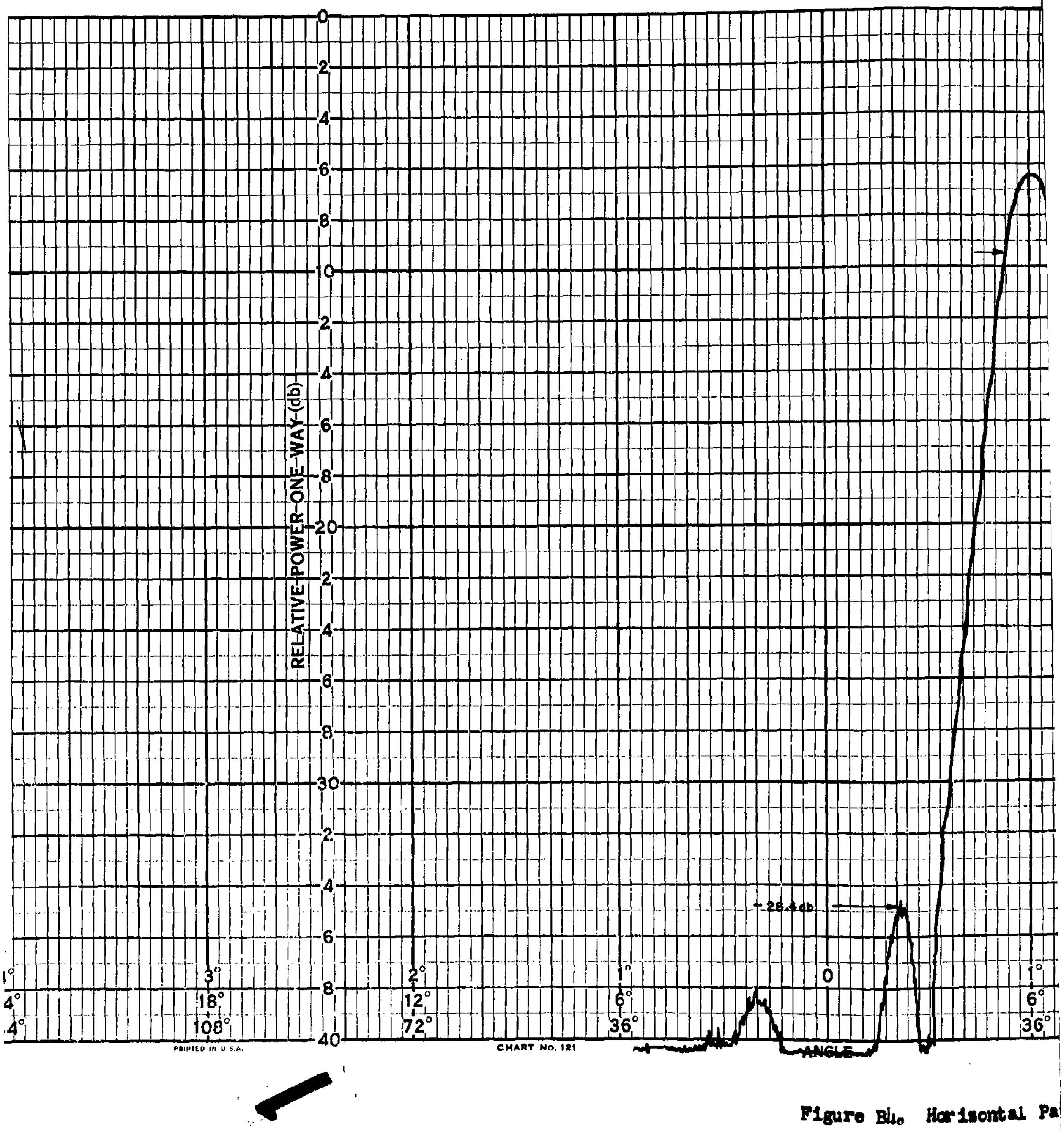

Figure B4. Horizontal Pa

Figure B4. Horizontal Pattern, Search Radar, 1350 Megacycles

2

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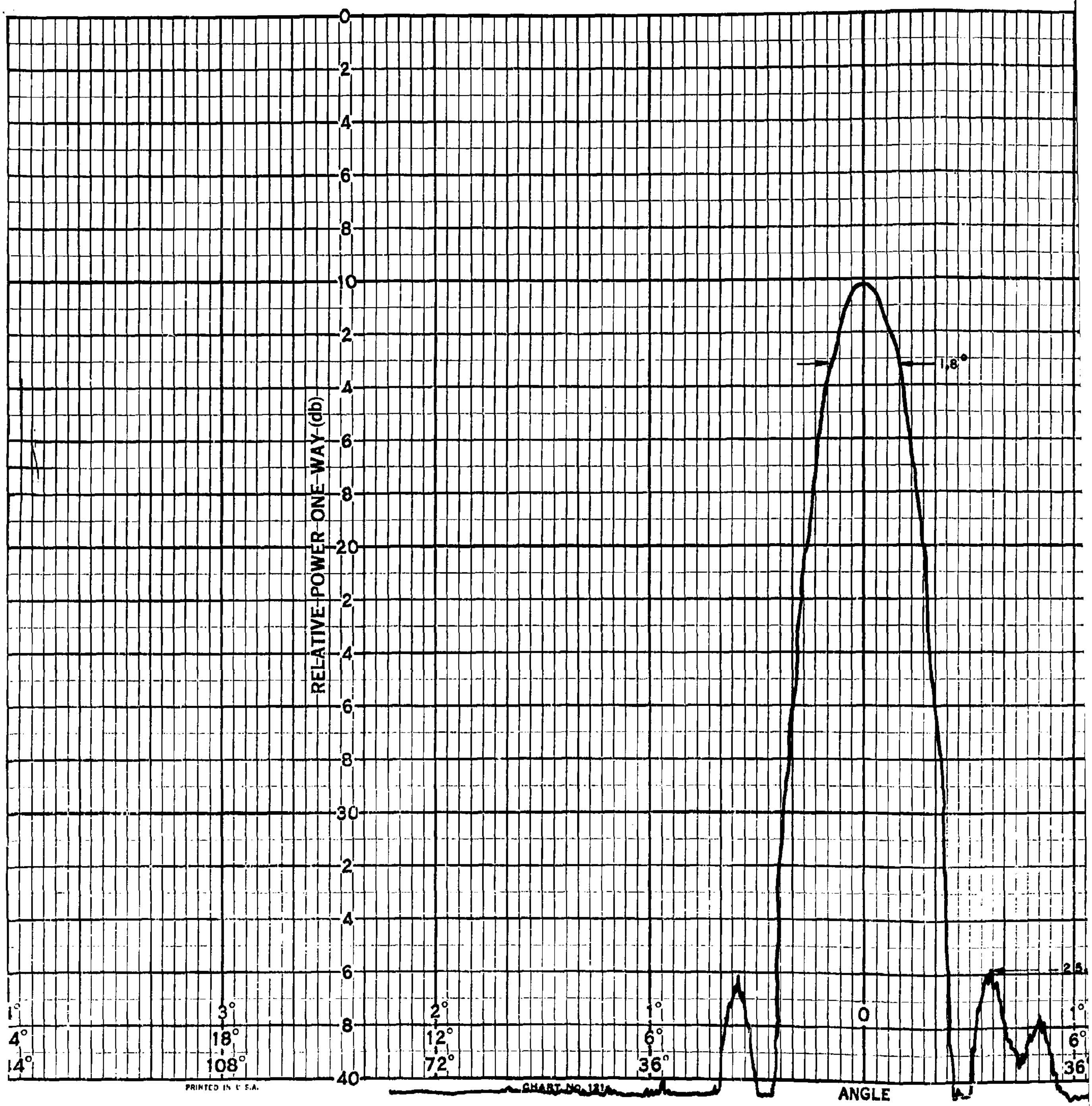

Figure B5. Horizontal P

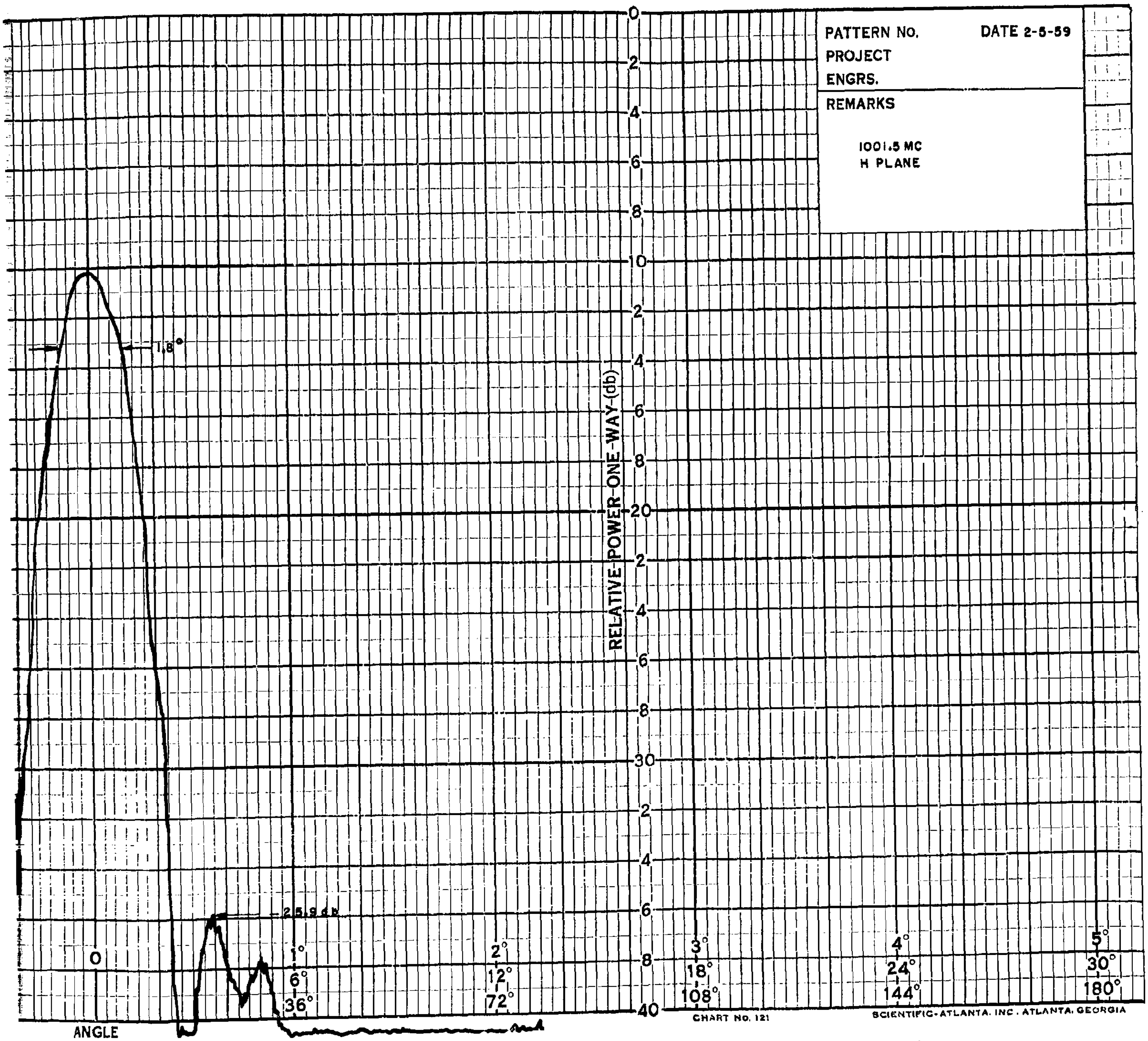

Figure B5. Horizontal Pattern, IFF Radar, 1001.5 Megacycles

2

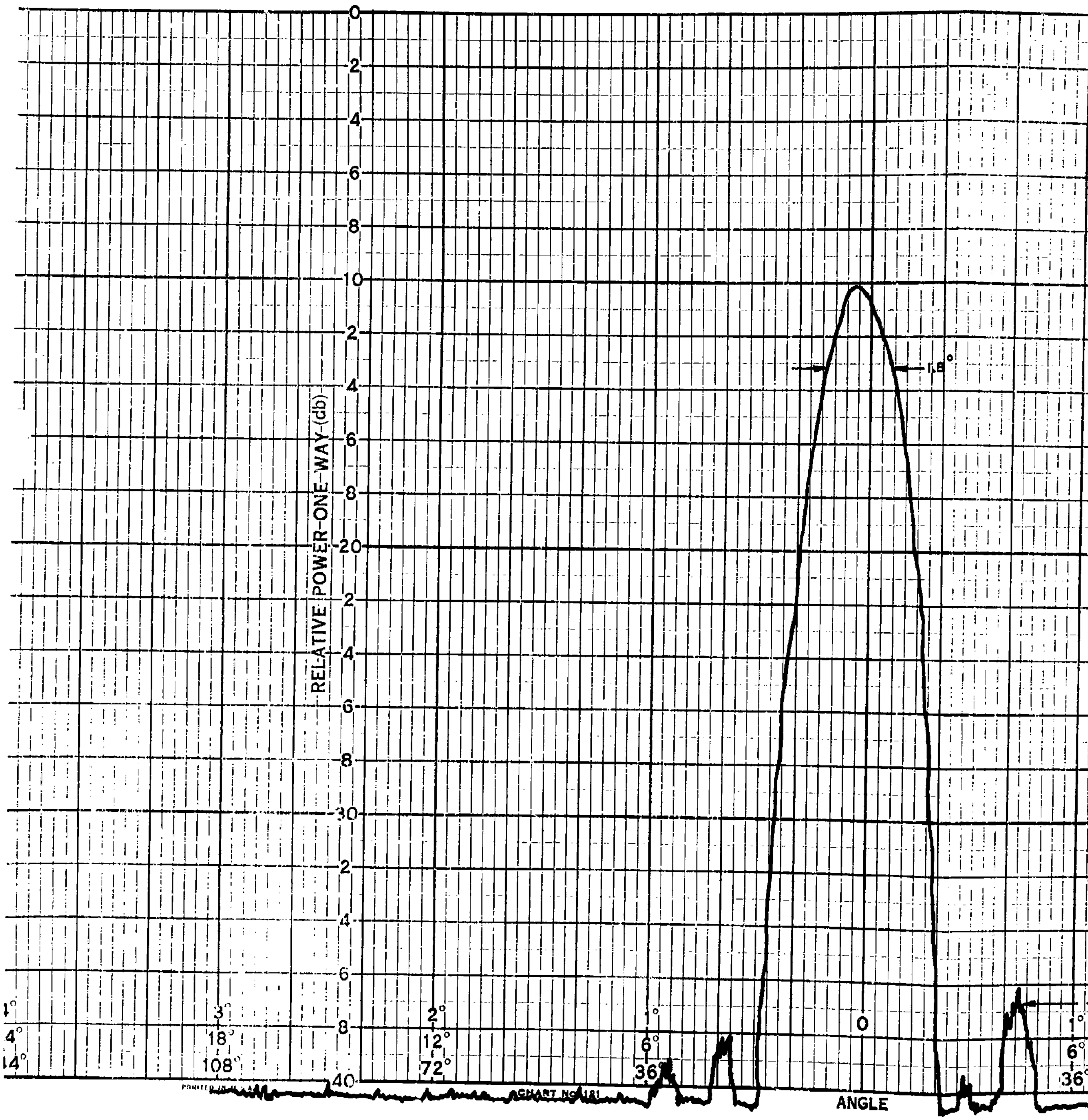

Figure B6. Horizontal F

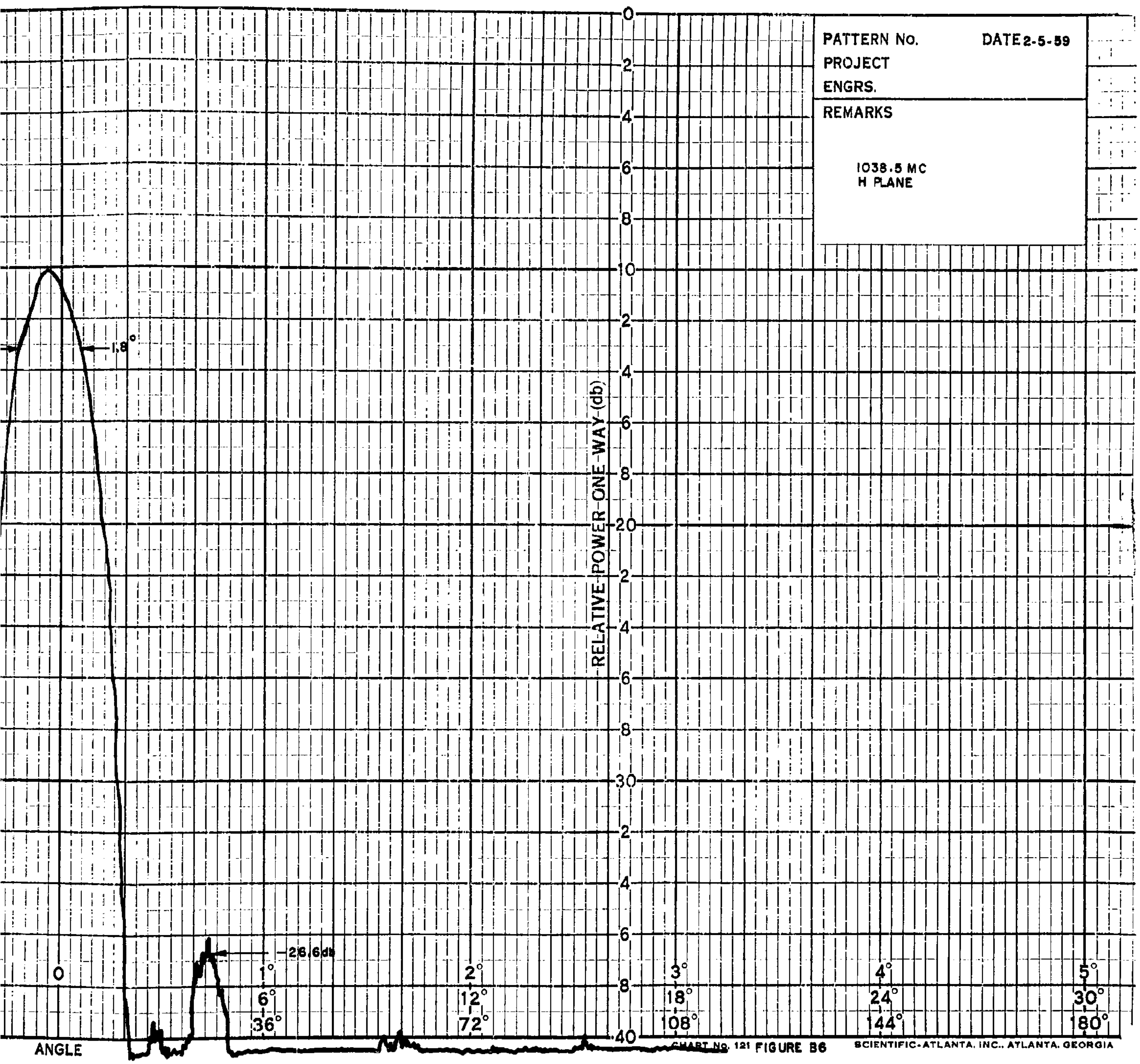

Figure B6. Horizontal Pattern, IFF Radar, 1038.5 Megacycles

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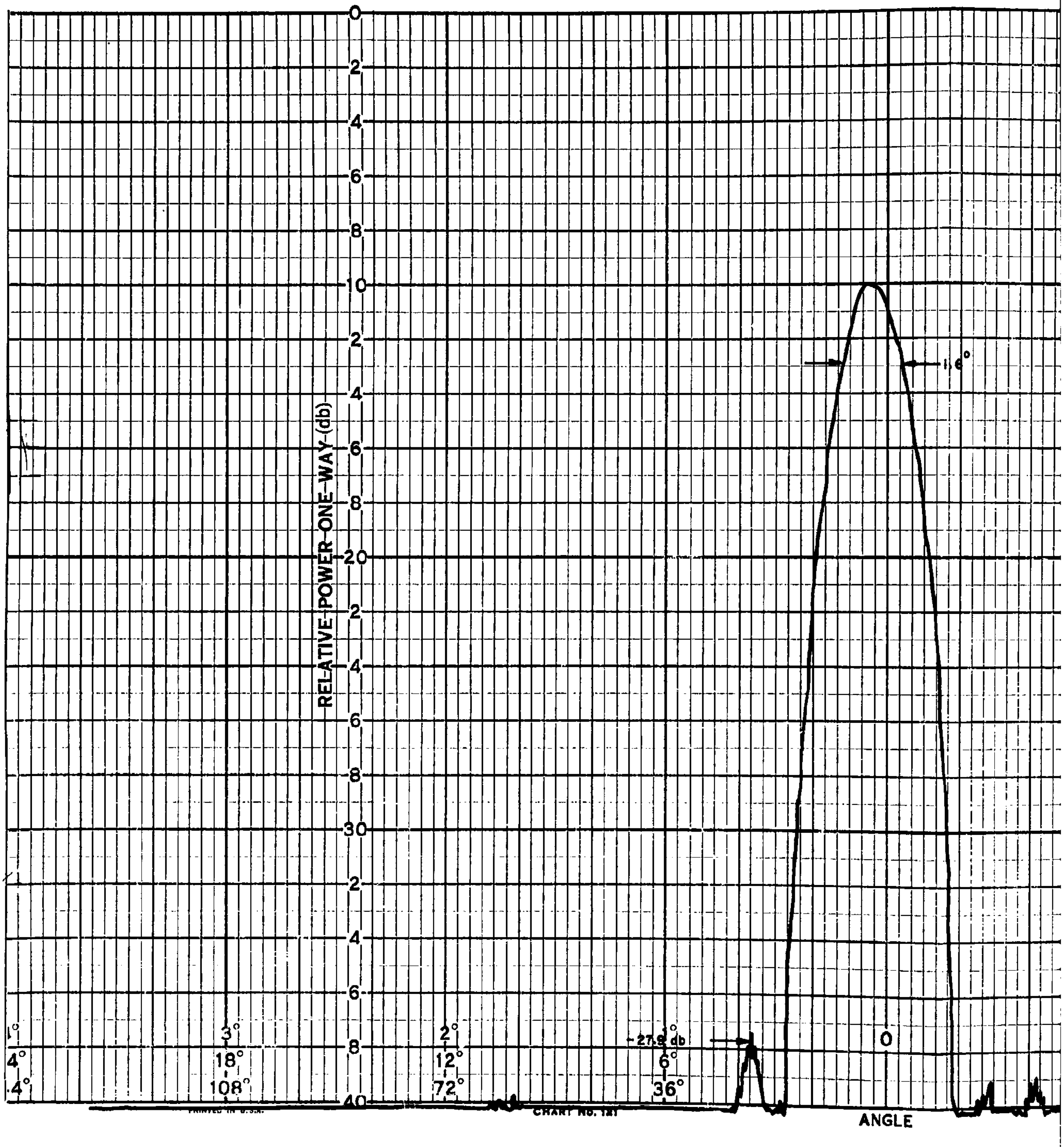

Figure B7. Hori

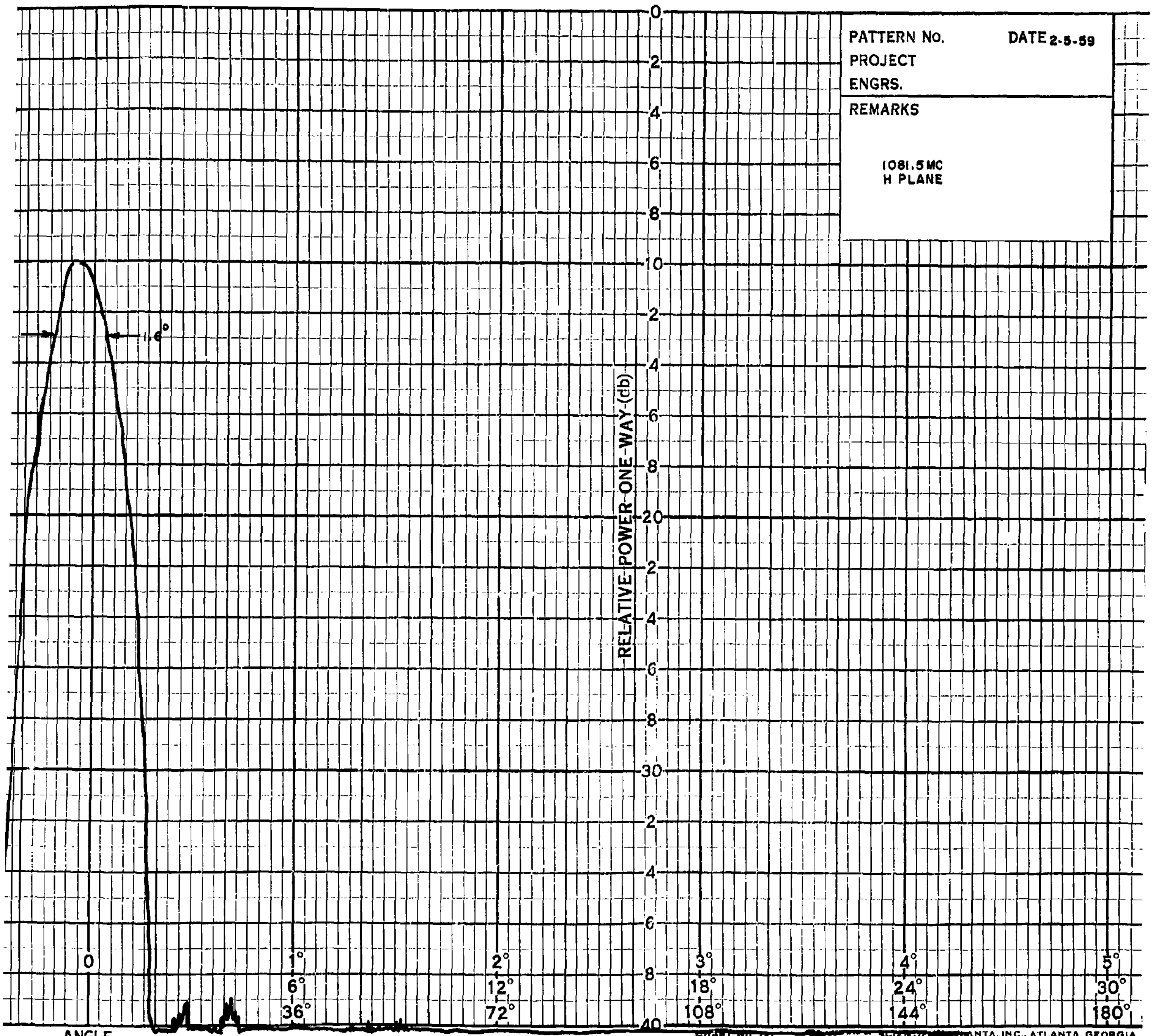

Figure B7. Horizontal Pattern, IFF Radar, 1081.5 Megacycles

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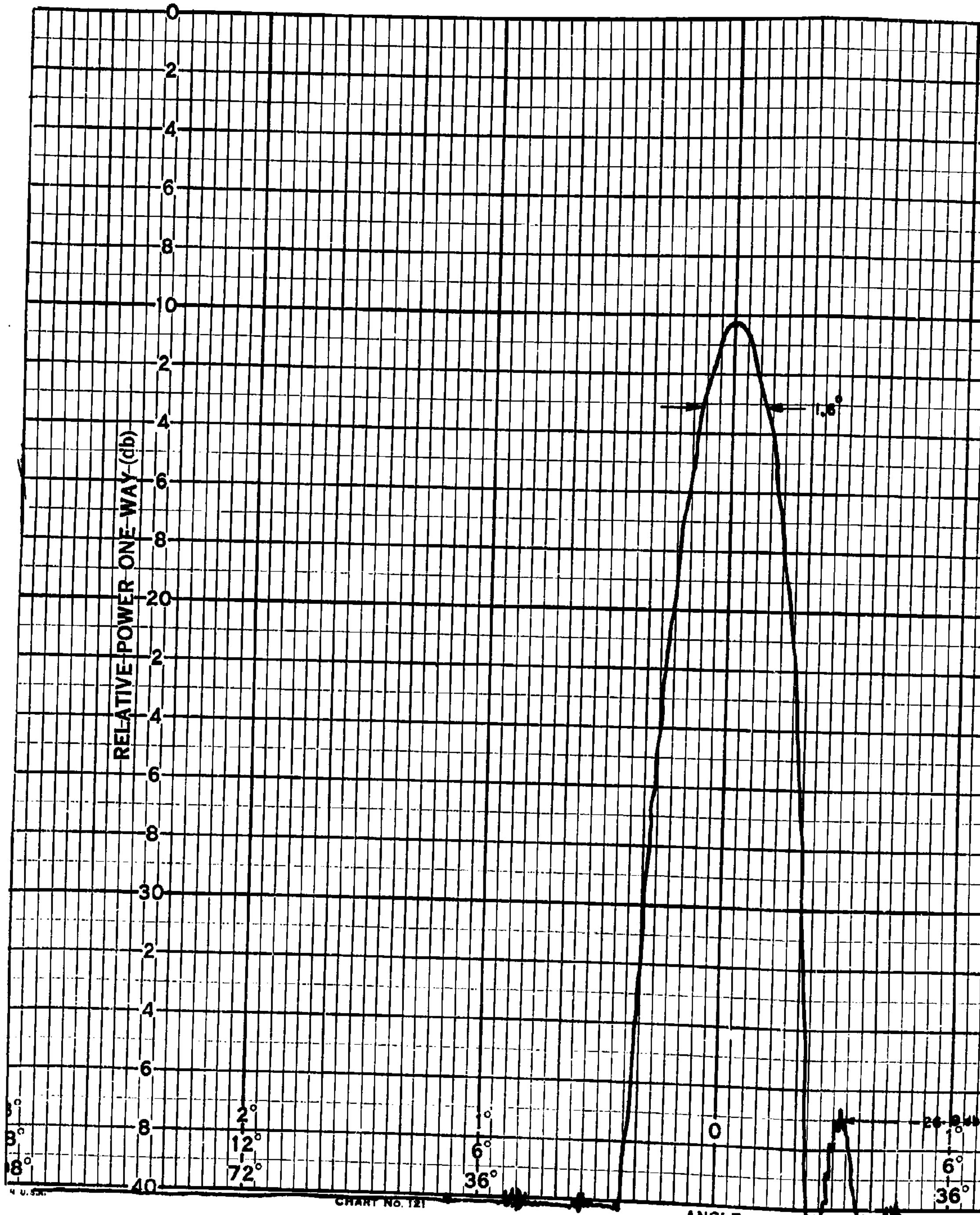

Figure B8. Horizontal

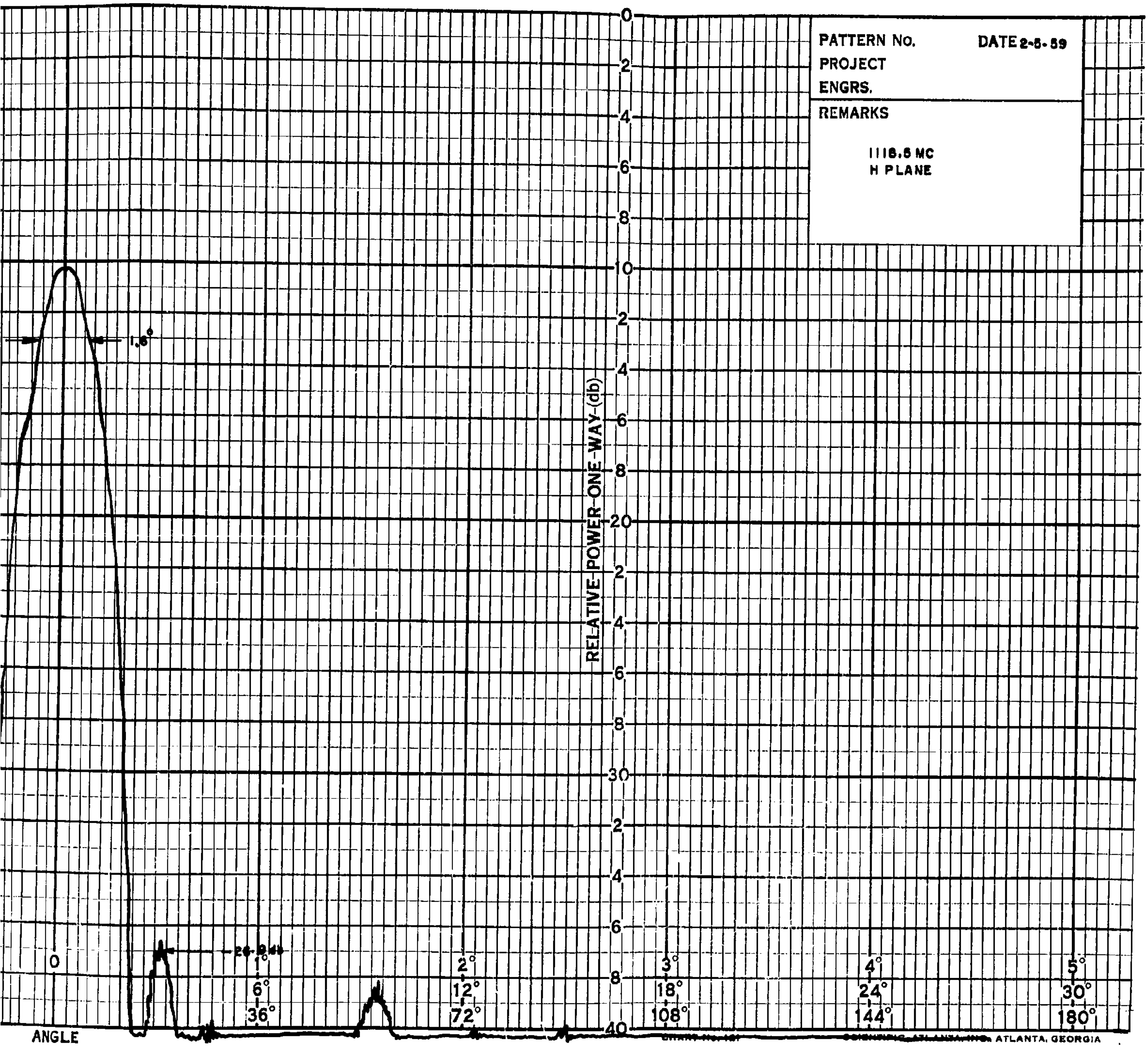

Figure B8. Horizontal Pattern, IFF Radar, 1118.5 Megacycles

2

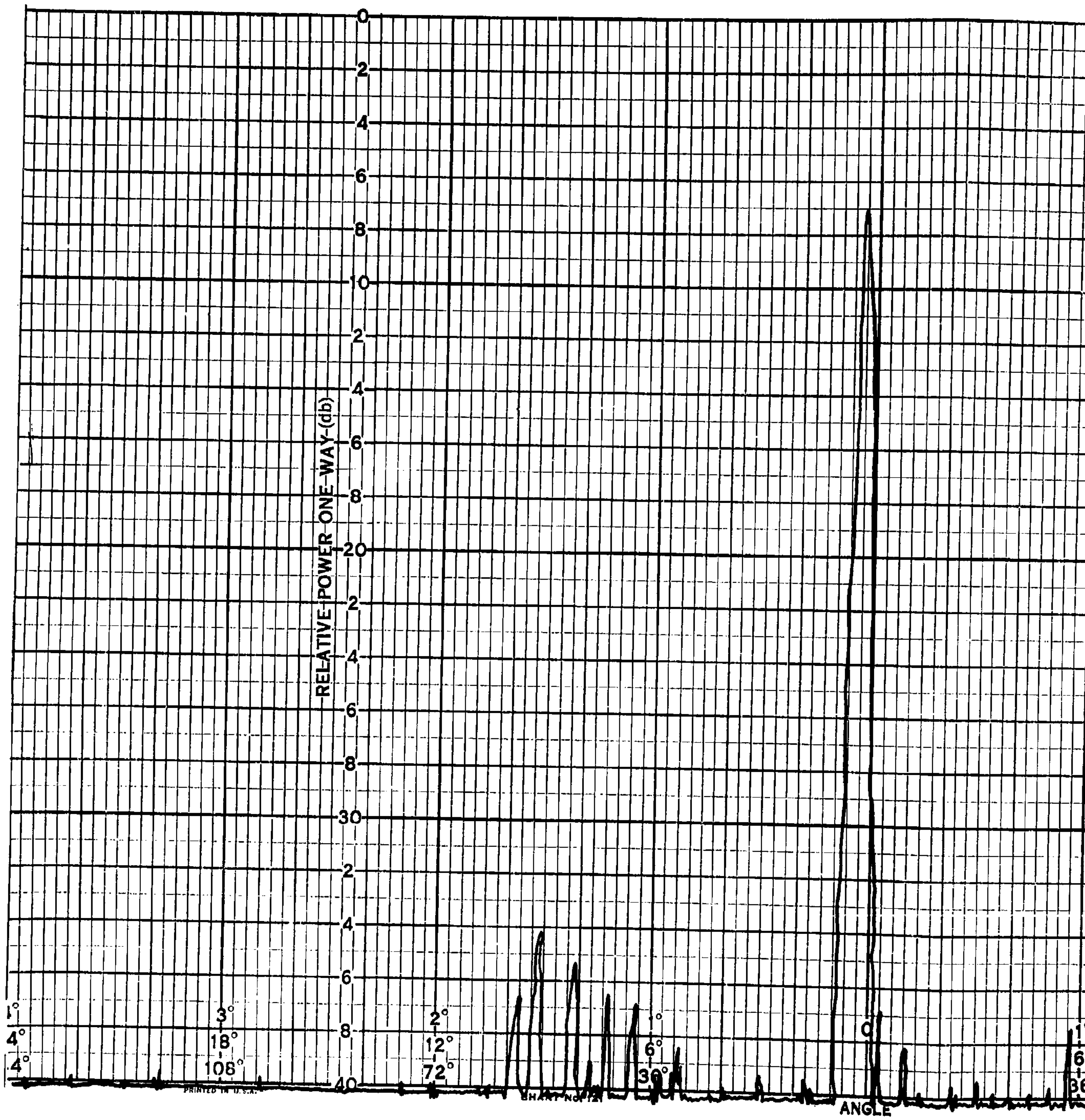

Figure B9. Back R

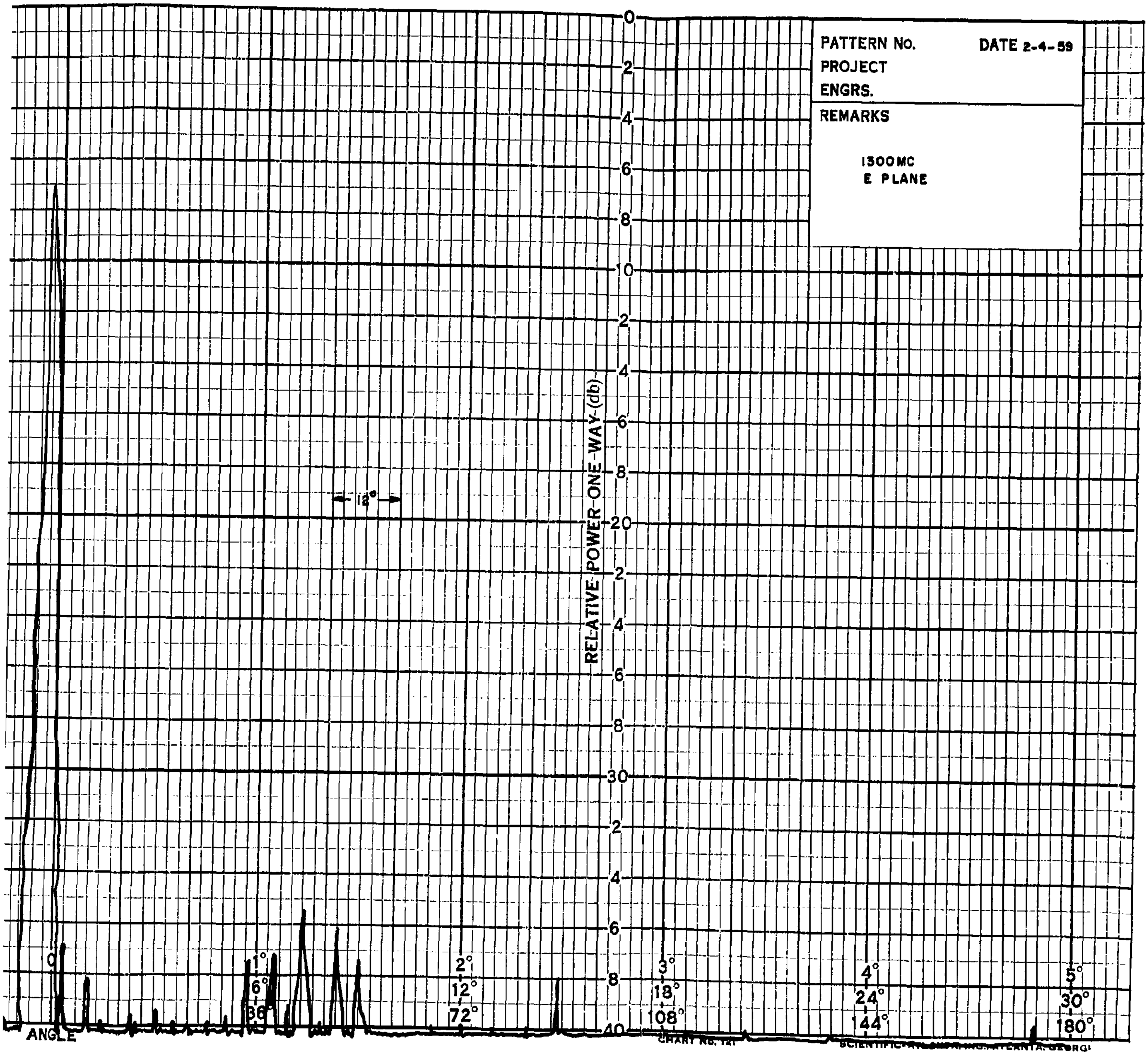

Figure B9. Back Radiation Pattern, Search Radar

2

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Figure B10. Azimuth Sid

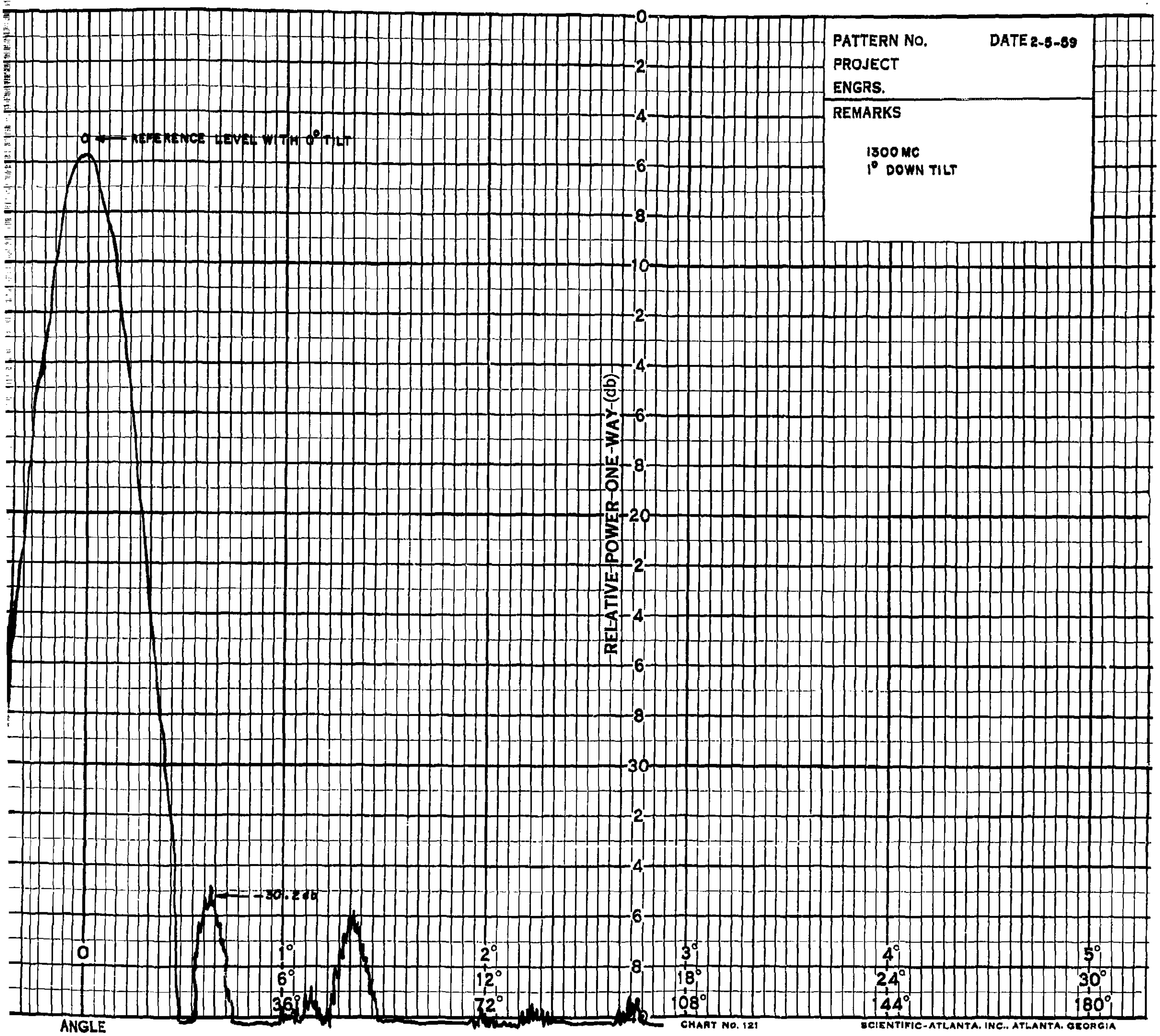

Figure B10. Azimuth Side Lobe Pattern, 1° Down Tilt

2

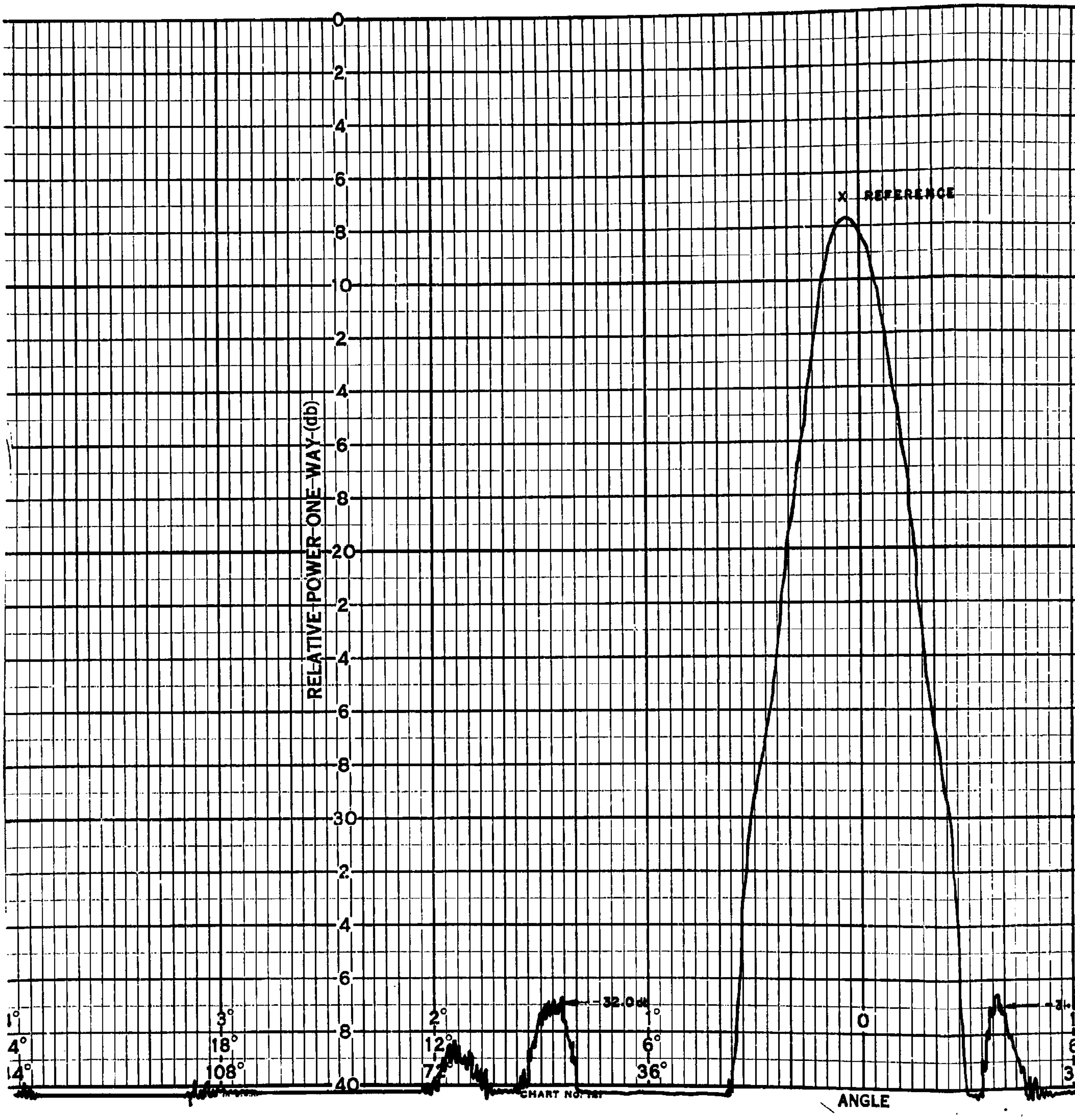

Figure B11. Azim

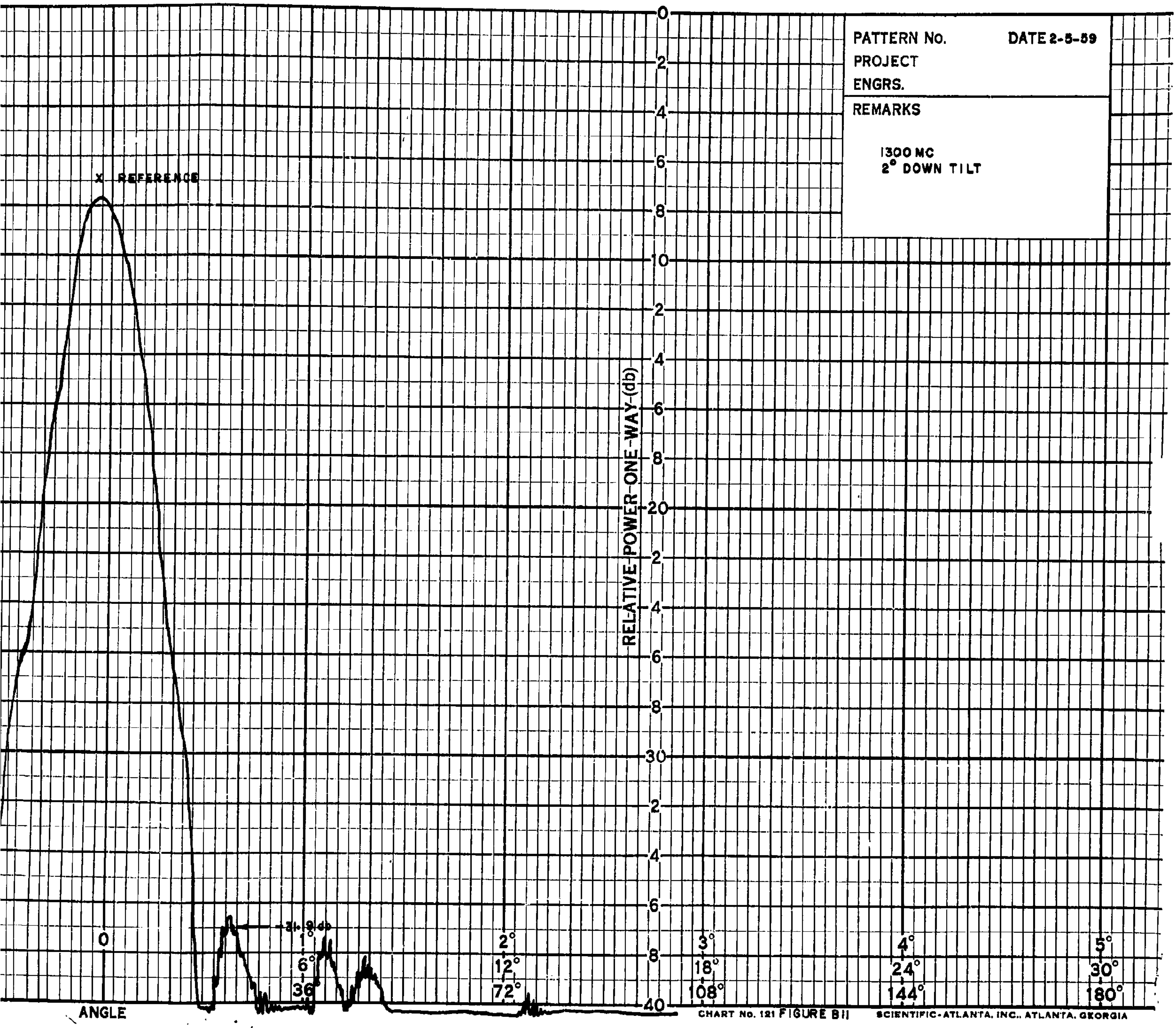

Figure B11. Azimuth Side Lobe Pattern, 2° Down Tilt

2

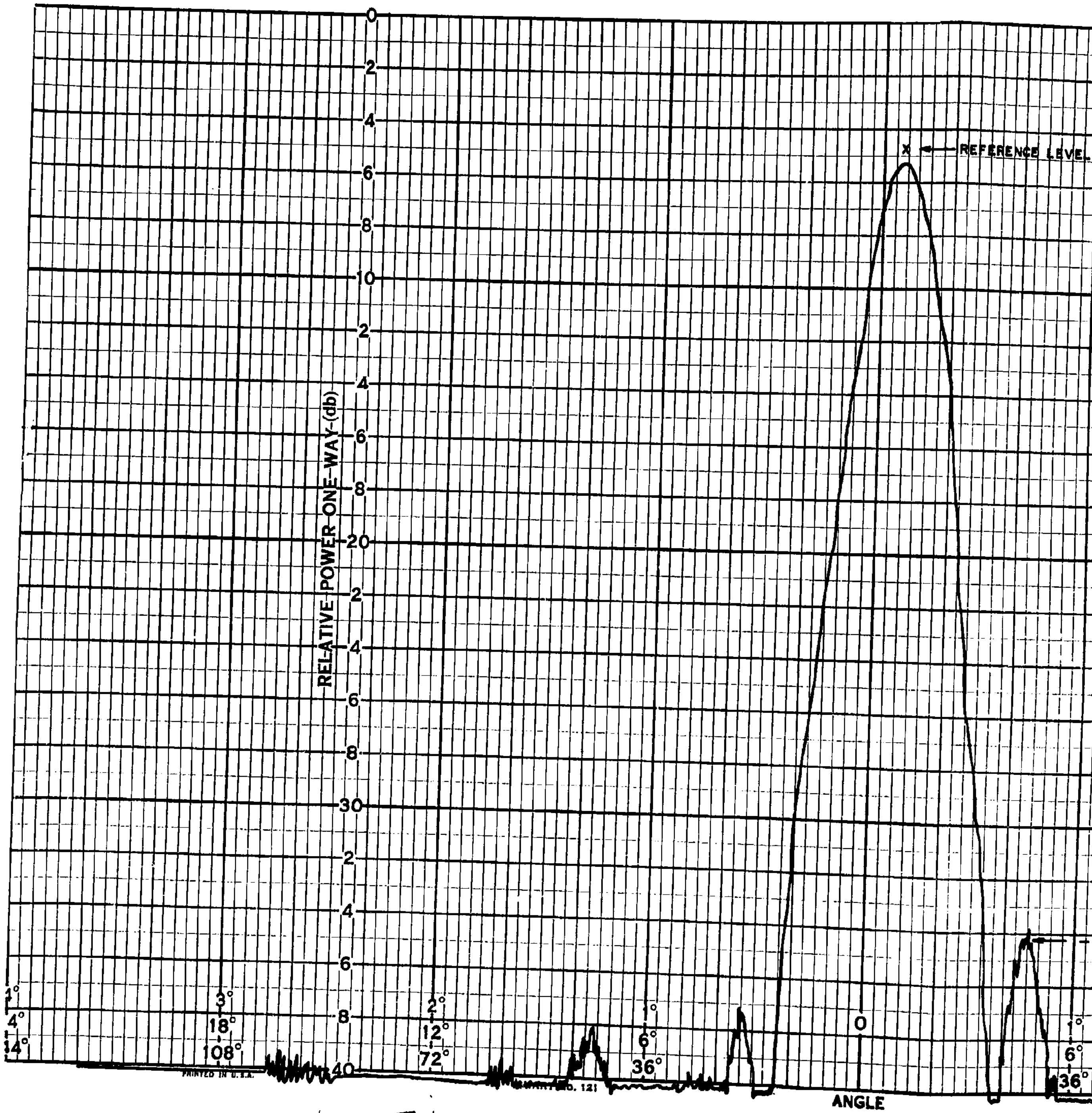

Figure B12. Azimuth Si

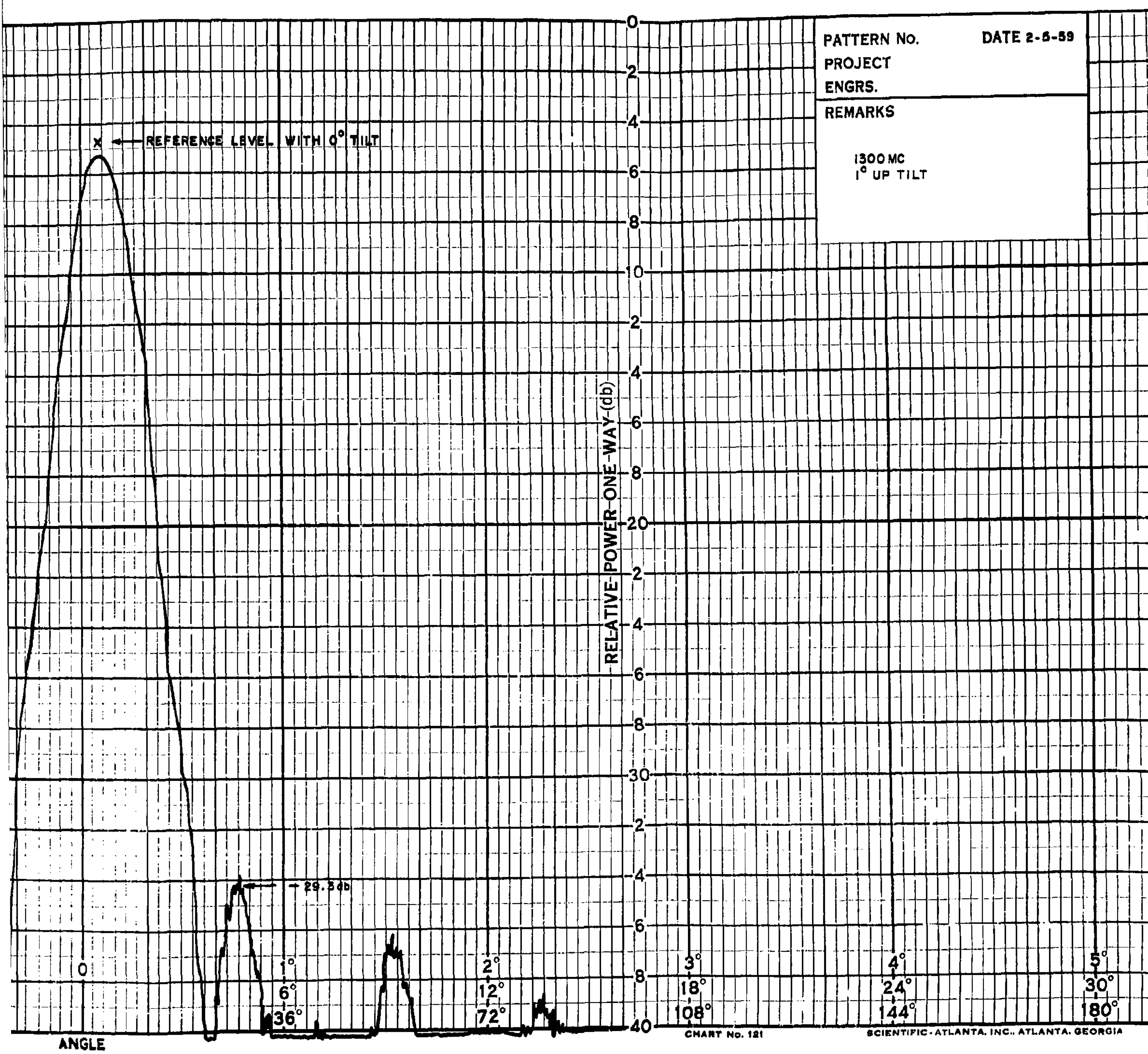

Figure B12. Azimuth Side Lobe Pattern, 1° Up Tilt

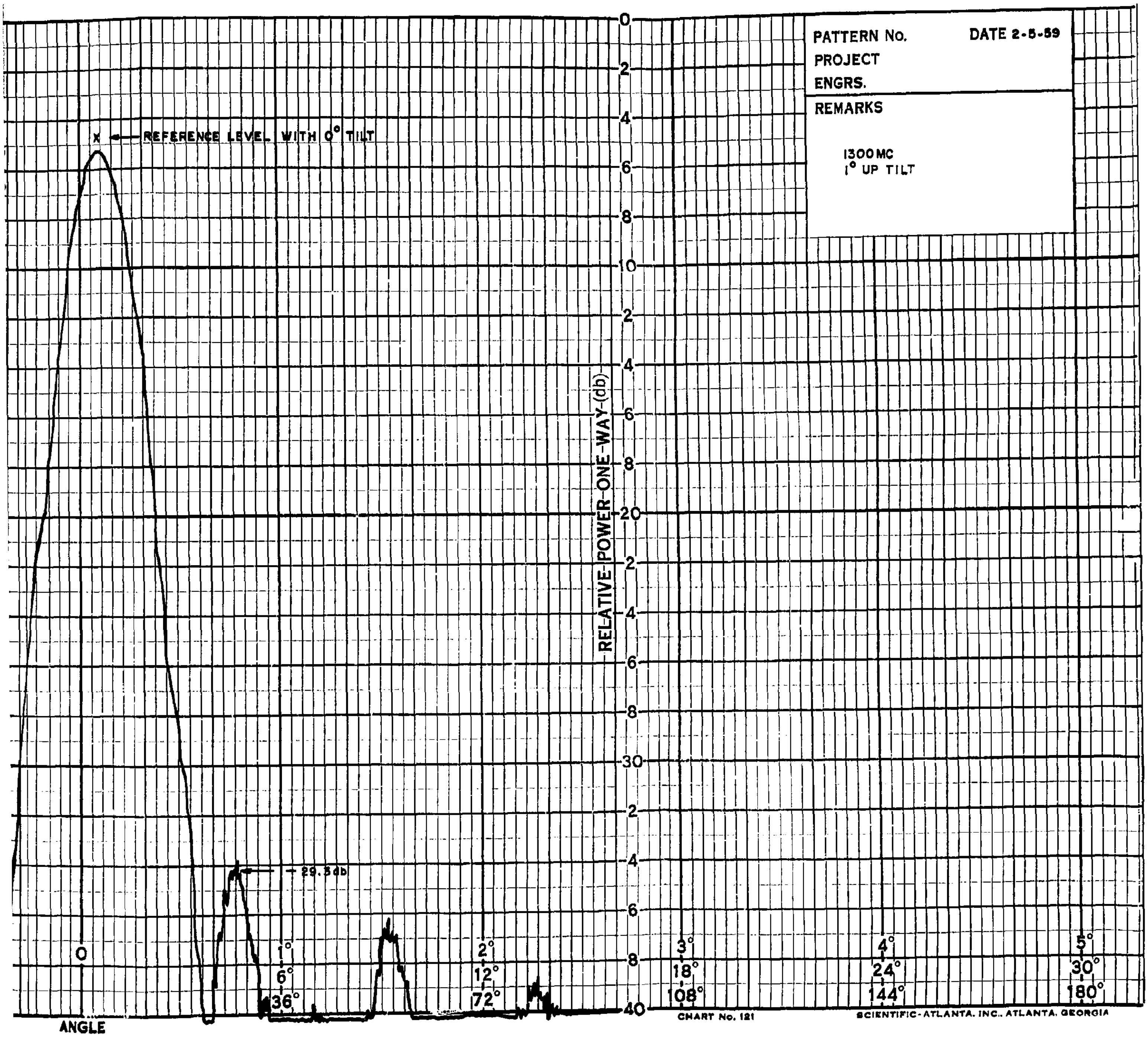

Figure B12. Azimuth Side Lobe Pattern, 1° Up Tilt

2

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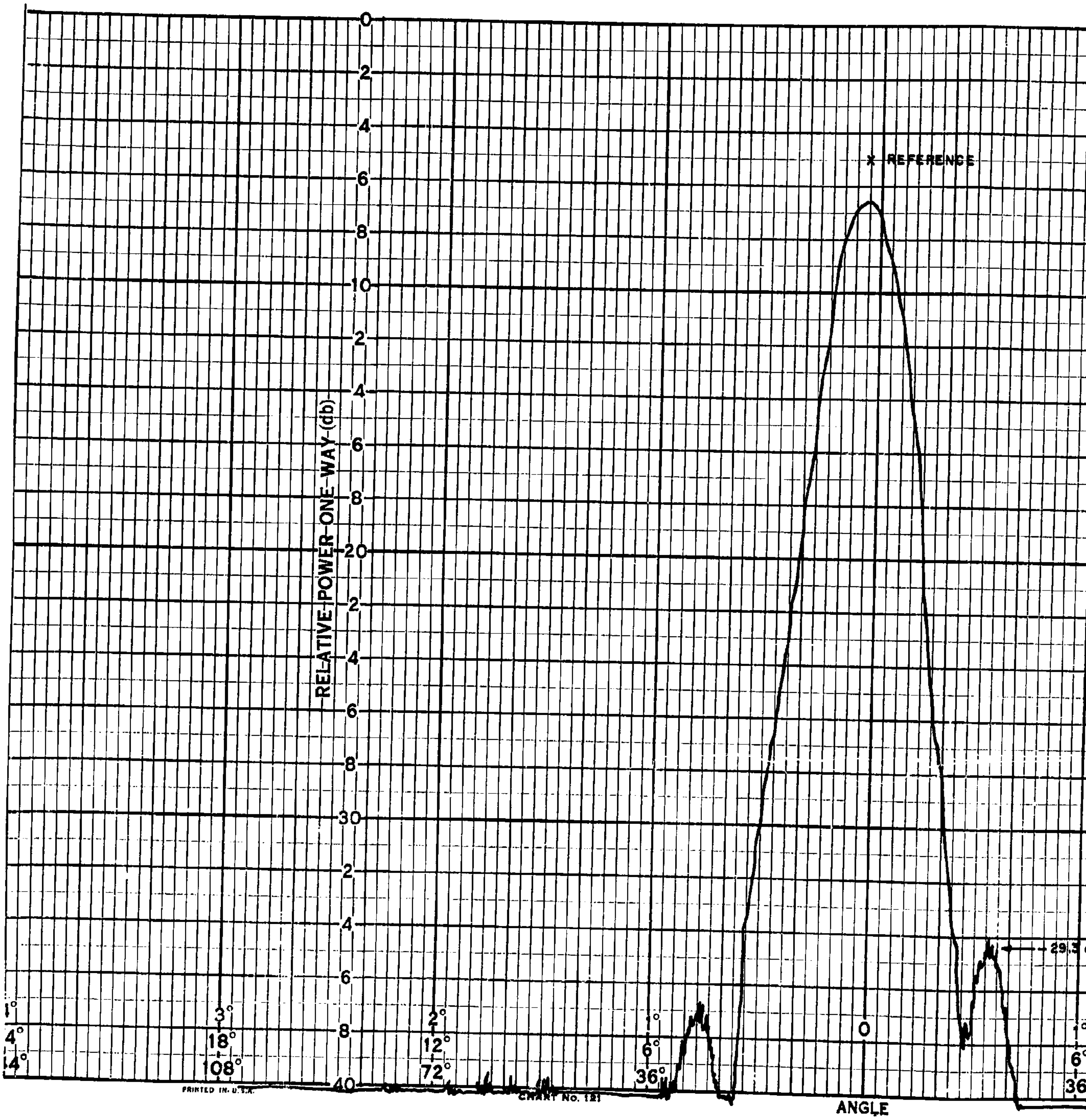

Figure B13. Azi

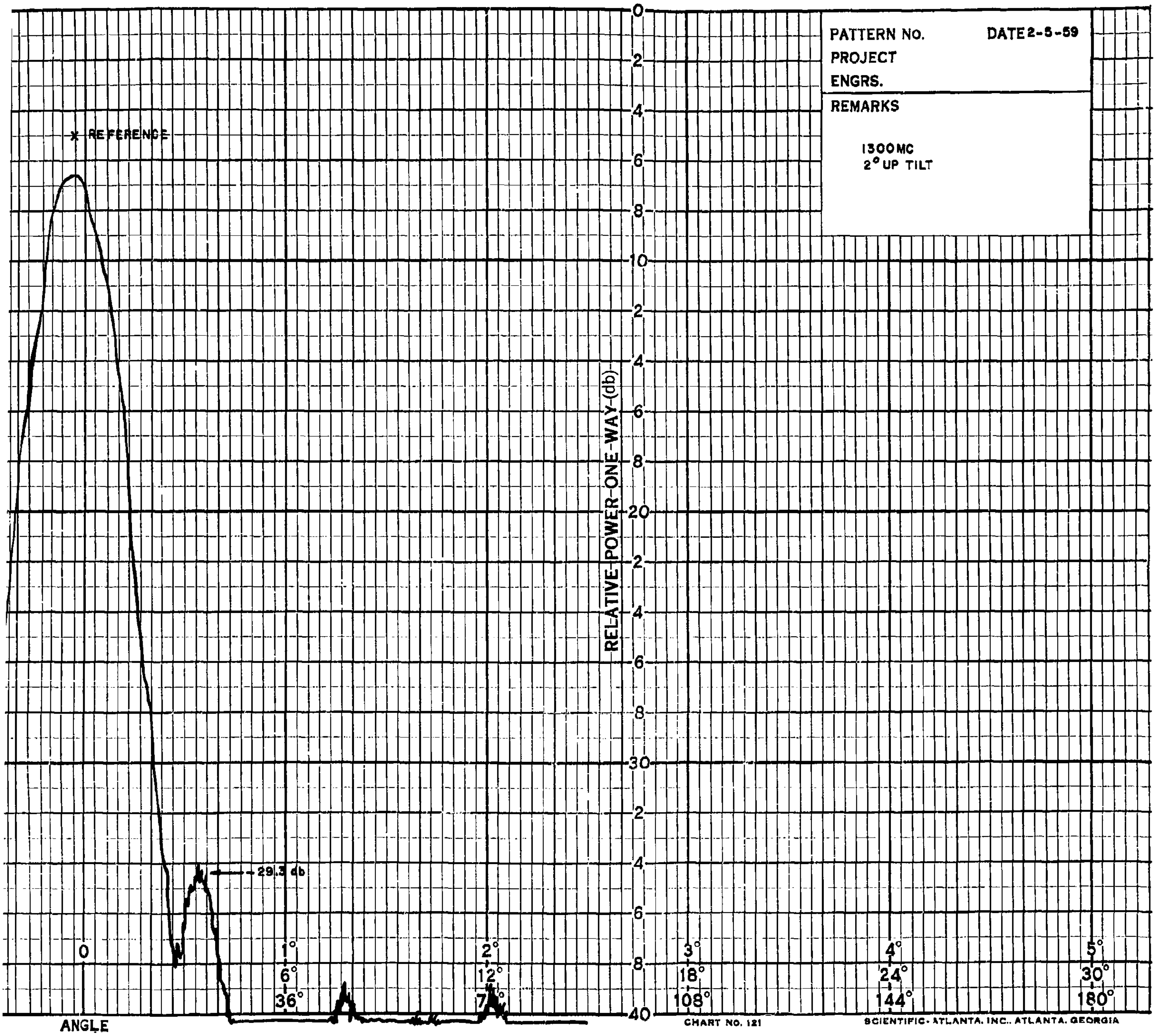

Figure B13. Azimuth Side Lobe Pattern, 2° Up Tilt

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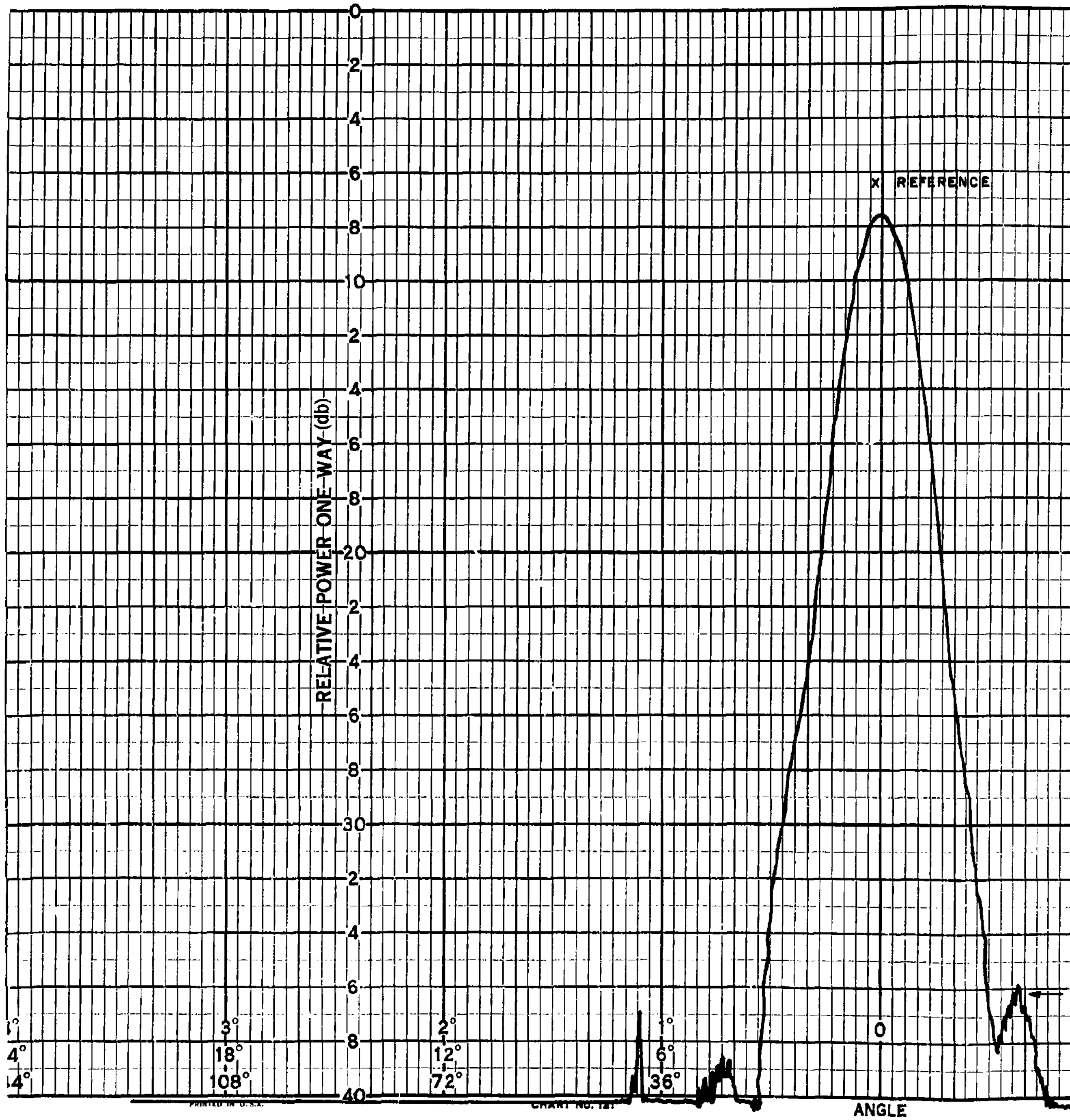

Figure B1

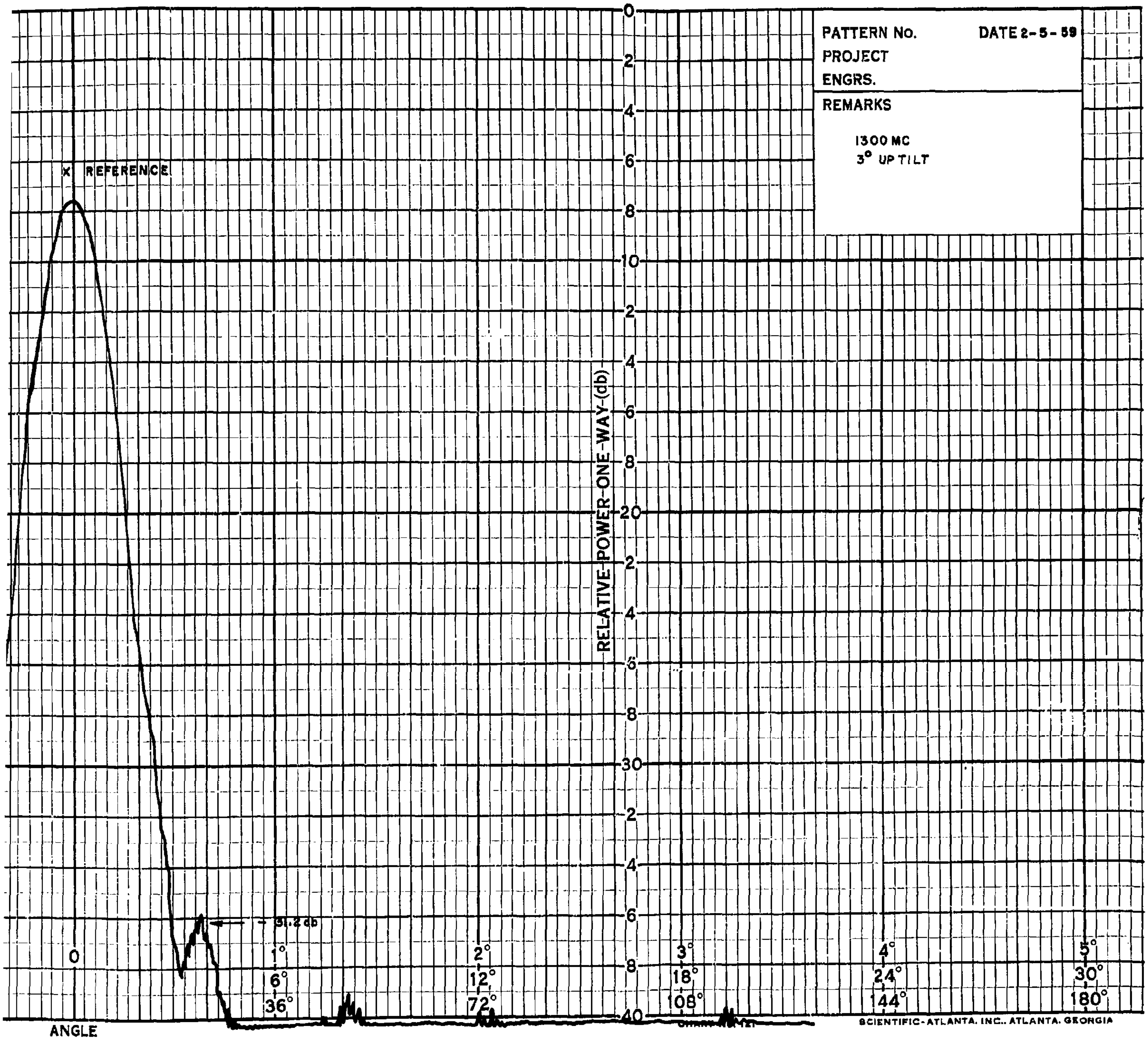

Figure B14. Azimuth Side Lobe Pattern, 3° Up Tilt

2

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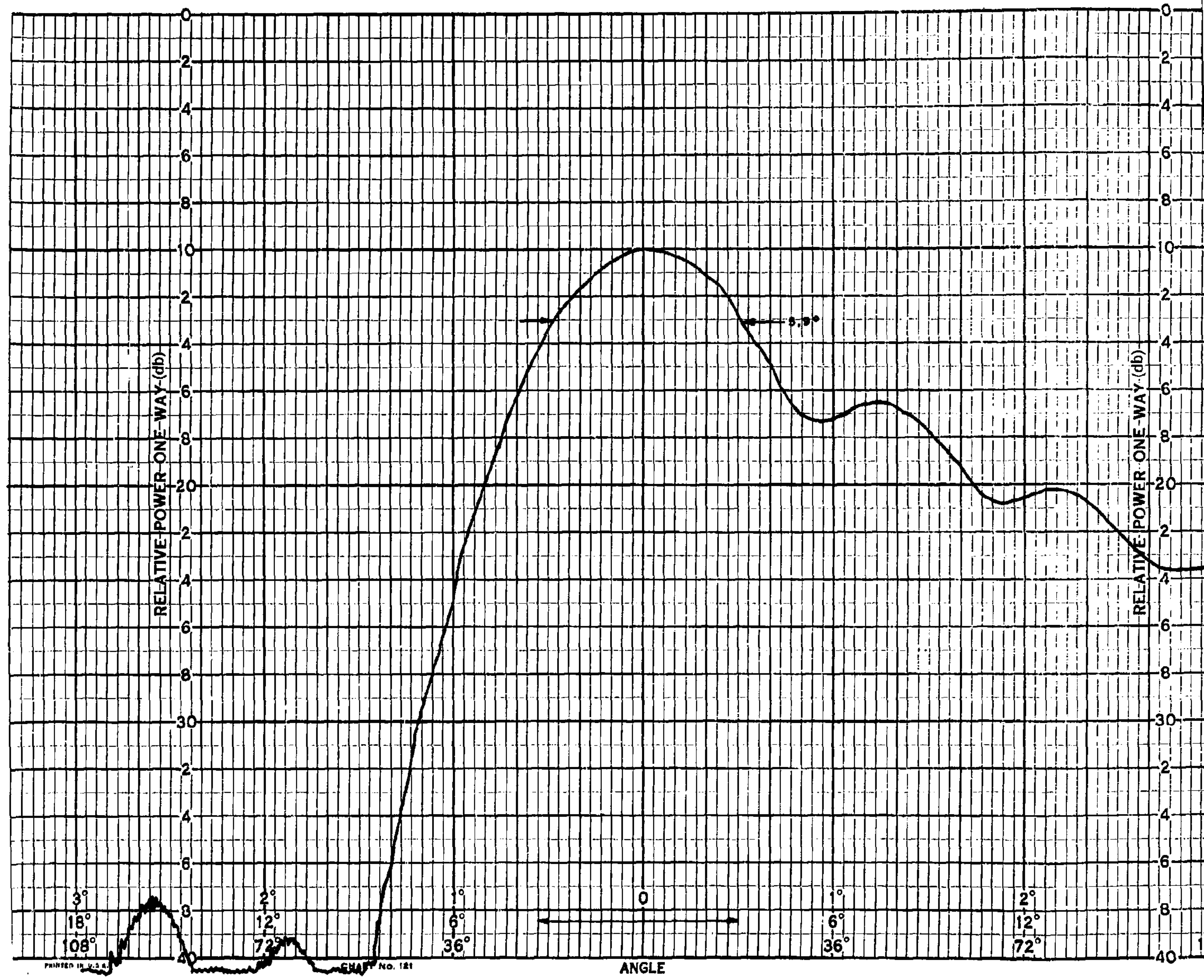

Figure B15. Elevation

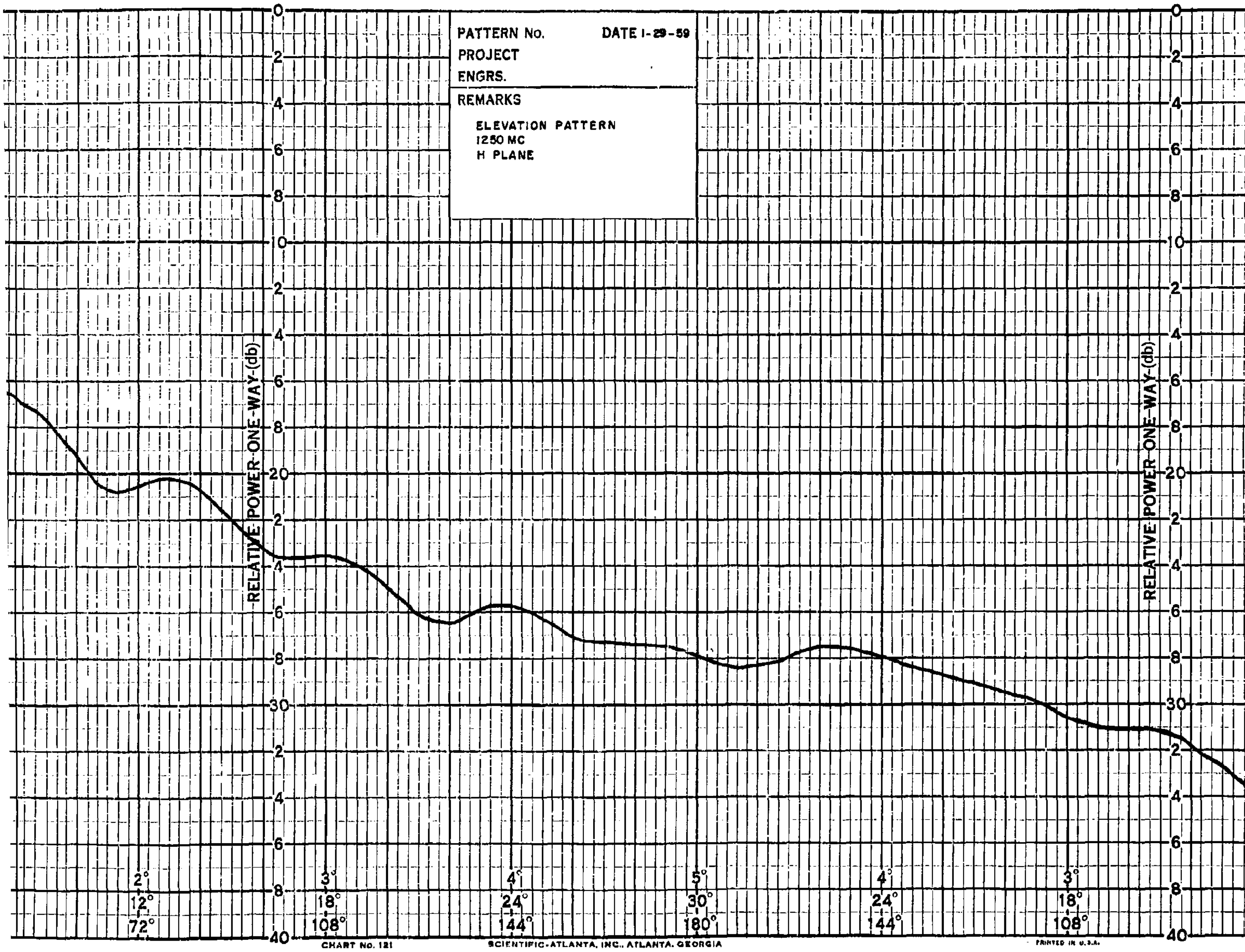

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Figure B15. Elevation Pattern, Search Radar, 1250 Megacycles

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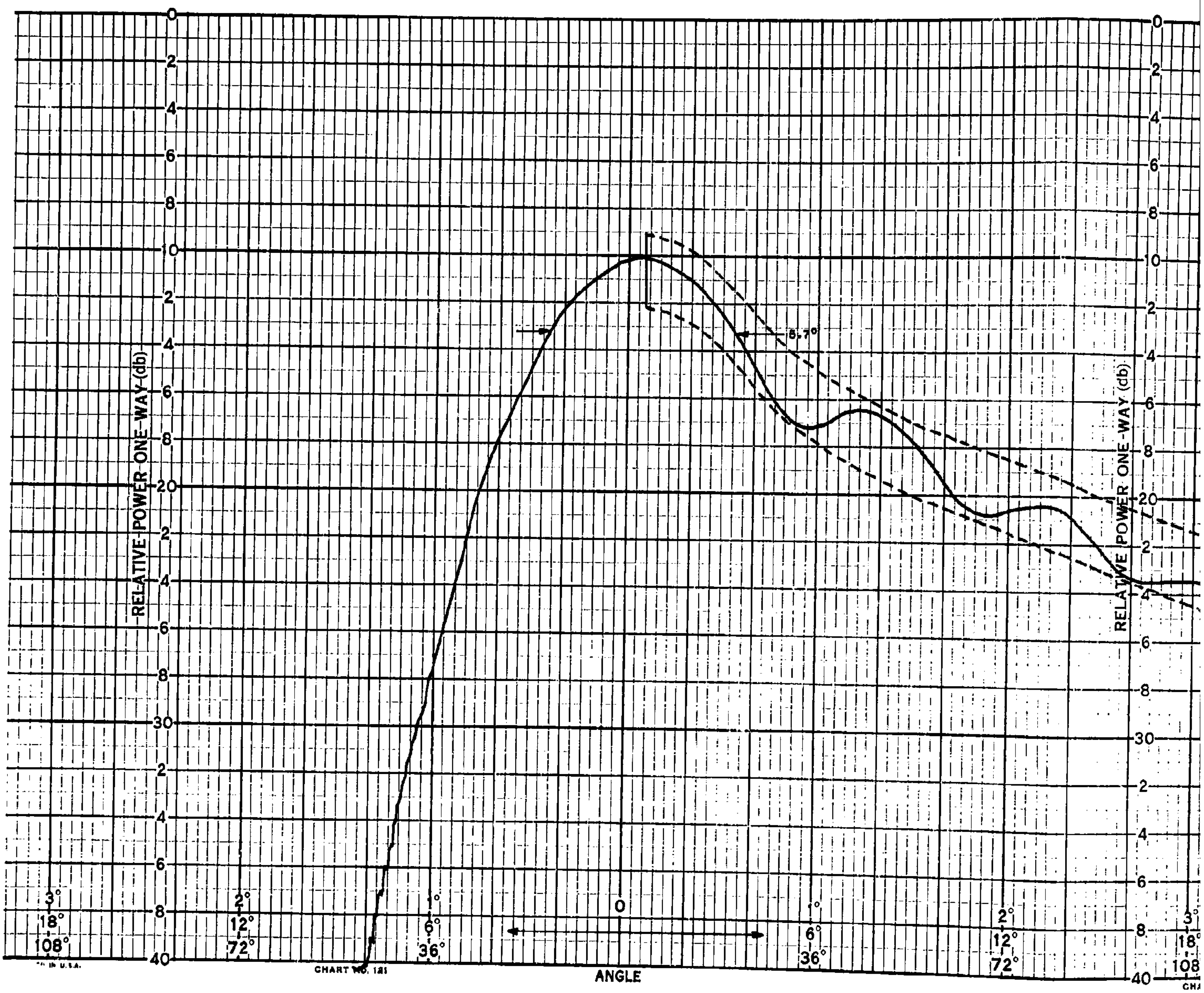

Figure B16. Elevation

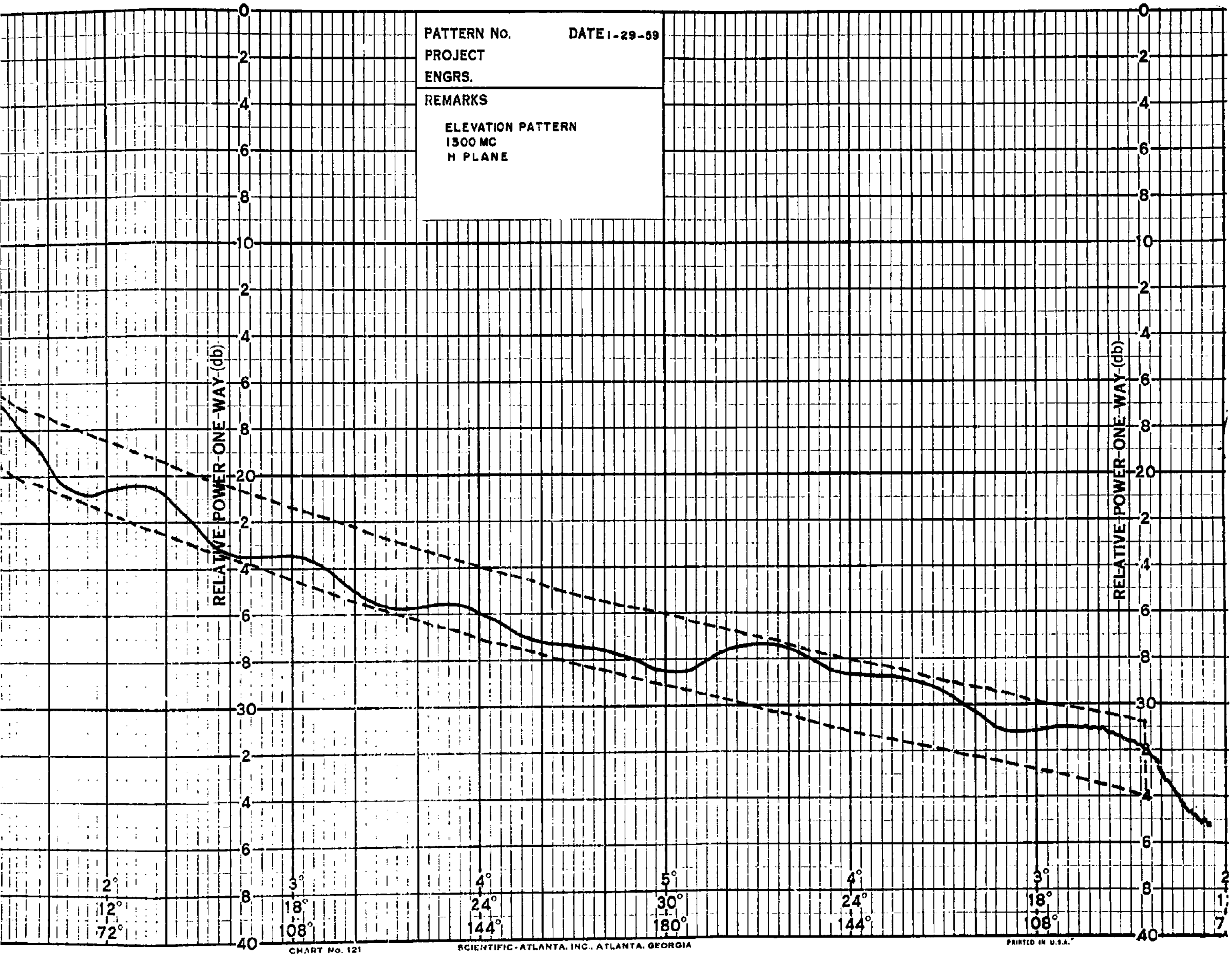

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Figure B16. Elevation Pattern, Search Radar, 1300 Megacycles

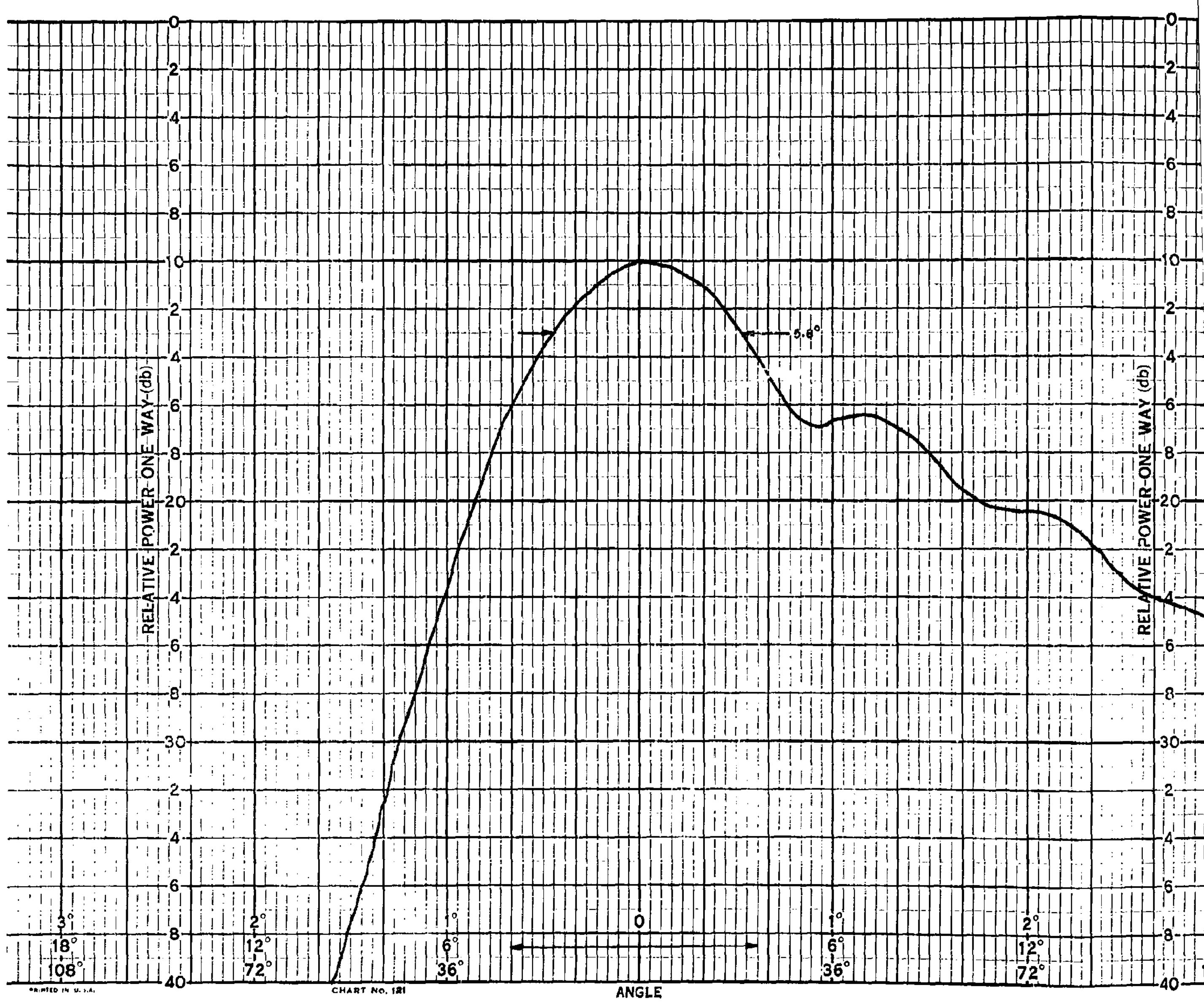

Figure B17. Elevatio

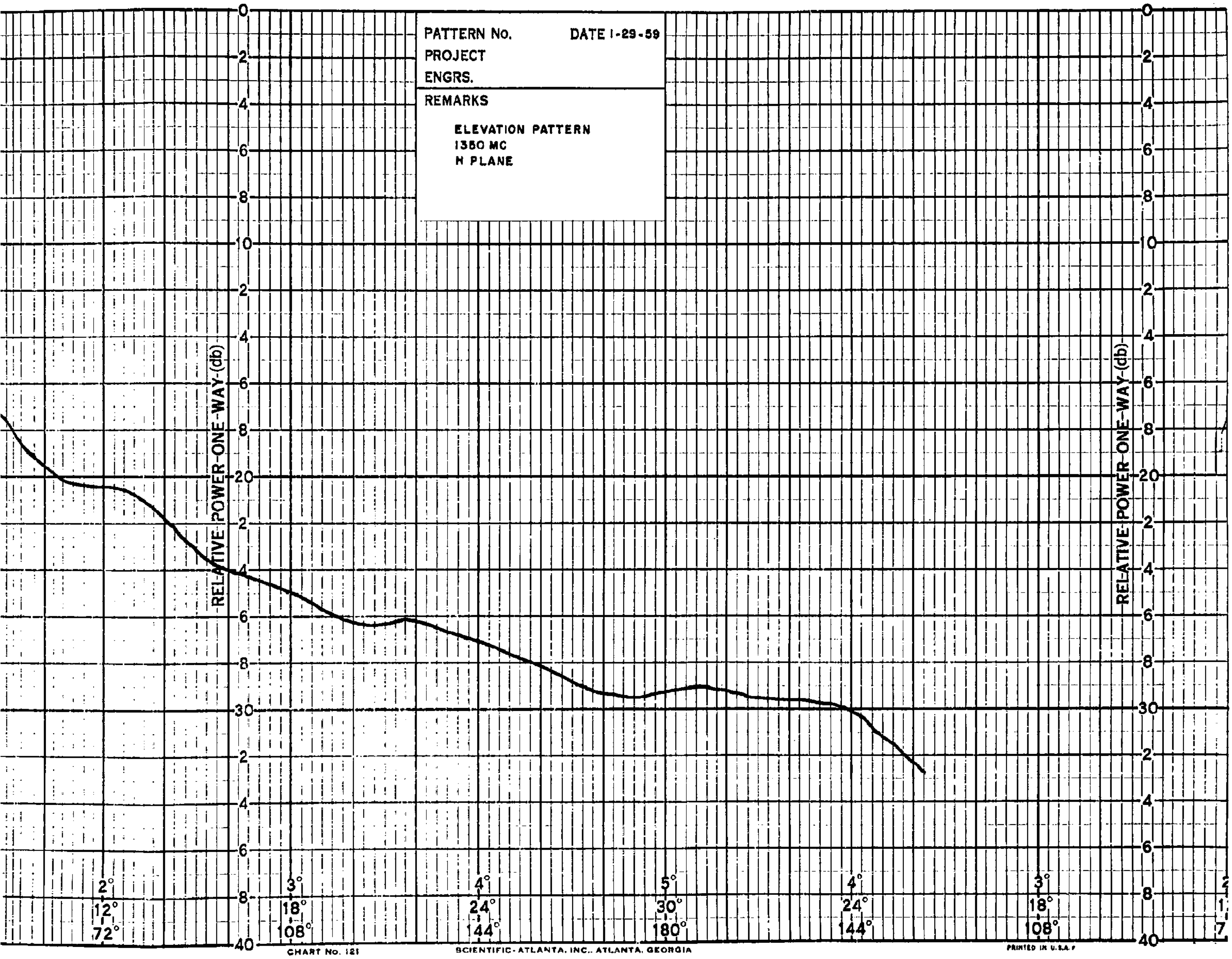

Figure B17. Elevation Pattern, Search Radar, 1350 Megacycles

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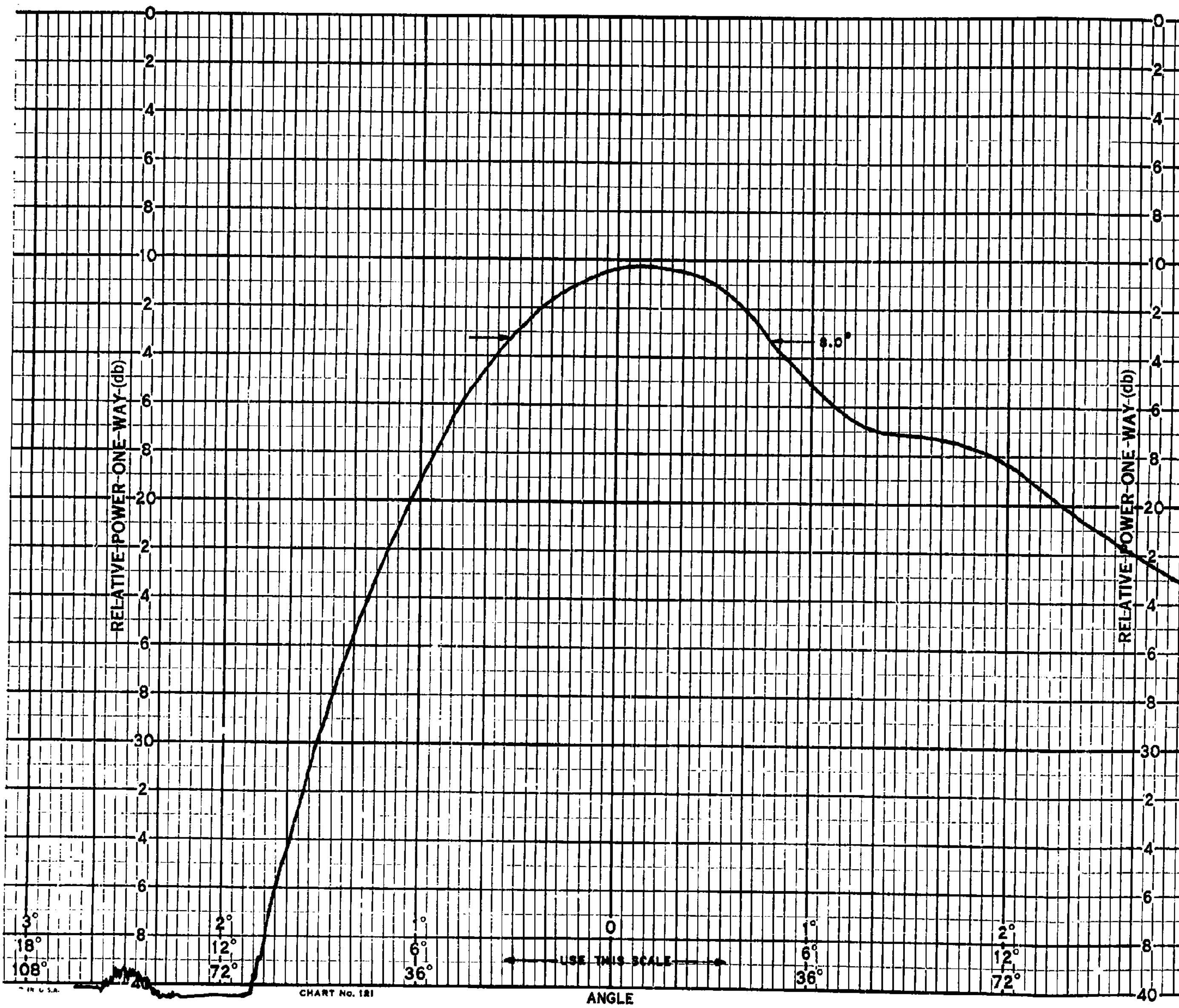

Figure B18. El

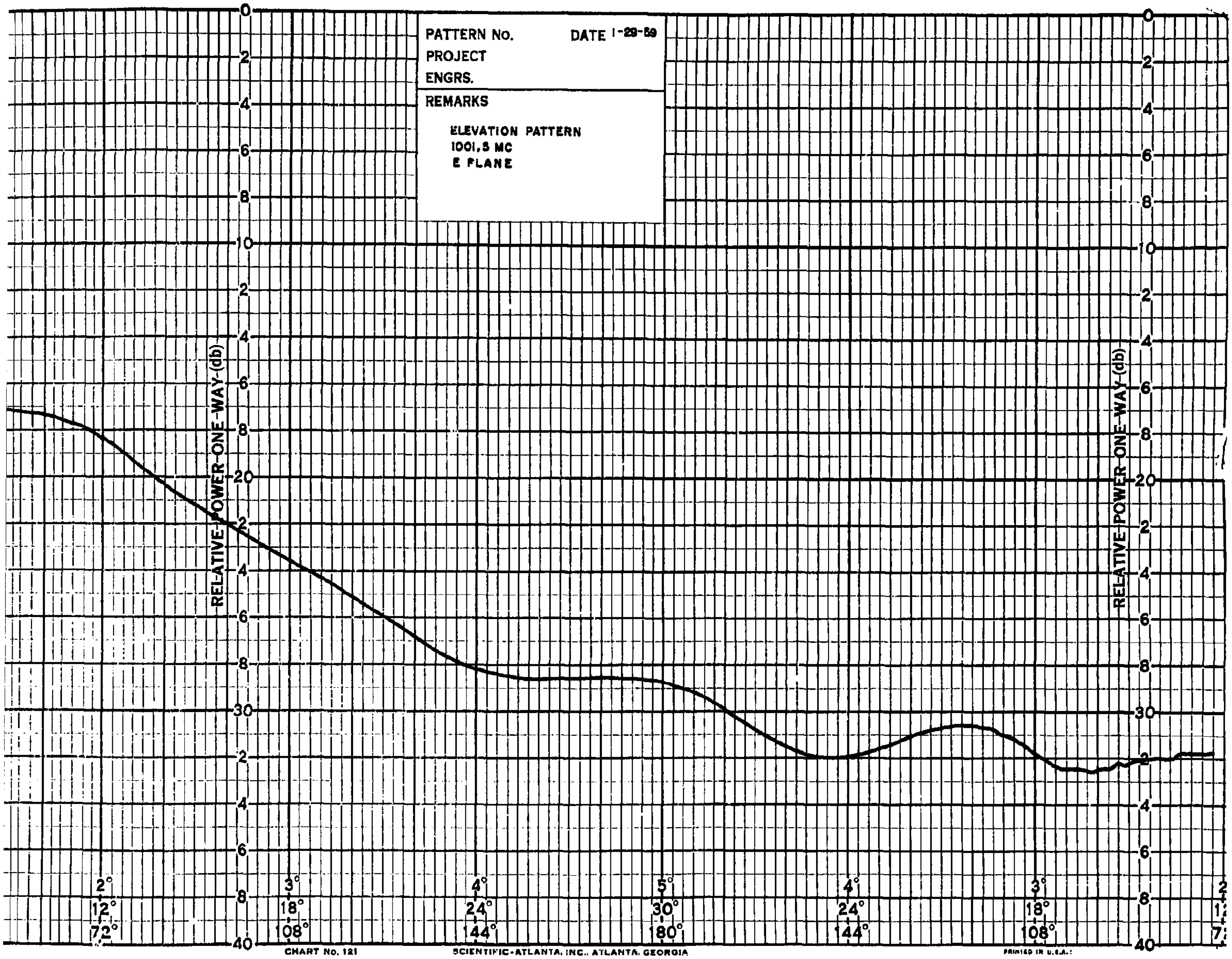

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Figure B18. Elevation Pattern, IFF Radar, 1001.5 Megacycles

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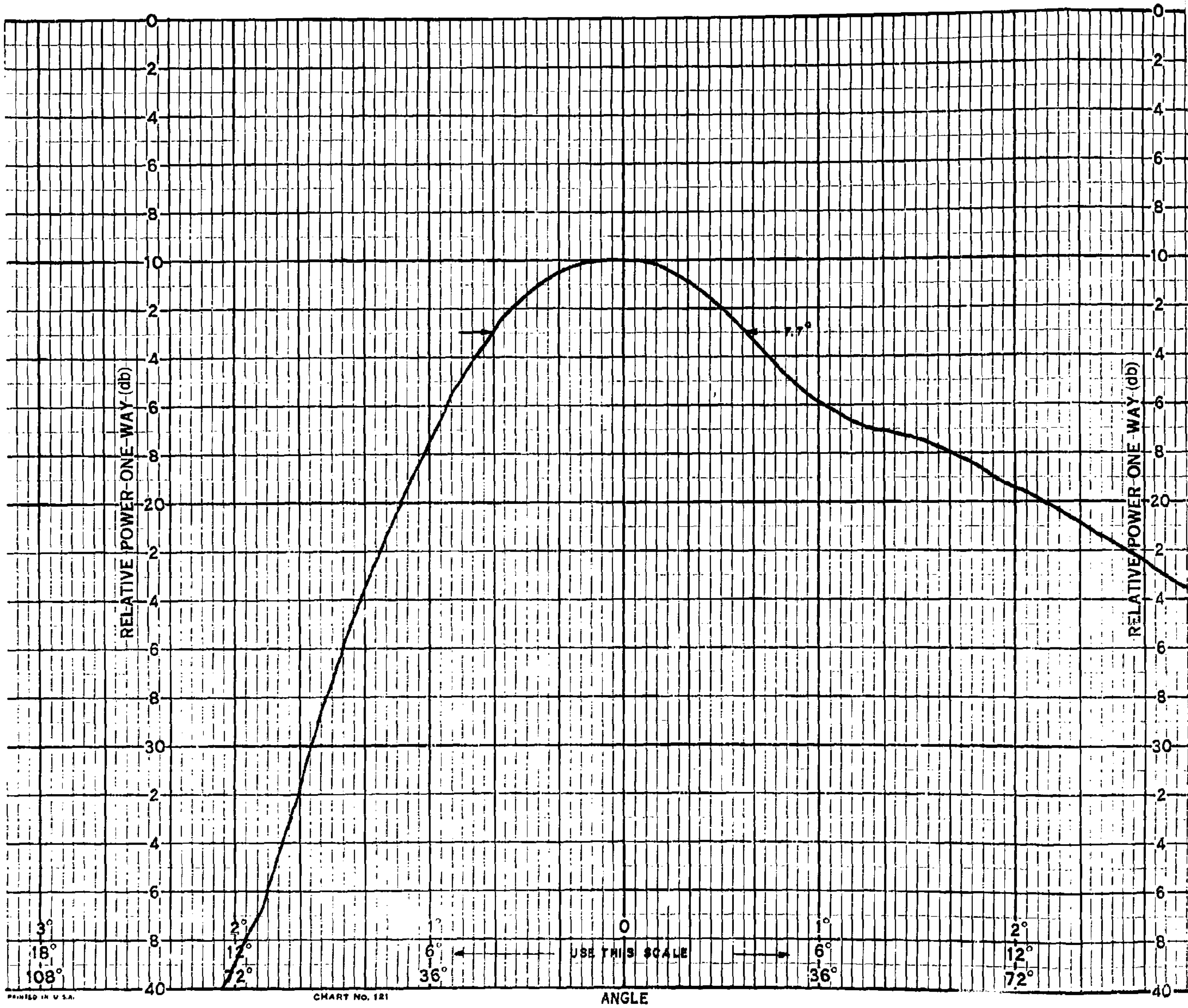

Figure B19. Elevation

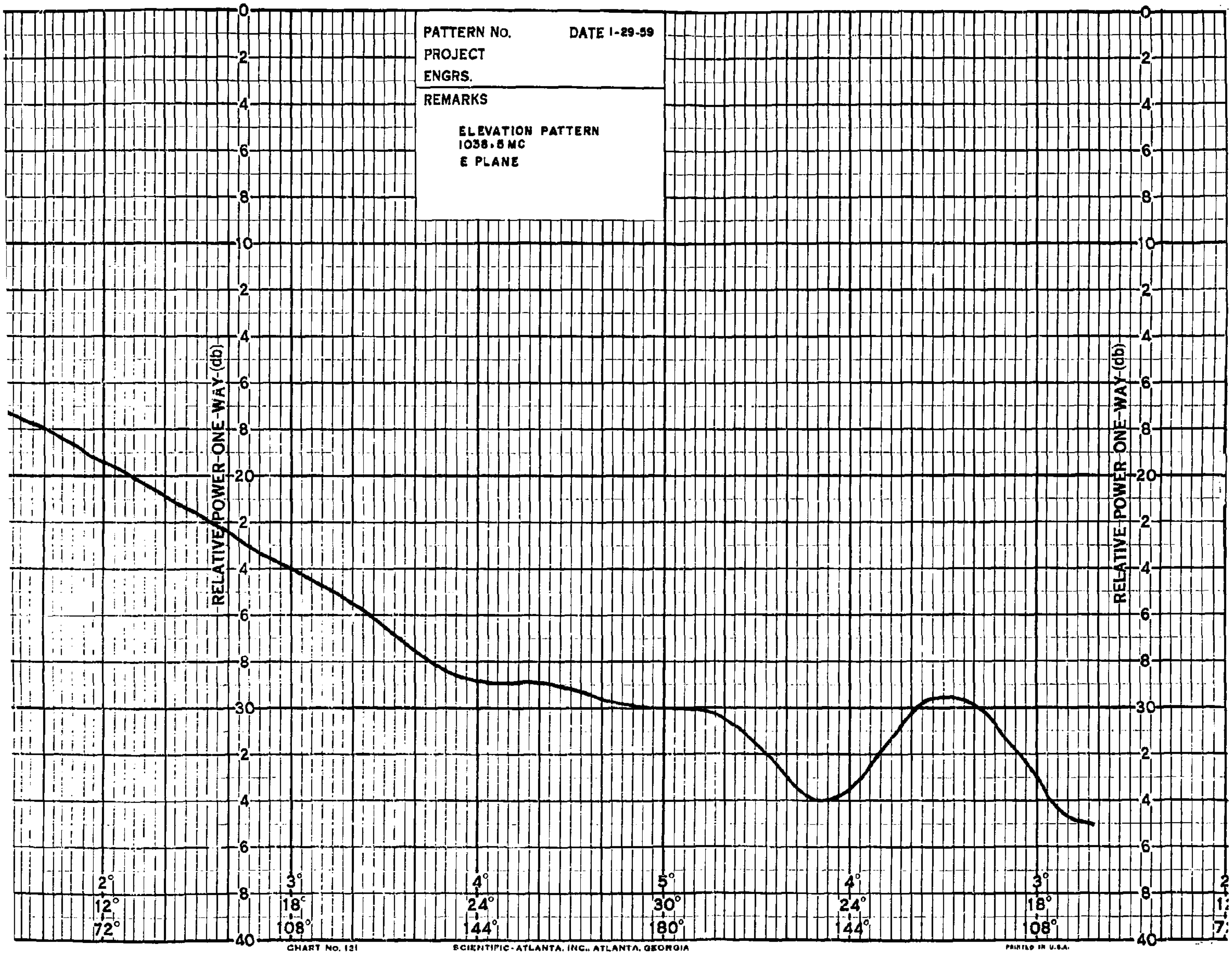

Figure B19. Elevation Pattern, IFF Radar, 1038.5 Megacycles

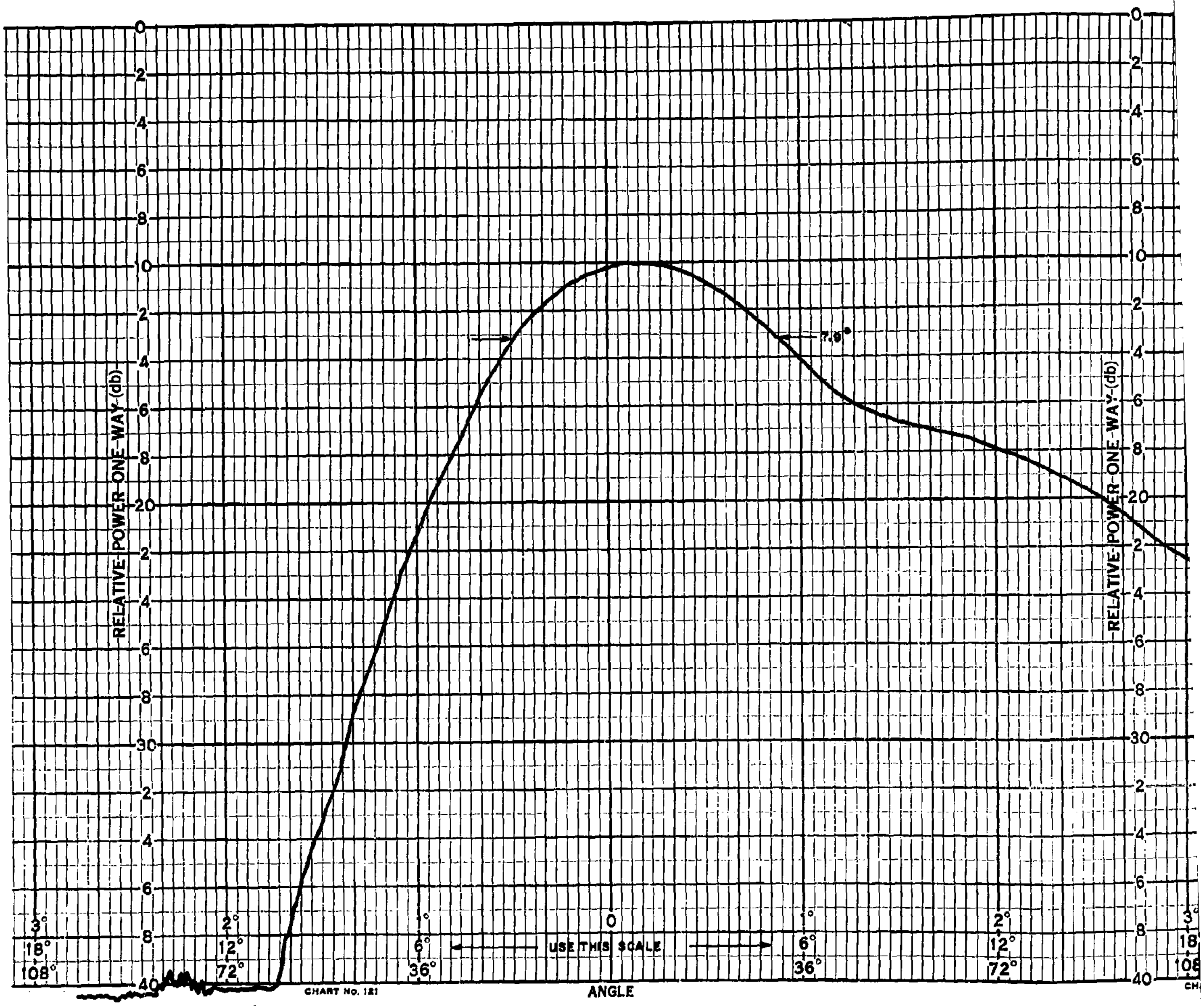

Figure B20. Elevation Pa

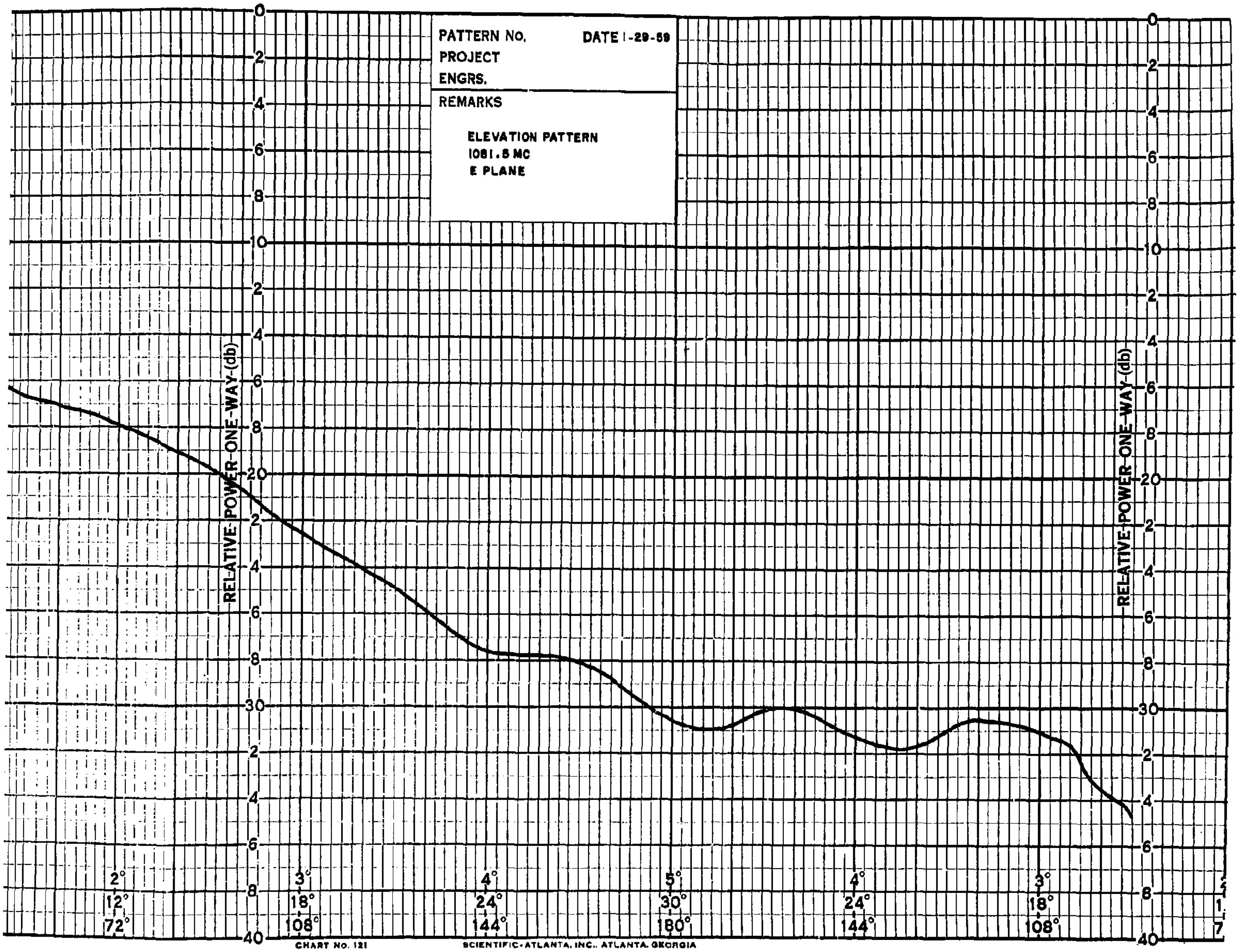

Figure B20. Elevation Pattern, IFF Radar, 1081.5 Megacycles

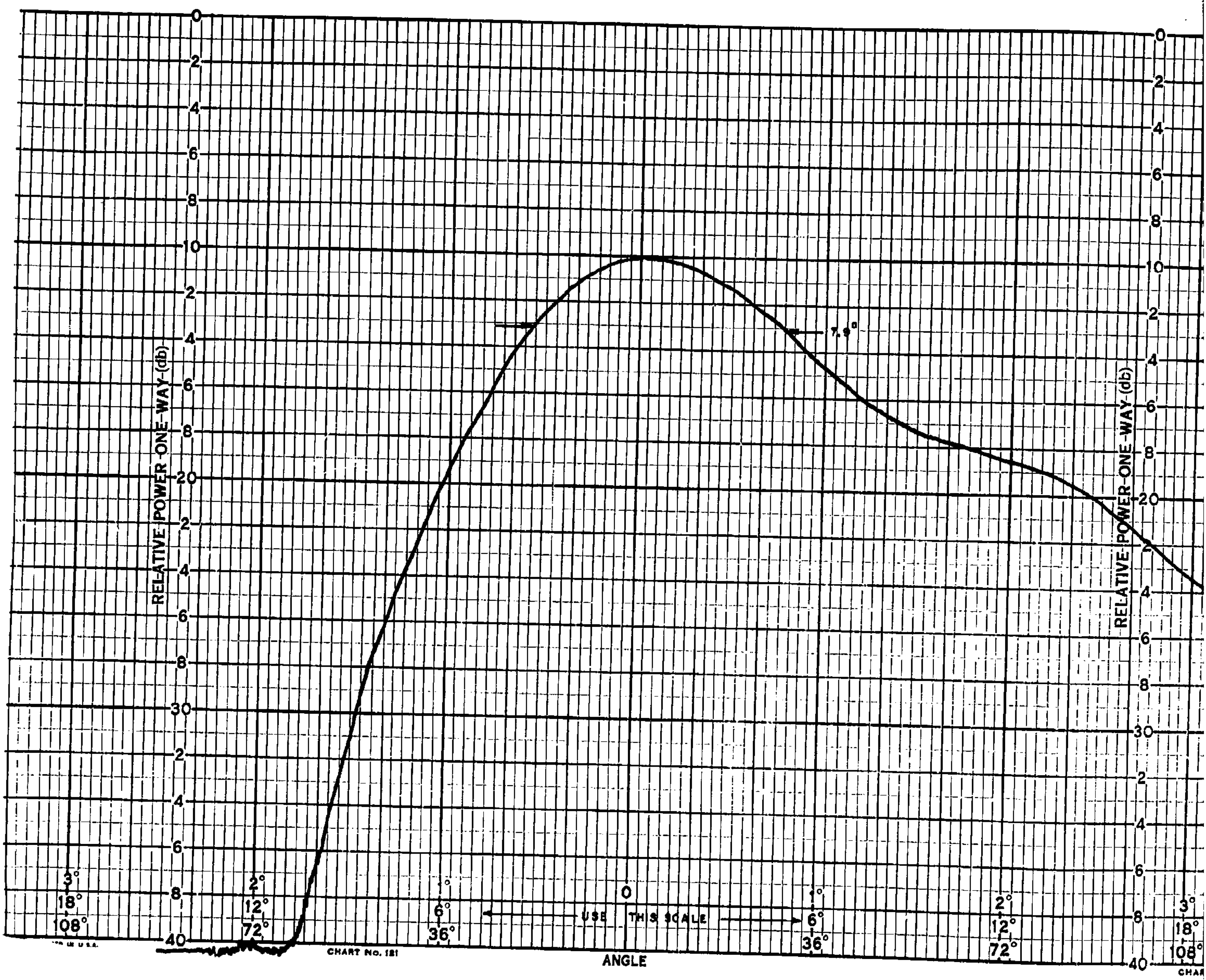

Figure E21. Elevatio

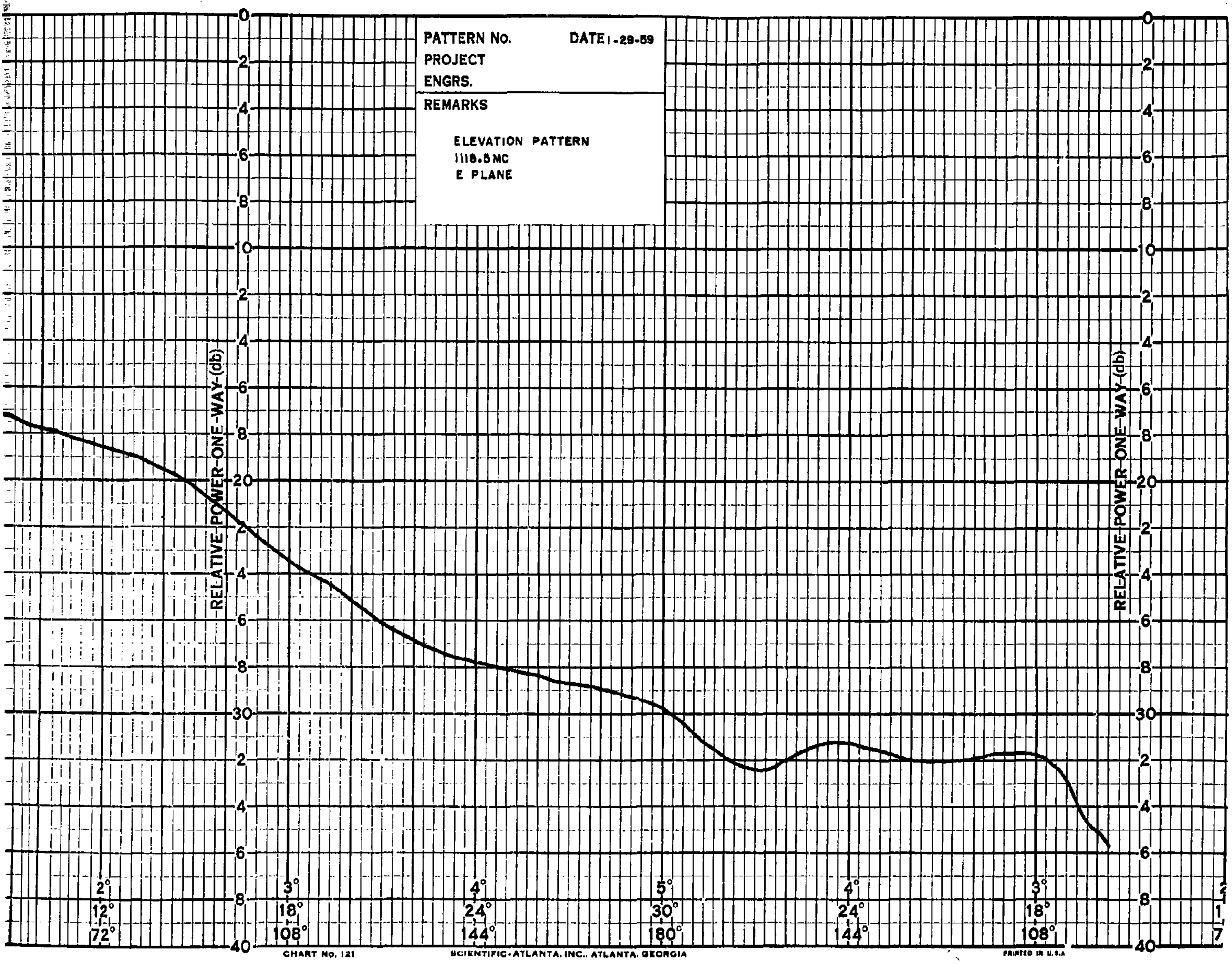

Figure B21. Elevation Pattern, IFF Radar, 1118.5 Megacycles

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ADDENDUM C

ROTARY JOINT HIGH POWER TEST

This test was conducted at the Rome Air Development Center, Rome, New York. The rotary joint was connected as shown in figure C1. The high power radar signal was supplied by a magnetron with an RG-69 waveguide output. The power divider on the output of the magnetron provided power output control. Power was measured with a high-power water-calorimeter load.

A double stub waveguide tuner was connected on the rotary joint output and the water-calorimeter load connected to the waveguide tuner. The output of a test generator was fed, through a 10 db attenuator and a waveguide slotted line, to the input of the rotary joint. The double stub tuner was then adjusted to provide a VSWR of 1.3 in the slotted line. The test generator and slotted line was then removed and the input of the joint connected to the magnetron.

The magnetron trigger circuit was adjusted to produce pulses 4 microseconds wide at a PRF of 225 pulses per second. Therefore, to obtain a peak power of 2 megawatts, an average power of 1800 watts must be obtained, since:

$$P_p = \frac{Pav}{\text{(Pulse width) (Pulse Repetition Rate)}}$$

where, P_p is peak power in megawatts and Pav is average power in watts

then pulse width can be given in microseconds, that is, $P_p = \frac{1800}{(4)\ (225)} = 2$ megawatts

The average power of 1800 watts would be measured on the calorimeter by a temperature difference on the two thermometers of 3.4 degrees C in accordance with the following calorimeter formula:

$$Pav = Q\ (\Delta T)\ 264$$

where Q = water flow (gallons/minute)

ΔT = temperature difference (degrees C)

Pav = average power (watts)

therefore, Pav = (2) (3.4) (264) = 1795 watts

In

P 22 AV 2 R

The rotary joint was evacuated to a gauge pressure of 6.7 psi (representing an altitude of 16,000 feet above sea level). Power was transmitted through the joint for a period of 15 minutes, during which time no adverse conditions were noted.

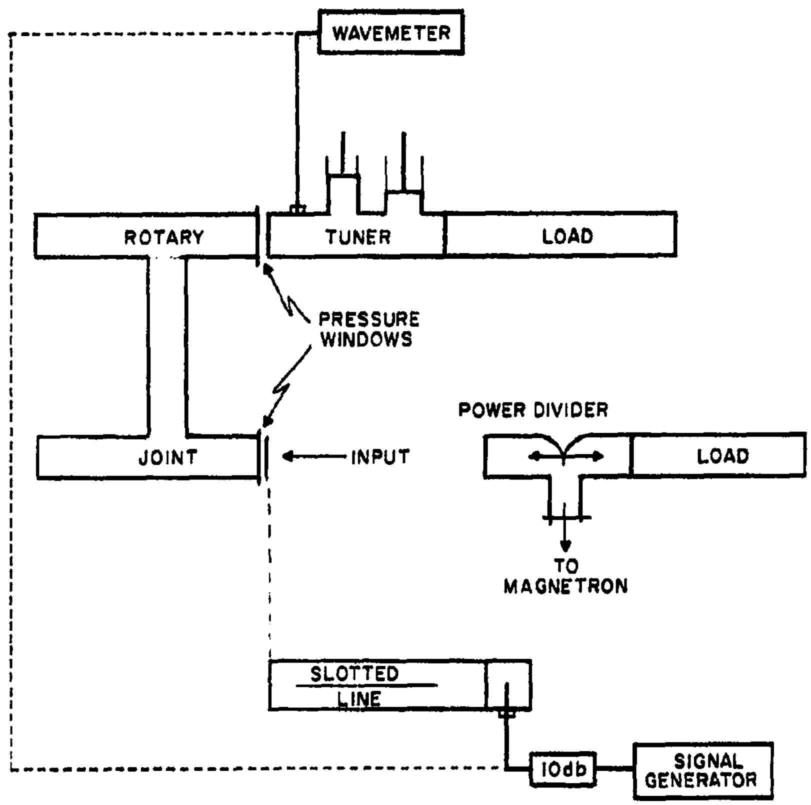

Figure C1. Rotary Joint, High Power Test

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ADDENDUM D

DESIGN OF OVERSIZED "O" RINGS

During pressure tests of the rotary joints, air leakage occurred at the pressure window flanges, with the pressure windows mounted in the normal manner. Flat plates, when substituted for the pressure windows, showed no evidence of leakage. A subsequent investigation of the pressure window and "O" ring groove design revealed that the leakage could be considerably slowed by placing a shim in the groove beneath the "O" ring.

According to manufacturer's design standards, the groove for this size "O" ring (.139 ± .004 inch) should be cut to such depth as to provide a minimum "squeeze" on the "O" ring of 0.025 inch. The groove, being 0.100 inch deep, should provide a minimum of 0.035 inch of "squeeze," more than adequate for proper sealing. However, this assumes that the window acts as a rigid flat plate. Since the leakage has persisted, it has been attributed to some deflection phenomena existing in the pressure window assembly.

In the worst case, the leakage was completely halted by placing a 0.030 inch shim beneath the "O" ring, thereby decreasing the groove depth by this amount. Thus, the theoretical "flat plate squeeze" would be increased to 0.065 inch.

The same amount of "squeeze" may be obtained by increasing the cross sectional diameter of the "O" ring to 0.165 inch and using it in the standard groove of 0.100 inch depth. By using this non-standard size "O" ring, the amount of squeeze required for proper sealing was obtained and the need for an extraneous part, the shim, has been eliminated.

The oversized "O" ring was molded by the Mechanical Rubber Company, Warwick, N. Y. in accordance with the optimum dimensions calculated by Hazeltine Electronic Division as follows:

"O" RING DESIGN CALCULATIONS

Cross Sectional Areas:

1 - Standard sized groove $A_G = .171 \times .100 = 0.0171 \text{ in}^2$

2 - Standard "O" ring (.139 dia.) $A_O = \dfrac{\pi(.139)^2}{4} = 0.0152 \text{ in}^2$

3 - Shims: $A_s \text{ (max.)} = .171 \times .0030 = 0.0051 \text{ in}^2$

$A_s \text{ (med.)} = .171 \times .0025 = 0.0043 \text{ in}^2$

$A_s \text{ (min.)} = .171 \times .0020 = 0.0034 \text{ in}^2$

Ratio of Excess Rubber to Available Space (R):

1 - **Maximum case:**

$$R = \frac{A_O - (A_G - A_S\,(max.))}{A_G - A_S} = \frac{0.0152 - (0.0171 - 0.0052)}{0.0171 - 0.0052}$$

$$R = \frac{0.0152 - 0.0120}{0.0120} = \frac{0.0032}{0.0120} = 27\%$$

2 - **Median case:**

$$R = \frac{0.0152 - 0.0128}{0.0128} = \frac{0.0024}{0.0128} = 19\%$$

3 - **Minimum case:**

$$R = \frac{0.0152 - 0.0137}{0.0137} = \frac{0.0015}{0.0137} = 11\%$$

SYNTHESIZED DESIGN CRITERIA

Method I: Design oversized "O" ring for median case of excess rubber to available space ratio.

$A_G = 0.0171$ in.^{2} $\quad R = 19\%$

$$A_O = A_G + 19\%\,A_G = A_G\,(1 + 0.19) = (0.0171)\,(1.19)$$

$$A_O = 2.04 \times 10^{-2}\ \text{in.}^2 = \frac{\pi d^2}{4}$$

$$d = \sqrt{\frac{4\,A_O}{\pi}} = \sqrt{\frac{(4)(2.04 \times 10^{-2})}{3.14}} = \sqrt{2.6} \times 10^{-1}$$

$d = 0.160$ inch

Method II: Design oversized "O" ring for median case where its cross sectional area is equal to the total cross sectional area of the standard "O" ring and median shim.

$$A_O = A_O + A_S(\text{med.}) = 0.0152 + 0.0043 = 0.0195 \text{ in}^2$$

$$d = \sqrt{\frac{(4)(1.95 \times 10^2}{3.14}} = \sqrt{2.48} \times 10^{-1} = 1.575 \times 10^{-1}$$

$$d = 0.158 \text{ inch}$$

The diameters computed in both methods I and II are approximately equal and should constitute the median size required for sealing the pressure windows. Therefore, a diameter of 0.150 ± .005 inch has been selected for the oversized "O" ring.

Subsequent air pressure immersion tests, using the oversized "O" ring in conjunction with the rotary joint and Airtron pressure windows, showed positive results.

In all cases they:

1. Fit the groove quite well without extruding when the flange bolts were taken up tightly.

2. Effectively eliminated the leakage of air through the pressure windows. This fact is supported also by the results of the heat run conducted on antenna pedestal serial number 3, in which the rotary joint was sealed at the pressure window flanges with the oversize "O" rings.

During the 24-hour period from 10:30 A.M. 9/26/60 to 10:30 A.M. 9/27/60, the total pressure drop due to air leakage was only 0.15 psi, well within the specified limit of 1.68 psi/day.

PERFORMANCE DATA SHEET
OA-1227 ANTENNA PEDESTAL
SERIAL NUMBER 3, 24-HOUR HEAT RUN

Time	Temp. °C	Pressure (PSIA)
10:30	17	15.05
11:30	19	15.1
12:30	21	15.4
1:30	22	15.6
2:30	24	15.8
3:30		
4:30	26	16.0
5:30	25	16.0
6:30	24	15.9
7:30	23	15.7
8:30	20	15.4
9:30	20	15.2
10:30	19	15.1
11:30	17	15.0
7:30 am	18.5	15.0
8:30	17	14.9
9:30	17	14.9
10:30	17	14.9

Results

Since the terminal temperature readings are equal, the air leakage loss is the difference in the terminal pressures, no temperature correction being required.

$$\frac{\text{P initial} - \text{P final}}{\text{Elapsed time}} = \text{Leakage Rate}$$

$$\frac{15.05 \text{ psig} - 14.90 \text{ psig}}{24 \text{ Hours}} = 0.15 \text{ PSI/Day}$$

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ADDENDUM E

ROTARY JOINT, ANTENNA GROUP OA-1227(XN-1)/TPS
SUMMARY OF 24 HOUR CONTINUOUS PRESSURE TEST

In compliance with paragraph 3.3.2.10 of SCL-5296, provisions have been made for pressurizing the rotating joint to 30 pounds per square inch, absolute pressure.

The rotary joint has been tested while continuously rotating over a period of 24 hours, and its leakage rate calculated as follows:

The initial air pressure within the system (p_1) was 15.0 psig and the air temperature (t_1) was 29.3°C. At the conclusion of the test (p_2) was 13.7 psig and (t_2) was 26.5°C. Although the pressure gauge readings show a pressure drop of 1.3 psi, it must be noted that there was also an accompanying drop in temperature. In correcting for the temperature differential, the readings must all be converted to absolute scales. Thus:

$$P_1 = p_1 + 14.72 = 29.72 \text{ psia} \qquad T_1 = t_1 + 273 = 302.3^\circ K$$

$$P_2 = 13.7 + 14.72 = 28.42 \text{ psia} \qquad T_2 = 26.5 + 273 = 299.5^\circ K$$

then the ideal final pressure ($P_2{}^1$) corresponding to (T_2) is calculated using the constant volume equation for changing conditions in one gas.

$$P_2{}^1 = \frac{P_1 T_2}{T_1} = \frac{(29.72 \text{ psia})\,(299.5^\circ K)}{302.3^\circ K} = 29.45 \text{ psia}$$

By subtracting the actual final pressure gauge reading from this value, the amount of air leakage in psi may be found. Hence:

$$\text{Air Leakage} = P_2{}^1 - P_2 = 29.45 - 28.42 = 1.03 \text{ psi.}$$

Article 18 of the second amendment (February 2, 1958) to paragraph 3.3.2.10 specifies: "The leakage rate for the joint shall not allow a loss of pressure in excess of 0.4 psi in a 24 hour interval." Obviously the tested leakage of 1.03 psi was much greater than the 0.4 psi per day allowed by the above amendment. In view of the basic design factors influencing the performance of the joint, it was Hazeltine's contention that the above specification is overly stringent and disproportionate with reasonability of design. Following are some of the design limitations which must be respected:

1. By stiffening the bellows, which would increase the sealing pressure, the leakage could be reduced. However, there is a maximum limiting surface pressure at which seal wear life is sacrificed for a little less leakage. We have designed our rotary seal bellows joint to function safely below this limit.

2. Conceivably, the fine finish on the sealing surfaces (flat within two light bands) could be made finer, but a harder, closer grained material would be needed. A compromise was effected here by using a high grade graphite ring with superior qualities of long wear and natural lubrication.

3. Another design factor involved is that of the total life of the air tank at a given leakage rate. Rather than impair the effectiveness of the rotary joint by obliging it to operate under critical conditions, Hazeltine chose to design for a tank with a longer service life, and a substantial factor of safety for pressure loss. The air tank has a capacity of 386 cu. inches, and a service pressure of 2100 psig. Assume that during the service life of the tank, that its pressure will drop from 2100 psig to equalize with that of the system at 30 psia.

Air Tank	Rotary Joint
$P_1 = 2100$ psig $= 2115$ psia	$P_2 = 30$ psia
$V_1 = 386$ in^3	$V_2 = 750$ in^3

Assume that atmospheric pressure = 15 psia

Since the air will stop exhausting from the tank when its pressure reaches equilibrium with that of the rotary joint its pressure differential (P_d) will be 2115 psia - 30 psia = 2085 psi

$$P_d V_d = P_2 V_2$$

$$V_d = \frac{P_d V_1}{P_2} = \frac{(2085 \text{ psi})(386 \text{ in}^3)}{30 \text{ psia}} = 26{,}800 \text{ in}^3$$

V_d is the volume that the air in the tank would occupy at 30 psia inside the rotary joint. Initially the rotary joint contains 750 in^3 air at 15 psia which is equivalent to 375 in^3 at 30 psia. Therefore, at 30 psia, the air tank is capable of completely filling the rotary joint 35 times.

$$\frac{26800 - 375}{750} = \frac{26425}{750} = 35.25$$

At the specified leakage rate of 0.4 psi per day, which is equal to 10 cubic inches of air lost from the rotary joint per day, the rotary joint may be completely exhausted in 1800 hours.

$$\frac{(0.4 \text{ psi/day})\ (750 \text{ in}^3)}{30 \text{ psia}} = 10 \text{ in}^3\text{/day air lost}$$

$$\frac{750 \text{ in}^3 \times 24 \frac{\text{hrs}}{\text{day}}}{10 \text{ in}^3\text{/day}} = 1800 \text{ hours}$$

Since the rotary joint will never exhaust below atmospheric pressure (15 psia) only one half of its volume of air will leak out. Thus the time to exhaust the rotary joint to equilibrium with the atmosphere will be 900 hours at the specified rate of leakage.

$$1800 \text{ hours/vol.} \times \frac{\text{vol.}}{2} = 900 \text{ hours}$$

Thus when the rotary joint is being filled to 30 psia, the tank must supply one-half of its volume of air. The tank has already been shown to be capable of completely filling the rotary joint 35 times. Therefore, it is capable of half-filling the rotary joint 70 times.

Thus the tank would become exhausted to equilibrium with the rotary joint at 30 psia in:

$$L = 900 \text{ hours} \times 70 = 63{,}000 \text{ hours}$$

Hence at the specified leakage rate of 0.4 psi/day, the tank is capable of a service life of 63,000 hours running time.

As specified in paragraph 3.7.2(a) of SCL-5296, the overhaul life of the unit is 5,000 hours of operation at 23 hours per day. By comparing the two figures, it becomes apparent that the tank life, at the specified allowable leakage rates is 12.6 times greater than the overhaul life of the unit.

At the tested rate of 1.03 psi per day, the air tank life would be:

$$L = \frac{63{,}000 \times 0.4}{1.03} = 24{,}450 \text{ hours}$$

which is 4.9 times the overhaul life.

To account for the inevitable variations in resilience and finish among the seals (1 ± 15% tolerance in surface pressures) a leakage factor of 30% may be added, assuming that the optimum seal, as tested, leaks at a rate of 1.03 psi per day. The leakage rate would then be:

$$r = 1.03 + (0.30)(1.03) = 1.34 \text{ psi/day}$$

A consideration must be made of the varying conditions encountered in operating and maneuvering in the field. The unit is designed to operate continuously for 23 of 24 hours per day for 5000 operating hours. We realize, however, that there will most likely be periods of intermittent shut down when the equipment isn't operating, but the seal is under pressure and will lose air through static leakage. Then too, it may at time become necessary to reposition the unit in the field for tactical reasons. This would necessitate closing off the tank air supply and breaking the unit down for transportable packaging. During transportation, the unit would be subjected to extended and irregular vibration, enhancing the possibility of additional leakage. Thus, a liberal factor of safety for pressure loss, as we now have, is advisable to offset the possibility of unanticipated loss of air; and as well, the adverse leakage affects of operation in elevated tropical temperatures.

To account for these variable factors, the leakage rate has been further increased by 25% giving a figure of:

$$r = 1.34 + (.25)(1.34) = 1.68 \text{ psi/day}$$

The tank life at this rate would be

$$L = \frac{63,000 \times 0.4}{1.68} = 15,000 \text{ hours}$$

The factor of safety for tank life over equipment overhaul life is now:

$$S.F. = \frac{15,000 \text{ hours}}{5,000 \text{ hours}} = 3.0$$

A realistic evaluation of the leakage problem has shown that the use of 1.68 psi/day as a criterion for rotary joint seal performance is of great advantage. This allows the rotary joint to operate effectively within its range of optimum conditions, while affording a liberal safety factor on leakage loss of 3.0.

Pursuant to this recommendation, article 18 of amendment 2 to paragraph 3.3.2.10, SCL-5296 was accordingly revised by, and upon approval of, the Signal Corps.

REPORT 10075 ADDENDUM F

CALCULATION OF HORSEPOWER REQUIRED TO ROTATE ANTENNA

Assuming the antenna to be a flat plate of the dimensions shown below, the horsepower may be calculated using the formula:

$$HP = .457\ N^2\ \rho\ C_D\ VD^4 \times 10^{-6}\ *$$

where: N = rotational speed (rpm)

D = length of antenna (ft)

V = wind velocity (ft/sec)

C_D = drag coefficient

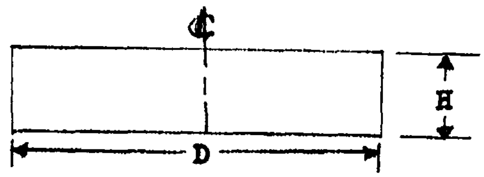
ρ = air density (slugs/ft^3)

For operating conditions:

V (max.) = 60 mph = 88 ft/sec

ρ (-65°F) = 0.00312 slugs/ft^3

C_D (noise) = 0.53



H = 11 ft.
D = 40 ft.
N = 6 rpm

$$HP = (.457)(6)^2\ (.00312)(.53)(88)(40)^4 \times 10^{-6}$$

$$HP = 6.15$$

The azimuth drive efficiency is assumed to be 85%

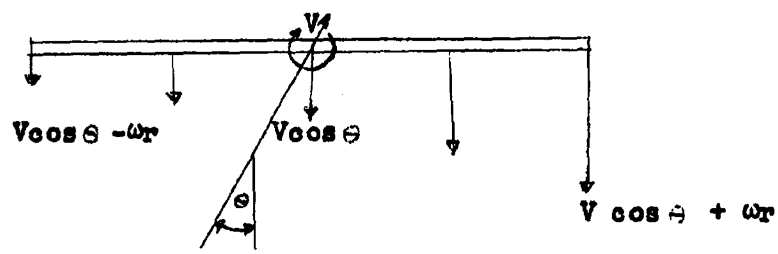
then $HP = \dfrac{6.15}{.85} = 7.24$

Using next higher standard size, HP = 7.5 HP

Derivation of Wind Torque Equation

(a) Wind Velocity Distribution:

A flat plate rotating about its center has a relative wind velocity distribution as shown below:



where: ω = rotational speed of plate (radians/sec)

V = wind velocity (ft/sec)

θ = angle of incidence of wind from perpendicular(°)

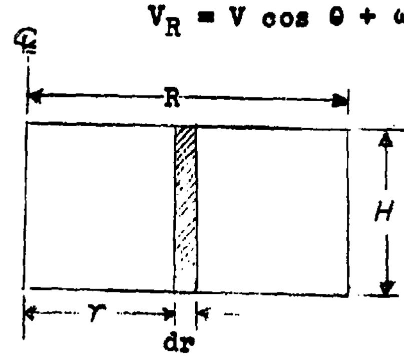
R = radius of plate (ft.)

Left side: wind aids rotation (negative torque)

$$V_L = V \cos \theta - \omega r$$

Right side: wind inhibits rotation, effects are additive

$$V_R = V \cos \theta + \omega r$$



Area:

$$A = \int_0^R H\,dr$$

(b) Torque (T):

$T = F\ r$ where: r = elemental radius

F = drag force on plate

Drag force:

$$F = 1/2\ \rho\ C_D\ AV^2$$

where: C_D = drag coefficient

ρ = air density (slugs/ft.^{3})

A = area of plate (ft.^{2})

V = wind velocity (ft/sec)

$$T = 1/2\ \rho\ C_D\ AV^2\ r$$

Left side: $V_L = V \cos \Theta - \omega r$ $\quad A \int_o^R H\ dr$

$$T_L = -1/2\ \rho\ C_D\ (\int_o^R H\ dr)\ (V \cos \Theta - \omega r)^2\ (r)$$

Right side: $V_R = V \cos \Theta + \omega r$

$$T_R = 1/2\ \ C_D\ (\int_o^R H\ dr)\ (V \cos \Theta + \omega r)^2 (r)$$

Total torque: $T = T_L(-) + T_R(+) = T_R - T_L$

$$T = 1/2\ \rho\ C_D\ (\int_o^R H\ dr)(r)\ \left[(V\cos \Theta + \omega r)^2 - (V \cos \Theta - \omega r)^2\right]$$

$$T = 1/2\ \rho\ C_D\ (\int_o^R H\ dr)(r)\ (4 \cos \Theta\ \omega r)$$

$$T = 2\ \rho\ C_D\ H\ V \cos \Theta\ \omega \int_o^R r^2\ dr$$

$$T = \frac{2}{3}\ \rho\ C_D\ H\ V \cos \Theta\ \omega\ R^3$$

Assume wind acting perpendicular to plate:

Angle of incidence $\theta = 0$

$\cos \theta = \cos 0^{\circ} = 1$

Substituting:

$R = \frac{D}{2}$ (D = total length)

$H = \frac{11}{40} D$

$\omega = \frac{2\pi N}{60}$ (N = rpm)

$T = 2.4\ N \rho C_D V D^4 \times 10^{-3}$

(c) Horsepower

$HP = \frac{2\pi NT}{33,000}$ substituting for T

$$HP = .457\ N^2 \rho C_D V D^4 \times 10^{-6}$$

ADDENDUM G

GEAR CALCULATIONS

1. AZIMUTH DRIVE GEAR DESIGN DATA SUMMARY AND CALCULATIONS

Gear Fig. G1	D.P.	P.D. in	No. Teeth	Face Width in.	Beam Strength W_b(lbs.)	Dynamic Load W_d(lbs.)	Limit Load for Wear W_w(lbs.)	R.P.M.	M.S. $\frac{W_b}{W_d}$
A	5	32.0	160	2.50	19,220	8200	9850	6.1*	2.35
B	5	4.0	20	2.75	12,750	8200	9850	49	1.56
C	8	12.5	100	2.00	7,810	2922	3120	49	2.67
D	8	2.50	20	2.25	5,610	2922	3120	245	1.92
E	10	9.1	91	1.50	4,610	1760	1940	245	2.52
F	10	2.6	26	1.75	3,630	1760	1940	862.5	2.06
G	16	5.0	80	1.25	2,365	1470	1520	862.5	1.61
H	16	2.5	40	1.50	2,130	1470	1520	1725	1.45

*The output speed of 6.1 RPM is a result of the choice of gear ratios. Although the nominal specified output is 6.0 RPM, the design speed of 6.1 RPM is within the requirements of the design specification.

AZIMUTH DRIVE GEAR CALCULATIONS

The design of the gear drive system is based on the following parameters:

Drive motor to be 7.5 H.P. at 1725 RPM, stall H.P. 15 (Ref: Drive motor calculations)

All gears to be 20° P.A. Spur, for maximum load capacity at minimum cost.

All gears except gear "A" to be made of SAE 4140 alloy steel, heat treated and surface hardened as shown in table above.

Gear "A" to be made of SAE 8660.

Tooth beam strength to be based on stall (maximum) torque of the drive motor.

Dynamic load and limit wear load to be based on 150 percent (approximately) of rated H.P.

Face widths to be approximately 5 times the circular pitch of the gear. (Ref: Machinery's Handbook.) Gear "B" will be crown shaved and shaft parallelism will be closely controlled to insure full tooth mating.

All gears to be class 3 commercial in order to obtain the required precision at minimum cost.

Overall gear reduction is 287.5:1

a. DYNAMIC LOAD CALCULATIONS: (MAXIMUM OPERATING LOADS)

(1) Dynamic load calculations are based on the following formula. (Ref: Manual of gear design-E. Buckingham.)

(2) Gear designations as per figure G1 and table G1.

$$W_d = \frac{.05\ V\ (FC + W)}{.05\ V + \sqrt{FC + W}} + W$$

Where: V = Pitch line velocity in F.P.M. =

$$\frac{\pi \times P.D. \times R.P.M.}{12}$$

F = Face width of gear in inches

C = Deformation factor (A constant based on gear accuracy, tooth form and modulus of elasticity.) (From chart in Buckingham.)

W = Total applied load = $\frac{33,000 \times H.P.}{V}$

GEARS A and B

Where

$W = 7.9 \times 10^3$ lbs.

$V = 50.3$ F.P.M.

$F = 2.5$ in.

$C = 1660$

$$W_d = \frac{.05\ V\ (FC + W)}{.05\ V + \sqrt{FC + W}} + W$$

$$= \frac{.05 \times 50.3(2.5 \times 1660 + 7.9 \times 10^3)}{.05 \times 50.3 + \sqrt{2.5 \times 1660 + 7.9 \times 10^3}} + 7.9 \times 10^3$$

$$= 8.2 \times 10^3 \text{ lbs.}$$

GEARS C and D

Where

$W = 2460$ lbs.

$V = 161$ F.P.M.

$F = 2.0$ in.

$C = 1660$

$$W_d = \frac{.05\ V\ (FC + W)}{.05\ V + \sqrt{FC + W}} + W$$

$$= \frac{.05 \times 161(2.0 \times 1660 + 24.6 \times 10^2)}{.05 \times 161 + \sqrt{2.0 \times 1660 + 24.6 \times 10^2}} + 24.6 \times 10^2$$

$W_d = 2922$ lbs.

GEARS E and F

Where

$W = 680$ lbs.

$V = 583$ F.P.M.

$F = 1.5$ in.

$C = 1660$

$$W_d = \frac{.05\ V\ (FC + W)}{.05\ V + \sqrt{FC + W}} + W$$

$$W_d = \frac{.05 \times 583\ (1.5 \times 1660 + 680)}{.05 \times 583 + \sqrt{15 \times 1660 + 680}} + 680$$

$$W_d = 1760 \text{ lbs.}$$

<u>GEARS G and H</u>

Where

W = 350 lbs.

V = 1130 F.P.M.

F = 1.0 in.

C = 1660

$$W_d = \frac{.05\ V\ (FC + W)}{.05\ V + \sqrt{FC + W}} + W$$

$$= \frac{.05 \times 1130 (1.0 \times 1660 + 350)}{.05 \times 1130 + \sqrt{1.0 \times 1660 + 350}}$$

$$W_d = 1470 \text{ lbs.}$$

b. BEAM STRENGTH CALCULATION (MAXIMUM ALLOWABLE TOOTH LOAD)

Beam strength calculations are based on the Lewis Equation

$$W_b = S_t\, P\, F\, Y$$

Where

S_t = Safe static bending stress of gear material

P = Circular pitch $\frac{3.14}{D.P.}$

F = Face width

Y = Tooth form factor (obtained by considering the gear tooth as a beam, fixed at one end and loaded at the other) (Ref: Manual of gear design-E Buckingham.)

GEAR A $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70{,}000$ lbs/in.^{2} $W_b = 7 \times 10^4 \times .628 \times 2.5 \times .175$
$P = .628$ in.
$F = 2.5$ in. $W_b =$ <u>19.22×10^3 lbs.</u>
$Y = .175$

GEAR B $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70{,}000$ lbs/in.^{2} $W_b = 7 \times 10^4 \times .628 \times 2.5 \times .116$
$P = .628$ in.
$F = 2.5$ in. $W_b =$ <u>12.75×10^3 lbs.</u>
$Y = .116$

GEAR C $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70{,}000$ lbs/in.^{2} $W_b = 7 \times 10^4 \times .393 \times 2 \times .142$
$P = .393$ in.
$F = 2.0$ in. $W_b =$ <u>78.1×10^2 lbs.</u>
$Y = .142$

GEAR D $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70{,}000$ lbs/in.^{2} $W_b = 7 \times 10^4 \times .393 \times 2 \times .102$
$P = .393$ in.
$F = 2.0$ in. $W_b =$ <u>56.1×10^2 lbs.</u>
$Y = .102$

GEAR E $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70{,}000$ lbs/in.^{2} $W_b = 7 \times 10^4 \times .314 \times 1.5 \times .140$
$P = .314$ in. $W_b =$ <u>46.2×10^2 lbs.</u>
$F = 1.5$ in.
$Y = .140$

GEAR F $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70,000\ \text{lbs/in.}^2$ $W_b = 7 \times 10^4 \times .314 \times 1.5 \times .110$
$P = .314$ in.
$F = 1.5$ in. $W_b = \underline{36.3 \times 10^2\ \text{lbs.}}$
$Y = .110$

GEAR G $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70,000\ \text{lbs/in.}^2$ $W_b = 7 \times 10^4 \times .196 \times 1.25 \times .138$
$P = .196$ in.
$F = 1.25$ in. $W_b = \underline{23.65 \times 10^2\ \text{lbs.}}$
$Y = .138$

GEAR H $W_b = S_t\ P\ F\ Y$

Where: $S_t = 70,000\ \text{lbs/in.}^2$ $W_b = 7 \times 10^4 \times .196 \times 1.25 \times .124$
$P = .196$ in.
$F = 1.25$ in. $W_b = \underline{21.3 \times 10^2\ \text{lbs.}}$
$Y = .124$

c. MARGIN OF SAFETY

Assuming pulsating load conditions: $\frac{W_b}{W_d} \geq 1.35$

Which means that for peroply designed gears, the tooth beam strength should be equal to or greater than 1.35 times the operating load. (Ref: Manual of gear design-E. Buckingham.)

The following is a tabulation of the M.S. of the gears:

GEAR	$MS = \frac{W_b}{W_d}$
A	$\frac{19.22 \times 10^3}{8.2 \times 10^2} = 2.35$
B	$\frac{12.75 \times 10^3}{8.2 \times 10^3} = 1.56$
C	$\frac{78.1 \times 10^2}{29.22 \times 10^2} = 2.67$
D	$\frac{56.1 \times 10^2}{29.22 \times 10^2} = 1.92$

GEAR	$MS = \frac{W_b}{W_d}$
E	$\frac{46.2 \times 10^2}{17.6 \times 10^2} = 2.52$
F	$\frac{36.3 \times 10^2}{17.6 \times 10^2} = 2.06$
G	$\frac{23.65 \times 10^2}{14.7 \times 10^2} = 1.61$
H	$\frac{21.3 \times 10^2}{14.7 \times 10^2} = 1.45$

d. LIMIT LOAD FOR WEAR

(1) Limit load for wear is based on the formula: $W_w = DFKQ$
(Ref: Manual of gear design-E. Buckingham.)

Where: D = Pitch diameter, pinion, inches
F = Face width, inches
K = Load-stress factor = $\frac{S_w^{\,2} \sin\alpha}{1.4} \left(\frac{1}{E_1} + \frac{1}{E_2}\right)$

Where: S_w = Surface endurance limit
α = Pressure angle
E = Modulus of elasticity of pinion and gear
Q = Tooth ratio = $\frac{2N}{N+n}$ (spur gear); $\frac{2N}{N-n}$ (internal gear)

Where: N = No. of teeth in gear
n = No. of teeth in pinion

(2) For acceptable gears $W_w \geqq W_d$

To provide continuous service without appreciable wear.

GEARS A and B $W_w = DFKQ$

Where: $D = 4.0$ in. $W_w = 4.0 \times 2.5 \times 430 \times 2.29$

$F = 2.5$ in.

$K = 430$ $W_w = \underline{9.85 \times 10^3 > W_d = 8.2 \times 10^3}$

$Q = 2.29$

GEARS C and D $W_w = DFKQ$

Where: $D = 2.5$ in. $W_w = 2.5 \times 2.0 \times 375 \times 1.67$

$F = 2.0$ in.

$K = 375$ $W_w = \underline{31.2 \times 10^2 > W_d = 29.2 \times 10^2}$

$Q = 1.67$

GEARS E and F $W_w = DFKQ$

Where: $D = 2.6$ in. $W_w = 2.6 \times 1.5 \times 318 \times 1.56$

$F = 1.5$ in.

$K = 318$ $W_w = \underline{19.4 \times 10^2 > W_d - 17.6 \times 10^2}$

$Q = 1.56$

GEARS G and H $W_w = DFKQ$

Where: $D = 2.5$ in. $W_w = 2.5 \times 1.25 \times 366 \times 1.33$

$F = 1.25$ in.

$K = 366$ $W_w = \underline{15.2 \times 10^2 > W_d = 14.7 \times 10^2}$

$Q = 1.33$

TABLE G1. GEAR MATERIAL HARNDESS DATA

GEAR	HEAT TREAT	SURFACE HARDNESS
A	See report on main bearing	
B	R_c 32-34	R_c 57-59
C	R_c 32-34	R_c 42-44
D	R_c 32-34	R_c 46-48
E	R_c 32-34	R_c 36-40
F	R_c 32-34	R_c
G	R_c 32-34	R_c 4-42
H	R_c 32-34	R_c 4-44

e. GEAR LIFE EXPECTANCY

The calculation of the life expectancy of the gears is based on the statement in "The Manual of Gear Design," by E. Buckingham, that the life of a gear is equal to the inverse proportion of the load raised to the $\frac{10}{3}$ power. Using load-stress factor "K" of 430 for gears "A" and "B" (the lowest life expectancy gears), and basing the life on 100 x 106 cycles, a maximum compressive stress, S_c, of 163,000 is obtained from the chart in the referenced text. The actual compressive stress is found from:

$$S_c = \sqrt{\frac{35W \left(\frac{1}{R_1} + \frac{1}{R_2}\right)}{L \left(\frac{1}{R_1} + \frac{1}{R_2}\right)}}$$

Where: W = Tooth load

R_1 = Pitch radius of gear x Sin Pressure Angle

R_2 = Pitch radius of pinion x Sin Pressure Angle

L = Face width

E_1 and E_2 = Modulus of elasticity of gear and pinion

Therefore: $S_c = 184,600$ lbs./in.^{2}

The life factor then becomes:

$$F_L = \left(\frac{163,000}{184,600}\right)^{\frac{10}{3}} = .66$$

To find the life expectancy at the actual load:

$$L = .66 \times 100,000,000 \text{ cycles}$$

$$= \underline{66 \times 10^6} \text{ cycles}$$

or

$$L = \frac{66 \times 10^6}{\text{RPM} \times 60}$$

$$L = \underline{10,000 \text{ hours}}$$

Since gears "A" and "B" have the smallest life expectancy, calculation for only those gears where shown.

2. SYNCHRO DRIVE BACKLASH CALCULATIONS
(Refer to Figure G2)

DATA	GEAR					
	J	L	M	N	K	R
No. Teeth	396	66	128	32	32	192
Dia. Pitch	20	20	32	32	32	32
Pressure Ang.	20°	20°	14-1/2°	14-1/2°	14-1/2°	14-1/2°
Pitch Dia.	19,800	3,300	4,000	1,000	1,000	6,000

To find the backlash at the individual meshes we have:

$$\text{Backlash (inches)} = \frac{\Delta C}{1/2 \text{ COT } \phi}$$ Van Keuren pg. 116

or

$$\text{Backlash (inches)} = 2 (\Delta C) \text{ TAN } \phi$$

To find lost motion of the driving gear with the synchros stationary we have:

$$\theta = \left(\frac{B_{1\&2}}{R_2}\right) \frac{D_3}{D_2} + \frac{B_{3\&4}}{R_4}$$

(Radians)

L. D. Martin
Machine Design
May, 1953

Nomenclature

B = Backlash

C = Center

D = Testing Diameter

R = Testing Radius

∅ = Pressure Angle

θ = Arc of Lost Motion

Lost Motion of driving gear from 1:1 synchro:

$B_{JL} = 2(\Delta C) \tan \emptyset$
$2(.0207)(.364) = .0151''$

Tol. on dist.*	= .014
Comp error (prec.cl 1)	= .001
Shaft tol.	= .0005
Bearing tol.	= .0002
Distortion due to load on large brg	= .005
ΔC	= .0207

$B_{KR} = 2(.0035)(.258) = .0018''$

Tol on dist.*	= .0018
Comp. error (prec.cl.1)	= .001
Shaft tol.	= .0005
Bearing tol.	= .0002
ΔC	= .0035

$$\theta_{JR} = \left(\frac{B_{KR}}{R_K}\right) \frac{D_L}{D_K} + \frac{B_{JL}}{R_J}$$

$$\frac{.0018}{.5} \quad \frac{3.3}{1} + \frac{.0151}{9.9}$$

$.0119 + .0015 = .0134$ Radians

$.0134 \times 3438 = 46.07$ Minutes

$\theta_{JR} \quad \pm 23$ minutes

*Tolerance on ℄ distance due to machining tolerances on bull gear bearing, upper and lower bearing support castings and location of synchro gear box.

Lost motion of driving gear from 24:1 synchro:

$B_{JL} = .0151$

$B_{MN} = .0018$ (same as B_{KR})

$$\Theta_{JN} = \left(\frac{B_{MN}}{R_M}\right) \frac{D_L}{D_M} + \frac{B_{JL}}{R_J}$$

$$\Theta_{JN} = \frac{.0018}{2} \frac{3.3}{4} + \frac{.0151}{9.9}$$

$$(.0009)\ .825 + .0015$$

$$.0007 + .0015 = .0022 \text{ radians}$$

$$.0022 \times 3438 = 7.56 \text{ minutes}$$

$\Theta_{JN} = \pm 4$ minutes

3. SUMMARY OF GEAR SHAFT BEARING DESIGN DATA AND AZIMUTH BEARING LOAD CALCULATIONS

a. Summary of Gear Shaft Bearing Design Data

BRG. IDENT. FIG.G3	BEARING NUMBER	ACTUAL LOAD	ACTUAL LOAD CORRECTED FOR LIFE & SPEED	RATED LOAD
I	499503	350 at 1725 RPM	378 lb.	440 lb.
II	499503	111 at 1725 RPM	120 lb.	440 lb.
III	43305	16.1 at 825 RPM	17.5 lb.	1111 lb.
IV	499605	412 at 862.5 RPM	445 lb.	1110 lb.
V	499508	1435 at 245 RPM	1556 lb.	2150 lb.
VI	#R-345-LL	4030 at 245 RPM	6860 lb.	9450 lb.
VII	#R-365-LL	13000 at 48 RPM	12,750 lb.	22,800 lb.
VIII	#R-365-RR	20,030 at 48 RPM	19,650 lb.	26,600 lb.

NOTES: 1. All bearings "New Departure" except those marked #.
2. #Denotes "Norma-Hoffman" bearings.

NOTES (Cont'd)

3. a. New Departure bearings corrected for life and speed by formula $R_c = R_a \times L$, where
R_c = corrected load
R_a = actual load
L = correction factor (N.D. Handbook Pg. 10)

b. Norma-Hoffman bearings corrected for life and speed by formula $R_c = R_a \times L \times S$ where
R_c = corrected load
R_a = actual load
L = life factor (Ref. Norma-Hoffman Catalog
S = speed factor Pg. 126)

b. AZIMUTH BEARING LOADS

Bearing load calculations are based on transmission of 15 H.P. Since only a negligible thrust load (the weight of the gears and shafts) exists, the load on the bearings is a function of the resultant of the separation and tangential forces developed by the gears, and the distance of that resultant from the supporting bearings of a given shaft. See figure G4 and G5.

Terminology

$$Q = \text{Torque transmitted} = \frac{\text{H.P.} \times 63025}{\text{RPM}}$$

$$P = \text{Tangential force} = \frac{Q}{\text{Pitch Radius}}$$

$$S = \text{Separation force} = P \times \text{TAN}\,\alpha \quad (\text{where } \alpha = \text{tooth pressure angle})$$

$$L = \text{Resultant force} = \sqrt{P + S^2}$$

(1) Bearing and gear identification is shown in figure G3.

(2) Roman numeral subscripts to P and S indicate bearing reactions.

(3) Arabic numeral subscripts to Q, P and S indicate gear mesh number.

BEARING I

$$Q_1 = \frac{15 \times 63025}{1725} = 548 \qquad P_I = \frac{438 \times 4.94}{6.50} = 333$$

$$P_1 = \frac{548}{1.25} = 438 \qquad S_I = \frac{159 \times 4.94}{6.50} = 121$$

$$S_1 = 438 \times .364 = 159 \qquad L_I \sqrt{(333)^2 + (121)^2}$$

$$L_I = 350 \text{ Lbs.}$$

BEARING II

$$P_{II} = \frac{438 \times 1.56}{6.50} = 105 \qquad L_{II} = \sqrt{(105)^2 + (38)^2}$$

$$S_{II} = \frac{159 \times 1.56}{6.50} = 38 \qquad L_{II} = \underline{111 \text{ Lb.}}$$

NOTE: Sub-subscripts denote gear mesh, e.g., P_{III} is the tangential force reaction of the first mesh on Bearing III.

BEARING III N.B. in calculating L_{III} - S_{III_1} and P_{III_1} are shown negative since they are acting in a direction opposed to S_{III_2} and P_{III_2}

$$Q_2 = \frac{15 \times 63025}{863} = 1090 \qquad P_{III_1} = \frac{438 \times 3.87}{5.25} = 323$$

$$P_2 = \frac{1090}{1.3} = 838 \qquad S_{III_1} = \frac{159 \times 3.87}{5.25} = 117$$

$$S_2 = 838 \times .364 = 303 \qquad P_{III_2} = \frac{838 \times 2.12}{5.25} = 338$$

$$L_{III} = \sqrt{(323 - 338)^2 + (125 - 117)^2}$$

$$S_{III_2} = \frac{303 \times 2.12}{5.25} = 125$$

$$L_{III} = \underline{16.1 \text{ Lb.}}$$

BEARING IV

$$P_{IV_1} = \frac{438 \times 1.38}{5.25} = 115 \qquad P_{IV_2} = \frac{838 \times 3.13}{5.25} = 500$$

$$S_{IV_1} = \frac{159 \times 1.38}{5.25} = 42 \qquad S_{IV_2} = \frac{303 \times 3.13}{5.25} = 181$$

$$L_{IV_1} = \sqrt{(500 - 115)^2 + (181 - 42)^2}$$

L_{IV} = 412 Lbs.

BEARING V

NOTE: In the calculation of loads on Bearings V and VI the angular displacement of the shafts #2 and #3 was considered by using the components of the forces.

$$Q_3 = \frac{15 \times 63025}{245} = 3860$$

$$P_3 = \frac{3870}{1.25} = 3095$$

$$S_3 = 3095 \times .364 = 1126$$

$$P_V = \frac{838 \times 12}{10.5} + \frac{3095 \times .358 \times 2.25}{10.5} + \frac{1126 \times .934 \times 2.25}{10.5}$$

$$P_V = 1421$$

$$S_V = \frac{303 \times 12}{10.5} + \frac{1126 \times 358 \times 2.25}{10.5} + \frac{3095 \times .934 \times 2.25}{10.5}$$

$$S_V = -185$$

$$L_V = \sqrt{(1421)^2 + (-185)^2}$$

L_V = 1435 Lbs.

BEARING VI

$$P_{VI} = \frac{838 \times 15}{10.5} + \frac{3095 \times .358 \times 12.75}{10.5} + \frac{1126 \times .934 \times 12.75}{10.5}$$

$$P_{VI} = 2740$$

$$S_{VI} = \frac{303 \times 1.5}{10.5} + \frac{1126 \times .358 \times 12.75}{10.5} + \frac{3095 \times .934 \times 12.75}{10.5}$$

$$S_{VI} = -2957$$

$$L = \sqrt{(2740)^2 + (-2957)^2}$$

$$L = 4030 \text{ Lbs.}$$

BEARING VII

$$Q_4 = \frac{15 \times 63025}{48} = 19.720 \qquad P_{VII_3} = \frac{3095 \times 6.5}{4} = 5020$$

$$P_4 = \frac{19720}{2} = 9860 \qquad S_{VII_3} = \frac{1126 \times 6.5}{4} = 1830$$

$$S_4 = 9860 \times .364 -3590 \qquad P_{VII_4} = \frac{9860 \times 2.94}{4} = 7250$$

$$S_{VII_4} = \frac{3590 \times 2.94}{4} = 2640$$

$$L_{VII} = \sqrt{(5020 + 7250)^2 + (1830 + 2640)^2}$$

$$L_{VII} = \underline{13,000 \text{ Lbs.}}$$

BEARING VII

$$P_{VIII_3} = \frac{3095 \times 2.5}{4} = 1932$$

$$S_{VIII} = \frac{1126 \times 2.5}{4} = 705$$

BEARING VIII (Cont'd)

$$P_{VIII_4} = \frac{9860 \times 6.94}{4} = 17,100$$

$$S_{VIII_4} = \frac{3590 \times 6.94}{4} = 6230$$

$$L_{VIII} = \sqrt{(1932 + 17100)^2 + (705 + 6230)^2}$$

$$L_{VIII} = \underline{20,030 \text{ Lbs.}}$$

4. AZIMUTH GEAR SHAFTS

a. General

All shaft calculations are based on the shafts being made from S.A.E. 4140 alloy steel, heat treated to 150,000 p.s.i.

Allowable torsional shear stress is taken to be 18 percent of the U.T.S. (see above). (Ref. Marks Handbook.)

Shaft designations are shown in figure G1.

Minimum shaft diameter calculations are found by using the formula:

$$D = \sqrt[3]{\frac{16}{\pi S_s} \sqrt{(K_m M)^2 + (K_t T)^2}} \quad \text{(Ref. Marks Handbook)}$$

where: S_c = shear stress 18% U.T.S. = .18 x 15,000 = 27,000 p.s.i.

K_m = shock and fatique factor = 3

K_t = shock and fatigue factor = 3

(Assuming worst loading conditions)

M = maximum bending moment (see derivation below)

The value of M for shafts #1 and #2 is obtained from the beam formula for a beam supported at the ends with a concentrated load at any point.

$$M = \frac{Pab}{L}$$

b. <u>Shaft No. 1</u>

P = 159 lbs. (S_1 of bearing load calculations)
a = 1.56
b = 4.94
L = 6.0

$$M = \frac{159 \times 1.56 \times 4.94}{6.0}$$

$$M = \underline{204 \text{ in. lb.}}$$

T = 548 (Q of gearing load calculations)

$$D_1 = \sqrt[3]{18.9 \times 10^{-5} \sqrt{(3 \times 204)^2 + (3 \times 548)^2}}$$

$$D_1 = \underline{.715 \text{ in.}}$$

c. <u>Shaft No. 2</u>

N.B.: The total bending moment on shaft No. 2 is the algebraic sum of the moments of S_1 and S_2 which are acting in opposite directions.

P_1 = 159 lbs. (S_1 of bearing loads)

$$M_1 = \frac{159 \times 1.38 \times 3.87}{5.25}$$

a = 1.38 in.
b = 3.87 in.
L = 5.25

$$M_1 = \underline{161 \text{ in. lb.}}$$

P_{II} = 303 lbs. (S_2 of bearing load calculations)

d = 2.12 in.
b = 3.13 in.
L = 5.25 in.

$$M_2 = \frac{303 \times 2.12 \times 3.13}{5.25}$$

$$M_2 = \underline{381 \text{ in. lb.}}$$

$$M_{TOT} = M_2 - M_1$$

$$= 381 - 161$$

$$M_{TOT} = \underline{220 \text{ in. lb.}}$$

$T = 1090$ (Q_2 of bearing load calculations)

$$D_2 = \sqrt[3]{18.9 \times 10^{-5} \sqrt{(3 \times 220)^2 + (3 \times 1090)^2}}$$

$D_2 = \underline{.856 \text{ in.}}$

d. <u>Shaft No. 3</u>

N.B.: The total bending moment on Shaft No. 3 is the algebraic sum of the moments of S_2 and S_3 which are acting in opposite directions. The bending moments are equal to the force S times its distance from its supporting bearing.

$P_1 = 303$ lbs. (S_2 of bearing loads)

$L_1 = 1.5$ in.

$M_1 = P_1L_1 = 303 \times 1.5$

$M_1 = 455$ in. lb.

$P_2 = 1126$ lbs. (S_3 of bearing loads)

$L_2 = 2.25$ in.

$M_2 = P_2L_2 = 1126 \times 2.25$

$M_2 = 2525$ in. lbs.

$M_{TOT} = M_2 - M_1 = 2525 - 455$

$T = 3860$ (Q_3 of bearing loads)

$$D_3 = \sqrt[3]{18.9 \times 10^{-5} \sqrt{(3 \times 2070)^2 + (3 \times 3860)^2}}$$

$D_3 = \underline{1.33 \text{ in.}}$

e. <u>Shaft No. 4</u>

N.B.: Since the physical arrangement of shaft No. 4 is the same as shaft No. 3, calculation of the total bending moment on shaft No. 4 follows that of Shaft No. 3.

$P_1 = 850$ lbs. (S_3 of bearing loads)

$L_1 = 2.88$ in.

$$M_1 = P_1L_1 = 1126 \times 2.88$$

$$M_1 = 3240 \text{ in. lb.}$$

$P_2 = 3590$ lbs. (S_4 of bearing loads)

$L_2 = 2.94$ in.

$$M_2 - P_2L_2 = 3590 \times 2.94$$

$$M_2 = 10{,}550 \text{ in. lbs.}$$

$$M_{TOT} = M_2 - M_1 = 10{,}550 - 3240$$

$$M_{TOT} - 7310 \text{ in. lbs.}$$

$T = 19{,}720$ (Q_4 of bearing loads)

$$D_4 = \sqrt[3]{18.9 \times 10^{-5}\sqrt{(3 \times 73.1 \times 10^2)^2 + (3 \times 19.72 \times 10^2)^2}}$$

$$D_4 = \underline{2.28 \text{ in.}}$$

f. Conclusion

The foregoing shaft diameters are minimum requirements. Actual shaft diameters will be affected by the bore size of the supporting bearings required, but will always be equal to or greater than the calculated minima.

The following safety factors indicate the ratio of the allowable working stress to the assumed working stress of the actual shafts.

$$S.F. = \frac{D^3 \text{ actual}}{D^3 \text{ calculated}}$$

Shaft No. 1 $\quad S.F. = \frac{(.937)^3}{(.715)^3} = \underline{2.25}$

Shaft No. 2 $\quad S.F. = \frac{(1.25)^3}{(.856)^3} = \underline{3.11}$

Shaft No. 3 $\quad S.F. = \frac{(1.77)^3}{(1.33)^3} = \underline{2.35}$

Shaft No. 4 $\quad S.F. = \frac{(2.55)^3}{(2.28)^3} = \underline{1.46}$

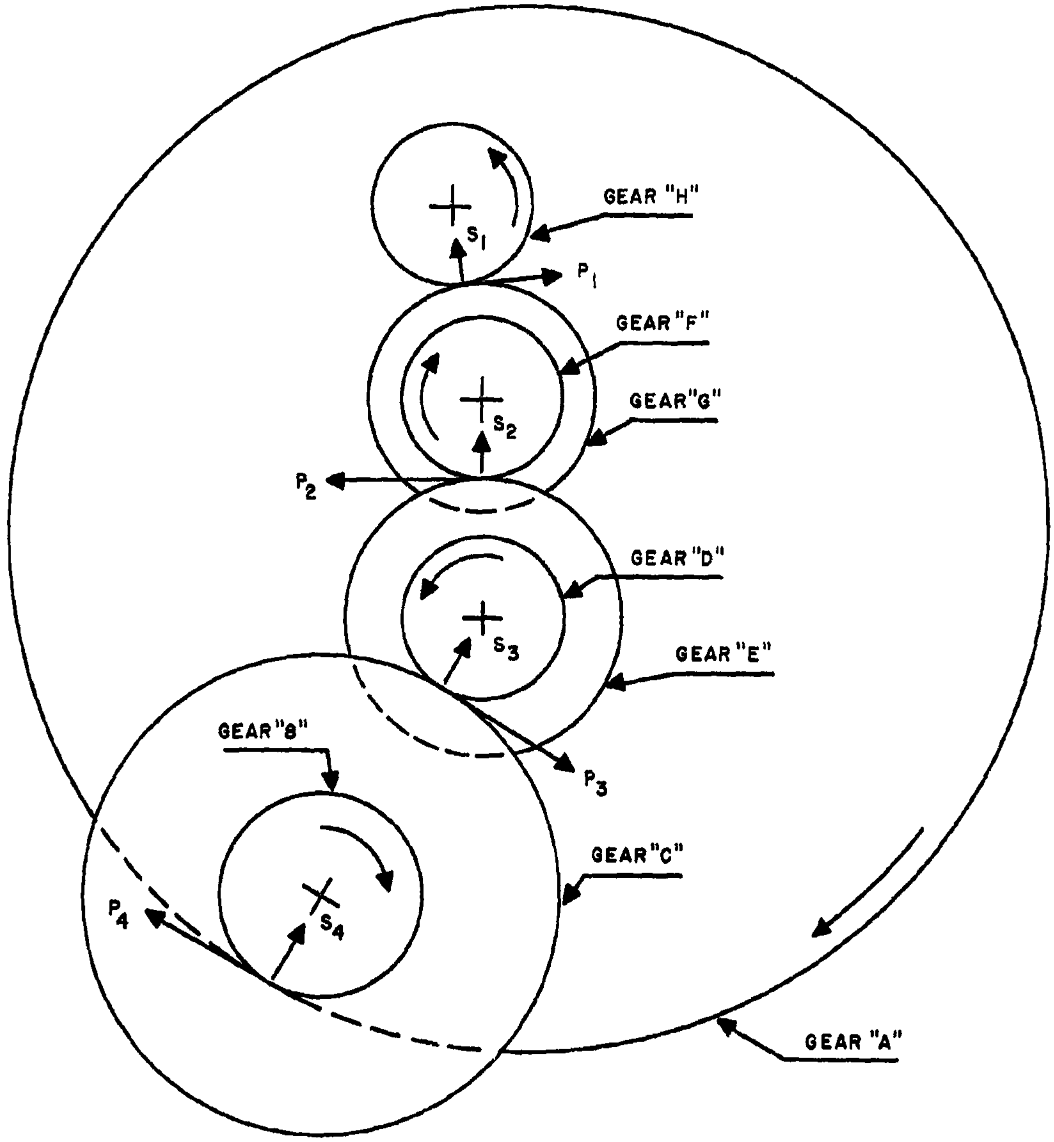


Figure G1. Azimuth Drive Gearing

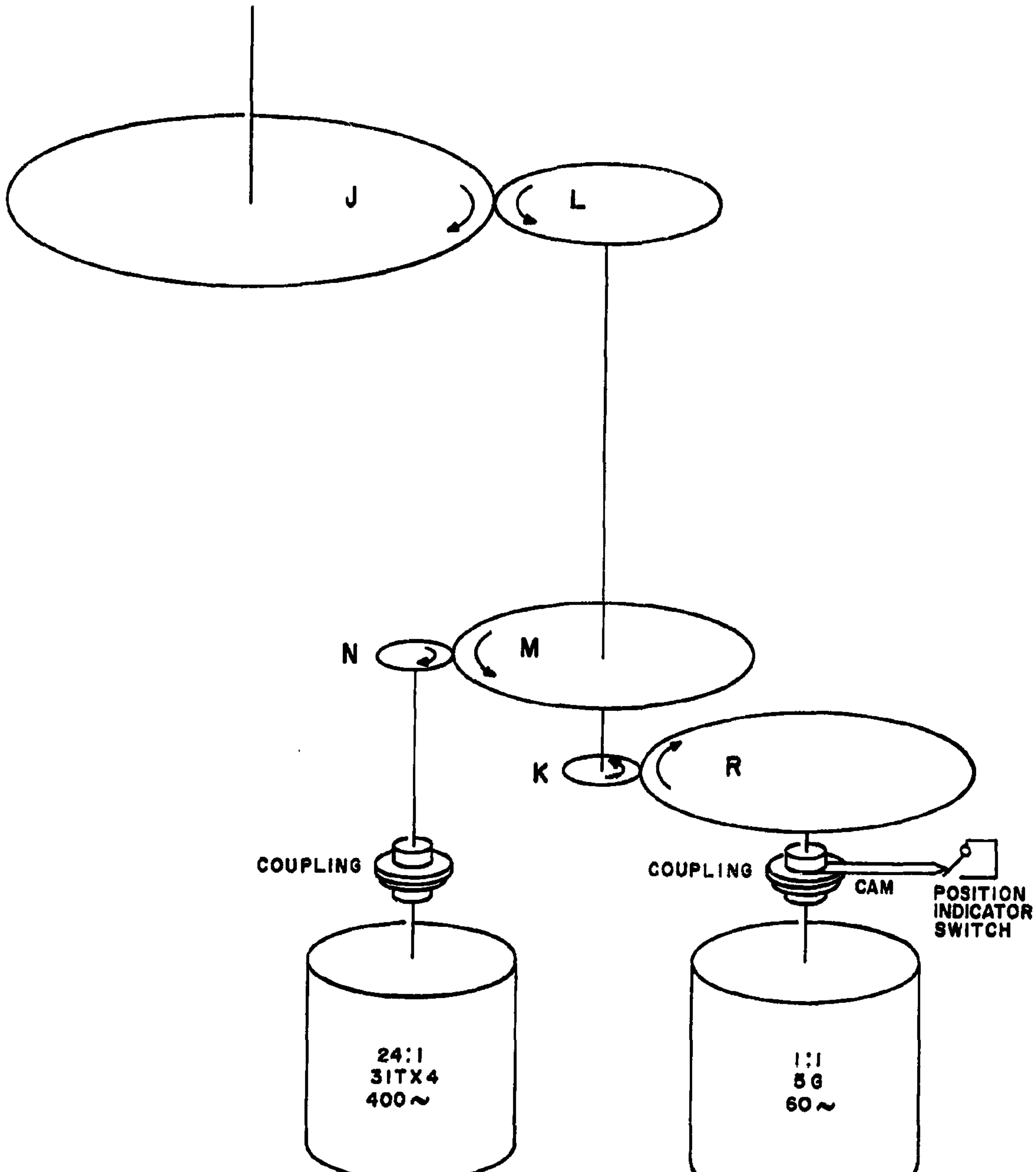


Figure G2. Synchro Drive System

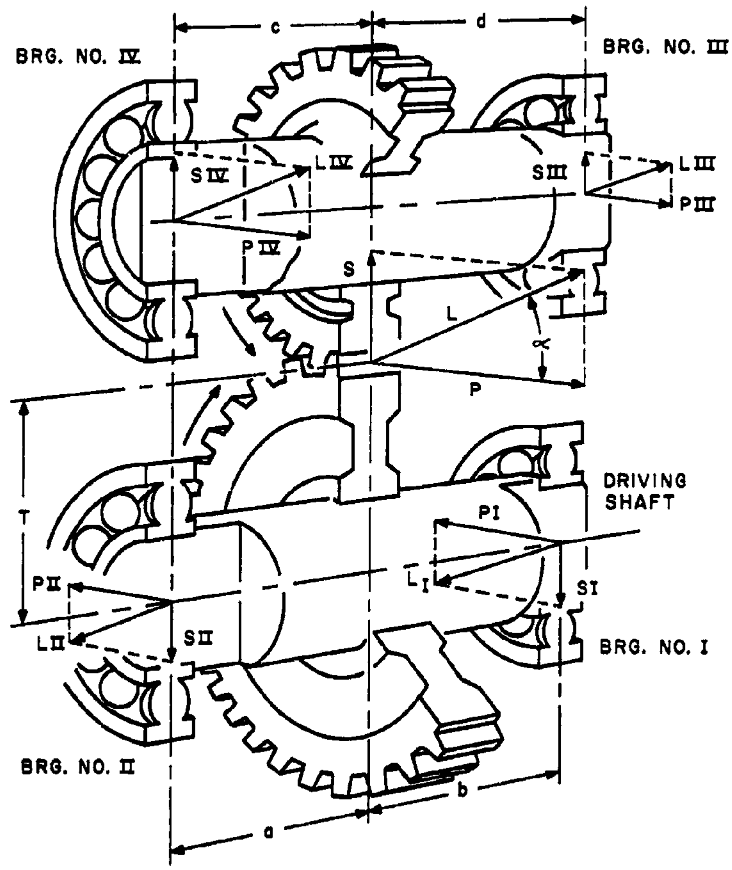
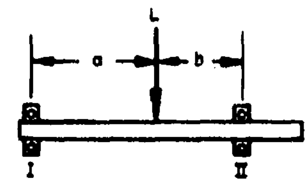


Figure G3. Typical Azimuth Bearing Load Analysis



LOAD ON BRG. NO. I $= \frac{L + b}{a + b}$

LOAD ON BRG. NO. II $= \frac{L + a}{a + b}$

Figure G4. Basic Bearing Load Analysis

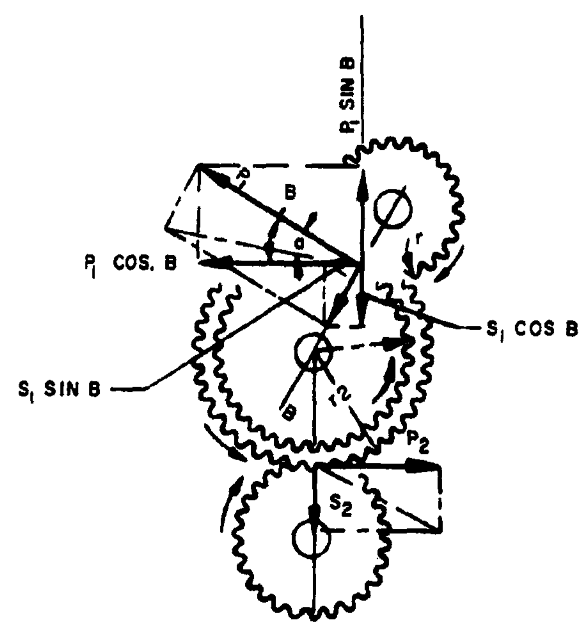


Figure G5. Typical Force Analysis of Angularly Displaced Shafts

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ADDENDUM H

WIND FORCE ON REFLECTOR ASSEMBLY

The drag force produced when relative motion exists between a fluid and a solid body is described by the formula:

$$F_d = \tfrac{1}{2} \rho \, C_d \, A \, V^2$$

where:

F_d = drag force (lbs)

A = projected area of solid body (ft^2)

V = velocity of relative motion (ft/sec)

C_d = drag coefficient (dimensionless quantity dependent upon configuration and surface conditions of solid body)

ρ = density of fluid (slugs/ft^3, dependent upon temperature and pressure conditions of the fluid)

For Air at Sea Level

$$\rho = \frac{1.325}{(Ta)}(g)$$

where:

Ta = absolute temperature (degrees rankine)

Po = standard air pressure at sea level = 29.92" Hg

g = 32.2 ft/sec^2

at $-65^\circ F$

$Ta = 395^\circ R$

$$\rho = \frac{(1.325)(29.92)}{(395)(32.2)} = .00312 \text{ slugs/ft}^3$$

Survival Conditions

Case I:

$$V = 90 \text{ mph} = 132 \text{ ft/sec}$$

* C_d (5/8" squarex mesh, without ice) = 0.53

$A = 364 \text{ ft}^2$

$\rho = .00312$ slugs/ft3

$F_d = \frac{1}{2} (.00312)(0.53)(364)(132)^2 = 5340$ lbs.

Derived from wind tunnel tests, applies to this size squarex mesh only.

Case II.

$V = 40$ mph $= 58.6$ ft/sec

C_d = (antenna iced, approximates flat plate) = 1.0

$A = 364 \text{ ft}^2$

$\rho = .00312 \text{ slugs/ft}^3$

$F_d = \frac{1}{2} (.00312)(1.0)(364)(58.6)^2 = 1950$ lbs.

Obviously, the most severe wind loading conditions occur when the wind velocity is 90 mph, the reflector being free of ice. The force produced is 5340 lbs., equivalent to a uniform static load of 14.7 lbs/sq. ft.

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ADDENDUM I

STATIC LOAD AND DEFLECTOR TESTS

Test Procedure

The reflector, reflector support and feed support were suspended from a heavy steel frame, with the reflector's vertex pointed towards the zenith. Male and female self-reading aluminum gauges were attached to the reflector and reflector support, to record deflections at any loading condition (see figure I1). A representative sample of concrete blocks was weighed on a calibrated scale and found to be 61 pounds per block.

A summation of the moments in comparing the load on the reflector support and pedestal bolting arrangement for, first, the reflector in its vertical location and, secondly, the reflector in its test location shows that 800 lbs of the antenna panel sections must be included as part of the effective loading to obtain valid results. In addition, 160 lbs. of 1/4-inch masonite was used to protect the screen surface of the reflector.

Concrete blocks simulated the load and were oriented so both halves of the reflector were symmetrical, and distributed to obtain uniform loading.

Test Results

Test results showed that the antenna easily surpasses its requirements. In interpreting the results of the deflection readings (tables I1 and I2), it must be remembered that the basic calibration was established with 800 lbs. of panels and 160 lbs. of masonite on the structure. This represents a distributed load of 960 lbs/365 ft.^{2} for 2.63 psf. This is, therefore, the zero datum for all deflection readings.

The Operation Average Net Deflection (table I3) uses the vertex location after deflection, as a datum to show the surface tolerance. Where there are symmetrical readings, these have been averaged: e.g., 1-1A, 2-2A, 3-3A, 5-7.

The Survival Average Net Deflection (table I4) uses the "W Brace" (i.e., 5 & 7) pick-up points as a datum to show the actual movement of the reflector itself with respect to the support structure. It should be noted that there is basically no deflection at the vertex. This was anticipated, for there is a negative bending moment and deflection in the central section. This is counterbalanced by the force of the local distributed loadings; therefore, little or no deflection occurs. These values have again been averaged as discussed in the previous paragraph.

Finally, it should be noted that there is an angular deflection of the beam, in addition to the surface tolerance deflection. This angular movement can be approximated by taking the difference of the 4 and 4A rods = 0.0013/132 inches = Tan Θ = 0.00227; therefore Θ = 0° -08' for the operational loading.

TABLE I1. FIELD DATA DEFLECTION READINGS

LOCATION	INITIAL READING	OPERATIONAL READING	SURVIVAL READING	CLIMAX READING	OPERATIONAL READING	FINAL READING
1A	0.1"	0.6"	1.0"	1.25"	0.7"	0.25"
2A	0.0"	0.35"	0.4"	0.45"	0.5"	0.05"
7	0.1"	0.4"	0.75"	0.9"	0.45"	0.1"
3A	0.1"	0.6"	0.05"	1.45"	0.7"	0.2"
4	0.1"	0.1"	0.4"	0.45"	0.15"	0.0"
6	0.1"	0.3"	0.1"	0.2"	0.3"	0.15"
4A	0.1"	0.4"	1.2"	1.4"	0.5"	0.0"
2	0.0"	0.2"	0.45"	0.4"	0.25"	0.05"
5	0.1"	0.45"	0.75"	0.9"	0.5"	0.15"
3	0.1"	0.55"	1.05"	1.2"	0.7"	0.25"
1	0.2"	0.55"	0.85"	1.0"	0.65"	0.2"

TABLE I2. TEST DATA DEFLECTION READINGS

NUMBER	TOTAL LOAD LBS.	TOTAL P.S.F.	DEFLECTION LOAD LBS.	DEFLECTION P.S.F.
1) Inital Reading	960	2.63	0	0
2) Operational Readings	3220	8.82	2260	6.19
3) Survival Readings	5380	14.7	4420	12.1
4) Climax Readings	5810	15.9	4850	13.3
5) Additional Loadings	6260	17.15	no readings	-
6) Operational Reading #2	3220	8.82	2260	6.19
7) Final Readings	960	2.63	0	0

TABLE I3. OPERATION AVERAGE NET DEFLECTION

LOCATION	OPERATIONAL READING	INITIAL READING	GROSS DEFLECTION	VERTEX (BASE) DEFLECTION	NET DEFLECTION	AVE NET DEFLECTION
1A	0.6	0.1	0.5	0.2	0.3	0.22
2A	0.35	0.0	0.35	0.2	0.15	0.07
7	0.4	0.1	0.30	0.2	0.2	0.17
3A	0.6	0.1	0.5	0.2	0.3	0.27
4	0.1	0.1	0.0	0.2	-0.2	-0.2
6	0.3	0.1	0.2	0.2	0.0	0.0
4A	0.4	0.1	0.3	0.2	0.1	0.1
2	0.2	0.0	0.2	0.2	0.0	0.07
5	0.45	0.1	0.35	0.2	0.15	0.17
3	0.55	0.1	0.45	0.2	0.25	0.27
1	0.55	0.2	0.35	0.2	0.15	0.22

The surface tolerance is a function of the total deflection of any point less the vertex deflection of 0.2 inches. The left side of the reflector deflected more than the right side; therefore, the average of corresponding points has been taken as a true reading.

TABLE I4. SURVIVAL AVERAGE NET DEFLECTION

LOCATION	SURVIVAL READING	INITIAL READING	GROSS DEFLECTION	"W" BRACE (BASE) DEFLECTION	NET DEFLECTION	AVE NET DEFLECTION
1A	1.25	0.1	1.15	0.8"	0.35	0.17
2A	0.45	0.0	0.45	0.8"	0.35	-0.35
7	0.9	0.1	0.80	0.8"	0	0
3A	1.45	0.1	1.35	0.8"	0.55	0.42
4	0.45	0.1	0.35	0.8"	-0.45	-0.45
6	0.2	0.1	0.1	0.8"	-0.70	-0.70
4A	1.4	0.1	1.3	0.8"	0.50	0.50
2	0.45	0.0	0.45	0.8"	-0.35	-0.35
5	0.9	0.1	0.8	0.8"	0	0
3	1.2	0.1	1.1	0.8"	0.30	0.42
1	1.0	0.2	0.8	0.8"	0	0.17

The surface tolerance is a function of the total deflection of any point, less the base deflection at the outboard "W Brace" pick up points. The left side of the reflector deflected more than the right side; therefore, the average of corresponding points has been taken as a true reading.

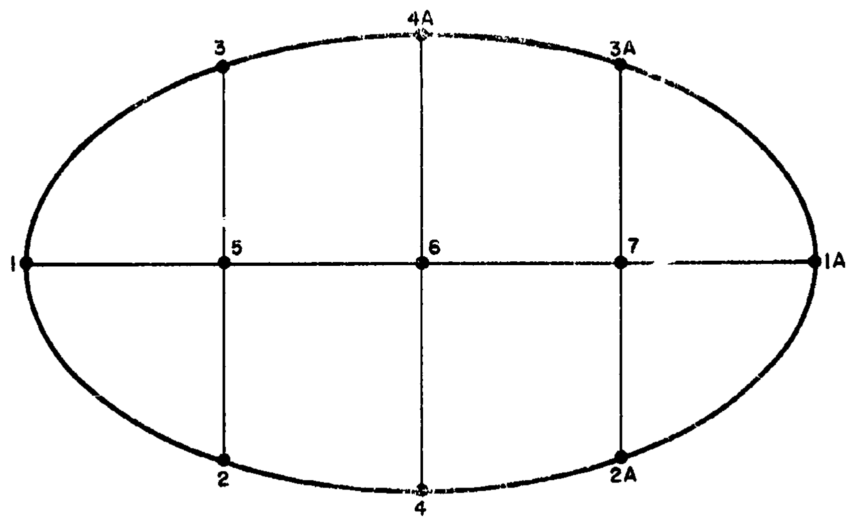

Figure I1. Location of Deflection Gauges

ADDENDUM J

MOISTURE RESISTANCE TEST

Test Procedure

(a) Dry at 130° ± 5°F for 24 hours.

(b) Conditioned at 77° ± 5°F and 40 to 50% relative humidity for 24 hours.

(c) Pedestal performance measurements were taken as described in paragraphs 3.3.2.2 and 3.3.2.5 of the Technical Requirements. The following cycle was performed five times. Temperature tolerance was maintained at ±5°F. Relative humidity was maintained between 90 and 98%.

(d) The temperature started at 86°F and was increased to 149°F in four hours at a rate of not less than 15°F per hour.

(e) The temperature was maintained at 149°F for eight hours.

(f) The temperature was returned to 86°F in four hours at a rate of not less than 15°F per hour.

(g) The temperature was maintained at 86°F for 21 hours. Pedestal performance measurements were taken during this step between the eighth and 12th hours.

(h) The temperature was lowered to 68°F in one hour.

(i) The temperature remained 68°F for four hours.

(j) The temperature was raised to 86°F in one hour.

(Note: The change in temperature from step (i) to step (j) was not less than 180°F.)

(k) The temperature was maintained at 86°F for five hours after which pedestal performance measurements were made. (Note: End of cycle.)

(l) The pedestal was conditioned for 24 hours at 77° ± 5°F and 40 - 60% relative humidity after which pedestal performance measurements were made.

Test Results

The data sheet on the following page provides the test results using the procedure above.

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DATA

Observed Data

	CONDITIONING	FIRST CYCLE		SECOND CY
	STEP (c)	STEP (g)	STEP (k)	STEP (g)
Operating Voltage------V_1	120	126	126	126
V_2	120	126	126	126
V_3	120	126	126	126
Starting Current-amps--I_1	87	88	82	83
I_2	87	82	82	83
I_3	87	82	86	86
Time to Attain Speed	Inst.	Inst.	Inst.	Inst.
Operating Current-amps-I_1	6.0	8.2	8.0	8.1
I_2	7.0	8.7	8.4	8.4
I_3	6.5	8.7	8.4	8.2
Power-watts-----------W_1	250	220	200	200
W_2	230	210	220	200
W_3	120	170	140	140
Pedestal Speed-RPM	6.2	6.2	6.2	6.2
Thermocouple Readings (deg. F) oF				
Position Chamber	80	90	86	87
Outer Casting	75	88	86	87
Thermostat Well	75	88	86	87
Inner Casting Level #1	75	88	86	87
Inner Casting Level #2	75	88	86	87
Rotary Joint-Floating NT.	75	88	86	87
Oil Filter-Top	75	88	86	87
Heater	75	88	86	87
Oil-On Dipstick	75	88	86	87
Gear Mesh Oil Tube	75	88	86	87

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Addendum J

DATA SHEET, MOISTURE RESISTANT TEST

I CYCLE	SECOND CYCLE		THIRD CYCLE		FOURTH CYCLE		FIFTH CYCLE		FINAL CONDITIONING
STEP (k)	STEP (g)	STEP (k)	STEP (g)	STEP (k)	STEP (g)	STEP (k)	STEP (g)	STEP (k)	STEP (l)
126	126	126	126	126	126	126	126	126	126
126	126	126	126	126	126	126	126	126	126
126	126	126	126	126	126	126	126	126	126
82	83	92	84	83	92	90	92	92	90
82	83	92	84	83	92	90	92	92	92
86	86	92	84	83	92	90	92	92	92
Inst.	Inst.	Inst.	Inst.	Inst.	Inst.	Inst.	Inst.	Inst.	Inst.
8.0	8.1	8.1	8.2	8.3	8.0	8.2	8.2	8.2	7.7
8.4	8.4	8.4	9.2	9.2	8.2	8.4	9.2	9.2	9.2
8.4	8.2	8.2	8.2	8.3	8.2	8.4	8.4	8.5	8.6
200	200	200	240	240	220	220	200	200	240
220	200	220	160	160	200	202	220	202	190
140	140	140	140	140	160	120	120	140	140
6.2	6.2	6.2	6.2	6.2	6.25	6.2	6.2	6.2	6.2
86	87	86	88	87	100	87	90	88	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78
86	87	86	88	85	90	85	90	86	78

2

ADDENDUM K

TEMPERATURE TESTS

Test Procedure

(a) The ambient temperature was set to 77° ± 3°F and the relative humidity set to less than 75 percent. Pedestal performance tests were taken.

(b) The temperature was raised to 163° ± 3°F for a period of four hours. The relative humidity was uncontrolled.

(c) The temperature was lowered to 153° ± 3°F and the relative humidity remained uncontrolled. Pedestal performance tests were made within 30 minutes of the time these conditions were met.

(d) The temperature was lowered to -68° ± 3°F for a period of 24 hours with the relative humidity uncontrolled.

The results of the next 48 hours of the test are given in log form. Table K1 shows temperature readings taken during that interval of time.

October 28, 1958

* 9:00 AM	Heater off. Thermocouple readings as shown. Attempted to start motor. Ammeter needle went off scale (100A) came back to approximately 90A for a second or two and then fuse blew. Replaced fuse and tried to turn pedestal with hand crank. Found immovable. Turned heater switch on. No heat as evidenced from No. 13 thermocouple reading. Check of control box showed voltage to be present across relay. Removed access door and contactor box door to check voltage to relay and heater continuity. From this time on (approx. 10:30) the above doors were left off. Heater trouble was traced to the relay being stuck and it was shorted out.
* 11:50 AM	Heater turned on.
12:00 PM	Turned slightly and with great difficulty by hand crank.
12:10 PM	Tried to start and blew fuse.
2:00 PM	Pedestal allowed to cool for over an hour. An investigation then indicated that the fuses being used in the contactor box were not slow blow fuses. Since none were

P 22 AM 2 R

*	2:00 PM (cont'd)	available, steel slugs were substituted for the fuses. However 40A slow blow fuses were in series with the line at the power box. It was decided to wait the full 30 minutes after heaters were turned on.
*	3:45 PM	Turned heater on.
*	3:55 PM	Thermocouple readings taken to see if there was any appreciable difference over the 12:00 noon figures because of the slightly higher initial temperatures. The heater was then left on for 20 minutes more before attempting to start.
	4:15 PM	Attempted to start motor by holding start switch 5 to 10 seconds. Ammeter needle went off scale, returned and held at approximately 90A. No start.
	4:25 PM	Same as 4:15.
	4:30 PM	Pedestal turned with hand crank. Effort expended approximately 1-1/2 times that at room temperature.
	4:35 PM	Same as 4:15.
**	4:45 PM	Pedestal started to turn very slowly and came up to speed in a few seconds. Ran for about 2-1/2 minutes with oil flow indicating normal and then shut down. Tried to start again and pedestal came right up to speed but shut down after about two minutes. Meters showed momentary peak of 90A and then returned to normal running current. Motor cut out traced to -10 thermostat.
	7:00 PM	The pedestal was allowed to cool down before trying again.
*	10:05 PM	No heat, pedestal started right up, oil flow indicated normal, and motor was cut out approximately two minutes after start. Indications were that either the oil level dropped below the thermostat exposing it to the cold air or, with the oil circulating, cold oil was being brought into contact with the thermostat. Another gallon of oil was added to the pedestal, but with the same results. Left to soak overnight.

October 29, 1958

*	9:08 AM	No heat, pedestal started right up, ran for a couple of minutes and then was shut off. The motor control thermostat was shorted out. Pedestal performance tests were made.
	9:11 AM	Heater turned on.
**	9:40 AM	Minus 10 thermostat closed. Thermocouple No. 17 read 68°F. Number 12 read -13°F.
*	9:55 AM	Pedestal started. Number 17 read +85°F. Pedestal stopped after two minutes. Number 17 read 35 to 40°F.
*	10:10 AM	Pedestal started. Number 17 read +91°F. Pedestal stopped after 3 minutes. Number 17 read +30°F.
*	10:35 AM	Pedestal started. Number 17 read +84°F. Pedestal stopped after 3-1/4 minutes. Number 17 read +30°F.
*	10:55 AM	Heater was shut off for about two minutes, during the period between this and the last reading. Pedestal started and ran for 3-1/2 minutes. Number 15 read -49°F reached a peak of +55°F and dropped to 30°F when the motor shut down.
*	11:15 AM	Ran 4 minutes. Number 15 started at -40 rose to +40 then dropped to 28 as thermostat opened.
*	11:32 AM	The pedestal was started and allowed to run until it shut down. Then after a one minute period, it was started again and allowed to run until it was tripped by the thermostat. This procedure was continued for 40 minutes with the pedestal being unable to rotate continuously for any great length of time. At the same time, thermocouple readings were taken every five minutes.
*	12:12 PM	The thermostat was shorted and the pedestal allowed to rotate during the lunch break.
*	1:44 PM	Reading taken after lunch.
*	2:00 PM	Last reading taken during this particular run.

* Temperature reading taken before start of test.

** Temperature reading taken after start of test.

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(e) The temperature was raised to 77°F ± 3°F with relative humidity less than 75 per cent. Pedestal performance tests were performed within 30 minutes of the time that these conditions were reached.

Test Results

Table K2 tabulates the results of the temperature tests from the procedure above. Malfunctions found, and solutions provided are given in part d of Phase 3 in Section 4.

TABLE K. COLD TEMPERATURE TEST THERMOCOUPLE READINGS

THERMOCOUPLE NO. / POSITION	7 OUTER CAST-ING	8 THERMO-STAT WELL	9 INNER CASTING LEVEL #1	10 INNER CASTING LEVEL #2	11 ROTARY JOINT	12 OIL FILTER TOP	13 HEATER	14 OIL ON DIP STICK	15 GEAR MESH OIL TUBE	17 THERMO-STAT OIL
OCTOBER 28, 1958										
9:00 AM	-62°F	-62°F	-62°F	-62°F	-62°F	-62°F	-62°F	-62°F	-62°F	-62°F
11:50 AM	-62	-62	-62	-62	-62	-62	-62	-62	-62	-62
11:55 AM						-60	+55			
12:00 NOON	-60	-48	-48	-46	-60	-43	+74	-55	-60	+23
12:07 PM	-59	-37	-37	-37	-60	-26	+88	-50	-60	+49
12:19 PM	-58	-20	-20	-20	-57	- 8	+112	-42	-60	+69
3:10 PM	-58	-55	-55	-55	-58	-50	-52	-59	-60	-52
3:55 PM	-59	-42	-43	-40	-58	-44	+84	-52	-60	+30
4:25 PM	-60	+ 5	0	0	-54	+ 9	+109	-30	+29	+33
10:05 PM	-61	-60	-60	-60	-60	-60	-60	-60	-60	-60
OCTOBER 29, 1958										
9:08 AM	-65	-65	-65	-65	-65	-65	-65	-65	-65	-65
9:41 AM	-63	-23	-25	-24	-60	-13	+109	-50	-64	+68
9:55 AM	-62	-12	-11	-10	-58	+ 2	+115	-44	-64	+85
10:10 AM	-60	+ 3	0	0	-55	+20	+120	-38	-62	+91
10:33 AM	-58	+10	+ 8	+10	-52	+35	+124	-33	-55	+84
10:55 AM	-55	+14	+11	+14	-49	+36	+116	-30	-49	+80
11:15 AM	-58	+15	+11	+14	-47	+34	+118	-35	-40	+79
11:32 AM	-50	+ 9	+ 5	+12	-45	+20	+60	-40	+14	+16
11:37 AM	-50	+ 5	+ 1	+ 8	-45	+15	+45	-41	+ 9	+13
11:42 AM	-49	+ 3	- 1	- 5	-45	+10	+45	-41	+ 7	+11
11:47 AM	-49	- 3	+ 3	-45	-45	+ 9	+43	-42	+ 6	+10
11:52 AM	-49	- 2	- 5	0	-45	+ 8	+42	-44	+ 7	+10
11:57 AM	-46	- 3	- 6	0	-45	+ 7	+41	-42	+ 6	+10
12:02 PM	-46	- 5	- 7	- 1	-45	+ 7	+42	-43	+ 6	+10
12:07 PM	-45	- 5	- 8	- 1	-44	+ 7	+42	-43	+ 6	+10
12:12 PM	-45	- 5	- 8	- 1	-44	+ 7	+43	-43	+ 6	+ 9
1:44 PM	-36	+ 1	- 1	+ 2	-40	+11	+57	-29	+13	+12
2:00 PM	-34	+ 3	0	+ 5	+38	+13	+57	-27	+15	+14

n
P 22 AM 2 '1

TABLE K2. TEMPERATURE TEST DATA SHEET

Observed Data

STEP	(a)	(c)	(d)	(e)
TIME	10/24/58 2:30pm	10/25/58 8:05am	10/29/58 9:08am	10/30/58 9:20am
Operating Voltage------V_1	126	126	115	120
V_2	126	126	115	120
V_3	126	126	115	120
Starting Current-amps-I_1	90	92	92	86
I_2	92	92	92	88
I_3	92	92	92	86
Time to Attain Speed	Inst.	Inst.	Inst.	Inst.
Operating Current-amps-I_1	7.7	8.2	8.4	6.8
I_2	9.2	9.2	10.0	8.0
I_3	8.6	8.5	8.6	8.0
Power-watts------------W_1	240	220	230	220
W_2	190	170	200	200
W_3	140	120	140	110
Pedestal Speed-RPM--max.	6.2	6.2	6.2	6.2

Thermocouple Readings (deg F)

Position	No.	(a)	(c)	(d)	(e)
Outer Casting	7	78	150	-65	77
Thermostat Well	8	78	150	-65	77
Inner Casting #1	9	78	150	-65	77
Inner Casting #2	10	78	150	-65	77
Rotary Joint-Floating NT.	11	78	150	-65	77
Oil Filter Top	12	78	150	-65	77
Thermostat (Oil)	13		150	-65	77
Heater	14	78	150	-65	77
Oil-Dip Stick	15	78	150	-65	77
Gear Mesh Oil Tube	16	78	150	-65	77

ADDENDUM L

OVERTURNING MOMENT, ANTENNA ERECTION

When the reflector is assembled about its pivot point with its top on the ground, its center of gravity has a moment arm of 72 inches from the pivot point. See figure I-1.

Resisting Moment:

$$M_R = (\bar{X}_R)\ (W) = (72'')\ (1600^{\#}) = 115{,}200 \text{ inch-lbs.}$$

$$M_R = \frac{115{,}200 \text{ inch-lbs}}{12} = 9600 \text{ ft-lbs.}$$

Cable Tension:

$$(52.5)F_1 = M_R$$

$$F_1 = \frac{115{,}200 \text{ inch-lbs.}}{52.5 \text{ inches}} = 2200 \text{ lbs.}$$

When the reflector is passing thru the "transfer point," the load is gradually transferred from the lifting cable to the lowering cable of the bridle. The overturning moment goes from 9600 foot-lbs., thru zero, then gradually rises to a positive maximum just before the reflector is erected.

Positive Moment:

$$M_P = (\bar{X}_P)\ (W) = (63'')\ (1600^{\#}) = 100{,}800 \text{ inch-lbs.}$$

$$M_P = \frac{100{,}800}{12} = 8400 \text{ foot-lbs.}$$

Cable Tension:

$$F_2 = \frac{100{,}800 \text{ inch-lbs}}{67 \text{ inches}} = 1500 \text{ lbs.}$$

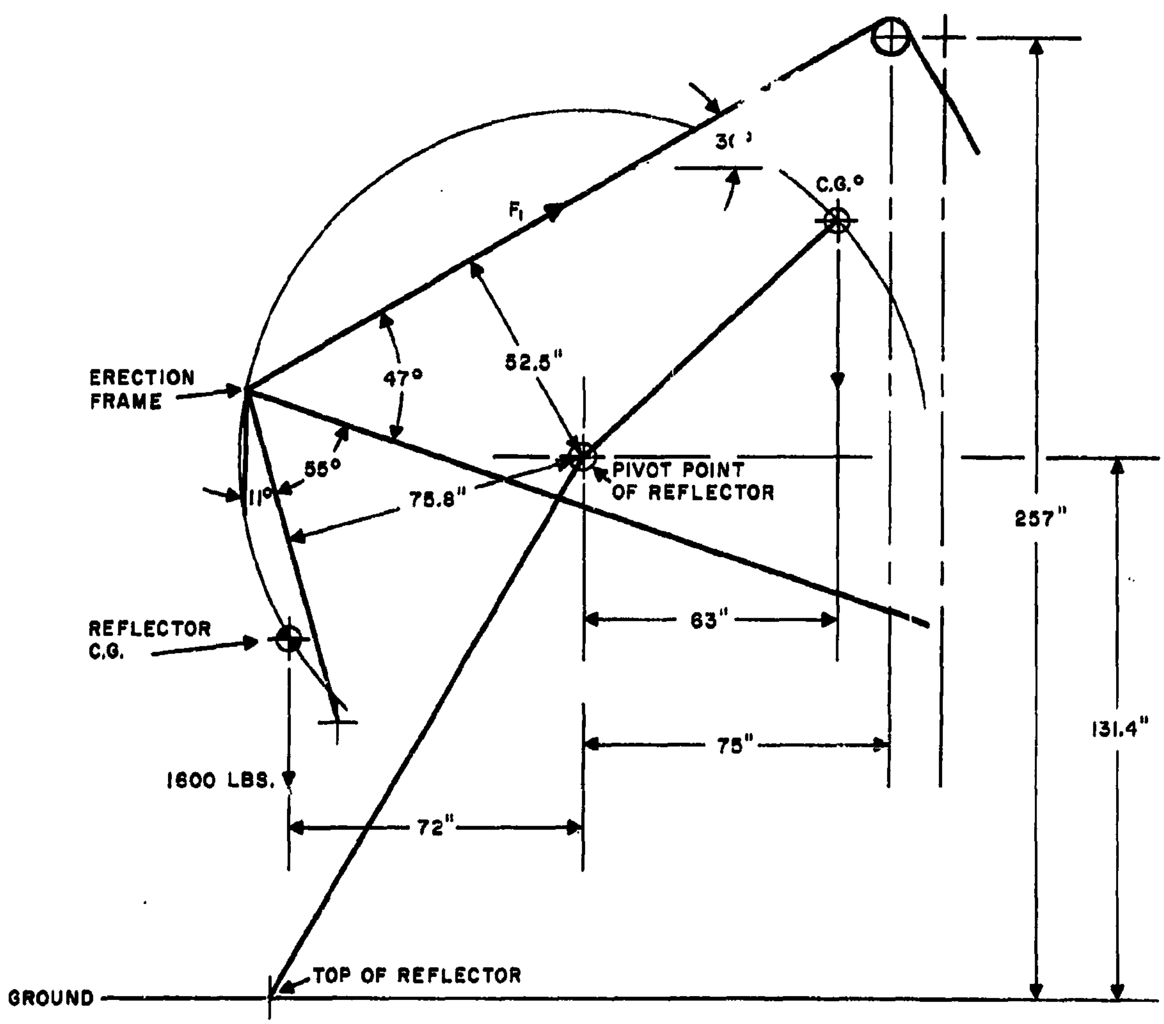

Figure L1. Overturn Moment Diagram

ADDENDUM M

TESTS ON GROUND ANCHORS

1. Screw Type Anchors

The earth anchor selected for the most common soil conditions (Classes 5 & 6) is A.B. Chance Company's No. 10146, a 10 inch diameter screw anchor with a 66 inch long shank. Its holding power is rated at 10,000 pounds and 8,000 pounds for soil classes 5 and 6 respectively.

The ease with which this anchor could be set in position was demonstrated for attending USASRDL personnel at Hazeltine Corp. Manufacturing facility, Greenlawn, N.Y. The anchor was rotated and driven into the ground (Class 5 soil) in approximately six minutes. Removal was accomplished with greater ease in approximately four minutes.

To test its holding power in this soil, the anchor was driven into the ground and subjected to a pull test. Between the shank eye of the buried anchor and the cable of an M-35 truck winch, a dynamometer of 10,000 pound capacity was shackled. With the winch operating, a 10,000 pound load was slowly applied to the screw anchor. With the load maintained at 10,000 pounds, the anchor remained firmly embedded.

2. Arrow Head Anchor

The arrow head anchor is recommended for use in soil Classes 3 and 4. It is rated at 11,000 pounds holding power when used in hardpan. In order to use these anchors, holes must be dug and the anchors inserted and buried. Either this, or the screw anchor previously described, may be used in Class 4 soil, whichever is more expedient under field conditions. Generally speaking, this anchor may be used wherever the field conditions are unfavorable for the use of the screw type anchor. This type of anchor must be dug out to remove, whereas the screw type anchor can be backed out by turning its rod counter-clockwise.

3. Rock Anchor

The anchor, recommended for Class 1 soil conditions, is the expanding type manufactured by A.B. Chance Co. The use of this anchor requires the drilling of a hole 1 7/8" in diameter and one foot deep in the rock. The anchor is slid down into the hole in the rock, and the rod turned clockwise, causing the anchor to expand and wedge tightly against the inside of the hole.

In this case it would be necessary to include a power impact tool (electric or pneumatic) and cutting tools to drill the anchor holes. Since the cutting tool is a perishable tool, a means of re-sharpening or replacement at the site should be considered. Operation in rocky terrain will also be limited by the capabilities of the supporting vehicles, and for these reasons is generally not considered feasible.

4. Operation of the equipment in swamps and marshes (Class 8) was not considered feasible owing to the inability of the supporting vehicles to negotiate this type of terrain.

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